UNIVERSITA' DI GENOVA
Istituto Politecnico di
Ingegneria Navale

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WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

DIPARTIMENTO DI INGEGNERIA NAVALE
UNIVERSITA' DEGLI STUDI DI GENOVA
NAV 017 006

Part 1

RESISTANCE AND PROPULSION

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WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL 1983
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SHIP DESIGN FOR FUEL ECONOMY

SCHEDULE

8th WEGEMT SCHOOL "Ship Design for Fuel Economy"

The School will follow the three key topics:

"Resistance and Propulsion" - 30/8, 31/8, 1/9
"Propulsion Machinery" - 2/9, 5/9, 6/9
"Design and Operation" - 6/9, 7/9, 8/9
Discussion on "Team Design Work" - 9/9

Registration:

For two weeks or for the first week:
Monday August 29th 14.15 - 17.15

For the second week:
Monday September 5th 09.00 - 09.45

Registration to take place at the Department of Naval Architecture (see map).

Chairmen:

Lars Larsson	30/8	Constantin Gallin	5/9
Gilbert Dyne	31/8	Jan Stefenson	6/9
C A Johnsson	1/9	Herbert Schneekluth	7/9
Lars Collin	2/9	Jan Hee	8/9

Date	Time	Title	Lecturer
Mon 29/8	14.15-17.15	Registration (first and second week)	
	19.30-	Reception	
Tue 30/8	09.00-	Introduction, organization	C Falkemo
	10.00-10.45 11.00-11.45	Ship resistance reduction - a survey	L Larsson
	11.45-13.00	Lunchbreak	
	13.00-13.45 14.00-14.45	Empirical optimization of hull forms including the influence of trim and draught	Å Williams
	15.00-15.45	Resistance reduction by turbulence control	M Landahl
	18.30-20.30	Team design subject	

Coffee is served every day during the breaks 09.45 - 10.00 and 14.45 - 15.00.

Date	Time	Title	Lecturer
Wed 31/8	08.00-08.45 09.00-09.45	Resistance increase due to hull roughness and fouling, including long-range effects of new paints	R L Townsin
	10.00-10.45 11.00-11.45	Resistance increase due to wind, waves and steering	W van Berlekom
	11.45-13.00	Lunchbreak	
	13.00-13.45 14.00-14.45	Ship propulsion improvement - Principles and a survey of alternative propeller systems	G Dyne
	15.00-15.45	Low speed propellers	C Pronk
	16.00-16.45	Controllable pitch propellers	O Björheden
	18.30-20.30	Team design subject	
Thu 1/9	08.00-08.45	Ducted propellers including ducts ahead of the propeller	W van Gent
	09.00-09.45	Propeller and vane-wheel, contra rotating propeller	O Grim
	10.00-10.45 11.00-11.45	Propellers with reduced tip loading and unconventional blade form and blade sections	C A Johnsson
	11.45-13.00	Lunchbreak	
	13.00-14.00 14.00-14.45	Visit to Swedish Maritime Research Center SSPA	
	15.00-15.45 16.00-16.45	Propulsion in service conditions Influence of hull detoriation, waves, wind and steering	K J Minsaas
	18.30-20.30	Team design subject	

Date	Time	Title	Lecturer
Fri 2/9	08.00-08.45 09.00-09.45	Comparison of power plant performance	L Collin
	10.00-10.45 11.00-11.45	Present and future marine fuels	R V Thompson
	11.45-13.00	Lunchbreak	
		End of week 1	
Sat 3/9	Free		
Sun 4/9	Free		
Mon 5/9	09.00-09.45	Registration (second week)	
	10.00-10.45 11.00-11.45	Economic and technical studies of fuelsaving ships	C Gallin
	11.45-13.00	Lunchbreak	
	13.00-13.45 14.00-14.45	Advanced marine power transmission	H M Hiersig
	15.00-17.00	Team design subject	

Date	Time	Title	Lecturer
Thu 8/9	08.00-08.45 09.00-09.45	Maintenance and repair; Ship's manning	H Glimberg
	10.00-10.45	Automatisation and fuel consumption	G Mau
	11.00-11.45	Paint and docking	A Milne
	11.45-13.00	Lunchbreak	
	13.00-13.45 14.00-14.45	Electricity production on board	H Klintorp
	15.00-15.45	Propeller hull interaction, derating engines and propeller layout	J Hee
	16.00-16.45	Ship routeing	J A Spaans
	18.30-20.30	Team design subject	
Fri 9/9	08.00-08.45 09.00-09.45 10.00-10.45 11.00-11.45	Discussion on team design works including conversation of existing ships	H Schneekluth C Gallin C O Larsson S Ekholm H Pöls J Stefenson T Gell K Sallén
	11.45-13.00	Lunchbreak	
	13.00-13.45	Closing speach	C Falkemo

Date	Time	Title	Lecturer
Tue 6/9	08.00-08.45 09.00-09.45	Interaction between propulsion machinery and hull structure	J Stefenson
	10.00-10.45 11.00-11.45	Wind as an aid for ship propulsion	P Schenzle
	11.45-13.00	Lunchbreak	
	13.00-13.45 14.00-14.45	The role of economics in ship design for fuel economy	H Benford
	15.00-15.45 16.00-16.45	Design choice from shipowners point of view	E Vossnack
	18.30-20.30	Team design subject	
Wed 7/9	08.00-08.45	Propulsion plant choice for a medium-sized container ship	C Gallin
	09.00-09.45	Which bunkerfuel engines for smaller ships?	C Gallin
	10.00-10.45	The optimisation of the principal dimensions of the ship	H Schneekluth
	11.00-11.45	The influence of ship operation on design	I L Buxton
	11.45-13.00	Lunchbreak	
	13.00-13.45 14.00-14.45	Fuel quality and treatment on board	O Samuel
	15.00-15.45 16.00-16.45	Coal fired ships of new generation	E Derdini
	18.30-	Mould-Loft dinner	

WEST EUROPEAN GRADUATE EDUCATION IN MARINE TECHNOLOGY

EIGHTH GRADUATE SCHOOL 1983

SHIP DESIGN FOR FUEL ECONOMY

List of participants

August 29th - September 9th

Name	Title	Organization
Burns, David	Design Engineer	Harland & Wolff Ltd, Belfast Northern Ireland
Costa, Ronaldo D'Antonio	Naval Engineer	Verolme Estaleiros Reunidos do Brasil S/A, Rio de Janeiro
Craveiro, Joad	Mechanical Engineer	Rinave Portuguesa, Blommenholm, Norway
Faber, E G	Dr	Sulzer, Bros. AG, Winterthur, Switzerland
Goncalves, Franklin de O	Naval Architect	Det Norske Veritas, Hövik, Norway
Lee, S H	Naval Architect	Hyundai Heavy Industries Co, London, Great Britain
Lindbom, Björn		Chalmers University of Technology, Gothenburg, Sweden
Loldrup, Henrik	Naval Architect	Burmeister & Wain Shipyard, Copenhagen, Denmark
Lorén, Styrbjörn	Engineer	Götaverken Motor AB, Gothenburg, Sweden
Mildner, Folke	Dr	Escher Wyss Gmbh, Ravensburg, Federal Republic of Germany
Park, Y D		Hyundai Heavy Industries Co, London, Great Britain
Park, Y K		Korea Shipbuilding N Eng Co, Pusan, Korea
Pettersson, Bo		Rederiaktiebolaget Gotland, Visby, Sweden
Sangberg, Bengt	Naval Architect	Dannebrog Rederi AS, Rungsted Kyst, Denmark
Son, B R		Korea Shipbuilding N Eng Co, Pusan, Korea
Suply, Denis	Engineer	Daz de France, Paris, France
Thoresson, Anders	Naval Architect	J W Berg AB, Öckerö, Sweden
Ventillard, Pascal		Chantiers Nord Mediterranee, La Seyne, France
Verleg, J H George	Naval Architect	Wysmuller Engineering BV, Ijmuiden, The Netherlands
Yang, Seung-Il		Ship Research Station, Kimm, Chungnam, Korea

Tot. 20

August 29th - September 2nd

Arnesen, Knut		Sig Bergesen D Y & Co, Oslo, Norway
Barrett, M	Naval Constructor	Ministry of Defence, Bath, Great Britain
Baumes, Gerard	Engineer	Chantiers de L'Atlantique, St Nazaire, France
Caby, Raymond	Naval Architect	Boelwerf N.V., Temse, Belgium
Clayton, J A	Research Manager	Stone Manganese Marine Ltd, Merseyside, Great Britain
Eskola, Harri	Scientific Officer	Helsinki University of Technology, Espoo, Finland
Grothues-Spork, Hermann	Dr	Berlin Ship Model Basin VWS, Berlin, Germany
Hansen, Kurt Erling	Mechanical Engineer	Alpha Diesel, Fredrikshavn, Denmark
Heed, Thomas		FMV, Stockholm, Sweden
Hämäläinen, Raimo	Naval Architect	Wärtsilä Turku Shipyards, Turku, Finland
Kalianos, Dimitris	Engineer	Penisular Maritime Ltd, London, Great Britain
Kingsley-Smith, T	Naval Architect	Blue Star Line Ltd, London, Great Britain
Kim, Doo Gyoon	Manager	Samsung Shipbuilding and Heavy Ind Co, Kyung Nam, Korea
Kurimo, Risto		Wärtsilä Ab, Helsinki Shipyard, Helsinki, Finland
Kämäräinen, Jorma	Naval Architect	Valmet Corporation, Helsinki Shipyard, Helsinki, Finland
Lapierre, H	Naval Architect	Chantiers du Nord et de la Mediterranee, Dunkerque, France
Lööf, Erik		Götaverken Arendal AB, Gothenburg, Sweden
Marshall, J R	Senior Lecturer	Plymouth Polytechnic, Plymouth, Great Britain
Olkinuora, Pauli	Naval Architect	Hollming Ltd, Rauma, Finland
Olsson, Thomas		KaMeWa, Kristinehamn, Sweden
Rantanen, Antti	Research Scientist	Technical Research Centre Of Finland, Espoo, Finland
Rydén, Lars		FMV, Stockholm, Sweden
Tangvald, Jon		A/S Mosvolds Rederi, Farsund, Norway
Tuovinen, Pekka		Helsinki University of Technology, Espoo, Finland
Vainio, Pyry Kalvero	Naval Architect	Hollming Ltd, Rauma, Finland

Tot 25

Name	Title	Organization
September 5th - September 9th		
Bakke, Hans-Jacob		Sig Bergesen D Y & Co, Oslo, Norway
Brown, Owen S		British Shipbuilders, Tyne & Wear, Great Britain
Cottin, Jean Lue	Engineer	Chantiers de L'Atlantique, St Nazaire, France
Evans, John	Research Officer	British Ship Research Associat, Tyne & Wear, Great Britai
Flarup, Jens Chr		Odense Steel Shipyard Ltd, Odense, Denmark
Haagensen, Per		A/S Mosvolds Rederi, Farsund, Norway
Hopeavuo, Timo	Naval Architect	Valmet Corporation, Helsinki Shipyard, Helsinki, Finland
Hughes, C N	Engineer	CP Ships, London, Great Britain
Kirk, M		British Shipbuilders, Tyne & Wear, Great Britain
Kylä-Kaila, Teuvo	Engineer	Wärtsilä Ab Turku Shipyards, Turku, Finland
Laakso, Jarmo	Naval Architect	Wärtsilä Ab Helsinki Shipyards, Helsinki, Finland
Lefebvre	Engineer	Chantiers du Nord et de la Mediterranee, Dunkerque, Franc
Poulsen, Jörgen	Engineer	Aalborg Vaerft, Aalborg, Denmark
Rusten, Jon	Ships Engineer	Fred Olsen & Co, Oslo, Norway
Sipilä, Heikki	Naval Architect	Wärtsilä Ab Turku Shipyards, Turku, Finland
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Ylhäinen, Pertti	Naval Architect	Hollming Ltd, Rauma, Finland

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WEST EUROPEAN GRADUATE EDUCATION
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EIGHTH SCHOOL 1983
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SHIP DESIGN FOR FUEL ECONOMY

SCHOOL STAFF

H Benford	University of Michigan Dept of Naval Architecture & Marine Engineering North Campus, Ann Arbor MI 48109 USA	G Dyne	Swedish Maritime Research Center SSPA Box 24001 400 22 Gothenburg Sweden
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WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

SHIP RESISTANCE REDUCTION - A SURVEY

by

Lars Larsson

Swedish Maritime Research Centre SSPA

Abstract

The present paper is an introduction to the resistance session of the 8th WEGEMT School on Ship Design for Fuel Economy. Its main purpose is to introduce the various components of the total resistance of a ship, i e to explain the physical background of each component, to assess its relative importance and to discuss (very briefly) the possibilities of reducing it. The main emphasis of the paper is placed on the resistance components of a ship moving in calm, unrestricted water.

1. Introduction

The simplest way of dividing the total fluid force on a moving ship is to split it into forces acting tangentially to the hull and forces acting normally. Of course these may in turn be divided into hydrodynamic forces, acting on the submerged part of the hull, and aerodynamic forces, acting above the water. This division, although physically correct, is not convenient for describing means of reducing the resistance of a ship. For this purpose, it is better to make a finer subdivision, more related to the physical phenomenon causing the resistance component in question. Frequently, the hydrodynamic components of resistance for a bare hull (i.e. without appendices), moving in calm, unrestricted water are considered the basic resistance components, the other ones being considered "additional". We shall in the following use this division.

2. The resistance components of a bare hull moving in unrestricted, calm water.

In Fig:s 1 a, the resistance of a 100 000 m^3 tanker at 16 knots has been split into components, and the same components for a 29 000 m^3 Ro-Ro ship at 21 knots are shown in Fig 1b. The components, and the possibilities of minimizing them will now be discussed one by one.

2.1 The flat plate resistance.

Ever since the early days of model testing, more than 100 years ago, ship hydrodynamicists have used the resistance of an "equivalent" flat plate as a means of scaling the, usually, most important part of the resistance, namely the one caused by viscosity. By "equivalent" is usually meant a plate of the same area, and moved at the same Reynolds number, R_n, as the ship.

$$R_n = \frac{V \cdot L}{\nu} \qquad (1)$$

where V is the speed, L is the length and ν the kinematic viscosity.

The plate should be hydraulically smooth, i.e. the roughness elements should be sufficiently small that their influence on the resistance can be neglected.

Since the only way longitudinal forces can be exerted on the plate is via tangential stresses the introduction of the plate is a means of separating the two types of forces from one another. However, this is not quite correct, since the tangential forces on the three-dimensional ship form will depart to a certain extent from the ones on the plate, thereby giving rise to the component "form effect upon skin friction". The flat plate resistance may thus be considered as the tangential, or frictional, component, without the influence of the shape of the hull, i.e. without any influence of the pressure distribution.

The resistance, R_{F_0}, of the plate may be computed from

$$R_{F_0} = \frac{A}{L} \int_0^L \tau_w(x)\,dx \qquad (2)$$

where $\tau_w(x)$ is the wall shear stress, L is the length and A is the wetted surface. According to Stokes' law, τ_w may be obtained from

$$\tau_w = \mu \frac{\partial u}{\partial y} \qquad (3)$$

where μ is the dynamic viscosity, u is the fluid velocity relative to the surface and y is the distance from the surface.

From (2) and (3) it appears that there are three ways of reducing the resistance for a plate of length L, namely by reducing

- the wetted surface, A
- the dynamic viscosity, μ
- the velocity gradient, $\partial u/\partial y$

The first one of these possibilities is trivial, but nevertheless of great practical importance. We shall not consider this any further.

In principle, the dynamic viscosity close to the surface may be changed either by changing the properties of water, or by introducing another fluid, separating the water from the surface. The first possibility has been explored by heating the surface, thereby reducing the viscosity of the adjacent water. This has been shown to have a very significant effect on the resistance, but the main reason for the positive effect is not the change in μ, but in $\partial u/\partial y$, caused by the stabilizing effect on the velocity profile. Very promising results have been obtained also by introducing air as a convenient medium for separating the water from the surface. Since the dynamic viscosity of air is only about 1.8×10^{-5} kg/ms as compared with about 10^{-3} kg/ms for water, considerable changes are theoretically obtainable. In practice about 80% reductions have been achieved.

In Fig. 2 the velocity profile in a laminar boundary layer at $R_n = 10^6$ is compared with the turbulent profile at the same Rn (based on the length from the virtual origin of the turbulent boundary layer). It is seen that in spite of the fact that the laminar boundary layer is thinner, its velocity gradient, $\partial u/\partial y$, close to the surface, is considerably smaller. According to (3) the laminar flow must then have a much smaller wall shear stress. From Fig. 3 it appears that <u>if</u> the laminar flow could be maintained at sufficiently high Reynolds numbers, the difference in skin friction coefficient C_{F_0}, compared with the turbulent case, would be tremendous.

$$C_{F_0} = \frac{R_{F_0}}{0.5\rho V^2 A} \qquad (4)$$

where ρ is the water density.

A large number of methods have been tried to reduce the turbulence in the boundary layer, and very promising results have been obtained by some of them. Thus, adding heavy polymers, even at very small concentrations, has been shown to reduce the resistance in laboratory experiments byas much as 90%. Other possibilities, which have been tried, are to design a special surface texture or to make the surface flexible. The latest idea is to manipulate the large scale eddies in the outer part of the boundary layer by introducing a device similar to a tandem wing

in that region. 20% net drag reduction has been achieved by this means.

The merits and demerits of the various drag reduction methods will be discussed in more detail in Professor Landahl's lecture.

2.2 Resistance due to roughness.

In Fig. 4 the measured roughness height of a large number of different ships of varying age is shown graphically [1]. The curve represents an approximate average of the results, which are in reality spread about ±50%. It is seen, that on the average, new ships have a roughness height, k, of about 100 μ, while after several years the surface has deteriorated to more than 600 μ. This holds, even if the hulls have been regularly maintained (but not shot-blasted).

The resistance increase, ΔC_F, associated with such roughness heights may be estimated from a diagram produced at SSPA [2], see Fig. 5. Assuming a length L of 100 m and a speed of 7 m/s, the Reynolds number becomes 7.10^8, while L/k is $100 \cdot 10^4$ (100 μ) and $16 \cdot 10^4$ (600μ). This yields C_{F_0}:s equal to $1.72 \cdot 10^{-3}$ resp. $2.07 \cdot 10^{-3}$. Since the "smooth" C_{F_0} at this Reynolds number is only $1.59 \cdot 10^{-3}$ the ΔC_F:s become $0.13 \cdot 10^{-3}$ resp. $0.48 \cdot 10^{-3}$, representing an increase of 8% resp. 30 %. For the tanker in Fig. 1a this corresponds to additional bunker costs of about $150 000 resp. $600 000. The economical importance of the roughness resistance is thus tremendous.

$/year

In the past, roughness has been considered in several different ways, the simplest one being the addition of a ΔC_F equal to $0.4 \cdot 10^{-3}$ for the service condition of a new ship. A very simplified formula for ΔC_F, Bowden's, is used today in the ITTC performance prediction procedure. More exact methods are, however, available, and the discussion of these and of a number of other aspects of the problem such as roughness geometry and drag, fouling, new paints, optimum docking intervals etc. will be subjects of Dr Townsin's lecture.

Here we will only briefly discuss one of the new methods, recently developed at SSPA, and not yet reported internationally.

According to classical literature on roughness effects, for instance Clauser [3], the local skin friction coefficient, c_f, in a turbulent

boundary layer may be obtained from the relation

$$\frac{2}{\sqrt{c_f}} = A \log \frac{u_\tau \delta^*}{\nu} + B - \Delta B \qquad (5)$$

where

$$c_f = \frac{\tau_w}{\frac{1}{2}\rho u_e^2} \qquad (6)$$

and u_τ is the shear velocity

$$u_\tau = u_e \sqrt{\frac{c_f}{2}} \qquad (7)$$

δ^* is the boundary layer displacement thickness, u_e is the velocity at the edge of the boundary layer, A and B are constants and ΔB represents the influence of roughness.

If δ^* and c_f can be measured for varying roughness Reynolds numbers, $u_\tau k/\nu$, ΔB can be determined for the particular type of roughness and the formula (5) may be used in a turbulent boundary layer calculation method for obtaining the skin friction. This is the procedure adopted at SSPA.

Karlsson Pr/ROUF KARL 1978

In the latter part of the seventies a device for measuring the frictional force on a surface was developed. This device is of the so called "floating element" type and its principle is explained in Fig. 6. The force so measured on a 200 x 200 mm^2 plate, cut out from the wall, but flush with it. When studying a particular type of roughness, not only the cut out part, but also the surrounding wall have to have the same roughness. In order to determine ΔB, the displacement thickness δ^* is obtained from simultaneous velocity profile measurements.

Pr/ROUF3 1978

Having obtained ΔB, this is introduced into a three-dimensional turbulent boundary layer calculation method developed by Larsson [4] and the local increase in c_f as compared with the smooth case can be computed. Recently, this procedure has been used for obtaining the local influence of roughness [5]. By such calculations the economics in smoothing various parts of the hull can be studied. In Fig. 7 the result of a calculation for a tanker is shown. The graph shows the reduction in C_F which is obtained by extending the smooth part (which starts at the bow)

another metre in the downstream direction. It is seen that the economics of grinding the surface is reduced aftwards. The reduction is not too large, however.

2.3 The form effect on skin friction.

On a flat plate, the pressure and the velocity outside the boundary layer are very nearly constant. This is not the case for a three-dimensional body like a ship. If it had not been for the small displacement effect of the boundary layer, the stream lines passing a flat plate would have been straight, while outside a ship hull the stream lines are pushed aside considerably, thereby giving rise to large pressure and velocity gradients. One example of this is given in Fig. 8, which shows some results for the SSPA Model 720. The distributions of pressure and external velocity (at the edge of the boundary layer) along two lines are given in the same figure. In the bow and stern regions the pressure is high and the velocity low while the opposite is true near the fore and aft shoulders of the hull. For this particular case the maximum speed is about 10% higher than the undistributed velocity.

The uneven velocity distribution gives rise to an uneven skin friction distribution. In regions, where the external velocity is high, the skin friction is also high, and vice versa. This is apparent from Fig. 9, which shows the local skin friction coefficient, c_f, for the same lines as shown in Fig. 8. For comparison the flat plate C_{F_0} is also given.

Generally, the average velocity around the hull is larger than the undistributed one, which means that the total skin friction coefficient, C_F, will be larger than the one for a flat plate. This effect is, however, usually quite small, only a few per cent.

2.4 The viscous pressure resistance.

In a potential flow without a free surface the pressure near the stern of a body will balance exactly the pressure at the bow, yielding zero pressure resistance (d'Alembert's paradox). This is not true in a viscous flow, where the boundary layer near the stern will disturb the pressure recovery, resulting in a net force directed backwards. The nature of this disturbance is explained in more detail in Fig. 10. In a potential flow the stream-lines will follow more closely the hull

contour when the sectional area of the hull is reduced backwards. The aftermost point is a stagnation point. Since the velocity is zero at this point the stream-line passing through it can be bent sharply. Even just outside the hull quite strong curvatures are found.

In the real flow the boundary layer causes a reduction in velocity near the hull, and since this layer is quite thick close to the stern the velocity loss is considerable. For continuity reasons the stream-lines will therefore depart from the hull contour and move on backwards without strong curvatures. The variations in pressure will then be much smaller, and no stagnation point will appear. In Fig. 8a the pressure loss at the stern of the SSPA Model 720 is indicated.

Separation of the flow will also cause a change in the pressure. If massive two-dimensional (reverse flow) separation occurs, the stream-lines near the stern will be displaced even further out and the pressure recovery will be very small. This causes a large pressure resistance. Fortunately, however, this type of separation, also called "bubble separation", is not very common. Experiments [6] indicate that the regions where reverse flow occurs are quite small even for very full forms. More common is the three-dimensional type of separation, leading to the formation of bilge vortices. These will also give rise to a pressure drag, although probably smaller than the one caused by the reversed flow.

In prediction methods based on towing tank data, both components just described, i.e. the form effect on the skin friction and the viscous pressure drag, are lumped together in a form drag, represented by a form factor k. This factor is the relative increase in resistance due to the two components, as compared to the (smooth) flat plate resistance. There are several ways of computing the form factor based on emperical ralations, but these are not very useful for minimization of the two components. To be able to do this theoretically, boundary layer calculations are necessary. Methods for calculations of these types are explained in the first half of Dr Gadd´s lecture. It is shown that there are now available several boundary layer methods, which work well over the main part of the hull, while near the stern only very few new methods of a more complex nature can be used. For an evaluation of different methods see also Larsson [7]. It should be noted that only very recently calculations of this type have been used for systematic optimization studies [8]. Such calculation can, however, be expected to increase dramatically in the near future.

2.5 The wave pattern resistance.

The ship moving along the free surface represents a pressure disturbance, which causes waves. The waves contain an energy, which is equal to the work done on the water by a force equal but opposite to the wave resistance component. This component appears through normal forces, i.e. it is a pressure drag. It is also a potential flow phenomenon, so d´Alembert´s paradox does not apply in the presence of a free surface.

As described in 2.4 the viscous part of the flow (i.e. the boundary layer and wake), may modify the pressure distribution also in the case of a free surface. Thus, the wave formation and the wave resistance are affected. Contrary to the case of 2.4, however, viscosity may in this case be beneficial. The reduction in the stern pressure causes a reduction in the stern waves and may affect the flow as if the hull was elongated (the displacement thickness extends behind the ship).

Investigations taking viscosity into account are relatively rare, but the amount of research in this field, neglecting viscosity, is tremendous. In fact, the wave resistance problem seems to be the most popular one for theoretical work in ship hydrodynamics, and already in 1898 a relevant theory was presented (Michell´s theory). This theory, which is frequently referred to even today, is based on the assumption that the hull is thin and the waves small. Without modifications it seems to be of limited interest for computing the wave resistance of real ships but it has been the basis of many attempts to develop a theory for wave resistance minimization. In principle, it should be much easier to achieve great reductions in the wave resistance than in the viscous resistance. The reason for this is that the various parts of the hull, each one acting as a pressure disturbance, may be designed so that the resulting waves interact in a beneficial way at a desired speed. In fact it is quite possible to design hulls for which the waves from the various parts completely eliminate each other, resulting in zero wave resistance. Such hulls are, however, extremely impractical and, since viscosity has not been considered, their wave resistance would not be zero in reality. The theory has, however, been useful in designing bulbs at the bow and, recently, at the stern.

Wave resistance calculations, using simpler theories, such as the ones

developed from Michell's, as well as considerably more advanced ones, developed during the computer age, will be described in more detail in the second half of Dr Gadd's lecture.

2.6 The wave breaking resistance.

If the waves at the bow get too steep they break, see Fig. 11, and the regular orbital motions of the water particles are partly converted into small irregular eddy motions. The corresponding energy is not transferred away from the hull in the wave system, but is found in the form of eddies in the wake of the ship.

The phenomenon of wave breaking has been known for about 15 years now, but no good theory for its prediction has been presented. An assumption, which is useful, and probably not too approximate, is that the sum of the wave pattern and wave breaking resistance components is equal to the computed wave pattern resistance. Thus, minimization of the sum of the two components can be made using wave resistance theories.

3. Other resistance components

Other resistance components of hydrodynamic origin are the ones due to spray generation, drag of appendages (brackets, bilge keels, etc), restricted waters, sea waves and steering. The aerodynamic component is of course due to the wind load on the part of the ship which is above the water.

Spray resistance and appendage drag are usually of relatively little importance except for very fast ships, like naval vessels. Neither of these, nor the restricted water resistance, will be dealt with in this course.

3.1 The resistance due to sea waves

The added resistance of a ship in a seaway stems essentially from two sources. For short waves in particular, and when the hull is moving in head seas, the waves will be reflected from the bow. The corresponding resistance component is quite dependent on the fullness of the ship. Faltinsen considered this component proportional to a fullness coefficient

$\overline{\sin^2\theta}$, defined as

$$\overline{\sin^2\theta} = \frac{1}{B}\int_{-B/2}^{B/2} \sin^2\theta \, dy \qquad (8)$$

where B is the beam and θ is the waterline angle. To minimize the component fine entrances are thus required.

The wave reflection resistance is, however, usually of less importance than the resistance component due to the ship motions. When the ship moves vertically relative to the surface, waves are generated and radiated. As for the waves of the wave pattern, these must give rise to a resistance component. Good estimates of this component can be made, knowing the added mass and damping coefficients plus the vertical relative velocity of each section of the hull. These quantities may be obtained from strip theory.

Typical figures for the added resistance due to sea waves are of course extremely dependent on the ship's route, speed and dimensions, but 10 % must be considered quite low. In most cases 15-30 % may be realistic.

3.2 The air resistance

The aerodynamic resistance when the ship moves in still air is usually quite small, but since the resistance is proportional to the relative velocity squared, considerable forces are generated in strong head winds. In fact, since the projected area of the hull and superstructure increases rapidly when the angle of incidence relative to the ship's longitudinal direction is increased, the maximum drag may occur at angles of 20-30°. The largest air resistance relative to the hydrodynamic resistance appears for slow ships. At 20 m/s as much as 40 % may be found for such ships. Streamlining of the superstructure might, however, reduce this figure considerably.

3.3 The resistance due to steering

The resistance increase due to steering is due partly to the induced resistance on the hull when it is not moving in its longitudinal direction, partly to the induced resistance on the rudder when at an angle of attack. For a directionally stable ship this resistance component is negligible in calm water, while for an unstable ship it may amount to a few per cent of the calm water resistance. In a seaway, and when the wind is

strong, larger rudder angles and drift angles of the hull may be encountered, giving rise to larger resistance increases.

The resistance components due to wind, waves and steering will be discussed by Mr van Berlekom.

4. Conclusion

The total resistance of a ship is composed of a large number of components, each one being possible to reduce by theoretical or experimental investigations. In this course all components will be discussed in some detail and the participants will hopefully be given the knowledge necessary for optimizing future designs.

5. References

[1] TOWNSIN R L, BYRNE D, MILNE A & SVENDSEN T: "Speed Power and Roughness: The Economics of Outer Bottom Maintenance". The Naval Architect, RINA, Sept 1980

[2] JOHANSSON L-E: "Measurements and Calculations of Frictional Resistance. Smooth Plate and Kockum's Painting No 4". SSPA Report No 2357-2, 1980 (in Swedish)

[3] CLAUSER F H: "The Turbulent Boundary Layer". Advances in Applied Mechanics 4, pp 1-51, 1965

[4] LARSSON L: "Boundary Layers of Ships". PhD Thesis, Chalmers University of Technology, 1975. Also available as Reports Nos 44, 45, 46, 47 and 49 of the Swedish State Shipbuilding Experimental Tank, SSPA

[5] JOHANSSON L-E: "Ship Resistance - The Importance of Roughness on the Various Parts of the Hull". Paper presented at the Colloquium on Fuel Efficient Ships, organized by SSPA 29 April 1982 (in Swedish)

[6] GROTHUES-SPORK H: "Separation Phenomena on Full Afterbodies". Research Center of the German Shipbuilders, Report No 50/75, 1975 (in German)

[7] LARSSON L: "The SSPA-ITTC Workshop on Ship Boundary Layers 1980". SSPA Report No 90, 1981

[8] Contributions of Nagasaki Experimental Tank to Design of Hull Form and Propeller. Part II: Study on Viscous Resistance and its Application. Nagasaki Experimental Tank, Mitsubishi Heavy Industries, Japan 1983

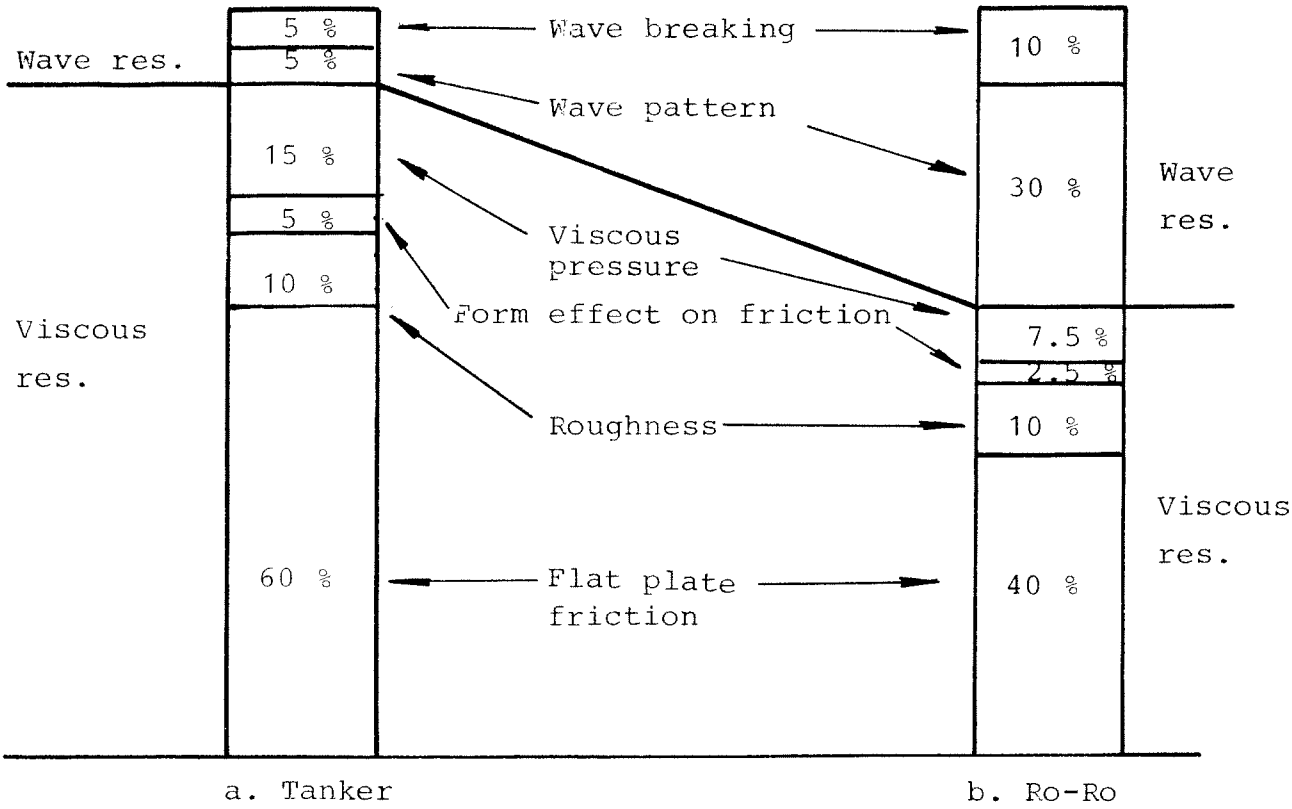

FIG 1 RESISTANCE COMPONENTS OF A 100 000 M^3 TANKER (16 KNOTS) AND A 30 000 M^3 RO-RO SHIP (21 KNOTS)

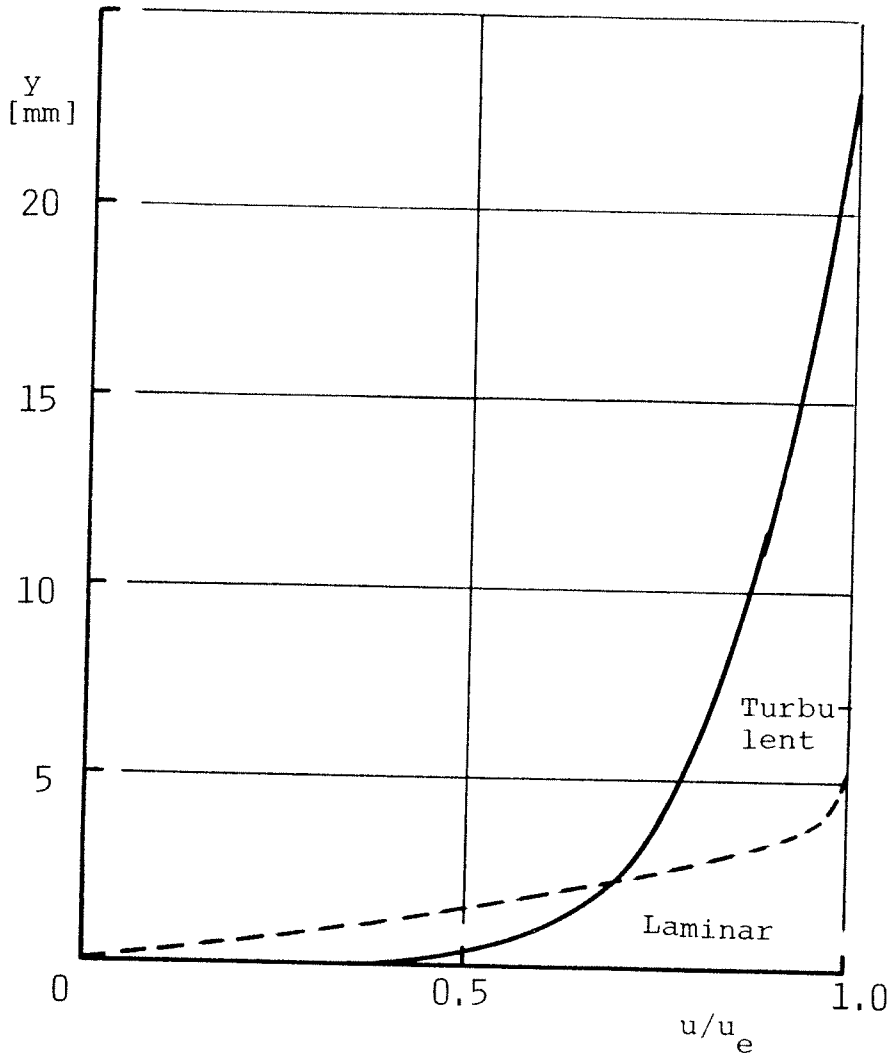

FIG 2 LAMINAR AND TURBULENT VELOCITY PROFILES
(1 M FLAT PLATE, 1 M/S, WATER)

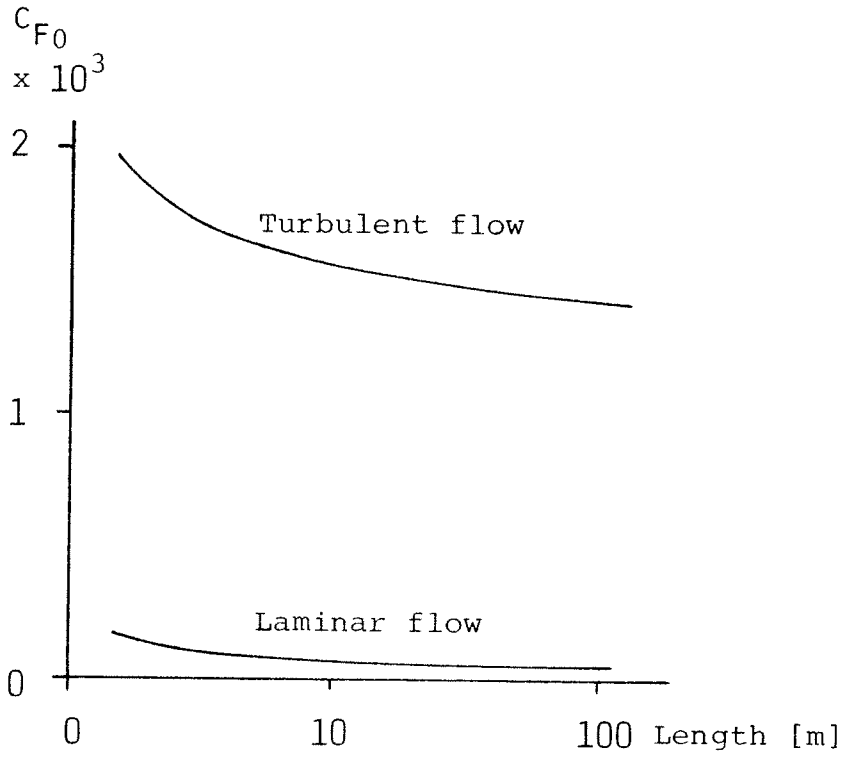

FIG 3 THE VARIATION WITH PLATE LENGTH OF THE TOTAL RESISTANCE COEFFICIENT. WATER VELOCITY 7.5 M/S.

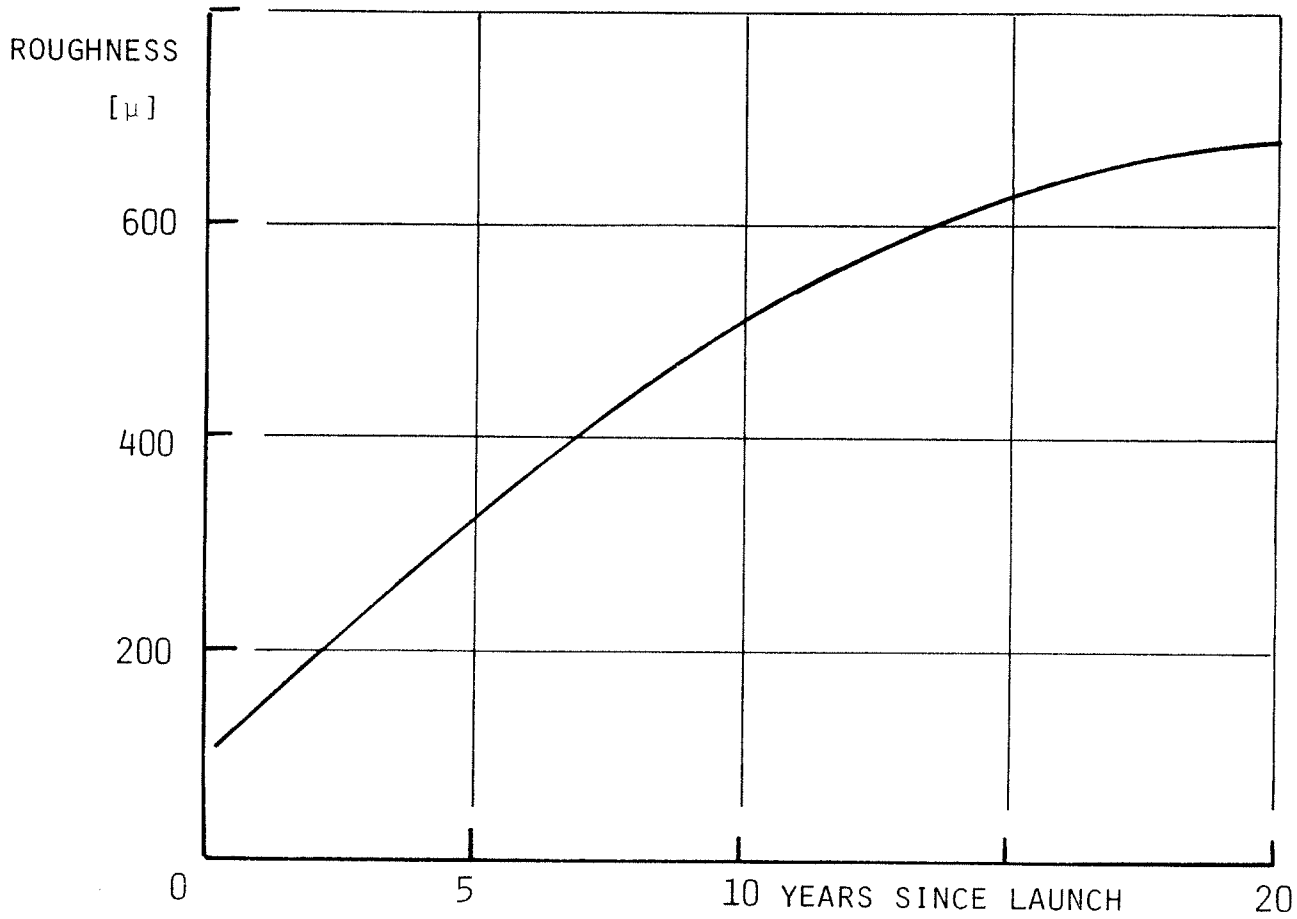

FIG 4 MEAN CURVE FOR HULL ROUGHNESS AS A FUNCTION OF AGE. MEASUREMENTS BY TOWNSIN ET AL [1]

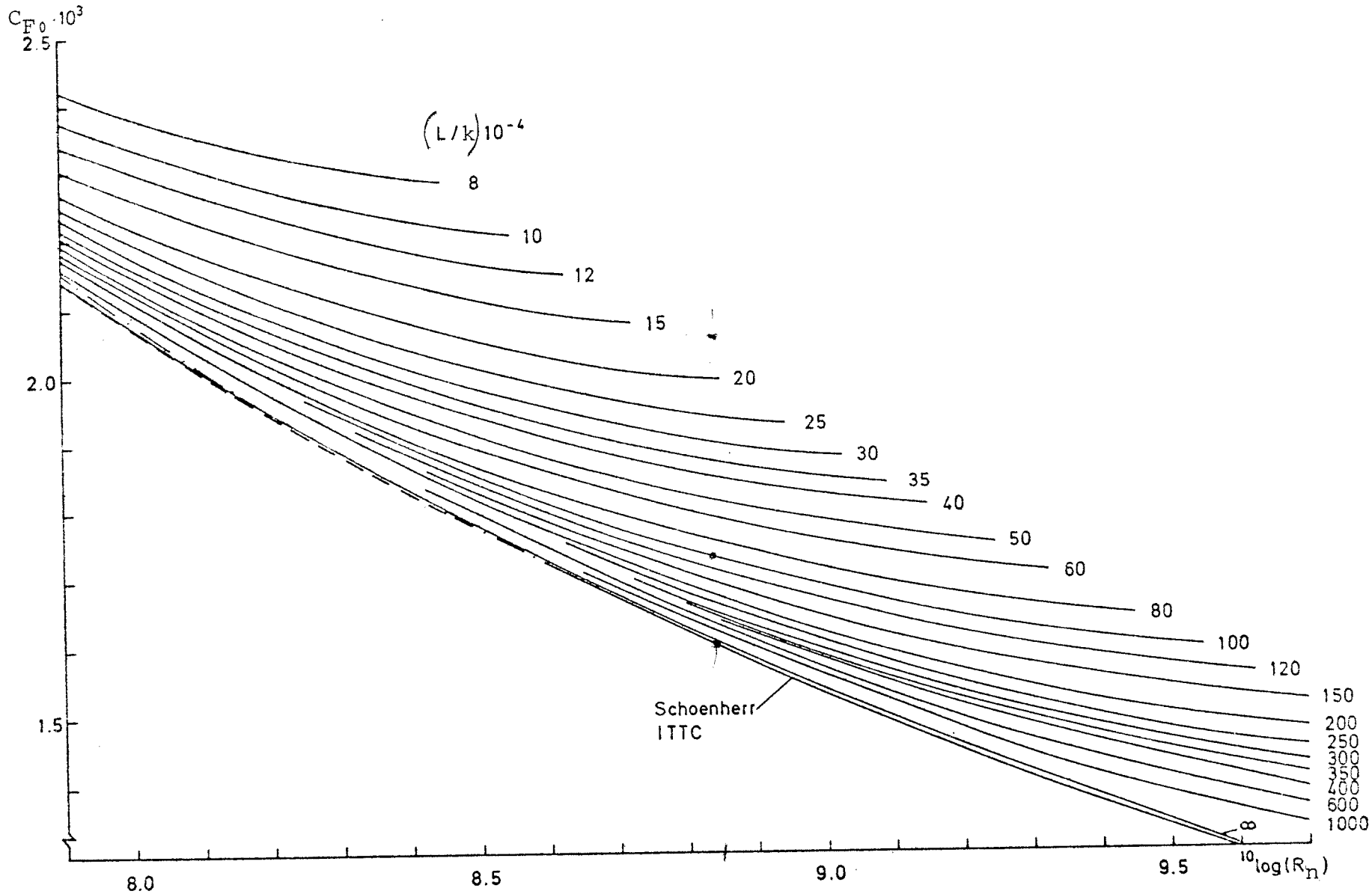

FIG 5 FLAT PLATE RESISTANCE FOR ROUGH SURFACES BASED ON MEASUREMENTS AND CALCULATIONS AT SSPA [2]

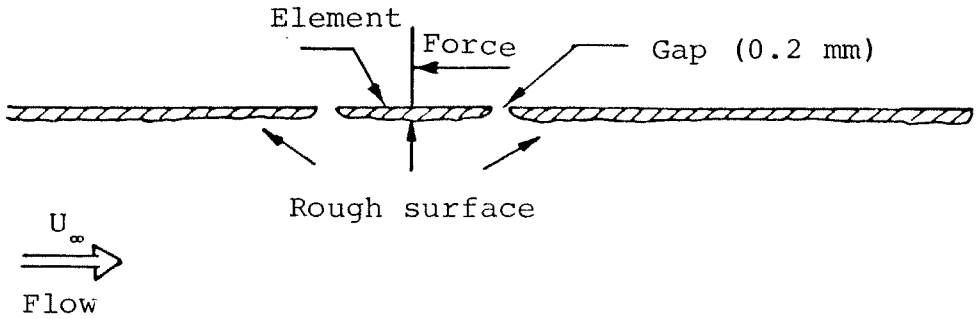

FIG 6 THE FLOATING ELEMENT BALANCE (PRINCIPLE)

Karlman
Göteborg

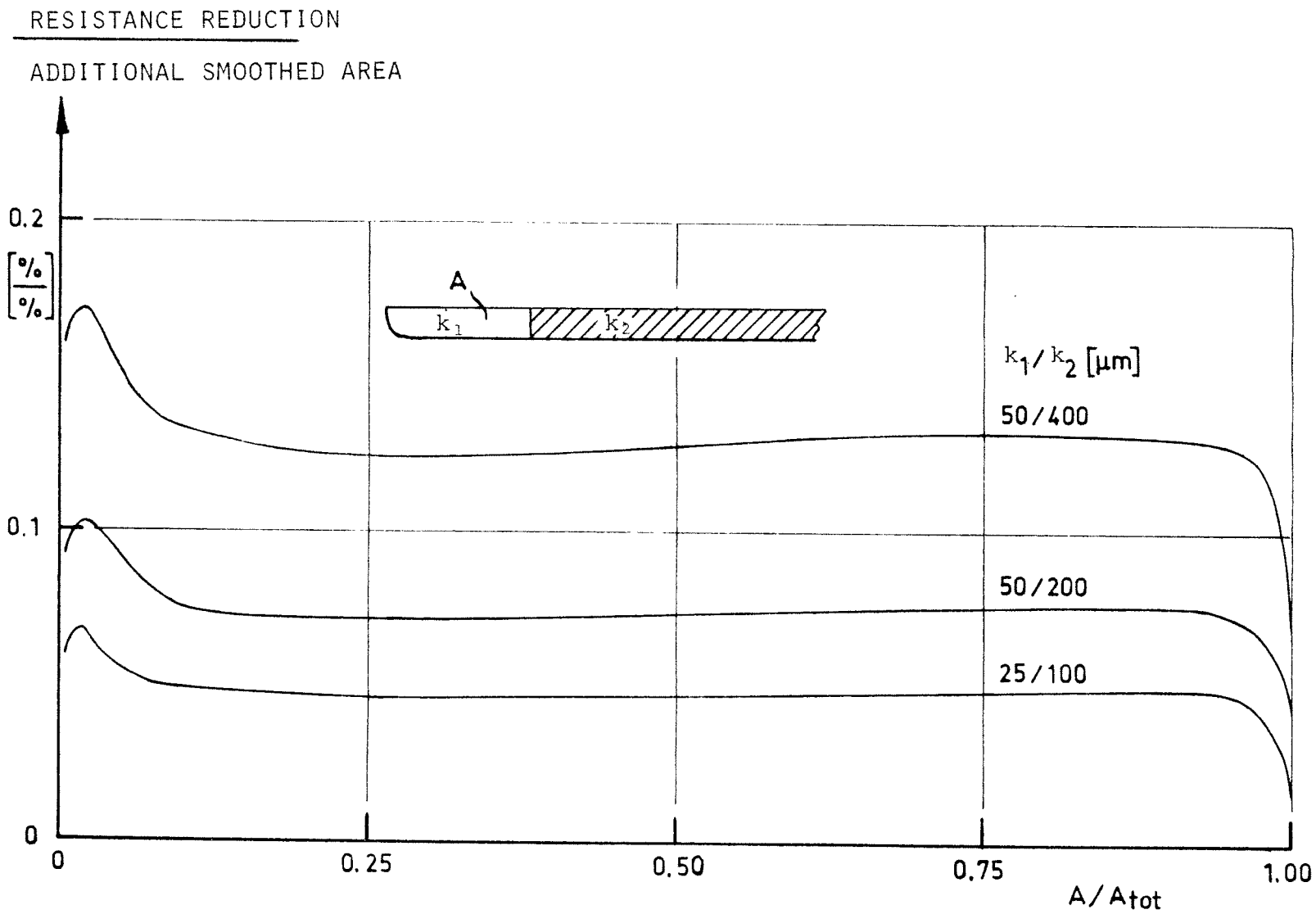

FIG 7 RESISTANCE REDUCTION PER ADDITIONAL SMOOTHED AREA UNIT. FROM [5]

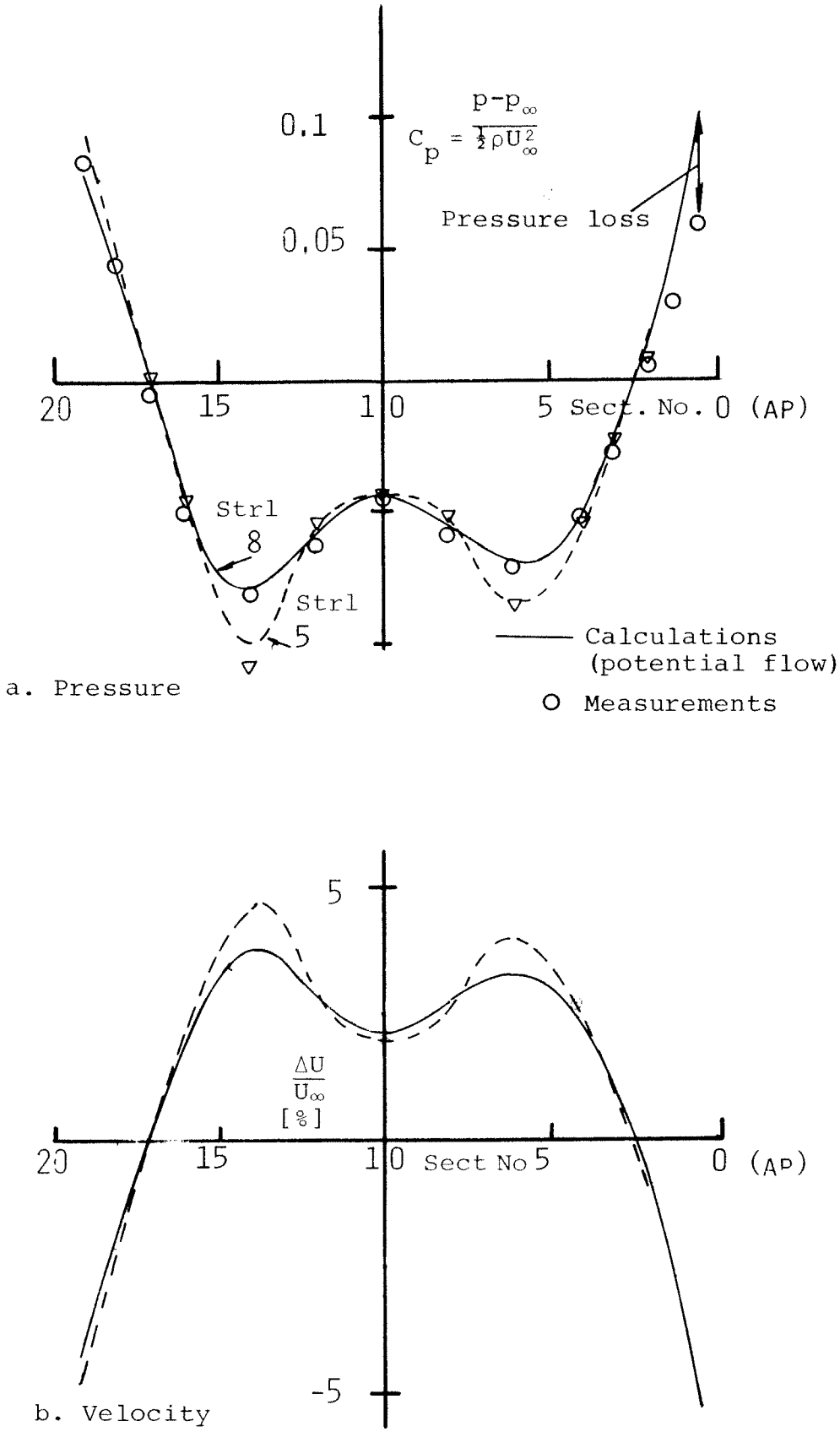

FIG 8 PRESSURE AND VELOCITY DISTRIBUTIONS ALONG TWO STREAMLINES. SSPA MODEL 720. [4] (SEE NEXT PAGE)

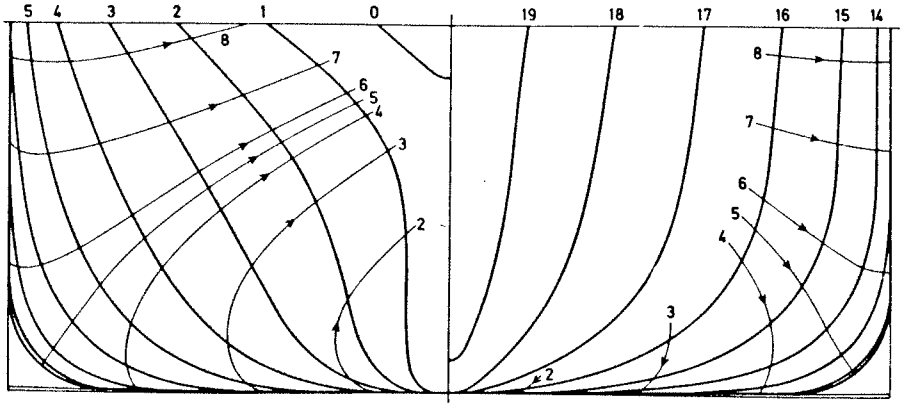

FIG 8C THE SSPA MODEL 720 WITH STREAMLINES

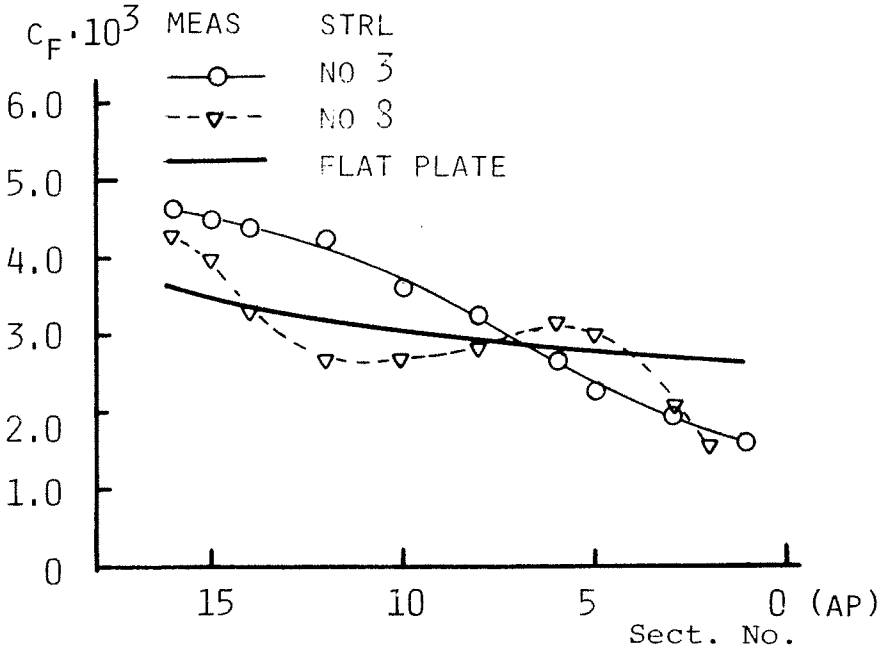

FIG 9 SKIN FRICTION DISTRIBUTION ALONG TWO STREAMLINES.
SSPA MODEL 720 [4]

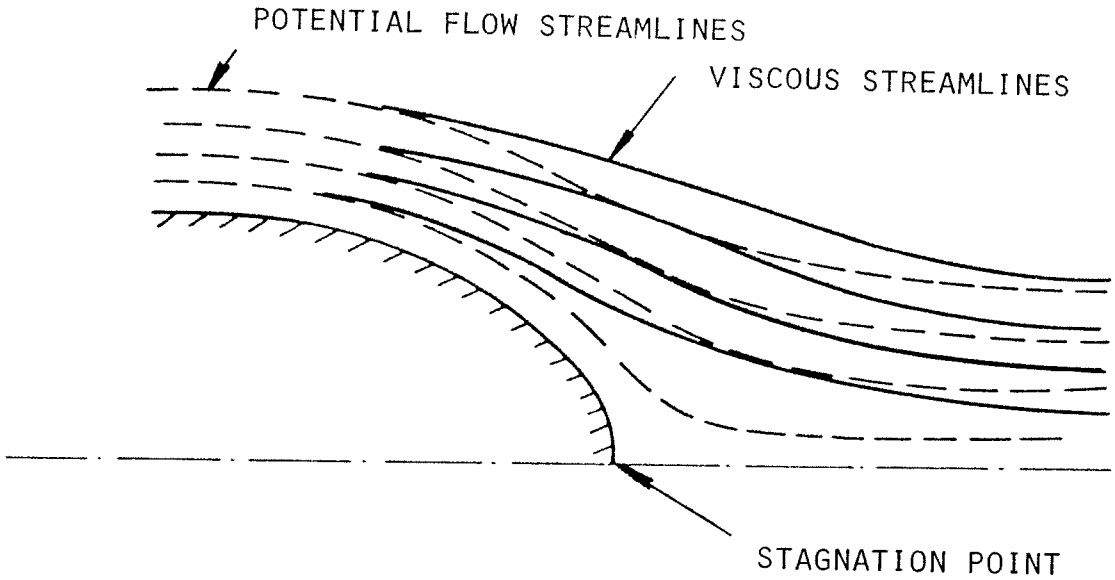

FIG 10 THE BOUNDARY LAYER DISPLACEMENT EFFECT ON THE STREAMLINES NEAR THE STERN

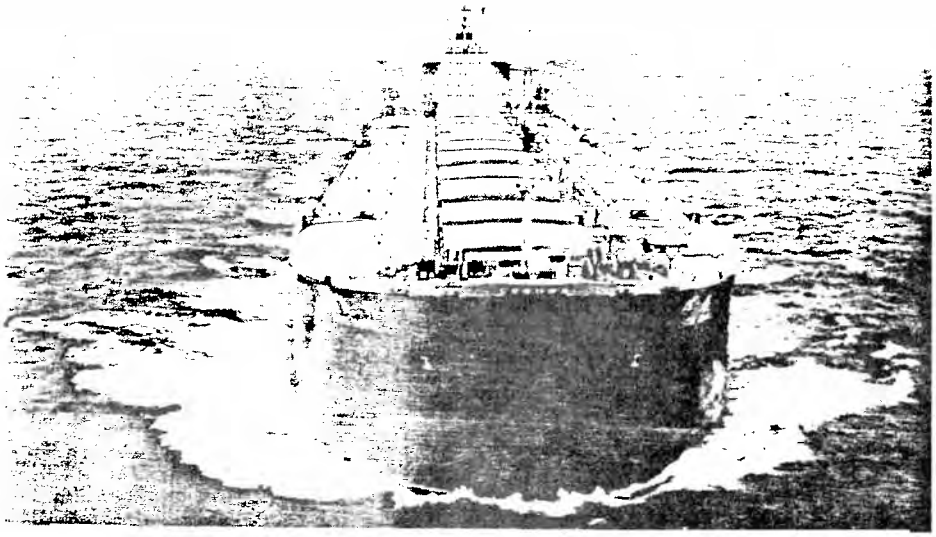

FIG 11 WAVE BREAKING

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SHIP DESIGN FOR FUEL ECONOMY

EMPIRICAL OPTIMIZATION OF HULL FORMS INCLUDING THE INFLUENCE OF TRIM AND DRAUGHT

by

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ABSTRACT

The design of ships for fuel economy does not entirely mean hull forms of minimum resistance. Problems in connection with the construction and operation of the ship may turn the attention also to other areas than resistance, e g ship dimensions, hull fullness, stability and propulsion by screw propeller.

At the initial design stage it must be made clear if the shipyard is to build an inexpensive hull (shorter and more blunt than standard) or if the shipowner is to operate a fuel-saving ship (longer and finer hull). These elements in the design process are contradictory, the final compromise being an important task.

Before starting hull form optimization the main particulars of the hull must be settled. An early selection of these may be corrected by a sensitivity analysis during which small systematical variations of hydrodynamical parameters can reveal beneficial or disadvantageous influences.

There are a number of criteria for limited wave and frictional resistance, good propeller efficiency and for the avoidance of propeller-induced hull vibrations. This guidance is valuable for the selection of hull form dimensions and fullness but will also lead to a first choice of the hull form. In this procedure it is necessary to have available the results from systematical standard hull form tests. The final optimization of the hull form begins with the knowledge (more or less established) of how to transform the standard hull forms to meet the unique requirements set by every new ship project.

When sailing at extreme draught and trim the behaviour of the ship is very dependent on how its stem and stern parts are designed. In this connection voluminous bulbous bows, large (slow-running) propellers and deep transom sterns should not be encouraged.

SHIP DESIGN FOR FUEL ECONOMY - WHAT IS IT ABOUT?

The fuel economy of a ship is very dependent on its hydrodynamic qualities. The size and speed of the ship give basic contributions to the fuel consumption, which are unavoidable, but there are other factors - among them ship hydrodynamics - which regulate the final fuel consumption.

These differences, small or large as they may be, are decisive as to whether or not the ship has an acceptable fuel economy.

What is meant by hydrodynamics in ship economy? Basically it has to do with the advance of the ship through the water, the water flow around the hull and the propeller action behind the hull, all of which absorb the available propulsive power.

From this we can venture to say that hydrodynamics, and with that the fuel economy, comes in as a significant design factor regarding the choice of:

- Ship main dimensions
- Hull form and fullness
- Propeller and propulsive arrangement

Likewise, ship hydrodynamics comes in as an important operational factor regarding

- Fuel consumption
- Propeller cavitation
- Vibrations

Generally it can be said that by a careful design of hull and propulsive arrangement one can improve the economy of

the ship. This is carried out more or less for all ship projects. What is important, however, is to judge the revenue of the extra efforts regarding fuel conservation that can be made in the initial design stage, during the ordinary project period and under the lifetime of the ship. A general diagram of income and expenditure is to be found in Fig 1.

Before anything more can be said about the importance of fuel costs one has to get a picture of the different costs of the ship. The limited and general examination carried out here, Fig 2, serves only as a visualization of the fuel costs compared with other costs.

From formerly being only a small fraction of the running costs, the fuel now stands for well over half of these costs, at least for larger ships. According to common conception the delivered power from the machinery is spent exclusively on overcoming the hull resistance, wind and waves. Less observed is that one third (for smaller and faster ships up to 50 %) of the power output disappears through propulsive losses due to a propeller efficiency of less than 100 % and interaction between propeller, hull and appendages.

It is understood that the cost distribution varies from one ship to another. The fuel costs in relation to other running costs or referred to transport volume may differ significantly between a ULCC and a car/passenger ferry, Fig 3, but it remains that the fuel costs in terms of pounds or dollars by necessity have to be reduced for both ships.

WHAT PROPULSIVE COMPONENTS ARE TO BE ATTACKED?

If the components making up the propulsive power are to be further discussed, a more detailed division is required. Which hydrodynamical elements can be influenced and which are forever fixed by nature? What can be influenced at the construction of a new ship and what can practically be achieved at the conversion of an existing ship?

The division of the propulsive components, as indicated in Fig 4, is still rather approximate. The contribution of roughness and fouling (of underwater shell plating) refers to a supposed "average" condition between two drydockings. This part of the viscous resistance can vary extremely, from a few per cent to half of the flat plate friction. The form and fullness of the underwater hull create an additional frictional resistance due to the viscosity of the water.

A second resistance component which varies extremely is the wind and sea resistance. The commonly used "sea margin" of 10 % of the total propulsive power is introduced here. Resistance from waves (created by the ship), wind and sea has by definition not to do with the viscosity of the water and is therefore called "residuary resistance".

A closer separation of the calm water resistance is presented in Fig 5. The main components are defined as viscous resistance, wave making resistance and spray resistance. The influence of flat plate friction and hull form on viscous resistance is substantially higher for a tanker than for a ro-ro vessel. The "hull form influence" is rather an influence coming from the hull fullness. This component is not so easily altered by the hull shape if the block coefficient is to remain unchanged. More affected by the hull form is the wave breaking and wave pattern resistance (from the forebody most) and, further, viscous pressure and separation resistance (afterbody most). These components are more predominant for the ro-ro vessel.

Components making up the total propulsive power are listed in Fig 6. The components can be approximately connected to hull and propeller respectively.

FIRST CHOICE OF HULL MAIN DIMENSIONS

The optimization of the hull form starts late in the initial design process. After some introductory consideration of the hydrodynamics of the project, mentioned in the foregoing

part, the next step is to make a first choice of hull main dimensions and fullness.

To start with the influence from hull length and fullness upon required propulsive power, it is evident from Figs 7 and 8 that the tanker is the most affected of the two exemplified ships. Required power for the extremely full and short tanker is nearly three times as high as for the ship with the finest and longest hull!

A number of criteria are occasionally used for the choice of hull length and fullness, some of which are associated with the development of wave resistance. The main parameter is Froude's number, $V/\sqrt{gL}$ (V = speed, L = length of the ship), as this figure is significant for the wave pattern created by the ship. The "economical speed" for a ship is said to be reached when the wave resistance suddenly increases due to some wave interference. In diagrams, Figs 9 and 10, ro-ro vessels and tankers tested at SSPA have been plotted together with known resistance-based criteria.

We can conclude from the diagrams that, for ro-ro vessels, the ship designers have not been consistent. This can be explained by the previously mentioned fact that the required propulsive power for this type of ship is not so sensitive to fullness. For tankers the judgement is just the opposite, the criteria are widely spread while the ships are assembled.

SENSITIVITY ANALYSIS OF HULL MAIN DIMENSIONS

Having preliminarily settled hull main dimensions and fullness it is now worth-while to check these by a sensitivity analysis, which also can comprise, for instance, some propeller parameters. The question is how much the required propulsive power is influenced by small changes of the various geometric parameters. If a minor correction of a dimension gives a potential reduction of required SHP, this ought to be considered.

The diagram Fig 11 illustrates that the required SHP will be more influenced for the tanker than for the ro-ro vessel for nearly all changes. For both vessels, however, the attainable savings in SHP are considerable, much more than is commonly known. It should also be observed that a reduction of propeller RPM may be as efficient as an increased hull length. A propeller RPM lowered by 25 % is quite realistic while the designer can have difficulties in obtaining approval for an increase in the hull length by a few per cent.

The resistance criteria for the choice of hull fullness are occasionally violated without other consequences than a gradually (for tankers rapidly) increased required propulsive power. Now it will be demonstrated that too high a hull fullness cannot be allowed with regard to the afterbody flow. The problem is serious for tankers where the afterbody fullness has to be limited, if a satisfactory propulsive behaviour is to be maintained.

One of the most immediate signs of bad propulsive behaviour of a ship is vibrations, mostly occurring in the afterbody and coming from the propeller. One talks about propeller-induced pressure fluctuations working on the shell plates near the propeller. If these impulses can be predicted by calculations or model tests this would be one of the best tools for checking ship vibration.

A simplified criterion for maximum allowable excitation has been established at SSPA based on the present ISO-limit of 6.4 mm/sec for allowed vibrational velocity at a reference point on deck above the propeller. This velocity can be represented by a certain allowed amplitude of pressure fluctuations from the propeller at a point in the stern frame just above the propeller.

The expected amplitudes are calculated from tests in a cavitation tunnel. Important parameters are among others cavitation number, wake distribution, propeller diameter and propeller RPM.

It is possible to calculate level curves for expected contra allowed pressure fluctuations for standard conditions regarding hull form, wake distribution, propeller size and RPM etc, Fig 12.

Known ships of 50 000 - 150 000 m^3 displacement with reported behaviour regarding propeller-induced vibrations are also indicated in Fig 12. Some of the ships with severe problems have been tested at SSPA after delivery and some period of service, the original model tests carried out elsewhere. Other unsuccessful ships have originally been given dimensions and hull form and fullness which have not been altered, notwithstanding recommendations made after the first model tests.

FIRST CHOICE OF HULL FORM

There is need for a reference when discussing hull forms. What is normal? What can be called conventional? It is evident that some sort of standard hull forms are helpful at the start of a preliminary ship design. There are in fact many reasons for having systematical series of standard hull forms as reference for hull form design, speed and power, stability, etc. Ship hull form series at SSPA are presented in Fig 13. A new hull form series consists of modern hull shapes for a number of ship classes, among which ro-ro vessels are centrally located, Fig 14.

The parent form of the SSPA new hull form series is shown in Fig 15. It serves as a reference hull form for judging the hydrodynamic characteristics of various ship projects but can also be used as a first choice of hull form for a project.

OBJECTIONS AGAINST THE STANDARD HULL FORM

Transverse stability, also the static stability, is most important for many types of ships due to cargo handling, stowage, seaworthiness, etc. To what extent will the need for greater stability influence other characteristics such

as main dimensions, fullness, hull form and margin against propeller-induced vibrations? Also here it is useful to refer to standard, which may be the location of the metacentre above keel, KM, for the standard hull form, Fig 16.

If the standard hull form cannot fulfil the stability requirements, and the breadth of the ship is for some reason restricted, a modification of the standard form must be made, Fig 17. A higher resistance can be expected, however.

Heavy cargo handling gear at the stern of the ship will require a more aft location of the centre of buoyancy, LCB, with fuller lines in the afterbody, Fig 18. Also in this case the alteration of the standard hull form can cause problems, perhaps more serious than in the previous case, as propeller-induced vibrations can occur.

The aim of the series of standard hull forms is that those who are designing ships can conveniently use these forms, and feel confident. Many vessels, specially tankers and bulkers, have been built with the standard hull form as a model. On the other hand one has to be attentive to what can happen at, for instance, an alteration of the LCB, Fig 19.

CONSIDERATIONS OF TRIM AND DRAUGHT

The propulsive properties of ships at various trims and draughts depend to a certain degree on the shape of the bow and the stern of the vessel. The existence of a bulbous bow has a great influence on the required SHP when the draught fore is varied.

Bulbous bows are fairly common, at least on the forebodies of ro-ro and container vessels. They are expensive, however, and shipyards try to escape building them. On many bulkers and tankers the bulb has already been abolished. Still the bulbous bow on fast vessels, operating at the designed waterline, can be considered effective, Fig 20.

When the draught aft is substantially reduced, the air suction to the propeller is most decisive for the performance of the ship. From Fig 21 it can be seen how the ship speed (at constant SHP) drops when the air suction to the propeller is fully developed.

Model tests in ballast for the exemplified tanker in Fig 22 discloses a considerable speed reduction at small mean draught and large trim by stern, evidently due to the bulbous bow breaking the water surface.

MODEL TESTS - ANALYSIS - CONCLUSIONS

As mentioned before, the model tests come in rather late in the ship initial design process. A number of considerations regarding hull main dimensions, form and fullness have to be made before the lines of the ship are drawn with the purpose of ordering model tests.

The judgement of a hull form starts with an analysis process immediately following the model tests, Fig 23. The required propulsive power for the topical hull over that of a standard hull form is estimated, Fig 24. The topical project and the standard ship have the same dimensions, fullness and propeller data. The relation P(ACTUAL) / P(STANDARD) expresses the quality of the full form with regard to fuel consumption.

Moreover, by use of standard hull form series, it is possible to find the minimum required propulsive power P(MIN) for a standard ship having the same displacement and speed as the topical one. The relation P(STAND)/P(MIN) illustrates how much of the required propulsive power is due to the choice of main dimensions and fullness.

The above-mentioned power relationships for exemplified ro-ro vessels and ferries are to be found as Figs 25 and 26.

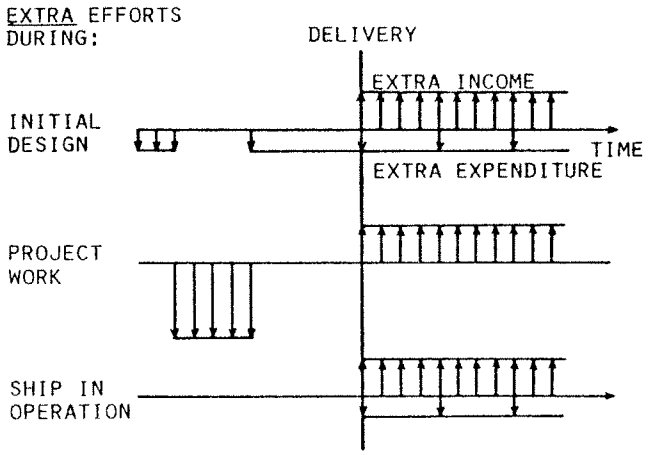


Fig 1. Income and expenditure for various extra efforts to conserve fuel.

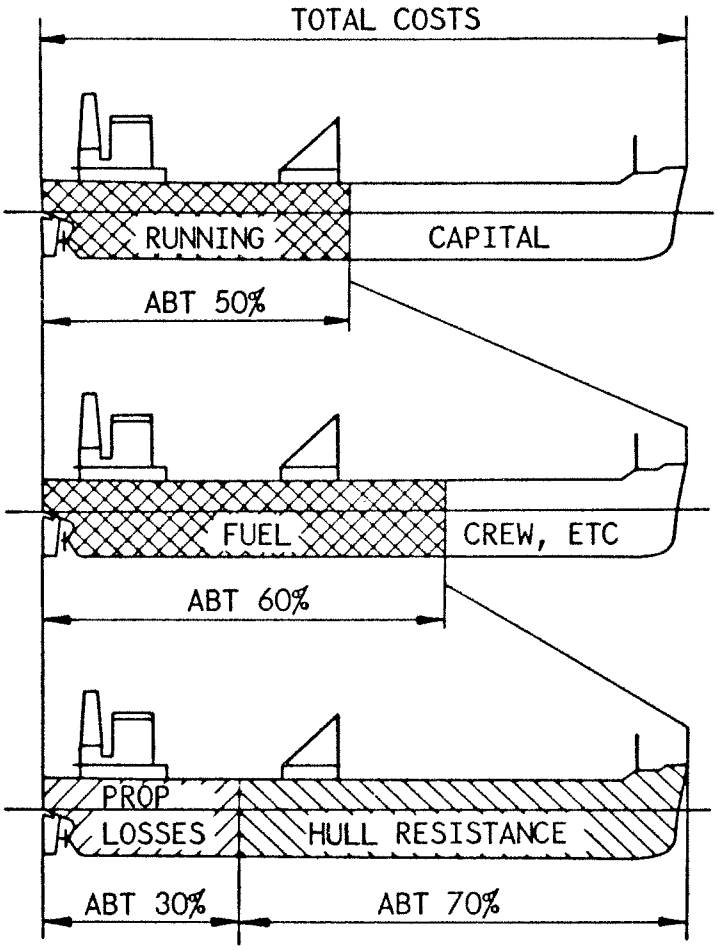


Fig 2. Components of total costs for abt 80 000 tdw tanker.

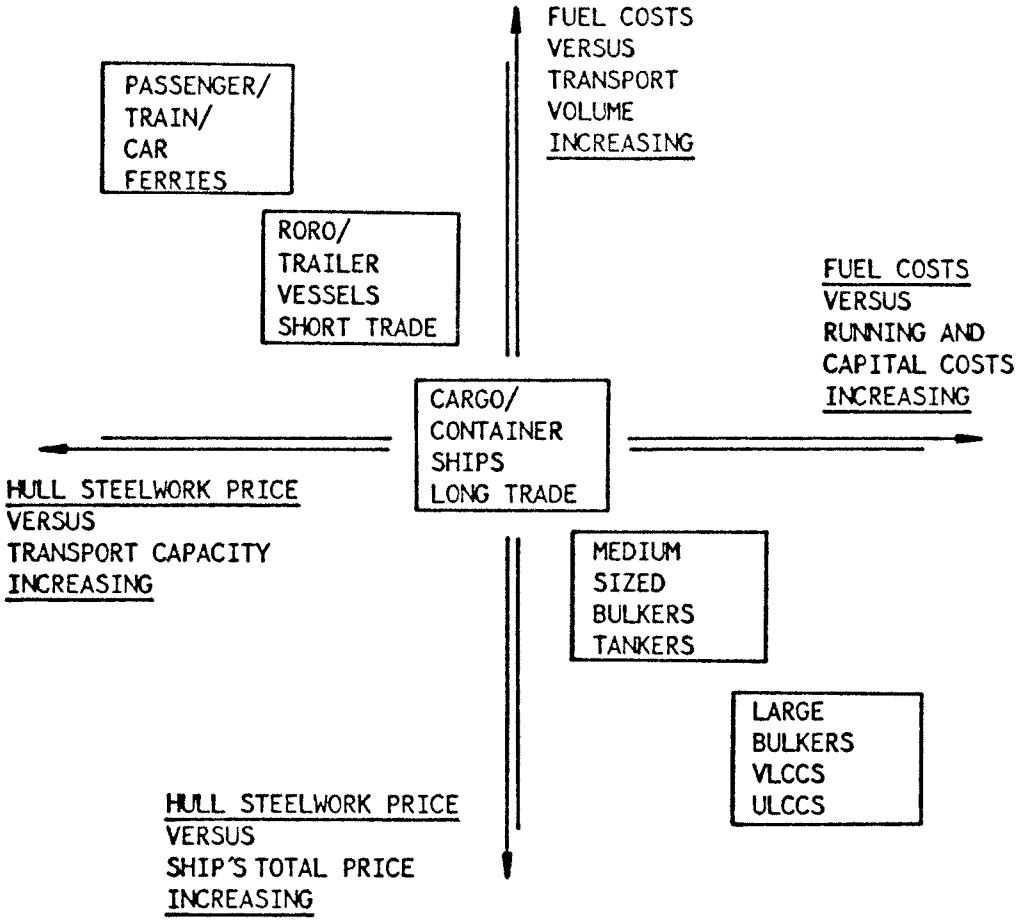
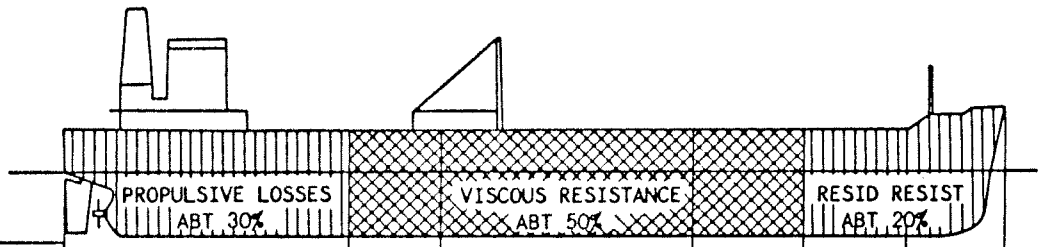


Fig 3. The tendency of relative fuel costs and hull steelwork price for a number of ship types.



COMPONENT:	PROPELLER EFFICIENCY	ROUGHN FOULING (AVERAGE)	FLAT SMOOTH PLATE FRICTION	FORM RESIST, ETC	WAVE RESIST	WIND AND SEA (AVERAGE)
ITEM:	PROPELLER, STERN ARRANGEM	HULL SURFACE CONDITION	UNDERWATER SHELL PLATING	HULL DIMENS, FORM, FULLNESS		WEATHER CONDITIONS
AT NEW-BUILDING:	TO BE OPTIMIZED	TO BE CONSIDERED	SMALL INFLUENCE	TO BE OPTIMIZED		INTRODUCE ROUTEING!
AT CONVERSION:	CAN BE IMPROVED	CAN BE IMPROVED	UN-CHANGEABLE	UN-CHANGEABLE		BETTER ROUTEING ?!

Fig 4. Components of propulsion for abt 80 000 tdw tanker and relation to various items.
How energy can be conserved at newbuilding and conversion.

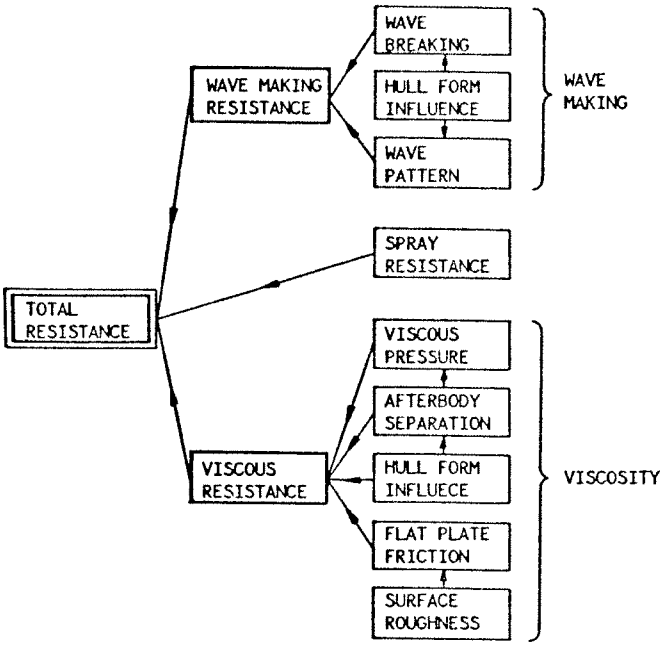


Fig 5. Components of calm water resistance.

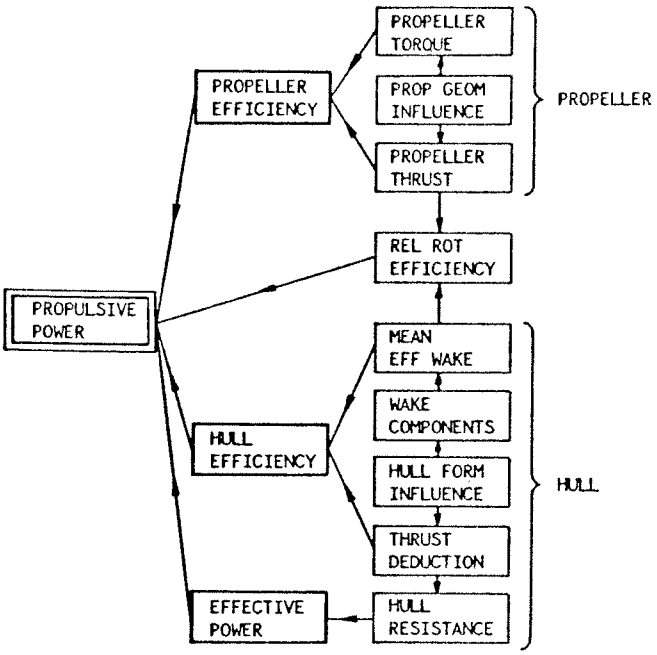


Fig 6. Components of propulsive power.

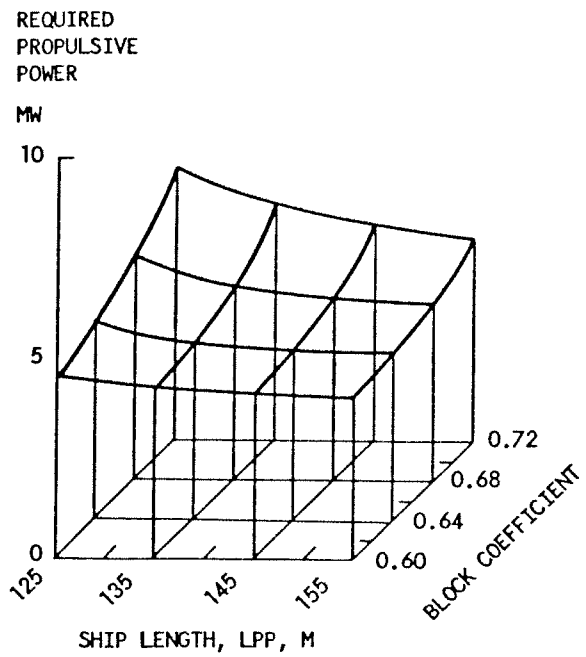


Fig 7. 12 000 m^3 displacement roro. Required propulsive power versus hull length and fullness. V = 18 knots.

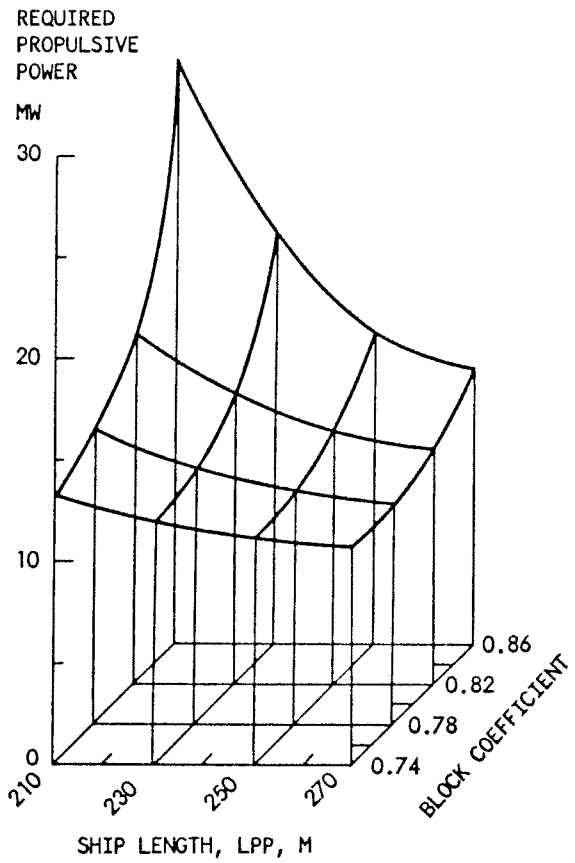


Fig 8. 100 000 m^3 displacement tanker. Required propulsive power versus hull length and fullness. V = 16 knots.

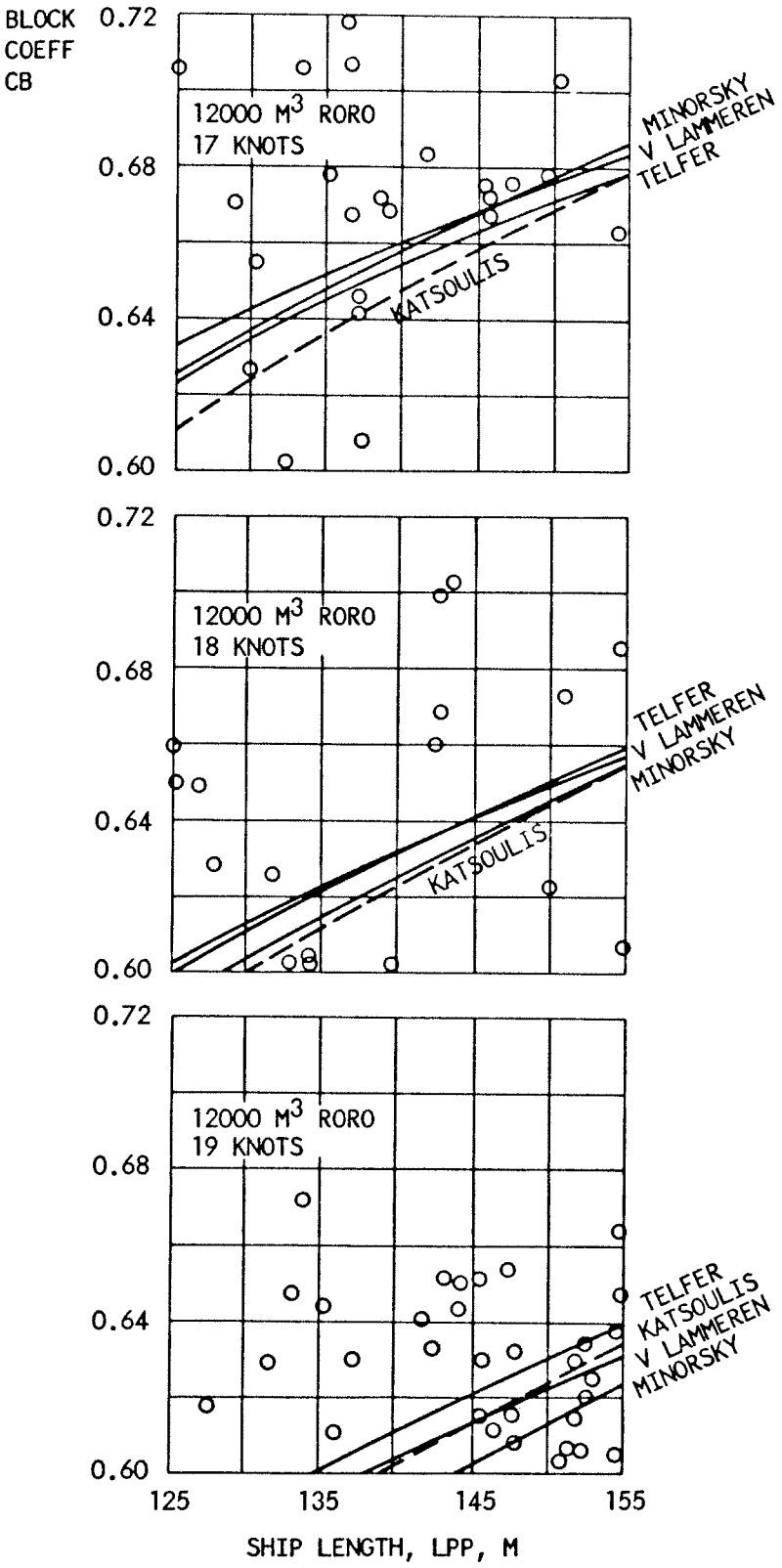


Fig 9. Criteria based on resistance development. Indicated roros of 6 000 - 18 000 m^3 displacement tested at SSPA. Displacement and dimensions reduced to 12 000 m^3.

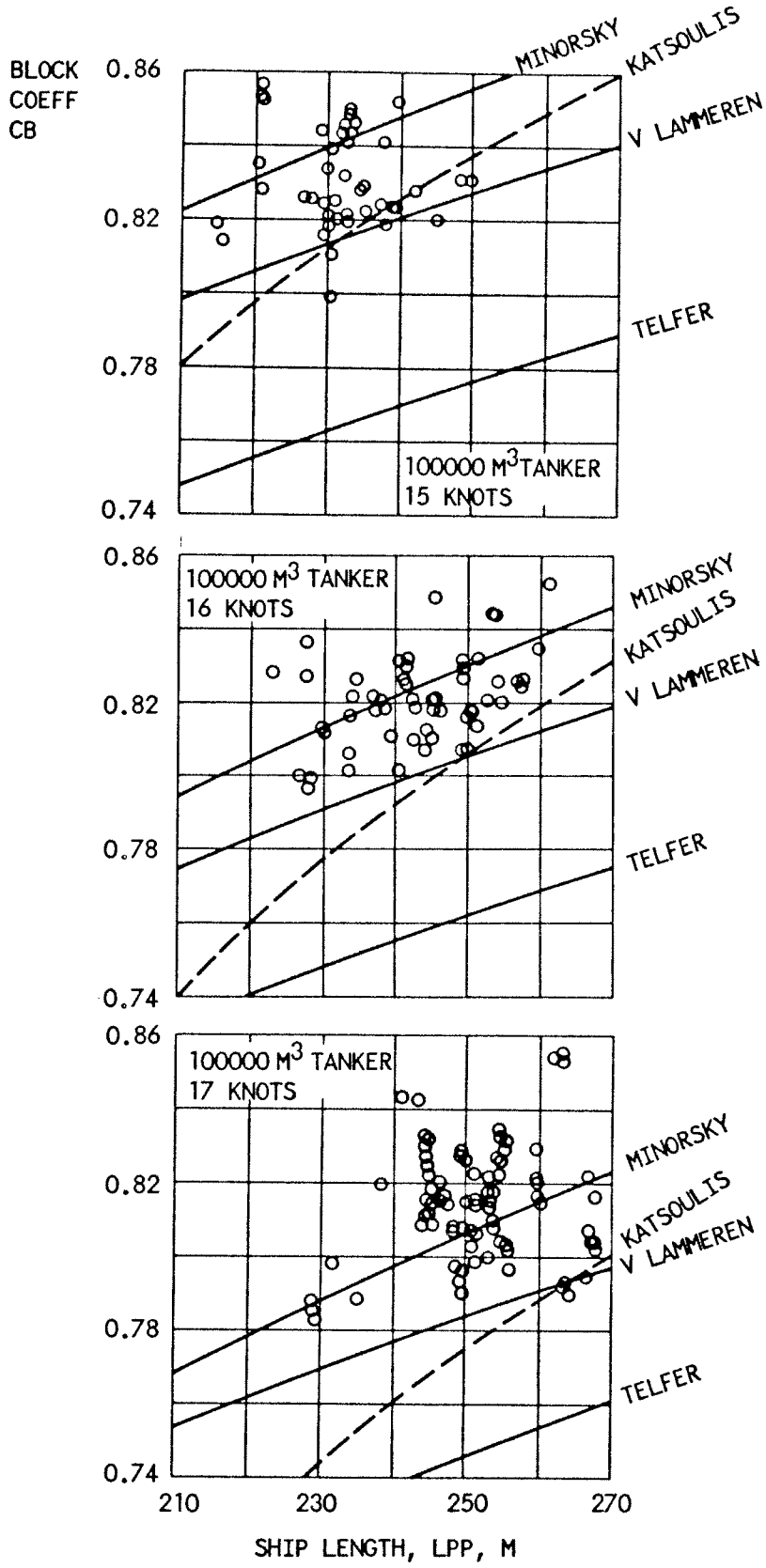


Fig 10. Criteria based on resistance development. Indicated tankers of 50 000 - 150 000 m^3 displacement tested at SSPA. Displacement and dimensions reduced to 100 000 m^3.

CHANGES	HULL MAIN DIMENSIONS "VARIABLE BLOCK COEFFICIENT"				
	LENGTH	BRE-ADTH	DRAFT	DISPL	BLOCK COEFF
PARENT	L	B	T	∇	CB
LENGTH CHANGE	XL	B	T	∇	$X^{-1}CB$
BREADTH CHANGE	L	XB	T	∇	$X^{-1}CB$
DRAFT CHANGE	L	B	XT	∇	$X^{-1}CB$
DISPL CHANGE	L	B	T	X∇	XCB

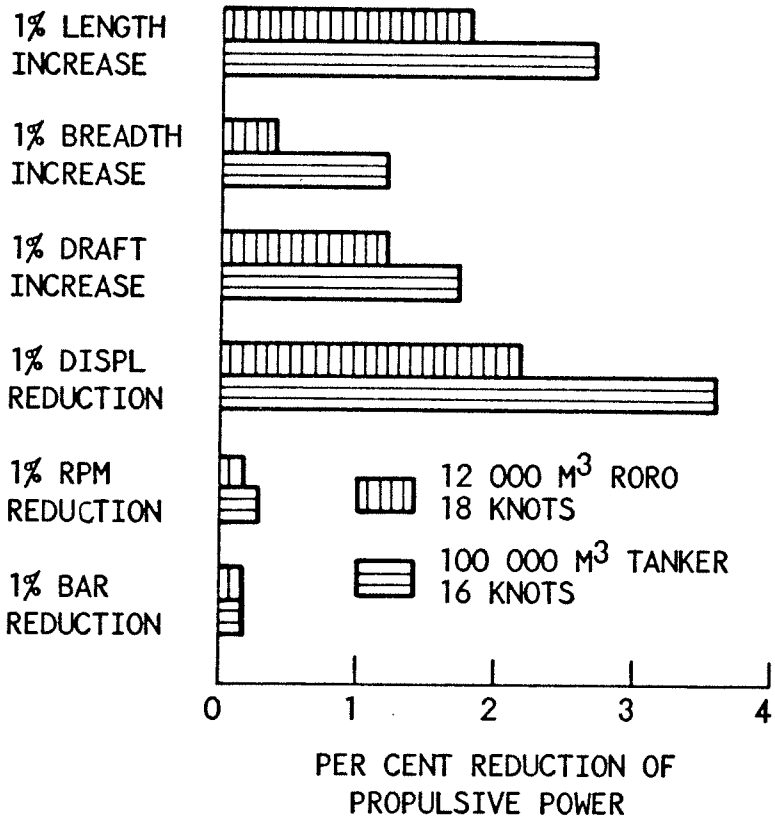


Fig 11. Reduction of propulsive power due to change of hull main dimensions, displacement, propeller rate of revs and propeller blade area. Variable block coefficient.

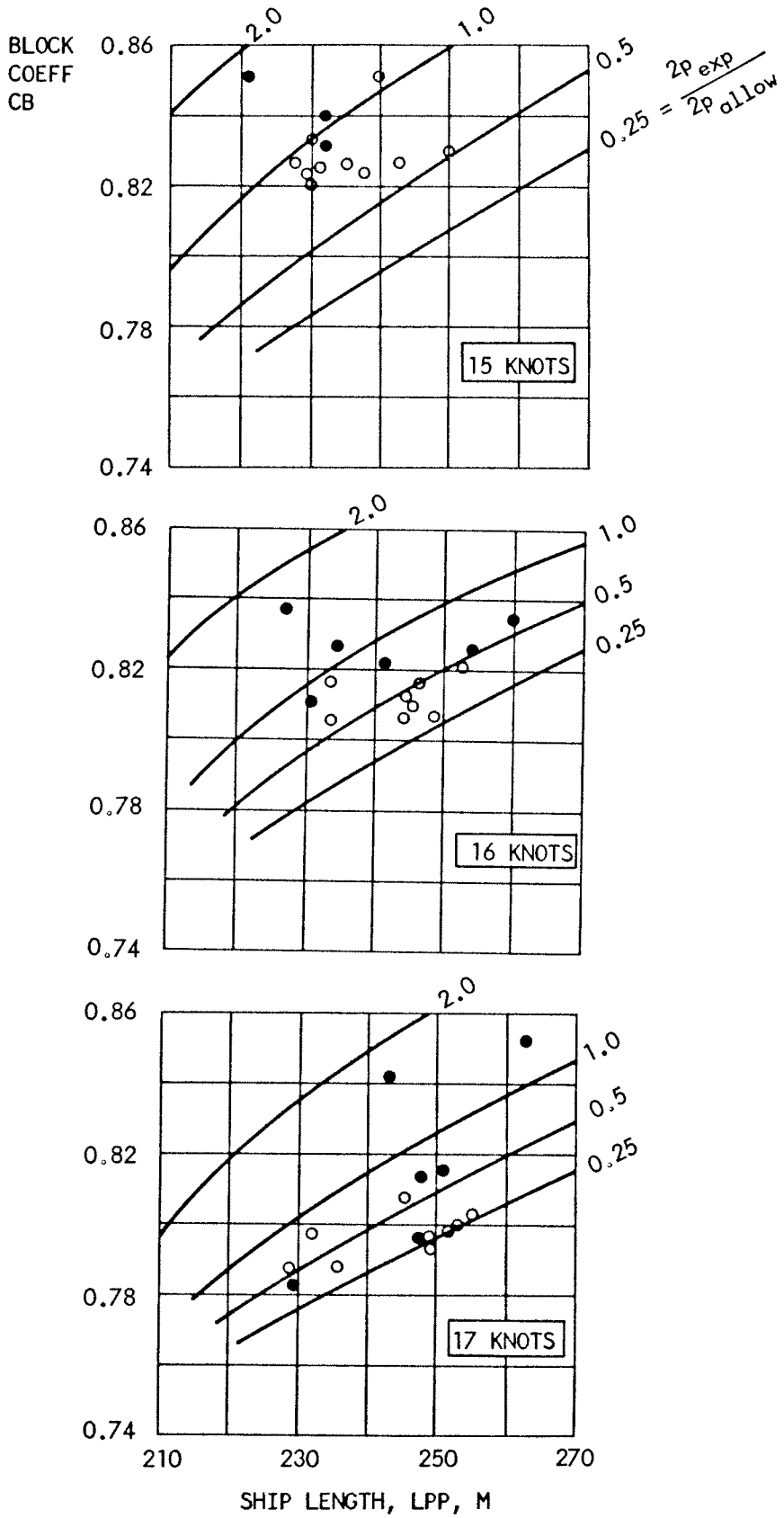


Fig 12. Criterion based on propeller-induced vibrations and reported ship behaviour. Displacement and dimensions reduced to 100 000 m^3. Open circles indicate no problems, filled circles severe problems. Curves refer to standard hull form and propeller.

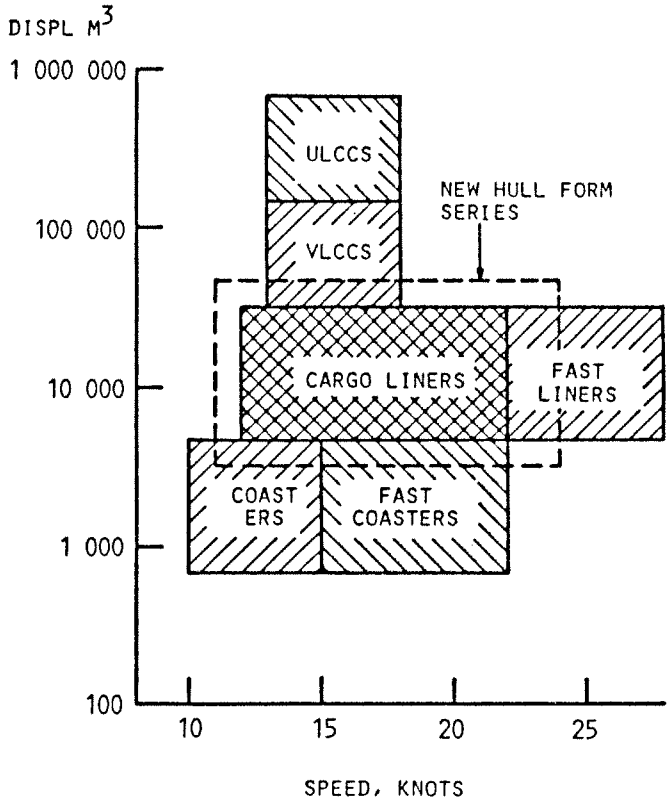


FIG 13. SHIP HULL FORM SERIES AT SSPA

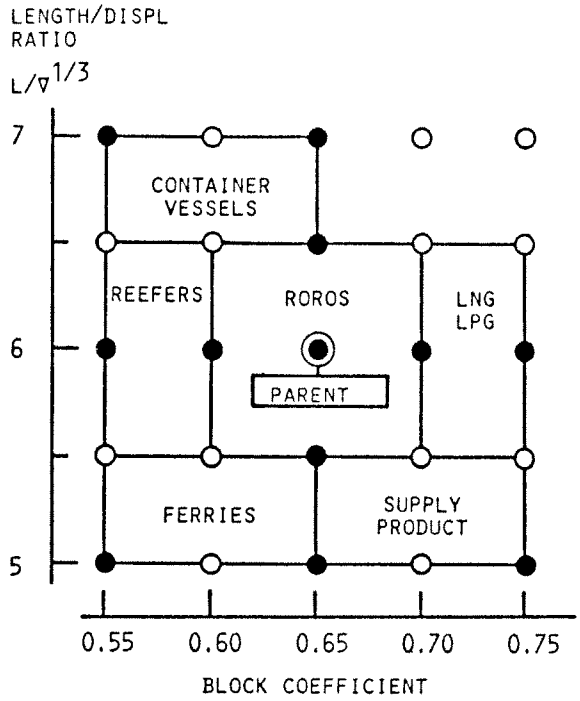


FIG 14. SSPA NEW HULL FORM SERIES. SHIP TYPES COVERED

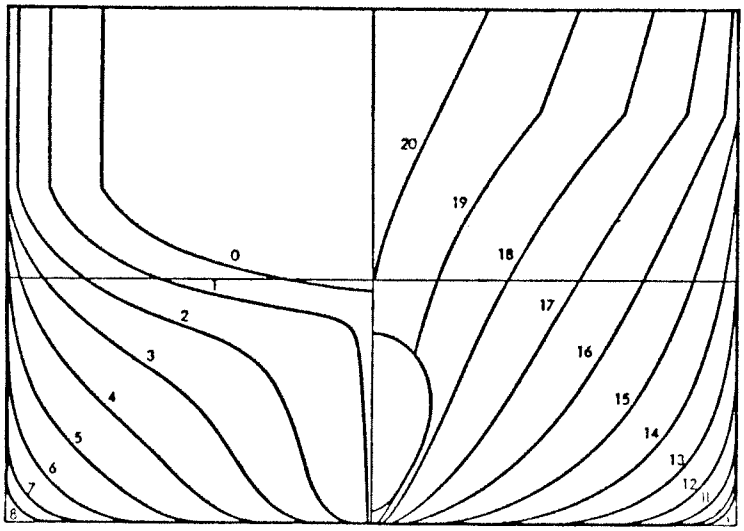


FIG 15. SSPA NEW HULL FORM SERIES. PARENT HULL FORM. BLOCK COEFFICIENT = 0.65

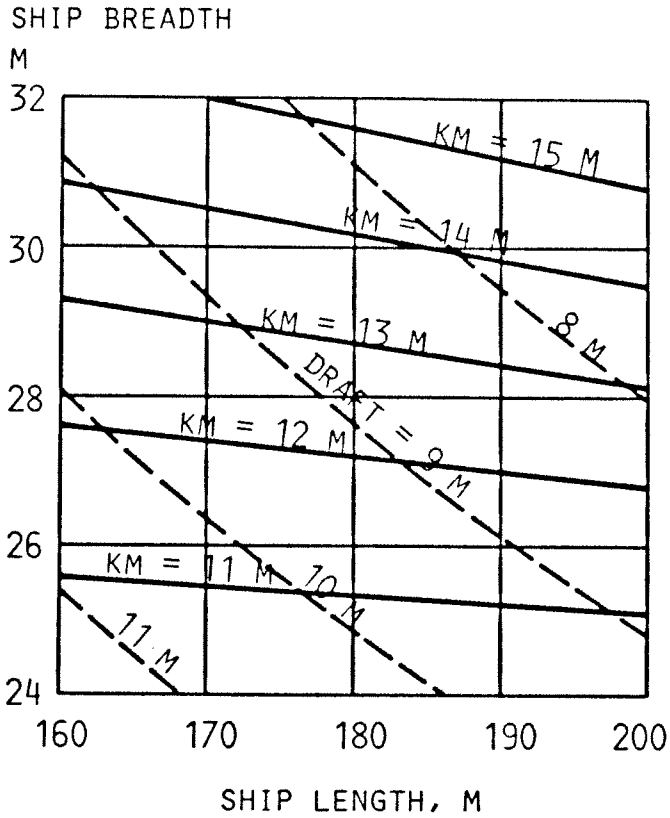


FIG 16. METACENTRE ABOVE KEEL (KM) FOR 30 000 M^3 DISPL RORO VESSELS WITH STANDARD HULL FORM. BLOCK COEFFICIENT = 0.65.

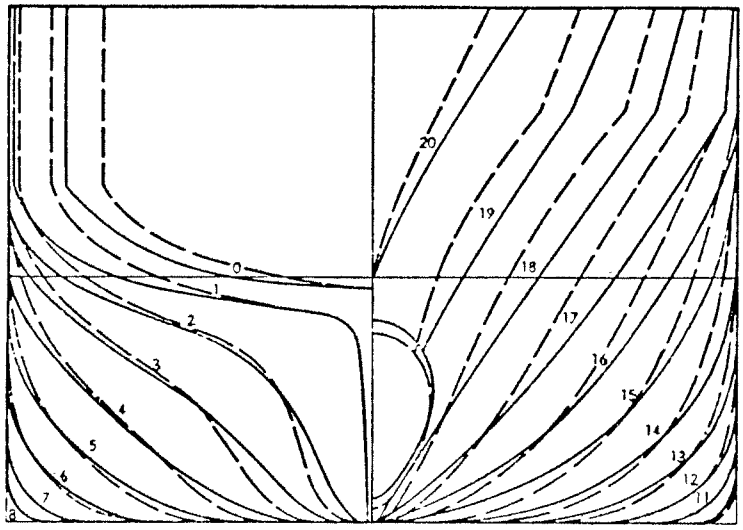


Fig 17. Modification of standard hull form to higher KM (solid lines). Block coefficient = 0.65.

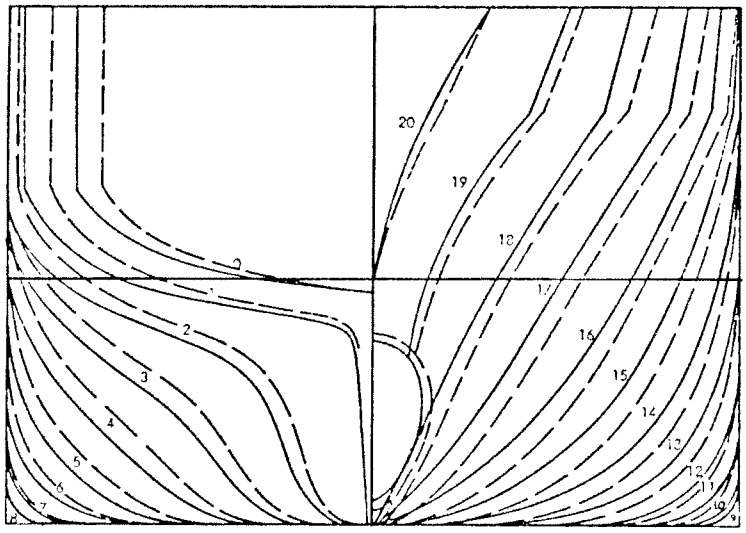


Fig 18. Modification of standard hull form to more aft LCB (solid lines). Block coefficient = 0.65.

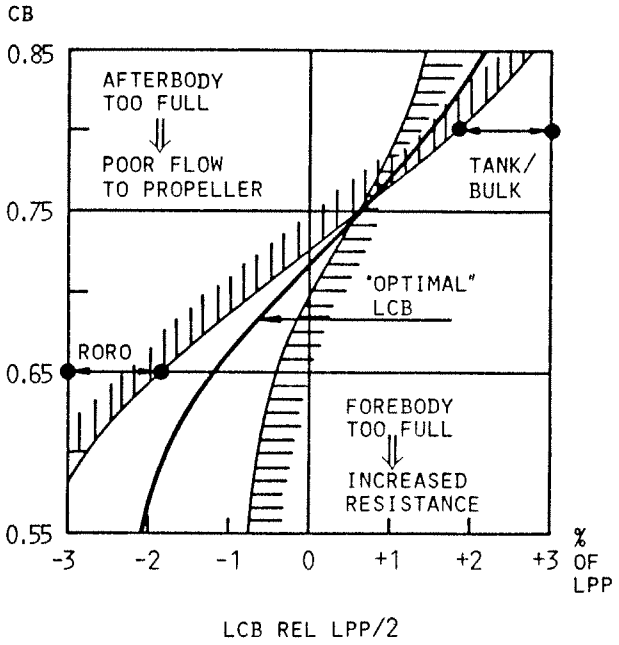


Fig 19. "Optimum" location of centre of buoyancy (LCB) for various block coefficient (CB). Standard hull form.

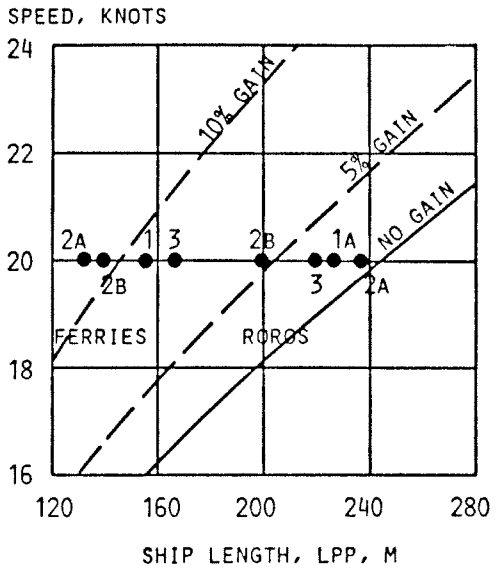


Fig 20. Per cent gain (power reduction) obtained by bulbous bow on a standard hull form. Speed and length of exemplified roros and ferries are indicated.

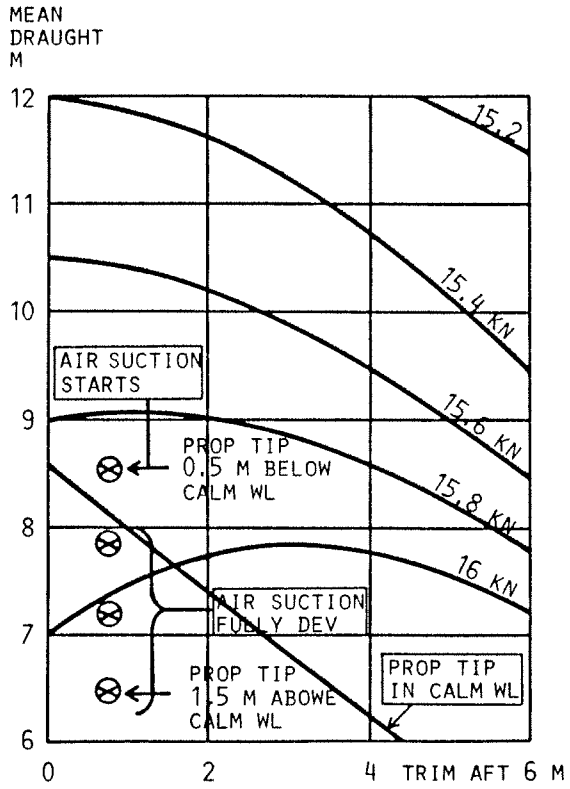


Fig 21. Model tests on speed and air suction to the propeller. 150 000 tdw tanker in ballast. Full draught 17 m. 18 000 SHP.

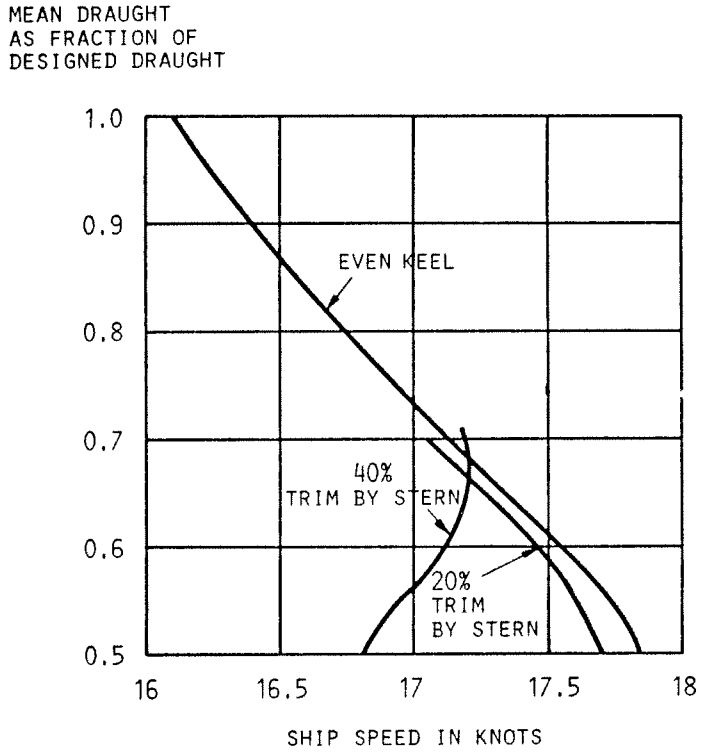


Fig 22. Model tests on speed for a 120 000 tdw tanker in ballast. Forebody with bulbous bow. 22 800 SHP.

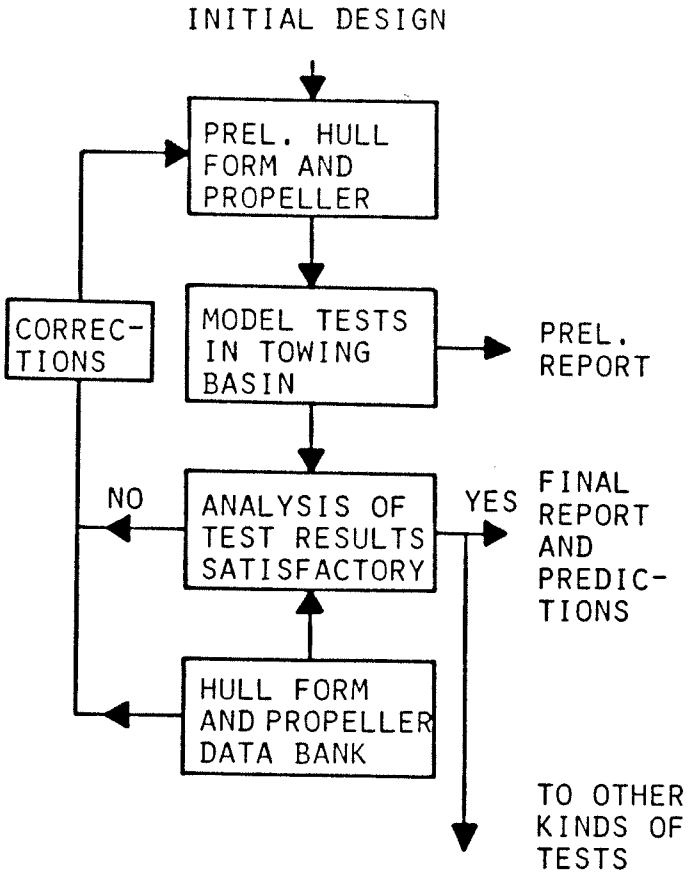


FIG 23. SCHEDULE OF MODEL TESTS

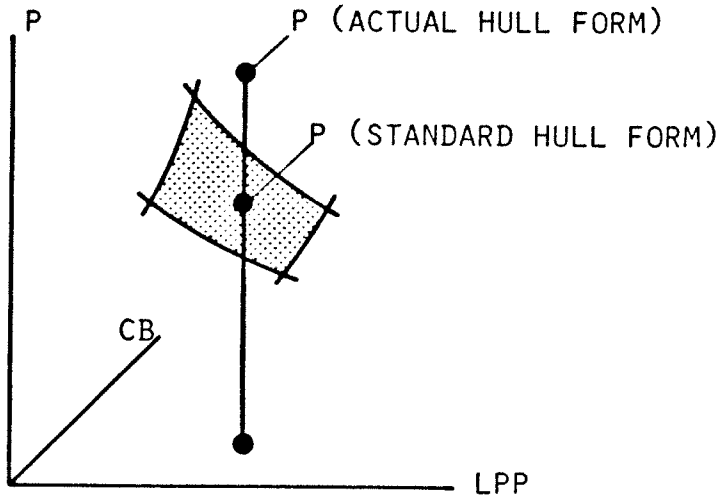


FIG 24. PLOTTING OF PROPULSIVE POWER P FOR STANDARD HULL FORM (ON CARPET) AND ACTUAL FORM

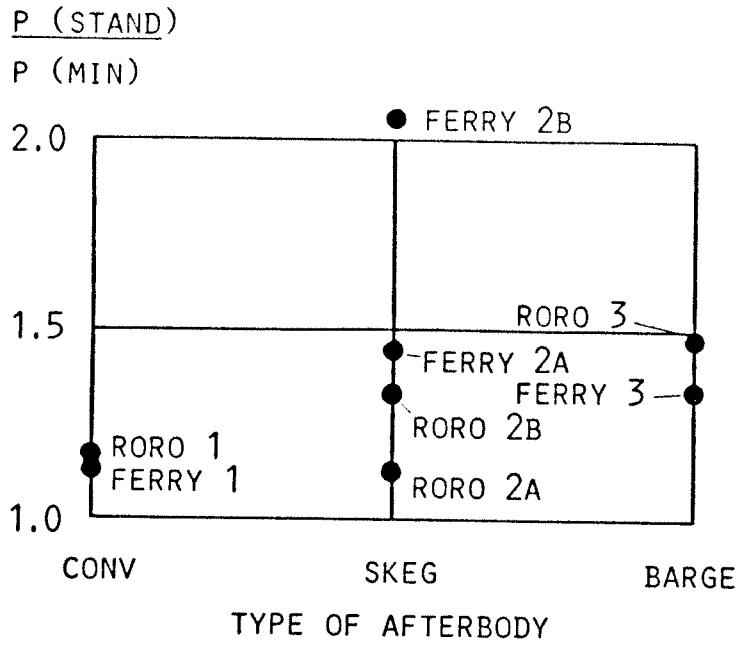


FIG 25. STANDARD PROPULSIVE POWER FOR EXEMPLIFIED ROROS AND FERRIES IN RELATION TO MINIMUM PROPULSIVE POWER.

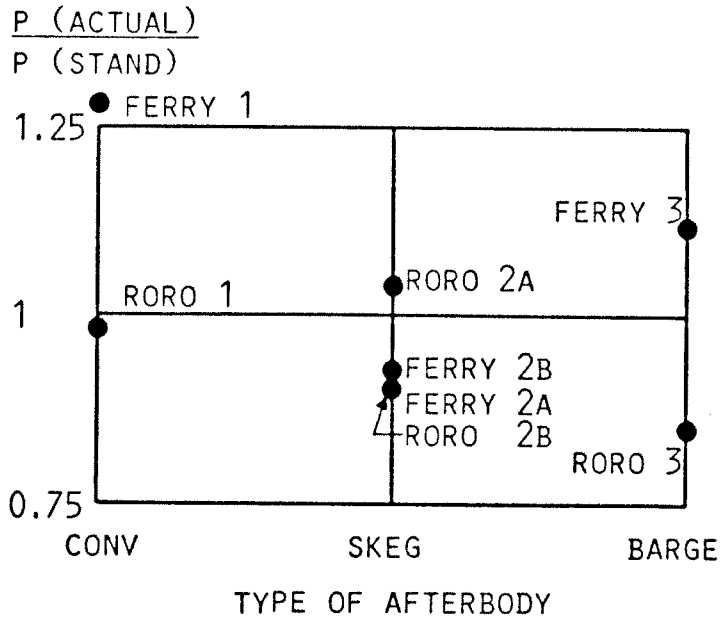


FIG 26. ACTUAL PROPULSIVE POWER FOR EXEMPLIFIED ROROS AND FERRIES IN RELATION TO STANDARD PROPULSIVE POWER.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

RESISTANCE REDUCTION BY TURBULENCE CONTROL

by

MARTEN LANDAHL

ROYAL INSTITUTE OF TECHNOLOGY, STOCKHOLM

and

MASSACHUSETTS INSTITUTE OF TECHNOLOGY, CAMBRIDGE

RESISTANCE REDUCTION BY TURBULENCE CONTROL

Marten T.Landahl
Royal Institute of Technology, Stockholm, Sweden
and
Massachusetts Institute of Technology, Cambridge, Mass.

SUMMARY

A number of different methods are reviewed whereby it has been found possible to reduce turbulent skin friction drag. The most effective one is through addition of certain polymers, the best ones (e.g., polyethylene oxide) being able to reduce the drag by up to about 80% with only a few parts per million of additives by weight. Laboratory studies indicate that the polymers affect the small-scale turbulent structures in the near-wall region. For reasons of costs, polymer additives are only interesting for special applications, however (such as to increase the capacity of a pipe line, for example). Newly explored methods of drag reduction have employed surface structure modifications to influence the small-scale eddies, and honeycomb structures and the like in the boundary layer with the aim of breaking up the large-scale turbulent structures. In particular, methods of the latter kind offer hope for practical applications to ships and aircraft. With large-eddy breakup devices, sizeable net drag reductions (about 20%) have been realized in laboratory experiments. The lecture includes a discussion of the mechanisms that might be at work with these devices.

1. Introduction

The large increases in the cost of fuel in recent years have intensified the search for methods to reduce the flow resistance of moving vehicles on land, in the air and on the sea. Traditional methods of drag reduction, such as streamlining to reduce flow separation and hence lower the form drag, have their limits, and in order to proceed further one needs to find means to reduce the skin friction.

Since in a turbulent boundary layer the momentum transport is controlled primarily by turbulent mixing processes, lowering of the skin friction requires the modifiction of the turbulence structure in such a manner that the momentum transport by the turbulent fluctuations are reduced. In order to achieve such a modification in an effective manner a thorough knowlegde of the mechanisms controlling the momentum carrying turbulent fluctuations is needed. However, the understanding of the turbulence mechanism operating in turbulent shear flows is unfortunately still rather incomplete, despite extensive experimental and theoretical research efforts over many years. The difficulties stem primarily from the problem of establishing cause-and-effect relationships in the complicated turbulent processes.

Nevertheless, by experimentation a few methods have been discovered that have been found effective in reducing skin friction in a turbulent boundary layer. The most efficient ones for liquids all involve additives such as certain long-chain polymers (max. drag reduction about 80%), fibers (50%) and gas bubbles (30%). Methods that do not involve additives are the so called large eddy breakup devices (20% drag reduction reported) and surface structure modifications (10%). Also very effective is laminarization of the boundary layer through suction and favorable pressure distribution, but these are impractical for the high Reynolds numbers of interest for ships and will therefore not be discussed. Some of the proposed methods, such as the use of compliant walls, have so far not shown any clearly demonstrable drag reductions.

In this lecture we shall review drag reduction methods of potential applications to ships, the effectiveness of the methods, and what is understood about how they work. We shall leave out what is obviously the most efficient and important drag reduction method, namely to keep the surface clean and smooth so as to minimize roughness drag. Neither shall we discuss the gains in turbulent skin friction that might be achievable through tailoring of the pressure distribution. As a start we shall first give a short review of what is understood about boundary layer turbulence and what has been learned from drag reduction experiments.

2. The structure of boundary layer turbulence

In recent years very extensive efforts have been devoted to investigations of the structure of shear flow turbulence. This area of research is of great interest both from a fundamental fluid dynamical point of view and for engineering applications. Most flows of engineering interest are turbulent, and if one could learn how the properties of such flows could be altered and controlled in favorable ways, tremendous economical benefits could be realized thereby. Turbulent drag reduction is one important aspect of turbulence control with promising potential applications to vehicular transport and chemical and process engineering.

We shall here give a brief review of the present understanding of boundary layer turbulence structure with particular emphasis of those aspects that might have bearing on drag reduction methods. It is common to subdivide the turbulent boundary layer in essentially two different regions, a near-wall region consisting of the viscous wall region and buffer region in which viscosity is believed to be of importance, and an outer region, consisting of a logarithmic layer and a wake or velocity defect region, the latter also incorporating the outer region of intermittency that arises because of the fluctuating boundary between turbulent and nonturbulent flow. In the outer region viscosity is believed to play little role in the momentum transfer process.

The inner region is best described in terms of wall related variables based on the wall friction velocity

$$u_* = \sqrt{\tau_w/\rho} \qquad (1)$$

where τ_w is the wall shear stress and ρ the density. From this and the kinematic viscosity, ν, one can form the viscous length scale

$$\ell^+ = \nu/u_* \qquad (2)$$

The viscous wall plus buffer region extends to a distance of about $y^+=y/\ell^+=70$ from the wall. The turbulence intensity and the turbulent kinetic energy production have their maxima in the wall region at around $y^+ =15$. At high Reynolds numbers, the boundary layer thickness may be of the order of tens of thousands in wall units and more. Hence, much of the interesting dynamical activity takes part in a thin layer near the wall with a thickness of one per cent or less of the total thickness of the boundary layer. It is clear from such simple estimates that the exerimental study of boundary layer turbulence structure is extremely difficult. The theoretical difficulties are of course also severe, since one has to model a three-dimensional randomly fluctuating flow field controlled by nonlinear processes.

Practically all of the detailed experimental information available today on the structure of turbulence in the near wall region have been obtained in low speed thick shear flows of fairly low Reynolds number. It is generally assumed that the near wall structure scales with inner variables so that such results are of universal validity for all Reynolds numbers, as well. However, from a simple analysis it can be shown that outer scale Reynolds number independence can only be approximately true. Therefore, drag reduction results obtained in the laboratory at modest Reynolds numbers may not necessarily apply to the full scale situation, something that must be born in mind when considering possible engineering applications.

The innermost region of the wall layer, the so called viscous wall layer (y^+ less than about 5) the most characteristic dynamical feature is the longitudinal streaks of alternating low- and high-speed fluid. Measurements of their average spacing gives a spanwise distance between streaks of about 100 in wall units. The low-speed streaks are seen to lift up intermittently, to become unstable, and subsequently to break up in a chaotic fashion. This sequence of events is known as bursting. Most of the Reynolds stress production takes place during bursting. Despite the very considerable experimental efforts spent on investigating the wall layer turbulence structure there is still considerable disagreement concerning the detailed structure and their importance for the dynamics of different observed features. One which is well established is the occurance of high speed "sweeps", which are high speed regions approaching the wall at a shallow angle and separated from the low speed fluid in the streaks by a thin internal shear layer. The sweeps appear to initiate the bursting of the low speed streaks.

Of importance in connection with drag reduction is the different scales of motion associated with the turbulence production. Typical spanwise scales are of the order 20 to 30 in wall units, which is a typical spanwise extent of the low speed region. The streamwise scales of momentum carrying eddies is generally considerably larger, typically several hundreds in wall units. The

dimension in the direction normal to the wall is not so clearly established; one finds strong y-coherence (Blackwelder & Kaplan, 1976), and it appears that the effects of energetic wall eddies are felt far up into the inviscid regions of the boundary layer, possibly all the way into the outer wake region. Indeed, Head and Bandyopadhyay (1981) have proposed that the turbulent fluctuations are highly correlated along 45 degree lines throughout the boundary layer, which would indicate that the effects of the wall layer might penetrate througout the boundary layer. From this one might conclude that the dynamic processes everywhere in the boundary layer are completely controlled by the wall layer. However, Kline (1978) has pointed out that, in view of the observations in the visual experiments carried out at Stanford, which showed that fluid is interchanged between the wall layer and the outer regions (buffer and log regions), there must be interaction between the inner and outer layers, so that both must play a role in the dynamics. Klines' opinion is supported by recent experiments by Alfredsson and Johansson (1982), in which it was found that the time between bursts scales with a time scale intermediate between the inner and outer time scales. It might therefore be possible to achieve turbulent drag reduction either by manipulating the outer or the inner layers. The experience so far, however, indicates that manipulation of the inner region is the more efficient, as will follow from the description below.

3. Methods for turbulent drag reduction in liquids

We shall here review a number of methods for turbulent drag reduction in liquids which have been demonstrated in laboratory experiments, and in some cases also used in engineering applications. We will list them in order of decreasing efficiency.

a) Polymer additives

The most efficient and most thoroughly investigated method of turbulen drag reduction in liquids is that of adding certain long-chain polymers to the liquid. Polyethylene oxide (Polyox) of high molecular weight is an efficient such additive, which gives sizable drag reductions at such small concentrations as a few parts per million by weight. Other effient drag reducers are polyacrylamide (Separan) and polyacrylic acid. The most efficient drag reducer reported on in the literature thus far (Oliver and Bakhtiyarov, 1983) is Magnafloc LT 25, a polyacrylamide of very high molecular weight (15-20 million) which gives a measurable drag reduction already at a concentration of 0.1 ppm. Typical of most of the highly efficient polymeric drag reducers is their sensitivity to degradation, both through aging when in dilute concentrations and through mechanical wear. This property partly accounts for the spread in experimental data reported; differeces in the manner the polymer solutions are prepared, for example, may lead to differences in the mechanical degradation. A puzzling phenomenon, first documented by Virk et al. (1970) is the existence of a maximum drag reduction asymptote (se figure 1), which is the lowest drag coefficient that can be achieved regardless of the additive used.

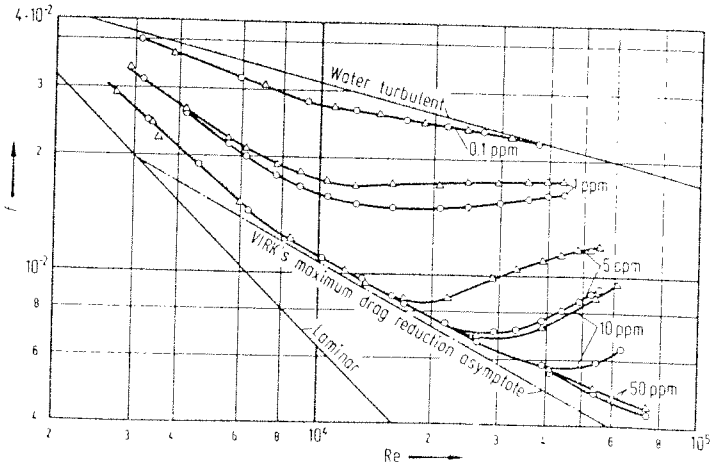


Figure 1. Pipe flow drag data for solutions of Polyox (from Paterson and Abernathy, 1970) (f = friction coefficient).

The drag at this asymptote is higher than the laminar value, indicating that the flow is still turbulent but with a reduced momentum transfer rate. Detailed investigations of the mean velocity profile shows that a new logarithmic layer develops, one with a different slope.

There has been a great deal of discussions in the literature about what mechanisms might be involved in polymer drag reduction, but no completely satisfactory explanation of the drastic effects observed has yet appeared. The experiments by Wells and Spangler (1967), who obtained large drag reductions by injecting polymers only in the wall region, seem to indicate that the turbulent momentum transfer is strongly controlled by processes in the wall region. For rheological reasons one would expect that the polymer additives would exert their influence primarily on the small-scale eddies. It is generally believed that the polymers impart viscoelastic properties on the fluid, and the departure from Newtonian properties would be greater for the smallest eddies which have the shortest natural time scales. However, viscoelasticiy as such appears not to be necessary for drag reduction, since large drag reduction has been observed also for suspensions of long slender but comparatively inelastic fibres, which show little elastic properties. More likely, it is the ansisotropy in fluid rheological properties produced by the additives that is responsible for the large dynamical effects. The (fully or partially) extended long-chain molecules tend to align themselves in the direction of elongational fluid motion and thus may inhibit vortex stretching (Landahl,1981). Such a stretching is believed to be essential for the maintenance of three-dimensional turbulence.

Although large drag reductions are possible with very small concentrations of polymer additives, the savings in fuel may nevertheless not justify their use for commercial ships. This can be seen from the following simple estimates:

The power required to overcome the frictional resistance can be directly related to the properties of the boundary layer at the trailing edge of the wetted surface. Per unit width of the

trailing edge it is

$$P = \rho U^3 \theta \tag{3}$$

where θ is the momentum thickness of the boundary layer, the fluid density and U the velocity. The fuel consumed per unit time and unit width of the boundary layer is

$$M_f = \rho U^3 \theta / \eta E \tag{4}$$

where E is the energy content of the fuel per unit weight and the overall efficiency factor of the propulsive unit. Assume that the additive concentration required to achieve a fractional drag reduction, ΔD, is c. It is reasonable to assume that the additive is needed only in the region of the boundary layer near the wall. For simplicity we shall assume that the additive is only needed wihin a region of thickness equal to the momentum thickness of the boundary layer. The amount of additive consumed per unit time and width of the boundary layer is therefore

$$M_a = \rho c \theta U \tag{5}$$

From (4) it follows that the amount of fuel saved, Ms, divided by the additive spent is

$$\frac{M_s}{M_a} = \frac{\Delta D}{D} \frac{U^2}{c \eta E} \tag{6}$$

As an example, take E=40MJ/kg, η =0.25 and U=10 m/s (20 knots). With these values (6) gives

$$\frac{M_s}{M_a} = \frac{\Delta D}{cD} \cdot 10^{-5}$$

In order for the drag reducing additive to be economical from a fuel savings point of view, the amount of fuel saved must substantially exceed the amount of additive used. A good drag reducer such as Polyox might give a 50% drag reduction at a concentration of 10 ppm. From (5) it follows that the amount of fuel saved is only half of what is spent in additive. Since the additive is an order of magnitude more expensive than the fuel (at least!) to use it would be highly uneconomical. To make any saving in overall fuel economy one would need additives which are orders of magnitude more efficient than the best ones available today. It is therefore highly unlikely that polymer additives would come into use for fuel savings within the forseeable future. However, for special applications, for example, in order to increase the speed of a vehicle which is power limited, might still be feasible. Other possible non-vehicular uses would be to increase the capacity of a pipe line, for example.

b) Gas bubbles

The idea of reducing the frictional resistance of a ship by introducing a gaseous layer along the surface as a lubricating layer is an old one with many patents dating back to the last century. The gas layer may be produced either by pumping air into the boundary layer or by electorlysis. The major difficulty with

this method is to keep the gas layer stable. There are two effects that produce breakup of the gas layer. Buoyancy tends to lift the large bubbles away. Also, dynamic instability of the Kelvin-Helmholtz type will tend to break up the layer into smaller bubbles. The bubbles may in turn act as roughness elements for the boundary layer and thereby negate any gain realized by having a fluid layer of lower density and hence lowered effective viscosity near the wall.

A somewhat new approach has been tried recently by the Russians. Bogdevich et al. (1977) injected air through a porous surface with microscopic holes which generated very small bubbles, with diameters of the thickness of the viscous wall layer. They were able to obtain drag reductions of the order 80%, however at the expense of large power requirement for the production of the bubbles. Nevertheless, when account was taken of this power loss, a net drag reduction of the order 30% resulted. This method seems quite interesting for ship applications, but several practical engineering problems need to be solved before the method can be used commercially.

c) Large eddy breakup devices

The idea with large eddy breakup devices is to try to inhibit the large eddies in the boundary layer primarily responsible for the momentum transfer in the inviscid portions of the boundary layer and thereby decrease the skin friction. This concept was first suggested by Yajnik. The original device explored by Yajnik and Acharya (1978) consisted simply of a screen set normal to the mean flow in the boundary layer. It produced a sizeable local skin friction reduction downstream (up to about 50%) but the drag of the device was too large to give a net drag reduction. Better results have been obtained with a device consisting of two horizontal thin ribbons located at a distance from the wall of about 80% of the boundary layer thickness and separated horizontally by a distance of about 8 ribbon widths (figure 2) which has been investigfated extensively by a group under Nagib at Illinois Institute of Technology (IIT). Corke et al. (1980) reported a substantial net drag reduction at distances downstream of the device greater than about 40 boundary layer thicknesses. Later, Corke et al. (1982) have reported a net drag reduction of up to about 20%. However, to date no verifications of these very promising results has yet appeared. The experiments by Bertelrud et al. (1982) gave no net reduction (see figure 2), possibly because a too short plate was used which did not allow measurements to be carried out sufficiently far downstream of the device to see whether the local skin friction reduction could be maintained sufficiently far downstream to compenste for the device drag.

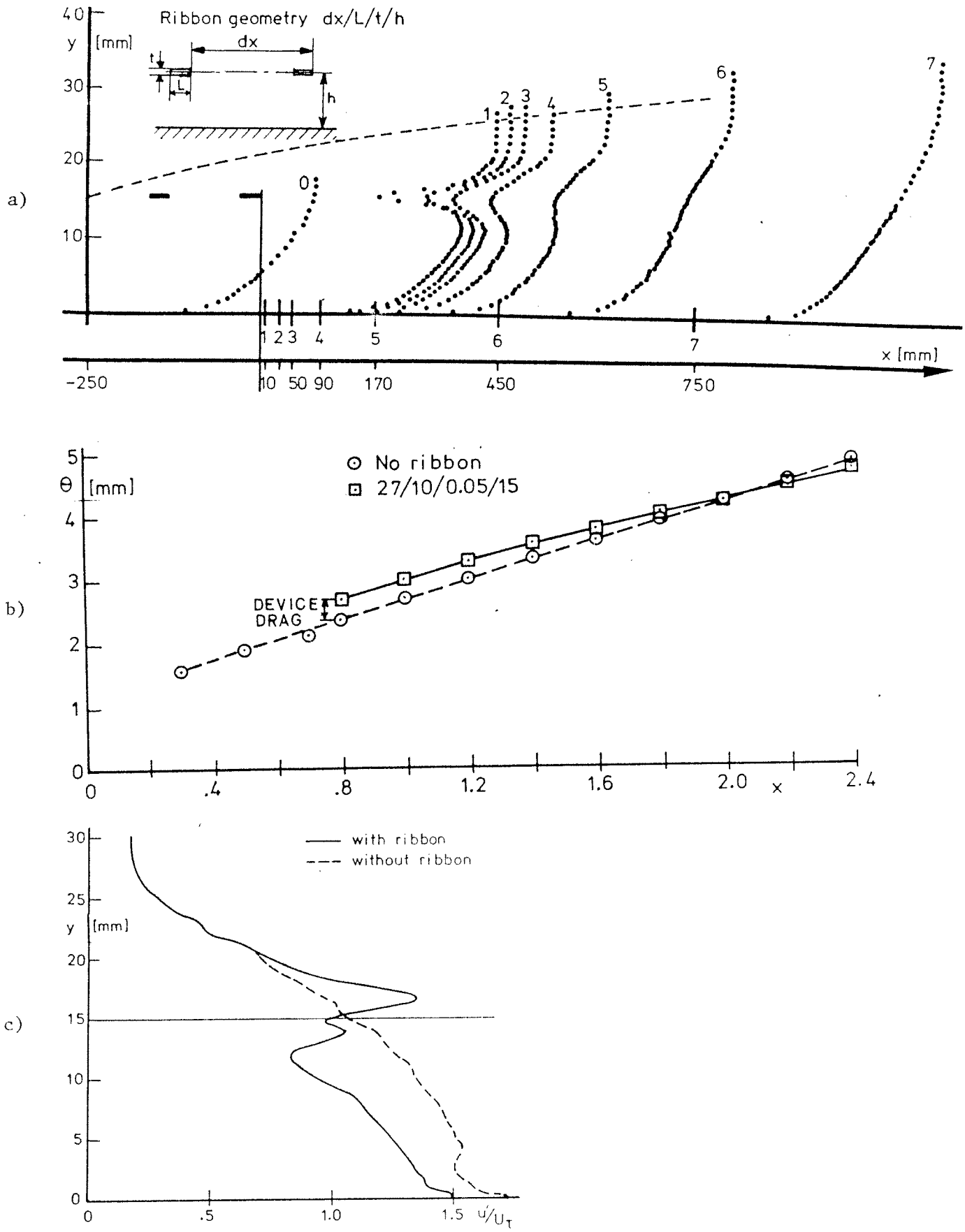


Figure 2. Results from investigations of drag reduction properties of transverse ribbons by Bertelrud et al. (1982)
a) Mean velocity profiles
b) Momentum thickness
c) Turbulence intensity

Also, experiments at NASA-Langley (Hefner et al. 1983) gave only a small net reduction (5% or less).

The experiments by Bertelrud et al.(1982) show that downstream of the device there is a reduction of the turbulence intensity in the region near the wall associated with the reduction in local skin friction. Thus, the turbulence is clearly modified by the device. However, all the experiments carried out so far have been made at comparatively modest Reynolds numbers and it remains to be seen whether it is possible to obtain net drag reductions at the extremely high Reynolds numbers of interest for ships.

d) Riblets and other surface structure modifications

The idea behind the use of riblets and other surface structure modifications is to influence the spacing and structure of the low-speed streaks in the viscous wall layer and thereby try to diminish the intensity of turbulent bursting. This concept has been extensively studied at NASA-Langley, where a number of different riblet configurations have been tested. The best configuration, which gave a drag reduction of about 10%, was one having a saw-tooth shape with a spanwise spacing and height between 10 and 25 in viscous wall units. Apparently, in order to affect the streaky structure the dimensions of the riblets must be of the order or smaller than the individual streak dimension, which is about 30 in viscous wall units.

The small dimensions of such riblets required at large Reynolds numbers will make application to ships rather difficult, if not impractical. With a friction velocity of 0.3 m/s (corresponding approximately to a forward speed of 20 knots) the spanwise dimension of the riblets would have to lie in the range of 0.03 - 0.08 mm. Even if one were to find methods to manufacture such surfaces, the problem of keeping them clean in ocean and harbor environments appears exceedingly difficult.

e) Compliant walls

A great deal of interest was generated about 20 years ago in a concept for drag reduction proposed by M. Kramer (1960), in which a specially designed compliant wall was supposed to stabilize the boundary layer and thereby make it possible to maintain a laminar boundary layer to Reynolds numbers far beyond that for natural transition. Kramer made experiments with coated plates towed behind a motor boat and claimed to obtain a substantial reduction in drag interpreted by him as due to an increase in the laminar portion of the boundary layer. His surfaces were designed to have high admittance at the frequences typical of the Tollmien-Schlichting instability waves and thereby be highly dissipative for such waves. Subsequent theoretical investigations by Benjamin (1960) and by Landahl (1962) showed, however, that dissipation in the flexible wall would lead to destabilization of the Tollmien-Schlichting waves. Thus, boundary layer stabilization by the flexible wall in the manner proposed by

Kramer was highly unlikely. Also, many laboratory experiments were undertaken to investigate the effects of compliant walls, both with laminar and turbulent boundary layers. However, apart from some recent Sovjet investigations (Kanarskiy and Teslo, 1980), there has been no experiments to date that have clearly demonstrated a drag reduction. This does not of course prove that wall flexibility cannot reduce the drag, but the possibility of discovering wall flexibility parameter ranges that produce sizeable drag reduction appears remote.

4. Conclusions

The review of the various methods for turbulent skin friction reduction through modification of the boundary layer turbulence structure that have been explored shows that only two at present appear feasible for application to ships, namely injection of air bubbles and large eddy breakup devices. All the others are either uneconomical (polymers), questionable as to their efficiency (compliant walls) or too difficult to construct and maintain (riblets). Further research might narrow the possibilities even further.

The fundamental difficulty in achieving turbulent drag reduction is that manipulation of the turbulence in the near wall region, where most of the energetic and momentum transfer processes appear to be controlled, would require methods to affect the small scales, which in practice can only be done through changing the rheological properties of the fluid such as what is done with polymer additives. Controlling the large scale eddies through the use of large eddy breakup devices does have an effect on the local skin friction, but it has been found difficult so far to hold the device drag low enough to give a net drag reduction. The most straightforward method appears to be to reduce the effective viscosity of the fluid in the region near the wall through injection of air bubbles, but in order for the bubbles to remain stable in suspension they need to be very small, and methods to produce small bubbles requires substantial power expenditure which would reduce the net drag gain. Nevertheless, this method appears promising enough to explore further.

References

Alfredsson, P. H. and Johansson, A. V. 1982 Rep. TRITA-MEK-82-03, Royal Inst. Technology, Stockholm.

Benjamin, T. B. 1960 J. Fluid Mech. 9, 513.

Bertelrud, A., Truong, T. V., and Avellan, F. 1982 AIAA Paper No. 82-1370

Blackwelder, R. & Kaplan,R.E. 1976 J Fluid Mech. 76, 89.

Bogdevich, V. G., Evseev, A. R., Malyuga, A., G. and Migirenko, G. S. 1977 in Second International Conference on Drag Reduction, D25.

Corke, T. C., Guezennec, Y. and Nagib, H. M. 1980 in Progress in Astronautics and Aeronautics, AIAA, 72, 128.

Corke, T. C., Nagib, H. M. and Guezennec, Y. G. 1982 NASA Contractor Report 165861.

Head, M. R. and Bandyopadhyay, P. 1981 J. Fluid Mech. 107, 297.

Hefner, J. N., Anders, J. B. and Bushnell, D. M. 1983 AIAA Paper No.83-0293

Kanarskiy, M. V. and Teslo, A. P. 1980 Fluid Mechanics, Soviet Research, 9, No. 5.

Kline, S. J. 1978 in Workshop on Coherent Structure of Turbulent Boundary Layers, Smith & Abbott Ed., AFOSR/Lehigh., 340.

Kramer, M. O. 1960 J. Amer. Soc. New. Eng. 72, 25.

Landahl, M. T. 1962 J. Fluid Mech.13, 609.

Landahl, M. T. 1980 Progress in Astronautics and Aeronautics AIAA, 72, 300.

Oliver, D. R. and Bakhtiyarov, S., I. 1983 J Non-Newtonian Fluid Mech. 12, 113.

Paterson, R. W. and Abernathy, F. H. 1970 J. Fluid Mech. 43, 689.

Virk, P. S., Merrill, E. W. and Mollo-Christensen, E. L. 1970 J. Fluid Mech. 30, 305.

Wells, C. S. and Spangler, J. G. 1967 Phys. Fluids 10, 1890.

Yainik, K. S. and Acharya, M. 1978 Lecture Notes in Physics, Structure and Mechanisms of Turbulence 1, 249, Springer Verlag.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

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SHIP DESIGN FOR FUEL ECONOMY

BOTTOM CONDITION AND FUEL CONSERVATION

by

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Delivered
Date: 31st August

BOTTOM CONDITION AND FUEL CONSERVATION

1. INTRODUCTION

In another lecture (O 6 - 8th Sept.) Milne will describe the characteristics and measurement of hull surface roughness. The purpose of these notes is to discuss methods for the calculation of the resistance penalty associated with hull and propeller surface roughness.

Hard and soft, shell and weed fouling will not be considered in these notes since self-polishing copolymer antifoulings have the potential to keep ships foul-free over any inter-dock period. Biological slimes will however be referred to since they can inhabit all toxic surfaces to some extent.

It is important to distinguish between roughness in service and new ship roughness for the purposes of this course on ship design for fuel economy. The designer has little control over the deployment and maintenance of the ship in service but will have an interest in the new painting specification and the ship performance on trial. New ship roughness values at each position on the hull (M.H.R.), should lie within a relatively narrow band and is due solely to paint rheology and quality of application. The mean hull roughness is given by

$$MHR = \sum R_t(50) \div \text{number of 50mm samples (say 12)}$$

Ships in service however have a wide range of MHR values due to surface damage in particular locations e.g. anchor and cable damage at the fore end, fender damage on the vertical sides and bottom damage due to grounding. Under the heading of damage, paint system failures in service may be included.

It follows from the above that a single number description of a hull's surface roughness, is more adequate for a new hull than one which has been in service for some time. The internationally adopted measure is the BSRA Average Hull Roughness (AHR) as derived from their Hull Roughness Analyser and

$$AHR = \sum MHR \div \text{number of measuring stations (say 100)}$$

For a standard method of measurement and histogrammatic presentation of grouped and ungrouped roughness data, $R_t(50)$ values, see ref.(130)*.

2. ADDED RESISTANCE CALCULATIONS USING SINGLE NUMBER ROUGHNESS DEFINITION

2.1 Telfer's reanalysis of Froude sand roughened plane results

Professor E.V.Telfer urged the profession to view the coefficient of smooth surface friction as a linear function of $R_n^{-1/3}$. The view was not universally accepted. In the same paper (112) he argued that the coefficient of rough surface friction should be a similar function but with the slope being dependent upon roughness height Reynolds number since C_{FR} is independent of Reynolds number, so that

$$C_{FR} = A\left(\frac{k}{L}\right)^{1/3} + B \qquad (1)$$

and since

$$C_{FS} = C + D\,R_n^{-1/3} \qquad (2)$$

the roughness increment

$$\Delta C_F = A\left(\frac{k}{L}\right)^{1/3} + (B-C-D.R_n^{1/3}) \qquad (3)$$

*Numbers in ( ) refer to items in the appended WEGEMT bibliography: MARINE ROUGHNESS & DRAG

Whilst this is both speed and length dependent, for high Reynolds numbers we may write

$$\Delta C_F = a\left(\frac{k}{L}\right)^{1/3} + b \qquad (4)$$

Telfer was not able to take this work beyond the sand-roughened surface data available at the time viz. the Prandtl-Schlicting intepretation of Nikuradze's pipe tests and the Froude sand roughened planes. He finally interpreted sand roughened surface friction as

$$C_{FR} = 75\left(\frac{k_s}{L}\right)^{1/3} + 1.2 \qquad (5)$$

As has been pointed out recently, particularly by Grigson (40, 41), rough ship surfaces are not sand-like but fall into the general category of engineering rough surfaces as studied by Colebrook and White (20). If the one third power law for ΔC_F applies to ship surfaces the constants need to be re-evaluated.

2.2 The ITTC correlation allowance

The 1978 ITTC correlation allowance formula was developed by Bowden (7) from the analysis of good quality model experiments and new ship trials for ten single-screw ships with lengths within the range 157 to 267m and having measured roughnesses between 144μm and 206μm, AHR. The analysis was sensitive to the choice of form factor and three were tried viz. those of Lindgren, Tagano and Watanabe. Each ship trial prediction was made by two methods - firstly by assuming the correlation allowance to be added to the ITTC line and secondly by assuming it added to the total viscous resistance. The correlation allowances which made the model prediction agree with the trial results were then plotted to a base of $\left(\frac{k}{L}\right)^{1/3}$ for the ten ships and for each of the form factor methods. The results showed considerable scatter but finally a mean representative line was chosen following the Telfer form:

$$10^3\,\Delta C_F = 105\left(\frac{k}{L}\right)^{1/3} - 0.64 \qquad (6)$$

It is important to recognise that in a correlation allowance length has an independent effect upon ΔC_F and that other factors of form and speed which are relevant are not included in the formula above. These considerations and the inherent inaccuracy of the analysis mean that the ITTC correlation allowance should not be taken as a roughness penalty predictor. Indeed various authorities believe that the ITTC correlation allowance overestimates roughness effects alone. In ref.(130) it is advised to reduce ΔC_F (ITTC) by 1/3rd to arrive at the roughness effect alone.

2.3 A roughness penalty predictor

A computer based integral method for the calculation of ship friction where there is a distribution of roughness over the hull is described in ref.(131): it will be outlined later in these notes. The method was used to calculate the roughness increment ΔC_f for four ship types each with a range of roughness between AHR = 100μm and 500μm and each travelling at a number of speeds. Some of the results are shown in fig.(1) as ΔC_f against $(k/L)^{1/3}$.

i. All 11 curves are nearly linear
ii. They all have approximately the same slope
iii. The mean slope is less than the ITTC correlation allowance formula
iv. ΔC_f values are less than ΔC_fITTC except for very long ships at low roughness
v. Speed makes a difference to ΔC_f
vi. Ship type (size and speed) makes a difference to ΔC_f.

Observations v. and vi. recall eqn.3 above where it was noted that from Telfer's work ΔC_f would be speed dependent. Using the form of eqn.3 the constants have been evaluated to give a good fit to the results in fig.(1):

$$10^3 \Delta C_f = 44 \left[\left(\frac{k}{L}\right)^{1/3} - 10 R_n^{-1/3} \right] + 0.115 \qquad (6)$$

This formula may be used as a roughness penalty predictor for hulls with a uniform roughness. For vessels with extreme surface damage in one location the prediction may be less accurate.

3. ADDED RESISTANCE CALCULATIONS FOR DISTRIBUTED ROUGHNESS USING INTEGRAL METHODS

3.1 Integral methods

If more detailed calculations of roughness effects are required than are possible with a simple predictor (eqn.6) then account must be taken of the following:

i. Pressure and velocity distribution over the hull and their changes with speed due to wavemaking.

ii. Variation in roughness over the hull.

iii. Improved roughness definition viz. measures of texture as well as height.

The preliminary to the solution of the momentum integral equations for boundary layer flow is the determination of the pressure gradients over the hull. A rigorous method is difficult to achieve but is required if accurate predictions of smooth ship friction C_{FS} are to be made. Some relief is possible in the present circumstances however since only the difference between rough and smooth friction is required $\Delta C_F = C_{FR} - C_{FS}$. The roughness function which is to be used in the momentum integral equation to predict C_{FR} must however be the best possible.

One approximate method (131) of obtaining the hull pressure distribution is to calculate the distribution on bodies of revolution defined by the rotation of five representative streamlines over the hull surface. The streamlines are obtained from experiment or theoretical calculation and five are chosen to suit the distribution of roughness data in a standard hull survey.

The boundary layer integration is based on a 'Thin axisymmetric' method with the cross-flow momentum equation uncoupled from its streamwise counterpart. The equations differ from the two dimensional (aerofoil) case by the inclusion of a streamline convergence term K_1. Further details of the integration procedure are given in Appendix I and (131). Each side of the wetted surface area of the hull, port and starboard, is divided into five horizontal strips, each one of which is associated with one of the streamlines. Each strip is divided at 20 stations into panels and the drag force on each panel is calculated.

3.2 The roughness function

Local friction needs to be calculated in association with local roughness measures which may have to include texture parameters. Fundamentally, this requires prediction of the boundary layer velocity distributions as influenced by the local roughness.

The logarithmic law satisfactorily describes measured velocity distributions through the boundary layer, except at the inner and outermost regions:

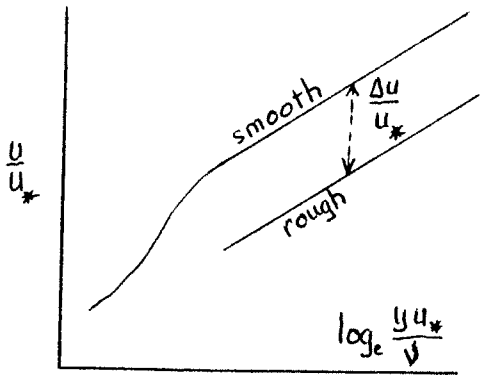
$$\frac{u}{u_*} = \frac{1}{K} \log_e \frac{yu_*}{\nu} + B$$

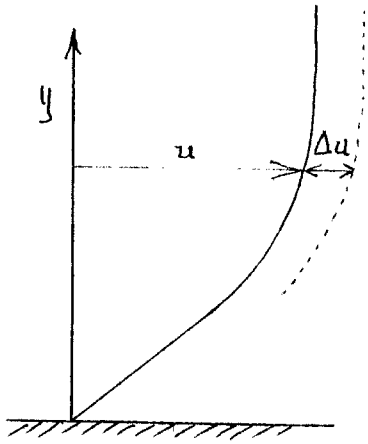
where $u_* = \sqrt{\frac{\tau_o}{\rho}}$

u = mean velocity in x direction

y is measured perpendicular to the boundary surface.

For rough surfaces the plot of $\frac{u}{u_*}$ against log $\frac{yu_*}{\nu}$ is moved vertically downwards by an amount $\frac{\Delta u}{u_*}$





Hence

$$\frac{u}{u_*} = \frac{1}{K} \log \frac{yu_*}{\nu} + B - \frac{\Delta u}{u_*}$$

where now τ_o has the rough boundary value.

Over the region of validity of the logarithmic law therefore the velocity u is augmented by Δu.

$\frac{\Delta u}{u_*}$ is one form of the <u>roughness function</u>

Another form is

$$\chi = B + \frac{1}{K} \log_e \frac{hu_*}{\nu} - \frac{\Delta u}{u_*}$$

whence

$$\frac{u}{u_*} = \frac{1}{K} \log_e \frac{y}{h} + \chi$$

and assuming that the roughness can be characterised by a height measure h. The problem therefore is to relate the roughness function $\frac{\Delta u}{u_*}$ to some measure of the roughness through the medium of experiments.

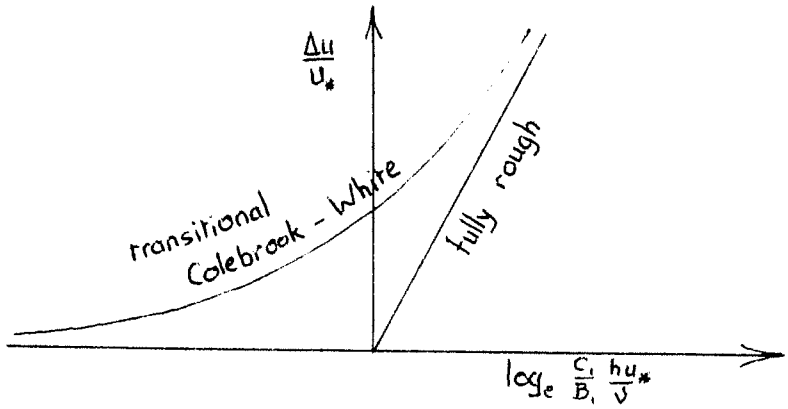
From experiments with rough pipes Colebrook & White showed that for industrially rough surfaces

$$\frac{\Delta u}{u_*} = \frac{1}{K} \log_e \left[\frac{C_1}{B_1} \frac{hu_*}{\nu} + 1 \right]$$

When $\frac{\Delta u}{u_*}$ is plotted against $\log_e \left[\frac{C_1}{B_1} \frac{hu_*}{\nu} \right]$ the curve is asymptotic to:

$\frac{\Delta u}{u_*} = 0$ ... hydraulically smooth

and $\frac{\Delta u}{u_*} = \frac{1}{K} \log_e \left[\frac{C_1}{B_1} \frac{hu_*}{\nu} \right]$ .... "fully rough" flow



By fully rough flow we mean that each roughness element has an effect by reason of its form drag i.e. a pressure effect only.

Generally ship hull roughness does not cause a fully rough flow. The satisfactory establishment of a Colebrook-White type transitional flow roughness function depends upon the adequacy of the height measure h to characterise the rough surface.

Musker found (83, 85, 72) that for 5 sample hull surfaces in the range good to bad, M.H.R. was an inadequate measure for h. He then introduced other measures of surface topography to provide an equivalent h. Thus

$$h' = \sigma_r \, (1 + a \, s_p)(1 + b \, s_k \, k_u)$$

where σ_r = standard deviation of roughness profile about the mean

s_p = average local slope

s_k = skewness of height distribution (if s_k (+ve) the peaks more dominant than valleys)

k_u = kurtosis (flatness or peakedness) of the distribution i.e. departure from Gaussian.

A more satisfactory relationship was then found between $\frac{\Delta u}{u_*}$ and h'. We may note however that a sample of 5 from the whole population of ships hulls is small and although 2 of Musker's surfaces were inadequately represented by M.H.R. this may not generally be true.

Figure (2) shows ΔC_F for the 5 Musker surfaces plotted against $(\text{M.H.R.})^{1/3}$ together with ΔC_F values from NSRDC 21 ft. test plane results (116) with a wide variety of ship paint surfaces. Whilst there is some scatter a trend is discernable among all the painted surfaces but with two of the Musker surfaces as 'rogues'. M.H.R. therefore may not be as inadequate a measure as is often supposed and we may use it for ship surface description whilst research progress is awaited.

4. ADDED POWER DUE TO ROUGHNESS

In Ref.(127) power increase due to roughness was considered on the basis of the ITTC correlation allowance: it will now be re-considered on the basis of eqn.6 - the present state of the art.

The increment in power ΔP due to roughness k as a percentage of smooth power P is

$$100 \frac{\Delta P}{P} \% = 100 \frac{\Delta C_F}{C_T} \% = \frac{44}{10 \, C_T} \left[\left(\frac{k}{L}\right)^{1/3} - 10 \, R_n^{-1/3} + \frac{.115}{44} \right] \%$$

Usually the % power change at constant speed due to a change in roughness k_1 to k_2 is being considered, then

$$100\,\frac{\Delta P_1 - \Delta P_2}{P}\,\% = \frac{44}{10\,C_T}\left[\left(\frac{k_1}{L}\right)^{1/3} - \left(\frac{k_2}{L}\right)^{1/3}\right] \qquad (7)$$

It will be noted that the slope of eqn.6 is now all important. Referring to fig.1 the slope of eqn.6 is much less than that of the ITTC correlation allowance and hence the power increment is less.

In a particular case with C_T known at, say, service speed, the power penalty for increasing roughness may easily be estimated.

In the general case advantage may be taken of Telfer's idea that C_F and hence C_V could be a linear function of R_n -1/3. From these beginnings in ref.(127), the coarse assumption was tested that at normal service power and speed most ships could be represented by a relationship between C_T and $L^{1/3}$. The relationship derived from a large sample of single screw ships over 100m in length was

$$\frac{1}{C_T} = 55.2\ L^{1/3} \quad \ldots\ L \text{ in (m)}$$

Thus

$$\frac{\Delta P_1 - \Delta P_2}{P} \times 100\% = \frac{44 \times 55.2}{10^3}\ (k_1^{1/3} - k_2^{1/3}) \quad \ldots\ k(\mu m)$$

$$\%\ \text{power change} = 2.5\ (k_1^{1/3} - k_2^{1/3}) \quad \ldots\ \text{say, with } k \text{ in } (\mu m) \quad \ldots\ (8)$$

Thus in making an effort to improve newbuilding roughness from 125μm 'good practice' to say 85μm 'best practice' (see Table 1 lecture 06 Sept.8 - Milne) there is a potential gain of 1.5% in power. This is rather less than Lackenby's rule of thumb 1% power per 10μm.

APPENDIX 1

The basic equation is the thin axisymmetric momentum equation of Cumptsy and Head*

$$\frac{1}{2} C'_f = \frac{d\theta}{ds} + \theta \left[(H+2) \frac{1}{U_s} \frac{dU_s}{ds} - K_1 \right]$$

together with the entrainment equation

$$C_E = \frac{d(\delta - \delta^*)}{ds} + (\delta - \delta^*) \left[\frac{1}{U_s} \frac{dU_s}{ds} - K_1 \right]$$

Standen's approximation to Head's curve (Stamford Conference Vol.I) is used for C_E

$$C_E = 0.0306 \ (H^* - 3.0)^{-0.653}$$

For the skin friction equation Coles Law of the Wall/Law of the Wake (21) is used in the form

$$\frac{U_s}{u^*} = \frac{1}{K} \ \ell_n \frac{\delta u^*}{\nu} + B_o + \frac{2\pi}{K} - \frac{\Delta u}{u^*}$$

$\frac{\Delta u}{u^*}$ is the roughness function.

Coles' two-parameter family of velocity profiles, the similarity laws, are used in the modified form derived by Musker (83, 85). The integral thicknesses δ^* and θ and the shape parameters H and H* are expressed as functions of $\sqrt{\frac{C'_f}{2}}$ and Coles' wake strength parameter π.

The method of solution of the differential equations is that of Lewkowicz et al (73).

*Cumptsey, N.A. & Head, M.R.: The calculation of 3-dimensional turbulent boundary layers - Pt.I: flow over the rear of an infinite swept wing. Aero.Q. Vol.18 1967, pp.55-84.

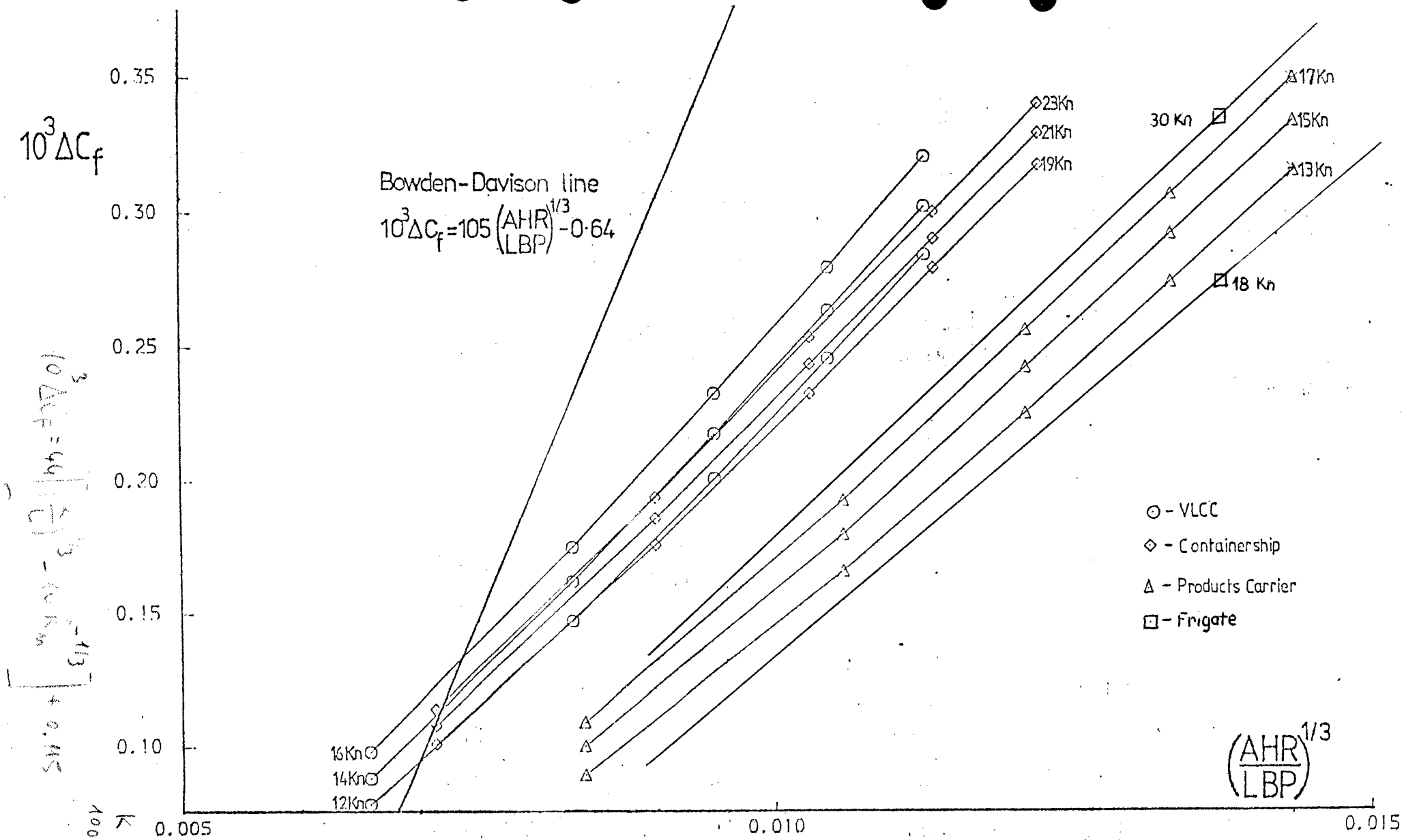


FIG. 1 - ΔC_f - $\left(\frac{AHR}{LBP}\right)^{1/3}$ relationship for 4 ship types: evenly distributed roughness, 100 - 500μm MAA

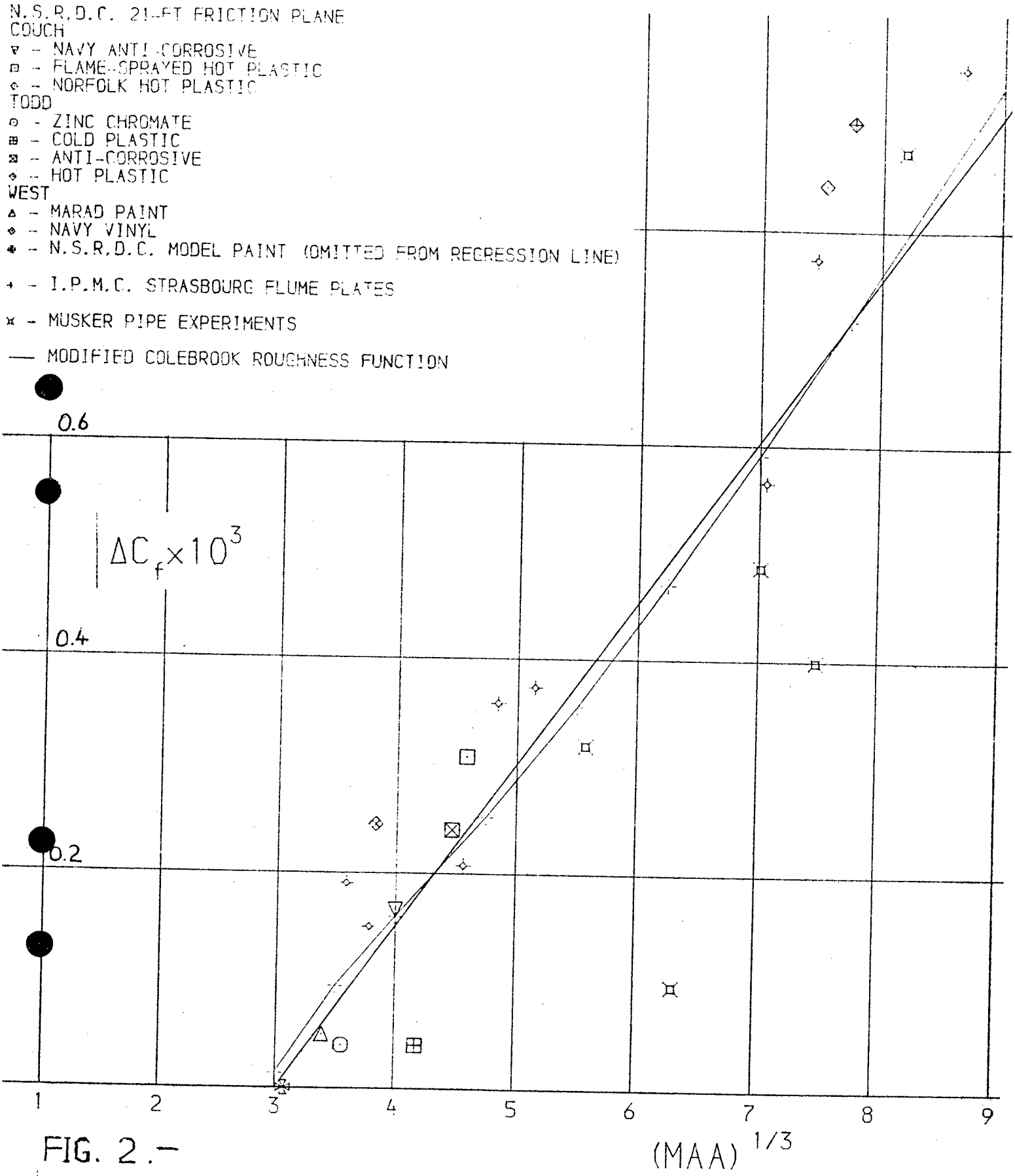


FIG. 2.—

CALIBRATION

OF ROUGHNESS FUNCTION - DTMB 21 FT FRICTION PLANE

BOTTOM CONDITION AND FUEL CONSERVATION - PART II

4. THE ROUGH HULL, ROUGH PROPELLER PROBLEM

Again, the designer has little control over hull and propeller maintenance and since the propeller is never rough in the new ship case, this section is only included for the sake of completeness. For more detailed accounts see (13) (110), (130) and (142).

4.1 The rough hull effects on the propulsive efficiency

i. Added resistance at constant speed generally results in a decrease in propulsive efficiency.

ii. Hull roughness increases boundary layer thickness and hence changes the wake fraction.

Following Sasajima the wake fraction ω_T may be divided into two parts

$$\omega_T = \omega_P + \omega_F$$

where ω_P is independent of Reynolds number but ω_F is a function of the viscous resistance C_V. From these ideas ITTC adopted the following wake scaling formulation in going from model (M) to ship (S) values:

$$\omega_{TS} = (t + 0.04) + (\omega_{TM} - t - 0.04) \frac{C_{VS}}{C_{VM}}$$

This formula may be used to estimate wake changes due to hull roughness increments ΔC_F to C_V.

Svensen has shown for a number of single screw ships (142) that although there is a significant decrease in open water efficiency due to increased loading on the propeller the effect is nearly cancelled by the increase in hull efficiency due to the increase in effective wake.

4.2 Rough propeller effects

The cost of maintaining propeller smoothness is considerably less than maintaining hull smoothness for the same penalty, there is therefore less pressure to undertake techno-economic investigations of propeller roughness. The surface of new propellers is finished to a standard, currently ISO R384,1966. This standard is now being revised to use a Centre Line Average measure R_a, based on a long wave cut off of 0.8mm. Such a standard measure although adequate to define machining standards is deficient for hydrodynamic purposes and in particular the long wave cut off of 0.8mm is concealing important surface information.

Apart from the early measures reported by Wellman (137) there is little information on propeller surface condition in service. Recent publications (130), (13) have improved our knowledge and there has also been useful linking of the roughness of Rubert comparator gauges into contemporary calculations (110). Since there is no agreed standard measure of propeller in-service surface roughness as there is for hulls, BSRA (R_t 50), the roughness function must be estimated from a more elaborate measure and that of Musker (see section 3.2 of Part I) can be used.

Using an integral method, such as that described in section 3.0 of Part I, 2-dimensional calculations of propeller section rough drag may be made and incorporated into standard propeller vortex analysis methods. The results of such a calculation are shown in Fig.3 taken from (142). It is interesting that as with hull roughness there is a linear relationship between the power penalty and (roughness height)$^{1/3}$. In this case the 'roughness height' is that of Musker, h'.

Also in Fig.3 is shown the equivalent Rubert gauge letter with surfaces described as:

A - new propeller
B - recondition, 'polished' propeller
C - F in-service roughness with F 'very bad'

5. TECHNO-ECONOMIC CONSIDERATIONS

The most detailed analysis of the economic consequences of hull and propeller roughness and various maintenance strategies may be found in ref.(142).

The problem may be studied as a roughness penalty at constant speed with the consequent increase in fuel consumption or at constant power with attendant loss of speed and trading profit: the latter case may only be studied with the benefit of commercial data and the analysis can be complex. In ref.(142) a method is offered for converting from a constant speed study to a constant power result, using,

$$\frac{C_S}{C_P} = 3.1 \frac{F}{I}$$

where
C_S = cost of added resistance at constant speed
C_P = cost of added resistance at constant power
F = daily fuel cost at sea
I = daily average income after deductions for cargo handling and port changes

The constant 3.1 applies to diesel engined ships with a lesser value for steam turbine machinery between 1.85 and 2.33.

A worked example of the simplified method is given in (142). Fig.4, also taken from (142), shows the results from more detailed calculations for four ship types:

SHIP A - 37000 tdwt Parcel carrier
B - 3000 teu Container ship
C - 350,000 tdwt V.L.C.C.
D - Panamax bulk carrier

The plot shows the annual cost per unit area of wetted surface for a given increment in hull roughness from an assumed standard of 125 μm AHR. These values may be interpreted as the maximum amounts available for maintenance procedures in order to reduce the roughness by given amounts.

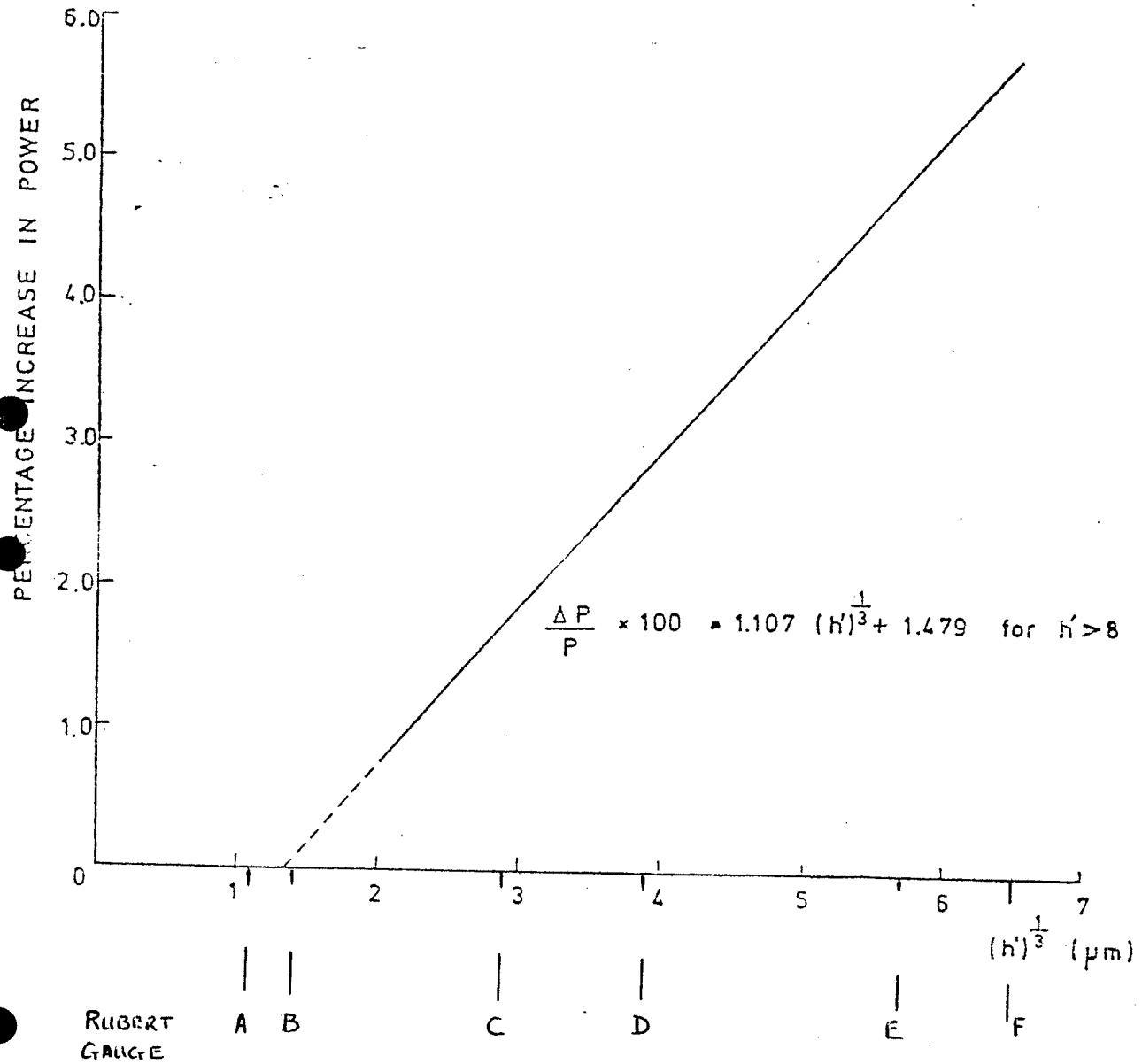

PERCENTAGE INCREASE IN POWER
6.0
5.0
4.0
3.0
2.0
1.0
0
1
2
3
4
5
6
7
$\frac{\Delta P}{P} \times 100 = 1.107\ (h')^{\frac{1}{3}} + 1.479$ for $h' > 8$
$(h')^{\frac{1}{3}}$ (μm)
RUBERT GAUGE
A
B
C
D
E
F

FIG 3.

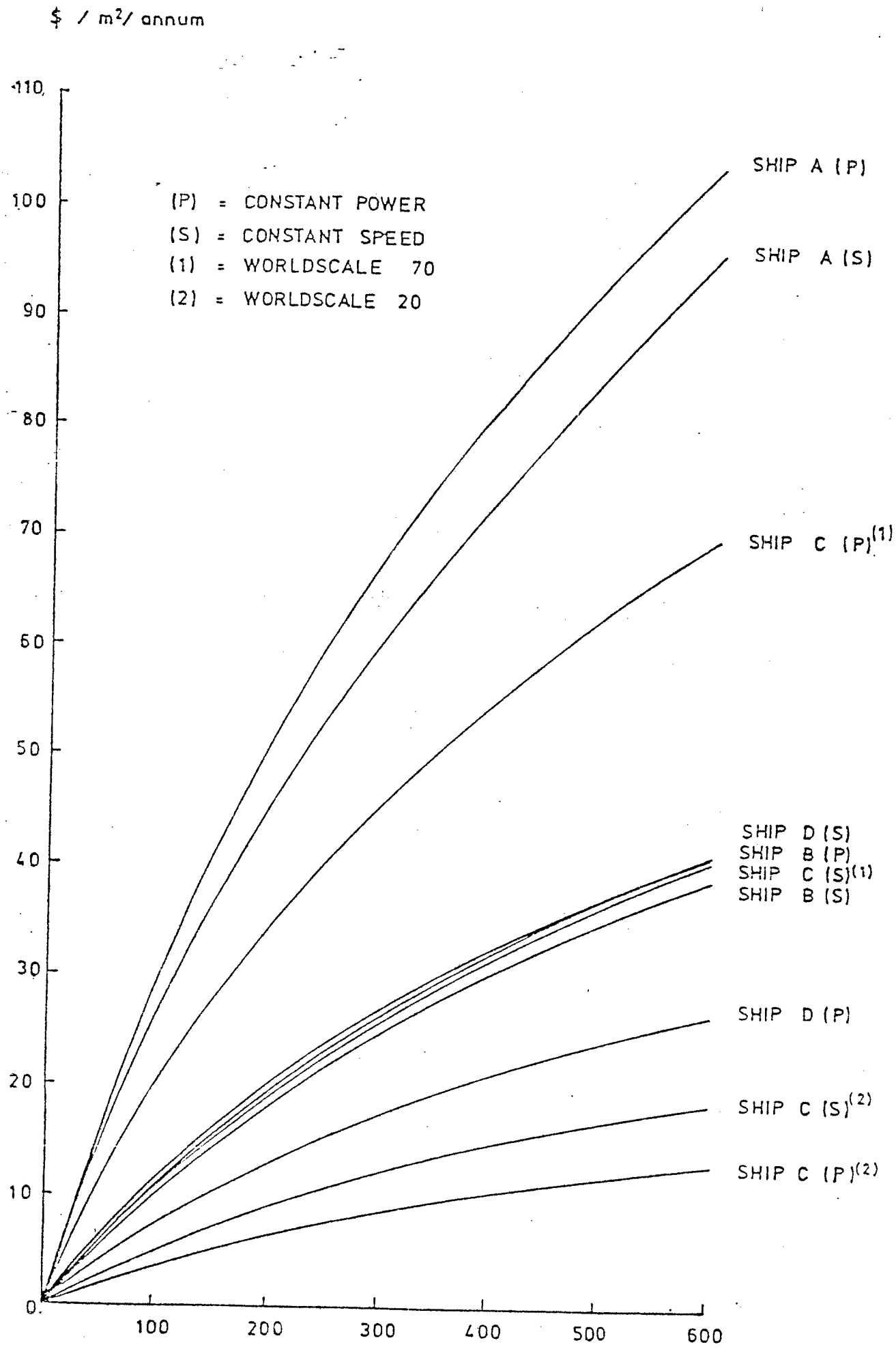

$ / m²/ annum
110
100
90
80
70
60
50
40
30
20
10
0
(P) = CONSTANT POWER
(S) = CONSTANT SPEED
(1) = WORLDSCALE 70
(2) = WORLDSCALE 20
SHIP A (P)
SHIP A (S)
SHIP C (P)(1)
SHIP D (S)
SHIP B (P)
SHIP C (S)(1)
SHIP B (S)
SHIP D (P)
SHIP C (S)(2)
SHIP C (P)(2)
100
200
300
400
500
600
INCREMENT TO A.H.R.(μm)

FIG. 4.

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February 1983 Edition.

1. Allan,JF. & Cutland,RS. The effect of roughness on ship resistance. In: Trans NECIES. 1955-56.

2. Altshul,AD. Frictional resistance of plates with production process roughness at a turbulent boundary layer. BSRA, translation no. 1967, from Proc. USSR Academy of Sci.

3. Antonia,RA. & Luxton,RE. The response of a turbulent boundary layer to a step change in surface roughness. Part 1: smooth to rough. In: J. Fl. Mech. 48 1971 pp721-761 .

4. Antonia,RA. & Luxton,RE. The response of a turbulent boundary layer to a step change in surface roughness. Part II : rough to smooth. In: J. Fl. Mech. 53 1972 pp737-757.

5. Baba,E. & Tokunaga,K. Study of local roughness effect on ship resistance for effective cleaning and protection of hull surface. In: Symp. Shipboard Energy Conservation '80. SNAME 1980.

6. Boersma,G. & Tasseron,K. Propeller maintenance, propeller efficiency and blade roughness. In: ISP. 1972 Sept.

7. Bowden,BS., & Davidson,NJ. Resistance increments due to hull roughness associated with form factor extrapolation methods. NMI, Ship TM 380, Jan 1974 (correction to TM 356).

8. Brown,DK. Fouling and economic ship performance. In: Inter Naval Corrosion Conf., 5th, April 1975.

9. Burrill,LC. Calculation of marine propeller performance characteristics. In: Trans NECIES 60 1943

10. Burrill,LC. The optimum diameter of marine propellers : a new design approach. In: Trans NECIES 72 1956.

11. Byrne,D. Hull roughness of ships in service. NECIES, Junior Section, paper read 1980 Feb 15th.

12. Byrne,D. The hull roughness of ships in service. Univ Ncle u Tyne, DNAS, MSc thesis, 1980, 2 vols, 330+440pp.

13. Byrne,D., Fitzsimmons,PA., & Brook,AK. Maintaining propeller smoothness : a cost effective means of energy saving. In: Symp. Shipboard Energy Conservation '82. SNAME 1982.

14. Canham,HTS., & Lynn,WM. The propulsive performance of a group of intermediate tankers. In: Trans RINA 104 1962.

15. Chaplin,PD. The analysis of hull surface roughness records. In: European Shipbuilding 2 1967.

16. Chen,DK. & Robertson,JA. The structure of turbulence in the wakes of roughness elements. In: ASCE, Hydraul.Div., Speciality Conf. 1971. Iowa Univ, Iowa City.

17. Christie,AO., Evans,LV., & Callow,ME. A new look at marine fouling. Pts 1-4. In: Shipping World and Shipbuilder. 1975 Oct.

18. Clauser,FH. The turbulent boundary layer. In: Adv. Appl. Mech. 4 1956 pp1-51.

19. Colebrook,CF. Turbulent flow in pipes, with particular reference to the transition region between the smooth and rough pipe laws. In: Jou.Inst.Civ.Eng. II 1938/39.

20. Colebrook,CF. & White,CH. Experiments with fluid friction in roughened pipes. In: Proc.Royal.Soc. London. A161 1937 pp367-381.

21. Coles,D. The law of the wake in the turbulent boundary layer. In: Jou. Fl. Mechs. 1 (2) 1956.

22. Conn,J. et.al. BSRA resistance experiments on the 'Lucy Ashton' Pt. II. In: Trans RINA. 1953. (see also Pt. I in 1951 and Pt. IV in 1955).

23. Couch,RB. Preliminary report of friction plane resistance tests of anti-fouling ship bottom paints. DTMB, NSRDC, Rep. 789, 1951.

24. Czarnecki,KR. & Monata,WJ. Roughness drag data due to two dimensional fabrication type surface roughness of an ogive cylinder from force tests at transonic speeds. NASA, TN D-5004, 1969.

25. Dashnaw,FJ. et al. Development of protective covering systems for steel and bronze ship propellers. In: Trans SNAME 88 1980.

26. Davis,RE. On the turbulent flow over a wavy surface. In: J. Fl. Mech. 42 1970 pp721-731.

27. Dvorak,FA. Calculation of turbulent boundary layers on rough surfaces in pressure gradients. In: AIAA J. 7 1969 pp1752-1759.

28. El-Samanoudy,MA. Turbulent boundary layers on an irregularily rough surface. Univ. Liverpool, Ph.D. thesis, 1974.

29. Ellingsen,PA. et.al. Improving fuel efficiency of existing tankers. In: SNAME 1977 May.

30. Emerson,A. Roughness and scale effect on propellers. In: Int. Shipbdg. Prog. 5 (43) 1958 March.

31. Ferguson,JM. The effect of surface roughness on the performance of a model propeller. In: Trans RINA. 1958.

32. Gaudet,L. & Winter,KG. Measurements of the drag of some characteristic aircraft excrescences immersed in turbulent boundary layers. RAE, TMA 1538, 1973.

33. Gawn,RWL. Roughened hull surface. In: Trans NECIES 58 1942.

34. Granville,P. A modified law of the wake for turbulent shear flows. In: J. Fluids Eng. ASME I 98 (3) 1976.

35. Granville,P. A prediction method for the viscous drag of ships and underwater bodies with surface roughness and/or drag reducing polymer solutions. NSRDC, Rep. SPD 797-01, 1977.

36. Granville,P. The torque and turbulent boundary layer of rotating discs with smooth and rough surfaces, and in drag reducing polymer solutions. In: JSR 17 (4) 1973.

37. Granville,PS. Drag-characterisation method for arbitrarily rough-surfaces by means of rotating disks. In: ASME 1981 June.

38. Granville,PS. The frictional resistance and turbulent boundary layer of rough surfaces. In: JSR 2 (3) 1958 Dec.

39. Grigson,CWB. Propeller roughness, its nature and its effect upon the drag coefficients of blades and ship power. In: Trans RINA 124 1982.

40. Grigson,CWB. The drag coefficients of a range of ship surfaces. In : Trans RINA 123 1981.

41. Grigson,CWB. The drag coefficients of a range of ships surfaces - II. In press: Trans RINA 125 1983 or Naval Architect 13 1983

42. Gronwall,PE. & Zink,PF. A containership operators program of bottom maintenance for reducing fuel consumption. SNAME, STAR Symp., 1982.

43. Guern,B.Le. Aspect Economique du traitement des surfaces de carenes. In: Bull. Tech. ATMA 1982.

44. Gumbel,EJ. Statistical theory of extreme values and some practical applications. In: Nat.Bu.Stds.App.Maths. 33.

45. Gyorgyfalvy,D. Effect of pressurization on airplane fuselage drag. In: J. Aircraft 2 (6) 1965.

46. Hacking,H. The economy of smooth hulls. In: Int. Ship Painting and Corrosion Conference, London, 1974. Proc.

47. Hacking,H. & Rippon,AH. Hull roughness and surface deterioration of ships built since 1965. BSRA, Rep. NS455, 1976.

48. Hama,FR. Boundary layer characteristics for smooth and rough surface. In: Trans SNAME 62 1954.

49. Hama,FR. Boundary layer characteristics for smooth and rough surfaces. In: Trans SNAME 62 1954.

50. Hamlin,NA. An exploratory study of the effect of arbitrarily distributed roughness on the frictional resistance of ship forms. Webb Institute, Centre for Maritime Studies, Rep. to SNAME Panel H-2, 1979.

51. Head,MR. Entrainment in the turbulent boundary layer. ARC, R&M 3152, 1960.

52. Head,MR. Improved entrainment method for calculating turblent boundary layer development. ARC, R&M 3643, 1963.

53. Hinze,JO. Turbulence, an introduction to its mechanism and theory. McGraw-Hill, 1959.

54. Hoerner,SF. Fluid-dynamic drag.

55. Hoerner,SF. The effect of roughness concentration upon the frictional drag. In: Amer.Soc. Naval Eng. (2) 1974.

56. Hoyt,JW. Use of Porphyridium Aerugineum as a scaling aid in towing tanks. In: ITTC, 11th. Proc.

57. ITTC, 1972. Perf. Committee App. 2, Sect. 5. Influence of friction drag on the propeller characteristic. In: ITTC 1972.

58. ITTC, 1978. Effect of blade roughness on propeller section lift and drag. In: ITTC, 1978. Proc.

59. ITTC, 1981. Section 3.2 Resistance increase due to roughness. In: ITTC, 1981. Proc.

60. ITTC, 1981. Section 4. Effects of surface roughness. In: ITTC, 1981. Proc.

61. Jones,R. & Williams,DH. The effect of surface roughness on the characteristics of the aerofoils NACA 0012 and RAF 34. ARC, R&M 1708, 1936.

62. Kans,S. et.al. Effect of fouling of a ship's hull and propeller upon propulsive performance. In: ISP 5 (41) 1958 Jan.

63. Karlsson,RI. A note on the roughness allowance W C_F. Chalmers Tekniska Hogskola, Nr. 80/9, June 1980.

64. Karlsson,RI. Studies of skin friction in turbulent boundary layers on smooth and rough walls. Parts I II III IV. Chalmers Tekniska Hogskola, Nr. 78, Jan 1978. Also: Summary of thesis 1980.

65. Karlsson,RI. The effect of irregular surface roughness on the frictional resistance of ships. In: Int. Symp. on Ship Viscous Resistance, Gothenburg. 1978.

66. Kempf,G. The effect of roughness on the resistance of ships. In: Engineering 143 1937.

67. King,MJ. The measurement of ship hull roughness. In: Conf. on Metrology & Properties of Engineering Surfaces, Leicester, 1982.

68. King,MJ. & Thomas,TR. Stylus measurement of the microgeometry of a coated surface. In: Journal Coatings Technology 50 (643) 1978 Aug.

69. Lackenby,H. Resistance of ships with special reference to skin friction and hull surface condition. In: Proc. I.Mech.E. 1961-62. Thomas Lowe Gray Lecture.

70. Lap,AJW. Frictional drag of smooth and rough ship forms. In: Trans RINA 98 1956.

71. Lerbs,HW. On the effects of scale and roughness on free running propellers. In: J.Amer.Soc.Naval Eng. 63 1951.

72. Lewkowicz,AK. & Musker,AL. The surface roughness and turbulent wall friction on ship hulls : interaction in the viscous sublayer. In: Int. Symp. on Ship Viscous Resistance, Gothenburg, 1978.

73. Lewkowicz,AK., Hoadley,D., Horlock,JH. & Perkins,HJ. A family of integral methods for predicting turbulent boundary layers. In: AIAA Jou. 8 (1) 1970.

74. Liu,CK. An experimental study of turbulent boundary layer on rough walls. Stanford Univ, Palo Atlo. Calif., Doctoral dissert. (Univ. Microfilms 67-4395) 1966.

75. Loeb,GI. Drag enhancement of microbial slime films on rotating discs. NRL, MR 4412, March 1981.

76. Loeb,GM. & Smith,N. Slime analysis of painted steel panels immersed in Biscayne Bay, Miami Beach, Florida. NRL, MR 4411, March 1981.

77. Malone,JA., Little,DE., & Allman,M. Effects of hull foulants and cleaning/coating practices on ship performance and economics. In: Trans SNAME 88 1980.

78. McNaught,., Parker,., Milne,A., Trotman,., & Pascoe,. BSRA Conf. Hull Surface Maintenance : its Role in Operating Economics. In: Motor Ship. 1978 April.

79. Milne,A. Self-polishing coatings in marine anti-fouling paints. In: R.I.C. Annual Chemical Congress, Durham, April 1980.

80. Mobius,H. Experimental investigations of the resistance and the velocity distribution in pipes with regularly arranged roughness and with turbulent flow (in German). In: Phys.Z. 41 1940 pp202-225.

81. Moody,LF. Friction factors for pipe flows. In: Trans ASME. 66 1944 p671.

82. Musker,AJ. Tabulate experimental data for flow in pipes with irregularly rough surfaces. Supplement to PhD thesis. Liverpool Univ., Ph.D. thesis supplement.

83. Musker,AJ. Turbulent shear flow near irregularly rough surfaces with particular reference to ships' hulls. Liverpool Univ., Ph.D. Thesis.

84. Musker,AJ. & Sarabachi,K. Wall friction and profileometry aspects of coating on irregularly rough surface. In: ISP 27 1980 Feb.

85. Musker,AL. & Lewkowicz,AK. The effect of ship hull roughness on the development of turbulent boundary layers. In: Int. Symp. on Ship Viscous Resistance, Gothenburg 1978.

86. Nayak,PR. Random process model of roughness surfaces. In: Trans ASME 83F Jou.Lub.Tech. 1971.

87. Nikuradse,J. Laws of flow in rough pipes. NACA, TM 1292, 1933. (Also in: Trans VDI Fors. 361)

88. Norwegian Skiptecknisk Forstrings-Institut Roughness grades for ships. NSFI.

89. Patience,G. The contribution of the propeller to energy conservation in ship operation. In: Trans IMarE (C) 94 1982.

90. Perry,AE. & Joubert,PN. Rough-wall boundary layers in adverse pressure gradients. In: J. Fl. Mech. 17 1963 pp193-211.

91. Perry,AE., Schofield,WH., & Joubert,PN. Rough wall turbulent boundary layers. In: J. Fl. Mech. 37 1969 pp383-413.

92. Peterson,EW. Relative importance of terms in the turbulent energy and momentum equations as applied to the problem of surface roughness change. In: J.Atmo.Sci. 29 1972 p1470.

93. Rough surfaces. Thomas,TR. (ed). Longmans, 1982

94. Roy,JF. Principe d'exploitation directe des essais de cylindre tourant and precisions quantitatives sur l'explortation des essais du cylindre tournant. Bassin d'Essais de Carenes Paris, Rep. 3584/3834, 1974.

95. Rubert,MI. Confusion in measuring surface roughness. In: ASNE Journal. 1960 May.

96. SNAME, Panel 0-23. Effect of bottom maintenance on frictional resistance of ships. SNAME, TRR 18, 1975.

97. Sasajima,H. & Himeno,Y. Scale correction for roughness effect. In: J. Soc. N.A. Japan 118 1965.

98. Sasajima,Y. Frictional resistance of wave roughened surfaces. In: Int.Shipbldg.Prog. 2 (13) 1955.

99. Sayles,RS. & Thomas,TR. Surface topography as a non-stationary random process. In: Nature 271 (5644) 1978 Feb.

100. Sayre,CL. & Duerr,RJ. Boundary layer investigation on USS Timmerman. DTMB, Rep. 1170, 1960.

101. Schlichting,H. Boundary layer theory. 6th ed. McGraw-Hill, 1968

102. Schlicting,H. Experimental investigations of the roughness problem. (in German). In: Ing.Arch. 7 1936 pp1-34. (see also Proc.ASME, 1936 and NACA, TM-823, 1936.

103. Sheng-Tien Hsu., & Kennedy,JF. Turbulent flow in wavy pipes. In: J. Fl. Mech. 47 1972 pp481-502.

104. Sillerud,BO. The effect of speed loss in service on ship freight income and fuel economy. In: Norwegian Maritime Research 2 1980.

105. Simpson,RL. A generalised correlation of roughness density effects on the turbulent boundary layer. In: AIAA J. 11 (2) 1973 pp242-244.

106. Stanier,MJ. Fuel savings through weld bead removal. In press:Naval Architect 1983.

107. Streeter,VL. Frictional resistance in artificially roughened pipes. In: Proc.ASCE. 61 1935 pp163.

108. Svensen,TE. Monitoring of speed and power performance in-service. In: Ship speed and economics, short course, Univ. Newcastle upon Tyne, 1983. Lecture 9.

109. Svensen,TE. Techno-economic reasons for selecting fuel-saving priorities. In: TransIMarE. C94 pprC95

110. Svensen,TE. & Medhurst,JS. The hydrodyamic and economic penalties of propeller roughness. In press:Marine Technology 1983.

111. Tack,EO. & Von Kerozek,C. Streamlines and pressure distribution on arbitarary ship hulls at zero Froude number. In: JSR 1968 Sept.

112. Telfer,EV. Further ship resistance similarily. In: Trans RINA 93 1951

113. Telfer,EV. On taking the rough with the smooth. In: Trans IESS. 1969 March.

114. Telfer,IE. Some aspects of the external maintenance of tankers. In: Trans RINA 114 1972.

115. Theodorsen,T. & Regier,A. Experiments on drag of thodring discs, cylinders and streamline rods at high speeds. NACA, Rep. 793, 1944.

116. Todd,FH. Skin friction resistance and the effects of surface roughness. In: Trans SNAME 59 1951.

117. Tokunaga,K. Local roughness effect on self-propulsion factors. In: Trans W. Japan Soc. NA. 63 1982.

118. Tokunaga,K. & Baba,E. A study of local roughness effect on ship resistance. In: Trans W. Japan Soc. NA. 59 1980.

119. Tokunaga,K. & Baba,E. Approximate calculation of ship frictional resistance increase due to surface roughness. In: J. Soc. NA. Japan 152 1982 Nov.

120. Townes,HW., Gow,JL., Powe,RE., & Weber,N. Turbulent flow in smooth and rough pipes. In: Trans ASME, J.Basic Engrg. 94 1972 pp353-362.

121. Townsend,AA. Flow in a deep turbulent boundary layer over a surface distorted by water waves. In: J. Fl. Mech. 55 1972 pp 719-735.

122. Townsin,RL. Economic consequences of fouling and roughness. Oyez. In: One Day Seminar on Marine Fouling, April 1979, London.

123. Townsin,RL. The speed loss budget. UNUT, DNAS, 1978. In: WEGEMT'78 Module H3 Hydodynamic Design.

124. Townsin,RL. The speed of our ships at sea. RINA/IMarE Western Joint Branch, paper read, April 1978.

125. Townsin,RL. & Robinson,CA. The monitoring of ship performance. GCBS, Tech Rpt J-10-04-042, 1975.

126. Townsin,RL. & Svensen,TE. Monitoring speed and power for fuel economy. In: Symp. Shipboard Energy Conservation '80, New York, Sept 1980, SNAME.

127. Townsin,RL., Byrne,D., Milne,A., Svensen,T. Speed, power and roughness : the economics of outer bottom maintenance. In: Trans RINA. 123 1981.

128. Townsin,RL., Moss,B., Wynne,JB., Whyte, I.M. Monitoring the speed performance of ships. In: NECIESTrans. 91 1975 pp159-178,D37-D43.

129. Townsin,RL., Wynne,JB., Milne,A., Hails,G. Hull condition : penalties and palliatives for poor performance. In: International congress on marine corrosion and fouling, 4th, Juan-Les Pins, Antibes, 14-18 June 1976. pp(3-42)-(3-71).

130. Townsin,RL., Byrne,D., Svensen,TE. & Milne,A. Estimating the technical and economic penalties of hull and propeller roughness. In: TransSNAME. 89 1981

131. Townsin,RL., Medhurst,JS., Hamlin,NA. & Sedat,R. Calculating the resistance of ships with distributed roughness. In: Proc. Centenary Conference NEC 100. NECIES May 1984.

132. Uberoi,SBS. Viscous resistance of ships and ship models. Tech. Univ. Denmark, Hydro og Aerodynamik Laboratorium Rep. Hy.13, 1969.

133. Vincenti,WG. Non equilibrium flow over a wavy wall. In: J. Fl. Mechs. 6 1959.

134. Ward,G. Ship speed and power performance In: Fitness for Sea - Seaworthiness, Newcastle upon Tyne, 10th Sept 1980. Proc.

135. Watanabe,S., Nagamatsu,N., Yokoo,K., & Kawakami,Y. The augmentation in frictional resistance due to slime. In: Journal Kansai Soc.NA. 131 1969 (English language translation).

136. Watson,DGM. Designing ships for fuel economy.. In: Trans RINA 123 1981. Also in: Naval Architect 11 (6) 1981 Nov

137. Wellman,F. Survey of hull and propeller roughness data. In: Shipbldr. & Mar.Eng. Bldr. 70 1963.

138. West,E. The effect of surface preparation and painting procedures on the resistance of old ship bottom plates as predicted from NSRDC friction plane model 4125. NSRDC, Rep. 4084, May 1973.

139. West,J. Protection of ships hulls. RINA, MTM 2, 1976.

140. Yolles,MI. et al. Walsh moment analysis of engineering surfaces. In: Conf. on Metrology and Properties of Engineering Surfaces, Leicester, 1982.

Dr.R.L.Townsin, WEGEMT Bibliography : MARINE ROUGHNESS AND DRAG

Addendum August 1983

141. Cumptsey, N.A. & Head, M.R.: The calculation of 3-dimensional turbulent boundary layers - Pt.I : flow over the rear of an infinite swept wing. Aero.Q.Vol.18 1967, pp.55-84.

142. Svensen, T.E.: The economics of hull and propeller maintenance examined in the face of uncertainty. Trans.NECIES Vol.100, Oct.1983.

143. Byrne, D.: Hull roughness and the impact of outer hull maintenance decisions on ship efficiency. Athens 1983.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

RESISTANCE INCREASE DUE TO WIND, WAVES AND STEERING

by

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RESISTANCE INCREASE DUE TO WIND, WAVES AND STEERING

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1. Introduction

Ship performance in service condition is affected by many factors, some of which can be controlled as hull form, bottom condition etc, but some cannot be directly controlled, as the action of wind and waves. The added resistance due to the direct and indirect action of wind and waves can be divided as follows

a) wind resistance of the ship's above-water parts
b) resistance caused by wind induced waves
c) rudder resistance
d) yawing resistance

Components a) and b) are directly a consequence of wind and wave action, components c) and d) are indirectly dependent. This division also provides a rational approach for calculating the added resistance and the resulting speed loss.

There are several reasons why fairly accurate estimates are needed for the added resistance due to wind, waves and steering. The most important ones are:

- to make appropriate allowances for weather conditions causing speed loss or power increase, for example for a ship sailing according to a time-table.
- to correct results from speed trials which are not made in calm weather conditions.

In addition, it is also of interest to compare the involuntary speed loss (due to wind and waves) with the speed loss due to other factors as fouling etc.

The approach to the added resistance due to wind, waves and steering will be made as follows.

- Forces on the ship's hull, the equations of motions in the horizontal plane, relevant terms in the equations.
- Wind coefficients for ships, estimates for such coefficients, calculation methods, relative and true wind.
- Wave induced resistance.
- Yawing resistance, why it is obtained, and some useful estimates.
- Drift resistance.
- Rudder resistance.
- The total added resistance, comparison of the components of added resistance, relative importance of these components.
- Methods for correcting the speed for added resistance.

2. Forces on the Ship's Hull, Equations of Motions

The forces on the ship can be described by the equations of motions (see (1)). For a complete description six equations are needed. For the estimates of a mean value of the added resistance, simplified expressions for the equations of motions can be used. These include the three equations describing the forces and moments in the horizontal plane.

Axial force:

$$X_{uu}u^2 + (1 - t)T + X_{vv}v^2 + X_{cc\delta\delta}c^2\delta^2 + X_{WIND} + X_{WAVES} + X_{YAWING} = 0 \quad (1)$$

Resistance in calm water	Propulsive force	Induced resistance due to drift	Induced resistance due to rudder	Wind resistance of above-water parts of ship	Induced wave resistance	Induced yawing resistance

Lateral force:

$$Y_{uu}uv + Y_{cc\delta}c^2\delta + k_{Ty}T + Y_{WIND} + Y_{WAVES} = 0 \quad (2)$$

Lateral force due to drift	Lateral force due to rudder	Unbalance force due to prop	Lateral wind force on above-water parts of ship	Lateral force due to waves

Yawing moment:

$$N_{uv}uv + N_{cc\delta}c^2\delta + k_{TN}T + N_{WIND} + N_{WAVES} = 0 \qquad (3)$$

Yawing moment due to drift	Yawing moment due to rudder	Unbalance yawing moment due to prop	Yawing moment of wind on above-water parts of ship	Yawing moment due to waves

where: u ship's velocity in axial direction

v ship's velocity in lateral direction

c water velocity at rudder

δ rudder angle

t thrust deduction factor

T propeller thrust

X_{uu}, X_{vv} etc. hydrodynamic coefficients depending on geometry, size etc.

The equations are given for quasi-steady conditions, and it can be noted that the wave resistance X_{WAVES} includes the pitching-heaving resistance component, and that the X_{YAWING} is the yawing-drifting resistance component.

3. Wind Coefficients

Direct wind forces on the above-water structure of a ship are usually obtained from wind tunnel tests. Several such investigations have been published and a selection is given in references (2) to (7). References (8) and (9) give results from towing the above-water parts of the ship upside-down in a towing basin. Isherwood (10) has published a calculation method for wind forces based on results from several wind tunnel tests with different types of ships.

In order to obtain more systematic data for wind forces, investigations in a wind tunnel were initiated by SSPA and included tankers, container and ro/ro ships, and small craft. The aims of these tests can be summarized as follows

(a) to obtain data for modern ship forms

(b) to obtain estimates for changes in deckhouse configurations as regards size and position

(c) to obtain some information on scale effects, detailed representation of the superstructure, wind boundary layer above the sea.

Results from these investigations have been summarized in (11).

Some differences in the wind conditions for the ship in the open sea and the model tested in a wind tunnel can be observed.

The wind speed experienced by the ship (relative wind V_R) will be the vector sum of true wind speed V_T and the ship speed V. In Fig 1 diagrams of the relation between relative and true wind are given. As the true wind speed will vary with the height above the sea level, the relative wind will also vary both as regards magnitude and direction. The velocity profile for the relative wind will thus deviate from a true boundary layer velocity profile and also be twisted. The deviation and twist will be largest for small values of true wind speed to ship speed. This is illustrated in Fig 2.

The results from the wind tunnel tests are also affected by the actual boundary layer in the tunnel. The problem of correcting for the boundary layer is discussed in (7) and the conclusion reached was that if a thickened boundary layer is used in the wind tunnel, the best resemblance with the natural wind boundary layer will be obtained. The wind coefficients from the SSPA investigations are thus based on the free stream velocity in the tunnel and are not corrected for the influence of the boundary layer. Different methods of correcting for the boundary layer can be used (see (7)) but in general the resulting differences in wind coefficients are in the order of 10 per cent or less. This should be borne in mind when wind coefficients from different sources are compared.

The presentation of wind coefficients causes also some other problems of methodology, especially when data from different sources are compared.

In general wind coefficients are given as two force coefficients, C_X and C_Y (longitudinal and lateral force) and one moment coefficient C_N (yaw moment). These coefficients are usually obtained by non-dimensionalising with relative wind speed V_R squared and transverse projected area A_T for C_X and lateral projected area A_L for C_Y and C_N.

The wind coefficients are then presented as functions of relative wind direction γ. This method is not suitable if results from different above-water configurations and different values of projected areas A_T and A_L are to be compared.

As was proposed in (7) it seems more rational to use the ship length L squared as the reference area, and to present the wind coefficients as functions of the ratios A_T/L^2 and A_L/L^2 (for C_X respectively C_Y and C_N) and with the relative wind direction γ as a parameter. Using this method comparisons between different investigations are facilitated.

This idea for the presentation of wind coefficients was further developed in (12) where the concept of principal shapes of wind coefficient curves was introduced. The wind coefficients as functions of relative wind direction are given, as illustrated in Fig. 3, as straight lines between a number of angles γ. For these angles of γ values of wind coefficients are given as functions of A_T/L^2 of A_L/L^2.

In Figs. 4-6 levels of wind coefficients are given for tankers, container and ro/ro ships, and small craft.

In conclusion the investigations of wind forces and moments show that for modern forms of tanker, container and ro/ro ships with deckhouses aft the longitudinal force coefficient C_X are well correlated by the ratio front area (A_T) to the ship's length (L) squared, and the transverse force coefficients C_Y and yaw moment coefficients C_N are correlated by the ratio lateral area (A_L) to the ship's length squared. Modifications in deckhouse shape etc. result only in minor differences in the wind coefficients.

The small craft investigations show some differences from the tanker and container ship investigations, which may be explained by differences in deckhouse position and different proportions of the main dimensions (deckhouse, hull length, width and height).

4. Wave Induced Resistance

The effect of waves on the added resistance is discussed in (13) and (14). In the approach outlined here the wave induced forces X_{WAVES} and Y_{WAVES} as well as the yawing moment N_{WAVES} are needed. X_{WAVES}, Y_{WAVES} and N_{WAVES} can be calculated or obtained from model tests. However, calculation methods are rather elaborate and model tests are in general not carried out to give the wave induced forces and moment, but rather the motions of the model.

Data on wave induced forces are rather scarce, but in order to give some estimates of the added resistance X_{WAVES} data from (14) are shown in Fig. 7, together with data from the discussion of (11).

For direct estimates of the speed loss due to wind and waves (13) is useful, which gives the speed loss as a function of Beaufort number (BN) and where also the influence of the ship's heading relative to the wind may be included.

The formula for speed loss given in (13) is

$$\frac{\Delta V}{V} = a \cdot BN + \frac{BN^{6 \cdot 5}}{b \cdot \nabla^{2/3}}$$

where BN = Beaufort number

∇ = volume of displacement (in m^3)

a,b = coefficients

The following values are given for a and b

	a	b
Tankers laden	0.5	2.7
" ballast	0.7	2.7
Container ships	0.7	22

Wind direction (relative to the ship) is introduced by a reduction factor μ as follows

$$2\mu_{bow} = 1.7 - 0.03(BN - 4)^2 \qquad (30^o - 60^o)$$
$$2\mu_{beam} = 0.9 - 0.06(BN - 6)^2 \qquad (60^o - 150^o)$$
$$2\mu_{following} = 0.4 - 0.03(BN - 8)^2 \qquad (150^o - 180^o)$$

5. Yawing Resistance

The effects of wind and waves are not steady and will thus cause the ship to perform a yawing motion. The combination of yawing and drifting motions will give an added resistance as may be seen from the complete equations of motion which include a vr term in the X-equation. v is the drift speed and r the yaw velocity.

An expression for the added resistance due to yawing motions may be obtained from (15) and using some estimates for the hydrodynamic coefficients and the frequency of the yawing motion, so that

$$X_{YAWING} = 1.2 \cdot \frac{V^2}{L} \cdot \nabla \cdot \rho \cdot (\psi_a)^2$$

where ∇ = displacement of the ship (m^3)
ψ_a = amplitude of the yawing motion

6. Drift Resistance

The induced drift resistance is the term $X_{vv} \cdot v^2$, which can be estimated as follows:

For a low aspect ratio wing in ideal flow the induced drag in the wind direction is $C_{Di} = C_L \cdot \beta/2$ (C_{Di} induced drag coefficient, C_L lift coefficient, β angle of incidence), which means that in the wing plane direction a forward acting force of $C_L \cdot \beta/2$ is obtained. The under-water hull of a ship may crudely be described as a low aspect ratio wing, which means that a positive value of X_{vv} may be expected from theoretical considerations.

In general X_{vv} is evaluated from model tests at small drift angles and X_{vv} is obtained as the difference between two large numbers,

i.e. the total resistance at a drift angle and the total resistance at the drift angle zero. It must thus be recognized that results for X_{vv} from model tests should be analysed critically. In Fig. 8 data for the added resistance due to side-slip $\Delta X(v^2)$ have been compiled from a number of tests including tankers, conventional freighters and container ships. It is evident that there is much scatter in the data. For a few cases even positive values of $\Delta X(v^2)$ have been found. The mean curve (drawn in Fig. 8) has a slope $X'_{\beta\beta} \simeq -0.200$ (β in radians $\beta \simeq v/V$). Using this value the following expression is obtained for $X_{vv} \cdot v^2$ ($= X_{DRIFT}$)

$$X_{vv} \cdot v^2 = -0.2 \cdot \tfrac{1}{2}\rho \cdot L \cdot T \cdot 10^{-3} \cdot v^2$$

$$\simeq 0.1 \cdot \rho LT \cdot v^2 \cdot 10^{-3}$$

(T = the draught of the ship)

7. Rudder Resistance

The rudder resistance term $X_{cc\delta\delta} \cdot \delta^2$ ($= X_{RUDDER}$) may be expressed according to Norrbin (12) as follows

$$X_{cc\delta\delta} \cdot c^2 \cdot \delta^2 = f(AR) \cdot \tfrac{1}{2}\rho c^2 \cdot A_R \cdot \delta^2$$

where $f(AR)$ is a function of the rudder aspect ratio AR
c flow velocity at the rudder
A_R rudder area

For normal rudders with aspect ratios about 1.5 to 3 the value of the function $f(AR)$ will be about 1.

The flow velocity (c) at the rudder can be calculated using the momentum theory and the propeller characteristics (see (16)). This velocity will vary over the height of the rudder and will have a higher value inside the propeller race. For most ships a reasonable estimate is to put $c = V$, that is equal to ths ship's forward speed.

The induced drag due to the rudder is thus

$$X_{cc\delta\delta}c^2\delta^2 = \tfrac{1}{2}\rho V^2 \cdot A_R\delta^2$$

This estimate is compared in Fig. 9 with a number of data from model tests and the agreement is rather good and might be improved if the effect of flow velocity (c) is included.

For the calculations of induced resistance rudder angles (δ) and drift angles (β) are needed. These angles can be obtained from solving the equations of motion ((1)-(3)).

The following expressions are obtained after some rearranging and neglecting the influence of the propeller unbalance force:

$$\beta = \frac{(Y_{WIND} + Y_{WAVES})\,l_d - (N_{WIND} + N_{WAVES})}{Y_{uv}\,(l_d - l_\beta)} \cdot \frac{1}{V^2}$$

$$\delta = \frac{(N_{WIND} + N_{WAVES}) - (Y_{WIND} + Y_{WAVES})\,l_\beta}{Y_{cc\delta}\,(l_d - l_\beta)} \quad \frac{1}{V^2}$$

where $l_\delta = N_{cc\delta}/Y_{cc\delta}$

$l_\beta = N_{uv}/Y_{uv}$

If non-dimensional quantities are used the following expressions for β and δ are obtained:

$$\beta = \frac{C_{Yww} \cdot \xi_\delta - C_{Nww}}{Y''_{uv} \cdot (\xi_\delta - \xi_\beta)} \cdot \frac{L^3}{2\nabla} \cdot \frac{\rho_a}{\rho} \cdot \left(\frac{V_R}{V}\right)^2$$

$$\delta = \frac{C_{Nww} - \xi_\delta - C_{Yww}}{Y''_{cc\delta} \cdot (\xi_\delta - \xi_\beta)} \cdot \frac{L^3}{2} \cdot \frac{\rho_a}{\rho} \cdot \left(\frac{V_R}{V}\right)^2$$

C_{Yww} and C_{Nww} are the combined wind + wave lateral force and yawing moment coefficients.

Y''_{uv} = non-dimensional static force coefficient
$Y''_{cc\delta}$ = non-dimensional rudder force coefficient
ξ_δ = $N''_{cc\delta}/Y''_{cc\delta}$
ξ_β = N''_{uv}/Y''_{uv}
$N''_{cc\delta}$ = non-dimensional rudder yawing moment
N''_{uv} = non-dimensional static yawing moment coefficient

The following estimates are valid for merchant ships

$-1.1 > Y''_{uv} > -1.5$

$\xi_\beta \simeq 0.35$

$0.20 > Y''_{cc\delta} > 0.30$

$-0.4 > \xi_\delta > -0.5$

8. The Total Added Resistance

In the preceding chapter the different components of added resistance have been discussed and estimates are given.

Summarizing the components of added resistance:

$$X_{WIND} = C_X \cdot \tfrac{1}{2}\rho_a V_R^2 \cdot L^2$$

$$C_X = C_X\left(\frac{A_T}{L^2};\ \gamma\right) \qquad \text{(Figs. 4-6)}$$

$$X_{WAVES} = C_{XWAVE} \cdot \tfrac{1}{2}\rho V^2 \cdot \nabla^{2/3}$$

$$C_{XWAVE} = C_{XWAVE}(BN) \qquad \text{(Fig. 7)}$$

(Speed loss due to wind and waves can also be directly obtained from (13).)

$$X_{YAWING} = 1.2 \cdot \frac{V^2}{L} \cdot \nabla \cdot \rho\ (\psi_a)^2$$

$$X_{DRIFT} = 0.1\rho \cdot L \cdot T \cdot v^2 \cdot 10^{-3}$$

v from solving equations (2) and (3)

$$X_{RUDDER} = \tfrac{1}{2}\rho V^2 \cdot A_R \cdot \delta^2$$

δ from solving equations (2) and (3)

In order to solve (2) and (3) estimates for wind and wave induced lateral forces and yawing moments are needed. From Figs (4) - (6) the wind induced lateral force and yawing moment can be obtained.

Two examples will be given for the added resistance components, one for a tanker and one for a container ship:

		Tanker	Container
Length	L	254 m	190 m
Displacement	∇	156,000 m^3	28,000 m^3
Speed	V	8 m/s	10 m/s
True wind speed	V_T	10 m/s	10 m/s
" " direction (relative to the ship)	γ_T	45°	45°
Relative wind speed	V_R	16 m/s	18 m/s
" " direction	γ	35°	30°
Front area ratio	A_T/L^2	0.010	0.016
Lateral area ratio	A_L/L^2	0.035	0.075
Wind coefficient	C_X	-0.006	-0.009
" "	C_Y	-0.012	-0.032
" "	C_N	0	-0.002
Wind resistance	X_{WIND}	0.0609 MN	0.0647 MN
Wave induced resistance	X_{WAVES}	0.105 MN	0.021 MN
Drift angle	β	0.1^o	0.4^o
Rudder angle	δ	0.4^o	1.6^o
Drift resistance	X_{DRIFT}	0.000 MN	0.000 MN
Rudder resistance	X_{RUDDER}	0.0001 MN	0.0015 MN
Yawing amplitude (assumed)	ψ_a	1°	1°
Yawing resistance	X_{YAWING}	0.0147 MN	0.0055 MN
Total added resistance	ΔR	0.181 MN	0.093 MN
Calm weather resistance	R_0	1.696 MN	0.740 MN
Relative added resistance	$\Delta R/R_0$	0.107 ~~MN~~	0.126 ~~MN~~

From the examples shown the importance of the induced wave resistance, especially for the tanker, can be observed.

9. Correcting the speed

The total added resistance can be written as

$$\Delta R = X_{WIND} + X_{DRIFT} + X_{RUDDER} + X_{YAWING} + X_{WAVES}$$

For the prediction of the performance of the ship it is of more interest to calculate the effects on the speed of the ship.

Two cases of speed loss will be considered; speed loss at constant thrust and power increase at constant speed.

For the constant thrust case, the speed loss is obtained from the expression

$$R(V_0) + \Delta R = R(V_0 + \Delta V)$$

Here $R(V_0)$ = calm wather resistance at the speed V_0
ΔR = added resistance due to wind
$R(V_0+\Delta V)$ = calm weather resistance at the speed $V_0+\Delta V$
ΔV = speed loss

If the calm weather resistance can be approximated by a V^2-dependency the speed loss ΔV will be

$$\frac{\Delta V}{V_0} = \sqrt{1 + \frac{\Delta R}{R_0}} - 1$$

where ΔR = added resistance
R_0 = calm weather resistance at the speed V

The second case - power increase at constant speed - is perhaps of more interest, as this is the case when the speed trials are analysed.

The required power P_0 at the speed V_0 in calm weather can be written as

$$P_0 = \frac{T_0 \cdot V_0}{(\eta_0)_0}$$

where η_0 = propeller efficiency

If the ship experiences an added resistance the required power $P_0 + \Delta P$ will be

$$P_0 + \Delta P = \frac{(T_0 + \Delta T)V_0}{(\eta_0)_0 + \Delta\eta_0}$$

Thus propeller thrust as well as propeller efficiency will differ from the calm weather condition. If thrust deduction as well as wake fraction are assumed independent of the propeller loading, the following expression for the power increase P is obtained

$$\frac{\Delta P}{P_0} = \frac{1 + \frac{\Delta R}{R_0}}{\left(1 + \frac{\Delta\eta_0}{(\eta_0)_0}\right)} - 1$$

The propeller efficiency depends thus on the propeller loading (or indirectly on the total resistance of the ship) and examples are shown in Fig 10.

For the examples given above the following values of speed loss and power increase are obtained.

		Tanker	Container
Relative added resistance	$\Delta R/R_0$	0.107	0.126
Speed loss at constant thrust	ΔV	0.42 m/s	0.61 m/s
Power increase at constant speed	$\Delta P/P_0$	14.5 %	15.4 %

10 References

1. NORRBIN, N H, "Theory and Observations on the Use of a Mathematical Model for Ship Manoeuvring in Deep and Confined Waters". SSPA Publ. No. 68, 1971

2. WAGNER, B, "Windkräfte an Überwasserschiffen". Jahrbuch der Schiffbautechnischen Gesellschaft 61, 1967

3. TSUJI, T et al, "Model tests of wind forces acting on ships". Ship Research Institute of Japan Reports, Vol. 7, 1970

4. AAGE, C; "Wind coefficients for nine models". Hydro- and Aerodynamics Laboratory Report No.A-3, 1971

5. SHEARER, K D A and LYNN, W M, "Wind tunnel tests on models of merchant ships". Trans. NECI, Vol. 76, 1960

6. AERTSSEN, G and COLIN, P E, "L'effet du vent sur trois types de navires en mettant l'accent sur le porte-conteneurs". Bulletin de l'Association Technique Maritime et Aéronautique No. 73, 1973

7. VAN BERLEKOM, W, TRÄGÅRDH, P and DELLHAG, A, "Large Tankers - Wind Coefficients and Speed Loss due to Wind and Sea". RINA Spring Meeting, 1974, Paper No. 9

8. BRUMMER, G M A and others, "Simulation of the steering and manoeuvring characteristics of a second generation container ship". Netherlands Ship Research Centre TNO, Report No. 170S, 1972

9. BEUKELMAN, W and BUITENHEK, M, "Full scale measurements and predicted seakeeping performance of the container-ship Atlantic Crown". International Shipbuilding Progress, Vol. 21, No. 243, Nov. 1974

10. ISHERWOOD, R M, "Wind resistance of merchant ships". Trans. RINA, Vol. 115, 1973

11. VAN BERLEKOM, W, "Wind Forces on Modern Ship Forms - Effects on Performance". Trans. NECI, Vol. 97, 1980-81

12. VAN BERLEKOM, W, "Wind resistance of surface ships". Appendix 8, Report of Performance Committee, 14th ITTC, 1975

13. TOWNSIN, R L and KWON, Y J, "Approximate Formulae for the Speed Loss Due to Added Resistance in Wind and Waves". RINA Preprint W S (1982)

14. NAKAMURA, S and FUJII, H, "Nominal Speed Loss of Ships in Waves". PRADS - International Symposium on Practical Design in Shipbuilding, Tokyo, October 1977

15. NORRBIN, N H, "On the added resistance due to steering on a straight course". Appendix 8, Report of Performance Committee, 13th ITTC, 1972

16. VAN BERLEKOM, W, "Effects of propeller loading on rudder efficiency". 4th Ship Control Systems Symposium, The Hague 1975

11. Nomenclature

A_L	lateral projected area (above-water structure)
A_T	front area (above-water structure)
A_R	rudder area
BN	Beaufort Number
c	water velocity at rudder
C_X, C_Y, C_N	wind coefficients
L	length of the ship
P	propeller power
R	resistance of the ship
ΔR	total added resistance
t	thrust deduction
T	propeller thrust
T	draught of the ship
u	ship speed in x-direction
v	ship speed in y-direction
V	ship speed
V_R	relative wind speed
V_T	true wind speed
x, y, z	co-ordinate axes
X_{uu}, X_{vv}..	hydrodynamic coefficients
β	drift angle
γ	direction of relative wind
γ_T	direction of true wind
δ	rudder angle
η_0	propeller efficiency
ρ	density of water
ρ_a	density of air
ψ_a	yaw amplitude
AR	rudder aspect ratio
∇	displacement of the ship

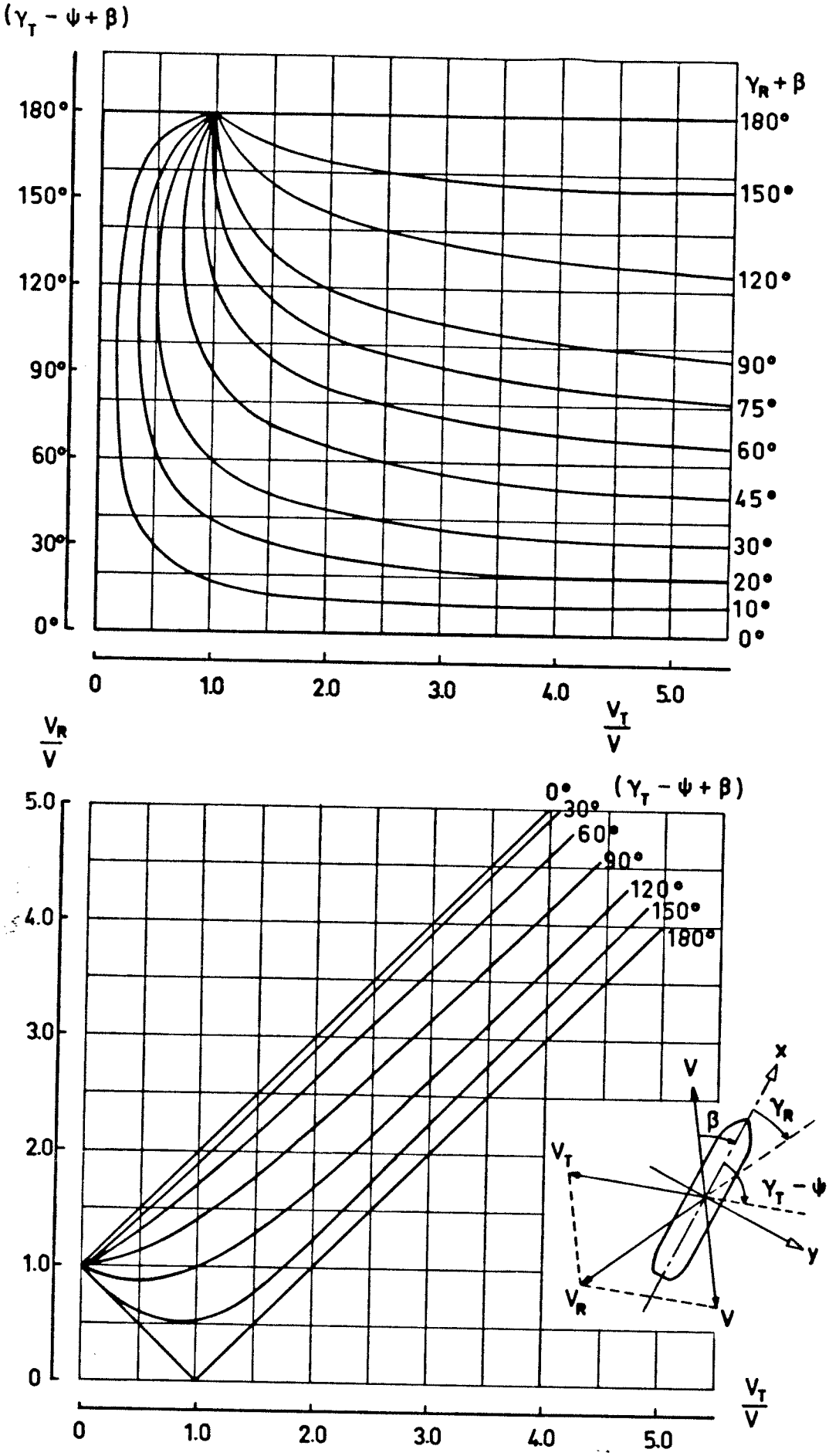


FIG 1 DIAGRAMS FOR CALCULATING RELATIVE WIND V_R

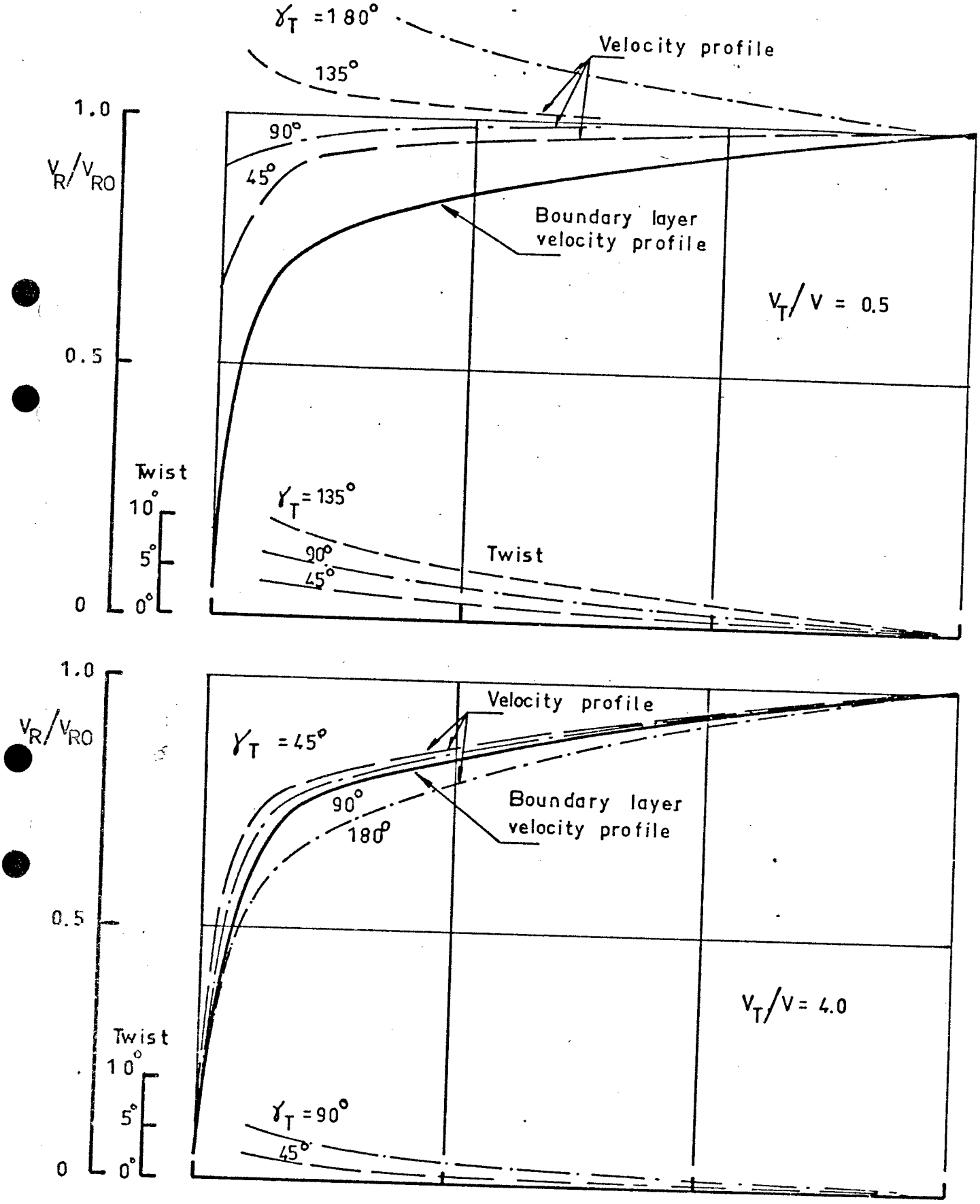


FIG 2 VELOCITY PROFILES AND TWIST FOR RELATIVE WIND

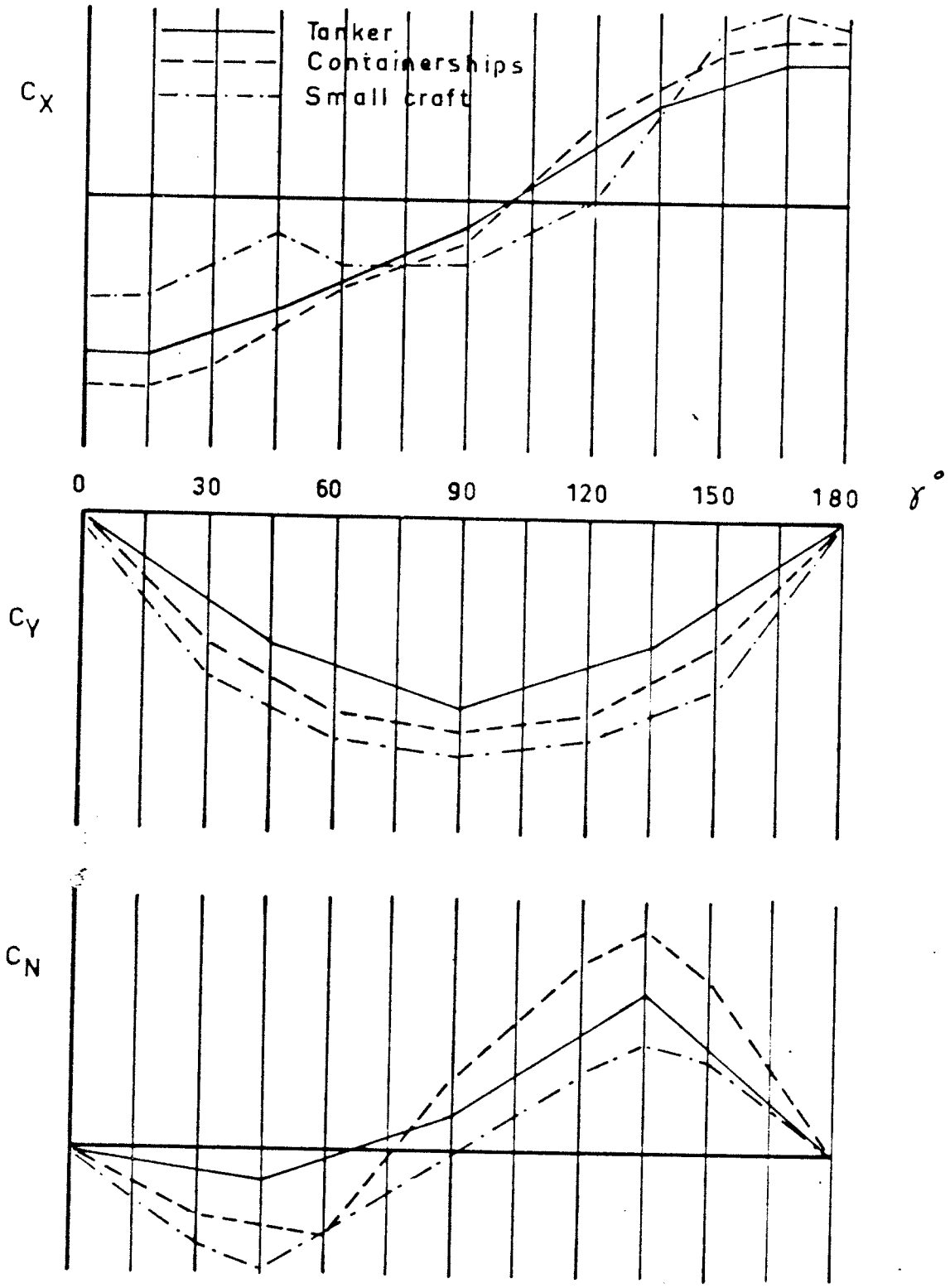
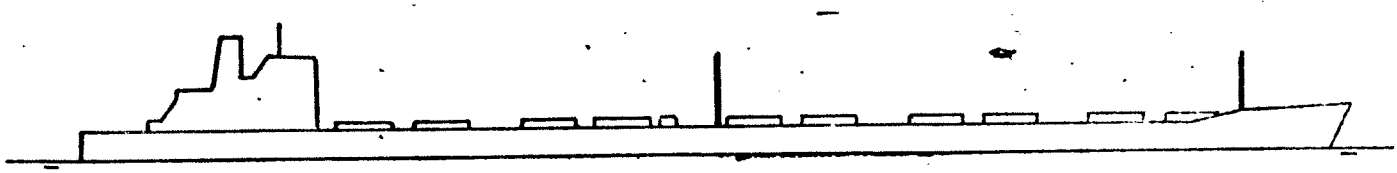


FIG 3 PRINCIPAL SHAPES OF WIND COEFFICIENT CURVES



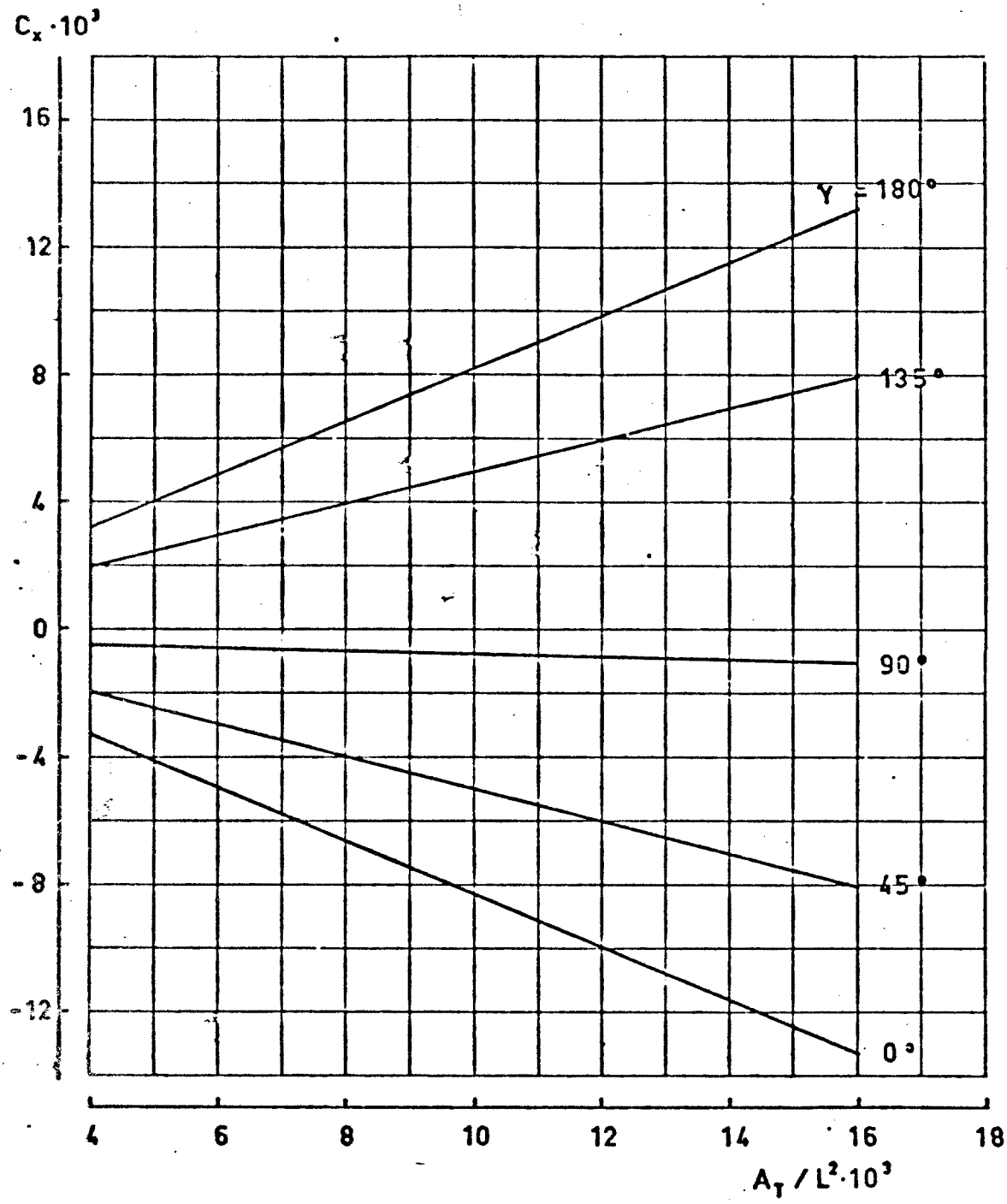


FIG 4 (1) LEVEL OF WIND COEFFICIENT C_X FOR TANKERS

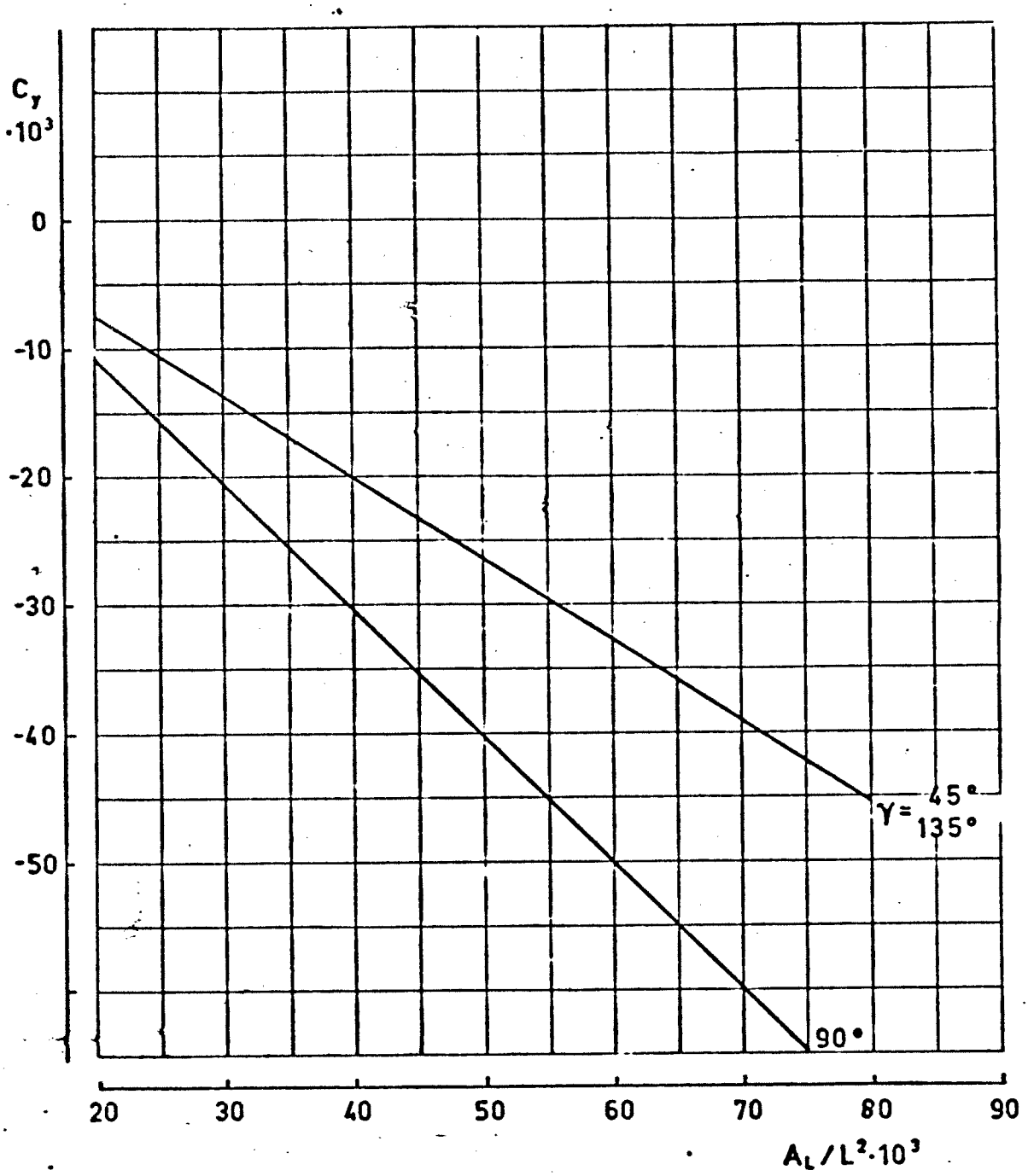


FIG 4 (2) LEVEL OF WIND COEFFICIENT C_Y FOR TANKERS

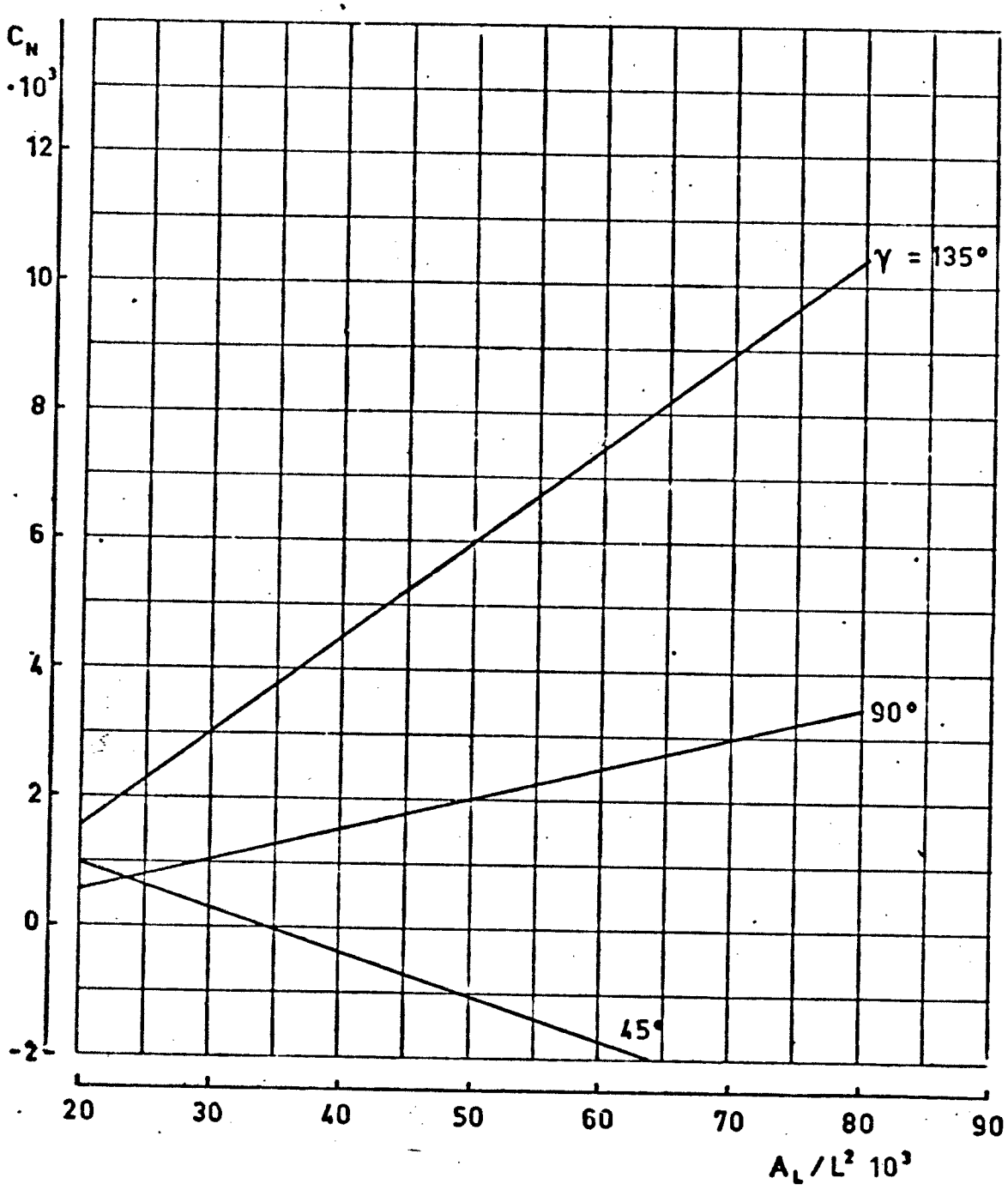
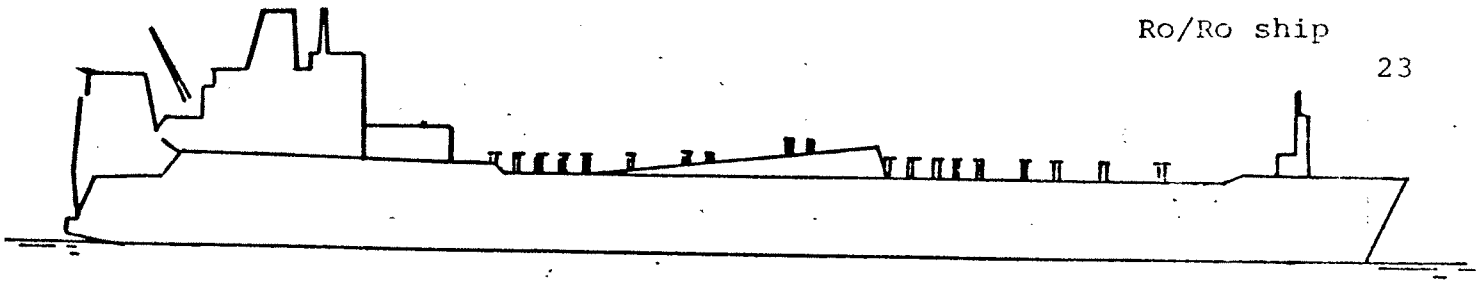
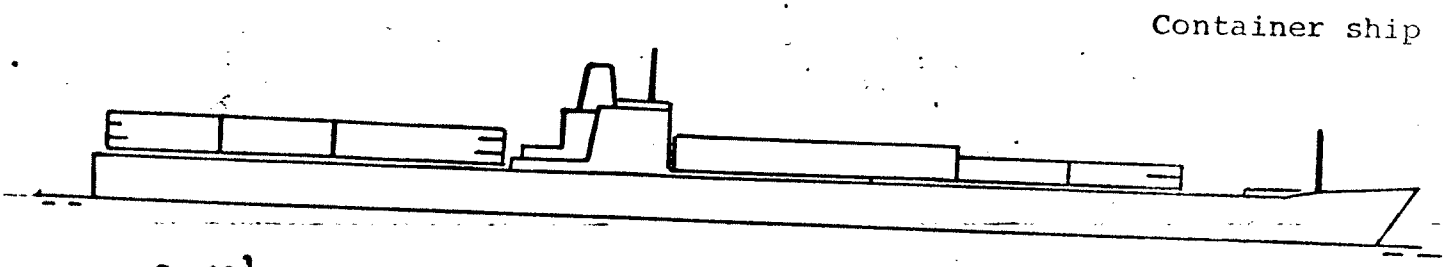


FIG 4 (3) LEVEL OF WIND COEFFICIENT C_N FOR TANKERS





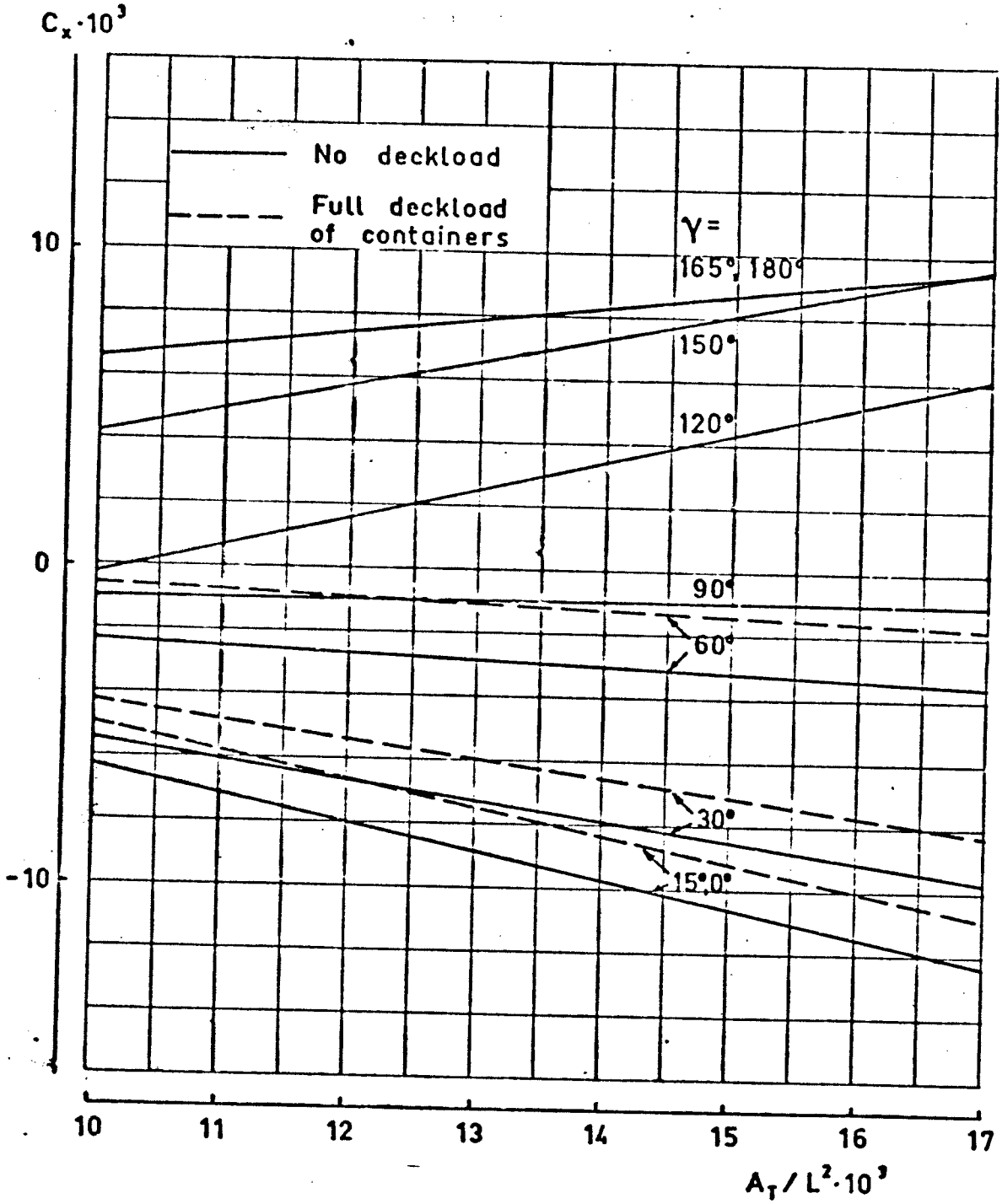


FIG 5 (1) LEVEL OF WIND COEFFICIENT C_X FOR CONTAINER SHIPS

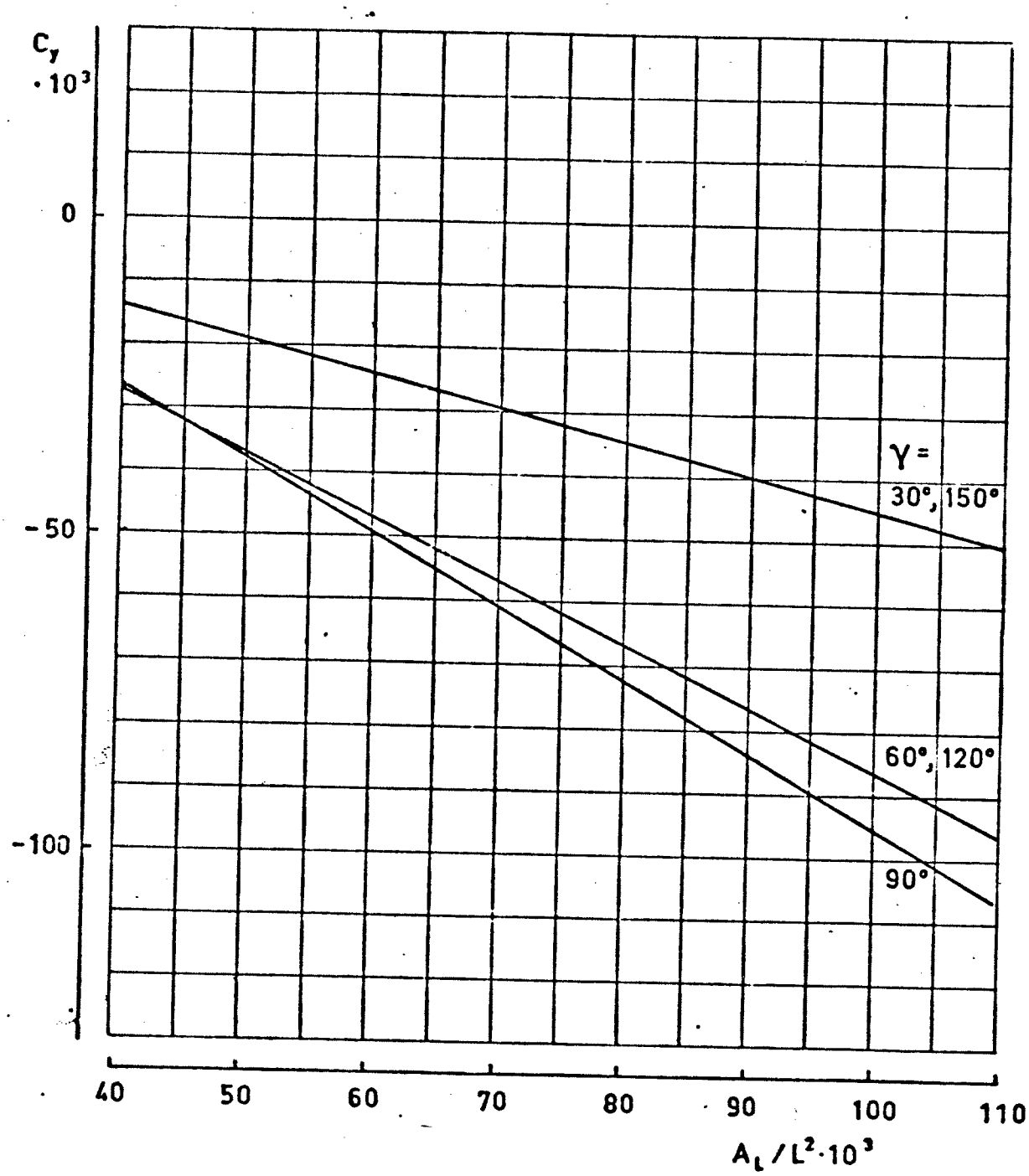


FIG 5 (2) LEVEL OF WIND COEFFICIENT C_γ FOR CONTAINER SHIPS

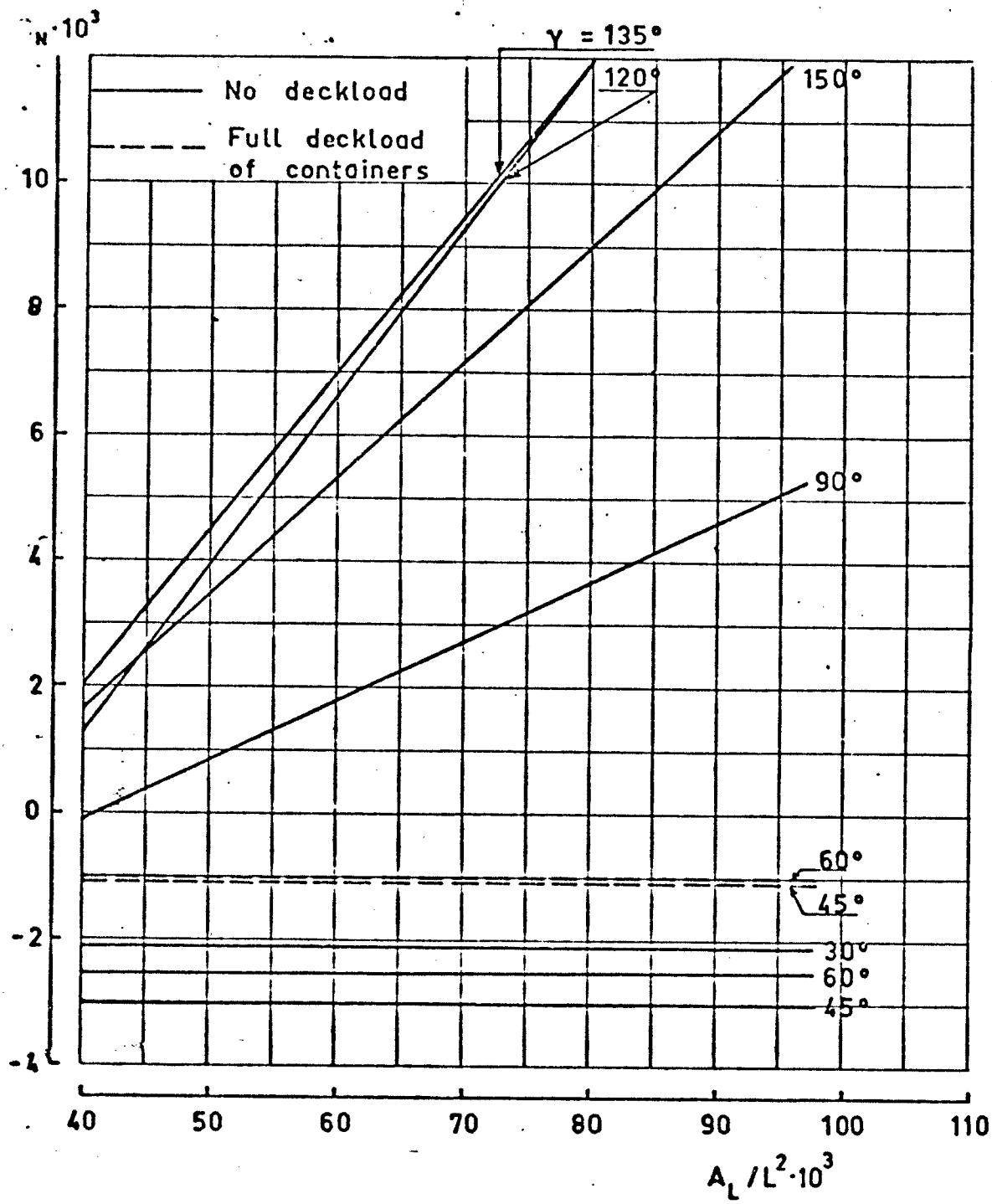


FIG 5 (3) LEVEL OF WIND COEFFICIENT C_N FOR CONTAINER SHIPS

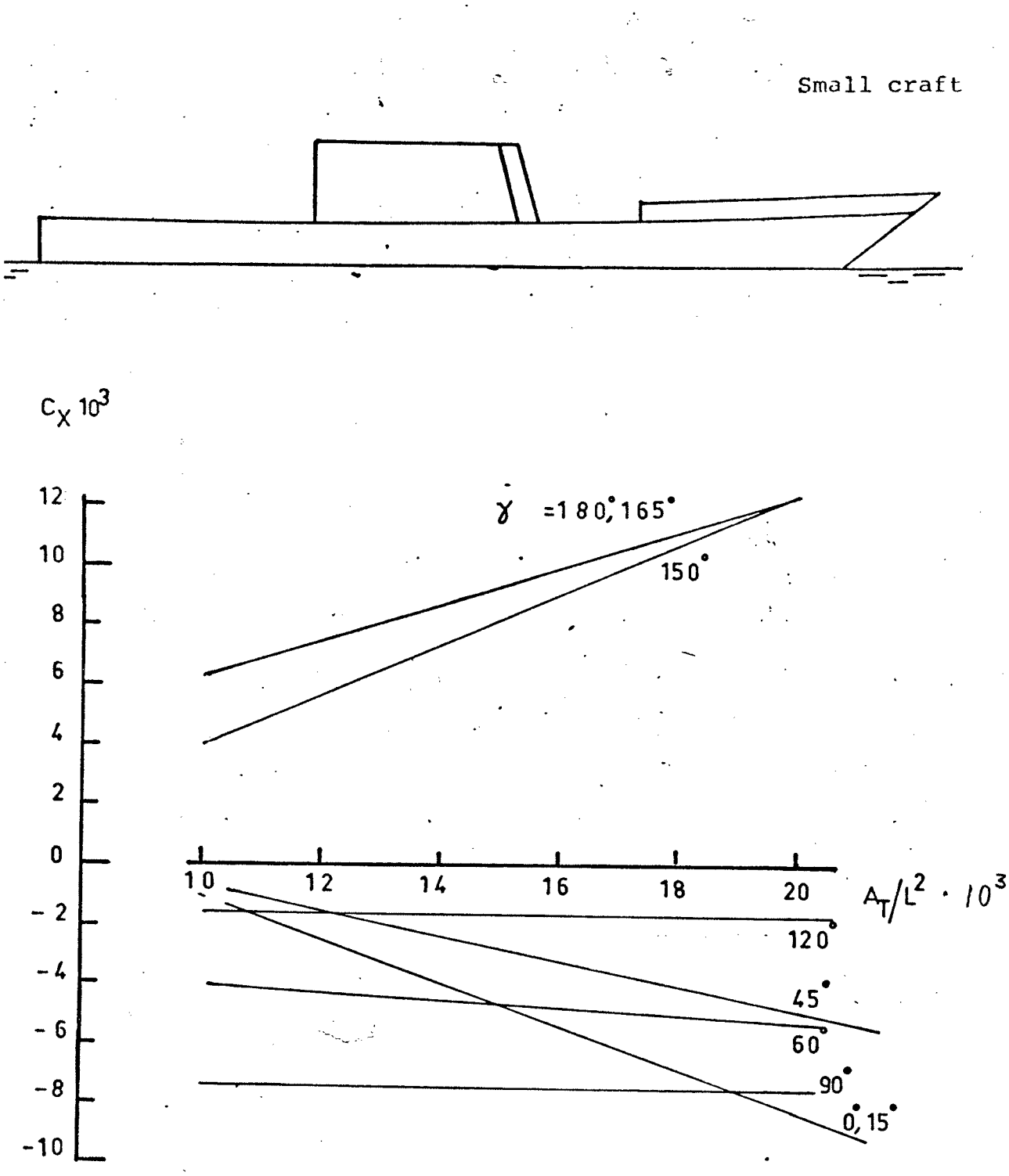


FIG 6 (1) LEVEL OF WIND COEFFICIENT C_X FOR SMALL CRAFT

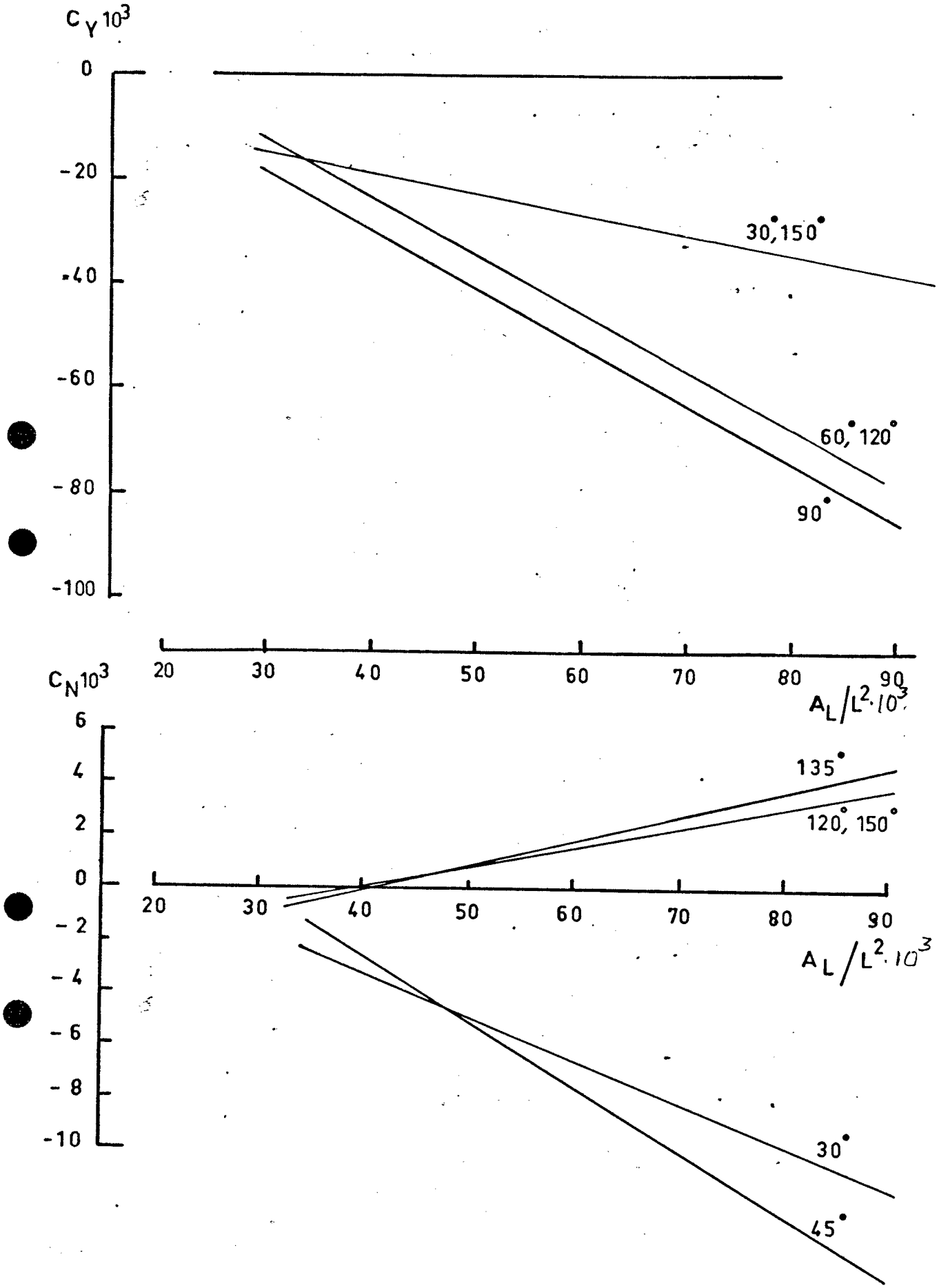


FIG 6 (2) LEVEL OF WIND COEFFICIENTS C_Y AND C_N
FOR SMALL CRAFT

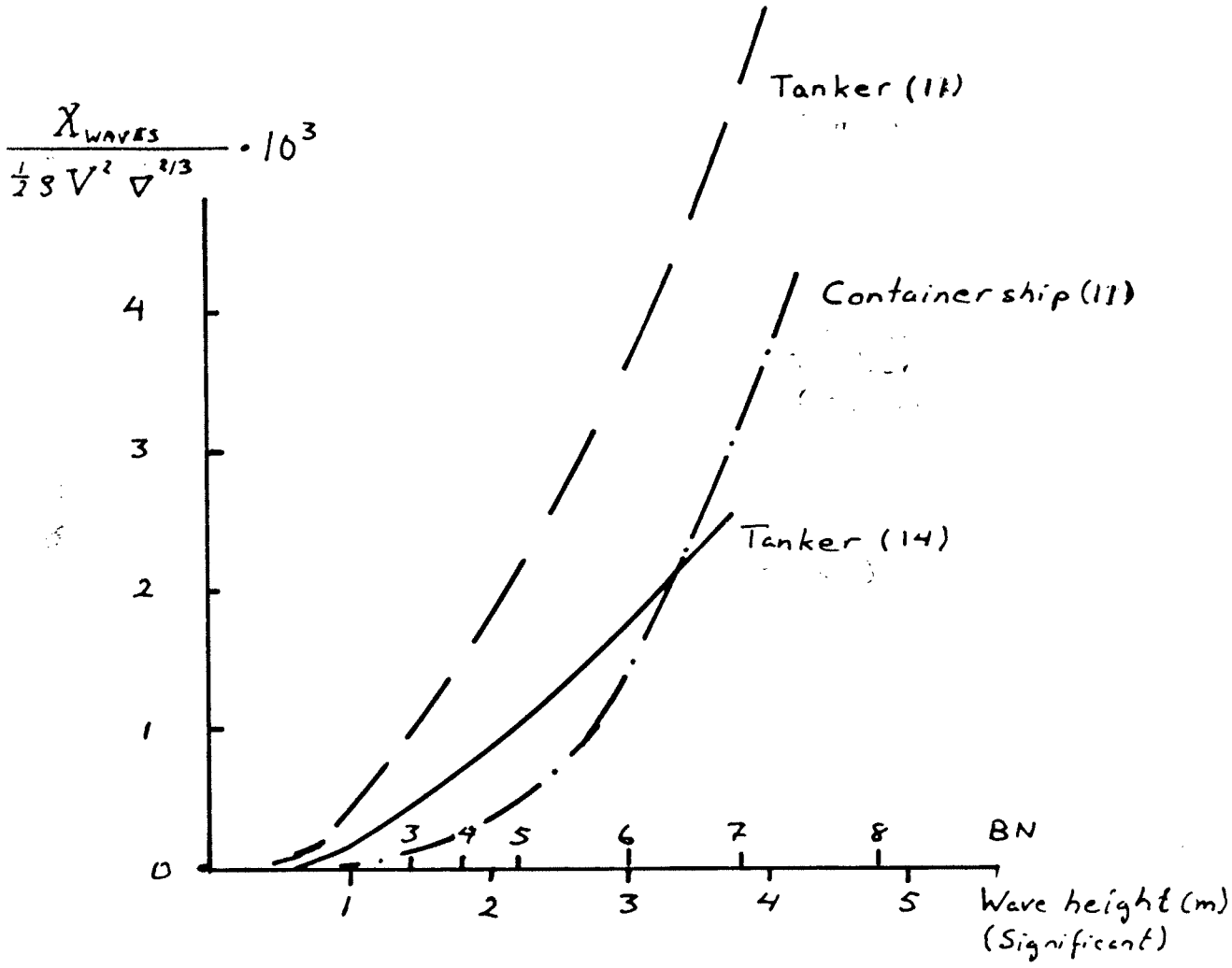


FIG 7 ADDED RESISTANCE INDUCED BY WAVES

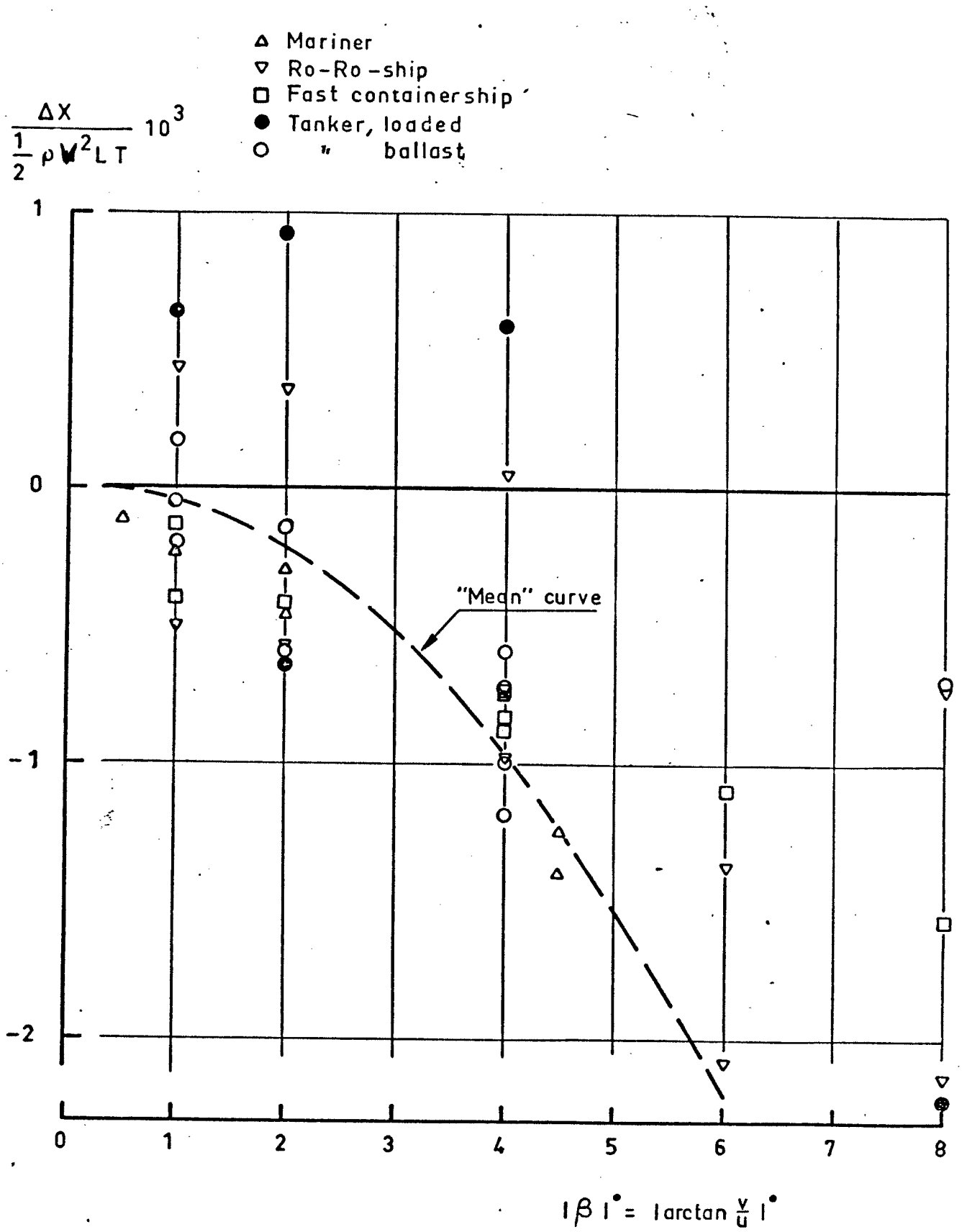


FIG 8 ADDED RESISTANCE DUE TO DRIFT

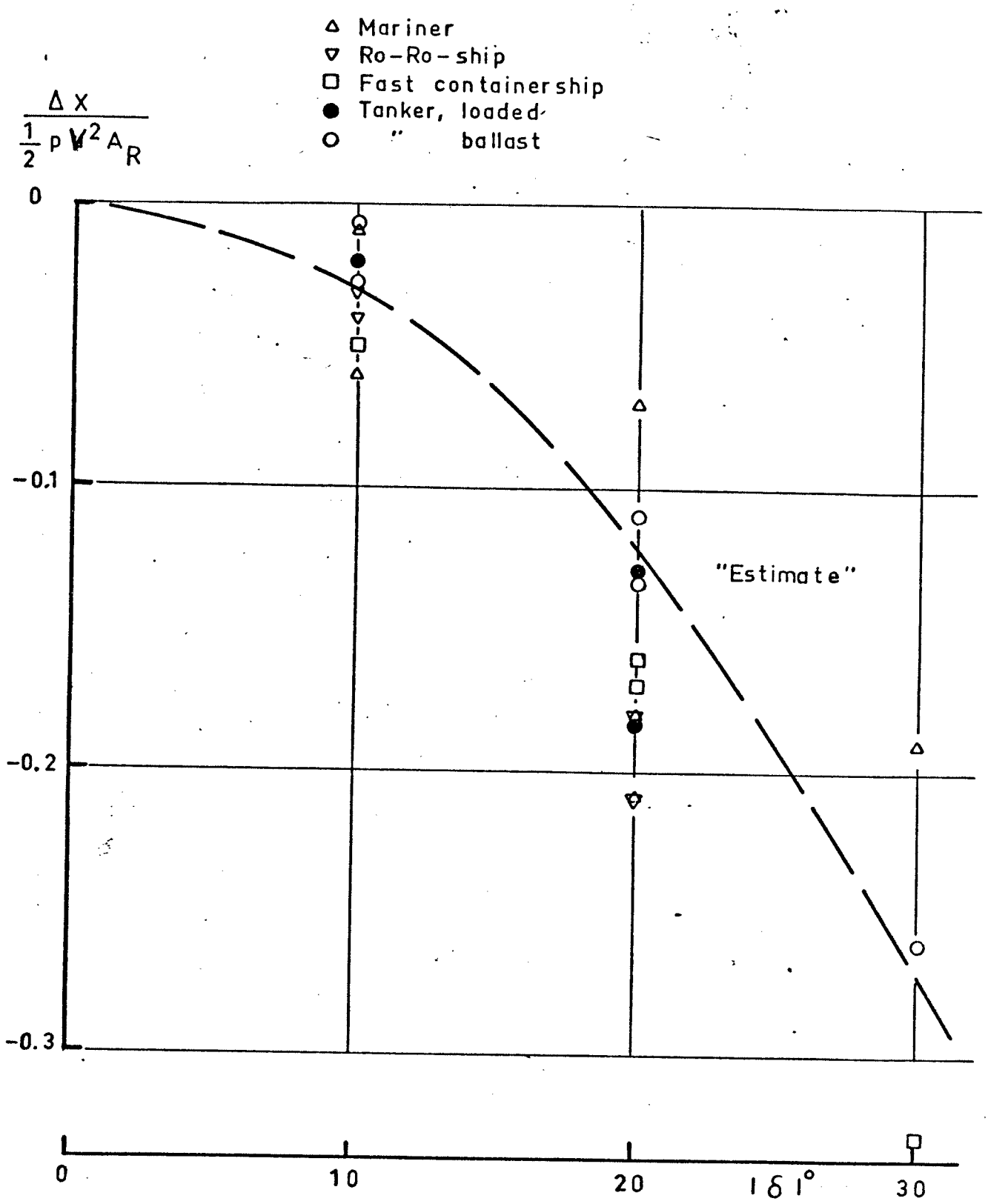


FIG 9 ADDED RESISTANCE DUE TO RUDDER

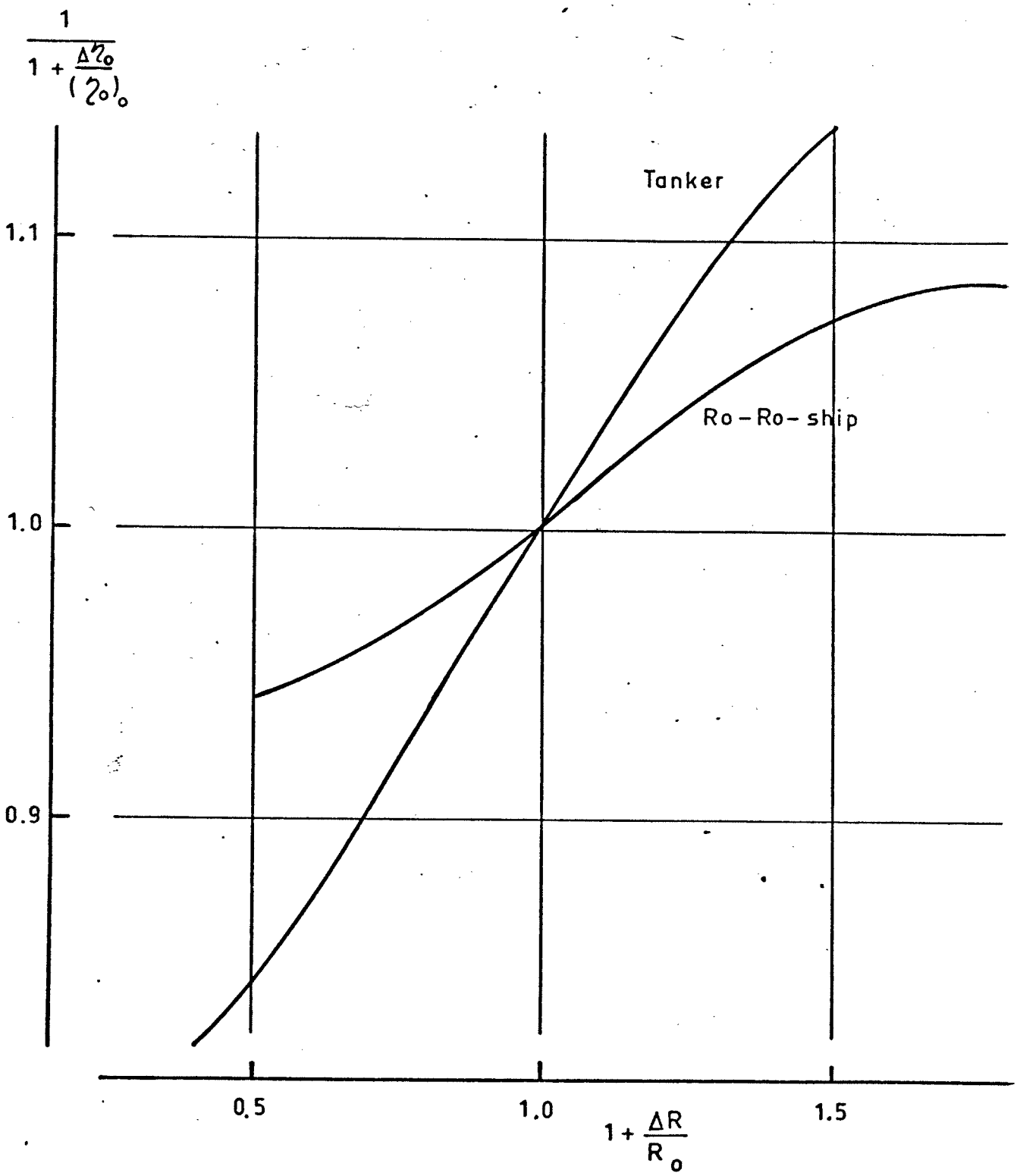


FIG 10 EFFECT OF ADDED RESISTANCE ON PROPELLER EFFICIENCY

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SHIP PROPULSION IMPROVMENT - PRINCIPLES AND A SURVEY OF ALTERNATIVE PROPULSION SYSTEMS

by

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The possibilities of reducing fuel costs by decreasing the resistance of the ship were discussed in the first lectures of this eighth WEGEMT school. The following six lectures will deal with the propulsion aspect of the problem.

The first of these lectures, presented below, is of a general nature in contrast to the remainder, which pay attention to more specific propeller problems and propulsion systems. Attempts are made to classify the different propulsion alternatives according to the reason for their power-reducing capacity. After a description of the principles of a propeller in uniform flow different possibilities to decrease the shaft power by reducing the losses due to axial momentum, slipstream rotation, blade number and blade friction are discussed. Of interest are propulsion systems, such as large slow-rotating propellers, ducted propellers, propellers with vane wheels, contra-rotating propellers and propellers with stator vanes ahead of or behind the propeller. The influence of blade area and form, radial load distribution and number of blades is also examined.

In the second part of the lecture the interaction between propeller and hull is treated and the possibilities to utlize this interaction to reduce the shaft power are reviewed. Stern shapes adapted to large propellers, multiple-screw propulsion, the twin skeg-twin screw concept, the overlapping propellers, different hull-duct-propeller combinations and systems for preventing slipstream rotation are examples of propulsion systems of interest in this connection.

For more detailed information on different systems of current interest the reader is referred to the other lectures.

A. Propulsion Systems in Uniform Flow

A1. Principles

A propeller is a propulsion device which converts the shaft power P_D into a thrust force T by a sternward acceleration of the fluid in which it works. If

Q_0 = the quantity of fluid passing the propeller disk in unit time (m^3/s)

ρ = density of the fluid (kg/m^3)

V_A = the velocity of the fluid approaching the propeller (m/s)

$u_{A_{00}}$ = increase in velocity in slipstream far behind the propeller (m/s)

and if the blade friction, the finite blade number and the slipstream rotation are ignored the momentum law gives

$$T = \rho \cdot Q_0 \cdot u_{A_{00}} \quad (N) \tag{1}$$

The shaft power P_D is equal to the increase in kinetic energy of the fluid

$$P_D = \rho \cdot Q_0 \cdot u_{A_{00}} (V_A + \frac{1}{2} u_{A_{00}}) \quad (W) \tag{2}$$

and the propeller efficiency can be written

$$\eta_0 = \frac{T \cdot V_A}{P_D} \quad \frac{1}{1 + \frac{1}{2} \frac{U_{A_{00}}}{V_A}} = \frac{1}{1 + \frac{1}{2} \frac{T}{\rho \cdot Q_0 \cdot V_A}} \tag{3}$$

Since often both T and V_A are given the first rule for shaft power reduction is:

Rule No 1: Increase the quantity of fluid passing the propeller disk in unit time

For a conventional propeller where the propeller induced velocity u_A at the propeller disk is half as large as in the ultimate wake $u_{A_{00}}$, Q_0 can be written

$$Q_0 = \frac{\pi}{4} \cdot D^2 (V_A + \frac{1}{2} u_{A_{00}}) \tag{4}$$

giving

$$\eta_0 = \frac{2}{1 + \sqrt{1 + C_{Th}}} \tag{5}$$

where

$$C_{Th} = \frac{T}{1/2\, \rho \cdot V_A{}^2 \cdot A} = \frac{8}{\pi} \cdot \frac{K_T}{J^2} = \text{thrust loading coefficient}$$

$$K_T = \frac{T}{\rho\, D^4 \cdot n^2} = \text{thrust coefficient}$$

$$J = \frac{V_A}{D \cdot n} = \text{advance ratio}$$

$$A = \frac{\pi}{4} \cdot D^2 = \text{disk area } (m^2)$$

$$D = \text{propeller diameter } (m)$$

The ideal optimum efficiency η_0 given in eq (5) is obtained if the blade circulation is constant along the radius, i e the increase in total head for the fluid passing the propeller disk is the same at all radii.

The variation of η_0 with K_T/J^2 is shown in Fig 1. As seen η_0 is increased when the propeller diameter is increased, if the thrust T and the advance velocity V_A are constant.

At the same time as a conventional propeller accelerates the fluid sternwards, it creates a certain slipstream rotation. The rotation has two main effects:

1. The thrust of the propeller decreases due to the reduced static pressure caused by centrifugal forces in the slipstream

2. The energy left behind the propeller increases

The rotational losses depend to a very high degree upon the propeller rate of revolution n, Thus, if the blade friction is ignored, K_T/J^2 is given and the circulation is constant along the radius, η_0 increases with increasing $\frac{1}{J}$ (increasing n), as indicated in Fig 2. This tendency is, however, counteracted by an increased friction and η_0 will therefore in reality reach a maximum at a certain rpm, see discussion below.

Another important factor in regard to the rotational losses is the radial load distribution. These losses are much higher if the inner parts of the blades are more loaded than the outer ones than vice versa. It is therefore an advantage to increase the tip loading. Since, however, the axial momentum losses increase when the constant radial load distribution is abandoned the tip loading must be moderate to obtain optimum efficiency. Fig 2 gives some examples of how η_0 varies with 1/J for different load distributions at $K_T/J^2 = 0.37$. At the rate of revs which gives the optimum efficiency, i e $J \cong 0.7$ and $K_T \cong 0.18$, see below, the percentual rotational losses are approximately

11.5% if the load distribution is	constant
6.5% "-	normal
3.5% "-	optimum

The second rule for shaft power reduction can now be formulated:

Rule No 2: Decrease the slipstream rotation simultaneously as other losses are kept down or reduced

Up to now the fluid has been assumed to be ideal. In reality the viscocity creates drag forces on the blades, which reduce the propeller efficiency. Since this reduction is quite important the third rule for shaft power reduction can be written:

Rule No 3: Decrease the drag forces on the propeller blades

The frictional forces depend upon the drag coefficient C_D, the blade area, the blade form and the local velocity. Its influence is demonstrated in Figs 3 and 4, where η_0 with and without friction is plotted against $\frac{1}{J} = \frac{D \cdot n}{V_A}$. A variation of $\frac{1}{J}$ means a variation of

- The rate of revs n for constant diameter in Fig 3, where K_T/J^2 = constant = 0.37 and

- The propeller diameter for constant rate of revs in Fig 4 where $\sqrt[4]{K_T/J^4}$ = constant = 0.93.

In both cases the blade area ratio is A_D/A_0 = 0.50 and the circulation distribution "normal", see Fig 2. As seen in Figs 3 and 4 an increase in 1/J in both cases means that the losses due to blade friction increase. Since the rotational losses, as shown above, have an opposite effect, the efficiency curves have an optimum at a certain rate of revs (K_T/J^2 given) or at a certain diameter ($\sqrt[4]{K_T/J^4}$ given). This

optimum appears at a K_T-value which is of the same order of magnitude, $0.1 \lesssim K_T \lesssim 0.2$, at all propeller loads. The optimum shifts slightly towards a lower K_T when the frictional forces decrease.

It is interesting to note that the optimum efficiency appears at a higher K_T-value if the slipstream rotation is taken away. This means that the optimum diameter of a contrarotating propeller is smaller than for a single propeller.

The calculations described above are all based on the assumption that the propeller has an infinite number of blades. Since the efficiency is lower for a propeller with a finite than with an infinite number of blades, see Figs 3 and 4, it is tempting to formulate the next rule for shaft power reduction by saying: increase the number of propeller blades! Experience has, however, shown that the influence of blade number is more complicated than that, the reason probably being that a change in blade number with given blade area also means a change in blade friction. Thus a study of test results from systematic propeller series [1] shows that the optimum propeller efficiency for a given blade area ratio

- is almost independent of the number of propeller blades if K_T/J^2 is constant but

- increases slowly with decreasing blade number if $\sqrt[4]{K_T/J^4}$ is constant (optimum efficiency at lower K_T-values for propellers with few blades)

The calculations used to determine the propeller efficiency in Figs 2-4 were carried out with one of the design methods for propellers in use at SSPA [2]. Let us summarize the results:

- The ideal propeller efficiency decreases with increasing propeller load K_T/J^2 (decreasing propeller diameter)

- The slipstream rotation means a loss of efficiency which decreases with increasing n if K_T/J^2 is given, see Fig 2 or with increasing D if $\sqrt[4]{K_T/J^4}$ is given.

- The loss in efficiency due to blade friction has an opposite tendency to the slipstream rotation. Curves showing the relationship between η_0 and K_T/J^2 or $\sqrt[4]{K_T/J^4}$ have therefore an optimum, see Figs 3 and 4.

- The efficiency is reduced further if the number of blades becomes finite, especially at lower $\frac{1}{J}$-values and the optimum shifts to a higher $\frac{1}{J}$-value (lower K_T-value).

- The optimum efficiency appears at a K_T-value of 0.12-0.20 if $\sqrt[4]{K_T/J^4}$ is given and 0.18-0.26 if K_T/J^2 is given.

- The percentual losses in efficiency at optimum conditions, based upon the ideal efficiency, eq (5), are approximately 5-7% due to slipstream rotation (normal circulation distribution), 8-11% due to blade friction ($A_D/A_0 = 0.5$; $C_D = 0.008$), 3% due to finite blade number ($Z = 4$) for $J \cong 0.70$. For $J \cong 0.35$ the corresponding figures are: 7-9%, 10-14% and 2% respectively.

A2. Reduction of losses due to axial velocities in the slipstream

After having established the magnitude of the different factors influencing the propeller efficiency, let us now look at different means of reducing the shaft power. First methods used to increase the quantity of fluid passing the propeller disk in unit time will be reviewed.

The most effective way to achieve this goal is to increase the propeller diameter and in this way reduce the propeller load K_T/J^2 or $\sqrt[4]{K_T/J^4}$. To get an optimum propeller also at

the new load the K_T-value must, as shown above, be almost unchanged. Since

$$K_T = \frac{T}{\rho \cdot D^4 \cdot n^2} \qquad (6)$$

this requirement means that an increase ΔD of the propeller diameter D must be accompanied by a reduction Δn of the rate of revs n of

$$\frac{\Delta n}{n} \cong - 2 \cdot \frac{\Delta D}{D} \qquad (7)$$

Let us take an example, where we utilize test results from one of the systematic propeller series of SSPA [1]. The optimum efficiency η_0 and the corresponding advance ratio J for SSPA 4:53 propellers are plotted against the propeller load $\sqrt[4]{K_T/J^4}$ in Fig 5. Assume that the

- o propeller thrust $T = 2.0$ MN (200 tonnes)
- o ship speed $V_S = 8.7$ m/s (16.9 knots)
- o mean wake fraction $w_T = 0.40$
- o propeller diameter $D = 7.5$ m
- o rate of revolutions $n = 2.0$ r/s (120 rpm)

which means that $\sqrt[4]{K_T/J^4} = 1.81$. Fig 5 shows that the J-value 0.35 agrees with the optimum and that the propeller efficiency is $\eta_0 = 0.50$.

Let us now reduce the rate of revs to n = 1.33 r/s (80 rpm). The propeller load $\sqrt[4]{K_T/J^4}$, which is independent of the diameter, will then be $\sqrt[4]{K_T/J^4} = 1.48$. The propeller efficiency is as shown in Fig 5 increased to $\eta_0 = 0.56$, provided that advance ratio is increased to J = 0.42, i e the propeller diameter is increased to 9.4 m. If the other propulsion coefficients are unchanged the larger propeller means a reduction in shaft power of 12%. In reality the gain will in most cases be less, see below.

Another way of increasing the quantity of fluid passing the propeller disk in unit time is to surround the propeller with a flow accelerating duct. The propeller load will then decrease and the duct takes over 20-30% of the thrust. An example of the capacity of the duct is given in Fig 5, where the optimum efficiency and the corresponding J-value for ducted propellers in SSPA systematic series [3] are plotted against $\sqrt[4]{K_T/J^4}$ together with the corresponding curves for a conventional propeller. As seen the gain in efficiency due to the duct increases with increasing propeller load. If we in the example discussed above want to surround the propeller with a duct instead of increasing the diameter, the efficiency η_0 would increase from 50% to 53%, i e a reduction in shaft power of 6%. The diameter could at the same time be reduced from 7.5 to 7.2 m.

Also a ring propeller, which is a propeller with a rotating ring (duct) mounted on the blade tips can increase the quantity of fluid passing the propeller disk. The ring can also reduce the losses due to the finite number of blades. Unfortunately the increased frictional forces outweigh the advantages and the efficiency of a ring propeller is therefore less than that of a conventional propeller [4].

Another, more unusual way to increase the propeller diameter is to mount a large free-running vane wheel behind a smaller highly loaded propeller [5], see lecture P6. The inner part of the vane wheel works as a turbine, which reduces the energy of the propeller slipstream and the outer part works as a slow-rotating propeller. The invention has been described as a hydrodynamically geared large slow-rotating propeller. It is, however, possible that it also has a beneficial effect on the slipstream rotation, see below.

A3. Reduction of Losses due to Slipstream Rotation

The rotational losses depend, as mentioned above, upon the radial load distribution. To obtain the best efficiency this distribution must be optimum, which means that the outer parts of the blades are more loaded than the inner ones. There are, however, two factors which make the choice of optimum load distribution questionable. The first one is that the high tip loads may deteriorate the cavitation and vibratory properties of the propeller. The second one is that the gain may be less than what it seems from Fig 2. The reason for this is that the influence of finite blade number seems to increase when the tip load is increased. Thus at optimum conditions the rotational losses are about 3.5% for an infinite number of blades, see Fig 2, but 9% for a four-bladed propeller [6]. Calculations carried out at SSPA indicate that the losses due to rotation and finite blade number are only slightly higher, 11%, if the blade tips are more unloaded (see normal distribution in Fig 2).

A rational way to decrease the slipstream rotation is to use contrarotating propellers [7]. These propellers have the following important characteristics:

- o They increase the efficiency by about 5%, if K_T/J^2 is given, but decrease the efficiency by about 2%, if $\sqrt[4]{K_T/J^4}$ is given, see Fig 5

- o The optimum diameter at a given $\sqrt[4]{K_T/J^4}$-value is about 15% less than for a conventional propeller, i e $\Sigma K_{Topt} \cong 0.30$ instead of 0.15-0.20, see Fig 5

- o The optimum load distribution means that the inner and outer parts of the blades are almost equally loaded

Let us use the calculation example in Chapter A2 to illustrate the advantage of contrarotating propellers. Assume that we want to decrease the rate of revs from 2.0 r/s (120 rpm) to 1.33 r/s (80 rpm) simultaneously as we replace the single propeller with a set of contrarotating propellers. The gain in efficiency will then be about 10%, which is slightly less than for a single propeller. The optimum diameter is, however, only 8.1 m to be compared with 9.4 m for a single propeller. The smaller diameter is of vital importance to ships with restricted draught. Furthermore, the fact that the blade tips can be less loaded may improve the cavitation properties.

It is possible that one of the reasons why a freely running vane wheel mounted behind a highly loaded propeller decreases the shaft power is a reduction of the slipstream rotation.

Two other methods used to get a similar effect as the contrarotating propellers is to mount stator vanes ahead of or behind the propeller. The vanes ahead of the propeller, which are also called reaction fins [8], give the flow approaching the propeller a prerotation of such a magnitude that the slipstream rotation more or less disappears. Since the tangential velocities of this prerotation are larger at the inner than at the outer radii the maximum of the radial load distribution will be moved inwards, which may contribute to the increased efficiency.

Also the stator vanes behind the propeller take away the slipstream rotation, but the higher local velocities at the vanes increase the frictional losses, which makes the gain in efficiency less. Furthermore, these vanes have no possibility to change the radial load distribution of the propeller.

Although a rudder mounted behind a propeller takes away part of the slipstream rotation, its influence upon the propeller efficiency in open water seems to be small. It increases, however, the efficiency in behind conditions somewhat [9].

A4. Reduction of Losses due to Blade Friction and Finite Number of Blades

Since the blade area ratio A_D/A_0 is one of the factors, which influence the frictional losses, it is important to keep it as low as possible. A reduction in A_D/A_0 from 0.7 to 0.5 means for example an improvement in propeller efficiency of about 3%. The minimum value of A_D/A_0 is in most cases determined by cavitation criteria. These criteria give in general a higher blade area for a propeller with a large number of blades than one with a low number. Furthermore, the frictional coefficient is larger for a propeller with a large number of short blades than for a propeller with few long blades. From an efficiency point of view it is therefore an advantage to have a propeller with few blades [10]. Often the number of propeller blades is, however, selected in regard to vibration criteria to avoid critical resonances in the ship structure.

In the calculations presented in Figs 3-4 the mean drag coefficient C_D was assumed to be $C_D = 0.008$, which gives good agreement with experiments with model propellers with smooth blades. In full scale, where the blade roughness is about 0.01 mm at the trial tests, the corresponding figure is $C_D \cong 0.007$. The blade surface, however, deteriorates with time and measurements have shown that the roughness after some years in service can be about 0.2 mm, which means a drag coefficient of $C_D \cong 0.010$. Since this means that the shaft power is increased 3-4% it is clear that a regular inspection and cleaning of the propeller blades is quite important [1].

B. Propulsion Systems in Behind Condition

B1. Principles

In the discussion above the flow approaching the propeller was assumed to be uniform with constant velocity, V_A. Now the more realistic condition that the propeller is working in the complicated flow at the stern of a ship will be treated. The difference between the speed of the ship V_S and the local water velocity V_A at the propeller plane is expressed in the nondimensional wake fraction w as follows:

$$w = 1 - \frac{V_A}{V_S} \tag{8}$$

This wake can be divided into two parts of fundamentally different natures: the frictional wake w_f and the displacement wake w_d. The frictional wake is a measure of the loss of total head due to the frictional forces inside the boundary layer of this ship [12]. It can be calculated from

$$w_f = 1 - \sqrt{\frac{P_1 - p_0}{\frac{1}{2}\rho V_S^2}} \tag{9}$$

where

P_1 = local total head measured by the centre hole of a pitot tube

p_0 = undisturbed static pressure

The displacement wake w_d, which is present also if the fluid is ideal is caused by the fact that the propeller is working in a region with increased static pressure at the ship stern. It can be written

$$w_d = w - w_f \tag{10}$$

Due to the action of the propeller, the static pressure in the stern region is changed, with the following two consequences:

1. The nominal wake, w_N, which is the wake without propeller, is decreased to the effective wake w_E

2. The resistance of the ship increases from R to R/(1 - t) where t is the thrust deduction

The effective wake is in most cases obtained by measuring the speed V_A at which an open water test must be performed to get the same thrust T and rate of revs n as in the corresponding self propulsion test. The wake fraction w_T determined by this so called thrust identity method is often named the Taylor wake.

The relationship between the shaft power P_D and the effective power $P_E = R \cdot V_S$, where R is the resistance of the ship, is expressed in the propulsive efficiency η_D

$$\eta_D = \frac{P_E}{P_D} \tag{11}$$

which can be divided into the following components:

$$\eta_D = \eta_0 \cdot \eta_H \cdot \eta_R \tag{12}$$

where

$$\begin{aligned} \eta_0 &= \text{propeller efficiency} \\ \eta &= \frac{1 - t}{1 - w_T} = \text{hull efficiency} \\ \eta_R &= \text{relative rotative efficiency} \end{aligned} \tag{13}$$

Of these efficiencies only η_0 is a true efficiency with a maximum value of 1.0. Both η_H and η_R are mostly above 1.0 and also the propulsive efficiency η_D can at best exceed 1.0, for example for bodies of revolution fitted with extremely well-designed contrarotating propellers.

To be able to understand the principles of shaft power reduction in behind condition, let us first study a ship

propelled by an ideal propeller without rotation and blade friction and with an infinite number of blades. The propeller is first assumed to be placed so far away from the ship that both the wake and the thrust deduction are zero and the relative rotative efficiency is 1.0. The propulsive efficiency will in this case be equal to the propeller efficiency in eq (3):

$$\eta_D = \eta_0 = \frac{2}{1 + \quad 1 + \frac{8}{\pi} \cdot \frac{T}{\rho D^2 \cdot V_S{}^2}}$$

If $\frac{T}{\rho D^2 \cdot V_S{}^2} = 0.55$

then

$$\eta_D = \eta_0 = 0.78$$

Now move the propeller into the frictional wake w_f far behind the ship. The velocity of the water approaching the propeller will then be $V_A = (1 - w_f)V_S$ and the propeller efficiency

$$\eta_0 = \frac{2}{1 + \quad 1 + \frac{8}{\pi} \cdot \frac{T}{\rho D^2 (1 - w_f)^2 V_S{}^2}}$$

which, if w_f is assumed $w_f = 0.35$, means

$$\eta_0 = 0.65$$

Since the propeller is far behind the ship the thrust deduction is zero and the hull efficiency η_H becomes

$$\eta_H = \frac{1}{1 - 0.35} = 1.54$$

The relative rotative efficiency is still assumed to be $\eta_R = 1.0$ and the propulsive efficiency η_D is

$$\eta_D = 0.655 \cdot 1.538 \cdot 1 = 1.00$$

If, finally, the propeller is moved forwards to the stern of the ship the interaction between propeller and hull means that the wake is increased by the displacement wake w_d and a thrust deduction is created. Since w_d and t are of the same order of magnitude the effect of moving the propeller forward, on the hull and propulsive efficiencies is relatively small.

Although the propeller efficiency decreases when a propeller is moved from the undisturbed flow outside the ship to the frictional wake behind the ship the increase in hull efficiency is so large that the propulsive efficiency is increased considerably. The reason for this is that the propeller regains part of the energy, which would otherwise have been left in the wake of the ship. A fourth rule for shaft power reduction can be formulated on the basis of this result:

Rule No 4: Move the propulsion units to areas where the frictional wake is high

Let us now study an ideal propeller in the axisymmetric wake field shown in the upper diagramme of Fig 6. Maximum propulsive efficiency is obtained if the resulting axial velocities far behind propeller and ship are constant, i e the total head is independent of the radius. Since the total head of the flow approaching the propeller is lower (larger w_f) at the inner than at the outer radii the propeller blades must in behind condition be more loaded at the hub than at the tips to get optimum efficiency [13], see lower diagramme of Fig 6, and compare with the corresponding uniform flow conditions in Fig 2.

If the slipstream rotation is taken into consideration the load distribution is optimum when the pitch of the trailing vortices far behind propeller and ship is independent of the radius. An example of an optimum load distribution in behind condition is given in Fig 6. A comparison with the

corresponding curve in uniform flow, see Fig 2, shows that also in this case the maximum blade load is shifted inwards in behind condition. The gain in propulsive efficiency obtained if the "normal" load distribution in Fig 6 is replaced by the optimum distribution is in the calculation case used about 2%.

The influence of load distribution on the shaft power discussed is probably true for bodies of revolution. It is much more uncertain if it is valid also for conventional ships, where the wake is irregular and the interaction between propeller and hull is very complicated.

One of the most efficient methods to improve the propeller efficiency is, as shown in Section A2, to increase the propeller diameter and decrease the rate of revs. Let us now use one of SSPA's computer programmes to find out if the gain is the same in behind condition. The propeller is assumed to work in the axisymmetric wake field given in Fig 6. The load circulation is "normal", see Fig 6, the thrust of the propeller T = 2.0 MN (200 tonnes) and the ship speed V_S = 8.7 m/s (16.9 knots). At a rate of revs of n = 2.0 r/s (120 rpm) the optimum diameter is 7.5 m. If n is reduced to n = 1.33 r/s (80 rpm) the optimum diameter is increased to 9.1 m. A comparison of the propulsive coefficients calculated for the two cases is given in the following table:

Propulsion Coefficient		n = 2.0 r/s D = 7.5 m	n = 1.33 r/s D = 9.1 m
Effective wake	w_E	0.40	0.34
Thrust deduction	t	0.10	0.09
Propeller efficiency	η_0	0.528	0.602
Hull efficiency	η_H	1.49	1.39
Propulsive efficiency	η_D	0.787	0.837

The gain in efficiency due to decreased rate of revolution is as seen about 6% to be compared to 12% if the calculations are based only upon the open water characteristics, see Section A2. The reason for the lower gain figure is that the increased propeller diameter for the axisymmetric case used means a reduction of the hull efficiency. This tendency has often been noticed also for propellers behind conventional ships.

The calculations were carried out using an improved version of the method described in [2]. The nominal wake and the static pressure at the propeller disk are given and the effective wake and the thrust deduction are determined simultaneously when the propeller is designed. A short description of the method is given in [13].

The axisymmetric wake assumption used above is of course a simplification. In most cases the wake is irregular and the interaction between propeller and hull very complicated and in many respects not yet fully understood. The influence of the propeller on the irregular flow field and flow separations at the stern caused or suppressed by the propeller are examples of unsolved problems, which have made a rational theoretical treatment of the minimum shaft power problem in behind condition difficult. In many cases we must therefore still rely upon empirical investigations. It should, however, be pointed out that empirical studies, comparing for example different propulsion systems, must be treated with certain caution. The gain figures are often so small that experimental errors, erroneous flow conditions and uncertain prediction methods may very well spoil the comparison and wrong conclusions may be drawn.

B2. Single Screw Propulsion

The propeller of a single screw ship is in general already working in a region, where the frictional wake is high. One way to increase the propulsive efficiency further may be to redesign the stern so that a larger part of the

frictional wake passes the propeller disk. This can for example be done by introducing a stern bulb, which creates bilge vortices. This stern modification must, however, be made with caution, otherwise an increased resistance may outweigh the improved propulsive efficiency.

There is, as shown above, a certain risk that the gain in efficiency obtained by replacing a small propeller with a larger slow-rotating one becomes lower than expected due to a decrease in hull efficiency. Furthermore, the large propeller diameter increases the risk of air-drawing in ballast conditions. To avoid these drawbacks the stern above the propeller is often altered and more or less pronounced tunnel stern have been introduced [14]. If these tunnels are properly designed they can be fairly beneficial, in any case for some types of ships.

B3. Multiple Screw Propulsion

The first rule for shaft power reduction formulated in Section A1 was to increase the quantity of fluid passing the propeller disk. Besides increasing the propeller diameter this can also be done by dividing the thrust on two or three propellers. Take the example used in Section A2 that we have a ship with a speed of 8.7 m/s (16.9 knots), a wake of 0.40 and a propeller thrust of 2.0 MN (200 tonnes). Let us see what we could gain in efficiency, if we divide this thrust on two propellers, but keep the rate of revs unchanged, n = 2.0 r/s (120 rpm). The propeller load $\sqrt[4]{K_T/J^4}$ is then decreased from 1.81 to 1.52, which means, see Fig 5, that the efficiency is increased from $\eta_0 = 0.50$ to $\eta_0 = 0.55$, i e 10% and the propeller diameter is decreased from D = 7.5 m to D = 6.3 m. If the two propellers are placed in a conventional twin-screw arrangement the assumption made above that the wake was the same as for the single screw is wrong. A control calculation shows that a decrease in effective wake from w = 0.40 to w = 0.34 is enough to cancel the whole gain. In reality the decrease in

effective wake is often larger than that and a twin screw arrangement of this type is therefore in general less efficient than the single screw alternative. Its ability to compete is, however, improved if the ship size and the propeller rate of revolution are increased. The required shaft power depends to a high degree upon the boss angle and shape. Also the direction of rotation plays an important role.

An improvement of the propulsion efficiency of a twin screw arrangement may be obtained if the ship is fitted with twin skegs. The two propellers and their rudders are then placed behind each skeg and the flow approaching the propellers reminds one of the flow at a single screw stern. The reduction in shaft power recorded for twin skeg - twin screw arrangements is mainly caused by an increased frictional wake fraction and a decreased resistance due to an improved flow around the propeller bosses.

Tests with tankers and bulk carriers carried out at SSPA indicate that a power reduction compared to single screw propulsion of well over 5% can be expected in the ship's main dimensions, hull fullness and propeller rate of revolutions are constant [15]. Just as for conventional twin screws the advantage of twin skeg - twin screw is increased with increasing ship size and rate of revs.

Another alternative to the conventional twin screw propulsion is the overlapping propeller system [16]. This system consists of two propellers, which have been moved inwards and sternwards so far that their propellers disks partly overlap. Because of their aft location they will work in a frictional wake field, which is almost as good as for a single propeller. Model tests carried out to investigate the characteristics of the overlapping propellers have shown that a power reduction of about 6% can be obtained compared to the single propeller systems. Other model tests have not been so successful, which may indicate that the

gain is questionable or that the design problems are large and not yet fully understood.

B4. Ducted Propellers

Self propulsion tests as well as full scale tests have revealed that a ducted propeller can give a shaft power reduction of 4-9% compared with a conventional single propeller, provided that the propeller load is sufficiently high, see Fig 5. The interaction between hull and duct is, however, very complicated and in some cases the gain expected from open water tests is not obtained in self propulsion. Another difficult problem is that there seem to be scale effects, which make the gain somewhat larger in full scale than at model tests. Thus if the 1978-ITTC prediction method for single screw ships [17] is used to convert the model test results to full scale the roughness allowance ΔC_F must in general be assumed lower than recommended [17].

To increase the propulsive efficiency of a ducted propeller further and to improve its ability to smooth out the wake irregularities attempts have been made to adapt the duct to the nonuniform wake field by making the duct nonsymmetrical [18]. Ducted propellers of this kind have been used with good results on several ships built in the Soviet Union. The cross sections of these ducts are circular at the plane of the propeller, but elliptical at the trailing edge with the major axis vertical.

Many Japanese ships fitted with ducted propellers have suffered from cavitation erosion inside the duct. To avoid this problem Mitsui have designed a new propulsion system, the Mitsui Integrated Duct Propeller (MIDP), where the duct is mounted ahead of the propeller. MIDP is said to achieve considerable improvements in propulsive performance in ships with full hull form [19]. It is in principle possible to move the duct forward without losing efficiency as

long as the propeller remains inside the duct. If, however, the trailing edge of the duct is well ahead of the propeller the open water efficiency is decreased considerably. The fact that a duct mounted ahead of a propeller sometimes can decrease the shaft power of full-shaped ships probably depends upon the fact that the ducts improve the flow around the stern. The duct can in this case be treated more as an appendage to the hull than as part of a propulsion unit. Mitsui have adapted a special prediction method to the MIDP, which is based upon these ideas.

Another new propulsion system, where the duct plays an important role is the so called Tip Vortex Free Propeller [20]. The propeller is placed immediately behind a duct and the blade tips are fitted with end plates. The propeller blades are highly loaded at the tips, a fact which is said to improve the efficiency considerably. The high gain figures claimed from this cause are questionable, but it is quite possible that the duct can give a certain improvement. A special prediction method is used to calculate the full scale shaft power from model test results.

B5. Systems Preventing Slipstream Rotation

The efficiency of a set of contrarotating propellers in open water, is as shown in Section A3, slightly lower than for a conventional propeller at the same rate of revolution - same $\sqrt[4]{K_T/J^4}$ - the main reason being a smaller optimum diameter, see Fig 5. In behind condition on the other hand the smaller propeller diameter has a considerable advantage. Thus, a change over from a conventional propeller to a set of contrarotating propellers implies that

- the hull efficiency and the propulsive efficiency are increased for a given rate of revs

- the rate of revolution can be reduced to a lower value when the size of the propeller is restricted due to for example restricted draught

There are also model tests indicating that the relative rotative efficiency of contrarotating propellers may be higher than for conventional propellers. The reason for this is not yet fully understood. Examples of the shaft power reductions due to contrarotating propellers in uniform flow and in behind conditions are given in [25].

Prerotating guide vanes or reaction fins placed ahead of the propeller in behind condition have two beneficial effects. First the prerotation of the flow approaching the propeller decreases the losses due to slipstream rotation. Secondly the tangential velocities ahead of the propeller may also decrease the rate of revs which means that also the frictional losses are slightly decreased. Due to the irregular flow behind the ship it is advantageous to use different guide vane angles on the starboard and port sides of the ship [8].

Attempts have also been made to create a prerotation by modifying the stern shape. The most radical attempt in this direction made recently is the so called Cochlea-channelled stern proposed by Tommasi for twin screw [21] and single screw propulsion [22]. The idea is to let the stern direct the boundary layer into a swirl, cancelling the slipstream rotation. If the high gain figures claimed (11-19%) are true other effects than the slipstream rotation must be involved.

Also Collatz has reported power-saving by using an asymmetric aftbody for a single screw ship [23]. Velocity measurements at the propeller disk showed, however, that the swirl created by the hull was insignificant and could not explain the 7% gain recorded. The real reason for the shaft power reduction is still to be found.

Many ideas aiming at a reduction of the slipstream rotation by introducing different devices behind the propeller have been tried in the past. Among them stator vanes and

the so called contrarudder can be mentioned. Another device of this kind, the Additional Thrusting Fin, has recently been tested by IHI [24]. The A.T. Fins are attached to the rudder horn and are mounted horizontally, but set at a certain angle of attack. Model tests are said to give a power reduction of 4-5%.

C. Summary

On the basis of theoretical considerations four rules for shaft power reduction are formulated:

1. Increase the quantity of fluid passing the propeller disk in unit time

2. Decrease the slipstream rotation simultaneously as other losses are kept down or reduced

3. Decrease the drag forces on the propeller blades

4. Move the propulsion units to areas where the frictional wake is high

These rules form a rational basis for the understanding of the principles of shaft power reduction for propellers in uniform flow and in axisymmetric wake fields. They must, however, be used with some caution, when the propeller is working in the irregular flow behind a conventional stern, where the interaction between propeller and hull may be large. In these cases one must in the end rely upon empirical investigations. The results from such investigations must, however, be judged with sound scepticism and a good rule is:

5. Do not belive everything you are told!

The high gain figures reported in the literature may be correct, but there is always a certain risk that they

instead are a result of that

- o the propulsion system used as reference is far from being optimal
- o measuring errors have distorted the results
- o questionable prediction methods have been used
- o the models have been too small

It is also possible that the power reduction reported is valid only for the special case tested.

References

1. Lindgren, H & Bjärne, E: "The SSPA Standard Propeller Family. Open Water Characteristics". SSPA Publ No 60, 1967

2. Dyne, G: "A Design Method for Wake Vorticity Adapted Propellers in a Frictional Wake". SSPA Publ No 91, 1981

3. Dyne, G: "Systematic Studies of Accelerating Ducted Propellers in Axial and Inclined Flows". Symposium on Ducted Propellers; RINA, 1973

4. van Gunsteren, L A: "Ring Propellers". Trans Inst Mar Eng 83 (1971) p 21-35

5. Grim, O: "Propeller und Leitrad". JSTG 1966

6. Kramer, K N: "Induzierte Wirkungsgrade von Best-Luftschrauben endlicher Blattzahl". Luftfahrtforschung, Vol 15, 1938

7. Bjärne, E: "Systematic Studies of Contra-Rotating Propellers for Merchant Ships". PROC IMAS 1973

8. Takekuma, K: "Evaluation of Various Types of Nozzle Propellers and Reaction Fin as the Device for Improvement of Propulsive Performance of High Block Coefficient Ships". Shipboard Energy Conservation 80 SNAME, 1980

9. Lindgren, H: "The Influence of Propeller Clearance and Rudder upon the Propulsive Characteristics". SSPA Publ No 33, 1955

10. Fagerjord, O & Andersen, K: "Are the Methods to Obtain Maximum Propulsion Efficiency Appropriate?" Paper No E3 SNAME Costs and Energy -82 Symposium, Sept 1982

11. Grigson, C W B: "Propeller Roughness, its Nature and its Effect upon the Drag Coefficient of Blades and Ship Power". RINA 1982

12. Dyne, G: "On the Scale Effect on Thrust Deduction". Trans RINA, 1973

13. Dyne, G: "On Optimal Wake Vorticity Adapted Propellers". PRADS 83, Tokyo 1983

14. - "The Ideas behind B and &'s Economy Hull Design". Motor Ship (1977):688, Nov, p 70-72

15. Williams, Å: "Single and Twin-Screw Propulsion of Tankers and Bulk Carriers". SSPA Publ No 74, 1975

16. Pien, P C & Ström-Tejsen, J: "A Proposed New Stern Arrangement". NSRDC Rep 2410, 1967

17. - Report of Performance Committee, 15th International Towing Tank Conference (ITTC), the Hague, 1978

18. Turbal, V K: "Theoretical Solution of the Problem on the Action of a Non-Axisymmetric Ducted Propeller System in a Non-Uniform Flow". Symposium on Ducted Propellers, RINA, 1973

19. - "Mitsui Integrated Duct Propeller". Pamphlet from Mitsui Engineering and Shipbuilding Co Ltd

20. Ruize-Fornelles Gonzales, R, Peres Gomez G & Baquierizo Briones, I: "Fuel Savings from a New Type of Propellers". Marine Propulsion International 1981/82 Dec, Jan, p I-XVI

21. Tommasi, G B: "Experimental Evaluation of a New Stern Shape". ISP Vol 23 No 258, Febr 1976

22. Tommasi, G B: "First Model Tests of a Cochlea-Channelled Stern for a Signle Screw Ship". ISP Vol 29 No 331, March 1982

23. Collatz, G: "Treibstoffeinsparung durch asymmetrische Hinterschiffsformen für Einschrauben-Schiffe". Hansa - Schiffahrt - Schiffbau - Hafen, 119. Jahrgang, 1982, Nr 1/2

24. - "Power Savings from Additional Thrusting Fins". The Motor Ship, Sept 1982

25. Lindgren, H, Johnsson, C A & Dyne, G: "Studies of the Application of Ducted and Contra-Rotating Propellers on Merchant Ships". ONR 7th Symposium on Naval Hydrodynamics, Rome 1968

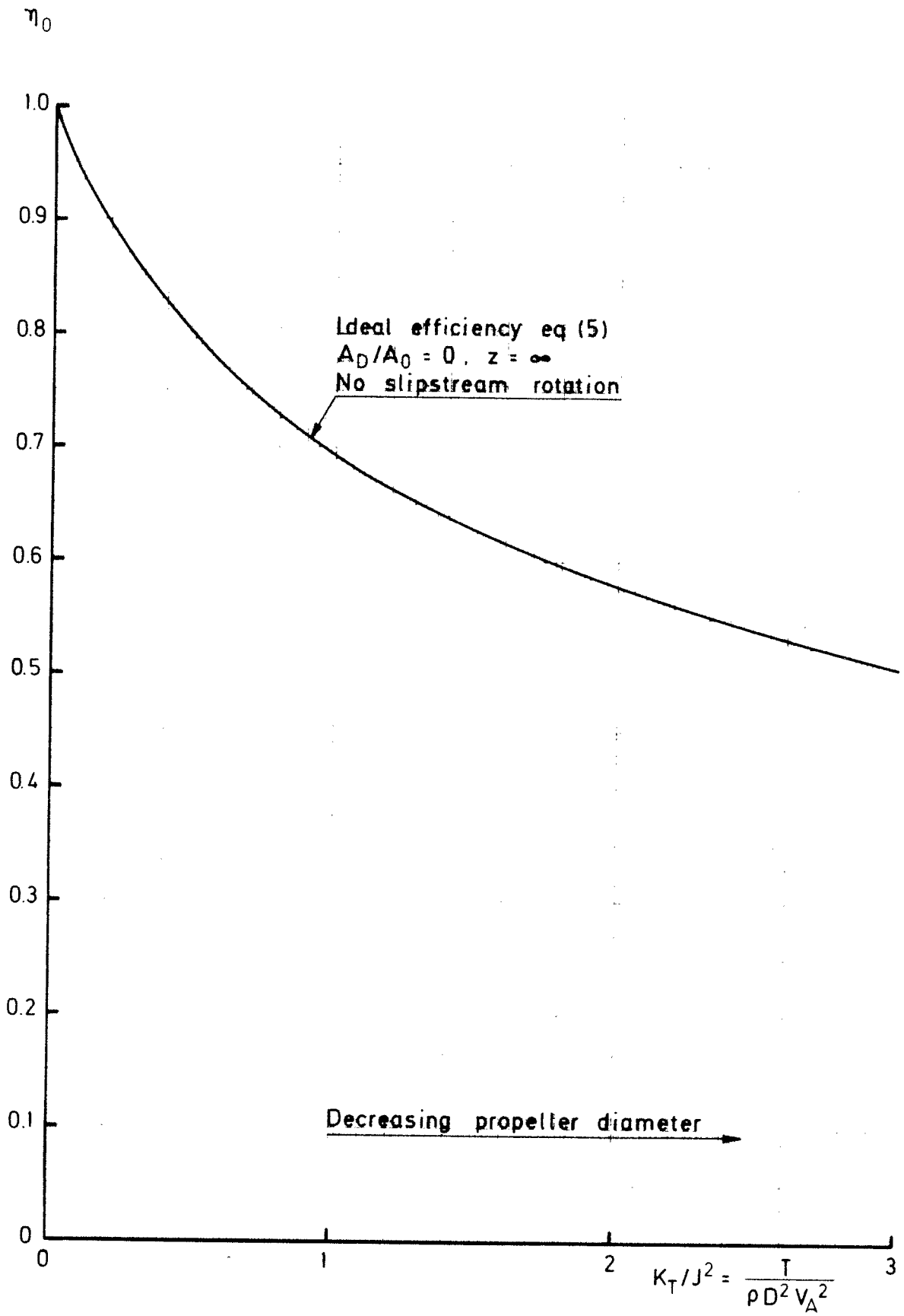


Fig 1 Ideal propeller efficiency at different K_T/J^2

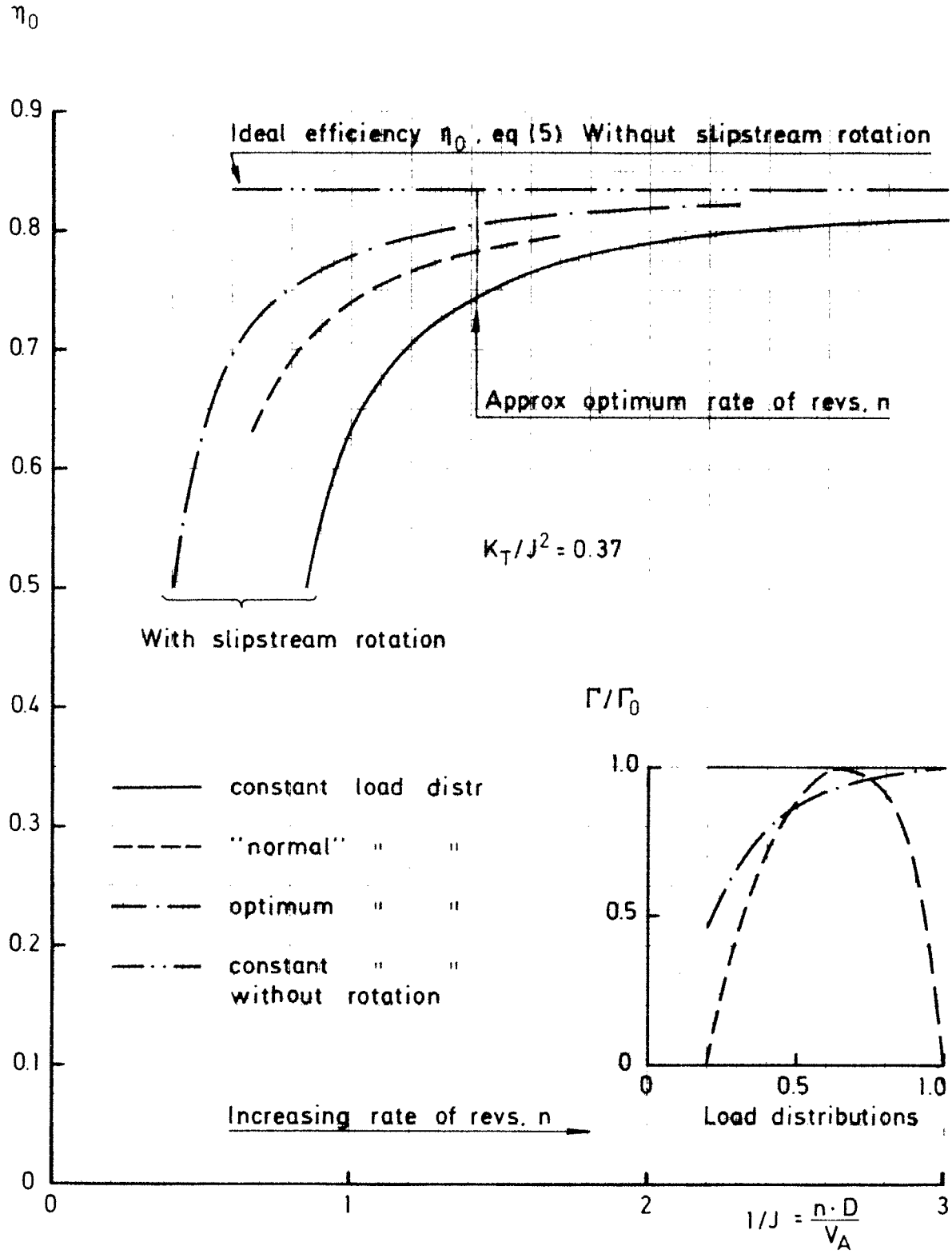


Fig 2 Influence of rate of revs and blade load distribution on propeller efficiency; No friction; infinite number of blades.

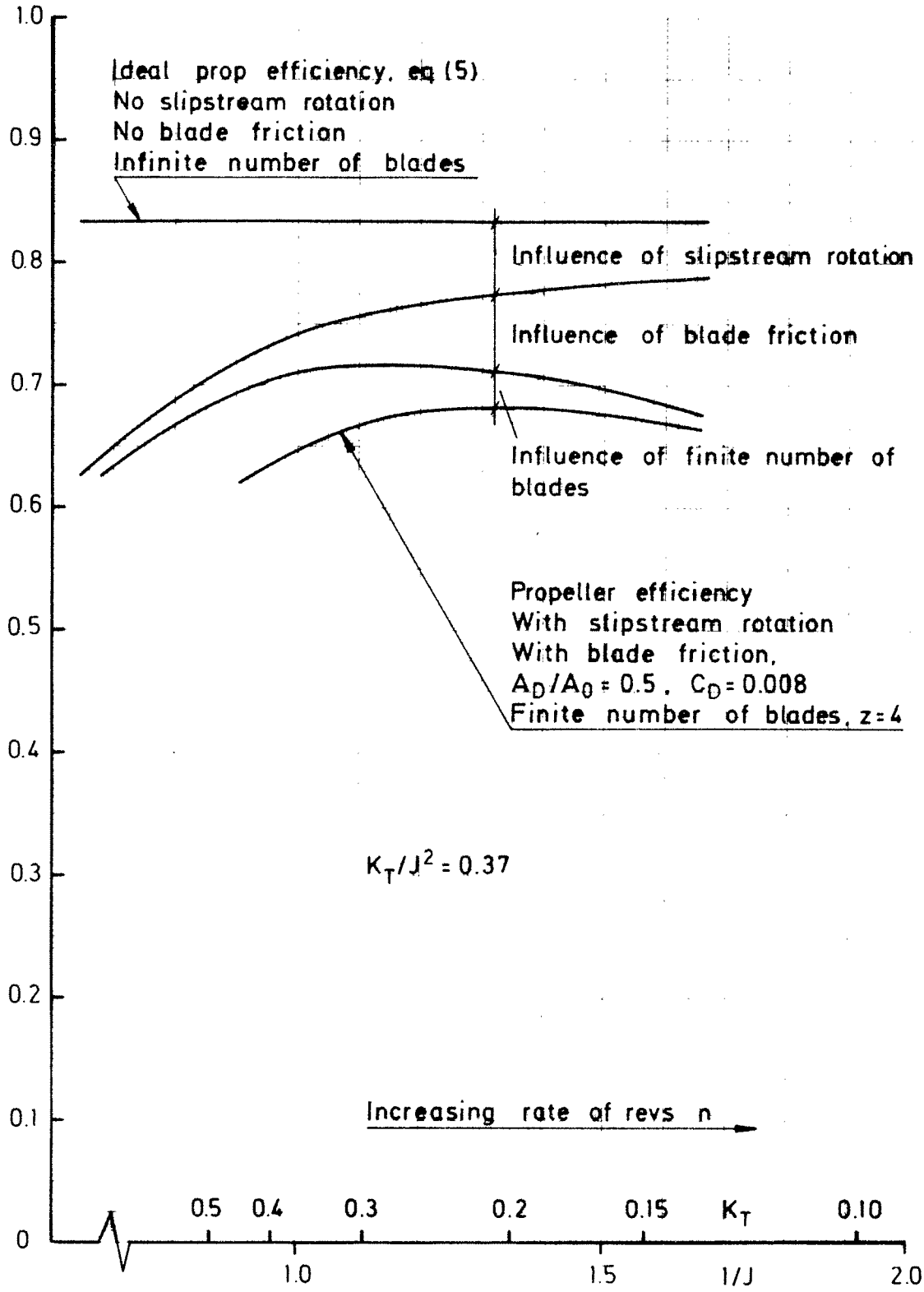


Fig 3 Influence of slipstream rotation, blade friction and finite number of blades at varying rate of revs

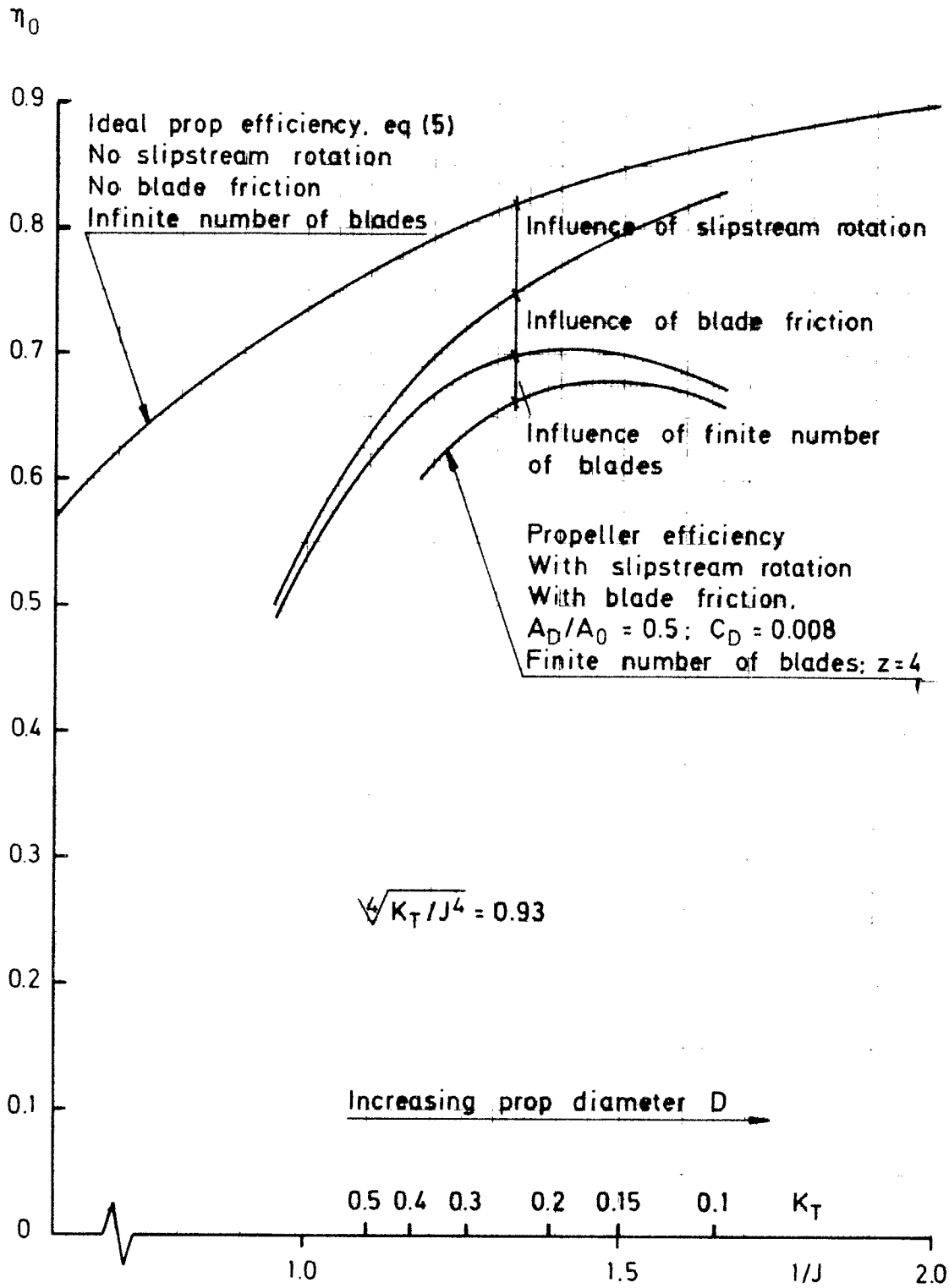
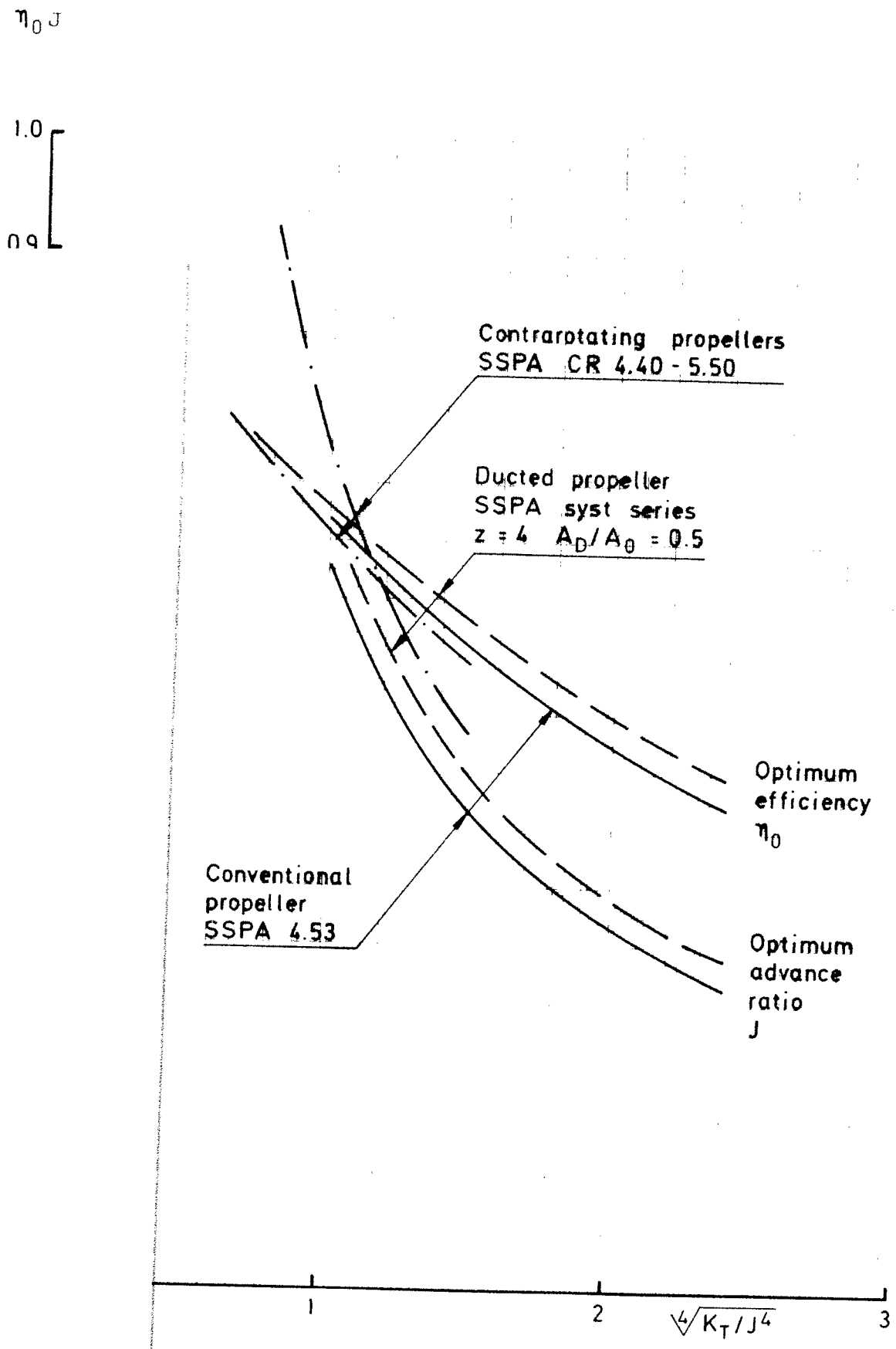


Fig 4 Influence of slipstream rotation, blade friction and finite number of blades at varying prop diameter



timum efficiency and advance ratio of conventional ducted
d contrarotating propellers at different $\sqrt[4]{K_T/J^4}$

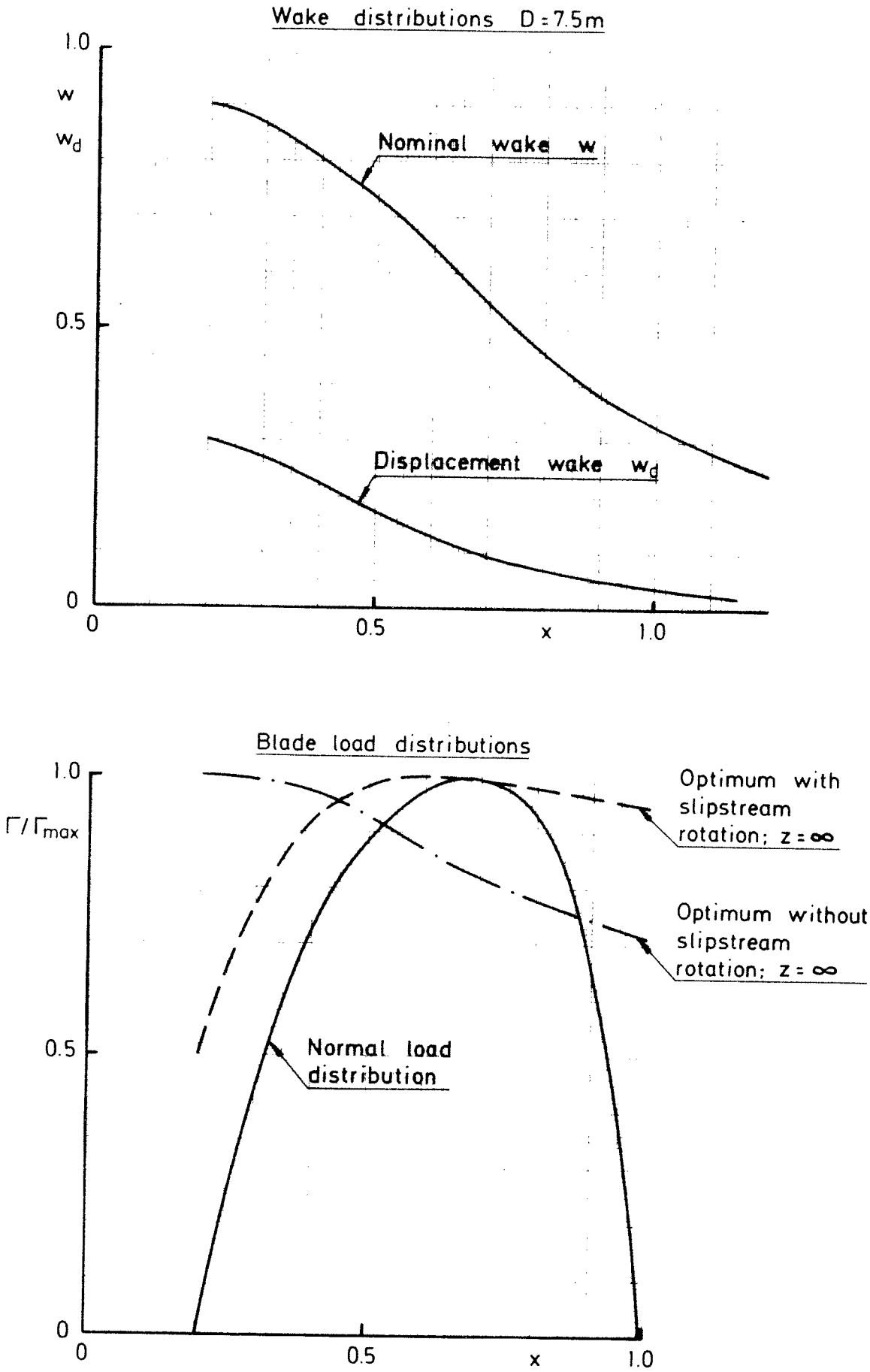


Fig 6 Wake and blade load distributions

WEST EUROPEAN GRADUATE EDUCATION
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MARINE PROPELLER DIMENSIONS AND THEIR EFFECTS ON PROPELLER EFFICIENCY

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Introduction

Wake adapted propellers were introduced some decennia ago. At that time it was understood that propellers work in a variable flow field caused by the potential flow around the ships hull and by the boundary layer around the aft body. In the beginning the denomination wake adapted was used when only the radial pitch distribution was adapted to the inflow conditions. However, the name wake adapted suggested more than only the variation of one parameter. Nowadays wake adaption means much more and we have reached the situation that all parameters describing the geometry of the propeller are determined in such a way that the influence of the wakefield is taken into account.

A similarity with wake adapted propellers can be observed when the denomination slow runners is used for propellers. Not only the adaption of one parameter, in this case the rotational speed, is sufficient to obtain the desired result, being the highest possible efficiency. In essence, so called slow speed propellers do not differ from other propellers. They are designed with the same methods according the same theories as other propellers. But, since the ratio of fuel cost to operating cost of the ships drastically increased, limitations for propeller speeds and dimensions were shifted. Although basic characteristics such as specific loading (C_T, B_P) and geometrical proportions (P/D, f/c, A_E/A_0) did not change to such an extent that their values come outside ranges in which already propellers were operating.

The first slow speed propellers were applied to tankers and container vessels where reduced cruising speed and ample clearances allowed larger propellers running at rotational speeds lower than used in the original designs. Later new designed ships appeared where absolute low numbers of revolutions were used. This explains the denomination slow speed, but it should be realized that no new developments for the propeller itself took place. So slow speed propellers are lying within the existing experience with respect to both design and manufacturing. One aspect may be an exception. The propeller hull interaction for large propellers and adapted aft bodies can cause unexpected effects, because the knowledge on propeller ship interaction is more based on empiricism than on theory.
What can be derived from the introduction of slow speed propellers is that altering restrictions can increase the propulsion efficiency. Therefore it will be shown what propeller losses consist of and how they are influenced by restrictions.

Order of magnitude of propeller efficiencies

The simplest representation of a propeller is the so called actuator disk. An actuator disk is an infinitely thin circular disk which accelerates the water, disregarding how this is achieved. The (simplest) theory of this simplest representation shows that the efficiency of a propeller is

$$\eta_p = \frac{2}{1 + C_T + 1} \qquad (1)$$

where C_T = specific loading = $\frac{T}{\frac{1}{2}\rho V^2 \frac{\pi}{4} D^2}$

T = thrust

V = intake velocity

D = diameter

ρ = specific density

This of course only estimates the axial losses of the propeller and neglects both the tangential losses, represented in the propeller slipstream by slipstream rotation, and the frictional losses due to friction forces on the propeller surface.

The order of magnitude of these losses can be derived from comparison with results of systematic experiments with propeller models as have been performed with the well-known Troost B series.

In Figure 1 the theoretical curve for the actuator disk as well as lines for optimum B series propellers are given. Optimum means in this respect that the rotational speed is determined in such a way that the rotational and frictional losses are minimal. As can be seen the rotational and frictional losses depend only slightly on the propeller loading. Since the B series has been designed with high attention for the efficiency, the efficiency of the B series can be considered as a good yardstick for what propeller efficiency can be obtained when the propeller loading is given. Also in the same figure over a hundred points are given representing designs made by the design department of Lips B.V. in 1982. Some of these propellers are sailing and others are in the manufacturing stage. It can be seen that most of them do not have the maximum obtainable efficiency. This must be explained by the existence of restrictions, for example non optimal number of revolutions, moderate to strong variations of velocity in the wakefield and desired low level of radiated pressure pulses.

In Figure 2 the difference in real efficiency and what could be obtained (η_B) is shown along the horizontal axis and for each percent difference interval the total design power of all propellers showing such a difference is given.
The conclusion to be drawn from this figure is that potentially for these propellers 50000 HP could be saved in delivered power representing about 10 million dollars per year sailing.
The first conclusion to be drawn from Figure 1 is the well-known rule in propeller design which can be considered as the main relevant and can be formulated as follows: A propeller has the highest efficiency when the diameter is as large as possible under the condition that the number of revolutions is optimum. Also can be concluded from Figure 1 that the difference between the ideal or theoretical efficiency and the practical efficiencies achieved for the whole range of propeller loadings is fairly constant. To the authors knowledge there is no theoretical explanation known for this almost constant loss in efficiency. In frictionless flow the rotational losses can be reduced to very low values by increasing the rotational speed to extreme values. In Figure 3 from reference [1] this is illustrated by the horizontal behaviour of the efficiency curves as function of the advance ratio. The corresponding maximum efficiency values are in agreeement with the theoretical values of Figure 1.
By the same token the frictional losses increases with increasing rotational speed. The optimum apparently results in a constant loss, where both frictional and rotational losses are fairly constant. This is also indicated in Figure 1 where the rotational losses for optimal propellers are calculated with the same theory as used to derive Figure 3.
In Figure 4 the effect is shown of non optimal rotational speeds or advance ratios J on frictional and rotational losses. The tendencies of the curves are already explained above. The total loss has a clear but flat minimum. With the dotted line is shown what the optimum efficiency is for a B 4-70 series propeller operating at a C_T value equal 2 but at various advance ratios. The tendency of this curve is clearly the same as the tendency of the curve for frictional plus rotational losses which have been derived with theoretical means.
Also is shown in the same figure what the effect of increased frictional resistance will be. For a controllable pitch propeller where the rotational speed can be freely taken, provided no torque limits are exceeded, it can be derived that with a rough propeller with high frictional resistance a change in rotational speed can result in lower total losses in the order of magnitude of one percent.

Combining the formulaes (2) and (3) for $C_T = 2$ and an average hydrodynamic pitch angle of 30 degrees gives a change in efficiency as function of blade area ratio which is given in Figure 5 by the dotted line. It is clear that the tendency of this curve resembles the tendency of the experimental values. In other words blade area has to be considered as a parameter mainly causing the same effects as aspect ratio and only to a quite limited extent as a parameter which causes additional frictional losses for optimum propellers.

The efficiency values for the points representing actual designs in Figure 1 have been calculated in the flow field as met behind the ship. This means that thrust and torque have been calculated in a number of angular positions of the propeller and the corresponding values have been integrated over the full rotation. The efficiency in the varying flow field is definitely lower than in a homogeneous flow. The most important factor in this respect is the change of drag-lift ratio during varying angles of attack. Since the drag of a two dimensional section can be approximated by a parabolic function of the lift one can understand that during a part of the revolution the value of ε can have values of twice the average value. From detailed calculations of actual designs it appeared that a drop in efficiency of 3% is possible.

Another reason why efficiencies of actual design cases, as shown in Figure 1, are considerably lower than what can be obtained is the occurrence of cavitation. The present state of the art is that damage of the propeller surface due to collapsing cavities can very well be avoided. Also the theorectical means are available to compute the order of magnitude of the hull pressure fluctuations which are caused by the growing and disappearing cavity. Fluctuating pressures at the ship's hull, due to the passage of the propeller blade along the hull when cavitation is left out of consideration, has to be taken into account also since it contributes to the pressure level. However, the contribution of the changing volume of the cavity is normally dominant.

Consequence of avoiding negative effects of cavitation means a further reduction in efficiency. The various parameters which can be adapted to counteract the eventual oversized cavities such as pitch at the blade tip, skew, blade thickness and section camber are often used in combination. Therefore general rules or estimations on the reduction of efficiency due to acceptance of only restricted cavity dimensions cannot be given. However, examples show that a 2% reduction in efficiency can occur.

To illustrate the effect of varying inflow condition on the efficiency a design example has been made for a propeller absorbing 14700 KW at 87.5 revolutions per minute while the ship sails 18.4 knots. When the restrictions are taken into account step by step, the following design conditions can be distinguished. The starting point is a propeller without the ship's presence. Since the final design is a 5-bladed propeller, the first estimate of the efficiency can be made from a B5-45 series diagram. It follows than that for the same power to reach the same ship's speed a propeller of 12 m diameter running at 40 revolutions per minute the efficiency will be over 75 percent. This leads to an irrealistic diameter for this case as soon as the clearances between propeller and ship are considered. Behind the ship a diameter of only 8.0 m fits. In a homogeneous flow, so not taking into account the wakefield of the ship, the efficiency for this diameter drops considerably.
In Fig. 6 the various steps are schematically shown, introducing step by step: reduction of diameter to fit with the clearances, wakefield but highly submerged (no cavitation) and realistic situations. The effect on efficiency is shown in Figure 7. The final efficiency is only 57 percent. As can be seen the sequence of importance of the effects is as follows: diameter reduction, blade area increase to reduce cavitation and additional drag to extra angles of attack.

Ship hull interaction

After having mentioned the most important effects on the efficiency of the propeller it is good to realize that the total propulsive efficiency contains more than only the propeller efficiency. The mutual interference between propeller and hull can have a great influence on the total efficiency. Therefore, when applying a large propeller diameter for slow speed propellers, it is important to realize that the so called hull efficiency is influenced and mostly in a negative way.
The total efficiency is defined as

$$\eta_p = \eta_o \cdot \eta_H \cdot \eta_R \qquad (4)$$

where η_p = total efficiency
η_o = propeller efficiency
$\eta_H = \frac{1 - t}{1 - w}$ = hull efficiency
η_R = relative rotative coefficient
w = wake fraction
t = thrust deduction factor

The physical meaning of these parameters is generally known and will not be treated here.
Experience learns that the effect of diameter increase on the relative rotative coefficient is minor and can be neglected. Under the assumption that the rotational speed is taken optimum the change in efficiency due to diameter increase can be derived from equation (1):

$$\frac{d\eta_o}{dD} = \frac{d\eta_i}{dD} = \frac{d\eta_i}{dC_T} \frac{dC_T}{dD}$$

combined with equation (1) this leads to

$$\frac{d\eta_p}{dD} = \eta_R \cdot \frac{1-t}{1-w} \cdot C_T \frac{d\eta_i}{dC_T} \left\{\frac{2}{1-w} \frac{dw}{dD} + \frac{1}{1-t} \frac{dt}{dD} - \frac{2}{D}\right\}$$

$$+ \eta_R \cdot \eta_o \frac{1}{1-w} \left\{\frac{1-t}{1-w} \frac{dw}{dD} - \frac{dt}{dD}\right\} \qquad (5)$$

Experiments teach us that the effect of diameter increase on thrust deduction is less important than the effect or wake. Neglecting terms in equation (5) with the derivative of the thrust deduction factor leads to the following approximation formula.

$$\frac{\%\ \text{fuel}}{\%\ \text{diameter}} = \frac{d\eta_i}{dC_T} \cdot \frac{2C_T}{\eta_o} \left(\frac{D}{1-w} \frac{dw}{dD} - 1\right) + \frac{D}{1-w} \frac{dw}{dD}$$

or $$\frac{\%\ \text{fuel}}{\%\ \text{diameter}} = a \left(\frac{bD}{1-w} - 1\right) \frac{C_T}{\eta_o} + \frac{bD}{1-w} \qquad (6)$$

Coefficient a is the gradient of the function in Figure 1 and coefficient b can be estimated from experimental results. See Figure 8. The value of b lies between - 3/L and - 6/L, where L is the ship's length. It turns out that up to half of the gain in efficiency, due to diameter increase, can be cancelled by the change in hull efficiency.

Effect of diameter increase on other parameters

The optimum rotational speed for a propeller can in first instance be estimated from a series diagram. It then turns out that optimum revolutions are obtained for almost constant value of torque coefficient K_Q. This coefficient is defined as:

$$K_Q = \frac{Q}{\rho n^2 D^5}$$

where Q = torque
n = rotational speed

Of course this constant value of K_Q is depending on the type of propeller, but as soon as the type of propeller with an estimated blade area has been determined, one can estimate the required change in rotational speed by means of the following formula:

$$n = \text{const. } D^{-5/3} \tag{7}$$

This also means that the cavitation number σ_n will change according

$$\sigma_n = \text{const. } D^{4/3} \tag{8}$$

As already shown above the magnitude of the blade area is dominated by suppressing unwanted cavitation phenomena. Detailed calculations with the velocity distributions in the wakefield as input have to be made to optimize the blade shape and blade area. However, in the course of the years many approximation formulaes and diagrams have been published. With these formulaes and diagrams one can estimate the blade area in an early design stage. In this context they can be used to illustrate the effect of diameter increase.
Three parameters play a role in these approximations: the propeller loading expressed in C_T, C_p or similar parameters, the cavitation number related to rotational speed, entrance speed or a combination and the variation in the wake fraction during a revolution at a relevant radius. Since these diagrams are mainly based on empirical data it is important to realize that each designer will use his own version. This means that the value of such diagrams is limited and mainly restricted to give indicative information.

For the present purpose a diagram based on C_T and σ_n is suitable. In Figure 9 three of these curves are given. Curve 1 is from reference [3], curve 2 is from reference [4] converted to C_T and σ_n for $\Delta w = .33$ and curve 3 is the well-known Burrill curve for merchant vessels with all type of suctions. In Figure 9 is also shown what the consequence is of a diameter increase. If point A represents the orginal propeller, a diameter increase then causes that the point, representing the propeller, follows in the diagram the direction of the arrow. Since this direction is rather close to the perpendicular on these boundary curves it can be concluded that diameter increase is one the most powerful tools to avoid cavitation problems. At the same time this conclusion can be translated as follows. When the diameter of the propeller is increased the blade area ratio can be reduced. The physical background of such diagrams giving the first indication on cavitation limits is the comparison of the average under pressure on the blade with the pressure drop before cavitation occurs. According such a global method it would mean that with an increase in diameter the blade area ratio varies linear with the reciprocal of the diameter.

Effect of roughness upon efficiency

Another effect important for the efficiency of the propeller is the roughness of the propeller blade. The roughness of the blade surface is strongly depending on the quality of the grinding and on the time spent in environmental conditions. Figure 10 shows some typical values for the centre line average roughness (R_a).

New propeller	Class 1 (ISO 484/1)	< 6 µm Ra
	Class S (ISO 484/1)	< 3 µm Ra
Propeller 12 to 24 months in service		20 µm Ra

Figure 10 Typical values for propeller roughness

At increasing roughness of the propeller blade the skin friction and therefore the drag of the blade increases. This increment of the skin friction has been investigated by Schlichting [5] and Nikuradse and Prandtl with the aid of sand roughness tests for plates.
This has resulted in the following empirical relation for the increase in drag coefficient of a blade section

$$\Delta C_D = 2 \cdot \left\{ \left(1.89 + 1.62 \log \left(\frac{c}{2\,kg}\right)\right)^{-2.5} - 0.455 \left(\log R_c\right)^{-2.50} \right\} \qquad (9)$$

with R_c - local Reynolds number $(\frac{cV}{\nu})$
c - chordlength
V - inflow velocity
ν - kinematic viscosity
kg - equivalent sand roughness

To estimate the effects of section drag upon the propeller characteristics the equivalent blade section approach of Lerbs [6] may be used. The equivalent sand roughness has to be related to the actual propeller roughness such that realistic changes in friction can be calculated. The usual method for circumventing this is to measure the drag of the rough surface at laboratory scale, compare it with that predicted by equation (9) and thereby determine an equivalent sandgrain size. However, some discrepancies exist, the Prandtl-Schlichting formula applies to surfaces having roughness elements which are all geometrically similar (characterized by one parameter). This is not usually the case for surfaces produced in an industrial process, which have textures different from that of uniform sand. The second reason follows from the fact that the first part of equation (9) does not involve the Reynolds number. For industrial surfaces the variation of the roughness is such that this difference should be taken into account. For a more theoretical treatment reference [7] can be useful.
To illustrate the effect of roughness upon efficiency, results of measurements with a B5-75 type propeller with different roughnesses are given in Figure 11 [8]. The results show that, at these Reynolds numbers, increasing k from 15 to 50 μm Ra decreases the efficiency with about 5 percent. Although this difference is expected to be smaller on full scale, the loss in efficiency will be large enough to pay for a repolishing of the propeller blades.

References

[1] Beek, T. van and Verbeek, R.: "Design aspects of efficient Marine propellers", Lips Technical Report no. 5330-8308 to be published in Jahrbuch der Schiffbautechnische Gesellschaft, 1983.

[2] Mises, R. von: "Theory of Flight", Dover publ. New York, 1958.

[3] Gunsteren, L.A. van and Pronk, C.: "Propeller design concepts", 2nd Lips Propeller Symposium, 1973.

[4] Jonk, A. and Beek, J. van de: "Some aspects of propeller-hull interaction", 5th Lips Propeller Symposium, 1983.

[5] Schlichting, H.: "Boundary-layer theory", MacGraw-Hill Book company, sixth edition, 1968.

[6] Lerbs, H.W.: "On the effects of scale and roughness on free running propellers", Journal of the Society of Naval Architects, 1951.

[7] Grigson, C.W.B.: "Propeller roughness, its nature and its effects upon the drag coefficients of blades and ship power", The Naval Architect, 1982.

[8] Oosterveld, M.W.C. (ed.): 15th International Towing Tank Conference, Report of the propeller committee, The Hague, 1978.

[9] Harvald, S.A.: "Wake of merchant ships", Doctor's thesis, The Danish technical press, Copenhagen, 1950.

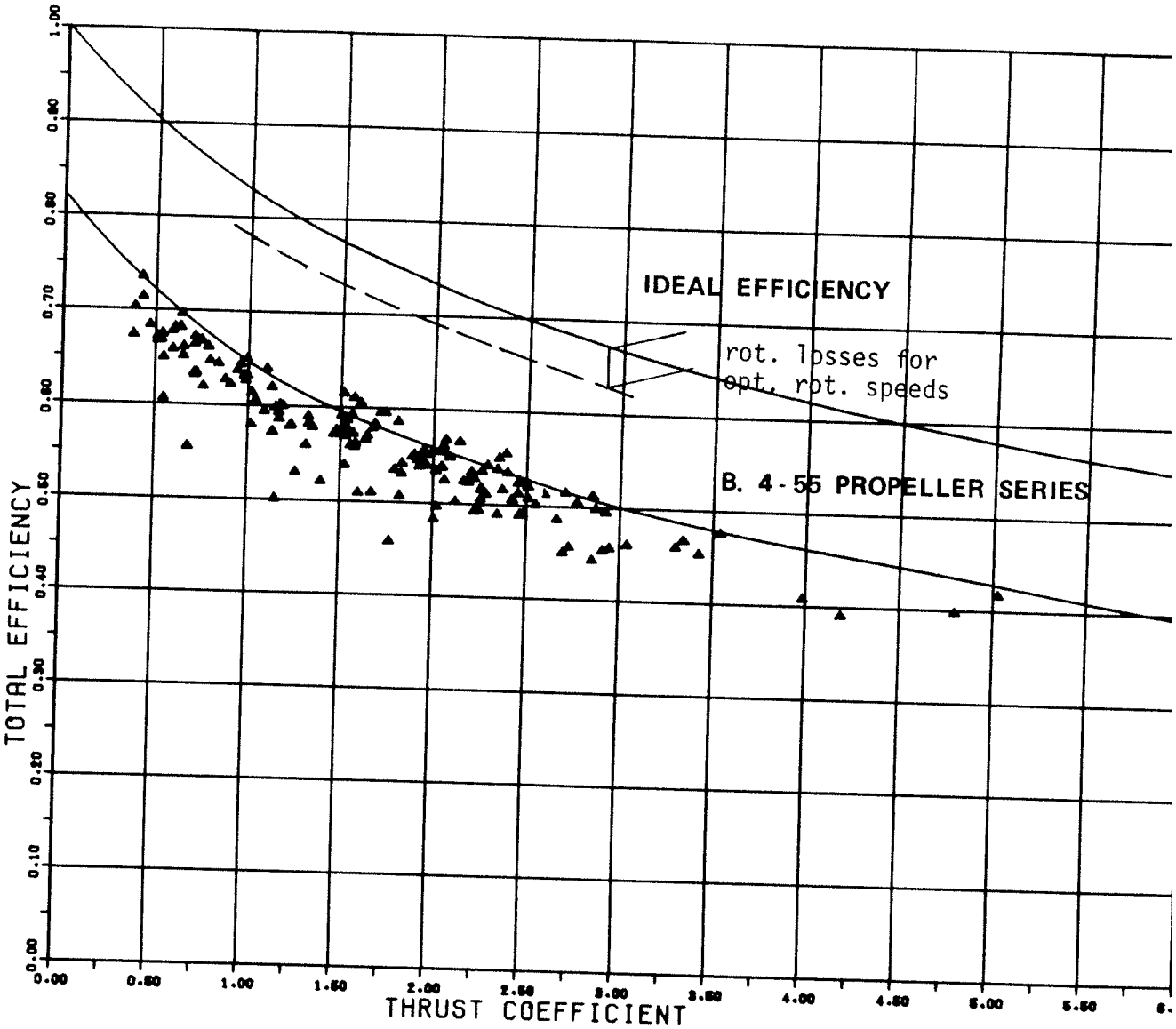


Figure 1 Optimum theoretical efficiency, maximum efficiency for B455-series and actual designs made in 1982

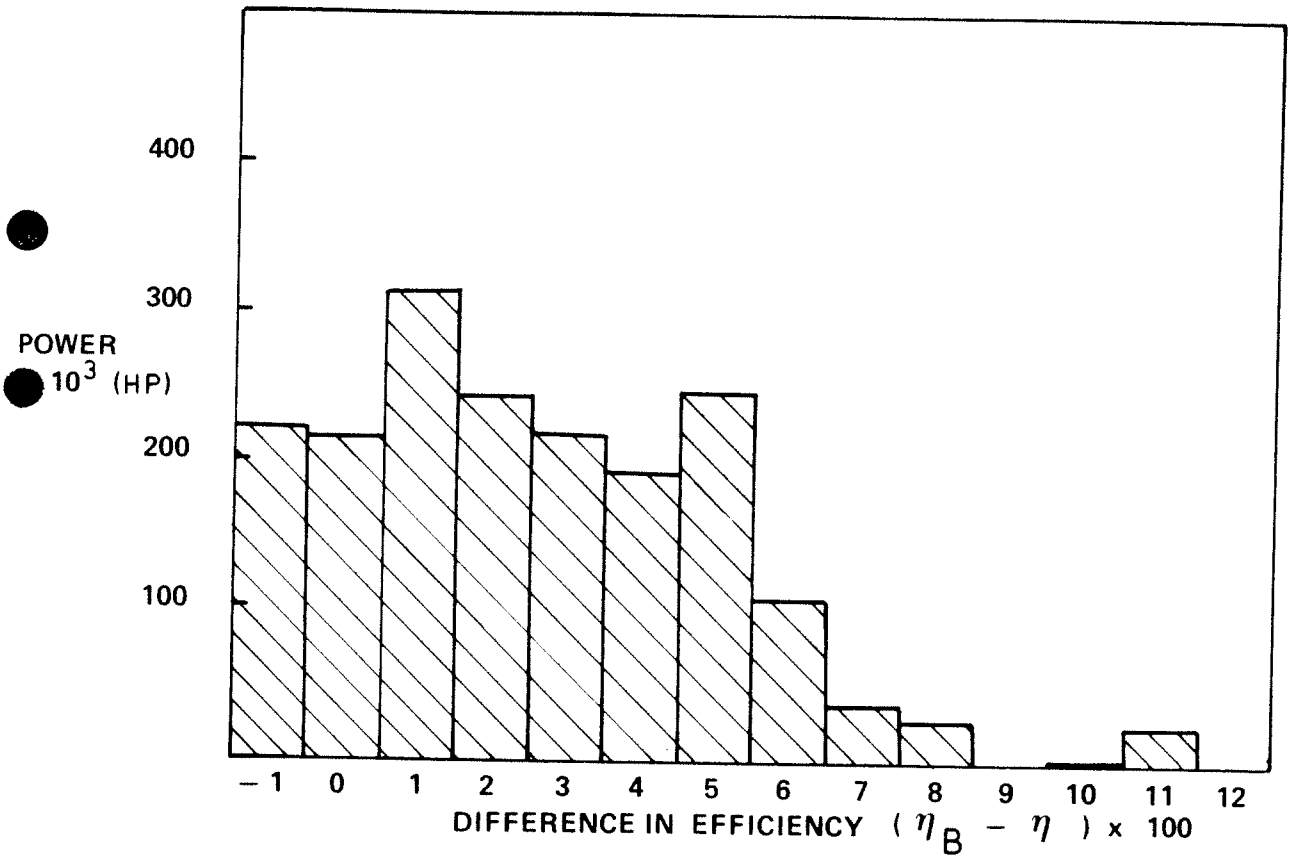


Figure 2 Differences in real efficiency and what could be obtained versus installed horsepower

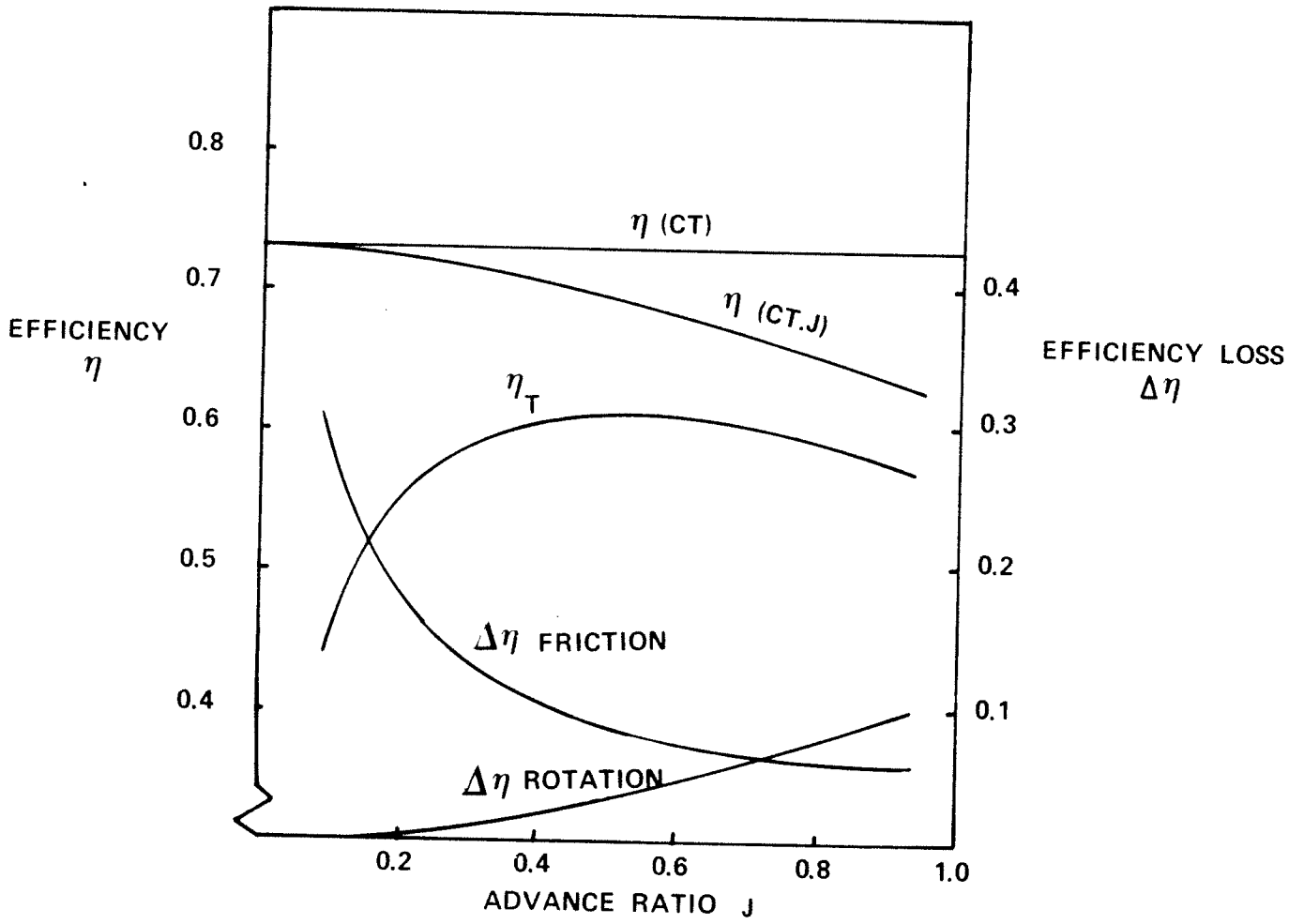
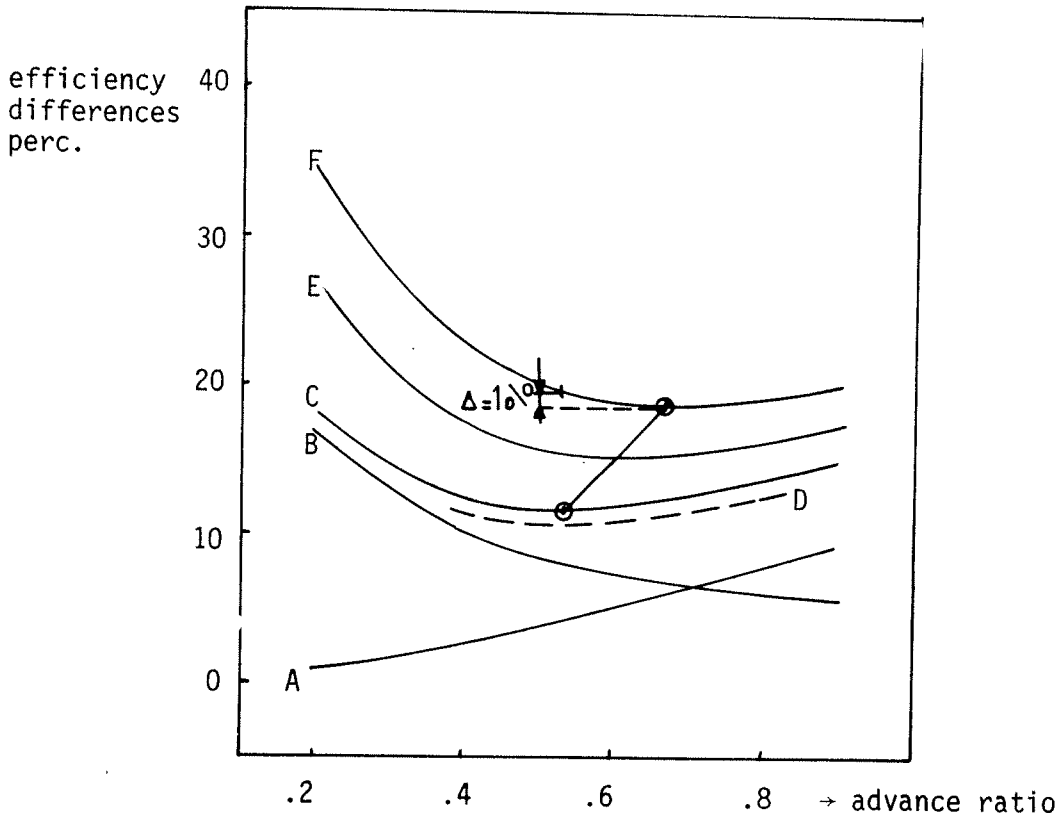


FIG. 3 ROTATIONAL AND FRICTIONAL LOSSES AS FUNCTION OF THE ADVANCE RATIO



A Rotational losses

B Frictional losses

C A + B

D Rotational + frictional losses derived from B4.70 series diagram

E Rotational losses + 1.5 x frictional losses

F Rotational losses + 2 x frictional losses

Fig. 4 Effect of frictional losses on optimum rotational speed for $C_T = 2$.

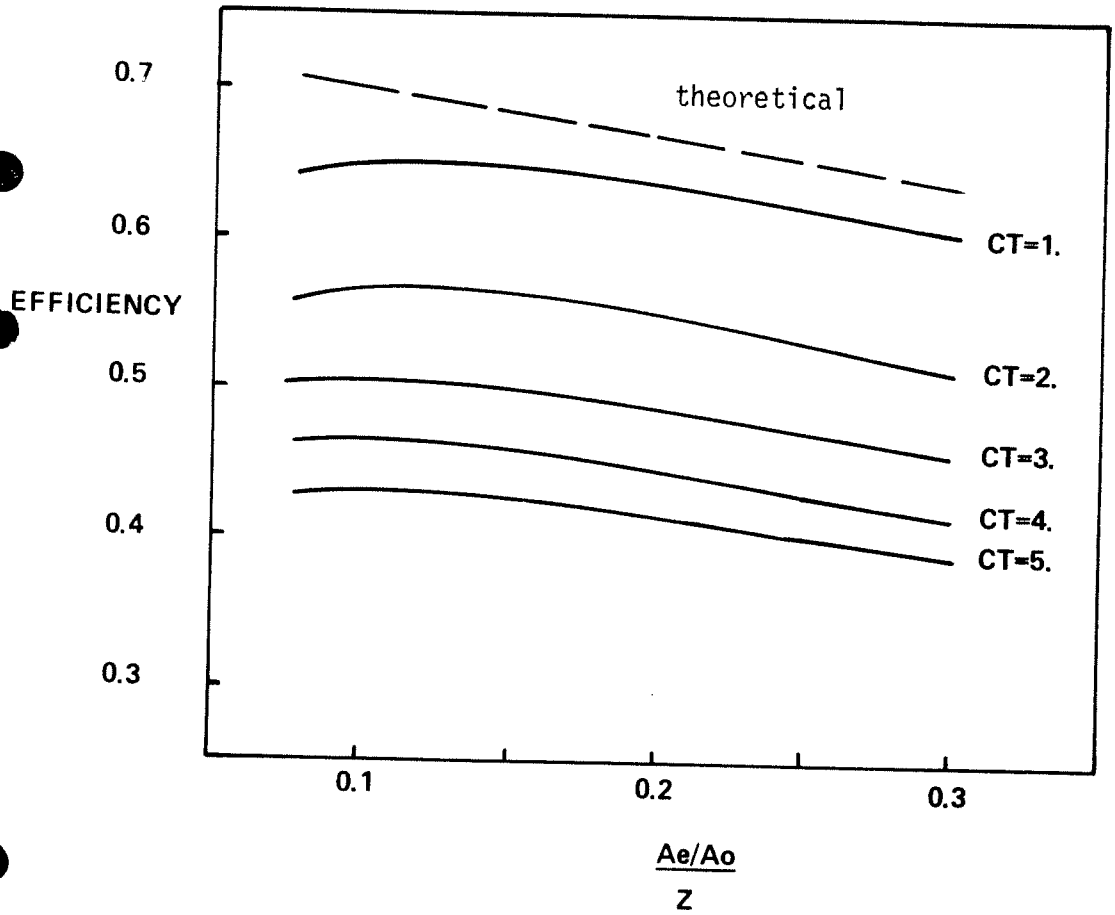
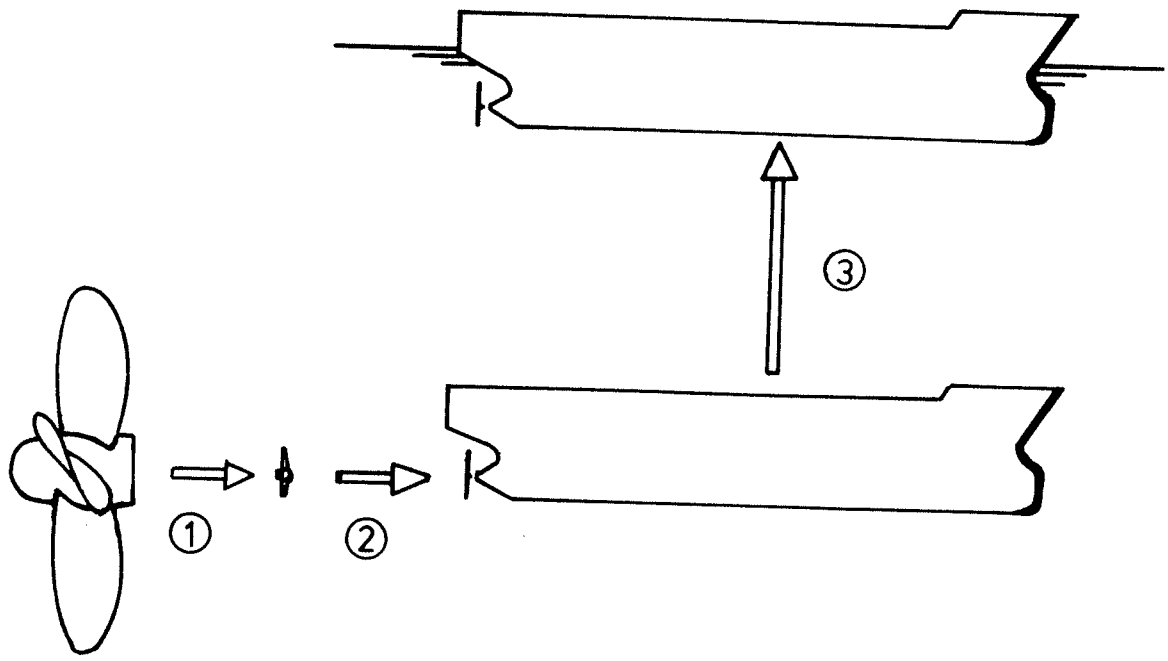
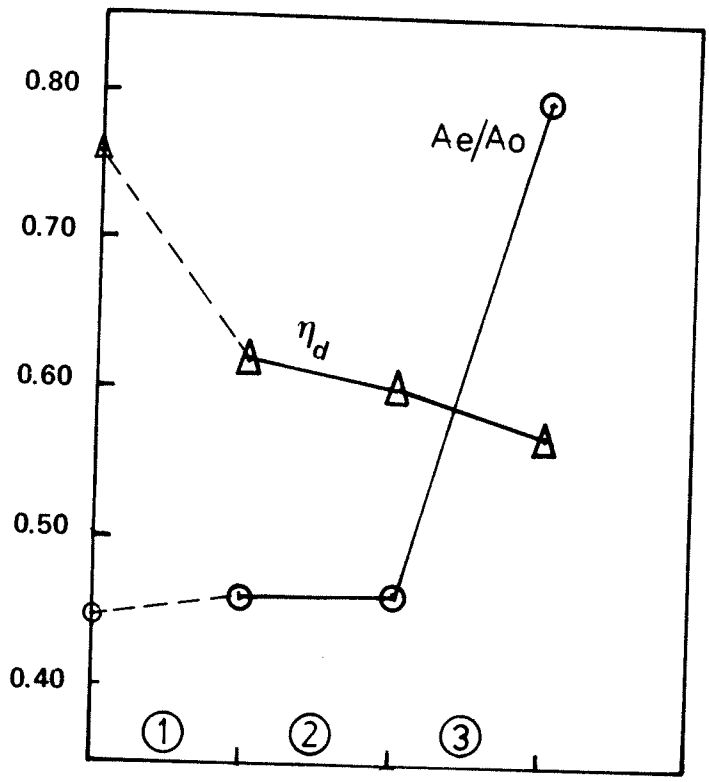


FIG. 5 EFFICIENCY DERIVED FROM B–SERIES PROPELLER



① HOMOGENEOUS FLOW; INFINITE DRAUGHT
② PROPELLER BEHIND SHIP; INFINITE DRAUGHT
③ PROPELLER BEHIND SHIP; FINITE DRAUGHT

FIG.6 PROPELLER DESIGN CONDITIONS



DESIGN CONDITION

DESIGN EXAMPLE FOR

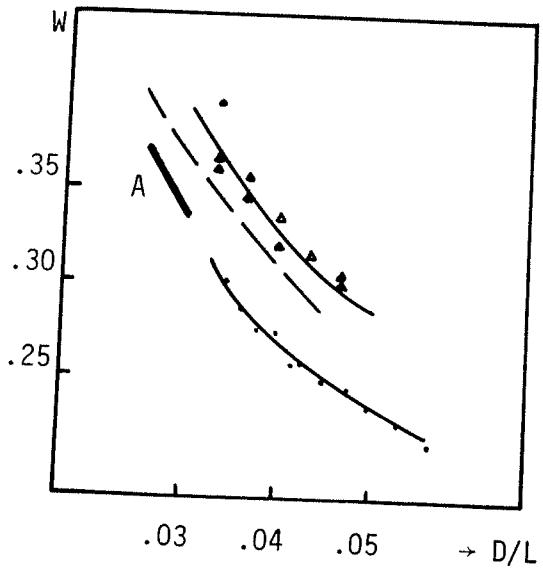
Pe	=	14700	kW
N	=	87.5	RPM
Z	=	5	
V_s	=	18.4	kn

1 UNIFORM FLOW; NO CAVITATION

2 NON UNIFORM FLOW; NO CAVITATION

3 NON UNIFORM FLOW; CAVITATION

FIG.7 INFLUENCE OF NON–UNIFORM FLOW AND CAVITATION RESTRICTIONS UPON TOTAL EFFICIENCY.



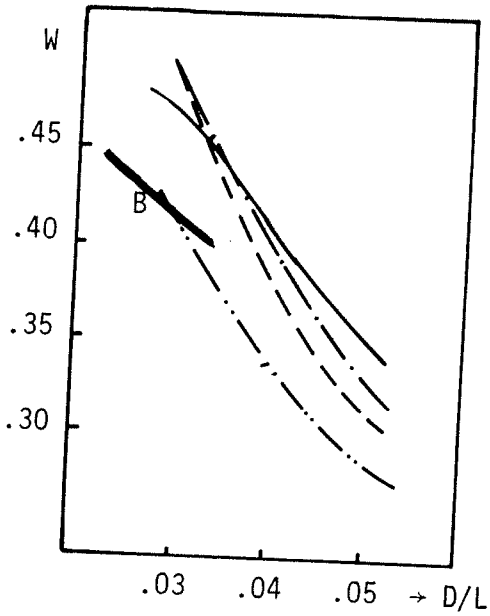


Fig. 8 Change in wake due to diameter increase according ref. [9] including recent experiments (A,B).

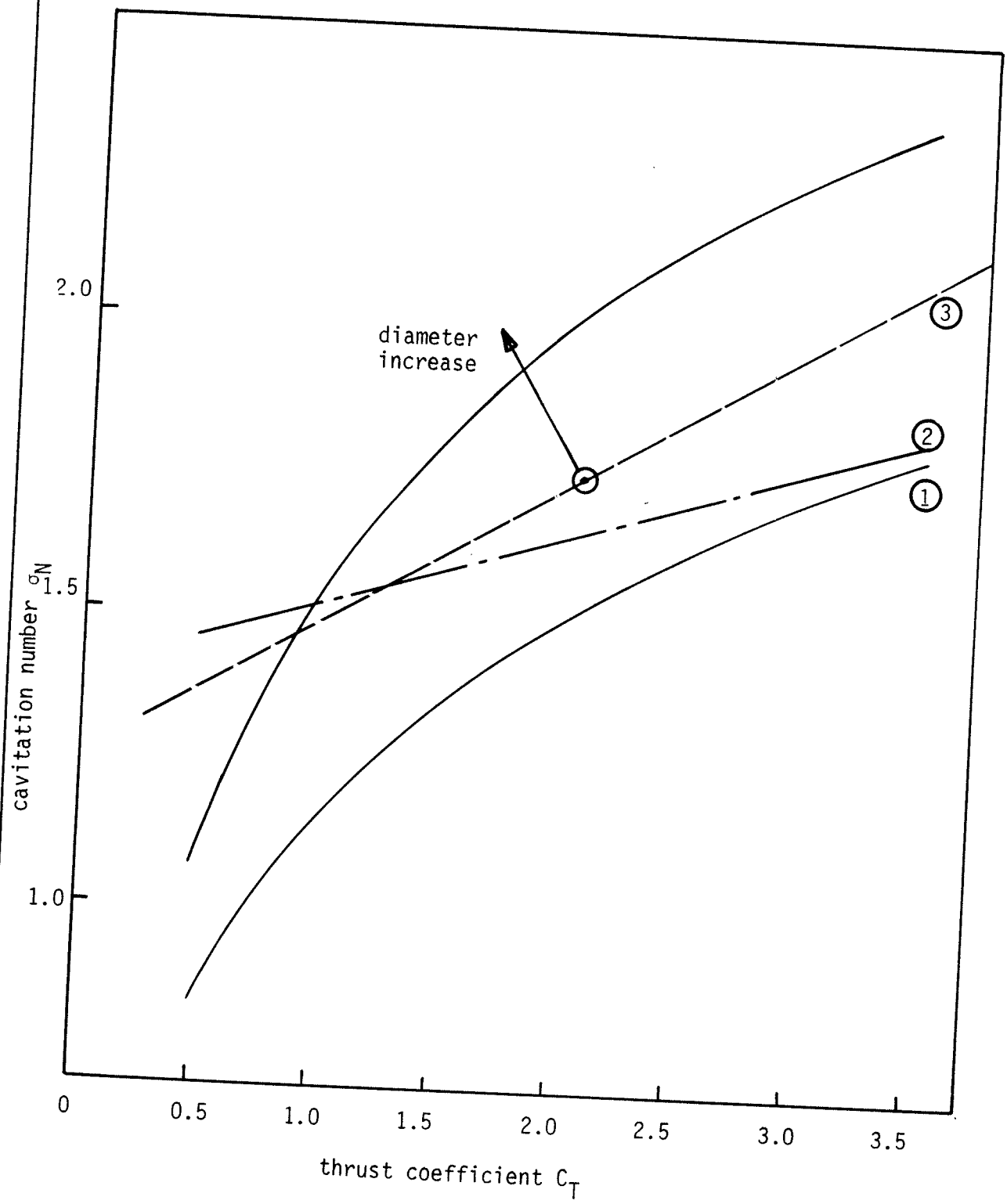
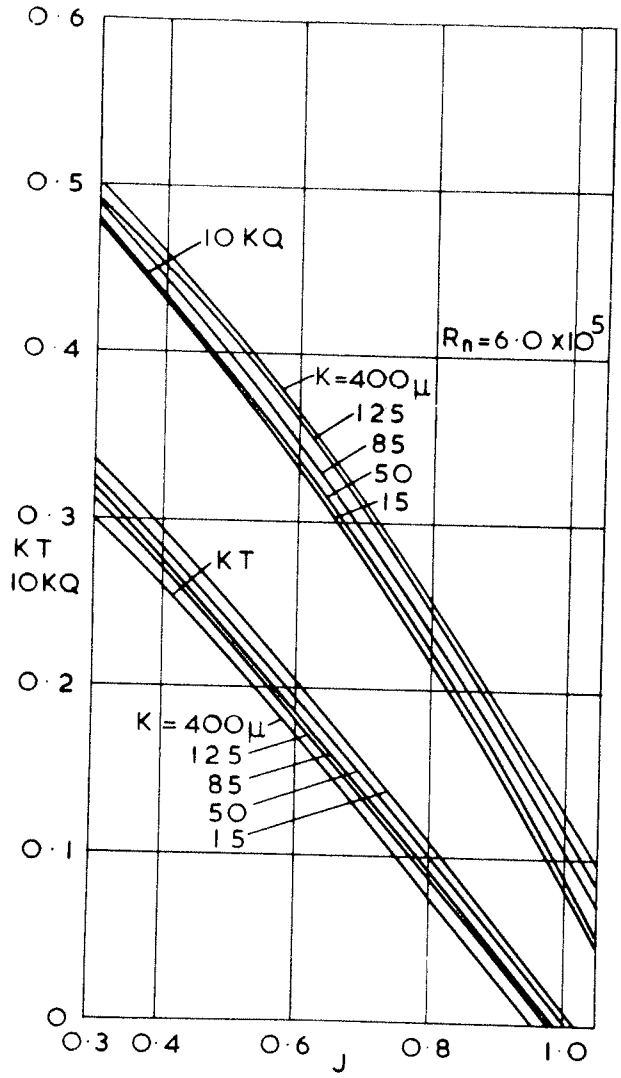


Fig. 9 Various cavitation limits



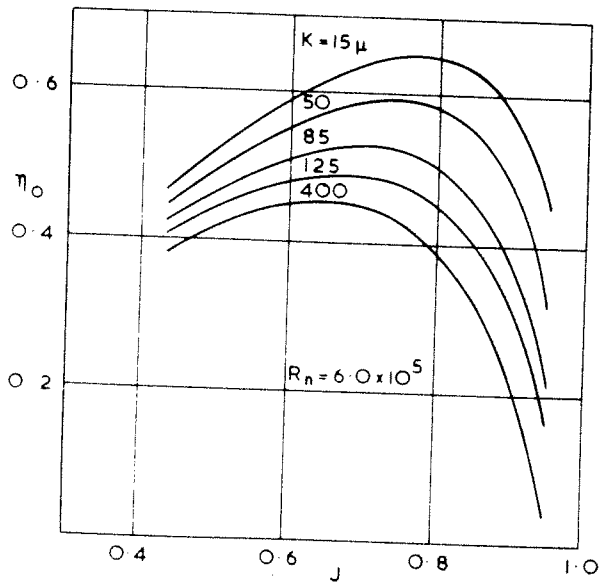


FIG.11 PROPELLER OPEN–WATER EFFICIENCY AS A FUNCTION OF BLADE ROUGHNESS

WEST EUROPEAN GRADUATE EDUCATION
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SHIP DESIGN FOR FUEL ECONOMY

PROPULSION IN SERVICE CONDITIONS
INFLUENCE OF HULL DETORIATION, WAVES, WIND AND STEERING

by

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12) Power Increase due to Loss of Propeller Efficiency

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1) Resistance Components and added Resistance due to Wind and Steering

In this lecture, we will discuss the relative importance of propulsion and different components of added resistance due to waves and wind. In some examples, comparisons will be made with still water resistance. Below are given the definitions and the equations used in the examples. Many simplifications have been made regarding still water resistance but the equations are only meant as a rough reference system for comparison with added resistance. Propulsion in waves can not be treated separately from motions and added resistance due to waves, which therefore have been examined more closely than added resistance due to wind and steering. These effects have been examined in a separate lecture.

In the following, we will have a look on the different resistance components that appears:

If we apply the nondimensional resistance coefficient

$$C_T = \frac{R}{\rho/2\, V_s^2 \cdot S}$$

where

R = resistance

V_s = ship speed

S = wetted surface $\approx (C_B \cdot B + 1.7 \cdot T) \cdot L$

the total resistance of the ship is:

$$C_{TS} = (C_{FS} + \Delta C_{FS})(1+k_0) + C_R + C_{AA} + C_A + C_{TA} + C_{AR} + C_{AW} + C_{AS}$$

where:

$$C_{FS} = \frac{0.075}{(\log R_n - 2)^2}$$ = frictional resistance

ΔC_{FS} = increase in frictional resistance due to roughness

k_0 = form factor

C_R = residual resistance

$$C_{AA} = 0.001 \frac{A_{TS}}{S}$$ = air resistance

A_{TS} = front area of the ship

S = wetted surface

C_{TA} = appendix resistance

where:

$$\log(10^2 \beta + 2.48) = 0.21 \cdot \log(H \cdot V_s) - 0.155$$

H = roughness of the ship in μ (10^{-3} mm) measured at a wave-length of 50 mm

The added resistance due to waves may be divided into two components. One for small wave-lengths ($\lambda/L < 0.5$) where the added resistance is mainly caused by wave reflections from the bow and will be a function of the waterlinearea-fullness. For larger wave-lengths ($\lambda/L > 0.5$), these will be an increase in added resistance coupled to an increase in vertical motions. This part of the resistance is caused by the motions. For larger ships, the first component is the most important one.

The added resistance due to steering may be divided into the following parts:

a) Induced resistance due to drift

b) Induced resistance due to rudder

N. H. Norrbin [25] summarized the main approximate formulas for the added specific resistance as below:

Due to sinusoidal rudder oscillations:

$$\frac{\Delta R'}{\Delta} = \frac{\bar{\Lambda}}{4}\left[1 + \frac{6}{\bar{\Lambda}} + \frac{9}{\bar{\Lambda}^2}\right]\frac{A_r}{L \cdot T} \cdot \frac{L/B}{6} \cdot F_n^{\,2} \cdot \delta_a^{\,2}$$

Due to sinusoidal yawing:

$$\frac{\Delta R''}{\Delta} = 20(1 + X''_{vr})\ \frac{\overline{OP}}{L} \cdot F_n^{\,2} \cdot \left(\frac{\psi_a}{2l/L}\right)^2$$

Due to steady drift or sideslip:

$$\frac{\Delta R'''}{\Delta} = 200\ F_n^{\,4} \cdot \beta^3$$

In these equations, we have the following definitions:

$2l/L$ = yaw period of the ship (2 - 5)

$\bar{\Lambda}$ = rudder aspect ratio (1.5 - 4)

A_r = rudder area

δ_a = helm amplitude (radians)

ψ_a = head deviation amplitude (radians)

heading the wind with a speed of 16 knots, we get an idea of the importance of the steering resistance.

Ship	L	L/B	C_B	B/T	$A_w(m^2)$
I	300	5.5	0.75	2	520
II	105	5.5	0.55	4	182
III	73	5.5	0.55	2	64

Compared to the still water resistance, the wind resistance will be:

Sea state	2	4	6	7	8
V_W (knots)	10	20	30	40	50
$H_{1/3}$ (m)	0.6	2.2	5.0	8.9	13.9
Ship I %	1.4	3.6	6.4	9.9	14.1
Ship II %	6.4	15.8	28.3	43.9	57.7
Ship III %	2.7	6.6	11.9	18.4	26.2

We will later see how large the added resistance due to waves will be for these ships. However, we see that added resistance due to wind at sea state 2 and 6 which are close to Beaufort 3 and 7 are of the same order of magnitude as the added resistance due to steering. If we make a random selection of helm - angles - heading - deviations and Beaufort numbers for different ships, we will find that Beaufort 7 - 8 will give amplitudes of the order: $\psi_a = 8^o$ and $\delta_a = 16^o$. The actual angles experienced at sea are complicated functions of ship and rudder characteristics state of wind and waves, main heading and helm control.

correctly, the Bernoulli equation will give us added resistance due to heave and pitch motions and due to wave reflection from the stern or the bow. If the oscillating pressure given above is integrated over the mean of the wetted surface, we get the mean force.

$$\overline{F_{OX}} = 1/2 \quad \omega^2 \cdot m \cdot z_a \quad \theta_a \cos(\varepsilon_z - \varepsilon_\theta)$$

where

m = the mass of the ship

z_{ma} = the heave amplitude of the center of the mass

θ_a = the pitch angle amplitude

ε_z = phase angle of the heave motion

ε_θ = phase angle of the pitch motion

When the phase difference is

$$/\varepsilon_z - \varepsilon_\theta/ < \frac{\pi}{2}$$

the mean force will be negative. In head sea, this will always be the case. The harmonic oscillating pressure which occurs only in first order theory, was integrated over the mean of the surface. There is also a part of the wetted surface which is time dependent. On this surface there is a time dependent pressure

$$f(x, t) = 1/2\, g\rho \cdot S^2(x, t)$$

where s is the relative motion between the hull and the surface. If we take the mean of this pressure over time, we get:

$$\overline{f(x)} = 1/4 \quad \rho \quad g \cdot S_a^{\,2}(x)$$

where S_a is the amplitude of the relative motion. f(x) will give the following longitudinal force if we integrate over the ship length:

$$\underline{\overline{F_{1x}} = 1/4 \quad \rho \quad g \int_L \cdot S_a^{\,2}(x) \quad \frac{\delta B_{WL}}{\delta x} \, dx}$$

where B_{WL} is the breadth of the ship in the water line area. The total added resistance will then be:

$$\underline{\overline{R_{aW}} = \overline{F_{0X}} + \overline{F_{1X}}}$$

$$S(\omega_e) = S(\omega) \cdot \frac{1}{\sqrt{1 + \frac{4\omega_e \cdot V}{g}}}$$

or

$$S(\omega_e) = S(\omega) \cdot \frac{1}{1 + \frac{2v}{g}\omega}$$

$$T_1 = 24.171 \cdot \left(\frac{4g}{H_{1/3}}\right)^{-0.5} \cdot k$$

(feet-sec)

The wave parameter k is determined from the significant period T_1. The Pierson-Moskowitz sea spectrum is a subset of these two parameter representations ($H_{1/3}$-T_1) corresponding to k = 1. If we consider the $H_{1/3}$-T_1 relation as in fig. (4) from Ström-Tejsen, Yeh and Moran, partially developed seas lie to the left, and dying seas and swell to the right. Using such a two parameter representation for the sea, the added resistance of a ship can similarly be characterized in the $T_{1/3}$-$H_{1/3}$ plane as illustrated in fig. (5) from Ström-Tejsen et al. Fig. (4) and fig. (5) in combination can be useful for a gross judgement of the added resistance performance in the total environment.

Ström-Tejsen et al. used 307 different but actual wave spectra for the North Atlantic and calculated the added resistance for a Series 60 C_B = 0.70 hull form. The predictions were also made for the Bretschneider representation for $H_{1/3}$-T_1 of the actual spectra. The result is shown in fig. (6). It is clearly illustrated that we need the instantaneous and measured spectrum in order to have a reliable background for routing. An unrealistic amount of weather statistics would be required for the actual routes.

Beaufort	3	5	7	9	11
$H_{1/3}$ (m)	1.40	2.15	3.75	6.20	8.40
T_2	5.9	6.5	7.8	9.0	10
Wind speed	9	18	30	44	60

they calculated the average resistance for various seas. See fig. (12). Beaufort 5 may be considered to represent the average weather on the North Atlantic Ocean, Beaufort 7 to represent bad weather and Beaufort 9 extremely bad weather. It is observed that the bulbous bow has higher added resistance in bad weather than the conventional bow.

Sustained speed is limited by added resistance and propulsive efficiency in waves. Fig. (13) from [40] shows sustained speed in various sea conditions for different ship headings. The ship is a Victory ship in the North Atlantic. When the sea conditions worsen, large motions, high accelerations, slamming and propeller racing may force the captain to reduce the speed voluntary or to change course. In the figure, the dashed line indicates the boundary between involuntary and voluntary speed reduction. In particular when head seas are encountered, the speed is limited rather by the ship motions than by lack of power specially for fast cargo ships. We are dealing with involuntary speed reduction due to increased resistance and reduced propulsive efficiency in waves. Fig. (13) may lead to the assumption that the greatest speed loss is experienced in head seas. Model tests do normally give the same result for regular waves when the wave length (ship length ratio is larger than unity as shown on fig. (14) a, b and c. For smaller ratios, there seems to be a maximum at 120°. From [40] we have taken some examples showing the influence of wave direction and ship parameters on required shaft horsepower and speed loss. See fig. (15) and (16).

Model tests in regular waves may be used to predict the performance in various irregular seas. Fig. (15) from [40] shows an example where the speed loss in the North Atlantic Ocean has been calculated in various head seas. It is obvious that a ship with weights concentrated amidships suffer less speed loss as compared to a sister ship with the loads shifted towards the ends of the ship. Steaming against a wind force 5, which is a normal sea condition on the North Atlantic, the speed loss is 0.6 knots for a ship with a radius of gyration of 22% of its length whereas the ship with a radius of 26% will slow down 1.1 knots. Often, however, one has to slow down, not so much because of lack of power but rather as a result of slamming or deck wetness. The significant amplitude of the relative bow motion is shown in the upper part of fig. (15). In order to have an impression of the magnitude of the bow motion, the draught has been indicated. Since the relative motion is smaller for a ship with a smaller radius of gyration, the probability of exceeding the draught (slamming) or the bow height (deck wetness) will also be smaller. Consequently, a ship with a small radius of gyration will be able to maintain speed in adverse weather much better than a ship with a large radius. Decreasing the longitudinal moment of inertia has a favourable effect on the speed loss, either voluntary or involuntary.

$\lambda / L < 0.5$

The Pierson-Moskowitz spectrum has no energy at $\frac{\omega T_2}{2\pi} < 0.5$

This means that:

$$T_2 < 0.9 \sqrt{\frac{L}{g}}$$

We may extend the formula to larger wave periods if:

$$\lambda = \frac{g \cdot T_0^2}{2\pi} < 0.5 \cdot L$$

In the equations above, T_2 is defined as:

$$T_2 = 2\pi \sqrt{\frac{\int_0^\infty S(\omega)\, d\omega}{\int_0^\infty {}^2 S(\omega)\, d\omega}}$$

T_0 is the wave period corresponding to the maximum value of the spectrum. For a Pierson-Moskowitz spectrum T_0, T_1 and T_2 are related through:

$$T_0 = 1.408\, T_2$$

$$T_1 = 1.086\, T_2$$

As an example, we have chosen the following relations between $H_{1/3}$ and T_2:

$H_{1/3}$ (m)	0.427	1.402	2.438	4.572
T_2	2.90	4.60	5.70	7.59

We have calculated the added resistance for 3 ship lengths L = 200, 250 and 300 m for $C_B = 0.65$ and $C_B = 0.80$. The speeds 16 knots and 25 knots were chosen. The result is shown in fig. (47). In the calculations, we used $\overline{\sin^2\theta} = 0.50$ which is a high value. $\overline{\sin^2\theta} = 0.25$ is a more normal value. For the Pierson-Moskowitz spectrum

$$T_0 = 1.408\, T_2$$

which indicates that the calculations only are reliable up to $T_2 = 5.7$ sec for the shortest ship and up to $T_2 = 6.96$ sec for the largest. If we know that Beaufort 5 ($H_{1/3} = 2 - 2.5$ m) represents normal weather in the North Atlantic Ocean and Beaufort 7 ($H_{1/3} = 3.75$) represents bad weather, we see that the equation given above can be used for the majority of the ships crossing the Ocean, the added resistance due to the waves will be 10 - 30% of the still water resistance.

6) The Importance of added Resistance for Energy Conservation

How important is the added resistance compared to the still water resistance? How much power may be saved by reducing the added resistance? Is it possible to reduce the added resistance without increasing the still water resistance? These are very relevant questions. In order to answer the questions, we need weather data for the different routes the ships are going to operate on. We need information on frequency of periods and wave heights from different directions, and we must know the added resistance for different wave height - period - head angle combinations. In the North Atlantic for example, we may apply the following frequency of $H_{1/3}$ during a year:

$H_{1/3}$ (m)	0.6	1.0	1.4	1.7	2.2	3.0	3.8	3.9	6.1
a (%)	3	7	11	18	21	18	10	5	2

If we assume that weather is the same in all directions, the ship will have the sea from $\pm 30^o$, 16.7% of the year. 66.6% of the time the sea will be from 30° to 150°. The rest of the time we will have the following sea from 150° to 180°. If the added resistance is R_{aW0} for head sea, the resistance for side sea and following sea will be:

$$R_{aWS} = k_s \cdot R_{aW0}$$

$$R_{aWf} = k_f \cdot R_{aW0}$$

respectively where k_s and k_f are constants which may be determined from model tests or empirically. The added resistance over one year will then be:

$$R_{aW} = (0.167 + 0.667 \cdot k_s + 0.167 \cdot k_f) \cdot \Sigma \; R_{aW0} (H_{1/3}) \cdot a$$

If we use the examples shown in fig. (47) with $\overline{\sin^2\theta} = 0.343$ and for simplicity assume that $k_s = 0.4$ and $k_f = 0.1$, we get the following increases in added resistance due to waves compared to still water in average over one year:

L = 200 m, V_s = 15 knots

C_B	0.65	0.80
R_{aW}/R_0	10.3%	7.5%

L = 300 m, V_s = 15 knots

C_B	0.65	0.80
R_{aW}/R_0	6.9%	6.2%

savings in power. If we calculate the added resistance due to wind and waves as shown above with data from fig. (48) and (49), we get the following values in the North Atlantic:

Ship	I	II	III
R_a/R_0(%)	5.71	35.71	42.813
$\frac{\Delta V}{V}$ (%)	1.884	11.784	14.128

Aertssen gave the following simple formula for the speed loss in North Atlantic:

$$\frac{\Delta V}{V} \cdot 100 = \frac{490}{L} + 2.64$$

Applying this formula for the ships above, we get:

Ship	I	II	III
$\frac{\Delta V}{V}$ (%)	4.27	7.300	9.173

If we want to study the possibility of reducing the added resistance due to wave reflection from the bow, we must know the frequency of head sea. We must define a sector for head sea. In the examples above, we used a sector equal to 60°. With different sector angles α, the added resistance in head seas will be:

$$R_{aW} = 2.78 \cdot 10^{-3} \cdot \alpha \, \Sigma \, R_{aW0} \cdot a$$

If we take the ships used in our example, we get:

L	C_B	V_s (knots)	$\frac{R_{aW}}{R_0}$ ($\alpha = 20^o$)	$\frac{R_{aW}}{R_0}$ ($\alpha = 40^o$)	$\frac{R_{aW}}{R_0}$ ($\alpha = 60^o$)
200	0.65	15	1.3 %	2.6 %	3.9 %
200	0.80	15	0.9 %	1.9 %	2.7 %
300	0.65	15	0.9 %	1.8 %	2.7 %
300	0.80	15	0.8 %	1.6 %	2.4 %
200	0.65	25	0.5 %	1.0 %	1.5 %
300	0.65	25	0.4 %	0.8 %	1.2 %

These ships have $\overline{\sin^2\theta} = 0.343$. If we reduce $\overline{\sin^2\theta}$ by 50%, we may obtain a reduction between 0.5% and 1% over a year. However, $\overline{\sin^2\theta}$ can not be reduced to such an extent without serious changes in still water resistance. Remembering that it may be misleading to judge the merit of a ship from a standardized representation of the sea condition, it is difficult to make a design which is optimal for the average conditions.

7) Free Surface Effects on Propeller Characteristics

The propeller sets up a steady wave motion when it is working close to the surface. This leads to a reduction in the propeller thrust. This reduction is a function of the propeller loading and the Froude number $V/\sqrt{g\,h}$ where h is the immersion of the the propeller shaft. It will also be influenced by other factors. Let us study a propeller operating in the open water condition: Dickmann [18] showed that the self induced velocity of the propeller in the propeller plane due to its disturbance of the surface could be expressed as:

$$\frac{\bar{\delta}}{V} = 1/2 \cdot \left(\frac{R}{h}\right)^2 \cdot \Omega \cdot \left(\frac{V}{\sqrt{g\,h}}\right) \cdot \frac{U_a^*}{V}$$

where:

$\bar{\delta}$ = self induced velocity averaged over the propeller disk

R = propeller radius

V = advance speed of the propeller

U_a^* = self induced velocity in the slipstream

The potential Ω has the following values:

F_n	0.5	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4	1.5
Ω	0.050	0.030	0.065	0.105	0.142	0.180	0.205	0.210	0.205	0.195	0.185

If we assume that the propeller is lightly loaded and we apply the simple momentum theory, we get the following equations:

$$V + \bar{\delta} + \frac{U_a^*}{2} = 2\pi\, n \cdot r \cdot \mathrm{Arctg}\,\phi$$

for the propeller operating near the free surface and

$$V + \frac{U_a}{2} = 2\,\pi\, n \cdot r \cdot \mathrm{Arctg}\,\phi$$

when the propeller is operating deeply immersed

$P/D = x \cdot \mathrm{tg}\ \phi$ is the pitch of the propeller.

From the equation above:

$$\bar{\delta} = \frac{U_a}{2} - \frac{U_a^*}{2}$$

A typical model propeller has a diameter equal to 0.2 m and is operating at n = 5 - 20 revs./sec. If we assume that h = 0.10 m, we get:

J	0.3	0.5	0.7	0.9
F_n (n=5)	0.303	0.505	0.707	0.909
F_n (n=10)	0.606	1.010	1.414	1.818
F_n (n=20)	1.212	2.020	2.824	3.636

We know from model tests that K_T is approximately 15% lower at J = 0.5 for n = 10 than for n = 5 This is easy to understand if we study the table above and fig. (19). These considerations are only of importance in open water tests. In propulsion tests, the F_n of the ship is:

$$F_n^* = \frac{V_s}{\sqrt{g\,L}}$$

which may be used to express the Froude number of the propeller:

$$F_n = \frac{1-w}{\sqrt{\frac{h}{L}}} \cdot F_n^*$$

which shows that F_n are identical for ship and model if F_n^* and $\frac{h}{L}$ are identical. When referring to an open water diagram, we must be sure that there is no influence from the free surface on the results. At n = 20 and J = 0.3, we get:

h/R	1.0	1.3	1.6	1.9	2.5	3.0
Ω (F_n)	0.210	0.124	0.082	0.058	0.034	0.021
$\frac{\Delta T}{T_0}$	0.174	0.110	0.076	0.055	0.033	0.021

which shows that open water tests normally are considerably influenced by the immersion. Reduced influence from the surface may either be obtained by increasing h/R or decreasing n. But to decrease n is a bad solution due to R_n effects and reduced measuring accuracy.

When h/R > 1.0 the corrections due to wave effects are of particular importance but when h/R < 1, the thrust must be corrected due to ventilation and the emergence of the propeller. The ratio between the immersed disc area A_1 of the propeller and the disc area A_0 of the propeller is:

$$R' = R \sqrt{\beta_0}$$

$$h' = R \left[0.44 + 0.41 (h/R) + 0.16 (h/R)^{2.5} \right]$$

which are used instead of R and h. A comparison between the procedure described above and experimental results by Kempf [12] is presented in the fig. (22). The agreement is certainly too good. Note that we have not taken into account neither partly nor full ventilation which is of particular importance for smaller J = V/n D values than considered in the example. The effect of ventilation is illustrated in fig. (52) where the efficiency of a propeller is given for both the partly and fully developed ventilation as a function of immersion and loading. See also fig. (23) from [20]. If the propeller thrust is changed with immersion, there is also a change in the torque coefficient K_Q. But due to drag force on the propeller, the change in K_Q is not the same as the change in K_T. Model tests by Kempf [12] and Gutsche [20] indicate that

$$K_Q = \beta^m \cdot K_{Q0}$$

where m is constant between 0.80 and 0.85 and K_{Q0} is the propeller torque coefficient for the deeply immersed propeller. This may be explained by increased drag lift ratio as in the following example:

At the equivalent propeller section (x = 0.7), the lift coefficient may roughly be approximated by:

$$C_L = \frac{K_{T0}}{1.5 \cdot EAR}$$

We may also very roughly express β^{1-m} as:

$$\beta^{1-m} = \frac{tg\,(\theta + Arctg \frac{C_D}{C_L})}{tg\,(\theta + Arctg \frac{C_D}{C_L \cdot \beta})}$$

where

$$tg\,\theta = \frac{P/D}{x_0} \approx 0.455 \cdot P/D$$

Let us assume the following values:

$$EAR = 0.65,\ K_{T0} = 0.20,\ P/D = 0.80,\ C_D = 0.008.$$

8) Ventilation and Partly Submerged Propeller.

Ventilated propellers may according to Hadler and Hecker [21] operate in 3 typical conditions:

1. Fully ventilated condition at small advance ratios where the suction side of the propeller is dry also when the blade is in the lowest condition.

2. Partly ventilated propeller at high advance ratios where mainly the emerged propeller is dry.

3. Unstable condition between 1 and 2 with hysterizes .

When a propeller section is fully cavitating, the lift coefficient is:

$$C_L = \frac{\pi}{2}\alpha + \frac{p_0 - p_v}{\rho/2 \; V_\infty^2}$$

where $p_0 - p_v$ is the pressure difference between the static pressure and the pressure in the cavity on the suction side. The pressure in the cavity being p_v. When the propeller is fully ventilated, the suction side is open to the atmosphere. The pressure difference will therefore be:

$$\Delta p = \rho \, g \, h$$

where h is the immersion to the actual section. The lift coefficient is:

$$C_L = \frac{\pi}{2}\alpha + \frac{\rho \; g \, h}{\rho/2 \cdot V^2}$$

It is therefore possible to define a ventilation number analog with the cavitation number:

$$\sigma_V = \frac{2 \, g \, h}{V_\infty^2} = \frac{2}{F_n^2} \cdot \frac{J^2}{J^2 + (\pi x)^2}$$

The problem is to determine when the propeller goes from one condition to another. If the propeller is operating without ventilation at a given advance ratio and we reduce the advance ratio successively, ventilation will occur at some point depending on C_L, σ_V or F_n. As shown in fig. (24), two types of ventilation may occur. In type A we have

$$C_p = \frac{p(x)}{\rho/2\, V^2} = \left(\frac{V_p(x)}{V_\infty}\right)^2 - 1$$

For an arbitrary profile we have:

$$C_p(x) = [\Delta C_L(x) \cdot k_1(x) + C_{Li}(x) \cdot k_2(x) + k_3(x) \cdot t/c(x) + 1]^2 - 1$$

where

$\Delta C_L(x)$ = lift coefficient due to angle of attack

$C_{Li}(x)$ = lift coefficient at chock free entrance

t/c = thickness chord length ratio

k_1, k_2 and k_3 are constants dependent on camber and thickness distribution in the chordwise direction (profile type)

For a NACA a=0.8 profile C_{Li} is given by:

$$C_{Li} = \frac{15.15}{k_c(x)} \cdot f/c\,(x)$$

where

$f/c\,(x)$ = camber chord length ratio

$k_c\,(x)$ = camber correction factor due to 3 dimentional flow

We assume that the profile is NACA 16, a=0.8 and that the ventilation is established to such a degree that it gives a drop in lift. We do also assume that h/R = 1.0 and that we get the drop when the profile at x=0.7 in average over one revolution has the ventilation number:

$$\sigma_v = \frac{\rho\, g\, h}{\rho/2 \cdot V_\infty^2}$$

The drop must occur when σ_v is close to C_p near the leading edge. Experience shows that C_p at x = 0.05 to 0.075 gives results in accordance with the model test results. We have put k_1 = 0.55 and k_3 = 0 and estimated the C_L - C_p relation for different values of C_{Li}. It is observed that large camber/chord length ratios give high C_L values. At x = 0.7, the thrust coefficient is:

$$K_T = 1.5 \cdot EAR \cdot C_L$$

which indicates that a delay in breakdown is obtained by large blade area ratio and high camber. In fig. (25), we have plotted C_L at breakdown for different propellers tested at h/R. It is observed that C_L increases with decreasing EAR. The propellers

9) Open Water Tests in Waves

When the propeller operates in waves, the immersion is periodic. If β and β^m are averaged over one wave cycle, it should be possible to express the open water diagram for the propeller in waves with:

$$K_T = \bar{\beta} \cdot K_{T0}$$
$$K_Q = \bar{\beta}_m \cdot K_{Q0}$$

where the bar indicates average values of β and β^m over one wave period. If we from model tests or theoretical calculations know the relative vertical motion between the propeller shaft centre and the water surface, we can use the same procedure for the propeller operating behind the ship.

Two propellers were tested in open water and in regular waves with a wave length equal to 0.2 m [27]. The diameter was 0.205 and 0.305 m. The immersion to the propeller shaft without the waves was 0.09 m. The K_T, K_Q values in waves were estimated by averaging β and β_m and using the open water diagram for the deeply submerged propeller and alternatively by taking K_T and K_Q values from tests without waves at immersions corresponding to the immersions the propeller experienced over one wave period. The result is shown in fig. (27). It is seen that reasonable results are obtained.

Behind a ship there is a relative motion between the propeller and the surface. We may expect that this motion is proportional to the wave height. The periodic variation of immersion is in regular waves expressed by:

$$h/R = h_0/R + \frac{k \cdot \xi_a \cdot \sin(\omega t)}{R}$$

where k is a constant and ξ_a the wave amplitude. We showed that β could be approximated by:

$$\beta = 1 - 0.675 \, [1 - 0.769 \, (h/R)]^{1.258}$$

If we take the average of $\beta = f(h/R)$, we get the following relation:

$h_0/R = 1.0$

$\frac{k \cdot \xi_a}{R}$	0	0.467	0.933
$\bar{\beta}$	0.893	0.858	0.771

10) Propulsion with restricted Immersion

Edstrand [19], Kempf [12] and Fleischer [28] tested models at different draughts with propellers working more or less immersed. Fleischer discovered that there was a decrease in thrust deduction with reduced immersion. When the effective immersion was reduced from h/R = 1 to h/R = 0.70, t was reduced from 0.245 to 0.220 for one ship and from 0.100 to 0.08 for another ship. The thrust deduction of a ship is given by:

$$t = \frac{2}{1 + \sqrt{1 + C_T}} \quad \frac{w_p}{1 - w}$$

When h/R = 0.70, the effective propeller disk is reduced and C_T will increase. The wake w do normally increase with reduced draft. These two effects may to some extent explain the reduction in t. Ventilation of the same type as for open water was also observed in the behind conditions. The result was reduced propeller efficiency and propeller racing. At NHL, propulsion tests were performed at different draughts. Open water tests at different immersions were also run. During the propulsion tests, the effective propeller immersion was measured. In fig. (28) and (29), the results from the open water and the propulsion tests are shown. The propeller thrust is plotted on the advance ratio of the ship and the propeller respectively. It is observed that the stern wave gives sufficient immersion at the higher speeds (high J values) and the difference in the curves is due to the wake. At smaller speeds it is impossible to define an effective wake which will bring the two curves together. Normally there will be an increase in K_T from the open water diagram to the propulsion diagram equal to approximately:

$$\Delta K_T = 0.425 \cdot w \cdot J$$

at a given J value. With w = 0.4, we get:

J	0.2	0.4	0.6	0.7
ΔK_T	0.034	0.068	0.102	0.119

which seems to bring the two curves together. It is also possible to write:

$$\Delta K_T = \frac{2\ \delta}{U_a} \cdot K_{T0}$$

where δ is due to the interaction between propeller and hull. Putting δ/V equal to 0.145 and $\frac{U_a}{V} = \sqrt{1 + C_T} - 1$, we get:

11) Propulsion in Waves

Some towing tanks apply a tow rope force to compensate for the difference in friction coefficient between ship and model when they do propulsion tests in waves. Others do not. During the tests, the average power, revolution or torque are kept as constant as possible in still water and in different wave conditions. In this way the so called speed loss is obtained by comparing the wave test results with the still water results at constant torque, power or revolutions.

Main engines on full scale ships are in most cases steam turbine or diesel engines. Generally we may assume that the turbine engine has approximately constant power characteristics and the diesel engine constant torque characteristics. See fig. (50). The actual characteristics are more complicated and are intermediate combinations of the two. In propulsion testing, an alternative would therefore be to simulate some engine characteristics or to simulate the way the engine is used in a seaway. If the waves are high compared to the ship length or if the wave length is near to the ship length both ship and model will experience a relative motion at the propeller position which will reduce the effective immersion. The instantaneous immersion of the propeller may be reduced to such an extent that propeller racing due to ventilation occurs. There may be strong fluctuations in RPM, torque and thrust with voluntary and involuntary speed loss as a result. The air drawing is often combined with cavitation which is not discussed in this lecture.

Fig. (30) and (50) illustrate what happens with thrust, torque, RPM speed and power during typical propulsion tests performed in different ways (RPM, Q or P const). It is seen that the speed loss is dependent on the method which is used.

For calculating the increase in power, the direct power method is normally used (O. C. Murday, Proceedings 13 ITTC, Vol. 2).

$$\Delta HP(\omega) = \frac{2\pi}{75} \left[Q_0 + \Delta Q(\omega) \right] \left[n_0 + \Delta n(\omega) \right] - Q_0 \cdot n_0 \cdot \frac{2\pi}{75}$$

The index 0 refers to still water.

For a given spectrum and sea state:

$$\overline{\Delta HP} = \frac{2}{\xi_a^2} \int_0^\infty \Delta HP(\omega) \cdot S(\omega)\, d\omega$$

The increase in revolutions are:

$$\overline{\Delta RPM} = \frac{2}{\xi_a^2} \int_0^\infty \Delta RPM(\omega) \cdot S(\omega) \cdot d\omega$$

This example indicates that the wake reductions observed by Moor and Murday [15] to some extent are imaginary. Moor and Murday observed that the thrust deduction fraction did change with increasing wave height and reached a minimum near λ/L = 1.0 in regular waves, see fig. (35). This change in t was not always followed by a change in effective wake. Nakamura and Naito [22] came to similar conclusions.

Neglecting the wave wake and assuming a uniform sourch distribution over the propeller disk, we have the following simplified expression for the thrust deduction:

$$\Delta T = \rho \pi R^2 \; U_a \; w_p \; V_A$$

where

$$\frac{U_a}{V_A} = \sqrt{1 + C_T} - 1$$

$$C_T = \frac{T}{\rho/2 \; V_A^2 \; \pi \; R^2} = \frac{J_0^2}{2\pi} \; \frac{K_T}{K_Q}$$

w_p = potential wake

The thrust deduction fraction will then be:

$$t = \frac{2}{1 + \sqrt{1 + C_T}} \; \frac{w_p}{1 - w}$$

where

w = effective wake

Fig. (36) shows t for w_p = 0.17. During model testing, the propeller loading may reach C_T values 3 - 4 times as large as in still water as shown in fig. (37). This must lead to a decrease in t of the same order of magnitude as measured during model tests in waves in the most severe conditions. Overload tests in still water give a similar trend. When the added resistance is high, one may not expect to find a linear relation between thrust and resistance increase (see fig. (38)). If necessary, results from overload tests in still water could probably also be applied in waves. As seen from fig. (39), also the wake changes with loading. Moor and Murday [15] and later on Nakamura and Naito [22] showed that η_R sometimes changed in severe wave conditions compared to the still water values. Both increasing and decreasing values were reported. Due to the interference between the propeller and the surface, K_Q and K_T will be reduced compared to the values when the propeller is well immersed. This follows from a study of the open water diagram in the two conditions . We are now discussing the false change in η_R due to the use of an open water diagram for the deeply submerged propeller. If the K_{Q0}, K_{T0} relation is given by:

β , β^m values during propulsion tests will be a function of the instantaneous value of h/R. This function seems to be the same both for constant torque and constant revolution and very similar to the function for open water.

Case a:

Thrust and torque will for a given average speed and revolution vary according to:

$$T(h/R) = [\beta(h/R)] \cdot K_{T0}(J_0)\, \rho [\, n(J_0)]^2 \cdot D^4$$

and

$$Q(h/R) = [\beta\,(h/R)]^m \cdot K_{Q0}(J_0)\, \rho [\, n(J_0)]^2 \cdot D^5$$

The required power will be:

$$HP(h/R) = [\beta\,(h/R)]^m \cdot K_{Q0}(J_0) \cdot \rho [\, n(J_0)]^3 \cdot D^5 \cdot \frac{2\pi}{75}$$

The average values are obtained by averaging $\beta(h/R)$ and $\beta^m(h/R)$ over one period as shown for the open water tests in waves.

Case b:

For given average speed and torque, the revolutions will vary according to:

$$n(h/R) = n(J_0) \cdot [\beta\,(h/R)]^{-0.5m}$$

The thrust will vary as:

$$T(h/R) = [\beta\,(h/R)]^{1-m} \cdot K_{T0}(J_0) \cdot \rho [\, n(J_0)]^2 \cdot D^4$$

The power is:

$$HP(h/R) = [\beta\,(h/R)]^{-0.5m} \cdot K_{Q0}(J_0)\, \rho [\, n(J_0)]^3 \cdot D^5 \cdot \frac{2\pi}{75}$$

Case c:

For given average speed and power, the revolutions will vary according to:

$$n = n(J_0)\ \beta^{-1/3m}$$

The thrust will vary as:

$$T(h/R) = [\beta\,(h/R)]^{1-2/3m} \cdot K_{T0}(J_0) \cdot \rho [\, n(J_0)]^2 \cdot D^4$$

The equations derived above seems to be useful for estimation of propeller characteristics in the behind condition in a seaway. Average values of thrust, torque, horsepower and revolutions are obtained by averaging $[\beta\ (h/R)]^{f(m)}$ over one period where the instantaneous values of h/R and β are estimated from

$$\frac{h}{R} = \frac{h_0 + \Delta h}{R} + \frac{k \cdot \xi_a \ \sin(\omega t)}{R}$$

and

$$\beta = 1 - 0.675\ [\ 1 - 769\ (h/R)\ ]^{1.258}$$

where

Δh = swell up (see fig. (33))

$k \cdot \xi_a$ = amplitude of relative motion at A.P.

To illustrate the relative importance of propeller diameter, immersion and relative motions on propulsion in waves, we have chosen case a) or constant revolutions as example. Let us first assume that a ship at a given speed may operate at 4 different conditions:

1. Ship self propulsion point in still water
2. Ship self propulsion point in waves
3. Model self propulsion point in still water
4. Model self propulsion point in waves with the same absolute increase in thrust due to waves as in 2.

The propeller diagram is given by:

$$K_{T0} = 0.3834 - 0.3543\ J_A - 0.0924\ J_A^2$$

$$K_{Q0} = 0.04679 - 0.03013\ J_A - 0.02154\ J_A^2 \qquad (P/D = 0.79)$$

We have the following propeller loadings compared to the still water at ship self propulsion:

1) Ship self propulsion in still water: (1.0)
2) Ship self propulsion in waves: (1.3)
3) Model self propulsion in still water: (2.0)
4) Model self propulsion in waves: (2.3)
   Thrust increase due to waves: (0.3)

reduced. The resistance of the model and the ship are according to theory and experiments proportional to ξ_a^2. If we study the thrust, we will see that n decreases with increasing ξ_a. This is easy to understand specially if the model is operating without a tow force. For example:

ξ_a/L	0.050	0.025	0.017
C_T	8.651	2.163	1.000
w	0.500	0.500	0.500
w_p	0.150	0.150	0.150
$t = \frac{2}{1+\sqrt{1+C_T}} \frac{w_p}{1-w}$	0.146	0.216	0.249

which gives:

ξ_a/L	0.050	0.025	0.017
$\frac{R(\xi_a/L)}{R(\xi_a/L=0.017)}$ (n=2)	8.651	2.163	1.000
$\frac{T(\xi_a/L)}{T(\xi_a/L=0.017)}$	7.608	2.072	1.000

For the thrust, this corresponds to n = 1.88 which is more in correspondance with the model results.

motion compared to the wave amplitude? The figures (44) and (45) may give an idea but $\sqrt{\overline{S_a^2}}$ is given for irregular sea and as a function of $H_{1/3}$. A Pierson-Moskowitz one parameter spectrum was applied. On fig. (48) and (49), we showed the estimated added resistance for 3 different ships. Using the relative motion data for these ships which had the following characteristics:

Ship	L	L/B	C_b	B/T	V_s	R(m)
I	300	5.5	0.75	2	16	4.00
II	105	5.5	0.55	4	16	2.40
III	73	5.5	0.53	2	16	1.75

we get:

Ship I

$H_{1/3}$ (m)	2.2	5.0	8.9	13.9
$\sqrt{\overline{S_a^2}}$	0.165	0.413	0.818	1.238

Ship II

$H_{1/3}$ (m)	0.6	2.2	5.0	8.9	13.9
$\sqrt{\overline{S_a^2}}$	0.061	0.254	0.516	0.678	0.744

Ship III

$H_{1/3}$ (m)	0.6	2.2	5.0	8.9	13.9
$\sqrt{\overline{S_a^2}}$	0.083	0.551	1.251	1.569	-

These values can not be used directly in connection with fig. (43) which is for regular waves, but they show that $\sqrt{\overline{S_a^2}} \approx 0.400 - 0.500$ when the added resistance is equal to the still water resistance at 16 knots. At instantaneous moments, $\sqrt{\overline{S_a^2}}$ may increase far above these values. It is then obvious that the propeller will come out of the water and draw air. The average relative motions are:

$$\overline{S_a^2} = \frac{1}{2} S_{a1/3}^2$$

$$\overline{S_a} = \frac{\sqrt{2}}{4} S_{a1/3}$$

We do further know that:

$$\overline{S}_{a1/10} = 1.275 \quad \overline{S}_{a1/3}$$

and

$$S_{max} = \overline{S}_{a1/3} \sqrt{\frac{\ln N}{2}}$$

J	0.1	0.2	0.3	0.4
K_{T0}	0.343	0.309	0.269	0.227
K_{TV_0}	0.06	0.06	0.06	0.06
K_{TV} (EAR = 0.6)	0.065	0.079	0.103	0.135
K_{TV} (EAR = 0.8)	0.066	0.085	0.117	0.160

From intersection with the K_T^*, J parabola the actual β values are obtained. Fig. (46) shows that this situation is associated with radical speed losses and propeller racing. It is therefore important to know the exact position of thrust breakdown. Let us illustrate this with an example:

At $h_0/R = 1$, the propeller is operating at $K_T = 0.219$ and $J = 0.360$. Assuming that D = 0.20 mm and n = 10 rev/sec, we get $F_n = 0.727$ which gives:

$$\sigma_v = \frac{2}{F_n^2} \frac{J^2}{J^2 + (\pi x)^2} = 0.099$$

Two of the propellers in fig. (25) have the following K_T^{**} and C_L values:

EAR	0.35	0.80
C_L	0.30	0.15
$K_T^{**} = 1.5 \cdot EAR\ C_L$	0.158	0.180

Both propellers give propeller racing. The propellers were of the Troost type with camber and C_{Li} as given below:

EAR	0.35	0.80
C_{Li}	0.321	0.084
f/c	0.029	0.013

A doubling of f/c is identical to a doubling of C_{Li}. K_T^{**} is estimated for these two cases:

a)	EAR	0.35	0.80
	C_{Li}	0.321	0.084
	K_T^{**}	0.136	0.147
b)	EAR	0.35	0.80
	C_{Li}	0.642	0.168
	K_T^{**}	0.233	0.206

13) Roughness Definitions

The roughness height alone does not give the perfect description of a given roughness profile. The addition of a topographic parameter or a shape factor will improve the description. Byrne [5] suggested to apply the parameter:

$$\alpha = \frac{m_o \cdot m_4}{m_2^2}$$

where m_n is the spectral moment of the spectrum $S(\gamma)$ of the surface. The spectral moment is defined as

$$m_n = \int_0^\infty \gamma^n \cdot S(\gamma) \cdot d\gamma$$

The broadness of the spectrum is characterized by ε, which is defined by:

$$\varepsilon^2 = 1 - \frac{m_2^2}{m_0 \cdot m_4}$$

where $0 \leq \varepsilon \leq 1$. The spectrum is narrow-banded when ε tends to zero. The texture parameter α will then be equal to unity. This corresponds to regular roughness of the surface. If α is large, the spectrum is broad-banded which means that the surface is irregular

Grigson [4] defines the texture parameter as:

$$m = \frac{k \cdot \overline{\Lambda} \cdot U_\tau}{\upsilon}$$

where:

k = constant

$\overline{\Lambda}$ = the average wave-length of the Fourier spectra

$U_\tau = \sqrt{\tau_o/\rho}$ friction velocity

m is determined from experiments with ship plates towed in a model tank or hull replicas tested in pipes.

Sasajima and Yoshida [29] define the parameter p which is a function of the maximum ratio between wave-height H and wave-length λ when H is measured at different wave-lengths. The $p\ (H/\lambda)_{max}$ is shown in fig. (54). In fig. (55), some examples of such measurements are shown. When p has been determined, ΔC_F can be determined. See fig. (65) where β is equal to 3.

$$\bar{H} = \sqrt{2} \cdot \sqrt{m_o}$$

and

$$\bar{H}_{1/3} = 4 \cdot \sqrt{m_o}$$

where:

$\bar{H}$ = average wave height

$\bar{H}_{1/3}$ = significant wave height

$\sqrt{m_o}$ = standard deviation

Using the expression given above and Karlsson's roughnesses ($\bar{H}$), we get a plot as shown in fig. (56) for a 6 feet long plank with $R_n = 3.16 \cdot 10^7$. Todd [32] towed planks of different roughness in a towing tank and measured ΔC_F. The plank was 6 feet long. Fig. (56) shows that his results corresponds very well with Karlsson's results. If Todd and Karlsson's results are plotted in fig. (66), we get a curve which corresponds to curve a.

This means that both Karlsson's, Todd's and Grigson's results may be approximated by the relation:

$$\underline{\log (C_F + \Delta C_F) = 0.21 \cdot \log \left(\frac{\bar{H}(50) \cdot V_s}{\upsilon}\right) - 0.153}$$

at $R_n = 3.16 \cdot 10^7$ ($\log R_n = 1.5$).

But it must not be forgotten that a given value of $\bar{H}$ may give different ΔC_F values depending on the texture of the surface. For a given surface with the velocity V_s and the roughness $\bar{H}(50)$, ΔC_F is estimated as shown above. Extrapolation to full scale is done by:

$$\Delta C_F (R_n) = \Delta C_F (\log R_n = 7.5) \left[\frac{C_F (R_n)}{C_F(\log R_n = 7.5)}\right]^n$$

where $\underline{n = 1.85 - 2.00}$

This equation is also valid for sand roughness. According to Schlichting-Prandtl [6], the following formulation may be used for such roughness:

$$(C_F + \Delta C_F) = [1.89 + 1.62 \cdot \log (\frac{c}{k_s})]^{-2.5}$$

With the Schlichting formula for smooth surface

14) Propeller Surfaces

Taniguchi and Sasajima [34] investigated the relation between the roughness and the frictional resistance of surfaces equal to propeller surfaces. The roughness $\bar{H}$ was measured according to J.I.S. (B 0601 - 1970) and the following roughnesses investigated experimentally 1.9μ, 14μ and 47μ on a plank with a length equal to 3.5 m towed in a model basin. They concluded that the Prandtl-Schlichting formula could be used and that the relation between the roughness $\bar{H}$ and the equivalent sand roughness followed the formula:

$$k_s = 2\bar{H}$$

According to Schlichting, the flow is smooth when:

$$\frac{k_s}{c} < \frac{100}{R_n}$$

where R_n is the Reynolds number of the section based on the chord length c. The chord length will normally be equal to 2 - 2.5 m, which means that turbulent flow should be obtained for:

$$\bar{H} < 1.5 - 2.5\mu$$

if $R_n = 5.3 - 6.7 \cdot 10^7$

Grigson [9] calculated the drag increase on propeller surfaces due to roughness. He concluded that the nature of the roughness was Colebrook-White [8] and that m had to be less than 20 and not worse or smaller than 1.95 (or sand roughness in the worst case). We do still lack information on how the roughness on propellers develop with time. When the propeller is new, we may find roughnesses down to 4 - 5μ. If the propeller surface is fouled, the roughness may reach values as high as 200μ. Some investigators indicate roughness increases of the order of 50 - 60μ without any other specifications. Others indicate 50 - 70μ after 2 or 3 years in service. Fig. (68) gives some examples of roughness measurements on ships that have been in service for some time.

Yokoo [35] investigated the roughness increase on propeller blades. His results are shown on fig. (55). If we apply the wave-length λ = 10 mm, it seems that $\bar{H}$ is approximately 20μ, 50μ and 150μ. If $(H/\lambda)_{max}$ is plotted on $H(1/\lambda = 0.1)$ for the hull and the propeller surfaces, fig. (67), two types of surfaces appear, one which is soft and wavy and one which is sharp and "hilly". See fig. (67).

15) Changes in Propeller Characteristics due to Roughness

It has for a long time been generally accepted that an increase in propeller roughness is followed by an increase in torque and a decrease in thrust. Model propellers tested with different roughnesses show the same. Analyses of the results indicate that the changes in propeller characteristics is not only caused by increased drag, there is also a decrease in lift as shown in fig. (69). The lift and drag coefficients have been derived from the model test results applying Lerbs' method [7] .

If the change in drag and lift due to propeller roughness is known, it is possible to estimate the influence on the propeller characteristics: The efficiency at each propeller section is expressed as:

$$\eta(x) = \frac{d K_T}{d K_Q} \cdot \frac{J}{2\pi} = \frac{J}{\pi x \, tg (\beta_i + Arctg \frac{C_D}{C_L})}$$

By the assumption of a certain thrust distribution, a factor m which depends on x can be determined in such a way that at any radius x, the product of the quantity $\frac{d K_T}{dx}$, at this radius and m(x) equals the total thrust coefficient:

$$K_T = \int_{x_h}^{1} \frac{d K_T}{dx} \cdot dx = \left(\frac{d K_T}{dx} \right)_x \cdot m(x)$$

Therefore at any fixed value of x:

$$C_L(x) = \frac{4}{\pi^2 \cdot m(x) \cdot z} \quad \frac{K_T}{C/D(x)} \quad \frac{\sqrt{1 + (\frac{1}{\pi x})^2 \cdot (\frac{J}{\eta(x)})^2}}{x^2 + (\frac{J}{\pi})^2}$$

In this expression, we have the following definitions:

z = number of blades

C = chord length of the profile

x = r/R

J = $V_a/n D$

$K_T = \frac{T}{\rho n^2 D^4}$

$K_Q = \frac{Q}{\rho n^2 D^4}$

C_D = drag coefficient

The equivalent radius is obtained from:

$$\eta_0 = \eta(x)$$

For different combinations of α, n, a and b, x_0 and m(x) have been estimated. See fig. (71). The calculations indicate that the equivalent radius may vary from x = 0.55 to 0.70 depending on the load distribution. $m(x_0)$ varies from 0.50 to 0.60. If we as Lerbs, assume that:

$$m(x) = \frac{\pi}{16} \frac{1}{x^2 \sqrt{1 - x^2}}$$

we get:

$x_0 = 70$ $\qquad$ $x_0 = 0.75$

$m_0 = 0.561$ $\qquad$ $m_0 = 0.528$

It is now possible to estimate the change in K_T and K_Q if the drag coefficient changes. From the expression for C_L:

$$\Delta K_T = \frac{m(x_0)}{0.4053} \left[x_0^2 + \left(\frac{J}{\pi}\right)^2 \right] \frac{tg\,\beta_i}{\sqrt{1 + tg^2 \beta_i}} \quad \frac{C}{D}(x_0) \cdot z \cdot C_D$$

and

$$\Delta K_Q = \frac{m(x_0)}{0.4053} \frac{\left[x_0^2 + \left(\frac{J}{\pi}\right)^2 \right] x_0}{\sqrt{1 + tg^2 \beta_i} \cdot 2} \quad \frac{C}{D}(x_0) \quad z \quad C_D$$

If we apply the following mean values:

$$x_0 = 0.70, \quad J = 0.30, \quad tg\,\beta = \frac{J}{\pi x} = 0.136$$

$$\beta_i = \frac{tg\beta}{tg\beta_i} = 0.60 \quad \text{and} \quad P/D = \pi x_0 \cdot tg\,\beta_i$$

we obtain:

$$\Delta K_T = 0.306 \cdot \frac{C}{D}(x_0) \cdot z \cdot P/D \cdot \Delta C_D$$

$$\Delta K_Q = 0.236 \cdot \frac{C}{D}(x_0) \cdot z \cdot \Delta C_D$$

---

which are almost identical to the expressions used in the ITTC program (36).

Due to changes in C_D, the lift coefficient will also change as indicated in fig. (69).

Examples:

The drag of the propeller section may be expressed as:

$$C_D = 2\ [\ 1 + (\frac{t}{c})\ ] \cdot C_F\ (R_n)$$

The R_n of the section is:

$$R_n = \frac{n\ D\ C}{\upsilon} \sqrt{J^2 + (\pi\ x_0)^2}$$

where C is approximated by:

$$\underline{C = 2.073 \cdot EAR \cdot \frac{D}{z}}$$

The inflow velocity to the profile is:

$$V_\infty = \sqrt{(2\pi\ r\ n)^2 + (J\ n\ D)^2}$$

We apply the propeller used as example in "Propeller Surfaces" and the equations developed for correction of the propeller diagram. It is assumed that the propeller surfaces in fig. (55) is used.

We then get:

Propeller Surface	New	1 year	5 years
$10^3 \cdot \Delta C_D$	0.560	1.660	2.260
ΔK_T	0.0002	0.0007	0.0009
ΔK_Q	0.0002	0.0006	0.0008

For J = 0.30, we get:

η_0 (smooth)	0.338	0.338	0.338
K_{T0} (smooth)	- 0.313	-0.313	-0.313
K_{Q0} (smooth)	0.0442	0.0442	0.0442
η (rough)	0.339	0.333	0.331

For J = 0.40:

η (smooth)	0.441	0.441	0.441
K_{T0} (smooth)	-0.265	-0.265	-0.265
K_{Q0} (smooth)	0.0382	0.0382	0.0382
η (rough)	0.439	0.434	0.431

16) Hull Surfaces

Several studies on hull roughness and its influence on the resistance are available. Both experimental and theoretical approaches have been tried. Resistance and roughness have been measured in laboratories on replica of the ship surfaces. Roughness development and speed loss have been measured on ships in service.

On fig. (62), the increase in roughness for some ships with time is shown. The curve is taken from Townsin et al. [37]. The increase in roughness is dependent on the number of recoatings and type of coating, the damage frequency etc. It is observed that a small number of ships increased much more rapidly in roughness than the others (curve b) and that one group was maintained with little loss of smoothness. This curve (c) is a very important finding since it shows that it is possible to maintain bottoms in a virginal state. Probably this curve represents something close to an optimum. If the ship is sand blasted and recoated, the roughness is decreased considerable as shown in fig. (76). In general ships, built the last 5 - 7 years do not have such a drastical trend as shown on fig. (76). With the new coatings used today, one may expect small changes with time. One type of such coatings is the reactivated antifouling compositions where the steel plate first is covered by a primer and the primer with an antifouling which releases the poison from the surface of the film. As the top layer becomes depleted of poison, the leaching rate decreases until it finally falls below the rate necessary to prevent fouling. Provided the film's thickness is great enough, there may still be large quantities of poison left beneath the non-toxic top layer which consists of the porous and rough binder. If this layer is removed, the leaching rate increases to practically the original value. The removing is made by removing the inactive layer by use of specially designed brusher. By reactivisation, the formation of sandwich coating and bubble formation are omitted. Roughness measurements have shown that the surface has obtained a decrease in roughness with the brushing.

In the last years, the so called self polishing antifoulings have been developed where the poison and the binder form a unity. In water, the poison will desolve into the water of a constant rate. Due to the turbulence and the friction, the water will polish the surface and we may experience a decrease in roughness with time. This type of antifouling can only be applied in primers of supreme quality and when the plate is sand blasted and is not recommended on old systems or ships. Fig. (77) shows the principle for this system.

Self polishing and reactivatable systems have been in use for 5 - 7 years and the experience so far is good. They are probably the only systems that can keep the ship out of dock for more than 2 years without any risk of fouling. Fig. (78) shows the time history for a typical self polishing surface. It is indicated that there will be a decrease

Plate	7	9	10	12	14
H(33 mm)	900	120	150	400	600
ΔC_F (Sasajima)	0.55	0.02	0.03	0.20	0.42
ΔC_F (Todd-Karlsson)	0.66	0.16	0.21	0.43	0.54
$\Delta (\Delta C_F)$	0.11	0.14	0.18	0.23	0.12

(The deviations between the two methods is probably caused by different ways of measuring and defining the roughness).

In fig. (78), we defined a permanent roughness which would increase with the years depending on how the vessel is protected from corosion and how it is treated when drydocking. In addition, we have damages. In special periods and in special waters and harbours we may observe severe fouling both on hull and propeller. It is difficult to measure the roughness increase due to fouling, but the "equivalent" roughness may reach values as high as several thousands . Though scrubbing and antifoulings cures a lot further savings are possible. Increase in power after 20 - 30 months corresponding to changes in $\Delta C_F = 0.50 \cdot 10^{-3}$ have frequently been reported. As seen in fig. (86), this corresponds to equivalent roughness increases of approximately $\Delta H(50) = 600\,\mu$ - $700\,\mu$ if we assume that the speed loss is due to hull fouling. The situation is considerably improved by scrubbing the hull regularly between the drydock cycles, for example once every voyage. The influence of the fouling on an american war ship [33] is shown in fig. (87). The increase in power at constant speed is about 50%. If we study fig. (85), we understand that this will correspond to $\Delta C_F = 1 \cdot 10^{-3}$ which is a high value. After having obtained the maximum degree of fouling, the ship was cleaned at different places on the hull until the entire hull was clean again. Fig. (88) shows the importance of cleaning the different parts of the hull. It is interesting to see that the propeller in this case contributed with 32% of the total resistance increase. If we subtract the dome and the propeller, we get an increase in ΔC_F equal to $0.43 \cdot 10^{-3}$ after 39 months. If we subtract the dome from the total resistance, the propeller is responsible for 50% of the total power increase. Without the dome, the propeller would have given approximately 18.75% power increase which corresponds to an equivalent sand roughness of $k_s = 1000\,\mu$ - $1500\,\mu$.

The optimum interval between cleanings and dockings depends upon many factors. These include the regrowth rate of fouling between cleanings, vessel size, fuel prices etc. For example as seen from fig. (82), larger ships will suffer more from roughness increase and fouling than smaller ships. It is evident that constant control with the roughness and fouling development is important during the service period of the ship. There are several systems available for such control. Power and ship speed is measured as function of time. The weak point with such measurements is the speed and to distinguish between increase in roughness on hull and propeller. The Norwegian Ship Research Institute has some experience with such measurements. Some examples from [38] are shown in fig. (89) a, b and c. Fig. (89) shows the development for a 22000 tdw tanker. During docking, the hull was sand blasted and covered with conventional antifouling. The roughness before and after the docking was $600\,\mu$ and 250μ respectively. The docking resulted in a speed gain of 0.8 knots. After 8 - 9 months, the influence of fouling could be registrated and after 12 months approximately, the speed was the same as before the docking. Fig. (90) shows the development for a sister ship. The ship was during the docking sand blasted and covered with self polishing paint. The roughness before and after the docking was 500 and 270μ . The speed was increased by 0.4 knots. Some time later, the speed seemed to have improved by 0.2 knots, more

$$K_T = \frac{V_s^2 \, [\,(1+k)(C_F + \Delta C_F + \Delta(\Delta C_F)) + C_R\,]}{2(1-t)\, n^2 \cdot D^4}$$

If we experience a change in K_T, it must be due to $\Delta(\Delta C_F)$. If such measurements are done regularly over a long period it is possible to follow the roughness development.

Torque and speed measurements may be inaccurate. Thrust measurements are possible but still more inaccurate. We must therefore normally rely on K_Q measurements. When K_Q has been measured, we enter the diagram with K_Q and get K_T^* from which $\Delta(\Delta C_F)$ is estimated. But $\Delta(\Delta C_F)$ or K_T^* will be too high if the propeller roughness has increased. There is one way to avoid this problem: We assume that the scale effect on the wake is given by:

$$\Delta(\Delta w) = 0.04 + t + (w_m - 0.4 - t) \; \frac{C_{FS} + \Delta(\Delta C_F)}{C_{Fm}}$$

From $\Delta(\Delta C_F)$, $\Delta(\Delta w)$ is estimated, which gives:

$$J = \frac{V_s \, [\,1 - w_m - \Delta w - \Delta(\Delta w)\,]}{n \; D}$$

and the propeller efficiency:

$$\eta_0 = \frac{J}{2\pi} \; \frac{K_T}{K_Q}$$

The power of the ship is:

$$HP = \frac{1-w}{1-t} \cdot \frac{\rho \; S \; V_s^3}{150 \; \eta_0 \; \eta_R} \; [\,(1+k)(C_F + \Delta C_F + \Delta(\Delta C_F) + C_R\,]$$

For each arbitrary chosen value of $\Delta(\Delta C_F)$ there are corresponding $\Delta(\Delta w)$, J, η_0 and HP values. When HP is equal to the power measured on the ship, we have the correct value for $\Delta(\Delta C_F)$ and K_T. Entering the propeller diagram with K_T and K_Q we have the correct ΔC_D value. In this way it should be possible to distinguish between the two contributions and take the right decisions before docking and to choose the right time for cleaning of propeller and/or hull.

13. Yamazaki, R. "On the Theory of Screw Propellers in Non-uniform Flows," Memoirs of the Faculty of Engineering, Kyushu University, Vol. XXV, No. 2-1966

14. Johnsson, C. A. "On Theoretical Predictions of Characteristics and Cavitation Properties of Propellers", Publications of the Swedish State Shipbuilding Experimental Tank, Nr. 64. Gøteborg 1968

15. Moor, D. J. and Murdey, D. E. "Motions and Propulsion of single Screw Models in head Seas, Part II,". TINA. Vol. 112, 1970

16. Boese, P. "Eine einfache Metode zur Berechnung der Widerstandserhöung eines Schiffes im Seegang", Ifs-Bericht nr. 258, Febr. 1970

17. Fujii, H. and Takahashi, T. "On the Increase of a Ship in Regular Head Sea", Mitsubishi Heavy Industries Technical Review, Vol. 7 No. 2 (Jan. 1967) p.86 (in Japanese)

18. Dickmann, E. Wechselwirkung zwischen Propeller und Schiff unter besonderer Berücksichtigung des Welleneinflusses, Jahrb. STG 40 (1939), S. 234/282

19. Edstrand, H. Om Djupgåendets Inflytande på Propulsionsegenskaperna, SSPA - Berich Nr. 21, Göteborg 1952

20. Gutsche, F. Einfluss der Tauchung auf Schub und Wirkungsgrad von Schiffspropellern, Schiffbauforschung 5/6 (1967), S. 256/277

21. Hadler, J. B. and Hecker, R. Performance of partially submerged Propellers, 7th ONR Symposium on Naval Hydrodynamics, Rome 1968

22. Nakamura, S. and Naito, S. "Propulsive Performance of a Container Ship in Waves", The Society of Naval Architects of Japan, Naval Architecture and Ocean Engineering, Vol. 15, 1977

23. Ström-Tejsen, J., Yen, H. Y. H. and Moran, D. D. "Added resistance in waves", SNAME Transaction 1973

24. Faltinsen. O, Minsaas, K, Liapis, N. and Skjördal, S. O. "Prediction of Resistance and Propulsion of a Ship in a Seaway", Thirteenth Symposium on Naval Hydrodynamics, 1980, Tokyo

38. Brembo, J. C. "Bruk av Tilstandskontrollsystemet "Predikt" til Energiøkonomisering og Driftsoptimalisering for seilende Skip",
NSFI, Trondheim, January 1982

39. Ewing, J. A. "The Effect of Speed, Forebody Shape and Weight Distribution on Ship Motions",
Trans RINA 1967, London

40. van Sluijs and Ir Tan Seng Gie "Retrospection on 15 Years NSMB Seakeeping Activities"
International Jubilee Meeting on the Occasion of the 40th Anniversary of the Netherlands Ship Model Basin, August 30 - September 1, 1972

41. Wahab, R. "Research on Bulbous Bow Ships, the Behaviour of a fast Cargo Liner with a conventional and with a bulbous Bow in a Seaway",
International Shipbuilding Progress Vol. 13, 1966 - No. 142

42. Rinvoll, A. "Surface Treatment and Painting of Hulls. The Importance of keeping Ship's Bottoms smooth",
NSN No. 9, 1982

43. Loukakis, T. A., Chryssostomidis, C. "Seakeeping Standard Series for Cruiser - Stern Ships"
Soc. NAME. Trans vol. 83, 1975

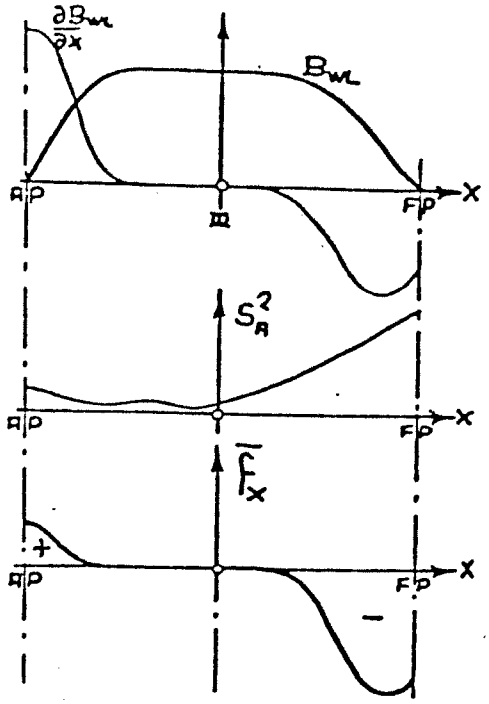
44. Swann, W. A. and Vossers, G. "The Effect of Forebody section Shape on Ship Behaviour in Waves",
Trans RINA, Vol. 103, 1961

45. Faltinsen, O. "Forelesningskompendium for Fag 81527". Sjøbelastninger og Bevegelser av Skip
Trondheim, januar 1982. Institutt for Marin Hydrodynamikk, NTH

46. Nakamura, S., Fujii, H. "Nominal Speed Loss of Ships in Waves",
Prads International Symposium on practical Design in Shipbuilding. Tokyo Oct. 1977

47. Joubert, P. N., Hoffman, P. H. "An Experimental Study on the viscous Resistance of a 0.564 - C_B Form",
Journal of Ship Research June 1979, Vol. 23, No. 2

Added Resistance due to Waves and Integration of Pressure along the Hull



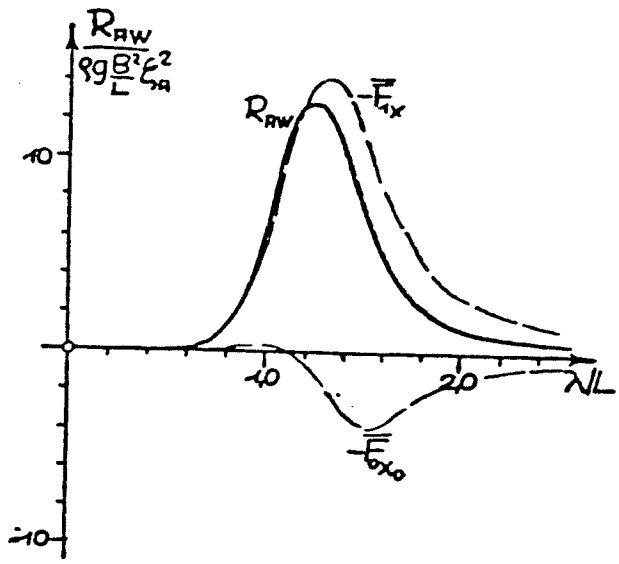


Fig.(1)
from Boese

Added Resistance due to regular Waves

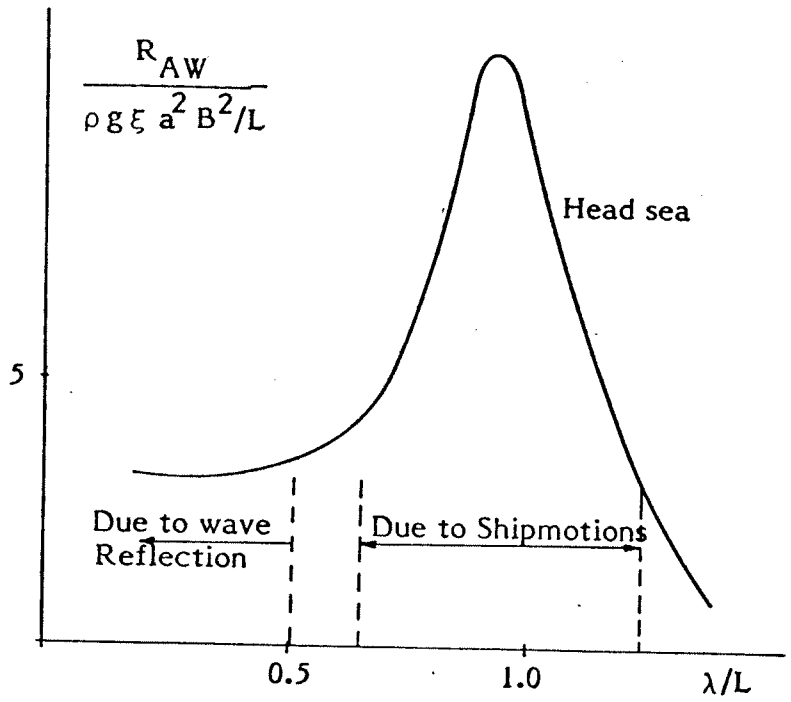


Fig. (2)

Different Spectra

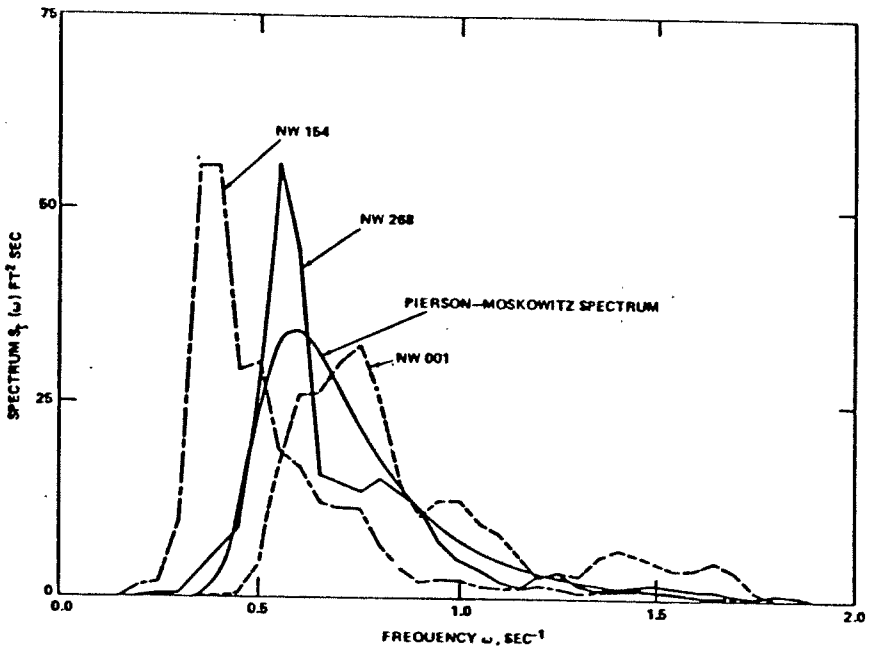


Fig. (3a)

Added Resistance in different Spectra

Ship Type	Ship Speed V Knots	Proude No. F_n	Added Resistance, R_{AW}, lb for Various Sea Spectra			
			NW 001	NW 154	NW 268	Pierson-Moskowitz
Destroyer	16.8	0.20	47,600	31,800	57,900	56,500
	21.0	0.25	49,300	40,700	74,000	65,000
	25.2	0.30	49,000	60,700	95,800	79,100
	29.4	0.35	43,000	60,700	95,800	79,100
High Speed Hull Form	16.8	0.20	58,500	33,600	62,700	67,400
	25.2	0.30	61,100	52,800	98,500	85,400
	33.6	0.40	44,500	74,200	105,600	88,300
	42.0	0.50	30,600	98,400	103,000	82,300
Series 60 $C_B = 0.60$	22.3	0.266	31,500	83,300	104,300	83,600
	23.8	0.283	28,400	90,600	102,900	82,300
Series 60 $C_B = 0.65$	19.9	0.237	32,400	75,200	100,100	81,000
	21.3	0.254	29,100	83,300	99,300	80,100
Series 60 $C_B = 0.70$	17.4	0.207	33,300	67,400	94,900	77,100
	18.7	0.222	30,400	74,200	95,200	76,900
Series 60 $C_B = 0.75$	14.9	0.177	34,500	60,200	88,900	72,100
	16.4	0.195	31,200	66,800	90,000	72,700
Series 60 $C_B = 0.80$	12.3	0.147	36,000	53,000	81,000	66,500
	13.9	0.165	32,800	58,600	83,200	67,900

Fig. (3b)

Wave Frequency in the North Atlantic Ocean for Beaufort 8 according to Walden's Data

Wave Height (m)	Wave Period (sec) 5	7	9	11	13	15	17		Sum over All Periods
0.75	0.07	0.63	0.42	0.28					1.40
1.75	1.26	4.21	2.25	0.98	0.14		0.07	0.07	8.98
2.75	4.49	25.75	26.60	6.46	1.68	0.07	0.14	0.49	65.68
3.75	4.63	57.40	94.87	36.14	6.95	1.26	0.35	0.42	202.02
4.75	1.61	47.22	124.07	57.60	13.19	3.09	0.35	0.84	247.97
5.75	0.70	18.10	82.18	48.42	11.22	2.74	0.84	0.35	164.55
6.75	0.35	16.00	57.48	45.33	12.35	2.18	0.28	0.07	134.04
7.75	0.42	6.99	29.54	30.95	9.96	0.77	0.14	0.42	81.19
8.75	0.07	6.81	14.55	25.19	10.31	2.04	0.21	0.14	59.22
9.75		3.79	6.18	9.62	6.74	2.53	0.63	0.07	29.56
10.75		0.14	0.35	0.42	0.35	0.28			1.54
11.75		0.07	0.28	0.35	0.77	0.14			1.61
12.75		0.21		0.35	0.63				1.19
13.75				0.28	0.28	0.07			0.63
14.75				0.07	0.14	0.07		0.07	0.35
15.75						0.07			0.07
Sum over All Heights	13.60	189.32	438.67	262.44	74.71	15.31	3.01	2.94	1000.00

Fig. (4a)

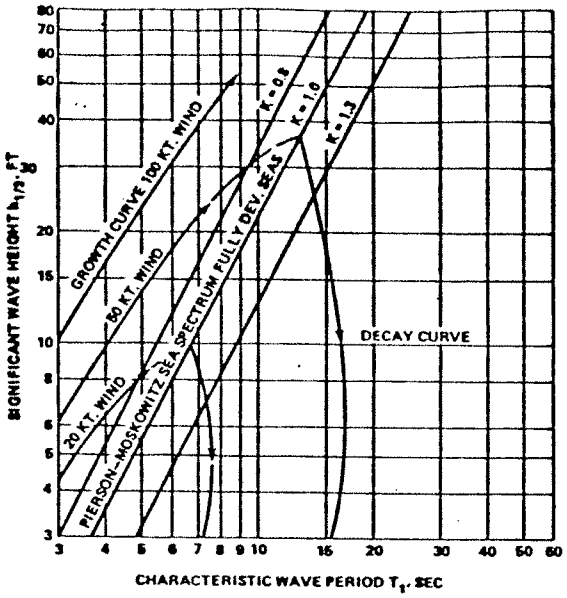


Fig. (4b)

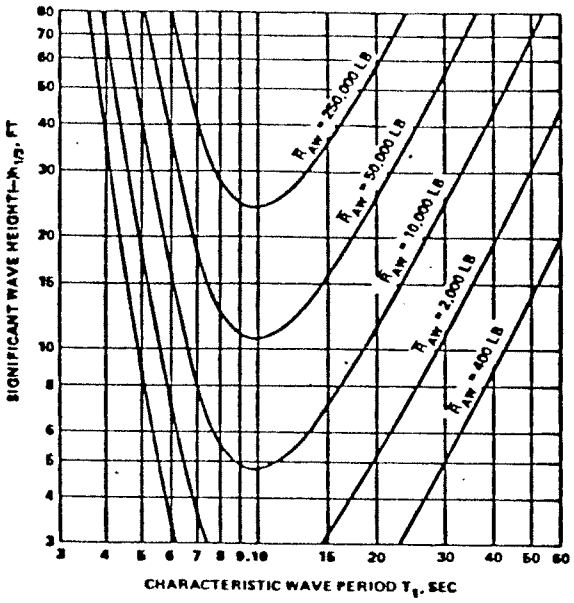


Fig. (5)

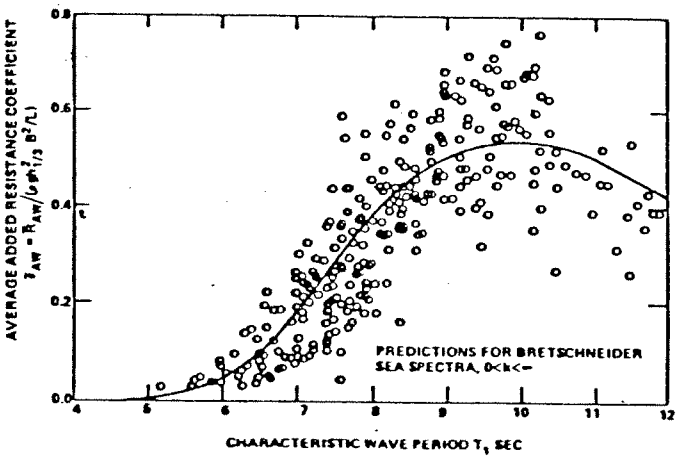
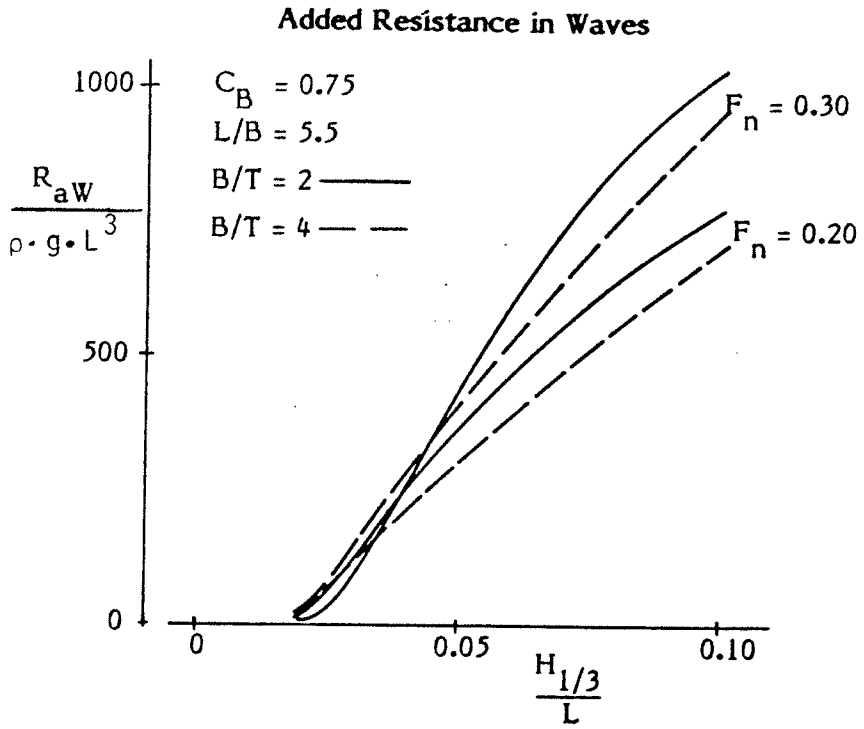
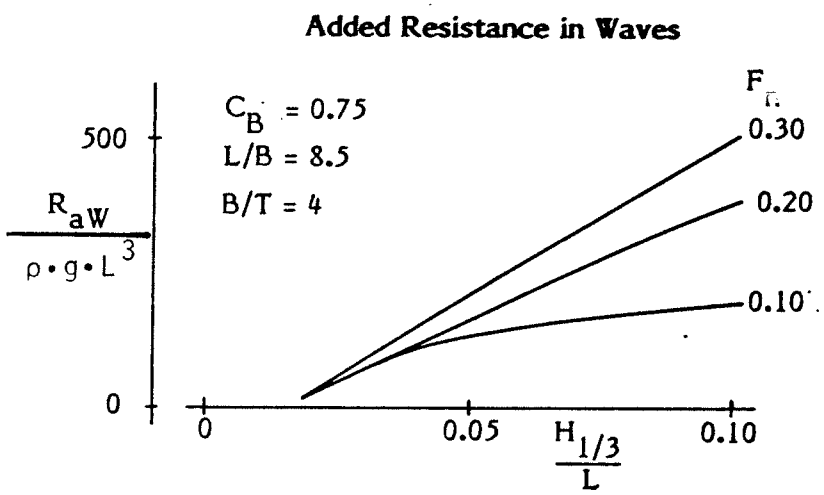


Fig. (6)


Added Resistance in Waves
1000
500
0
$\frac{R_{aW}}{\rho \cdot g \cdot L^3}$
C_B = 0.75
L/B = 5.5
B/T = 2
B/T = 4
F_n = 0.30
F_n = 0.20
0
0.05
0.10
$\frac{H_{1/3}}{L}$


Added Resistance in Waves
500
0
$\frac{R_{aW}}{\rho \cdot g \cdot L^3}$
C_B = 0.75
L/B = 8.5
B/T = 4
F_n
0.30
0.20
0.10
0
0.05
0.10
$\frac{H_{1/3}}{L}$

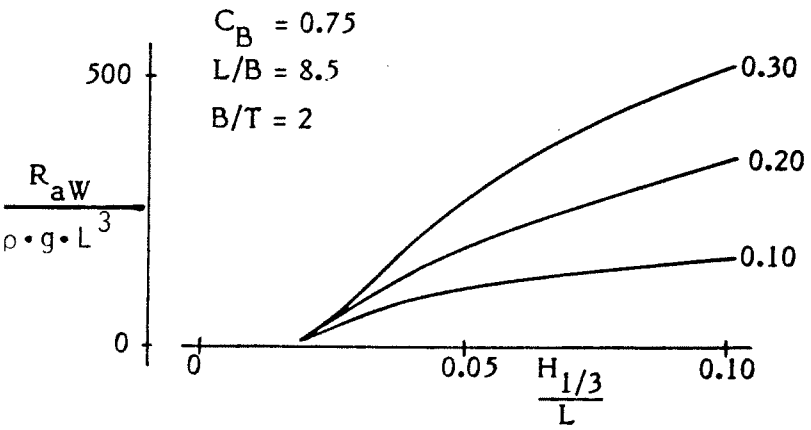
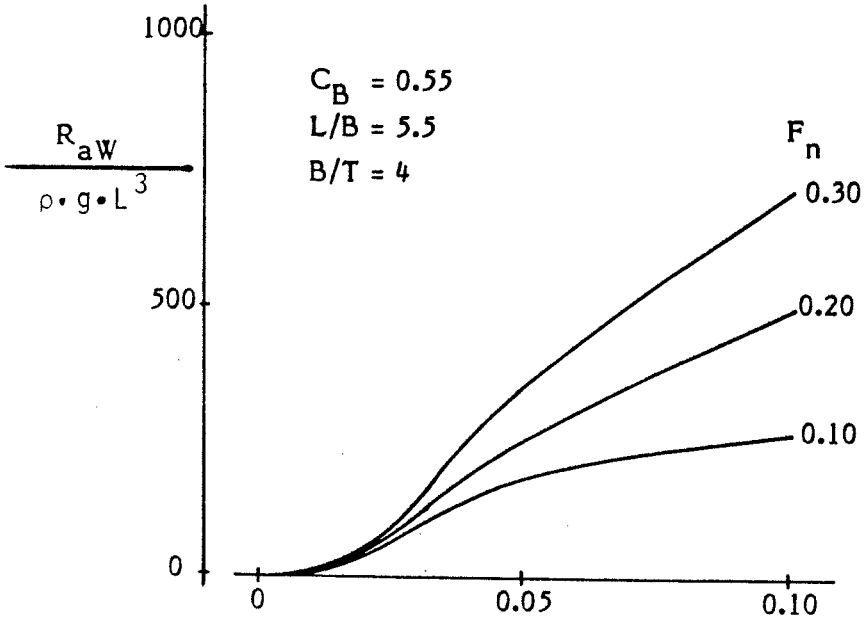
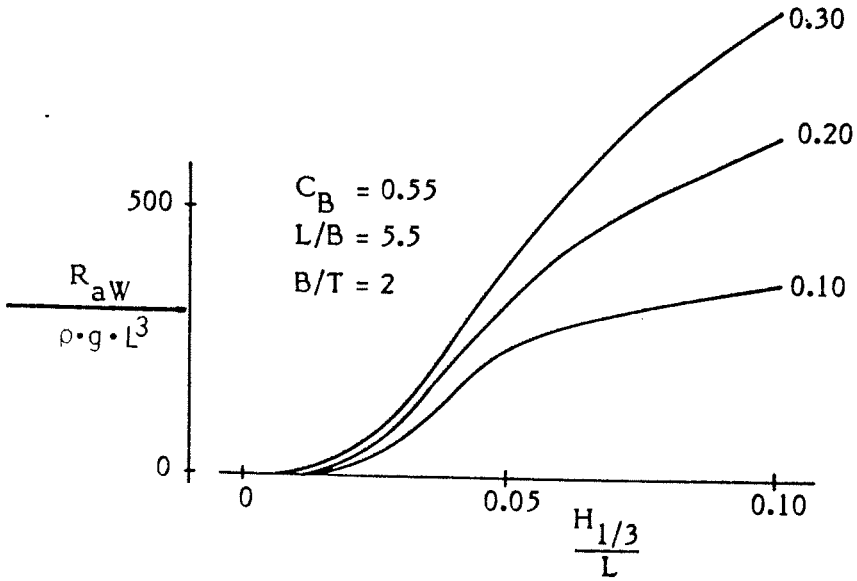

500
0
$\frac{R_{aW}}{\rho \cdot g \cdot L^3}$
C_B = 0.75
L/B = 8.5
B/T = 2
0.30
0.20
0.10
0
0.05
0.10
$\frac{H_{1/3}}{L}$

Fig. (7)

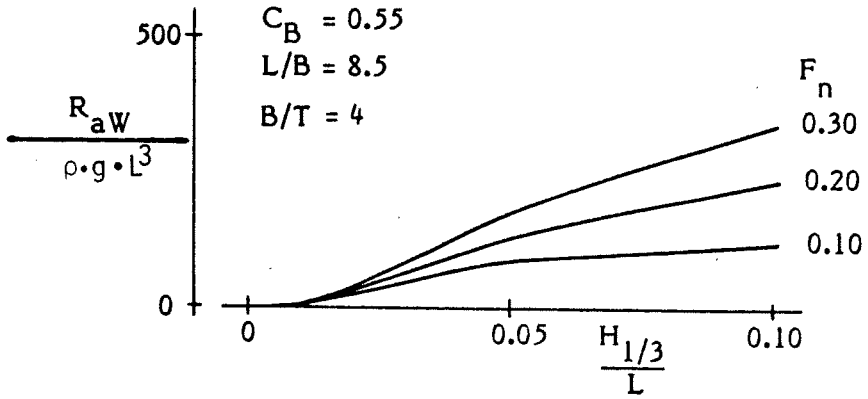
Fig. (8)

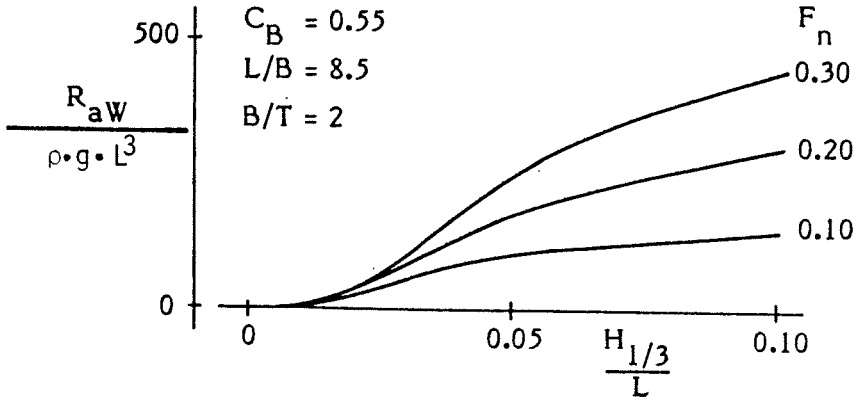
Added Resistance in Waves





Added Resistance in Waves





Influence of Forebody Shape on Motions in head Waves

Series 60. $F_n = 0.15$

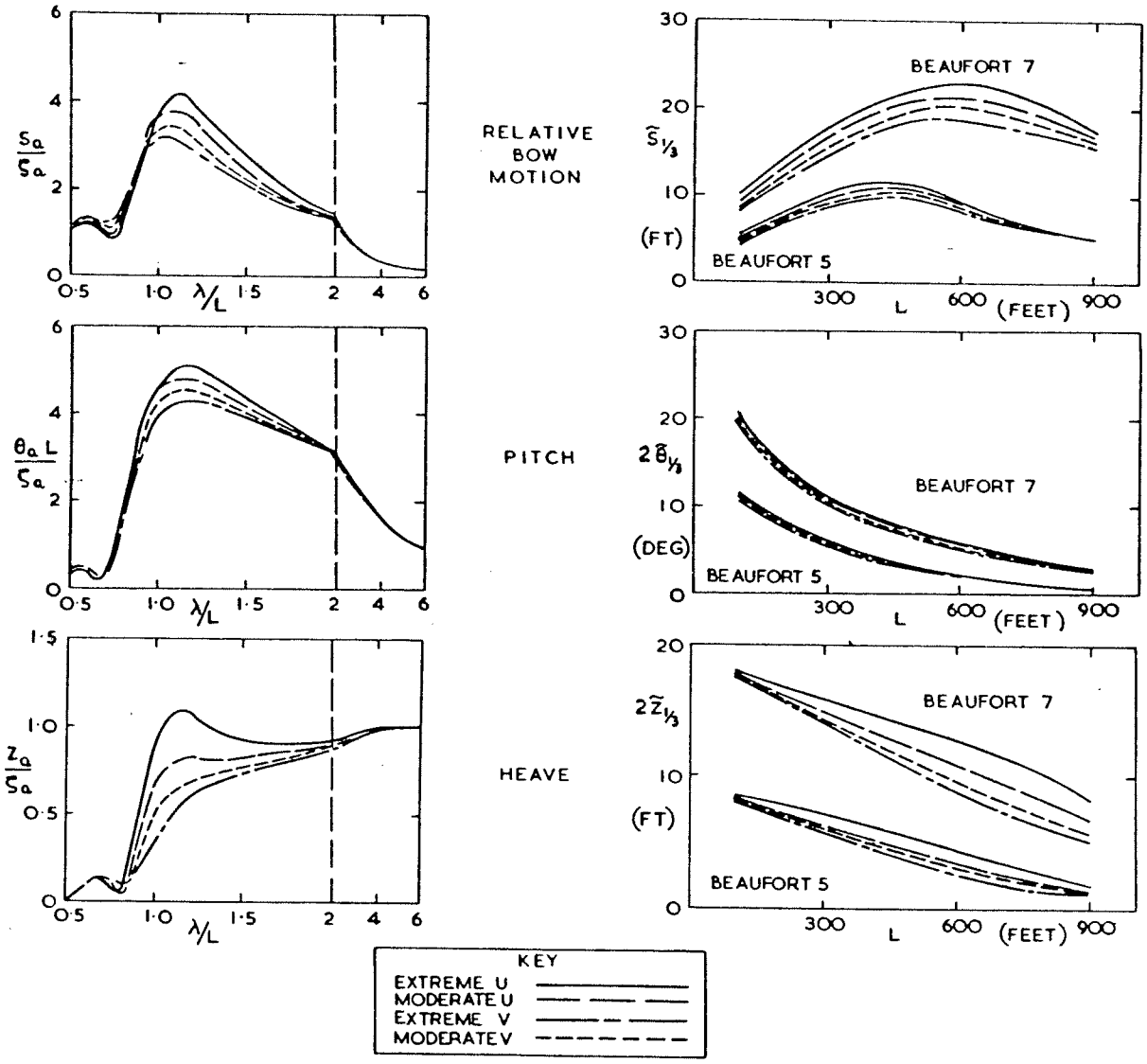


Fig. (9)

Influence of Speed on Motions in head Waves. Series 60

Extreme U-form

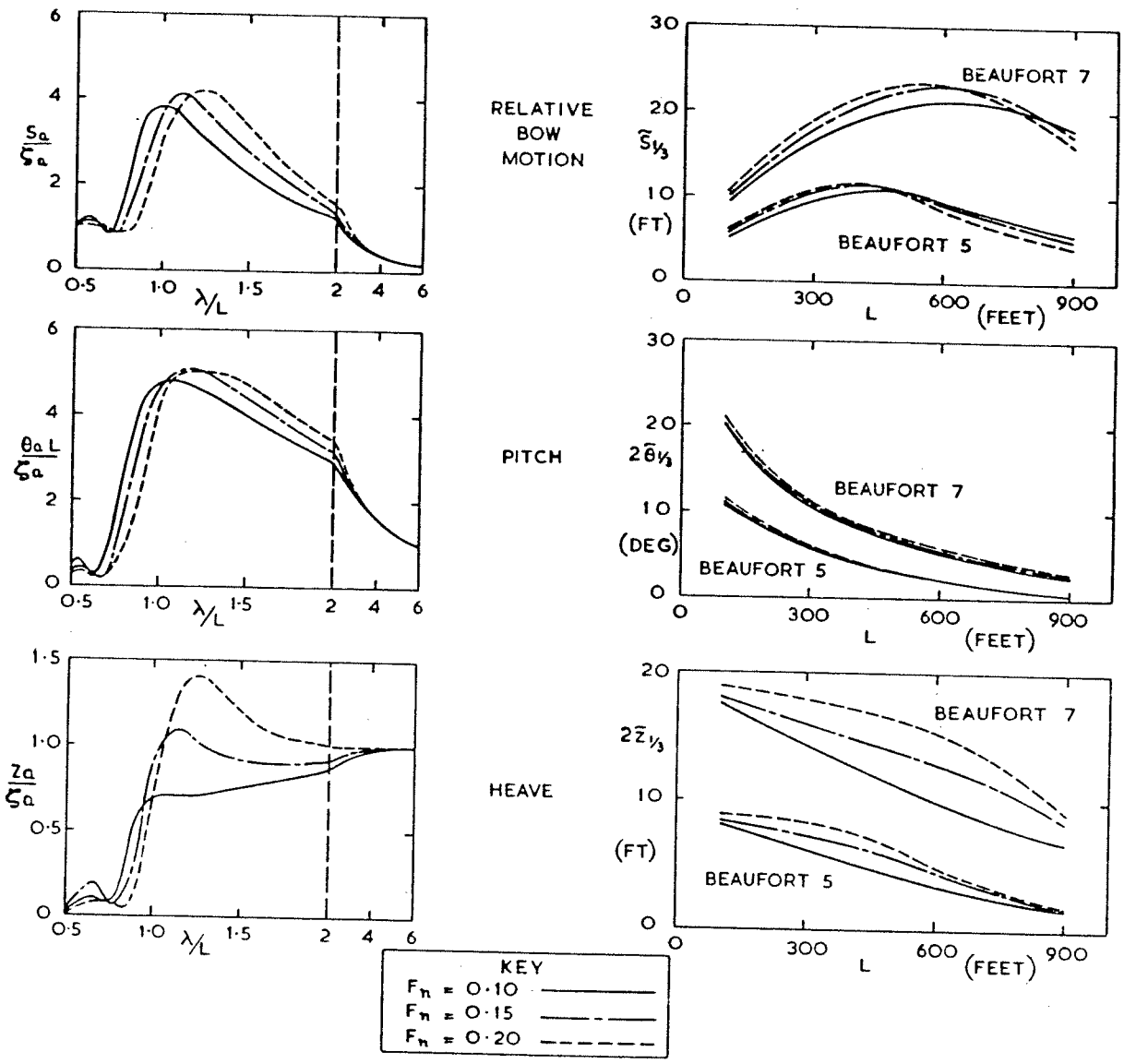
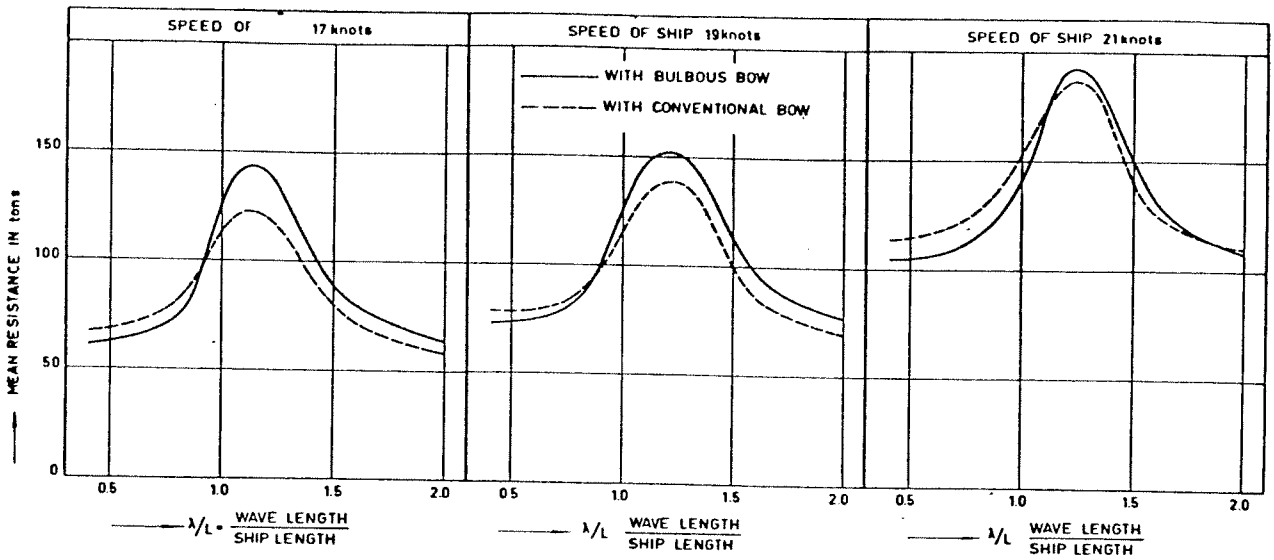


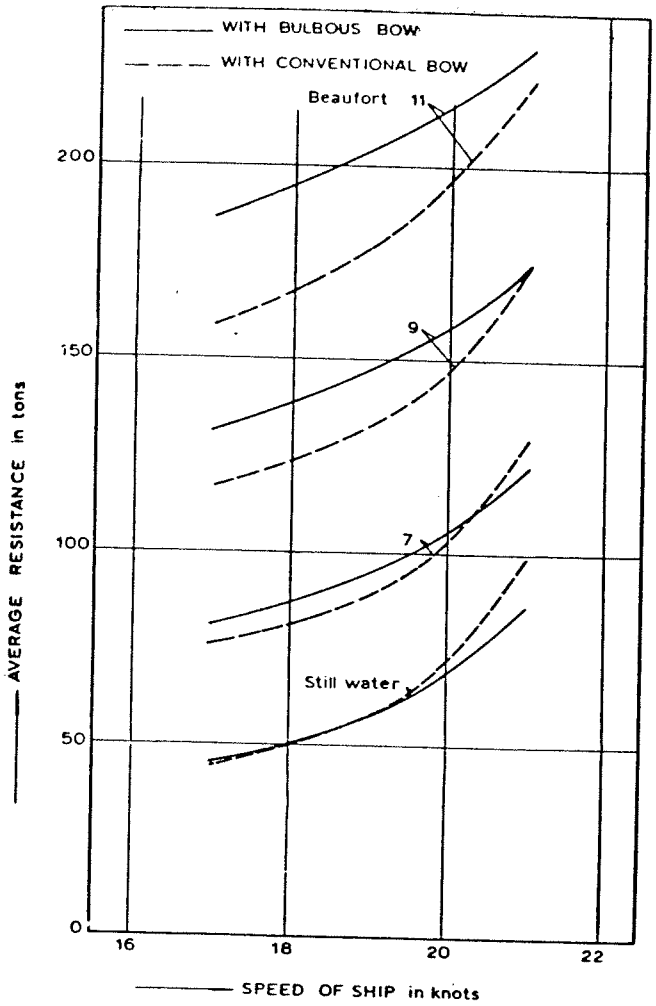
Fig. (10)

Influence of Bow Shape on Added Resistance and Power



Mean resistance in regular waves, wave height $2\bar{h} = 3$ m

Fig. (11)



Calculated average resistance for various seas, based on test results in regular seas. For still water the measured values are given

Fig. (12)

Trend of speed with sea conditions for Victory ships

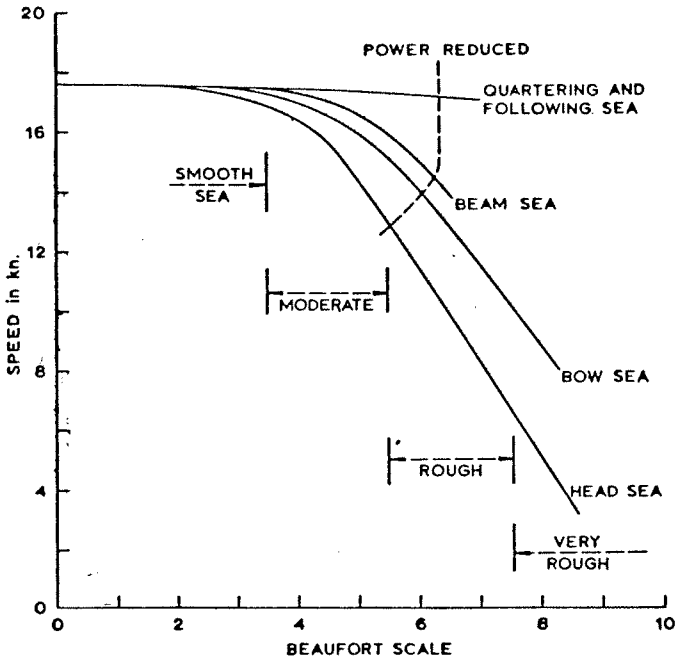


Fig. (13)

Effect of wave direction on power increase for Victory ships

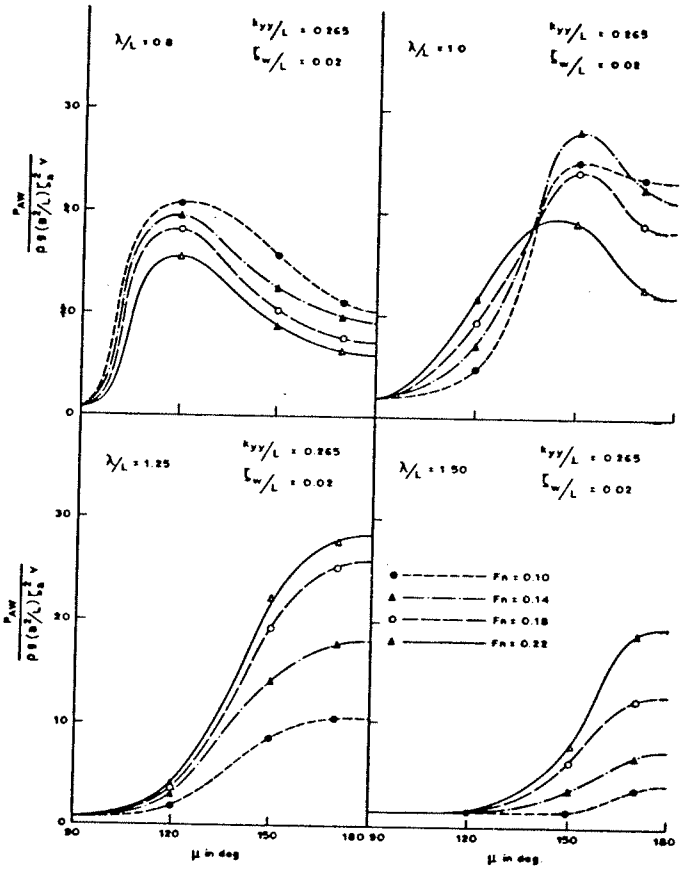
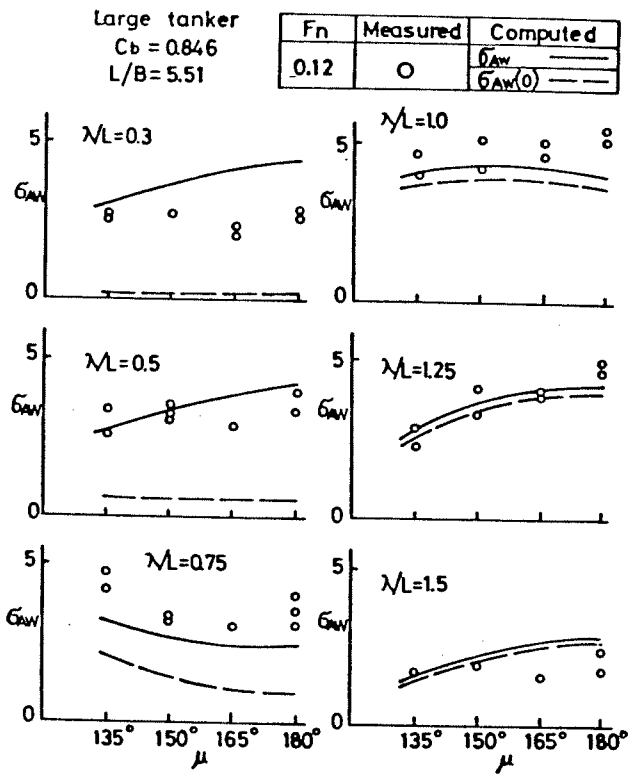


Fig. (14a)

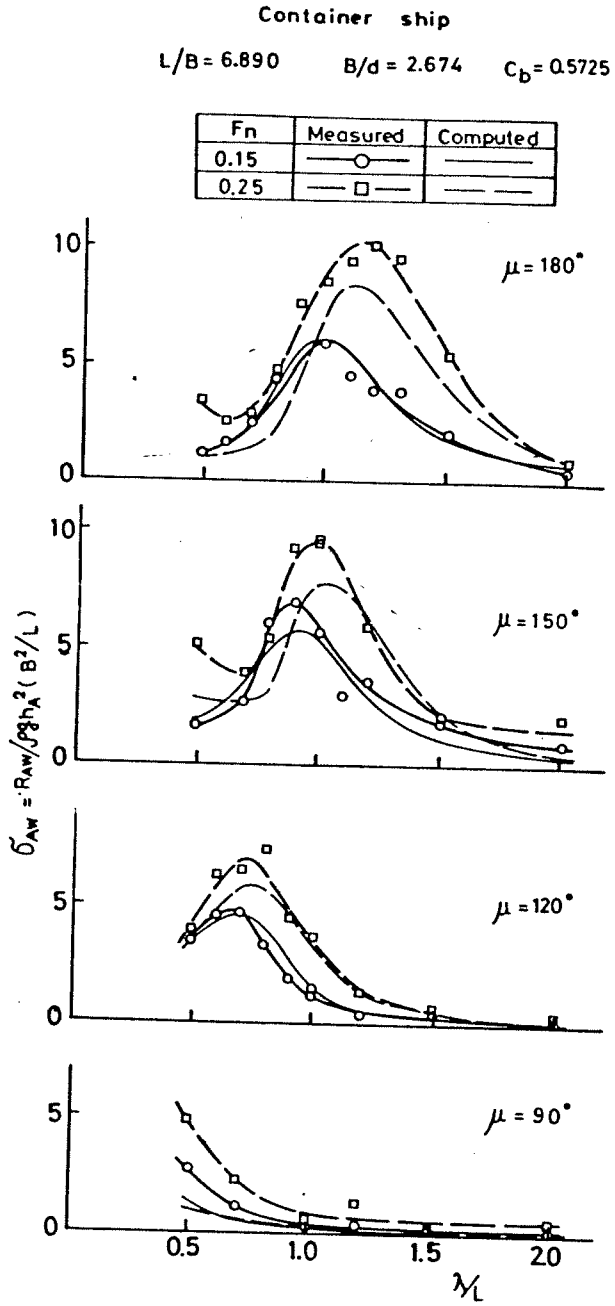
Influence of Wave Direction



Measured and added resistance coefficients in regular head and bow waves.

From Fujii and Nakamura [46]

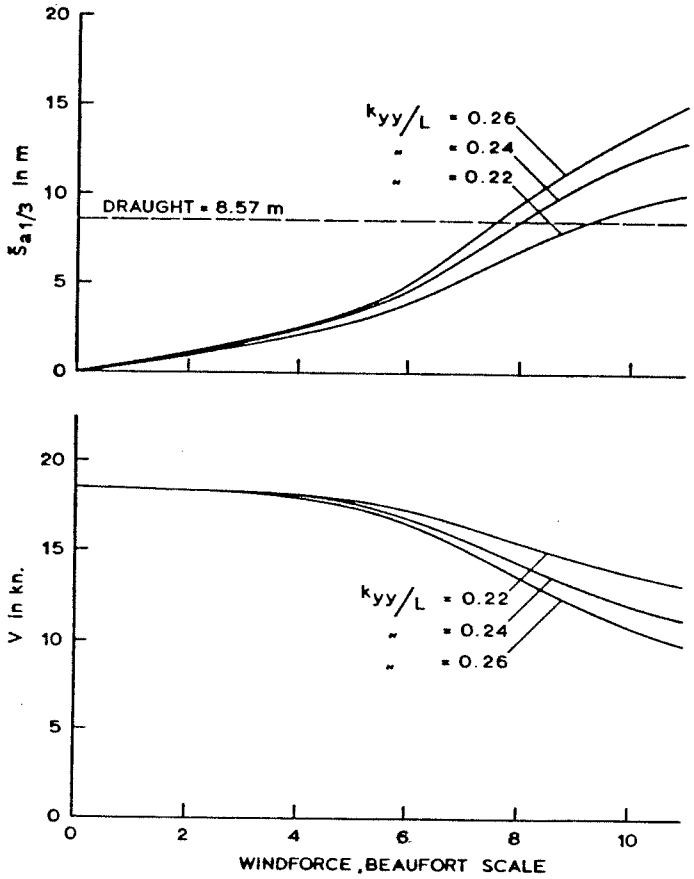
Fig.(14b)



Comparison of measured and computed added resistance coefficients in regular oblique waves

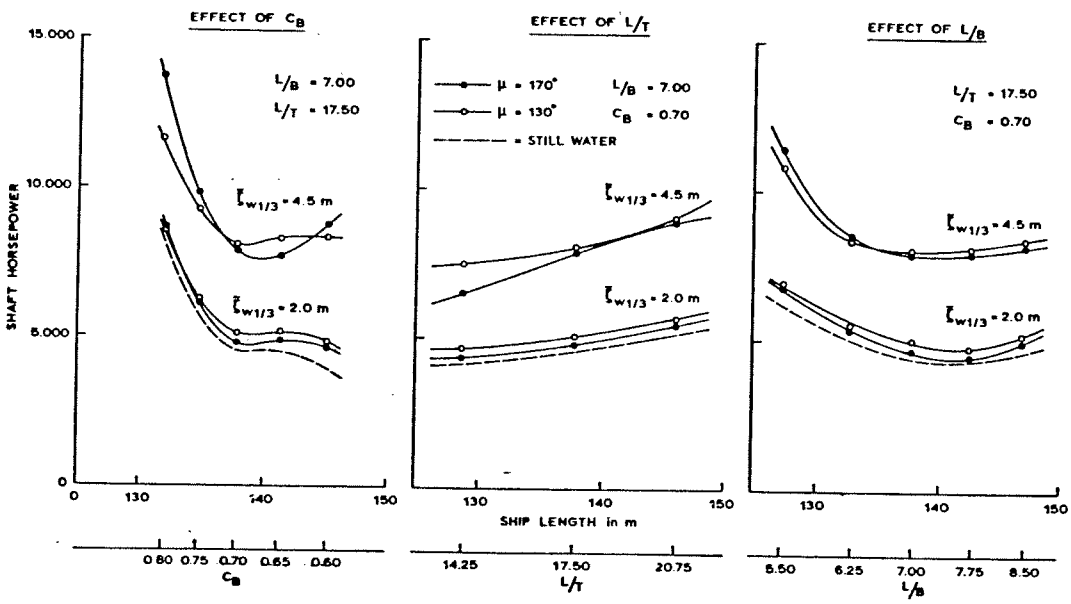
From Nakamura and Fujii [46]

Fig. (14c)



Speed loss and relative bow motion in head seas for a 150 m ship at 7.830 HP

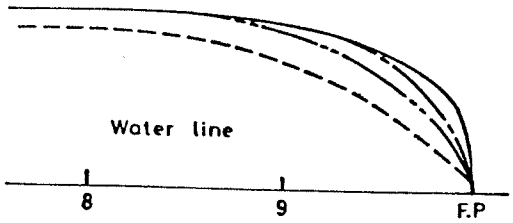
Fig. (15)



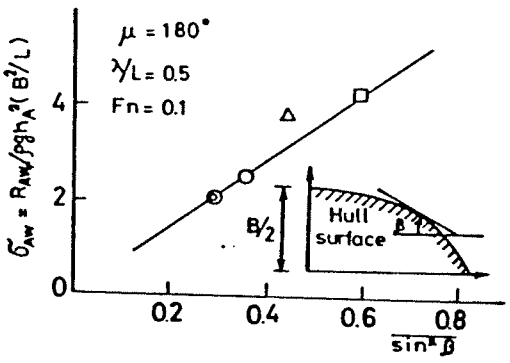
Required shaft horse power for 15.000 tons displacement ship configurations for a sustained speed of 15 knots

Fig. (16)

Added Resistance - Bluntness Coefficient Dependence



$\sin^2\beta$	Water line	Measured
0.592	————	□
0.438	—— - ——	△
0.356	—— -- ——	○
0.292	— — — —	⊙



Relation between bluntness coefficients and added resistance coefficients

From Fujii and Nakamura [46]

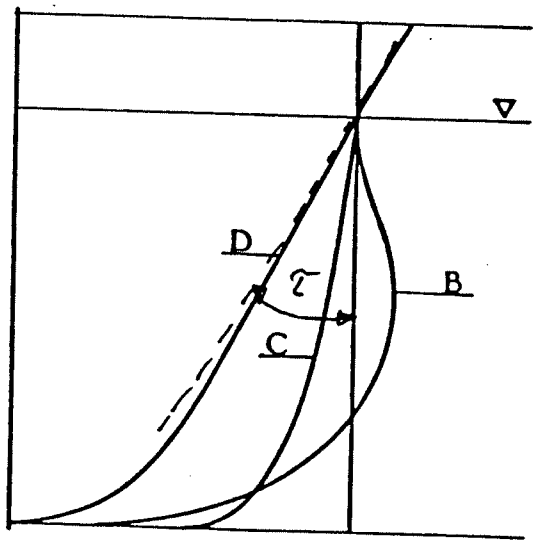


Fig. (17)

Added Resistance at Small Wave Lengths

from Faltinsen, Liapis, Minsaas and Skjördal [24]

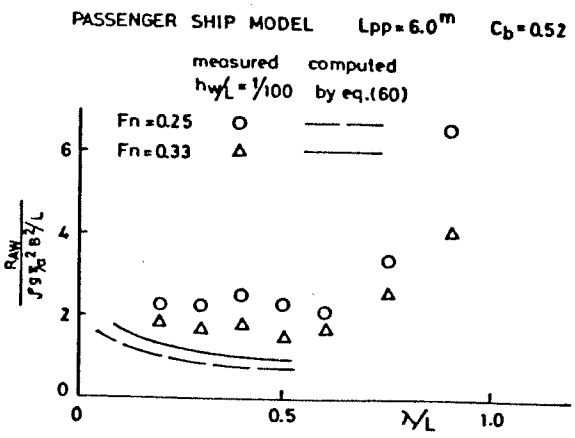


Fig. 1

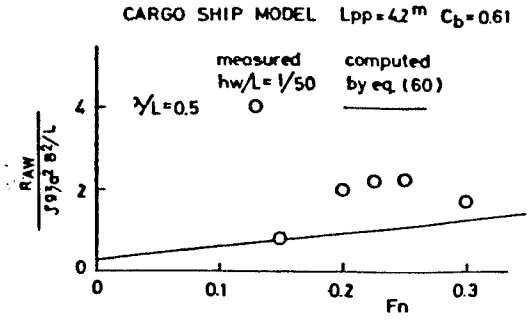


Fig. 2

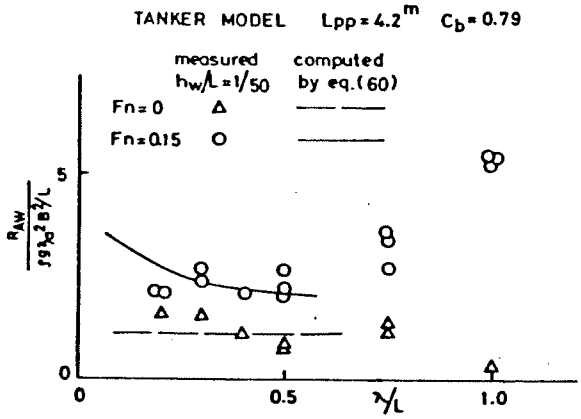


Fig. 3

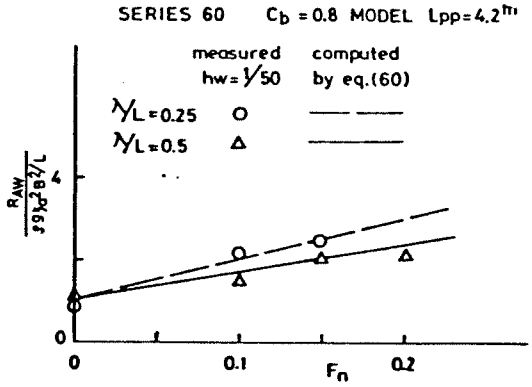


Fig. 4

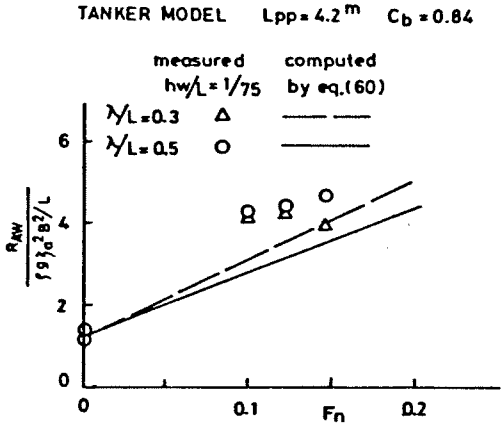


Fig. 5

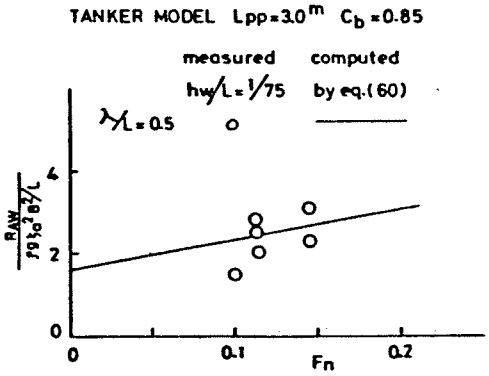


Fig. 6

Comparison of added resistance at small wave lengths

computed values: by the authors' asymptotic formula

measured values: obtained in Nagasaki Exp. Tank

Fig. (18)

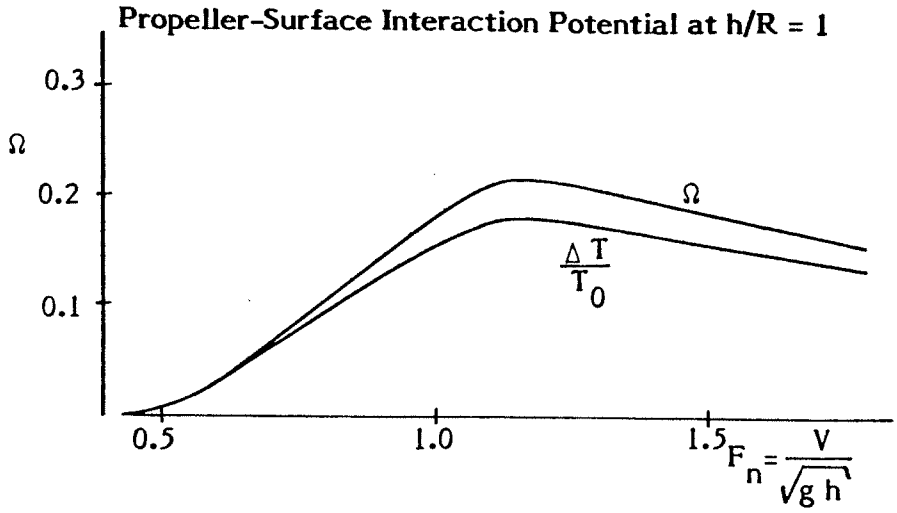


Fig. (19)

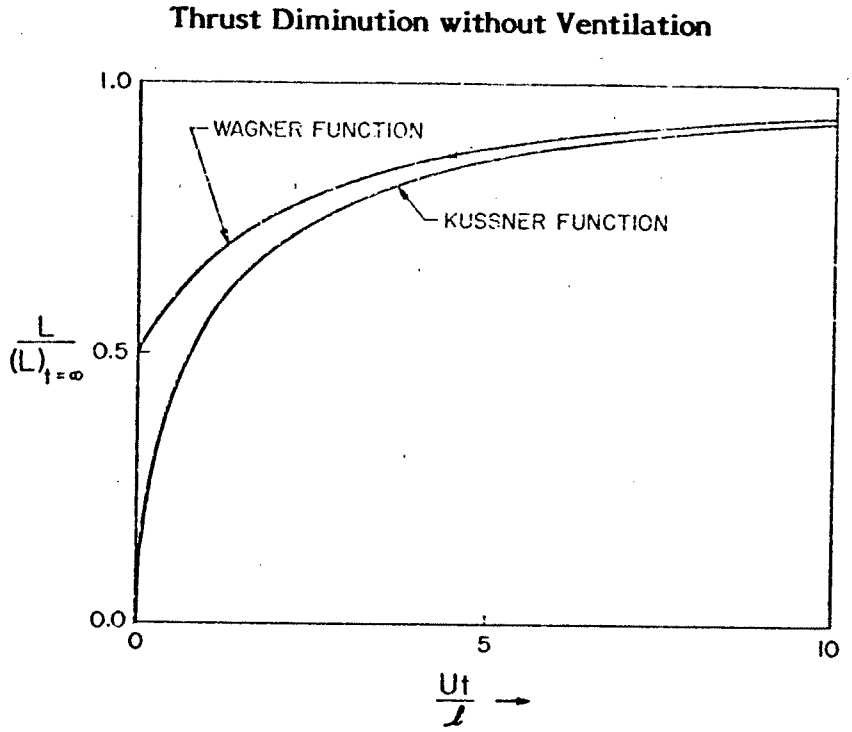


Fig. (20)

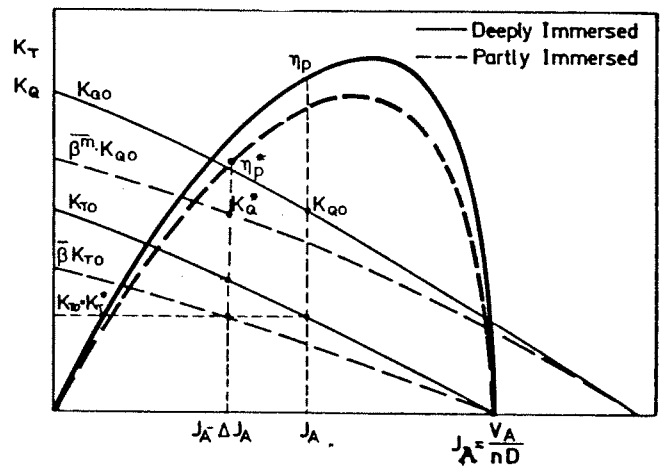
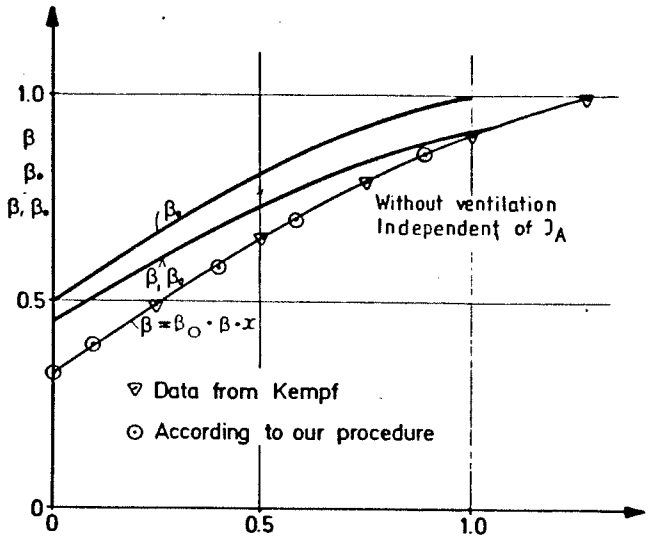


Fig. (21)

Open water diagram for partly and deeply immersed propeller



β -diagram for a propeller h/R

Fig. (22)

Propeller Diagram

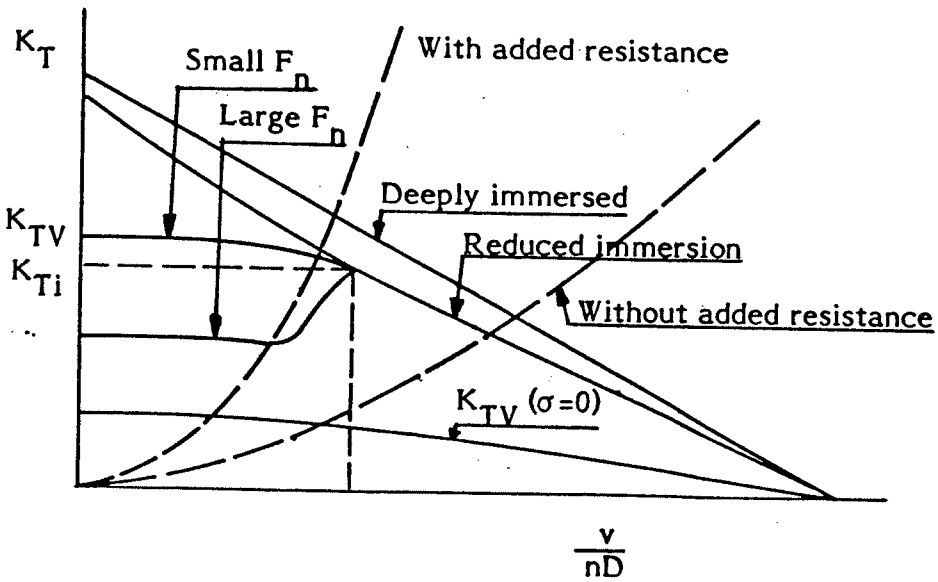


Fig. (24)

Influence of Immersion and Ventilation on the open Water Diagram

from Fleischer [28]

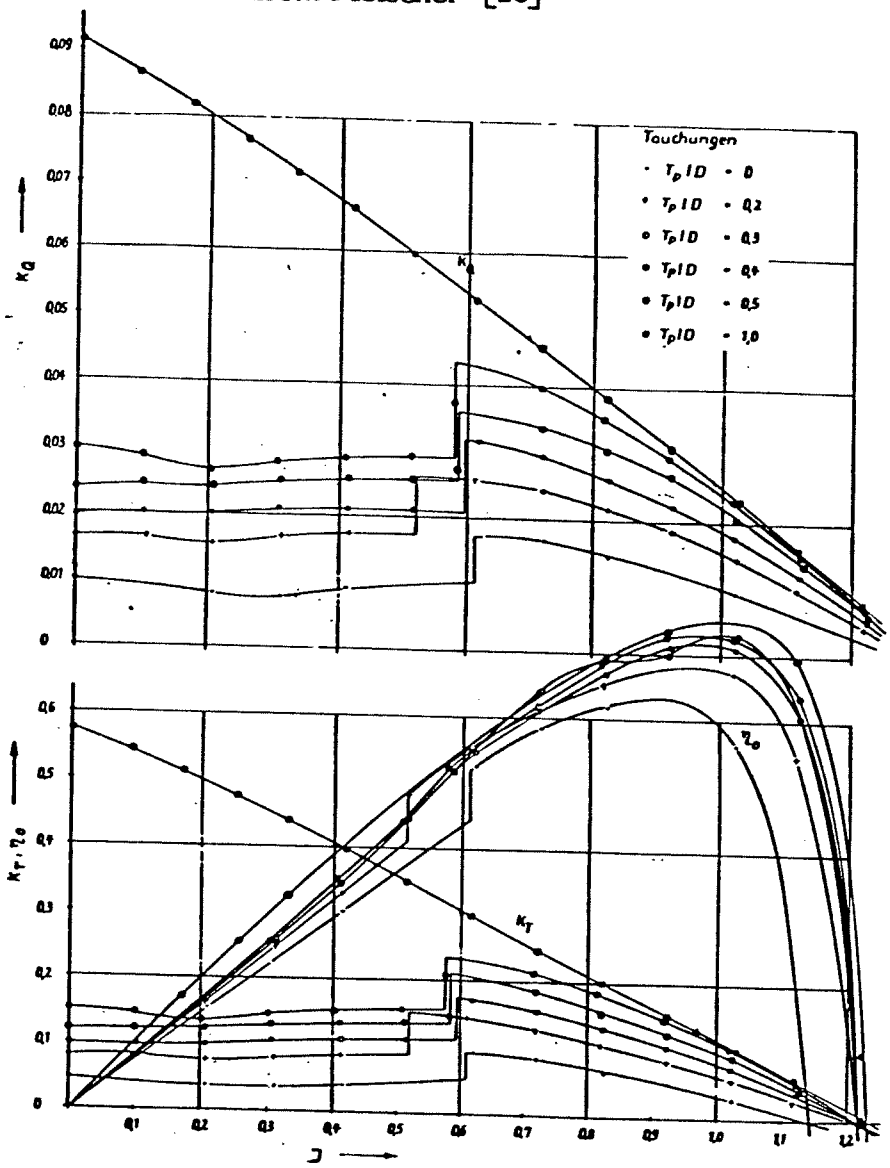


Fig. (23)

Lift Characteristics at Thrust Breakdown

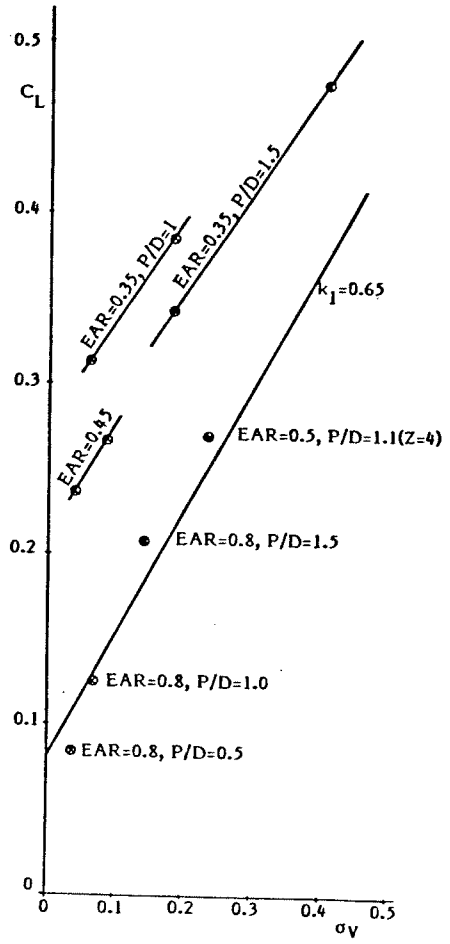


Fig. (25)

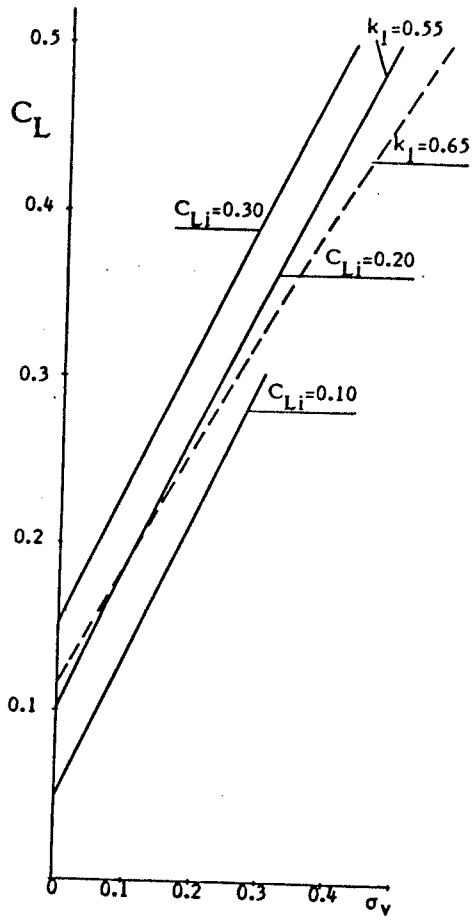


Fig. (26)

Estimated and Measured Propeller Characteristics in Regular Waves

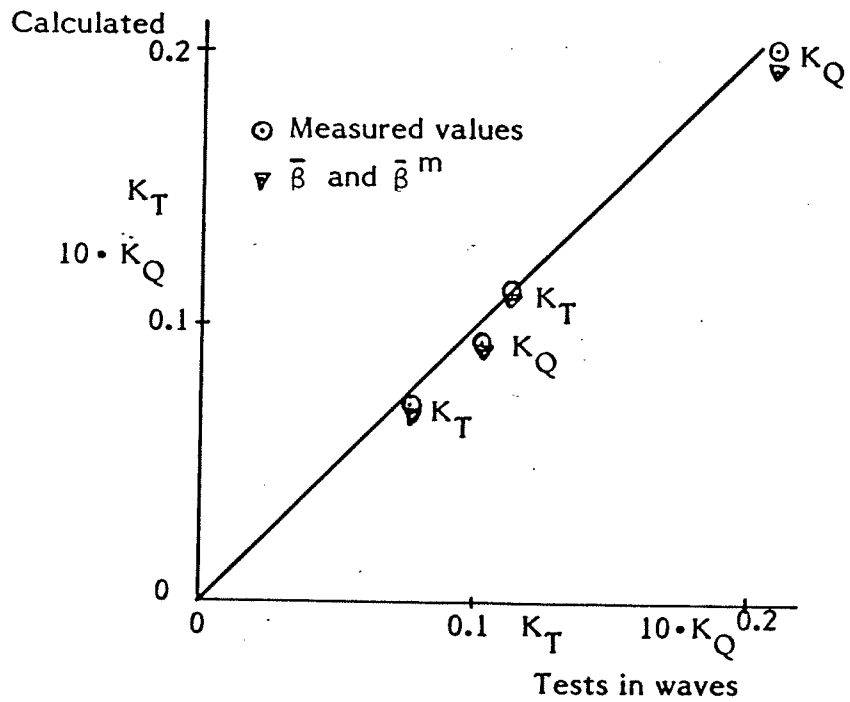


Fig. (27)

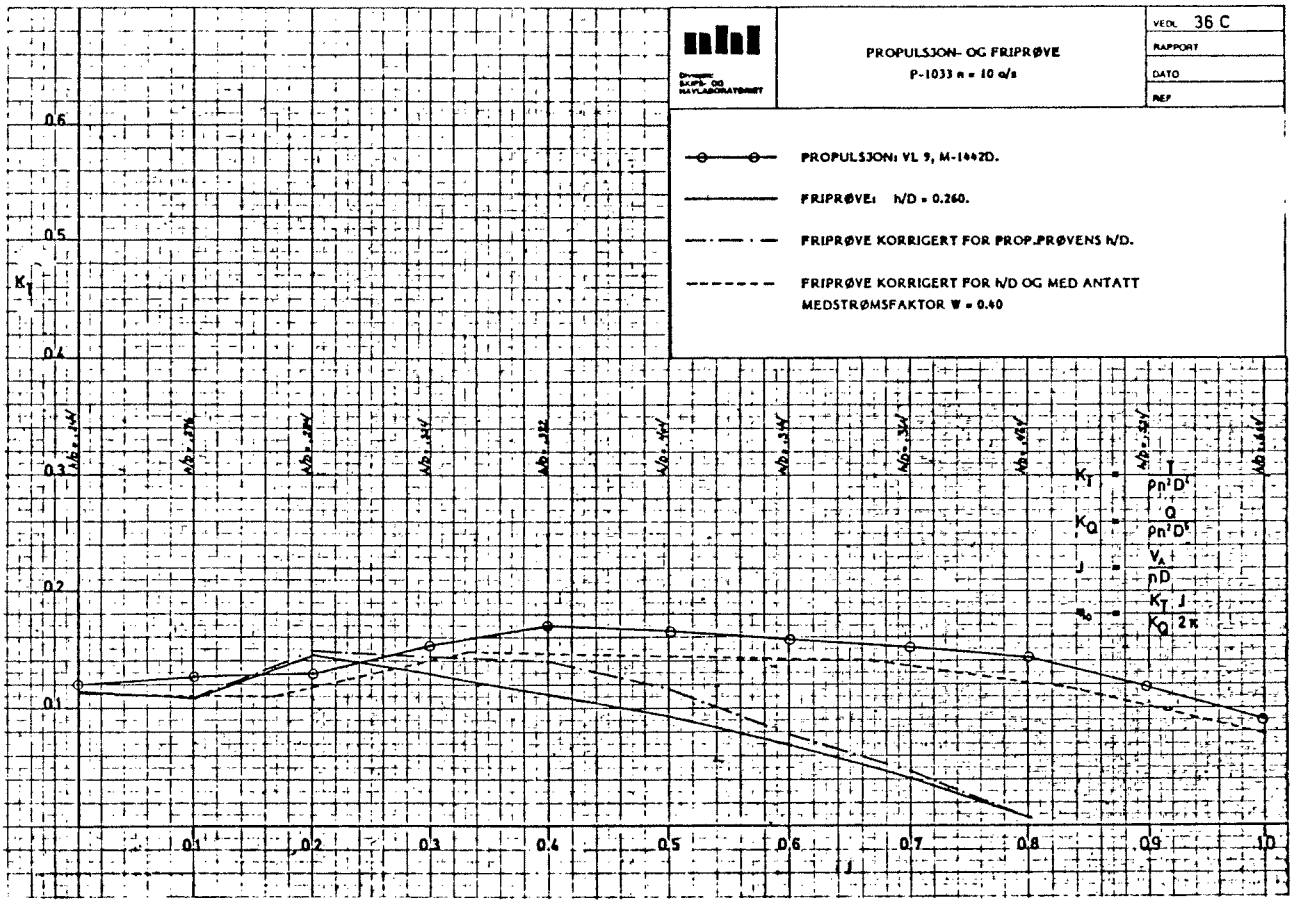

PROPULSJON- OG FRIPRØVE
P-1033 n = 10 o/s
VEDL. 36 C
RAPPORT
DATO
REF
PROPULSJON: VL 9, M-1442D.
FRIPRØVE: h/D = 0.260.
FRIPRØVE KORRIGERT FOR PROP.PRØVENS h/D.
FRIPRØVE KORRIGERT FOR h/D OG MED ANTATT MEDSTRØMSFAKTOR W = 0.40
K_T
K_Q
J
η_o
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0

Fig. (29)

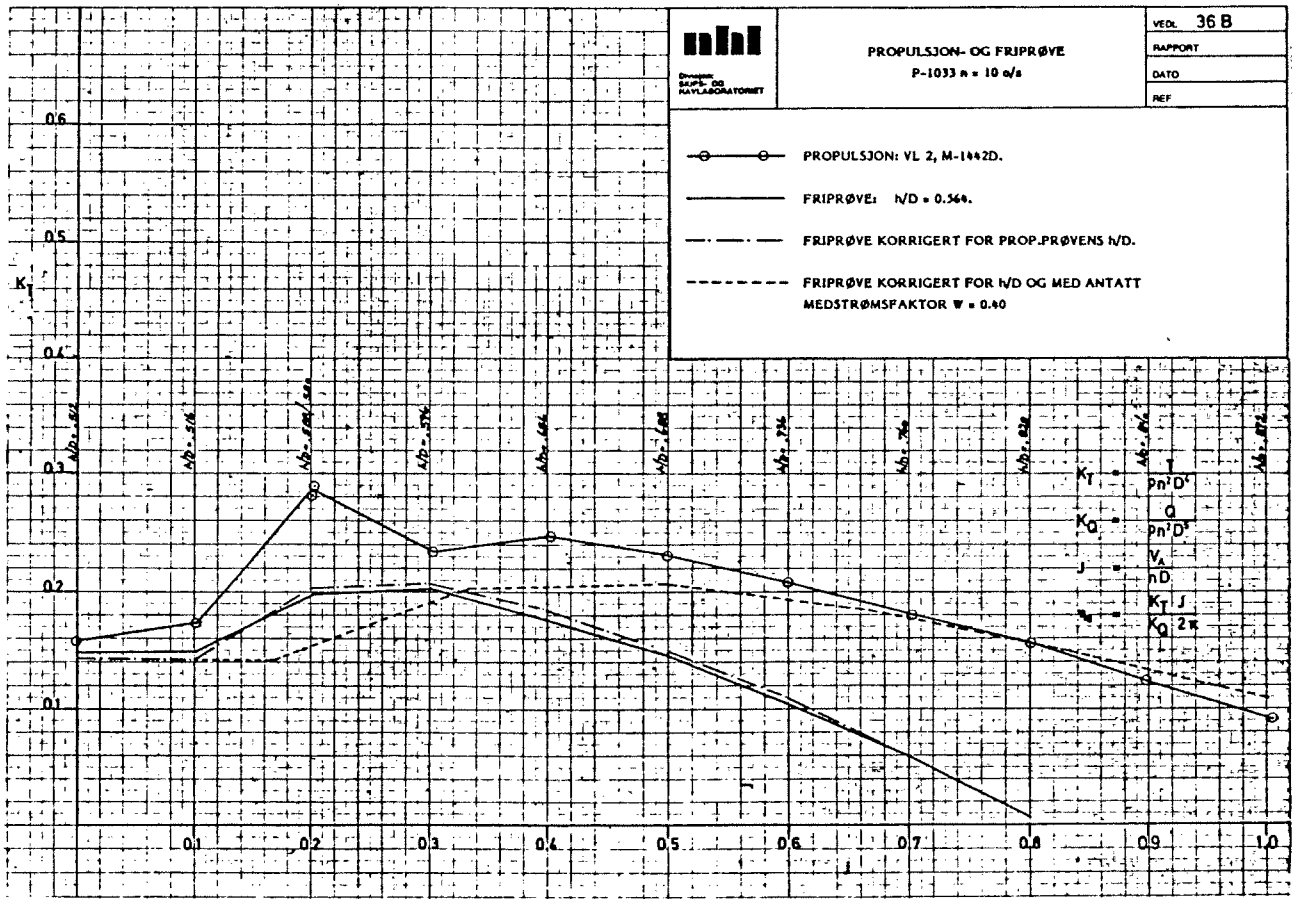
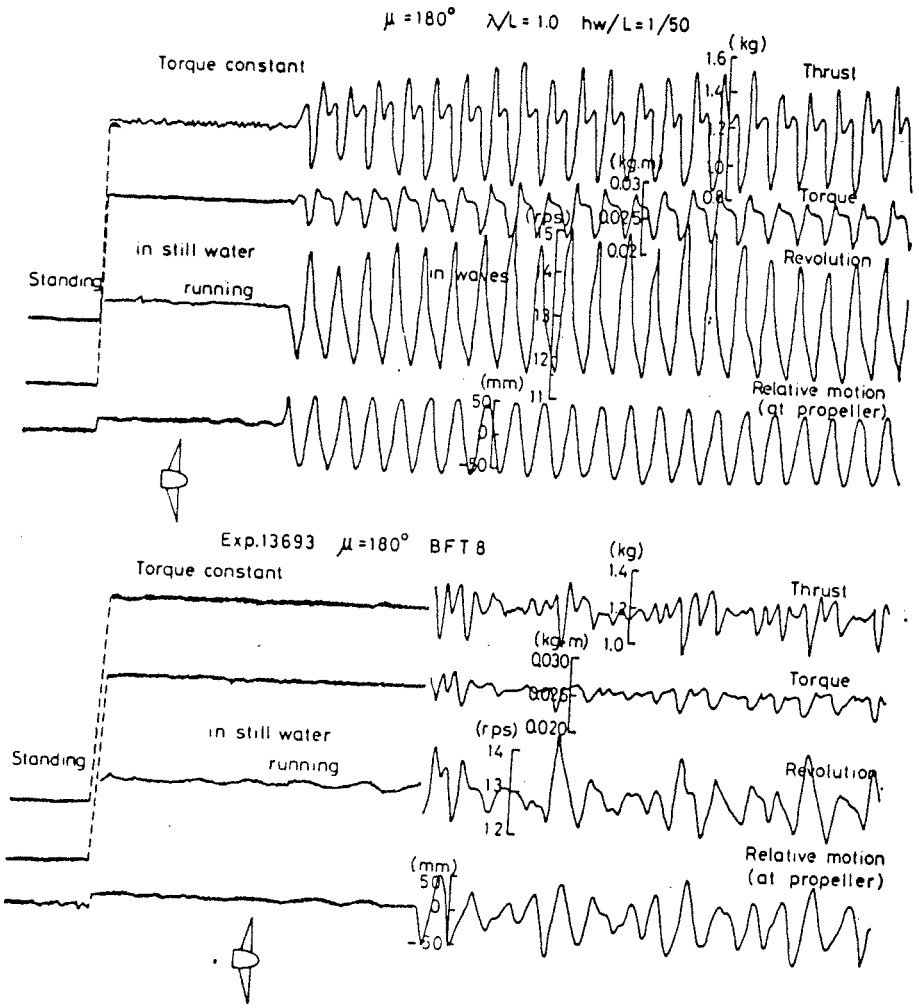
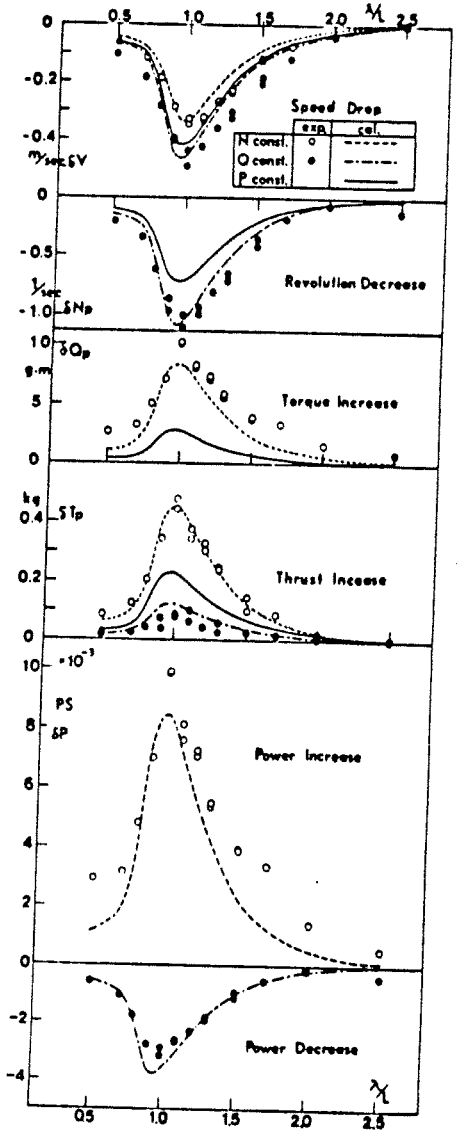

PROPULSJON- OG FRIPRØVE
P-1033 n = 10 o/s
VEDL. 36 B
RAPPORT
DATO
REF
PROPULSJON: VL 2, M-1442D.
FRIPRØVE: h/D = 0.564.
FRIPRØVE KORRIGERT FOR PROP.PRØVENS h/D.
FRIPRØVE KORRIGERT FOR h/D OG MED ANTATT MEDSTRØMSFAKTOR W = 0.40
K_T
K_Q
J
η_o
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0

Fig. (28)



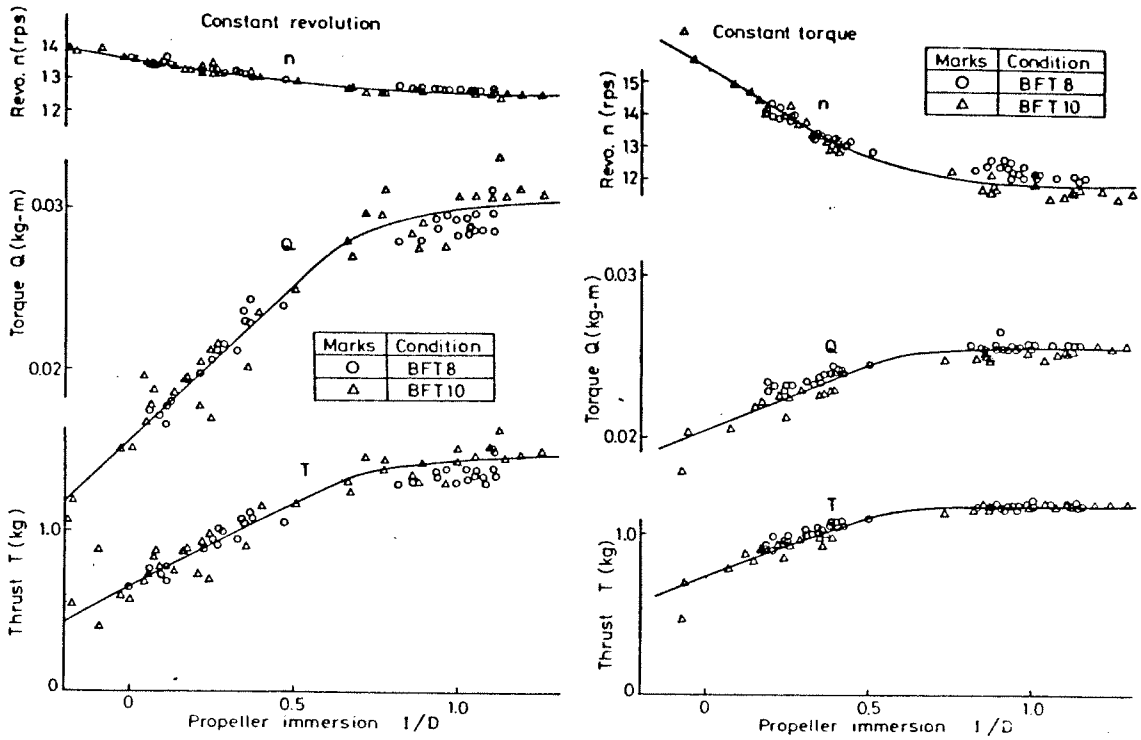
Example of records of propeller load fluctuations at racing condition

Fig. (31)



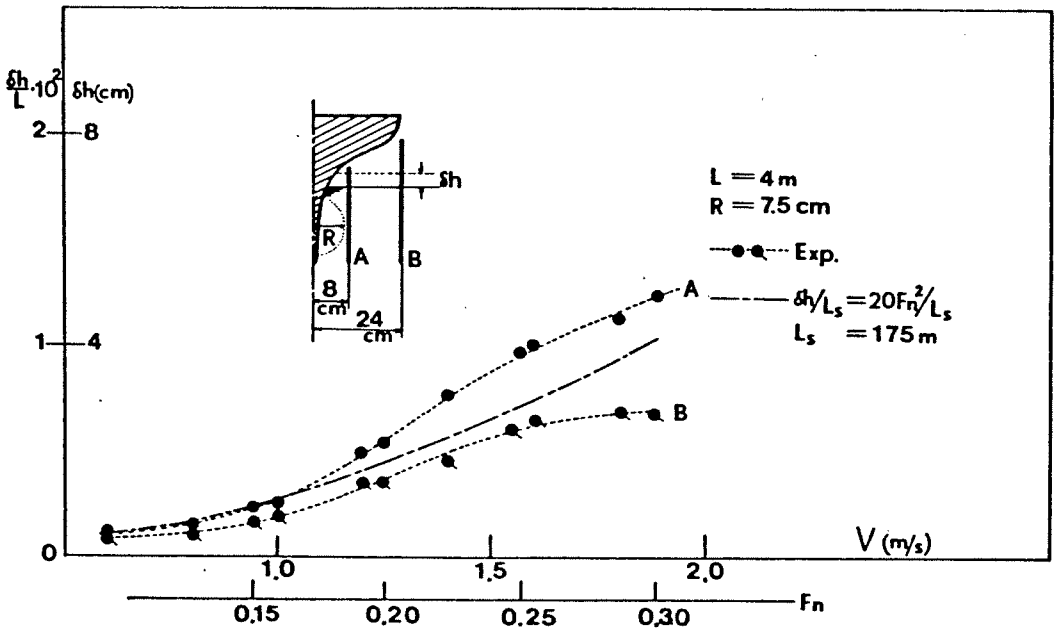
Comparison of measured and computed speed drop in regular head waves.
From Nakamura and Fujii [46]

Fig. (30)



Instantenous values of propeller thrust, torque and revolutions versus propeller immersion

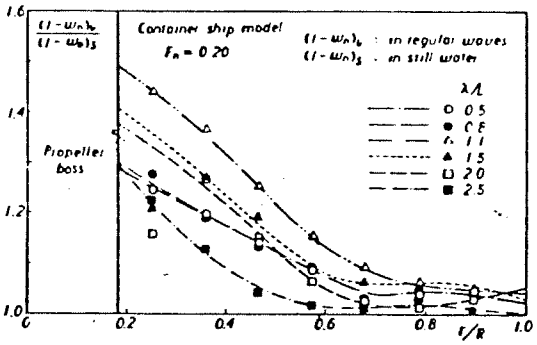
Fig. (32)



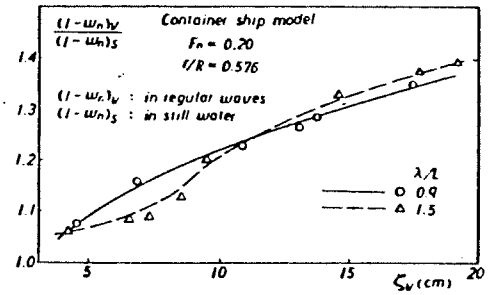
Swell up of water level at propeller position

Fig. (33)

Influence of Waves on Propulsion Factors

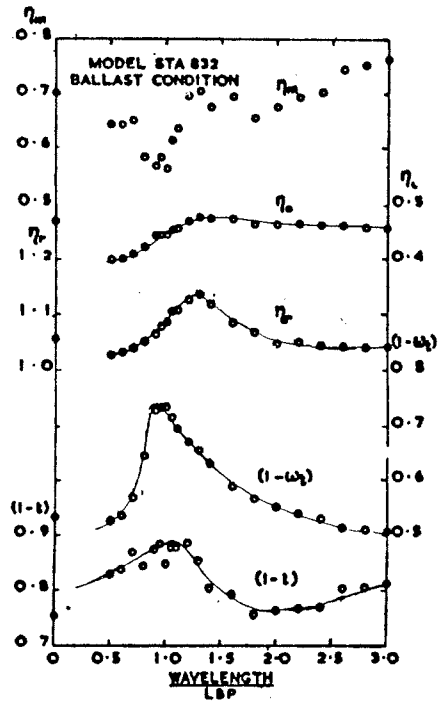


Ratio of $(1-w_n)$ in propeller disk in regular head waves to that in still water



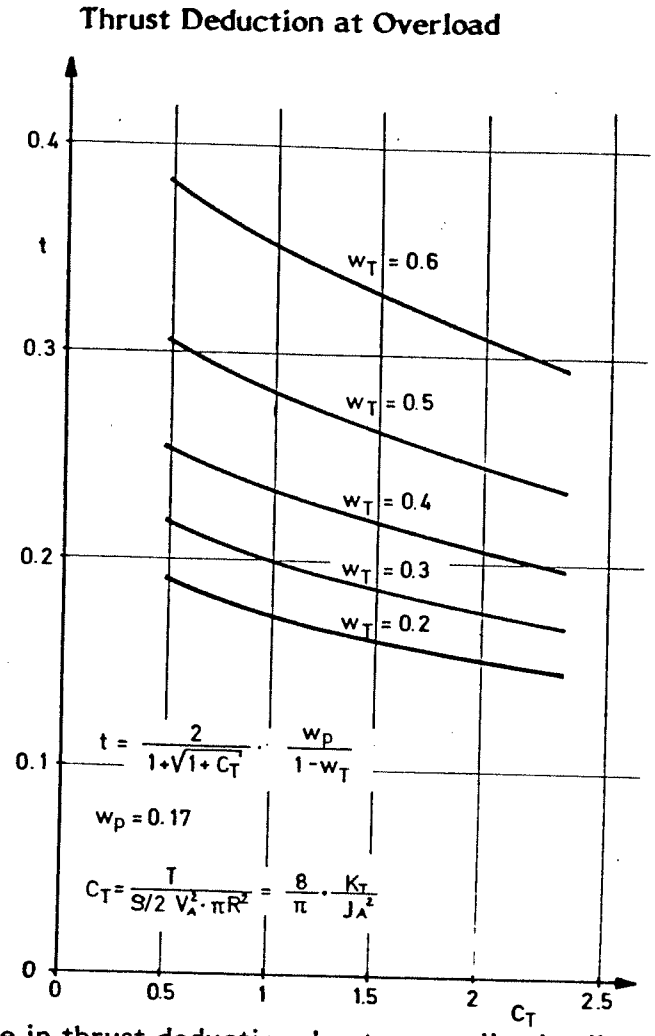
Ratio of $(1-w_n)$ in propeller disk in regular head waves to that in still water (effect of wave height)

From Nakamura and Naito [2] Fig. (34)



from Moor and Murday [15]

Fig. (35)



Change in thrust deduction due to propeller loading from Faltinsen. Liapis, Minsaas and Skjördal [24]

Fig.(36)

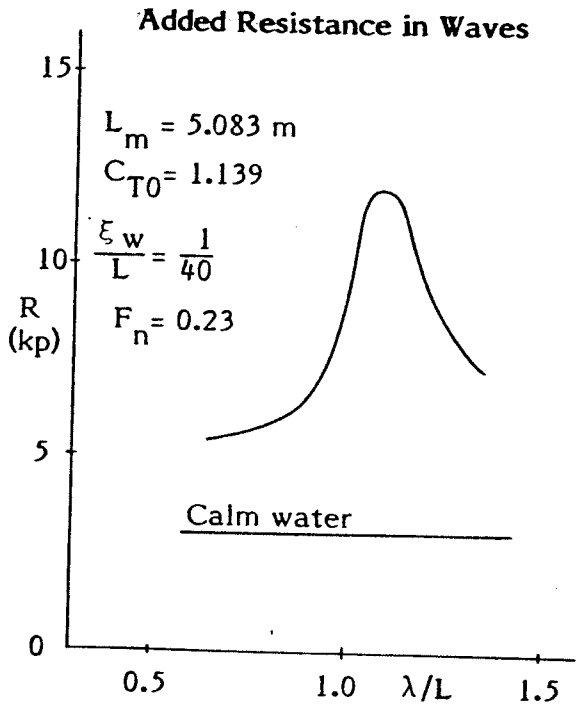
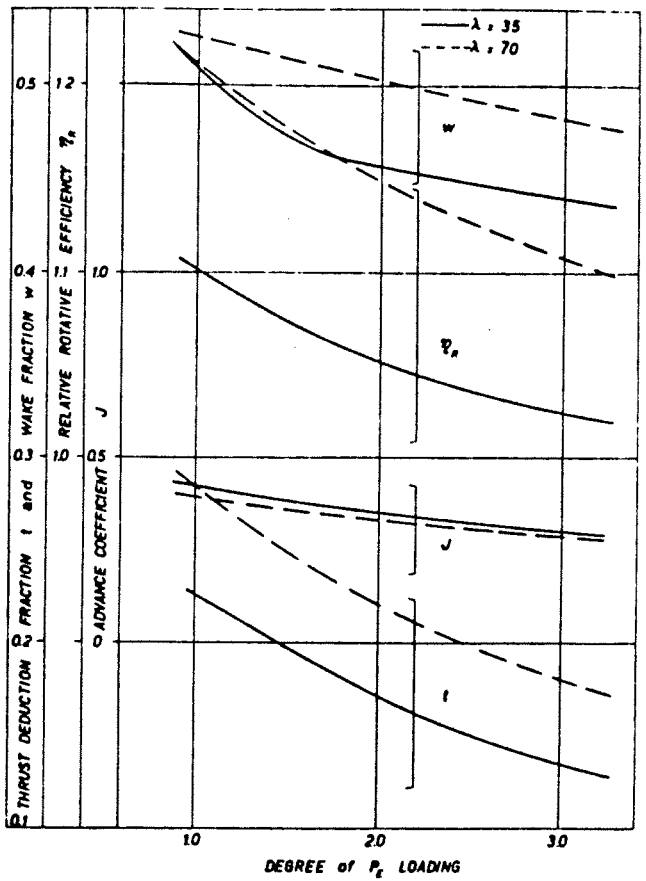
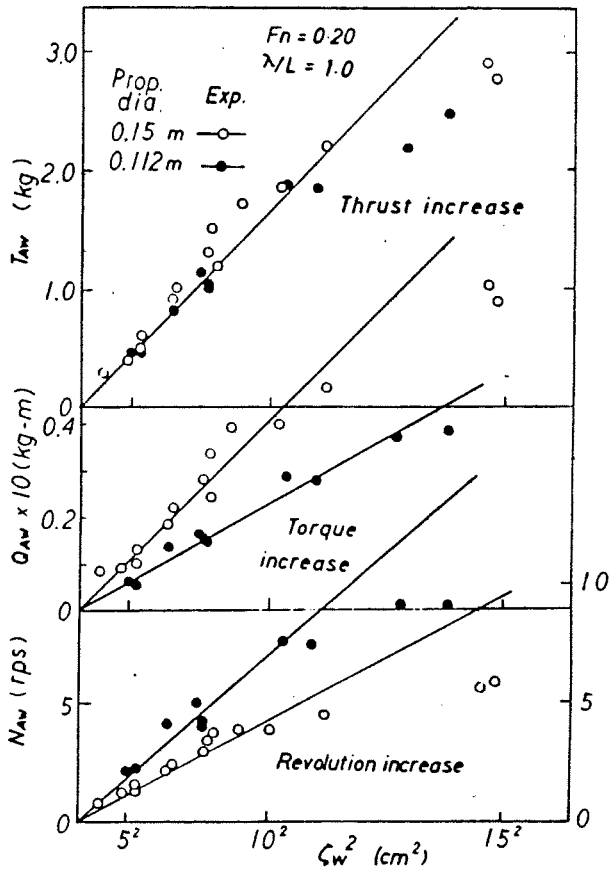


Fig. (37)



Propulsion coefficients at F_n=0.13 in still water with different propeller loadings.

Fig.(39)



Effect of wave height on mean increase of propeller thrust, torque and revolutions in regular head waves

From Nakamura-Naito [22]

Fig. (38)

Imaginary Increase in η_r due to Surface Effects

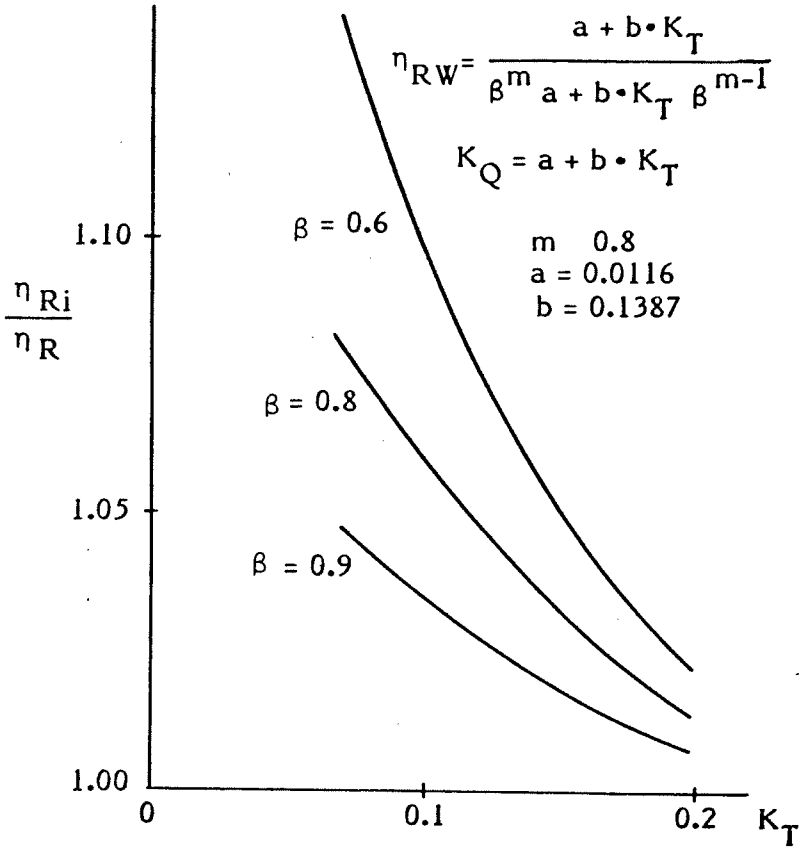
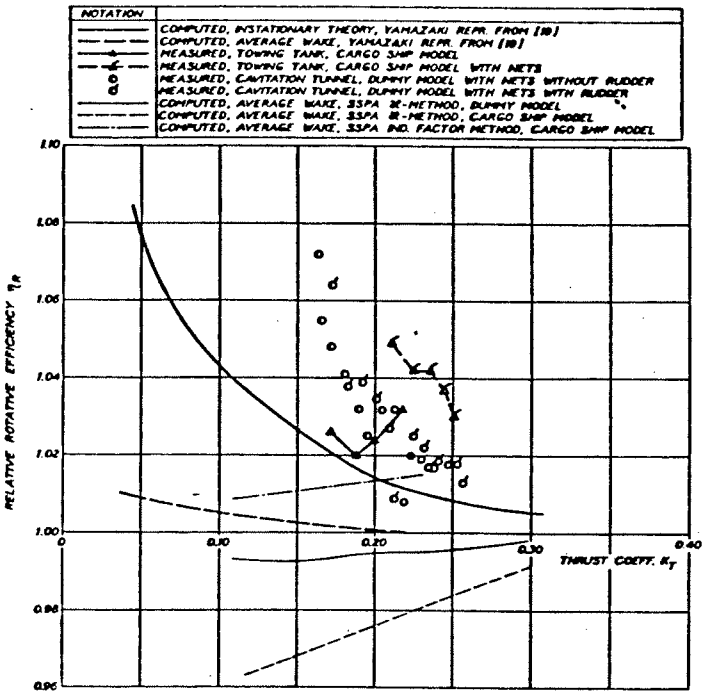


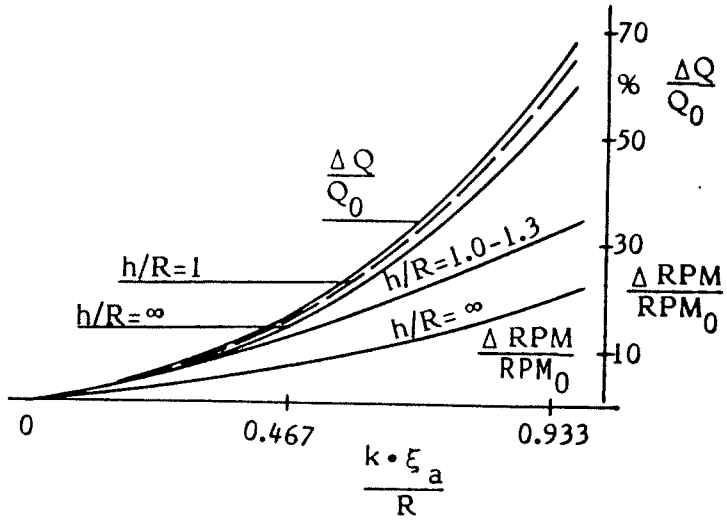
Fig. (40)



Variation of relative efficiency with K_T. Comparison between calculations and experiments.

Fig. (41)

Increase in Power, RPM and Torque due to Waves



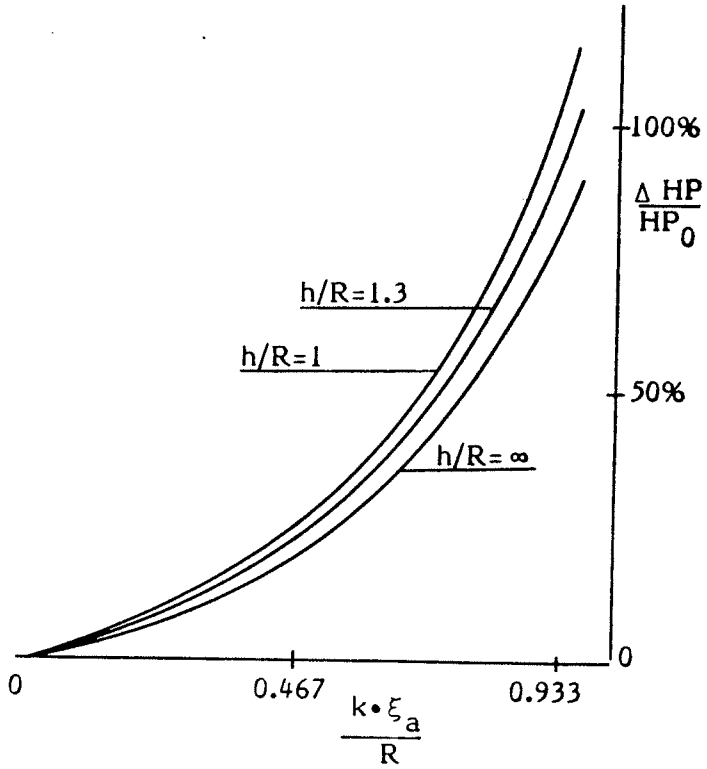
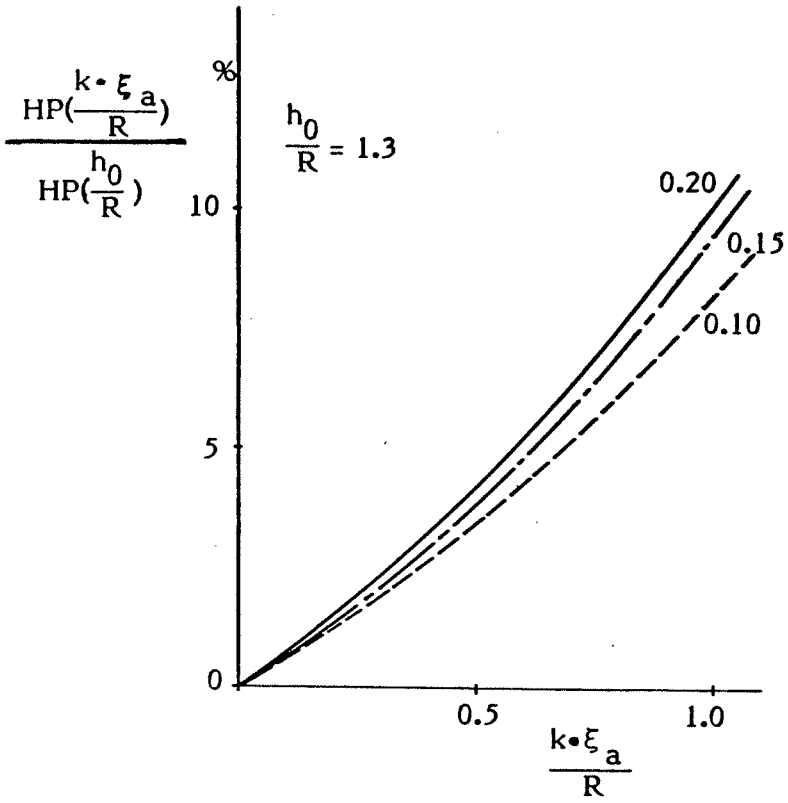


Fig. (42)

Increase in Power due to Variable Propeller Immersion



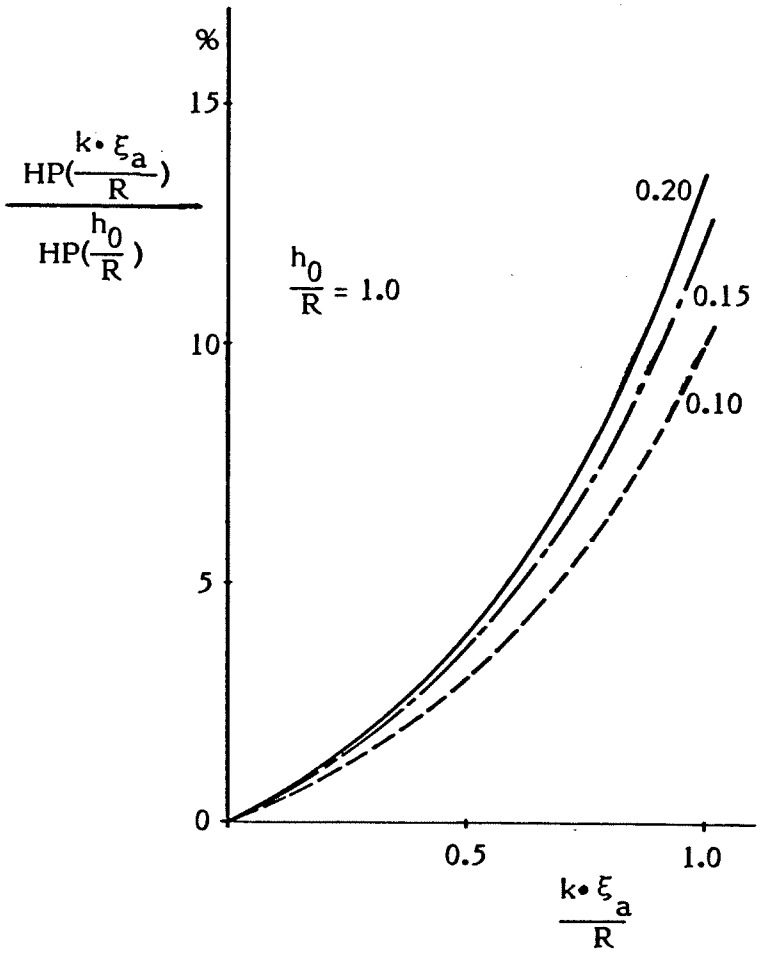
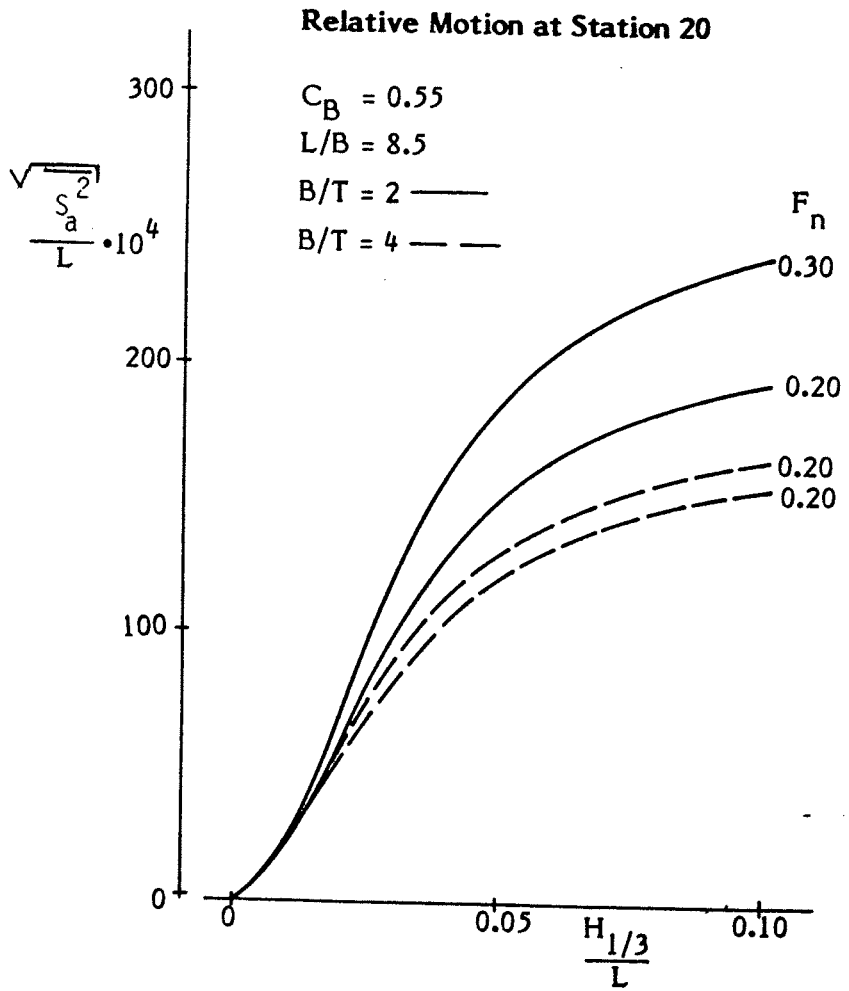


Fig. (43)


Relative Motion at Station 20
C_B = 0.55
L/B = 8.5
B/T = 2
B/T = 4
F_n
0.30
0.20
0.20
0.20
$\sqrt{\overline{S_a^2}}/L \cdot 10^4$
300
200
100
0
0
0.05
0.10
$H_{1/3}/L$

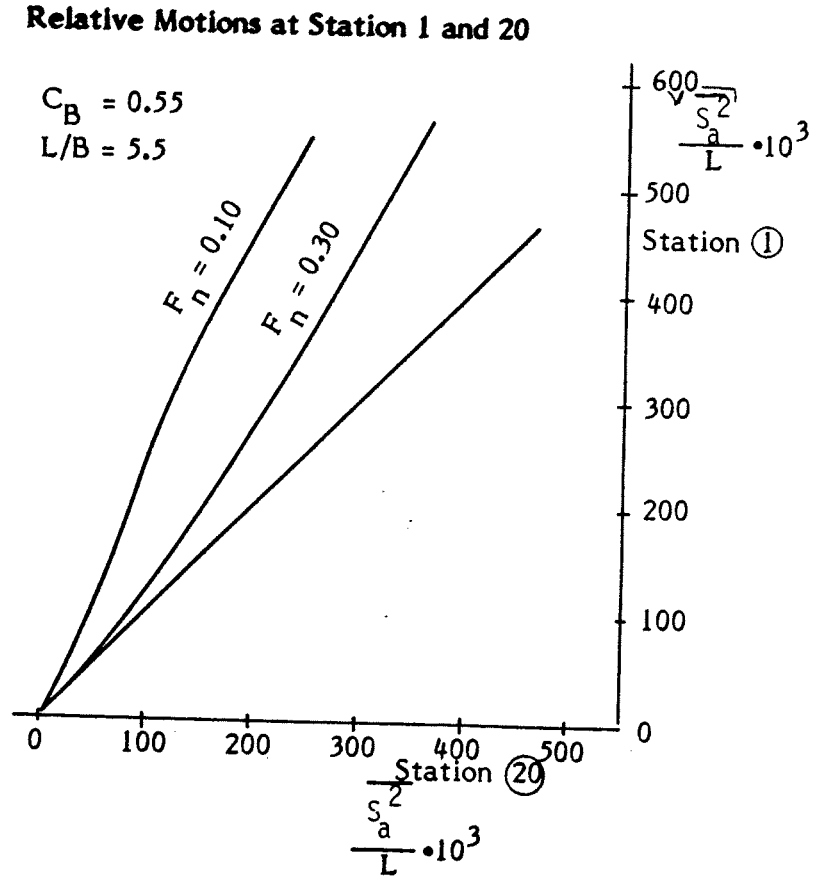
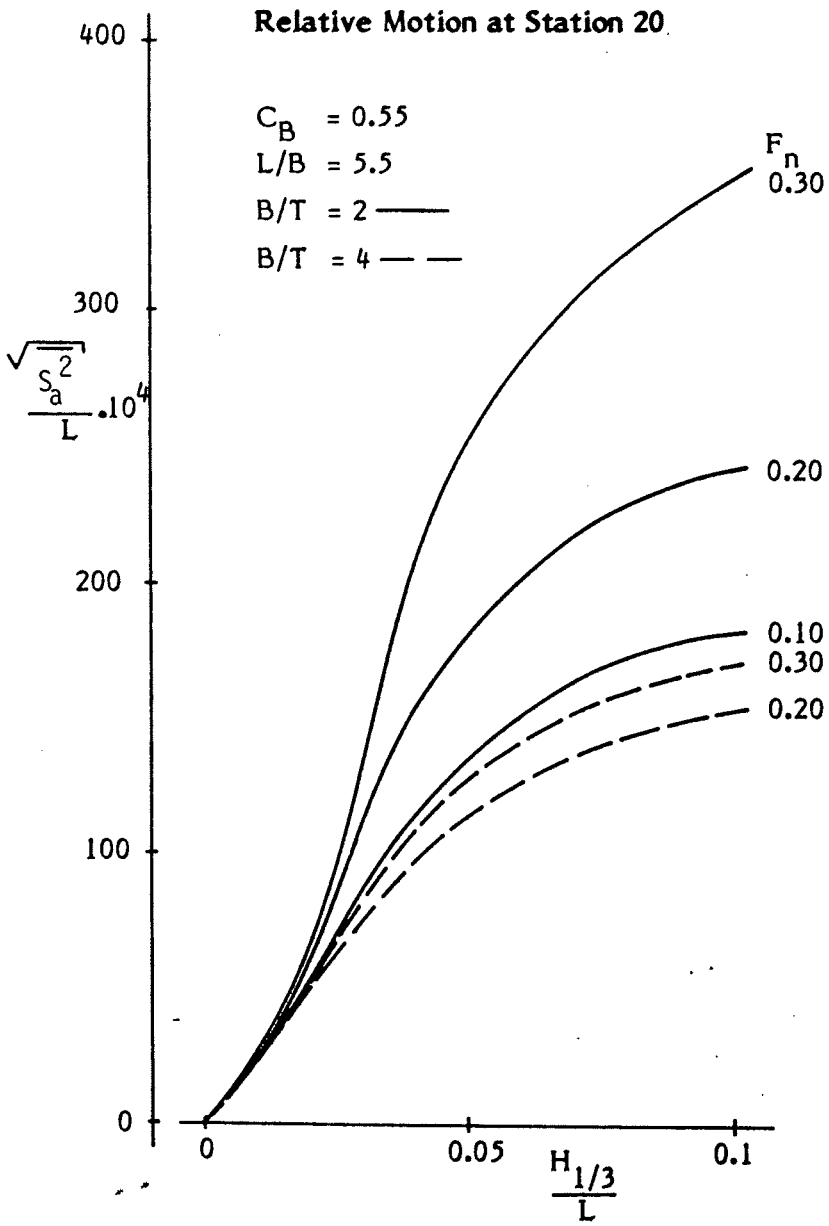

Relative Motions at Station 1 and 20
C_B = 0.55
L/B = 5.5
F_n = 0.10
F_n = 0.30
$\sqrt{\overline{S_a^2}}/L \cdot 10^3$
600
500
400
300
200
100
0
Station 1
0
100
200
300
400
500
Station 20
$\sqrt{\overline{S_a^2}}/L \cdot 10^3$

Fig. (44)


Relative Motion at Station 20
C_B = 0.55
L/B = 5.5
B/T = 2
B/T = 4
F_n
0.30
0.20
0.10
0.30
0.20
400
300
200
100
0
$\sqrt{\overline{S_a^2}}/L \cdot 10^4$
0
0.05
0.1
$H_{1/3}/L$

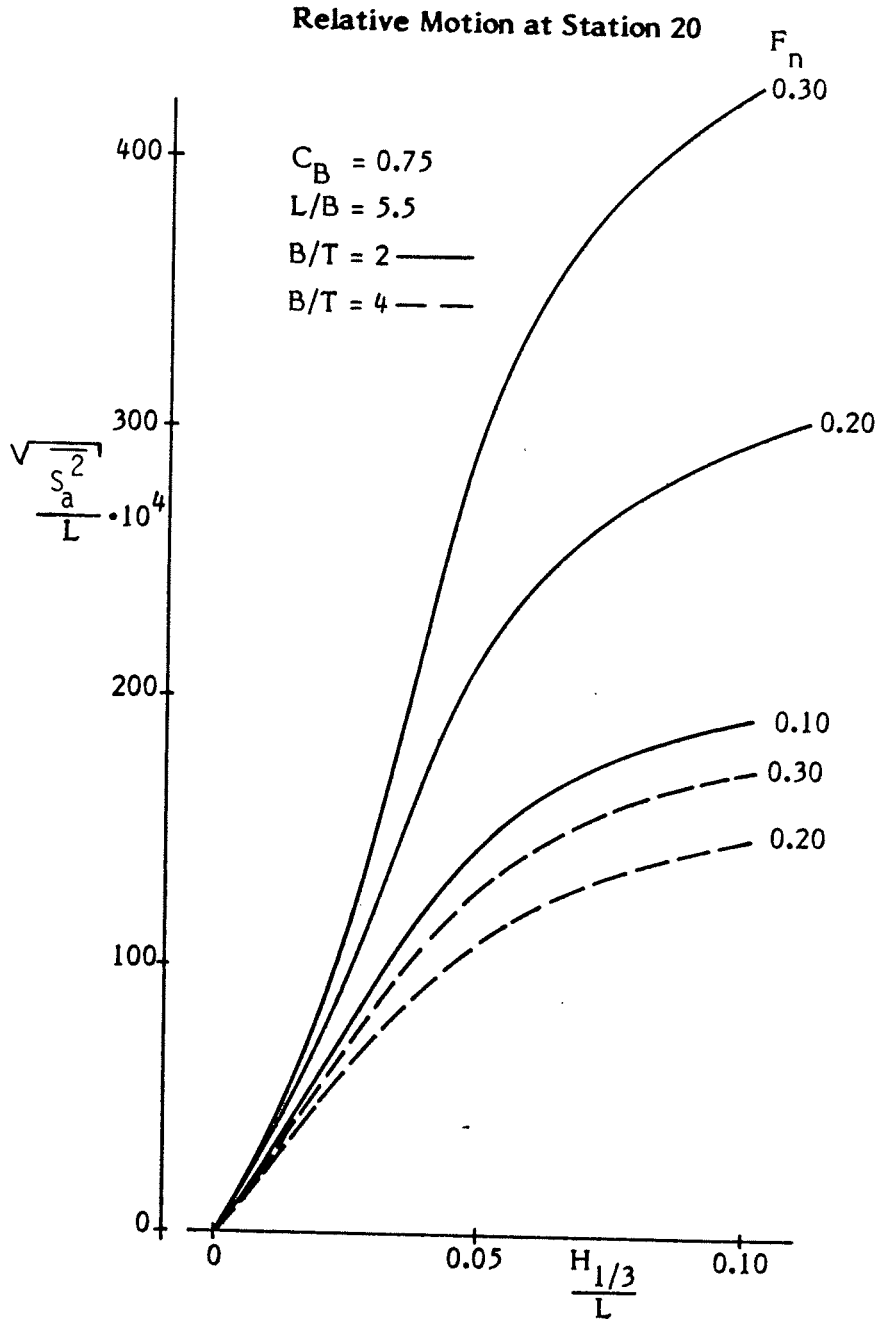

Relative Motion at Station 20
C_B = 0.75
L/B = 5.5
B/T = 2
B/T = 4
F_n
0.30
0.20
0.10
0.30
0.20
400
300
200
100
0
$\sqrt{\overline{S_a^2}}/L \cdot 10^4$
0
0.05
0.10
$H_{1/3}/L$

Fig. (45)

Changes in RPM and HP due to Relative Motions at constant Speed and Resistance

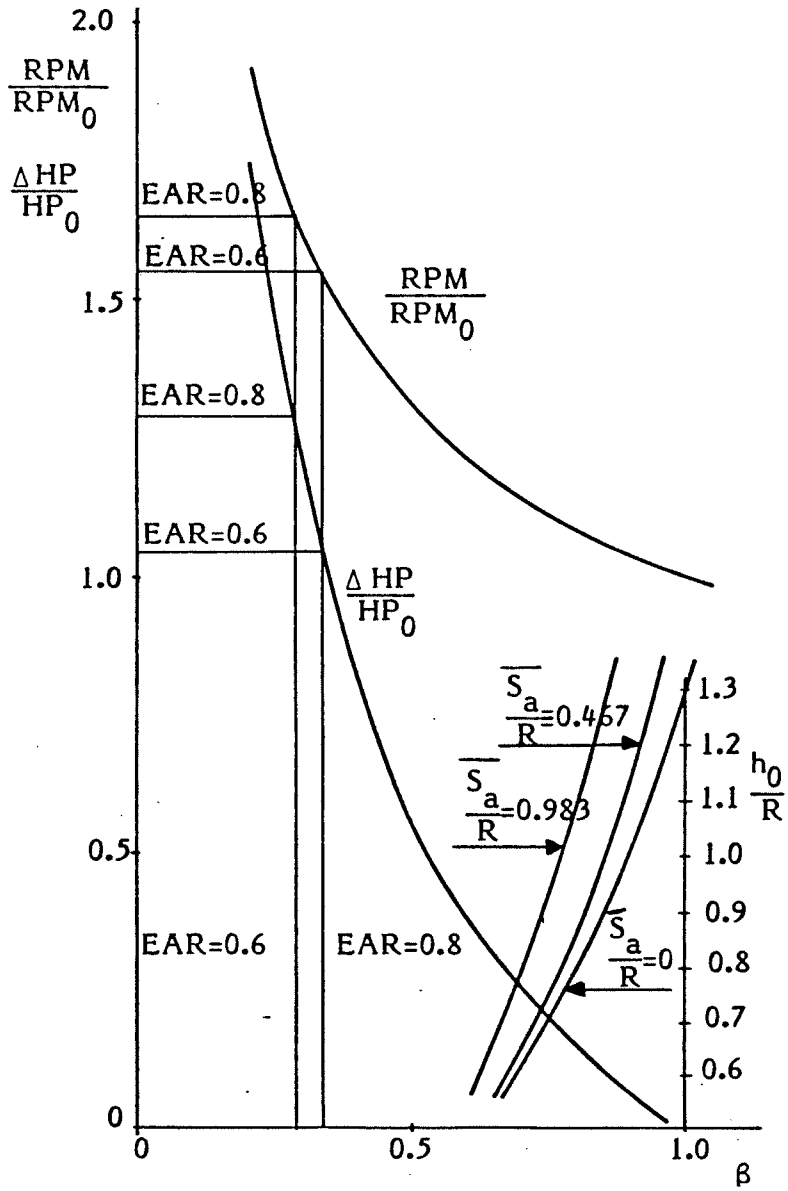


Fig. (46)

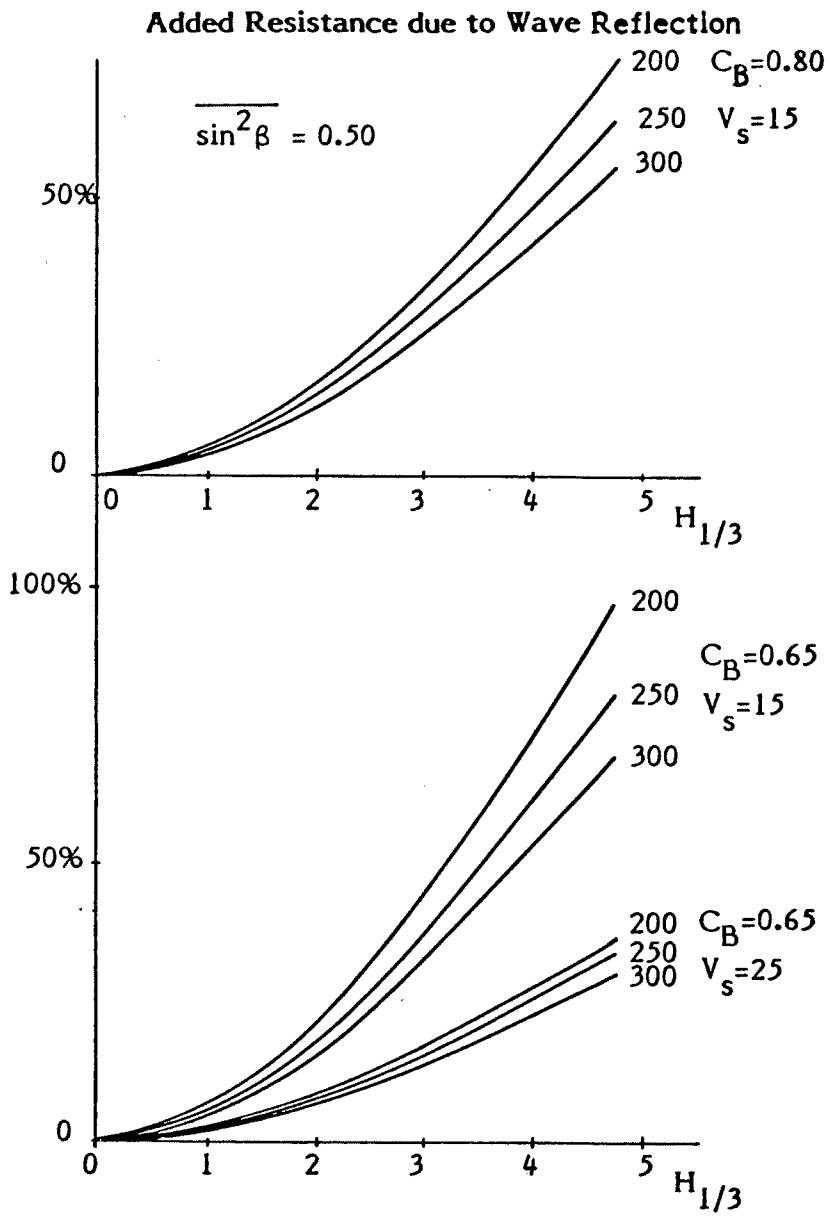

Added Resistance due to Wave Reflection
$\overline{\sin^2\beta}$ = 0.50
200
250
300
C_B=0.80
V_s=15
50%
0
0
1
2
3
4
5
$H_{1/3}$
100%
200
250
300
C_B=0.65
V_s=15
50%
200
250
300
C_B=0.65
V_s=25
0
0
1
2
3
4
5
$H_{1/3}$

Fig. (47)

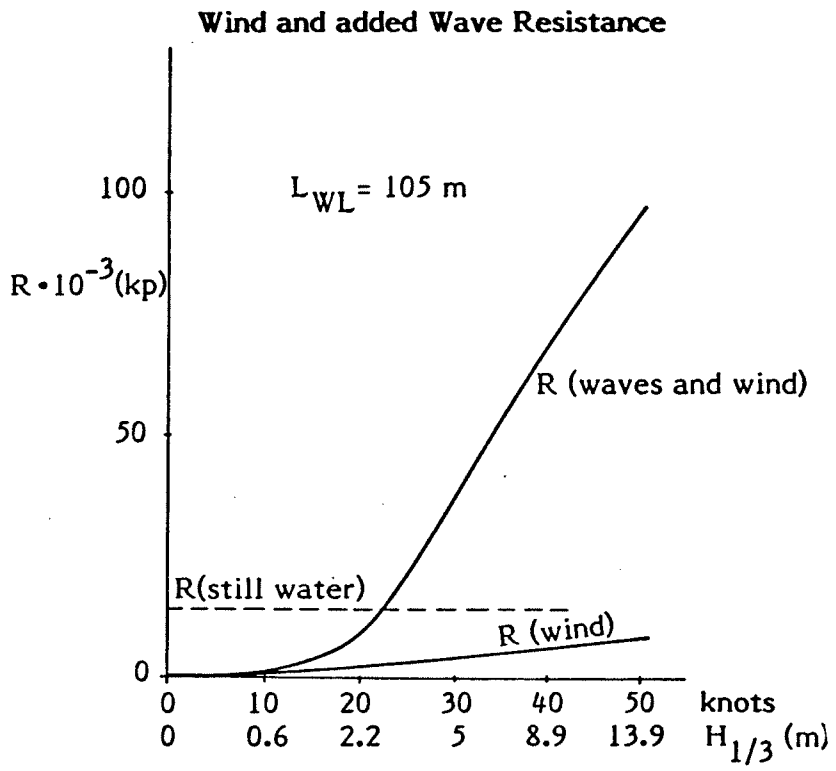
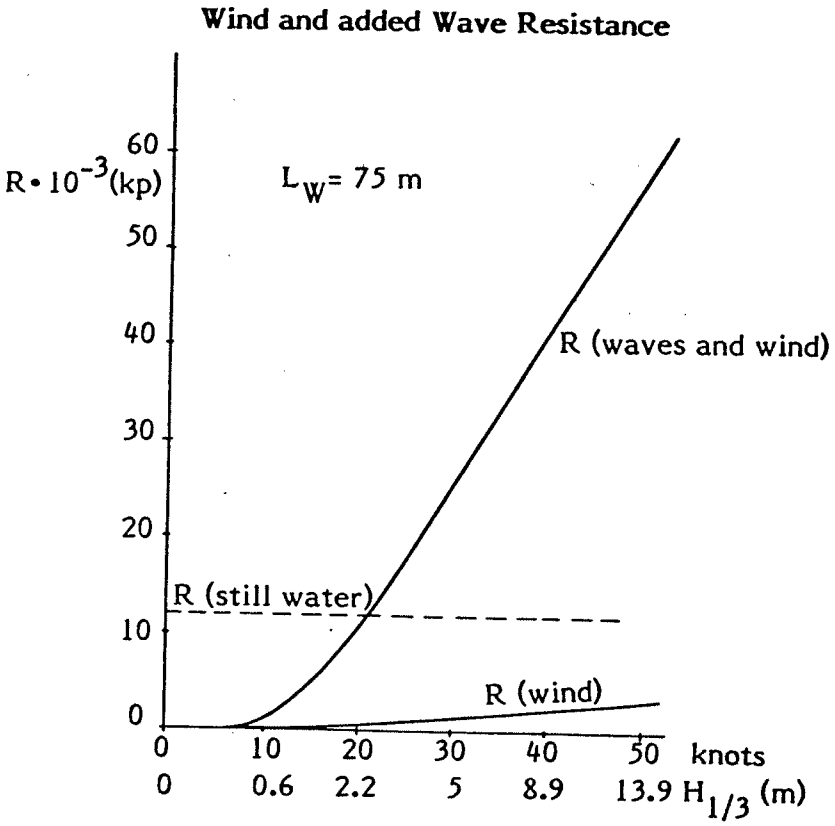

Wind and added Wave Resistance
L_{WL}= 105 m
100
$R \cdot 10^{-3}$(kp)
R (waves and wind)
50
R(still water)
R (wind)
0
0
10
20
30
40
50
knots
0
0.6
2.2
5
8.9
13.9
$H_{1/3}$ (m)

Fig. (48)


Wind and added Wave Resistance
R • 10^-3 (kp)
60
50
40
30
20
10
0
L_W = 75 m
R (waves and wind)
R (still water)
R (wind)
0 10 20 30 40 50 knots
0 0.6 2.2 5 8.9 13.9 H_1/3 (m)

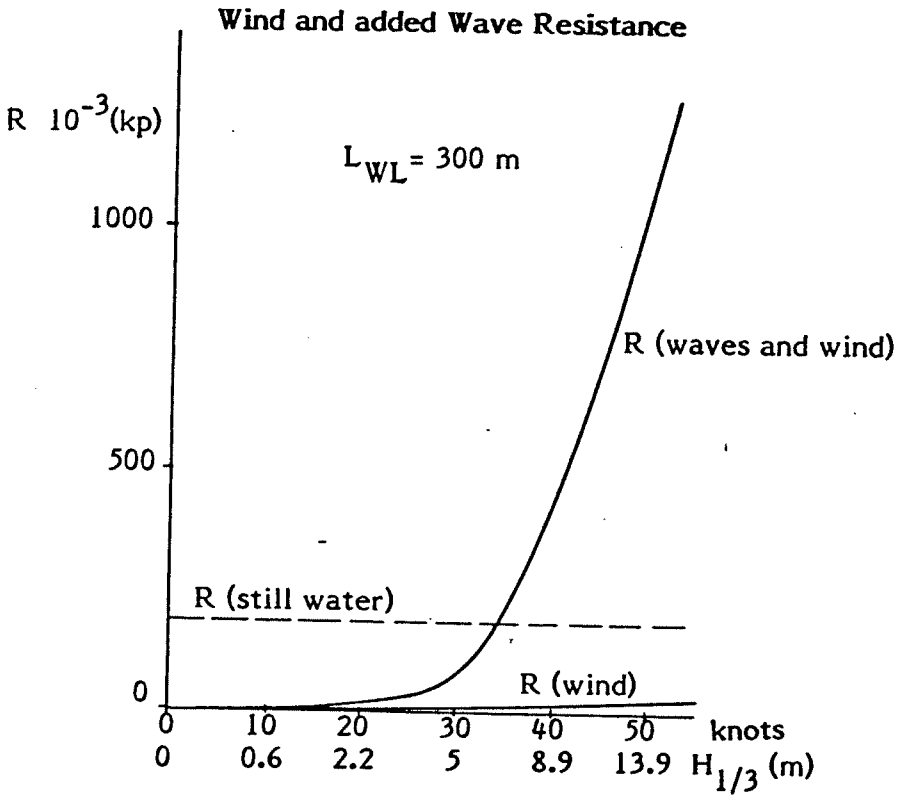

Wind and added Wave Resistance
R 10^-3 (kp)
1000
500
0
L_WL = 300 m
R (waves and wind)
R (still water)
R (wind)
0 10 20 30 40 50 knots
0 0.6 2.2 5 8.9 13.9 H_1/3 (m)

Fig. (49)

Power - RPM Diagram

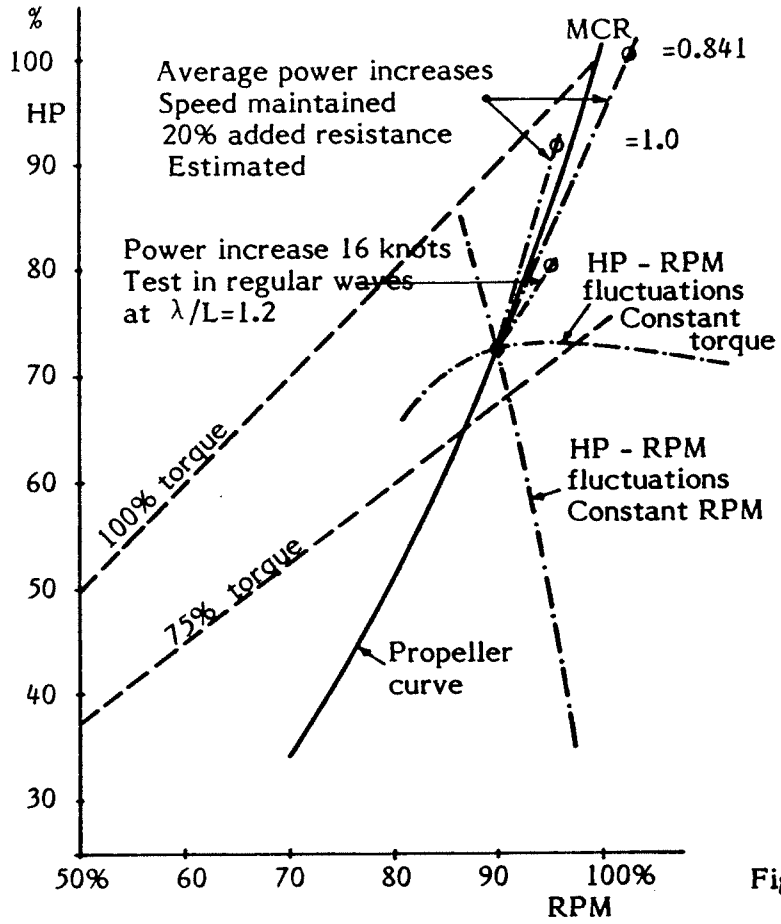


Fig. (50)

Increase in RPM due to Relative Motions at constant Resistance

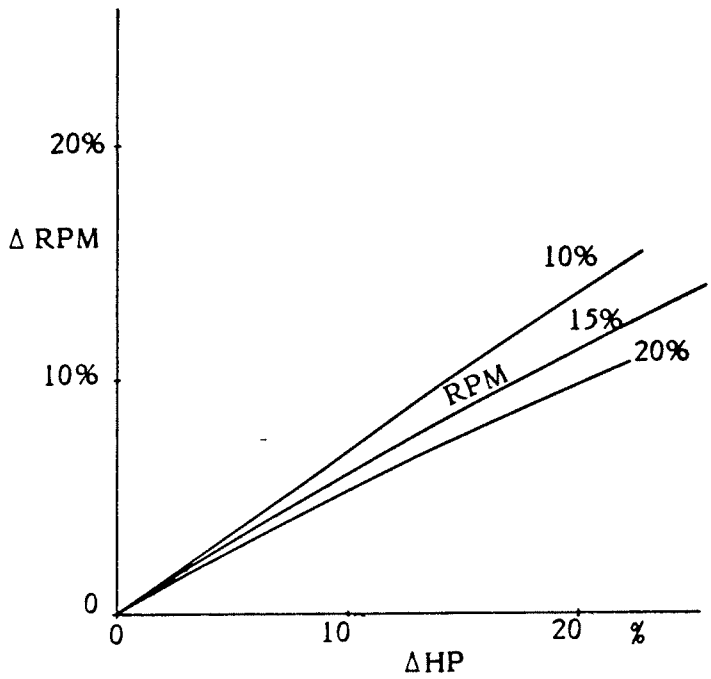
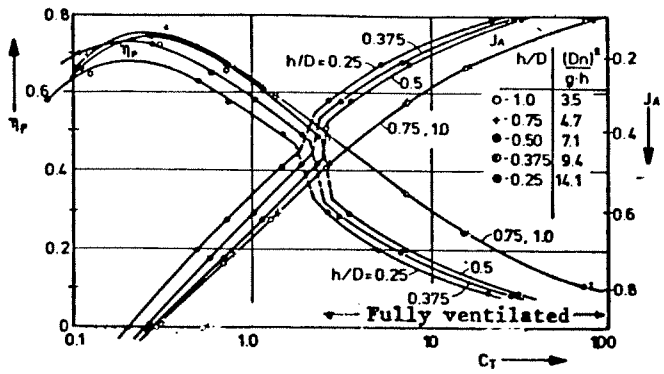


Fig. (51)

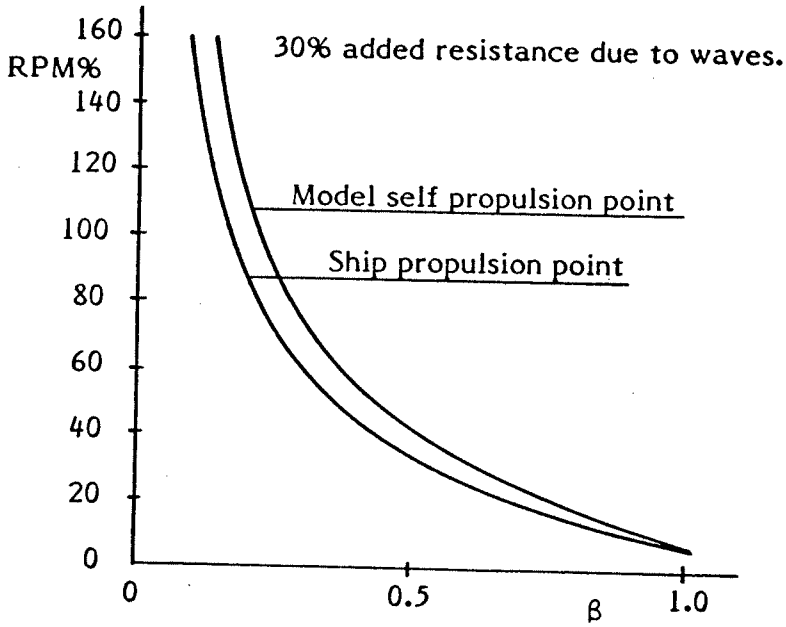


Change in propeller efficiency due to propeller loading, immersion and ventilation (from Gutsche)

Fig. (52)

Increase of Power and RPM due to Surface Effects

Constant Thrust and Speed



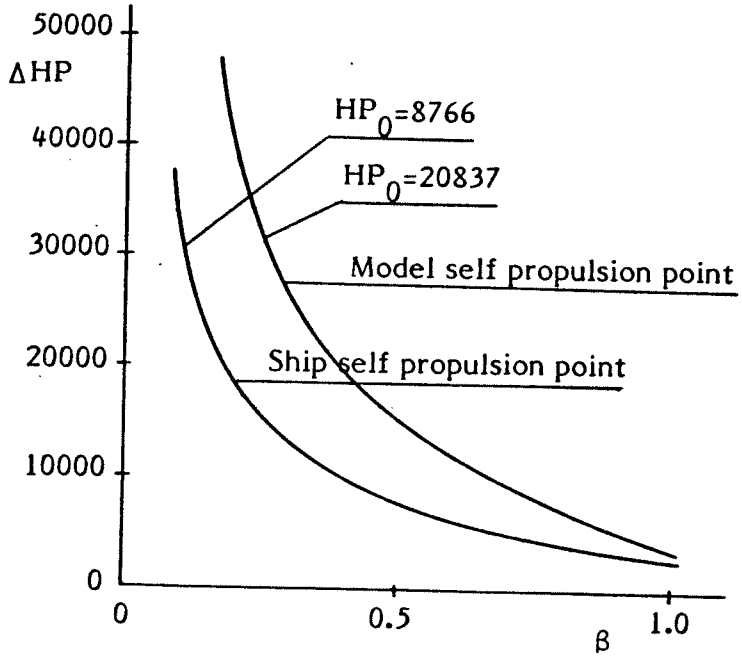


Fig. (53)

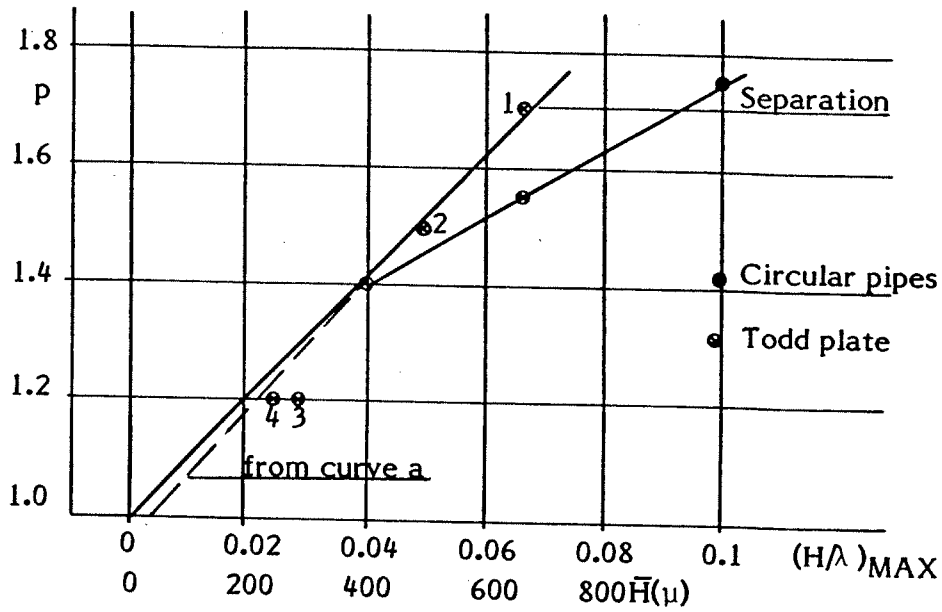

1.8
P
1.6
1.4
1.2
1.0
1
2
4 3
Separation
Circular pipes
Todd plate
from curve a
0
0.02
0.04
0.06
0.08
0.1
(H/λ)MAX
0
200
400
600
800H̄(μ)

Fig.(54)

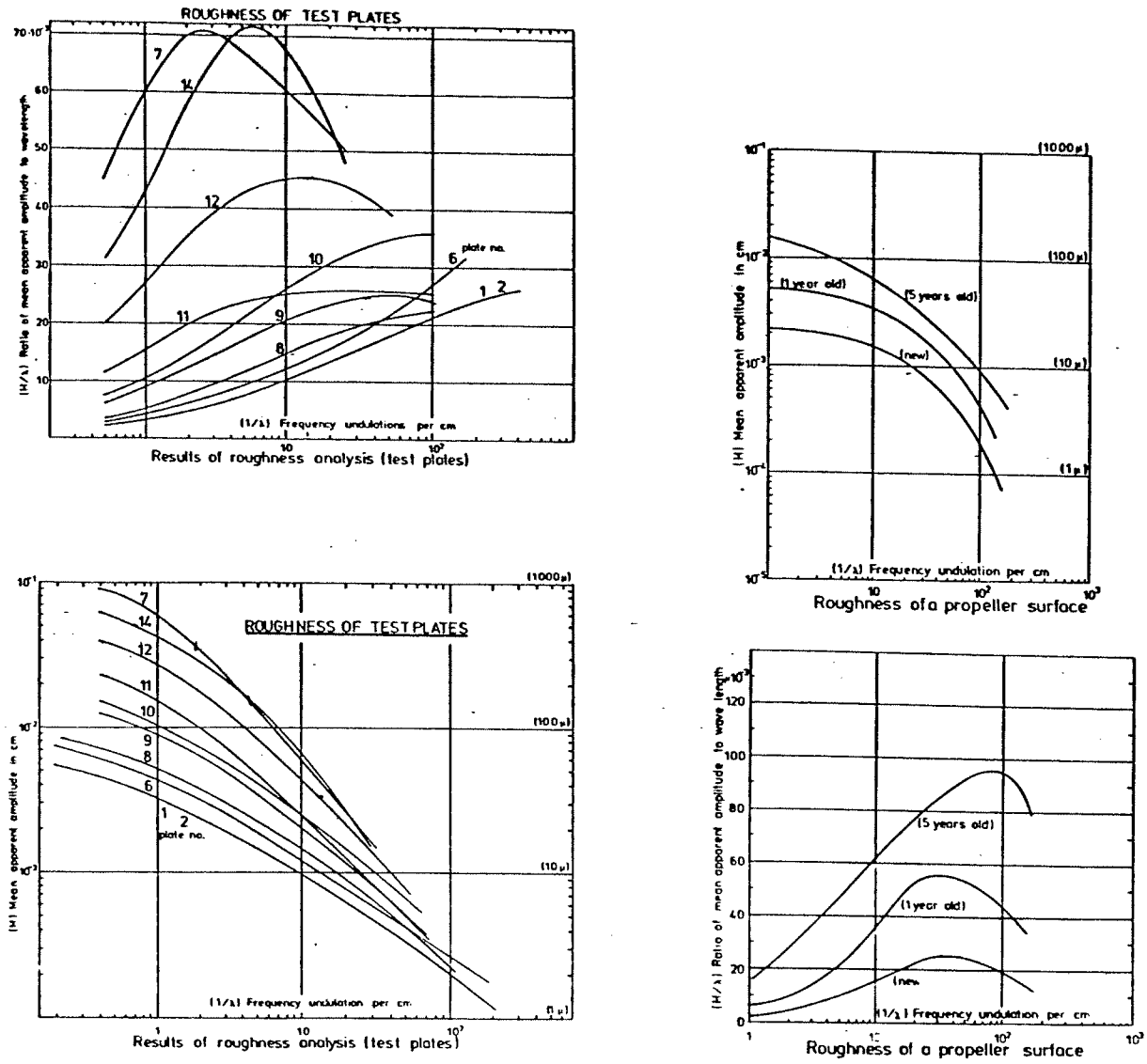
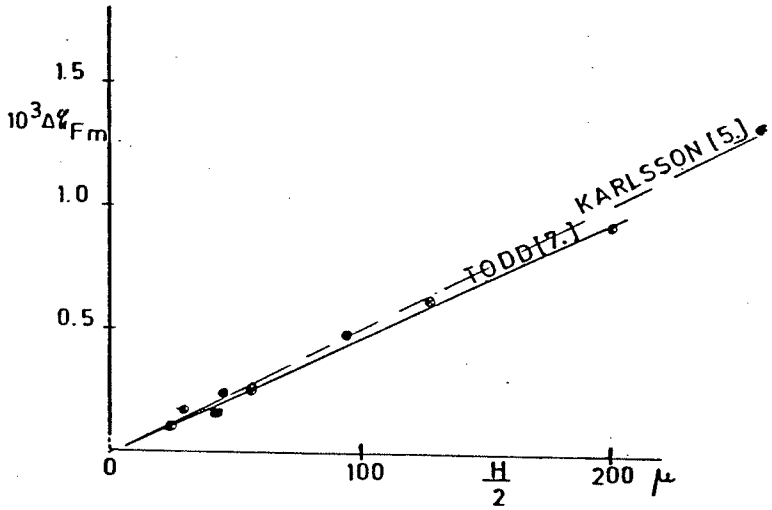
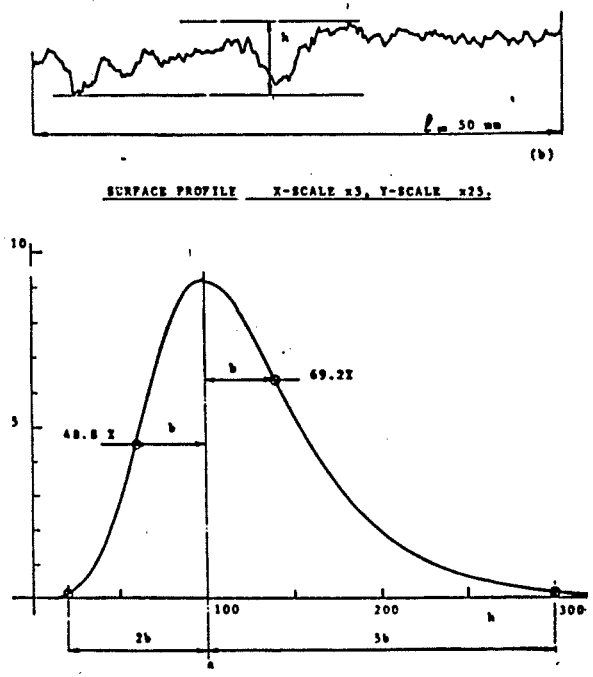


Fig. (55)



LOG R_n = 7.5 λ = 52.5 mm

Fig. (56)



From Grigson

Fig. (57)

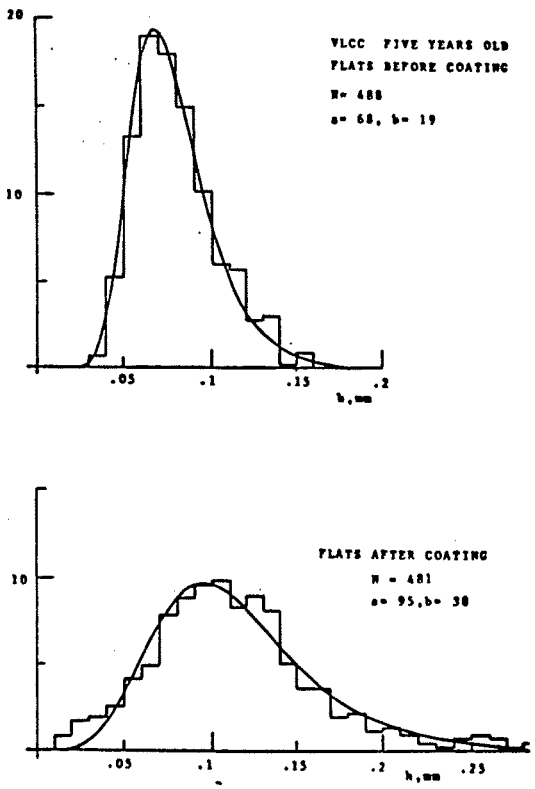
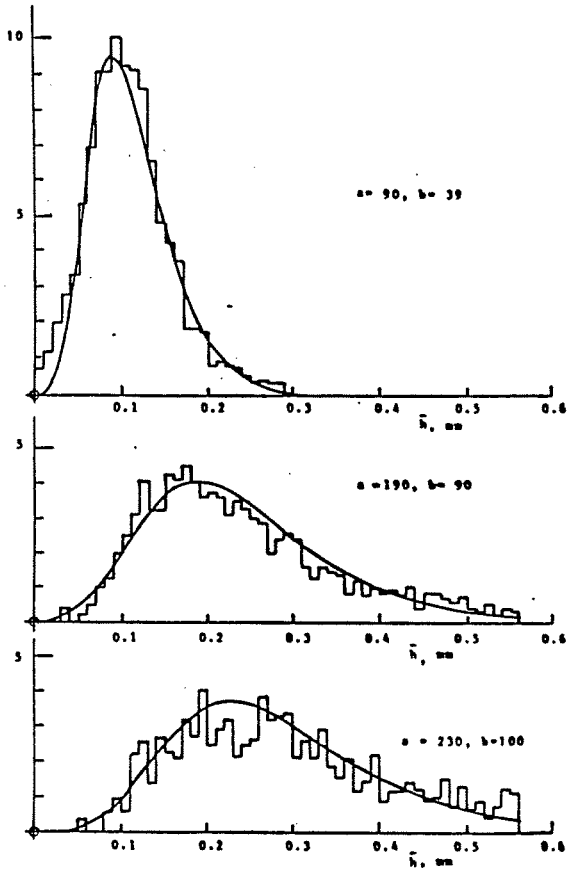


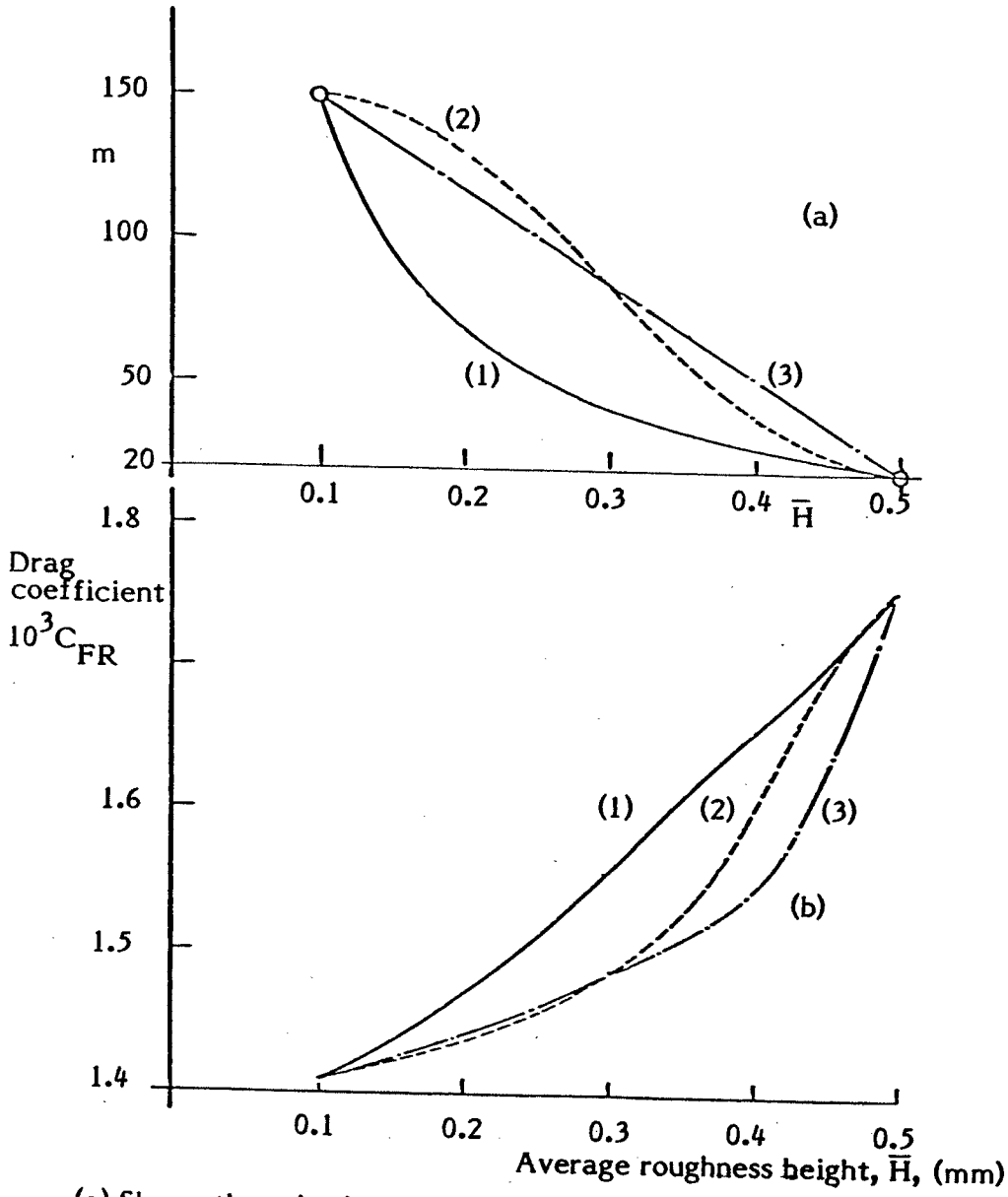
Fig. (59)



THREE NEW-BUILDINGS COATED BY AIRLESS SPRAY UNDER COVER.

Fig. (58)

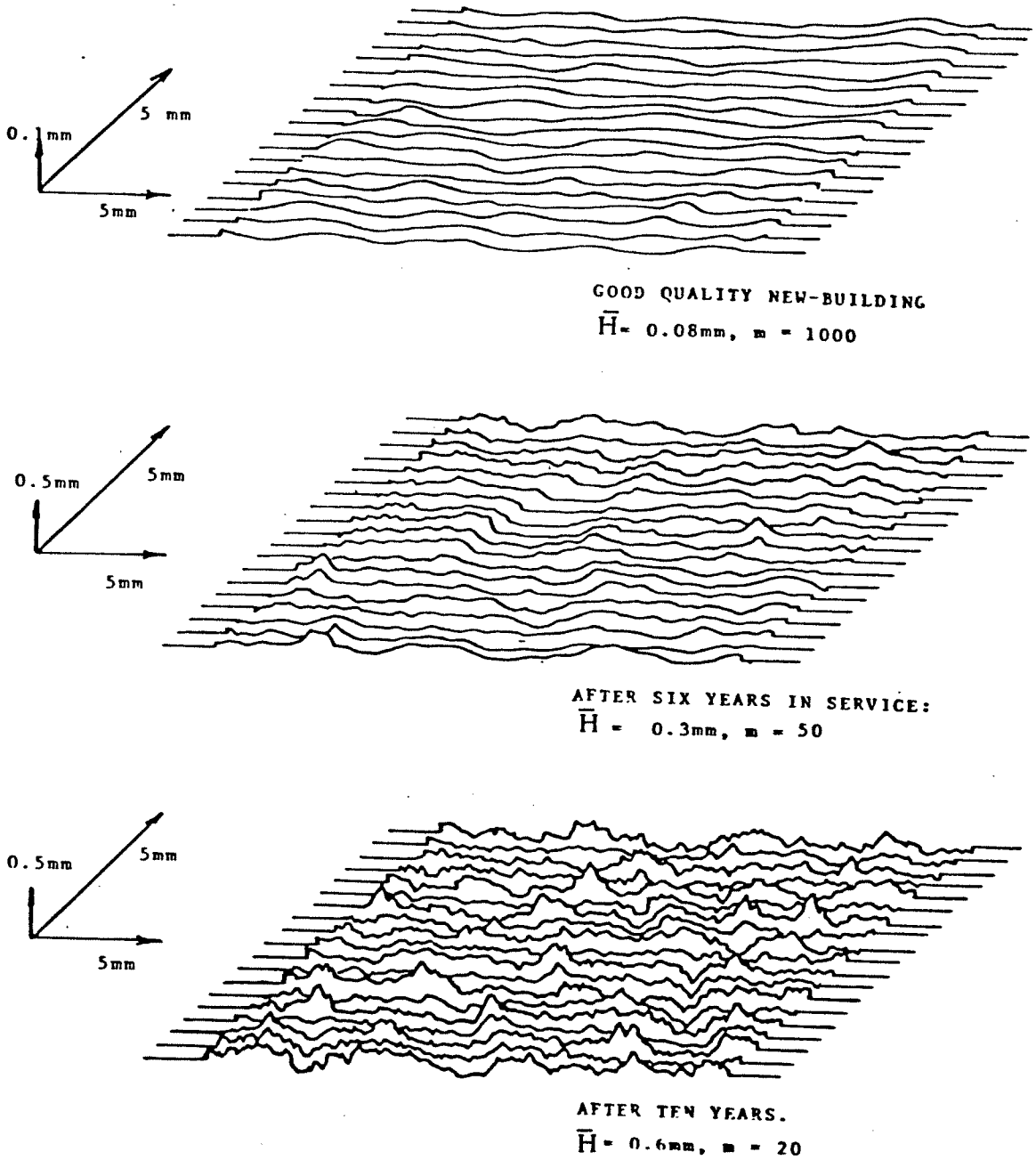
Relation Texture - Roughness



(a) Shows three basic ways in which m might change as $\bar{H}$ increases.
(b) The corresponding changes in the drag coefficient.
Vessel of length 250 m in tropical seas at 30°C., $R_L = 2.05 \times 10^9$.

Fig. (60)

Build Up of Roughness and Change in Texture

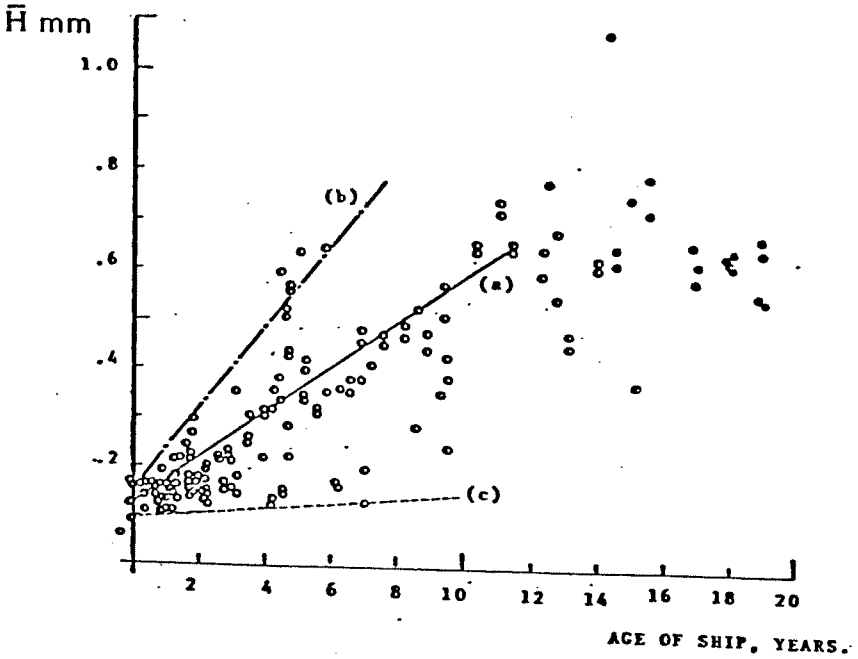


Illustrating the build up of roughness by increase in height and change of texture

From Grigson [4]

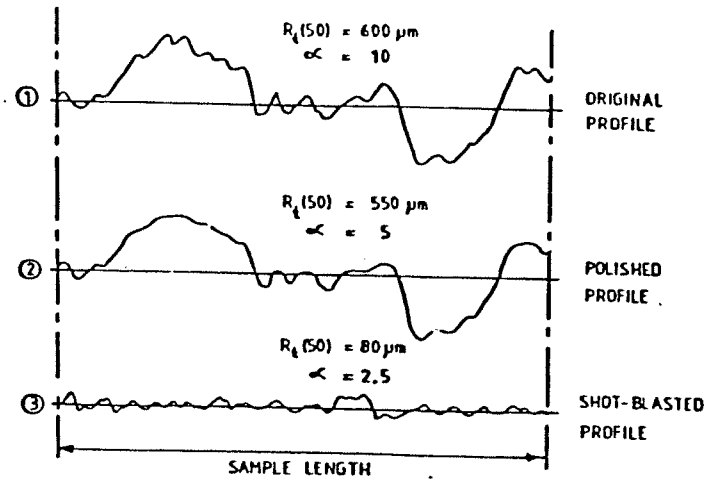
Fig. (61)

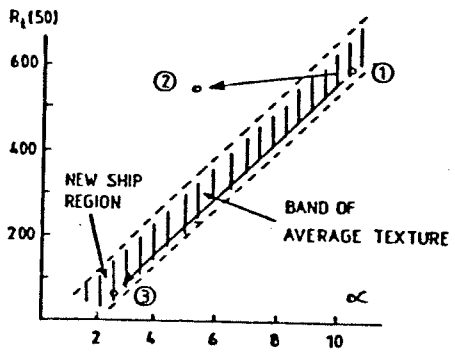
AVERAGE HULL ROUGHNESS, HEIGHT



INCREASE OF THE HEIGHT OF HULL ROUGHNESS WITH AGE.
(Data from Townsin et al.)
Solid circles indicate older ships not likely to have been built with shot-blasted plate.

Fig. (62)





BSRA roughness measure $R_t(50)$ and the texture parameter α

From Townsin [37]

Fig. (63)

Sasajima - Yoshida

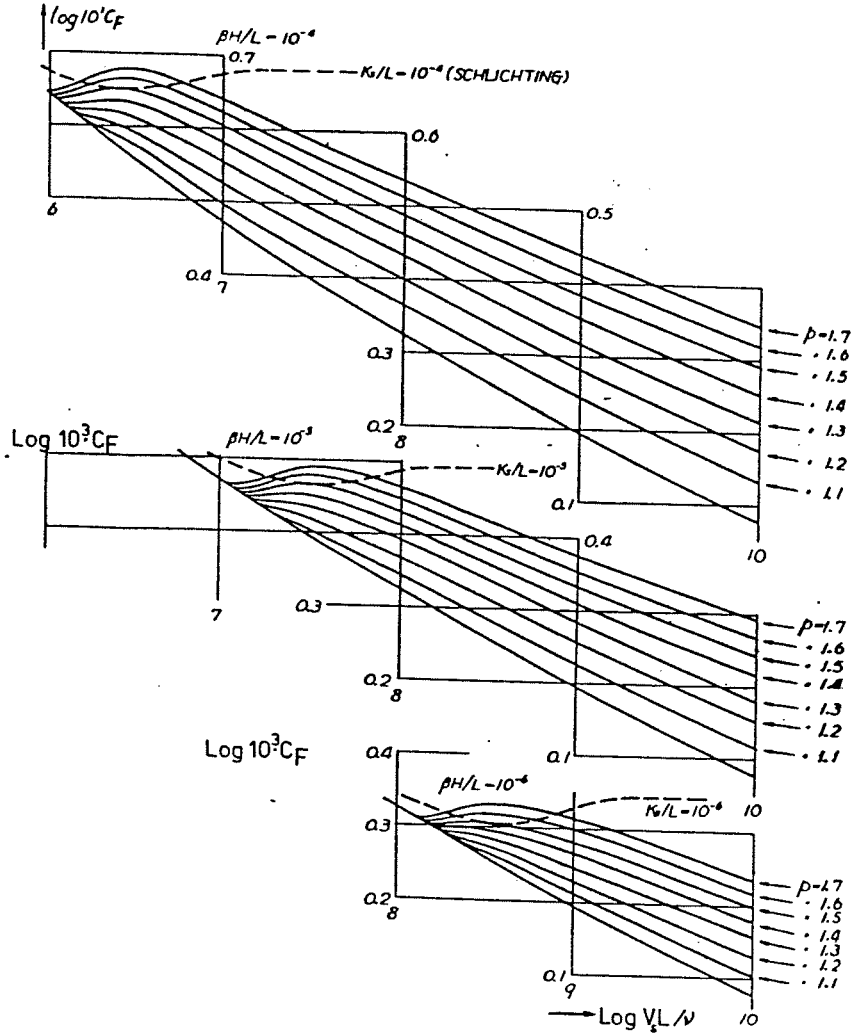


Fig. (65)

Karlsson CTH

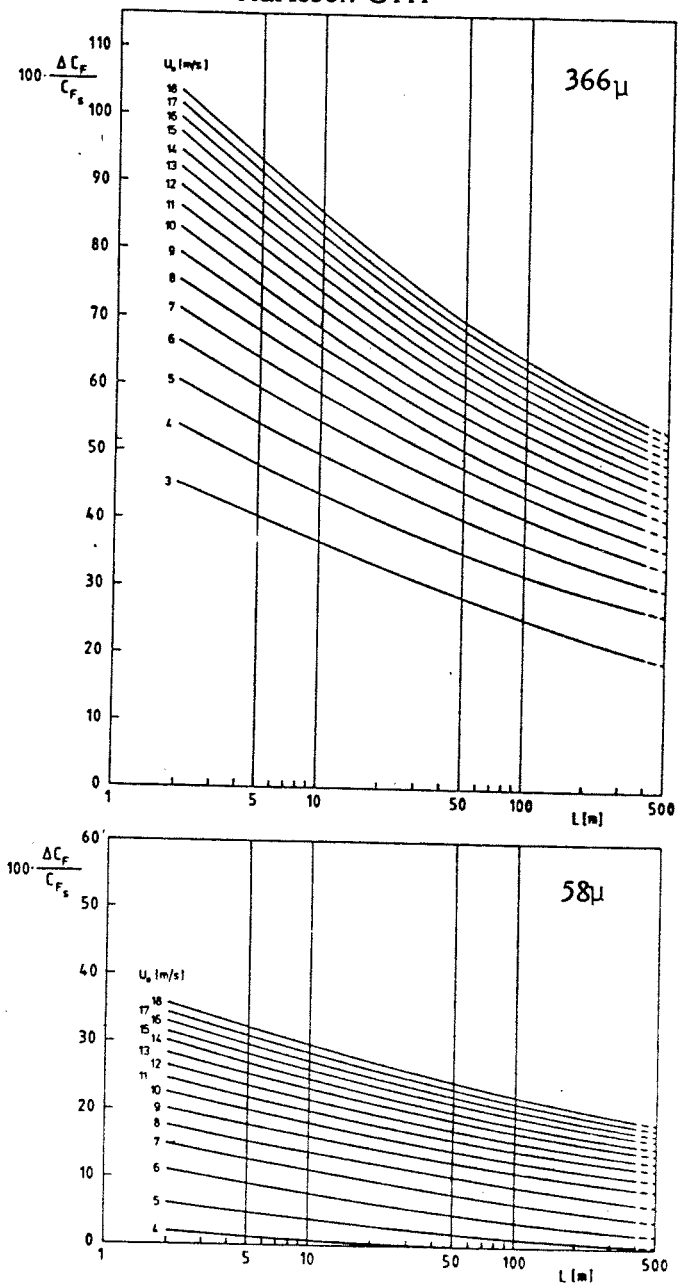


Fig. (64)

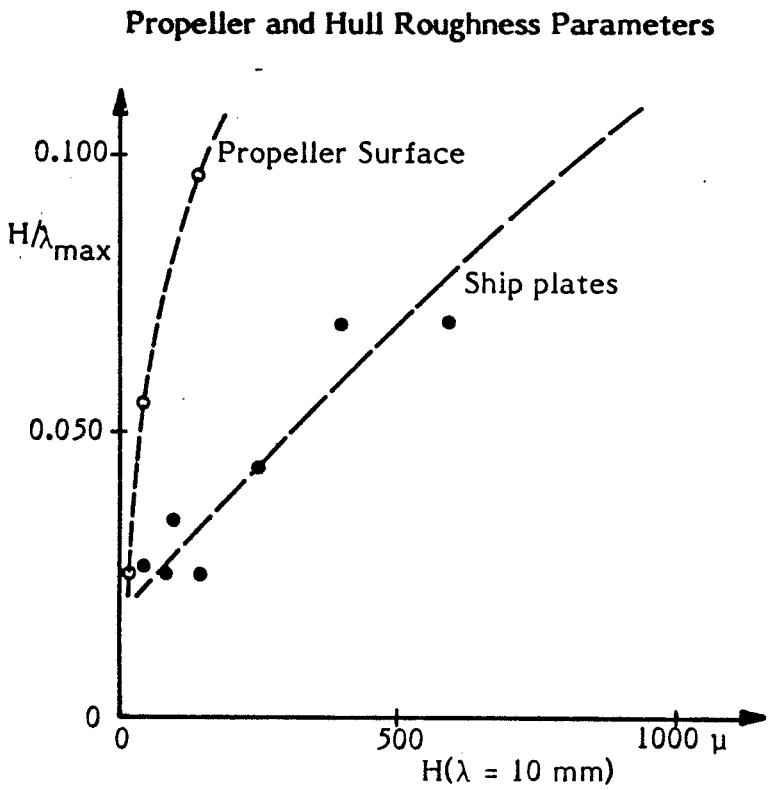

Propeller and Hull Roughness Parameters
0.100
0.050
0
H/λmax
Propeller Surface
Ship plates
0
500
1000 μ
H(λ = 10 mm)

Fig. (67)

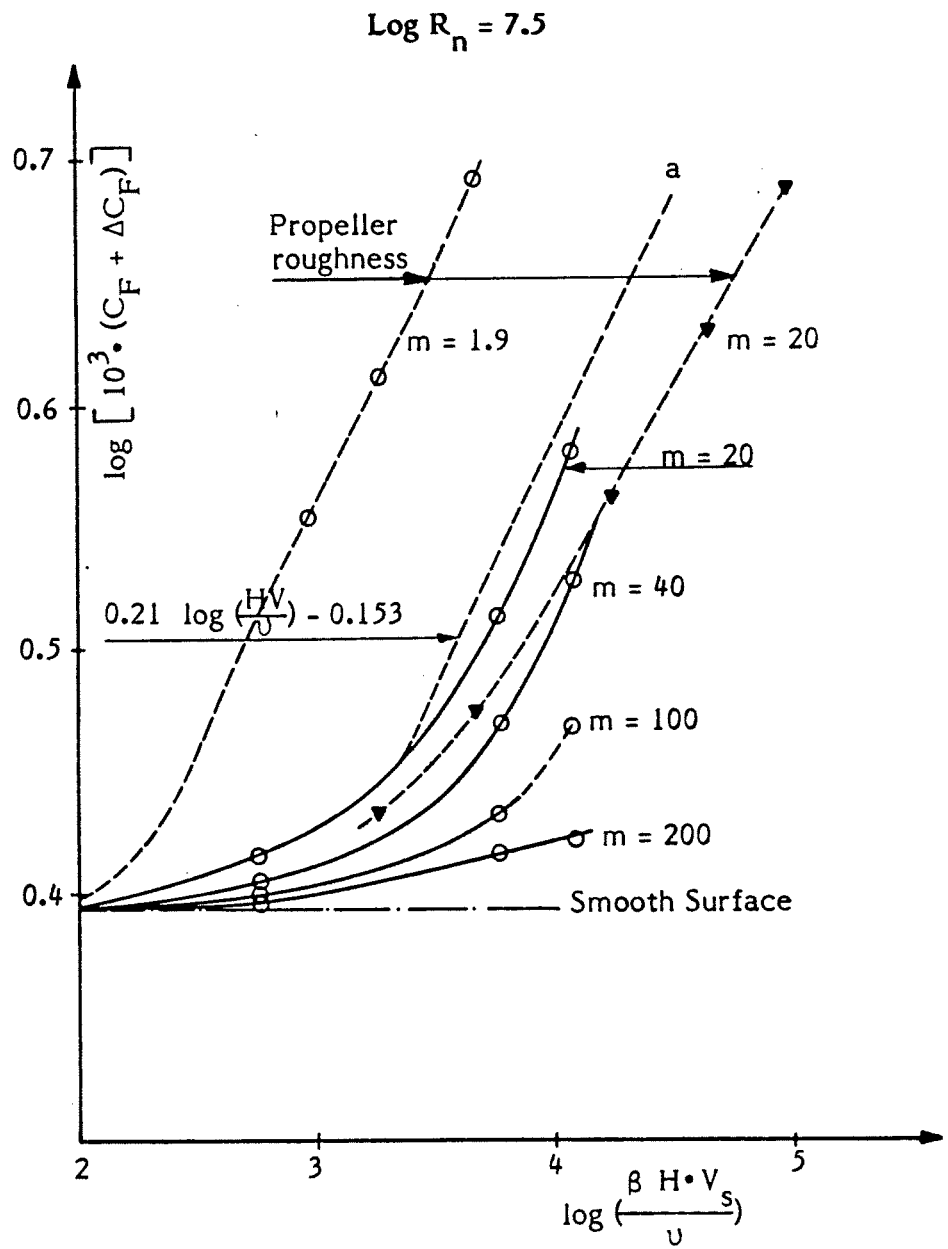
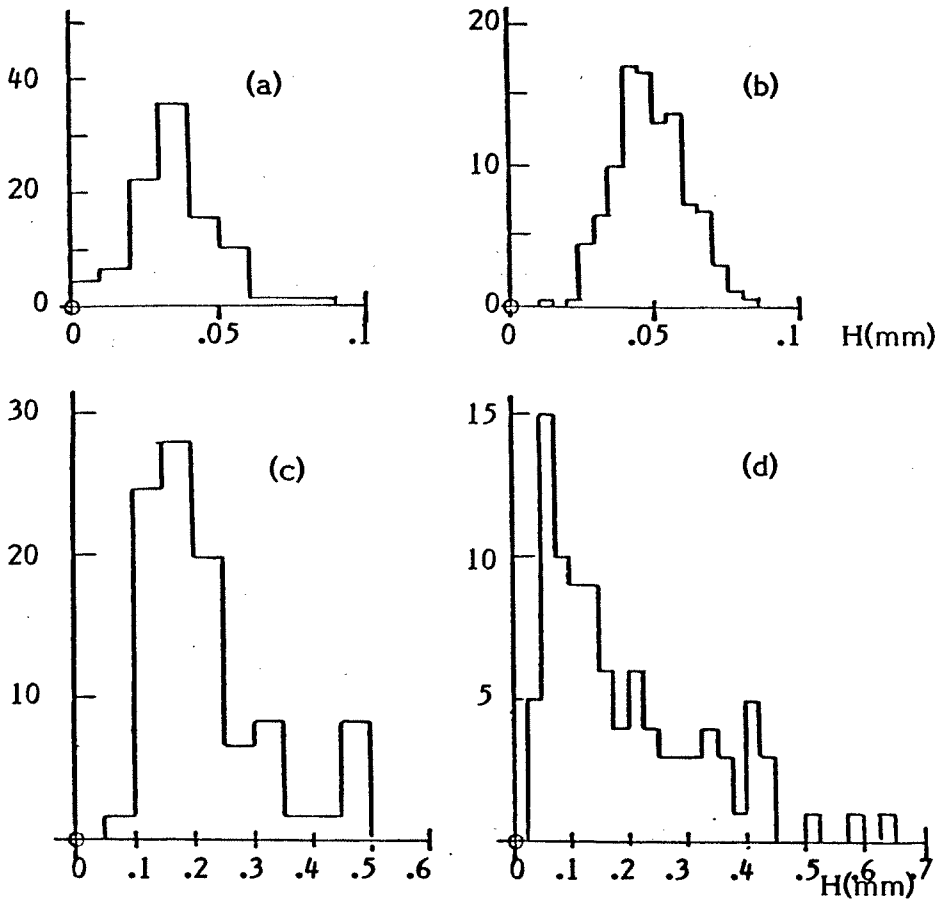

Log Rn = 7.5
0.7
0.6
0.5
0.4
log [10³ • (CF + ΔCF)]
Propeller roughness
a
m = 1.9
m = 20
m = 20
m = 40
m = 100
m = 200
0.21 log (HV/υ) - 0.153
Smooth Surface
2
3
4
5
log (β H • Vs / υ)

Fig. (66)

Histograms of Roughness for four High-power Screws



(a) and (b) 32.000 hp with reconditioned surfaces, 5 and 6 years.
(c) 40.000 hp, 2-1/2 years, high block coefficient hull.
(d) 45.000 hp, 7 years, slender hull.
Measurements by A. Milne.

Note that the true values of H must all be higher than those of the histograms, because of the large radius of the measuring probe.

Fig. (68)

Influence of Propeller Roughness (Gutsche)

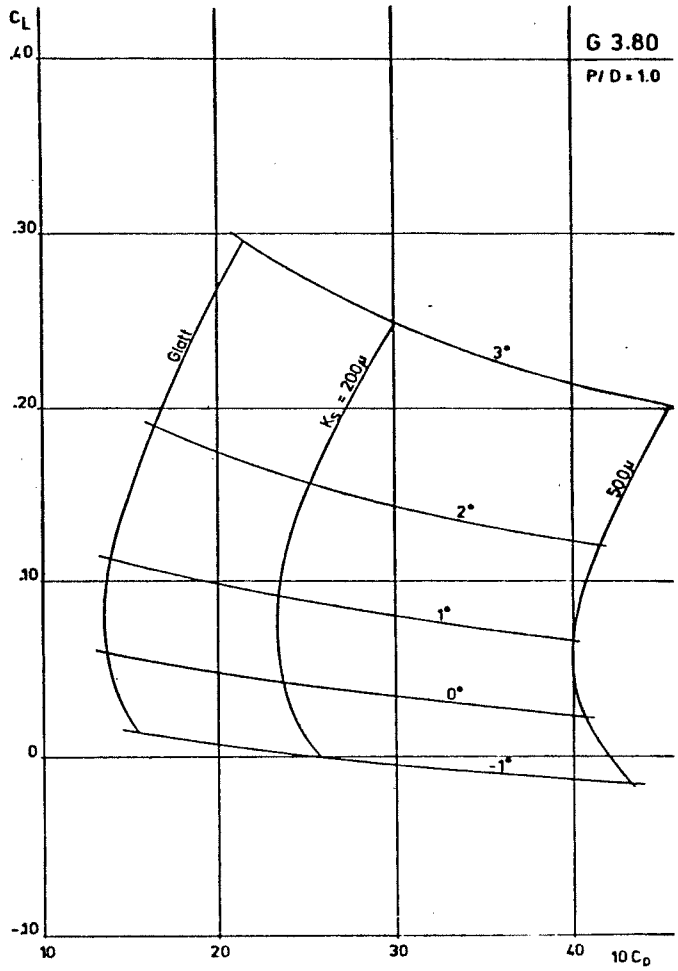


Fig. (69)

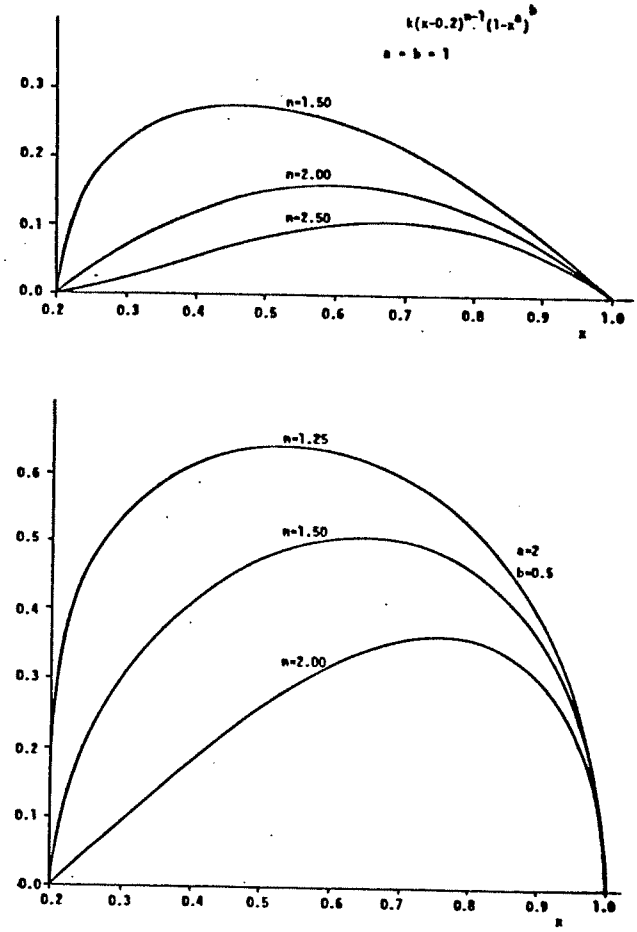
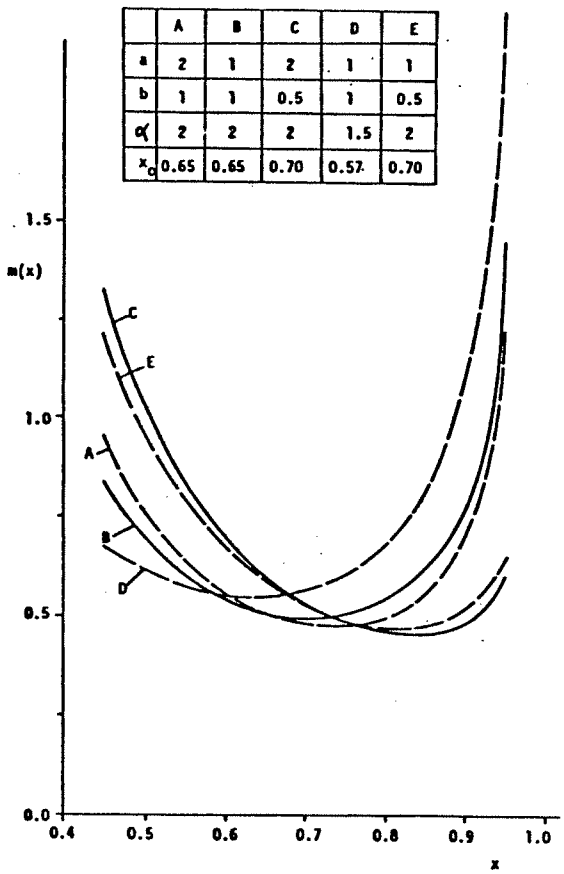


Fig. (70)



	A	B	C	D	E
a	2	1	2	1	1
b	1	1	0.5	1	0.5
α	2	2	2	1.5	2
x_c	0.65	0.65	0.70	0.57	0.70

Fig. (71)

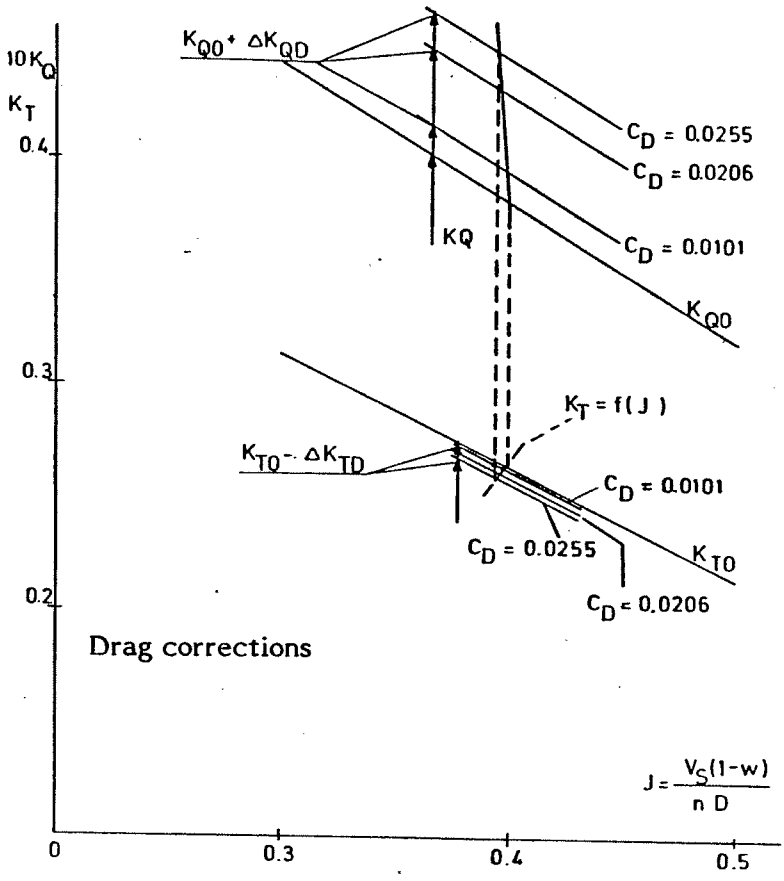

10K_Q
K_T
0.4
0.3
0.2
K_QO + ΔK_QD
K_Q
C_D = 0.0255
C_D = 0.0206
C_D = 0.0101
K_QO
K_T = f(J)
K_TO − ΔK_TD
C_D = 0.0101
C_D = 0.0255
K_TO
C_D = 0.0206
Drag corrections
J = V_S(1-w) / n D
0
0.3
0.4
0.5

Fig. (72)

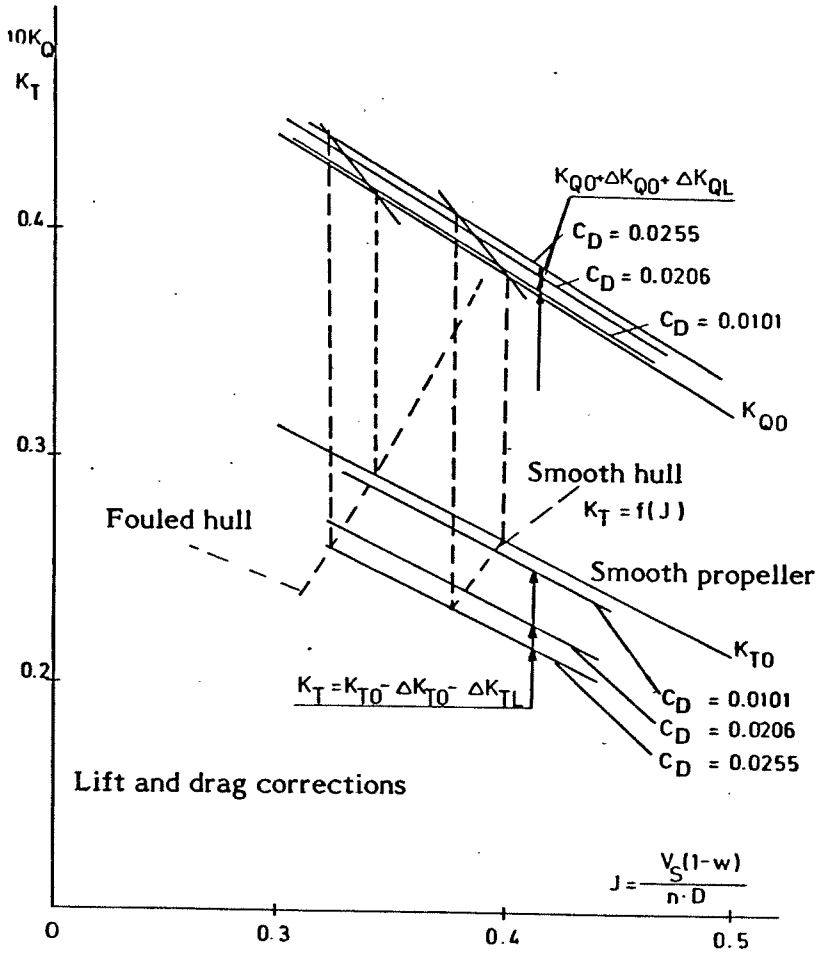

10K_Q
K_T
0.4
0.3
0.2
K_QO+ΔK_QO+ ΔK_QL
C_D = 0.0255
C_D = 0.0206
C_D = 0.0101
K_QO
Smooth hull
K_T = f(J)
Fouled hull
Smooth propeller
K_TO
K_T = K_TO − ΔK_TO − ΔK_TL
C_D = 0.0101
C_D = 0.0206
C_D = 0.0255
Lift and drag corrections
J = V_S(1-w) / n·D
0
0.3
0.4
0.5

Fig. (73)

Influence of Propeller Roughness on Power
(Drag and Lift Correction)

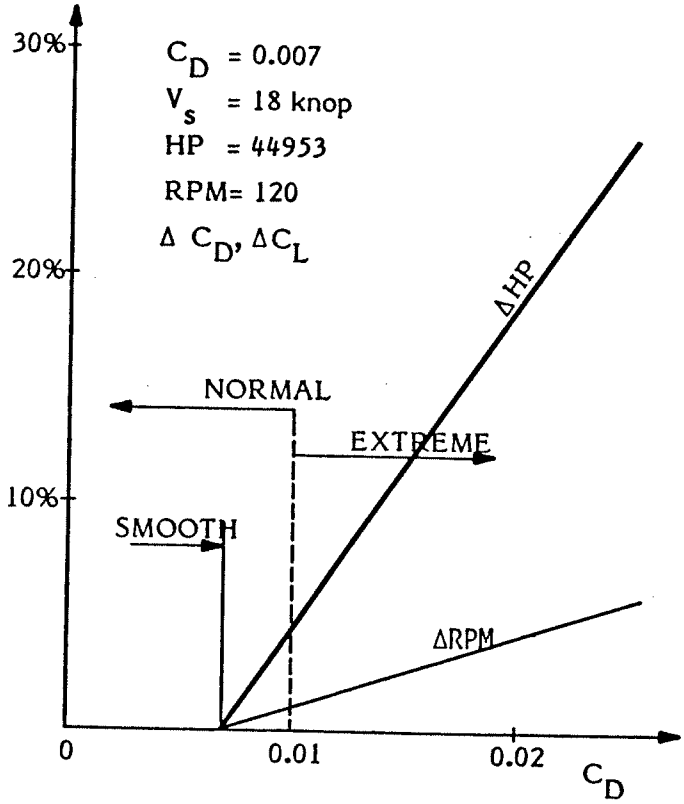


Fig. (74)

Influence of Propeller Roughness on Power
(Drag Correction)

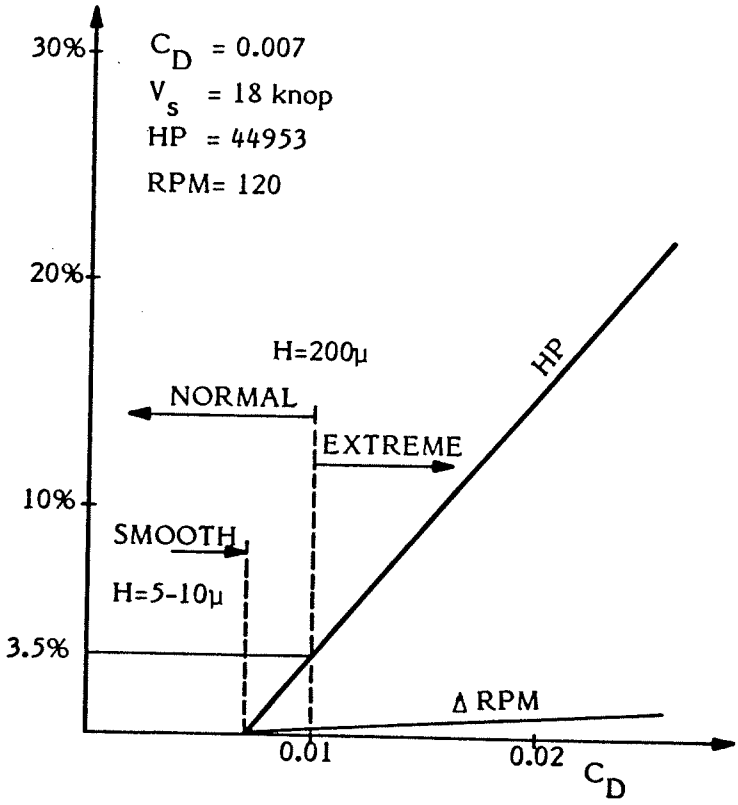


Fig. (75)

ROUGHNESS DIAGRAM.

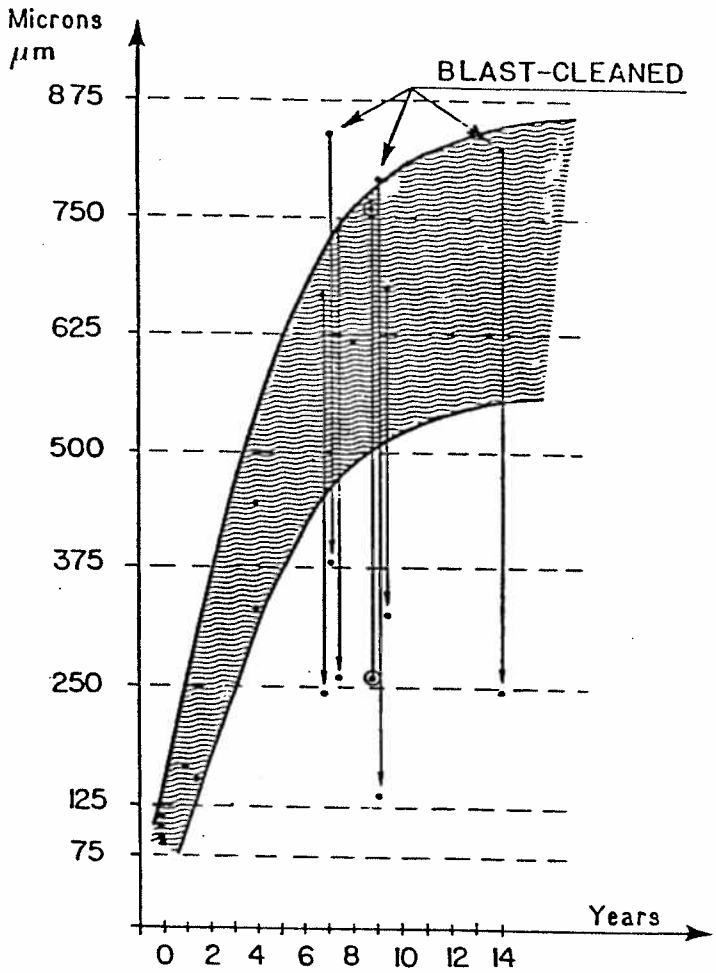
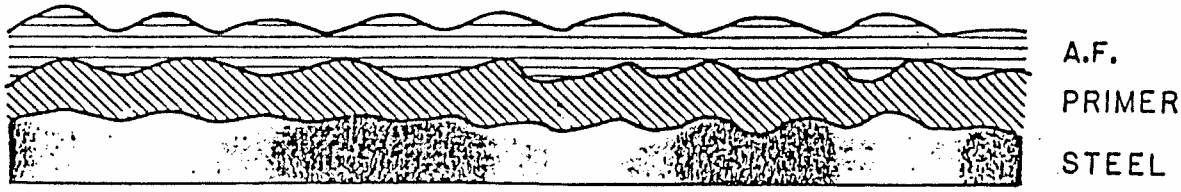


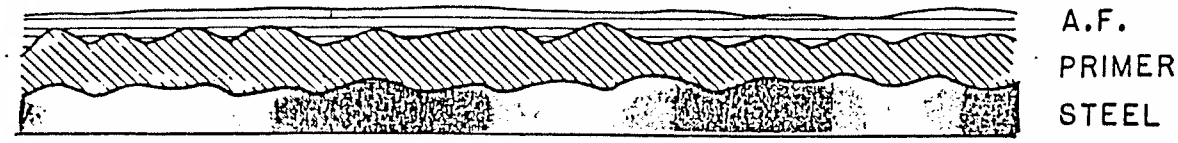
Fig. (76)

SELFPOLISING ANTIFOULINGS

1. New painted



2. After some months in service



3. After long time in service

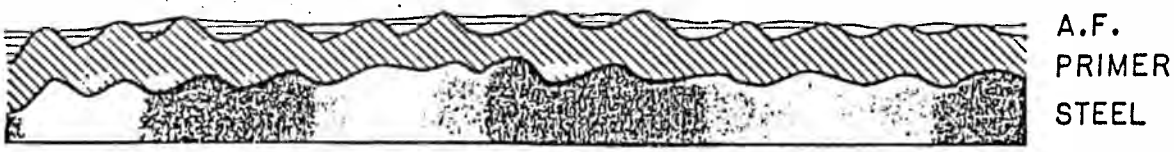


Fig. (77)

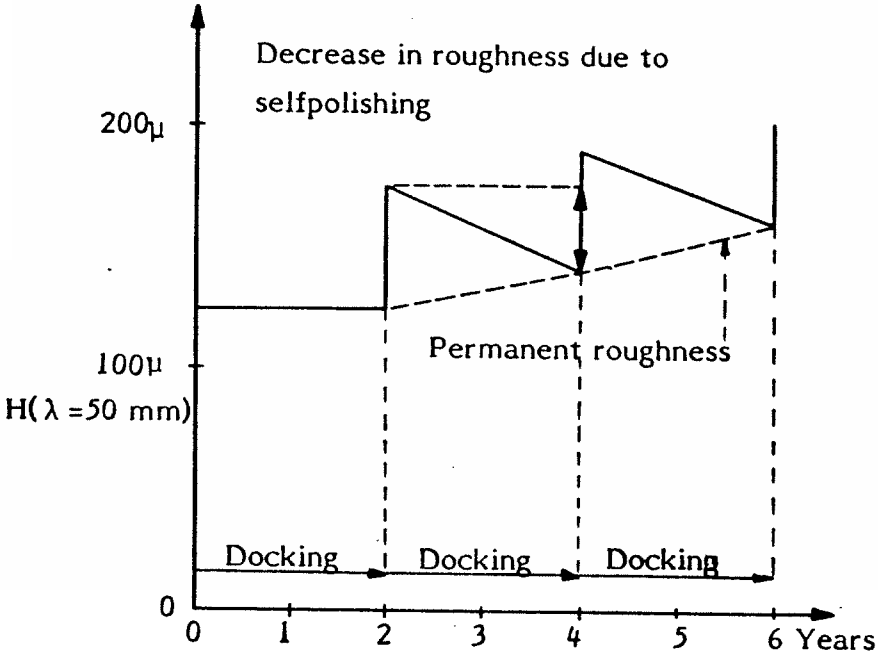
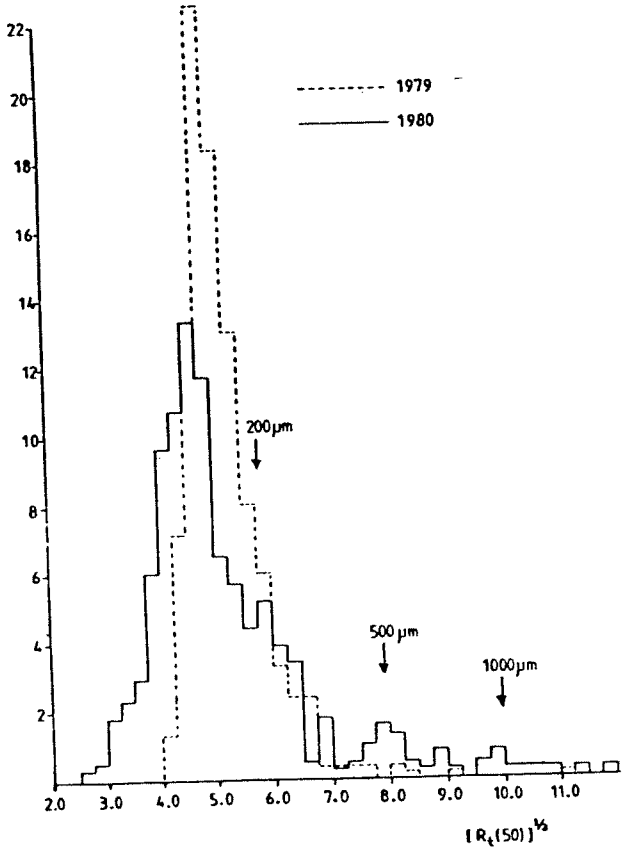


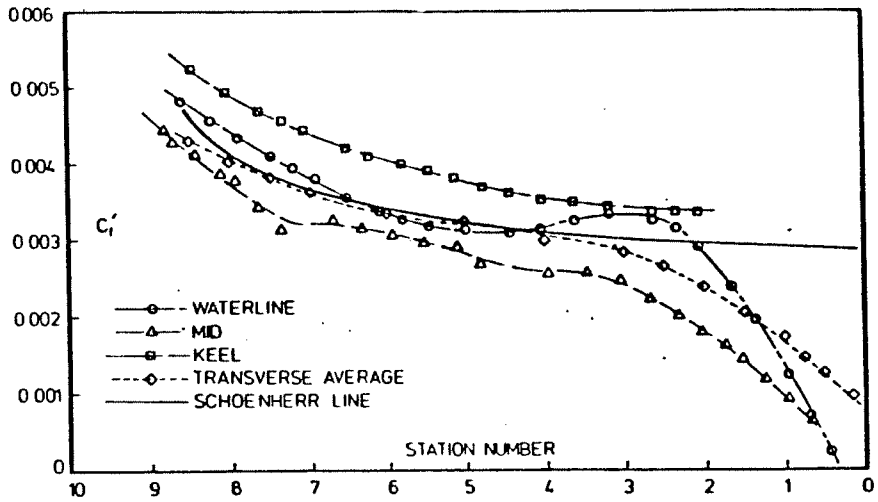
Fig. (78)



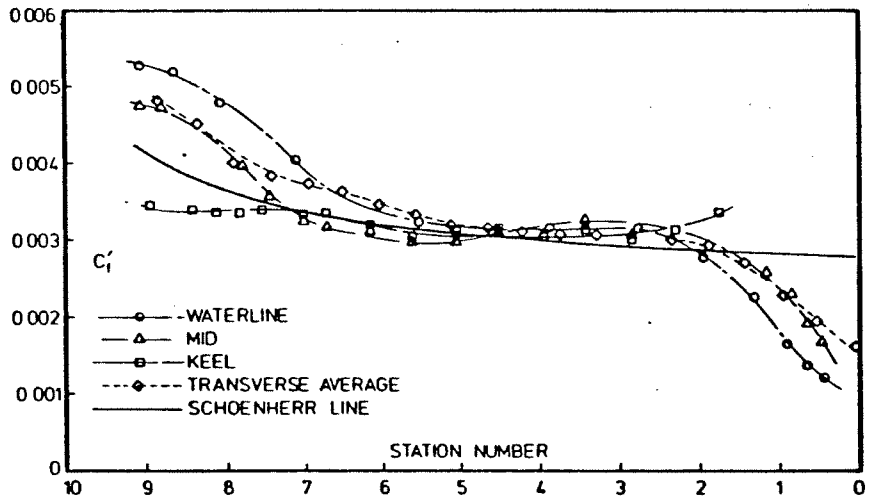
Histograms of ungrouped roughness data—flats, Ship A: 1979—547 readings at 47 locations; 1980—384 readings at 26 locations

From Townsin [37]

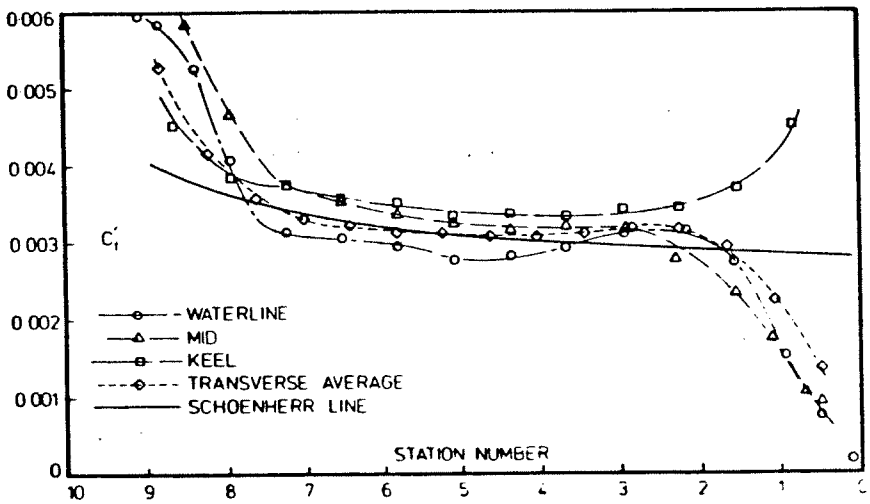
Fig. (79)



(a)



(b)



(c)

Streamwise variation of transverse average and local skin friction coefficients along three streamlines at $R_n = 4.59 \times 10^6$ for three models: (a) 0.564-C_B model, (b) 0.685-C_B (lucy Ashton), (c) 0.80-C_B model

From [47]

Fig. (80)

Distribution of Local Frictional Coefficient

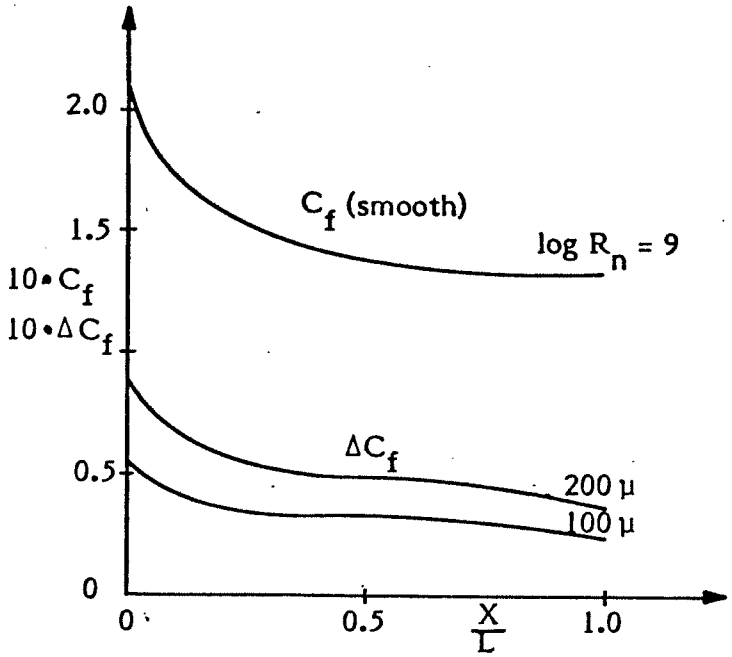


Fig. (81)

Frictional Resistance - Rest Resistance

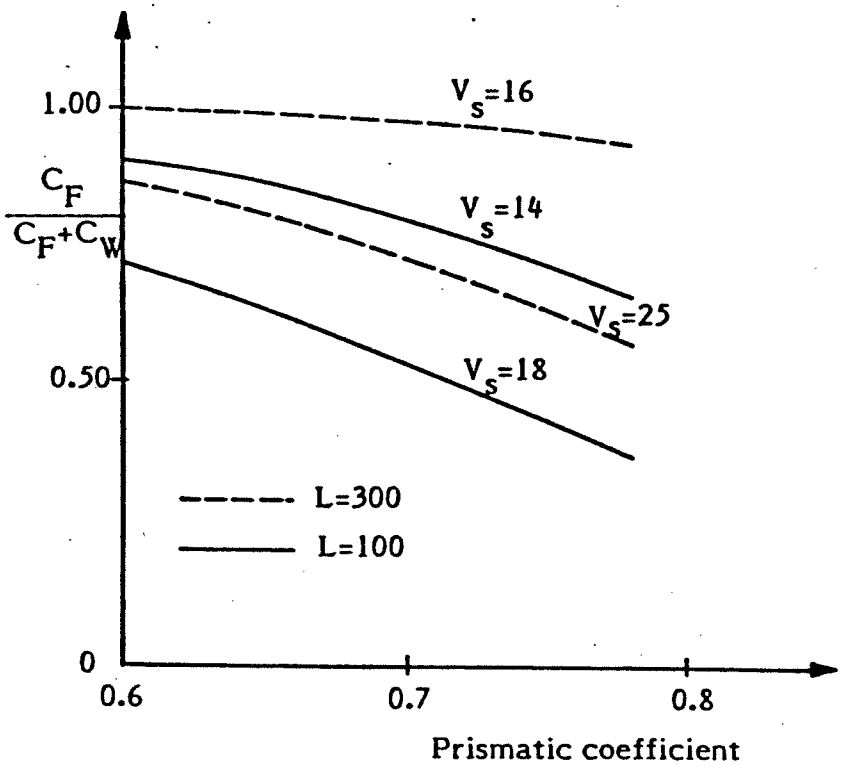
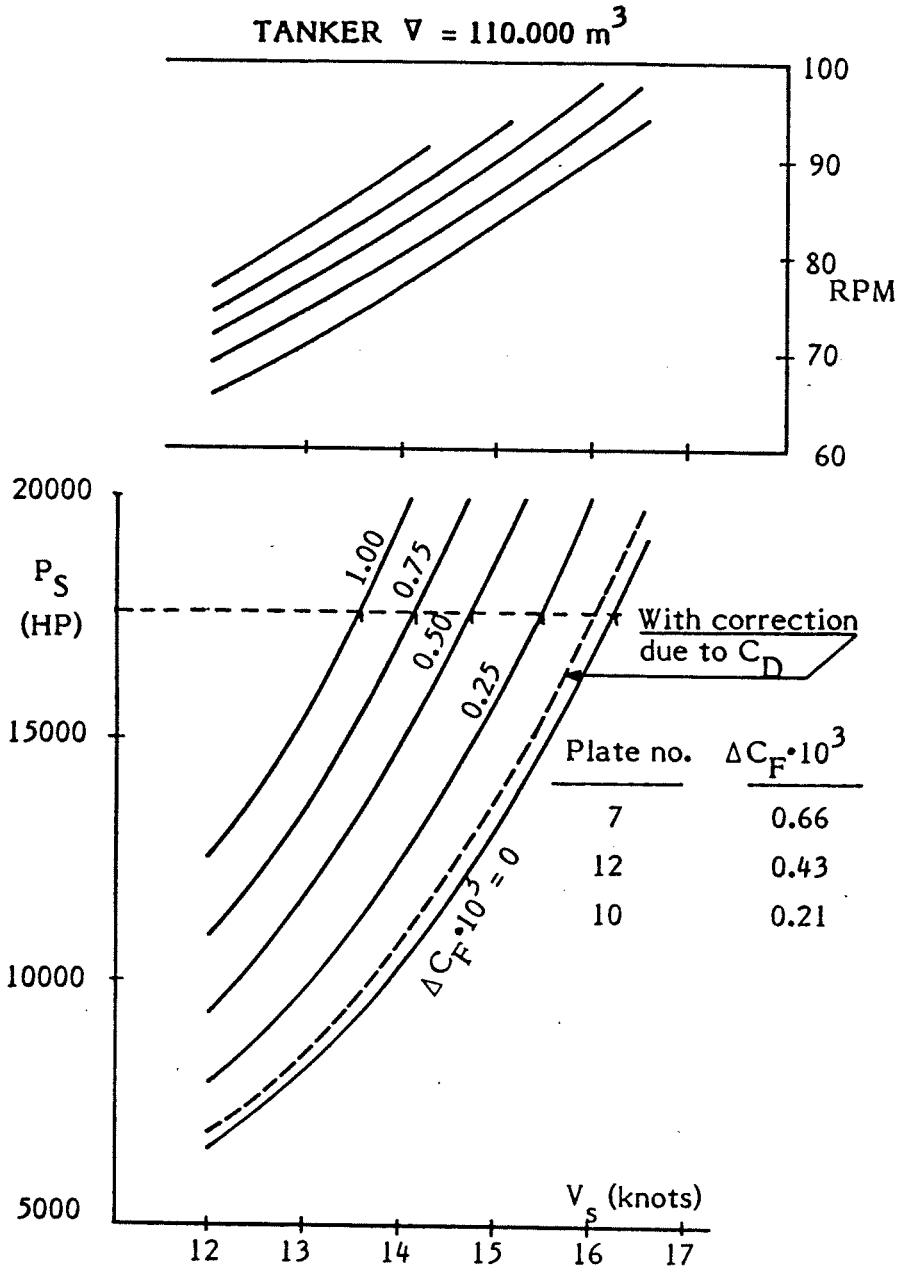


Fig. (82)


TANKER ∇ = 110.000 m3
100
90
80
RPM
70
60
20000
PS
(HP)
15000
10000
5000
1.00
0.75
0.50
0.25
ΔCF·10^3 = 0
With correction
due to CD
Plate no. ΔCF·10^3
7 0.66
12 0.43
10 0.21
VS (knots)
12 13 14 15 16 17

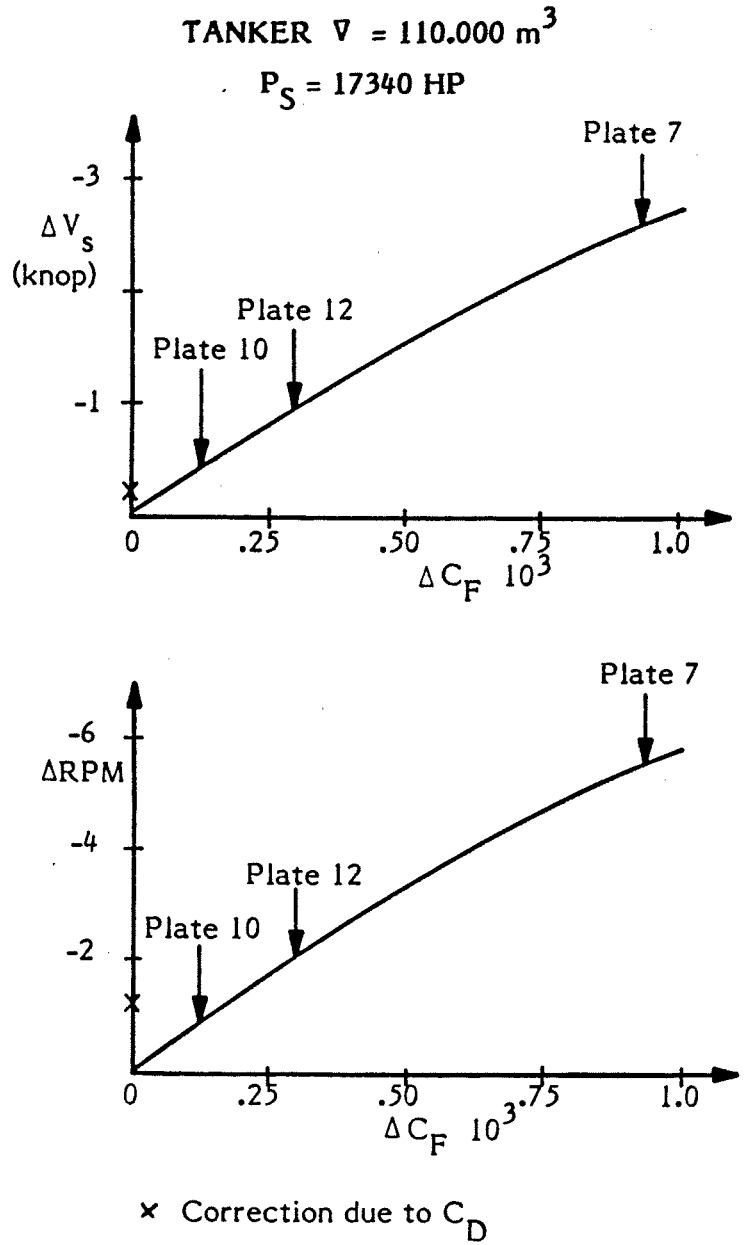

TANKER ∇ = 110.000 m3
PS = 17340 HP
Plate 7
-3
ΔVS
(knop)
Plate 12
Plate 10
-1
0 .25 .50 .75 1.0
ΔCF 10^3
Plate 7
-6
ΔRPM
-4
Plate 12
Plate 10
-2
0 .25 .50 .75 1.0
ΔCF 10^3
× Correction due to CD

Fig. (83)

Power Increase due to Roughness and Fouling

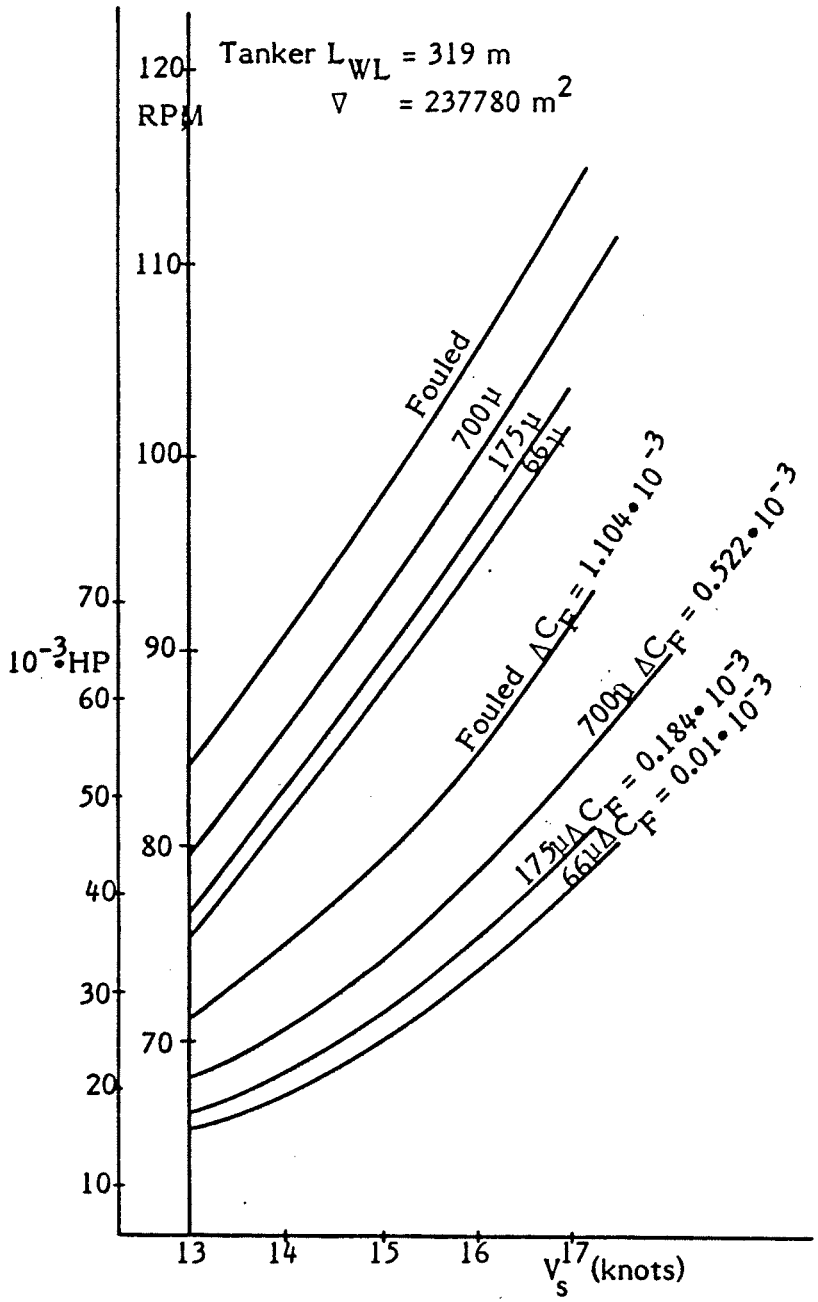


Fig. (84)

Power Increase due to Roughness and Fouling

LPG Tanker

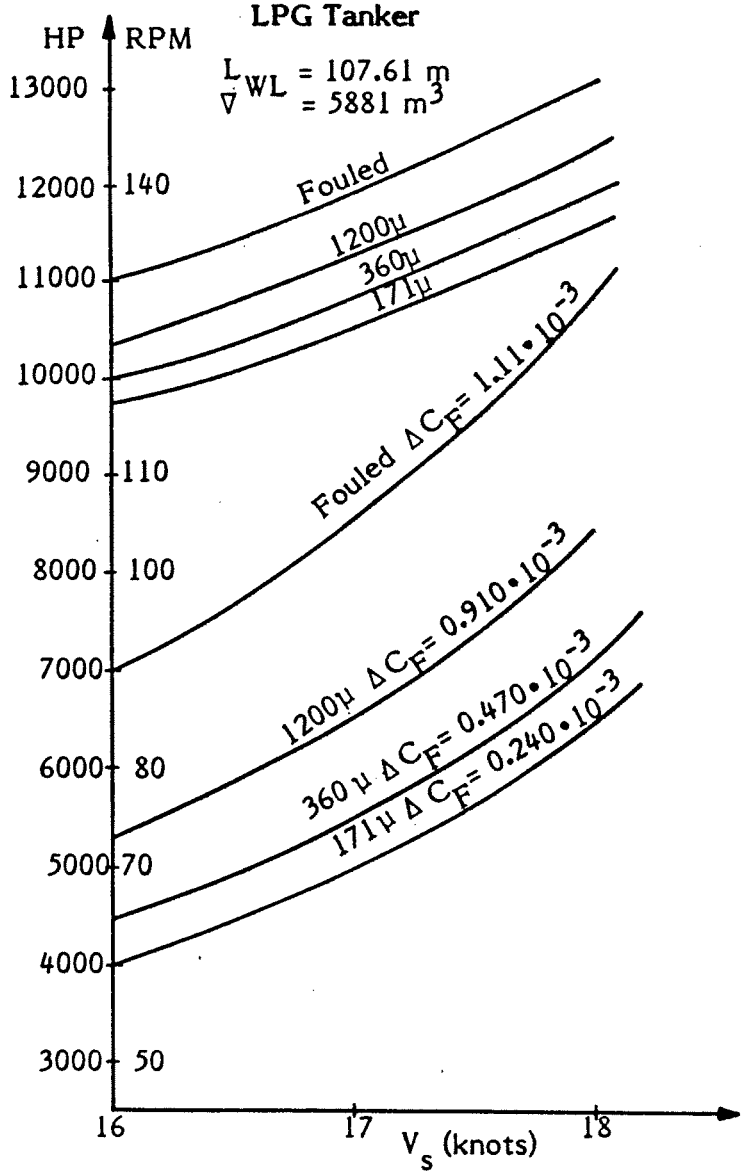


Fig. (85)

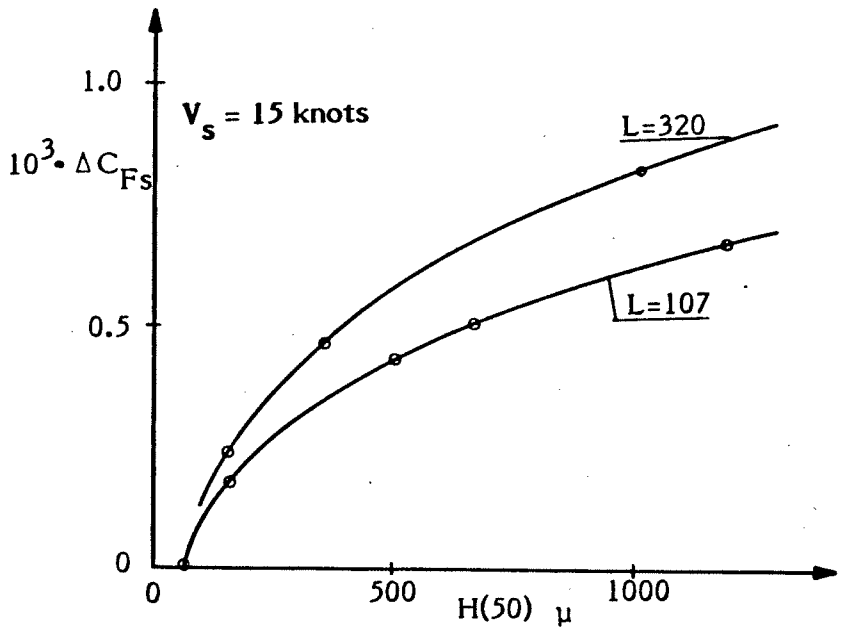


Fig. (86)

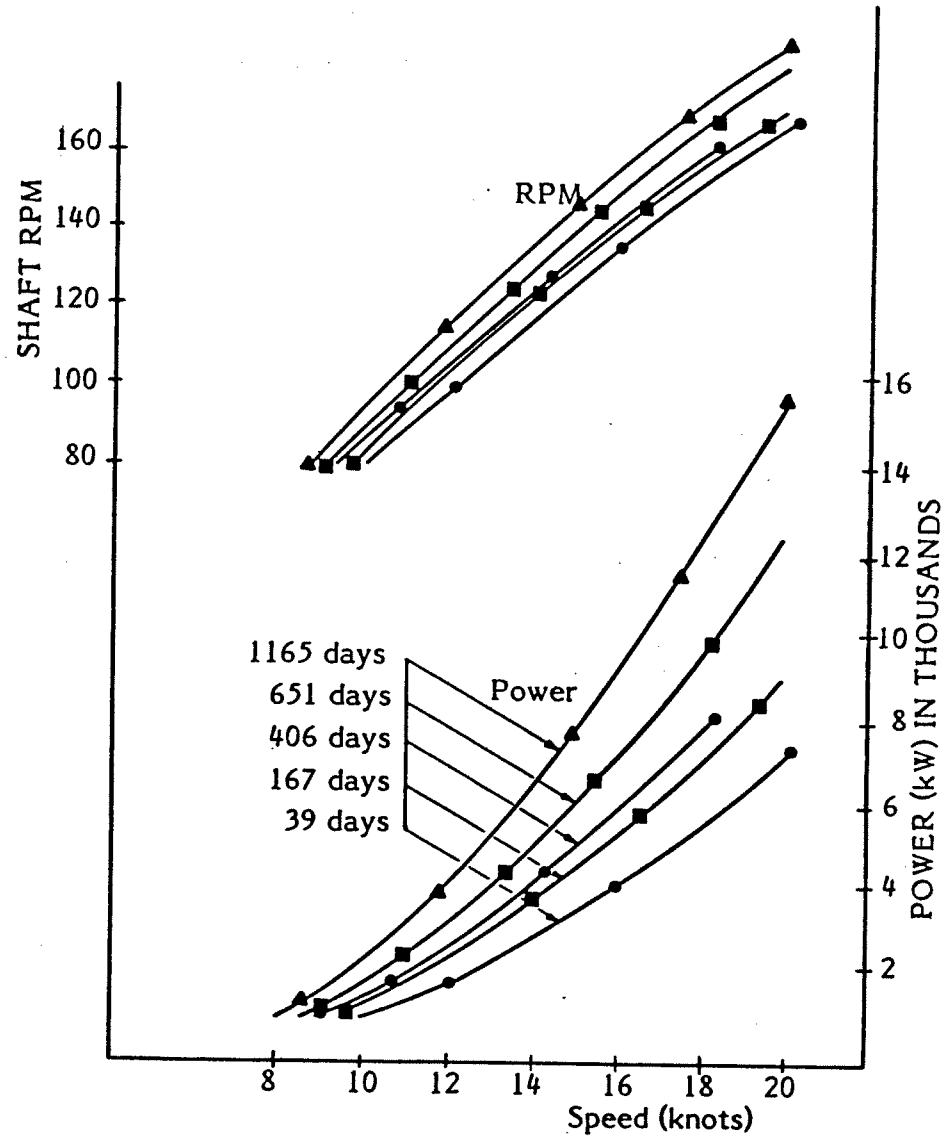
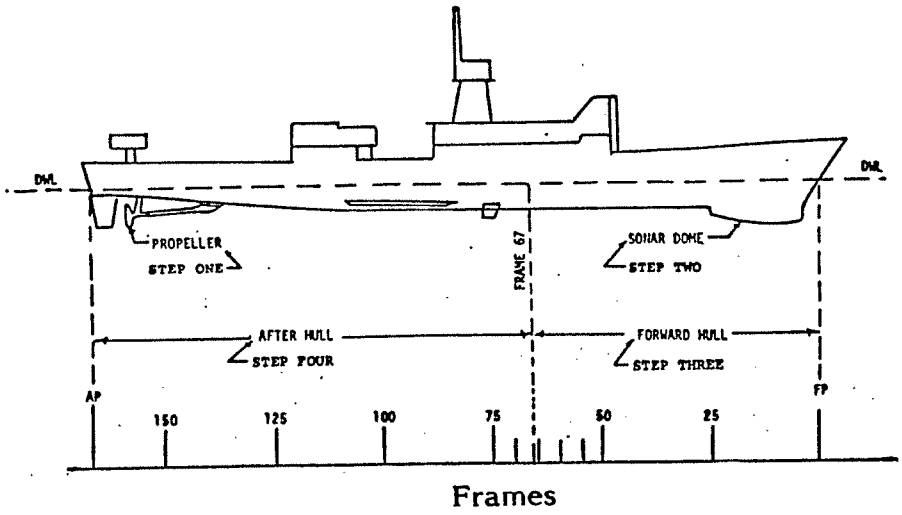

SHAFT RPM
160
140
120
100
80
RPM
Power
1165 days
651 days
406 days
167 days
39 days
16
14
12
10
8
6
4
2
POWER (kW) IN THOUSANDS
8
10
12
14
16
18
20
Speed (knots)

Fig. (87)


DWL
DWL
PROPELLER
STEP ONE
FRAME 67
SONAR DOME
STEP TWO
AFTER HULL
STEP FOUR
FORWARD HULL
STEP THREE
AP
FP
150
125
100
75
50
25
Frames

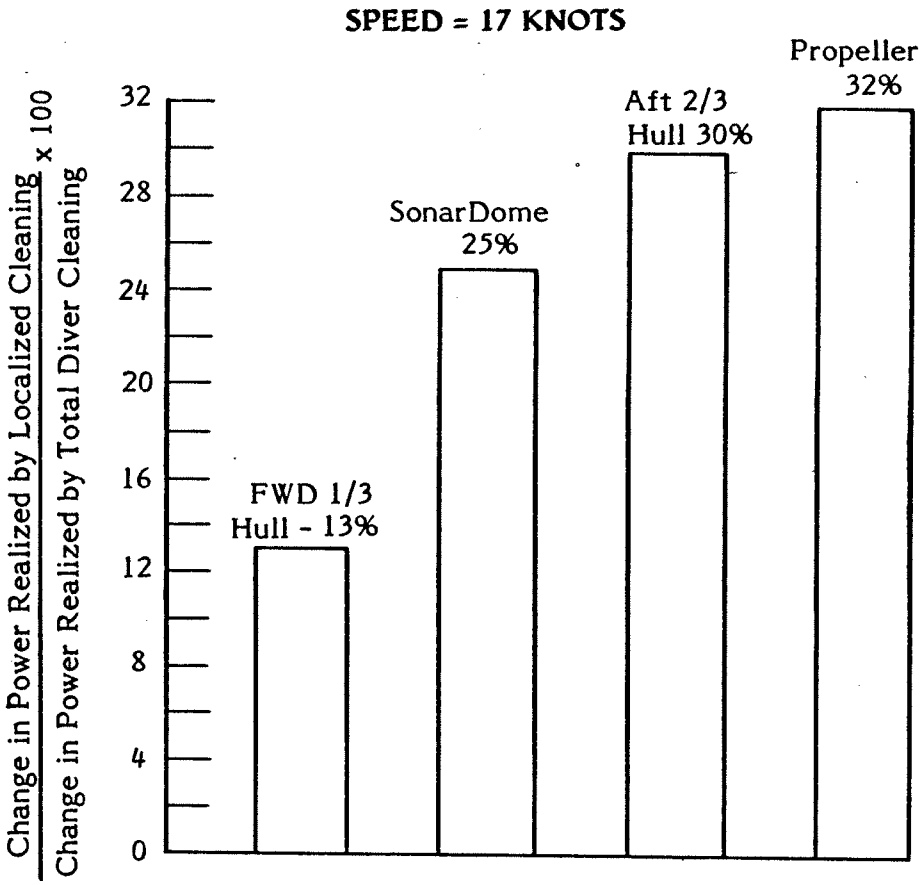

SPEED = 17 KNOTS
Change in Power Realized by Localized Cleaning / Change in Power Realized by Total Diver Cleaning x 100
32
28
24
20
16
12
8
4
0
FWD 1/3
Hull - 13%
SonarDome
25%
Aft 2/3
Hull 30%
Propeller
32%

Fig. (88)

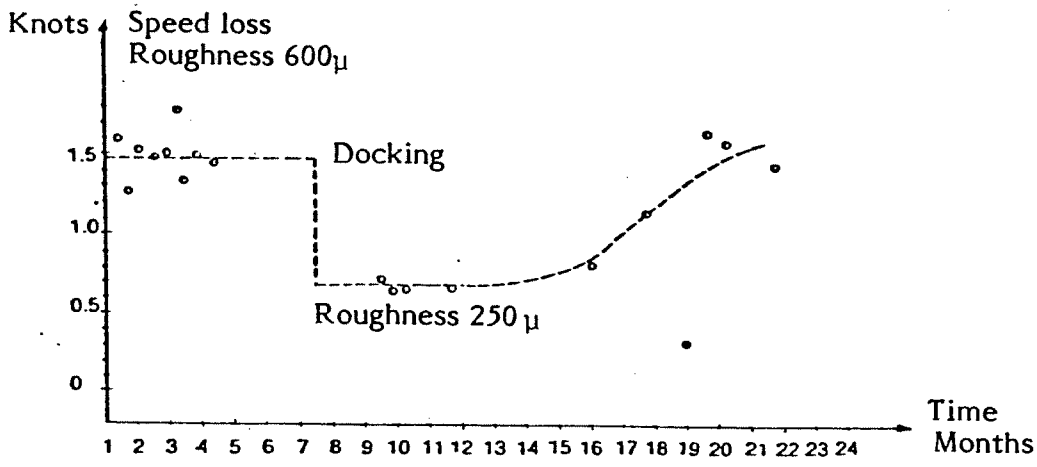

Knots
Speed loss
Roughness 600μ
Docking
Roughness 250 μ
1.5
1.0
0.5
0
Time
Months
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24

Fig. (89)

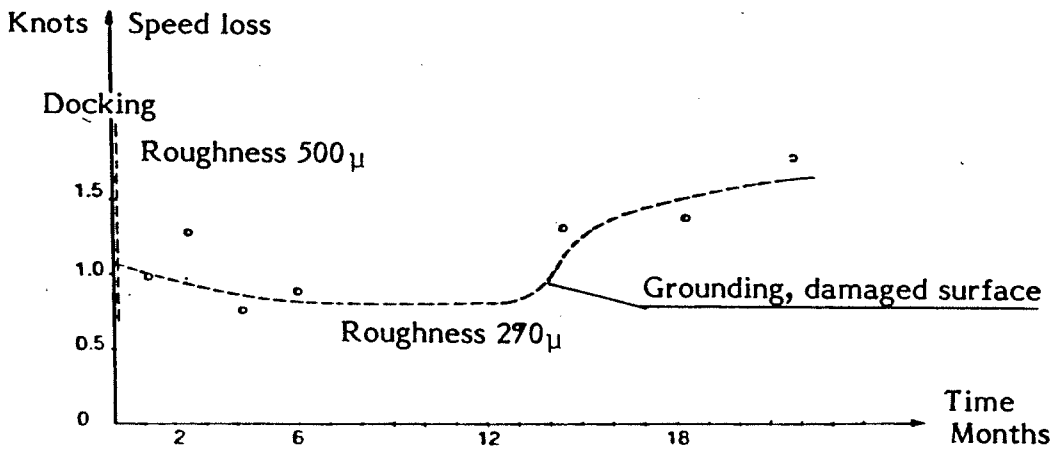
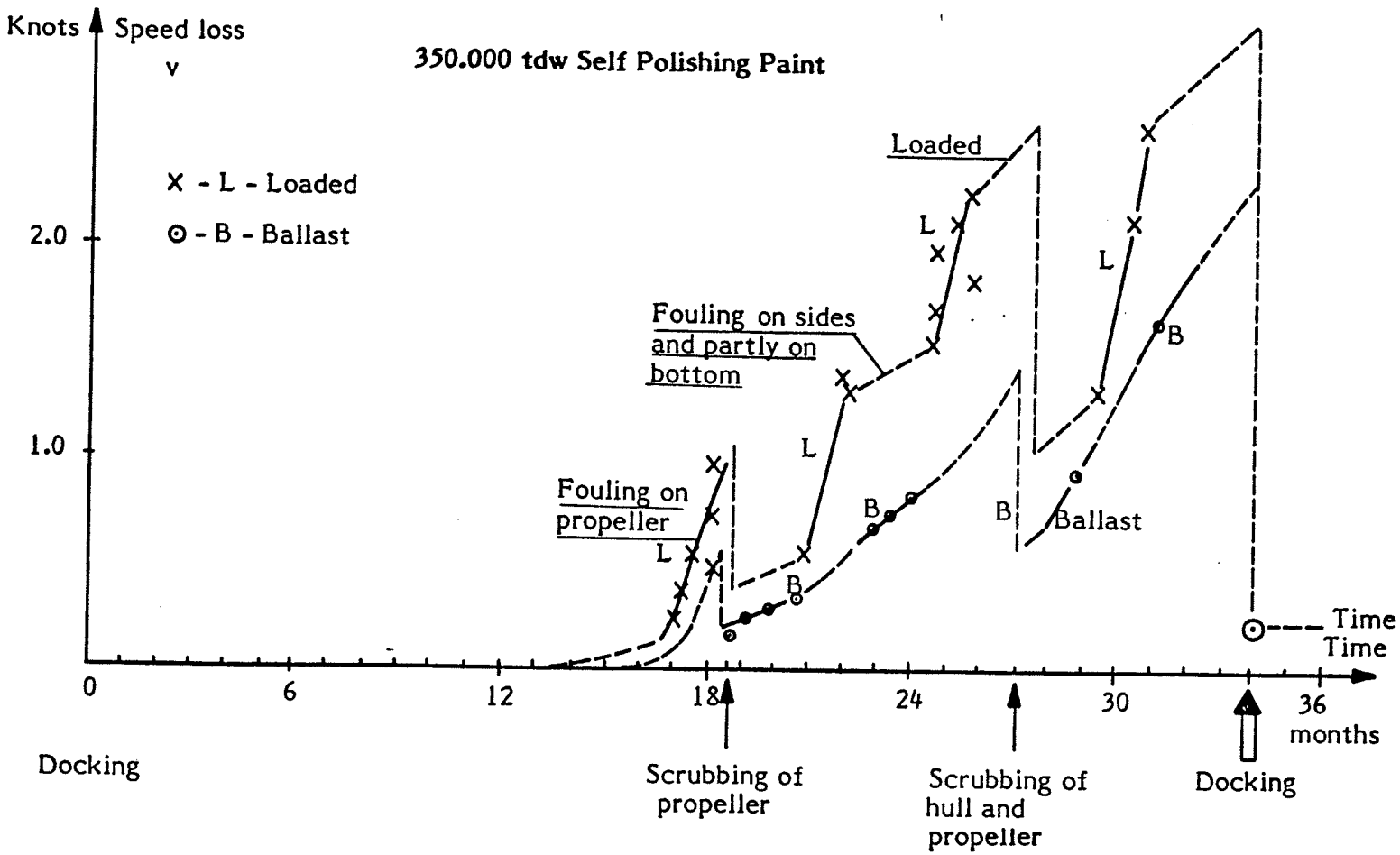

Knots
Speed loss
Docking
Roughness 500 μ
Grounding, damaged surface
Roughness 270μ
1.5
1.0
0.5
0
Time
Months
2
6
12
18

Fig. (90)



Speed Loss for a 350.000 tdw Tanker

Fig. (91)

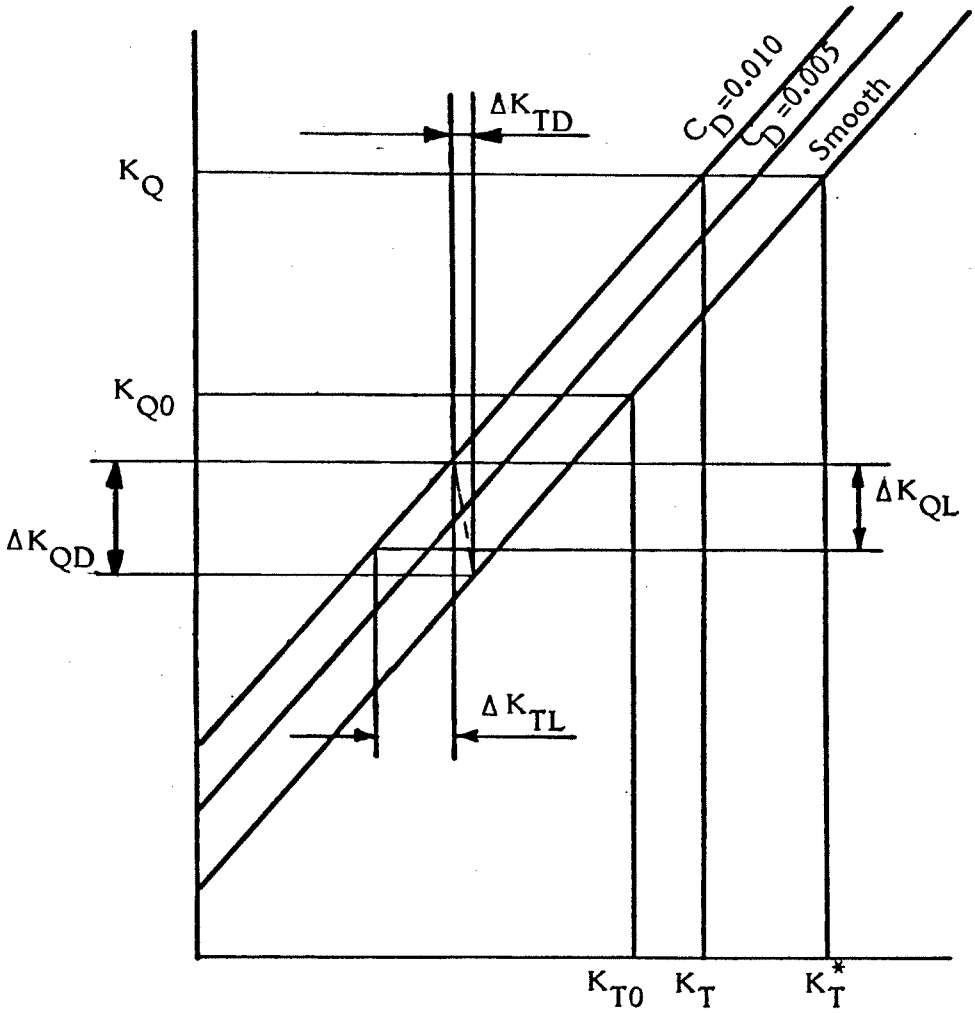

ΔK_{TD}
C_D=0.010
C_D=0.005
Smooth
K_Q
K_{Q0}
ΔK_{QL}
ΔK_{QD}
ΔK_{TL}
K_{T0}
K_T
K_T^*

Fig. (92)

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

Controllable Pitch Propellers

by

O Björheden

KaMeWa AB

Delivered

Date 83-08-31

Controllable Pitch Propellers

by O Björheden
KaMeWa AB

Summary

In the paper with appendices short descriptions are given of the CPP (Controllable Pitch Propeller) system, its mechanicl design and working principle as well as controls available for its operation. Current methods for the projecting, hydrodynamic design and testing of CPPs are outlined and some of the basic principles for optimum utilization of propellers and engines are briefly discussed. The merits of Highly Skewed CP Propellers are summarized and a study is presented of a project case where the inherent better vibration performance of highly skewed CPP blades is utilized for the purpose of improving the propulsive performance of a Ro-Ro ship by means of increased propeller diameter and lowered shaft speed. From calculations and comparative propulsion tests it is concluded that a large diameter Highly Skewed CPP in combination with a modern superlong stroke diesel engine will result in fuel savings in the order of 8-10 per cent in comparison with a conventional type of machinery.

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CP Propellers with Highly Skewed Blades	15
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Introduction

Controllable pitch propellers (CPP) have been known since the beginning of this century but had their break-through in the 1930's, at first on ships with high demands on manoeuvrability such as ferries or on ships where the full engine power was to be utilized at high as well as low speeds such as tugs and trawlers. Over the years other advantages such as their flexibility when operating with one or more diesel engines on ships with multi-engine installations, the possibility of utilizing shaft generators with constant rpm, reduced engine wear and simplified automation have considerably broadened the market for controllable pitch propellers. Today CPPs are extensively used on all kinds of both merchant ships and warships.

In short, the general mission of any propeller is to propel the ship with a maximum of reliability and efficiency whilst generating a minimum of disturbing vibrations and noise. For particular ships, such as certain warships, the call for low noise levels also involves the hydro-acoustic noise which is radiated from the propeller to the surrounding water.

The necessity for reliability includes the demands for strength in addition to resistance against corrosion and cavitation damage. For the propeller designer the strength and cavitation problems are aggravated by the uneven wake field behind the ship, in particular on single screw vessels where the variation in axial velocity to the propeller may well amount to 60-80 % of the ship speed.

The resistance against corrosion is mainly related to the material and today a fairly large variety of bronze and stainless steel qualities with adequate corrosion and fatigue properties are available to the propeller manufacturer.

The fuel consumption is directly related to the propulsive efficiency. In order to achieve the highest possible efficiency, the propeller diameter should be selected as large as possible with a corresponding optimum propeller shaft speed. A clear development trend, initiated by today's high fuel prices, is

a more frequent use of slow-running large diameter propellers. On ships with highly loaded propellers a nozzle of the accelerating type is often used in order to improve the efficiency. In addition to the application of large propeller diameters and/or nozzles with the purpose of minimizing the axial momentum losses in the propeller slipstream, various devices aiming at reducing the rotational losses may also be considered. In the first hand, already a conventional rudder placed behind the propeller will to some extent act as a guide vane and save some of the kinetic energy in the rotating propeller slipstream. By shaping the rudder profiles more properly in terms of angle of attack and section camber (contra-rudder) or by fitting more complete guide vane devices (reactor fins) in front of or behind the propeller further improvements in the propulsive efficiency can be gained. The very best propulsive performance at a given propeller diameter should be expected with a pair of counterrotating propellers but with regard to its mechanical complexity this configuration has not yet been used in practice on full scale ships.

For fixed pitch propellers (FPP) the optimum criterion is normally clear and related to a certain propulsive condition whilst for CPPs different conditions such as operation with alternatively one or two engines per shaft at multi-engine installations or propulsion with a varying number of propellers in multiple screw ships may result in somewhat conflicting requirements on various parameters. Frequently, these so-called off-design conditions lead to compromises in the choice of propeller diameter, radial distribution of pitch and camber etc.

Independent of propeller type, the evolution at sea towards increased engine powers and higher speeds as well as larger ships, sometimes with unorthodox lines, has increased the problems of propeller-induced vibrations and noise in the ships' afterbodies. The hull vibrations are primarily caused by the uneven wake field in which the propeller has to operate and by pulsating forms of propeller cavitation induced by the fluctuating blade angle of attack. In general, the problems are increasing with increased specific propeller loading and increased velocity. Among the hydrodynamic measures which may be taken in order to reduce the propeller-induced vibrations are:

- Design efforts pertaining to the hull lines and the arrangement of the propeller on the hull.

- Design efforts pertaining to the propeller and in particular the propeller blades.

- Installation of accelerating fins, semi-ducts and similar devices which aim at improving the wake field.

In the first hand measures according to the two first alternatives should of course be chosen. The third category may be considered as an emergency solution to be utilized if and when the design efforts have not resulted in a satisfactory improvement. Normally all kinds of fins are connected to some disadvantage, such as installation and maintenance costs, increased ship resistance, susceptibility to slamming etc.

To some extent the demands on high propulsive performance on one hand and low propeller-induced vibration levels on the other are conflicting with each other. To obtain the best efficiency the propeller should have the largest possible diameter and moreover in order to obtain the best propulsive interaction between propeller and hull (the best hull efficiency) it should operate in a region with a maximum of frictional hull wake. These demands will normally result in small clearances between propeller blade tips and hull as well as large circumferential wake variations within the propeller disc area. Both these latter circumstances will in principle contribute to increasing the propeller-induced vibrations. The best compromise solution is hence to apply the biggest possible propeller which will still result in acceptable hull vibration levels.

When designing the propeller such as to minimize the propeller-induced vibrational impulses in order to facilitate a large propeller diameter a few different grips may be considered:

- to increase the propeller blade number and/or blade area above normal values

- to unload the propeller blade tips in order to suppress blade tip and tip vortex cavitation

- to give the propeller blade a suitable amount of skew or warp in order to obtain a more soft interaction with the uneven hull wake field.

With regard to strength and cavitation safety an increase in the propeller blade number above what is commonly used (Z = 3 - 5, Z = 4 predominantly) has to be accompanied by an increase in the propeller blade area which in turn tends to increase the frictional losses, thereby reducing efficiency. As can be realized, an increase in blade number or in the blade area alone is not the ideal way of enabling a larger propeller diameter since most of the gain in efficiency from the increase in diameter will be consumed by increased frictional losses.

To some extent the same reasoning is valid also for the second method involving blade tip unloading. A moderate unloading of the propeller blade tips is applied by most designers and will not influence the efficiency noticeably. When deviating considerably from the optimum load distribution, however, the drop in ideal propeller efficiency will soon become larger than what can be gained from an increased diameter enabled by the reduction in vibration excitation attributed to the tip unloading.

The most effective method by far to obtain propellers with low vibration excitation levels without sacrificing anything in terms of efficiency is to apply propeller blades with a large degree of skew. In recent years highly skewed (HS) CP propellers have also become more and more popular not only on passenger vessels but also on many cargo ships where their inherent better vibration performance may be utilized to improve the propulsive performance by allowing larger propeller diameters with accordingly smaller propeller - hull clearances.

In the present report the main object is to illustrate how CP propellers with conventional or highly skewed blades can be used to improve the fuel economy of the ship. The principles for project calculations as well as hydrodynamic design and testing of the CPP are described. For the sake of completeness brief notes are also given on the mechanical design characteristics of a CPP system as well as controls available for its operation.

The CPP System

Basically a CPP system consists of the following components:

Propeller hub with blades which can be adjusted from full ahead to full astern pitch

Propeller shafting with oil distribution box

Hydraulic system

Remote control system

In addition to the above, various control equipment, such as automatic load control, joystick control (combining the operation of several propulsion and manoeuvring devices such as propellers, thrusters and rudders), fuel economy control etc. may be included in a ship installation.

As regards the attachment of propeller blades to the hub body and the mechanism for pitch adjustments a variety of designs and working principles are at hand. On smaller ships with moderate engine powers, such as tugs, trawlers etc. CPPs of the push-pull rod type are frequently used. In the brochure, app. 1, details are given of one particular CPP of this type. The push-pull rod CPP is mainly characterized by a simple blade attachment, often with a split hub body and integrated blade flange crank pin ring, a crosshead with slots carrying the sliding shoes attached to a push-pull rod fitted in a boring of the propeller shaft and activated by a hydraulic servomotor in the shaftline inside the ship hull plating. In addition to low price the main advantage put forward for the push-pull rod concept is that it minimizes the number of mechanical components in the hub body and makes parts, such as hydraulic servo, control valves etc. accessible for overhaul without drydocking of the vessel. For both technical and economical reasons, however, the push-pull rod principle becomes less attractive at large powers.

On large vessels the predominant type of CPP has the entire pitch setting mechanism including the hydraulic servomotor in the propeller hub. With this solution, the high pressure oil

required for operation of the servomotor is transferred through a pipe in a central boring of the propeller shaft via an oil distribution box (sealing box) fitted on the shaft.

The brochures, app. 2 and 3, give details of one particular CPP of this type. On this design the control valve required for regulating the oil flow is located in the hub and the high pressure oil pipe is serving as valve rod and pitch indicator as well. The hub body is solid with integrated bearing rings carrying the blade loads. The servo in hub-type of CPP has proven reliable for shaft powers in excess of 40.000 hp and propeller diameters in excess of 10 metres.

Most CPPs are reversible, i.e. they have a piston stroke permitting a blade angular motion of 55-60 degrees which is sufficient for utilization of the full engine power from full ahead to full astern ship speed. Special hubs with extra long stroke (110-120°) permitting the blades to be set in fully feathered position are also available. This feature is valuable on multiple screw ships when it is desired to minimize the drag from propellers temporarily not engaged in the propulsion.

The high pressure oil flow required for pitch setting is generated by hydraulic pumps which can be driven either electrically or mechanically from the propeller shaftline. On most ships both types of pumps are installed in order to enable propeller pitch adjustments both before and after starting up of the main engine.

In order to prevent water from leaking into the hub a special gravity tank is included in the hydraulic system of most servo in hub-type CPP installations. By this arrangement a certain overpressure is always maintained relative the surrounding water.

For safe operation of propeller and engine from the bridge a remote control system is always included in the CPP system. A large variety of designs from simple mechanical ones via pneumatic to electric or electronic control systems are available oon the market. Appendix 4 shows one example of a modern control system with a number of options as regards built in functions.

Basically there is always some sort of combinator control, i.e. a device which combines suitable values of propeller pitch and shaft speed from full ahead pitch - full rpm via zero pitch - idling rpm to full astern pitch - full rpm by different positions of one single control handle. The layout of the combinator curve is of great importance for the ship's fuel economy as will be discussed later on.

Hydrodynamic Design, Testing and Characteristics of the CP Propeller

Project Calculations

The hydrodynamic design of CP propellers is based on the same principles as those of fixed pitch propellers but a few additional parameters must be considered, i.a. the blade spindle torque required for pitch adjustments.

At KaMeWa rather extensive calculations are performed already at an early project stage in order to settle fundamental parameters such as propeller diameter and shaft speed, blade number, blade area and blade thickness ratio, propeller hub size and shaft dimensions. These project calculations are fully computerized permitting a large number of projects to be studied every day. Figure 1 shows in a block diagram the various steps of computation and iteration involved. Hydrodynamic calculations such as determination of optimum propeller diameter or shaft speed, propeller pitch ratio and efficiency are based upon systematic model test results stored in the computer. As discussed above, the best possible efficiency is obtained with the largest possible propeller diameter and the corresponding optimum shaft speed. The propeller blade area should be selected as small as possible in order to reduce frictional losses. However, a lower limit for the blade area is determined by the requirement for cavitation safety margins. For CP propellers there is also an upper limit around EAR = 0.80 which is dictated by the requirement for reversibility. In practice, all merchant ship propellers fall well beneath this upper limit.

The propeller hub size is determined by strength calculations as well as calculations of hydraulic pressure required to turn the

blades, bearing ring pressures etc. For this purpose a large amount of data for the various hub types and sizes are likewise stored in the computer.

Finally, besides the propeller blade and hub calculations, the shaft dimensions, required oil distribution (OD) box size and pump capacity, a number of weights, flywheel effects and propeller coefficients are also calculated.

Blade Design Calculations

The hydrodynamic design of a KaMeWa propeller blade is based on vortex theory calculations according to lifting line (1) as well as lifting surface theory (2).

For blades of conventional shape lifting line calculations are carried out for optimum and non-optimum free running as well as wake-adapted propellers with optional criteria for the optimum circulation distribution. The lifting surface effect is corrected for, based on the works by (3) and (4). The Naca 16 profile thickness distribution in combination with mean line a = 0.8 mod. (5) is adopted as standard. Profile drag coefficients are thoroughly calculated at each radius for the full scale as well as a corresponding model propeller taking surface roughness into account. This procedure permits the designer to choose whether to base the design on thrust or power and it also provides valuable information on scale effect and influence from surface roughness. For highly skewed blades complete lifting surface theory calculations are carried out in addition to the computations above. Through the years a large number of propellers designed according to lifting line and lifting surface theory methods above have been tested in model as well as full scale and generally a good agreement between calculated data and test results has been achieved. Moreover it can safely be stated that the calculation methods result in good designs regarding efficiency as well as cavitation performance.

Regarding blade strength, a basic criterion is of course to fulfil the requirements laid down by the classification society. In addition, stresses are calculated at various blade sections utilizing elementary beam theory, or in the case of HS blades,

finite element methods. Static as well as dynamic blade loads caused by the irregular wake field are taken into account and the risk of fatigue is judged on basis of well proven criteria. In principle, the same strength philosophy involving fatigue considerations is applied also to the blade - hub attachment as well as several parts of the hub interior.

Upon completion of the design calculations the propeller blade drawing is produced by the aid of a numerically controlled plotter (fig 2).

Off-Design Considerations

For an ordinary merchant ship e.g. a cargo liner or a tanker more or less constantly operating at full power and speed the propeller blade design is fairly straight. The radial pitch distribution is chosen close to the optimum, however, with some reduction at the blade tip in order to suppress tip vortex cavitation and to reduce the propeller-induced pressure pulses acting on the ship's hull. The blade section length distribution is chosen as to provide ample margins against back bubble cavitation and from the circulation calculation the blade section camber is determined to generate the required blade lift at ideal angle of attack. The blade thickness distribution is chosen with regard to cavitation performance as well as static and dynamic strength considerations.

However, among the merits of the CP propeller is its flexibility in adapting smoothly to various propulsive conditions by pitch adjustments permitting the ship to be operated over a wide variety of power and speed combinations. Classical examples are tugs and trawlers on which the full engine power can be utilized at either light free running or heavy towing conditions. On modern cargo vessels and ferries the medium speed diesel engines are now being used to an increasing extent and multi-engine arrangements with two or more diesels driving the CP propeller over a collecting gear have become common (fig 3). Depending on the route and timetable the shipowner may wish to run the vessels with one, two or more engines engaged and always with full shaft speed. The shaft driven AC-generator is also applied on many ships in order to reduce the amount of fuel required

for the production of electricity on board, and this arrangement in principle implies constant shaft speed operation at all propulsive conditions that may occur.

On ships with two, three or more propellers conditions occur with one, two or three propellers engaged and the remaining ones disconnected or locked with the blades in maximum ahead or fully feathered position.

In the propeller blade design work, of course, due consideration must be paid to any off-design condition and quite often compromises have to be made as regards the choice of diameter, pitch distribution and blade section camber in order to achieve high efficiency as well as good cavitation performance over the whole speed range. In case of tugboats and trawlers a large propeller diameter together with a fairly low design pitch and large camber is often chosen in order to obtain a high bollard pull and normally this can be done without sacrificing any of the efficiency at the free running condition. For ships with multi-engine arrangements or AC-generators which are to operate with reduced shaft power over prolonged periods of time it may be necessary to reduce to some extent the diameter and blade section camber in order to obtain high efficiency and acceptable cavitation performance at both full and reduced power. One particular item is the risk of erosive face cavitation at conditions with low power and high shaft speed. Sometimes local modifications of the blade leading edge are required to prevent this type of cavitation.

Hydrodynamic Testing - Propeller Characteristics

At the KaMeWa works in Kristinehamn, a new propeller laboratory with two cavitation tunnels was built in 1970-71 and put into commercial operation in 1972. Some information as regards technical data, instrumentation etc. is given in the brochure, app. 5.

Both tunnels are mainly devoted to the testing and development of KaMeWa CP propellers and only to some extent tests are conducted on order from outside. In the conventional tunnel, a major part of the routine testing of single screw merchant ship

propellers takes place whilst in the free surface tunnel tests with propellers for twin screw ships, high speed propellers, water jet propulsion devices etc. are predominating. In the following the various steps in a normal test program for a merchant ship propeller will be briefly described.

The normal start of a test series is the free running test aiming at determining the basic propeller characteristics. In homogeneous parallel flow the propeller thrust and torque is measured over a range of advance ratios and cavitation numbers. For CP propellers a number of different pitch settings are normally tested and for tugs, trawlers, icebreakers etc. the bollard pull condition is penetrated as well. When required, the astern performance of the propeller or the risk of overspeeding with the propeller operating in the turbine range (windmilling propeller) may also be studied. Test results are evaluated into the normal K_T-K_Q-η form and are then plotted in the test diagram as functions of the advance ratio J and the cavitation number σ. Figure 4 shows an example of free running as well as bollard pull cavitation test results obtained with a KaMeWa CP propeller. Based on the model test data, thrust, torque and efficiency of the full scale propeller may be determined for the design as well as off-design conditions applying the normal scaling laws. The propeller test diagrams may be used for a variety of project and design calculations; one example is the common P_D-n diagram derived from propeller test results plus results of towing tank resistance test of the ship. Fig 5 shows one example of a P_D-n diagram for a twin screw ship with a twin-engine installation.

Upon completion of the free running tests follows the cavitation test in a simulated wake field. In the KaMeWa tunnels the technique with a dummy model of the ship's afterbody is normally applied when simulating the wake pattern. Several types of afterbody dummies are available and can be adapted to the special requirements of the test in question. In fig 6 a normal model arrangement for afterbody tests is shown. The axial velocity distribution behind the ship afterbody as obtained in the towing tank and possibly corrected for scale effect is simulated in a step by step procedure by altering the vertical wire grids in front of the propeller. The geometrically correct shape of the afterbody in the vicinity of the

propeller ensures an approximately correct pattern of the tangential and radial wake components as well.

Finally the propeller is tested in the simulated, varying velocity field. Normally the propeller model is driven by an electric motor over an angle gear unit built into the hull dummy with a strain gauge dynamometer in the propeller shaft. The whole model arrangement is then mounted on a large plate fitted into a hatch on top of the cavitation tunnel test section. This technique has proved to be most practical and implies a minimum of test section occupation for mounting work. A few examples of model configurations are shown in fig 7.

Based upon the systematic free running test results the pitch setting of the propeller is determined for all propulsive conditions concerned and then the afterbody test is performed on basis of torque identity, i.e. in the simulated wake field the propeller model is run at correct torque coefficient and cavitation number.

Observations of the cavitation pattern are carried out either in stroboscopic light or with high speed film technique. The risk of erosion is judged on basis of the observations and from erosion tests carried out with the propeller blades covered with a soft paint. If the test indicates a risk of erosion the blade shape, of course, has to be modified in order to eliminate the risk of erosion. In most cases only small leading edge modifications are required, e.g. to prevent face cavitation or unstable back sheet or cloud cavitation.

Apart from the cavitation observations several measurements are also performed during the afterbody test. Mostly dynamic phenomena such as propeller-induced vibrations and noise are being studied. The mechanically transferred vibrational impulses can be measured either by special strain gauge hub or blade dynamometers. The water-borne fluctuating pressure pulses acting on the ship's hull surface are measured with the aid of small pressure gauges built into the dummy model in the region concerned. Harmonic analysis of the signals is performed directly upon measurements. On basis of the results obtained the risk of serious propeller-induced vibrations and noise on board the ship

can be judged.

Besides the cavitation testing in the KaMeWa tunnels discussed above, a large amount of open water and self propulsion tests with KaMeWa propellers is performed in various model basins throughout the world every year, test results being reported also to the propeller manufacturer.

As regards full scale performance a ship trial trip report is submitted to the designer for every new KaMeWa installation put into operation. Data are given on ship load as well as weahter conditions and water depth on the measured mile. The ship's speed obtained at the various powers and shaft speeds tested is reported together with data on propeller pitch setting, hydraulic oil pressure required to turn the propeller blades etc. Notes are made concerning noise and vibration performance as well. On tugboats and supply ships a bollard pull test is often conducted in connection with the trial trip. The full scale trial results, of course, bring valuable information to the designer and give him the opportunity of checking his design in various respects.

On some ships more extensive full scale tests including propeller cavitation observations, pressure pulse and noise measurements are carried out as well. In principle the same techniques are available as for the model tests. Through plexiglass windows fitted into the hull plating above the propeller cavitation is observed and photographed, often with the aid of a high speed film camera (fig 8).

Of course, the full scale test when thoroughly performed and correlated to the corresponding model test brings about the most valuable information and contributes to a better understanding of all the complex physics involved in ship propulsion. This feed-back is necessary in order to obtain improved designs for the ships of tomorrow.

Optimum Operation of Propeller and Engine

The P_D-n diagram (fig 5) reflects the performance of a CPP in propelling the ship at one particular condition in terms of weather, water depth, ship draught, amount of fouling etc. To cover the whole mission profile of the ship similar diagrams have to be calculated for all different operating conditions concerned. As can be seen from the P_D-n diagram the propeller normally has its best performance when operating at pitch ratios near its design value. For determination of the fuel consumption, however, not only the ship resistance and the propulsive efficiency but also the efficiency of the engine must be taken into account. In figures 9 and 10 specific fuel consumption curves of the engines have been plotted in the P_D-n diagram for conditions with one and two engines in operation respectively. Based on this type of diagram the fuel consumption of the entire machinery and the corresponding optimum modes of operation may be determined as shown in figure 11. It should be noticed, however, that not only the fuel economy but a variety of parameters such as the engine load limit curve, occurrence of critical shaft speeds, requirements on acceleration capability etc. have to be considered in the practical layout of the combinator curves.

With the increasing fuel prices it has become more and more important for the shipowner to ensure optimum operation of the ship's machinery. For that reason computerized fuel optimizing systems are now being developed in various quarters. Fig 12 shows the outline of one such system designated APO (Advanced Propulsion Operation) developed by the NKK company in Japan for their most recent classes of energy saving ships (6).

In fig 13 is shown the working principle of the KaMeWa FEC (Fuel Economy Control) system of which two prototype units have been built and installed on board ships for testing.

With the new computerized control systems it will be possible in the near future to advise the ship operator not only about the optimum propeller pitch ratio - rpm combination for a certain operating condition but also regarding what is the most economical speed at given conditions of weather - ice - water

depth, when to start up one more engine or one more propeller etc. No doubt, such systems will contribute to improve the fuel economy of the ships of tomorrow.

CP Propellers with Highly Skewed Blades

KaMeWa activities within the field of highly skewed propellers started in the summer of 1976 in connection with the project work for a new European class of twin screw frigates. The first full scale installation of a KaMeWa HS propeller ran trials in December 1977. This propeller was fitted on a single screw Ro-Ro ship which had until then suffered unacceptable vibrations with propeller blades of conventional design. The new HS blades completely solved the problem, reducing the vibrational impulses by more than 50 per cent, thereby bringing the hull vibrations to a level which was considered normal and fully acceptable. The improvement in vibration performance was achieved without sacrificing anything in terms of propulsion and manoeuvring performance.

Since then orders have been taken for altogether 167 KaMeWa HS propellers distributed on various ship types as follows:

Single screw merchant ships	56
Twin and triple screw merchant ships	61
Naval ships	50
Total	167

The above installations represent practically all types of military and merchant ships with the exception of icebreakers, tugs and supply ships for the offshore industry, the latter two types mostly being fitted with nozzle propellers. Of the 167 propellers ordered about 130 have run trials, in general with nothing but good results. Most of the propellers are installed on newbuildings but on 25 ships representing about 40 propellers old blades of conventional design have been replaced by new HS blades in order to improve the comfort level in the ship.

At the start of the HS development it was expected that HS propellers would mainly be requested for single screw ships

with poor hull wake distributions in order to solve a present or potential vibration problem. Very soon, however, the new blade design became quite as popular on many twin screw vessels, such as ferries and cruise liners where an improved comfort level for passengers has a direct impact on the commercial success of the ship. Today one can safely claim that on large passenger ships HS propellers are standard whilst non-skewed type of propeller blades are used only by exception. Hence, it is motivated to put the question which type of blades should be referred to as conventional.

A most interesting aspect on the HS propeller blade design is the possibility to utilize its inherent better vibration performance for the purpose of achieving improved propulsive performance by allowing a larger propeller diameter and accordingly smaller clearances between propeller blade tips and hull. This possibility which seems particularly attractive in combination with today's super long stroke low-speed diesel engines will be further discussed later on.

In the competition with fixed pitch propellers it should be noted that in general FPP blades cannot be designed with as large degree of skew as CPP blades due to the very high blade tip stress levels that will occur during stopping and backing manoeuvres with reversed direction of rotation of the propeller. In recent years several cases of blade tip failures on large highly skewed fixed pitch propellers have taken place during stopping manoeuvres whilst no single case of similar damage has been reported on CP propellers in spite of the much larger number of HS CPPs being in operation.

In conclusion one may claim that the HS propeller blades have a very pronounced natural direction of rotation and preferably they should not be rotated the other way at high speed or loading. Accordingly they are much more ideal for uni-directional CP propellers than for FPPs.

A short summary of the merits of the KaMeWa HS-CPP design is given in the enclosed article from the Motor Ship, August 1981 (app. 6).

Fuel Saving by means of Large Slow-Running CP Propellers

The trend towards larger propeller diameters, often in combination with geared ship machineries is now very noticeable throughout the world shipbuilding. In the table, fig 14, are summarized data of nine large CP propellers installed on medium size energy saving bulk carriers and tankers recently built in Japan. On most of these ships in addition to fitting a low speed - large diameter propeller a number of steps such as installation of exhaust gas economizers, shaft-driven generators etc. have been taken in order to improve fuel economy and for some vessels fuel savings in the order of 35 per cent are claimed in comparison with previous types of similar size ships.

To meet the demands for lower propeller shaft speeds two of the world's leading manufacturers of large crosshead diesel engines, MAN-B&W and Sulzer, have both developed new series of superlong stroke engines characterized by low shaft speed and low specific fuel consumption plus adaptability of fuel and scavenge air systems in order to optimize the operation. With some of these new engines, service propeller shaft speeds around 60 rpm are now available without any gearing of the machinery. However, on many ships, such as container ships, Ro-Ro ships, reefers, produc carriers etc. with limited space in the stern aperture, the lowest shaft speeds and corresponding low specific fuel consumption available cannot be fully utilized with propellers of conventional type, the main reason being the physical limitation in propeller diameter set up by the requirement on propeller - hull clearance, i.e. the minimum clearance between propeller blade tips and hull required in order to avoid unacceptable propeller-induced hull vibrations. As discussed above and in app. 6 this difficulty can be overcome by the use of highly skewed propeller blades which normally yield 50-75 % lower excitation in comparison with conventional blades.

The principle for the increase in propeller diameter and the corresponding changes in shaft speed, propulsive efficiency and fuel cost reduction related to the improvement in propulsion alone are roughly indicated in figs 15 and 16.

In order to illustrate more clearly what can be gained in a practical case, a study has been carried out for a new class of large single screw Ro-Ro ships presently under construction at a European shipyard. The prerequisits for this study were as follows:

The ships in question will be equipped with fixed pitch propellers which have been designed with the largest possible diameter considering vibration excitation and the corresponding optimum shaft speed. The blades of this FPP are of moderately skewed design, i.e. they have got the maximum degree of skew which is considered acceptable with regard to the risk of overstressing the blade tips during stopping and backing manoeuvres with reversed direction of rotation.

Comprehensive model tests including resistance, propulsion and cavitation tests with measurements of the propeller-induced pressure pulses had been performed at the Swedish Maritime Research Institute SSPA with actual models of the above FP propeller and hull. Based on given ship data and results from these tests a highly skewed CP propeller was designed with about 10 per cent larger diameter than the FPP (D = 7.9 m as compared to D = 7.2 m). The CPP alternative was designed such as to generate the same maximum level of pressure pulses against the hull with its propeller centre raised 350 mm (= half the increase in propeller diameter) above that of the FPP. Data for both propeller alternatives together with estimated full scale values of propulsive factors, power, fuel consumption and fuel cost savings at the ship's MCR and service conditions are given in the table of fig 17.

As can be seen from the table, an improvement in propulsive efficiency of about 4 relative per cent was expected with the CPP alternative. Moreover, the considerably lowered shaft speed would motivate the choice of a more modern and efficient diesel engine than the one which has been chosen for the FPP. Altogether this would result in a total fuel saving of about 9 per cent for the CPP as compared to the FPP alternative, which in turn would correspond to a cost saving of 1.5 - 2.3 million Swedish crowns (SEK) per year depending on the ship's service profile. Since the added price for the CPP and the

MCE engine would be in the order of SEK 6.7 million, a pay-off time of 4-5 years could be expected with an interest rate of 20 %.

With the data from the theoretical study above looking quite promising it was decided to manufacture also a model of the CP propeller (fig 18) and to conduct comparative propulsion tests at SSPA. The results of these comparative tests are summarized in figure 19 for the 16-21 knots speed range. As can be seen from the diagram, the improvement in propulsive performance with the CPP alternative became 2.8 relative per cent at the MCR speed 20 knots but as much as 5-6 per cent in the 16-18 knots speed range. Hence, for an average, the anticipated performance of the CPP alternative was well achieved. To follow up the propulsion tests, comparative cavitation tests including pressure pulse measurements with both propeller models are planned to be carried out in the near future.

Conclusions

In the chapters above the basic characteristics and some of the merits of the controllable pitch propeller have been briefly described. Many aspects on CPP propulsion have not been touched upon at all. In short, however, the main advantages of a CPP system may be summarized as follows:

CP propellers are well known since long and have proven reliable for the propulsion of all types and sizes of ships.

By the installation of a CPP system, the ship's manoeuvrability in terms of stopping, accelerating and docking is considerably improved in comparison with that of FPP ships.

Thanks to its continuous, unidirectional way of operation and readiness for load control by means of pitch adjustments the CPP system implies reduced engine and stern tube wear as well as simplified automation.

By its flexibility in adopting smoothly to different operating conditions (towing vs free running, shallow vs deep water, calm vs rough weather, fully loaded vs light ship, clean vs fouled

hull etc.) the CPP enables full utilization of the installed engine power at any service condition of the ship.

On many ships, more flexible and/or more economic machineries such as multi-diesel engine arrangements, combined diesel and gas turbine arrangements etc. may be installed thanks to the CPP principle. Constant speed shaft-driven electric generators, pumps etc. can easily be installed.

On ice-breaking ships, the ice-breaking capability is considerably improved with the CPP system thanks to its better low speed thrust performance and quicker manoeuvres.

On offshore vessels, precise manoeuvring and/or dynamic positioning is achieved much more easily with CPP than with FPP systems.

On CP propellers blades of highly skewed design can be fitted without any limitations in the power to be utilized ahead and astern. In comparison with blades of conventional design the HS blades yield superior vibration qualities.

Highly skewed CP propellers can be utilized for the purpose of improving ship's fuel economy by allowing smaller propeller - hull clearances with accordingly larger propeller diameter and reduced shaft speeds. The large diameter - low shaft speed concept yields improved propulsive efficiency as well as possibilities to use more efficient low speed diesel engines.

References

1. Lerbs: "Moderately Loaded Propellers with a Finite Number of Blades and an Arbitrary Distribution of Circulation", SNAME 1952.

2. Kerwin, J E: "Computer techniques for propeller blade section design", International Shipbuilding Progress Vol 20, 1973.

3. Morgan, Silovic, Denny: "Propeller Lifting Surface Corrections", SNAME 1968.

4. Minsaas, Slottelid: "Lifting Surface Corrections for 3-Bladed Optimum Propellers", NSFI publ No 107, Dec. 1970.

5. Abbot, v Doenhoff: "Theory of Wing Sections".

6. Yamashita, F: "Advanced Propulsion Operation System. New Operation System in NKK's New Energy-Saving Ships", NKK 1982.

Appendices

1. Brochure: Seffle Controllable Pitch Propellers
2. Brochure: KaMeWa CP Propeller - Main Principle
3. Brochure: KaMeWa XF1-type Propellers
4. Brochure: KaMeWa Electronic Control System for Main Propellers
5. Brochure: KaMeWa Marine Laboratory
6. O Björheden: The Design and Performance of Highly Skewed Controllable Pitch Propellers. The Motor Ship, August 1981.

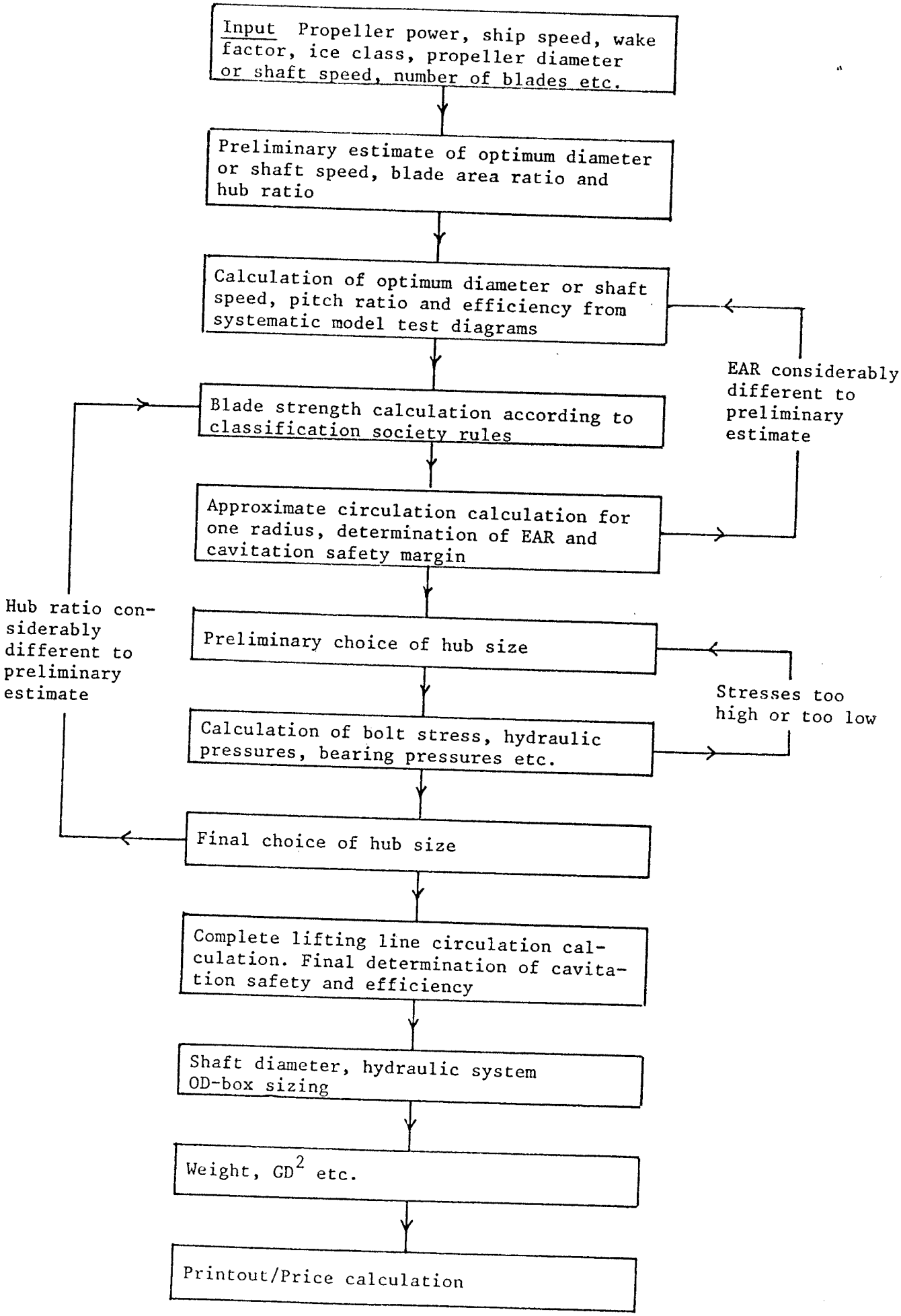
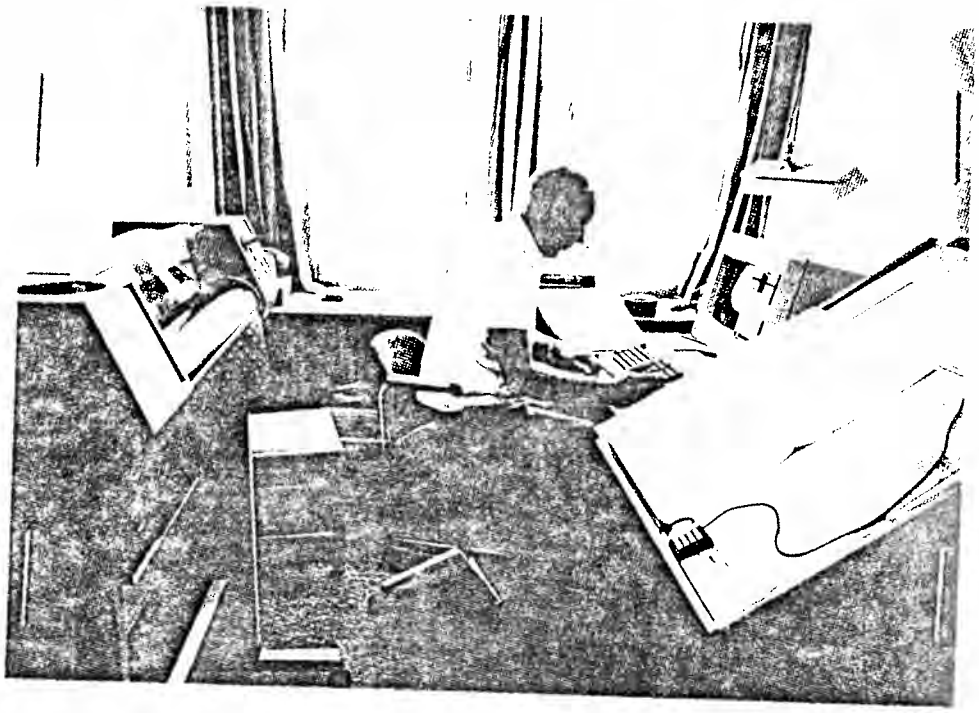


Fig 1. Block diagram, propeller project calculation



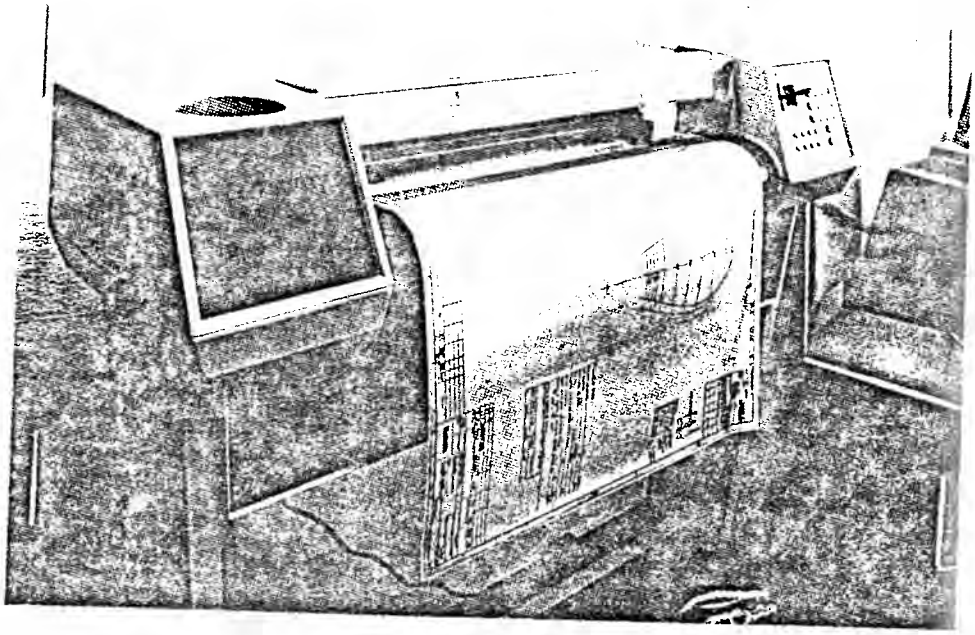


Fig 2. CAD working station with computer, digitizer and numerically controlled plotter

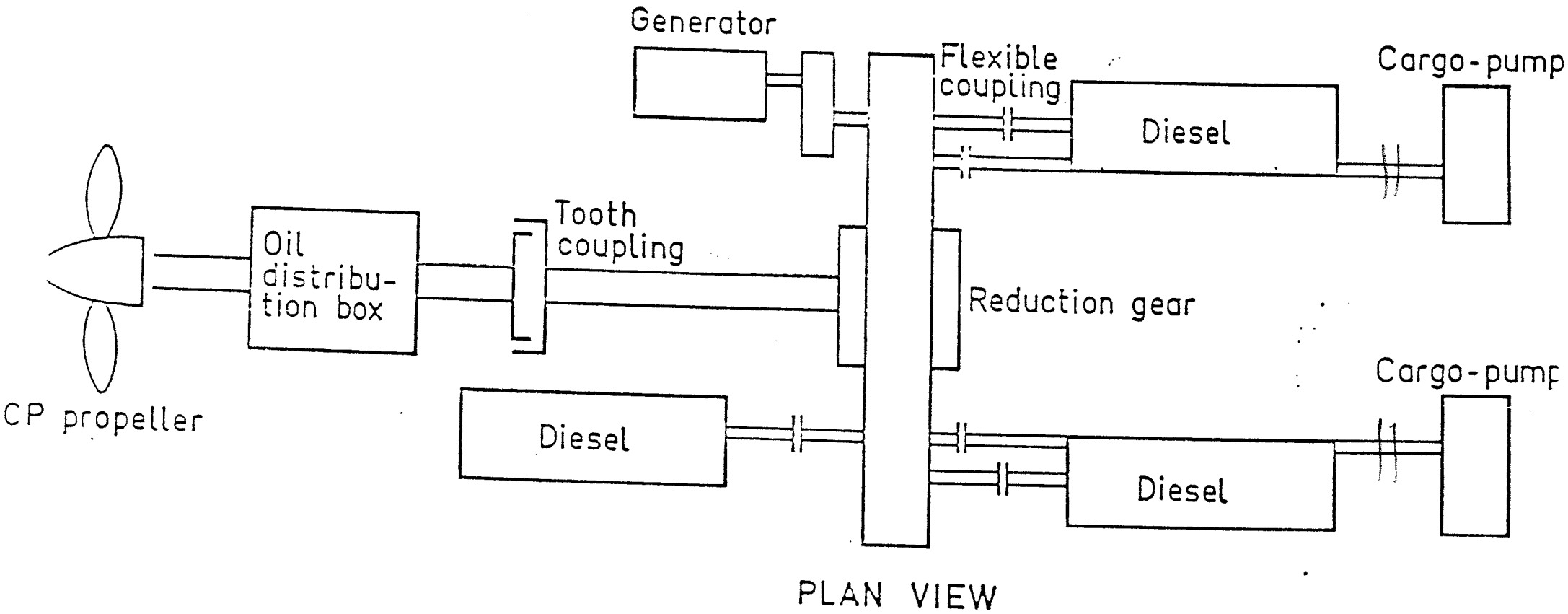
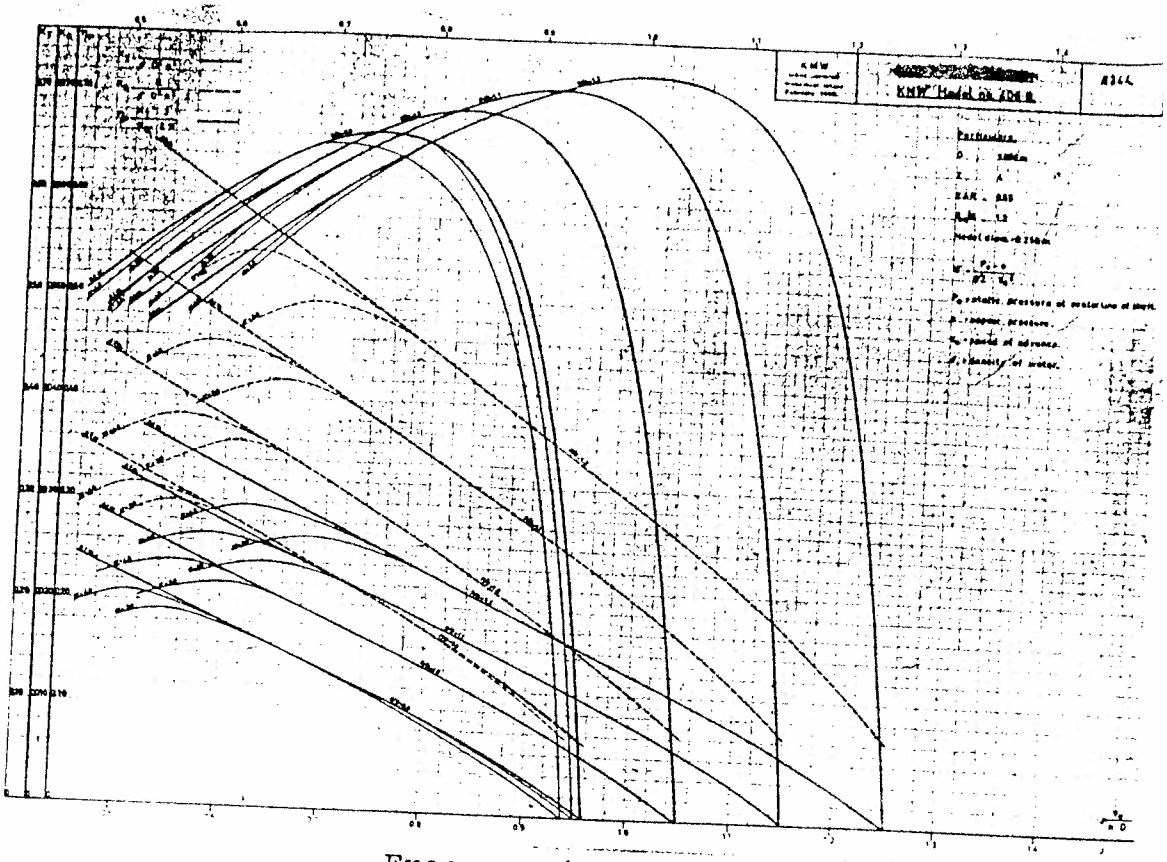


Fig 3. Multi-engine installation with CPP on a 130000 tdw tanker



Free running ahead

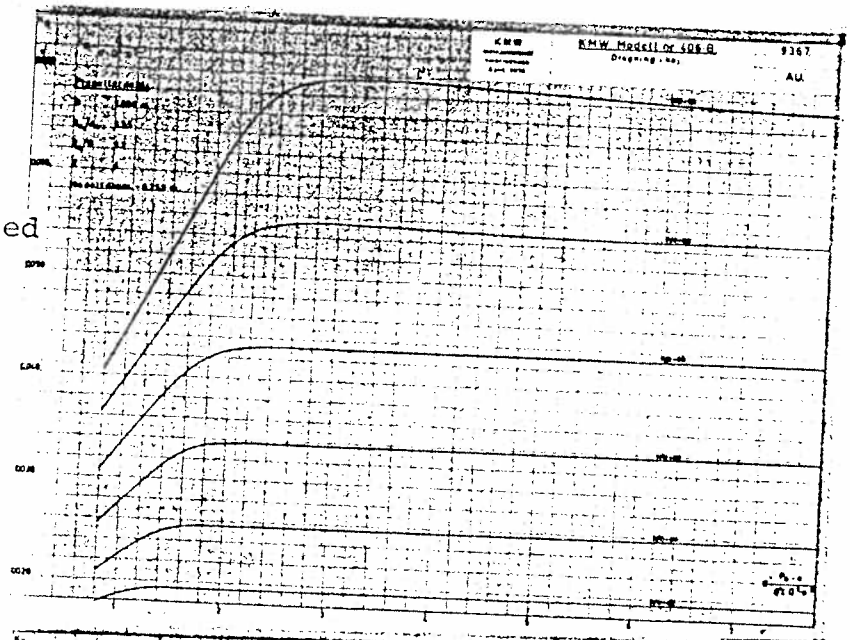
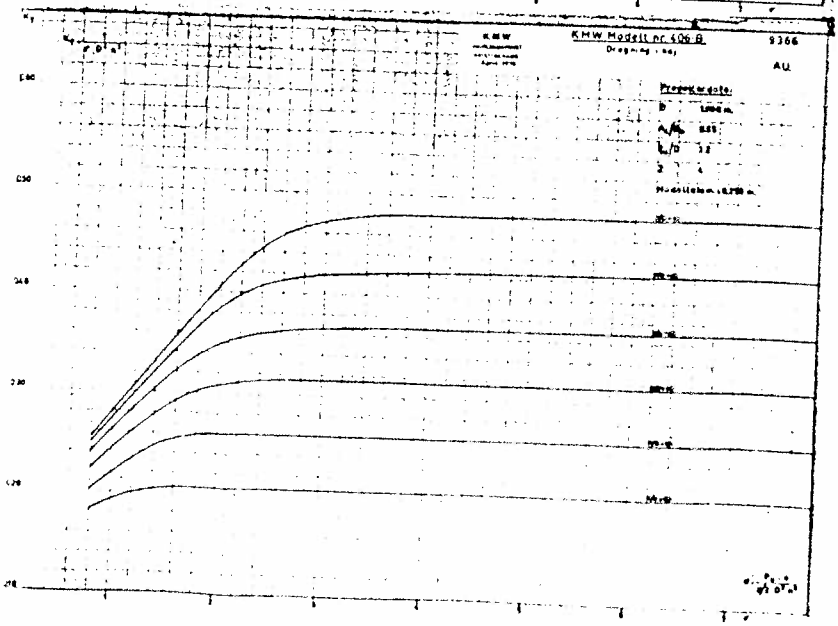


Fig 4. CP propeller characteristics obtained in cavitation tunnel

Bollard pull ahead



Bollard pull astern

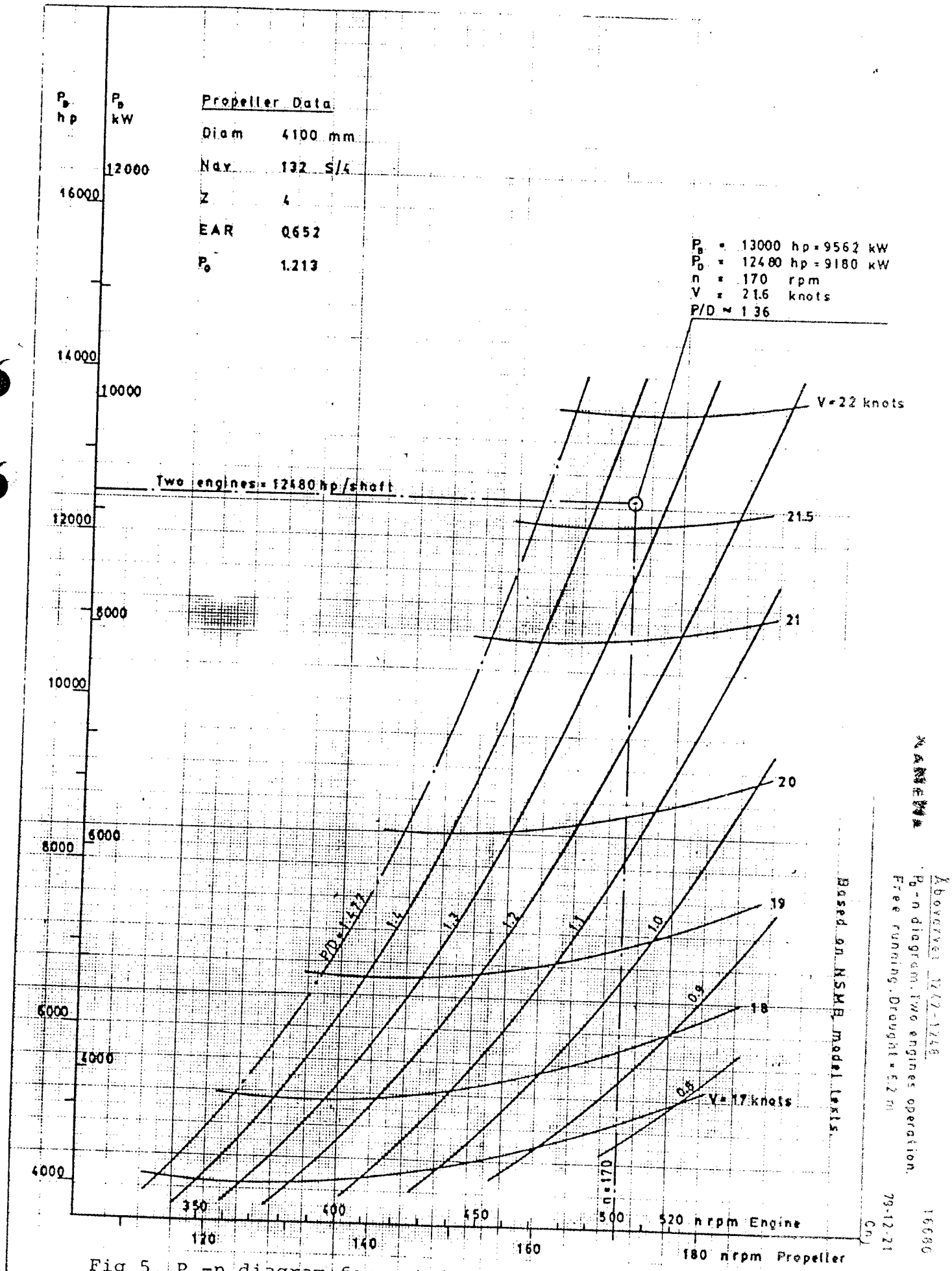


Fig 5. P_D-n diagram for a twin-screw ferry with two engines per shaft

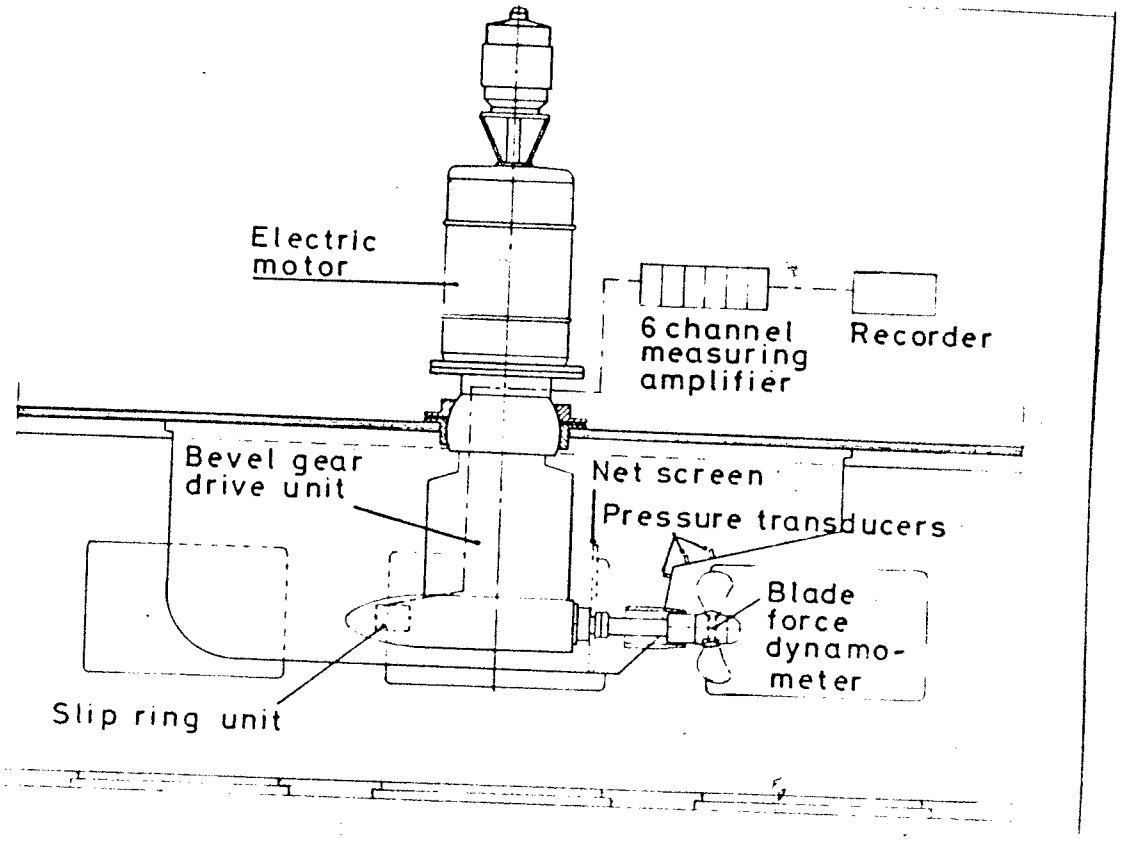
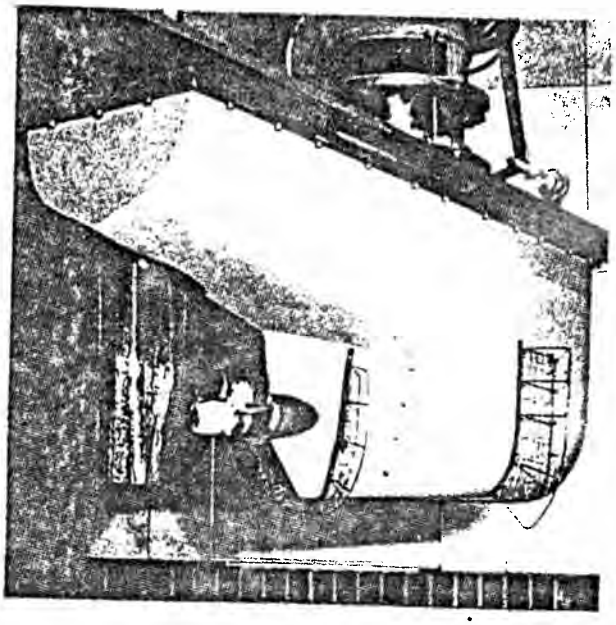


Fig 6. Model arrangement for afterbody tests in cavitation tunnel



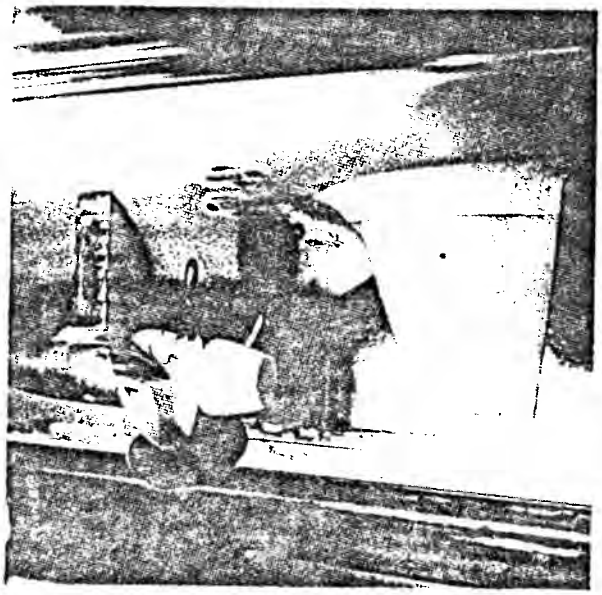


Fig 7. Afterbody arrangements for a single and a twin-screw ship

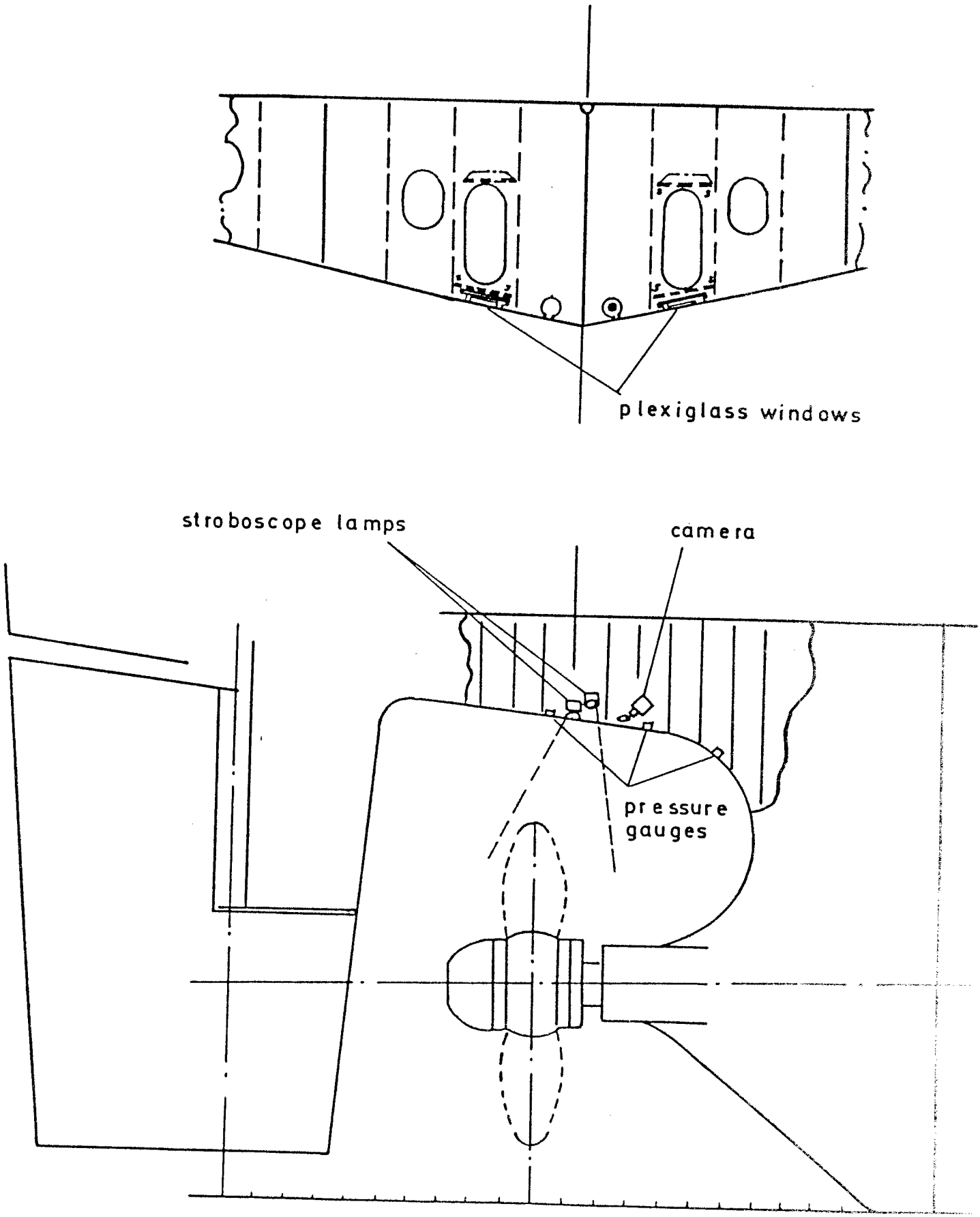


Fig 8. Principal arrangement for full scale cavitation test

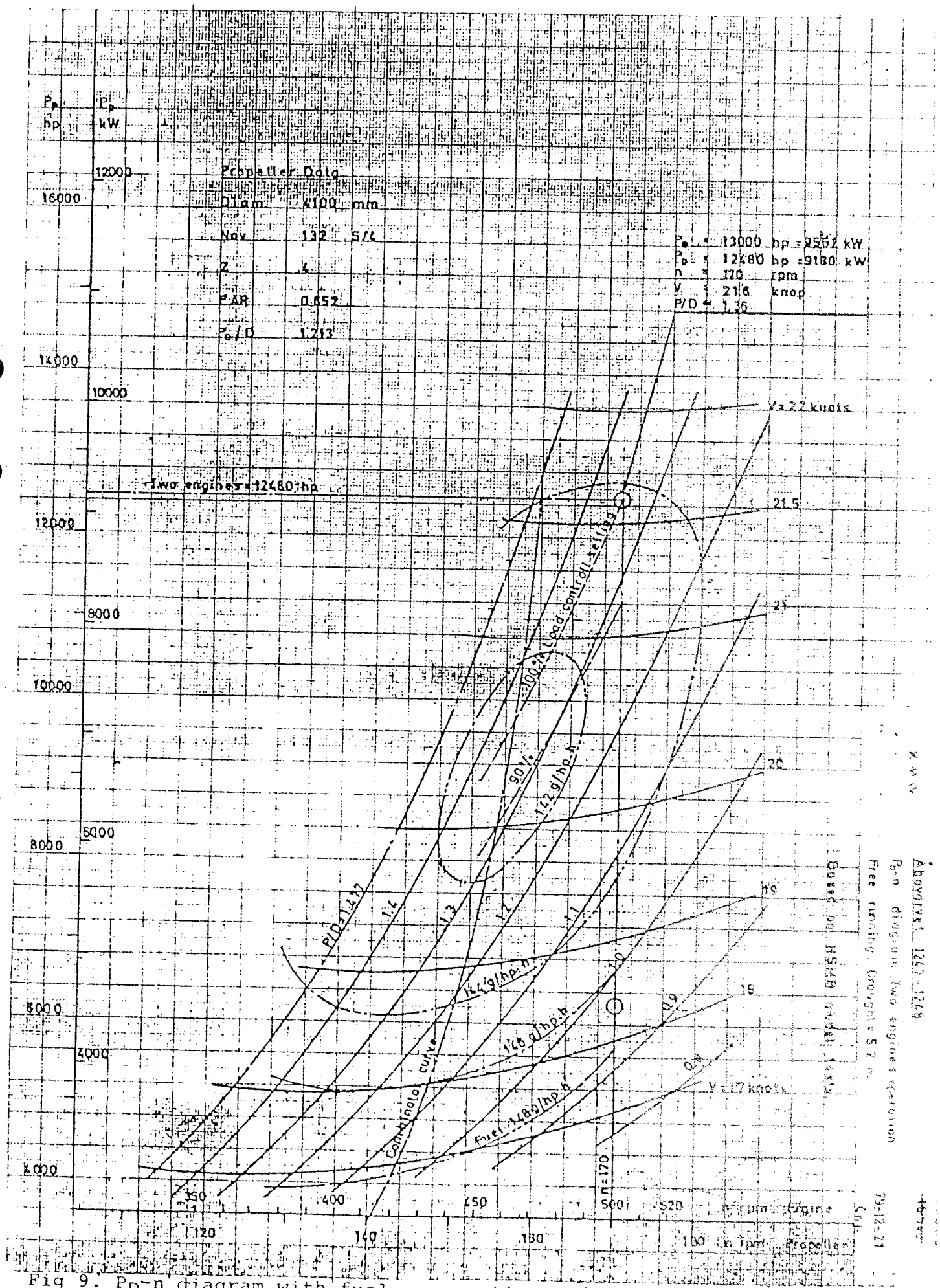


Fig 9. P_D-n diagram with fuel consumption curves, two engines per shaft

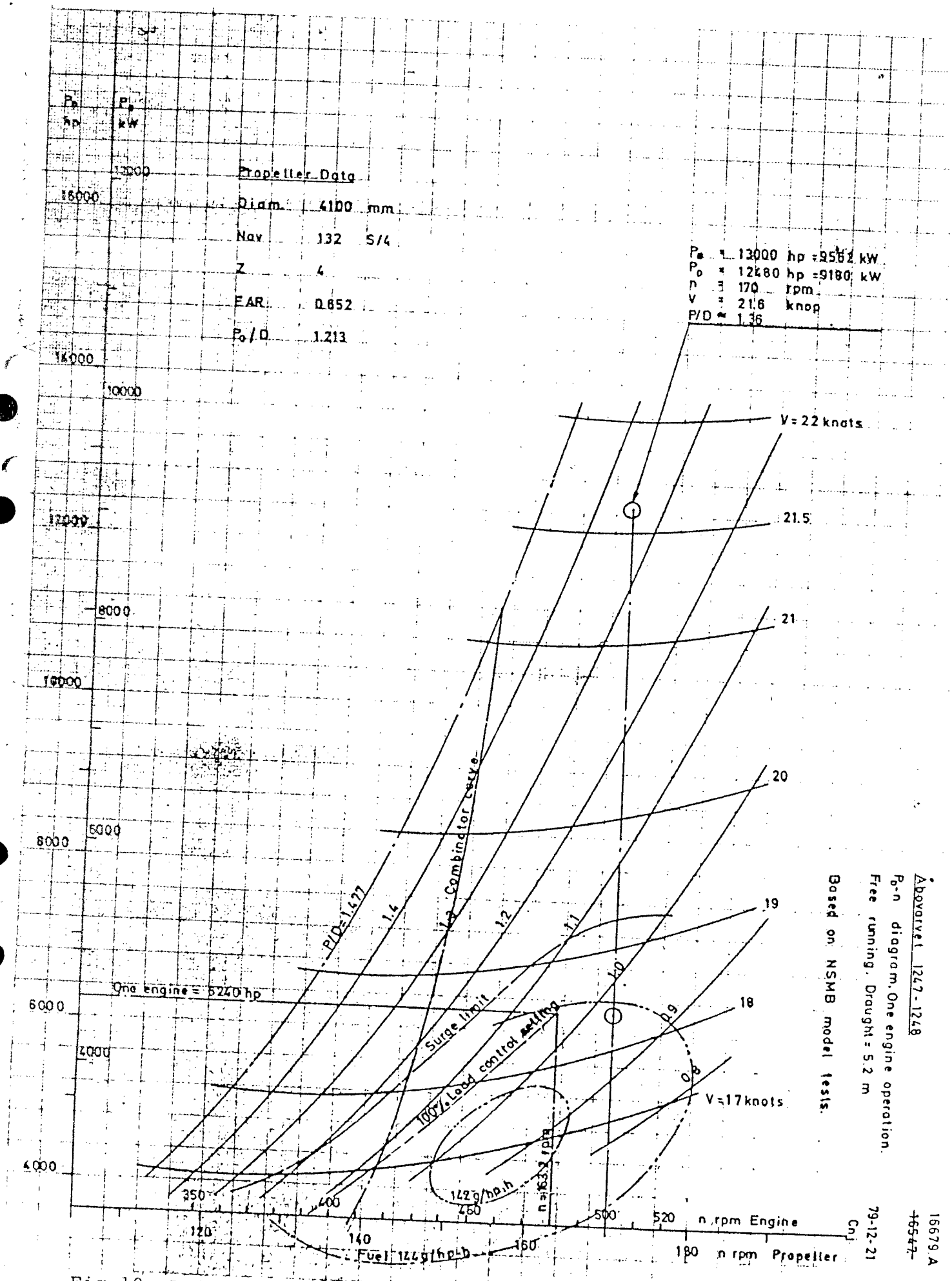
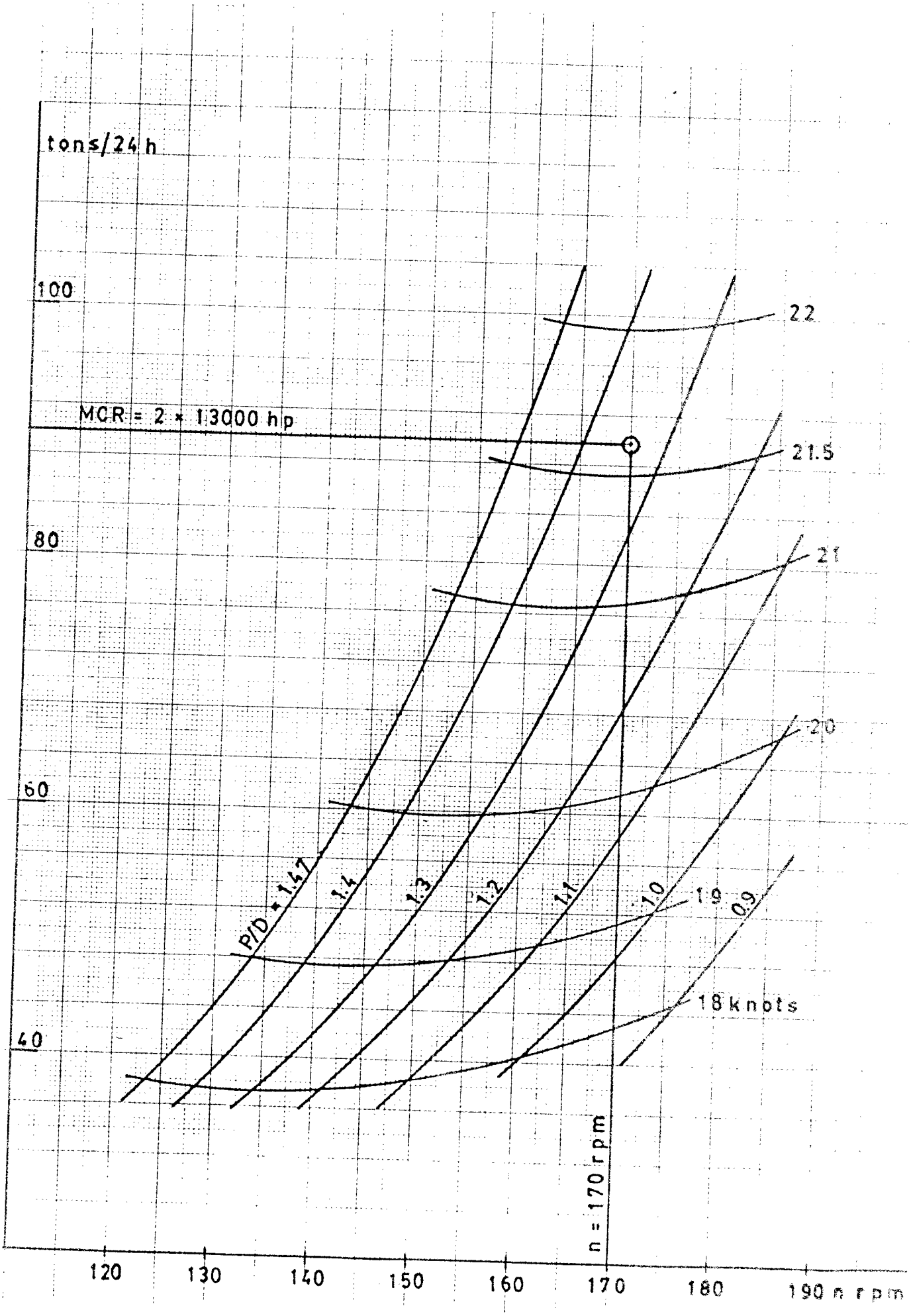


Fig 10. P_D-n diagram with fuel consumption curves, one engine per shaft

Fig 11. Fuel consumption of a twin screw ferry with two engines per shaft driving



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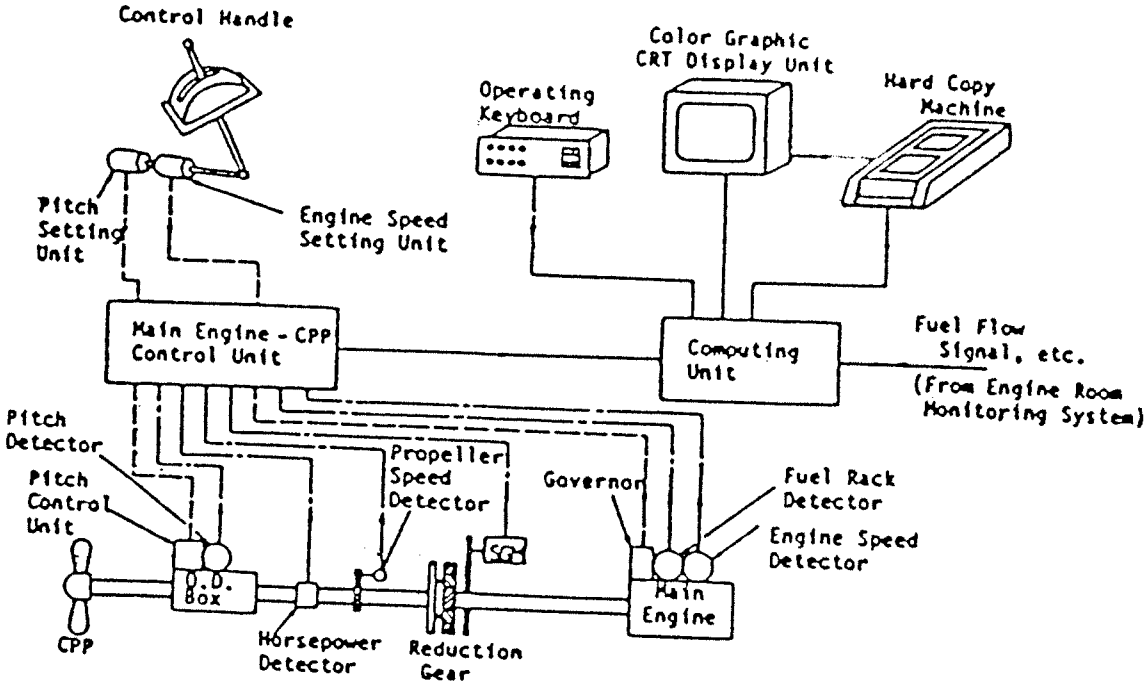


Fig 12. Outline of the NKK APO (Advanced Propulsion Operation) system

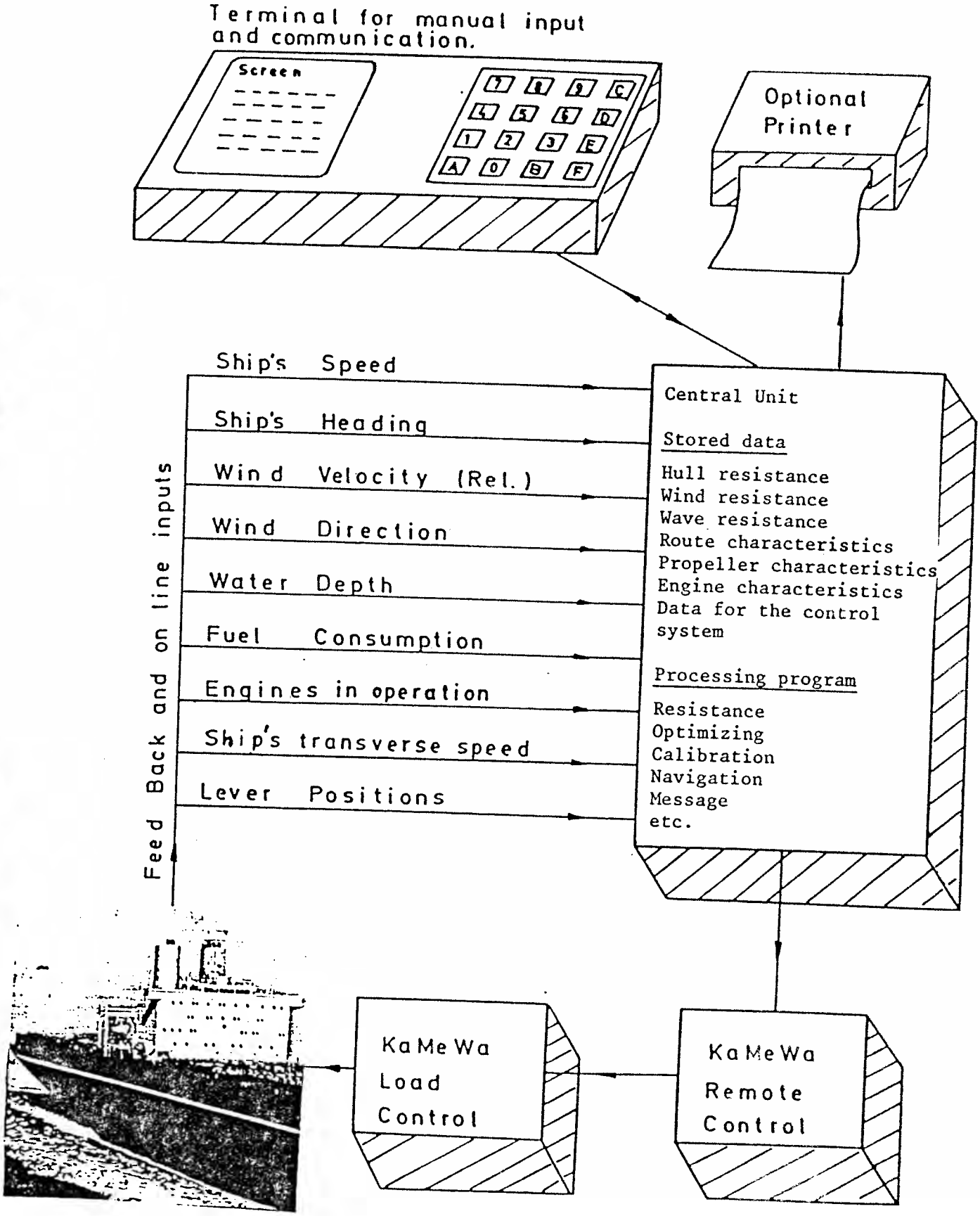


Fig 13. Working principle of the KaMeWa FEC (Fuel Economy Control) system

FIG 14

PARTICULARS OF LARGE CPP OF ENERGY SAVING SHIPS
(LARGER THAN 7.7M PROP. DIA)

NAME OF VESSEL	SHIP A	SHIP B	SHIP C	SHIP D	SHIP E	SHIP F	SHIP G	SHIP H	SHIP J
DELIVERY	JULY. 1982	SEP. 1982	JULY. 1982	AUG. 1982	MAR. 1983	—	—	SEPT. 1982	MAR. 1983
TYPE OF VESSEL & DWT	145 ORE/BC	145 ORE/BC	89 ORE/BC	209 BC	89 ORE/BC	145 ORE/BC	238 TANKER	208 ORE/COAL	224 ORE/COAL
PS × SHAFT RPM MR	16700 × 64	16700 × 64	14200 × 79	18900 × 60	14200 × 79	16400 × 62	24000 × 69.3	15500 × 45	16770 × 53
PS × SHAFT RPM SR	15000 × 62	15000 × 62	12800 × 76	16060 × 57	12800 × 76	14750 × 60	21600 × 66.9	13200 × 43	—
SHAFT GNE. & MOTOR (KW)	250	250	—	280	—	250	—	150	—
PROP. DIA × BLADE No.	9000 × 4	9000 × 4	7700 × 4	9300 × 4	7700 × 4	9200 × 4	9200 × 4	11000 × 3	9150 × 5
DIRECTION OF ROTATION	LEFT	LEFT	LEFT	LEFT	LEFT	LEFT	LEFT	RIGHT	RIGHT
TYPE OF PROPELLER	CPP, MITSUBISHI — KaMeWa	CPP, MITSUBISHI — KaMeWa	CPP, MITSUBISHI — KaMeWa	CPP, MITSUBISHI — KaMeWa	CPP, MITSUBISHI — KaMeWa	CPP, MITSUBISHI — KaMeWa	CPP, MITSUBISHI — KaMeWa	CPP, KAWASAKI — ESCHER WYSS	CPP, KAWASAKI — ESCHER WYSS
PROP. WEIGHT (TON)	63.2	63.2	43.3	66.6	63.2	65	68.1	76	—
PROP. SHAFT DIA.	930	930	795	880	795	930	870	890	—
EXH. GAS T/G	800 KW	800 KW	800 KW	850 KW	800 KW	800 KW	1500 KW	580 KW	—

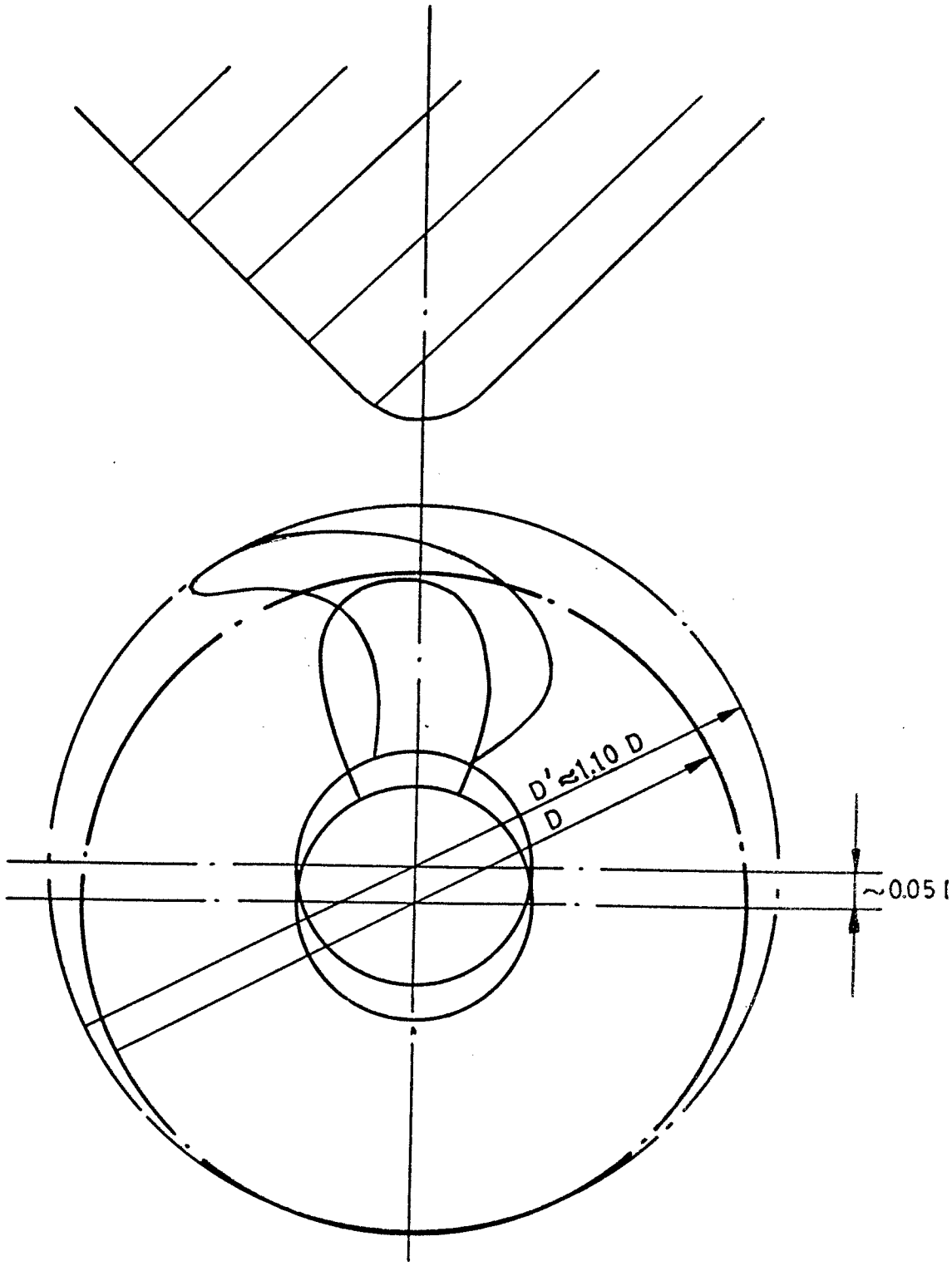


Fig 15. Principle of increased propeller diameter with Highly Skewed Propeller

PROPELLER AND ENGINE DATA

ALTERNATIVE	1	2
PROPELLER	FPP	CPP
DIAMETER D, M	7.20	7.90
HUB DIAMETER D, M	1.44	2.20
HUB RATIO	20.0	27.8
BLADE NUMBER Z	5	5
BLADE AREA RATIO EAR	0.69	0.48
PITCH RATIO P/D, ABT.	1.00	1.17
EFFECTIVE POWER, HP	16000	16000
PROPELLER SHAFT SPEED N, RPM	97	74
SPEED V_S, KNOTS	20.1	20.1
WAKE FACTOR W	0.238	0.232
THRUST DEDUCTION T	0.168	0.172
HULL EFFICIENCY $\eta_H = \frac{1-T}{1-W}$	1.092	1.078
RELATIVE EFFICIENCY	0.996	0.996
PROPELLER EFFICIENCY η_O	0.639	0.673
TOTAL EFFICIENCY η_D	0.695	0.723
PROPELLER POWER P_D, HP	23022	22126
ENGINE POWER P_B, HP	23260	22350
ENGINE (B&W)	6L90GBE	6L90MCE
SPEC. FUEL CONSUMPTION, G/HPH	128	121.3
FUEL CONSUMPTION, TONS/YEAR AT FULL POWER 300 DAYS/YEAR	21436	19520
CORRESPONDING FUEL COST, MKR (1200 KR/TON)	25.72	23.42
FUEL COST SAVING REL. ALT. 1, MKR	-	2.30
FUEL CONSUMPTION, TONS/YEAR AT 80 % POWER AND 250 DAYS/YEAR	14290	13013
FUEL COST, MKR	12.15	10.61
FUEL COST SAVING REL. ALT. 1, MKR	-	1.54

Fig 17. Data of two alternative propeller and engine installations on a single screw Ro-Ro ship

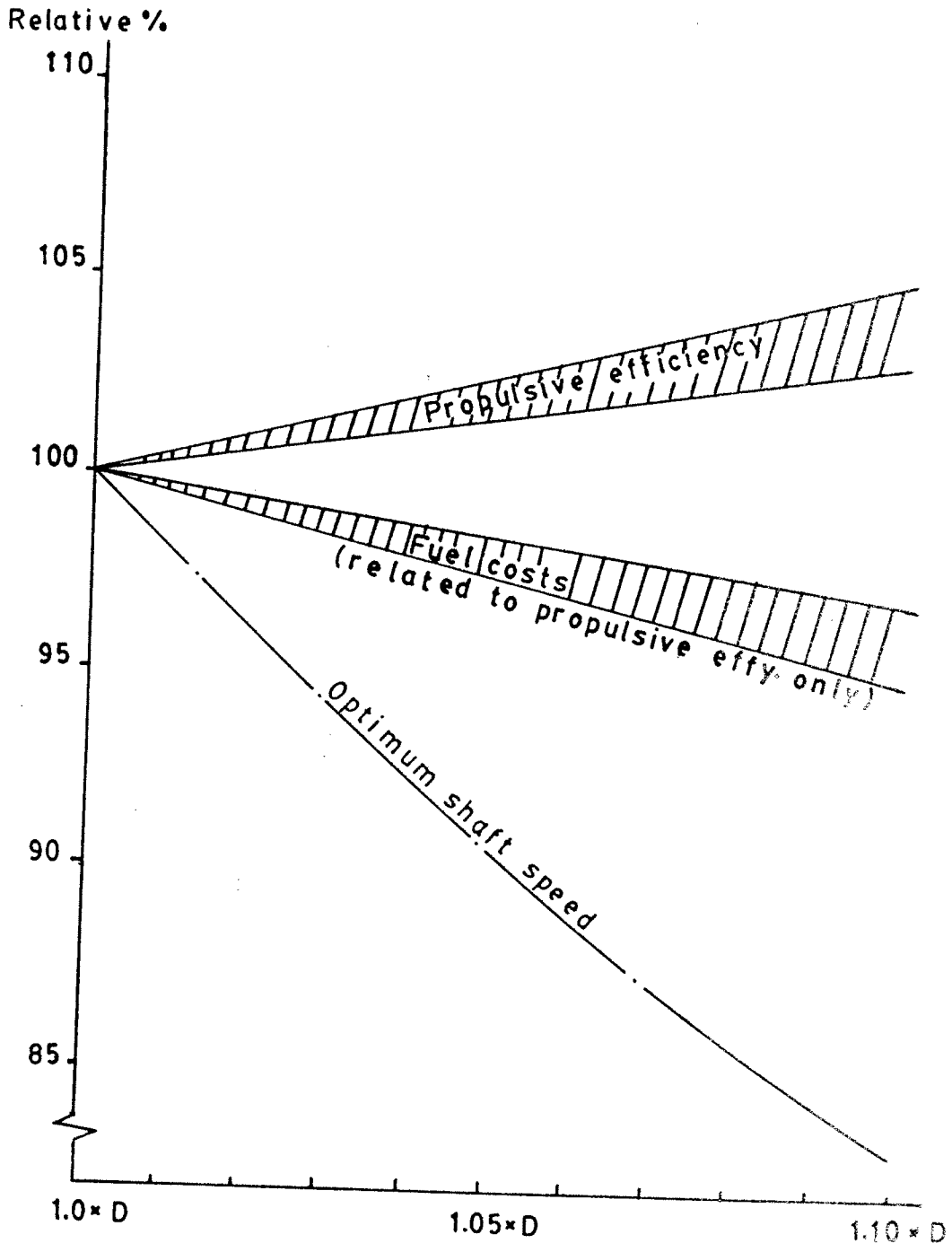


Fig 16. Improved fuel economy with increased propeller diameter

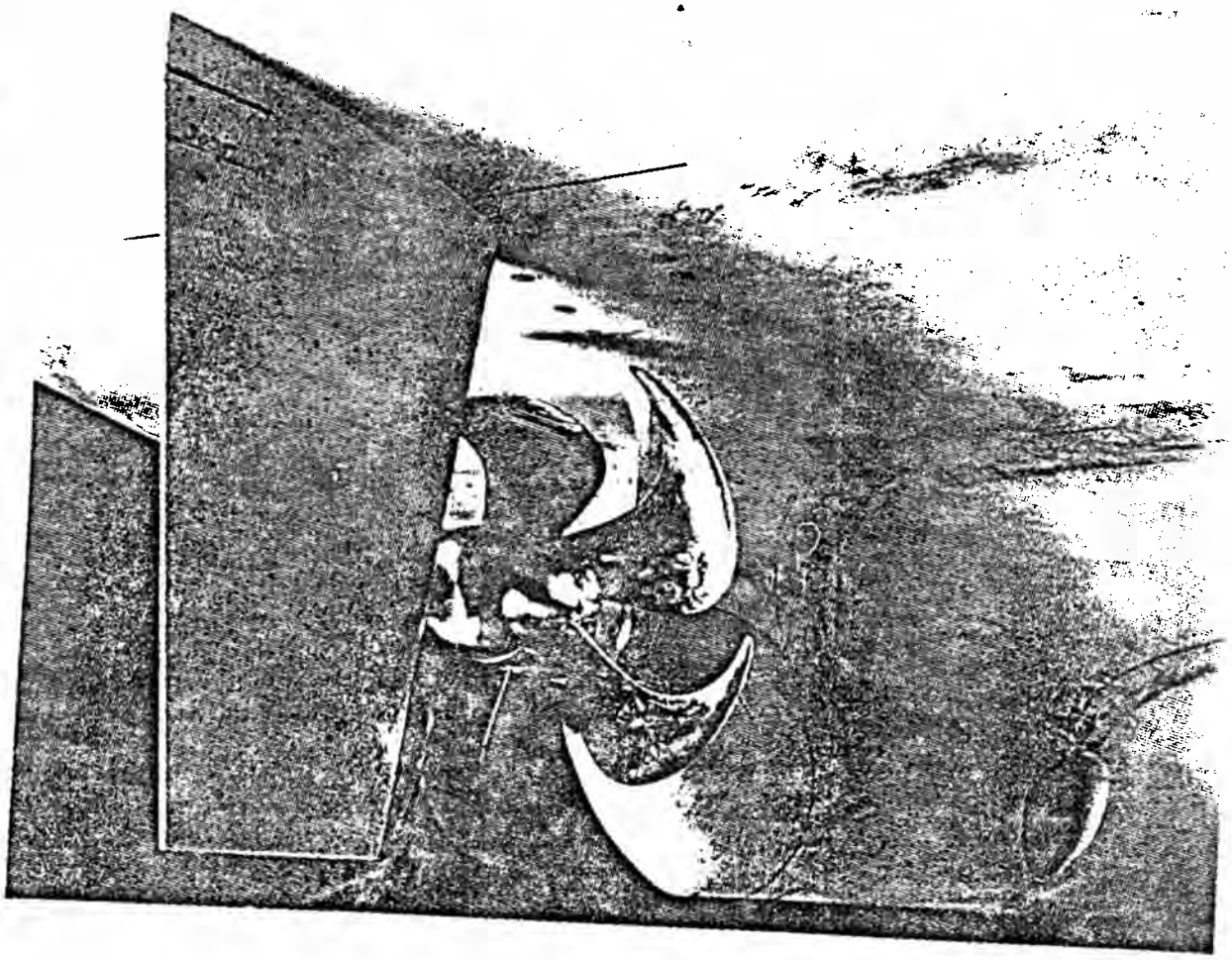


Fig 18. Model of large diameter Highly Skewed CPP

Single Screw Ro-Ro ship
Model test results and
comparison between CPP and FPP

83-06-06
Cnj..

		CPP	FPP
Propeller diameter	=	7900 mm	7200 mm
Blade shape	=	High skew	Moderate skew
Hub ratio	=	0.278	0.20
EAR	=	0.48	0.69
Z	=	5	5

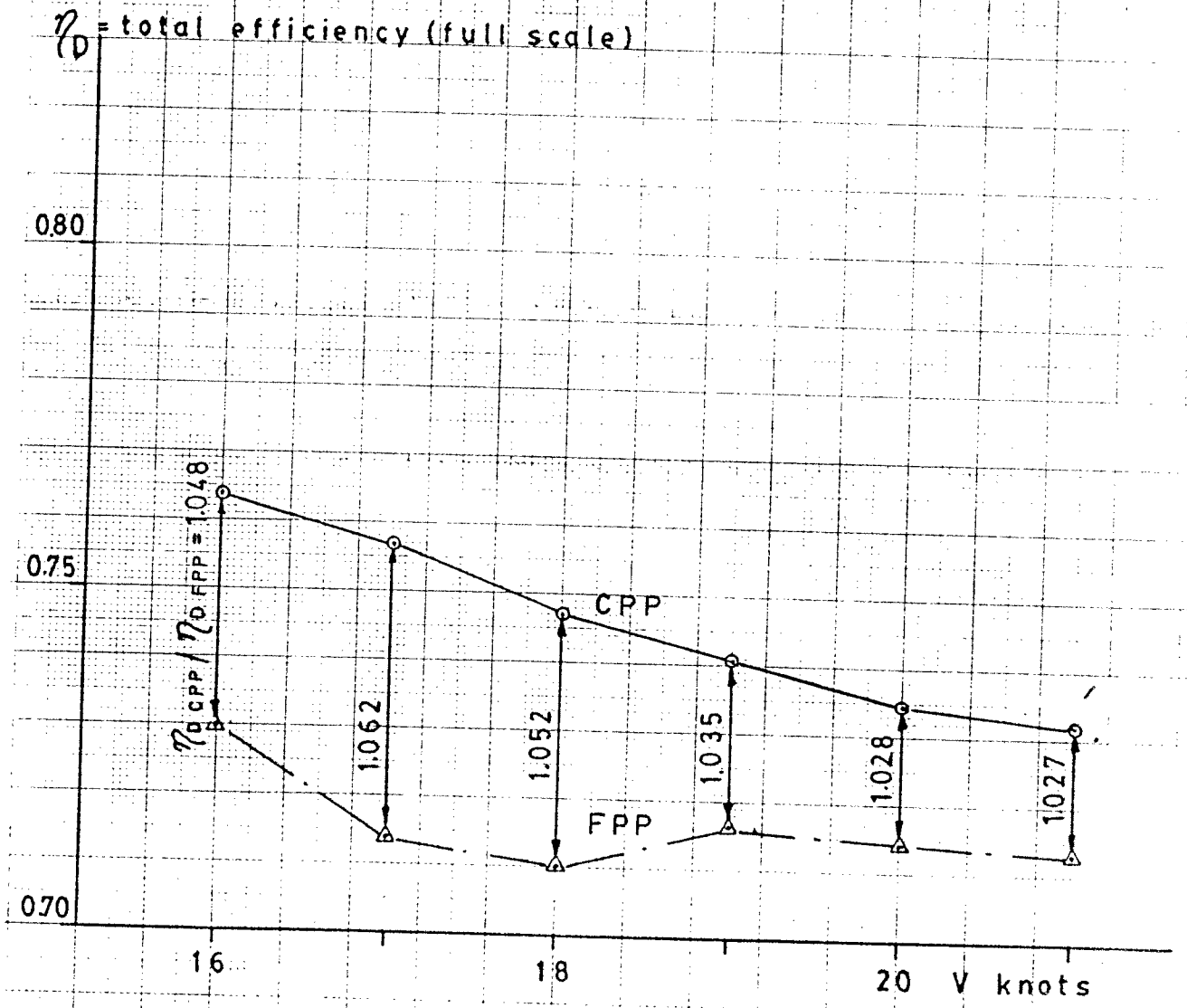
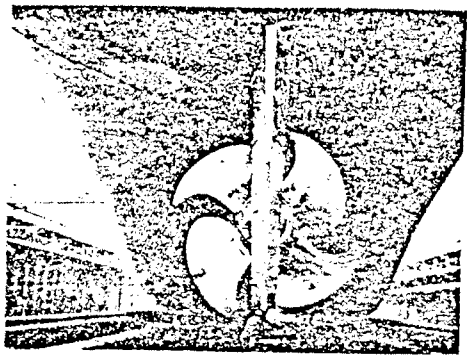


Fig 19. Results of SSPA self-propulsion tests with two alternative propeller installations on a single screw Ro-Ro ship

THE DESIGN AND PERFORMANCE OF HIGHLY SKEWED CONTROLLABLE PITCH PROPELLERS

Reprinted from THE MOTOR SHIP, August 1981



CONTROLLABLE PITCH propellers with highly-skewed (HS) blades are now being extensively used to ensure low noise and vibration levels in ships' accommodation combined with maximum propulsive efficiency and manoeuvrability, or even as a means to improve efficiency with maintained vibration and noise qualities. During the past three to four years more than 120 HS propellers representing an aggregate power of about 1·2 million hp have been ordered from the KaMeWa works in Sweden.

The HS propellers are fitted on virtually all kinds of vessel from low speed single-screw tankers to twin- or triple-screw naval craft with speeds in excess of 30 knots. Because of its outstanding vibration performance the HS propeller has become particularly popular on large twin-screw ferries and cruise liners, of which a considerable number have been built in Scandinavia in the past few years. The first KaMeWa propellers of the HS type in Canadian waters will come into operation in early 1982, fitted to a series of twin-screw British Columbia ferries now being reconstructed and repowered on the west coast. On these ships it has been possible to lower the vibration excitation by about 25 per cent in spite of a 50 per cent increase in propeller power. In the U.S.A. a successful replacement of propeller blades was made recently on a Great Lakes bulk carrier suffering from a vibration problem. With new blades of the HS type, propeller-induced vibrations were reduced by about 60 per cent.

Here, Orvar Björheden of KaMeWa's Marine Laboratory, Kristinehamn, Sweden, details the merits of highly-skewed c.p. propellers and reports on the excellent in-service performance.

Fig. 6 (Above) Highly-skewed KaMeWa propeller installed on the large reefer ship "Winter Moon", one of a series of six similar vessels operated by the Salen group.

The design and performance of *highly-skewed* controllable pitch propellers

IT IS well-known that propellers give rise to vibration and noise in the ship's afterbody, mainly due to the uneven hull wakes in which the propellers have to operate. Traditionally, the propeller-induced vibration impulses are divided into two categories: dynamic propeller forces and moments being mechanically transferred to the hull *via* propeller shaft and bearings, and fluctuating pressures acting on the hull plating adjacent to the propeller. The water-borne pressure impulses are largely influenced by wake-induced pulsating cavitation on the propeller blades and constitute on most ships the main source of excitation.

Highly-skewed propeller blades have been developed with the purpose of reducing propeller-induced vibrations through a softer interaction between the propeller and the ship's wake. To achieve this goal the skew curve has to mismatch the hull wake in such a way that various parts of the blade do not simultaneously enter the high wake regions behind the ship stern. A further aim with the skewed blade shape for c.p. propellers is to minimise the blade spindle torque during pitch adjustments.

Blade design

The HS blade design of a four-bladed KaMeWa propeller for a single-screw merchant ship is shown in Fig. 1. The design is an example of a propeller with balanced skew, which implies that the mid-part of the blade is drawn somewhat forward of the blade spindle axis whilst the tip is strongly swept backwards. Moreover, the blade has a forward rake, the blade angle being of such an order as to put the blade tip in a plane through the propeller centre when the pitch is adjusted at its design value.

The hydrodynamic design of the blade is based on vortex theory calculations according to lifting line as well as lifting surface theory. The radial distribution of bound circulation is chosen partly with the purpose of suppressing blade tip cavitation as well as tip and root vortices. The unloading of these parts of the blade is not of such a magnitude, however, as to influence noticeably the propeller efficiency.

Model testing

At the KaMeWa Marine Laboratory comprehensive model tests have been carried out in the cavitation tunnels in order to investigate the characteristics of the HS propeller (Ref. 1). The investigations have involved the determination of efficiency and thrust performance at free running and bollard pull conditions, ahead and astern, by tests in homogeneous parallel flow. An example of test results is shown in Fig. 2. Propeller cavitation and vibration performance have been evaluated by tests with model propellers operating in simulated wake fields behind ship afterbody models.

The "behind" tests have included measurements of the propeller-induced pressure impulses acting on the hull plating as well as measurements of the dynamic propeller and propeller blade forces and moments. For several projects the dynamic propeller blade stress distribution has also been mapped out with the aid of strain gauges fitted to the model propeller blade (Fig. 3). In all, about 50 propeller models have been investigated and for more than 20 projects pressure impulse measurements with HS blades as compared with conventional blades have been carried out. A summary of measured impulses for the first seven ships is given in Fig. 4.

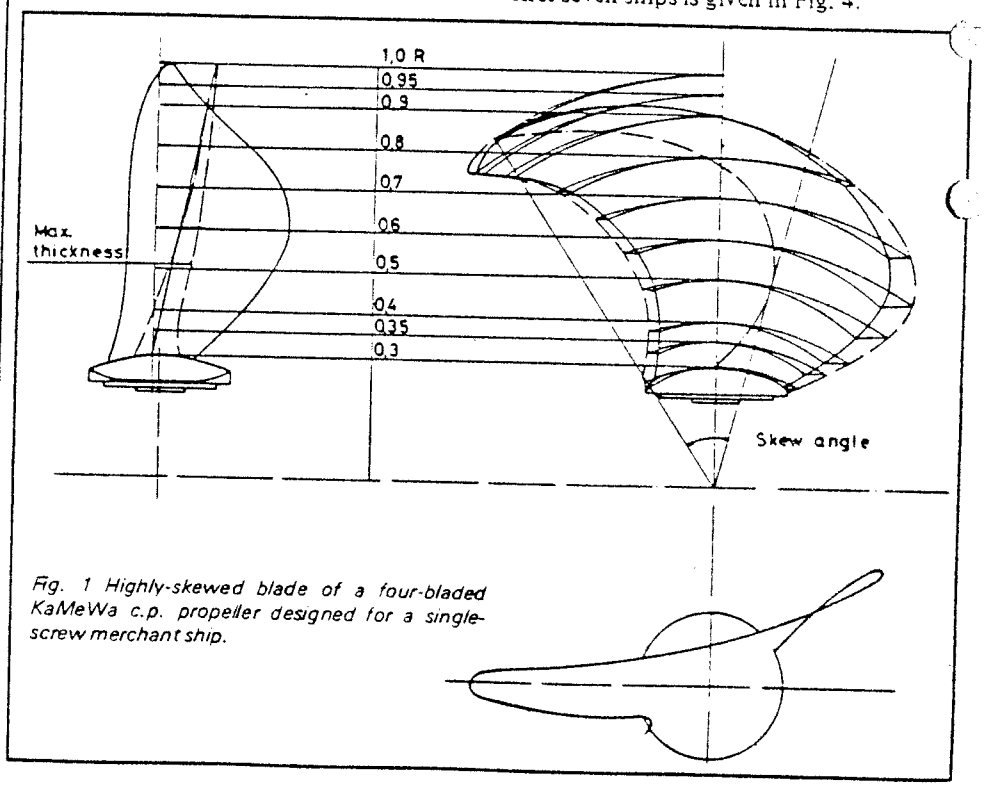
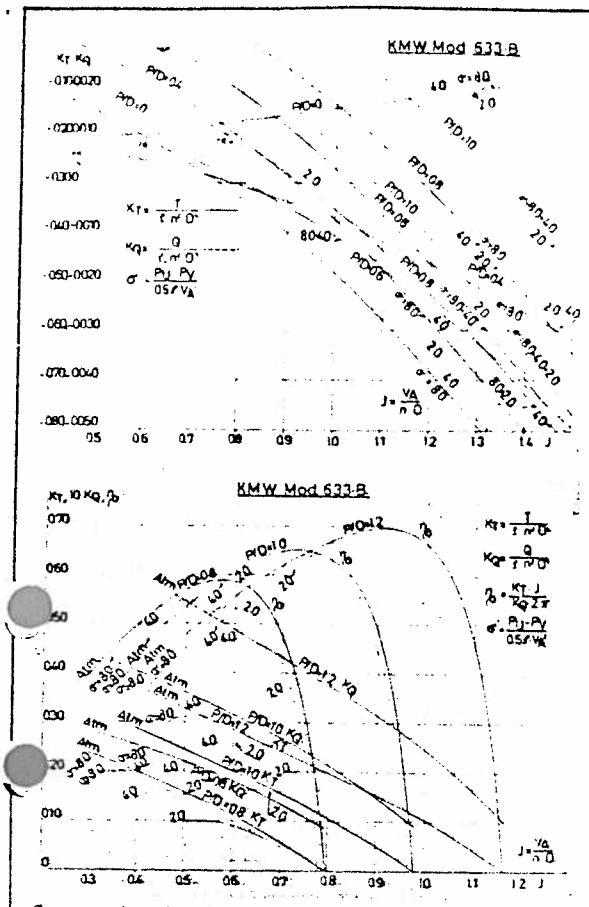
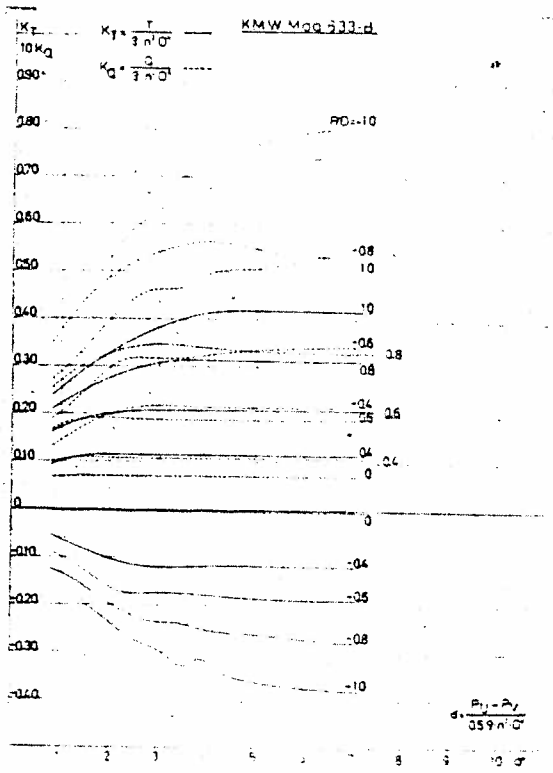


Fig. 1 Highly-skewed blade of a four-bladed KaMeWa c.p. propeller designed for a single-screw merchant ship.



(Left) Propeller-turbine range.

(Right) Bollard pull ahead and astern.



Free running ahead.

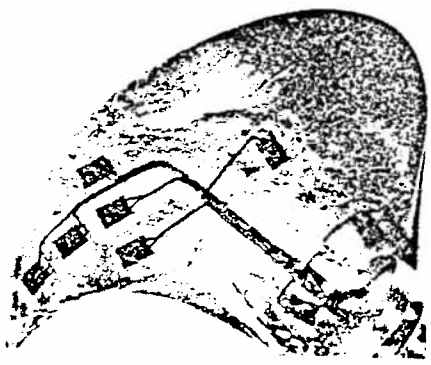
Fig. 2 Characteristics of KaMeWa model propeller 533-B designed for a single screw 20 500 hp car carrier: Diameter D 6·1 m; Blade area ratio EAR 0·554; Design pitch ratio P_0/D 1·177; Thickness ratio t_0/D 0·0565; Skew angle ϑ 48°.

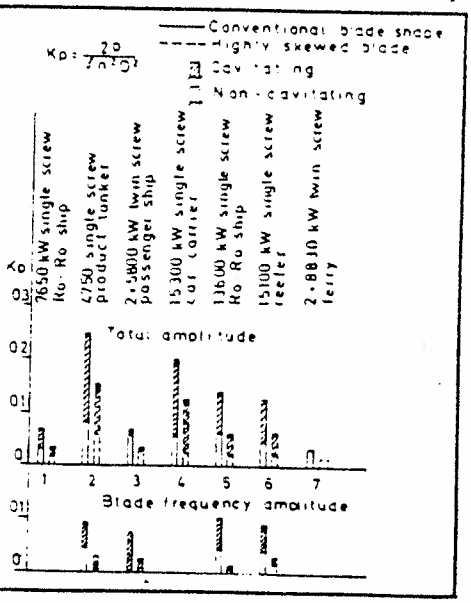
To illustrate more clearly the influence of blade skew on the propeller-induced pressure impulses against the hull a series of four propeller models with systematically varied degree of skew have been manufactured and tested. All four models have been designed for a certain single-screw merchant ship with identical conditions as regards propeller power, shaft speed, ship speed, wake distribution and propeller diameter, and with identical radial distributions of load, blade section lengths and thicknesses. For the design condition corresponding to the normal service condition of the ship the propeller-induced pressure amplitudes have been recorded with the propeller models operating in the wake field behind a hull dummy. The result is shown in Fig. 5. Briefly, the findings of the model investigations may be summarised as follows:

- Propeller blades of the highly-skewed type are applicable to c.p. propellers and will result in considerably reduced vibration impulses without sacrificing propulsive and manoeuvring performance in comparison with blades of conventional shape. The improved vibration performance is achieved mainly due to considerably reduced fluctuating pressures on the ship's hull surface, but the dynamic propeller and propeller blade forces and moments are normally reduced as well.
- At the bollard pull condition the HS c.p. propeller normally yields a somewhat lower thrust ahead but, on the other hand, a considerably increased thrust astern compared with that of a conventional c.p. propeller, which is considered a benefit on most ships.
- The HS blades may also be designed in such a way that the blade spindle torque required for pitch adjustment becomes somewhat lower than that of more conventional blades.
- HS blades yield improved qualities with respect to incipient cavitation, thereby enabling higher cavitation-free speeds to be achieved than with blades of conventional design (naval craft).
- The main disadvantages of the HS blade design are related to the elaborate design work and complicated manufacture which call for a somewhat higher price than that of conventional blades. Moreover, during reversal of c.p. propellers the HS blades require somewhat more longitudinal space in the stern aperture than blades of conventional design.

Service experience

So far 64 KaMeWa propellers with HS blades have come into operation, all of them with successful results. Fig. 6 shows the propeller installed on a series of six large reefer ships built by the Götaverken shipyard for the Salén Group in Sweden (*The Motor Ship*, April and June 1979). On about a dozen ships full-scale measurements of the propeller-induced pressure impulses against the hull plating have been carried out, normally in conjunction with comprehensive surveys of the ship vibration pattern. In all cases the measurements have confirmed the model results as regards the propeller excitation level and on many





(Above) Fig. 3 Model propeller blade instrumented for strain measurements.

(Right) Fig. 4 Maximum propeller-induced pressure amplitudes on hull plating measured during model tests with both conventional and highly-skewed c.p. propellers.

ships maximum vibration velocity levels as low as 1-2 mm/s have been recorded at full engine power (Refs. 2 and 3). Moreover, the full-scale trials have fully demonstrated the good propulsive and manoeuvring performance of the HS propellers.

On a number of vessels the existing conventional blades have been replaced by new ones of the highly-skewed type. On six single-screw cargo vessels the propeller-induced pressure amplitudes with the original blades (not of KaMeWa make) have been of such an order as to prevent exploitation of the full engine power due to unbearable vibration levels or even cracking of the hull plating in areas above the propeller. After retro-fitting HS blades all vessels are now considered completely normal from a vibration point of view and the full engine power can be utilised.

On about 10 ferries and cruise liners existing blades of the conventional type have been replaced by HS blades in order to improve comfort in passenger and crew cabins. The first example was the "Song of Norway" on which considerably reduced noise levels were recorded in the aftermost accommodation after the retrofit.

Efficiency gain potential

The inherent better vibration performance of the HS blade design may be applied for the purpose of achieving improved propulsive performance by allowing a larger propeller diameter and accordingly smaller clearances between propeller blade tips and hull. A suitable criterion could be to aim at the same vibration level in the ship as that resulting from a conventional propeller design with a normal tip clearance. With the assumption that the HS propeller design nominally yields 50 per cent less excitation and, moreover, that the pressure impulses against the hull decrease inversely proportional to the tip clearance to the power of 1·5, calculations indicate that a propeller diameter can be chosen about 10 per cent larger with HS blades than with conventional blades for approximately the same vibration level in the ship. The optimal propeller shaft speed should be chosen in both cases, which implies a shaft speed about 15-20 per cent lower with HS blades than with the conventional type. The propeller blade area in turn may be somewhat reduced with a larger diameter and will become 12-15 per cent lower with the HS blade installation.

An examination of a number of representative project cases shows that the gain in propulsive efficiency with the larger diameter will be between 3 and 5 per cent, depending to a certain extent on the propeller loading.

One may of course discuss the priority of comfort level versus a possible efficiency gain when choosing larger propeller diameters and smaller tip clearances. In times of increasing fuel prices, however, the issue can certainly arise and the potential of the HS propeller in this respect should be of interest to many shipowners.

So far service experience with HS propellers on ice-going ships is relatively scarce. On a number of Baltic ferries and cargo vessels for the traffic between Finland and Sweden five propellers have now been in operation during two normal winter seasons and eight more propellers have been in service for one season. All the propellers have been designed to fulfil the Finnish-Swedish ice class 1A requirements and Ni-Al bronze propellers as well as propellers in stainless steel are represented. No malfunction of any of these installations has been reported (such as vibrations) which could be related to ice damage and on those vessels which have entered drydock no damage has been found at the inspections.

For light ice conditions involving breaking of thin ice or traffic in broken channels there is reason to expect some advantages with the HS blade shape compared with more conventional shapes, mainly with regard to the slope of the blade leading edge at the outer propeller radii. The HS blade shape thus brings about a reduced angle of the leading edge contour relative to the incoming flow as compared with more symmetrical blade shapes, which in turn should imply less violent acceleration of and shock loads from ice blocks colliding with the edge.

Finite element calculations of stresses and deformations of propeller blades being exposed to extreme ice loads over the blade tip region have been carried out at a few institutes in Scandinavia. The calculations show that under these conditions, corresponding to crushing or milling of solid ice, blades of the HS type will get a different stress distribution with considerably higher maximum stresses than corresponding blades of conventional type. Recent investigations at the KaMeWa Marine Laboratory, including FEM-calculations as well as comparative stress measurements on model propellers, have demonstrated, however, that this drawback can be overcome by a particular modification of the blade shape without sacrificing any of the good qualities of the basic HS design. Hence, there are also good prospects for using the HS blade design for operation in severe ice conditions in the future. □

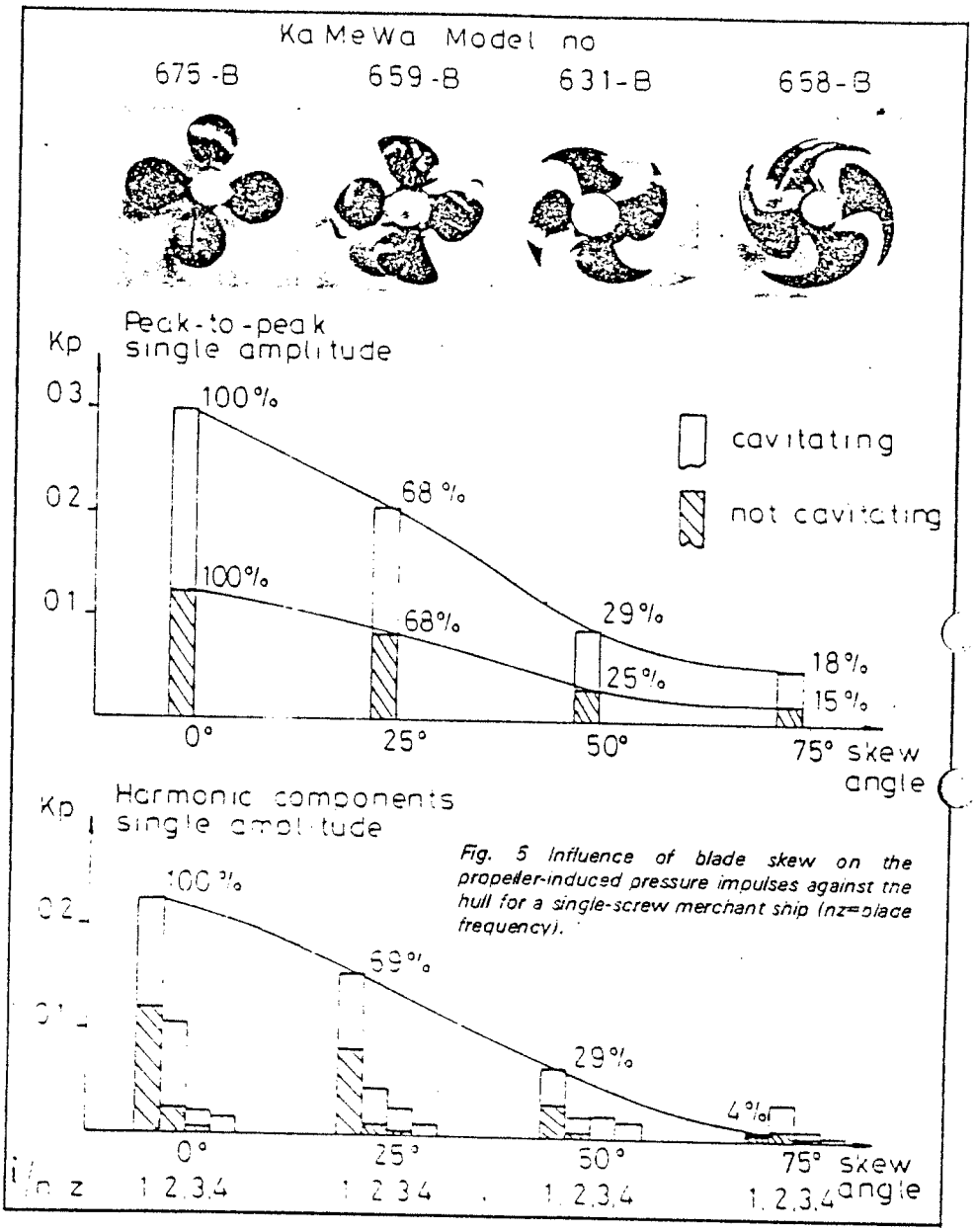


Fig. 5 Influence of blade skew on the propeller-induced pressure impulses against the hull for a single-screw merchant ship (nz=blade frequency).

(1) O. Björheden: Highly-skewed controllable pitch propeller. German Ship Technical Society, STG, Berlin, November 1978 (in German).

(2) O. Björheden: Vibration performance of highly-skewed c.p. propellers. RINA, London, December 1979.

(3) O. Björheden: Experiences of controllable pitch propellers with highly-skewed blades. Nordic Ship Technical Meeting, NSTM, Lyngby, 1980 (in Swedish).

DUCTED PROPELLER SYSTEMS AND ENERGY SAVING

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Lecture notes prepared for
8th WEGEMT GRADUATE SCHOOL
August 29 - September 9, 1983
GOTHENBURG

ABSTRACT

Modern applications of ducts in ship propulsion systems show a great variety. Most remarkable is the trend to locate the duct ahead of the propeller. Energy savings are compared with those of other devices. The combination of propeller and duct is discussed on the base of hydrodynamic aspects.

It is indicated that these considerations also apply to the design of non-axisymmetric ducts or other devices which exert lifting forces.

Design methods for conventional applications are reviewed and possible further development is suggested.

Energy saving with ducted propeller systems can be obtained by mutually adapted design methods and model scale information.

CONTENTS:

1. INTRODUCTION
2. HYDRODYNAMIC DESCRIPTION
2.1. Circulation around duct
2.2. Onset flow conditions
2.3. Viscous effects on the duct
2.4. Propeller and duct induction and interaction with shear flow
3. DESIGN PROCEDURES
3.1. Design based on systematic series
3.2. Design of propeller for given duct shape
3.3. Design of ducted propeller systems
4. MODEL TESTING
4.1. Flow measurements
4.2. Extrapolation of model test results to ship scale
5. FINAL CONSIDERATIONS
References
Appendix

1. INTRODUCTION

The main requirements for a ship propulsion system are:
- a high power efficiency,
- minimum danger of cavitation erosion,
- minimum propeller induced vibration,
- good stopping abilities,
- favourable interaction with rudder for manoeuvring,
- dependability, minimum vulnerability,
- low building and maintenance costs.

In many cases the ducted propeller fullfils, for various ranges of operating conditions, these requirements. The type of duct which has found wide application is flow-accelerating, which qualification indicates that its effect is an increase of the flow velocity through the propeller disk. This leads primarily to the possible increase of efficiency if compared with an open propeller. Application of the flow - decelerating type aims at the increase of pressure, associated with the decrease of velocity, which may lead to retardation of cavitation phenomena. Both types are schematized in Figure 1.

Only the flow - accelerating type of duct will be dealt with in this paper. In practice the ducted propeller has been found very effective especially when the propeller loading is relatively large. In Fig. 2 an indication of the power savings is given as a function of B_p which is a coefficient based on given speed of advance V_A in knots, delivered power P_D in horse powers and propeller revolutions per minute N according to:

$$B_p = NP_D^{1/2} V_A^{-5/2} = 33.07\ K_Q^{1/2}\ J^{-5/2} \qquad (1)$$

(B_p is not dimensionless!)

The savings are given in percentages of the shaft power P_S required for a conventional single propeller. Typical values of B_p for the propeller of various ships are:

twin screw ships	10-15
fast war ships	10-25
single screw cargo ships	15-35
tankers	35-70
towing vessels	80

The above reductions have been found for the conventional ducted propeller configurations, i.e. the propeller located at the mid-duct position. One of the big drawbacks of the configuration is the danger of erosion on the innerside of the duct, due to heavy implosion pressures of cavitation. In view of this problem the installation of the duct just forward of the propeller can be attractive. See Fig. 3.

Recent applications of such duct configurations have been successful and proved to be good energy savers, while other benefits as reduction of propeller cavitation and hull vibrations and improved manoeuvrability have been found.
In the model testing of such configurations it is found that fitting the duct to the hull during the resistance test, without propeller,leads to a reduction in most cases if compared with the bare hull resistance, i.e. without duct. Apparently the presence of the duct improves the flow along the afterbody in such a way that a reduction in hull drag is achieved which is larger than the additional duct drag. Another recent trend in the application of additional devices aiming at energy saving are socalled reaction fins or stators. Although the conceptual thought behind its application is different from the duct principle, viz. to compensate for the kinetic energy losses of the propeller slipstream, their effect in practice may come close to that of a duct in non-uniform flow ahead of the propeller. Both, reaction fins and ducts have to be designed as streamline shaped profiles with a high lift/drag ratio.
An example of a reaction fin is given in Fig. 4.

Additional fins can also be fitted to the rudder to improve its interaction with the propeller slipstream. A comparison of the energy savings to be expected for various arrangements is given in Table 1 which is taken from Ref. 1.

TABLE 1

Savings in required power in per cents of power with conventional single propeller. Energy saving propulsion arrangements:

Low RPM - large diameter	5-18
Skew or stern tunnel	1- 5
Propeller retrofit	2-10
Propeller with vane wheel	9-12
Reaction fins	4- 8
Additional thrusting fins	2- 5
Asymmetric gondola	5-10
Asymmetric aft body	5- 8
Ducted propeller	5-12
Duct ahead of propeller	5-10
Overlapping twin screw	5-16
Overlapping twin screw + LRPM	10-21→ ?

In this paper the hydrodynamic principles are reviewed and the possibilities for design considered.
Also the results of recent research in this field will be reviewed.

2. HYDRODYNAMIC DESCRIPTION

In the basic model for the application of a duct around a propeller, already an explanation for possible energy saving is comprised. For instance, Küchemann and Weber (Ref. 2) show that by using an accelerating duct a gain in thrust, over the propeller thrust, is achieved without a loss in efficiency, or the total thrust, equal to the thrust of the propeller only, is achieved at a higher efficiency.

These conclusions are reached by considering the axial forces in a uniform flow, in which the primary effect is an increase of the mass flow through the propeller disk. See Fig. 5.

When the ratio between propeller thrust and total thrust is τ (for positive duct thrust: $\tau \leq 1$) the ideal efficiency is given by:

$$\eta_i = 2/(1 + \sqrt{1 + \tau C_{th}}) \qquad (2)$$

where

$\eta_i = TV_A$

$C_{Th} = T/(\frac{1}{2} \rho V_A^2 \pi R^2)$

T = thrust

R = propeller radius

V_A = advance speed

This relation shows that for constant C_{Th} a decrease of τ leads to higher η_i.

In Ref. 2 also the effect of the axial position of the propeller in the duct has been studied. Based on simple representations, the duct by a cylinder and the propeller by a disk with uniform loading, flow calculation show that when the propeller is in a central position, the increase of mass flow by the duct is nearly twice the increase when the propeller is at the duct entry or exit. This suggests that a central position is the best choice to obtain the maximum effect of the duct.

Recent trends in application of ducts for propeller indicate, however, that configurations where the propeller is not in the central position, even outside the duct, can be beneficial as well. Such applications are in the behind ship condition, where other

effects than those present in the ideal, uniform flow occur.
A direct consequence of this more complex situation is, that the duct thrust is no longer simply related to the additional mass flow through the propeller disk, via momentum theory. Now the duct thrust has to be derived from the flow conditions at the duct itself. The duct around the propeller is an annular hydrofoil, on which the main hydrodynamic force is a lift force acting normally to the flow direction. In general, there will be always such a lift force, which is related to the circulation around the duct section and controlled by a Kutta condition at the trailing edge. In case of uniform flow, without propeller, the circulation can only be created by the duct shape, i.e. its deviation from a thin cylindrical tube form. The lift force will now be directed radially, there is no axial force.
A positive axial force on the duct, i.e. the thrust, has to be considered as a component of the lift force, which comes into being by a rotation of the lift force vector, which in turn is caused by any deviation in the flow from the axial direction. See Fig. 6. If this concept of the action of any duct section is adapted it is applicable to the action of reaction fins as well.

More recent studies based on momentum theory applied to ducted propellers are presented in Ref. 3.

From the foregoing observations it can be concluded that for a further understanding of the various devices, the following hydrodynamic aspects have to be considered:

- circulation around the duct or fin section,
- onset flow conditions,
- viscous effects on the duct,
- propeller and duct induction and interaction with shear flow.

2.1. Circulation around duct

For a profile placed in an arbitrary flow at not too large angle of incidence with the mean flow direction, the main force is related to the circulation around the profile. By definition this is an integral along a closed line of the velocity component tangential to this line.

A parallel relationship exist between the pressure distribution along this line and its integral: the lift force. The integral quantities, i.e. circulation and lift force, are controlled by the flow conditions at the trailing edge (Kutta condition). This means that the shape of the profile can be varied in a certain range, thus modifying the distributions of velocities and pressures, but not their integrated values.

The magnitude of the circulation can be influenced by modifying the flow at the trailing edge. An imported effect is achieved by the radially inward directed induced velocity component of the propeller. This component is maximum in the blade tip region. The Kutta condition at the trailing edge of the duct creates such a circulation that the radial velocity is compensated. Thus when the propeller is located at the exit of the duct, a maximum circulation and lifting force can be expected. This is confirmed by theoretical studies by Küchemann and Weber (Ref. 2), but it also explains why the effectiveness of the duct can be maintained when it is placed ahead of the propeller.

For a NSMB 19A duct the effect is demonstrated by a calculation of the pressure distribution in Fig. 7, which is taken from Ref. 4.

It can be seen that when the propeller is moved afterwards (higher x_p/c) the inside suction pressure is strongly increased in the exit region but hardly influenced in the entrance region. Consequently, in view of the duct geometry, the radial force will change considerably, while the axial force remains approximately constant.

2.2.Onset flow conditions

When the duct is placed far ahead of the propeller the onset flow conditions will primarily govern the effect of the duct. Improvement of the flow along the hull can be expected if the duct accelerates the flow, even without propeller. In that case the experience from model testing that the resistance of hull plus duct can be less than the bare hull resistance is understood. The magnitude of this difference will depend on the quality of the onset flow. If flow separation is suppressed by the duct the beneficial effect is large. A completely different case appears when the lifting force on the duct (without propeller) in the onset flow is directed outward and consequently the flow is decelerated. An increase of the hull plus duct resistance can be expected. For duct positions closer to the propeller, also the effect of propeller induced velocities becomes stronger. Nevertheless in many cases the optimisation of the duct shape should be primarily based on the onset flow conditions aiming at favourable flow acceleration. When the duct is ahead of the propeller the designer is free to deviate from the axisymmetric form. The effect of the propeller will be that the duct circulation is increased, while also the radial velocities will enlarge the thrust component of the lift force. Of course the limits for proper hydrodynamic action of the duct profile should not be exceeded.

2.3. Viscous effects on the duct

The viscous flow effects on the performance are twofold. Firstly there are the viscous drag forces on the duct. These forces directly reduce the effectiveness of the duct as they counteract the thrust forces. The main parameter of a propeller duct, which governs this reduction is the length/diameter ratio of the duct. From the extensive testing of systematic ducted propeller series (Ref. 5) it can be concluded that for a duct length larger than 0.5 through 1.0 diameter the duct drag overrules the efficiency improvement. For estimates of the drag in practical cases the

coefficients for wing sections, as a function of Reynolds number, are sufficiently accurate. The second effect of viscosity is the development of the boundary layer and the subsequent flow condition at the trailing edge (Kutta condition). Especially when the trailing edge is rather rounded, like the NSMB duct 37, this effect becomes important.
In Fig. 8 a schematic representation is given of the character of the boundary layer.

Paint tests and theoretical calculations, reported in Ref. 4, have been carried out to find a proper indication about the trailing edge condition and thus on the circulation control mechanism. This is a requirement to facilitate theoretical predictions of the duct performance. When one is able to perform inviscid pressure distribution calculations the best choice for the Kutta condition is the equality of the peaked pressure minima on the duct inside and outside close to the trailing edge. An improved condition can be obtained, however, when boundary layer calculations are possible. Then the equality of the pressure at the locations of separation has to be required. In Fig. 9 the improvement in theoretical results is shown by comparison with experimental values (Ref. 4).

2.4. Propeller and duct induction and interaction with shear flow

For many propulsion systems the loading is rather heavy, which means that the velocities induced by the loading are of the same order of magnitude as the onset flow. Therefore these velocities have to be taken into account in a refinement of the forementioned hydrodynamic aspects. For that purpose the induced velocities can be suitably described by the vortex systems which are created at the propeller and by a vorticity distribution over the duct corresponding to its circulation. The flow field around the propulsor behind the ship (total velocity field) is not just the sum of the onset flow (nominal wake field) and the induced

velocities. Actually a rather complicated interaction takes place for which a complete description is not yet available. It has become clear, however, that in the ship's wake a substantial part of the interaction is due to the transport of the vorticity in the onset flow due to the induced velocities. It is clear that this interaction is not present in uniform onset flow and consequently also is of vital importance for the understanding of differences between experimental results obtained in open water and behind ship with ducted propeller systems. For axisymmetric wakes, which can be considered as an approximation of the ship wake based on its radial distribution of circumferential means along circles around the propeller axis, the theoretical models are developed so far that an estimate of the interaction velocities can be made.
Falcao de Campos and Van Gent (Ref. 6), using the socalled large-shear small disturbance approximation, derived that the interaction can be described by a new vorticity distribution. This is simply related to the vorticity distribution of the nominal wake via the stream functions of the induction. For the present case of a ducted propeller this can be written as:

$$\gamma_i = \frac{1}{U} \frac{d}{dr} \left(\frac{\gamma}{r}\right) (\Psi_p + \Psi_d + \Psi_i) \qquad (3)$$

where

Ψ_p = streamfunction of the propeller induced flow
Ψ_d = streamfunction of the duct induced flow
Ψ_i = streamfunction of the interaction flow
γ = vorticity in the nominal wake
γ_i = vorticity of the interaction flow
U = velocity of the nominal wake
r = radial coordinate

A closed solution for the interaction flow can be obtained as from the continuity equation also a relation between γ and Ψ_i follows. Then by application of the expression of Biot and Savart the velocity of the interaction follows.
It is noteworthy that in eq. (3) the induction velocities do not appear, only the streamfunction. This means that the details of the

induced velocity distribution are not so important. The stream function is an integral value, which is not so sensitive for differences in distributions.
An application of the solution of eq. (3) has been given in Ref. 4, where also the presence of an axisymmetric afterbody is taken into account. Comparisons with experiments are given as well. Here some results of calculations of the interaction velocities will be presented for the propeller and duct separately placed in a shear flow. See also Ref. 7.
From the calculations in Refs. 6 and 7 it can be concluded that both interactions, viz. propeller induction resp. duct induction, with the shear of the non-uniform wake in general lead to a remarkable increase of the axial flow through the propeller disk. In some regions the interaction velocities are larger than the induced velocities. The radial components are most important with respect to the effect of the propeller on the duct. The radial velocities of the propeller alone in uniform flow, have a rather peaked maximum (directed towards the axis) in the vicinity of the blade tip. Outside this region, especially for radii slightly larger than the propeller radius, the radial velocities have a rather constant magnitude. In this region also the radial components of the interaction velocities are of the same magnitude as the radial components of the induced velocities.
Therefore it can be concluded that in shear flow the relative position of propeller and duct becomes less critical than in uniform flow. The radial velocities, which are important to create a axial component of the lift force, exist in a larger part of the flow field. To maximize the circulation around the duct, its trailing edge has to be placed in the region where the radial velocities have their (peaked) maximum.

3. DESIGN PROCEDURES

In the design of ducted propeller systems, three levels can be distinguished. The first one is the design based on the results of experimental investigation of systematic series. Secondly the conventional wake-adapted propeller design can be applied in combination with a duct of given shape. Finally a system design can be considered, in which the shape of the duct is designed as well, in regard of the onset flow.

3.1. Design based on systematic series

A widely known systematic series of ducted propeller systems are the NSMB-series. The accelerating duct type is covered by a series of four different propellers in duct no. 19A and a series of three other duct shapes, nos. 22, 24 and 37, with one of those propellers. A sketch and the dimensions of the geometries of duct no. 19A and duct no. 37 are given in Figs. 10 and 11. NSMB no. 19A has been developed on the base of extensive investigations. See Refs. 8 through 12. From the structural point of view a simple duct shape was chosen. It has a cylindrical part at the inner side in the mid-chord region, where usually the propeller is located. The trailing edge is not too sharp, which makes it less vulnerable while also the reverse action of the duct when stopping the ship will become better.

For towing vessels (especially push boats), the thrust which can be developed at bollard pull condition either with the propeller running ahead or astern is of utmost importance. For such cases the duct no. 37 has been developed with a relatively thick trailing edge. Oosterveld (Ref. 5) has reported the numerical test results for the systematic series covering the whole range of operating conditions of speed and propeller rotation rate for ahead and astern manoeuvring.

For the normal operation conditions of propulsion the numerical values of the test results are available in polynomial form,

especialy suited for comptuter aided design studies. Recently these data have been checked and re-analysed as in publications a number of errors was detected. In the Appendix the correct data are given. The most useful applications are the design problems of:

- finding the optimum diameter for given speed of advance, propeller rotation rate and power to be absorbed or required thrust,
- finding the optimum propeller rotation rate for given speed of advance, diameter and power to be absorbed or required thrust.

3.2. Design of propeller for given duct shape

This is at present the common approach to the problem. A review of the relevant literature is given here to show the theoretical background. (Ref. 13.)
Dickmann and Weissinger (Ref. 14) established the foundations of the theory of a ducted propeller considered as a propulsion unit. They assumed a propeller with infinite number of blades with constant load distribution and represented the duct by ring vortices on a cylinder with a length equal to the duct's length and a diameter representative of the duct's diameter. The propeller slipstream was described by a semi-infinite ring vortex cylinder extending from the duct's trailing edge. The strength of the slipstream vortex system was determined from the application of a dynamic boundary condition at the duct's trailing edge. The ring vortex distribution on the duct was assumed and the duct's shape was obtained as a result.

Morgan (Ref. 15) and Ordway et al (Ref. 16) considered a propeller with a finite number of blades represented by classical lifting line theory and used linearized theory for the duct. The theory includes the treatment of finite blade interference effects between duct and propeller. As shown by Hough and Ordway (Ref. 17) the steady part of the induced velocity field due to a finite bladed propeller, as obtained from lightly loaded lifting line theory, is

identical to the axisymmetric velocity field of an infinitely bladed propeller as given by classical actuator disk theory. Since it introduces considerable simplification in the required calculations, this procedure is often used when determining the steady load on the duct.

Extension of these theories to include effects of heavy loading and a non-linearized duct description was initiated by Chaplin (Ref. 18) who used ring vortex distributions located on the duct camber line and on the slipstream boundary tube springing from the duct's trailing edge.

Non-linearized theories including the effects of duct thickness, suitable for the representation of the thick ducts occurring in marine applications, have been developed based on surface singularity methods. (Refs. 19, 20). For a detailed account of the linearized theories and Chaplin's non-linear theory we refer to the review of Weissinger and Maass (Ref. 21).

Several investigations have shown that the above mentioned theories give, in general, fairly satisfactory predictions in uniform axial flow, of the thrust acting on the duct, duct pressure distribution and thrust ratio between propeller and duct provided that no significant separation occurs on the duct.

Most of the proposed design methods deal with the direct-inverse problem or the design of a propeller inside a duct of a given shape (Refs. 24, 25, 19).
Other methods consider the inverse-inverse problem and determine simultaneously the duct shape when designing the ducted propeller. In the design method (Ref. 28), the shape of the duct mean-line is determined from the continuity law for given duct vortex distribution, and an assumed thickness distribution represented by ring sources is superimposed. A similar method is followed by Oosterveld (Ref. 26), while Morgan and Voigt (Ref. 27) depart from

a given pressure distribution and determine the vortex and source distributions satisfying the linearized boundary conditions on the duct and the Kutta condition at the duct's trailing edge. From the strength of these singularities the camber, thickness and angle of attack of the duct profile are calculated.

Common to the various design methods is the representation of the propeller by conventional propeller lifting line theory. When accounting for the interaction between propeller and duct the assumption of an infinite number of blades is almost exclusively used. With this assumption the helicoidal vortex sheets trailing from the propeller blades degenerate into an orthogonal system of ring vortices and horse-shoe rectilinear vortices.Different methods have been used for the computation of the axial and radial velocities induced by the ring vortex system, namely: numercial integration (Refs. 23, 25), closed form expressions as derived by Ordway, (Ref. 17), in terms of Legendre functions or Gibson's, (Ref. 20), expressions in terms of elliptic integrals.

When designing the propeller with Lerbs induction factor method (Refs. 28, 23, 24, 25, 29), the choice of the hydrodynamic pitch is one of the basic design assumptions. It is taken in the ultimate wake by Dyne (Ref. 23), at the propeller plane in Caster's method or at an intermediate prescribed value by Minsaas (Ref. 25). The lifting line calculations are supplemented by camber correction factors from lifting surface effect, either derived for representative open propellers, (Ref. 30), in the design method, (Ref. 29), or by other methods (Ref. 21), with additional corrections to pitch and camber due to the duct and hub curved induced flow.

The influence of finite blade number interference effects between propeller and duct on the hydrodynamic pitch is also accounted for, in an approximate way, by Minsaas (Ref. 25), by mirrorring the propeller helical vortex sheets on the cylinder prolonging the inner surface of the duct.

This so-called "wall effect" is seen to be important near the tip where it decreases the hydrodynamic pitch. Minsaas concludes that this effect is partially cancelled by the velocities induced by the vortices shed from the blade tip if the propeller is designed with finite blade circulation at the tip.

As an example the design method developed at NSMB is given. The various steps in this procedure have recently been checked. (Refs. 4 and 13).

In the design method the following assumptions haven been considered:

- An axisymmetric duct and hub configuration is represented by a surface vorticity model on the assumption of inviscid and incompressible irrotational flow (Ref. 4)
- When accounting for the interaction between propeller and duct the propeller is represented by an actuator disk with a radial distribution of circulation identical to the finite bladed propeller.
- The strength of the ring vortices shed from the actuator disk is determined by using the hydrodynamic pitch in the ultimate slipstream under the assumptions of moderately loaded propeller theory. As shown by Dyne (Ref. 23) this procedure implies the agreement for the propeller thrust between general momentum theory and the Kutta-Joukowski law.
- The duct inviscid thrust is calculated by integration of the pressure distribution and the propeller inviscid thrust from Kutta-Joukowski law.
- The propeller is designed by Lerbs induction factor method, (Ref. 28), according to the current practise at the NSMB (Ref. 31). In the propeller lifting line calculations the pitch of the vortex lines is determined at the propeller plane under the influence of the average duct induced velocities. A tip clearance is assumed and the propeller blade circulation is considered to vanish at the hub and the tip.
- Finite blade propeller-duct interference effects are neglected.
- Viscous effects are accounted for by calculating the viscous drag on the duct and specifying the blade sections drag coefficient.

- Corrections to camber from lifting surface theory may be applied.

A flow chart of the design procedure for given total thrust (or power) and a desired hydrodynamic pitch distribution is given in Fig. 12.

3.3. Design of ducted propeller systems

In comparison with a conventional propeller, the performance of a ducted propeller behind a ship may substantially differ from its performance in uniform flow. (Ref. 32).

It has already been suggested in chapter 2 that, when evaluating ducted propeller performance behind a ship, attention should be paid to the influence of the radial wake on the inflow to the duct, which may cause a thrust distribution between propeller and duct different from the one obtained in uniform flow.
Similarly, the effect of the interaction between the wake vorticity and the duct and propeller vortex systems is likely to have a considerable influence on the ducted propeller performance.
See also Ref. 33.

However, the investigation on these aspects of the complex ducted-propeller hull interaction have not been, sofar, carried out to such an extent to enable systematic improvement of the well established design procedures based on systematic experimental investigations and hydrodynamic theories in uniform flow as described in foregoing sections.

Additional options in the design of modern ducted propeller systems are:
- flexibility in the mutual position of propeller and duct,
- optimisation of the duct shape in the onset flow to achieve highest efficiency.

With respect to the first option it can be stated that the design procedure for the propeller, as described in the foregoing section, can be adapted. There is no information available based on systematic variation on the mutual position, however. In Ref. 34 a number of configurations, which have been investigated are reported. Behind a ship model, inclination angle, diameter and longitudinal position of the ducted propeller have been varied, which direct interfere with the hull. Problems related to the relatively small tip clearance of the propeller inside the duct, cavitation and consequently erosion, becomes less important.
The consequence of the second option is that more detailed information about the flow is required. This means that in the design process model testing and especially flow measurement become more involved. Furthermore non-axisymmetric duct shapes have to be considered, for which a design procedure has not yet been established.
A number of interesting studies are collected in Ref. 35.

Some progress has been made at NSMB with respect to computer aided design of duct shapes. A design method for airfoils, developed at the National Aerospace Laboratory NLR in the Netherlands (Ref. 36) has been modified for application on annular foils as well (Ref. 37.). The attractive feature of the method is that the geometry can be prescribed partly as well as the desired pressure distribution. The relative importance of both type of requirements can be adjusted by weight factors along the contour. Thus in one region the geometry can be kept unchanged, while at other locations a change in geometry is designed to satisfy optimisation requirements.
The method has been tested in uniform flow; its flexibel formulation facilitates, however, easy inclusion of propeller induced velocities and velocities due to other disturbances.

4. MODEL TESTING

4.1. Flow measurements

Although the modern means for flow measurement, like Laser-Doppler techniques, have become in use for application on board of ships, model scale still is best suited for this purpose.
The standard testing techniques for resistance and propulsion are useful for mutual comparison of configurations and final judgement of a design. Especially the design requirement of energy saving asks for a high accuracy of these measurements, in order to find the often marginal improvements.
Detailed flow measurements, not only in the propeller plane (nominal wake) but also further upstream or downstream, supply additional information. This information can be used either in the design, to adapt the local geometry of the propulsor to the local flow, or after testing a design to understand its performance and find out ways of further improvement. The Laser-Doppler velocimetry has been developed sufficiently, to enable it use for advanced designing.

4.2. Extrapolation of model test results to ship scale

The modern trend in propeller/duct configurations has raised the attention for the problem of extrapolation of model test data to the real ship situation. Two controversial opinions can arise when too simplified approaches are pursued. The problem is described suitably when the performance prediction of the ITTC-78 for ships with a conventional propulsor is taken as a base. In this method the duct has to be introduced as an element which in general interacts with the propeller and with the hull. Now in model testing the duct can be considered as a part of the propulsion unit, which approach implies open water tests with the ducted propeller and resistance test with the hull excluding the duct and propeller. But the alternative apprach is to consider the duct as an appendage to the hull in the resistance test and perform open water tests with the propeller alone.

It is clear that both approaches have their shortcomings, due to the neglection of interactions in the separate tests.
Which of them dominates will depend on the configuration, but this cannot be foreseen in general. Therefore it is attractive to have a method available in which the interactions are always taken into account. Such an extrapolation method has been investigated and compared with the others (Ref. 38). Model testing comprises open water test with ducted propellers and resistance tests with the duct installed at the hull. Now in order to treat the propeller as an open propeller, the appropriate wake fraction has to be found. This requires an estimate of the duct induced velocities at the propeller location. In this investigation theoretical models have been used to calculate this induction either by momentum theory or by the duct vortex system. It can be concluded that the alternative method based on "dual interaction" in model testing is satisfactory in natural cases.
A more rigourous point of view is taken, when the base of separate open water test for the propeller and resistance test for the hull is discarded. Instead only a propulsion/overload test is used in which the propulsion properties at one propeller loading at several speeds and several propeller loadings at one speed are measured (Ref. 39). By extrapolation to zero propeller thrust the required data for resistance and for average propeller inflow velocity are derived. Furthermore the scale effect on the duct resistance has been taken into account.
A first correlation with available full scale data shows a considerable improvement if compared with the more traditional approach. More correlation cases would be welcome.

5. FINAL CONSIDERATIONS

A propeller with duct is a propulsion system, which from a hydrodynamic point of view can be considered as a combination of an active and a passive part. The active part is the rotative propeller: its motion is essential to create in the flow through the propeller system an increase of momentum necessary to achieve a propulsive force on the system as a whole. The active part or impeller cannot be missed, contrary to the passive parts, which are not absolutely required. Passive parts alone cannot create a propulsive force or absorb power to do so. By hydrodynamic interference between active and passive parts, however, the passive parts can carry a part of the propulsive force and thus take over a part of the loading from the impeller. The design requirements for the impeller are lightened in this way. The basic principles of the interaction between active and passive parts are well understood (chapter 2).

Whether a system of rotator and stator will satisfy the requirement of minimum power absorbtion depends on several circumstances. In the practical case of ship propulsion the system operates in a non-uniform flow in the vicinity of another large body, the ship. Moreover, viscous effects, insofar not considered in the basic principles, often reduce the favourable effects. Nevertheless applications in shipbuilding have proved, that energy saving is possible, when the single propeller is replaced by a system comprising static lifting devices as well (chapter 1). There is no proper indication that for any ship no more than the minimum amount of energy is spent to propel it.

When designing a system of propeller and duct use can be made of the tools developed, experimentally by systematic series and theoretically by hydrodynamic models. These models can be used for cases of ducts ahead of the propeller as well. In their present state of development they are less suited to deal with the non-uniformity in the ship's wake and with a further optimisation of the duct shape, profiles and non-axisymmetric rings.

Further development of design tools based on hydrodynamic theory should take into account the non-axisymmetric structure of the wake. Thus the design of more general stators or reaction fins can be covered as well (chapter 3).
In addition to the common resistance and propulsion investigations at model scale, detailed experimental information on flow and loading distribution is required (chapter 4). Instrumentation is available to do so. Extrapolation of results on models scale to the ship can be done by means of improved methods if modern measurement techniques are used to collect data of flow and loading on the ship. Combination of advanced design tools and detailed experimental information will give a key to further energy saving.

References:

1. Muntjewerf, J.J.,
Propulsion research by model testing (1983).

2. Küchemann, D. and J. Weber,
Aerodynamics of propulsion
McGraw - Hill Book Company, Inc., 1953

3. Van Gunsteren, L.A.,
A contribution to the solution of some specific ship propulsion problems.
Thesis 1973, Techn. Univ. Delft.

4. Falcao de Campos, J.A.C.,
On the calculation of ducted propeller performance in axisymmetric flows.
NSMB Publication No. 696 (1983)

5. Oosterveld, M.W.C.,
Wake adapted ducted propellers.
NSMB Publication No. 345

6. Falcao de Campos, J.A.C., and W. van Gent
Effective wake of an open propeller in axsymmetric shear flow.
NSMB Report No. 50030-3-SR (1981)

7. Van der Vegt, J.J.W.,
Actuator disk in a two-dimensional non-uniform flow.
NSMB Report No. 50236-1-RD (1982)
To be published in International Shipbuilding Progress.

8. Manen, J.D. van: "Open-water test series with propellers in nozzles", ISP, Vol. 1, No. 3, 1954.

9. Manen, J.D. van: "Recent research on propellers in nozzles", I.S.R. Vol. 1, No. 2, 1957.

10. Manen, J.D., Superina, A.: "The design of screw-propellers in Nozzles". ISP, Vol. 6, No. 55, 1959.

11. Manen, J.D.: "Effect of radial load distribution on the performance of shrouded propellers", ISP, Vol. 9, No. 93, 1962.

12. Manen, J.D. van, Oosterveld, M.W.C.: "Analysis of ducted propeller design", Trans. SNAME, Vol. 74, 1966.

13. Falcao de Campos, J.A.C.
Ducted propeller design method based on lifting line model for moderately loaded propellers and on surface vorticity model for the duct.
NSMB Report No. 50030-6-SR (1981)

14. Dickmann, H.E., Weissinger, J.: "Beitrag zur Theorie optimaler Dusenschrauben (Kortdüsen)", Jahrbuch STG, Vol. 49, 1955.

15. Morgan, W.B.: "Theory of the annular airfoil and ducted propeller", Fourth Symposium on Naval Hydrodynamics, Office of Naval Research ACR-73, Vol. 1, 1962.

16. Ordway, D.E., Sluyter, M.M., Sonnerup, B.O.V.: "Three-dimensional theory of ducted propellers", Therm. Advanced Research, Ithaca/New York, TAR-TR.602, 1960.

17. Hough, G.R., Ordway, D.E.: "The generalized actuator disk", Developments in Theoretical and Applied Mechanics, Vol. II, 1965.

18. Chaplin, H.R.: "A method for numerical calculation of slipstream contraction of a shrouded impulse disk in the static case with application to other axisymmetric potential flow problems", DTMB Report 1857, 1964.

19. Ryan, P.G., Glover, E.J.: "A ducted propeller design method: A new approach using surface vorticity distribution techniques and lifting line theory", Trans. RINA, Vol. 144, 1972.

20. Gibson, I.S., Lewis, R.I.: "Ducted propeller analysis by surface vorticity and actuator disk theory", RINA Symposium on Ducted Propellers, 1973.

21. Weissinger, J., Maass, D.: "Theory of the ducted propeller. A Review", 7th Symposium on Naval Hydrodynamics, 1968.

22. Morgan, W.B., Caster, E.B.: "Comparison of theory and experiment on ducted propellers", 7th Symposium on Naval Hydrodynamics, 1968.

23. Dyne, G.: "A method for the design of ducted propellers in a uniform flow", SSPA Publ. No. 62, 1967.

24. Caster, E.B.: A computer program for use in designing ducted propellers", NSRDC Rept. 2507, 1967.

25. Minsaas, K.J.: "Ducted propeller-design theory", NSFI Rept. No. R-77.78, 1978.

26. Oosterveld, M.W.C.: "Series of model tests of ducted propellers", NSP Report No. 67-163-AH on Contract No. N62558-4555, 1967.

27. Morgan, W.B., Voigt, R.B.: "The inverse problem of the annular airfoil and ducted propeller", DTMB Report 2251, 1966.

28. Lerbs, H.W.: "Moderately loaded propellers with a finite number of blades and an arbitrary distribution of circulation", Trans. SNAME, Vol. 60, 1952.

29. Fitzsimmons, P.A.: "Propeller research at BSRA", Symposium on Advances in Propeller Research and Design, GDANSK, 1981, pp. 195, 238.

30. Morgan, W.B., Silovic, V. and Denny, S.B.: "Propeller lifting surface corrections", Trans. SNAME, Vol. 76, 1968.

31. Gent, W. van, Oossanen, P. van: "Influence of wake on propeller loading and cavitation", 2nd Lips Propeller Symposium, 1973.

32. Dyne, G.: "Scale effect experiments with ducted propellers", PRADS Symposium, Tokyo 1972.

33. Okamoto, H. and Kozawa, K.: "On the hydrodynamic characteristics of ducted propeller in uniform and non-uniform flow", Symposium on Hydrodynamics of Ship and Offshore Propulsion Systems, HØVIK outside Oslo, March 1977.

34. Oosterveld, M.W.C. and Van den Berg, W.
Research in a depressurized towing tank on ducted propeller-hull interaction.
Int. Shipb. Progress, Vol. 23, 1976,

35. Proceedings RINA Symposium on ducted propellers, Teddington 1973.

36. Labrujère, Th. E.,
Airfoil design by the method of singularities via parametric optimisation of a geometrically constrained least squares object function.
NLR Report TR 76139 U

37. Wegereef, J.W.,
Design method for an axisymmetric duct in a uniform flow using surface-vorticity, NSMB Report No. 50030-2-SR (in Dutch), 1981.

38. Stierman, E.J.
Extrapolation methods for ships fitted with a ducted propeller.
To be published in International Shipbuilding Progress.

39. Holtrop, J.,
Model-ship correlation for a ship with a ducted propeller.
MARIN Report No. 50306-2-VT, Feb. 1983.

Appendix

NSMB polynomials for propellers in nozzles.

Accelerating nozzles nos. 19A, 22, 24, 37
Propeller series Ka 3-65,
Ka 4-55, Ka 4-70,
Ka 5-75.

Definitions:

Total thrust coefficient $K_T = \sum_{x,y} C_T(x,y)\,(P/D)^x (J)^y$

Torque coefficient $K_Q = \sum_{x,y} C_Q(x,y)\,(P/D)^x\,(J)^y$

Nozzle thrust coefficient $K_{TN} = \sum_{x,y} C_{TN}(x,y)\,(P/D)^x\,(J)^y$

PROPELLER	Ka 3-65		NOZZLE 19A	
x	y	C_T	C_{TN}	C_Q
0	0	.028100	.154000	.006260
	1	-.143910	.115560	0.0
	2	0.0	-.123761	-.017942
	3	-.383783	0.0	0.0
	4	0.0	0.0	-.008089
	5	0.0	-.741240	0.0
	6	0.0	.646894	0.0
1	0	0.0	-.542674	0.0
	1	-.429709	-.749643	0.0
	2	0.0	0.0	-.016644
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	-.162202	0.0
2	0	.671268	.972388	0.0
	1	0.0	0.0	0.0
	2	.286926	1.468570	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	-.182294	-.317644	.040041
	1	0.0	0.0	0.0
	2	0.0	-1.084980	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	-.032298	0.0
4	0	0.0	0.0	0.0
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	.199637	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	0.0
	1	0.0	.060168	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	0.0	0.0	-.003460
	1	-.017378	0.0	-.000674
	2	0.0	0.0	.001721
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
0	7	0.0	0.0	0.0

PROPELLER	Ka 4-55		NOZZLE 19A	
x	y	C_T	C_{TN}	C_Q
0	0	-.375000	-.045100	-.034700
	1	-.203050	0.0	.018568
	2	.830306	0.0	0.0
	3	-2.746930	-.663741	0.0
	4	0.0	-.244626	-.195582
	5	0.0	0.0	.317452
	6	.067548	0.0	-.093739
1	0	2.030070	.244461	.158951
	1	-.392301	-.578464	-.048433
	2	-.611743	1.116820	0.0
	3	4.319840	.751953	.024157
	4	-.341290	0.0	0.0
	5	0.0	0.0	-.123376
	6	0.0	-.089165	0.0
2	0	-3.031670	0.0	-.212253
	1	0.0	-.146178	0.0
	2	0.0	-.917516	0.0
	3	-2.007860	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	2.836970	.068186	.156133
	1	0.0	.174041	0.0
	2	0.0	.102331	0.0
	3	.391304	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
4	0	-.994962	0.0	0.0
	1	0.0	0.0	.030740
	2	0.0	0.0	.073587
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	-.031826
	1	.015742	0.0	-.014568
	2	0.0	0.0	-.109363
	3	0.0	0.0	0.0
	4	0.0	0.0	.043862
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	.043782	-.008581	.007947
	1	0.0	0.0	0.0
	2	0.0	0.0	.038275
	3	0.0	0.0	0.0
	4	0.0	0.0	-.021971
	5	0.0	0.0	0.0
	6	0.0	0.0	.000700
0	7	0.0	.088319	.022850

PROPELLER	Ka 4-70		NOZZLE 19A	
x	y	C_T	C_{TN}	C_Q
0	0	.030550	.076594	.006735
	1	-.148687	.075223	0.0
	2	0.0	-.061881	-.016306
	3	-.391137	-.138094	0.0
	4	0.0	0.0	-.007244
	5	0.0	-.370620	0.0
	6	0.0	.323447	0.0
1	0	0.0	-.271337	0.0
	1	-.432612	-.687921	0.0
	2	0.0	.225189	-.024012
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	-.081101	0.0
2	0	.667657	.666028	0.0
	1	0.0	0.0	0.0
	2	.285076	.734285	.005193
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	-.172529	-.202467	.046605
	1	0.0	0.0	0.0
	2	0.0	-.542490	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	-.016149	0.0
4	0	0.0	0.0	-.007366
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	.099819	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	0.0
	1	0.0	.030084	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	0.0	0.0	-.001730
	1	-.017293	0.0	-.000337
	2	0.0	-.001876	.000861
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
0	7	0.0	0.0	0.0

PROPELLER	Ka 5-75			NOZZLE 19A
x	y	C_T	C_{TN}	C_Q
0	0	.033000	-.000813	.007210
	1	-.153463	.034885	0.0
	2	0.0	0.0	-.014670
	3	-.398491	-.276187	0.0
	4	0.0	0.0	-.006398
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
1	0	0.0	0.0	0.0
	1	-.435515	-.626198	0.0
	2	0.0	.450379	-.031380
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
2	0	.664045	.359718	0.0
	1	0.0	0.0	0.0
	2	.283225	0.0	.010386
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	-.162764	-.087289	.053169
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
4	0	0.0	0.0	-.014731
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	0.0
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	0.0	0.0	0.0
	1	-.017208	0.0	0.0
	2	0.0	-.003751	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
0	7	0.0	0.0	0.0

PROPELLER	Ka 4-70		NOZZLE 22	
x	y	C_T	C_{TN}	C_Q
0	0	.008043	.001317	.032079
	1	0.0	0.0	0.0
	2	-.208843	0.0	-.020219
	3	-.902650	0.0	-.021294
	4	0.0	-.937036	0.0
	5	.369317	0.0	0.0
	6	0.0	.682898	0.0
1	0	0.0	0.0	-.102805
	1	-.661804	-.559885	0.0
	2	.752246	0.0	0.0
	3	0.0	.951865	0.0
	4	0.0	-.376616	0.0
	5	-.159272	0.0	0.0
	6	0.0	0.0	0.0
2	0	.720632	.371000	.140281
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	-.202075	-.096038	-.026416
	1	0.0	.011043	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
4	0	0.0	0.0	0.0
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	0.0
	1	0.0	0.0	-.008516
	2	0.0	0.0	0.0
	3	0.0	-.093449	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	0.0	0.0	0.0
	1	0.0	0.0	.005229
	2	0.0	0.0	0.0
	3	0.0	.045373	0.0
	4	0.0	0.0	-.000195
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
0	7	0.0	-.244550	.001334

PROPELLER	Ka 4-70		NOZZLE 24	
x	y	C_T	C_{TN}	C_Q
0	0	-.056541	-.026195	.023557
	1	.001197	0.0	0.0
	2	0.0	0.0	-.016989
	3	-1.234240	-.838832	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	.555129	.082386
1	0	.235791	.109624	-.072021
	1	-.705042	-.681638	0.0
	2	0.0	.773230	0.0
	3	-.159436	0.0	-.037596
	4	0.0	0.0	-.034871
	5	.908131	0.0	0.0
	6	-.840424	0.0	0.0
2	0	.454377	.259217	.103364
	1	0.0	0.0	0.0
	2	1.254499	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	-.131615	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	-.110537	-.058287	-.013447
	1	0.0	0.0	0.0
	2	-.392827	0.0	0.0
	3	-.205174	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	.013180
4	0	0.0	0.0	0.0
	1	0.0	0.0	0.0
	2	0.0	0.0	-.012173
	3	0.0	0.0	.046464
	4	0.0	0.0	-.035041
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	0.0
	1	-.022669	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	-.044629	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	0.0	0.0	0.0
	1	.011161	0.0	0.0
	2	0.0	0.0	0.0
	3	.023147	.026228	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
0	7	.452712	-.239044	-.049039

PROPELLER	Ka 4-70		NOZZLE 37	
x	y	C_T	C_{TN}	C_Q
0	0	-.162557	-.016806	.016729
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	-.077387	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	-.099544	.030559
1	0	.598107	0.0	-.048424
	1	-1.009030	-.548253	-.011118
	2	0.0	.230675	-.056199
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
2	0	.085087	.460206	.084376
	1	.425585	0.0	0.0
	2	0.0	0.0	.045637
	3	0.0	0.0	-.042003
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
3	0	0.0	-.215246	-.008652
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	-.021044	0.0	0.0
	6	0.0	0.0	0.0
4	0	0.0	.042997	0.0
	1	0.0	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
5	0	0.0	0.0	0.0
	1	-.038383	0.0	0.0
	2	0.0	0.0	0.0
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
6	0	0.0	0.0	0.0
	1	0.0	0.0	-.001176
	2	.014992	0.0	.002441
	3	0.0	0.0	0.0
	4	0.0	0.0	0.0
	5	0.0	0.0	0.0
	6	0.0	0.0	0.0
0	7	0.036998	.051753	-.012160

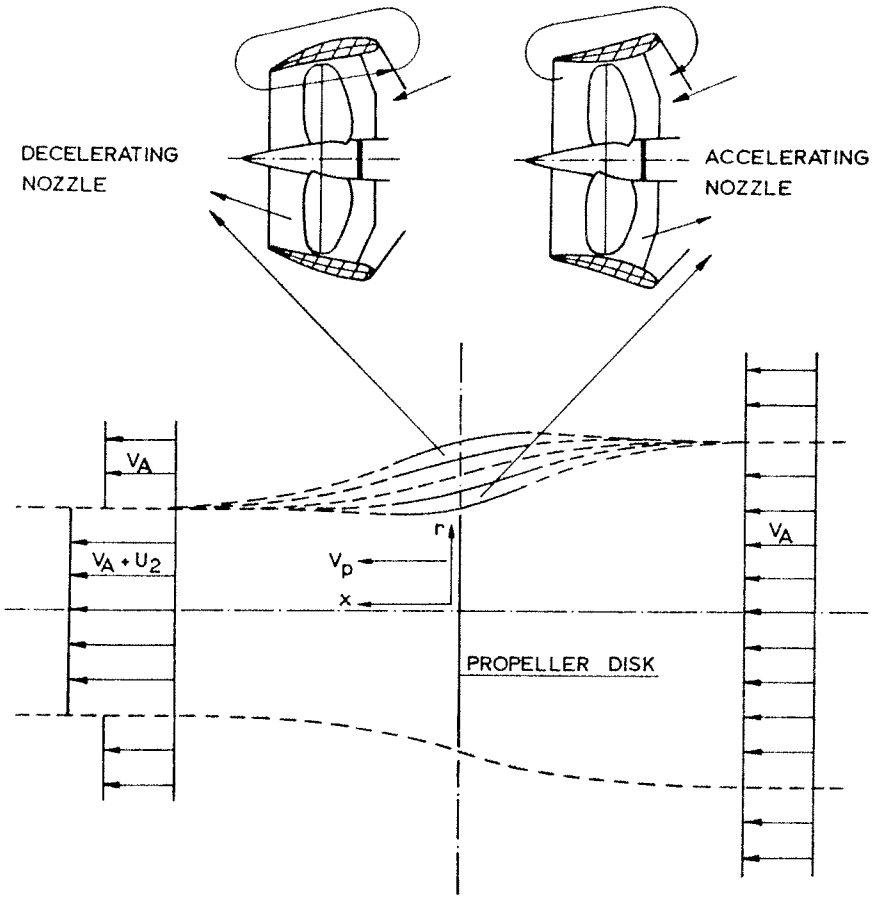


FIG. 1. DUCT ACCELERATING AND DECELERATING FLOW TYPES

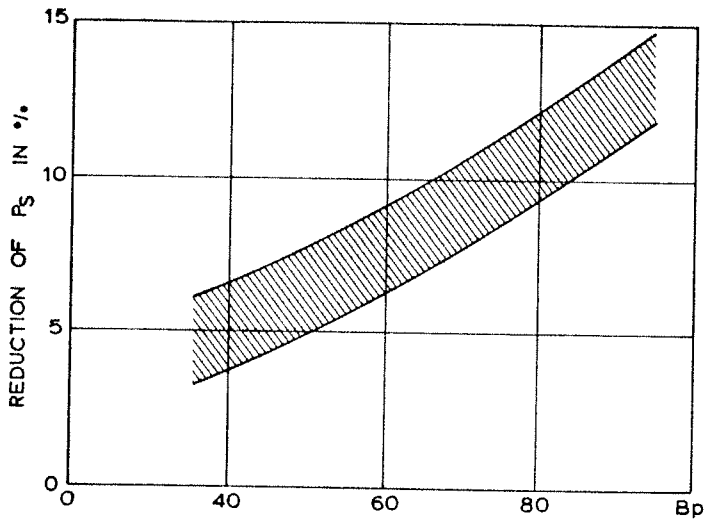


FIG. 2. REDUCTION OF SHAFT POWER DUE TO APPLICATION OF A DUCT AROUND THE PROPELLER

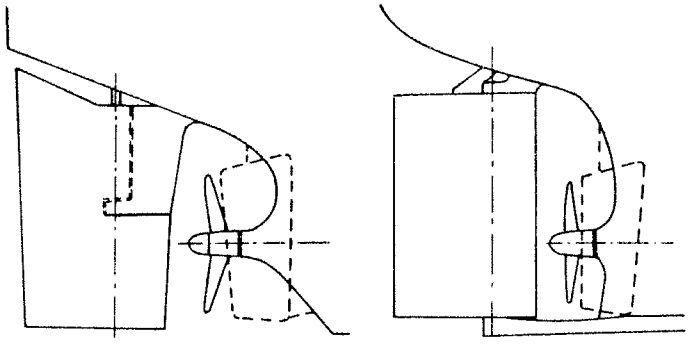


FIG. 3. DUCTS AHEAD OF THE PROPELLER

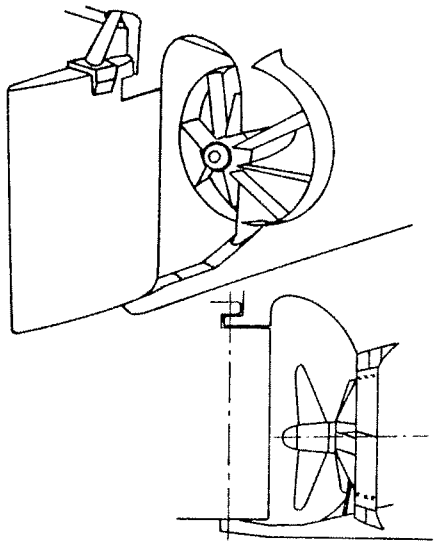


FIG. 4. REACTION FIN

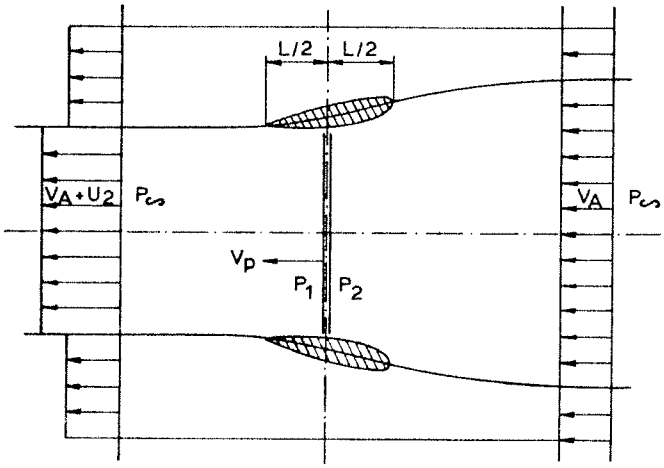


FIG. 5. MOMENTUM SCHEDULE FOR DUCTED PROPELLER

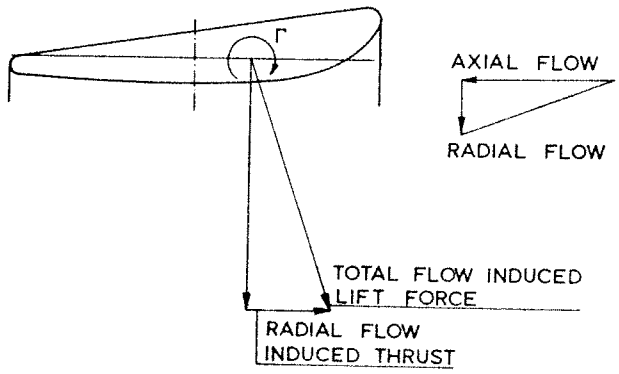


FIG. 6. HYDRODYNAMIC FORCE SCHEDULE
FOR DUCT SECTION

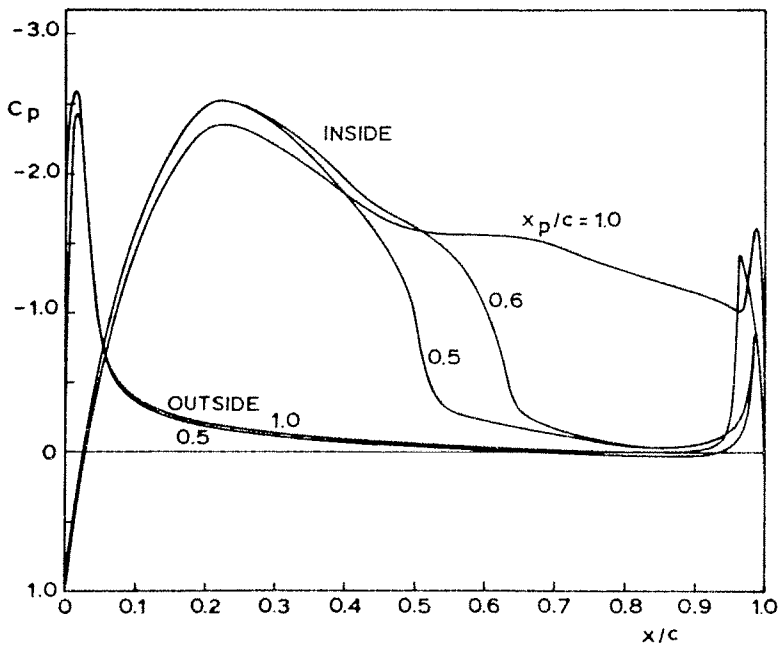


FIG. 7. EFFECT OF LOCATION OF PROPELLER PLANE ON THE PRESSURE DISTRIBUTION ON DUCT NSMB 19A.
$J = 0.50$, $C_{T_p} = 1.245$

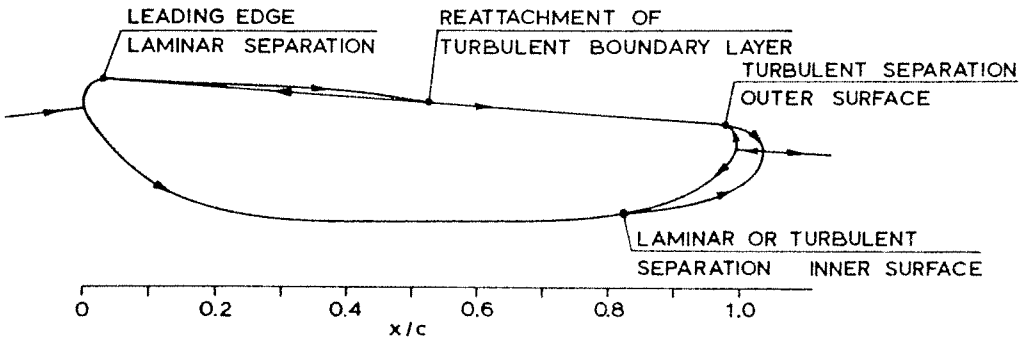


FIG. 8. SCHEMATIC REPRESENTATION OF THE BOUNDARY LAYER CHARACTER ON DUCT 37 IN UNIFORM FLOW REYNOLDS NUMBER $Re_c = 1.0 \times 10^5$ TO 4.0×10^5

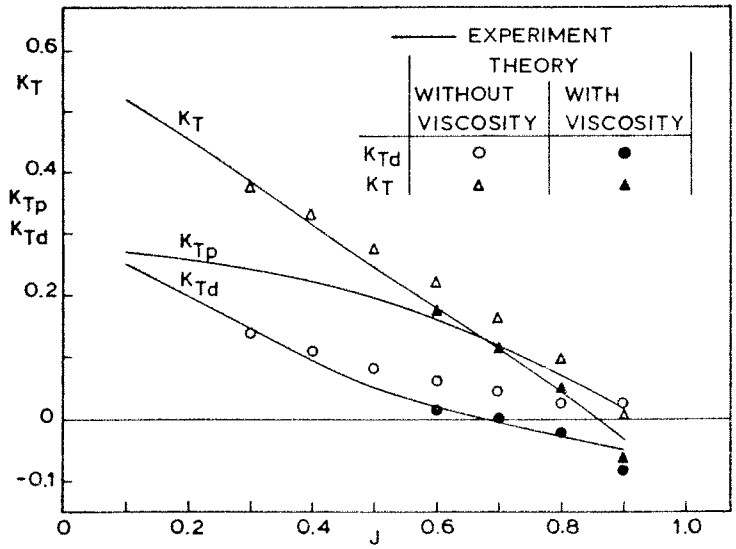
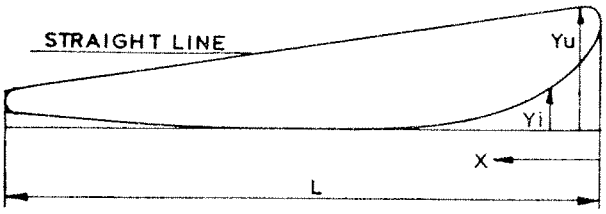


FIG. 9. CORRELATION FOR THE INVISCID AND VISCOUS CALCULATIONS OF THE THRUST ACTING ON THE DUCT 37

DUCT No. 19A PROFILE

L/D = 0.5



PERCENTAGES OF LENGTH		
100X/L	100Yi/L	100Yu/L
0	18.25	18.25
1.25	14.66	20.72
2.5	12.80	21.07
5	10.07	20.80
7.5	8.00	STRAIGHT LINE
10	6.34	
15	3.87	
20	2.17	
25	1.10	
30	0.48	
40	STRAIGHT LINE	
50		
60		
70	0.29	
80	0.82	
90	1.45	
95	1.86	
100	2.36	± 6.36

FIG. 10. PROFILE COORDINATES OF DUCT No. 19A

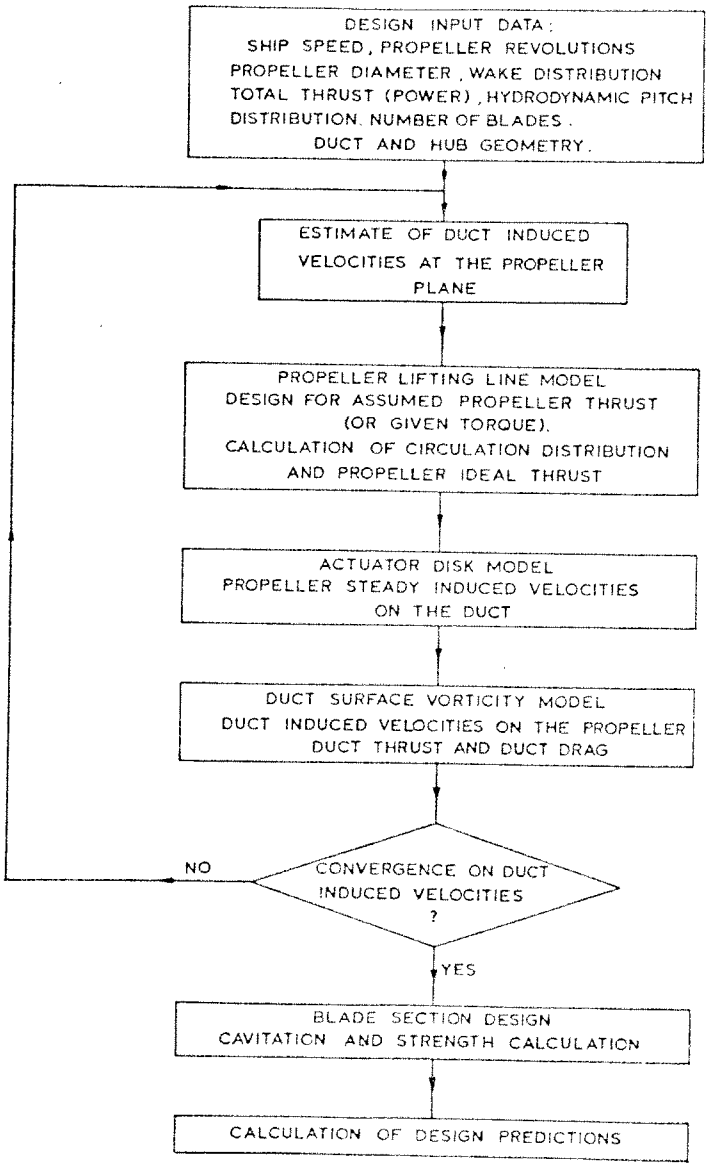


FIG. 12 FLOW CHART OF DUCTED PROPELLER DESIGN PROCEDURE.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

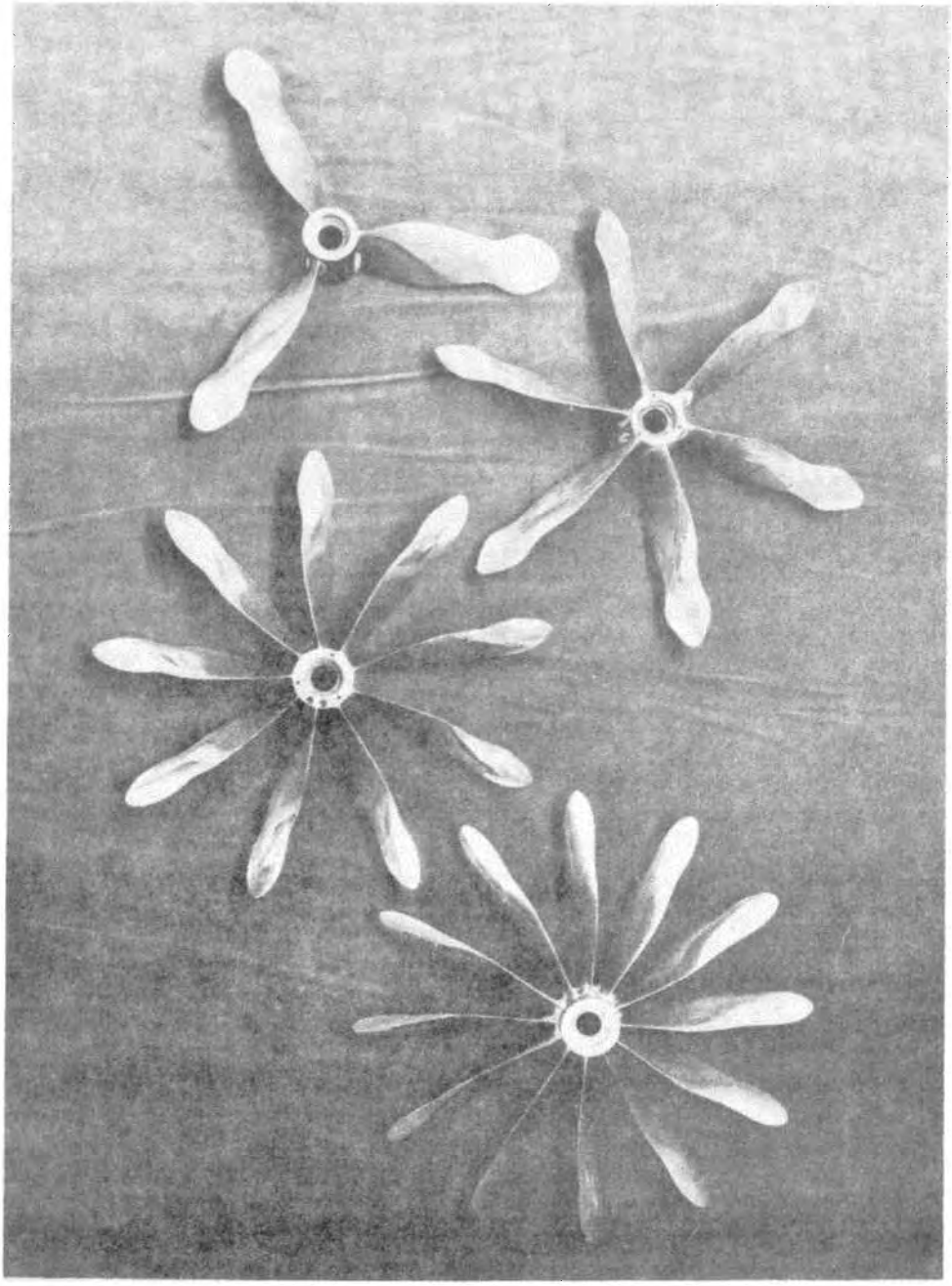
Propeller and Vane Wheel, Contrarotating Propeller.

by

Otto Grim

Prof.emer. Dr.Ing. Dr.Ing.E.h.

Universität Hamburg, Hamburg



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1. Introduction

In 1973 the oil price started to rise to an unexpected level: the energy crisis developed.
The resulting profit losses of the ship owners gave rise to discussions and actions which purpose it had been and still is to save energy also on ships. These discussions and actions also include the propulsion unit, the propeller. Units of a new type were suggested; it was also the beginning of the tendency to favour the big slow-running propeller of which it was well known what one would have to expect. How far this tendency is going is shown in a publication by Kawasaki Industries Limited, entitled: "Super Energy Saving Bulk Carrier". A propeller with adjustable blades is described, which has a diameter of 11 m and a number of revolutions of 43 min^{-1}. The consequences resulting from a so extreme low number of revolutions on the driving shaft, the bearings, gears etc. are obvious.

The new-type propulsion unit - propeller and vane wheel (PVW)and subject of this lecture - is picking up a large quantity of water like the big slow-running, conventional propeller (BSLP) and is granting an efficiency at least as good as with the BSLP, but it can be operated with a higher number of revolutions. In 1966 this new device appeared in public of the first time [1]. There was, however, very little interest, especially since at that time the oil price was very low. In the meantime the unit is installed on board the German research ship "GAUSS". During the - up to now - three years operation it has been quite a success and the safety in operation has been proved.

2. Idea and Main Characteristics

The idea is to install behind the propeller a vane wheel with a diameter larger than that of the propeller, Fig.1. This wheel is not mechanically driven but freely rotating. Its function is to take power from the slipstream which otherwise would get lost and to transform it into additional thrust power. The layout of the vane wheel is, therefore, the following: the inner part acts as a turbine and the outer part as a propeller.

Fig.2 shows model vane wheels with 3, 6, 9 and 12 blades, which were used for model tests in the Hamburg Ship Model Tank. The vane wheel rotates, of course, with a smaller number of revolutions than the propeller. Its number of revolutions is determined by the pitch and the distribution of the pitch above the radius and it is adjusted in such a way that the torque is balanced i.e. the resulting torque is equal to zero at this wheel. The torque supplied by the turbine part of the wheel is spent in the propeller part. Subsequently, with this wheel the consequences of a small error of the pitch are not as unpleasant as with the propeller.

The pitch distribution of the vane wheel above the radius is completely different from that of a propeller. In order to explain this in a simple way two speed diagrams and the corresponding hydrodynamic forces will be discussed, one diagram for an inner radius in the turbine part and a second diagram for an outer radius in the propeller part, Fig. 3. In addition, it is here assumed that the incident flow is rotational symmetrical and that the induced speeds and the resulting hydrodynamical forces are standing perpendicular to the resulting incident flow of the blade sections.

Then the proportions thrust- to torque component are following for both radii

$$\frac{T}{Q} = \frac{1}{r \cdot \tan \beta_i} \qquad (1)$$

whereby in the turbine part thrust T and torque Q are negative magnitudes. Since with the wheel the resulting torque must be balanced, the widths Δr on both radii are chosen so big that the two moments are just eliminated. Then follows:

$$T_{prop} = -T_{turb} \cdot \frac{(r \cdot \tan \beta_i)_{turb}}{(r \cdot \tan \beta_i)_{prop}} \qquad (2)$$

The thrust produced in the propeller part, therefore, only exceeds the resistance produced in the turbine part (negative thrust), if the hydrodynamical pitch - $r \cdot \tan \beta_i$ - is bigger in the turbine part than in the propeller part. The simplifying assumptions here applied are, of course, not maintained for the calculation and the design of the vane wheel. The result of this explanation, however, remains valid.

A further characteristic of the vane wheel is the number of blades and the number of revolutions. The number of blades of the propeller is between 2 and 6. The induced efficiency of a propeller is increasing with the rising blade number, but the friction losses are also increasing. From this are resulting the blade numbers for propellers with regard to a possibly good efficiency. This is, however, different for a vane wheel: firstly the number of revolutions is extreme low and secondly, the circulation above the radius is changing the sign. Both characteristics demand a big number of blades. Fig. 4 shows for <u>one</u> given propeller the result of a systematical calculation, a variation of the number of revolutions and blades of the vane wheel, located behind the propeller. The figure does not only show that the number of blades

must be big, it also shows that the vane wheel must rotate much slower than the propeller. This indicates that the number of revolutions of the vane wheel is even lower than that of a slow running propeller, being applied for the same propulsion task whose diameter would be the same as that of the vane wheel. The extreme low number of revolutions of the vane wheel can be realised without any difficulties, as it is not mechanically conditioned but only by the choice of the pitch.

3. Model Tests

Results of model tests are available; systematical tests have, however, not been carried out and can hardly be considered due to the big number of freely eligible parameters.In addition, one difficulty must be mentioned and discussed, which reduces the value of the results from tests with models of the PVW.

It is known that in tests with model propellers the Reynold's number for one blade section on the representative radius .7 R shall not be smaller than 3.10^5. Otherwise a laminar flow would arise over a large part of the blade surface and a conversion of the test results to a full scale unit would become problematical. Subsequently, the diameter of propeller models should not be smaller than about 200 mm and for propulsion tests, for which Froude's similarity law has to be observed, the ship models must be at least 6 m long. The Reynold's number of a blade section of a vane wheel amounts to only a fraction of that of a blade section of the propeller, because the number of revolutions of the vane wheel is smaller and the chord lengths of the blade sections are smaller than those of the propeller.

It is, therefore, hardly possible, especially if the Froude's law must be observed, to chose the models so big that the Reynold's number for the vane wheel lies above 3.10^5. The diameter of the vane wheel would then have to amount to 400 mm and the length of the model to at least 15 m. A big scale effect must, therefore, be accepted at tests with PVWs and a somewhat inexact conversion to full scale must also be taken into account.

Fortunately, the propeller theory is highly developed and the elements of this theory are adequate to deal also with the vane wheel. Nevertheless, it is not possible to do without a model test. This remains necessary, in order to comprehend the interrelationship between ship and propulsion unit, i.e. in order to determine the wake- and thrust deduction numbers - w and t - as well as the efficiencies η_H, η_R.

The hydrodynamical reaction of the vane wheel on the propeller and the ship can be assumed to be very small. This is confirmed by the model tests and that makes it possible to suggest the following procedure for model tests:

a) In the first place the conventional tests are carried out: resistance-, propulsion-, propeller open-water tests and wake field measurements with a suitable stock propeller or with a propeller specially designed for the PVW. These tests supply the above mentioned factors of the interrelationship and they allow a sufficient estimation of the flow behind the propeller in the prospective vane wheel area. It is and must be sufficient, to determine the mean value of this flow via the circumference and to consider a rotational symmetrical flow for the vane wheel.

b) After that the vane wheel can be calculated. The result is the profit gained by the vane wheel. Subsequently small corrections can be made at the values determined under a), namely the values of velocity, number of revolutions, propeller pitch and efficiency. If this calculation of the vane wheel shall be controlled by a model test - as far as this is possible - an open-water test can be carried out with a model of PVW - as large as possible - and at a speed as high as possible. With a test of this kind the Froude's law does not have to be observed.

In order to underline the usefulness of this method I would like to refer to the following, always confirmed results of open-water tests, Fig. 5.

aa) The coefficient of the torque is changed by the vane wheel behind the propeller to such a small extent that it is hardly measurable. This seems to indicate that there is no hydrodynamical reaction of the vane wheel on the propeller and only a very small friction in the vane wheel bearing.

bb) The curve of the thrust coefficient decreases more steeply for the PVW than for the propeller alone. The thrust coefficient is higher for small and mean advance ratios and lower for large advance ratios than for the propeller alone. This is, of course, related to the fact that with high advance ratios only little energy is contained in the propeller slipstream and that then the friction losses at the vane wheel exceed the profit which might still be possible.

If an operating point is determined by the model tests under a) - perhaps point A in the diagram of Fig.5 - the new operating point B for the PVW is obtained by putting a square curve through A (or a curve of a higher degree corresponding to the known resistance curve of the ship). For the new operating

point B a smaller coefficient of the torque is valid as well as a bigger coefficient of the thrust, advance ratio and efficiency.

Fig. 15 originating from the full scale unit also confirms that the reaction of the vane wheel on the propeller is small and that there are no difficulties to correct the pitch of the propeller. In both cases no additional model tests were carried out. In the first case - a motor launch - the flow behind the existing propeller was estimated and the vane wheel designed. This figure shows that an influence of the vane wheel on the power (and by this also on the torque) does not exist. According to the above statements the torque should become smaller by the vane wheel at an equal number of revolutions. Here this influence should be especially small since the motor launch is running with very high Froude numbers up to .4 and the resistance curve is subsequently extremely steep. In the second case - the research ship "GAUSS" - the pitch was slightly corrected for the propeller of the unit PVW in accordance with the conventional propeller, which was already manufactured. Thus in both cases the same power at an equal number of revolutions was obtained. How this was achieved, is shown in Fig. 15 for the motor launch.

4. Remarks concerning the Design and Optimum Performance of Propeller and Vane Wheel.

The task remains, of course, to design the propulsion unit and to chose the involved parameter in such a way that the optimum effect is achieved. The propeller theory offers two methods for the design of a propeller: The method based on the Goldstein mean value factors and a further method which applies the induction factors developed by Lerbs. It is perhaps more simple to use the first mentioned method if the pitch over the radius remains equal or almost equal. This vastly applies to conventional propellers. The second method does not demand this supposition, it can be applied for any pitch distributions. Both methods are of the same value for a pitch which remains equal above the radius. For the calculation of the vane wheel only the Lerbs' induction factor method can be applied. It remains valid and useful for the special incoming flow, the special pitch distribution and for the change of the sign of the circulation of the vane wheel. It is necessary to chose much more supporting places above the radius than it is necessary and usual for propeller calculations and design.

The Lerbs' induction factor method delivers good results also for the vane wheel; it supplied, for instance, the results shown in Fig. 4. This method is here not described in detail. A more simple method is discussed, which

can only be applied for an infinite large number of blades and naturally for a rotational symmetrical flow. This method allows, however, a good insight in the conformities and steps of computation.

4.1 Conventional Propeller

I would like to begin with some introductory remarks on the conventional propeller.

This is done because different systems are to be compared with each other; in particular, the qualities of the propeller and vane wheel will be compared with those of the conventional propeller and of other propulsion units. Kramer's well known diagram, Fig. 6, shows for the free-running propeller the mathematical interrelationship existing between the advance ratio λ, the thrust coefficient C_T, and the induced efficiency η_i. The diagram is even valid for an arbitrary number of blades, the influence of the blade number appears as a shift of the entrance advance coefficient (not plotted in Fig.6). This influence can here be regarded as small and may serve as a reason for the use of the aforementioned simplification - according to which an infinite number of blades is assumed; the more so since it is not the object of this work to provide new design data, to present new findings for the conventional propeller, or to replace well-known design procedures. The brief discussion of the conventional propeller shall provide only a basis for comparisons. Kramer's diagram can be used only as an interim result. In this diagram no viscosity losses are taken into account, but only exit losses leaving the system in the form of kinetic energy in the propeller slipstream. The viscosity losses are now introduced by means of the well-known concept of the drag-lift ratio. As usual in propeller theory, the drag-lift ratio is estimated at a blade section according to the formula

$$\varepsilon = \frac{0.008}{C_L} \qquad (3)$$

The lift coefficient C_L and thus the length of a chord of a blade section are essentially determined by the danger of cavitation. The lift coefficient and thus the drag-lift ratio are estimated with a view to Burrill's diagram, known to all propeller experts as

$$C_L = 0.2 \cdot \sigma^{0.55} \qquad (4)$$

where σ is the cavitation number of the blade section.

The drag-lift ratio thus amounts to

$$\varepsilon_p = \frac{0.008}{0.2 \cdot \sigma^{0.55}} \tag{5}$$

and for a velocity of V = 8 m/s, for instance, about

$$\varepsilon_p = 0.016 \cdot \left[1 + \left(\frac{r \cdot \omega_p}{V}\right)^2\right]^{0.55} \tag{6}$$

The drag-lift ratio, which becomes higher for peripheral blade sections than for inner ones, is applied herein to the propeller. For the vane wheel the coefficient will be assumed to be somewhat higher, that is, somewhat less favourable

$$\varepsilon_L = 0.020 \cdot \left[1 + \left(\frac{r \cdot \omega_L}{V}\right)^2\right]^{0.55} \tag{7}$$

Table 1 shows the magnitude of the drag-lift ratio and lift coefficient for different radii according to this estimation. This estimation, which seems realistic, permits comparison between different propulsion devices under equal cavitation conditions. The blade area of a wheel is not explicitly treated in this paper. Instead of the blade area, the velocity (V) is the parameter, given as 8 m/s throughout. Here, I would like to repeat that no design data are offered; instead, different propulsion devices are compared. It is considered important that the drag-lift ratio is chosen to be dependent on the cavitation number; thereby it is smaller for lower circumferential speeds than for larger speeds.

Table 1 Lift coefficients and drag-lift ratios for a propeller with an average ratio $\lambda = 0.20$ and for a velocity of 8 m/s

r/R	0.2	0.4	0.6	0.8	1.0
C_L	0.342	0.273	0.233	0.206	0.187
ε_p	0.023	0.029	0.034	0.039	0.043

Thus all conditions are provided for the determination of the efficiency of the propeller in real fluid. The plotting is done analogous to Kramer's diagram (see Fig. 7); however, the real efficiency not the induced efficiency, is given as the parameter of the curves.

Only within a small range are the curves of practical meaning, from the point where straight lines with a gradient of 2 touch the curves up to the maximum of the curves, where horizontal lines touch them. At given values of thrust and velocity, a straight line with a gradient of 2 represents a constant number of revolutions but varying diameters, while a horizontal line with a gradient of zero represents a constant diameter but varying number of revolutions. The diagram replies directly, therefore, to questions concerning the optimum number of revolutions at a given diameter or concerning the optimum diameter at a given number of revolutions. The replies are given by the determination of the touching points of a horizontal line or a straight line with a gradient of 2 with the set of curves.

Fig. 7 represents, of course, the same facts as the known graphs, for instance, the series diagrams. This one is, however, more suitable for purposes of the present work.

4.2 Propeller and Vane Wheel

Fig. 8 shows preliminary results for an efficiency of 0.70, one curve for the propeller alone and a series of curves for the propeller and vane wheel, valid for different diameter ratios: DD = DL/DP 1.2, 1.3, 1.4 and 1.5. It can easily be seen that with the propeller and vane wheel larger thrust coefficients can be produced, since the total momentum necessary to generate the thrust is given to a larger quantity of water. For the larger thrust of the combined propeller and vane wheel device, a larger power must naturally be delivered to the propeller. This means that with increasing diameter ratio the propeller needs wider blades, and the increase of the chord length of the blades is limited by the problem of fixing the blade area.

Nevertheless, it is not the most important task to determine this limit, as will soon be shown. A presentation of the results - as in Fig. 8 - does not automatically allow a judgement. A different presentation would be more suitable for a judgement, where a reference diameter for the thrust coefficient and advance ratio, that is, the diameter of the vane wheel - the largest diameter of the system - should be used, and not the diameter of the mechanically driven propeller (as in Fig. 8). The curves of Fig.8 are then shifted obliquely downward the more the diameter ratio increases; see Fig. 9. The curves are then close together so that they even form an envelope. This means that for a given total diameter of a device there exists an optimum diameter ratio. It also means that often it is not necessary and not even optimal to go up to the aforementioned limit, caused by the necessity of fixing the propeller area. The points of the envelope could be marked with

data of the diameter ratios - the touching points of the individual curves with the envelope. The envelope represents the optimum solutions. In the following, only these envelopes are given; they are deliberately stopped at the left end - in the direction of decreasing advance ratios and increasing number of revolutions, respectively - at a diameter ratio of 1.4.

The recognition that the curves for different diameter ratios (Fig.9) form an envelope and that there is an optimum diameter ratio for each touching point, provided the total diameter and the number of revolutions are fixed, leads to the problem of the optimization of the propulsion devices. This optimization can be taken as a central point of this lecture.

4.3 Optimization

An optimum layout of a propulsion device within the limiting conditions given is a natural demand. Optimization means in this case that the propulsion device must be designed in such a way that for the given values of the thrust, velocity, diameter and number of revolutions the least propulsion power will be required; that is, the highest efficiency will be obtained. Here, we are concerned with a little more than the well-known questions, mentioned earlier, as to the optimum number of revolutions for a given diameter, and vice versa, the optimum diameter for a given number of revolutions. These certainly important questions are answered within the general optimization sought. Optimum design of blade section profiles, blade contours, etc. is not of concern here either, as important as it might be. Nothing new can be contributed here; these aspects have been accounted for in this paper by the choice of the drag-lift ratios. But, then, what can still be optimized ? Table 2 gives the answer very clearly.

Table 2

Device	What Shall be Brought to an Optimum ?		Method	Criterion	Remarks
Conventional propeller	radial distribution of the induced velocity		variation of this velocity	$\frac{\delta\, TP(r)}{\delta\, QP(r)} = CP$	.....
Propeller plus vane wheel	radial distribution of the velocity	induced by the vane	variation of this velocity	$\frac{\delta\, TL(r)}{\delta\, QL(r)} = CL$	QL = 0
		induced by the propeller	variation of this velocity	$\frac{\delta\, TP(r) + \delta\, TL(r) - CL \cdot \delta\, QL(r)}{\delta\, QP(r)} = CP$	
	ratio of diameters		variation of the propeller diameter	$\frac{\delta\, TP + \delta\, TL}{\delta\, QP} = CP$	
	ratio of loads		trial and error	.....	
	ratio of number of revolutions		trial and error	.....	

TP(r), TL(r), QP(r), QL(r) = thrust and torque for an annular element of radius r and of beam 1.
TP, TL, QP, QL = resulting thrust and torque amounts.

For the conventional propeller, only the optimization of the radial distribution of the pitch (or of the induced velocity) is left. This optimization problem was first dealt with (as known to all experts) by Betz in 1919. He found that the hydrodynamic pitch for a weakly loaded propeller in an ideal liquid must be a constant for all radii. A little later the problem was also treated for the propeller in a viscous fluid. The optimization of the propeller and vane wheel system requires a corresponding number of conditions to be met according to the larger number of free design variables.

First it must be ensured in each case that the resulting hydrodynamical torque is equal to zero at the non-mechanically driven vane wheel

$$Q_{Lres} = \int_{\text{over all radii}} QL(r)dr = 0 \qquad (8)$$

(Friction in the bearing is neglected). This condition does not belong to the optimization conditions, since it must be fulfilled in any case. The radial distribution of the velocity induced by the vane wheel is optimized by the condition given in Table 2 in the first place. The condition is valid in the same way for the turbine, and for the propeller part of the vane wheel.

For the second condition, leading to the radial distribution of velocity induced by the propeller, it has to be taken into account again that a variation of this velocity is followed by a variation at the vane wheel. The variation of the thrust, therefore, consists of three parts:

$$\delta T_{PL}(r) = \delta T_P(r) + \delta T_L(r) - CL \cdot \delta Q_L(r) \qquad (9)$$

- of the variation of the thrust element $T_P(r)$ produced directly at the propeller,
- of the variation of the thrust element $T_L(r)$ produced by the same quantity of water at the vane wheel, as well as
- of the variation of the thrust element produced in the propeller part of the vane wheel, which was caused by the torque element $Q_L(r)$ produced in the turbine part, by the same quantity of water as with the two thrust elements as mentioned in the foregoing.

The variations $T_L(r)$ and $Q_L(r)$ are different from the variations in the first condition, since here they are caused by a variation of the velocity induced by the propeller, while previously they are caused by a variation of the velocity induced by the vane wheel.

By means of the third condition the propeller diameter (and thus also the diameter ratio) can be optimized at a fixed total (vane wheel) diameter. The propeller diameter is varied and the variations occurring in the condition are the alterations of the total thrust and the total torque caused by the diameter variation.

In order to optimize the load ratio, the ratio of the constants of the first two conditions CL:CP must be chosen correspondingly. It might be the best way to repeat the computations for different ratios of the constants mentioned.

An explicitly formulated condition for determination of the optimum ratio of number of revolutions does not exist. It remains only to calculate optimum devices for different revolution ratios with regard to the first four conditions. The optimum ratio of number of revolutions will then be chosen according to the results of these calculations. It turns out that the optimum ratio

revolutions of vane wheel
to
revolutions of propeller

is lying between 0.35 and 0.45. This was already found in a different way (see Fig.4).

5. Analysis of Results of Point 4

The maximum values are all only weakly distinguished. Therefore, only strong deviations from the optimum conditions are significantly noticeable as decreases in thrust coefficients and efficiencies. This helps to overcome one difficulty caused by the second condition. The second condition leads to a strong shifting of the load to the inner radii and to the difficulty of placing the blade area necessary for this loading at these radii. In the calculations, the results for the propeller and vane wheel are presented; the condition is, therefore, broken at the inner radii in the sense of a discharge of these blade sections. The following detail might be of interest.

The conditions - in particular the first two - supply for the vane wheel also the limits of the turbine and propeller part. They supply for the torque elements at the vane wheel $Q_L(r)$ negative values at the inside and positive ones at the outside of the vane wheel. Where, moving from the inner

region to the outer one, the negative torque element becomes zero, the limit of the turbine is reached, and where the torque element starts to become positive, the propeller begins. Both limits may be within the propeller slipstream. They do not coincide, however, and a small unloaded part of the vane wheel is left between them for which the first optimization condition cannot be applied. The reason for this unsteady transition is that the hydrodynamical lift force changes the direction suddenly by about 180 deg at the alteration of the sign of the angle of attack of a profile, that is at the transition from a turbine to a propeller profile, while the direction of the friction force remains almost unchanged, Fig. 10.
Fig. 11 shows the results of the calculations and - in order to facilitate the comparison - once again the results for the propeller. It shows that essential differences exist between the performances of the devices compared. This comparison shows in particular a superiority of the propeller and vane wheel at small advance ratios (high number of revolutions).

In order to gain further insight, the following analysis of the results is made. One point chosen deliberately in the diagram belongs to two sets of curves. It represents a possible operating point for two propulsion devices, whereby the devices produce the same thrust at the same velocity, the same diameter, and the same number of revolutions. Power and efficiency are, however, different.

An operating point ($C_t = 1.32, \lambda = 0.212$) is chosen for a moderate loading, whereby the performance cannot be improved by a Kort nozzle. Fig. 12 shows for this operating point the velocities in the slipstream far behind the corresponding propulsion devices. Fig. 13 shows for the propeller and vane wheel system the same velocities and in addition the velocities between the two parts of the device. According to the continuously applied simplifications, it is here assumed that the propeller slipstream is fully developed at this point of the gap, while the influence of the vane wheel is not yet noticeable. The influence of the turbine, of the unloaded intermediate part of the vane wheel is clearly shown in this diagram. The differences between the two curves (a and b) in the inner region represent the velocities induced by the turbine of the vane wheel.

Knowledge of these velocities allows us to analyze the losses. Table 3 gives the results of this analysis.

Table 3 Comparison of Propulsion Devices

	Conventional Propeller	Propeller plus Vane Wheel	Propeller plus Vane Wheel
Operating Point	CT=1.32,λ=0,21	CT=1.32,λ=0.21	CT=0.917,λ=0.175 diameter larger by 20%
Efficiency, %	62.27	65.00	67.99
	Losses, %		
Exit losses by axial and by tangential velocities in the slipstream	17.83 4,18 22.01	19.25 3.00 22.25	15.54 1.43 16.97
Friction losses at the propeller and vane wheel	15.72 15.72	10.30 2.45 12.75	11.27 3.77 15.04
		Improvement in Efficiency, %	
		4.38	9.19

For the propeller and vane wheel the energy loss by rotation of the slipstream is decreased.

The amount of the friction loss, however, is unexpected and noticeably smaller than with the conventional propeller. In this case the superiority of this device is even based - and that almost completely - on the decrease of the friction loss, which can easily be explained. It is based on the decrease of the diameter and thus of the circumferential speed of the mechanically driven part of the device. For the "frictional efficiency" of a propeller, the following estimation is known:

$$\eta_{\varepsilon} = \frac{1 - 2 \cdot \varepsilon \cdot \lambda_i}{1 + \frac{2}{3} \cdot \frac{\varepsilon}{\lambda_i}} \qquad (10)$$

In the case in question the drag-lift ratio ε is smaller and the induced advance ratio λ_i is larger than for the conventional propeller. This is caused by the decrease of the circumferential speed of the mechanically driven part. Both changes cause an improvement of the "frictional efficiency". The friction loss of the vane wheel is admittedly added to the friction loss at the propeller. It is, however, small due to the small number of revolutions of this wheel.

Table 3 contains a third column, which is valid for a propeller and vane wheel system which produces the same thrust and is operated with the same number of revolutions as the other devices, but in which the diameter is - corresponding to the original idea - larger by 20 percent. This device is working with an efficiency of 67.99 percent. While Device No.2 of the same diameter as the conventional propeller achieves an improvement in efficiency of only 4.3 percent, this improvement amounts to 9.1 percent for Device 3. Both improvements are related to the efficiency of Device 1, the conventional propeller.

The result of another device is added, a propeller and vane wheel device which differs from that given in Column 3 of Tabe 3 only in that the mechanically driven propeller is not optimized for the combined system but for the conventional one. This device achieves an efficiency of 66.0 percent, which means an improvement of 5.9 percent compared with Device 1. This example shows the importance of the optimization of the propeller of the combined device.

Fig. 11 shows that in a wide range of applications, even for low or moderately loaded devices, the propeller and vane wheel system can be superior (regarding efficiency) by more than 5 percent. It also shows that improvements of 10 percent and more can be obtained with more highly loaded propellers. Optimization of the propeller and vane wheel system with regard to efficiency and the subsequent results have been sufficiently discussed. The diagram in Fig.11 can easily be applied; it is clearly arranged. Calculations for systems with a finite number of blades which were also carried out to a large extent, using vortex theory, as well as almost all model tests, confirm that the differences described between both devices are qualitatively and to a certain degree also quantitatively maintained. Since the power is larger in the slipstream of a propeller with a finite number of blades at equal thrust and advance ratios than with an infinite number of blades, the advantages of the propeller and vane wheel system can even be greater.

An additional property concerning the efficiency of the propeller and vane wheel system remains to be mentioned. An increase of loading behind the ship due to a seaway, or fouling of the ship's hull or similar circumstances, reduces the efficiency of the propeller. If a fixed blade propeller operated by a diesel engine is used, the power delivered by the engine decreases due to the reduction of the number of revolutions. Both effects become smaller for the system propeller and vane wheel. This can be recognized from the open-water diagram, Fig.5, which shows the curve of the thrust coefficient increasing more steeply with decreasing advance ratio than for the propeller alone.

For the design the real conditions must naturally be considered as far as possible. The data of the interrelationship between ship and propulsion unit must be known and the Lerb's induction factor method must be applied. The method as described under 4.2 has proved to be useful also for a finite number of blades in order to determine the optimum pitch distribution. The diagram in Fig. 11 is recommended for a first estimation of the profit to be achieved by the PVW. For this purpose it is attempted to find the operating point of the propeller behind the ship. Starting from this point the improvement of the efficiency caused by the vane wheel depending on the diameter can then be roughly estimated. Here it has to be observed - and it is repeated here - that for the PVW in this diagram the biggest diameter of the system is employed for the applied parameters, i.e. the diameter of the vane wheel.

6. Full Scale Units

Because of the difficulties with model tests - mentioned under 3. - it was particularly important to obtain results from full scale units. In addition, further questions regarding manoeuvering properties, operating safety etc. can only be answered by full scale trials.

In 1969 it was already possible to carry out a test on a motor launch. [2] The diameter of the vane wheel amounted to 1.35 m. Fig. 14 shows the arrangement on a motor launch of the Howaldtswerke-Deutsche Werft AG, shipyard, prepared for a test which was promoted by the Research Centre of German Shipbuilding. It is remarkable that in this case, only the propeller shaft had to be extended, in order to fit a pivot for the bearing of the vane wheel, and that otherwise no further changes to the ship or engine were necessary. It must also be mentioned that for this test a new propeller was not manufactured.

Fig. 15 shows the results of this test, plotting the power and the thrust of the propulsion device versus the number of revolutions, once with and once without the vane wheel. It shows in particular that at the same propulsion power the thrust was increased by 10 to 13 percent with the vane wheel.

A continuous trial was not carried out on the motor launch. The bearing of the vane wheel was only improvised. Besides the motor launch was running at full power with the Froude number .4, at which the resistance curve rises extremely steep and an improvement of the efficiency of the propulsion unit can hardly lead to a remarkable increase of the velocity. The research ship "GAUSS", which is used by the Deutsches Hydrographisches Institut for research

and investigation tasks, was completed in April 1980. After some trial- and measuring trips with a conventional propeller the ship was equipped with the new-type propulsion system "Propeller and Vane Wheel" in May 1980. Since that date the ship is operating with this unit, this means an operation period of three years.

The design of the "GAUSS" was concluded and the building started before the suggestions could be introduced to equip the ship with the PVW. The plans had, therefore, been prepared for the conventional unit: propeller with fixed blades. Model tests had been carried out in the Hamburg Ship Model Tank for the conventional unit. From the test report of this institute (WP 12/79) only one prognosis is here reported:
For the trial trip a velocity of 13.54 kn at a shaft power of 1177 KW is predicted, assuming a headwind of Beaufort 2. Further data for this state are given as follows:

a total efficiency η_D	of 70,7 %
a propeller efficiency η_o	of 61,6 %
a wake number	of 20.7 % and
a thrust deduction number	of 14.2 %.

On the basis of these data the conventional propeller was designed and produced by Messrs. Schaffran, Propeller Company, Lehne Limited:

diameter	2,40 m
mean pitch	2,10 m
developed area	3,10 m^2
number of blades	5
pitch ratio	0.875
area ratio	0,685
mass	1850 kg

In March 1979 the suggestion to equip the ship with propeller and vane wheel has been made for the first time. The decision to realise this suggestion and provide propeller and vane wheel as an alternative beside the conventional propeller was only made in January 1980, after the launching and only three months before the ship was completed.

Soon after the first contact it was decided to lift the rear end of the propeller shaft by 150 mm, in order to be able to locate the vane wheel, (diameter 2,75 m) at a later date, when occasion arises. This remained the

only structural change. Rudder, propeller aperture etc. remained unchanged, as also the position of the propeller, apart from the above mentioned lift. Even the propeller shaft remained unchanged, since the pivot for the bearing of the vane wheel is not fixed to the shaft but screwed on the hub of the propeller, in order to allow to dismantle propeller and vane wheel, without having to remove the rudder. The location of the propeller shaft in an elastic tube had been prescribed right from the beginning. The result of it was a free place for the propulsion units. The elastic tube could so be dimensioned that for both alternatives sufficient strength and good vibration properties could be granted, Fig. 16. The following conditions for the design and construction of the system propeller and vane wheel had to be observed:

- It must also be possible to use the mechanically driven part of the unit - the propeller - as a free-running spare propeller.
- This propeller must be produced from the already casted raw propeller.
- The diameter of the vane wheel is not allowed to exceed 2,75 m, the number of blades shall not be more than 7.
- The total unit must meet the demands of the ice class "E 2" of the Germanischer Lloyd.

These conditions partly resulted from the state of building of the ship at that time and of its parts. To a greater part it resulted,however, from the wish to keep the risk of the first realisation on a ship as limited as possible. Additional model tests could not be carried out, the available time was too limited and the ship model too small.

Corresponding to these conditions the unit propeller and vane wheel had been designed and manufactured:

Propeller

diameter	2.32 m
mean pitch	2.14 m
developed area	3.04 m^2
number of blades	5
pitch ratio	0.92
area ratio	0.72
mass	1800 kg

Vane Wheel

diameter	2,75 m
mean pitch in the propeller part	4,15 m
in the turbine part	6,29 m
developed area	2,39 m^2
number of blades	7
pitch ratio	1,51 / 2,29
area ratio	0,40
mass	1550 kg

It can be taken for granted that the unit could not be layed out to an optimum because of the before mentioned condition and due to lack of experience.

6.1 Trials

All measurements and evaluations, which are here discussed, were carried out by members of the Hamburg Ship Model Tank and the ship's management. Measured mile trips were made: with the conventional propeller on 9th May 1980 and with propeller and vane wheel on 27th May 1980. The weather was favourable on both days, at wind forces of Beaufort 3 most of the time, which only occasionally increased to 4 to 5. All tests were made which seemed to be important for the judgement of the propulsion units. Essential differences between the two units were only established at the velocity- and power measurements. For this reason no complete report on the other trials will here be given but just a short information.

6.2 Results

6.2.1 Velocity- and Power-Measurements

The velocities were determined by means of land marks and besides the resistance log of the Hamburg Ship Model Tank. The torque was measured behind the driving engine at the shaft by means of a Philips torsion indicator. Unfortunately, a thrust measurement could not be carried out. For each pair of trips - with and against the current - the current velocity parallel to the mile could be determined from the "stopped speeds above the ground" and at the same time from the velocities "compared to water" which were determined with the log. Current velocities of 0.12 to 0.22 kn were measured. After the correction of the stopped velocities differences remained between

the mile out and the return mile, which could only be caused by the different wind- and sea strengths. From these corrected velocities - corrected only with regard to the current velocity - the speeds were determined in the well-known way, which are valid when running in calm water.

Fig. 17 shows a comparison: power versus velocity, which is valid for calmwater free of current and waves. The most important result of the whole trial is a power saving of about 9 percent, this means an increase of the velocity of 0.3 kn at the same power.

6.2.2 Standard Manoeuvering Test

There were no remarkable differences between the behaviour with a conventional propeller and that with the propeller and vane wheel. In Fig.18 the time to attain the switching value, the reach time and the time for a full swing are shown as function of the rudder angle.

6.2.3 Turning Circles

Table 4 includes the measured data for a rudder angle of 35° and a rudder setting time of 22 sec for a rudder angle of 65°.

6.2.4 Stop Tests

The results of the stop manoeuvres at a starting speed of 13.4 kn are given in Table 5.

6.2.5 Going Astern

In both cases the ship could be drawn astern in a straight direction. There were no measurements carried out.

6.2.6 Pressure Fluctuation Measurements

Five pressure gauges were firmly fixed to the hull in the propeller range. Two of them failed on the first day, so that only the results of three pressure gauges could be compared. In Fig. 19 the result of the harmonic analysis is shown. For five speed ranges the figure shows the harmonic components of the 5th, 10th, 15th and 20th order of the number of revolutions of the propeller. The amplitudes of fluctuations with the blade frequencies of the vane wheel, which are not shown in Fig. 19, amounted to a maximum of 43 percent of the amplitudes of the propeller blade frequency.

Table 4

	Conventional Propeller Port	Conventional Propeller Starboard	Propeller and Vane Wheel Port	Propeller and Vane Wheel Starboard	
Speed at the beginning of the manoeuvre	13,0	13,0	13,4	13,4	kn
Time for the full turning circle	148	152	143	144	sec
Diameter	152	161	143	153	m

Table 5

	Conventional Propeller	Propeller and Vane Wheel	
Stop time	88	85	sec
Stop way	290	280	m

The pressure amplitudes are lower than on board of many other ships.

6.2.7 Vibrations

The vibrations are low everywhere: on the bridge, in the engine room and in the living quarter.

6.3 Summary of the Measurement Results achieved on board of the "GAUSS"

The following was established:

- Within the investigated speed range of 10 to 13.8 kn the ship requires at equal velocity with a conventional propeller a power which is higher by 9 percent than with a propeller and vane wheel. In this connection it should once more be referred to the fact that - unlike the conventional propeller - in this case an optimum design of the system propeller and vane wheel was not possible due to the circumstances as mentioned before. The author is of the opinion that an optimum design would have resulted in a power reduction of 11 to 12 percent.
- The manoeuvering properties are almost equal in both cases.
- The stop properties are slightly improved by the system propeller and vane wheel.
- The properties of the ship - when running backwards - are about equal in both cases.
- The pressure fluctuations at the hull are also about equal in both cases.
- The vibrations did not cause any claims or difficulties.

6.4 Operating Experience

The propulsion system propeller and vane wheel is installed on the "GAUSS" since May 1980. The conventional propeller is used as a spare propeller. The ship was twice docked in. Propeller and vane wheel remained undamaged and in good operating condition. The vane wheel could easily be turned by hand, in the dock it was already turned by a strong wind. During the docking in April 1982 the vane wheel was dismantled, in order to check the bearings, and they were also undamaged and in good operating condition. They were cleaned, filled with new grease and again installed.

The "GAUSS" is operating as research ship with many outboard devices. The danger that foreign bodies could come into the propulsion unit must be expected to be higher than with other ships. Twice a hawser did in fact get into the unit. In both cases the consequences were not too bad, the hawsers were cut and fall out of the unit after a slight turning and the ship continued its work.

In the winter 81/82 the ship came repeatedly into heavy ice. The ship's management reported strong strokes in the propulsion unit. The ship could, however, continue its service. The PVW has resisted the heavy stress as it could be noted in the dock in April 1982.

The ship's management considers the handling of the ship to be normal, i.e. to be the same as with the conventional propeller. They only wish to obtain a device which shows the number of revolutions of the vane wheel, in order to be able to judge whether it is working well. On account of these experiences the unit can be regarded as safe in operation. In addition, it can be concluded that the risk of an installation on other ships can be regarded as limited and estimable. According to the author's view this risk will certainly remain much smaller than that which was connected to the development and introduction of the propeller with adjustable blades.

7. Propeller and Vane Wheel - Slow-Running Propeller

In the introduction it was already mentioned that the PVW guarantees an efficiency which is at least as high as with the BSLP, but it has a higher number of revolutions. This can also be seen in Fig. 11, and even more explicitly in Fig. 20 which was calculated for a special case. This is, at first, only valid for the same diameter. Apart from that it is probably possible to chose a larger diameter for the PVW and to lead the tops of blades nearer to the hull than this is allowed with the BSLP, since the circumferential speed, the danger of cavitation, the loading of the tops of the blades are smaller and the number of blades greater than for the BSLP.

Literature

1 Grim, O., "Propeller and Leitrad", Jahrbuch der Schiffbautechnischen Gesellschaft, Band 60, 1966.

2 Chirila, J., Engle, G., and Grim, O., "Untersuchung eines Vortriebsorgans - Propeller und Leitrad auf einer Barkasse", Forschungszentrum des Deutschen Schiffbaus, Bericht No. 10, 1969.

3 Grim, O., "Propeller und Leitrad, Propulsionsversuche mit einem völligen Modell", Forschungszentrum des Deutschen Schiffbaus, Bericht No.22, 1971.

4 Wagner, R., "Rückblick und Ausblick auf die Entwicklung des Contra-propellers", Jahrbuch der Schiffbautechnischen Gesellschaft, Band 30, 1929.

5 Suhrbier, K.R., "An Experimental Investigation on the Propulsive Effect of a Rudder in the Propeller Slipstream", International Ship Building Progress, Vol.21 (1974).

6 Grim, O. "Propeller and Vane Wheel. J. Ship Research 24 (Dec 1980) No.4

7 Grim, O. "Forschungsschiff "GAUSS" - Erprobung des Antriebsorgans "Propeller und Leitrad", Hansa 117 (1980), Nr. 24.

Symbols

Dimensional Data :

r	radius
V	velocity of the incident flow
D	diameter
DP	diameter of the propeller
DL	diameter of the vane wheel
ω_P	angular velocity of the propeller
ω_L	angular velocity of the vane wheel
Γ	circulation
ρ	density
a, b	beam and length of the rudder

Non-dimensional Data :

RP	radius of a blade section of the propeller
RL	radius of a blade section of the vane wheel
RPM	maximum radius of the propulsion device
DD = DL/DP	diameter ratio
T	thrust
TP	thrust of the propeller
TL	thrust of the vane wheel
Q	torque
QP	torque of the propeller
QL	torque of the vane wheel

Cont'd

Non-dimensional Data :

ε	drag-lift-ratio of a blade section
ε_p (= EP)	drag-lift-ratio of a blade section of the propeller
ε_L (= EL)	drag-lift-ratio of a blade section of the vane wheel
C_L	lift coefficient for a blade section
σ	cavitation number for a blade section
λ	advance ratio for a blade section
λ_i	advance ratio, induced
CT	thrust coefficient
η	efficiency
η_i	efficiency, induced
η_ε	efficiency, frictional
XP, YP	coordinates of the velocity induced by a blade section
XL, YL	coordinates of the velocity induced by a blade section of the guide mechanism or vane wheel

APPENDIX

There is no need to apply the vortex theory as long as only rotational symmetrical flows are presumed. It is sufficient to base the computations on theorems of momentum. All computations and results should, however, agree with corresponding results of the vortex theory applied to systems with an infinite number of blades. Because the free vortices are deformed by the vane wheel or the guide mechanism it may even be questionable to what an extent the vortex theory can be applied in these cases. For the direction of the induced velocities the normal condition, known from the propeller theory is assumed, which rationally is applied to the guide mechanism and the vane wheel. This is an assumption which for more general cases, as occurring here, is not proved and which probably is not quite exact, but which is perhaps applicable as an approximation. In an earlier work of the author this assumption was not used; finite numbers of blades were assumed at that time and the computations were based on the vortex theory as well as on the method of the induction factors according to Lerbs. There were no obvious discrepancies observed between the results of the two computation methods. The method described here is more simple, more transparent and, therefore, more easily to re-examine.

An unconventional presentation and an unconventional standardization are chosen, in order to obtain formulae as simple as possible. The quantities were normalized as follows:

The velocities with the incident (-flow-) velocity V,

the forces with $\pi \frac{\rho V^4}{\omega_p^2}$

the torques with $\pi \frac{\rho V^5}{\omega_p^3}$

the radii with $\frac{V}{\omega}$

ω_p = angular velocity of the mechanically driven propeller

If the power is accordingly normalized with $\pi \frac{\rho V^5}{\omega_p^3}$, it is identical with the corresponding normalized force. In particular, the normalized

thrust power is identical with the thrust and the normalized torque power with the normalized torque.

1.· Conventional Propeller

The well known velocity diagram for a radius r or normalized RP $= \frac{r\,\omega_P}{V}$ is then given (Fig.22)

by the incident flow velocity $\overline{AB} = 1$

the circumferential velocity $\overline{BC} = RP = \frac{r\,\omega_P}{V}$

and the induced velocity $\overline{EA} = \frac{U_{ind}}{V}$

In the propeller plane the half induced velocity $\overline{DA} = \frac{1}{2}\frac{U_{ind}}{V}$ appears and according to the supposed normal condition the induced velocity makes a right angle with the resulting relative velocity $\overline{DC}$. A rectangular coordinate system x,y with the origin in B is laid into the velocity plan. The coordinates of point E are designated with XP and YP, where XP is less than zero and YP greater than one. Due to the normal condition the points A and E are on a circle, whose centre is in C. Therefore, the equation of the circle is:

$$(RP + XP)^2 + YP^2 = RP^2 + 1 \qquad 1.1$$

The hydrodynamical force acts in the ideal fluid in the direction of the induced velocity. In a real fluid an additional friction force, which results from the multiplication with the drag-lift-ratio ε, is acting in the direction of the resulting velocity. For an annular element with the radius RP and the width dRP the normalized thrust element is

$$dT = RP \cdot (YP + 1) \cdot (YP - 1 + EP \cdot XP) \cdot dRP \qquad 2.1$$

and the torque element is

$$dQ = RP^2 \cdot (YP + 1) \cdot (-XP + EP \cdot (YP - 1)) \cdot dRP \qquad 3.1$$

where the drag-lift-ratio of the propeller is designated with EP.

In order to obtain the optimum pitch distribution the induced velocity is varied. The following variations then result for the annular element :

From the equation of the circle

$$(RP + XP) \cdot \delta XP + YP \cdot \delta\ YP = 0 \qquad 4.1$$

From the force equations:

$$\delta(dT) = RP \cdot [\, 2 \cdot YP \cdot \delta\ YP + EP \cdot (XP \cdot \delta\ YP + (YP + 1) \cdot \delta XP)] \cdot dRP \qquad 5.1$$

$$\delta\,(dQ) = RP^2 \cdot [\, - XP \cdot \delta YP - (YP + 1) \cdot \delta XP + 2 \cdot EP \cdot YP \cdot \delta\ YP\,] \cdot dRP \qquad 6.1$$

The optimum radial distribution of the induced velocity is obtained if

$$\frac{\delta\,(dT)}{\delta\,(dQ)} = CP \qquad 7.1$$

is the same value for all radii. Otherwise a higher thrust could be obtained at the same torque, the same diameter and the same number of revolutions. The four equations 4.1 up to 7.1 are satisfied for a given CP and thus the problem is solved.

The hydrodynamical pitch angle is then obtained from

$$\tan\beta_i = \frac{YP + 1}{2 \cdot RP + XP}\ , \qquad 8.1$$

the total thrust and the total torque by integration over all radii

$$T = \int dT\ , \qquad Q = \int dQ\ , \qquad 9.1$$

the efficiency from

$$\eta = \frac{T}{Q}\ , \qquad 10,1$$

the thrust coefficient amounts to

$$CT = \frac{\varrho\, T}{RPM^2} \qquad 11.1$$

and the advance ratio

$$\lambda = \frac{1}{RPM} \quad ,$$

where the maximum normalized radius is designated with RPM

$$RPM = \frac{RM \cdot \omega}{V}.$$

If this problem is solved for different given values CP and different radii RPM, the complete diagram - Fig.6 - is determined for a perfect fluid, i.e. for EP = 0, the Kramer diagram for $z = \infty$ is obtained in this way, Fig.7.

2. Propeller and Vane Wheel

The vane wheel encounters the velocity $\overline{EB}$ (Fig. 22) in the fully developed propeller slipstream ~~as the fixed guide mechanism above~~, and outside the propeller slipstream the axially directed velocity $\overline{AB} = 1$. It rotates with the angular velocity ωL.

The vane wheel induces the velocity $\overline{LE}$ in the propeller slipstream. This velocity is differently directed in the turbine- and propeller part.

XL, YL are the coordinates of point L

XP, YP are the coordinates of point E

A normal condition being supposed - like above - the points E and L are on a circle with the center H

$$(UE \cdot RP + XL)^2 + YL^2 = (UE \cdot RP + XP)^2 + YP^2 \qquad 1.2$$

Since the contraction of the propeller slipstream is not taken into account the same fluid mass is running through an annular element of the vane wheel as through the corresponding annular

element in the propeller plane.

$$RP \cdot (YP + 1) \cdot dRP \qquad 2.2$$

The thrust force is acting on the annular element of the vane wheel

$$dTL = RP \cdot (YP+1) \cdot [\, YL-YP \mp EL \cdot (XL-XP) \,] \cdot dRP \qquad 3.2$$

and the torque element

$$dQL = RP^2 (YP+1) \cdot [\, XP-XL \pm EL \cdot (YP-YL) \,] \cdot dRP \qquad 4.2$$

where in both formulae the upper sign (of the friction term) is valid for the turbine part and the lower for the propeller part. The torque element is negative in the turbine- and positive in the propeller part. Because the vane wheel is rotating with a smaller angular velocity than the propeller the formulae for the normalized torque and for the normalized rotational power of the vane wheel are no longer identical. They differ from each other by the factor UE, the ratio of revolutions. Instead of the formula 4.2 for the torque element the following formula for the rotational power is applied

$$dQL = UE \cdot RP^2 \cdot (YP + 1) \cdot [XP - XL \pm EL \cdot (YP - YL)] \cdot dRP \qquad 5.2$$

The formulae 3.2 to 5.2 are, however, only valid for that part of the vane wheel, which is in the propeller slipstream, for that part for which the fluid mass moving through an annular element is given by 2.2.

For the part outside the propeller slipstream 2.2 is replaced by

$$RP \cdot (YL + 1) \cdot dRP \qquad 6.2$$

and for the thrust or thrust power and rotational power 3.2 and 5.2 are replaced by

$$dTL = RP \cdot (YL + 1) \cdot [\, YL - 1 + EL \cdot XL \,] \cdot dRP$$

$$dQL = UE \cdot RP^2 \cdot (YL + 1) \cdot [\, -XL - EL \cdot (1 - YL) \,] \cdot dRP \qquad 7.2$$

The transition from 2.2 to 6.2 is causing an unsteadiness, due to the neglect of the contraction of the slipstream. This is not significantly disturbing within the frame of the introduced simplifications. It appears to be important, however, that by retention of 2.2 for the inner range it is guaranteed that the exact amount of energy flow of the propeller slipstream is transmitted into the vane wheel.

For the propeller itself the velocities and force elements remain the same as under 1.

It must, of course, be granted in all cases that the resulting rotational power of the vane wheel equals to zero.

$$\int dQL = 0 \qquad 8.2$$

Then the optimization can begin and it would be best to begin with the vane wheel. A given propeller slipstream is supposed and the question for the optimum vane wheel is asked. There are two optimization conditions, which can be found in the table 2 on page 10 in the first and fourth places. The first condition, which is formulated analogous to the already known conditions

$$\frac{\delta\,(dTL)}{\delta\,(dQL)} = CL \qquad 9.2$$

guarantees the optimum radial distribution of the velocity induced by the vane wheel. If CL would not be of the same value for all radii of the vane wheel, the thrust produced by the vane wheel could be increased for a constant diameter of the vane wheel, maintaining, of course, condition 8.2, just by changing the induced velocities in such a way that at the end the CL according to 9.2 will be of the same value for all radii.

Equation 9.2 is valid for the turbine- as well as for the propeller part.

For the inner range, where 2.2 is valid for the fluid mass, this condition can still be expressed explicitly.

The velocity induced by the vane wheel is varied:

At first the circular equation 1.2

$$(UE \cdot RP + XL) \cdot \delta XL + YL \cdot \delta YL = 0 \qquad 10.2$$

and then the equations 3.2 and 5.2 for the power elements

$$\delta (dTL) = RP \cdot (YP+1) \cdot [\delta YL \mp EL \cdot XL] \cdot dRP$$
$$\delta (dQL) = RL \cdot RP \cdot (YP+1) \cdot [-\delta XL \mp EL \cdot \delta YL] \cdot dRP \qquad 11.2$$

If 11.2 is inserted in 9.2 and afterwards XL and YL eliminated by means of 10.2, the following is obtained

$$CL = \frac{-(RL+XL) \mp EL \cdot YL}{RL \cdot -YL \pm EL \cdot (RL+XL)} \qquad 12.2$$

or

$$\frac{RL+XL}{YL} = \frac{RL \cdot CL \mp EL}{1 \pm EL \cdot RL \cdot CL} \qquad 13.2$$

RL means = UE·RP

If 13.2 together with the equation of the circle 1.2 - applying the upper signs of the friction terms - result in a value XL, which is greater than XP, one is in the turbine part and the thus computated coordinates XL and YL remain valid. If with the lower signs a value XL, less than XP, is obtained, one is in the propeller part, and the thus computated coordinates remain valid. There is, however, a gap for which both conditions are not valid; then one is neither in the turbine- nor in the propeller part but in the unloaded gap.

Then

$$XL = XP \quad \text{and} \quad YL = YP \qquad 14.2$$

are set and the optimization condition 9.2 is loosing its sense for this part.

Herewith the discussion of the first optimization condition is concluded.

Now the condition will be discussed, as shown in Table 2 on place 2, which is resulting from the variation of the velocity induced by the propeller and leading to the optimum radial distribution of this velocity.

At first, the equations 4.1 to 6.1 can be adopted from 1.

Only a P shall be added to each of the symbols T and Q in 5.1 and 6.1 in order to be able to recognize that these forces and moments are directly generated by the propeller. The forces at the vane wheel are changing, of course, by a change of the velocity induced by the propeller. Thus the variation of the thrust element according to 3.2 due to a variation of the propeller induced velocity amounts to

$$\delta\,(dTL) = RP \cdot \left[\delta YP \cdot (YL - YP \mp EL(XL - XP)) + (YP+1) \cdot (\delta YL - \delta YP \mp EL(\delta XL - \delta XP))\right] \cdot dRP \qquad 15.2$$

and analogous the variation of the torque according to 5.2, where δXP and δYP are a direct consequence of the variation of the induced velocity, whereas δXL and δYL result directly from condition 9.2 and can be computed from 1.2 and 13.2. The total thrust alteration consists of three parts

$$\delta\,(dT) = \delta\,(dTP) + \delta\,(dTL) - Cl \cdot \delta\,(dQL) \qquad 16.2$$

the direct change at the propeller itself $\delta\,(dTP)$
the indirect change at the vane wheel $\delta\,(dTL)$

and the change of the torque produced at the vane wheel, which - according to 9.2 - is transformed in the propeller part into the thrust $-CL \cdot \delta(dQL)$

The torque produced at the propeller is also changed according to 6.1. Analogous to the above considerations, the optimum distribution of the velocity induced by the propeller is obtained if CP in

$$\frac{\delta\,(dT)}{\delta\,(dQP)} = \frac{\delta\,(dTP) + \delta\,(dTL) - CL \cdot \delta\,(dQL)}{\delta\,(dQP)} = CP \qquad 17.2$$

is the same for all radii of the propeller.

A further optimization condition is obtained when the diameter of the propeller or the diameter ratio is varied at a constant total diameter of the vane wheel.

If, for instance, the radius of the propeller is decreased by δRP

$$\delta TP,\ \delta QP,\ \delta TL_1,\ \delta QL_1 \qquad 18.2$$

are omitted.

The first two values are computed directly from 2.1 and 3.1, the last two from 3.2 and 5.2, whereby in all formulae dRP is replaced by δ RP and all further values have to be inserted for the radius of the propeller. The outer range of the vane wheel thus increases by the amount δ RP and the values

$$\delta TL_2 = RP \cdot (YL+1) \cdot [YL-1+EL \cdot XL] \cdot \delta RP \qquad 19.2$$

$$\delta QL_2 = RP \cdot RL(YL+1) \cdot [-XL-EL \cdot (1-YL)] \cdot \delta RP \qquad 20.2$$

are added.

In total the resulting change of the thrust is obtained

$$\delta T = \delta(TP) + \delta(TL)_1 - \delta(TL)_2 - CL \cdot [\delta(QL)_1 - \delta(QL)_2], \qquad 21.2$$

where the values of the vane wheel according to 18.2 are designated with the index 1. The change of the torque at the propeller is also obtained. Obviously, also for this variation

$$\frac{\delta T}{\delta QP} = CP \quad , \qquad 22.2$$

is valid, since otherwise an increase or decrease of the resulting thrust would be obtained by supplying the torque to other radii of the propeller according to 17.2, whereas in the optimum case no change may be obtained in this way.

All optimization conditions are herewith discussed with the exception of the ratio CP:CL and of the ratio of revolutions. But these were done by trial and error.

In practice the computations are carried out as follows:

Four values are arbitrarily presumed:
UE - the transmission ratio
CP - the constant according to 17.2 and 22.2 23.2
CL - the constant according to 9.2

and a radius RP for the blade tips of the propeller.

Herewith the computations can be carried out. The total radius of the vane wheel then results from demand 8.2. The conditions of Table 2, places 3 and 4, are not yet fulfilled. The computations are repeated for different radii RP

of the blade tips of the propeller under retention of the first three values 23.2, until condition 22.2 (Table - place 3) is fulfilled. Then only the conditions concerning place 4 and 5 of the table are not explicitly fulfilled. They can be fulfilled if computations are also carried out for different ratios CL:CP and different transmission ratios UE.

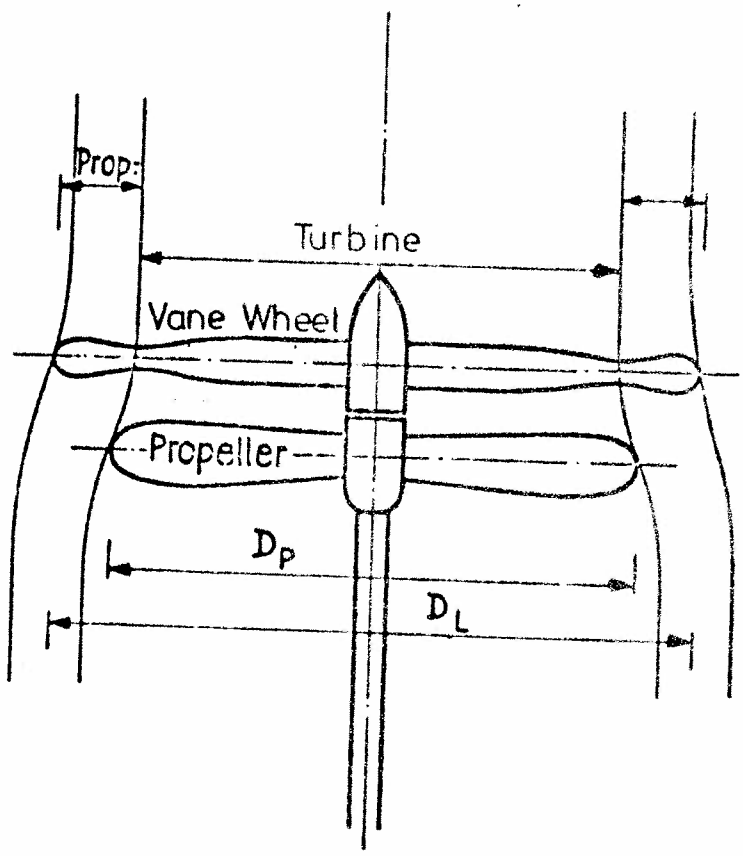


Fig. 1 Propeller and rotating vane wheel

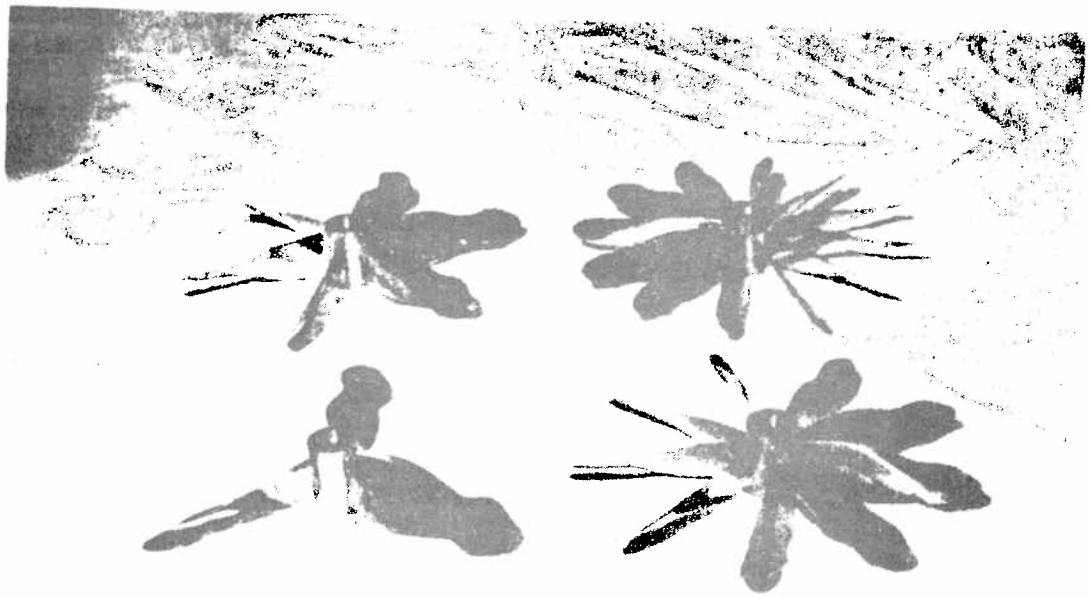


Fig. 2 Vane wheels with 3; 6; 9; 12 blades

Fig. 3

V Speed of the flow, incoming to the unit

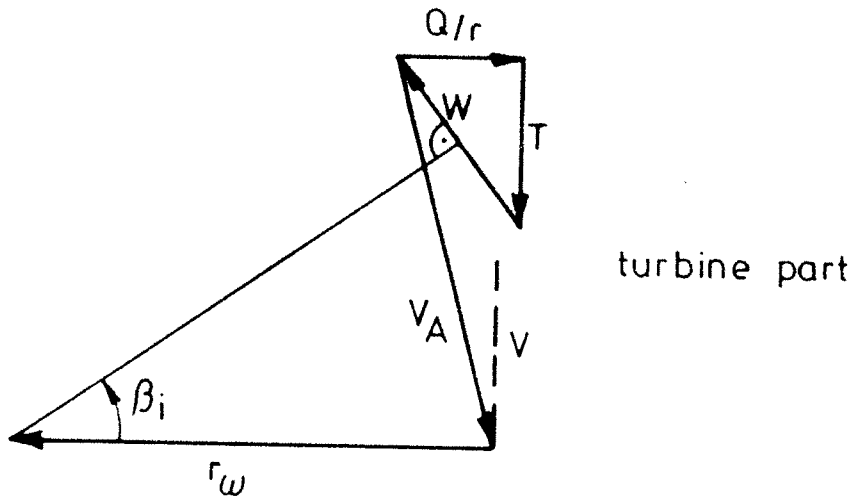
V_A Speed, incoming to the vane wheel, that means for the turbine part speed in the slip stream of the propeller

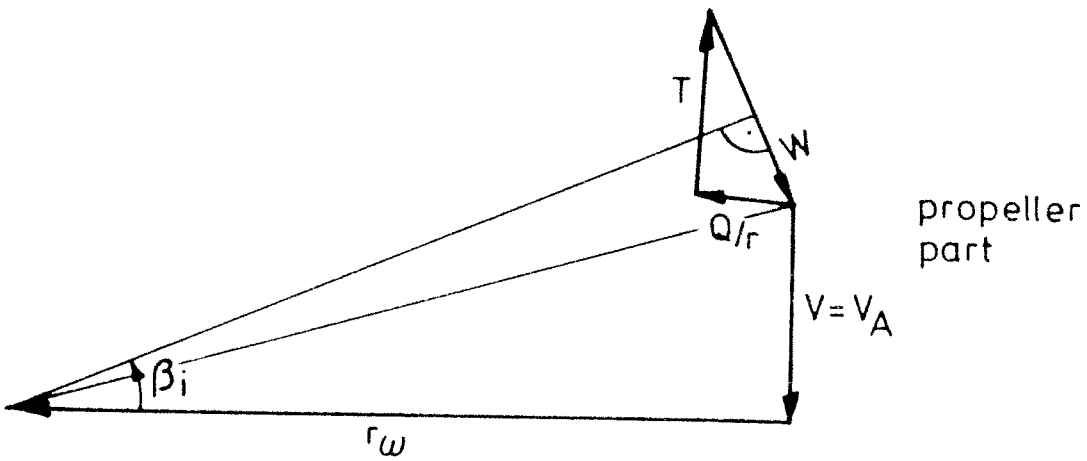
$r\omega$ Circumferencial speed

W Induced speed

T Thrust

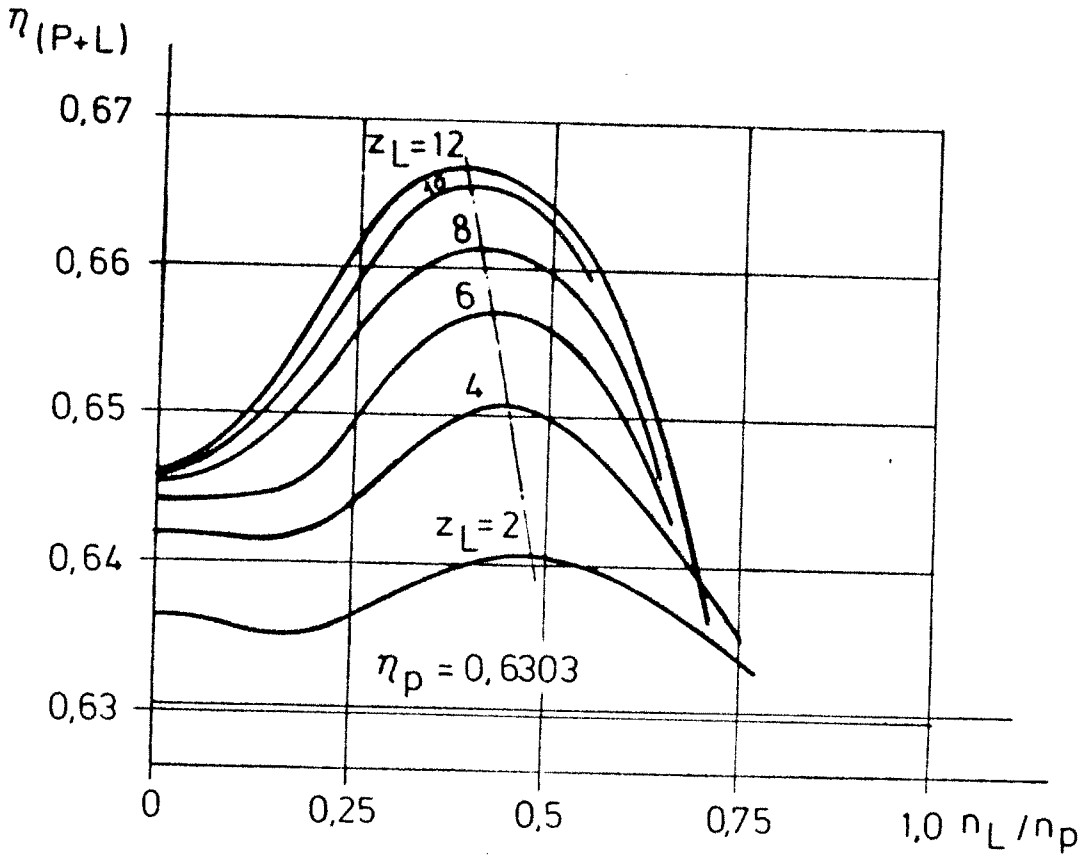
Q Torque





$$\frac{Q/r}{T} = \tan \beta_i$$

Fig. 4



Influence of the number of revolutions and blade number on the efficiency.

(Index L means vane wheel)

Fig. 5

Open - water test

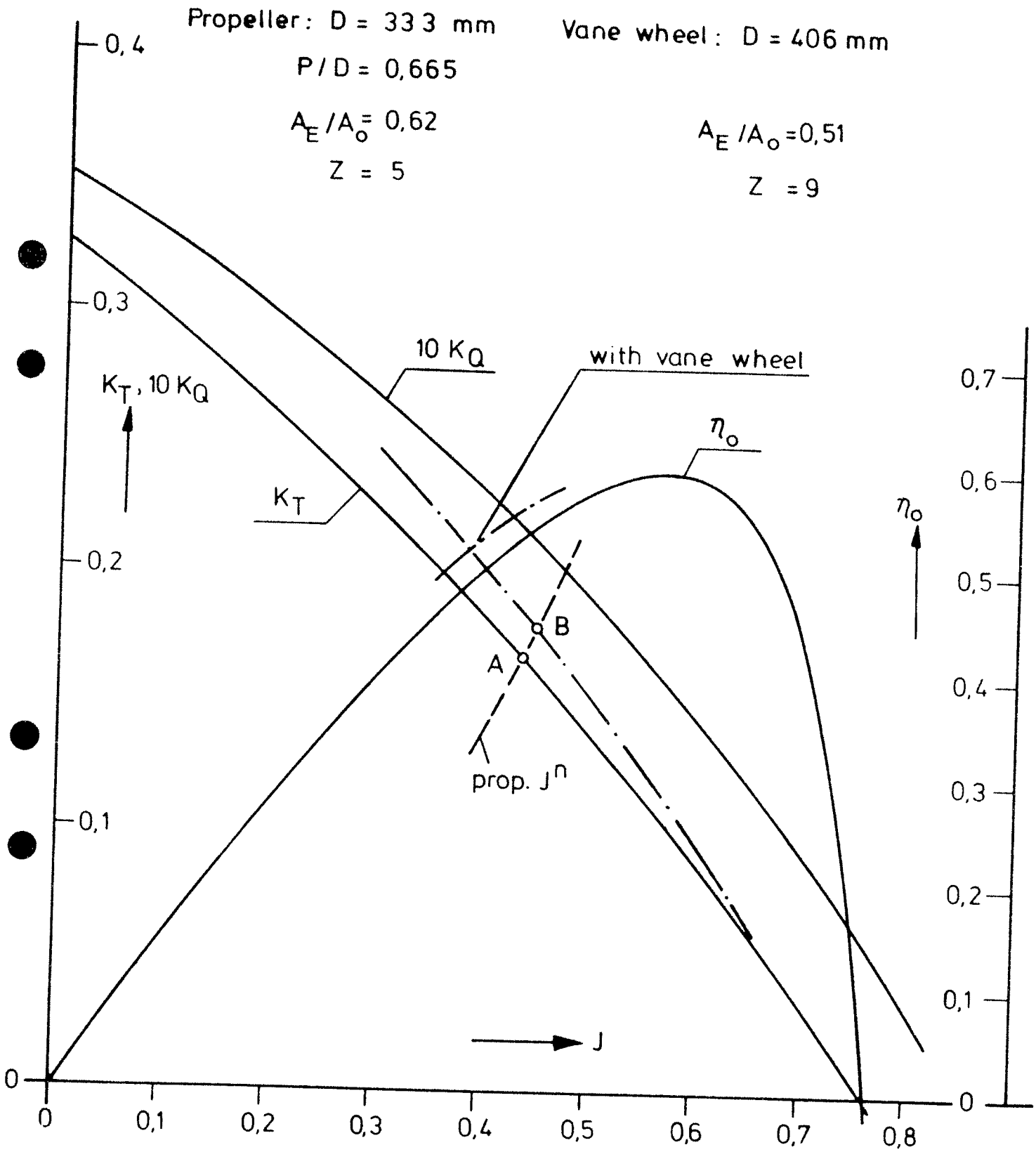
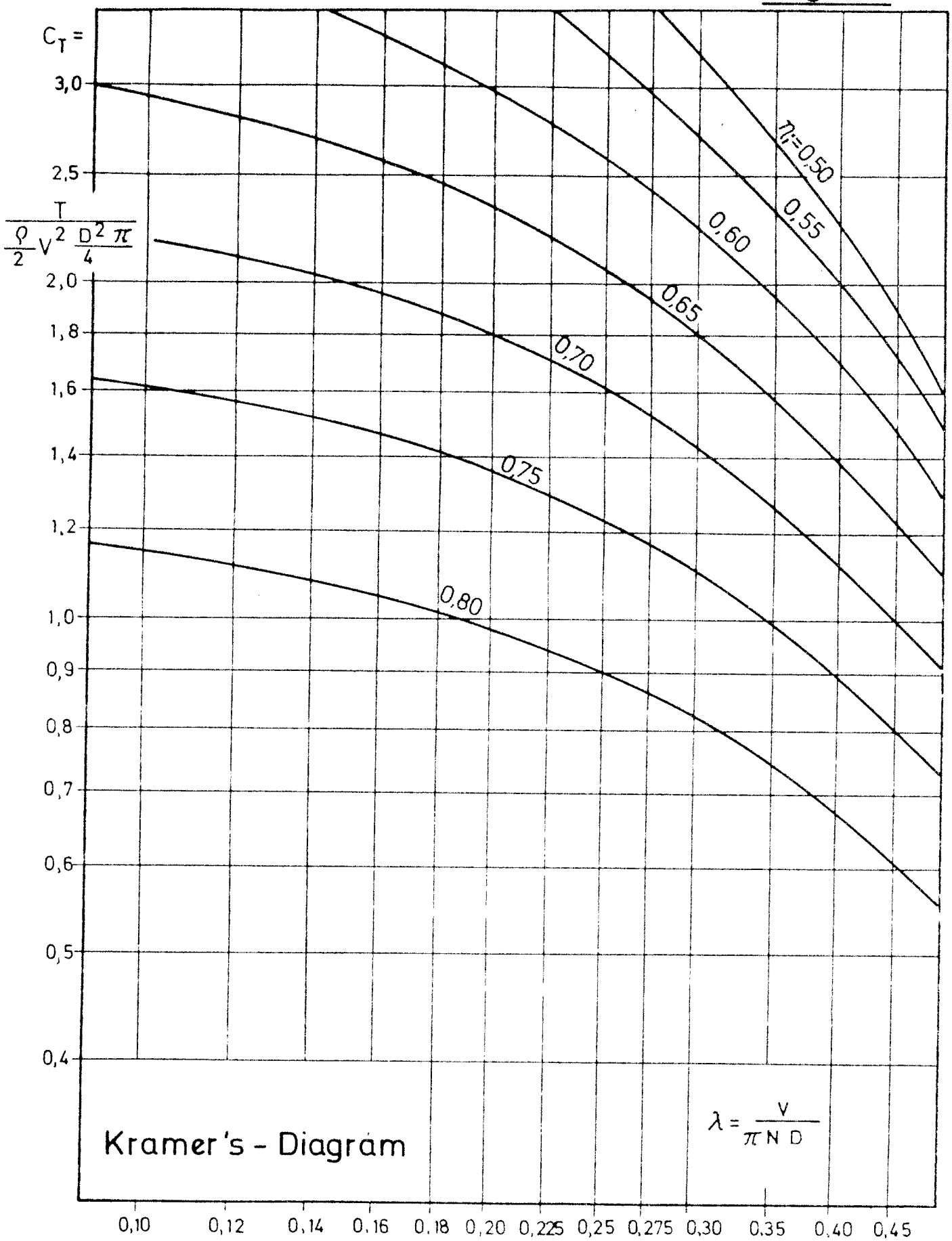


Fig. 6



Kramer's - Diagram

Fig. 7

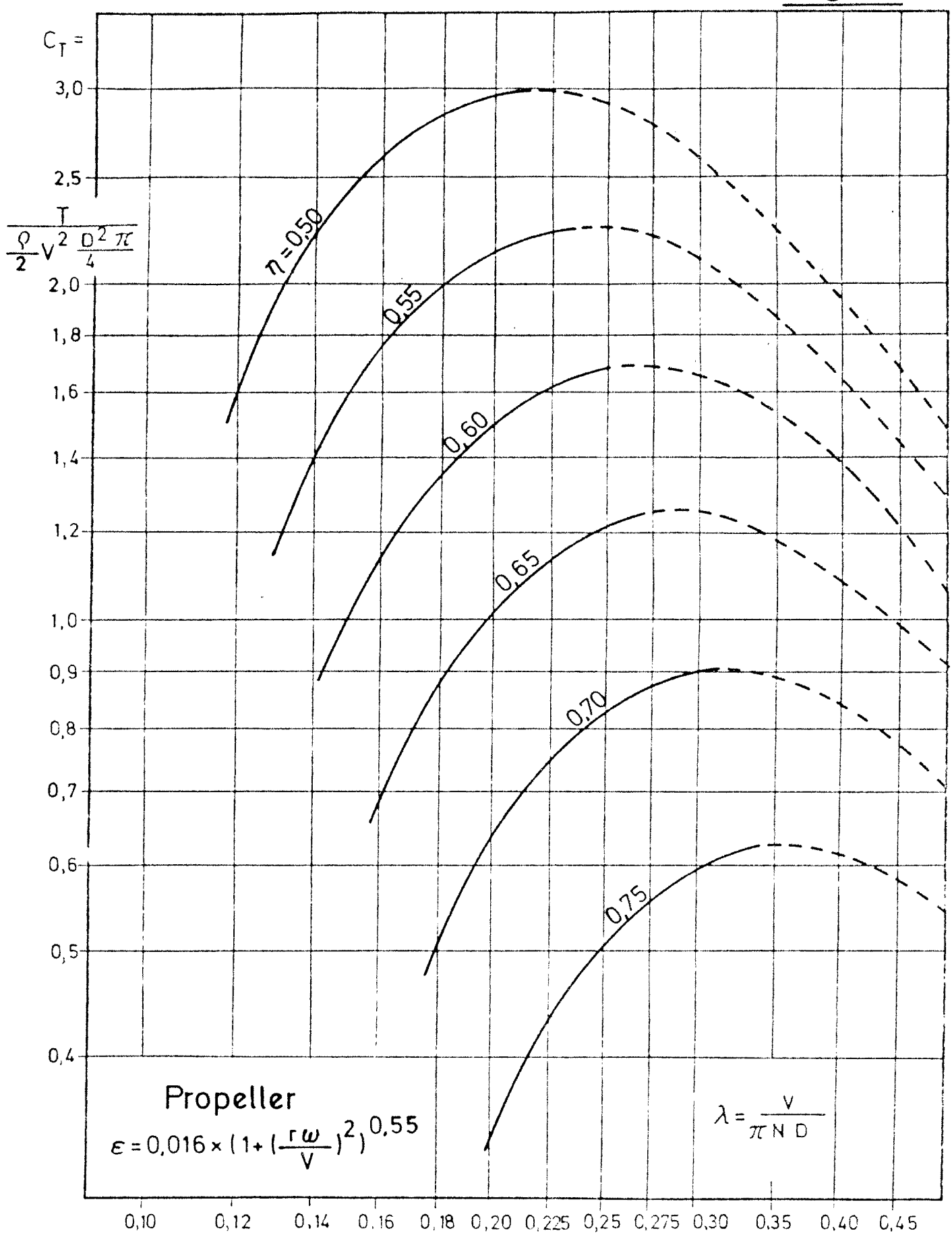

$C_T = \frac{T}{\frac{\rho}{2} V^2 \frac{D^2 \pi}{4}}$
3,0
2,5
2,0
1,8
1,6
1,4
1,2
1,0
0,9
0,8
0,7
0,6
0,5
0,4
η = 0,50
0,55
0,60
0,65
0,70
0,75
Propeller
$\varepsilon = 0{,}016 \times \left(1 + \left(\frac{r\omega}{V}\right)^2\right)^{0{,}55}$
$\lambda = \frac{V}{\pi N D}$
0,10
0,12
0,14
0,16
0,18
0,20
0,225
0,25
0,275
0,30
0,35
0,40
0,45

Fig. 8

Propeller and vane wheel $\eta = 0{,}70$
CT and λ related to the propeller diameter

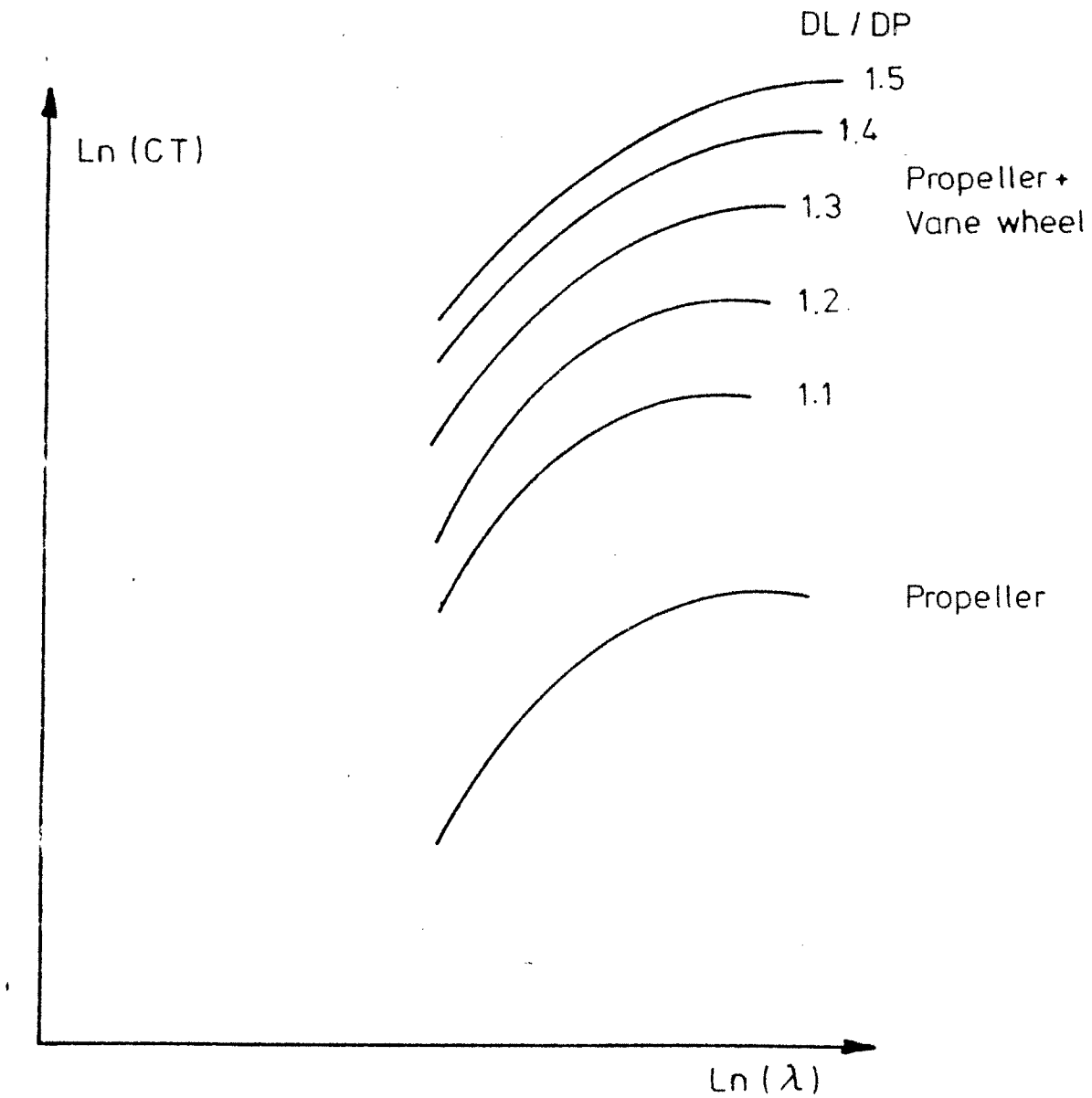


Fig. 9

Propeller and vane wheel $\eta = 0{,}70$
CT and λ related to the diameter of the vane wheel

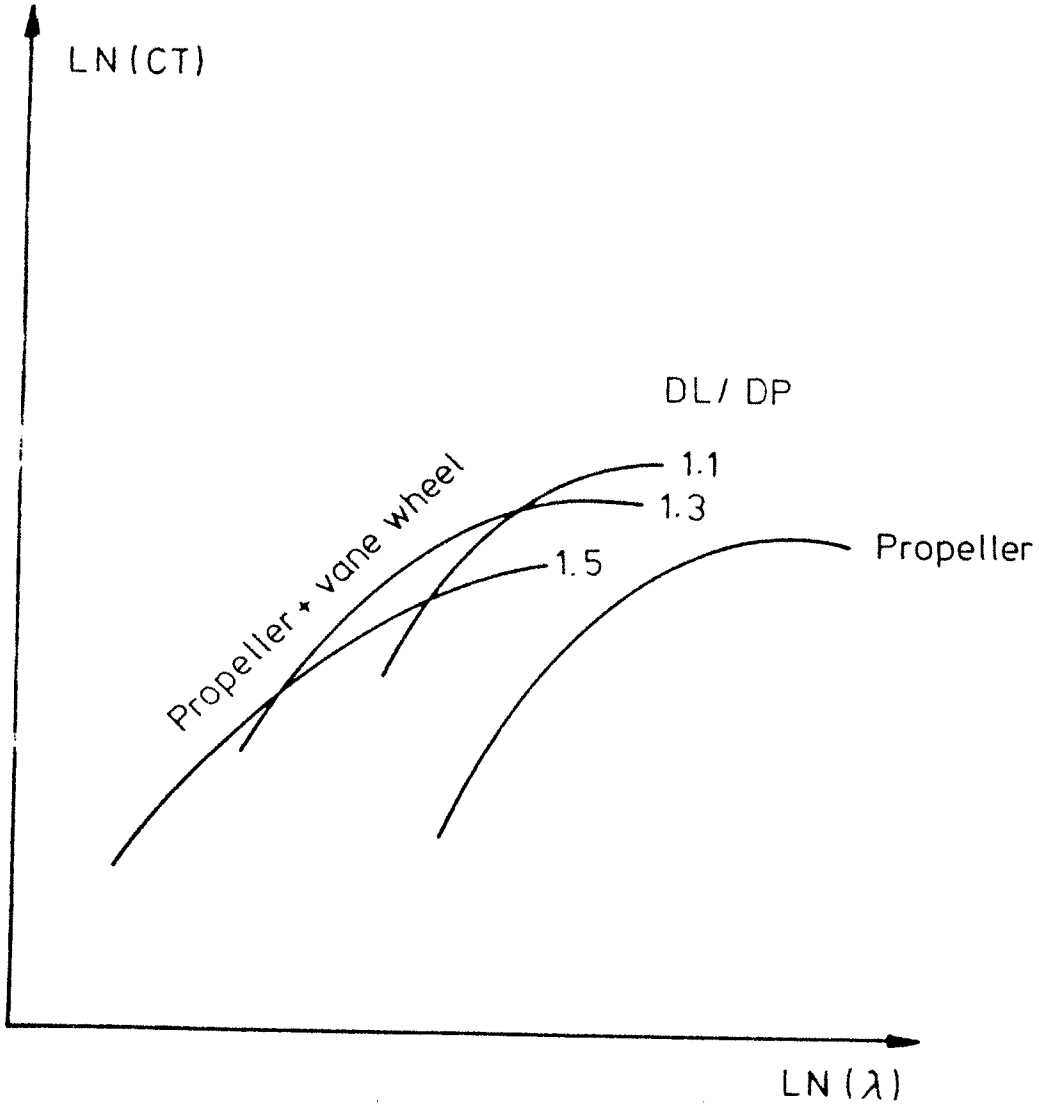
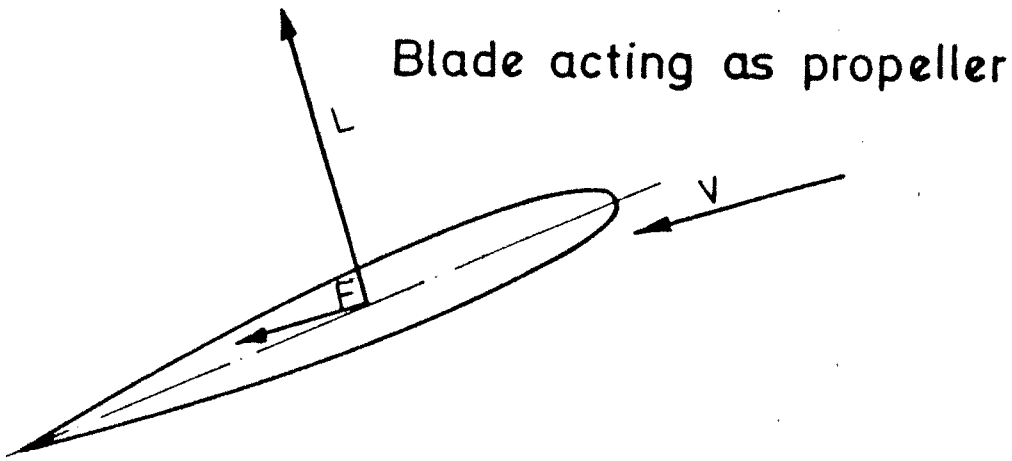
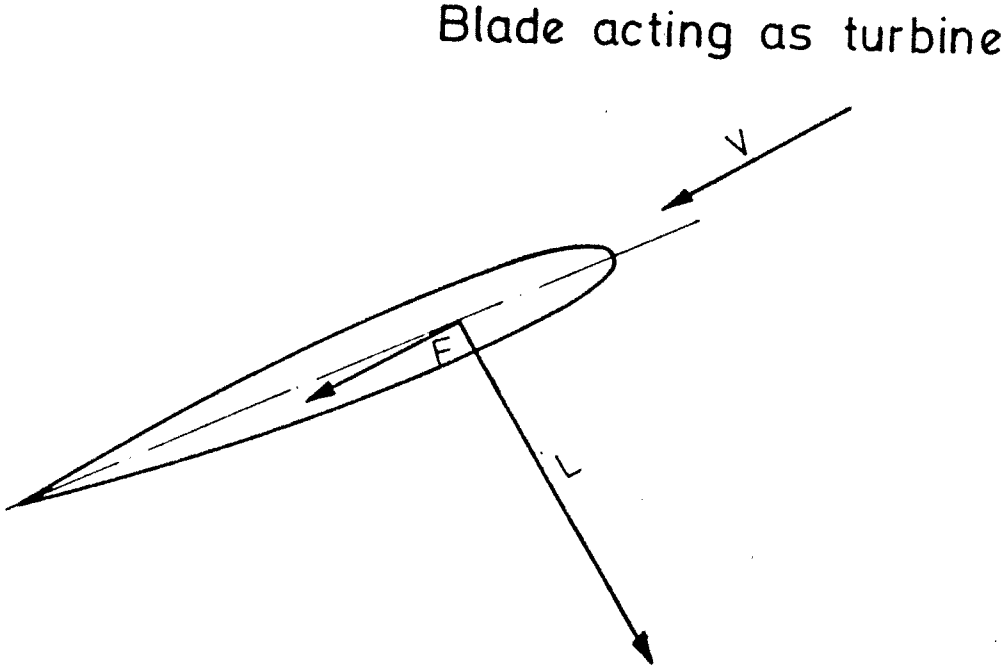


Fig. 10





L = Lift

F = Drag

Fig. 11

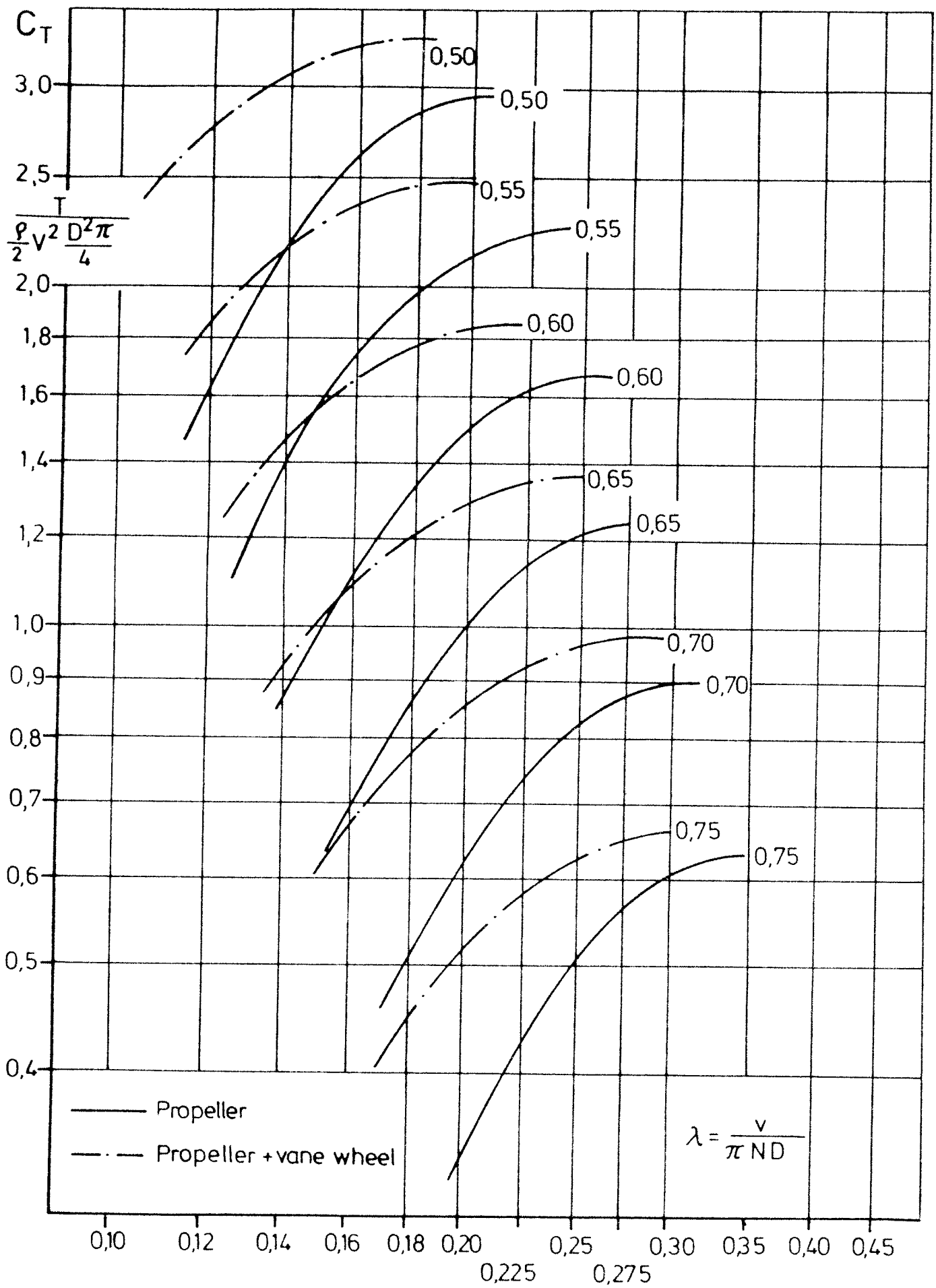
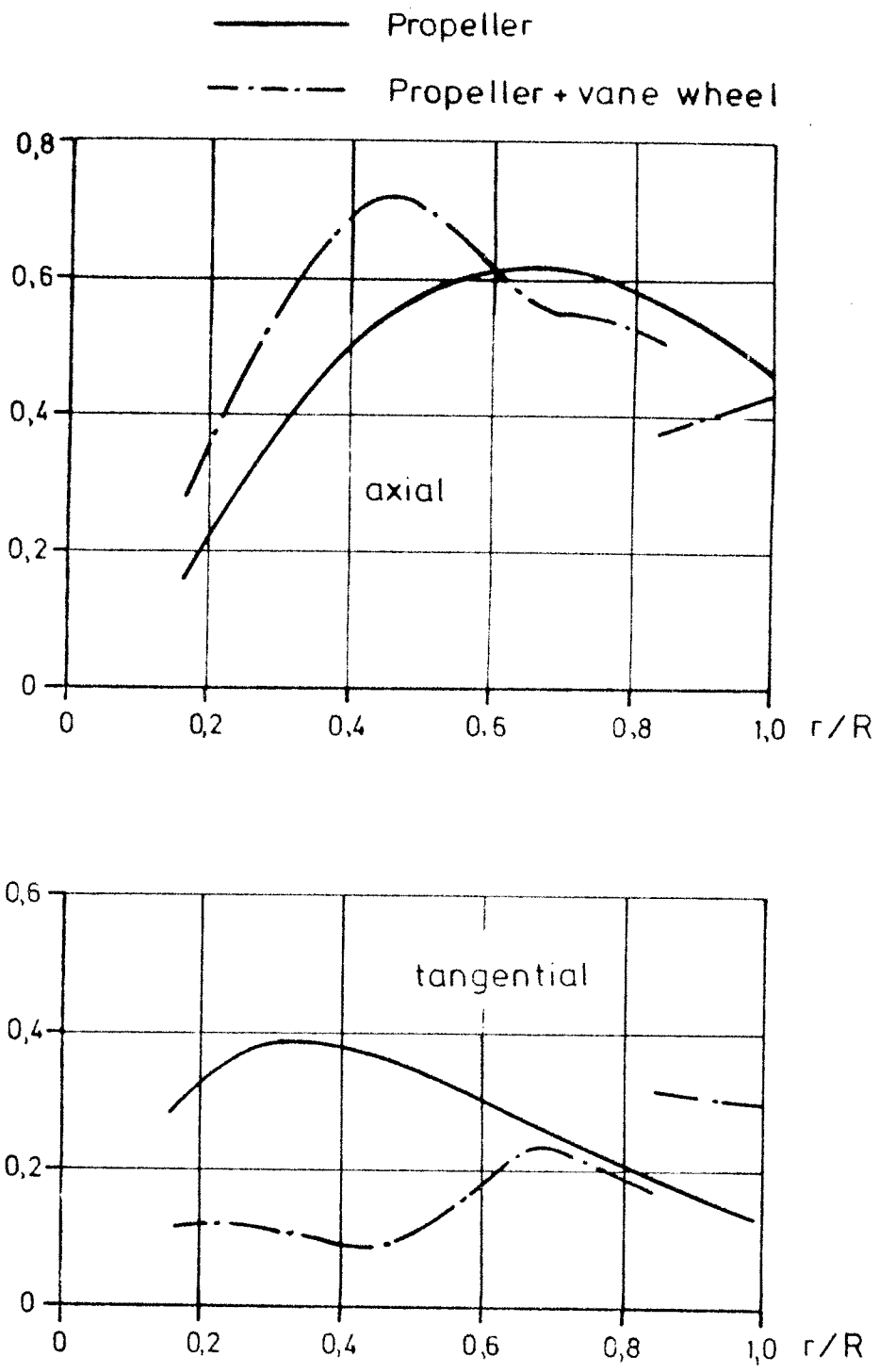

C_T
$\frac{T}{\frac{\rho}{2} V^2 \frac{D^2 \pi}{4}}$
3,0
2,5
2,0
1,8
1,6
1,4
1,2
1,0
0,9
0,8
0,7
0,6
0,5
0,4
0,50
0,50
0,55
0,55
0,60
0,60
0,65
0,65
0,70
0,70
0,75
0,75
Propeller
Propeller + vane wheel
$\lambda = \frac{v}{\pi N D}$
0,10
0,12
0,14
0,16
0,18
0,20
0,225
0,25
0,275
0,30
0,35
0,40
0,45

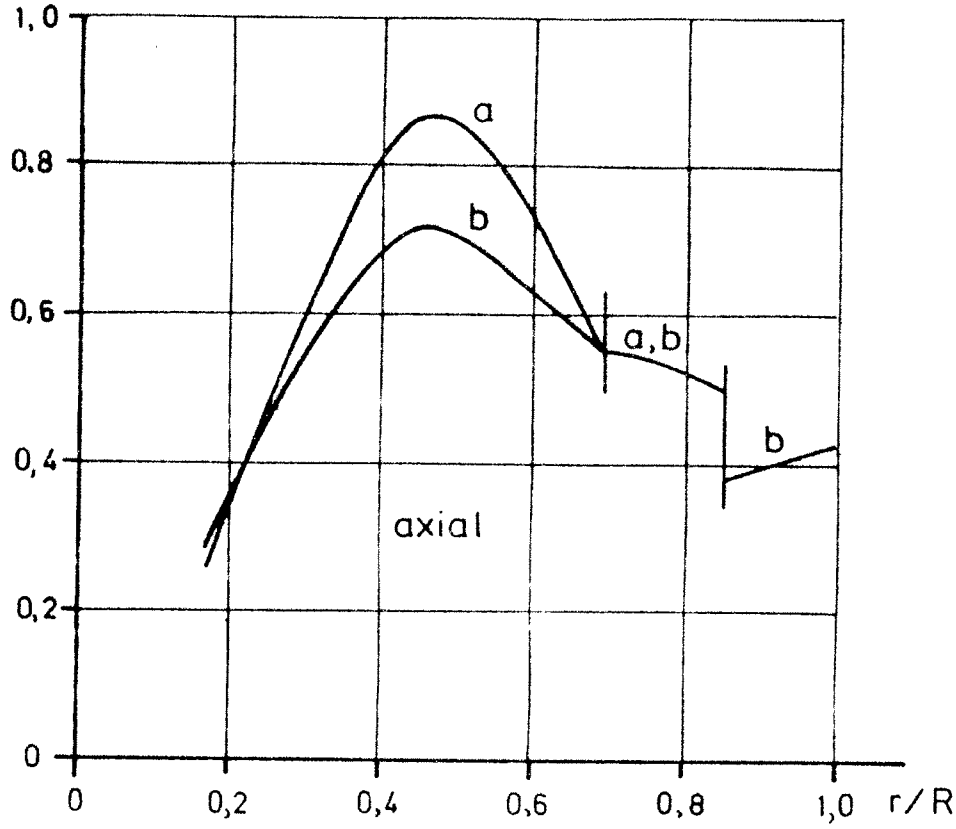
Fig. 12

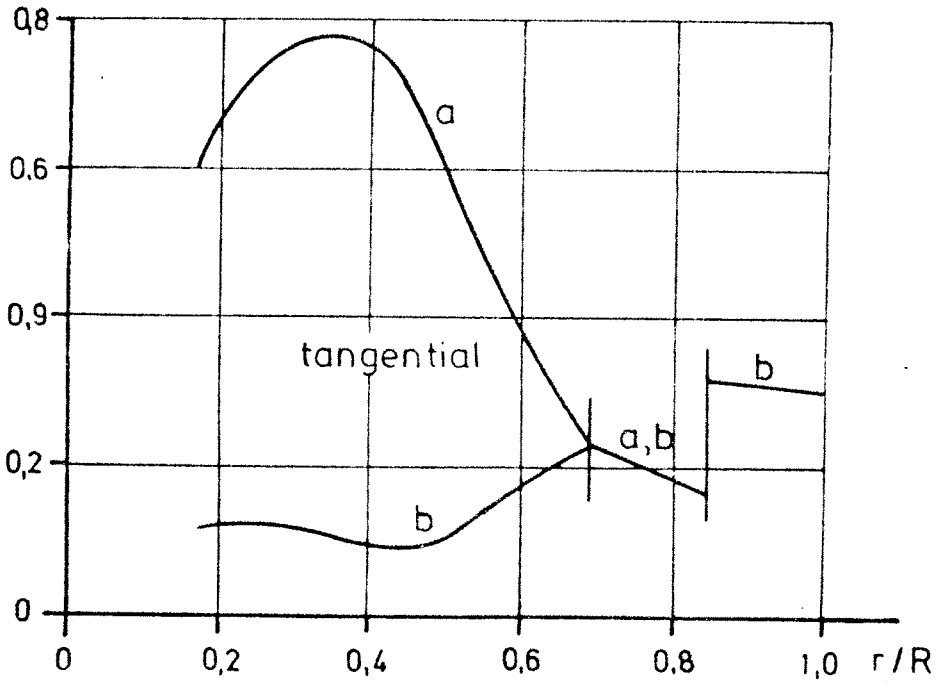


Velocities induced within the slipstream far behind the propulsion devices

$C_T = 1{,}32$ $\lambda = 0{,}212$

Fig. 13

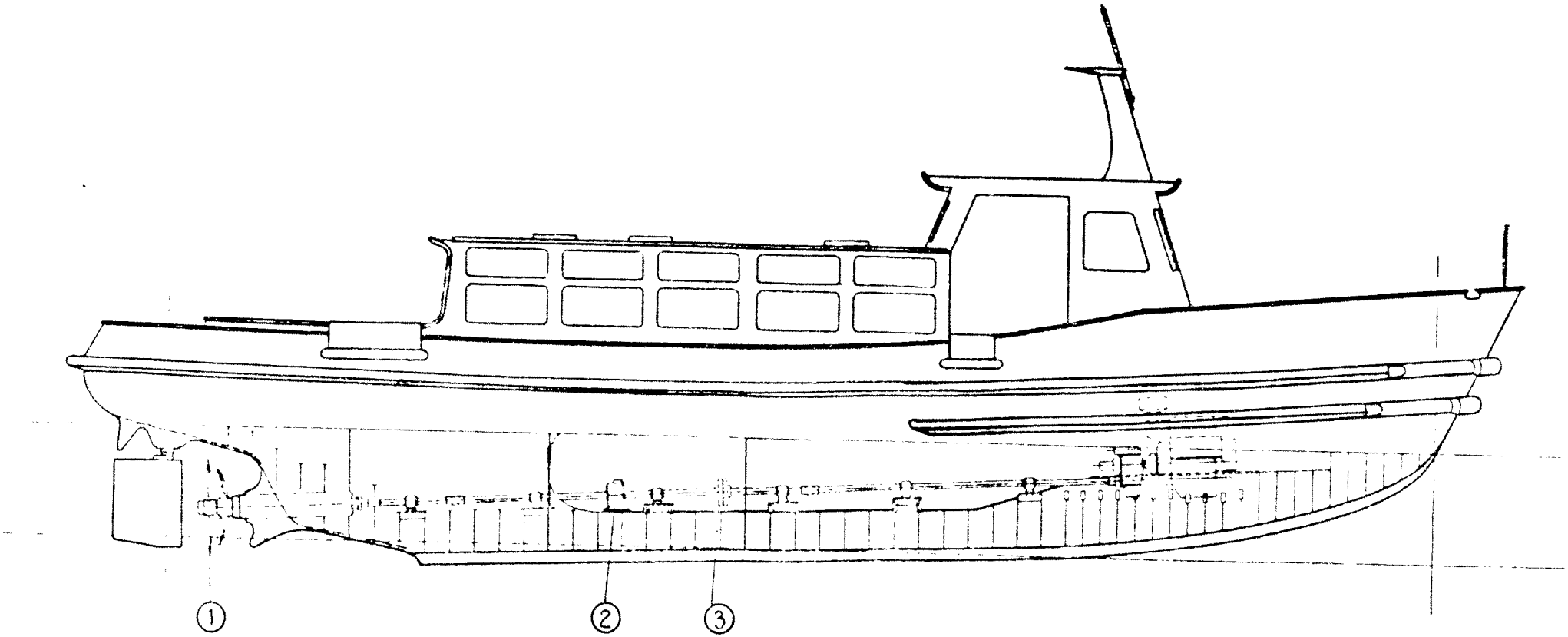




Velocity induced by propeller and vane wheel

a) between propeller and vane wheel

b) behind the vane wheel



Launch "HDW 1" with propeller and vane wheel

① = vane wheel ② = SIMPLEX-thrust meter

③ = MAIHAK-torque meter

Fig. 14

Fig. 15

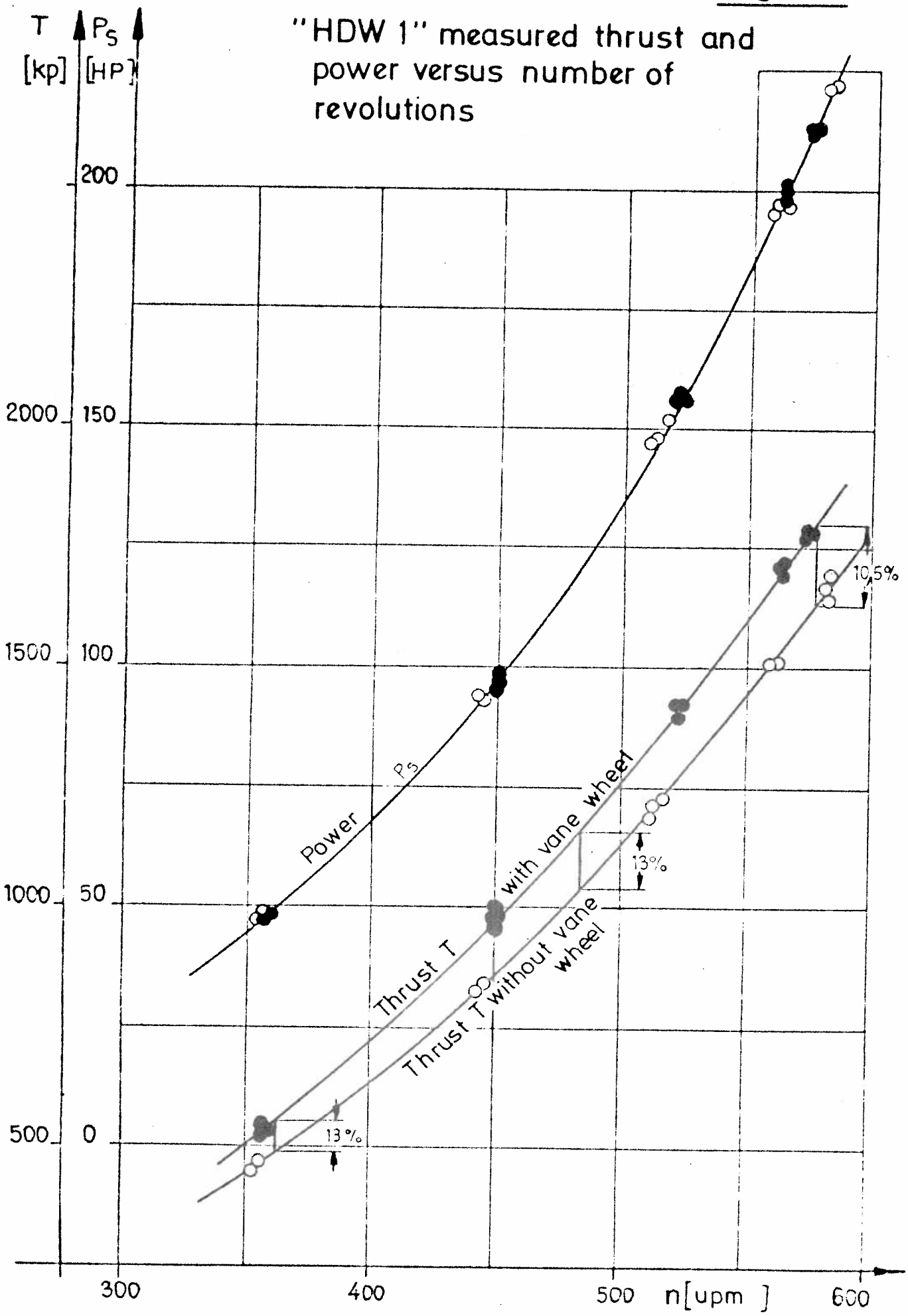

"HDW 1" measured thrust and power versus number of revolutions
T [kp]
P_S [HP]
2000
1500
1000
500
200
150
100
50
0
300
400
500
600
n[upm]
Power P_S
Thrust T with vane wheel
Thrust T without vane wheel
13%
13%
10,5%

Fig. 16

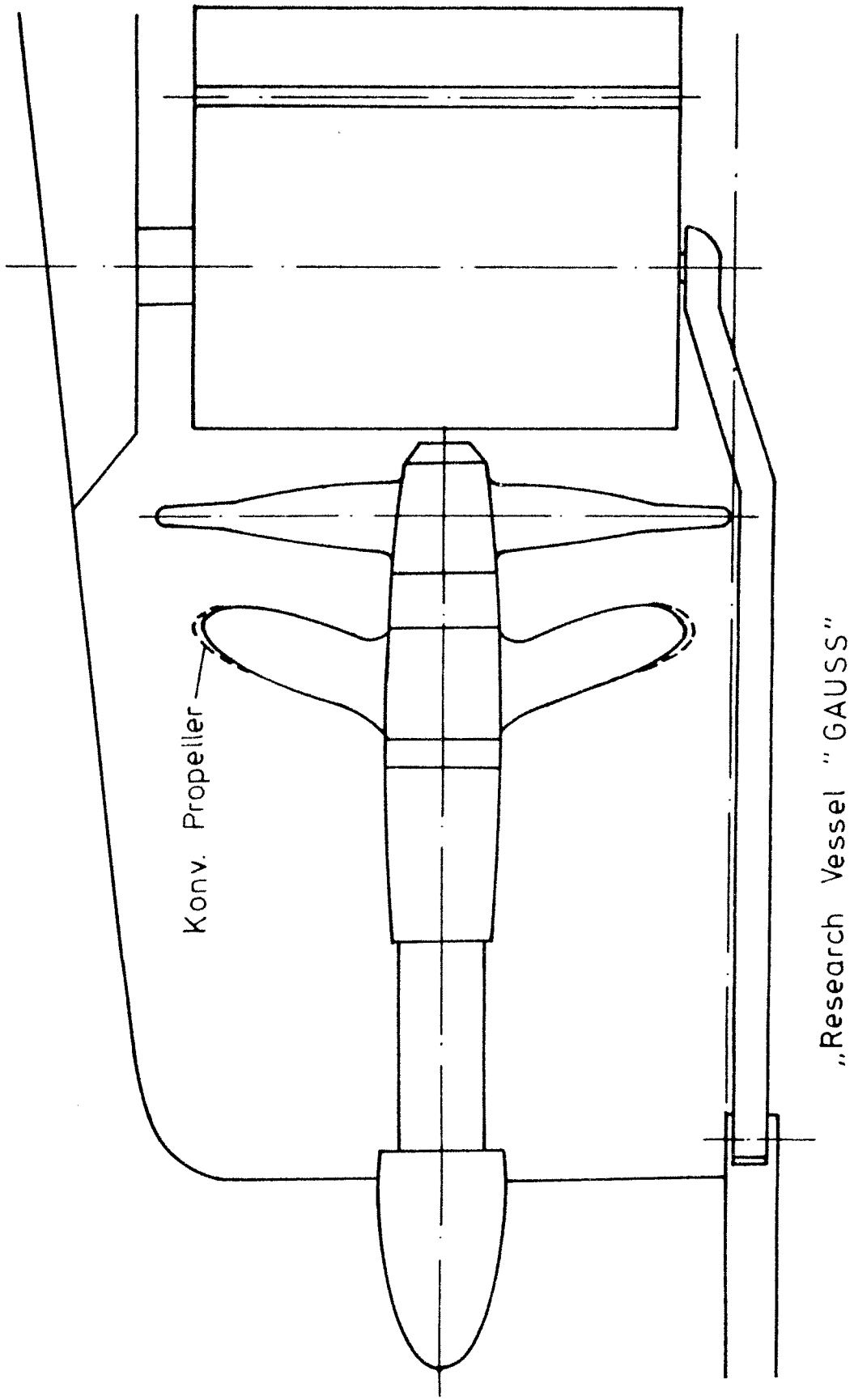


Fig. 17

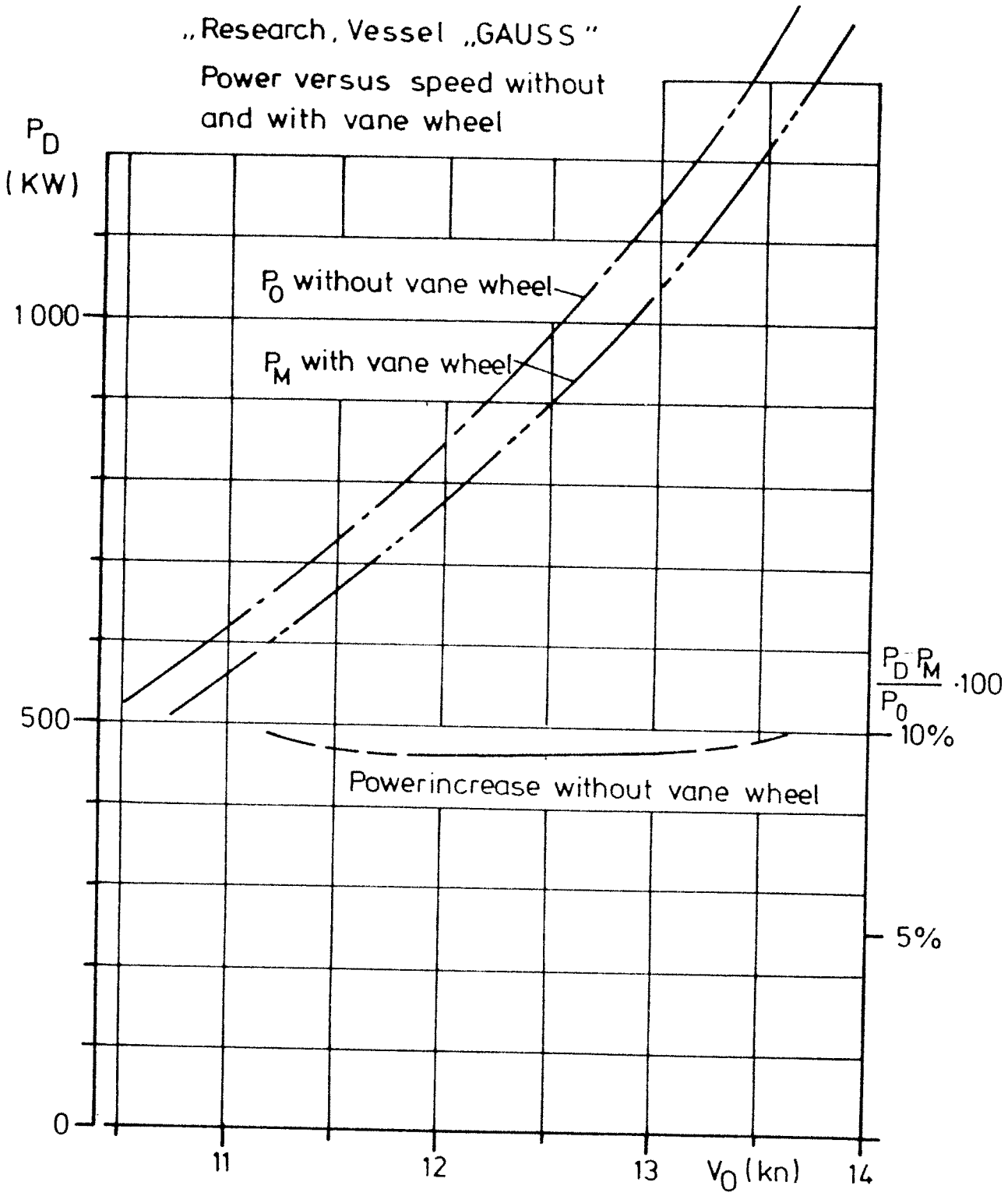
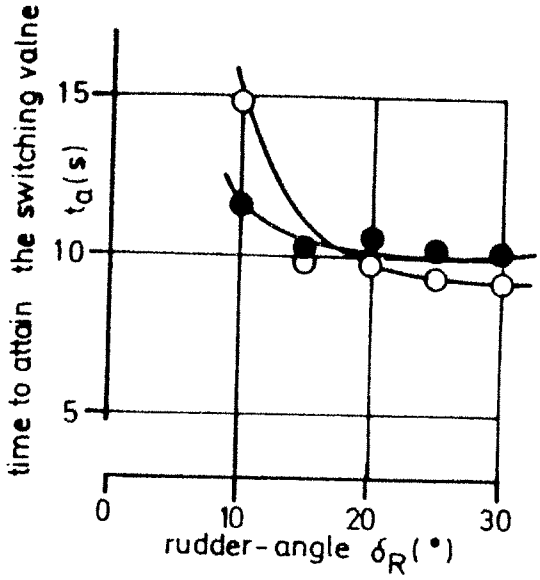

„Research, Vessel „GAUSS"
Power versus speed without
and with vane wheel
P_D
(KW)
1000
500
0
P_0 without vane wheel
P_M with vane wheel
$\frac{P_D - P_M}{P_0} \cdot 100$
10%
5%
Powerincrease without vane wheel
11
12
13
V_0 (kn)
14

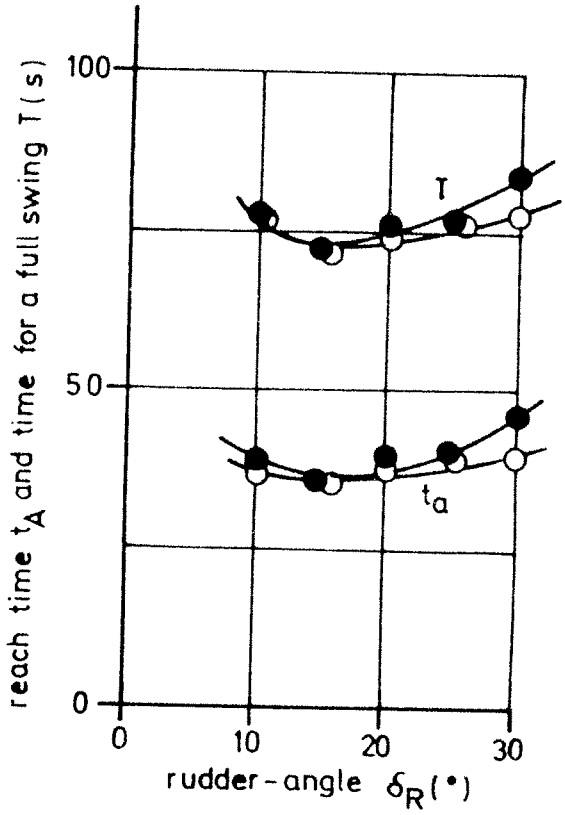
Fig. 18



Standard Manoeuvring Test

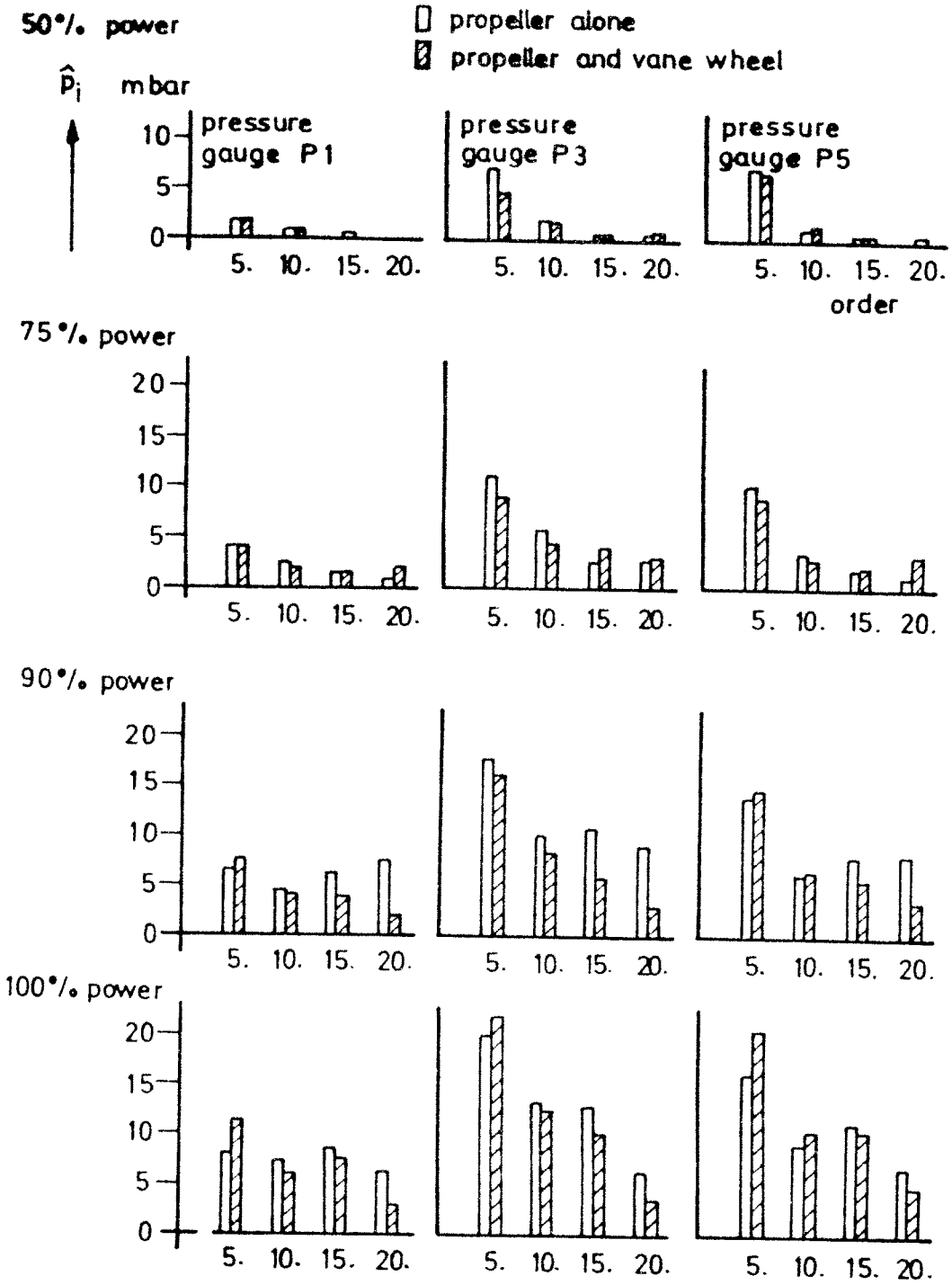
time to attain the switching value t_a (s) as funktion of rudder – angle δ_R (°)

● without vane wheel
○ with vane wheel



reach time t_A (s) and time for a full swing T (s) as funktion of rudder – angle δ_R (°)

Fig. 19



Harmonic components of the 5th, 10th, 15th and 20th order

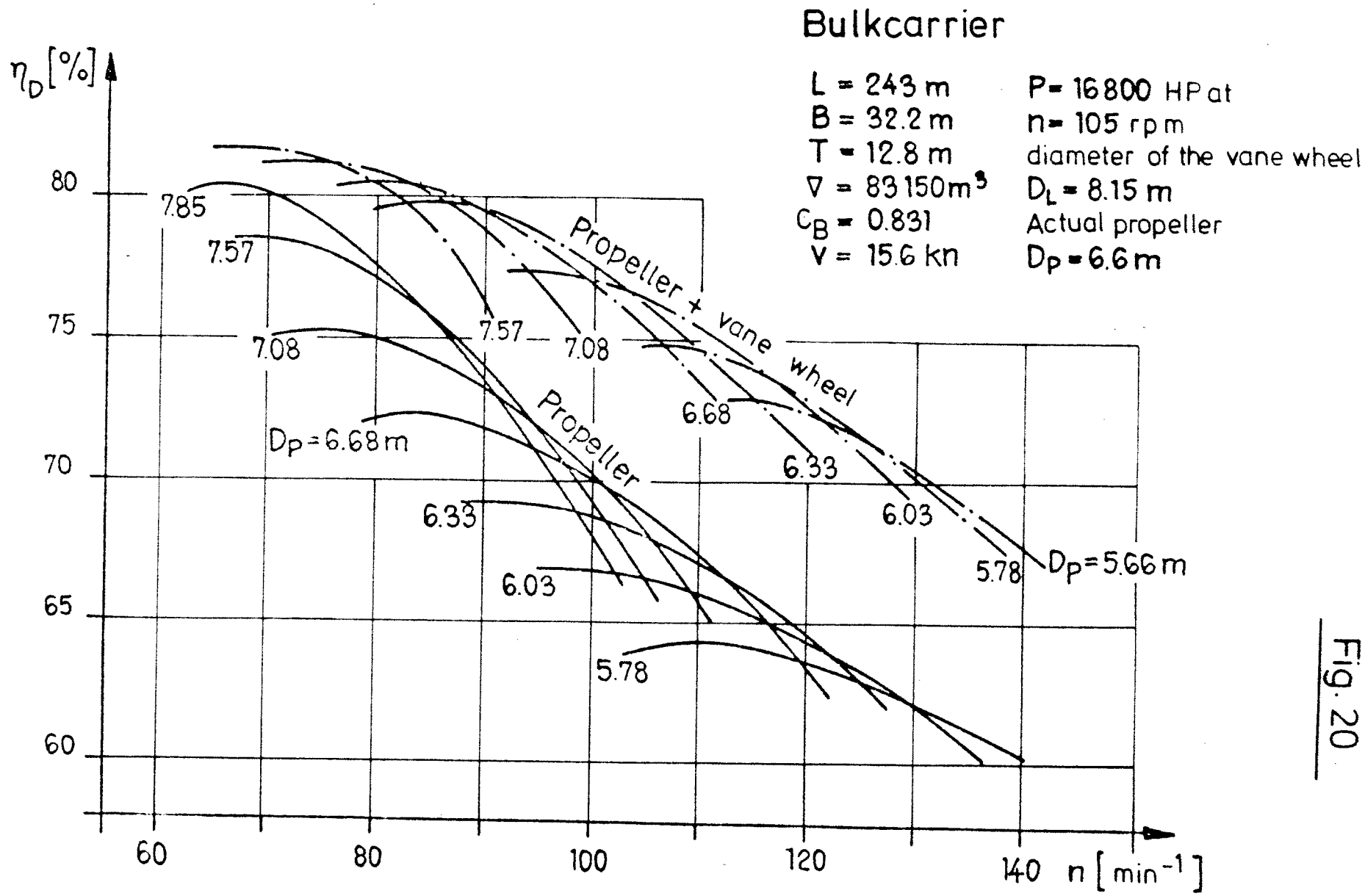
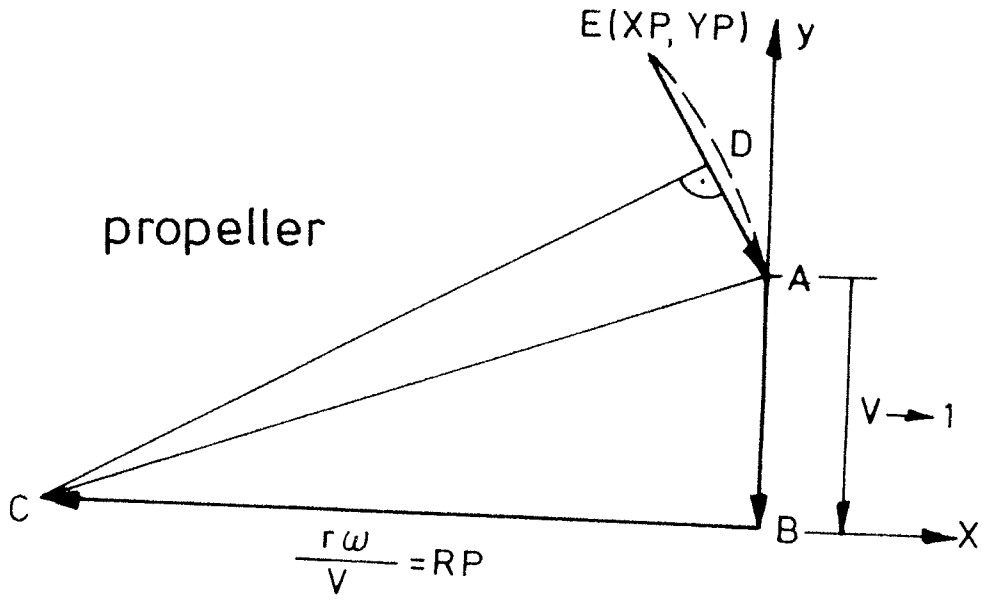
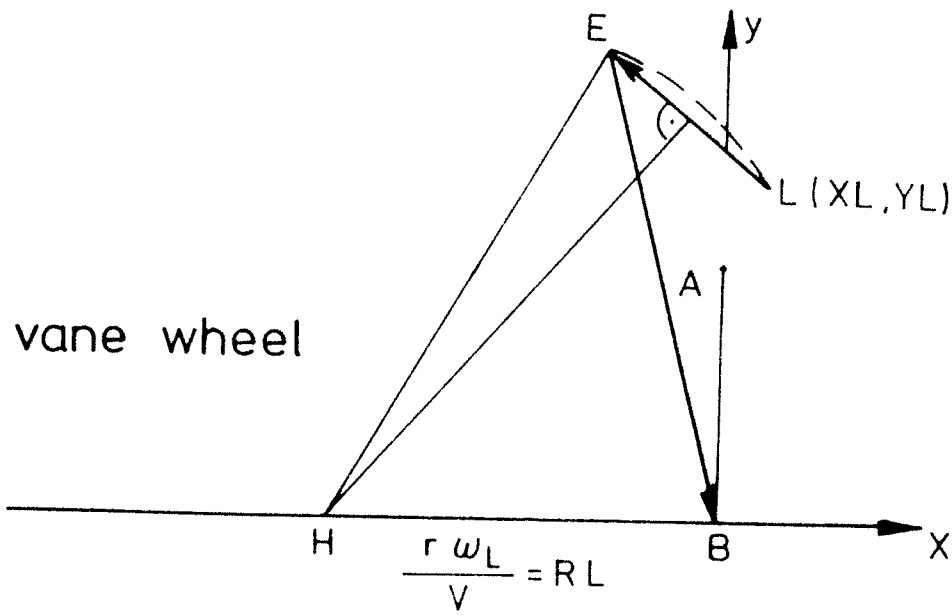

Bulkcarrier
L = 243 m
B = 32.2 m
T = 12.8 m
∇ = 83150 m³
C_B = 0.831
V = 15.6 kn
P = 16800 HP at
n = 105 rpm
diameter of the vane wheel
D_L = 8.15 m
Actual propeller
D_P = 6.6 m
η_D [%]
80
75
70
65
60
60
80
100
120
140
n [min⁻¹]
Propeller + vane wheel
Propeller
7.85
7.57
7.08
Dp = 6.68 m
7.57
7.08
6.68
6.33
6.03
5.78
Dp = 5.66 m
6.33
6.03
5.78

Fig. 20

Fig 21


E(XP, YP)
y
D
propeller
A
V → 1
C
B
X
$\frac{r\omega}{V} = RP$


E
y
L (XL, YL)
A
vane wheel
H
B
X
$\frac{r\omega_L}{V} = RL$

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

PROPELLERS WITH REDUCED TIP LOADING AND UNCONVENTIONAL BLADE FORM AND BLADE SECTIONS

by

Carl-Anders Johnsson
Swedish Maritime Research Centre SSPA
Gothenburg

PROPELLERS WITH REDUCED TIP LOADING AND UNCONVENTIONAL BLADE FORM AND BLADE SECTIONS

by

Carl-Anders Johnsson, SSPA

Summary

The importance of using adequate methods when designing propellers with unconventional load distribution and blade form is emphasized and illustrated by results of design calculations. The main features of the design methods are briefly discussed. Using results of calculations and experiments the magnitude of the influence of tip unloading, blade form and shape of blade sections on efficiency, vibration and noise excitation, cavitation damage and strength is discussed. The experimental technique and analysis used when determining these properties are briefly discussed and finally further possibilities of optimizing propellers are discussed.

1. Introduction

The following design conditions are the most important ones to be fulfilled, when designing a propeller for a ship:

1) High efficiency
2) Adequate strength
3) No cavitation damage
4) Low vibration and noise excitation

One difficulty in reaching the best compromise is that, when trying to improve the design in one respect, the result is very often bad behaviour in some other respect. Thus the present emphasis on high efficiency has sometimes led to propellers showing unacceptable bahaviour as regards cavitation damage, vibration and noise excitation. This is still more to be regretted as the possibilities of affecting the properties of the design have lately increased by the introduction of modern propeller design methods, based on vortex theory.

In spite of their complexity also the modern design methods

have however their shortcomings and experimental verification is necessary to check that the desired properties of the propeller are obtained.

Below some features of modern propeller design methods will be breifly described and some examples given on the advantages obtained, when using them in the design work. Further, the experiments relevant in this context will be discussed, the design and experiment work being illustrated by results of a series of propellers for which some design parameters were varied.

2. Propeller design methods, some basic concepts

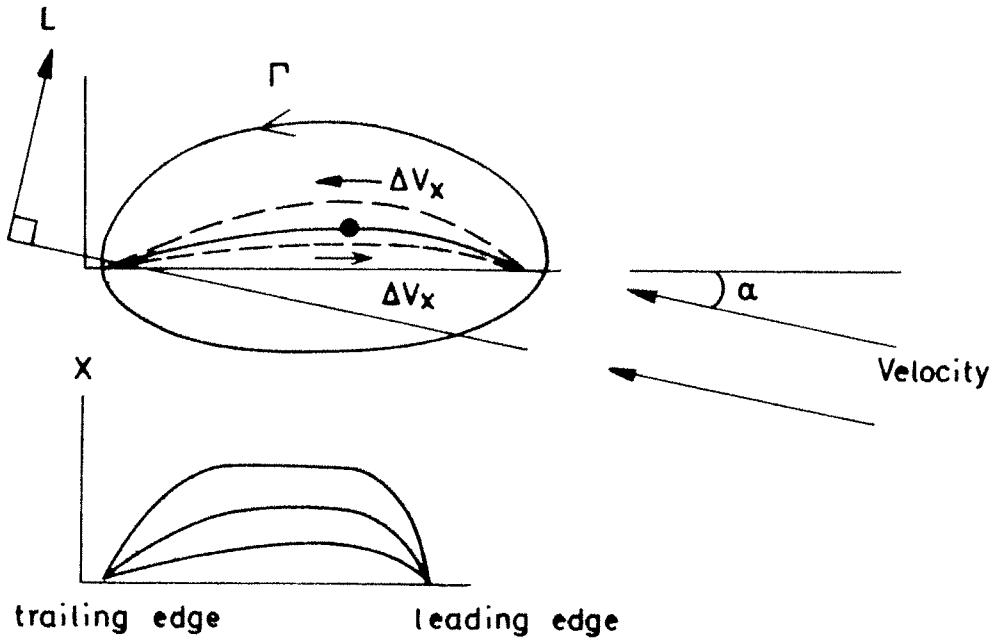
The basic concepts circulation, lift, vortices and induced velocities are briefly introduced and illustrated in Figs. 1, 2 and 3.

The main purpose of the propeller calculation is to determine the induced velocities at different points of the blades, as these affect the geometry (pitch and camber) of the blade sections. In order to determine these velocities the circulation distribution has to be known. However the relation between circulation, thrust and torque is affected by the induced velocities. Thus an iterative procedure is necessary, the starting values being the magnitude of the thrust, the number of revs and the speed and the shape of the radial circulation distribution

$$\Gamma_r = k \cdot \Gamma_0(r)$$

The value of k is determined by iteration.

A complete lifting surface calculation is a rather complicated procedure. Accordingly, propellers are even today often designed using lifting line calculations and approximate 3-dimensional calculations (procedure 2 of Fig 3a). Also designs based entirely on results of systematic series are sometimes used for merchant ships. Below some illustrations will be given as to the advantages of using a rigorous design procedure.



Local velocity, upper side, $V+\Delta V_x$

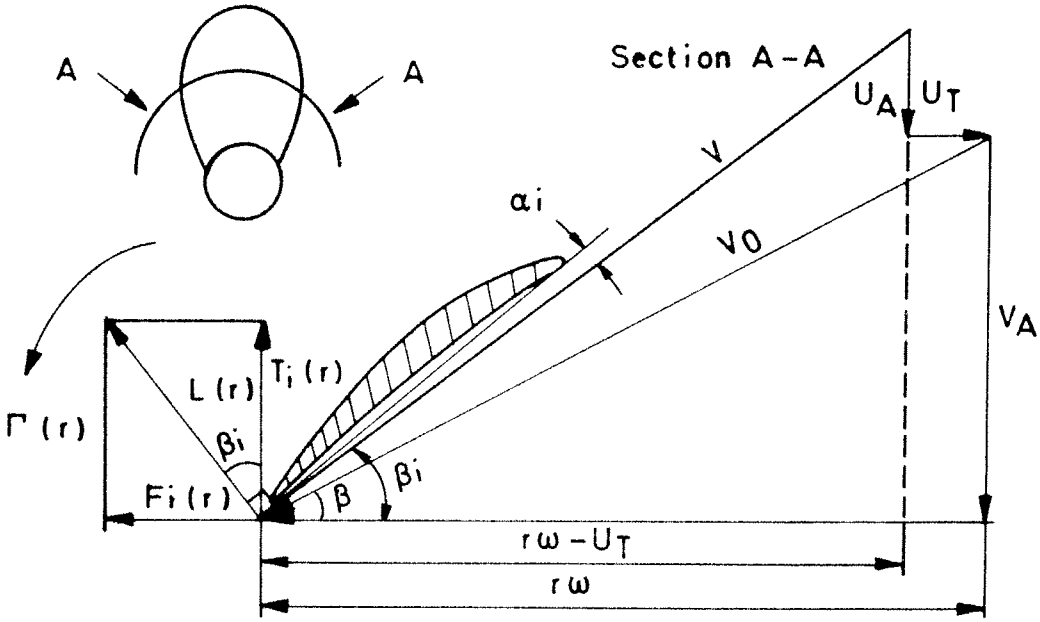
-"- -"- lower side $V-\Delta V_x$

Local circulation $\gamma(r,x) = 2\Delta V_x$

Total -"- $\Gamma(r) = \oint \gamma(r,x)\,ds \approx \int \gamma(r,x)\,dx$

Lift $L(r) = \rho V \Gamma(r)$

Fig. 1 The conceptions lift and circulation.
Hydrofoil or propeller blade section.



U_A, U_T induced velocities

$$\mathrm{tg}\ \beta = \frac{V_A}{r\omega}$$

$$\mathrm{tg}\ \beta i = \frac{V_A + U_A}{r\omega - U_T}$$

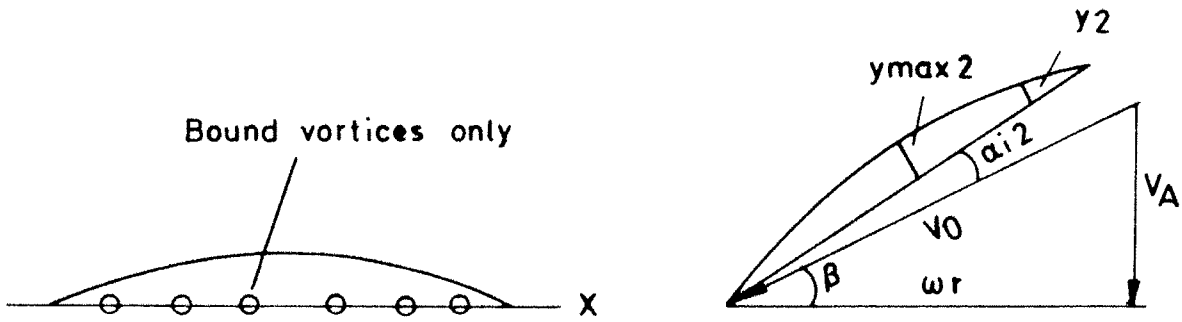
$$L(r) = \rho V \Gamma(r)$$

$$T_i = L \cdot \cos \beta i \text{ (thrust)}$$

$$F_i = L \cdot \sin \beta i \text{ (gives torque and power)}$$

Fig. 2 Blade section of a propeller blade.
Velocities and forces.

Two-dimensional calculation



y max 2 = maximum camber, two-dimensional flow

αi2 = ideal angle of attack, two-dimensional flow

Two-dim + approx. three-dim. calc. (lifting line)

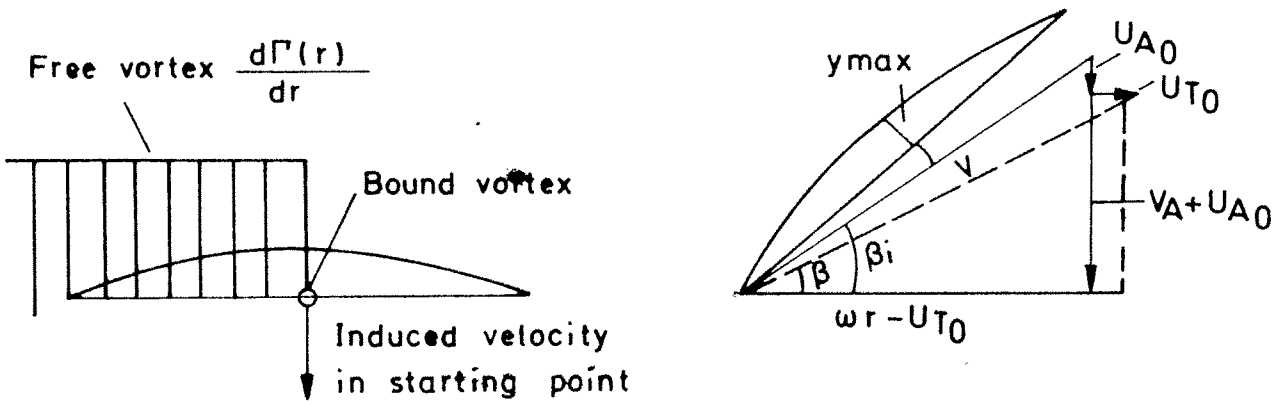
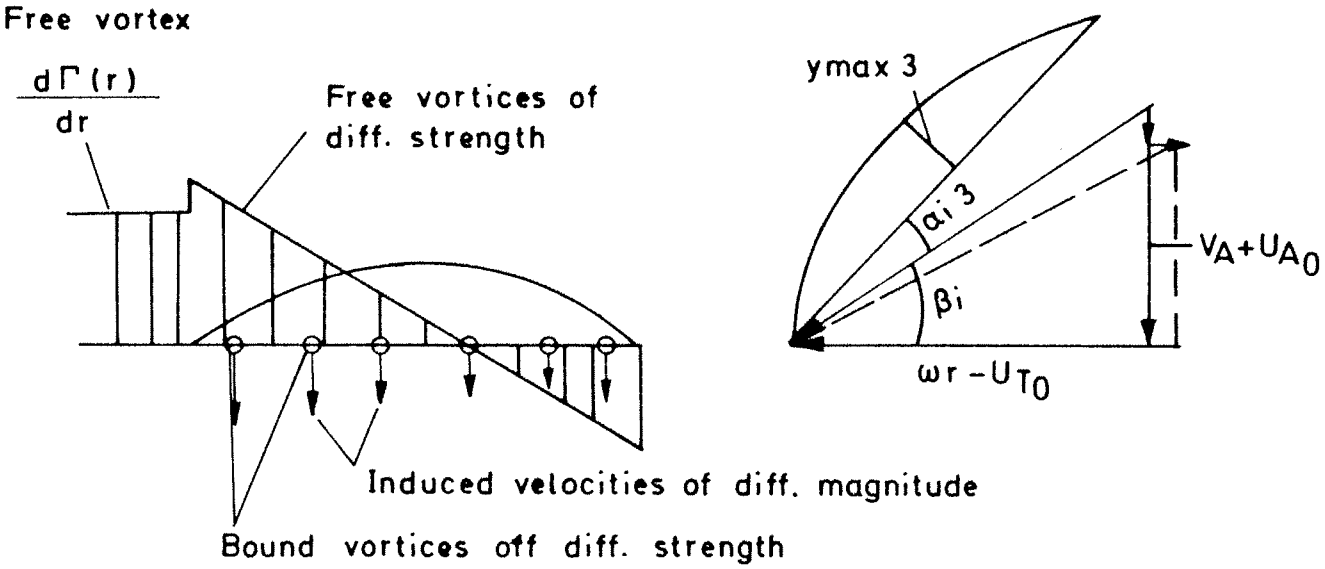


Fig. 3a Different stages of a propeller calculation: Two-dimensional and approximate three-dimensional calculation.

Complete three-dim. calc. (lifting surface)



Camber correction $k_c = \frac{ymax\,3}{ymax\,2}$

Corr. of ideal angle of attack $k_{\alpha i} = \frac{\alpha i\,3}{\alpha i\,2}$

Fig. 3b Complete three-dimensional propeller calculation (lifting surface).

3. Advantages of using modern propeller design methods

3.1. Determination of optimum diameter and blade area ratio of a propeller

A widely accepted way of determining the optimum diameter of a propeller is to use results of a systematic series (e.g. those of SSPA or Wageningen), the blade area ratio being determined by a simple cavitation criterion (e.g. that of Burrill).

Having a modern design method available, the optimum diameter can instead be determined by running the lifting line part of the program for different diameters and plotting the calculated efficiencies in the way shown in Fig. 4. This procedure has several advantages:

1) As the pressure distribution along the blade sections can be calculated, a certain margin against cavitation can be prescribed, the result being that different blade area ratios are obtained for different diameters, which affects the efficiency and thereby the optimum diameter.

2) This is also true for the hub diameter. When using results of systematic series as a basis it has to be assumed that the ratio hub diam./prop. diam. is constant.

3) When using systematic series as a basis it is assumed that the propellers have the same geometry for all diameters. When theoretical methods are used, different propeller geometries (different camber and blade length) are obtained for each diameter. As might be expected, a less peaky optimum is then obtained, when the optimum diameter is determined by theoretical calculations, than when a systematic series is used as a basis.

3.2. Determination of the radial circulation distribution over the propeller blades

It is well known (and can be proved) that the best efficiency is obtained if the propeller is given a load distribution characterized by a large amount of load on the outer sections.

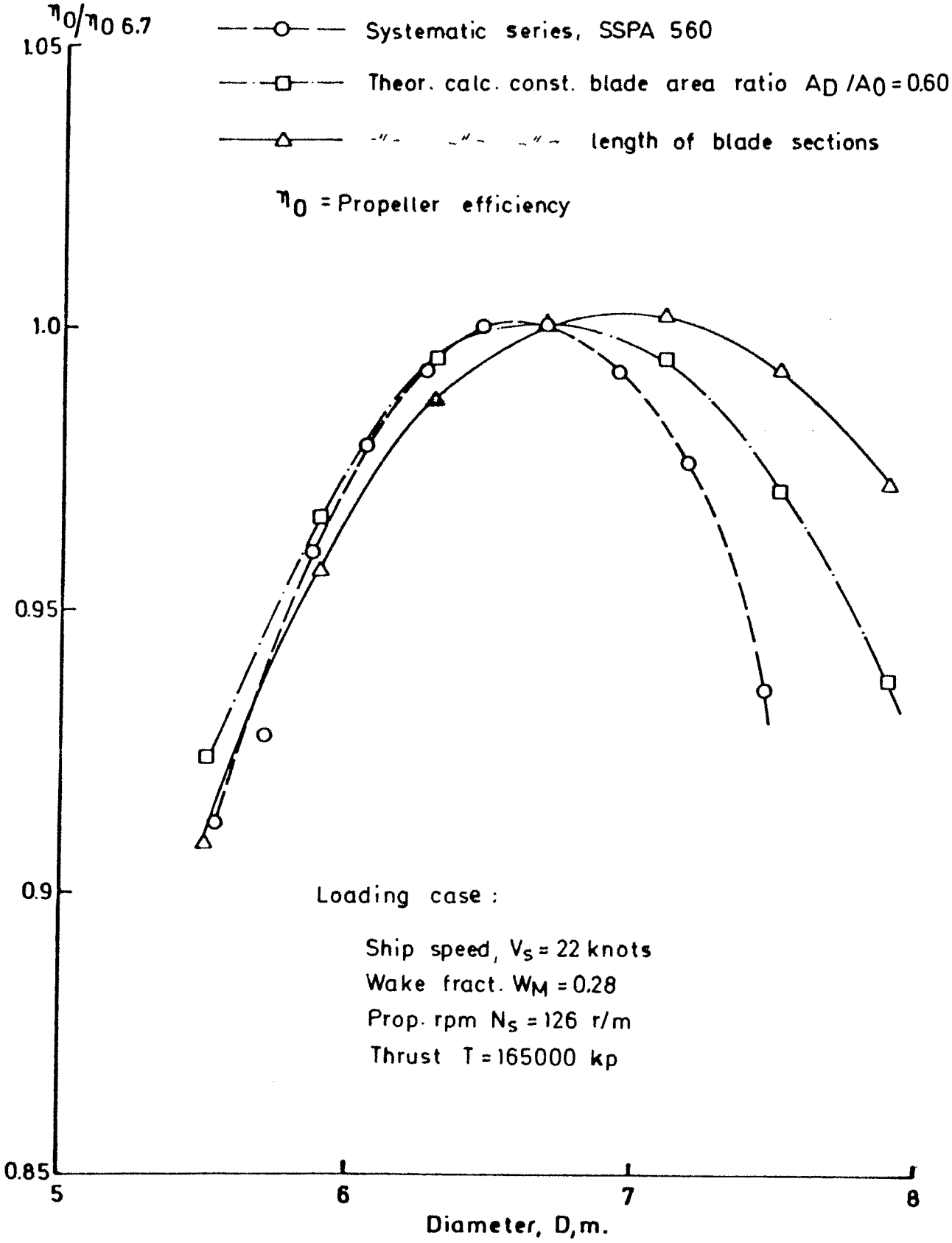


Fig. 4 Determination of the optimum propeller diameter.

Such a load distribution results in a nearly constant pitch or a larger pitch for the outer sections. Such pitch distributions are very often given (without design calculations) to propellers comprising the systematic series found in the literature. However, to obtain acceptable vibration properties for such propellers it is necessary that they work in a favourable wake field, i.e. one characterized by small spatial velocity variations. As this is not very often the case, such a propeller cannot, in many cases, be considered as an optimum compromise.

One way of improving such a propeller from the point of view of vibration excitation is to reduce the loading of the outer blade sections by modifying the radial pitch distribution. If not earlier, a theoretical design method has then to be used in order to secure an adequate relation beween camber and pitch. What can happen, if such a procedure is not used, is illustrated in Fig. 5.

In the case referred to in Fig. 5 the original propeller, which was picked out of a systematic series, was not accepted by the ship owner, the main reason being that unacceptable vibration and noise was experienced at the acceptance trials with the ship. To improve the situation in these respects, the pitch distribution and blade form were changed, but the flat face of the blade sections was maintained, i.e. the radial camber distribution was not changed, see Fig. 5. The results of the full scale tests with the new propeller (P1868), which were confirmed at model tests in the SSPA cavitation tunnel, showed that a certain reduction of the vibration excitation was obtained at the blade frequency. The tendency to cavitation erosion which, according to the model tests, was present already at the original propeller, was however seriously aggravated, the result being that the new propeller was regarded as unacceptable in this respect. An analysis of the design shows that the camber for the outer sections, being determined from the hydrodynamically irrevelant condition that the face of the blade sections should be flat, was too large compared to the camber of the inner sections. As a result of this, the extension of cavitation became large in the middle part of the blade, which in turn gave a cavity having a convex trailing edge. This is known to promote cavitation

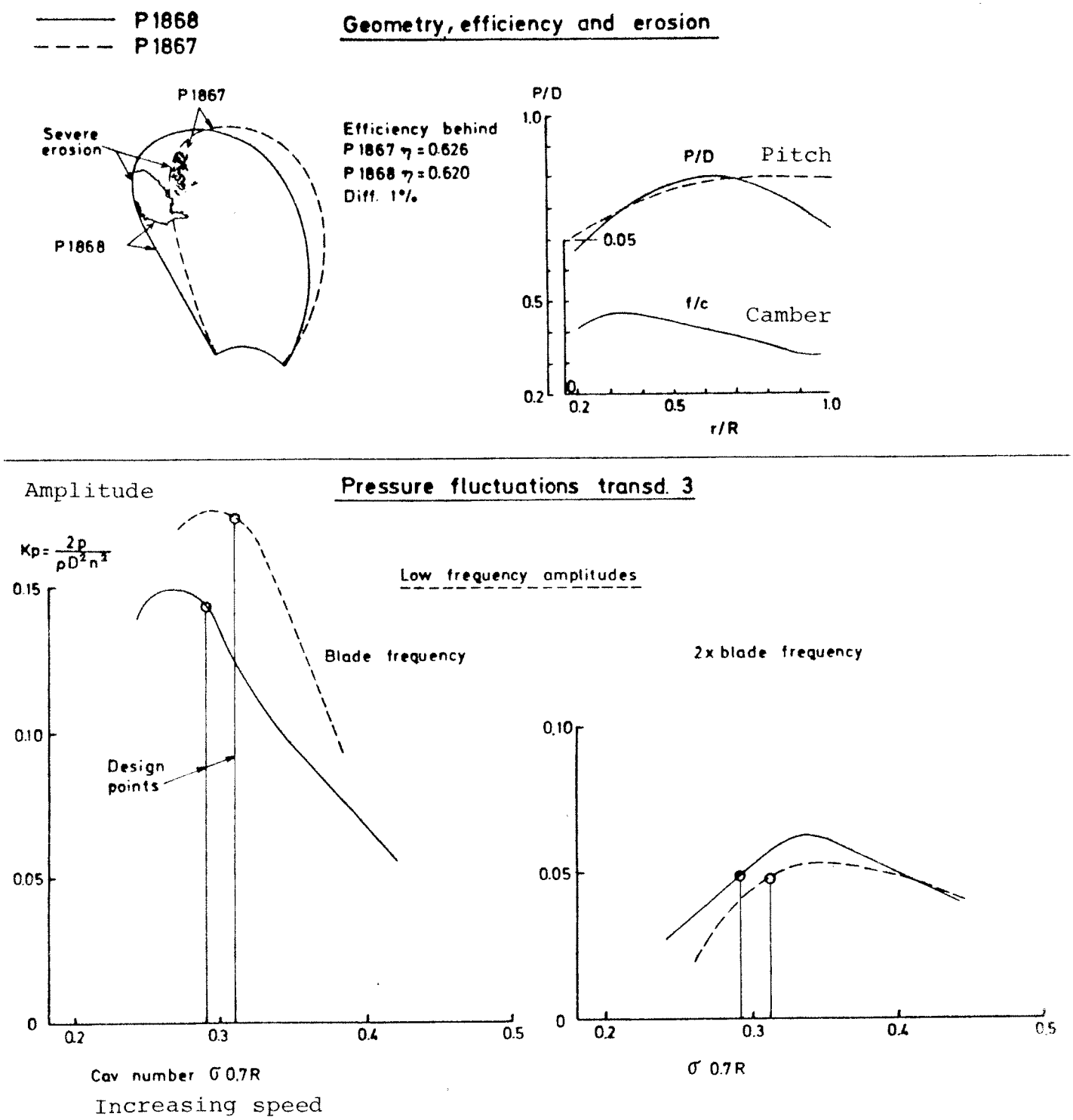
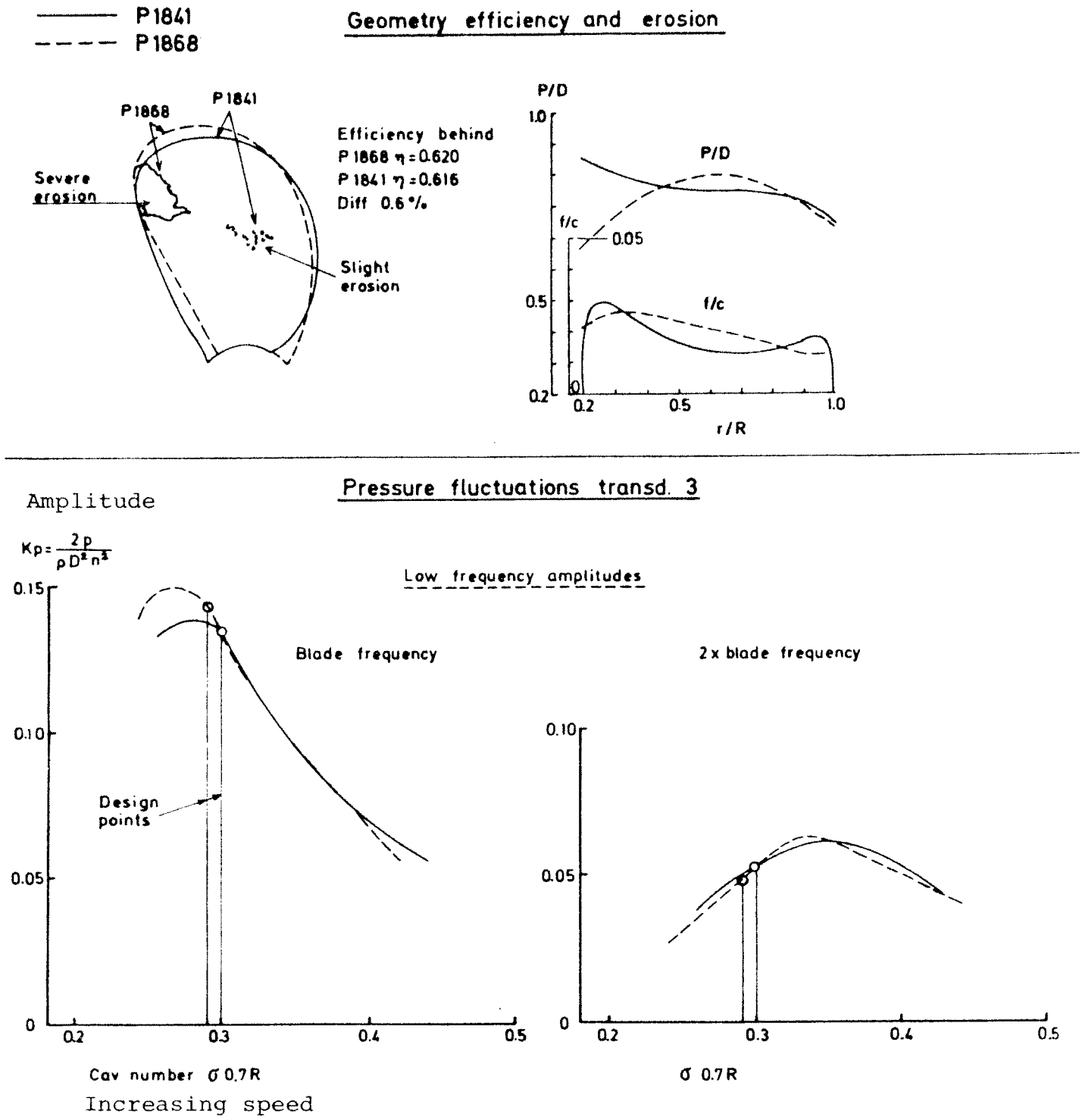


Fig. 5 Comparison of two empirical propeller designs.
Results of measurements in cavitation tunnel.



P1868 empirical propeller design, see Fig. 5.
P1841 designed by lifting surface theory.

Fig. 6 Comparison of propeller designs.

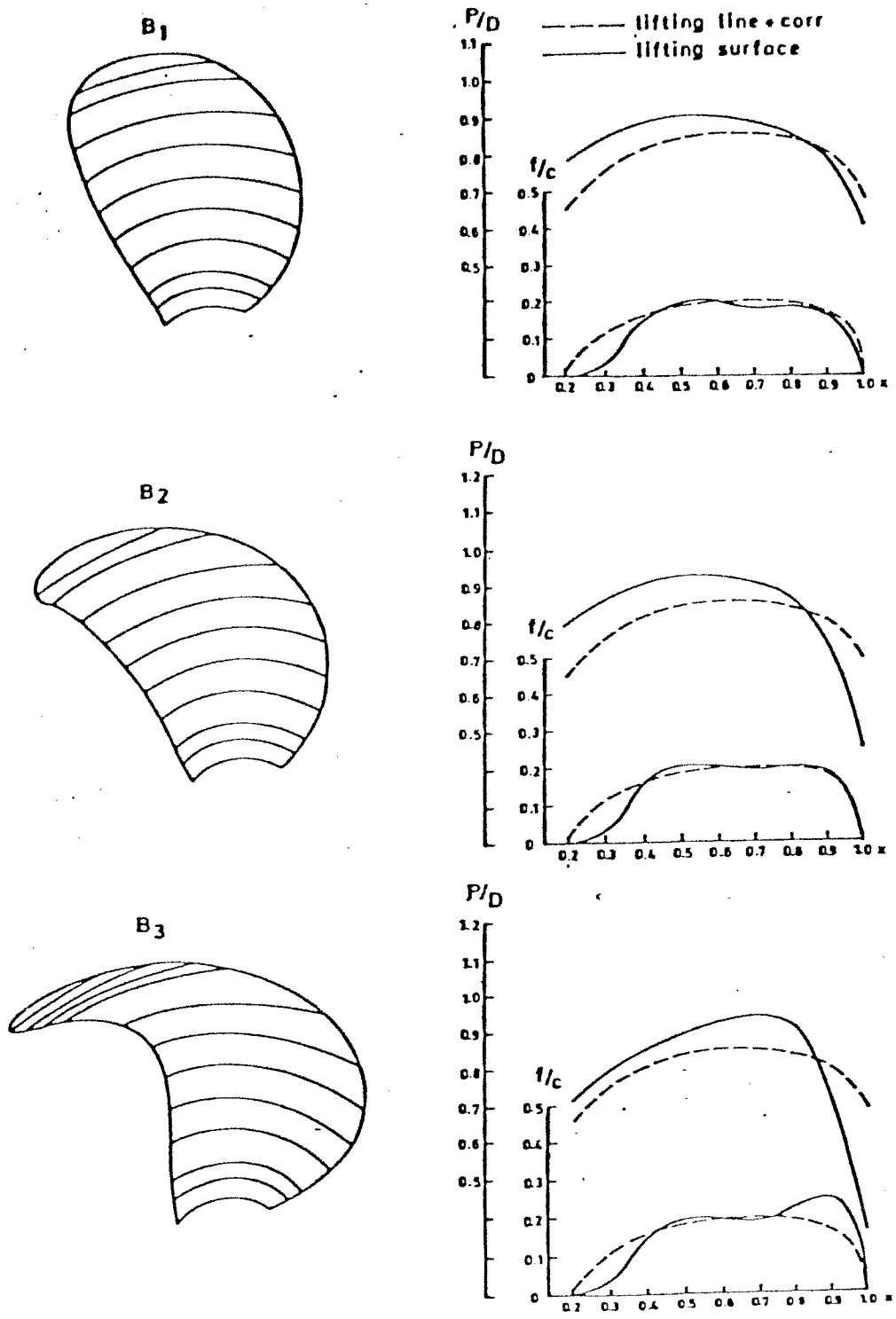


Fig 7.- Influence of propeller design method on radial pitch and camber distributions. Different blade forms. Radial circulation distribution G_2 in Fig. 8.

damage. This effect was present already at the original propeller but was amplified when reducing the pitch of the outer sections, without making the corresponding reduction of the camber.

A more successful design for the same case is shown in Fig. 6. For this propeller design calculations were made, using lifting surface theory, the result being that the pitch and camber fit each other. It is evident from Fig. 6 that this propeller has much less cavitation damage than the empirically designed propeller, the vibration excitation level and efficiency being the same.

3.3. Influence of blade form on propeller design

One way of reducing the vibration excitation from the propeller is to use extreme skew back, another one is unloading of the blade tips. Very often these two concepts are combined. This is a situation where a rigorous calculation is necessary, as a drastic change of the blade form affects the relation between the radial distributions of loading (circulation), pitch and camber. This is illustrated by Fig. 7, which shows three propeller designs, characterized by the same radial circulation distribution, but different blade forms. The pitch and camber distributions obtained by two different design methods are shown in the diagrams. The approximate method is based on lifting line calculations, camber correction factors from the literature being used for determining the final values of pitch and camber for the blade sections. Using this procedure, the same radial distributions of pitch and camber are obtained for all three designs. If, on the other hand, a rigorous lifting surface procedure is used, different distributions of camber and, particularly pitch, are obtained for the three blade forms. From Fig. 7 it is evident that, if an approximate method is used for the design, the propeller with extreme skew back will get too large a pitch for the outer sections, the result being more loading near the tips than prescribed, which means that the beneficial influence of skew back on vibration excitation is reduced and sometimes eliminated.

Based on the lifting surface calculations illustrated in Fig. 7

two model propellers with blade forms 2 and 3 were manufactured and tested in a cavitation tunnel. The following beneficial effects of increased skew were found at these tests:

A) The excitation level was reduced over the whole frequency range investigated, the reduction at blade frequency being about fifty per cent.

B) The extension of cavity damage was reduced.

These advantages were obtained without any reduction in efficiency. However, strength calculations carried out by finite element methods showed that high stresses could be expected in some parts of the blades, particularly when reversing and backing.

3.4. Influence of radial load distribution on propeller design and properties

One possibility of affecting the compromise between high efficiency and good cavitation behaviour is to play around with the radial load distribution. It is well known that, in principle, unloading of the blade tips means a efficiency loss but lowering of vibration excitation at blade frequency. It is however difficult to quantify these changes by theoretical calculations. Accordingly, an experimental investigation was recently completed at SSPA, including a variation of the radial load distribution and blade form.

The results of this investigation were summarized in [1] and some of them will be discussed below. The starting point of the investigation was to make twelve propeller designs for the same loading case, the number of blades, the diameter and the blade area ratio being the same, see Fig. 8.

A problem, not mentioned earlier, is indicated in Fig. 8, i.e. how the propeller should be adapted to the wake field. This is a very complex problem, which will not be discussed in detail here. As is evident from Fig. 8 it affects the relation between the camber and pitch distributions. It has to be mentioned that

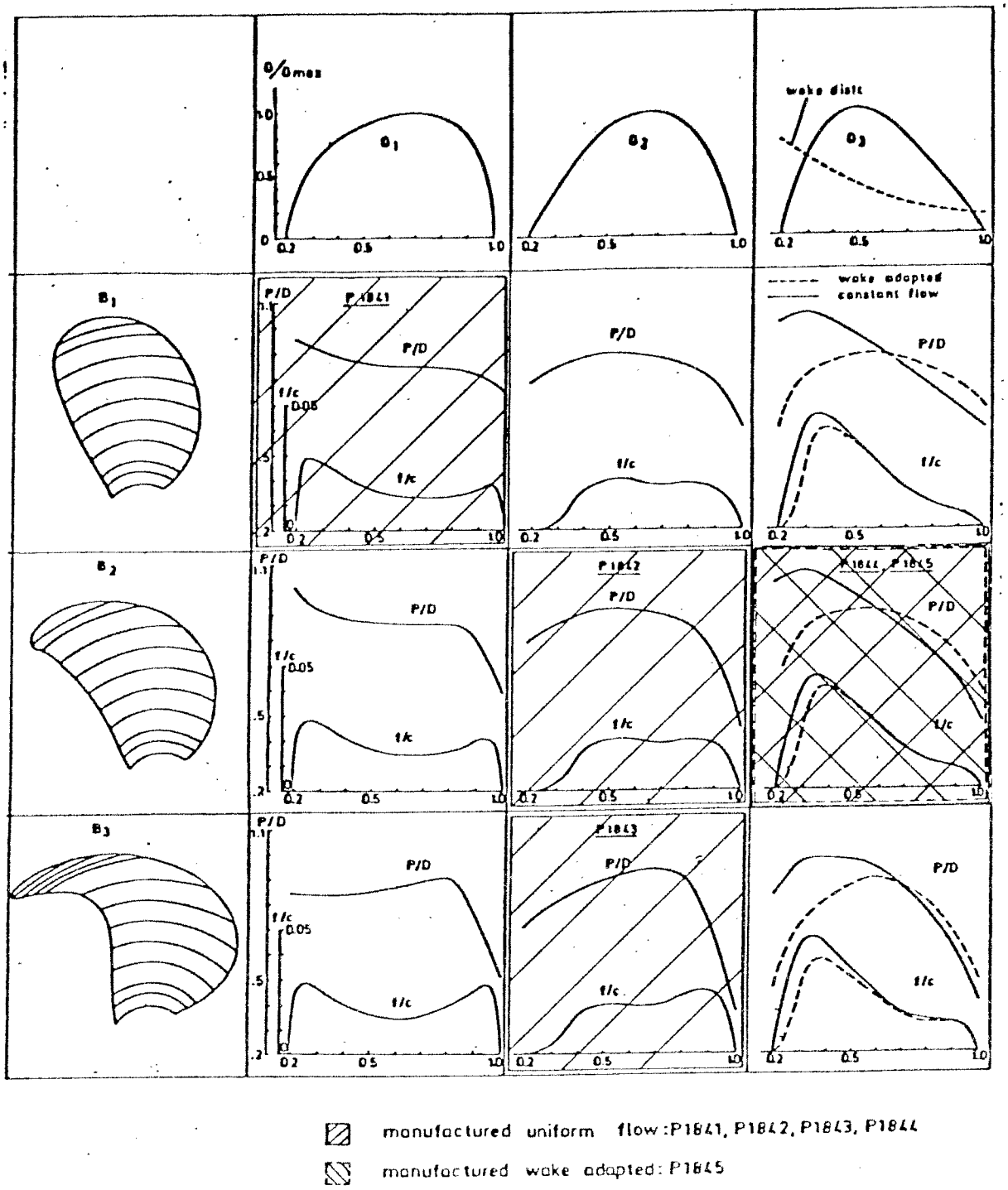


Fig. 8 Different propeller designs for the same loading case (5-bladed container ship propellers). Variation of radial circulation distribution and blade form.

the majority of the designs were made, assuming a steady, uniform flow. For one circulation distribution however, designs were also made, assuming a conventional adaption to the mean wake of each radius.

To determine the influence of the two main design parameters, model propellers were manufactured in accordance with five of the designs and extensive model tests were carried out in the towing tank and the cavitation tunnel.

The designs were made for a loading case corresponding to that of an existing Ro-Ro-ship. By courtesy of the ship yard models of the two full scale designs discussed earlier (see Fig. 5) were borrowed and tested together with the propellers of the series. The results of these tests will be discussed below.

4. Experiments carried out with models of different propeller designs

4.1. Types of experiments

Below the types of model tests carried out with the different designs are listed, and the results are discussed. The extension of these tests is what can be regarded as reasonable when evaluating a propeller design for a merchant ship. As was done in this case, the results of the model tests should be checked by full scale measurements and observations.

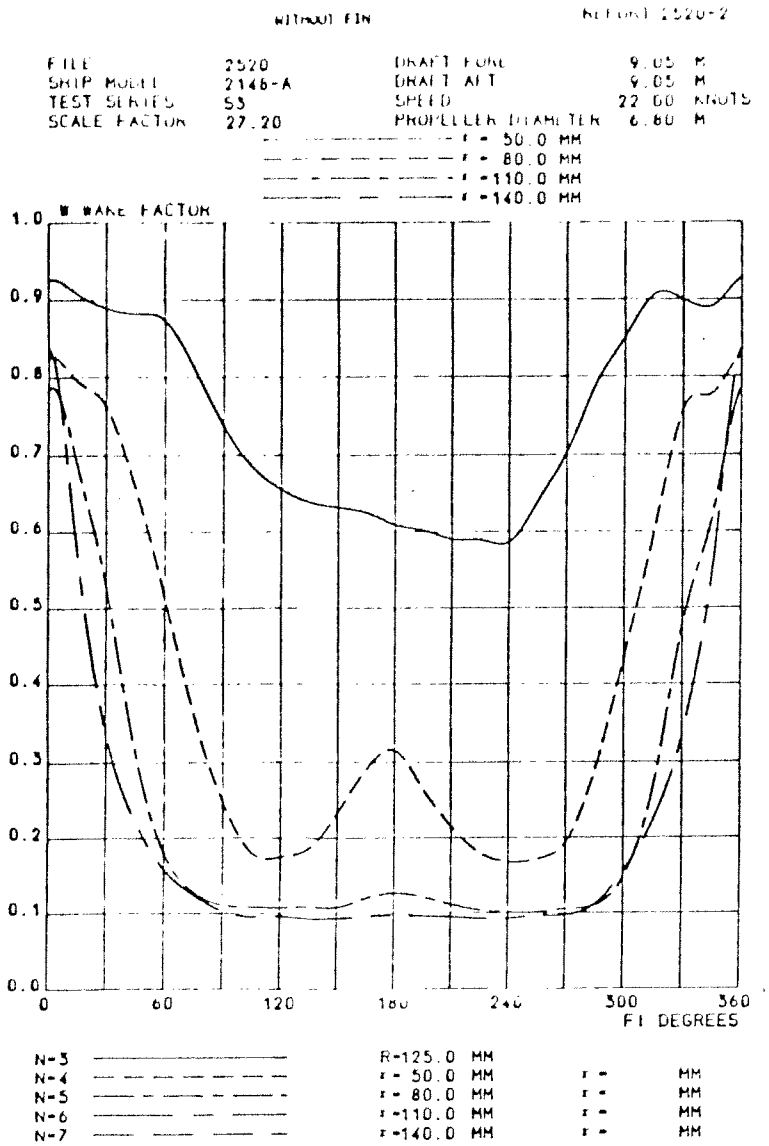
The following kinds of tests were carried out:

1) Open water tests in the towing tank for determination of propeller characteristics.

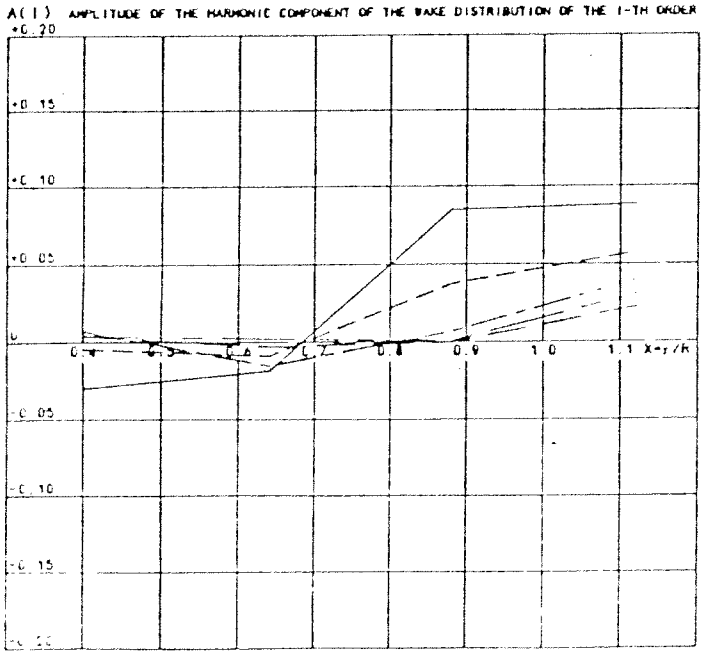
2) Measurement of the wake field behind the ship model (see Fig. 9).

3) Self propulsion tests in the towing tank and similar measurements in the cavitation tunnel.

4) Visual observations of cavitation patterns on the propeller blades in the tunnel.



Measured peripheral wake distribution. Example



Result of harmonic analysis of wake distribution

Fig. 9 Wake distribution measured for a Ro-Ro-ship model used for propeller series.

5) Paint tests for the determination of the risk of cavitation damage, see Fig. 10.

6) Measurement of the pressure fluctuations induced by the propeller in different points of the stern of the ship model, see Fig. 11.

The cavitation tunnel tests, items 4-6, were carried out in SSPA cavitation tunnel No. 2, using the same ship model as was used in the towing tank, see Fig. 12.

4.2. Results of the experiments

Some results of the experiments, referred to above, are given in Figs. 5, 6 and 13-15.

In Figs 5 and 6, which were discussed earlier, some of the individual propellers were compared. In Figs. 13-15 the properties of the different designs are summarized, the degree of unloading of the tips being characterized by the parameter (pitch 0.95 R)/(pitch 0.4 R).

The influence of the different parameters investigated on the properties of the propellers can be summarized as follows:

Efficiency (see Fig. 13)

A) The radial pitch distribution is the most important parameter as regards efficiency for propellers with conventional blade form and the same blade area ratio.

B) The efficiency loss due to tip unloading is smaller in behind condition that at open water tests.

C) The efficiency loss due to unloading is predicted reasonably well by theoretical calculations.

D) A propeller having extreme skew has the same efficiency as the corresponding propeller with conventional blade form (the same radial circulation distribution).

Pressure fluctuations

A) Also when pressure fluctuations are concerned the radial pitch distribution is the most important parameter, design principles and blade area ratio being of less importance.

B) The amplitudes of blade frequency can be reduced appreciably by unloading the blade tips. For twice the blade frequency and higher frequencies, unloading of the tips is not a very efficient way of reducing the pressure amplitudes.

C) When incorporating extreme skew in the design the pressure amplitudes are reduced significantly over the whole frequency range.

Cavitation damage

A) The relation between the radial distributions of pitch and camber is very important for the intensity and extension of the cavitation damage. This is evident from Figs. 5 and 10. The comparison in Fig. 5 indicates an increasing extension of cavitation damage when the pitch of the outer blade sections is reduced without the corresponding reduction of the camber.

B) In the present case the extension of the cavitation damage is slightly reduced with increased tip unloading and skew.

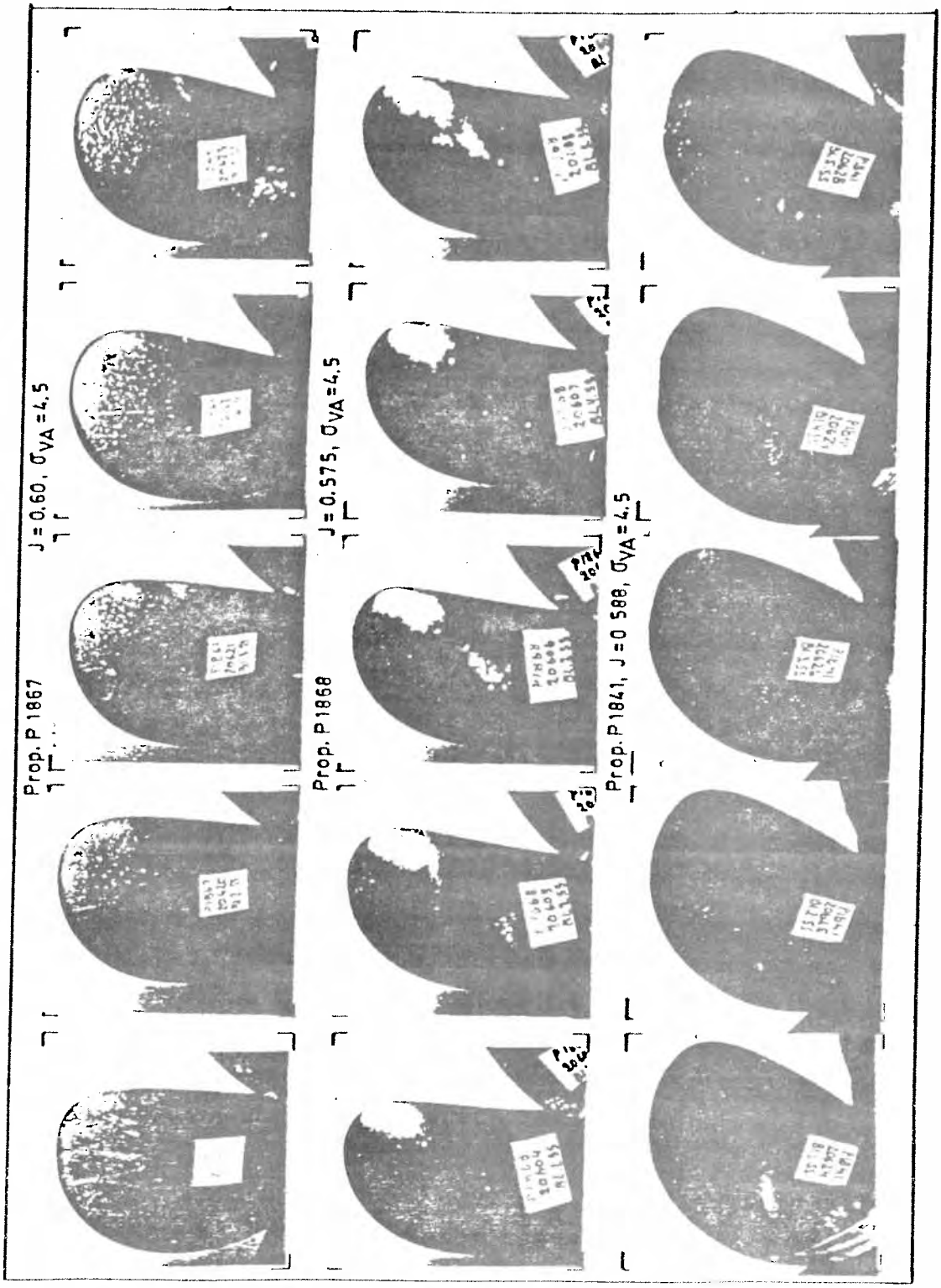
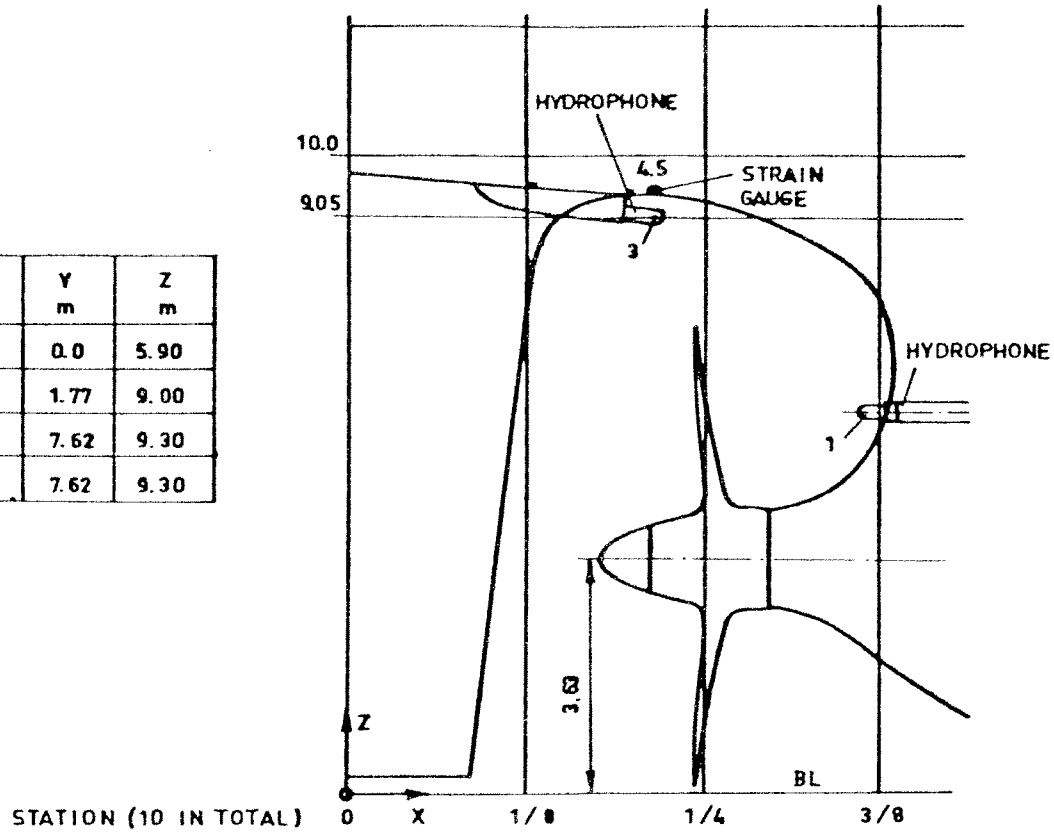


Fig. 10 Results of paint tests for detection of risk of cavitation damage. Photos of suction side (back) after 30 minutes' run.

Transd.	X m	Y m	Z m
1	8.08	0.0	5.90
3	4.58	1.77	9.00
4	"	7.62	9.30
5	"	7.62	9.30



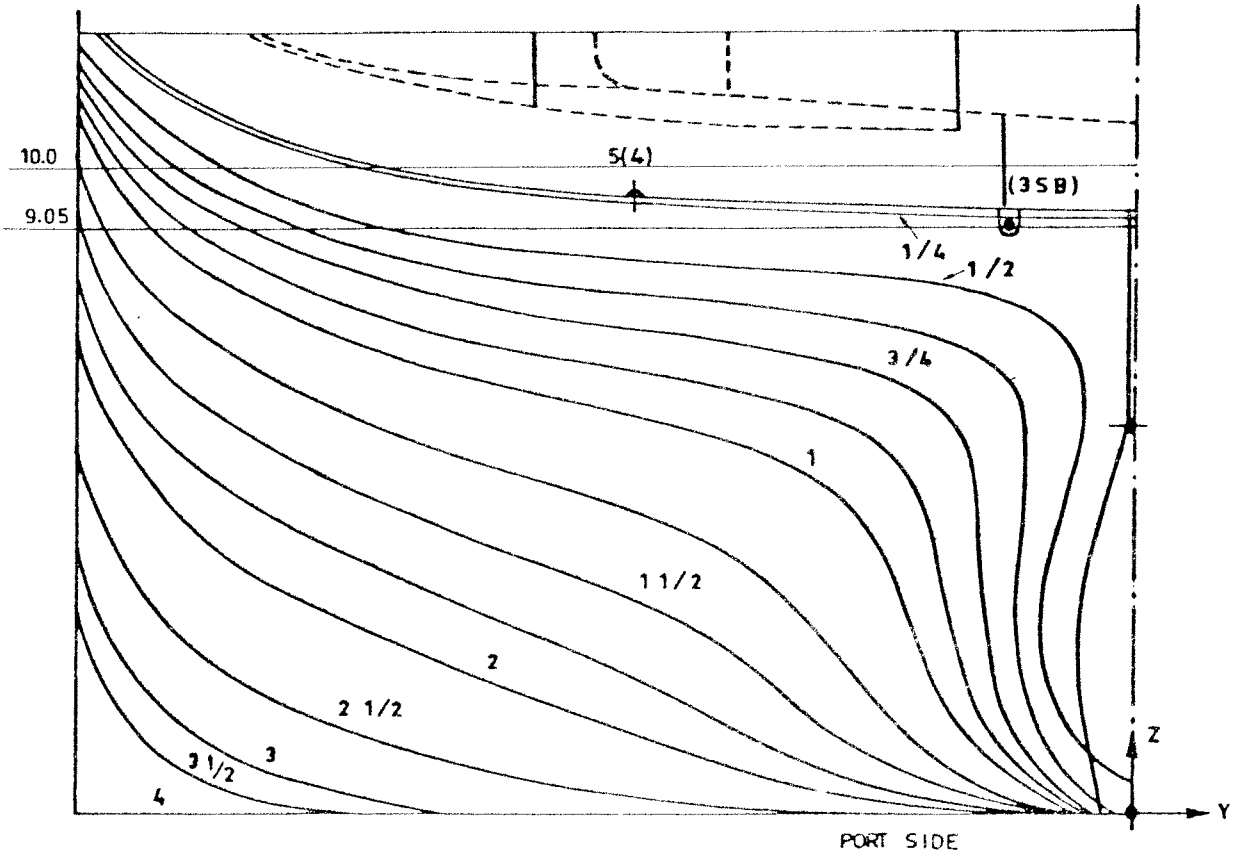


Fig. 11 Positions of hydrophones and transducers for the measurement of pressure fluctuations.

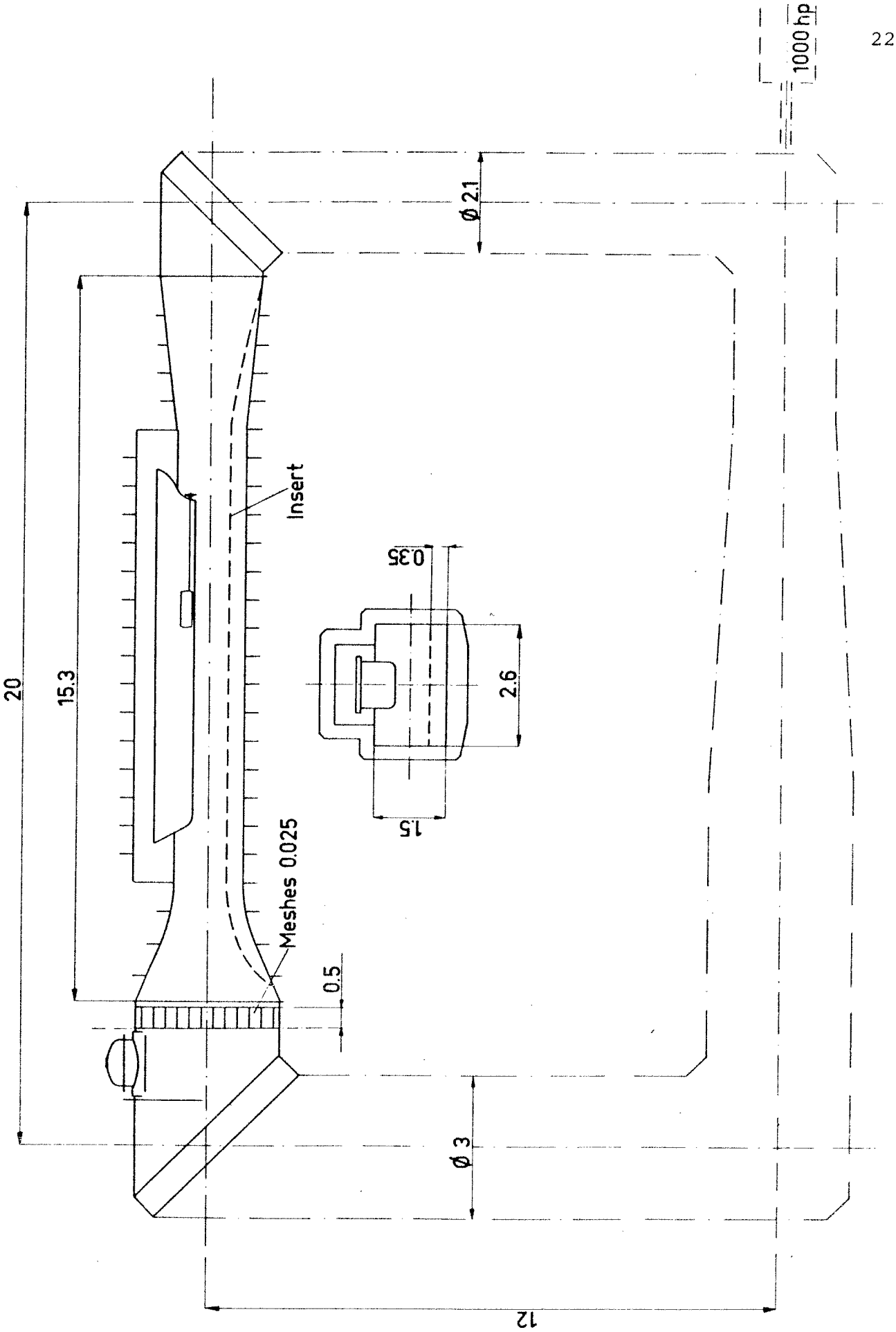


Fig. 12 SSPA Cavitation Tunnel No. 2. Low speed test section in place.

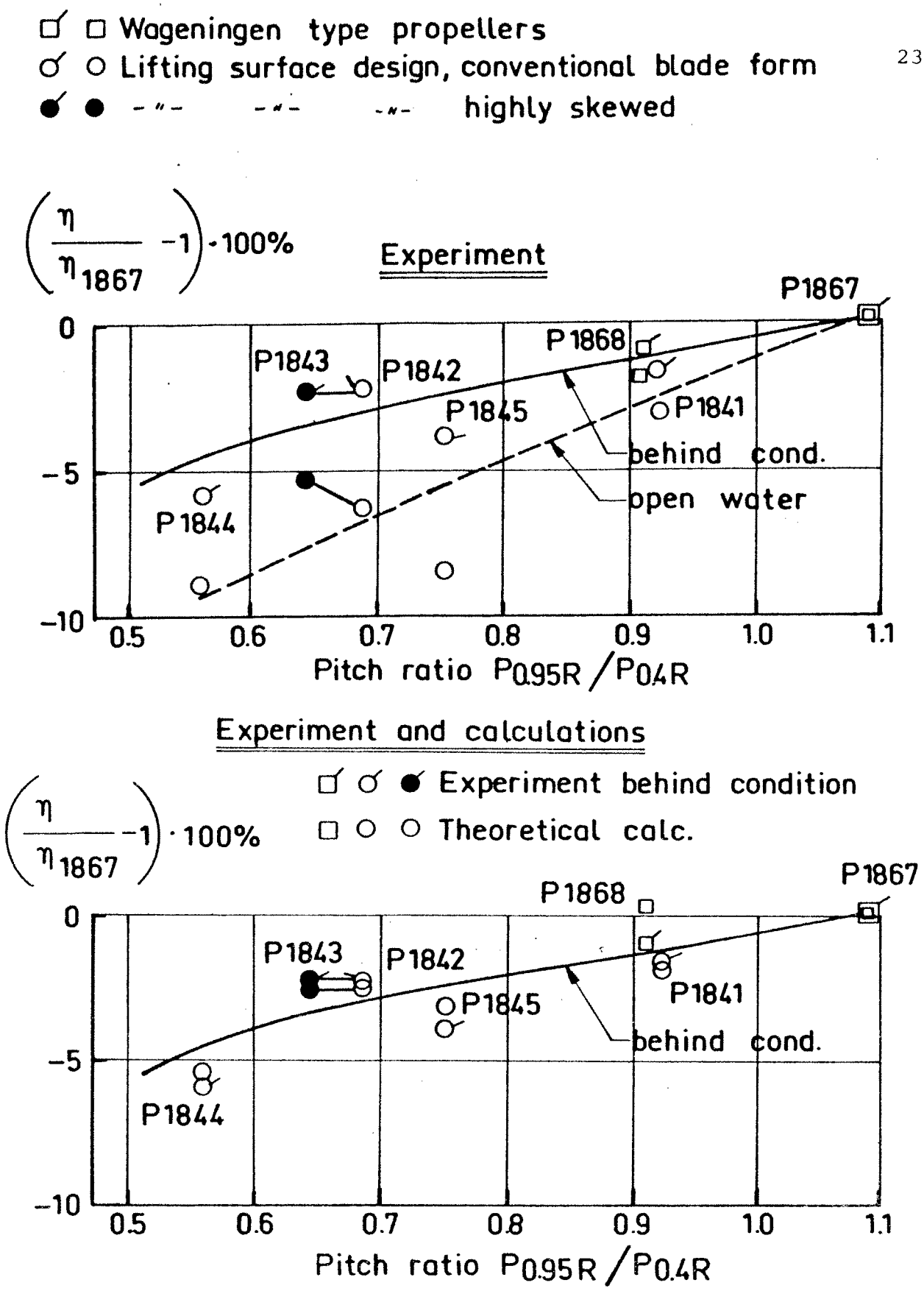


Fig. 13 Influence of radial load distribution and skew on efficiency. Calculations and experiment.

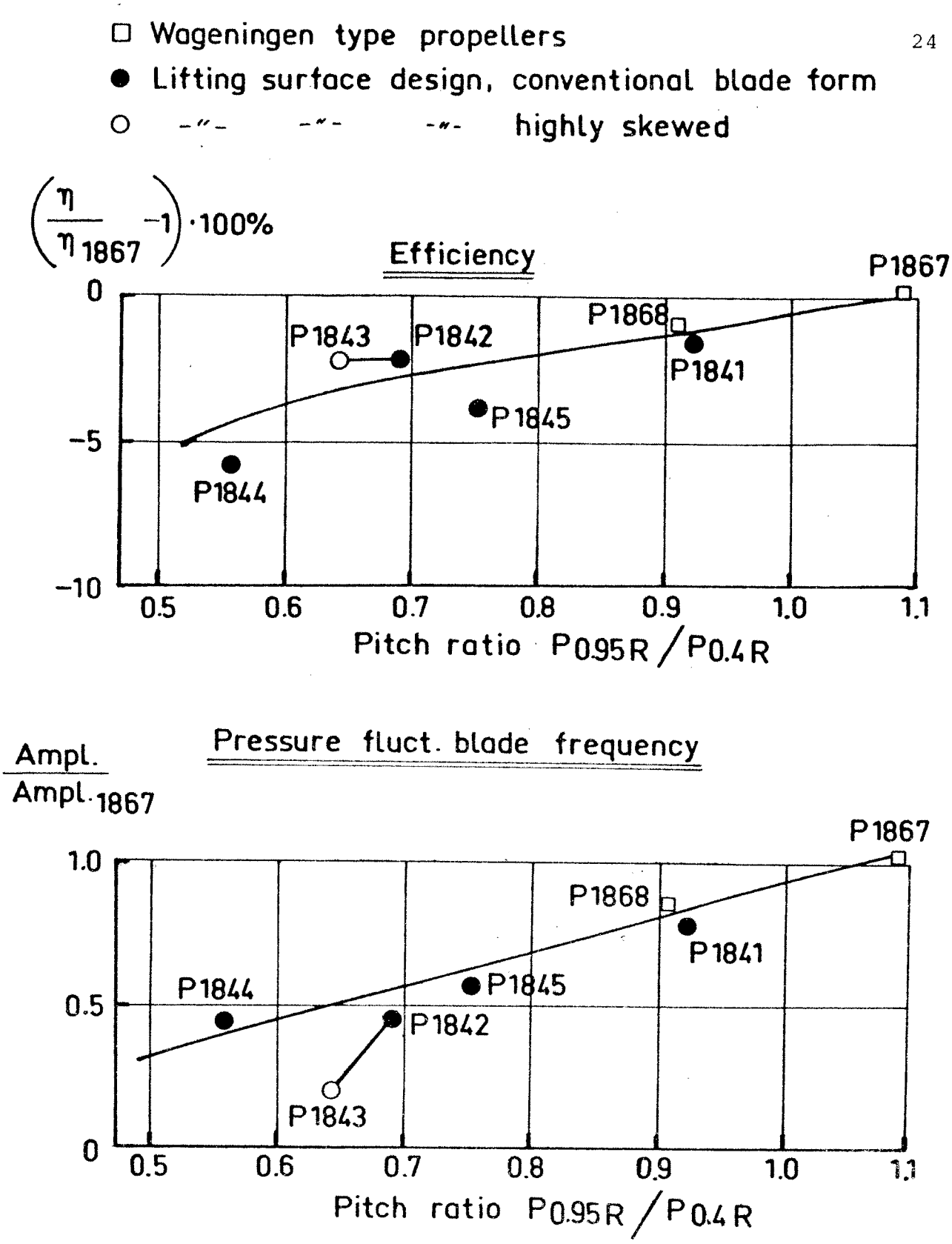


Fig. 14 Influence of radial load distribution and skew on efficiency and pressure fluctuations of blade frequency.

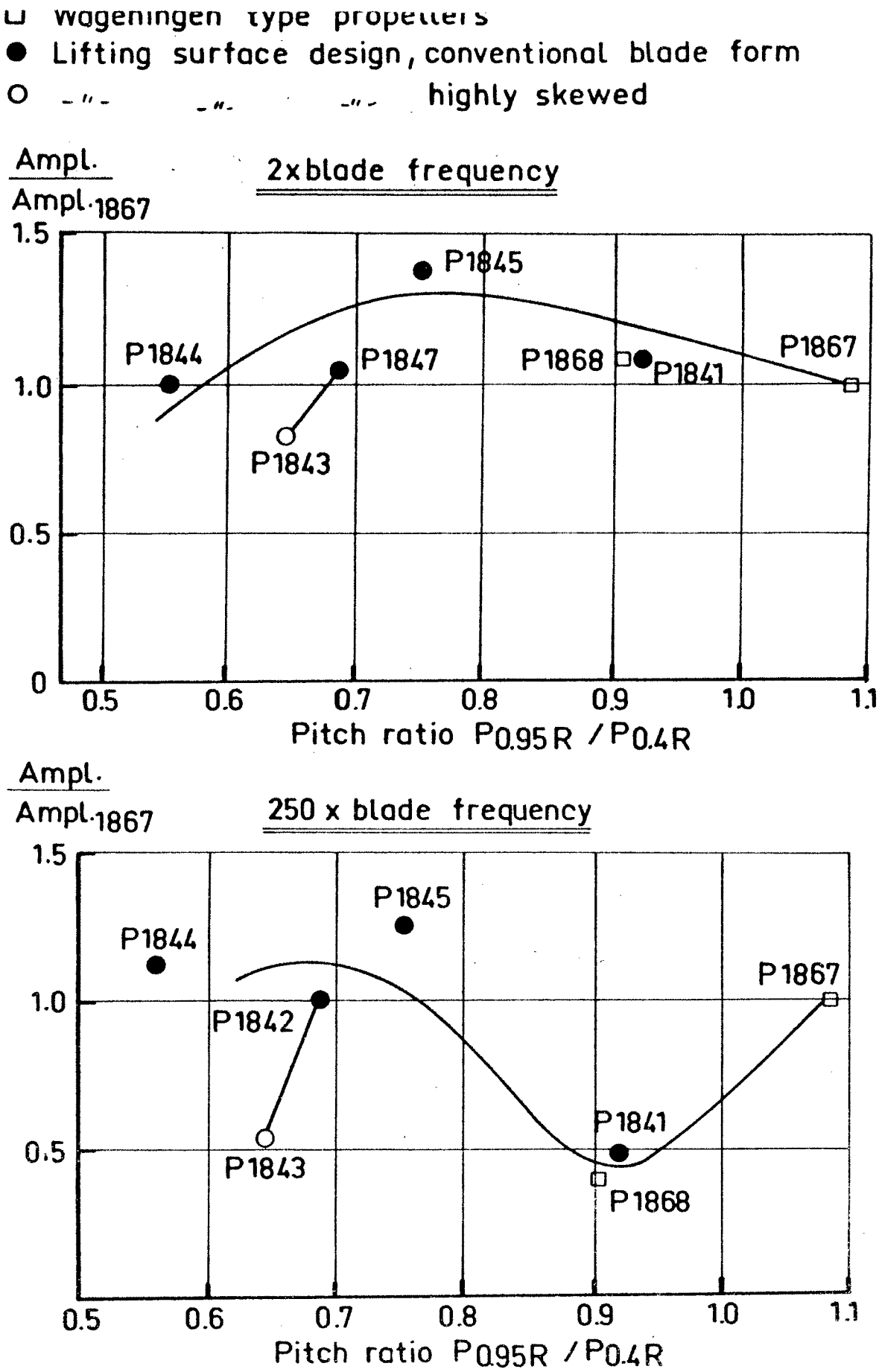


Fig. 15 Influence of radial load distribution and skew on pressure fluctuations. Twice and 250 times the blade frequency.

5. Other important parameters

5.1. Optimizing thickness and length of blade sections

From profile tables or theoretical calculations it is evident that, if a propeller blade section works at an ideal angle of attack, a better cavitation margin (and slightly better efficiency) is obtained for a thin profile than for a thick one. If, on the other hand, the section works at a large angle of attack, the deterioration of the pressure distribution is larger for the thin profile and the final result might in some cases be that the cavitation properties turn out to be better for the thick profile. This situation is illustrated by the pressure distributions shown in Fig. 16.

Thus, to find a good compromise in the merchant ship case, calculations of efficiency and cavitation behaviour should be carried out for the propeller working in the non-steady flow behind the hull. Such calculations are however very complicated and the results of this kind published hitherto have to be regarded as somewhat unreliable.

5.2. Optimizing the blade section shape

For almost all modern ship propellers, designed by theoretical methods, the blade sections are built up by NACA profiles, the most widely used type being composed of the a=0.8 mean line and the 16-type thickness distribution or a purely elliptic thickness distribution which is very similar to the NACA 16. The blade sections obtained in this way have however some practical disadvantages, one being due to the fact that the outer blade sections of a propeller are in most cases very thin. Very often the thickness/length ratio of such a section is much smaller than those covered by the NACA tables, and the formulas in these tables give values of the leading edge radius and the ordinates of the forward part of the profile which are much too small to satisfy strength and manufacturing requirements. Thus the forward part of the profile has to be modified, the result being a pressure distribution which may be unfavourable from the point of view of cavitation, as well as efficiency, see Fig. 17 a.

This is a field for potential improvements and by the use of modern profile theory, some attempts to improve the situation have been made at SSPA and elsewhere.

It could be expected that both the shape of the mean line and the thickness distribution might be improved. In Fig. 17b the forward part of a type of thickness distribution is shown, which has been tried at SSPA with some success. The chordwise distribution of thickness is composed of two ellipses, the maximum thickness being rather close to the leading edge in case of a thin profile. A diagram showing the chordwise pressure distribution for three blade sections of this kind is also given in Fig. 17b. It is evident from Fig. 17b that the pressure distribution is more favourable for the section having the maximum thickness close to the leading edge than for the other sections. This result has been confirmed at model tests with propellers having similar profiles, the improvement being manifested as lower vibration excitation and slightly higher efficiency than obtained with the corresponding propeller having modified NACA blade sections. Further work is going on at SSPA in this area, using both theoretical and experimental analysis and design methods and also investigating the influence of the mean line shape on the efficiency and cavitation properties of a propeller.

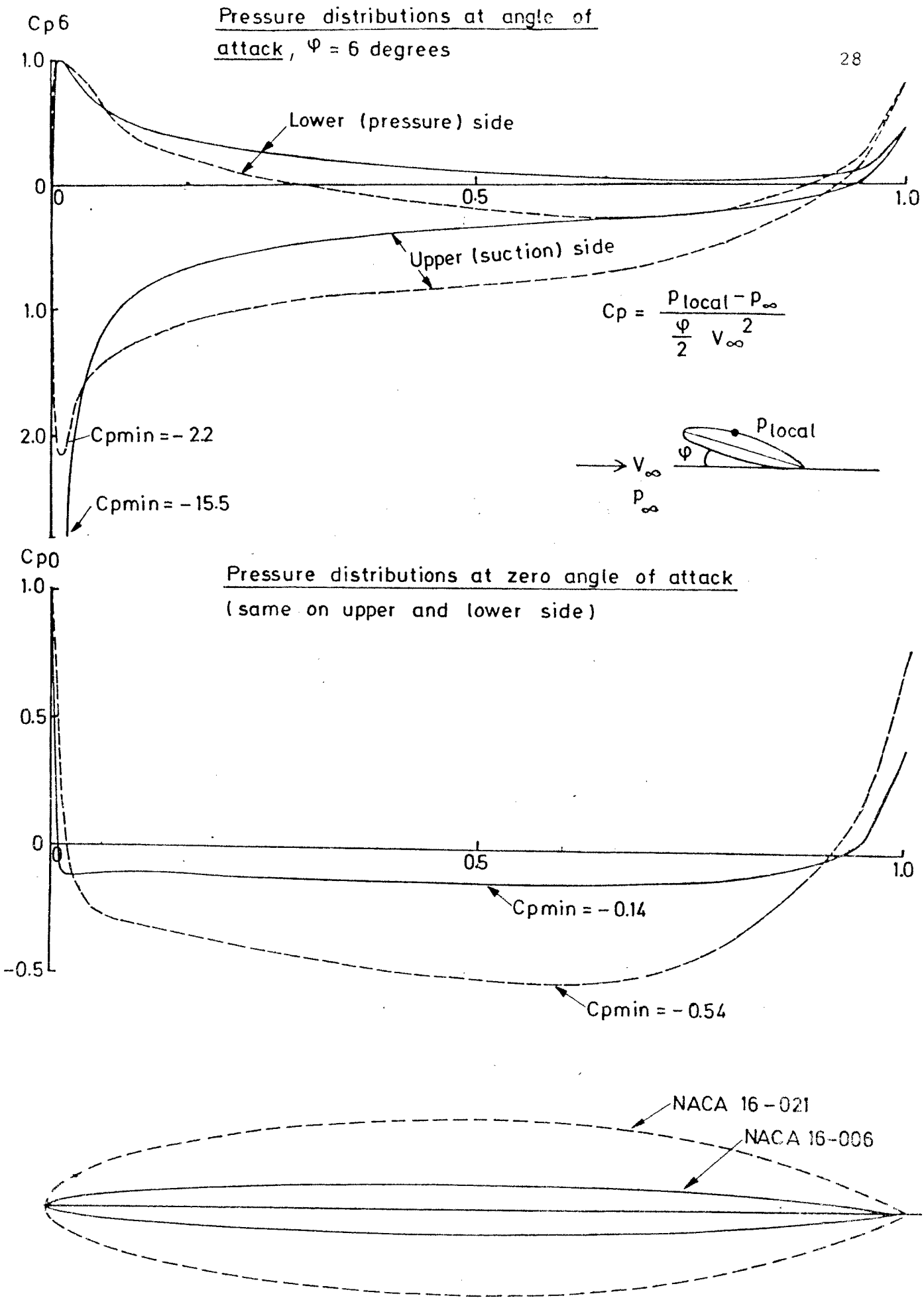


Fig. 16 Pressure distribution on blade sections having different thickness ratios.

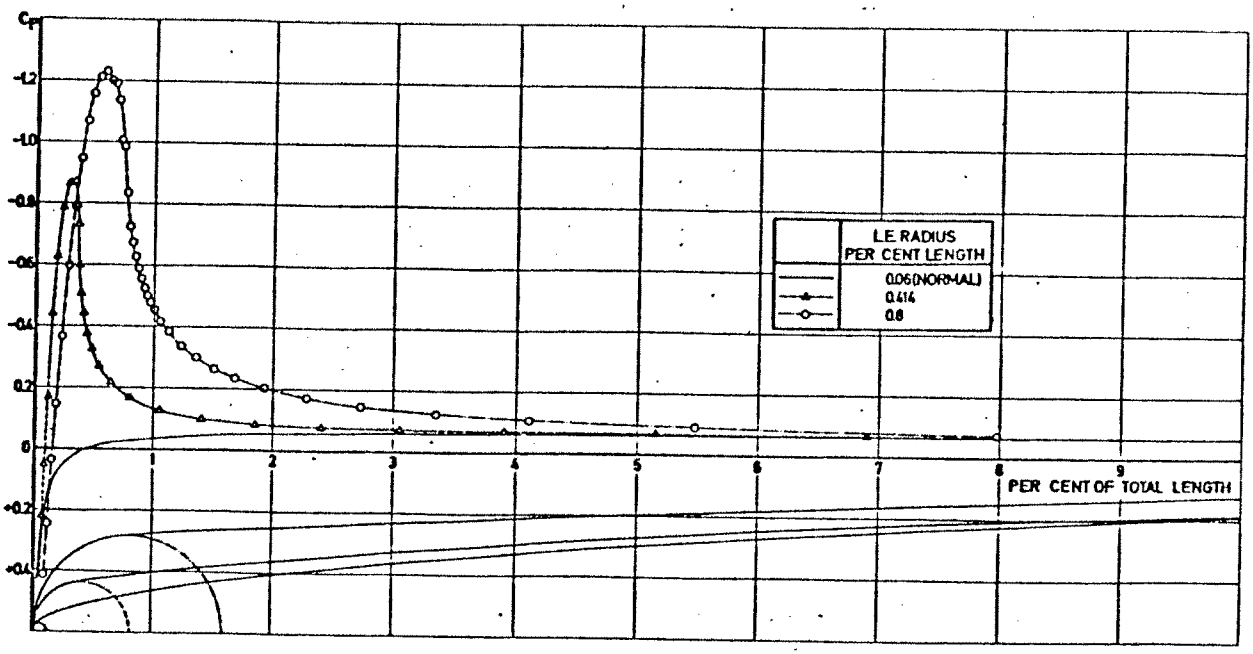


Fig. 17a Influence of leading edge radius on pressure distribution at zero angle of attack. Symmetric profile of elliptic shape. Thickness length ratio s/l = 0.0346. (Blade section at r/R = 0.95 of tanker propeller.)

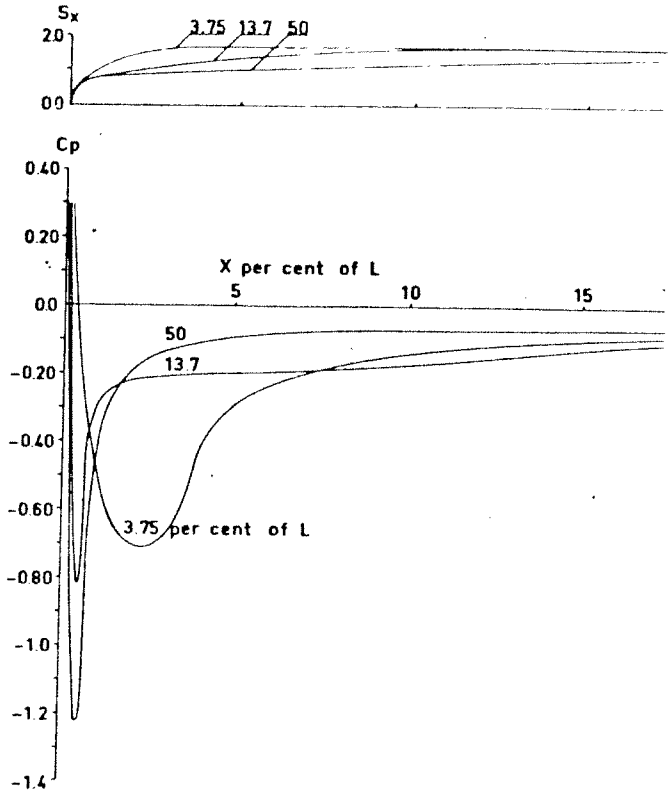


Fig. 17b Influence of position of maximum thickness on pressure distribution. Leading edge radius 0.8 per cent of L. The same profile as in Fig. 17a.

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WEST EUROPEAN GRADUATE EDUCATION
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EIGHTH SCHOOL 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

Part 2

MACHINERY

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SHIP DESIGN FOR FUEL ECONOMY

Comparison of power plant performance

by

Lars Th Collin and Kjell AC Bergström

Chalmers Technical University
Gothenburg

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1 BACKGROUND

From its very beginning development of marine propulsion machineries has been a very stimulating task both for individuals, industries, and ship owners. During most of this century the main interest has been concentrated on the low speed direct coupled diesel engine and the steam turbine. Most probably these two machineries had been considered to being developed to their optimum evolution limits uptil in the middle of the nineteenseventies, as the conditions they had been working under during the pre-war period and the post-war period had been very stable and they had been supposed to have developed a kind of balance status between each other.

The economic conditions under which they had been competing had not changed for several decades and the competition criteria to which they had been exposed during that period had only altered in a relatively slow way during the years. That had in its turn brought about a live expectancy for marine propulsion units where there had been a rather large time available for research, development and marketing and the cost recovery for both the manufacturer and the utilizer had equally had an extended time for their balance. If trying to draw a time sequence being typical for the middle decades of this century, an evolution and development period of around five years from the first idea to the first engine in series production might be considered as reasonable. After that period a period of production and marketing including time for alternations of the original design was likely to follow and to last for about twenty to twentyfive years, whereafter a period of about twenty years was foreseeable for the operation of that particular machinery on board under economical and favourable conditions. Summing up these periods of time we find that the successful machinery of a certain basic design was assumed to have a length of life of about forty to fifty years from its very first idea to the time when the last production unit will be abandoned due to service economical considerations. If we compare that lifetime to the knowledge and facts in our possession today on the future oil and energy supply facilities we could immediately state that the conditions for a lifelength of machinery and evolution as such, must be and have been strongly affected by what has happened since 1973 and as well might happen after 1983.

2 FUEL CONSIDERATIONS

As a marine machinery utilizes fuel which to a great extent also is used for power generation and heating it has been possible to take advantage of the vast range of fuel qualities and quantities marketed. However, the marine transportation only absorbs two to four per cent of the total hydrocarbon production all over the world and therefore the qualities available for marine transportation purposes in the future must be determined by the way how other and more dominating consumers of oil products are going to develop their future needs, both as far as qualities and quantities are concerned. We have therefore from the mid seventies experienced a quick deterioration in fuel quality for marine purposes as well as a steep increase in price. The stability of the situation which was referred to earlier has drastically changed during the later years, which has made the basis for estimations and predictions far more difficult today than even ten years ago.

The general situation when it comes to the pricing of the different fuel qualities did tend during the earlier period to form a price difference between bunker sea and marine diesel fuel of roughly fifty per cent and additional cost of twenty per cent for a gas oil (see Diagram 1). That means that if the bunker C costed onehundred units the marine diesel fuel costed one hundred and fifty and the gas oil costed one hundred and seventy. During the seventies the price relation was far more turbulent (see Diagram 2). At the time when the crude price per barrel was about two dollars the different refinery processes might have given sufficient reasons for that price difference, but it could also have been influenced by the fact that to keep a stimulating competing climate between the steam turbine and the direct coupled diesel engine, a price variation of that kind might have been useful to build up an equilibrium inbetween transportation cost using steam turbines and diesel engines, considering the lower efficiency of the steam turbine, considering the superior ability of the steam turbine at that time to burn even the worst fuel qualities, and also completing the cost situation with a price level on lubricating oil which the diesel engines used to a rather large extent just to make it possible for both machineries to compete on slightly equal terms. My own opinion is that the latter reason has been the dominating reason especially when you consider how the price development of the relative prices of the different qualities has been during the later years.

During the sixties the dominating main propulsion systems, the steam turbine

and the diesel engine, were supplemented by other converters of thermal energy, the diesel engine by the medium speed geared diesel engine and the steam turbine by the gas turbine, either derived from the jet engine type or from the industrial gas turbine. These three new alternatives in conjunction with the two earlier ones made further prognostics still more complicated than they else would have been, but even if we go back ten years we would see that the tendencies which could be recognized even then have merely been enlarged today, and that the real competition will not be inbetween all those five but inbetween two of these five in the future.

3 ECONOMICAL CONSIDERATIONS

In order to be able to get some further information about what the future trends might be, the economical situation has to be more thoroughly analyzed, not only the direct or the indirect machinery cost may be taken into consideration but also all foreseeable facts effecting the economic picture. In accordance with praxis from other tends of technical-economical analysis the basic costs are split up in the following four main categories:

1 Capital costs and costs related to them. Our approach refers here to costs for interest and amortization of the ship including its machinery. The same costs for container equipment if any, insurance costs, port fees, and miscellaneous costs.

2 Service costs. Here we refer to costs for crew, service maintenance of hull and containers as well as service and spare parts costs for machinery.

3 Fuel and lubricating oil costs. This means fuel for machinery, main machinery and lubricating oil, as equivalent machineries are being compared where only the main machinery differs in the individual case auxiliary power requirements have not been considered.

4 Cargo capital costs. An average cargo value has been assumed, interest for which is charged as cost during transportation, thus pushing the economical transportation optimum towards higher speeds or lower speeds all depending upon what kind of freight goods there are.

It will very quickly be realized that the price of a ship is far from being stable in time. Particularly during the last decade we have learnt that the large surplus of laid up ships have severely depressed the market to a state where it is very difficult to define the price for a new ship or a ship brought in to service. Not only the price as such is essential, the amortization rate and the interest which has to be paid is also extremely influential on the total cost. The subsidizing of the ship building industry in different countries has on top of that created a situation where the connection to the governmental bodies is as essential as anything else in the shipping industry. The financing of the ship and the bookkeeping value of the ship is a very essential part of the cost for the transportation and to a certain extent the financing of the ship including any subsidizing might be more essential than the choice of the main machinery. But, in the long run, when a healthy economy will be achieved hopefully in the future, the realistic different costs are very essential for a judgement how to build up a fleet which will be competitive in the future. The following approach which will be made is therefore devided into segments. The first one will in detail discuss two different ships, one 80.000 ton tanker, and a 1500 container ship where the conditions from today minus ten years are considered. The early seventies was the last period when stable conditions were prevailing. We will there, in those two ships, analyze the total cost for their utilization in the way which we earlier described and find ways to define the minimum transportation cost under the different conditions. We would anyhow avoid to have the cost in absolute figures and we will try to get them in relative figures only. When establishing calculations as relative calculations instead of absolute calculations, a large number of parameters controlling the propulsion power requirement to different hulls might be eliminated as well as the varying influence of the propeller arrangement on the propulsion efficiency on the main machinery. In order to maintain good reliability for the calculations the hulls, speeds and so on are chosen from actual ships being in service. The variation in economical optimal speed is calculated from the varying conditions considered during the study.

4 INPUT DATA ON THE MACHINERY SIDE

In order to be able to compare the influence of different machineries

on the operational characteristic of ships during transport the characteristics of such machineries must first be determined as realistically as possible. The input data chosen can be seen from Table 1 where the characteristics generally accepted during the first part of the nineteenseventies for directly driven and medium speed diesels, steam turbines, light gas turbines and industrial gas turbines have been indicated. As far as prices per horse power of installed output are concerned they have been expressed in the table as cost fractions of the direct driven diesel machinery cost, which in the calculation had been assumed to be 100 US $ per horse power for the machinery and 108 US $ per horse power for the supplementary machinery. The latter figure in this programme is reduced by 50 per cent for the portion of the output which is required for propelling of the ship by more than 16 knots. In relation to this price per output the medium speed machinery has been assumed to be 15 per cent cheeper, the steam turbine 5 per cent more expensive, the light gas turbine 10 per cent more expensive. In this complex it must be noted that it has been assumed that an extra gas generator must be available as replacement unit on board complied with requirements of the availability of the light gas turbine ship. Finally the industrial gas turbine has been judged to be about 20 per cent more expensive than the directly driven diesel. These values have been chosen in accordance with literature studies and interviews from the early seventies, but also here there is a relatively large fluctuation between actual costs and costs charged due to sales situation and financing condition. It is obvious that the higher the outputs are the more decreasing specific weights are likely to be obtained in most machinery cases. From the complicity point of view, anyhow, the indicated specific weight for the total machine has been adopted straight through the calculation. The thermal efficiencies have been indicated in a relatively conservative way, and the transmission efficiencies have been assumed as being 1 per cent loss for the shafting, and 1 per cent loss for each gear step. The relative propulsion efficiency has been set equal to 1.00 and will only be reduced for the directly driven diesel engine in the large tanker by 7 per cent and for the smaller tanker by 3.5 per cent compared with the gear optimalized propeller speeds. The fuel quantities have been so evaluated that the directly driven diesel engine and the steam turbine are able to burn bunker sea without difficulties, and the medium speed engine without difficulties can burn 1000 sec Redwood I ('RI), while the light gas turbine outfit ought to utilize qualities between marine diesel

oil and gas oil. The fuel spectrum of the industrial gas turbine lies between 1000 'RI and Marine Diesel Fuel. The relative fuel prices are also indicated in the table and it is well known that the diesel engine and, to a certain extent, the steam turbine are burdened by lubricating oil consumption. If this is converted into fuel cost .5 g/hPh cylinder oil and .33 g/hPh system oil correspond to an increase of fuel prices of 14 per cent and 1 g/hPh cylinder oil for the medium speed engine will be turned into 19 per cent of the fuel price. The steam turbine is only burdened by 1 per cent increase due to lubricating oil consumption. Finally service maintenance and spare parts costs have been evaluated in Table 1 from literature and other sources. They have been related to costs per horse power for the whole machinery and are stated in millions of these per service hour. It must be pointed out that the directly driven and medium speed diesels as well as the steam turbines are based on statistically founded figures while a light and industrial gas turbine mainly has estimated values as base. Experience gained during the last years from light gas turbine plants indicate however a quadrupling of these values but these might not be seen significant for the future.

5 INPUT DATA ON SHIPS

This type of machineries has then been adopted for the ships defined in Table 2. One of those are a tanker, while the others represent container ships, able to carry 1500 containers. The table indicates maximum dead weights installed machinery and output for continuous service rating. It also indicates the service speed, number of crew member, etc., as well as typical transportation routes and the probable cost for such a ship in the early seventies.

6 INPUT COST DATA

In order to be able to convert crew cost, insurance cost, etc. into annual amounts, these have to be tied to the ship capital cost as can be seen from Table 3 in accordance with estimation published. Basic cost for bunker sea at that time is set to 14 US $ per ton. The basic capital value for the cargo during tanker transportation has also been set to 14 US $ per ton

while capital values during container transport have been set to a hundred fold thereof, i.e. 1400 US $ a ton.

7 OPTIMIZATION SEQUENCE

In accordance with approach made now the earlier defined portion of the transport cost for a given hull is estimated with variations in transport speeds around the service speed projected for the original ship. It is assumed that the ship maximum cargo capacity has been utilized after machinery weight and bunker requirements of the originally chosen machinery has been considered. As a result of that an optimizing curve is obtained with transportation cost per nautical mile cargo ton as a function of the ship speed. It must be pointed out that in connection with speed variations the machinery output is adopted to this so that the main machinery output varies just as its weight and bunker requirements, capital costs etc.. As the cargo capacity of a given hull is affected by the variation in machinery weight the transportation optimum is directly and indirectly given by the capital density of the cargo. At speed variations also the hull prices are varying by about 3.5 per cent for each knot of variation in speed. A deviation, if any, between the optimum point that is calculated and the order points is probably in the first place due to pure transportation cost optimum having been calculated and the relationship between transportation capacity and cost per nautical mile not having been considered. Any deviation might also indicate the realism or lack of realism in the data utilized for the technical-economical situation as well as the assumed capital density. The same optimum curve is devised for the alternative transportation distances indicated in Table 2. Only variation in optimum point positions have been indicated on the curve for the medium distance in the diagram series in Diagram 3. The optimum curve which the ordinary machine has obtained with a variation of transport distances is now completed with four other machinery types in each one of the two ships as well as variations in transport distance are indicated by plotting on the optimum position for the longer and shorter distance respectively.

8 OPTIMIZATION RESULTS

What can be seen from the calculated graphs dealing with conditions about ten years ago. First of all the difference in cost between steam turbine machineries and the direct coupled diesel engine is small. Allthrough the study of these ships and others we have considered in no case the transportation cost difference was more than 1 per cent. Changing the routing the cost for tanker and container ship will give different indications. A reduction of the transportation distance for tankers will increase the transportation cost as the total time in port will increase while for the container ship the reduced bunker quantity influence will dominate the change in port time and reduce the transportation cost. The influence on container ships is also essentially stronger. When looking at the more recent alternatives the base values from Table 2 show a marked advantage for the medium speed engines in the order of the magnitude of 3 per cent compared with the superior ones of the two established machinery types. On the other hand the gas turbine machineries show in all cases less profitability compared with steam or diesel, in the container ship 3 per cent higher transportation cost and in the tanker ship 6 per cent higher cost. In the tanker case the difference between the light gas turbine and the industrial gas turbine is not significant but in the case of the container ship the values chosen give the industrial gas turbine 1 per cent in advantage compared with the light gas turbine.

9 VARIATION IN INPUT DATA

The conclusions made here are not to be assessed more value than what the adequacy of the input data permits for which reason the variation control of the input data has been made as per Table 4. A safe conclusion can anyhow be made from the calculations. Optimizing implies adoption of the optimum speed to the machinery and as for a given hull and displacement the power requirements are assumed to vary with the third power of the ship speed. Also the output is affected by the choice of machinery. The variations are well seen in the lower series of curves in the diagrams. The result of the mentioned variations of the basic data is illustrated in Diagram 4. Doubling of service and spare parts costs as well as lubricating

oil costs for a direct coupled diesel immediately balances off the difference compared to the steam turbine while the reduction of efficiency reported for the steam turbine during a ten year period, here marked by 11 % increase in fuel consumption widens the gap between the two established machineries beyond what has been shown in the first approach. The same type of change was made for the geared machinery in addition to an increase of the capital value from 85 to 95 per cent of the cost of the direct driven machinery. Even if the increase of the cost level gives a strong indication, the total effect of the increasing capital costs as well as a doubling of service and lubricating oil costs will not increase the cost level to the direct coupled diesel engine range.

As far as gas turbine machineries are concerned the fuel quality assumed in the beginning for the industrial gas turbine was reduced from marine diesel fuel to 1000 seconds Redwood 1. All other characteristics remain unchanged. This did not reduce the costs to the earlier stated range of the steam turbine in the tanker case but in the container case an overlapping achieved at near a somewhat higher speed in the gas turbine case. The light gas turbine which has been assumed to be able to burn marine diesel fuel requires, however, an anticipated increase in efficiency to 34 per cent maintained fuel qualities to reach up into the steam turbine class for container ships. If such an increase in efficiency would require jet fuel the competitiveness would be severely limited. The reported increase in cost for service and spare parts during the first year of operation will affect the optimum position but not more than if the higher efficiency can be obtained with marine diesel fuel, the alternative would still be of interest.

10 SUMMARY OF THE SITUATION TEN YEARS AGO

Summing up the results obtained by the chosen basic data and their variation, it can be said that no especially surprising results have been indicated than possibly the potentiality of the geared diesel machinery. As, however, the aim has not been to analyse the present competition situation at that time and the correctness of the basic data chosen does not permit too strict evaluations it can only be said

that the results indicated with further variations as to relative costs and specific weight might be carried to a still more reduced difference in cost. That indicates that the situation ten years ago was in favour of the medium speed engine according to these investigations and gave a rather even situation between the steam turbine and the direct coupled diesel engine. On the other hand the gas turbine did prove to have large difficulties in being able to compete. Before going into the situation of today we could play with the results already achieved. We could from the now established optimum points with the five different prime movers let the fuel cost increase from two fold to four fold to eight fold up to twelve times the present cost level. We see there that the cost for transportation with ship B and ship D increases for B more than 50 per cent and for the container ship roughly 35 per cent. We could see that the difference between the direct drive diesel and the medium speed diesel has diminished but also that the relative competitiveness of the steam turbine has been a little diminished compared with the earlier situation. The gas turbine cases are becoming so expensive that they will have difficulties really to compete. When considering the really highest price increase the optimum speed has been reduced by about 30 per cent and the installed horse power has been reduced by more than 50 per cent. When considering the cargo capital density increasing with the fuel price the optimum speed will develop as to Diagram 6. We could also play a little with the financial side and consider that the life expectancy varies from fifteen to five years with tempers and interest and that does increase the cost and asks for a reduction in speed when the life time is decreased. If on the other hand the short life time five years is maintained and the interest increases to 15 per cent we have a sharp increase in transportation cost and an increase in speed and installed output is definitely there which means that the installed horse power is not only depending upon the cost of fuel but also upon the financial situation and that a high interest level does demand a higher speed than a low interest area. Finally the transportation cost development depends on the layout of the machinery output as to Diagram 8.

11 THE PRESENT SITUATION

Having considered the situation ten years ago with the relatively well defined costs for crew, fuel, interest, financing, and so on, we could try

to project the model we have to the present situation. We know that the price increases on the different new buildings are different. Let us for simplicity assume the following increases: ship prices have increased by 150 per cent, as well as engine prices. The cost per man-year has increased by 250 per cent but that is partly counteracted by reduction in the crew number by roughly 25 per cent. The fuel cost has increased by 1100 per cent and the relative cost per hp of medium speed engines has decreased compared with the direct coupled diesel engine from .85 per cent of the direct driven in 1973 to .70 in 1983 and, last but not least, the efficiencies of both the direct coupled diesel engine and the medium speed engine have been assumed to be 50 per cent which has been achieved recently. Considering only the ship, engine and man-year costs the average increase is by 183 per cent. If the already mentioned changes are applied to the present situation on B- and D-ships we would find that from 1983 the B-ship speed should be reduced by around 12 per cent and the cost being increased by roughly 3.3 times. For the container ship (D) the speed demand has decreased to about 84 per cent of the earlier state, the cost increase is roughly 2.7 times. The relative values are defined in Diagram 9.

In the following argument around the present situation we do exclude the two gas turbine cases and look only upon the steam turbine, the direct driven diesel engine and the geared diesel engine. We have now reached the basic point corresponding to the today's cost level that means that we now have ships of 1973 year's model with 1983 year's efficiencies, with 1973 year's interest and with 1973 year's service and maintenance costs, and with the relative bunker cost of 1983. That means that we could study some further parameters to find out what the competitive power of the three different machineries would be in the present situation.

First of all in Diagram 10 you see the optimization of the hull with the three different engines. You could directly see there that the difference between the medium speed engine and the steam turbine has increased for the tanker up to about 12 per cent and for the container ship it is about 9 per cent. The difference between the medium speed engine and the direct driven engine is about 4 to 5 per cent on the tanker and about 2 to 3 per cent on the container ship. The theoretical advantage of the medium speed engine has thus been still increased during the past ten years even if the direct driven diesel is still dominating. On the other hand you could state that very few steam turbine ships have been ordered in recent time and that

that market is far more limited than what it was in the early seventies. In Diagram 11 you could see the influence on service costs. As mentioned earlier the service cost is related to the buying cost of the engine as part per millions of that per service hour. The nominal service cost has been step by step increased according to a logarithmic scale and it could be directly seen that the service costs for the tankers if it would be increased by 16 times its normal value will not cover more than 5 to 6 per cent increase in transportation cost and on the medium speed engine the influence is slightly less. On the container ship the values are lower but it should be remembered that the difference between the medium speed and the steam turbine has a larger magnitude than what these very severe increases in service costs would bring with it. In Diagram 12 a variation around the present fuel cost represented by point no 2 is given. The fuel cost range goes from .75 to 1.50 times the present level. That wide range of fuel cost does not impose more than about 25 to 30 per cent increase in transportation cost but imposes strong reduction in speed on both the tanker and the container ship which means a far stronger reduction in installed horse power.

Finally, we have studied the financing and from Diagram 13 you would find what a variation around the present conditions 10 per cent interest and 10 years amortization means. If the interest rate is increased to 15 and 20 per cent and even 25 per cent it imposes cost increase on the direct driven diesel by about 50 to 55 per cent on the tanker and on the container ship about 75 to 80 per cent increase. If the amortization period extend to 25 respectively 50 years a slight relaxation could be traced and is indicated on the container ship side, but as could be seen the interest rate as such is a dominating item. That leads directly to a question. How would the annuity influence the cost of a ship. We have in Diagram 14 indicated an annuity varying from 2 per cent meaning an annuity of about 10 per cent on a ship being bought at 20 per cent of its normal market value up to 32 per cent meaning that you would like to have your ship amortized in 4 to 5 years with the present interest rate. As could be seen the influence on the transportation cost as well as on the transportation speed of the given annuity is tremendous. From the present basic value indicated by the points 4 (10 per cent interest and 10 years amortization) the transportation cost could drop 40 per cent or increase 130 per cent which indicates that the economical handling of the purchase of a ship on the running cost of a ship is today extremely essential.

It gives directly the background for the present situation on the ship market where slightly modified ships being produced at an earlier state but being sold close to scrap-costs keep the freight level so low that new ships can only be amortized in a short time with extremely great difficulties. For a technician this is an extremely sad conclusion of the last diagram, as we could see that,whatever you have done on the technical side, you only had the possibility to influence fractions of what could be done through the financing. On the other hand when the market turns better and more realistic again the advantages and disadvantages, which have been indicated here with different machineries, will play an important role in the future development and the future selection of machineries. We could anyhow, from what we have seen, directly state that the future competition will be mostly between the different applications of diesel engines, which means that we could proceed with some discussion of technical qualities of those.

12 TECHNICAL QUALITIES

We have in the earlier part of this lecture studied the economical situation 1973 and we have reproduced also the present relative situation from 1983. We have there also found what kind of difference in transportation cost we have on a ship with a high efficient diesel engine in comparison with a less efficient steam turbine. We also have found that the real competing situation will be within the diesel engine field and thus it might be worth-while to study some common and dominating problems when laying out the machinery and what kind of service problems could be expected from the fuel deterioration and from the corresponding increase in heat load. The first point should anyhow be the reliability influence on the economical situation. We see that the difference between the two main competitors is of the magnitude of a few per cent. The mathematical model which we have used here enables us to transfer the gain of a per cent in transportation cost into a comparable equivalence of other types, for example availability.

Of course the secondary costs involved in ships being off hire , such as possible harbour fees, costs for repair and damages to the charter

contract, or possible further delays due to crowded situation in ports, can not be considered, but from a purely mathematical/economical point of view we could state that 1 per cent in gain is roughly outdated by two to three days off hire when the profit margin is moderate down to zero as today, and with a higher profit margin on the transportation these days will be diminished to half a day or so. As a conclusion you could say that a few per cent in advantage of the medium speed engine versus the direct coupled engine might mean the acceptance of two to eight days off hire per year, not more but possibly far less, if the off hire period is combined with extensive treatment of the unit or loosing the freight schedule. It means that an installation which shows a good theoretical/economical value must not lack in the reliability but has to be available during at least 350 days a year or more in order to be able to compete at all. As the reliability is very strongly linked with the complexity of the machinery we can from Diagram 16 see that the direct coupled diesel engine is simpler, has fewer transmission components than the geared medium speed engine with not only couplings and gear boxes but also variable pitch propellers possibly included. Of course the utilization of two engines coupled to the same propeller enables the ship to move even if one engine should fail but if the gear box fails the ship is stuck. The highest possible reliability feature is of course included in a twin or triple screw machinery, each possibly consisting of a plurality of engines coupled or gear-boxed to the same. This gives very high availability factors but also rather complicated installations.

As figures on the reliability of ship machineries are rather scarce two arguments have been compiled. In the first, in Diagram 16, the reliability which could be found in the literature from different machinery layouts are indicated assuming failure intensities where every per cent deviating from 1.00 means one critical failure in hundred accumulated ship years. A critical failure is regarded as a failure when the ship has to be towed into harbour for reparation. The published values give a slight advantage of a four stroke trunk engine to the crossed engine but the difference is not big enough to be significant. It could anyhow be seen that the direct coupled crosshead machinery tends to have the same reliability as two medium speed engines geared to a single propeller and that a single engine geared

unit has a lower reliability. The highest reliability has a two propeller machinery of direct coupled engines or a two propeller geared engine where each propeller utilizes two medium speed engines. In the next diagram, Diagram 15, it is assumed that the required reliability for the main engine, the transmission line, and the propeller, that means a system reliability shall be .95 with an assumed reliability of each component in the transmission line of .99. The necessary reliability of the prime mover to reach the .95 in total varies between .989 and .562. The point 562 is what is expected from the medium speed engine, where the machinery consists of two propellers, each with two medium speed engines. The engine reliability demand has thus been cut down to an extremely low value not sacrificing the machinery total reliability, and I assume that that fact will be an extremely interesting fact for the future selection of main machineries.

13 HEAT TRANSFER ARGUMENTATION

As a diesel development has been concentrated on two-stroke large bore low speed engines and four-stroke medium speed engines it might be of interest to study, if there are any inherited advantages or disadvantages of the two types. This can possibly be made in a very simple manner. As the internal combustion engine has a closed working medium as soon as scavenging ports or valves are closed the pressure inside the engine is directly linked to the temperature inside the same. This means that as soon as a pressure diagram is given the temperature diagram can be derived from the same. Using simple assumptions about the heat transfer from the working medium inside the engine like the Eichelberger formula, this permits us to calculate the instantaneous heat transfer rate as a function of the crank angle. That heat transfer can be integrated during the complete working cycle and the mean value will be established. Furthermore, the heat flow intensity can be calculated. That means that the instantaneous product of temperature and heat transfers which value also can be integrated and be given a mean value. By dividing the integrated heat transportation intensity with the integrated heat transfer capacity an equivalent mean temperature for the working cycle will be established, which temperature together with an average heat transfer will act in a heat flow matter exactly as the chosen working cycle.

This means that we not only for a two-stroke engine but also for a four-stroke engine can calculate the average heat transfer value and the equivalent temperature of the cycle. Due to the scavenging stroke of the four-stroke engine, the mean temperature as well as the mean heat transfer rate will be lower on a four-stroke engine. If we carry these calculations through for the two-stroke engine of today's general layout, with the current accepted maximum pressure and mean effective pressures we can choose mean effective pressures on the four-stroke engine giving the same output per piston surface area. This means that the mean effective pressure of the four-stroke engine shall approach the double mean effective pressure of the two-stroke engine. If we do this, we can first study how the temperature flow through the two stroke engine will look when you have a given dimension of the material dividing the combustion space from the cooling medium. In the chosen scale with equivalent heat flow capacities the temperature gradiant will be a straight-line through the cooling medium flow, through the metal part and through the heated side to the gas temperature. If we now double the wall thickness we find that we can easily assume that the thermodynamical cycle does not know what kind of size of engine it is working in. Through the fuel injection equipment, combustion intensity and so on, all designers are trying to keep the working cycle equivalent in spite of changes in dimensions. We will from Diagram 17 find that the temperature drop through the material is increasing, even if the increase is not proportional to what it was in the thickness one case as the total heat flow resistance from the equivalent temperature to the cooling temperature has been increased by increase in metal dimension. If we make the same with the four-stroke engine, Diagram 18, we will see that through the smaller heat transfer rate and through its smaller equivalent temperature, the temperature gradient will be smaller through a given thickness of material between the cooling medium and the working medium. If we increase the thickness in the same manner as we did on the two-stroke, the temperature gradient will of course increase but also here at a smaller rate than in the original part. If we assume the given size in the two-stroke and the four-stroke engine in the Diagram 19 we will find that we have lowered the temperature gradient and thereby the temperature of the inner surfaces as well as some stresses in the material by about 20 to 25 per cent by turning from two-stroke to four-stroke. If on the other hand we consider that in praxis we are comparing

two-stroke engines of sizes 1.0 and four-stroke engines of sizes .5 we will find that we diminish the surface temperature of the heated part of the inner side of a four-stroke engine by more than 50 per cent compared with the temperature of the two-stroke engine. This will in turn lead to lower stresses and also to lower temperatures on the surface to be lubricated or kept kleen. It has to be kept in mind that part of those margins can be utilized in different designs by increasing the maximum pressure in order to reduce the fuel consumption or by adopting still higher mean effective pressure than the double in order to reduce the material involved in the prime mover, but basically two similar engines given the same specific output per piston surface area give great advantage from a thermal stress point of view for the four-stroke engine. We have now found a case where the two-stroke direct coupled diesel engine has advantages and another case where the four-stroke engine has advantages but we have not yet studied the energy release question and how it might be influenced by the future fuel situation in the marine world.

14 INFLUENCE OF FUEL QUALITY

Before we go into that I would repeatedly point out that the marine sector does utilize 3 to 4 per cent of the total petroleum production in the world. That means that the marine sector is not a dominating part but has to adopt itself to larger consumers of intermediates and heavy fuels which are the heating and power generating sectors. With the present tendencies towards a limitation of the crude oil supply in the future we have to assume that the demand for lighter fractions from the barrel will increase at the same time as the power production and heating gradually will be taken over by other types of energy sources. As long as the internal combustion engine remains a dominating prime mover for road transportation, larger parts of the barrel will be absorbed by those types of consumers. That means that what remains in the barrel and will be available for marine transportation will gradually increase in viscosity and decrease in availability as well as quality as any contamination in the crude will be more and more concentrated in the marine fuels available. That leads us to look for a simple description of what happens in an internal combustion engine when the fuel is injected.

The P-V diagram indicates a given environment during the upper turning point of the piston, an environment which is given by the scavenging pressure and the scavenging temperature as well as the scavenging efficiency to which the engine is subjected. On top of that we have the compression ratio with defined increase in temperature until the fuel is injected. We will assume a given compression ratio and a scavenging temperature and pressures of today's standard. We now introduce a droplet of fuel into this environment. If we assume a given rpm of the engine, this means a given angle of velocity. We also have a time included in the Diagram 20. However, the droplet which is introduced at a given angle before the top dead centre is subjected to a heat transfer which can be calculated and that heat transfer induces a temperature on the droplet more or less quickly. We can see that the smaller droplet below 40 μ is more or less instantaneously given high temperatures but bigger droplets need longer time for heating. We can from Diagram 21 see how a droplet of a given size will be heated under different speed conditions. The speed is changed in logarithmic steps and as can be seen in angular time the heating of the droplet is more rapid when the speed is lower than when it is high. This means that the time available for the heating, the evaporation, and the combustion, is more generous on the slow speed engine which of course is well-known, but if we turn back to the same typical droplet and consider this droplet to consist only of fuel of only one fraction, i.e. one single hydro carbon chain type, then we can assume that the barrel is relieved from the lighter fractions and that this droplet will gradually consist of heavier fractions and in Diagram 22 it is shown how far different hydro carbon chains of paraffin type have to be heated in order to give a partial pressure of fuel in the surrounding air, compiling with the criteria of combustion mixtures. Thus the droplet has to be heated to a temperature where the evaporation rate is large enough to form a combustible mixture. What is seen here is anyhow that when we assume that the droplet size consists of a chain of 12 to 20 carbon atoms they will with normal droplet sizes very quickly get heated and form a combustible mixture but when we go to intermediate fuels up to heavy fuels to chains of 50 to 60 carbon atoms, the temperature in the air fuel mixture itself tends to be too small to give

the necessary droplet temperature. This means that the physical ignition delay will be increased and statistically when the time sequence during which you have your energy release will be elongated. That can of course be cured by utilizing a higher compression ratio in the engine. A higher compression ratio gives the higher peak pressure, and a higher peak pressure will very quickly influence the thermal load of the engine, while at the same time it demands a thicker surrounding around the combustion space as well as it gives an increase in the heat transfer rate.

The simplest solution is of course to control the droplet size and to reduce the droplet when the heavier fuel is introduced, but at the same time this means a demand for better control of the injection equipment, higher injection pressures, and possibly higher wear rates on the injection equipment, which might influence the reliability and the stability of the combustion system in itself. Of course the delayed combustion in a diesel engine will mean combustion on the back of the expansion diagram, and higher load on exhaust valves etc which in turn opens up the material to corrosion by vanadium pentoxides or sodium or other types of contaminations.

15 CONCLUSIONS

We can as a conclusion say that a general economic picture for transportation ten years ago did favour the diesel engine in comparison with other types of prime movers. The conditions of 1983 will continue to favour the diesel engine of the two types, even if the fuel situation will give more and more pronounced availability problems. The diesel engine as such has reached a very well developed stage giving efficiencies not reached earlier, and through that it has attained a very high competing power, but it has also to be given a very high reliability, if possible, as we could see from the earlier discussions. The better economy can very quickly be ruined by a few days off hire a year, especially if costs and damages are involved together with that. We can find from the reliability point of view that the two-stroke engine as such has a high reliability potential and will only be challenged and surpassed by a multipropeller multi-engine unit. On the other hand a four-stroke engine has advantages from

thermal load point of view which also includes the ability to keep the piston rings in good condition, and other hot parts relatively cool. Finally, when the fuel deteriorates further, the slow speed engine will have the highest possibility to keep intact and to maintain its reliability some years more on the future heavy fuel. The competition from the geared engines will anyhow be more and more severe.

Type	Mach. price / Mach. price DD	Mach. weight kg/HP	Thermal efficiency	Transmission efficiency	Relative propulsion efficiency	Fuel quality	Relative fuel price	Lub. oil cost in % of fuel cost	Service & Maint. Spare Part costs ppM and hour
DD	1.00	75	38 %	99 %	A 93 % B 96.5 % CD 100 %	Bunker C	1.00	14 %	3.0
MSD	0.85	55	40 %	98 %	100 %	1000°R1	1.03	19 %	2.8
ST	1.05	45	32 %	97 %	100 %	Bunker C	1.00	1 %	1.2
LGT	1.10	25	28 %	97 %	100 %	MDF-GO	1.5-1.7	-	1.8
IGT	1.20	35	32 %	97 %	100 %	MDF - -1000°R1	1.03-1.5	-	0.6

Table 1. Machineries studied

Designation	Type	Max DW	Installed output shp CSR	Engine type	Service speed knots	Crew no	Typical cargo routes 10^3 naut. mil	Estimated price 1972 10^6 US $
B	Tanker	80 000	18600	DD	15.8	35	4-7-11	15
D	Container (1500)	29 560	32000	ST	22.0	33	4-9-13	24 + 4.4

Table 2. Ship types studied

Ship	Crew cost US$ year and man	Insurance cost ship+container	Maintenance ship except machinery	Maintenance containers	General costs	Port fees
B	14 000	2 %	1.2 %	-	1 %	1.2 %
D	14 000	1.5 %	4.0 %	33 %	1 %	1.2 %

Basic cost for Bunker C 14 $/ton
Basic value for capital density at transportation by tanker, 14 $/ton
Basic value for capital density at transportation by container ship, 1400 $/ton
Basic value for amortization 10 years, Basic value for interest 10 %

Table 3. Assumed data concerning costs 1973

DD Service Maintenance and Spare Parts + Lubr. Oil Cons. 0.5 → 1.0 → 2.0 times input data

MSD Service Maintenance and Spare Parts + Lubr. Oil Cons. 0.5 → 1.0 → 2.0 times input data
Capital cost 0.95 x DD

ST Efficiency $\eta \rightarrow 0.9 \times \eta_0$

LGT Fuel quality MDF → Jet Ker (1.5 → 2.5 times basic costs)
Efficiency $\eta = 0.28 \rightarrow 0.34$
Service Maintenance and Spare Parts 1.0 → 4.0 times input data

IGT Fuel quality MDF → 1000°R1 (1.5 → 1.03 times basic costs)

Table 4. Variations in input data (dia. 4)

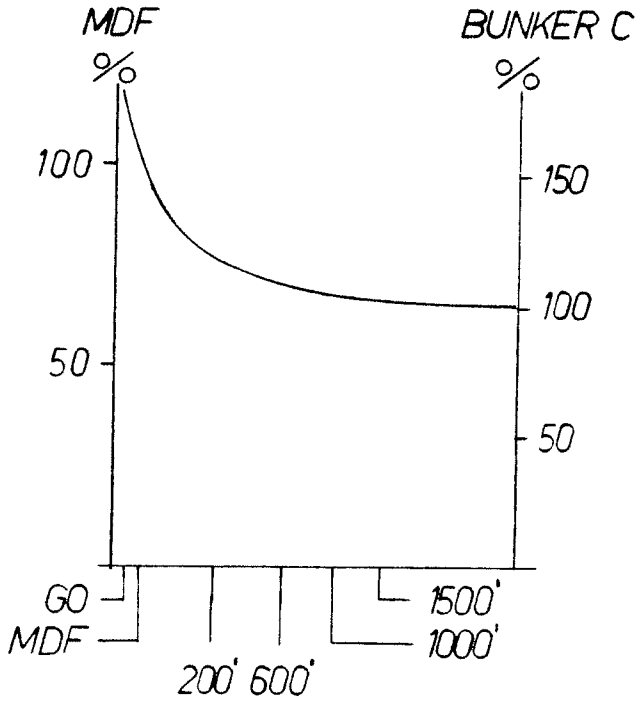


Diagram 1. Marine Bunker Cost-Quality Relationship up to 1975

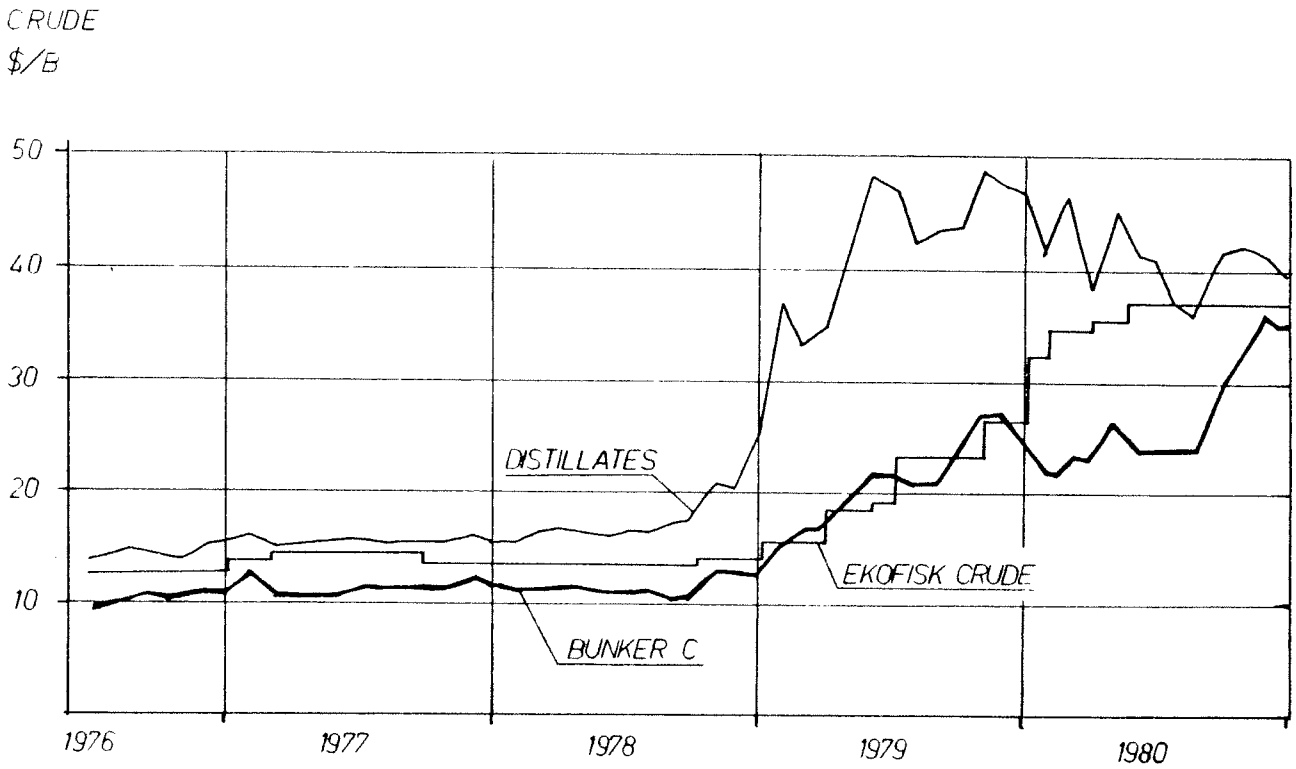


Diagram 2. Marine Bunker Cost-Quality Relationship 1976 - 1980

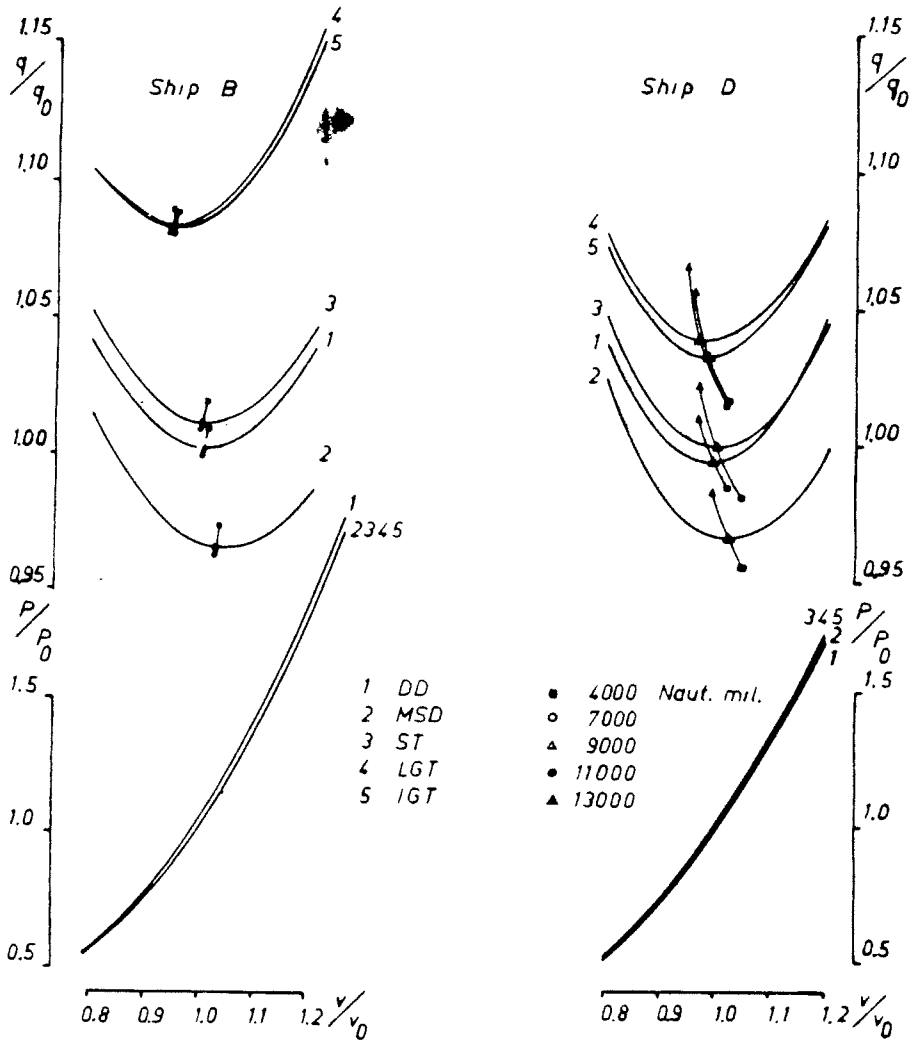


Diagram 3. Optimization as to speed, output and basic transportation cost for the vessels and machineries

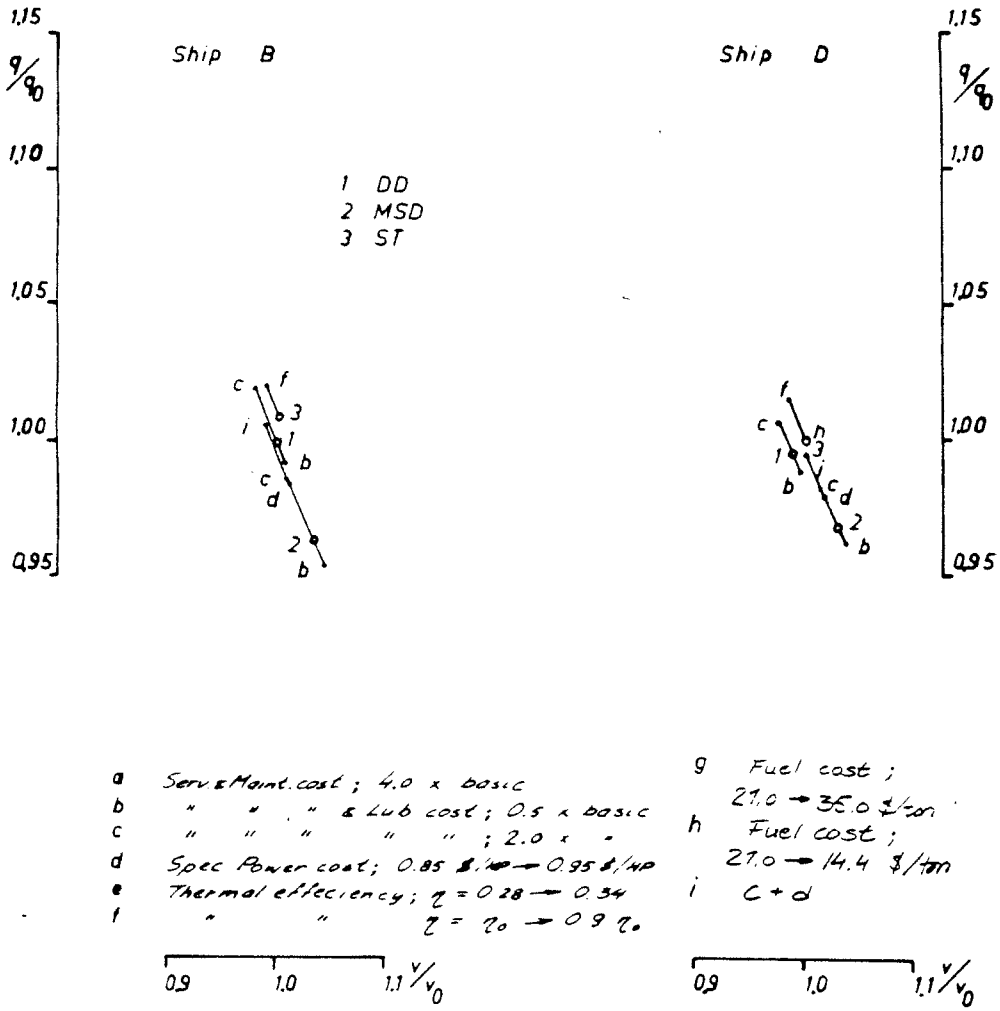


Diagram 4. Variation in optimum basic transportation costs when variations from table 4 are applied

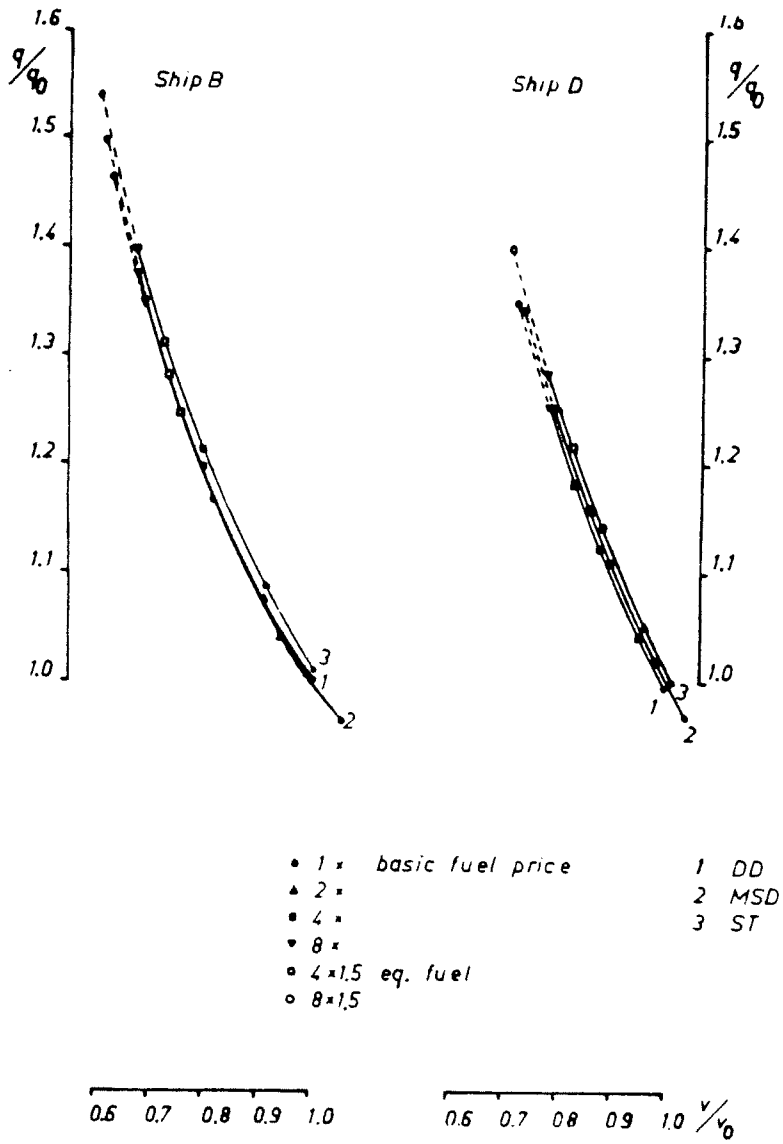


Diagram 5. Variation in optimum transportation costs and conditions when fuel price increases

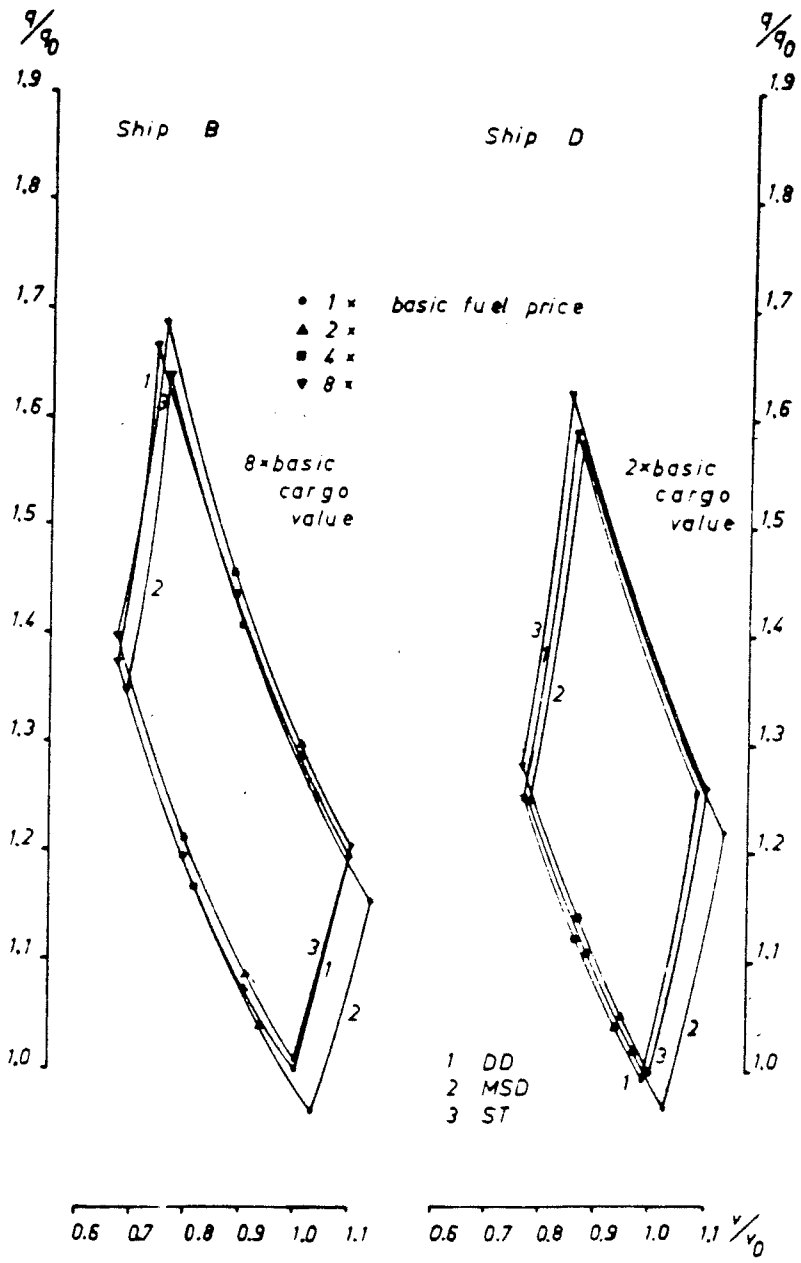


Diagram 6. Variation in optimal transportation costs and conditions when fuel price increase as well as capital density of the cargo

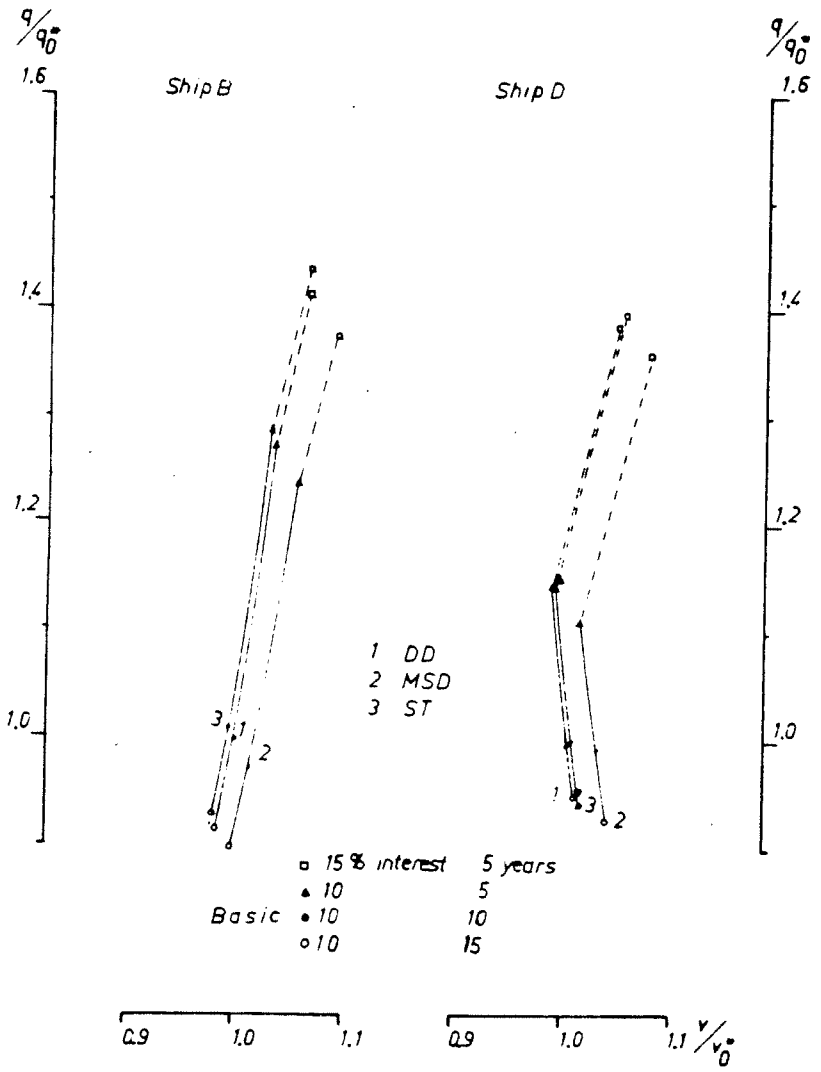


Diagram 7. Variation in optimum transportation cost and conditions when the capital recovery factor changes

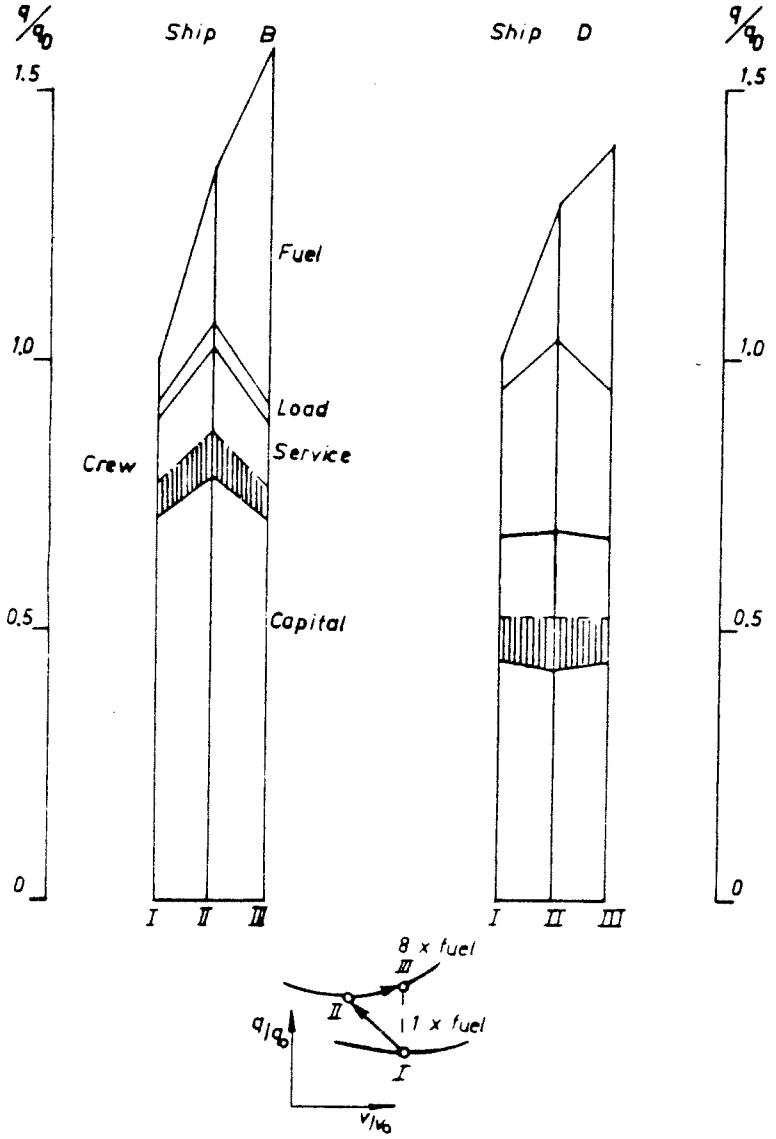
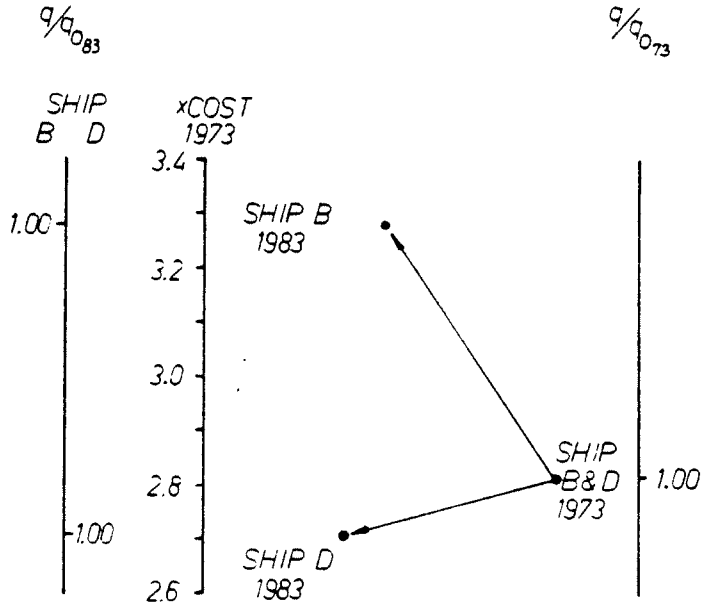


Diagram 8. Cost distribution in the two basic cases when fuel cost rises eight times with optimization of speed or with maintained earlier praxis



$COST_{83} = 2.83 \times COST_{73}$

$SPEED_{83} = 0.905 \times SPEED_{73}$

The assumed variations 1973 - 1983 are:

1 Ship prices + 150 %
2 Engine prices + 150 %
3 Cost for a man year + 250 %
4 Crew number - 25 %
5 Fuel cost + 1100 %
6 Medium speed engine cost 0.7 x DD 1983
from 0.85 x DD 1973
7 Direct and medium speed efficiencies
50 % 1983, 38 - 40 % 1973

Diagram 9. Cost and optimal speed development 1973 - 1983 for the two studied ships

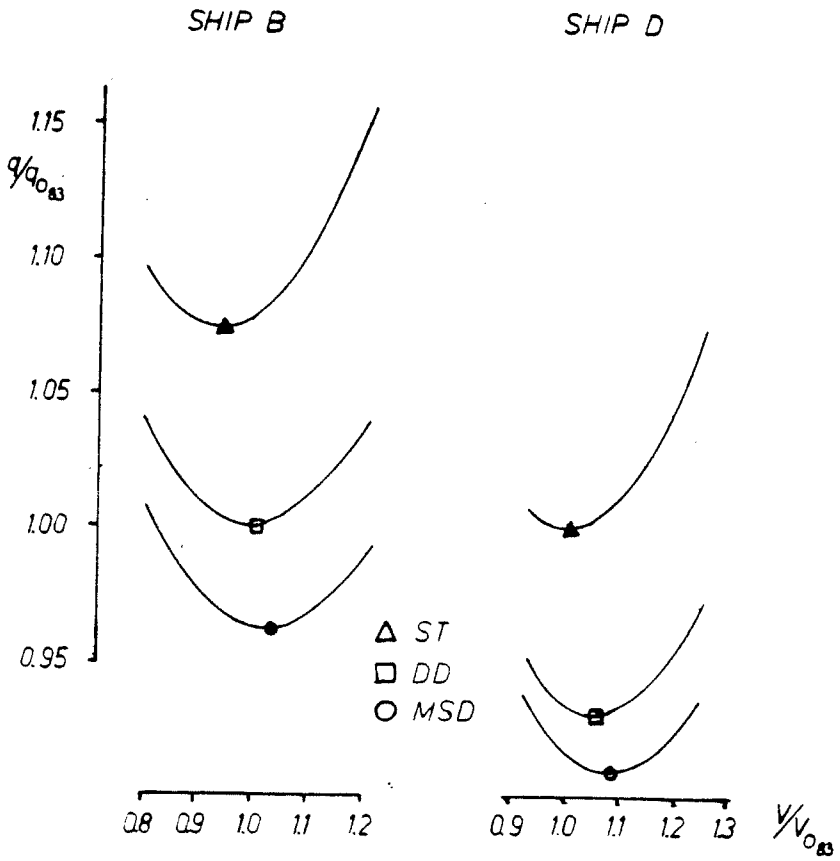


Diagram 10. Optimizing 1983 machinery. The differencies between the three alternatives are much larger than 1973

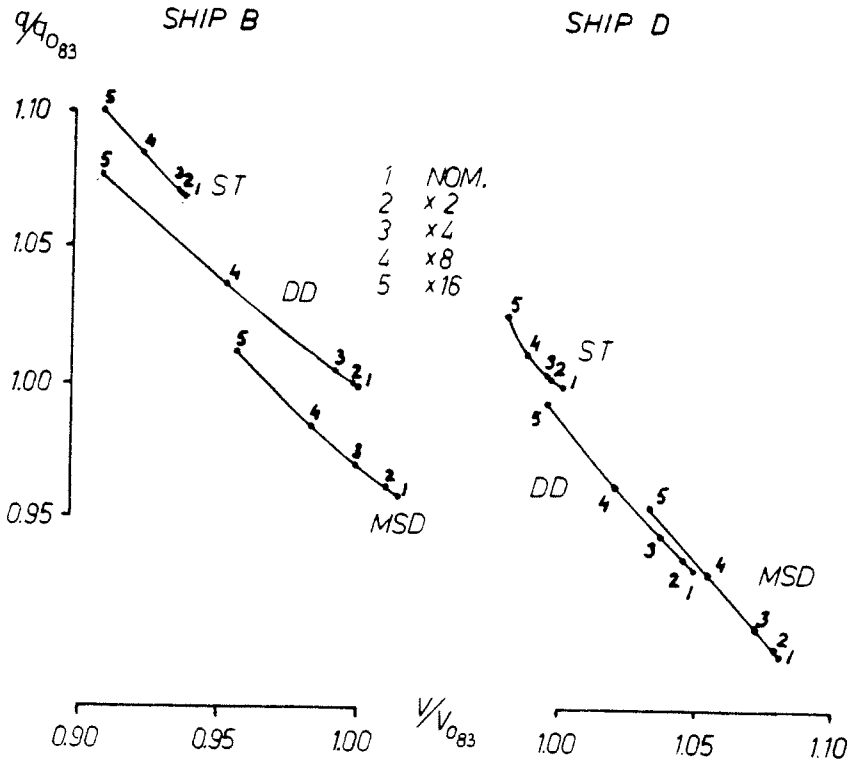


Diagram 11. Total cost and speed layout development when service costs are drastically increased

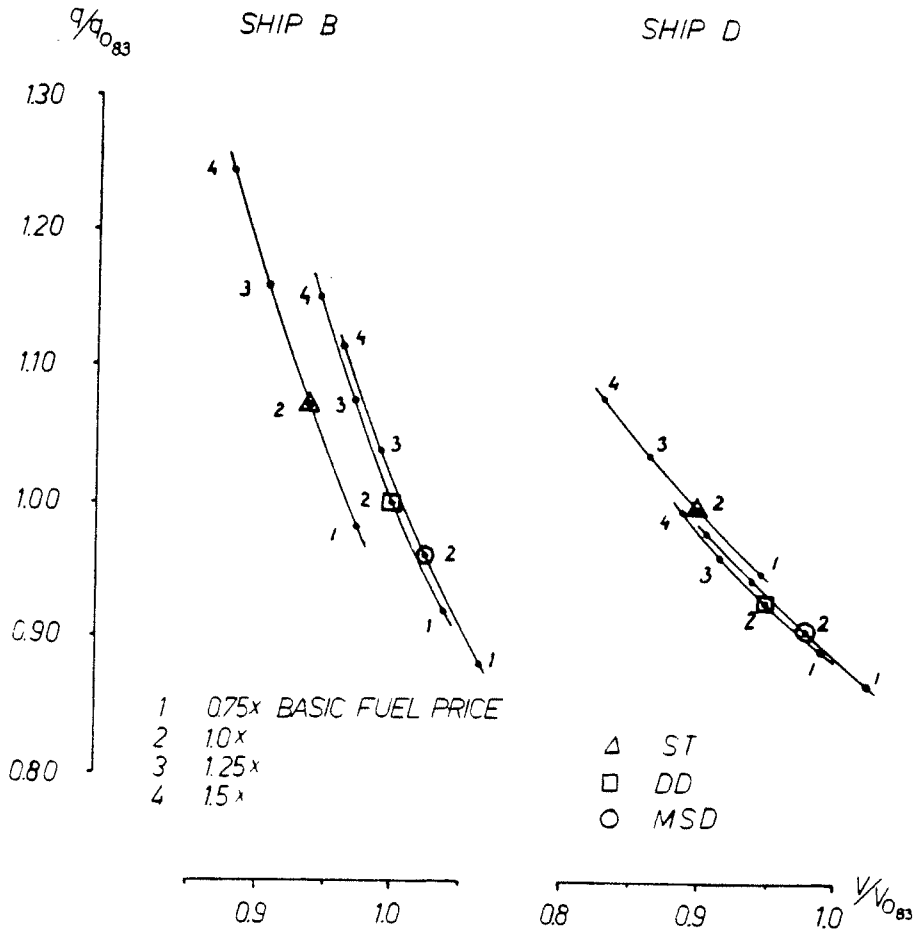


Diagram 12. Speed and cost development lavout as a funktion of further bunker cost variations

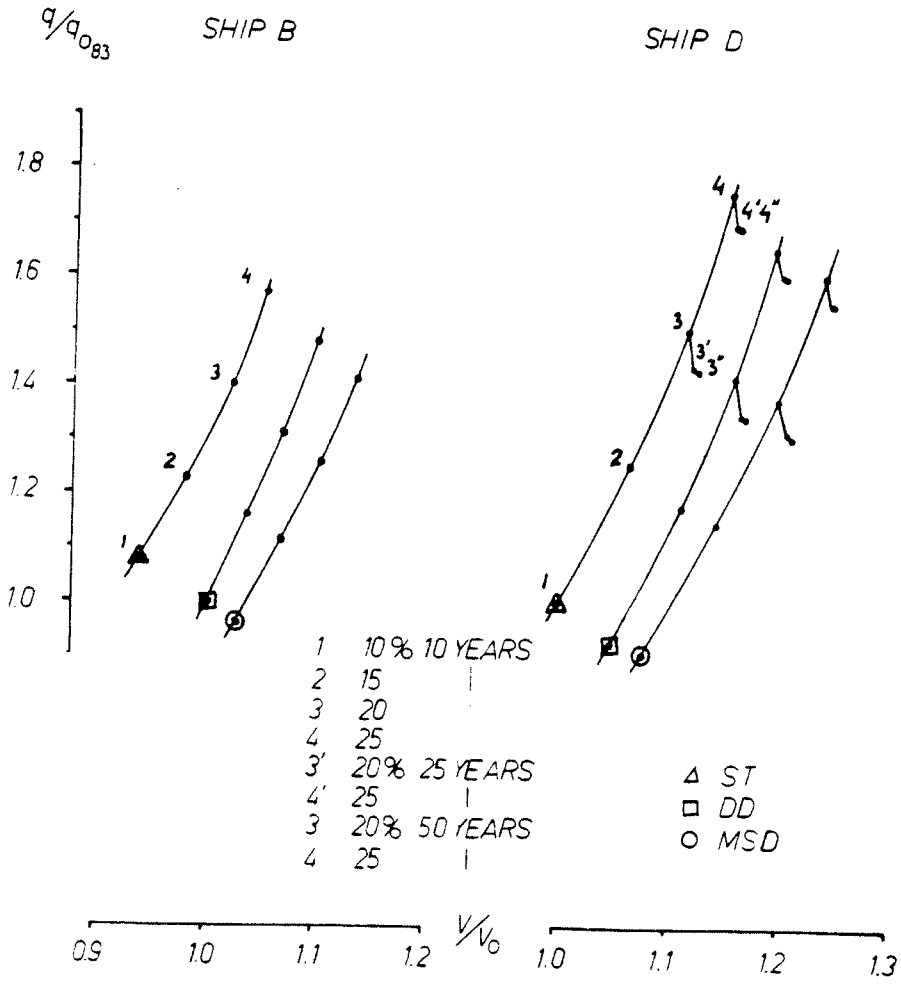


Diagram 13. Influence on optimal speed from the interst level and amortisation period

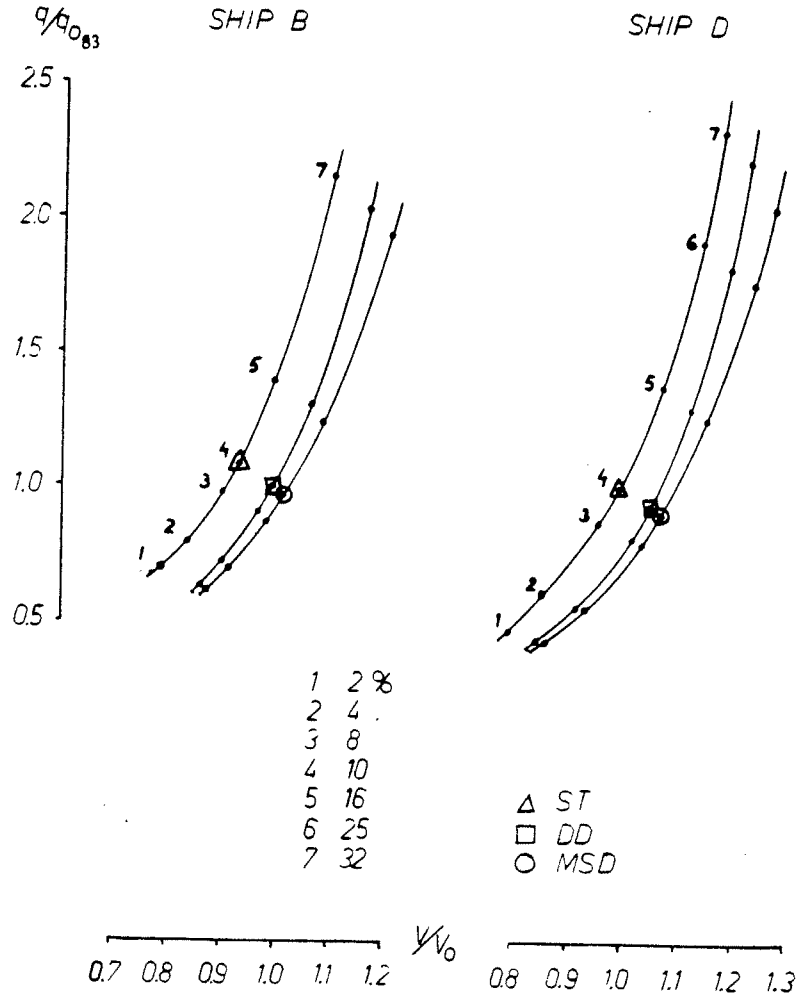


Diagram 14. Impact on optimal speed layout from the annuity

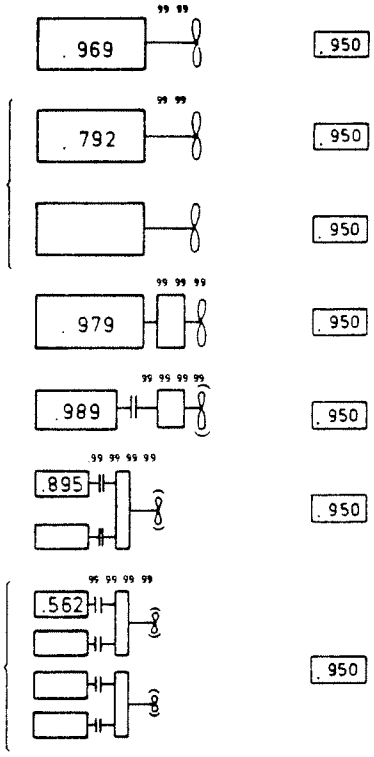


Diagram 15. Necessary engine reliability for equivalent system reliability

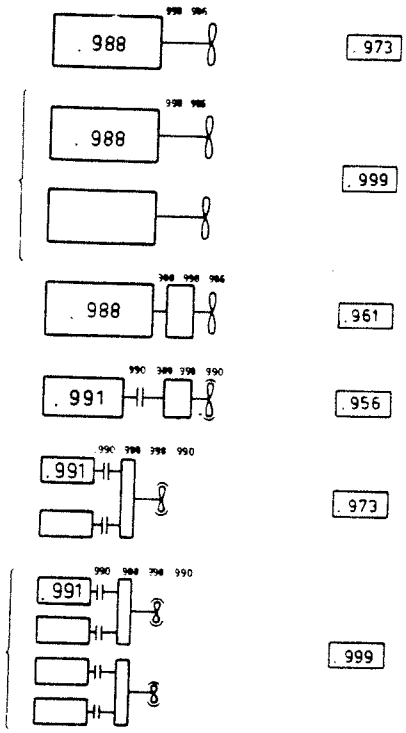


Diagram 16. Total system reliability

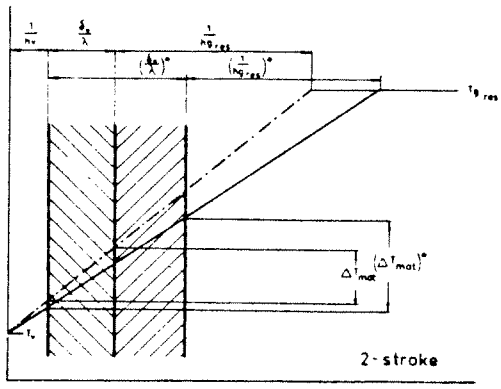


Diagram 17. Changes in temperature distribution with increased wall thickness

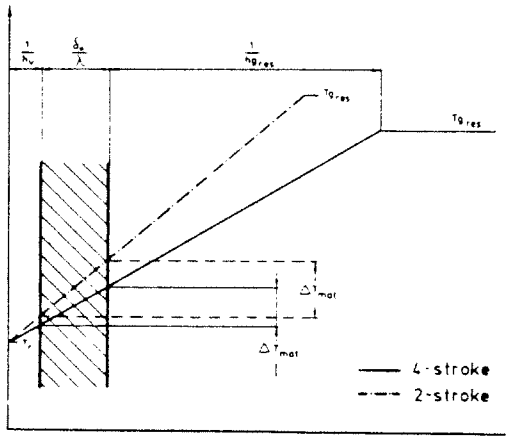


Diagram 18. Temperature distribution comparation of 2- and 4-stroke engines

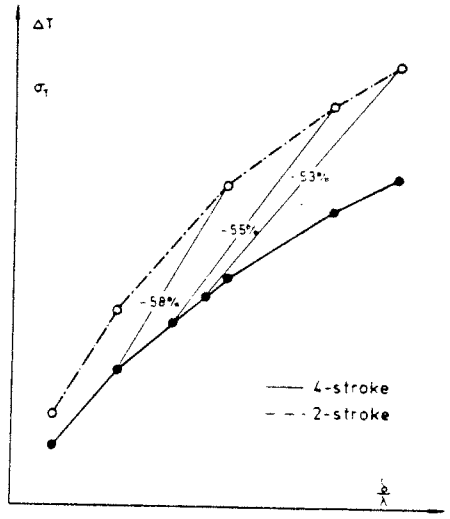


Diagram 19. Temperature level comparison of 2- and 4-stroke engines

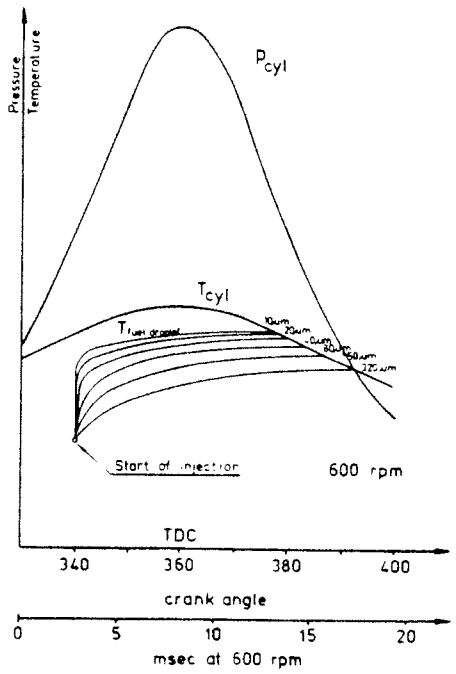


Diagram 20. Influence of fuel droplet size on fuel droplet temperature

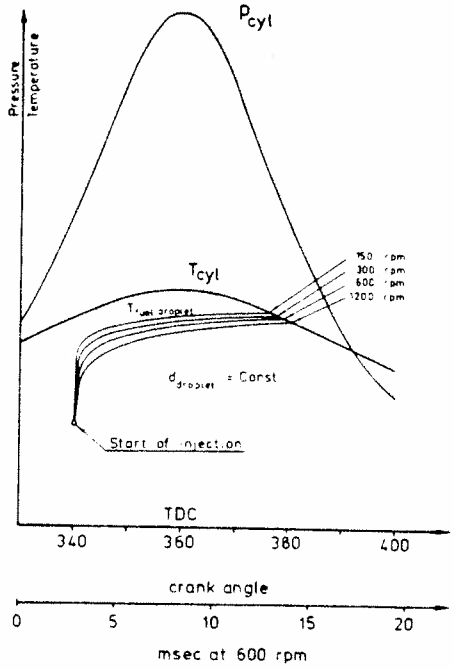


Diagram 21. Influence of engine speed on fuel droplet temperature

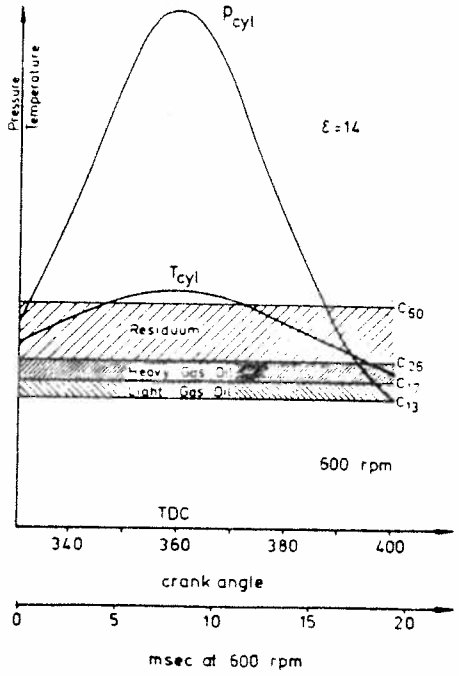


Diagram 22. Influence of fuel quality on fuel droplet temperature

WEST EUROPEAN GRADUATE EDUCATION
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SHIP DESIGN FOR FUEL ECONOMY

PRESENT AND FUTURE MARINE FUELS

by

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8th WEGEMT SUMMER SCHOOL

Lecture by Professor R. V. Thompson

PRESENT AND FUTURE MARINE FUELS

SUMMARY

An attempt is made in this presentation to forecast the types of fuel which will be available to the marine market at the turn of the century.

The presentation is divided into four distinct areas:

1. Future fuel availability.
2. World trading position.
3. Related power plant requirements.
4. Future fuel synopsis.

The format is designed to ensure that a sensible impression of propulsion requirements in the early part of the next century may be formalised, and hence the appropriate fuel availability predicted. Naturally, it is necessary to determine demand in both volume and form in order to provide a worthwhile analysis of availability - hence this approach.

1. FUTURE FUEL AVAILABILITY

1.1 Introduction

The forecasting of energy requirements and supplies twenty years ahead has always been a difficult exercise and at the present time is even more so due to the general political and economic unrest throughout the world. In what follows such a forecast is attempted by the forward projection of the historical trends of sea trading requirements coupled with reasonable and flexible assumptions.

Between 1974 and 1980 the cost of oil increased by 300% while at the same time the cost of living rose by 230% and 160% in the UK and USA respectively. Furthermore, oil supplies 55% of the energy needs of Europe and at least 45% of that of America - the latter accounting for 30% of the world's daily usage.

Energy costs in general are expected to increase in spite of transient 'dips' and in particular, within the period considered, the real cost of oil is expected to rise by 200% on today's prices and of even more significance will be the reduction of its availability.

1.2 Liquid Hydrocarbons - naturally occurring oil

An estimate of world recoverable hydrocarbon reserves made at the end of 1972 is shown in Table I and although 'world oil' is now conservatively estimated at 600 billion tonnes, only 35 to 40% is available for use with current technology. Fig. 1 shows how the annual production of oil and the estimated reserves have varied up to the present. The USA and Canada are within ten years or so of exhausting their own indigenous supplies, and in view of the intense exploration activity within those lands, the discovery of further reserves is unlikely and their reaction to this situation is of vital importance to the rest of the world.

Woods compares discovery rates between the Middle East and other parts and examines the influence of intense methods of exploration on the ratio of 'giant' to other fields found. As a result it is concluded that world oil recoverable reserves may be as much as three times as large as the present accepted figure. If world energy demand grows at 3.5% per annum, then the associated production is expected to peak at about the year 2030, assuming feasible rates of recovery. Political and economic stresses will probably extend this date.

1.3 Other Liquid Hydrocarbons

Oil shale and tar sands are a possible source of oil fuel and represent a reserve of liquid hydrocarbon fuel which is additional to the reserves referred to in 1.2.

Oil shale is a sedimentary rock which is fully impregnated with a solidified hydrocarbon wax, kerogen, which may be decomposed at elevated temperatures to form a refinable oil based product.

World reserves of oil shale are estimated to be in excess of 30×10^4 Mtoe distributed as indicated in Fig. 2 with by far the largest deposits being found in the USA. These deposits are capable of producing 100 litres/tonne of rock and are relatively accessible. Production is labour intensive and requires huge quantities of water and demands substantial development of explosive production techniques to permit large scale economic recovery.

Tar sands are found extensively in Canada and are extremely rich in bitumen. Although total reserves in place are estimated at 10^5 Mtoe, ecological problems will probably limit production to 1.6×10^4 Mtoe.

The resulting material is low in sulphur content and has a typical specific gravity of 0.84. The viscosity of the fuel produced from these sources is of the order of 320 secs. Redwood No.1 at 38°C (40 cSt at 50°C).

At present the estimated cost of oil extraction from oil shale/tar sands is about 25% above the regulated price of crude oil. The spot price of crude is often in this same region and the spot market now constitutes 10% of the world exchange in crude. Thus as the indigenous reserves of liquid crude in the USA disappear within the next ten years, it can be anticipated that considerable effort and investment will be made to develop an economical method of oil extraction from oil shale and tar sands.

This development, when successfully achieved, will have a marked effect on the situation and can be expected to relieve the present anxiety and tension regarding oil acquisition and distribution, thereby permitting a sensible return to acceptable planning procedures regarding production and transportation requirements. The latter should produce a much needed and salutory effect on world shipbuilding capacity and distribution.

1.4 Coal

The actual quantity of coal considered to be available today varies with source of the assessment. However, it is generally agreed that world coal in place is vast. Reserves known to be economically viable in terms of recovery exceed availability of crude in the ratio of 4 to 1, i.e. approximately 0.4×10^{12} Mtoe. Total reserves, not necessarily recoverable due to ecological and related pressures probably exceed 1.74×10^{12} Mtoe, and are distributed as shown in Table II. The ratio of economically recoverable to total available will probably increase with demand, although in any event currently available supplies are sufficient to meet total world energy demand for the next 100 to 300 years depending upon forecast of world growth and capacity to retrieve.

The relative cost of energy from a variety of sources is shown in Fig. 3. It is anticipated that oil will soon cost more than coal in gross terms.

Amongst the industrialised nations, Japan is presently restructuring its energy utilisation from oil to coal (see Fig. 4), largely supplying its

needs from Australian sources. Some smaller nations are also doing the same restructuring.

As a direct consequence of the expected shift in emphasis by 'Third Supply Nations' from oil to coal based economies, coal will tend to establish an economic position in world energy supply which suggests that the crossover point, indicated in Fig. 3, between the two primary energy sources may not be met this century.

With respect to use at sea: handleability is the largest single obstacle to the value of coal as an alternative bunkering element to oil. It may be used 'as is', pulverised or in the form of a coal-oil slurry.

Lumped coal has obvious disadvantages which are not altogether obviated by pulverisation. Optimisation of particle size and slurry viscosity could result in a 'fluid' fuel with a substantial retention of inherent mixture stability permitting shore rather than at-sea preparation.

The use of coal at sea in either the solid or pulverised state except for certain specific applications, e.g. Far East coal trade, is most unlikely due to its relatively low heating value (30,000 kJ/kg) necessitating a 30-50% increase in bunkering weight or 100% increase in volume. Aesthetic difficulties also apply, particularly the need for implementation of stringent fire safety legislation at the design stage.

1.5 Liquid Hydrocarbons from Coal

Oil has been produced from coal for the past 55 years or so, usually out of necessity rather than for purely economical reasons. The German war effort (1939-45) depended substantially on her ability to produce synthetic crude. For similar strategic but not belligerent reasons, South Africa produces substantial quantities of oil from coal thereby reducing its vulnerability to politically based pressure from oil producing nations.

Apart from developments of the original Fischer-Tropsch type, other processes are proving of substantial interest, these being Coal Oil

Energy Development (COED) and Solvent Refined or H-Coal (SRC). The former process is reported to yield up to $0.24m^3$ syncrude per tonne of coal as well as useful by-products.
Table III tabulates typical characteristics associated with syncrude. However, for marine use the highly volatile constituents have to be removed to meet flash point regulations, circa 60^oC.

The SRC product currently available in relatively small quantities in the USA is low in sulpher, but exhibits the disadvantage of a low diesel index and high melting point. Essentially, the problem of retarded ignition may be redressed by the addition of hydrogen in a specialised process with a more than modest increase in cost. (20% additional hydrogen required). The fuel burns well when ignited but causes an increase in exhaust gas temperature which, with additional cooling requirements, tends to increase specific fuel consumption. A tentative solution demands that the SRC be 'doped' with normal diesel oil (circa 3.5%) to facilitate initial combustion.

Other problems need to be resolved before the fuel may be released for general use; it is highly aromatic and therefore objectionable to humans and also incompatible with commonly used materials, particularly those based on rubber or synthetic compounds.

Consequently, substantial research in prime mover technology is required before oil from coal compounds may be used with equanimity in marine propulsion plant, particularly with respect to diesel engine applications. The attendant aesthetic problems also require the investment of time and effort to ensure complete safety at sea.

Reference to Table IV indicates, however, that the expenditure of effort in this direction could produce substantial economic returns in the immediate and long term future - assuming availability of coal in substantial quantities. Fig. 5 summarises the coal position.

1.6 Other Fuels

Interest, in a gross sense, is being shown in the development of alternative fuels for power generation; however their impact on marine propulsion technology in the intermediate and short term is unlikely to be of definitive proportions.

Apart from LNG which is too valuable a commodity to use in other than specialised applications at sea, methanol and hydrogen show significant promise but require substantial technological advance in production techniques to reach viability. The former has a calorific value of 20,000 kJ/kg and a specific volume double that of heavy fuel oil. It may be handled with equanimity as with conventional hydrocarbons and can be burned in boilers either on its own or in conjunction with other fuels. The 'pseudo' diesel index is poor and hence use in a conventional compression ignition engine installation is not currently possible or advisable. The material is mildly toxic and may be produced from coal in conjunction with large quantities of heat, possibly supplied from nuclear sources.

Hydrogen may be burned in all forms of power plant with modifications both to 'combustion space' configuration and intrinsic sensing and control instrumentation. Production is currently inefficient and costly which, together with its capacity for spontaneous ignition, tends to colour the product with a 'to be developed' syndrome, apropos general use in the marine field.

Table V lists the approximate relative properties of the best known 'alternative fuels'.

Fig. 6 summarises the overall possibilities.

1.7 Summary

(i) Estimates of liquid hydrocarbon resources obtainable worldwide are considered to be pessimistic to the extent of perhaps 100%.

(ii) Availability is limited principally by political rather than conservational considerations.

(iii) Although fuel costs appear to be high, they are not 'unrealistic' when compared with equivalent values a decade ago.

(iv) Alternative fuel sources are vast, particularly with respect to coal and shale deposits; however, with the exception of Far East developments, they will remain expensive in terms of overall combustion economics in comparison with general crude oil in the short and intermediate period.

(v) Production of liquid fuels from alternative sources, i.e. coal, shale, etc. will continue to gain momentum due to political rather than purely economic considerations and will satisfy shortfalls of crude availability on a progressive basis.

(vi) Marine transport, with the exception of certain specialised applications, will continue to burn hydrocarbons as its primary fuel irrespective of price and, to some extent, quality.

2. GENERAL TRADING POSITION - FORECAST

2.1 Introduction

Two fundamental alternatives are available for the prediction of world growth as applied to specific commodities. The first involves the postulation of an econometric model implying the development of a scenario upon which to base predictions which is itself developed from first principles. This method of analysis is commonly used in myriad forms by governmental predictive bodies based upon sophisticated computer models in conjunction with substantial manpower resources and at best yields short term data.

The approach which is to be used here is relatively unsophisticated, involves an intuitive consideration of individual commodity demand and supply worldwide over an extensive period, and will not withstand microscopic evaluation.
Although some assessment of both governmental and structural policies are included, dramatic short term impulse or step functions are subject to a 'wide band' filter represented by simple time averaging.

The results, although subject to those limitations alluded to above, are nevertheless interesting.

2.2 Assumptions

The results detailed below are derived on the basis of the following assumptions each of which exhibit controversial tendencies!

(a) No major war or economic recession will occur within the period of interest, as in 1975 when total world production declined 2%, resulting in a volumetric reduction in world trade of 5%.

(b) That economic growth, particularly in industrially orientated countries can be achieved without serious difficulties in balance of payments or shortage of liquid assets leading to import restrictions.

(c) Technological innovation follows similar growth paterns to the previous period reviewed, i.e. no material change in commodity demand or form being realised.

(d) Politics governing import and export control will not change to any measurable degree.

(e) Commodity transportation by sea is a direct function of demand and response of production.

Historically, demand growth in 1969-70 was approximately half that experienced as an average in the 1960s, i.e. 2.2%. Considerable improvement was shown through 1970-72 (3.5%) and 1972-73 (5.8%) peaking in mid 1973 to the extent that in 1974-75 a net decline of approximately 2% was experienced globally. Some recovery occurred in 1976-77 overall, although only to a marginal extent in developing countries.

Although the world manufacturing nations are said to be in a state of current recession, recession in modern times is symptomatic with a decline in the rate of growth as opposed to a linearised increase in production or a holding of a stable volume position. Assumption (a) above leans more towards the latter definition rather than the former, i.e. it assumes that no significant rate of change of growth may be expected in the next two decades.

With demand recovering in the major developed market economies together with a relative surge of imports being planned by the East European Bloc, world trade should exhibit a marginal upward momementum, thereby eliminating the setback of 1975.

Naturally, the trading position with Eastern Bloc nations is subject to short term inconsistencies; however, in the long term, it is predicted that these will be time averaged assuming that no major conflict results from international entrenchment. In any event, it appears most unlikely that the pre-recession growth rate of 9% per annum will be regained in this or any other decade. A detailed literature review indicated 4-4 % to be a 'reasonable' assumption of growth in world turnover in the immediate future and 4 -5% in the longer term. In view of the apparent consumer saturation in manufactured goods, together with a perceptible reduction in the rate of growth of world population and continuing energy restrictions, these figures are considered to be over-optimistic. Basic 'life support' material transportation will continue to grow and to some extent obviate some of the 'looseness' in demand for secondary products; however, overall growth is thought likely to average 2 % rather than 4-5%.

With respect to seaborne trade, this has tended to follow world demand in terms of accumulated average but has led a more chequered career when

reviewed on a short-term basis.

Although important, the prediction of seaborne trade growth and form is subject to many random functions as illustrated over the past ten years. However, it is reasonable to assume that future performance will not be required to meet world gross expansion for the following reasons:

(i) Export curbs on Japan.
(ii) EEC closed market developments.
(iii) Re-opening of Suez Canal.
(iv) Slow but significant industrialisation of developing nations using their own raw materials for internal consumption.
(v) Improvements in capacity and distribution of oil based products via pipelines, Persian Gulf to Mediterranean, Alaska, USA, etc.
(vi) Marginal growth in airborne trading.

Two other factors affect the overall scenario, particularly if weighted toward Western world ship manufacture/repair capacity; growth of the Eastern Bloc merchant fleet and its attempt to transport not only its own indigenous requirements but also a steady encroachment on the percentage of trade carried by traditiona merchant fleet owners.

A further contributory factor leading to the acceptance of a pessimistic world transportation growth assessment is the capacity of countries such as the USA, Canada, UK and Germany to achieve a measure of self sufficiency in energy supplies should the need arise and time permit. Currently, the pressure to develop other energy resources is primarily economic; however, it is not inconceivable that in the comparatively near future a political withdrawal of supply could make procrastination fatal, hence an intensity in effort to develop alternative intrinsic sources must be expected.

2.3 Oil Shipment

Growth in oil consumption reached a zenith of 8% per annum during 1973 declining to 3% in 1975 with some marginal increase in growth being exhibited since then. A review of published predictions indicate average growth rates varying from 2.9% to 6.7%; however, no account is taken of 'offshore' oil discoveries or development, shale oil production, etc., which are scheduled to disturb historical trends in the import/export cycle.

Assuming a relatively conservative figure, based upon details leading to the above statements, the average growth rate of oil transportation is assumed to be 2.5% until the year 2000. Table VI indicates the movement of commodities at sea in 1977 which when compounded at 2.5% suggests that some 3,170 million tonnes of oil will require movement in the year 2000.

At the present time there is a surfeit of large tankers and growth in the demand for new tonnage of this type is not expected until 1985 or so.

2.4 Dry Bulk Transport

A detailed evaluation of all the various sub-groups which together form bulk trading, suggest as a conservative figure of predictive growth 3.3%, resulting in the possible shipment of some 1,600-1,900 million tonnes by the year 2000. Assuming that ship design and utilisation will improve by some 10 to 15% over this period, it is estimated that the corresponding plant requirement at the end of the period wil be of the order of 215-250 million dwt. The equivalent size and numbers involved are shown in Table VI.

The above involves the assumption that the increase in expected coal utilisation will stimulate a need for respective carriers, which will more than recompense for the decline in the growth of steel production and corresponding transportation of iron ore (currently 1,580 x 10^9 tonne miles per annum). It could be argued that those countries which

have large coal resources may be reluctant to export them for short-term benefit against long-term scarcity - this is unlikely to occur within the next 20 years due to the prolification of the production.

2.5 General Cargo Trading

Currently, the general cargo trade has an annual accumulative haulage index of 1,500 billion tonne miles averaging 6,000 miles per voyage. It accounts for some 15% of all seaborne tonnage and 5% of total tonne miles achieved with a gross monetary value approaching 80% of total sea trading and can, therefore, exercise a substantial influence on new building demand and selection of associated prime movers. The expected composite growth potential is expected to follow a compounded percentage rate between 3.4 and 4.2% suggesting, in turn, a cargo movement in the region of 560 to 620 million tonnes.

Current fleet size corresponds to 105 million dwt which, based upon the above figures, should increase to a world demand in the year 2000 of approximately 120-140 million dwt. This prediction is based upon the premise that the bulk of general cargo trade vessels will follow the container ship concept thereby improving overall carriage efficiency.

2.6 Specialiised Applications

Liquefied gas carriers have proved to be a successful innovation born out of necessity and indications are that potential for growth will continue to exist for some considerable time. The USA, for example, imports 4 million tonnes of LNG every year and historically exhibits an insatiable demand for continuous growth in energy consumption. A conservative estimate for fleet demand in 2000 is 30 to 40 million dwt.

High speed trafficking in high value goods such as electronic, scientific equipment, medical supplies, etc. could show some potential in the future as too could barge carrying, integrated, multipurpose vessels, etc. As offshore sources produce 20% of the world's total oil requirements, support vessels, specialised short route tankers, etc.

could demonstrate some marginal growth. Any further comments regarding Ro-Ro, catamaran pipelaying vessels, etc. would be purely speculative and, in consequence, are best left to fill the percentage remaining unspecified in estimating world tonnage requirements.

Fig. 7 summarises the overall estimated position relating to ship demand to the year 2000.

2.7 Summary

(i) Assuming no major worldwide conflict there will be a distinct decline in the rate of growth of the industrial nations.

(ii) Overall growth in sea transportation may reasonably be expected to average 3%.

(iii) The oil tanker market will show some growth but this will not be truly identifiable until approximately 1985. Some additional growth will be seen towards the end of the century.

(iv) Initially, interest will be shown in vessels up to 120,000 tonnes, although specialised yards will still be able to maintain production of ULCCs.

(v) Product tanker demand should continue to improve due primarily to the diversification of manufacture of oil derivatives and the international dispersal of refinery plant.

(vi) Industrialisation of the developing nations will increase the demand for special products vessels.

(vii) Dry bulk transport requirements will continue to grow in relation to population growth and, to some extent, in response to some shift of emphasis to coal as an energy source (3-4%).

(viii) Growth in specialised products will assist demand in the general cargo trading position demanding a total fleet tonnage

of approximately 140 million dwt.

(ix) Technological evolution will encourage the design and development of vessels for specialised applications, particularly for high speed trafficking of high value goods and the supply of offshore developments.

(x) The expansion of the USSR merchant fleet if permitted to continue unabated will absorb a substantial proportion of the growth predicted, particularly in the area of dry bulk carriers.

(xi) World manufacturing capability exceeds demand.

3. POWER PLANT REQUIREMENTS TO THE YEAR 2000

3.1 Introduction

Fig. 8 indicates the factors involved in the selection of the main propulsion system for any given application. Substitution of the necessary data leads to the conclusion that fuel cost is the most significant economic factor to be considered over the life cycle of a ship. As more and more emphasis is placed on the scarcity and hence value of hydrocarbon fuel sources, it is plausible to suggest that any techno-economic assessment of ship purchase and deployment in the future will lead to the conclusion that fuel economy rather than first cost will form the basis of any decisions made and subsequent orders placed.

Fig. 9 predicts the 'power to weight' ratio for the range of vessels considered projected over the next 20 years. Using these historically placed ratios in conjunction with assumed annual fleet demand, an approximate machinery requirements schedule may be derived for each class - Figs. 10, 11 and 12.

The total power plant requirement in 2000 should approximate to 16×10^6 kW of which 70% could be diesels, 20% steam turbines and the

rest gas turbines. Obviously, hybrid systems could be involved in any or all of the three main classifications where the heading used refers to the principal power unit.

3.2 Fuel Quality

Recently there has been a marked tendency towards the combustion of relatively poor quality fuels in propulsion plant at sea in a further attempt to minimise operational costs. Reliability of the prime propulsion element has been, to some extent, affected by the use of these fuels due to the incompatibility of intrinsic contaminants with engine components and lubricating fluids.

In addition to the 'usual' contaminating materials such as silicon, rust and water, these 'residual' fuels contain large quantities of asphaltines - up to 10% by weight. Apart from being highly aromatic, these materials tend to block filters because of their ability to congeal with other undesirable problem particles. Removal of these aromatic materials tends to reduce the calorific value of the fuel by a perceptible margin and therefore chemical treaatment is generally preferred as a means of reducing the overall detrimental effect. (1 part to 2,000 or 4,000 depending upon degree of contamination.)

Other significant problems include fuel stratification leading to bunker incompatibility and the effect of wax contamination on filters and heating coils, etc. Additional storage requirements needed to combat the relatively high viscosity of the material aso tends to the overall expression of 'unsociable fuels'.

Typical unwanted but included constituents of heavy fuel oils are shown in Table VII and a contaminant 'weight count' is shown in Table VIII.

An assessment of the effect of specialised fuel treatment plant on the associated system is shown in Table IX where it would appear that the use of centrifuge based systems offers distinct advantages in terms of engine wear, all other aspects being equal. However, the associated equipment development and installation costs could modify the economic

picture appreciably.

Even with the inclusion of fuel treatment plant, the use of 'heavy' fuel still presents difficulties associated with ignition delay and subsequent relatively large rates of change of pressure when used in diesel engines and demand detail development and specialised research attention. These and other aspects are referred to below.

3.3 Diesel Propulsion - Status

Figs. 13 and 14 indicate why diesel drive systems have in recent years achieved a monopolistic position in the marine propulsion market, particularly in the low to medium power ranges. The current aim of the diesel engine manufacturing industry is to improve reliability, reduce fuel consumption, noise and vibration and to offer engines capable of efficient operation using an ever widening spectrum of fuels.

3.3.1 Reliability

The need to use low grade fuels while maintaining or increasing the present level of bmep, aggravates some of the problems of the diesel engine, particularly if reduced power slow speed running has to be maintained for extended periods.

The wear-rates of piston rings (particularly the top one) and liners are adversely affected (increases of 100% have been reported), and require further development of special lubricating oils and new material combinations.

The use of fuels with low cetane value increases the shock loading within the engine, with particularly adverse influence on the top piston ring. At slow speed conditions, the atomisation is relatively poor, maximum combustion temperature is lower and there is increased tendency to precipitate carbon residues leading to a marked reduction in reliability. There is thus need to develop a constant pressure injection system which, together with modifications to combustion

chamber design, should serve to improve reliability. The incorporation of scavenge air heaters (to heat air to 80°C) will also help improve combustion conditions for slow speed running below 45% MCR.

For full power operation, as bemp is increased, so too are thermal stresses and continued development in the distribution of material in the neighbourhood of combustion space, cylinder head and piston crown is necessary to maintain the standard of reliability.

The present trend of medium speed engines to increase power per cylinder in order to reduce the number of cylinders for a given output will undoubtedly continue, using increased density of air charge which, in turn, requires further developments of exhaust gas turbines and compressors. As developed these engines will continue to expand into the domain of both slow speed diesels and steam plant due to the flexibility and reliability of operation afforded by multi-engine installations. Fig. 15 gives an indication of current and anticipated diesel power distribution.

3.3.2 Fuel Consumption

Factors directly affecting improvements in specific fuel consumption are:

(a) Development of high pressure injection systems together with improved mixing processes.

(b) Improvements of turbo-charger efficiency.

Present development work on high pressure injection systems indicates that savings of 5 to 6 g/kWh can be expected at normal full load with even greater proportional savings at part load. Careful attention will have to be paid to nozzle design and manufacture, and to operational temperature control.

The constant pressure system of turbo-charging is the most efficient for pressure ratios in excess of 2.2 to 1: it is widely used at present and

will continue to be used. For pressure ratios in excess of 3.5 to 1, new materials may have to be used to permit higher speeds of rotation.

The injection of emulsions, ie. homogenous fuel oil water mixtures, has shown improvement in fuel consumption and reduction in exhaust gas temperatures particularly at part load conditions (4-7% on medium speed engines). Modification of the fuel system is minimal and in consequence sensible return may be obtained for a modest initial cost particularly in the retrofit market.

Requirements associated specifically with medium speed engines include the ability to operate on heavy fuel oils together with a reduction in lubricating oil consumption. Successful realisation of the former depends upon the resolution of exhaust valve problems and turbine fouling.

Exhaust valve difficulties, i.e. plating, corrosion, erosion and burning of seats, etc. can be solved technically, but economic aspects predominate. Exhaust gas temperatures above 550°C cause catastrophic fouling of exhaust gas turbines, particularly when using heavy or residual fuels. Valve seat temperatures are limited to 450°C. Attention to detailed design and selection of materials should permit lubricating oil consumption levels below 1 g/kWh to be realised.

3.3.3 Waste Heat Recovery

A considerable percentage of available heat in fuel is lost to exhaust gases - 28-40% and in consequence the predicted austerity in fuel supplies has kindled interest in exhaust gas systems. Use of a waste heat boiler in conjunction with a turbogenerator can prove economically viable on the longer trade routes. Duel pressure systems, although more complicated and hence expensive, have been designed to produce energy equivalent to 9% of main engine power. These have been widely used in the USA on land based installations and on 16 BP tankers using 5,600 kW main engines.

A recent development has been the optimised integrated waste heat recovery system - the Seajoule package which, it is claimed, results in a net improvement of some 3 to 4% of work output. A similar system manufactured by Hitachi claims a 6.1% reduction when used in conjunction with a 13,000 bhp installation. This assumes that diesel engine performance and subsequent propeller matching characteristics may be modified to optimise overall capability.

Finally, the utilisation of a Rankine Cycle as a bottoming plant for diesel engine exhaust heat recovery can show improvements of up to 10% in overall energy utilisation.

3.3.4 Diesel Engine Demand

Table X indicates the number, size and range of engines manufactured or ordered up to and including 1977. Table XI attempts to project the industry's requirements over the next 20 to 25 years.

A cursory glance suggests that, after some initial growth in the low power range, demand should stabilise through the period under consideration. Although growth is projected in the larger power range, it will experience a time lag of some 10 to 15 years with pre-1977 trends expected in the 1990s.

Currently the slow speed diesel holds some 75% of the total market, however, expected developments with the medium speed diesel with respect to residual fuels, increase of power to weight to cylinder ratio, etc. could enhance its relative position to some 35 to 40% at the turn of the century.

Developments with coal or shale derived fuels should also harden the diesel manufacturers' hold on the market.

Heat recovery systems will be recommended extras by many prime mover manufacturers but sales resistance based on first cost, size and weight could severely limit application initially.

3.4 Steam Plant

The staggering rate of change of fuel price has had disastrous repercussions on the steam propulsion plant industry. In 1978 only 32 ships were completed worldwide, utilising steam plant, representing less than 3% of total number of propulsion plant installed. Sixteen of these ships were built in the USA for national carriers. Figs. 13 and 14 compare the performance charcteristics of various types of propulsion machinery indicating that steam plant economics improve with increase in output demand. Fig. 16, although linearised, demonstrates the effect of fuel economics on the industry as a whole but more specifically emphasises the plight of the steam plant specialist

Current applications include:

Containerships	USA only
Bulk Carriers	150,000 dwt and above
Tankers	150,000 dwt and above
LNG Carriers	20,000 dwt

Even in these specialist areas, selection of steam propulsion plant is minimal, as with the exception of LNG carriers, the diesel engine is emphasising its monopolistic position. It is also interesting to note that even at large power levels, the slow speed diesel is under pressure from its medium speed partner - a 280,000 dwt tanker being converted from steam to medium speed diesels in 1979 purely on the basis of economy. Subsequent trials indicated that other advantages above and beyond that of fuel economy could also be realised, i.e. reduction in stopping distance and time, thereby enhancing operational safety (twin drive, etc.).

Manufacturers are attempting to secure at least a small corner of the market by sounding the virtues of design advances, viz. tip seal techniques, increase in blade length and mean diameter to reduce LP turbine speed, minimisation of steam entry losses, etc. which in total produce fuel savings of approximately 1.6%. Although of value this figure is insufficiently attractive to generate serious interest from ship owners and in consequence more sophisticated systems are being

voiced, i.e.increase in mean operating temperatures and pressure, reheat cycles and additional bleed stages. Fig. 17 shows the variation of specific fuel consumption with power for various standard cycles.

Although attractive from the point of view of fuel economy, all these methods result in further complexity or raise queries regarding reliability with prospective users, i.e. material compatibility, superheater tolerance to failure, aspects of safety and reheater protection during manoeuvres often demanding a twin furnace system or complex damper interaction and control.

The expected reduction in available fuel quality has, however, injected new vigour in the minds of 'steam men'. Intense interest and activity has been applied to the fluidised bed concept, used but largely ignored in the 1930's, which exhibits the advantages of virtual immunity to fuel contamination, high temperature operation, exhaust gas pollution control and superheater protection.

A system currently receiving serious attention involves a simple non-reheat boiler, fluidised bed superheater/reheater, together with an integrated turbine reduction gear unit. Although the main boiler operates at 500°C, the overall specification calls for 600°C at 122 bar producing a fuel consumption of 0.22 kg/kWh. Should tests with the prototype system prove conclusive, enthusiasts are convinced that propulsion units as small as 10,000 bhp could be economically served by steam plant.

Some direct coal-fired ships will be ordered for specialised applications, i.e. coal carriers for Far East trading; however, no general application is expected.

3.5 Gas Turbines

Claims for gas turbines are itemised in Table XII, and underline the reason for their popularity in military and fast long haul trading vessels.

Regenerative cycles with reheat and intercooling offer considerable advantages over simpler systems in terms of power and efficiency, but exhibit penalties including weight, size, complexity, etc.

Use of the gas turbine as a prime mover is limited by the following:

(a) Unproven capability to burn bunker C in an efficient and reliable manner.

(b) Reversing difficulties.

(c) Poor part load performance.

Progress in recent years in material development and transpiration cooling techniques, etc. have made it possible to increase maximum operating temperatures by approximately $10^{o}K$ per year with an appropriate improvement in fuel consumption. Controllable pitch propellers are increasing in use and size, therefore, the reversing problem is eroding with time.

Part load performance can be improved by using variable geometry turbine guide vans, etc. and further gains in this area may be expected although at a price.

Reflection on the above suggests that gas turbines could present a viable alternative to more traditional power plants but probably in combined cycle operation and only in specialised applications, viz. LNG carriers and general military use.

3.5.1 Waste Heat Recovery Gas Turbine Systems

In a simple gas turbine cycle, up to 75% of the total available heat is lost to exhaust, the efflux containing a substantial proportion of air (80%) due to the inherently high air-fuel ratio demanded by the turbine for sensible combustion. Improvement in overall efficiency can be obtained by utilising the energy of the exhaust gases at $340^{o}C$ to produce steam in a waste heat boiler to deive a simple turbo-generator

and/or evaporator to provide service power and heat. Improvements of approximately 2.5% in fuel consumption can be realised.

Alternatively, if the available excess power is supplied to a propulsion motor connected via the gearbox to the propeller shaft an improvement of 10% can be expected, or if the turbogenerator is connected directly to the primary pinion via an hydraulic coupling fuel consumption falls to 0.247 kg/kWh.

A further concept involves a 'helper' turbine to provide additional motive power for the compressor which in conjunction with variable areas second stage nozzles in the gas turbine itself increases power output and reduces fuel consumption expectancy to 0.246 kg/kWh.

3.6 Combined Gas Steam Cycles

The gas and steam turbines can be combined to offer attractive power generation cycles, the simplest concept utilising a waste heat boiler producing sufficient energy to drive a steam turbine.

A duel pressure waste heat boiler with supplementary burner, low pressure evaporator configuration, could offer economic advantages if fuel costs escalate during the period under review. The fuel rate for a 20 MW plant is estimated at 0.228 kg/kWh. Although incomplete, Table XIII attempts to provide some indication of the comparative performance between rival concepts of combined cycles. Of particular interest is the fluidised bed/gas turbine combined cycle which offers substantial economies in fuel consumption - expected to be 0.214 kg/kWh with total fuel flexibility. The gas turbine contributes 55% of total power for a 20 MW plant configuration. Expected steam conditions are 32 bar at 603°C leading to the system being aluded to as the Supercharged Steam Generator Propulsion Concept of STAG form.

3.7 Summary

(i) Although there has been a marked tendency towards the combustion of relatively poor fuels, this tendency is expected to stabilise within the next two years.

(ii) Alternative fuels will become available in approximately five to ten years' time; however, it is anticipated that assuming no major catastrophes, sufficient fuel of acceptable quality will be available to the marine field for the foreseeable future.

(iii) The diesel engine will consolidate its position as the world leading prime mover. There will be a perceptible shift of emphasis from the slow speed to the medium speed diesel propulsion power plant with the latter being in the majority.

(iv) Considerable research will be applied to problems associated with reliability including the combustion of relatively low quality and non aesthetic fuels.

(v) The constant pressure turbo-charger will predominate, pressure ratios will increase. Overall fuel consumption will decrease as a result of the higher specific power to weight ratio.

(vi) Overall electronic control will be commonplace at the turn of the century.

(vii) Waste heat recovery systems will be fully integrated. Energy system design will be effective within the next two years.

(viii) Steam plant will remain in decline, the majority of systems being manufactured in the US for its own shipping although specialised applications will occur particularly for Far East trade.

(ix) Although more sophisticated alternatives of steam plant will be offered, their inherent complexity will maintain the trend toward diesel utilisation.

(x) Continued development of application of the fluidised bed reheat systems should assist in the maintenance of some steam plant at sea.

(xi) With the exception of specialised applications, ie. military warships, LNG carriers, etc., the gas turbine will play a subservient role to other forms of plant. Some considerable appication will occur in waste heat recovery gas turbine systems, ie. as auxiliary power generation.

4. FUTURE FUEL SYNOPSIS

It is abundantly clear that the current relatively 'ad hoc' method of bunkering fuels will have to be considerably modified. As indicated previously, although there should not be a shortage of liquid hydrocarbons the quality available to the Marine Industry will be, at best, suspect. Additional catalytive cracking procedures are being developed to produce a larger proportion of distillates thereby ensuring that the residue will be even more unpalatable than at present. Consequently purchase on the basis of viscosity and specific gravity will not be sufficient to guarantee compatibility with main engine requirements.

The constituent parts of a fuel which it is suggested will characterise its performance with suggested acceptable values are shown in the Table.

The maximum values of density have been selected on the basis of current water extraction capability using conventional centrifuging equipment. Viscosity values ensure compatibility with existing fuel systems and flash point follows international safety legislation.

Most of the other suggested determinants provide an indication of the performance of the fuel in the combustion process with particular emphasis on the effect of impurities including carbon residues on engine related contamination and are self explanatory. What is recognised is the problem encompassed by the term Ignition Quality for which no standard test exists.

In addition to the above "Users" have to be aware of the effect of additional impurities such as Ashphaltenes & sodium, the latter causing deposits to occur on important engine functional components to operational detriment.

Finally, it is to be expected that in addition to the classification of fuel it will be necessary to improve on board treatment plant and facilities to facilitate reliable combustion. Blending is becoming fashionable and will have to be sensibly controlled to prevent problems associated with sludge and sediment formation. Rules regarding compatibility will have to be defined and carefully adhered to to prevent the development of wax like sludge which has been shown to disable almost the entire fuel treatment facility.

Review of the above indicates that the immediate future for the marine technologist is one of challenge particulary with respect to fuel classification, engine development and treatment compatibility.

5. CONCLUDING COMMENTS

One of the most striking results of this study is the conclusion that world resources of liquid hydrocarbon, whilst finite, are larger than previously anticipated. Assuming a growth in energy consumption proportional to the expansion of world population, it may be suggested that demand will exceed possible supply some time after the end of the century, possibly by as much as 30 years. This assumes that exploration will be intensified and to a reasonable extent rewarded. The advancing real price of oil will result in a perceptible decrease in the ratio of marginal to profitable fields, thereby increasing the possible yield. International politics as opposed to supply/demand economics will also limit world usage of liquid hydrocarbons, thereby extending its terms of availability.

Far Eastern and Australian coal production will increase, thereby easing the burden on 'Third Supply' nations such as Japan which, in turn, will have a rationalising effect on Middle East crude supplies. Oil from coal will be both economic and widespread in the 1990s.

The production of shale oil deposits will accelerate appreciably in the next ten years and is prolific in those countries which currently exhibit high energy demand. Land based nuclear derived energy will develop more slowly than anticipated due to ecological resistance, but economic pressure will ensure that power is generated from the use of bulk fuels such as coal and general waste products created by civilisation.

The resulting conclusion, assuming that world politics maintain some modicum of self control, is therefore that the 3-4% of world oil demand required to maintain trading at sea will be seen to be both a fundamental requirement and cost effective and hence assured.

With respect to the trading scenario, the overall picture is one of growth but only in keeping with the expected expansion in world population. Substantial rates of change of growth are not expected in the foreseeable future. There wil be perceptible changes in emphasis; tankers maintaining, essentially, the status quo, i.e. zero real growth, with expansion being limited to dry bulk and specialised product carriers.

The diesel engine is expected to maintain its position of superiority over steam plant with a perceptible shift towards the medium speed product. Fluidised bed combustion will probably assist this trend, being used in waste heat concepts initially, although some movement towards steam plant may be expected in the 1990s. The construction and use of coal burning ships will be confined to specialised applications and specialist carriers.

Emphasis will be placed on the design, construction and use of a ship as an integrated system, thereby minimising waste and maximising economy. Overall ship speed will decrease with a sensible reduction in power to displacement ratio. Power plant reliability will be enhanced and the trend towards burning poor quality fuels arrested - ship availability being preferred to a doubtful reduction in apparent fuel and operational costs.

World ship construction capacity still exceeds demand and will continue to contract, the degree of contraction depending to a large extent on the capacity and resolution of the Western World to limit Soviet fleet and trade expansion.

Table: Suggested grades and specifications of fuels to be used by the marine trade when ordering fuels

GRADE	MA		MB		MC		MD		ME		MF		MG		MH		MI		MJ	
INSPECTION	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
Density at 15°C	—	—	—	0.90	—	0.920	—	0.990	—	0.990	—	0.990	—	0.990	—	0.990	—	0.990	—	—
Viscosity Kinematic cSt at 40°C	1.5	5.5	—	11.0	—	14.0	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Viscosity Kinematic cSt at 80°C*	—	—	—	—	—	—	—	15	—	25	—	45	—	75	—	100	—	130	—	130
Flash Point PM (Closed) °C	43	—	60	—	60	—	60	—	60	—	60	—	60	—	60	—	60	—	60	—
Pour Point (Upper) °C 1 Dec–31 Mar	—	—	—	0	—	0	—	24	—	30	—	30	—	30	—	30	—	30	—	30
Pour Point (Upper) °C 1 Apr–30 Nov	—	—	—	6	—	6	—	24	—	24	—	30	—	30	—	30	—	30	—	30
Cloud Point °C	—	−16	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Ramsbottom Carbon wt%	—	0.20	—	0.25	—	2.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Conradson Carbon wt%	—	—	—	—	—	—	—	12.0	—	14.0	—	20.0	—	22.0	—	22.0	—	22.0	—	—
Sulphur wt%	—	1.0	—	2.0	—	2.0	—	3.5	—	4.0	—	5.0	—	5.0	—	5.0	—	5.0	—	5.0
Ash wt%	—	0.01	—	0.01	—	0.05	—	0.10	—	0.10	—	0.15	—	0.20	—	0.20	—	0.20	—	0.20
Sediment by Extraction wt%	—	0.01	—	0.02	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Sediment (Total Existent)	—	—	—	—	—	†	—	†	—	†	—	†	—	†	—	†	—	†	—	†
Water Volume %	—	0.05	—	0.25	—	0.30	—	0.50	—	0.80	—	1.0	—	1.0	—	1.0	—	1.0	—	1.0
Cetane Index	45	—	35	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Ignition Quatity	—	—	—	—	†	—	†	—	†	—	†	—	†	—	†	—	†	—	—	—
Metal Contents ppM Vanadium	—	—	—	—	—	100	—	250	—	350	—	500	—	600	—	600	—	600	—	600
Aluminium ††	—	—	—	—	—	—	—	30	—	30	—	30	—	30	—	30	—	30	—	—

NOTES: * An indication of the approximate equivalent in kinematic viscosity at 50°C is given below:—

Kinematic viscosity at 80°C	15	25	45	75	100	130
Kinematic viscosity at 50°C	40	80	180	300	500	700

† Considered important but currently no standard test method available

††An acceptable test method has to be agreed.

ENCLOSURE

Professor R V Thompson, University of Newcastle-upon-Tyne

Table I
World Recoverable Hydrocarbon Reserves, End 1972

TYPE	BILLION TONS		
	Recoverable	*Possible Additions*	*TOTAL*
Crude Oil	75	145	220
Natural Gas	53	147	200
Tar Sands, Heavy Oil	35	185	220
Shale Oil	55	445	500
Coal	700	1,800	2,500

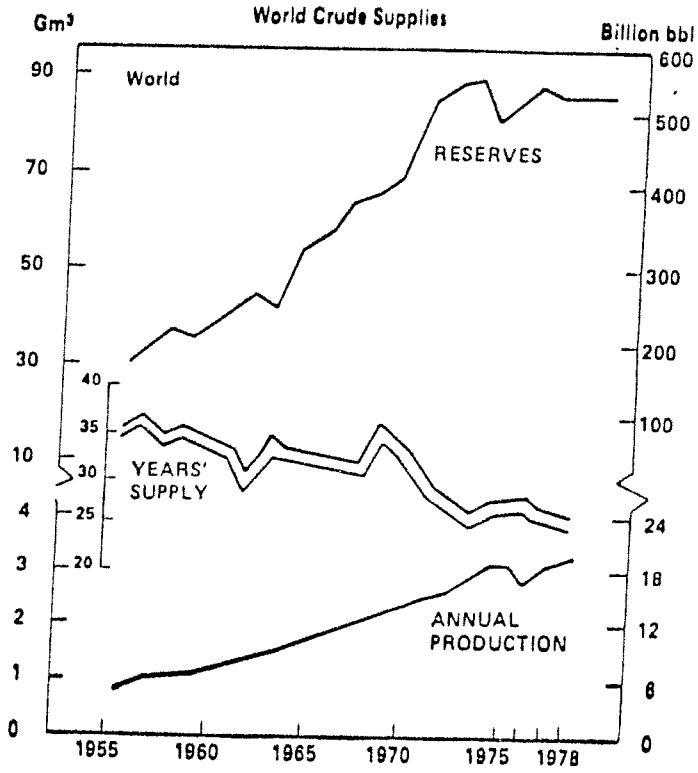

Fig. 1

RELATIVE COSTS OF ENERGY SOURCE

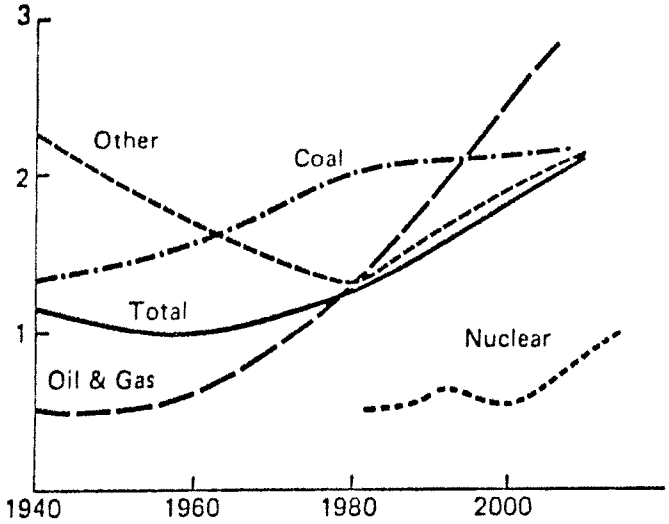

Fig. 3

Table II
Ultimate World Coal Reserves
Units—Mtoe

Africa	9,530
Asia	179,210
Europe	98,245
USSR	924,000
N America	490,600
S America	5,335
Oceania	32,300
TOTAL	1,739,220

FORECAST CHANGEOVER FROM OIL TO COAL BURNING IN JAPAN

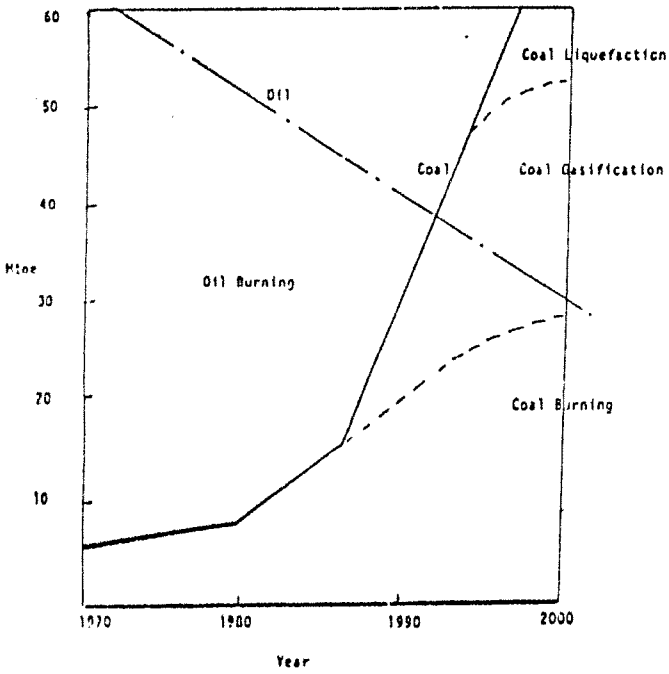

Fig. 4

Table III
Properties of Synthetic Crude Oil
COED Process

Typical values only—considerable variation is likely depending on coal source.

Specific gravity	0.900
Viscosity at 38°C	3.4 cSt (34 sec Redwood 1)
Flash point (closed cup)	8°C
Pour point	2°C
GCV	44,000 KJ/kg
Sulphur	0.0001% by weight
Ash	0.005% weight
Cetane rating	Approx zero

SRC Process

Specific gravity—varies from light to heavy depending on the extent of final hydrogenation process.

GCV	37,000 KJ/kg—Constant regardless of the source of the coal feedstock.
Sulphur	0.8% weight
Ash	0.1% weight

Table IV
Synthetic Oil Costs (1978 prices)

Feedstock	*Process*	*Typical Plant Capacity (tonne per day)*	*Plant Cost $M*	*Oil Cost $/barrel*
Coal	Fischer-Tropsch	12,000	1,275	31.80–42.40
Coal	H – Coal	4,000	300	19.08–23.32
Coal	COED	4,000	570	23.32–27.56
Natural Gas	Fischer-Tropsch	—	—	19–22

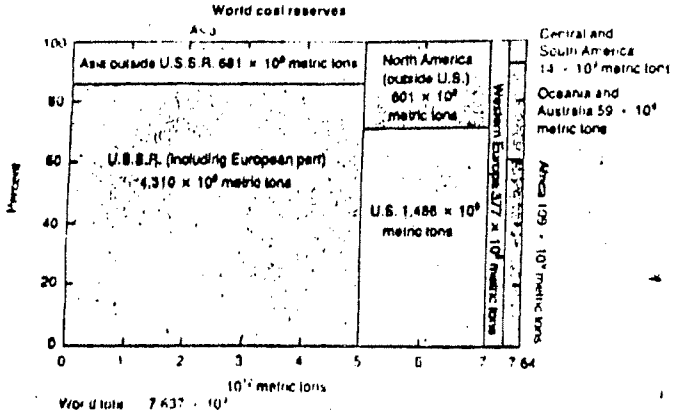

Fig. 5

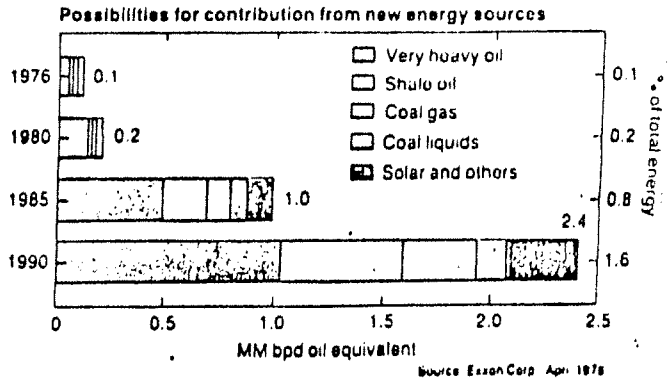

Fig. 6

Table V
Properties of some Alternative Fuels

Fuel	*Specific Weight†*	*Specific Volume†*	*Freezing Point °C*	*Boiling Point °C*	*Storage Sys. Req.†*		*Relative Storage Costs/Unit Energy**
					Weight	*Volume*	
Heavy Fuel Oil	1	1	—17	460	1	1	1
Methane	0.86	1.67	—184	—162	1.8	6.4	70
Methanol	2.1	2.2	—98	65	2.1	2.2 (as liq.)	1.5
Hydrogen	0.36	4	—259	—252	2.9	15.4	150
Hydrazine	2.9	2.4	1	114	—	—	—

† Relative to oil
* Based on tank size of approximately 1 tonne oil equivalent

Table VI
Sea Transport in the Year 2000

Types	*Types Dwt Thousands*	*Average Dwt Thousands*	*Quantity to be Shipped Million Tonnes*	*Fleet Demand Million Dwt*	*Fleet Demand Number of Vessels*	*Fleet Demand Dwt % of Total*
Oil Tankers	350–400 100–150 60–70		2,000–2,330 1,000–1,170	285–330 145–170	800–825 1,150–1,450 135–165	
	TOTAL	200–500	3,000–3,500	430–500	2,085–2,440	50%
Dry Bulk Carriers	150–180 70–90 30–40		960–1,140 400–475 240–285	129–150 54–62 32–38	830–860 700–760 950–1,050	
	TOTAL	120–130	1,600–1,900	215–250	2,480–2,670	25%
General Cargo			600–720	120–140		14%
LNG				30–40		4%
Others				65–70		7%
TOTAL				860–1,000		100%

WORLD FLEET DEMAND ON THE YEAR 2000

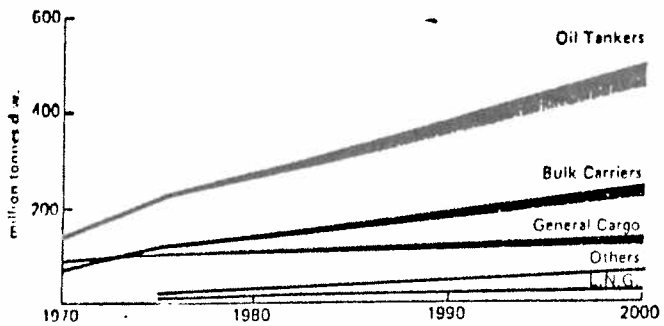

Fig. 7

METHOD TO CALCULATE N.P.V. FOR MAIN ENGINE

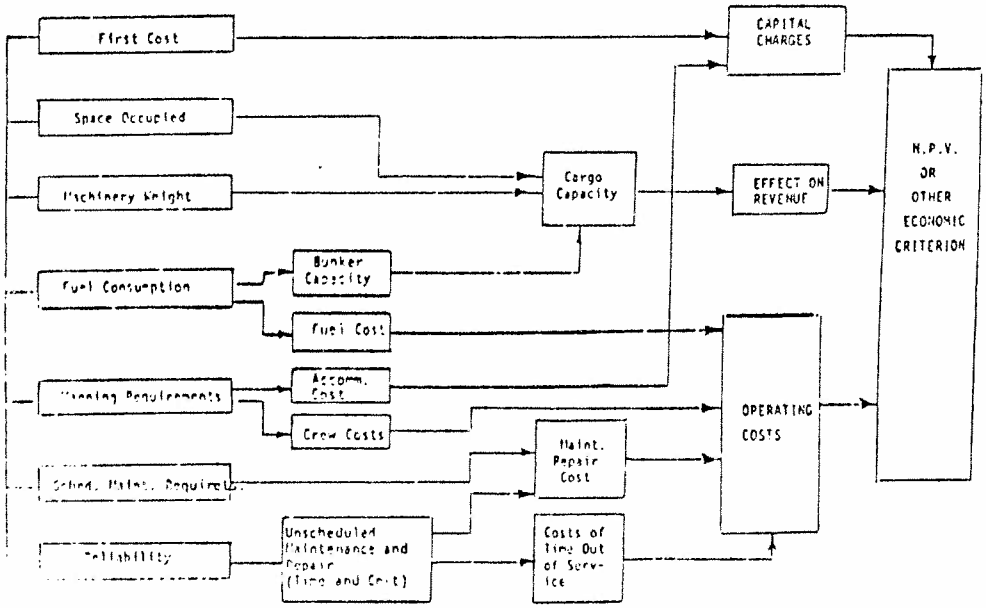

Fig. 8

SPECIFIC POWER TRENDS

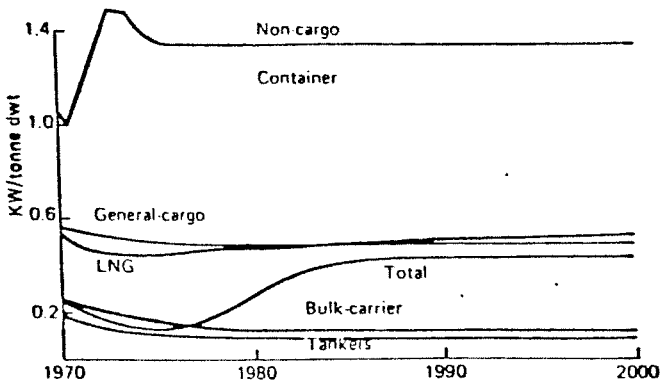

Fig. 9

MACHINERY TRENDS IN BULK CARRIERS AND GENERAL CARGO SHIPS

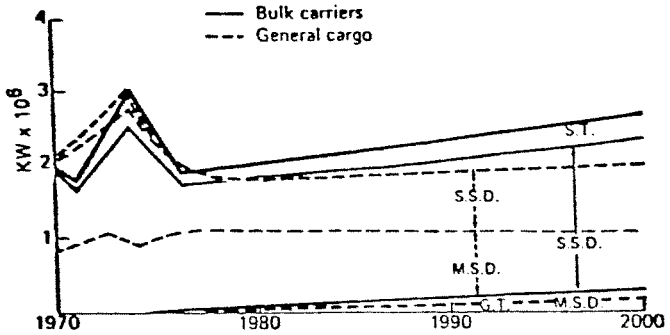

Fig. 11

MACHINERY TRENDS IN TANKERS

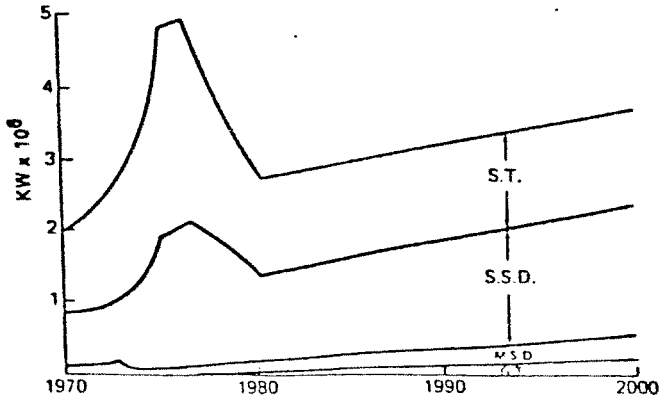

Fig. 10

MACHINERY TRENDS IN CONTAINERSHIPS

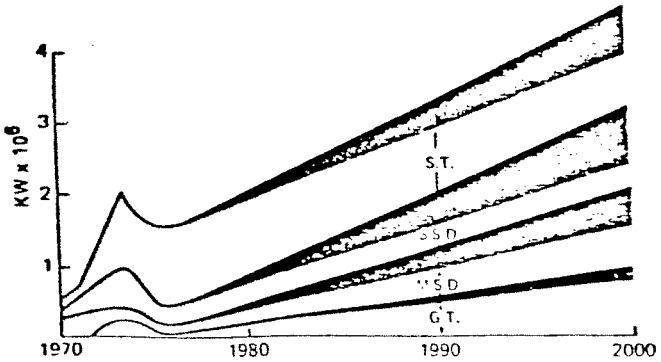

Fig. 12

Table VII
Corrosive and Abrasive Constituents in Heavy Fuel

		Melting Point °C
Vanadium Pentoxide	V_2O_5	690
Sodium Metavanadate	$Na_2O.V_2O_5$	630
Sodium Pyrovanadate	$2Na_2O.V_2O_5$	640
Sodium Orthovanadate	$3Na_2O.V_2O_5$	850
Sodium Vanadate—IV, V	$Na_2O.V_2O_4.5V_2O_5$	625
Pentasodium Vanadate—IV, V	$5Na_2O.V_2O_4.11V_2O_5$	535
Ferric Metavanadate	$Fe_2O_3.V_2O_5$	860
Ferric Vanadate	$Fe_2O_3.2V_2O_5$	855
Sodium Sulphate	Na_2SO_4	880

Table VIII
Composition of Turbo-Charger Blade and Nozzle D[illegible]si

	% Weight
Vanadium	28
Calcium	10
(as $CaSO_4$)	(20)
Sodium	>1
Sulphur	8.2
Nickel	4.2
Iron	>1
Zinc	>1
Magnesium	Trace

Table IX
Comparative Wear Data for Various Fuel Treatments
All for 160 cSt at 50°C Fuel
(Nominal 1,500 sec. Redwood 1 at 38°C)

Engine Type	*Power per Cyl. (kW)*	*Fuel Treatment*	*Average max. liner wear (mm/1,000 hours)*
B & W K74EF	1,270	Homogeniser Centrifuge Filter	0.33 (8)* 0.09 (32) 0.25 (16)
B & W 84VT2BF	1,710	Centrifuge	0.12 (27)
Sulzer RD76	1,200	Centrifuge	0.07 (12)
Unspecified	1,400	Filter	0.31 (8)

* Wear rates averaged over ( ) cylinders

ANNUAL OPERATING COSTS OF
VARIOUS MARINE POWER PLANTS

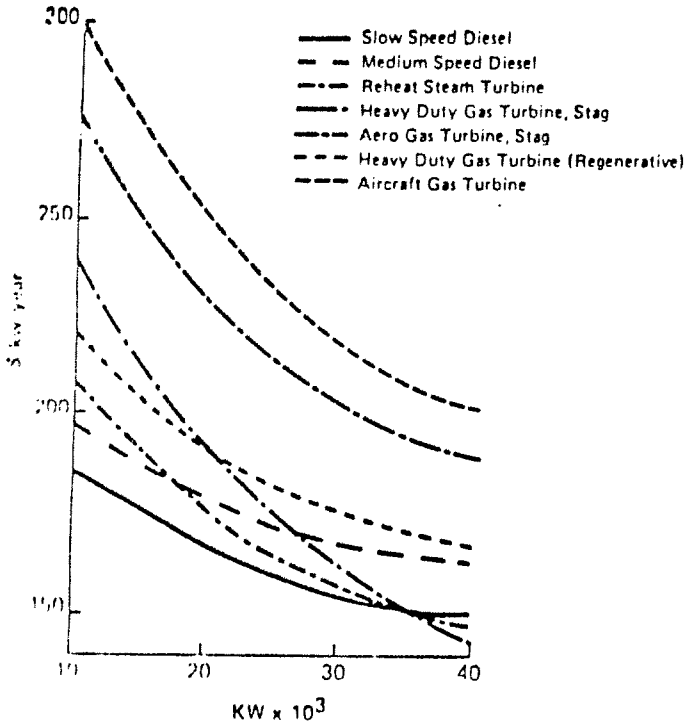

Fig. 13

OVERALL SPECIFIC FUEL CONSUMPTION

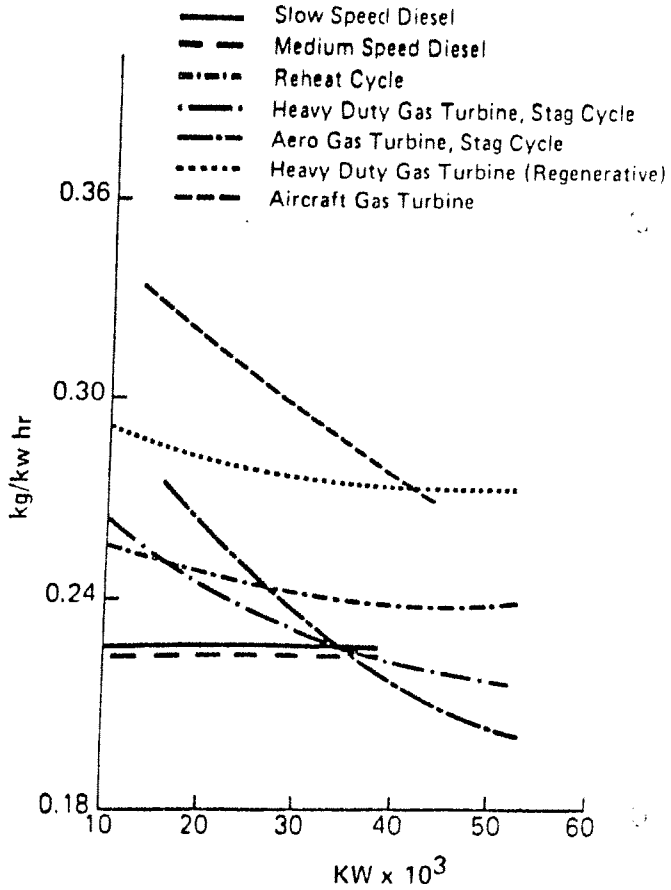

Fig. 14

Table X
Distribution of Diesel BHP on Various Power Ranges

	4 – 9 × 10³ bhp		9 – 15 × 10³ bhp		15 – 20 × 10³ bhp		20 – 25 × 10³ bhp		> 25 × 10³ bhp		TOTAL
	3 – 6.75 MW		6.75 – 11.25 MW		11.25 – 15 MW		15 – 18.75 MW		> 18.75 MW		
YEAR	%	kW × 10⁶	%	kW × 10⁶	%	kW × 10⁶	%	kW × 10⁶	%	kW × 10⁶	kW × 10⁶
1975	17	1.2	32	2.28	12	0.85	20	1.42	19	1.35	7.13
1976	17	1.29	32	2.44	12	0.91	20	1.52	19	1.45	7.63
1977	18.5	1.33	38	2.74	13.5	0.97	15.5	1.12	14.5	1.04	7.22

Table XI
Projected Distribution of Diesel BHP on Various Power Ranges

	4 – 9 × 10³ bhp		9 – 15 × 10³ bhp		15 – 20 × 10³ bhp		20 – 25 × 10³ bhp		> 25 × 10³ bhp		TOTAL
	3 – 6.75 MW		6.75 – 11.25 MW		11.25 – 15 MW		15 – 18.75 MW		> 18.75 MW		
YEAR	%	kW × 10⁶	%	kW × 10⁶	%	kW × 10⁶	%	kW × 10⁶	%	kW × 10⁶	kW × 10⁶
1980	27	2	38	2.85	12	0.91	11	0.83	12	0.91	7.5
1985	25	2	40	3.3	12	0.99	11	0.92	12	0.99	8.2
1990	23	2.1	34	3.05	16	1.45	14	1.25	13	1.15	9.0
1995	22	2.2	30	3.0	20	2.0	14	1.4	14	1.4	10
2000	20	2.25	28	3.15	22	2.5	15	1.65	15	1.65	11.2

PERCENTAGE DISTRIBUTION OF DIESEL B.H.P. ON VARIOUS POWER RANGES

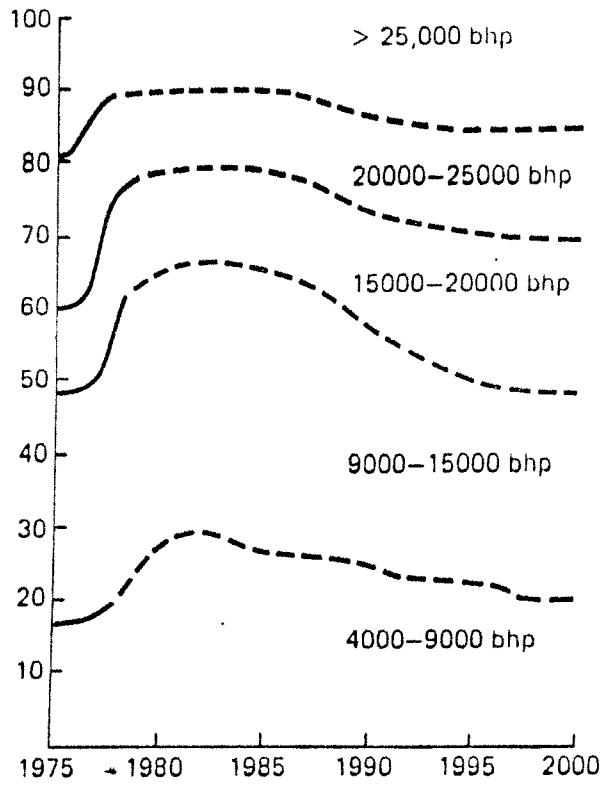

Fig. 15

SHIPS ON ORDER—WORLDWIDE

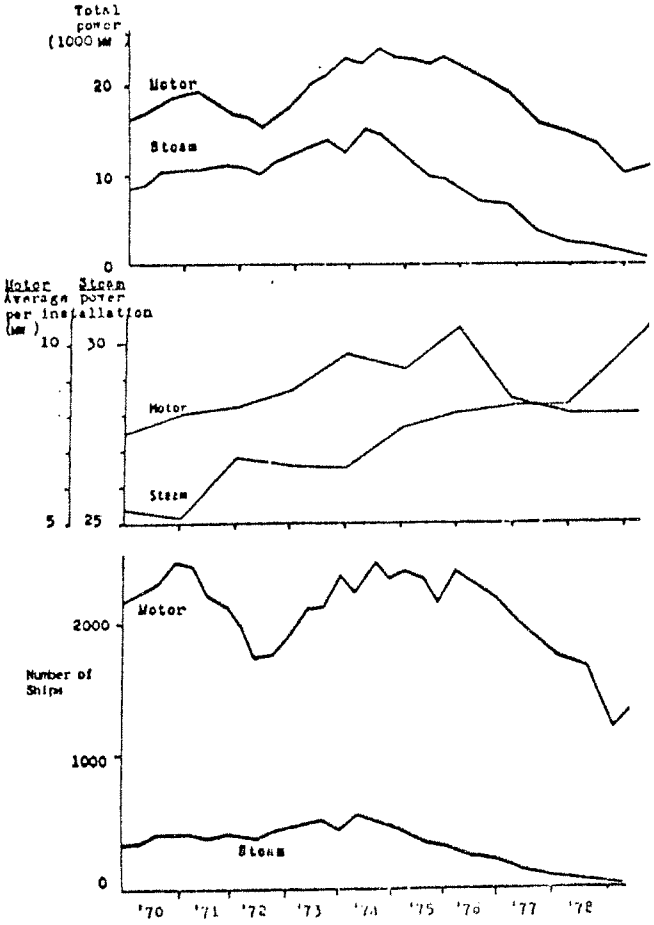

Fig. 16

FUEL RATE FOR VARIOUS STANDARD STEAM CYCLES

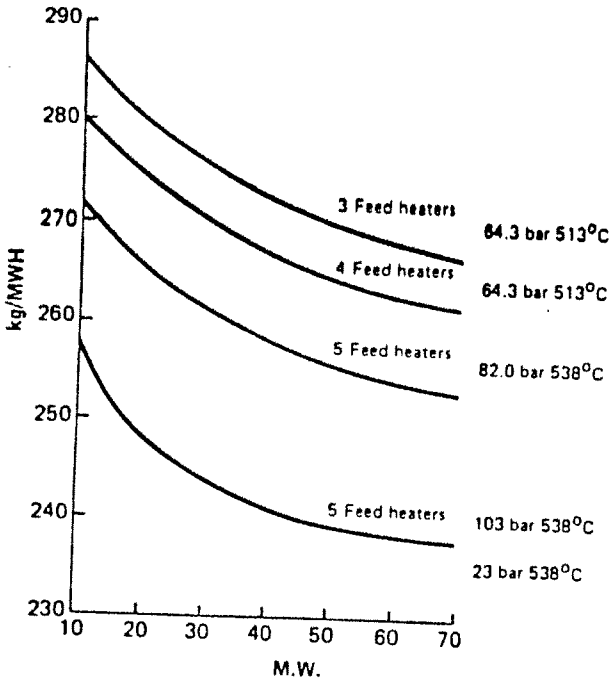

Fig. 17

Table XII
Gas Turbine Characteristics

1. Low specific weight (kg/kW)
2. Low specific volume (m^3/kW)
3. Low specific first cost
4. Minimum requirements for external auxiliary systems
5. Environmental acceptability (emissions, thermal pollution etc)
6. Vibration free
7. Very low lubricating oil consumption
8. Low installation cost since no heavy foundations are necessary
9. Flexibility of installation due to low weight, small size, and no vibrations
10. Ease of maintenance, and low manning requirements
11. Ease of automation

Against the above stated advantages of the gas turbine consider the following main disadvantages:—

1. Higher fuel consumption
2. Sensitivity to fuel contamination
3. Poor part load performance

Table XIII
Characteristics of Combined Power Plants

Plant Description	*Gas Turbine Output MW*	*Steam Turbine Output MW*	*Power Share GT/ST*	*Total Output MW*	*Fuel Consumption kg/MWh*	*Thermal Efficiencies %*	*Gas Turbine Exhaust Temp. °C*	*Stack Temp. °C*	*Initial Steam Condition*
Arrangement with helper gas turbine					246.8			194	
General Electric proposed combined cycle arrangement	20	11.3	64/36	31.3	233.5		498	171	454°C
Dual pressure cycle CPC				20	228				20–40 bar 353 C
Stag 3000				13.75		33		149	
Stag 5211				26		35.3		110	
Stag 5251				31.8		35.1		110	
Stal-Laval Recuperation Cycle	2 × 90	70	72/28	250		47	452	114	28 bar 409°C
Steam Plant with supercharged steam generator (fluid bed combustion)			55/45	20	214				82 bar 603°C

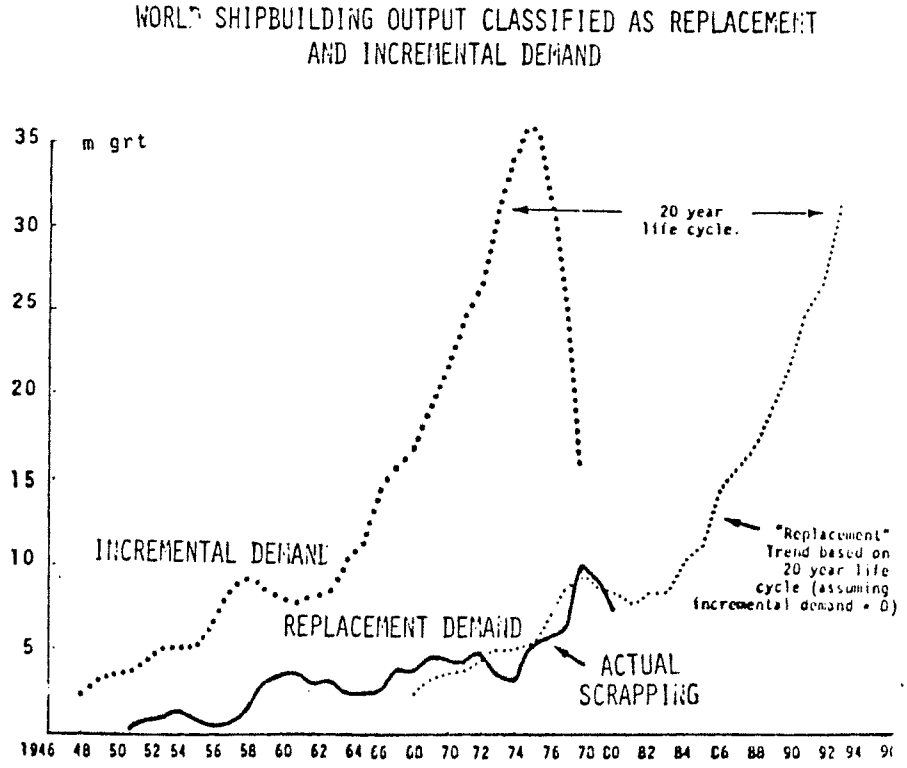

Fig. 18

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

ECONOMIC AND TECHNICAL STUDIES OF FUEL SAVING SHIPS

by

C GALLIN & O HEINDRICH

DELFT UNIVERSITY OF TECHNOLOGY

DELFT

ECONOMIC AND TECHNICAL STUDIES OF FUEL SAVING SHIPS

by

Prof.Dr.Ing. C. Gallin and O. Heiderich

Contents

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   3.3 Container ship 2200 TEU containers
   3.4 Shuttle tanker "Jarena"
   3.5 Cruising car liner "Scandinavia"
   3.6 Cruise ship "Nieuw Amsterdam"
4. Critical survey and Conclusions
5. Acknowledgements

*) each example consists of
   a) a brief description
   b) the ship and its design
   c) the alternative(s)
   d) conclusions

1. Necessity for, and Object of, the Study

As far as shipping and shipbuilding are concerned, these are hard times in which we live. Stagnation and recession in the world economy have brought about a marked reduction in the need for transportation capacity, whilst fuel prices have shown a dramatic upward trend during the seventies. Although there has been a noticeable downward movement in oil prices over the last two years, it should however not be forgotten that these are currently fifteen hundred per cent higher than in 1973, the year of the first oil crisis.

Shipowners are encountering great difficulties in finding employment for their ships, and can barely operate at a profit. They think long and hard before ordering and then expect, quite rightly, to receive an economical vessel. The natural desire for optimization of the ship design has become an imperative.

This necessity for optimization also represents a challenge to the shipbuilding and ancillary industries. A further challenge now comes from the Far East, where wages are low and ships built economically, quickly and efficiently. For this reason, many shipyard directors are in even worse straits than the shipowners. Design engineers in our neighbouring countries have to step up their efforts. But necessity is the mother of invention , and it is said that hard times for the economy are good times for research.

Given these conditions, it is clear that an approach based purely on engineering considerations is no longer sufficient, however effective it may be. The design of a ship or propulsion system must be evaluated in both technical and economic senses, and in the end, it is the economic aspect which forms the basis for the optimization criterion. Not all design engineers allow themselves to be guided by this.

Indeed, it is advisable to deal with the economic side of the design first. In so doing, much worthless technical work, exploring alternatives which stand no hope of being adopted, can be avoided. The title of this essay was chosen deliberately, as an indication of the methodology used in the work.

2. Method and Scope

For the shipowner, the crucial factor is what the ship earns overall, i.e. the final balance. Hence there is little point in highlighting the advantages of individual components in the system "ship", in this case the gears or the clutches. The possibilities offered by the various component combinations and the results which ensue are ultimately decisive. In order to survive in a world of harsh competition, the shipowner should possess qualities of courage and a healthy curiosity when engaged in the risks of his business. This applies similarly to the shipyards. Needless to say, operational safety should never be disregarded.

We must view the whole propulsion system of a ship as one unit. Today, this has come to include, in particular, the production of electric energy by the main engines, i.e. the shaft (alternators. ~~generators.~~ A consideration of other ways of saving energy on board, such as using the exhaust heat or cooling water

of the diesel engines, is beyond the scope of this essay, although the authors do not wish to detract anything from the valuable developments in this area. For similar reasons, ways of decreasing the underwater resistance or increasing propulsion efficiency have also been omitted here. Propeller efficiency, largely dependent on the rpm, and fuel consumption, directly associated with the propeller efficiency, could not be disregarded, however.

The design of a propulsion unit together with its useful and economical application is dependent on ship type. In order to form an opinion, several ship types have to be considered. Here, too, limits to the length of the paper have to be set. The authors have concentrated on three ship types: the container ship, the shuttle tanker and the cruise ship. These have been chosen, firstly, because of their relevance to modern shipping and secondly, because they each require a relatively high amount of electrical energy. This will then give rise to the question of which type of shaft ~~generator~~ alternator to use for direct or indirect main propulsion. Several examples applied to the same ship type will support the conclusions reached.

For each example, an interesting, currently existing ship or a corresponding ship design has been found. The ship or design is called Alternative A. Using the principal of challenger and defender, where A is the defender, one or more challengers have been proposed, and these are labelled Alt.B or Alt.C. The authors must confess to a certain degree of pleasure in carrying out this "constructive" provocation.

This is rather a dangerous game, it being all too easy to come into collision with deeply-rooted opinions or become inadvertently involved in competition, and thus "smash porcelain" unintentionally. This is not the intention nor the purpose of this work, and in order to overcome this hurdle, the components used in each example, e.g. main engines, all come from the same manufacturer (with the exception of one example, where this was impossible).

The authors have also endeavoured to make the comparison of alternatives as objective as possible. For each ship example, the same assumptions were always made so that, for example, the same ship's speed, state of technical development, price situation, etc. are used. Prices are quoted at the December 1983 level, and exchange rates against the US Dollar at the 30.10.82 figures. Prices are given in US Dollars, as this is the currency customarily used for price rating in shipping. The terms employed are also uniform, for example the "continuous service rating (CSR)", set at 0.85 of the "maximum continuous rating (MCR)". Similarly, the same method of calculation is applied to all alternatives. The output data and results are compiled in identical tables. Each table is divided into four groups from top to bottom: the technical data, the initial costs, the operating costs and the economic efficiency criteria. The data for the various alternatives appear from left to right in each table. Alternative A serves as a reference basis for the comparison, while the differences between initial costs and operating costs form the input data for the calculation of the economic criteria of merit.

The following three economic criteria of merit have been used:

1) The "Pay-out Period" (POP), i.e. the time it takes for savings made on fuel to cover the additional capital expenditure. Conversely, given a more economical alternative but higher fuel costs, the time taken for the higher fuel costs to nullify the advantage of low capital costs is also determined as the POP.

2) The "Internal Rate of Return" (IRR), i.e. the internal percentage attained by the corresponding alternative. The higher the IRR, the better the alternative.

3) The "Net Present Value" (NPV), i.e. the present value (cash value) of all savings made on fuel costs over the entire period of service after deduction of additional capital costs. A positive NPV means a good, a negative NPV a poor (loss-making), business enterprise.

Each table includes a list containing explanatory footnotes and a diagram. The explanations of the symbols used in the diagrams are taken from the book "Ships and their propulsion systems" by Gallin, Hiersig, Heiderich.

3. Examples

3.1 Containership 1400 TEU Containers

The first example is a series of four container ships at present under construction at the Chantiers de l'Atlantique shipyard in Saint-Nazaire for the Societé Navale Chargeurs Delmas-Vieljeux and the Cie de Navigation Denis Frères, both Paris-based companies.

CONTAINER SHIP 1400 TEU

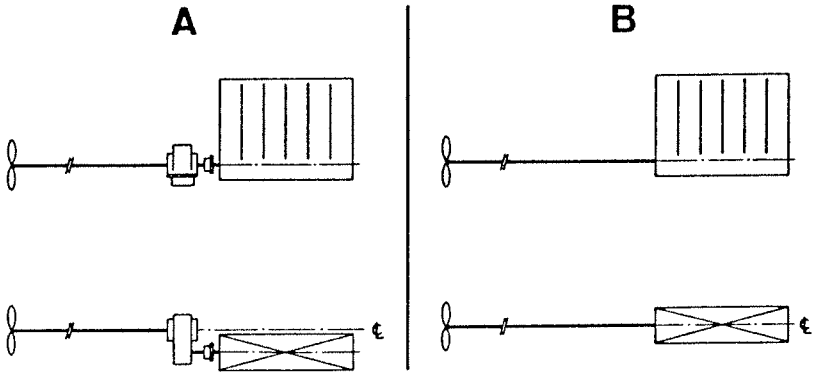

Abb. 1

The design is particularly interesting. The propulsion unit consists of a Sulzer two-stroke diesel engine driving a propeller (with fixed pitch) via a torsionally-elastic Spiroflex coupling KJR 530 and a single-input reduction gearbox GUH 1500, both manufactured by L & S. Details of the unit are given in table 1 and diagram 1. The engine speed is 140 rpm and the propeller speed 77 rpm, the latter being determined in order to accommodate the propeller at the given ship's draught.

These vessels will operate between France and the West Coast of Africa, calling at several ports and often working in estuaries, so that the shipping company has decided to have shaft ~~generators~~ alternators installed.

After this series was ordered, however, two-stroke engines with long strcke and considerably lower engine speeds came on the market. At Sulzer, this new type of engine is commonly known as "RTA".

The question which we can pose in our example is: is there an RTA engine of roughly the same power and a nominal speed of around 77 rpm which could be used in direct-drive without foregoing the advantage of the large, slow-speed, high-efficiency propeller? If there is, what would such a propulsion unit cost and what savings could be expected in operating costs?

As table 1, Alternative B, clearly shows, a positive answer is provided by the basis 5 RTA 84 engine. The entire propulsion unit admittedly

CONTAINER SHIP 1400 TEU				TAB I
ALTERNATIVE			A	B
MAIN ENGINE	NUMBER	–	1	1
	MAKE	–	SULZER	SULZER
	TYPE	–	7 RLB 66 (MCR 1)	5 RTA 84 (R2)
	MCR	KW	11200	12400
	SPEED	RPM	140	82
PROP.	TYPE	–	1xFPP	1xFPP
	SPEED	RPM	77	82
ALTER-NATOR	ON ENGINE	KW	–	–
	GEAR-PTO	KW	–	–
	RID. GEAR	KW	–	–
	INDEPEND.	KW	–	–
POWER TRANS-MISSION	COUPLING	–	KJR 530	–
	PROP. GEAR	–	GUH 1500	–
	RID. GEAR	–	–	–
MAIN ENGINE		1000 $	4934	6136
AUX. ENG./ALTERNAT.		1000 $	–	–
COUPL./CLUTCH/GEAR		1000 $	693	–
SHAFT (SUPPL.)		1000 $	–	65
PROPELLER + RES.		1000 $	APPROX. SAME PRICE	
TOTAL		1000 $	5627	6201
DIFF. INVESTMENT		1000 $	0	+ 574
CSR		KW	9520 1)	9330 2)
SFC		G/KWxH	182	167
FUEL	3)	T/DAY	43,66	39,26
CONSUMPTION	4)	1000 $/YEAR	1856	1669
DIFF. OPERATION		1000 $/YEAR	0	– 187
POP	5)	YEARS		3,7
IRR	6)	%		32,1
NPV	5)	1000 $		848

1) CSR = 0,85 MCR

2) CSR = CSR of Alt. "A"
minus 2 % red. gear losses

3) Inclusive 5 % allowance for L.C.V.

4) 250 sailing days per year
and 170 $ one ton fuel

5) at 10 % interest and
15 years operational life

6) 15 years operational life

applicable for TAB I and TAB Ia

CONTAINER SHIP 1400 TEU				TAB Ia
ALTERNATIVE			A'	B
MAIN ENGINE	NUMBER	–	1	1
	MAKE	–	SULZER	SULZER
	TYPE	–	8 RTA 58 (R1)	5 RTA 84 (R2)
	MCR	KW	11280	12400
	SPEED	RPM	123	82
PROP.	TYPE	–	1xFPP	1xFPP
	SPEED	RPM	77	82
ALTER-NATOR	ON ENGINE	KW	–	–
	GEAR-PTO	KW	–	–
	RID. GEAR	KW	–	–
	INDEPEND.	KW	–	–
POWER TRANS-MISSION	COUPLING	–	KJR 550	–
	PROP. GEAR	–	GUH 1500	–
	RID. GEAR	–	–	–
MAIN ENGINE		1000 $	4740	6136
AUX. ENG./ALTERNAT.		1000 $	–	–
COUPL./CLUTCH/GEAR		1000 $	700	–
SHAFT (SUPPL.)		1000 $	–	65
PROPELLER + RES.		1000 $	APPROX. SAME PRICE	
TOTAL		1000 $	5440	6201
DIFF. INVESTMENT		1000 $	0	+ 761
CSR		KW	9588 1)	9396 2)
SFC		G/KWxH	173	167
FUEL	3)	T/DAY	41,80	39,54
CONSUMPTION	4)	1000 $/YEAR	1777	1680
DIFF. OPERATION		1000 $/YEAR	0	– 97
POP	5)	YEARS		16
IRR	6)	%		9,5
NPV	5)	1000 $		NEGATIVE 23

costs $574,000 more, but this can be regained in three years and ten months by an annual saving of $187,000. At 32.1% the internal rate of return produced by this additional expenditure is relatively high; the present value (NPV) of all savings after deduction of capital expenditure stands at $848,000.

So, seen in purely economic terms, Alternative B is better. Sulzer should be congratulated on the development of the RTA. The shipping company cannot be blamed in any way for not ordering it as it was not available at the time. For this reason, too, the comparison between Alt.A and Alt.B is not entirely appropriate, since this is being made between two different generations of diesel engines with different specific fuel consumptions.

In contrast, the comparison made in table 1a is a correct one. Here, Alt.A has been modernized and the 8 RTA 58 engine has replaced the 7 RLB 66. Alt.A is now Alt. A'. Alt.B is $761,000 more expensive than Alt. A' but the savings made in operation only amount to $97,000/year. It therefore takes 16 years to cover the additional capital costs, an unattractive proposition to any shipowner in the current atmosphere of market instability. Given the normal 10 to 15 years service life of a ship, Alternative B is not economically viable (NPV is negative). A shipowner would always give preference to the $761,000 cheaper Alt.A', i.e. the company of Delmas Vieljeux made the right decision at the time. It is regrettable that Sulzer has not yet brought out a series of two-stroke diesel engines with increased speeds, since these would go even further

towards improving the economic advantage of indirect drive in such cases as these.

3.2 Container ship 1450 TEU containers

The example here is of a container ship ordered by a German shipowner which is still in the design stage. The shipyard offered to reproduce an existing ship and equip it with a propulsion unit of the specification given in table II under "Alt.A". The propulsion unit consisted of a two-stroke diesel engine operating a fixed-pitch propeller in direct-drive. Three diesel ~~generators~~ alternators and a shaft ~~generator~~ alternator fitted in the shafting were installed to provide electric energy. The shaft ~~generator~~ alternator was equipped with a thyristor unit to keep the mains frequency constant when shaft speed varies.

The shipowner asked whether there were alternatives which could produce the same power (or more) but at cheaper initial or operating costs. He also enquired whether it was possible to instal more powerful propulsion units in the same engine room in order to increase the ship's speed.

In this exceptional case, the result was the simultaneous installation of two and four-stroke engines made by different manufacturers.

Owing to the limited scope of the work, we will concentrate only on the first question. Table II provides a general synopsis, and diagram 2 expresses this visually.

CONTAINER SHIP 1450 TEU

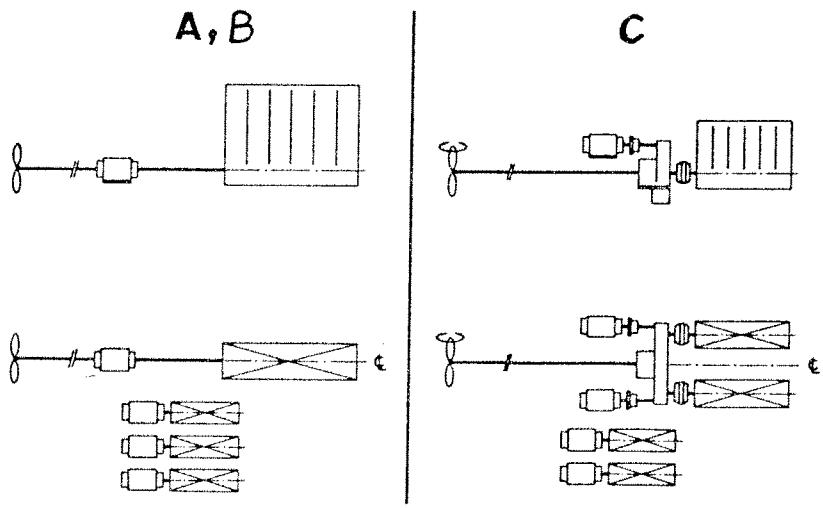

Abb. 2

CONTAINER SHIP 1450 TEU						TAB II
ALTERNATIVE			A	B	C1	C2
MAIN ENGINE	NUMBER	–	1	1	2	1
	MAKE	–	MAN-B&W	SULZER	MaK	
	TYPE	–	5L80GFCA	6RLB76 (MCR2)	6M601	
	MCR	KW	11300	12000	12000	6000
	SPEED	RPM	106	110	425	
PROP.	TYPE	–	1xFPP	1xFPP	1xCPP 1)	
	SPEED	RPM	100	104	70	
ALTER-NATOR	IN SHAFT	KW	1x790 2)	1x790 2)	–	
	GEAR-PTO	KW	–	–	2x790	
	RID. GEAR	KW	–	–	–	
	INDEPEND.	KW	3x790 3)	3x790 3)	2x790 3)	
POWER TRANS-MISSION	CLUTCH	–	–	–	KAA380	
	PROP.GEAR	–	–	–	GVA1900	
	RID. GEAR	–	–	–	–	
MAIN ENGINE		1000$	3552	3552	2537	
AUX.ENG./ALTERNAT.		1000$	846	846	354	
COUPL./CLUTCH/GEAR		1000$	–	–	592	
SHAFT (SUPPL.)		1000$	–	–	–	
PROPELLER + RES.		1000$	144+144	156+156	683+29	
TOTAL		1000$	4686	4710	4195	
DIFF. INVESTMENT		1000$	0	+24	-491	
CSR		KW	9600	9700 4)	9550 5)	5200 6)
SFC		G/KWxH	186	184	182	
FUEL		T/DAY	45,00 7)	45,00 7)	43,13 8)	23,18 8)
CONSUMPTION	9)	1000$/YEAR	2065	2065	1980	1064
DIFF. OPERATION		1000$/YEAR	0	0	-85	-1001
SHIP'S SPEED		KNOTS	18,1	18,1	18,1	15,0
	10)					
NPV		1000$			2531	

1) Propeller diameter D_p = 7,3 m
2) Frequency controlled alternator
3) Diesel engine MaK 6 M 332 (830 kW at 720 rpm)
4) CSR = CSR of Alt. "A"
+ 1 % eff. loss (higher prop. rev.)
5) CSR = CSR of Alt. "A"
+ 2 % red. gear losses
+ 2 % CPP efficiency loss
- 4,5 % effic. gain (lower prop. rev.)
6) CSR = 0,85 MCR of one engine
+ 2 % red. gear losses
+ 2 % CPP efficiency loss
- 2 % effic. gain (lower prop. rev. but max. rpm = ct.)
7) Inclusive 5 % allowance for L.C.V.
8) Inclusive 5 % allowance for L.C.V.
minus 0,67 t/day difference in fuel consumption for one auxiliary engine less
9) 270 sailing days per year
and 170 $ one ton fuel
10) at 10 % interest,
15 years operational life, with
both engines in oper. 220 days (80 %) of sea time
one engine in oper. 50 days (20 %) of sea time

In principle, Alt. B is the same as Alt.A, - only the engine manufacturer is different. This shows how keen the competition is between manufacturers: six cylinders instead of five and 700 kW more power are offered at the same price.

This particular shipowner has had satisfactory experiences with four-stroke engines, and is now extremely interested in the possibility of operating economically with partial loading (slow-steaming).

So the third alternative consisted of two four-stroke Mak 6M 601 diesel engines each driving a controllable-pitch propeller via highly-elastic Pneumaflex clutches ~~couplings~~ KAA 380 and a ~~multi-~~ twin-input single-output L & S reduction gearbox GVA 1900. This third alternative is subdivided into Alt. C 1, where both engines, and Alt. C 2, where only one engine, is in use. It is widely known that the controllable-pitch propeller is essential in order to be able to use the power of one engine to the full, as in Alt. C 2. The controllable-pitch propeller also enables operation using partial loading with constant rpm, thus rendering superfluous the installation of an expensive shaft alternator ~~generator~~ with the thyristor unit. This was replaced by an alternator driven via a power take-off (PTO) on the reduction gearbox. A second PTO was added, thus both replacing and saving the cost of a further diesel unit (see table II).

As expected, Alt. B differs only slightly from Alt. A in initial and operating costs.

In contrast to Alt.A, Alt. C is almost half a million dollars (491,000) cheaper, despite the reduction gearbox and the couplings and clutches.

This fact can be attributed to the reasonably-priced main engines and shaft (alternators ~~generators~~) and to the smaller number of auxiliary units.

The operating costs are also more favourable ($85,000/year with Alt. C1 and $1,001,000/year with Alt.C2), so that the savings made here must be continuously added to the savings on initial costs. Between twin and single-engine operation (Alt. C1 and C2) the ship's speed decreases from 18.1 to 15.0 kn, i.e. by only 17%. In any case, a container ship of this size operates between nearby ports and in river service on reduced power. If we assume that the vessel operates for 80% of the time on two and 20% of the time on only one engine, the savings on fuel will amount to $268,000/year. Given the 15 year operational life of these ships, the present total value (NPV) of all savings comes to $2,531,000!

The propulsion unit with a two-stroke engine can certainly also operate partially loaded when slow-steaming. A CSR = 0.5 x 0.85 = 42.5% is however not good for the engine in the long run. Neither is the specific fuel consumption at an optimum with 42.5% partial loading. In the case of Alt. C 2, however, the engine runs at an optimum speed on 85% power.

CONTAINER SHIP 1450 TEU					TAB. IIa
SYM-BOL	ITEM	UNIT	ALTERNATIVE		
			A	B	C
a	MAIN ENGINE(S)	MAKE	B&W	SULZER	MaK
		TYPE	5 L 80 GFCA	6 RL B 76	2 x 6 M 601
		1000 $	3552[1]	3552[1]	2537
b	COUPLINGS	TYPE	–	–	2 x KAA 380
		1000 $	–	–	97
c	REDUCT. GEAR	TYPE	–	–	GVE 1900
		1000 $	–	–	495
d	CONTROLLABLE PITCH PROPELLER	Ø MM	6400	6400	7300
		1000 $	517[2]	555[2]	683[2]
	RES. BLADE	1000 $	26	27	29
e	SHAFT GENERATOR	KW	790	790	790
		1000 $	178[3]	178[3]	25[4]
GRAND TOTAL		1000 $	4273	4312	3866
	ITEMS a+b+c	1000 $	3552	3552	–
d'	FIXED PITCH PROPELLER	Ø MM	6400	6400	–
		1000 $	144	156	–
	RES. PROPELLER	1000 $	144	156	–
e'	SHAFT GENERATOR	KW	790	790	–
		1000 $	338[5]	338[5]	–
f	PROPELLER SHAFT	Ø MM	485	490	–
		1000 $	67	60	–
GRAND TOTAL		1000 $	4245	4262	–

1) inclusive balancer
2) inclusive prop. shaft
3) riding gear on the shaft
4) power take off + clutch
5) generator in the shaft
+ frequency control

Particularly worthy of mention in this example is table IIa. At the top of the table the initial costs of the units A, B and C are again given, and Alt. A and B have controllable-pitch propellers. The lower section of the table shows the initial costs of Alt. A and B with fixed-pitch propellers. The most striking feature here is that there is virtually no difference in price between the units with fixed-pitch and those with controllable-pitch propellers. This can be attributed to the considerable difference in price between the various types of shaft (alternator, ~~generator~~, and to the fact that a reserve blade is sufficient for the controllable-pitch, whilst for the fixed-pitch propeller a complete spare propeller has to be provided.

In other words, taking the initial costs of the propulsion unit as a whole, it is possible to make a free choice between fixed-pitch and controllable-pitch propellers at the same price. A controllable-pitch propeller offers great advantages in manoeuvrability and contributes towards maintaining the scheduled service speed even in inclement conditions. The 2% loss in efficiency caused by the larger hub has to be tolerated, and this has been taken into account in the tables.

(page 13 is deleted)

3.3 Container ship 2200 TEU

The third example is "hot". It is the enquiry for a 2200 TEU container ship, a current project that is at present in circulation on the shipbuilding market. The vessel is to be engaged between ports on the European Atlantic coast (Felixstowe, Hamburg, Rotterdam, Antwerp, Le Havre) and Asia.

A service speed of 20.5 kn is anticipated for a propulsion power of 32,000 hp, i.e. 23,530 kW. Under discussion as a possibility is a Sulzer 8 RTA 84 engine, operating at point R1 on the Sulzer layout diagram,

producing 23,680 kW at 87 rpm in direct drive. Apart from the usual independent diesel units, a turbo unit fed by exhaust heat and an alternator with thyristor control attached to the main engine will be used to produce electric energy. The costs of the propulsion units based on these plans are given in table III below Alt.A. Diagram 3/a shows a schematic representation of the unit.

In accordance with the previous example and the indisputable advantages of slow-steaming (as a continuous operational condition if possible), Alternative B is a twin-engine system. Engines of the same type (two-stroke) with long stroke and manufacturer (Sulzer) have been chosen for Alt.B so as to keep the conceptions of the shipping company intact and to limit the number of design parameters to be investigated to one (the number of engines). Given these limitations and the high power, the choice is not so wide: the engines in Alt.B run (in the case of R1) at 105 rpm, a very low speed for indirect drive, and this causes a great increase in the cost of the unit.

In addition, the draught of this ship was given as 12.2m. With a ratio of propeller diameter (Dp) to ship's draught of 0.70 - 0.75, a

$$Dp = 8.54 - 9.15 \text{ m}$$

results. For a 9.0m propeller, the optimum propeller speed for maximum continuous rating (MCR) is 78 rpm. For a fixed-pitch

CONTAINER SHIP 2200 TEU					TAB III
ALTERNATIVE			A	B1	B2
MAIN ENGINE	NUMBER	–	1	2	1
	MAKE	–	SULZER	SULZER	
	TYPE	–	8RTA84 (R1)	6RTA68 (R1)	
	MCR	KW	23680	23400	11700
	SPEED	RPM	87	105	
PROP.	TYPE	–	1xFPP	1xCPP 1)	
	SPEED	RPM	82	70	
ALTER-NATOR	ON ENGINE	KW	1x1100 2)	–	
	GEAR-PTO	KW	–	2x1100	
	RID. GEAR	KW	–	–	
	INDEPEND.	KW	3x1100 3)	2x1100 3)	
POWER TRANS-MISSION	CLUTCH	–	–	KAA550	
	PROP. GEAR	–	–	GVA2450	
	RID. GEAR	–	–	–	
MAIN ENGINE		1000 $	8962	10030	
AUX. ENG./ALTERNAT.		1000 $	1247 4)	536	
COUPL./CLUTCH/GEAR		1000 $	–	1852 5)	
SHAFT (SUPPL.)		1000 $	–	–113	
PROPELLER + RES.		1000 $	288+288	1160+86	
TOTAL		1000 $	10785	13551	
DIFF. INVESTMENT		1000 $	0	+2766	
CSR		KW	20130 6)	20520 7)	10540 8)
SFC		G/KWxH	170	171	
FUEL 9)		T/DAY	87,18 10)	88,42	45,42
CONSUMPTION 11)		1000 $/YEAR	4001	4058	2085
DIFF. OPERATION		1000 $/YEAR	0	+57	–1916
SHIP'S SPEED		KNOTS	20,5	20,5	~17
NPV 12)		1000 $		102	

1) Propeller diameter appr. 9,5 m

2) Frequency controlled alternator

3) Diesel engine MaK 6 M 332 (1200 kW at 900 rpm)

4) Inclusive one frequency controlled alternator

5) Inclusive two alternators

6) CSR = 0,85 MCR

7) CSR = CSR of Alt. "A"
+ 2 % red. gear losses
+ 2 % CPP losses
– 2 % efficiency gain at slow running

8) CSR = CSR of one engine
+ 2 % red. gear losses
+ 2 % CPP losses
+ 2 % (constant prop. rev. at partial load)

9) Inclusive 5 % allowance for L.C.V.

10) Inclusive 0,94 t/day difference in fuel consumption for one more auxiliary engine.

11) 270 sailing days per year
and 170 $ one ton fuel

12) at 10 % interest and
15 years operational life, with
both engines in oper. 210 days (78 %)
one engine in oper. 60 days (22 %)

propeller, the propeller speed would be approximately 74 rpm at a service rating of CSR = 0.85 MCR. Careful designing and an appropriate stern form could lead to speeds as low as 70 rpm for indirect drive (Alt.B). Admittedly, direct drive (Alt.B) speeds of 87 and 82 rpm come close to this. The approximately 2% increase in propulsion efficiency for the engine speed, reduced by the L & S ~~double~~ twin-input reduction gearbox GVA 2450, is of lesser significance to Alt.B.

Therefore we can ask the question: Are the advantages of a twin-engine propulsion unit "slow-steaming" (see example 3.2), i.e. increased operational reliability, possibility of maintenance work at sea without operational interruption, better manoeuvrability (controllable-pitch propeller) etc, financially justifiable? Are the results of the economic comparison correct?

As every shipowner is discovering, being able to run fast now on schedule and fully laden may be over-mortgaging the future.

The doors must remain open to the possibility of "catching smaller fry", i.e. operating optimally but on reduced schedules.

Table III shows that Alt.B is in fact ~~$5,766,000~~ $ 2,766,000 more expensive in terms of initial costs. The operating costs ($57,000/year) are somewhat higher when operating on both engines at full speed (Alt.B 1). For single-engine operation (Alt.B 2) however, the operating costs prove to be considerably less ($1,916,000/year). In this case, the additional capital costs would be recovered in under two years.

Given a mixture of the two, and the assumed operational life of the ship, the financial balance can be achieved using one engine for 22% of the time. In inter-port operation, at each end of the service route, this time ratio between Alt.B1 and B2 can always be expected. In the case of Alt. B 2, the vessel nevertheless operates at a speed of approx. 17 kn.

SHUTTLE TANKER „JARENA“

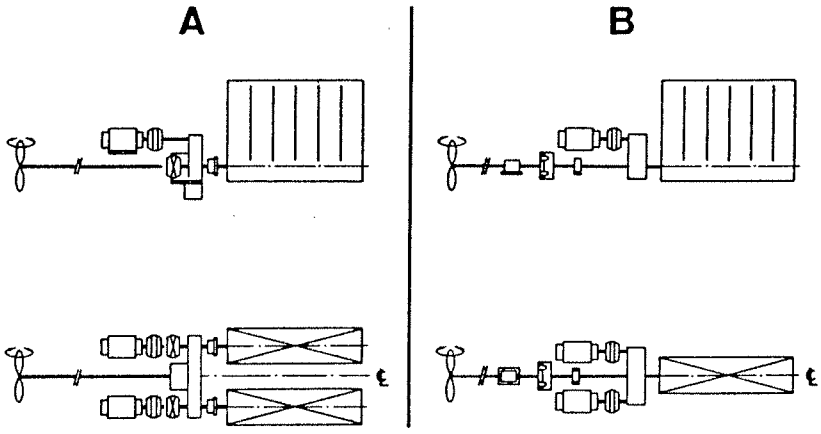

Abb. 4a

3.4 The Shuttle tanker "Jarena"

This ship was selected for study because its twin-engine propulsion unit fulfills an extreme design requirement: each of its two shaft ~~generators~~ alternators can absorb up to 70% of the power of one of the main engines (see table IV, Alt. A).

To explain this requirement, we must also explain the type of service in which the ship is engaged. The Norwegian state-owned oil company "Statoil" has designated the ship for the regular transportation of crude oil from the Statfjord field to the refineries on the North Sea coast. The total distance of 1000 NM (approx. 150 hours) for outward and return journeys is unusually short for a crude oil tanker. For this ferry-like type of service it is the turn-round times which determine the economic efficiency of the vessel, since large amounts of power are needed to operate the pumps.

The propulsion unit comprises two two-stroke diesel engines connected to the ~~multi-~~ twin-input single-output L & S ~~double-input~~ reduction gearbox GVS 2400 (diagram 4/A) each via a non-clutchable elastic Spiroflex coupling KJT 490 and a torsionally-rigid external Pneumastar clutch KUG 490. The gearbox is equipped with two high-performance power take-offs. Each PTO can drive an alternator at 1200 rpm via elastic Pneumaflex clutches KAD 320. The quillshaft design of the ~~double-~~ twin-input reduction gearbox enables this even with an idle propeller.

For the 'Challenger' (Alt.B) was proposed a single-engine direct drive system operating the ~~generators~~ alternators via a riding L & S GUG 60/225 gearbox with two power take-offs ~~(diag.~~ (diag. 4A).

SHUTTLE TANKER „JARENA"				TAB IV
ALTERNATIVE			A	B
MAIN ENGINE	NUMBER	–	2	1
	MAKE	–	MAN-B&W	MAN-B&W
	TYPE	–	8L55 GBE 1)	6L90GBE 1)
	MCR	KW	16240	17490
	SPEED	RPM	155	97
PROP.	TYPE	–	1xCPP	1xCPP
	SPEED	RPM	80	97
ALTER-NATOR	ON ENGINE	KW	–	–
	GEAR-PTO	KW	2x5500	–
	RID. GEAR	KW	–	2x5500
	INDEPEND.	KW	–	–
POWER TRANS-MISSION	CLUTCH	–	KJT/KUG490	KAD320
	PROP.GEAR	–	GVS2400	–
	RID. GEAR	–	–	GUG60/225
MAIN ENGINE		1000$	5256	4642
AUX. ENG./ALTERNAT.		1000$	–	–
COUPL./CLUTCH/GEAR		1000$	1680	770
SHAFT (SUPPL.)		1000$	–	63
PROPELLER + RES.		1000$	APPROX. SAME PRICE	
TOTAL		1000$	6936	5475
DIFF. INVESTMENT		1000$	0	– 1461
CSR		KW	13800 2)	14070 3)
SFC		G/KWxH	175	170
FUEL 4)		T/DAY	60,86	60,28
CONSUMPTION 5)		1000$/YEAR	2587	2562
DIFF. OPERATION		1000$/YEAR	0	– 25
NPV 6)		1000$		1651

1) Type "GBE" in place of installed "GFCA"

2) CSR = 0,85 MCR

3) CSR = 0,85 MCR of Alt. "A"
- 2 % red. gear saving
+ 4 % eff. loss (higher prop. rev.)

4) 5 % allowance for lower L.C.V.

5) 250 sailing days per year
and 170 $ one ton fuel

6) 10 % interest and
15 years operational life

So as to make a proper comparison, the engines chosen in both alternatives were from the same manufacturer (MAN - B & W) and generation. Thus Alt. A was modernized, and engine 8 L 55 GBE adopted instead of 8 L 55 GFCA. In both cases the controllable-pitch propeller was retained as necessary to the service requirements of the ship, which include regular mooring at buoys and in port. In Alt.B, a Separex KTA disconnecting coupling and a separate thrust bearing in the propeller shaft are needed in order to drive the shaft (alternators ~~generators~~ when the propeller is idle.

Table IV shows that Alt.B is in fact considerably more favourable ($1,461,000) than Alt.A. Its operating costs are also marginally more advantageous ($25,000/year), so that the value of total savings increases even further (NPV $1,651,000).

So is Alt.B preferable? Probably not, because it lacks the operational flexibility essential to the ship. If both (alternators ~~generators~~ simultaneously, or the controllable-pitch propeller, were not fully loaded, the main engine would often run partially loaded in continuous operation.

In offshore enterprises, operational reliability is of decisive importance, and in this respect Alt. B is not necessarily quite comparable with Alt.A, in which the various combinations of operational possibilities make it particularly attractive. Diagram 4b clearly illustrates this.

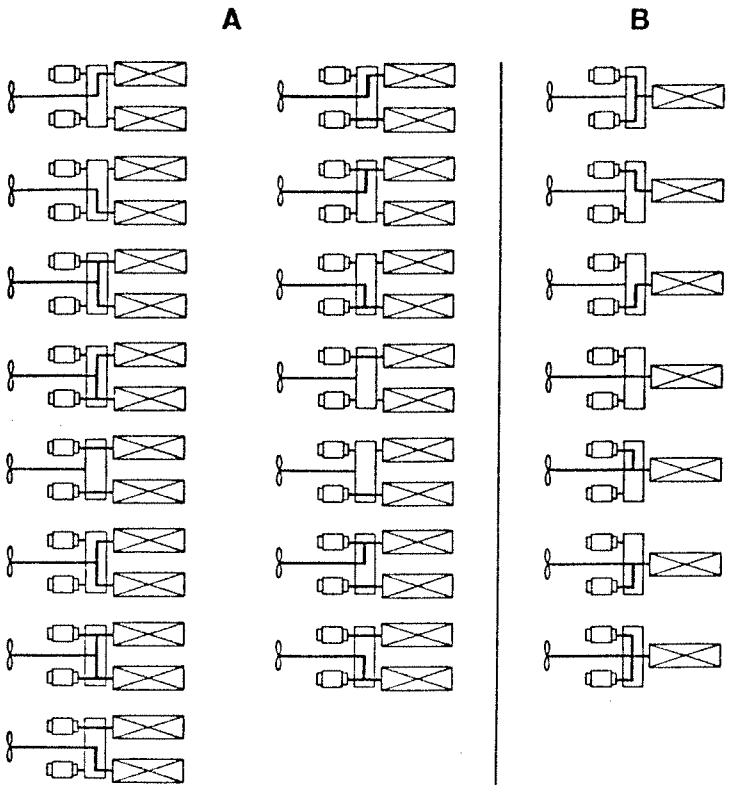

Abb. 4 b

Finally, the time involved in disconnecting the Separex coupling installed in the propeller shafting, a process repeated for every discharge operation, should also be taken into account.

In view of the particular service duties of the vessel, the shipping company's decision to give preference to the operational reliability and flexibility over the cost factor can be readily understood.

3.5 The Cruising Car-liner "Scandinavia"

Unlike the tanker trade, cruising is still a lucrative business in the shipping world. The passenger capacity is evidently insufficient, for older ships are being lengthened and new ones ordered. A newbuilding was recently delivered to the Dubigeon Normandie SA yard in Nantes, France, for the "Scandinavia World Cruise", Nassau (a subsidiary of DFDS, Copenhagen). Two other cruise ships, the "Nieuw Amsterdam" and the "Noordam", are at present under construction at Chantiers de l'Atlantique in Saint Nazaire for the Holland America line.

There are good grounds therefore for studying the cruise ship as a ship type currently very much in vogue. This is also true of the propulsion unit, as two-stroke diesel engines in direct drive have recently come to be used in cruise ships, including the two above-named newbuildings from France. The rather limited draught of these two ships (7.15 and 7.40 m respectively) excludes the option of very low propeller speeds.

CRUISING CAR-LINER „SCANDINAVIA"

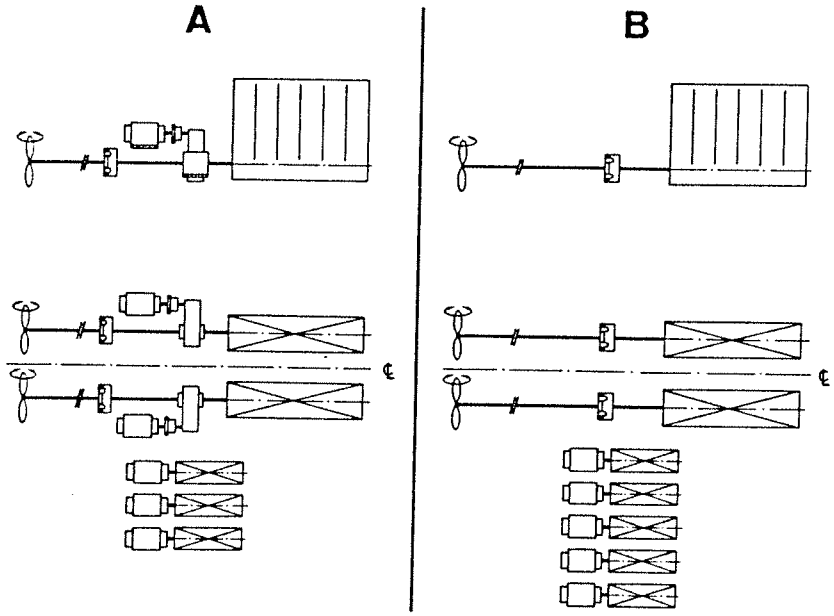

Abb. 5/a

Like the other ship types discussed, cruise ships need large amounts of energy to cover the hotel requirements on board, i.e. energy for light, ventilation, air-conditioning, heating, water preparation, the fin stabilisers, bow thrust unit, etc. Here, too, the optimum way of producing energy is worthy of investigation.

The frequent port calls and cruising speeds of these ships necessitate controllable-pitch propellers. These obviate the need for shaft (alternators ~~generators~~ fitted with thyristors. Twin-screw propulsion is therefore given preference.

Details of the "Scandinavia's" propulsion unit can be seen in table IV, Alt. A, whilst its design concept is shown in diagram 5/A.

Electric energy is produced at sea by two shaft (alternators ~~generators~~ powered via riding L & S GUG 42/190 gearboxes. The gearboxes increase the speed of the main engines (155 rpm) to 900 rpm for the alternators. These have individual outputs of 2100 kW.

Three separate diesel units cater for port operations and reserve needs. Each unit produces 2100 kW and is driven by a MaK 8M 453 four-stroke engine (2210 kW at 600 rpm).

CRUISING CAR-LINER „SCANDINAVIA"					TAB V
ALTERNATIVE			A	B	
MAIN ENGINE	NUMBER	–	2	2	
	MAKE	–	MAN-B&W	MAN-B&W	
	TYPE	–	9L55GB 1)	7L55GB	
	MCR	KW	21330	16590	
	SPEED	RPM	155	155	
PROP.	TYPE	–	2xCPP	2xCPP	
	SPEED	RPM	155	155	
ALTER-NATOR	ON ENGINE	KW	–	–	
	GEAR-PTO	KW	–	–	
	RID. GEAR	KW	2x2100	–	
	INDEPEND.	KW	3x2100 2)	5x2100 2)	
POWER TRANS-MISSION	COUPLING	–	KJR/KTA	KTA114/17/4	
	PROP. GEAR	–	–	–	
	RID. GEAR	–	GUG42/190	–	
MAIN ENGINE		1000 S	5728	4744	
AUX. ENG./ALTERNAT.		1000 S	1575 3)	2625 3)	
COUPL./CLUTCH/GEAR		1000 S	595	132	
SHAFT (SUPPL.)		1000 S	–	18	
PROPELLER + RES.		1000 S	APPROX. SAME PRICE		
TOTAL		1000 S	7898	7519	
DIFF. INVESTMENT		1000 S	0	–379	
CSR		KW	18130 4)	13710 5)	4420 6)
SFC		G/KWxH	180	180	195
FUEL 7)		T/DAY	82,24	62,19	21,72
CONSUMPTION 8)		1000 S/YEAR	3495	2643	923
DIFF. OPERATION		1000 S/YEAR	0	+71	
DIFF. INVEST. LOST		YEARS		8	
NPV 9)		1000 S		NEGATIVE 161	

1) Type "GB" in place of installed "GFCA"

2) Diesel engine MaK 8 M 453 (2210 kW at 600 rpm)

3) Price inclusive foundation but exclusive alternator

4) CSR = 0,85 MCR

5) CSR = 0,85 MCR of Alt. "A"
minus $\frac{2 \times 2100}{0,95}$ = 4420 kW

6) CSR of one auxiliary diesel 4420 kW

7) Inclusive 5 % allowance for L.C.V.

8) 250 sailing days per year at sea (70 % of service time) and 170 $ one ton fuel

9) at 10 % interest and
15 years operational life

Given the boundary conditions together with the logical and simple design concept of this propulsion unit we should now proceed to examine whether the installation of shaft alternators ~~generators~~ is really a sensible proposition financially - since in any case the power of the shaft alternators ~~generators~~ amounts to around 20% of the power of the main engines. The heavy and expensive two-stroke propulsion engines would decrease in power by this percentage through the installation of separate diesel units. The faster and therefore lighter four-stroke engines of the auxiliary units must be cheaper. Against this, however, their specific consumption is higher, i.e. they incur higher fuel costs, although at present the auxiliary units can also run on heavy oil.

Alt.B in table V shows the situation when, instead of two shaft alternators, ~~generators,~~ two separate diesel units (five altogether) are installed.

Since this ship will operate like a ferry between New York and the Bahamas, a total of approximately 250 days at sea per year has been assumed.

The initial cost of the entire Alt.B system is in fact $379,000 lower, but fuel costs $71,000/year higher. The advantage gained in initial costs will therefore be eroded within 8 years by the higher fuel costs (at an interest rate ~~a rate of return~~ of 10%). Assuming an operational life of 15 years, the net result will be a loss of $161,000 (negative NPV).

CRUISING SHIP „NIEUW AMSTERDAM“

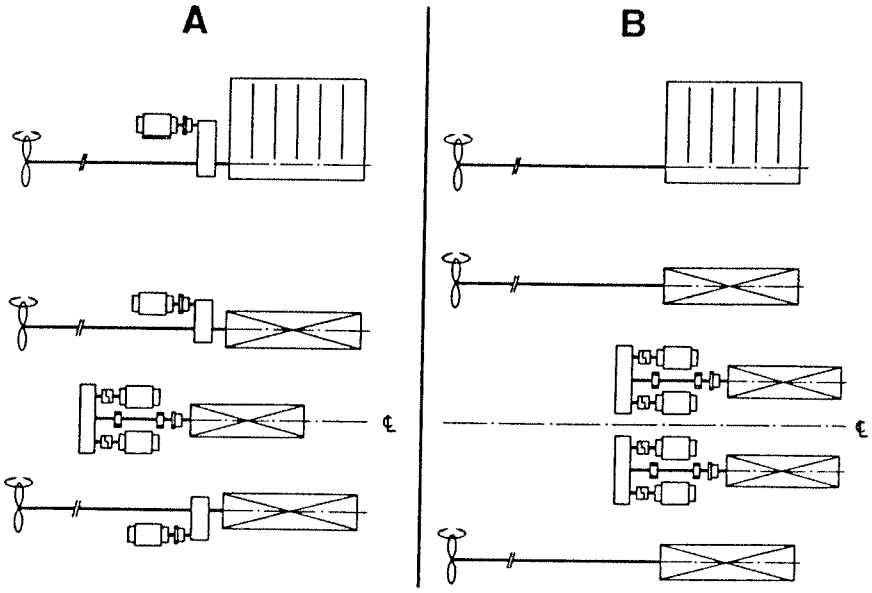

Abb. 6/a

It should not be forgotten that in Alt. A the auxiliary units on board comprise a total of 3 x 8 = 24 cylinders, whereas Alt.B has 5 x 8 = 40; this means that all cylinders have to be serviced.

In summary : Alt. B is only a viable proposition if the shipowner intends to keep the vessel for no more than 8 years.

Although the drawing shows the unit as being simple in design, in reality it is quite sophisticated, e.g. the propeller has a "feathering position", the propeller shaft a disconnecting coupling, and so on.

3.6 The Cruise Ship "Nieuw Amsterdam"

The design concept of the propulsion unit in this case is similar to that of the "Scandinavia", although here shipowner and yard introduced a new aspect to ship design by fitting the auxiliary unit also with a two-stroke diesel engine (diag. 6/A). A Sulzer engine type RLB 56 with only four cylinders simultaneously drives two 2000 kW alternators via an L & S GJZ 1020 branched gearbox. The achievement of this design concept is to have an auxiliary diesel with a high power output and hence an even lower specific fuel consumption.

The question posed in connection with the "Scandinavia" also applies to the "Nieuw Amsterdam", i.e. do shaft (alternators ~~generators~~ with riding gearboxes, in view of the new situation, still have a purpose?

Shaft (alternators ~~generators~~ are used for sea operations and the auxiliary unit for port operations. What is the result, when an auxiliary unit of this type

CRUISING SHIP „NIEUW AMSTERDAM"					TAB VI
ALTERNATIVE			A	B	
MAIN ENGINE	NUMBER	–	2	2	
	MAKE	–	SULZER	SULZER	
	TYPE	–	7 RLB 66 (ERP1)	6 RLB 66 (ERP1)	
	MCR	KW	20300	17400	
	SPEED	RPM	135	135	
PROP.	TYPE	–	2 x CPP	2 x CPP	
	SPEED	RPM	135	135	
ALTER-NATOR	ON ENGINE	KW	–	–	
	GEAR-PTO	KW	–	–	
	RID. GEAR	KW	2 x 2400.	–	
	INDEPEND.	KW	1 x 4000 1)	2 x 4000 1)	
POWER TRANS-MISSION	COUPLING	–	KJ 200/KJ 430	KJS 430	
	ALT. GEAR	–	1 x GJZ 1020	2 x GJZ 1020	
	RID. GEAR	–	GUG 45/150	–	
MAIN ENGINE		1000 $	9868	7054	
AUX. ENG./ALTERNAT.		1000 $	2054 2)	4108 2)	
COUPL./CLUTCH/GEAR		1000 $	477	412	
SHAFT (SUPPL.)		1000 $	–	20	
PROPELLER + RES.		1000 $	APPROX. SAME PRICE		
TOTAL		1000 $	12399	11594	
DIFF. INVESTMENT		1000 $	0	– 805	
CSR		KW	17260 3)	13050 4)	4210 5)
SFC		G/KW x H	178	178	182
FUEL 6)		T/DAY	77,42	58,54	19,31
CONSUMPTION 7)		1000 $/YEAR	2303	1742	574
DIFF. OPERATION		1000 $/YEAR	0	+ 13	
NPV 8)		1000 $		706	

1) Auxiliary diesel engine Sulzer 4 RLB 56 (MCR 1) 4400 kW at 163 rpm

2) Exclusive alternators and foundations

3) CSR = 0,85 MCR

4) CSR = 0,85 MCR of Alt. "A"

$$\text{minus } \frac{2 \times 2000}{0,95} = 4210 \text{ kW}$$

5) CSR = 4210 kW for one auxiliary group

6) Inclusive 5 % allowance for L.C.V.

7) 175 days per year at sea (50 % of service time) and 170 $ one ton fuel

8) at 10 % interest and 15 years operational life

is also employed for sea operations instead of using the two shaft alternators? ~~generators?~~

Table VI, Alt.B provides the answer. The time spent in port by this vessel, which must be considered a pure cruise ship, is considerable. According to the shipping company, the ratio anticipated between time in port and time at sea is roughly 50/50.

Alt.B, with two new-type auxiliary units instead of the two shaft alternators, ~~generators,~~ has an initial cost of $805,000, and is therefore considerably more favourable. This case shows that, as far as price is concerned, the auxiliary unit with only four cylinders and two simultaneously driven alternators is a fine conception.

The fuel costs of Alt.B are only slightly higher ($13,000) since there is also no great difference in specific consumption between main and auxiliary engines, and the time which this ship - a cruise ship in the purest sense - spends at sea is shorter than that of the "Scandinavia". The investment advantage is therefore not eradicated - and the NPV of savings made over the operational life of the ship amounts to as much as $706,000.

The shipowner, by introducing this new aspect (Alt. A), can be said to have made a good decision. In the end, we can only ask why he failed to take one further logical step to Alt. B.

But experience has shown that novelties in ship design are introduced with great care, by taking one small step at a time. This study throws light on only one aspect, the financial aspect, whereas sea operations on board are complex and dependent on several factors.

4. Critical Survey and Conclusions

After a subsequent reading through of the work, the authors have realized how limited and one-sided the investigations have been. Nevertheless, two important questions have been brought to light from the complexity of present-day systems:

a) indirect as opposed to direct ships' propulsion
   and
b) the production of electric energy using the main engines as opposed to the conventional way using auxiliary units.

Quantitively, the comparisons have been made only from a financial point of view, whilst other aspects have been mentioned only in a qualitative sense.

The study was limited to only three types of ship, and the ships used as examples were chosen more or less arbitrarily.

Nevertheless it is necessary at this point (not as an apology, but rather to reflect the reality) to highlight the difficulties by which one is

confronted when attempting to make objective comparisons.

Apart from the difficulties encountered when trying to procure price specifications, it should be said that these prices are all relative. In the first place, these are standard prices with no indication of the margins contained therein. These margins differ radically from manufacturer to manufacturer and depend on place, time, economic situation, etc. Each supplier has his own external selling strategy, but sometimes within the firm there are products, e.g. a new engine type, which the supplier wants to bring onto the market but undervalues or even overvalues in price. Similarly, the fluctuations in oil prices and the change in the standing of the dollar, which give the shipowner so many headaches, must also be mentioned.

All these factors contribute towards falsifying the results and making a "scientific" study extremely difficult. Furthermore, the results are only valid as absolute values for a short time.

So is this type of quantitative investigation to be recommended at all? The authors are convinced that the answer should be positive!

Naval architects and marine engineers have known about the qualitative relationships between the various parameters in ship design for a long time now. For example, the fact that a slow-speed propeller is more efficient is a "truism". Whether it can be applied and is worthwhile applying is a different matter. Decisions, however, are usually judged

quantitively according to the criterion of profitability. Fortunately, moreover, crises and competition ensure a flow of new products onto the market which have to be investigated.

The following priorities (not necessarily in order of importance) have emerged from this study:

a) the stage in development of diesel engines as influencing the choice of propulsion unit, and the advantages which they bring.

b) the necessity for an overall study of the propulsion system when discussing prices of components such as controllable-pitch propellers, shaft (alternators, ~~generators,~~ etc.

c) the difference in initial cost between four- and two-stroke diesel engines with the same power.

d) the significance of the twin-engined propulsion unit in relation to slow-steaming and partial loading.

e) the possibilities of indirect drive in relation to power branching for shaft generators.

f) the rational production of energy on board ships,

and last but not least (we hope)

g) the importance of such studies.

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WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug. 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

Advanced Marine Power Transmission

by

Professor Dr.-Ing. H. M. Hiersig

Ruhr-University Bochum

Shipping world has become accustomed to geared propulsion power transmission. Only the slow-speed crosshead engines develop speeds which directly match those of the propeller. Even in cases such as these, however, the tendency towards increasing propeller efficiency by lowering propeller speeds encourages consideration of combining slow-speed engines with gearing /1/. Where turbines are involved, be they steam or gas, gearing is essential since turbine speeds are far higher than those of the propeller /2/. Medium-speed engines required the development of ships' gearboxes with correspondingly lower speed reduction /3/.

The toothed wheel gear has unquestionably become established in preference to the belt-drive, hydrostatic gear and electric transmission in power transmission systems in ships. In spite of the disadvantage of fixed gear ratios, variable at best in stages, and several other undesirable characteristics such as, for example, noise emission, toothed wheel gears are always a natural first choice. The reasons for this are: high specific output, i. e. small dimensions, weights and moments of inertia, and unequalled efficiency. The gear losses are almost negligible, as also is the heat generation from toothed wheel gears.

The ship's propulsion itself poses a challenge to the engineering and technology of the toothed wheel gear: the strict requirements of reliability and availability for continuous operation, the power outputs to be transmitted and the output torques and, finally, the imperative need for quiet running, all necessitate expert knowledge on the part of the design and manufacturing engineers. Those few manufacturers who were capable of constructing this type of gear have become somewhat exclusive. Large investment is needed in particular in the area of mechanical and inspection equipment, as this is essential to meet the high engine speeds and considerable outputs produced /4/.

At the time when steam turbine propulsion was prevalent in larger units, the task of the turbine manufacturer was indeed a complicated one. He would combine turbines and gears into one construction unit and then, naturally, regard the gear section as being his responsibility. This gave rise to the two- or multi-stage toothed gearing, typical on turbine ships, some of which were fitted with two or more power paths (locked trains) /5/. Their wheels are made of quenched and tempered alloyed steel and incorporate teeth which have been hobbed in temperature-controlled shops on high-precision and often gigantic hobbing machines and then, after the flanks of the teeth have been shaved, checked over. This ensures the greatest possible tooth durability, as well as making allowance for bending and twisting occurring in the often slim pinion shafts and of other changes in position which may arise. The relatively small tooth load capacity, judged by today's standards of advanced technology, made the large dimensions essential. With diameters of the largest main wheels being in the range from 4 to 6 metres, having tooth widths often up to two times one metre, it is difficult to envisage the full extent of the system. To avoid any changing of overlap, thus optimizing quiet running, large helix angles should be adopted. If heavy axial thrust bearings are to be avoided, the resulting axial forces must be balanced out by means of double helical gearing. Thus, over several years, a type of gear was developed which combines a maximum amount of technical know-how. See Fig. 1 /6/.

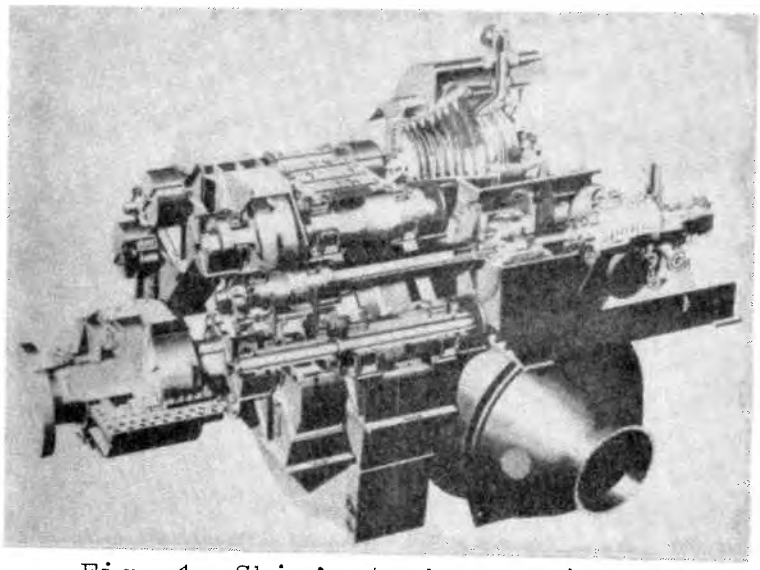

Fig. 1 Ship's turbo-machinery

More recently, however, two factors have contributed to the decrease in the number of turbo-gears of the original type being used in ship gear construction. The relatively high fuel consumption of oil-fired steam turbine systems (coal-fired ones are now being reconsidered) encourages a switchover to diesel engines as propulsion machinery. On fast ships, space for the machinery system is limited. This reason alone calls for the application of another technology for the toothed wheel gear, and to bring about a conversion to case-hardened wheels. Only in this way can a noticeable reduction in the dimensions of the gearing be achieved /7/. For this, there are numerous examples in defence technology /8/, involving the use of gas turbines, often combined with diesel engines, as propulsion machinery. In the course of time, this will also apply to steam turbine propulsion systems, where the steam is produced in nuclear reactors.

GTS "Finnjet", however, is one instance in civil shipping where the two turbo-gearboxes are fitted entirely with helical gears which are case-hardened and ground at the tooth flanks. This was necessary to keep the gearbox weight to a minimum, despite the output of 2 x 37 500 hp. It is well known that with high ship's speed, the propulsion power depends considerably on the weight and displacement. See Fig. 2.

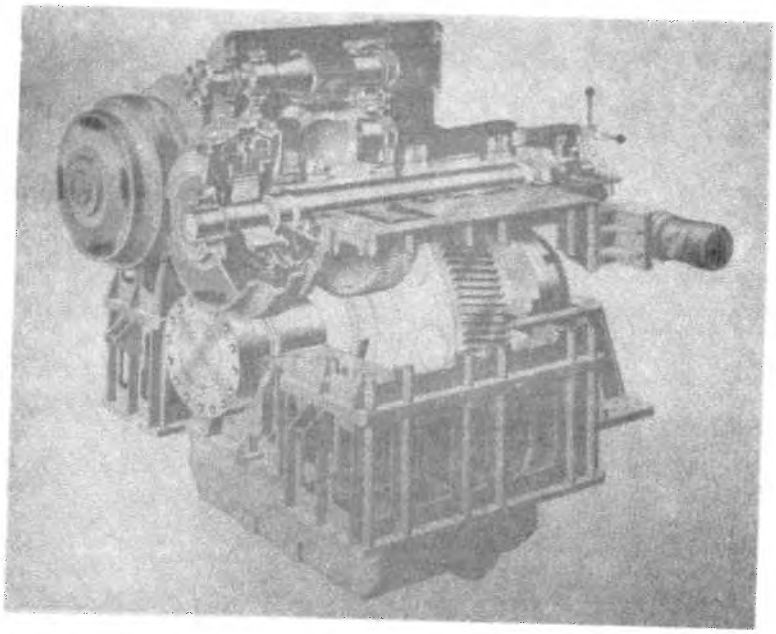

Fig. 2 Spatial representation of one of the two three-stage Navilus-turbo-gearboxes of GTS "Finnjet" equipped with friction clutches (P = 27 600 kW, i = 380/172)

The output required would have necessitated quenched and tempered wheels of around 4000 mm in diameter, whereas the modern wheels which transmit the same power measure only 2600 mm and have a small tooth width /9/. There have always been arguments against case-hardened toothed wheels which assert that case-hardening leads to severe brittleness and that the high specific tooth loads could eventually cause damage. Nevertheless, the good record of availability of this ship, engaged on its daily schedule, confirms irrefutably the reliability of this technology and the soundness of the construction concept of this type of gear.

In view of the fuel prices, the "Finnjet" will symbolize a special case by which other vessels, and in particular freighters, cannot easily be compared. On these latter ships, conversions are currently being carried out from gas turbine propulsion to medium-speed diesel engines, an indication that the diesel engine (medium or slow-speed) is favoured. Hence, gears for propulsion systems with diesel engines will be given preferential treatment in the following exposition.

1. Systematic classification

There is a bewildering variety of gears which transmit diesel engine outputs to propellers. A systematic summary will therefore help to clarify the general survey.

1.1 Single reduction gears

Fig. 3 shows schematic diagrams of the various single reduction gears. The basic forms (G 1.1 and 1.4) with cylindrical and conical wheels lead to variations involving intermediate wheels, which may be either without stages (G 1.1.1) or in stages (G 1.1.2 and 1.4.2). Designs with internal gears (G 1.2.1 and 1.2.2), seldom made nowadays, represent a transition to the planetary gears (G 1.3.1 and 1.3.2) which are becoming increasingly important as components of propeller drive, and elsewhere in power systems technology.

1.2 Distributing and summarizing gears and gears with multiple power paths

Fig. 4 provides a general survey of distributing and summarizing gears and such with multiple power paths. The distributing gears (G 2.1.1, 2.1.2 and 2.1.3) are of vital significance to vessels and in naval ships in which two controllable-pitch propellers are driven by a single engine. In this case, a simple machinery system is combined with high manoeuvrability. To avoid lateral drift, an intermediate wheel (G 2.1.2) must normally be provided to make the two propellers counter-rotate.

More common are summarizing gears (G 2.2.1), usually in the form of twin input-single output gears, although triple- and even quadruple-input gears do also occur.

The many merits of this configuration, where two engines drive a single screw, clearly account for the recent increase in its application: redundancy if an engine fails,

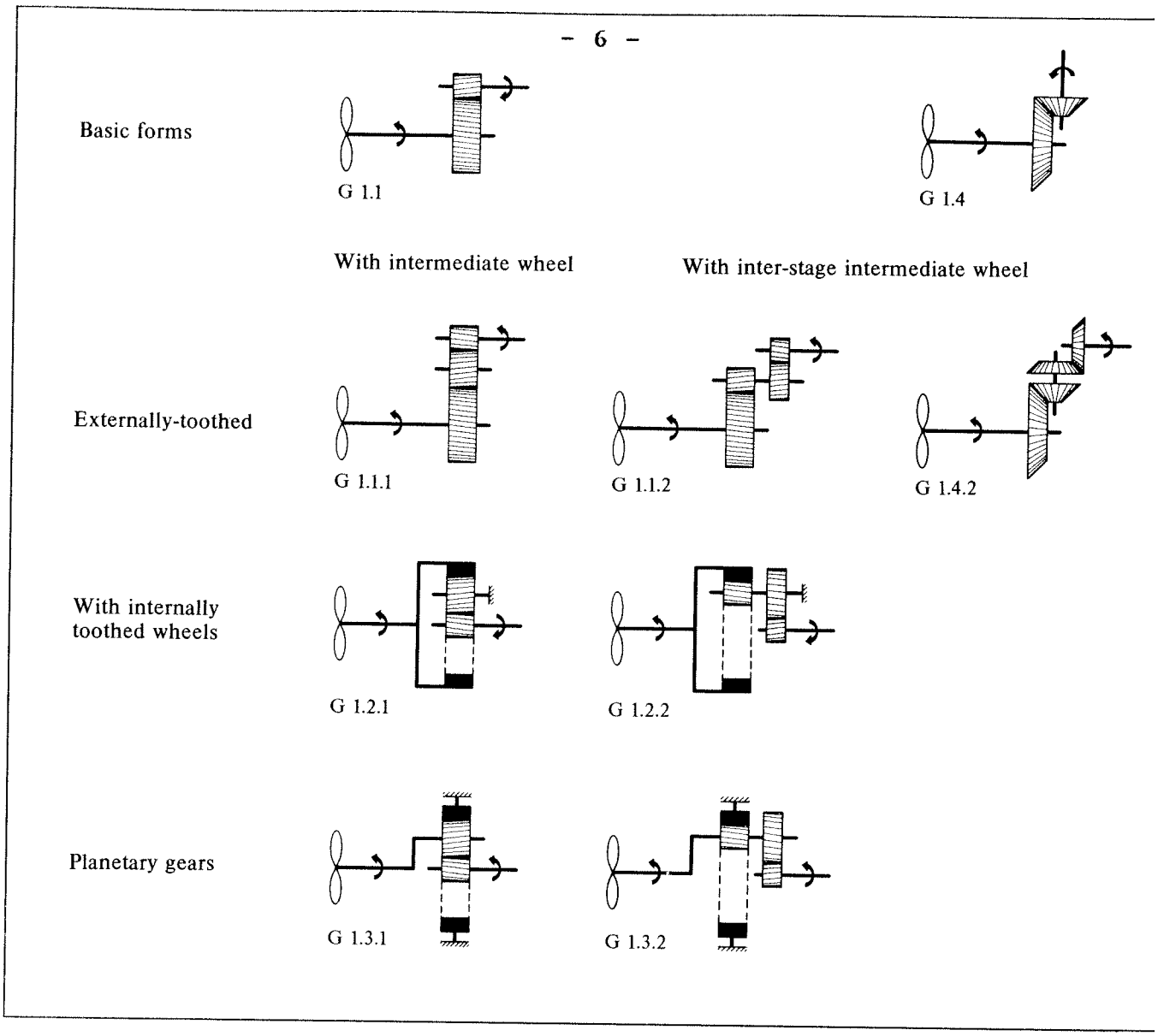

Fig. 3 Single path reduction gears

savings in fuel consumption when operating in confined waterways; quick reversing of fixed-pitch propeller by alternative engaging of the ahead or reverse running engine.

Even twin-screw ships are equipped with twin systems if this means keeping the vessel's deck low or the engine room short.

If the two engines in a twin-system require a substantial distance between shaft axes, e. g. in the case of vee-form or crosshead engines, the distance can be bridged using intermediate wheels (G 2.2.2) for small gear ratios and, for greater ratios, intermediate stages (G 2.2.3) as well.

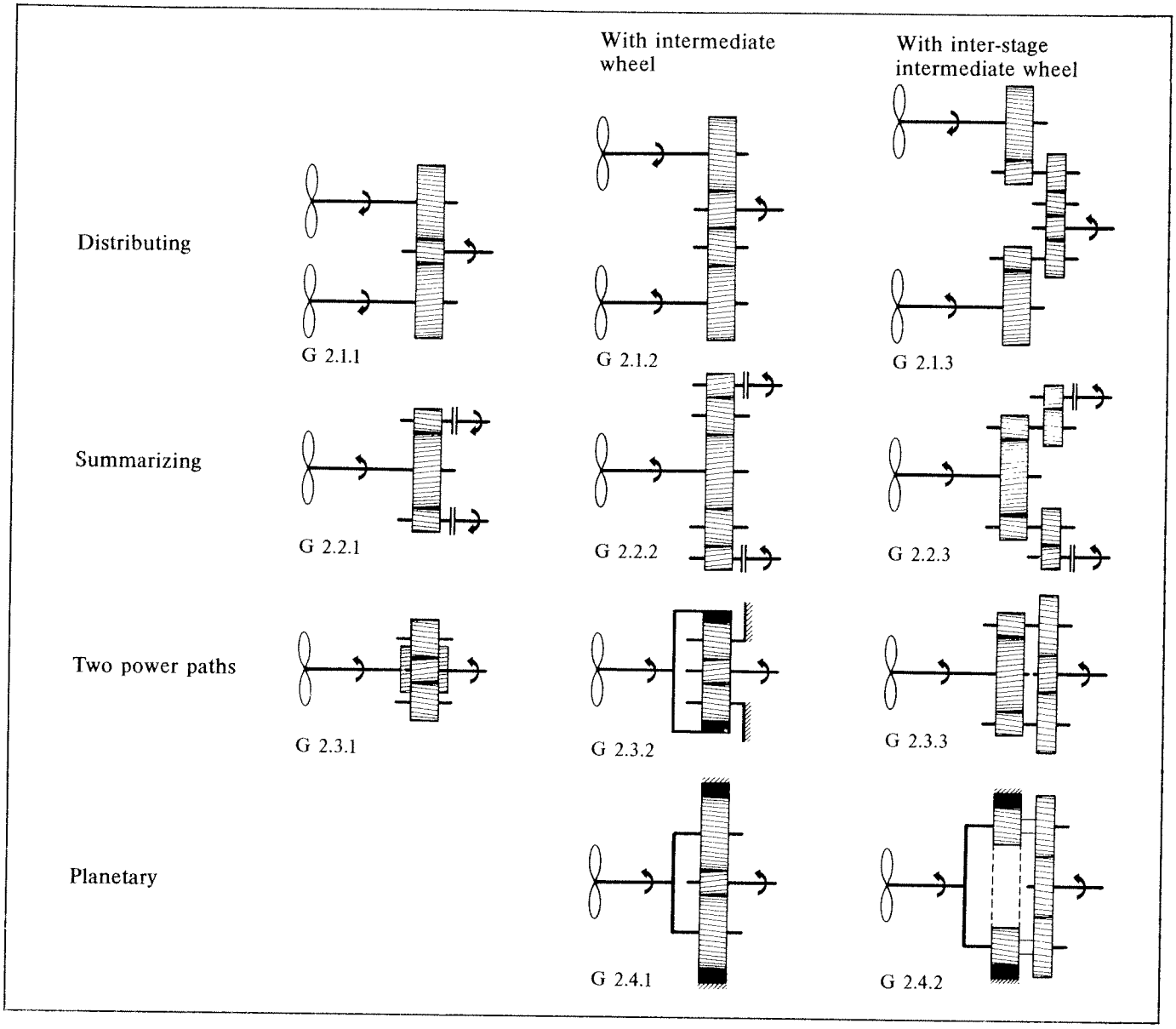

Fig. 4 Distributing and summarizing gears and gears with multiple power paths

As a rule, summarizing gears are provided with clutches which facilitate the engaging and disengaging of one of the engines.

Ships' gear dimensions can be considerably diminished by way of power branching. As shown in G 2.3.1, the output can be divided over two intermediate wheels which mate with a common large wheel. Should one wish to profit from the advantages offered by the coaxial type of design, G 2.3.2 with ring gear on the output shaft, and G 2.3.3 with an externally-toothed spur or helical gear, can be applied. In these latter two configurations, the output

is distributed across two, three or five branches before merging again at the output wheel. Where greater gear ratios are involved, epicyclic or planetary gears (G 2.4.1 and 2.4.2) are used, the planet carrier combining the multiple power paths.

1.3 Variable-speed and reverse gears

Several gears used in ships can be classed as variable-speed gears, see Fig. 5. Their clutches, symbolically represented by discs, enable engines with unequal powers and speeds to be engaged or disengaged. These types of "father and son" systems (G 3.1.1 and 3.1.2) are encountered both in fishing and other marine vessels. They allow the engines to drive the propellers at full engine power but at different speeds, thus providing two ranges of operation. When, however, neither a reversible engine nor a controllable-pitch propeller is fitted, reversal of the direction of rotation must be performed by the reverse reduction gear.

Reverse-reduction gears have either a reversing pair of wheels (G 3.3.1) or one reverse pinion, arranged as an intermediate wheel (G 3.3.2), unless they are designed as planetary gears with a Ravigneaux stage (G 3.3.3), where two planets are geared together in each case. This same method can be used to construct three-speed gears, which normally consists of two ahead stages and one astern stage (G 3.4.1 and 3.4.2). In principle this can also be applied to planetary gears (G 3.4.3), although these are seldom produced for such purposes.

1.4 Auxiliary gears

Finally, auxiliary gears for Power Take-Off (PTO) complete this tableau of possible ships' gear design. See Fig. 6.

The fact that it is an advantage to have electricity produced by the main engine, owing to its small consumption of fuel and of fuel of inferior quality, has meant that auxi-

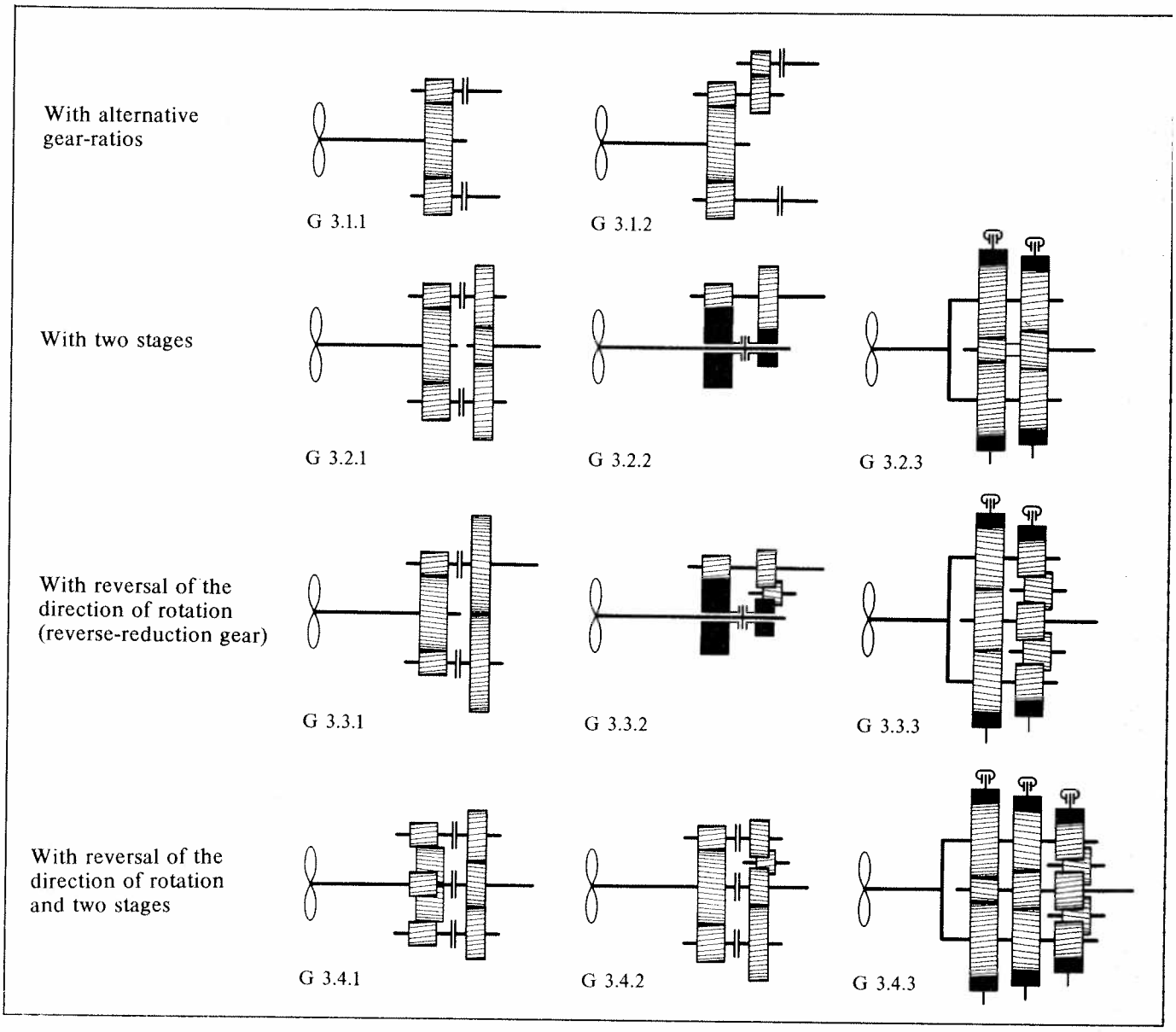

Fig. 5 Variable speed and reverse gears

liary gears for PTO have become increasingly important. Here, shaft generator gears (G 4.1) operate continuously with the propeller shaft (the so-called secondary PTO), whilst in the case of a primary PTO (G 4.2.1 and 4.2.2) the ouput shafts are driven by the main engine, even in cases where the propeller is idle, e. g. in harbour.

Generators should always operate at constant speed. For this reason there exist auxiliary gears which can be adjusted in steps (G 4.2.1.1), or steplessly, for instance hydrostatically, (G 4.2.1.2) so that when the main engine speed decreases, by stepping up its speed the nominal speed of the generator can again be maintained.

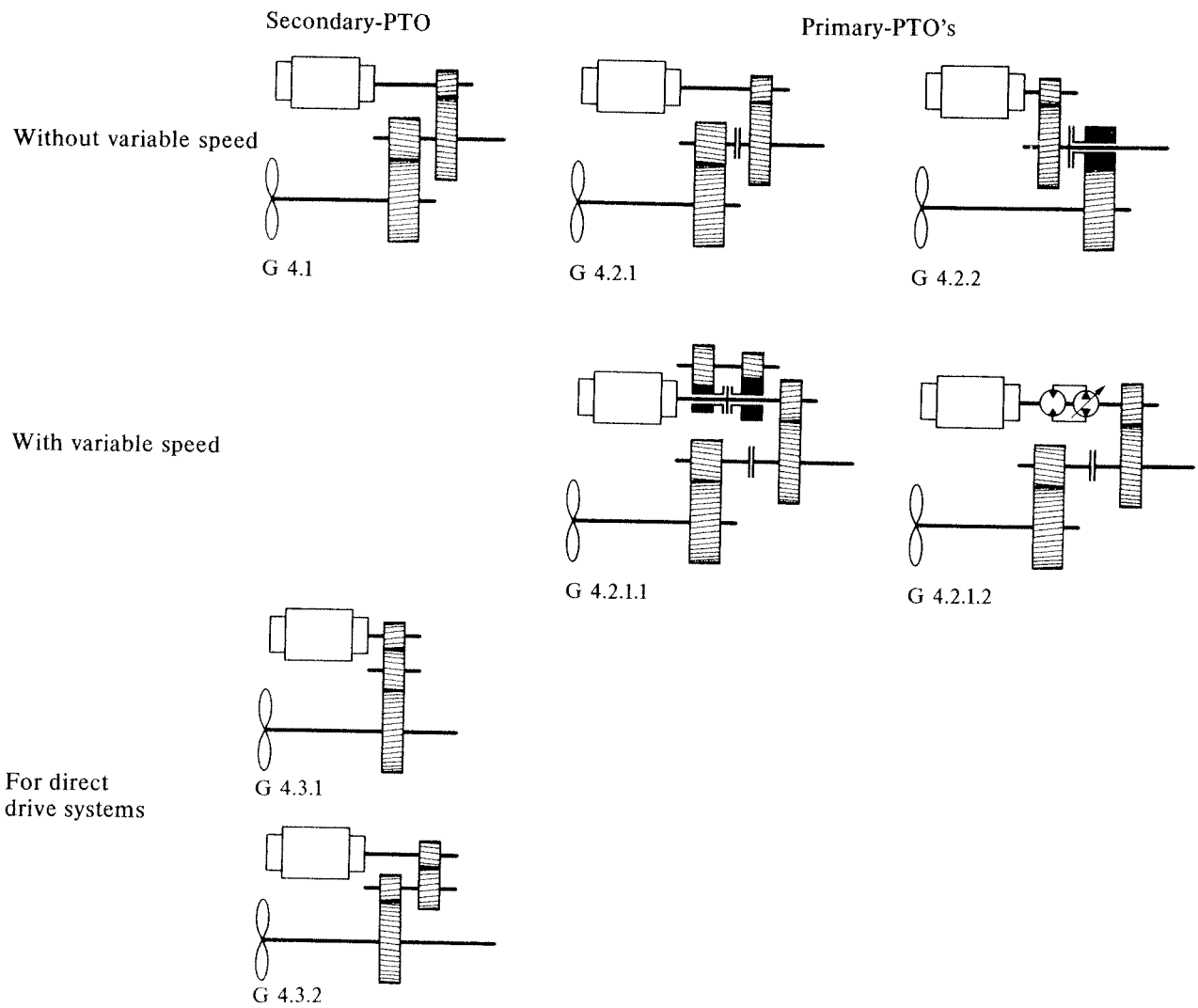

Fig. 6 Auxiliary gears

Gears designed for the true shaft generators (G 4.3.1 and 4.3.2), which "ride" on the propeller shaft in direct drive systems, conclude this systematic classification of gears used in diesel engine propulsion systems on ships.

2. Couplings and Clutches for Diesel Engine drives

In power transmission trains usually the components are connected by couplings. They are mostly designed to compensate transverse deviations deriving from mounting errors, displacements of shafts after the application of load or

even under thermal influences. Some of them, moreover, are torsionally elastic in order to lessen impacts in torque or to displace the natural frequency of the power transmission train or to separate one part of the system from the other one.

Couplings can be torsionally rigid and still however, compensate for shaft displacements. They can also be torsionally elastic to a greater or lesser degree through variation of the stiffness of the elastic elements, and they can in case of resonance act as damping devices by converting vibration energy into heat. Fluid couplings - as a rule - require exact mounting of shaft ends; they, however, maintain the torque even if the secondary part is blocked, what makes them suited for icegoing ships. Thus, in any discussions concerning ship propulsion, the couplings always play a significant part in the proceedings. This concerns two particular functions, both of which are performed by the couplings: firstly, controlling the natural frequency of the propulsion system so that no harmful torsional vibrations occur; and secondly, compensating the often inevitable shaft displacement so that both the engine bearings and the crank web deflections are unimpaired. This brings the following features into the foreground: torsional stiffness (defined by relationship between the torsional angle and a nominal torque); and radial, angular and axial stiffness.

Values for these are provided by the manufacturers in their specification catalogues. Practical specifications for the damping properties of the coupling, an important determinant of the torque increase in case of resonance, are also usually available.

Clutches (if they are not combined with elastic elements) bring a different set of problems to the fore. Among these are the heat capacity, the heat conduction and the temperatures which can be tolerated at the rubbing surfaces. Other aspects relevant to the functioning of clutches are

the mode of activation and the time lapse involved in making connection. Engaging too hard is detrimental to the mechanical components of the propulsion system, whilst too soft a clutch leads to considerable generation of heat.

2.1 Systematic classification

Unfortunately it is not possible, by strict reference to the constructional forms, to classify couplings as was done for ships' gears. The following classification criteria have therefore had to be based directly on design characteristics.

2.1.1 Couplings

Figures 7 to 9 show the main couplings. The tooth coupling (K 1.1.1) accommodates small shaft displacements, yet is torsionally-stiff but permits some backlash /14/. In contrast, the Vulaflex coupling (K 1.1.2) has no backlash and permits large shaft displacements as are required with elastic engine mountings. Very large shaft distances are bridged by the cardan shaft (K 1.1.3), rarely used, however, in shipbuilding, except on smaller units or vessels with vee-drive. Couplings with membranes have no backlash and are transversely elastic. Normally, these are designed (like tooth couplings) as double couplings (K 1.1.4 to K 1.1.6). Owing to its good inertia characteristics the latter is particularly suitable for high-speed propulsion machinery.

Bolt or dog couplings (K 1.2.1 and K 1.2.2) in Fig. 8, of which the Holset and KHD types (K 1.2.3 and K 1.2.4) have substantial elastomer elements, were developed especially for small gears which can be flanged on to the flywheel casing of the diesel engine. Diagram K 1.2.5 shows a coupling fitted with tangentially acting rubber torsion springs known as Metalastic. Diagram K 1.2.6 shows a coupling suitable for the biggest torques. The tension spring elements made of rubber filled with strong textile fibres (K 1.3.1) in Fig. 9 are used in GFL couplings to obtain the greatest possible torsional angle. An all-steel coupling manufac-

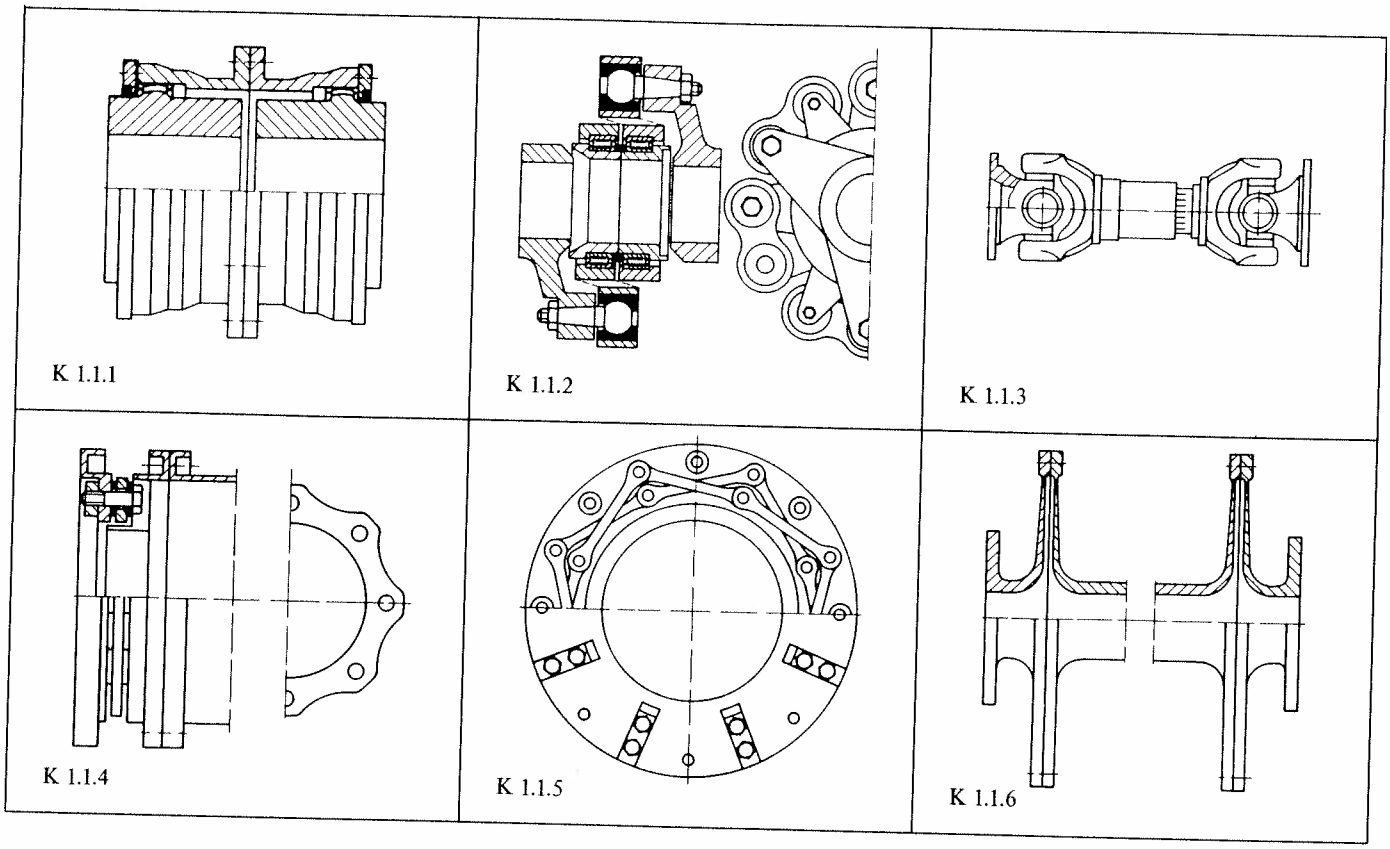

Fig. 7 Torsionally stiff, axially compensating couplings

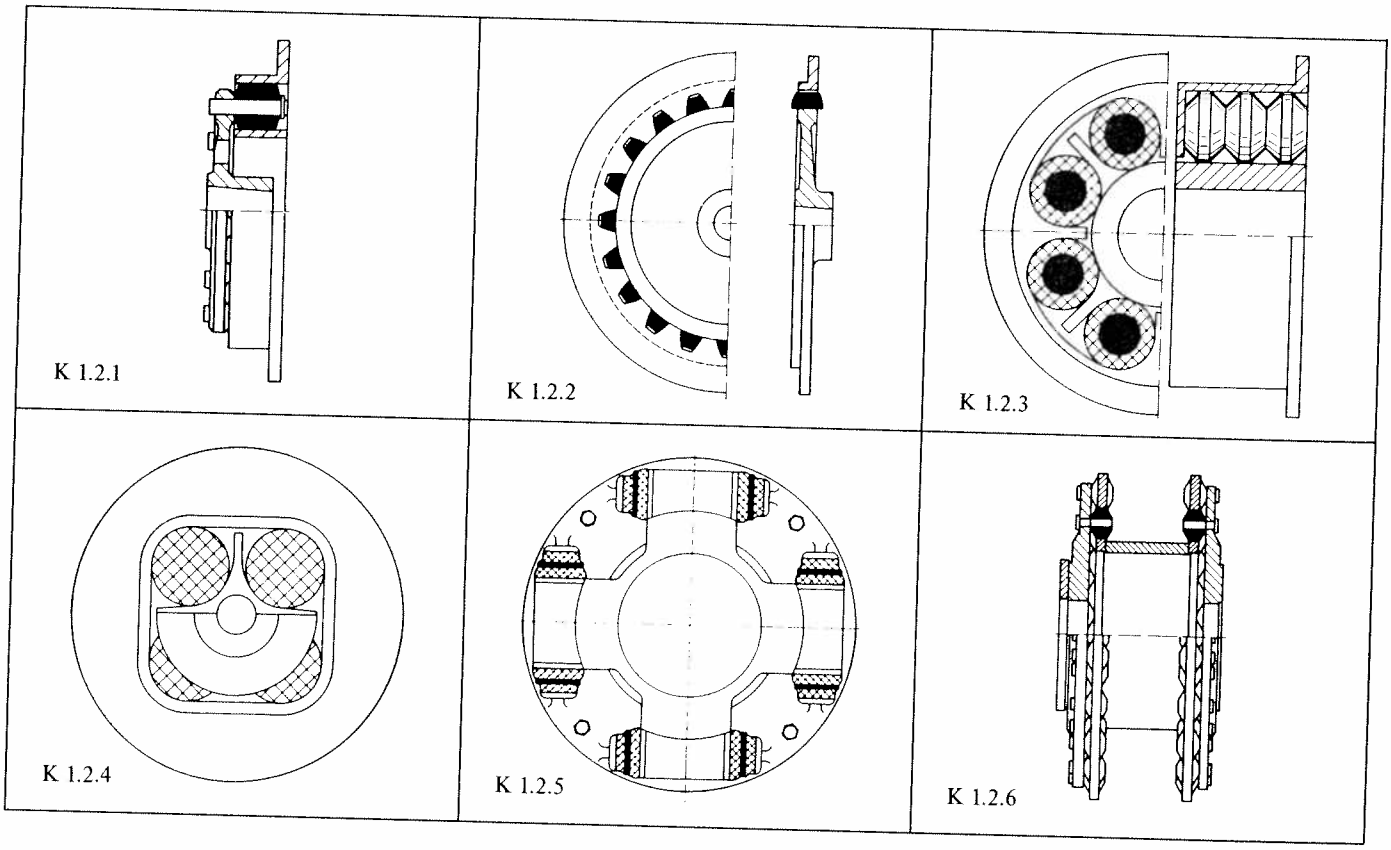

Fig. 8 Resilient couplings

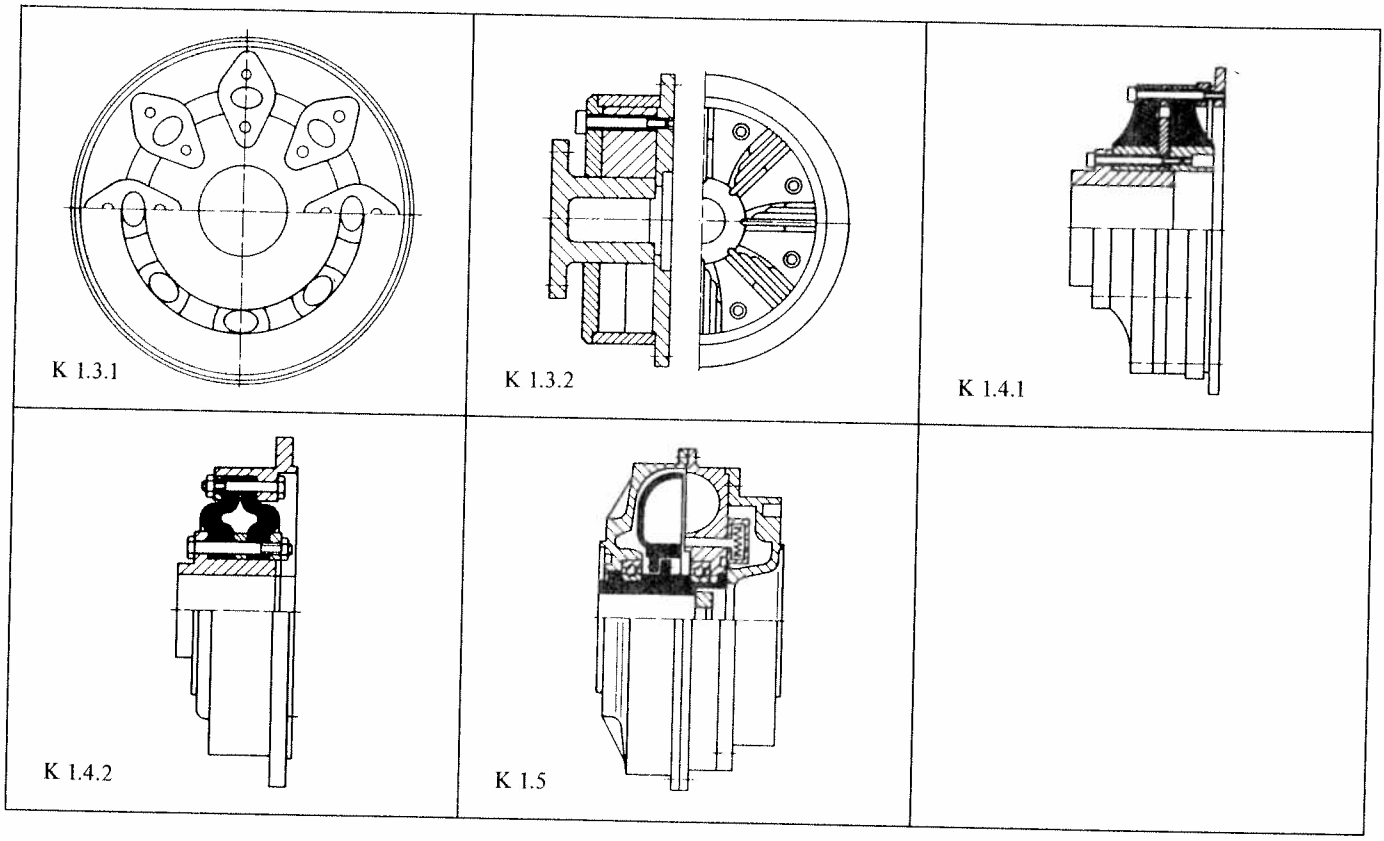

Fig. 9 Torsionally-elastic and slip couplings

tured by Geislinger (K 1.3.2) transfers the torque through steel spring elements which also act as pistons to displace oil from one chamber to the other. In addition to achieving a high torsional elasticity, this method also ensures good damping.

In the couplings K 1.4.1 and K 1.4.2 in Fig. 9, the torque is transferred entirely by rubber elements. The former coupling, the Spiroflex /15/, makes use mainly of natural rubber, whilst the latter, the Vulkan-EZ, employs textile-filled rubber membranes. Both produce large torsional angles of the order of 5° to 15° at the nominal torque, this having a great influence on the torsional stiffness of a torsionally-vibrating propulsion system, thus causing significant alteration in the natural frequency. Finally, K 1.5 in shows a slipping hydrodynamic coupling ideally suitable for avoiding torsional vibrations: this coupling does, however, cause a loss in power /16/.

On the other side it proves advantageous, as it keeps the propeller torque at lowest speeds, even at standstill.

2.1.2 Clutches

The shaft connections hitherto described are referred to as "couplings". The term "clutches", however (as will be shown in this section), refers to disengageable connections. Clutches, Figs. 10 and 11, can have either solid contact or frictional operation. The former clutches (K 2.1.1 and 2.1.2 Separex) are normally only to engage when stationary or, like the SSS clutch, as overrunning clutches (K 2.1.3) in the synchronous running condition.

In this respect, the friction clutches, built as integrated multi-plate clutches or as clutches with cylindrical or conical rubbing surfaces, are different. The oil-lubricated integrated multi-plate clutches need relatively small force to be engaged mechanically (K 2.2.1) or are controlled hydraulically (K 2.2.2) or electro-magnetically (K 2.2.3).

Dry-running clutches built onto the gearbox exterior are usually controlled pneumatically. Here, clutch discs can be provided if the pressurizing force acts in an axial direction as in Fig. 11 (K 2.2.4 Wichita); or a friction cylinder installed as K 2.3.1 (Airflex), if the pressurizing force acts radially inwards. Conical surfaces require acial pressurizing forces (K 2.4.1 Pneumastar or K 2.4.2 Pneumaflex) /17/.

Finally, in order to become a clutch, the hydrodynamic coupling, represented in K 2.5, can be connected to an emptying and filling system.

3. Gears and gear units

Ships' gears and, more specifically, gears used in diesel engines functioning as main engines, have much in common with other toothed wheel gears, but also differ from these

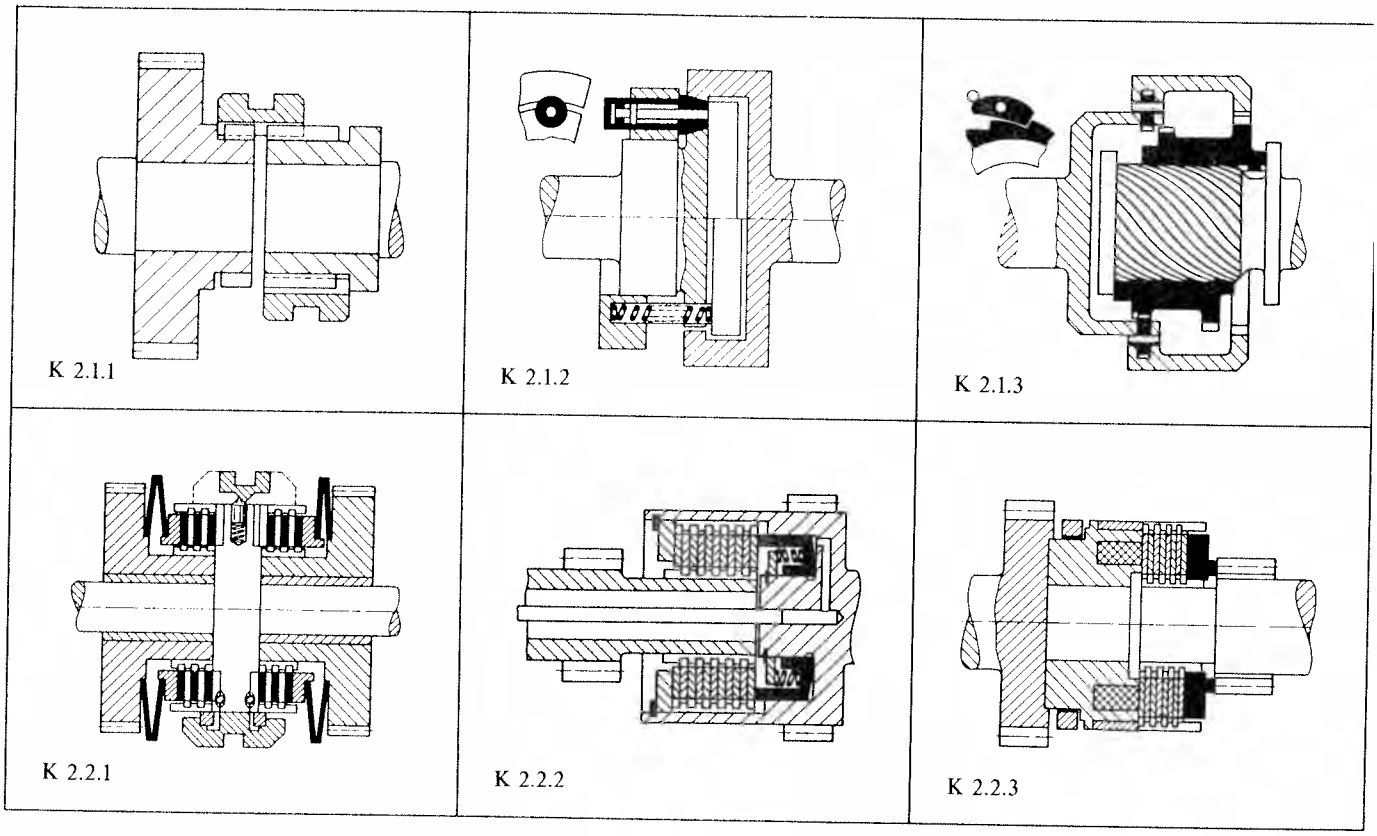

Fig. 10 Solid contact and oil flooded clutches

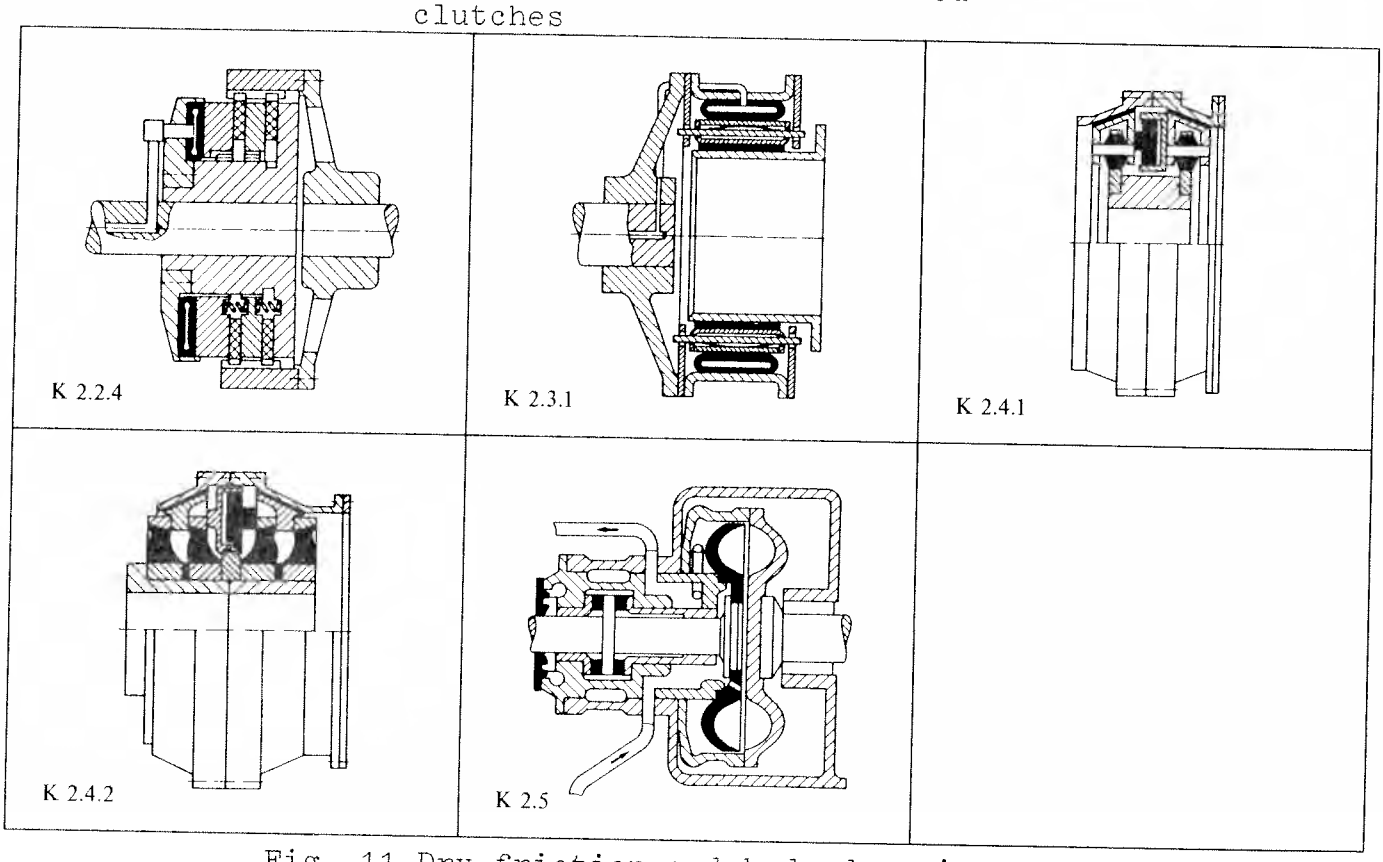

Fig. 11 Dry friction and hydrodynamic (fluid) clutches

by virtue of their particular application. As the system of classification has already shown, there are certain construction types which do not occur other than in this form /10/. However, even the simple cylindrical gears have, at the very least, a special casing which usually also accommodates the thrust bearing whereby the propeller thrust is transferred into the ship's hull. The design, which generally involves high concentration of space, calls for forced circulation lubrication and an external cooling system in conditions of continuous operation. With the engine room frequently unmanned, there is good reason to equip the ship's gears with a series of sensors and gauges for transmitting signals. Many of these features are self-evident; some, however, are the subject of much controversy. For this reason, certain important aspects will be discussed more fully in the following sections.

3.1 The gear wheels

Toothed wheels are the principal components of gears. Their dimensions are the prime determinants of the construction size and its configuration. Hence the design (or size dimension) of the gear wheels is a decisive criterion. It is dependent upon the transmitted power, the gear meshing, its profile and precision, and upon the material and its treatment.

Whilst ship's gears which are driven by machinery (e. g. turbines) with a high degree of uniformity, come under the category headed "smooth operation", gears which are coupled to diesel engines are subjected, as part of a system capable of torsional vibrations, to torque fluctuations which can involve considerable variations form the nominal torque. With its relatively low degree of uniformity, the diesel engine is a source of vibrations of various amplitudes and modes /11/. The torsional vibration situation can be calculated quite accurately, and there are certain measures

which may be taken to influence it. Hence, the occurrence in the gear area of resonances containing high amounts of energy should always be avoided. This sometimes proves problematic, however, since with irregular running of the engine, e. g. in the event of a cylinder misfiring, excitations of orders other than those occurring during normal running become predominant, often causing resonances and torque increases, even "hammering", i. e. the emergence of negative torques. Ships' gears must be able to endure such working conditions, even if only for a limited period of time.

When calculating the dimensions, this is taken into account by way of a dynamic magnification factor, which represents the normal torque increases relative to the nominal torque. The classification societies specify values for this, just as they also give procedures for the dimensioning of gear teeth according to engine ouput, speed and transmission.

Approved ships' gears are characterized by the optimizing improvements made to the gear meshing, by addendum modification, tip relief and helix angle correction. Of great significance, however, is the choice between single and double helical gearing. This is directly connected with the absorption of the axial tooth forces, which result from the division of the normal force into tangential and axial components. With high speeds, it is the axial thrust bearing which results in relatively large losses. These are the reasons, therefore, why with high-speed turbo-gear pinions the axial forces on the helical gearing are balanced out by opposing the helix angles. This does not apply to the same extent to the slow speed pinions in diesel engine gears.

In this case, the free axial forces can be carried by axial bearings or, in the case of small gears, simply by roller bearings. This arrangement enables the single helical gear, which is less difficult to produce, to be used with helix angles which facilitate the use of even overlap ratios. In this way, the total length of the con-

tact lines in each tooth position is constant and the tooth stiffness variation, which is an important factor in noise emission, remains practically constant during one passage of the tooth. See Fig. 12 /12/.

Gears with roller bearings are usually provided with an overlap ratio of 1, so as not to lessen the durability of the roller bearings through excessive axial forces. Gears with plain bearings, generally governed by strict requirements for quiet operation, are capable of accommodating large axial forces and can thus also accept greater helix angles of up to an overlap ratio of 2. Single helical gearing in ships' gears has the advantage that, to a certain extent, unavoidable axial propeller shaft vibrations cause less additional stressing of the gear wheels than is the case with double helical gearing.

With double helical gearing the pinions are not axially restrained. They must therefore follow each axial movement of the main wheel. Their mass and the parts of the coupling attached to the pinions, however, produce considerable forces opposing the accelerations and decelerations. In a gear with single helical gearing, the axial vibrations of the large wheel give rise only to additional rotational motions, which are counteracted by much smaller secondary inertia forces.

The material from which the toothed wheels are made has an important relation to the dimensions. The introduction of case-hardening produced a remarkable increase in the strength of tooth flanks compared with a heat-treated gear wheel made from high alloy refined steel. Here, the critical load limit is governed by the strength of the tooth flanks, whereas with case-hardened and ground teeth, damage, e. g. through pitting of the flanks, seldom occurs. The load capacity limit then depends on the strength at the root fo the tooth or in the tooth itself. This is why toothed gear wheels with case-hardened teeth normally have fewer but larger teeth.

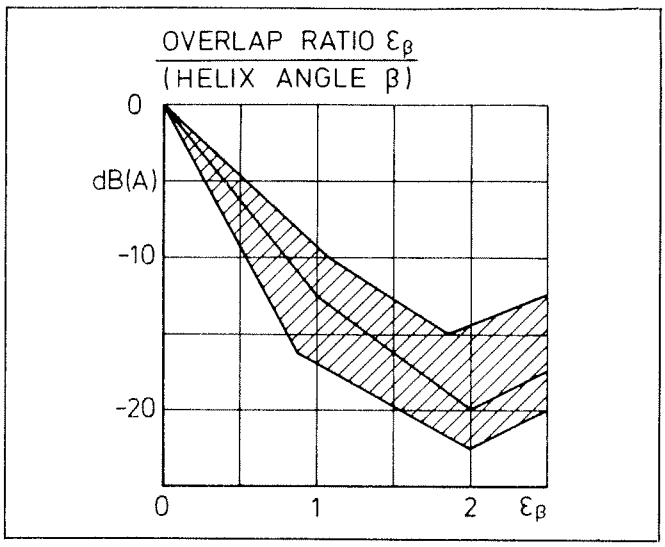

Fig. 12 Influence of overlap ratio on noise reduction

Gear wheels can up to a certain size, be produced from forgings. However, if the diameter exceeds approximately 1500 mm it is more economical to make the gear rim and the wheel body separately and then join them together /13/.

A method involving bolting on the wheel rims has provided a good solution to this. See Fig. 13.

Since, however, distortions may arise when heat-treating the gear rims, it is preferable to design the large wheel for welded construction. In such cases, the wheel body is constructed with a hub and steel side plates with reinforcements, and welded onto the gear rim. See Fig. 14. The method employed here originates from the principle used for the large wheels in turbine gears, and is indeed, most successfully applied with large wheels incorporating case-hardened teeth. Owing to the rigid connection of hub and gear rim only slight distortions occur, so that grinding of the tooth flanks requires only limited cutting, this being beneficial to the uniformity of the hardened case on the tooth flanks.

3.2 Shaft supports

Gear wheel shafts, particularly in small ships' gears, are usually mounted in roller bearings. Roller bearings have the advantage of simplicity of installation and also

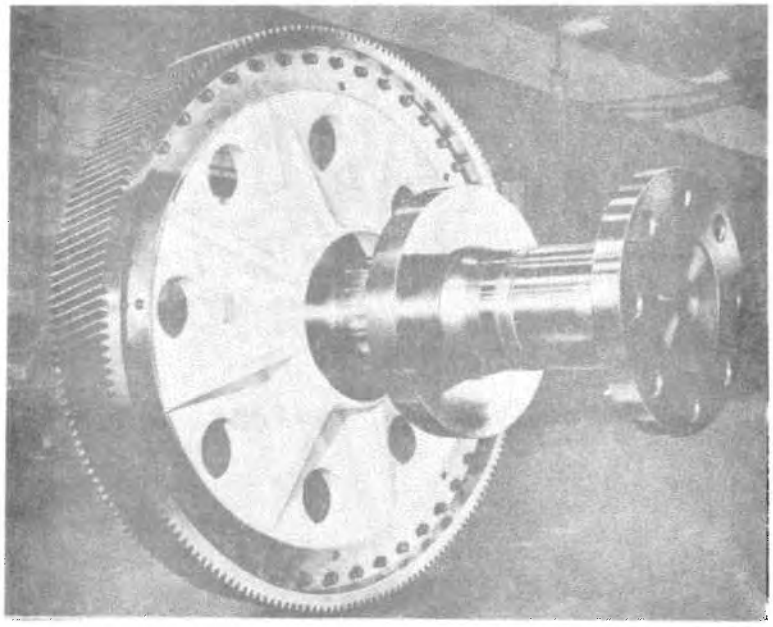

Fig. 13 Main gear wheel with bolted gear rim

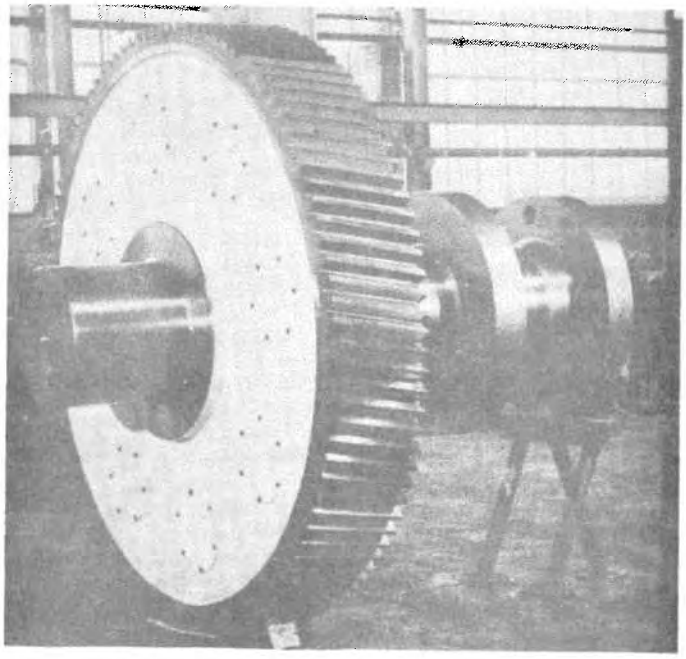

Fig. 14 Main gear wheel of fabricated construction

enable both radial and axial forces to be absorbed in one bearing. See Fig. 15. In large gears, however, plain bearings are preferred. Unlike large roller bearings, these are easier to replace, and neither produce nor ease the transmission of sound. The same applies to the thrust bearings which absorb the propeller thrust. Here, large thrust forces are taken by the axial bearing with tilting thrust pads. See Fig. 16.

3.3 Casings

Whether the casing is made of grey cast-iron or steel depends largely upon amortizing the pattern costs and hence upon the number of casings to be produced. For a corresponding design, the two materials can be regarded as being of equal value. Hybrid constructions, where grey cast-iron is used for the complex shape of the middle section, whilst steel is employed for the sump and the cover, have also proved successful.

The way in which the gearbox seating is connected to the ship's hull is, however, of great importance. Where large areas are in contact, unavoidable hull distortions may impair the relative position of the shafts to each other. For this reason, it is beneficial to keep the bottom surface area of the gearbox small. A strengthened substructure, which transfers a minimum of distortion onto the gearbox, should be provided at its seating.

Associated with this is the problem of where to position the thrust bearing: integrated in the gearbox casing or arranged separately. The question essentially becomes one of how the thrust bearing can be given a high axial stiffness. For if a very large main wheel necessitates a deep incision in the double bottom, great difficulty will be encountered in giving the integrated thrust bearing high axial rigidity. In this case (as is normal in turbine gears) the thrust bearing is mounted separately. If, however, the dimensions of the large wheels can be reduced to the ex-

tent that the gearbox causes no interruption or weakening of the longitudinal hull structure, the thrust bearing can be connected directly with the gearbox without causing any adverse effects. The advantages of this style of solution are in simpler construction and ease of installation.

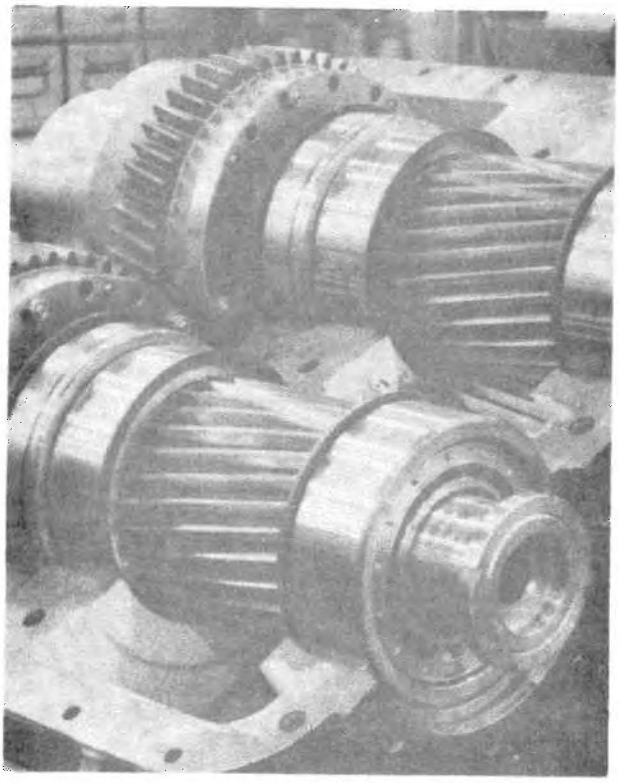

Fig. 15 Gear shafts mounted in roller bearings

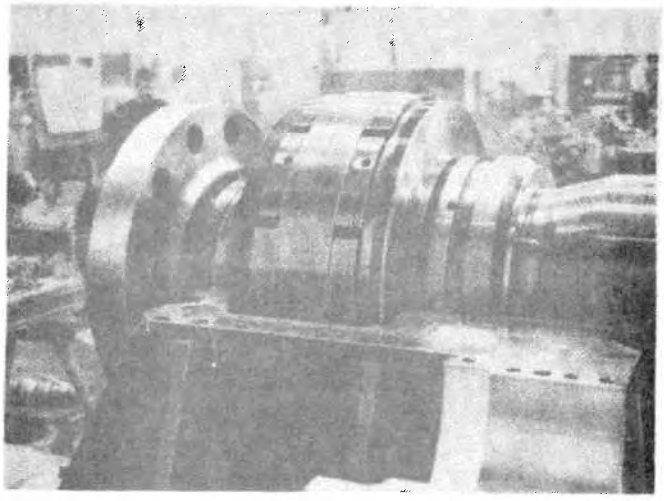

Fig. 16 Output shaft mounted in plain bearing and thrust bearing with tilting pads

Even if a favourable gear tooth configuration results in minimising noise, the material and shape of the casing are still significant factors as regards sound emission. Preference is given to casings made of grey cast-iron, with its good material damping, in order to limit noise. The shape of the casing should be such that frequencies containing high amounts of energy emanating from the toothing do not incite casing walls to resonance. If it is not possible to make use of curved surfaces, irregular thickening of the inner walls of the casing is an alternative and very adequate solution. With regard to noise emission, in order to match the standard of the grey cast-iron casings, welded cases require relatively thick plating. Should the weight make this inpracticable, the casings are designed with double walls, for which stiffeners provide reinforcement between the inner and the (possibly perforated) outer wall. See Fig. 17.

It is particularly in ships that special attention should be given to the reduction of vibrations. If a stiff-elastic mounting cannot be considered, it is worthwhile ensuring that there is sufficient elasticity between the bearing supports and mounting surfaces of the casing itself. Heavy masses on the bearing structures and elastic connections between these masses and the substructure help to reduce structureborne vibrations.

3.4 Standardized gear units

The gear industry offers gear units, along with complete documentation, for several applications. Rational and economical manufacturing, as well as specifications concerning the permissible power at a given engine speed, reliably tested and approved by the classification societies, give the user the advantage of being able to pass over the many time-consuming considerations which are essential to a full understanding of the performance of special gears. In addition, short delivery times and more favourable prices can be expected.

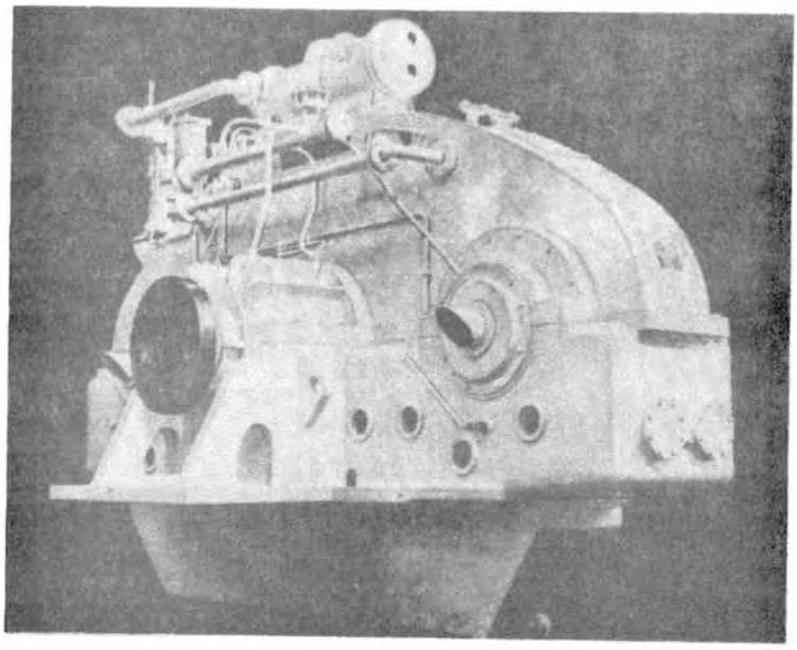

Fig. 17 Gear case with double walled frame

Using the systematic classification of ships' diesel engine gears presented in Figs. 3 to 5, the most common standard gears can likewise be listed in order.

The table of Figure 18 and the diagrams in Figures 19 to 21 give an initial overview of the standard ships' gears available, without making any pretensions to completeness. The field is relatively well covered, so that several different kinds of application can be catered for.

Above all, it is to accommodate the larger outputs that there is a wide variety of gears of special design configuration. These will be discussed separately.

3.5 High-power transmission

The maximum rating of a single stage offset gear unit - type G 1.1, Fig. 19.3 and 19.6 - amounts to 31,6 kW/rpm at the ratio 5:1. With an engine speed of 400 rpm the output of 12 640 kW resp. 17 190 hp can be transmitted by this big unit. Its output torque results to 1,5 MNm. The case-hardened and flankground bull gear of this unit reaches already a pitch circle diameter of 2500 mm. Another gear unit - type G 2.3.3, Fig. 20.2 - is rated with 54 kW/

Type	Centre Distance mm	Power Range (kW/rpm) i = 2 : 1	i = 4 : 1	i = 4,5 : 1	i = 5 : 1	i = 6 : 1	Fig. No.	Diagram No.
G 1.1 vertical with								
roller bearings	450 ... 650	2.6 ... 4.7	1 ... 2.15					1
plain bearings	600 ... 1000	6.3 ... 31.0		2.35 ... 11				2
plain bearings	1050 ... 1500	31.0 ... 95.4			12.5 ... 31.6		19	3
horizontal with								
roller bearings	450 ... 650	2.6 ... 4.7	1 ... 2.15					4
plain bearings	600 ... 1000	6.3 ... 31.0		2.35 ... 11				5
plain bearings	1050 ... 1500	31.0 ... 95.4			12.5 ... 31.6			6
G 2.2.1 medium weight	1000 ... 2050*)	3.1 ... 26.0				1.25 ... 3.1		1
heavy weight	1000 ... 2050*)	4.0 ... 34.0				1.40 ... 11.0		1
G 2.2.3	2150 ... 2250*)	36 ... 84				14 ... 39	20	2
G 2.3.3	900 ... 1150	23 ... 114			12 ... 54			.3
G 2.4.1			21 ... 98			6.8 ... 32		4
G 3.3.1 light weight	160 ... 280	0.07 ... 0.23	0.07 ... 0.23					1
medium weight	250 ... 500	0.18 ... 1.03			0.18 ... 1.03		21	2
heavy weight	300 ... 660	0.90 ... 7.36				0.27 ... 3.08		3

*) Centre distance between the engines is in each case twice as high
**) Relation hp/kW = 1.36

Fig. 18 Range of standardized ships gears for diesel engines

rpm. Herewith the output increases to 21 600 kW resp. 29 376 hp and to the torque of 2,6 MNm, despite the bull gear has even less diameter than that of the above mentioned unit. The difference is caused by the application of the "locked train" design.

In the strict sense of the word dividing and summarizing of power signifies the locked train. Typical examples for power dividing and summarizing gears are the epicycling or planetary gears, Fig. 22. Their sun gear divides the power (or the torque) to the intermediate (or planet) wheels. They convey their tangential forces to the annulus (or ring gear). The output provides the planet carrier

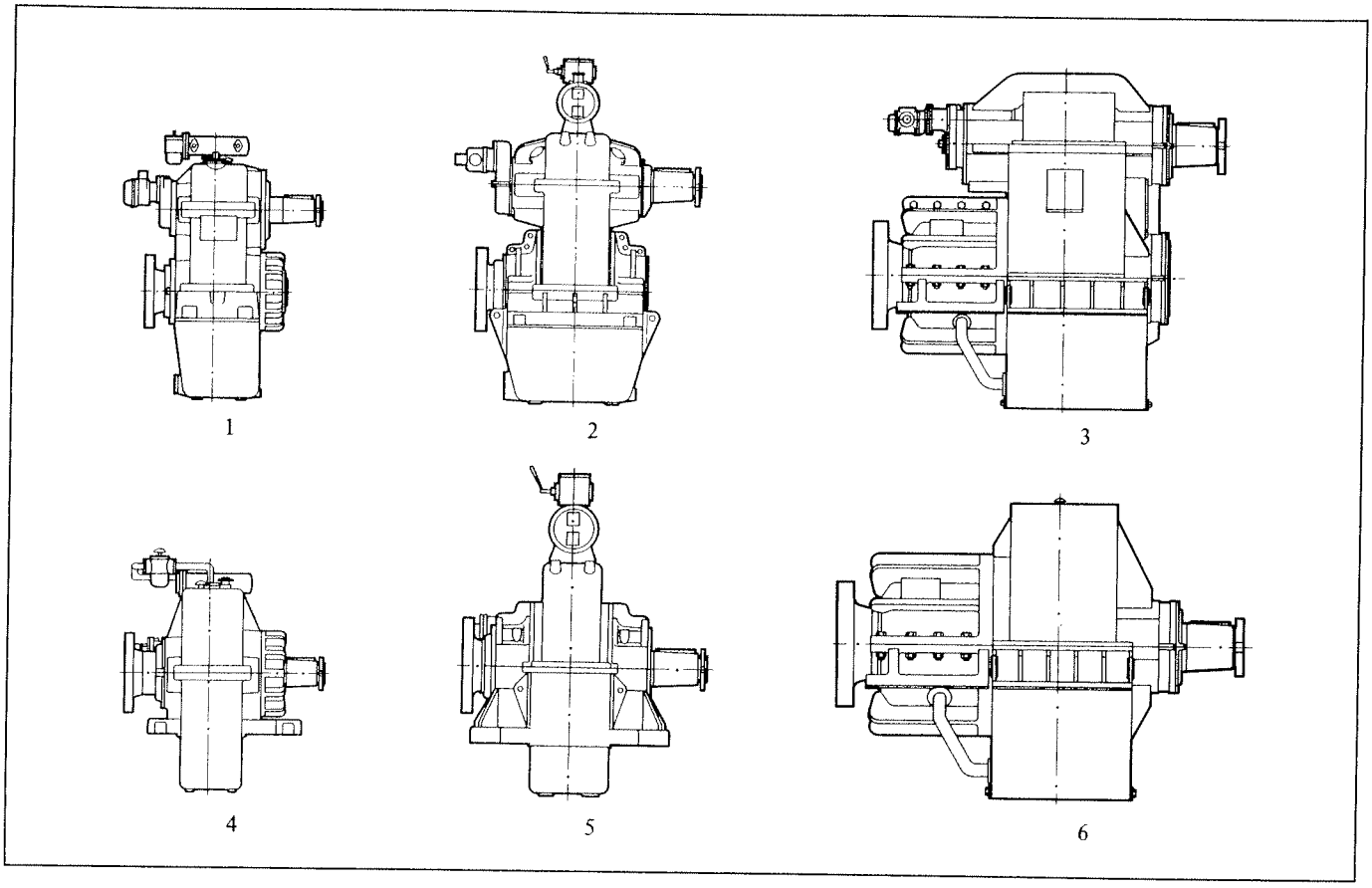

Fig. 19 Single stage gearboxes with shaft displacement (G 1.1)

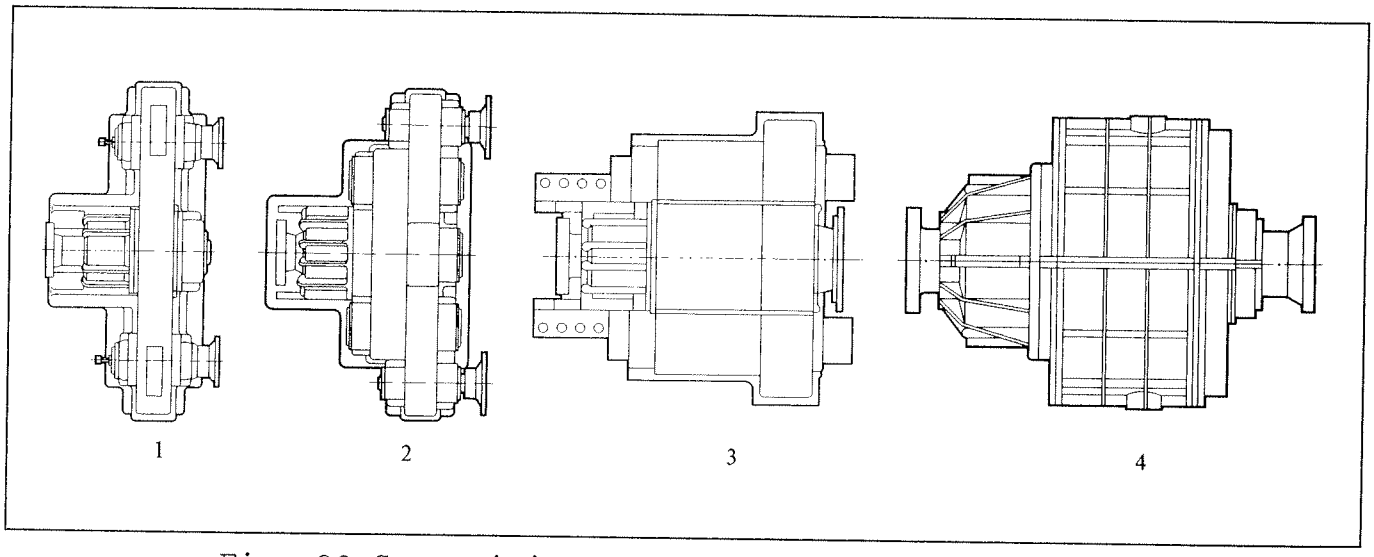

Fig. 20 Summarizing, two paths and planetary gearboxes (G 2.2.1, G 2.2.3, G 2.3.3, G 2.4.1)

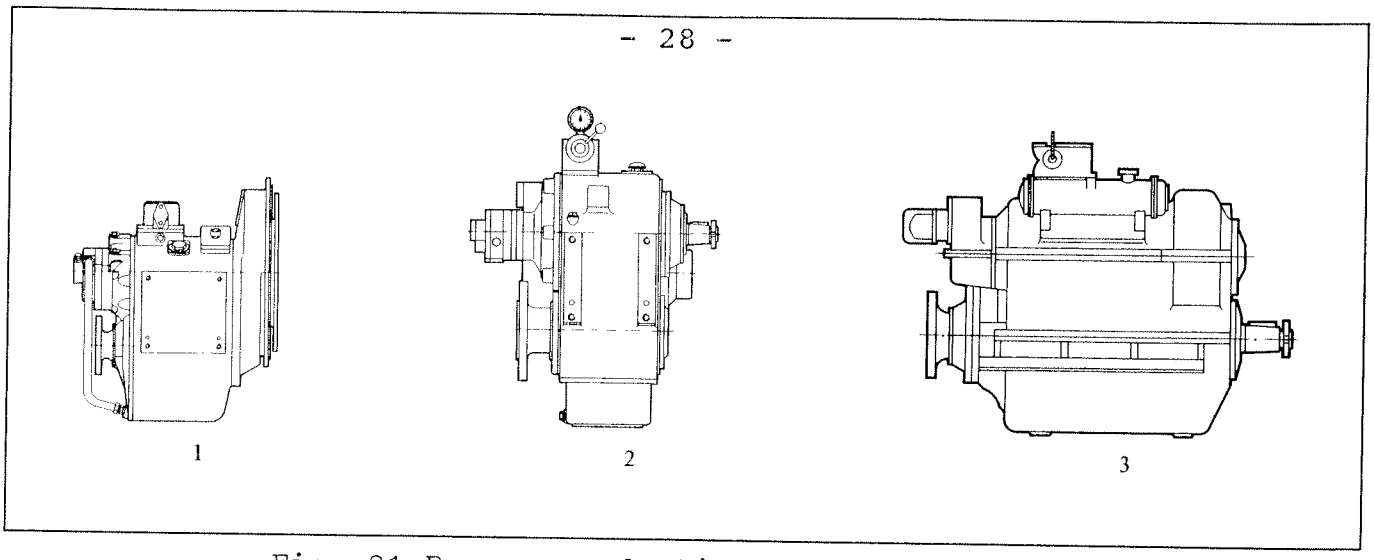

Fig. 21 Reverse-reduction gears
(G 3.3.1)

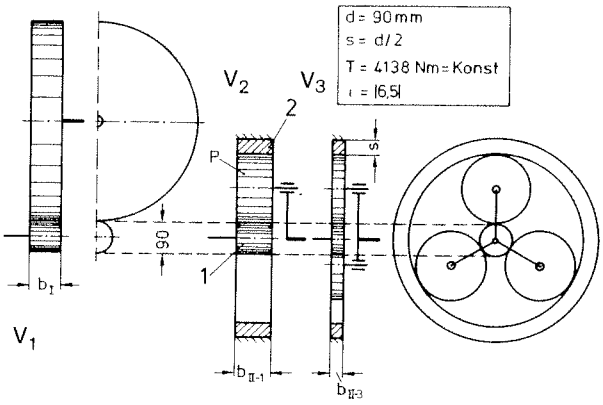

Fig. 22 Relation of gear volumina between offset and epicycling gears ($i = 6,5$; $T_1 = 4138$ Nm; $N = 1500$/min; $P = 650$ kW 16 Mn Cr 5 case-hardened; $V_1/V_2/V_3 = 1/0,48/0,25$)

(or cage), which get two times the tangential forces to build up its torque. Though the teeth of the intermediate wheels have to withstand forces of changing direction, though the bearings of that wheels get twice the tangential forces to bear, finally result small dimensions because the advantage of the distribution of power is so prevalent. In Fig. 22 the relation of the gear's volumina gives an impression of the favourable design, which applies the locked train configuration.

The expression "locked train" has been coined in connection with large turbine gearing, particularly in the field of ship propulsion. In order to diminish the diameter of the big wheel to feasible sizes, the input power had to be divided in two trains: the input pinion transmits the power to two wheels and to the connected pinions which drive the summarizing bull gear, Fig. 23. The necessary equilibration of torque at these turbo gears regularly has been achieved by torsionally elastic shafts, as the are described at "c" in Fig. 24, which shows different equilibration methods. A spatial view of a realised gear unit with power division to two paths is given in Fig. 25.

The combination of gears an the intermediate shafts requires exact phase angle positions of the two wheels between all the shafts of one locked train. This can be achieved by adjusting of the wheels, either with the application of oilpressure joints (as in the unit of Fig. 25) or with friction clutches (as in the unit of Fig. 2). In Fig. 26 /18/ the two methods are compared. As in ship propulsion plants the torque is always small if the propeller turns with low speed both the clutches receive small torque when they are engaged. Whenever in this engaging moment exists a slight difference in the transmitted torque, this difference will be diminished when with increasing speed the torque reaches high values. Thus, the equilibration between the two power paths remains within the permissible tolerances. In respect of clutches-rigid ones or highly torsionally elastic ones-there seems no practical restriction in transmissible torque: the biggest standardized highly elastic clutches are rated to 1,4 MNm.

In a wider sense the term "locked train" also can be applied to all closed circuit systems with two or several power paths. To a certain extent even two or more combustion engines can fall under the same headline. Particularly heavy installations with high power output use multiple prime movers. In such plants the bull gear only summarizes the

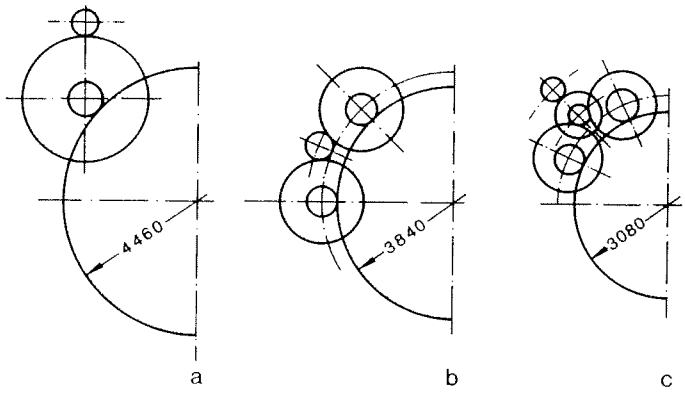

Fig. 23 Tandem and dual tandem configurations of turbine gears

a without power distribution
b with two power paths
c with two power paths and three stages

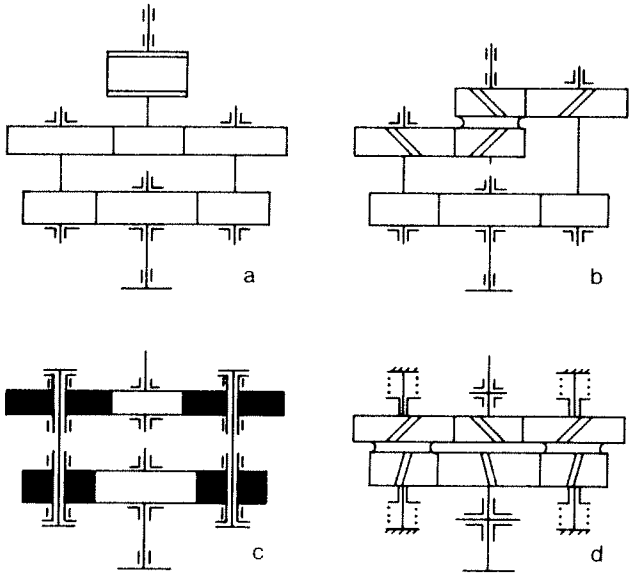

Fig. 24 Methods for balancing tooth forces

1 toothed coupling, radially adjusting pinion
2 axially adjusting pinion
3 torsionally elastic shafts
4 axially adjusting intermediate wheels

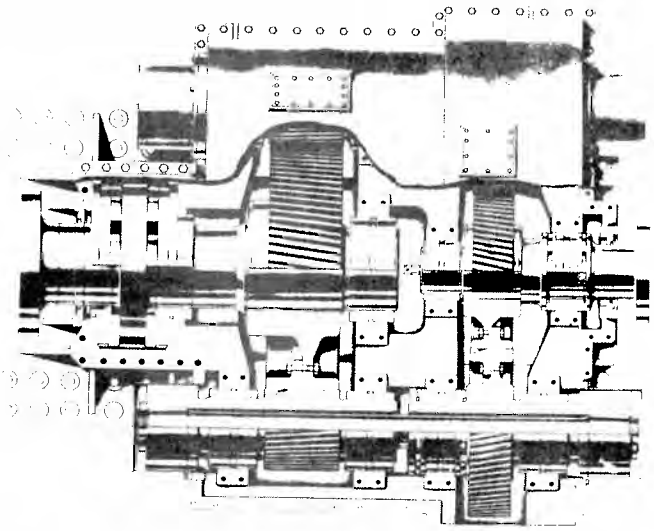

Fig. 25 Big reduction gear unit with two power paths and power distribution with torsionally elastic shafts

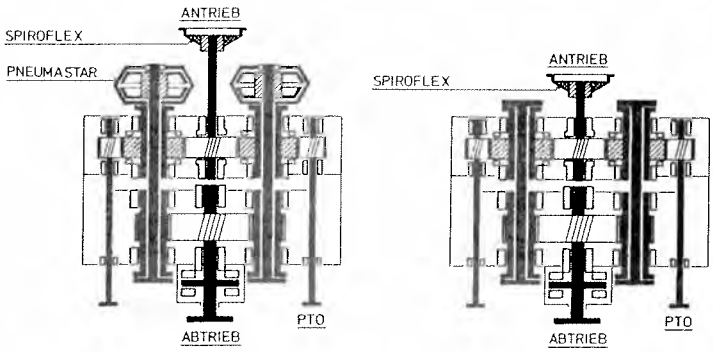

Fig. 26 Reduction gears with torsionally elastic shafts

a without friction clutches (secondary power-take-off)

b with friction clutches (primary power-take-off)

power of the engines, for instant of four engines, Fig. 27. Thus, the output of the most powerful diesel engines can be combined and transmitted by one shaft, which may lead to torques in the range of 4 to 5 MNm or propulsion power up to about 40 000 kW at 80 rpm, what certainly will be sufficient for all actual purposes.

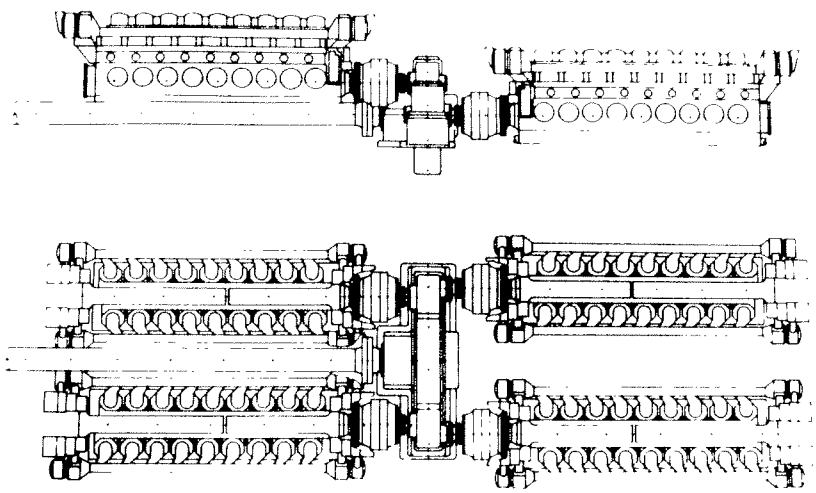

Fig. 27 Summarizing the output of four engines by one gear unit

4. Gearings in interaction with the environment

In the range of the gear case mounting the substructure has to be very sturdy, particularly if the thrust bearing is integrated into the gear case. The foundation shall be free from deflections caused by different loading conditions or rough sea. Thus, gearings of small dimensions, i. e. gears with bull gears of limited size are desirable. They do not need deep intersections into the double bottom and do not interfere with the reinforced substructure. If deformations of the hull are inavoidable, then they should take place where they cannot harm the parallelity of the bearing supports, i. e. between gearing and engine at the position of the elastic coupling or clutch. On the other side the range between the gear unit and the first propeller shaft supporting bearing should be reinforced similar as the gear unit's substructure. All bending of the propeller shaft has to take place in large distance from the gearing. With these precautions the deterioration of the achieved tooth contact can be avoided.

When a gear unit is running under load the generated heat will warm up the housing. This leads to the extension of original dimensions. Alterations of this kind are mostly different to the deviations that occur at the engine or

the engines. All this has to be kept in mind at the installation process. But everything would be more difficult if a tank with warm lubrication oil is installed near the gear case. Thermal extension of the substructure can by no means be compensated. Very much devotion should be paid to these problems in order to beware the tooth contact from any deterioration. Test with gear units of actual size have proven that this can be achieved.

Besides the diminuition of tooth contact which would concentrate the tangential forces to a smaller width, exists the danger of exceeding torque loads. They can be the consequence of torsional vibrations of the total transmission system with its many inertias and elasticities. Particularly the diesel engine can incitate the system to vibrations. Certainly the procedure according to which the dimensions of the involved parts are chosen gives space for torque increments, but the extent of the deviations from the nominal value and the time during they occur is limited. Fortunately the electronic calculators now-a-days easens the investigation into the torsional vibration situation and the decision how to act against any excessive torque peaks. Great importance have the torsionally highly elastic couplings, the couplings with great dampening effect and the slip couplings, each of which in special fields and applications.

Very different from the normal ship propulsion is the propulsion in ice. Here totally unforeseeable conditions can arise: the propeller can blow on ice floes and it can be stuck in the solid ice. Thus, great shock loads can meet the teeth and all other parts of the transmission train. Besides high service factors (which derive from the classification rules) it is a question of the clutches which are applied whether the icegoing ship performs well or not. Quick and reliable action is required. Mechanical clutches and hydrodynamic ones compete in the stage of development of to day and it is open what kind of clutch performs in practice to full content.

Also axial movements of the bull gear depending on elasticities in the thrust bearing or its support can lead to additional tangential loads, particularly at double helical toothing. Then the pinions have to follow the movement of the bull gear and with them the masses to which they are connected. The values of acceleration and deceleration during the axial vibration then determine the additional forces which affect the teeth.

An outstanding example of gear units may close the survey on the gearing technology in the field of ship propulsion /19/. Two crosshead diesel engines type 8 L 55 GUCA manufactured by Hitachi - B & W together produce 15 400 kW at 155 rpm. By highly elastic couplings type Spiroflex KJI 490 are the engines connected with the summarizing gear type Navilus GVS 2400, Fig. 28. Two clutches of the type Pneumastar KUG 490 permit to engage one of the two engines or both of them through the quillshafts of the input pinions. The ratio of the gear reduces the already slow speed of the crosshead engines to 80 rpm. The propeller shaft drives the 8,2 m diameter propeller of this shuttle tanker of 128 000 t. Also two generators can be driven by the engines with the speed of 1200 rpm, Fig. 29. The whole unit has the weight of 170 t. It resemples a masterpiece of the art of advanced gear technology and as well a contribution to the campaign to save fuel quantity and fuel quality /20/.

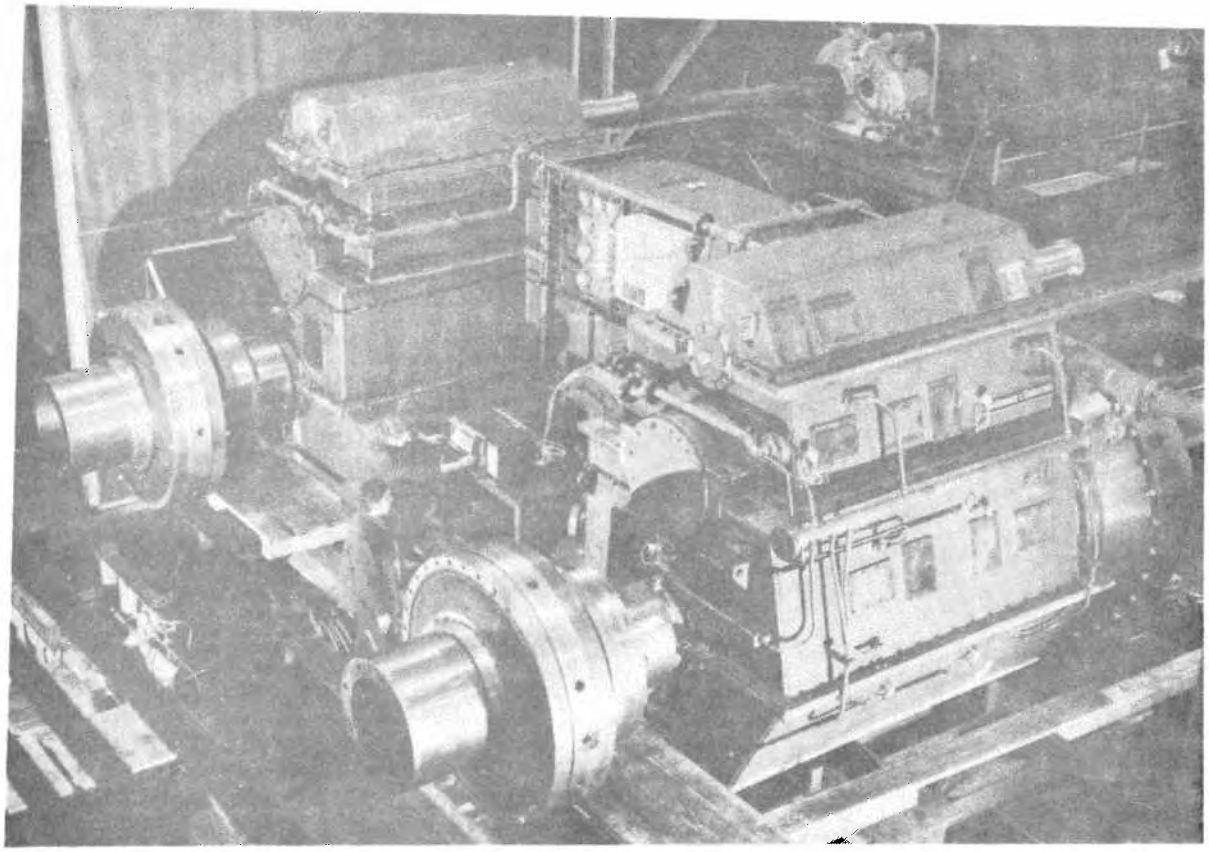

Fig. 28 Summarizing gear unit Navilus GVS 2400
transmitting 15400 kW at 155/80 rpm

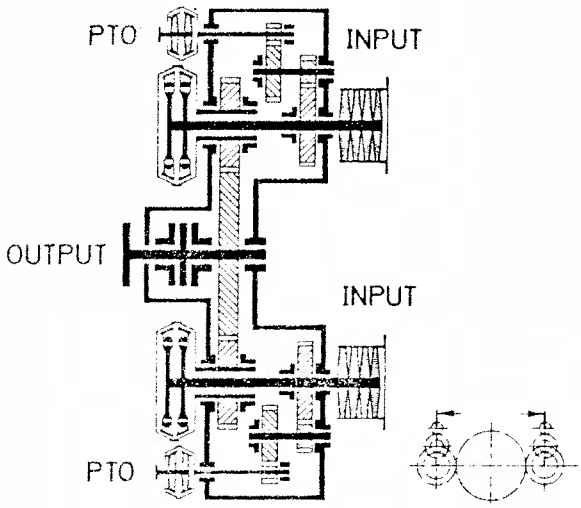

Fig. 29 Schematic of the shuttle tankers
giant gearing

Bibliography

/1/ Hiersig, H. M. and Siefert, K.-H.
Marine gears in combination with crosshead engines,
Shipbuilding & Marine Engineering International,
July/August, 1980, pp. 309-314

/2/ Archer, S.
Marine Propulsion with special reference to Transmission of Power,
Institut Marine Engineering, London, 1964.

/3/ Hiersig, H. M. and Burkhardt, W.
Kompakte Getriebe für mittelschnell-laufende Dieselmotoren,
MTZ 31, 1970, pp. 466-469.

/4/ Neufeld, J.
Bauarten großer Schiffszahnrad-Getriebe,
Konstruktion 6, 1954, Heft 11, pp. 417-428.

/5/ Jung, I., Larsen, G., Larsson, P.-E.
Der Schiffsdampfantrieb von heute und morgen,
STG 1966, pp. 290-308.

/6/ Shannon, J. F.
Marine Gearing,
Institut Marine Engineering, London, 1977.

/7/ Hiersig, H. M.
Einsatzgehärtete Stirnräder für Schiffsgetriebe in Schweißkonstruktion,
MTZ 36, 1975, pp. 332-334.

/8/ Hiersig, H. M.
Bauformen von Gasturbinen-Getrieben insbesondere für Schiffe,
MTZ 38, 1977, Heft 11, pp. 503-508.

/9/ Hiersig, H. M.
Sondergetriebe für Schiffshaupt- und Nebenantriebe,
Institut für Schiffsmaschinenbau Universität Hannover,
Fortbildungskurs 1980.

/10/ Hiersig, H. M.
Getriebe und Kupplungen für den Antrieb von Schiffen durch mittelschnelle Dieselmotoren,
Hansa 106, 1969, Heft 19, pp. 1723-1730.

/11/ Fleischmann, E.
Drehschwingungen in Schiffs-Haupt-Antriebsanlagen,
MTZ 32, 1971, Heft 6, pp. 202-207.

/12/ Ill, M.
Lärmarme Getriebe,
VDI-Berichte 239, 1975, pp. 109-115.

/13/ Hiersig, H. M.
Große einsatzgehärtete Stirnräder als Schweißkonstruktion,
World Congress on Gearing, Paris, 1977, pp. 1227-1239.

/14/ Fleiß, R.
Nichtfluchtende Zahnkupplungen,
Konstruktion 32, 1980, Heft 3, pp. 96-100.

/15/ Hiersig, H. M.
Hochdrehelastische Gummikupplungen für Schiffsantriebe,
ant 9, 1970, pp. 247-279.

/16/ Meier-Peter, H.
Flüssigkeitskupplungen,
Institut für Schiffsmaschinenbau Universität Hannover,
Fortbildungskurs 1980.

/17/ Pinnekamp, W.
"Spiroflex" und "Pneumaflex" - Zwei elastische Kupplungen zur Überwindung von Drehschwingungen in Schiffsantrieben,
Schiff & Hafen 16, 1964, Heft 7.

/18/ Schäfer,W.
Große Schiffsgetriebe unter anlagetechnischen Aspekten,
Hansa 119, 1982, Heft 18, pp. 1197-1200.

/19/ de Jong, E.
Super-Untersetzungsgetriebe für Großdiesel
Antrieb, April 1982, pp. 13 und 14.

General reference:

/20/ Gallin/Hiersig/Heiderich
Ships and their Propulsion Systems
Lohmann & Stolterfoht GmbH, Witten
F. Rep. Germany

WEST EUROPEAN GRADUATE EDUCATION

IN MARINE TECHNOLOGY

EIGHTH SCHOOL August 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

INTERACTION BETWEEN PROPULSION MACHINERY AND HULL STRUCTURE

by

Jan Stefenson

Chalmers University of Technology
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Summary

Interaction between machinery and hull will be observed in a large number of operational features of a ship from steering and stopping the ship to automation of engine plants. Interaction with special reference to ship design for fuel economy is in this paper restricted to a few areas where engines, applied for the purpose, create special problems or require modified criteria for the installation. The areas selected cover alignment of stiff shafting with or without gear transmission, balancing of four and five cylinder engines and some reference to torsional vibration of these engines. A simplified method for estimating the double bottom deflection is proposed to facilitate an alignment suitable for all operating conditions Further, a description of the balancing of the above mentioned engines is included and the devices illustrated. Special method to obtain the necessary balancing of torsionals have been indicated and typical situations for torsional vibration of four and five cylinder engines with short propeller shafting described.

Introduction

In connection with ship design for fuel economy, new or modified types of propulsion machinery will be considered, also new transmission systems and propeller designs. This will influence the design criteria for interaction between machinery and hull.

To obtain low propeller speeds, gears will be installed connected to medium speed diesel engines as well as slow speed engines. The shafting will be more rigid for the lower speeds. These features influence the alignment of the shafting with regard to both accuracy and design criteria.

The introduction of shaft generators to save fuel for auxilary purposes and to operate the complete plant on heavy fuel oil will also favor gears in the transmission system and call for highest quality of shafting alignment.

The race for fuel economy has brought new diesel engine designs to the market. As the bore size increases, the fuel consumption decreases. Consequently, the larger bores are favorable and engines with few cylinders and large bores are preferred. The engines with few cylinders require special means for balancing and the interaction machinery-hull has to be observed with regard to vibration.

Also, for small ships, fuel economy is essential and operation on heavy fuel oil is attractive which in turn favors engines with lower speeds, larger bores and few cylinders compared with high speed diesels operating on destillate fuel (marine diesel fuel). This indicates that also four stroke engines with few (four) cylinders have to be taken into consideration with respect to vibration interaction.

The torsional vibrations for the attractive engines with few cylinders have different features compared with conventional engines; higher additional stresses and resonance at higher rpm. This may also influence the interaction problem.

Superlongstroke engines are now available with low shaft speed. Also for this reason, the larger engine types with lower speed will be attractive. This favors the selection of engines with few cylinders as well.

According to the above introduction, interaction machinery-hull with reference to fuel economy will be covered by the following sections:

1. Alignment of engine gears and shafting.
2. Balancing of engines with 4 and 5 cylinders.
3. Torsional vibration of engines with 4 and 5 cylinders.

1. Alignment of engine, gears and shafting

This chapter describes the efforts to build up a simplified model for the alignment of engines and shafting meeting the accuracy demanded in practical applications (Ref. 1).

An alignment procedure has to define the different aspects of the alignment problem area and propose practical solutions as necessary. The following items will not cover all tasks in design, engineering and construction of the alignment of shafting but emphasizes a few problems which are frequently overlooked. They will form the base for this paper with regard to alignment of the types of engines referred to above.

1. Estimate the double bottom distortions probable from alignment condition (installation) to different operating modes.

2. Transform these distortions into terms of alignment.

3. Adjust alignment criteria during operation into new criteria for the installation procedure.

4. Estimate possibilities to exceed acceptable limits during operation. This will be the most difficult part since we have to compare values from different types of estimates.

5. Check from one set of measurements along the engine frame that the deformation is generally in accordance with our simple deformation geometry.

According to item 1 the most important problem is the double bottom distortion. A simple method for the treatment of this problem will be proposed below.

If the double bottom problem can be given a practical solution, a simple theory and yet a satisfactory accuracy, even all the other problems of the procedure could be treated.

Adoption of the model to practical alignment

The alignment is physically connected to many phenomena of the shafting and engine components.

A short summary is given below of the factors influencing the reliability of the propulsion system and having connection with the alignment.

- The crank shaft and the main bearings may be subject to additional stresses (forces) larger than the mean value of those emanating from the normal operation. See Fig. 1.

- Critical phenomena may arise when malalignment causes unloaded bearings or bearing loads in a direction fatal to the bearing within the engine itself. The dynamic forces on a bearing may cause unacceptable vibration, if the alignment does not provide for a proper minimum bearing load.

- Couplings and clutches are generally very sensitive and must operate within their narrow tolerances of misalignment and axial/radial forces.

- The gears in all geared installations put restrictions on alignment and alignment forces. The gear teeth require far more accuracy to operate properly than normal journal bearings.

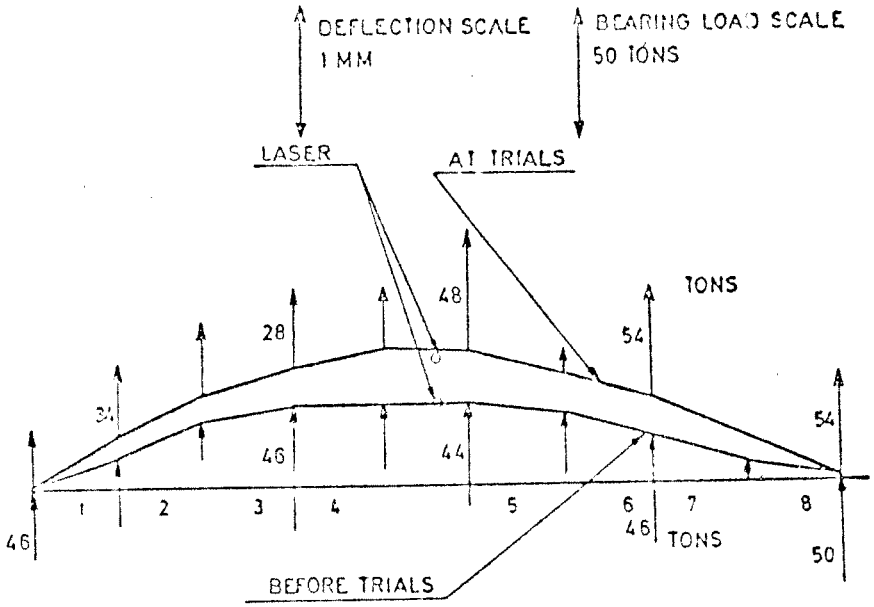
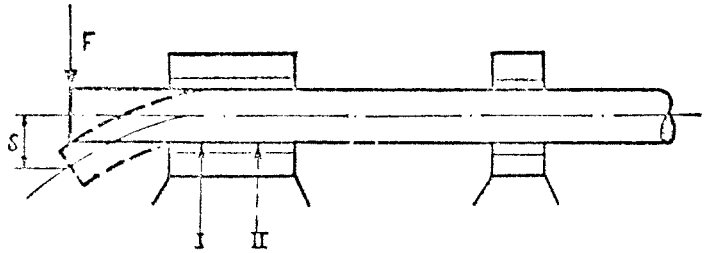


FIG. 1 Crank shaft indication evaluated as deflection and bearing loads.



$F/\delta \sim$

I $0.5 \cdot 10^9$ N/m

II $0.3 \cdot 10^9$ N/m

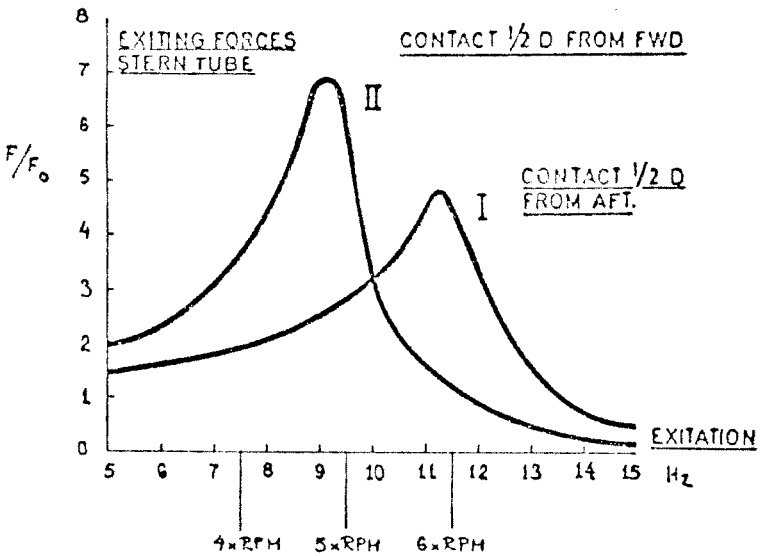


FIG. 2 Dynamic forces on stern tube caused by propeller forces at different points of support.

This is obtained by proper alignment and has been very well introduced in turbine practice.

- The line shaft bearings must operate within their load tolerances at all cargo and sea conditions.

- Dynamic forces caused by the propeller may create undue vibration, if the stern shaft is not properly supported in the bearings. Fig. 2 shows the increase in dynamic response in the stern due to a small change in the point of support, when the stern bearing is worn.

Considering all these phenomena it must be a very difficult task to the designer to find a compromise in the alignment satisfying all the criteria. When in addition the alignment changes considerably during operation it might seem impossible to fulfill all the requirements. The amount of deflection of the system is very essential to know for all operating modes, if the design with respect to alignment shall be successful.

Different patterns of bottom stiffness evaluation

The first idea was to use a very simple model with the center line described as a circular line and the magnitude of deflection given by the radius, shorter for large deflection and longer for small deflection. This model was later developed by introducing different radius for different parts of the double bottom. Practical experience did not support the circular assumption and a standard formula for plate deformation has been introduced.

After some comparisons with measured or thoroughly investigated hulls, the plate formula was combined with hull beam deflection superimposed on the local plate deformation.

In finding the most suitable standard formula, various checks were made also for different types of ships with different engine room locations. It has been realized that the hull beam shear deformation has a vital influence. Also other types of models have been investigated and compared in the first instance with

FEM-methods, since the experience available showed that finite element calculations were very expensive. To develop estimating methods could thus be directed towards simplification of finite element estimates and also to design a simple calculating method based on the normal theories in stress analysis.

The calculating method proposed here, designed to calculate the deformations, is composed by two different parts:

1. Method for main hull beam deformation
2. Method for double bottom deformation

In the development of a simple model for the double bottom deflection, the plate model has been selected in spite of the uncertainty of

- the dimensions of an equivalent triangular plate and
- the supporting conditions at its three edges.

The plate model still offers advantages, because it has been found to require a simple calculation procedure, if only a set of FEM computations applicable to the problem concerned are executed.

Another reason for considering the plate model is that it may consider both transverse and longitudinal stiffness of the double bottom.

The ship hull beam is calculated only with regard to its shear deformation.

This shear beam model for a hull may be criticized for ignoring the bending deformation and the effect of hull side deformation caused by water pressure in describing the deformation of the double bottom structure.

Anyhow this model procedure appears to be greatly appreciated for the following reasons:

- Omitting the bending deformation makes the calculation procedure extremely well defined.
- In most ship hulls the shear deformation obviously is much greater than the bending deformation.
- The effects of the hull bending deformation and water pressure on the hull sides are contributing to the total deformation with opposite signs.

Thus the total deflection of the double bottom is to be estimated as a sum of the two kinds of deformation treated.

Basic concepts of the calculation method

Deformation of double bottom itself

(1) The double bottom structure can be transformed into an equivalent triangular plate with a thickness that gives the same bending stiffness as the beam in its longitudinal direction (Fig. 3).

(2) The three edges are considered to be supported without torque.

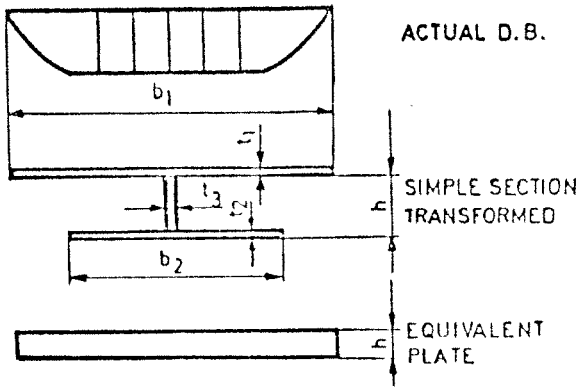
(3) A constant pressure that causes deflection upwards is applied to the whole area of the triangular plate.

(4) The deflection line of the double bottom along the center line can be calculated and plotted by applying a set of FEM.

Deformation of ship hull beam

(1) A ship hull beam can be transformed into a shear beam, the cross sectional area of which is changing stepwise in its longitudinal direction (Fig. 4). The forces by water pressure are applied in the same manner as above.

(2) The boundary conditions of the supports at each end of the beam is assumed to be a simple support, not with the real ship in consideration, but because of the characteristics of the shear beam.



Introduction of equivalent plate for d.b.

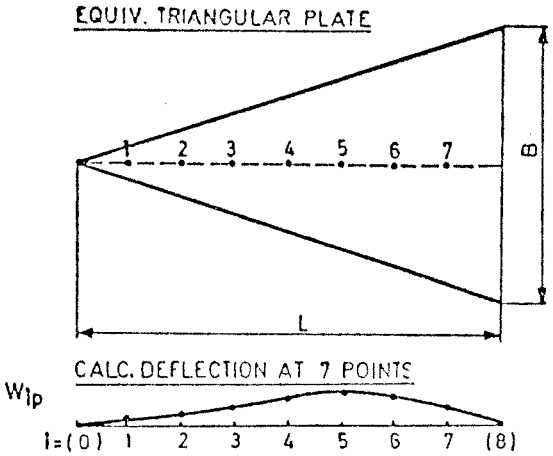


FIG. 3 Bending deformation of double bottom.

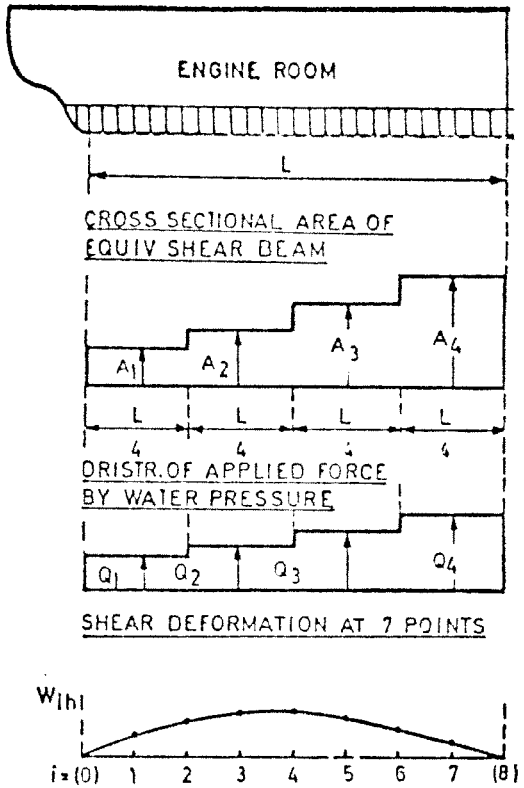


FIG. 4 Shear deformation of hull beam.

Calculation procedures in details

Deformation of the double bottom itself

Input data and output

Input: B: breadth of d.b.
L: length of d.b.
h: thickness of equivalent plate to d.b.
E: Youngs' modulus = $2.0 \cdot 10^{10}$ kg/m^2
ν: Poisson's ration = 0.3
p_o: constant water pressure

Output: W_{ip}: deflection of each point due to the plate's deformation (i = 1.2---7). See Fig. 3 .

Basic formula for deflections

W_{ip} (i = 1.2---7) is to be given by the formula

$$W_{ip} = \varepsilon_i p_o L^4/D$$

where: $D = Eh^3/12(1-\nu^2)$,

ε_i is given by the Table 1 obtained from a separate calculation by FEM and presented in tabular form.

The assignment of output data

(1) B Breadth of d.b.
B is to be the bredth of the d.b. at the front end of the engine room.

(2) L Length of d.b.
L is to be the distance between the front and back end of the engine room.

(3) h Thickness of the equivalent plate to d.b.
h can be given so that the bending stiffness is equal to that in the actual section of the double bottom.

B/L	ε_1	ε_2	ε_3	ε_4	ε_5	ε_6	ε_7
0.5	0.0003	.0060	.0260	.0657	.1147	.1405	.1017
0.6	.0007	.0141	.0549	.1256	.2004	.2278	.1569
0.7	.0020	.0285	.0996	.2086	.3098	.3327	.2211
0.8	.0045	.0509	.1614	.3138	.4398	.4521	.2924
0.9	.0092	.0825	.2403	.4391	.5872	.5832	.3693
1.0	.0167	.1241	.3351	.5812	.7482	.7227	.4501

The deflection of W for each point (i=1,2 ---- 7) it to be calculated by the expression of

$$W_{ip} = \varepsilon_i (P_0 L^4/D) \times 10^{-3}$$

where:

P_0 = constant water pressure

L = length of double bottom

B = width of d.b.

D = $Eh^3/12(1-\nu^2)$

h = thickness of the plate

E = Young's modulus

ν = Poisson's ratio

TABLE 1. THE DEFLECTION LINE OF THE CENTER LINE OF D.B.

(4) $\underline{P_o}$ Water pressure applied to d.b.

P_o is to be given at the bottom of the d.b., which is equal to γd, where γ being the specific weight of sea water and d the draught.

(5) $\underline{\gamma}$ Specific weight of sea water

$\gamma = 1.035 \cdot 10^3$ kg/m^3

(6) $\underline{d}$ Draught difference

This is the difference between the reference condition and the actual case.

Shear deformation of the ship hull beam

Input and output

Input: L: length of the shear beam concerned

A_1---A_4: cross sectional areas of four segments of the beam

G: shear modulus = $0.83 \cdot 10^{10}$ kg/m^2

Q_1---Q_4: water pressure force density at each segment of the beam specified by the value of force per length unit including the weight of structure

Output: W_{ih}: deflection at each point due to shear (i = 1---7)

Basic formula

W_{ih} (i = 1.2---7) is to be given by the formulas

$$W_{ih} = (Q_r L^2/GA_r) \sum_j^4 \zeta_{ij} q_j \quad (i = 1.2---7)$$

where: A_r: reference area of cross section

a_j: $= A_r/A_j$ (j = 1,---4)

Q_r: reference density of applied force

q_r: $= Q_j/Q_r$ (j = 1,---4)

and ζ_{ij} being influence factors defined as

$$\zeta_{ij} = (R_i/1024) \sum_{k=1}^{4} S_{ijk} a_k$$

where R_i and S_{ijk} are given by Table 2.

The assignment of input data

(1) L Length of shear beam
L is identical to the length of d.b.

(2) A_1---A_4 Sectional areas of beam
These are estimated as a mean effective area at four points to be, for example, A_4 = (area at the point 6 + area at the point 8)/2.

(3) Q_1---Q_4 Pressure force density and weight of structure
Q_i may be given from the displacement volume of part i, which is equal to $4\gamma V_i/L$, while γ is the specific weight of sea water and V_i is the displacement volume at the part i calculated by the draught. Structure weight of part i is deducted.

Comparison between calculations with finite elements and simplified method

There has been an opportunity to compare test calculations according to the above methods and available extensive calculations by finite element methods. The results indicate a general conformity between the calculations with an accuracy that seems acceptable for most practical cases. However, when the simplified method indicates smaller deformations than the FEM-method, there is a risk of underestimating the bearing loads and shaft stresses in the operation and the safety margins must be set accordingly in the design of the system.

i	j	R_i	S_{ij1}	S_{ij2}	S_{ij3}	S_{ij4}
1	1	1	17	1	1	1
1	2	1	15	-1	3	3
1	3	1	9	-3	1	5
1	4	1	3	-1	-1	3
2	1	2	9	1	1	1
2	2	2	15	-1	3	3
2	3	2	9	-3	1	5
2	4	2	3	-1	-1	3
3	1	1	15	-1	3	3
3	2	1	25	9	9	9
3	3	1	15	3	3	15
3	4	1	5	1	-3	9
4	1	4	3	-1	1	1
4	2	4	5	1	3	3
4	3	4	3	3	1	5
4	4	4	1	1	-1	3
5	1	1	9	-3	1	5
5	2	1	15	3	3	15
5	3	1	9	9	9	25
5	4	1	3	3	-1	15
6	1	2	3	-1	-1	3
6	2	2	5	1	-3	9
6	3	2	3	3	-1	15
6	4	2	1	1	1	9
7	1	1	3	-1	-1	3
7	2	5	5	1	-3	9
7	3	1	3	3	-1	15
7	4	1	1	1	1	17

$$\zeta_{ij} = (R_i/1024)(S_{ij1}a_1+S_{ij2}a_2+S_{ij3}a_3+S_{ij4}a_4)$$

$$= R_i\lambda_{ij}$$

TABLE 2. INFLUENCE FACTORS TO GIVE SHEAR DEFORMATION.

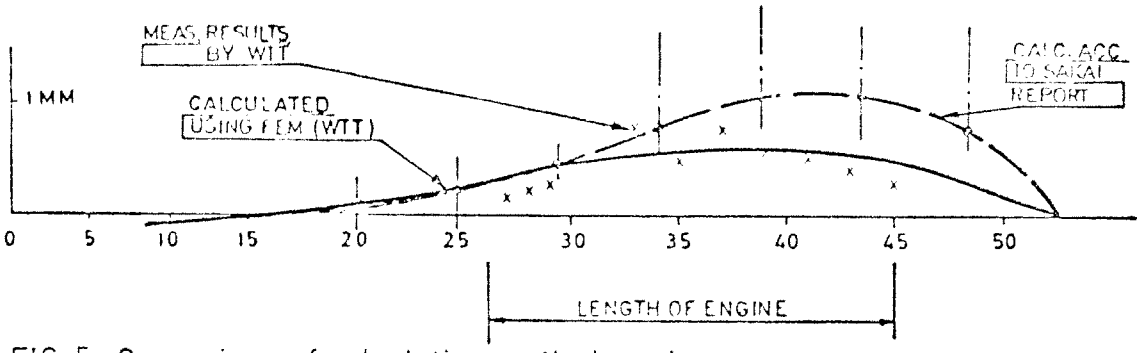


FIG. 5 Comparison of calculating methods and measurements for d.b. LPG-tanker (Wärtsilä)

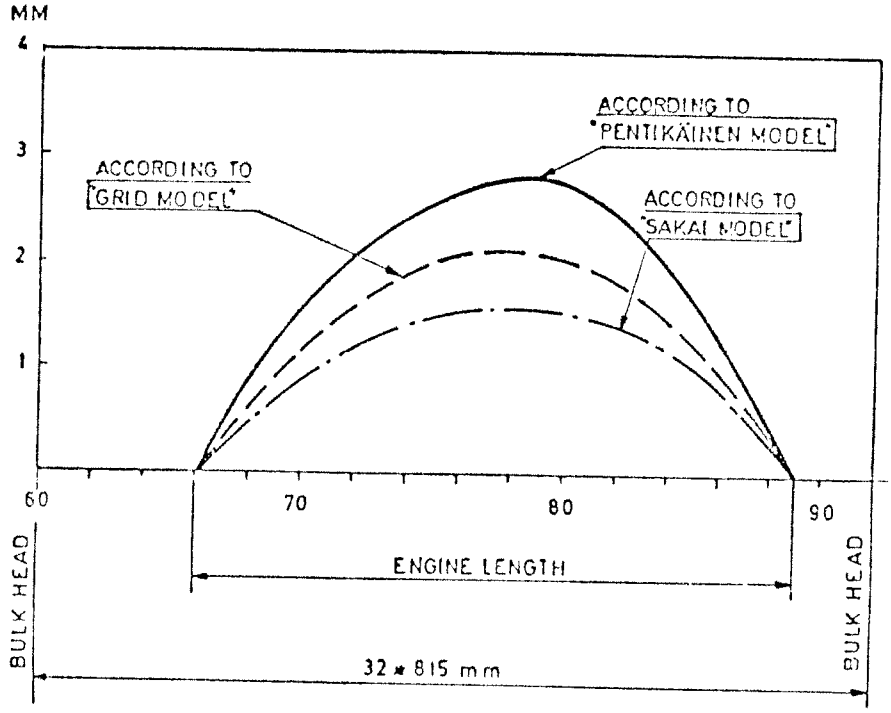


FIG.6 Comparison of methods for d.b. calculation (Sulzer)

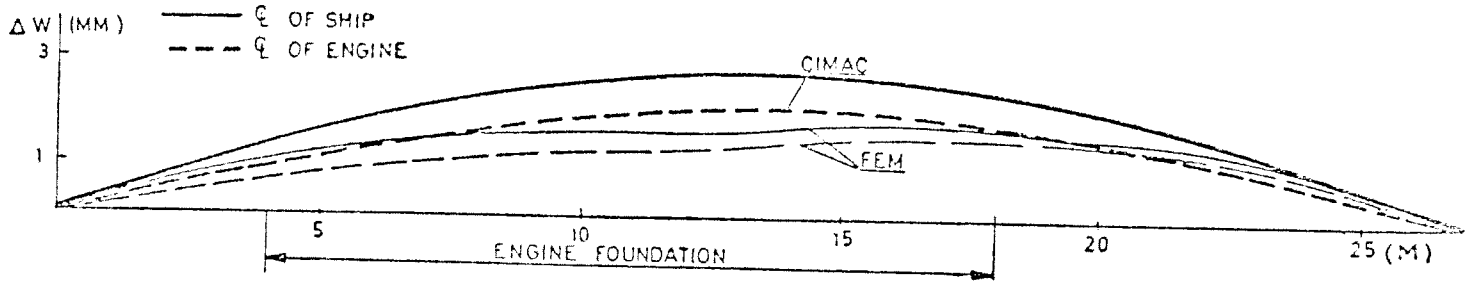


FIG. 7 Comparison of methods for d.b. calculation. Container v. (MAN)

Wärtsilä have investigated a LPG-vessel and as shown in Fig. 5 , the proposed model gives larger deformation in the forward part of the engine room. Over the length of the engine the results conform very well. Also measured results have been indicated being of the same magnitude as the calculated.

Sulzer have provided one interesting comparison for a container ship according to Fig. 6, where both a transverse beam model (Pentikäinen model) and triangular plate and shear beam model have been tested. A more elaborate grid model gives values in between. In this case, however, the proposed model indicates the smallest distortion.

For the maximum deflection of the double bottom, there are the following variations from the FEM calculation:

Fig. 7 : + 35% (center line of shafting); 55% (c.l.)

Fig. 8 : - 17% (c.l.)

Fig. 9 : - 21% (c.l.)

The values show that the magnitude of deflection has been covered quite well by the model, so that, in MAN opinion, a good basis for a simple calculation method has been created.

In case of the twin screw container ship, Fig. 7 , it may be observed that the engine room is situated in a more forward position, where the double bottom has an almost rectangular shape.

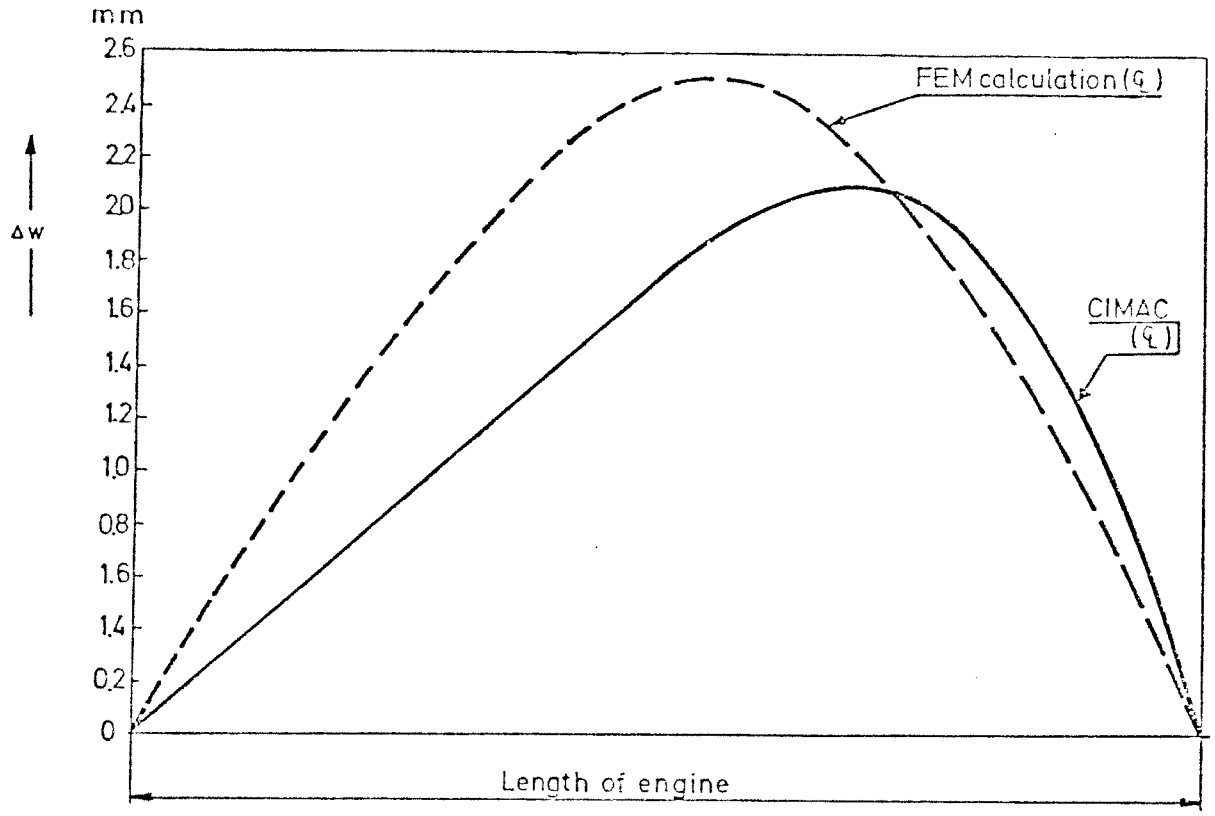


FIG 8 Comparison methods for d.b. calculation. Turbin tanker (MAN)

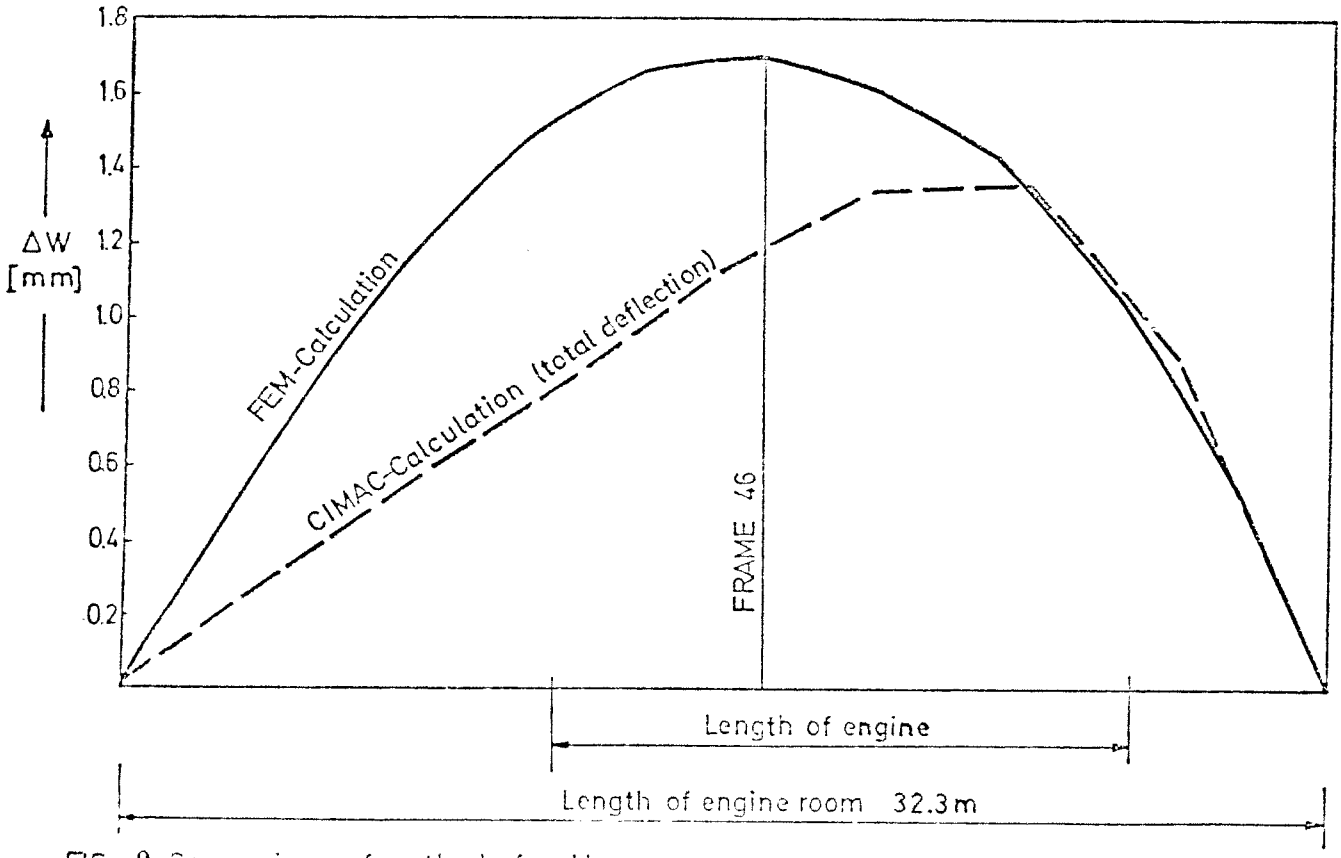


FIG. 9 Comparison of methods for d.b. calculation. Container ship (MAN)

A practical model designed according to the above principles would be used and applied in far more cases and within a much broader register of ship types than has been the case with the complicated models used on larger ships at very high costs. In addition, a simplified model has the advantage that it may be applied at several stages of the project and design work, thus enabling a comparison of different alternatives. It is believed that the method proposed will be suitable for fast accumulation of experience in the field, for optimum utilization of the material in the hull structure supporting the propulsion engine and shafting and for obtaining a realiable machinery with respect to alignment.

2. Balancing of engines with 4 and 5 cylinders

The balancing of slow speed diesel engines with few cylinders has recently been covered by Sulzer and MAN-B&W in the current technical press ref (2) and (3). From these papers, the following paragraphs have been abbreviated.

The engine's inertia forces originating from unbalanced rotating and reciprocating masses, as illustrated in Fig. 10, will create unbalanced external moments of both first and second order. This means moments varying periodically with a frequency equal to the engine speed and equal to twice the engine speed, respectively. Since these external moments have significant magnitude they might under adverse conditions give rise to disturbing vibrations of the ship's hull.

It should be noted, however, that for the four cylinder engine the size of the first order moment is only about half the second order moment. Furthermore, a number of ships have already entered service and none has had any problems with hull vibrations excited by the first order external moments.

The following will deal with the first order external moment, represented by M1V, the maximum value of the alternating moment acting in the vertical plane; and M1H, the maximum value of the alternating moment acting in the horizontal plane. Methods will be examined to control these external moments so that excessive hull vibrations excited by first order external moments is prevented.

M1V and M1H are the resultants of the three separate moments: M1rot, M1rec and M1bal as illustrated in Fig. 11.

(I) M1rot is a rotating moment originating from centrifugal forces from the unbalanced rotating masses of crankthrows, crankpin bearings and the part of the connecting rod which rotates with the crankshaft. The plane in which this moment is acting rotates with the crankshaft and the moment (M1rot) is acting successively in all fore and aft orientated planes of the ship.

Particular attention is paid to the moment when it is acting in the vertical plane and when it is acting in the horizontal plane.

Only a moment of first order can occur.

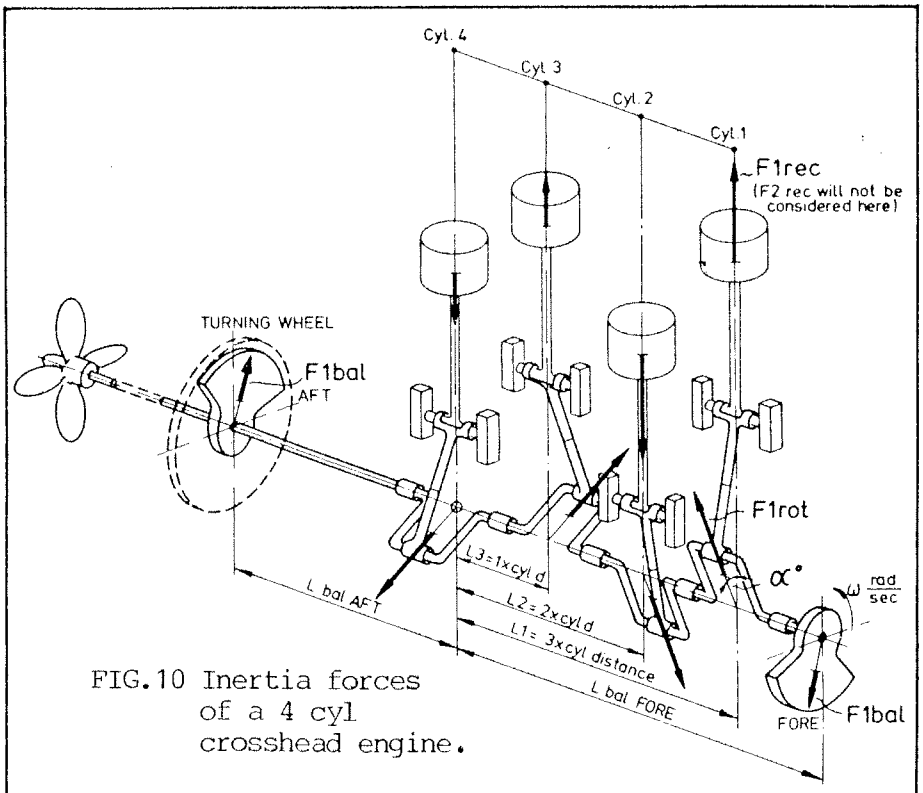


FIG.10 Inertia forces of a 4 cyl crosshead engine.

Inertia forces at engine angular speed ω (radians per second) are as follows:

$F1rot = m_{rot} \times r \times \omega^2$ m_{rot} being the unbalanced mass of crankthrow, crankpin-bearing, and lower part of connecting rod. All masses transferred to crank radius r used as the gravity radius.

$F1rec = (m_{rec} \times r \times \omega^2) \times \cos\alpha°$ m_{rec} being the mass of piston, crosshead, and upper part of connecting rod. F1rec can be considered as the vertical projection of a constant vector: $m_{rec} \times r \times \omega^2$ rotating with same orientation as F1rot (of same cylinder).

$F1bal = m_{bal} \times r_G \times \omega^2$ m_{bal} being the counterweight mass with gravity radius r_G.

Each of these force categories is balanced for the engine as a whole, but they produce the unbalanced external moments; M1rot, M1rec and M1bal which appear from Fig. 2. These external moments can be determined as vectorial resultants originating from the individual moments from each single inertia force with respect to, for instance, cyl. 4.

The numerical values of the individual moments are:

from cylinder 1 : M1 = F x L1 = F x 3 x cyl. distance
" " 2 : M2 = F x L2 = F x 2 x " "
" " 3 : M3 = F x L3 = F x 1 x " "
" " 4 : M4 = F x 0 = 0
" counterweight FORE : M FORE = F1bal x Lbal FORE
" " AFT : M AFT = F1bal x Lbal AFT

Note: F is an abbreviated designation for F1rot and F1rec, respectively.

Crankshaft positions of special interest, seen from FORE

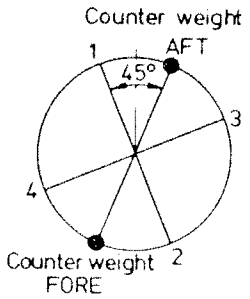
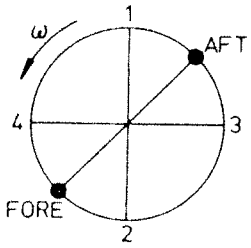
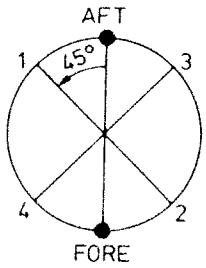
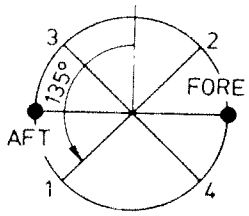
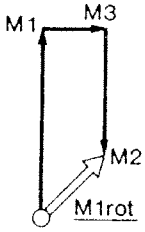
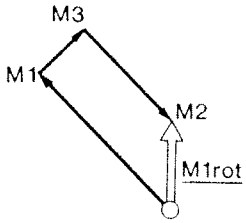
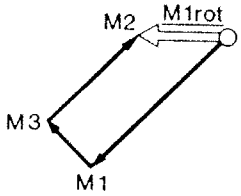
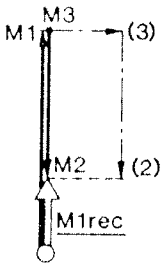
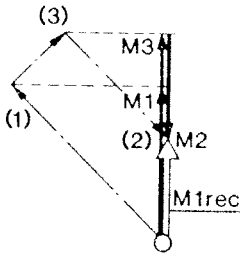
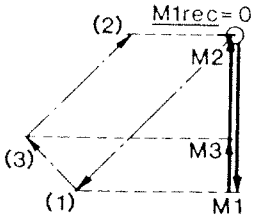
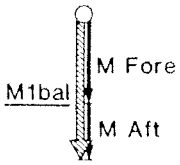
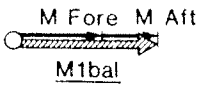
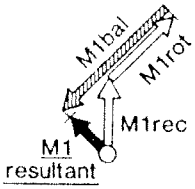
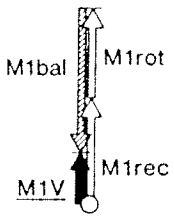
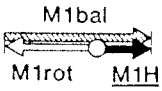
	 Crankshaft position as in Fig. 1	 Initial position: crank cyl. 1 Top Dead Centre	 Position for Max. vertical external moment: **M1V**	 Position for Max. horizontal external moment: **M1H**
I) M1rot is the resultant rotating moment with respect to cyl. 4 produced by the centrifugal forces of the unbalanced rotating masses from each cylinder.				
II) M1rec is the resultant reciprocating moment with respect to cyl. 4 produced by 1st order inertia forces of reciprocating masses from each cylinder. The actual inertia force is considered as the vertical projection of a constant size rotating vector.				
III) M1bal is the resultant balancing moment with respect to cyl. 4 produced by the centrifugal force of counterweight FORE and the centrifugal force of counterweight AFT (integrated in turning wheel)				
IV) **M1V** and **M1H**, transferred to the ship's hull via engine main bearings and engine foundation. **M1V** and **M1H** are the resultant moments from M1rot, M1rec and M1bal illustrated above.				

Fig 11. The external moments M1V and M1H resulting from M1rot, M1rec and M1bal.

(II) M1rec is a moment originating from the inertia forces of the reciprocating masses of pistons, piston rods, crossheads and the reciprocating part of connecting rods. This moment only acts in the vertical plane of the ship, and reaches its maximum value simultaneously with the above mentioned rotating moment M1rot acting solely in the vertical plane. For a four cylinder engine with firing order 1-3-2-4 this will occur when the crankthrow of No 1 cylinder is 45 deg after top.

After a further 90 deg rotation of the crankshaft, M1rot is acting solely in the horizontal plane and M1rec will now be zero.

Because of the complex movement of the reciprocating masses, a second order external moment (M2rec) will be created as well. But M2rec, acting only in the vertical plane and sometimes counteracted by a second order moment compensator, will be dealt with separately. M2rec is generally designated M2V.

(III) M1bal is a balancing moment originating from centrifugal forces produced by integrating one counterweight in the engine turning wheel and mounting another counterweight on the front end of the crankshaft. The counterweights are placed so that the resultant rotating balancing moment, M1bal, produced by them always acts in the opposite direction of the rotating moment M1rot mentioned in item (II). Only a moment of first order can occur.

The relationship between M1V, M1H and M1rot, M1rec and M1bal can be illustrated by the following four different balancing examples. Example No 4, also used as an illustration in Fig. 11, shows the B&W standard balancing providing the same maximum external moment in the vertical plane as in the horizontal plane = half of M1rec.

All the examples follow the basic relations:

Maximum moment in vertical plane:

$M1V = M1rec + M1rot - (M1bal)$

Maximum moment in horizontal plane:

$M1H = 0 + M1rot - (M1bal)$

Example (1) No balancing counterweights at all will provide:

$M1V = M1rec + M1rot - (0)$

$= M1rec + M1rot$

$M1H = 0 + M1rot - (0)$

$= M1rot$

Example (2) M1bal with same value as M1rot will provide:

$M1V = M1rec + M1rot - (M1rot)$

$= M1rec$

$M1H = 0 + M1rot - (M1rot)$

$= 0$

Example (3) M1bal equal to (M1rot + M1rec) will provide:

$M1V = M1rec + M1rot - (M1rot + M1rec) = 0$

$M1H = 0 + M1rot - (M1rot + M1rec) = (-)M1rec$

Example (4) Illustrating B&W standard balancing

M1bal equal to (M1rot + half of M1rec) will provide:

$M1V = M1rec + M1rot - (M1rot + 0{\cdot}5 \times M1rec) = 0{\cdot}5 \times M1rec$

$M1H = 0 + M1rot - (M1rot + 0{\cdot}5 \times M1rec) = (-)0{\cdot}5 \times M1rec$

Adjustable counterweights

If disturbing hull vibrations should arise either in the vertical or the horizontal plane at a particular engine speed, it is normally sufficient to change this speed by a few revolutions to eliminate the vibrations. This, of course, is not practicable at engine speeds close to maximum continuous rating, but in that case the hull vibrations can instead be eliminated by reducing the size of the external moment acting in the hull vibration plane concerned.

It is unlikely that vibration problems should simultaneously occur in the other plane, as the natural frequencies in the vertical and horizontal hull planes do not normally coincide.

From the above balancing examples, Nos 2 and 3, it can be seen that it is possible to achieve a particularly small external moment in either the vertical or the horizontal plane by transferring the moment to the opposite plane by adjusting the size of the balancing moment: M1bal. Based on this fact B&W has developed an additional adjustable balancing system for four cylinder engines comprising a pair of adjustable counterweights attached to the main counterweight at the front end of the crankshaft and a similar pair of adjustable counterweights attached to the turning wheel, which is also provided with a main counterweight.

As shown in Fig. 12, each pair of adjustable counterweights consists of two angular segments guided in a circular slot and each fixed with three screws. The position of the segment can be changed.

First order compensator

As already mentioned, the connection between the natural hull frequencies in the vertical and horizontal planes is normally such that for the engine running in the vicinity of nominal speed, the probability of resonance vibrations exists only in the one plane. Hence occasional hull vibration problems can normally be cured by minimising the external moment in the plane concerned by means of the adjustable counterweights. However, in case of quite special hull vibration conditions, a request to eliminate any possibility of excitation of first order vibrations in both the vertical and the horizontal planes can be met by introducing a special moment compensator driven from the aft end of the engine.

The manner of complying with this demand can be illustrated as follows:

Fig. 13, illustrates the relationship between the natural frequencies, node numbers and ship sizes for commonly available ship types.

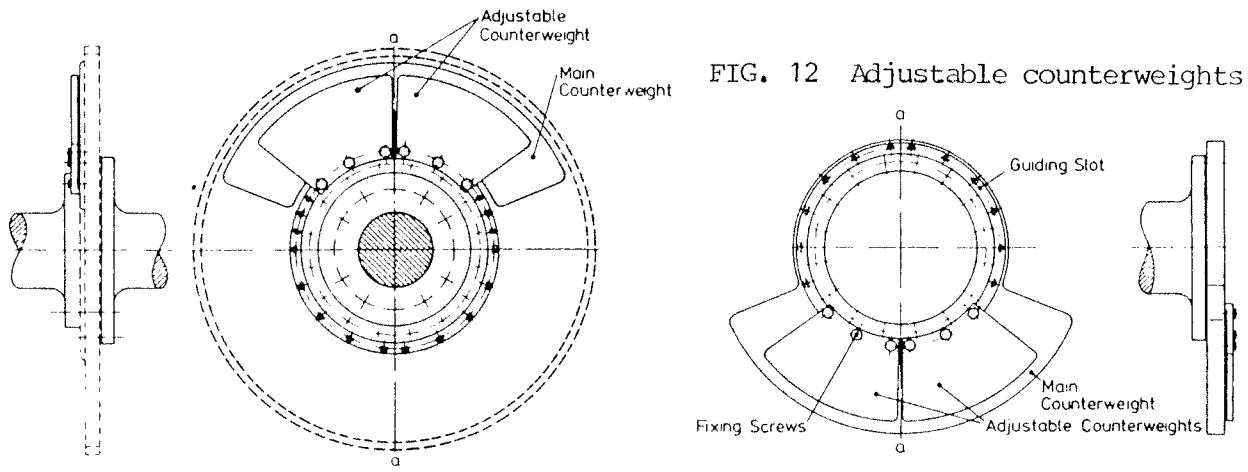


FIG. 12 Adjustable counterweights

The adjustable counterweights can be fixed in seven different positions to provide seven different balancing moments. For each position the segments are mounted symmetrically to the axis a-a.

Left. Main counterweight (aft) integrated in the turning wheel with a pair of adjustable counterweights attached. Right. Main counterweight (fore) with a pair of adjustable counterweights attached.

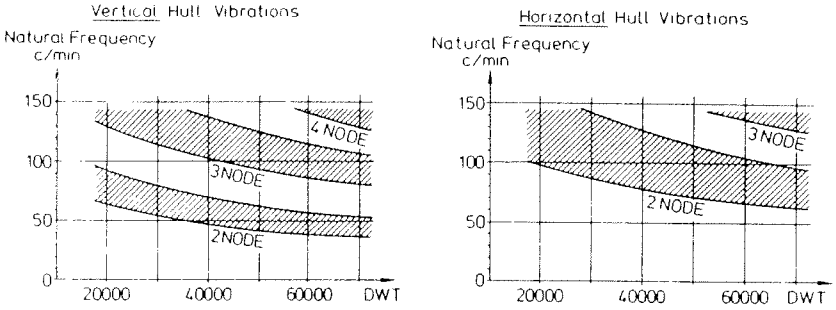
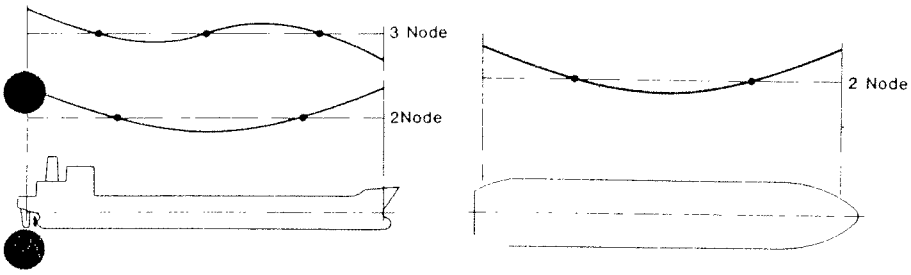
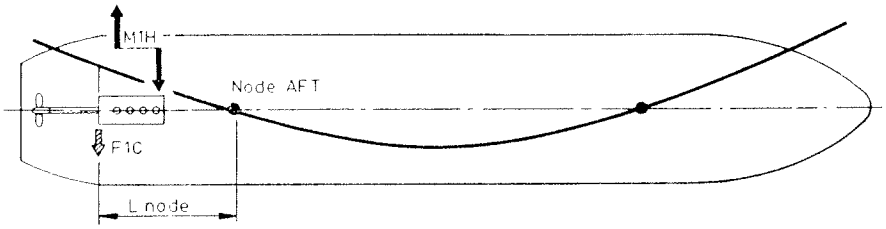


Illustration of ship hull natural frequencies for various sizes of tankers and bulk carriers which could be equipped with four cylinder L-MC and L-GB low speed engine types having the following powers and engine speeds:

4L90MC: 13 760kW at 74 rev/min
4L80MC: 10 840kW at 83 rev/min
4L70MC: 8 320kW at 95 rev/min
4L60MC: 6 120kW at 111 rev/min
4L90GB: 13 500kW at 97 rev/min
4L80GB: 10 400kW at 106 rev/min
4L67GB: 7 350kW at 123 rev/min

FIG. 13 Ship hull natural frequencies





Moment compensating method for two-node horizontal hull vibrations. The compensating force F1C shall create a moment M1C with respect to node aft of the same magnitude as the horizontal external engine moment M1H. Hence M1C = F1C × Lnode out balances M1H, and the value can be determined accordingly.

FIG. 14 Method for compensating horizontal hull vibrations

It will be seen that vertical hull vibration resonance, excited by the first order vertical engine moment, might occur for both 2 node and 3 node hull vibrations when using one of the above mentioned four cylinder low speed engines as a prime mover, bearing in mind that the excitation frequency is equal to the engine speed. It is obvious, however, that in the horizontal plane resonance vibrations can only occur for 2 node hull vibrations, because the natural frequency for 3 node horizontal vibrations is too high to be excited by the horizontal engine moment alternating with engine speed frequency.

Based on this knowledge the adjustable counterweights are moved to the position providing the minimum size of M1V by which, as a first step, hull vibrations in the vertical plane will be reduced to a harmless level. However, this balancing will cause the horizontal moment M1H to increase to it maximum value.

As the next step the ability of M1H to create two-node horizontal hull vibrations is then eliminated by introducing a counteracting horizontal moment by means of a horizontal alternating force of an adequate size antiphasing M1H. This force, with its maximum value designated F1C, only acts in the horizontal plane and provides the counteracting moment with respect to the vibration node aft, see Fig. 14.

Half of F1C is produced by a counterweight driven by the camshaft chain drive and rotating in the opposite direction of the crankshaft. The other half of F1C is a force produced by an imaginary counterweight rotating with the engine turning wheel. This latter force is created simply by moving the adjustable counterweights to a new position fulfilling the demand for obtaining this force, together with the demand for minimum M1V.

As two-node hull vibrations have well predictable node positions, and the aftermost node has a significant distance from the engine site in aft end installations (being almost universal now), only a small and fairly well defined force, F1C, is required.

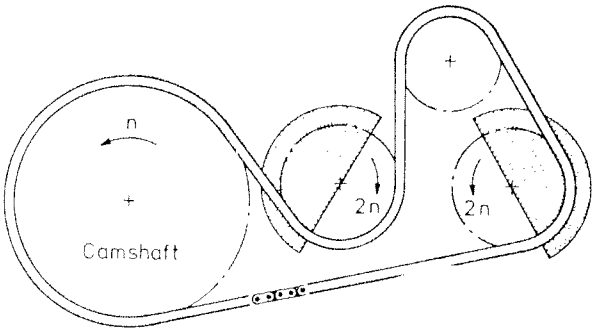
As will be seen from the above explanation, the first order balancing of four-cylinder two stroke diesel engines is well under control, with the possibility of transferring the external unbalanced moment from the horizontal plane to the vertical plane, and vice versa, in a very simple way and with the additional possibility of introducing a compensator in extreme situations.

Second order vertical couple

The second order vertical couple tends to "rock" the engine around a transverse axis at twice the engine speed. Only if the engine is located on or in the vicinity of a node of the hull vertical vibration and the natural frequency coincides with the frequency of the couple can this cause harmful or disturbing vibrations in the hull. The countermeasure (Ref. 3) is to create a synchronous couple in opposite direction to balance the engine generated moment. A pair of Lanchester balancers, driven off the camshaft at both ends of the engine (Fig. 15), each create a vertical reciprocating force in opposite direction (generated by counter-rotating excentric masses) forming together a couple. The chain-drive off the camshaft keeps this couple in strict phase opposition to the engine unbalanced couple; the dimensions of the masses provide for the correct magnitude.

The Lanchester balancers compensate the second-order vertical couple within the engine; no couple is thus transmitted to the hull. This system offers an effective solution to second-order unbalance. Table 3 shows for which engines such balancers are recommended.

Although such balancers add to the mechanical complexity of the engine, they may operate with high reliability if installed with quality engineering and the design properly integrated with the engine design.



SULZER
0781 3193

5 11 81
3-107 128 219

Fig. 15 – Principle of a *Lanchester balancer* (two counter-rotating excentric masses rotating at twice the engine speed) for complete compensation of second order vertical unbalanced couples of the engine.

	number of cylinders							
	4		5		6		>6	
	balancer*		balancer*		balancer*		balancer*	
	1	2	1	2	1	2	1	2
RTA 84	B	A	D	A	D	A	D	D
RTA 76	B	A	D	A	D	B	D	D
RTA 68	B	B	D	B	D	B	D	D
RTA 58	C	C	D	B	D	C	D	D
RTA 48	C	C	D	C	D	C	D	D
RTA 38	C	C	D	C	D	C	D	D

* *balancer type :*

1 : *combined 1st/2nd order balancer*

2 : *2nd order balancer*

A : *balancer is needed*

B : *balancer may be needed in some cases*

C : *balancer will only exceptionally be needed*

D : *balancer is not needed*

Table 3 – Cylinder numbers of *RTA* engines where balancers may be needed.

	1	2	3					4					5					6				
								engine standard ratings														
1			R1 = engine MCR					R2					R3					R4				
2	engine type	number of cylinders	engine speed	external couples 1st order M1V	1st order M1H	2nd order M2V	ΔM	engine speed	external couples 1st order M1V	1st order M1H	2nd order M2V	M	engine speed	external couples 1st order M1V	1st order M1H	2nd order M2V	ΔM	engine speed	external couples 1st order M1V	1st order M1H	2nd order M2V	ΔM
			rev/min	± kNm	± kNm	± kNm	± kNm	rev/min	± kNm	± kNm	± kNm	± kNm	rev/min	± kNm	± kNm	± kNm	± kNm	rev/min	± kNm	± kNm	± kNm	± kNm
3	RTA 84	4	87	***0	***0	***378	1,602	82	***0	***0	***336	1,457	74	***0	***0	***274	1,640	70	***0	***0	***245	1,486
		5		481	481	**579	1,258		427	427	**515	1,147		348	348	**419	1,253		311	311	**375	1,140
		6		0	0	**0	875		0	0	**0	797		0	0	**0	871		0	0	**0	792
		7		283	283	731	645		251	251	650	590		205	205	529	643		183	183	474	586
		8		235	235	0	406		209	209	0	376		170	170	0	405		152	152	0	374
		9		520	520	1,226	266		462	462	1,089	249		376	376	887	265		337	337	794	248
		10		245	245	863	227		218	218	767	211		177	177	625	227		159	159	559	211
		12		0	0	0	164		0	0	0	150		0	0	0	163		0	0	0	149
4	RTA 76	4	95	***0	***0	***280	1,202	90	***0	***0	***252	1,091	81	***0	***0	***204	1,229	76	***0	***0	***179	1,125
		5		356	356	**429	942		329	329	**385	858		259	259	**312	938		228	228	**275	861
		6		0	0	1,867	655		0	0	1,675	595		0	0	1,357	652		0	0	1,195	597
		7		213	213	542	483		191	191	489	440		155	155	394	481		136	136	347	441
		8		715	715	0	304		642	642	0	282		520	520	0	303		458	458	0	282
		9		385	385	908	199		346	346	815	186		280	280	660	198		246	246	581	186
		10		181	181	639	171		162	162	574	159		132	132	465	170		116	116	409	159
		12		0	0	0	123		0	0	0	113		0	0	0	123		0	0	0	113
5	RTA 68	4	105	0	*1,610	1,516	877	99	*0	*1,432	1,348	796	89	0	*1,156	1,090	899	84	*0	*1,030	971	823
		5		255	255	1,888	686		227	227	1,678	624		183	183	1,356	684		163	163	1,208	628
		6		0	0	1,313	477		0	0	1,168	434		0	0	944	476		0	0	841	436
		7		157	157	382	351		140	140	340	321		113	113	275	351		100	100	245	323
		8		510	510	0	222		453	453	0	205		366	366	0	221		326	326	0	205

remarks : *M1V = external couple* — *1st order vertical*
M1H = external couple — *1st order horizontal*
M2V = external couple — *2nd order vertical*
ΔM = reaction of torque variation
all free forces 1st and 2nd order are totally balanced

** engine with non-standard counterweights (max. possible reduction of M1V).*
**** engine generally equipped with 2nd order balancing gears.*
***** engines equipped with combined 1st/2nd order balancing gears.*

Unbalanced couples of the Sulzer engine types RTA 84, 76, 68

Table 4 - *RTA 84, 76, 68 :* **unbalanced couples.**

Table 4 indicates the magnitudes of the external couples for Sulzer engines of the larger long stroke types with 3 to 6 cylinders. Unbalanced couples may be accepted to a certain extent according to practical experience. Unbalanced external forces on the other hand may not be accepted.

3. Torsional vibration of engines with 4 and 5 cylinders

The torsional vibrations of the shafting system have of course to be calculated for any engine. With today's largely prevailing aft engine rooms and consequently short shafts, the major critical torsional frequency of four-, five- and six-cylinder engines tends to fall, without countermeasures, at or near the engine's service speed range (Fig. 16). This situation is prevailing for a wide range of different diesel engines making it possible to apply the experience gained with earlier engines.

Two solutions are normally applied:

- For four and five cylinders and very short shafts, the common procedure is to increase the shaft diameters (particularly for the intermediate shaft) and to use a very light flywheel; as a consequence, the natural frequencies of the shaft system will be increased, thus pushing the major critical above the engine maximum speed, leaving the entire service range clear of detrimental vibrations and of any forbidden speed range.

- The opposite procedure is applied to somewhat longer shafts and normally to six-cylinder engines; the thinnest allowable shafting and a heavy flywheel reduce the natural frequency; the critical frequency now lays at low speed, where the excitation and thus the amplitude are smaller; a forbidden speed

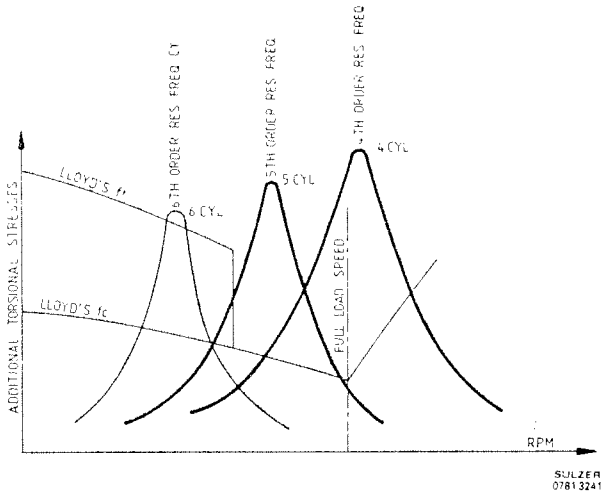


Fig. 16 – Torsional vibrations of 4-, 5- and 6-cylinder engines with short shafting : additional stresses at the resonant frequency exceed the classification societies' limits.

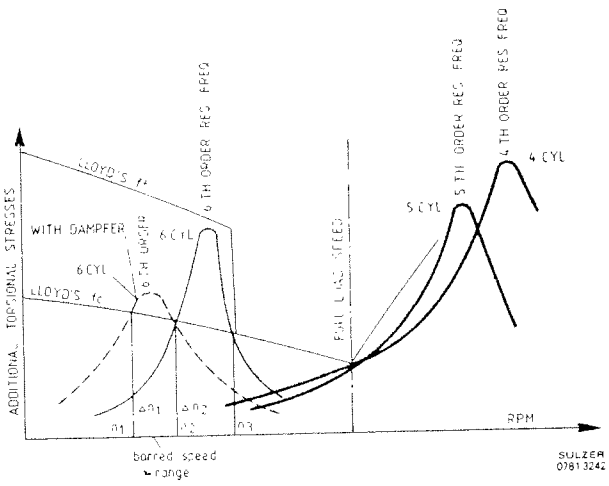


Fig. 17 – Torsional vibrations of 4-, 5- and 6-cylinder engines with short shafting : solutions with *reinforced shafts* (4; 5 cylinders) and with *damper* and *minimum diameter shafts* (6 cylinders) meet the stress requirement.

range (where continuous operation is prohibited) is common, sometimes in connection with a torsional vibration damper to further reduce the amplitude.

Fig. 17 shows both solutions. Sophisticated computer programs provide speedy and highly accurate results.

One point should be remembered about torsional vibrations: they are a design stage problem, which can be properly calculated; they do not cause hull vibration problems and may be best termed an internal shafting problem.

REFERENCES

1. Collin, L., Sakai, H., Stefenson, J., "Alignment model for shafting systems of diesel engine plants". CIMAC Conference, 1979, Vienna.

2. Lindquist, M., "Balancing the first order external moments of MAN-B&W four cylinder low speed engines". The Motor Ship, June, 1983, p 47 ff.

3. Schneider, P., "Installation aspects of the new Sulzer super-longstroke engine". Bulletin Technique du Bureau Veritas, April, 1983, p 76 ff.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

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SHIP DESIGN FOR FUEL ECONOMY

WIND AS AN AID FOR SHIP PROPULSION

by

Peter Schenzle

HAMBURGISCHE SCHIFFBAU VERSUCHSANSTALT

HAMBURG SHIP MODEL TANK

HAMBURG

1. Introduction

After at least 6000 years during which the wind systems of the oceans and of the sea shores were continually utilized as the principle source of energy for ship propulsion, commercial shipping is now completely relying on fossile fuelled engine propulsion since at least 60 years. During this period we have got used to considering the wind as a more or less serious trouble maker in ship operation acting directly by increasing the aerodynamic resistance and indirectly by the additional resistance due to the wind induced waves. Stimulated by a growing awareness of ecological problems and by the recent increase of fuel prices, the idea of wind propulsion, not only for pleasure but also for commercial shipping, became again an object of discussion and some research activities since about 10 years [1,2,3,4].

Can the technical progress of the recent 60 years in aerodynamics, structural, mechanical and control engineering lead to an ecologically and economically useful contribution of wind energy to commercial ship propulsion? And which out of the confusing variety of proposed technical concepts is the best solution for a given type of ship, route and trade?

Since commercial ships today are unthinkable without any engine propulsion for safety reasons, the question can only be as to the ratio of installation and application of motor and wind propulsion (FIG.1).

A 'Motor-Assisted Windship' would be depending on wind conditions in ship speed and routing, but has the option of saving a substantial percentage of fuel, say ~75 - 95 %.

A 'Wind-Assisted Motorship', on the other hand, can be operated, similar to a normal motorship, at largely constant service speed but at reduced engine power whenever favourable winds are contributing to the propulsive force, thus saving a minor percentage of fuel, say ~5 - 25 %.

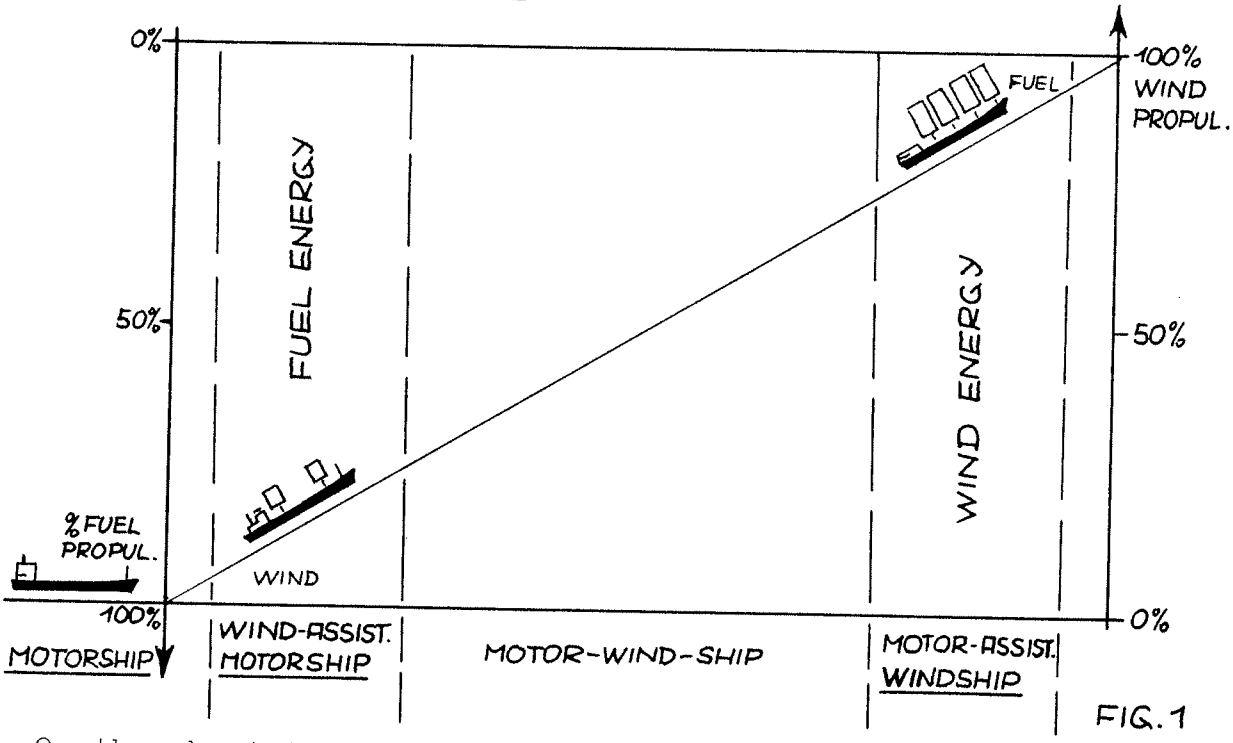


FIG. 1

On the short term the concept of wind assistance seems to be more easily acceptable to ship owners and charterers in the industrialized countries, and it is in this field that first practical experiments with systems of different types installed on normal coastal motorships are under way in Japan, U.S.A. and U.K.

2. Wind conditions over sea

One problem in assessing the potential merits of wind assistance or wind propulsion is a realistic estimation of the wind condition likely to be encountered on an intended trade route.

A simple estimate of a single average windspeed cannot account for the typical variety of relative wind conditions contributing to an average drag or driving force and power loss or gain. The reason is a highly nonlinear relation between windspeed, shipspeed and relative course on the one hand and the generated forces on the other.

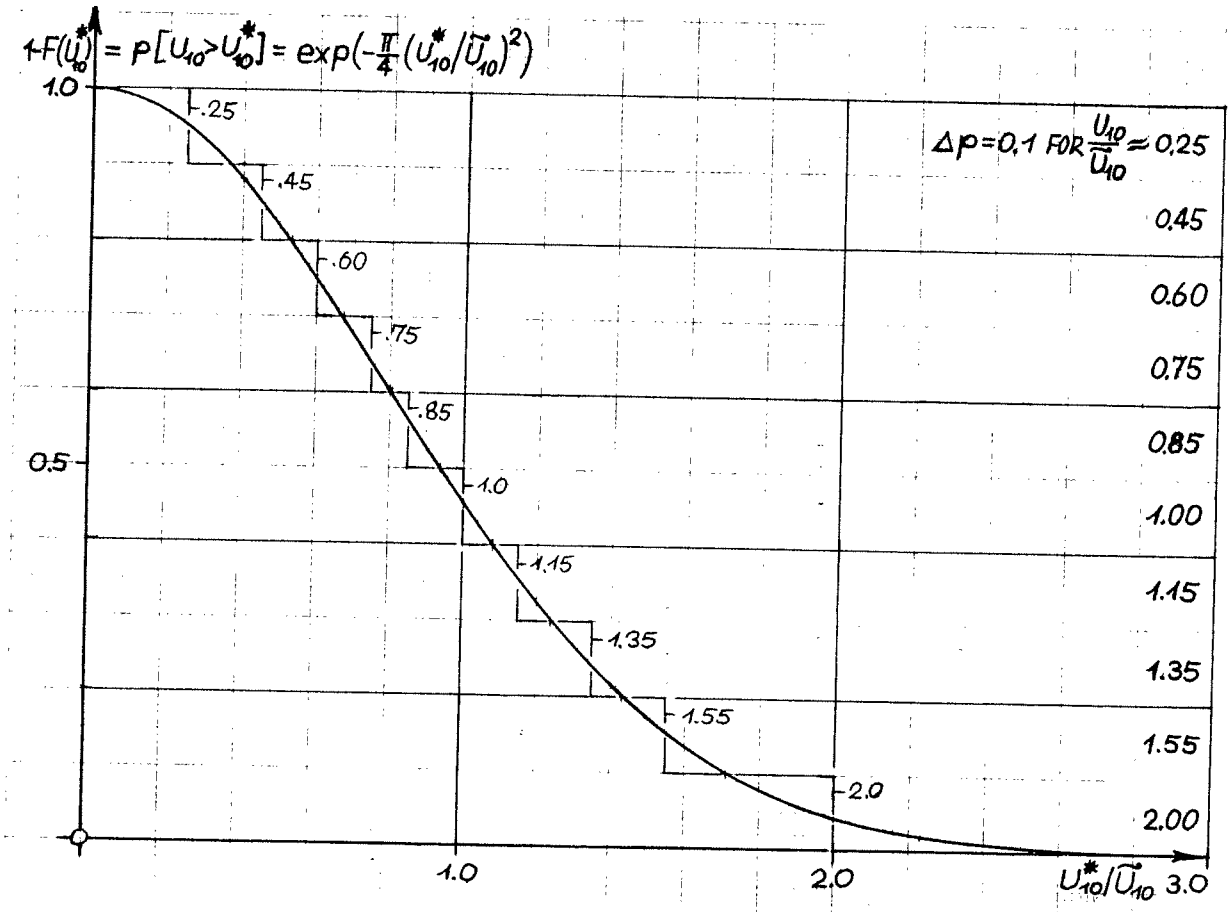
There are tropical areas with systems of light to moderate winds but seasonally steady and predictable in velocity and direction, such as Tradewinds or Monsoons. There are our temperate latitudes with strong short-term broad-band variations in wind velocity and direction, largely governed by the

passing cyclones during at least half of the year. And there are certain areas with extreme cyclones transforming into the dangerous hurricanes and typhoons, which need special attention in safety considerations for wind loaded structures.

Statistical material of wind observations in temperate latitudes of variable winds may be well represented by Weibull distributions of the wind velocity U_{10} on a standard level of $10\,m$ above the sea surface, the probability of U_{10} exceeding a limit U_{10}^* being: $p[U_{10} > U_{10}^*] = 1 - F(U_{10}^*) = e^{-\left(\frac{U_{10}^*}{U_a}\right)^k}$

Since the exponent k indicating the 'narrowness' of the distribution is in most cases close to $k = 2.0$, this special case, the 'Rayleigh distribution' may be used as a standard distribution of wind speed containing only one free parameter, the average wind speed $\tilde{U}_{10}$: which can easily and reliably be gained from observations:

$$p[U_{10} > U_{10}^*] = 1 - F(U_{10}^*) = e^{-\frac{\pi}{4}\left(\frac{U_{10}^*}{\tilde{U}_{10}}\right)^2}$$



STANDARD (RAYLEIGH) DISTRIBUTION OF WINDSPEED IN TEMPERATE LATITUDES

FIG. 2

Values of annual average windspeed $\tilde{U}_{10}$ are for:

the Caribean Sea	$\tilde{U}_{10} \approx 6-7\,m/s$
the North Sea	$7-8\,m/s$
the North Atlantic Ocean	$8-10\,m/s$

A simple discretisation of the above distribution is to assign a probability $\Delta p = 0.1$ or 10% to each of ten discrete windspeeds U_{10} given as a ratio $U_{10}/\tilde{U}_{10}$ to the average windspeed $\tilde{U}_{10}$ (Fig.2).

A simple distribution of wind direction μ which may be assumed as fairly independent of windspeed U_{10} is of simple cosine form around an average value $\tilde{\mu}$:

$$f(\mu-\tilde{\mu}) = \frac{1}{2\pi}(1+a\cdot\cos(\mu-\tilde{\mu}))$$

or as discrete probabilities Δp to be assigned to small sectors of width $\Delta\mu$:

$$\Delta p(\mu-\tilde{\mu}) = \Delta\mu\cdot f = \frac{\Delta\mu^\circ}{360^\circ}(1+a\cdot\cos(\mu-\tilde{\mu}))$$

In our latitudes the average wind direction $\tilde{\mu}$ is normally from the west and the variation parameter $a \approx 0.6$ which means a maximum density (from the west) of four times the minimum density (from the east). For round trips, consisting of two opposite courses Ψ in arbitrary absolute direction, the above type of wind direction distribution yields all relative directions of true wind to the ship's head $\chi = \mu - \Psi$ with constant probability from 0° to 360°: $f(\chi = \mu - \Psi) = \frac{1}{2\pi} = const.$

If the wind data are gained by anemometer measurements, then the reported U_{10} values are averaged over 10 minutes to eleminate fluctuations with periods $t \leq 10$ min. The maximum values during the 10 min. measurement are about 50 % higher than the average. The peak frequency $\hat{n}$ of the fluctuation spectrum in longitudinal direction is depending on average windspeed U_z and height over sea level z, for neutral condition:

$$\hat{n}_x \approx \frac{1}{20}\frac{U_z}{z} \quad or \quad \hat{t} \approx 20\frac{z}{U_z}$$

If the wind data are given in terms of visually estimated Beaufort numbers B_n according to the appearance of the sea surface, we need a reasonable conversion scale to assign an average equivalent windspeed $U_{10}(B_n)$ to each Beaufort number. There are a number of conversion scales the most common of which is the international standard scale of 1949:

$$U_{10} = 0.836\,\frac{m}{s}\cdot B_n^{3/2}$$

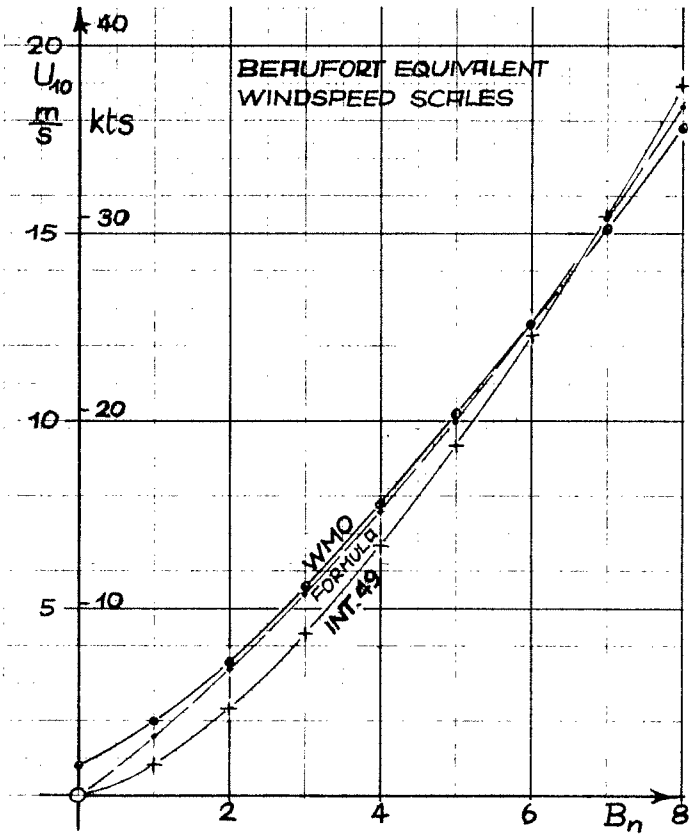
Recent comparisons of wind speed measurements with simultaneous observations of Beaufort number indicate that this simple formula leads to underestimate the windspeeds related with low observed Beaufort numbers $B_n \leq 5$. New proposals by WMO and local weather offices account for these results but are not yet standard. An old scale according to Seilkopf which was in use at the turn of the century is almost of the same quality as the new proposals and can easily be kept in mind:

$$U_{10} = 2\,\frac{m}{s}\cdot B_n \quad \Big|\; B_n \leq 5$$

$$U_{10} = 3\,\frac{m}{s}\cdot B_n - 5\,\frac{m}{s} \quad \Big|\; B_n \geq 5$$

or approximated by a single formula:

$$U_{10} = 1.5\,\frac{m}{s}\cdot B_n + 0.1\,\frac{m}{s}\cdot B_n^2 \quad .$$



B_n	U_{10} [m/s] INT. 47	WMO	SEIL-KOPF	FORMULA
1	0.84	2.0	2	1.6
2	2.36	3.6	4	3.4
3	4.34	5.6	6	5.4
4	6.69	7.8	8	7.6
5	9.35	10.2	10	10.0
6	12.29	12.6	13	12.6
7	15.48	15.1	16	15.4
8	18.92	17.8	19	18.4
9	22.57	20.8	22	21.6
10	26.43	24.2	25	25.0
11	30.50	28.0	28	28.6
12	34.75	32.3	31	32.4

FIG. 3

From the wind speed data U_{10}, valid for the standard height of 10 m above sea level, we must derive an effective or local windspeed U from the wind profile in the athmospheric boundary layer. For neutral thermal conditions of the athmosphere the wind profile is of the simple logarithmic form [5]:

$$\frac{U_z}{U_{10}} = \frac{\log \frac{z}{z_0}}{\log \frac{10m}{z_0}} \quad ,$$

where z_0 is a surface roughness parameter depending only on the small ribbles of the sea surface and therefore practically constant $(z_0 \approx 10^{-4})$ for all stronger winds:

$$\frac{U_z}{U_{10}} = \frac{5 + \log_{10}\left(\frac{z}{10m}\right)}{5} = 1 + 0.2 \log_{10}\left(\frac{z}{10m}\right) .$$

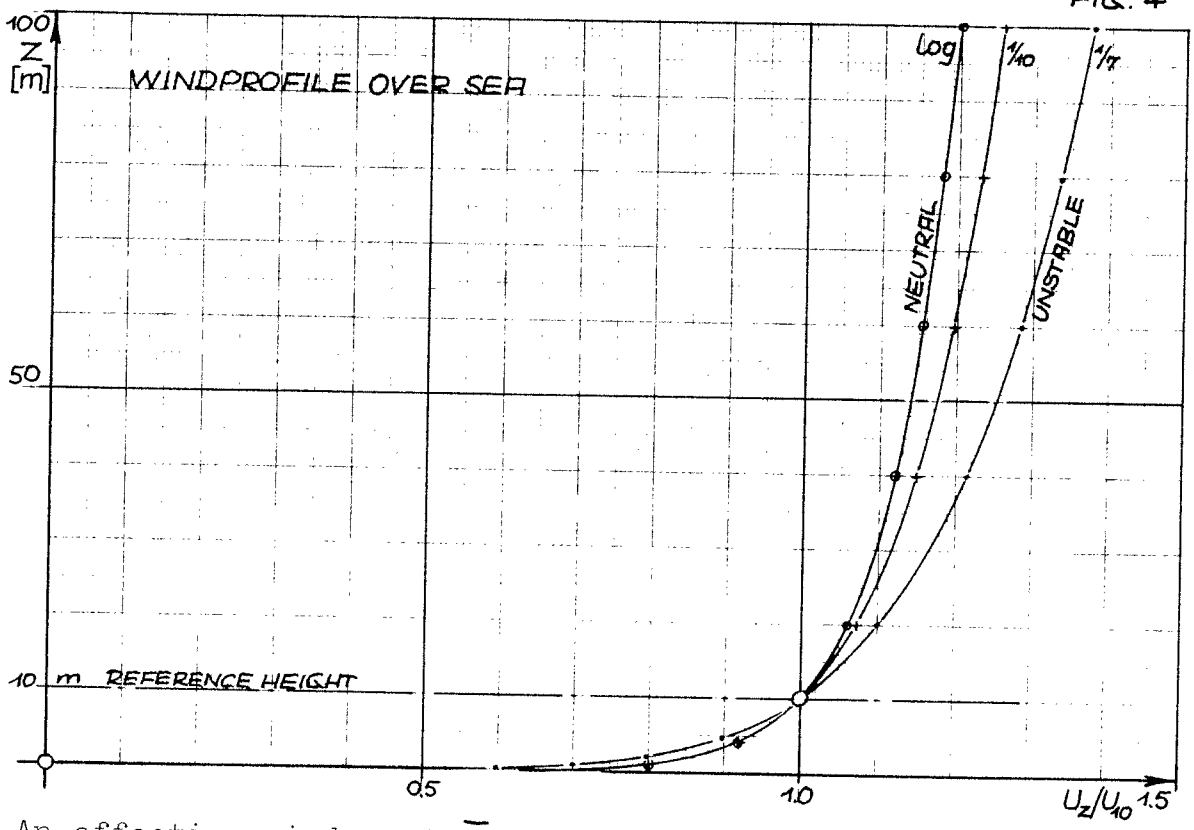
This can be approximated by a simple power formula:

$$\frac{U_z}{U_{10}} \approx \left(\frac{z}{10m}\right)^{1/10}$$

with less than 1 % error between $1m < z < 10m$

and less than 5 % error between $0.1m < z < 100m$.

FIG. 4



An effective windspeed $\overline{U}_E$ for estimating windforces on a ship's hull can be gained by averaging the dynamic pressure over the average height z_E of the hull over water:

$$\overline{U}_E^2 = \frac{U_{10}^2}{z_E} \int_0^{z_E} \left(\frac{z}{10m}\right)^{2/10} dz = \frac{U_{10}^2}{1.2}\left(\frac{z_E}{10m}\right)^2 = \frac{U_{z_E}^2}{1.2}$$

$$\overline{U}_E \approx 0.91 \cdot U_{z_E} .$$

The effective windspeed $\overline{U}_E$ over a height of z_E is 91 % of the local windspeed U_{z_E} on the level of the upper edge of the body. For objects over deck, such as sails a local value $U_{z_{CENTRE}}$ on the level of the centre of the object is a sufficient approximation for $\overline{U}_E$.

If, under light-wind condition $(B_n < 5)$ the air stratification on sea may be labile, the profile will be less full and may be better approximated by a power between 1/8 and 1/6, yielding a stronger decrease below 10 m and a stronger increase above 10 m.

3. Wind forces on ships

Wind forces on overwater part of nearly all imaginable ship-types have been measured in windtunnel tests, and many publications give data for particular configurations [9 - 12]. Some authors, especially Isherwood [8] have published systematic evaluations of wind force data allowing the estimation of forces on many configurations of hull, superstructure and loading gear according to characteristic form parameters.

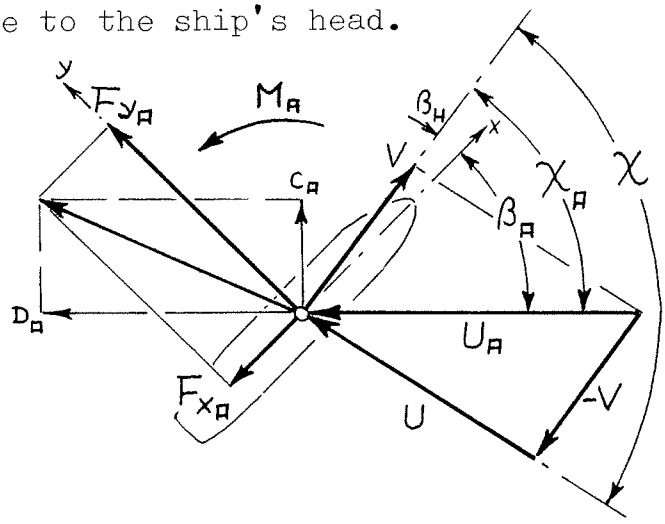
A favourable presentation of such data is in the form of coefficients of longitudinal force C_x refering to projected transverse area A_T and of side force C_y and yawing moment C_M refering to projected lateral area A_L :

$$C_{x_{AT}} = \frac{F_{x_A}}{\frac{\varrho}{2} U_A^2 A_T}$$

$$C_{y_{AL}} = \frac{F_{y_A}}{\frac{\varrho}{2} U_A^2 A_L}$$

$$C_{M_{AL}} = \frac{M_A}{\frac{\varrho}{2} U_A^2 A_L \cdot L}$$

as functions of the angle of inflow β_A of the apparent wind relative to the ship's head.



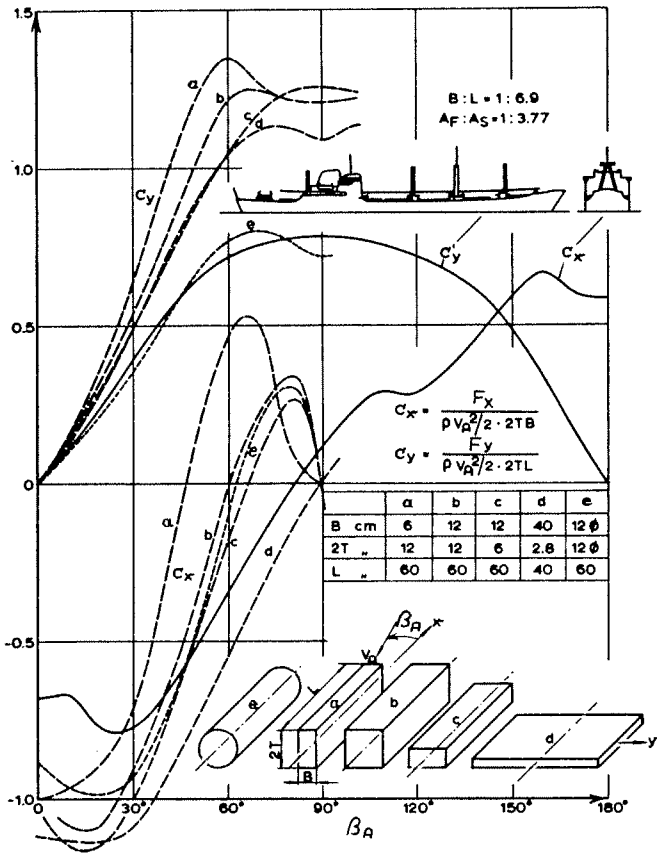
If the true windspeed is U at an angle χ against the ship's course at shipspeed V , the apparent windspeed U_A and direction χ_A against the ship's course are:

$$U_A^2 = (V + U\cos\chi)^2 + (U\sin\chi)^2$$

$$\tan\chi_A = \frac{U\sin\chi}{V + U\cos\chi} \quad , \quad \beta_A = \chi_A - \beta_H \quad ,$$

the angle of inflow of the apparent wind relative to the ship's head β_A is reduced by the drift angle β_H which is often negligible, especially when the shipspeed V is not too low.

A typical example of windforce coefficients for a general cargo ship is presented together with data for simple prismatic bodies in Fig. 5 according to Wieghardt [5] .



	a	b	c	d	e
B cm	6	12	12	40	12 ⌀
2T "	12	12	6	2.8	12 ⌀
L "	60	60	60	40	60

Fig. 5 Coefficients of the longitudinal and transverse wind force on the superstructure of a cargo ship model versus wind direction according to [5] compared with those of prisms without a center plane simulating the water surface.

The variation of the coefficients with the apparent wind angle β_A can be expressed as Fourier series, but if we are only interested in the order of magnitude and the principal trend, it is sufficient to adopt the principal terms.
Average values for Tankers and Bulkers are:

$$C_{x_{AT}} \approx -0.8 \cdot \cos(\beta_A)$$
$$C_{y_{AL}} \approx 1.0 \cdot \left(1 - \frac{L^2}{100 A_L}\right) \cdot \sin(\beta_A) \quad ,$$

and for Containerships:

$$C_{x_{AT}} \approx -0.6 \cdot \cos(\beta_A)$$
$$C_{y_{AL}} \approx 1.1 \cdot \left(1 - \frac{L^2}{100 A_L}\right) \cdot \sin(\beta_A) \quad .$$

A tentative formula for a more detailed representation of the typical trend of the longitudinal force component for many container- and cargoships is

$$C_{x_{AT}} \approx -0.1 - 0.6\cos(\beta_A) - 0.15\cos(3\beta_A) + 0.2\cos(5\beta_A)\,.$$

If we are talking about wind resistance of ships, we have to be aware of three components, the direct longitudinal aerodynamic force on the above water part $-F_{x_A}$ and the additional hydrodynamic resistance on the underwater hull due to the wind-induced sideforce F_{y_A} and on the rudder to ballance the yawing moment M_A (see Appendix):

$$R_{A\,WIND} = -F_{x_A} + \frac{F_{y_A}^2}{\rho_H V^2 A_{C_H}}\left(\frac{k}{2}-1\right) + \frac{|M_A|}{L/2}\left(\frac{D_i}{C}\right)_{RUDDER}\,,$$

but in most practical cases the indirect (hydrodynamical) components may be neglected relative to the direct wind force. The relative order of magnitude of the direct windresistance $-F_{x_A}$ to the basic hydrodynamic ship resistance at low Froude numbers is:

$$\frac{-F_{x_A}}{R_{T_H}} = \frac{-\rho_A \cdot U_A^2 \cdot A_T \cdot C_{x_A}}{\rho_H \cdot V^2 \cdot S_W \;\; C_{T_H}}$$

$$\approx \frac{1}{800}\cdot\left(\frac{U_A}{V}\right)^2 \frac{B^2}{1.25BL}\cdot\frac{.6}{.003} \approx 0.2\,\frac{B}{L}\left(\frac{U_A}{V}\right)^2 \approx 0.03\left(\frac{U_A}{V}\right)^2$$

That means in still air $(U_A/V=1)$ an airdrag in the order of 3 % of the basic ship resistance. For headwinds the factor $(U_A/V)^2 = (1+U/V)^2$ is increasing rapidly. For a fresh headwind of twice the shipspeed the factor is 9 yielding a resistance increase in the order of 30 %.

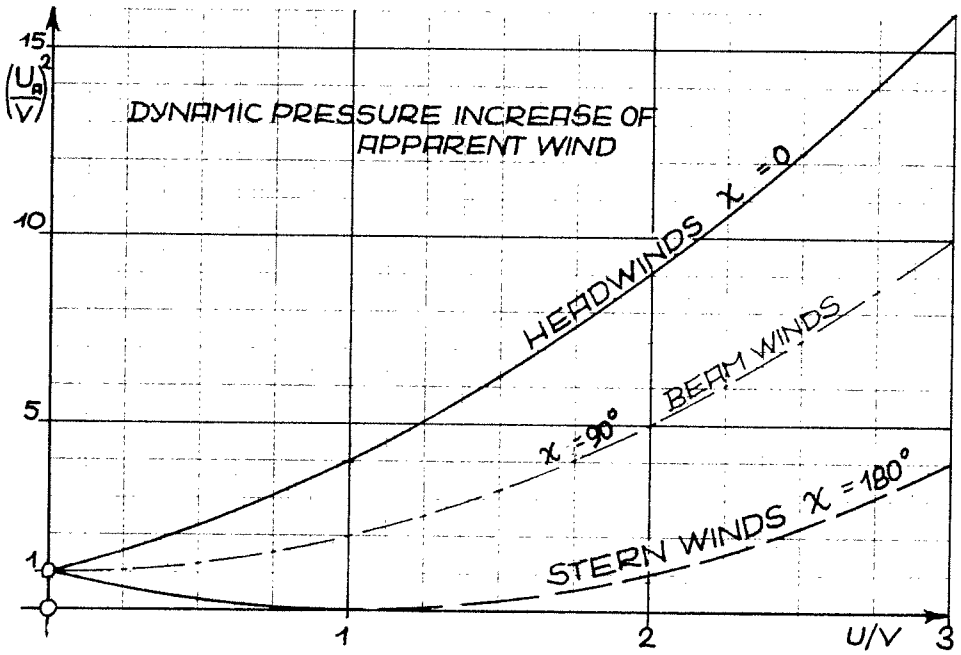


FIG. 6

It is obvious that, thinking about fuel saving in connection with wind, it might be paying if we could reduce the windage of the superstructure an loading gear by simple measures.

Prismatic deckhouses show the highest drag coefficients of about 1.2 if they are in the shape of 'transverse walls' (wide and high but short). The same applies to transverse arrangements of deck containers or to elevated RO-RO ramps. In addition, on very high structures where the increase of windspeed and dynamic pressure in the wind profile over sea becomes effective, of course, reduction of height is more effective than reduction of width. But narrow elongated houses would not only have less transverse area A_T but also lower drag coefficients of about 0.8 , e.g. if it would be possible to change the position of a deckhouse with $b/L = 2$ from transverse to longitudinal, the drag would be reduced by a ratio of:

$$\frac{D_L}{D_T} = \frac{C_{DL}}{C_{DT}} \frac{L}{b} = \frac{0.8}{1.2} \cdot \frac{1}{2} = \frac{1}{3}$$

In addition rounded front corners of prismatic bodies with radius in the order of $0.1 \cdot b$ may reduce the drag by up to 50 %.

Circular masts show high drag, especially in the sub-critical speed range $U_a \lesssim 4 \frac{m^2}{s}/d$ the drag coefficient is $C_D = 1.1$. Here elongated oval or U-shapes show much lower longitudinal forces and most effective are elliptic or parabolic 'noses' which may produce even a forward driving force in a certain sector of bow wind inflow.

Whether such measures might be economically paying should be decided not on the basis of calm weather'air drag' but on the basis of a statistical estimation of average wind resistance taking into account the expected frequencies of windspeeds and relative courses under realistic service assumptions.

4. Traditional and novel Wind propulsion systems

The spectrum of possible and proposed technical systems for the utilization of wind energy for ship propulsion is of remarkable but sometimes confusing width. There are mainly nautical arguments for a revival of traditional gaff- and square rigs to avoid technical risk and to adopt a reliable and proven system with marginal improvements in materials and mechanized winches [13] despite all the shortcommings in efficiency and in manning requirement, suitable to the social and economic conditions of the 19 th century.

There is a great variety of suggestions for consequent developments of the traditional rigs of the sailing cargo ships according to the technical development since the turn of the century. On this line are the proposals for a stepwise mechanisation and aerodynamic streamlining of the square rig [14] via the curved yards and the roller-furling of the 'Dyna Ship' [15] to the symmetrically cambered rigid panels, which are tested in Japan as wind assistance for motor ships [16]. Similar proposals for the developement of the fore and aft rig lead via improved and mechanized variants of the rigs of our modern sailing yachts to the soft, cambered airfoil of the 'Princeton Sailwing' and eventually to the 'Wingsail', a rigid airfoil with trailing edge flaps [17].

Typical development steps are:

- aerodynamic shaping of soft sails and their leading edges,
- mechanized (roller-) reefing and sail trimming,
- self supporting rotatable masts,
- adopting of rigid airfoil wings with flaps.

The superiority of the rigid airfoil wingsail over a well-shaped soft sail is restricted to the extremely close-hauled range of courses. Such a rig is especially suitable for light-weight, high speed craft or as wind assistance for motor ships where the engine propulsion provides a relatively high shipspeed with respect to most of the wind conditions.

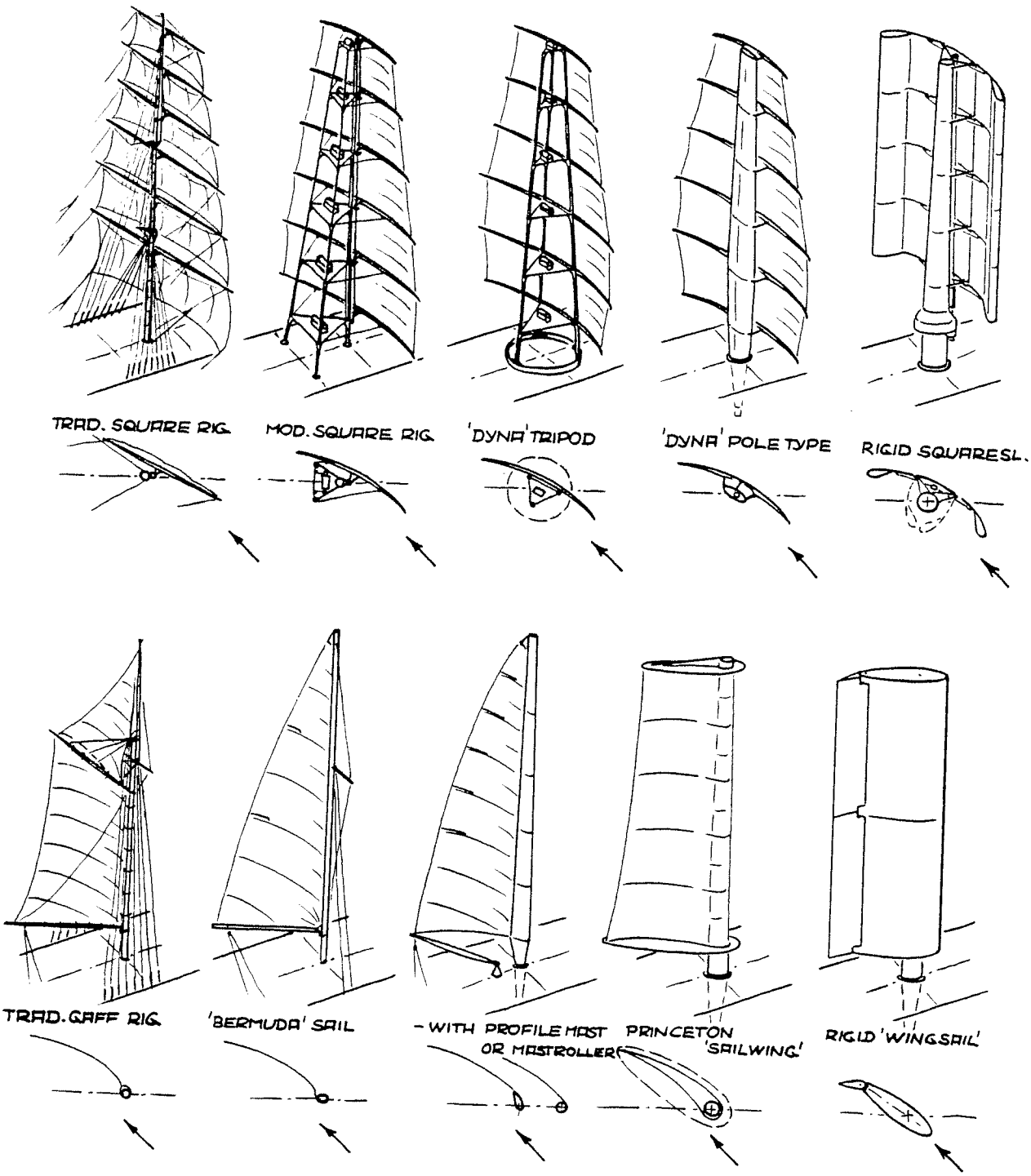
But reefing of rigid wings is impossible or extremely difficult and the integration of loading gear into such a rig may cause serious problems. Self-supporting cantilever masts tend to high structural weight and costs and the manifold subdivision of the sail area of square rigs seems worth wile only in case of very large units.

Novel wind propulsion systems such as Flettner-Rotors, horizontal - or vertical axis wind turbines or kites show some interesting performances characteristics, especially in view of applications to wind assistance, but a comparative valuation requires further research and development effort.

Traditional square rigs are efficient on a broad reaching course but handicapped in the close-hauled condition due to low effective aspect ratio and restricted yard angle. Modern proposals avoid these shortcommings.

Fore- and -aft rigs are more efficient close to the wind but restricted in boom angle on the broad reach. A critical problem is the mast in front of the sensitive leading edge of the mainsails. The tendency of flow separation is reduced by a flow stabilizing effect of the jib, acting as a leading edge slat. Further improvements could be a favourable (parabolic) shape of the mast cross section, an adjustable profile mast or a rotatable mast (mastroller) where the sail is always adjusted smoothly to the leeward side respectively, thus reducing the parasitic drag and increasing maximum lift considerably.
The Ljungström Rig has a double sailcloth on both sides of a rotatable mast, and similarly the 'Princeton Sailwing' uses a double cloth slung loosely around the mast and held by a tensioned leech rope between cantilever boom and gaff, both forming an effective cambered airfoil with round nose, showing high maximum crossforce and low parasitic drag.

Rigid airfoil 'wingsails' show extremly low parasitic drag but maximum crossforce is limited since only symmetrical (uncambered) profiles can be used on both tacks. By the use of trailing edge flaps maximum lift is greatly improved. The



POSSIBLE DEVELOPMENTS OF TRADITIONAL WIND PROPULSION SYSTEMS

FIG. 7

lacking possibility of reefing can be replaced by feathering the wing to the position of the very low minimum drag, but it must be made sure that this is always possible and that dynamic instability is prevented.

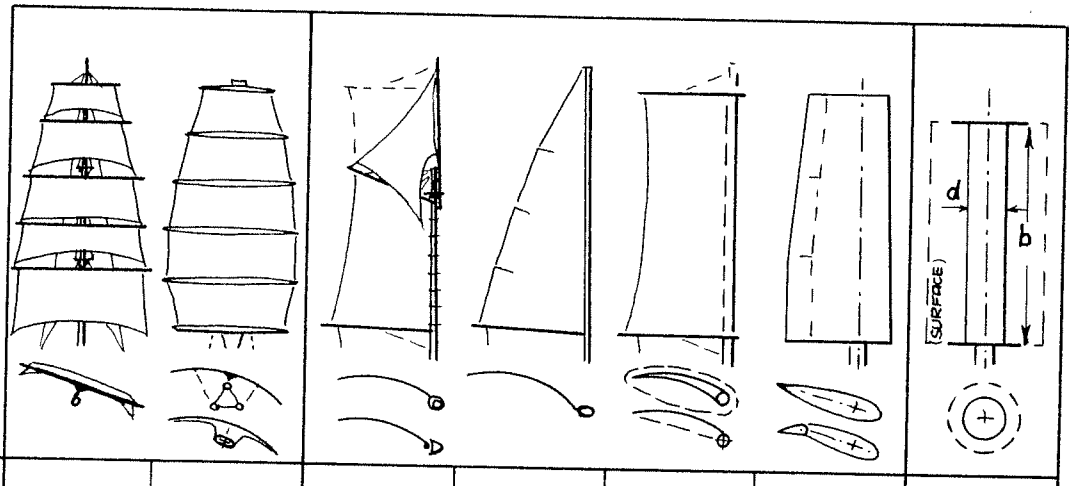
The 'Flettner Rotor' is a very interesting servo mechanism to generate very high lift with a relatively small device[18]. But the ultra-high maximum lift coefficient in the order of 10 is partly due to the arbitrary choice of the reference area, which is virtually only the projection of the (somewhat clumsy) mast, while the sail is the rotation. The minimum drag coefficient $C_{D_0} \approx 0.5$ however, is fairly large when refering also to mast projection. A tentative suggestion for a more suitable reference area may be the surface of the cylinder, which may be a better index for weight and cost in comparison with a sail rig, thus reducing the coefficients by a factor $1/\pi$. The coefficients are still high, however, and a rotor may be smaller than a comparable sail or wing, but in order to achieve a comparable close-hauled performance the capture area or the effective span must be the same. If for wind assistance a rotor is considered as an alternative to a sail or wing, its height may be only 1/2 diameter lower (thanks the end disks) than the effective height of the sail or wing. Finally we should not forget the requirement of servo power for driving the rotor which may be, very roughly, in the order of 20 % of the reduction of propulsive power. An interesting variante is the 'blown rotor' of Wolfson Unit Southampton, which is not rotating but generating circulation by blowing compressed air through tangential slots, with the advantage of mechanical simplicity and easy control at less weight and cost.

With modern wind turbines or 'windmills' extracting energy from the wind and transfering it to an ordinary ship propeller it is, in principle, possible to 'sail' directly into the eye of the wind without tacking. Since the energy extract is directly related to high reaction-(drag-)forces on the turbine, the propeller is in a heavy load condition and eventually, in stronger winds and increasing resistance due to the wind-induced waves, this mode of operation will be prevented by decreasing efficiency of the whole system and limitations in absolute power level.

Special attention must be paid to the structural dynamics of a large turbine mounted on a floating platform moving in six degrees of freedom.

The aerodynamic performance of alternative systems is described by the relation between drag and crossforce (or lift) as plotted in the lift-versus-drag polar diagram and characterized by the four parameters (see appendix):

minimum drag coeff.	C_{D_0}	(parasitic drag) ,
maximum drag coeff.	C_{DM}	(separated flow),
maximum lift coeff.	C_{CM}	(crossforce before stall),
effective aspect ratio	AR_E	(capture area).



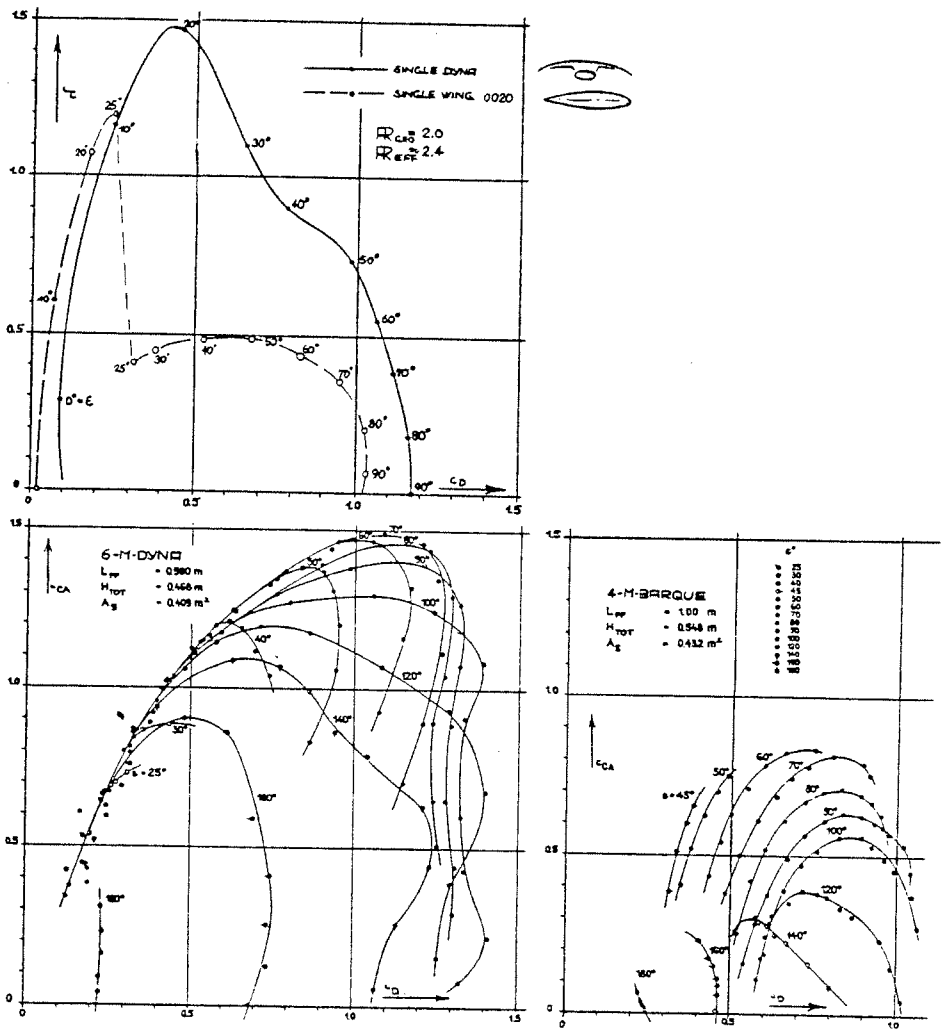
RIG TYPE	TRADIT. SQUARE RIG	MODERN SQUARE (DYNA-)	TRADIT. GAFF R. (MOD. G.)	BERMUDA RIG	SAILWING (MASTROLLER)	WINGSAIL (T.E.FLAP)	FLETTNER ROTOR (/SURFACE)
C_{D_0}	~0.13	0.10 (0.07)	0.10 (0.07)	~0.08	0.05 (0.06)	~0.02	0.50 (0.16)
C_{DM}	~1.0	1.2	1.0 (1.1)	~1.2	1.2	1.0	~5 (1.6)
C_{CM}	~0.9	1.5	1.1 (1.2-1.5)	~1.0-1.3	1.7 (1.6)	1.1-1.4 (~2.0)	~10 (3.2)
AR_E	~0.5 $\approx AR_{1\,SAIL}$	~2.0 $\approx AR_{GEO}$	~1.6 (2-3) $\approx AR_{1\,SAIL}$	2-3 $\approx 2/3\, AR_{GEO}$	2-3 $\approx AR_{GEO}$	2-3 $\approx AR_{GEO}$	~6 (2) $\approx \frac{b+d/2}{d(\cdot\pi)}$

TYPICAL AERODYNAMIC PARAMETERS OF SOME BASIC RIG TYPES FIG.8

If a single device (e.g. sail rig or wing) can be adjusted as a whole according to the inflow the relation between drag and crossforce depending on a single parameter, the angle of attack β relative to the chord of the wing or sail, is adequately described by a single C_C-C_D-polar diagram.

If, however, the relative position of several elements of a rig is depending on their mounting on deck of the ship the aerodynamic interaction will be depending on an additional parameter, the inflow relative to the ship's axis β_A . The performance of such a system is described by a series of C_C-C_D-polar diagrams, the parameters of which are depending on the apparent wind angle β_A relative to the ship's head.

The reason may be the variable interaction between mast and sail, between fore and mainsail, between the sails on different masts or between neighbouring wings or rotors....

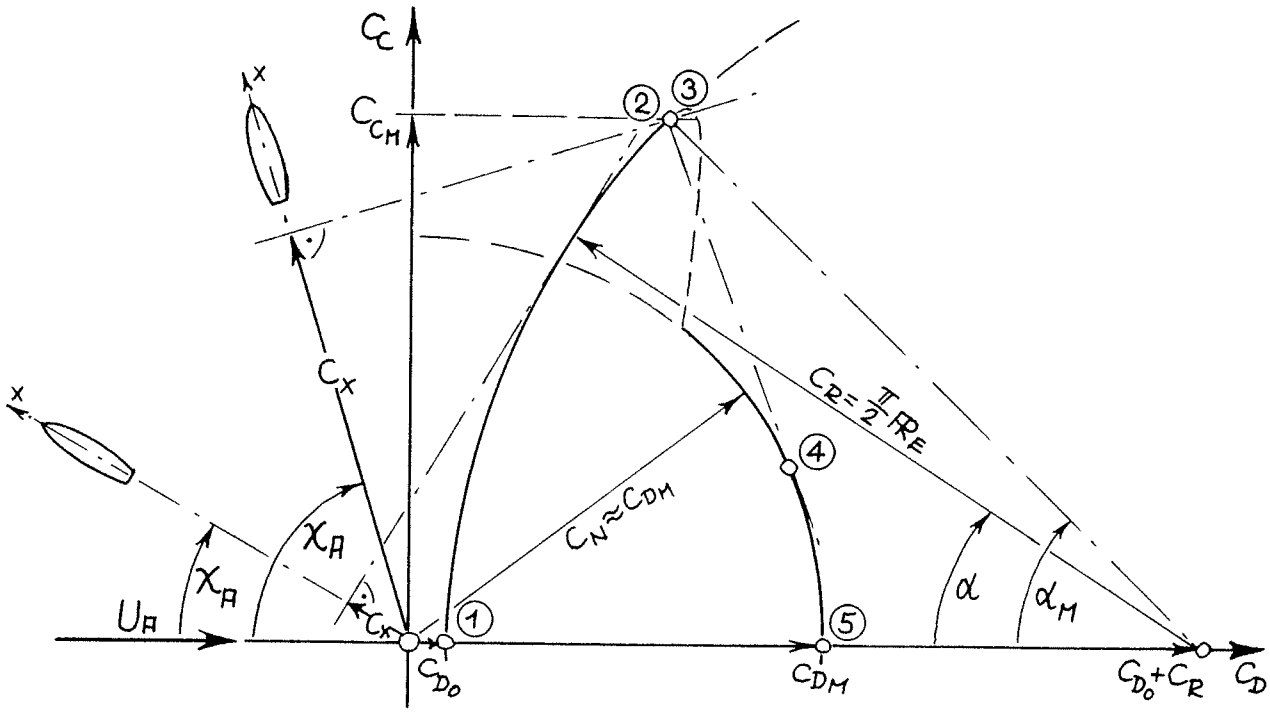


WINDTUNNEL RESULTS:
WIND FORCE COEFFICIENTS FOR SINGLE SAILS AND COMPLETE SHIP MODELS [19]

FIG. 9

5. Wind assistance for motor ships

For wind assistance, however, there will be normally enough space on deck to install sails, wings or rotors with some distance between them, and the performance will be sufficiently described by a single C_C-C_D-polar diagram (see appendix):



If we consider only wind assistance for a motorship at constant 'full' shipspeed the driving force ΔF_x from the wind will be most of the time only a smaller percentage of the total thrust required to maintain the constant speed, the propeller and engine will be discharged accordingly, and the accompanying crossforce will be easily ballanced by the hull at full speed at a negligible drift angle β_H. Under these assumptions the effective driving force is directly the longitudinal component of the resulting wind force on the wind assistance system [20]. The maximum possible driving force for each angle χ_A of apparent inflow is given by the distance from the origin of a tangent line to the C_C-C_D-polar diagram at right angle to the desired relative course $\chi_A \approx \beta_A$, with $C_R = \frac{\pi}{2} AR_E = 2A_C/A_S$

and $\sin \alpha_M = C_{C_M}/C_R$.

The reference area A_S is the sail- or wing-area.

The maximum driving force C_x is then according to the range of inflow:

close hauled with less than max. crossforce ①-②:

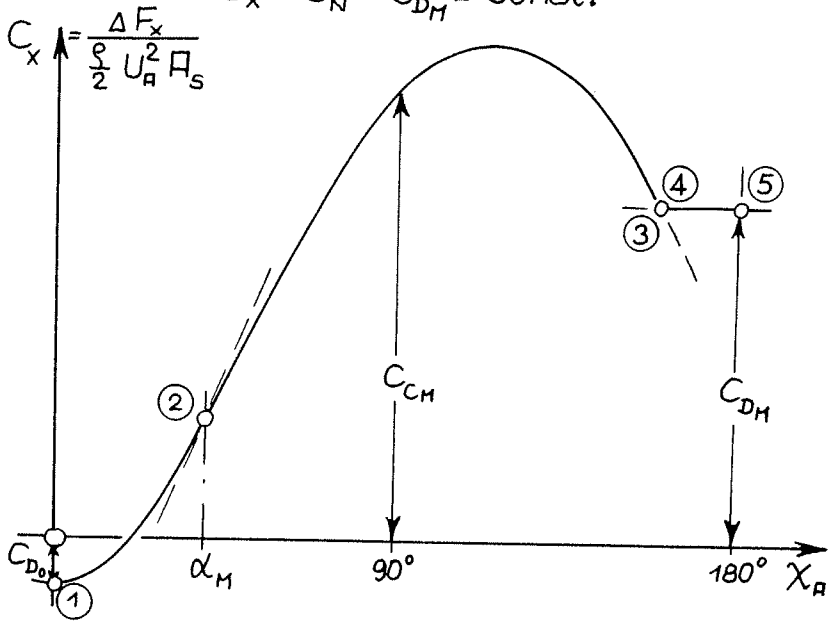
$$C_x = C_R - (C_{D_0} + C_R)\cos\chi_A \quad | \chi_A < \alpha_M \quad ,$$

reaching with max. crossforce C_{CM} , just before stall ②-③:

$$C_x = C_{CM}\sin\chi_A - (C_{D_0} + C_R(1-\cos\alpha_M))\cdot\cos\chi_A \quad | \chi_A \geq \alpha_M$$

and running downwind with pure normal force ④-⑤ , if this yields higher driving force than above:

$$C_x = C_N \approx C_{D_M} \approx const.$$



By this procedure the maximum achievable driving force coefficient C_x as a function of the angle of apparent inflow is derived from the C_C-C_D polar diagram as a function of the four parameters characterising the aerodynamic performance of the wind propulsion system, sail, wing or rotor. It may be noted that the driving force, in a remarkably wide range of reaching courses ②-③, is completely determined by the single point of maximum lift (or crossforce) and the corresponding drag just before stall.

From this driving force coefficient $C_x(\chi_A)$ we can calculate the driving force ΔF_x generated by the wind assistance for every apparent wind speed U_A and direction χ_A . With shipspeed V we

can calculate the effective power $\Delta P_E = \Delta F_x \cdot V$ and with some knowledge on the trend of propulsive efficiency we can even estimate the expected reduction of delivered power to the propeller

$$\Delta P_D = \frac{\Delta P_D}{\Delta P_E} \cdot \Delta F_x \cdot V$$

for each case of apparent wind. For a comparative valuation of alternative wind assistance systems, however, we would like to know how likely each of these apparent wind conditions is to be encountered by the ship. This probability can be estimated for a given shipspeed according to statistical distributions of the true windspeed U and the course χ relative to the true wind. On the basis of the above suggested standard (Rayleigh) distribution of windspeed U with parameter average windspeed $\tilde{U}$ and the assumption of all relative courses χ to the true wind to be equally likely to be sailed (as e.g. on round trips) we can calculate a coefficient $\tilde{C}_{x_{\tilde{U}}}$ of average driving force $\Delta\tilde{F}_x$ refering to the average true windspeed $\tilde{U}$ and depending only on the ratio of constant service speed V to the average true windspeed to be expected in the operation area:

$$C_x(\chi_A) \equiv \frac{\Delta F_x}{\frac{\rho}{2} U_A^2 A_S} \rightsquigarrow C_{x_{\tilde{U}}}\left(\frac{V}{\tilde{U}}\right) \equiv \frac{\Delta \tilde{F}_x}{\frac{\rho}{2} \tilde{U}^2 A_S}$$

$$\tilde{C}_{x_{\tilde{U}}}\left(\frac{V}{\tilde{U}}\right) = \sum_{ij} C_x\left(\chi_A\left(\frac{V}{U}, \chi\right)\right) \cdot \left(\frac{U_A}{V}\right)^2 \left(\frac{V}{U}, \chi\right) \cdot \left(\frac{V}{U_A}\right)^2 \cdot \Delta p\left(\frac{U}{\tilde{U}}\right)_i \cdot \Delta p(\chi)_j$$

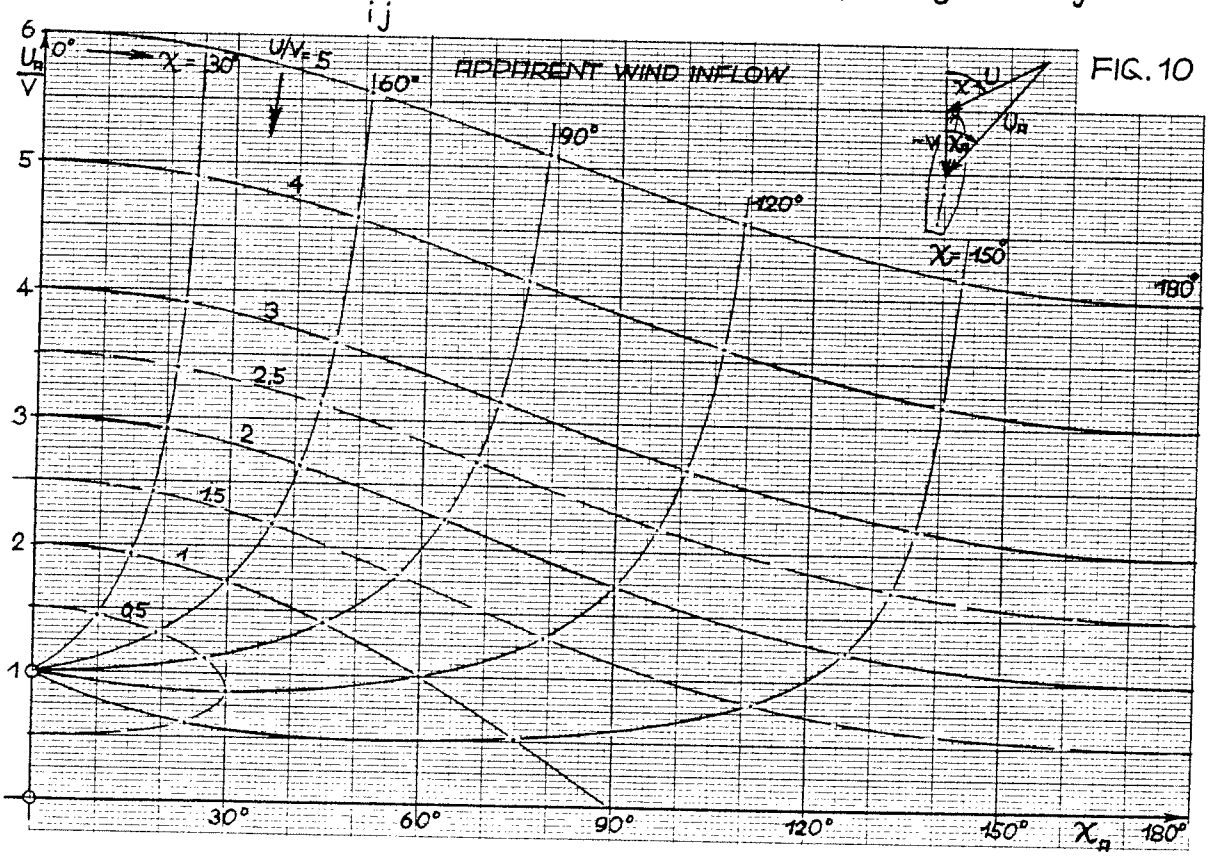
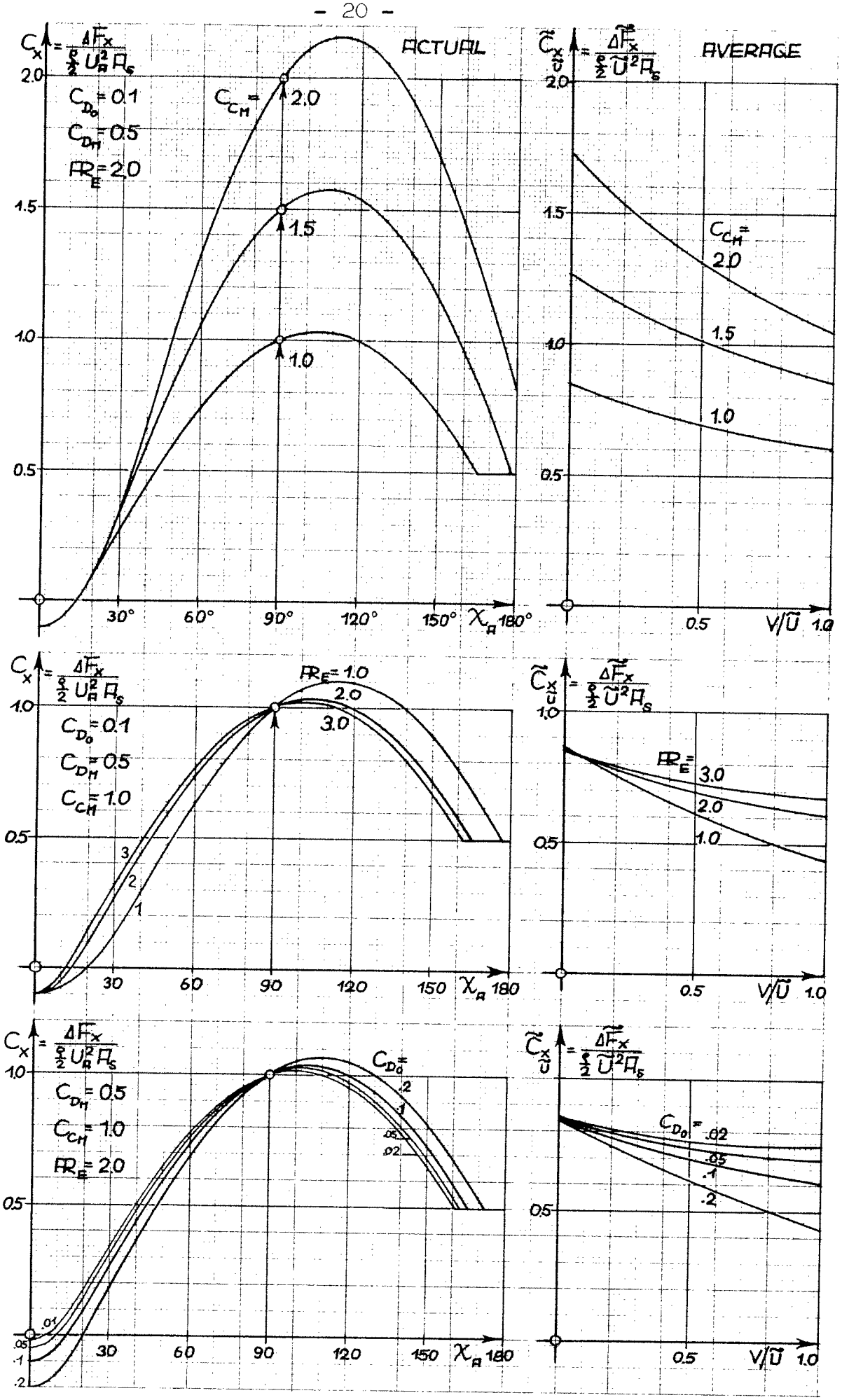


FIG. 10



ACTUAL AND AVERAGE DRIVING FORCE FROM WIND ASSISTANCE FIG.11

This calculation of average driving force coefficient has been performed for a standard set of aerodynamic parameters:

$$C_{D_0} = 0.1$$
$$C_{D_M} = 0.5$$
$$C_{C_M} = 1.0$$
$$AR_E = 2.0$$

and, as a sensitivity study, for variations of one parameter respectively, for: $C_{D_0} = .02, .05, (.10), .20;$

for: $C_{C_M} = (1.0), 1.5, 2.0$

and for: $AR_E = 1.0, (2.0), 3.0.$

The resulting driving force coefficient C_x versus apparent wind angle χ_A and the expected average driving force coefficient $\tilde{C}_{\tilde{x}_{\tilde{U}}}$ versus speed ratio $V/\tilde{U}$ are presented in Fig. 11 . This parametric calculation of average driving force is valid under the assumption that drift angle and resistance increase due to drift are negligible, and that neither reefing is necessary nor the constant service speed is ever exceeded under full wind power.

The expected average driving force coefficient $\tilde{C}_{\tilde{x}_{\tilde{U}}}$ may be well approximated within the above range of parameters by the following formula:

$$\tilde{C}_{\tilde{x}_{\tilde{U}}} \equiv \frac{\Delta\tilde{F}_x}{\frac{\rho}{2}\tilde{U}^2 A_S} \approx C_{C_M}\left(0.85 - \left(1.9\frac{C_{D_0}}{C_{C_M}} + 1.13\frac{C_{C_M}}{\pi AR_E}\right)\frac{V}{\tilde{U}} + 0.13\left(\frac{V}{\tilde{U}}\right)^2\right) \qquad 0<\frac{V}{\tilde{U}}<1$$

$$= 0.85\,C_{C_M} - \left(1.9\,C_{D_0} + 1.13\frac{C_{C_M}^2}{\pi AR_E}\right)\frac{V}{\tilde{U}} + 0.13\,C_{C_M}\left(\frac{V}{\tilde{U}}\right)^2 \qquad \frac{1}{3}<\frac{C_{C_M}}{AR_E}<1 \quad 0<\frac{C_{D_0}}{C_{C_M}}<\frac{1}{5}$$

$$\Delta\tilde{F}_x \approx \frac{\rho}{2}A_S\left(0.85\,C_{C_M}\tilde{U}^2 - \left(1.9\,C_{D_0} + 1.13\frac{C_{C_M}^2}{\pi AR_E}\right)\tilde{U}V + 0.13\,C_{C_M}V^2\right) \qquad 0<\frac{C_{D_0}}{AR_E}<\frac{1}{10}$$

The first term shows that the average driving force at zero shipspeed (bollard pull) is practically proportional to the maximum crossforce and independent of parasitic drag and aspect ratio. The second term shows a reduction in average driving force proportional to shipspeed and due to parasitic and induced drag becoming more important when 'sailing' close to the apparent wind. And the third term shows a quadraticly increasing benefit from shipspeed due to increase of apparent windspeed in bow winds. The average effective power expected

from wind assistance is simply the expected average driving force multiplied by the (constant) shipspeed:

$$\tilde{C}_{P_{\tilde{U}}^{E}} \equiv \frac{\Delta \tilde{P}_E}{\frac{\rho}{2}\tilde{U}^3 A_S} = \frac{\Delta \tilde{F}_x \cdot V}{\frac{\rho}{2}\tilde{U}^3 A_S} = \tilde{C}_{X_{\tilde{U}}} \cdot \frac{V}{\tilde{U}}$$

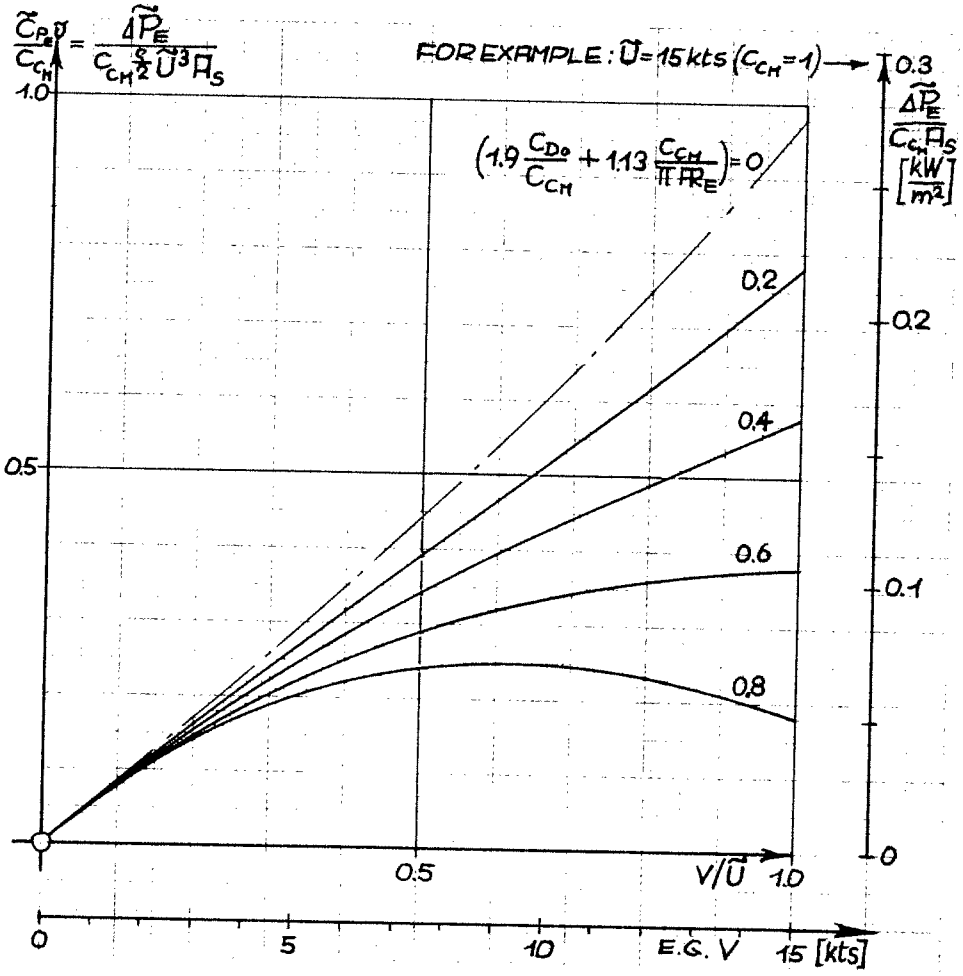
$$\approx 0.85\, C_{CM}\frac{V}{\tilde{U}} - \left(1.9 C_{D_0} + 1.13\,\frac{C_{CM}^2}{\pi AR_E}\right)\left(\frac{V}{\tilde{U}}\right)^2 + 0.13\, C_{CM}\left(\frac{V}{\tilde{U}}\right)^3$$

$$\Delta \tilde{P}_E \equiv \Delta \tilde{F}_x \cdot V \approx \frac{\rho}{2} A_S\left(0.85\, C_{CM}\tilde{U}^2 V - \left(1.9 C_{D_0} + 1.13\frac{C_{CM}^2}{\pi AR_E}\right)\tilde{U}V^2 + 0.13\, C_{CM} V^3\right)$$

If the effective power coefficient $\tilde{C}_{P_{\tilde{U}}^{E}}$ is normalized with the maximum crossforce coefficient C_{CM} it can be plotted versus speed ratio $V/\tilde{U}$ and with the drag-term $\left(1.9\frac{C_{D_0}}{C_{CM}} + 1.13\frac{C_{CM}}{\pi AR_E}\right)$ as a parameter (see Fig. 12):

$$\tilde{C}_{P_{\tilde{U}}^{E}}/C_{CM} \equiv \frac{\Delta \tilde{P}_E}{C_{CM}\frac{\rho}{2}\tilde{U}^3 A_S} \approx 0.85\frac{V}{\tilde{U}} + 0.13\left(\frac{V}{\tilde{U}}\right)^3 - \left(1.9\frac{C_{D_0}}{C_{CM}} + 1.13\frac{C_{CM}}{\pi AR_E}\right)\left(\frac{V}{\tilde{U}}\right)^2$$

As an example, additional scales are plotted to read average effective power in KW/m^2 for an average effective windspeed of $\tilde{U}_E = 15\,kts$ (≙ North Sea).



AVERAGE EFFECTIVE POWER FROM WIND ASSISTANCE FIG. 12

On an analogous way as the above estimation of the expected average effective power gain from wind assistance we can estimate the average effective power loss due to wind forces on the ship (see chapter 3). For a longitudinal aerodynamic force coefficient:

$$C_{x_{A_T}} \equiv \frac{F_{xA}}{\frac{\rho}{2} U_A^2 A_T} \approx C_{x_M} \sin \chi_A \qquad \chi_A \approx \beta_A \quad -0.5 \gtrsim C_{x_M} \gtrsim -1$$

the above model yields an average effective power coefficient of:

$$\tilde{C}_{P_{EA}} \equiv \frac{\tilde{P}_{EA}}{\frac{\rho}{2} \tilde{U}^2 A_T} \approx C_{x_M}\left(1.3\left(\frac{V}{\tilde{U}}\right)^2 + 0.5\left(\frac{V}{\tilde{U}}\right)^3\right),$$

or to the above approximation: $\approx 1.9\, C_{x_M}\left(\frac{V}{\tilde{U}}\right)^2$.

Some typical examples of average effective power gain or loss to be expected under wind conditions of 15 kts average effective windspeed (at the centre of the wind assistance system) are listed below:

$\tilde{U}_E = 15$ kts	TRADIT. GAFFSL.	MODERN GAFFSL.	SAIL-WING	WINGSL. T.E.-FLAP	FLETTNER -ROTOR	HULL AND SUPERSTR.
C_{D_0}	.10	.07	.05	.02	.16	C_{x_M} = .5 – 1.0
C_{C_M}	1.1	1.4	1.7	2.0	3.2	–
AR_E	1.6	2.5	2.5	2.5	2.0	–
$\Delta\tilde{P}_E/A_S$ [kW/m²]					(-20% $P_{ROT.}$)	$\tilde{P}_{EA}/A_T$ [kW/m²]
V= 8 kts	.10	.15	.19	.22	.22	-.12 -.24
10 kts	.13	.18	.22	.26	.23	-.15 -.31
12 kts	.15	.21	.26	.30	.24	-.19 -.38
15 kts	.18	.26	.31	.36	.25	-.25 -.49

The Flettner rotor, comparing its surface with sail area and with 20 % rotation power subtracted, shows the highest effective power only at low shipspeed. Its power increase with shipspeed, however, is only weak due to its relatively high parasitic and induced drag. Therefore the rotor is outrun in the medium speed range by the hard wingsail with trailing edge flap, and at shipspeed equal to average effective windspeed eventually the expec-

ted average effective power from modern soft sails is in the same order or even higher than from the rotor. The hard wingsail is clear champion in the high speed range, while the output of the traditional gaffsail is the lowest over the whole speedrange at about 50 % of the wingsail.

It may be noted that the loss of effective power due to the windforces on hull and superstructure are to be expected in the same order of magnitude per square meter transverse area as the expected gain due to wind assistance per square meter 'sail area'. That means, if the 'sail area' A_S of the wind assistance systems is not more than the projected overwater transverse area A_T of the ship and superstructure the effect of wind assistance may be not more than a compensation for (considerable) wind resistance of the ship. On the other hand it may be well worth-while considering a reduction of the windage by appropriate shaping of the superstructure in the project stage before considering active wind assistance.

The above comparison on the basis of expected average effective power per square meter, even if converted into delivered power and fuel saving, is only one important input element for a comparative valuation of wind assistance systems, other elements being:

cost of investment and operation per m^2,

as well as practical operational aspects like:

weight and stability,
interference with cargo handling,
reliability, maintenance and repair,
qualification of operator,
remote control and automation.

An economic valuation of wind assistance is difficult and can hardly be of general validity because of uncertainties in the oil price as well as in the costs of investment and operation, the few existing systems being prototypes and still in the experimental phase of operation. On this stage, however, the expected annual fuel saving $\widetilde{\Delta F_a}$ can be estimated, the cost of which is a limiting value for the annual capital and operation costs of the wind assistance system.

To convert the reduction in effective power into a reduction in power to be absorbed by the propeller we must consider the reduction in thrust loading on the propeller and the corresponding propulsive efficiency before and after the discharge. The effect can be expressed by a discharge efficiency η_Δ depending on the level of propeller loading and on the amount of discharge:

$$\eta_\Delta = \frac{\Delta P_E}{\Delta P_D} \approx 0.5 - 0.65 \text{ (tanker, bulkers...)}$$

$$\approx 0.6 - 0.75 \text{ (containerships...)},$$

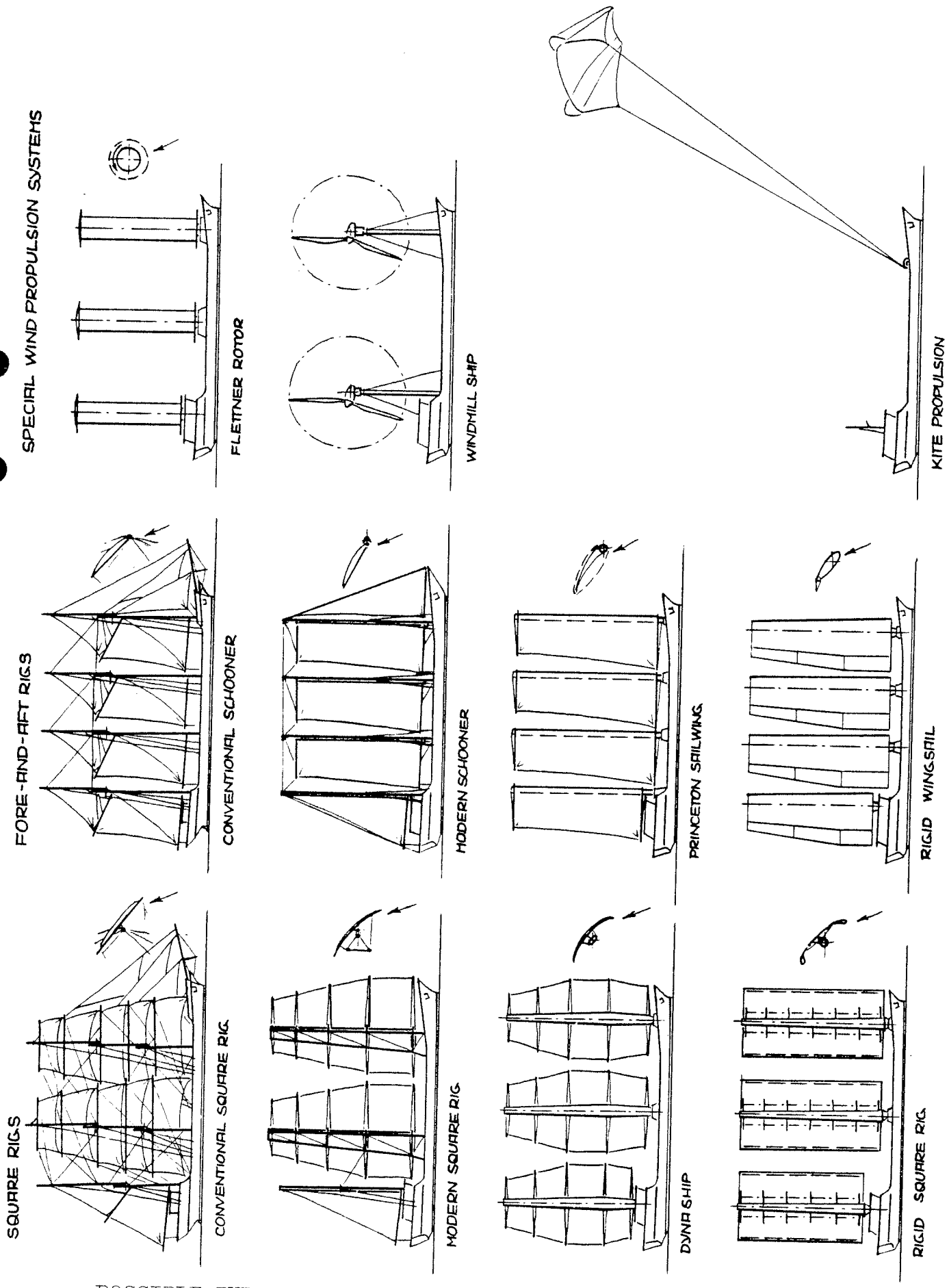
where the upper limit is the quasi propulsive coefficient η_D at full engine propulsion and here valid for full discharge by wind assistance, whereas the lower values are valid for small power reductions. Since the ideal wind assistance will not be always achievable in practice, due to non-optimal adjustment and interference with superstructure or to limiting the forces in strong winds, a reduction factor r for the wind assistance effect is introduced tentatively in the order of about $0.8 - 0.9$. The efficiency of wind assistance may be influenced, in addition, by the lower limit of power depending on engine and propeller type.

With a specific fuel consumption $f_E \approx 0.2\ kg/kWh$ and an annual sea time $t_a \approx 6000\,h \mathrel{\hat{=}} 250\,d$ the expected average annual fuel savings $\widetilde{\Delta F_a}/A_S$ per unit sail area would be:

$$\frac{\widetilde{\Delta F_a}}{A_S} = t_a \cdot f_E \cdot \frac{r}{\eta_\Delta} \cdot \frac{\widetilde{\Delta P_E}}{A_S}$$

$$\text{e.g.} \quad \approx 6000\,h \cdot 0.2 \frac{kg}{kWh} \cdot \frac{0.8}{0.6} \cdot 0.25 \frac{kW}{m^2} = 400 \frac{kg}{m^2}$$

If the annual capital and operation costs of the wind assistance system shall be ballanced by fuel saving alone in the above example one square meter of reference area of the system may not cause annual costs higher than the price of 400 kg fuel respectively, and the depreciation rate and period must be selected accordingly.



POSSIBLE TYPES OF REAL WINDSHIPS

FIG. 13

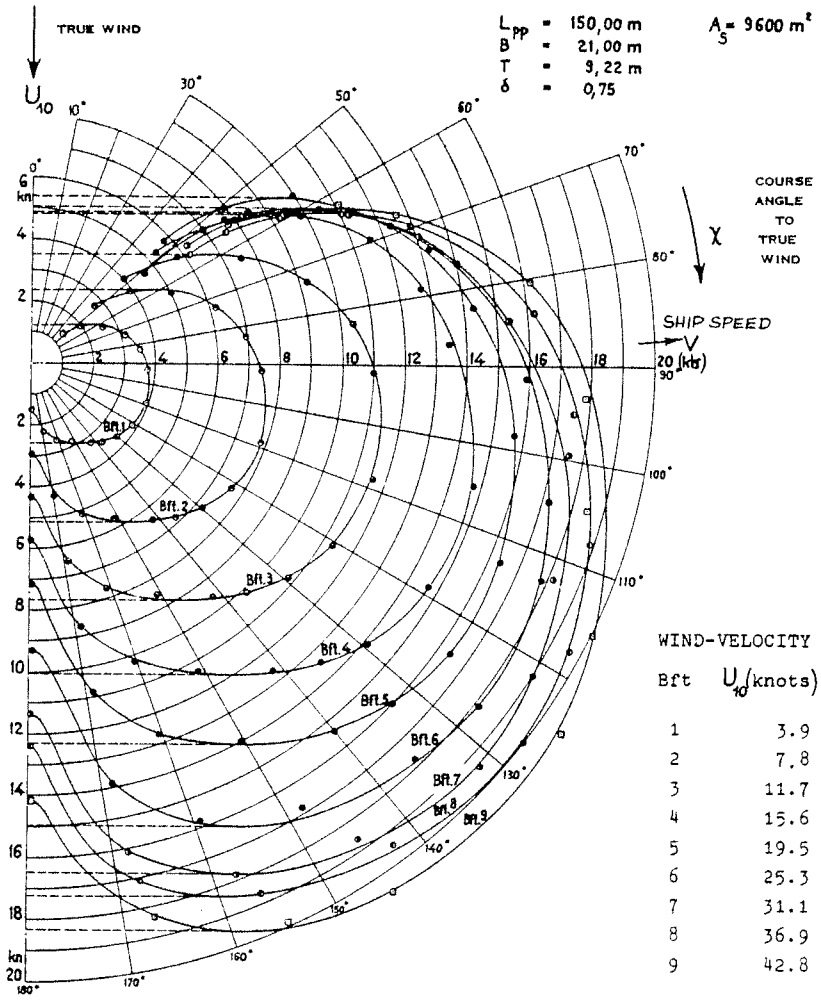
6. Motor-assisted windships

If wind propulsion should help to a substantial fuel saving, we would have to consider, at least for some trades or climatic regions, also the other end of the spectrum of combinations of motor and wind propulsion (Fig. 1): the 'motor-assisted windship' or conventionally called the 'auxiliary sailing-ship'. This ship would, most of the time at sea, be driven by a full size wind propulsion system or sailing rig. The auxiliary propulsion engine of relatively low power would serve only to maintain a certain floor speed in light weather, for port entry and exit and to maintain a save manoeuvrability against strong adverse winds in confined waters.

Some typical proposals for full wind propulsion systems - traditional, modernized and novel - are sketched in Fig. 13. Such a real 'windship' could achieve a substantial fuel saving but would be depending in schedule and in routing, to some extend, on the long-term and short-term weather conditions. While the long-term (seasonal) variations in average (service-) speed can be accounted for by seasonal (climatological) planning of schedule and route, the short-term variations can be largely reduced by weather information and weather-routing, which should be even more important for windships than already proved for motorships.

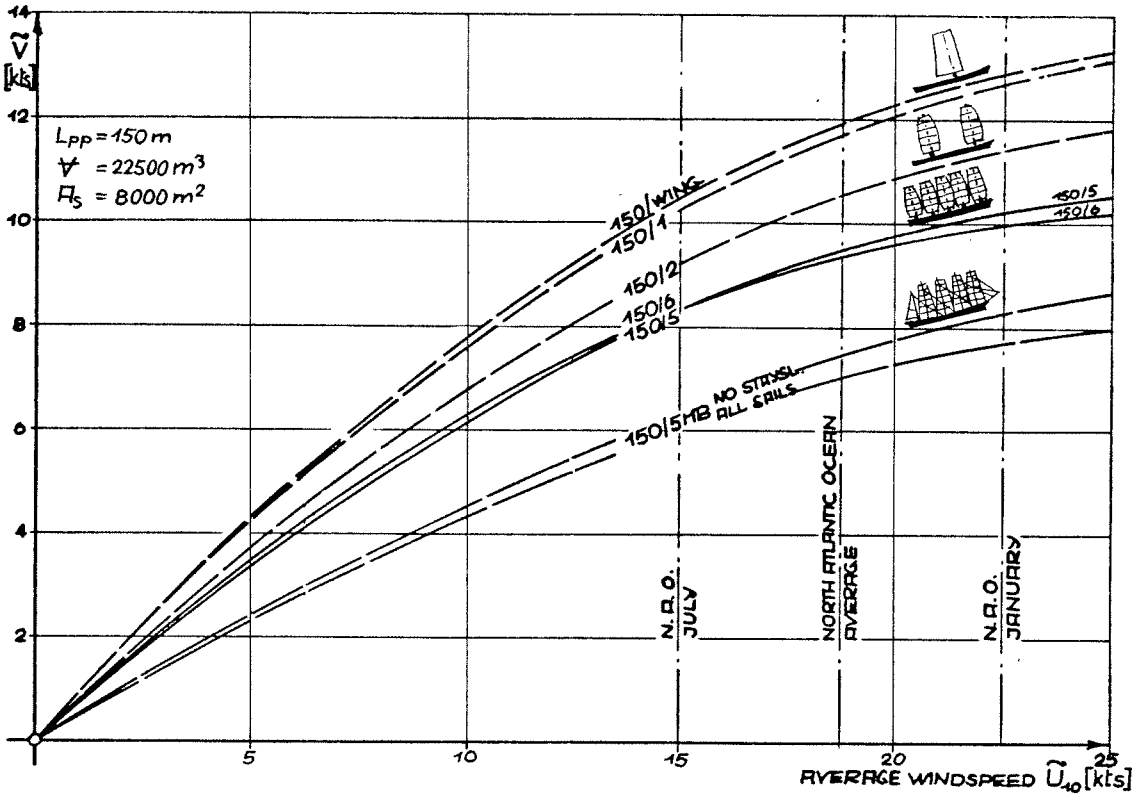
In some trades, e.g. bulk cargo, the absolute long-term schedule keeping seems to be not so important as the relative short-term information to the agent: 'when is the ship expected to arrive at the next port', to be able to make arrangements for ship and cargo, say two days in advance.

A precondition for all short-term planning of schedule and route is the prediction of achievable shipspeed depending on windspeed and relative course. This can be done iteratively by establishing the equilibrium of forces and moments on the hull and on the rig [21]. The hydrodynamic forces on the hull are due to forward speed, drift, heeling, rudder angle and waves, while the aerodynamic forces on the rig and the over-water hull are depending on the inflow of the apparent wind,



VELOCITY DIAGRAM FOR A DYNASHIP SHOWING SPEEDS OF TRAVEL (V) AS A FUNCTION OF THE COURSE ANGLE TO THE TRUE WIND (χ) FOR VARIOUS WIND INTENSITIES ON THE BEAUFORT SCALE (SEILKOPF). [21]

FIG. 14



AVERAGE SPEEDS FOR EQUAL SHIPSIZE AND DIFFERENT RIGS FOR STANDARD WIND DISTRIBUTIONS (ROUNDTRIPS, NO ROUTEING, MAX. HEEL. φ_{MAX} = 10°) [22]

FIG. 15

the adjustment (and reefing) of the rig and the heeling angle under appropriate restrictions by rig strength and stability.

For the estimation of the achievable 'service'-speed and for long-term (seasonal) planning we need an estimation of the expected average speed on a certain route and under the expected wind conditions of the corresponding sea area. This long-term average speed $\tilde{V}$ can be estimated on the basis of the predicted short-term speed assuming standard distributions of wind speed and relative course [22] on a similar way as above discribed for the estimation of the expected average power from wind assistance.

Weather routing is practiced in its simplest (shortest-term) form on every sailing boat by tack upwind or downwind whenever it is necessary or yields a better progress towards the objective than a more direct course. This can be done even on rather short sea turns or under fairly steady weather conditions of trade-winds or monsoons.

On longer sea turns in the variable weather of our temperate latitudes medium-term weather routing may require further and one-sided deviations of the objective course, when this yields an even better progress towards the far distant objective and the deviation can be compensated later when the wind will have changed, e.g. after having passed a depression.

In case of combined propulsion by wind and engine power P the criterion for weather routing is not only maximum speed-made-good towards the objective or minimum time per distance or per voyage but a combined criterion minimizing e.g. the sum of time depending (fixed) costs C_{t_o} and energy-(or fuel-)costs C_E per distance d or per voyage: [23, 24]

$$\frac{C_t + C_E}{d} = c_{t_o} \cdot \frac{t}{d} + c_E \frac{E}{d} \overset{!}{=} min$$

$$= c_{t_o}/V + c_f \cdot f_E \cdot P/V \overset{!}{=} min$$

$$= \frac{c_{t_o}}{V(U,\chi,P)} \cdot \left(1 + \frac{c_f}{c_{t_o}} \cdot f_E \cdot P\right) \overset{!}{=} min$$

The optimum speed V to be sailed by combined propulsion or the optimum percentage of full power P/P_0 to be used is depending on the ratio of fuel costs at full power $C_{f_0 t}$ to fixed costs C_{t_0} :

$$\frac{C_t + C_E}{d} = \frac{C_{t_0}}{V(U,\chi,P)}\left(1 + \frac{C_{f_0 t}}{C_{t_0}} \frac{P}{P_0}\right) \overset{!}{=} min$$

and on the windspeed U and relative course χ .

Whether or not wind propulsion will be applied - as primary propulsion, wind-assistance or in a combined mode with engine - in our industrialized or in developing countries - in steady light-to-moderate wind areas or under strong but variable wind conditions - may be depending not only on simple operational economics but also on a number of additional aspects as:

independence of fuel supply and reduction of pollution,
extended range of operation and stable platform for small research and fishing craft,
or simply the emotional attraction of sailing on cruising vessels.

7. References

[1] -- Symposium on the Futur of Commercial Sail, RINA, London, Nov. 1975

[2] -- Symposium on the Technical and Economical Feasibility of Commercial Sailing Ships, Liverpool Polytechnic, Febr. 1976

[3] -- Symposium on the Viability of Commercial Sailing Ships, Ship and Marine Technol. Requirements Board, Dept. of Industry, London, June 1979

[4] -- Symposium on Wind Propulsion of Commercial Ships (Comsail '80), RINA, London, Nov. 1980

[5] Wieghardt, K.: 'On wind resistance', Intern. Jubilee Meeting, NSMB, Wageningen 1973

[6] Hoerner, S.F.: 'Fluid Dynamic Drag', Hoerner Fluid Dynamics, Bricktown, N.J. 1965

[7] Mc. Cormick, B.W.: 'Aerodynamics of V/STOL Flight', Academic Press, New York, 1967

[8] Isherwood, R.M.: 'Wind Resistance of Merchant Ships', Trans. RINA, 1973

[9] Wagner, B.: 'Windkräfte an Überwasserschiffen', Trans.STG, Hamburg, 1967

[10] Aertssen, G.: 'L'effet du vent sur trois types de navires..', Bull. de l'Ass. Technique Maritime et Aeronautique No. 73, 1973

[11] van Berlekom, W., et al: 'Large Tankers - Wind Coefficients and Speed Loss due to Wind and Sea', RINA, Spring Meeting, London 1974

[12] Böhme, H.; Thiemann, H.: 'Ermittlung der Windkräfte an einem Tanker und drei Containerschiffen', FDS-Rep. No. 48, Hamburg, 1975

[13] Willoughby, R.M., et al: 'Design Problems of a Commercial Ship', Comsail '80 [4] , London, Nov. 1980

[14] Woodward, I.B., et al: Feasibility of Sailing Ships for the American Merchant Marine', prep. for U.S. Mar. Adm., Univ. of Mich. Rep. No. 168, Feb. 1975

[15] Prölss, W.: On the Economic Possibilities of Windpropelled Merchant Ships, Transl. from Trans. STG 1967, IfS-Transl. No. 4059, Hamburg, 1967

[16] Sudo, M., et al: 'Operation Performance of Sail Equipped Small Tanker', Nippon Kokan Techn.Rep. Ov.No.33,1981

[17] Bergeson, L., et al: 'Wind Propulsion for the American Merchant Marine', Prep. for U.S. Mar. Adm., Windship Devl.Corp., Norwell Mass., March 1981

[18] Wagner, C.:'Technische und wirtschaftliche Aspekte für einen Windzusatzantrieb mit Rotoren...', Schiff und Hafen, 34, H. 9, 1982

[19] Wagner, B.:'Sailing Ship research at Hamburg University', Symposium Liverpool [2] , IfS-Schrift Nr. 2249, Hamburg, 1976

[20] Remmers, K.: 'Die Leistung von Hilfssegeln', Schiff und Hafen H. 1, 1949

[21] Wagner, B.:'Fahrtgeschwindigkeits-Berechnungen für Segelschiffe', Trans. STG, Hamburg, 1967 Translation by Liverpool Polytechnic 1976

[22] Schenzle, P.:'Standardised Speed Prediction for Wind Propelled Merchant Ships', Comsail'80 [4] , London, Nov. 1980

[23] Mays, J.H.:'Sailing Ship Weather Routing', Comsail'80 [4] London, Nov. 1980

[24] Letcher, J.S.:'Optimal Performance of Ships Under Combined Power and Sail', Journal of Ship Research, 26, Sept. 1982

Appendix: Fluid dynamic drag and crossforce (lift)

Wind forces on ships are consisting of the two basic types of fluid dynamic forces: drag or resistance in the direction of the undisturbed relative inflow and crossforce or lift under a right angle to the inflow. Since we are only interested in forces in the horizontal plane, the term 'crossforce' seems to be preferable here to the term 'lift' for the horizontal force component at a right angle to the inflow.

Usually fluid dynamic forces are presented in the form of coefficients made nondimensional by the stagnation pressure of the inflow and a sometimes arbitraty reference area A, e.g. the transverse or frontal area A_T, the lateral area A_L, the 'wetted surface' area S_w or simply a characteristic length squared L^2:

Drag coefficient $$C_D = \frac{D}{\frac{\rho}{2} V^2 A}$$

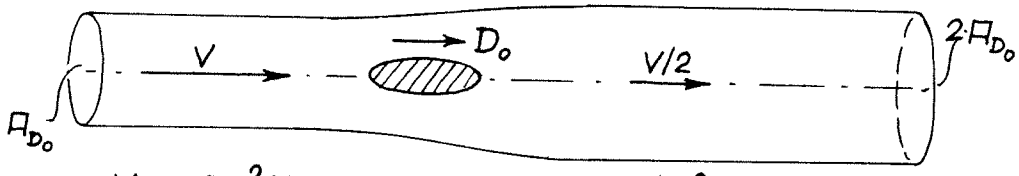
Crossforce coefficient $$C_C = \frac{C}{\frac{\rho}{2} V^2 A}$$ (= C_L lift coefficient)

The arbitrary choice of a reference area A can be avoided, if it is included in a corresponding force coefficient with the dimension of an area.

'Drag area' $$A_D = \frac{D}{\frac{\rho}{2} V^2} = C_D \cdot A$$

'Capture area' $$A_C = \frac{C}{\rho V^2 \sin\alpha} = C_C \cdot \frac{A}{2\sin\alpha}$$ with α 'downwash angle' or 'deflection angle'.

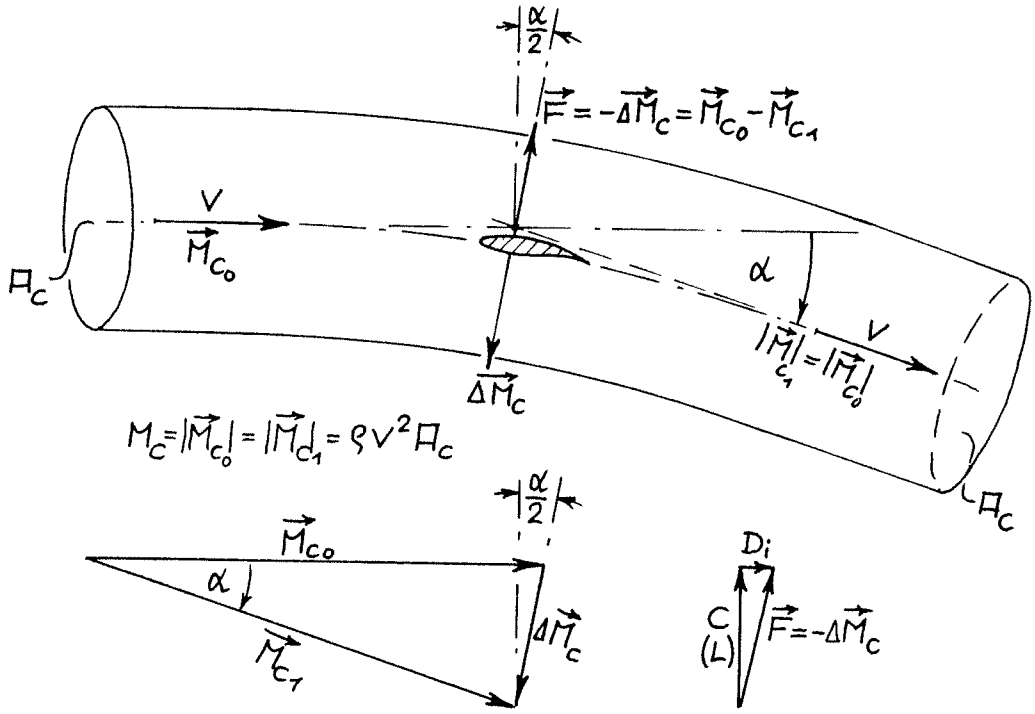
The 'drag area' A_D can be thought of as of a fictive transverse area, which, when loaded with the full stagnation pressure $q = \frac{\rho}{2} V^2$ of the undisturbed inflow, would produce the parasitic drag force D_o. An equivalent interpretation of A_{D_o} is the cross section area of a stream tube in the undisturbed inflow which, while passing the drag body, looses half of its initial momentum flux $M_{D_o} = \rho V^2 A_{D_o}$:



$$M_{D_o} = \rho V^2 A_{D_o} \qquad M_{D_1} = \rho \left(\frac{V}{2}\right)^2 \cdot 2 A_{D_o} = \frac{\rho}{2} V^2 A_{D_o} = M_{D_o}/2$$

$$D_o = M_{D_o} - M_{D_1} = \frac{\rho}{2} V^2 A_{D_o}$$

Similarly the 'capture area' A_c is a fictive cross section area of a stream tube in the undisturbed inflow, which is deflected by an angle α while passing a lifting body:



The corresponding momentum flux $M_c = \rho V^2 A_c$ is changed not in magnitude but in direction by the angle α. At the location of the lifting body the flow has been deflected just by the mean angle $\alpha/2$ between the undisturbed 'inflow' $\vec{M}_{c_0}$ and the eventual direction α of the 'downwash' $\vec{M}_{c_1}$. The differential momentum $\Delta\vec{M}_c = \vec{M}_{c_1} - \vec{M}_{c_0}$ is at a right angle to the mean flow direction $\alpha/2$ and corresponding to an oppositely directed resulting force $\vec{F} = -\Delta\vec{M}_c$ on the body with the components

'crossforce' (or 'lift') $C(=L) = M_c \cdot \sin\alpha = \rho V^2 A_c \cdot \sin\alpha$

'induced drag' $D_i = M_c \cdot (1-\cos\alpha) = \rho V^2 A_c \cdot (1-\cos\alpha)$

In reality however, the momentum flux influenced by the body is not bounded by the fixed stream tubes with the cross sections of 'drag area' A_D and of 'capture area' A_c, but rather smoothly distributed and gradually vanishing at greater distances from the body. But the ballance of forces and momentum is correctly represented by the model. Especially in case of the crossforce the model yields the correct relation to the induced drag D_i due to the downwash α which is increasing with decreasing capture area A_c:

$$\sin\alpha = \frac{C}{\rho V^2 A_c} \qquad D_i = C \cdot \tan\left(\frac{\alpha}{2}\right) \approx \frac{C^2}{2\rho V^2 A_c}$$

For a single wing the capture area A_c is equal to a circular area with diameter equal to the effective 'span' b_E of the wing:

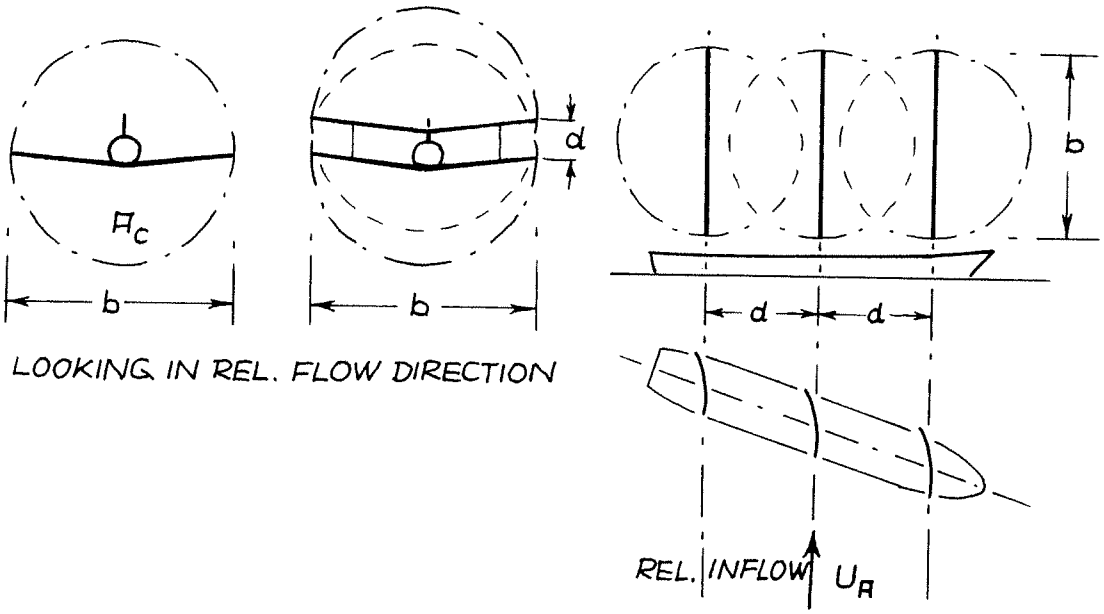
$$A_c = \frac{\pi}{4} b_E^2 \quad ,$$

and the relation to the commonly used effective 'Aspect Ratio' AR_E is then:

$$AR_E \equiv \frac{b_E^2}{A} = \frac{4}{\pi}\frac{A_c}{A}$$

$$\text{or} \qquad A_c \equiv \frac{\pi}{4} b_E^2 = \frac{\pi}{4} \cdot AR_E \cdot A \quad .$$

For a <u>multi-wing arrangement</u>, as e.g. a biplane or a multi-mast sailing rig, the capture area or the aspect ratio of each wing (or sail) is reduced due to the interaction with the neighbouring wing(s) according to the relative distance d/b, measured transversely to the flow direction:



From biplane theory [6] we know that, when we move a second identical wing from infinity to a position at a close distance d over a first wing, the capture area of the combined system A_{c_2} is reduced to less than twice the area A_{c_1} of the single wing. Or the effective aspect ratio AR_2 of the combination is less than the value AR_1 for the single wing:

$$\frac{A_{c_2}}{A_{c_1}} = 1 + a\left(\frac{d}{b}\right)$$

$$\frac{AR_2}{AR_1} = \frac{1}{2} \cdot \left(1 + a\left(\frac{d}{b}\right)\right)$$

The reduction factor $a\left(\frac{d}{b}\right)$ accounts for the reduced amount of momentum that can be captured between the wings, whereas the portions above the upper and below the lower wing can be regar-

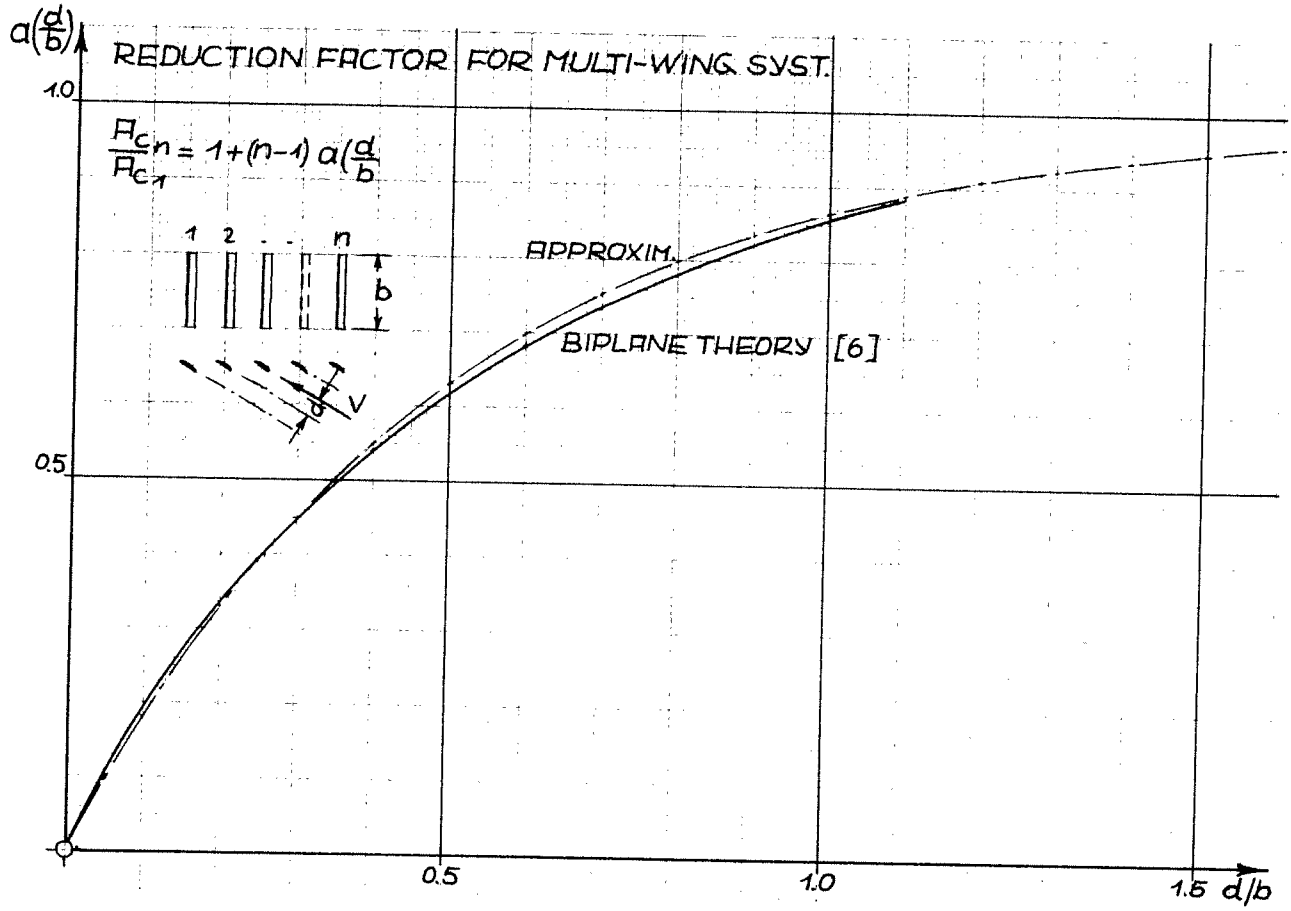
ded as remaining unreduced. If this procedure is extended, as an approximation, to multi-wing or multi-mast arrangements the corresponding formulae are:

$$\frac{A_{c_n}}{A_{c_1}} = 1 + (n-1) \cdot a\left(\frac{d}{b}\right)$$

$$\frac{AR_n}{AR_1} = \frac{1}{n} + \left(1 - \frac{1}{n}\right) \cdot a\left(\frac{d}{b}\right)$$

A simple approximation for the reduction function $a\left(\frac{d}{b}\right)$ is:

$$a\left(\frac{d}{b}\right) \approx 1 - e^{-2\frac{d}{b}}$$

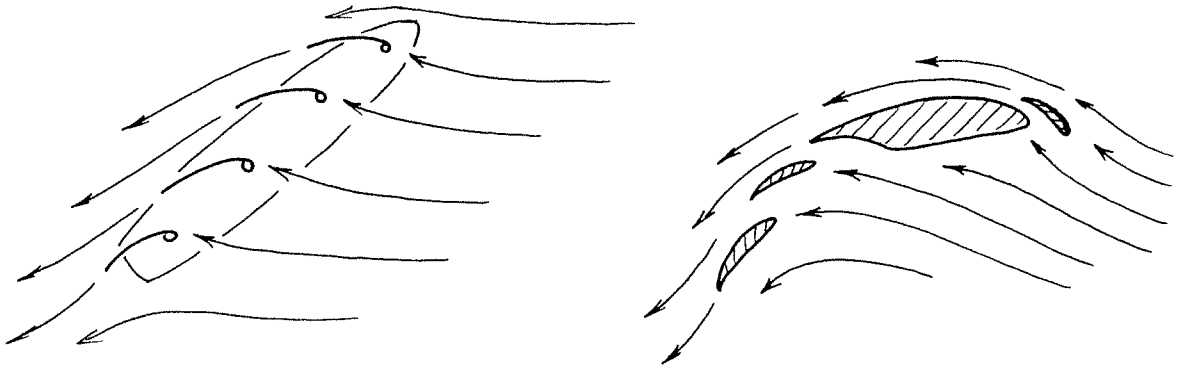


A few numerical examples for double wing arrangements are:

distance d/b	∞	1.0	0.5	0.25	0.1
capture area A_{c_2}/A_{c_1}	2	1.85	1.6	1.4	1.2
aspect ratio AR_2/AR_1	1	0.9_3	0.8	0.7	0.6

If wing span is unlimited a single wing with large span has a big capture area and a good efficiency in generating a given lift at low induced drag. If span or rig height is limited, however, from geometrical or structural reason, it is better to split up the available area into a series of wings or masts of better effective aspect ratio AR_n to achieve an increase in

capture area by a factor A_{c_n}/A_{c_1} instead of a single wing with a low aspect ratio of AR_1/n . If in such a multi-wing arrangement each element is carefully adjusted to the local angle of attack, the attached lifting flow can be maintained up to remarkably high maximum crossforce or lift coefficients $C_{C_{MAX}} = C_{L_{MAX}} = C_{L_X} \gtrsim 2.0$ with gentle stalling characteristics similar to landing aircraft splitting up its wing into a system of carefully adjusted slats and slotted flaps.



In summary we have now established a <u>simple model for fluid dynamic forces</u> being made up of three principle components represented by characteristic areas or coefficients:

the parasitic drag mostly due to viscous and separation effects:

$$D_o = \frac{\rho}{2} V^2 A_{D_o} \qquad C_{D_o} \equiv \frac{D_o}{\frac{\rho}{2} V^2 A} = \frac{A_{D_o}}{A}$$

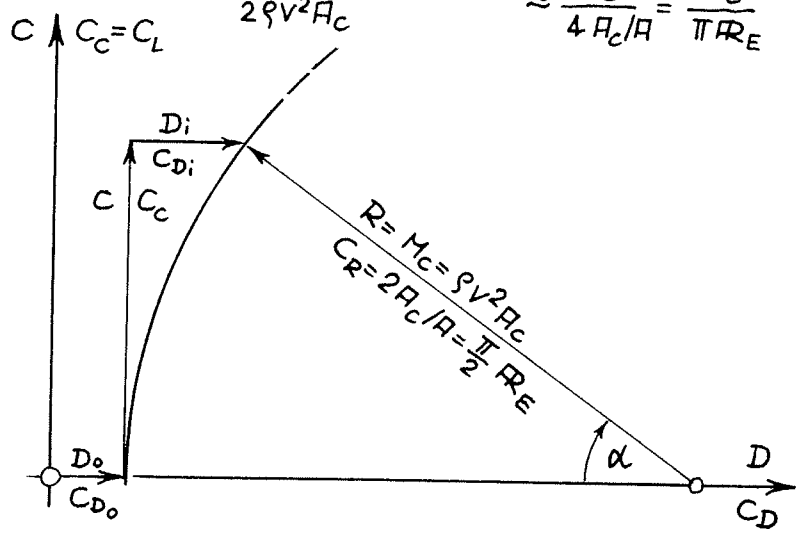
the crossforce or lift:

$$C = \rho V^2 A_c \sin\alpha \qquad C_C \equiv \frac{C}{\frac{\rho}{2} V^2 A} = \frac{A_c}{A} \cdot 2 \sin\alpha = \frac{\pi}{2} AR_E \sin\alpha$$

and the drag due to lift or induced drag:

$$D_i = \rho V^2 A_c (1-\cos\alpha) \qquad C_{D_i} \equiv \frac{D_i}{\frac{\rho}{2} V^2 A} = \frac{A_c}{A} \cdot 2(1-\cos\alpha) = \frac{\pi}{2} AR_E (1-\cos\alpha)$$

$$\approx \frac{C^2}{2\rho V^2 A_c} \qquad \approx \frac{C_C^2}{4 A_c/A} = \frac{C_C^2}{\pi AR_E}$$



The line of possible combinations of fluid dynamic drag and crossforce or lift on a body in a lift -versus- drag polar diagram is a circle with radius:

$$R = M_C = \varrho V^2 A_C$$

equal to the capturedmomentum flux and determined by the capture area A_C . The centre is shifted on the drag axis to $D = D_0 + R$ so that the minimum drag at zero crossforce is the parasitic drag D_0 .

This is a sufficient model only for low lift or crossforce coefficients $C_C \lesssim 1$ and small deflection angles $\alpha \lesssim 0.5 \lesssim 30°$.

<u>High crossforce coefficients are limited by stalling</u> to more or less high values $C_{C\,MAX}$ or C_{LX} between ~ 9 and 2.5 depending on the stalling characteristics of the wing or sail. After stall in the separated flow condition the resulting force is directed almost normal to the plane of the chord of the wing or sail with an almost constant coefficient $C_N = C_{D\,MAX}$ between 1 and 1.5 for sharp edged plates or sails. Consequently our model of fluid dynamic forces is completed by the stalling point at $C_C = C_{CM}$:

$$C_{CM} = C_{C\,MAX} = C_{LX}$$

$$\sin\alpha_M = C_{CM}/(2\,A_C/A) = C_{CM}/(\tfrac{\pi}{2} AR_E)$$

$$C_{DM} = C_{D_0} + 2\,A_C/A \cdot (1 - \cos\alpha_M)$$

$$= C_{D_0} + \tfrac{\pi}{2} AR_E\,(1 - \cos\alpha_M)$$

and by the normal force C_N in the separated flow condition after stall:

$$C_C = C_N \cos\beta$$

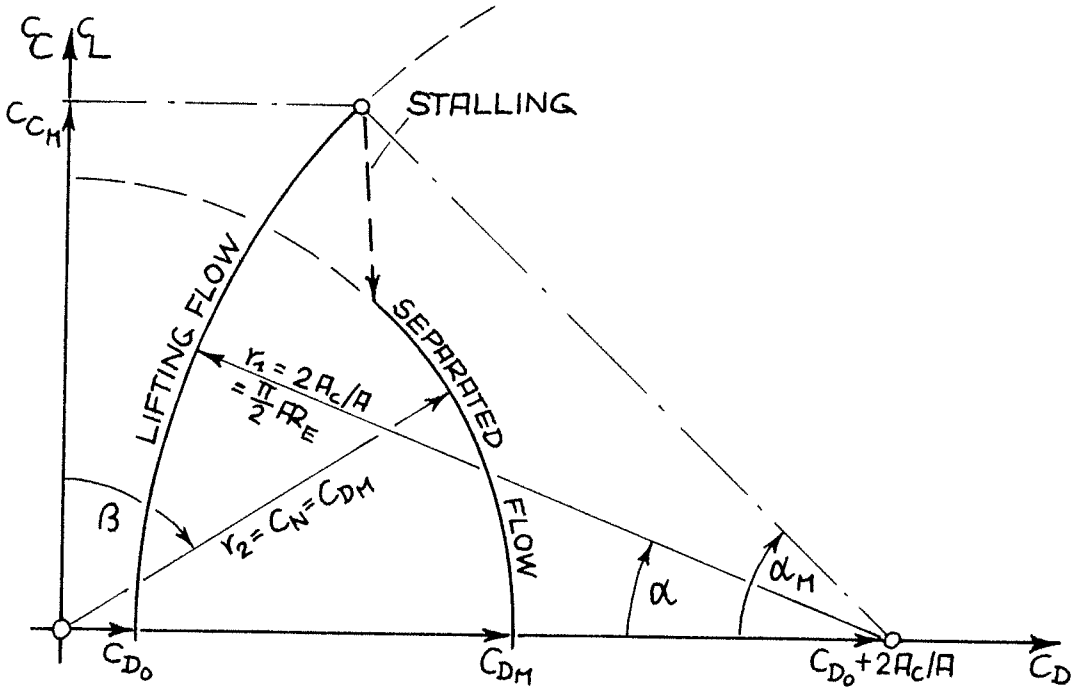
$$C_D = C_N \sin\beta \;,$$

where β is the angle of attack of the inflow relative to the chord of the wing or sail. In the region of attached lifting flow condition the relation between the deflection angle α and the effective angle of attack β_E relative to the zero lift inflow is with good approximation:

$$\beta_E = \frac{\alpha}{2}\left(1 + \sqrt{1 + \left(\frac{2}{\pi}\frac{A_C}{A}\right)^2}\right) = \frac{C_C}{4A_C/A}\left(1 + \sqrt{1 + \left(\frac{2}{\pi}\frac{A_C}{A}\right)^2}\right)$$

$$= \frac{\alpha}{2}\left(1 + \sqrt{1 + \left(\frac{AR_E}{2}\right)^2}\right) = \frac{C_C}{\pi AR_E}\left(1 + \sqrt{1 + \left(\frac{AR_E}{2}\right)^2}\right)$$

with the limits $\beta_E \longrightarrow \alpha \cdot AR_E/4$ for $AR_E \longrightarrow \infty$
and $\beta_E \longrightarrow \alpha$ for $AR_E \longrightarrow 0$



The completed polar diagram of fluid dynamic forces is now consisting of two circular lines for the regions of attached lifting flow and of separated flow respectively.

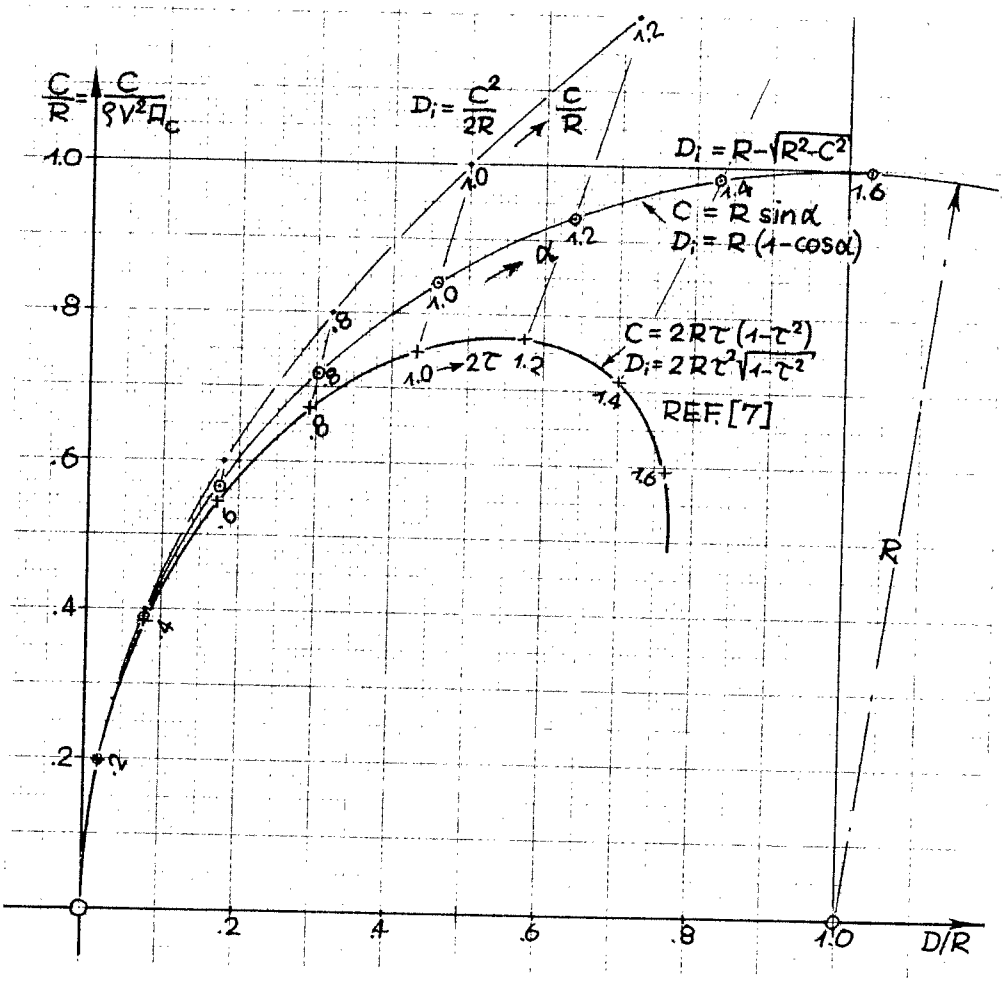
For <u>high deflection angles</u> $\alpha \gtrsim 0.5 \hat{=} 30°$ the deflected momentum is gradually reduced by a deformation of the free vortex system in the downwash. The crossforce is reduced and the line of fluid dynamic forces gradually deviates below the circular line above. The crossforce and induced drag including this effect is given according to Mc Cormick [7] :

$$L = C = \rho V^2 A_c \cdot 2\tau \cdot (1-\tau^2)$$

$$D_i = \rho V^2 A_c \cdot 2\tau^2 \cdot \sqrt{1-\tau^2},$$

where the parameter $\tau = \frac{\Gamma_0}{2bV}$ indicates the relative circulation in the centre of an elliptically loaded wing and is, for small values, equal to the half deflection angle $\alpha/2$:

$$\tau \approx \frac{\alpha}{2} \quad \text{for} \quad \alpha \lesssim 0.5 \hat{=} 30°$$



The above model can be applied also to the drifting underwater hull of a ship under the action of a side force, caused by wind. In the low-speed range of negligible wavemaking the parasitic drag can be accounted for by a constant drag area A_{D_0} in the order of $0.003 \cdot S_W$, the wetted surface.

$$A_{D_0} = C_T \cdot S_W \approx 0.003 \cdot S_W$$

In the high speed range the drag area is increasing like the total resistance coefficient as a funktion of Froude number. For the determination of crossforce the effective span b_E including the mirror effect of the water surface is nearly twice the ships maximum draft:

$$b_E \approx 1.9\, T_M$$

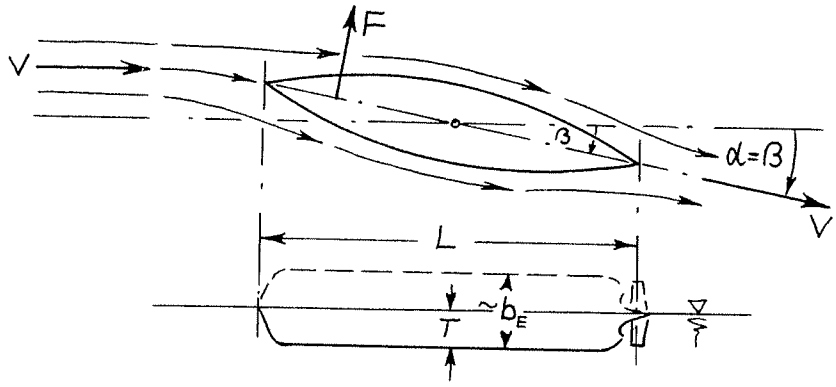
but the effective capture area A_C is, of course, only the lower half circle with diameter b_E, which is actually existing under water:

$$A_C = \frac{1}{2} \cdot \frac{\pi}{4} b_E^2 \approx \frac{1}{2} \cdot \frac{\pi}{4} (1.9\, T_M)^2 \approx 0.9 \cdot \frac{\pi}{2} T_M^2 \approx 1.4\, T_M^2$$

For a normal commercial ship hull on even keel the aspect ratio:

$$AR_E \equiv b_E^2 / (T \cdot L) \approx 1.8\, T/L$$

is extremely small and the flow deflection at small drift angles β is already completed in the forward part of the hull while in the rear part the fully deflected flow is parallel to the hull axis:



Consequently the deflection angle α is equal to the (small) drift angle β and the crossforce is concentrated close to the ship's bow and must be ballanced by a permanent rudder angle for course keeping. For a given side force F_y or crossforce $C \approx F_y$ the drift angle is $\beta = \alpha \approx \frac{C}{\rho V^2 A_c}$ and the induced drag is $D_i \approx k \frac{\beta}{2} \cdot C \approx \frac{k}{2} \frac{C^2}{\rho V^2 A_c}$,

where k is a correction factor in the order of 2 , accounting for reduced leading edge suction and depending on the shape of the ship's bow. For a normal sharp-edged stem k is between ~2 and 2.5 , while for bulbous bows and especially for elliptical bows of full slow-speed tankers and bulkers k may be less than 2 . In this case the induced longitudinal force parallel to the ship's forward axis may be positive, that means a 'driving force':

$$
\begin{aligned}
F_{x_i} &= C \sin\beta - D \cos\beta \\
&\approx C\left(\beta - k\frac{\beta}{2}\right) = C \cdot \beta \left(1 - \frac{k}{2}\right) \\
&= \rho V^2 A_c \beta^2 \left(1 - \frac{k}{2}\right) \\
&= \frac{C^2}{\rho V^2 A_c} \left(1 - \frac{k}{2}\right)
\end{aligned}
$$

for $k < 2 \rightsquigarrow F_{x_i} > 0$ (forward)

for $k > 2 \rightsquigarrow F_{x_i} < 0$ (backward)

For larger drift angles $\beta \gtrsim 5^\circ$ of low aspect bodies a second nonlinear term of the crossforce becomes important, which is due to the crossflow around the ship's keel and distributed over the length according to the shape of the lateral area and the ship's cross sections:

$$F_y \approx \rho V^2 A_C \cdot \beta + \frac{\rho}{2} V^2 C_2 A_L \cdot \beta \cdot |\beta|$$
$$= \rho V^2 A_C \cdot \beta \cdot \left(1 + \frac{C_2}{2} \frac{A_L}{A_C} |\beta|\right)$$
$$= \rho V^2 A_C \cdot \beta \cdot \left(1 + 2 C_2 \frac{|\beta|}{\pi A R_E}\right),$$

where the coefficient C_2, resembling to a drag coefficient against crossflow, is in the order of ~ 1.5 for round sections to ~ 2.5 for a sharp keel.

A trim to the stern or a rake of keel is effective in shifting the linear part back from the bow, thus unloading the rudder or enabling the hull to carry sails also on the rear part.

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1842

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

NAVOSH 008

EIGHTH SCHOOL 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

Part 3

DESIGN AND OPERATION

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SHIP DESIGN FOR FUEL ECONOMY

THE ROLE OF ECONOMICS IN SHIP DESIGN FOR FUEL ECONOMY

by

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University of Michigan
Ann Arbor

ABSTRACT

The point of this paper is that the real theme of the conference is not to save fuel, but to save money. As we consider various ways to increase fuel efficiency we must also consider their effects on capital and operating costs, and on revenues. For each alternative proposal we must predict the cash flow pattern (i.e., how much cash will flow in or out of the firm and when) as a result of adopting that particular proposal. We then consider how to select between competing alternatives, based on these cash flow projections.

In going through the above, we must recognize complicating factors such as taxes, inflation, borrowed capital, and decision making in the face of an unpredictable future. While these complicating factors are dealt with rather superficially, attention is called to more thorough treatises.

Because of the wide diversity in background of the conference's participants, no effort is made to present reliable data. The aim, rather, is to develop an understanding of the principles involved in the use of engineering economics as an aid in making rational decisions in designing ships for fuel economy.

ABBREVIATIONS

A:	A uniform annual cash flow
A':	A uniform annual after-tax cash flow
AAB:	Average annual benefit
AABI:	Average annual benefit index
AAC:	Average annual cost
ACCR:	Annual cost of capital recovery
C:	Annual transport capability
CA:	Compound amount factor
CC:	Capitalized cost
CR:	Capital recovery factor
CR':	Capital recovery factor after tax
D:	Depreciation
ECT:	Economic cost of transport
F:	A future sum of money
g:	Gradient (incremental annual amount)
i:	Annual interest rate
i':	Annual interest rate applied to after-tax cash flows; also yield on an investment
LCC:	Life cycle cost
P:	A single initial amount; also investment
PBP:	Pay back period
PV:	Present value (same as PW)
PW:	Present worth (same as PV)
N:	Number of years in future
NPV:	Net present value
NPVI:	Net present value index
Rev.:	Annual revenue
RFR:	Required freight rate
SCA:	Series compound amount factor
SF:	Sinking fund factor
SPW:	Series present worth factor
t:	Tax rate
Y:	Annual operating costs

Notes

1. (SPW-10%-5): Signifies the series present worth factor for 10 percent interest and 5 years.

   Similar arrangements are used for all the interest relationships.

2. Other abbreviations are explained where used.

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1. INTRODUCTION

The theme of this conference, Ship Design for Fuel Economy, is highly appropriate in this era of high oil prices. Yet, we must keep in mind that the central aim of fuel economy is ship economy. In basic terms, as we make decisions about designing and operating ships, we want to make the best possible use of scarce resources (among which we include fuel). That is what economics is all about.

At this conference you will hear many proposals for improved designs or operating methods, all aimed at saving fuel. In judging each proposal, you should always ask if the future cash benefits justify the greater initial cost that will usually be required. In simplest terms, economic analysis almost always boils down to that question. The question is simple, but the solution is somewhat less so. We cannot just add future savings and see if they exceed the investment. Too many complicating considerations intervene. These include:

*The time value of money
*Taxes
*Risks and uncertainty
*Inflation or deflation
*Inventory costs
*Financing methods
*Timing of payments

My intent in preparing these notes is to introduce the practicing engineer to the elements of engineering economics. Supplied with these tools and a bit of common sense, one should be able to handle all those complicating factors in weighing the relative merits of competing proposals for saving fuel.

In all of this, I beg you to keep in mind that the principles are clear-cut and easily understood. The analysis may look horribly complex, but it is in most cases methodically assembled from simple components. Taken one step at a time, the task is easy.

Everyone can agree with the dictum that (a) a merchant ship is a capital investment that earns its returns as a socially useful instrument of transport, and (b) the best measure of engineering success is profitability. The only trouble is that business managers cannot agree on how to define profitability. We shall not take sides on this issue, but explain how to derive each of the more widely accepted, logical measures of economic merit.

2. SYSTEMS ANALYSIS

Engineering economics is closely akin to systems analysis, an organized approach to decision making. This is a systematic way of attacking a problem, using the following dicrete steps:

*Clearly define the objective in functional terms (e.g., move coal from A to B).

*State clearly under what constraints the system is to operate (e.g., national flag requirements, classification society requirements, port and canal requirements, etc.)

*Decide on what economic measure(s) of merit are to apply, and what affecting values (e.g., tax rates) are to be assumed.

*Generate a list of all conceivable strategies for attaining the objective in the face of the constraints.

*Predict the quantitative value(s) of the measure(s) of merit likely to be attained by each of the alternative strategies.

*Append some discussion of any important influencing considerations that cannot be reduced to monetary terms (e.g., political implications).

* * *

In preparing to carry out the next-to-last step we must predict as best we can how each decision will affect the cash flowing in or out of the organization. What are the added costs, what are the future net benefits, and when will they occur?

Computers may be brought to bear if large numbers of alternatives are to be considered. Whether we use computers, hand-held calculators, or backs of envelopes, however, one rule applies: The decision will be made by some person, or group of persons, and will not hinge simply on the best numerical projection of some measure of merit. Like nearly all else in our business, there is art as well as science in this. Indeed -- and roughly speaking -- the more important the decision the greater is the reliance upon art. That is what makes ship design so fascinating.

Until relatively recent years naval architects and marine engineers were not instructed in practical economics as part of their formal education. As an unfortunate result, most of the big and important studies bearing on ship design were (and often still are) made by accountants. I refer to this as unfortunate. Accounting is an admirable and necessary profession. Those reared in its complexities are not, however, ideally suited to the task of analyzing alternative design proposals -- at least not by themselves. Accountants of course do not pretend to understand the technical matters involved in ship design. More than that, they are trained to look back, not ahead, and they allow the arbitrary strictures of bookkeeping rules to distort their thinking. Three examples:

*Accountants tend to ignore lost-opportunity costs because they are not entered in the books.

*Accountants tend to treat imaginary depreciation costs as though they actually existed.

*Accountants normally accept money at face value just as though inflation did not exist.

Wise ship design decisions require teamwork between engineers, business managers, and operating personnel.

Three additional observations are pertinent at this point:

*Decisions are between alternative opportunities. In making comparisons, concentrate your attention on assessing those cost factors that would be different between the alternatives. Ignore past history except as an aid in predicting the future.

*Since much guess work is involved in predicting future conditions, cost projections are bound to be crude. Moral: do not carry computations beyond three or, at most, four significant figures.

*Most engineering decisions should be made on a basis of simple economic analysis. Prudent business managers usually select their options, and decide yes or no, on straightforward economics. Only after that will they seek the best method of financing. By that time, however, the big design decisions have been made. In short, do not adulterate your economic studies with confusing financial intricacies.

3. TIME VALUE OF MONEY

Money does you no good until you exchange it for immediate personal needs or desires. Human nature being what it is, we almost invariably prefer to have our needs or desires met as soon as possible. This means that a given amount of cash that exchanges hands today is more important than the same amount of cash exchanging hands in the future. This is a natural, human trait and one that must be recognized if we are to make proper decisions in satisfying human needs -- which is, after all, the central aim of engineering.

In more mundane terms, a sum of money in the hand today need not be spent, but could be put to work and allowed to generate rent money (i.e., interest). In so doing you have sacrificed the goods or services you could have acquired today in the hope of acquiring more or better in the future. Again, the concept is solidly based on human instincts and refutes those who claim to see evil in profitable enterprises.

We come, then, to a fundamental concept of economics: We must consider not only <u>how much</u> money flows in or out of an organization, but <u>when</u>.

The preceding paragraphs have explained, qualitatively, why we must assign some time value to money. We must also consider relative risks, recognizing that expectations may or may not be fulfilled. Riskier proposals naturally place greater emphasis on the time value of money.

Inflation is an altogether different matter that will be dealt with separately. See Section 13.

The quantitative recognition of time value of money is handled by means of standard compound interest formulas. I assume you are at least vaguely aware of these, and I shall make no attempt to prove the relationships. See References 8, 14, or 20 if you are curious.

We can think about interest in three distinct forms, although the mathematics remain the same regardless. The three forms are:

*Some contractural arrangement, such as the interest rate (and frequency of falling due) agreed upon before you borrow money from a bank.

*Lost opportunity costs, such as the potential gains rejected when you hoard cash, or fail to invest in a profitable enterprise when the chance is yours.

*Internally generated cash from a profitable enterprise, sometimes called <u>yield</u>.

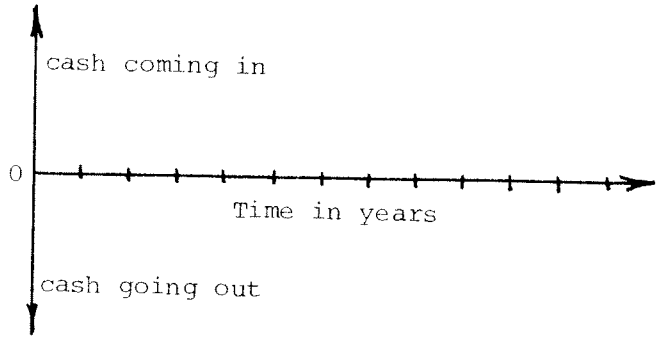
Interest relationships are used to weigh the relative merits of cash flows that occur at different times. A basic concept here is to equate a future cash flow to an equivalent amount right now. We call this imaginary immediate figure <u>present worth</u>, PW , or <u>present value</u>, PV . To find it, we discount the future amount by a factor that is affected by (a) the time value of money (i.e., interest rate), and (b) the remoteness of the event. The mathematics of this are easy, as explained in Section 5.

4. STANDARDS

Before discussing interest relationships we should understand a few standard assumptions and abbreviations. These are common to most engineering economy studies, although variations do exist.

Assumption number one is that all cash flows occur on the last day of the year. This assumption simplifies our mathematics. Although in fact cash may change hands almost continuously, any errors that result from our simplifying assumption are likely to be common to all alternatives under study. Common errors are neutral in their effects on decision making.

To help us visualize the amount and timing of cash entering or leaving the organization, we frequently use what we call a cash flow diagram. Time is represented on the horizontal scale. Annual cash amounts are shown on the vertical scale.



Zero on the time scale can be arbitrarily selected. It may mean:

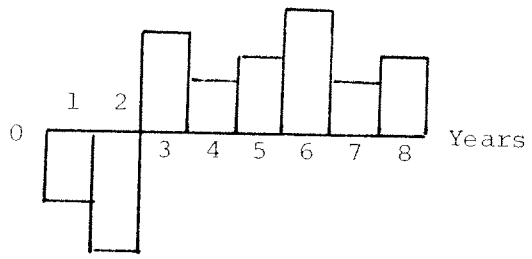
Now,

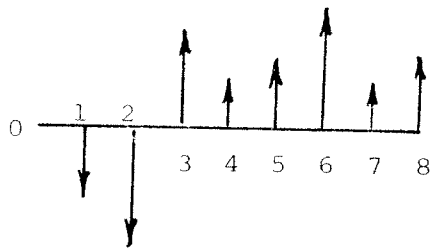
Time of decision,

Time when unit goes into service,

Etc.

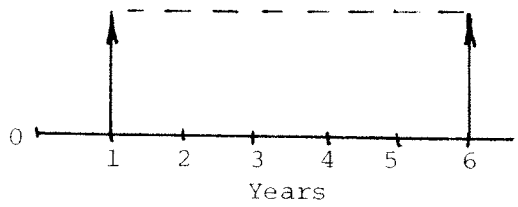
Cash flows may be represented by bars or by arrows:





In the above, each end-of-year arrow represents the amount of cash that flowed in or out during that year.

Where annual cash flows are uniform we may use this convention:



Numerical values of the compound interest relationships will hinge on (a) the interest rate per compounding period and (b) the number of compounding periods. Under our assumption of year-end cash flows, we are talking about annual interest rates, i, and number of years, N. One standard shorthand method for specifying i and N is shown by this example. Suppose we are talking about the present worth factor for 12 percent interest and 15 years. We show it thus:

(PW-12%-15)

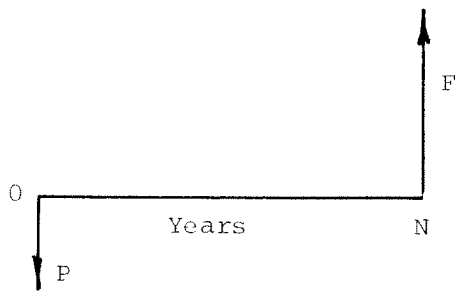
On occasion, in order to save time and space, we may append the numerical value, which in this case happens to be 0.1827:

(PW-12%-15)0.1827

This can be interpreted as, "the present worth factor for 12 percent annual interest and 15 years, which has a numerical value of 0.1827..."

5. INTEREST RELATIONSHIPS

5.1 Single Amounts



Here we have the basic relationship between a single amount at time zero and another single amount N years in the future.

$$F = (CA\text{-}i\text{-}N)P \qquad (1)$$

and

$$P = (PW\text{-}i\text{-}N)F \qquad (2)$$

where

F = future amount

P = present amount, principal, present worth, or present value

CA = compound amount factor

PW = present worth factor

Our algebraic relationships are:

$$(CA\text{-}i\text{-}N) = (1+i)^N \qquad (3)$$

$$(PW\text{-}i\text{-}N) = \frac{1}{CA} = \frac{1}{(1+i)^N} \qquad (4)$$

I shall assume that you own a programmable pocket calculator and so have no need for interest tables. If that is not the case, see References 8, 14, or 20.

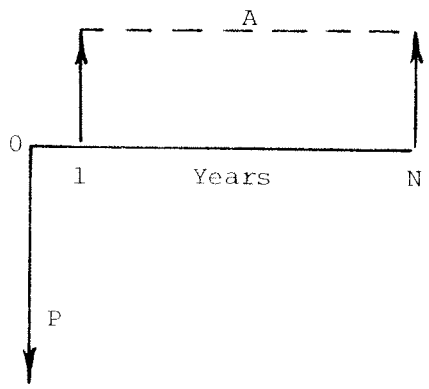
Note: In all of these cases, the cash flow arrows could be reversed without changing our relationships.

Here is a numerical example: Find the present worth of an expected $1000 payment 15 years in the future using an interest rate of 12 percent:

$$PW = (PW\text{-}12\%\text{-}15)0.1827\ \$1000 = \$182.70$$

5.2 Uniform Periodic Amounts

In many economic projections we assume uniform annual cash flows. We know that uniformity will not really occur, but we are unable to predict the ups and downs with any degree of accuracy. Again, any errors that result from this assumption are likely to be the same for all alternatives. If we also assume a single invested amount made on the day before the unit enters service, we have this common cash flow pattern:



If we know the amount of the investment, P , and want to find the corresponding uniform annual amount, A , we use the capital recovery factor, CR :

$$A = (CR\text{-}i\text{-}N)P \tag{5}$$

Conversely, if we know the uniform annual amount and want to find its present worth, we use the series present worth factor, SPW :

$$P = (SPW\text{-}i\text{-}N)A \tag{6}$$

The algebraic values are:

$$(CR\text{-}i\text{-}N) = \frac{i(1+i)^N}{(1+i)^N-1} \tag{7}$$

$$(SPW\text{-}i\text{-}N) = \frac{1}{(CR\text{-}i\text{-}N)} = \frac{(1+i)^N-1}{i(1+i)^N} \tag{8}$$

Here is a numerical example: A $1000 loan is to be repaid over a 10-year period with an interest rate of 15 percent. Find the annual payments.

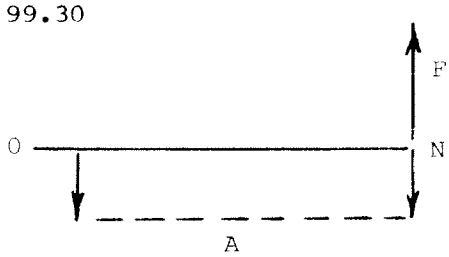
First use Equation 7 to find the capital recovery factor:

$$(CR\text{-}15\%\text{-}10) = \frac{i(1+i)^N}{(1+i)^N-1} = \frac{0.15(1.15)^{10}}{(1.15)^{10}-1} = 0.1993$$

$$A = (CR\text{-}15\%\text{-}10)P \tag{5}$$

$$A = 0.1993\ \$1000 = \$199.30$$

We may also have a pattern in which uniform annual amounts are matched against a single future amount, F . If we know the uniform amounts, A , and want to find the future amount, F , we use the series compound amount factor, SCA :



$$F = (SCA\text{-}i\text{-}N)A \tag{9}$$

Conversely, if we know the future amount and want to find the corresponding uniform annual amounts, we use the sinking fund factor, SF :

$$A = (SF\text{-}i\text{-}N)F \tag{10}$$

The algebraic values are:

$$(SCA\text{-}i\text{-}N) = \frac{(1+i)^N-1}{i} \tag{11}$$

$$(SF\text{-}i\text{-}N) = \frac{1}{(SCA\text{-}i\text{-}N)} = \frac{i}{(1+i)^N-1} \tag{12}$$

Here is a numerical example. Find the uniform annual amounts you will need to bank at 10 percent interest in order to withdraw $1000 in 5 years.

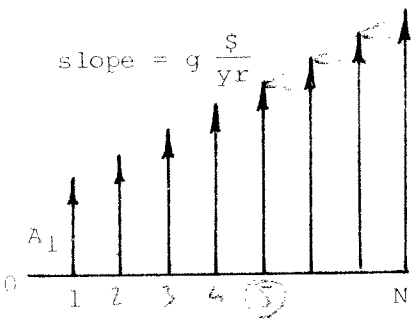
Our first step is to use Equation 12 to find the sinking fund factor:

$$(SF\text{-}10\%\text{-}5) = \frac{i}{(1+i)^N-1} = \frac{0.10}{(1.10)^5-1} = 0.1638$$

Then:

$$A = 0.1638\ \$1000 = \$163.80$$

5.3 Gradient Series



We may be able to predict a cash flow pattern showing a uniformly increasing amount, g , per year. The equivalent uniform annual amount of such a cash flow can be found as follows:

$$A = A_1 + \frac{g}{i} - \frac{Ng}{i}(SF\text{-}i\text{-}N) \tag{13}$$

The present value of the series can then be found using the series present worth factor based on the same values of i and N .

If the pattern shows a uniform downward slope, then the equivalent uniform amount will be:

$$A = A_1 - \frac{g}{i} + \frac{Ng}{i}(SF\text{-}i\text{-}N) \tag{14}$$

Here is a numerical example: Find the present worth of a cash flow that starts at \$1000 the first year and then increases by \$200 per year for another 4 years (i.e., $g = \$200$ and $N = 5$). Use 15 percent interest.

First find the sinking fund factor from Equation 12:

$$(SF\text{-}15\%\text{-}5) = \frac{i}{(1+i)^N-1} = \frac{0.15}{(1.15)^5-1} = 0.1483$$

Then apply the equation for an increasing gradient series:

$$A = A_1 + \frac{g}{i} - \frac{Ng}{i}\,(SF\text{-}i\text{-}N) \qquad (13)$$

$$A = \$1000 + \frac{\$200}{0.15} - \frac{5\ \$200}{0.15}\,(0.1483)$$

$$A = \$1345$$

To find the present worth, first find the series present worth factor from Equation 8:

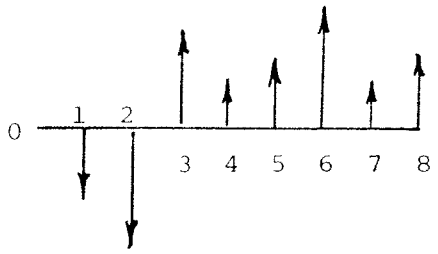
$$(SPW\text{-}15\%\text{-}5) = \frac{(1+i)^N-1}{i(1+i)^N} = \frac{(1.15)^5-1}{0.15(1.15)^5} = 3.352$$

Then:

$$P = 3.352\ \$1345 = \$4508$$

5.4 Random Series

To find the present worth of an irregular cash flow you must discount each amount individually to time zero and then find the cumulative present value. This cumulative amount can be converted to an equivalent uniform annual amount using the capital recovery factor. Such computations are best shown in tabular form.



As a numerical example, let us check the values of P and A found in the gradient problem that we solved a moment ago. Table 1 shows the step-by-step procedure.

TABLE 1

GENERAL SOLUTION FOR PRESENT WORTH AND EQUIVALENT UNIFORM ANNUAL AMOUNT

Year N	Cash Flow	(PW-15%-N) $\frac{1}{(1.15)^N}$	PW
1	\$1000	0.8696	\$869.60
2	\$1200	0.7561	\$907.32
3	\$1400	0.6575	\$920.50
4	\$1600	0.5718	\$914.88
5	\$1800	0.4972	\$894.96
		P =	\$4507.26

(This compares with \$4508 found before)

To find A , first find CR from Equation 7:

$$CR = \frac{i(1+i)^N}{(1+i)^N-1} = \frac{0.15(1.15)^5}{(1.15)^5-1} = 0.2983$$

Then:

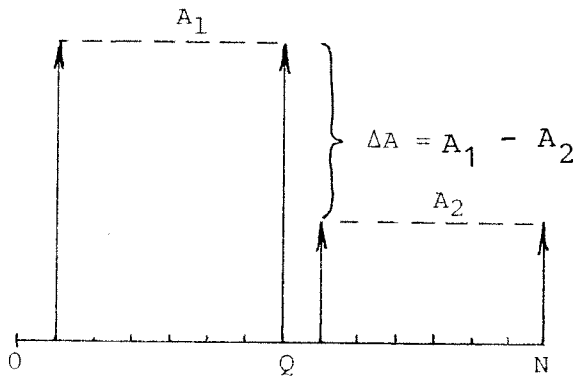
$$A = 0.2983 \ \$4507.26 = \$1345$$

(This agrees exactly with the value found before.)

5.5 Stepped Cash Flows

We frequently find cash flow patterns that exhibit a step up or down (or perhaps several such steps). There are alternative ways to find the present worth. Perhaps the easiest to understand is to:

a) Find the present worth of A_2 for N years, then
b) Add the present worth of ΔA for Q years:



In short:

$$PW = (SPW\text{-}i\text{-}N)A_2 + (SPW\text{-}i\text{-}Q)\Delta A \qquad (15)$$

5.6 Non-Annual Compounding

If interest payments fall due at intervals other than once a year, all of our basic interest relationships still hold true. Simply treat i as the interest rate per compounding period and N as the total number of compounding periods involved. Be warned, however, that monthly compounding at one percent per month places a higher time value on money than does 12 percent compounded annually. References 14 and 15 explain the mechanics of relating various compounding periods.

6. TAXES

Most traditional maritime nations impose an annual tax on corporate earnings. Typical tax rates range from 40 to 50 percent of the before-tax cash flow minus certain tax shields. Notable among the latter is the depreciation allocation, D. In its simplest form (which we call straight-line depreciation), the annual allocation remains constant and is taken as the invested cost, P, divided by the anticipated money-earning life, N:

$$D = \frac{P}{N} \tag{16}$$

Figure 1 shows the distribution of cash based on this simple definition of depreciation, assuming that no other tax shields are in effect. This leads to the following relationship between cash flows before and after tax:

$$A' = A(1-t) + t\frac{P}{N} \tag{17}$$

or

$$A = \frac{A' - t\frac{P}{N}}{1-t} \tag{18}$$

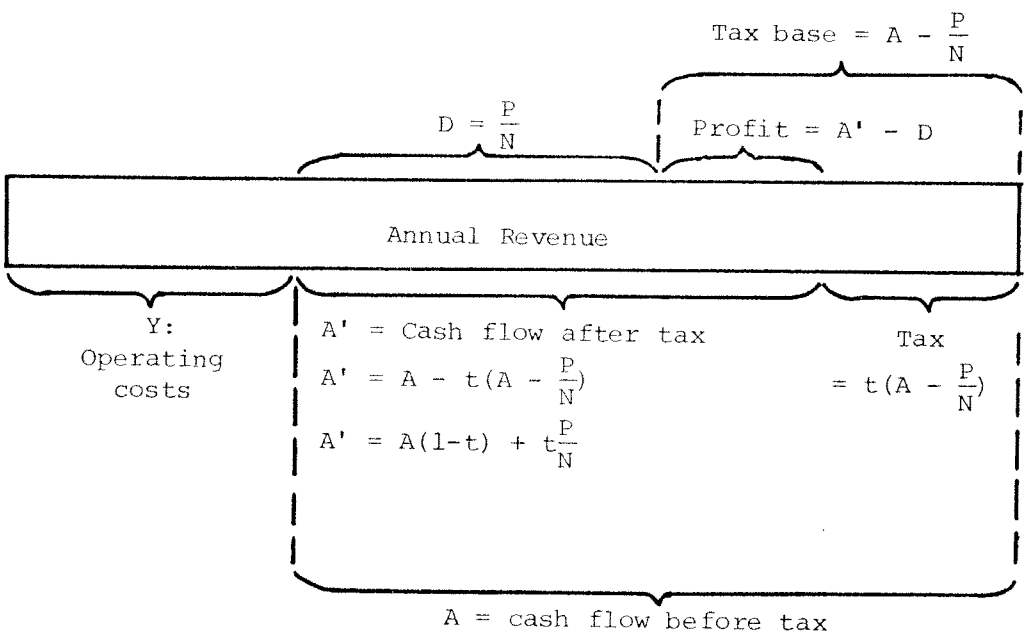


Figure 1: Distribution of Cash Assuming Straight-line Depreciation and No Other Tax Shields

These relationships can be extended to assessing the benefit of an incremental investment intended to reduce operating costs without affecting revenues:

$$\Delta A' = \Delta Y(1-t) + t\,\frac{\Delta P}{N} \qquad (19)$$

where

$\Delta A'$ = incremental gain in cash flow after tax
ΔY = annual saving in operating cost
ΔP = incremental investment.

An important fact to note is that the measures of merit that we shall discuss later are all based on the after-tax cash flow (A') and not on profit (A'-D). Profit is misleading in that it distorts the question of when the cash changes hands.

There are many complications that we could explore with respect to taxes. These include other depreciation plans, tax credits, tax deferral schemes, and the tax shields often associated with fixed interest payments on borrowed capital. As implied in the section on Systems Analysis, however, most engineering decisions are based on the simple concepts embodied in Figure 1. Business managers will almost certainly apply more complex tax and financial arrangements, but not until after they have made their technical decisions based on the simple straight-line, all-equity (i.e., no borrowed capital) approach. For details see Reference 8.

Here is a numerical example in illustration of the simple concept explained above. A fuel-saving device is expected to save $75,000 per year over the ship's remaining life of 7 years. It will add $3000 per year to maintenance costs and $2000 per year to insurance costs. The tax rate is 40 percent. If the owner's time value of money is 9 percent, how much should he be willing to pay for the device, fully installed?

Our first step will be to find the net annual gain before tax: ΔA

$$\Delta A = \$75{,}000 - \$3000 - \$2000 = \$70{,}000$$

Next, we convert that to the after-tax level using a variation on Equations 17 and 19:

$$\Delta A' = \Delta A(1-t) + t\,\frac{\Delta P}{N} \qquad (20)$$

$$\Delta A' = \$70{,}000(1-0.40) + 0.40\,\frac{\Delta P}{7}$$

$$\Delta A' = \$42{,}000 + 0.0571\Delta P$$

To find how much the owner should be willing to pay (ΔP) for that annual gain, we simply find its present worth. First we shall need to use Equation 8 to find the series present worth factor for 9 percent interest and 7 years:

$$(SPW\text{-}9\%\text{-}7) = \frac{(1.09)^7 - 1}{0.09(1.09)^7} = 5.033$$

$$\Delta P = 5.033\ \Delta A'$$

$$\Delta P = 5.033\ (\$42{,}000 + 0.0571\Delta P)$$

$$\Delta P = \$211{,}400 + 0.2874\Delta P$$

$$0.7126\Delta P = \$211{,}400$$

$$\Delta P = \$296{,}700, \text{ say } \$300{,}000.$$

That is the maximum amount the owner should be willing to pay.

7. MEASURES OF MERIT

My purpose here is to define the more commonly used, valid measures of merit in ship design. I shall not present an exhaustive discussion of their relative virtues, nor shall I take time to show what is wrong with various other misleading criteria with which our technical literature is infested. References 2, 4, 6, 7, 9, 12, and 13 discuss these matters in detail, for those of you who are interested.

Any valid measure of merit for a proposed investment will fall into one of three categories depending on whether the analyst wants to assign (vs derive) an interest rate and assign (vs derive) a level of income. Table 2 illustrates this.

TABLE 2

THREE MAJOR CATEGORIES OF MEASURES OF MERIT

Required Assumptions		Primary Measure of Merit	Surrogates or Derivatives
Revenue	Interest Rate		
yes	yes	NPV	NPVI, AAB, AABI
yes	no	Yield	CR, CR', PBP
no	yes	AAC	LCC, CC, RFR, ECT

In this table, the abbreviations stand for:

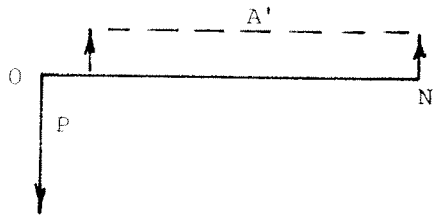
NPV: Net present value
NPVI: Net present value index
AAB: Average annual benefit
AABI: Average annual benefit index
CR: Capital recovery factor before tax
CR': Capital recovery factor after tax
PBP: Pay-back period
AAC: Average annual cost
LCC: Life cycle cost
CC: Capitalized cost
RFR: Required freight rate
ECT: Economic cost of transport

See text for definitions.

The rest of this section is given over to presenting bare-bone definitions of the measures of merit shown in Table 2.

7.1 Net Present Value and Its Derivatives

Net present value, NPV , is one of the most widely accepted and easily understood economic criteria in use today. It requires an estimate of future revenues and the assignment of an interest rate for discounting future cash flows. The discount rate is usually taken as the minimum rate of return acceptable to the business managers. As implied by its name, NPV is simply the present value of the projected after-tax cash flow, including the investments.



In the simple cash flow pattern shown above,

$$NPV = (SPW\text{-}i'\text{-}N)A' - P \tag{21}$$

Net present value has two inherent weaknesses: It tends to favor massive investments and it can be misleading if alternatives have different lives. The first weakness may be overcome by using the net present value per dollar of investment. We call this the net present value index (NPVI) or profitability index:

$$NPVI = \frac{NPV}{P} \tag{22}$$

If alternatives have different lives, NPV will tend to favor the longer lived. That distortion can be eliminated by multiplying each NPV by a capital recovery factor based on the same discount rate, but appropriate to the individual life expectancies. We call this criterion the average annual benefit, AAB :

$$AAB = (CR\text{-}i'\text{-}N)NPV \tag{23}$$

If we take the AAB per dollar invested, that will eliminate both weaknesses in the use of NPV . We call this third variation the average annual benefit index, AABI :

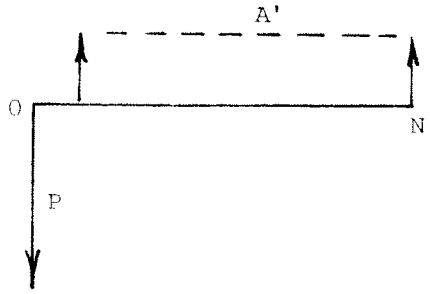
$$AABI = \frac{AAB}{P} \tag{24}$$

7.2 Yield and Its Surrogates

Another widely accepted measure of merit is called here yield. You will also find it referred to as discounted cash flow rate of return (DCF), internally generated interest, rate of return, internal rate of return, profitability index, percentage return, investor's method, equivalent return on investment -- and others.

Like NPV , yield requires a prediction of future revenues. Rather than assigning an interest rate, however, yield asks you to find that rate that will make the present worth of future after-tax cash flows equal to the present worth of the investments. In short, yield is that interest rate that leads to an NPV of zero.

If the cash flow pattern is complex, we are forced to use trial-and-error to find yield. In simple patterns, however, there is an easier way.



Given the simple pattern shown above, we can base yield on the after-tax capital recovery factor, CR' :

$$CR' = \frac{A'}{P} \tag{25}$$

Figure 2 can be used to convert CR' to interest rate, i' , or yield.

Yield avoids the shortcomings of NPV in that it does not give unfair advantage to larger investments or those with longer lives. There are, however, cases where it can be misleading. This is particularly true where the attainable yield differs markedly from the organization's actual time value of money. See Reference 2 for a fuller explanation.

If all alternatives have equal lives, then an examination of Figure 2 will show that the alternative with highest value of CR' will automatically have the highest yield. Moreover, the alternative with the highest capital recovery factor before tax (CR) will normally enjoy the highest capital recovery factor after tax (CR'). This means that CR may be a surrogate for yield.

$$CR = \frac{A}{P} \tag{26}$$

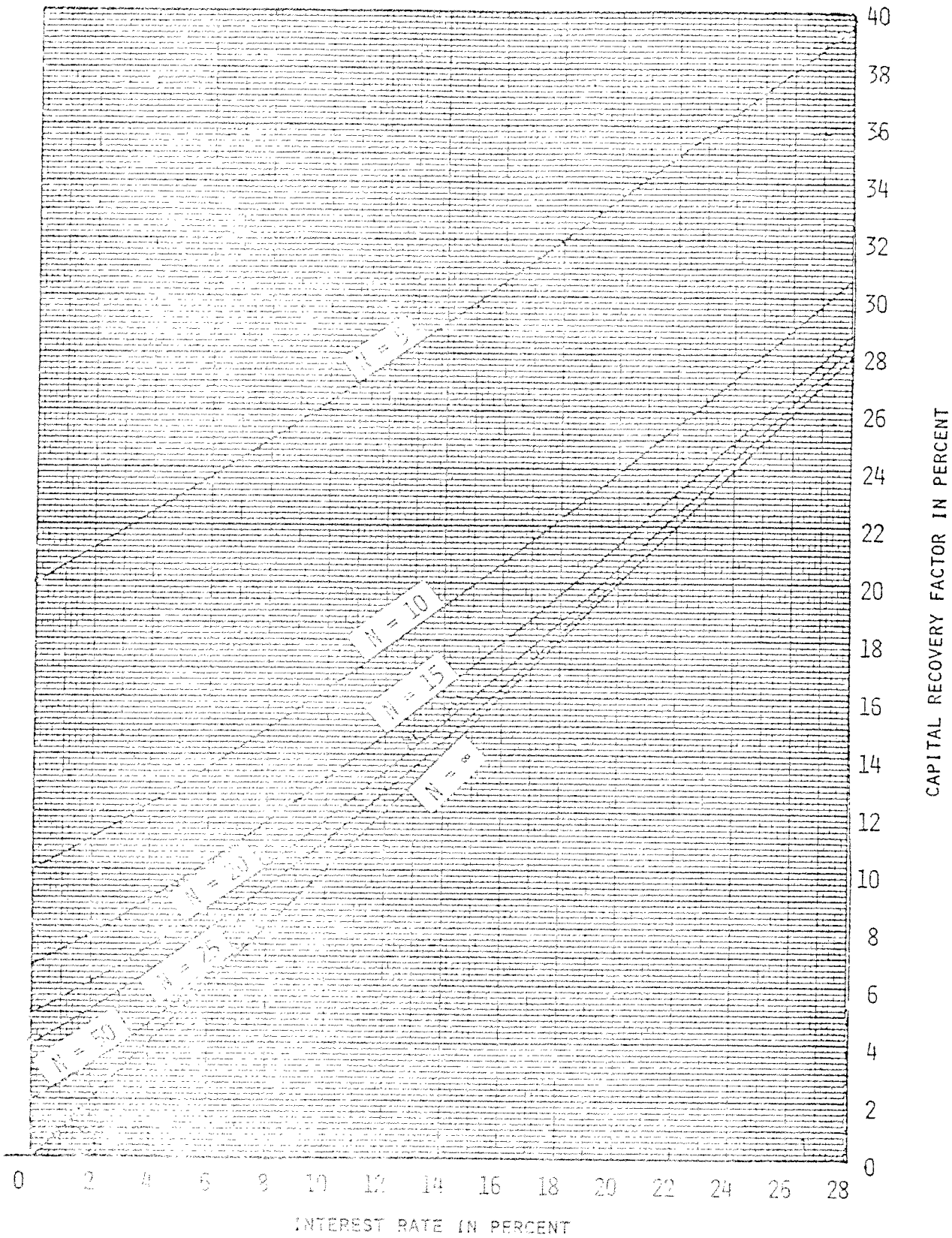


Figure 3. Capital Recovery Factor Versus Interest Rate

The truth of the above assertion is shown next. Recalling Equation 17:

$$A' = A(1-t) + t\frac{P}{N}$$

Dividing through by the investment, P :

$$\frac{A'}{P} = \frac{A}{P}(1-t) + \frac{t}{N}$$

but

$$\frac{A'}{P} = CR' \tag{25}$$

and

$$\frac{A}{P} = CR \tag{26}$$

so

$$CR' = CR(1-t) + \frac{t}{N} \tag{27}$$

and

$$CR = \frac{CR' - \frac{t}{N}}{1-t} \tag{28}$$

Since t and N are the same for all alternatives, CR will vary directly with CR' which will, in turn, vary with the yield, i' .

Another surrogate for yield is the pay-back period, PBP , the number of years required to regain the initial investment:

$$PBP = \frac{P}{A'} \tag{29}$$

As is clear, PBP is the reciprocal of the capital recovery factor after tax, CR' . As such, it shares that criterion's strengths and weaknesses.

Let me stress that these surrogates for yield will be misleading unless all alternatives have equal lives and the standard, simple cash flow pattern (as shown on page 14) holds true. Other assumptions implied in that cash flow pattern are as follows:

1) The tax depreciation period equals the economic life.

2) Taxes are based on straight-line depreciation.

3) There are no bank loans or bonded debts.

4) The investment is made in a single lump sum on the day the ship is delivered.

5) No working capital is required.

6) The ship's net disposal value will be zero.

7) No tax-deferral privileges are used.

8) No investment tax credit is used.

9) Revenues and operating costs will remain uniform throughout the economic life of the ship (in constant-value dollars).

10) There are no major components (e.g. containers) with economic lives that differ from that of the ship.

These are admittedly bold assumptions. Yet, in the majority of ship economic studies they are reasonably safe because the errors induced tend to be much the same for all alternatives. You will find in dealing with many accountants a strong desire to introduce every conceivable real-life complication in infinite varieties and combinations thereof. For the sake of your computer budget and your own sanity, however, you must try to confine such baroque efforts to the final few alternatives. Your initial winnowing should as much as possible be based on simplified models.

7.3 The Nexus Between Yield and NPV

If all alternatives have equal lives, and if all of our above mentioned simplifying assumptions are accepted, then yield and NPVI will point to the same decisions. Here is why: by definition

$$NPVI = \frac{NPV}{P} \tag{22}$$

but $NPV = (SPW-i'-N)A'-P$

so $$NPVI = \frac{(SPW-i'-N)A'-P}{P}$$

$$NPVI = \frac{(SPW-i'-N)A'}{P} - 1$$

but $\frac{A'}{P} = CR'$

therefore $$NPVI = (SPW-i'-N)CR' - 1 \ . \tag{30}$$

Since the series present worth factor should be the same for all alternatives (i' and N being equal), it is clear that the alternative promising maximum CR' will automatically promise maximum $NPVI$. This helps explain a peculiarity of the $NPVI$ criterion, which is that it shows the same optimum regardless of the discount rate used -- which is certainly not true of NPV.

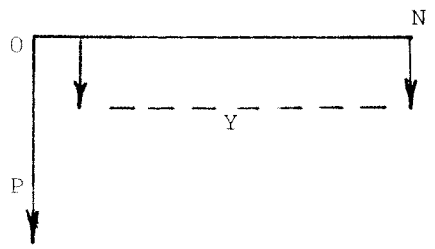
Equation (30) may upon reflection seem rather anomalous. Since SPW and CR are normally thought of as reciprocals, you might conclude that NPVI should always come out equal to zero. That is not the case, however, because as here defined CR' is derived (A'÷P) while SPW is based on an assigned interest rate, and the two will seldom be reciprocals.

7.4 Average Annual Cost

Average annual cost, AAC , is defined as that uniform annual cash flow that is equal in present value to some non-uniform pattern of largely negative cash flows. It usually looks only at investments and operating costs, ignoring revenues. If all alternatives have equal revenues (which includes the possibility of that being zero), then the one with minimum AAC will be the most desirable provided all are of equal capability.

The interest rate used in these calculations is usually taken as some target rate. This would normally be a few percentage points higher than the minimum acceptable rate used in NPV .

In its simplest form, our pattern of costs will be:



$$AAC = (CR-i-N)P + Y \tag{31}$$

The expression (CR-i-N)P is called the annual cost of capital recovery, ACCR . Note that it is based on before-tax interest rates (in case revenues and taxes are both involved). To find CR , we start with the owner's target yield, i' , use that to find CR' (see Equation 7) and then convert that to CR by means of Equation 28.

In more complex cash flows, simply find the present value of the entire cash flow and multiply that by a capital recovery factor based on the same interest rate and for a number of years equal to the unit's life span.

7.5 Derivatives of Average Annual Cost

The present value of the entire cash flow, mentioned just above, is another useful criterion. It is usually called the life cycle cost, LCC . This criterion is closely akin to AAC :

$$LCC = P + (SPW-i-N)Y \tag{32}$$

or:

$$LCC = (SPW-i-N)AAC \tag{33}$$

Life cycle cost suffers one disadvantage in comparison to AAC in that it may be misleading if alternatives have unequal lives. That can be overcome by the use of the capitalized cost, CC , criterion. In this approach we assume

an endless replication of each cash flow pattern and find the present worth of providing such a perpetual service:

$$CC = (SPW\text{-}i\text{-}\infty)AAC \tag{34}$$

but

$$(SPW\text{-}i\text{-}\infty) = \frac{(1+i)^{\infty}-1}{i(1+i)^{\infty}} = \frac{1}{i} \tag{35}$$

so

$$CC = \frac{AAC}{i} \tag{36}$$

Average annual cost, life cycle cost, and capitalized cost are valid only when all alternatives have equal functional capabilities. Where capabilities differ, we must recognize those differences by taking the average annual cost per unit of capability. In a cargo ship, for example, we can find the cost per ton of cargo delivered on a given trade route. This we call the required freight rate, RFR :

$$RFR = \frac{AAC}{C} = \frac{ACCR + Y}{C} = \frac{(CR)P + Y}{C} \tag{37}$$

where

C = annual transport capacity

This is a valuable criterion, much used in the marine industry for studying the feasibility of new engineering concepts or optimizing the details of any particular concept. It tells us, in effect, how much the shipowner must charge his customer if the shipowner is to earn a reasonable after-tax return on his investment. The theory behind RFR is that the best ship for any given service is the one that provides that service at minimum cost to the customer. Implicit here are the assumptions that (a) free market forces predominate in the trade, and (b) all competitors operate within the same frame of capital and operating costs.

In using RFR , you must remember to put capital costs on a before-tax basis. Here is a numerical example in illustration of that point.

A shipowner expects his proposed ship to carry 450,000 tons of cargo per year on a given trade route. The ship will cost $60 million and is expected to last for 20 years. Its annual operating costs will average $1.7 million. The tax rate is 42 percent. How much must the owner charge per ton of cargo if he is to earn a 12 percent yield on his investment? (Note: We shall accept all those simplifying assumptions shown on pages 16 and 17.)

To solve this problem we start by finding the after-tax capital recovery factor, CR' , corresponding to the stipulated 12 percent yield and 20-year life. Equation 7 applies:

$$(CR'\text{-}12\%\text{-}20) = \frac{0.12(1.12)^{20}}{(1.12)^{20}-1} = 0.1339$$

Then we use Equation 28 to convert CR' to CR :

$$CR = \frac{CR' - \frac{t}{N}}{1-t} = \frac{0.1339 - \frac{0.42}{20}}{1 - 0.42} = 19.47 \text{ percent}$$

Now we are ready to find how much the owner must charge, i.e., the required freight rate:

$$RFR = \frac{(CR)P + Y}{C} \tag{37}$$

$$RFR = \frac{(0.1947)\$60,000,000 + \$1,700,000}{450,000}$$

$$RFR = \frac{\$11,680,000 + \$1,700,000}{450,000}$$

$$RFR = \$29.74 \text{ per ton}$$

Where valuable cargoes are involved, we may want to recognize that speed of delivery will affect the inventory cost of the goods in transit. One way to do this is to add the inventory cost per ton of cargo to the required freight rate. We call this modified value the economic cost of transport, ECT .

$$ECT = RFR + \frac{ivd}{(1-t)365} \tag{38}$$

where

i = Interest rate reflecting cargo owner's time-value of money
v = Value of cargo, per ton, as loaded aboard
d = Days in transit

Reference 2 explains the derivation of this concept.

7.6 Comparative Results

Within the marine industry today the three leading measures of merit are NPV , yield , and RFR . Each of these criteria will usually steer us toward slightly different designs. This is largely because each uses a different time-value of money:

NPV uses a minimum acceptable rate,

RFR uses a (somewhat higher) target rate, and

Yield uses whatever rate drops out.

The three criteria will agree only when these conditions are met:

1. NPV and RFR both use the same interest rate, and

2. The revenue used in finding yield coincides with RFR .

7.7 The Incremental Approach

Suppose you have to choose between two alternatives both of which have equal annual incomes -- perhaps zero or perhaps unknown, but still the same. The one with higher first cost is expected to have lower operating costs. Suppose further that each has uniform annual costs and both lives are the same. We can then, if we wish, examine the difference in investment (ΔP) from select-

ing the more expensive unit and its saving in annual operating costs (ΔY). We can derive the corresponding yield or NPV, or any of the other criteria that assume a known income. That is because any saving in operating cost will produce an equivalent increase in cash flow. But be careful. If taxes are involved, the saving in operating cost will be tempered by the tax, which will in turn be tempered by a difference in the depreciation tax shield. As we showed in Section 6 (Taxes), the net annual gain, A', from a saving in operating costs, ΔY, is:

$$\Delta A' = \Delta Y(1-t) + t\frac{\Delta P}{N} \qquad (17)$$

The same concept can be extended to multiple-choice situations, such as optimization studies. To do this, rank the alternatives in order of ascending first costs. Then use NPV or yield (or other) to analyze the incremental cost and incremental cash flow in going from the first to the second alternative. If that meets your standard of profitability, go on and compare the third to the second. Continue this procedure until the incremental cash flow is no longer great enough to justify the incremental investment. That is, for example, until the NPV of the difference becomes negative or the yield becomes less than the cut-off rate.

The analysis-of-increments method as outlined above is sufficient for selecting designs where there is a relatively smooth series of steps in advancing from one alternative to another. Where the progression is untidy, however, you should not quit as soon as you reach the less-than-acceptable measure of profitability. Perhaps benefit is to be gained by going even further. That being the case, examine the differences between the new alternative and the last acceptable one (<u>not</u> the one or ones that failed to meet the standard).

8. SELECTING AN INTEREST RATE

In real life, whenever an interest rate must be specified, that responsibility falls on management. On occasion, however, design engineers find it necessary to take on that task themselves. It is therefore important to develop some feeling for what prudent business managers look upon as reasonable rates. In this, we must keep clear that we are talking about numbers that have been corrected for inflation, i.e., based on constant-value dollars. (See Section 13 for comments on inflation.)

A baseline is the so-called cost of capital. For example, suppose a company raises 60 percent of its capital from the sale of common stock (i.e., equity) and the rest from bank loans. Suppose further that the company aims to earn, as a minimum, a 10 percent return on its equity and get its bank loan at 6.5 percent. Then its cost of capital is found by taking the weighted average of those amounts:

Equity:	$0.60 \times 0.10 =$	0.060
Borrowed:	$0.40 \times 0.065 =$	0.026
Cost of capital		0.086

Given that cost of capital, the typical business manager would round it up to 9.0 or 9.5 percent as his minimum acceptable rate. That would then be the number he would use in finding NPV for some minimum-risk investment. If greater risks are involved, he would assign higher rates. Be sure to note here that any investments chosen are likely to have high positive values of NPV. That implies that the actual rate achieved (i.e., yield) would <u>exceed</u> the cut-off rate.

Assigning the target rate used in RFR is more of an art. In theory, it is the rate that will effect a balance between supply and demand in the trade. Higher levels of profitability will attract more ships and drive down freight rates. Lower levels of profitability will see ships drawn to other trades. A yield (after tax, of course) of 10 percent is normally thought of as a minimum. In relatively risky ventures a figure of 15 percent would be easily justified.

Please be assured, all of the numbers cited above are highly debatable. Use them accordingly.

9. OWNER'S COSTS

The total investment in a ship includes not only the shipyard bill, but a highly variable increment for a variety of other costs. Reference 23 cites these approximate figures for large merchant ships (1978 dollars):

Spare parts	$600,000
Owner-furnished materials	$250,000
Plan approval	$1,000,000
Owner's supervision	$1,500,000
Administration and legal fees	$400,000
Total	$3,750,000

10. PAYMENT SCHEDULES

Remember, too, that the shipyard bill is not paid on the day before the ship starts in service. It is, rather, spread out over the entire building period -- which may be a year or two. There may be some studies in which you should recognize this reality. The principles explained in Section 5 are all you need.

Reference 23 shows some typical payment schedules.

11. LEVERAGE

Many business managers supplement their equity capital with borrowed funds, usually from a bank. This expansion of capital is referred to as leverage. Regardless of the source of these supplementary funds, they normally require periodic repayment of capital plus some fixed level of interest. These claims take precedence over dividends paid out to stockholders. This means that any significant degree of leverage adds to the stockholders' risk. This leads to the common sense conclusion that leverage, when employed, should be aimed at increasing the rate of return on equity capital commensurate with the added risk. In short, this approximate relationship should obtain:

$$f_o i_o + f_B i_B = i' \qquad (39)$$

where

f_o = proportion of equity capital

f_B = proportion of borrowed capital = $1 - f_o$

i' = overall yield on total capital

Solving for i_o :

$$i_o = \frac{i' - f_B i_B}{f_o} \tag{40}$$

Suppose, for example, that a 10 percent yield is thought to be reasonable for an all-equity investment. If 75 percent of the investment is to be obtained from a bank at 7 percent interest, what should be the aimed-for yield on equity, i_o ? Applying Equation 40:

$$i_o = \frac{0.10 - 0.75 \times 0.07}{0.25} = 19\%$$

The main point of this section is that for design economics, we can usually ignore the presence or absence of bank loans. It is true that interest paid to a bank is usually treated as an operating expense, which means it is tax-free. Like those other tax advantages dealt with in Section 6, however, such gains are likely to be ignored by prudent managers in their initial economic assessments. In summary, design decisions should be based on sound economics, not finances.

12. UNCERTAINTY

We must be frank to recognize that all of our cost and income projections involve a good deal of guess-work about the future. We cannot eliminate all risks associated with those uncertainties, but there are logical ways of understanding how much risk is involved. There are also ways of making design decisions that will supposedly minimize risks. References 8, 15, 16, 17, and 22 may be consulted for help in this. You should be aware, however, that truly successful managers cannot explain exactly how they go about making decisions. Intuition clearly plays a part -- and for that there are no equations.

13. INFLATION

The continuing, world-wide deterioration in the purchasing power of paper money lends confusion to economic studies. Much of this confusion is needless. The best way to clarify your mind is to train it to think about constant-value (real) units, rather than face-value (nominal) units.

Consider the basic case where we expect all future costs (operations, taxes, and future replacement) to rise together on a common tide of inflation. Will an expectation of 20 percent inflation lead to a design decision different from an expectation of 10 percent? Let us think in constant-value terms, and assume that the owner will be allowed to let his freight rates float up with inflation. We can then easily conclude that our projected cash flows will be uniform when corrected for inflation. If that is the case we can stick to today's cost levels and apply with confidence any one of our economic criteria that assume uniform cash flows.

There is one frequent exception to the assumption of uniformly rising costs. Many governments do not recognize inflation when figuring depreciation. In constant-value terms that tax shield keeps shrinking. We can quantify this by double discounting to find its present value. The same idea can be applied to any other cost that is changing at a unique rate.

Here is an example. Find the net present value for a ship projected to cost $20 million and to produce annual revenues (in constant-value dollars, CV$) of $6 million. Annual operating costs are $1.7 million, in CV$. The tax rate is 40 percent. Expected life is 20 years. The owner stipulates a

discount rate, r, of 9 percent in CV terms. The expected rate of inflation, d, is 15 percent. Inflation is not recognized in figuring depreciation.

To solve this, we must find the compound discount rate for those cost elements that remain constant in face-value dollars, FV$. In this case, the following rate would apply to the depreciation:

$$i = (1 + d)(1 + r) - 1 \tag{41}$$

$$i = (1.09)(1.15) - 1 = 25.35\%$$

Our expression for NPV then becomes:

$$NPV = (SPW\text{-}9\%\text{-}20)A(1-t) + (SPW\text{-}25.35\%\text{-}20)t\frac{P}{N} - P \tag{42}$$

$$(SPW\text{-}9\%\text{-}20) = \frac{(1.09)^{20}-1}{.09(1.09)^{20}} = 9.129 \tag{8}$$

$$A = Rev - Y = \$6M - \$1.7M = \$4.3M$$

Note: M = million

$$(1-t) = (1-0.40) = 0.60$$

$$(SPW\text{-}25.35\%\text{-}20) = \frac{(1.2535)^{20} - 1}{.2535(1.2535)^{20}} = 3.902 \tag{8}$$

$$\frac{P}{N} = \frac{\$20M}{20} = \$1M$$

$$NPV = 9.129\$4.3M\times0.60 + 3.902\times0.40\$1M - \$20M$$

$$NPV = \$5.114 \text{ million}$$

Another approach to this problem is to prepare a year-by-year table showing all cash flows in constant value terms. Use this to derive NPV, yield, or whatever measure of merit you wish. This method becomes increasing appropriate as scenarios become more complex. Examples are shown in Reference 1.

14. EXPANDED HORIZONS

Throughout this paper we have directed our thoughts to but one ship at a time. In many instances that is a valid approach. In other settings, however, optimizing a single new unit may lead to inefficiencies of the overall fleet. Or, it may lead to inefficiencies in the door-to-door transport system in which the ship is but a link.

As an example of the latter situation, consider the relationship between ships and terminals. What is good for the ship (i.e., large size) may be bad for the terminal, where great costs may be required for dredging and pier construction. Large ships, too, require large terminal storage capacity and that leads to high cargo inventory costs. See References 3 and 18.

Converting from the micro-economics of a single ship to the macro-economics of an entire fleet is seldom a precise operation. Simplifying assumptions are in order, and careful judgment is required. See References 6, 7, 11, 13 and 19.

15. FINALE

I have tried here to present the highlights of what a ship designer should know about practical economics. I have deliberately erred on the side of simplicity so as to make the principles easily understood. In most real-life settings these simple approaches are entirely adequate. Where more elaborate structures are required, these principles should still provide all the building blocks you will need. Guided by common sense and a firm resolve to keep extraneous details in perspective, you should be able to analyze even relatively complex real-life problems. There will be occasions, of course, where additional background knowledge will be required. The appended list of references and additional bibliography should provide much of what is needed.

I have made no attempt to supply factual cost information or details regarding tax structures. Those facts and figures are obviously necessary in any economic analysis. I can only urge you to develop such data on your own, and to keep them current as best you can.

Economic analysis is the vital link between engineering and business management. The naval architect or marine engineer who develops his or her cost estimating techniques and knows how to apply simple economic principles will always find useful and interesting work in the marine industry. Moreover, for the ambitious young engineer who aspires to high managerial position, engineering economics is the first sure step to success.

REFERENCES AND ADDITIONAL BIBLIOGRAPHY

1. Benford, Harry, "Of Dollar Signs and Ship Designs," SNAME STAR Symposium, 1975.

2. Benford, Harry, "A Second Look at Measures of Merit for Ship Design," University of Michigan, Department of Naval Architecture and Marine Engineering, Report No. 223, August 1980.

3. Benford, Harry, "Bulk Cargo Inventory Costs and Their Effect on Design of Ships and Terminals, " SNAME Marine Technology, Vol. 18, No. 4, Oct. 1981, pp. 344-349.

4. Buxton, Ian, "Engineering Economics Applied to Ship Design," Trans. RINA, Vol. 114, 1972, pp. 409-428.

5. Buxton, Ian, "Economic Comparisons of Ships and Their Propulsion," Stone Manganese Marine Conference, 1974.

6. Buxton, Ian, "Application of Engineering Economics to Ships and Fleets," WEGEMT, Sept. 1978.

7. Buxton, Ian, "Ships in the Marine Economic Environment," WEGEMT 1978.

8. Bierman, Harold, Jr., and Smidt, Seymour, The Capital Budgeting Decision, Macmillan, New York, 1971.

9. Chapman, Kenneth R., "Economics and Ship Design," Trans. North East Coast Inst. of Engineers and Shipbuilders, Vol. 83, 1967, pp. 31-46.

10. Erichsen, Stian, "Low Paying versus High Paying Cargo in the Liner Trades," University of Michigan, Department of Naval Architecture and Marine Engineering, Report No. 126, 1972.

11. Erichsen, Stian, "Optimizing Containerships and Their Terminals," SNAME Spring Meeting Papers, 1972, pp. 4.1-4.11.

12. Goss, Richard, "Economic Criteria for Optimal Ship Design," Trans. RINA, Vol. 107, 1965, pp. 581-600.

13. Goss, Richard, Studies in Maritime Economics, Cambridge University Press, 1968.

14. Grant, Eugene L., and Ireson, W. Grant, Principles of Engineering Economy, Ronald Press, New York, 1960.

15. Jelen, F.C., Editor, Cost and Optimization Engineering, McGraw-Hill, New York, 1970.

16. Klausner, Robert F., "What an Engineer Should Know about Capital Budgeting," University of Michigan, Department of Naval Architecture and Marine Engineering, Report No. 083, May 1970.

17. Krappinger, Odo, "Relationships between the Profitability and Safety of Ships," University of Michigan, Department of Naval Architecture and Marine Engineering, Report No. 068, 1969.

18. Mentz, Paul B., "What Price Speed?" SNAME Chesapeake Section, Sept. 1975.

19. Nachtsheim, John, "Ship System Economics -- Less the Ship," SNAME Spring Meeting Papers, 1972, pp. 1.1-1.17.

20. Newman, Donald G., Engineering Economic Analysis, Engineering Press, San Jose, 1976.

21. Scher, Robert M. and Benford, Harry, "Australian Bulk Carrier Design and Operations in an Era of High Fuel Costs," Sea Transport Technology 1981, Univ. of New South Wales, Kensington NSW, Australia, 1981, pp. 117-139.

22. Sen, Pratyush, "Optimal Ship Choice under Uncertain Operating Conditions," Trans. RINA, Vol. 120, 1978, pp. 137-148.

23. Swift, Peter M., "A Shipowner's View of Engineering Economics," WEGEMT, Sept. 1978.

24. Walsh, Robert G., Jr., "Estimated Return-on-Investment of Oil Tankerships," Marine Technology, SNAME Vol. 15, No. 1, Jan. 1978, pp. 51-74.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

DESIGN CHOICE FROM SHIPOWNERS POINT OF VIEW

by

E Vossnack

Nedlloyd Fleet Service

Rotterdam

PART I DESIGN CHOICE FROM SHIPOWNERS POINT OF VIEW.

How to cope with the demands of transportation ?

Liner Fleet Retonnaging.

The container is dominating

- The container,whether door to door, or pier to pier, is now generally accepted and in many instances demanded by the customer. Even for trades which in fact are not yet ready for this concept.

Dying cargoliner

- The sophisticated high-speed-cargoliner has phased out; a small amount of boxes and a high fuel consumption, insufficient stability, no containergear, no space on deck, too weak hatchcovers, no square space in holds.
  Union purchase derrick gear, heavy derrick, deeptanks, reeferspaces are of no use anymore.
  3 to 4 Days average stay in port is too long for efficient transportation.

Large cell. container vessels

- Introduction of the large cellular containervessels operating on trunklines between terminals equipped with container gantries, resulted in a far higher transport production per year and consequently a lower transportation cost per cubft.

Coöperation

- In order to offer a regular pattern of sailings with rationalised frequency to cope with the cargo flow, shipping companies have to co-operate closely which may lead to formation of consortia by which the results can be maximized. Shipowners individuality is, however, more or less absorbed by such consortia.

Weekly service

- In case the speed is sufficient to call 6 ports in Asia and 6 ports in Europe in a 70 days roundtrip, than a weekly service is offered to the customer.
- Only this system allows for economy of scale and reduction of costs.
- Feederlines connect wayports with the mainports of the trunklines.

Large RoRo container vessel

- Meanwhile large RoRo containervessels have been introduced as well which can cope also with cargo commodities like steelplates, pipes, ingots, paper, drums, pallets, rolling stock, which cargo is less suitable for containerisation but can easily be carried on platform or trailers.
  These ships are more flexible in the trade but require more skill and ingenuity of the crew to obtain the maximum benefit.

M.P.vessels

- Since in many developing countries the infrastructure is not (yet) geared to the container era, this type of vessel - suitable to carry considerable numbers of container and/or bulk (gen.cargo as well as loose bulk commodities) - is probably for the transitional period the best answer for

  trades between industrial countries on one side and under-developed countries on the other side of the world (for example Japan - W.Africa). One way traffic with loaded containers Westbound to W.Africa will accummulate empty boxes in W.Africa.
  Eastbound - carriage of tropical products will probably go in the oldfashioned way.

Problems with stowage	- The empty containers should be returned to Japan, but helas, there is on a multipurpose vessel hardly any space for stowing empties if the vessel is filled with logs and cotton. Adding an extra midbody to the vessel could solve the problem for a limited period of time.
Too long in port	-Breakbulk and boxes are in conflict with each other on board and on the quay,which after all causes extra costs and many delays, results in a long portstay and a long turnaround time.
Expensive vessel	- The production of the multipurpose vessel will always be less in comparison to fully cellular and RoRo, meanwhile the m.p.vessel is complicated and relatively expensive (per Teu). The m.p. vessel will therefore loose on the long run.
Waiting for port development	- The first requirement which has to be fulfilled is the development of ports in West and East Africa - India, Indonesia - West and East America. We are awaiting for dredging in ports lacking depth and length for turning the vessel. We are waiting for suitable quays, containercranes and parking areas.
Switch to another trade	- If these facilities or part thereof have been provided for the trade will switch over from conventional to container transport very quickly. Considering the severe competition between liner operators it is for the shipowner of enormous importance that he can change one vessel from one trade into another in a very short time.
Flexibility of vessel	- Vessels features should be such that the ship can serve almost all over the world and cope with the normal liner trade requirements. - The smaller type cellular vessel for the future should be very flexible and try to match or supplement her RoRo sister. - This means that at a maximum capacity a small draft and a short length overall should be persued. - A maximum beam will allow for a huge deckload containers because of vessels stability. - Maximum manoeuvrability should be obtaind by bow- and sternthrusters plus a large rudder.
Gear on/off	- Installation of a gantry crane or alternatively 2 or 3 "plug in bay" jibcranes will give the ship her own gear and independance from underdeveloped port equipment. - After a certain period of time the port might be equipped with a portainer or the vessel should shift to another trade and can be converted to gearless within one to altern, 3 days.
Reefer containers	- The "worldwide" containervessel should have special facilities for containers loaded with frozen cargo; chilled cargo, or cargo which needs to be ventilated, temperature and moisture controlled. - The vessel should have space below deck to carry these reefers undisturbed as the alternative of stowage on hatchcovers will cause many shiftings of the container during loading/discharging operations in the intermediate ports.

- The integral reefer container is to be preferred over the ducted porthole box since the integral box is independent of the vessel and needs only to be plugged in. Provided sufficient aircooling is available in the hold (altern. watercooling), the integral reefer can be kept cold during ports stay and feeder-transport by plugging in only. Meanwhile the capital investment in the reeferbox is independant from the capital investment in the ships which makes the use of such equipment more flexible.
  Condition is however, that before the loaded integral reefer container goes on board, her machinery should be carefully checked (PTI)
- To the contrary, when a large quantity of fruit or frozen cargo is to be carried regularly and no terminal stay, and/or feeder transport in a hot climate will occur, then the system of insulated porthole boxes, connected to the ships cooling ducts is to be preferred, because of the simple reliability.
  In that case, however, we design a special ship suited to a special trade which is contrary to the required flexibility by management.

Breakbulk

- Special facilities for breakbulk below deck on tanktop can be considered underneath the container stacks to improve flexibility.

Deckload

- Hatchcovers to be strong and rigid enough to carry modules and long and heavy lifts which could be loaded by floating cranes.

Lashing

- Container stacking on hatchcovers up to 4 (5) tiers high. and preferably lashing by "twistlocks" only if the stackload upon the lowest box does not exeed 30 Ton. Over 30 Ton diagonal rodlashings are necessary to prevent racking on the lowest box.
- It must be realised that on many ships 50% of the cargo is carried upon hatchcovers as "deckload" and lashings are of enormous importance to safety of cargo, crew and vessel.

Rolling

- In this respect rolling should be avoided which should be done as far as possible by loading the vessel up to the right metacentric height (0,6 - 0,8m).

Range of stability

- Sufficient freeboard (upto 35° heeling angle) is to be strived after in order to give the vessel a sufficient range of dynamic stability.
  Too many smaller type container vessels do not comply with these basic requirements.

Special shiptypes.

From page 46 - 150 many special shiptypes are described, which have to take over the transportation which was carried out upto 1970 by the conventional cargoliner.

- container RoRo for protected sea and limited waterdepth.
- open coaster.
- heavy lift vessel.
- bow ramp landing craft.

- container feeder for coastal trade.
- container transporter over rivers and canals.
- small chemical tanker.
- 38000 product chemical tanker.

Ship Dedicated to the trade.

From page 150 - 163.

- Main figures necessary for a feasibility study what type of ship is most suitable for the trade.

Improving economy of the ship itself.

From page 167 - 230.

- Physical aspects of the liner vessel and improvements on economy,
  - speed
  - stability
  - course keeping
  - block coefficient
  - smoothness of hull
  - hullform
  - rudder
  - thrusters
  - fin stabilizers
  - propeller
  - shaftgenerator.

Example of Universal Containerschip "Nedlloyd Clarence".

PART II AFTERBODY DEVELOPMENT.

Will regard to propulsion, fuel consumption and vibration + noise problems.

- Many single screw vessels suffer from propeller exited vibration, causing cracks, damage to instruments and hindrance to the crew (in particular with the accommodation deckhouse "All-Aft").

- By application of a larger propeller diameter in combination with lower r.p.m., a higher propulsive efficiency can be obtained, however, what about the vibration troubles?
  Generally these troubles will become smaller when lowering the propeller tipspeed and tipload.
  It is a must to adapt the afterbody to the larger "wheel".
  How to shape the hull,however, how to keep the propeller immersed in ballast condition?
  How much tip clearence should be given in relation to wakepeak and tiploadpeak of the propeller ??

- Systematic step-by-step research should be carried out to obtain the influence on hull resistance, propeller efficiency, hull efficiency, cavitation pattern and hull presure fluctuations.

- The flow pattern around the afterbody is dominant to the problem and in our opinion flow separation should be avoided by all means (which is very difficult on full body ships).

- The application of pram shaped hulls with a slender gondola underneath, accommodating the dieselengine is promising.
  The weakpeak at 12 o'clock should be minimal.
  The propeller should turn in the least disturbed flow.
  On the real ship, (scale 1:1) this arrangement will require minimum horsepower and excite minimum vibrations generally speaking.

- Good and bad shapes are laid down in figures :
  Single screw, full body, slow ship (bulkcarrier - tanker - coaster).
  " " finer body (cargo liner) (tunnel - fins).
  " " container vessel.
  " " roro vessel.
  Twin screw , ferry (single skeg versus twin skeg).

- Fuel consumptions figures "an service" of container ships fitted with 2-3 propeller - diesel and steam are given.

- Single screw and twin screw are compared for the future.

PART III ABOUT STEERING.

Described are steering problems with twin screw ships having differences in arrangements :

- Twin screw, single balance rudder.
  Twin screw, single skeg rudder.
  Twin screw, twin rudder.
  Single screw, single rudder, open skeg, propeller on pipe.
  Triple screw, single rudder.
  Outward/inward turning propellers : swinging with stern.
  C.P. propellers.
  Bow- and stern thrusters.

FNB/E. Vossnack
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Propulsion Plant Choice for a Medium-Sized Container Ship

by

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PROPULSION PLANT CHOICE FOR A MEDIUM-SIZED CONTAINER SHIP

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1. Introduction

The dramatic development of fuel prices in the last decade and the consequences for shipping are generally known. Energy-saving ship propulsion plants are urgently required by shipowners. A detached judgement of all relevant factors is the precondition for the right choice. In view of the complexity of ships' design and the severe competition, decisions are not always that easy to make, in particular, when besides the operating economy of a propulsion plant other parameters, like e.g. cost price, reliability, space requirements etc. could play an important role.

The latest studies of Prof. Gallin have to be looked at in this connection. He started in 1979 (Fig. 1) with the question of employing reduction gearboxes in marine propulsion plants with two-stroke diesel engines to achieve better propeller efficiencies [1]. Logically, the application of reduction gearboxes and the efforts to reduce not only the operating cost, but also the purchase cost, lead to considering the four-stroke engines as well [2].

The favourable results of this engine type in an economic comparison lead to consider twin-engine installations. The subsequent third study [3] on the basis of a concrete example, a "Panamax" bulk carrier, analysed the twin-engine installation with four-stroke diesel engines, reduction gearbox and CP propeller. Apart from the slowly running propeller, the economy of the waste heat recovery was investigated:among all positive characteristics of the examined alternatives the operational flexibility of the twin-engine installation stood out, i.e. the possibility to economically use only one engine when sailing in ballast, or for "slow steaming".

The consequences from one type of ship not necessarily apply to other types, as the design and economy parameters may be very different. Furthermore, the requirements of the marine market are constantly changing.

So the demand arose to investigate other types of ship in a fourth study. Following the trend of the marine market, the topic "The Container Ship" was chosen.

2. The Container Ship

The considerable time saving in container handling would be of no sense without a sufficient speed of the ship. Generally it can be stated that for container ships a speed, which exceeds that of a normal freighter is still a demand, in spite of the high fuel cost. This, however, requires a higher propulsion power. Higher output, however, implicitly consumes considerable amounts of fuel, the propulsion plants therefore have to be optimized, and they have to save energy. Neither must they strain the profitability of a ship, which already is expensive, even more by being very costly.

Furthermore, it is not easy to find sufficient space for bunker tanks in the slim form of a container ship, as considerable water ballast has to be taken on board for reasons of stability.

On container ships, the containers are loaded and unloaded by the lift-on lift-off method. This requires a short engine room. For the same reason

superstructures are located above the engine room. As further space is required below deck for the guides for container stowage, and as space losses between the round ship's form and the rectangular container blocks are unavoidable, the space requirements have priority. This type of ship therefore must have an engine room as small as possible.

Due to the quick loading and unloading, the laying-days in port are rather short for a container ship. This is favourable from the economic point of view, creates however some difficulties. Besides social problems with the crew, the time for maintenance and repairs of the ship or the propulsion plant in the harbour has become shorter. Constant supervision and maintenance, which have to be effected during operation at sea, therefore have become more important and can be another essential argument when choosing the propulsion plant. Also in this regard, twin-engine installations are of particular advantage.

Another factor in determining the engine plant of a container ship is the relatively high required auxiliary power. The reasons are as follows: the container ship is designed according to space requirements. This leads to large dimensions and an excessive freeboard. The deck cargo - several layers of containers - opposes large surfaces to the wind. This complicates manoeuvering in port and at sea, so that efficient bow thrusters have to be provided. Another reason for the increased generator output is the demand which is heard more and more often nowadays for refrigerated containers. Furthermore, the cargo, which was packed into containers on land is not always suitably stowed for a sea voyage. In particular containers arranged on deck, and their lashings, shall not be exposed to too strong rolling motions. On large ships, active stabilizing devices with fins are used against rolling, which also need auxiliary power. Finally, trimming and heeling may occur during quick loading and unloading of containers, which affects the operation itself. To avoid this, compensation tanks are provided, which require high pump capacity over a short time. The cheap generation of electrical energy by means of shaft generators therefore has become a standard for modern container ships.

The most important of these contradicting factors for the design of a container ship are shown in Fig. 1. The present study is based on an actual example, concerning space and investment requirements.

3. An actual example

As an example, a medium-sized container ship (1450 TEU) was taken. According to the yard's offer the ship was to be equipped with a two-stroke crosshead diesel engine of 11.300 kW (15.370 BHP) at 106 rpm. This should result in a trial speed of more than 18 kn at 82 % max. continuous rating (MCR).

The corresponding ship's form is slim, the block coefficient is between 0,61 and 0.62 depending on loaded condition. The ship's lines taper off to the aft. The stern bulb, which was fitted to obtain a uniform wake pattern, at the same time creates more room for the main propulsion plant in the aft body.

4. The task

Among others, there are two important questions the shipowner may ask:

1. Are there propulsion plants which fulfil the requirements, but are lower in initial and operating cost?
2. The charter revenues of the ship are determined according to the actual speed. Is there a possibility to install an even more powerful propulsion plant in the same engine room, and if so, how much would it cost, what will be the ship's speed, and what is the fuel consumption rate per day?

In both cases numerical results are expected.

Furthermore the owner does not want to overload the main engines and, in general, operate in optimum conditions with regard to specific fuel consumption i.e. at 85 % MCR. For service speed an allowance of 20 % above the output necessary at good weather should be considered. Finally, in each case 1130 HP have been branched off from the main engine(s) for a shaft generator of 790 kW.

5. The execution of this study

The present study investigates six propulsion plants, three each for each of the above two questions. These six alternatives have been marked with the letters A to F.

The first three alternatives A, B and C were supposed to have a similar output and answer the first question. Table I shows their technical data, recorded ship's speed and fuel consumption rates one besides the other. Table II shows the total purchasing cost of a propulsion plant, consisting of main engine, gearbox and clutch coupling (for indirect drive), shaft and propeller, for two versions, i.e. fixed or CP propeller. Naturally, only the CP propeller has been indicated for the twin-engine installation of alternative C, as otherwise the engine could not be run fully loaded in single-engine operation. It must not be overlooked that an economic operation with only one engine is possible with a twin-engine plant. Therefore, the two possibilities of alternative C are separately listed in Table I as alternatives C1 and C2.

For the output range of Tables I and II a propeller diameter of approx 6.4 m results for direct drive. The propeller speed is open for indirect drive, but the maximum propeller diam. (Dp) is decisive. This again depends on the lowest draught (T) in the aft ship. In the present example this minimum draught is 8 m in ballast. As a container ship seldom is operating under ballast conditions only, a Dp/T ratio of 0.9 has been accepted, resulting in Dp_{max} = 7.3 m. For loaded conditions of the ship the Dp/T ratio therefore is approx 0.7, which can be regarded as optimum. This situation, which may differ from case to case, is utilized best by the indirect drive. The lower propeller speed of alternative C results in a higher propulsion efficiency than that of the alternatives A and B.

Efficiency gains or losses are shown in Table I and - to simplify the matter - incorporated into the specific fuel consumption. Furthermore the specific consumption is corrected for the lower energy contents of the fuel in practice, and thus the daily fuel consumption of the ship is calculated. For a more exact comparison the consumption rates would have to be referred to the same ship's speed (last line in Table I). In this example, however, the owner was dependant on absolutely achievable values.

Alternatives D,E and F (with increased output) were handled similarly. The results are shown in Tables III and IV. Remarkable in this group is alternative E, which was made of two four-stroke engines with different output, according to the "father-and-son" principle. Thus three output ranges result, and each can be run at optimum conditions (alternatives E1, E2 and E3).

Finally alternative F represents the installation with the highest output that can be accommodated in the existing engine room.

Fig. 2, 3 and 4 give an impression of the space conditions in the engine room. The three figures show overlapped alternatives D and E very clearly.

6. Discussion on the results

6.1 Lower output range

The results of the first three alternatives A, B and C, for the lower performance range, show that the ship's trial speeds very between 18.1 and 18.4 kn, i.e. only by 0.3 kn. Alternative B consumes 3.1 t/day or 810 BHP at 85 % MCR more. Alternative C1 (indirect drive) consumes only 0.3 t/day more for a ship's speed higher by 0.2 kn. This is due to the lower propeller speed (70 instead of 100 rpm), in spite of losses in the gear (2 %).

It should be noted that when operating alternative C2 with only one engine, the ship's speed decreases by 3.1 kn (approx 17 %), whereas the fuel consumption is reduced to 51 % (approx half). This again proves the economic advantages of "slow steaming".

For a correct comparison the same ship's speed has to be assumed in all three cases. In this study, 18.1 kn was taken, in accordance with alternative A. This means the same DHP_2 or BHP_2 all over, meaning that indirect drive (alternative C1) is increased by 0.02 x DHP_2. The different propeller speeds are already incorporated into the corrected specific fuel consumption. The daily consumption rate of alternative B is 0.4 t/day above that of alternative A, and that of alternative C1 is 1.6 t/day below that of alternative A.

What about the initial costs? The prices in this study were inquired to the best of knowledge and belief, but have to be considered with care. All values apply for a delivery of the parts in the first half of 1983.

It should be stressed that prices can be negotiated to a wide extent, depending on the situation of the bidder, so that the results of a comparison of prices of different bidders have to be put into the correct relations.

For the three alternatives A, B and C and in case the ship is equipped with a CP propeller, Table II proves that alternatives A and B cost about the same, whereas alternative C (indirect drive) is less expensive by US $ 407.000, as the extra price for the two-stroke engines and the shaft generator gearbox exceeds by far the additional costs for two clutch couplings, the reduction gearbox, the bigger shaft and the larger propeller.

It is remarkable that there is hardly any price difference between the installations with fixed or CP propeller (as Table II shows). The reasons are

first the far more expensive shaft generator with thyristor rectifier of the propulsion plant with fixed propeller, and second the fact that one spare blade is sufficient for the CP propeller, whereas the owner in this case requires an absolutely equivalent spare fixed propeller. Generally, and the more in case of price equivalence, the owner prefers a propulsion plant with CP propeller. For manoeuvering advantages of a CP propeller and the possibility to relief the engine he puts up with 1 - 2 % efficiency loss. Also a higher service speed of the ship, achievable by adequate propeller pitch setting according to the increased resistance in bad weather or for higher output respectively, can be recorded. At part load, however, the propeller speed of the propulsion plant with CP propeller, which is coupled to the PTO because of the shaft generator remains constant. Therefore the lower propeller efficiency and the higher specific fuel consumption of the main engine have to be taken into account.

6.2 Higher output range

Alternatives D, E and F for the higher output range (Tables III and IV) have to be regarded in the same way.

The ship's speed achieved with alternatives D and E1 are nearly equal (19.0 or 19.1 kn respectively). Alternative F, however, permits substantial increase (19.7!). In this example the block coefficient of the ship is even optimum for the increased speed.

Nevertheless the daily consumption rate of alternative E1 is lower by 1,0 t. The losses in the gear are covered by the more favourable propeller speed of 76 rpm compared to 100 rpm.

For 0.7 kn alternative F needs 2890 BHP more. This means 5.7 t fuel additional daily consumption, including the more favourable efficiency of the slow speed propeller (80 rpm). It has to be remembered that the higher the speed, the harder the conditions.

Also in this group of installations with increased output, the alternatives E1 and F with the twin-engines are of great interest at part load. The alternative E1, the so-called "father-and-son" plant, can be run under optimum conditions, i.e. 16.4 kn with the "father" (Alt. E2) and 15.0 kn with the "son" (Alt. E3). The fuel consumption rates are 29.0 and 23.3 t/day.

The alternative F, at single-engine operation, is acting like alternative E2 with only the "father" engine running.

Also for this performance range, the correct comparison has to be made with the same ship's speed (last line of Table III).

Daily consumption rates of 52.0, 50.2 and 50.0 result for the alternatives D, E1 and F. For alternative F, running at part load (73 %) at this speed, a specific fuel consumption higher by 1 g/BHPh has been assumed.

For the same ship's speed (19.0 kn) the consumption rates of the plants with four-stroke engines (E1 and F) are more favourable.

Regarding initial investment (Table IV) also in this performance range the versions with fixed or CP propeller cost the same. Thus, this important statement is proved again.

Among the propulsion plants with a CP propeller, alternative E is cheaper by US $ 658.000, compared with alternative D. The reasons are the same as for the lower performance range. It is of particular importance that the by far stronger alternative F costs more or less the same, or even slightly less, than alternative D. The owner receives free of charge - as regards the construction cost - the additionally installed 3400 BHP (at 100 % MCR) or 2890 BHP

(at 85 % MCR) respectively, and thus a possible increase of 0.7 kn ship's speed.

7. Aspects of space

The Fig. 2, 3 and 4 clearly show the space conditions.

Only alternatives D and E were marked, but on the basis of the mentioned figures it is quite easy to imagine that also the other alternatives A to F will fit into the same engine room.

The two four-stroke engines naturally are shorter, but with the necessary couplings and gears they have nearly the same length as the slow-speed two-stroke engine (Fig. 2).

The twin-engine four-stroke installation requires a bit more width than the single-engine two-stroke installation (Fig. 3 and 4), however, the situation is not critical, which is also due to the stern bulb. This is shown in Fig.3 with the outlines of frames 24 and 27.

There is sufficient space between the hull plating and the back edge of the engines as well as of the reduction gears.

The important difference lies in the required height for both kinds of propulsion plant (Fig. 2 and 3). The necessary cylinder removal height of the two-stroke engine surpasses the main deck (15.8 m above the base). This requires room up to the first superstructure deck. Compared to this, the four-stroke engine only requires space up the the ship's lower deck (9.6 m above the base). The outlines of the gained engine room space for the latter propulsion plant can easily be recognized in the drawing. Roughly calculated, the volume is as follows:

$$12 \times 9 \times 9 = 972 \text{ m}^3$$

Assumed that a four-stroke plant requires only

$$4{,}5 \times 9 \times 9 = 362 \text{ m}^3$$

then

$$\text{approx } 600 \text{ m}^3$$

remain for other purposes.

Naturally the space above the propulsion plant cannot be used for every purpose, but by re-arranging the auxiliary engines and the aggregates, free space can be provided somewhere else. This would give way for bunker tanks in the engine room. 600 m³ are approx one third of the total bunker capacity of the ship! Apart from the space requirements on board of a container ship, such a re-arrangement is particularly important for the disposition of bunker tanks in or near the engine room, in connection with the energy-saving heating of the tanks with the waste heat of the cooling water (extended waste heat recovery). Furtheron, in case of more space available, the form of the settling and day tank can be made more favourable.

8. Summary

This study has shown that the choice of the propulsion plant should not be made thoughtless , nor with prejudice.

For the majority of ship types - and in particular for a container ship - the propulsion plant is the most important factor, the "heart" of the system "ship". The complexity of the propulsion plant, the advantages and disadvantages and last not least the initial costs deserve a careful analysis, based on exact figures.

This study shows that the twin-engine installation with large-bore four-stroke diesel engines, on the above conditions, can offer real functional and economical advantages for a medium-sized container ship over a single-engine two-stroke installation.

9. Words of gratitude

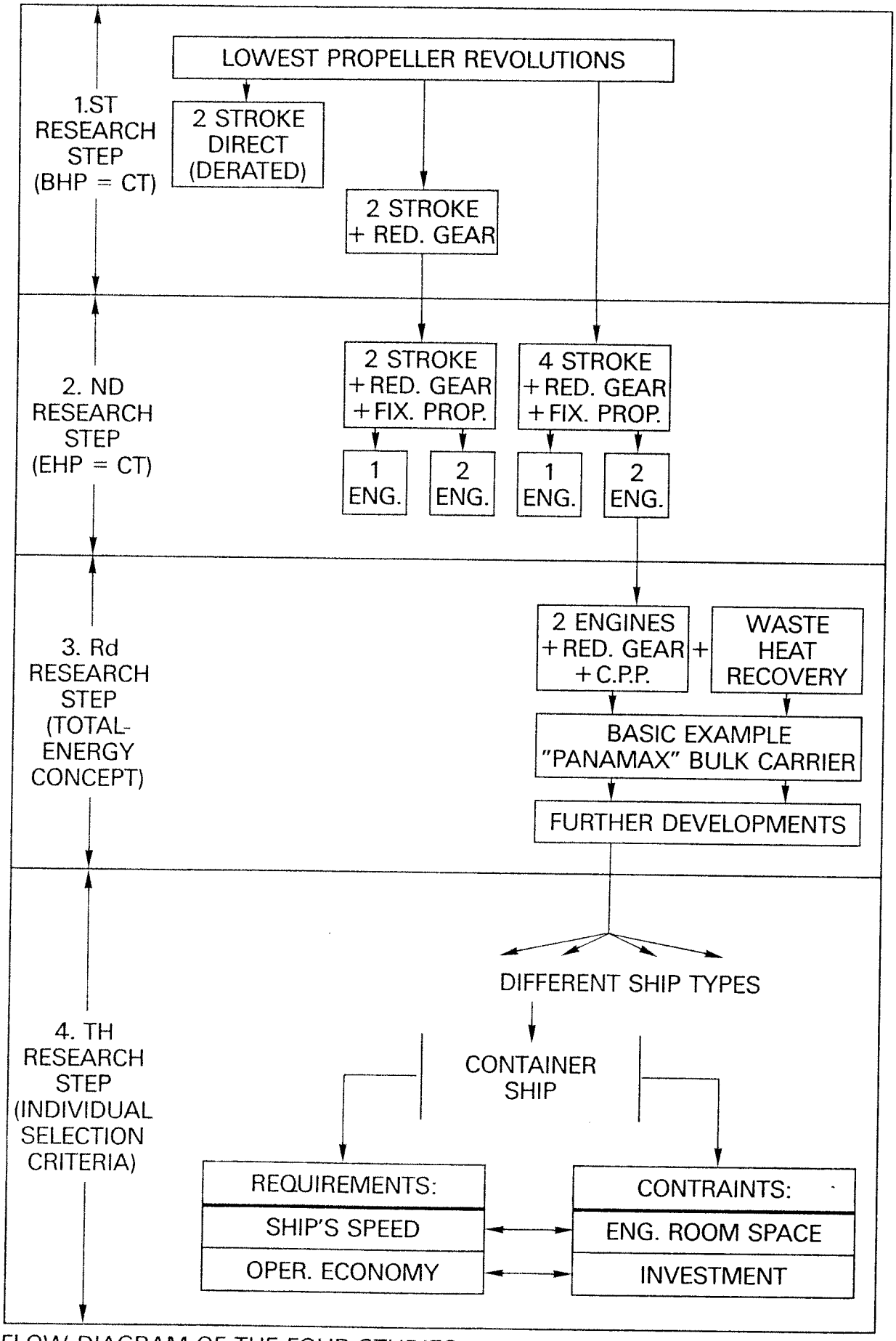
The author owes many thanks to Mr. Horst Krüger, owner of the shipping company Hans Krüger GmbH in Hamburg, for his help and skillful advice in shipping matters. Sincere thanks to Messrs. Rolf Schiffmann of the shipyard Thyssen Nordseewerke GmbH in Emden, Peter Nissen of Krupp MaK Maschinenbau GmbH in Kiel, K.-H. Siefert of Lohmann & Stolterfoht GmbH in Witten, H. Wolf and J. Tornblad of KaMeWa in Kristinehamn and H.P. Fransen and P.W.C.H. van Oirschot of Lips B.V. in Drunen. Thanks also to Mrs. P. Littek for the fast and exact translation and typing.

10. Reference List

1. Gallin, C., "Fuel Economy, Propulsion Efficiency and Diesel Engine Installations", The Motor Ship, September 1980.

2. Gallin, C., K.-H.Siefert, Heiderich, O., "Alternatives for Economical Diesel Ship Propulsion", The Third International Marine Propulsion Conference, London, March 1981; The Motor Ship, May 1981.

3. Gallin, C., "Advanced Energy Saving Concepts". Paper presented before the Danish Society of Naval Architects and Marine Engineers, Copenhagen; before the Norwegian Shipowners Association, Oslo; before the Marine Engineering Society in Japan, Tokyo and Kobe; as well as before the "Ship Costs & Energy '82" symposium, New York 1982; published in "Schip en Werf", No. 25, 1982.



FLOW-DIAGRAM OF THE FOUR STUDIES

FIG. 1

ITEM		UNIT	ALTERNATIVE			
			A	B	C_1	C_2
MAIN ENGINE(S)		MAKE	B&W	SULZER	MaK	MaK
		TYPE	5 L 80 GFCA	6 RLB 76	2 x 6 M 601	1 x 6 M 601
ENGINE POWER	100 %	BHP_1	15370	16320	16320	8160
	85 %	BHP_2	13060	13870	13870	6940
ENGINE SPEED	100 %	RPM	106	110	425	425
	85 %		100	104	425	425
SHAFT GENERATOR[1]		KW	790	790	790	790
TRANSMISSION EFFIC.		–	0,98	0,98	0,96	0,96
POWER AT PROPELLER		DHP_2	11690	12490	12230	5580
EXCL. SEA MARGIN[2]		DHP_3	9740	10410	10190	4650
SHIP'S SPEED	TRIAL[3]	KN	18,1	18,4	18,3	15,0
	SERVICE[4]		17,3	17,6	17,5	13,7
PROP. REV. AT BHP_2		RPM	100	104	70	70
C.P. PROP. EFFIC η_0		–	0,660	0,655	0,690	0,675
EFFIC. GAIN/LOSS[5]		%	0	– 0,8	+ 4,5	+ 2,3
FUEL CONSUMPTION	S.F.C.	$G/BHP_2 \times H$	132	132	131	131
	S.F.C.'[6]		132	133	125	128
	S.F.C."[7]		139	140	131,5	134,5
	DAILY	T/DAY	43,5	46,6	43,8	22,4
	DAILY[8]		43,5	43,9	41,9	–

[1]POWER FOR SHAFT GENERATOR $\frac{790}{0,95} = 830$ KW = 1130 HP

[2]20 % SEA MARGIN $DHP_3 = \frac{DHP_2}{1,2}$

[3]ON 10,5 M DRAUGHT FOR DHP_2

[4]ON 10,5 M DRAUGHT FOR DHP_3

[5]VERSUS ALTERNATIVE "A"

[6] S.F.C.' = S.F.C. CORRECTED FOR EFFIC. LOSS OR GAIN

[7]S.F.C." = S.F.C.' x $\frac{10200}{9700}$

[8]SHIP'S SPEED 18,1 KN = CT.

FUEL CONSUMPTION OF PROPULSION PLANTS WITH LOWER OUTPUT

Table I

SYM-BOL	ITEM	UNIT	ALTERNATIVE		
			A	B	C
a	MAIN ENGINE(S)	MAKE	B&W	SULZER	MaK
		TYPE	5 L 80 GFCA	6 RL B 76	2 x 6 M 601
		1000 $	3552[1]	3552[1]	2537
b	COUPLINGS	TYPE	–	–	2 x KAA 380
		1000 $	–	–	97
c	REDUCT. GEAR	TYPE	–	–	GVE 1900
		1000 $	–	–	495
d	CONTROLLABLE PITCH PROPELLER	Ø MM	6400	6400	7300
		1000 $	517[2]	555[2]	683[2]
	RES. BLADE	1000 $	26	27	29
e	SHAFT GENERATOR	KW	790	790	790
		1000 $	178[3]	178[3]	25[4]
GRAND TOTAL		1000 $	4273	4312	3866
	ITEMS a+b+c	1000 $	3552	3552	–
d'	FIXED PITCH PROPELLER	Ø MM	6400	6400	–
		1000 $	144	156	–
	RES. PROPELLER	1000 $	144	156	–
e'	SHAFT GENERATOR	KW	790	790	–
		1000 $	338[5]	338[5]	–
f	PROPELLER SHAFT	Ø MM	485	490	–
		1000 $	67	60	–
GRAND TOTAL		1000 $	4245	4262	–

[1] INCLUSIVE BALANCER
[2] INCLUSIVE PROP. SHAFT
[3] RIDING GEAR ON THE SHAFT
[4] POWER TAKE OFF + COUPLING
[5] GENERATOR IN THE SHAFT + FREQUENCY CONTROL

INVESTMENT OF PROPULSION PLANTS WITH LOWER OUTPUT Table II

ITEM		UNIT	ALTERNATIVE				
			D	E_1	E_2	E_3	F
MAIN ENGINE(S)		MAKE	B&W	MaK	MaK	MaK	MaK
		TYPE	6 L 80 GFCA	1x8 M 601 1x6 M 601	1x8 M 601 –	– 1x6 M 601	2x8 M 601
ENGINE POWER	100 %	BHP_1	18360	19040	10880	8160	21760
	85 %	BHP_2	15610	16180	9250	6940	18500
ENGINE SPEED	100 %	RPM	106	425	425	425	425
	85 %		100	425	425	425	425
SHAFT GENERATOR[1]		KW	790	790	790	790	790
TRANSMISSION EFFIC.		–	0,98	0,96	0,96	0,96	0,96
POWER AT PROPELLER		DHP_2	14190	14450	7790	5580	16680
EXCL. SEA MARGIN[2]		DHP_3	11830	12040	6490	4650	13900
SHIP'S SPEED	TRIAL[3]	KN	19,0	19,1	16,4	15,0	19,7
	SERVICE[4]		18,2	18,3	15,6	13,7	18,9
PROP. REV. AT BHP_2		RPM	100	76	76	76	80
C.P. PROP. EFFIC η_0		–	0,660	0,680	0,675	0,650	0,680
EFFIC. GAIN/LOSS[5]		%	0	+ 3,0	+ 2,3	– 1,5	+ 3,0
FUEL CONSUMPTION	S.F.C.	$G/BHP_2 \times H$	132	128,7	127	131	127
	S.F.C.'[6]		132	125	124	133	123,5
	S.F.C."[7]		139	131,5	130,5	140	130
	DAILY	T/DAY	52,0	51,0	29,0	23,3	57,7
	DAILY[8]		52,0	50,2	–	–	50,0

[1]POWER FOR SHAFT GENERATOR $\frac{790}{0,95} = 830$ KW = 1130 HP
[2]20 % SEA MARGIN $DHP_3 = \frac{DHP_2}{1,2}$
[3]ON 10,5 M DRAUGHT FOR DHP_2
[4]ON 10,5 M DRAUGHT FOR DHP_3
[5]VERSUS ALTERNATIVE "D"
[6] S.F.C.' = S.F.C. x $\frac{'o}{o}$
[7]S.F.C." = S.F.C.' x $\frac{10200}{9700}$
[8]SHIP'S SPEED 19,0 KN = CT.

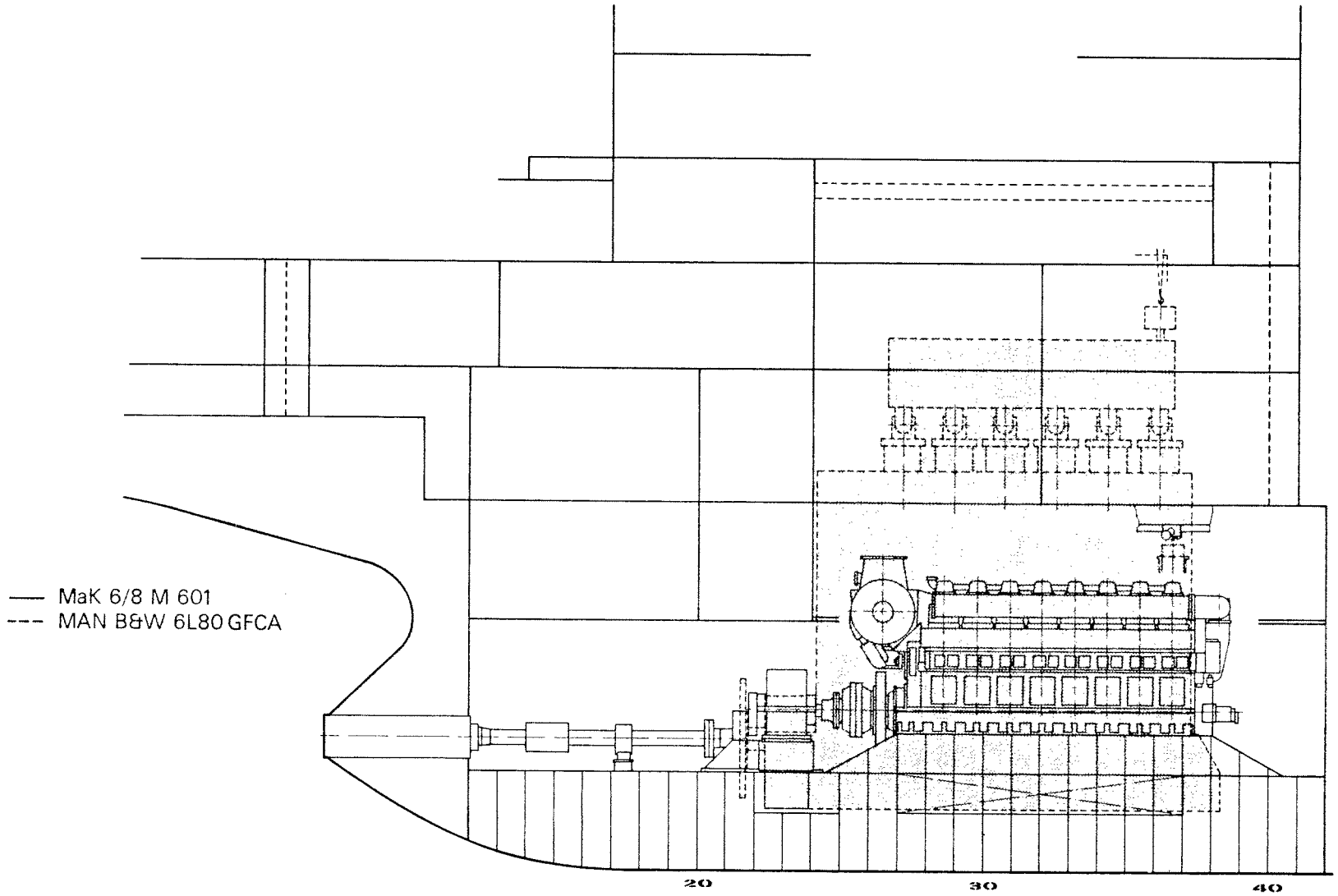
FUEL CONSUMPTION OF PROPULSION PLANTS WITH HIGHER OUTPUT

Table III

SYM-BOL	ITEM	UNIT	ALTERNATIVE		
			D	E	F
a	MAIN ENGINE(S)	MAKE	B&W	MaK	MaK
		TYPE	6 L 80 GFCA	1 x 8 M 601 1 x 6 M 601	2 x 8 M 601
		1000 $	4228[1]	2960	3383
b	COUPLINGS	TYPE	–	1 x KAA 410 1 x KAA 380	2 x KAA 410
		1000 $	–	118	140
c	REDUCT. GEAR	TYPE	–	GVE 1950	GVE 1950 A
		1000 $	–	558	575
d	CONTROLLABLE PITCH PROPELLER	Ø MM	6700	7300	7300
		1000 $	610[2]	696[2]	740[2]
	RES. BLADE	1000 $	30	31	35
e	SHAFT GENERATOR	KW	790	790	790
		1000 $	178[3]	25[4]	25[4]
GRAND TOTAL		1000 $	5046	4388	4898
	ITEMS a+b+c	1000 $	4228	–	–
d'	FIXED PITCH PROPELLER	Ø MM	6700	–	–
		1000 $	178	–	–
	RES. PROPELLER	1000 $	178	–	–
e'	SHAFT GENERATOR	KW	790	–	–
		1000 $	338[5]	–	–
f	PROPELLER SHAFT	Ø MM	515	–	–
		1000 $	64	–	–
GRAND TOTAL		1000 $	4986	–	–

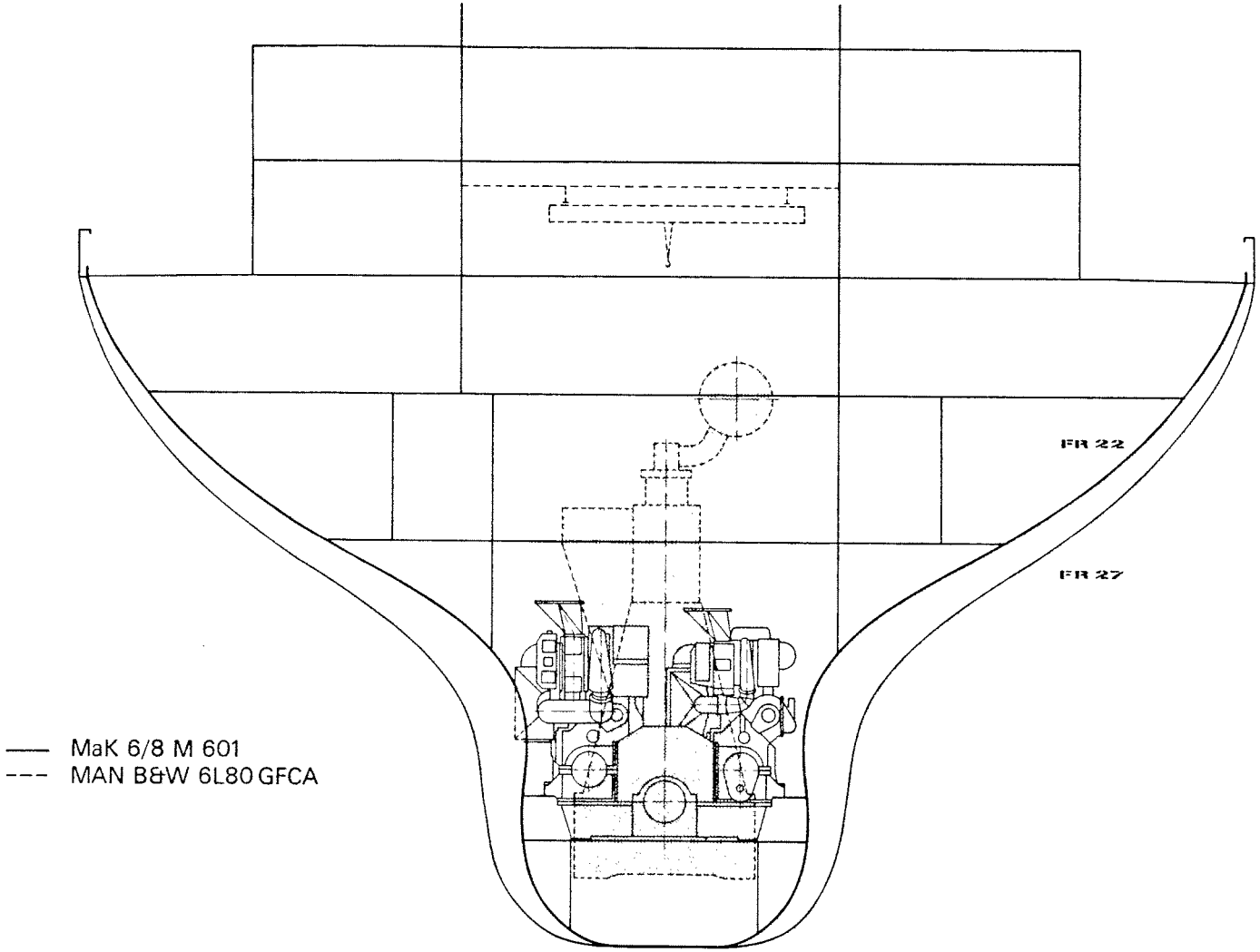
[1] INCLUSIVE BALANCER
[2] INCLUSIVE PROP. SHAFT
[3] RIDING GEAR ON THE SHAFT
[4] POWER TAKE OFF + COUPLING
[5] GENERATOR IN THE SHAFT + FREQUENCY CONTROL

INVESTMENT OF PROPULSION PLANTS WITH HIGHER OUTPUT Table IV



LONGITUDINAL SECTION LOOKING TO PORT

Fig. 2



SECTION AT FR 22 LOOKING FORWARD

Fig. 3

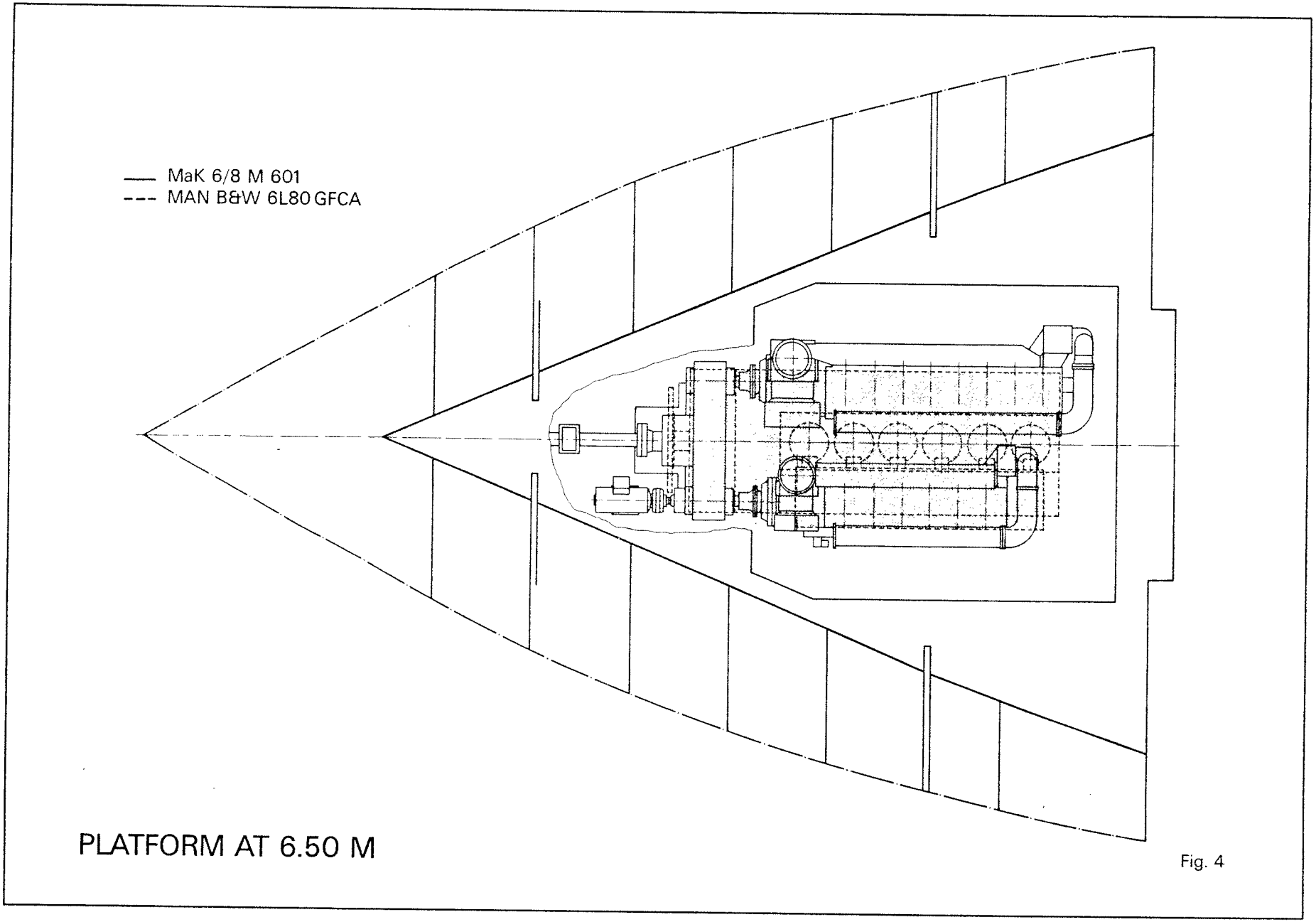


Fig. 4

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

Which Bunkerfuel Engines for smaller ships?

by

Prof. Dr. Ing. C Gallin

Delft University of Technology
Delft

Which bunkerfuel engines for smaller ships?

by

Prof. Dr. Ing. C. Gallin and
Dipl.-Ing. P. Nissen

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2. An actual example
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4. Discussion of the results
5. Summary
6. Bibliography

1. Introduction

The competition between four-stroke and two-stroke diesel engines for ship's propulsion exists as long as the engines themselves. Much was written about this, but the present study does not intend to be an addition to those publications. However, as new situations arise in shipping - and shipbuilding - new technical developments have to be analysed.

The strong increase in fuel prices has led to using heavy fuel oils with poor qualities even in smaller diesel engines down to 1000 HP. Four-stroke trunk-piston engines of this size have been especially designed for this application, to enable auxiliary engines and small ships to be driven with heavy fuel oil.

One example for the above mentioned development is the new heavy fuel engine M 35 introduced by Krupp MaK, Kiel in 1982, with a bore of 350 mm and a stroke of 380 mm, and a present output of 2650 kW (6-cylinder engine) or 3500 kW (8-cylinder engine) at 750 rpm. The specific fuel consumption is 189 g/kWh (139 g/HPh) [1].

On the other hand, manufacturers of two-stroke crosshead diesel engines are looking for new markets and are at present offering so-called "mini-bore" two-stroke engines, with bores below 400 mm and outputs down to 2000 HP. This way they try to enter the market for smaller ships, a market which so far was exclusively reserved to four-stroke trunk-piston engines, which are less heavy and require less space.

The first "mini-bore" two-stroke diesel engine was introduced by Mitsubishi three years ago [2] ; their model UEC 37/88 has a bore of 370 mm. They were soon followed by MAN-B&W with the engine L 35 GB (E) with 350 mm bore, and finally Sulzer with their 380 mm-bore engine RTA 38.

One special design feature of these engines - in particular of the last two models - is a bore/stroke ratio of approx 1:3. By this it was possible to reduce the speed to 150 - 200 rpm. Such an engine speed is not far from the optimum propeller speed of smaller ships with draughts between 4 and 6 m. A reduction gear can be omitted, these engines can be directly coupled to the propeller. Another advantage of these two-stroke engines with long stroke is a favourable specific fuel consumption, which is between 175 g/kWh (129 g/HPh) and 184 g/kWh (135 g/HPh) depending on the engine type.

The general disadvantage of all two-stroke crosshead engines is the considerable weight and price, and the large dimensions, which particularly applies to the height. These

disadvantages are even greater for long-stroke engines. Regarding the length, however, the two-stroke engines with less cylinders are shorter, whereas the four-stroke engines have to have a reduction gearbox. On the other hand it is possible to install shorter twin-engine installations with high operational safety for the same investment amount when using four-stroke engines.[3]

The developments on either side did not ease the dilemma which the owners are often in! To have the choice is not always an easy task. The owners of smaller ships, too, are now confronted with information from both sides.

The present study mainly tries to objectively analyse one decisive factor - economy. After all this is the main criterion for the owner's decision. When choosing a propulsion plant, the general question is whether later operational results will justify additional investments in the beginning. In other words: Are "mini-bore" two-stroke engines more efficient than four-stroke engines with the same output and also running on heavy fuel? It is this study's task to find the answer.

2. An actual example

An economic comparison between alternatives can only be based on concrete figures. This fact is sufficient to assume an example for which price indications were inquired. Price indications are relative, depending on time, place and manufacturer, and which therefore may be different. There is, however, no other way.

As an actual example for this study a 460 TEU container feeder ship, actually being under construction in Western Germany, was chosen. The main data of this ship which are relevant for this study, are shown in Fig. 1. This newbuilding can be regarded as small cargo vessel with a length between perpendiculars of just over 100 m, a width of 19.2 m and a mean draught in loaded condition of 6.20 m. The installed output of approx 3500 kW (approx 4800 HP) resulting - after deduction of 260 kW (350 HP) for a shaft alternator - in a trial speed in loaded condition of 13.7 kn can also be regarded as representative for this ship size.

For the main drive a Krupp MaK 8 M 35 engine was chosen, by the way the first of this series which is installed in a ship. The engine will drive a CP propeller via a single reduction gearbox.

The reports on the prototype tests made in the Berlin "Versuchsanstalt für Wasserbau und Schiffbau" were made available to the authors of this study with the kind assistance of the owners, the shipping company Horst Bartels in Hamburg. Therefore, all results of the resistance and propulsion measurements, in particular the measured wake factor, are known.

Presuming as a sound basis a propeller (Dp) to draught (T) ratio of the loaded ship of $\frac{Dp}{T} = 0.7$, a maximum propeller diameter results as:

$$Dp(max) = 0.7 \times 6.20 = 4.34 = \text{approx } 4.30 \text{ m}$$

In ballast conditions, the propeller must be well immersed. Fig. 1 shows in this operation condition an aft draught of 4.80 m, the $\frac{Dp}{T}$ ratio will then be approx 0.9, which is still acceptable. For a propeller

diameter of 4.30 m and the given conditions the optimum propeller speed is approx 150 rpm, which can easily be realized with the indirect drive by means of the reduction gear.

The installation with the Krupp MaK engine 8 M 35, as described above and shown in Fig. 1, was taken as a basis for further comparison.

3. Method of comparison

A clearly defined and strictly adhered to method of comparison belongs to a correct comparison.

3.1 Propeller layout

For a CP propeller this problem is not that acute, as by pitch setting the propeller can be made "light" or "heavy". In Fig. 2, showing the output (P_B) as a function of the propeller speed (rpm), this means that the propeller curve may move to the right or to the left. Therefore it is mainly a question of strength, i.e. a question of dimensioning whether a certain operational aim can be reached or not. There are other points of view, like cavitation and efficiency effects. These two features are, however, covered in the design stage. Point "A" in Fig.2 shows the operational point 100 % output at 100 % propeller speed, the so-called maximum continuous rating (MCR), in this case without the shaft alternator. If the owner - as is common practice - wants to reserve the possibility to dispose of the full power, if necessary, the strength of the CP propeller has to be in accordance with the load at this point "A".

The line AB shows in a simplified manner the characteristic and well-known curve of the diesel engine at maximum and constant torque. The engine curve for 85 % maximum torque, i.e. for reduced mean pressure, is shown as a dotted line in Fig. 2. Due to the favourable specific consumption rates owners prefer to operate on this last mentioned line.

If the engine is equipped with a low-cost shaft alternator without frequency governor but with a CP propeller, as in this example, the CP propeller must keep at constant speed. If the maximum propeller speed is chosen, one moves towards the vertical line AB when changing the propeller pitch. If the owner decides in favour of an engine continuous service rating (CSR) of 85 %, and if the shaft alternator power requirement is deducted, the operational point "B" in Fig. 2 results as ship's propulsion. For the propeller calculation the losses in the reduction gear (2 %) (only for indirect drive) and the shaft line (2 %) have to be deducted from the P_B-value of point "B" for all alternatives (see list in Fig. 1).

As, in the long run, the operation with 85 % engine output is required and - as generally known - a lower speed has a favourable influence on the propeller efficiency, one could tend to set a lower speed, which has to be kept constant for the shaft alternator, e.g. also to 85 % of the maximum propeller speed. The operational point would then move on the vertical (dotted) line to around 85%rpm. This design can however not be recommended because the engine would thermally be overloaded above point "C". That would mean that the engine could no longer be operated at full load. If the ship is, however, forced to request higher power by certain circumstances, this can only

be made by increasing the revolutions. In such a case, the shaft alternator would always have to be switched off and the auxiliary engine be switched on which would probably be a handicap for operation on board.

3.2 Engine layout

The design possibilities of a diesel engine for ship's propulsion are shown in Fig. 3. The diagram was taken from a Sulzer specification [4].

Other engine manufacturers like MAN-B&W or Mitsubishi published similar diagrams [5] [6] .

The principle itself is known and therefore need not be repeated. Only a few words to make it clear: A diesel engine for ship's propulsion can either be derated to approx 85 % full load (point "R1") by decreasing the mean effective pressure (mep) along the propeller curve (point "R2" in Fig. 3) or along the line of the constant mean effective pressure (point "R3").

The operational point "R2" reduces the specific fuel consumption more than the operational point "R3".

However, the best possibility to reduce the specific fuel consumption is a combination of both means (point "R4"). The engine output available is then reduced to approx 70 % full load. In case that unchanged propulsion power is required, a more powerful engine has to be installed (larger number of cylinders or larger bore), which then would become more expensive. The achievable savings in fuel consumption are confronted with higher investments also in a case of engines from the same design family. When selecting the engines to be compared, these possibilities were taken into account.

3.3 Engine alternatives

Fig. 4 gives a survey of diesel engines under comparison, indicating the manufacturer, type, output in kW and speed in rpm, the last two each for MCR. On the left you will find the data for Krupp MaK's four-stroke engine 8 M 35 which is the basis for this comparison. On the right you find the data of the selected two-stroke engines, listed according to manufacturers (Sulzer, MAN-B&W and Mitsubishi), one below the other. The alternatives have been marked I to X.

As the Krupp MaK engine is the only one requiring a reduction gearbox, 2 % for the losses in the gear were deducted from the beginning from its MCR output of 3520 kW. 3450 kW remain, and the other engines were selected accordingly.

The two-stroke engines in comparison must have an MCR of around or above 3450 kW. If the MCR is above this value, the engine in question should be derated to the same level for a correct comparison.

In this case, same condition means the same achievable ship's speed at contracted draught. In other words: the propeller thrust required has to be the same for all analysed alternatives. This makes it necessary to integrate the propeller efficiency into the calculation of to what extent the engine has to be derated. As the engines compared have different speeds at MCR - and the propeller efficiency depends on this - the necessity to derate is different for each alternative, given constant propeller thrust.

Under the aspect of a desirable reduction in fuel consumption the selection of engines may follow a different course.

The first ships equipped with these Sulzer engines cover the three possibilities "R2", "R3" and "R4". For "R4" (alternative II), however, one more cylinder is needed.

For the group of the MAN-B&W engines the design for maximum rating is called "MC" (previously "GB"), whereas the fuel saving design (with lower mep) is called "MCE" (or "GBE"). Not only one more cylinder for the more economical version (alternative VI compared to alternative V and VIII to VII) is compared, but also a larger bore, 450 instead of 350 mm (alternative VII to V, and VIII to VI).

Finally, the third group of the Mitsubishi engines stands out at a relatively high speed at MCR (210 rpm). Therefore both alternatives were taken with a maximum mean effective pressure ot this engine type (14.72 kg/m²). Alternative IX has one cylinder more to make a higher speed deration possible (rather high propeller efficiency), and alternative X to make available the required output at all.

3.4 Method of calculation

The method of calculation applied can be seen in tables I, II and III. An engine with an output above 3450 kW at MCR and a speed of more than 150 rpm was derated by means of a complicated iteration process, not down to this output value, but to an MCR' between both. E.g, the alternative V in Table II was derated from 4000 kW at 200 rpm to 3720 kW at 186 rpm. The remaining difference between 3720 kW of alternative V and 3450 kW of alternative I serves to balance the worst propeller efficiency of alternative V, caused by its higher speed (186 rpm) compared to alternative I (150 rpm). For two alternatives (alternative II and IV in Table I) even a slight increase of the speed at MCR' (compared to MCR) was necessary to achieve the required output at all.

The operational point MCR' of the engines calculated this way serves the design and dimensioning of the CP propeller. The speed corresponding to MCR' is also decisive for the selection of the shaft alternator. This speed will therefore be kept constant during the further propeller calculation process. In this example (alternative V) 186 rpm remain constant.

To calculate the fuel consumption rate, the continuous service rating has to be found. It is:

$$CSR' = 0.85\ MCR'$$

This means, e.g. for alternative V the CSR' is approx 3160 kW. From CSR', 260 kW for the shaft alternator and then 2 % for the losses in the shaft line are deducted. The output delivered to the propeller (P_D') is e.g. 2840 kW for alternative V. For this value and for the constant speed at MCR' the efficiency was calculated for the propeller designed for MCR', e.g. the alternative V achieves 0.5314 propeller efficiency. For this operation mode, too, it has to be assumed that the propeller thrust has to be the same as for alternative I to reach the required ship's speed of 13.6 kn for the contracted draught. To simplify matters, in this study the P_D'-value of alternative I is taken for each alternative (e.g. alternative V), i.e. 2620 kW, and is multiplied with the ratio of efficiency between alternative I and the alternative in question. For alternative V the ratio would be:

$$\frac{0.5672}{0.5314} = 1.0674$$

It results a propeller output (P_D") corrected according to the different efficiencies; e.g. 2800 kW for alternative V. The losses in the shaft line (2 %) and 260 kW of the shaft alternator are added to this output. By this, CSR" results, for which value the fuel consumption rate can finally be calculated.

It was possible to effect all above described propeller calculations for the ten alternatives only with the kind assistance of Messrs. Lips in Drunen. For the iteration process approx 800 operational points were calculated by the computer program of this company.

Now, the further calculation process is very easy. By means of the specific fuel consumption of the engine alternative in question and the calculated necessary output CSR" the daily fuel consumption rate was calculated, on the basis of 300 days/year at sea. A 2 % higher consumption was added for alternative I for losses in the reduction gear.

To calculate the fuel cost, the fuel price at the port of Rotterdam at the date of this study (4th quarter of 1982) was taken as a basis. The price then was 170 $/t. The tables I, II and III are issued in a way that price corrections can later be made without problems.

The difference between the fuel cost per year of alternative I and that of all other alternatives results in annual savings. Only alternatives IX and X have higher consumption costs, due to the high speed or low propeller efficiency respectively.

This is followed in Tables I, II and III by the calculation of the different investment and profitability criteria.

The prices for the engine including the spare parts required by the classification society, and the price for the corresponding CP propeller including the propeller shaft were added to each other. For alternative I the prices for the gear and the coupling have to be added, for the calculation of which we thank Messrs. Lohmann & Stolterfoht GmbH in Witten.

All prices were fixed for end of 1983/ beginning of 1984.

The alternative I was also the basis for the investments, and the additional investments for all other alternatives were calculated accordingly.

The "Capital Recovery Factor (CRF)" and the "Pay-Out-Period (POP)" were taken as profitability criteria.

The CRF is the "Returns (R)" to investment "P" ratio.

$$CRF = \frac{R}{P}$$

In this study R means the savings in fuel cost for each alternative, and P the additional investment.

POP is the time necessary to pay back the additional investment. Without interest this would be easy:

$$POP = \frac{P}{R}$$

As, however, interest has always to be paid, the POP is calculated with the formula

$$(CRF)_N^i = \frac{R}{P}$$

"i" meaning the interest rate (assumed: 8 %) and "N", P and R given, is the POP.

The results are once more clearly shown in Fig. 8.

4. Discussion of the results

Considering the Tables I, II and III and Fig. 8, it is shown that the additional investment for alternatives II to VIII is rather high, between approx 450.000 and 900.000 $, compared to relatively low annual fuel savings of approx 10.000 and 60.000 $. The CRF is therefore rather poor, between 0.020 and 0.084, i.e. between 2 and 8.4 %. Not even the interest rate of 8 % could be paid back, except for alternative VII. The POP is therefore infinite, except for alternative VII, where it is 40 years, which is absolutely of no interest for the owner.

The situation is different for the alternatives IX and X. The additional investment is less costly (134.000 or 40.000 $ respectively), but they can never be paid back as there result no fuel savings but additional costs. The POP is therefore infinite for alternatives IX and X, too.

Considering the results of the manufacturers it is worth mentioning that in the group of the Sulzer alternatives, Table I, the alternative IV, i.e. with the "R2" design, requires the lowest investment, whereas alternative II with "R4" shows the highest CRF. In the MAN-B&W group, Table II, the alternative VII with the largest bore (450 mm) but lowest number of cylinders (5) is the cheapest alternative. This alternative has the highest CRF, although the additionally invested money of the owner in this alternative, too, cannot pay out within the 15 or 20 years of the normal operation time of the vessel.

In the third group of the Mitsubishi engines, Table III, finally the alternative IX with one more cylinder is the most favourable. The alternative X is cheaper, but becomes more expensive after approx 3 years by the higher fuel cost.

This study did not take maintenance and repair cost into account. There are different opinions on this. In a discussion paper for [7], N. Weichhold [8] listed up the cost for a four-stroke and a two-stroke propulsion plant with the same output, without finding appreciable differences as a final result.

This study did not deal with the effect of the very strongly deviating dimensions and weights of the investigated alternatives. They strongly depend on the ship's speed and size. However, to gain a visual image, the contours of the alternative requiring the least space of each group (alternative IV for Table I and alternative V for Table II) were drawn over those of alternative I in Fig. 5, 6 and 7. Exception: the 9 UEC 37/88 engine (alternative IX for Table III). By this all cylinder numbers were presented.

5. Summary

It is the general conclusion of this study that the additional investment for the "mini-bore" two-stroke diesel engines cannot be recovered during the 15 - 20 operational years of a ship, in spite of the more favourable fuel consumption and higher oil prices. This conclusion is only based on one example, which may however be regarded as representative for small cargo vessels.

6. Bibliography

1. "Project M 35"
   MaK marine propulsion plants,
   7/1982 issue.

2. "Mini-bore low-speed diesel engines"
   The Naval Architect, May 1982,
   page E 121.

3. Gallin, C., Siefert, K.-H.,
   Heiderich, O., "Alternatives for
   Economical Diesel Ship Propulsion",
   The Third International Marine
   Propulsion Conference, London,
   March 1981; The Motor Ship,
   May 1981.

4. "General Technical Data for RTA
   Marine Diesel Engines", Sulzer,
   January 1982.

5. "Specification L 35 MC, L 35 MCE",
   B&W Diesel A/S Engineering,
   Edition 1, 1982.

6. "MHI Improves Efficiency of HA
   Series Mitsubishi UE Diesels",
   Leaflet, 1982.

7. Gallin, C., "Advanced Energy
   Saving Concepts", Ship Costs &
   Energy '82 Symposium, New York,
   October 1982; Schip en Werf,
   No. 25, 1982, page 395-402.

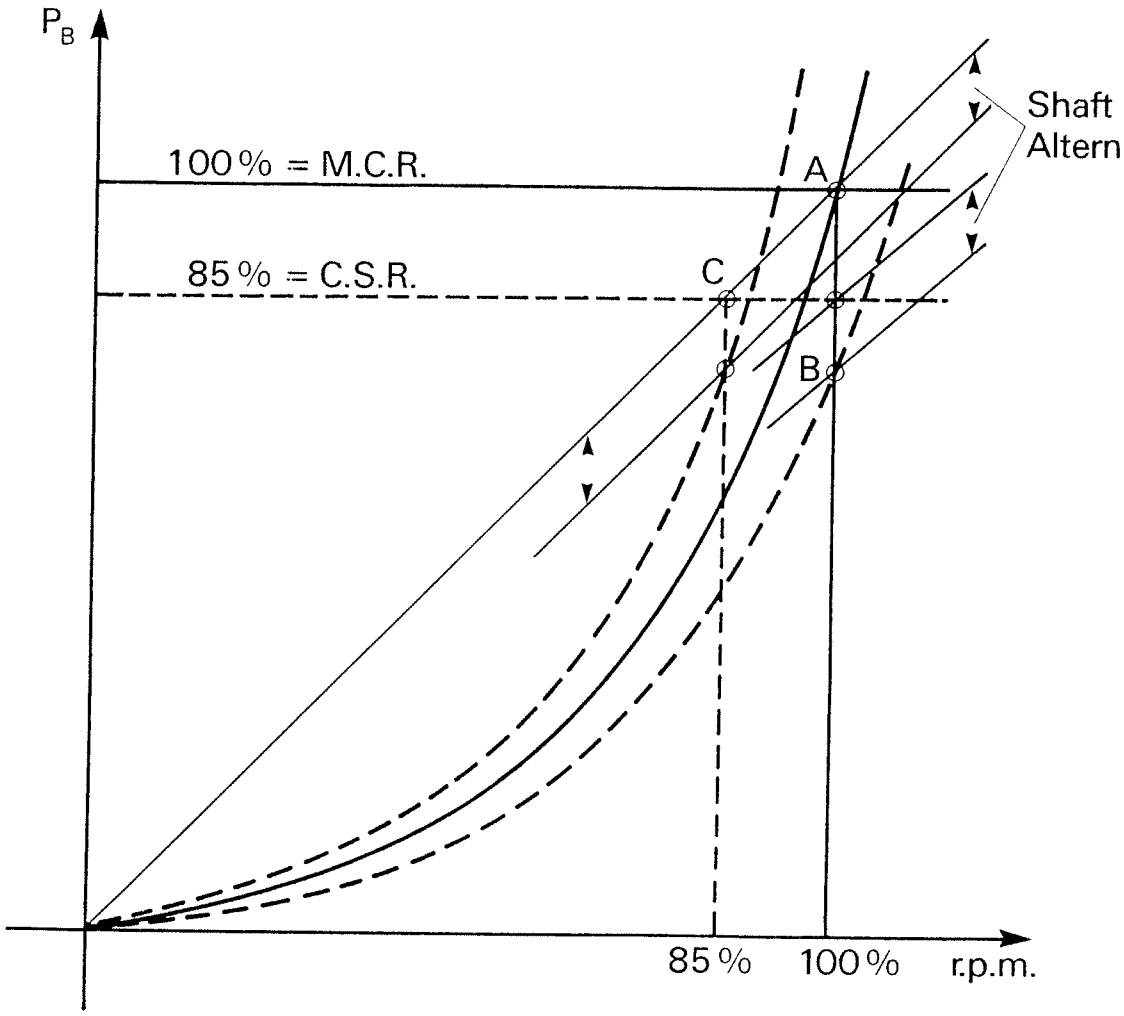
8. Weichhold, N., "Reflections on
   the Maintenance Cost as Criteria
   for the Propulsion Plant Choice",
   New York, October 1982.

PARTICULARS OF THE 460 TEU CONTAINER SHIP

LENGTH BETWEEN PERP.	108,50 m
BREADTH (moulded)	19,20 m
DEPTH (moulded)	8,15 m
DRAUGHT	
in Loaded Condition (mean)	6,20 m
in Ballast Condition (mean)	4,20 m
in Ballast Condition (aft)	4,80 m
PROPELLER DIAMETER (maximum)	4,30 m
POWER OF MAIN ENGINE	3520 KW
REDUCTION GEAR LOSSES	- 70 KW
OUTPUT REDUCTION GEAR AT:	
Maximum Continous Rating (100 %)	3450 KW
Continous Service Rating (85 %)	2930 KW
SHAFT ALTERNATOR	- 260 KW
PROPULSION POWER	2670 KW
DELIVERED AT PROPELLER	2620 KW
SHIP'S SPEED	13,7 KW
PROPELLER REVOLUTION (optimum)	150 r.p.m.

Fig. 1

C. P. PROPELLER LAYOUT



POINT A FOR PROPELLER LAYOUT
AND FOR CONSTANT
REVOLUTIONS NUMBER

POINT B FOR EFFIENCY CALCULATIONS

POINT C NOT RECOMMENDABLE
FOR LAYOUT

Fig. 2

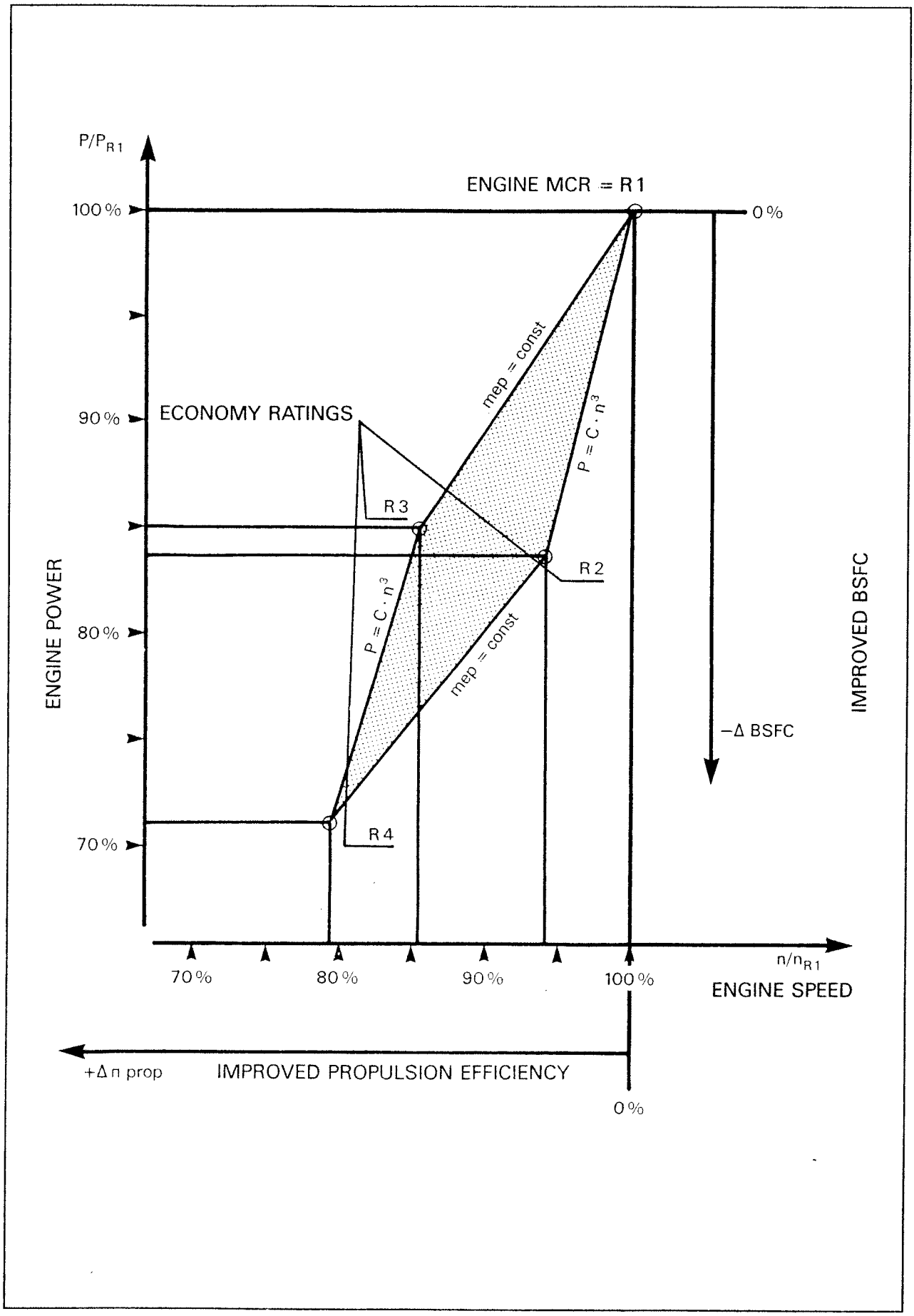

P/P R1
ENGINE MCR = R1
100 %
0 %
90 %
ECONOMY RATINGS
mep = const
P = C · n³
R 3
R 2
80 %
ENGINE POWER
P = C · n³
mep = const
IMPROVED BSFC
−Δ BSFC
70 %
R 4
70 %
80 %
90 %
100 %
n/n R1
ENGINE SPEED
+Δ n prop
IMPROVED PROPULSION EFFICIENCY
0 %

Fig. 3

SCHEME OF COMPARISONS

ALT.	I
MAKE	MaK
TYPE	8 M 35
KW	3450 *
RPM	750/150

* 3520 KW MINUS 2 %
FOR RED. GEAR LOSSES

ALT.	II	III	IV
MAKE	SULZER		
TYPE	8 RTA 38(R4)	7 RTA 38(R3)	7 RTA 38(R2)
KW	3440	3640	3570
RPM	150	162	179

ALT.	V	VI	VII	VIII
MAKE	MAN – B & W			
TYPE	8 L 35 MC	9 L 35 MCE	5 L 45 GB	6 L 45 GBE
KW	4000	3870	3900	4020
RPM	200	200	175	175

ALT.	IX	X
MAKE	MITSUBISHI	
TYPE	9 UEC 37/88	8 UEC 37/88
KW	4305	3825
RPM	210	210

Fig. 4

ALTERNATIVE		I	II	III	IV
MAKE	UNIT	MaK	SULZER		
TYPE		8 M 35	8 RTA 38	7 RTA 38	7 RTA 38
MCR	KW	*3450	3440	3640	3570
SPEED	RPM	750/150	150	162	179
MCR'	KW	3450	3460	3520	3585
SPEED'	RPM	750/150	153	157	180
CSR'	KW	2930	2940	2990	3050
P_D'	KW	2620	2630	2680	2730
PROP. EFF.	——	0,5672	0,5655	0,5577	0,5425
P_D''	KW	2620	2630	2660	2740
CSR''	KW	2930	2940	2970	3060
SFC	G/HP X H	139	129	132	130
	G/KW X H	189	175	179	177
FUEL CONSUMPT.	T/DAY	**13,56	12,35	12,76	13,00
	10^3 x $/YEAR	692	630	651	663
SAVINGS	10^3 x $/YEAR	0	62	41	29
ENGINE	10^3 x $	631	1.654	1.476	1.476
GEAR + CLUTCH	10^3 x $	146	——	——	——
CCP + SHAFT	10^3 x $	216	233	233	230
TOTAL	10^3 x $	993	1.887	1.709	1.706
SUPPL. INVESTM.	10^3 x $	0	894	716	713
CRF	——	0	0,069	0,057	0,041
POP	YEARS	0	∞	∞	∞

NOTE: * MINUS 2 % RED. GEAR LOSSES ** PLUS 2 % RED. GEAR LOSSES

TABLE I

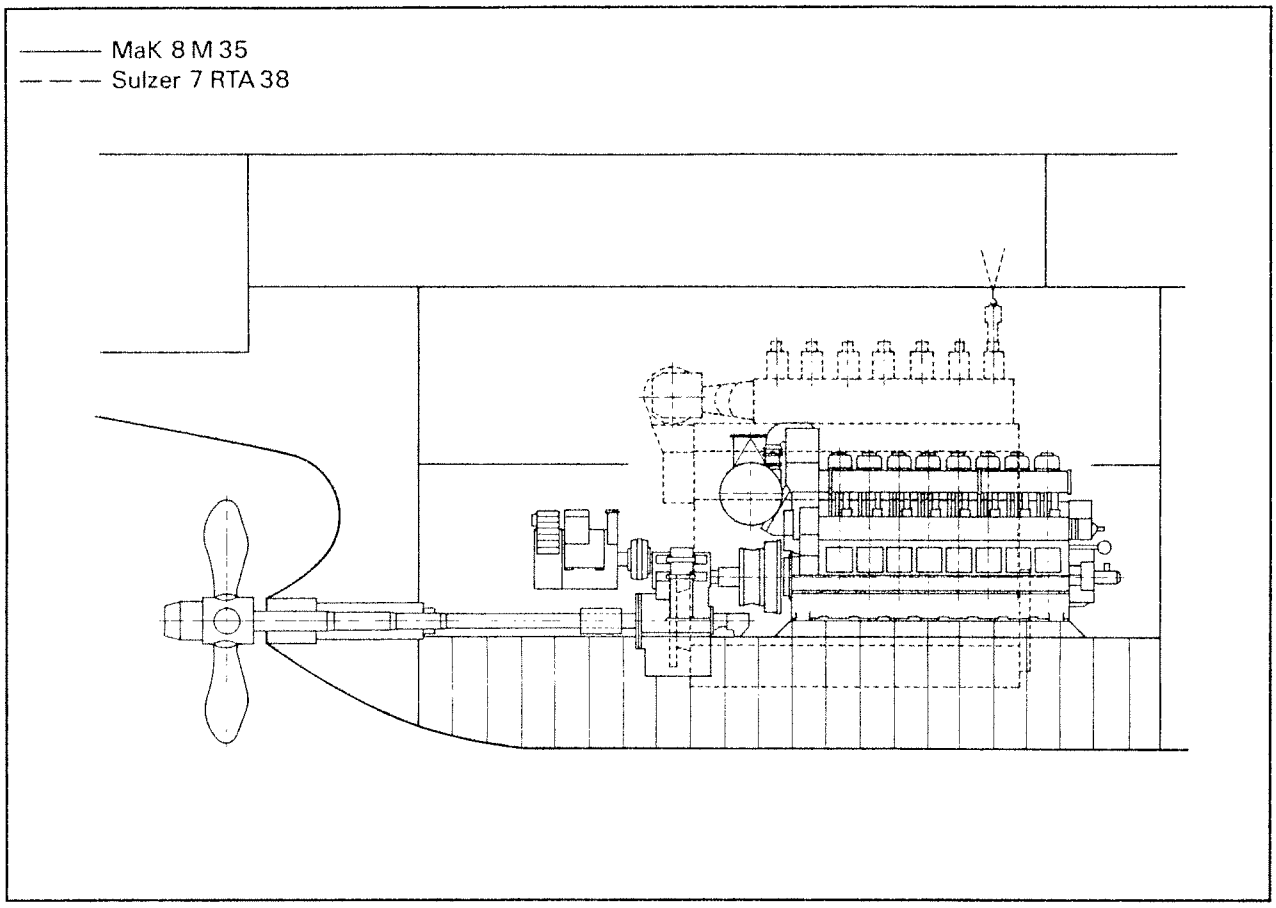
ALTERNATIVE		I	V	VI	VII	VIII
MAKE	UNIT	MaK	MAN – B & W			
TYPE		8 M 35	8 L 35 MC	9 L 35 MCE	5 L 45 GB	6 L 45 GBE
MCR	KW	*3450	4000	3870	3900	4020
SPEED	RPM	750/150	200	200	175	175
MCR'	KW	3450	3720	3720	3600	3600
SPEED'	RPM	750/150	186	192	161	156
CSR'	KW	2930	3160	3160	3060	3060
P_D'	KW	2620	2840	2840	2740	2740
PROP. EFF.	——	0,5672	0,5314	0,5282	0,5499	0,5524
P_D''	KW	2620	2800	2810	2700	2690
CSR"	KW	2930	3120	3130	3020	3000
SFC	G/HP X H	139	132	130	132	130
	G/KW X H	189	179	177	179	177
FUEL CONSUMPT.	T/DAY	**13,56	13,40	13,30	12,97	12,74
	10^3x$/YEAR	692	683	678	661	650
SAVINGS	10^3x$/YEAR	0	9	14	31	42
ENGINE	10^3 x $	631	1.222	1.302	1.130	1.302
GEAR + CLUTCH	10^3 x $	146	——	——	——	——
CCP + SHAFT	10^3 x $	216	227	226	230	234
TOTAL	10^3 x $	993	1.449	1.528	1.360	1.536
SUPPL. INVESTM.	10^3 x $	0	456	535	367	543
CRF	——	0	0,020	0,026	0,084	0,077
POP	YEARS	0	∞	∞	ca. 40	∞

TABLE II

ALTERNATIVE		I	IX	X
MAKE	UNIT	MaK	MITSUBISHI	
TYPE		8 M 35	9 UEC 37/88	8 UEC 37/88
MCR	KW	*3450	4305	3825
SPEED	RPM	750/150	210	210
MCR'	KW	3450	3750	3750
SPEED'	RPM	750/150	183	205
CSR'	KW	2930	3190	3190
P_D'	KW	2620	2870	2870
PROP. EFF.	——	0,5672	0,5270	0,5204
P_D''	KW	2620	2820	2860
CSR''	KW	2930	3140	3180
SFC	G/HP X H	139	135	135
	G/KW X H	189	184	184
FUEL CONSUMPT.	T/DAY	**13,56	13,86	14,0
	10^3 x $/YEAR	692	707	738
SAVINGS	10^3 x $/YEAR	0	−15	−46
ENGINE	10^3 x $	631	899	809
GEAR + CLUTCH	10^3 x $	146	——	——
CCP + SHAFT	10^3 x $	216	228	224
TOTAL	10^3 x $	993	1.127	1.033
SUPPL. INVESTM.	10^3 x $	0	194	40
CRF	——	0	——	——
POP	YEARS	0	∞	∞

NOTE: * MINUS 2 % RED. GEAR LOSSES ** PLUS 2 % RED. GEAR LOSSES

TABLE III



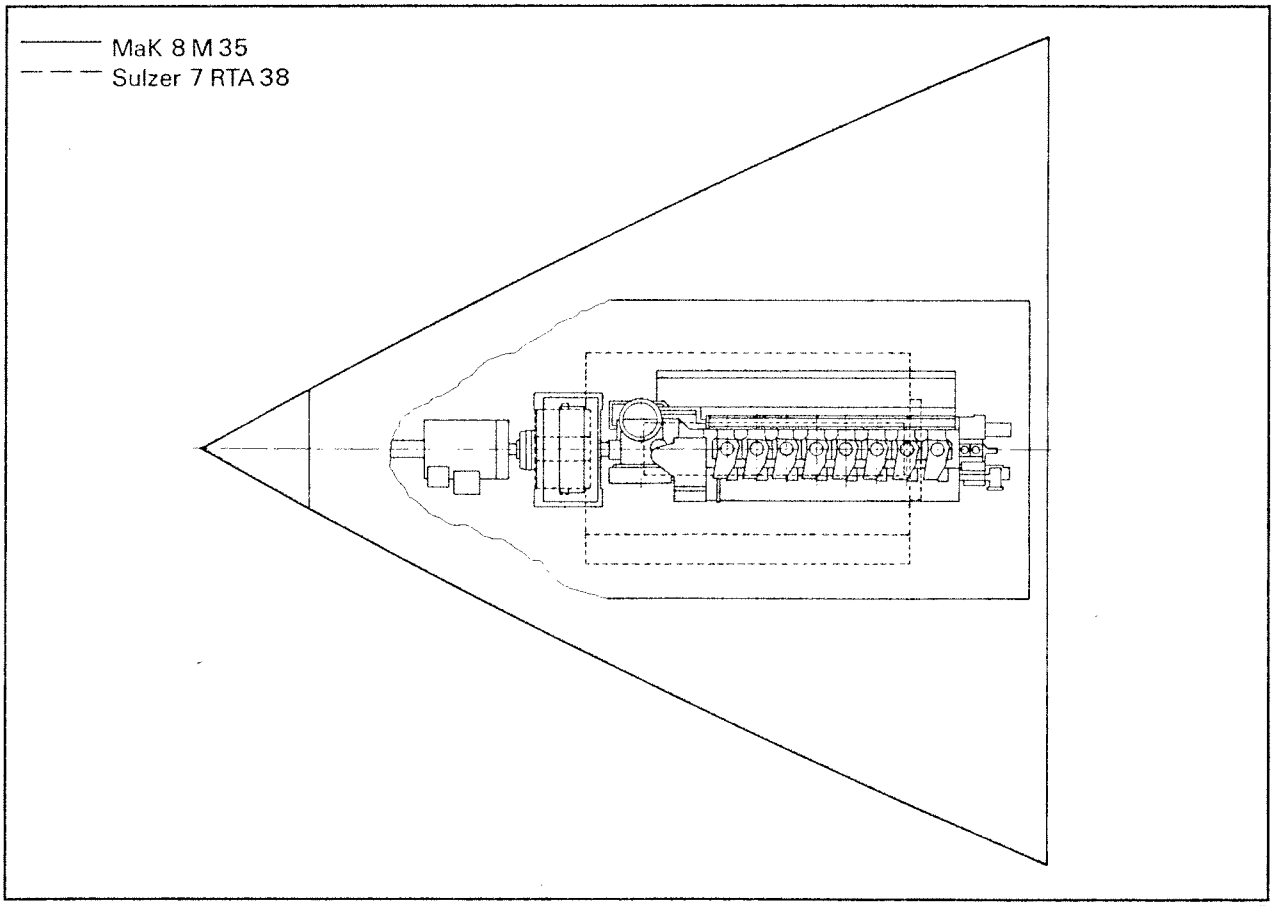
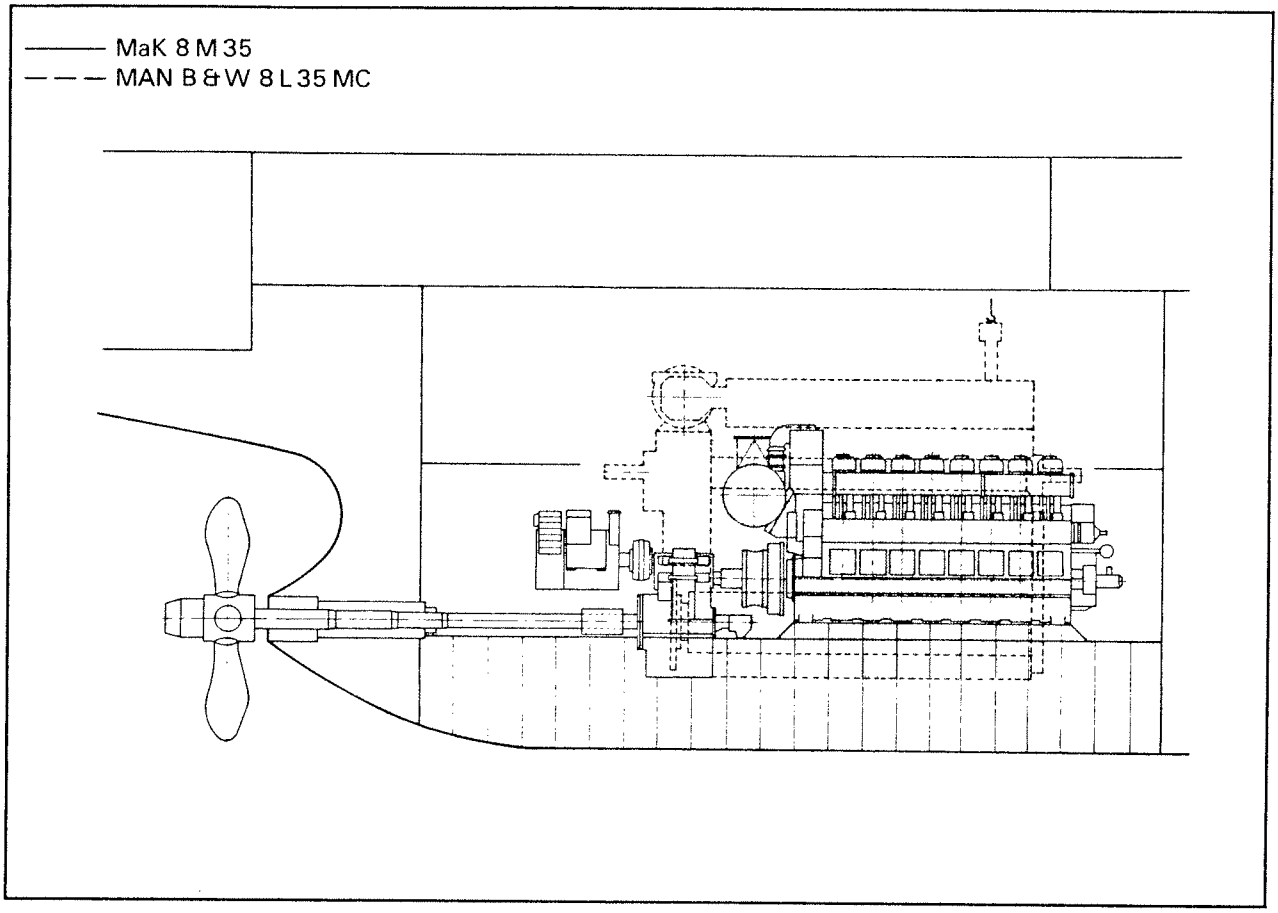


Fig. 5



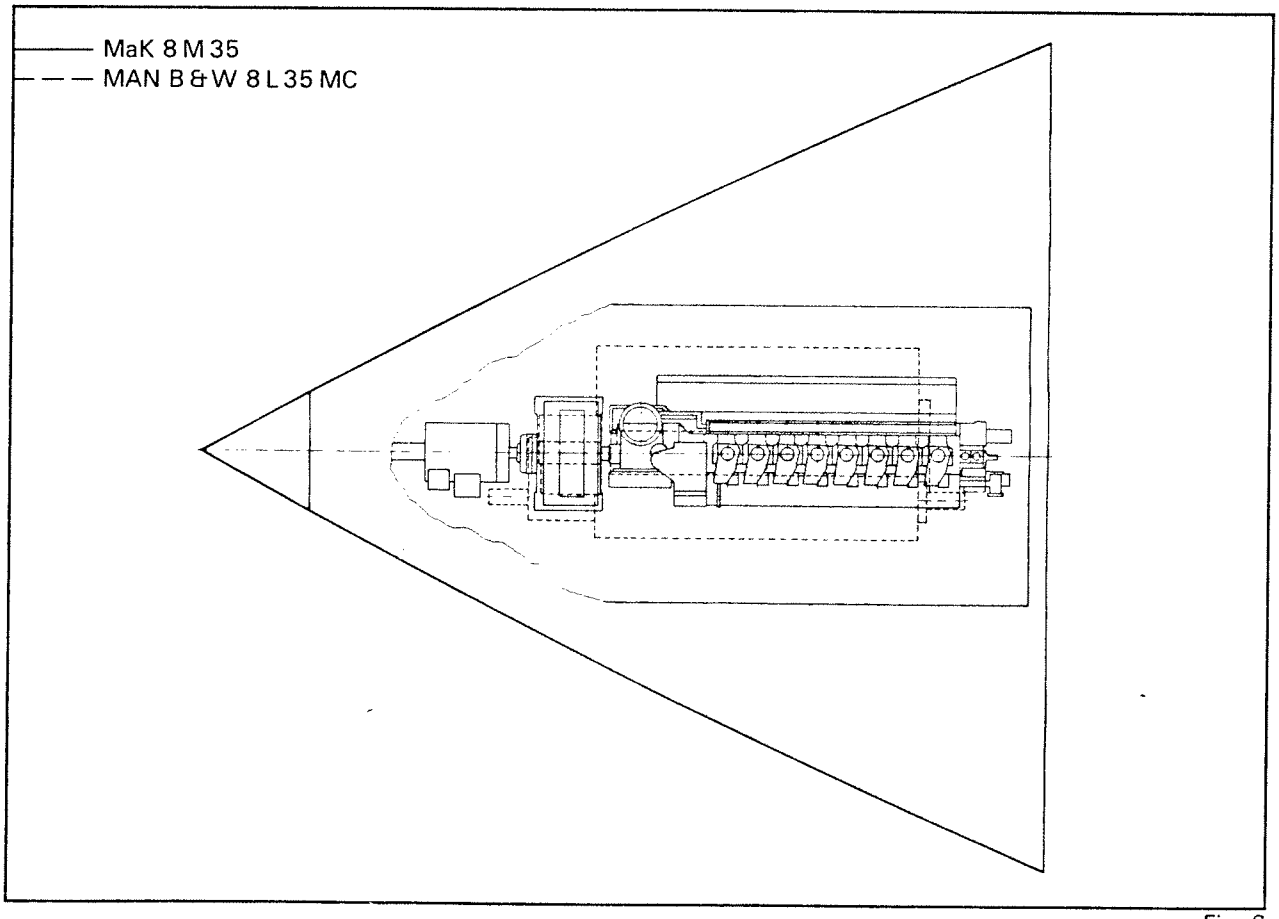
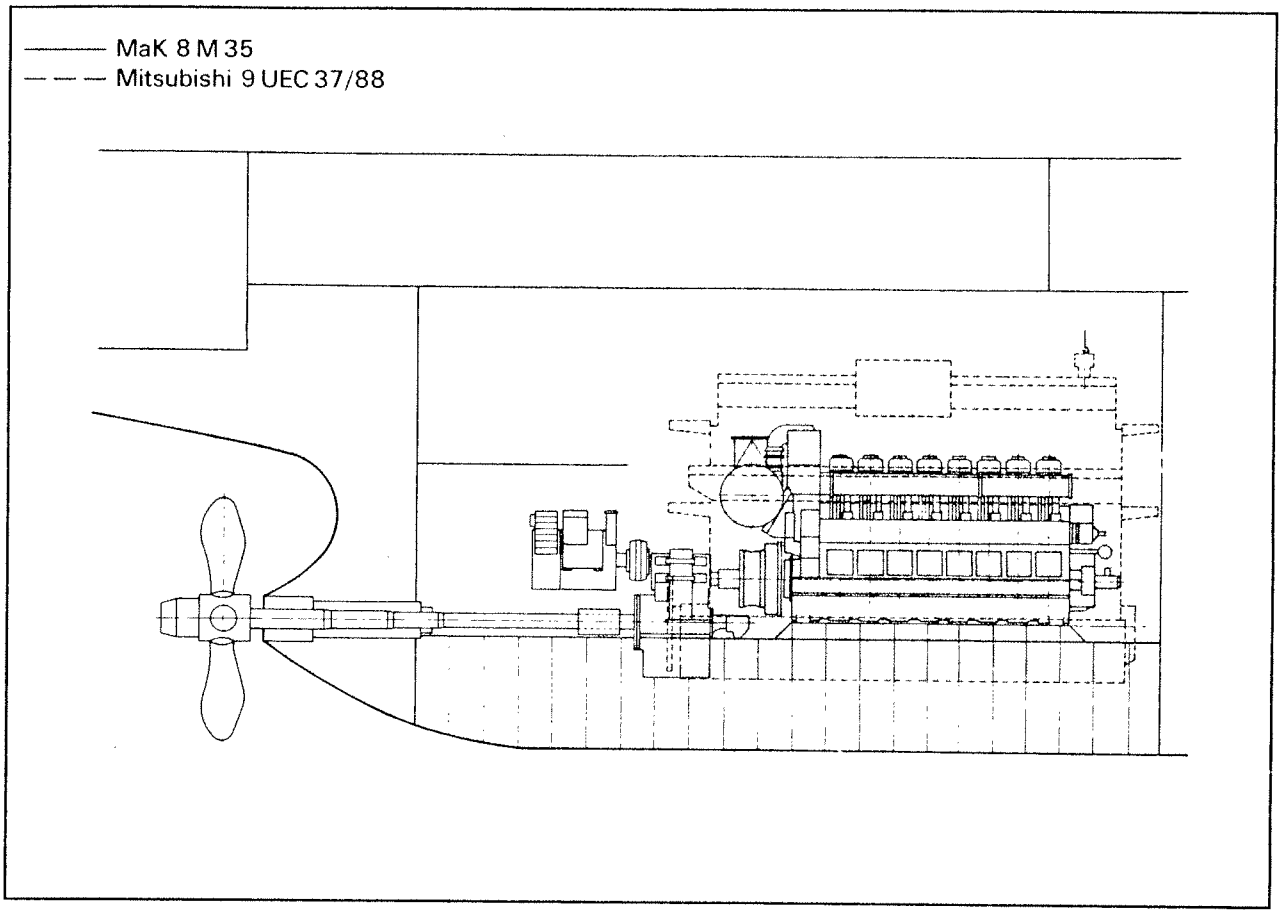


Fig. 6


MaK 8 M 35
Mitsubishi 9 UEC 37/88

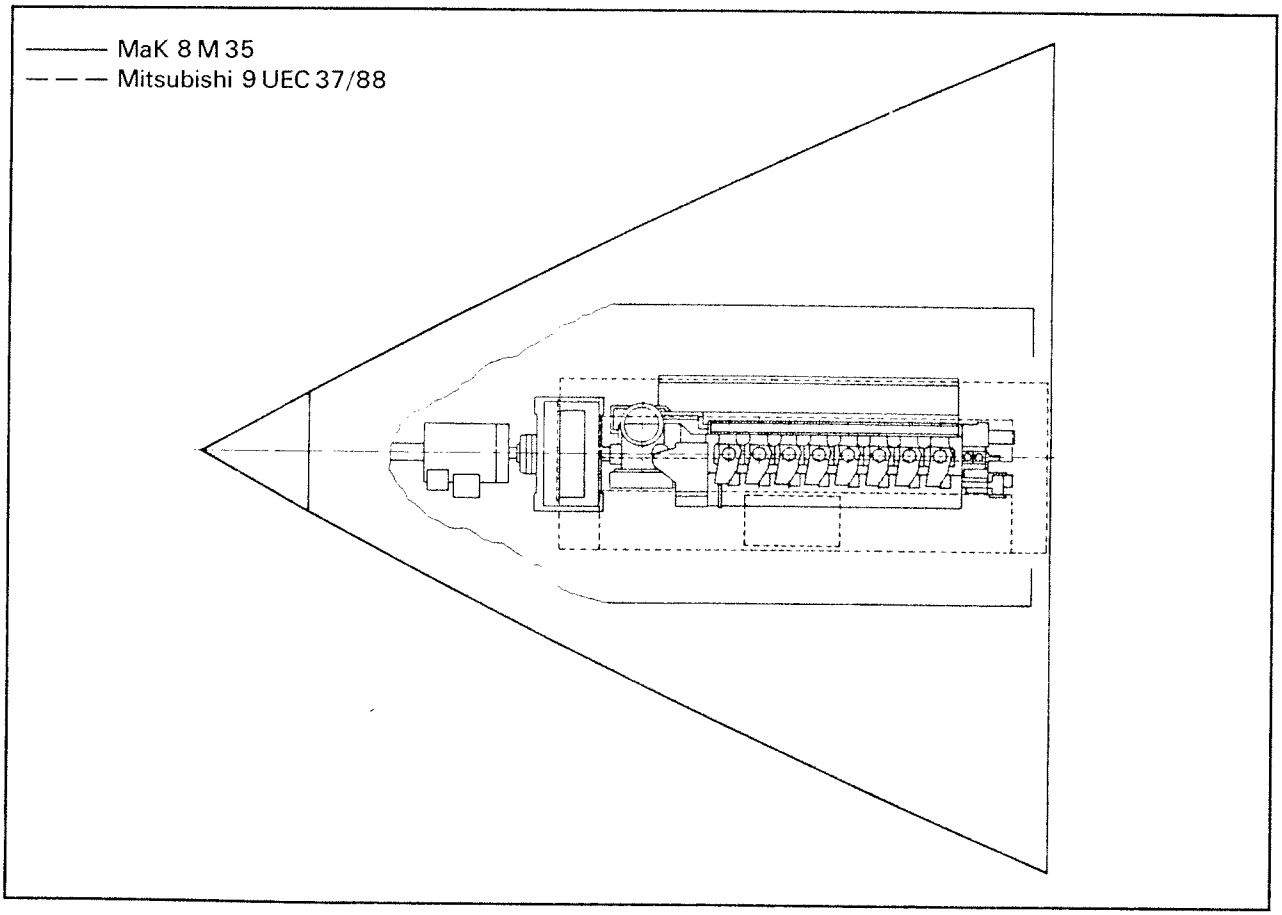

MaK 8 M 35
Mitsubishi 9 UEC 37/88

Fig. 7

REVIEW OF RESULTS

ALT.	I
MAKE	MaK
TYPE	8 M 35
CRF	0
POP*	0

* AT 8 % INTEREST

ALT.	II	III	IV
MAKE	SULZER		
TYPE	8 RTA 38(R4)	7 RTA 38(R3)	7 RTA 38(R2)
CRF	0,069	0,057	0,041
POP*	∞	∞	∞

ALT.	V	VI	VII	VIII
MAKE	MAN – B & W			
TYPE	8 L 35 MC	9 L 35 MCE	5 L 45 GB	6 L 45 GBE
CRF	0,020	0,026	0,084	0,077
POP*	∞	∞	∞	∞

ALT.	IX	X
MAKE	MITSUBISHI	
TYPE	9 UEC 37/88	8 UEC 37/88
CRF	(—)	(—)
POP*	∞	∞

Fig. 8

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THE OPTIMIZATION OF THE PRINCIPAL DIMENSIONS OF THE SHIP

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SIGNIFICANCE OF OPTIMISATION

Considering the manifold efforts of shipyards and shipowners towards improving their economical success it is basically true that the optimisation of the principal dimensions and form ratios can be effectively pursued independent of possible other aims of optimisation such as rationalization of production.

Therefore, it is not opportune to say that instead of optimisation one improves the economy by production organisation.

With the optimisation of the ship's form the attainable benefits indeed keep within relatively close limits - about 3 % of the production costs -, but the same can be realized with negligible risk and with minimal expense. Therefore they should "be picked up", whenever the possibility is given. Related to the total revenue an additional gain amounting to 3 % of the production costs is worth attention.

If a shipyard obtains an average saving of costs of 3 % with the aid of optimisation of principal dimensions, it benefits from this gain independent of the future order situation. If in contrast the shipyard rationalizes its production lines, for instance by changing over from slip to dock production, this investment may turn into a loss if anticipated orders fail to appear.

These savings are partly within the margin of attainable accuracy in forward cost estimates, so that the real savings may not become evident on making the final accounts. This fact, however, is due to the deficiency in making forward estimates and not associated with optimisation, and, therefore, cannot be used as an argument against the introduction of optimisations.

DESCRIPTION OF PROBLEM

The hull design of ships is characterized by the ratios of the principal dimensions and by the hull lines. Both fields are of great significance for the technical quality and cost of the ship.

I should like to confine myself here to the choice of principal dimensions:
Length, Breadth, Depth, Draught, and to the selection of principal ratios, and assume suitable lines as given. The principal ratios are the relations of the principal dimensions and of the block-coefficients.

I assume further that the owners want a ship with given data for deadweight, hold capacity and speed, meaning that the specifications on whose basis the owners ask for quotations from the shipyards are given. It is not intended hereby to check whether these specifications themselves are opportune.

These specifications of the owners can be fullfilled by ships of very different dimensions, wherefore it is beneficial to choose such dimensions which are economically optimal.

For this the procedure to follow is to start with a conventional design and modify this while adhering to owner's specifications in such a way that it gives other principal dimensions and other principal ratios than the first one. These alternatives are named in the following as "Variants".

VARIATION OF THE SHIP'S PRINCIPAL DIMENSIONS

For varying the principal dimensions and ratios of the ship for the purpose of optimisation two premises are to be mentioned:

1) The aim of the calculation is only the optimisation of the principal dimensions and ratios in economical sense without, however, executing a profit-and-loss-account for the shipyard or the shipping company.

2) The exact specifications of the owners with respect to the performance of the ship are given either as contained in the tender enquiry or as building stipulations, meaning that relevant values such as the deadweight, hold capacity, speed, outfit, equipment, type of ship, size of superstructures and deckhouses, classification, steel construction, stability, etc. are specified by the owners.

When in the following considerations the principal dimensions are varied for optimisation, this is done, as far as possible, for individual parameters independent of others in order to assess the effects of the variation more readily and also to draw conclusions for other projects. However it must be stated that a certain interdependance among the dimensions exists so that independent variation is only partially practicable. If, for instance, the length is varied while remaining dimensions are kept constant, deadweight and capacity would change too.

Within this scope the variation of three parameters shall be investigated:

1) The Breadth of the Ship

2) The Length of the Ship

3) The Blockcoefficient

As first approximation the variations may be carried out with constant values for displacement and underdeck-space. If necessary, a compensation for the change of displacement and underdeck-space is to be applied for the variations of the dimensions. Furthermore for the variation care must be taken that the stability is sufficient and appropriate.

THE VARIATION OF THE SHIP'S BREADTH

Definition

The variation of the breadth of the ship is effectively carried out in the way that the ship's length and the blockcoefficient remain constant. At constant midship section area the ratio of breadth and draught is varied so far that displacement and underdeck-space remain constant.

The effect of the variation of breadth

If the breadth of the ship is varied at constant values of displacement, hold capacity, length and blockcoefficient, e.g. made bigger to influence the stability, the following effects occur:

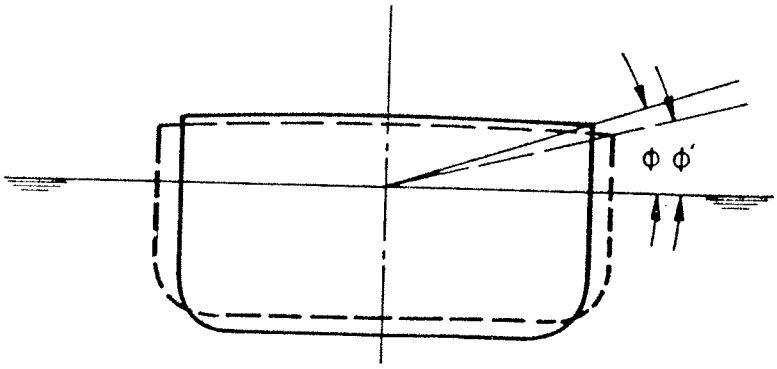
1) Increase in steelweight due to increased length to depth ratio. That means a reduction of the depth as associated with the height of a girder.

2) Increase in necessary power and in engine weight due to the higher resistance and-mostly-due to the lower propeller efficiency as a result of the required reduction of propeller-diameter necessitated by the draught reduction.

3) Increase in fuel consumption as a result of the increase in propulsive power.

4) Smaller freebord, which can cause a discrepancy between the legally required and the possible freebord.

5) Earlier immersion of the side of deck.

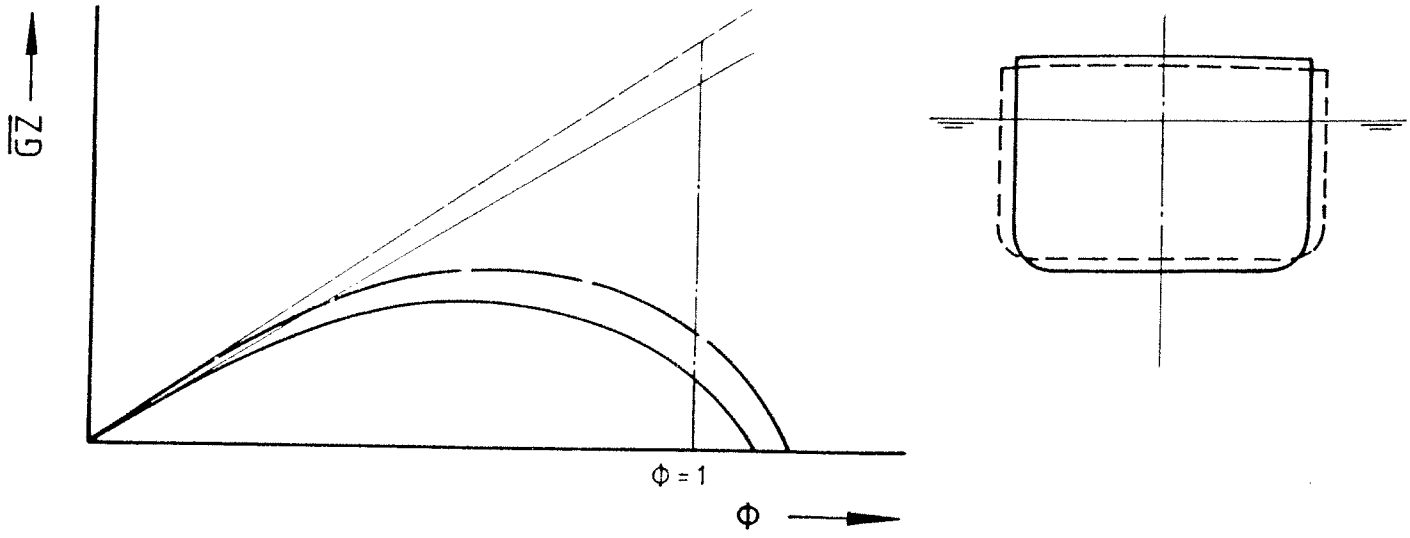


Assumed Variation of the breadth

The angle ϕ varies to ϕ'. This is a reduction of that angle which characterizes the immersion of the side of deck.

Change in righting arm curve

The curve of righting arms of the widened ship has not only a steeper initial tangent due to the greater metacentric height but also a greater range in spite of the earlier immersion of the side of deck.



Righting arms for varied breadth

Conclusions for the dimensioning the ship's breadth

Besides the increase in the stability the resulting facts are only negative so that one may say without doing any economic-calculation: The breadth must be of such a size that the safety is sufficient. Beyond this every further increase in breadth will only produce disadvantages unless there exists a restriction of draught.

THE VARIATION OF THE SHIP'S LENGTH

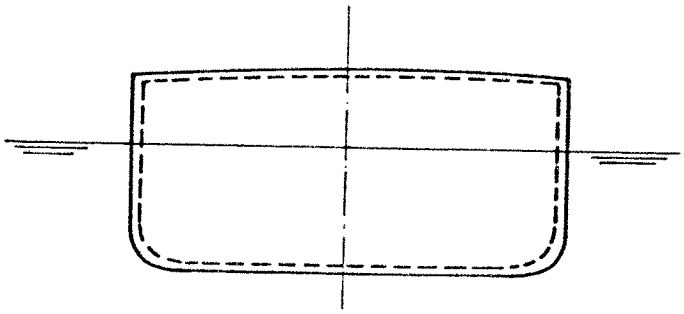
Definition

With the variation of the ship's length the midship section area will be changed in inverse proportion to the length.

The ratio breadth to draught remains constant. So as approximation a constant initial stability at constant values of displacement and under-deck-space will be preserved.

Effects of variation of length

The influence of the length on the design of a ship is very versatile. One may best recognize this influence by an imagined variation of length. For this the length of a ship at constant weight of payload or deadweight and at constant hold capacity is varied in a way that the midship section area changes in inverse proportion to the ship's length and that the ratios breadth to draught and breadth to depth are preserved. Thereby the constancy of displacement and underdeck-space can verify approximately the constancy of weight of payload and cargo space.



Variation of midship-section schematically

If, for instance, one increases the length by lo %, the scheme of variation requires a reduction of the midship section area by lo %, too. The depth, breadth and draught will be reduced by about 5 % each. The ratios breadth to draught and depth to draught as well as the fullness of displacement are preserved. The ratios length to breadth and length to depth will be increased by about 16 %. In detail the lengthening has the following consequences:

1) Increase in the necessary freebord according to the rules

2) Reduction in the initial stability

3) Better course steadiness

4) Poorer steerage

5) Increase in the steel weight

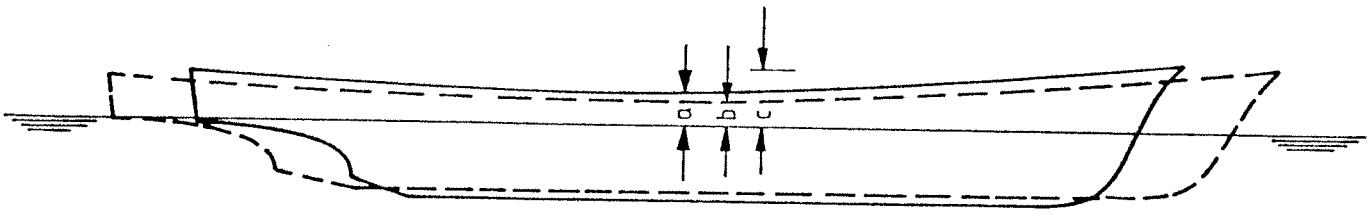
6) Reduction in the engine power and weight - independent of the Froude range

7) Reduction in the fuel quantity for the same cruising range

This listing can be considerably extended by including smaller effects.

On 1) Increase in the freebord according to the rules

With this lengthening the real freebord is reduced. But the freebord-requirement will be increased according to the rules. Hence these tendencies are opposite.



Influence of length-variation on freebord

a = Freebord of the initial form
b = Freebord of the strechted ship
c = Nominal value of freebord after lengthening

A shortening is always possible in consideration of the freebord-rules. A lengthening is practicable only in a limited way.

First of all only such cases shall be examined, where no discrepancy occurs between the freebord due to the rules and that freebord which results from the variation modul. The following types of ships come in question for variation:

a) Full scantling vessels with freebord

b) Shelter-deckers

c) For full scantling vessels shortening or lengthening with possibility of compensation of the "freebord according to the rules" by a smaller blockcoefficient.

On 2) Reduction in the initial stability

A compensation of the slight reduction in the initial stability caused by variation is possible through a slight increase in the breadth to draught ratio. With this, depth and draught will be reduced still further. The consequence will be an additional increase in steel weight and a decrease in power-saving.

Conclusion of the effects

Summing up, one can say that 3 groups of effects of the length-variation exist:

On 1) The compatibility with the freebord-rules may be endangered. By that limits of the variation are given.

On 2)-4) The slight changes in initial stability, course steadiness and steerage can be restored, if required, to the former values of the original form by small technical modifications.

On 5)-7) The changes in hull steel weight, in propulsive power and in fuel consumption affect considerably the three optimisation-criterions, namely:

1) Production costs

2) Profits and

3) Capital Recovery

Therefore particular attention has to be paid in the optimisation-calculation to the corresponding changes in steelweight, propulsive power and fuel consumption.

The economical optimum of the ship's length is not identical with any technical optimum as it is otherwise the case with many technical problems.

THE VARIATION OF THE BLOCKCOEFFICIENT

The realization of the variation

When varying the blockcoefficient it is necessary to take care that the design conditions are adhered too, especially that stability and underdeck-space remain constant over the variation. The ship's length remains constant, too. The midship section area will be varied in inverse proportion to the blockcoefficient. As first approximation the ratio breadth to draught will be kept constant to retain a given value of initial stability.

To keep the underdeck-space constant, the product $L \cdot B \cdot D \cdot C_{B,D} = \nabla_D$ has to be kept constant. (The differences due to the round of beam and sheer of deck are neglected here).
Then $C_{B,D}$ can be calculated by the following equation:

$$C_{B,D} = C_B + c \cdot \frac{D - T}{T} (1 - C_B) \qquad c = 0{,}3 \text{ for U-shaped sections}$$
$$= 0{,}4 \text{ for V-shaped sections}$$

This formula is also useful for many design purposes.
We assume here as first approximation that at constant underdeck-space the size of cargo space also remains constant.

With this simplification the change in the required machinery space and the variability of the badly utilizable spaces at the ship's ends remains unconsidered initially.

To keep the initial stability constant, the breadth must be varied according to the following formula developed by MÜHLBRADT:

$$B = \frac{B_O}{C \cdot [\frac{C_B}{C_{B,O}} - 1] + 1}$$

In this formula C has the value:

$C \approx$ o,12 for passenger ships and container ships
$C \approx$ o,16 for cargo ships and tankers

The change of the qualities by the variation

If a variation of the blockcoefficient is executed according to the above-mentioned scheme, a series of qualities will be changed with this step. Following changes result from a reduction of the blockcoefficient:

1) Reduction in the freebord according to the rules as far as the variation is carried out in the range $C_B >$ o,68.

2) Reduction in the area under the curve of righting arms, if starting from the same initial stability.

3) Slight reductions in the hull steel weight, if calculating with constant initial stability. The hull steel weight changes only a little, if the variation is carried out without a restriction on draught or without a constant draught.

4) Reduction in the necessary propulsive power and in the weight of the engine plant, if starting with usual cargo-ship-conditions

5) Reduction in the fuel quantity corresponding to the reduced propulsive power

6) Smaller resistance in seaway

7) Better seaworthiness reduced effects of slamming

8) Less "harbour-friendly", as there is only a shorter parallel middle-body available and the flare of frames is greater at the ship's ends.

9) Bigger hatches, heavier and more expensive

10) More unfavourable form of the cargohold

11) The limits of slipway, docks and locks are reached earlier

12) A more uniform and smaller wake. Less cavitation-danger of the propeller and less vibrations caused by the propeller

13) Greater lengths of derricks, if the breadth of ship is relevant for them.

14) More upperdeck-area

AIMS OF THE OPTIMISATION

As optimisation aims there are those in favour of the yard and those in favour of the owners.

For the yard those ship-dimensions are economically optimal which cause the least production costs.

A possible aim for the owner is the "optimisation of profits". Apart from the production costs the running costs are important for the owner. Thereby the former appear as capital costs, i.e. as interest and amortisations.

Assuming that all possible variants just correspond to the owner's requirements, it follows that all these variants have the same earning chances. At the same earnings the variant with the least expense has the most profit.

That means a maximizing of profit can be attained simplifyingly by minimizing of expense.

Mostly the maximizing of profitability or the maximizing of capital recovery factor (CFR) is regarded as the optimisation aim, the CFR being the quotient of profit to invested capital.

Normally the orderer is aware of the suitable choice of the optimisation aim.

The aim of the calculation is to choose that alternative from many possible designs fulfilling the technical requirements of the shipping company, which corresponds to the desired economical optimisation purpose. It is, however, not easily possible and also not necessary to give a predication of the absolute amount of the production costs, profits or capital recovery factor.

With these assumptions for such calculations it is possible to make use of some extensive simplifications.

COSTS DEPENDING ON VARIATION

If a ship design is varied in its principal dimensions and principal ratios in such a way that the shipowner's requirements are fullfilled in any case, the production costs vary as follows:

1) Hull steel
   On lengthening the hull steel becomes heavier and more expensive. With a reduction of the blockcoefficient the hull-steel only becomes a little heavier but more expensive in manufacturing.

2) Starting with normal cargo ship's forms and speeds the engine plant becomes smaller and cheaper on lengthening a ship and

with a reduction of the blockcoefficient

3) The other costs, in particular those of the outfit and equipment, are influenced only a little by the variation. But often it is worth to consider also the hatches, which change with the dimensions of the ship, as variation dependent item.

For the running costs this variability in production costs is important-in so far as the capital costs vary with it.

Further variable costs which come in question here are above all the fuel- and lubricating oil costs, which vary with the varying propulsive power requirements. The remaining costs are largely independent of variation.

Only in exceptional cases the machinery space and the qualification of the engine room staff may vary with the variation in necessary propulsive power.

For the described optimisation it is not suitable to compile the steel costs - as is usual at the yards - in a sum of raw material costs, labour man-hour and overhead costs. On the contrary it is beneficial, with due regard to the multitude of variants to compile the steel costs as the product of steel weight multiplied by unit price per manufactured ton of steel.

For the engine plant costs two methods may be efficient:

1) The product of propulsive power multiplied by standard price per power unit
   or
2) the costs of the main engine plus an existing reduction gear multiplied by a factor to take the "remaining engine parts" into account.

Methods of determination of hull steel weight and propulsive power

The hull steel weight and the required propulsive power are to be calculated for obtaining the variable costs for all variants. Therefore methods must be chosen which need less effort but are sufficiently accurate. Above all care must be taken as far as possible that for all variants the same method for calculation of hull steel weight and propulsive power is used. If it is necessary to change the method as for example because of exceeding the range of validity, at least one point ought to be calculated according to both methods. The results are to be harmonized by the use of coefficients in such a way that the curves of the outcome have no discontinuities.

POSSIBLE SIMPLIFICATIONS OF CALCULATION

As this optimisation deals with the technical and economical assessment of a number of ship designs, the effort can be enormous - even with the use of electronic data processing. To shorten the calculation following simplificating steps as have partly been mentioned earlier are available:

1) Choice of appropriate optimisation aim - for instance optimisation of the production costs only.

2) Limiting the economical calculation to costs - without considering the earnings.

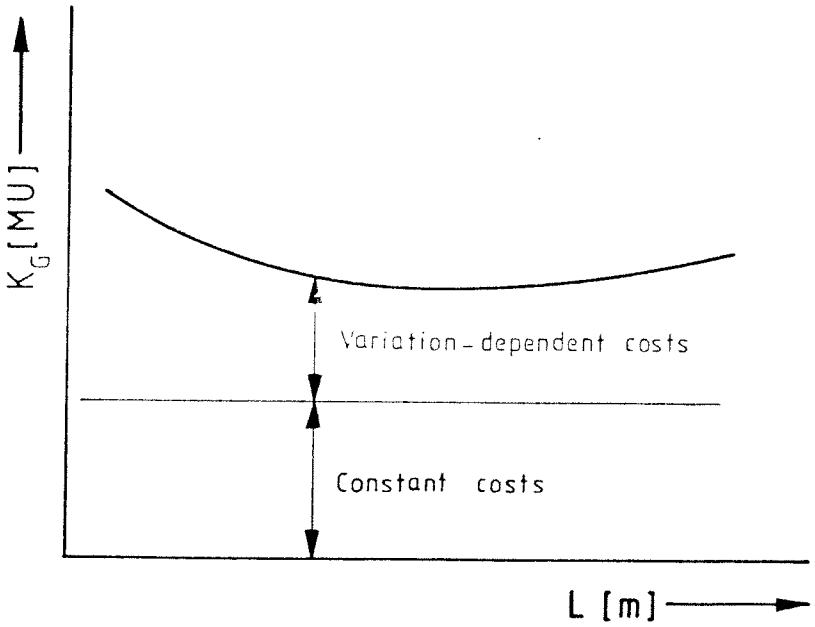
3) Limiting to variation-dependent costs. On subdividing the costs into those dependent on length and those independent of length, the minimun of total costs lies at those dimensions, for which the variation-dependent costs show a mini-

mum. Therefore it is sufficient to analyse only such costs which depend on variation. This splitting of the costs in those depending on variation and those being constant involves a reduction of the necessary input values apart from simplifying the calculation. - The practicability of such a costsplitting is given for all treated optimisation aims.

To work with "difference costs" is a special application of the conception of marginal costs, which has been introduced in public and industrial economics.

Marginal costs are defined as "cost increment on increasing the production by one unit".

Whereas in the case of an optimisation of the principal dimensions of a ship the size and performance capability remain constant according to definition, the components determining the size get changed. The difference costs can be imagined as a sum of the marginal costs, which result from the several increments and reduction applied in the process of form optimisation.



Splitting of the costs in those dependent on length and those independent of length

4) Application of difference costs

One does not need to consider the whole group of the variable costs of hull steel and machinery. It is sufficient to consider the difference in the costs compared with the "initial variant", the starting design.

The difference costs give the savings or increments in costs for the several variants. To work with difference costs has the following advantages:

- The difference costs show clearer figures, which are easier to compare.

- The calculation with difference costs can have a simplifying effect, since the scope to be considered is smaller. In an optimisation based on owners' interests for example one can calculate with a uniform rate of interest.

For executing the calculation one should start, as far as possible, from a suitable basic model having conventional dimensions and ratio values. For the succeeding variation attention must be paid to the fact that the shipowner requirements are not overfulfilled without inevitable reasons - as for instance greater stability due to restriction of draught -, because this might disturb the comparison of results.

If we had a mathematical equation of the costs instead of a few points of the cost-curve, we could find the minimum of the costs by differentiation. In the first derivation the constant factors disappear. For calculating the capital recovery factor one has to refer the profit to the whole invested capital. Here, too, the calculation can be carried out with this simplification as will be shown.

5) Calculations with technical simplifications, which are permitted for this problem:

- In the first place the complete variation can be executed using a constant displacement.

- For the variation of length the stability can be kept nearly constant by keeping the breadth to draught ratio constant.

- Execution of a freebord calculation only for critical cases.

- Calculation with steady alteration of propulsive power and steel weight.

With these simplifications of the calculation two categories are to be differentiated:

I Those simplifications, which do not contain errors, as for example:

- Choice of a suitably limited aim of optimisation

- Restriction to the variation-dependent costs

- Use of difference costs

II Simplifications,which exist in a difference between the real ship and the mathematical model, and at the expense of accuracy. These simplifications may influence the result. Therefore for these simplifications it has to be considered as to how far they are acceptable.

For technical simplifications it is possible to accept them, if the error arising from them is sufficiently small. This is done, for example, with the minor alterations in the equipment.

They can also be considered later. This comes in question for example in step-wise alteration of the steel costs when the number of bulkheads alters or in step-wise alteration of the power plant costs when the number of cylinders of engine alters.

AIM - FUNCTIONS

For the optimisation of the principal dimensions and principal ratio values one needs for all optimisation aims technically an as exact as possible knowledge of the hull steel weight and the propulsive power requirement as a function of the principle dimensions and principle ratio values.

An as exact as possible relationship between these values as function of the dimension variation is important hereby.

Furthermore the future prices and interest rates enter into the calculation for the shipowner-oriented optimisation aims. Plausible assumptions have to be made or so-called sensibility studies have to be carried out in this case. That means alternative calculations with very high and very low estimated values have to be executed which will localize the result.

AIM - FUNCTION FOR PRODUCTION COSTS OPTIMISATION

The production-difference-costs are composed of the sum of hull-steel-difference-costs and power-plant-difference-costs.

$$\Delta K_G = St_o \cdot k_{St,o} - St_n \cdot k_{St,n} + \Delta K_M \cdot C_M$$

For the difference costs dependent on power one may also write

$$+ \Delta P_B \cdot k_M \cdot C_M$$

Nomenclature:

Item	Description	Measuring Unit
ΔK_G	Production difference costs	Monetary Unit = MU e.g.: DM, $, £
St_o	Hull steel weight of the initial variant	t
St_n	Hull steel weight of the variant n	t
k_{St}	The specific costs of the installed steel. They may be assumed constant in the length variation, but they are variable for the variation of the blockcoefficient. Thus it follows $k_{St}=k_{St;o,8}+\frac{k_{St;o,5}-k_{St;o,8}}{o,8-o,5}(o,8-C_B)$ The second index means the respective blockcoefficient.	MU/t
ΔK_M	Difference costs for the main engine	MU
C_M	Factor taking into account the production costs of the "remaining parts" of the power plant	—
ΔP_B	Difference of the required propulsive power	kW
k_M	Costs of a propulsive power unit	MU/kW

AIM - FUNCTION FOR PROFIT OPTIMISATION

(Shipowner-oriented problem)

Here profit is the difference between returns and costs concerning the ship in an annual period. As already mentioned, not the profit itself is determined, but the technical data of those variants, for which the profit is a maximum.

Since by definition the earning chances of all variants are taken to be as equal, that variant is optimal with respect to profits, which shows the lowest operation costs. As mentioned above for the production costs, it is therefore sufficient here also, to minimize only the costs.

Since for determining the cost minimum it suffices to seek the variation dependent values, only the following cost groups are to be taken into account:

1) Capital-related costs for the hull steel

2) Capital-related costs for the power plant

3) Fuel- and lubricating oil costs

The production costs of the power plant depend on the trial trip conditions because of the rating of the main engine. The fuel costs have to be referred to the service speed. The sum of the variation-dependent annual costs amounts to:

$$k = St \cdot k_{St} \cdot (a_{St} + v_{St} + r_{St}) + K_M \cdot C_M \cdot (a_M + v_M + r_M) + P_{B,D} \cdot F \cdot (k_T \cdot b + k_{Sch} \cdot s)$$

Nomenclature:

Item	Description	Measuring Unit
$P_{B,D}$	Brake power at service speed	kW
F	Annual running time	h
k_T	Costs of one ton of fuel (or heavy oil)	MU/t
K_M	Costs of the main engine - if necessary with reduction gear	MU
b	Specific fuel consumption	t/(kW · h)
k_{Sch}	Costs of one ton of lubricating oil	MU/t
s	Specific lubricating oil consumption	t/(kW · h)
a	Annuity factor	—
v	Assurance - and tax-factor	—
r	Repairs-factor	—

For the profit-optimisation it is also advantageous to work with difference costs because of more clarity. Furthermore the work becomes easier from the fact that for the capital-related costs, depending upon how the capital is raised, one may calculate either with an averaged rate of interest or with that of the last mortgage, which is the marginal rate of interest. So the difference costs are:

$$\Delta k = [St_o \cdot k_{St,o} - St_n \cdot k_{St,n}] \cdot (a_{St} + v_{St} + r_{St})$$
$$+ \Delta K_M \cdot C_M \cdot (a_M + v_M + r_M)$$
$$+ \Delta P_{B,D} \cdot F \cdot (k_T \cdot b + k_{Sch} \cdot s)$$

In words:

The annual cost-differences are equal to the sum of the differences of the enlarged capital-related costs for hull steel and power plant as also the differences of the fuel- and lubricating oil costs. In this formula the difference-costs for hatch covers, other outfit parts and possibly altering staff-requirements are not considered.

AIM - FUNCTION FOR CAPITAL RECOVERY OPTIMISATION

The capital recovery may be defined as follows without considering the origin of the capital:

$$\text{Capital Recovery Factor [CRF]} = \frac{\overbrace{\text{annual returns - annual costs}}^{\text{annual profits}}}{\text{invested capital}} \times 100$$

CRF is measured in percent (%) per year (a).

Here the factors depending on variation can also be split in a constant and a variable part. For the annual costs you may say: "Annual costs of the zero-variant plus/minus difference-costs". Accordingly for the invested capital the sum of the invested capital of the zero-variant and the difference investment can be inserted. The equation for CRF considering the difference costs is the following:

$$CRF = \frac{E - (k_o \pm \Delta k)}{K_{G,o} \pm \Delta K_G} \times 100$$

Nomenclature:

Item	Description	Measuring Unit
CRF	Cost Recovery Factor	%/a
E	Annual returns	MU/a
$K_{G,o}$	The capital to be invested for the zero-variant	MU

In this equation the simplification as mentioned before to take only the variable costs into account is only partially practicable. If for the annual profit of the zero-variant $E - k_o$ as simplification the value zero is inserted and if for the production costs of the zero-variant an estimated value is assumed, the calculated CRF has its maximum at the "dimensions of optimal CRF". Therefore additional rough estimated values for the production costs and for the annual profit of the zero-variant are to be introduced into the calculation.

RESULTS OF THE CALCULATION

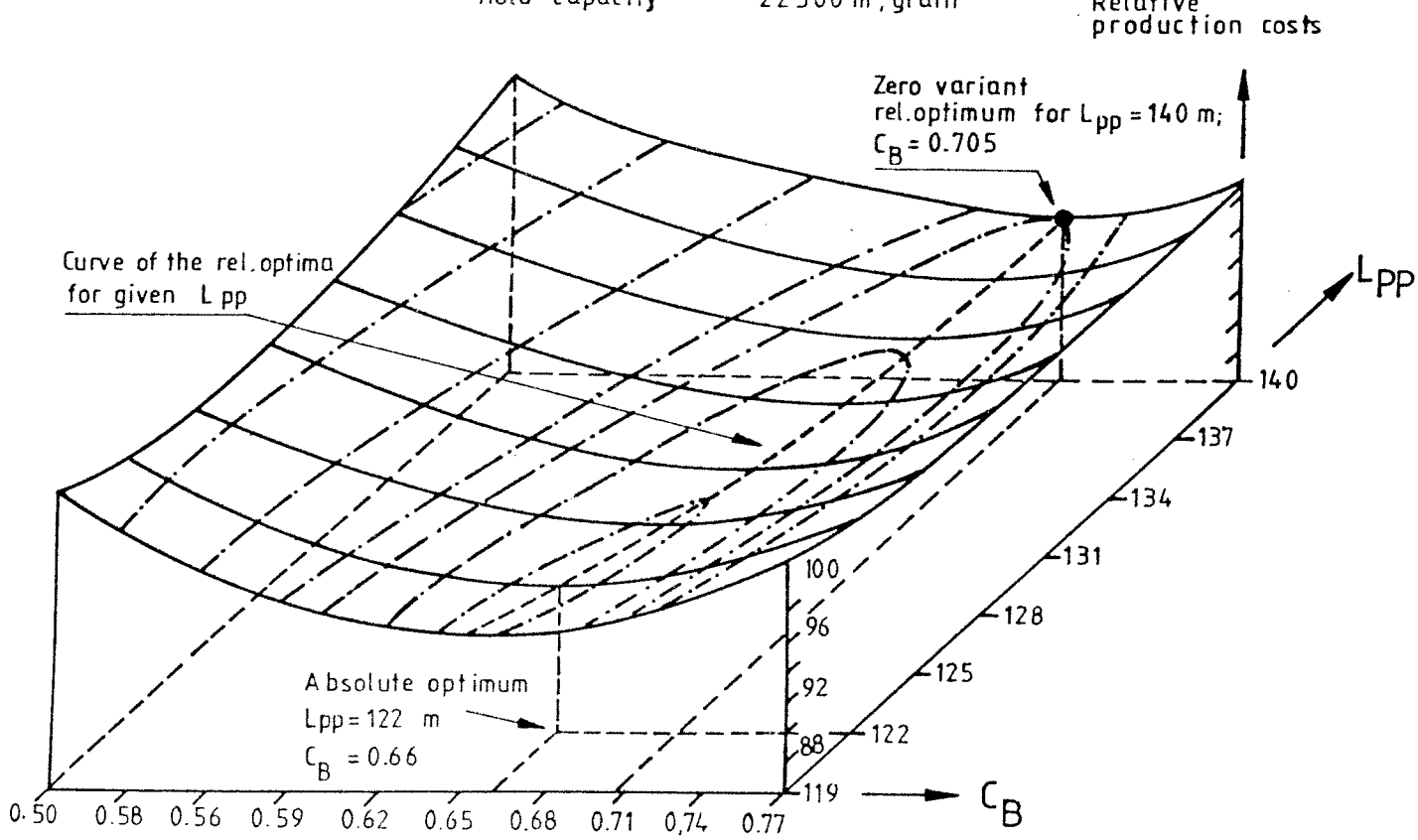
If the task is to optimize length and blockcoefficient, first of all every interesting length can be calculated and again for each length the blockcoefficient can be varied. One has to imagine a cost-area $K = f\ (C_B, L)$, of which the spatial minimum is to be determined.

Such a procedure means a big effort of calculation which is practicable only with the aid of EDP-plants. Suitable for such

problems is for instance the FORTRAN-PROGRAM OPTIMA, developed at the University of Aachen.

Example:

Multi purpose cargo ship 16300 tdw
- Trial speed 16,3 kn
- Hold capacity 22300 m^3, grain



Total costs as function of length and blockcoefficient

DIFFERENCES BETWEEN THE DIMENSIONS OF BUILT SHIPS AND THE RESULTS OF THE OPTIMISATION

The optimized main data diverge a little from the usual principal dimensions and principal ratio values of built ships, mostly towards shorter ships or less full ships respectively. One now has to ask:

- What are the reasons for this tendency?
- Is such a divergent result correct?

There are several reasons for the optimisation results differing from the statistical values:

1) Usually no optimisation is made of the principal dimensions when designing ships. One relies largely on previous facts and statistical values. Hence, one copies something, over which one knows that it is all right in technical respect, but over which one doesn't know if it corresponds in economical respect to the present requirements.

2) In the course of the last decades ships have become faster in general. Often one has simplified the designs of faster ships by installing a more powerful engine in an existing design of a slower ship. In this way many ships appear in the statistics as too full in relation to the speed and are accordingly fitted with a relatively high propulsive power.

3) Often it is to be found that fuller ships are more popular than less full ones:

   a) The fuller ship with the greater parallel midshipbody is considered as more tailor-made for harbours.

   b) The fuller ship has bigger hold floor-spaces, the hold sides run more vertically. The cargo hold of a fuller ship is sometimes better to load.

   c) At the yard the greater degree of difficulty in constructing the less full ship is more evident than the reduced costs of the respective power plant.

   d) Psychologically the shipowner may have the impression that a less fine ship as opposed to a fine ship of the

same size offers bigger carrying capacity and thus more freight, which he may be throwing away with the decision for a less full ship.

4) Finally the optimisation result too may be subject to deficiencies despite faultless calculation, which arise from the inaccuracy of the components. When calculating the resistance and steel weight, for instance, the alteration-tendencies and absolute values may be different depending upon the method used,i.e. even on fixing the input data, the cost curves can have different run depending upon the method used.

By varying the initial values and by using different calculation methods for the steelweight and power requirement the contour lines of the cost curves deliver a more or less broad band over the principal dimensions and the blockcoefficient. The scatter within the band width is also to be explained by insufficiencies of the methods.

For comparing the calculation results with the data of comparative ships it is pointed out once more that the economical disadvantages on deviating from the economical optimum increase progressively and that small deviations in the vicinity of the optimum hardly affect the economy. Moreover it has to be observed that the dimensions for optimal capital recovery factor do not diverge too far from those of the smallest production costs. Hence, if one executes an optimisation according to the criterion of the smallest production costs - starting with the usual dimensions - , one has already done an important step towards an optimal capital recovery factor.

EVALUATION OF THE CALCULATION RESULTS

The cost curves plotted over the principal dimensions and principal ratio values are apt to give more or less wide ranges, if one varies the unknown and the remaining indefinite initial values by the aid of a sensibility study. A certain range of tolerance is based on the fact that the sub-routines which enter into the complete procedure and which are interchangeable, have certain shortcomings. For instance the commonly used resistance calculation methods show a varying dependence on the fullness. Differences in the gradients of the resistance curve often lead to bigger errors than the differences in the absolute values of resistance.

Generally it can be stated that some uncertainties give the optimisation-result some margin of tolerance, which however is small. Within this margin one can lean on the following aspects in practical application:

- Tendency towards usual values, if deviations exist.
- Take into account navigational advantages, for instance the choice of a ship with smaller draught.
- Tendency to other optimisation-aims. If the capital recovery, for instance, is optimized one may choose the ship with the smallest production coats within the mentioned range.
- Tendency to a ship with the smallest loss of speed in seaway. This is mostly a ship with the smallest blockcoefficient.
- Tendency to a ship with the smallest fuel consumption.

This range, which arises from different reasons during the process of optimizing, demands similar further decisions as the conventional choice of principal dimensions.

The appropriate choice of the principal ship dimensions even when using a program, should be carried out by a naval architect

who has an understanding of the functional relationships of input data and design considerations.

An optimisation-calculation cannot relieve the project engineer of his responsibility in determination of technically appropriate and economically advantageous principal particulars and form ratios. It only offers him a greater certainty as basis for his decisions.

In conclusion it can be stated that the optimisation of the dimensions according to the method in question leads to more economical ships such that the effort involved is relatively small as against the attainable benefit.

In many cases it is possible to reduce the production costs of a ship by several percent points. Thereby 5 % is not a rarely attained value.

As far as methodics are concerned the given procedure is rather a think and proceed mode for the optimisation of principal ship dimensions. This procedure contrasts with other optimisation methods in so far as terms like production costs and capital recovery as usually used by yards and shipowners are not calculated, but rather decision criteria specially derived for this purpose are made use of. These functions are kept in such a simple form that the naval architect is able to select the optimum according to the desired criterion, for instance the production cost-or capital recovery optimum. For the production costs the simplification is realized by taking only the variation dependent and not all costs into account. For the optimisation of the capital recovery factor which in fact must include all production costs, the latter are substituted by a sum of exactly calculated variable costs and a uniform estimated sum representing the remaining costs for the whole variation series.

With this brief review it is as such not possible to carry out the optimisation in a given case. As reference, therefore, I draw the attention to my book:

"Entwerfen von Schiffen"("Design of ships"),
Publisher: Koehler, Herford, FRG.

Furthermore I mention the FORTRAN-Program OPTIMA at the Technical University of Aachen, Lehrstuhl für Schiffbau, Entwurf und Dynamik.

WEST EUROPEAN GRADUATE EDUCATION MARINE TECHNOLOGY

SHIP DESIGN FOR FUEL ECONOMY

Eighth Graduate School, Gothenburg

THE INFLUENCE OF SHIP OPERATION ON DESIGN

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SUMMARY

The emphasis in this lecture is mainly on the ship designer (whether in a shipyard or shipowner's or consultant's office), who needs to estimate how the overall performance of alternative ship designs is affected by operational aspects. Time in port and cargo handling influence overall economics strongly, but the designer is concerned more with long term trends than with short term predictions more suitable for ship and port operators. The proportions of port time (including waiting time) to sea time will be discussed for various ship types and trades, with consideration of the utilisation of time in port. The statistical variation in port time on different calls at the same port, and between different ports, will be indicated with some data.

For many vessels, consideration of a range of operating drafts is important, since even on 'loaded' voyages, a wide range of drafts may be experienced in service. Overall the emphasis will be on the need to use realistic data on draft limitations, cargo stowage factors, steaming distances loaded and ballast, cargo handling rates etc. Together these affect both technical and economic performance, by influencing both the revenue generating potential and the operating costs.

THE INFLUENCE OF SHIP OPERATION ON DESIGN

Introduction

Ship Design for Fuel Economy means not only reducing power requirements and fuel consumption, but also maximising the difference between total income and total costs. The designer therefore needs to address his attention to developing a design of ship that over its anticipated life and anticipated operating pattern will be more profitable than all the other alternative feasible designs. In the case of closed system vessels (Ref. 1) (i.e. those vessels dedicated to a service with little outside competition, such as LNG carriers), the overall system may be optimised, in the choice of type of vessel, number of vessels, capacity, speed, terminal facilities etc. In the case of open system vessels (e.g. tramp type vessels subject to maximum competition) or semi-closed system vessels (e.g. cargo liner types subject to limited competition), the designer must try to establish ship characteristics which will be tuned to a range of anticipated operational requirements over a life of 15 or more years. The designer therefore needs to take a long term view as to the expected level and variation in such aspects as cargo handling rates, voyage distances, port time, availability of part cargoes, amount of ballast steaming etc. In particular, the ratio of sea time to port time has a marked influence on the potential value of fuel saving designs.

Where a ship is being designed for a limited range of trades, e.g. a deepsea cellular container ship, or oil products tanker, owner and designer (who may be employed by the owner, the prospective builder, or a consultant) may be able to establish the required operational performance within limits. A ship intended for more general trading, e.g. a multi-purpose bulk carrier, will have less clear-cut requirements, while a similar vessel offered as a standard product by a shipbuilder needs to appeal to a wide range of potential owners, and charterers.

Operational Factors Influencing Ship Revenue

Most owners will be examining the potential profitability of a new design and alternatives by forecasting revenue earning potential, and total operating costs. Annual revenue will depend on anticipated annual cargo carried, and forecast average freight rates. The designer's crystal ball will be no better than anyone else's at forecasting freight rates; although it is possible to identify ship technical characteristics which attract above average freight rates - see Ref. 1, which identifies amongst other things low fuel consumption as important for bulk carriers. So the designer is likely to concentrate on features which are likely to maximise cargo carried. One can write:-

Annual cargo = Max. cargo payload x Load factor x Round trips per annum x 2

N.B. 2 because there are two revenue generating legs on each round trip

Maximum cargo payload may be an obvious figure for weight limited ships like ore carriers, or volume limited ships like liquefied gas carriers, but may vary with cargo type or stowage factor for other ship types. Fig. 1 shows two histograms of oil products specific gravities for clean and dirty products; thus the performance of oil product tankers needs to be evaluated in two or three conditions if there is insufficient cubic capacity to load a full deadweight of every cargo. Langenberg (Ref. 2) shows that for container ships, payload may be limited by weight (draft), or by capacity (TEU slots) or by stability - see Fig. 2.

Unless the anticipated cargo varies within only narrow limits over a shipload, or over a year, it would be wise to investigate the economic performance of container ships over a range of assumed cargo availabilities. Either a weighted average of the different payloads may be taken, or in more extensive investigations, a simulation model developed. Fig. 3 shows a histogram of actual container cargo weights for a particular trade. Discussion with operators will often provide such data, although care is needed to distinguish between slots for 20ft and for 40 ft. containers, since mean values are different as shown in Table I.

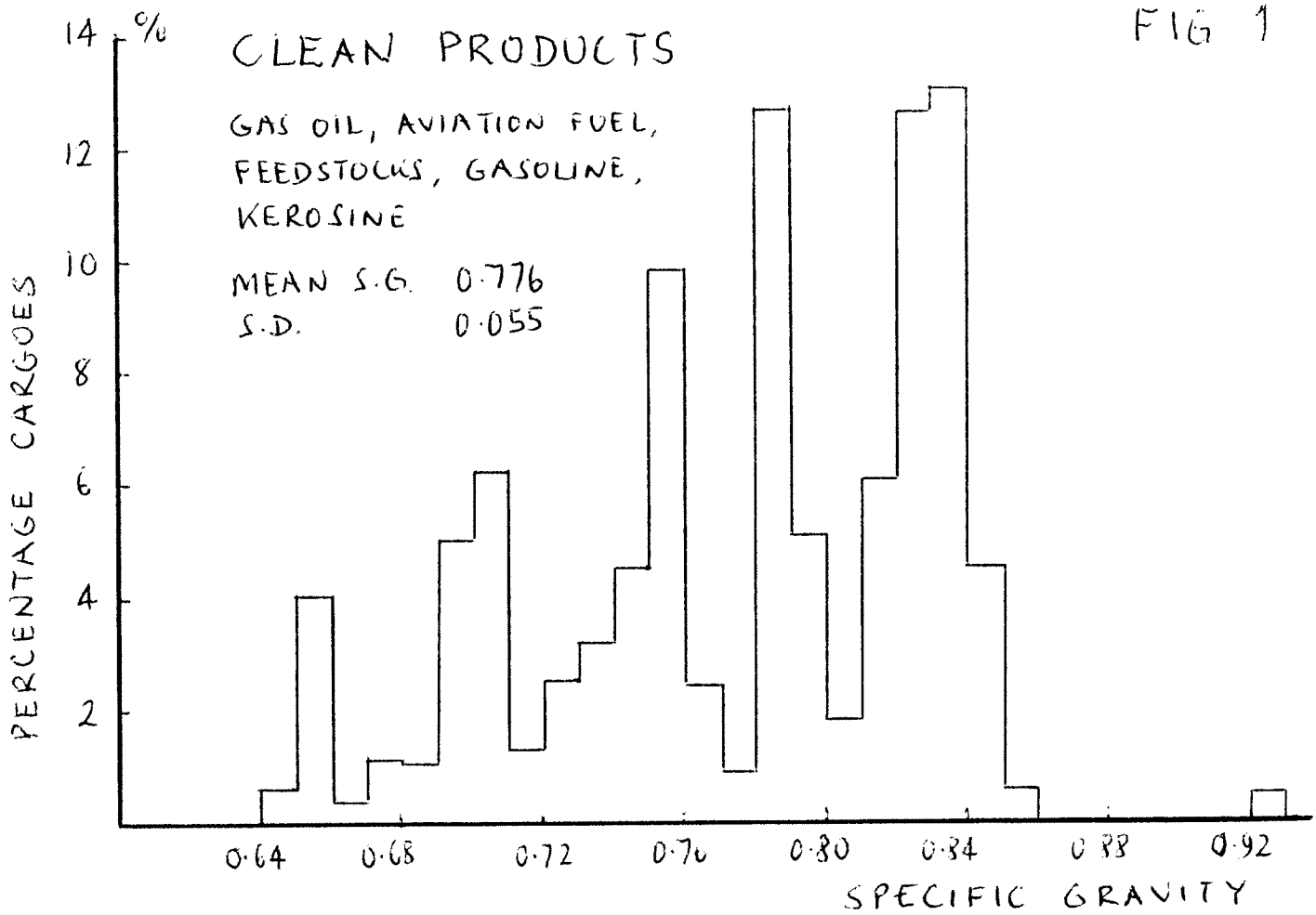
TABLE 1

TYPICAL DEEP SEA CONTAINER STATISTICS

	20 foot dry cargo	40 foot dry cargo
Dimensions, ft.	20 x 8 x 8½	40 x 8 x 8½
m	6.06 x 2.44 x 2.59	12.19 x 2.44 x 2.59
Maximum weight, tonnes	20.3	30.5
Empty weight, t	2.2	3.6
Maximum payload, t	18.1	26.9
Mean weight, t	13.2	16.2
Standard deviation, t	5.6	5.9
Mean cargo, t	11.0	12.6
Mean cargo per TEU, t	11.0	6.3
Cubic capacity, m^3	33	67
Nominal stowage factor, m^3/t	1.82	2.49
cu.ft/ton	65	90
Mean stowage factor, m^3/t	3.00	5.32

N.B. Excludes empty containers, and specials (reefer, tank etc).

FIG 1


CLEAN PRODUCTS
GAS OIL, AVIATION FUEL, FEEDSTOCKS, GASOLINE, KEROSINE
MEAN S.G. 0·776
S.D. 0·055
%
14
12
10
8
6
4
2
PERCENTAGE CARGOES
0·64
0·68
0·72
0·76
0·80
0·84
0·88
0·92
SPECIFIC GRAVITY

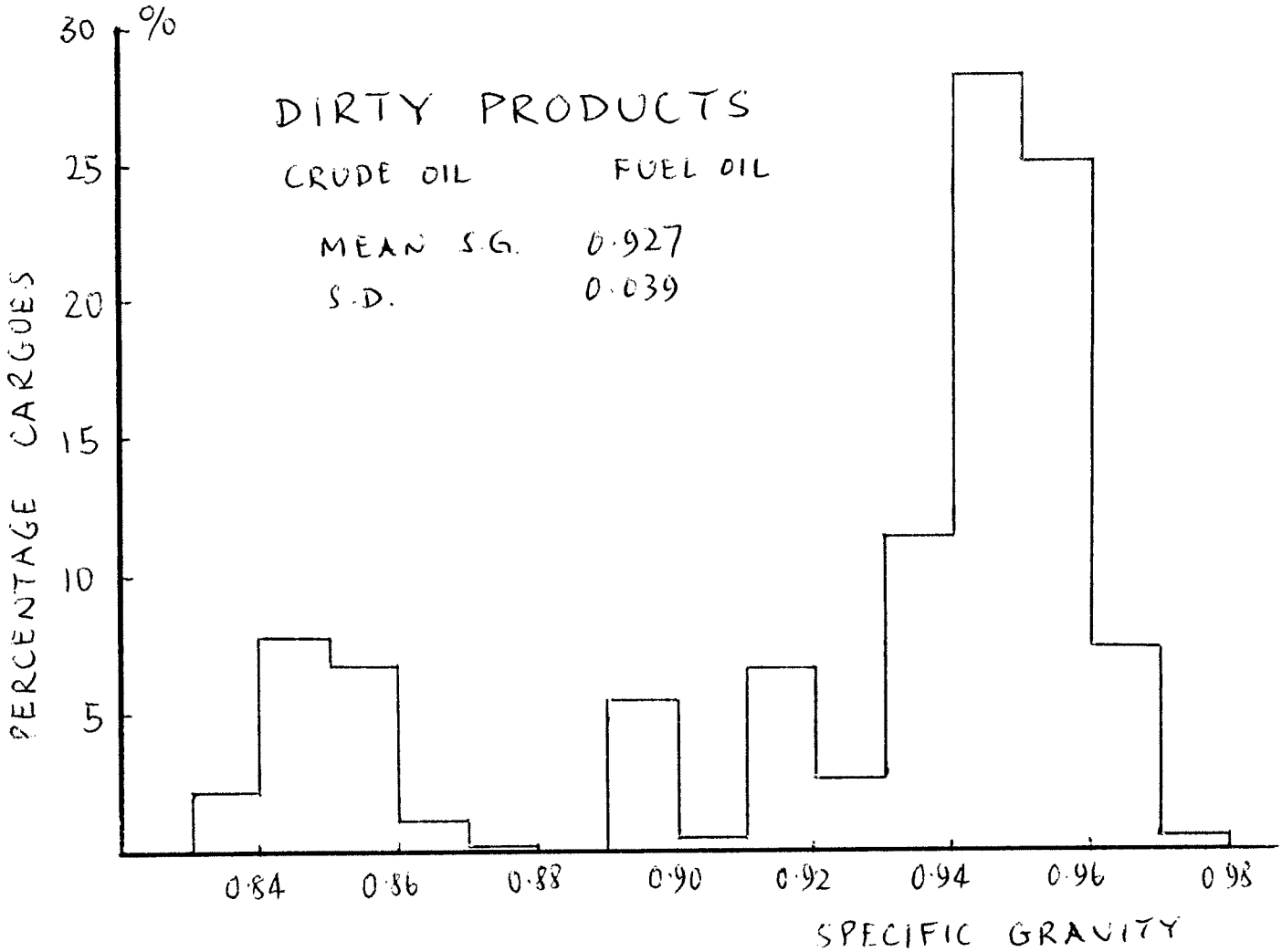

DIRTY PRODUCTS
CRUDE OIL
FUEL OIL
MEAN S.G. 0·927
S.D. 0·039
%
30
25
20
15
10
5
PERCENTAGE CARGOES
0·84
0·86
0·88
0·90
0·92
0·94
0·96
0·98
SPECIFIC GRAVITY

FIG. 2

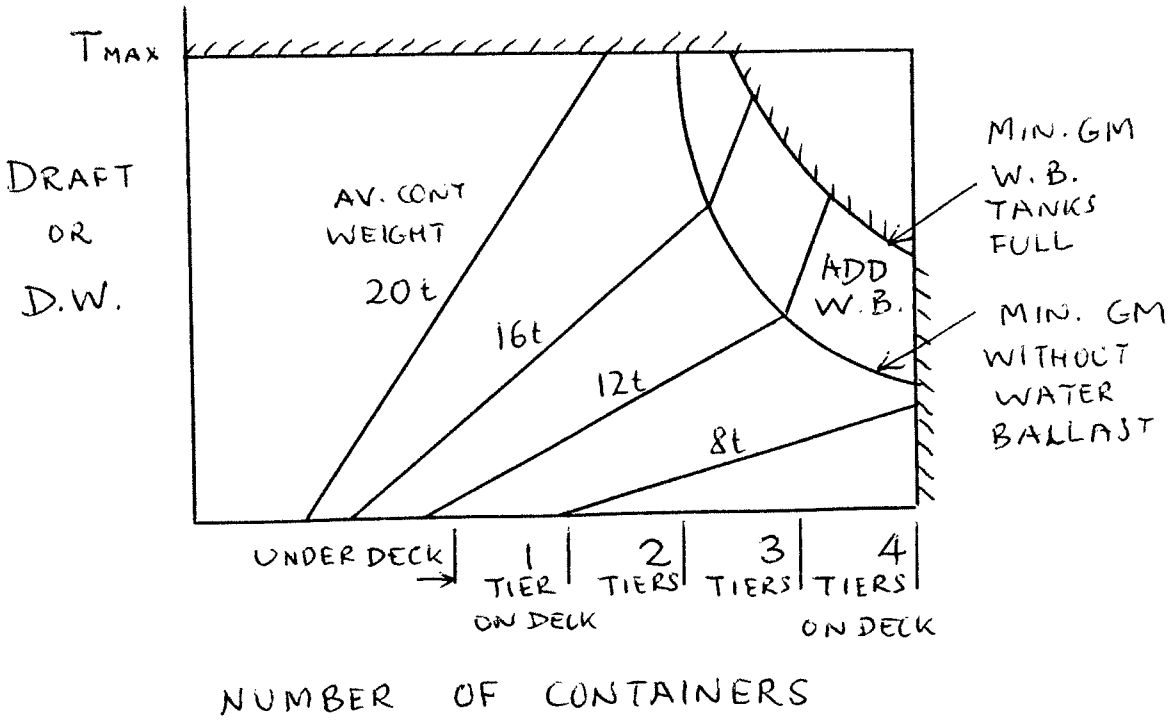
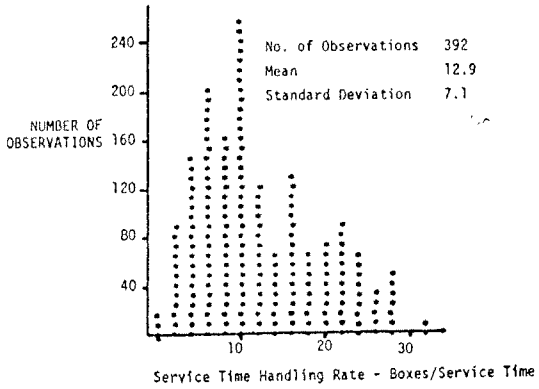
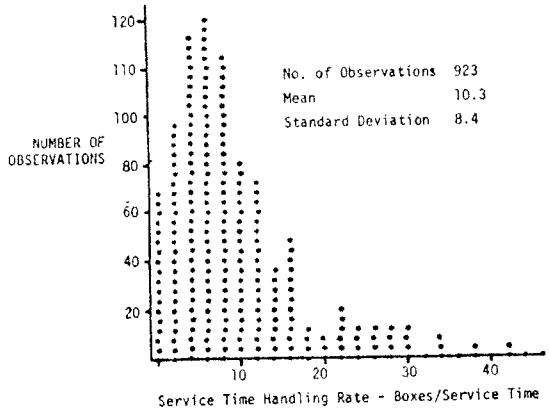


FIG. 4

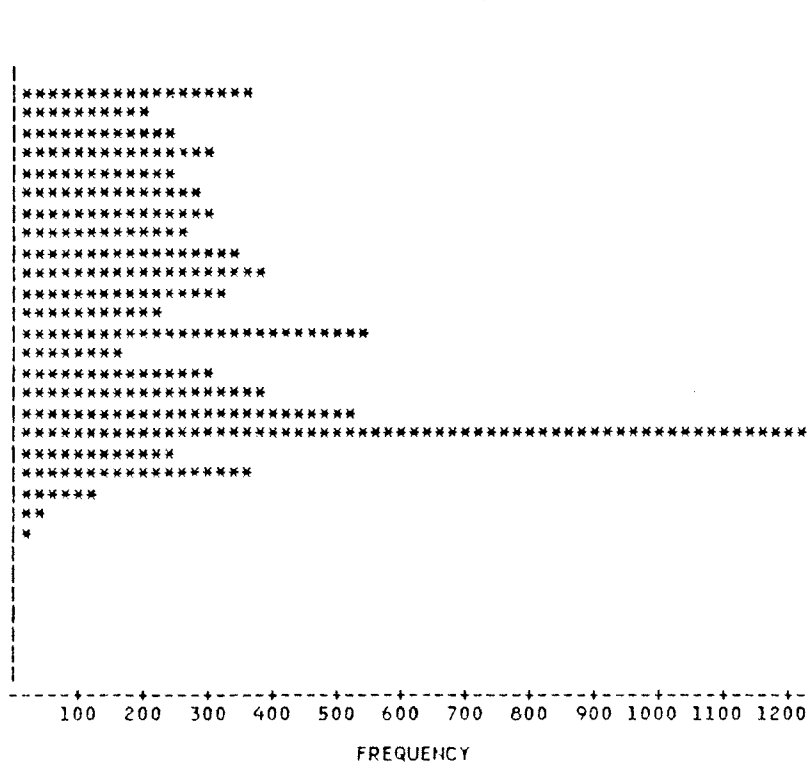


Distribution of service handling rates for ships more than 300 TEU capacity.



Distribution of service handling rates for ships less than 300 TEU capacity.

FREQUENCY BAR CHART



MIDPOINT WEIGHT	FREQ	CUM. FREQ	PERCENT	CUM. PERCENT
1	364	364	4.95	4.95
2	196	560	2.66	7.61
3	237	797	3.22	10.83
4	293	1090	3.98	14.82
5	249	1339	3.38	18.20
6	286	1625	3.89	22.09
7	300	1925	4.08	26.17
8	260	2185	3.53	29.70
9	336	2521	4.57	34.27
10	385	2906	5.23	39.50
11	312	3218	4.24	43.74
12	221	3439	3.00	46.74
13	536	3975	7.29	54.03
14	158	4133	2.15	56.18
15	301	4434	4.09	60.27
16	375	4809	5.10	65.37
17	528	5337	7.18	72.54
18	1227	6564	16.68	89.22
19	247	6811	3.36	92.58
20	362	7173	4.92	97.50
21	115	7288	1.56	99.06
22	31	7319	0.42	99.48
23	21	7340	0.29	99.77
24	5	7345	0.07	99.84
25	0	7345	0.00	99.84
26	1	7346	0.01	99.85
27	0	7346	0.00	99.85
28	2	7348	0.03	99.88
29	0	7348	0.00	99.88
30	9	7357	0.12	100.00

FIG. 3. Container cargo weight distribution of European exports to Far East. 6321 20 ft plus 518 40 ft containers. Weight per TEU; i.e. 40 ft divided by two. For total weight, add empty weight of container, approx. 2 tonnes.

Mean weight:	12.17 tonnes	S.D. 6.05
Mean cubic volume:	19.78 cu.m.	S.D. 7.37

Mean values, standard deviations and actual weight distributions are useful design figures, not only in estimating the likelihood of exceeding stacking weights in cells or on hatch covers, but also assessing limits on ship strength, ship motions and ship stability. Of course, the operator is controlling the container weights and stowage pattern on each voyage, allowing for loading and discharging ports, but the designer needs aggregated information in the earlier stages to define the long term performance, as well as operating limits which can usually be extended by the use of water ballast. Such differences in payload may also affect operating draft, and consequently ship speed and fuel consumption - see later.

Load factor measures the average payload on board on a round trip as a proportion of the maximum possible. Although there does not seem to be a universal definition of load factor, I split it into two components:-

$$\frac{\text{Average cargo payload on loaded legs of voyage}}{\text{Maximum cargo payload}}$$

$$\times \quad \frac{\text{Average miles steamed with cargo}}{\text{Total miles steamed}}$$

The latter term is essentially (1 - proportion of miles in ballast), so that a ship like a bulk carrier which loads on average 90% of maximum payload, and steams 40% of total miles in ballast would have a load factor of 54%. Long term average load factors for actual ships can vary from over 70% for efficient cargo liner types to below 50% for ships subject to a high degree of part loads and ballast steaming, e.g. some products carriers. Table 2 gives an indication of figures which may be used for a range of ship types in the absence of more precise data.

Round trips per annum are estimated for a typical operating profile; adjusted to twelve months to link in with operating and financing costs which are usually related to annual periods, i.e.

$$\frac{365 - \text{annual days off-hire}}{\dfrac{\text{Round trip miles}}{24 \times \text{av. service speed}} + \text{Port days per round trip}}$$

Off-hire includes time when the ship is under repair, docking or broken down at sea. While Table 2 is based on a typical 15 days off-hire each year, for specific studies it is usually better to adopt a figure related to the long term average for the ship type in question. Fearnley & Egers used to publish an annual off-hire survey (last in 1971) showing the following averages:

TABLE 2

TYPICAL SHIP VOYAGES AND PRODUCTIVITIES FOR NEW SHIPS

Ship Type	Summer Deadweight	Max. cargo payload tonnes	Av. speed knots	Av. miles per R.T. (p.a.)	Port calls per R.T. (days/call)	Port days per R.T. (% time)	Sea days per R.T. (p.a.)	Total days per round trip	RTPA 350 days	Load Factor % miles loaded % full	Load Factor % miles loaded Overall	Ship Productivity Tonnes cargo p.a. (per t DW)	Ship Productivity Tonne-miles p.a./10^6 (per t DW)
Large tanker	260,000	249,000	14	16,000 (106,000)	2 (2.5)	5 (9)	48 (317)	53	6.60	50 96	48	1,580,000 (6.1)	12,600 (49,000)
Products carrier	30,000	28,500	15	5,000 (88,000)	3 (2.0)	6 (30)	14 (245)	20	17.5	50 90	45	450,000 (14.9)	1,120 (37,000)
LNG Carrier	40,000	37,000 80,000m^3	19	6,000 (123,000)	2 (2.0)	4 (24)	13 (268)	17	20.6	50 100	50	760,000 (19.0)	2,290 (57,000)
Ore carrier	110,000	106,000	15	10,000 (97,000)	2 (4.0)	8 (22)	28 (272)	36	9.73	50 100	50	1,030,000 (9.4)	5,150 (47,000)
Bulk carrier	26,000	24,000	14.5	11,000 (82,000)	3 (5.0)	15 (32)	32 (243)	47	7.45	70 90	63	225,000 (8.7)	1,240 (48,000)
Break-bulk cargo (two decks)	16,000	14,000	14.5	15,000 (73,000)	5 (5.8)	29 (40)	43 (209)	72	4.86	80 75	60	82,000 (5.1)	610 (38,000)
Fast cargo liner	15,000	11,000	18	18,000 (73,000)	11 (4.0)	44 (51)	42 (171)	86	4.07	95 75	71	64,000 (4.3)	570 (38,000)
Large container ship 2300 TEU	36,000	27,000 11.7t/cont	22	12,000 (127,000)	6 (1.7)	10 (30)	23 (244)	33	10.6	100 75	75	430,000 (11.9)	2,530 (72,000)
Barge Carrier	35,000	27,000 75 barges	18	14,000 (114,000)	8 (1.2)	10 (23)	33 (269)	43	8.14	100 70	70	308,000 (8.8)	2,150 (62,000)
Deep sea RoRo	20,000	14,000	20	20,000 (106,000)	12 (2.0)	24 (36)	42 (223)	66	5.30	100 70	70	104,000 (5.2)	1,040 (52,000)
RoRo freighter (short sea)	4,500	3,000 90 40ft trailers	17	800 (93,000)	2 (0.5)	1 (33)	2 (233)	3	117	100 60	60	420,000 (94)	168 (37,000)
Coaster (single deck)	3,000	2,800	12	1,200 (51,000)	2 (2.0)	4 (49)	4.2 (179)	8.2	42.7	60 90	54	129,000 (43)	77 (26,000)

days off-hire:

584 tankers over 60,000 dwt		20.0 days
of which:	steam turbine	19.3
	diesel	20.7
	oil company	21.1
	independents	19.5

In the absence of published data, operators may sometimes provide information, though it is necessary to ensure that a large enough sample of ships is included to give a realistic long term figure, and then to allow for any special features, e.g. novel machinery.

Typical voyages may be determined either from discussions with operators, or from analysis of published information, e.g. Lloyds Voyage Record (Ref. 3). With some background knowledge of the actual ship and principal

BREMEN EXPRESS —cont.
Southampton Oct 1/4
Suez Pd Oct 11
Singapore Roads Oct 21/22
Tokyo Oct 28/30
Kobe Oct 31/Nov 2
BREMEN MARU (Ja)
Kobe Oct 4/6
Abt 16m off Tomisaki Oct 7
Hong Kong Oct 9/10
Singapore Roads Oct 13/14
Tanga Oct 22/24
Mombasa Oct 24/30
Dar-es-Salaam Oct 31/Nov 6
Mombasa Nov 12
BREMER URANUS (Ge)
Kenitra Oct 2/5
Figueira Oct 7/9
Honfleur Oct 12/14
London Oct 15/20
Kenitra Oct 25/26
Vianna Oct 28/31
Honfleur Nov 3/5
Lowestoft Nov 8/10

BRIGHT VEGA —cont.
Melbourne Oct 3/3
Yokohama Oct 23/25
Nagoya Oct 31/31
Brisbane Nov 13/13
Sydney Nov 15/15
Melbourne Nov 17/18
BRIGIT (Da)
Ardersier —/—
Rouen Sep 17/17
Antwerp Sep 21/28
Dover Strait Pd Sep 29
Gibraltar Pd Oct 5
Suez Pd abt Oct 15
Aden —/Nov 5
Port Said Pd Nov 12
BRIGITTE GRAEBE (Ge)
Bremen Oct 10/11
Rostock —/—
Londonderry Oct 21/23
Glasgow Oct 23/27
Mantyluoto Nov 6/8
Helsinki Nov 9/10
Bremerhaven Nov 14/15

BRITANIA (Ge)
Tripoli(Lebanon) Oct 23/24
Beirut Oct 25/25
Mersin —/—
Marseilles Nov 1/4
Valencia Nov 5/6
Barcelona Nov 6/6
Leghorn Nov 9/10
Beirut Nov 15/16
BRITANIS (passenger) (Pa)
New York Sep 9/9
New York Sep 11/11
New York Sep 13/13
New York Sep 14/14
New York —/Sep 18
New York Sep 18/20
Providence Sep 22/—
New York Sep 27 1982 - Laid up
BRITANNIA (Da)
Rotterdam Aug 16/18
Umea Aug 22/24
Svendborg Aug 26 1982 - Laid up / Sep 24
Hamburg Sep 25/28

BRITISH PROGRESS —cont
Brunei Bay(Sabah) Jne 15 1978 - Laid up
BRITISH PROMISE (tank) (Br)
Cape St Vincent Anch off Sep 5/—
Gibraltar Pd Oct 1
Malaga Oct 1
Renamed SANANDAJ (Ir)
BRITISH PURPOSE (tank) (Br)
Singapore Roads Jne 8 1978/Jne 15 1978
Brunei Bay(Sabah) Jne 18 1978 - Laid up
BRITISH RANGER (tank) (Br)
Arzanah Island Jly 27/28
Das Island Jly 28/29
Hovic Sep 12/15
Bullen Bay Sep 24/27
BRITISH RELIANCE (tank) (Br)
Khor Fakkan Oct 2/—
Mina al Fahal Oct 28/28
Fateh Terminal Oct 29/30
Dubai Pd Oct 31

trades, it is often possible to judge whether ships are loaded or in ballast, as well as determine approximate steaming distances, sea time and port time. Owing to uncertainty about the exact route followed, weather and other delays, and possible waiting time outside ports, it is not recommended to use the sea days as a means of establishing average speed. This should be estimated on a more consistent basis from the normal methods of service speed prediction based on long term expectations of power margins related to average weather, average roughness and fouling and average draft and trim - as discussed by other lecturers. It is also important to allow for any periods of steaming time at reduced speed, e.g. canal transits, river passages or extended manoeuvring time. On some trades, such periods may amount to a quarter of total sea time.

Minimisation of <u>time in port</u> is of course one of the keys to successful ship operation. Much effort has been expended to improve cargo handling methods, streamline port operations and apply operational research techniques to the problem. A distinction needs to be made between the needs of the

operator and port to predict and control turnround times on day to day operations, and the designer's needs to incorporate realistic cargo handling rates and waiting times, based on long term expectations of performance from a proposed design of ship and shore facility. It is well known that average handling rates and times in port vary significantly between the best and the worst ports (factors of 4 or 5 to 1 are not unusual, e.g. better container ports 700-1000 TEU per day, worse 180 - 250).

Fig. 4 shows some data for container cargo handling rates from Ref. 4. Fig. 5 and Table 3 show data for an efficient bulk grain discharging terminal where it can be seen that although overall discharge rates increase with ship size, there is a large standard deviation (around half the mean time).

TABLE 3

DISCHARGING TIMES FOR GRAINS AND OILSEEDS

Cargo Quantity, Tonnes	10,000-20,000	20,000-40,000	40,000-80,000
Ships in sample	39	40	39
Mean quantity, tonnes	15,400	26,400	58,600
Mean discharge rate t/h	280	460	750
Mean hours alongside	54	57	79
Standard deviation, h	34	33	33
Mean waiting time, h	26	61	43
Mean port time, h	80	118	122

It should be remembered that cargo is not worked all the hours a ship is in port, and that there will be breaks for nights, weekends, holidays, as well as at start and finish of operations, and for meal times, weather, equipment malfunction etc.

Table 4 gives and indication of breakdown of time in port; in general

TABLE 4

BREAK DOWN OF TIME IN PORT FOR TYPICAL GENERAL CARGO VESSELS

	Break-bulk	Unit Load
Handling cargo	42%	66%
Waiting for berth, start & finish work	14	17
Nights, holidays, meal-breaks, breakdowns etc.	38	9
Weather, berth shift, repairs etc.	6	8
Total port time	100	100

it is rare for break-bulk vessels to have cargo worked more than 40% of the time in port, or 70% for unit load ships, and 80% for bulk vessels. Analysis of actual ship data such as from Lloyds Voyage Record, or actual ship records,

shows that average time in port is often longer than that quoted by ship and port operators, who tend to emphasise the best rather than average figures. Care also needs to be taken in assessing statistics; for example some port authorities may not include time waiting for a berth as 'port time', yet to the shipowner is assuredly is, if 'port time' is considered realistically to be 'non-sea-time', i.e. non-productive time. (For those also concerned with port development, Ref. 5 is recommended).

Estimation of long run port times is therefore a judicious blend of past experience of similar trades (see Table 5), consideration of the proposed cargo handling equipment and its likely performance (on ship and on shore), and the anticipated general performance of the ports considered, given the expected cargo throughputs (allow for load factor), e.g. what shifts are worked, whether berth congestion is likely, and whether significant developments are expected in the next few years (long term plans are less important, since with discounted cash flow calculations, the possible benefits have a low present value).

Total port time needs of course to be estimated after including all the ports on a typical round trip - possibly only two on a bulk shuttle trade, but up to a dozen on a multi-port itinerary for some cargo liner types. Table 2 gives some average sea and port times, from which it can be seen that port time may constitute between one quarter and one third for vessels like large unit load ships, but as high as one half for break-bulk cargo vessels or single deck coasters. This proportion or its complement the number of days at sea a year, is also important in assessing the potential benefit of fuel saving devices, e.g. shaft driven alternators. Fig. 6 from Ref. 6 shows that the rate of return is significantly influenced; the ranking or order of merit of different fuel saving strategies may thus be completely altered. Even if it is not possible to predict annual sea days for every application, it is useful to know for example that Design A is more economic than Design B as long as sea days are above say 200 per annum. It is also desirable to couple such evaluations with alternative fuel prices (see later) and differentials between grades of fuel, since the crossover points of sea days and equal economy may well change.

Part loading of cargo ships is an important element in their economics. Not only is load factor and earning potential affected, but ship speed and fuel consumption as well. Part loading may occur for a number of reasons:

- lack of cargo availability
- port or canal draft limitations

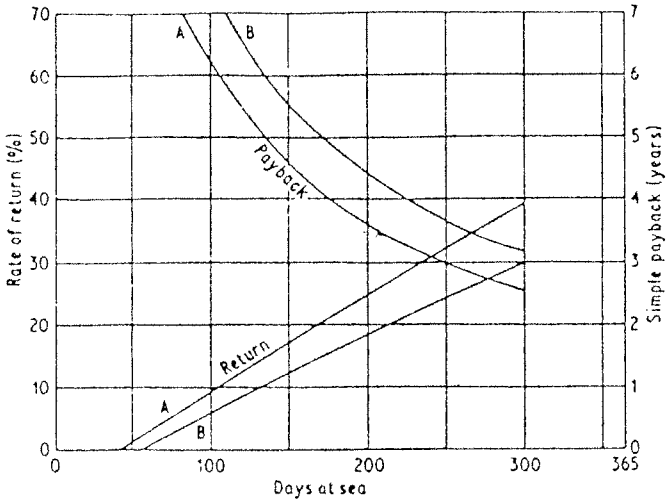
TABLE 5

TYPICAL CARGO HANDLING RATES

	Tonnes per Ship Working hour	Typical/Max. Tonnes per Day
Break-bulk general cargo	20-60	400/1000
Palletised cargo	80-100	900/2000
Steel, forest products etc.	100-400	1500/5000
Roll-on/roll-off	200-600	4000/10000
Containers	300-800	5000/12000
Barge carrier	500-1500	10000/25000
Dry bulk : discharge	1000-5000	20000/80000
load	2000-10000	30000/120000
Liquid bulk	1000-15000	100000/300000

Note that working hours per day are usually less than 24. Actual average rates of cargo handling equipment working are typically only about half the nominal designed rate, due to the effect of slower rates at start and finish of a hold, breakdown of equipments, interference between neighbouring equipment, delays on shore, etc.

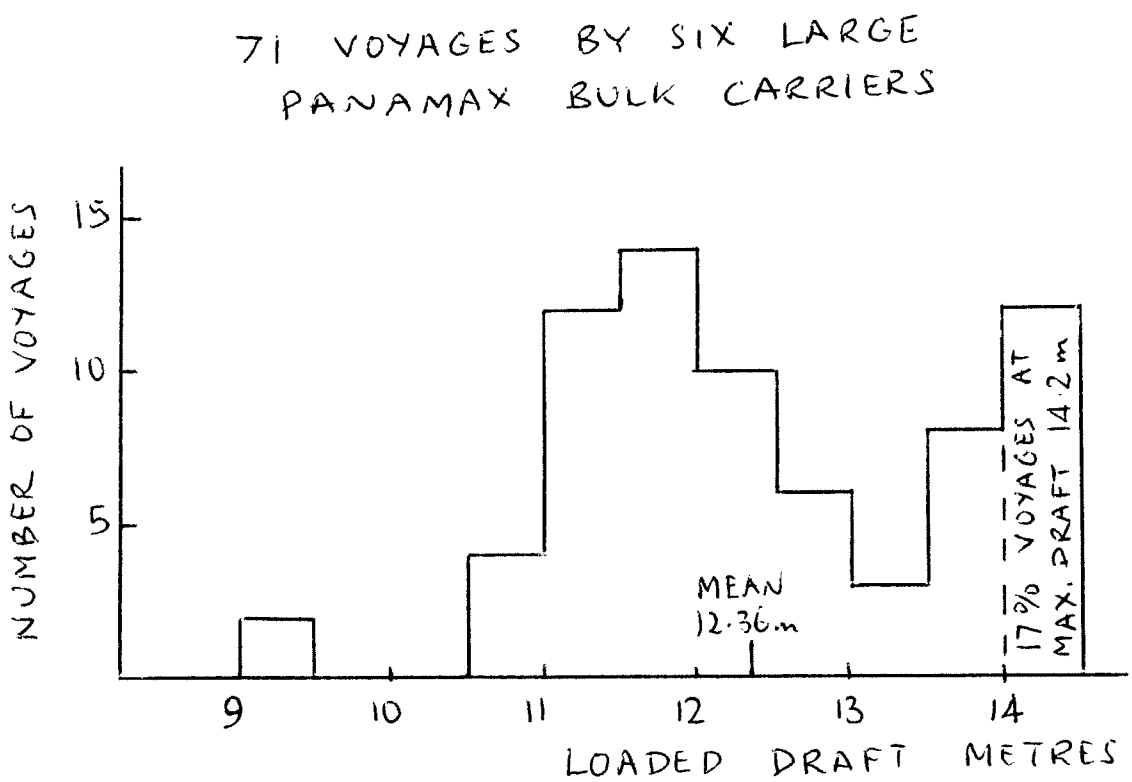
FIG 6



Curve A: Compared to diesel alternator on marine diesel oil at £170/tonne
Curve B: Compared to diesel alternator on blended fuel at £140/tonne

FIG 6 Rate of return and simple payback waste heat recovery system and hydraulically driven pumps

FIG. 7



- insufficient cubic capacity to load ships to maximum draft (cargoes with high stowage factors, m^3/tonne)
- early or late stages of voyage on multi-port itineraries.

Fig. 7 shows a histogram of actual loaded drafts of a number of sister bulk carriers, where average cargo was about 85% of the maximum; maximum draft was only used on 17% of voyages. Such figures emphasise the importance of designing hull form and propeller to be most efficient at the drafts most often used in practice: the concept of a 'design draft' less than a 'scantling draft'. For typical full forms, the power required at 80% of maximum draft is reduced by about 10-15% for the same speed. Similarly, for ships which are likely to spend a high proportion of their time in ballast, it is important that performance in the ballast condition - in smooth and rough water - is given full consideration, both from the choice of hull and propeller, and from the design of the ballast system.

Operational Factors Affecting Ship Costs

The main cost elements considered in evaluating the profitability of freight earning vessels are:

- capital charges, associated with the acquisition cost of the ship
- manning costs, including wages, benefits, victualling and other personnel costs
- maintenance, repair and stores
- fuel costs
- port and canal charges
- cargo handling costs
- miscellaneous costs, e.g. administration, commission.

For cargo liner vessels, there will be 'service costs' associated with cargo agents, provision of containers, feeder and positioning costs; while passenger vessels are a special case, but constitute less than 4% of marine vessels.

The actual techniques of making economic evaluations are well understood, (e.g. Ref. 7) while the designer is unable to influence many of the operator's costs e.g. port charges. My comments are therefore mainly concerned with the need to use realistic operational data to assist the designer estimate the costs of alternative designs.

Capital charges depend on the acquisition cost of the ship, financing terms, expected life and expected rate of return. The normal methods of cost estimating should give the differences between the alternative designs being studied. When converting initial costs into annual figures, e.g. via the capital recovery factor, it is desirable:

(a) to use an economic life which is likely to be shorter than the physical life, say 15 to 20 years, to allow for economic obsolescence

(b) to work consistently either in real terms (money of constant purchasing power, e.g. 1983 dollars) or in money terms, allowing for cost escalation. In the latter case, target rates of return will be higher than those in real terms, since approximately:

Money rate of return % = Real rate of return % + rate of inflation %

Real long term rates of return in shipping are rarely above 5 to 6%.

It is well known that operating costs vary widely between different shipowners, due to different policies on manning, maintenance, insurance etc. Some cost reductions are counter balanced by additions under other headings, e.g. minimum maintenance policies may incur more breakdowns and off-hire time. For general studies of alternative ship designs, it is usually sufficient to take typical values for operating costs, but ensure that any differences between designs are reflected in corresponding cost differentials, e.g. additional cargo handling gear is likely to incur higher maintenance costs. Of course studies for a particular operator can utilise data relating to that operator's experience. It should be noted that actual maintenance and repair costs are often higher than manufacturer's estimates - as any car owner will agree.

Cargo handling costs are a significant proportion of shore to shore costs, usually in the region of 20-40% of total costs. In many trades however, such costs are not paid by the ship operator, or are at a fixed rate over which the operator has little control. In general, detailed figures are only needed where the design alternatives are likely to incur different cargo handling costs.

Fuel costs. This whole course is addressing the problem of maximising fuel economy. Assuming that engineering studies have already evaluated the differences in daily fuel consumption, the two important remaining factors are:

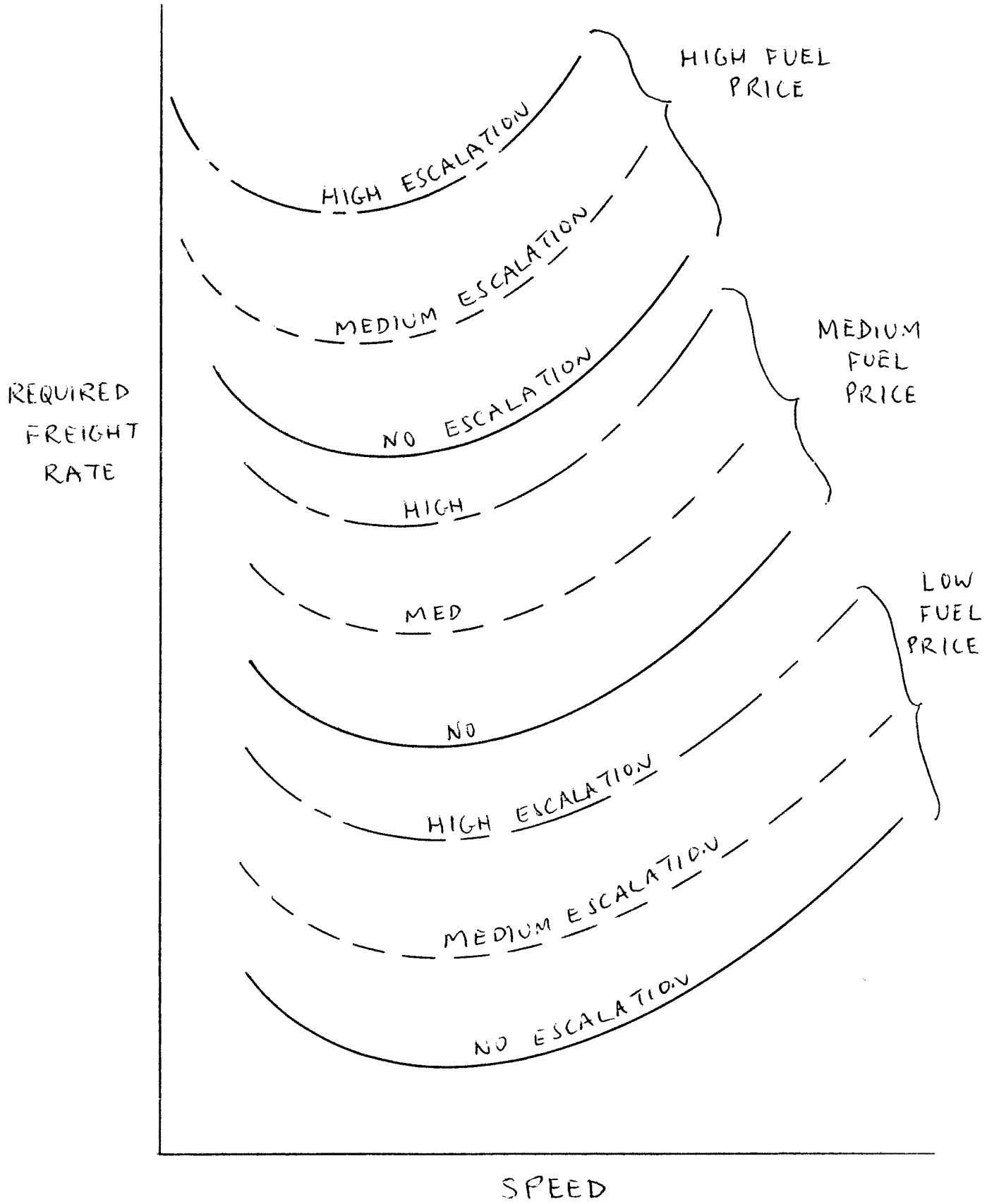
- proportions of sea and port time, since a difference in daily consumption of 10 or more to 1 is quite normal between sea and port.
- fuel price and escalation.

The former has already been discussed. The estimation of fuel price over the life of a ship is never easy. It is thus usually desirable to take a typical (local or worldwide as appropriate to the ship) price for each grade of fuel, based on an average of say a year, and then to take at least two other prices, usually one higher and one lower. Since evaluations are based on a notional life of ship, it is also necessary to estimate possible price escalation over the period, either in real or in money terms. The designer will probably select an escalation rate derived from various published studies, by oil companies and others. It is usually desirable to couple this with a higher and a lower rate, to produce a matrix of nine fuel price/escalation rate pairs, and a corresponding sensitivity study, whose results may be presented as Fig. 8. The sensitivity of the results to the assumptions is then made clear, which usually helps in making the final judgement of which design to select. This becomes particularly important where there are large differences between fuel consumptions and fuel prices for each alternative, e.g.coal fired versus diesel ships. Where fuel differences are small e.g. between slow speed and medium speed diesel ships, the influence of fuel price is less important and only a few combinations need be examined.

Conclusion

In most cases, it is sufficient to use realistic data reasonably typical of actual operations, rather than to seek excessive precision. The examples presented show that operational data exhibits large variances. It is thus usually more important to ensure that the correct orders of magnitude are being used, e.g. on port time, and that differences between alternatives are consistently evaluated. In some cases it may be necessary to take a range of values (high/medium/low), at least when identifying the most promising design alternatives. If projects proceed to a more detailed study, then is the time to consider more sophisticated techniques, including simulation. However, the data and effort required for such more extensive studies should not be underestimated. The 'steady state' (deterministic) models can provide the designer with useful results at a fraction of the cost of developing stochastic models. Finally, all designers should have a good appreciation of the actual problems of ship operation, and take every opportunity to observe ships and ports in every day service.

FIG. 8


HIGH FUEL PRICE
HIGH ESCALATION
MEDIUM ESCALATION
NO ESCALATION
MEDIUM FUEL PRICE
HIGH
MED
NO
LOW FUEL PRICE
HIGH ESCALATION
MEDIUM ESCALATION
NO ESCALATION
REQUIRED FREIGHT RATE
SPEED

References

1. Buxton, I.L. Matching Merchant Ship Designs to Markets. Trans. N.E.C.I.E.S. 1981-82.

2. Langenberg, H. A Diagram Showing The Main Operational Data of a Containership. Special Ships, July 1979.

3. Lloyd's Voyage Record. Published weekly by Lloyds of London Press.

4. Edmond, E.D. and Maggs, R.P. Container Ship Turnaround Times at UK Ports. Maritime Policy and Management, No. 1, Vol. 4, 1976.

5. Port Development. A Handbook for Planners in Developing Countries. United Nations Conference on Trade & Development. UN Publication Sales No. E.77. II.D.8

6. Meek, M. and Craig, M.F. Design Considerations in the Development of a Fuel-economic Ships. Trans. I.Mar.E., Paper C101, Vol. 94, 1982.

7. Advanced Ship Design Techniques. Economic Design Course, Module E.1.
West European Graduate Education in Marine Technology, Newcastle University, September 1978.

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FUEL QUALITY CONTROL AND TREATMENT ON BOARD

by

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NORDSJTERNAN AB, STOCKHOLM

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INTRODUCTION

ACCORDING TO STATISTICS ABOUT 5-6% OF ALL PETROLEUM PRODUCTS ARE USED AS BUNKER FUEL OIL FOR SHIPS' PROPULSION.

LANDBASED INDUSTRIES INCLUDING POWER STATIONS ARE FACING A NUMBER OF RESTRICTIONS WITH REGARD TO THE EXHAUST GAS EMISSIONS. SULPHUR ACID CONTENTS AS WELL AS SMOKE PARTICLES ARE IN THE FOCUS IN MOST COUNTRIES.

AT SEA, THIS TYPE OF LIMITATIONS DOES NOT EXIST ALTHOUGH THERE HAVE BEEN DISCUSSIONS REGARDING SHIPS' EXHAUST GAS EMISSIONS IN SOME PORTS.

SINCE THE FIRST OIL CRISIS IN 1973 THE SHIPPING INDUSTRY HAS FACED A CONSTANTLY RAISING PRICE LEVEL FOR BUNKER FUEL OILS. ALTHOUGH IT HAS FLATTENED OUT THE LAST YEAR, WE CAN EXPECT FURTHER INCREASES IN PRICE DUE TO

- INCREASED PRODUCTION COST FOR CRUDE OIL
- GENERAL INFLATION
- SHORTAGE OF HEAVY FUEL OIL DUE TO CHANGES IN REFINERY TECHNIQUE ON ACCOUNT OF INCREASED DEMAND FOR GASOLINE AND GAS OILS.

FOR CERTAIN SHIPS THE FUEL BILL CAN BE AS HIGH AS ABT. 40% OF THE TOTAL OPERATIONAL COST. THIS FACT HAS FURTHER INCREASED THE PRESSURE ON THE ENGINE BUILDERS TO DEVELOP ENGINES WITH HIGHER EFFICIENCY AND BETTER POSSIBILITIES TO BURN FURTHER DETERIORATED FUEL QUALITIES. SIMULTANEOUSLY WITH THE DEVELOPMENT OF THE DIESEL ENGINES, THE SHIPS' FUEL HANDLING PLANTS HAVE UNDERGONE A SIMILAR DEVELOPMENT WITH REGARD TO THE DETERIORATED FUEL OILS.

QUALITY OF TO-DAY'S BUNKER FUEL OILS

THE QUALITY OF A BUNKER FUEL OIL IS DEPENDING ON

- ORIGIN OF CRUDE OIL
- REFINERY TECHNIQUE
- BLENDING WITH OTHER PETROLEUM PRODUCTS IN ORDER TO REACH A DESIRED VISCOSITY.

THE MIDDLE EAST CRUDES GENERALLY CONTAIN MORE SULPHUR BUT LESS VANADIUM. DUE TO THEIR PARAFFINIC STRUCTURE THEY HAVE A HIGHER POUR POINT. SOME NORTH AFRICAN AND NORTH SEA CRUDES HAVE LOW SULPHUR CONTENTS AND ARE THEREFORE MORE IN DEMAND FOR INDUSTRIAL USE WITH REGARD TO LIMITATIONS IN POLLUTION OF SULPHUR OXIDES.

CRUDES FROM THE VENEZUELAN AREA HAVE IN GENERAL MODERATE SULPHUR BUT HIGH VANADIUM CONTENTS. DUE TO THE GENERAL NAPHTENIC STRUCTURE THE POUR POINT IS LOW. THE ASPHALTENES ARE NORMALLY HIGH WHICH MAKES THEM VERY INTERESTING FOR THE REFINERIES INTENDED FOR PRODUCTION OF BITUMEN AND BASE LUBRICATION OILS. FOR SHIPS HAVING NONE OR POOR HEATING FACILITIES IN THE BUNKER TANKS HEAVY FUEL OILS ORIGINATING FROM VENEZUELAN CRUDES ARE VERY INTERESTING, PARTICULARLY WHEN OPERATING IN COLD WATERS DUE TO THE LOW POUR POINTS.

IT IS TO-DAY A WELLKNOWN FACT THAT THE DEMAND FOR GASOLINE AND GAS OILS HAS INCREASED DURING THE LAST TEN YEARS. IN ORDER TO GET A BETTER ECONOMY IN THIS PRODUCTION THE REFINERIES HAVE BEEN REBUILT OR NEW REFINERIES HAVE BEEN CONSTRUCTED, HAVING NOT ONLY ATMOSPHERIC AND VACUUM DISTILLING BUT ALSO DIFFERENT STAGES OF CRACKING THE RESIDUAL OILS.

THE REFINING TECHNIQUES ARE SCHEMATICALLY SHOWN IN FIG. 1. THE MAXIMUM GASOLINE YIELD IS OBTAINED BY THE USE OF VACUUM DISTILLATION OF THE ATMOSHPERIC RESIDUE AND PASSING THE HEAVY DISTILLATE FROM THIS DISTILLATION THROUGH THE FLUID CATALYTIC CRACKING UNITE (F.C.C.). THIS PROCESS DERIVES ITS NAME FROM THE CATALYST WHICH IS IN SUCH A FINE POWDER FORM THAT IT APPEARS TO BEHAVE LIKE A FLUID.

THE CRACKING PROCESS TAKES MORE OF THE LIGHT PARTS, THAT IS TO SAY, MORE OF THE HYDROGEN CONTENTS FROM THE FUEL LEAVING A HIGHER PROPORTION OF CARBON WHICH ADVERSELY AFFECTS THE IGNITION AND COMBUSTION QUALITIES OF THE FUEL.

VIS-BREAKING ACTS ON THE HEAVY RESIDUAL OIL FROM THE VACUUM DISTILLATION TOWER. BY A LOW DEGREE OF THERMO-CRACKING THE VIS-BREAKER REDUCES THE VISCOSITY OF THE VACUUM RESIDUE AND THUS SAVES THE NEED FOR ADDING GASOIL TO REDUCE THE VISCOSITY. THUS UP TO 10% OF GASOIL IS SAVED AND CAN BE USED FOR OTHER PURPOSES, WHEN A VIS-BREAKER IS FITTED. ADDITIONALLY, THERE IS A DANGER OF A SMALL AMOUNT OF CATALYST FINES FINDING THEIR WAY VIA THE FCC SLURRY OIL INTO THE MARINE HEAVY FUEL OIL. THE CATALYST CONSISTS OF ALUMINIUM-SILICONE COMPOUNDS IN FINE ABRASIVE PARTICLE FORM WHICH MAY CAUSE HIGH WEAR IN DIESEL ENGINE PARTS, IF NOT REMOVED BY PURIFICATION.

VIS-BREAKING AND CRACKING PROCESSES WILL HAVE THE FOLLOWING EFFECT ON THE FUEL OIL

- REDUCTION OF THE AMOUNT OF RESIDUAL FUEL
- INCREASE OF DENSITY
- INCREASE IN CONRADSON CARBON VALUE
- POSSIBLE INCREASE IN COMPATIBILITY SEDIMENT
- POSSIBILITY OF CONTAMINATION BY CATALYST FINES.

THE TABLE IN FIG. 2 GIVES THE CHARACTERISTICS OF RESIDUALS FROM DIFFERENT REFINERY PROCESSES.

THE TABLE IN FIG. 3 GIVES SOME INFORMATION REGARDING THE PROPERTIES OF BUNKER FUEL OILS.

QUALITY CONTROL OF BUNKER FUEL OILS

IN THE GOOD OLD DAYS MOST OF THE BUNKER FUEL OIL WAS SUPPLIED THROUGH ONE OR ANOTHER OF THE BIG MAJOR OIL COMPANIES SUCH AS ESSO, SHELL, GULF ETC. THIS PATTERN HAS TO-DAY CHANGED COMPLETELY. A VERY LARGE PORTION OF THE BUNKER FUELS ARE SUPPLIED BY SMALL INDEPENDENT BUNKER SUPPLIERS.

THE SUBSTANTIAL INCREASE IN BUNKER FUEL OIL PRICES DURING THE 1970'S MEANT THAT CONSIDERABLY MORE WORK HAD TO BE SPENT ON THE DIESEL ENGINES IF THEY SHOULD CONTINUE TO BE THE MAIN PRIME MOVER FOR SHIPS.

OWNERS, IN THEIR DRIVE FOR LOWER RUNNING COSTS, HAVE BEEN FORCED TOWARDS HIGHER VISCOSITY RESIDUAL FUELS AND THIS CONTRIBUTED TO SOME OF THE PROBLEMS EXPERIENCED. FOR MANY YEARS IT HAD BEEN POSSIBLE, WITHOUT ANY DIFFICULTY, TO USE ALMOST ALL THE LOW VISCOSITY RESIDUAL FUELS, WHETHER THE FUEL HANDLING EQUIPMENT ON BOARD WAS OPERATED IN CORRECT ORDER OR NOT. THIS STATE OF THINGS STARTED TO CHANGE IN THE SECOND HALF OF THE 70'S. MORE AND MORE PEOPLE BEGAN TO REALIZE THAT VISCOSITY ALONE WAS NOT SUFFICIENT TO DEFINE THE QUALITY OF A RESIDUAL FUEL QUALITY.

FACING THIS SITUATION THE CIMAC WORKING GROUP ON FUTURE FUELS WAS COMMISSIONED IN 1979 TO DRAW UP NEW FUEL STANDARDS IN COOPERATION WITH THE BRITISH STANDARDS INSTITUTION (BSI) AND THE INTERNATIONAL CHAMBER OF SHIPPING (ICS). DISCUSSIONS BETWEEN BSI, ICS AND SOME OIL COMPANIES REVEALED THAT THEY COULD NOT ACCEPT ENGINE BUILDERS' OPINIONS ON INTERMEDIATE FUEL OIL GRADE SPECIFICATIONS. COMPROMISE PROPOSALS WERE MADE WHICH FINALLY RESULTED IN FIXING CONTAMINANT LEVELS OF VANADIUM, CARBON AND ALUMINIUM FOR VISCOSITY RELATED GRADES. ENGINE OPERATORS BELIEVE THAT THESE DATA ARE THE ABSOLUTE MINIMUM TO ALLOW OLDER ENGINES TO RUN ON RELATIVELY CHEAP INTERMEDIATE FUELS. THE ENGINE BUILDERS' (CIMAC) MINIMUM TARGET SETTING IS TO SEE THAT, WHEN GENERAL FUEL QUALITY DETERIORATES TO THE EXTENT CAUSING ENGINE OPERATION DIFFICULTIES, THERE IS AT LEAST ONE

INTERMEDIATE FUEL SUITABLE FOR OLDER ENGINES. UNFORTUNATELY, THE CIMAC REQUIREMENTS FOR SPECIFICATIONS OF INTERMEDIATE MARINE FUELS FIG 4 WERE NOT INCLUDED IN THE NEW BSI STANDARD. HOWEVER, THE NEW ISO WORKING GROUP FOR FUTURE FUELS (WG-6, TC 28/SC 4) HAS INCORPORATED CIMAC:S MINIMUM REQUIREMENT INTO THE NEW ISO STANDARDS WHICH WILL REPLACE THE BSI SPECIFICATION IN ABOUT TWO YEARS AND THEREAFTER BECOME THE INTERNATIONAL STANDARD.

THE ICS WORKING GROUP HAS CONCENTRATED IN PROMOTING A MORE COMPREHENSIVE BUNKER DELIVERY NOTE CONTAINING SEVERAL IMPORTANT FUEL QUALITY PARAMETERS, FIG. 5. THIS NEW BUNKER DELIVERY NOTE WILL NO DOUBT INCREASE THE KNOWLEDGE OF FUEL OIL QUALITY BEING DELIVERED ONBOARD. PRACTICAL EXPERIENCE HAS SHOWN THAT THE SUPPLIERS ARE VERY NEGATIVE TO THIS BUNKER DELIVERY NOTE.

IN SPITE OF THE BSI STANDARD AND ICS BUNKER DELIVERY NOTE THERE WAS STILL NEED FOR INFORMATION REGARDING THE ACTUAL QUALITY OF FUEL SUPPLIED TO THE ACTUAL SHIP. IN THIS SITUATION DET NORSKE VERITAS IN 1980 INTRODUCED ITS "FUEL QUALITY TESTING PROGRAMME". THE MAIN FEATURES OF THIS PROGRAMME ARE

- A KNOWLEDGE OF THE QUALITY OF THE FUEL SUPPLIED AT EACH BUNKERING
- OPPORTUNITIES FOR TAKING APPROPRIATE ACTION TO ENSURE THAT THE FUEL IS HANDLED CORRECTLY
- AN INDEPENDANT CHECK ON THE ACCURACY OF QUALITY INFORMATION PROVIDED BY THE SUPPLIER, EITHER IN FORM OF A SPECIFICATION OR BY ACTUAL QUALITY LEVELS.

AT THE LABORATORY IN OSLO THE SAMPLES ARE ANALYSED TO DETERMINE

- DENSITY AT 15°C
- KINEMATIC VISCOSITY AT 50°C
- POUR POINT °C
- CONRADSON CARBON RESIDUE %
- SULPHUR %
- WATER %
- ASH %
- METAL CONTENTS P P M (VANADIUM, SODIUM, ALUMINIUM, SILICON)
- COMPATIBILITY WITH DISTILLATE FUEL
- CALORIFIC VALUE (CALCULATED)

THE OWNERS OR THE OPERATORS ARE ADVISED BY TELEX THE RESULT OF THE ANALYSIS, IN GENERAL WITHIN 3 DAYS. THE PROGRAMME WILL BE OF GREAT VALUE FOR ALL PARTICIPANTS WITH REGARD TO THE POSSIBILITY TO ACCUMULATE STATISTICS IN COMPUTERS FOR FUTURE USE.

ALTHOUGH DNV WAS FIRST TO START UP THIS SERVICE FOR THE OWNERS AND OPERATORS IT MAY ALSO BE MENTIONED THAT LLOYDS REGISTER OF SHIPPING ALSO IS PROVIDING A SIMILAR SERVICE.

QUALITY CONTROL ON BOARD

ALTHOUGH THE PARTICIPANTS IN DNV TESTING PROGRAMME WILL RECEIVE AN ANALYSIS IN DUE COURSE THIS MAY BE TOO LATE FOR I.E. THE FOLLOWING REASONS:

- THE BUNKERED FUEL OIL DOES NOT AT ALL COMPLY WITH THE SPECIFICATION
- WATER CONTENTS ARE TOO HIGH
- TOO HIGH LEVEL OF ABRASIVE PARTICLES
- THE BLENDED FUEL MAY BE INCOMPATIBLE.

FOR ECONOMICAL OR TECHNICAL REASONS IT MAY BE NECESSARY TO MAKE A PROTEST TO THE QUALITY OF THE BUNKERED FUEL DURING THE ACTUAL BUNKERING.

IN ORDER TO MAKE A FIRST ANALYSIS MANY OWNERS AND OPERATORS ARE PROVIDING ADEQUATE LABORATORY FACILITIES ONBOARD THE SHIPS. NORMALLY SUCH A LABORATORY WILL MAKE IT POSSIBLE TO CARRY OUT THE FOLLOWING ANALYSES:

- DENSITY
- VISCOSITY
- WATER - FRESH OR SALT
- COMPATIBILITY
- ABRASIVE PARTICLES

IN ADDITION TO THE TESTS FOR FUEL OIL THERE MAY ALSO BE INCLUDED THE FOLLOWING ADDITIONAL TESTS FOR THE LUB OIL:

- COMPARATIVE VISCOSITY TEST
- PRESENCE OF STRONG ACID

SUCH A TESTING PROGRAMME CARRIED OUT ONBOARD MAY NOT BE AS ACCURAT AS IN A PROFESSIONAL LABORATORY ASHORE. NEVERTHELESS, IT HAS PROVED TO BE OF GREAT VALUE TO PREVENT ANY GREATER BREAKDOWNS BY MEANS OF THESE FIRST ANALYSES.

CONTROL OF BUNKERED QUANTITY

CAREFUL CONTROL OF BUNKERING PROCEDURES ONBOARD DIFFERENT SHIPS HAS REVEALED THAT IT IS NECESSARY TO APPLY A VERY STRICT CONTROL THAT THE QUANTITY ORDERED ALSO REALLY WILL BE RECEIVED ONBOARD THE VESSEL.

NORMALLY A BUNKER CONTRACT IS STIPULATING THAT MEASUREMENT OF THE VOLUME IS TO TAKE PLACE EITHER AT THE SELLERS' TANK INSTALLATION OR ONBOARD THE BUNKER BARGE IN THE SELLERS' OPTION. ASSUME THAT A SHIP IS BUNKERING 4,000 TONS BUNKER FUEL OIL SIX TIMES A YEAR AND THERE IS 1% MISSING EACH TIME. WITH A PRICE OF USD 170.-/TON THIS WILL BE A LOSS OF USD 40.800.- PER YEAR OR ABOUT SEK 325.000.- PER YEAR.

EARLIER, MOST OF THE OIL COMPANIES HAD THEIR OWN BUNKERING BARGES BUT TO-DAY PRACTICALLY ALL THE BUNKER BARGES ARE PRIVATELY OWNED.

IN ORDER TO HAVE A GOOD CONTROL OF THE BUNKERING A COMPUTERIZED BUNKERING CONTROL SYSTEM IS UNDER TEST ONBOARD A SWEDISH VESSEL. AFTER SOME TEETHING PROBLEMS THE SYSTEM NOW SEEMS TO WORK SATISFACTORILY. THE MAIN PARAMETERS THAT ARE CONTINUOUSLY MEASURED DURING THE BUNKERING ARE

- VOLUME
- TEMPERATURE
- DENSITY
- WATER CONTENTS

MEASUREMENT OF VOLUME AND TEMPERATURE ARE OBVIOUS, BUT DENSITY IS NECESSARY TO KNOW IN ORDER TO AVOID THAT AIR-OIL EMULSIONS WILL BE CALCULATED AS SOLID OIL IN THE VOLUME METER. A CONTINUOUS MEASUREMENT OF WATER IS ALSO NEEDED IN ORDER TO FIND THE ACTUAL BUNKERED QUANTITY OF FUEL OIL.

HANDLING OF FUEL OILS ONBOARD

THE PIONEERING WORK REGARDING THE ARRANGEMENTS ONBOARD WHEN OPERATING THE MAIN DIESEL ENGINE ON HEAVY FUEL OILS WAS DONE BY JOHN LAMB IN THE SECOND HALF OF THE 1940'S. MOST OF THE PRINCIPLES REGARDING FUEL OIL HANDLING IN STORAGE TANKS ETC. ARE STILL IN USE. THE TWO STAGE CENTRIFUGAL PURIFICATION OF THE FUEL OIL WAS ALSO APPLIED.

STORAGE AND TANK ARRANGEMENT

THE TANK ARRANGEMENT FOR STORAGE OF FUEL OILS ON A TANKER OR BULK CARRIER IS GENERALLY A COMBINATION OF DEEP TANKS FORWARD OF THE CARGO COMPARTMENT AND CROSS BUNKER TANKS AFTER THE CARGO COMPARTMENT BUT FORWARD OF THE FORWARD ENGINE ROOM BULKHEAD. THOSE TANKS ARE NORMALLY VERY HIGH COMPARED TO THE BOTTOM AREA. THIS MEANS THAT SETTLING OF IMPURITIES TAKES VERY LONG TIME. A MORE FAVOURABLE TANK SHAPE FOR NATURAL SETTLING PROVIDES THE DOUBLE BOTTOM BUNKER TANKS IN A DRY CARGO VESSEL SUCH AS A RO-RO- OR REEFER VESSEL. NORMALLY THE BUNKER TANKS ARE FITTED WITH HEATING COILS FOR STEAM HAVING AN AMPLE CAPACITY TO KEEP THE FUEL PUMPABLE IN ALL KINDS OF CLIMATES AND WATERS. THE UPPER LIMIT FOR HEATING IS SETTLED BY THE FLASH POINT FOR THE BUNKER FUEL OIL. NORMALLY THE UPPER LIMIT IS 60°C.

UP TO NOW THE COILS HAVE NORMALLY BEEN DESIGNED FOR STEAM AS HEATING MEDIUM. WITH THE INCREASING INTEREST FOR HEAT RECOVERY AND THE NECESSITY TO UTILIZE AS MUCH AS POSSIBLE OF THE STEAM IN THE

TURBO GENERATING SET AN INCREASED INTEREST IN USING JACKET COOLING WATER AS HEATING MEDIUM FOR THE BUNKER OIL HAS BEEN NOTICED. EACH PROJECT HAS TO BE EVALUATED FROM ECONOMICAL POINTS OF VIEW AS THE HEATING COILS HAVE TO BE ENLARGED CONSIDERABLY WHEN USING JACKET WATER AS HEATING MEDIUM. CIRCULATING PUMPS HAVE ALSO TO BE INCLUDED IN THE COMPARISON.

BLENDED FUEL OIL PROPERTIES

BUNKER FUEL OILS WHICH BASICALLY ARE HEAVILY CRACKED ARE CONTAINING A HIGH DEGREE OF DISPERSED ASPHALTENES. DEPENDING UPON THE BLENDING AND THE ORIGIN OF THE BLENDED COMPONENTS THE MIXTURE CAN BE CONSIDERED AS STABLE. THIS MEANS THAT NO FLOCCULATION OR AGGLOMERATION WILL TAKE PLACE WITHIN REASONABLE LIMITS.

INCOMPATIBLE OR UNSTABLE ARE OILS IN WHICH ASPHALTENES PRECIPITATE (FORMING SLUDGE), EITHER IN ONE OF THE COMPONENTS OR AFTER THE BLENDING OF TWO OR MORE COMPONENTS.

AS MENTIONED BEFORE IT IS EXTREMELY IMPORTANT TO BE OBSERVANT TO THE PROBLEM OF INCOMPATIBILITY ALREADY WHEN BUNKERING BY MEANS OF ANALYSING THE BUNKER OIL.

THE KNOWLEDGE OF THE PROBLEMS REGARDING UNSTABLE BLENDS IS STILL NOT SUFFICIENT ENOUGH TO GIVE A DEFINITE ADVISE TO CURE THE PROBLEM

FUEL OIL CLEANING ARRANGEMENTS

IN A NUMBER OF INVESTIGATIONS REGARDING CYLINDER LINER WEAR THERE IS STATED THE SUPERIORITY OF THE PURIFICATION OF FUEL OILS BY MEANS OF CENTRIFUGAL SEPARATORS. ALTERNATIVES OF DIFFERENT KINDS HAVE BEEN TESTED WITHOUT ANY SUCCESS. THE MEASURED WEAR OF LINERS AND PISTON RINGS HAVE PROVED THE QUALITY OF THE CENTRIFUGAL PURIFICATION OF HEAVY FUEL OILS. AS MENTIONED EARLIER THIS CLEANING PROCESS WAS PART OF THE SUCCESS OF JOHN LAMB'S EARLIER TESTS.

A NORMAL LAY-OUT OF A HEAVY FUEL OIL TREATMENT PLANT ARRANGEMENT IS SHOWN IN FIG. 6.

THE SETTLING TANK IS NORMALLY FILLED UP ONCE PER 24 HOURS BY A TRANSFER-PUMP. THIS PUMP IS NORMALLY OF SCREW TYPE AND OPERATES WITH A SPEED OF 1450 OR 1750 R P M. THE BELLMOUTH OF THE SUCTION PIPE IS GENERALLY VERY CLOSE TO THE BOTTOM OF THE BUNKER TANK. ANYTHING THAT MAY HAVE BEEN SETTLED TO THE BOTTOM DURING THE STORAGE OF THE HEAVY FUEL OIL IN THE BUNKER TANK WILL BE DRAWN TO THE TRANSFER-PUMP AND VERY VIGOROUSLY MIXED INTO THE FUEL AGAIN. IF SOMETHING OF THE SEDIMENT WOULD HAVE A CHANCE TO SETTLE AGAIN IN THE SETTLING TANK, THE TANK HAS TO HAVE A VERY LARGE AREA AND A VERY SMALL DEPTH. UNFORTUNATELY, IT IS NOT POSSIBLE TO ACCOMMODATE SUCH A NICE SETTLING TANK ONBOARD A MODERN VESSEL. THE TANK WILL NORMALLY BE HIGH WITH A MODERATE BOTTOM SURFACE. AS THE TEMPERATURE IN THE SETTLING TANK NORMALLY MUST NOT EXCEED 60°C THE SETTLING SPEED WILL BE EXTREMELY LOW WHEN OPERATING ON HIGH VISCOSITY FUEL OILS. THE SETTLING SPEED OF A PARTICLE CAN BE CALCULATED ACCORDING TO THE WELL KNOWN STOKES FORMULA:

$$V = \frac{(R_P - R_O)\, D^2}{18 \cdot \eta} \cdot G.$$

V = SETTLING SPEED
R_P = DENSITY OF THE PARTICLE
R_O = DENSITY OF THE OIL
D = PARTICLE DIAMETER
η = DYNAMIC VISCOSITY OF THE OIL
G = GRAVITY ACCELERATION

ACCORDING TO TO-DAY'S EXPERIENCE MOST OF THE FINE DISPERSED SLUDGE WILL BE RETAINED IN THE FUEL OIL IN A COLLODIAL FORM. THIS IS OF GREAT IMPORTANCE WHEN CONSIDERING DIFFERENT POSSIBILITIES TO CLEAN THE OIL FROM CATALYST FINES.

FOR THE FUTURE IT MUST BE CONSIDERED TO USE THE SAME TECHNIQUE AS FOR CLEANING OILY WATER. AS THE SEPARATION OF OIL FROM BILGE OR BALLAST WATER IS A MATTER OF AVOIDING THE FORMATION OF EMULSION WHEN PUMPING THE OILY WATER, IT SEEMS LOGICAL TO BUILD IN SLOW RUNNING PUMPS TO FEED THE PURIFIERS DIRECTLY FROM THE BUNKER TANKS INSTEAD OF TO-DAY'S SYSTEM WITH HIGH SPEED TRANSFER PUMPS.

SOME ASPECTS ON USING CENTRIFUGAL SEPARATORS FOR CLEANING OF FUEL OILS

WHEN STOKES FORMULA IS APPLIED TO A CENTRIFUGAL SEPARATOR WITH CONICAL DISCS THE FOLLOWING FORMULA WILL BE THE RESULT:

$$Q = \left(\frac{(R_P - R_0) \cdot D^2}{18\,\eta} \right) \cdot \frac{2}{3}\,\pi\, N \cot\alpha (R_2^3 - R_1^3)\,\omega^2 \qquad (1)$$

Q = SEPARATOR CAPACITY
ω = ANGULAR SPEED OF THE SEPARATOR BOWL
R_1, R_2 = INNER AND OUTER RADIUS OF THE DISCS
N = NUMBER OF THE PASSAGES BETWEEN THE DISCS
α = HALF CONE ANGLE

THIS FORMULA HAS BEEN PUBLISHED BY A NUMBER OF AUTHORS AND IS BASICALLY USED BY ALL MANUFACTURERS OF CENTRIFUGAL SEPARATORS TO DESCRIBE THE CAPACITY OF THEIR MACHINES.

FROM THE FORMULA (1) IT IS POSSIBLE TO STATE THAT THE CAPACITY OF A SEPARATOR IS INVERSLY PROPORTIONAL TO THE DYNAMIC VISCOSITY WHEN IT IS DESIRED TO OBTAIN IDENTICAL SEPARATION (D = CONSTANT) FOR OILS OF DIFFERENT VISCOSITY.

ANOTHER CONCLUSION THAT CAN BE DRAWN FROM EQUATION (1) IS, THAT THE THROUGHPUT IS DIRECTLY PROPORTIONAL TO THE EFFICIENCY OF THE SEPARATOR.

IF THE SAME FUEL OIL IS SEPARATED IN TWO DIFFERENT SEPARATORS WITH THE EFFICIENCY E_1 AND E_2 THE SAME FINAL RESULT WILL BE OBTAINED WHEN

$$\frac{Q_1}{Q_2} = \frac{E_1}{E_2} \qquad (2)$$

WHERE

$$E = \frac{2}{3} \cdot N \cdot C \, TG \alpha \, (R_2^3 - R_1^3) \omega^2 \qquad (3)$$

WITH THE SAME FINAL RESULT IT MEANS THAT THE SAME PARTICLE WITH A DIAMETER OF D WILL BE SEPARATED FROM THE FUEL OIL.

IF THE THROUGHPUT WILL BE KEPT CONSTANT IN BOTH CASES IT WOULD BE POSSIBLE TO EXPRESS THE FINAL RESULT AS

$$\left(\frac{D_1}{D_2}\right)^2 = \frac{E_2}{E_1} \qquad (4).$$

PRACTICAL EXPERIENCE AT ALFA LAVAL HAS LED TO A MODIFICATION OF THE FORMULA (1) IN SUCH A WAY THAT THE EXPONENTS OF R AND ω ARE MODIFIED FROM 3 TO 2,75 FOR R AND FROM 2 TO 1,5 FOR ω .

WESTFALIA IS USING THE UNCHANGED FORMULA (1) WITH A GENERAL MODIFICATION FACTOR CALLED"LEISTUNGSFAKTOR".

THE CENTRIFUGAL SEPARATOR MAY BE ARRANGED AS A PURIFIER OR AS A CLARIFIER.

THE PURIFIER MAY SEPARATE WATER FROM OIL CONTINUOUSLY AND AT THE SAME TIME REMOVE SOLID PARTICLES (FIG. 7).

THE CLARIFIER MAY ONLY REMOVE SOLID PARTICLES CONTINUOUSLY (FIG. 8)

WITH THE DIFFERENT CRACKING PROCESSES IN THE REFINERIES THE HEAVY FUEL OIL HAS GOT A HIGHER DENSITY. UP TO NOW THE LIMIT FOR A PURIFIER HAS BEEN SET AT 991 KG/M^3 AT 15°C.

SUCH A LIMIT DOES NOT EXIST FOR THE CLARIFIER AS IF THERE IS ANY WATER PRESENT IN THE OIL IT WILL BE SEPARATED AND RETAINED IN THE CENTRIFUGE TOGETHER WITH THE SOLIDS.

IN ORDER TO OVERCOME THE PROBLEM WITH THE MAXIMUM DENSITY OF 991 KG/M^3 FOR THE OIL, ALFA LAVAL HAS INCORPORATED A WATER DETECTOR IN THE CLEANED OIL OUTLET AFTER A CLARIFIER. WHEN DETECTING RAISING WATER CONTENTS IN THE CLEANED OIL IT IS ASSUMED THAT THE SEPARATOR BOWL IS FILLED WITH WATER AND HAS TO BE EMPTIED WITH THE NORMAL SELF-CLEANING MECHANISM.

WITH THIS NEW ARRANGEMENT CALLED ALCAP IT IS CLAIMED THAT HEAVY FUEL OILS UP TO A DENSITY OF 1010 KG/M^3 MAY BE CLEANED. SEE ENCLOSED PAMPHLET.

SEPARATION OF ABRASIVE SOLIDS

AS STATED EARLIER IN THE INTRODUCTION OF THE "FLUID CATALYTIC CRACKER" THERE IS A GREAT RISK THAT A SMALL PORTION OF THE CATALYST FINES WILL BE RETAINED IN THE HEAVY FUEL OIL. THE CATALYSTS ARE A CRYSTALLINE ALUMINIUM-SILICATE PARTICLE WHICH IS HIGHLY POROUS WITH A VERY LARGE SURFACE. THIS PROPERTY IS NECESSARY IN ORDER TO GIVE A GOOD FUNCTION IN THE CRACKING PROCESS. THE SIZES ARE GENERALLY BELOW 50 MICRON. WITH THOSE PROPERTIES, IT IS NORMALLY VERY DIFFICULT TO SEPARATE THE CATALYST FINES UNLESS A VERY LOW THROUGHPUT IS USED IN THE SEPARATOR. THIS PARTICULARLY REQUIRES A VERY ACCURATE AND CAREFUL CONTROL OF THE PURIFIER AT SINGLE STAGE TREATMENT. ANY INCOMPLETE ADJUSTMENT OF THE PURIFIER WILL BE AUTOMATICALLY TAKEN CARE OF IN THE CLARIFIER IN CASE OF A TWO-STAGE CLEANING PROCESS. WITH THE INTRODUCTION OF THE ALCAP-SYSTEM A GREAT STEP HAS BEEN TAKEN TO OVERCOME THE PROBLEM.

DIFFERENT TESTS HAVE SHOWN THAT PARTICLES BETWEEN 5 TO 10 MICRONS ARE DIFFICULT TO TRAP. EVEN MUCH BIGGER PARTICLES ARE FOUND REMAINING IN THE HEAVY FUEL OILS. IN ORDER TO MINIMIZE THE OPERATIONAL RISK FILTERS HAVING CLEANABLE CRNI-STEEL MESH ARE INSTALLED AS A SAFETY DEVICE AFTER THE PURIFIER. REPORTS FROM DIFFERENT INSTALLATIONS ARE INDICATING SATISFACTORY FUNCTION.

THE WHOLE PROBLEM COMPLEX ABOUT DIFFERENT COMBINATIONS OF CENTRIFUGAL SEPARATORS AND FILTERS ARE AT PRESENT THE OBJECT TO A SPECIAL SURVEY BY THE RESEARCH DEPARTMENT AT DET NORSKE VERITAS.

AN INSTALLATION FOR CLEANING OF HEAVY FUEL OIL INCORPORATING A SEPARATOR AS WELL AS FILTER IS SHOWN IN FIG. 9.

WATER PROBLEMS WITH FUEL OILS

RESIDUAL FUEL OILS THAT ARE DELIVERED FROM A REFINERY ARE NORMALLY WITHOUT ANY WATER AT ALL. WATER MAY BE INTRODUCED INTO THE OIL BY CONDENSATION IN STORAGE TANKS ASHORE. ON BOARD TANKERS AND BARGES THE OIL MAY BE MIXED WITH REMAINING WATER IN THE TANKS AFTER CLEANING OR BALLASTING. AS LONG AS THE WATER IS KEPT IN LARGE DROPLETS IT MAY BE SETTLED OR SEPARATED OUT.

AFTER SOME PUMPING, WATER MAY BE EMULSIFIED INTO THE OIL AND IT IS THEN MUCH MORE DIFFICULT TO GET A SEPARATOR TO REMOVE THE WATER.

IN THE LAST YEARS A DISCUSSION HAS TAKEN PLACE BETWEEN DIFFERENT ENGINE BUILDERS AND BUILDERS OF HOMOGENIZING EQUIPMENT REGARDING THE POSSIBILITY OF FURTHER EMULSIFYING THE WATER INTO THE OIL AND THUS MAKING IT MORE HARMLESS IN THE ENGINE.

ONE REASON FOR THESE IDEAS IS THAT NORMALLY THE COMBUSTION AIR WHICH IS TAKEN INTO THE ENGINE SOMETIMES MAY BE RATHER WET AND CONTAIN MUCH MORE SALT THAN THAT SALT WHICH IS SOLVED IN THE WATER IN THE BUNKER TANKS.

FURTHERMORE, TESTS WITH CONTINUOUS WATER INJECTION IN THE FUEL OIL HAS TAKEN PLACE. HEREBY IT IS CLAIMED THAT WITH THE AID OF THE WATER THE FUEL ATOMAZATION IN THE CYLINDER HAS BEEN MUCH MORE EFFICIENT AND THE COOK FORMATION HAS BEEN REDUCED. A BETTER FUEL CONSUMPTION IS ALSO REPORTED. BETWEEN 5-10% OF WATER AT FULL LOAD GIVES ABOUT 2-3% FUEL REDUCTION.

ANOTHER TYPE OF FUEL TREATMENT IS THE INTRODUCTION OF ADDIVITES LIKE "CLEAN POWER" OR SIMILAR, WHICH ARE CLAIMED TO IMPROVE THE COMBUSTION AND THUS REDUCE FUEL CONSUMPTION.

FUEL OILS WITH HIGH DENSITY AND HIGH CONTENTS OF ASPHALTENES HAVE NORMALLY A POOR CETANE NUMBER WHICH ALSO MAY BE CORRECTED BY A SUITABLE ADDITIVE.

UNDER ALL CIRCUMSTANCES THE USE OF ADDITIVES MEANS THAT DOSING AND MIXING EQUIPMENT HAS TO BE INCORPORATED IN THE FUEL HANDLING PLANT.

IT IS NECESSARY TO FURTHER STUDY ALL THOSE NEW PRODUCTS MORE CLOSELY BEFORE A MORE VALID OPINION CAN BE EXPRESSED.

FUTURE FUEL HANDLING AND CLEANING PROCESSES

UP TO NOW THERE HAVE BEEN DIFFICULTIES TO RECEIVE INFORMATION FROM THE MANUFACTURERS OF CENTRIFUGAL SEPARATORS REGARDING THEIR POSSIBILITIES TO CLEAN HEAVY FUEL OIL IN FORM OF GUARANTEE FOR WATER CONTENTS AND IMPURITIES AFTER THE FUEL OIL HAS BEEN PURIFIED. IN MOST CASES THEY PREFER TO GIVE INFORMATION IN FORM OF PRACTICAL EXPERIENCES FROM THE FIELD.

THE GENERAL TREND TO-DAY IS POINTING TOWARDS MORE PRECISE INFORMATION ABOUT PRESTANDA AND PERFORMANCE OF THE EQUIPMENT FOR SHIPBOARD USE. PERHAPS COMBINATIONS OF CENTRIFUGES, FILTERS AND HOMOGENIZERS WILL BE THE SOLUTION FOR THE NEAR FUTURE. WE MAY ALSO FACE THE INTRODUCTION OF NEW PURIFICATION TECHNIQUE SUCH AS ELECTRO-STATIC FILTRATION, OR OTHER CLEANING PROCESSES.

AS LONG AS THE DIESEL ENGINE WILL BE THE MOST ECONOMICAL PRIME MOVER FOR SHIPS' PROPULSION THE NECESSITY OF AN EFFICIENT CLEANING PROCESS WILL BE FURTHER EMPHASIZED WITH THE EXPECTED DETERIORATION OF THE HEAVY FUEL OILS.

DIAGRAM OF CRUDE OIL REFINING

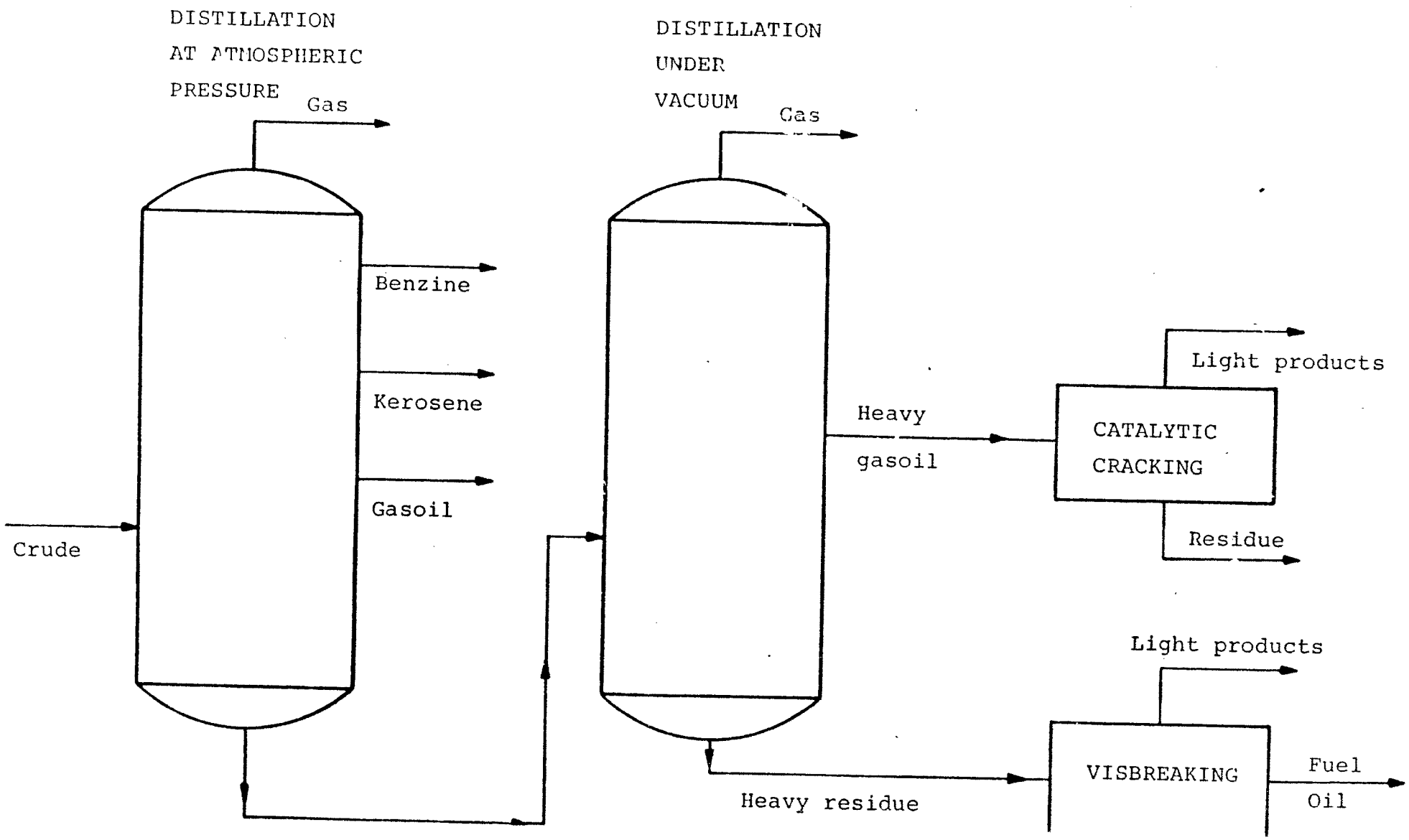


Fig. 1

Fig. 2

CHARACTERISTICS OF RESIDUALS FROM REFINERY PROCESSES

Process	Straight Run Residual Present	Future	F.C.C.	Vis-Breaker
Characteristics			Slurry Oil Residual	
Density	0.96-0-98	0-96-0.98	0.98-1.1	0.98-1-1
Viscosity cSt at 50°C	230-370	250-500	1000	3000
Pour Point °C	10-24	1-40	20-30	-
Carbon Residue % wt.	8	20	-	12-17
Asphaltenes % wt.	4	6	-	10-20

FIG. 3

ANALYSIS OF THE MOST COMMON BUNKER OIL QUALITIES

	ARUBA	CALIFORNIA	AG HEAVY	AG LIGHT	CHINA
DENSITY G/CM^3 15°C	0.976	0.979	0.969	0.950	
FLASH POINT ^{o}C	78	109	95	178	
POUR POINT ^{o}C	3	15	4	12	
WATER VOL. %	0.3	TRACE	< 0.1	0.8	
SEDIMENT WT %	0.01	0.03	0.02	0.09	
CARBON RESIDUE (CCR) WT %	12.9	10.5	8.0	7.8	
ASH WT %	0.085	0.01	0.036	0.024	
SULPHUR WT %	2.30	0.70	3.55	2.62	0.15
NAPPM	60	15	33	101	0.5
NIPPM	51	13	22	11	0.7
VPPM	336	300	67	47	1.0
PB PPM		44	43		
VISCOSITY SEC. R. 1.38^{o}C	2.933	3.960	5.580	1.400	
VISCOSITY KINEMATIC MM^2 /S 38°C	722	354(50°C)	1.373	344	

FIG. 4

CIMAC requirements for specification of intermediate marine fuels

new BSI old	DM 10C CIMAC 1 MC	RM 10C CIMAC 2 MD IF 40	RM 10B CIMAC 3* MM IF 40	RM 10A CIMAC 4* MP IF 40	RM 15D CIMAC 5 ME IF 80	RM 25E CIMAC 6 MO IF 180	RM 25F CIMAC 7 MF IF 180	RM 35H CIMAC 8 MG IF 380	RM 35K CIMAC 9 MK IF 380	RM 45M CIMAC 10 MH IF 500	RM 45K CIMAC 11 ML IF 500	RM 55H CIMAC 12 MI
Density at 15°C (max)	0·920	0·991	***0·991***	0·970	0·991	***0·991***	0·991	0·991	–	***0·991***	–	0·991
Kinematic visc 40°C cSt (max)	14·0											
Kinematic visc 100°C cSt (max)		10	***10***	10	15	***25***	25	35		***45***	45	55
Viscosity Redwood No 1 (max)	80	300	***300***	300	600	***1 500***	1 500	3 000		***5 000***	5 000	7 000
Flash point °C (min)	60	60	***60***	60	60	***60***	60	60		***60***	60	60
Pour point (upper) °C												
Dec 1 - Mar 31 (max)	0(1)	24	***24***	0(1)	30	***30***	30	30		***30***	30	30
Apr 1 - Nov 30 (max)	6	24	***24***	6	30	***30***	30	30		***30***	30	30
Carbon residue mass % (max)	2·5											
Conradson carbon mass % (max)		14	***10***	10	14	***15***	20	22		***22***	22	22
Ash mass % (mass)	0·05	0·10	***0·10***	0·10	0·10	***0·10***	0·15	0·20		***0·20***	0·20	0·20
Water vol % (max)	0·30	0·50	***0·50***	0·50	0·80	***1·0***	1·0	1·0		***1·0***	1·0	1·0
Sulphur (4) mass % (max)	2·0	3·5	***3·5***	3·5	4·0	***5·0***	5·0	5·0		***5·0***	5·0	5·0
Vanadium (4) ppm (max)	100	300	***150***	150	350	***200***	500	600		***600***	600	600
Requirements for which suitable ISO methods are not available for intermediate marine fuels												
Total existing sediment (2)	(2)	(2)	***(2)***	(2)	(2)	***(2)***	(2)	(2)		***(2)***	(2)	(2)
Ignition quality (3)	(3)	(3)	***(3)***	(3)	(3)	***(3)***	(3)	(3)		***(3)***	(3)	(3)
Aluminium (2) ppm (max)	30	30	***30***	30	30	***30***	30	30		***30***	30	30

* *Best IF quality according to CIMAC*

(1) Purchasers should ensure that the pour point is suitable for their trading pattern.
(2) Considered important, but currently no ISO-test method at all available, but suitable method already under preparation.
(3) Considered important but currently no official test method available.
(4) Pending preparation of ISO-standard, requirements for limiting content to be determined by suitable measure-method of an available national standard.

Columns in bold are recommended with preference

FIG. 5

INTERNATIONAL CHAMBER OF SHIPPING

RECOMMENDED BUNKER DELIVERY NOTE

NAME OF VESSEL: @ (PORT)

Product name or general designation	
Quantity in tonnes	
Volumetric Quantity (m^3)	
Temperature of Product (°C)	

Density at 15°C (kg/litre)	
Kinematic Viscosity (cSt) at corresponding temperature (°C)	@
Flash Point (P–M) Closed Cup (°C)	
Pour Point (°C)	
Conradson Carbon Residue (percentage by mass)	
Sediment (percentage by mass)	
Water Content (percentage by volume)	
Cetane Index (distillate fuels only)	
Sulphur (percentage by mass)	
Vanadium (mg/kg)	

Note: Specific energy (calorific value) (MJ/kg) can be calculated from the known density and sulphur contents in the formulae quoted in BS 2869 Amendment 2 "Petroleum Fuels for Oil Engines and Burners".

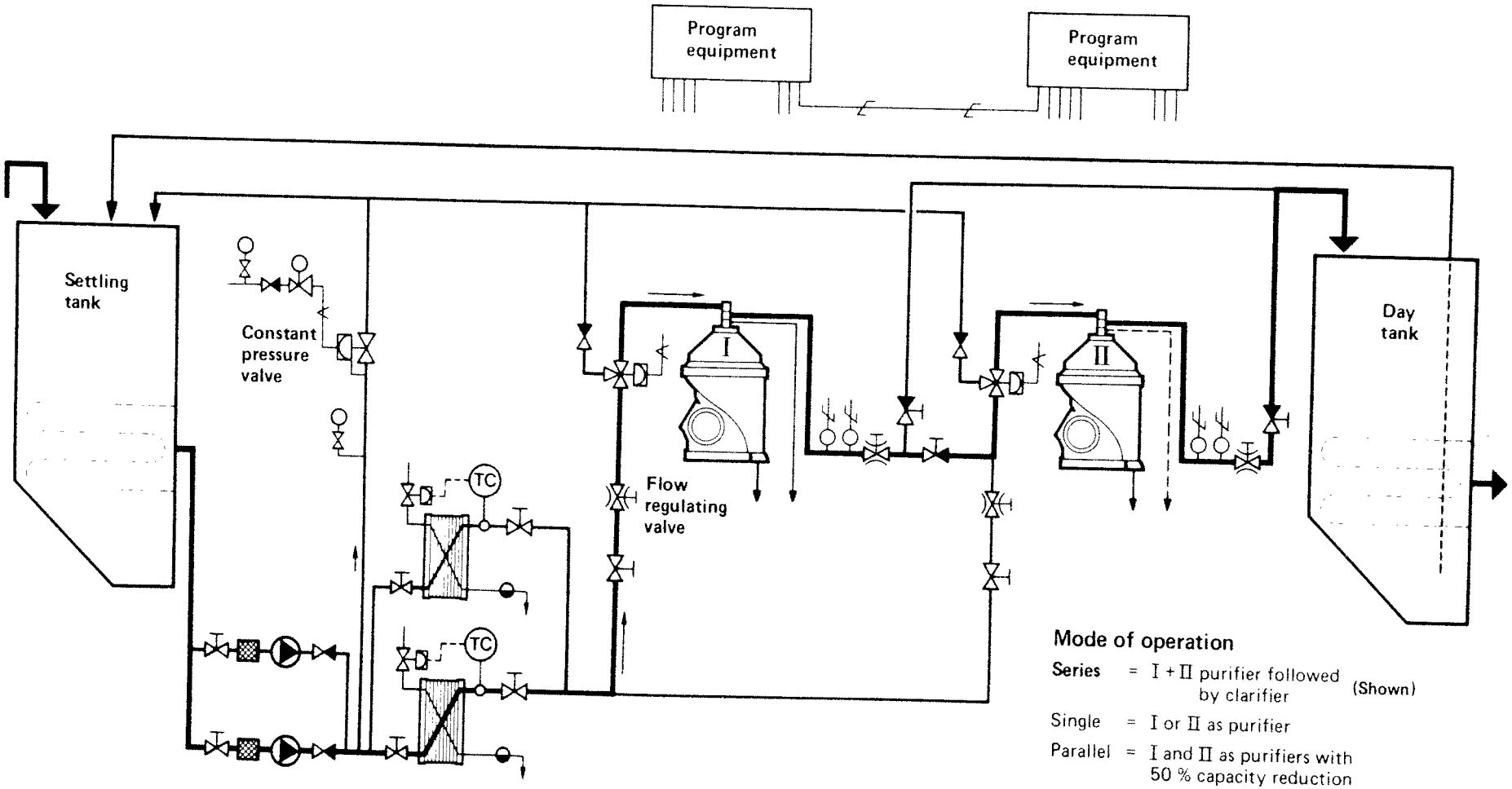
Bunker Supplier: _____

Signature of Suppliers' Representative or Agent: _____

Date of Delivery: _____

Drawing no. 1761081

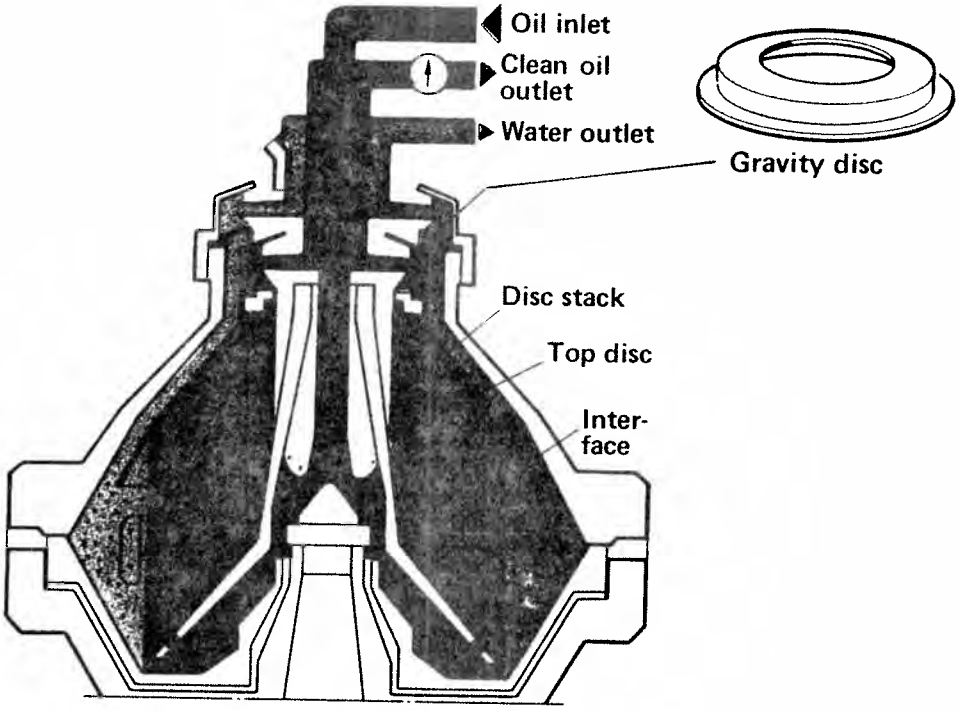
HFO cleaning system, separate feed pumps.
Operation in series



ALFA-LAVAL

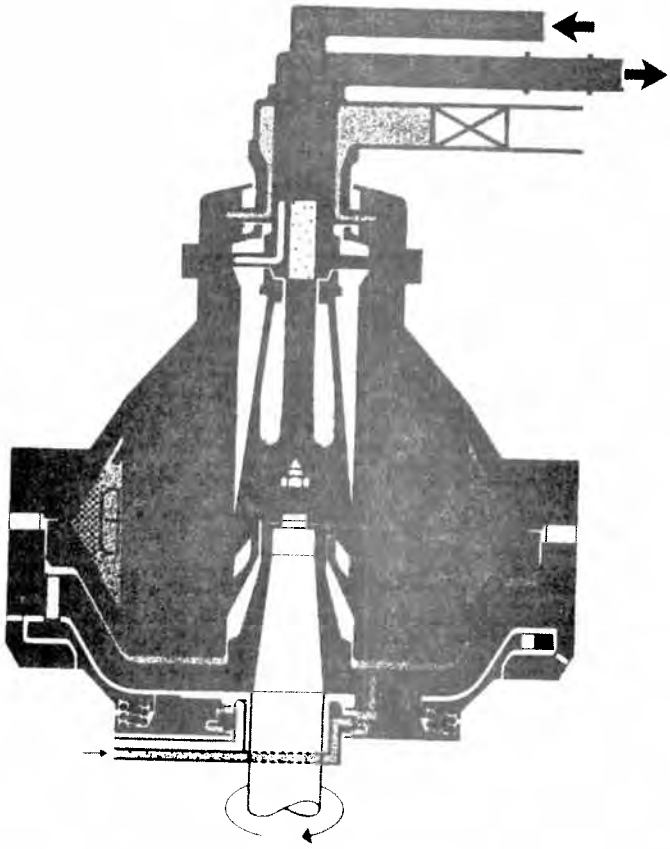
Fig. 6

FIG. 7



Purifier – correct interface position.

FIG. 8



CLARIFIER

FIG. 9

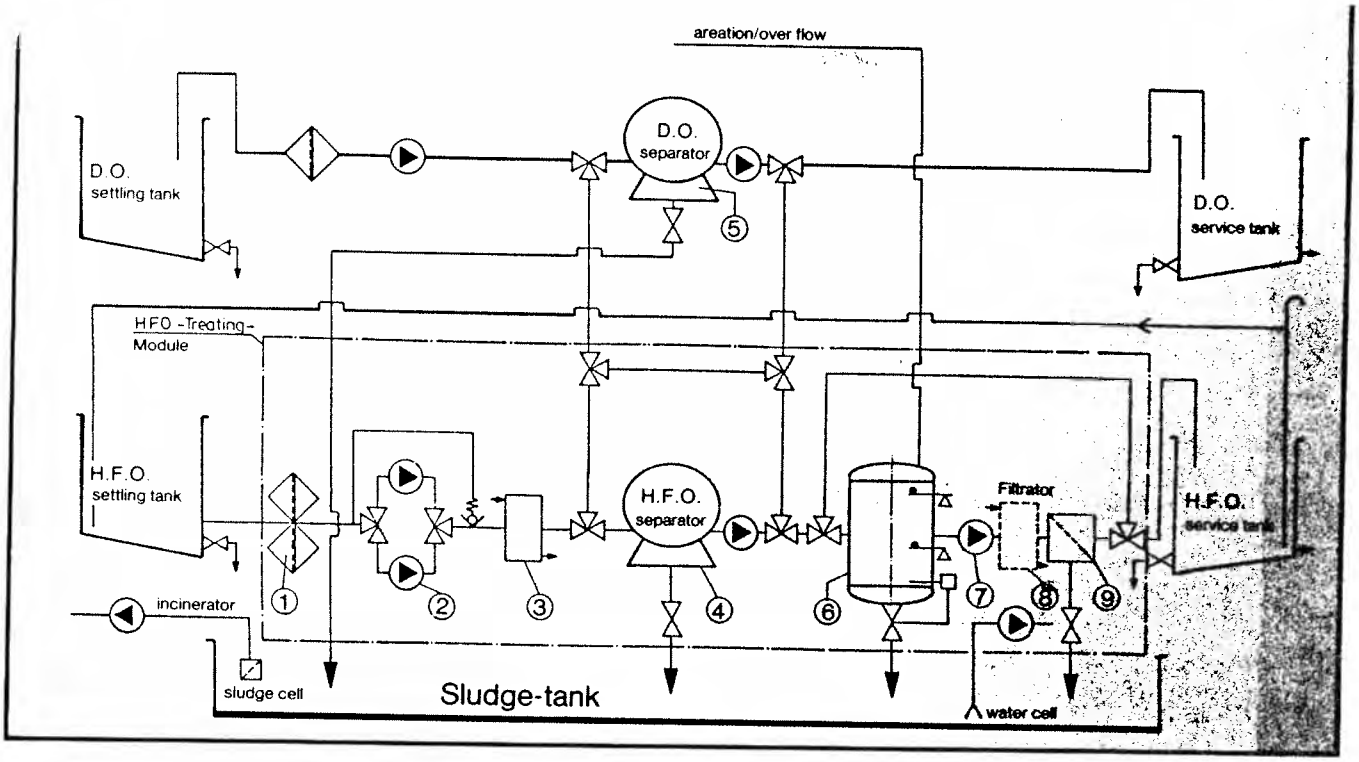


Diagram of a two-stage heavy fuel oil treatment system combining separation and filtration. 1) Filter to protect pumps, 2) pumps, 3) heat exchanger, 4) heavy fuel oil separator, 5) diesel oil separator, 6) heavy fuel oil buffer tank, 7) Filtrator's pumps, 8) additional heat exchanger, 9) Filtrator's automatic filter for fine filtration.

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

COAL FIRED SHIPS OF NEW GENERATION

by

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ITALCANTIERI SPA

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COAL FIRED SHIPS OF NEW GENERATION

ing. E. Derdini

Mod. 90.0163.02 ARD/Mult. 4.74

COAL FIRED SHIPS OF NEW GENERATION

INTRODUCTION

The prevailing role that energy plays, was highlighted by the oil crisis which affected, in one way or another the entire wirld's economy during the seventies. In fact, before the crisis broke out, easy operation, reduced maintenance, high performance and so on, had a premium over fuel consumption. Nowadays, these values have been completely inverted, having energy saving become one of the most important tasks of modern science.

The aspect of energy saving is mainly referred to oil. In the marine field, these problems are tackled in various ways, i.e. reduction of the ship's speed, greater propulsion efficiency with low rpm propellers, diesel engine having increasingly higher efficiencies, more and more sophisticated systems for waste heat recovery, etc. and lastly, recurring to fuels alternative to oil.

In this context, coal is now gaining a revival, although still rather limited. Consequently, steam propelling machineries which are identified as the most flexible plants, as far as the type of fuel is concerned, are also coming back.

The return of coal as marine fuel has been made possible by modern technologies which have allowed to obtain reasonably high efficiencies and therefore rather low consumptions. In addition, the fully automated operation of coal firing propelling plants, similar to oil firing, was made possible perhaps with even a lower degree of pollution.

In comparison with the past, the reduction of fuel rate deserves the merit of reducing both running costs and bunkering space on board. In fact, to this effect, we can immediately realize how bunkering space may well represent an actual problem for coal fired ships and must therefore be carefully studied.

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As we are all aware, the fuel consumption of a steam plant depends on several factors. The most important among there are:

- calorific value of fuel,
- boiler efficiency,
- selected heat cycle.

It is on the heat cycle that valuable improvements can be attained like, for example, through regenerative cycles with an increased number of bleedings which could also be combined with reheating systems.

Obviously, a plant of this type, which has already been used in oil fired ships, is more difficult to realize in the case of coal firing. As we shall see later on, the first coal fired ship built in our yard, which includes the classic 3 bleedings cycle has been adopted with boilers having an economizer and a steam air heater. A future task is that of developing more sophisticated and more efficient systems as mentioned beforehand.

With regard to the new installations, typical of the coal fired ships, the key plants are certainly the coal handling and feeding as well as the ash evacuation disposal. Moreover, adequate provision have to be taken to face the slower response of coal combustion under manoeuvering conditions.

We shall now examine in general, the various implications of coal firing. We shall then pass on to the description of the Propelling Machinery of the two coal fired ships built by ITC.

TECHNICAL ASPECTS OF MODERN ERA "COAL FIRED SHIPS"

As we shall see further ahead, the type of boiler at present adopted in ship's propulsion is the "travelling stoker" type, although a pulverized coal system may seem to be more rational at first sight, as it is generally used in land installations.

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The fluidized bed boiler, which represents the most attractive solution, due to the fact that it allows different qualities of solid fuel to be burnt, has not yet reached an operatively safe stage suitable for installation on board ships.

Let us now summarize the fundamental aspects of coal fire propulsion at the present time and the most important technical problems which can be summarized more or less as follows :

- Larger sized boilers, far more complex than oil fired boilers with a notably major number of auxiliary devices, such as : driving systems for stoker, feeder and coal spreader, forced and induced fans with further addition of overfire air, cyclones for the collection of soot and fly ashes etc.

- Coal conveying and feeding systems, which must, among other things, avoid coal to be broken with consequent reduction of size.

- Ash storage and disposal system.

- Fire and explosion prevention and safety problems.

- Air pollution and ecological problems that could be caused by coal dust and ashes etc.

- Automation and monitoring problems like, for example, the installation of closed circuit television cameras for the control of the condition of the stoker inside the furnace.

- Coal bunker storage (e.g. coal cannot be kept inside the coal storage units for lengthyperiods without adequate movement due to the danger of spontaneous combustion).

- Shape and geometry of coal bunkers with the elimination of recesses where possible coal clogging may occur.

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- Regulations of Classification Societies still under-development but however already quite advanced.

Foremost, it must be observed that it is not possible at present, to obtain a coal bunkering in every port of the world. What is more, boilers with coal firing on stokers are necessarily designed to burn a specific type of coal, of a given composition, calorific value and ashes and are not suitable to burn coal of different qualities.

In order to maintain low costs, it would also be logical for bunkering to be carried out in coal-producing countries. As the availability of a certain type of coal will surely be limited, fixed routes between two ports, one of which having bunkering facilities, appears to be more attractive.

Furthermore, as the calorific value of coal is lower than of fuel oil and the total efficiency of the steam cycle is lower than that of diesel, the coal storage bunkers would have to be much larger than those for oil.

So roughly, a volume increase of 3 ÷ 3,5 times of the storage bunkers can be estimated.

On the contrary to fuel oil, one must further note that coal requires bunkers of a geometrically defined form.

The greater weight and volume of solid fuel has therefore a notable influence on the loading capacity of the ship in relation to her type, duration of voyages and speed.

As already mentioned, the boilers must, in fact, be designed for a certain well defined type of coal.

On a travelling stoker boiler, coal sizes must be kept within certain limits.

A typical specification is, for example, the following :

- 95% of the coal must be able to pass through a sieve having holes which measure 3/4";

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- not more than 50% must pass through a sieve having holes which measure 1/4".

The reasons for these severe limitations can be summarized as follows :

Excess of small sized coal :

- excessive deposit of coal in the furnace in proximity of the rotating spreader;
- increase of particles emitted with gases due to the major quantity of fuel burnt in suspension;
- reduction of boiler efficiency due to a major loss of coal;
- loss of coal due to its passing through the stoker bars;
- irregular distribution of the coal along the stoker with an irregular bed thickness;
- pulsations inside the furnace;
- overheating of the coal distribution system;
- black smoke.

Excess of large size coal :

- slow ignition;
- ashes with a high carbon content due to incomplete combustion;
- flame instability;
- blake smoke.

Bunkering can be carried out by means of various methods. The choice of the most suitable depends on many factors among which, time availability for refuelling, port facilities etc.

In general pneumatic-type systems are preferred, similar to the ones relative to coal transfer from the bunker to the ready use hopper which can be installed together with the normal hatches for alternative loading by means of grabs.

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COAL COMBUSTION SYSTEMS

Coal can be burnt in the boilers according to various systems among which, however, the following are the more pratical and the more widely used:

- fluidized bed combustion;
- pulverized fuel combustion;
- coal combustion on stokers with mechanical distribution.

Let us try to briefly examine the characteristics of the first two systems, stopping a while longer on the stoker system which is in fact, the one that interests us the most.

FLUIDIZED BED COMBUSTION

Coal combustion occurs on a bed of inert granular material which could be sand or even the ash itself.

The coal is thoroughly mixed with the mass of inert particles.

A bundle of vaporizing and superheating tubes is submerged into the forementioned fluidized bed directly absorbing part of the heat produced during combustion.

At maximum load, combustion occurs at a temperature which generally does not exceed 850° C.

A highly favourable characteristic of this system (besides obtaining higher efficiency) is that of being able to burn fuels of different qualities even of a lower standard and of different sizes, in the same boiler.

Low temperature combustion avoids the melting of "clinker".

However, in consideration of the limited experience, acquired exclusively in land installations, sufficient guarantees are not yet available as regards the possibility of employing the existent technologies in safe and efficient marine systems.

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In fact, this system has been studied for not more than ten years.

Anyhow, due to its favourable characteristics, fluidized combustion has a consistent progress and therefore concrete applications can be expected.

PULVERIZED COAL COMBUSTION

This is the most widely adopted coal combustion system for large boilers in land installations.

It foresees the use of finely pulverized coal which is injected into the furnace by means of burners through appropriate airflows.

Problems which are caused by the use of pulverized coal on board ships are essentially: that of the difficulty of transportation in bunkers due to the serious danger of spontaneous combustion and that of the ashes, which are all emitted through the funnel together with combustion products.

In land installations, in fact, complicated and expensive collecting systems, both mechanical and electrostatic, are provided which, among other things, are rather cumbersome.

On the other hand, when burning on the stoker the ashes are gathered underneath the stoker itself.

An alternative to the bunkering of the pulverized coal could be that of milling it on board the ship itself; of course this is a practice which cannot, at least for the time being, be taken into consideration.

On the contrary to what could intuitively be thought regarding the danger of spontaneous combustion or even explosion, this is much higher for coal than for fuel oil.

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COMBUSTION OF COAL IN PIECES ON THE MECHANICAL STOKER

This is the problem, we are mainly concerned with and shall therefore be more deeply examined.

First of all, the dimensions are notably larger in respect to an equivalant oil fired boiler, particularly with regard to the furnace, having a rather greater volume, in addition to the arrangement of the travelling stoker.

As regards water and steam drums and tubes, these are similar to the oil fired boiler, keeping in mind, however, the necessity of major space to increase the accessibility between tubes due to greater quantity of combustion products.

Another substantial difference is that of the combustion air which is necessary to the coal both in higher quantity and pressure.

In fact, the excess of air which is approx 5% for oil, reaches 30% for coal whilst the air/gas circuit offers a notably higher resistance also because of the presence of the cyclones.

Instead of forced draught fans only we have:

- forced draught fans for air underneath the stoker;
- high head overfire fans;
- induced draught fans.

Solid particles which abandon the furnace together with the gases are formed partly by incombusted particles that must be collected in special hoppers and reinjected into the boiler so as to obtain a combustion as complete as possible.

The coal fired boiler is also fitted with an oil burner for emergency and for particular manoeuvring needs.

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There is no need to provide for the regulation of the superheated steam temperature in a coal fired boiler due to the fact that design limits are normally not overcome.

On the contrary, it is more probable that the steam temperature drops underneath the design limits.

The installation of an economizer to improve the efficiency is provided, whilst for the combustion air, a steam/air heater is usually installed to improve the efficiency of the steam cycle.

However, the air temperature must never reach high levels (180 °C max) so as to minimize the danger of "clinker" formation (hard agglomerate of ash which has reached fusion point).

Let us now see how coal feeding and combustion occurs.

The essential components are the feeder/spreader and the travelling stoker.

The first is made up of a rotary feeder which is fed by gravity from a ready use hopper and a rotary spreader which is composed of a shaft with a series of adjustable radial blades, driven by a motor at variable speed.

The distribution is controlled by regulating the length of the blades so as to establish the point of impact with the coal and therefore its trajectory.

A correct coal feeding/spreading can be obtained in the furnace by setting the speed variation with the adjustment of the blades and therefore of the trajectory. Each boiler can be equipped with three or more coal feeders/spreaders.

The other part of the stoker is formed by the travelling grate.

This is composed of transversal rectangular bars or plates of heat-resistent cast iron, fixed on longitudinal chains and driven at variable speed by means of gears and electric motor.

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The interstices between the bars are closed off by overlapping keys, so as to force the air to pass through Venturi calibrated holes, drilled in the bars so as to establish a correct distribution.

A primary air flow, supplied by forced draught fans and corresponding to about 80% of the total combustion air, is injected underneath the stoker cooling it down before passing through the above mentioned calibrated holes to take active part in the combustion. A secondary air-flow supplied by an overfire fan and corresponding to approx 20% of the combustion air is injected directly above the stoker.

The stoker, which is made up of one or two separate sections, is designed to travel at a speed which is normally adjustable between Ø and 6 m/h, towards the front of the boiler, on which the stoker/spreader is located.

The transversal bars are selfcleaning and thus the ashes fall into the front side; the clinkers break up and are collected in the main hopper.

Other hoppers, for the collection of ashes are fitted at the back of the stoker underneath the superheater and the boiler smoke box in addition to the cyclone installed on the gas exhaust duct.

During combustion, the smaller coal particles tend to burn in suspension without reaching the stoker. Instead, the large pieces fall on the stoker and are distributed so that the bigger ones fall further away whilst the pieces of smaller dimensions fall progressively nearer to the front of the boiler.

Approximatively 40-50% of the coal burns in suspension whilst the remainder makes up a layer which burns on an ash bed above the bars of the stoker.

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By regulating the travelling speed of the stoker, the coal burns completely within the time employed for it to reach the front, so as to discharge only the ashes into the hopper.

The system provides a degree of flexibility which is acceptable for normal navigation conditions and a sufficiently rapid response to the necessities of normal load variations. In order to increase this flexibility in port, one of the two stoker sections is stopped, whilst to satisfy immediate requirements of load variation during manoeuvring, a steam dumping system is provided, directly connected with the condenser.

The quantity of coal supplied is directly controlled by the feeder which has an almost immediate effect on the heat produced by coal that burns in suspension, whilst the remaing layer of coal burning on the stoker has a low thermal inertia and the quantity of heat produced varies very rapidly in function of air flow of the forced draught.

Among the various systems, the spreader/stoker is, anyhow, the one that offers major guarantee of good operation in a marine boiler, inasmuch as it has already been widely used in land installations, in addition to its sturdiness and operative flexibility compatible with an automatic control.

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As it is certainly well-known, my company, the Italcantieri of Trieste, has at present two coal fired ships under completion. I am certain you will find it interesting to proceed with the examination of these already existing installations.

Main characteristics of the ships :

- length overall		255,00 m.
- breadth moulded		35,30 m.
- depth moulded		19,00 m.
- design draught, moulded		12,20 m.
- corresponding deadweight	abt.	75.750 tons.
- cargo capacity (100% full)		67.800 m3
- main propulsion M.C.R.		19.000 SHP
- trial speed with ship fully loaded, in shallow water (32 m depth) at M.C.R. of propulsion unit		15,08 knots
- propeller rpm		85 rpm
- normal navigation rating		17.000 SHP
- calculated fuel rate at 19.000 SHP (1200 Kw E.L. load)		481 g/HPh
- calculated fuel rate at 17.000 SHP (1200 Kw E.L. load)		488 g/HPh
- total fuel consumption at 19.000 SHP		9139 Kg/h
- daily fuel consumption at 19.000 SHP		219,3 tons/day

PROPELLING MACHINERY ARRANGEMENT

Before going into more detail about the various installations, I find it appropriate to take a look at the general arrangement of the propelling machinery shown in fig. 1, 2 and 3.

The coal bunker (which has a capacity of approx. 2.400 tons of coal) is located forward of the propelling machinery room, adjacent to the boiler, in order to keep the distance for fuel feeding to a minimum. Loading pipe-lines, sensors and air filters are located in the upper section. Screw conveyors and pneumatic denseveyors for the transfer of coal are found in the bottom section. The boiler is located immediately aft.

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The ready use hopper, which supplies the boiler feeders with coal by gravity, has been installed between the coal bunker and the boiler. The "plenum chamber" for the combustion air, the ash hopper and the pipe-lines for the removal of the ashes, can be seen under the stoker.

The hoppers for the storage of the ashes are located at the sides of the boiler.

The arrangement of the main condensing unit can be clearly observed on the plans.

It consists of one unit GE MST 14 designed by General Electric and manufactured by Cantieri Navali Riuniti.

The Control Room is situated on the third flat and also includes the main switchboard.

HEAT CYCLE

The heat cycle is represented at least in its essential lines on figure 4.

The superheated steam feeds directly at full pressure and temperature, the main turbines the turboalternator and the two feed pumps.

A three bleedings regenerative cycle has been adopted, with a feed water temperature of 149°C, suitable for an economizer recovery system.

Considering that the response time of the coal fired boiler may not be sufficiently rapid (approx. 2 minutes for passage from the maximum to minimum load) in respect to the requirements for quick-manoeuvering, an automatic dumping system has been installed with two desuperheating stage : one in the upper drum of the boiler and an external one with feed water atomization.

MAIN BOILER

Let us firstly note that for better efficiency, easier operation, less weight and space requirements, a single boiler installation has been chosen.

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The boiler (see fig. 5) is of Combustion Engineering design and built in Italy by Franco Tosi.

It is of the membrane water tube type, with two drums natural circulation and a two-stage superheater.

The normal Continuous Rating of the boiler is 62000 Kg/h of superheater steam at 6 MPa and 491°C with coal firing, while with diesel-oil firing 50% of M.C.R. can be reached.

As regards water and steam sections, they are similar to the ones of an oil fired boiler: while the combustion air and exhaust gas sides are more complex considering a balanced draught system is used and for this purpose the following fans have been installed:

- 2 two-speed forced draught fans
- 1 single-speed overfire fan of high head
- 2 two-speed induced draught fans

Approximately 80% of the combustion air is introduced underheath the stoker in the plenum chamber which is sub-divided into two parts, the same as the stoker. One of these is cut off during port service.

A secondary air flow fed by the overfire fan and corresponding to the remaining 20% of combustion air is directly injected over the stoker without pre-heating.

A slightly negative pressure is maintained by the induced draught fans in the furnace so as to ensure the absence of leakage in the boiler room.

An economizer has been fitted at the gas outlet, whilst a steam heater has been installed for heating the combustion air.

Fly ashes are caught by mechanical cyclone separators fitted with automatic dampers at the inlet and outlet of the three cyclone sections in order to maintain a good efficiency at all conditions.

The stoker is made up of five sections and moves towards the front of the boiler where it discharges the ashes into the hopper underneath.

The coal is distributed on the stoker by six groups of feeders and spreaders which are supplied by gravity through two ducts of 800 mm diameter.

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Automatic regulation allows a turn-down of 1 to 4. The lower limit corresponds to approx. 15.500 Kg/hr of steam. A higher ratio is obtainable by means of manual adjustments. The minimum response time to pass from maximum load with all stoker and feeder sections functioning automatically, is of about 2 minutes.

To meet the rapid variations of steam requirements during manoeuvres, an automatic dumping system to the condenser has been provided, for a maximum of 16.000 Kg/hr of steam adequately desuperheated. This solution, suitable for manoeuvering conditions, is not sufficient for port service.

For this condition, four of the six feeder groups and the three corresponding stoker sections can be turned off.

THE COAL BUNKER (see figg. 6 ÷ 7)

The main coal bunker having a capacity of about 3500 cm has been designed applying the theory of the "mass-flow" of bulk materials. In particular, theoretical studies and tests have been carried out with the aim of determining the optimum number, angles and form of the lower chutes.

These chutes are made of corten steel, to limit the wear caused by coal abrasion. A certain number of air cancer on the sloped walls have been installed for the removal of possible arching or blockages by means of compressed air jets.

In the upper section of the bunker and in the ready use hopper, CO detectors have been installed for the constant control of the concentration of this gas in several places through the main monitoring system.

In case of sportaneous ignition of coal, seawater mass injection is provided on top of the bunker.

COAL HANDLING

From ashore, the coal can be loaded by means of a pneumatic conveyor system or by grabs through the two hatches (see fig. 6 and fig. 7). The loading dump valves, which have an automatic sequence control, permit the coal to be evenly distributed on the entire surface of the storage hold.

The sensors, fitted on top of the bunker, close each single drop valve when the service level is reached.

The system, which conveys the coal from the main bunker to the ready use hopper, must satisfy the following fundamental conditions :

- adequate use of the available space
- reduced maintenance
- absence of ambient pollution
- minimum coal degradation
- service reliability even in rough sea
- low level of noise
- possibility of automatic operation

Among the various solutions examined, in agreement with the shipowner's requirements, a pneumatic conveyor system of the "dense phase" type has been chosen. The system is made up of six pneumatic units, each one fed by two screw conveyors, as clearly illustrated on the plan.

The screw conveyors are composed of a tapered hub and variable pitch flies so that the coal flow from the hopper is uniform along the entire discharge opening.

The ready use hopper is fed by denseveyors by means of independent lines; in this way a high degree of reliability can be attributed to the feeding system.

The system has been carefully tested in our shipyard where a test rig has been installed reproducing a line identical to one fitted on board.

All the effects of the coal condition have have been checked, in particular, the moisture content. These tests have permitted to determine all the interventions capable of overcoming a great variety of situations

ASH HANDLING (see table 8)

On the basis of an average content of ashes in coal equal to 15% a maximum of 33 tons of ashes could be produced daily, the major part of which will be continuously discharged by the stoker in the hopper situated underneath, whilst the remaining part will be eliminated by the cyclones.

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If the ship operates in deep water, the ashes can be evacuated directly into the sea, otherwise they are stored in appropriate lateral hoppers having a total volume of approx. 600 cubic meters, equal to 10 days of full power navigation for subsequent discharge into the sea or ashore.

For ash handling, a vacuum system supplied by National Conveyors USA, has been installed. The vacuum in the lines is produced by steam ejectors. The output is around 15 tons/hr.
The lines are made of special cast iron alloy, particularly resistant to abrasion with bends that can be rotated 180° or renewed when necessary after a prolonged period of operation.

Discharge overboard is carried out by a steam ejector which throws the ash down one meter diameter vertical duct from the main deck right to the sea. A certain number of nozzles, around the circumference of the duct, provide tangential water jets for the complete wetting of the ashes and their rapid elimination.
The system is completely automatic. After having selected the operation mode (direct discharge overboard, storage in main hoppers, evacuation from the hoppers etc.), the various sequences are controlled automatically. The automation system explores the condition of each hopper in a cyclic mode; when a certain level is detected, the preset sequence is then activated.

TAKE HOME MOTOR

Taking into account the single boiler installation and the necessity of being able to manoeuvre the ship in any emergency circumstance as a consequence of the particular routes alongside the barrier reef, a take-home electric motor has been installed.

This emergency propulsion system utilizes a wound rotor asynchronous motor of 1500 kW and 4 poles which can be started in 14 steps; this solution mainly allows the limitation of the torque required for starting so that it results to be compatible to that developed by the dieselgenerator.
The take-home motor can be engaged very quickly to the first reduction L.P. pinion. A speed of about 6 knots can be reached.

AUTOMATION SYSTEM

The completely automatic management of the propelling machinery, from the Engine Control Room or from the bridge, is provided, in line with the L.R. U.M.S. requirements.

The control and monitoring are based on the Honeywell TDC 2000 digital system. This system is made up of four peripheral units, of microprocessors appropriately distributed in the Engine Room and of a unit for the control and central processing in the Engine Control Room which can be operated also from the bridge.

Each peripheral unit can continue functioning automatically, even in case of damage to the others and this is connected to the central unit with two different coaxial cables and thus the system has a high degree of realiability.

The central unit is doubled, that is, apart from assuring a certain redundancy, permits the use of a video for the signalling the alarms whilst the other is used to examine the condition of the systems and of the single controlled parameters.

The central system controls the boiler but it is always possible to take over the local control on the front of boiler.

The main propulsion unit is controlled by an independent electro-hydraulic GECO system interfaced with the central system.

In addition to the boiler and propulsion systems, also other plants are fitted with individual systems of automatic control interfaced with the TDC 2000 for data monitoring and recording.

- Electric Power Station : an ASEA-NIFE GENA system supervises the functioning of main and auxiliary generators
- Coal Handling : it is equipped with a local and centralized control system which guarantees the coal level in the ready use hopper activating the various operational sequences
- Ash Handling : the control system cyclically explores the filling up state of the hoppers and according to the functioning mode chosen (discharge into storage hoppers, to sea or on land), operates the various components of the installation in suitable sequence.

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The distributed control system is able to record the data and alarm and to calculate trends and display a great variety of average values and statistics.

CONCLUSION

Lastly, I should mention that the aim of this rather concise lecture is that of going through all the main peculiarities dealing with the coal firing installations. Obviously, each problem must be further deepened by specialists in the field.

I furthermore, feel that, more useful than anything else, was the review of the installations which were realized by our shipyard. To date, the first ship is ready for how sea trials. I shall therefore be able to give further information/details during the occasion of the forthcoming lecture.

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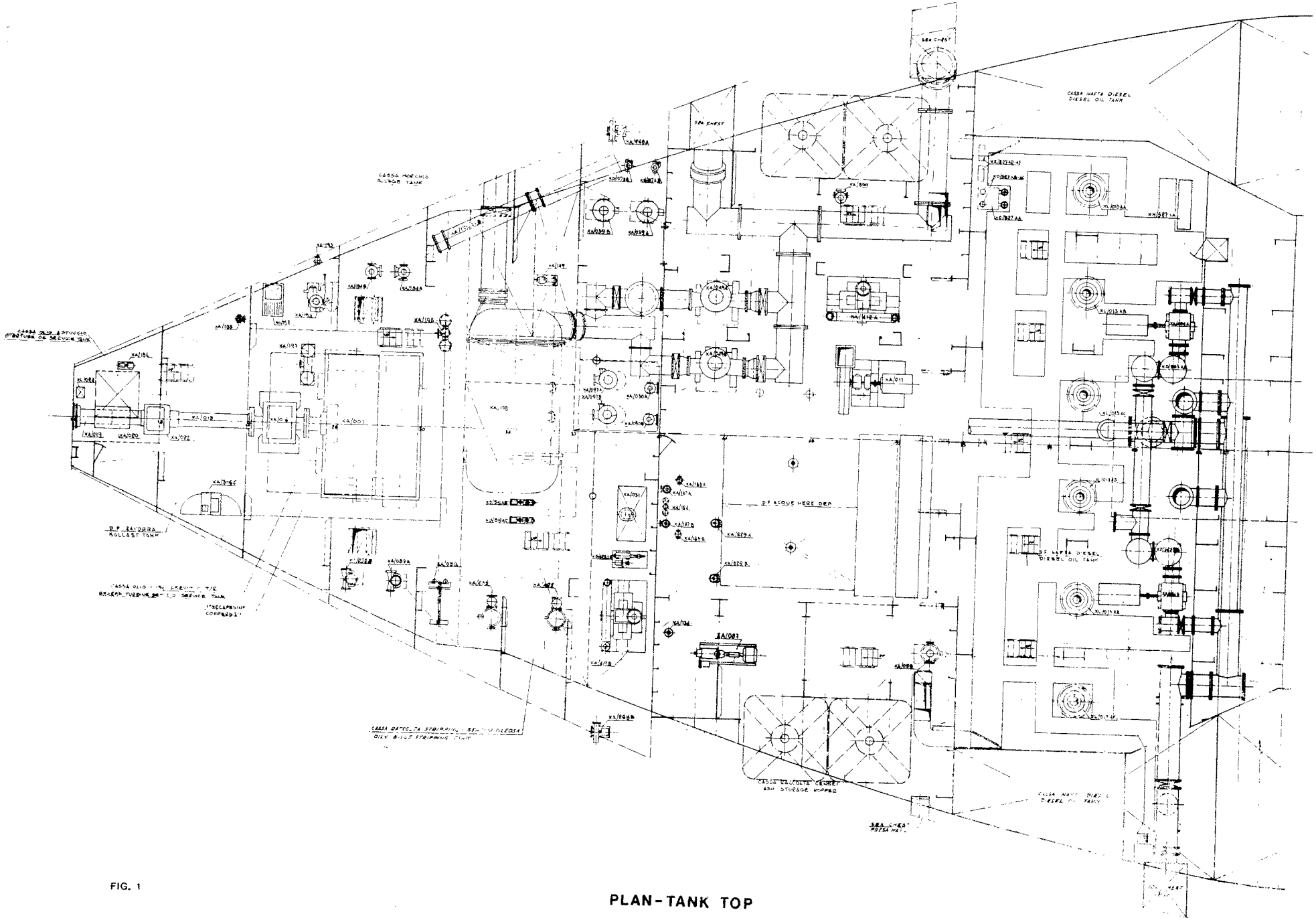


FIG. 1

PLAN-TANK TOP

ELEVATION

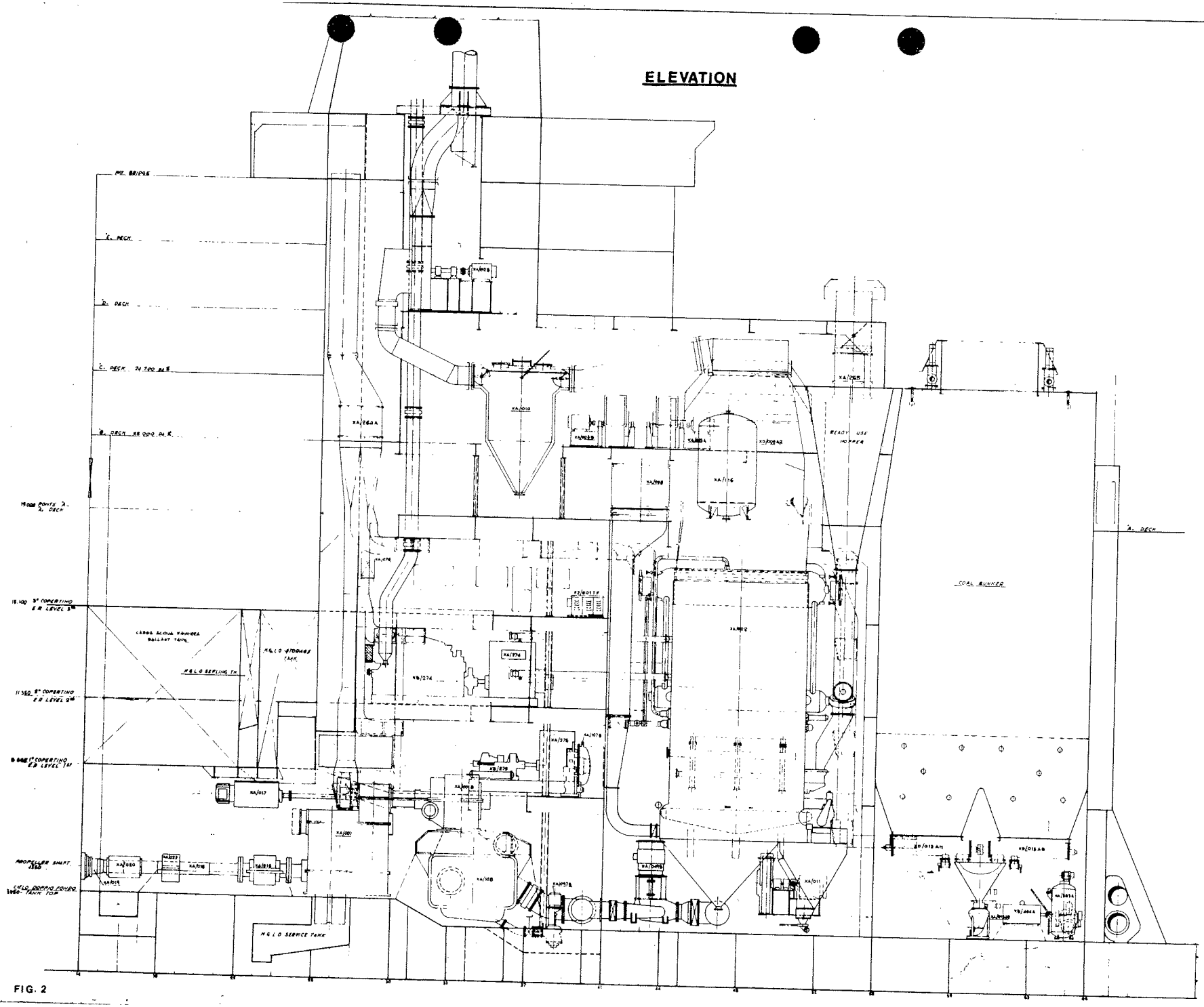
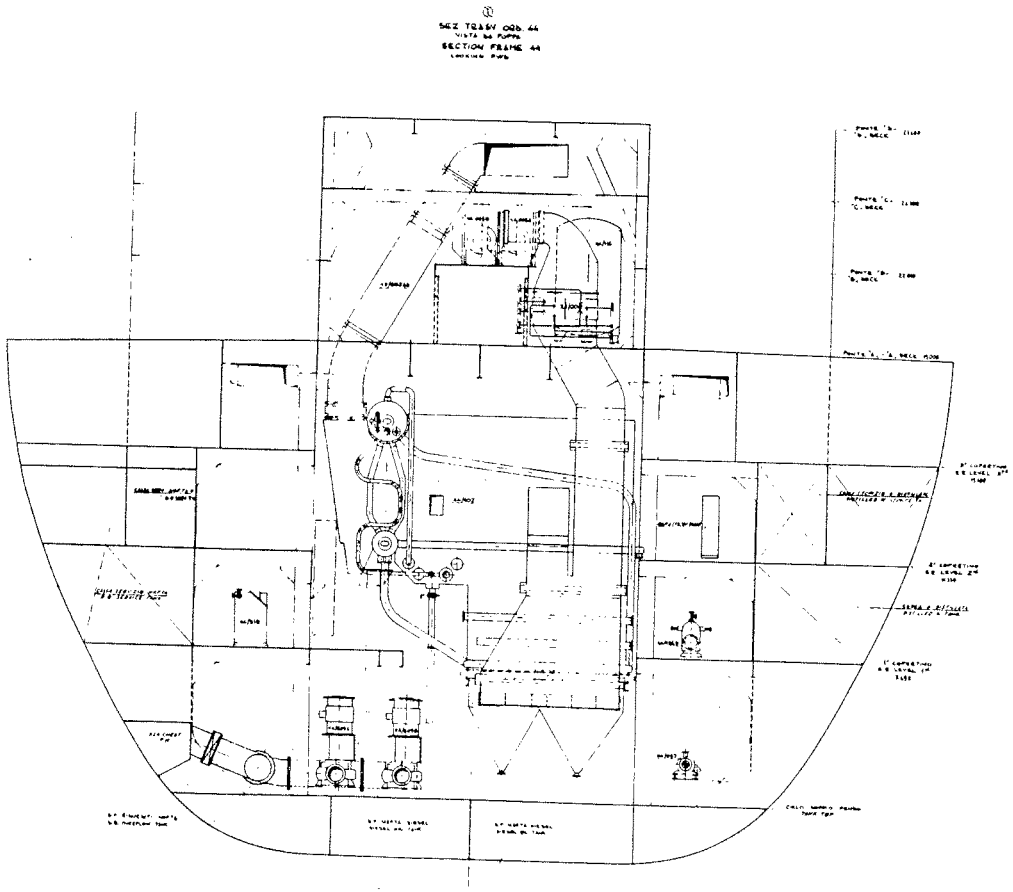


FIG. 2


SEZ. TRASV. OGD. 44
VISTA DA POPPA
SECTION FRAME 44

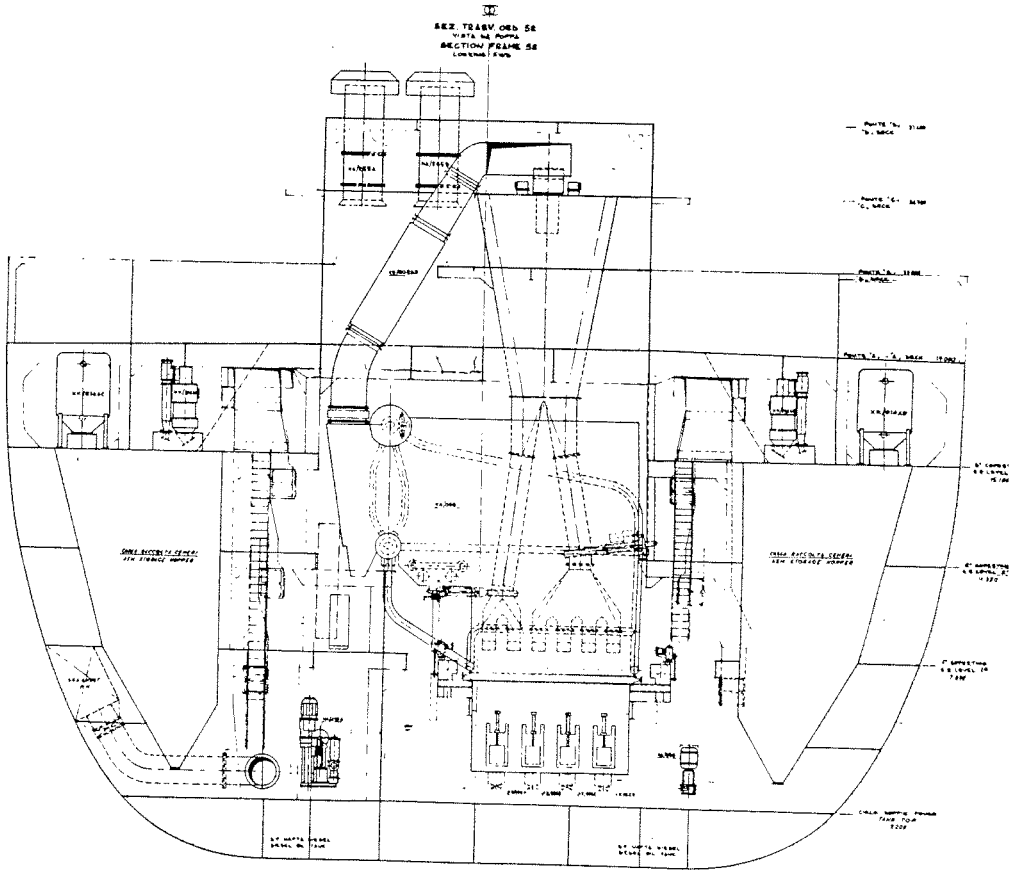

SEZ. TRASV. OGD. 58
VISTA DA POPPA
SECTION FRAME 58

FIG. 3

HEAT CYCLE DIAGRAM

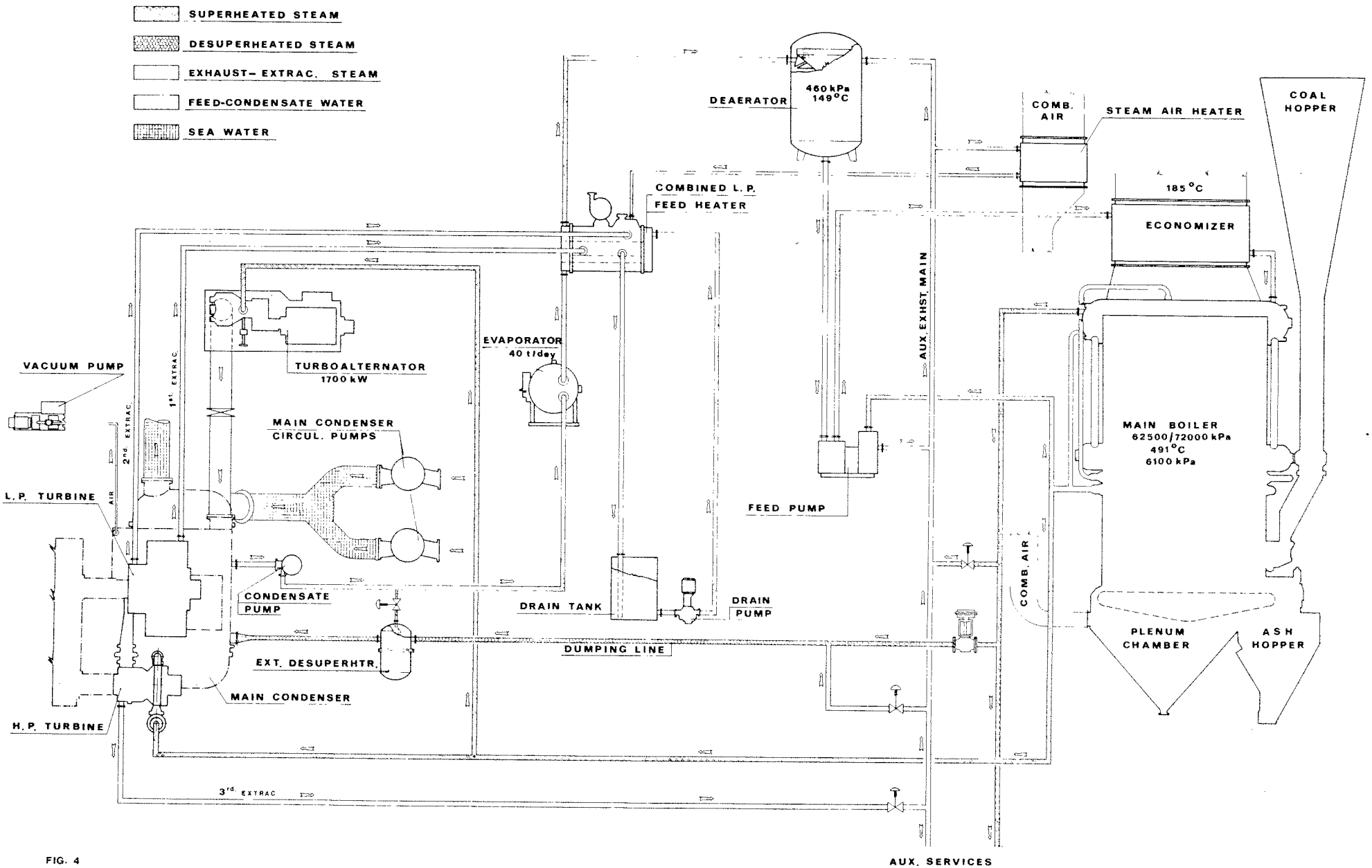


FIG. 4

AUX. SERVICES

COAL-FIRED BOILER

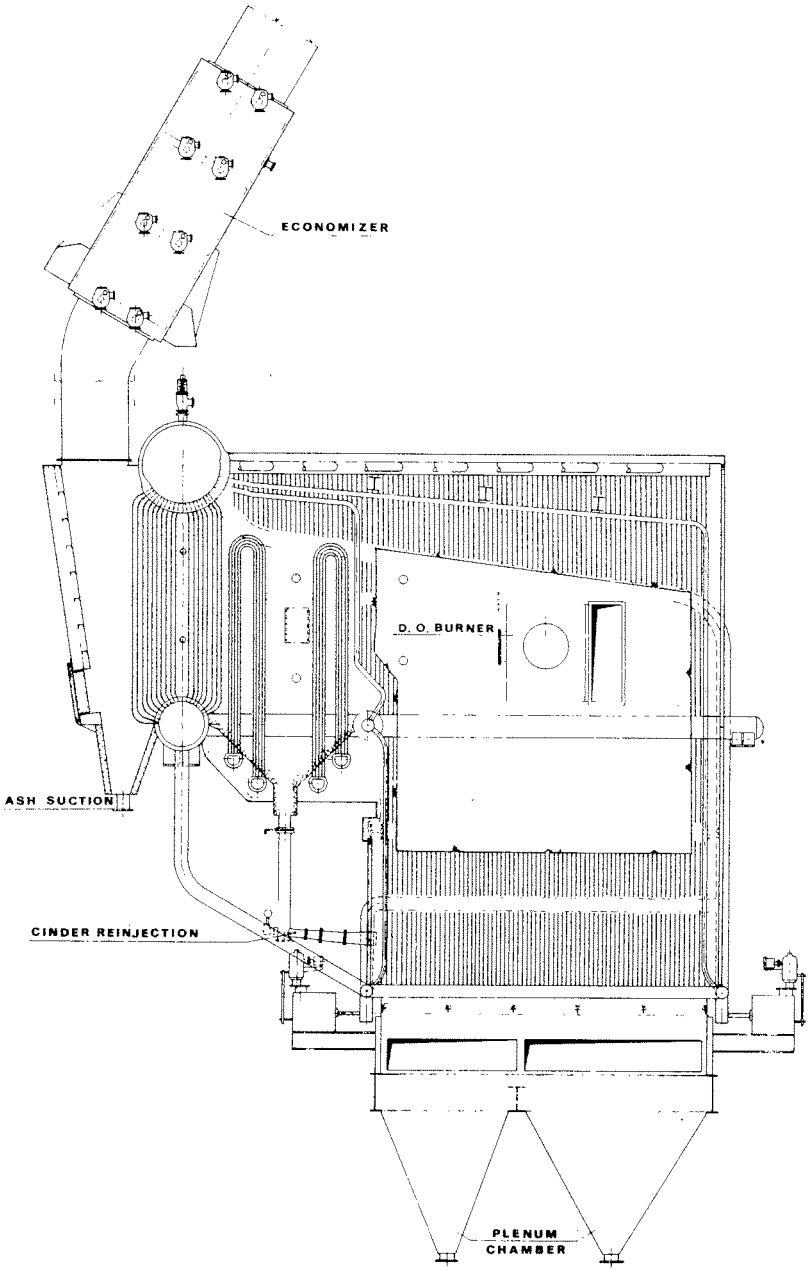
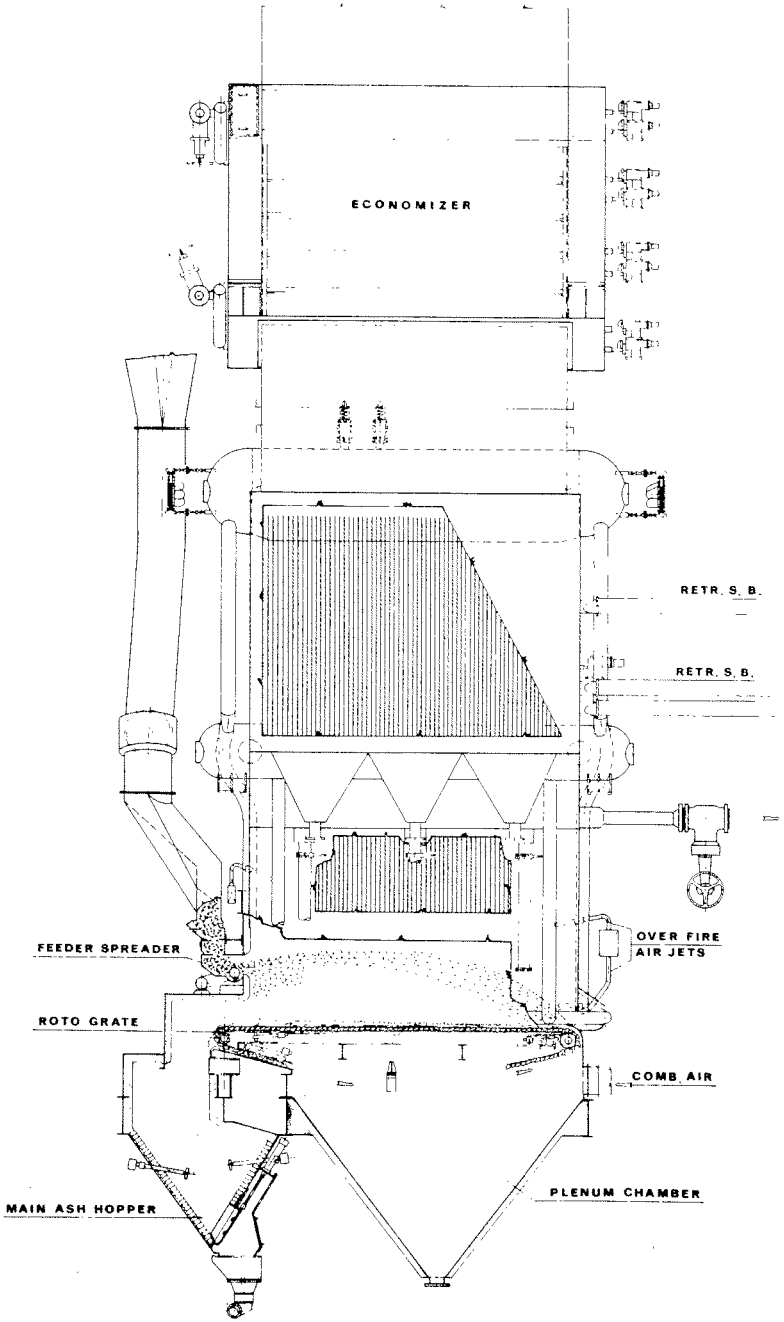
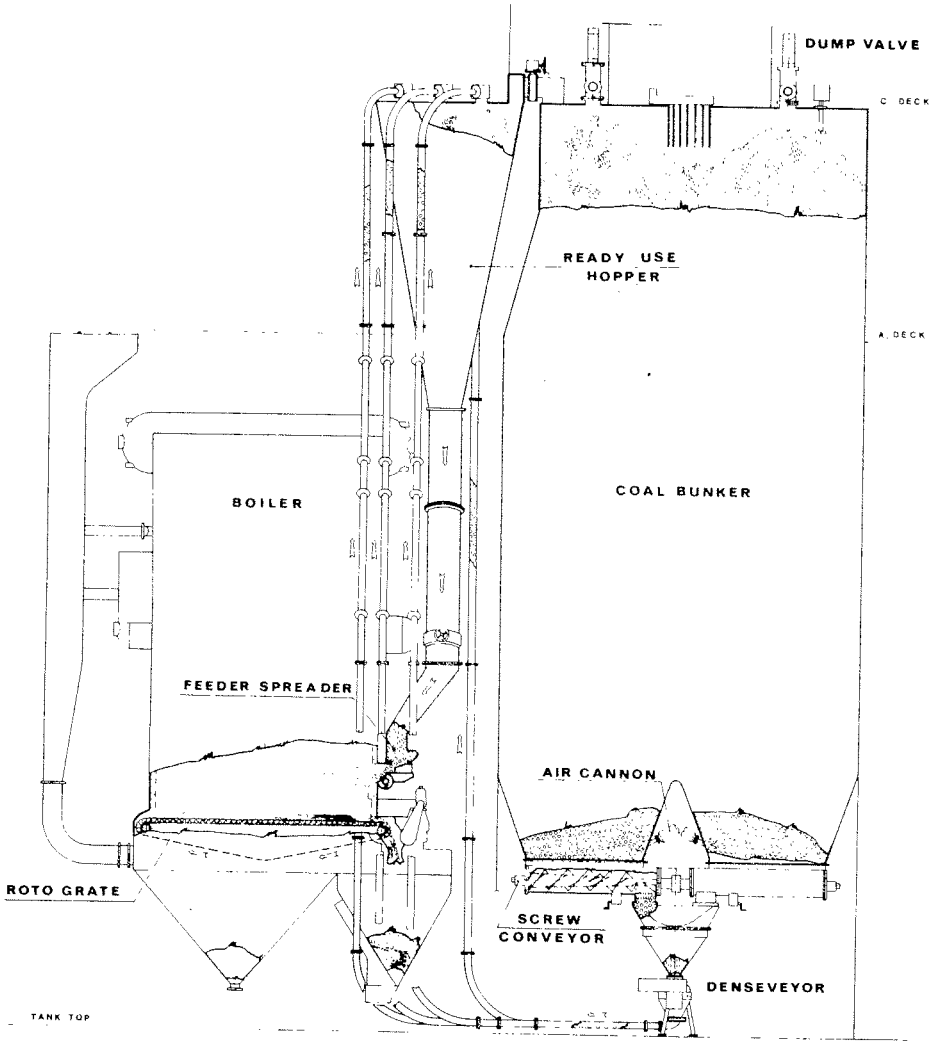


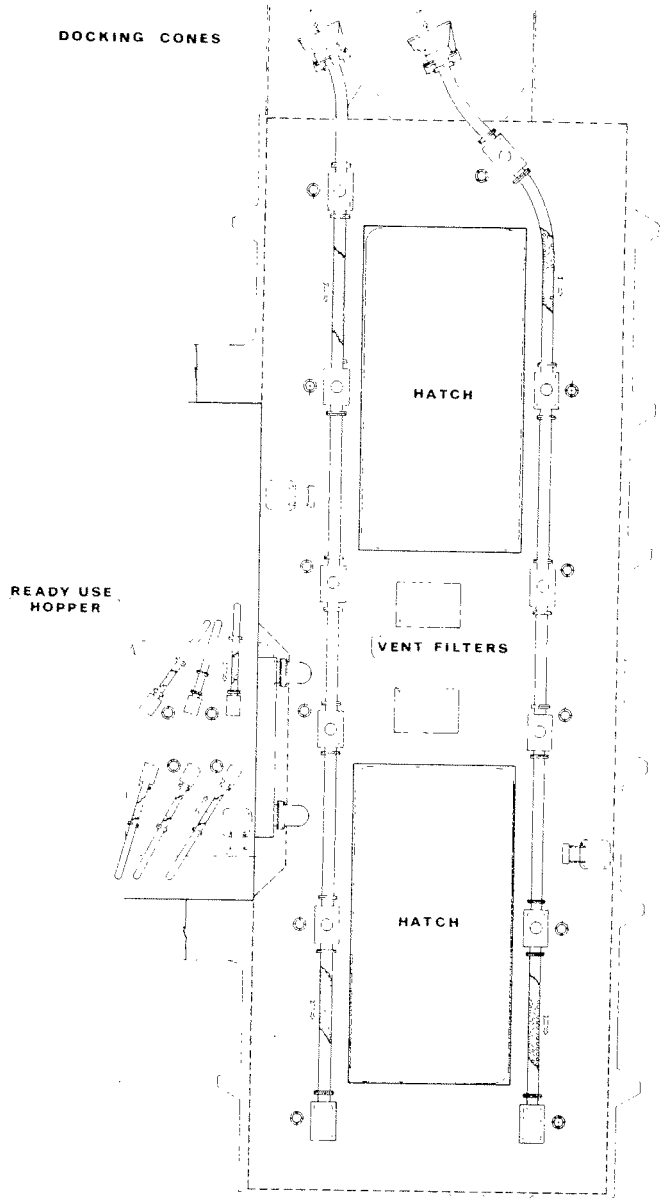
FIG. 5



PNEUMATIC COAL HANDLING SYSTEM



ELEVATION



PLAN 'C' DECK

FIG. 6

PNEUMATIC COAL HANDLING SYSTEM

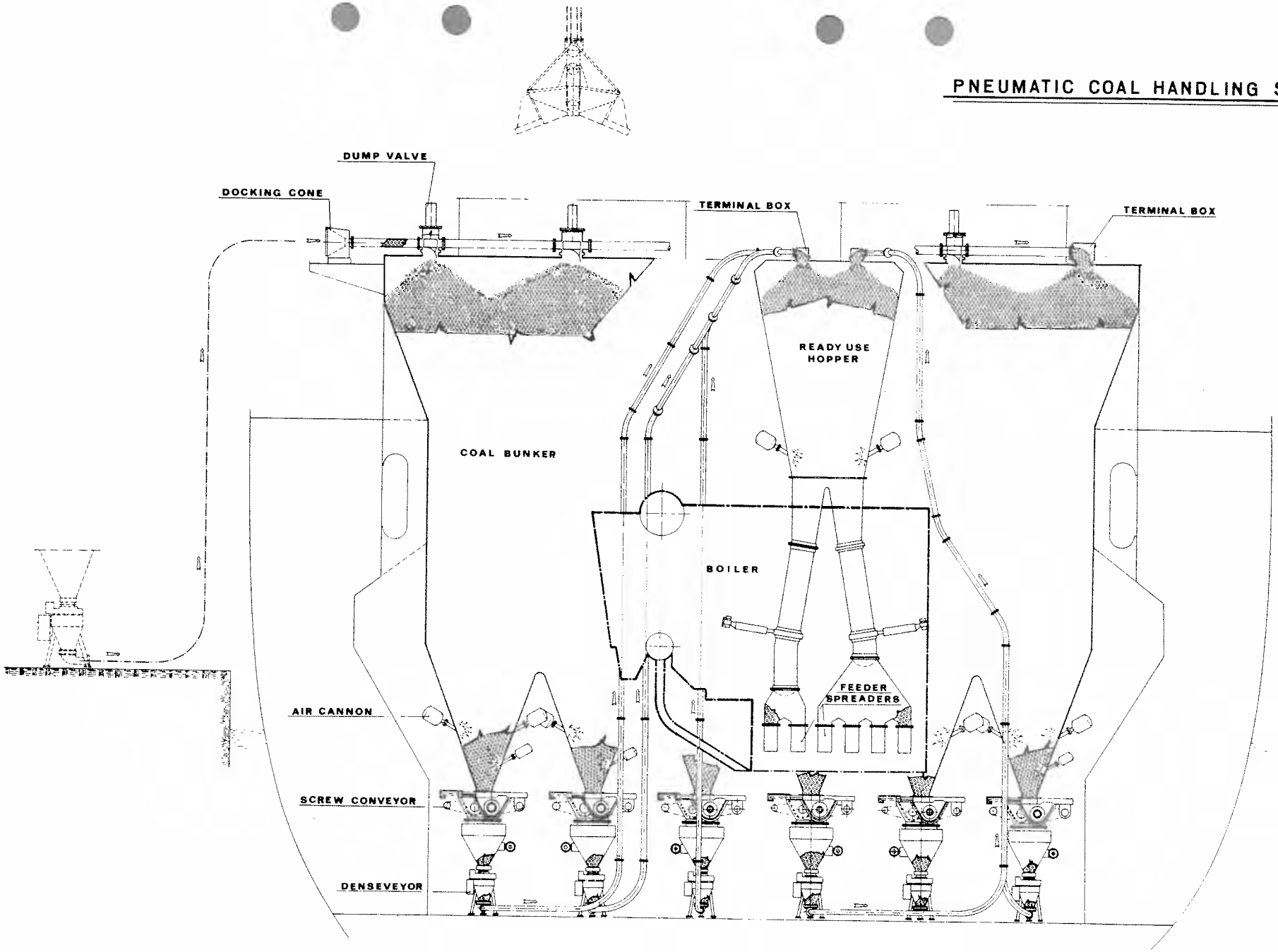


FIG.7

ASH HANDLING SYSTEM

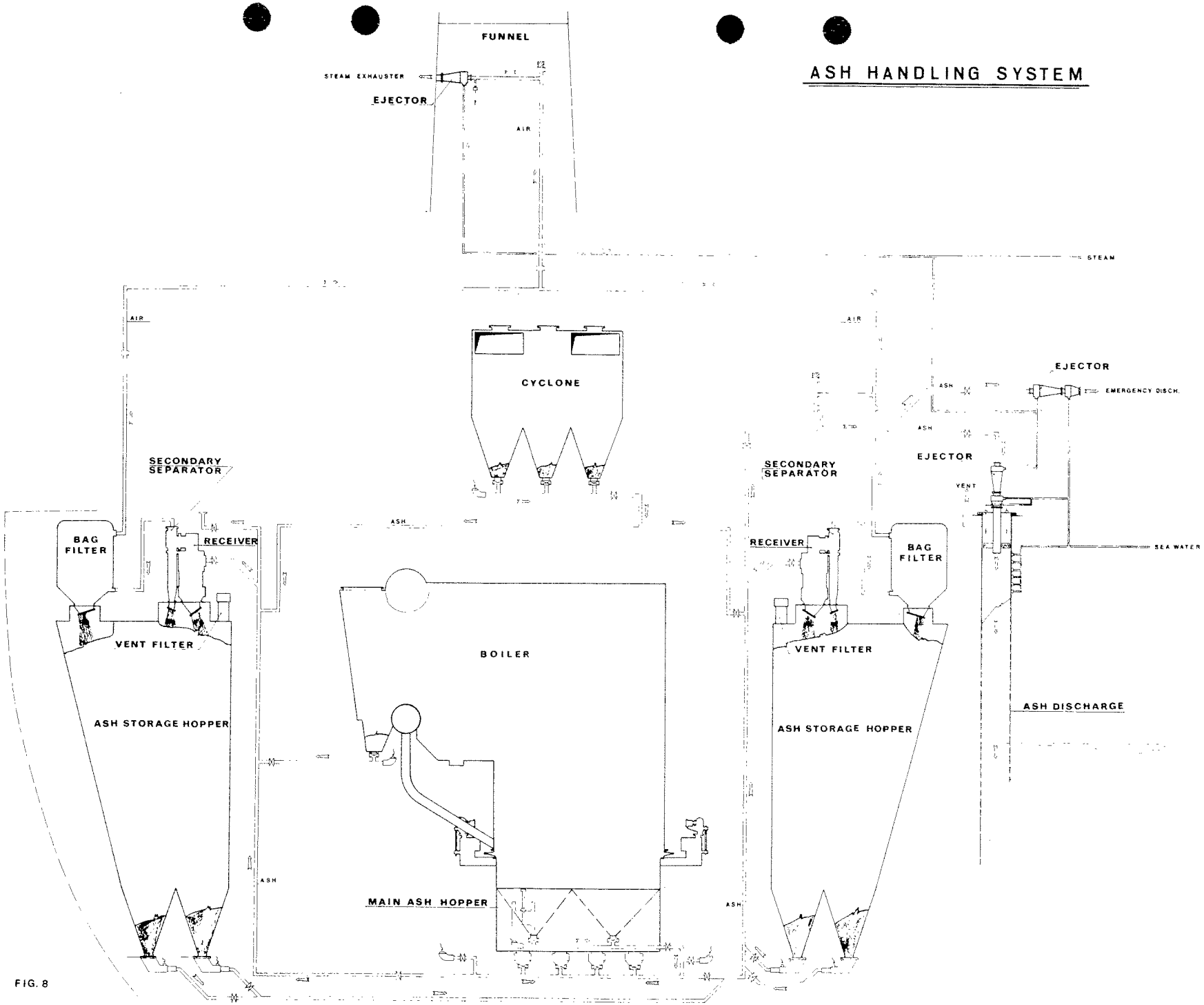


FIG. 8

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

MAINTENANCE & REPAIR; SHIP'S MANNING

Fuel-saving measures on ships operating with reduced manning

by

Henry Glimberg

Broström Shipping Company Ltd , Gothenburg

MAINTENANCE & REPAIR; SHIP'S MANNING

FUEL-SAVING MEASURES ON SHIPS OPERATING WITH REDUCED MANNING

In 1977, Broström Shipping Co ordered a series of six ro/ro vessels. At that time, the high manning cost under Swedish flag was our main concern. An agreement was reached with the Union to operate these ships with a crew of 16 instead of the normal 23-25.

Soon after delivery, fuel prices doubled and fuel savings once again became our highest priority.

This paper describes the maintenance procedures developed as a consequence of the reduced manning, the fuel savings measures which have been introduced and actual operating experience.

Description of the vessels

The vessels have an overall length of 165 metres, a breadth of 25.5 metres and a deadweight of about 12,000 tons at a draught of 8 metres. Container capacity of 800 TEU's. The main propulsion is two long-stroke diesel engines with a maximum power of 21,000 BHP and trial speed is 21 knots. Three ships were delivered in 1978 and three in 1979.

Manning

During the third quarter of 1977, an agreement was reached with Unions and Swedish Board of Shipping and Navigation on the following manning:

Master	1
Officers	3
Radio Operator	1
Chief Eng	1
Engineers	2
GP ratings	6
Cook/Steward	1
Messman	1
Total	16 men

The first manning proposal put forward in the negotiations was more radical. For instance, it did not have a Radio Operator.

Workload calculations

The abovementioned crew is intended to carry out all the duties involved in the operation of a ship including the maintenance needed for a safe voyage and supervision of cargo handling in ports. It was, however, understood that the 16 man crew was not dimensioned to handle all the maintenance and repair work.

To clarify the man-hours needed for operation and maintenance & repair, a detailed workload calculation was made. The calculation was done by a Master and a Chief Engineer and then distributed for discussions to the superintendents and the officers intended to man the newbuildings.

Maintenance intervals and man-hours were based partly on the manufacturers' recommendations but mainly on our own experience in achieving good reliability. The man-hours were calculated in detail for each component and type of work. One page of the calculation is translated and enclosed for your reference (Encl 1).

The result of the workload calculations was that about 31,600 man-hours/year were needed for operation and about 18,800 man-hours/year for repair & maintenance. These figures represent an average during a five year period. The total available man-hours for the 16 men were estimated at about 44,800 hours/year.

Evalution of different maintenance solutions

About 1,900 of the 18,800 man-hours for maintenance were considered suitable for specialist firms.

The remaining man-hours could be solved either by reinforcing the crew or by using shorebased workshops. These alternatives were evaluated for each type of work, from an economical and practical point of view.

Five year maintenance contracts were quoted by the different manufacturers for all major equipment. A contract for maintenance of the complete ship was also quoted by the building yard.

Each quotation was evaluated against having the maintenance done in the conventional way (i.e. crew and specialist firms and own purchase of spare parts).

With the help of the workload calculations and our statistics on spare part costs, it was thereby possible to find the optimum combination.

The conclusion was that a maintenance contract for the main engines was the best economical solution, while maintenance of other equipment should in principle be carried out by the crew.

A maintenance contract for the main engines was signed with Götaverken Motor Co, which covers classification work, all maintenance and repair work and the supply of all necessary spare parts for a fixed price during a five year period. A paper published by Götaverken Motor in March 1980 is enclosed (Encl 2).

What type of equipment was installed as a result of the reduced manning?

Equipment installed as a direct result of the manning reduction was fairly limited. Too much and too sophisticated equipment may instead give a higher manning level as the man-hours required for maintenance will increase and specialist knowledge not normally available among the crew may be required. It is rather a question of avoiding peak loads and stress situations. Examples are: a good mooring arrangement with remote controls; a microcomputer-based alarm and monitoring system which gives full information in clear reading on TV-screens on the bridge and in Chief engineer's office and a good internal communication system. Automation equipment should in principle be designed so that the engineers can operate and repair it without a specialist (el.engineer) in the crew.

It is, of course, also important to have reliable equipment and a high quality paint system.

Maintenance principles

With a reduced crew, attention has to be paid to finding systems with minimum paperwork and much flexibility, in order to allow the men on board to adjust it according to actual experience.

Some of our guide lines are given below.

* The maintenance planning should be done on board.
* Planning should be based on a combination of running hours and condition monitoring and revised according to actual experience.
* Sufficient man-hours shall be available to allow for preventive maintenance.
* Priority is given to equipment which can cause offhire in case of failure.
* Touch up of paint damages to be done at earliest convenience but general overpainting (cosmetics) should be avoided.
* Only failures (or deviations from normal operation) are reported to head office.
* Annual budget to be made up in close cooperation between senior officers and superintendent.
* Copies of all bills to be sent to the ship.
* Actual budget results to be sent each month to the ship.

Maintenance planning system

Based on these guide lines a new maintenance planning system was developed.

It consists of planning boards with follow-up cards covering a five year rolling period for all the major work plus a routine list for work with shorter intervals than three months. A short description of the system is enclosed (Encl 3).

Operating experiences

The whole idea with a reduced crew is of course to improve the economic result. It is no secret that a lot of scepticism regarding the economics existed in the beginning among ourselves in the Technical Department as well as among the officers at sea. During the theoretical economic evaluation of different maintenance solutions it however became clear that the savings in personnel costs will be larger than assumed extra costs for maintenance and repair.

The ships have now been in operation for 4-5 years and the actual result is rather surprising.

With a reduced crew it was natural to assume that other costs, and especially the maintenance and repair costs, would be higher. However, actual results show that not only the personnel costs have been reduced but other costs as well.

The maintenance and repair costs for the six vessels show the following development (average per ship and year).

	Calculated Project stage 1977 level	**Actual**			
		1979	**1980**	**1981**	1982
MUS$	0.360	0.130	0.225	0.305	0.270

This positive result can be explained as follows:

* Shortage of manpower makes it necessary to establish priorities such as avoiding unnecessary work; more condition-monitoring based maintenance; simplifying routines; minimizing paperwork; quick feedback to head office if equipment or systems need to be modified to decrease workload etc.

* Every man is important in a small crew and this improves the working morale as well as the general quality of the crew.
* The detailed calculation of man-hours was worked out and discussed by the senior officers. This resulted in good pre-planning, where everybody knew the expected workload.
* The maintenance contract for the main engine removes the very high peak load that will arise if the maintenance of the main engine is carried out by the crew during the short port calls. With a specialist team from the contractor, who knows the ship, the work will be done with minimum disturbance to the crew. The crew can then work at a normal speed on the planned maintenance of all other equipment, which has resulted in an efficiency per man-hour that is considerably higher than originally expected.
* The quality of the ship and its equipment has also proved to be higher than expected.

FUEL-SAVING MEASURES

Fuel costs doubled shortly after delivery of the ships, which is why several measures have been taken to reduce fuel costs, such as:

Blended oil system for aux. engines

The four aux. engines were originally designed for diesel oil. In the autumn of 1979, prices in Rotterdam were 320 US$/ton for diesel oil. It was then decided to modify the aux. engines to 800 Red Sec oil and install a blending unit. Estimation at that time was that we could at least run the modified engines with 200 Red Sec. The saving would then be about 150,000 US$/year and ship. Calculated modification cost was 110,000 US$/ship.

The actual result was that the estimated modification costs were correct, but it was in fact possible to use a 30/70 DO/FO-ratio instead of assumed 50/50-ratio. On the other hand, price difference DO to FO has narrowed, which is why calculated savings in US$ are close to the original calculation.

Fuel oil to boiler

Also, the thermal oil boiler were originally designed for diesel oil. The burner and fuel system were modified to fuel oil (3.500 Red Sec) at a cost of about 18,000 US$, which corresponded roughly to the yearly saving.

Modifications to the main engines

Three of the ships now have a new type of fuel pump, which gives max. cylinder pressure from 70% to 100% load. The savings are about 2-3% in fuel and the installation cost is about 60,000 US$/ship.

The main engines of the remaining three ships have been derated, which gives about 1.5-2% savings but limits the max. power to 75%. Installation costs are about 10,000 US$/ship.

B&W have also recently introduced new fuel nozzles, which allow the main engines to operate continuously between 35% and 100% load. We have tested these for some months on one ship and are now installing these slow-speed nozzles on all the ships.

Energy-saving programme

A comprehensive energy-saving programme was introduced in 1979. The following examples can be mentioned:

* Aux. engines should run between 70% and 90% load when fuel consumption is about 230g/kWh. 25% load gives 320g/kWh i.e. an increase of about 40%!

* Avoid unnecessary lighting. One chief engineer reported that it costed 3,667 SEK per year to have an "enlightened chief" (i.e. the cost of having ceiling lamps on, the whole year in his bedroom, dayroom, hall and bathroom). An other example is only to have emergency lighting on in the engine room when unmanned.

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* Avoid unnecessary running of cargo ventilating fans. If the cargo fans on these ships are running continuously with full power, the actual cost per year and ship is more than one million SEK.

Others

* Improved cargo handling has allowed the mean speed to be lowered by about 1 knot.
* Instructions to use a mean speed between ports without margins.
* Instrument on bridge showing kg fuel/naut mile.
* Comprehensive model tests carried out in order to find optimum trim curves.
* Installed satellite navigation which, especially in the Mediterranean area, improves ability to keep a correct mean speed.
* Installed adaptive autopilot.
* Instructions to minimize ballast.
* Hull roughness measurement/improved A/F-paint.
* Two speed pumps installed for port use.

WHAT ABOUT THE FUTURE?

Although fuel costs are constant for the time being, the development of new fuel-efficient engine has been dramatic. Also the paint manufacturers, have made tremendous progress with the improved self-polishing paints. A lot of research has been spent on finding energy-saving hull forms. New types of slow-running propellers are developing, etc, etc. All this is already here, which is why we will see a lot of very fuel-efficient ships coming out from the shipyards in the next few years.

Do not be surprised if some of these ships can be operated safely and economically with a crew of 11.

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COMPONENT	SPECIFICATION OF WORK	RUN HRS YEAR	INTER VALL BETW OVER-HAULING	NUMBER OF UNITS	MANHOURS OVER HAULING	MANHOURS CONDITION CHECK	MANHOURS AVERAGE TOTAL PER YEAR	MUST BE DONE IN PORT	CAN BE DONE AT SEA	SUIT FOR
TRANSPORT							1.926			
CROSS HEAD BEARING	OPEN UP FOR LLOYD'S SURVEY	5.250	20.000	24	10		63	63		W/Y
CROSS HEAD BEARING	CHECKING OF CLEARANCES CROSS HEAD BEARING/GUIDE	5.250	3.000	24		0,5	21	21		OB
CROSS HEAD BEARING	RENEWAL LOW HALFES	5.250	40.000	24	30		95	95		W/Y
CRANK SHAFTS	CRANK SHAFT DEFLECTION	5.250	3.000	24		0,5	22	22		OB
MAIN BEARINGS	OPEN UP FOR LLOYD'S SURVEY	5.250	20.000	26	16	0,25	109	109		W/Y
MAIN BEARINGS	CHECKING OF CLEARANCES	5.250	3.000	26			11	11		OB
THRUST BEARING	OPEN UP FOR LLOYD'S SURVEY	5.250	20.000	2	20		10	10		WY
TURNING GEAR	INSPECTION	5.250	12.000	2		4	4	4		OB
CRANK CASES	INSP/CONTR OF BOLT TIGHTNESS	5.250	2.000	24		1	63	63		W/Y
SAFETY VALVES C.C.	FUNCTION CONTROL	5.250	6.000	24		0,25	5	5		OB
SCAVANGING BELTS	CLEANING OF BELTS & GATES	5.250	3.000	24	2		84	84		W/Y
AIR COOLERS	CLEANING (CIRC W PUMP)	5.250	2.000	8	4		84	84		OB
AIR COOLERS	OVERHAULING FOR CHANGING COOLER NEST & LLOYD'S SURVEY	5.250	20.000	8	10		21	21		W/Y
TURBO CHARGER	DISMANTL/CLEANING/SURVEY	5.250	12.000	8	20		70	70		S
TURBO CHARGER	RENEWAL OF BEARING AND PUMPS	5.250	12.000	8	4		14	14		S
TURBO CHARGER	CLEANING AIR/GAS SIDES & FILTERS	5.250	2.000	8	4		84	24	60	OB
							2.686			

W/Y = WORKSHOP/YARD
OB = ONBOARD
S = SPECIALIST FIRM

Encl 1

Encl 2

ENGINE MAINTENANCE CONTRACTS

INTRODUCTION OF GÖTAVERKEN MOTOR AB

Products

Götaverken Motor AB is one of the companies in the Swedyard Group. The company has emerged from the Diesel Department of the previous Götaverken. The traditional product is diesel engines for ship propulsion where the company has been active since 1915. The company had its own crosshead diesel design from the 1930's to the early 1970's. Nowadays Götaverken Motor AB is a licensee of B&W. The company manufactures the whole range of B&W 2-stroke crosshead diesel engines for propulsion as well as stationary power plant applications. In connection with this the company also has a very substantial after sales business supplying spares and technical service not only to engines built by the company but also to all Eriksberg/B&W engines, where a substantial number of units are still in operation. The latter were added to the line following the merger of Eriksberg into the Götaverken Group when the Machinery Department was marged with Götaverken Motor AB.

In the later years Götaverken Motor AB has launched several new products in an effort to deversify its operations away from the heavy dependence on marine engines. In addition to the traditional products the company now produces process components such as heat exchangers (a previous product, now revived and developed further), machine tools under licence and even a line of pipe fitting based on a revolutionary Swedish invention.

Götaverken Motor AB employs about 850 people and has a yearly turnover of about 300 million Swedish Kronor.

After sales capabilities

This paper will however deal with main engine maintenance under contract, which is one of our after sales activities. It may be essential here to point out that the after sales activities and capabilities of Götaverken Motor AB are considerably bigger than that of a normal licensee-engine builder. This is due to the fact that we had our own engine design for a number of decades, close to 700 of which are still in service. The merger with Eriksberg added more than 250 units to our market and additionally we have the Götaverken/B&W engines built during the later half of the 1970's. The basis of about 1000 engines constitute a market that is big enough to support the costs of a genuinly active spare part marketing effort and a sizeable technical service department. We have about 35 service engineers available in Gothenburg, specializing in different technical fields, and there is a quite heavy demand for their services.

Additionally Götaverken Motor AB has its own subsidiary company in Singapore. Operations within this service company are managed by two Swedish service engineers and the local personnel comprises about 20 people. The company has a modern workshop and an extensive stock of engine spares.

Maintenance Contract efforts

During the first half of the 1970's the Götaverken Group marketed a complete operation and maintenance package for shipowners. This program was primarily intended to assist the many new non-traditional shipowners entering the market before the oil crisis. This quite ambitious program did however not yield any direct results in the form of maintenance contracts.

MAIN ENGINE MAINTENANCE AGREEMENT WITH BROSTRÖMS REDERI AB

Origin

When planning its new vessels for the Mediterranean and North Sea trades the Broström Rederi AB aimed at considerable reductions in manning of the vessels. Thus there would be enough crew to operate the vessels but much of the maintenance works normally performed by the crew would have to be done in harbour by personnel other than the crew. This has been compared with the operating principles used by airlines.

Since the vessels were first discussed with yards within the Swedyard Group it was only natural that talks started between Broströms and Götaverken Motor AB at an early stage regarding the possibilities of establishing maintenance support from Götaverken Motor AB on the basis of fairly fixed commitments. Even after it was decided that the vessels were to be built in Japan we felt at Götaverken Motor AB that the business was still worthwhile going after, since the vessels were to be equipped with engines of the B&W design. This led to the signing in 1978 of maintenance contract for 6 RoRo-vessels on the Scandinavian-Mediterranean trade and one bigger RoRo-vessel sailing for the Barber Blue Sea Line. This paper deals mainly with the Mediterranean vessels.

The Mediterranean vessels are equipped with two Mitsui/B&W 12L45GF two-stroke crosshead engines and the Barber Blue Sea Line vessel with one Mitsui/B&W 9K90GF engine.

Scope of supply

The contracts cover maintenance, repair and spares supply for the main engines including such equipment as turbo chargers and piping on the engines but excluding all auxiliary equipment. Under the contract we have committed ourselves to a number of services and product deliveries:

- Establishing of a maintenance planning and scheduling file comprising all units and parts of the engines subject to maintenance.

- Visits onboard at sea by one of our service engineers 4 times a year, mainly for condition check of the engines, but also as a part of the maintenance planning.

- Performance of all maintenance work, scheduled overhauls as well as repair work. All maintenance work should preferrably be performed in what is defined as the base harbour.For the vessels on the Mediterranean trade the base harbour is Gothenburg and for the Barber Blue Sea Line vessel it is Singapore. Maintenance work shall however also be performed outside the base harbour when necessary and under certain conditions.

- As far as engine spares are concerned the vessels are equipped with a stock of initial spares, this stock being agreed between Broströms and ourselves. Spares from this stock are used continuously during overhaul and repair work and after using the spare part we shall soonest possible replace it by a new or reconditioned part. Consequently at the expiration of the maintenance agreement the spare part stock will be exactly the same as when the vessel was brand-new.

Term

There is one maintenance agreement signed for each vessel and they are all valid for a period of 5 years from the delivery of the vessel. The possibility of a prolongation is foreseen but the conditions for such a possible prolongation are not settled. The parties recognize that the experience gained during the term of the contract will determine conditions for prolongation.

Treatment of uncertainties

With the maintenance contract being a new feature both parties face several uncertainties which may influence the economical result of the contracts. Some efforts have been spent by the parties when drafting the contracts in order to reasonably well balance those uncertainties between them:

One basic task has been to define in a satisfactory manner the standard of maintenance work and the level of workmanship to be expected. This has been taken care of in general terms by referring to the maintenance standard that can be expected from a well renowned shipowning company and more specificly by placing a ceiling on the number of involuntary engine stops at sea that may occur each year.

Estimating beforehand the total amounts for work to be performed and spare parts to be consumed during such long period as 5 years has of course represented considerable uncertainty to us as the contractor.

Generally we have based our assumptions on past experience regarding specific costs per hp and year for such activities. From this experience data we have however had to deduct an anticipated increased efficiency that we estimate to realize with this method of performing the maintenance work and some allowance for the expected generally lower maintenance costs with these very modern engines. Price for our total commitment under the maintenance contract is fixed in the price level of the year when the contract became effective. Like for all contracts extending over several years the inflation represents a considerable uncertainty. Therefore it has been necessary to use a price revision clause, whereby the yearly costs are changed with the changes in appropriate Swedish official labour and material cost indexes.

Performance of maintenance work outside the base harbour represents extra costs, the extent of which is quite difficult to estimate. In order for the maintenance contract to be of a genuine value to the shipowner he must be able to have maintenance as required performed anyway along the route of the vessel. On the other hand, if this facility was completely free of extra charges there would not be an incentive on the part of the shipowner to minimize such assistance. In these contracts this has been solved by our supplying personnel anywhere along the route without extra costs for the labour time including travel time whereas Broströms reimburses us for travelling and living expenses incurred on such occasions.

A major uncertainty is the risk that sometime during the term of the agreement a major break down should occur involving very substantial costs for repair work and replacement spares. This situation with a small risk of a high cost is a typical insurance situation. Since Götaverken Motor AB is by no means an insurance company, this situation has been dealt with by placing a ceiling on the costs being involved in any one specific incident. This agreement has been based on a principle of cover for costs beyond the ceiling through the shipowner's insurance policy.

Reporting

To facilitate planning and monitoring of the maintenance work performed the contracts stipulate mutual regular reports. We obtain reports on operational data from the vessels, oils used etc. We issue reports comprising our maintenance work plans for the immediate future and of work performed .

THE MAINTENANCE CONTRACT AT WORK

Operating experience

Of the six vessels on the Mediterranean trade three were delivered in late 1978 and the remaining three in the fall of 1979. The big vessel served in Singapore also began it service in 1979.

The amount of experience in serving all six vessels in Gothenburg is thus still limited. The first three have however recorded about 8.000 running hours for their engines. The approximate yearly engine running time is 6.000 hours.

Sailing schedule

The vessels on the Mediterranean trade have a round trip time of about 1 month. Usually they call at Gothenburg twice during each around trip, both on their way from the Mediterranean and when outbound after having completed the Baltic portion of their route. This provides for a total of approximately 10 calls a month in Gothenburg.

Availability time

Like most RoRo-ships, these vessels spend only a short time in harbour. The berth time in Gothenburg is usually from 06.00 in the morning until 20.00 or 22.00 in the evening. This means the availability for our service personnel to perform their maintenance work is approximately one working day. Quite often this day has happened to be a Saturday or Sunday.

Maintenance planning

With such a short time of availability it is essential that overhauls and other scheduled maintenance is reasonably evenspread over the year. This schedule has been made up as a normal routine within the framework of the planning manual forming part of our contract. The four times a year visits onboard by our service engineer also provide part of the maintenance planning based on condition checks of the engine and on discussions with the chief engineer. Usually the service engineer joins the vessel on one of the coastal legs within Scandinavia. A final schedule of maintenance to be performed during a particular call at Gothenburg is determined about a week beforehand and communicated to the shipowner.

Workforce

For the managing and general supervision of our work under the maintenance contracts we have set aside two service engineers. By more or less dividing the number of vessels between them they gain a very good knowledge of the individual engines.

The direct performance of the maintenance work is handled by a number of fitters from the Erection and Assembling Department of our engine manufacturing plant. Usually a team of 1 foreman and about 6 fitters is used. The exact composition of this group varies over time but there is a nucleus developing that participates on most occasions.

Between their days working for the maintenace contract, this workforce is used in our regular engine production. This is of course one of the key factors why a maintenance organized along these lines can be an attractive alternative in economical terms to the conventional maintenance by the ship's crew.

Component overhaul

In order to meet the requirements from the short availability times, component overhaul is performed on a rotation basis. Complete subassemblies such as exhaust valves are removed and replaced by exchange assemblies. The removed items are brought to our workhop where they are stripped and the components such as exhaust valve spindles and seats are overhauled or reconditioned. In order to keep the number of exchange assemblies within reasonable limits only of couple of subassemblies or items are removed at a time.

Overhaul intervals

One of the basic ideas behind the maintenance contract concept is that the complete responsibility for both maintenance work and spare part supply will provide us as the contractor with a very good insentive to really try and find the optimum by minimizing the total cost of work and spares. Consequently the contracts do not stipulate any fixed overhaul intervals. Starting out from the recommendations of the engine designer we have been able to stretch the overhaul intervals based on service experience and continuous monitoring of the state of the engines. Although these vessels are not equipped with any sophisticated eletronic equipment for automatic condition check and forecasting of maintenance requirements, we think the experience gained by our personnel will quite adequately compensate for this. We will however need a longer time of operating the maintenance contracts before we can make an accurate assessment of how much can really be gained through the efforts of optimizing the total maintenance costs.

Maintenance outside base harbour

As was foreseen in the contract, maintenance has been required outside the base harbour of Gothenburg a number of times. This has been handled without problems and we have been able to meet the requirements of Broströms.

Repair works

So far no major repair work has been required. A number of mostly minor guarantee items have appeared during the guarantee year. On behalf of Mitsui, our people have performed the guarantee repair on most of those occasions.

BRIGHT FUTURE PROSPECTS

Experience summary

With the natural reservation that 18 months is after all a limited period we can certainly state that our experience of this type of contracted main engine maintenance is quite positive. We have generally been able to work within the frames of our pre-contract estimates and the prospects of achieving cost reductions through further optimizing and rationalization appear to be there.

As far as we know our customer, Broströms, is also satisfied with the way we perform under the contracts. One indication of this fact is that Broströms have shown a considerable interest in obtaining similar contracts for other vessels which are already in service. Discussions are in progress to determine the conditions for such additional contracts.

Ferry engine maintenance

For the new Rederiaktiebolaget Gotland ferries now being built by Öresundsvarvet we have signed analogue maintenance contracts with that shipowner. Those ferries are equipped with 4 engines of type 8K45GUC which are geared together two by two. Those engines are manufactured by Götaverken Motor AB. To perform maintenance under that contract we will have to send our personnel across to the Baltic coast of Sweden and consequently we are planning the maintenance in such a way as to require less frequent service and maintenance work. Activities under those contracts will start this summer when the first ferry enters service.

Flexibility

We have a number of inquiries for maintenance contracts similar to what has been described here or variations thereof. The spectrum ranges from maintenance and overhaul of one single type of component to complete maintenance supervision of diesel power plants. In our marketing efforts for diesel power plants we have found that the ability to offer sustained maintenance or maintenance support and supervision during a number of years after delivery of the plant will be of essential importance to the operator.

The exact scope of supply of services and products under any one contract can be tailored with a good deal of flexibility to meet the individual needs of any particular customer. Of course we have to investigate our total risk exposure carefully for different possible contract designs. For instance we could not make a contract for a fixed price yearly spare part supply unless we have some influence on how the engines are operated and maintained.

Additionally we will be able to include the mechanical auxiliary equipment in the engine room under a maintenance contract.

Advantage summary

Obviously maintenance under contract could not be feasable for every vessel,trade,shipowner or type of machinery. However, the concept has such obvious inherent advantages and can be applied with such a flexibility that we are convinced it will have a bright future. Some of the advantages to the shipowner of having this type of maintenance agreement can be summarized as follows:

The shipowner is relieved from the administrative task of planning maintenance works and spare part supply and procurement.

The maintenance agreement provides the shipowner with an almost completely known cost for his maintenance during the term of the contract.

The shipowner benefits from continuously utilizing the very experienced and specialized personnel as well as the machining capability of a well known engine manufacturer.

Most of all, the shipowner will find that maintenance can be performed at a competitive low cost to him. The confidence in our ability to offer competitive prices is based mainly on three factors:

- The general experience and skill of our personnel and an efficient spare part administration.
- No idling time since the personnel is occupied in the regular production in between maintenance work periods.
- The continuous efforts to achieve the minimum for all maintenance costs through an optimum balancing of overhaul work and spare part consumption.

SHORT DESCRIPTION OF MAINTENANCE PLANNING SYSTEM

The system consists of planning boards for preventive maintenance and classification work; follow-up cards for recording work carried out and instruction/note-cards for special work procedures and notes; routine list for maintenance with shorter interval than three months and deviation report for reporting to head office.

Planning boards (page 3)

Preventive maintenance and classification work are planned on "Modulex-boards" covering a classification period of 5 years.

Classworks are marked in red; overhauling with in yellow and work that requires docking in blue.

When the work is completed the red/yellow or blue marking is covered with a green marking.

When a failure is noted on a component a black mark is used.

Follow up cards (page 4)

Each component has a follow-up card (white) which is kept in a pocket at the left side of the planning board. The cards are used for short descriptions of work carried out and recording of any failures.

Instruction/note cards (page 5)

These cards (yellow) are used when special attention or instructions are needed for a certain component, and they are placed behind the follow-up card for the component in question.

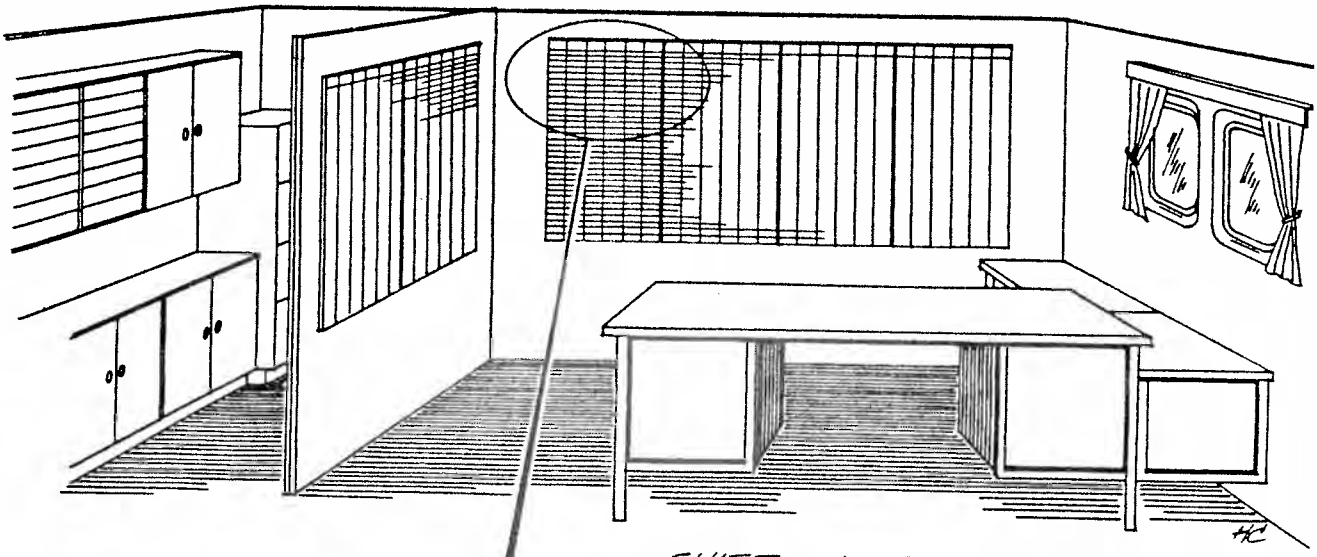
Routine list

Maintenance and checks which are carried out at shorter intervals than 12 weeks are planned in the routine list.

Deviation reports

Only failures are reported to head office. These reports are stored in a common database which is used for comparing different designs and makes when ordering new ships or components.

MAINTENANCE PLANNING BOARDS



CHIEF ENG. OFFICE

FOLLOW UP CARDS

RED □ = CLASSIFICATION WORK
YELLOW □ = OVERHAULING WORK
BLUE □ = DOCKING WORK
BLACK □ = FAILURES
GREEN □ = WORK COMPLETED

	1980				1981				1982				1983				1984			
	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4
CYLIND. 1 STB ME				□						□						□				
" 2 "								□						□						□
" 3 "								□						□						□
" 4 "						□						□						□		
" 5 "									□						□		□			
" 6 "					□						□									
" 7 "										□						□				
" 8 "							□						□						□	
" 9 "									□						□					
" 10 "									□						□					
" 11 "								□						□						□
" 12 "							□							□					□	
CYLIND. 1 PORT ME					□		□				□		□				□		□	
" 2 "						□						□						□		
" 3 "									□						□					
" 4 "						□						□						□		
EXHAUST V. 1 STB ME				□			□			□			□			□			□	
" 2 "					□			□			□			□			□			□
" 3 "						□			□			□			□			□		
" 4 "							□			□			□			□			□	
" 5 "							□		□			□			□			□		
" 6 "					□			□			□			□			□			
"							□			□			□			□				
"							□			□			□							
"						□			□			□			□					

Maint. No. 7 1 1 0 0 0

Main cooling S.W. pump stb.

FOLLOW UP CARD

DATE	OPERATION TIME	WORK CARRIED OUT	REPORT.
81-03-14	1840	Changed mech. seal complete (type Crane 42 mm)	X
81-10-10	3760	Changed bottom ball-bearing and mech. seal compl. except spring.	X
		(THIS CARD = WHITE)	

Main cooling s.w. pump stb.

INSTRUCTION/NOTE CARD

1.) Use special tool when removing rotor.
Plac. aft of s.w. pump center.

2.) Shift allways packing 1186-1 when overhaul.

Pittings in pumphousing retauched with Devon carbid no. 5
1981.10.10

(THIS CARD = YELLOW)

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

SOME ASPECTS OF PAINT AND DOCKING

by

A MILNE

INTERNATIONAL PAINT

GATESHEAD

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12. Micrometer Measurement of Film Thickness.
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14. Cumulative Frequency Thickness Distribution of One Coat of Copolymer Antifouling.
15. Statistical Model of Two or Three coat application to Nominal 300 micron Total Dry Film Thickness.
16. Film Thickness Distribution of a Two Coat Self-Polishing Antifouling to a Products Carrier.

1. Introduction:

The maintenance of the outside hull of a vessel is ultimately an investment decision on the part of the owners, to be evaluated against many other possible investments, and the techno-economic analysis of ship performance requires accurate information of several kinds on the condition of the hull. The ideal underwater hull surface is one which starts, and remains, hydraulically smooth for the lifetime of the vessel. Such a surface would be easy to model economically since the only information required would be the first cost. The real situation is much more complex. The only property of the hull which is of economic importance is its smoothness, since the consequence of hydrodynamically significant roughness is an increase in frictional drag on the vessel, influencing either ship speed or ship power, or both. Information is required therefore on the development of roughness throughout the history of the vessel; the relationship between the roughness measured and the performance of the vessel is required, and this, in turn, requires that we be sufficiently knowledgeable about the relationship between roughness and drag.

The roughness itself can be considered of two types, the physical or geometrical 'hard' roughness of the surface and the fouling or biological roughness, due to the settlement and growth of organisms on the surface.

The growth of fouling on surfaces has a catastrophic effect on performance, and it is right that the emphasis should be on techniques by which the outside hull should be protected as completely and for as long a period as practical from fouling.

Physical roughness, however, is also a substantial component, in the long term, of ship performance, and the recognition that the whole process of drydocking, repair and maintenance of the outer hull is a major contributor to the development of roughness leads to the conclusion that an integrated approach to the problems of roughness, fouling and maintenance is required.

Much work on the techno-economic analysis of investment in maintenance, in the measurement of ship performance and in the analysis of roughness has been covered in considerable detail elsewhere and will be referred to in passing; the emphasis here is on the practical aspects of ship maintenance and the role they play in the outturn of the vessel.

Improved materials, particularly improved antifouling compositions, play an important role in keeping vessels out of dock longer, and apart from a greatly improved lifetime the self-polishing antifoulings have other important advantages.

Since the important property of the outside hull is smoothness, the contribution of cleaning, repair and maintenance techniques, methods of application, access etc. to the overall quality, is equally important, and the role of the importance of techniques will be stressed in the following discussion.

2. Roughness of Ships, Roughness Development and Analysis of Ship Roughness:

2.1 The Roughness Measure. How do we quantify the roughness of a ship?

The measure of roughness with which, to date, ship performance has been correlated, is the Average Hull Roughness (A.H.R.), based on a "mean-of-means" average of some 100 locations pseudo-randomly distributed, on an equal area basis, (Fig.1) over the hull of the vessel. At each location a string of 10-15 samples is taken, each sample representing 50mm of the hull surface, (Fig.2). For each sample the numerical value recorded is the apparent amplitude, that is, the change in height between the highest peak and the deepest trough within the 50mm sample, peak and trough not necessarily being adjacent.

The reasons for the choice of roughness measure were partly empirical and practical, and the necessity and sufficiency of the measure have been much re-analysed in recent years. The original choice was partly dictated by the fact that the original instrument was mechanical (B.S.R.A. Wall Gauge), and the analysis by eye and by hand were labour intensive, requiring the construction of an envelope round the roughness trace and sample-by-sample measurement. When the instrument was modernised (B.S.R.A. Hull Roughness Analyser), the same roughness standard, which had, by then, become general, was adopted. The questions which arise from the original choice are:

Is the amount of information collected sufficient to describe say, 35,000m^2 of ship surface?

Given that the choice of sample was dictated empirically, and filters out all information shorter than 2.5mm (1.6 x the size of the ball stylus), and greater than 50mm, (the sample length determined by the length of the skid), does this spectral window contain sufficient information about the hydrodynamically significant roughness?

Is a simple height parameter (h), which seems to ignore the spectral complexity (texture) of the profile, adequate?

Is a roughness measure, intended originally for the correlation of performance of new-building with speed and power, adequate at the roughness of older and much rougher vessels, and as a quality control instrument in drydock situations? Is the measure statistically valid compared with the more familiar and mathematically tractable centre-line-average (C.L.A.) and (R.M.S.) or root mean square measures?

The statistical validity was analysed by Chaplin (1) and more recently by Byrne (2) and Grigson (3), starting from the consideration of the roughness as a random stationary waveform as in Cartwright and Longnet-Higgins (4). The latter derived an expression for the distribution of maxima of such a waveform and found it to be a mixture of Rayleigh & Gaussian distributions the proportions depending on Longuet-Higgins parameter ϵ, which itself depends on the proportion of negative maxima (r). The relation between ϵ and r is

$1 - \epsilon^2 = (1 - 2r)^2$ and when r is small the p.d.f. (probability distribution function) is Rayleigh and the spectrum is narrow banded.

FIG. 2. THE ROUGHNESS SAMPLE

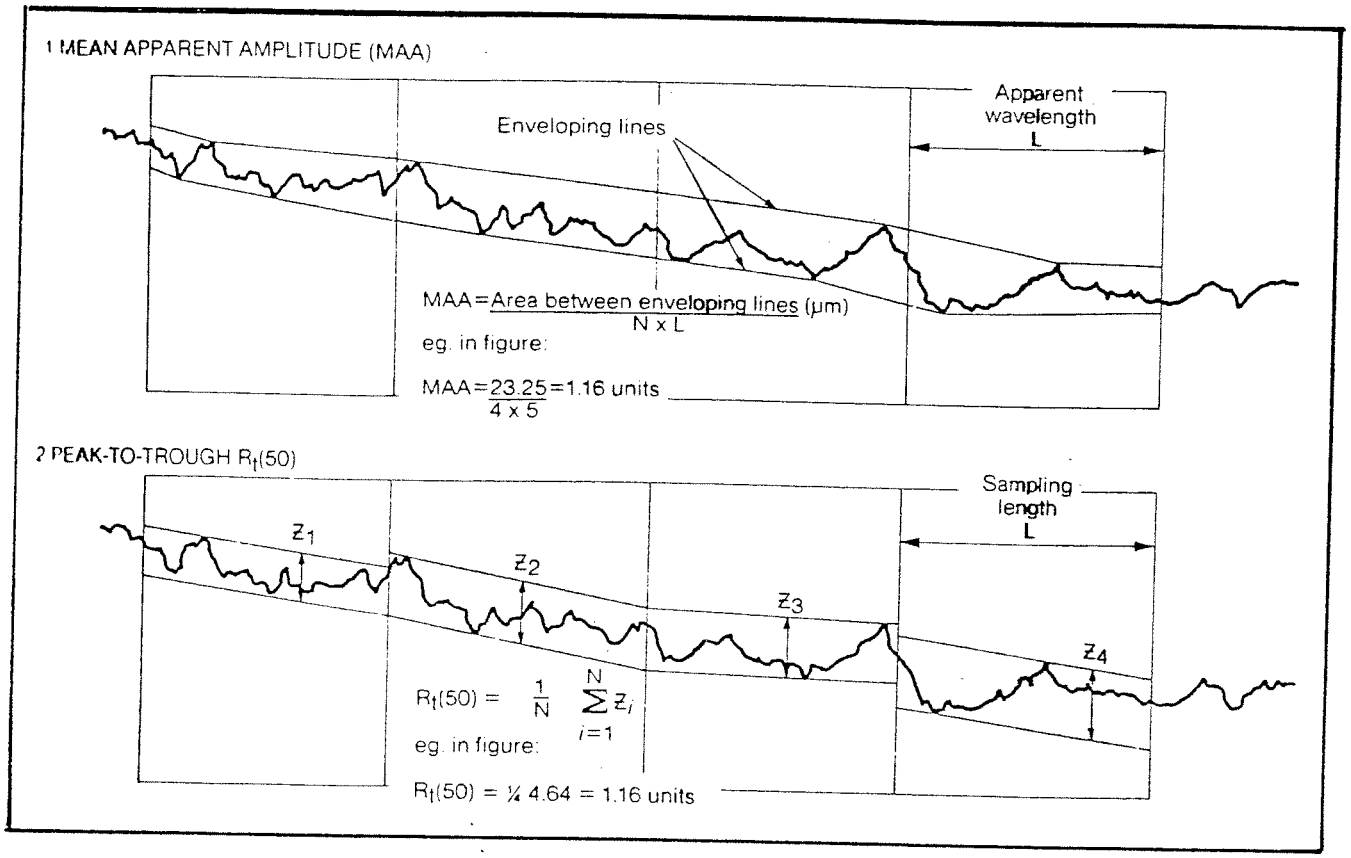
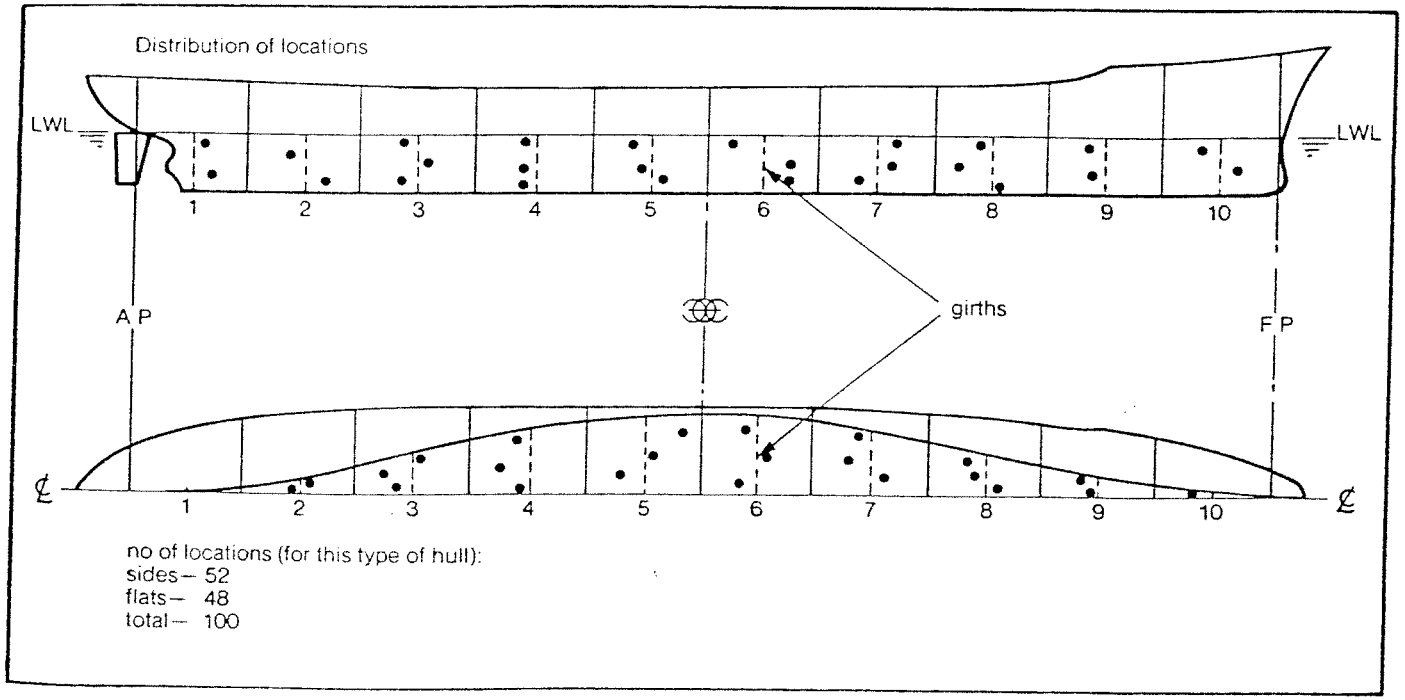


FIG 1 DISTRIBUTION OF LOCATION OF LOCATIONS FOR HULL SURVEY



2.1 A roughness survey involves the measurement of a large number of samples, in each of which we measure the maximum-maximum and the minimum-minimum, of n possible maxima or minima.

The average value of the maximum-maximum measured over a large number of samples was:-

$$H_{max} = \sqrt{2}\sqrt{m_0}\left(\sqrt{\psi} + 0.5\gamma/\sqrt{\psi}\right)$$

$$\psi = \ln\left[n(1-2r)\right]$$

$$\gamma = \text{Euler's Const} \simeq 0.58$$

If the profile is symmetrical $H_{max} = H_{min}$ and the differences averaged over all the samples is $H_{max} + H_{min}$ or

$$h = \sqrt{2}\sqrt{m_0}\left(\sqrt{\psi} + .29/\sqrt{\psi}\right)$$

Ochi (5) came to similar conclusions but used a wavelength base rather than a number n and came to the conclusion that

$$h = \sqrt{2}\sqrt{m_0}.\sqrt{\phi} \qquad \phi = \frac{1}{2\pi}\sqrt{\frac{m_2}{m_0}}$$

with the assumptions therefore that the spectrum is narrow banded and that each sample contains a sufficient number of extrema, the M.A.A. is a valid measure of roughness.

A full description of the profile requires information on the texture of the profile, i.e. in addition to the height parameter or vertical distribution, one also requires some information (in principle) on the longitudinal distribution, i.e. some measure of wavelength.

There are a variety of parameters by which the texture may be described, more or less rigorously. Grigson uses a texture parameter $\bar{m}$, a measure of separation of maxima. Byrne, following Rice and Longuet-Higgins uses a parameter $\alpha = m_0 m_4/m_2^2$. m_4 = fourth spectral moment or Kurtosis and is intuitively satisfying, as well as being compartively easy to measure since it can be interpreted in terms of the zero crossing density of the spectrum d_3, and the density of extrema d_e.

$$m_2 = \pi^2 . m_0 . d_3^2$$

$$m_4 = \pi^4 . m_0 . d_e^2$$

and is thus visibly a measure of spectral complexity.

Musker (6) included the third spectral moment (skewness) of the distribution to produce a modified height parameter.

h^1 = (1 + 0.5S) (1-0.2 Sk.Ku) where

S = Slope
SK = Skewness
Ku = Kurtosis

admittedly on a different sample length.

Byrne went on to show that although in principle any combination of height and texture are possible some are much more likely than others. In particular for new-buildings the possible variation in texture is very small. Indeed for shot-blasted or new vessels α was usually 2-3. Since α for a sine-wave is 1, (one extreme per zero crossing) the spectra of new-building roughness is extremely simple. Impressions to the contrary are an illusion conditioned by the fact that to illustrate roughness or topography of any sort it is customary to magnify the vertical axis X10-X100. The true slope and texture of well applied paint is comparatively modest, compared say with sand grain roughness. The reason for this comes from the way the random waveform is produced. The final roughness is caused by the superposition of several coats of paint, which, for a short period are viscous liquids with a limited capacity for flow and levelling. During the time for which they do flow the rate of levelling is proportional to $1/\lambda^4$, where λ is the wavelength of the disturbance, only wavelengths much longer than the film thickness survive therefore, during film formation.

2.2 Film Formation. The contribution of Rheology to Roughness:

The initial flow and levelling are governed by the equation:-

$$T_{1/2} \propto \eta \lambda^4 / \gamma h^3$$

where η is the viscosity.
γ is the surface tension.
λ the wavelength of the disturbance.
h the film thickness.
$T_{1/2}$ is the half-life for the decay of a disturbance.

For most high build solvent based compositions η, γ do not vary very much so we have

$$T_{1/2} = K.\lambda^4/h^3$$

and wavelength, become comparable when

$$K h^3 = \lambda^4$$

and the effects of film thickness, (where K is a constant)

This is the source of the characteristic texture of airless sprayed paint, normally described as "orange peel", and the practical importance of obtaining adequate coating thickness. It is readily noted that in thin areas of paint application that flow and levelling are inhibited, and in the extreme, bad texture turns to overspray or incomplete coalescence of the film.
On the other hand the scale of sizes governing the coalescence of films is:

Parameter	Size (μm)	$\lambda^4 \propto T^{\frac{1}{2}}$
Smallest Airless Spray Particles	1	1
Average " " "	10	10^4
Greatest " " "	100	10^8
Film Thickness (max)	1000	10^{12}

2.2

The quality of application has to be grossly defective before it contributes significantly to the roughness of the vessel. The contribution of orange peel to the practical outturn is shown later (Section 2)

2.3 The Statistical Analysis of Hull Roughness:

The histogram of a roughness survey is a probability distribution of maximum-maximum to minimum-minimum differences; it is an extreme value, following extreme value statistics. This only became clear when the practice of averaging the n data for each location and averaging the N locations for the vessel was abandoned in favour of computing the roughness in terms of the mode and width and displaying the distribution as a p.d.f. of the individual readings.

When this is done, it is found, as expected, that a new, intact ship, gives an excellent fit to an extreme value distribution of the form.

$$F(x) = e^{-e^{-y}}$$ representing the cumulative frequency.

$$f(x) = e^{-y} \cdot e^{-e^{-y}}$$ the p.d.f., or first derivative of $F(X)$

$$y = (x-a)/\theta$$

a = Mode of the distribution.

θ = Width of the distribution.

which are related to the mean and standard deviation of a Gaussian (Normal) distribution by

$$\text{Mean } (\bar{x}) = a + \gamma\theta$$

$$\text{Standard Deviation } \sigma = \pi\,\theta/\sqrt{6}$$

where γ = Euler's Constant.

The main implications of the correct choice of distribution are that the probabilities of low values are much less than that implied by a Normal distribution. Indeed the search for a better fit for the data was first investigated when it was noticed that the size of the standard deviation of ship roughness surveys frequently implied negative roughness (Fig.5), a physical impossibility.

On the other hand the distribution tends to unity much more slowly than a normal distribution. Roughness which looks like anomolous outliers of a Normal Distribution are anticipated and accounted for by an Extreme Value distribution. In a "mean-of-means" of course, they are simply obscured by truncating the histograms.

In effect the extreme value distribution also contains information about the texture of the surface in terms of the width parameter θ , since it measures the distribution of the extremes. Were all the extremes the same size (as in a simple sine wave) we would have, not a distribution, but a line spectrum of zero width. The width of the extreme value distribution is therefore a good measure of the quality of painting.

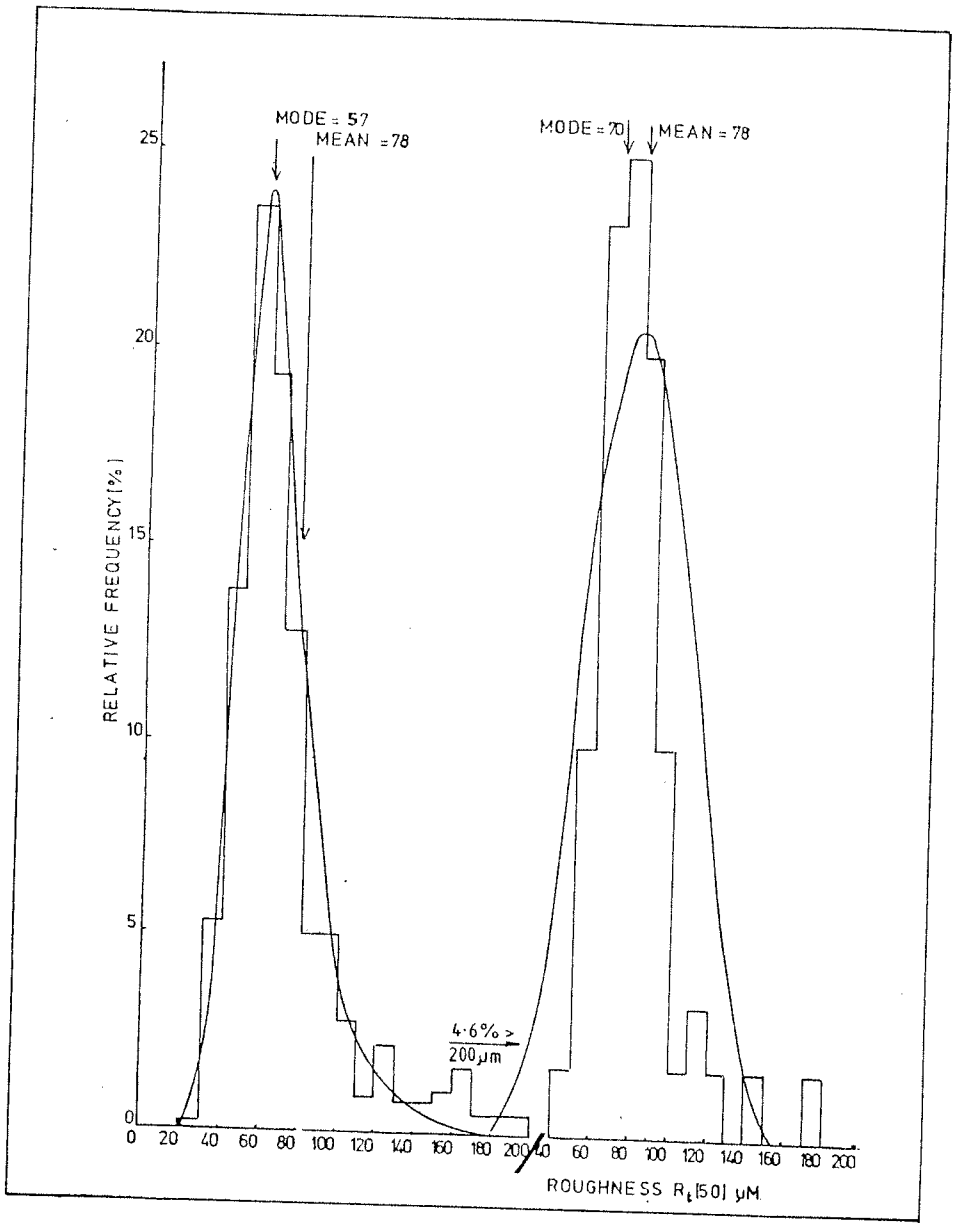


FIG. 3 COMPARISON OF EXTREME VALUE AND NORMAL [grouped] DATA FOR A NEW BUILDING.

2.4 Change of Roughness with Age:

The limitations of a mean and standard deviation as a description of hull roughness become more conspicuous as a vessel ages. While the new ship, freshly painted can be considered as one, or at most two roughness populations (sides and flats), discrete processes, frequently non-random, contribute to the development of roughness on the hull as it ages. We can, therefore, expect the roughness p.d.f. to change with time. In particular, we would expect the mature ship roughness to comprise two populations, one representing the intact paint, and one the total of damage, defect and system deterioration. The intact paint from the foregoing should be well fitted by an extreme value distribution, since, like the new ship it is generated by the superposition of coats of paint. The damage, defects and deterioration of the system may be of an order of magnitude worse than roughness of the intact paint, which may have a modal roughness of 50-150 μm. The complete paint system may have a thickness of $>500 \mu m$. Detachment to metal therefore will generate roughness of this order, and detachment followed by pitting corrosion, roughness in excess of 2000 μm. Poor paint application, such as overspray, or blistering will also generate large roughnesses.

It is convenient to deconvolute the roughness histogram into two extreme value distributions, to extract as much information as possible from the roughness data, and this is done, by computer by a curve fitting routine (Fig. 4). Figure 5 shows how much information is lost by reliance on A.H.R. and standard deviation only.
It is not possible to specify, from the fitted curves, exactly the causes of damage and defect, and the filling of the 'rough' population is only approximate since the extreme value approach depends on there being a large number (n) of extrana in a given sample. For the larger defects, where the size of the defect is of the order, or greater than, the sample size, this is not the case, and if only one form of defect is present then a Gaussian distribution would be expected, but this is rarely so.

2.5 Ship Roughness New Building and In-Service:

Byrne, 1976-1979, [2] collected average hull roughness data on 147 surveys of 47 vessels, and these were supplemented by further results (internal reports and [7]) between 1979-1982, mainly from outside Japan, and mainly on non-self-polishing systems.

Table 1 , summarises 24 recent results from Japan from a total of 112 surveys, and mainly on new-buildings or reblasted vessels coated with self-polishing antifoulings. The A.H.R. (Average Hull Roughness) of acceptable new-buildings was judged by Byrne and BSRA to be 125 μm per 50 mm.

ROUGHNESS OF SOME RECENT JAPANESE NEW-BUILDINGS AND DRYDOCKINGS

	Date	Type	Treatment	Sides		Flats		Hull	
				In	Out	In	Out	In	Out
1		RB	3 x A/C + 3 x SPC9					243(99)	85(18)
2		ND	3 x A/C + 3 x SPC9	231(73)	198(69)	236(54)	168(74)	233(69)	186(72)
3		RB	3 x A/C + 3 x SPC4	250(74)	226(76)	245(88)	234(58)	248(80)	229(68)
4	6/80	NB	2 x A/C + 2 x SPC						103(29)
5		ND	1 x T/U 3 x SPC4	125(49)	153(64)	169(72)	174(38)	143(63)	162(56)
6	3/82	ND	1 x T/U 4 x SPC9	188(87)		204(98)		193(91)	
					144(65)		169(37)		151(54)
7	1/81	RB	3 x A/C + 3 x SPC9	238(113)	230(148)	181(54)	-4-(41)	219(101)	188(120)
8	5/80	NB	4SPC9(s)	250(134)		334(140)		301(152)	
			2 x A/C + 3 x SPC9		100(37)		134(49)		116(46)
	5/81			92(37)		104(31)			99(34)
			1 x SPC9				158(59)		
	1/82		1 x SPC9	108(31)		115(41)			111(35)
					109(37)		146(46)		123(44)
9	6/81	NB	3 x A/C 1 x SPC20)		81(27)		83(26)		82(26)
	9/82		2 x SPC4)	81(18)	87(24)	112(22)	106(36)	95(25)	97(32)
			2 x SPC4						
10		RB	3 x A/C + 3 x I.S.S.		185(58)		140(31)		167(54)
11		NB	2 x A/C + 1 x A/C +		113 (36)		128(40)		119(38)
12	5/82		2 x SPC9	233(100)	162(46)	89(44)	124(43)	204(90)	100(46)
13	10/81	ND	3 x TU. 3 x SPC9	274(160)	220(111)	198(89)	206(94)	248(144)	215(106)

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ROUGHNESS OF SOME RECENT JAPANESE NEW-BUILDINGS AND DRYDOCKINGS

	Date	Type	Treatment	Sides		Flats		Hull	
				In	Out	In	Out	In	Out
14	6/81	NB	3 x A/C only + 2 x SPC9(F) 3xSPC9(S)	106(33)	71(29)	89(24)	81(29)	99(31)	76(29)
15	8/82	RB	2 x A/C + 1 x SPC20 + 3 x SPC4	518(125)	130(127)	470(127)	135(48)	495(129)	132(56)
16	10/79	RB	2 x A/C + 4 x SPC20	304(195)	163(83)	337(125)	227(200)	317(204)	189(145)
17	9/81	RB	3 x A/C + 3 x SPC9					292(117) 98(23)	123(56) 129(22)
18	5/81	RB	*At shop primer stage After. SPC	318(116)	330(65)* 131(30)	300(91)	284(40) 131(22)	312(109)*	131(28)
19	7/82	RB	2 x A/C + 1 x SPC20 + 3 x SPC9	570(130)	151(45)	467(163)	170(40)	529(153)	158(44)
20	7/82	RB	2 x A/C + 1 x SPC20 + 3 x SPC9	583(210)	142(40)	422(167)	147(38)	508(209)	144(39)
21	9/80 9/81	RB	3 x A/C + 3 x SPC4	100(42)	85(131) 121(48)	119(39)	90(33) 150(144)	107(42)	87(32) 132(49)
22	3/82	NB	2 x A/C + 2 x SPC9(F) 3 x SPC9(S)		141(71)		147(43)		147(47)
23	1/81	RB	SP + 3 x A/C + 3 x SPC(F) 4 x SPC(S)	267(104)	84(23)	252(61)	89(21)	262(92)	85(23)
24	5/82	RB	2 x A/C + 2 x SPC + 2 x SPC4	405(102)	119(19)	408(155)	124(23)	406(26)	121(21)

CODE:
NB = New Building
RB = Reblast
ND = Normal Recoating

A/C Anticorrsive System x No. of Coats 3 x Usually Vinyl Tov
2 x Usually Pure Epoxy/Pitch Epoxy
Self-Polishing Compositions (SPC) = Intersmooth SPC 4,9,20
I.S.S. = Interspeed Special

2.5

The six new-buildings of the present study, vessels 4, 8, 9, 11, 14, 22 of Table 1, have got roughnesses on delivery, as follows:-

No.	AHR						
	Sides		Flats		Total		
	Mean	S.Dev.	Mean	S.Dev.	Mean	S.Dev.	6x
4					103	29	
8	100	37	134	49	116	46	
9	81	27	83	26	82	26	
11	113	36	128	40	119	38	
14	71	29	81	29	76	29	
22	141	71	147	43	147	47	
Average	101	40	115	37	107 ± 20	36 ± 7.4	

Equivalent Extreme Value Statistic:

Mode = $\tilde{x}$ = 91 (Mean = Mode + .58)

Width = θ = 28 (Standard Deviation = $1.28\,\theta$

Minimum = $\tilde{x} - 2\theta$ = 51.0 1% Level

Maximum = $\tilde{x} + 5\theta$ = 247.0 99% Level

These results are instructive. They suggest that the 125 μm standard for new-building A.H.R. is quite achievable even where the total film thickness of modern coatings is much greater than previously, and achieved in fewer coats.

We should therefore find, for a well painted new-building approximately 98% of the roughness between 50 microns and 250 microns and this is generally the case. Fig. 3 shows the typical new-building distribution.

2.6 Reblasted Vessels:

While the damage and defects of an underwater hull are minor and localised, they are readily repaired. After 5-10 years, depending on the system used and the original standard of preparation, the whole paint system will have deteriorated to the point where only total reblasting and recoating can restore it.

The decision to do this will usually be as a result of a measured fall in performance. This is of particular importance to constant speed vessels, and it is no coincidence that six of the eleven reblasted vessels are container ships, and that five of them were reblasted and recoated as part of a major reconstruction which involved the replacement of steam by diesel power for reasons of fuel economy and improved performance.

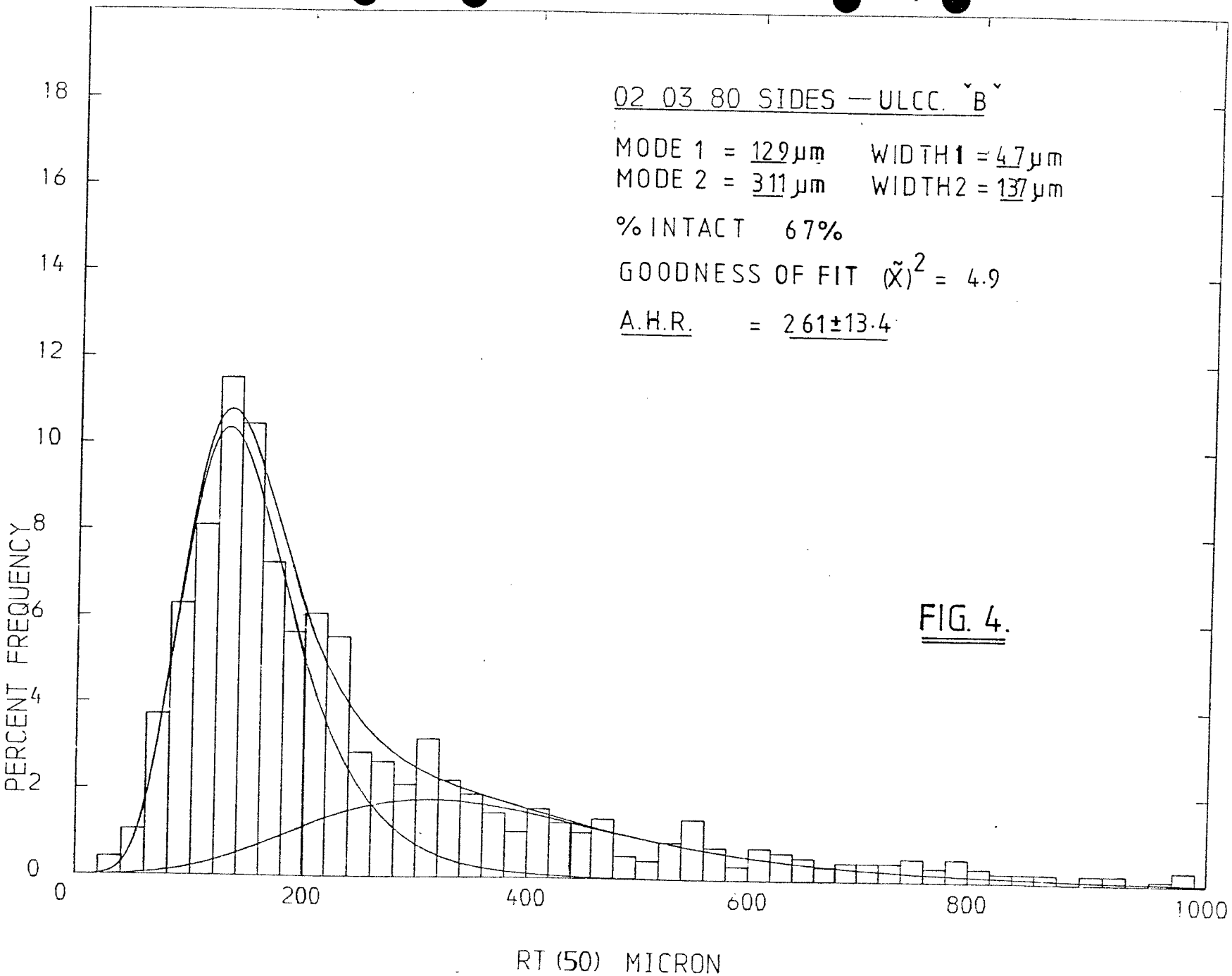
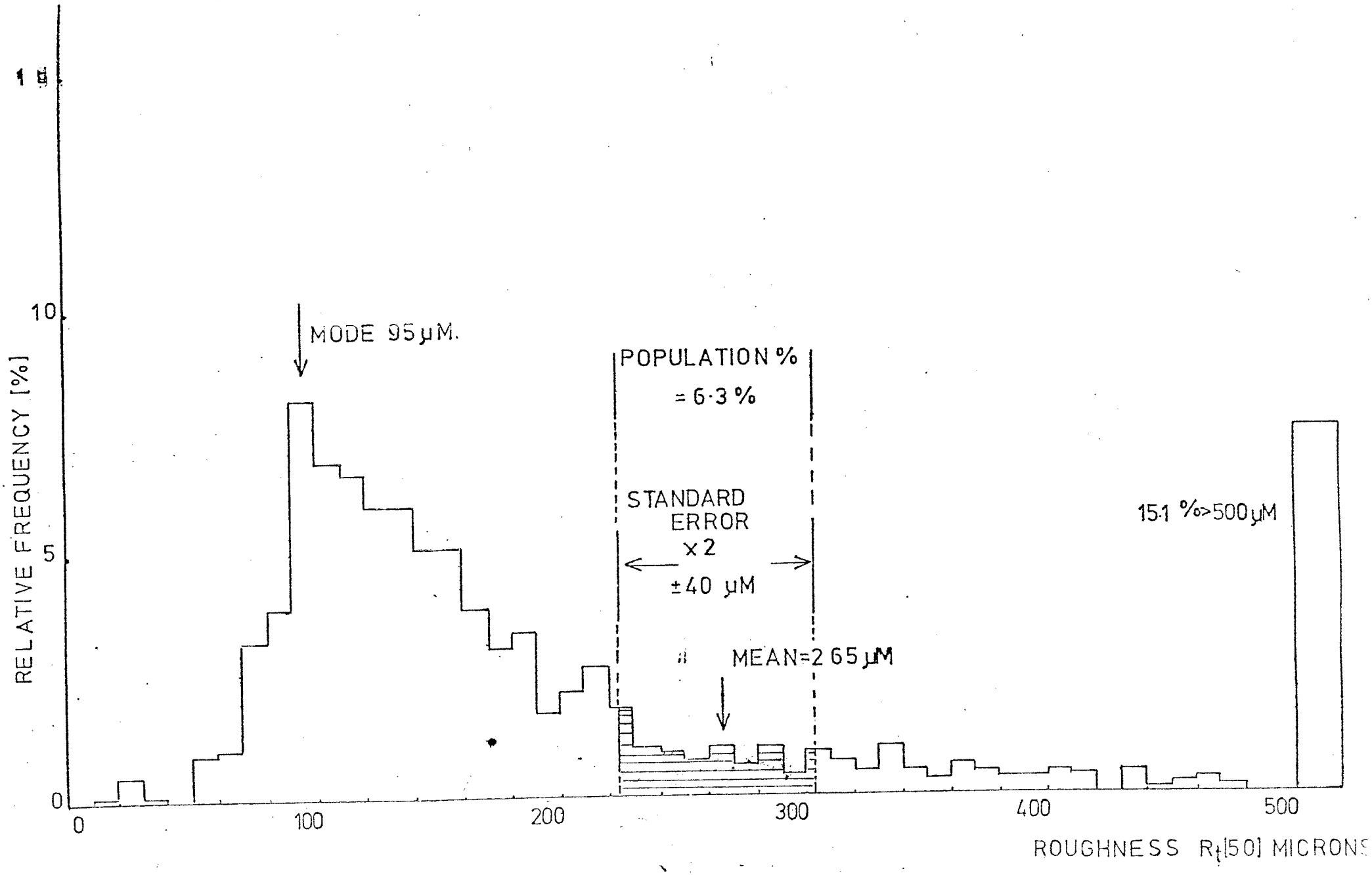

02 03 80 SIDES — ULCC. "B"
MODE 1 = 129 µm WIDTH1 = 47 µm
MODE 2 = 311 µm WIDTH2 = 137 µm
% INTACT 67%
GOODNESS OF FIT $(\tilde{X})^2$ = 4.9
A.H.R. = 261±13.4
PERCENT FREQUENCY
0
2
4
6
8
10
12
14
16
18
0
200
400
600
800
1000
RT (50) MICRON

FIG. 4.

FIG. 5.


RELATIVE FREQUENCY [%]
10
5
0
MODE 95 µM.
POPULATION %
= 6·3 %
STANDARD
ERROR
× 2
± 40 µM
MEAN=2 65 µM
15·1 %>500 µM
0
100
200
300
400
500
ROUGHNESS Rt[50] MICRONS

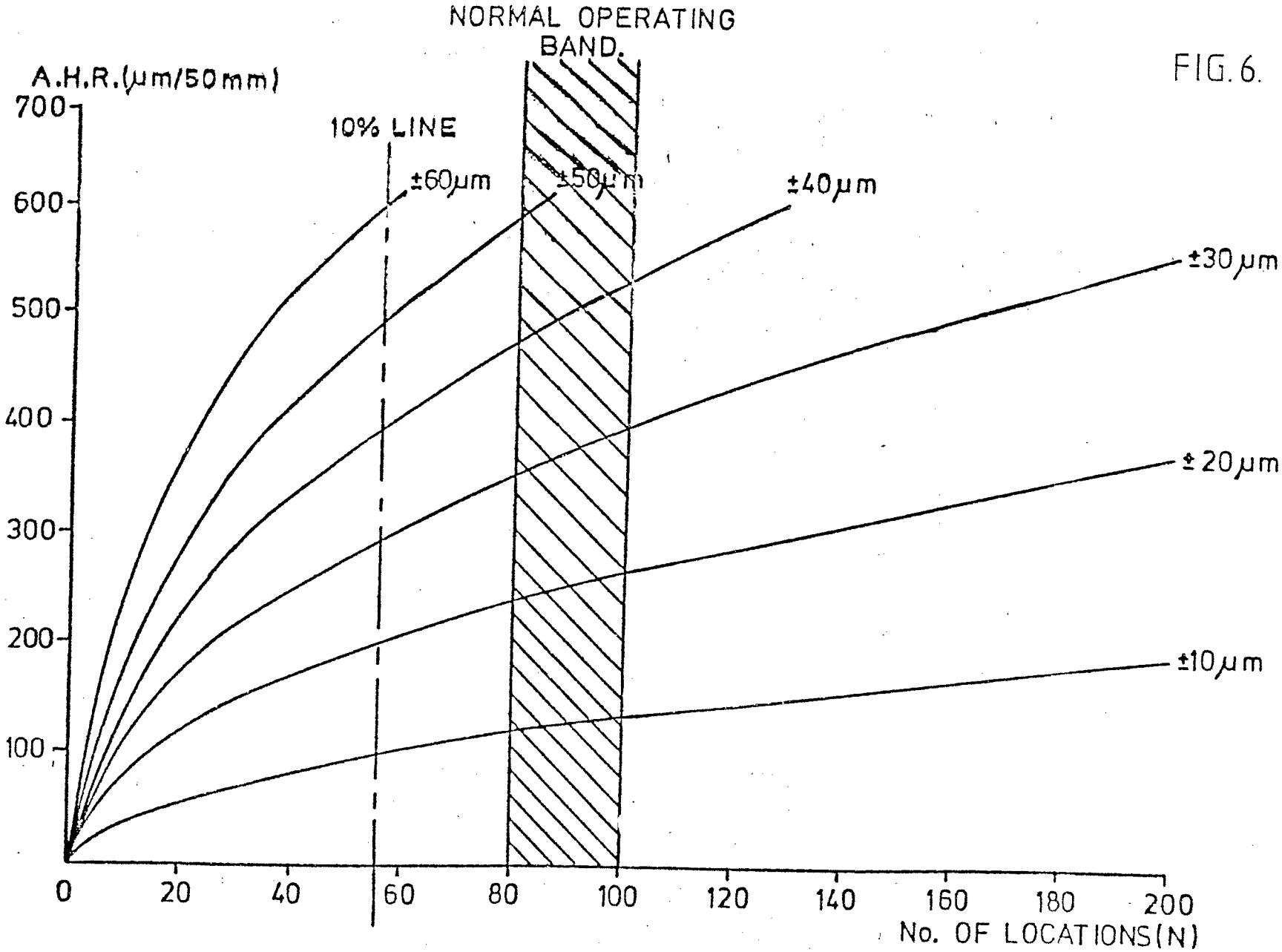


FIG. 6 STATISTICAL LIMITS TO THE PRECISION OF HULL ROUGHNESS SURVEYS.

2.6

The results of reblasting and recoating with S.P.C. were, for vessels 1, 7, 15, 16, 17, 18, 19, 20, 21, 23, 24:-

Average In-docking Roughness	358 ± 68
Average Out-docking Roughness	132 ± 19

On average the vessels benefited by 226 microns as a result of reblasting and recoating, and the recovery brings them back to a roughness of almost new-building quality; only residual corrosion pitting, a permanent addition to roughness, prevents complete restoration.

2.7 Sources and Causes of Ship Roughness:

The ways in which surfaces become rough and the extent and rate at which it happens are of considerable importance. Only by attention to them will the major benefits of modern coatings be realised. The prediction of ship performance depends on the assumption that the surface roughness is homogenous, isotropic and Gaussian. This may have formerly been the case for conventionally coated hulls, but it is no longer so. It is also clear that, given the rate of deterioration in the past (25-40 microns per annum), neither attempts at prevention nor cure have been particularly successful. It has already been shown that, as a vessel ages the roughness population becomes complex, the major part representing intact paint, which may be well or badly applied, and the remainder the aggregate of damage and defect.

The analysis of hull roughness, while it quantifies these effects, does not give much guidance as to their causes. In the course of roughness surveying we therefore note, at each location, the characteristic roughness of the area surveyed, under thirteen headings and on a base of severity of 0 to 3, 0 = no defect, 1 = slight, 2 = moderate, 3 = severe. It is subsequently possible to calculate the contribution for each of the defect and damage parameters. The categories of roughness into which damage and defect most observed fall are:-

		Code
1.	Damage by Abrasive to the coating on spot blasting	A
2.	Blistering of the coating system	B
3.	Corrosion at damage points	C
4.	Cracking of coatings	Cr or K
5.	Detachment of coatings	D
6.	Fouling residues from poor cleaning	F
7.	Grit and dirt inclusions from poor cleaning	G
8.	Mechanical damage from grounding, berthing, tugs and fenders	M
9.	Orange peel, typical airless spray texture	O
10.	Pitting damage, following detachment	P
11.	Runs, dribbles and spatter from poor application	R
12.	Dry spray, or overspray from poor application	S
13.	Wavy undulations from poor application	W

The above roughness sources belong to three main categories:-

1. The intrinsic properties of the paint 2,4,9,11,12,13 possibly affecting the hull extensively.

2. A combination of paint properies and method of application 6,11,12,13.

3. Intensive, external sources representing catastrophic events and producing large roughnesses over small areas, e.g. 1,3,5,8,10.

Analysis of Roughness Severity:

From a number of such surveys it is possible to evaluate the contribution of these defects to the overall roughness of the hull and these are summarised for both new buildings and vessels in service in the following tables, from which the following conclusions may be drawn:-

1. Although the surveys were done on a number of occasions, under different conditions and locations, there is a satisfactory agreement between measured height and visual assessment, where a large enough sample is present.

2. Where no other defect is present, 'orange peel' is the limiting factor for smoothness. The rheology of the paint, in terms of flow and levelling therefore govern how well a vessel may be painted. Since the average value of this parameter is about 109 microns, it is seen to correspond almost identically to the roughness of new ships.
What we do not know is whether this type and degree of roughness is of hydrodynamic consequence at ship Reynolds numbers, but poor texture (grade 3) can evidently account for an additional 100 microns of roughness, and the other roughness components, arising from the interaction of rheology and application, overspray, waviness, runs, drips and sags can add, respectively 150, 200, or 300 microns where present, to the roughness of a new building or reblasted vessel. If roughness of this order is significant, new concepts in materials and application methods will be required.

3. The contribution of the intensive sources of roughness, detachment, mechanical damage and pitting is localised but severe, and frequently contributes up to 70% of the roughening above new ship level.
Mechanical damage is local to the bow(cable chafing), to the parallel mid-body, and to the boottopping, where tug and fender damage are the principle causes of roughening. The forces involved in these forms of roughening are such as to destroy conventional coatings.

4. The maintenance of a smooth hull, therefore has to take into account the differing ways in which practical roughness is generated; the rheology of the paint, the methods of application, and the mechanical properties of the coating system, particularly the resistance to impact and abrasion.

5. Self-polishing antifoulings are a significant contribution to the maintenance of clean smooth hulls. Roughness acquired in the course of their application, as long as it is small in comparison with the coating thickness (20-50 microns, say) will polish, but the anticorrosion system needs to be applied with care, not only because of the contribution to the overall roughness but because asperities at this level lead to premature polish through of the system.

TABLE 2

New Building Roughness

Roughness Parameter	Roughness Grade	No of Locations	%	Mean Roughness $\bar{x}$	Std Dev. σ_{n-1}	Std Error S.E.	Largest Value per Location		
							Mean x	Std Dev. σ_{n-1}	Std Error ± S.E.
1 Orange Peel	1	29	33	109	39.6	± 7.4	211	119	± 22
	2	26	30	114	16.9	± 3.3	218	96	± 19
	3	4	4.5	178	123	±61.1	379		
2 Overspray plus Orange peel	1	17	19.3	107	14.7	± 3.6	170.0	50	± 12
	2	5	5.7	141	45	±20.0	370.0	290	±131
	3	1	1.0	114			172		
3 Wavy Roughness	1	6	6.8	101	14.5	± 6.0	152		
	2	-		-	-	-			
	3	-		-	-	-	-	-	-
No of Cases		88							
Total No of Locations		89							

Average Hull Roughness at Outdocking 113.6 μm/50 mm

TABLE 3

Mean Roughness Grades for Six Rough Sample Areas and one New Building

Roughness Parameter	Grade	Sample Areas						Mean Value	New Building 'O'
		A	B	C	D	E	F		
Orange Peel	0	139±12	115	147±16	109±5	-	-	128±9.2	-
	1	164±11	150±9	-	-	118±11	106±30	135±13.5	109±7.4
	2	168±7	193±13	193	132±14	142±9	132±14	160±12.0	114±3.3
	3	289±43	244±18	235		173±48	193±62	211±23	178±61.0
Wavy Roughness (W)	1	-	233	-	134±8	133	-	167	101.0±6.0
	2	234		-	-	173±36	-	213	-
	3			-	-	248		248	-
Overspray (S)	1	-	-	-		161	162±46	162	107±3.6
	2	-	-	151±24		-	241±30	196	141±20.0
	3	360	360	229±24		-	256±78	301	114(1 sample)
Runs/Drips sags (R)	1	-	170			-		170	
	2	-				-	216±47	216	
	3	393				-		393	
Grit etc. Inclusions (G)	1			154		182	-	168	
	2	305				-	-	305	
	3		466			-	-	466	
Detachment (D)	1	-	311±48				-	311±48	
	2	-	332±22				-	332±22	
	3	-	444±37				-	444±37	
Mechanical Damage (M)	1	399	278±42	230±48		160		265	
	2	-	-	284±17	164±25		361±59	270	
	3	400	338±26	321±46				353	
Pitting (P)	1	-	-	-		129	202±32	166	
	2	-	-	-			261±49		
	3	-	-	178±22					

3. Practical Roughness Measurement:

Information on the roughness of a vessel has several practical uses, the principal of which are:-

3.1 For techno-economic decision making, principally the decision as to whether it is necessary to completely reblast a hull.

3.2 For quality control at new building to comply with the owner's new building specification.

3.3 For quality control in drydock, in which case both in-docking and out-docking surveys will be required.

3.4 Precision. Whatever the reason for the survey it is important that the survey obtained is an objective measure of hull roughness, but it is also important to remember that even when carried out well, the measure is a statistical one and that the precision depends on the amount of data, and that a statistic implies an error band. Figure 6 shows the error associated with the estimation of average hull roughness A.H.R. for different levels of accuracy. In general, for a survey with 95% confidence, of a moderately rough vessel, 100 locations, (1000 - 1500 data points) will be required.

Objectivity. It is important to note that it is the roughness of the hull which is required, not the roughness of selected features of roughness. The most difficult attitude to teach a surveyor is the importance of a non-selective approach, particularly if he or she has some pre-conceived idea as to what is required.
The hull is considered as an equal area grid, by dividing it lengthwise into 10 stations, and at each station into equal bands round the girth at that location. Sides, flats and boottopping, which frequently have different paint systems, and suffer differently in terms of impact, abrasion and exposure are considered separately. (Fig. 1)
Within each equal area, the hull roughness analyser is placed at random, without consideration of the nature of the surface, whether it be considered rough, smooth, broken or intact, excepting only weld lines.
The same quantity of data should be collected at each location. Failure to do this can result in two kinds of bias; if the gauge is placed on a smooth or rough area at the beginning of each run, there will be anomalously high or low readings at beginning and end, and if the surveyor collects longer or shorter strings of data depending on roughness, bias will also be introduced; both types of bias have been found.

Weighting. The average hull roughness of a vessel, if the survey has been made on an equal area basis, will be unbiased. If, however, the data collected is not in proportion to the areas of flats, sides and boottopping then the result must be weighed in proportion to the area.

A.H.R. (Vessel) = (A.H.R. Sides x % Area Sides) + (A.H.R. Boottopping x % Area Boottopping) + A.H.R. Flats x % Area Flats)

3.4 Calibration. Calibration strips are now provided as standard with each gauge, and the gauge should be calibrated at the beginning and end of each survey and included with the survey data. The pick-up of the instrument is a sensitive strain gauge which can be damaged by misuse, and this has been a frequent source of error. As roughness becomes more and more specified, and the subject of guarantee, so the responsibility of the surveyor for an accurate and objective survey increases.

Timing. There are two other major sources of error in roughness surveying. At in-docking the surface must have been washed or water-jetted free of fouling and salt; even when apparently clean, dry salt crystals can be of 50-500 μm in size, and make a considerable difference to a smooth/ hull.
At out-docking, sufficient time must elapse between the completion of painting, and the carrying-out of a roughness survey.

4. The Role of Self-Polishing Antifoulings in Ship-Maintenance.

4.1 The self-polishing antifoulings, introduced in 1973, serve at least four major roles in improved ship performance.

1) Improved lifetime, giving longer docking intervals.

2) An avoidance of coatings build up and the concomitant roughening and deterioration, because the product polishes away.

3) A short term reduction in roughness, due to the "smoothing" of 'application' roughness.

4) Outstanding fouling control.

4.2 The improved lifetime is a consequence of the linear mechanism of controlled release. Previous antifoulings, of all sorts, had only poor control of the leaching of biocides. At best the release was, as a rate process, equivalent to exponential decay, and beyond a certain point, increased lifetime can only be obtained wastefully. The self-polishing copolymers, with a linear release mechanism, have a lifetime which depends only on the initial film thickness, which is, in principle unlimited, and systems for up to five-years lifetime have been specified.

4.3 It is recognised that the accumulation of old coatings, as a consequence of repeated dry-docking over a number of years, is the major source of underwater hull roughening. If the hull is adequately protected by cathodic protection then there is no steel loss; the most effective demonstration of this is the way it is possible to restore 7-10 year old vessels to almost new ship condition by reblasting and recoating.
A conventional system comprising 2-3 coats of anticorrosive, a fitting-out antifouling and 2 coats of service antifouling will have an initial thickness

of A/C	2 x 125	or	3 x 75	
Fitting out A/F	1 x 50	to	1 x 75	
Service A/F	2 x 50	to	2 x 75	for 18 months service.
	400	to	450	microns

increasing at every dry-dock by:-

4.3

1 coat of Sealer	1 x 75μm)	225 microns
2 coats of Antifouling	2 x 75μm)	

The total thickness of coating after 7-10 years is 1525 - 2025 microns.

This build up of composition with all the possibilities for detachment and breakdown is the most conspicuous source of the accumulation of roughness with age. Because of their outstanding film integrity and lack of porosity, the self-polishing antifoulings can be applied without sealing, avoiding the accumulation of sealing coats, and avoiding the inactivation of residual S.P.C. which remains. Over a period of years the rate of application and of polishing will be in equilibrium and the system thickness will not increase with time.

4.4 The smoothing of self-polishing materials, and the beneficial effects on ship performance have been demonstrated in several experiments. In [7], Milne showed that in a series of plates of self-polishing antifoulings, in ship-trials with roughnesses from 46 to 654 μm A.H.R., dramatic reductions in both roughness and drag resulted from five months exposure. (Table 4).

Table 4

Plate No.	Roughness (μm)	Drag Kgf @ 7.5mm sec.	Drag Ratio (Rough)/(Smooth)
1. Before	46	3.8	1.14
After	54	3.7	1.11
2. Before	351	4.7	1.41
After	113	4.2	1.26
3. Before	415	5.2	1.56
After	95	3.96	1.15
4. Before	654	5.65	1.69
After	136	4.25	1.27
Control	-	3.33	1.0

The results on panel 1, although not statistically significant are seemingly anomalous, but suggest that there is a limit to turbulent polishing; changes in other roughness parameters, R_a (C.L.A.), Rq (R.M S.) and the slope of asperities and valleys also tend to limiting values.

4.5 The earlier experiments were done using the first commercial grades of self-polishing antifoulings, which had rapid polishing rates, and rapid smoothing rates. The economy of the self-polishing systems are dependent on the balance between antifouling effectiveness, application thickness and polishing rate. The development of more effective, slower polishing materials over the last 10 years,(Intersmooth 1, 4, 9, 20) and materials designed for different vessels with widely different operational requirements made it necessary to re-analyse the polishing and smoothing characteristics of these materials under service conditions.

4.5 This experiment[1], involved four different commercial grades of self-polishing material, applied at three different levels of roughness, exposed for 9 months on special panels attached to the bilge keels of a U.L.C.C. from September 1981 to May 1982, during which the vessel did 44,000 miles @ ~ 4,900 miles per month.

Two techniques of topographical measurement were used:-

i) Random, statistical, profiling.
ii) Precision relocation.

The latter technique, designed in collaboration with and following the methods of 9 , involved machining the reverse sides of the panels with orthogonal grooves which sit on a set of matching spherical bearings on the relocation table.
The relocation results were checked against the statistical analysis of ten random profiles, and for the roughest panels of the set (B,C,), up to 25 profiles were examined.

Instrumentation. The roughness was measured before and after immersion on a Ferranti 30B Surfcom and the digitised data was processed on a C.B.M. microcomputer.

The roughness parameters calculated were:-

i) Maximum Peak - Trough Height.
ii) Root Mean Square Roughness.
iii) Mean Slope of Roughness.

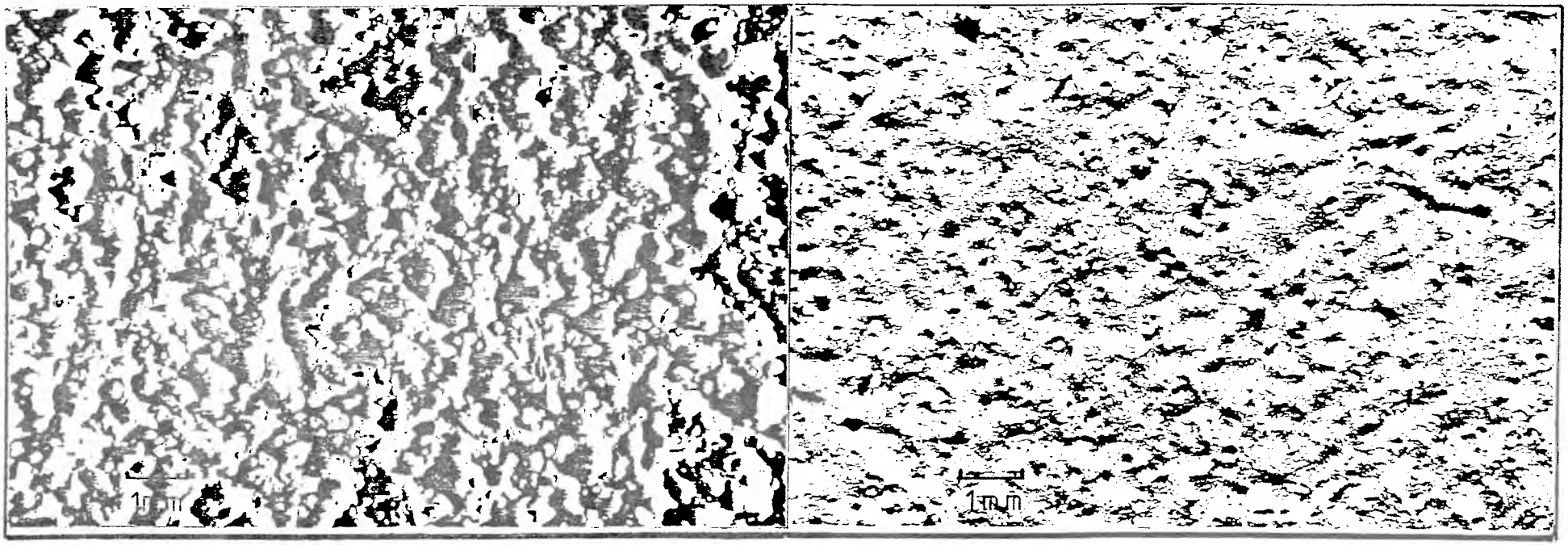
The long and short wavelength cut-offs are 20mm and 500μm; (this is as close as the bench measuring equipment was able to approximate to the standard R_t(50) of conventional ship measurement).
The changes in these roughness parameters over a nine-month period are summarised in Table 5 , and Figure 7 shows the superposition of two of the re-located traces for system 3B, before and after, and shows the dramatic change in texture, and amplitude, which are equally impressive, when original and final microphotographs are compared. (Fig. 8)

The conclusion drawn from the complete series of trials and measurements are:-

1. Where the paint is applied as well as it can be (A panels), i.e. $RMS_{20} \leqslant 10\mu m$, $R.T.M._{(20)} \leqslant 25\mu m$ $slope_{[20]} < 0.5^{o}$ there is little or no change with time, confirming the results of the previous experiment.

2. The rate of polishing (bulk removal of composition) and the rate of smoothing (reduction in roughness parameter) are largely proportional, i.e. system 4, intended for slow polishing is also on the basis of (figure 9) the slowest to smooth.

1. See acknowledgements.



BEFORE

AFTER

POLISHING OF DELIBERATLY ROUGHENED SYSTEM 3[INTERSMOOTH 4], AFTER 9MONTH TRIAL

INITIAL ROUGHNESS RTM = 281 μm FINAL = 151 μm.

RMS = 70 μm = 41 μm.

FIG 8

System 3 Before Immersion

System 3 After Immersion

Period:- 9 Months

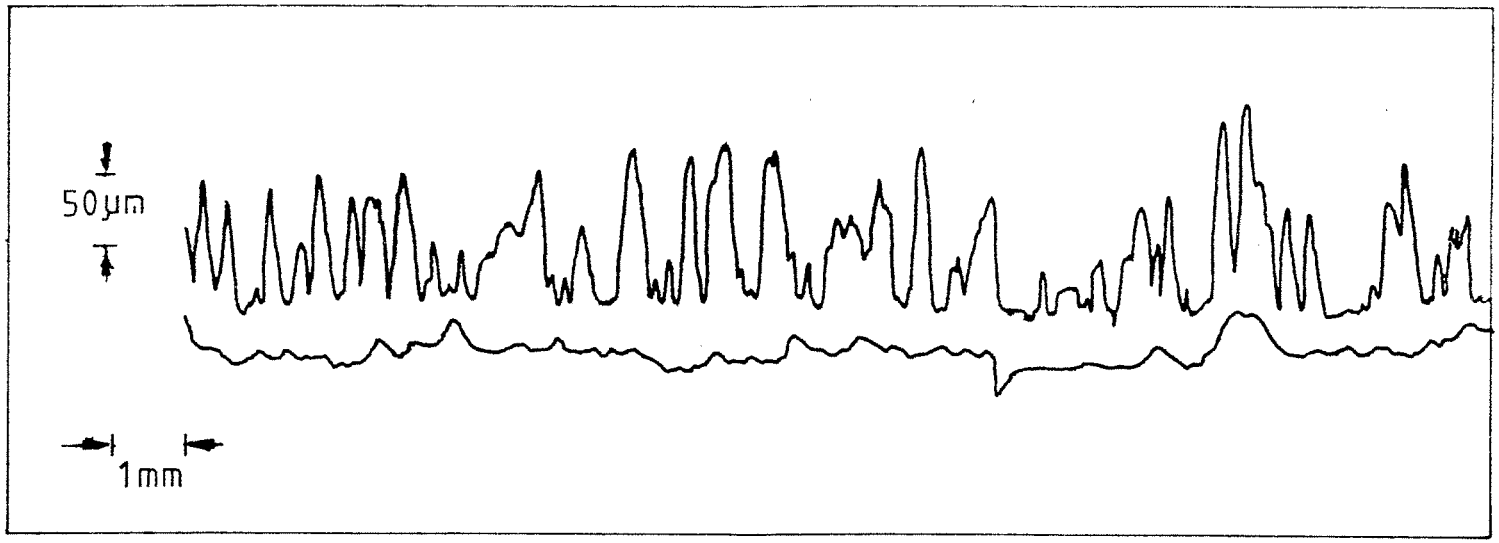


FIG. 7 Relocation Profiles of Copolymer System 3 after 9 Months Service.

Table: 5 Roughness Changes In-Service of a Series of Commercial Self-Polishing Compositions

System		$_0$R.M.S$_1$		Δ RMS	$_0$R.T.M$_1$		Δ RTM	$_0$SLOPE$_1$		Δ S
1	A^2	12	14	-2	22	27	-5	0.4	0.5	-0.1
	B	83	68	15	319	244	75	10.4	8.4	2
	C	56	29	26	211	94	117	7.3	2.8	4.5
2	A^2	13	12	1	35	27	8	.7	.7	0
	B	42	25	17	159	91	68	5.3	2.6	2.7
	C	58	36	21	216	128	88	7.8	4.3	3.5
3	A^2	8	10	2	19	36	-17	.5	.6	-.1
	B	41	15	26	152	48	104	5.3	1.3	4.0
	C	70	41	29	281	151	130	9.2	4.6	4.6
4	A^2	9	8	1	23	28	-5	.4	.4	0
	B	39	29	10	151	104	47	5.3	3.6	1.7
	C	68	51	17	266	201	64	9.1	6.6	2.5

System 1. The original SPC1 (1974) a fast polishing, non-copper, organo-tin copolymer anti-fouling.

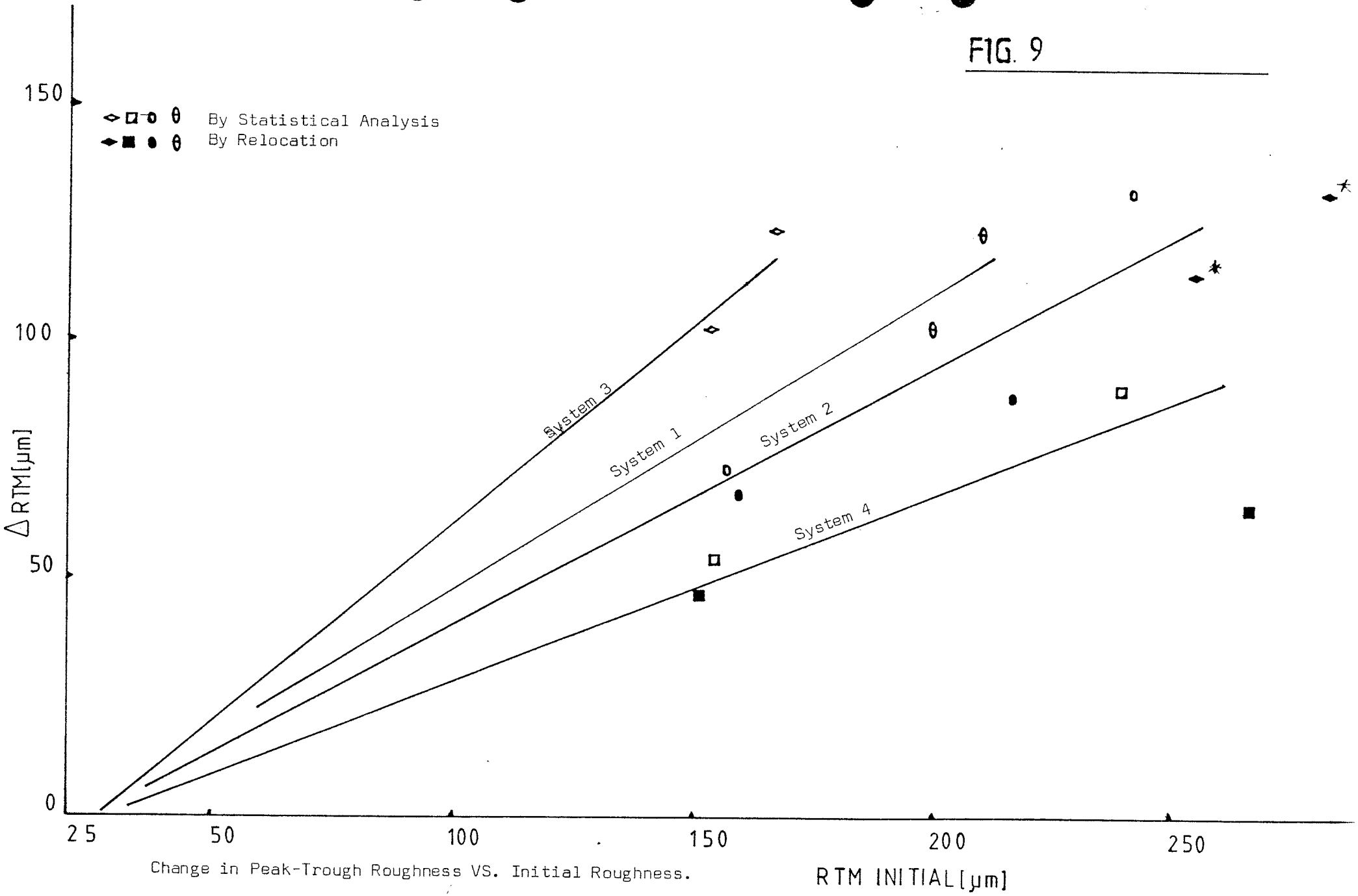
System 2. SPC4 moderate polishing copper containing, organo-tin copolymer anti-fouling.

System 3. SPC9 moderate polishing cuprous oxide organo-tin copolymer anti-fouling.

System 4. SPC20 slow polishing cuprous oxide organo-tin copolymer anti-fouling.

R.M.S. = Root Mean Square.
R.T.M. = Peak - Trough.
Slope = Roughness Slope (^{o})
Δ = Difference.
0 1 Superscripts = Before = (0)
After = (1)
A^2 By semi-relocation.
B,C, Random Survey.

FIG. 9



Change in Peak-Trough Roughness VS. Initial Roughness.

4.5

3. Whether on the basis of RTM, RMS, or Slope, the reduction in roughness tends to some non-zero value, which seems to be similar, independent of polishing rate or initial roughness; this again confirms previous work and ship experience.

5. Copolymer Antifoulings: Antifouling Performance.

The chemistry and mode of action of the self-polishing antifoulings have been described extensively elsewhere since their introduction ten years ago [10, 11] during which time continuous improvements have been made in the formulation of these compositions. Their widespread acceptance depends ultimately on their ability to provide almost complete fouling control over the now greatly extended dry-docking periods. The major difference in the mechanism of biocide release between conventional and self-polishing compositions is that the former cannot maintain an adequate leaching rate over an extended period; in the latter an approximately linear leaching rate is maintained as long as any composition remains on the surface.
The ultimate test of the performance of these materials is the behaviour of the systems in service.
As part of normal dry-docking procedure the major paint companies report on the condition of vessels, the treatment received, and the system re-applied. Fouling is always difficult to quantify but by providing staff with a carefully graded series of photographic standards, fouling on hulls can be reported quantitively in terms of type, size, density, and extent on the hull. Such data is now processed automatically on a computerised system, Dataplan, on which a large amount of comparative information has now been collected.
The fouling ratings are scaled, depending on the seriousness of the fouling to give a score, (from 100-0), representing a clean to completely fouled hull. A score of 90% or above is considered satisfactory.
Figure (10), shows the performance ratings for over 100 applications of one of the copolymer antifoulings, indicating that, at 12 months out of dock, 74% of vessels were fouling free and that 89% were considered satisfactory.
Figure (11), compares the performance of other a'f's: conventional antifouling (A), a high performance conventional antifouling (B), and a high performance chlorinated rubber antifouling (C), designed respectively for 12, 15 and 18 months service. These show that at the design lifetime only about 50% of vessels show satisfactory levels of antifouling performance. (The fact that we can still find vessels coated with these materials drydocking after twice the design lifetime, when 70-90% of them are heavily fouled, indicates the low awareness, even now, of the cost penalty of fouling in some areas of the business.)

FOULING RATING vs % OF RECORDINGS Sides U/W

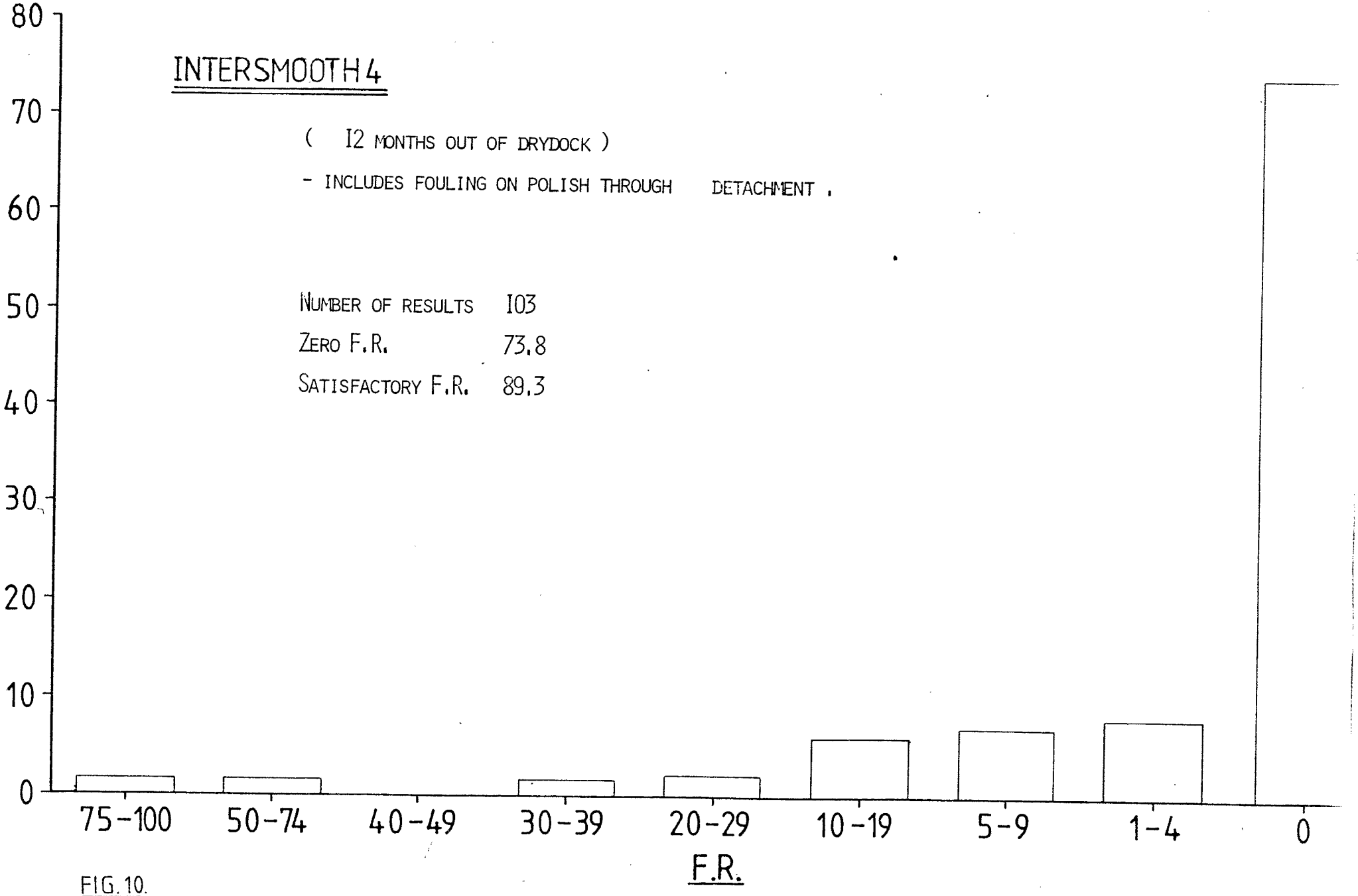


FIG. 10.

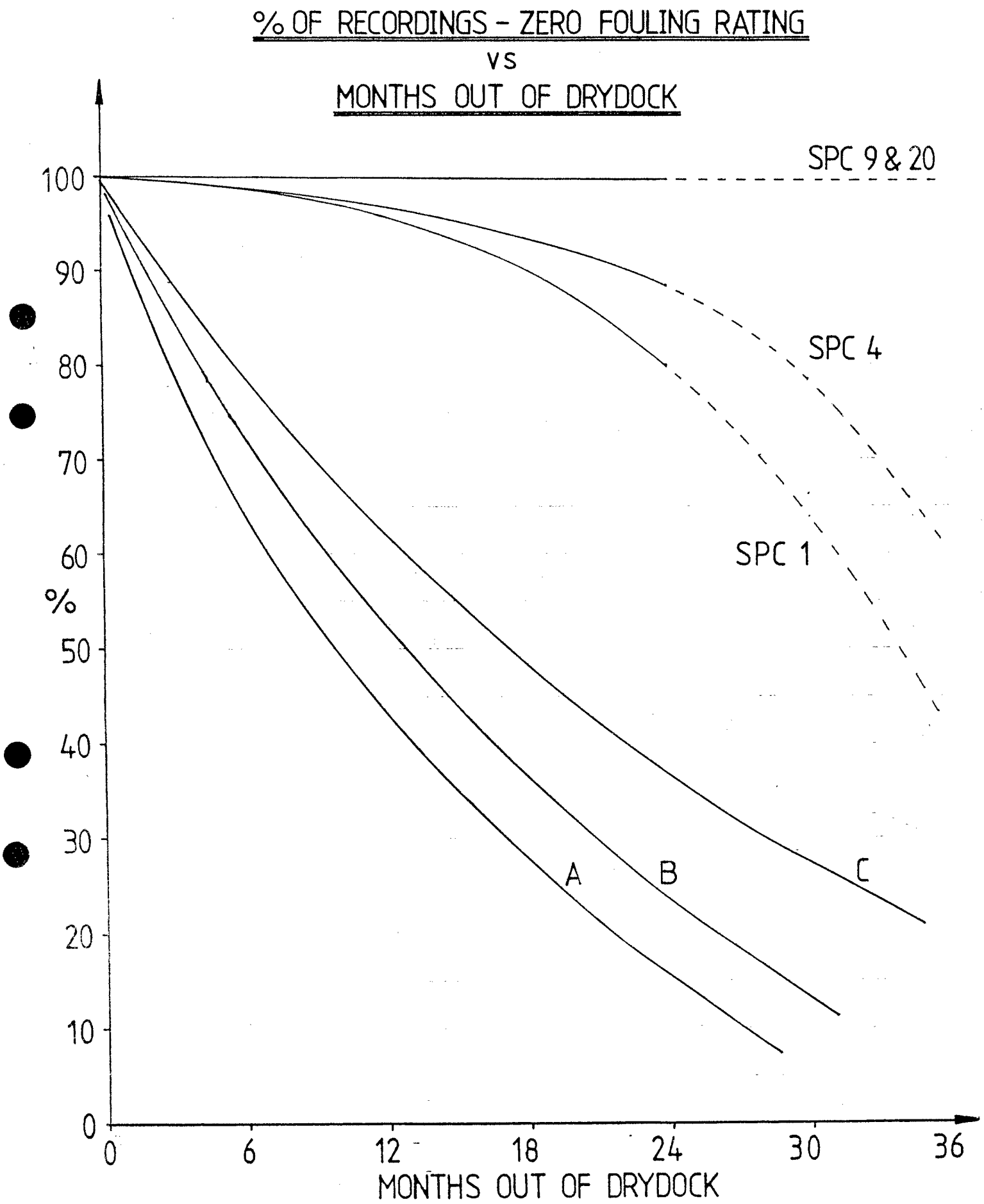

% OF RECORDINGS - ZERO FOULING RATING
VS
MONTHS OUT OF DRYDOCK
SPC 9 & 20
SPC 4
SPC 1
A
B
C
100
90
80
70
60
%
50
40
30
20
10
0
0
6
12
18
24
30
36
MONTHS OUT OF DRYDOCK

FIG.11.

5. The self-polishing antifoulings have quite a different pattern, showing excellent performance up to the design lifetime, which again was frequently exceeded by owners; the characteristic mode of failure of the self-polishing systems is the rapid onset of fouling following complete polish through to substrate, (covered in the next section), and the improving reliability of these materials is shown by the 100% success rate, brought about not only by increasing success in the control of microfouling and increasing ability to specify and control polishing rates, but also by an increasing willingness on the part of owners to carry out the necessary surface preparation to the necessary standard. It was very difficult to argue for a very high standard of surface preparation while the success of conventional antifoulings was 50% at 15 months, but for 100% success at 36 months and upward, a clean hull becomes a worthwhile investment.

6. Application of Paint Systems. Film Thickness Distribution.

In this section the distribution of film thickness is discussed mainly in terms of the distribution of thickness of self-polishing antifoulings, but, although the consequences are most conspicuous in this case, the methods of application of other coatings are similar, and similar distributions would be expected.
The need for this work followed the observation of 'premature' polish through of some of the earliest self-polishing compositions and the appearance of airless-sprayed vessels after a period at sea. The practice adopted, of painting successive coats of paint in contrasting shades, adopted for reasons of job-control in the building and repair yard, has the effect, after polishing of about one coat, that thick and thin areas become conspicuous. When the complete system of 2 - 5 coats has polished, the now non-toxic substrate fouls and this happens first where the film thickness is below specification.
The specification for a coat of paint is done in terms of a film thickness (say 100 μm), at a given application rate, and the polishing-rate will usually be given in terms of microns/month or microns per 10,000 sea-miles. It is tempting to express the lifetime of the system as:-

$$\text{Lifetime} = \frac{\text{Thickness } (X)}{\text{Rate}(y)} = \frac{x}{y} \text{ months}$$

More precisely however, we can express the probability of polish-through as:-

$$P(t) = P(x) . P(R) = \text{(Probability of a Given Thickness)} \cdot \text{(Probability of a given rate)}$$

We therefore need to know the probability distribution of paint thickness, and even more important than the 'average' film thickness, we need to know the minimum thickness, since this determines the minimum predicted life of the system.

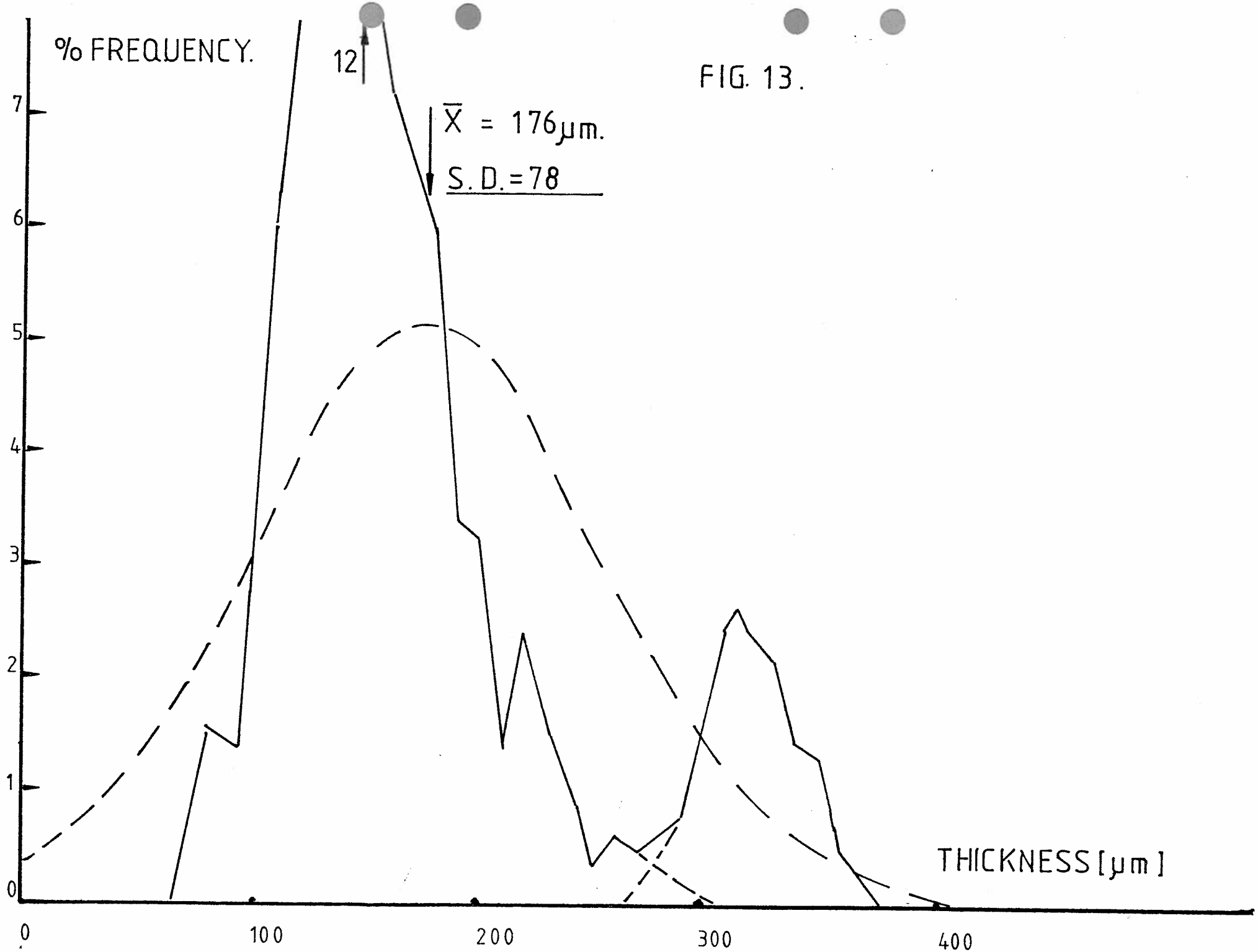

% FREQUENCY.
12
$\bar{X}$ = 176μm.
S.D.=78
7
6
5
4
3
2
1
0
THICKNESS [μm]
0
100
200
300
400

FIG. 13.

FIG. 14.

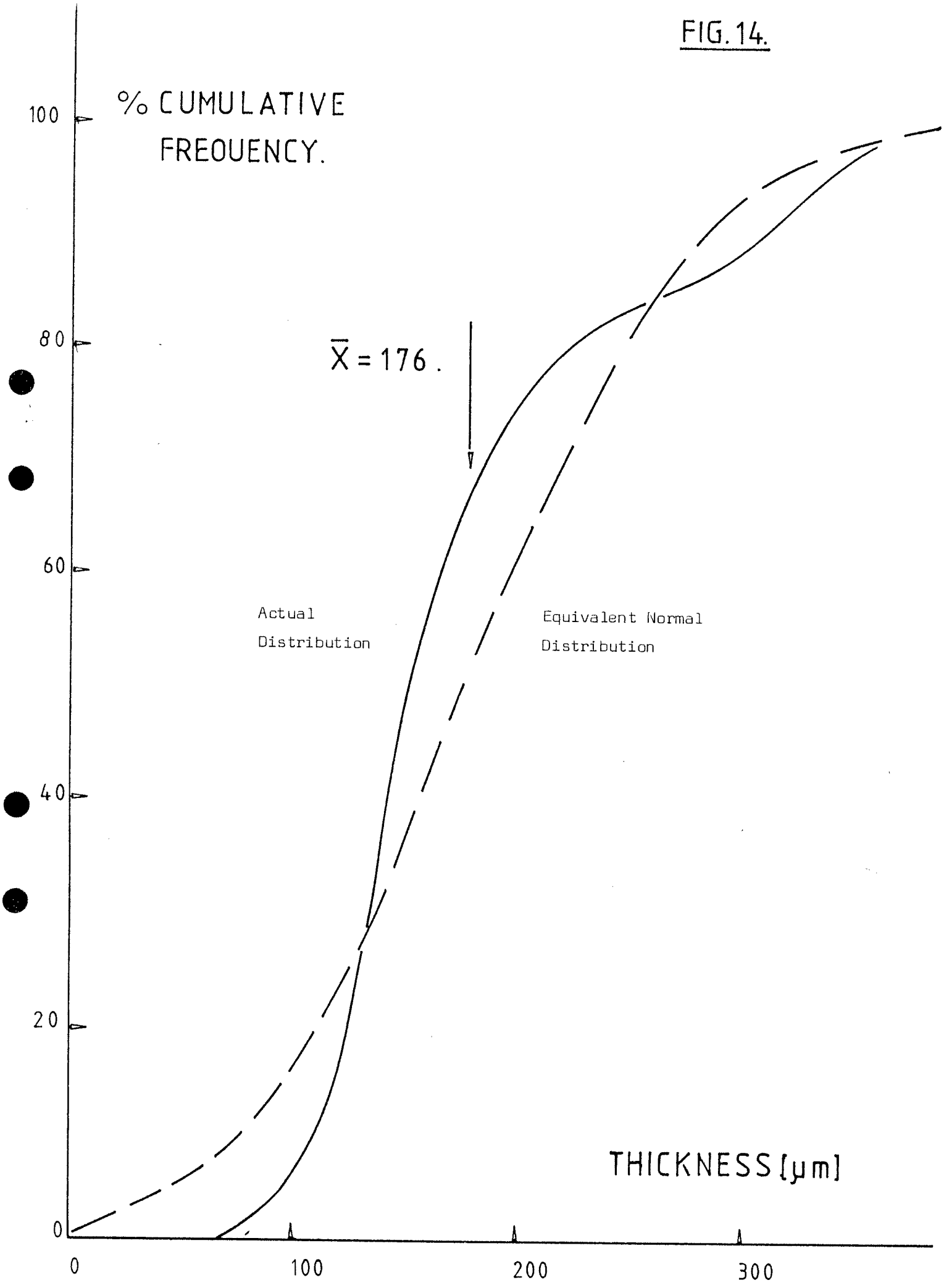
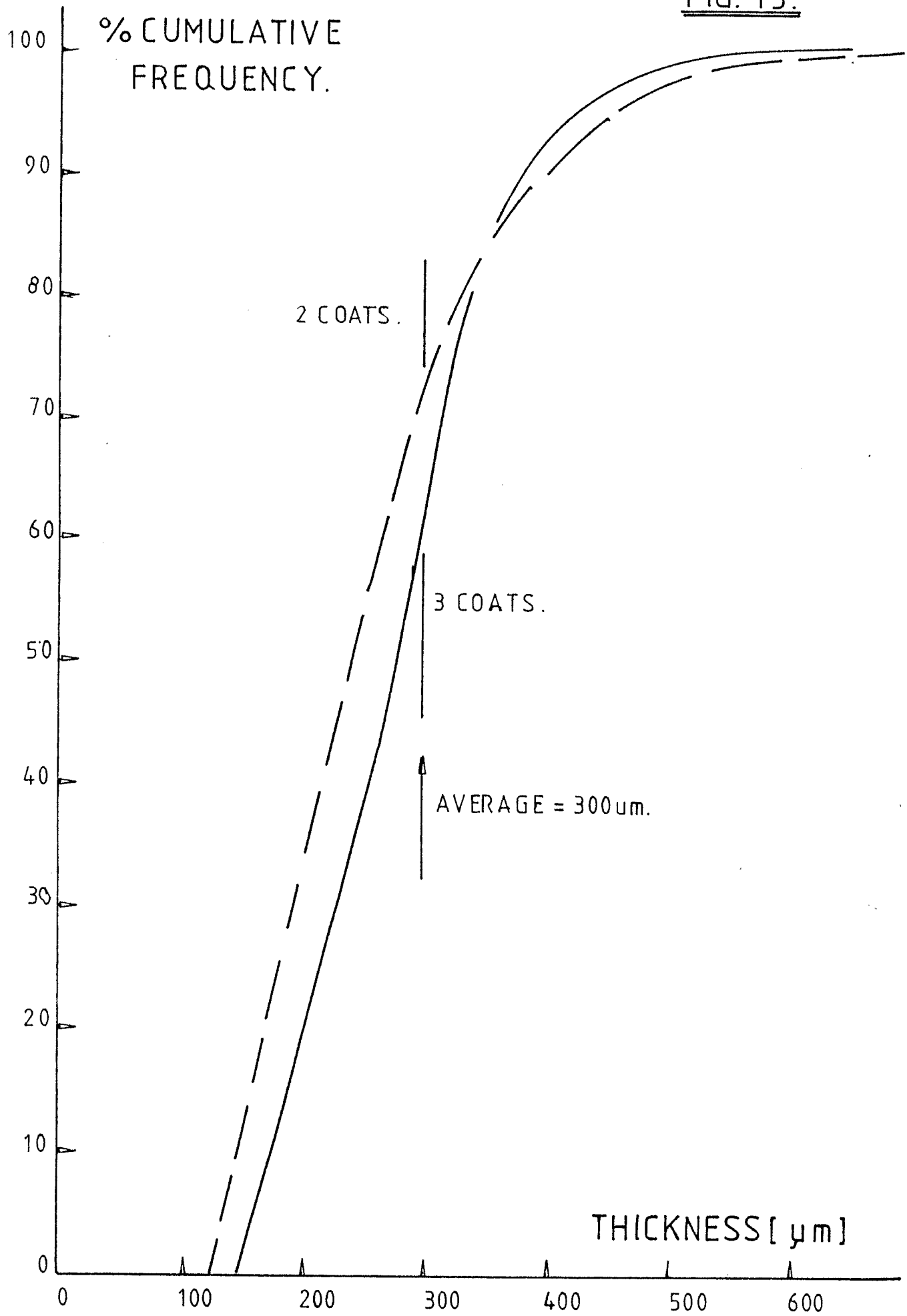

% CUMULATIVE FREQUENCY.
100
80
60
40
20
0
$\bar{X} = 176$.
Actual Distribution
Equivalent Normal Distribution
THICKNESS [μm]
0
100
200
300

FIG. 15.


% CUMULATIVE FREQUENCY.
100
90
80
70
60
50
40
30
20
10
0
2 COATS.
3 COATS.
AVERAGE = 300um.
THICKNESS [μm]
0
100
200
300
400
500
600

6. The film thickness measured is complicated by several factors. Even when we know the theoretical average thickness for a given rate of application, the 'practical' rate is subject to the amount of paint lost in the application process and the amount of solvent retention. Further complications are the roughness of substrate and the final surface, and the accelerated polishing which happens in the locus of asperities.
Most previous work in this area has been based on inadequate data, or on laboratory studies where ideal conditions exist, but usually on too few results.
The technique adopted in this work was to apply transparent adhesive tape, in lengths of 0.5 - 1.0m, randomly to the hulls of vessels. The lengths of tape were recovered, after painting, by peeling them from the hull while the paint was still plastic, usually 6 - 24 hours after application.
The film thickness was measured originally by digital micrometer after peeling the backing tape from the strip, with one measurement per centimetre. The instrument reads to the nearest micron, and was chosen for precision and speed. The instrument gives anomalously high results however, as shown in Fig. 12 . Since it measures thickness over a circular area of 8mm dia. it records, not the average thickness in this area, but the maximum thickness, i.e. it includes an element of roughness. We were therefore forced back on to the less precise (but more accurate!) and time consuming method of measurement on a microscope with graticule, although this does allow measurement of the individual coats of a multicoat system.
The experiments now cover a variety of vessels, drydocking world-wide, and over 30,000 measurements of film thickness, on both new-buildings and repair and maintenance drydockings, and show a consistent pattern.
In summary we can say that:-

1) The distribution of film thickness of non-Gaussian.

2) A given coat of airless-sprayed composition may vary in thickness from 0.4 - 2.4 of the average thickness.

3) Airless spray application is dominated by the pattern in which separate passes of the airless spray gun are overlapped.

4) The statistical distribution of thickness can be modelled by assuming a Binominal distribution.

5) The distribution is positively skewed.

6) The characteristics of airless spray application have considerable implications with respect to the efficiency and effectiveness of self-polishing systems.

The statistical model which was found to fit the data was of the form:-

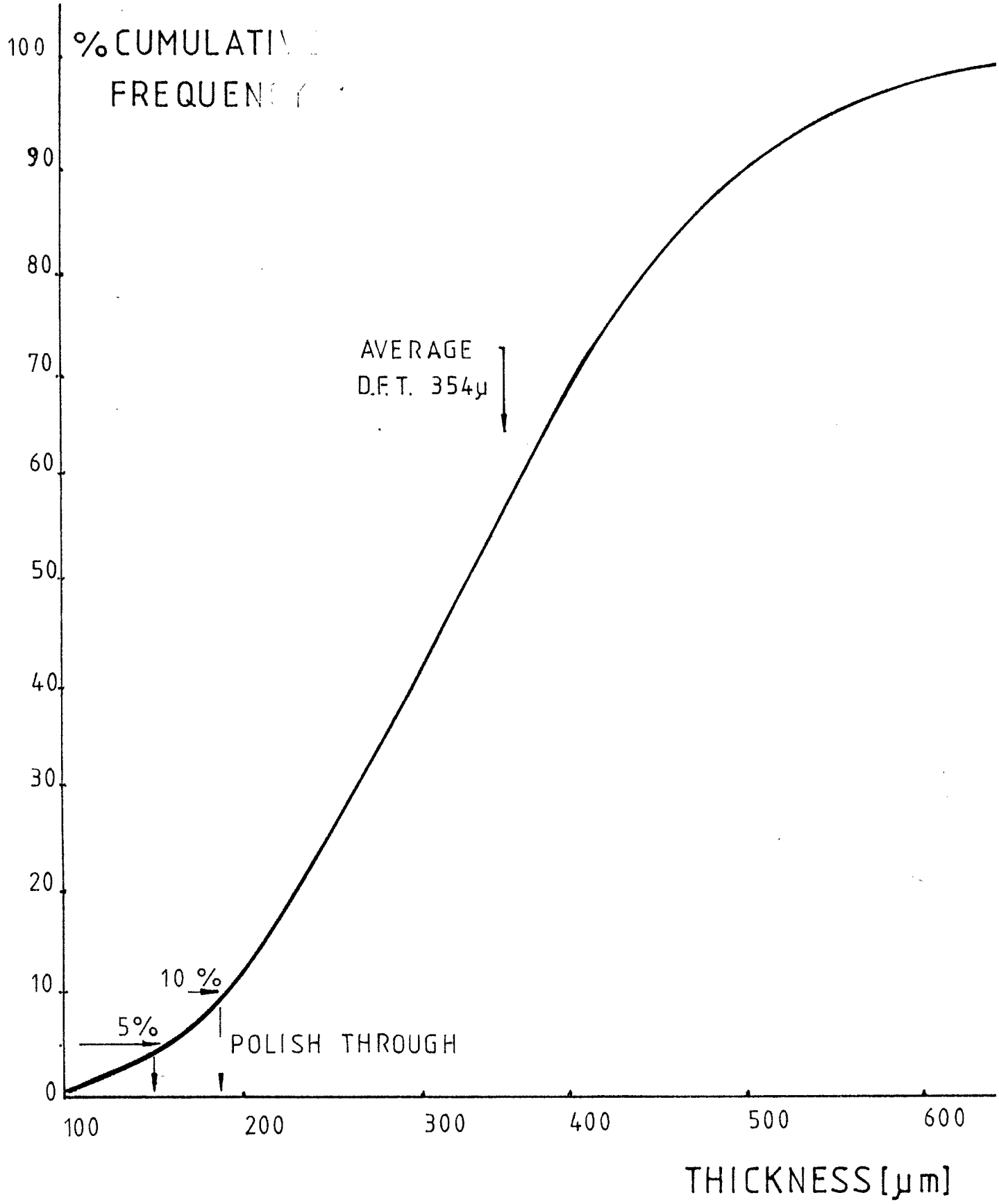
$1 = (a + b)^N$ at its simplest, where, in the course of application a is the fraction of the surface getting one pass.
b is the fraction of the surface getting two passes.
N is the number of coats.

If a_1, b_1 = horizontal overlap at each pass.
a_2, b_2 = vertical overlap at the edge of each new area.
a_3, b_3 = the way in which surplus paint is utilized by re-coating part of the area at random,

then the full model would be:-

$$1 = (a_1 + b_1)^N . (a_2 + b_2)^N . (a_3 + b_3)^N$$ where N is the number of coats.

FIG. 16.


% CUMULATIVE FREQUENCY
100
90
80
70
60
50
40
30
20
10
0
AVERAGE D.F.T. 354μ
10 %
5%
POLISH THROUGH
100
200
300
400
500
600
THICKNESS [μm]

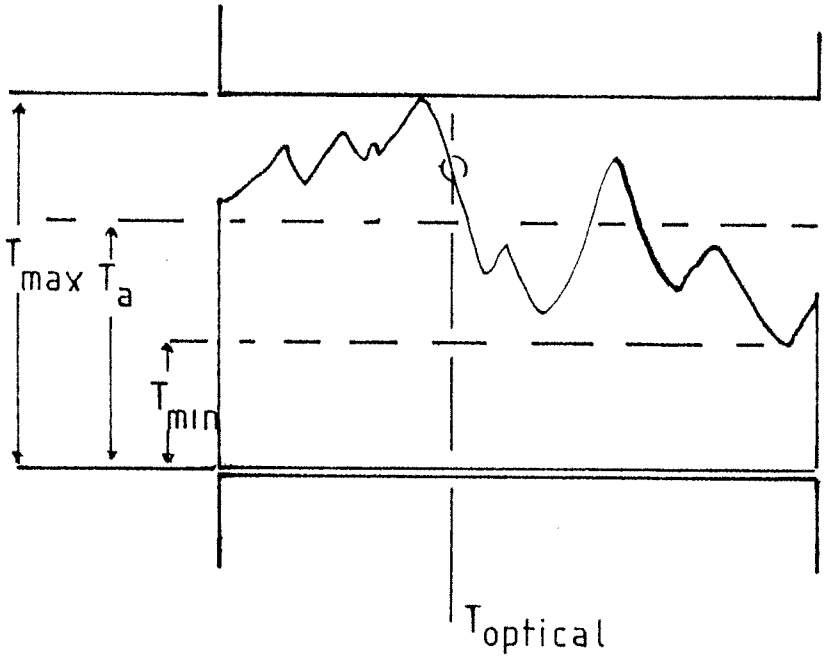


FIG. 12.

Film thickness measurement by micrometer.
In cross section the measured thickness (T max) contains an element of the roughness measured at a cut-off equal to the area of the anvil, and that the 'true' or 'average' thickness Ta, could be estimated if the roughness were known, at $R_{t(8)}$, or extrapolated from the ship roughness $R_{t(50)}$.

T_{opt} is a random measurement by microscope.
However the anvil is circular and measures the maximum height within a circular area. The two roughness elements are therefore not comparable.

6. However, provided a is large compared with b the simpler formula suffices. The Binomial distribution can also be expressed as:-

$$p(x) = {}^{n}C_{x} \cdot p^{x} (1-p)^{(n-x)}$$

Even in the course of a single coat the paint is overlagged in several ways. For example, if when we coat a vessel with a film of average film thickness 100 microns, and overlap by 25%, then 75% of the area gets 80 microns, and 25% gets 160 microns, and continue the process for four coats, then:-

$$1 = (0.75 + 0.25)^4$$

$$\text{Fractional Area} = (0.75)^4 + 4.(.75)^3.(.25) + 6.(0.75)^2.(.25)^2 + 4.(0.75)(25)^3 + (0.25)^4$$

% Area =	32	42	21	4.7	0.4
Thickness =	320	400	480	560	640

giving the positively skewed distribution of Fig.16

Even the simplest model suggests a possible variation in thickness by a factor of two.
Figs. 13 & 14 shows the actual distribution of a single coat of a self-polishing antifouling on a passenger liner which was fully staged for blasting and repainting, both as a probability distribution and as a cumulative frequency distribution, and including the distribution as the equivalent Gaussian for the same mean and standard deviation.
The effect of totally staging the vessel is to confer excellent access for the painters, avoiding the necessity of using pole guns, so a good standard of application is expected, but even here the range of thickness is made up of several overlapping populations with major overlaps at about 160 and 320 microns, with minor peaks at 80 and 240 microns confirming the binomial nature of the application. The cut-offs at minimum and maximum, relative to the Gaussian distribution, are also comparatively sharp.

Multi-coat Systems:

The minimum and maximum thicknesses are determined by the application rate per pass and the number of passes. For a system of N coats therefore, the minimum and maximum are N times the values for the single coat. What changes is not the range, relative to the mean, but the shape of the distribution, which should become less skewed as the number of coats increases.
Table 6 shows the way in which the distribution varies for a 2,3, or 4 coat system applied to the same average film thickness of 300 microns as:-

2 x 150 micron system.
3 x 100 micron system.
4 x 75 micron system.

Fig.15 shows the comparison between a two and three coat system as modelled, and Fig. 16 shows the comparison between model and actual distributions for a 2 x 150 micron antifouling system on a product carrier.

6. The major practical consequences of the positively skewed distribution and the 5:1 - 6:1 ratio between maximum and minimum thickness are the following:-

1. Because of the skewness, polish through of polishing systems takes place much more rapidly than for a normal distribution; for example, when 50% of the material has polished by volume, 70-80% may have polished by area of the hull.

2. Thickness specifications, normally quoted by average, (to the nearest micron, frequently) need to be thought of in terms of minima where this is important to the performance of the system. While this is especially so for self-polishing coating systems, where the consequences are very obvious, there is no reason to believe that the thickness distribution on other systems differs greatly. It is little wonder, therefore, that actual paint consumption so greatly exceeds the theoretical spreading rates.

3. Airless spray application utilises paint inefficiently, because of the distribution of material over such a wide range. Short of complete automation it is difficult to visualise dramatic improvements.

4. Because of the statistical basis of the model, and the assumption that the distribution is random, randomness is the best that can be achieved. Where the pattern of thin or thick areas coincide on overcoating, the distribution becomes worse, This is frequently the case, since the location and type of access, the position of blocks on flats, etc., tends to impose a pattern on the operators, so that errors are superimposed.

5. For high-build coatings better distribution is obtained by the use of narrower fans, increased overlap and cross-lapping where necessary.

TABLE 6

Binomial Thickness Distribution for 3 coat (300 µm) system in 2, 3 4 coats with 20% overlap and three overlaps

System 1 2 coats @ 150 µm				Mid Point of Class	System 2 (Std) 3 coats @ 100 µm				Mid Point of Class	System 3 4 coats @ 75 µm				Mid Point of Class
K=Passes	P(x)	ΣP(x)	K.P(x)		K=Passes	P(x)	ΣP(x)	K.P(x)		K=Passes	P(x)	ΣP(x)	K.P(x)	
0	–				0	–				–				
1	–				1	–				–				
2	.262	.262	.524	188	2	–				–				
3	.393	.656	1.179	282	3	.134	.134	.402	188	–				
4	.246	.901	.984	376	4	.302	.436	1.216	250	4	.069	.069	..276	188
5	.082	.982	.410	470	5	.302	.738	1.51	312	5	.206	.275	1.03	235
6	.015	.998	. 09	564	6	.176	.914	1.06	374	6	.283	.558	1.698	281
7	.0015	.9995	. 01	658	7	.066	.980	.462	437	7	.236	.794	1.652	328
8	–				8	.017	.997	.136	499	8	.133	.927	1.064	375
9	–				9	.003	1.000	.027	562	9	.053	.980	.477	422
10	–				10	–				10	.016	.996	.16	469
11	–				11	–				11	.003	.999	.033	516
12	–				12	–				12	.0005	.9995	.006	563
%	99.95					100.0					99.95			
	Σ K.P(x) = 3.199 = 300 µm					Σ K.P(x) = 4.813 = 300 µm					ΣKP(x) 6.396 = 300 µm			

Note: The model is of the form $1 = (a + b)^N$, of a binomial expansion; alternatively the probability of an event (in this case a pass with an airless spray gun) is $P(x) = {}^nC_x\ P^x(1 - P)^{n-x}$

The minimum probability is one pass per coat, and the additional passes or overlapping passes, and the associated probabilities are shown. The cumulative probability (the chance of an event happening or not) is = 1.
The average thickness is the sum of the discrete thickness times the probability. Average = ΣK.P(x) = 1

This gives a different total in each case, but as we want the total system thickness to be 300 µm in each case we set ΣK.P(x) = 300 and calculate the required thickness per pass from this, and the class width and interval.

REFERENCES

1. Chaplin. 'The Analysis of Hull Surface Records' European Shipbuilding 16, 40-47, 1967.

2. Byrne D.B. 'Hull Roughness of Ships in Service' M.S.C. Thesis and Paper to N.E.C.I.E.S. Junior Section. 1979.

3. Grigson C.W.B. 'Analysis of Roughness Surveys of Hulls' N.S.F.I. Report. Nov.81.

4. Cartwright & Longuet-Higgins. 'The Statistical Distribution of the Maxima of a Random Function. Proc. Roy.Soc.A. 237, 212-232,1956.

5. Ochi. 'Prediction of Extreme Values' J. Ship Res. 17,29-37,1973.

6. Musker A. 'Turbulent Shear Flows Near Irregularly Rough Surfaces with Particular Reference to Ship Hulls' Ph.D. Thesis 1978 University of Liverpool.

7. Milne A. 'Self-Polishing Antifoulings' Norwegian Ship Res.Inst. Trondheim. Jan.1982.

8. Townsin R.L., Byrne D., Milne A., Svensen T.E. 'Speed, Power and Roughness, The Economics of Outer Bottom Maintenance" Trans. I.N.A., Spring Meeting, 1980.

9. Thomas T.R. 'Recent Advances in the Measurement and Analysis of Surface Microgeometry' WEAR,33 (1975)

10. Milne A. 'Self-Polishing Coatings in Marine Antifouling Paints' Chemical Soc./Royal Inst.Chem. Annual Chemical Congress. April 1980.

11. Atherton D., Verborgt J, & Winkler, M.A.M. Journal of Coatings Technology. 51,657,1979.

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SHIP DESIGN FOR FUEL ECONOMY

Automatisation and fuel consumption

by

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Automatisation and Fuel Consumption

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Automatisation in shipping was introduced in the "sixties" to increase the economy and to enhance the safety of the ship propulsion plants.
Therefore it may be welltimed to devide up the subject in

- the term automatisation,
- the target of automatisation,
- the evolution of the fuel consumption,
- the fuel prices and its influence on shipping economy,
- automatisation and fuel consumption.

The Term Automatisation

Automatisation means the self-acting / automatic, suitable and successful operation of engines or engineplants.
Successful operation is to guarantee irrespective of changes in supposition as functions, initial conditions and disturbance variables.
Successful operation stands for optimal results as to reliability, safety and economy, and therefore for availability.
For ship propulsion plants availability is of importance in particular. Lack of availability / disposability leads to interruptions in ships time-tables, to "off-hire"-times, consequently to losses in economy; sudden lack of availability at sea may lead to further damages and to unsafeness for ship, cargo and crew.
There is no mystery to automatisation / automation, but its successful application depends firstly on the design of the system to be automated and, secondly, on the control equipment selected. The marine industry requires the benefit of a consultancy service with a marine background which will provide shipowners with guidance and advice in the selection of marine control systems and their economical justification.

The Target of Automatisation

In the 60'ies, when automatisation was introduced in shipping, the main targets of automatisation were to reduce crew and repair costs without increasing the finance costs at the same amount. The influence to the fuel bill has been of less impor-

tance at that time.

The first step in automatisation should be the setting up of a working program. Then follow the next steps:

- mechanization of operation,
- application of controlers to keep rated operation values in program sectors with correction of occurent disturbances and fina lly
- automatic correction of the whole program as to optimize the system.

The main targets of automatisation in shipping then are:

- lowering of costs
  - crew costs,
  - operation costs,
  - finance costs by better transshipment,
- improvement of transport offers,
- improvement of working conditions aboard

since 1973 a new target of automatisation is to be added:

- the use of automatisation to keep the engine conditions in their optimal ranges to run the engines with the lowest possible fuel consumption.

The Evolution of the Fuel Consumption

Because of the overwhelming of diesel driven ships in the merchant ships of the world the fuel consumption of diesel plants shall be considered. From the beginning of the development of diesel engines the fuel consumption stood in the center of the engineering efforts. So the diesel engine became the prime mover with the highest thermal efficiency.

Allreadyy the first diesel engines of Rudolf Diesel have shown low fuel consumption rates, about 50 % lower than other engines at that time:

1895: 351 g/BHPh (477 g/kWh); p_i = 7,6 kg/cm²,

1897: 238 g/BHPh (324 g/kWh); i.e. η_e = 26,2 %.

Today diesel engines for ship propulsion achieve specific fuel consumptions of about 128 g/BHPh (177 g/kWh), i.e. η_e = rd. 50%.

Fig. 1 shows the development at slow speed diesel engines of Sulzer as an example. The specific fuel consumption has been reduced by steps.

The newest generation of diesel engines is marked by the following criterias:

For 2-stroke-diesel-engines:

- uniflow scavenging,
- large stroke-/bore-ratio of about 3:1 and more,
- deep combustion chambers protecting the liner walls,
- 1 center exhaust valve,
- high injection pressure, at part loads too,
- high firing pressure,
- constant pressure turbocharging,
- very high turbocharger efficiency of about 70 %.

For 4-stroke-diesel-engines:

- larger stroke-/bore-ratios of about 1:1,1 to 1:1,4,
- deep combustion chambers protecting the liner walls,
- high injection pressure, at part loads too,
- optimization of the liner wall temperatures,
- high firing pressure,
- best turbocharger efficiency,
- good mechanical efficiency, obtained through favourable design of crankshaft and bearings, well designed frames, stiff cylinder liners, optimal piston guidance etc.,
- exhaust walves with cooled seatings or with exhaust valve baskets,
- special applications for long range part load operation, i.e. 2-stage turbocharging with heating at part load or reversing cam shaft to avoid contamination.

The success of these trends are specific fuel consumptions of 123 g/BHPh (166 g/kWh) or even optimized values of 118 g/BHPh (160 g/kWh) for ultra-long-stroke big 2-stroke-diesel-engines.

The Fuel Prices and its Influence at Shipping Economy

The oil-crises of 1973 and the price development in the following years up to now has resulted in a general trend to poorer fuel qualities inparticular in the marine field. On the other hand a general uncertainty has developed regarding the future availability of fossil liquid fuels.

The fuel prices increased extremely since 1973, see fig. 2, and constitute rising of the daily shipping costs, as shown in fig. 3.

An other paramount important aspect of fuel is its quality and availability. Since the midfifties more and more marine diesel plants are running with heavy fuel because of cheaper prices

for heavier fuels, intermediate fuels - IFO - , compared with marine diesel fuel (MDO) or marine gas oil (MGO).
So today all slow running marine engines and the bigger medium speed engines are burning heavy fuel up to a viscosity of 700 cSt at 50°C.
A proposal for the classification of marine fuels is now prepared by a ISO working group in fig.4.
The development of the mineral oil market led today to the consumption of fuels with very low qualities, as shown in fig.5.

Such residual fuels require treatment plants.
At my experience a system designed to improve cleaning of heavy fuels consists of 2 separators and a filtrator.
Due to the continuing deterioration in the quality of heavy fuels it is becoming increasingly difficult to adequately remove water and solids. Fuels with density 1,05 are not infrequent nowadays. Also, there are wide fluctuations in the solids-content, depending on the bunkering port and the fuel production at the refinery. Some separator systems can handle fluctuating operating conditions only with difficulties, see fig. 6, as well as with filtrators.
Fig. 7 shows an installation using a standard separator and a Boll + Kirch filtrator in series. In this arrangement the filtrator carries out 3 main functions:

- indication - by supervision the frequency of the filters back-flushing, an indication can be given of any malfunction (by inadequate maintenance, wrong adjustment, fluctuating solids-content of the oil due to bunkering or moving sea) in the separator,
- compensation - removal of solids and water in the event of malfunction of the separator or if the solids have a smaller or nearly the same density as the fuel and
- stand by - during maintenance of the separator the fuel is treated full-flow by the filtrator alone.

In the flow diagram in fig. 7 the filter to protect the pumps is shown as item 1; item 2 shows the pumps, which have by-passes to enable the rate of fuel delivery to be regulated. Fuel passes then through the heat exchanger (item 3) to the separator (item 4). The diesel oil separator (item 5) exists as stand by for the HFO separator. The separated heavy fuel flows into a buffer tank (item 6), necessary because fluctuating of back

pressure at the separators outlet is not invariably permissable on all separator layouts. The buffer tank is fitted with a water-probe, which both signals water in-rushes and causes the water to be drained away. The fuel then is delivered by the filtrator pumps (item 7) into the filtrators automatic filter (item 9) for fine filtration. An extra heat exchanger (item 8) can be installed.

The separator control equipment as well as the control of the filtrator is an excellent example for automatisation at ships installation. The sufficient function of any treatment in the fuel system of a diesel plant is assumption for minimum fuel consumption.

The Automatisation and Fuel Consumption

The importance of the fuel consumption for the daily shipping costs forces to minimize the fuel rates at all operations. That means the necessity to select a fuel saving plant in design and construction as well as to maintain the plant in fuel-optimal running conditions at sea.

This problem is to solve with automatisation only. Nothing but automatisation of the engine plant is enabled to keep optimal the various parameters required to obtain lowest fuel rates over long periods as usual in shipping.

Among the most important instruments used in ship's engine rooms are those needed for measuring the variables from which specific fuel consumption is calculated. These are the torsion-meter and tachometer or any other kind of power measuring system and the fuel flow meter.

Specific fuel consumption is the ratio of the hourly fuel consumption to the engine power output; $b_e = B_h/P_e$ in g/kWh.

In order to evaluate specific fuel consumption it is necessary to obtain accurate fuel flow measurements. The past practice of observing the level in a daily-use tank over a substantial period has given way to more accurate and more convenient methods of measurements. The flow meters now in general use for fuel flow applications are of the positive displacement type, which measure directly the volume of fluid passed through the meter, see fig. 8. As example the vane type flowmeter manufactured by V.A.F. is a common form of positive displacement meter. The rotor carries two vanes which pass through

$$t_1 \quad \frac{L}{c + (v \cos\alpha)}$$

and against the flow direction

$$t_2 \quad \frac{L}{c - (v \cos\alpha)}$$

L = distance between A and B
c = sonic velocity in the fluid without motion
v = velocity of the fluid in the pipe
α = angle between A/B and pipe axis

With this is

$$v = L/2 \cos\alpha \, (1/t_1 - 1/t_2)$$

$$\boxed{v = L/2 \cos\alpha \, (f_1 - f_2)}$$

A volume-meter then forms with singles from the processing unit C the actual flow in m^3 or in m^3/h or similar.

The power output of an engine can be found by measuring the cylinderpower with indicator device or continously in example with the ASEA-Cyldet-System, see fig. 10. In 1979, ASEA introduced this system which continously measures the combustion pressure in the cylinders of diesel engines. This system meanwhile is further developed to present informations in addition to the maximum (firing) pressures and time-pressure diagrams, namely mean indicated pressure p_i and cylinder load in kW or HP. Multiplying with the mechanical effiency one obtains the effective load .

The transducer, which is based on the PRESSDUCTOR^R load cell, operates continously directly against the diesel engines indicator take-off without any forced cooling whatsoever.

The principle is that a gaspressure, acting against a diaphragm having a movement of approximately 10 μm at a pressure of 150 bar, brings about a change in the permeability of the transducer. This change in permeability also brings about a change in the magnetic flux which is directly proportional to the pressure in the cylinder at any specific time.

By utilising modern technology, it is now possible to measure the pressure at any angular positionof the crankshaft provided that some form of crank-angle indicator is installed. In the CYLDET-MIP system, the rotation of the crankshaft is measured with the aid of a toothed belt and a contact-free transducer, 40 times per revolution. In addition, there is a transducer which provides a reference at the top dead centre of a given piston.

the rotor centre, mounted so that they can slide through the rotor. This assembly rotates within the meter body and, as the rotor is eccentric to the body, the vanes reciprocate in slots in the rotor. The measured fluid entering the meter exerts a force on the lower vanes causing rotation. The volume enclosed by the two lower vanes and the side of the body forms a metering chamber, and it follows that the volume passed by the meter in each revolution is four times that of the metering chamber.
The rotor spindle is used to drive a counting mechanism or tachogenerator, according to the output required. In some designs the spindle leaves the body casing through a stuffing box, and it is difficult to maintain a good seal. This problem is overcome in the VAF type instrument by means of a magnetic coupling. A permanent magnet rotor having twelve poles is mounted on the end of the spindle, and covered by a stainless steel cap. The rotation of this magnet is followed by a similar magnet which is mounted outside the pressure casing. The maximum torque which can be transmitted by this system is 4 kg/cm, although under steady flow conditions only 5 kg/cm is required. An additional advantage of this arrangement is that if a counting mechanism should seize, the magnetic transmission would slip, thus avoiding any damage to the instrument.
A very new equipment of flow meters from Danfoss is working with ultrasonic and has therefore no wear, see fig. 9. The measurement of the fuel flow is produced by ultrasonic waves transmitted from signal transformer A in fig. 9 through the fluid in the direction of the flow to the single transformer B. The transformer B is receiving the sonic impulse with a time-lag t_1. The transmission time of the sonic impulse is transformed by an oszillator to a frequence $f_1 = 1/t_1$ and then stored in the memory of the signal processing unit C. Now a next sonic impulse will be sent in the opposite direction from B to A and gives the frequence $f_2 = 1/t_2$, which also will go into C. This process repeats with a frequency of 1000 Hz.
In direction of the flow there will be

The position of the piston is then determined electronically since the crank radius and the length of the connecting rod are programmed in the central processing unit. It is then a simple matter for the microprocessor,which calculates the indicated mean pressure, to summate the pressure in the cylinder at each position of the piston. This value is then used to determine the cylinder power and for the presentation of the pressure-crank angle diagram.

The CYLDET-MIP also presents other parameters such as revolutions per minute, maximum pressure, compression pressure and firing angle, the latter in degrees. These parameters are presented in digital form directly on the operator's panel. Furthermore, the time-pressure diagram and the crank angle-pressure diagram can be displayed on the oscilloscope to enable closer studies of particularly the fuel-injection and combustion phase. Facilities can also be incorporated to enable the injection pressure or alternatively, the opening time of the fuel injection valve, to be displayed on the oscilloscope which incorporates a memory.

Parallel with the oscilloscope presentation , it is also possible to register the pressure curves on a X-Y-recorder with excellent resolution.

With this equipment of automatisation it is indeed possible to save fuel:
When the cylinders of a diesel engine are unevenly loaded, some of the cylinders will produce more horsepower than others. In a 8-cylinder engine, a difference in indicated mean pressure of 1 bar will increase fuel consumption by approxemately 1 per cent.
To run a diesel at too low a maximum pressure is undoubtedly the most costly of all methods. For each bar reduction in pressure the fuel consumption will be increased by approximately 1/4 gramme/horsepower hour.

Furthermore, if the fuel is injected too late, a considerable increase in fuel consumption will result. If, during the instant of fuel injection, the crankshaft angle deviates by 1° from the correct angle, the increase in fuel consumption will be approximately 2 %.

Example: A normal 20 000 h.p. diesel engine, running for a period of 300 days, will consume bunker oil for approximately US$ 13 000/day. A 1 % reduction or saving may not appear to be very much, but calculated over a period of one year and at an oil price of US$ 175 per ton, the savings would amount to approximately US$ 40 000.

Another example applies to a multi-engine arrangement where each shaft is driven by two engines. From the beginning, the maximum pressure and the exhaust temperatures constituted the only parameters for controlling the machinery. During a period of 6 months, three of the couplings were damaged and this caused the shipowner to direct his interest to the monitoring of the cylinder pressure. The correct distribution of the load between the cylinders is of vital importance if vibrations in the output shaft are to be avoided. By keeping the mean pressure in all the cylinders as uniform as possible, the risks of vibrations and destroyed couplings will be minimised. Since the turn of the year, an on-board pressure monitoring system has been used and no further coupling damage has been experienced. According to information received from the shipowner, each coupling costs approximately DM 50 000, to which must be added the costs for installation.

In addition to lower fuel consumption due to improved control of valve functions and injection renders better control of the preheating stage thus making the engine less sensitive to variations in the grade of fuel oil, better operational reliability due to earlier indication of faults which could otherwise result in irregular running or breakdown; therefore reducing maintenance costs and last not least easier detection of combustion irregularities through continously connected pressure transducers.

By completion the CYLDET-MIP-System with further sensors for surface temperature of the cylinder liners, temperature of the liner material approximately 10 mm from the piston side, wear monitoring and piston ring condition the system is then a cylinder monitoring system (CYLDET-M).

If the flowrate of the fuel is now compared with the shaft horsepower, the specific fuel consumption can then be derived. In addition, the distance travelled can also be determined. The numerical value can then be presented as litres/nautical mile, a value which should be minimized on all vessels. This is an easy matter on vesssels with CP propellers where the pitch and speed can be adjusted to minimize fuel consumption without influencing the speed of the vessel.

The specific fuel consumption is given by the makers in accordance with the ISO ambient reference conditions:

Air temperature before charger 27° C,
Pressure before charger 1000 mbar,
Temperature of cooling medium
for charge air cooler 27° C,
Caloric value (lower) 10200 kcal/kg
(42700 kJ/kg).

On the other hand the engine powers are valid up to tropical conditions at sea level:

Ambient air temperature 45° C,
Ambient air pressure 1000 mbar,
Charge air coolant temperature 32° C.

These datas are also according to ISO standard 3046/I.

The other method to obtain the effective load of a diesel engine is to measure shaft torque and shaft revolutions. The majority of torque meters rely on the angular deflection of the shaft over a known distance, this being directly proportional to the torque transmitted. The constant required to establish the torque/angular deflection relationship is determined by the shaft diameter and the modul of rigidity of the shaft material.

As an example for a power-, torque- and revolution-meter a system of Hoppe-Bordmeßtechnik, Hamburg, may be shown in fig.11.

This system operates without any brushes on the shaft. The torque angle of about 1° is first of all multiplied by the tooth gear to 8 to 15 fold, the shaft revolutions thereby transmitted without losses to small alternators giving the measuring currents, which deliver power, torque and revolutions according with fig. 12.

The methods currently used for shaft speed/revolution measurement are fairly straightforward. Two methods have been shown in fig. 10 and 11.

Where a more representative indication of specific fuel consumption is required the relevant data may be fed into a data logger or other computing system to record the results as an analogue trace or as a series of digital entries. In application of satellite communication it may be possible to deliver such data directly to the desk of the superintend engineer of a shipowner.

The further development of control systems for fuel consumption has to include the ship itself with its variables.

In this way operates e.g. the ASEA Economy-Sailing system. Practically all ships operate with fixed voyage times, it has been shown. Normally, to ensure a safety margin, the vessel is run at a high rate of knots during the initial part of its voyage and gradually slows down so that the time of arrival is as planned. This method of propelling the ship results in uneven load take-off, with unnecessrily high fuel consumption.
Fig. 13 shows these coherences. Instead of accomplishing the distance in T hours at a speed of 15 knots, two thirds of the distance is covered at a speed of 17 knots und one third at a speed of 11 knots, in which case the travelling time will be the same as the nominal time. Calculations show that when a ship is operated in this manner, the power consumption will be much higher, in this case 10 % higher than necessary.
Thus, if the load take-off can be kept constant, there is a good chance of saving considerable amounts of fuel.
ASEA's Economy Sailing system directs the ship economically from port to port taking into account voyage time and outside factors affecting the ship.
A number of outside factors influence fuel consumption during a voyage: Weather condition, wave height, current, depth below keel, trim of ship, water temperature, etc.. In order to run economically, with a constant load take-off, and so as to be able to select the most favourable route with regard to prevailing conditions, it is essential to be able to calculate the effect of the factors listed.
ASEA's Economy Sailing system, which the company has developed in co-operation with Salén Technologies (SALTECH), provides this facility. A demonstration will be given to the participants.

The system consists of a microprocessor unit on line with sensors that provide the necessary data.
It can be connected to the ship's remote control system for automatic control. Economy sailing includes two main programmes: prediction of arrival and route planning.
The computer calculates the most favourable average power at which the ship should be operated during the voyage to arrive at the correct time.
Route planning means that the microprocessor calculates the alternative routes using prevailing conditions for wind, current, shallow water, etc., and indicates the most economical route.
If a ship is fitted with Economy Sailing, significant fuel savings can be made. Investment costs can be covered within a year.
Considerable fuel savings are possible if a ship operates at correct trim for each speed and loading condition. ASEA and SALTECH have together created a system - OPTI-TRIM - that helps to operate a ship at correct trim.
A few minutes in front of the screen can be save a great deal of fuel.
OPTI-TRIM tells in figures how many tonnes of fuel/day can be saved if the ship is loaded to optimum trim at a given speed; helps to plan the loading of cargo, the bunkering of fuel oil and fresh water and the correct positioning of ballast in order to make maximum fuel savings; tells how to bunker fuel in the most economical manner. OPTI-TRIM is based on ship data specific for each ship or ship series.
The trend towards more automation in ship's engine rooms is proceeding at a moderate pace and the marine engineering industry is taking various steps to align itself to the needs of the future.

A [illegible] system was developed by BP, it only consists of two documents - an [illegible] manual and [illegible] the [illegible] the [illegible] calculator is used to collect engine data and how model curves are used to monitor performance, performance trends, and fault findings, are also introduced. It has been found that, on new sister ships, the engine parameters - assumed to be similar - are in fact quite different. Using a base model these differences [illegible] adjustment of timing and fuel system components.

The need for a monitoring device thus became apparent. But a monitoring system should be simple. Examples of service result in the BP fleet from an engine over 2000 h monitoring period show a clear performance improvement. From an inital 11 % deviation from the fuel consumption index curve, performance was improved to give a 5,4 % deviation.

A further possibility to save fuel is to introduce an adaptive autopilot. Adaptive head control devices are increasing in merchant fleets; such autopilots are advantageous, because the head influencing attributes of a ship are variable. Such attributes are wind, sea and eccentric thrust of the propeller.
It is the task of the autopilot, the head control, to counteract these disturbances and to hold the wanted head as exact as possible.
The output singnal of the autopilot influences the steering gear, moving the rudder of the ship. Through the rudder the ship will keep a torque and will go back to the wanted head. But by this an additional resistance will be caused and the ship speed will descend. With an optimal justified autopilot the additional resistance will be minimized. The behaviour of a ship changes continously through different draughts, increasing roughnes of the hull, effects of wind and sea. For an optimal operation it is therefor necessary, to syntonize the autopilot continously. That can be done automatically by an adaptive device.
The new requirement of today is not only to optimize the head steadiness but also to minimize the fuel consumption for a given distance. At the same time the ships speed should be uniform. That means that the disturbances specified before shall be balanced by higher engine outputs. For this target a control circuit becomes superposed over the head control circuit of the autopilot. This superposing control minimizes the momentary fuel consumption, measured from engine fuel rack position and speed, and keeps the ships speed constant influencing the engines fuel rack, see fig. 14.
In this fig.

F means the position of the fuel rack,
V_s means the pretended ships speed,
V means the actual ships speed
ψ means the momentary (actual) head angle,
ψ_{soll} means the pretended head angle,

Füllungsnachführung	means the fuel rack adjust,
n	means the speed of the main engine,
Hauptmaschine	means the main engine,
M_d	means the actual propeller torque,
Kursregler	means the head control device - autopilot,
δ_{sw}	means the adjust single of the autopilot,
δ	means the rudder angle,
Rudermaschine	means the steering gear,
Schiff	means the ship,
K	means the transfer factor for the ship in the model of Nomoto, K. (Response Analysis of Manoeuvrability and its Application to Ship Design, 60th Anniversary Series, Vol.11, Society of Naval Architects of Japan),
T_I	means intregation time of the circuit,
T_D	means differentation time of the circuit.

Scientific research works to this problem are running at the Institute for Ships Design at the University Hannover(Prof. Dr. Droste, Prof. Dr. Keil). Here in this lecture it only was my intention to draw the attention to the special connection between automatisation and fuel consumption.

Just now a large research program "Ship of the Future"is running in Germany. One of the research subjects is the developement of an intregrated navigation system - INA - . Responsible for this is TELDIX in Heidelberg / Germany. Also with this new developement it should be possible to form the required input singnals to an adaptive autopilot, see fig. 15.

Besides control and regulation just as important is supervision. Fig. 16 shows a modern supervision system from Siemens. This system is built up in a hierarchic manner. Each level, shown in fig. 16, works on top of the foregoing level. Level A and B are forming a conventional supervision system, nevertheless modern built up with microprocessor technique. Level C gives possibilities for remote representation of the data on screens and loggers or printers. From the keyboard at the screens one can choose the required data in special cases.

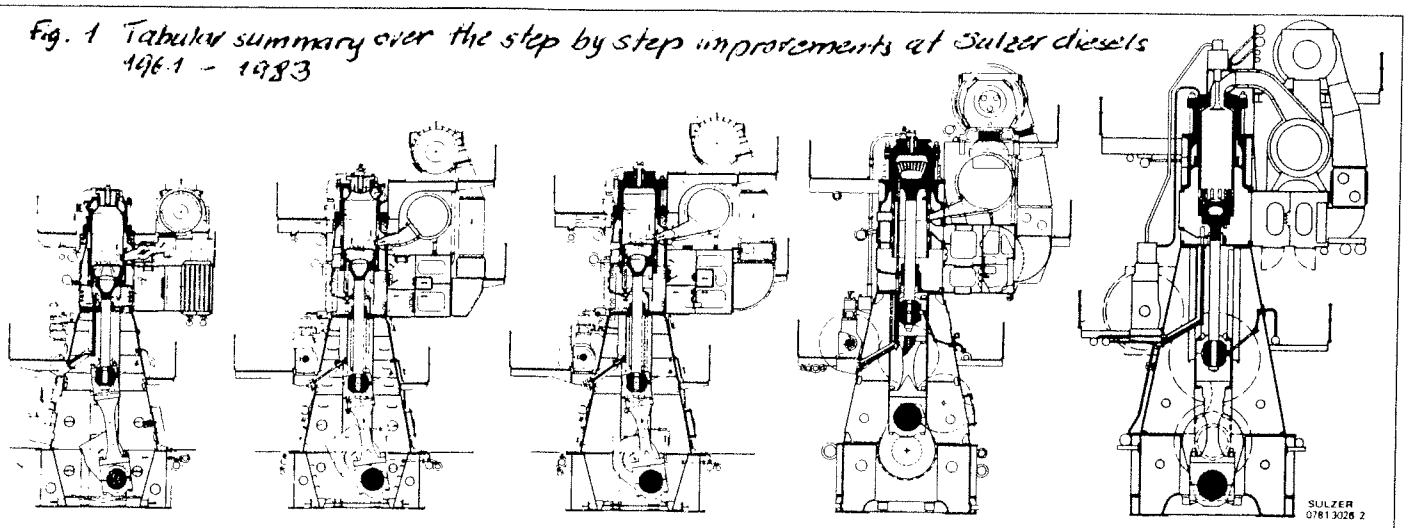
The sensors are at the engines in the engine room. There or in an engine control room may be the different measuring stations with their actual informations.

In level D is given a possibility to connect the engine room supervision system with an overplaced ships supervision system, e.g. with INA.

So a very high degree of automatisation including supervision will be obtained for future ships.

Finally it can be said that the automatisation contributes to minimize the fuel consumption.
The present state of automatisation in the engine room today is the unmanned operation. The control devices may be centraliced or decentraliced. The microprocessor is "going" aboard. Ship operation including engine, navigation, cargo and deck machinery will covered in the future by computer application.
More computer technique aboard must however lead to better working conditions, to more safety aboard for crew, ship and chargo, to increasing economy and finally to better shipping and more reliability of ships.

Fig. 1 Tabular summary over the step by step improvements at Sulzer diesels 1961 - 1983



Motorenbezeichnung	RD 90	RND 90	RND 90 M		RLB 90		RTA 84		
Baujahr	1961	1969	1976	1979	1980		1983		
Mittlerer Zylinderdruck	8,65	10,64	12,29	12,29	14,31	14,79	15,35	13,53	(bar)
Höchster Zylinderdruck	76	84	92	94	118	118	125	125	(bar)
Brennstoffverbrauch	208	208	203	193	182	184	173	167	(g/kWh)
Zylinderleistung	1692	2133	2465	2465	2940	2800	2960	2100	(kW/Zyl.)
Drehzahl	119	122	122	122	102	94	87	70	min^{-1}
Bewertungspunkte	MCR	MCR	MCR1	MCR1	MCR1	MCR2	R1	R4	

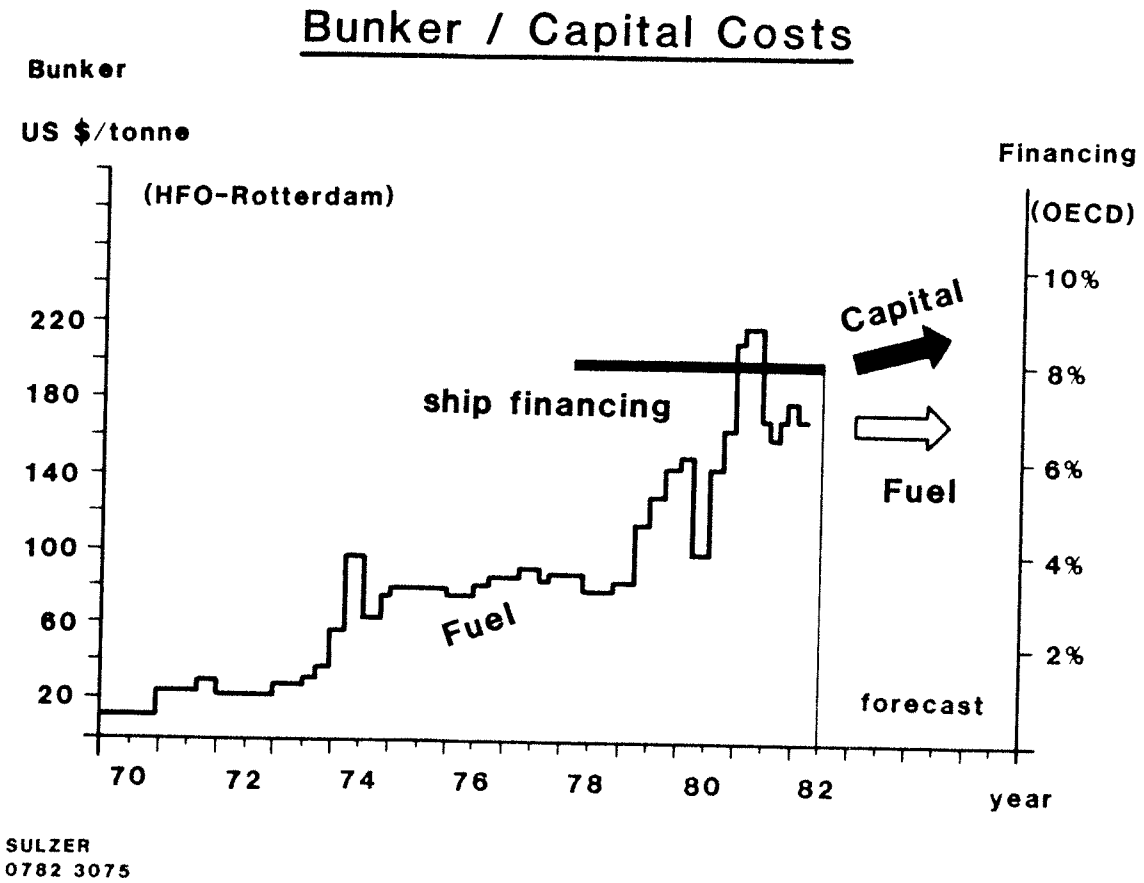


Fig. 2: Rising fuel costs 1973 to 1981.
In the course of the last year, fuel has become relatively less expensive, in comparison with capital costs.

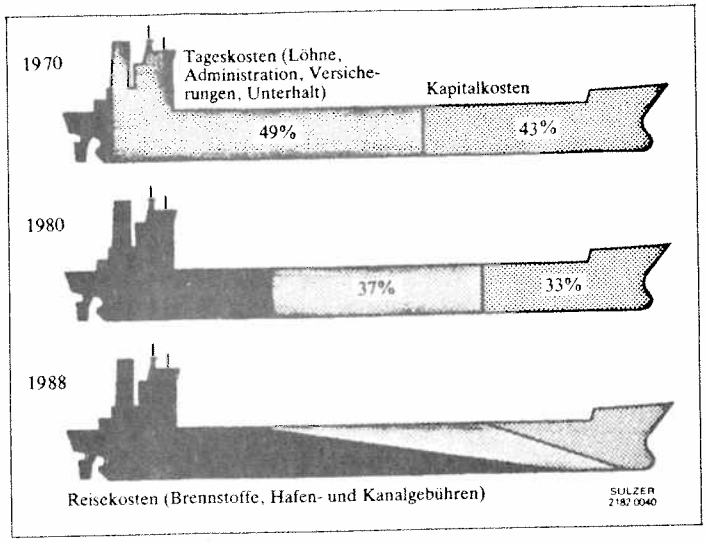


Fig. 3 Fuel- and harbour costs, crew-, insurance- and maintenance-costs, finance costs 1970 - 1980 - 1988

Klassifikationsvorschlag für Marine-Brennstoffe (ISO TC 28 / SC 4/ WG6 - Doc. N 56 (v. Juli 1982))

Type of fuel	Symbol		Remarks
	Class	Category	
Distillate fuel	ISO-F-	DM-X	Emergency purposes external to the machinery spaces
Distillate fuel	ISO-F-	DM-A	General purpose;shall contain no residuum
		DM-B	General purpose; may contain a trace of residuum
		DM-C	General purpose; will contain some residuum
Residual fuel	ISO-F-	RM-10A RM-10B RM-10C RM-15D RM-25E RM-25F RM-35G RM-35H RM-45H RM-55H	Limited density
Residual fuel	ISO-F	RM-35K RM-35L RM-45K RM-45L RM-55L	Unlimited density

Fig. 4 ISO Proposal for Marine fuel Standarts

Fig. 5

ISO - BSI - CIMAC Residual Marine Fuel Standards

Status as of July 1982

	Category															
ISO-F-RM	10A	10B		10C	15D	25E	25F	35G	35H	35K	35L	45H	45K	45L	55H	55L
BSI			M4		M5		M6		M7		M10	M8		M11	M9	M12
CIMAC	4	3		2	5	6	7		8	9		10	11		12	

Viscosity cSt 80 °C	15				25	45		75				100			130	
Viscosity cSt 50 °C	40				80	180		380				500			700	

Inspection		Limits															
Density 15 °C, kg/1	Max.	0.970	0.991			0.991	0.991		0.991		—	0.991		—	0.991		—
Kinematic Viscosity 100 °C, cSt	Max	10				15	25		35			45			55		
Flash Pt. °C	Min	60				60	60		60			60			60		
Pour Pt. (Upper) °C 1 Dec-31 March	Max	0 1)	24			30	30		30			30			30		
1 April-30 Nov	Max	6 1)	24			30	30		30			30			30		
Carbon Residue, Conradson, % Mass	Max	10		12	14	14	15	20	18	22	—	22		—	22		—
Ash, % Mass	Max	0.10				0.10	0.10	0.15	0.15	0.20		0.20			0.20		
Water, % Volume	Max	0.50				0.80	1.0		1.0			1.0			1.0		
Sulfur, % Mass	Max	3.5				4.0	5.0		5.0			5.0			5.0		
Vanadium, mg/kg	Max	150		250	300	350	200	500	300	600		600			600		

Total Existent Sediment	Max	2)				2)	2)		2)			2)			2)		
Ignition Quality	Min	2)				2)	2)		2)			2)			2)		
Aluminum	Max	30 3)				30 3)	30 3)		30 3)			30 3)			30 3)		

1) Purchasers should ensure that this pour point is suitable for their trading pattern, especially if the vessel is trading in both the Northern and the Southern hemispheres.
2) Considered important, but no limits have been set pending the development of a suitable method of test.
3) Pending development of a method of test.

Treatment unit / Requirement	Separator	Filtrator	Separator-Filtrator in series
Separation of the solids in the case of porous FCC residues		X	X
Removal of water when salts are present γ oil < 1.06 γ oil > 0.99	X	X	X
Removal of water at γ oil > 1.06		X	X
Only slight accumulation of sludge	X		X
Adaptability to solids content of oil entering		X	X
Liability to faults due to degree of mechanisation (min)		X	X
Low cost of spares and maintenance		X	X

Fig. 6— The problems which can arise both with separators of current designs and filtrators.

Fig. 7 — An installation using a standard separator and a Boll filtrator in series.

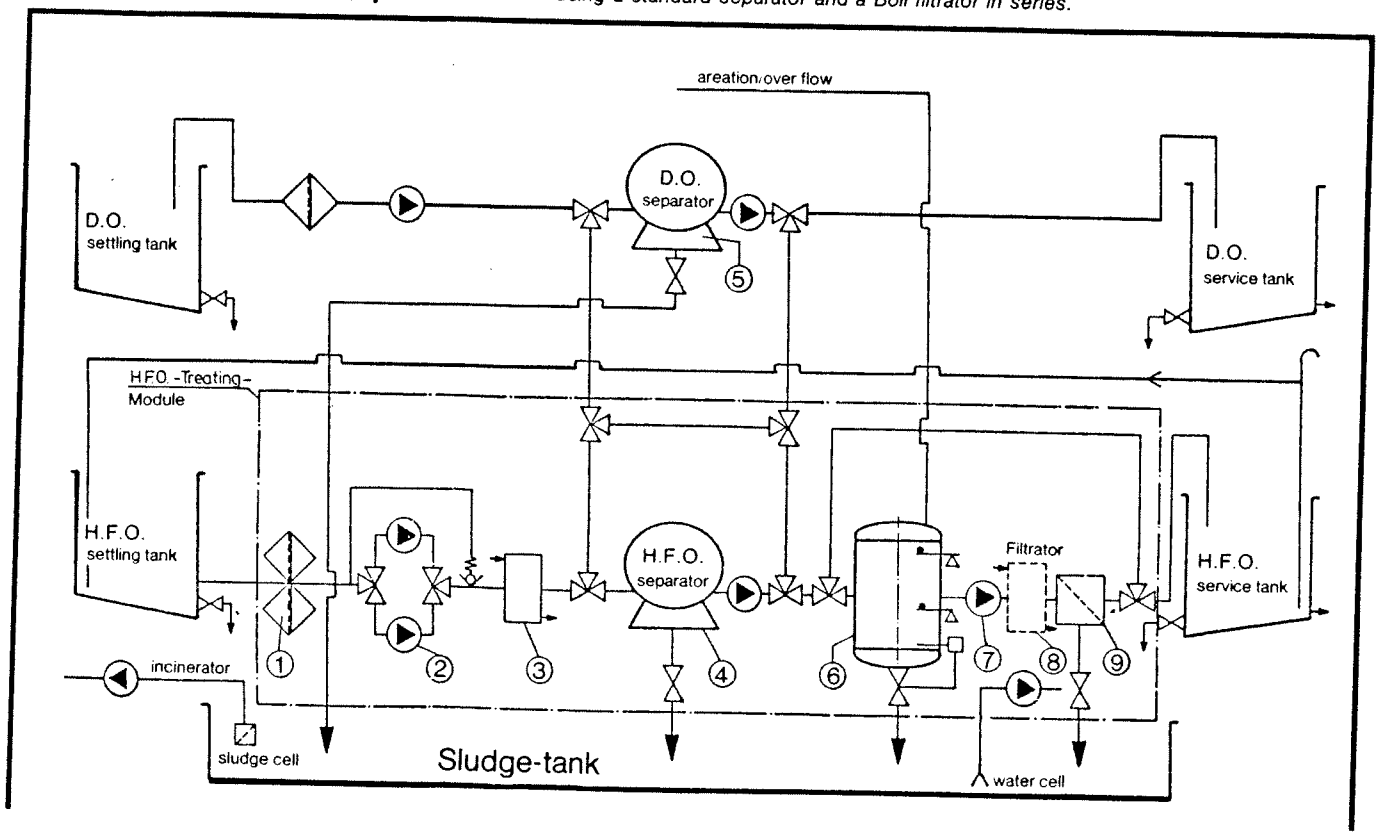
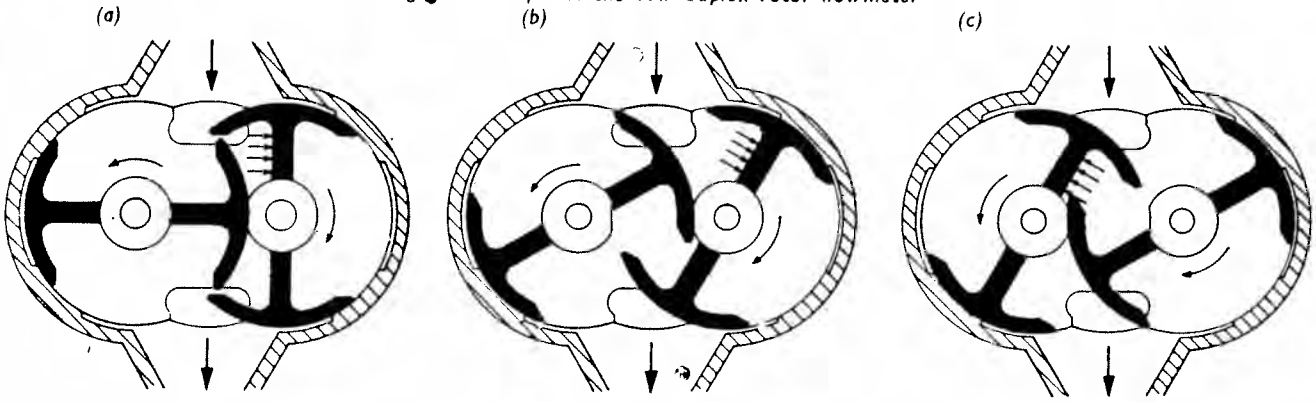


Fig. 8 : Principle of the VAF duplex rotor flowmeter



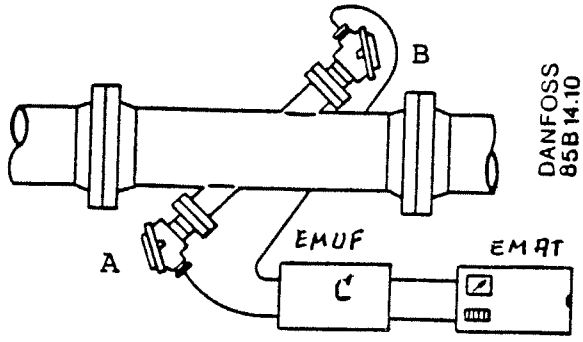


Fig. 9 Danfoss flow measuring pipe with flow meter EMFU and volume meter EMAT

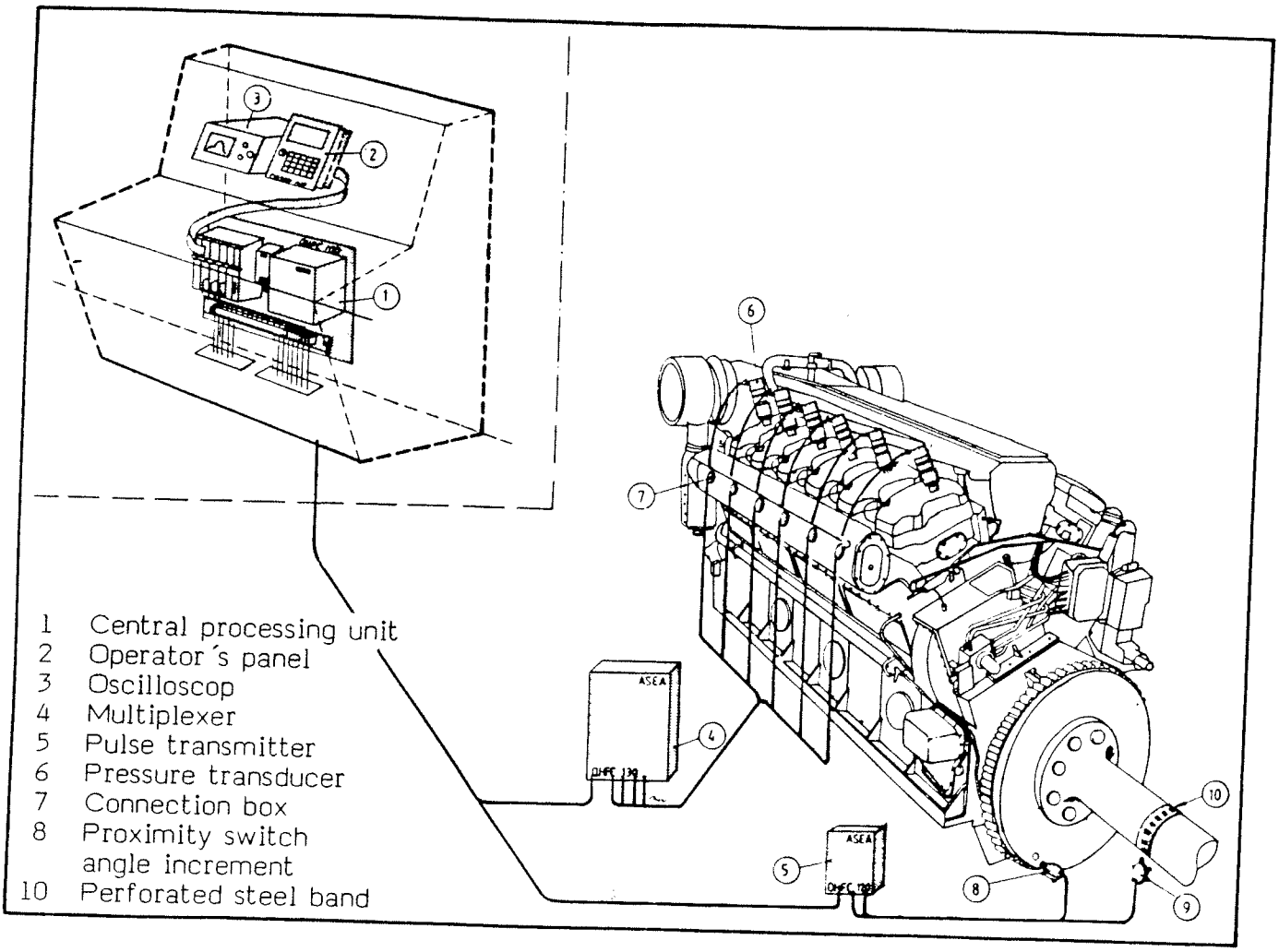


Fig 10 ASEA - Cyldet MIP - System

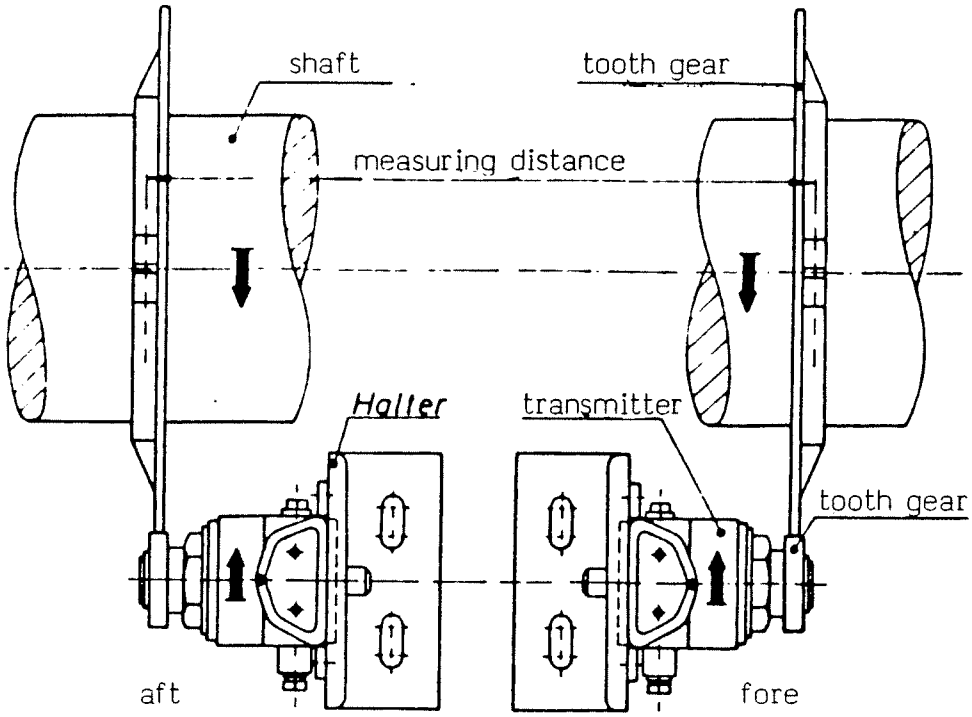
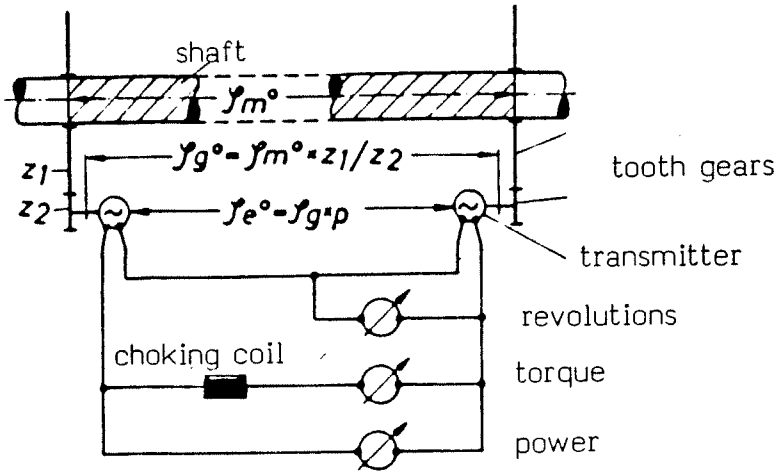


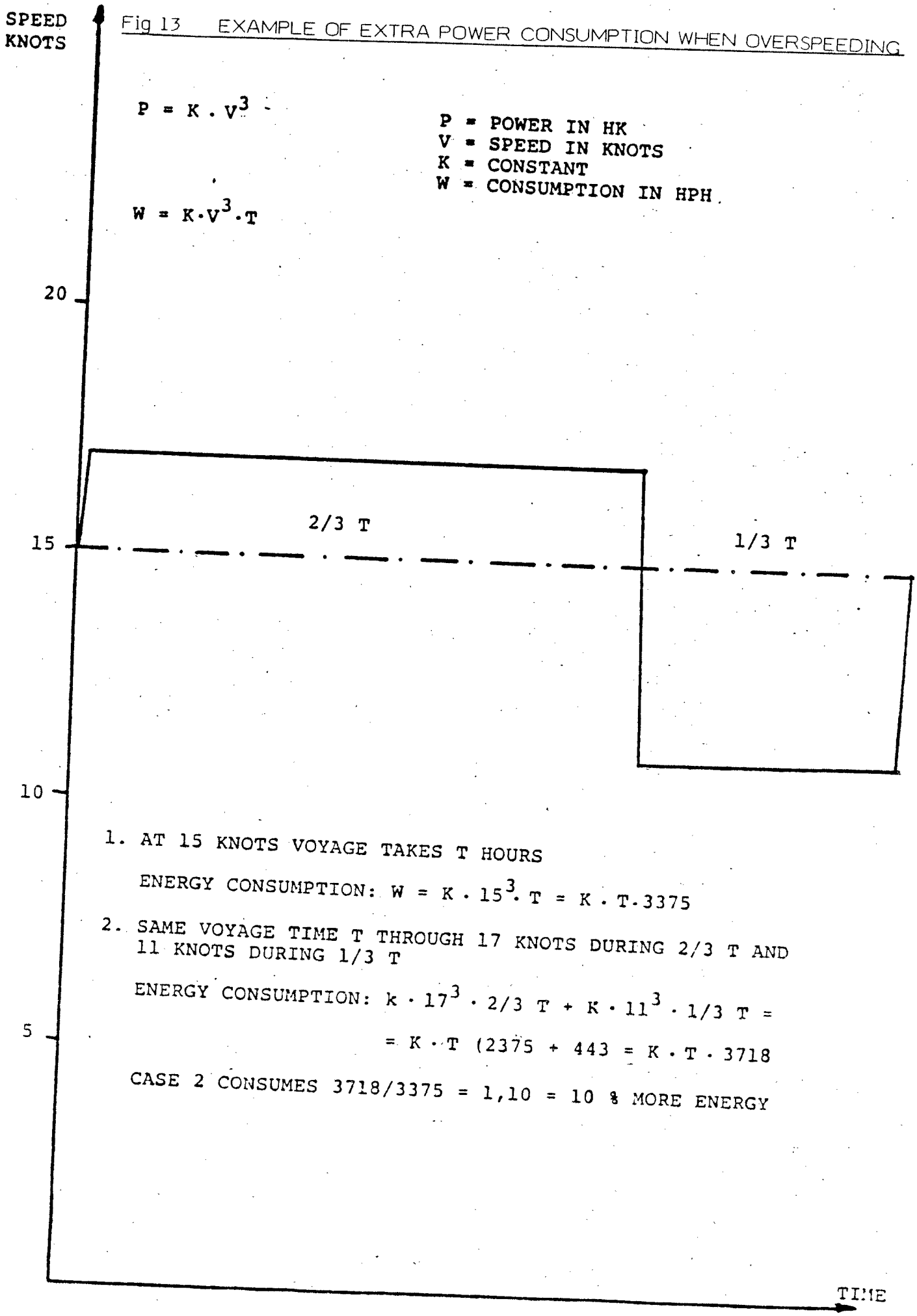
Fig 11 Hoppe - Power - Meter (System)



φm - torsion angle of the shaft
φg - torsion angle of the transmitter
φe - torsion angle of the phases

Fig 12 Wiring diagramm

Fig 13 EXAMPLE OF EXTRA POWER CONSUMPTION WHEN OVERSPEEDING



$P = K \cdot V^3$

$W = K \cdot V^3 \cdot T$

P = POWER IN HK
V = SPEED IN KNOTS
K = CONSTANT
W = CONSUMPTION IN HPH

1. AT 15 KNOTS VOYAGE TAKES T HOURS

   ENERGY CONSUMPTION: $W = K \cdot 15^3 \cdot T = K \cdot T \cdot 3375$

2. SAME VOYAGE TIME T THROUGH 17 KNOTS DURING 2/3 T AND 11 KNOTS DURING 1/3 T

   ENERGY CONSUMPTION: $k \cdot 17^3 \cdot 2/3\ T + K \cdot 11^3 \cdot 1/3\ T =$

   $= K \cdot T\ (2375 + 443 = K \cdot T \cdot 3718$

   CASE 2 CONSUMES 3718/3375 = 1,10 = 10 % MORE ENERGY

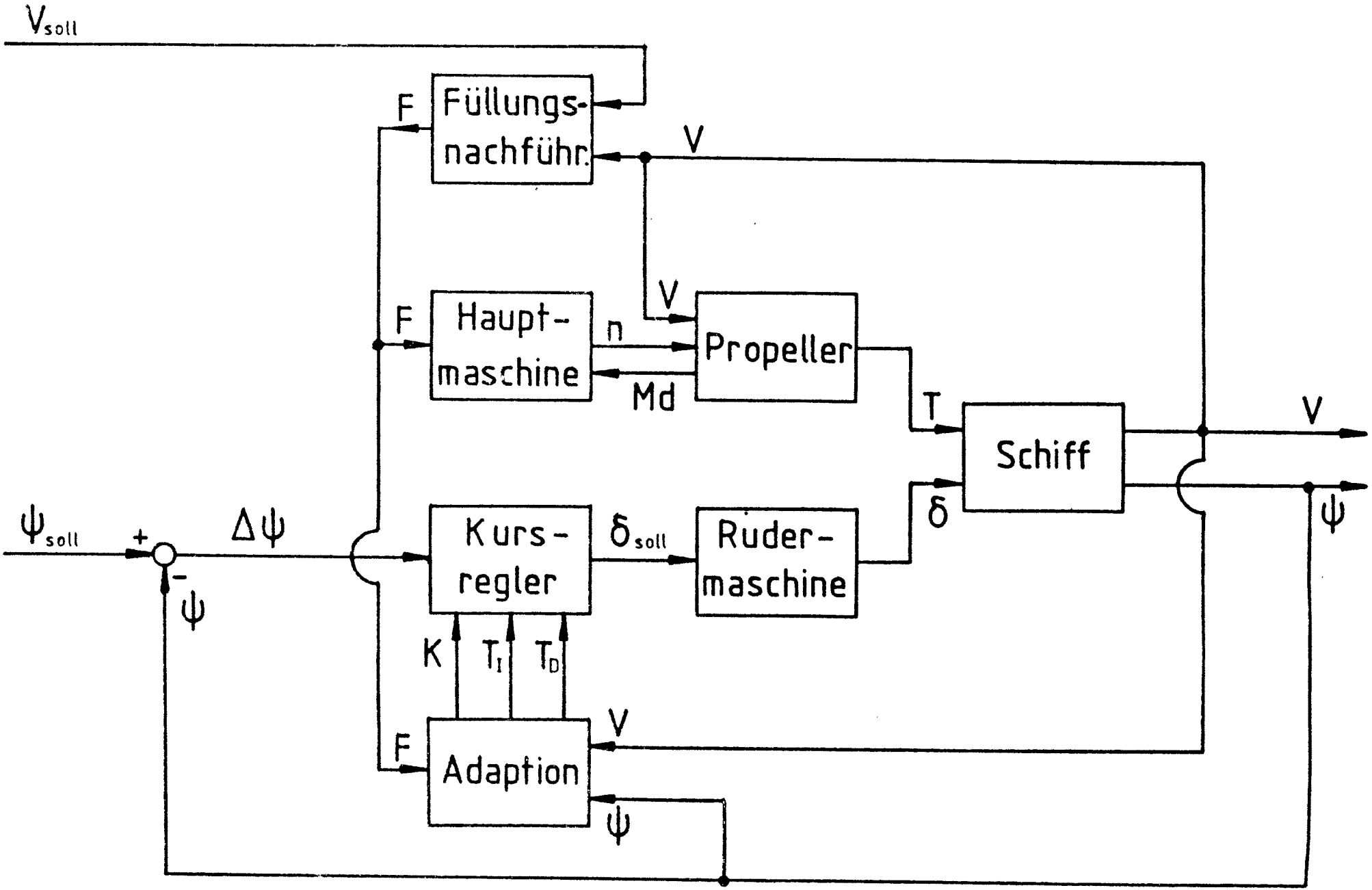
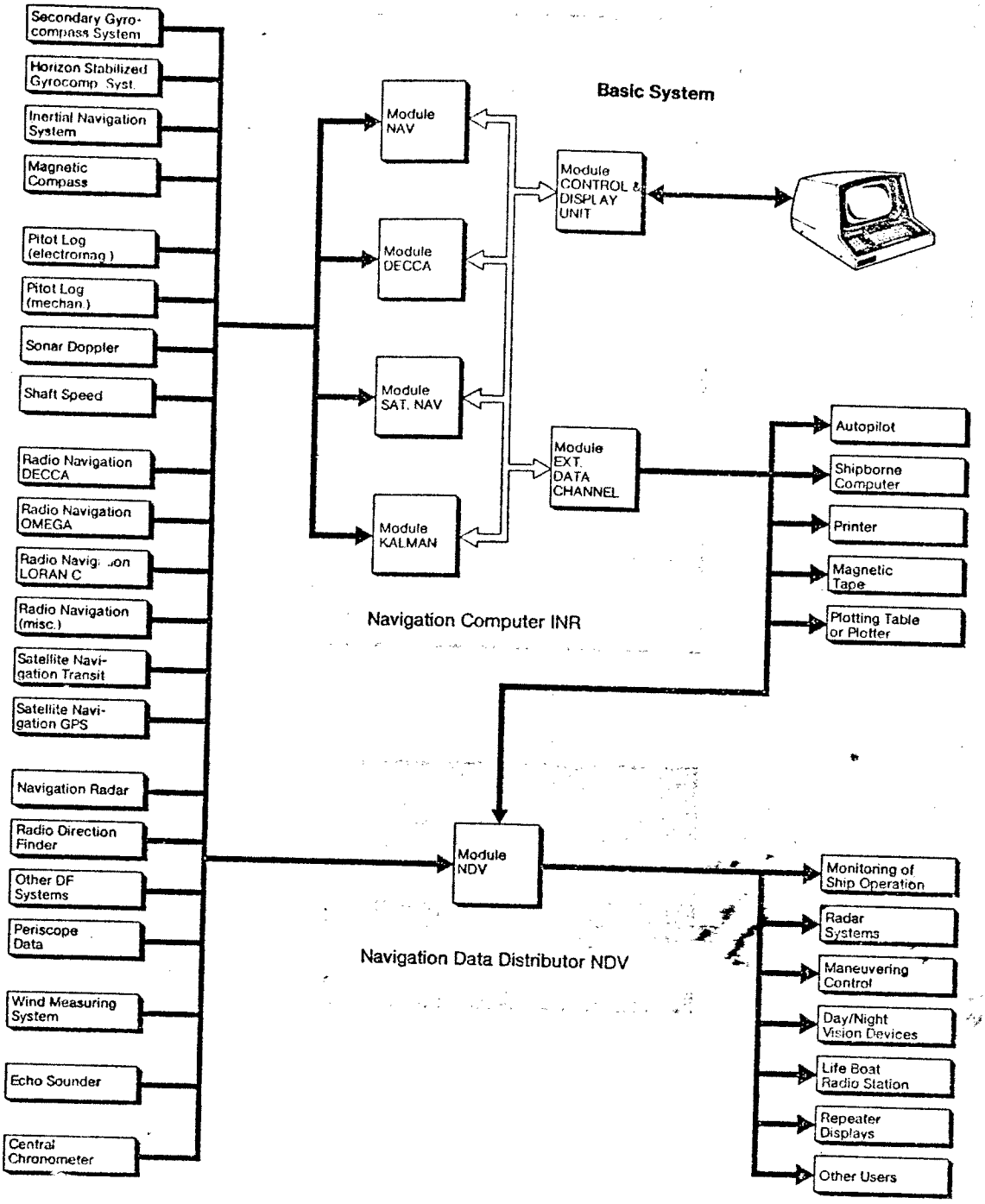


Fig 14 Blockschaltbild Adaptionskreis



TASKS

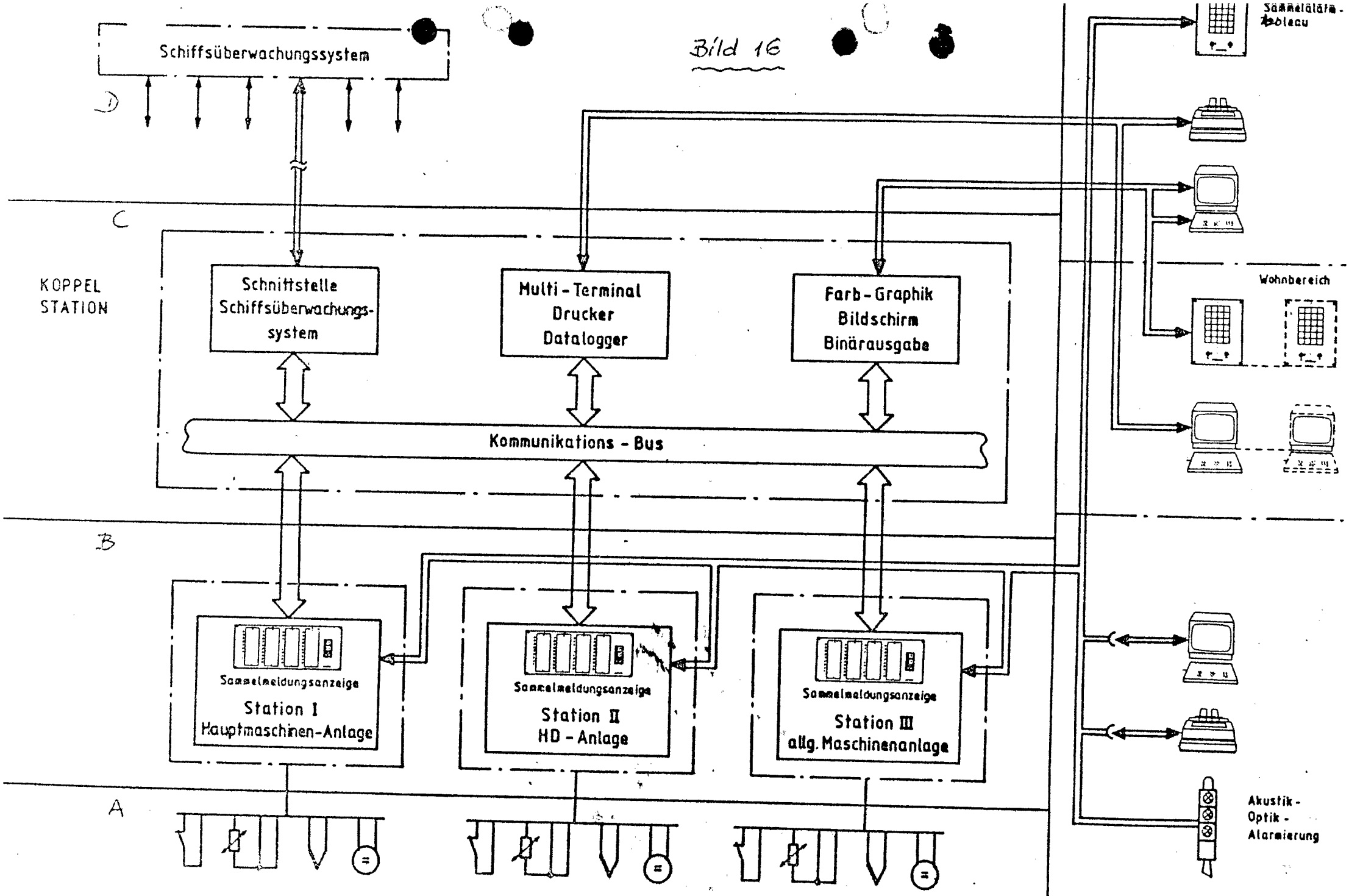
- Integration of navigation instruments into one computer-controlled system.
  - Easy operation and monitoring of the overall navigational area by a centralized control and display unit.
  - Higher density of navigational data inputs.
  - Automation of navigational tasks.
- Optimal availability of the navigation system by early detection of degrading functions.
  - Automatic monitoring of operating conditions of all integrated navigation instruments.
  - Redundant processing and distribution of navigational data.
- Optimal navigation and track steering by microprocessor-based navigation computer.
  - Computation and processing of navigational data by optimal use of sensor-furnished data.
  - Combining of various navigation methods.

ARITHMETIC OPERATIONS

- Error control of the sensor data, plausibility tests.
- Processing and transfer of status reports, error information, and test signals.
- Output of operating instructions and other information for optimal navigation and track steering.
- Automation of navigational tasks (chain selection with DECCA, LORAN, position fixes by SATNAV, etc.).
- Computation with respect to preselected waypoints for optimal track steering (course, deviation, great circle, etc.).
- Continuous processing of optimized position data, speed data and course data.
- Computation of wind data, current data and drift data.
- Continuous processing of other navigational data (wind, time, etc.).

Bild 15

Bild 16



		9.12.81	Optimierung der Überwachung von Maschinen und Anlagen	Siemens AG	Schiff der Zukunft	Konzept Blockschaltbild 1
		Erand				

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL AUG 1983
in Gothenburg
SHIP DESIGN FOR FUEL ECONOMY

ELECTRICITY PRODUCTION ON BOARD

by

HENRY KLINTORP & OLE GRØNE

B&W DIESEL A/S

COPENHAGEN

WEGENT '83, Gothenburg

ELECTRICITY PRODUCTION ON BOARD

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B&W Diesel A/S, Engineering Dept.,
Copenhagen, Denmark

Contents:

1. Introduction
2. Waste Heat Recovery
3. Main Engine Driven Generators
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7. Service Experience
8. Pilot Plants
9. The Unifuel Concept and Economy
10. Computer Based Assessments of Alternative Generator Sets
11. Conclusion

1. INTRODUCTION

The energy supplied to a ship is used primarily for propulsion purposes. Over the last decade, where the world has seen a radical change in the price of fuels, tremendous progress has been made in reducing the expenses for propulsion. With M.A.N.-B&W in the lead, a reduction in specific fuel consumption of diesel propulsion engines has been seen from some 155 g/BHPh even down to 118 g/BHPh, the latter related to our L90MCE being the lowest on the market.

Next to propulsion stands electricity production on board as an energy consumer. For the same reasons as for propulsion, i.e. the price of fuel, much progress has been seen in the electricity generating systems installed on board for the purpose of reducing the fuel bill.

Traditionally, electricity on board is produced by means of diesel generators. Other means are main engine driven generators, as well as steam driven turbo-generators, the steam being produced primarily by exhaust gas boilers (Fig. 1).

This paper will survey technically the above-mentioned alternative methods of generating electric power, especially in combination with M.A.N.-B&W main engines, as well as

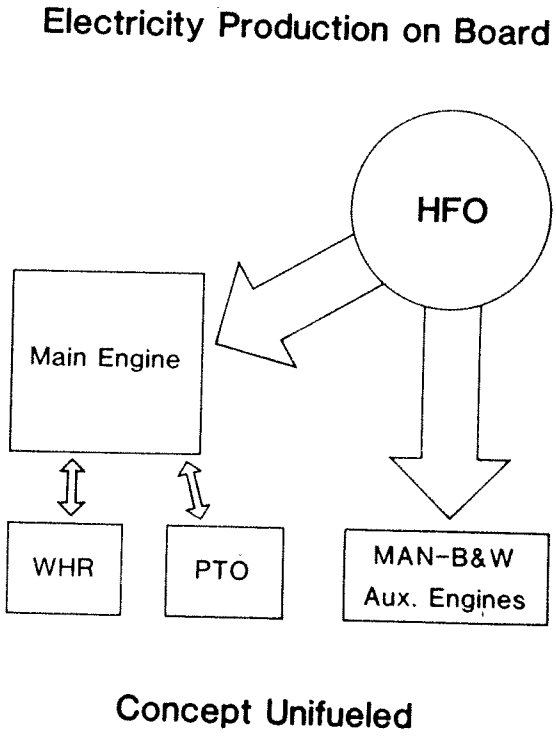


Fig. 1

outline in detail the development and the possibilities of operating modern M.A.N.-B&W diesel generators on the same heavy fuel as used for the main engine without blending, i.e. utilizing the "unifuel concept".

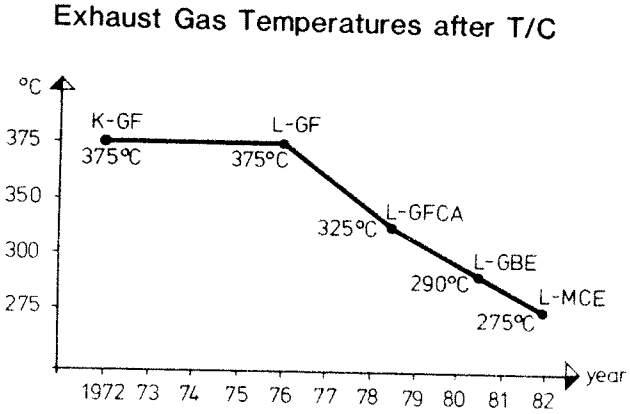
Design fuel quality for the engines is BS MA 100 Class 9 and CIMAC 10.

Using known power profiles from different types of ships, suggested combinations of electricity producing equipment will be given on the basis of the best operational economy.

2. WASTE HEAT RECOVERY

The use of an exhaust gas boiler to produce saturated steam for heating purposes and superheated steam to drive a turbo-generator is well proven and with quite good success.

Over the last ten years, the exhaust gas temperature after turbocharger for M.A.N.-B&W low speed engines has been reduced considerably as a consequence of the increased thermal efficiency of the engines. The temperature drop, together with the reduced fuel consumption, as illustrated in Fig. 2, clearly indicates that better system planning is necessary.



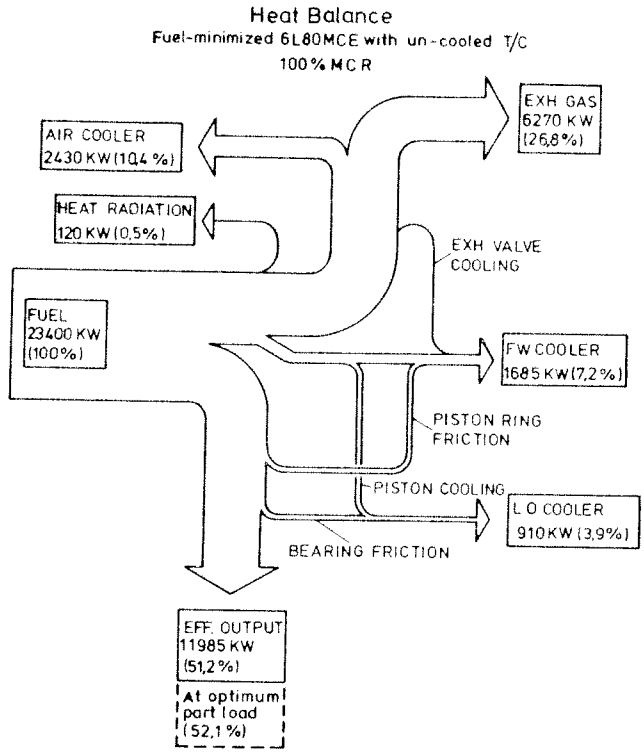
DEVELOPMENTS IN EXHAUST GAS TEMPERATURES for B&W low-speed diesel engines.

The temperatures refer to ISO ambient conditions and normal service running at sea.

Fig. 2

Whereas the exhaust gas boilers installed with the impulse turbocharged K-GF and L-GF engines could easily cover the electric power and steam demand down to about 80% part load, suitable for most trade patterns, a supplementary supply of heat is needed with the constant pressure turbocharged engines.

As indicated on the heat balance diagram for the L-MCE engines, having a shaft output above 50%, something like 2000 kJ heat at a temperature level of 275°C, corresponding to 0.55 kW, is available per kWh shaft power.



Heat balance for fuel-minimized diesel engine type 6L 80MCE optimized at 85% mep. The heat balance is based on ISO amb. cond. and valid for 100% MCR: 11985 KW at 83 RPM.

Fig. 3

For more details on the M.A.N.-B&W range of 2-stroke diesel engines, see ref. I.

For the large medium-speed engines used as propulsion plants, the trend has been the same, although with a somewhat higher temperature level.

For an M.A.N.-B&W V40-45, having a shaft output of some 43%, something like 2300 kJ heat at a temperature level of 365°C, corresponding to 0.64 kW, is available per kWh shaft power.

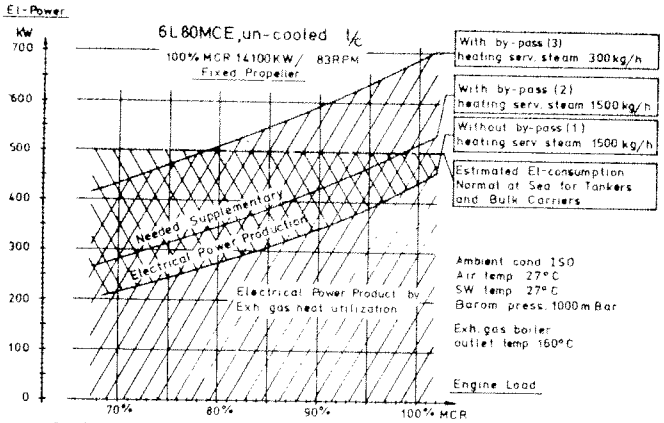
For more details on the M.A.N.-B&W range of 4-stroke diesel engines, see ref. II.

Whether a 2-stroke or a 4-stroke propulsion plant is concerned, the exhaust gas data depends primarily on the following factors:

a. Layout point of diesel engine (i.e. chosen MCR)
b. Fixed-pitch or controllable-pitch propeller (constant r/min)
c. Ambient conditions of anticipated trade routes
d. Time in service and actual load (i.e. operating profile).

A larger variety of systems can be designed - for example systems with single or dual pressure boilers, turbo-generators with back pressure or condensing turbines, bleed-off steam, low pressure steam turbines, etc.

To illustrate the steam economy, an example with 6L80MCE using a simple steam system for a bulker or tanker will be calculated.



TURBO-GENERATOR-ELECTRIC POWER PRODUCTION

The turbo-generator electric power production is shown for three alternatives in connection with exhaust gas heat utilization and based on available superheated steam after deduction of saturated steam for heating services, for alt. (1) and (2) 1500 kg/h and for alt (3) 300 kg/h (preh. hfo. bef. eng.).

Fig. 4

Fig. 4 shows the electric power that can be produced using the steam production for a single pressure boiler (steam pressure 7 bar, 165°C) from 70% to 100% MCR of the engine. With a traditionally designed engine room, about 1500 kg/h of saturated steam is required for heating purposes, under normal sailing at sea. An alternative, using only 800 kg/h for heating, is also shown.

With a modern turbo-generator, and moderate vacuum (0.07 bar), a specific steam consumption of about 7 kg/kWh is obtainable at 100% load.

From the figure it can be seen that only under high load conditions is it possible to cover all heat and electrical power requirements on board, which means that additional electricity generating is needed.

A detailed description is beyond the scope of this paper, but can be found in Ref. III.

Corresponding data is available for M.A.N.-B&W 4-stroke propulsion plants.

3. MAIN ENGINE DRIVEN GENERATORS

The use of a main engine driven generator enables the production of electricity to be based on the fuel oil used for the main engine. In fact, today we actually specify the same fuel for our auxiliary engines as for our low-speed diesels, but it should be borne in mind that heavy fuel operation is a relatively new concept to most other makers of medium-speed engines.

A number of pros and cons can be summed up in relation to main engine driven generators:

1) Low specific fuel oil consumption
2) Reduced overall maintenance costs
3) Reduced noise level in the engine room
4) Extra investment
5) Extra space requirements

The main engine driven generator can, in principle, be used in all types of ships. In some cases this generator can be used as an extra means of producing electricity, while, in other cases, it could be used to replace an otherwise necessary auxiliary engine.

When the generator is installed in connection with main engines with fixed-pitch propeller, frequency converters of either rotary or static type will normally be installed, resulting in a relatively low electrical efficiency.

The shaft generator can, as opposed to, for example, exhaust-gas-based turbo-generators, normally develop its maximum effect within a relatively large load range of the main engine. Generators with frequency converters normally give maximum output in the area of 100% down to 70-65% revolutions, which for ships with fixed-pitch propellers corresponds to an engine load from 100% down to approx. 28%.

As a rule, the shaft generator is so dimensioned that it is able to meet the electrical demand at sea.

4. GENERATOR POSITIONING

Traditional positioning of 2-stroke and 4-stroke engines comprises generators connected to the main engine via step-up gears.

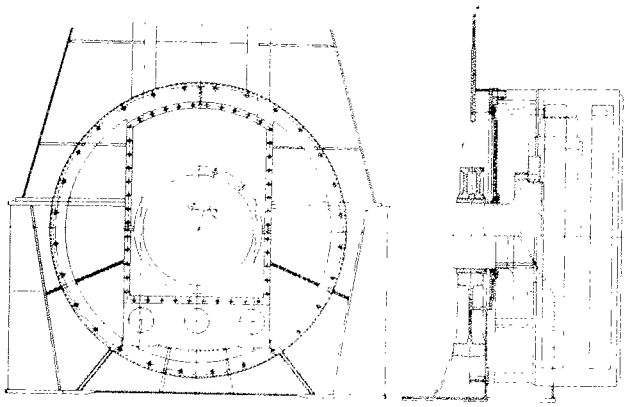
In a newer solution, primarily for 2-stroke engines without gearbox, the generator is integral with the intermediate shaft.

Recently the idea of mounting the generator directly on the fore end of the M.A.N.-B&W low-speed engine, as shown in Fig. 5, was conceived.

This is a space-saving design as seen from the fact that normally the generator will only increase the engine length by some 400 mm.

A further advantage is that interference with the vital camshaft drive is avoided. No separate foundation is needed either.

The generator is, in principle, an integral part of the engine, as the generator is driven directly by the crankshaft, which is



Face for generator at fore end of engine.

Cross section of fore end of engine with integral generator.

Fig. 5

extended and provided with a flange for mounting the rotor. Accordingly, the rotor is carried by the foremost engine main bearings, and the stator is mounted directly on the fore end of the engine. Extra foundations etc. on the tank top are thus avoided when using this concept.

The efficiency of the generator will be approximately equal to or, on account of a smaller air gap, slightly higher than when the generator is mounted on the intermediate shaft.

To give an idea of the size, the L90 engine can be equipped with a generator with a max. electrical output of about 1600 kW.

A generator arrangement which, in principle, is completely different from the previous types, comprises a combination of an epicyclic gear and a hydraulically operated frequency converter, i.e. a gear with a continuously varying ratio.

The gear system is designed for maintaining a constant output speed with an input speed varying from 100% down to approx. 65% engine revolutions.

As with waste heat recovery systems, a wide variety of systems using main engine driven generators and combinations of both systems are on the market.

It is beyond the scope of this paper to describe such systems in detail. However, one thing characterizes such systems - the more complicated they are, the more expensive and less user-oriented.

For more details on main engine driven generators, we can refer to Ref. IV.

5. DIESEL GENERATORS

Of the engines in the programme of M.A.N.-B&W, which comprises both 2-stroke and 4-stroke engines, all developed for heavy fuel operation, primarily the L23/30 and the L+V 28/32 engine types are advocated as heavy fuel burning auxiliary engines for the generator speed of 750/720 r/min. Fig. 6 illustrates the power range of these engines.

Auxiliary Engines Speed 720 rpm

		kW	HP
3L23/30	3 cyl.	315	430
5L23/30	5 cyl.	525	715
6L23/30	6 cyl.	630	860
7L23/30	7 cyl.	735	1000
8L23/30	8 cyl.	840	1140

Auxiliary Engines Speed 750 rpm

		kW	HP
3L23/30	3 cyl.	330	450
5L23/30	5 cyl.	550	750
6L23/30	6 cyl.	660	900
7L23/30	7 cyl.	770	1045
8L23/30	8 cyl.	880	1195

Auxiliary Engines Speed 720 rpm

		kW	HP
5L28/32	5 cyl.	875	1190
6L28/32	6 cyl.	1050	1425
7L28/32	7 cyl.	1225	1665
8L28/32	8 cyl.	1400	1905
12V28/32	12 cyl.	2100	2850
14V28/32	14 cyl.	2450	3330
16V28/32	16 cyl.	2800	3810
18V28/32	18 cyl.	3150	4280
6L28/32*	6 cyl.	1260	1715
8L28/32*	8 cyl.	1680	2285
9L28/32*	9 cyl.	1890	2570

* New rating

Auxiliary Engines Speed 750 rpm

		kW	HP
5L28/32	5 cyl.	925	1260
6L28/32	6 cyl.	1110	1510
7L28/32	7 cyl.	1295	1760
8L28/32	8 cyl.	1480	2010
10V28/32	10 cyl.	1850	2520
12V28/32	12 cyl.	2220	3020
14V28/32	14 cyl.	2590	3520
16V28/32	16 cyl.	2960	4020
18V28/32	18 cyl.	3330	4530
6L28/32*	6 cyl.	1320	1800
8L28/32*	8 cyl.	1760	2400
9L28/32*	9 cyl.	1980	2700

* New rating

Fig. 6

In the following, the main emphasis will be laid on the service background of these heavy fuel burning auxiliary engines, and on the basis of an economic evaluation it will be shown that for the vast majority of ships such engines are still by far the best option giving the simplest systems and the best operational economy.

6. DESIGN FEATURES OF THE 23/30 AND 28/32 ENGINES (Used for M.A.N.-B&W Gen-sets)

The L23/30, L28/32 and V28/32 types were designed to fill a gap existing in the production programme of Heavy Fuel Oil Engines, i.e. to provide a supplement to the well proven T23L-4, S28L-4 and U28L-4 types.

The mentioned engine types have, during service, proved their ability to run on any kind of heavy fuel, so the existing main design features were kept:

- r/min of 720 to 750
- turbocharged by pulse principle
- rigid and simple design
- good access for overhaul
- etc.

The efforts were therefore concentrated on:

- reducing the specific fuel oil consumption (S.F.O.C.)
- increasing the specific output of the engine.

The former target was reached by modifying the compression ratio, the pmax, and optimizing the fuel injection and valve timing, and the latter by increasing the specific output by approximately 20%.
The r/min of 720 for 60 Hz sets and 750 for 50 Hz sets were kept, as these speeds have proved to be ideal for heavy fuel operation, giving sufficient time for ignition and combustion.

For more specific information on engine design, see ref. II or the engine specification available from the maker.

When the owner makes an enquiry for a ship, the shipyard will often receive a "maker's list" covering the major items, including auxiliary engines, and from this list the yard can choose the type suited.

The owner's choice will in particular be based on the operating costs and reliability.

The first demand involves: engine designed for continuous heavy fuel oil operation and low fuel oil consumption - both are fully covered by the M.A.N.-B&W engines. The fuel oil consumption demand has often been misunderstood as being low consumption at full load, which is misleading, due to the fact that it is of the greatest importance for the owner to obtain low consumption within the running area, which for aux. engines is normally between 50 and 85% load. The minimum consumption of our engines is now found at about 75% load, as seen in Fig. 7, and illustrated in Fig. 8, showing installed auxiliary engines and load pattern for a number of ships, the average load being in the range mentioned.

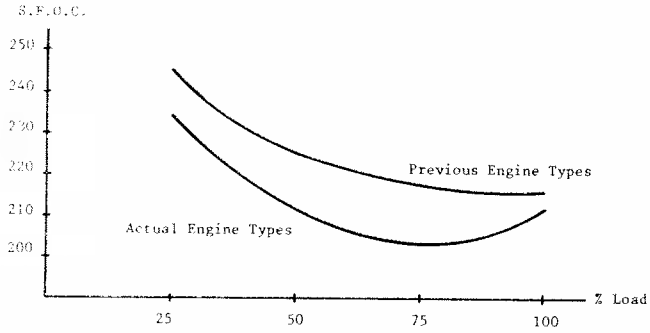


Fig. 7

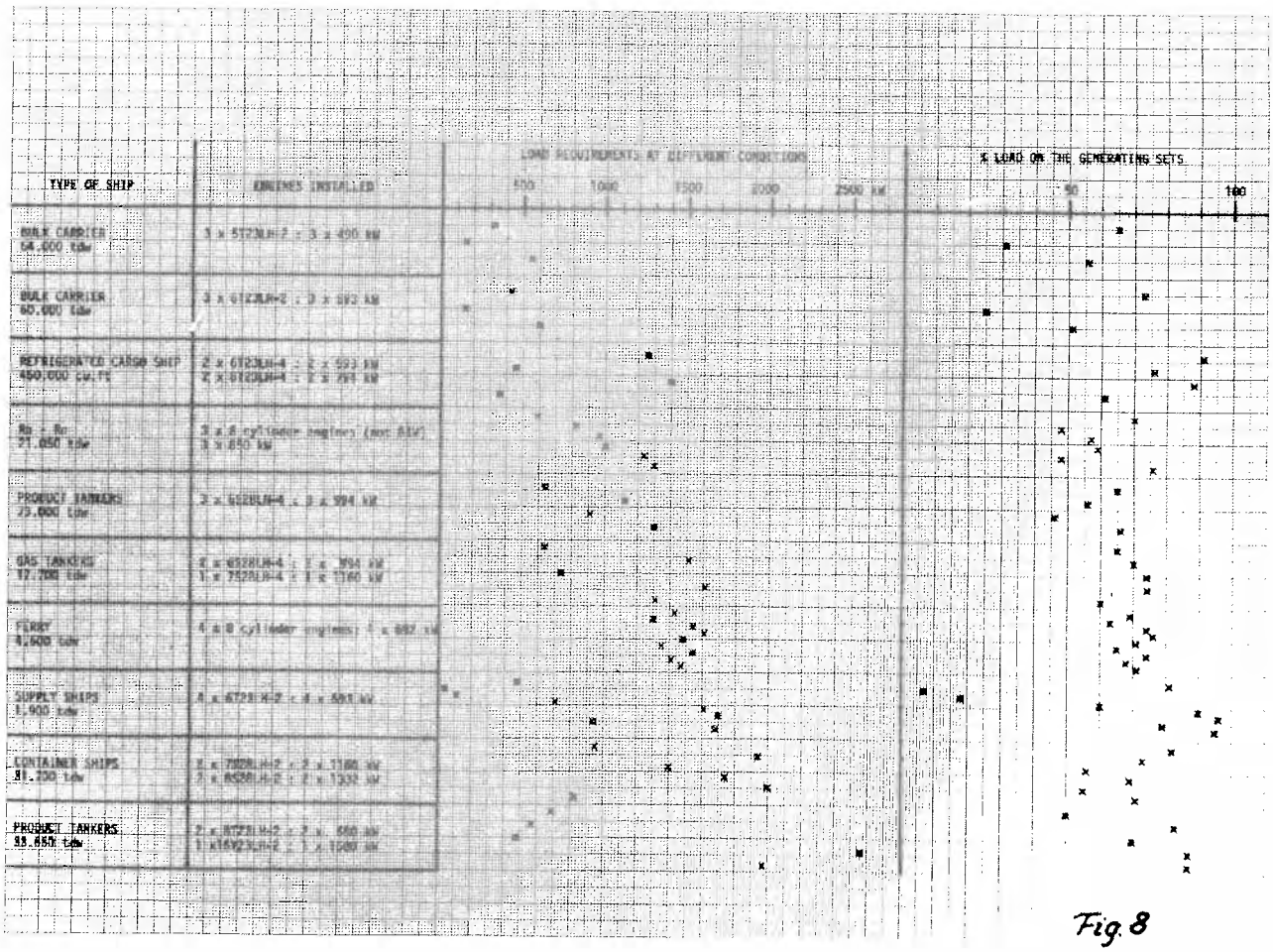


Fig. 8

This load range places diesel gen-sets in a more favourable position than the same engine types used as propulsion engines, the latter often being exposed to continuous operation at all loads from 100% MCR down to idling.

In order to avoid fouling of the gas-ways, we do not recommend to run the engines at loads below 20% MCR for longer periods than 20 hours.

Start and stop of the engine should take place on heavy fuel oil in order to prevent any incompatibility problems by changing to marine diesel oil.

Thus, marine diesel oil should only be used in connection with docking of the ship.

The second demand, reliability, can comprise for instance: "world wide service", "engines with good service results", and "long overhaul intervals", as described in the following - well-known features of all the M.A.N.-B&W gen-sets.

In addition to the above-mentioned demands, the individual shipyards have their own specified requirements, such as

- low price
- easy and cheap installation
- compact generating sets
- low overhauling height
- low noise level
- easy and good communication with engine maker.

The demand for easy installation is made by all yards, and with this in mind we have developed special fuel system units, i.e. blending units for in-line blending and, as a newcomer, a unifuel unit.

The blending unit is primarily designed for existing plants, but is also supplied for new engines when so preferred by the owners. The unifuel unit is designed for today's engines for the unifueled ship, which means that the same grade of heavy fuel is used for main and auxiliary engines, without any blending on board.

7. SERVICE EXPERIENCE

Testbed running is a normal part of an engine development programme in order to verify calculated stress levels and engine performance data, to optimize injection equipment, valve timing, turbocharger matching, etc.

However, it is our philosophy that the only place to test and prove the quality of an engine is during operation at sea, because we fully realize that testbed running has certain limitations for determination of wear rates and long-term performance.

Consequently, service experience is in fact the primary foundation for our engine design and development.

The 23-types and 28-types at today's design stage are based on proven designs born in 1965 and 1972, respectively, both types having been developed continuously through the years to cope with changing market demands.

Today, more than 3000 engines of the 23-types and more than 1000 engines of the 28-types are in operation on board ships.

Our experience with both the 23-types and the 28-types is, inclusive of propulsion engines and stationary engines, based on a total number of 20,000 cylinder units in service with an accumulated number of engine hours of 90×10^6.

Out of this figure, approximately 30×10^6 engine hours (one third), represents heavy fuel operation, mainly achieved as auxiliary engine operation.

Moreover, a similar number of engine hours on heavy fuel operation has been accumulated with earlier engine types which are no longer in production but still in service.

Our recommendations for overhauling intervals and information about expected service life of main components are calculated on the basis of average values of recorded service results.

Proper overhaul work is, on the basis of service experience, scheduled to take place for every 6000 running hours or every 12000 or more running hours.

There is one exception, viz. the fuel injection valves, which are scheduled to be cleaned and checked for opening pressure every 2000 running hours.

For marine diesel gen-sets, 6000 service hours is a typical value for annual utilization.

Our philosophy is that our diesel gen-sets, without any reduction in reliability and service availability, shall be able to be operated on heavy fuel for two years before any major overhaul has to take place. Service results have already confirmed that this is a realistic basis.

Generally, 8,000 service hours have been recorded between major overhauls, and some reference plants followed by our test engineers have achieved 12,000 service hours on heavy fuel without any overhaul.

8. PILOT PLANTS

In order to gain service experience from our latest types of engines, and to evaluate the most recent service experience, we have established a close cooperation with shipowners in operating standard plants as pilot plants. In particular, we have followed plants operated as "unifueled" plants, i.e. using the same type of heavy fuel for main engines and auxiliary engines.

As pilot plants are kept under close observation, with visits being paid on board every 1000-2000 hours, these plants give a good opportunity for long-term field testing of new components. For the same reason an excellent opportunity is created to introduce new maintenance methods and to optimize overhaul intervals.

An example of an earlier pilot plant is a diesel gen-set of type 6T23LH in the J. Lauritzen owned M/S "Irena Dan", a 70,000 tdw bulk carrier with three diesel gen-sets.

The engine type 23LH equipped with a 2-valve cylinder head was designed to operate on a fuel quality corresponding to IF 40, and all diesel gen-sets on board M/S "Irena Dan" were originally operated on IF 30.

In order to investigate how this engine type would perform on a more inferior fuel quality, it was decided to operate one of the engines on IF 180.

Today this engine has accumulated more than 20,000 service hours running on IF 180, mainly bunkered on the US East and Gulf Coasts and in Western Europe.

It was soon experienced that operating an engine originally designed for diesel oil or light fuel oil on an inferior fuel quality had some negative effects on the engine's components and performance, for which reason various modifications were introduced.

The following is a brief description of modifications introduced and the prolonged overhauling intervals achieved.

By introduction of Nimonic 80A spindles, improved valve spindle mechanism and new valve seat rings, the interval between exhaust valve overhauls was extended to 4000 service hours and the service life to 12,000 hours.

Decreased turbocharger performance, owing to heavy fouling of the nozzle ring and rotor in the gas turbine, resulted in increased exhaust temperature and burning of exhaust valves.

Introduction of water-washing equipment on the turbine side solved this problem and prolonged the overhaul interval for the turbocharger to 6000 service hours.

The piping of the fuel system was changed to enable continuous circulation of preheated fuel even with stopped engine.

An improved type of high pressure fuel pipes was tested on this engine, and later on introduced as standard.

Development and introduction of an improved procedure for plateau honing of the cylinder liners solved the problem of high wear rates, and the overhauling interval for piston, piston rings and cylinder liner was prolonged to 12,000 service hours.

Many of these modifications introduced on this engine have been incorporated as standard on engines in today's programme.

4-valve cylinder heads were introduced as standard on the basis of tests on board pilot plants.

The main purpose of introducing the 4-valve cylinder head was to achieve prolonged overhauling intervals in general and especially for the exhaust valves when operating on heavy fuel oil.

The main features of the 4-valve cylinder head are:

- exhaust valve spindles of Nimonic 80A
- rotocaps on exhaust valves
- new valve seat rings
- inlet valve spindles of martensitic steel with Stellite 12 as valve seat surface.

The first diesel gen-sets with 4-valve cylinder heads entered service in 1978 in the J. Lauritzen owned 12,570 dwt refrigerated cargo ship M/S "Canadian Reefer", and later in a sister ship, M/S "Equadorian Reefer".

The ships, which are operated according to the "unifuel" concept, have M.A.N.-B&W type L67GFC main engines and three aux. engines type 6S28LH-4.

A further four diesel gen-sets, also of type 6S28LH-4, installed on board the passenger/vehicle ferry M/F "Viking Sally" were followed as a pilot plant when entering service in 1980. This ferry in route between Sweden and Finland is also operated according to the "unifuel" concept, the propulsion machinery consisting of four M.A.N.-B&W type 40/45 medium-speed engines burning the same

type of heavy fuel as the auxiliaries, in this case IF 180.

During visits on board it was revealed that the appearance of the exhaust valve seats noted at 2000 running hours hardly changed during the following 6000 running hours and they were remounted without overhaul.

Today we have achieved our target, two years' operation on heavy fuel without overhaul.

4-valve cylinder heads are now standard on both our L23/30 and L28/32 gen-sets.

9. THE UNIFUEL CONCEPT AND ECONOMY

The economic advantage of heavy fuel operation of auxiliary diesel engines is determined by the price difference between marine diesel oil, currently priced at some 250 US$/t, and the most relevant alternative, heavy fuel oil, i.e. a price difference of some 80 US$/t.

The economic "drawback" is determined by the capital cost of the heavy fuel oil installation and by extra maintenance cost, if any.

To illustrate the advantages and drawbacks we will, as an example, look at the auxiliary engine plant of a ship which is operated as a unifueled ship. The auxiliary engines are of M.A.N.-B&W type 3x6S28LH-4.

In 1980 these engines consumed 1,400 tons heavy fuel oil (IF 180), corresponding to some 6.5 x 10^6 kWh. The saving compared to marine diesel oil amounted to 1,400 t. x 80 dollars/t = 112,000 dollars.

The capital cost of the heavy fuel installation is calculated to some 7,000 dollars/year and the spare parts consumption to 3,500 dollars/year out of which less than half is considered to be influenced by the fuel quality.

A self-explanatory annual saving of more than US$ 100,000 was secured.

The extra kW consumption for fuel heating and centrifuging has not been considered, but it is less than 2% of the total kW production.

As mentioned, the "Unifuel concept" means that the same heavy fuel is used under all conditions for propulsion and for electricity production, whether done by means of Waste Heat Recovery, Main Engine Driven Generators or Diesel Generators.

In the design of the "unifuel" ship, the shipyards have the following possibilities to combine with heavy fuel burning propulsion engines:

- M.A.N.-B&W auxiliary engines
- Waste Heat Recovery systems
- Main Engine Driven Generators.

Owing to the high cost of fuel, it is increasingly important for ships to be built to utilize the possibilities available and to select the right combination within the one progressing concept, i.e. the Unifuel concept.

The right combination will depend on many factors, some of which are well defined, and some of which are dependent on individual preferences and tradition.

Influencing parameters are:

- price of equipment
- installation costs
- rate of interest
- electrical efficiency
- fuel consumption
- predicted power profile
- maintenance cost and availability.

Examples of different combinations of electrical equipment - dealing with the ship's total consumption of heavy fuel - operating under sea conditions are shown in Fig. 9, the engines being a 6L80GBE installed in a ship with an actual electrical power requirement of some 450 kW at sea and a 4L90GB installed in a ship with an electrical power requirement of some 510 kW at sea.

In case A, where the utilization of waste heat can cover all the heat and electric power requirements under normal conditions at sea, the installation of a turbo-generator seems the obvious choice, while in case B, where supplementary steam from an oil fired boiler is needed, the choice of electrical equipment is less obvious. For elucidating the economical aspects, corresponding examples with tables showing the influence of first cost, fuel saving and maintenance cost are also shown.

In the examples, the different electrical equipment installed is assumed to be:

Case 1) 3 diesel generator sets, each with 80% MCR = actual el. power req.
One set is normally running at sea.

Case 2) 2 diesel generator sets, each with 80% MCR = actual el. power req.
1 turbo-generator set with 90% MCR = actual el. power req.
Only the turbo-generator is normally running at sea.

Case 3) 2 diesel generator sets, each with 80% MCR = actual el. power req.
1 main engine driven generator set with 80% MCR = actual el. power req.
Only the main engine driven generator is normally running at sea.

Furthermore, we have used the following approximated specific (MCR) first costs for comparison of the different electrical power systems:

SHIP SERVICE, NORMAL AT SEA

FUEL CONSUMPTIONS

ECONOMY

Fuels for	Viscosity c.St/s.Red.I	Lower Calorific Value	Price
Main engine Donkey boiler Diesel generator	same	41,000 kJ/kg	170 $/ton

6L80GBE - nominal rated
100% MCR: 18,360 BHP at 106 RPM
Service rating 80% MCR, i.e.
14,690 BHP at 98 RPM

	Case 1	Case 2	Case 3
	Main engine + Aux. Diesel Generat.	Main Engine + Turbo Generator (+ Donkey Boiler)	Main engine + Power Take Off
Main Engine Power	Service Knots 14,690 BHP 98 RPM	Service Knots 14,690 BHP 98 RPM	Service Knots 1) 15,430 BHP 98 RPM
SFOC of Main Engine LCV (Lower Calorific Value) 41,000 kJ/kg,	4) 131.4	131.4	132.3
Fuel consumption and cost per day of main engine	46.3 7870	46.3 7870	49.0 8330
Power take off, electric output	-	-	450
Available superheated steam for turbo generator	-	> 3150	-
Necessary supplementary steam production of donkey boiler	0	0	0
Fuel consumption and cost per day of donkey boiler (η = 80%) LCV = 41,000 kJ/kg	0 0	6) 0 0	0 0
Turbo generator el-production (7.0 kg steam/kWh)	-	450	-
SFOC of diesel generator LCV = 41,000 kJ/kg incl. equivalent lub.oil amount	3) 241	-	-
Fuel consumption and cost per day of diesel generator	2.6 440	0 0	0 0
Diesel generator el-production (normal el-consumption in ship)	450	0	0
Total fuel cost per day at ship	8310	7870	8330
Difference in fuel cost	+ 440	0	+ 460

4L90GB - Optimized
Service rating 15,000 BHP
at 91 RPM

	Case 1	Case 2	Case 3	
	Main engine + Aux. Diesel Generat.	Main Engine + Turbo Generator (+ Donkey Boiler)	Main engine + Power Take Off	
Main Engine Power	Service Knots 15,000 BHP 91 RPM	Service Knots 15,000 BHP 91 RPM	Service Knots 2) 15,840 BHP 91 RPM	Units
SFOC of Main Engine LCV (Lower Calorific Value) 41,000 kJ/kg,	5) 133.8	133.8	134.3	g/BHPh
Fuel consumption and cost per day of main engine	48.2 8194	48.2 8194	51.1 8687	tons/24h $/24h
Power take off, electric output	-	-	510	kW
Available superheated steam for turbo generator	-	3070	-	kg/h
Necessary supplementary steam production of donkey boiler	0	500	0	kg/h
Fuel consumption and cost per day of donkey boiler (η = 80%) LCV = 41,000 kJ/kg	0 0	6) 1.01 172	0 0	tons/24h $/24h
Turbo generator el-production (7.0 kg steam/kWh)	-	510	-	kW
SFOC of diesel generator LCV = 41,000 kJ/kg incl. equivalent lub.oil amount	3) 241	-	-	g/kWh
Fuel consumption and cost per day of diesel generator	2.95 501	0 0	0 0	tons/24h $/24h
Diesel generator el-production (normal el-consumption in ship)	510	0	0	kW
Total fuel cost per day at ship	8695	8366	8687	$/24h
Difference in fuel cost	+ 329	0	+ 321	$/24h

		Case 1	Case 2	Case 3	Case 1	Case 2	Case 3	Units
Equipment	Difference in first cost, estimated	0	+ 450,600	+ 129,400	0	+ 510,700	+ 146,600	$
Fuel consumption	Difference in fuel cost per day	+ 440	0	+ 460	+ 329	0	+ 321	$/day
	- - - - per year (250 days/year)	+ 110,000	0	+ 115,000	+ 82,250	0	+ 80,250	$/year
	Equivalent first cost - fuel consumption (Capitalization factor 6)	+ 660,000	0	+ 690,000	+ 493,500	0	+ 481,500	$
	Total equivalent first cost difference, incl. fuel saving	+ 209,400	0	+ 368,800	0	+ 17,200	+ 134,600	$
Maintenance	Difference in maintenance cost	+ 8,210	+ 1,500	+ 560	+ 9,310	+ 1,700	+ 640	$/year
	Equivalent first cost - maintenance (capitalization factor 6)	+ 49,300	+ 9,000	+ 3,400	+ 55,900	+ 10,200	+ 3,800	$
	Total equivalent first cost difference, incl. fuel saving and maintenance	+ 249,700	0	+ 363,200	+ 28,500	0	+ 111,000	$

1) PTO : η_{PTO} (incl. gen.) = 0.825; 450 kW el-prod. $\sim \frac{450}{\eta_{PTO}}$ kW = 545 kW = 740 BHP

2) PTO : η_{PTO} (incl. gen.) = 0.825; 510 kW el-prod. $\sim \frac{510}{\eta_{PTO}}$ kW = 618 kW = 840 BHP

3) Diesel Generator: $m_{fuel+equiv.} = \frac{m_{fuel} + m_{lub} \times 3}{\eta_{gen}} = \frac{210 + 1.6 \times 3 \text{ g/kWh}}{0.93}$ = 231 g/kWh ref. LCV = 42,700 kJ/kg
= 241 g/kWh ref. LCV = 41,000 kJ/kg

4) Main Engine : m_{fuel} = 126.2 g/BHPh ref. LCV = 42,700 kJ/kg
m_{fuel} = 131.4 g/BHPh ref. LCV = 41,000 kJ/kg

5) Main Engine : m_{fuel} = 128.5 g/BHPh ref. LCV = 42,700 kJ/kg
m_{fuel} = 133.8 g/BHPh ref. LCV = 41,000 kJ/kg

6) Donkey Boiler : 1 kg superh. steam $\sim \frac{h_{st.sh} - h_f}{LCV \times \eta} = \frac{2975-210}{41000 \times 0.8}$ = 0.0843 kg fuel

Fig. 9

Case 1) Unifueled diesel generator sets: US$ 230/kW installed.
Case 2) High-utilization waste heat recovery systems, incl. turbo-generator, but less the first cost of a normal low-heat-utilization system: US$ 1,160/kW installed
Case 3) Main engine driven generator sets: US$ 460/kW installed.

The difference in fuel consumption is capitalized, using the assumed capitalization factor 6. In example A, the equivalent first cost difference (incl. fuel saving) shows an obvious advantage in favour of the turbo-generator solution, whereas, in example B, the complete diesel generator set solution seems to be the best one.

Further to the above estimates, the maintenance cost for the various electrical equipment also has to be incorporated, although this is rather difficult to estimate. As a guidance, the following specific maintenance costs have been used:

Case 1)	Diesel generator: (US$ 10/BHP year)	US$ 14.6/kW year
Case 2)	Turbo-generator:	US$ 3/kW year
Case 3)	Main engine driven generator:	US$ 1/kW year

As can be seen from the examples, the turbo-generator and main engine driven generator solutions are now becoming more attractive, compared to the economical evaluations in which maintenance costs were not taken into consideration.

Reliability in service (i.e. availability) has not been incorporated in the economical evaluation, as this parameter depends on many other factors.

10. COMPUTER BASED ASSESSMENTS OF ALTERNATIVE GENERATOR SETS

For the purpose of going deeper into the selection so as to be able to give advice customers, M.A.N.-B&W Diesel have developed a computer based tool for an economic evaluation of alternative generator sets. This tool is utilized to provide the best suited technical solution for a given ship, based on an economic evaluation of the relevant basic parameters.

To show how this tool operates, an example in which three alternative generator combinations are available has been selected. It is assumed that three individual generators are necessary. The first configuration comprises three unifueled gen-sets, the second two unifueled gen-sets and one main engine driven gen-set (PTO) and, finally, the third is composed of two unifueled gen-sets and one waste heat recovery system with a turbo-generator, assuming that the main engine can provide the necessary exhaust gas heat.

The example is based on actual recorded consumptions, but the purpose here is not to go into a real case; it is rather to demonstrate how the tool and the evaluation process work.

Below are given the main input parameters for each of the three types of generator sets. The input can, of course, be chosen on the basis of actual experience.

Item	Unit	Unifuel Gen-Set	Main Engine Driven Gen-Set	Waste Heat Recovery Set
Power	kW	625	625	625
Max el. serv. output	%	80	80	90
SFOC	g/kWh	228	206	-
SLOC	g/kWh	0.95	0.5	0
Maintenance	US$/kWy	12.5	3	1
Spares	US$/kWy	10	6	3
Fuel cost	US$/T	180	180	0
Lub. oil cost	US$/T	1080	1080	1080
Inst. cost	US$/kW	220	450	900
Inflation rate	%/year	10	10	10
Increase fuel price	%/year	7	7	7

All plants are assumed to be financed with 80% of the installation costs by means of series loans over nine years at an interest at 12% p.a., and top-up financing at an interest rate of 8% p.a., with a pay-back period of 3 years.

The load structure used in the example is given below:

Manoeuvring	400 hours/year	700 kW
Ballast	3500 hours/year	500 kW
Loaded	3500 hours/year	400 kW
Port	1360 hours/year	330 kW

Using this load structure as input, as well as the input data already shown, we obtain the result shown in Fig. 10.

The figure gives the average total production costs (in per cent/kWh) for a 10 year period, calculated on the basis of a Net Present Value of the total costs.

It will appear that the first two alternatives give almost equal production costs, whereas the combination of one waste heat recovery unit and two unifueled gen-sets is advantageous in this case.

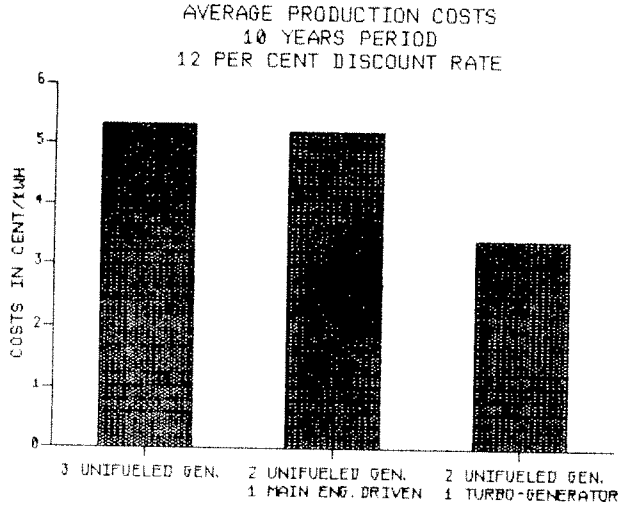


Fig. 10

However, a change in the fuel oil price and especially in the price level of the equipment, as well as in the load structure and the financial parameters, could easily alter the optimal solution. The tool described is designed in such a way that it opens up for a systematic examination of key-parameters and input data.

Fig. 11 shows an example of the complete output.

11. CONCLUSION

As illustrated in the foregoing, many considerations have to be weighed and evaluated when deciding on electricity generating equipment for a ship. Whatever the requirements for a ship's electrical needs may be, these needs can be fulfilled with the assistance and equipment of M.A.N.-B&W, whether the solution comprises main propulsion engines with power take-off and/or waste heat recovery, or heavy fuel burning auxiliary engines for "Unifuel" application.
The optimum solution is in our programme.

REFERENCES:

I: The M&G Programme of M.A.N.-B&W Diesel. Design Philosophy and Application of the Modern Long Stroke Crosshead Engine. ISME '83, Tokyo. (H. Klintorp, O. Grøne, B&W Diesel A/S).

II: Constant Innovation with a View to Enhancing Reliability and Durability. ISME '83, Tokyo. (R. Häfner, M.A.N.-Augsburg AG).

III: Waste Heat Recovery, B&W publication. (H. Klintorp, B. Jacobsen, B&W Diesel).

IV: Main Engine Driven Generators, B&W publication. (G. Mikkelsen, W. Brendorp, B&W Diesel A/S).

ALTERNATIVE POWER GENERATORS
COMPUTER BASED ASSESSMENT

GENERATORS
3 x 5T23LH-4

1.

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	YEAR 6	YEAR 7	YEAR 8	YEAR 9	YEAR 10
EL. PRODUCTION (KWH)										
DIESEL 1	3798800	3798800	3798800	3798800	3798800	3798800	3798800	3798800	3798800	3798800
DIESEL 2	80000	80000	80000	80000	80000	80000	80000	80000	80000	80000
DIESEL 3	0	0	0	0	0	0	0	0	0	0
TOTAL PRODUCTION	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800
COSTS IN 1000 USD										
OPERATION COSTS										
FUEL COSTS	159.19	170.33	182.25	195.01	208.66	223.27	238.90	255.62	273.51	292.66
LUB. OIL COSTS	3.98	4.26	4.56	4.88	5.22	5.58	5.97	6.39	6.84	7.32
SPARE PARTS	18.75	20.62	22.69	24.96	27.45	30.20	33.22	36.54	40.19	44.21
MAINTENANCE	9.70	10.67	11.73	12.91	14.20	15.62	17.18	18.90	20.79	22.87
OPERATING COSTS	191.61	205.88	221.23	237.75	255.53	274.66	295.26	317.44	341.33	367.05
FIXED COSTS										
INTEREST	46.20	39.60	33.00	26.40	22.00	17.60	13.20	8.80	4.40	.00
REPAYMENT	64.17	64.17	64.17	36.67	36.67	36.67	36.67	36.67	36.67	.00
FIXED COSTS	110.37	103.77	97.17	63.07	58.67	54.27	49.87	45.47	41.07	.00
TOTAL YEARLY COSTS	301.98	309.65	318.40	300.81	314.19	328.93	345.13	362.91	382.39	367.05

GENERATORS:
2 x 5T23LH-4
1 x MAIN ENGINE DRIVEN GENERATOR

2.

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	YEAR 6	YEAR 7	YEAR 8	YEAR 9	YEAR 10
EL. PRODUCTION (KWH)										
MAIN ENGINE DRIVEN	3150000	3150000	3150000	3150000	3150000	3150000	3150000	3150000	3150000	3150000
DIESEL 1	648800	648800	648800	648800	648800	648800	648800	648800	648800	648800
DIESEL 2	80000	80000	80000	80000	80000	80000	80000	80000	80000	80000
TOTAL PRODUCTION	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800
COSTS IN 1000 USD										
OPERATION COSTS										
FUEL COSTS	146.71	156.98	167.97	179.73	192.31	205.77	220.18	235.59	252.08	269.72
LUB. OIL COSTS	2.45	2.62	2.80	3.00	3.21	3.43	3.67	3.93	4.21	4.50
SPARE PARTS	16.25	17.87	19.66	21.63	23.79	26.17	28.79	31.67	34.83	38.32
MAINTENANCE	4.03	4.43	4.87	5.36	5.90	6.49	7.13	7.85	8.63	9.50
OPERATING COSTS	169.44	181.91	195.31	209.72	225.21	241.86	259.77	279.03	299.75	322.04
FIXED COSTS										
INTEREST	62.30	53.40	44.50	35.60	29.67	23.73	17.80	11.87	5.93	.00
REPAYMENT	86.53	86.53	86.53	49.44	49.44	49.44	49.44	49.44	49.44	.00
FIXED COSTS	148.83	139.93	131.03	85.04	79.11	73.18	67.24	61.31	55.38	.00
TOTAL YEARLY COSTS	318.27	321.83	326.34	294.76	304.32	315.04	327.02	340.34	355.13	322.04

GENERATORS
2 x 5T23LH-4
1 x TURBO-GENERATOR (WHR)

3.

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	YEAR 6	YEAR 7	YEAR 8	YEAR 9	YEAR 10
EL. PRODUCTION (KWH)										
TURBO-GENERATOR	3150000	3150000	3150000	3150000	3150000	3150000	3150000	3150000	3150000	3150000
DIESEL 1	648800	648800	648800	648800	648800	648800	648800	648800	648800	648800
DIESEL 2	80000	80000	80000	80000	80000	80000	80000	80000	80000	80000
TOTAL PRODUCTION	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800	3878800
COSTS IN 1000 USD										
OPERATION COSTS										
FUEL COSTS	29.91	32.00	34.24	36.64	39.21	41.95	44.89	48.03	51.39	54.99
LUB. OIL COSTS	.75	.80	.86	.92	.98	1.05	1.12	1.20	1.28	1.37
SPARE PARTS	14.38	15.81	17.39	19.13	21.05	23.15	25.47	28.01	30.81	33.90
MAINTENANCE	2.45	2.70	2.97	3.26	3.59	3.95	4.34	4.78	5.26	5.78
OPERATING COSTS	47.48	51.31	55.46	59.95	64.82	70.10	75.82	82.02	88.75	96.04
FIXED COSTS										
INTEREST	93.80	80.40	67.00	53.60	44.67	35.73	26.80	17.87	8.93	.00
REPAYMENT	130.28	130.28	130.28	74.44	74.44	74.44	74.44	74.44	74.44	.00
FIXED COSTS	224.08	210.68	197.28	128.04	119.11	110.18	101.24	92.31	83.38	.00
TOTAL YEARLY COSTS	271.56	261.99	252.74	188.00	183.93	180.28	177.06	174.33	172.12	96.04

Fig. 11

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SHIP DESIGN FOR FUEL ECONOMY

Propeller Hull Interaction, Derating Engines and Propeller Layout

By

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1. Introduction

As well known and often said: Before the big oil crisis fuel oil for main powering equipment in merchant ships was almost an unlimited commodity, the price of which was often considered as a minor, not to say neglected, cost factor of the operation of a vessel. At that time the naval architect designed the hull and the marine engineer the main engine and little was the understanding from either side about the other's work. Both parties optimized (or suboptimized) from the criteria of his area. The thing in between, the propeller, had then to be done the best way, with the naval architect's resistance at one side and the power and rpm of the marine engineer on the other.

There was not much flexibility from either side in order to produce a given speed at a given condition to the minimum use of the very cheap fuel oil. Also at that time a ship was most often designed to whatever maximum speed obtainable with the engine given.

This is not so anymore. The naval architect now requires a main engine he can adapt to the hull given (a hull with lowest possible resistance, but in today's world the hull is more often designed according to the function of the vessel). The main engine manufacturer has given him exactly that: An engine which the ship designer can adapt and optimize with his hull.

This lecture will be based on the assumption that today the layout of the main engine and propeller is based on the specification of a given speed at a given condition. The ship designer's job is then to come up with a design which will use less fuel than the competitive designs (of course an economical question will also be raised here, because even a fuel saving can be too

costly; but if first cost and other extra running costs are recalculated to equivalent fuel consumption then we are back on the trail).

This paper will also limit itself to the matching of propeller and two stroke slow speed diesel engines. Other lectures in this course has taken well care of hull design, resistance reduction, and other types of main propulsion systems. The ideas and the philosophy expressed in this paper can of course still be used.

2. The System: Ship Hull - Propeller - Main Engine

The ship, machinery and propeller can be considered as a system (see fig. 1), where the best possible balance has to be found. The system is built up by three part-systems: the ship, the machinery and the propeller. A change in one of the part-systems will usually mean changes in the two other systems as well. For instance if the ship is enlarged, bigger machinery is needed if the same speed is to be achieved. In order to get sufficient power transferred from the machinery to the water around the ship the propeller has also to be changed.

We will now discuss each part by itself and after that their interactions, but first some definitions.

2.1 Definitions

To ensure the correct nomenclature, some of the parameters which are of importance for the interaction between ship, machinery and propeller are described in this chapter.

The power needed (the effective power) to move the ship through the water or to tow the ship at the speed V is:

$$P_E = RV$$

where R is the resistance (see fig. 2).

The propeller delivers the power (the thrust power)

$$P_T = TV_A$$

where T is the propeller thrust and V_A is the velocity of the water flowing to the propeller or the speed of advance of the propeller.

The power delivered to the propeller is:

$$P_D = 2\pi Qn$$

where Q is the propeller torque and n is the rate of revolution.

The power delivered by the propulsion machinery (the shaft power) is:

$$P_S = 2\pi Mn$$

where M is the torque which has been transferred to the shafting from the propulsion machinery.

Instead of P_S, P_B (the brake power) will be used for internal combustion engines.

The individual components are connected through the equation

$$\frac{P_E}{P_B} = \frac{P_E}{P_T} \cdot \frac{P_T}{P_D} \cdot \frac{P_D}{P_B}$$

$$\eta_T = \eta_H \cdot \eta_B \cdot \eta_S \quad ,$$

where

η_T is the total efficiency

η_H is the hull efficiency

$\eta_H = \frac{RV}{TV_A} = \frac{R/T}{V_A/V} = \frac{1-t}{1-w}$, where t is the thrust deduction fraction and w is the wake fraction

η_B is the propeller efficiency for the propeller working aft of the ship and is defined as

$\eta_B = \eta_0 \eta_R = \frac{TV_A}{2\pi Qn}$

η_0 is the efficiency of the propeller working without ship, that is in a homogeneous field of flow

η_R connects η_B and η_0 and is called the relative rotative efficiency

η_S is the shaft efficiency.

2.2 The Hull

Concerning the hull we will limit ourselves to the fact that the hull creates a resistance against movement through the water and air (wind) and creates the flow to the propeller. These facts are normally given in a very simple way as a diagram or table with resistance

against speed. See fig. 2. Also the fact that the water flows at a lower speed to the propeller than the speed of the vessel is given by a number (sometimes speed dependent if one believes and dares), the wake fraction. See fig. 2.

The last thing of the hull we must know about is the fact that the presence of the hull in front of and close to the suction side of the propeller creates a suction in the aft end of the hull, thus giving rise to an extra resistance which means the propeller must deliver a thrust which is somewhat higher than the resistance of the hull. This effect is conventionally described by a number (sometimes speed dependent if one really dares) called the thrust deduction fraction, t. See fig. 2.

How this resistance, wake fraction and thrust deduction fraction is found is not within the scope of this lecture. We shall only mention that the first could be found by ref. (1) and (2) and the latter by ref. (3). Fig. 3 reproduced from ref. (3) shows Harvald's diagram to find w and t. If a project has reached a certain stage, model test will give it all. For a discussion of this see ref. (4).

One more factor must be mentioned at this place. It is to be expected that the hull when used in the real sea and not only in still water will react as if the resistance was higher. This extra resistance is often called "sea margin" and the interpretation of this idea is wide open. We will revert to this later, but will already now introduce sea margin as an added resistance meaning that the propeller has to produce even more thrust in order to keep the vessel's speed. What this means will be discussed in the following paragraphs among other things.

2.3 The Propeller

Although the propeller is a simple device for propulsion, it still represents one of the most difficult problems in hydrodynamics. Therefore the propeller is normally thought about very empirically rather than theoretically in the design phase. Models of propeller have been made and long series have been tested. The most famous and most often used at least in Europe is of course the Wageningen B-screw series. Results have been given in design charts, see fig. 4 and in formulas, see ref. (5). Note that fig. 4 is a logarithmic diagram.

The propeller geometry is described mainly by only four numbers:

Z:	The number of blades
D:	The diameter
P:	The pitch or the pitch ratio P/D
A_E/A_O:	The expanded area ratio.

The capability of the propeller can be measured by moving the propeller through the water and rotate it at the same time. The propeller will then produce a thrust, T and need a torque, Q in order to do it. T and Q are normally given non-dimensionally as the thrust and torque coefficient.

$$K_T = \frac{T}{\rho n^2 D^4} \qquad (2.3\text{-}1)$$

and

$$K_Q = \frac{Q}{\rho n^2 D^5} \qquad (2.3\text{-}2)$$

ρ is the mass density of water.

The speed through the water, V_A of the propeller and the revolution is normally combined in the advance number, J:

$$J = \frac{V_A}{nD} \qquad (2.3\text{-}3)$$

Now K_T and K_Q can be given as functions of J, Z, D etc. See f.ex. ref. (5). As can be seen from the diagram fig. 4. K_Q is not plotted, but the efficiency of the propeller is. The efficiency is given as:

$$\eta_0 = \frac{J}{2\pi} \frac{K_T}{K_Q} \qquad 2.3\text{-}4)$$

(develop this formula as an exercise).

So K_Q can be found from this formula rather than mess up the nice diagram with more curves.

Also two other non-dimensional quantities are given in fig. 4 on the two rather peculiar logarithmic axes:

The thrust loading coefficient which expresses the thrust per area unit of the propeller disk, an easily adaptable physical coefficient non-dimensionalised by Bernoulli:

$$C_{Th} = \frac{T}{\frac{1}{2} \rho V_A^2 \frac{\pi}{4} D^2} \qquad (2.3\text{-}5)$$

Also the abstract, but non-dimensional basic thrust

coefficient is given:

$$b_T = \frac{T\, n^2}{\rho\, V_A^{\,4}} \qquad (2.3\text{-}6)$$

Along lines perpendicular to each axis both C_{Th} and b_T are constant.

Remembering our policy spelt out in the introduction that we will lay out the system based only on a given speed at a given condition, we will see how handy these two basic coefficients come in.

With speed and condition (sea margin) known then V_A and thrust T are also known. With a choice of blade number and expanded area (which we will not discuss here) we only need to know the pitch and diameter of the propeller to have it fully defined.

But C_{Th} contains only diameter D as unknown and b_T only rps, n as unknown.

There are now two ways to find the optimum propeller under the given assumptions.

First if the diameter is known or given for instance because of limited draught or other conditions. Calculate C_{Th} and follow the line to where the efficiency is maximum (helped by the line "η_{max} for C_{Th}"). The pitch ratio P/D and the advance number J can then be read. n can be calculated from J by $n = V_A/(JD)$.

If n is given (main engine specified data for instance), calculate b_T. Follow the same procedure and end up calculating D from J by $D = V_A/(Jn)$.

Please observe here that the efficiency has a very little degree of sensitivity. J and P/D and therefore also n and D can vary quite much, before the efficiency changes significantly: The C_{TH} and b_T lines are nearly following the constant efficiency levels. We will discuss and use this fact later.

One more thing can be seen in fig. 4. There are two sets of curves for two different Reynold's numbers. This expresses the expectation that the propeller in full scale (high Reynold's number) is more efficient than the small model (of about 25 cm). This must definitely be true, but how far this can be carried is an open question. In ref. (5) there is such a Reynold's number scale effect built in and this is what is plotted on fig. 4; but at very high Reynold's numbers this means that the optimum propeller is found at rather low advanced numbers giving a quite high propeller diameter.

The full set of the B-series diagrams are to be published later from the Technical University of Copenhagen.

When the optimum propeller is found in the way described here then the performance of the propeller can be read on the basis of the diagram along the P/D-curve connected to the found propeller in this way:

a. For a given V_A, look up R, the wake fraction and the thrust deduction fraction in the hull data tables, see section 2.2.

b. Calculate C_{Th} according to formulas (2.3-5).

c. Intersect the line C_{Th} with the P/D curve found, fig. 5.

d. Read J.

e. Calculate n from (2.3-3).

f. Read also η_0.

g. Calculate K_Q from (2.3-4).

h. Calculate Q from (2.3-2).

The delivered power or propeller power can now be calculated by:

$$P_D = 2nQ\pi \qquad (2.3\text{-}7)$$

and the necessary brake power, P_B, on the main engine can be found by dividing with the shaft efficiency, η_S:

$$P_B = P_D/\eta_S \qquad (2.3\text{-}8)$$

By repeating this procedure (today of course in the easiest way in a computer or a personal computer if it is an often repeated task, otherwise quite simply by the diagram) one obtains a diagram of P_B versus n with indication of the speed. See fig. 6.

This we will call the demand curve for the ship with a propeller of given diameter and pitch.

We will read one more characteristic thing about the propeller out of fig. 4.

If the diameter, D is enlarged the C_{Th} becomes smaller as can easily be seen from (2.3-5), this means that the constant C_{Th} line moves to the right in the diagram and intersects at larger values of the efficiency with the "η_{max} for C_{Th}" curve. Also J becomes larger

and since V_A is the same, then n becomes smaller as can be seen off 2.3-3.

Try the same reasoning with b_T assuming n becomes smaller. That will mean that D becomes larger and the efficiency also becomes larger.

These are many words to express the well known fact:

A large slow running propeller is more efficient than a small fast propeller.

We will remember this propeller demand curve till later and discuss the main engine.

2.4 The Main Engine

A diesel engine of the kind we are talking about here is normally and traditionally described by the maximum brake power it can deliver at the maximum revolutions (often per minute, but since we use SI units, power will be in kW and revolutions in per sec, rps. See sec. 5 for relations between units). Earlier the engines were designed for this point and the naval architect had to accept it and find a propeller, of which the demand curve (see fig. 6) did pass through this design point or close to it. Then he had to accept the efficiency following from it and also the rather random speed of the vessel (but of course at least the contract speed). This did not matter when fuel efficiency was not a problem, but today things have changed. The main engine manufacturers have designed their engines in such a way that a high degree of flexibility is available for the ship designer.

For the discussion to follow we use the MAN-B&W engines as examples, but similar features exist with the SULZER engines.

The engines still come with a nominal maximum continuous rating, MCR and maximum revolutions per sec. rps, but the design is such that it allows this point (rps, MCR) to be moved to lower values of both rps and MCR. The area in which this movement is allowable is named as the layout area.

The load diagram, that is the diagram which shows the operating area of the engine: max rps, max load (i.e. max effective pressure) and min rps at given power, is still the same; it only moves within the layout area. (See fig. 7 and 8).

So when the load diagram is moved along the limit of the layout area we will create an area where the main engine can operate or is available. Let us call it an availability area. We will match this area with the demand curve of the propeller in section 2.6.

When the engine is laid out at lower power than the nominal one, this reserve can be utilized to improve the specific fuel oil consumption, sfoc of the engine. That is, the amount of fuel oil needed to produce one kW during one hour. It is normally measured in gram per kW-hour: g/kWh. Combined with this the propeller can be laid out at lower rps, thus becoming larger and more efficient.

The fig. 9 together with fig. 10 explain how to determine the sfoc. Both figures by the courtesy of MAN-B&W.

As can be seen in fig. 9 the sfoc will even be lower at part load of the specified MCR of the engine, this is because the engine by the design of the fuel oil pumps so to speak lays itself out at the lower load, thus improving the sfoc. This has a minimum of about 80% of the specified laid out MCR. Earlier this was called to "derate" the engine, but now one can consider it as it is up to the ship designer to specify where he wants the engine maker to optimize the main engine, so that the vessel operates at the design speed and condition using lowest possible amount of fuel oil.

2.5 The Propeller-Hull Interaction

We have already mentioned the two most important factors of the propeller-hull interaction: The wake fraction and the thrust deduction fraction. A third one to be mentioned is the relative rotative efficiency.

The wake fraction gives the average fraction the speed of water to the propeller is reduced as the propeller experiences it. This wake fraction is called the effective in comparison with the one measured without the working propeller by pitot tubes or laser velocity meters. This wake fraction is called the nominal. The wake obviously varies with speed, hull form, condition of the hull surface, trim and draught of the vessel. How this goes on is yet to be solved. The design chart as given by Harvald in (5) and reproduced in fig. 3 is probably still one of the best available today. Obviously when the propeller diameter grows larger a larger part of the blades will work in the free stream around the hull and the wake fraction will thus be smaller as also indicated in fig. 3. The wake fraction given in fig. 3 is the one which can be measured in a model tank and is therefore scaled. The full size wake

can be expected to be about 20-30% lower. Each shipyard with its own hull form characteristics will be able to set a suitable value.

The thrust deduction fraction indicates how much the propeller pressure field is transferred to the hull. Normally one expects a high value if the hull is close to the propeller. When the propeller diameter becomes larger it will normally come closer to the hull and larger thrust deduction fraction can be expected, which is also indicated in fig. 3. Sharp water lines, some rake on the propeller and good distance in general to the hull surface will improve the thrust deduction fraction. As full size resistance of a ship is very seldom known, the thrust deduction fraction scaling from model to full size is not known either, but is usually considered equal in model and full size.

The efficiency of the propeller working in open water can be measured in model, but the working condition of the propeller behind a ship hull is different due to the confused and often rather turbulent flow which prevails here. This is normally taken care of by multiplying a constant to the open water efficiency in the range of 0.95 to 1.05. This constant is regarded as another efficiency, the rotative one, merely because of the way it multiplies together with other efficiencies (see sec. 2.1), but could just as well be called an ignorance constant, since no acceptable theoretical explanation exists so far. Again each hull form designer will known by experience his rotative efficiency for the type of hull form he designs. No guidance can be given and if no experience is at hand use the value one.

How all these factors work together is already shown in sec. 2.3 where the propeller demand curve was intro-

duced. This curve shows how a given propeller with a given geometry demands a certain power in order to turn at a certain rps, at the same time producing a certain thrust to drive the hull against all types of resistance at a certain speed.

It is clear from the discussion in this section 2 that the Power, P_B, the rps, n, the pitch ratio P/D, and the diameter, D, all are interconnected, and if one wants a propeller with the highest possible efficiency to pass through a certain point (n, P_B) the P/D and D cannot be chosen arbitrarily. In section 3 we will discuss this.

2.6 The Propeller-Hull-Main Engine Interaction

We have now described the propeller demand curve in sec. 2.3 and 2.5 and also the availability area of the main engine in sec. 2.4

Obviously the demand curve of the propeller must fall within the area where the main engine is available. If this is not the case either the propeller must be laid out in a different way and thus maybe decrease in efficiency or another main engine with a more suitable availability area must be chosen.

Fig. 11 shows the combination of the demand curve with the availability area.

This is the name of the game to choose the propeller and main engine in such a way that the demand and availability match at the same time as the ship keeps the specified speed and the fuel consumption is the lowest possible.

Fuel consumption is the product of the power necessary for the speed and the specific fuel consumption at the specified layout of the main engine MCR and rps.

By the rules that a slow running propeller is more efficient than a fast running and by inspection of fig. 9 it is obvious that a large engine (low rps and high MCR) laid out in the part of the layout area which is lowest and most to the left, will be the optimum. But with this optimization problem as with all others there are limiting boundaries. Typically this is space in the engine room: there simply is a limit to the size of main engine which can be accommodated within the hull. Another is cost: larger bore, more slowly running engines are more costly per kW than the smaller more fast running ones. As already mentioned even fuel saving can be bought at a price which is too high and also today money is a scarce commodity to be obtained at a high price and therefore also to be saved.

The good ship designer will be able to combine these things and come up with the true optimum choice of propeller and main engine based on the conditions given to him.

3. The Layout of the Propeller and Main Engine

This section will discuss some of the conditions for propeller and main engine layout and shows the method to use when the optimization work should be done.

3.1 Conditions for Layout of the Propeller

Very often the discussion is raised of which condition

to use for the propeller:

Full load, ballast, intermediate draught? With or without sea margin or added resistance? In a trial condition (that is clean hull, still water and lowest wake fraction) or in a service condition with high extra resistance (how high?) and larger wake fraction due to fouling on the hull and propeller.

It is the author's opinion that the propeller of course should be optimized for a condition most typical for the intended operation of the ship and only the ship operator (or owner if he operates it) can define this. Very often that is not the case since the operator does not know or do not understand, and the ship designer must make the choice.

For ordinary cargo vessels a condition on full load with an added resistance of 15% and a wake fraction as for a clean hull (since self polishing coating, SPS, is now more and more used on new ships) seems the natural and physically well defined condition.

The often used and misunderstood way of doing it in trial and then layout 3 or 4 revolutions per minute "light" (that is 3 or 4 rpm higher) is very unscientific and highly unsatisfactory to logical and rational approach to this problem.

3.2 Limiting Diameter

With today's very slow running engines normally the largest possible propeller diameter is the one with the best efficiency. But certain restrictions can be laid on the diameter.

First of course if the ship is small with a small draught. The propeller in the service condition must be reasonably submerged between 25%-35% of the propeller diameter.

But even large ships can have a limited draught. Typically container ships, since the containers normally do not weigh very much compared to size.

Other ships like tankers and bulk carriers operate more than half their time in ballast with a low draught. Bulk carriers can be trimmed aft or take water ballast in one or more of the holds, but tankers today must comply with the IMO regulations. So with minimum segregated ballast the propeller must be submerged with a draught of 2% of the length plus 2 m and maximum trim of 1.5% of length. This rule does not create a large draught aft. A tanker of 750 feet or 228.6 m needs only to have a draught of 8.3 m aft which is very close to the propeller diameter one can obtain with very big slow running engines.

Normally this is considered to be so bad that a smaller more fast running engine with less favorable sfoc is chosen. But it is not always necessary to take this sad decision since the efficiency of the propeller is not so sensitive to the diameter as discussed previously (page 9).

As an example can be mentioned a 30,000 tdw tanker with an optimum propeller of 6.5 m, a speed of 15.6 knots and a pitch ratio of 0.67. If this is changed to a 6 m propeller (0.5 m less) speed is only reduced to 15.56 (0.25%), same power and rps, pitch ratio changes to 0.80. Such a change is well inside any measuring accuracy and is only interesting from a theoretical point of view, not a practical. The conclusion is that a li-

miting diameter is not necessary, a very crucial factor in the choice of engine.

If one tries to see how the speed of the ship varies with rps, on different optimal demand curves (see fig. 6) one will see that for constant speed, P_B will vary with n according to the following

$$P_B \sim n^p \qquad 3.2\text{-}1$$

p will for slender ships be around 0.2 and for fuller ships around 0.3. Each designer will soon get experience on his own hull form.

This idea comes in very handy when the engine and propeller should be matched. This we will discuss in the following section.

3.3 The Combined Layout of Propeller and Engine

Formula 3.2-1 tells us that when n becomes smaller so does the power needed for the same speed. This means an optimum propeller laid out at low rpm requires less power and therefore one should think less fuel consumption. By inspection of fig. 12 one sees that this is not the whole story since the specific fuel oil consumption could be higher. Take a look at fig. 12. Propeller I is laid out at 6% higher rps than the other propeller II at the points where the required speed is obtained and needs appx. 1.5% more power to obtain this speed than propeller II. On the lower part of fig. 12 one observes that propeller II has a higher sfoc than I: appx 1 g/bhph that is, with the engine in question appx. 0.8% higher spfoc. The curve A and B is drawn as the specified MCR was in the "A" and "B" point respectively. In total propeller II is only about 0.7% better

in fuel consumption than I. To this we must remember that one has to pay a larger propeller and shaft so a thorough consideration is appropriate.

Often the owner wants to operate the engine at 90% load (even if it for many years has been known that the C in MCR really means continuous) and therefore the specified MCR must lay 10% above the operating point, but at the same time within the layout area. In fig. 12 one can see this requirement is easily obtained in point "A", but not quite in point "B". A layout of the propeller should then be just to the right of propeller II (the broken line) and the engine MCR at the upper boundary of the layout area. The specified (MCR, rps) is then (93%, 93%) of the nominal or catalogue values. Also the load diagram is shown. The specific fuel oil consumption in the operating point "C" is also indicated in the lower part of fig. 12.

4. Symbols and Units

Even if not all of the symbols in the following list is used in this paper it is given here.

The symbols have been chosen in accordance with the "Standard Symbols", 1975, recommended by the Presentation Committee, ITTC, where applicable.

General:

g = acceleration due to gravity

C = constant.

Ship dimensions:

L = L_{WL} = length of waterline

L_{pp} = length between perpendiculars

B = breadth on waterline, moulded

T = draught, moulded

S = wetted surface (L x mean girth) (including rudder)

∇ = displacement volume

Δ = mass of displacement or mass of ship

Dw = deadweight (mass)

LCB = longitudinal centre of buoyancy (also used to denote the distance of the centre of buoyancy abaft amidship (or $L_{pp}/2$))

A_M = area of midship section.

Engine dimensions:

D = cylinder diameter

s = stroke length

c = number of cylinders.

Propeller dimensions:

Z = number of blades

D = propeller diameter

P = propeller pitch

A_0 = propeller disc area $\frac{\pi D^2}{4}$

A_E = expanded blade area.

Kinematic and dynamic symbols:

V = speed of ship

V_A = speed of advance of propeller

R = R_T = total resistance

T = propeller thrust

Q = propeller torque

n = rate of revolution

P = power

P_B = brake power

P_D = delivered power at propeller

P_E = RV = effective power

P_I = indicated power

P_S = shaft power

P_T = thrust power
P_I = indicated mean pressure
P_B = mean effective pressure
P_{max} = maximum pressure
ρ = mass density of water
ν = kinematic viscosity of water.

Dimensionless coefficients and ratios:

$\delta = \frac{\nabla}{LBT}$ = block coefficient

$\beta = \frac{A_M}{BT}$ = midship section coefficient

$\varphi = \frac{\nabla}{A_M L}$ = prismatic coefficient

$F_n = V/\sqrt{gL}$ = Froude number

$R_n = \frac{VL}{\nu}$ = Reynolds number

$\frac{L}{\nabla^{1/3}}$ = length-displacement ratio

$J = \frac{V_A}{nD}$ = advance number of propeller ($J' = \frac{V}{nD}$)

$K_Q = \frac{Q}{\rho n^2 D^5}$ = torque coefficient

$K_T = \frac{T}{\rho n^2 D^4}$ = thrust coefficient

$C_{TH} = \frac{T}{\frac{1}{2}\rho V_A^2 A_0}$ = thrust loading coefficient

$w = \frac{V - V_A}{V}$ = Taylor wake fraction

$t = \frac{T - R_T}{T}$ = thrust deduction fraction

η_B = propeller efficiency behind ship

$\eta_H = \frac{1 - t}{1 - w}$ = hull efficiency

η_M = mechanical efficiency of main engine

$\eta_0 = \eta = \frac{J}{2\pi} \cdot \frac{K_T}{K_Q}$ = propeller efficiency in open water

η_R = relative rotative efficiency

η_S = shaft efficiency

η_T = total efficiency

$b_T = \frac{Tn^2}{\rho V_A^4}$ = 1st basic coefficient

$b_Q = \frac{Qn^3}{\rho V_A^5}$ = 2nd basic coefficient

Système International d'Unités (SI) are used throughout.

The relation to earlier used units is as follows:

1 kW 1.360 metric horsepower (1 hp = 735.499 W)
1 kW 1.341 British horsepower (1 hp = 745.700 W)
1 t = 1000 kg ~ 0.984 ton (1 ton = 2240 lb = 1.01605 t)
1 m s^{-1} ~ 1.944 knots (1 kn = 1852 m h^{-1} ~ 0.5144 m s^{-1}).

5. References

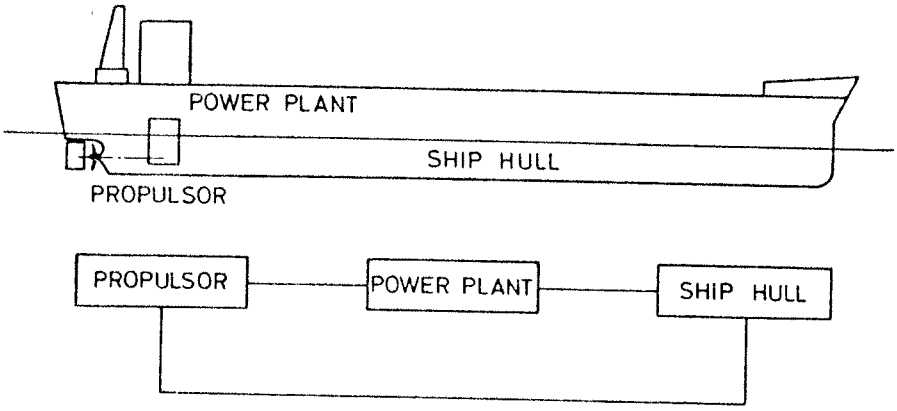
(1) GERTLER, M. 1954. A Re-analysis of the original test data for the Taylor Standard Series. Report 806, David Taylor Model Basin, Washington.

(2) GULDHAMMER, H.E. and HARVALD, S.A. 1974. Ship resistance, effect of form and principal dimensions, Akademisk Forlag, København.

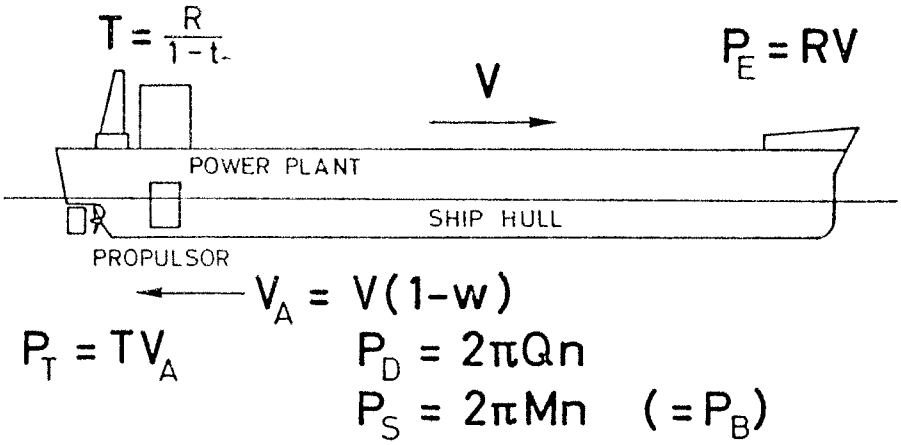
(3) HARVALD, S.A. 1977. Prediction of power of ships. Instituttet for Skibs- og Havteknik, Danmarks Tekniske Højskole, Lyngby.

(4) HARVALD, S.A. and HEE, J.M. 1978. An outline of a shipyard's need for model testing. Instituttet for Skibs- og Havteknik, Danmarks Tekniske Højskole, Lyngby.

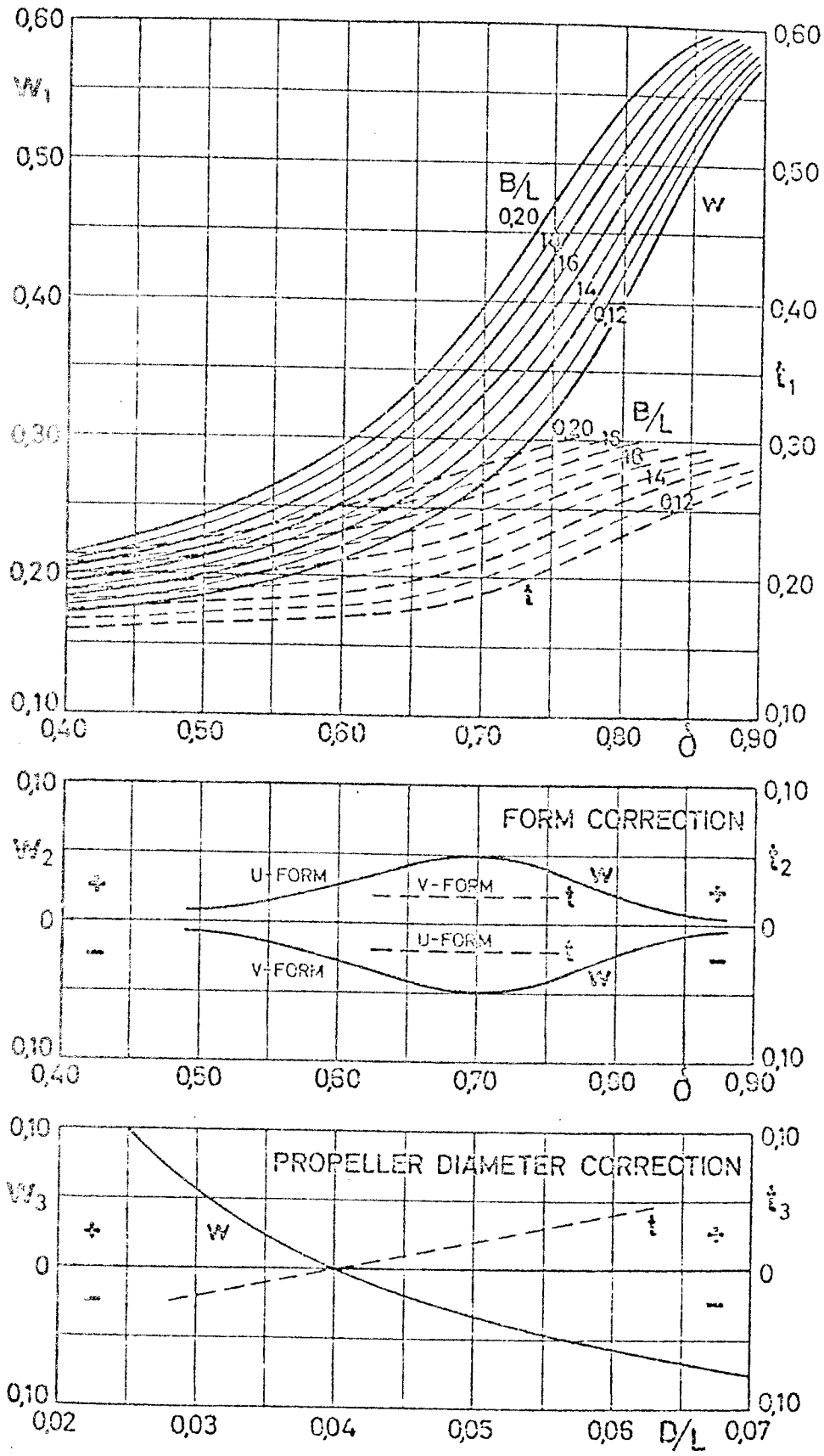
(5) OOSTERVELD, M.W.C. and VAN OOSSANEN, P. 1975. Further computer-analysed data of the Wageningen B-Screw series, Int. Shipbldg. Prog. No. 251.



Figur 1.



Figur 2.



The wake and thrust deduction coefficient for single screw ships. $w = w_1 + w_2 + w_3$. (Harvald)

$t = t_1 + t_2 + t_3$.

Figur 3.

ISH-Propeller Diagram B4:,50

Non-dimensional coefficients:

Advance coefficient: $J = \frac{V_A}{n \cdot D}$

Thrust coefficient: $K_T = \frac{T}{\rho \cdot n^2 \cdot D^4}$

Torque coefficient: $K_Q = \frac{Q}{\rho \cdot n^2 \cdot D^5}$

Thrust loading coefficient: $C_{Th} = \frac{T}{\rho/2 \cdot V_A^2 \cdot \pi/4 \cdot D^2} = \frac{8 \cdot K_T}{\pi \cdot J^2}$

Basic thrust coefficient: $b_T = \frac{T \cdot n^2}{\rho \cdot V_A^4} = \frac{K_T}{J^4}$

Propeller efficiency: $\eta_0 = \frac{P_T}{P_D} = \frac{T \cdot V_A}{Q \cdot 2 \cdot \pi \cdot n} = \frac{K_T}{K_Q} \cdot \frac{J}{2\pi}$

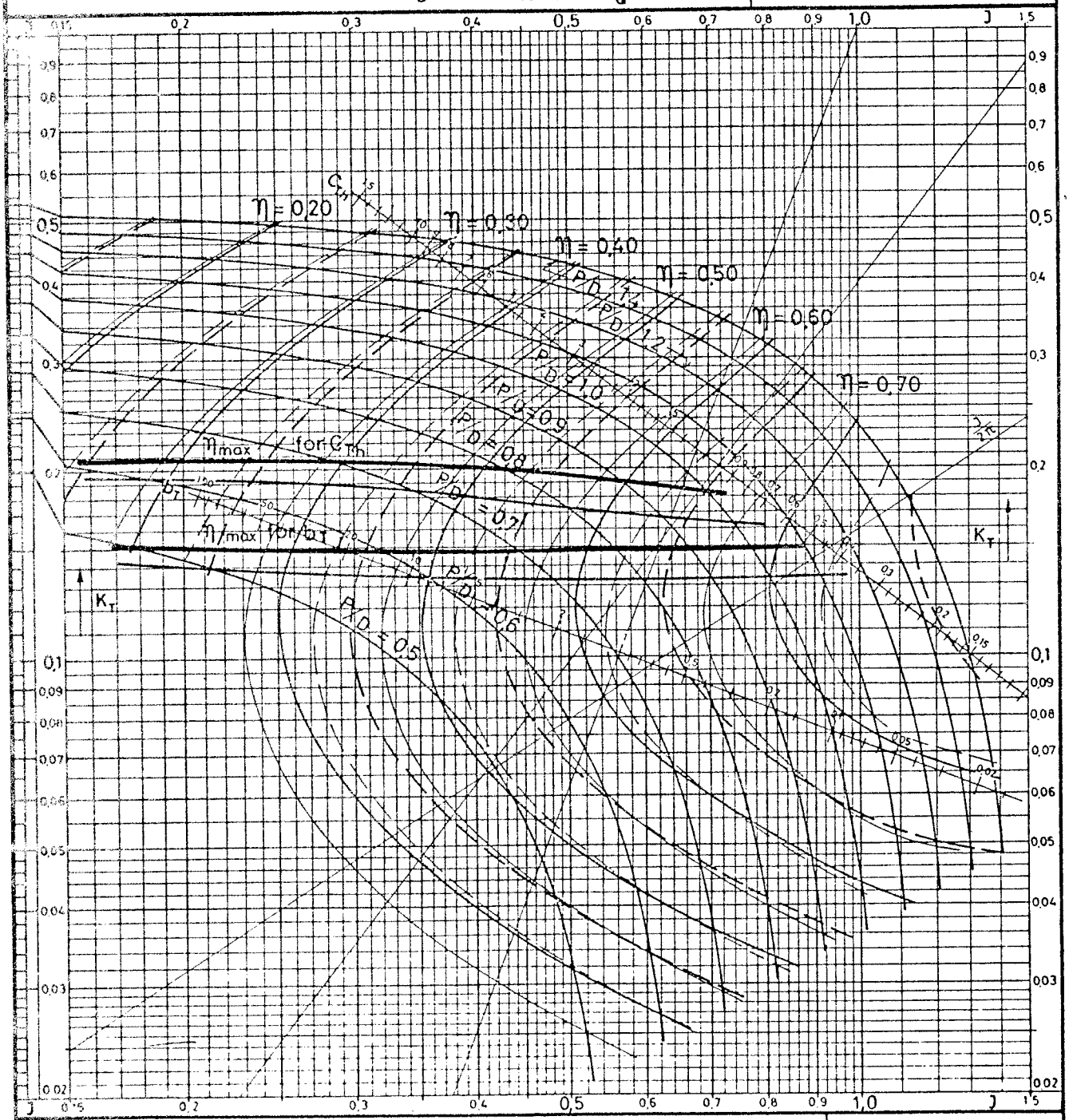
Wageningen B-series

Number of blades: **4**

Expanded area ratio: **0,50**

$Rn = 2 \cdot 10^6$ ———

$Rn = 2 \cdot 10^7$ ———



Instituttet for Skibs- og Havteknik
Department of Ocean Engineering

Danmarks Tekniske Højskole
LYNGBY, Danmark

B4:,50

Fig. 4.

ISH-Propeller Diagram B4:,50

Non-dimensional coefficients:

Advance coefficient: $J = \frac{V_A}{n \cdot D}$

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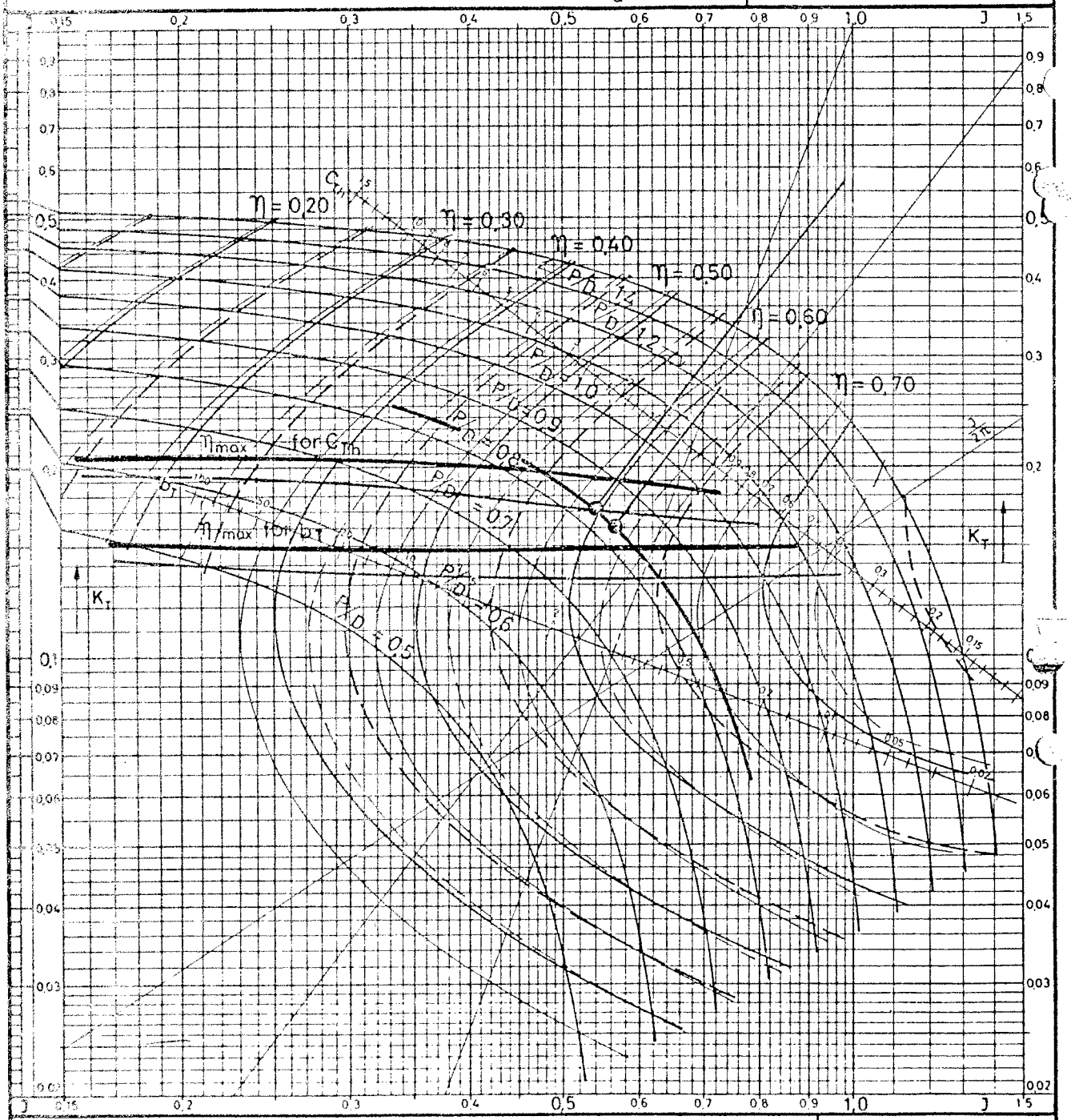
Rn = $2 \cdot 10^7$ ———



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Department of Ocean Engineering

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LYNGBY, Danmark

B4:,50

Fig. 5.

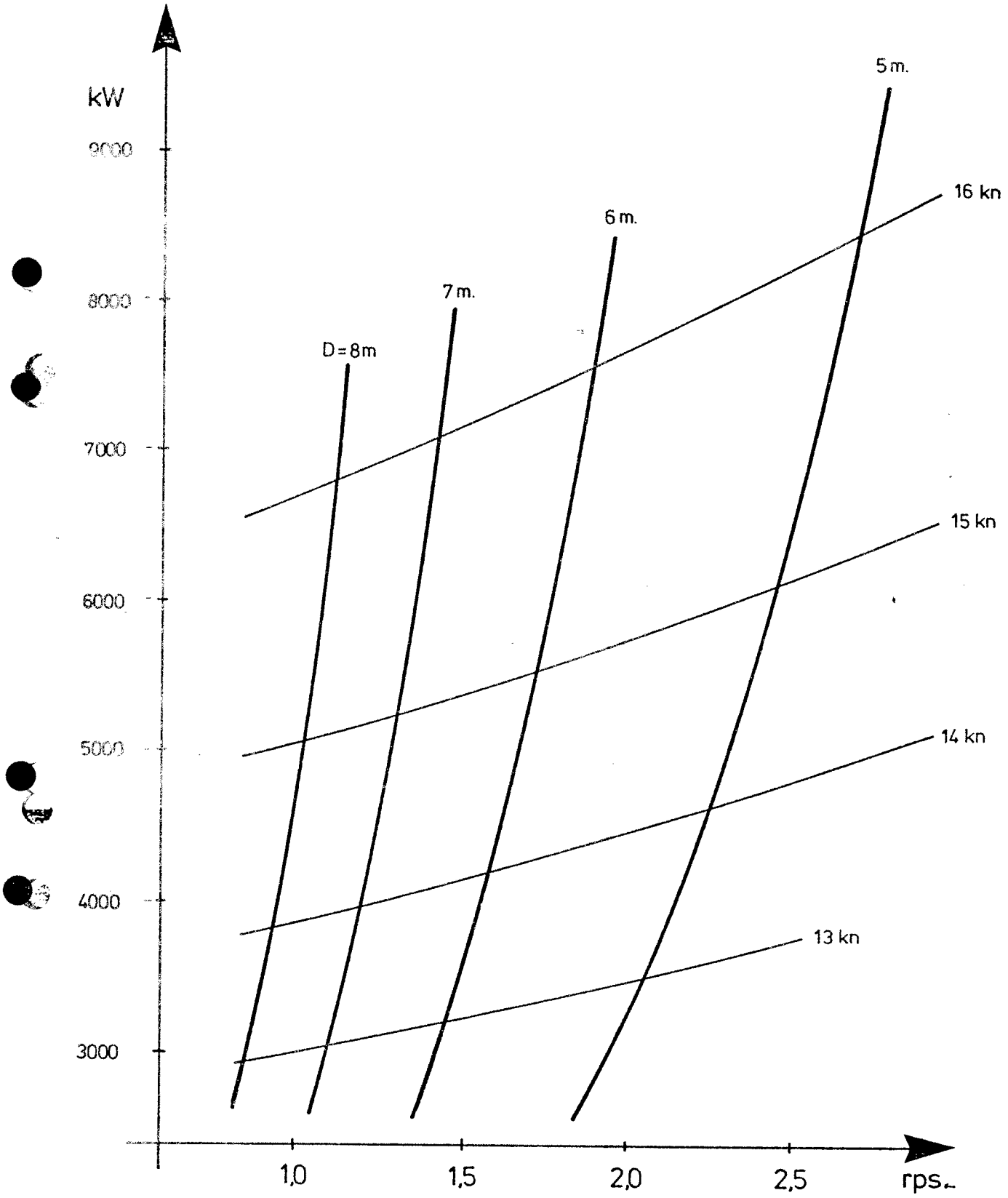

kW
9000
8000
7000
6000
5000
4000
3000
D=8m
7 m.
6 m.
5 m.
16 kn
15 kn
14 kn
13 kn
1,0
1,5
2,0
2,5
rps.

Fig. 6

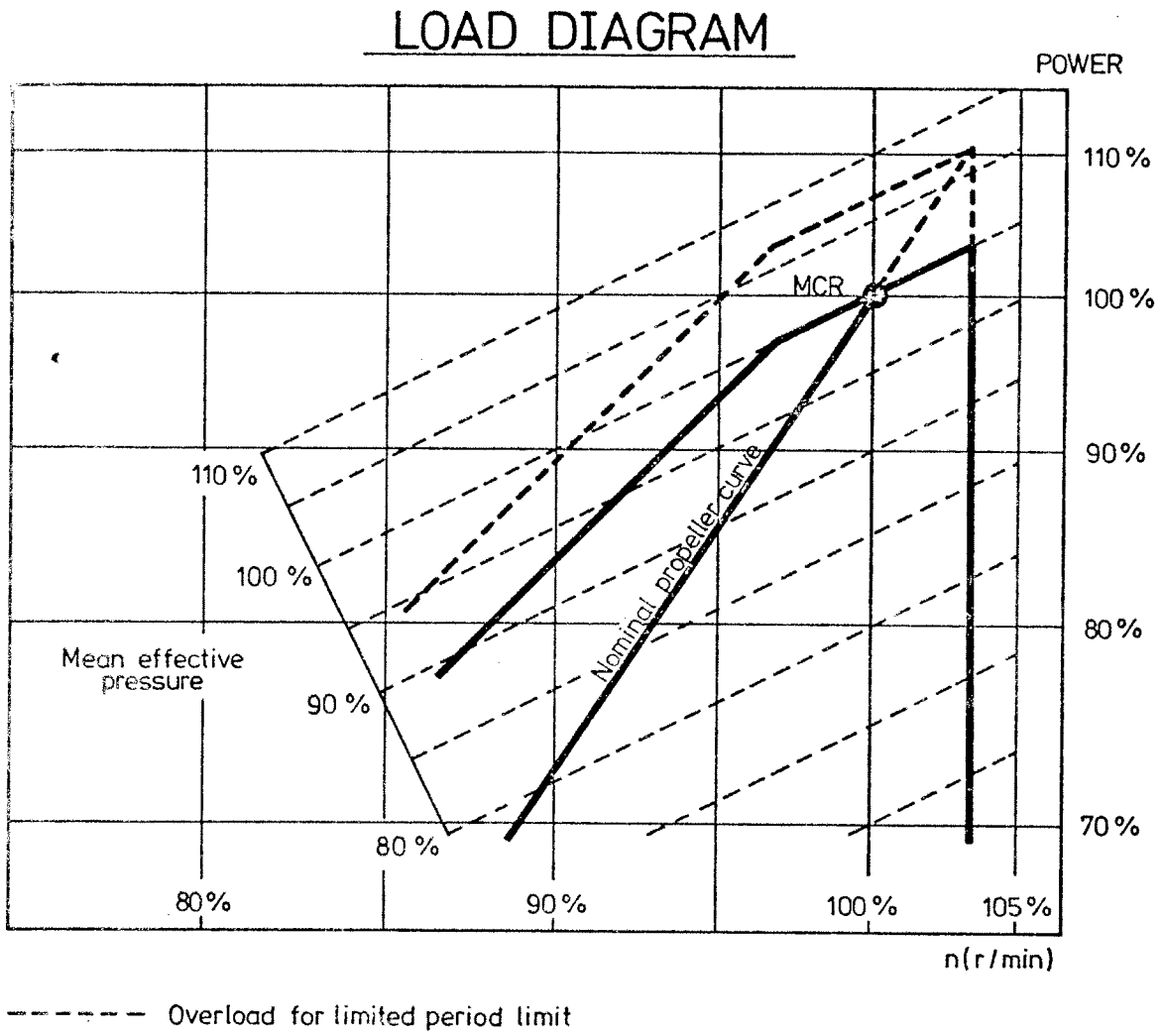

LOAD DIAGRAM
POWER
110 %
100 %
90 %
80 %
70 %
MCR
Nominal propeller curve
110 %
100 %
90 %
80 %
Mean effective pressure
80 %
90 %
100 %
105 %
n(r/min)
Overload for limited period limit

Fig. 7

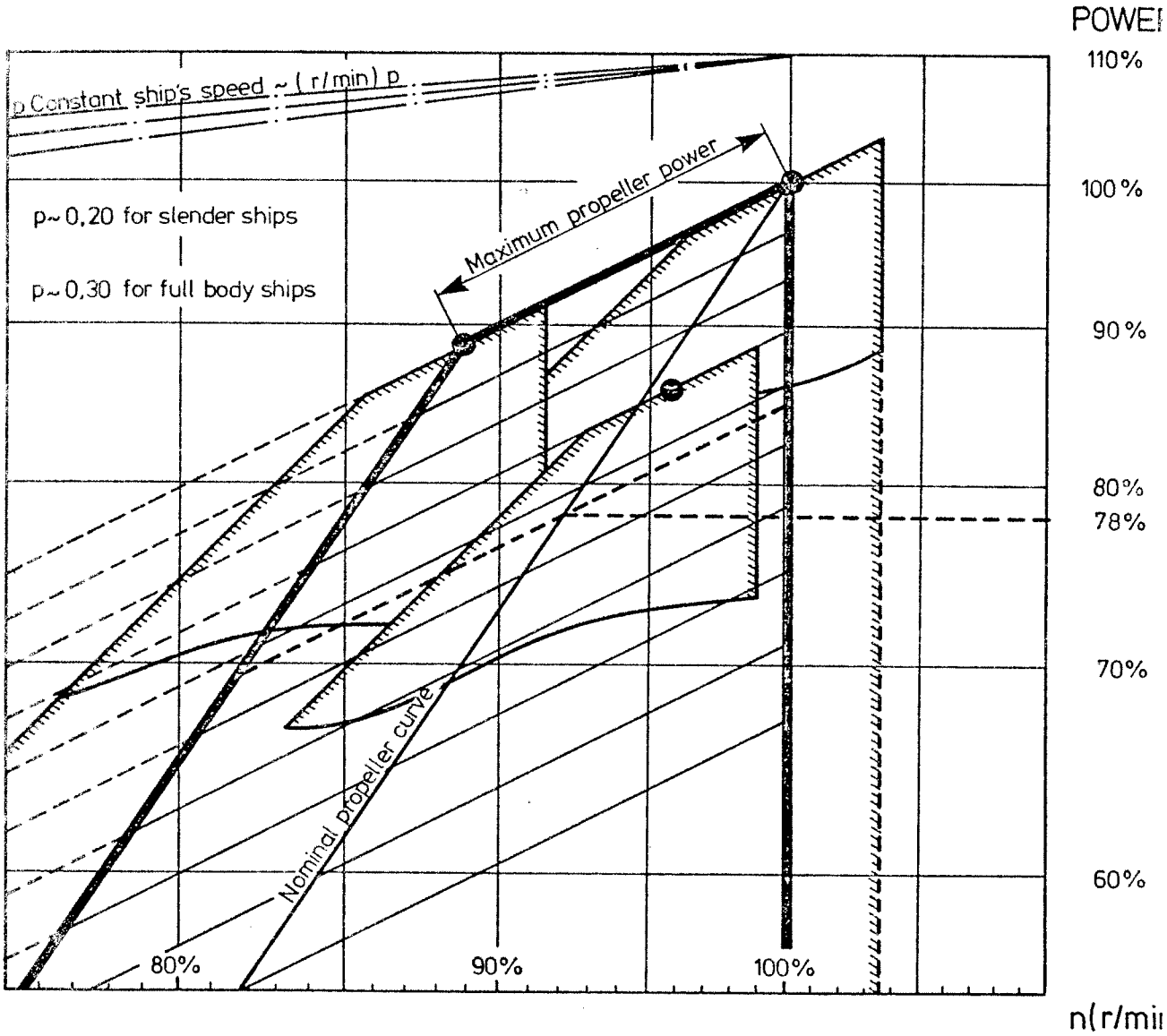
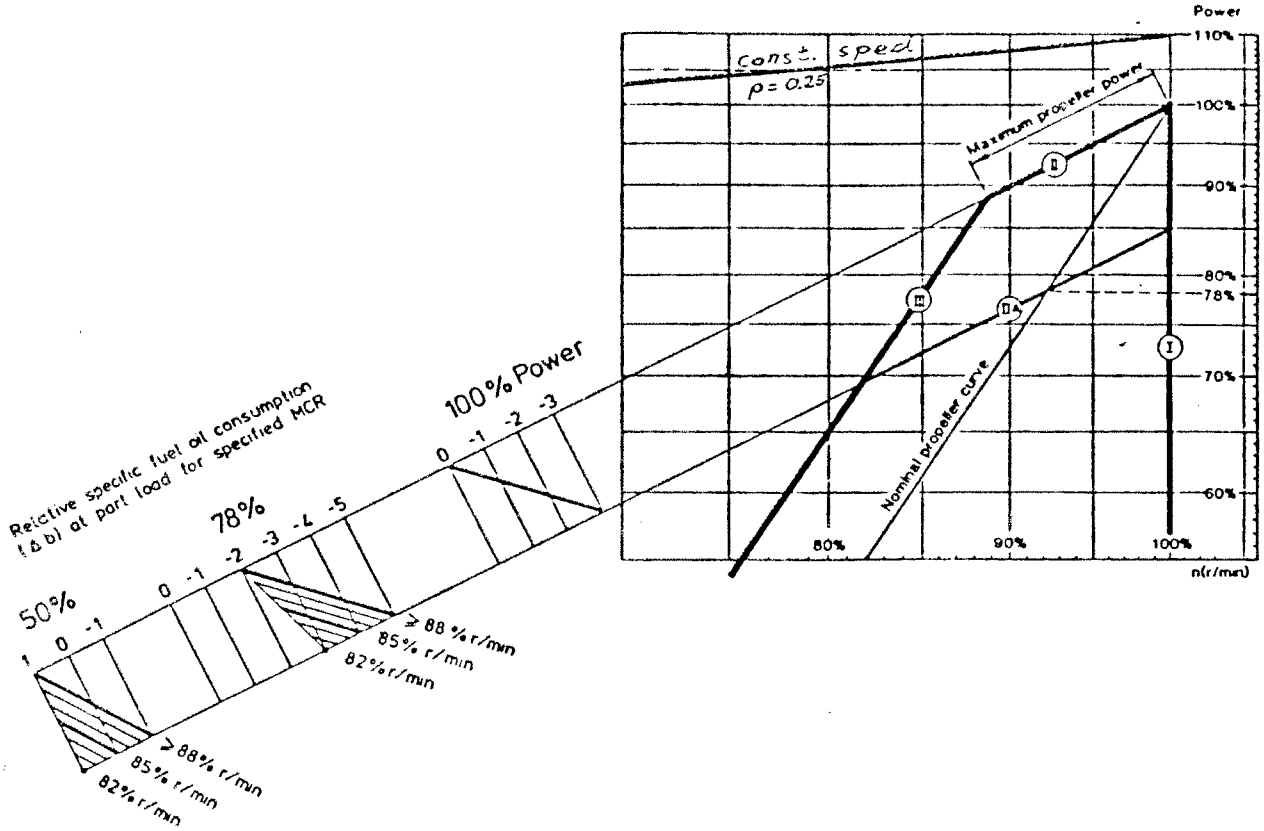
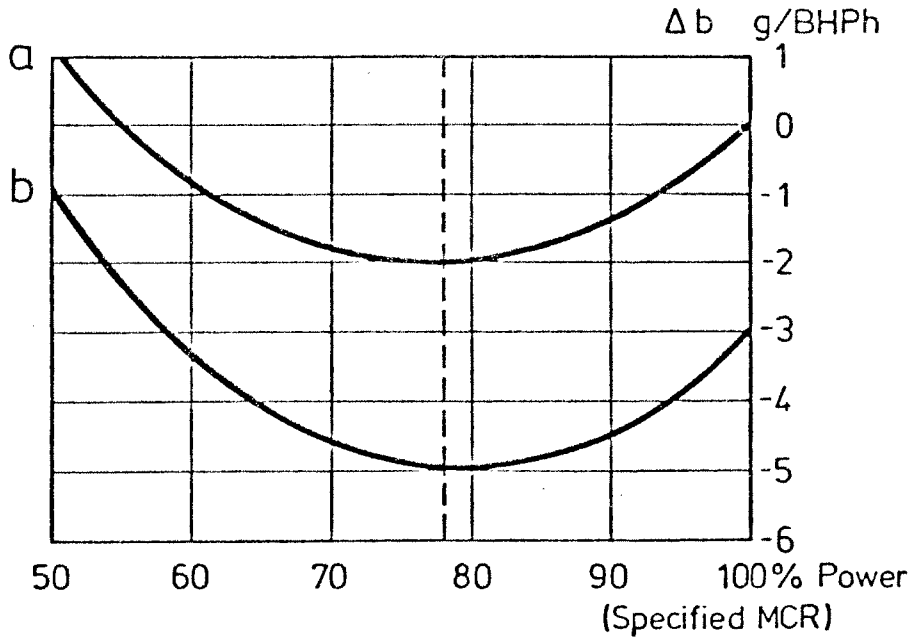

POWE
110%
100%
90%
80%
78%
70%
60%
p Constant ship's speed ~ (r/min) p
p~0,20 for slender ships
p~0,30 for full body ships
Maximum propeller power
Nominal propeller curve
80%
90%
100%
n(r/mi

Fig. 8.

Fuel oil consumption at specified MCR and Part Load

L-MCE





Examples:

a) MCR point on line II

b) MCR point on line IIA (r/min ≥ 88%)

Fig. 9.

Graphs for Part Load Consumptions

The specific fuel oil consumption at 100%, 78% and 50% of the specified MCR compared to the specific fuel oil consumption at nominal MCR (catalogue value) is given by heavy graphs to the left of the lay-out diagram.

Any point within the limits of the lay-out diagram determined by the lines I, II, II A and III can be chosen at specified MCR point.

Determination of the Part Load Consumptions

The consumption figures at the specified MCR and part load can be determined from the graph as follows:

A straight line parallel to line II is drawn through the specified MCR point.

This line intersects the lines for specific fuel oil consumption at 100%, 78% and 50% power. The intersection points show the figures for specific fuel oil consumption in relation to the nominal specific fuel oil consumption.

Please observe that the intersection points to be used at 78% and 50% power depend on the RPM of the specified MCR point (above 88% or between 88% and 82% of the nominal RPM of the engine).

Below the lay-out diagram the corresponding specific fuel oil consumption figures from the 100%, 78% and 50% graphs can be plotted and the part load curve can be drawn.

As an example is shown a consumption curve (a) for the MCR point placed on line II and (b) for the MCR point placed on line II A (for r/min = 88% of nom. r/min). The consumption figure "0" refers to the specific consumption at nominal MCR (the catalogue value).

Fig. 10

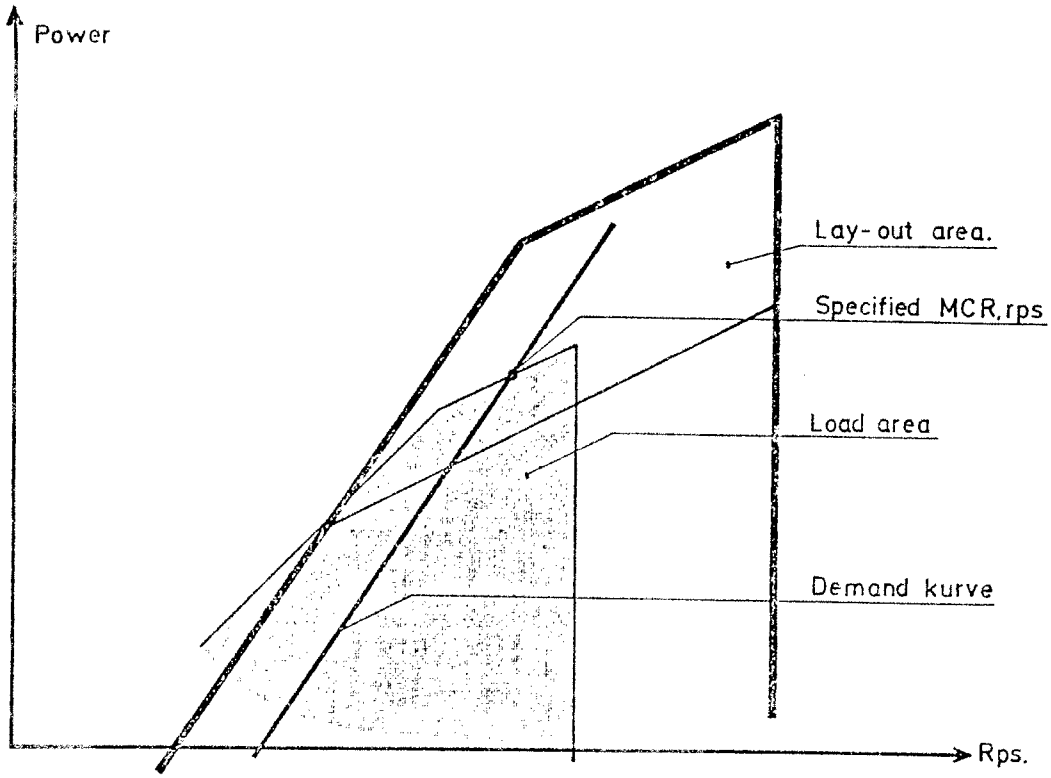
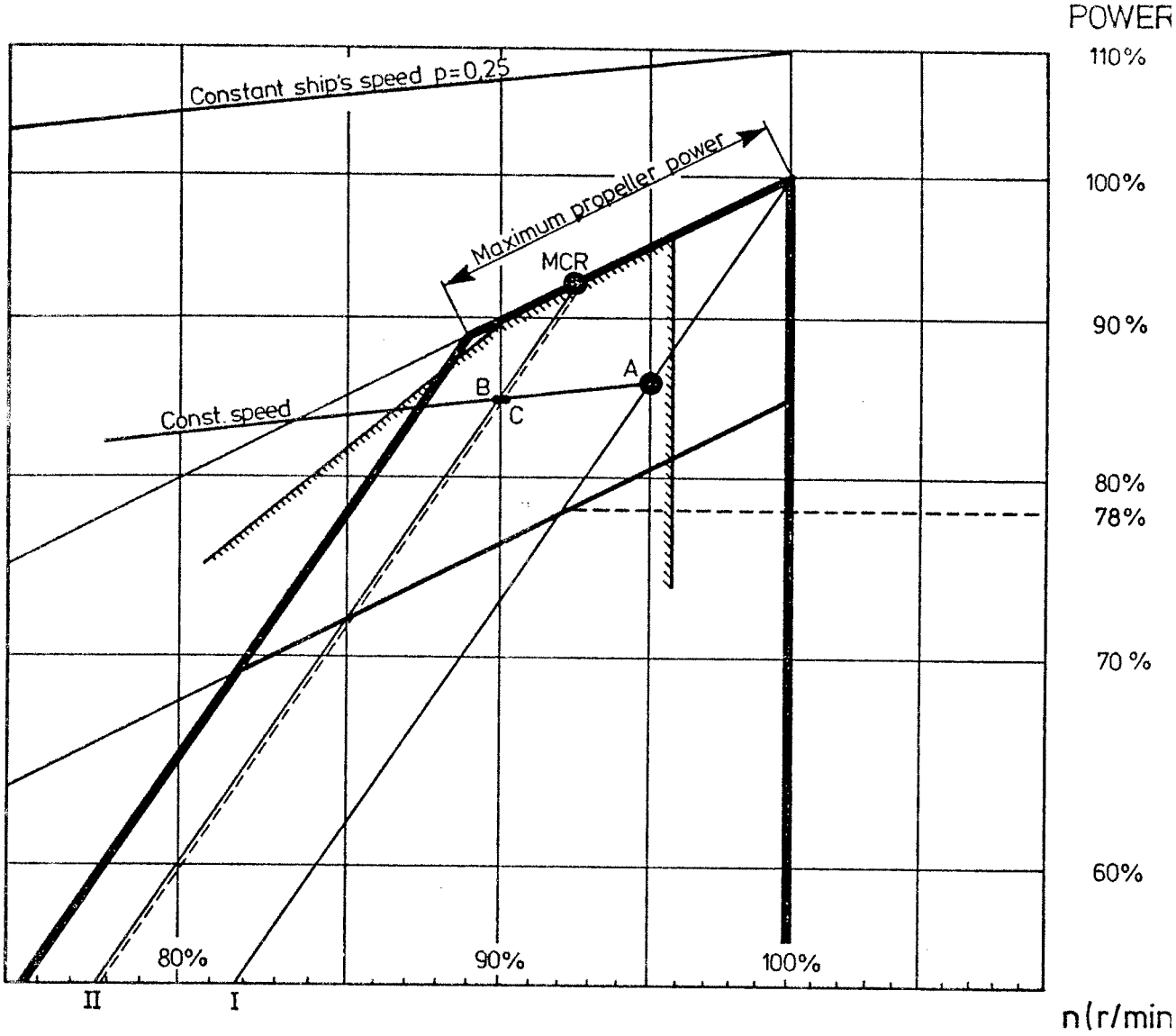

Power
Lay-out area.
Specified MCR,rps
Load area
Demand kurve
Rps.

Fig. 11.


POWER
110%
100%
90%
80%
78%
70%
60%
Constant ship's speed p=0,25
Maximum propeller power
MCR
A
B
C
Const. speed
80%
90%
100%
II
I
n(r/min

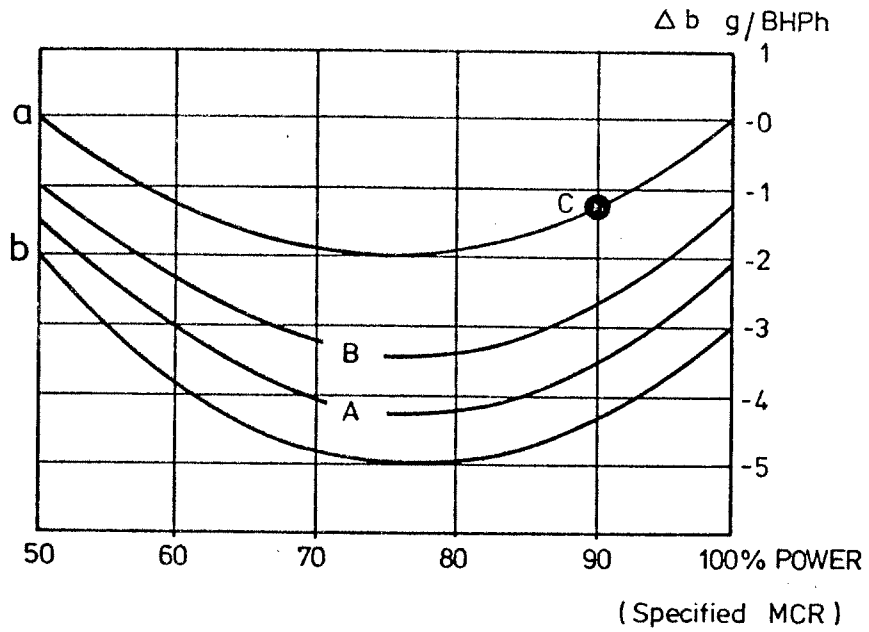

Δ b g/BHPh
1
-0
-1
-2
-3
-4
-5
a
b
C
B
A
50
60
70
80
90
100% POWER
(Specified MCR)

Fig. 12

WEST EUROPEAN GRADUATE EDUCATION
IN MARINE TECHNOLOGY

EIGHTH SCHOOL Aug 1983

in Gothenburg

SHIP DESIGN FOR FUEL ECONOMY

Ship routeing

by

Prof.ir. J.A. Spaans

Delft University of Technology
Dept. Marine Technology
Delft

SHIP ROUTEING

1. Definition of the problem

Ship routeing is a procedure where an optimum track is determined for a particular vessel on a particular run, based on expected weather, sea state and ocean currents.

Optimisation can be preformed in terms of
(i) minimum passage time
(ii) minimum fuel consumption within a specified passage time
(iii) minimum damage to ship and/or (deck)cargo by the sea
(iv) maximum comfort to passengers
(v) or a combination of the above criteria.

For most commercial purposes a combination of the criteria (i) and (iii) or (ii) and (iii) is preferable.

2. Historical review

For many centuries navigators have gathered climatological and hydrographic data to be used for later voyages.

The first published "Sailing directions" now known, are the 13th century "Compasso de Navigare" for the Mediterranean and the 15th century "Al Muhet" bij Ibn Mâjid for the Northern Indian Ocean and adjacent waters. This famous Arabic navigator was Vasco da Gama's pilot on his first trip across the Indian Ocean in 1498.

Prince Henry The Navigator (1394-1460) promoted the scientific approach to navigation and gathered the best cartographers, astronomers and other scientists of that time in his nautical centre in Sagres Portugal. Shipmasters collected information on their trips and had afterwards to report in Sagres where the information was gathered and processed into charts and pilot books.

In the 16th century the know-how of the Iberians spread over Europe, mainly by the standard work:
"Arte de Navigar" by Pedro de Medina
Valladolid 1545.
This work was translated and published in Italy, France, Holland, England and Germany [1] .

In the 17th century Dutch mariners and cartographers contributed considerably to the art of navigation, their charts and sailing directions were of world fame.

Benjamin Franklin (1706-1790) published the first Gulf Stream chart in 1770. He gathered his information mainly from the shipmasters of "New England whalers". The passage time of transatlantic mailboats was considerably reduced by Franklin's chart.

Matthew Maury (1806-1873) was the founder of Maritime meteorology and oceanography. He published the first "Pilot Charts" in 1845, containing seasonal wind and current information . The average passage time from New York to California round the Horn for instance was reduced from 183 days to 139 days by using Maury's season charts [2] .

At the end of the 19th century and in the beginning of the 20th century the "Deutsche Seewarte" published detailed Sailing directions (Segelhandbücher) for shipmasters of sailing vessels. Figure 1 gives the average timefronts (isochrones) for sailing vessels outward bound from the Lizard as given in [3] . The perfect voyage planning was certainly one of the major reasons why the Germans, like Ferdinand Laeisz' flying P-line, managed to operate sailing vessel profitably until far into the 20th century, when other seafaring nations had already totally switched to engine propulsion. (Only the Finish sailing fleet of Gustaf Erikson from Mariehamn managed to operate square rigged sailing vessels until the second world war, with cheap second-hand vessels, boyscrews and relatively low wages). A historical lesson is to be learnt from Ferdinand Laeisz: "The ship owner operating with optimal voyage planning will survive where others have to give up"!

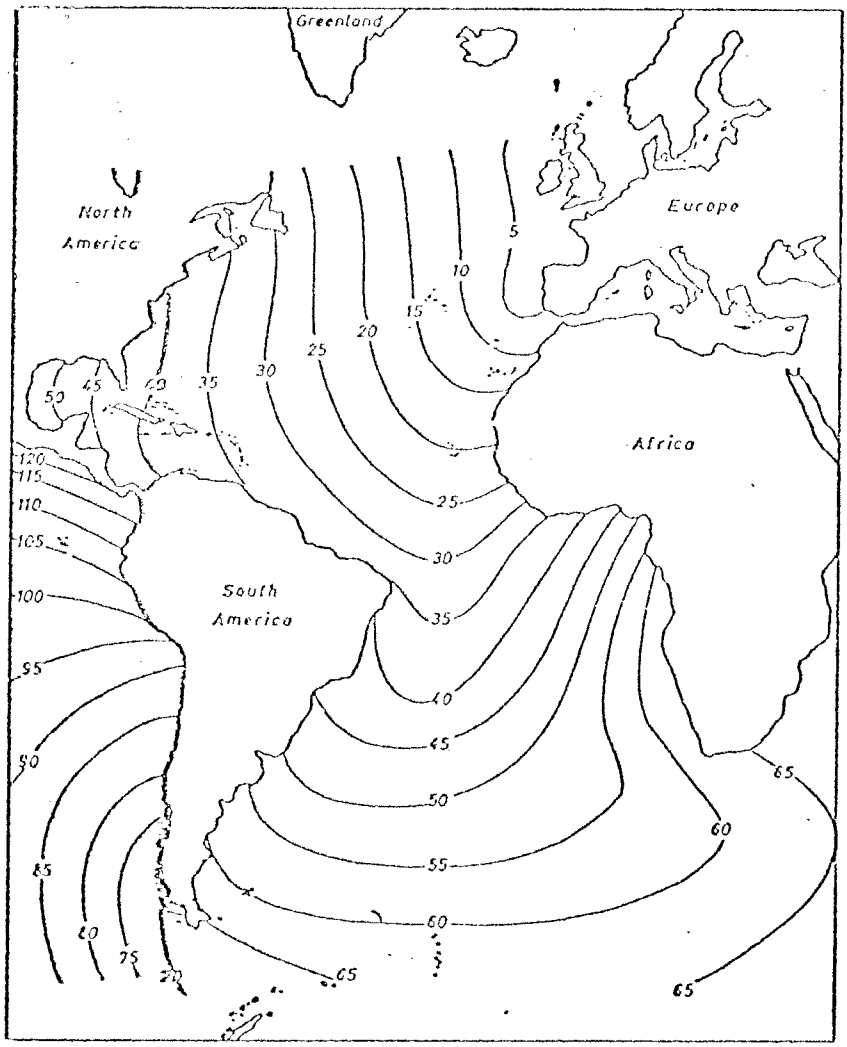


Figure 1. Isochrones outward bound from the Lizard.

In 1952 as first commercial shipping company the American President Lines started to recommend routes to bypass the worst storms for the Line's ships operating in North Pacific. A meteorologist was enlisted to provide forecasts in more detail than government predictions [4] . Simultaneously other shipping companies were independently thinking along the same lines. By the end of the 1950's two American consulting firms - Oceanroutes and Pacific Weather Analysis, which later became Weather Routing Incorporated - had been formed, later followed by Bendix Marine Science Services.

In 1960 the Royal Netherlands Meteorological Institute at De Bilt was the first office in Europe to provide routeing services for the North Atlantic, followed by Bracknel in the U.K. and the Marine Meteorological Office in Hamburg.

3. Climatological routeing

Average values or frequency distributions of winds, sea state, ice limits and ocean currents can be found in climatological atlases or special season or monthly charts. The mariner can find this information including recommended tracks in

- Ocean Passages for the World (B.A.)
- Pilot Charts (USA)
- Sailing directions and pilot books (B.A. and others)
- Climatological atlases published by Meteorological or Hydrographic offices.

"Ocean Passages" gives recommended routes for sailing vessels, low powered vessels and powered vessels, see figure 2.

The "Pilot Charts" give information of ocean currents, ice limits, wind distribution and recommended tracks, see figure 3. The B.A. Pilots give additional information in descriptive form.

Using climatological data for routeing is only justified in areas where the weather situation is stable like the trade wind and monsoon areas outside the cyclone season.

When carried out on board, climatological routeing is usually done without rigorous computer programs. The shipmaster assisted by his navigating officer compares loxodromic and great circle routes, takes advantage of currents, avoids areas with a high storm frequency especially when counterwinds are expected, and avoids areas with low visibility; the ice limits and the regulations of the load-line convention are taken into account when necessary. Often the recommended route or near-by one is taken as the best as these routes are the result of age-long experience of mariners. However, different types of vessels in operation will require different routes. For instance a fully loaded tanker is less subject to damage by heavy seas than a vessel with cargo on deck and consequently the latter will be routed on a track with a lower storm frequency.

Navigators are taught at nautical colleges how to tackle these problems. Case-studies of different types of vessels and different routes are carried out. In the following section it is shown that better results can be reached with weather routeing.

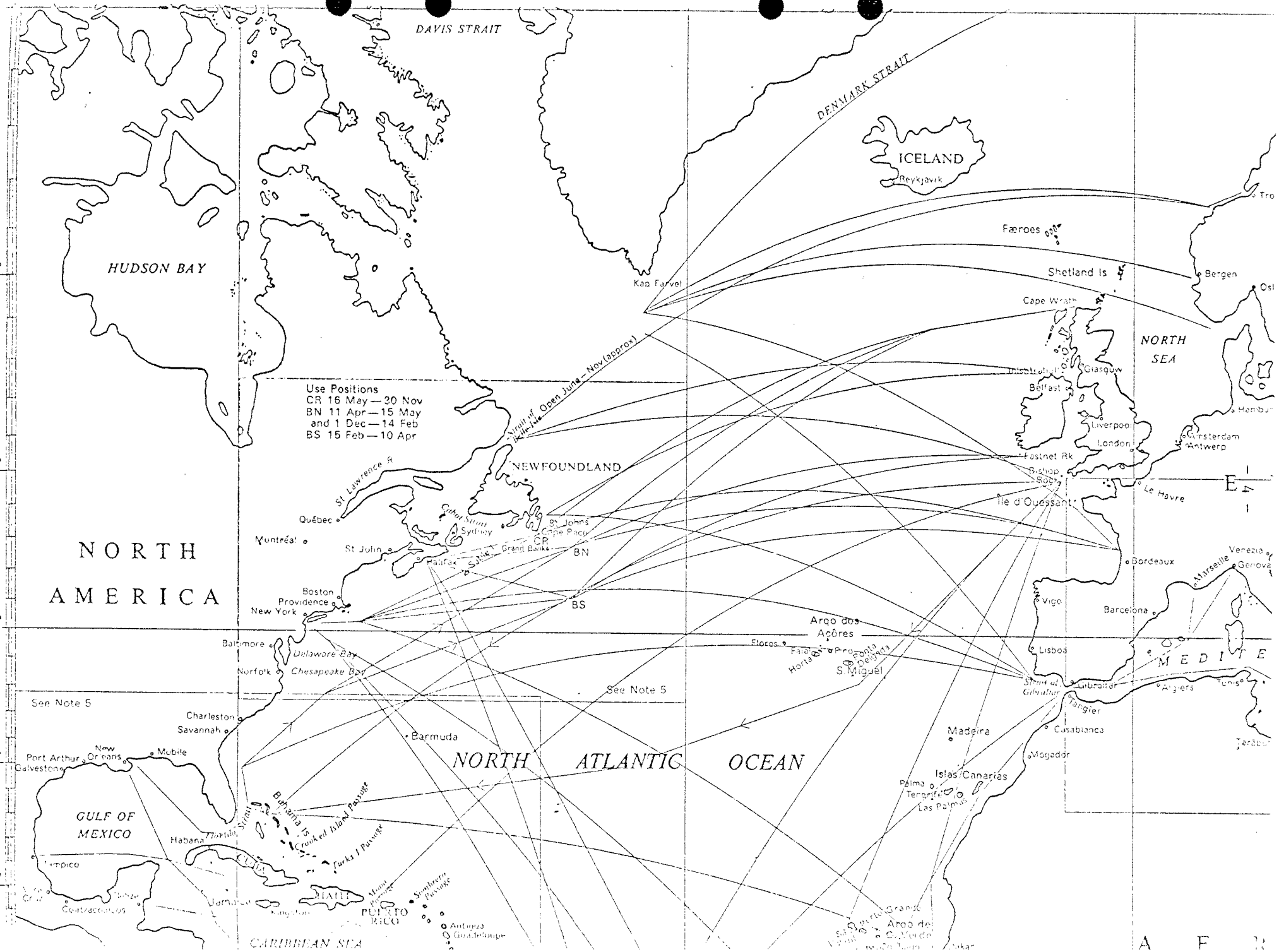


Figure 2. Ocean passages for the world.

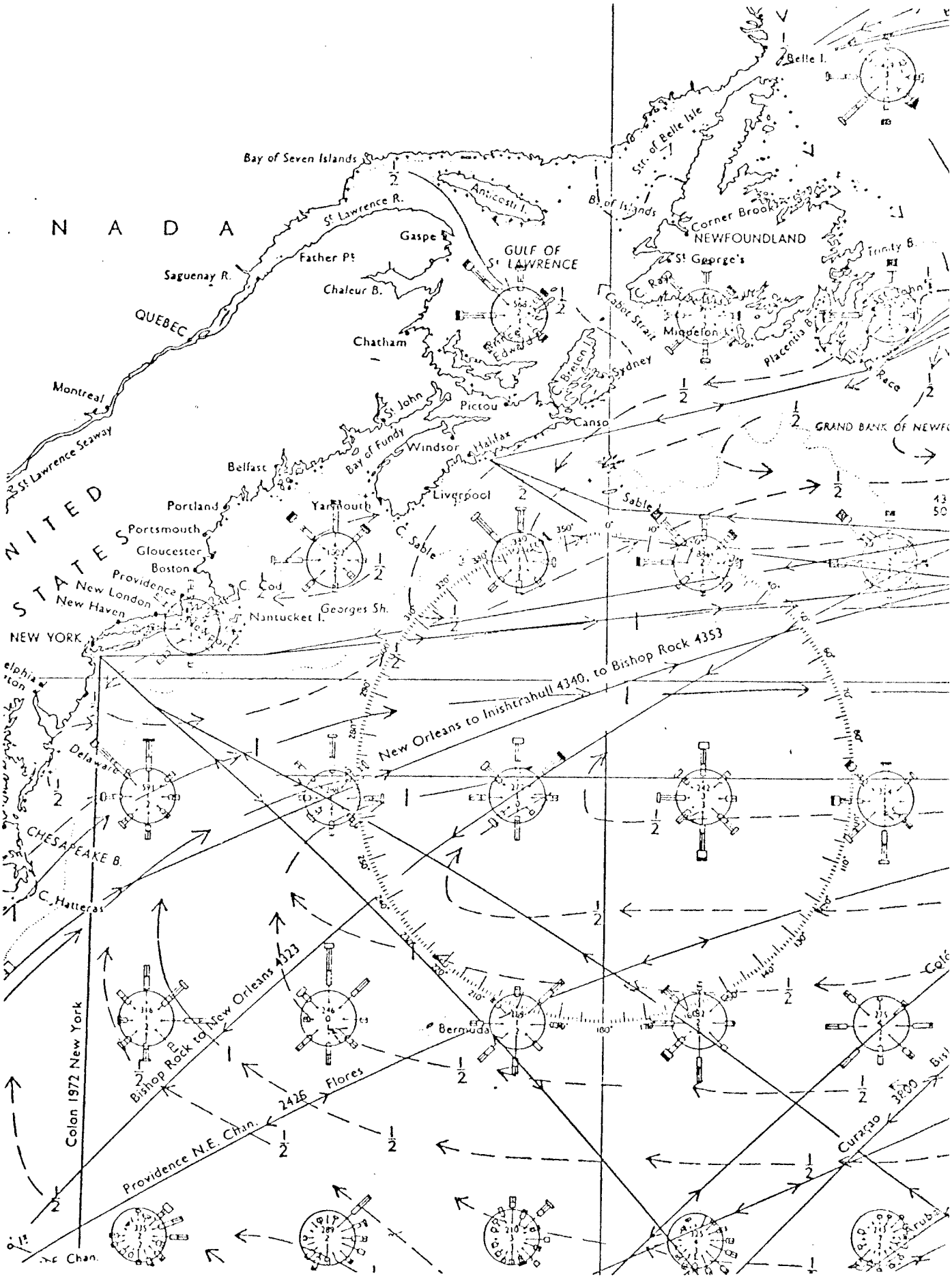


Figure 3. Pilot Chart North-Atlantic.

4. Weather routeing

4.1. Graphical method

In its infancy in the 1950's the tactics of weather routeing were only aimed at bypassing storm centres or tropical storms to reduce the damage to ship and cargo.

In 1957 R.W. James [5] introduced a graphic method to determine a "least-time-track" for an ocean passage. In many routeing offices this procedure is still practised. The ship's speed as function of the significant wave height for four different conditions of wave directions was used by James, as shown in figure 4.

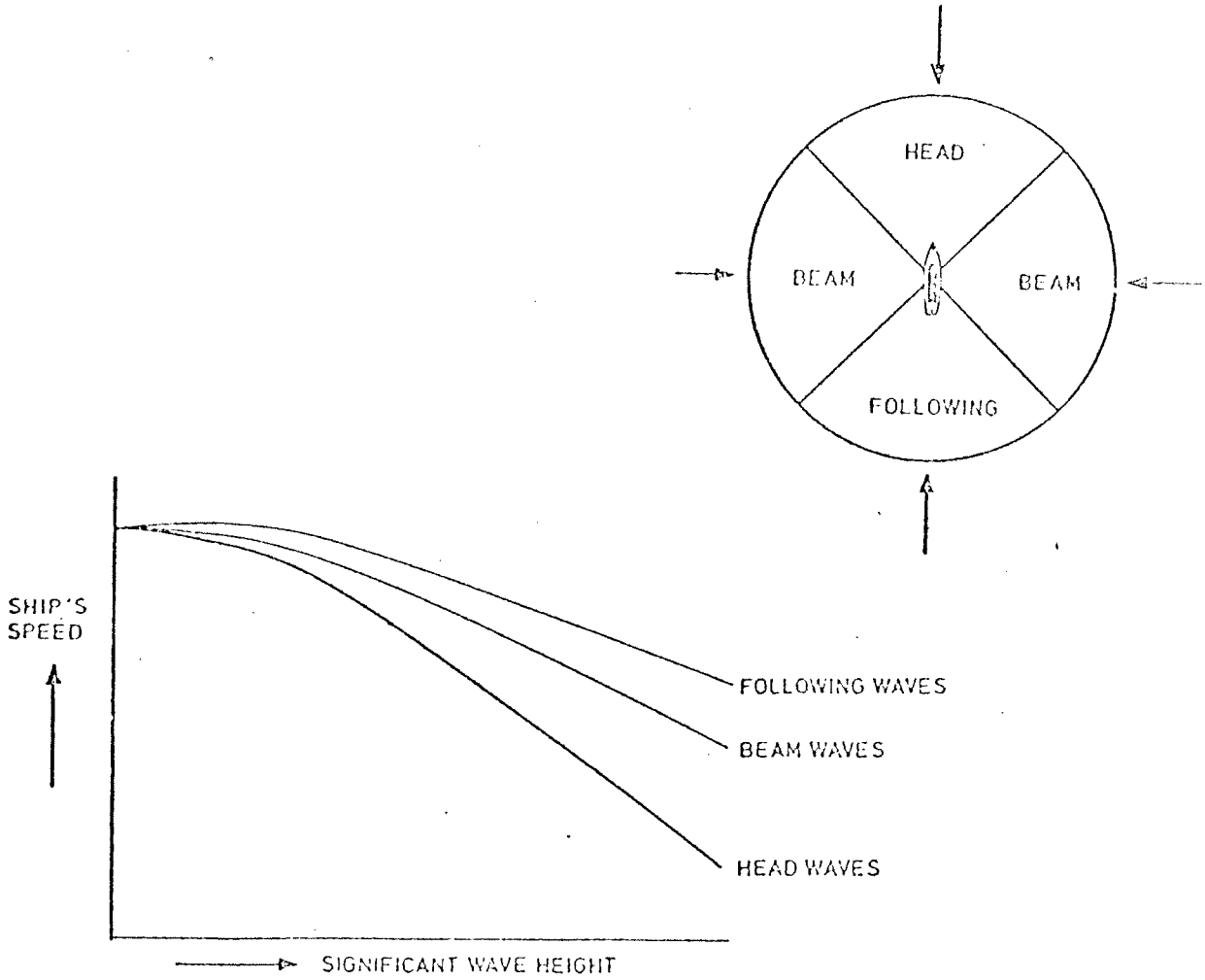


Figure 4. Ship's speed as function of seastate.

At the Royal Netherlands Meteorological Institute (KNMI) empirical data are used to plot the ship's speed in a polar diagram for different Beaufort wind forces and courses relative to wind and sea, see figure 5.

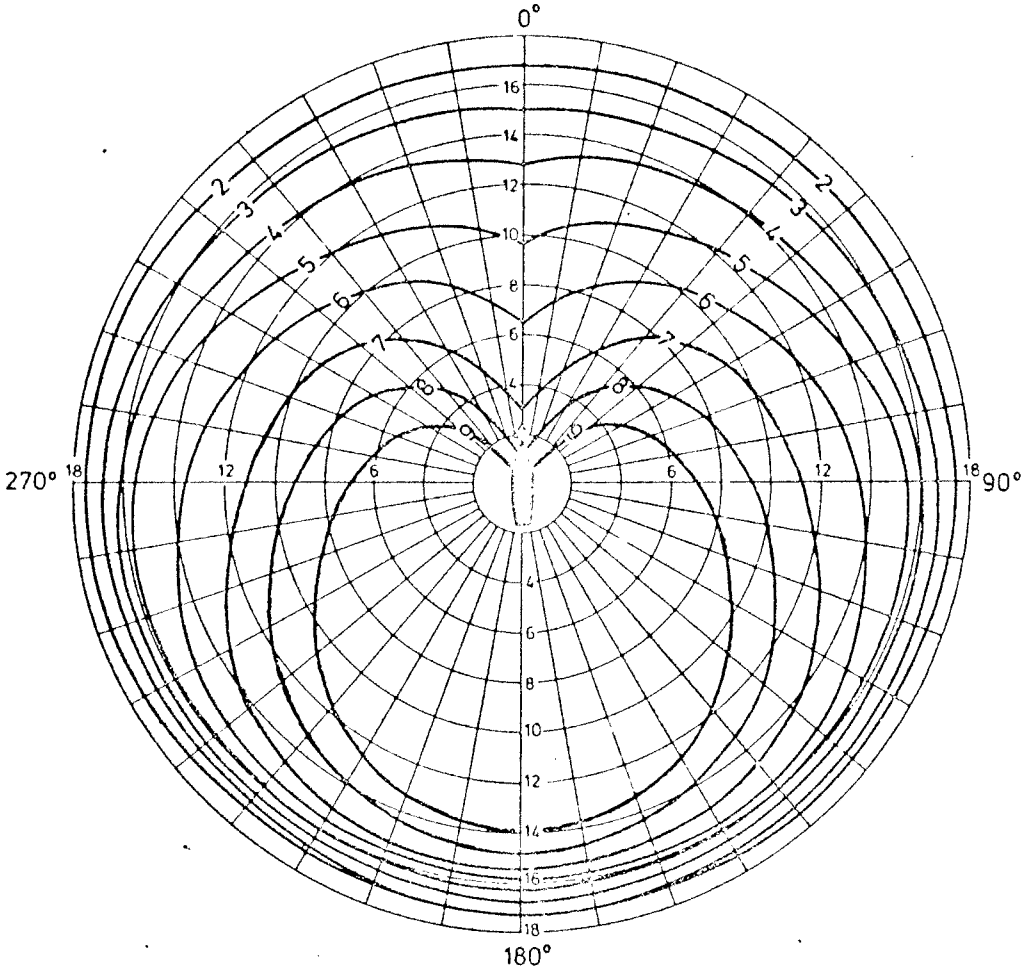


Figure 5. Polar speed diagram.

The influence of the significant period of swell and sea on the ship's speed is hardly taken into account at Routing Offices until now, although the fuel consumption in tons per mile is significantly higher in a specific interval of wave periods for a particular vessel. Figure 6 shows this for a 23000 tons containervessel according to Journee [11].

Before the computer took over, wave charts were hand-drawn using the information from the weather charts. The diagram of figure 7 is used by KNMI for this purpose, giving the significant wave height and period as function of wind speed, wind duration and wind fetch (= distance to windward where the wind is building up the waves).

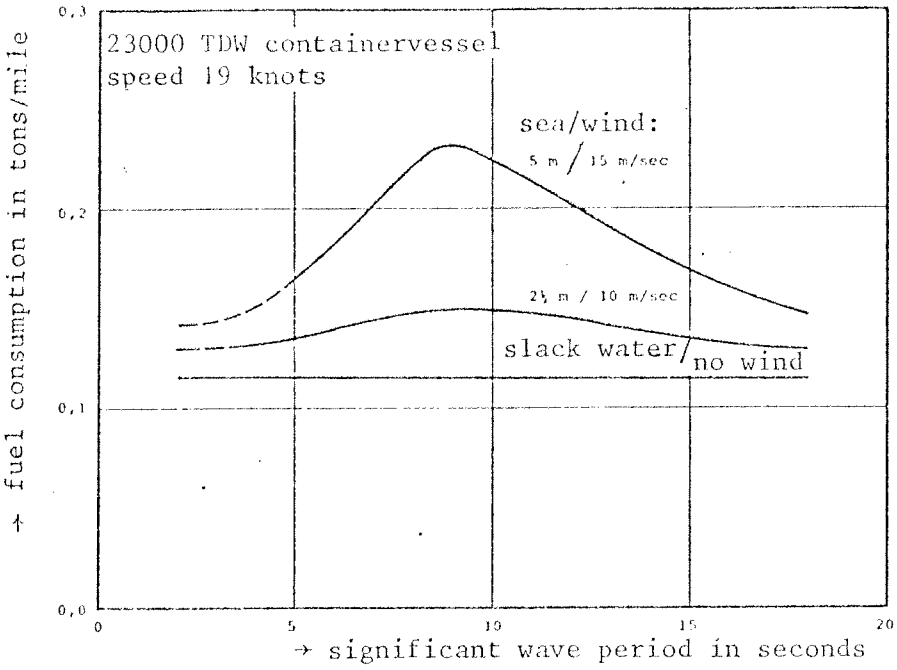


Figure 6.

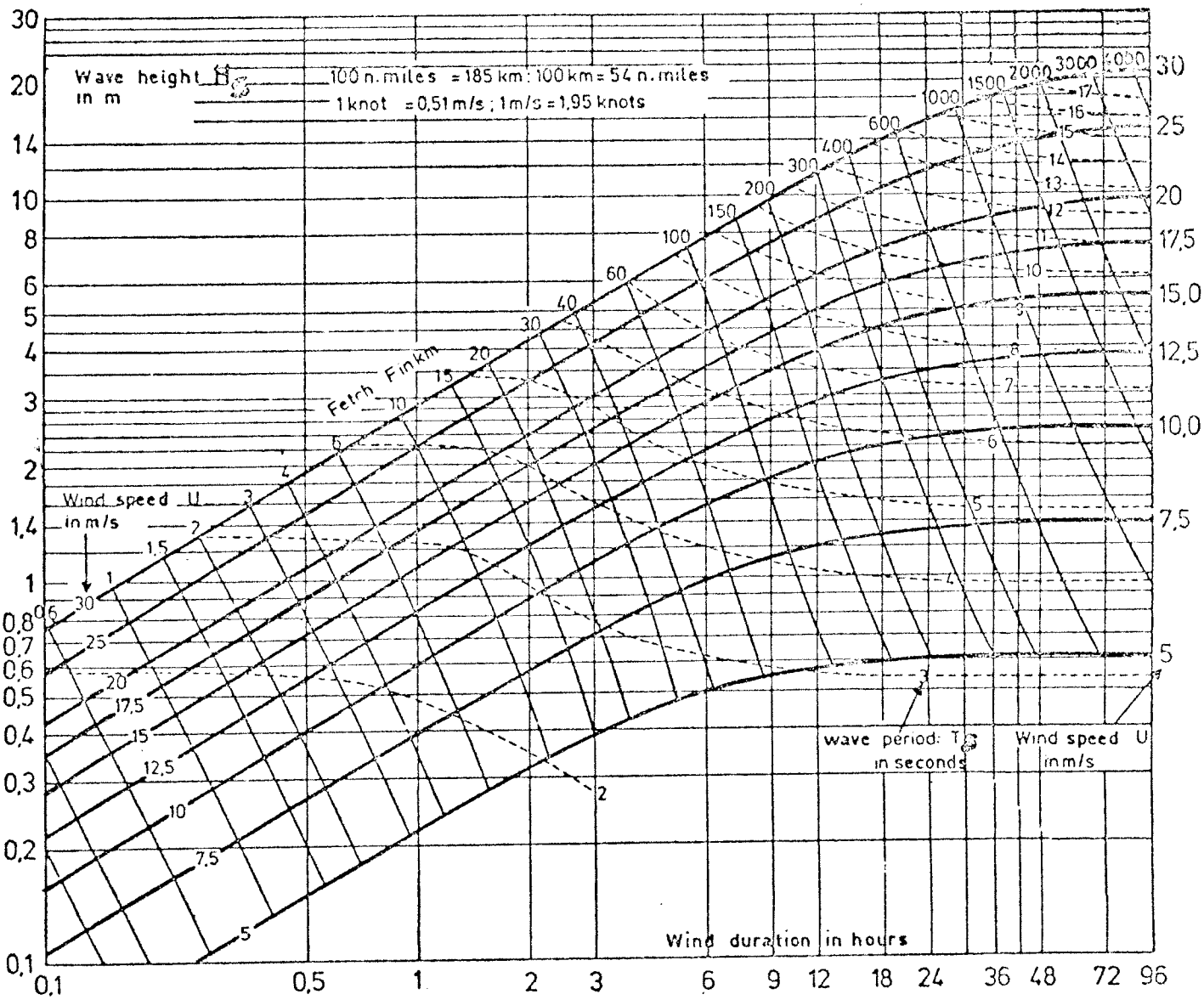


Figure 7. Significant wave height and period as function of wind speed, wind duration and wind fetch.

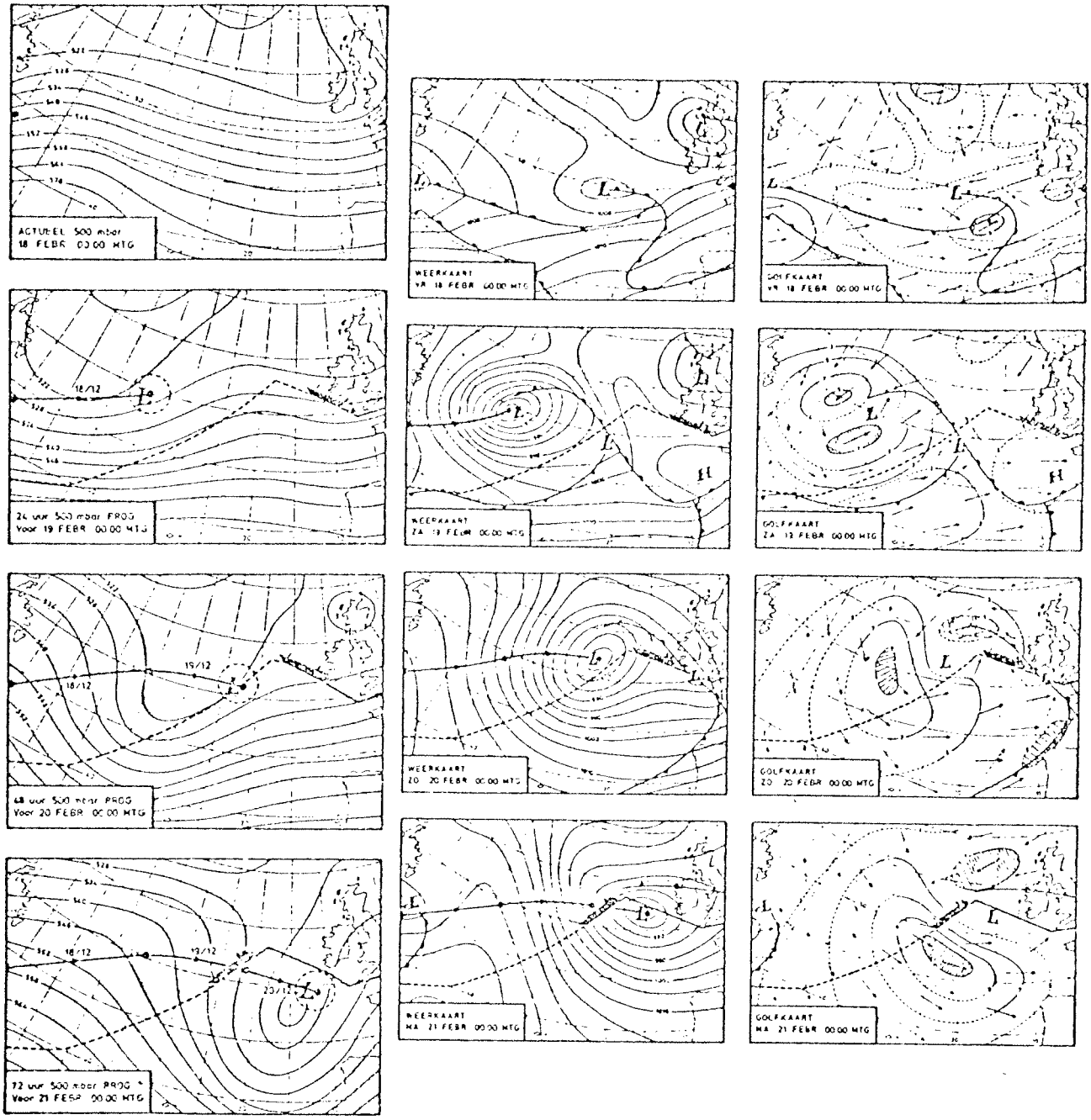


Figure 8. Actual and predicted 500 mb charts, actual weather and wave charts.

Nowadays, the enormous amounts of meteo data are processed by computers to produce the actual and predicted weather and wave charts. European Meteorological offices use a 6-day weather prediction for the North Atlantic mainly based on the data from the "European Centre for Medium Range Weatherforecasts" in Reading U.K. The wave charts are however manually corrected, on the basis of the wave information from weather reports from individual ships, which reports are transmitted from selected ships with 0600 hours GMT interval.

For the prediction of the track to be followed by a depression the 500 mb charts are used. The direction of a line of equal altitude of the 500 mb pressure level is roughly followed by the centre of a depression. In figure 8 the actual and predicted 500 mb charts, the weather chart and the wave chart for the actual situation on 18 February and for the situations 24 h, 48 h and 72 h thereafter are shown. It can clearly be seen in this figure that the area with the highest waves does not coincide with the centre of lowest pressure, but lies considerably south-easterly to southerly off this centre. The charts in figure 8 from the Netherlands Meteorological office show how a particular vessel was routed outside the area of highest waves.

The graphic method of determining the least time track to the destination is shown in figure 9. From the point of departure the distance covered in 12 hours is plotted in a number of directions. The envelope of these vectors is the 12 h isochone. The distance covered in 12 hours is determined, taking into account the speedloss in waves, see figure 4 and 5, and the wave chart as given in figure 8. From a number of points of the

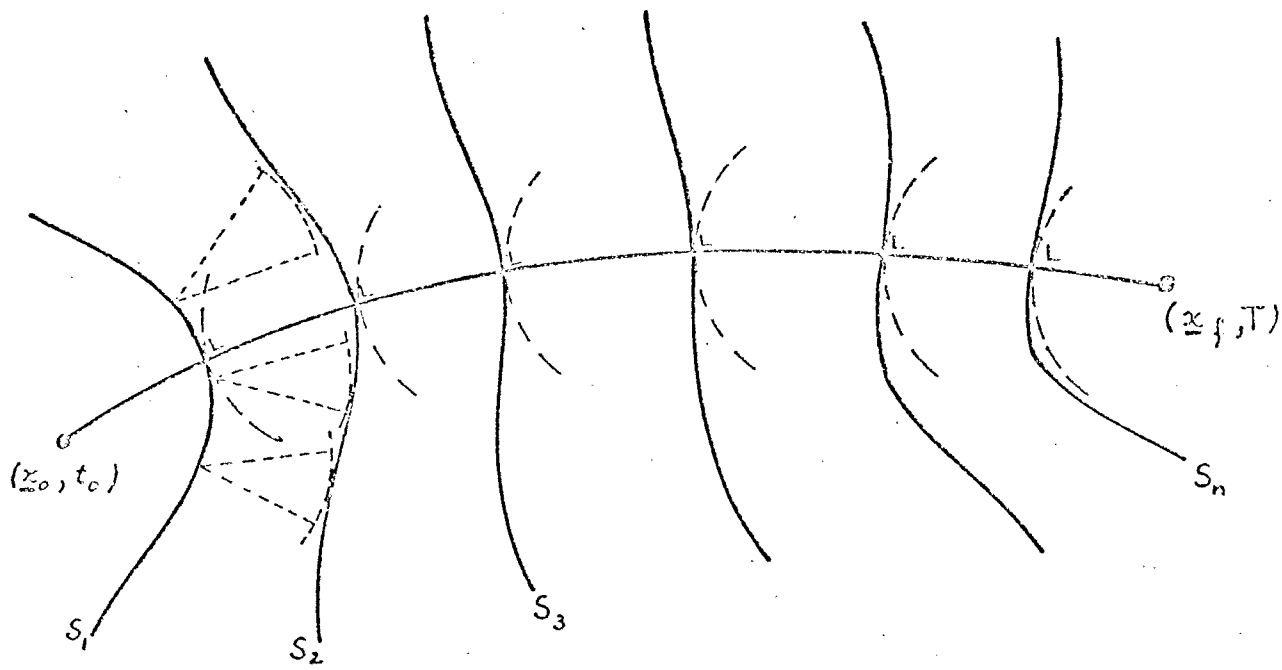


Figure 9. Least-time-track construction.

12 h isochrone a "local" 24 h isochrone is determined; the envelope of these local isochrones is the overall 24 h isochrone. As far as the weather prediction extends this procedure is continued as shown in figure 9.

From the destination backwards a line is drawn perpendicularly to all isochrones. This line is taken as the "least time track". Theoretically this is not correct; in each point of the least time track the projection of the distance travelled per unit of time on the perpendicular of the isochrone should be maximal, as shown in section 4.3. In practice, however, this manual method has proved to be very useful. Areas with high waves - especially head seas - are avoided because of the inherent speed loss. The consequent advantage is, that less damage to ship and (deck) cargo will be experienced, so safety and economy benefit simultaneously.

4.2. Practical results

Upon completion of a routed voyage, routeing offices evaluate the recommendations given. With the experienced weather an evaluation route is determined in the same way as described in section 4.1.

In the follow-up report which is also forwarded to the client,the time profits/losses of the track followed in regard to the great circle track and the loxodromic track are given and also the time loss of the track followed in regard to the "evaluation track" is provided.

In figure 10 the evaluation chart is shown for a particular case. The track actually followed shows to be very irregular. The actual least-time track is not shown in the figure, but lies somewhat more to the south. The time profit in this case of a 14 day voyage was about 2 days compared to the great circle track and 3 days compared to the loxodromic track.

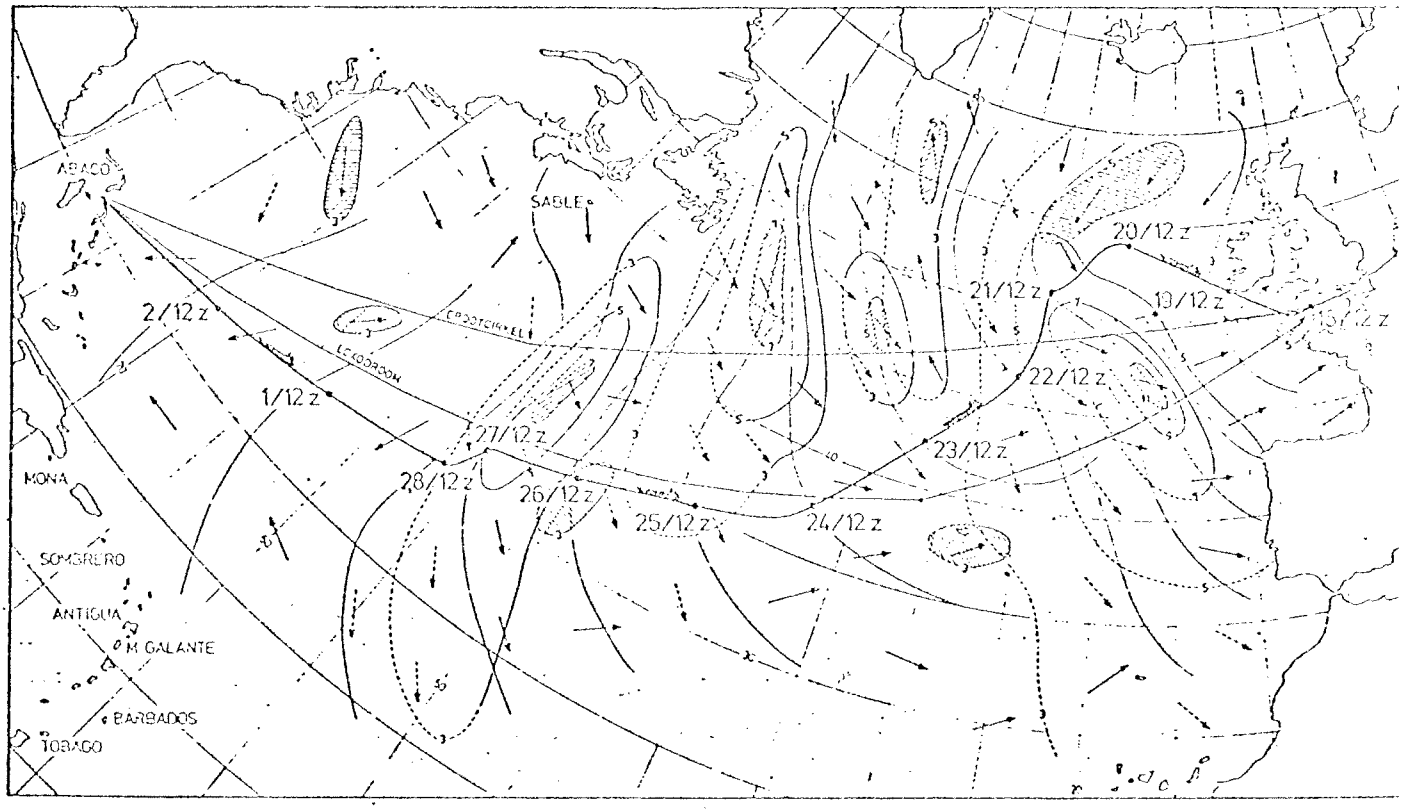


Figure 10. Evaluation chart.

Average profits are given in reports of routeing offices like [6], [7], [8] and [9].

In [9] the average results of the Dutch Meteorological office of 1971-1981 are published. The average time profit is different for 15 several geographical routes and varies between 1 hour and 10 hours.

For people who are suspicious of routeing offices publishing their own results, reference [8] is recommended which states that the results published by KNMI are even somewhat conservative.

In [4] a Shell spokesman confirms an average time profit of 4 hours on an Atlantic crossing, which equals about 10 tons of fuel or about $5000.- per transit.

The price of a routeing advice for the North Atlantic varies; in the Netherlands the charge is about $200.- plus telegram expenses. In view of the increased fuel costs, more ship operators are using the services of routeing offices.

ENROUTE TIMES PER SELECTED ROUTES
DECEMBER 1979

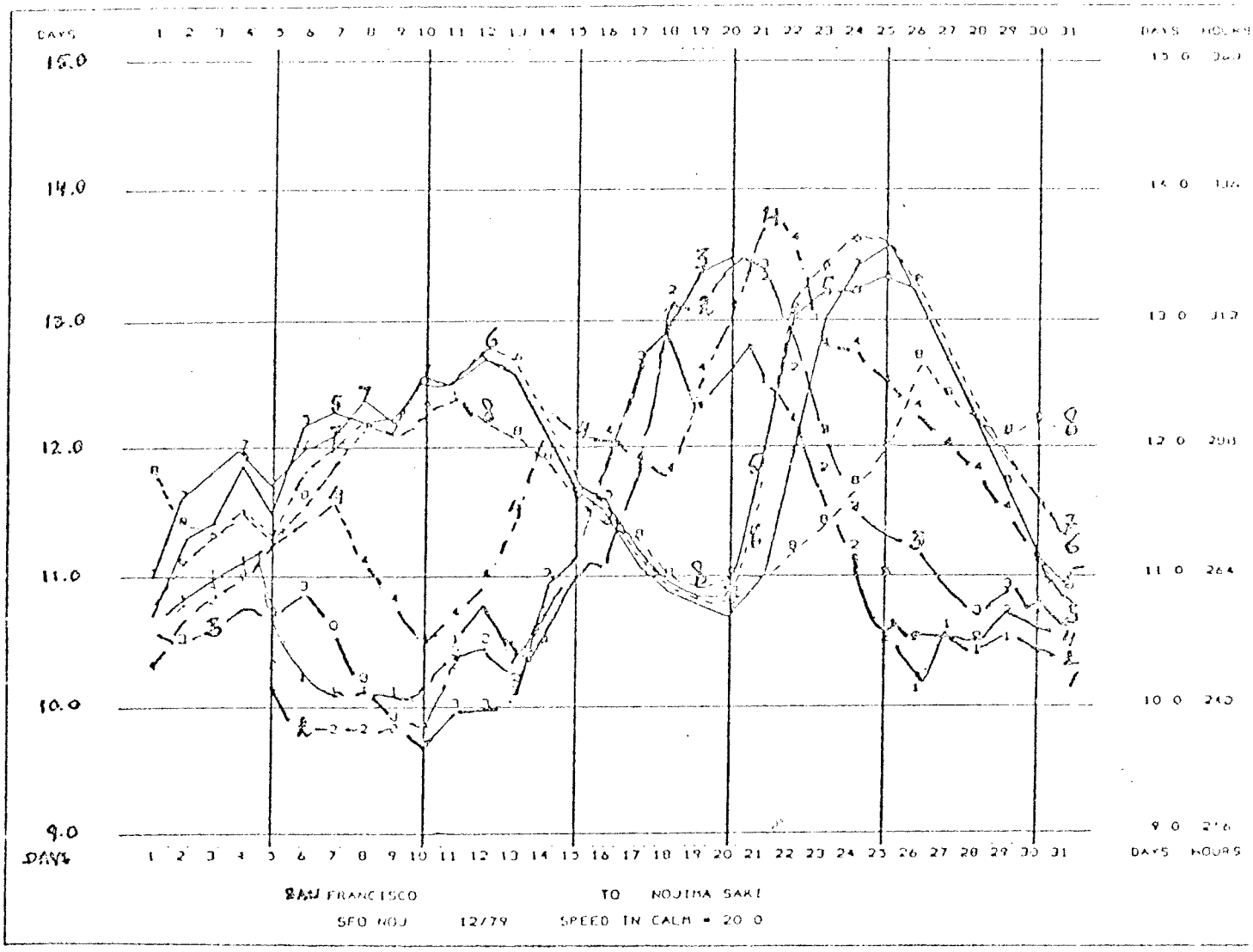


Figure 11A. Routeing results December 1979.

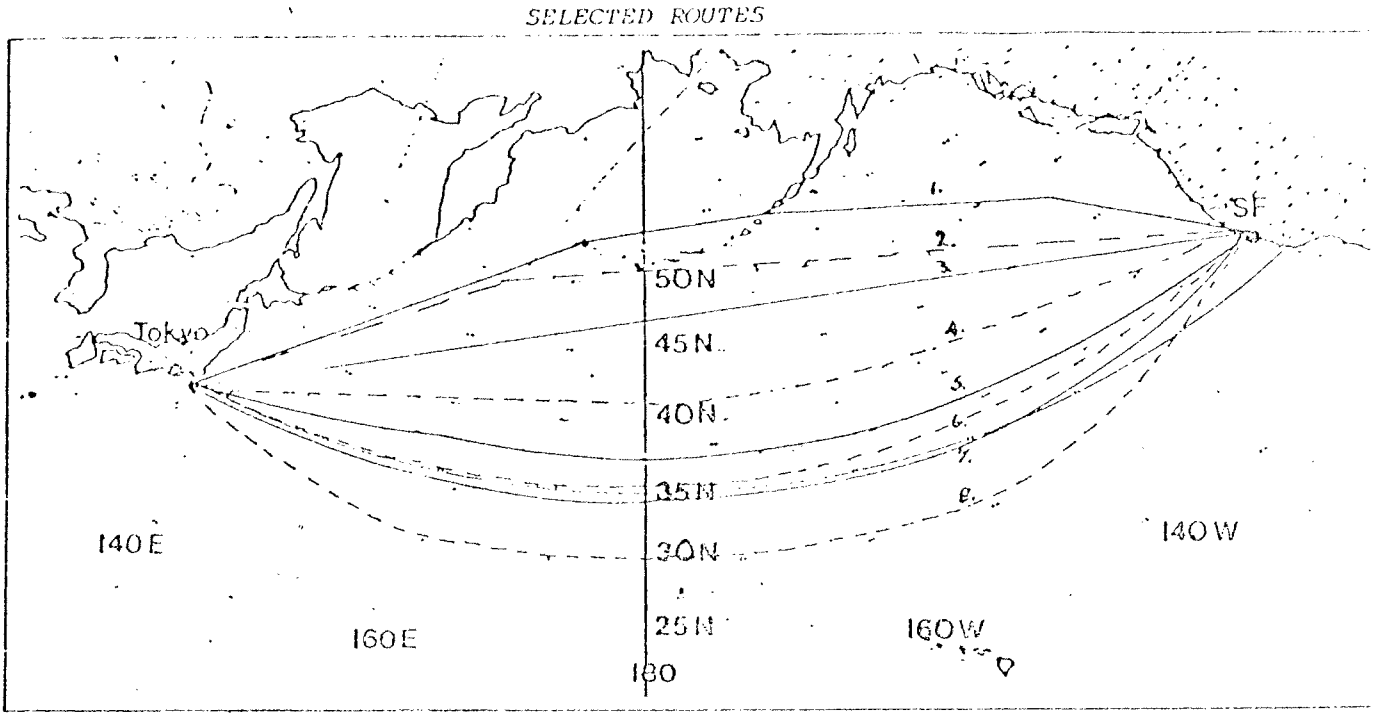


Figure 11B.

In reference [10] an interesting case study is mentioned. Oceanroutes Inc. routeing office simulated the passage of the north Pacific from San Francisco to Nojima Saki for the month of December for several years, where 8 different tracks were compared, see figure 11 where the results for 1979 are shown. For each day departure from San Francisco the enroute time is plotted vertically for each of the 8 tracks. Differences of about two and a half days passage time exist on the 10th December and 21st of December, but the choice of the route is reversed in both cases! This experiment clearly puts a query against routeing by Pilot Charts based solely on climatological considerations.

Apart from time/fuel savings, damage to ship and cargo is diminished by avoiding high (head)seas. In 1976 the Naval Weather Service (USA) contracted for a study covering 6 years of Atlantic experience in heavy weather and its damage effects. Figure 12 summarizes the results. An average ship will experience $1000.- per day damage to ship and cargo when proceeding in a significant wave height of 6 m, whereas this amounts to $10,000.- per day for waves of 8 m height; a dramatic increase!

An experiment conducted by Oceanroutes Inc. and mentioned in reference [10], was the analysis of the transmitted weather reports of January 1977 of 660 ships enroute in a trans-Pacific voyage of which 39% were routed by a routing office. The figure 13 shows the result, which is at first glance not very dramatic. However the sting is in the tail of the distribution, where it can be seen that about twice as many unrouted ships encounter significant wave heights of more than 6.5 m than routed ships. The same graph is made for ships in head seas in figure 14 where the distinction between routed and unrouted ships is even clearer. On an average the sea height was reduced by 14 per cent.

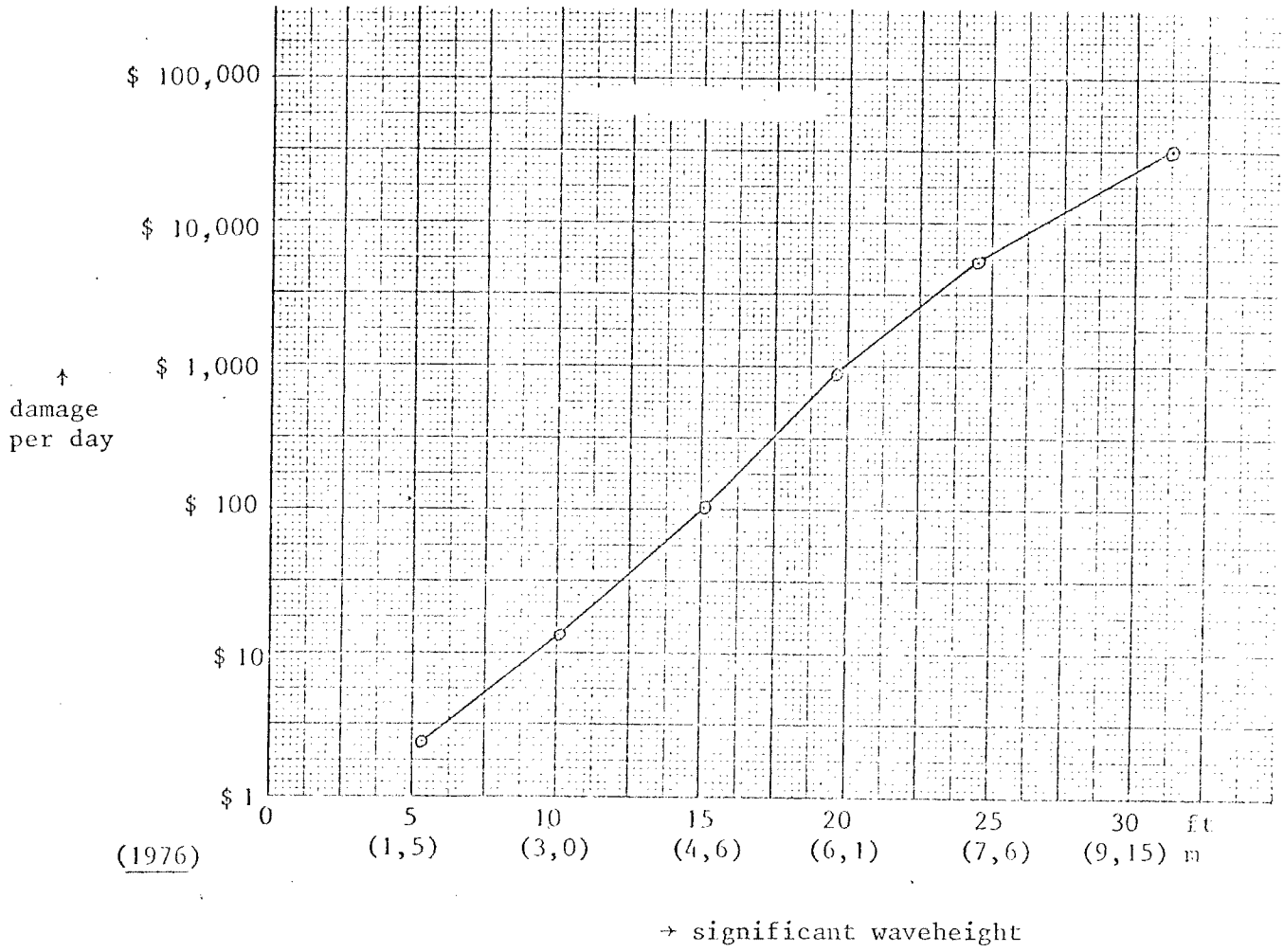

$ 100,000
$ 10,000
$ 1,000
$ 100
$ 10
$ 1
↑
damage
per day
0
5
10
15
20
25
30
ft
(1,5)
(3,0)
(4,6)
(6,1)
(7,6)
(9,15)
m
(1976)
→ significant waveheight

Figure 12.

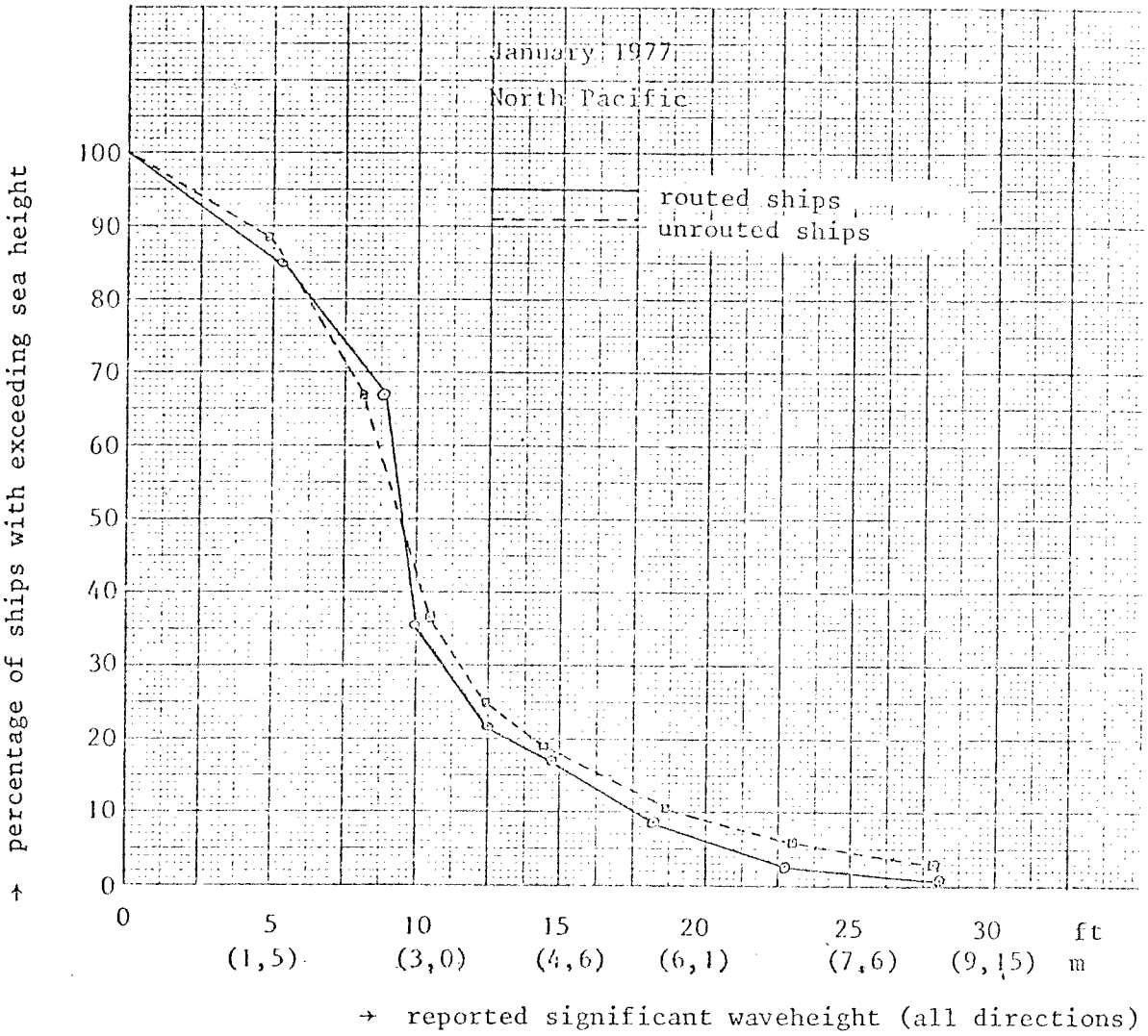

January 1977
North Pacific
routed ships
unrouted ships
100
90
80
70
60
50
40
30
20
10
0
→ percentage of ships with exceeding sea height
0
5
(1,5)
10
(3,0)
15
(4,6)
20
(6,1)
25
(7,6)
30
(9,15)
ft
m
→ reported significant waveheight (all directions)

Figure 13.

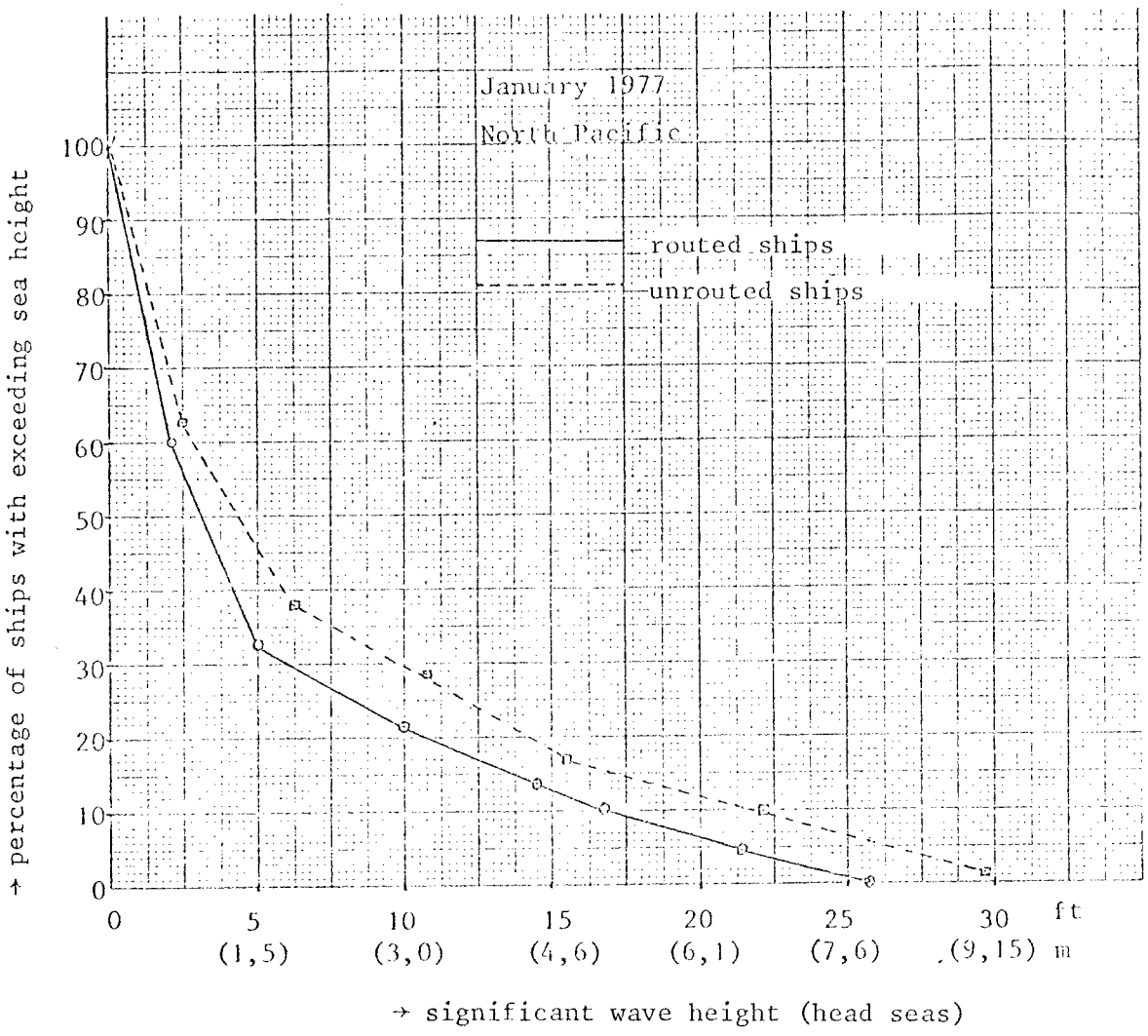

January 1977
North Pacific
routed ships
unrouted ships
→ percentage of ships with exceeding sea height
100
90
80
70
60
50
40
30
20
10
0
0
5
(1,5)
10
(3,0)
15
(4,6)
20
(6,1)
25
(7,6)
30
(9,15)
ft
m
→ significant wave height (head seas)

Figure 14.

Another noteworthy result is mentioned in [10]. A Japanese automobile shipper found that after using routeing recommendations for trans-Pacific voyages from a routeing office, the average damage per automobile shipped, dropped dramatically from \$2.95 to \$0.85 which is a considerable profit in view of the 350,000 vehicles exported annually from Japan to the USA.

4.3. Computer methods

First of all the predicted weather data are available in digital form, as the predicted weather charts are made by the computer.

The second major input for the computer program is the ship's speed in sea conditions which can generally be described for a specific vessel with a given draught, trim and hull condition as

$$\vec{V} = \vec{f}\ (n,\ \psi,\ H_s,\ T_s,\ \Theta_s,\ H_{sw},\ T_{sw},\ \Theta_{sw},\ V_w,\ \Theta_w) + \vec{S} \qquad (1)$$

where

n	= propeller revolutions per minute
ψ	= vessel heading
H_s	= significant wave height of sea
T_s	= significant period of sea
Θ_s	= average course of seawaves
H_{sw}, T_{sw}, Θ_{sw}	= parameters for swell
V_w	= wind speed
Θ_w	= wind direction
$\vec{S}$	= current vector
$\vec{V}$	= speed over the ground
$\vec{V}'$	= speed through the water

When meteorological and oceanographical data are known as function of the position and n is constant, formula (1) can be reduced to

$$\vec{V} = \vec{f}\ (t,\ \varphi,\ \lambda,\ \psi)$$

where φ = latitude and λ = longitude.

For a given situation $\vec{V}\ (\psi)$ is given by a speed indicatrix as shown in figure 15. The speed indicatrix in figure 15 at right is concave in a given sector due to an obligatory speed reduction when proceeding in waves entering from abaft the beam. By "symmetric cruising" a higher average speed could be attained. In a study by J.M.J. Journee of Delft University of Technology [11] sponsored by the Ministry of Economic Affairs as part of a general "energy saving" project, mathematical models are given for the ship's speed and the fuel consumption taking into account the relevant factors.

To develop a computer algorithm, an optimisation criterion has to be defined. In most programs the transit time is minimized [12], [13], [14] by computing $\psi\ (\varphi,\ \lambda,\ t)$ for the voyage, keeping n constant. Only in high seas n will be reduced to avoid damage.

Basically two methods are in use:

(i) The method based on the "maximum principle" of Pontryagin [15]

A state vector $x\ (t) \in R^n$ and a control vector $u\ (t) \in R^m$ are defined. The differential equation for x (t) is given by

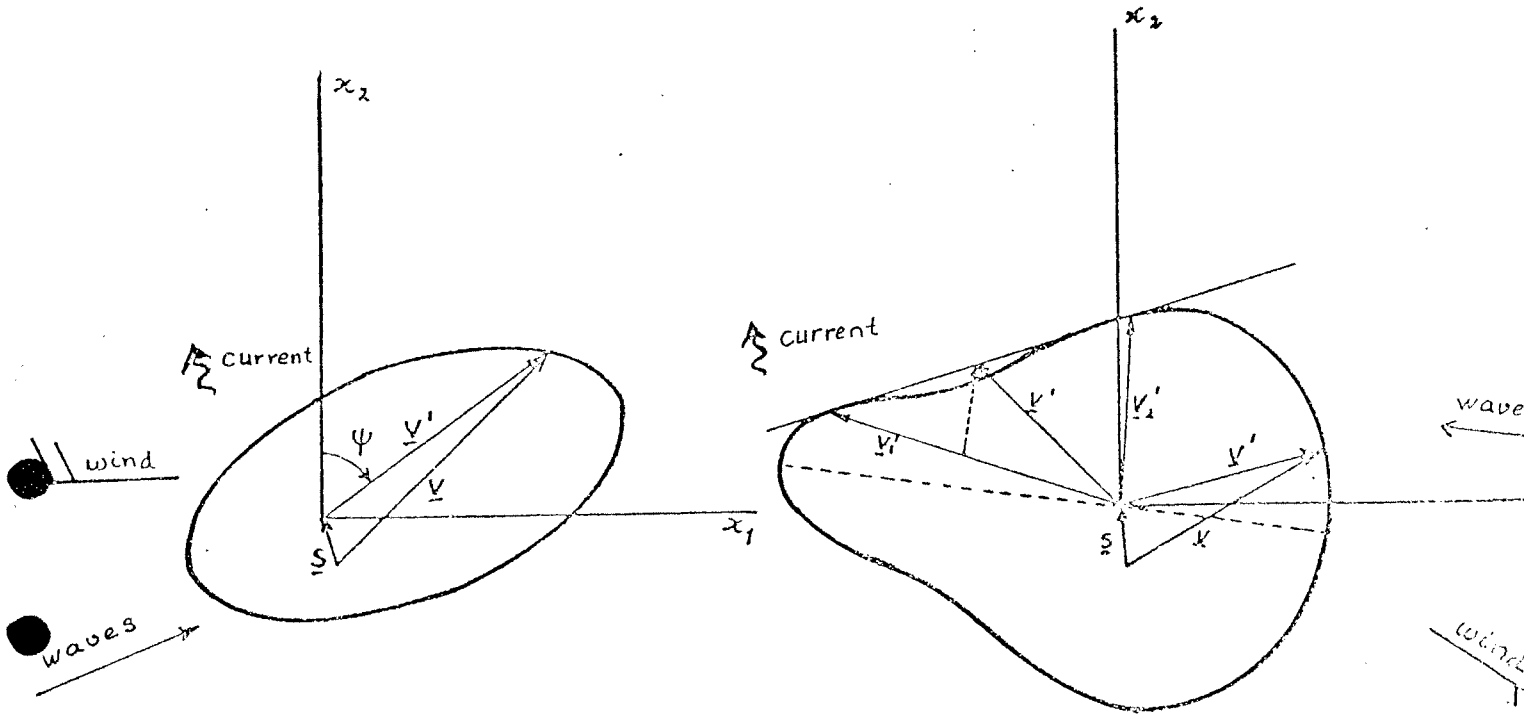


Figure 15. Speed indicatrix.

$$\dot{x}\ (t) = f\ (x\ (t),\ u\ (t),\ t)$$

$$x\ (0) = x_0 \qquad (3)$$

$$x\ (T) = x_1$$

find x (t) and u (t) to minimize the object functional

$$J = \int_0^T L\ (x,\ u,\ t)\ dt \qquad (4)$$

An adjunctive vector p is introduced and when L (·) and f (·) are twice continuously differentiable for x, u and t, the problem is solved by solving the simultaneous equations

$$\dot{x} = f\ (x,\ u,\ t)$$

$$\dot{p}^t = L_x - p^t\ f_x \qquad (5)$$

$$L_u - p^t\ f_u = 0$$

When the Hamilton function is introduced

$$H = -L + p^t\ f \qquad (6)$$

the solution reads

$$\left.\begin{aligned} \dot{x} &= H_p \\ \dot{p} &= -H_x \\ H_u &= 0 \end{aligned}\right\} \qquad (7)$$

In the case of minimum time routeing:

$$J = \int_0^T 1 \, dt$$

$$L = 1$$

$$H = -1 + p^t f$$

$$\dot{x} = f\ (x, \psi, t) \text{ (speed indicatrix)}$$

$$\dot{p}^t = -p^t f_x$$

$$p^t f_\psi = 0$$

$$x\ (0) = x_0$$

$$x\ (T) = x_1$$

$$H\ (x_1, T) = 0$$

As $J_x = p$, and the isochrones as mentioned in section 4.1 are contours of equal J, the vector p is the gradient vector of J and is perpendicular on a contour line.

According to Pontryagin $H = -1 + p^t f = -1 + p^t \dot{x}$ has to be maximized so the projection of the speed vector $\dot{x}$ on p has to be maximized, as used in section 4.1.

In figure 16 the timefronts for an Atlantic transit computed by the algorithm of Bijlsma [13] are shown.

When the rate of decrease of fuel is described by the equation

$$\dot{\chi} = f\ (t, \varphi, \lambda, v, \psi) \qquad (8)$$

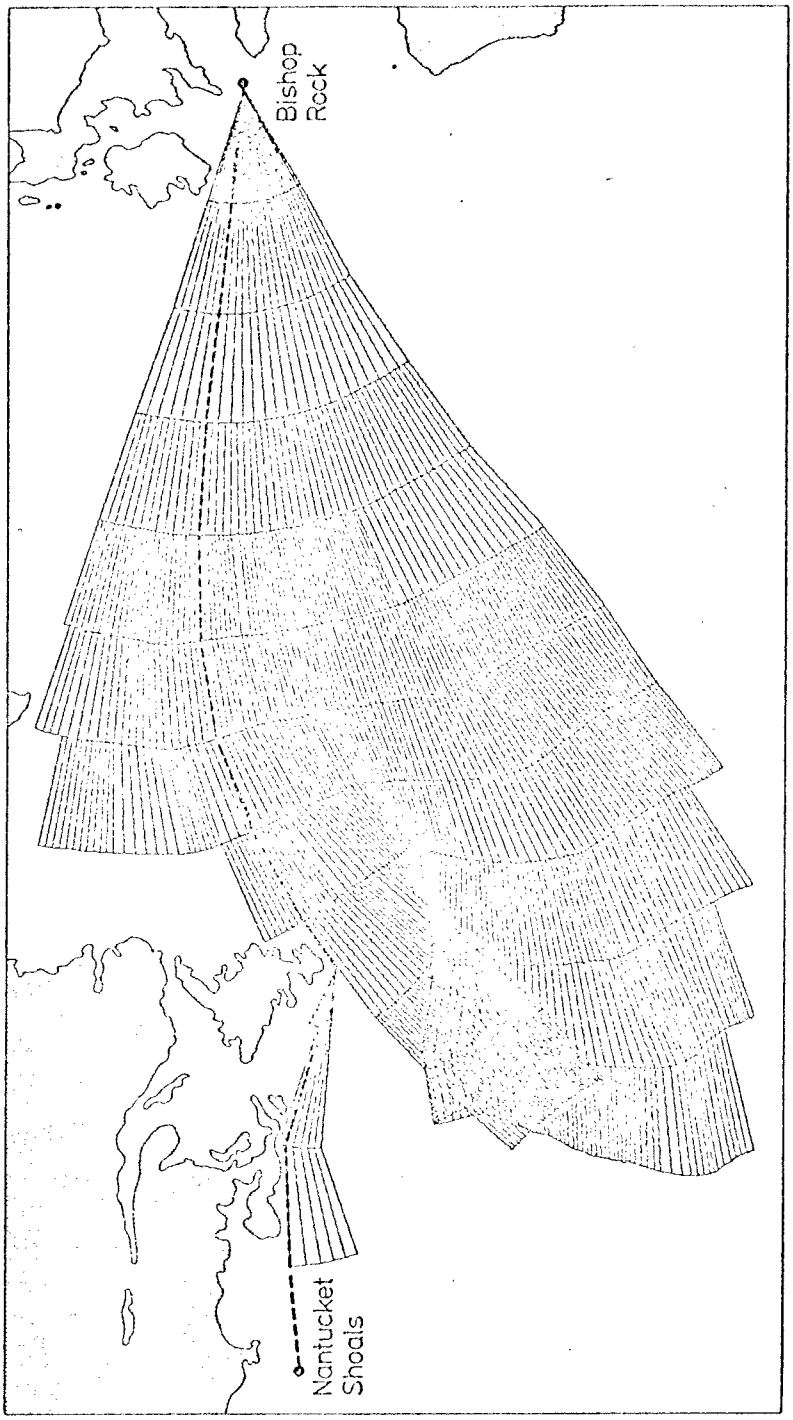
functions ψ (t) and v (t) have to be determined minimizing

$$J = \int_0^T f\ (t, \varphi, \lambda, v, \psi) \cdot dt \qquad (9)$$

In [13] this is pointed out; it is a subject of further research at KNMI.

(ii) The method of "forward dynamic programming"

In the ocean area under consideration a series of gridpoints is appointed equally distributed "along track" and "cross track" along the greatcircle route, see figure 17. From the starting point the enroute-time to the gridpoints on the first cross-track line is calculated with the expected weather and ship's parameters.



Computer produced by means of an incremental plotter using wave information over the period 17 January-23 January 1970, fictitious ship's data and a 12-hour time step. The least-time track is indicated by the dashed line.

Figure 16.

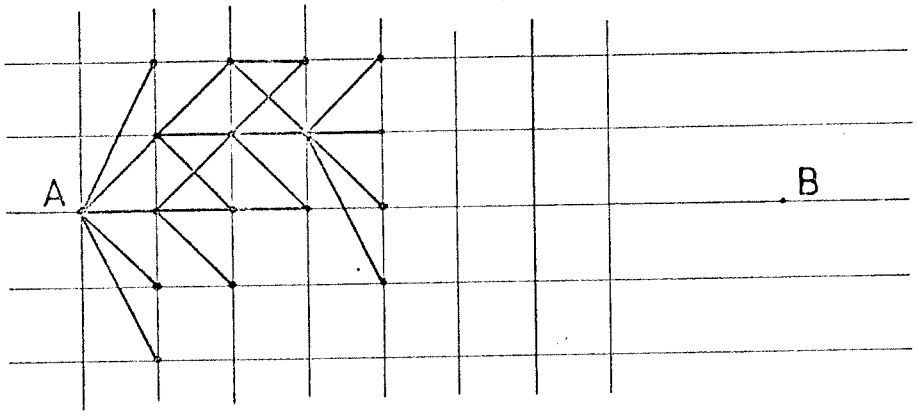


Figure 17. Forward dynamic programming.

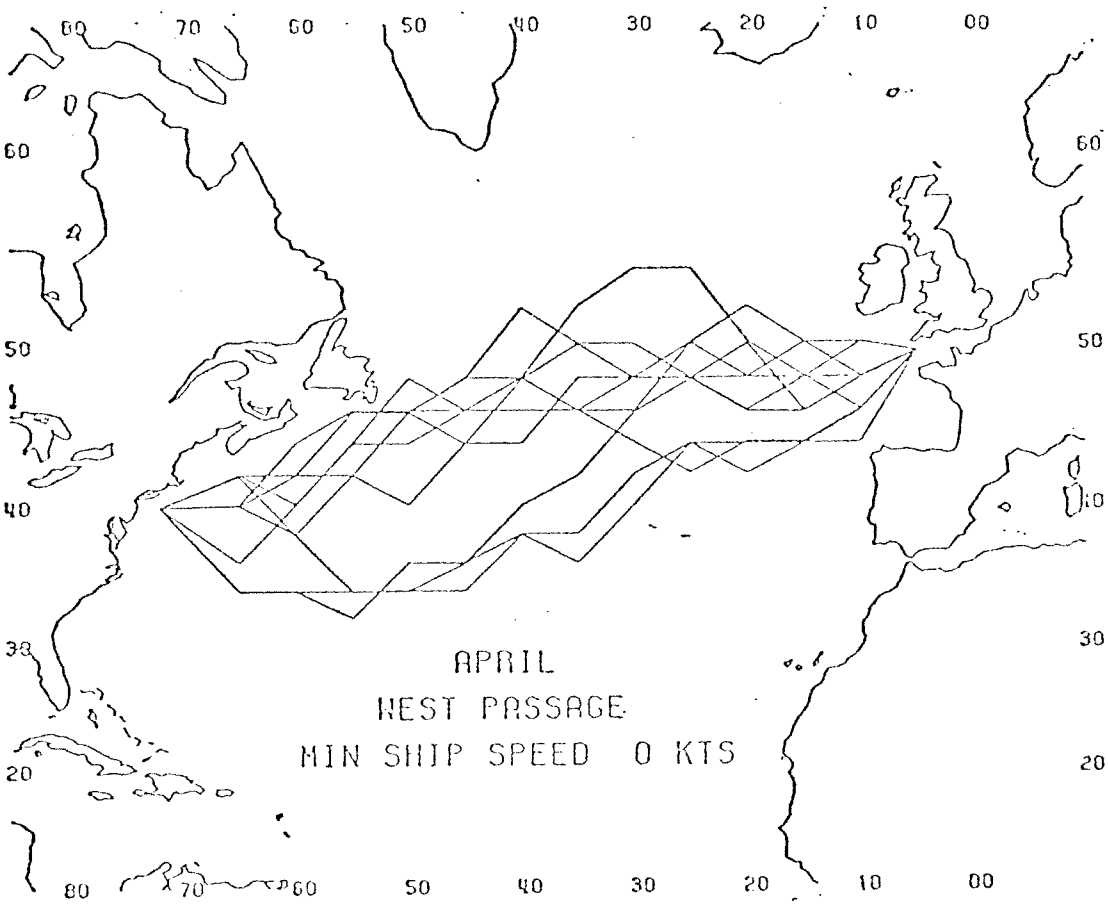


Figure 18.

From each gridpoint on the first line this same procedure is carried out giving different times of arrival at the second cross-track points for different routes. The earliest times of arrival and respective routes are stored, others are skipped. This procedure is continued until the destination is reached. The route belonging to the earliest arrival time is the "least-time-track". In figure 18 routes determined by this method are shown [16].

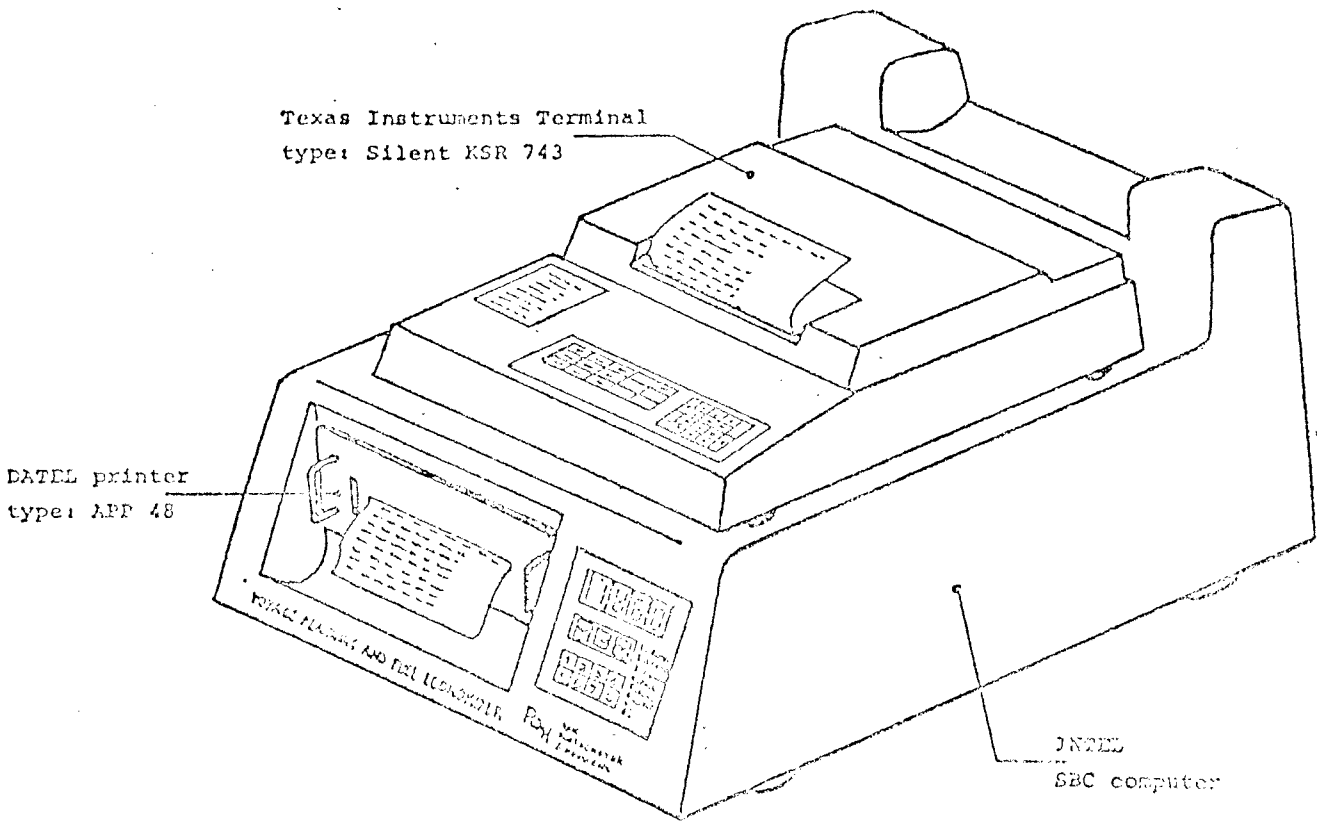
The second method is simpler, but a disadvantage is that interpolation in time for the meteo data is necessary to compute the passage times between the gridpoints, as meteorological data are available for the standard 12 h GMT intervals.

In [11] a system is described to present to the shipmaster continuous information about the performance of the ship, to face him directly with his fuel management. Average fuel consumption per hour, per mile and per kWh together with speed, power and rpm are printed after each selected period, for instance 20 minutes. Also is given information about the effect on the ship's performance by the degree of fouling of the ship's hull. An algorithm is developed which simulates the ship's performance depending on the ship's heading, the actual weather and the weather forecasts. This algorithm is used to determine the optimum speed to arrive just in time at the destination or to calculate the expected time of arrival at a fixed engine setting, rpm, etc. The total device including the computer was called "energy clock" and is commercially for sale as a PERSUS (Performance Surveillance System). The device is shown in figure 19.

Subjects of further research at routeing offices and research institutes are:

(i) development of better ship's performance data in waves, i.e. reliable and accurate speed indicatrices and thresholds for course and speed in extremely high waves
(ii) development of better object functions to minimize fuel and damage
(iii) increasing reliability of medium range weather forecast, including the energy distribution of the waves with respect to frequency and direction
(iv) routing of commercial sailing vessels.

As an example of the last subject, figure 20 is shown where an Atlantic crossing was simulated with real (winter) weather and with the speed indicatrix of a 17000 tons sailing vessel with a maximum sail area of 103,000 ft^2. The enroute time was 5 days, and the distance covered was 2034 n.m. Along the great circle track of 1800 n.m., 8 days would have been used.



COMMAND CODES
VOYAGE PLANNING

1 list positions
2 enter new positions
3 delete positions
4 insert positions
5 change data
6 calculate E.T.A
7 calculate R.P.M.
8 exit module

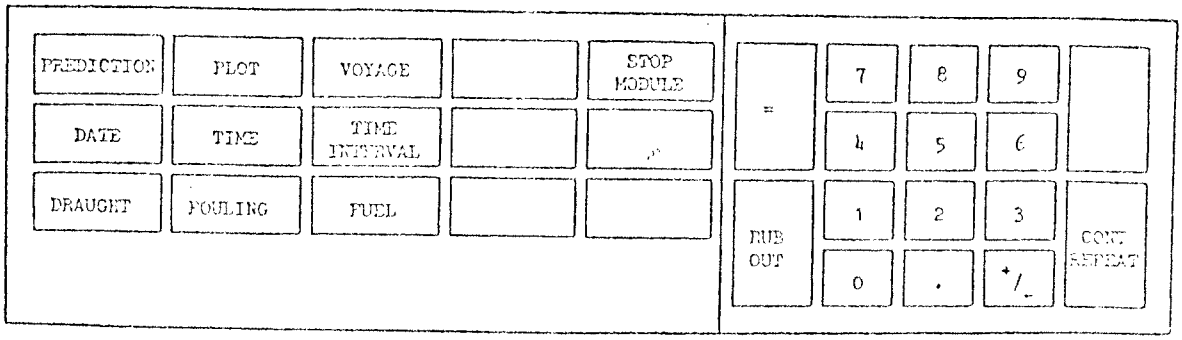


Figure 19.

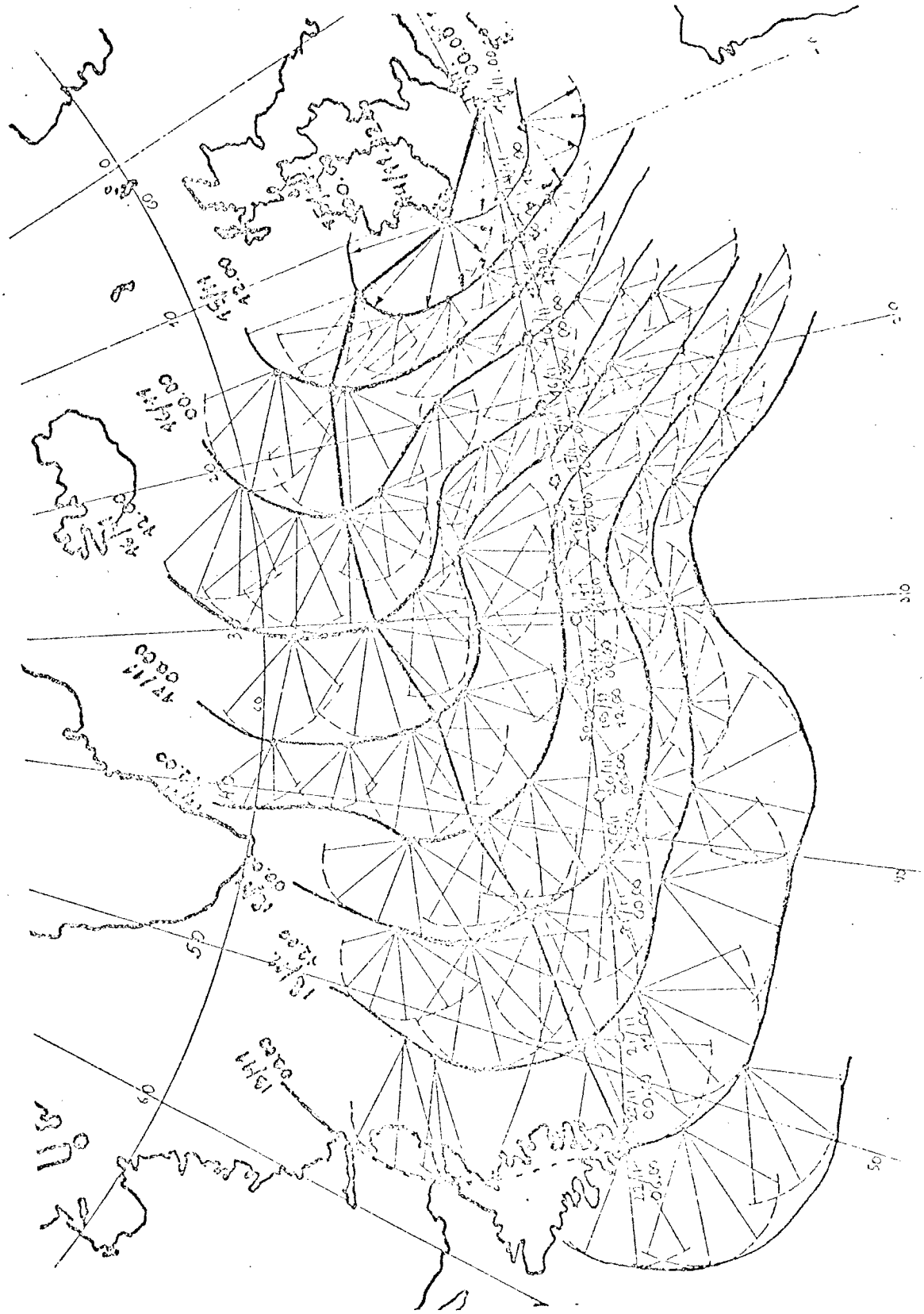


Figure 20. Least time track and great circle track of routed sailing vessel.

[1] Ernst Crone
Cornelis Douwes zijn leven en zijn werk
Tjeenk Willink, Haarlem, 1941

[2] American Practical Navigator
Originally by Nathaniel Bowditch
Defense Mapping Agency, USA, 1977

[3] Segelhandbuch für den Atlantischen Ozean
Deutsche Seewarte
L. Frederichsen & Co., Hamburg, 1910

[4] Dodging the weather
Quarterly journal of the American Bureau of Shipping
August 1982

[5] R.W. James
The present state of ships routing
Interocean '70 international conference

[6] G.C. Korevaar
Experiences and results of the ship routing of the Royal Netherlands Meteorological Institute
Scientific report 76-9
KNMI, De Bilt, 1976

[7] Results of Marine and Isthmian lines weather routed crossing 1961-1963
Weather Routing Incorporated, New York

[8] Report on the results of weather routing North Atlantic traffic by the Royal Netherlands Meteorological Institute
Shell International Marine Ltd.
MRP/22 July 1966

[9] D. Heyboer
KNMI Routering (Dutch)
NTT-De Zee, 1981, no. 11

[10] W.G. Constantine
Weather routing for safety and economy
Ship operation and safety conference
Southampton April 1981

[11] J.M.J. Journee
Fuel saving by surveillance and simulation of ship's performance (in Dutch)
NTT-De Zee, Novermber 1982

[12] C. de Wit
Optimal meteorological ship routing
Nederlands Scheepstudiecentrum TNO, report no. 142S, August 1970

[13] S.J. Bijlsma
On minimal-time ship routing
Royal Netherlands Meteorological Institute, report no. 94 (1975)

[14] G. Frankel, H. Chen (MIT)
Optimization of ship routing
Prepared for National Maritime Research Center
Kings Point, New York, 1978

[15] L.S. Pontrygin
Mathematical Theory of optimal processes
Interscience Publishers Inc.
New York, 1962

[16] James H. Mays
Sailing ship weather routing
Symposium on wind propulsion of commercial ships
London, 1980

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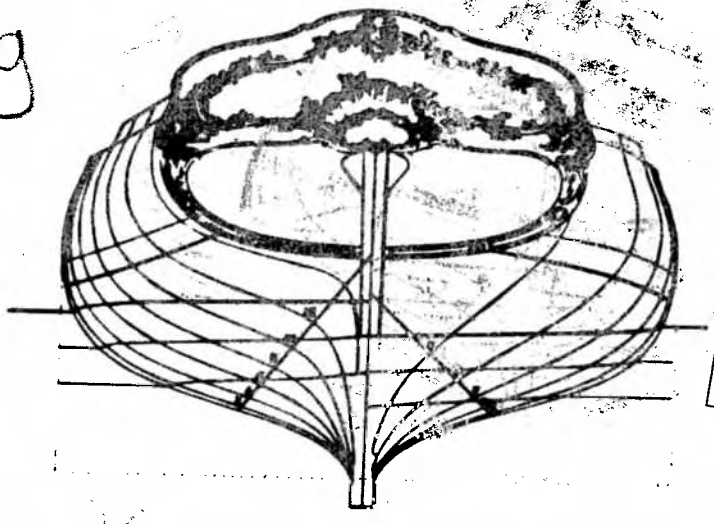

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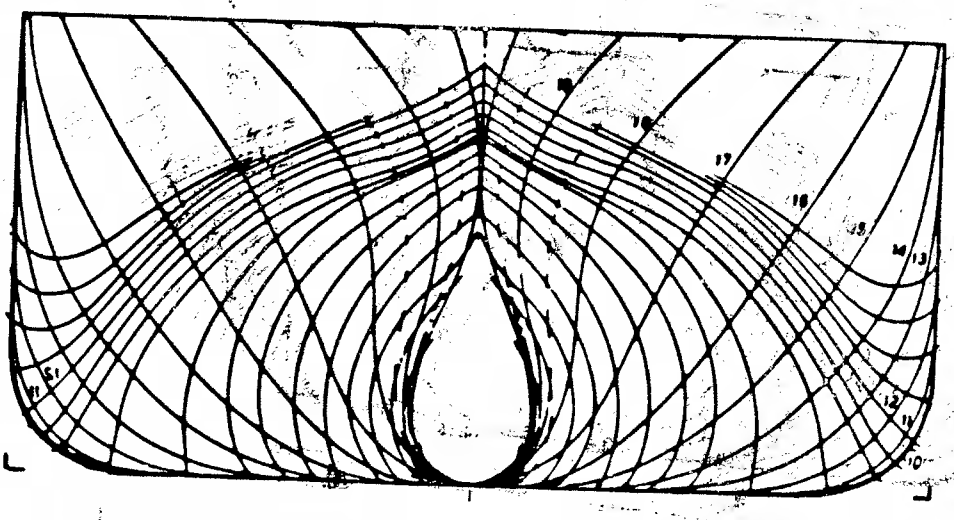
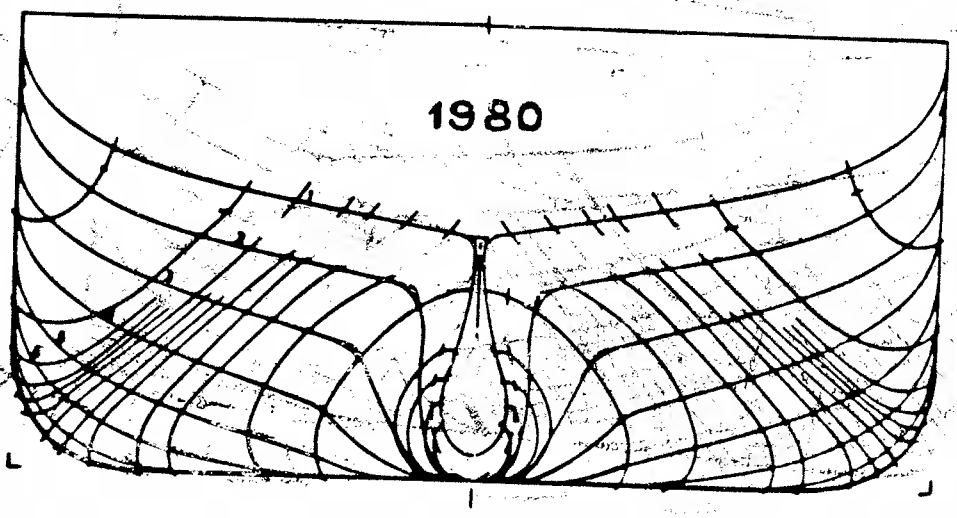
HULLFORM DEVELOPMENT



1980



HULLFORM DEVELOPMENT

WITH SPECIAL REGARD TO

PROPULSION
VIBRATION + NOISE PROBLEMS
FUEL CONSUMPTION
FULL BODY
FINER HULL
CONTAINER
RO-RO
STEERING

By

VOSSNACK	NEDLLOYD - R'DAM
BUYS	"
VRIEND	"
JONK	NSMB - WAGENINGEN
V.D. BEEK	"
STAPEL + WOOSSANEN	"
POLS	V/D GIESSEN DE N.
VOOGD	LIPS - DRUNEN
DE BRUYN + MOELKER	T.P.D. - DELFT
V.D. VELDEN + MAR. LELS +	V.D. MOLEN - H.T.S.
GROTHUS - SPORK	V.W.S. - BERLIN

DEC '83

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16 STEERING .....

17 CAVITATION-EROSION ON RUDDERPOST AND RUDDER

18 Literature. .....

PART III ABOUT STEERING.

Described are steering problems with twin screw ships having differences in arrangements :

- Twin screw, single balance rudder.
  Twin screw, single skeg rudder.
  Twin screw, twin rudder.
  Single screw, single rudder, open skeg, propeller on pipe.
  Triple screw, single rudder.
  Outward/inward turning propellers : swinging with stern.
  C.P. propellers.
  Bow- and stern thrusters.

FNB/E. Vossnack

23rd August 1983.

1 S
FULL BODY
SLOW SHIP

AFTERBODY HULL FORM VARIATIONS - INFLUENCE OF STREAMLINES.

- The influence of the fullness and curvature of the bilge around the engine aft-end (sections 2-3-4) on the flow lines and wake has clearly been ascertained.

The water particles running underneath the ship's bottom will be forced far outward by a "fat" gondola. Further aft, the particles separate from the hull surface and move towards the propeller in a downward motion at a much reduced velocity.

- This creates cavitation on the propeller blades, followed by pressure fluctuations on the hull surface above the propeller.
- Many ships show separation of flow just ahead of the upper section of the propeller ! (Ref. model 5141 D)
- Many single-screw vessels show a breaking wave at this point and have rattling afterbodies !

- It is at the very first moment when the building contract is being signed - and where length and deadweight then are fixed - that the baby is already doomed to vibrate all her life.

- Short length / beamy vessels, engine far aft, a fat afterbody and a narrow propeller aperture are causing this evil.

- The shipyard tries to offer the maximum carrying capacity at the lowest price and lowest fuel consumption which is quite understandable of course.

- Improvement of the hull form at a later stage by the Model Basin and adoption of wake-adapted (skew-back) propellers can hardly soften this evil if main dimensions stay in the wrong combination.

Wrong dimensions are to be prevented at the contract stage ! (I.e. FAR before tanktesting normally could be started.) Therefore:- tanktesting should be carried out b e f o r e signing contract!

REMARK 1:

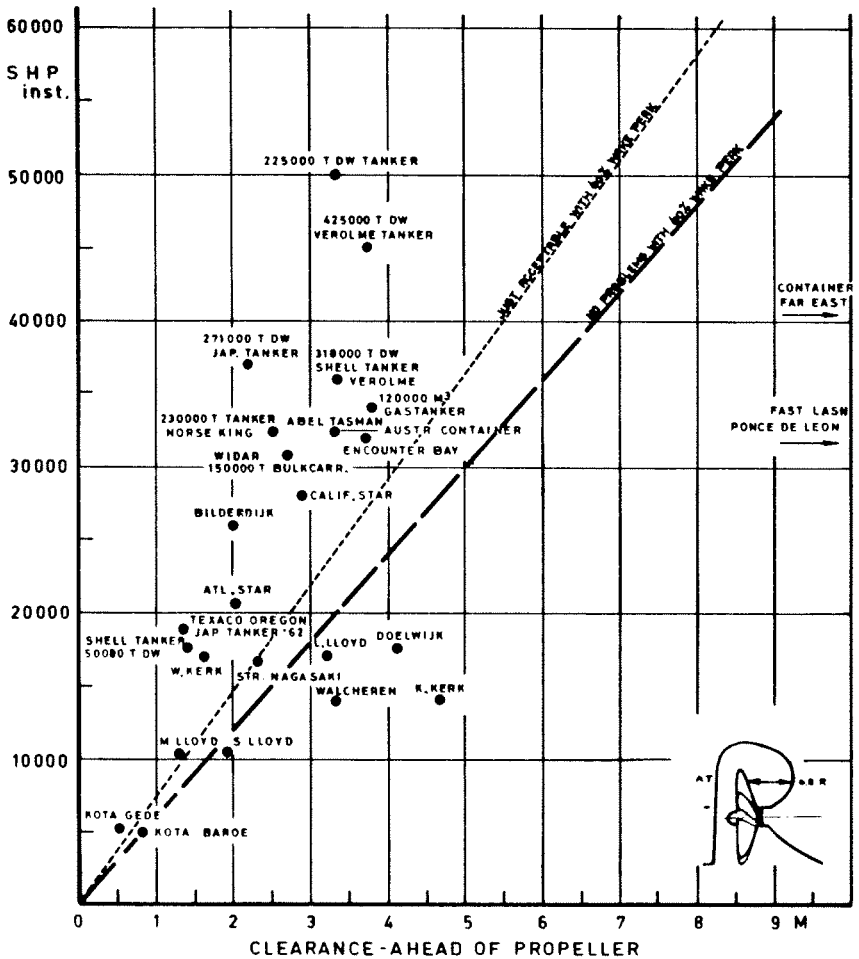
In the past many model basins were giving wrong suggestions to yards because they recommended to create wake in order to obtain a high propulsive efficiency.
(Small aperture, stern bulb) (counter rotation, bossing) (Twin skeg on twin screw).
The yard offered the owner a short cheap ship with a blunt afterbody - small aperture - high wake - high hull efficiency.
Result in basin : less horsepower for speed which seems favourable
Result in ship : heavy vibration.

Contrary the owner wishes a fine afterbody, a wide aperture, a low wake and consequently a small hull efficiency and should be aware of the fact that this gives less deadweight or asks for a longer ship. The owner must pay more for higher quality.
Result of ship : good speed, no vibrations as experienced by Nedlloyd. The scale effect works in favour for the free flow propeller on the real ship.

REMARK 2:

It is better to construct a fine ended propeller aperture on single screw and a slender bossing of elegant light construction on twin screw than to make the construction very strong and big and blunt which will create flow disturbance with consequently excited vibrations.

- Many single screw vessels have a radius in the propeller stern frame at the W.L. 6 m. above base of 250 mm and over.
- Many twin screw vessels vibrate heavily because bossings are strengthned by ring framing inside and consequently the outer diameter is over 2,5 m.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40000
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20000
10000
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425000 T DW
VEROLME TANKER
CONTAINER
FAR EAST
271000 T DW
JAP. TANKER
318000 T DW
SHELL TANKER
VEROLME
120000 M3
GASTANKER
230000 T TANKER
NORSE KING
ABEL TASMAN
AUSTR CONTAINER
FAST LASH
PONCE DE LEON
ENCOUNTER BAY
WIDAR
150000 T BULKCARR.
CALIF. STAR
BILDERDIJK
ATL. STAR
TEXACO OREGON
JAP TANKER '62
SHELL TANKER
50000 T DW
W. KERK
STR. MAGASAKI
L. LLOYD
DOELWIJK
WALCHEREN
K. KERK
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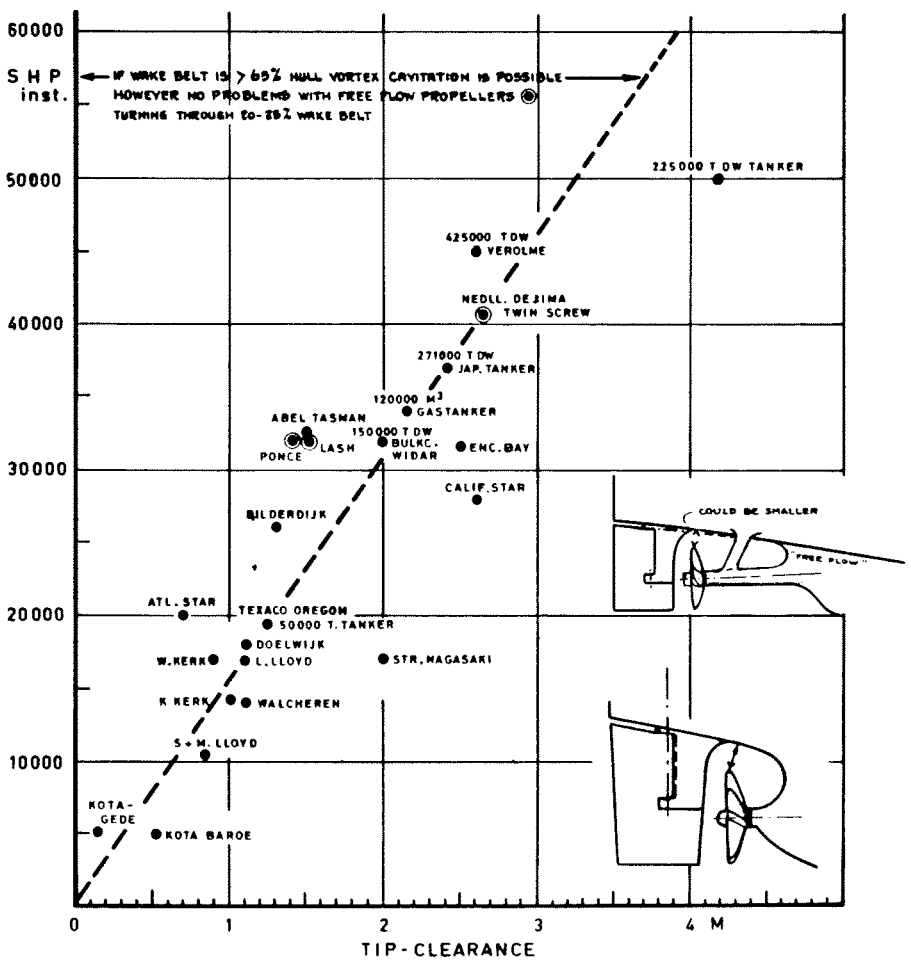
CLEARANCE-AHEAD OF PROPELLER


60000
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IF WAKE BELT IS > 65% HULL VORTEX CAVITATION IS POSSIBLE
HOWEVER NO PROBLEMS WITH FREE FLOW PROPELLERS
TURNING THROUGH 20-25% WAKE BELT
225000 T DW TANKER
425000 T DW
VEROLME
NEDLL. DEJIMA
TWIN SCREW
271000 T DW
JAP. TANKER
120000 M3
GASTANKER
ABEL TASMAN
PONCE
LASH
150000 T DW
BULKC.
WIDAR
ENC. BAY
CALIF. STAR
BILDERDIJK
COULD BE SMALLER
FREE FLOW
ATL. STAR
TEXACO OREGON
50000 T. TANKER
DOELWIJK
W. KERK
L. LLOYD
STR. MAGASAKI
K KERK
WALCHEREN
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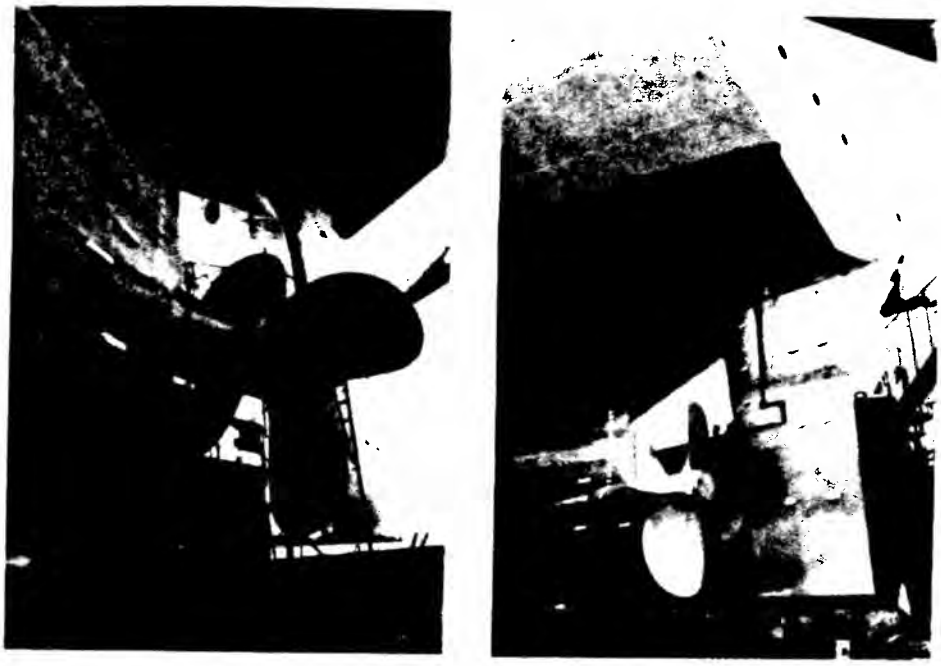
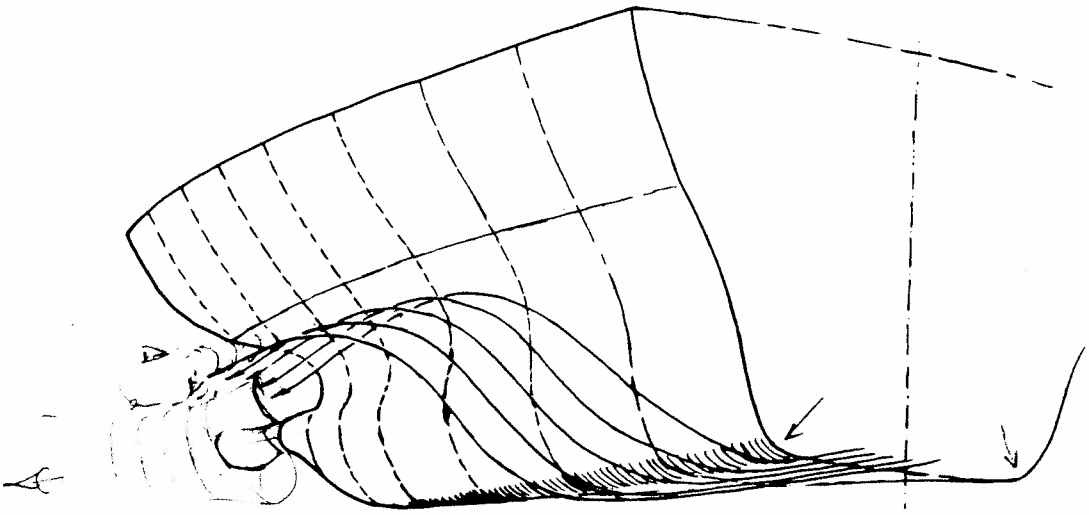
TIP-CLEARANCE

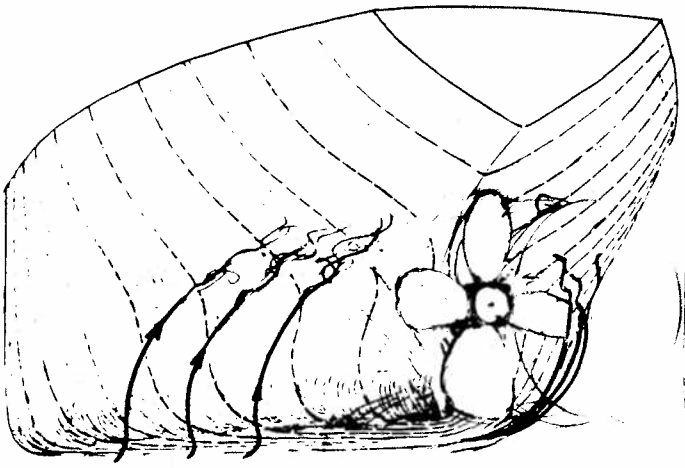


WELL DESIGNED AND PROPERLY CONSTRUCTED AFTERBODY OF MULTI-PURPOSE SHIP "NEDLLOYD BARCELONA"

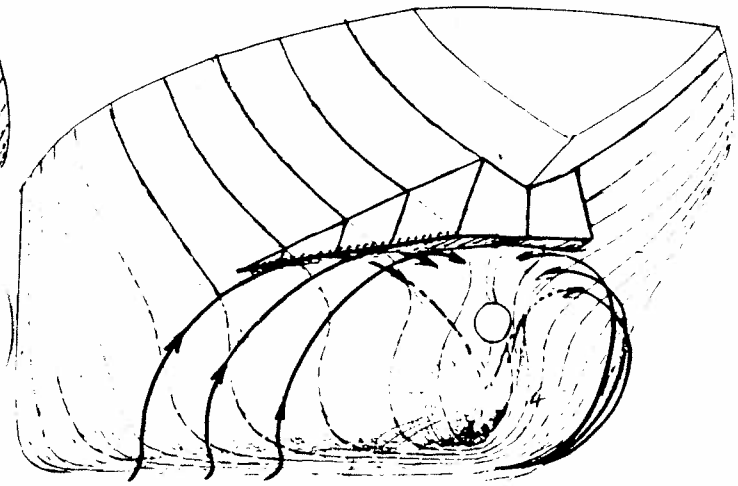
BLUNT AFTERBODY AND NARROW APERTURE OF STANDARD VESSEL



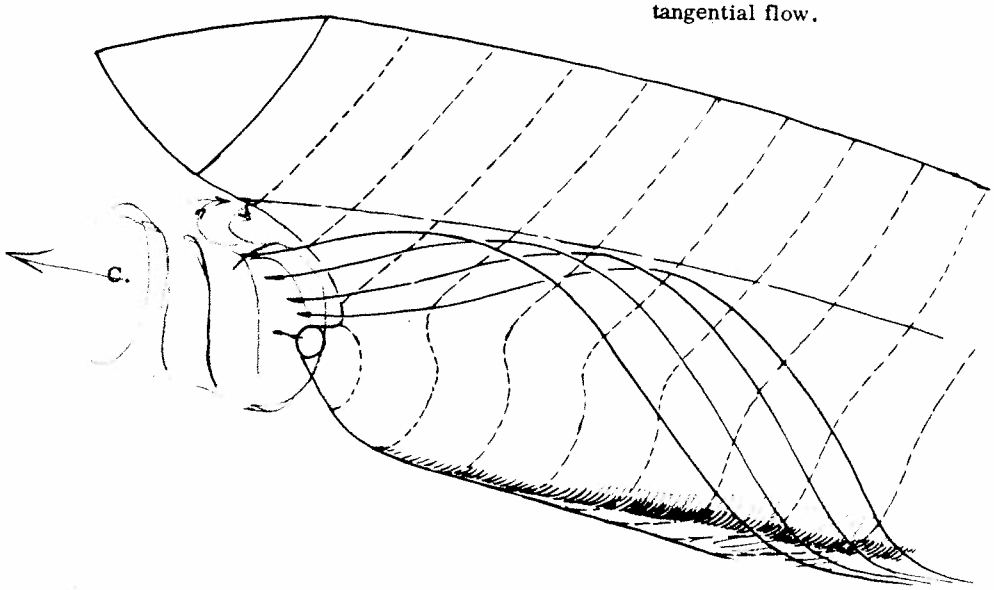
A. Excited vortex by small bilge roundings.

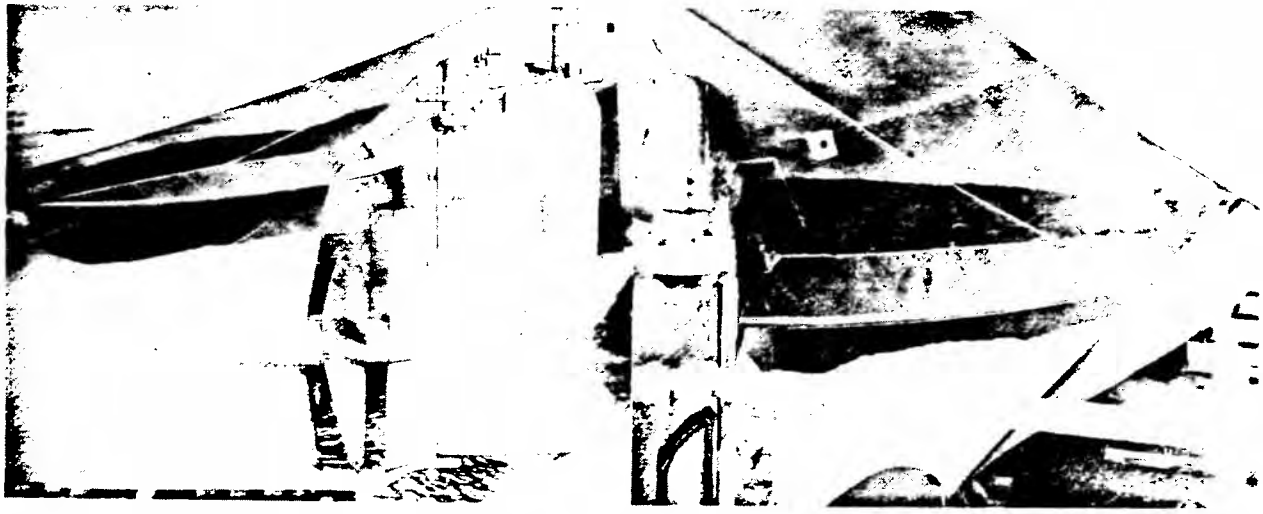


B. Excited vortices strongly influenced by trim and shallow water.

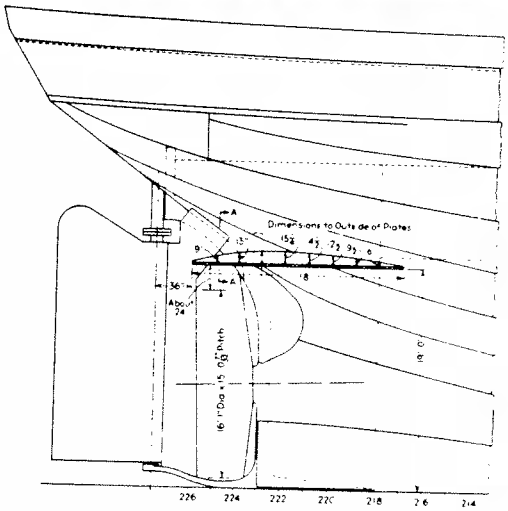


D. Function of fin or tunnel. Directing flow to propeller equalizing wake by reducing the wake peak in top position. However not influencing tangential flow.

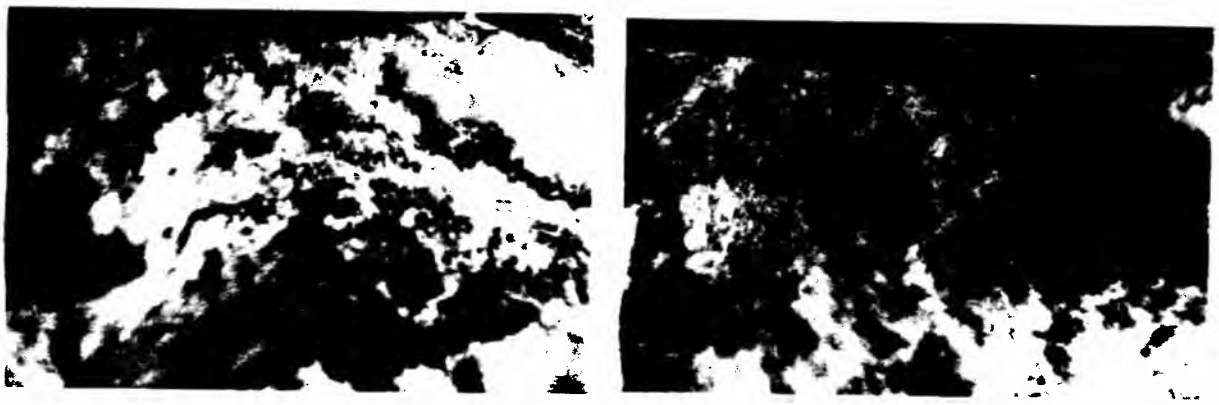




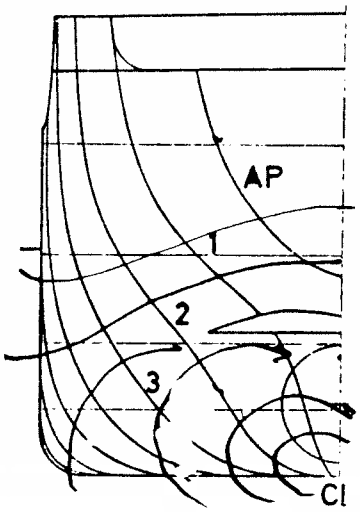
Fin installed on Great Laker 'Carl D. Bradley' which suffered from heavy fantail vibration SNAME 1952 [3].



FIN AS AFTERTHOUGHT

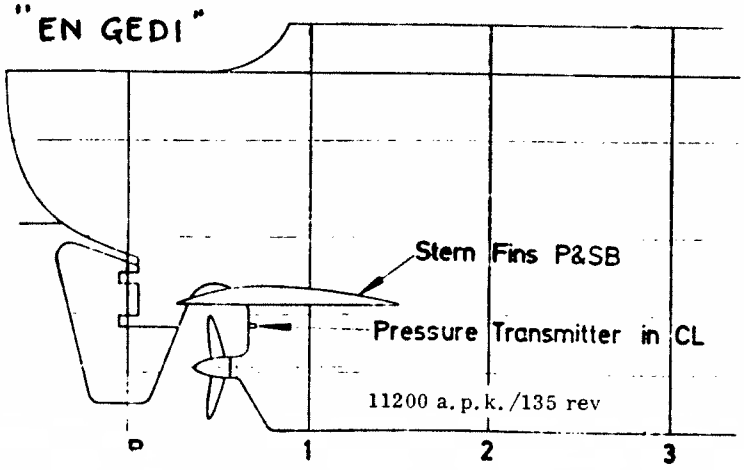


Water boiling up around the stern. Before ..... and ..... After fitting the fin.



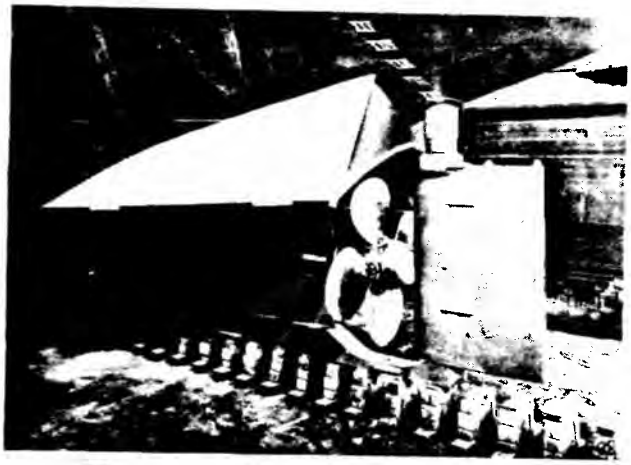
TOO BLUNT AFTERBODY

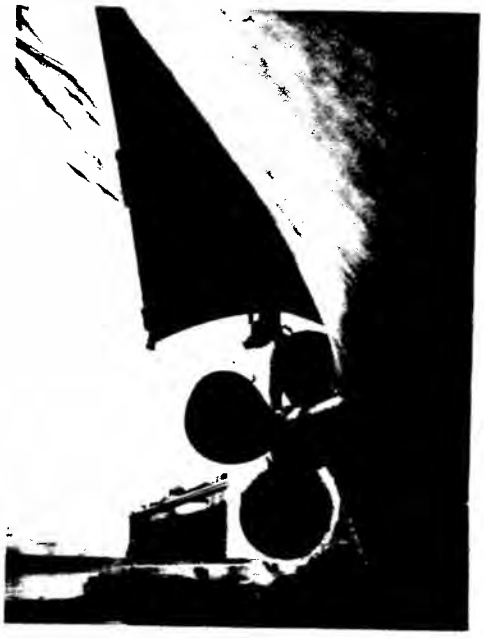
"EN GEDI"



. 1969 Installation of flow accelerating fins in 33000 ts bulkcarriers 'En Gedi' and 'Avedad', proposed by SSPA, S + SR March 6-1970.

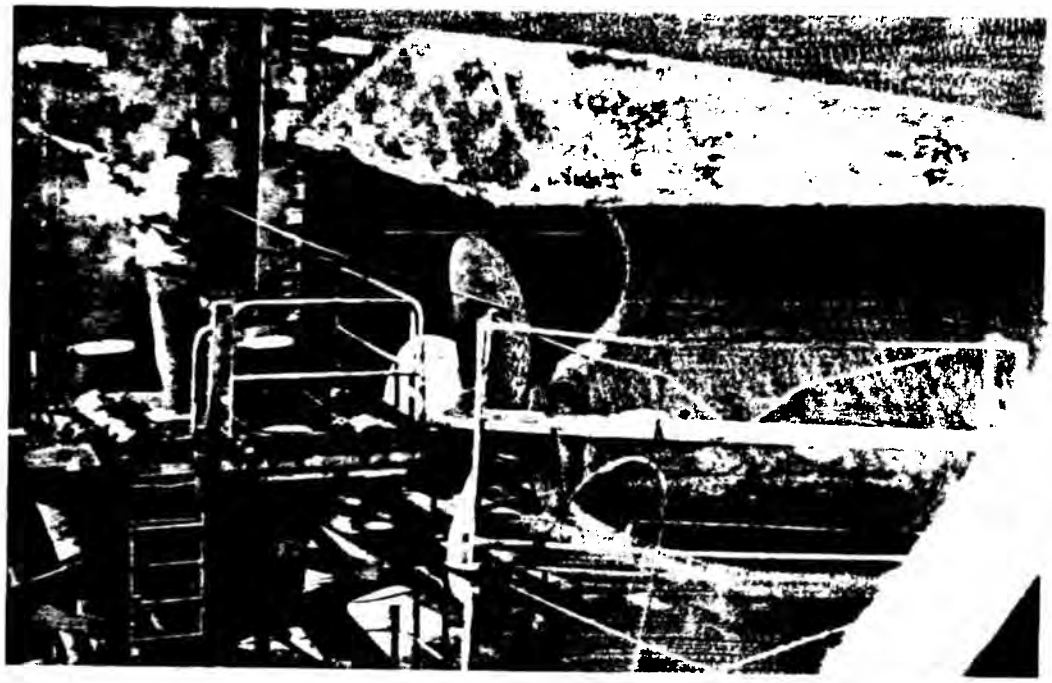
For certain points the amplitude of vibration is reduced to about 1/3.

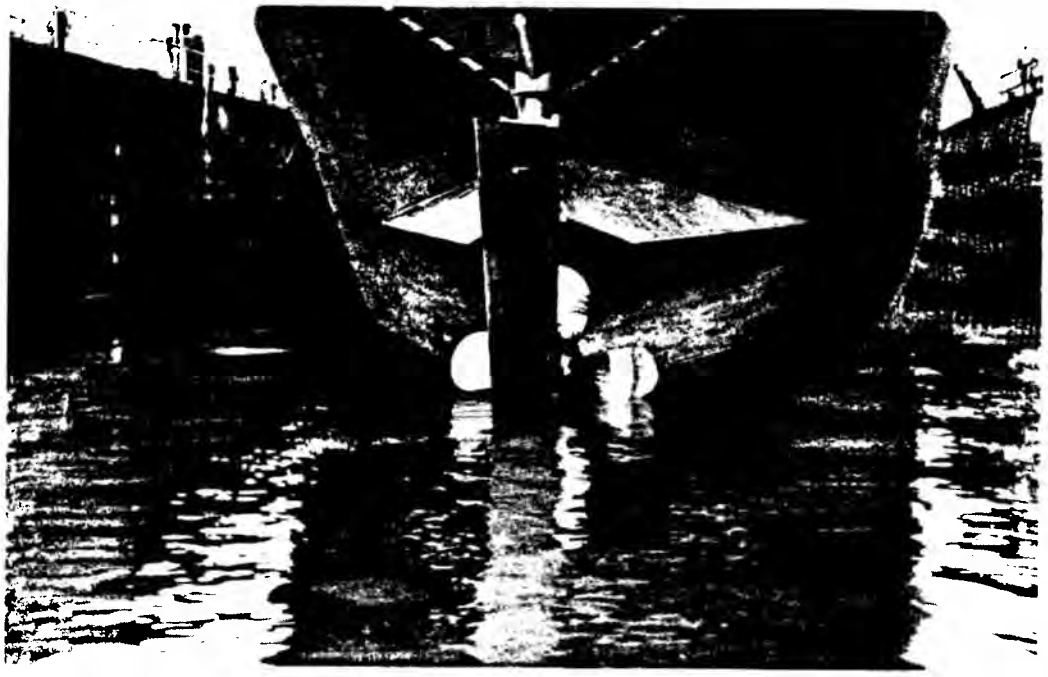




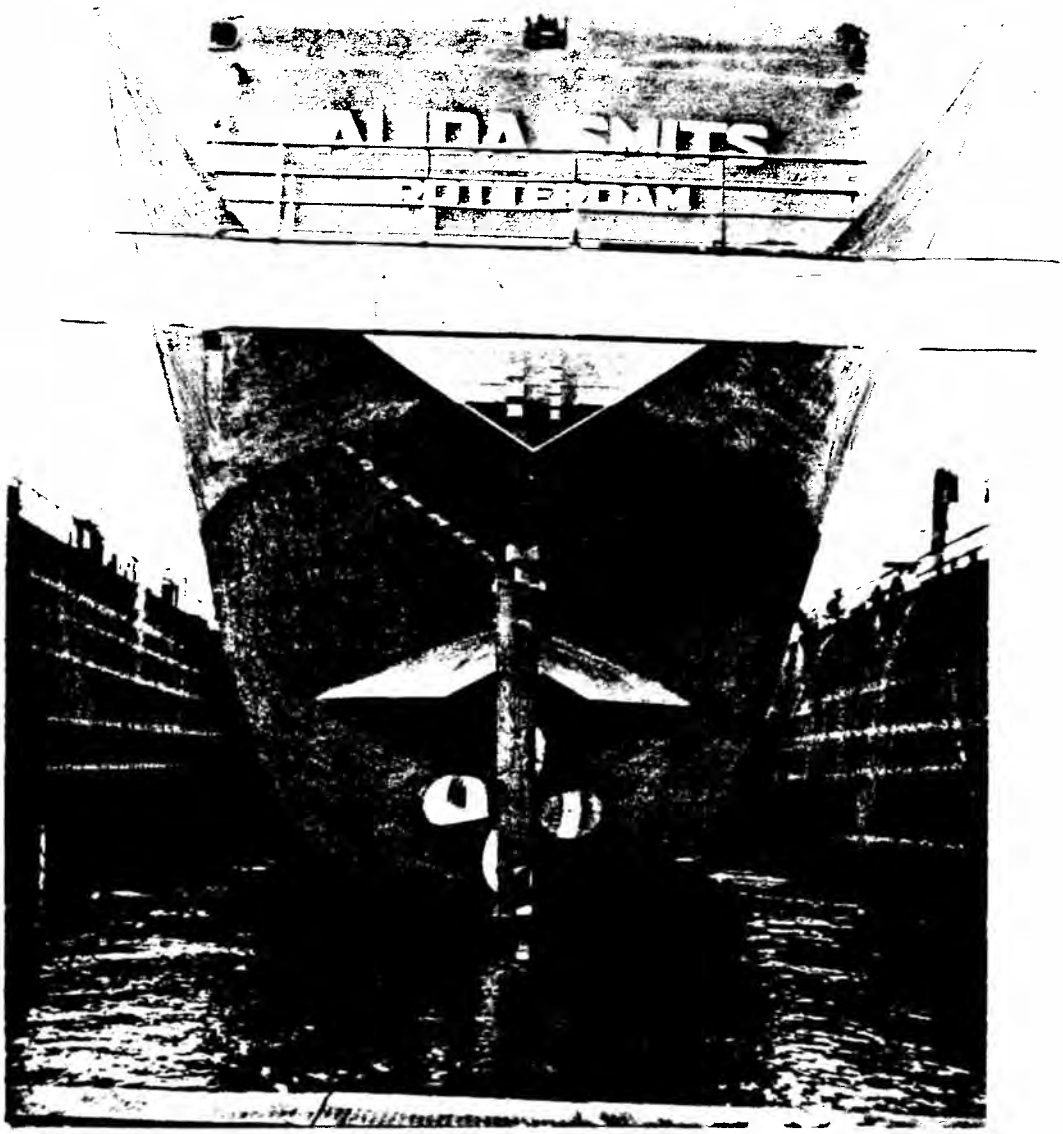
DREDGER "DEEPSTONE"

1972 Installation of tunnel on dredger Deepstone. Proposed by N. S. M. B.
A large reduction of vibration was obtained.

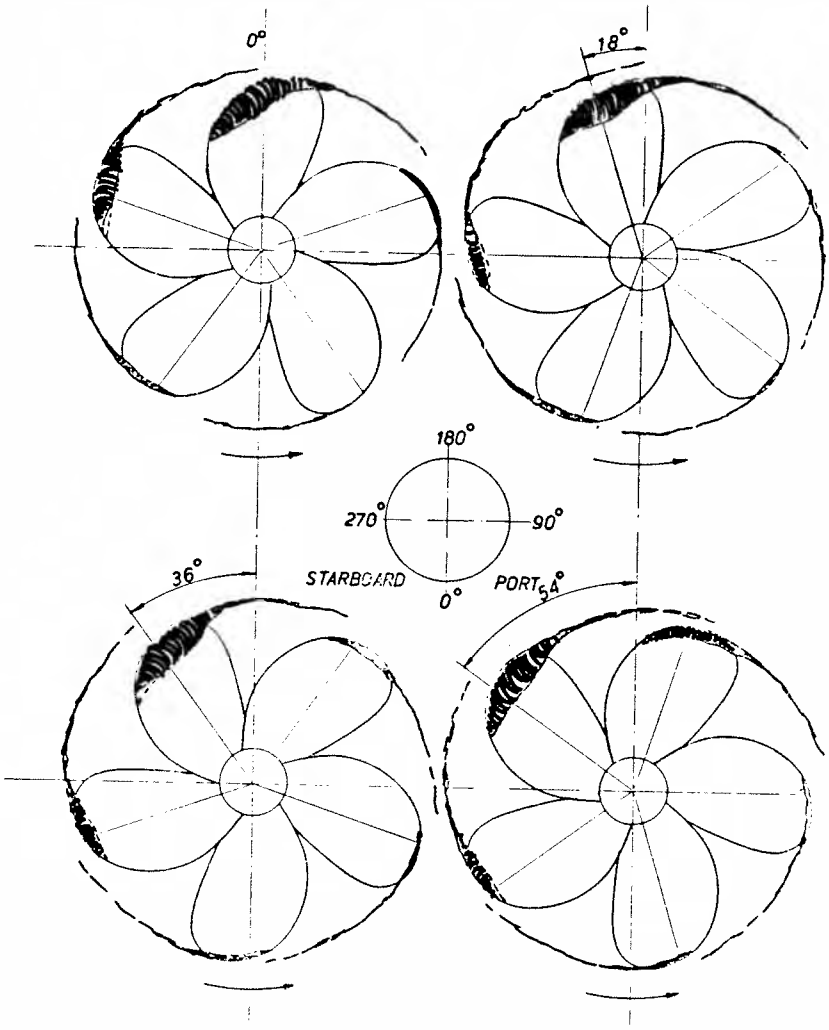




m.s. ALIDA SMITS - Anti-vibratie vinnen.
(foto: Mei 1983)


ROTTERDAM

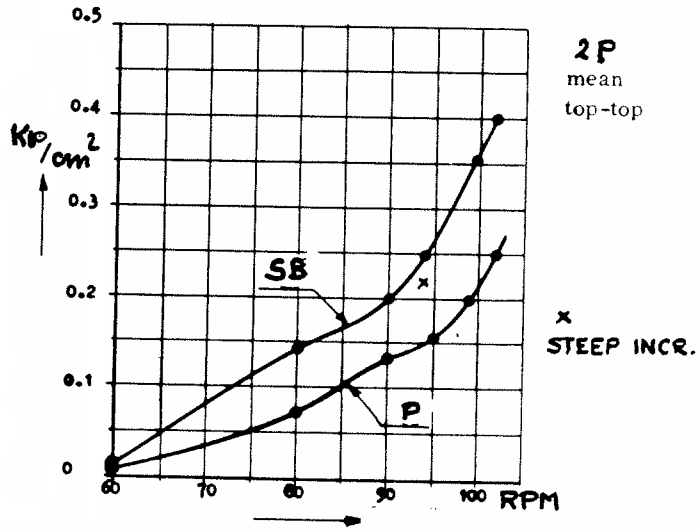
$K_T = 0.2125$; $\sigma_n = 1.77$ PROPELLER MODEL 4525



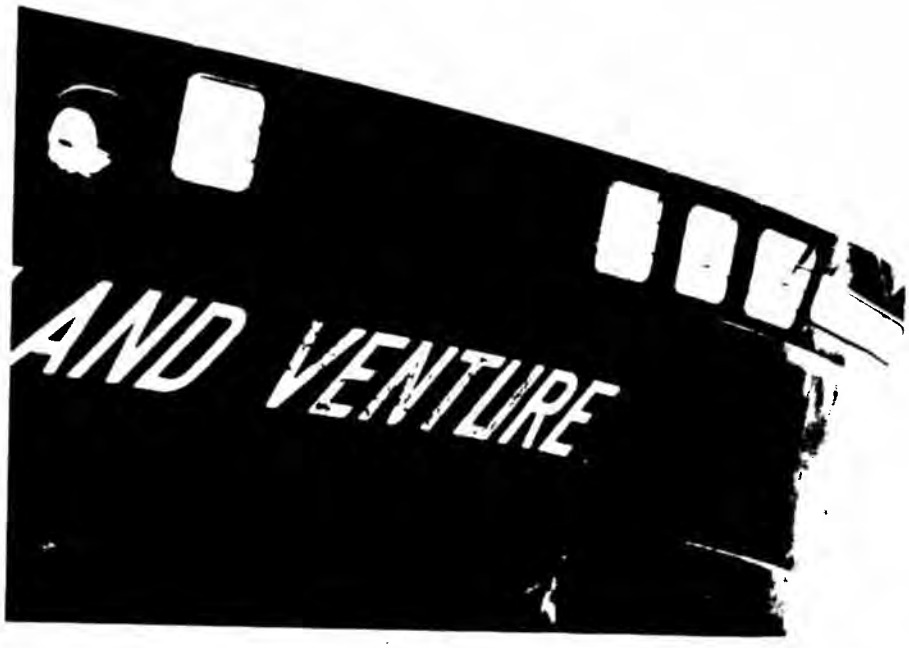
Cavitation pattern on model propeller. 32450 s.h.p. x 110 rev.

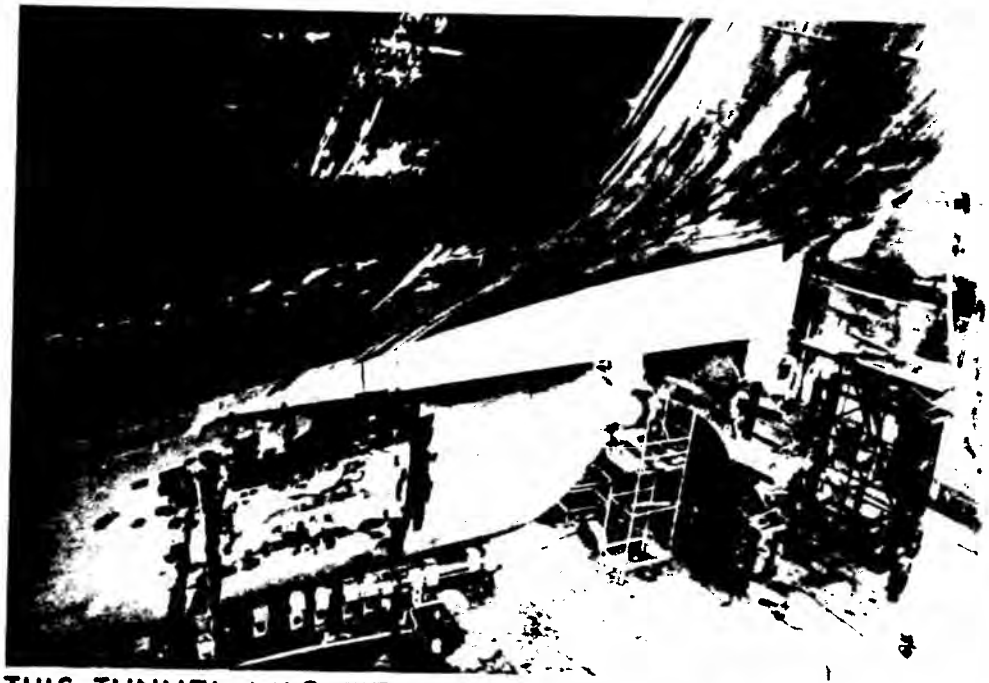
"ABEL TASMAN"

. Dummy of afterbody in cavitation tunnel (N.S.M.B.). Note pressure gauges above propeller.

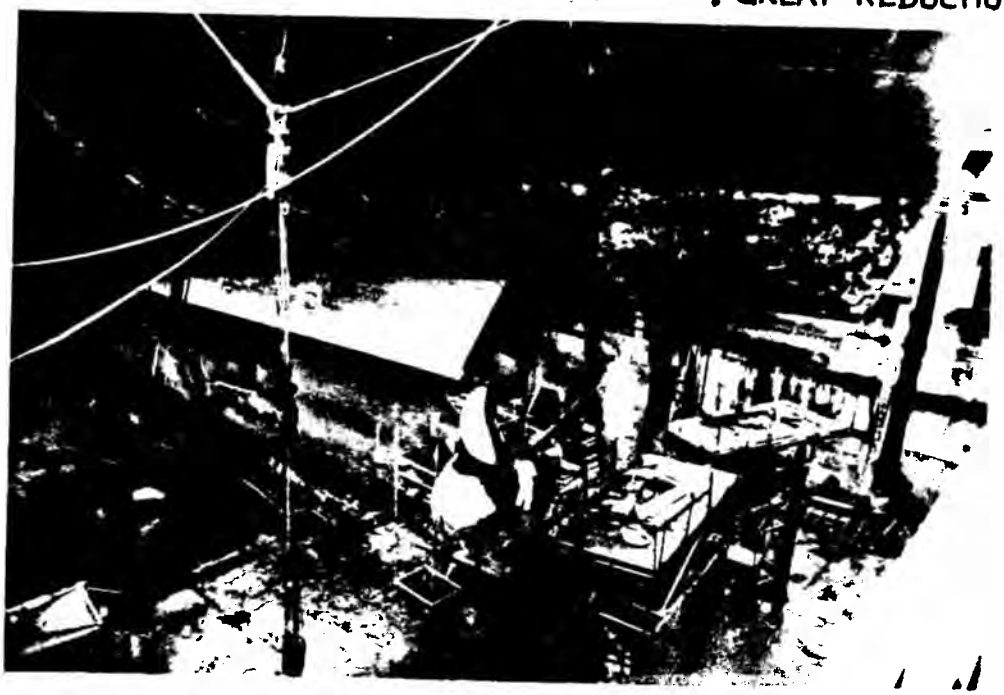


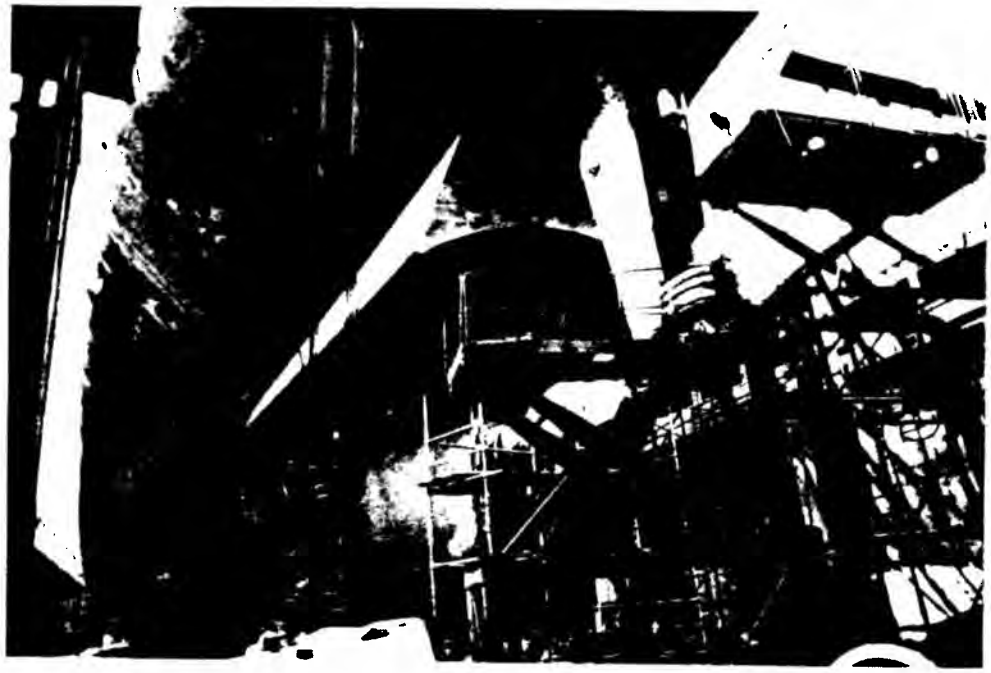
Full size pressure fluctuations. Because of contra-rotating vortex on SB side, pressure fluctuations are much stronger.

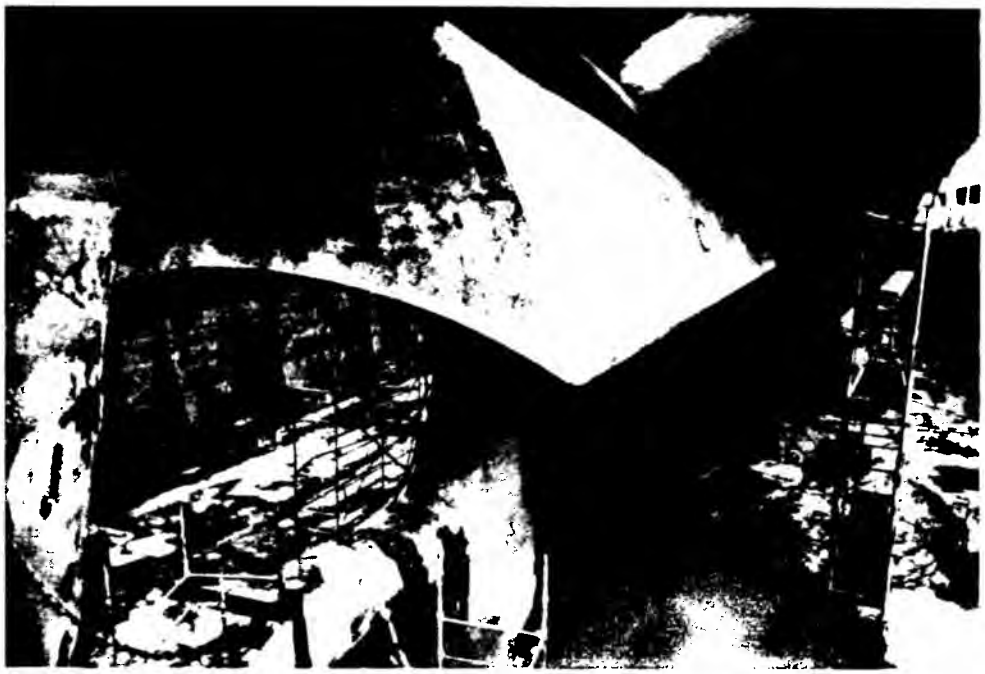


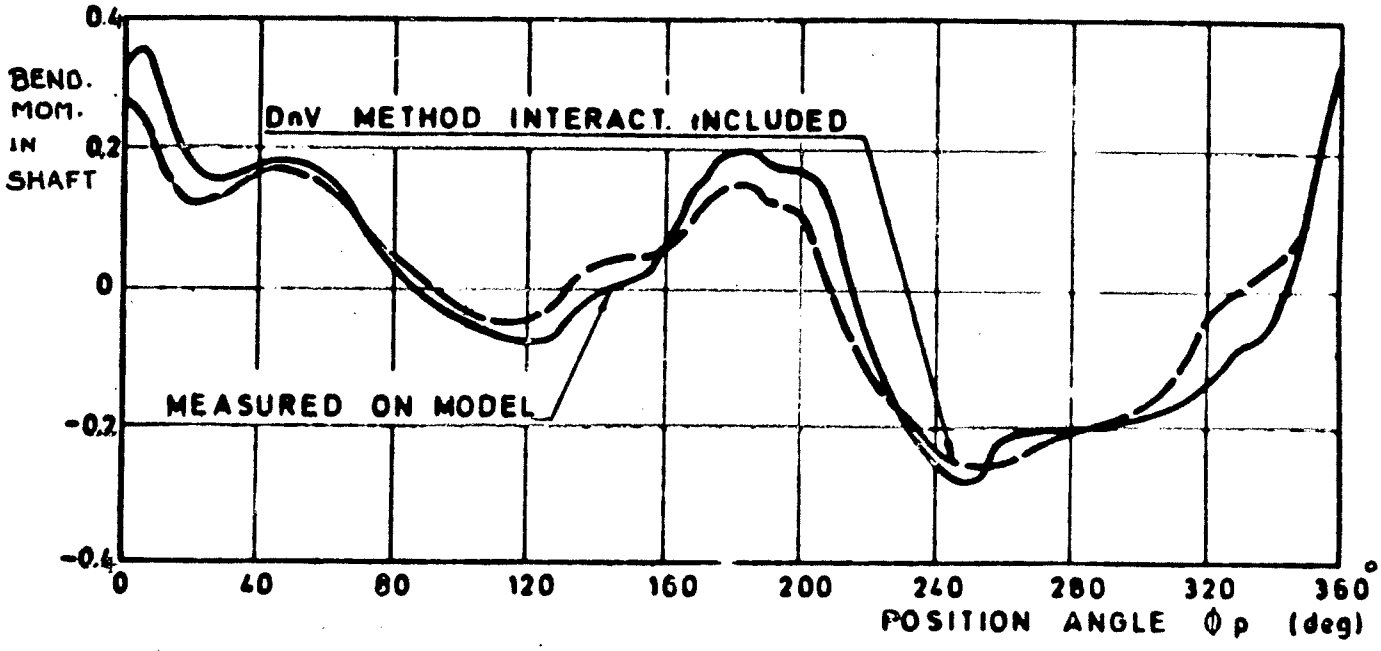


THIS TUNNEL WAS FITTED : GREAT REDUCTION OF VIBRATIONS









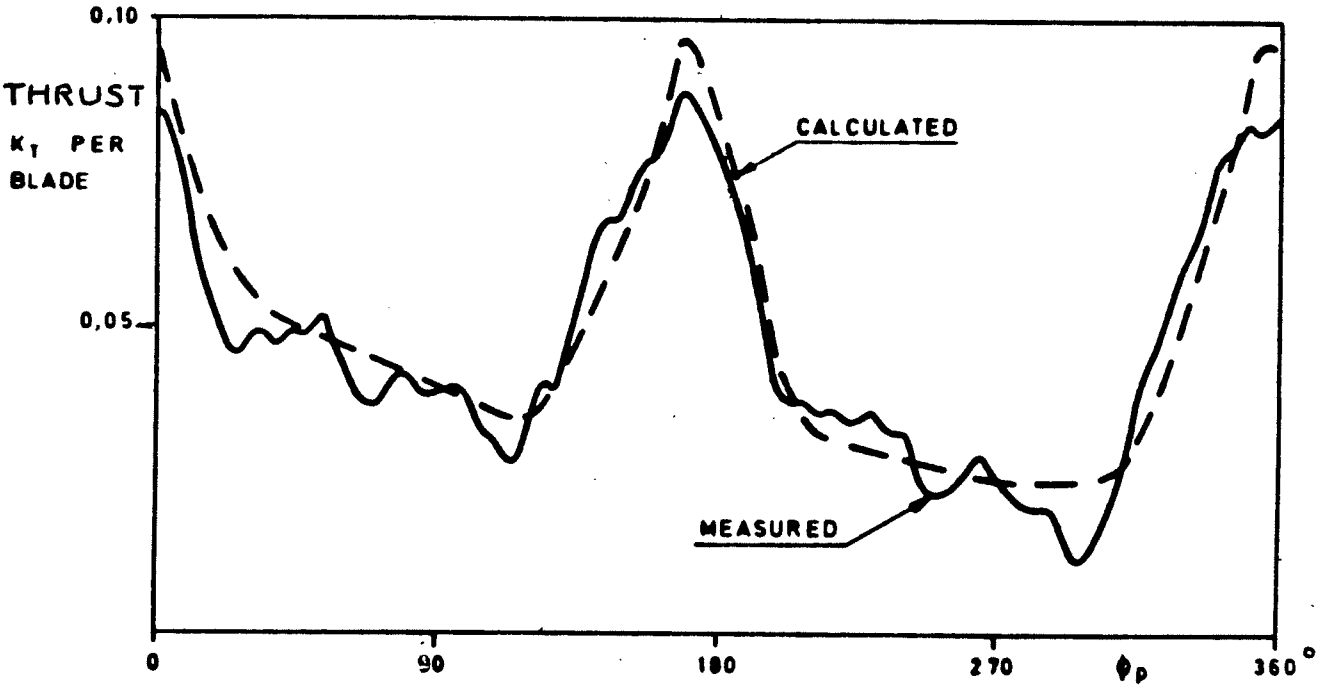
Variation of bending moment on a tanker propeller in a wake. Experiments and calculations

TANKER DnV

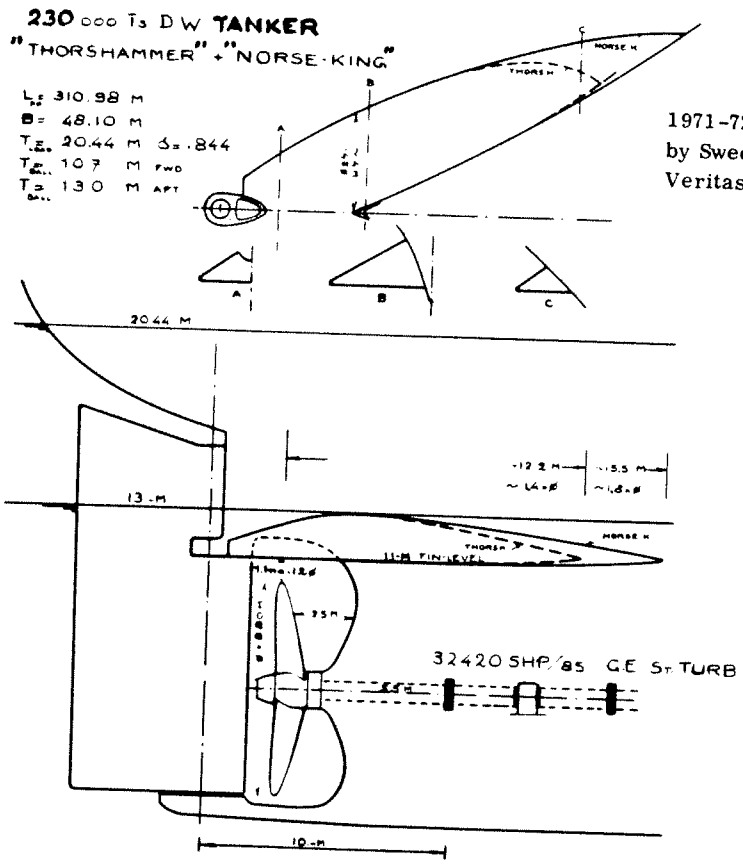
MEASURED AND CALCULATED K_T PER BLADE

$Z = 4$

$A_E / A_O = 0.585$

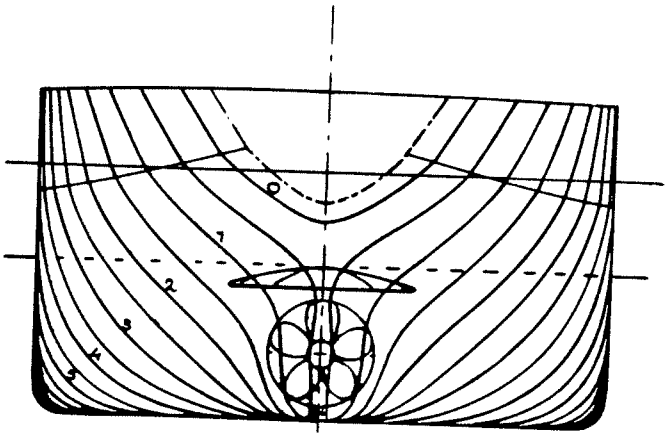


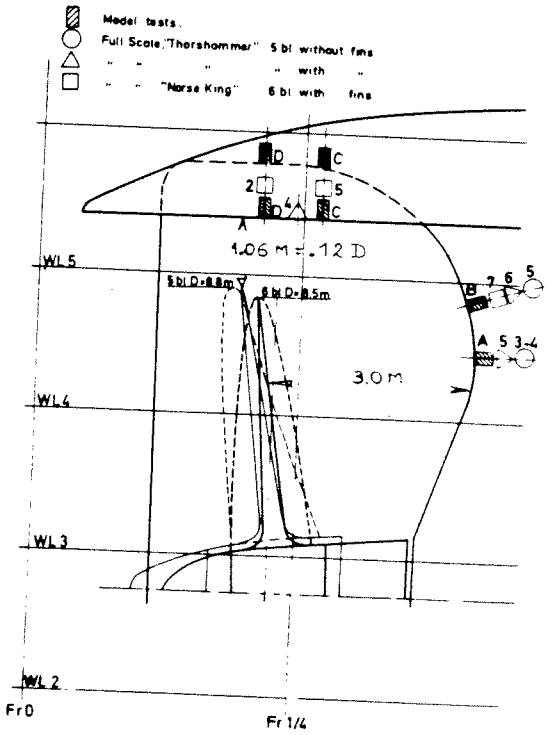
Thrust variation of model propellor in a wake. Experiments and calculations



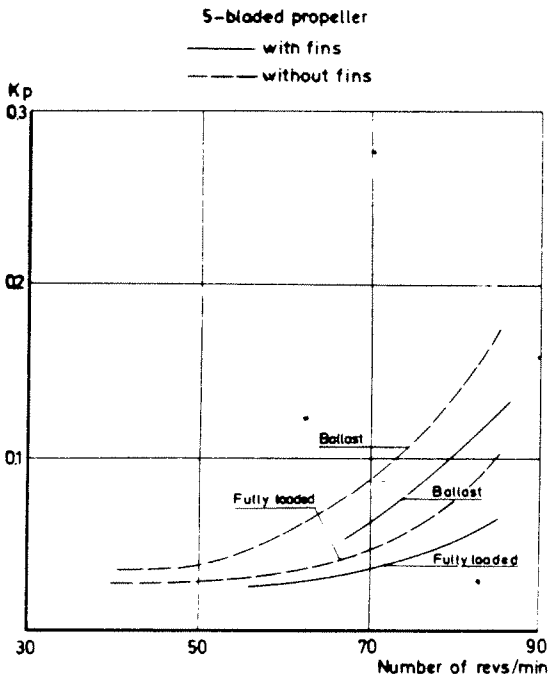
1971-72 Vibration investigation on 230 000 ts dw tanker by Swedish State Experimental Tank and Det Norske Veritas [4,4A].

FIN LATER INSTALLED

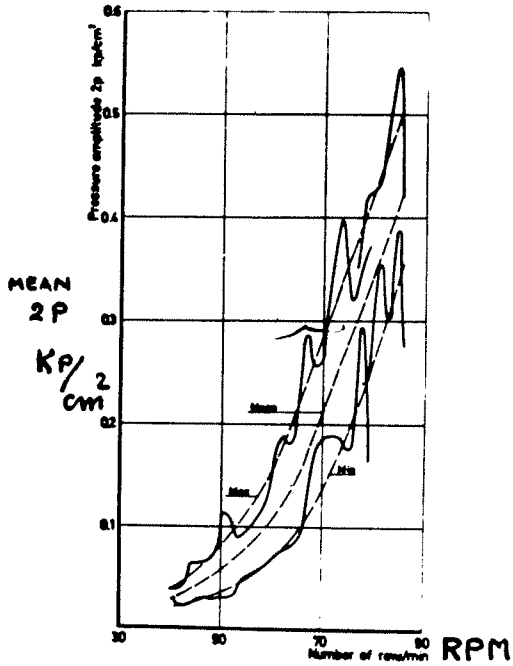




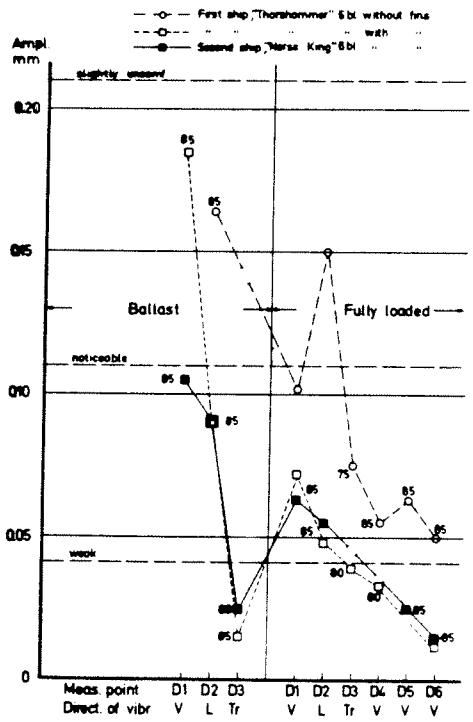
Positions of transducers. For measuring pressure fluctuations on the model and full scale ship.



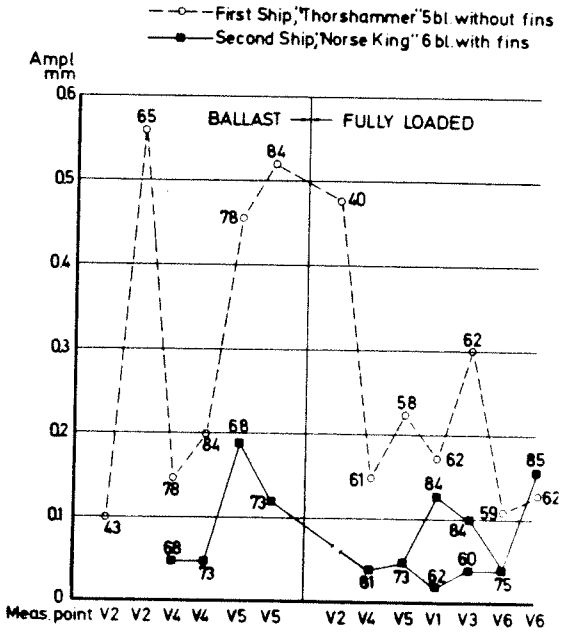
Pressure fluctuations in full scale. T/T 'Thorshammer' (first ship) with and without fins.



Pressure fluctuations in full scale. T/T 'Norse King' (with fins 6-bladed propeller).



Vibration levels in deckhouse. Numbers indicate approximate number of rev. per min.



Vibration levels in wing tank No. 5, SB. Numbers indicate approximate number of rev. per min.

Investigations on a tanker model by S. M. T. Trondheim.

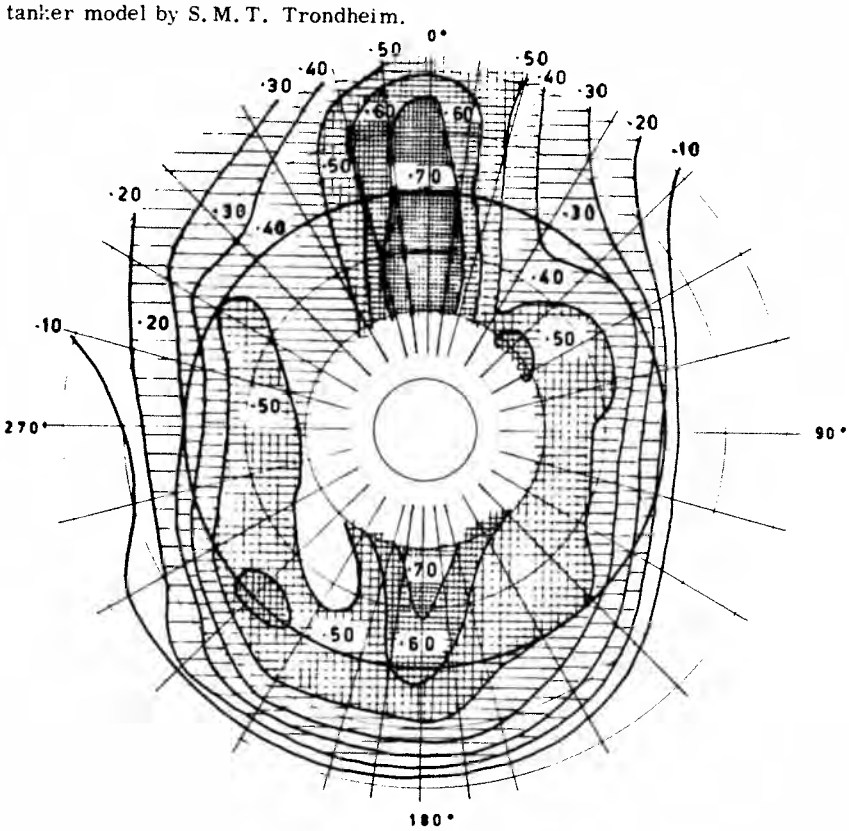


Figure Wake pattern without fin.

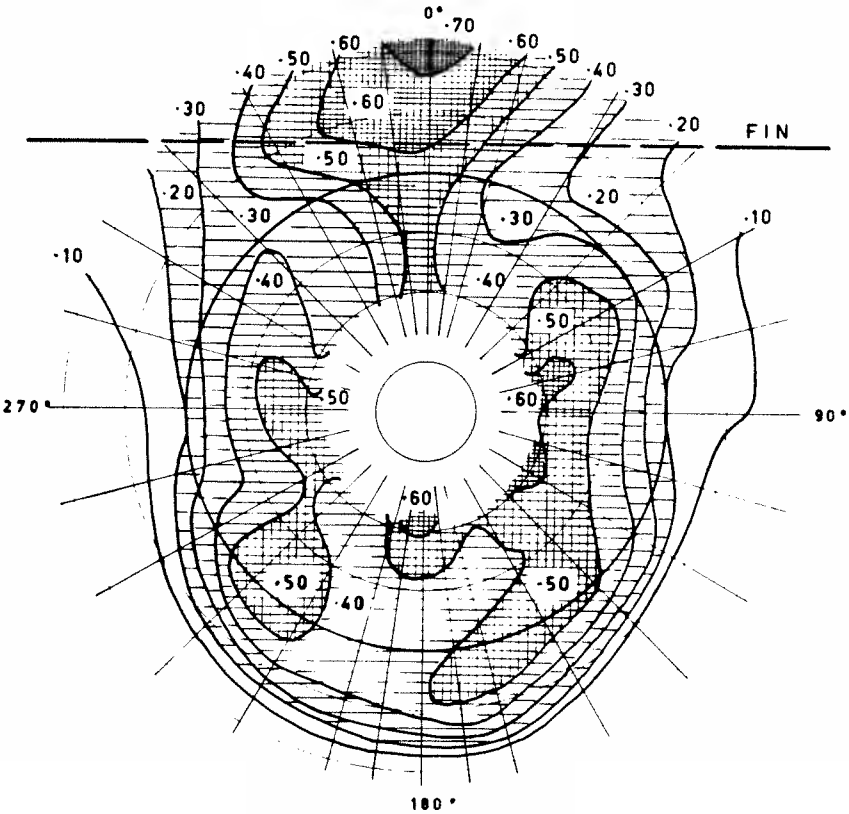
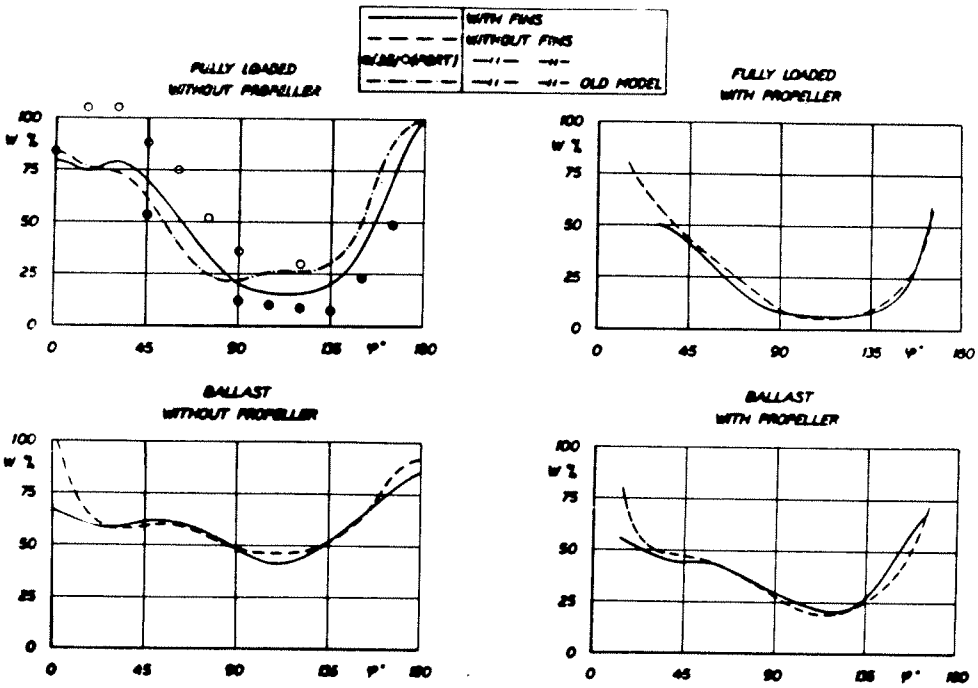
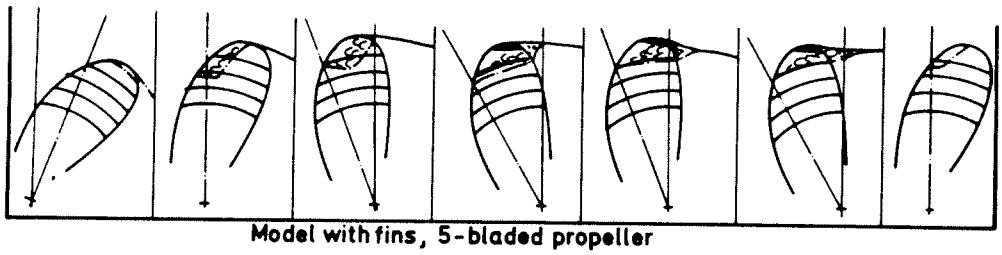


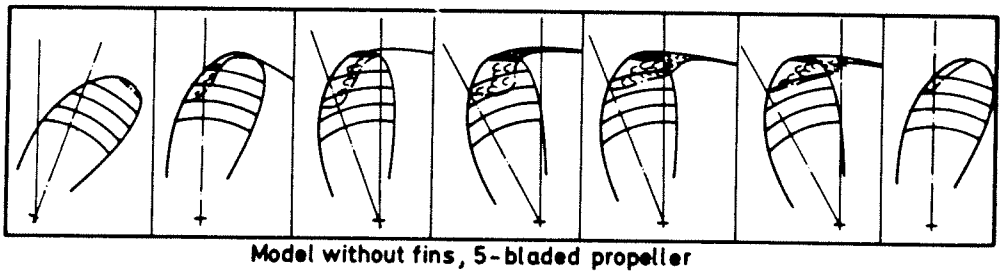
Figure Wake pattern with fin.



Results of wake measurements in towing tank. Radius, R = 70 mm. Results without propeller measured in the propeller plane. Results with propeller measured 68 mm forward of the propeller. Note reduction of wake-peak in top position (0°) by fin.

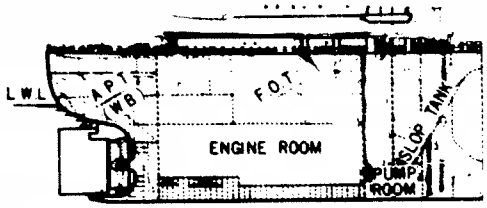


Model with fins, 5-bladed propeller

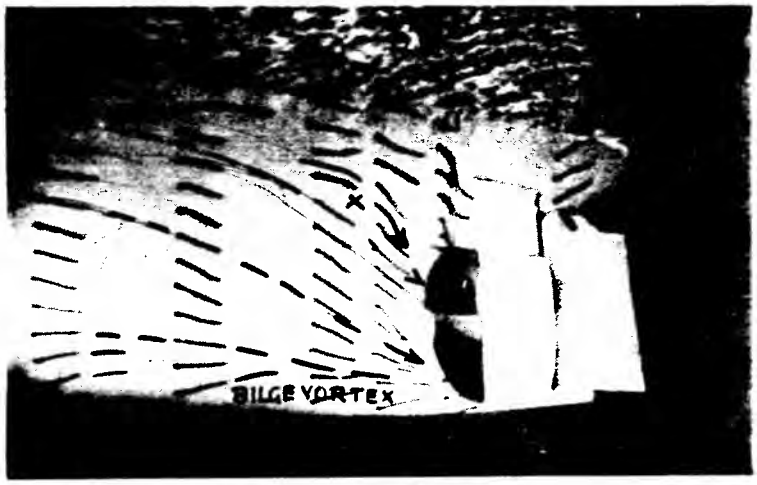


Model without fins, 5-bladed propeller

Tests in cavitation tunnel. Cavitation patterns in different blade positions. Different hull-propeller configurations. Fully loaded condition, 16 knots.



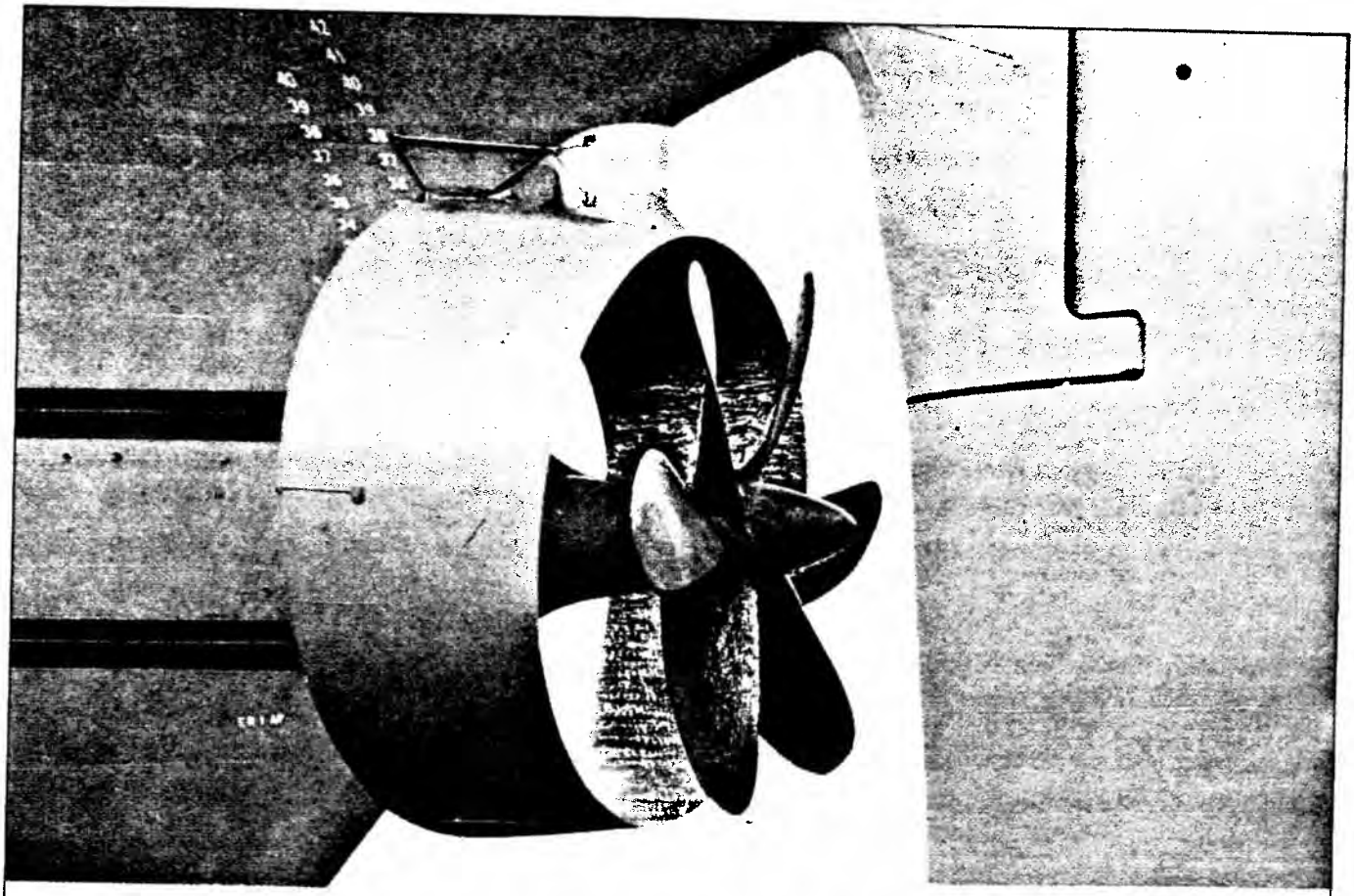
1970 'Golar Nichu' TS Tanker fitted with Kort nozzle. 215000 Ts d.w., 30000 s.h.p. x 90 rev.



× FLOW SEPARATION

AT "12-O'CLOCK" POS^N
SEVERE CAV. EROSION
IN SIDE NOZZLE →

Better steering and course keeping. Less variations in propeller revs in a rough sea. Clearance forward of propeller should be increased to obtain in conjunction with nozzle reduction of vibrations.

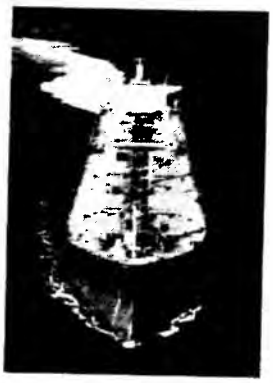


M.I.D.P. Reduces Fuel Consumption of Ships by 5-12%.

Mitsui Integrated Duct Propeller (M.I.D.P.) improves ship propulsion efficiency, lowers fuel consumption and decreases hull vibration.

It reduces hull resistance and is free from the duct's cavitation erosion by installing an asymmetrical duct between the propeller and hull, as a part of the hull. Lower fuel consumption can be achieved without changing the propeller of a vessel in service.

The percentage of fuel cost reduction increases with ship size. In the case of a VLCC or ULCC, a saving of somewhere between 5 and 12% has been confirmed from tank tests and tests using actual ships.



Successful results were obtained at the sea trials of Exxon's 250,000-dw/t tanker "ESSO COPENHAGEN" in 1978. Since then, inquiries for the M.I.D.P. are being received from many shipowners and petroleum companies, domestic as well as overseas.

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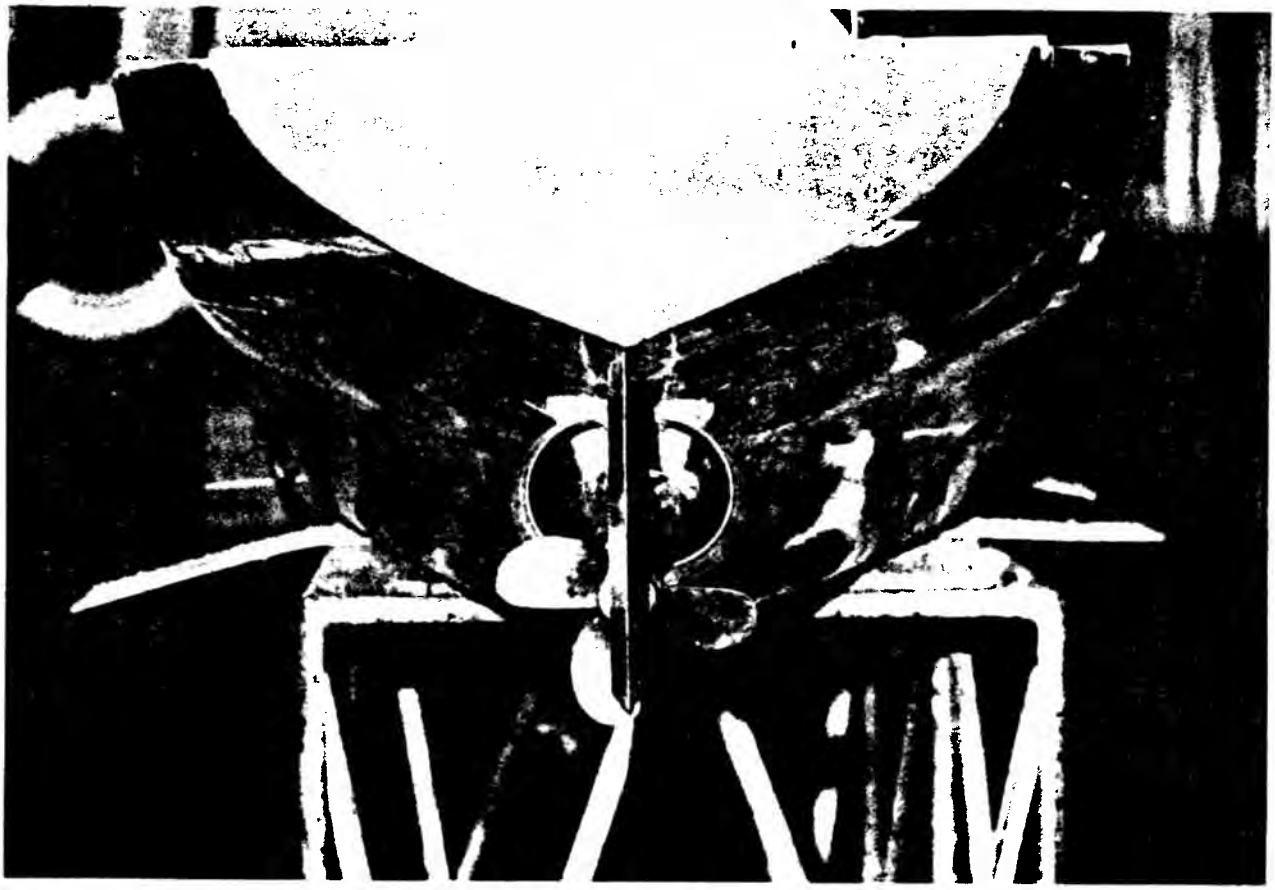
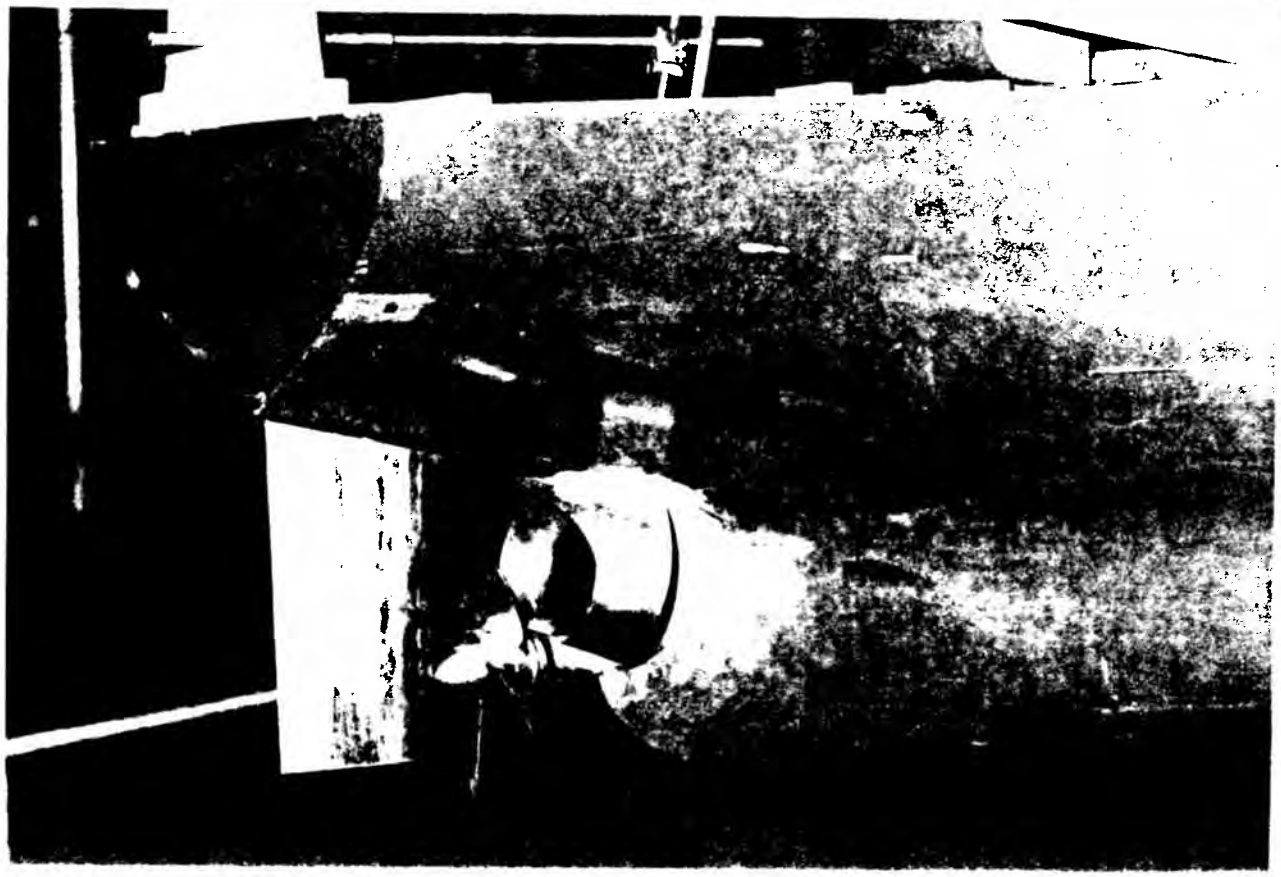
Overseas Offices: New York, Los Angeles, Mexico, London, Düsseldorf, Vienna, Singapore, Hong Kong, Rio de Janeiro

MODELL EINES MASSENGUTSCHIFFS MIT VORDÜSE

SEP 83 SCHNEEKLUTH

AACHEN-DUISBURG 83

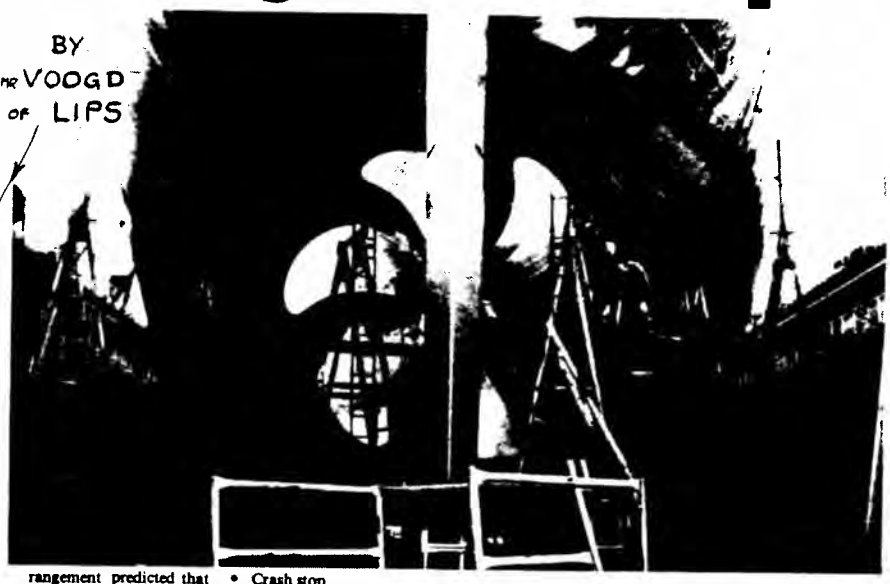




Skewed propellers overcome noise and vibration problems on "Range" class ships

The recent installation of skewed propellers in conjunction with small fins fitted to the afterbodies of "Flinders Range" and "Selwyn Range" has eliminated the source of severe noise and vibration problems and permitted the vessels to be operated at designed power and speed. This article outlines the steps taken to produce these results.

BY MR VOOGD OF LIPS



The Problem

When the "Flinders Range" first entered service in the latter part of 1979, severe localised vibration in the upper areas of the machinery space and in the superstructure appeared when the vessel was in the loaded condition.

In-service experience also highlighted the phenomenon that violent shuddering of the vessel developed when proceeding in ballast into a head sea when the stern tunnel became immersed.

As a result of the high level of noise and vibration both vessels were operated on r.p.m. reduced to permit a reasonable level of habitability.

An independent surveyor conducted a vibration survey of the superstructure when the vessel was in the loaded condition between Muscat and Abu Dhabi in September, 1979.

The surveyor's report indicated that the superstructure was resonant in the fore and aft direction at normal service engine revolutions due mainly to insufficient longitudinal structural steel bulkheads in the superstructure.

The existing structural steel bulkhead exhibited a lack of continuity.

The recommendations that followed suggested consideration be given to the fitting of either a four or six bladed propeller so that at normal engine r.p.m., exciting forces would occur out of the natural frequency range of the superstructure.

The Solutions

New Propellers, Development of Design: ANL proceeded with the investigation of a suitable propeller design which would permanently solve the current problems. Tenders were called for tank testing and the manufacture of two suitable propellers. The Nederlands Ship Model Basin (NSMB) at Wageningen, Holland, was commissioned by the propeller manufacturers to undertake model tests to confirm an optimum propeller design, that would include elimination of the existing stern tunnels.

A series of cavitation tests was commenced at NSMB's Depressurised Towing Tank in Holland with a model of the new six-bladed propeller, without the stern tunnel being fitted. These tests revealed that a propeller hull vortex (PHV) was still present, and, since the application of a stern tunnel was to be avoided, the next step was to investigate whether the application of fins only would lead to a solution of the PHV problem. Several different configurations and combinations of fins were tried until the right design was evolved.

Diagram "A" illustrates the positioning of the fins.

The results of the model tests with the new six bladed skewed propeller and small fin combination are summarised below:

- The propulsive performance of the six bladed propeller was predicted to be equivalent to that of the original propeller.
- The cavitation of the new propeller was expected to be harmless from the erosion point of view.
- The final four fin arrangement predicted that PHV cavitation would not occur.
- The hull pressure fluctuation levels indicated that ship vibration would be insignificant.
- The new propeller details would be of the skewback type with a diameter of 5500 mm and a mean pitch of 4290 mm.

New propellers were delivered to Australia early in 1982 for installation in May and June when the vessels were drydocked. Advantage was taken at the same time to completely gritblast the hulls and coat both vessels with a self polishing anti-fouling paint in order to get the underwater hull surfaces to the best possible condition and save fuel.

The photograph shows the installation of the propeller to "Flinders Range" in drydock. Three of the fins can be seen one on either side of the rudder and one horizontally attached to the rudder horn.

Sea Trials, Results

On completion of the docking on 23-25 May, 1982 the "Flinders Range" proceeded to Kiawa, N.S.W. where sea trials were conducted in accordance with B.S.R.A. comprising the following:

- Speed and power
- Manoeuvring trials
- Noise level recordings
- Vibrations of ship's structure and engines

The speed power relationship of the vessel is shown in diagram "B".

Manoeuvring trials included the following

- Turning circles
- "Z" manoeuvre
- Spiral manoeuvre
- Crash stop
- Inertia stop
- Williamson turn

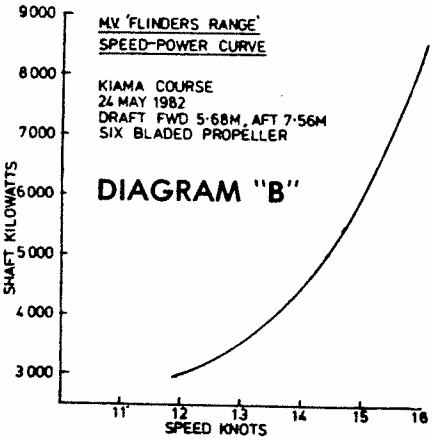
Vibration measurements were taken at similar positions as taken originally both in the accommodation and in the machinery space and were predominantly less than $0.2 m/s^2$ and well below I.S.O. standards.

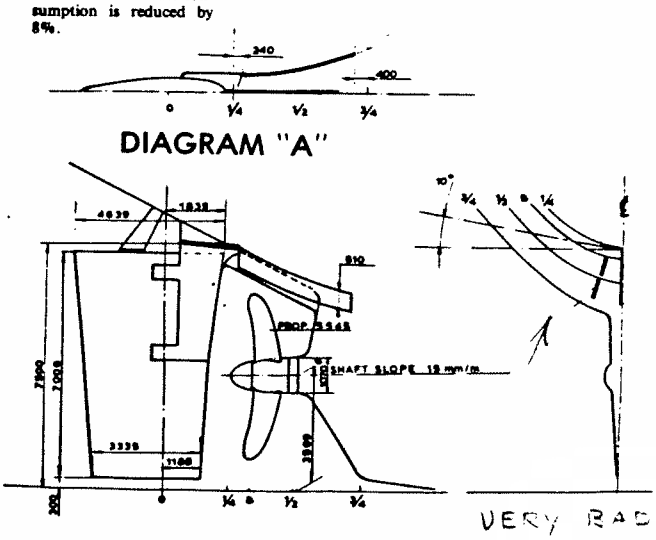
Noise — extensive recordings of noise levels were taken and showed an over-all reduction to a standard significantly below Australian requirements.

Conclusion

The redesigned propellers and stern fins have accomplished the objectives:

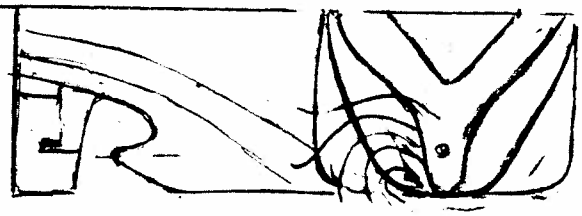
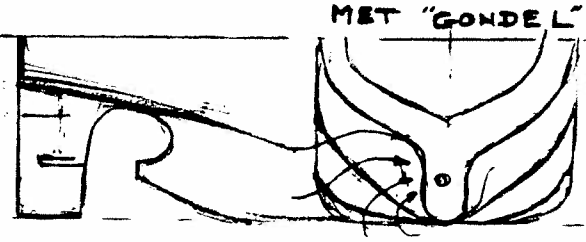
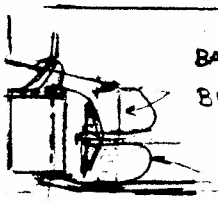
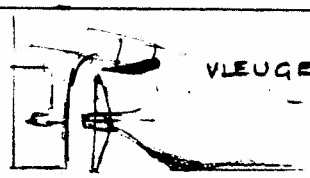
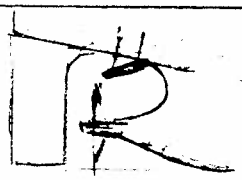
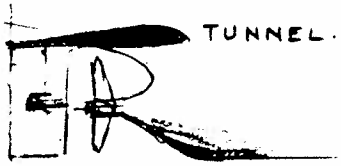
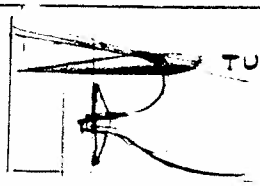
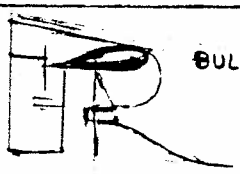
1. Noise and vibration levels have been reduced to acceptable standards.
2. The ships' seagoing performance is no longer constrained and fuel consumption is reduced by 8%.





VERY BAD HULL DESIGN

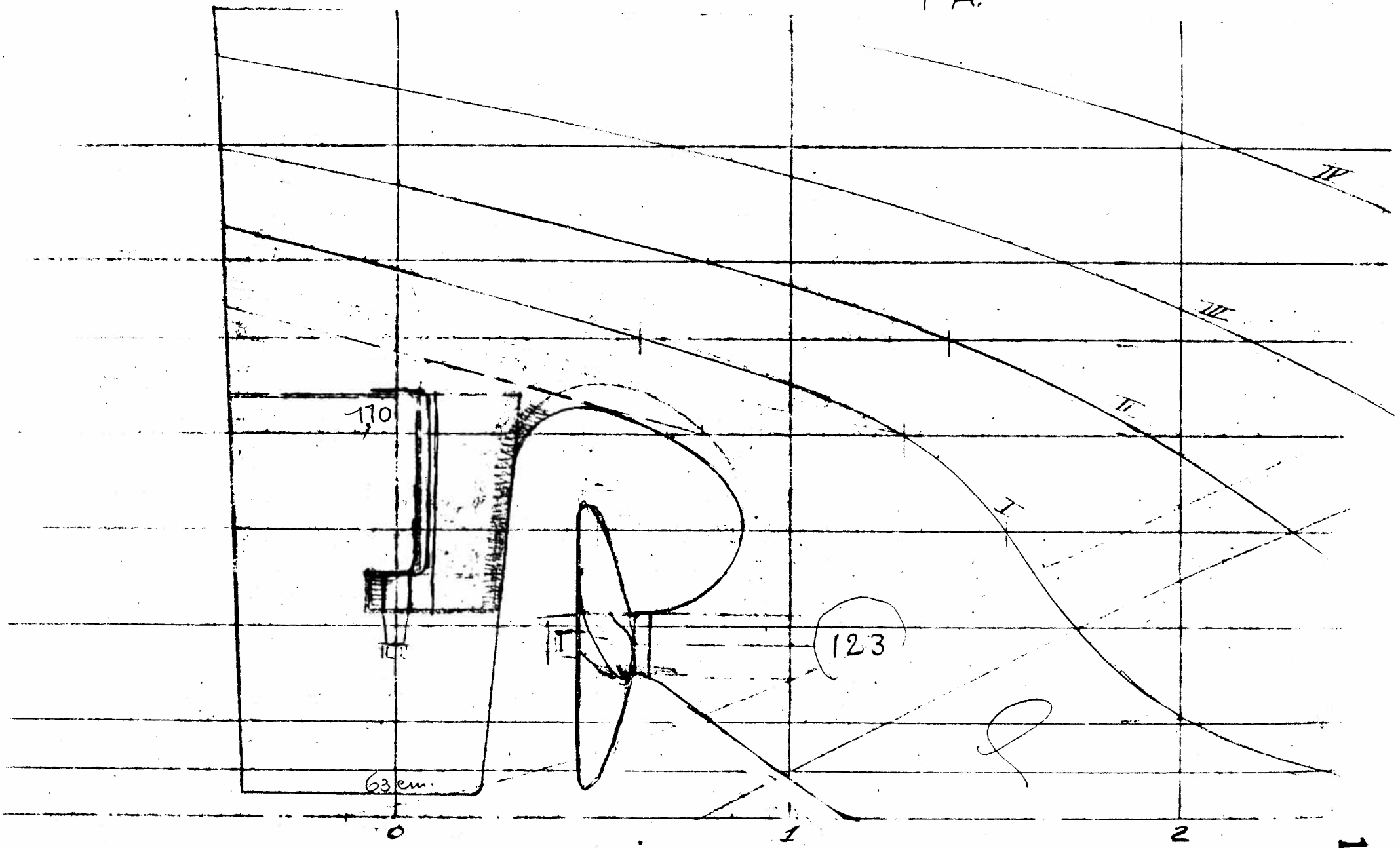
PR. TANKER.

	1 = BASIS VORM	2 = MATIG "PRAAM"
	ZAKSPANT 	MET "GONDEL" 
1-A	⊗ Ø 5,8 123 RPM. 6 cyl B+W 85% MCR = 11100 / 183 RPM.	Ø 5,8 123 RPM.
1-B	× noodsproeg. Ø 6,8 90 RPM. 85% = 10400 /	Ø 7,0 90 RPM.
1-C	⊗ BUIS 3 VARIANTEN Ø 5,8	BUIS. Ø 5,8
1-D	 BALANSROER + HAK BUIS. ook zonder buis te onder.	
1-E	×  VLEUGEL.	 VLEUGEL.
1-F	 TUNNEL DAK.	 TUNNEL DAK
1-G	LICHAAM.	 BULBLICHAAM.

MAY '80 TRYING TO REDUCE REQ. H.P BY: LOWER REVS, LARGER Ø AND APPENDAGES

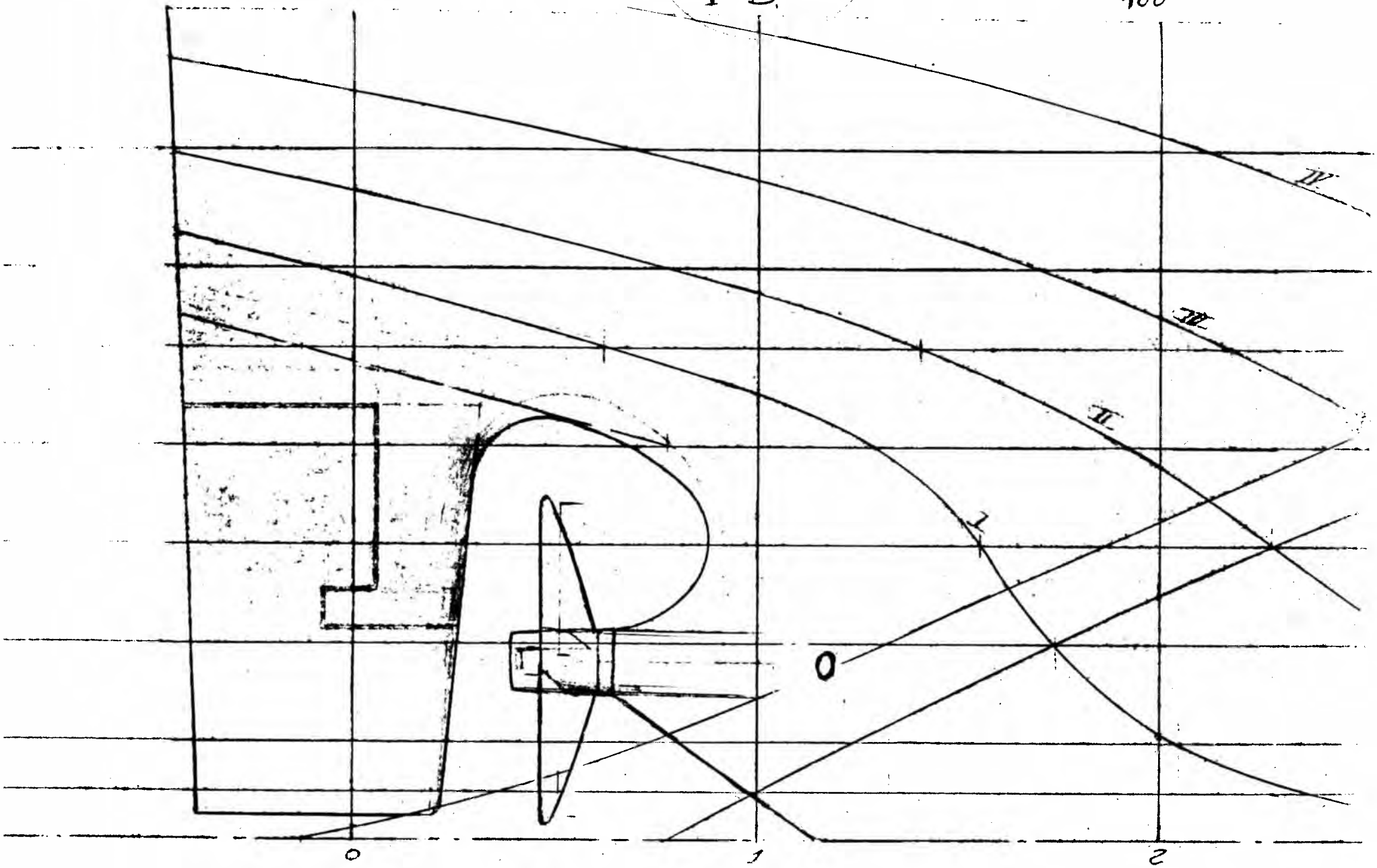
BASIS VORM, 123 RPM = Ø: 5,8

7 A.



model № 5734

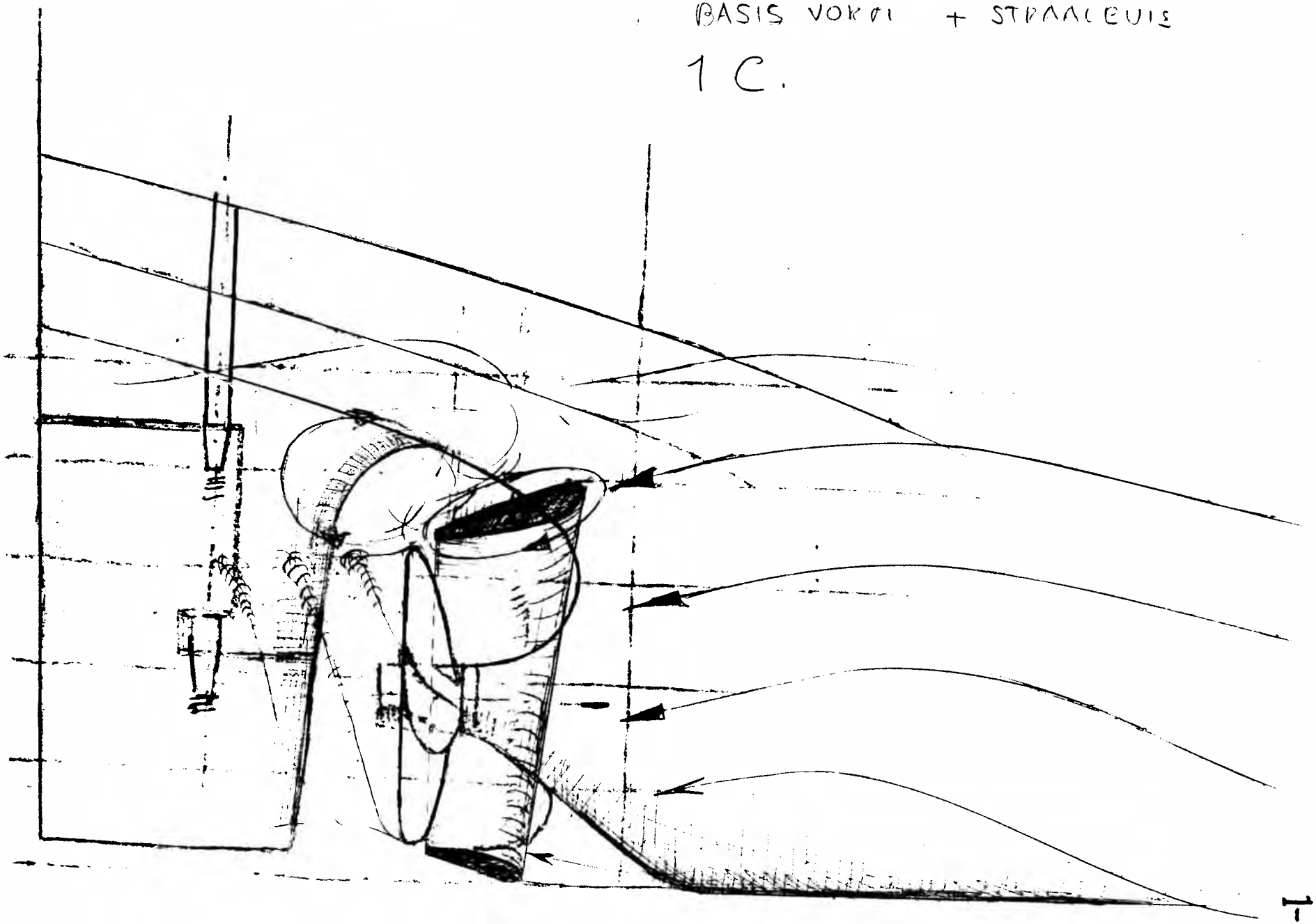
BASISVORM 1 B. LANGZAAM TOEREN 90 à 100 RPM – ∅ ~ 6.5 m



model № 5734

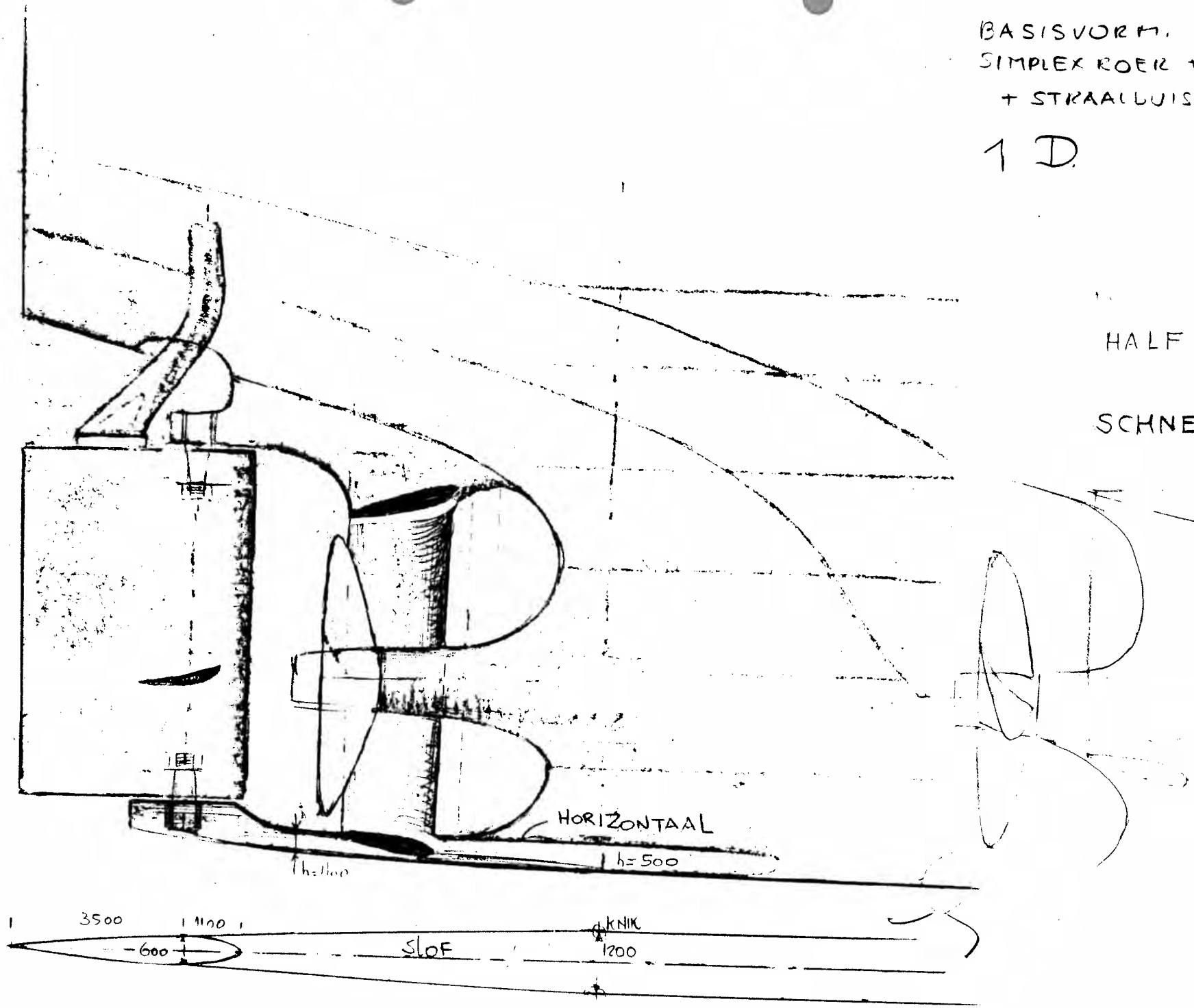
BASIS VOKOI + STRAALEUIS

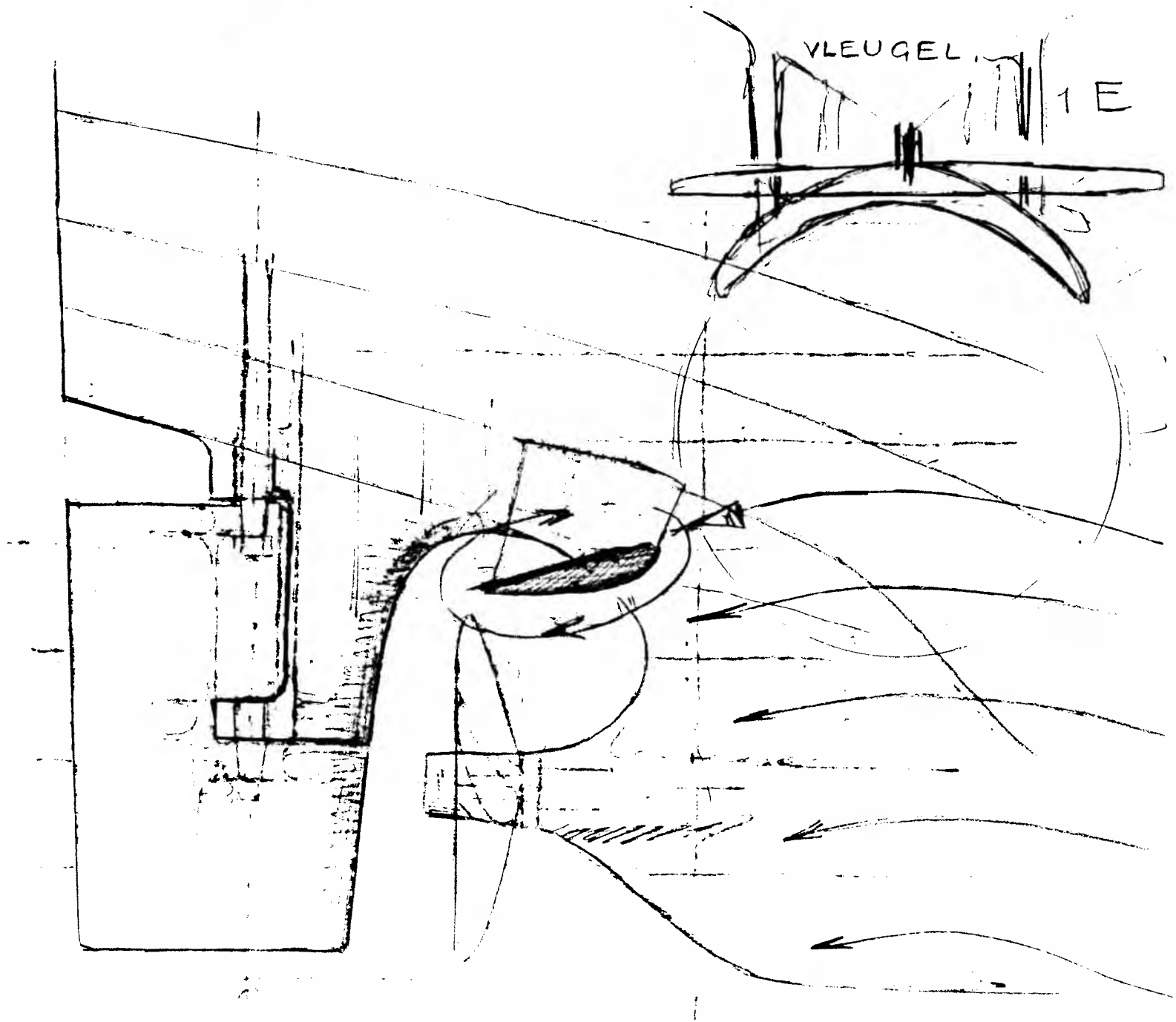
1 C.



BASISVORM.
SIMPLEX ROER + HAK
+ STRAALLUIS.

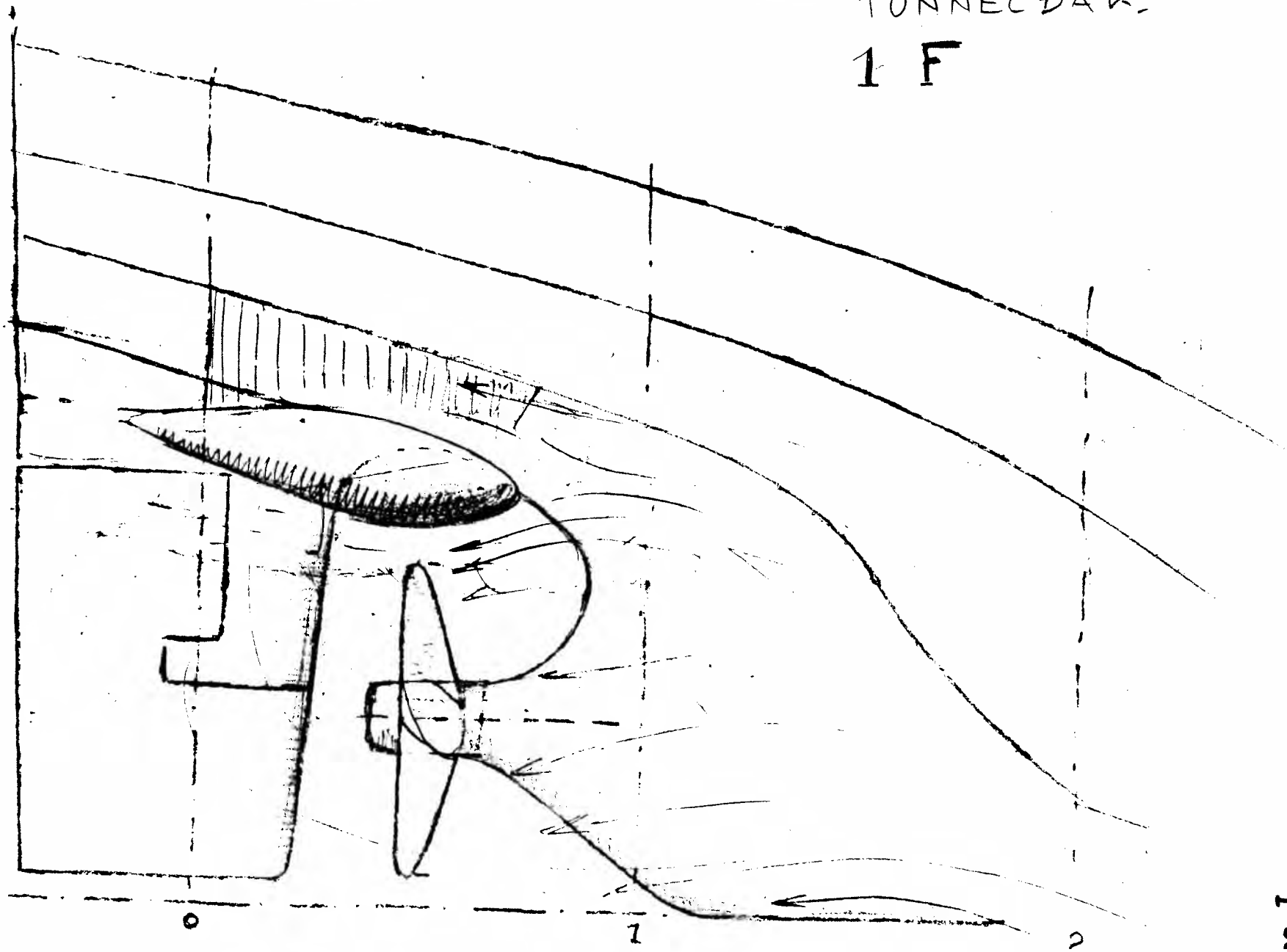
1 D.




VLEUGEL
1 E

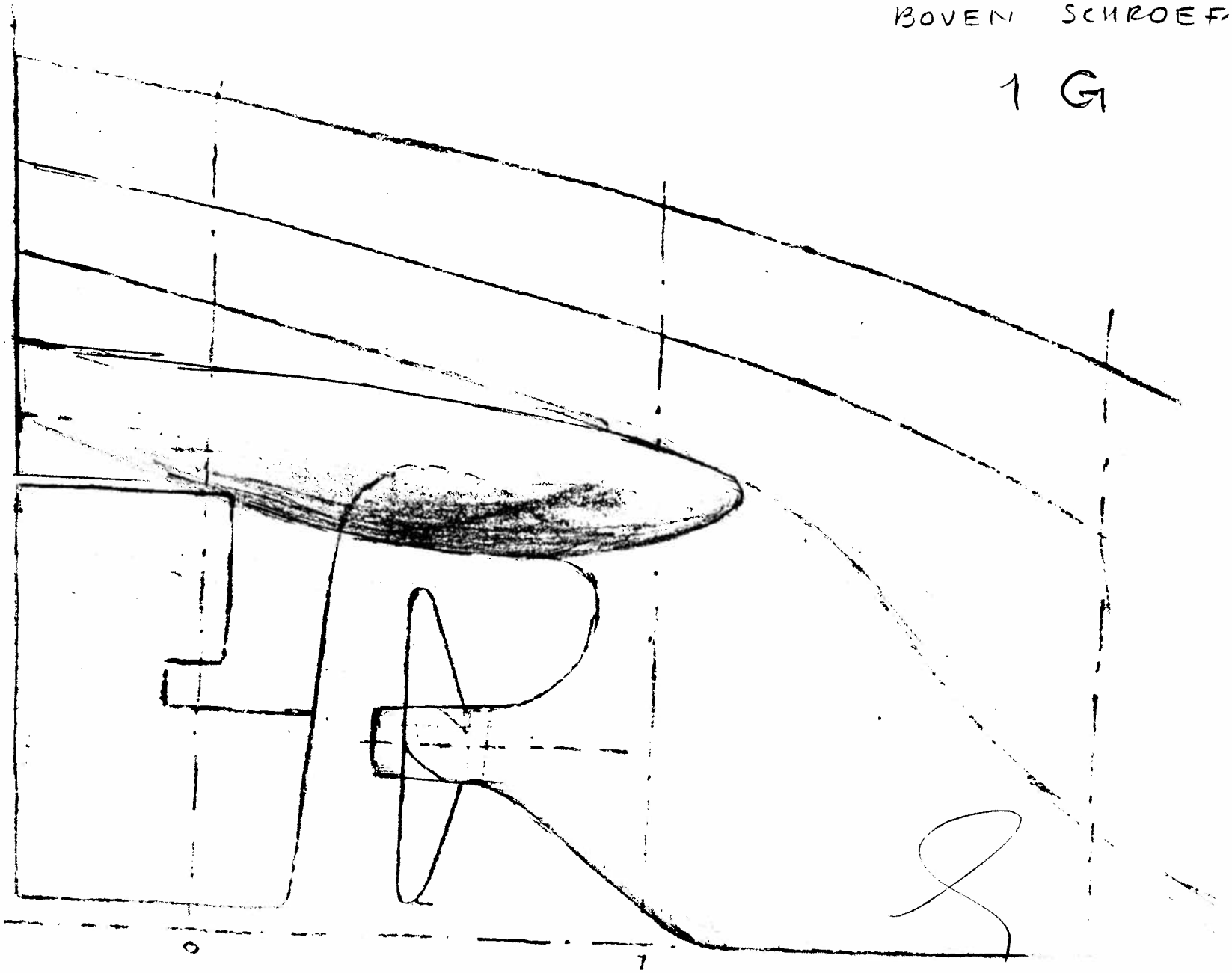
BASISVORM.
"TUNNELDAK."

1 F



LICHAAM.
BOVEN SCHROEF.

1 G



IHI

Vol. 16 No. 182 May, 1982 Ishikawajima-Harima Heavy Industries Co., Ltd., Tokyo, Japan

Additional Thrusting Fin developed for energy saving of ship's propulsion

The first A.T. Fin (additional thrusting fin) will be fixed on a VLCC keel-laid soon at IHI Kure Shipyard for Daikyo Tanker Co., Ltd., Tokyo. This newly developed A.T. Fin aims at producing additional thrust for propulsion from the energy of rotational flow by the propeller race. The rotational energy used to be wasted. A.T. Fin has shown 4-5% more recovery of energy involved in rotational flow, which has been lost so far.

While the screw propeller is widely used for ship's propulsion, which is an efficient application of the principle of lift, a certain amount of energy for rotating the screw propeller must be wasted in the propeller race as rotational flow; the screw propeller is always designed from the viewpoint to minimize the rotational energy, and yet the propeller rotation itself inevitably raises such a loss.

Loss ratio reaches 10-14% of the total energy when the screw propeller is rotating in the open water, but in fact about a half of such a lost energy is collected by the rudder in the propeller race. But some amount of energy is still wasted.

IHI A.T. Fin, consisted of two fins fixed horizontally on the rudder horn (support of rudder), can recover as much as 4-5% more of the wasted energy. This fact has been confirmed by the following tests and measurements as well as by theoretical calculations besides the resistance and self-propulsion tests that also have been performed at IHI Ship Model Basin in Yokohama.

A.T. Fin fixed on a rudder of scale model

- **Test and measurements**

1. Load worked on fins in propeller ahead and astern
2. Normal force and torque of rudder
3. Cavitation tests
4. Hydrodynamic force worked on fins by ship's motion in wave
5. Impact pressure if fins retouch the water surface after occasional exposure

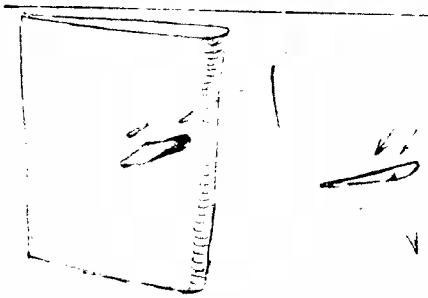
- **Calculations**

1. FEM three dimentional strength analysis for fins, rudder and stern
2. FEM vibration analysis of submerged fins, rudder and stern
3. Computer simulation of ship's maneuverability

IHI A.T. Fin has the following excellent features that make the clients easily estimate its cost-effectiveness:

1. Easy estimate of propulsive efficiency and power saving improvement because of fitting on the rudder horn in propeller race (other similar device arranged in complex flow before the screw propeller)
2. Sure estimate on its effect by the established theoretical propeller calculation in non-uniform flow
3. Simple structure
4. Feasible even for ship in service after study about their rudders

Patent of A.T. Fin is pending now and its first fitting on 238,-400 DWT VLCC driven by 24,000 PS diesel will bring in Daikyo Tanker about 40 million yen annual savings (about 3 tons of daily fuel saving).



(WAGNER FINS ~ 1907
STAR CONTRA ~ 1920
(OLD IDEA

2

ON THE ADJOINING PAGES :-

EXAMPLES OF FULL-BODY SHIPS

M.P. CARGO "GERMAN LIBERTY"			NEDLL. NIGER/NILE	VIBR.
PROD. TANKER	BOEL	30.000 DWT	MAASKERK	SOME VIBR.
PROD. TANKER	"	38.000 DWT	PROJECT	
PROD. TANKER	RSV	38.000 DWT	PROJECT	
PROD. TANKER	V.D.G	38.000 DWT	MAASSLUIS	O.K.
BULKCARRIER	VEROLM	45.000 DWT	MARGOT JACOB	O.K.
BULKCARRIER	GDYNIA	50.000 DWT	AMSTELMEER	VIBR.
VLCC JAPAN	IHI	270.000 DWT	MAASBREE	O.K.
VLCC	BOTLEK RSV	318.000 DWT	MAASBRACHT	O.K.
LPG	LA SEYNE CNIM	52.000 M^3	ANTILLA BAY	VIBR!
LNG	ST NAZ. CH. L'ATL.	120.000 M^3	GASTOR	?

↓

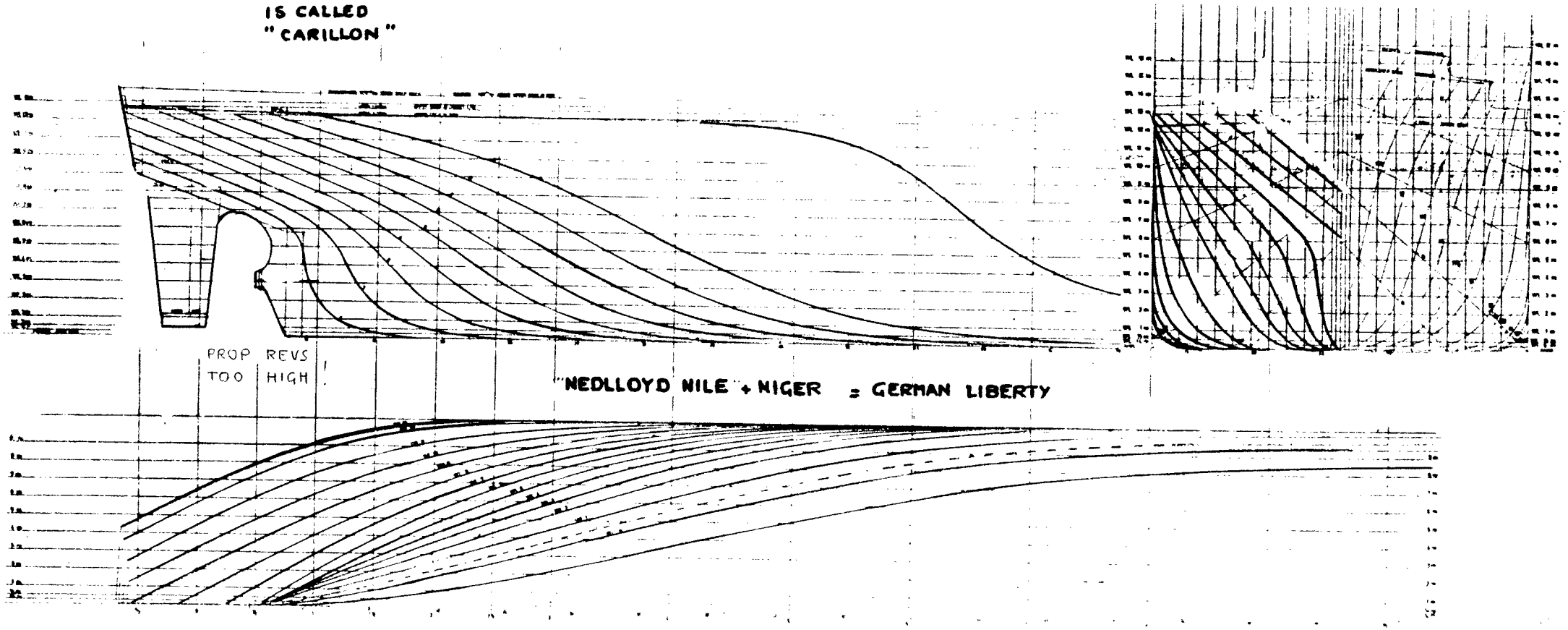
ARE THOSE SHIPS

ACCEPTABLE AS TO VIBRATIONS ? ↑

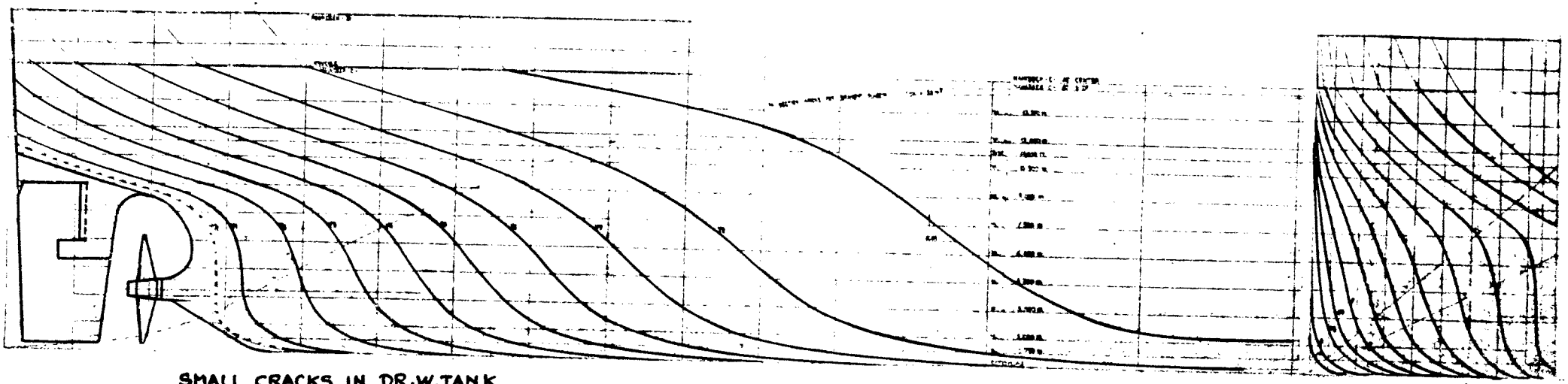
COMPARISON OF AFTERBODIES

LIMITS : HOW FAR CAN WE INCREASE <u>FULLNESS</u> OF AFTERBODY ?

RATTLING AFTERBODY
HIGH RPM OF PROPELLER
MESSROOM
IS CALLED
"CARILLON"



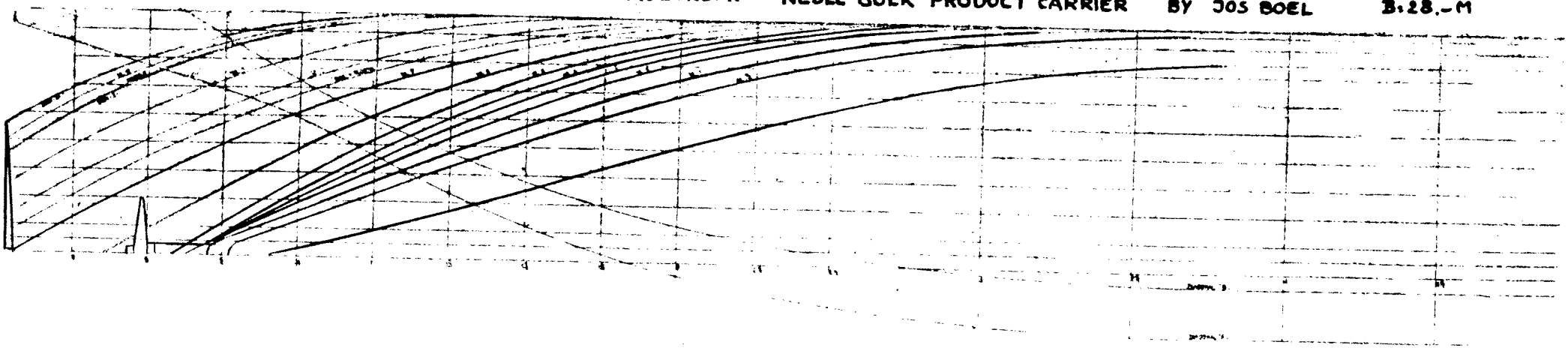
SOME RATTLING



SMALL CRACKS IN DR.W.TANK

EXTREMELY U SHAPE ↑

"Maaskerk" NEDLL BULK PRODUCT CARRIER BY JOS BOEL B:28.-M



BOELWERF

"PANMAX" PROD.TANKER PROJECT

L_{ll} = 168,00 m

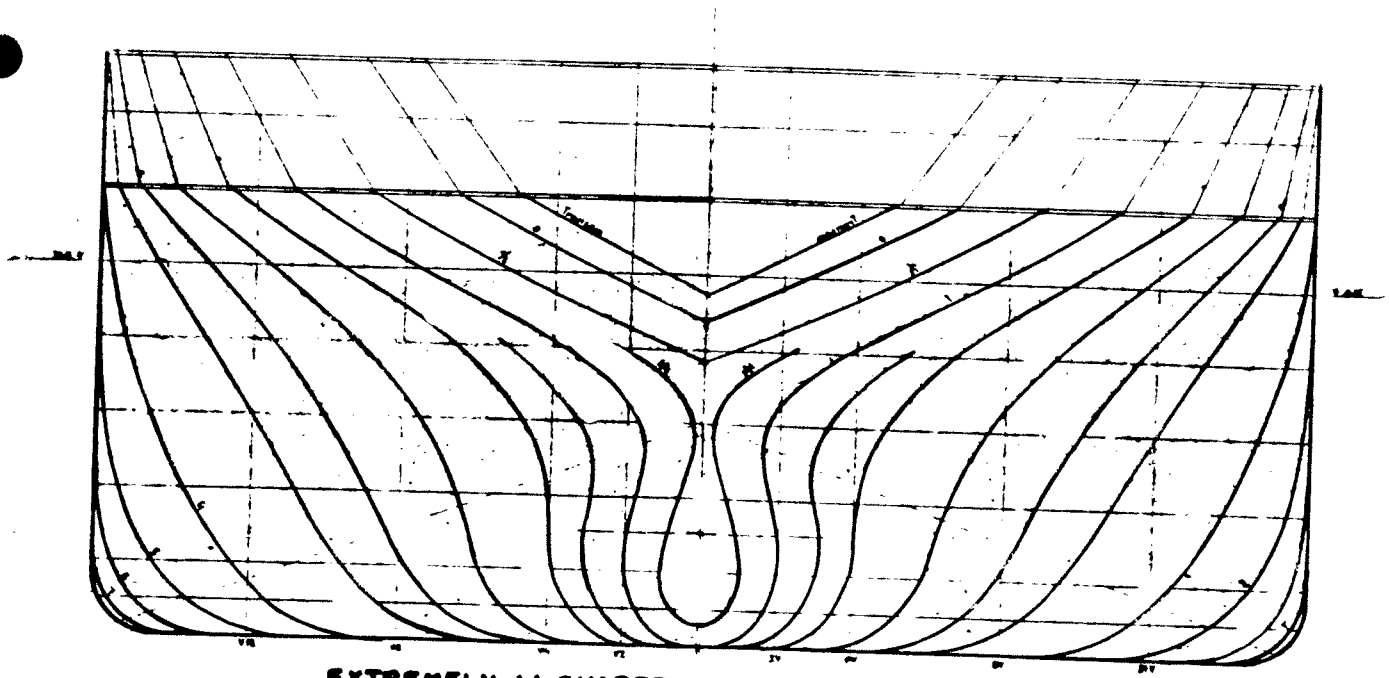
B_{mld} = 32,25 m

T = 10 m

C_b = .8092

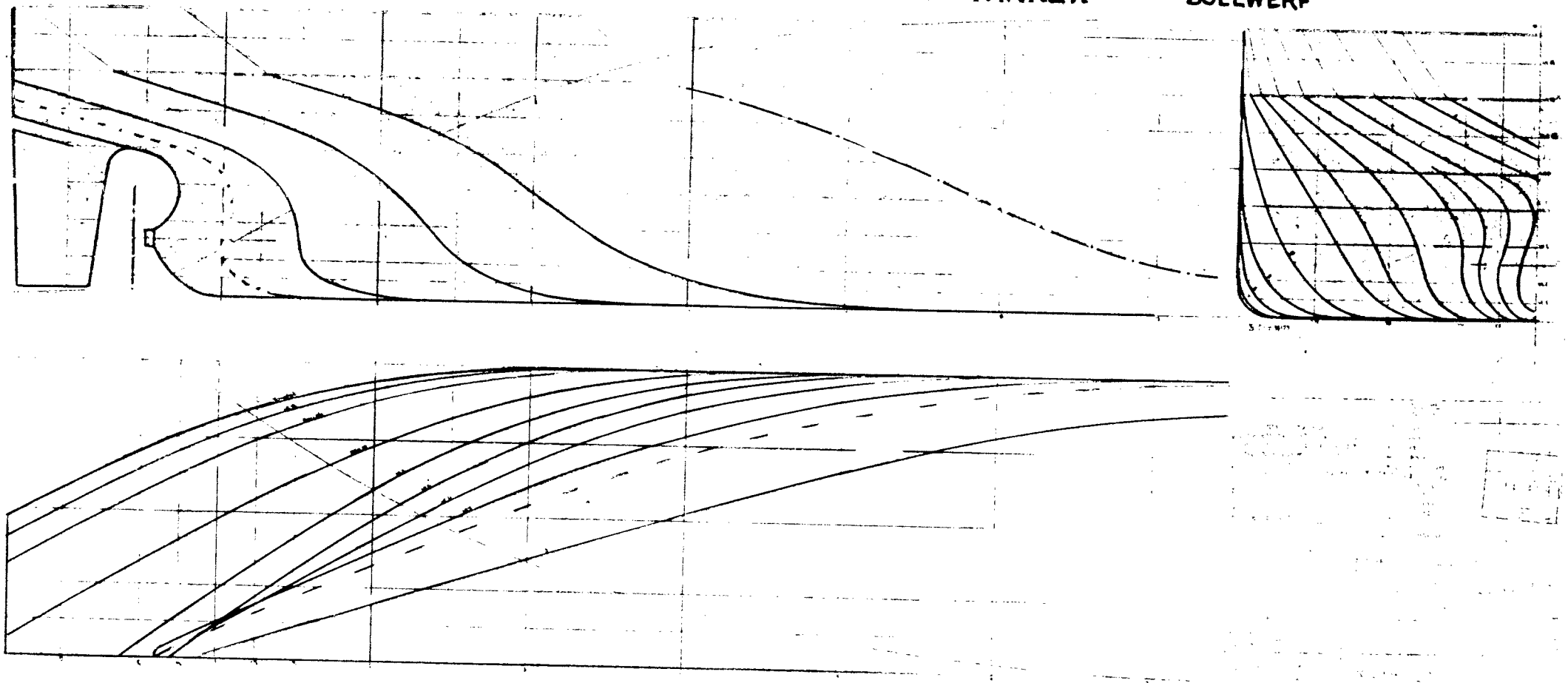
L/B = 5,209

B/T = 3,225



EXTREMELY U SHAPED

PROJECT "PANMAX" PROD. - TANKER BOELWERF



RSV

PANMAX PROD. TANKER PROJECT

L_{11} = 180,00 m

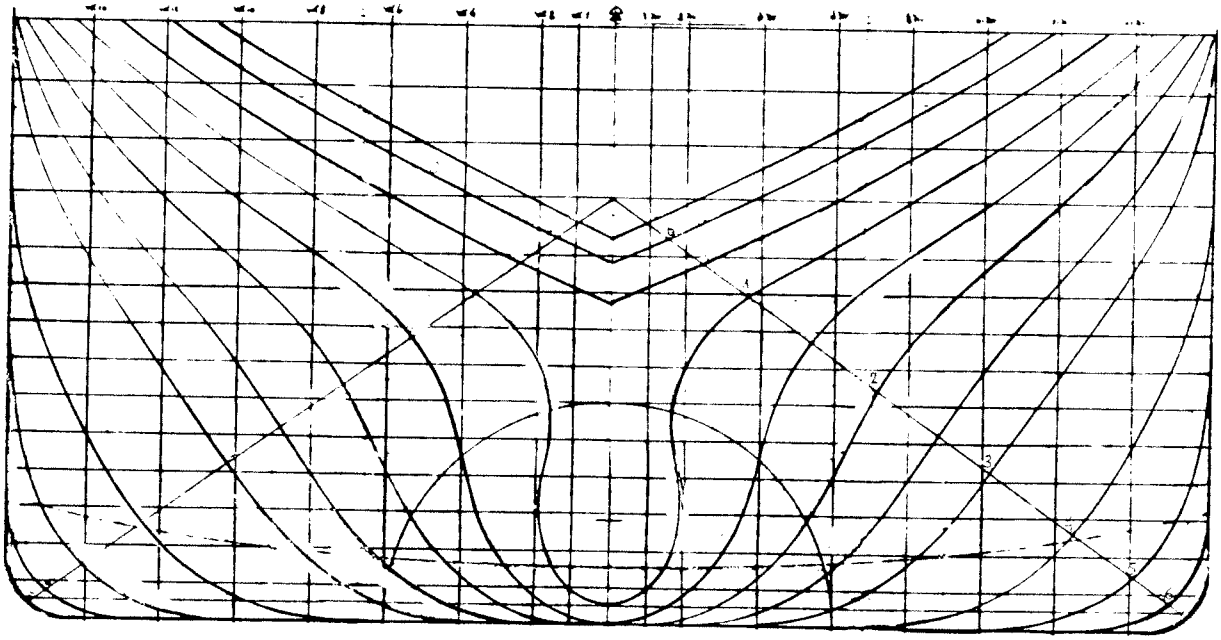
B = 32,25 m

T = 11,56 m

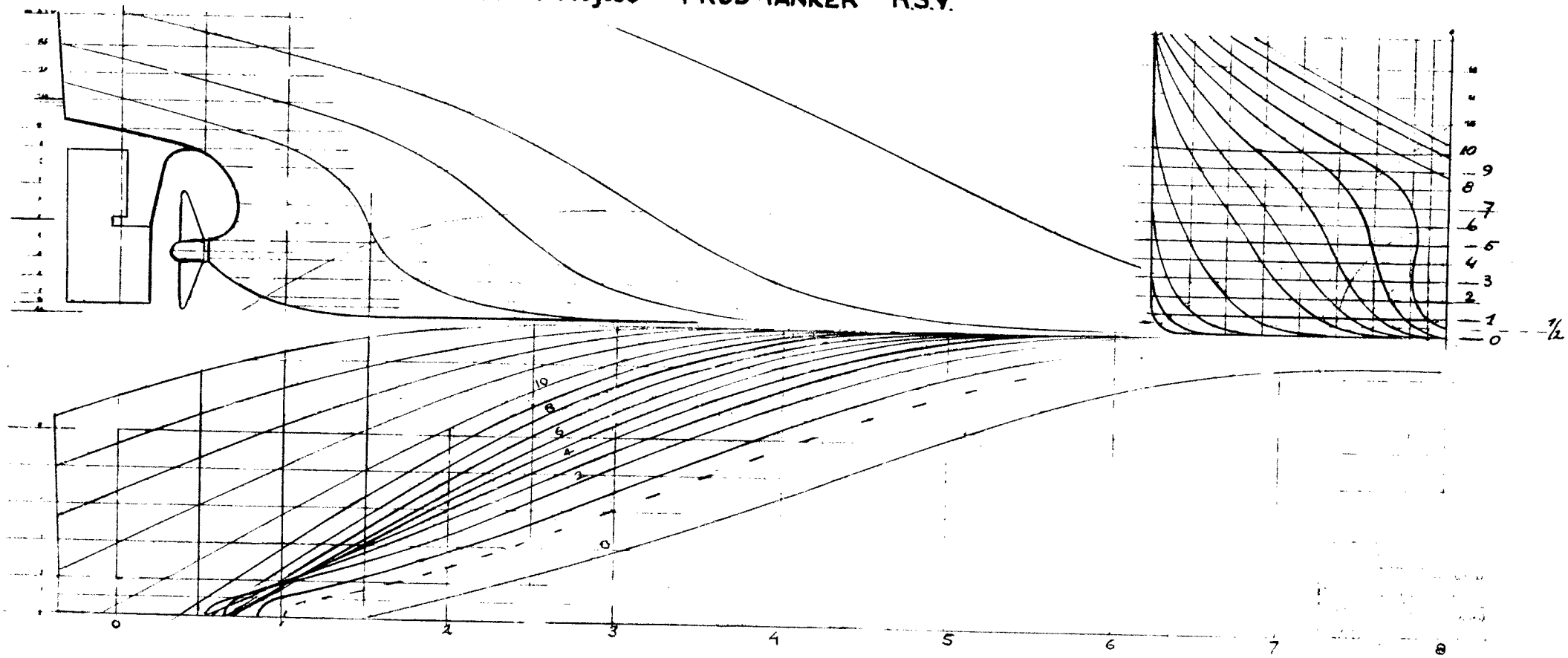
Cb =

L/B = 5,58

B/T = 2,79



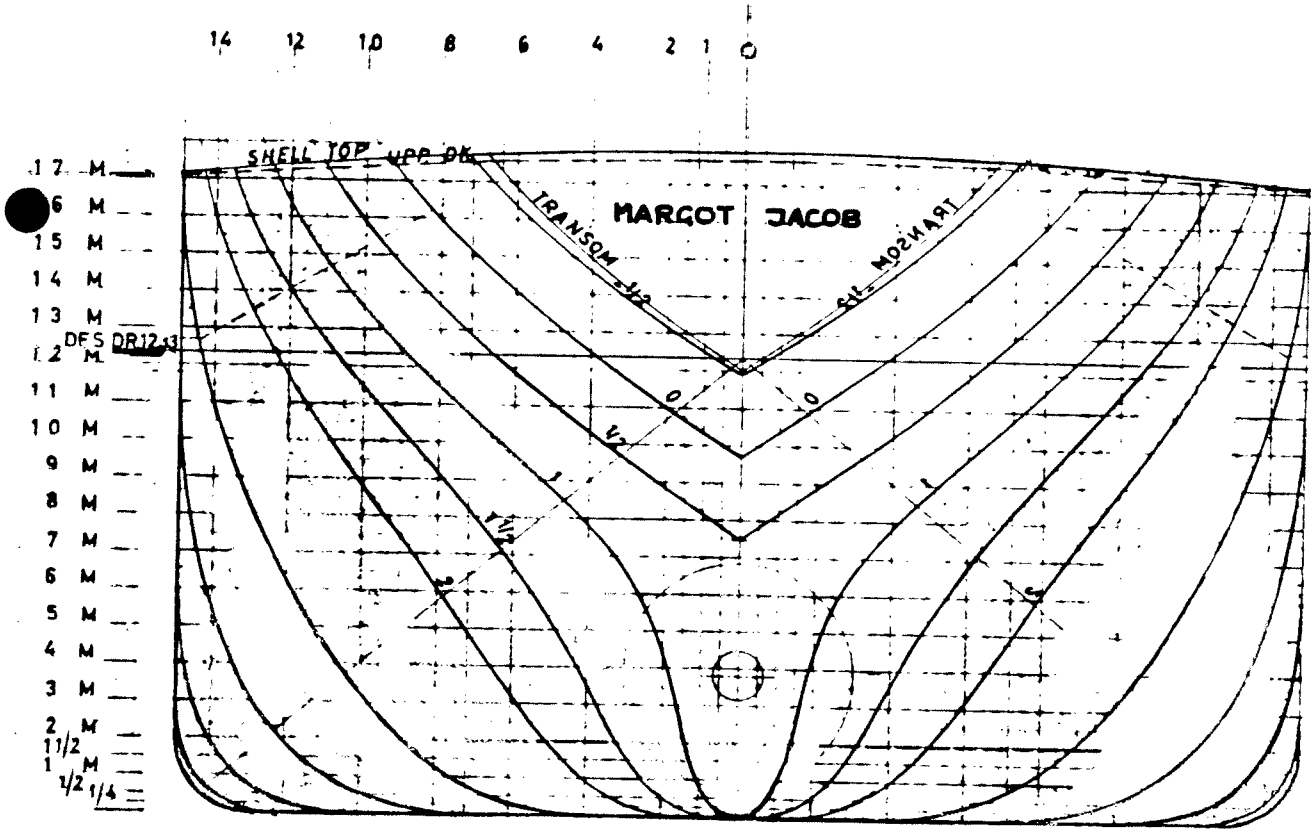
"PANMAX" Project PROD-TANKER R.S.V.



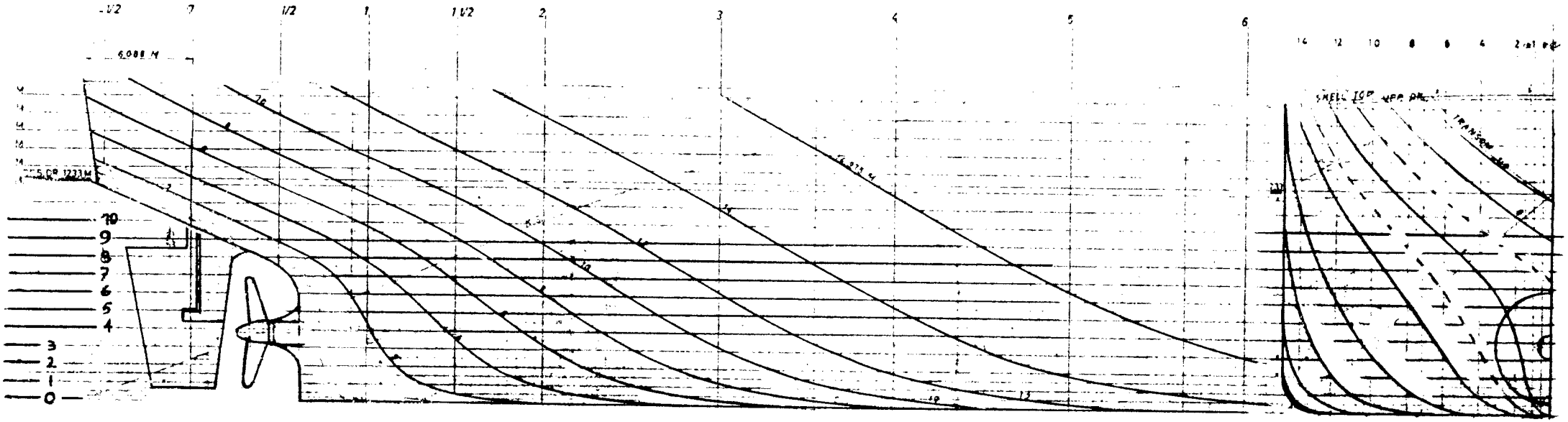
VEROLME CORK

BULKCARRIER

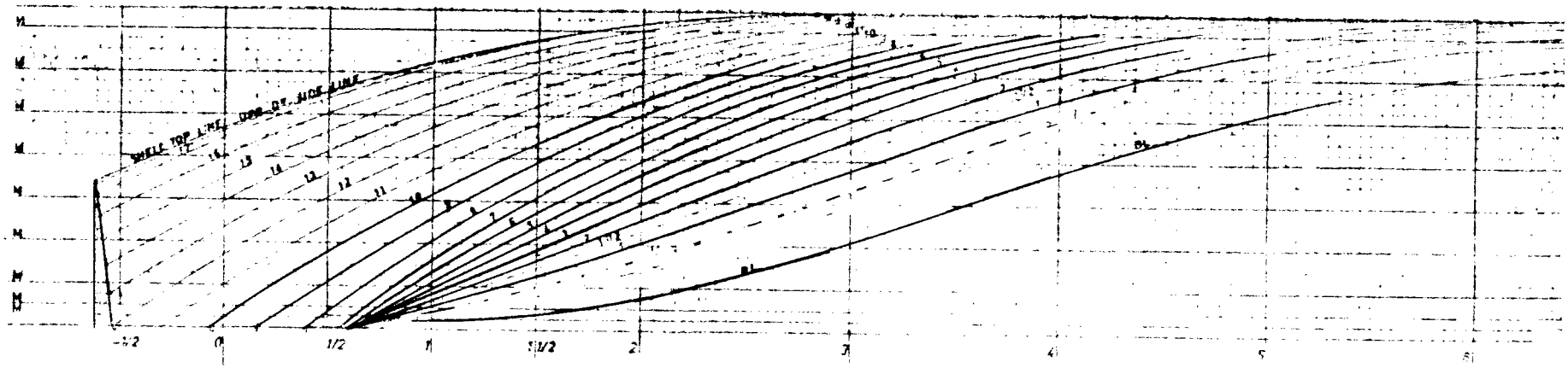
L_{11}	=	194,62	m	L/B	= 6,560
B_{mld}	=	29,67	m	B/T	= 2,406
T	=	12,33	m		
C_b	=	.822			



BULKCARRIER VEROLME CORK "MargotJacob"



SEEMS TO SAIL O.K.



DIMENSIONS:

LENGTH O.A.	ABT.	205.00 M
LENGTH B.P.		194.20 M
BREADTH MLD.		28.70 M
DEPTH TO UPPERDECK		17.70 M
DESIGN DRAFT		12.33 0 M
SUMMER DRAFT		12.69 M
MOULDED VOLUME		50926 M^3
Cb.		0.821502
Cp.		0.829502
Cw.		0.899500
Cm.		0.995200

STATION SPACING - - - 9.7.31 M

LINES
VEROLME CORK
COBH.
VEROLME UNITED SHIPY.

YARD Nº 838

2-5

AMSTELMEER/AMSTELMOLEN

50.000 DWT PANMAX BUKCARRIER - GDYNIA

L_{ll} = 205,50 m

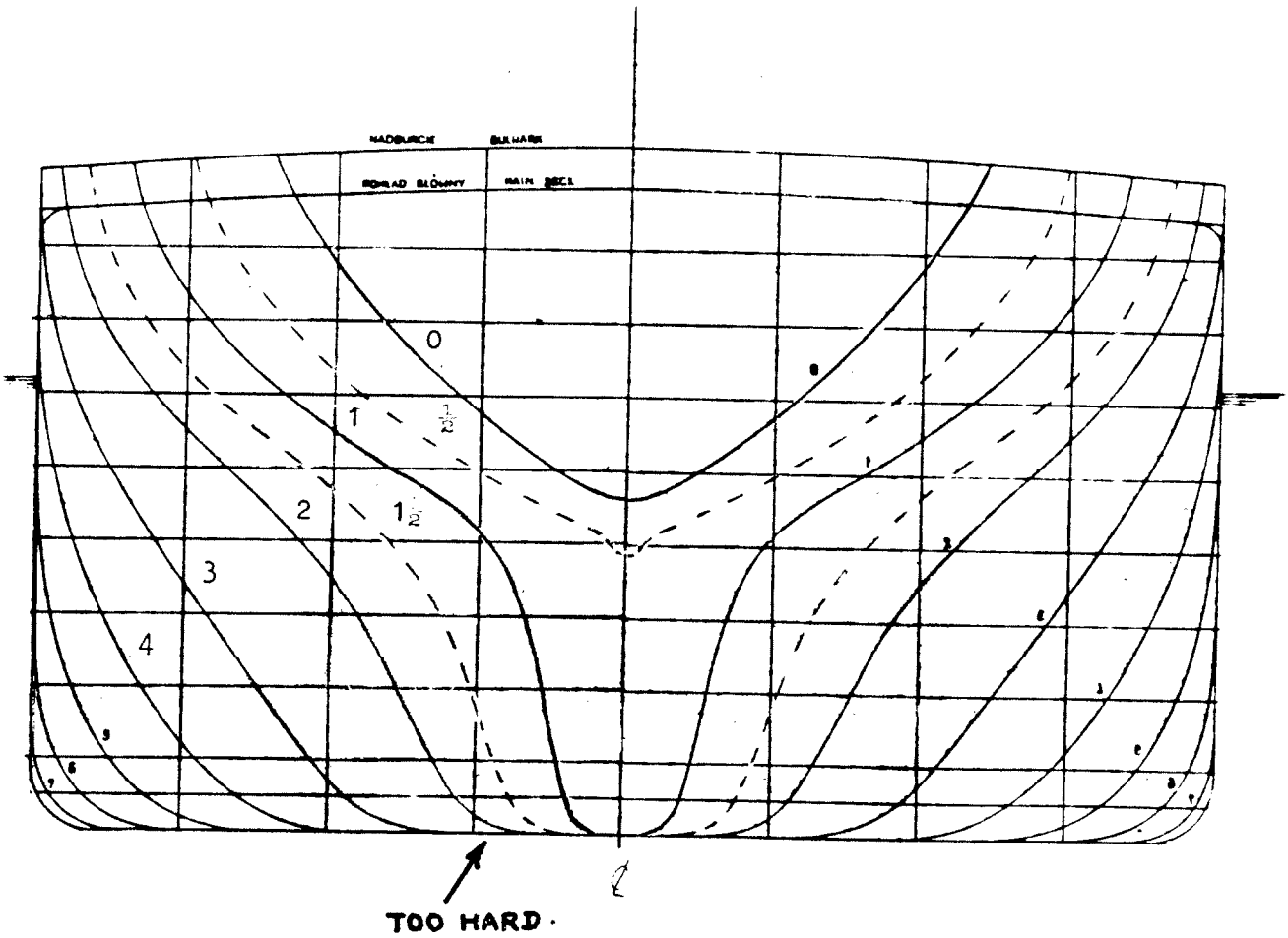
B_{mld} = 32,20 m

T = 12,40 m

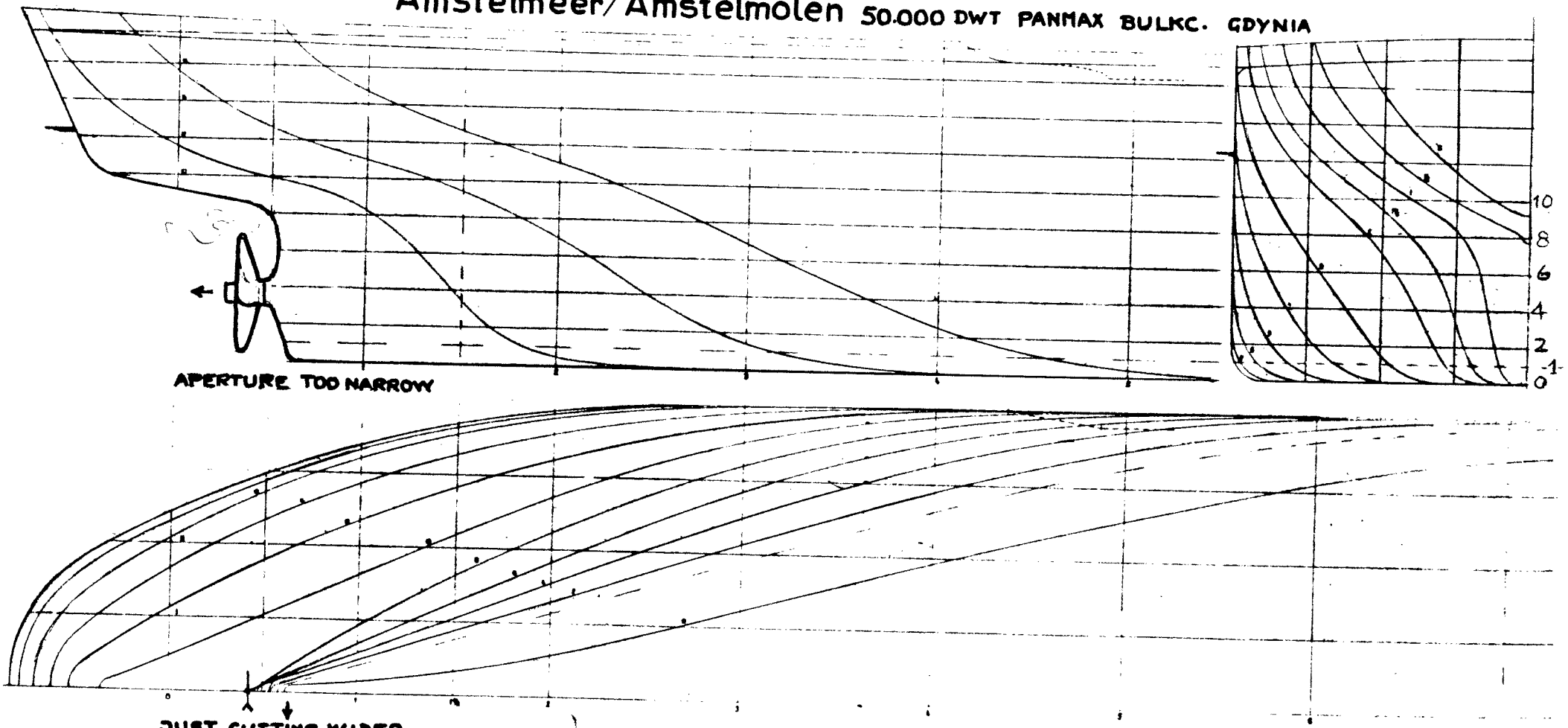
C_b = .820

L/B = 6,382

B/T = 2,597



Amstelmeer/Amstelmolen 50.000 DWT PANMAX BULKC. GDYNIA

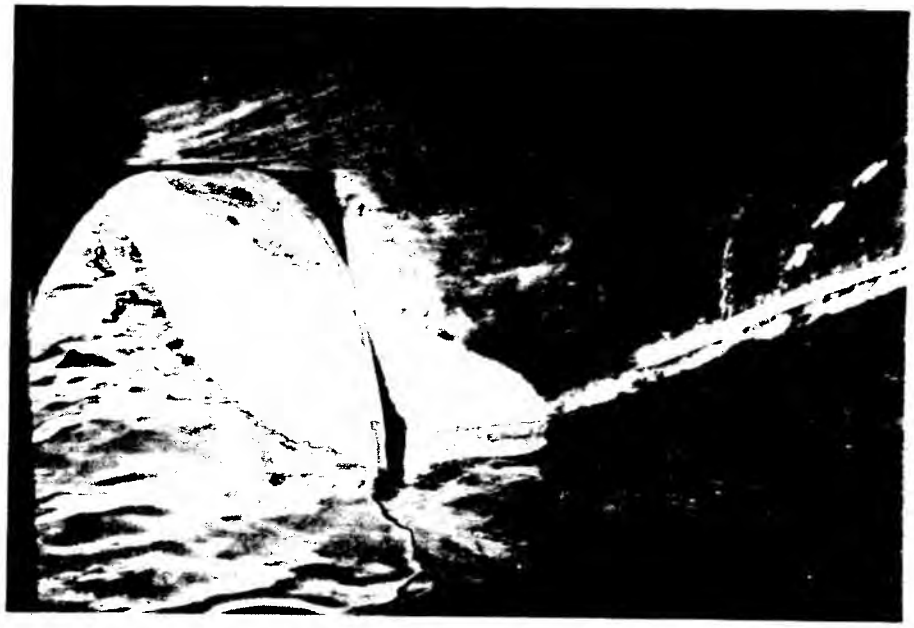


JUST CUTTING WIDER
AND ROUNDING UP
DOES NOT IMPROVE SITUATION!

RATTLING

HULLFORM GENERALLY VERY GOOD
HOWEVER: PROPELLER AND RUDDER SHOULD BE 2½ M (MORE) AFT
TOTAL L.O.A IS ADEQUATE

48



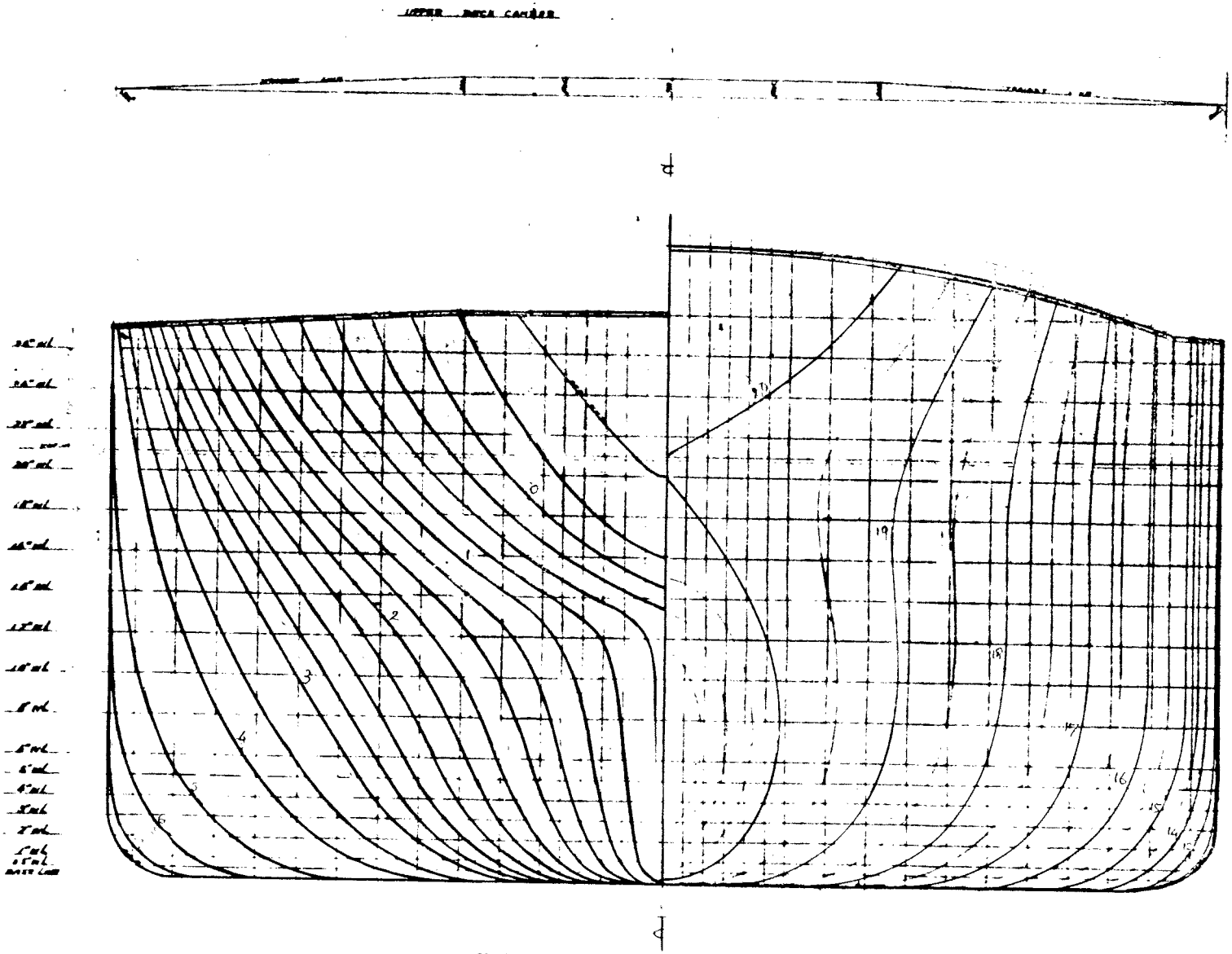
Typisch voorbeeld van een te dikke achtersteven bij een standaard schip.
Panamax bulkcarrier AMSTELMEER: 1976 (!).

Typical example of a too blunt stern of a so-called
off-the-shelve standard vessel ("much deadweight within short length").

PROPELLER AND RUDDER SHOULD BE MOVED MORE AFT ~ 2½ M
HULL PLATING TO BE EXTENDED AFT TO A FINE ENDING APERTURE

270.000 DWT VLCC I·H·I

ACCEPTABLE AT 14 KN AND DERATED SHP

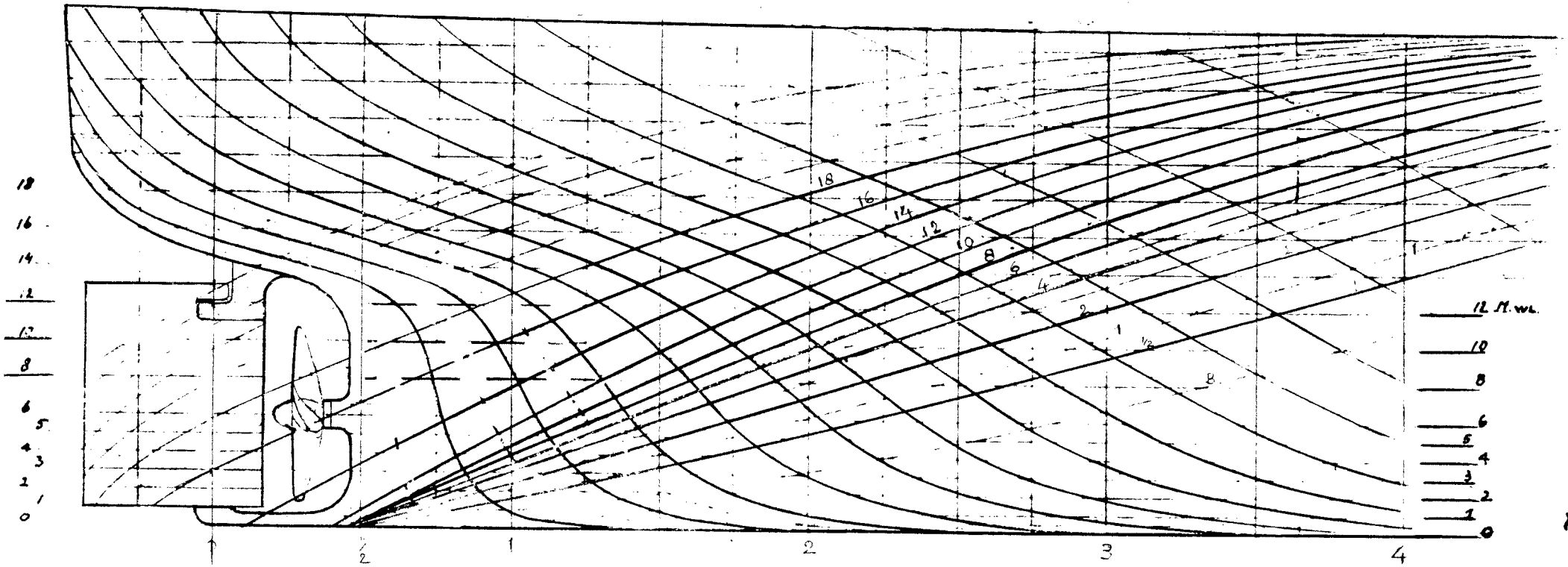


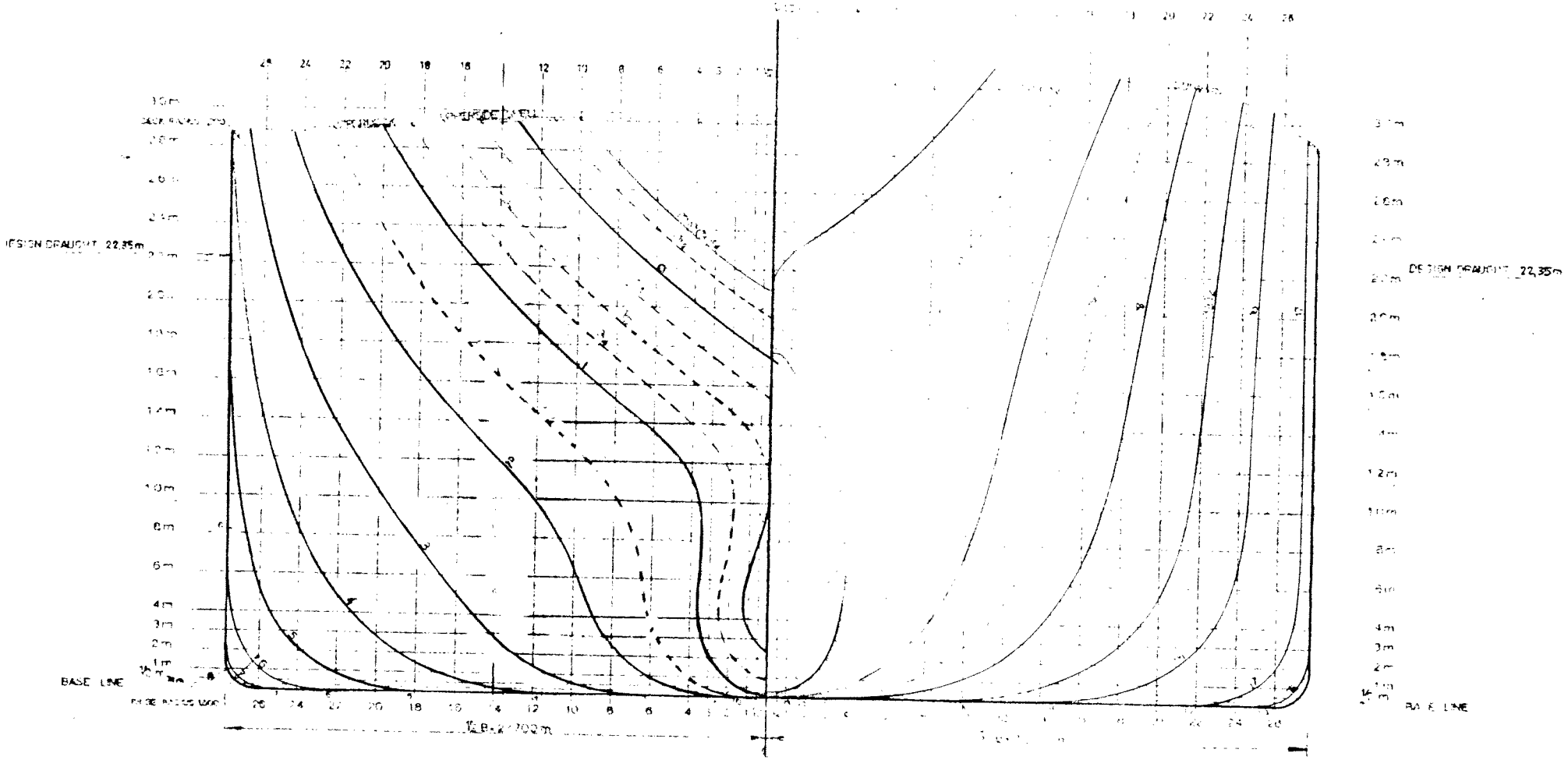
"MAAS BREE"

270.000 DWT VLCC IHI

"MAASBREE" ex "SINDE"

GOOD VESSEL



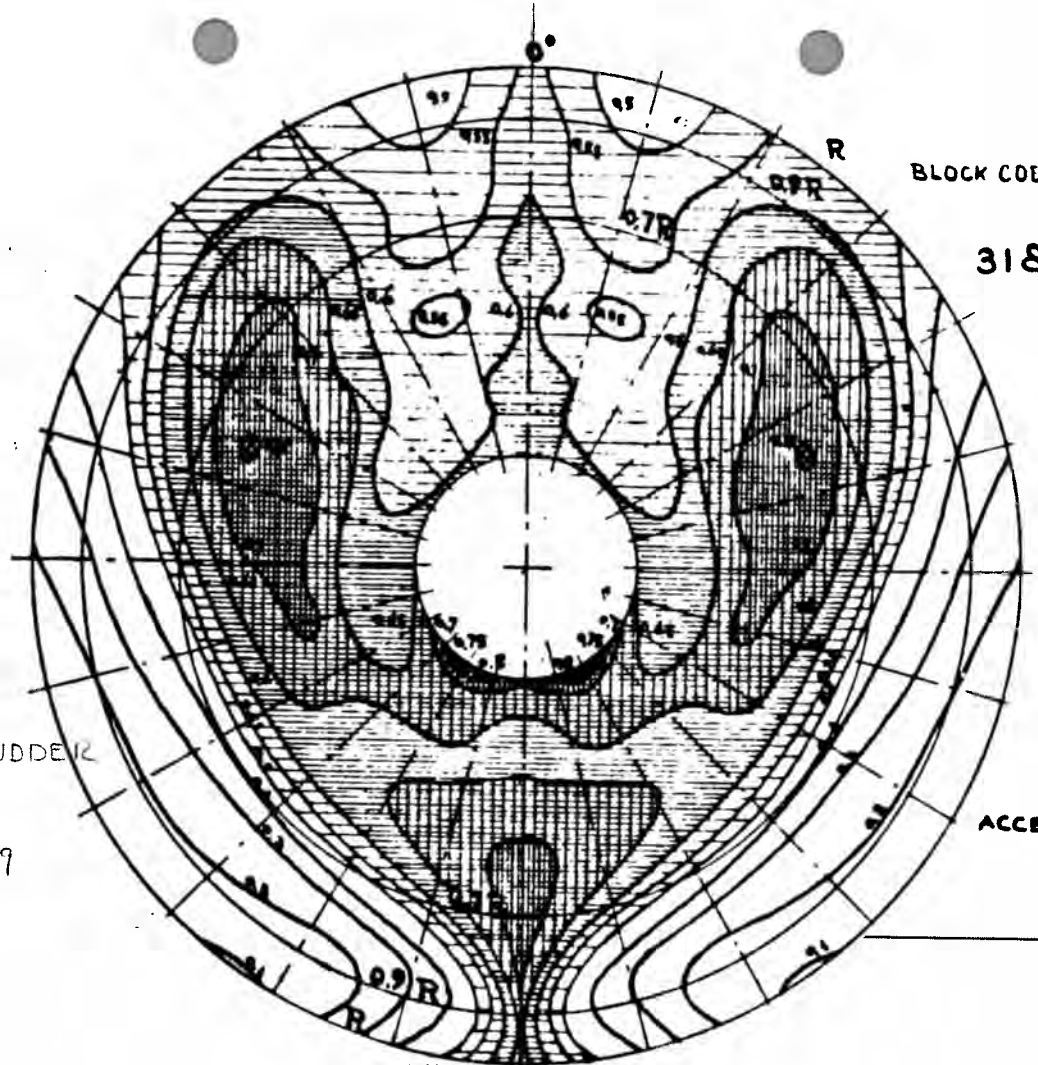


ACCEPTABLE AT 14 KN AND DERATED SHP

LENGTH OVERALL	352.091 m
LENGTH BP	336.000
BREADTH MOULDED	55.400
DEPTH (MOULDED	28.550
DESIGN DRAUGHT MOULDED	22.350
C_b	0.8398
C_{WL}	0.9091
$C_{\otimes}$	0.9991
C_p	0.8406
LCB	177.821 m FORWARD OF A.P.P.

318.000 DWT VLCC RSV BOTLEK

"MAASBRACHT"



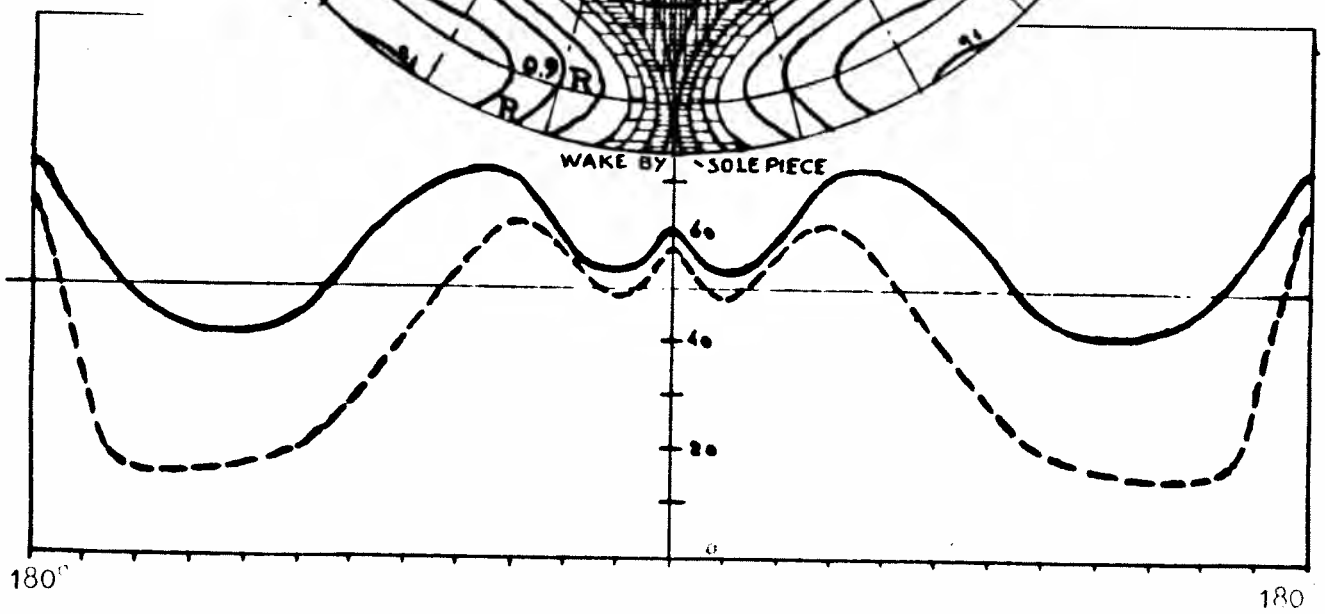
BLOCK COEFF. = 0.84

318.000 DWT "MAASBRACHT"
VLCC

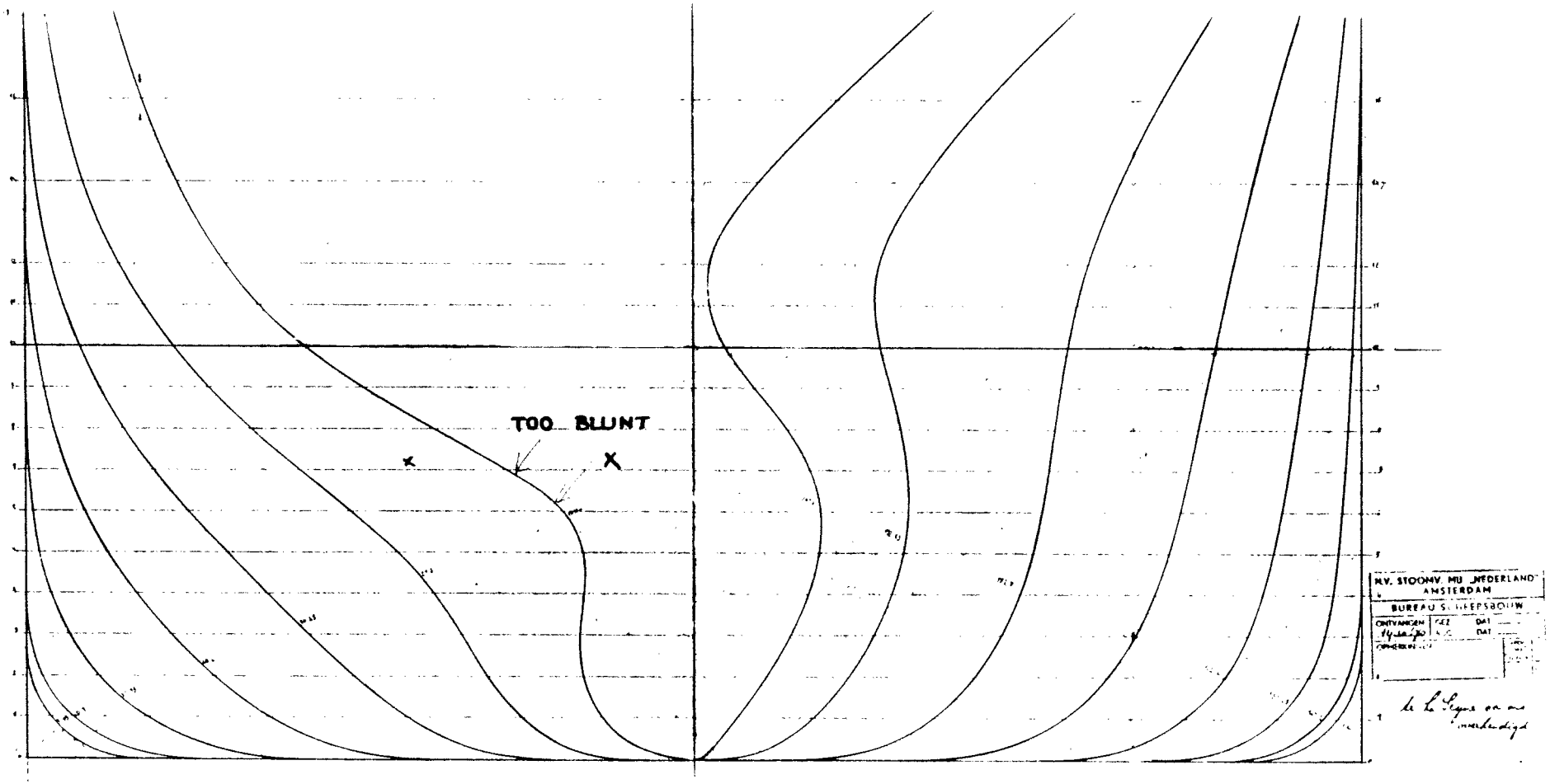
STERNBULB OF ESSO
+ SOLE PIECE AND BAL. RUDDER
SHELL

SEE PAGE 4-9

ACCEPTABLE AT 14 KN
AND DERATED SHP



52000 M^3 LPG CARRIER LA SEYNE CNIM



RATTLING VESSEL

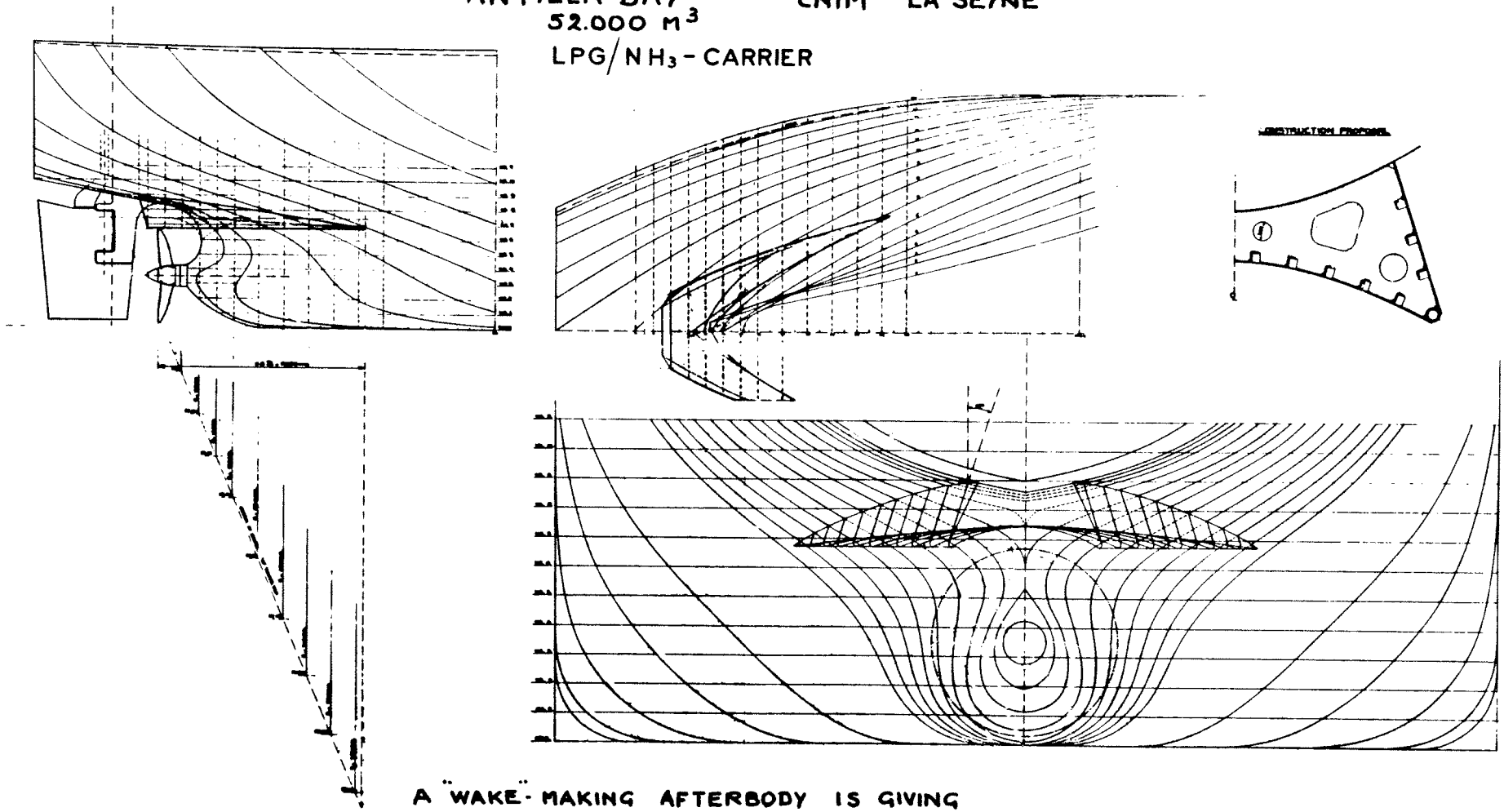
"ANTILLA - BAY"

CONSTRUCTIONS NAVALES & INDUSTRIELLES de la MEDITERRANEE

Travaux de LPG de 52000 m^3 Chons 1396 1397 1398

2-17

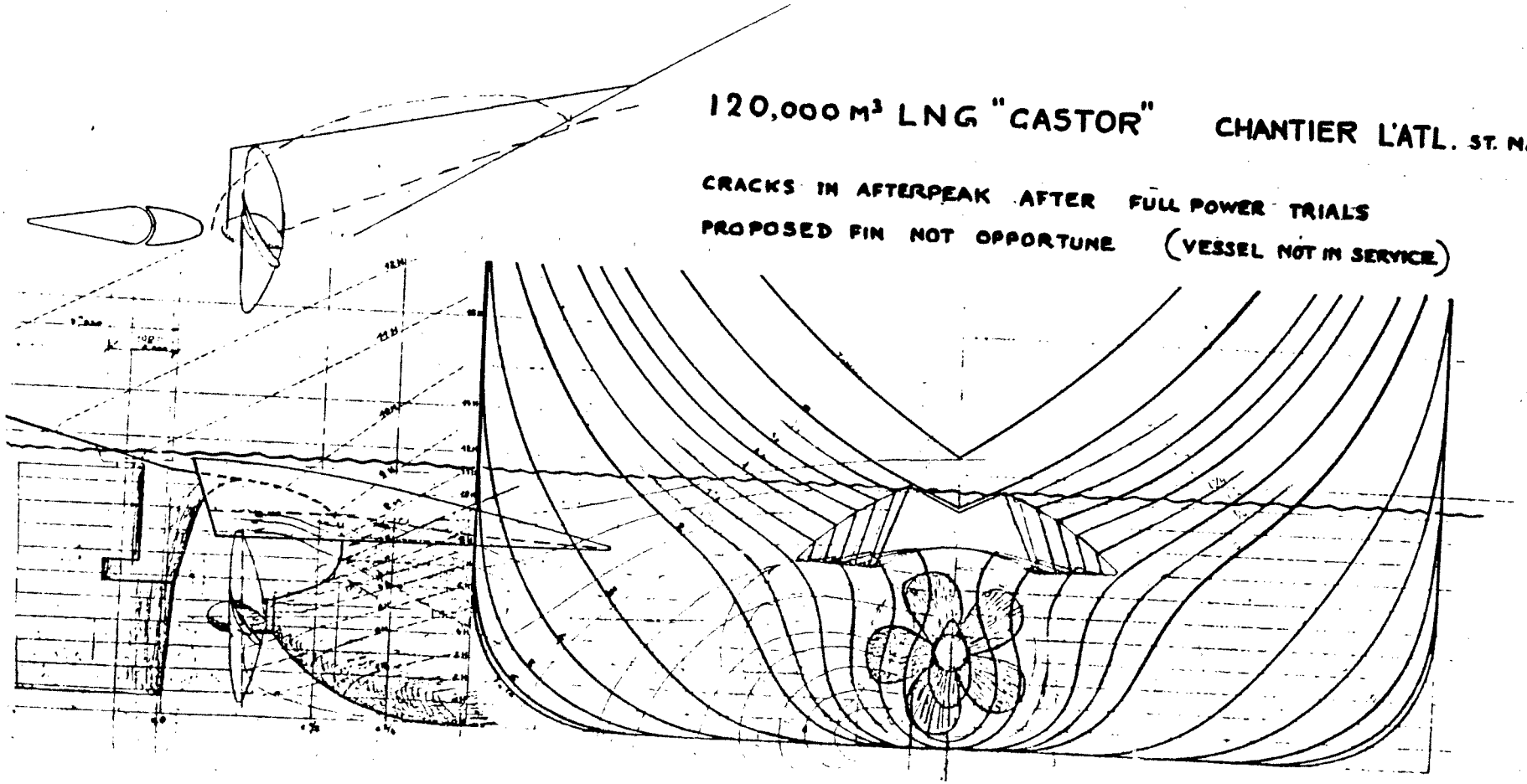
"ANTILLA BAY" CNIM LA SEYNE
52.000 m^3
LPG/NH_3 - CARRIER

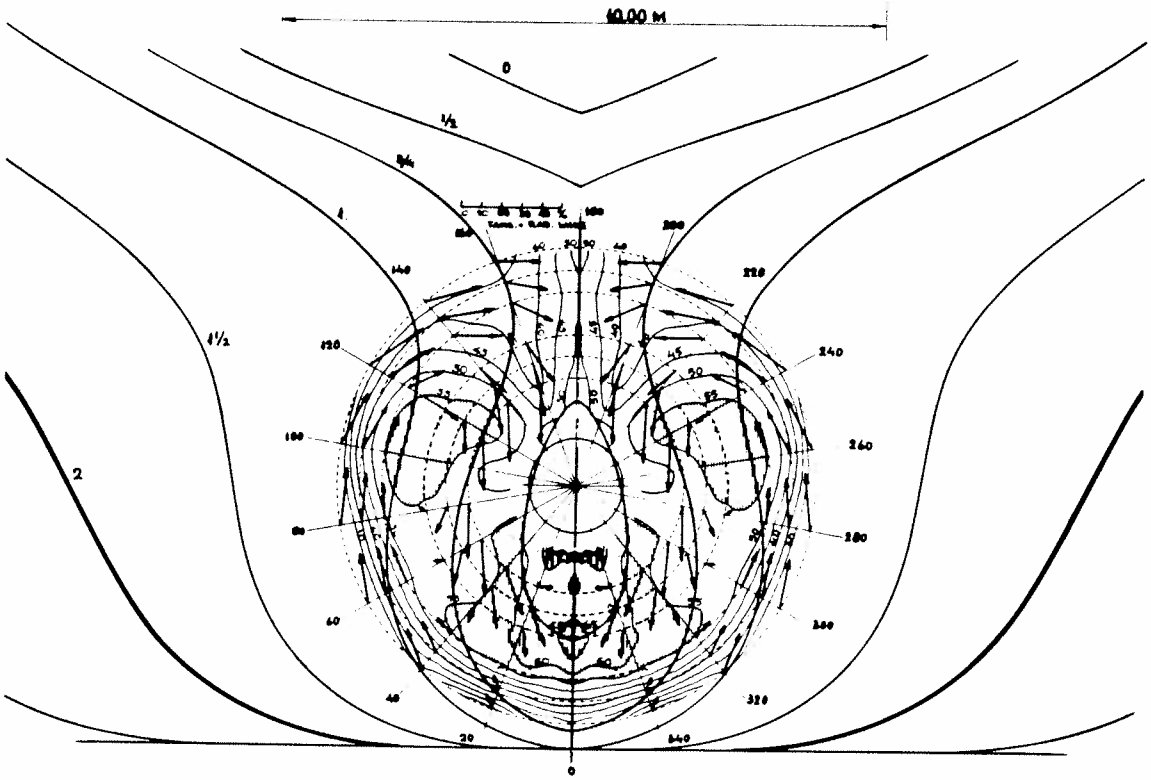


A "WAKE-MAKING" AFTERBODY IS GIVING
A HIGH PROPULSIVE EFFICIENCY (HULL EFF. $\frac{1-t}{1-\psi}$), IN MODEL BASIN AND MISLEADS NAVAL ARCHITECTS
BECAUSE VIBRATIONS CANNOT BE MEASURED.

Typisch voorbeeld van een slecht achterschip : zogenaamde "Wake Maker".
Bolle waterlijnen en schroefsenten naar boveneind van het schroefraam : Sterke Trillingen.
Toepassing van een tunnel zou verbetering kunnen geven, echter de waterlijneinden onder de tunnel blijven bol.

(Typical example of poorly shaped stern,
a so-called 'Wake-Maker'.
Convex waterlines .. strong vibrations.
Application of **FIN** gave no improvements)

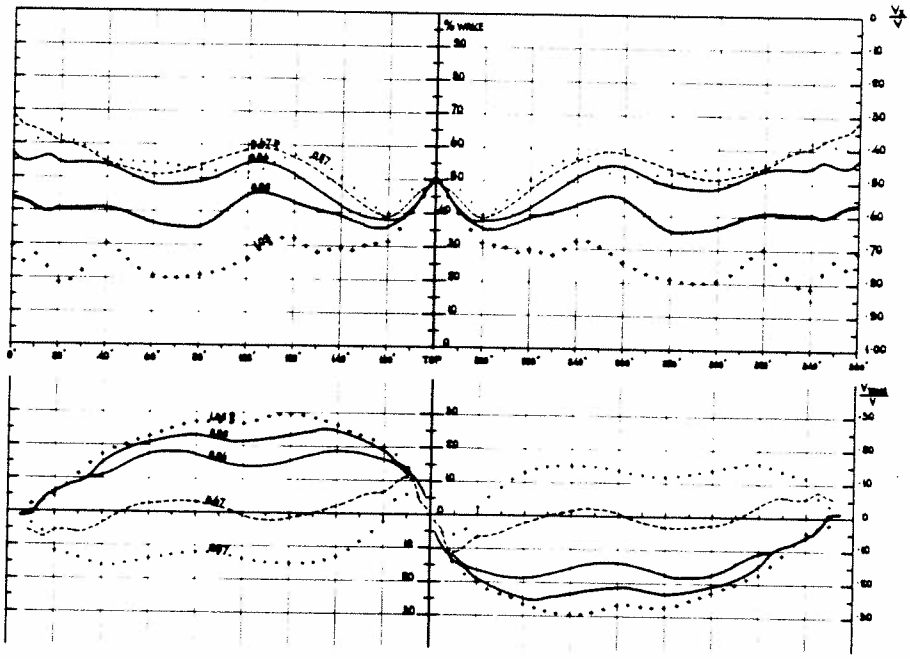

120,000 m³ LNG "CASTOR"
CHANTIER L'ATL. ST. NAZ.
CRACKS IN AFTERPEAK AFTER FULL POWER TRIALS
PROPOSED FIN NOT OPPORTUNE (VESSEL NOT IN SERVICE)



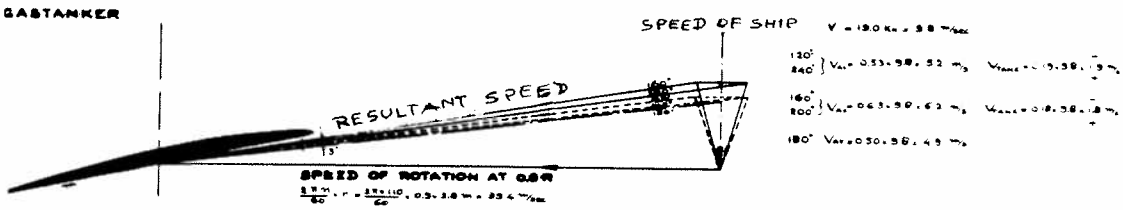
Gastanker 34000 s.h.p. x 110 revs. δ = 0.74

axial wake

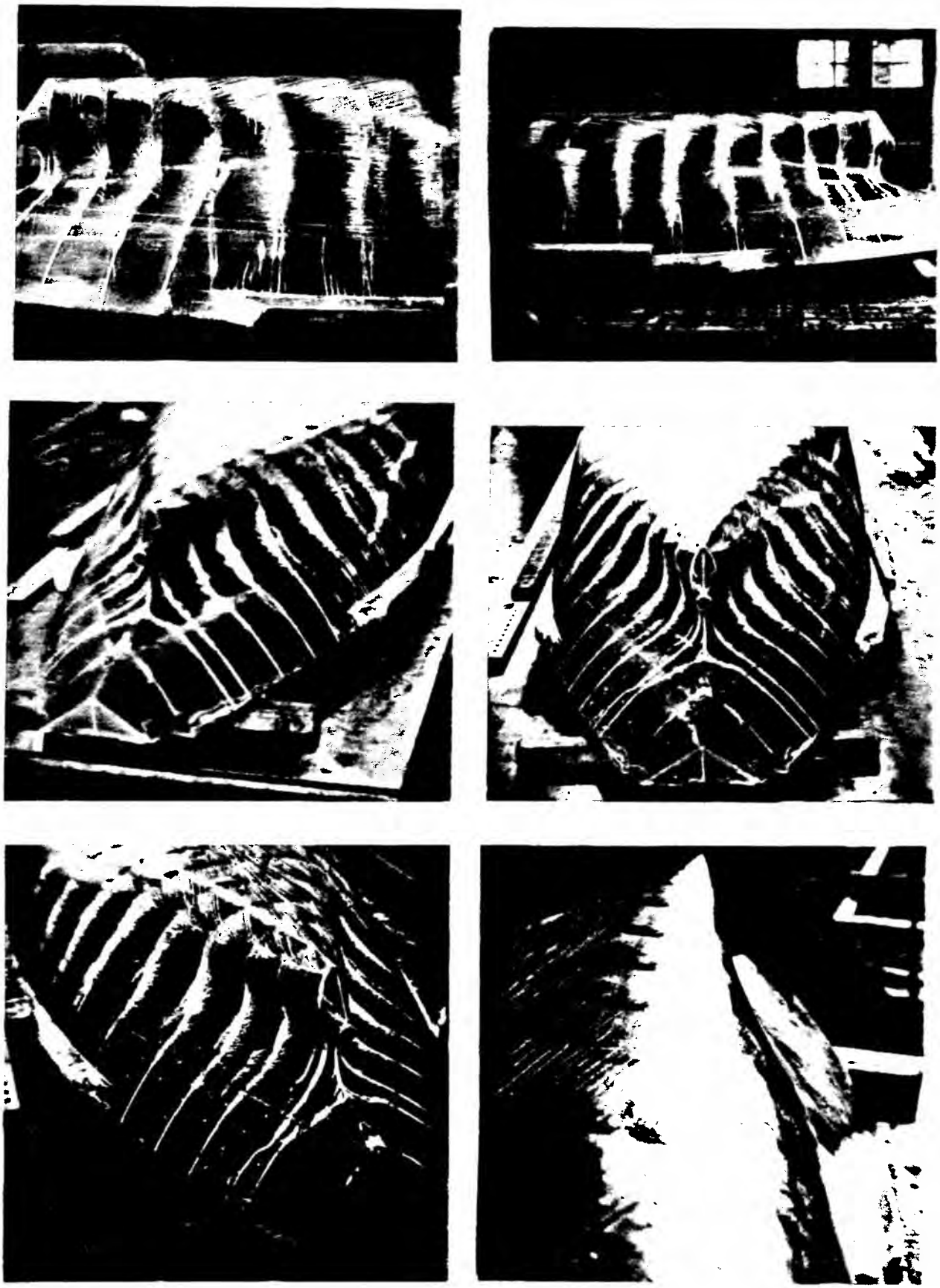
tangential wake



3 dimensional wake-field (5 hole pitot tube) N.S.M.B.



Variations of resultant speed. Small variations of angle of attack (as far as model tests).



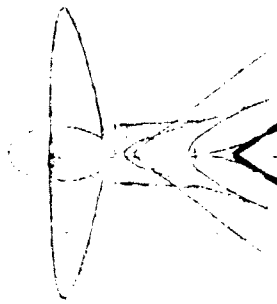
N.S.M.B. Streamline tests on gastanker model. Note separation of flow at bilge roundings.

120.000 M^3 LNG "GASTOR"

L.N.G. CARRIER
"GASTOR"

78600 T. DW.
120.000 M³

34000 SHP
110 RPM



31° ~ ACCEPTABLE

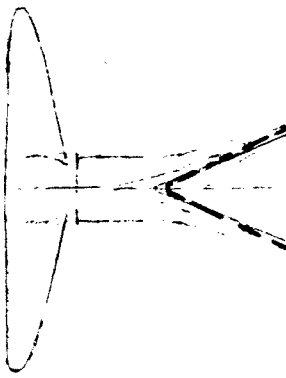
31°

---- WATER LINE
AT TOP OF PROPELLER.

2.9.76
NSU-S. T.A.B.
ROTTERDAM.

V.L.C.C.
"MAASBRACHT."
318.000 T. DW.

36 000 SHP
85 RPM

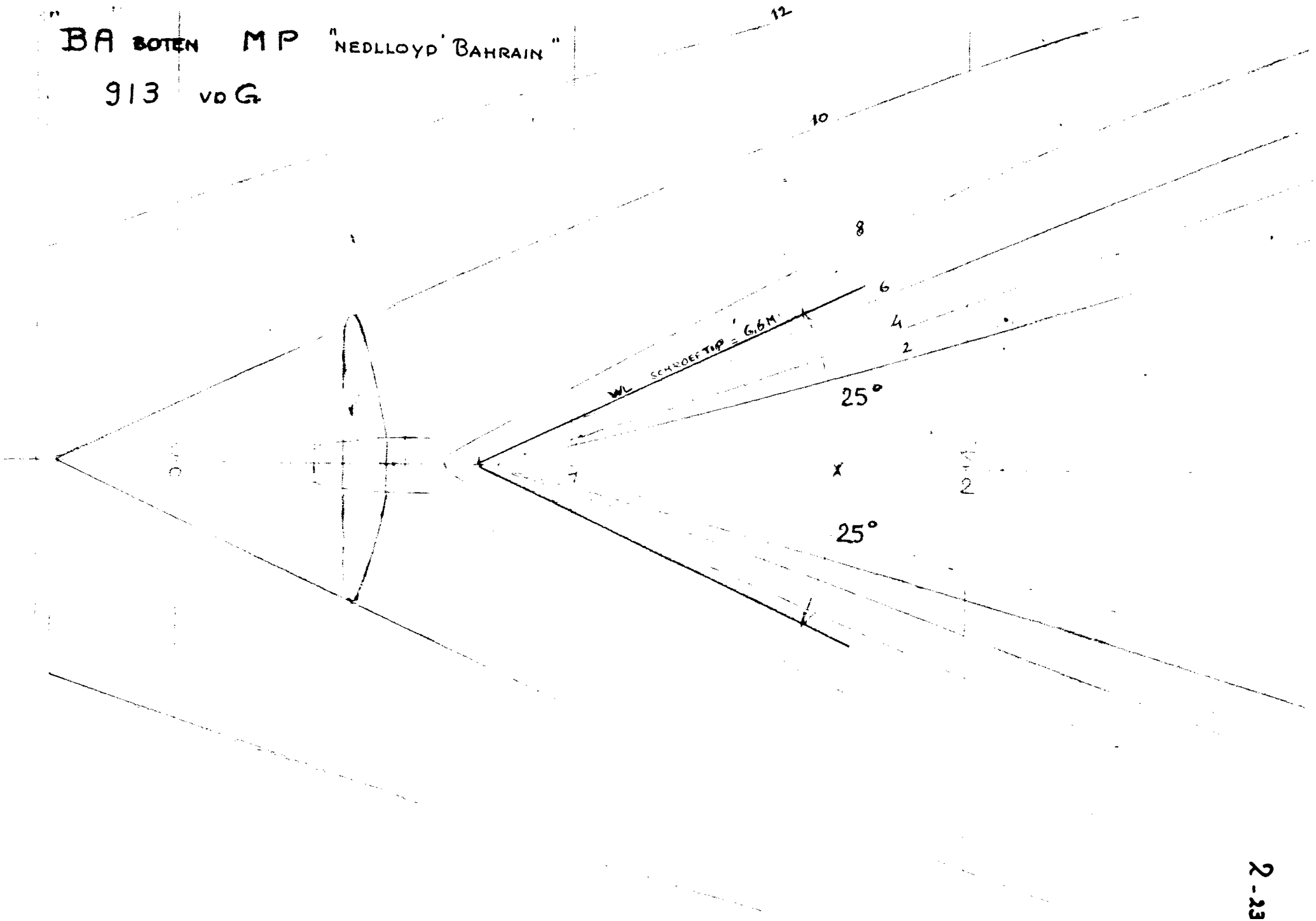


26° ACCEPTABLE

26°

---- WATERLINE AT
TOP OF PROPELLER.

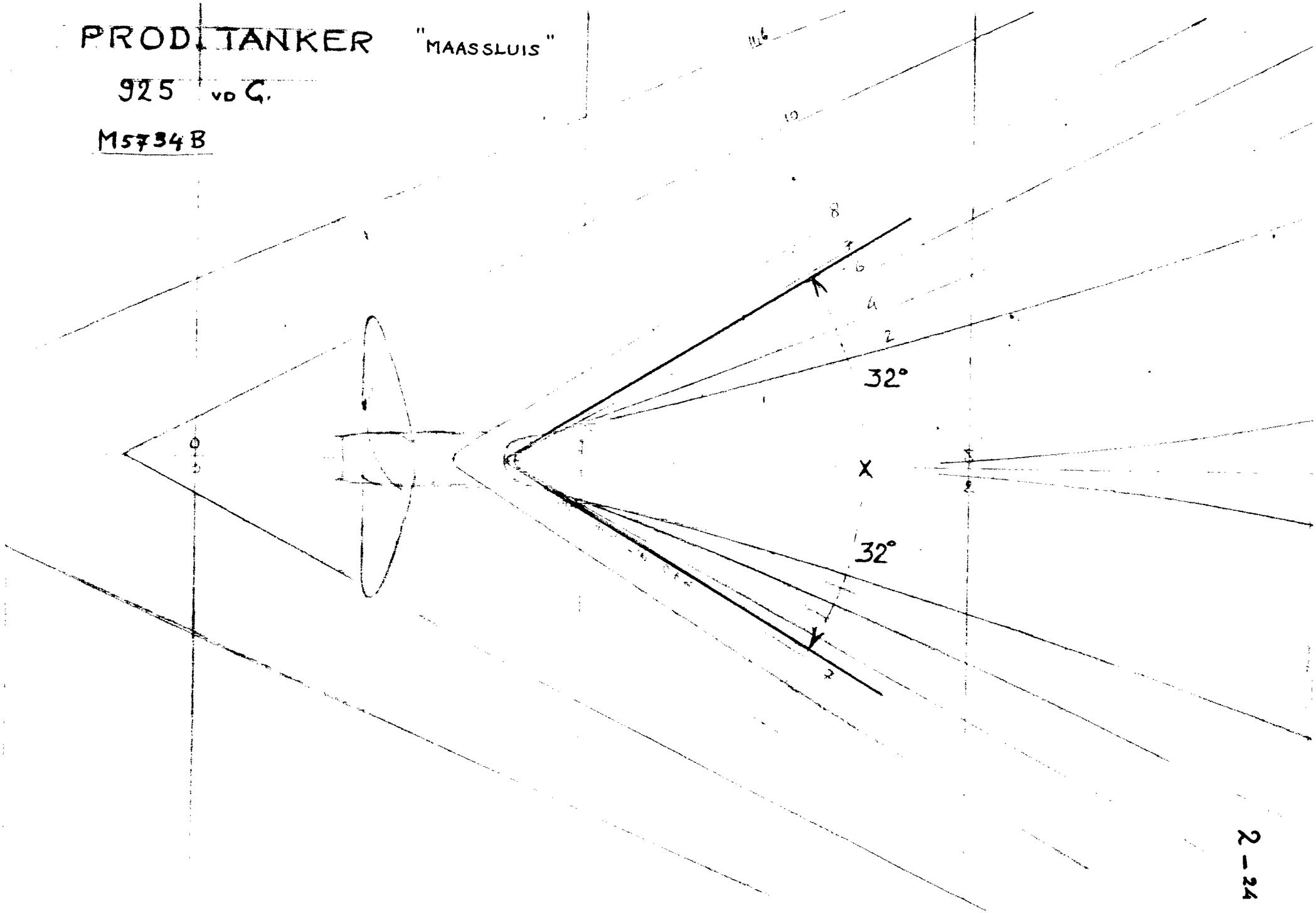
2.9.'76
NSU-S. T.AB.
ROTTERDAM


BA BOTEN MP "NEDLLOYD BAHRAIN"
913 VD G
12
10
8
6
4
2
WL SCHNEEF TOP = 6,6M
25°
25°

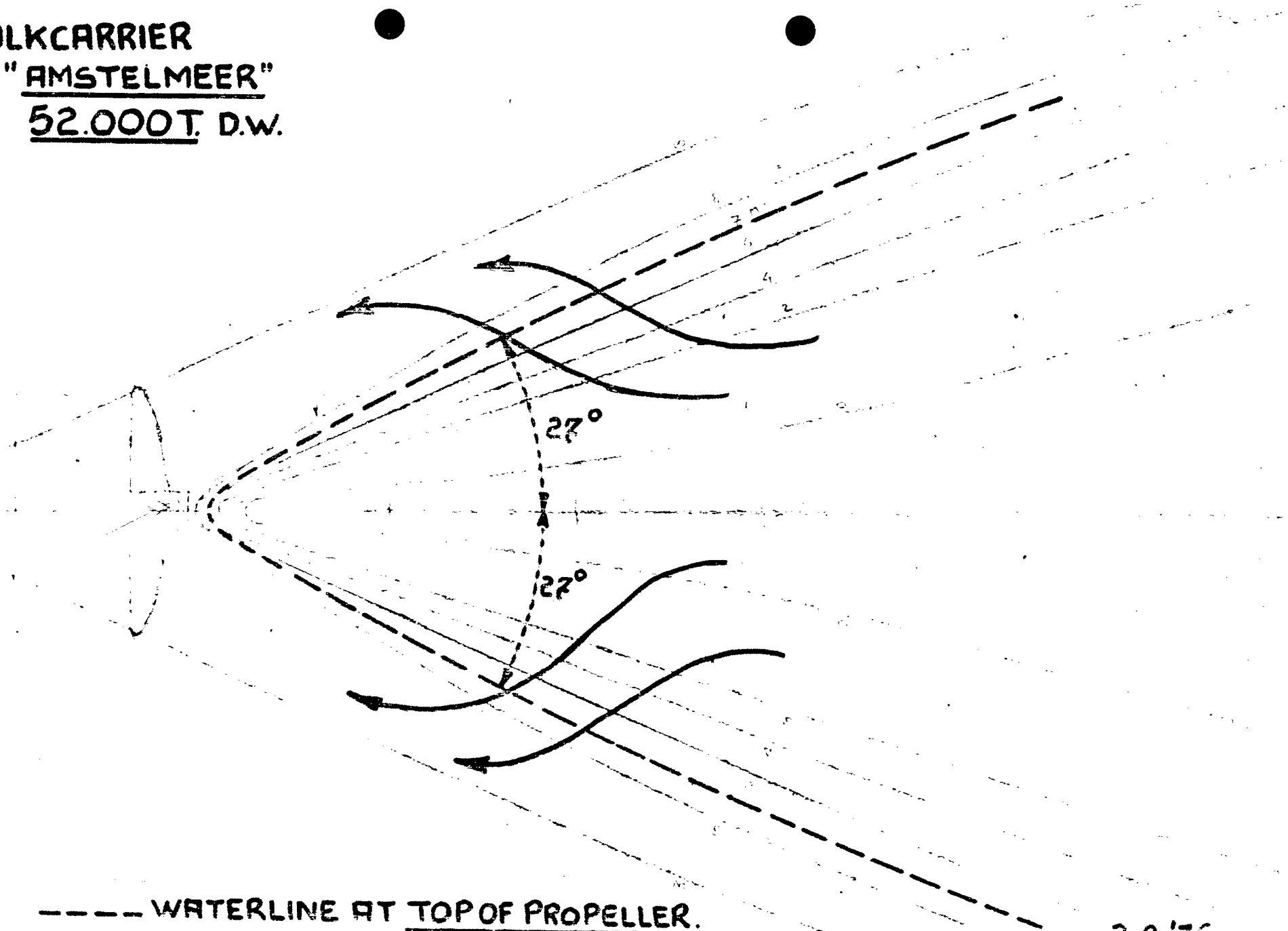
PROD. TANKER "MAASSLUIS"

925 vd G.

M5734B



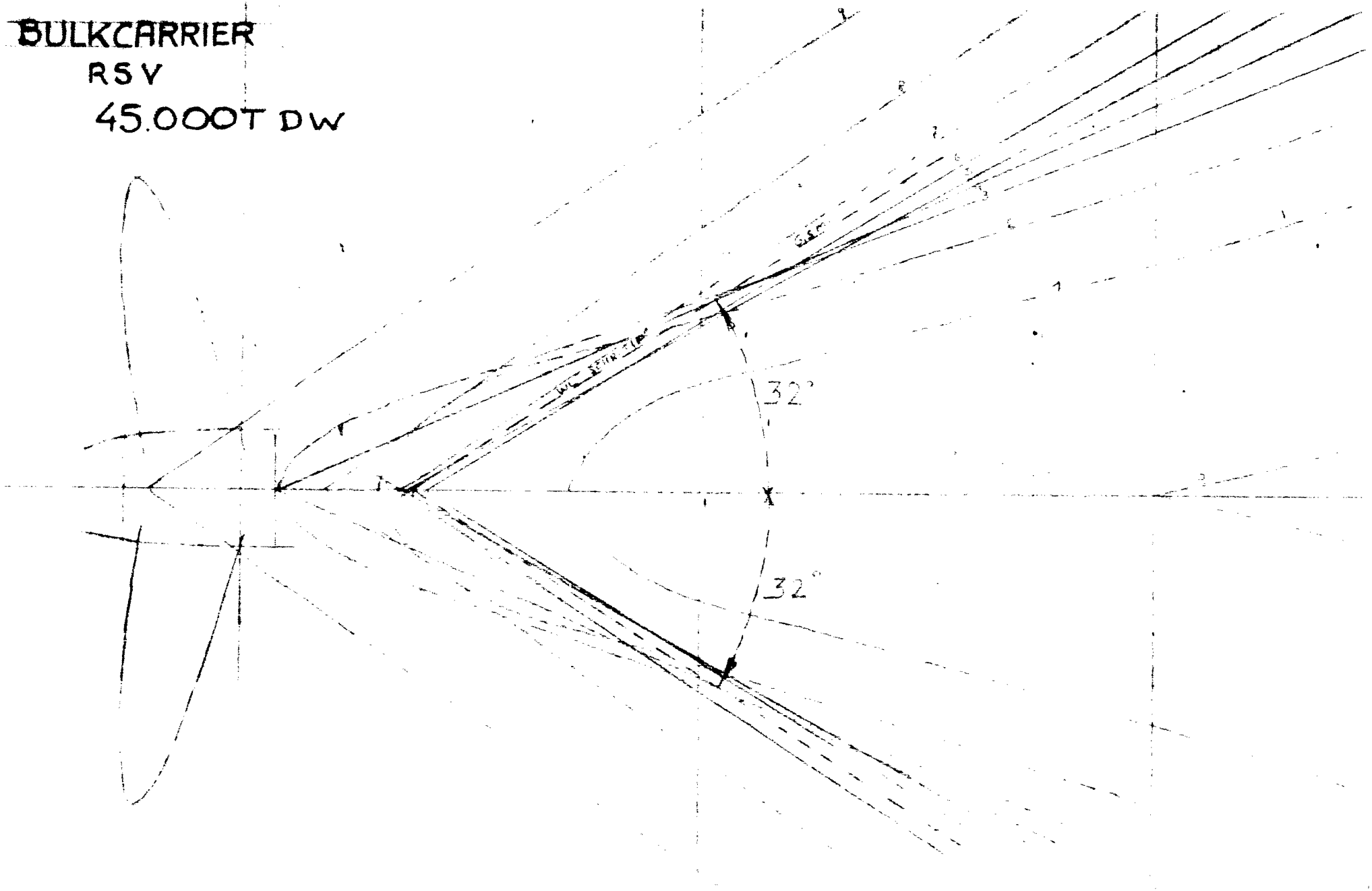
BULKCARRIER
"AMSTELMEER"
52.000T. D.W.



---- WATERLINE AT TOP OF PROPELLER.

2.9.'76
NSU_S T.A.B.

BULKCARRIER
RSV
45.000T DW



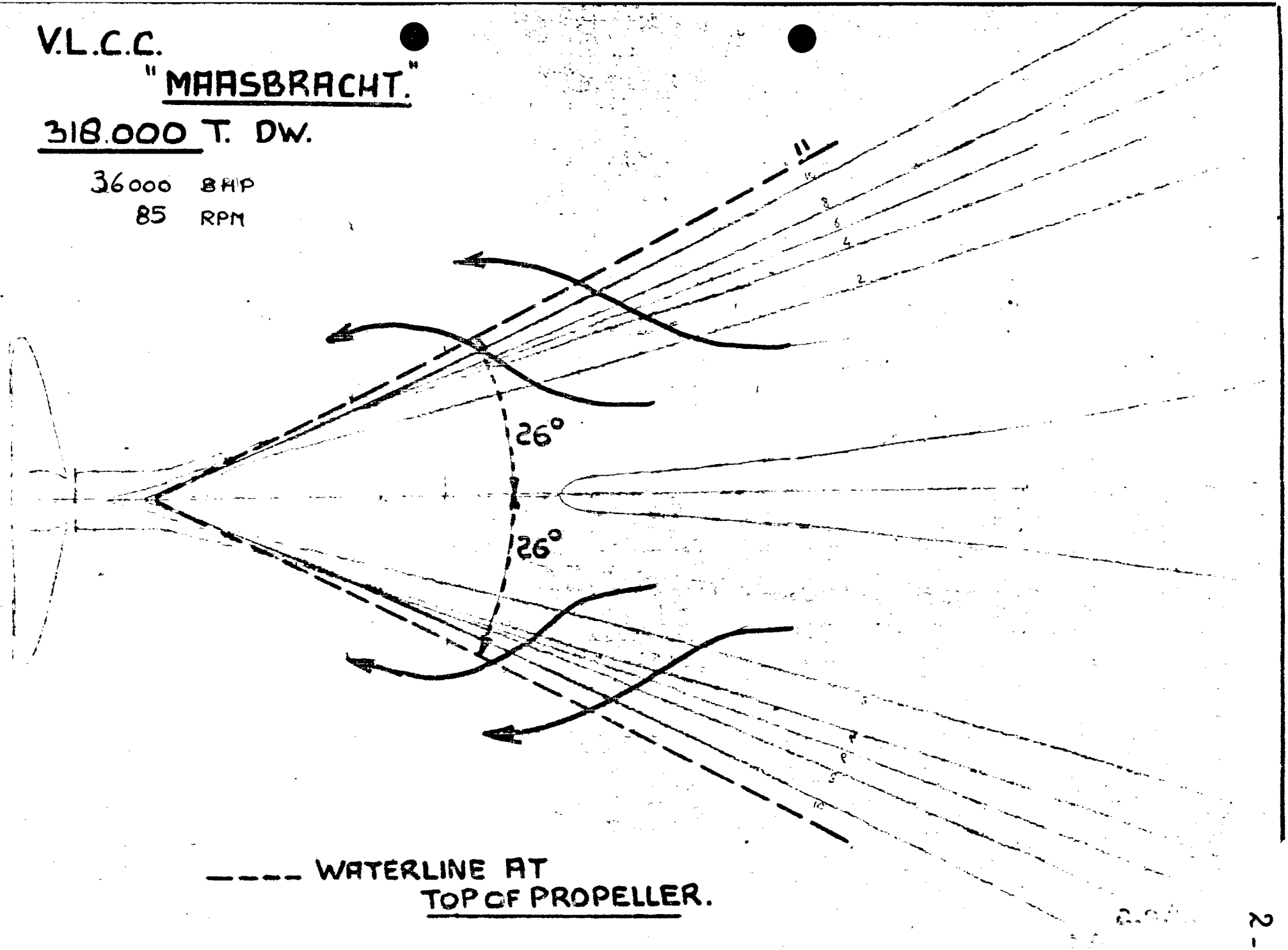
V.L.C.C.

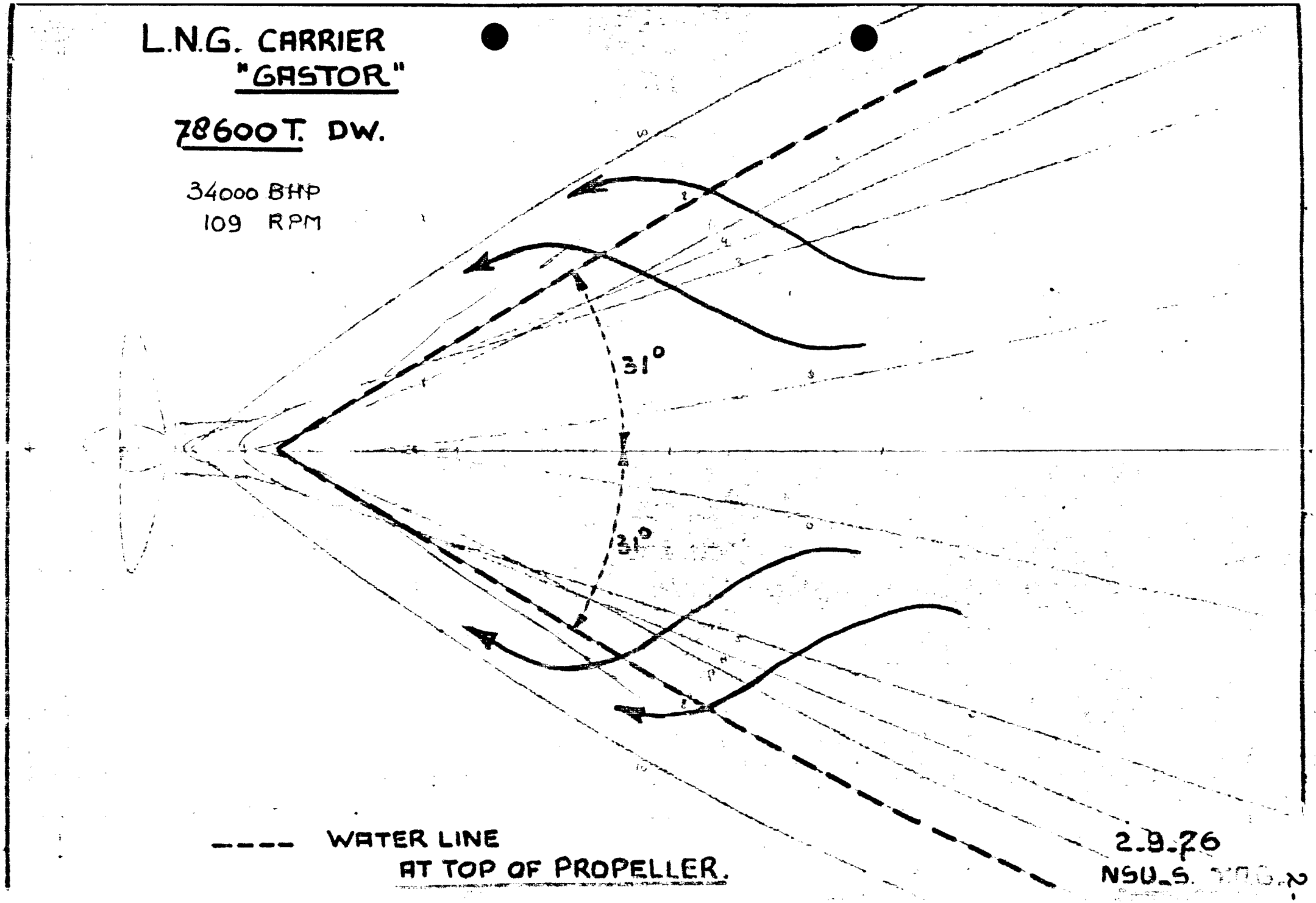
"MAASBRACHT."

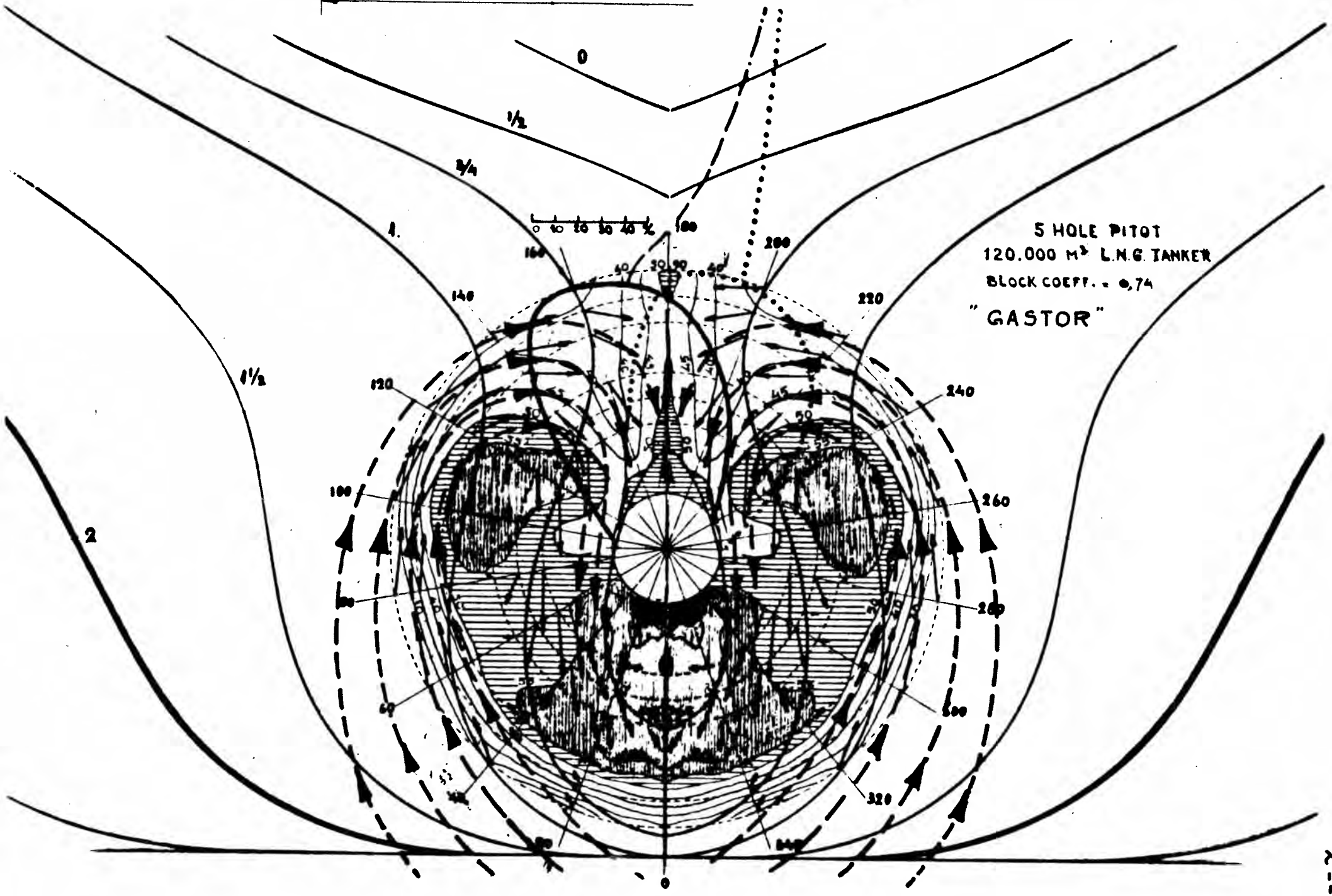
318.000 T. DW.

36000 BHP

85 RPM

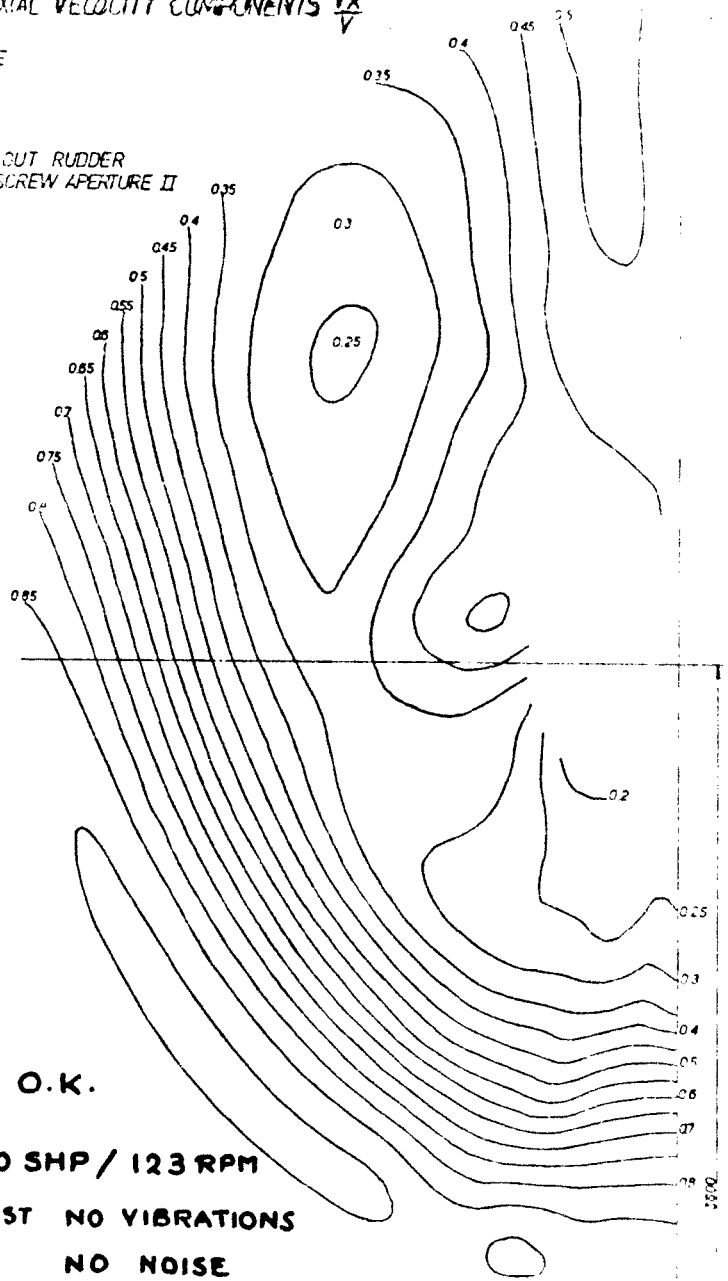



L.N.G. CARRIER
"GASTOR"
78600 T. DW.
34000 BHP
109 RPM
31°
31°
---- WATER LINE
AT TOP OF PROPELLER.
2.9.76
NSU_S.


5 HOLE PITOT
120.000 m³ L.N.G. TANKER
BLOCK COEFF. = 0,74
"GASTOR"
0
1/2
3/4
1.
1 1/2
2
0 10 20 30 40 %
100
120
140
160
180
200
220
240
260
280
300
320
340

LINES OF EQUAL AXIAL VELOCITY COMPONENTS $\frac{V_x}{V}$

HIP MODEL No. 5734 E
EST No 39158
PEED 1.37 KNOTS
RAFT 11.30 m
ODEL CONDITION WITHOUT RUDDER
WITH SCREW APERTURE II



O.K.

11100 SHP / 123 RPM

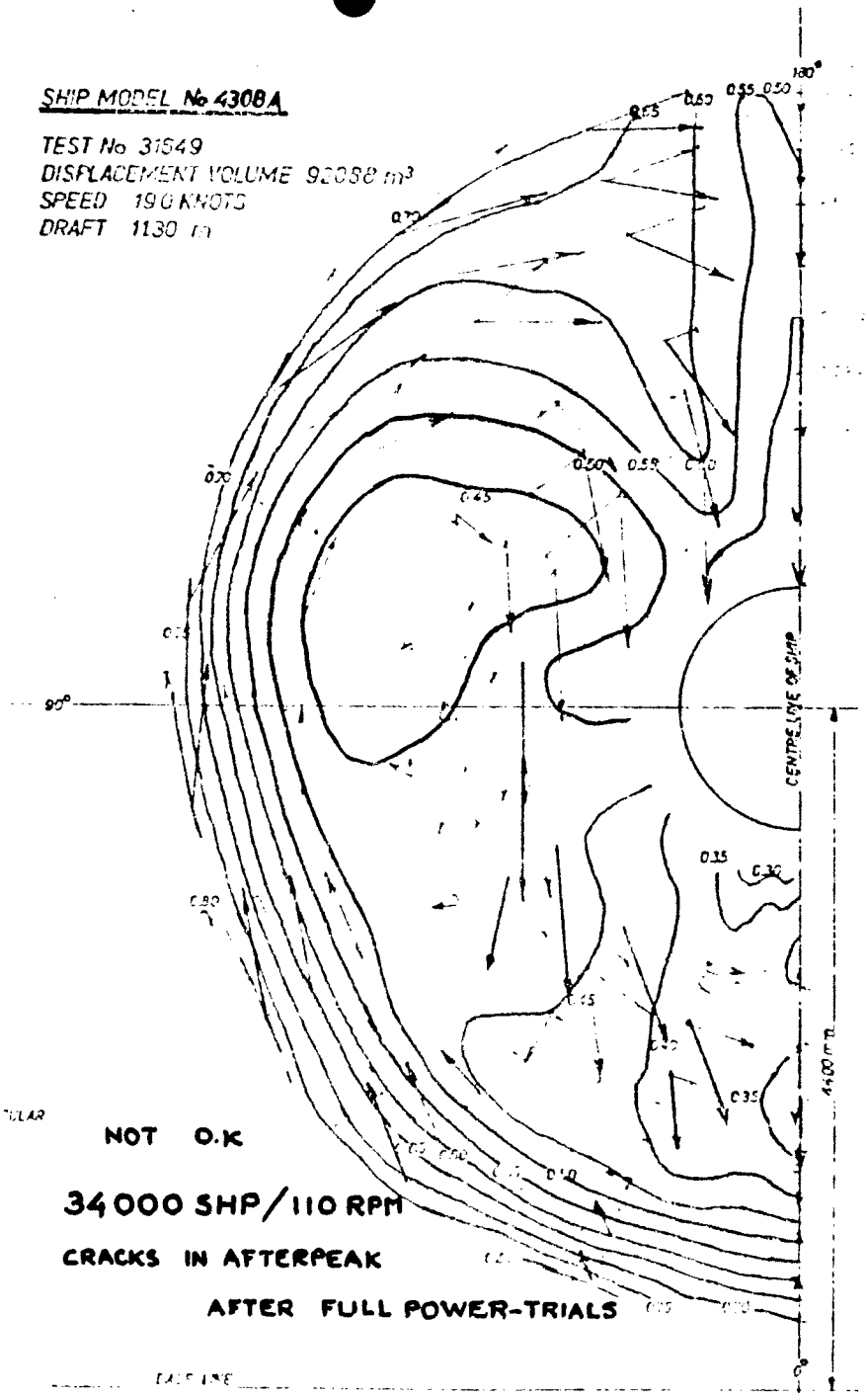
ALMOST NO VIBRATIONS

NO NOISE

"MAASSLUIS" 38000 DWT PR.TANKER

SHIP MODEL No 4308A

TEST No 31549
DISPLACEMENT VOLUME 92058 m³
SPEED 19.0 KNOTS
DRAFT 11.30 m



NOT O.K

34000 SHP/110 RPM

CRACKS IN AFTERPEAK

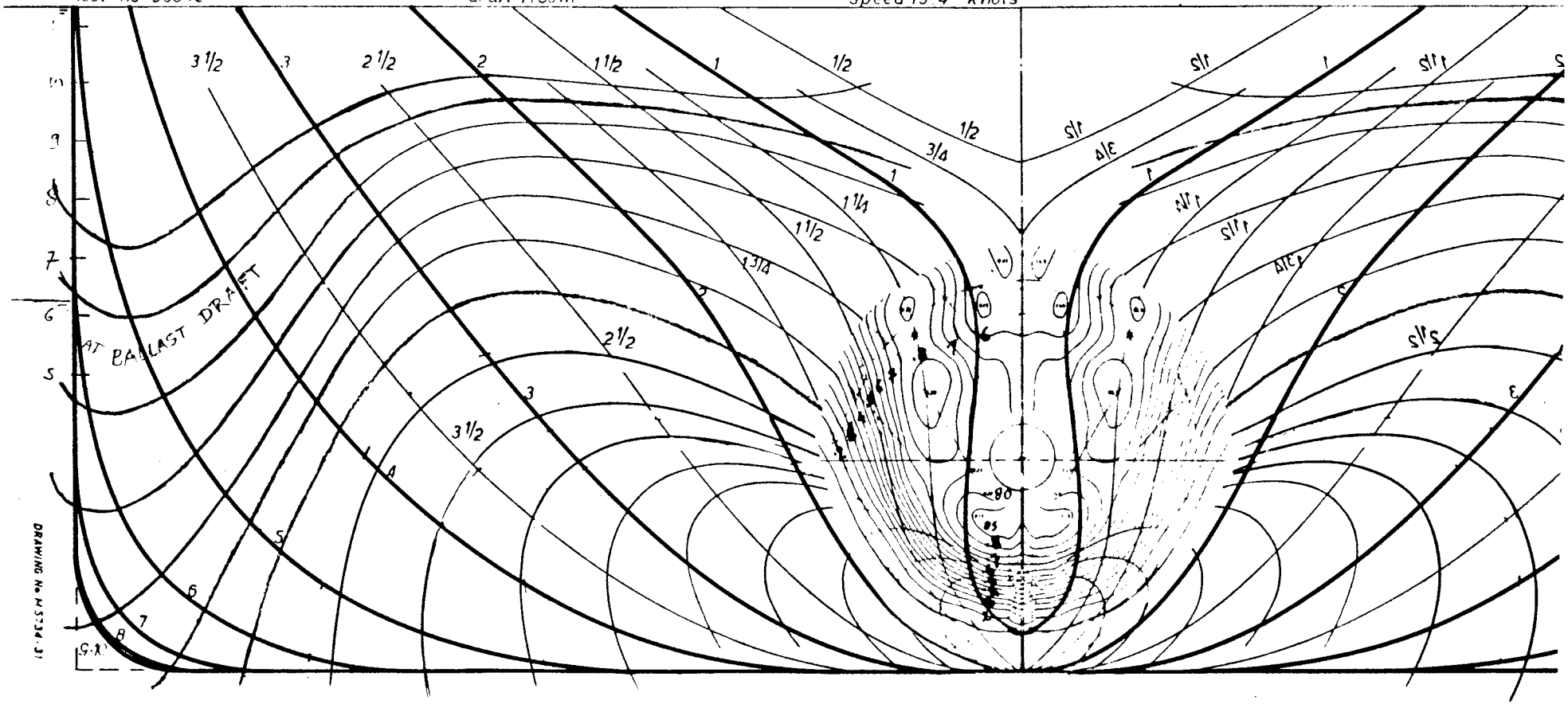
AFTER FULL POWER-TRIALS

"GASTOR" 120.000 M³ LNG

SHIP MODEL No. 5734B
STREAMLINES on AFTERBODY

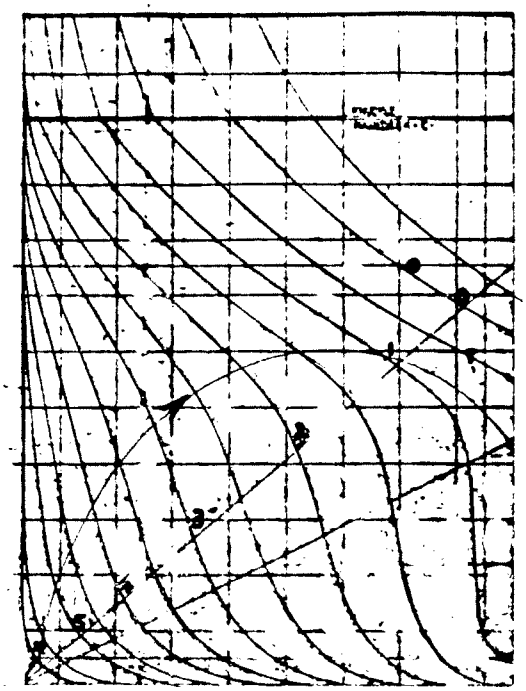
11.100 SHP/123 RPM

test no 38942 draft 11.30 m speed 15.4 knots

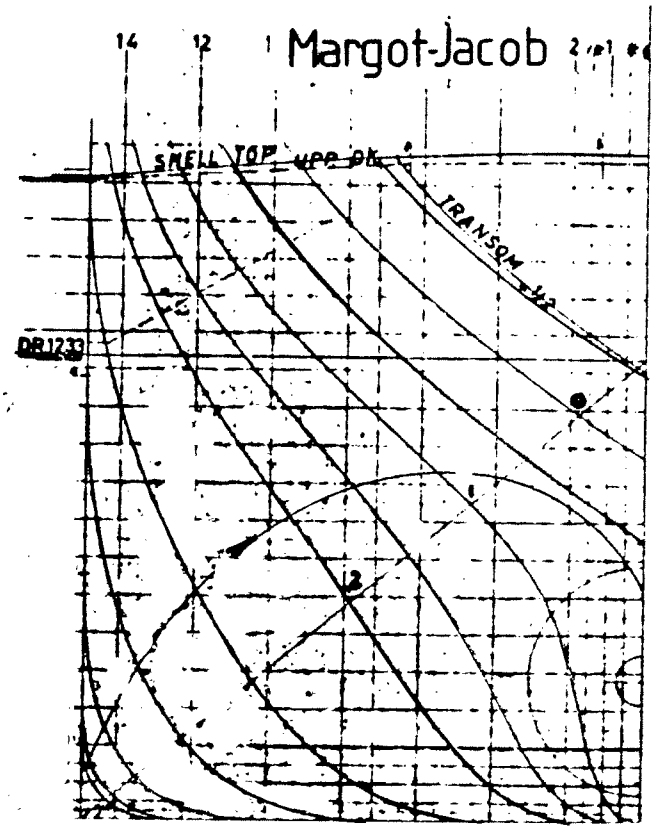


38000 D.W.T. PRODUCT-TANKER "MAASSLUIS"

"Maaskerk"



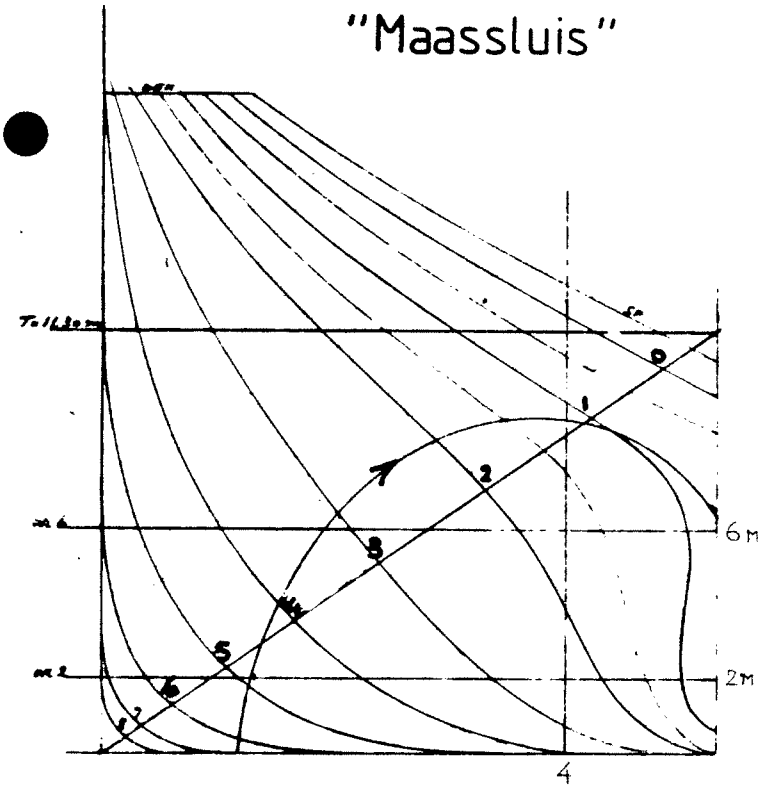
Margot-Jacob



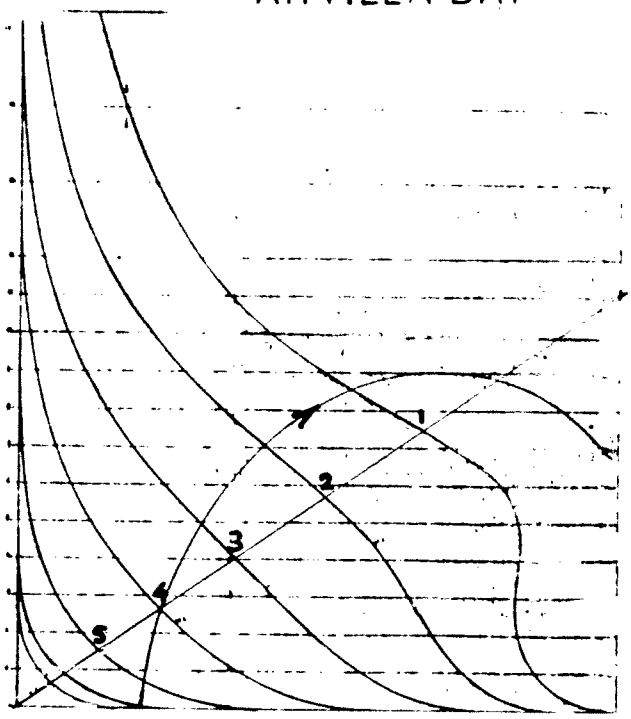
ESTIMATED FLOWLINE TO PROPELLER TIP IN 12 O'CLOCK POS.
0.8R

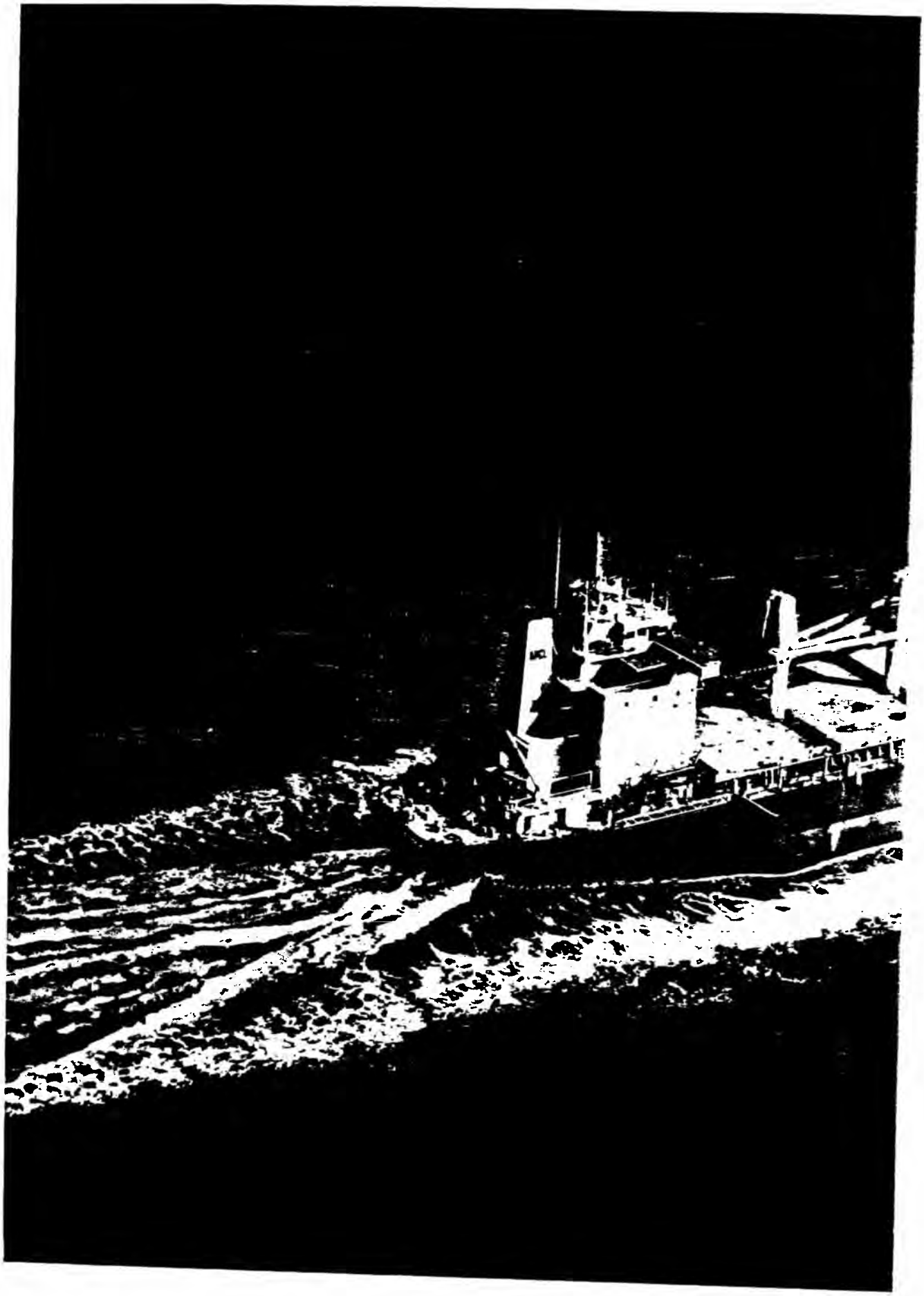
MORE KNOWLEDGE IS BADLY NEEDED!

"Maassluis"

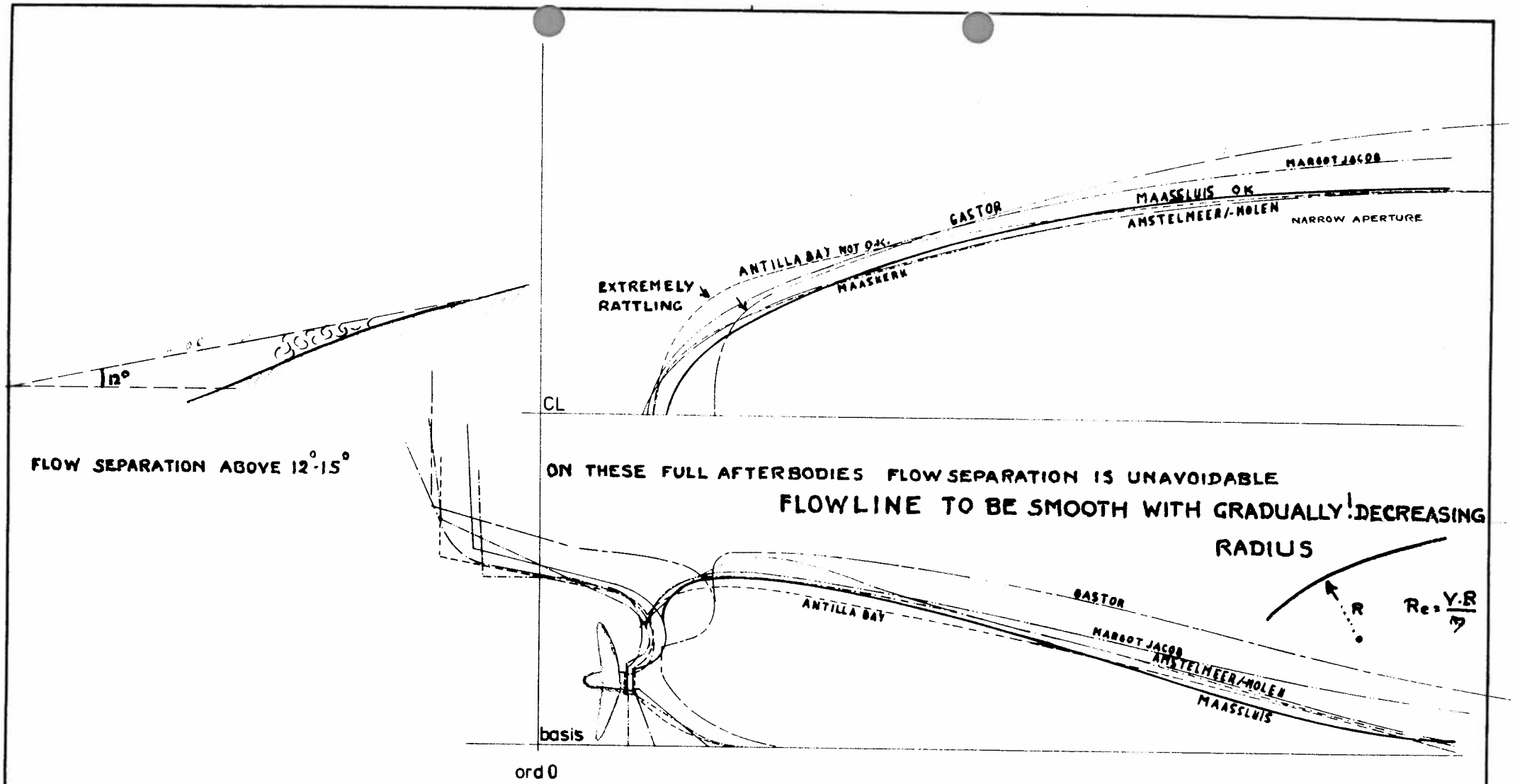


"ANTILLA BAY"





Example of stern wave - indicating poor flow lines.
WHICH IS GENERATED AT THE SIDES



——·——·——·——	Margot Jacob
——— · ——— · ———	Gastor
————————	Maassluis
——·——·——·——	Maaskerk
——— · ——— · ———	Amstelmeer/-molen
- - - - - - - - - -	Antilla Bay

Nedlloyd Fleet Services
Newbuilding Department

'Flowline' on different afterbodies.
TO PROPELLER TIP AT 12 O'CLOCK

Drawn Gus Mouw Date FEB 1983	Scale	No	
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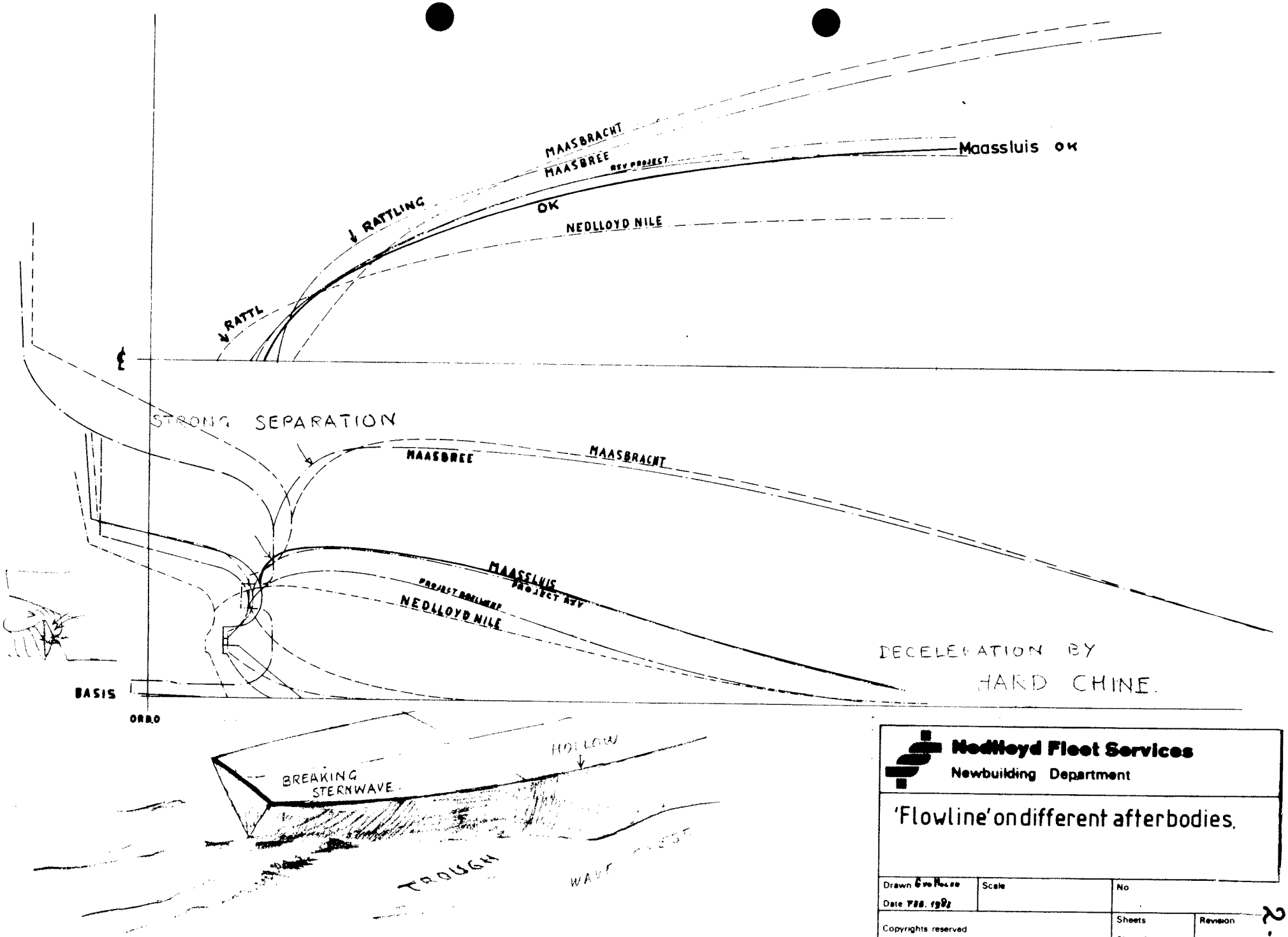
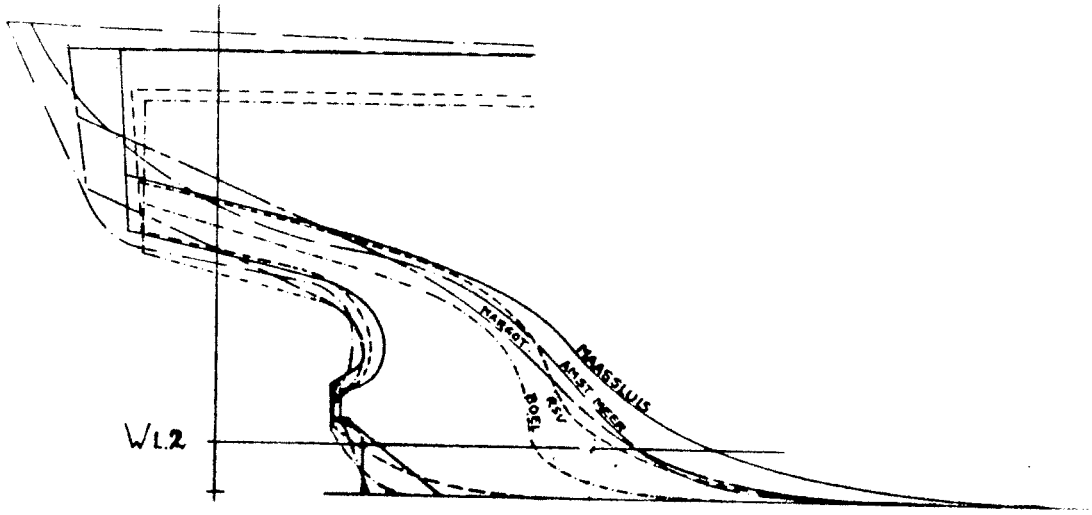
2-34

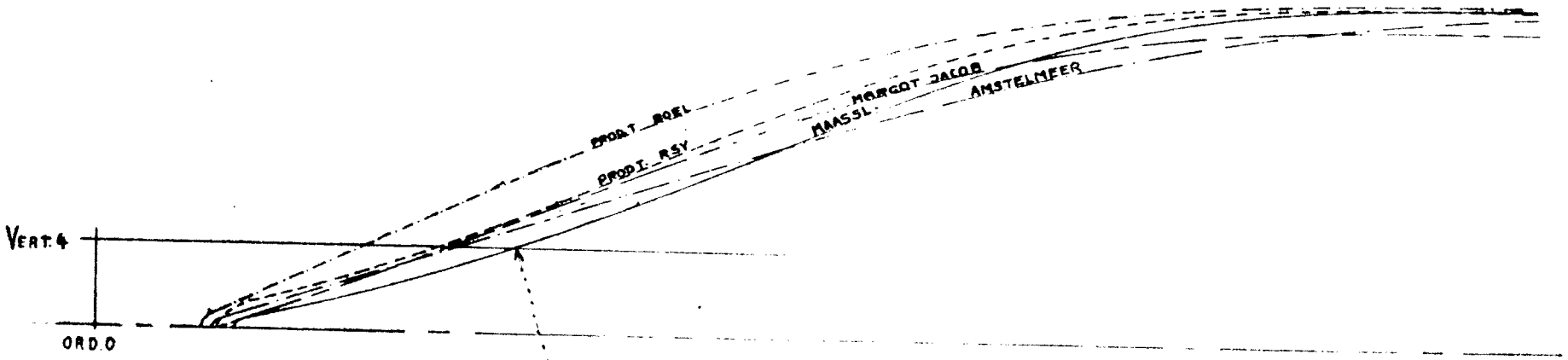


Nedlloyd Fleet Services
Newbuilding Department

'Flowline' on different afterbodies.

Drawn G.v.d. Hooven Date Feb. 1983	Scale	No	
Copyrights reserved		Sheets Sheet No	Revision





MAASSLUIS HAS SLENDER WL 2 DESPITE HER $\frac{L}{B}$ = 5.09

SHORT VESSEL!

"MAASSLUIS"

BOEL PROD TANKER

"AMSTELMEER"

VEROLME CDRK "MARGOT JAKOB"

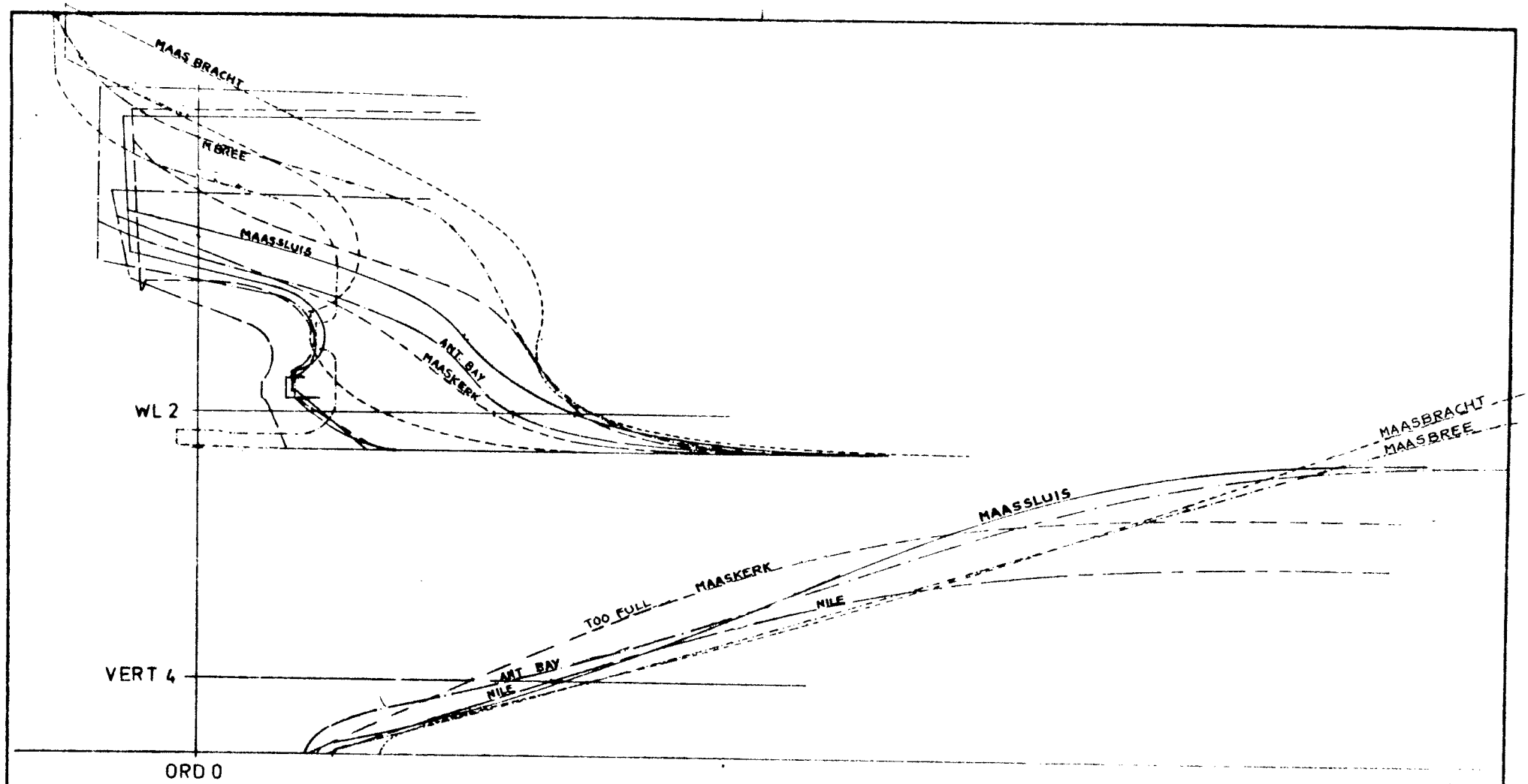
RSV PROD-TANKER

Nedlloyd Fleet Services

Newbuilding Department

COMPARING WL2 AND VERT4

Drawn G. van Meeuwen Date 1983	Scale 1:200	No 1
Copyright reserved	Checked	Approved



Maasbree
Maasbracht
Maassluis
Antilla Bay
Maaskerk
Nedlloyd Nile

Nedlloyd Fleet Services
Newbuilding Department

COMPARING WL 2 AND VERT. 4
SHIPS LINES PUT ON APP. 0

Drawn G.v.Maaren Date feb 1982	Scale	No
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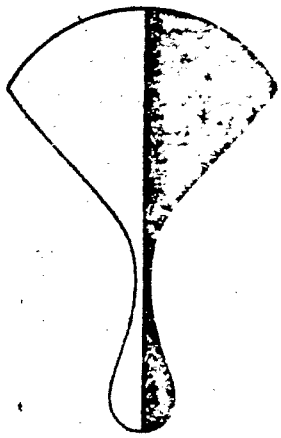
3

Bericht 50/75

FDS

FORSCHUNGSZENTRUM
DES
DEUTSCHEN SCHIFFBAUS



32910

FULL-BODY-STERNS, SEPARATION OF FLOW

Ablösungserscheinungen an völligen Hinterschiffen

MAX ALLOW. C_B - AFT ?

(THIS IS DEPENDENT ON SHIPS LENGTH.)

IT IS BETTER TO DEFINE AN ANGLE

SECTIONAL AREA IN M^2 ON BASE OF M.

H. GROTHUS-SPORK

VERSUCHSANSTALT FÜR WASSERBAU UND SCHIFFBAU

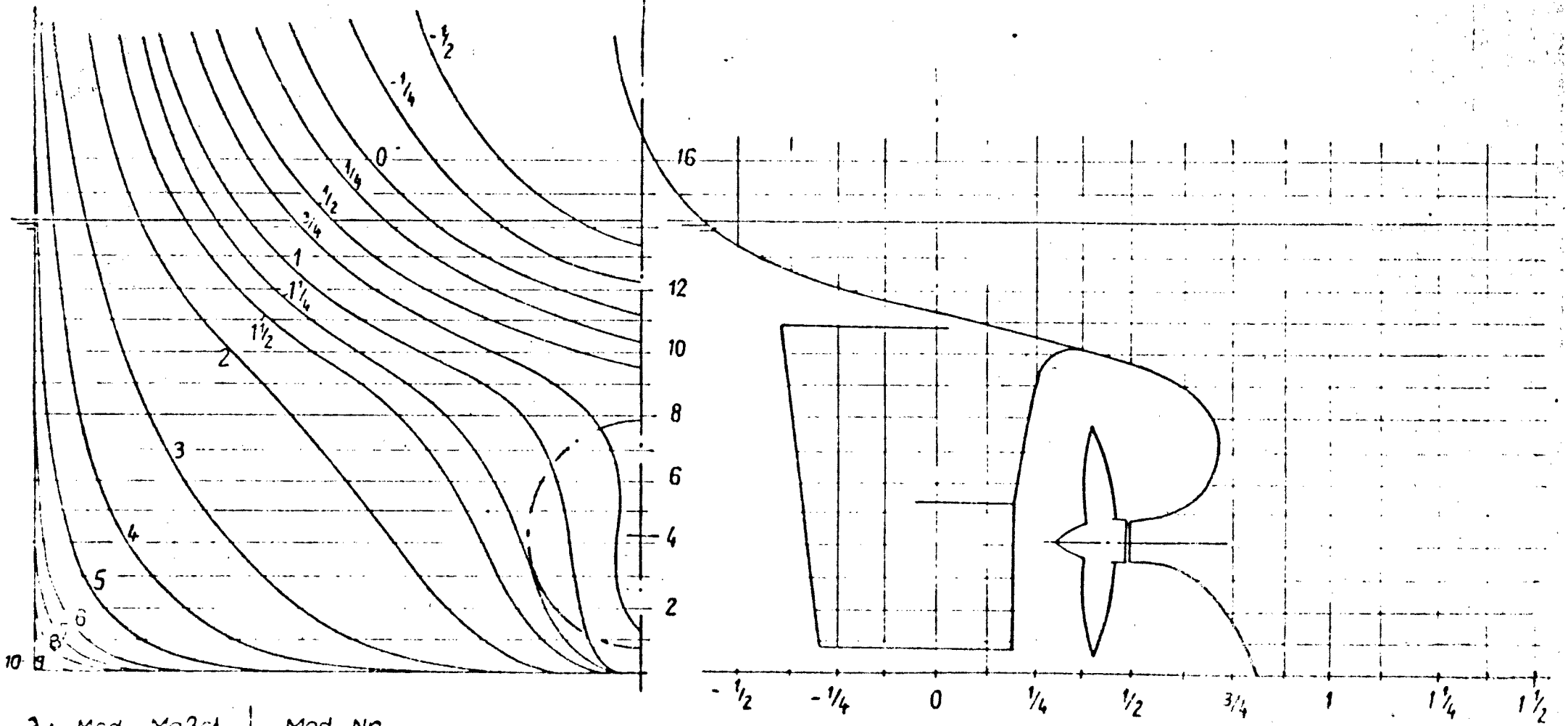
BERLIN

2 HAMBURG 1 AN DER ALSTER 1

Tabelle 1 - Hauptabmessungen der untersuchten Formen und Modelle

Form Nr.		1	1	1	2	3	4	5
Schiffstyp		gr.	Tanker		gr. Tanker	Tanker	kl. Tanker	Bulkcarrier
Modell Nr.		2231	2216	2232	2239	2184	2209	2112
Modell-Maßstab SCALE		55	42	33 1/3	44	40	27,5	42,5
L_{pp} Mod.	m	4,60	6,02	7,59	7,04	6,00	6,69	6,36
L_{pp}	m		253		310	240	184	270
B	m		38,33		51,81	40,0	28,40	42,5
T	m		14,20		22,32	14,49	11,58	16,39
∇	m^3		117109		304890	111590	48722	158924
C_B (δ)			0,8504		0,851	0,802	0,805	0,845
C_B hinten (δ_{hinten})			0,784		0,786	0,770	0,760	0,796
⊙ vor HL	% L_{pp}		0,531		0,531	0,515	0,524	0,525
L_{WL}/B			6,804		6,130	6,15	6,690	6,538
B/T			2,700		2,321	2,760	2,452	2,593
α_H (L_{pp}) CWL-Völligkeit im Achterschiff			0,917		0,904	0,883	0,905	0,925
α_H (WL) auf L_{WL} bezogen			0,864		0,861	0,841	0,850	0,875
$D_{Propeller}$	m		7,00		9,50	7,00	6,10	7,00
D_{Pr}/T			0,493		0,425	0,483	0,527	0,427
Hinterschiffsspantform		ausgeprägte V-Form geringe Anwulstung vor Propeller			V-Form weniger ausgeprägt als Form 1	Spant-form ähnl. Form 2	AG Weser Heckwulst	völligste Form mit ausgeprägten V-Spanten

L_{pp} = 253 m	∇ = 117100 m^3	$C_{B\,ges.}$ (TOTAL) = 0,850
B = 38,3 m	B/T = 2,70	$C_{B\,hinten}$ (AFT) = 0,784
T = 14,2 m	L/B = 6,80	



λ: Mod - Maßst	Mod Nr
33 1/3	2232
42	2216
55	2231

Form 1

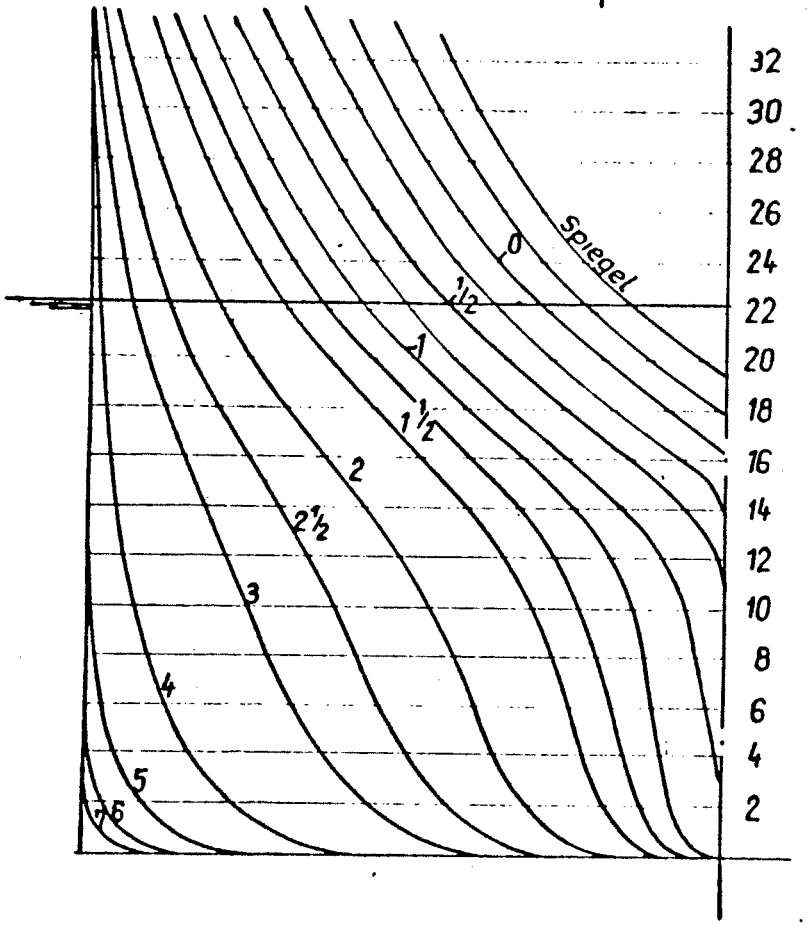
1

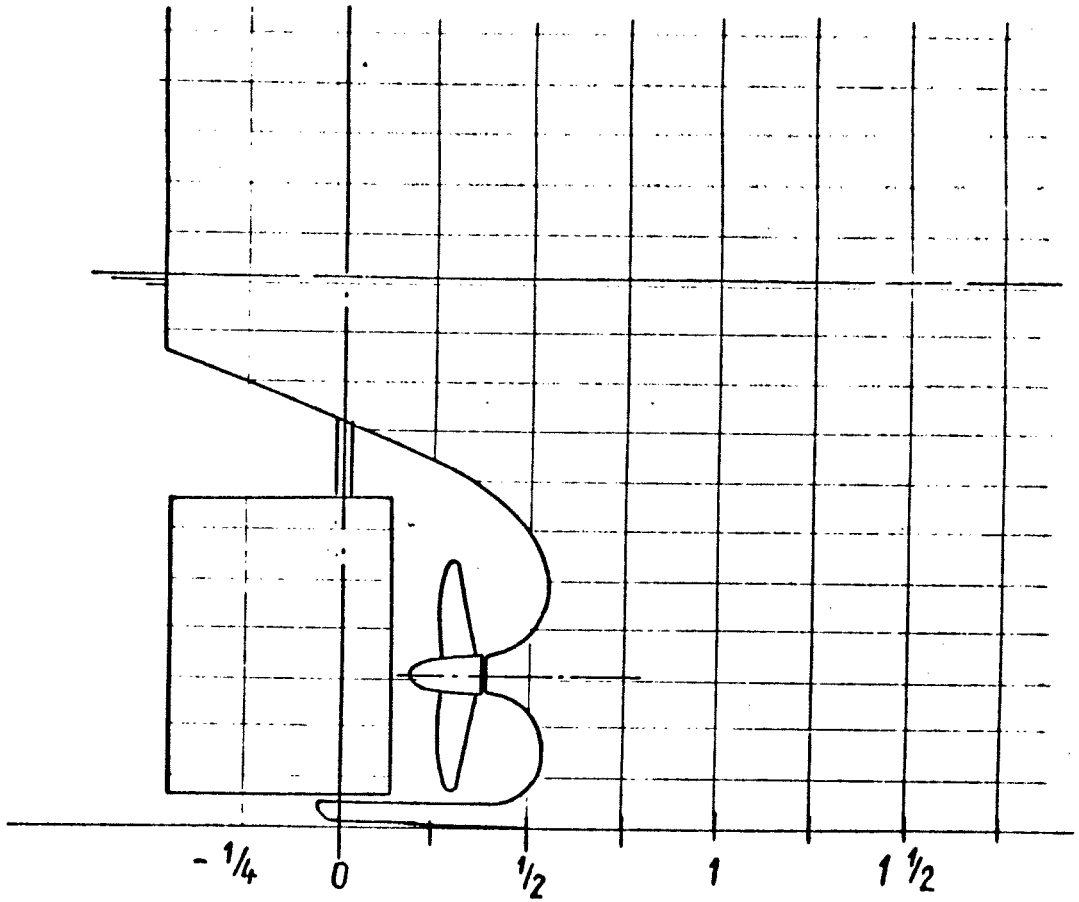
L_{pp} = 310 m
B = 51,8 m
T = 22,3 m

∇ = 304890 m³
B/T = 2,32
L/B = 6,13

$C_{B\,ges.}$ = 0,851
$C_{B\,hinten}$ = 0,786

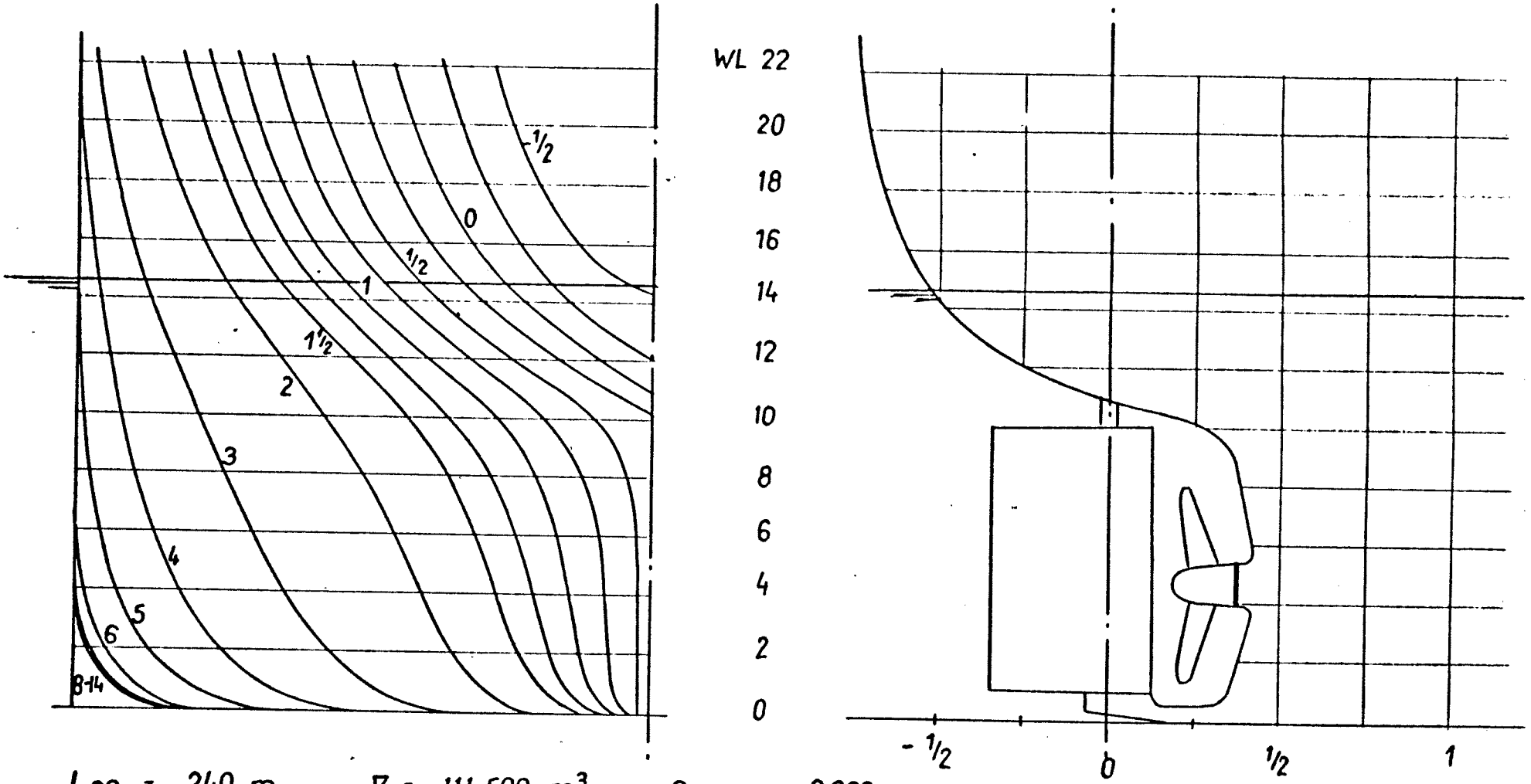
λ = 44





Mod. Nr. 2239	Form 2	2

3-4



L_{pp} = 240 m
B = 40 m
T = 14,49 m

∇ = 111 590 m³
B/T = 2,76
L/B = 6,15

$C_{B\,ges.}$ = 0,802

$C_{B\,hinten}$ = 0,770

λ = 40

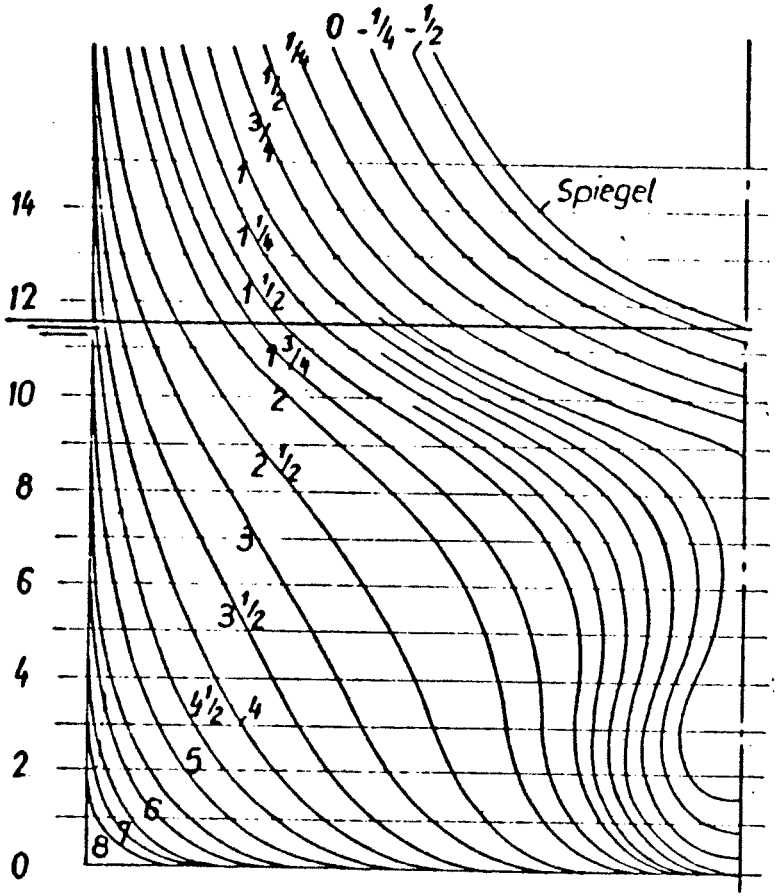
Mod. Nr. 2184	Form 3	3

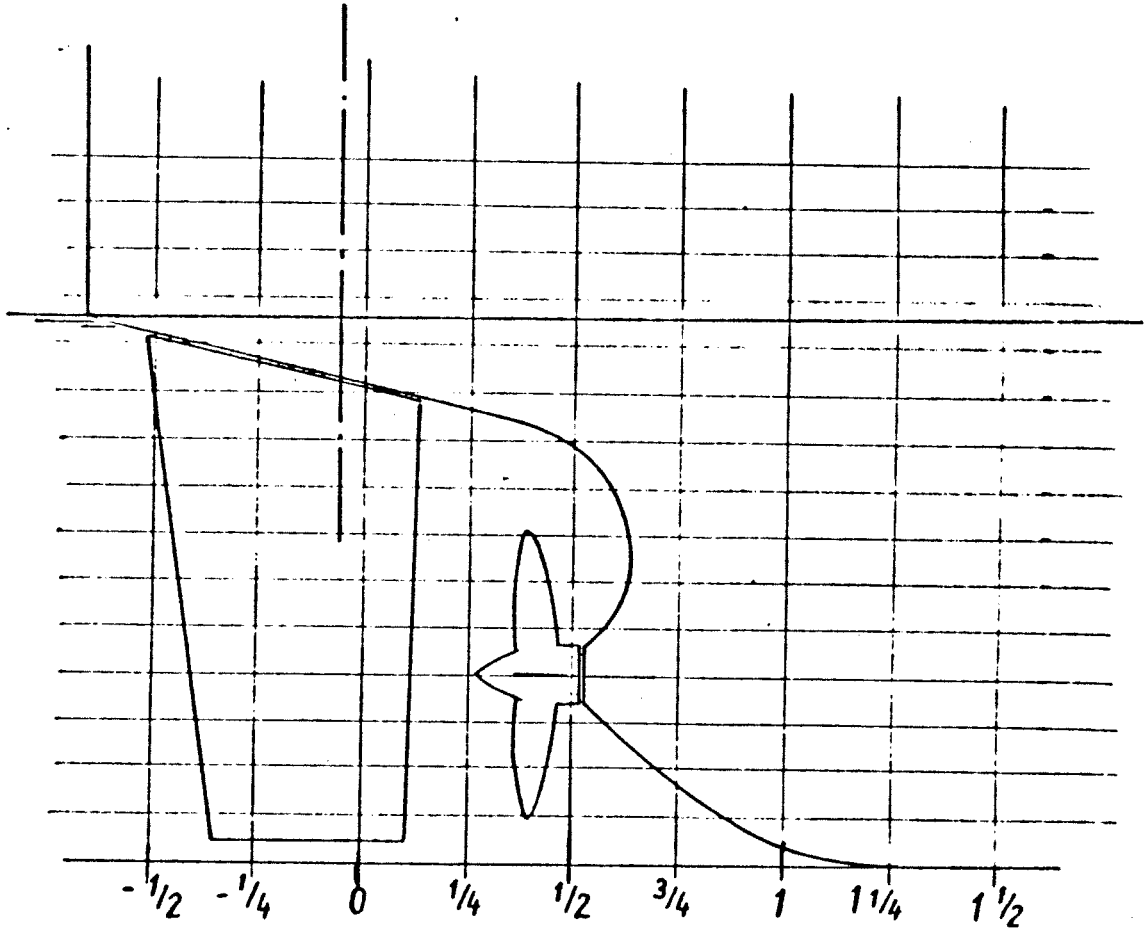
L_{PP} = 184 m
B = 28,4 m
T = 11,6 m

∇ = 48 722 m^3
B/T = 2,45
L/B = 6,69

$C_{B\ ges.}$ = 0,805
$C_{B\ hinten}$ = 0,760

λ = 27,5





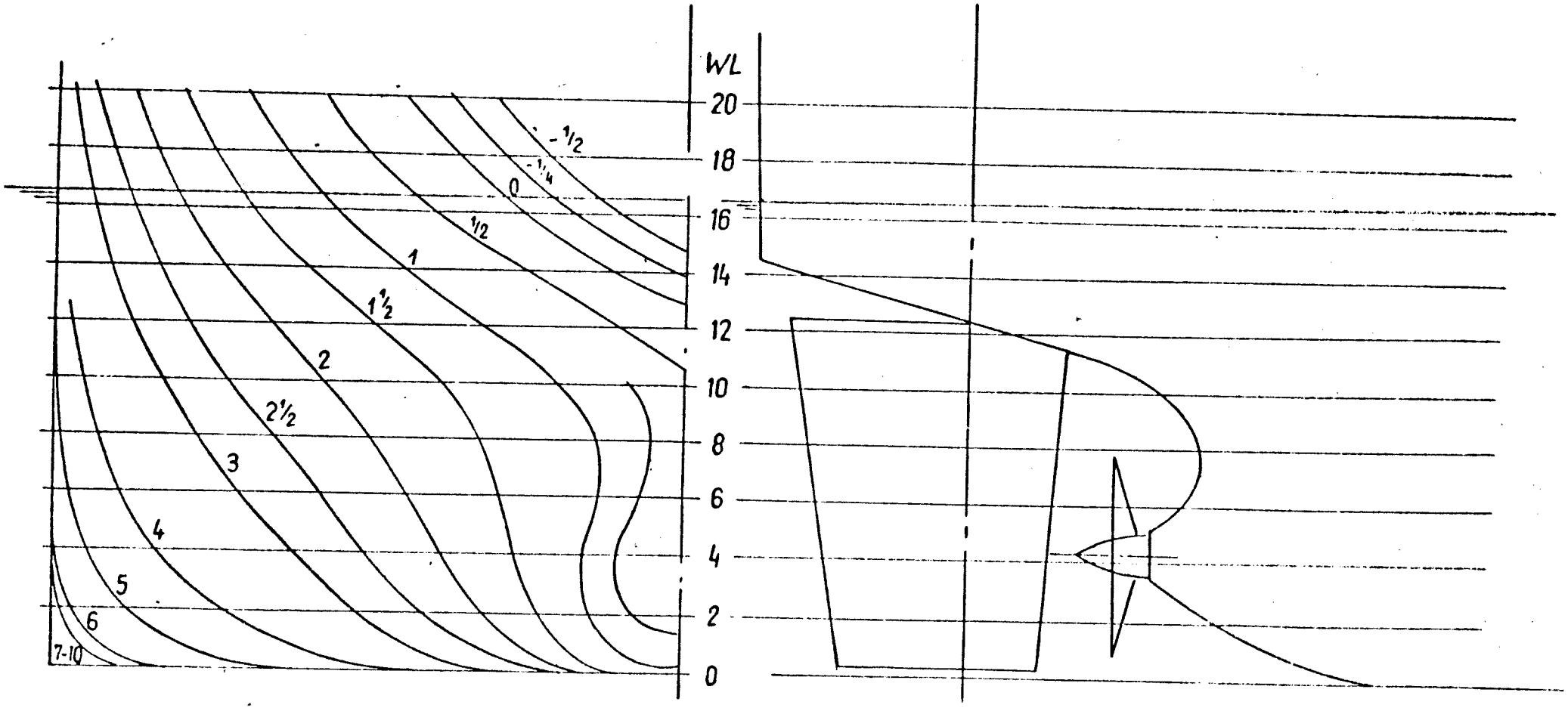
Mod. Nr. 2209	Form 4	4

3-6

L_{pp} = 270 m
B = 42,5 m
T = 16,4 m
∇ = 158925 m^3

B/T = 2,59
L/B = 6,54

$C_{B\,ges}$ = 0,845
$C_{B\,hinten}$ = 0,797
λ = 42,5



Mod. Nr. 2112	Form 5	5

Schwimmlage : CWL

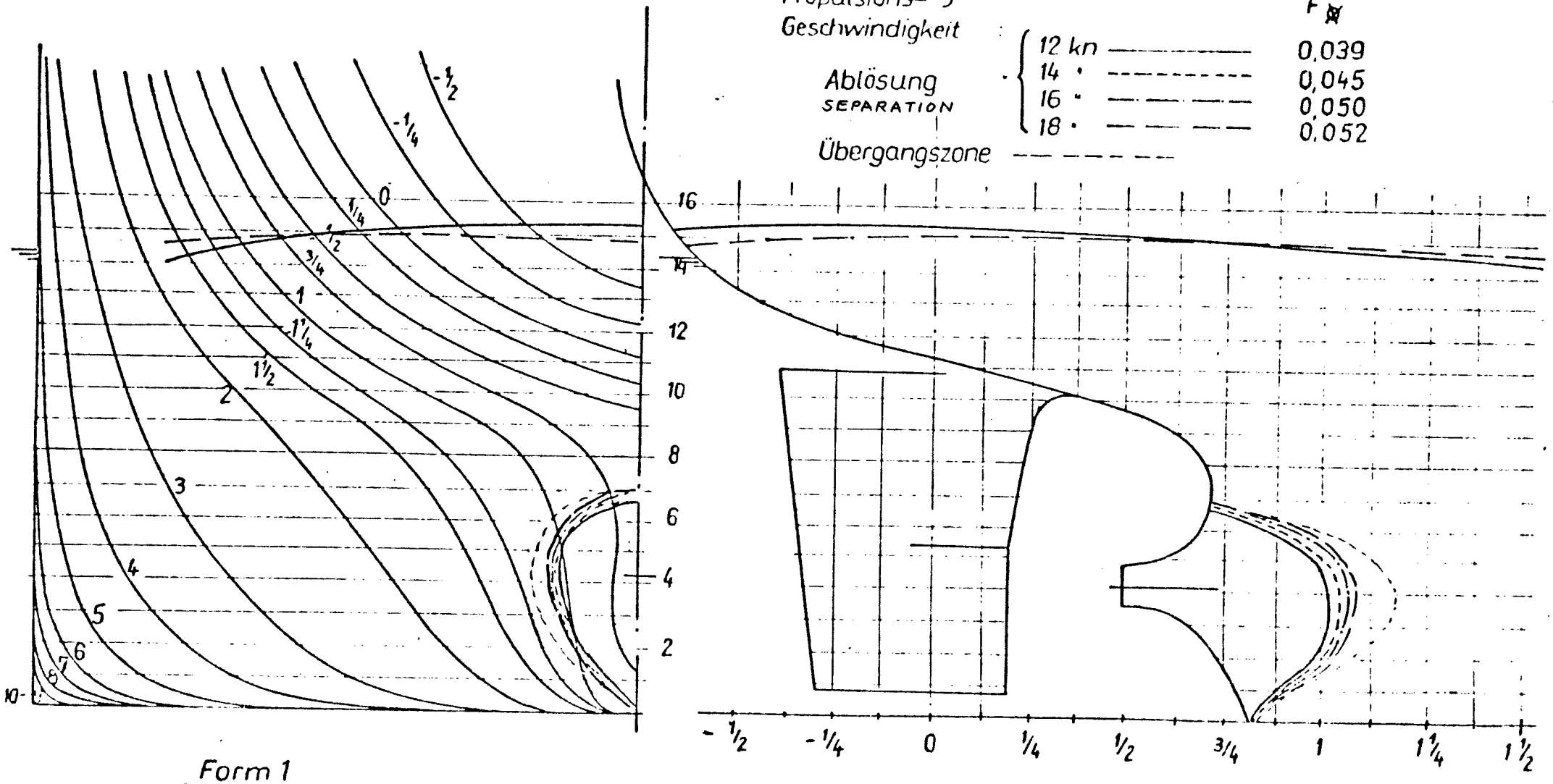
Widerstands - } Versuch – RESISTANCE
~~Propulsions-~~

Geschwindigkeit :

		$\frac{F_{Abl}}{F_{\boxtimes}}$
12 kn	————	0,039
14 "	- - - - - -	0,045
16 "	— — — —	0,050
18 "	— — — —	0,052

Ablösung
SEPARATION

Übergangszone — — — — — —



Form 1

CWL

1 : 55

Mod. Nr. 2231	Ablösung am Achterschiff	6

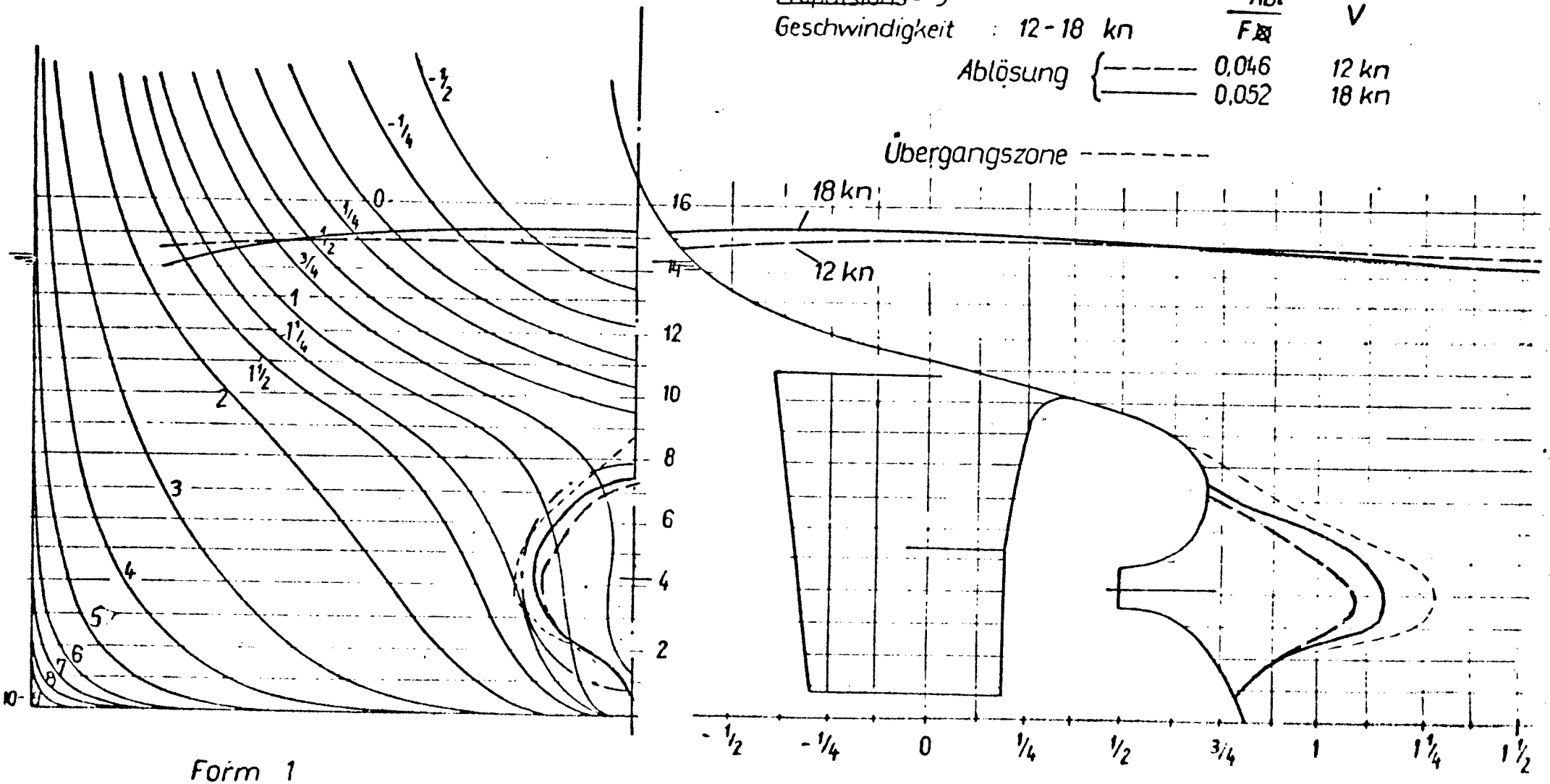
3-7 B

Schwimmlage : CWL

Widerstands - } Versuch - RESISTANCE
~~Propulsions~~ }

Geschwindigkeit : 12 - 18 kn

	$F_{Abl}/F_{\boxtimes}$	V
Ablösung ------	0,046	12 kn
Ablösung ———	0,052	18 kn



Form 1

CWL

1 : 42

Mod. Nr. 2216	Ablösung am Achterschiff	7

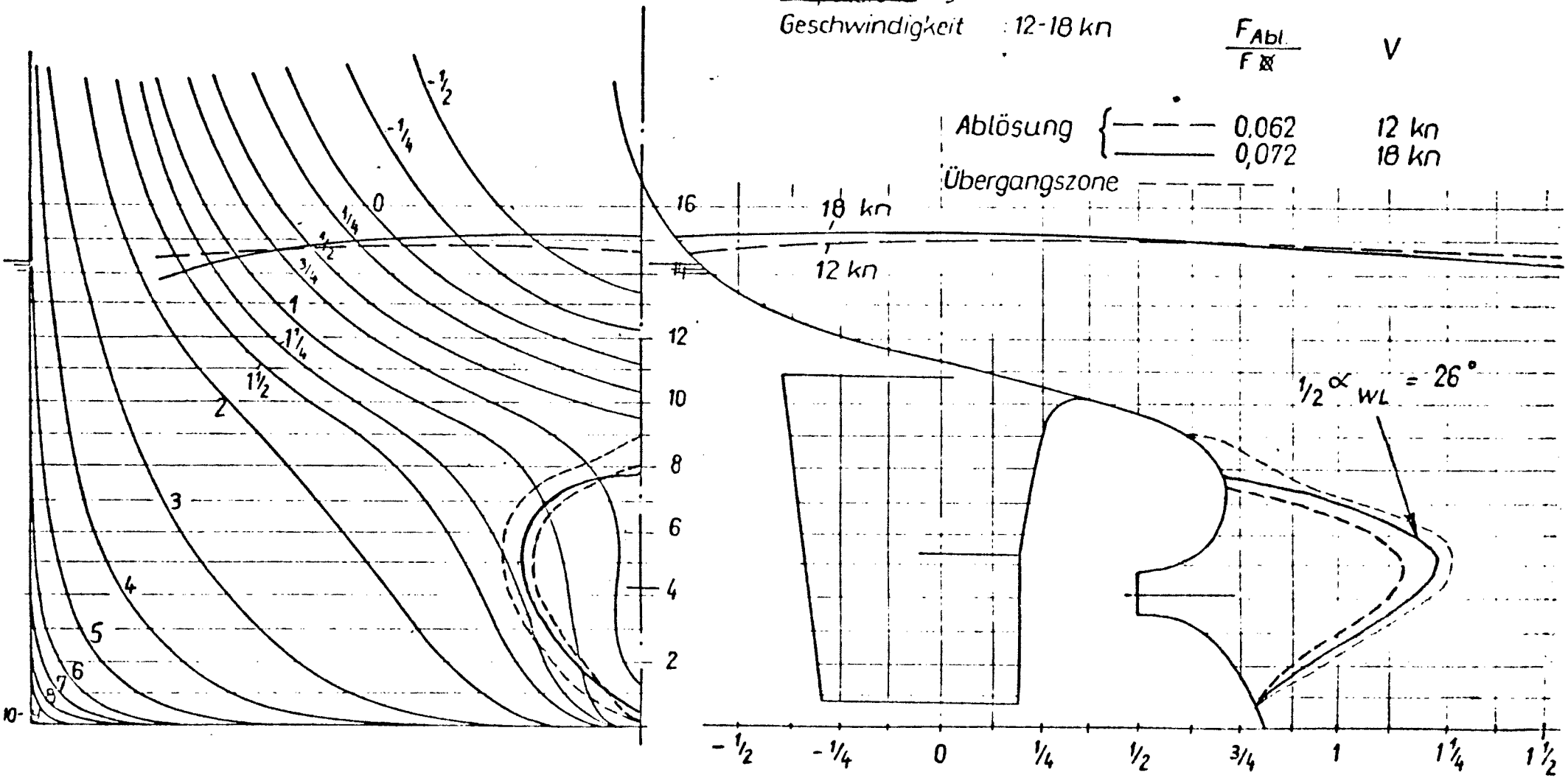
3.8

Schwimmlage : CWL

Widerstands - } Versuch RESISTANCE
~~Propulsions-~~

Geschwindigkeit : 12-18 kn

	$\frac{F_{Abl.}}{F_{\boxtimes}}$	V
Ablösung — — —	0,062	12 kn
———	0,072	18 kn
Übergangszone — — —		



Form 1

CWL

1 : 33 1/3

Mod. Nr. 2232	Ablösung am Achterschiff	8

3.6

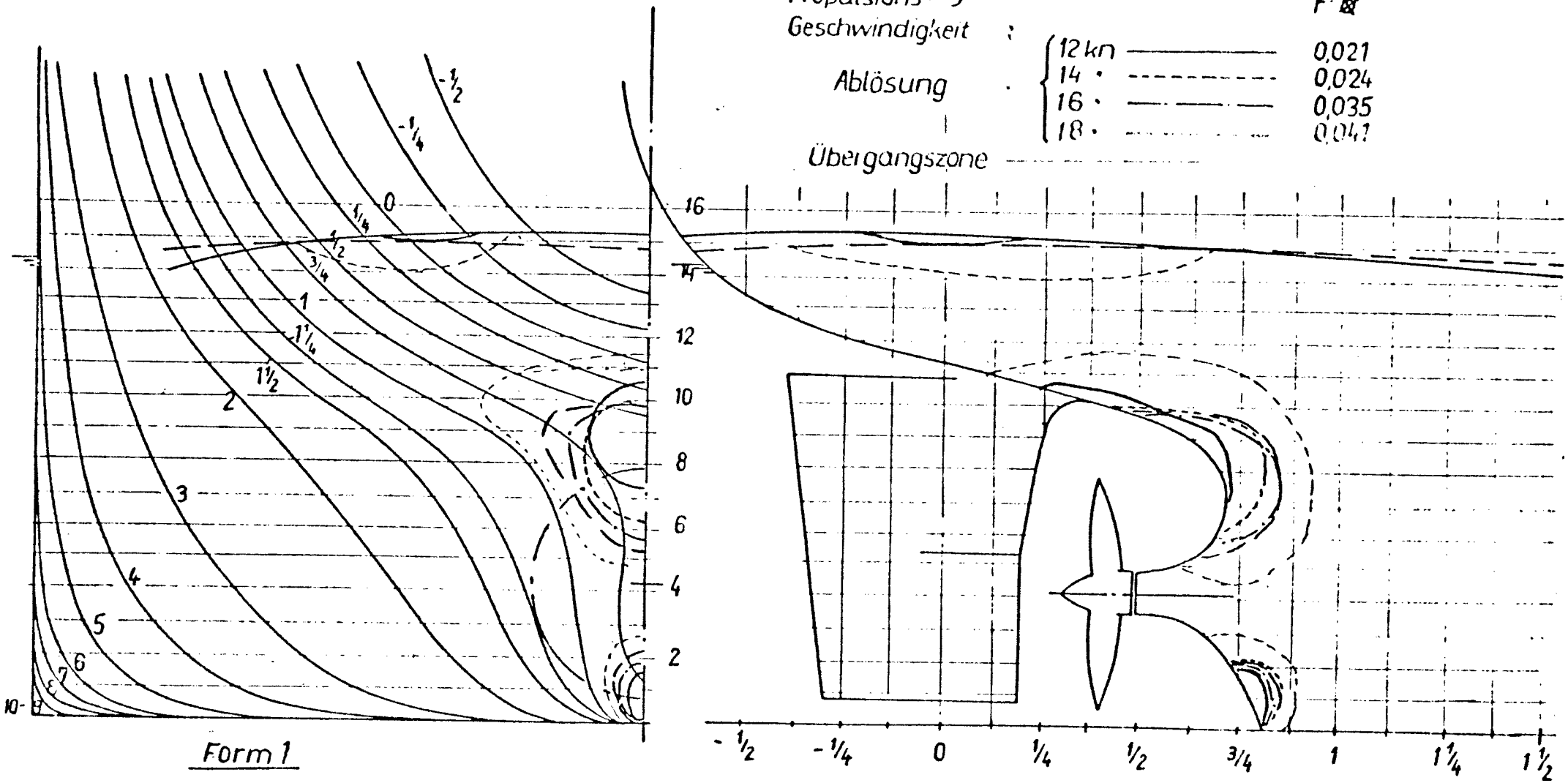
Schwimmlage : CWL

~~Widerstands-~~ Propulsions- } Versuch PROPULSION $\frac{F_{Abl}}{F_{\boxtimes}}$

Geschwindigkeit :

Ablösung . { 12 kn 0,021; 14 · 0,024; 16 · 0,035; 18 · 0,041

Übergangszone



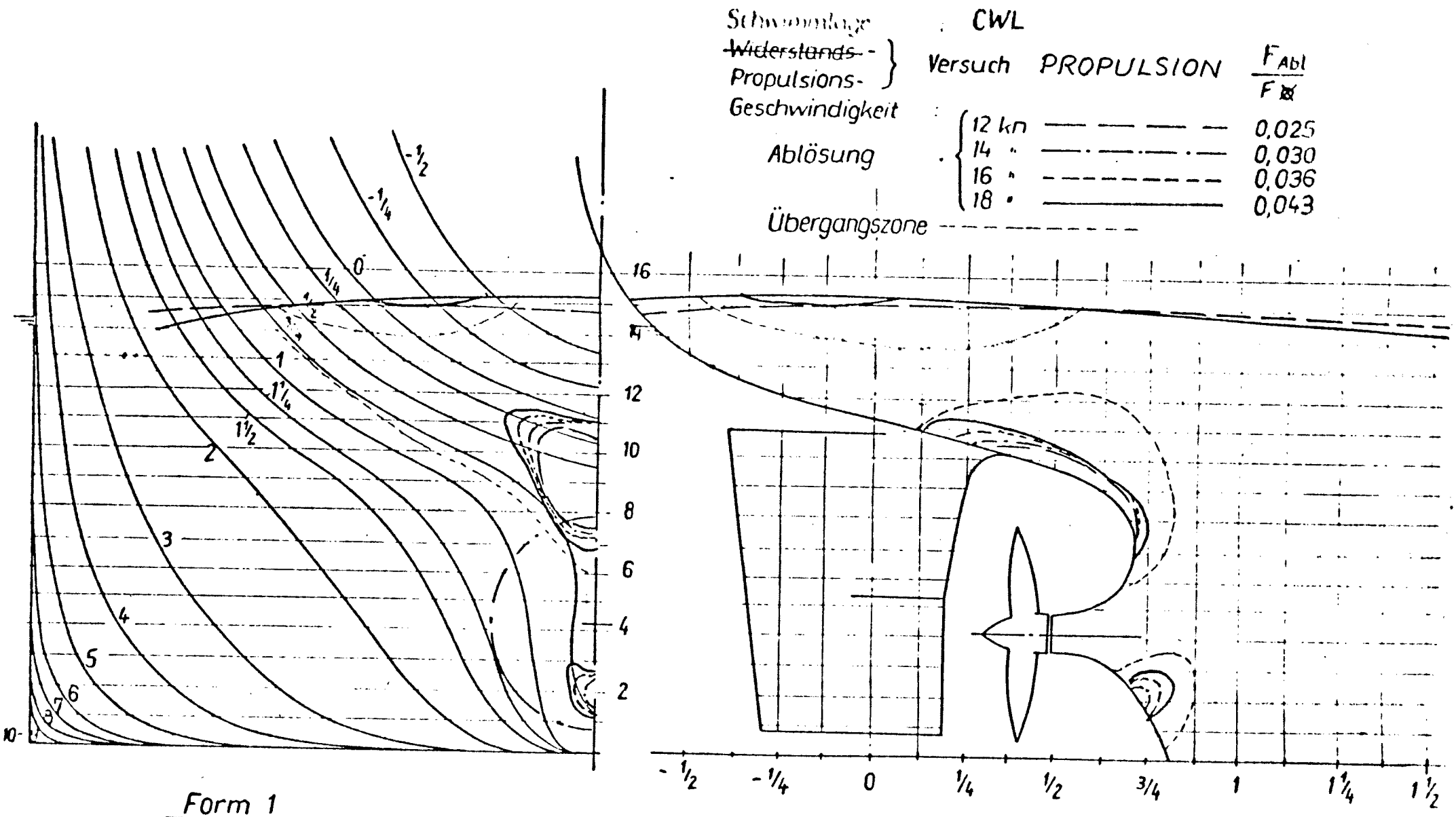
Form 1

CWL

1 : 55

Mod. Nr. 2231	Ablösung am Achterschiff	9

3-10



Form 1

CWL

1:42

Mod. Nr. 2216	Ablösung am Achterschiff	10

Schwimmlage CWL

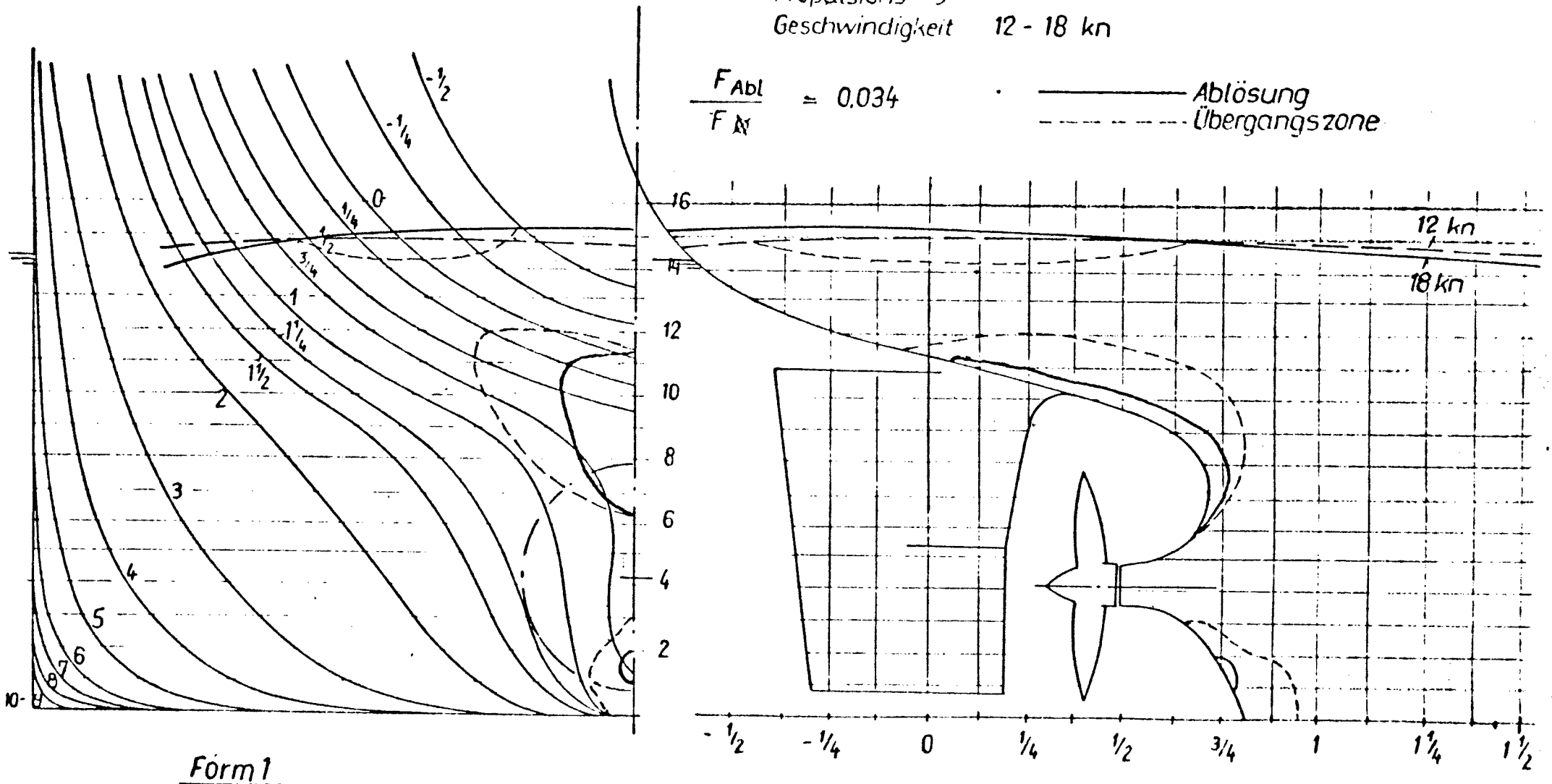
~~Widerstands-~~ Propulsions- } Versuch PROPULSION

Geschwindigkeit 12 - 18 kn

$\frac{F_{Abl}}{F_{\nabla}} = 0.034$

——— Ablösung

- - - - Übergangszone

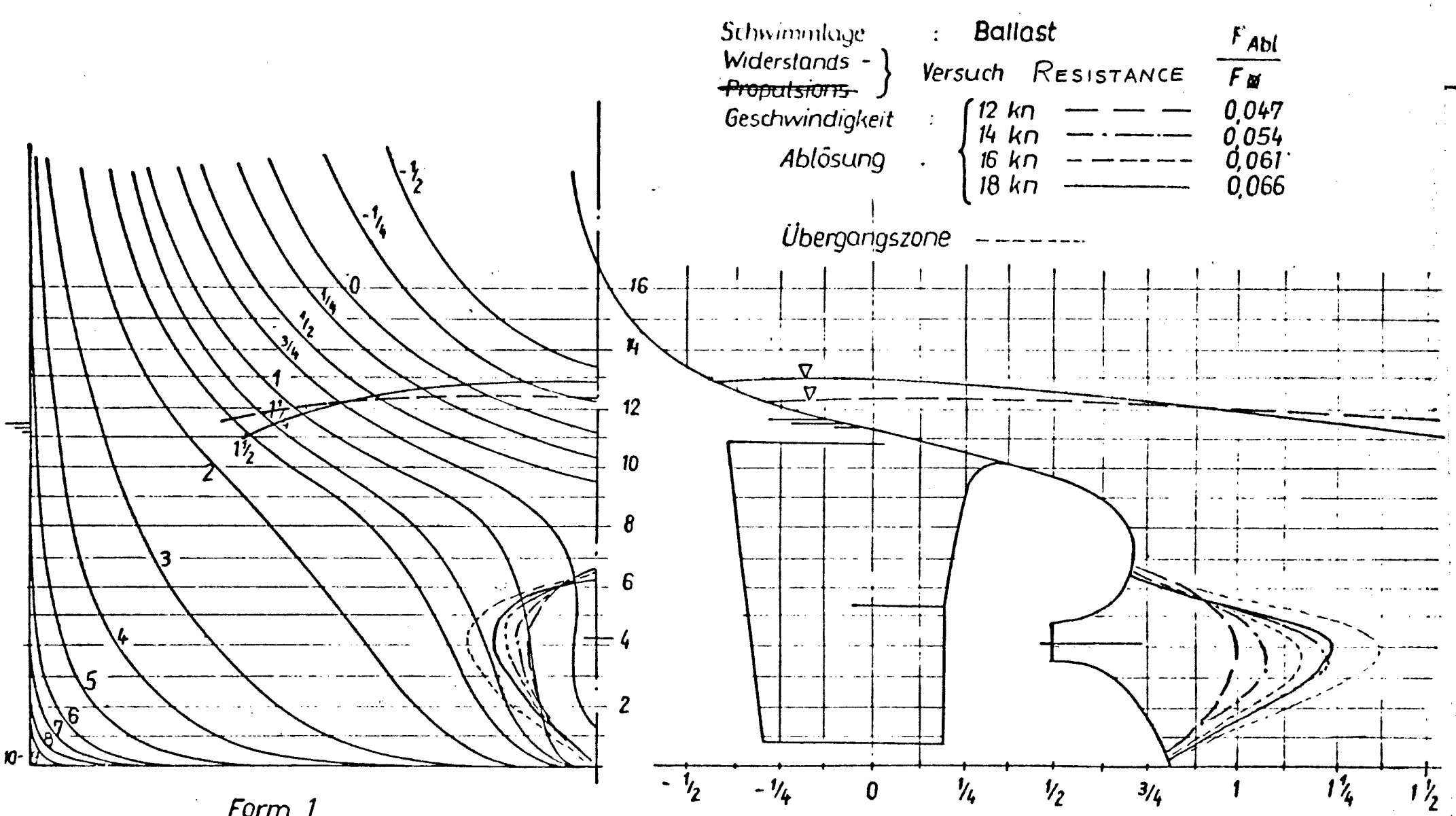


Form 1 / CWL

1 : 33 1/3

Mod. Nr. 2232	Ablösung am Achterschiff	11

3-18



Form 1

Ballast

1 : 55

Mod. Nr. 2231	Ablösung am Achterschiff	12

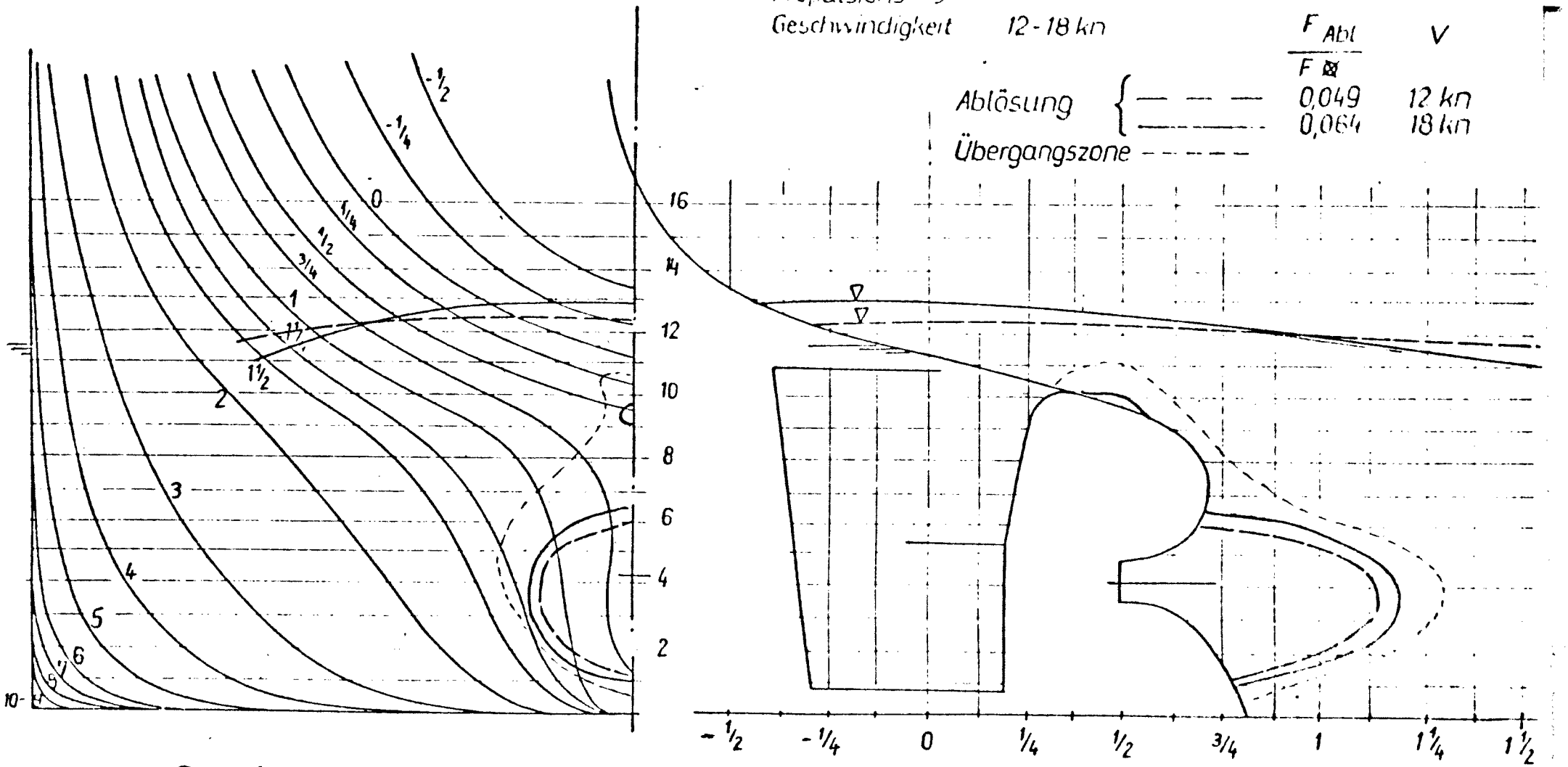
3-13

Schwimmlage : Ballast

Widerstands - } Versuch RESISTANCE
~~Propulsions-~~

Geschwindigkeit 12 - 18 kn

	F_{Abl} / $F_{\boxtimes}$	V
Ablösung — — —	0,049	12 kn
Ablösung ———	0,064	18 kn
Übergangszone - - - - -		



Form 1

Ballast

1 : 42

Mod. Nr. 2216	Ablösung am Achterschiff	13

3-14

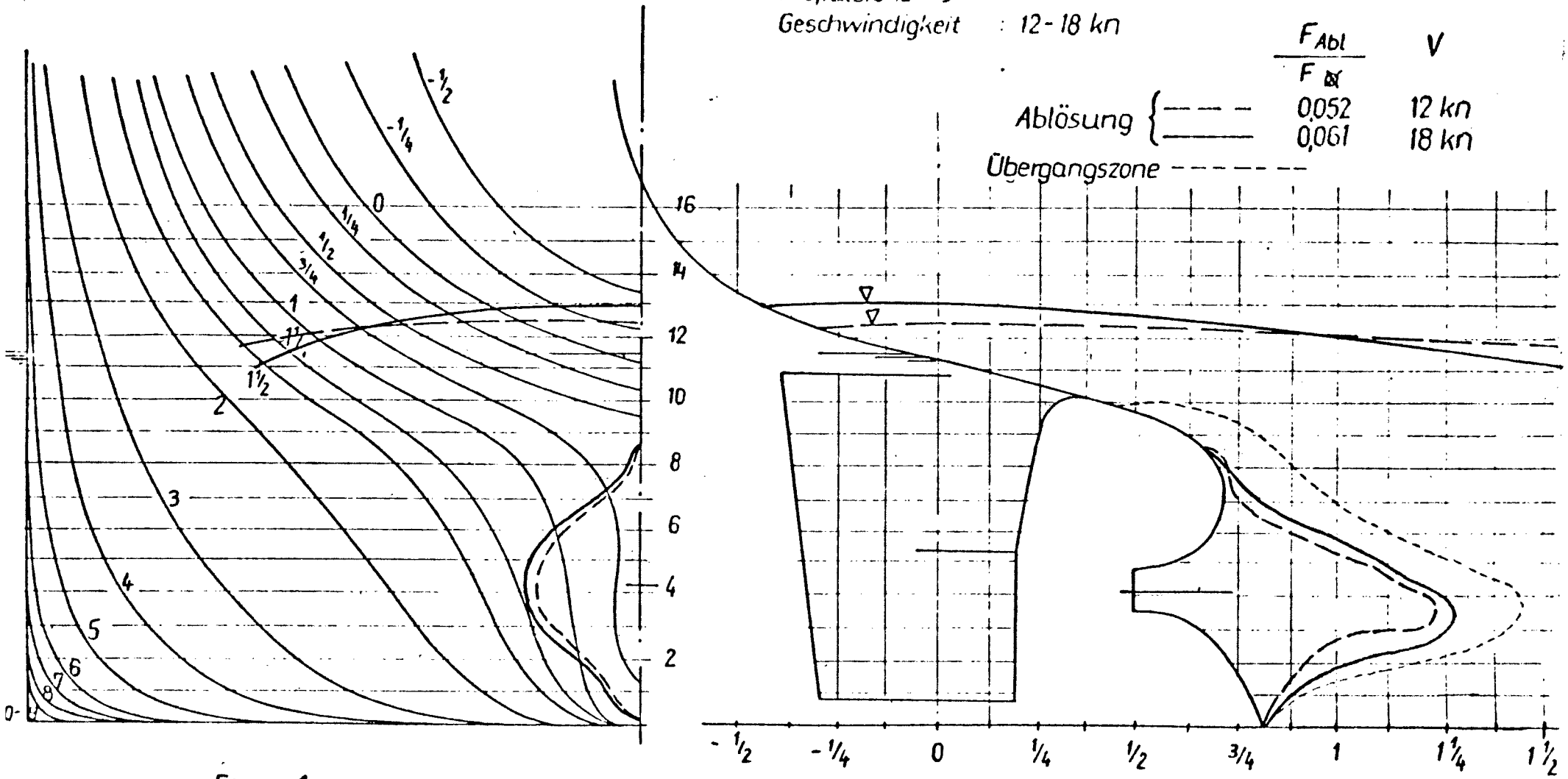
Schwimmlage : Ballast

Widerstands - } Versuch RESISTANCE
~~Propulsions-~~

Geschwindigkeit : 12 - 18 kn

	F_{Abl} / $F_{\boxtimes}$	V
Ablösung ----	0,052	12 kn
Ablösung ——	0,061	18 kn

Übergangszone - - - - - -



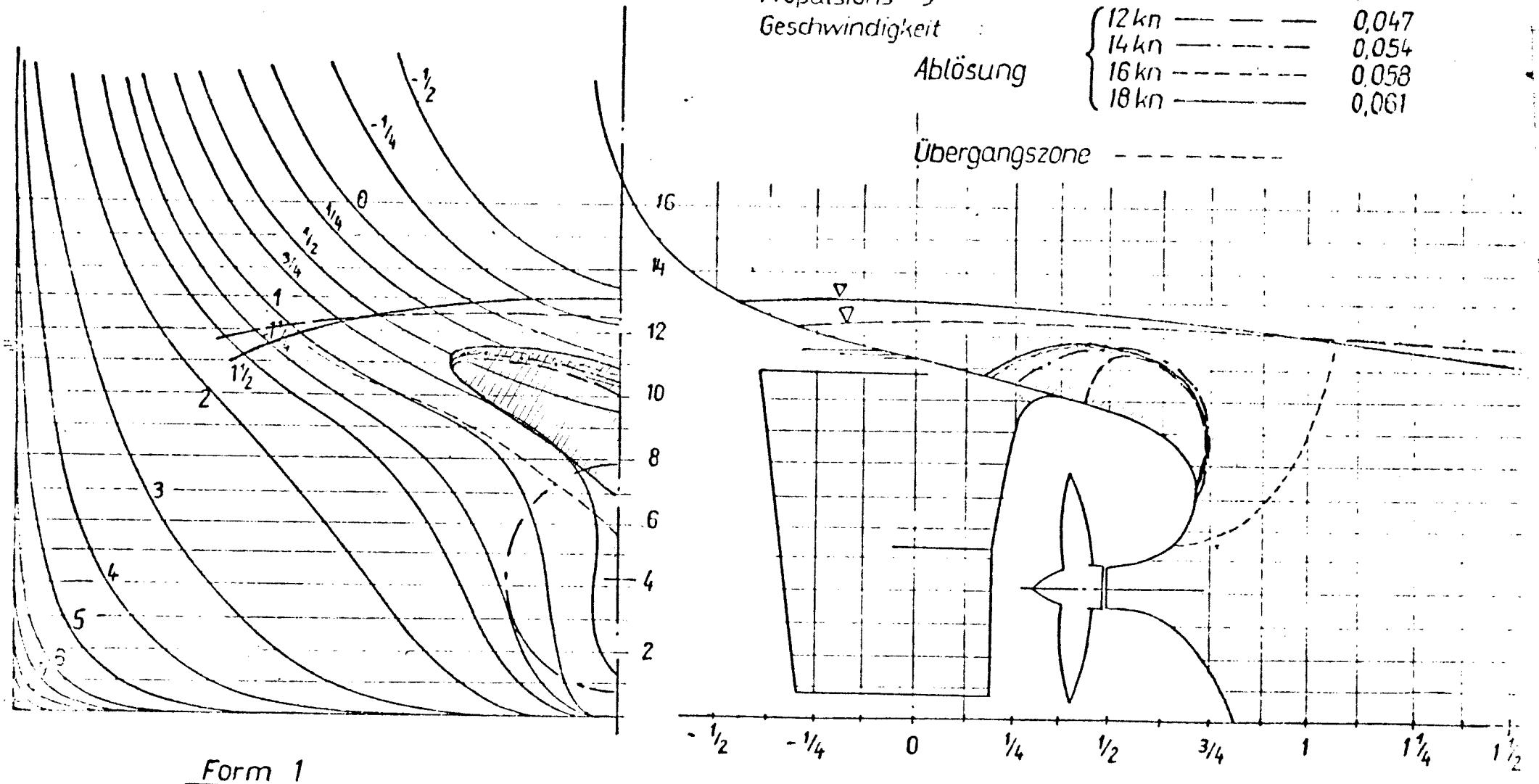
Form 1

Ballast

1 : 33 1/3

Mod. Nr. 2232	Ablösung am Achterschiff	14

3-15



Form 1

Ballast

1 : 55

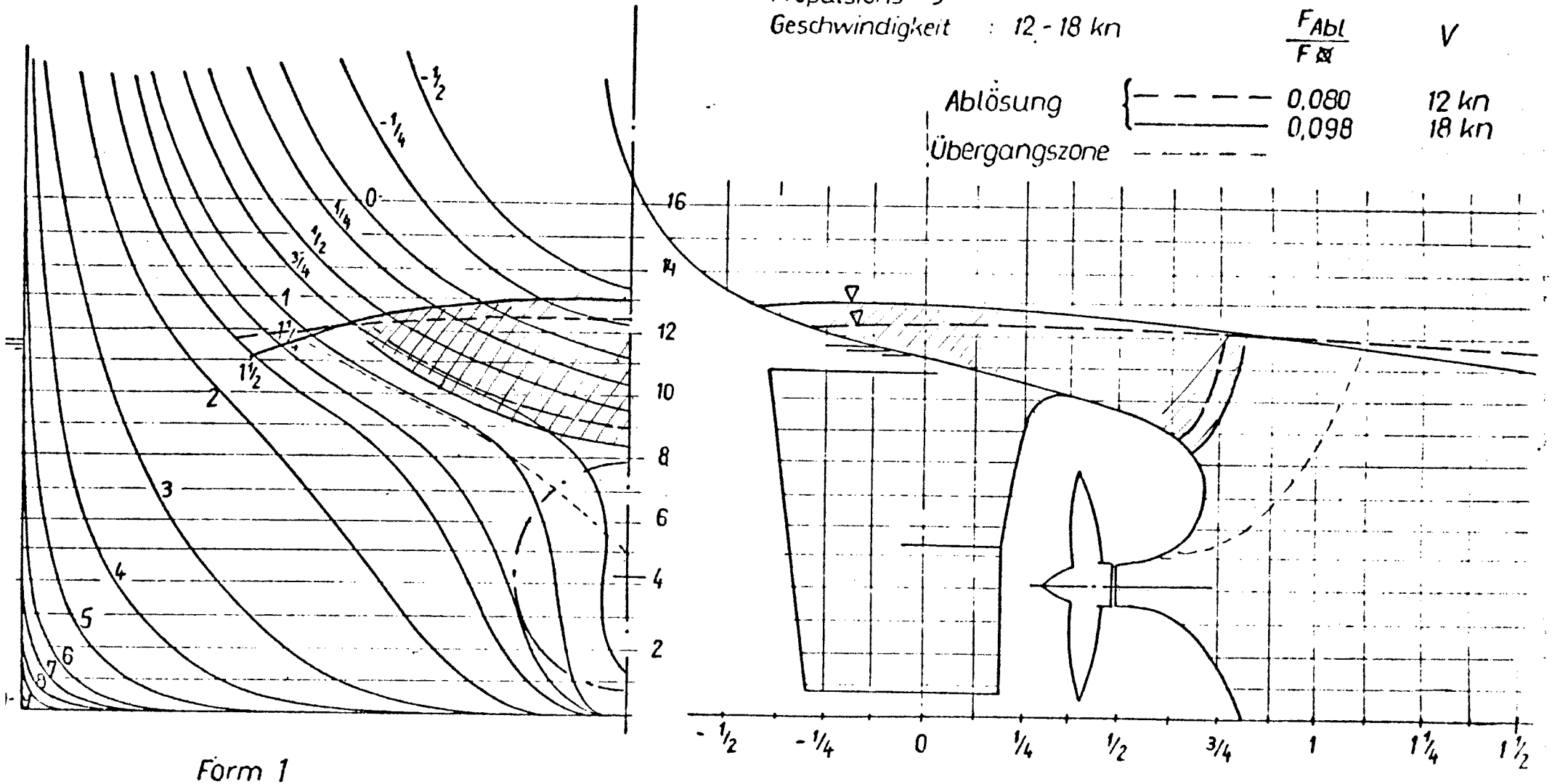
Mod. Nr. 2231	Ablösung am Achterschiff	15

Schwimmlage : Ballast

~~Widerstands-~~ } Versuch PROPULSION
Propulsions-

Geschwindigkeit : 12 - 18 kn

	$\frac{F_{Abl}}{F_{\boxtimes}}$	V
Ablösung	0,080	12 kn
	0,098	18 kn
Übergangszone		



Form 1

Ballast

1 : 42

Mod. Nr. 2216	Ablösung am Achterschiff	16

3-17

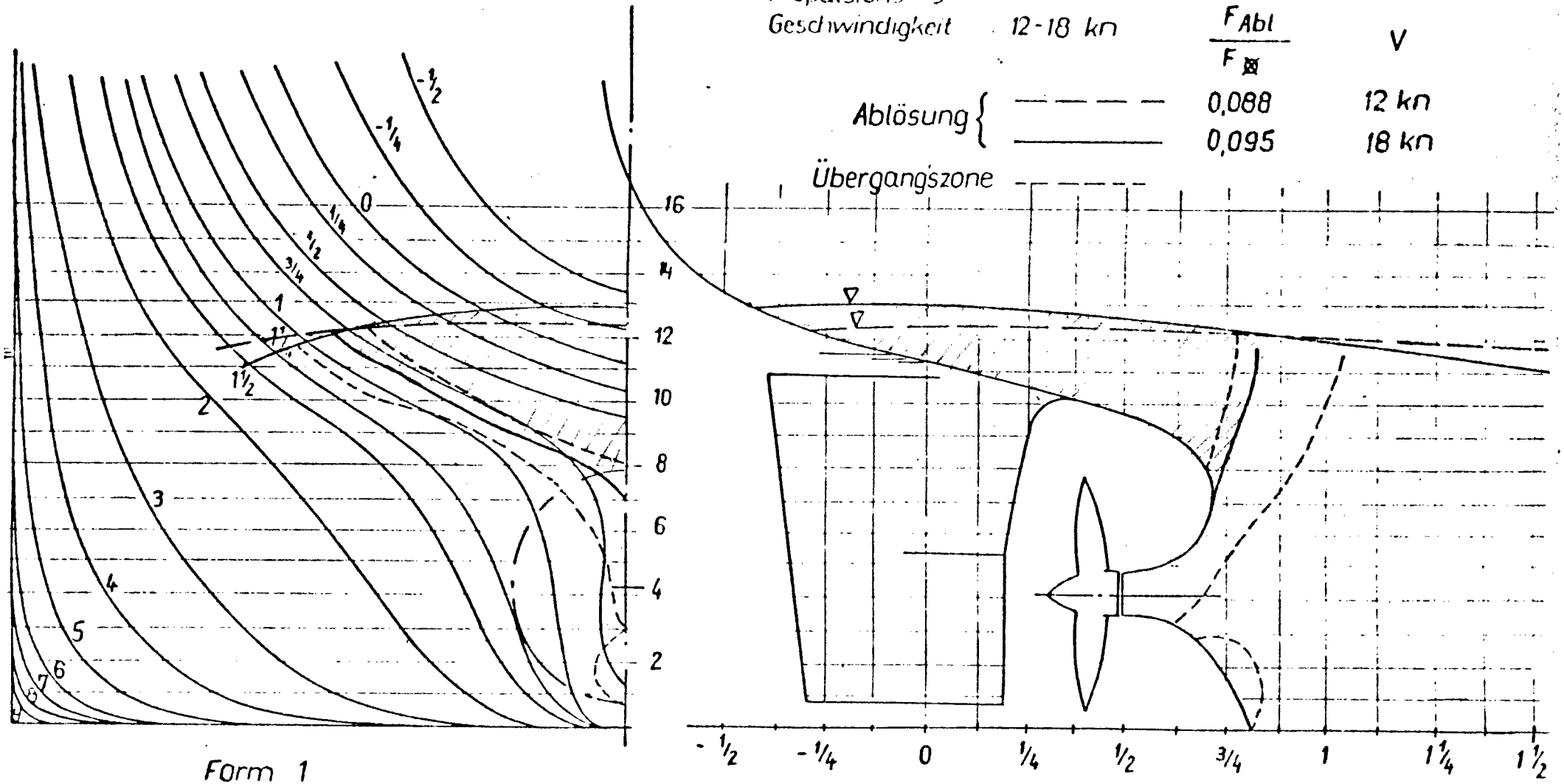
Schwimmlage : Ballast

~~Widerstands~~ - } Versuch PROPULSION
Propulsions-

Geschwindigkeit 12-18 kn

	$\frac{F_{Abl}}{F_{\boxtimes}}$	V
Ablösung { — — — —	0,088	12 kn
———	0,095	18 kn

Übergangszone - - - - -



Form 1

Ballast

1 : 33 1/3

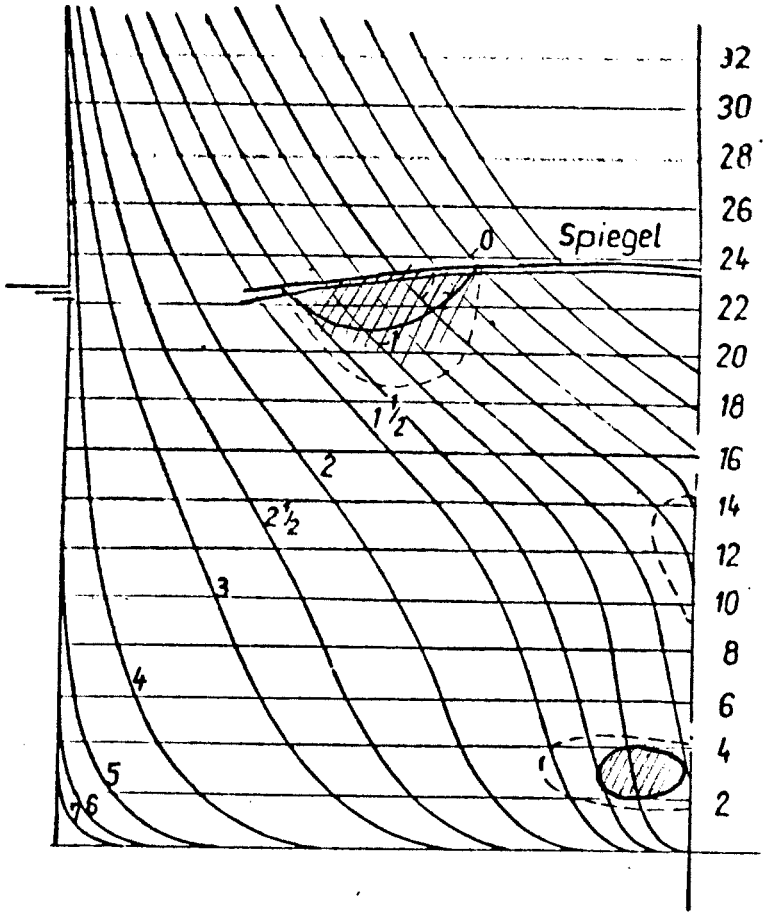
Mod. Nr. 2232	Ablösung am Achterschiff	17

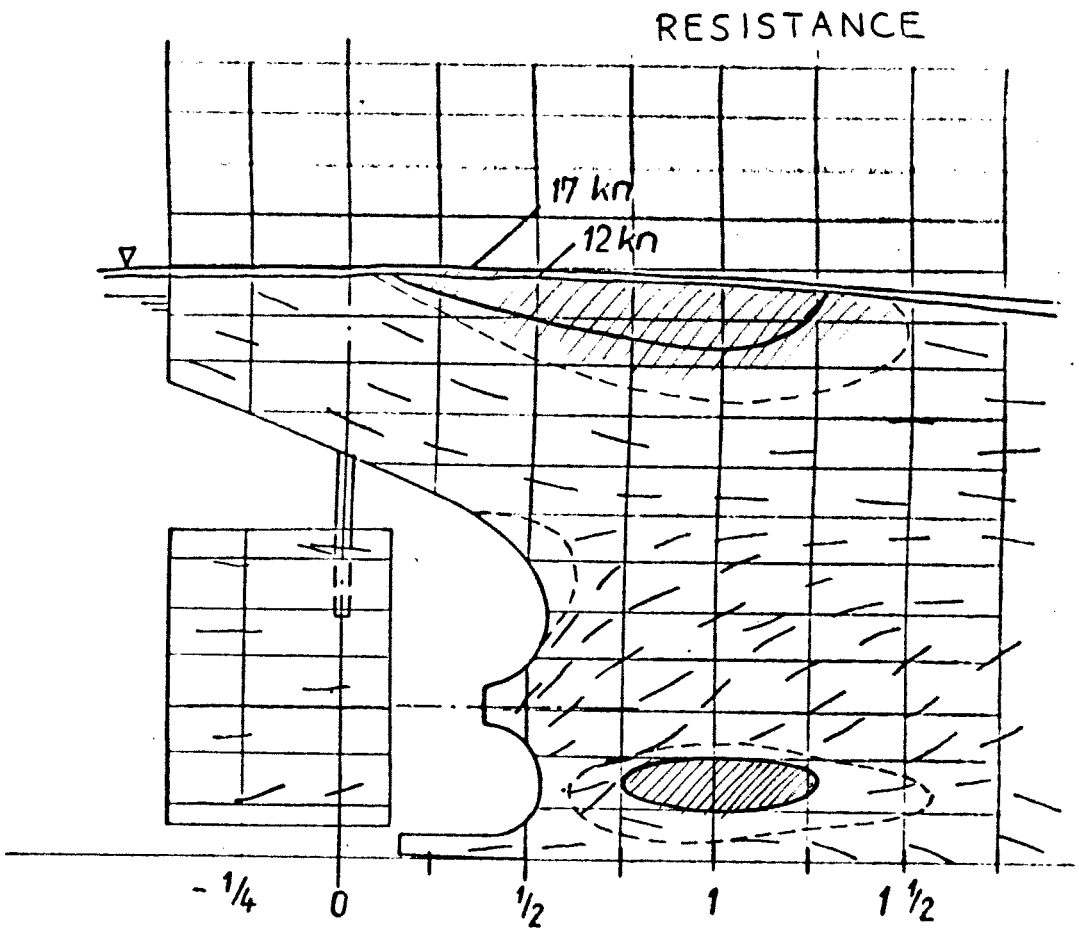
3-18

Modellmaßstab λ = 44
Widerstands - } Versuch
~~Propulsions~~ -

$\frac{F_{Abl}}{F_{\boxtimes}} = 0{,}036$

V = 12-17 kn





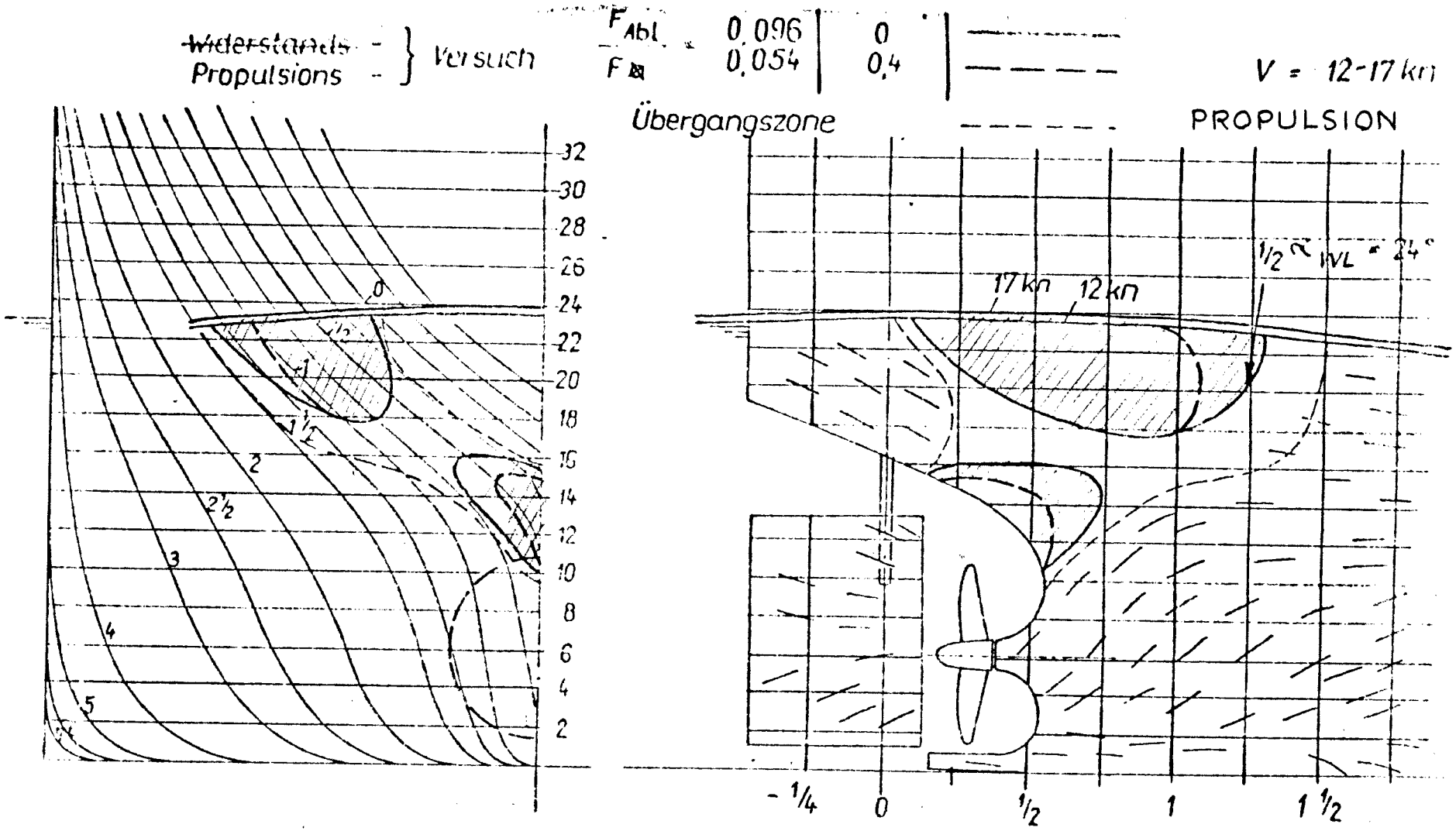
Form 2
CWL

——— Ablösung
– – – – Übergangszone

1:44

Mod. Nr. 2239	Ablösung am Achterschiff	18

3-19



Form 2

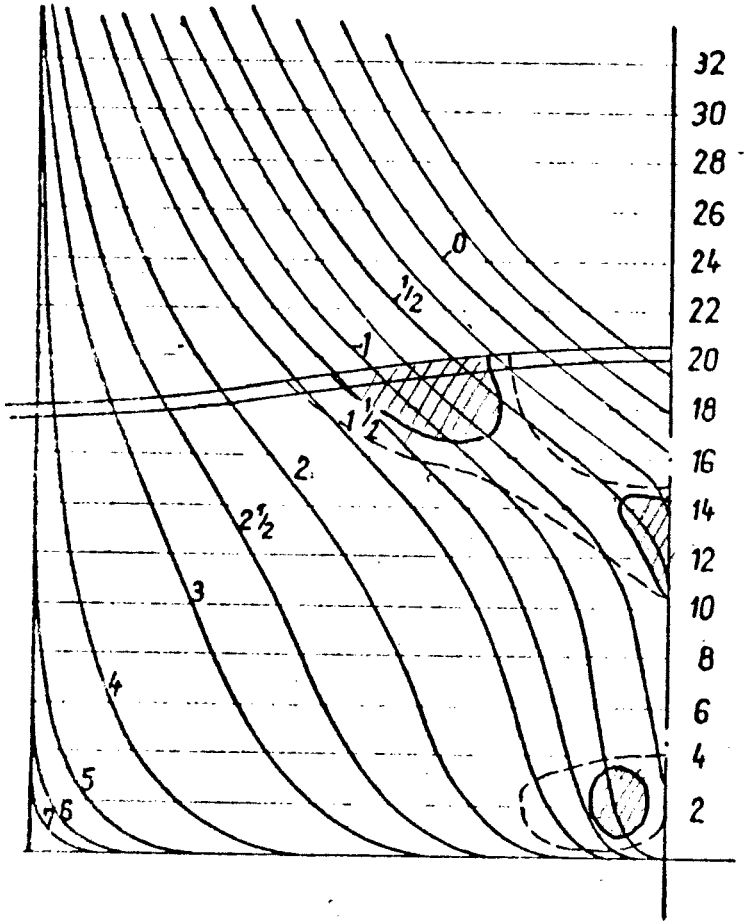
CWL

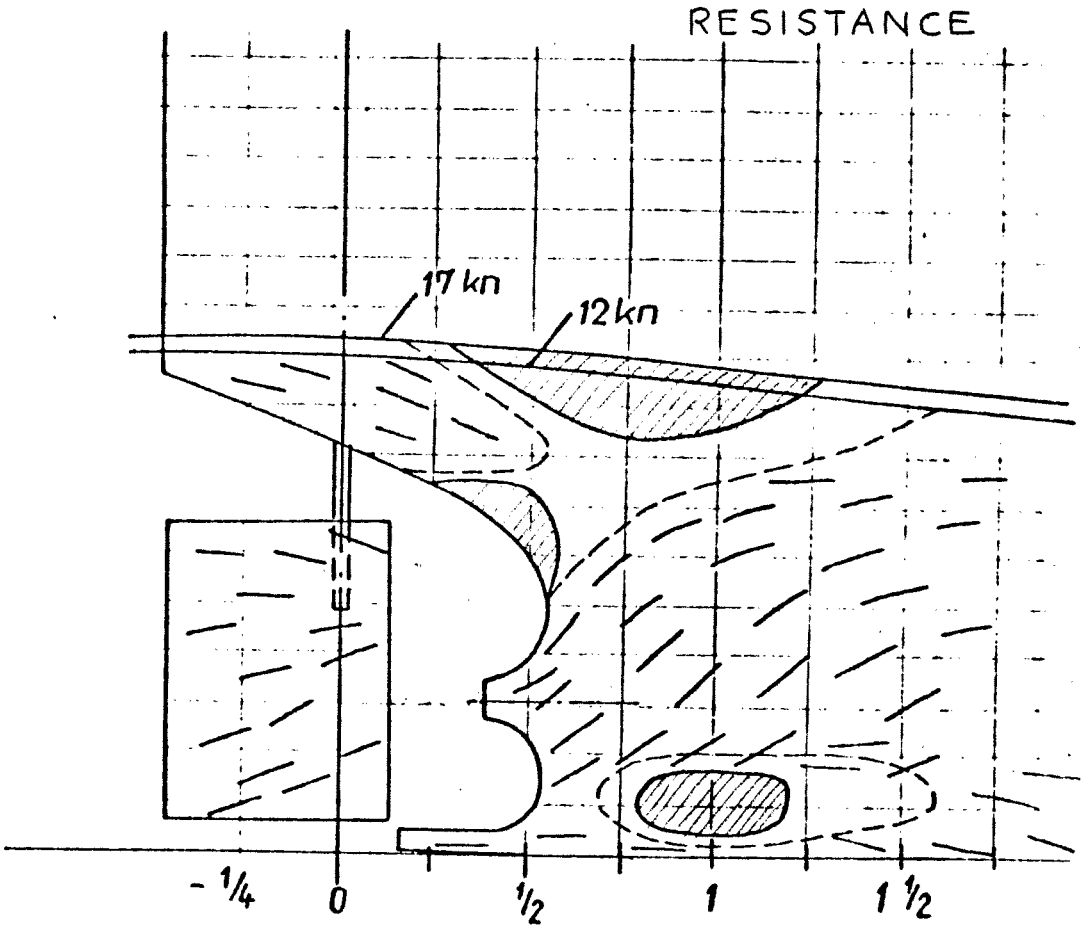
1 : 44

Mod. Nr. 2239	Ablösung am Achterschiff	19

Widerstands - } Versuch
~~Propulsions~~ -

$\frac{F_{Abl}}{F_{\otimes}} = 0{,}064$ V = 12 - 17 kn





Form 2
Ballast

——— Ablösung
- - - - Übergangszone

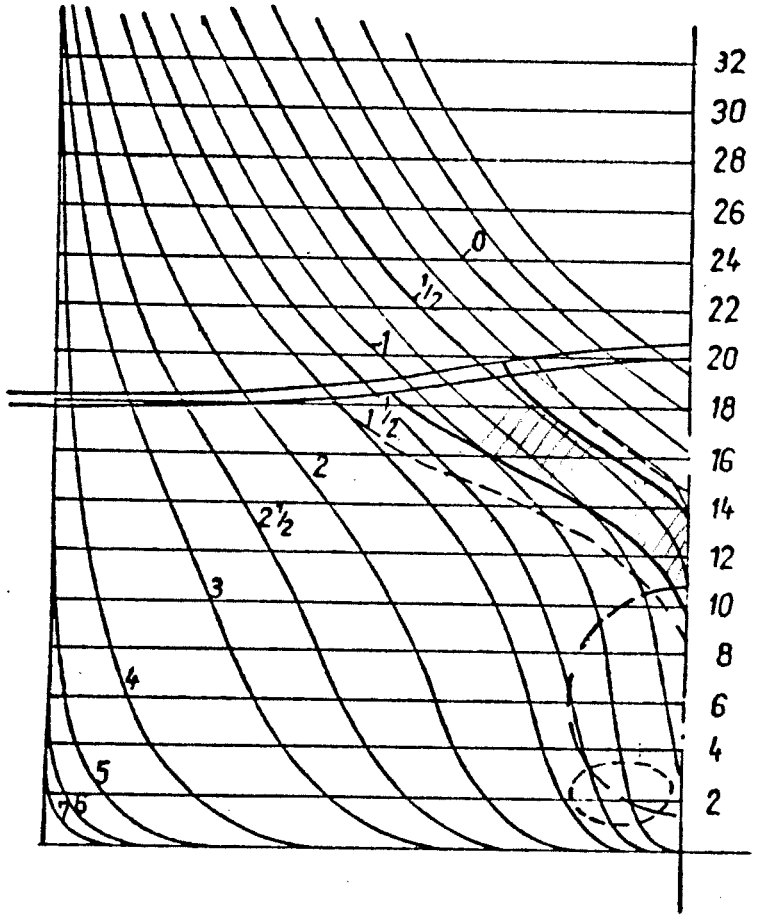
1 : 44

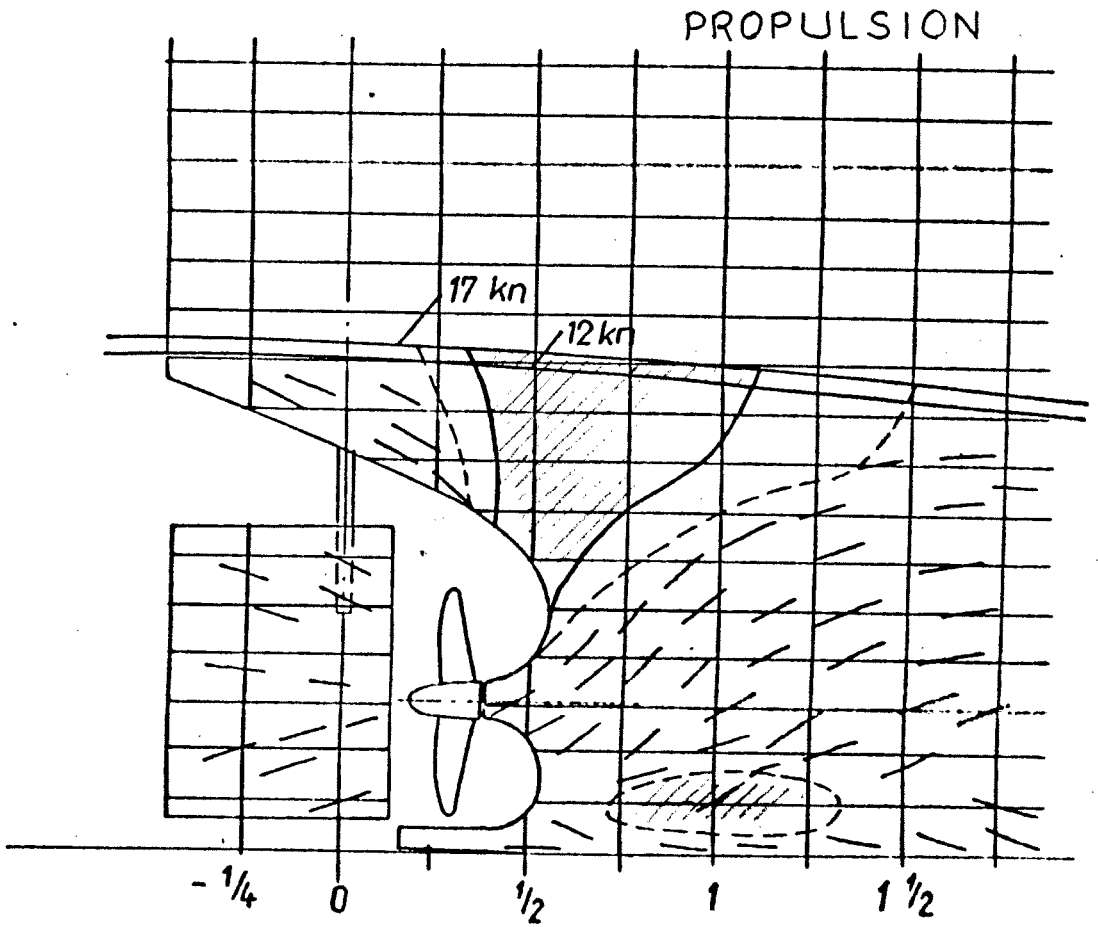
Mod. Nr. 2239	Ablösung am Achterschiff	20

3-21

Modellmaßstab $\lambda = 44$
~~Widerstands~~ - } Versuch
Propulsions -

$\frac{F_{Abl}}{F_{\boxtimes}} = 0{,}065$ $V = 12 - 17$ kn





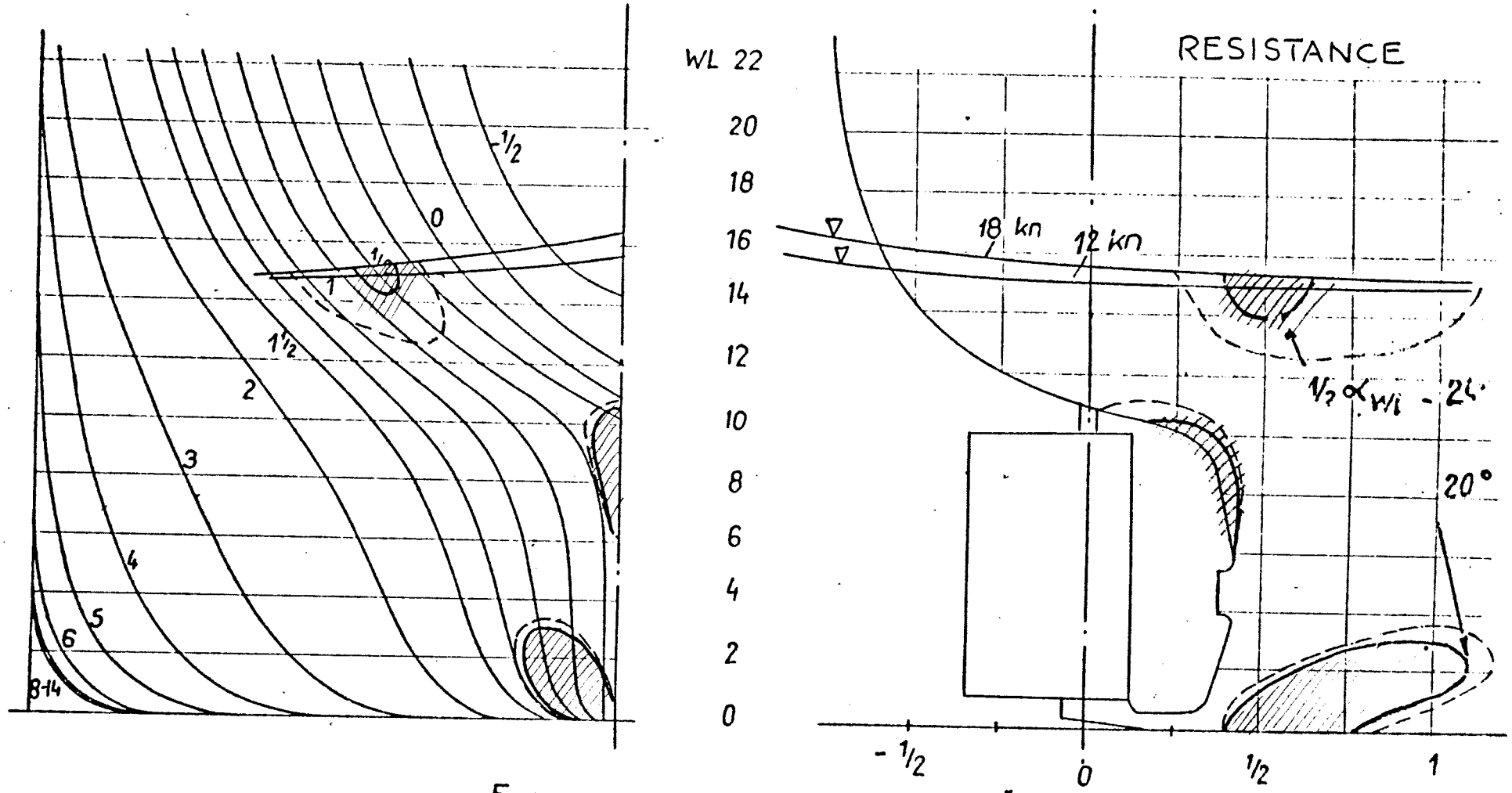
Form 2
Ballast

——— Ablösung
— — — Übergangszone

1 : 44

Mod. Nr. 2239	Ablösung am Achterschiff	21

3-22



Form 3
CWL

$\frac{F_{Abl.}}{F_{\otimes}} = 0{,}031$

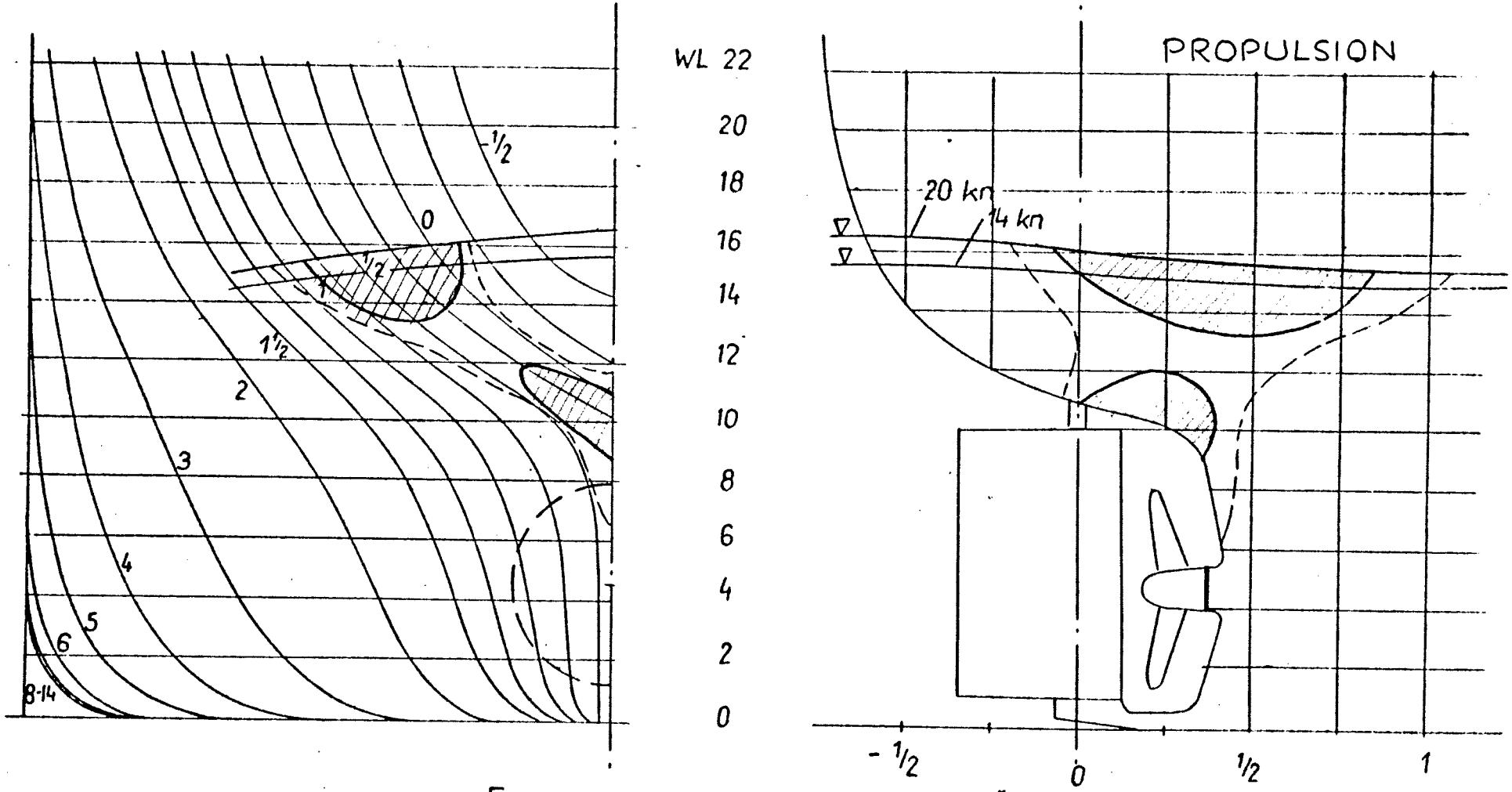
Ablösung
Übergangszone

V= 12-18 kn

1:40

Mod. Nr. 2184	Ablösung am Achterschiff	22

3-23



Form 3 / CWL

$\frac{F_{Abl.}}{F_{\otimes}} = 0{,}049$

Ablösung

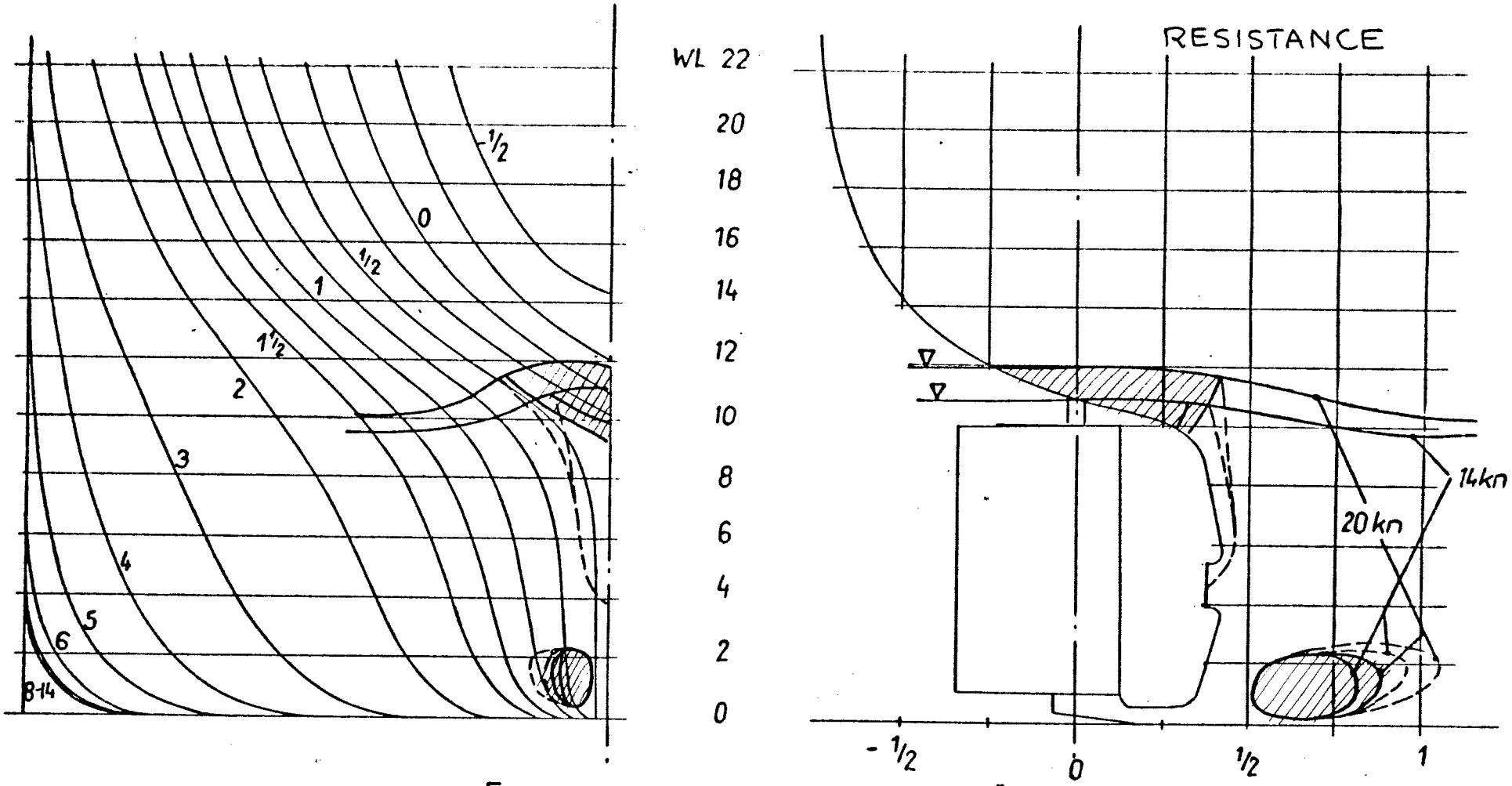
Übergangszone

V = 14 - 20 kn

1 : 40

Mod. Nr. 2184	Ablösung am Achterschiff	23

3-24



Form 3
Ballast

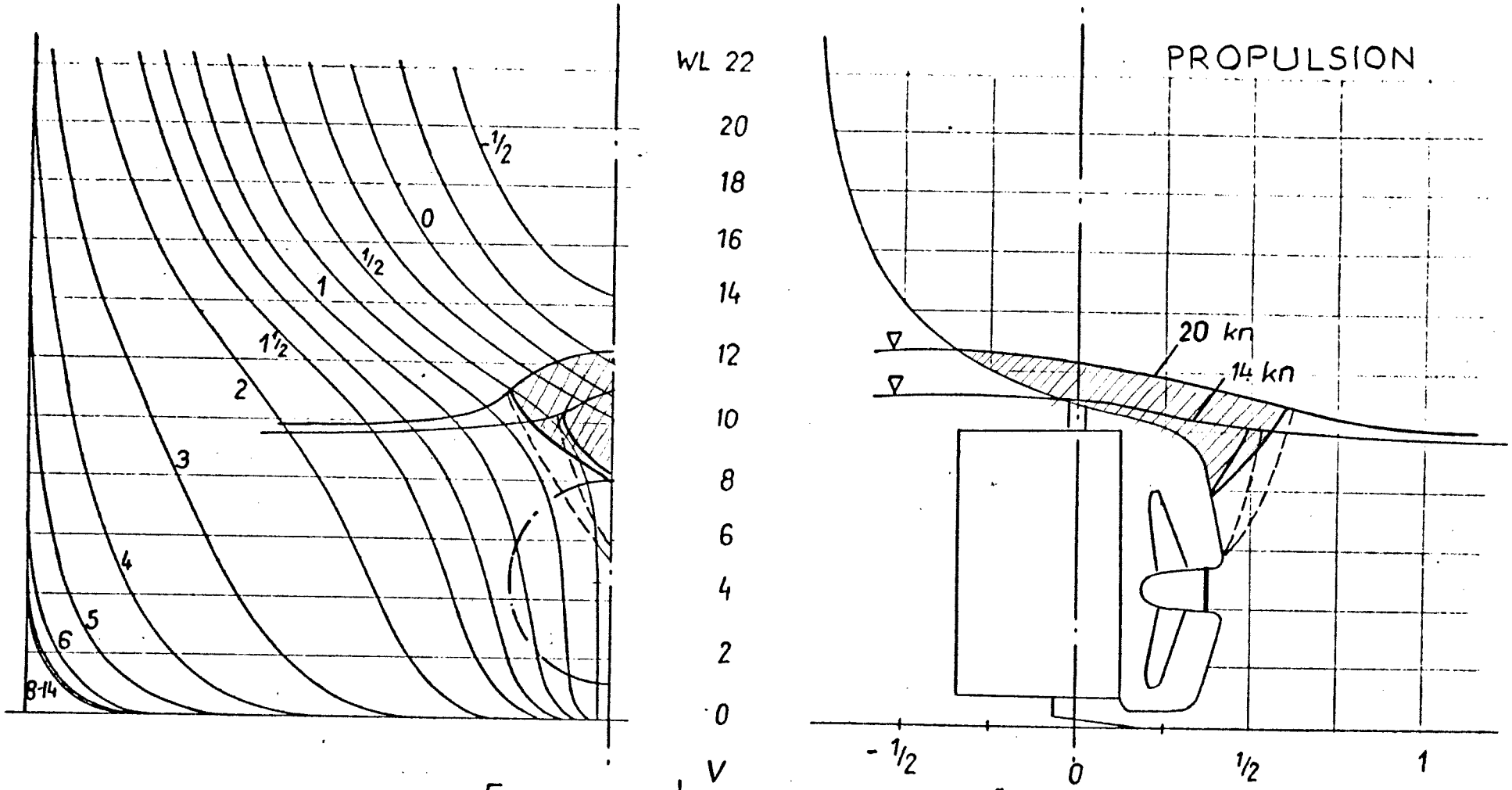
$$\frac{F_{Abl.}}{F_{\boxtimes}} = \begin{cases} 0{,}045 & 20\,kn \\ 0{,}022 & 14\,kn \end{cases}$$

——— Ablösung
– – – – – Übergangszone

V = 14 - 20 kn

1 : 40

Mod. Nr 2184	Ablösung am Achterschiff	24



Form 3
Ballast

$\frac{F_{Abl.}}{F_{\otimes}} = \frac{0{,}054}{0{,}020}$ | 20 kn / 14 kn

——— Ablösung
- - - - Übergangszone

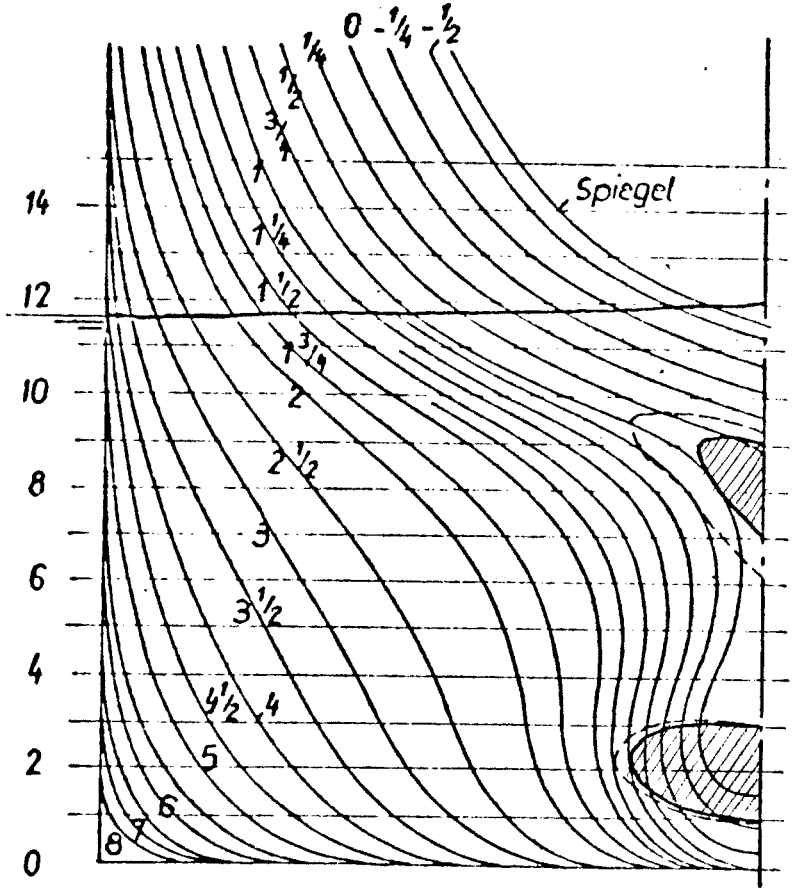
V = 14-20 kn

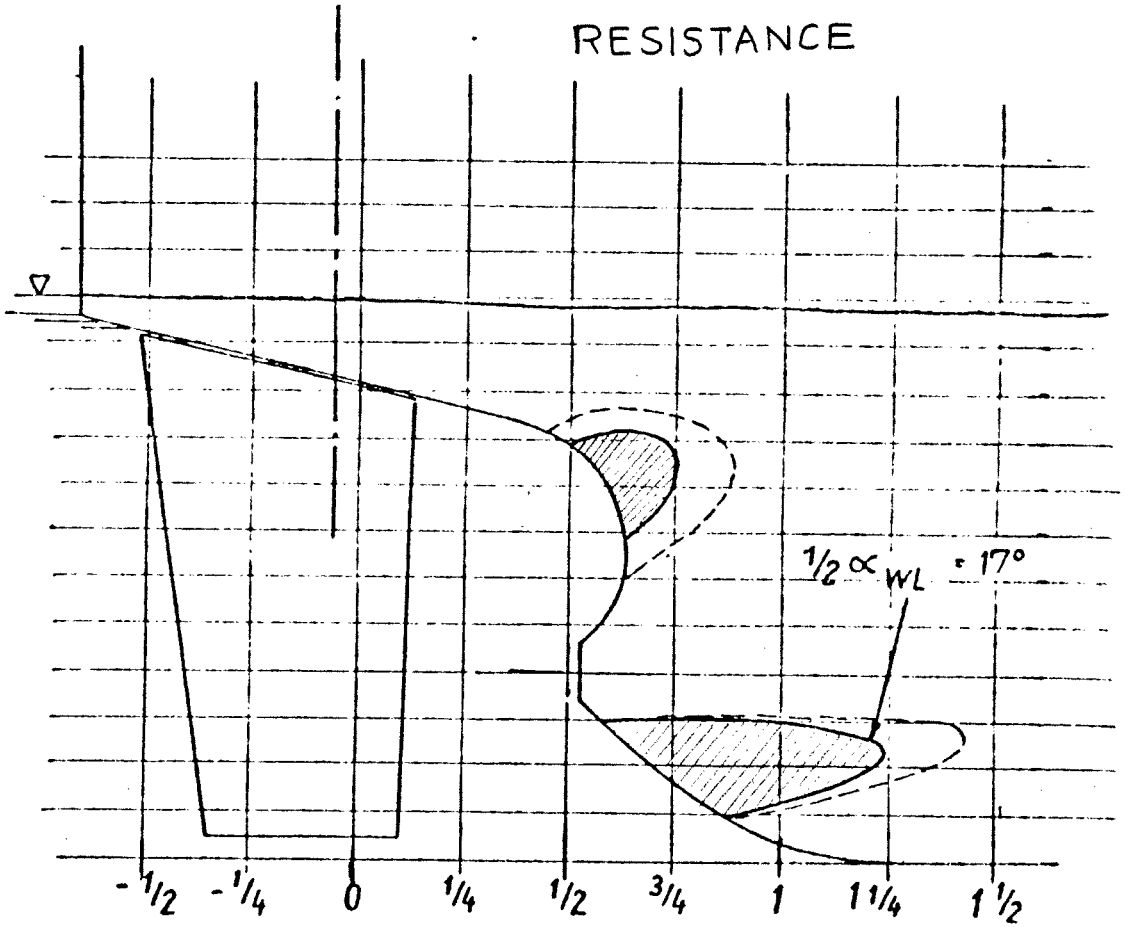
1 : 40

Mod. Nr. 2184	Ablösung am Achterschiff	25

3-26

$$\frac{F_{Abl}}{F_{\boxtimes}} = 0{,}0384$$





Form 4 / CWL

——— Ablösungsbereich
– – – – – Übergangszone

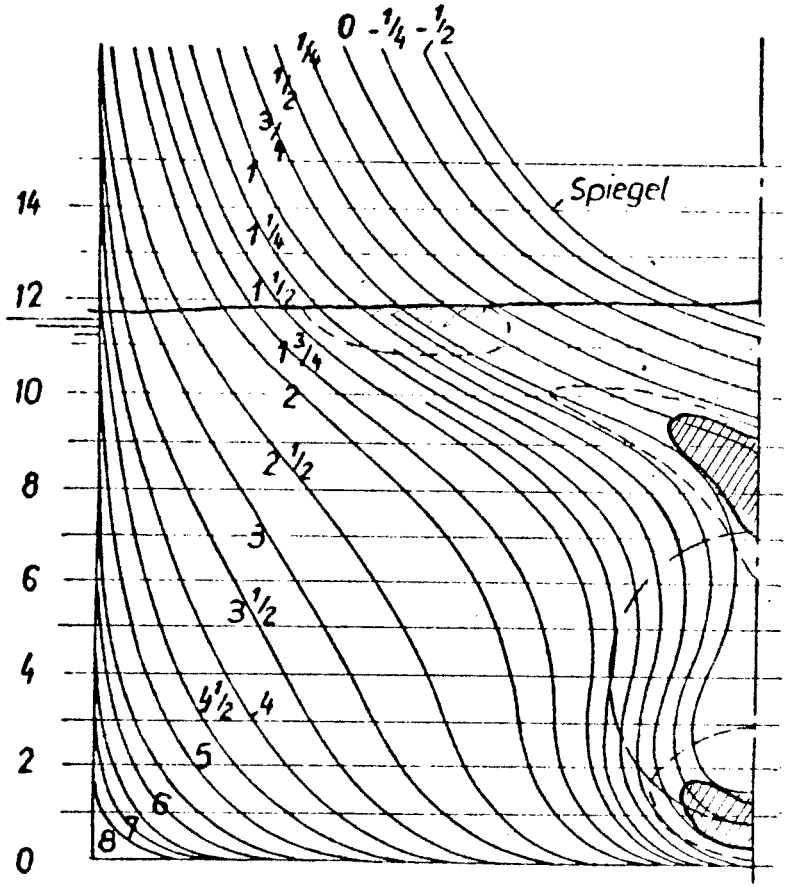
} V = 13 - 17 kn

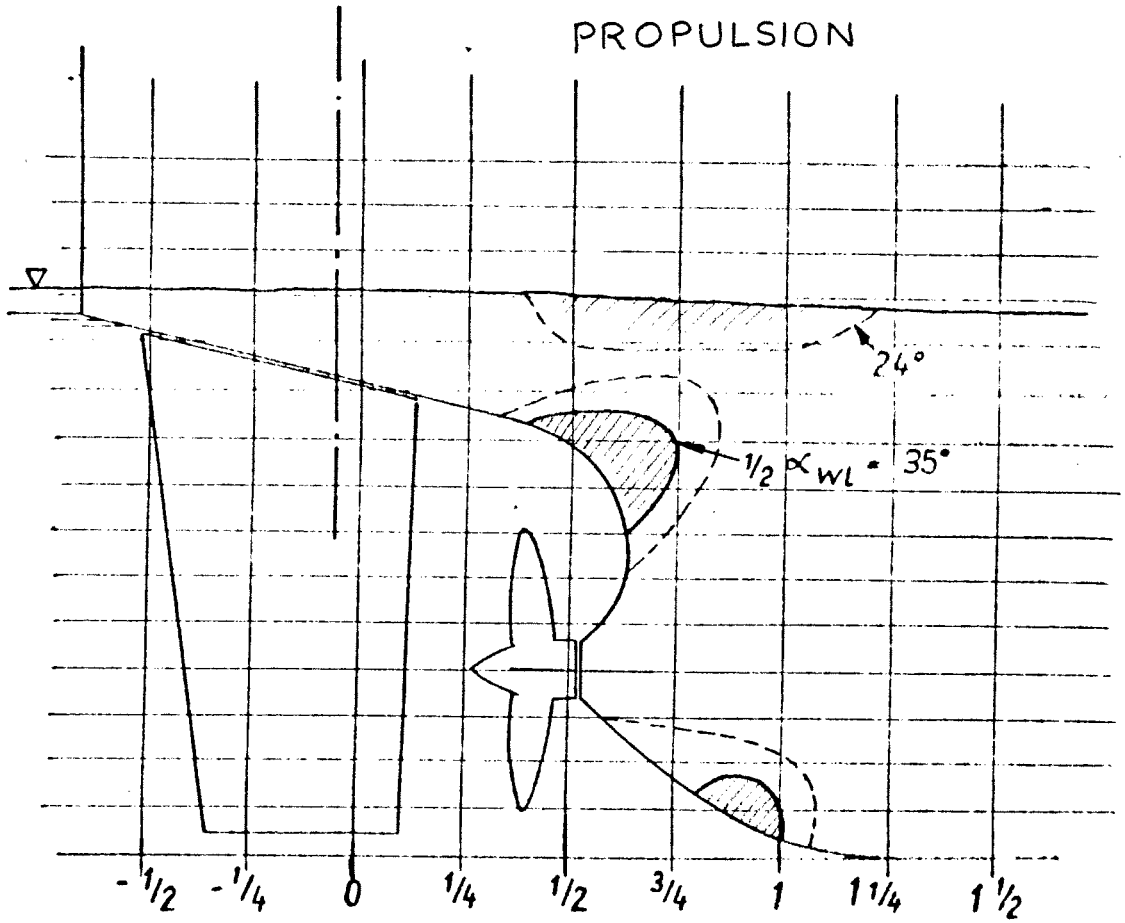
1 : 27 1/2

Mod. Nr. 2209	Ablösung am Achterschiff	26

3-27

$$\frac{F_{Abl}}{F_{\boxtimes}} = 0{,}0236$$





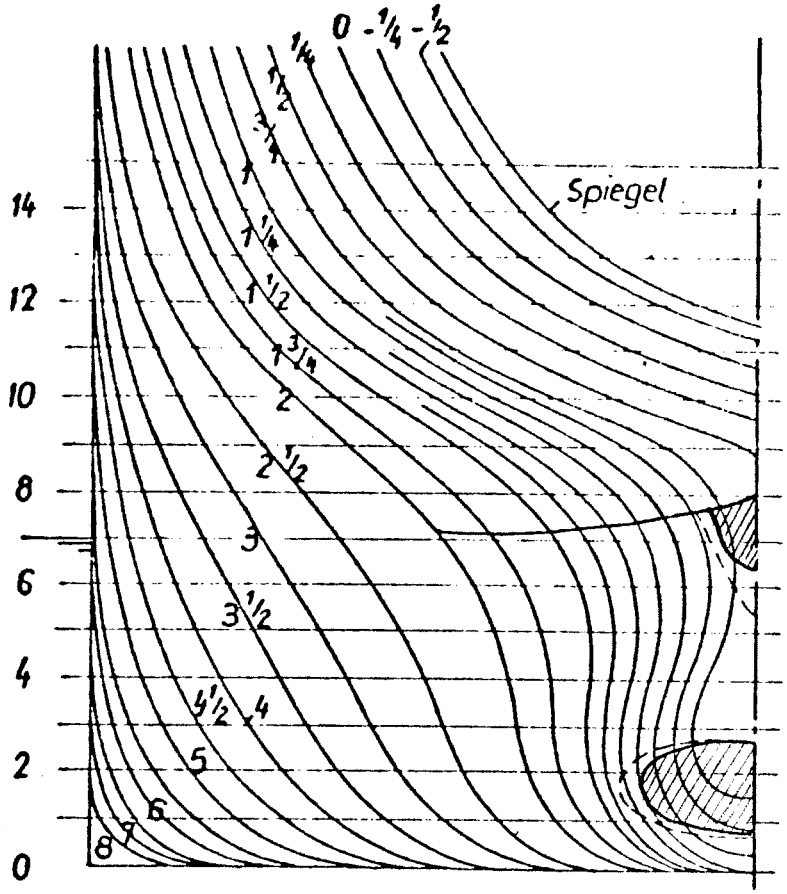
——— Ablösungsbereich
– – – – Übergangszone
} V = 13 - 17 kn

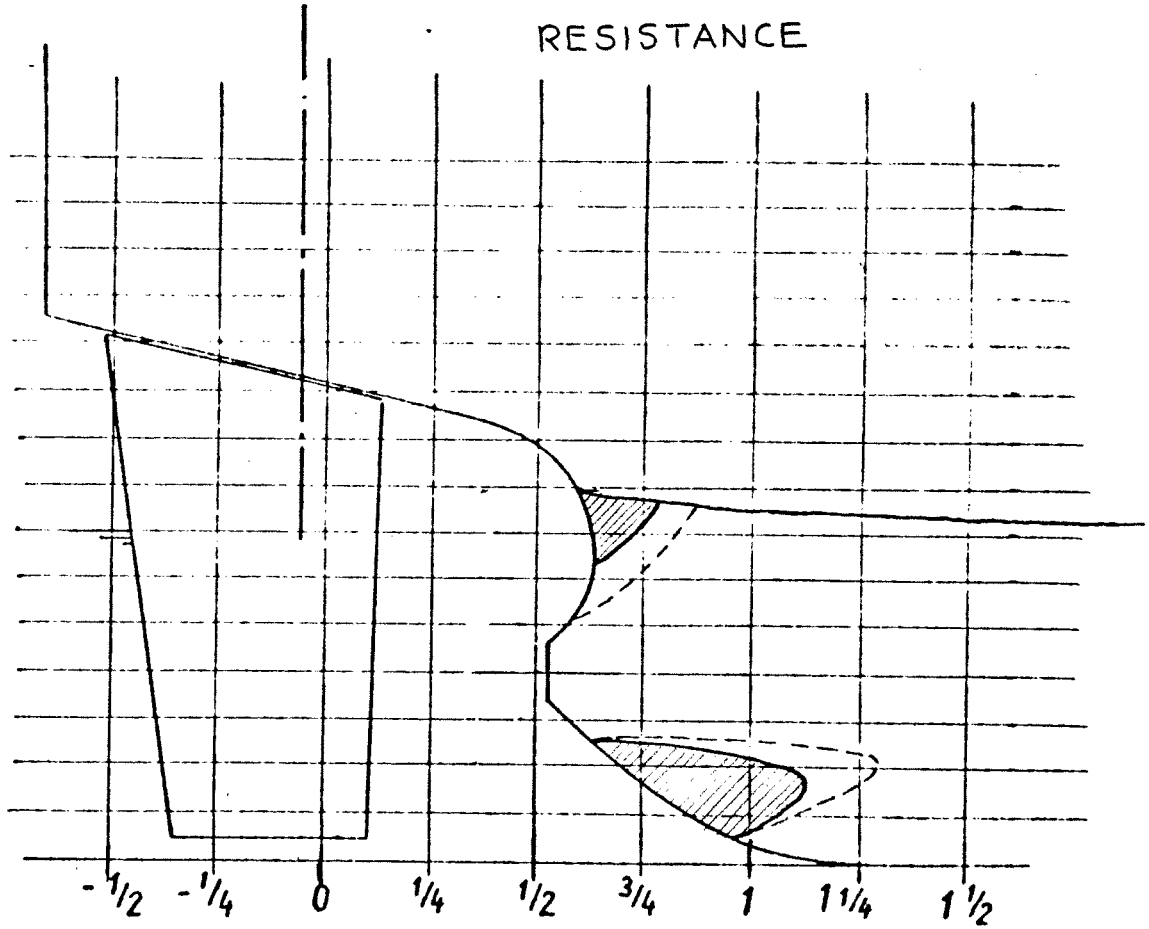
Form 4
CWL

1 : 27 1/2

Mod. Nr. 2209	Ablösung am Achterschiff	27

$$\frac{F_{Abl.}}{F_{\otimes}} = 0{,}0496$$





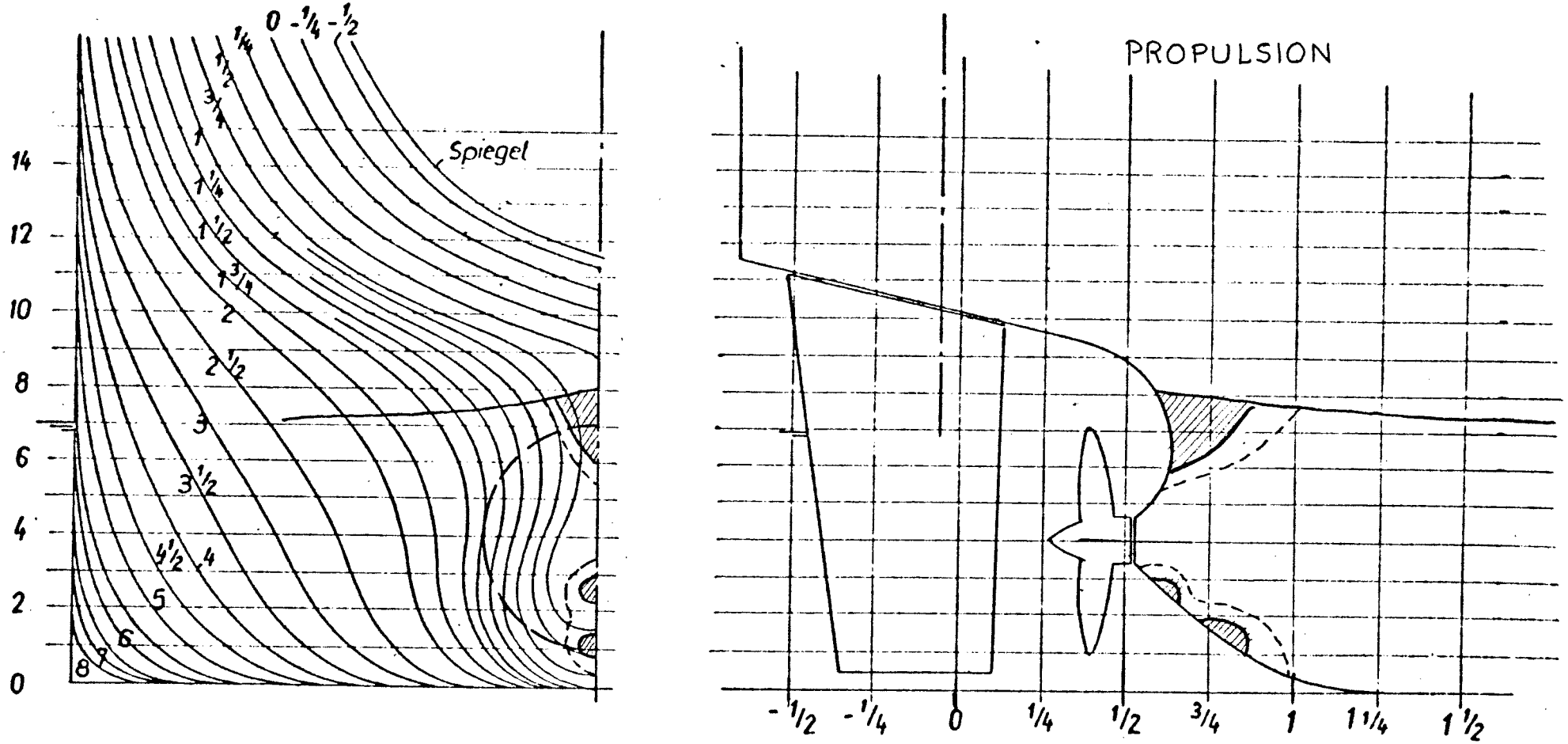
Form 4 / Ballast

1 : 27½

——— Ablösungsbereich
- - - - Übergangszone
} 14 - 18 kn

Mod. Nr. 2209	Ablösung am Achterschiff	28

$$\frac{F_{Abl}}{F_{\boxtimes}} = 0{,}0235$$



Form 4

Ballast

1 : 27 1/2

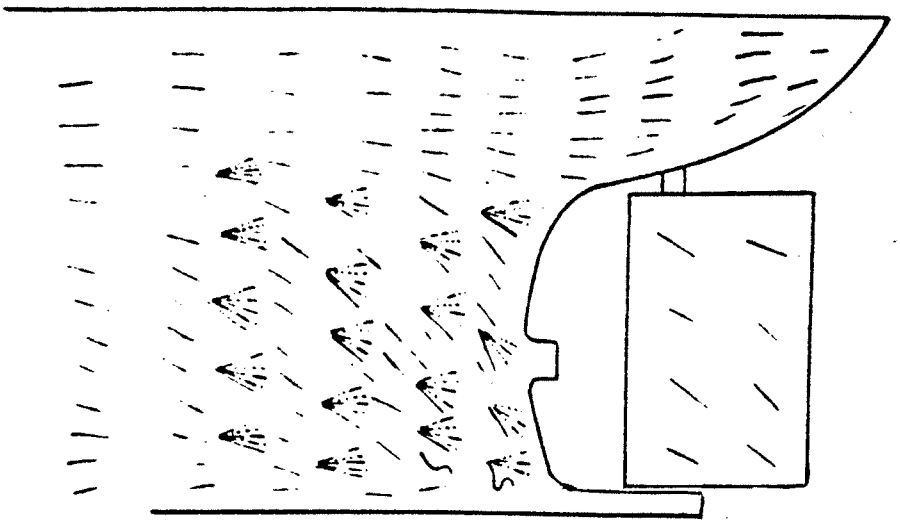
——— Ablösungsbereich

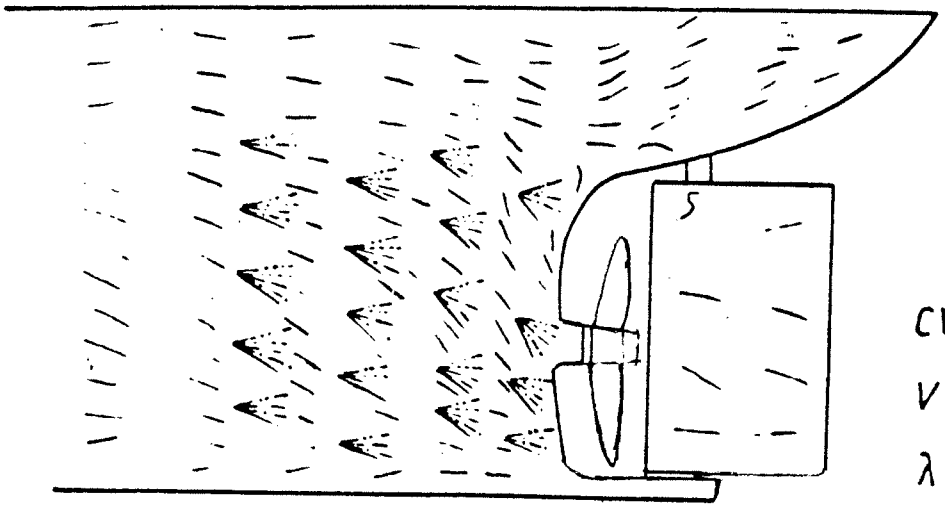
– – – – Übergangszone

V = 14 - 18 kn

Mod. Nr. 2209	Ablösung am Achterschiff	29

3-30

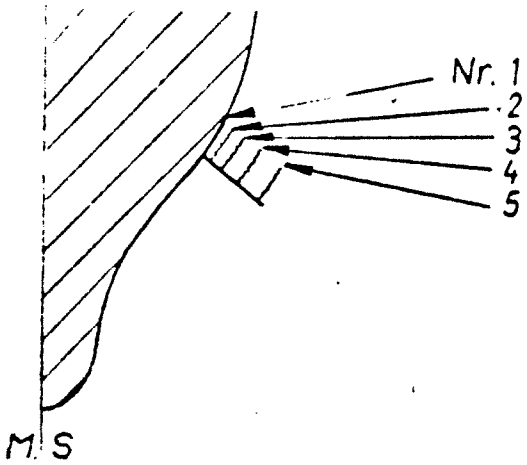




CWL

V = 16 kn

λ = 40

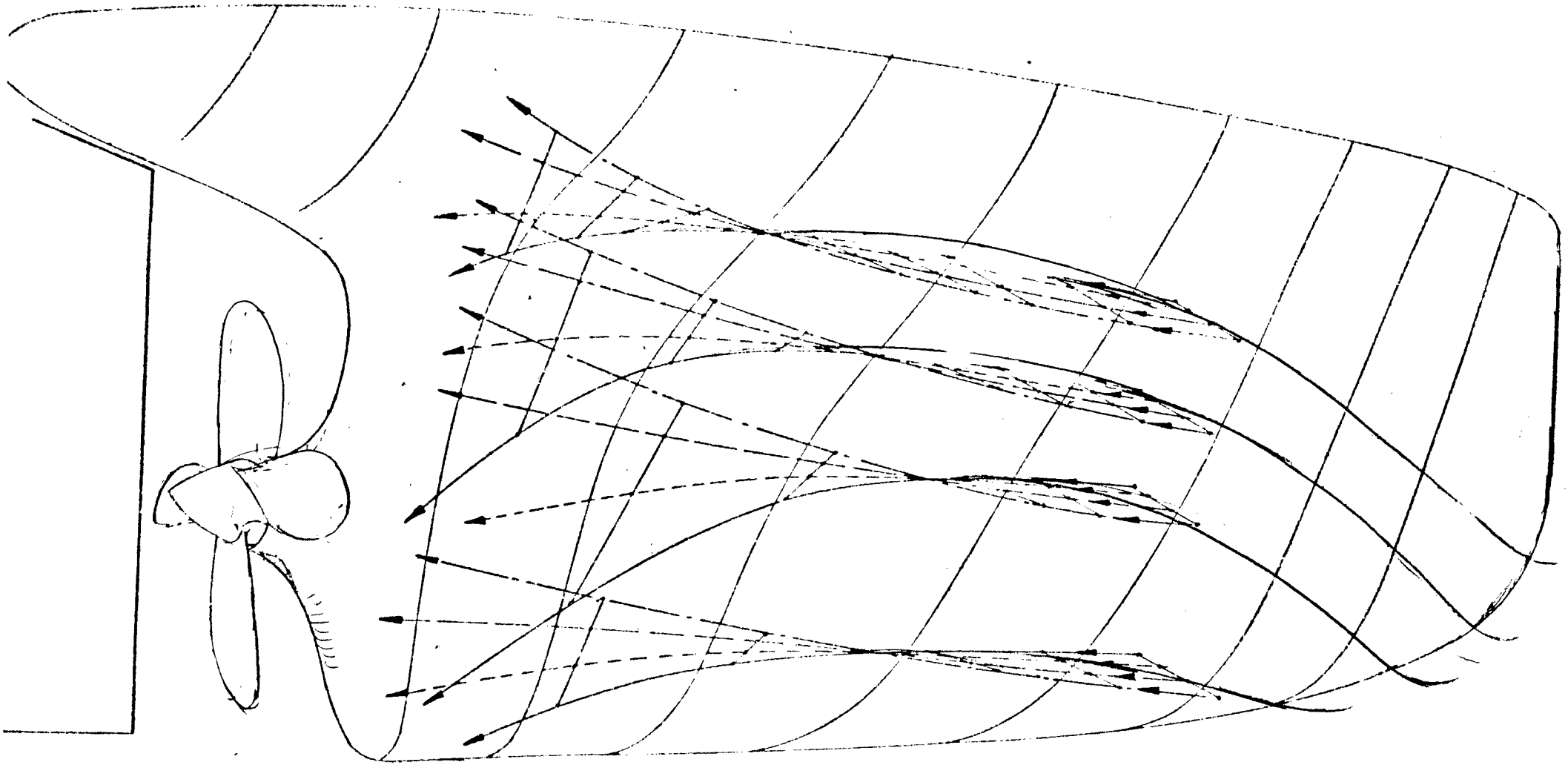


Abstand e der Fadensonden von der Außenhaut

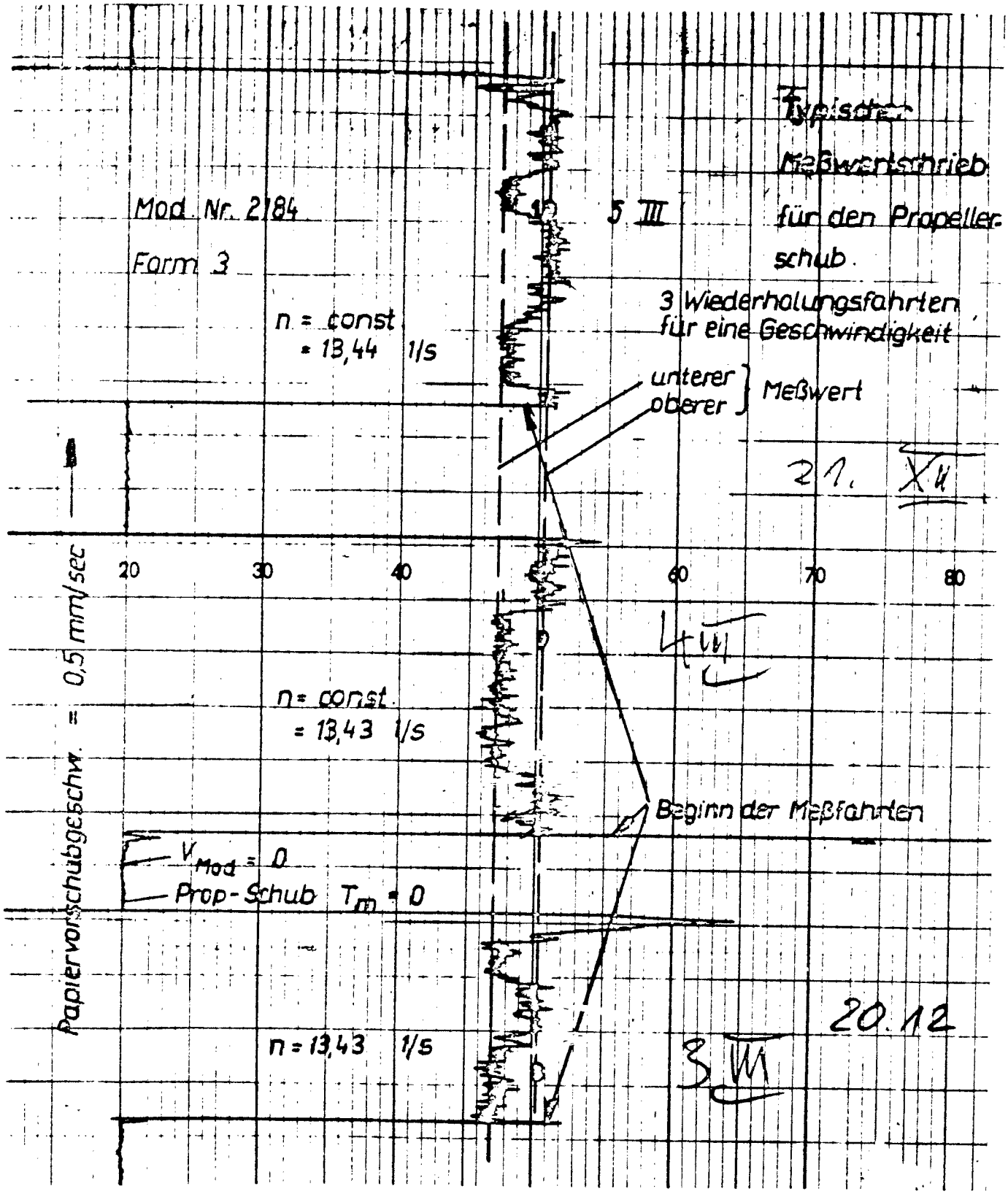
Nr.	$e_{Mod.}$	e_{Schiff}	Kennz.
1	0 mm	0 mm	———
2	5	200	—·—·—
3	10	400	— —
4	30	1200	- - - - - -
5	60	2400	—··—

Form 3

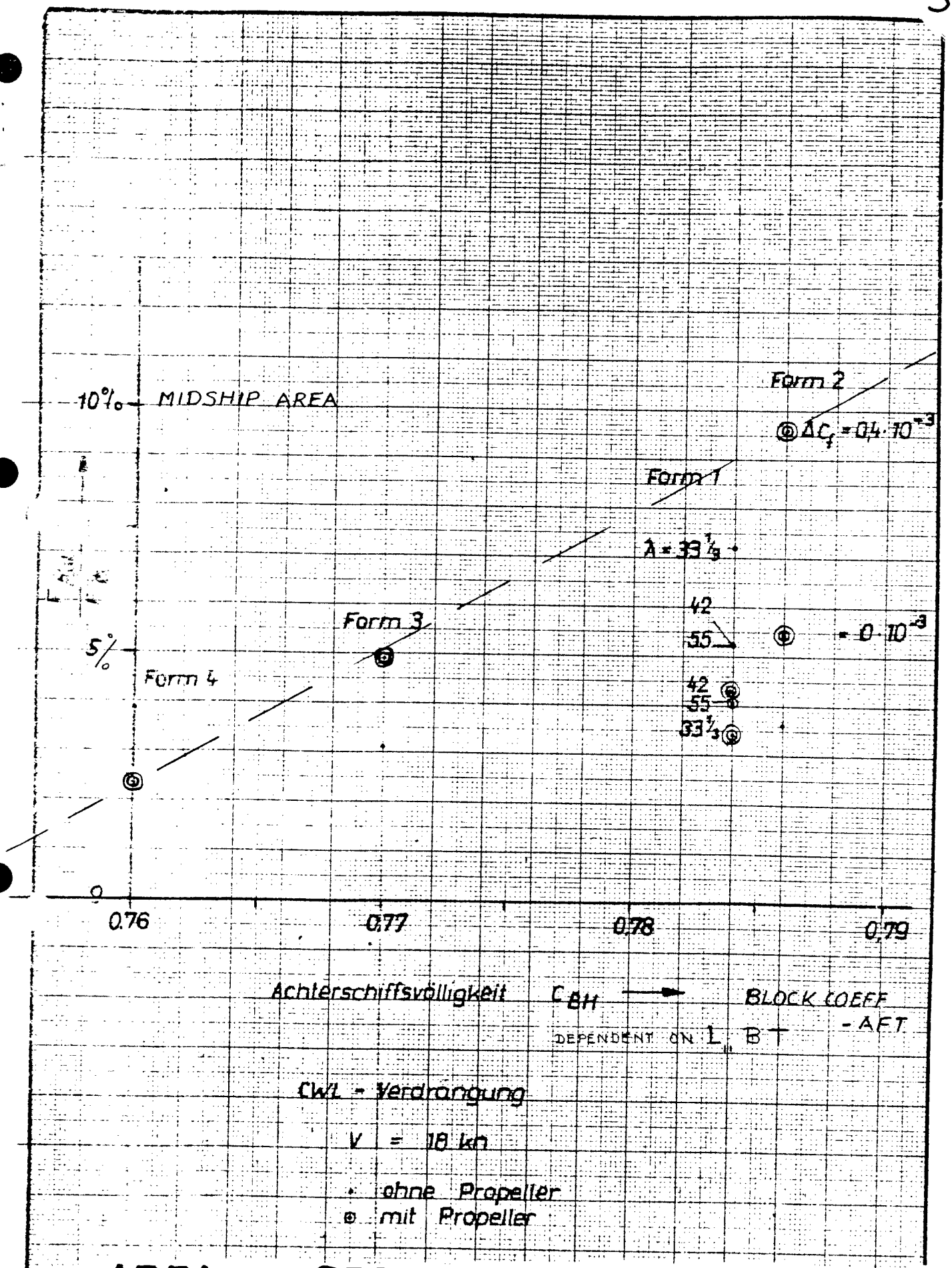
Mod. Nr. 2184	Strömungsverlauf am Achterschiff	30



	Aus den beobachteten Richtungen (Bild 30) hergeleitete Strömungstendenzen	31

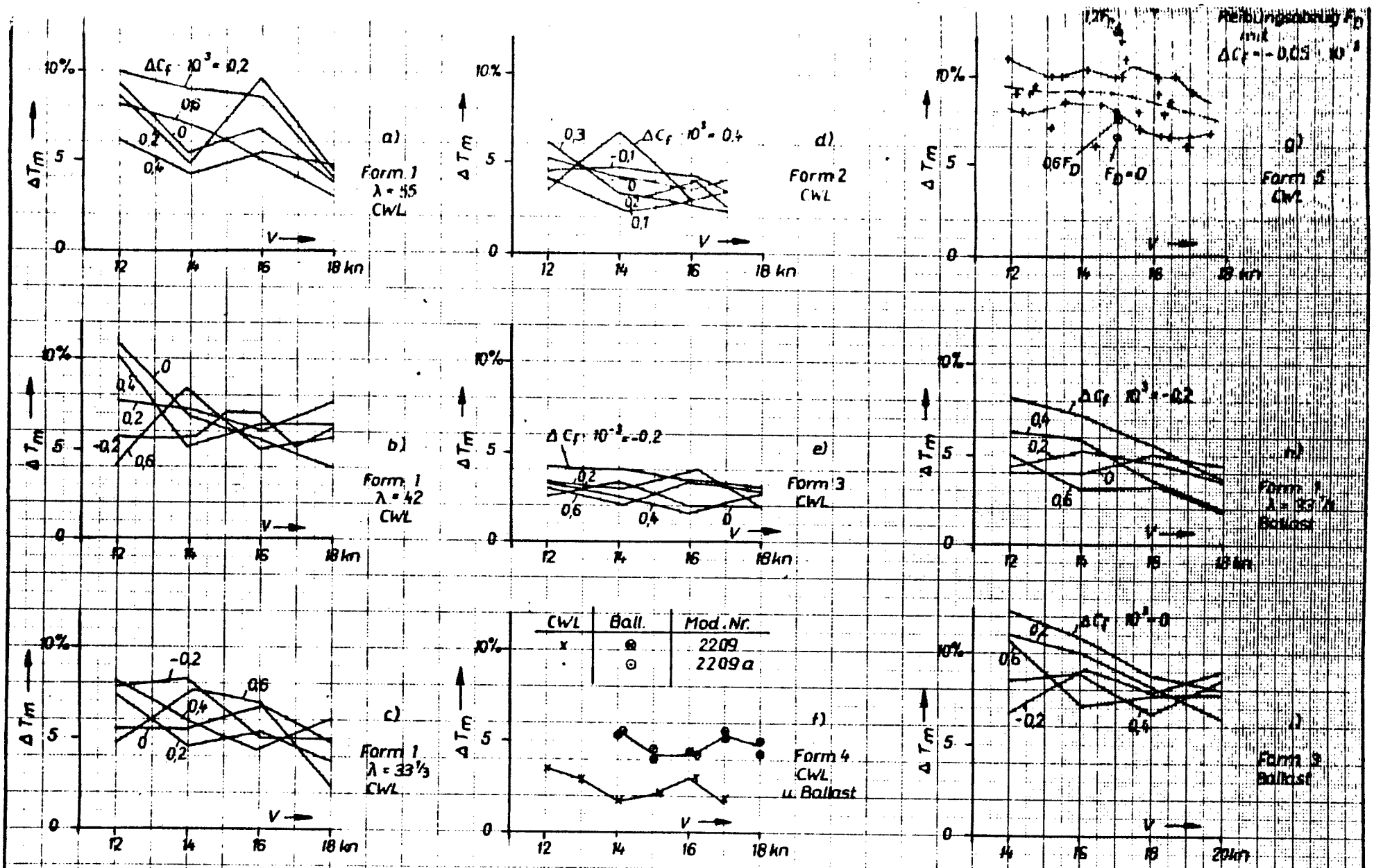


PROPELLER-THRUST NON-STATIONARY BECAUSE OF FLOW FLUCTUATION



AREA OF SEPARATION AS FUNCTION OF $C_{B\text{-}AFT}$

	Prozentuale Ablösungsfläche	33

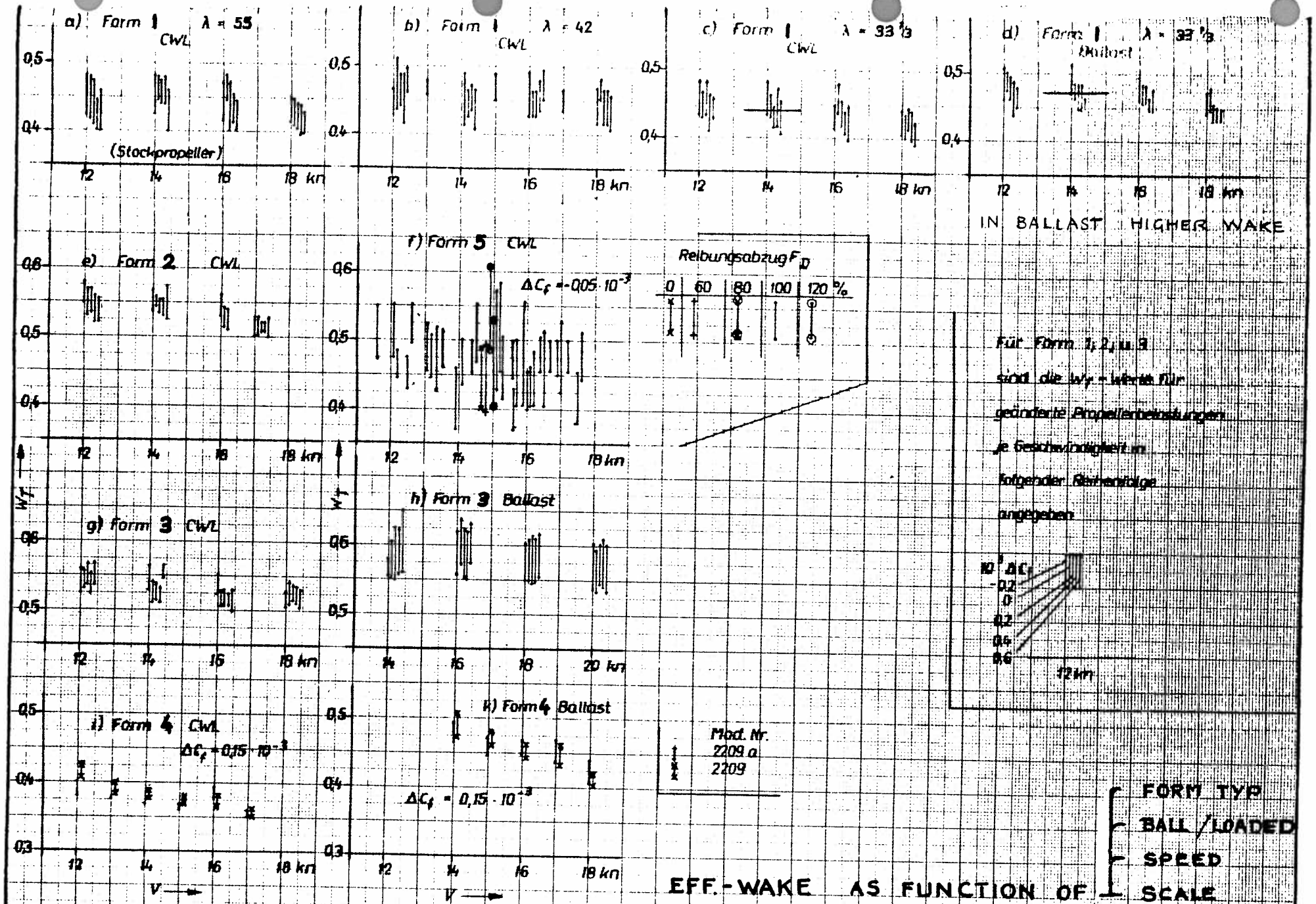


VARIATIONS OF PROPELLER THRUST

Gesamte Meßwertschwankungen des Modellpropellerschubes

34
a - i

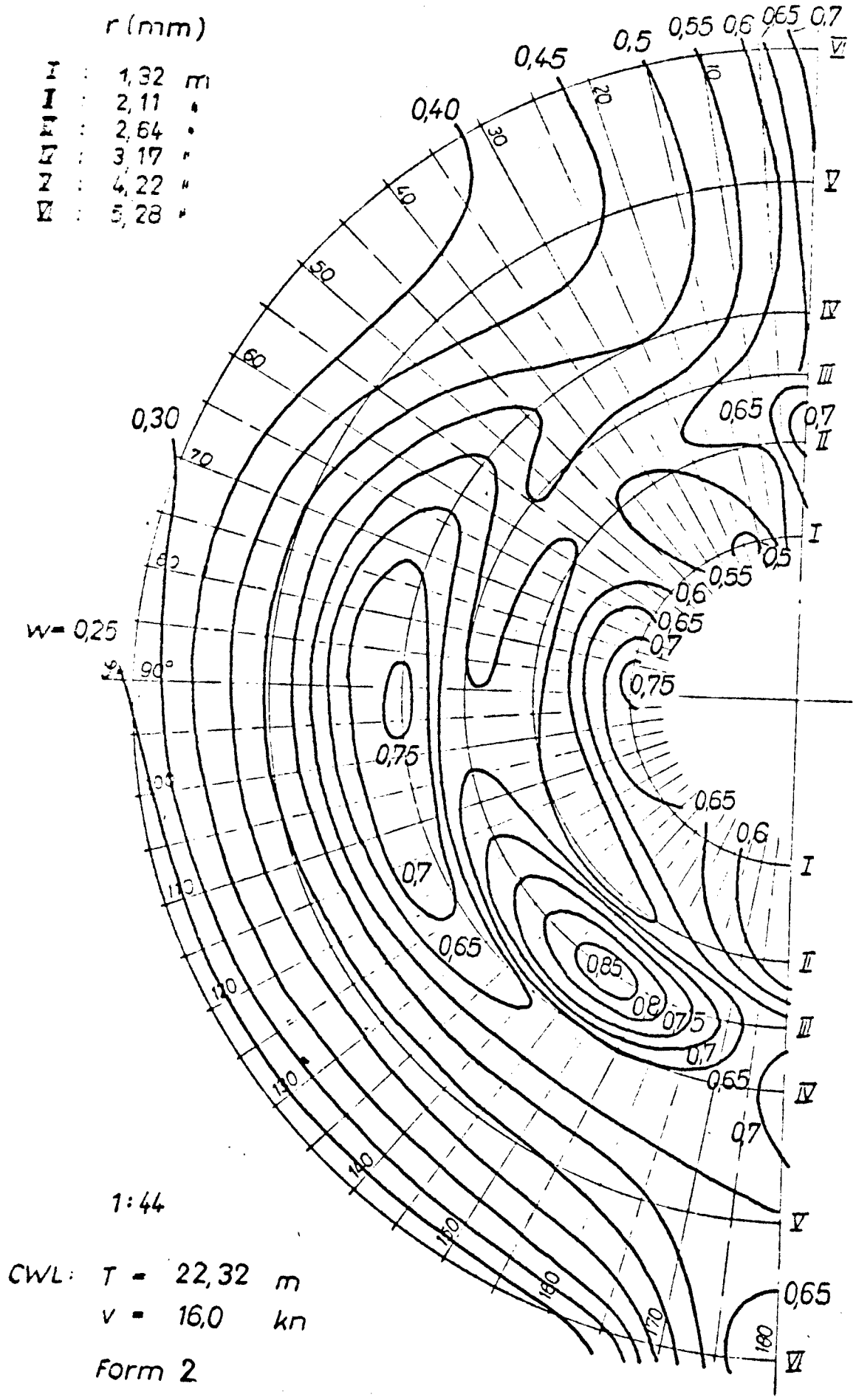
3-35



Effektiver Nachstrom w_T mit den Grenzen seiner Änderung am Modell

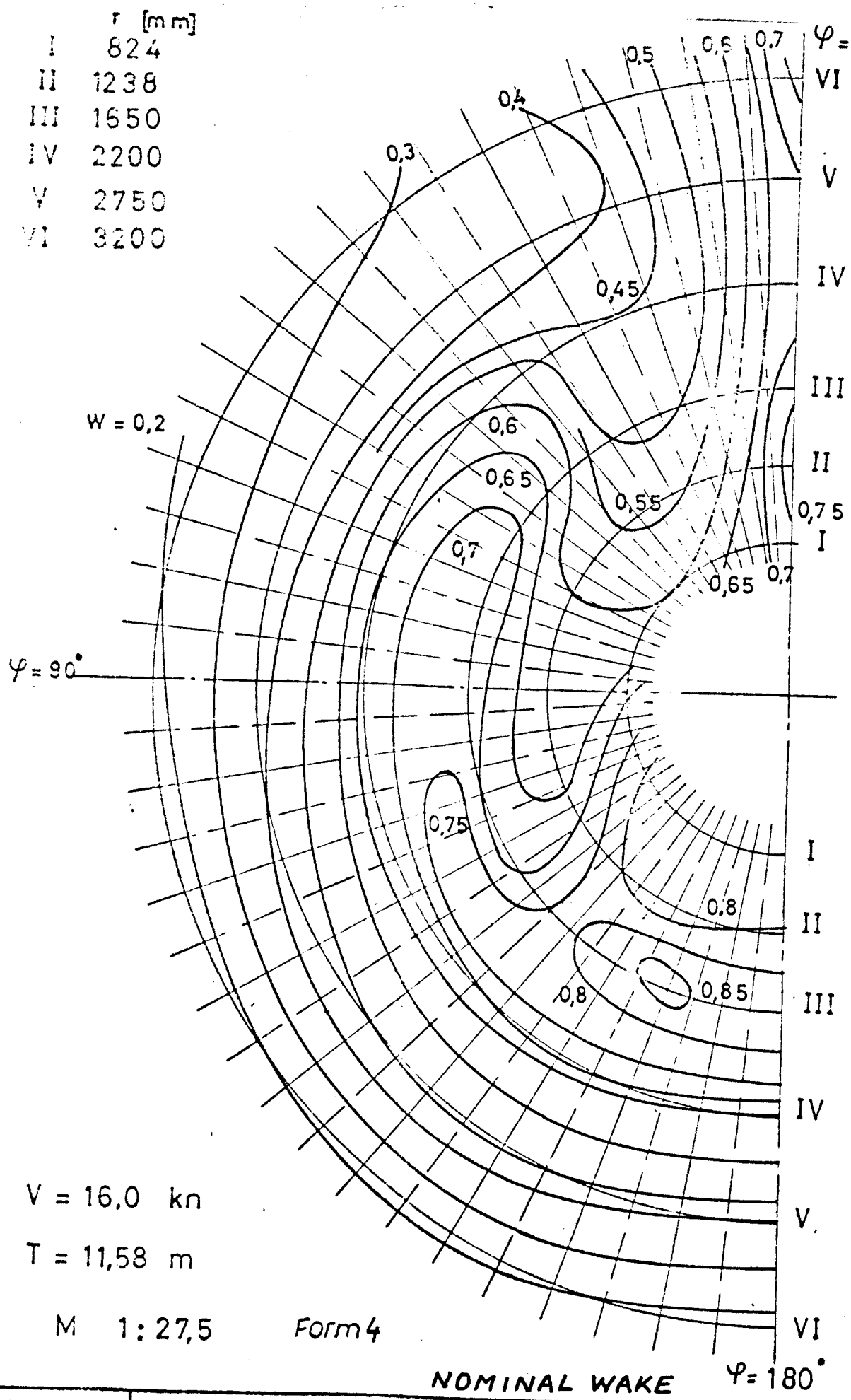
35
a-k

3-36

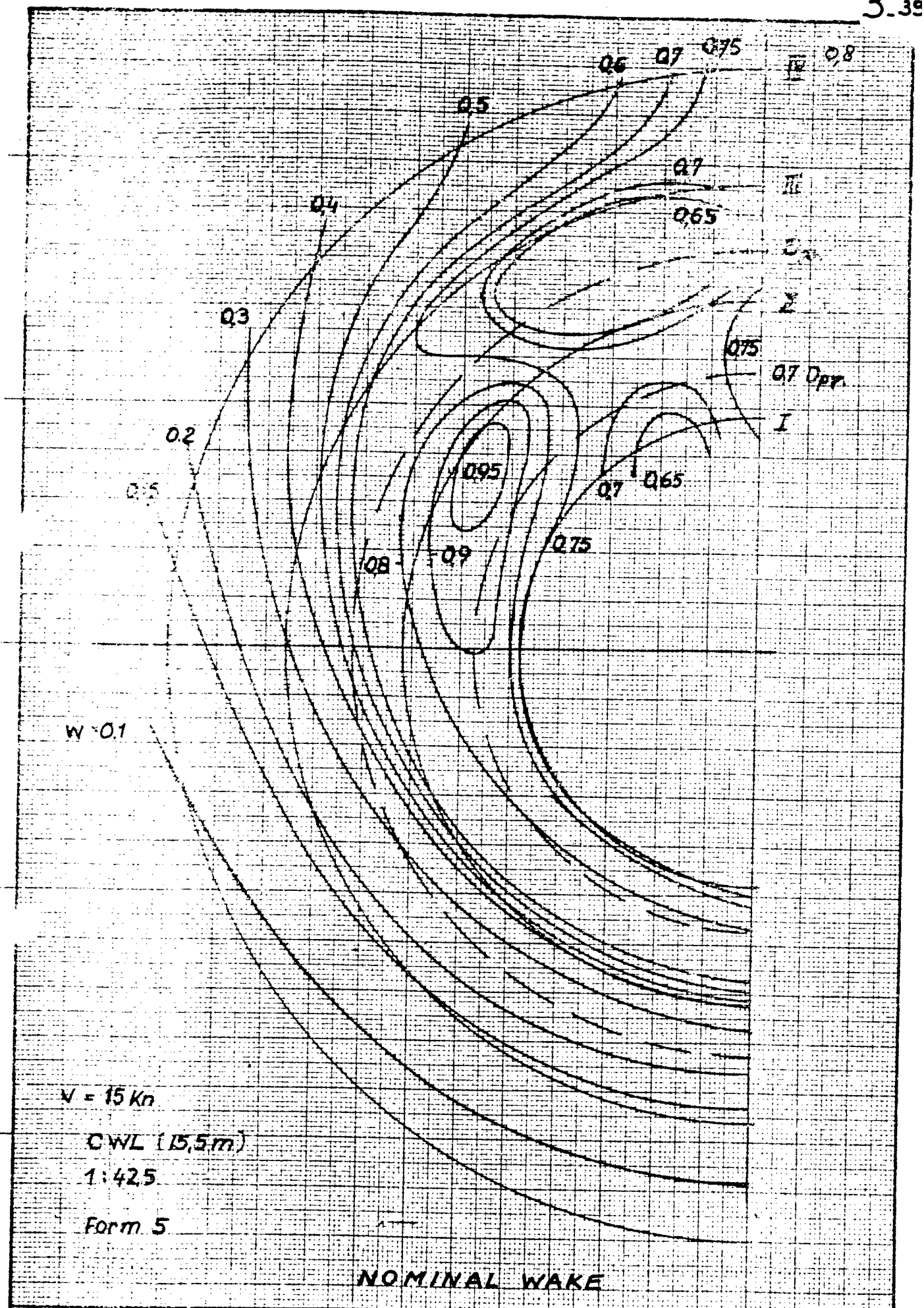


NOMINAL WAKE

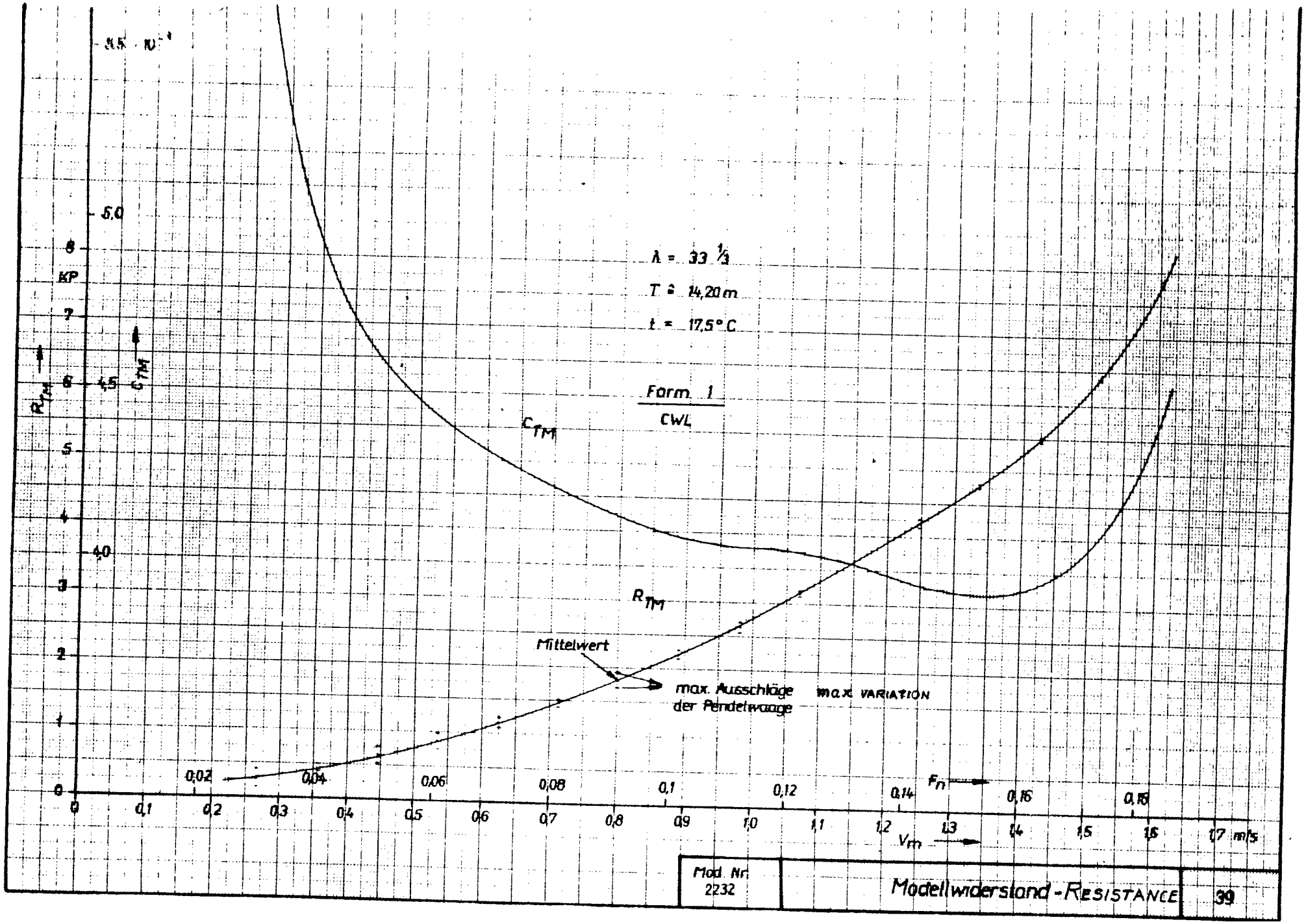
Mod. No. 2239	Verteilung des nominellen Nachstroms	36



Mod. Nr. 2209	Verteilung des nominellen Nachstroms	37



Mod. Nr.		
2112 a	Verteilung des nominellen Nachstroms	38

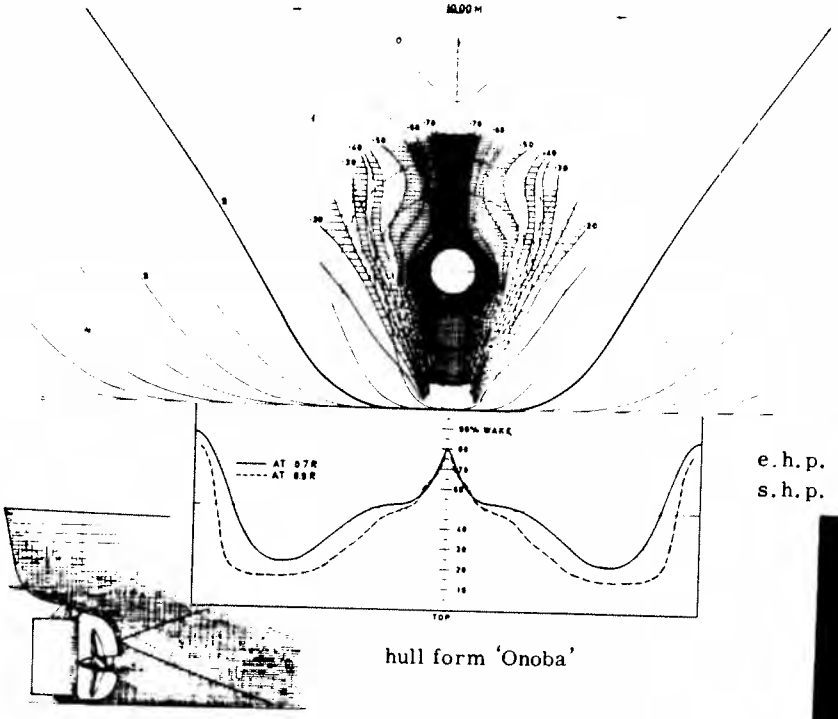

λ = 33 1/3
T ≙ 14,20 m
t = 17,5° C
Form 1
CWL
C_{TM}
R_{TM}
Mittelwert
max. Ausschläge der Pendelwaage
max VARIATION
KP
R_{TM}
C_{TM}
F_n
V_m
m/s
Mod. Nr. 2232
Modellwiderstand - RESISTANCE
39

3-4

4

TRYING TO DEVELOP A CONDOLA FOR SINGLE SCREW

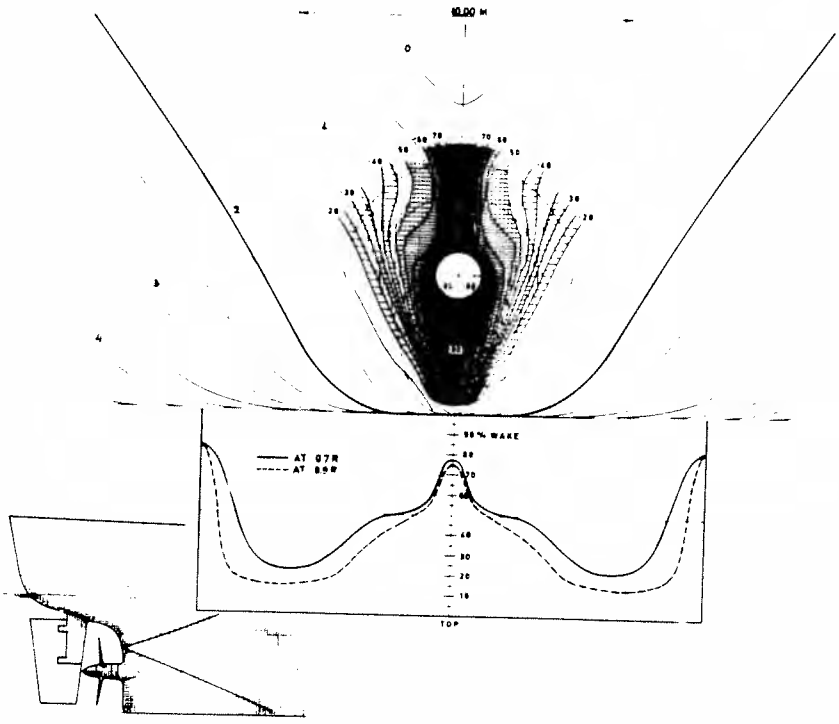
1960 ——> 1980



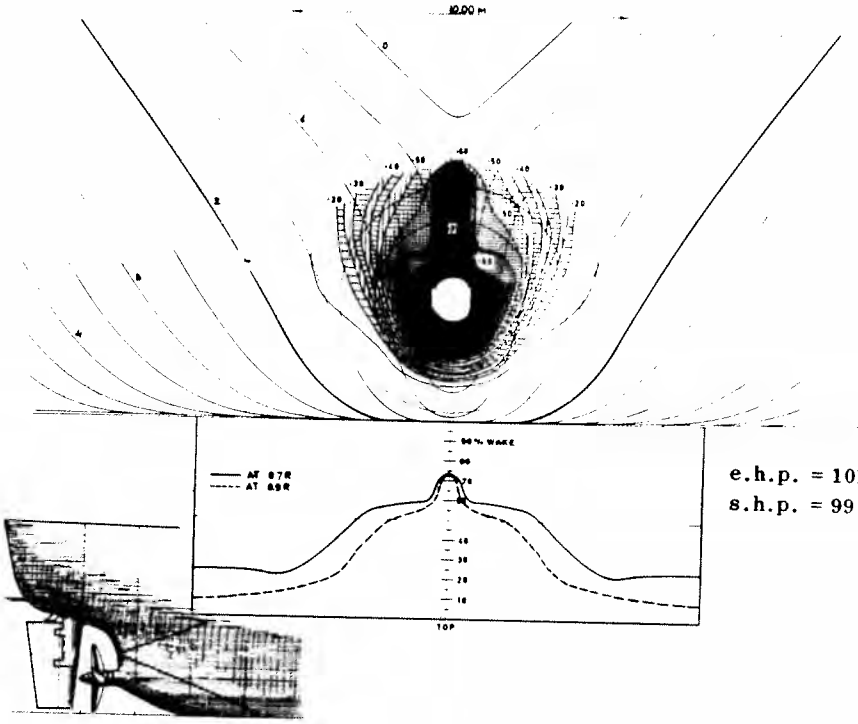
e.h.p. = 100% η = 0.729
s.h.p. = 100% (prop. 2503-5)

hull form 'Onoba'

High wake peak in top → cavitation. High wake peak in bottom position.
With 4 bl. prop. : strong variations in torque and thrust.
With 5 bl. prop. : strong variations in transverse forces on sterntube and bending moments in tailshaft.

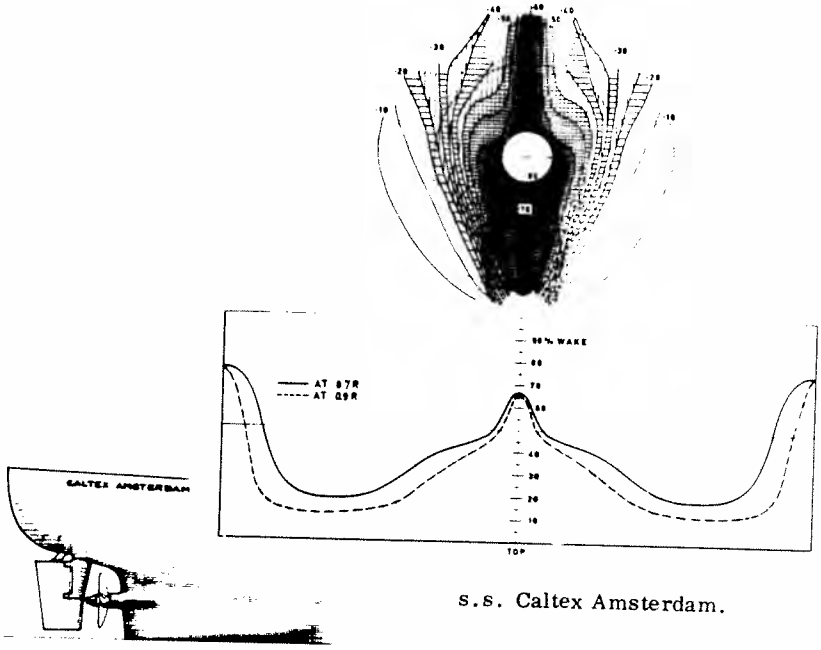


Sole piece removed. Not much improvement.



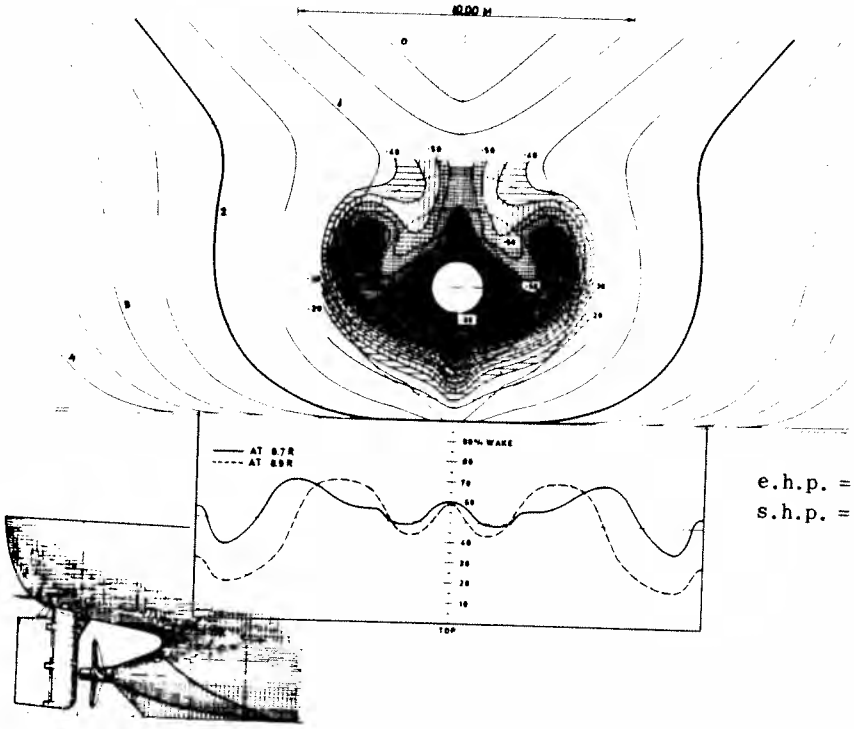
e.h.p. = 101% $\eta = 0.726$
s.h.p. = 99 1/2 %

Rounded hull shape below propeller shaft. Large improvement in bottom position. No improvement in top.



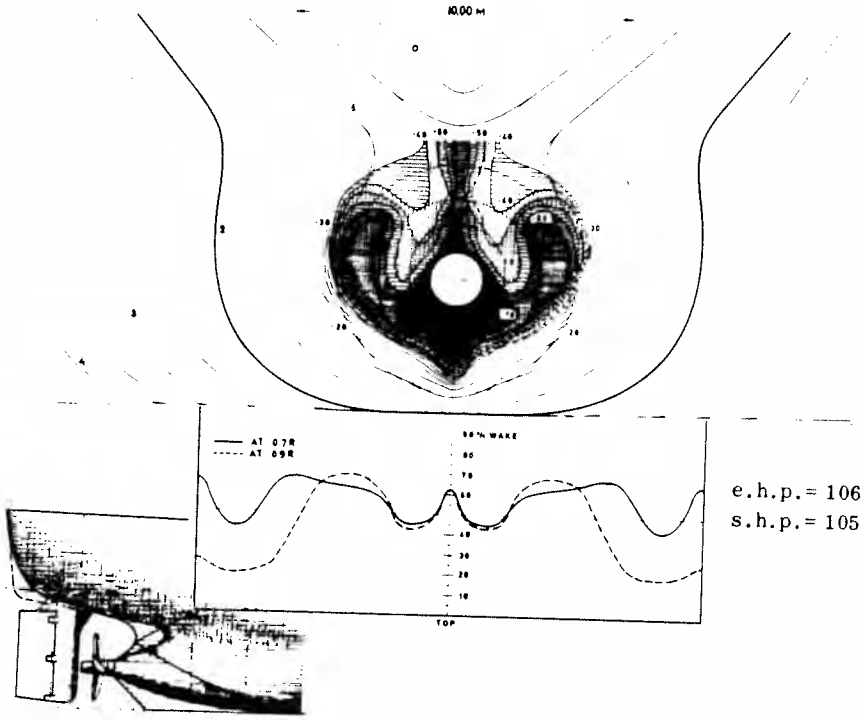
s.s. Caltex Amsterdam.

. Example of type ⋀. Not much improvement.
as done on "ONOBA" (previous page)



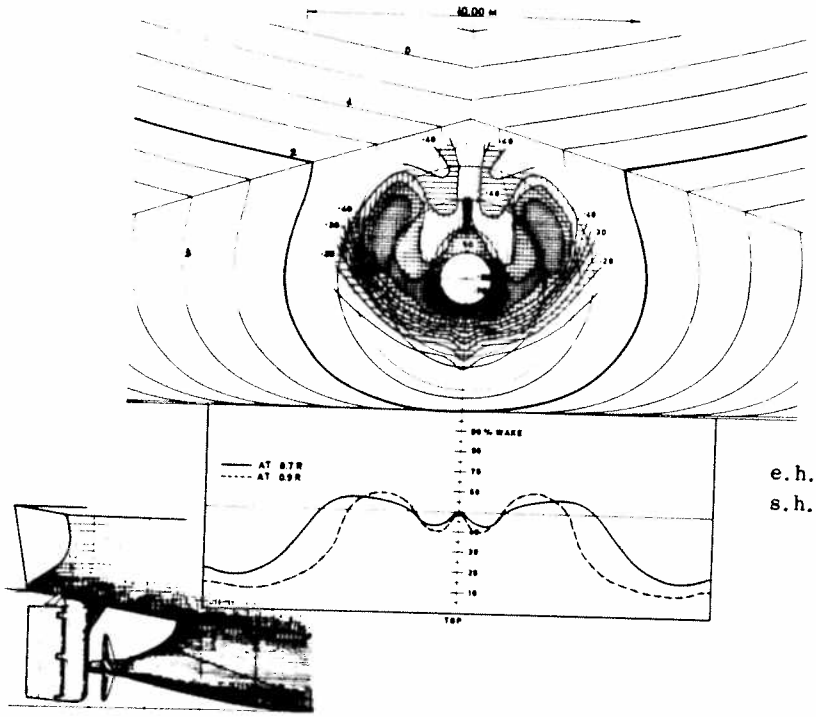
e.h.p. = 104% η = 0.680
s.h.p. = 112%

Fat Hogner cigar. Two vortexes excited, p and s, contra rotating tangential flow in propeller disk. Flow separation above propeller shaft, blunt waterline endings at 8 m did increase s.h.p. value.



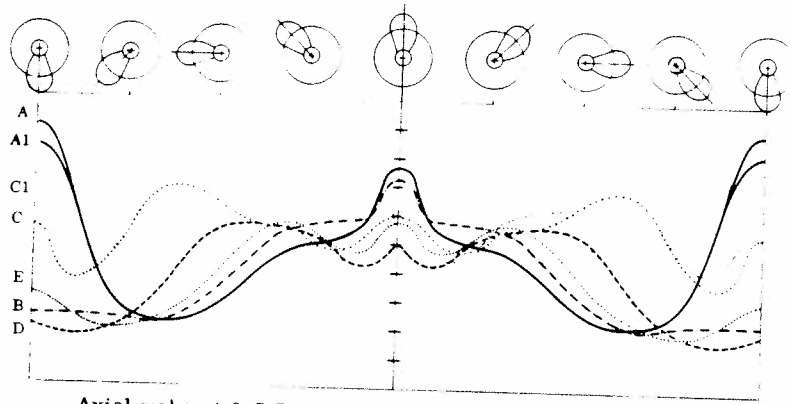
e.h.p.= 106 1/2 % η = 0.734
s.h.p.= 105 1/2 %

Same fat Hogner cigar. Waterlines above propellershaft extended. Less increased s.h.p.

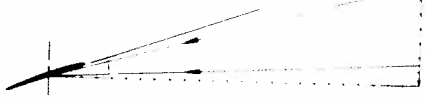


e.h.p. = 103% η = 0.706
s.h.p. = 106%

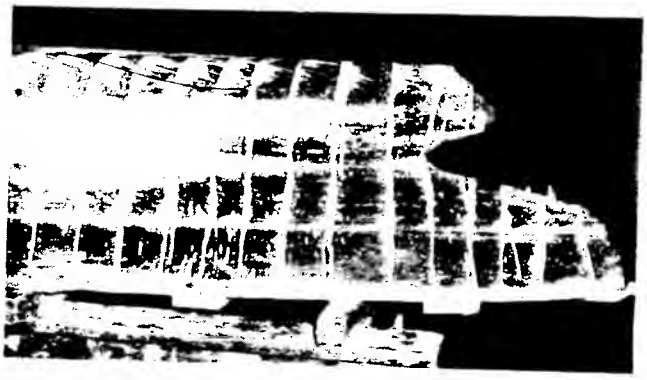
Slender Hogner cigar below transom stern. Two vortexes excited. Insufficient wake filling of wake field according to model test. How will be the wakefield on the ship?



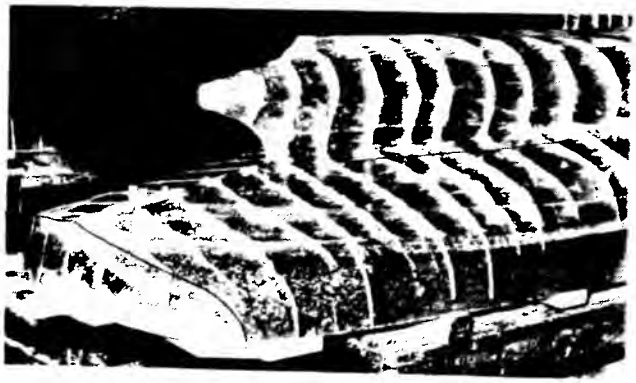
Axial wake at 0.7 R measured with normal pitot tube



variation of angle of attack
(without taking into account the tangential flow).

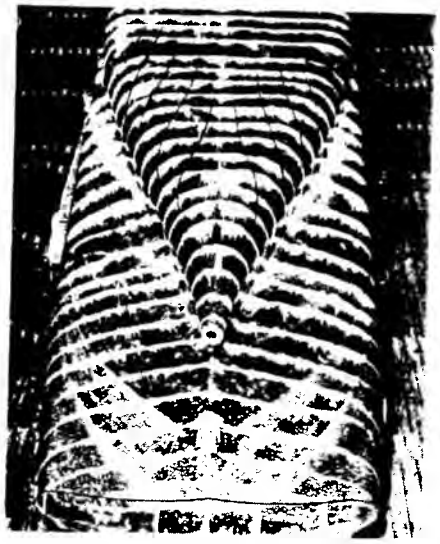


C. Hogner afterbody.



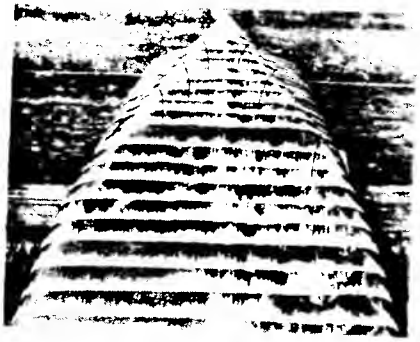
D. Hogner cigar + transom stern.

C. Hogner afterbody.



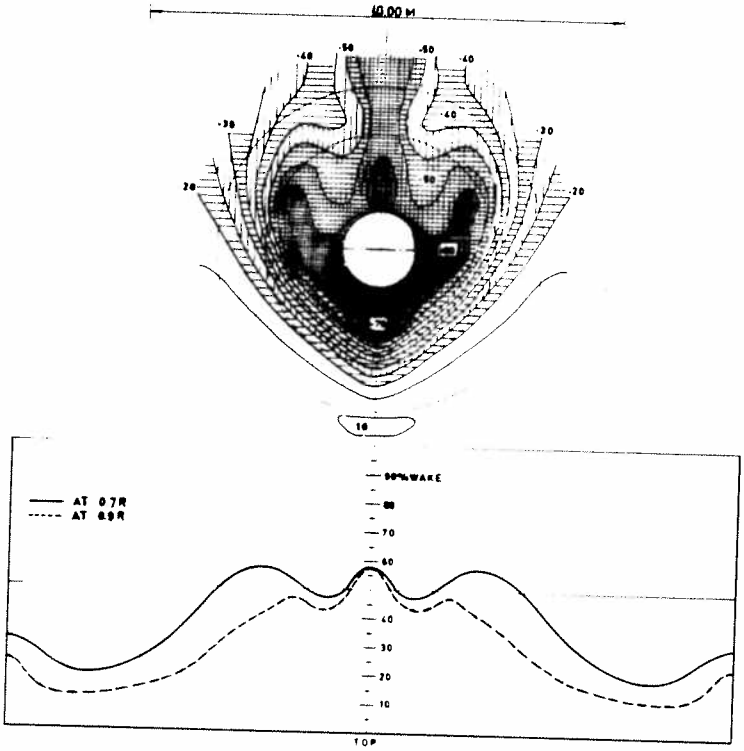
D. Hogner cigar + transom stern.
Note separation of flow at bilge rounding.

Type C. Hogner afterbody.

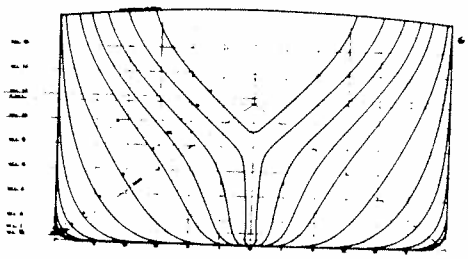


D. Hogner cigar + transom stern.

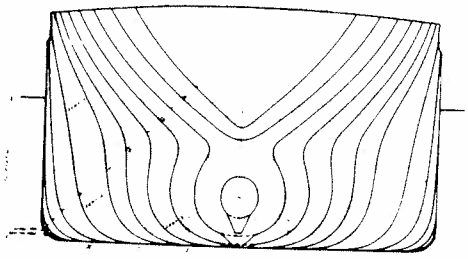
Models by Netherlands Ship Model Basin.



DOELWIJK



Normal afterbody A



Hogner afterbody C

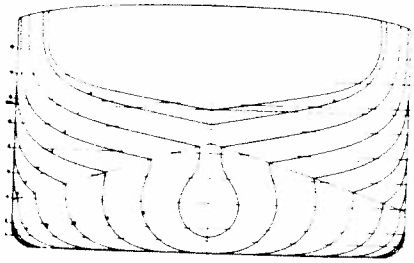
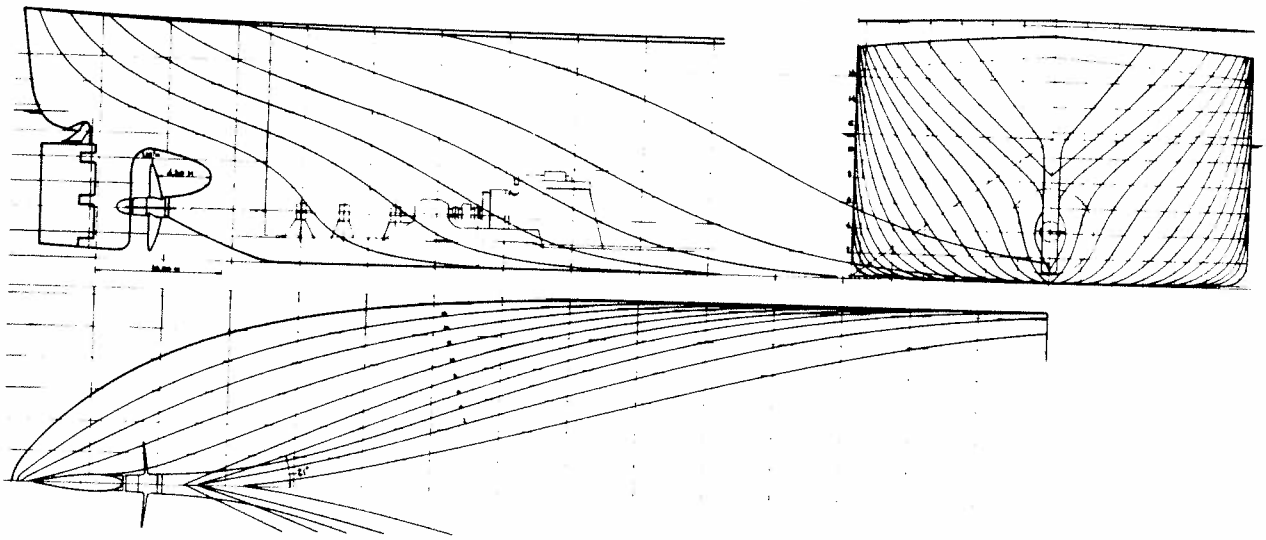
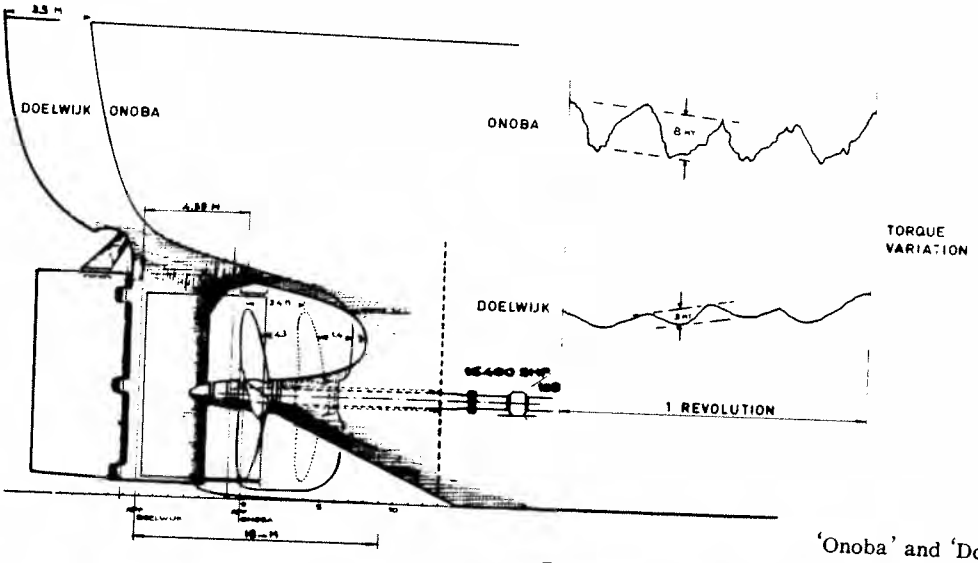


Figure 5. Hogner cigar and transom stern D



Afterbody E. Derived from A by increasing length 3.5 m. Type Doelwijk.

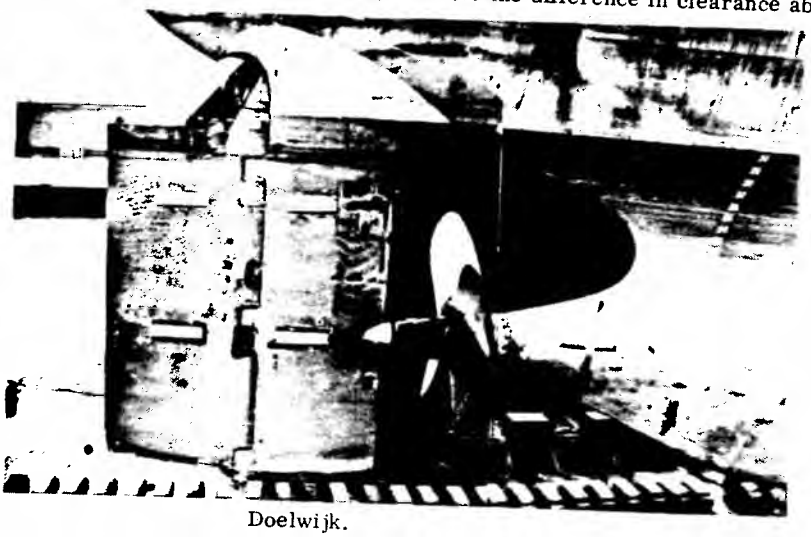


50 000 t.d.w. tankers

'Onoba' and 'Doelwijk'.
Reduction of vibrations by increasing clearance.

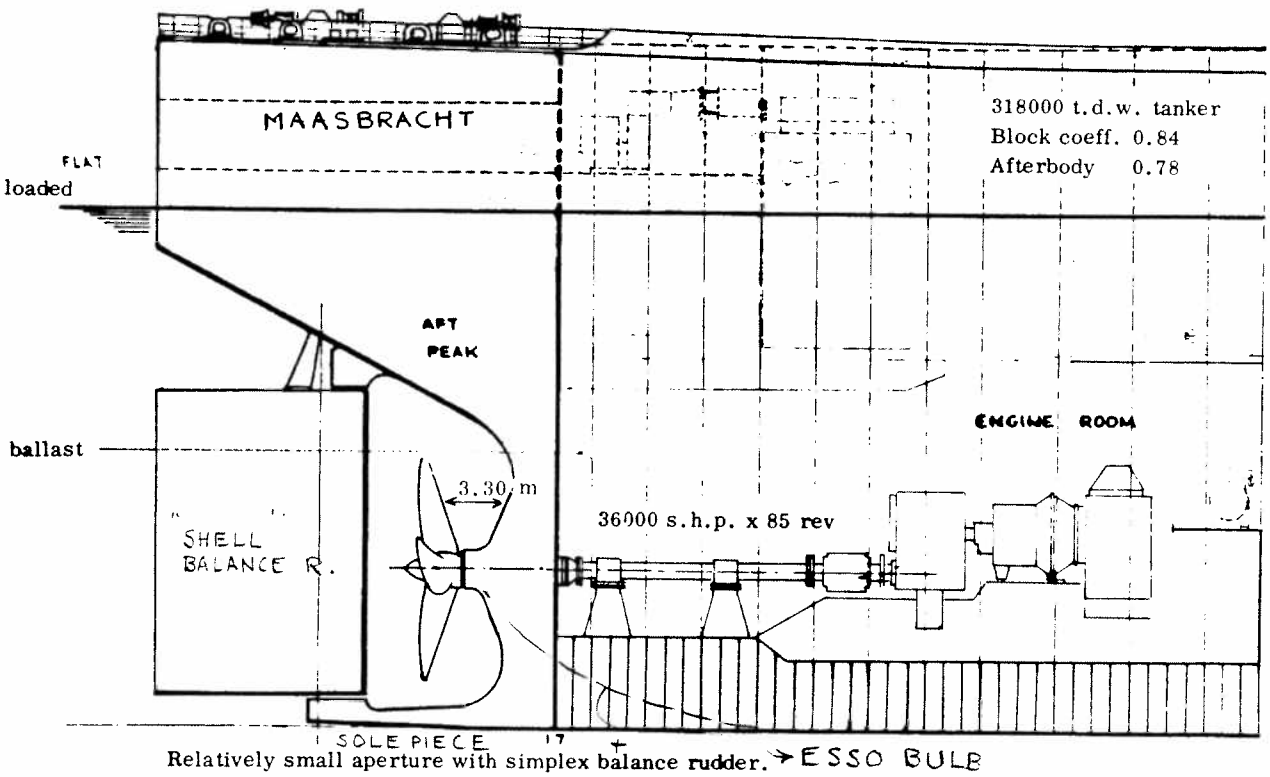
Esso Essen. A.G. Weser stern.
Note the difference in clearance above propeller shaft.

Doelwijk.
Note the difference in clearance above propeller shaft.



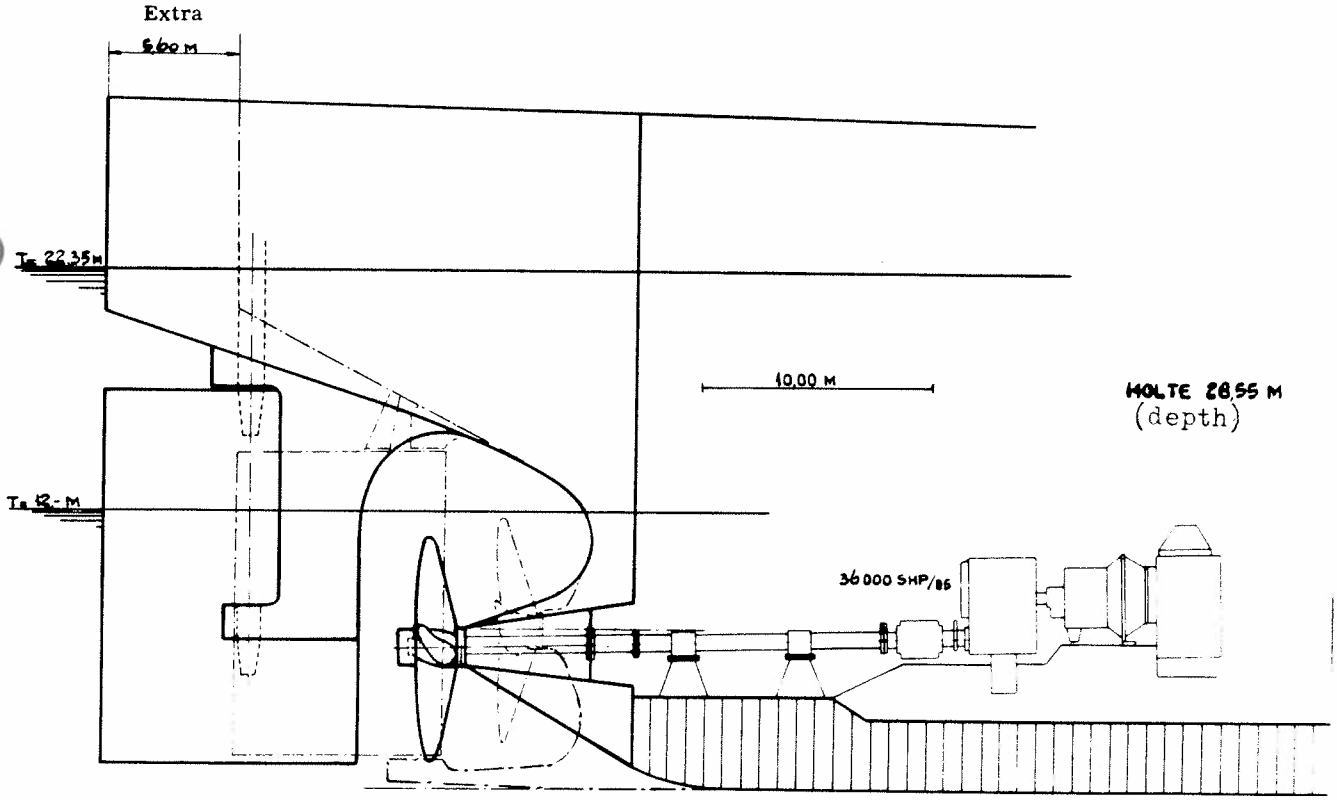
Doelwijk.
- Wide aperture - no vibration troubles.
- Free Oertz rudder - expensive.

Good steering and course keeping. Low maintenance costs.

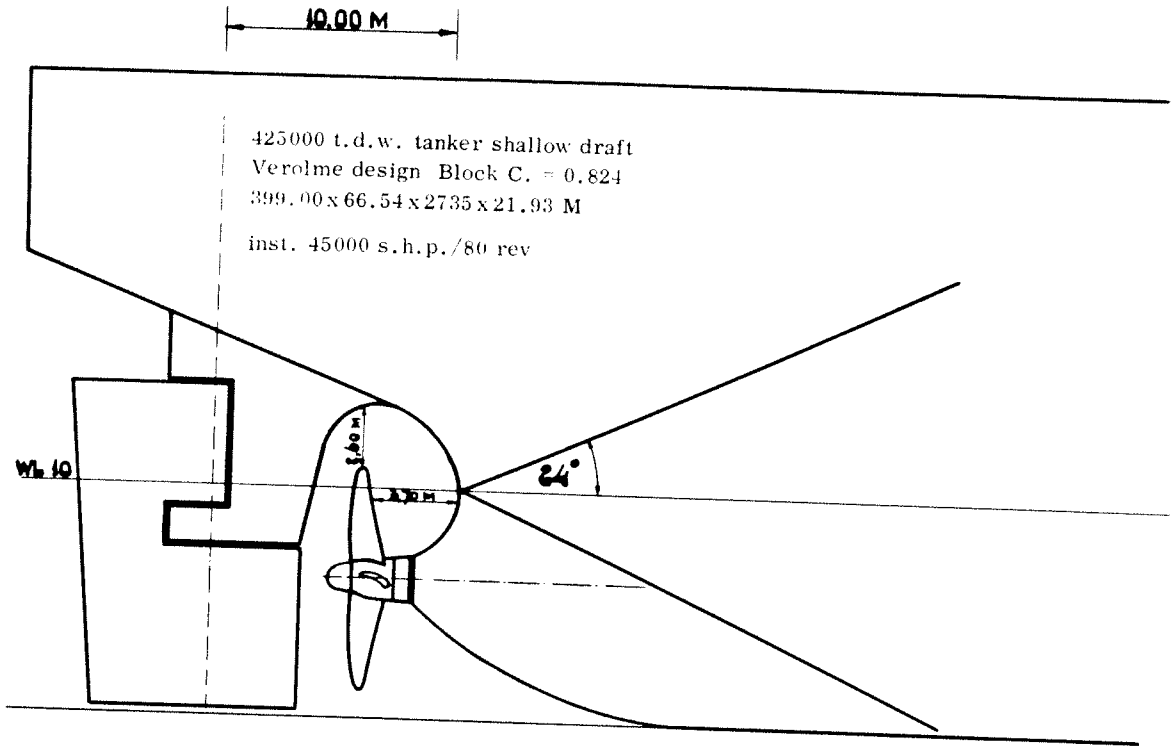


Relatively small aperture with simplex balance rudder.

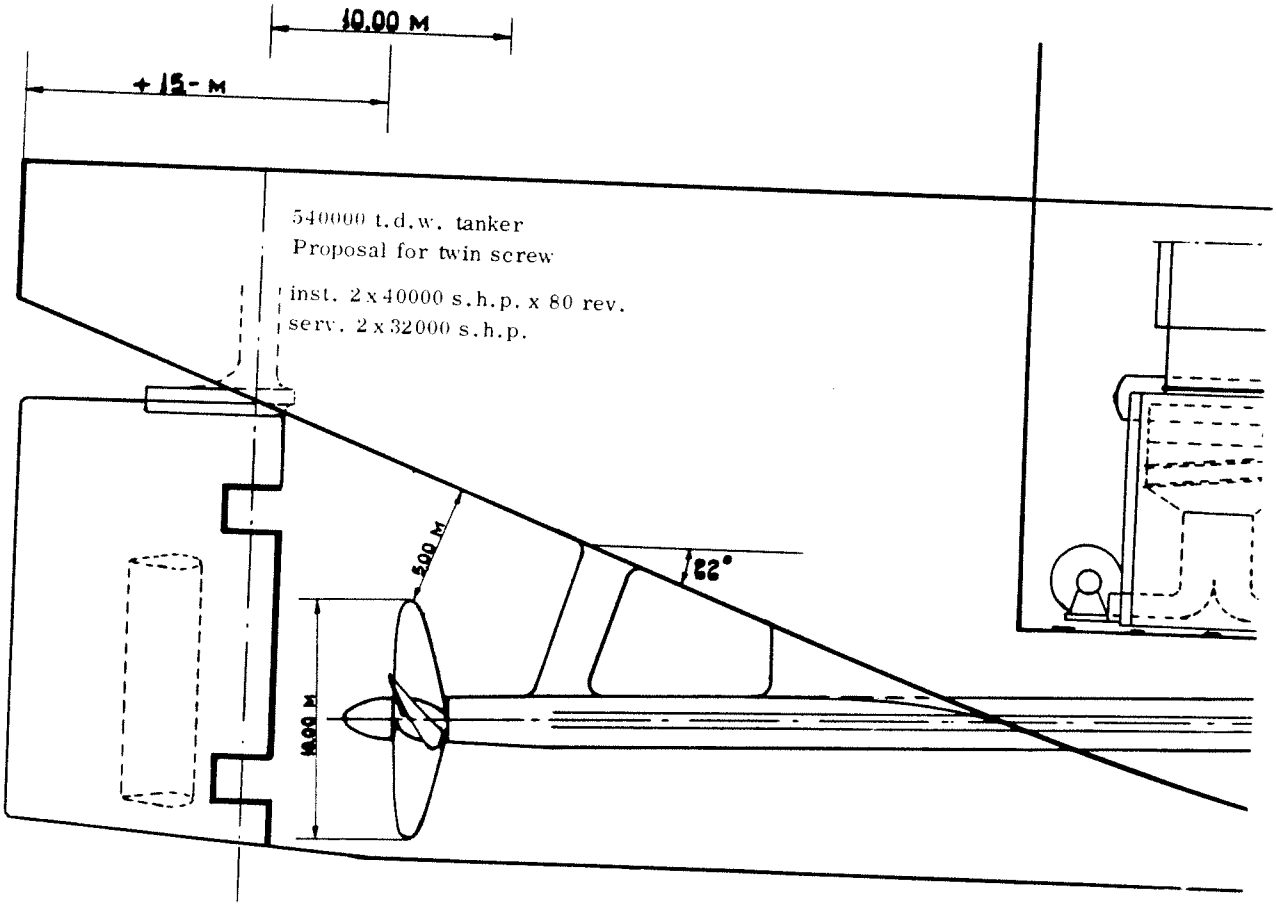
Cheap construction, small steering gear, good ste ring. Possibility of repairs on rudder: cone, bearings, couplingflange, sole piece. High wake peak (-75%) in bottom position. Heavy variations of torque and thrust with 4-bladed propeller are to be expected. (Gearing and thrustblock). Possibility of fore and aft movement of superstructure in ballast. Can full power be maintained in ballast? (Application of fin?). With 5 or 7 - bladed propeller, free vertical and horizontal forces on sterntube bearing, which might cause sterntube-wear-down.



Proposal for improvement (was considered to be impossible because of standard-ships-production-program of shipyard. "Owners have to accept standard designs".)

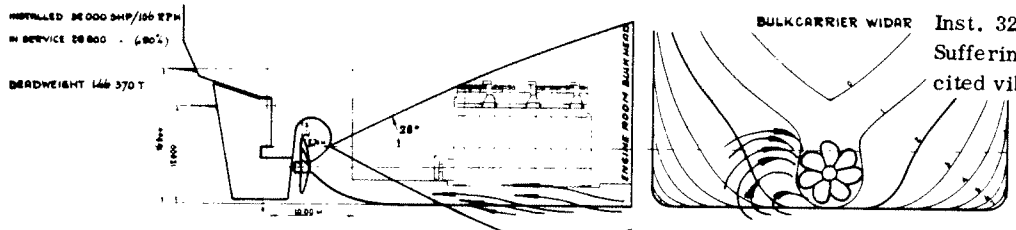


Considered to be utmost limit for power on single screw. Relatively slender waterline endings above propeller shaft.



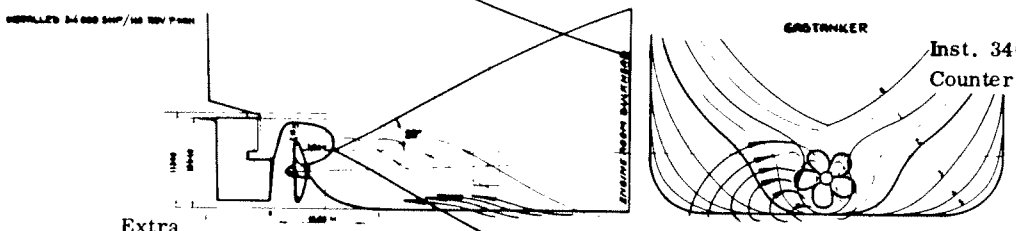
Pram with two free flow propellers. 22° angle to be adjusted according speed-length ratio. Oblique flow should be taken into account for cavitation investigations.

A



Inst. 32000 s.h.p. x 106 revs
Suffering from propeller excited vibrations (7-bl.prop.)

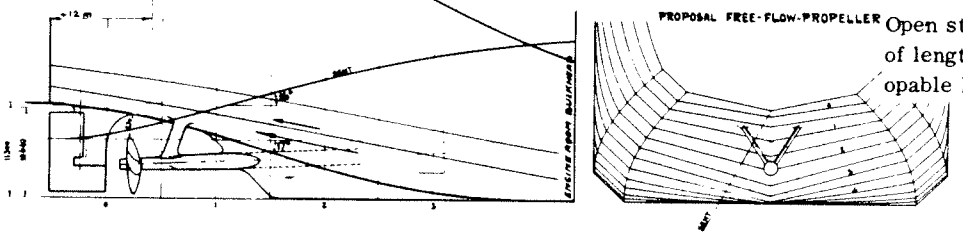
B



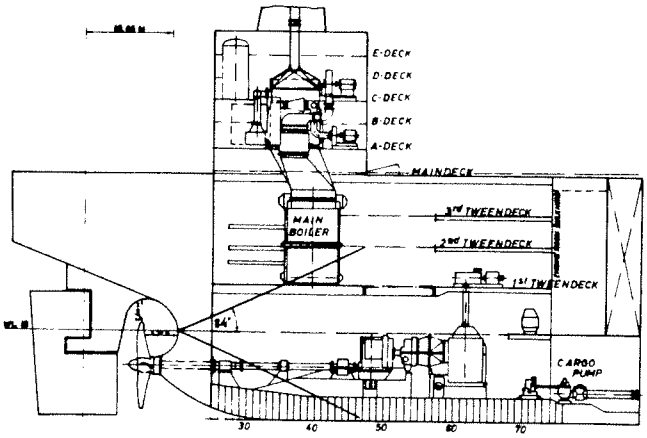
Inst. 34000 s.h.p. x 110 revs
Counter rotating-vortexes.

Extra

C

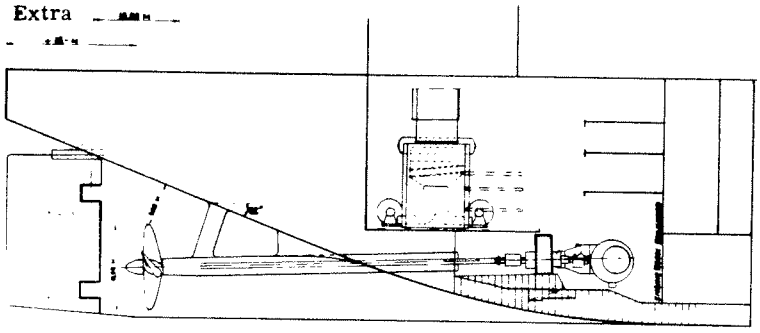


Open stern variant. Increase of length 12 m. Simple developable hull surface.

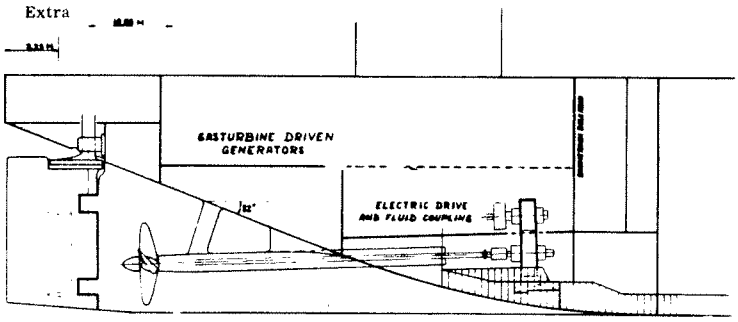


450000 t.d.w. shallow draft tanker
Verolme design.
inst. 45000 s.h.p. x 80 rev.p.m.

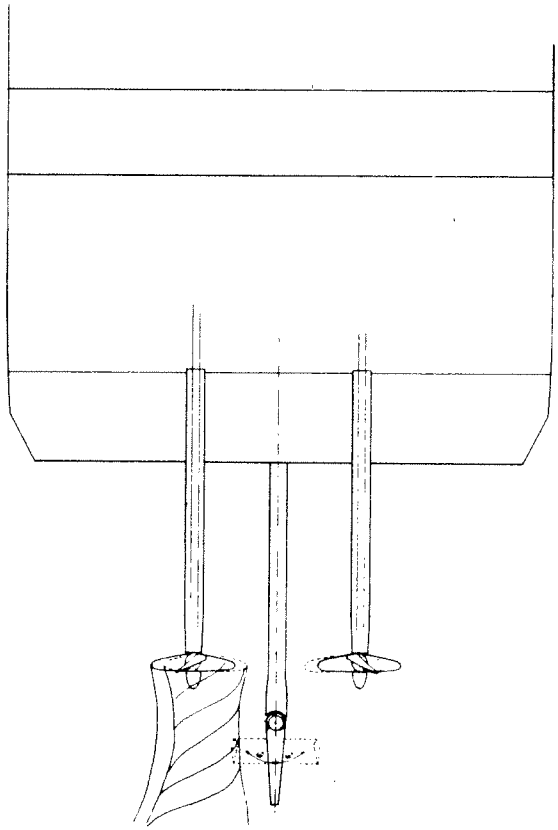
Extra

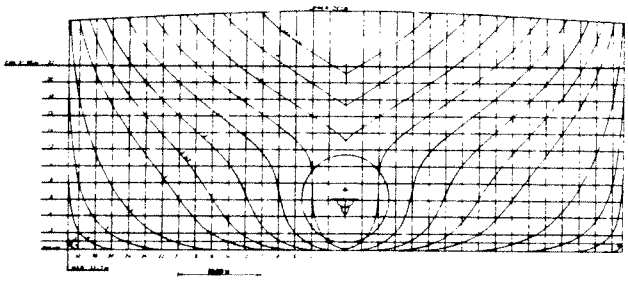


530000 t.d.w. tanker.
inst. 2 x 32000 s.h.p.
Proposal of pram-type stern with free flow propellers increase of length 15 m.



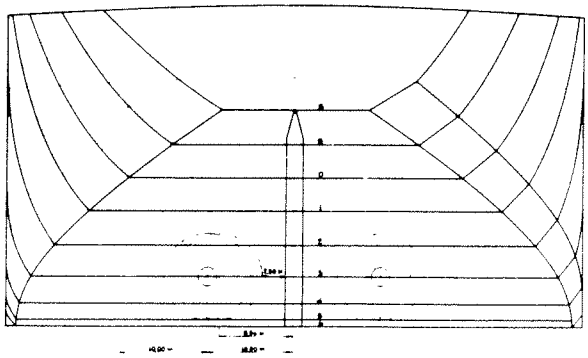
. Proposal of pram-type with small increase of length. Problems and extra costs are transferred to engine-room.

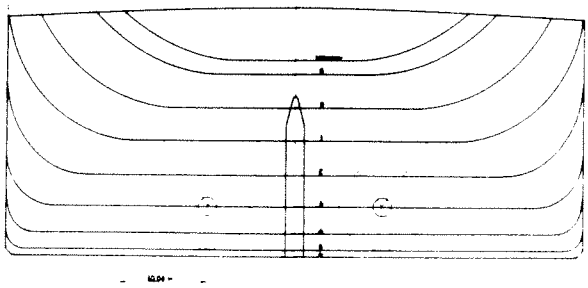




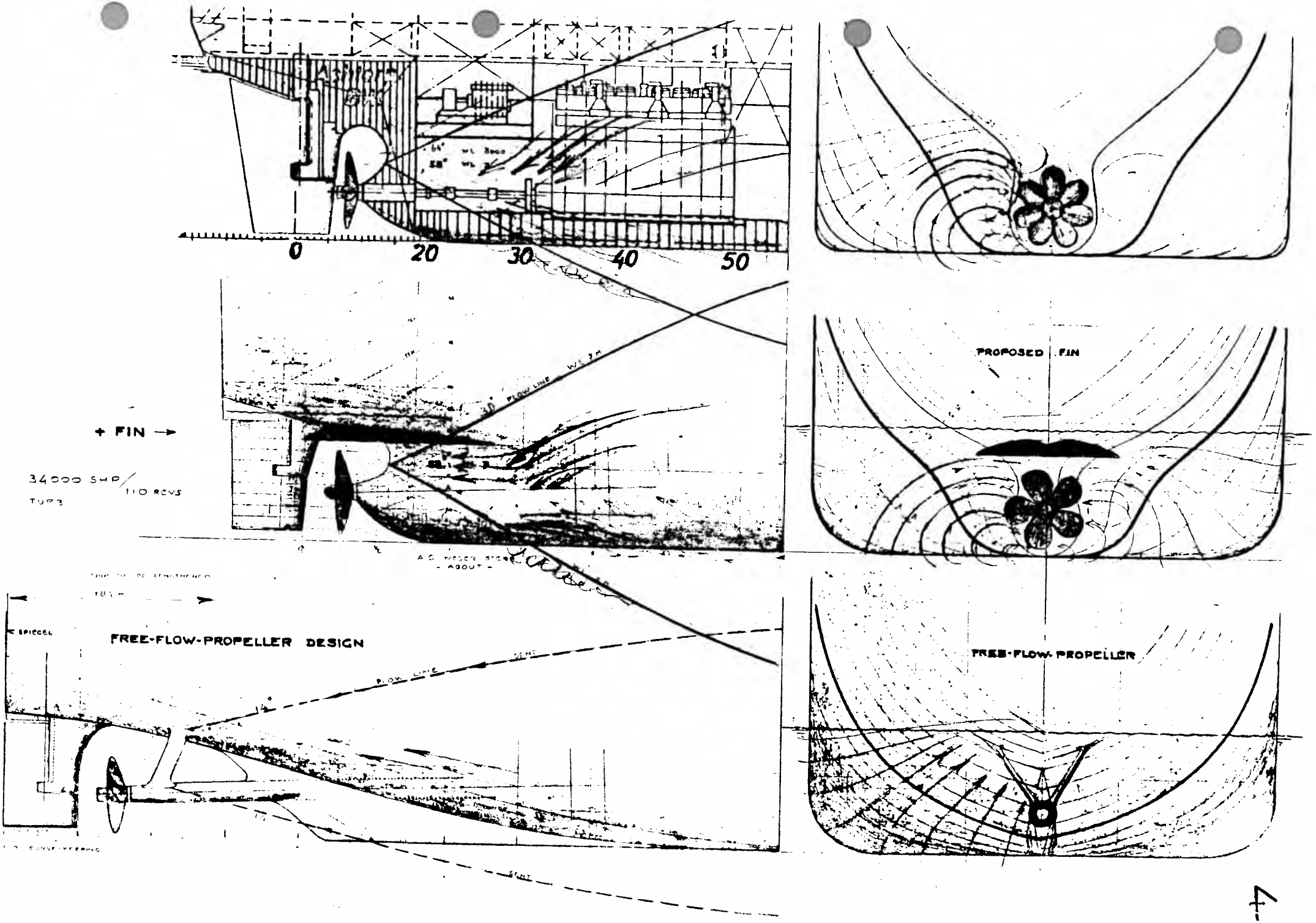
450000 t. tanker. Shallow draft. Verolme design.

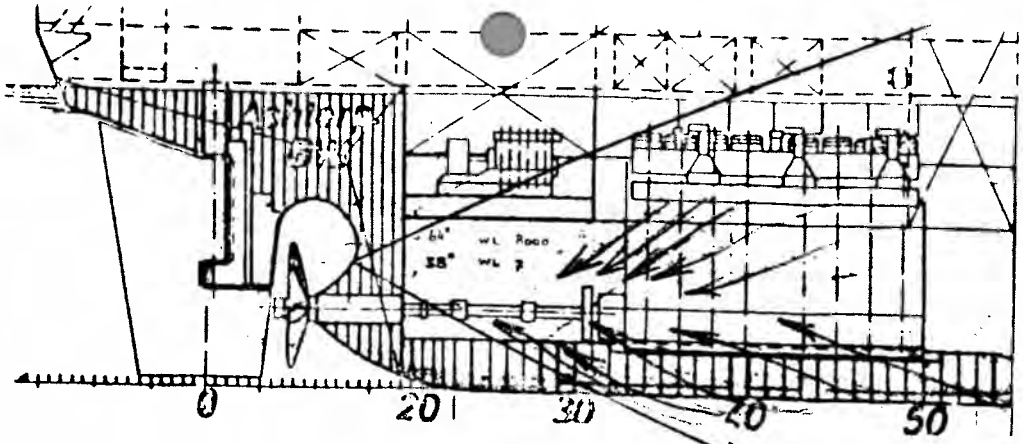
Extended skeg on centreline. For course keeping and good steering.

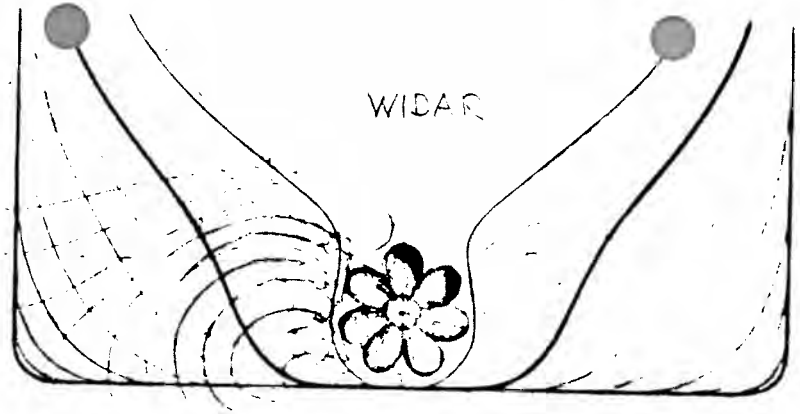


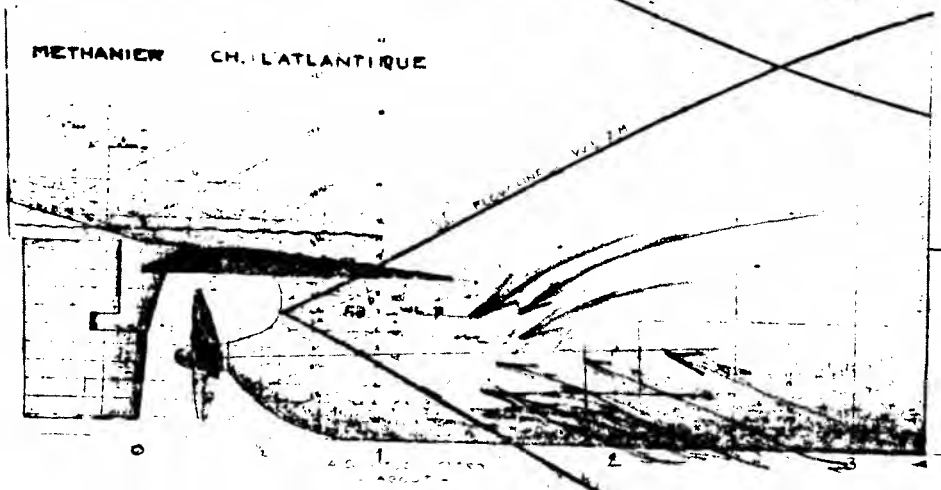


. Developable hull surface. Better course keeping. Small increase in s.h.p.

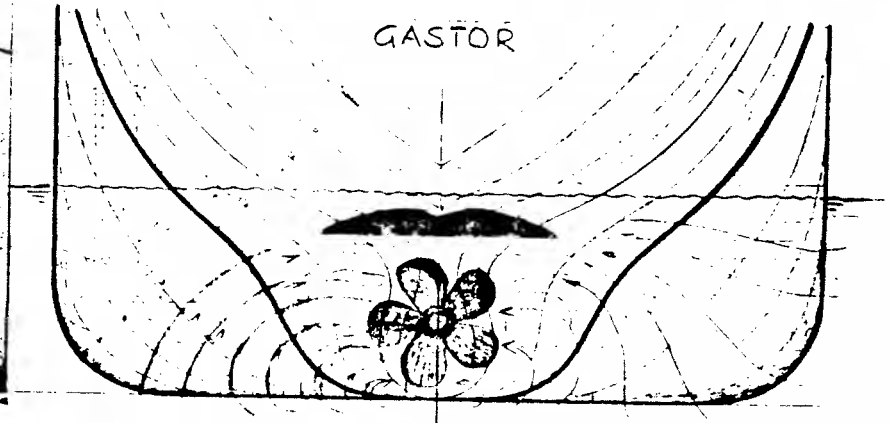

0
20
30
40
50
+ FIN →
34000 SHP/110 REVS
TUP3
FLOW LINE
PROPOSED FIN
FREE-FLOW-PROPELLER DESIGN
FLOW LINE
FREE-FLOW-PROPELLER

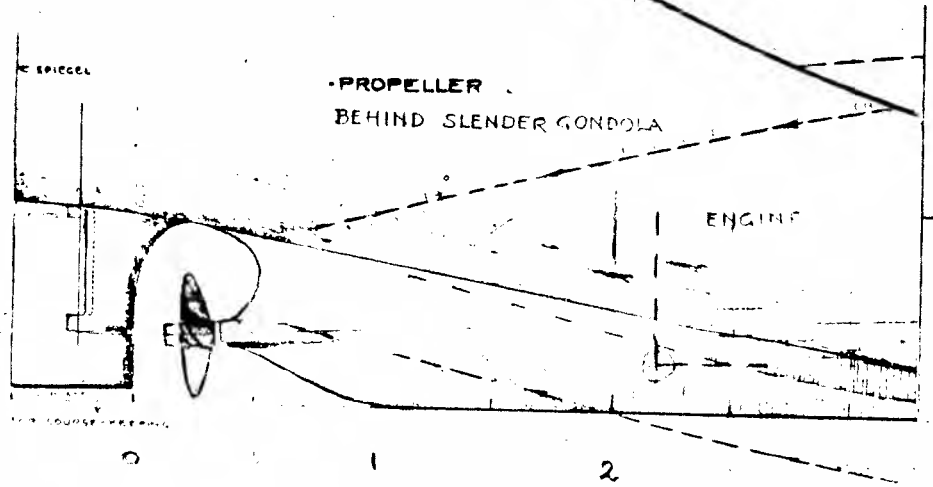


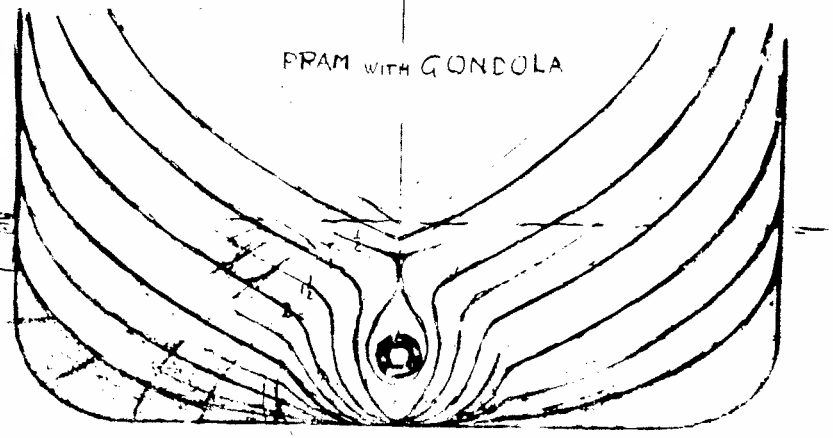




34000 SHP/110 REVS
TURB







1983

5

5-1

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Investigations into the propulsive performance of simplified full hull forms

A. Jonk and A. Rem

M 39
September 1980

FULL BODY SIMPLIFIED HULLS

JONK + REM

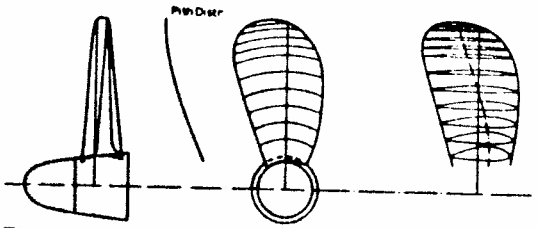


Fig. 6 Propeller for "simplified" configurations.

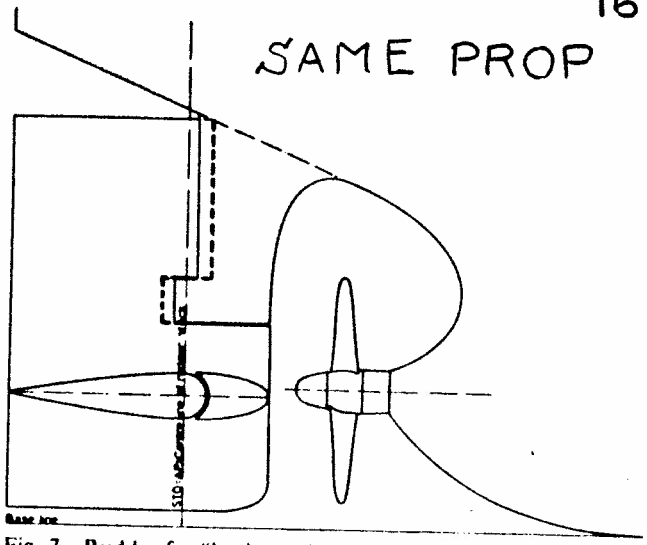


Fig. 7 Rudder for "basic model".

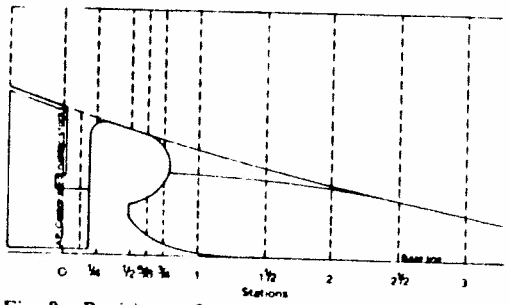
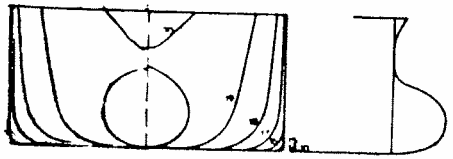
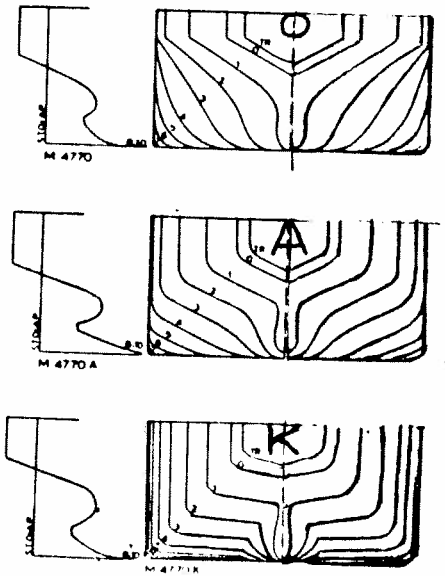


Fig. 8 Positions of tufts.

4770	O			
"	A			
"	K			



	P_E		P_D	
15 KN LOADED	100%		100 %	BASE
16 KN BALL	100		100	
	95		99	
	92		$91\frac{1}{2}$	BEST
	87		$100\frac{1}{2}$	
	90		$107\frac{1}{2}$?	



$\frac{L}{B} = 5.66$

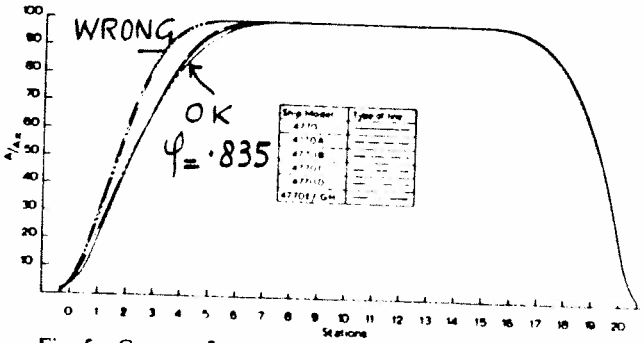


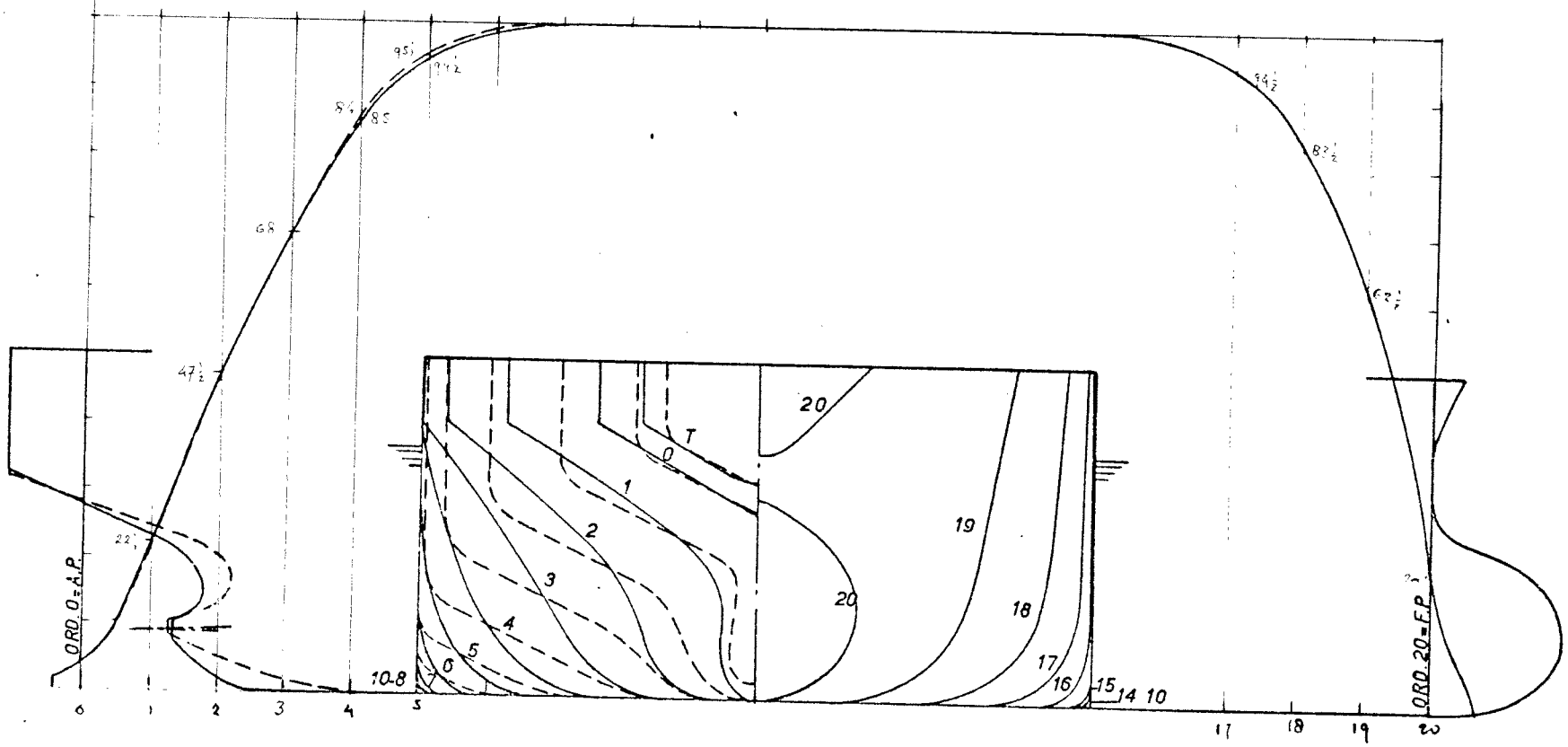
Fig. 5 Curves of sectional areas.

MODEL 4770 – 4770 A

BODYPLAN, STEM AND STERN OUTLINES AND CURVE OF SECTIONAL AREAS

MODEL 4770 ————————

MODEL 4770 A — — — — —

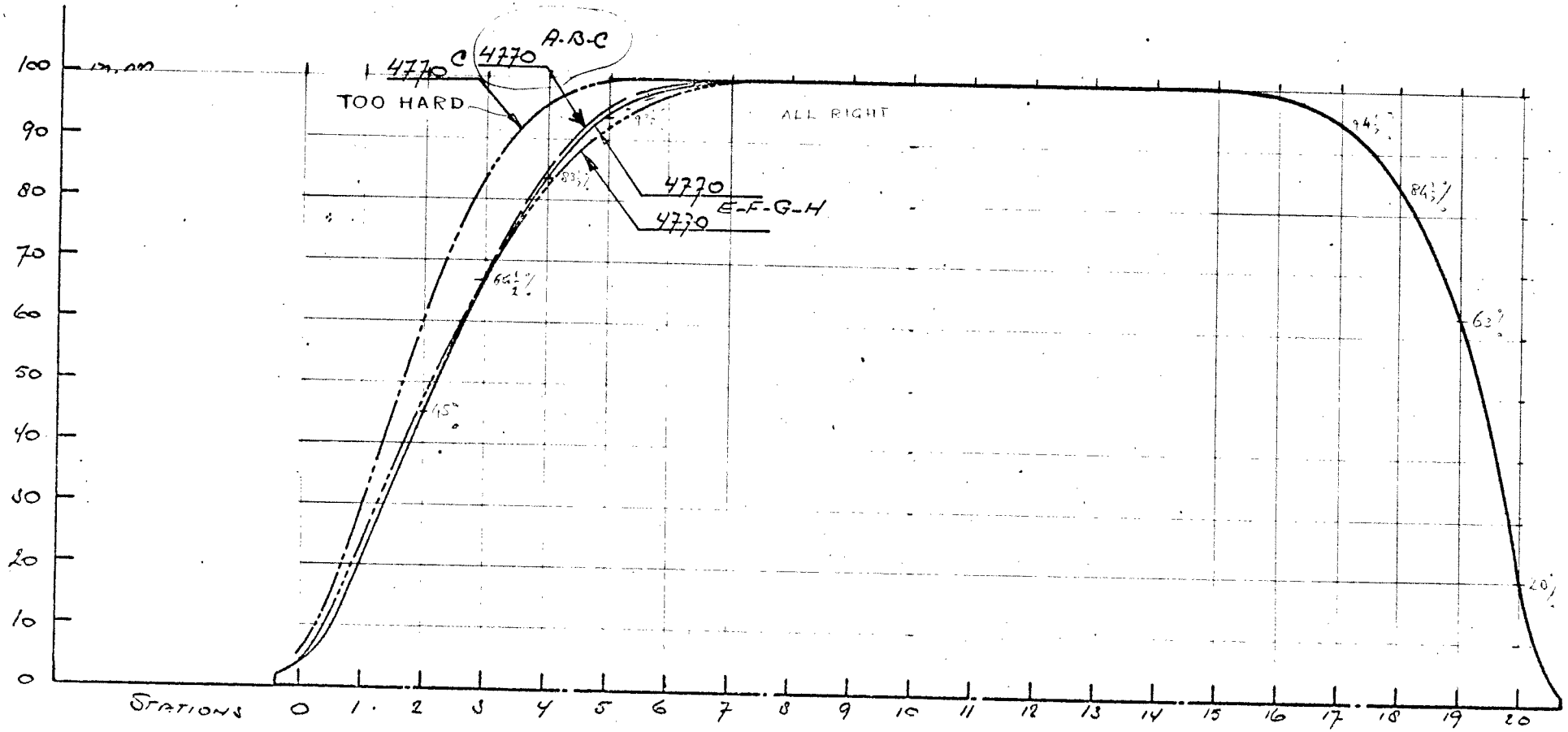


DRAWING NO. M4770-17

Ship Model 4770

Curves of Sectional Areas

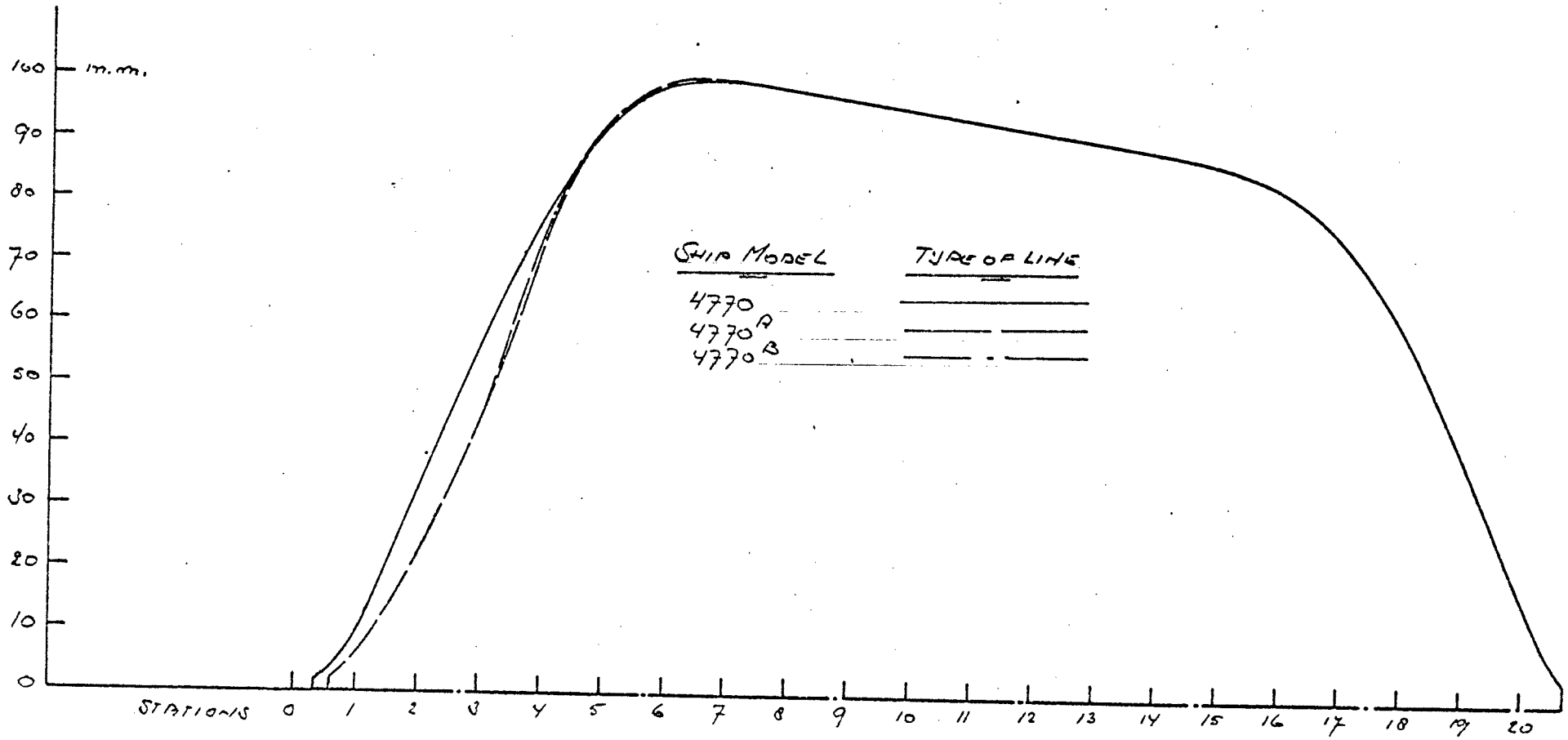
Draft. 24.23 m



SHIP MODEL 4770

CURVES OF SECTIONAL AREAS

DRAFT. TV = 7.90m / TA = 11.00m.



LOADED

Table I. Main particulars of the ship LOAD – condition.

Denomination	Symbol	Unit	Configuration										
			4770	4770A	4770B	4770C	4770D	4770E	4770F	4770G	4770H	4770J	4770K
Length between perpendiculars	L_{PP}	m	370.00	370.00	370.00	370.00	358.71	370.00	370.00	370.00	370.00	370.00	370.00
Breadth moulded	B	m	66.00	66.00	66.00	66.00	66.00	66.00	66.00	66.00	66.00	66.00	66.00
Draft moulded	T	m	24.23	24.23	24.23	24.23	24.23	24.23	24.23	24.23	24.23	24.23	24.23
Displacement volume	∇	m^3	494780	495783	496064	512816	494780	494697	494679	494688	495135	495983	493587
Displacement weight in sea water	Δ	tons	507150	508178	508466	525636	507150	507065	507046	507055	507513	508383	505927
Wetted surface without rudder	S	m^2	37396	37526	37645	38540	37262	38252	38148	38243	39252	38116	38113
Block coefficient	C_B	-	0.8362	0.8379	0.8384	0.8667	0.8625	0.8360	0.8360	0.8360	0.8368	0.8382	0.8342
Midship section coefficient	C_M	-	0.9993	0.9993	0.9993	0.9993	0.9993	0.9993	0.9993	0.9993	0.9993	0.9993	0.9993
Prismatic coefficient	C_P	-	0.8368	0.8385	0.8390	0.8673	0.8631	0.8366	0.8366	0.8366	0.8374	0.8388	0.8348
Longitudinal centre of buoyancy fwd. of midship (in % of L_{PP})	%	-	3.45	3.39	3.37	2.07	2.14	3.40	3.40	3.40	3.38	3.29	3.46
Breadth/draft ratio	B/T	-	2.72	2.72	2.72	2.72	2.72	2.72	2.72	2.72	2.72	2.72	2.72
Displacement length/breadth ratio	L_D/B	-	5.66	5.66	5.66	5.66	5.44	5.66	5.66	5.66	5.66	5.66	5.66

Table XI. Summary of resistance and self propulsion tests at 16 knots, loaded draft.

Denomination	Ship Model										
	4770	4770A	4770B	4770C	4770D	4770E	4770F	4770G	4770H	4770J	4770K
R_T in tons	345.86	325.47	328.39	349.40	340.31	314.48	302.83	321.70	341.35	307.55	299.15
P_E in HP	37958	35720	36040	38346	37349	34513	33236	35306	37462	33754	32832
T in tons	431.14	385.02	392.70	412.56	412.56	395.91	382.45	401.67	424.09	397.32	374.13
P_D in HP	56141	55713	57905	60317	60066	58827	57105	60152	63864	60596	56646
$\eta_D = \frac{P_E}{P_D}$	0.676	0.641	0.622	0.636	0.622	0.587	0.582	0.587	0.587	0.557	0.580
K_T	0.185	0.151	0.153	0.157	0.157	0.146	0.143	0.148	0.150	0.150	0.138
$10\ K_Q$	0.193	0.168	0.172	0.172	0.172	0.161	0.159	0.163	0.163	0.172	0.155
J	0.242	0.322	0.314	0.309	0.309	0.338	0.344	0.332	0.330	0.318	0.355
t	0.198	0.155	0.164	0.153	0.175	0.206	0.208	0.199	0.195	0.226	0.200
w_e	0.561	0.390	0.401	0.404	0.405	0.337	0.332	0.349	0.339	0.386	0.306
η_H	1.829	1.386	1.396	1.421	1.387	1.198	1.186	1.230	1.218	1.260	1.153
η_B	0.370	0.462	0.446	0.447	0.448	0.489	0.491	0.477	0.482	0.442	0.503
η_o	0.351	0.443	0.435	0.429	0.429	0.459	0.463	0.453	0.451	0.438	0.474
η_R	1.053	1.045	1.024	1.042	1.045	1.067	1.059	1.054	1.068	1.008	1.062

$$\frac{1-t}{1-\psi} \times \eta_B \times \eta_R = \eta_D = \frac{P_E}{P_D}$$

$$\frac{80.2}{43.9} \times 0{,}370 \times 1{,}053 = 0{,}676$$

BALLAST

Table II. Main particulars of the ship. Ballast condition.

Denomination	Symbol	Unit	Configuration					
			4770	4770A	4770B	4770F	4770J	4770K
Length between perpendiculars	L_{PP}	m	370.00	370.00	370.00	370.00	370.00	370.00
Breadth moulded	B	m	66.00	66.00	66.00	66.00	66.00	66.00
Draft moulded	T	m	7.90/11.00	7.90/11.00	7.90/11.00	7.90/11.00	7.90/11.00	7.90/11.00
Displacement volume	∇	m^3	180566	175761	175849	176829	178168	175684
Displacement weight in sea water	Δ	tons	185070	180155	180245	181250	182623	180076
Wetted surface without rudder	S	m^2	25281	25232	25281	26038	26010	26034

Table XIV. Summary of resistance and propulsion test results at ballast condition - speed 18 knots.

Denomination	Ship Model					
	4770	4770A	4770B	4770F	4770J	4770K
R_T in tons	295.12	266.20	275.29	274.08	263.36	262.84
P_E in HP	36438	32867	33989	33839	32516	32452
T in tons	380.53	329.28	346.58	372.84	352.98	347.86
P_D in HP	48963	45512	48076	55868	52112	52521
η_D	0.744	0.722	0.707	0.606	0.624	0.618
K_T	0.181	0.151	0.156	0.142	0.146	0.131
$10\ K_Q$	0.196	0.173	0.178	0.161	0.169	0.148
J	0.241	0.314	0.301	0.343	0.326	0.373
t	0.224	0.192	0.206	0.265	0.254	0.244
w_e	0.631	0.510	0.526	0.414	0.465	0.359
η_H	2.100	1.650	1.674	1.254	1.395	1.179
η_B	0.355	0.437	0.422	0.483	0.447	0.525
η_o	0.350	0.435	0.421	0.463	0.447	0.488
η_R	1.012	1.006	1.002	1.049	1.001	1.075

P_E - BALL

Table VII. Percentages P_t for the ballast condition.

V in knots	4770	4770A	4770B	4770F	4770J	4770K
14	100	91.3	92.9	94.7	87.8	88.1
15	100	91.8	93.9	95.4	88.7	89.6
16	100	92.3	94.4	95.6	89.1	90.4
17	100	91.1	94.2	94.7	89.3	89.9
18	100	90.2	93.3	92.9	89.2	89.1

P_E - LOADED

Table V. Percentages P_t for the loaded condition.

V in knots	4770	4770A	4770B	4770C	4770D	4770E	4770F	4770G	4770H	4770J	4770K
13	100	94.5	95.3	100.0	98.9	91.7	89.1	94.0	100.9	90.8	88.2
14	100	95.0	95.4	101.0	99.2	91.2	88.7	93.4	100.9	89.8	87.2
15	100	95.0	95.3	101.4	99.1	91.1	87.9	93.1	99.5	89.1	86.6
16	100	94.1	94.9	101.0	98.4	90.9	87.6	93.0	98.7	88.9	86.5
17	100	94.1	94.9	101.1	98.5	90.7	87.9	93.5	98.5	89.4	86.9

P_D BALL

Table XII. Percentages P_D for the ballast condition.

V in knots	4770	4770A	4770B	4770F	4770J	4770K
14	100	90.3	93.4	108.9	101.6	103.1
15	100	91.3	95.0	112.0	105.2	105.7
16	100	91.4	96.3	113.8	107.1	107.7
17	100	92.4	97.1	113.9	107.6	107.9
18	100	93.0	98.2	114.1	106.4	107.3

P_D - LOADED

Table IX. Percentages P_D for the loaded condition.

V in knots	4770	4770A	4770B	4770C	4770D	4770E	4770F	4770G	4770H	4770J	4770K
13	100	98.9	103.9	108.5	109.2	103.3	101.2	108.7	117.7	110.0	100.8
14	100	99.3	102.5	107.6	108.6	104.0	101.5	107.8	115.8	109.7	100.7
15	100	99.0	101.9	107.2	107.7	104.6	101.6	107.7	114.1	108.9	100.7
16	100	99.2	103.1	107.4	107.0	104.8	101.7	107.1	113.8	107.9	100.9
17	100	99.5	104.4	107.8	108.1	105.8	101.5	107.5	113.5	107.2	101.2

BEST

SIMPLIFIED HULLS

JONK + REM 1980

CE - BALL

Table VIII. Percentages C_E for the ballast condition.

V in knots	Ship Model					
	4770	4770A	4770B	4770F	4770J	4770K
14	100	108.9	105.2	90.6	97.5	95.3
15	100	107.6	103.4	88.0	94.4	92.9
16	100	107.4	102.0	86.7	92.6	91.2
17	100	106.3	101.2	86.7	92.1	91.0
18	100	105.7	100.0	86.3	93.0	91.5

CE - LOADED

Table VI. Percentages C_E for the loaded condition.

V in knots	Ship Model										
	4770	4770A	4770B	4770C	4770D	4770E	4770F	4770G	4770H	4770J	4770K
13	100	105.9	105.2	102.5	101.2	109.1	112.3	106.1	99.3	110.4	113.3
14	100	105.5	105.1	101.4	100.9	109.7	112.9	106.8	99.3	111.6	114.6
15	100	105.5	105.2	101.2	101.0	109.9	113.9	107.5	100.6	112.5	115.3
16	100	106.4	105.5	101.5	101.6	110.1	114.3	107.6	101.5	112.7	115.5
17	100	106.3	105.6	101.3	101.5	110.2	113.8	106.9	101.5	112.0	114.9

CA - BALL

Table XIII. C_A values in percentages. Ballast condition.

V in knots	Ship Model					
	4770	4770A	4770B	4770F	4770J	4770K
14	100	107.6	105.8	104.2	113.0	111.6
15	100	107.1	104.6	103.4	111.7	109.5
16	100	106.5	104.1	103.1	111.3	108.6
17	100	107.8	104.3	104.1	111.1	109.3
18	100	108.8	105.4	106.2	111.0	110.2

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PROBABLY A MISTAKE

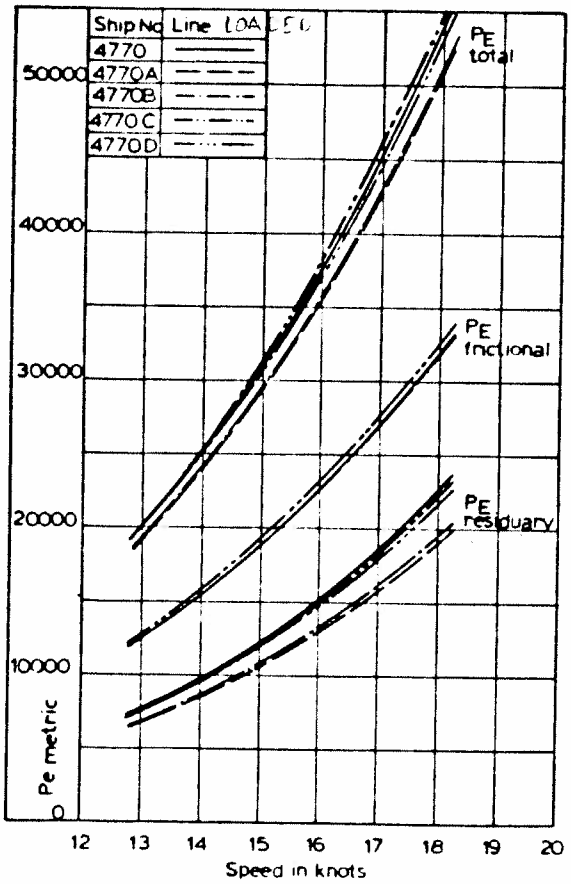
CA - LOADED

Table X. Percentages C_A for the loaded condition.

V in knots	Ship Model										
	4770	4770A	4770B	4770C	4770D	4770E	4770F	4770G	4770H	4770J	4770K
13	100	101.3	96.3	94.4	91.7	96.8	98.9	92.1	85.1	91.0	99.1
14	100	100.9	97.9	95.1	92.1	96.2	98.5	92.8	86.4	91.3	99.1
15	100	101.3	98.3	95.5	92.9	95.7	98.5	92.9	87.8	92.1	99.1
16	100	100.9	97.2	95.3	93.5	95.5	98.1	93.3	87.9	92.9	98.9
17	100	100.6	96.1	95.0	92.6	94.6	98.3	93.1	88.3	93.5	98.7

LOADED

P_E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4770B
4770C
4770D
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PE residuary
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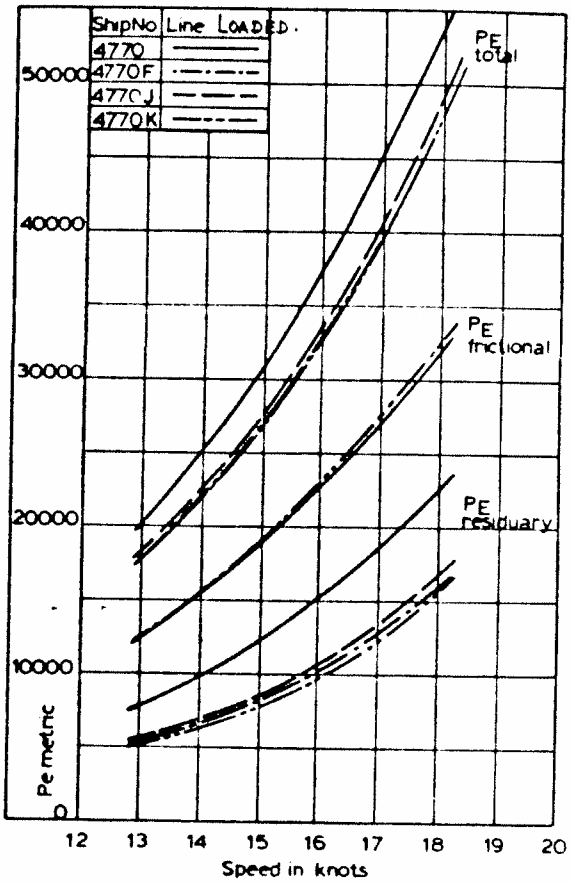
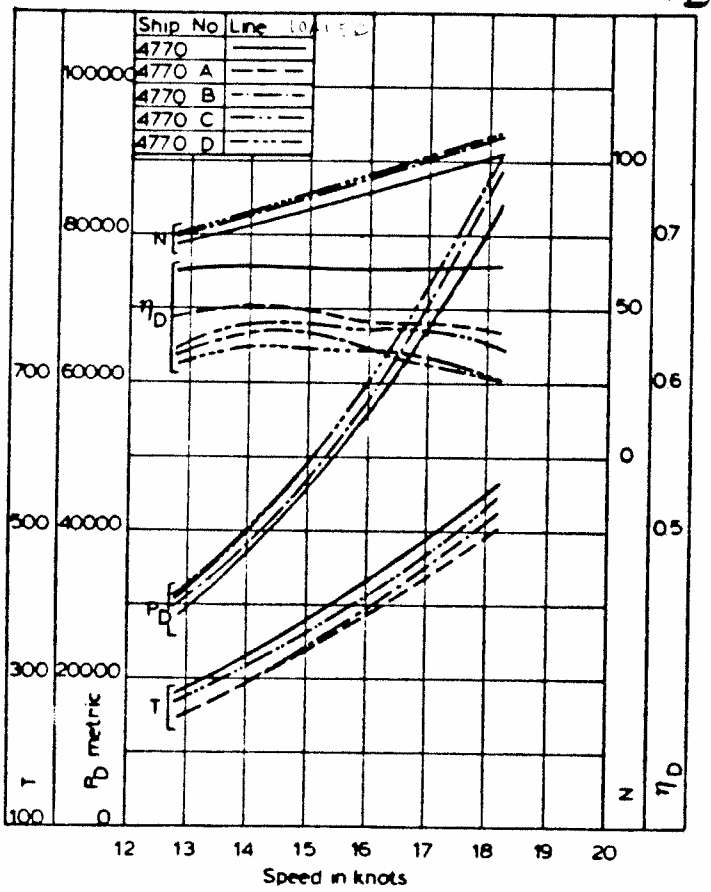
40000
30000
20000
10000
0
Pe metric
12 13 14 15 16 17 18 19 20
Speed in knots


ShipNo
Line
4770
4770F
4770J
4770K
PE total
PE frictional
PE residuary
50000
40000
30000
20000
10000
0
Pe metric
12 13 14 15 16 17 18 19 20
Speed in knots

P_D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4770 B
4770 C
4770 D
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PD
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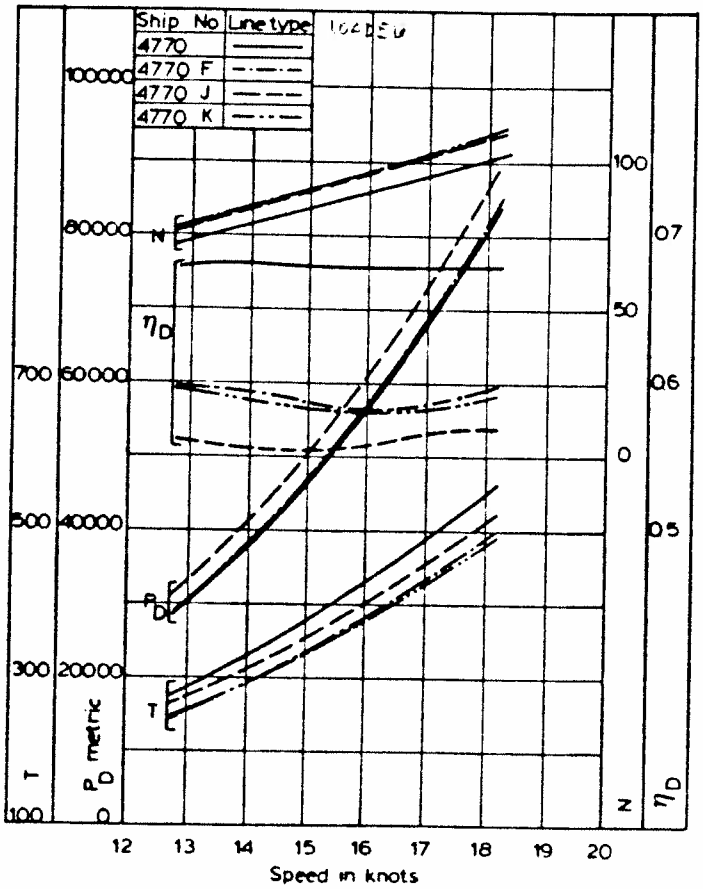
60000
40000
20000
0
PD metric
700
500
300
100
100
50
0
N
0.7
0.6
0.5
ηD
12 13 14 15 16 17 18 19 20
Speed in knots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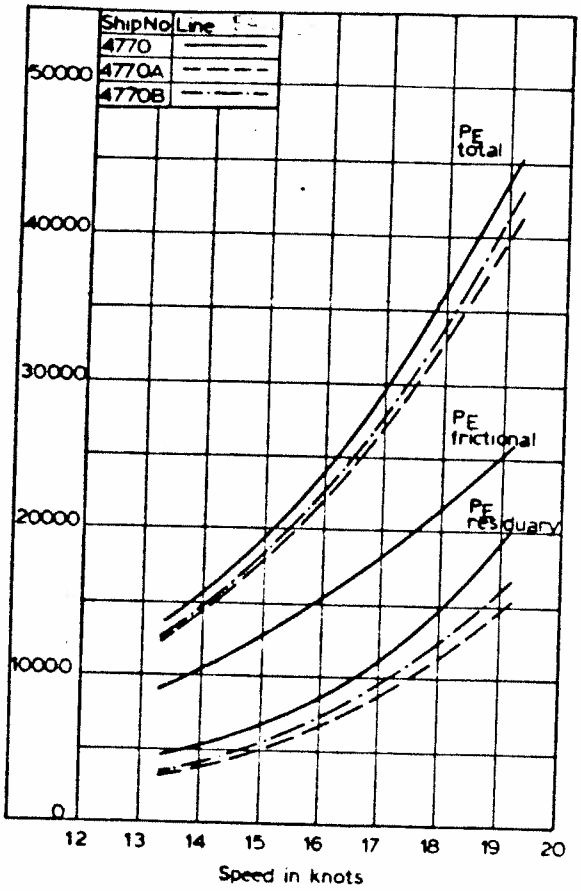
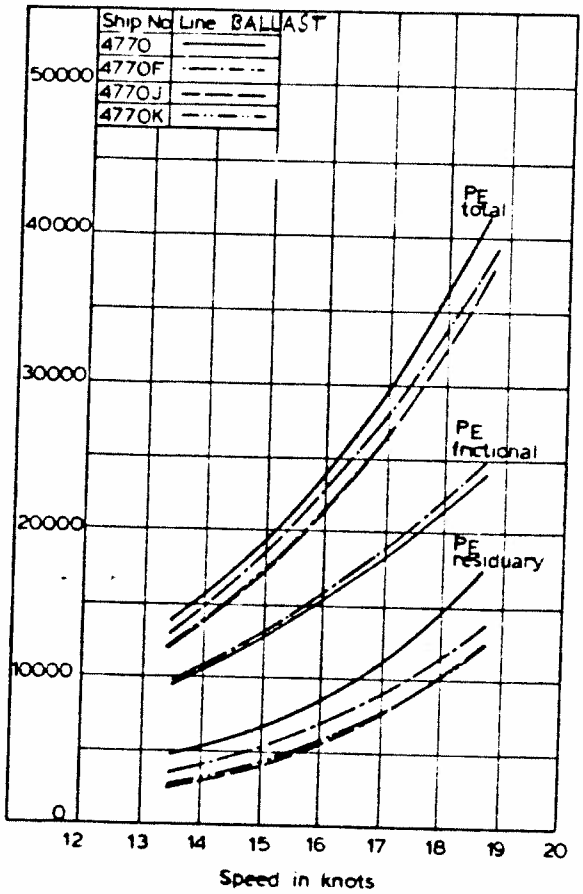
4770 K
N
ηD
PD
T
100000
80000
60000
40000
20000
0
PD metric
700
500
300
100
100
50
0
N
0.7
0.6
0.5
ηD
12 13 14 15 16 17 18 19 20
Speed in knots

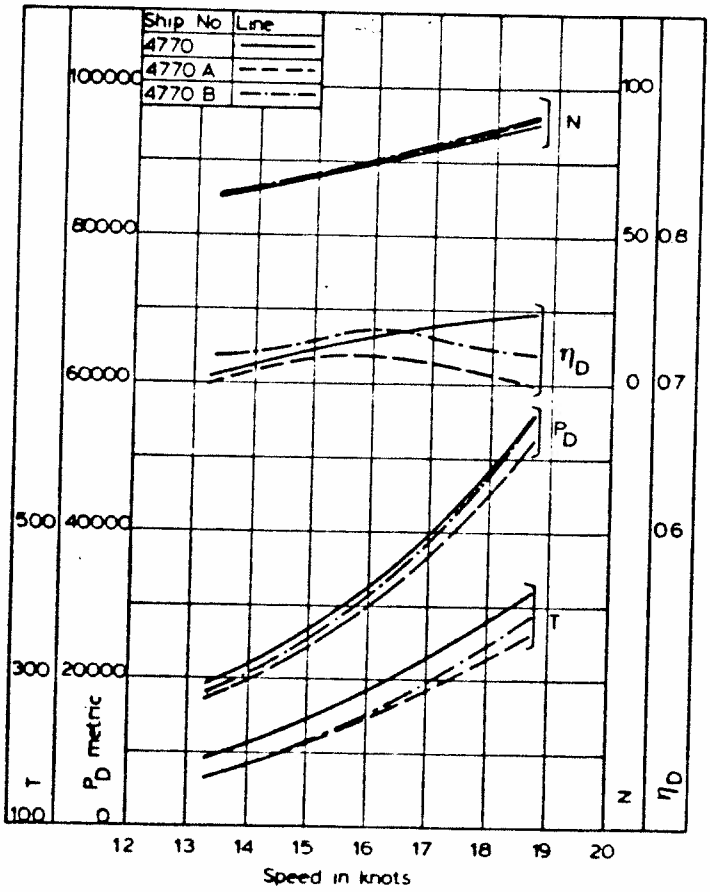
BALLAST

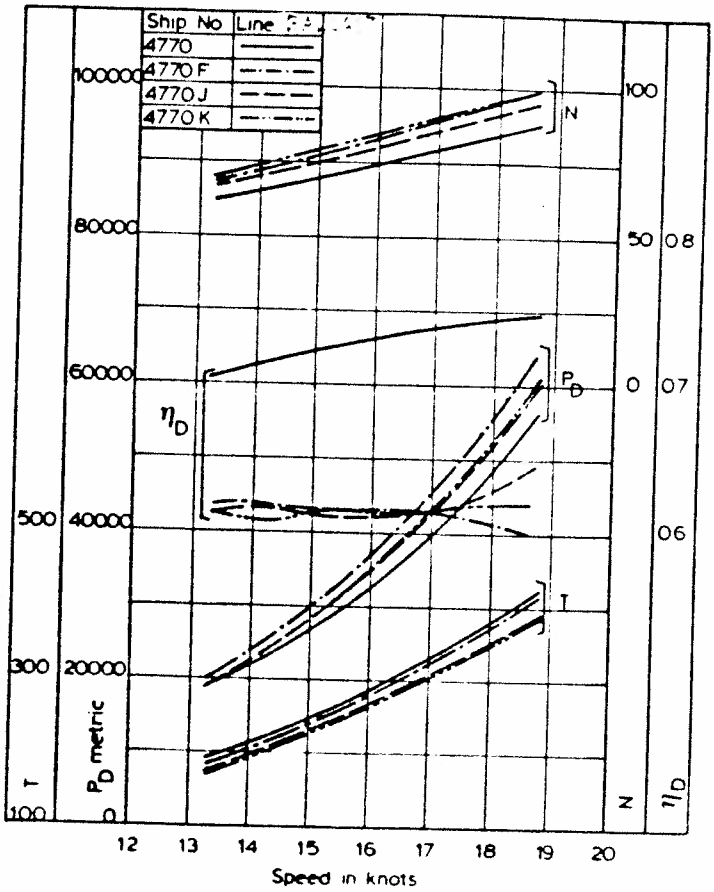
P_E

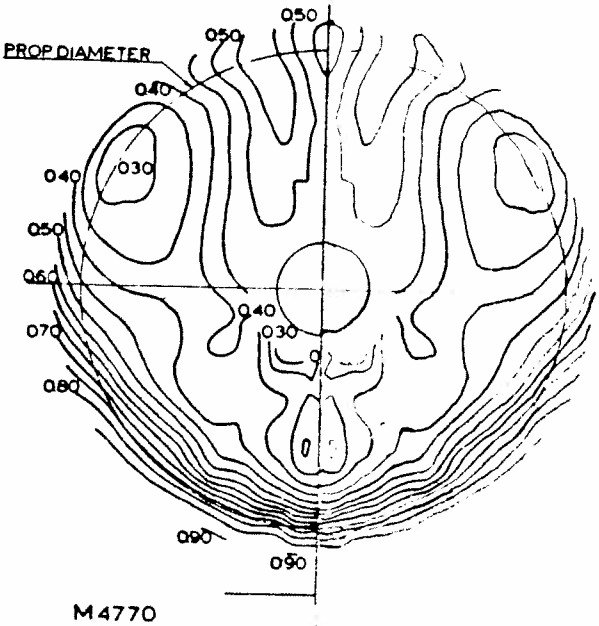




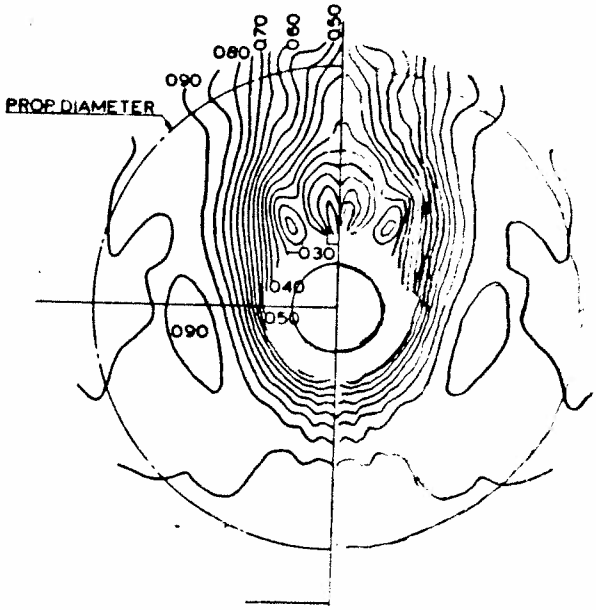
P_D







M4770



M.4770K

$\eta_{PROP} = 0.351$

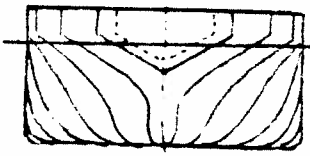
$\frac{1-0.198}{1-0.561} = 1.829$

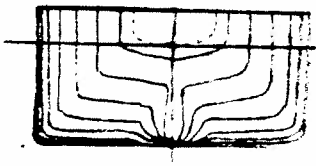
$\eta_R = 1.053$

$\eta_D = 0.676$

16 KN

LOADED



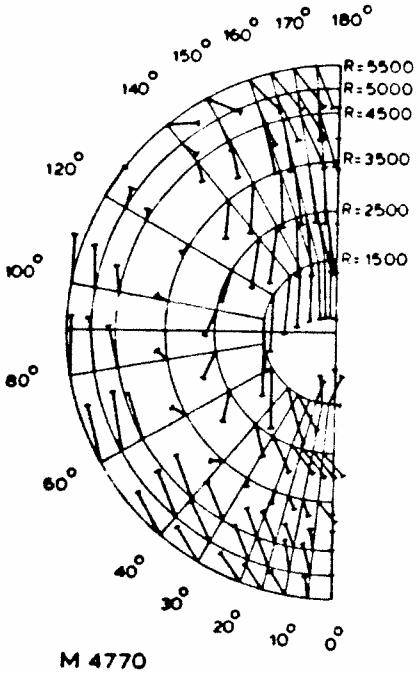


$\eta_{PROP} = 0.474$

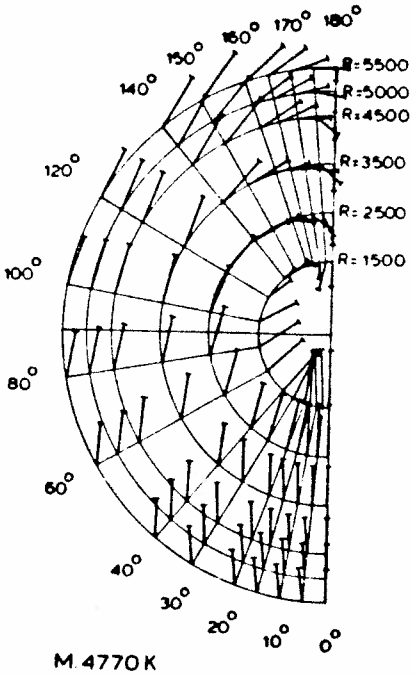
$\frac{1-0.300}{1-0.306} = 1.153$

$\eta_R = 1.062$

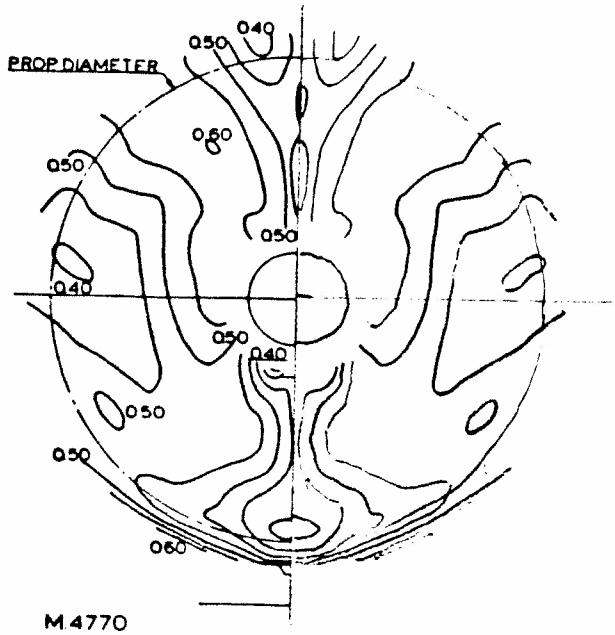
$\eta_D = 0.580$



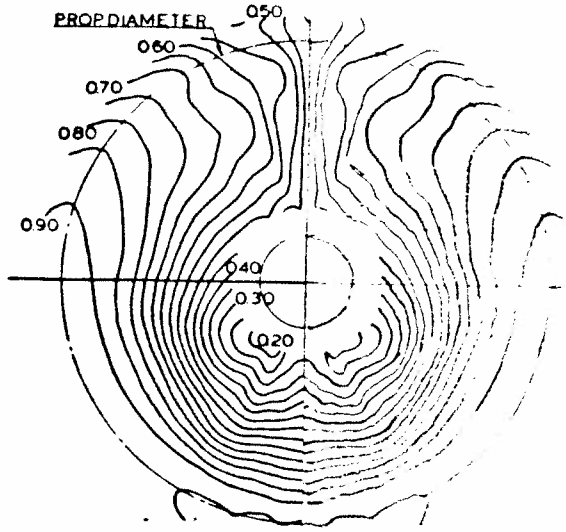
M 4770



M.4770K



M.4770



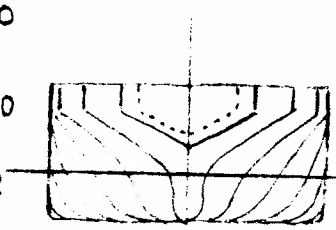
M.4770K

$\eta_{PROP} = 0{,}350$

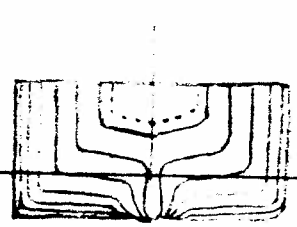
$\frac{1-0{,}224}{1-0{,}631} = 2{,}100$

$\eta_R = 1.012$

$\eta_D = 0.744$



18 KN
BALLAST

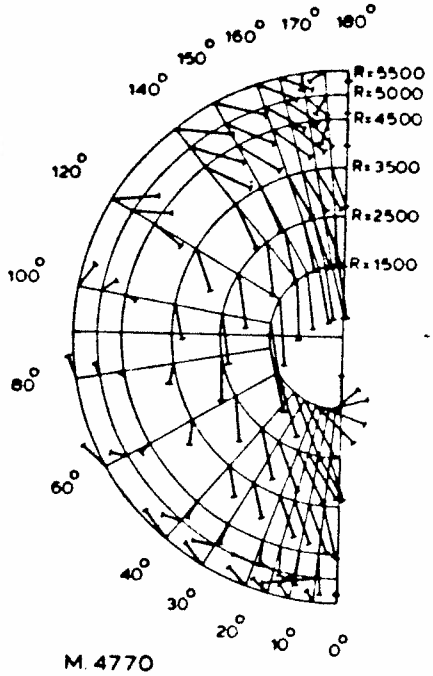


$\eta_{PROP} = 0.488$

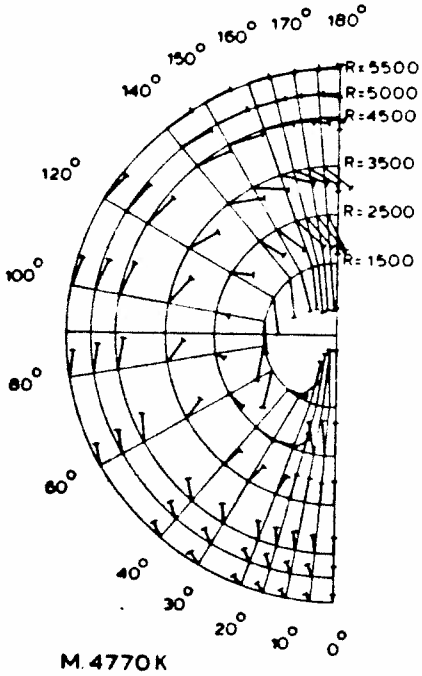
$\frac{1-0.244}{1-0{,}359} = 1.179$

$\eta_R = 1.075$

$\eta_D = 0{,}618$



M.4770



M.4770K

6 Conclusions

- The investigations have shown that simplified "pram" type afterbodies can be designed with an improvement in resistance of about 10% with regard to a conventional form, the resistance of which was considered to be satisfactory.
- The best simplified designs are about equal in propulsive performance with regard to the conventional hull in the loaded condition. In the ballast condition, however, a gain in required power of some 7% is obtained with the "pram" type hull 4770A with regard to the conventional hull 4770. Compared with the conventional hull the simplified versions all show the need for more required power in the ballast condition.
- The simplified hull forms show an interaction between the ship and the propeller which is completely different from that of the conventional form: ship models 4770A to D have lower wake fractions, resulting in a lower hull efficiency than the conventional form, whereas ship models 4770E to K show thrust deduction fractions equal to that of the conventional hull form and much lower wake fractions, resulting in very low hull efficiencies.
- As a result of the low hull efficiency, the total efficiency η_D of the simplified "pram" type is very low, resulting in a relatively poor propulsive performance.
- The simplified "pram" type hull forms lead to a peripherally more homogeneous nominal wake field with a pronounced wake peak at 12 o'clock position.
- Thrust deduction fractions, nominal and effective mean wake fractions, in loaded and ballast drafts, indicate the interaction between propeller and ship's hull and can give insight into suppression of flow separation in front of the propeller by the propeller and propeller generated flow separation above the propeller.
- More systematic research is necessary to study the possibilities for improving the propeller-hull interaction.
- The design of a "gondola" for simplified hull forms needs special attention in order to avoid a poor performance, especially in the ballast condition.

References

1. KISS, R. K., Aspect of simplified hull forms. Past, present and future. SNAME, San Diego, section February, 1972.

6

INCREASING PROPELLER Ø

IN ORDER

TO REDUCE FUEL CONSUMPTION

AS BASE:

PRODUCT TANKER "MAASSLUIS" 38.000 DWT - 11.3 M

MCR 13.100 BHP/	123 RPM		Ø = 5800	15 KN
	↓ 90	"	Ø = 7100	
	60	"	Ø = 8800	
?	40	"	Ø = 10 800	

SEE ALSO P - 1

KEPT IDENTICAL:

SHIPS PROPULSION-TAIL, INCL ENGINE FOUNDATION-LOCATION

SLOW-TURNING PROPELLERS,

- give savings on fuel consumption, especially for vessels having a relatively low sailing speed and large fulness (Blocks .75/.8/.85).

- Large propeller diameter in combination with low R.P.M. has a considerably higher propulsive efficiency.

- Problem, however, is "How to house the large propeller ?". The stern, due to receive that large propeller should yet be of sufficient sharpness. Whilst L.o.a. is not to be altered, this means that the vessel will loose in cargo deadweight capacity, because the waterflow towards the propeller requires a certain slenderness.

- Another problem is formed by the "I.M.O." requirement that the propeller must be totally submerged in ship's ballasted condition. This has as a consequence that (the shape with) the biggest propeller also needs an increased quantity of waterballast.

- Here now the dish-shaped hullform (so-called 'pram'-shape) comes into the picture, her Loaded Draft and Light Draft differing relatively little.

- The pram shape also offers more room for the large-diameter propeller. Problem here, however, is that the directly coupled diesel engine will need a separate 'gondola' protuberance properly lined onto the hullbottom. Vessels with geared engines need sufficient gondola for the gear-box (only).

- We thus arrive at the following alternatives as to R.P.M. and profitableness:-
  (1) Direct-coupled/slow-turning/Uniflow scavenging/few cylinders diesel eng.
  (2) Medium speed diesel engine(s) with allied reduction gear.

- The shaft-generator coupled to main engine and providing electric power at sea (thus saving on expensive D.O. fuel) has to be incorporated into the energy-balance. Questions to be answered are:-
  - In which case the fuel consumption is smallest ?
  - In which case the engine room is smallest ?

- Of great importance is the question: What clearance for this propeller ? The answer thereto may need measurements of pressure fluctuations by model testing tanks and full-size measurements. Also at varying skew.

Particulars of ship "MAASSLUIS"/ TYPE

Ship model No.		5734 B	5734 B
Length between perpendiculars	L_{PP}	164.00	164.00 m
Length on waterline	L_{WL}	167.68	160.75 m
Breadth moulded	B	32.24	32.24 m
Draft moulded on A.P.	T_A	11.30	4.50 m
Draft moulded on F.P.	T_F	11.30	7.00 m
Displacement volume moulded	∇	46167	21896 m^3
Displacement weight in sea water	Δ	47321	22443 m.t.
Wetted surface, bare hull	S_1	7397	5435 m^2
Wetted surface, rudder inclusive	S_2	7496	5500 m^2
L.C.B. position forward of midship station in % of L_{PP}	θ	3.10	2.99 %
Block coefficient	C_B	0.772	-
Midship section coefficient	C_M	0.996	-
Prismatic coefficient	C_P	0.775	-
Length-Breadth ratio	L_{PP}/B	5.09	-
Breadth-Draft ratio	B/T	2.85	-

Ship model details:

Ship model material: wood

Linear scale ratio λ = 23.7

Turbulence induced by studs.

Characteristics of propellers

Propeller No.		4540 Stock prop.	5546 final Lips prop.
Diameter	D	5925	5800 mm
Pitch at 0.7R	$P_{0.7}$	4278	4242 mm
Pitch ratio	$P_{0.7}/D$	0.722	0.731
Boss-diameter ratio	d/D	0.194	0.178
Expanded blade area ratio	A_E/A_O	0.670	0.586
Number of blades	Z	5	4

η_D

PROPULSION TEST RESULTS

Test No. 39502

Model condition: BULBOUS BOW I
SCREW APERTURE I
RUDDER I

Ship model No. 5734B
Propeller model No. 5546
Scale ratio λ = 23.7000

Draft moulded on FP = 11.300 m
on AP = 11.300 m
mean = 11.300 m

Displacement volume = ∇ = 46167 m^3

Skin friction correction F_D **from model to ship is according to** ITTC INCREMENTAL RESISTANCE COEFFICIENT CA = 0.000100

$P_{D\,TRIALS} = P_D \times 1.02$

	Ship Speed in knots V_S	P_D	P_E	$\eta_D = \frac{P_E}{P_D}$	C_A	Revs/min n_S	Resistance R_T in tons	Thrust T in tons	$t = \frac{T-R_T}{T}$
4690	12.00	4598	3467	0.754	492	91.4	42.13	51.85	0.187
5370	12.50	5215	3920	0.752	490	95.9	45.72	56.49	0.191
6030	13.00	5903	4422	0.749	487	100.0	49.59	61.47	0.193
6790	13.50	6651	4968	0.747	484	103.9	53.65	66.79	0.197
7660	14.00	7507	5567	0.742	478	108.1	57.97	72.39	0.199
8640	14.50	8468	6242	0.737	471	112.1	62.76	78.46	0.200
9780	15.00	9579	7020	0.733	461	116.6	68.23	85.42	0.201
11050	15.50	10825	7929	0.732	450	121.5	74.58	92.72	0.196
12690	16.00	12405	9053	0.730	432	127.0	82.49	101.52	0.187
14730	16.50	14440	10515	0.728	407	133.1	92.91	113.18	0.179
17250	17.00	17424	12616	0.724	369	139.6	108.19	130.51	0.171

VALUES ARE GIVEN IN METRIC UNITS

P_D = delivered HP at propeller.
P_E = effective HP.
n_S = RPM of the ship's screw under the below conditions, not corrected for difference between the wake of ship and model, without any allowance

$$C_A = \frac{\Delta_1^{2/3} V_S^3}{P_D \text{ metric}}$$

Δ_1 = Displacement weight in salt water, in metric tons.

British HP = $\frac{75}{76}$ metric HP

Note: Results of self-propulsion tests as stated above refer to the self propulsion point of ship.
The results are calculated direct from measured model values without any allowance for appendages not present in the model, wind and sea; so values for ship are for tank condition corrected for salt water (γ_1 = 1.025) and 15° centigrade.
Model and ship dimensions are moulded.

h_p

PROPULSION TEST RESULTS

Test No. 39503

Ship model No.5734B
Propeller model No.5546
Scale ratio λ = 23.7000

Model condition: BULBOUS BOW I
SCREW APERTURE I
RUDDER I

Draft moulded on FP = 4.500 m
on AP = 7.000 m
mean = 5.750 m

Displacement volume ∇ = 21896 m³

Skin friction correction F_D **from model to ship is according to** ITTC INCREMENTAL RESISTANCE COEFFICIENT CA = 0.000100

$P_{TRIALS} = P_D \times 1.06$

Ship Speed in knots V_S	P_D	P_E	$\eta_D = \frac{P_E}{P_D}$	C_A	Revs/min n_S	Resistance R_T in tons	Thrust T in tons	$t = \frac{T-R_T}{T}$
13.00	4109	3382	0.823	425	87.3	37.92	48.17	0.213
13.50	4701	3832	0.815	416	91.2	41.38	52.87	0.217
14.00	5371	4316	0.804	407	95.2	44.95	57.92	0.224
14.50	6116	4862	0.795	397	99.3	48.89	63.24	0.227
15.00	6944	5459	0.786	387	103.8	53.06	68.97	0.231
15.50	7861	6111	0.777	377	108.4	57.47	74.91	0.233
16.00	8883	6818	0.768	367	113.3	62.12	81.19	0.235
16.50	9987	7623	0.763	358	117.9	67.36	87.74	0.232
17.00	11207	8525	0.761	349	122.4	73.10	94.63	0.227
17.50	12513	9607	0.768	341	126.4	80.03	102.00	0.215
18.00	13926	10818	0.777	333	130.3	87.62	109.57	0.200

VALUES ARE GIVEN IN METRIC UNITS

P_D = delivered HP at propeller.
P_E = effective HP.
n_S = RPM of the ship's screw under the below conditions, not corrected for difference between the wake of ship and model, without any allowance

$$C_A = \frac{\Delta_1^{2/3} V_S^3}{P_D \text{ metric}}$$

Δ_1 = Displacement weight in salt water, in metric tons.

British HP = $\frac{75}{76}$ metric HP

Note: Results of self-propulsion tests as stated above refer to the self propulsion point of ship.
The results are calculated direct from measured model values without any allowance for appendages not present in the model, wind and sea; so values for ship are for tank condition corrected for salt water (γ_1 = 1.025) and 15° centigrade.
Model and ship dimensions are moulded.

PREDICTION

SHIP MODEL No. 5734 B

PROPELLER MODEL No. 5546

"MAASSLUIS"

6-6

TANK VALUES ARE BASED ON RESULTS OF PROPULSION TEST No. 39502

DRAFT 11.30 m

LOADED

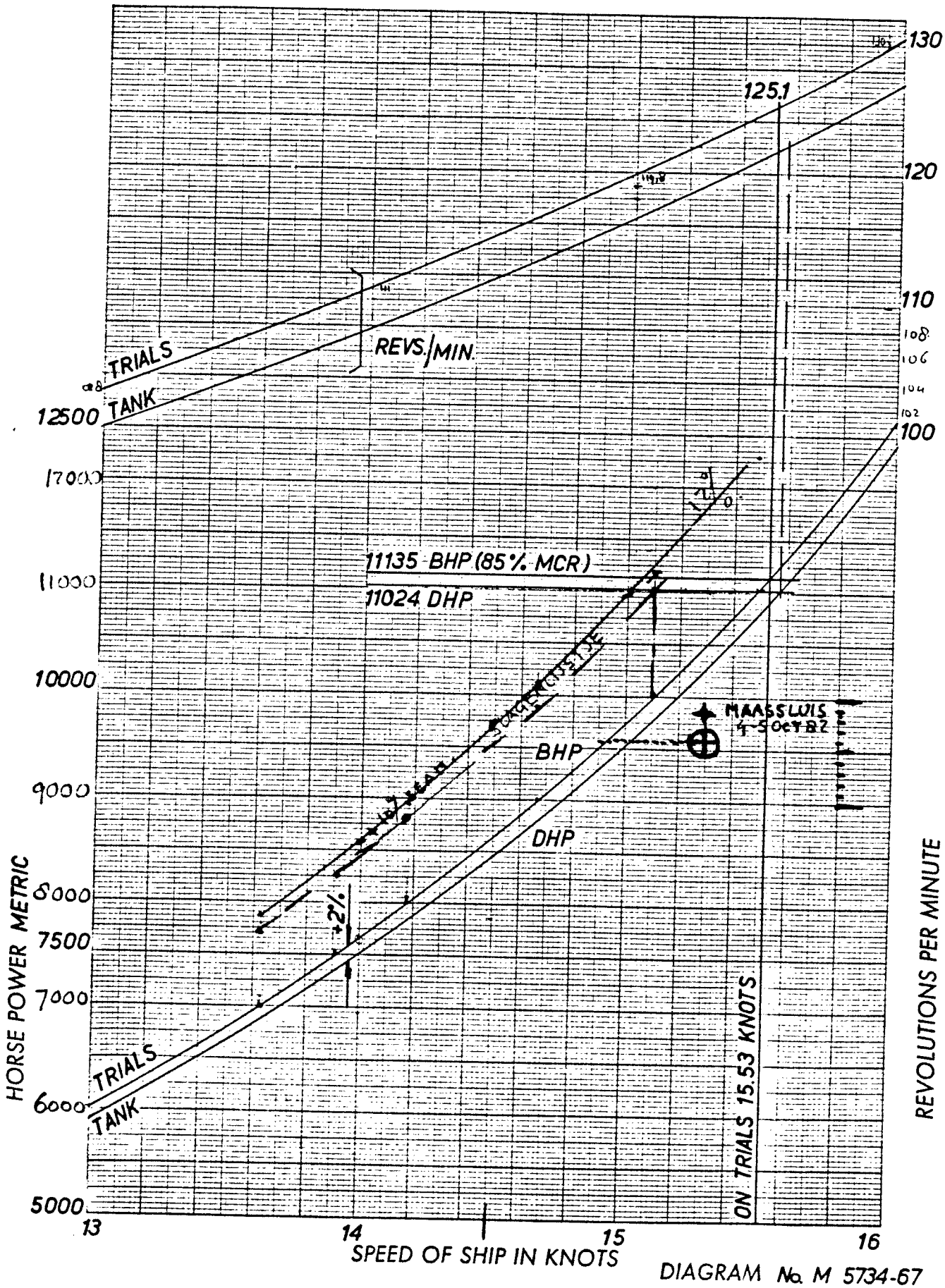


DIAGRAM No. M 5734-67

PREDICTION

SHIP MODEL No. 5734 B

PROPELLER MODEL No. 5546

MAASSLUIS

6·7

TANK VALUES ARE BASED ON RESULTS OF PROPULSION TEST No 39503

DRAFT 4.50/7.00 m

BALLAST

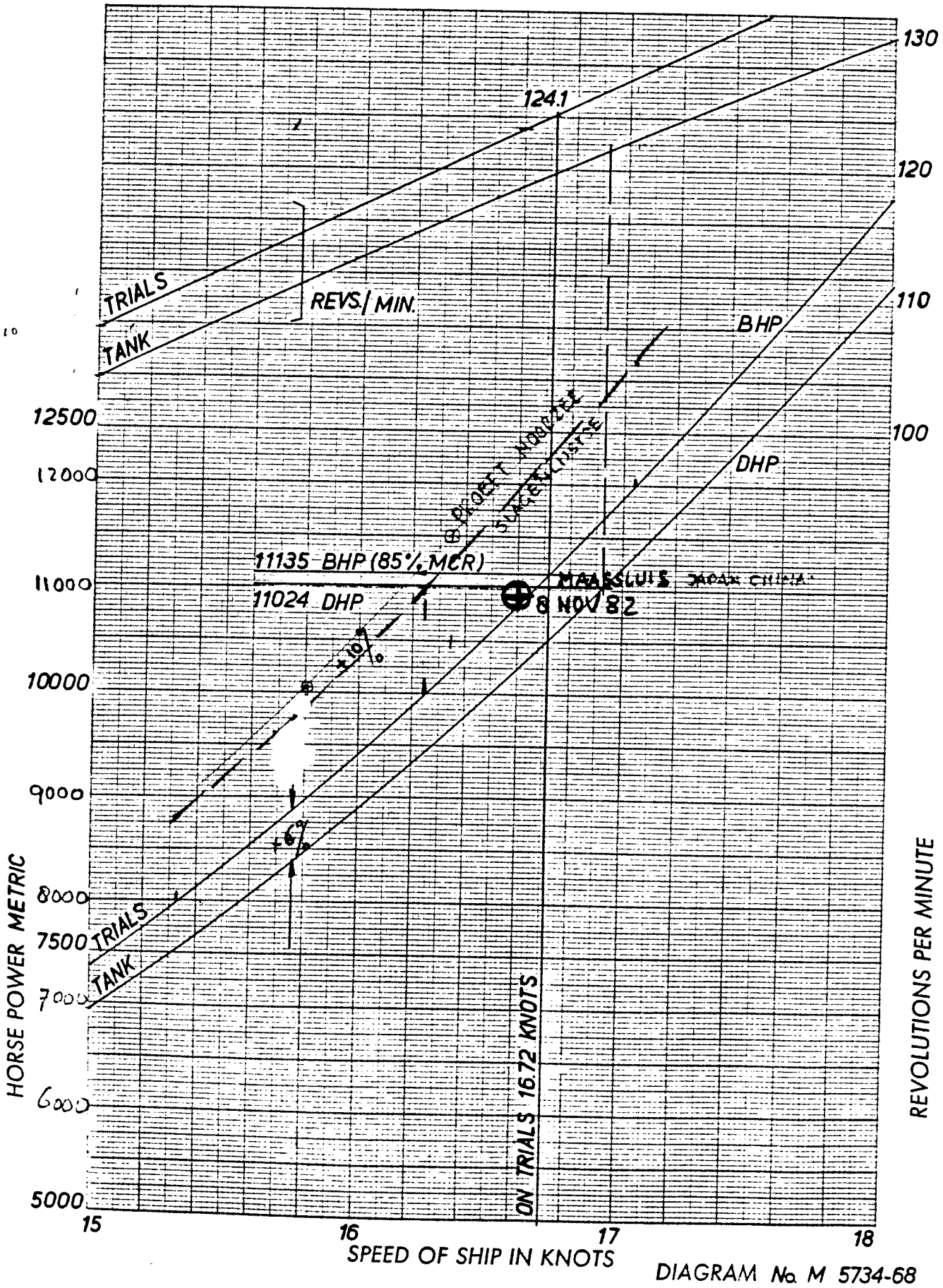


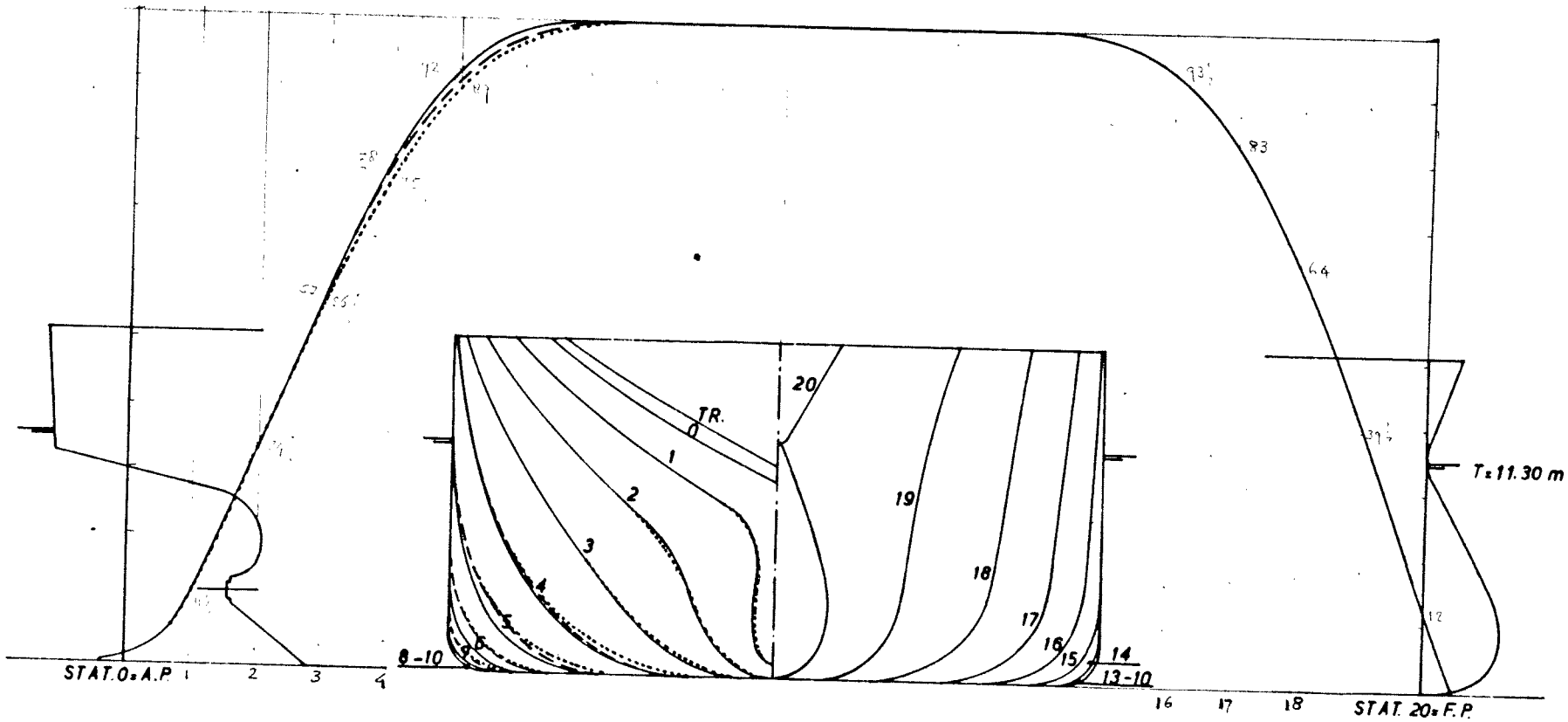
DIAGRAM No. M 5734-68

BODY PLAN, STEM AND STERN OUTLINES AND CURVE OF SECTIONAL AREAS

SHIP MODEL No.5734 ————

SHIP MODEL No.5734 A— — — —

SHIP MODEL No.5734B ..... K

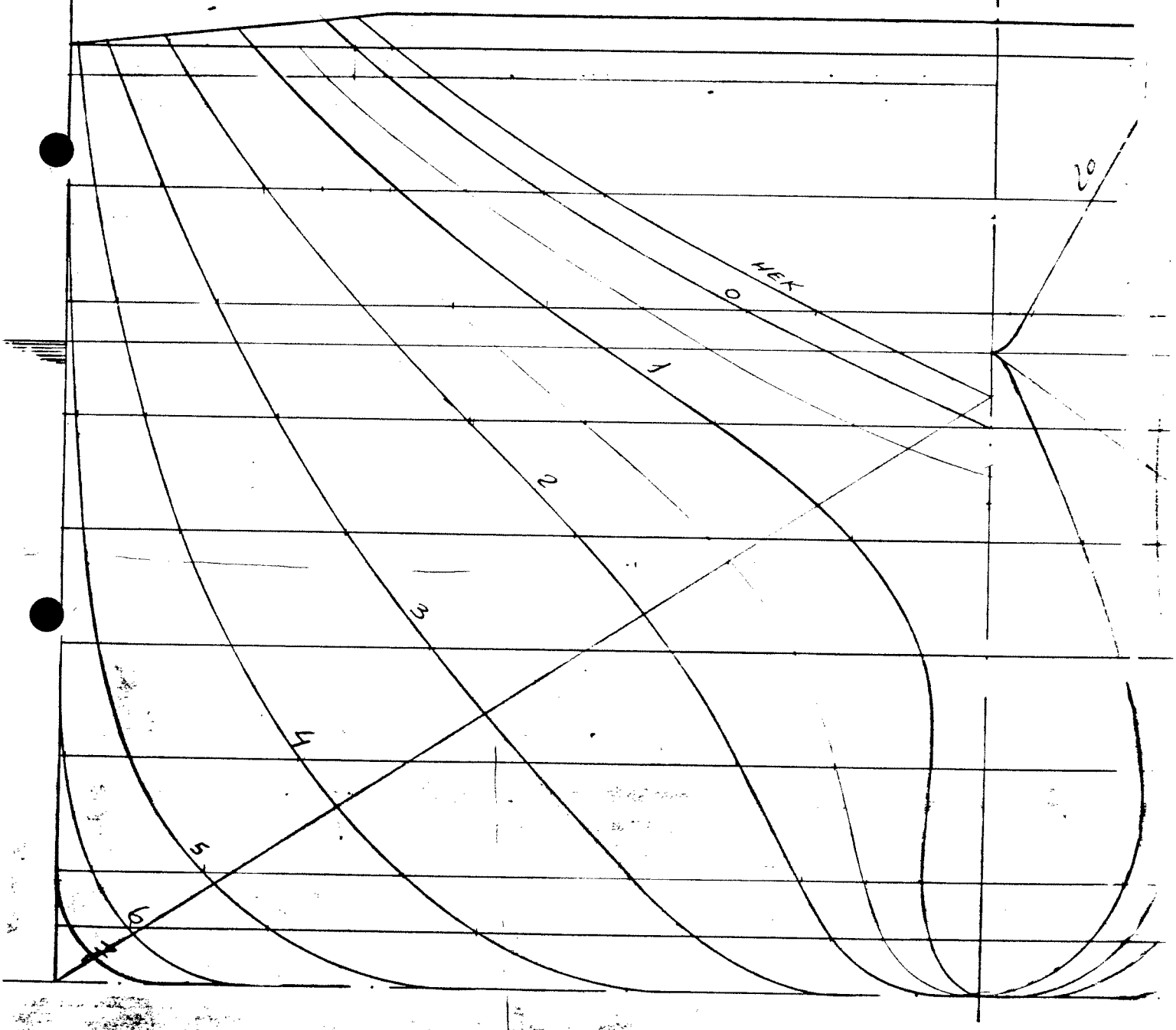


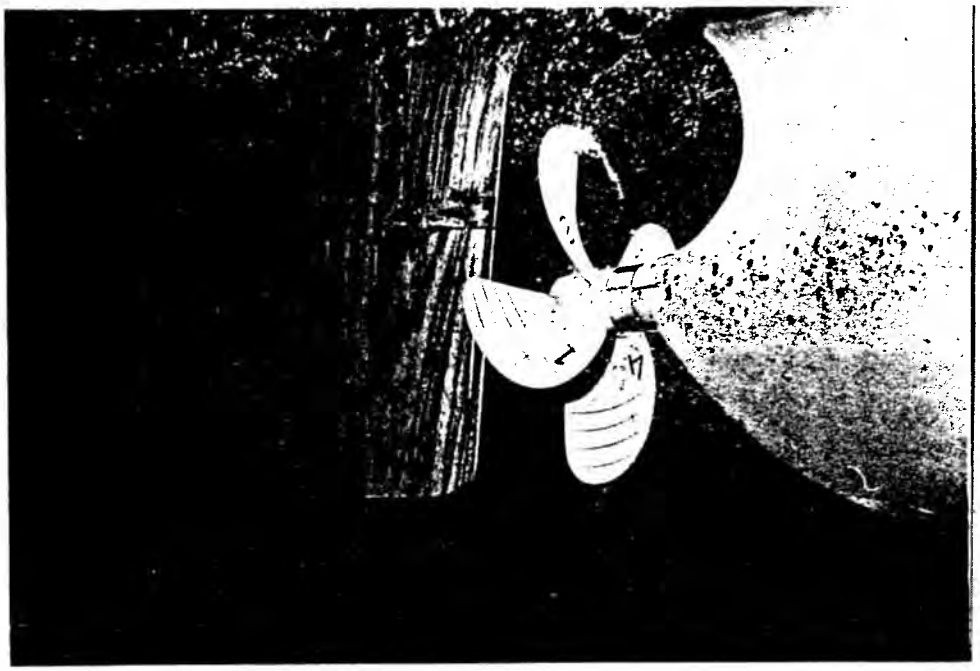
DRAWING No. M5734-19

5734 B

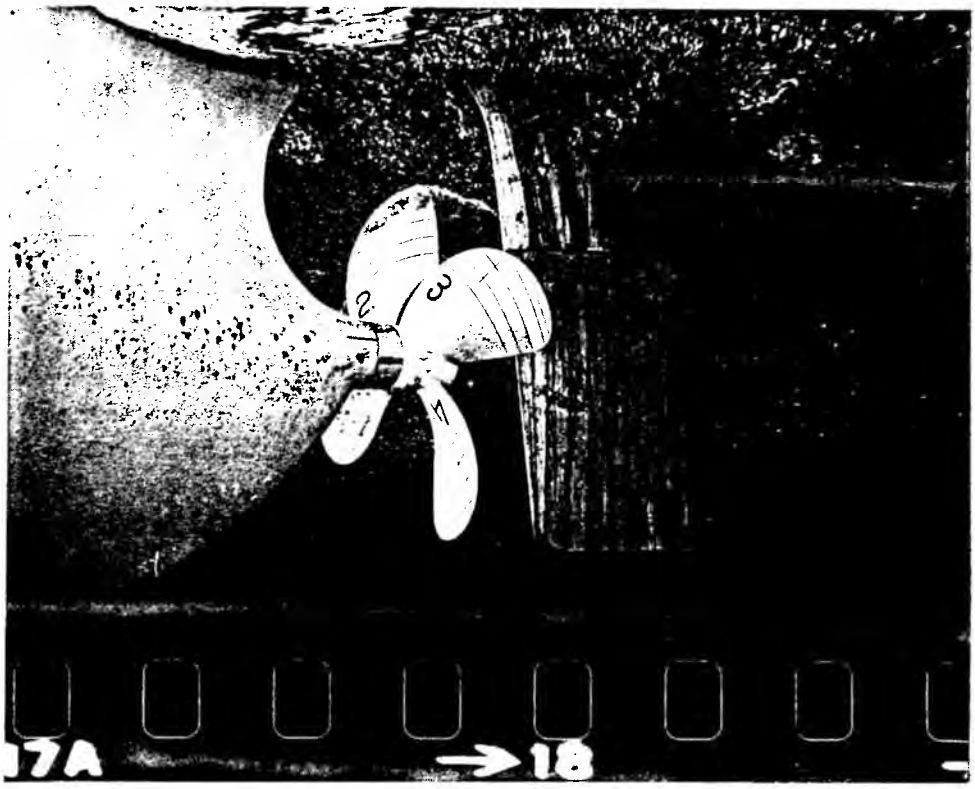
PROD. T " MAASSLUIS "

1:100





Ballast....36°



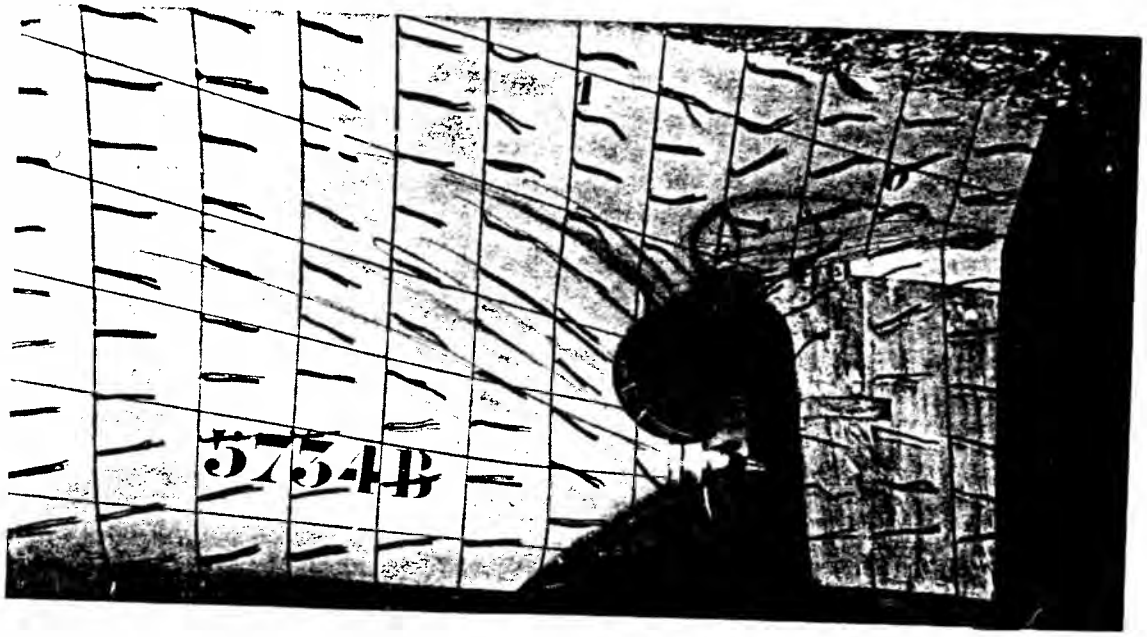
SHIP MODEL No. 5734B

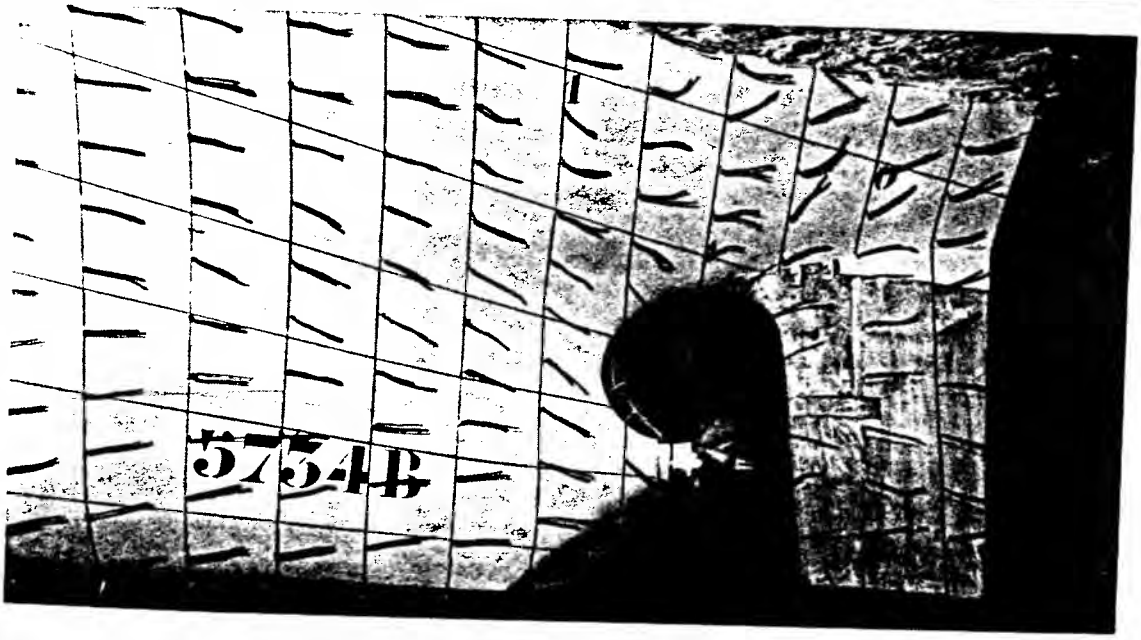
FLOW VISUALIZATION

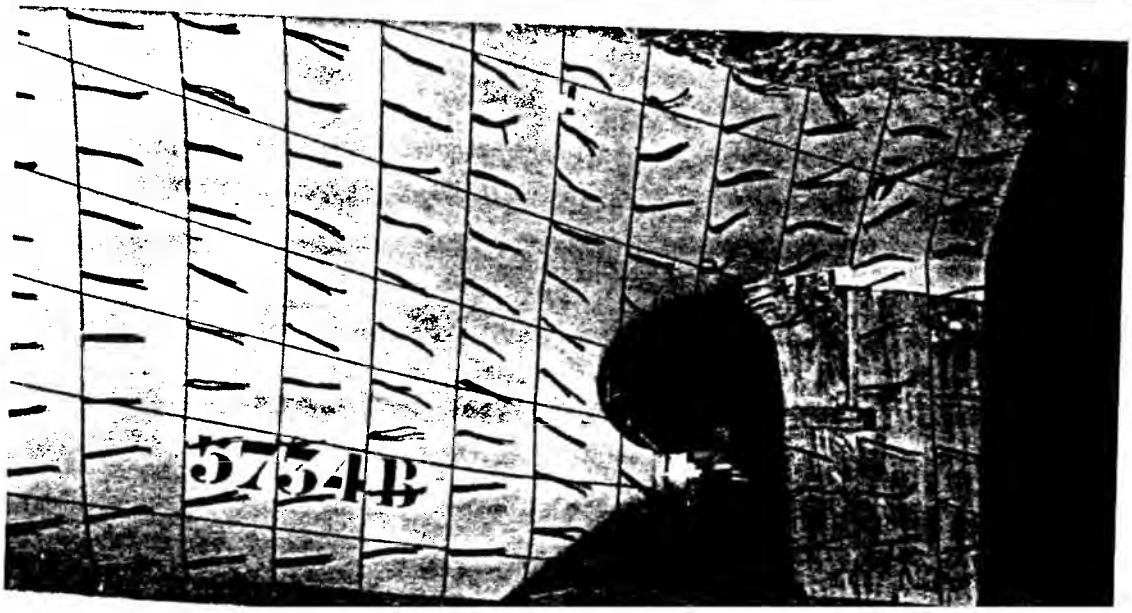
DRAFT 11.30 m

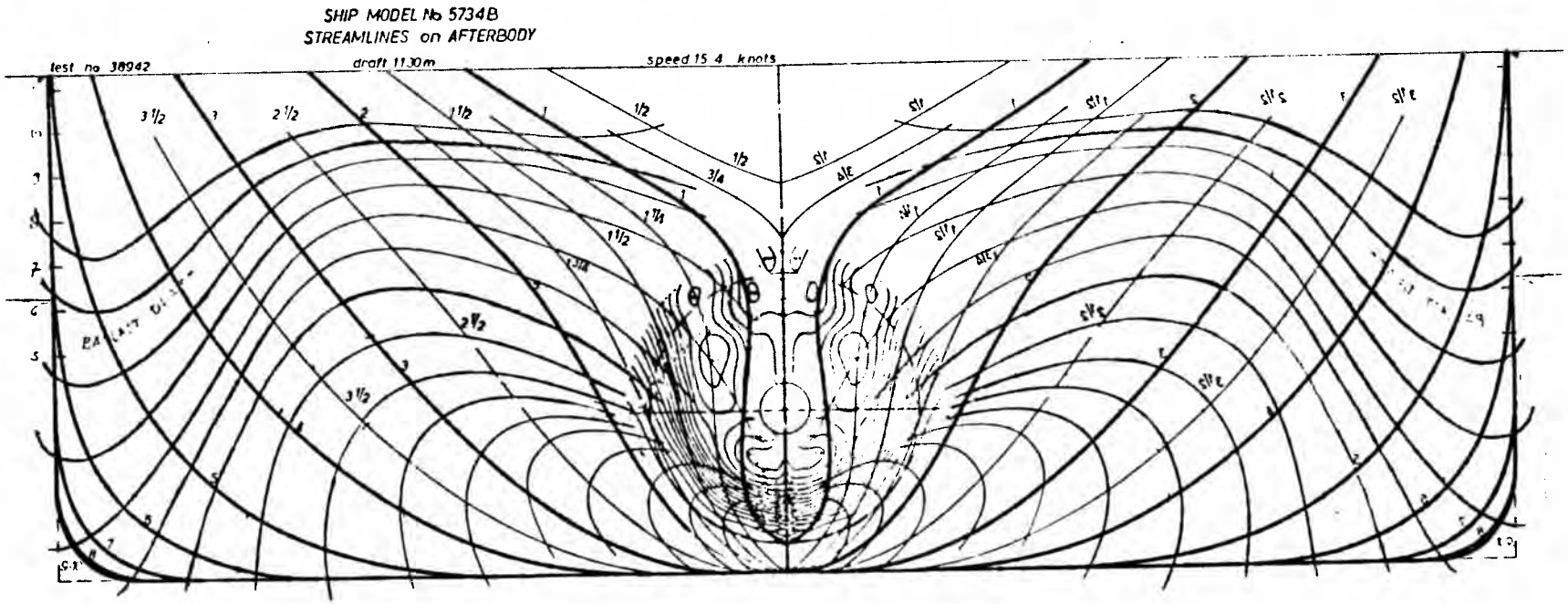
SPEED 15.42 KNOTS
WITH PROPELLER

TEST No. 38951







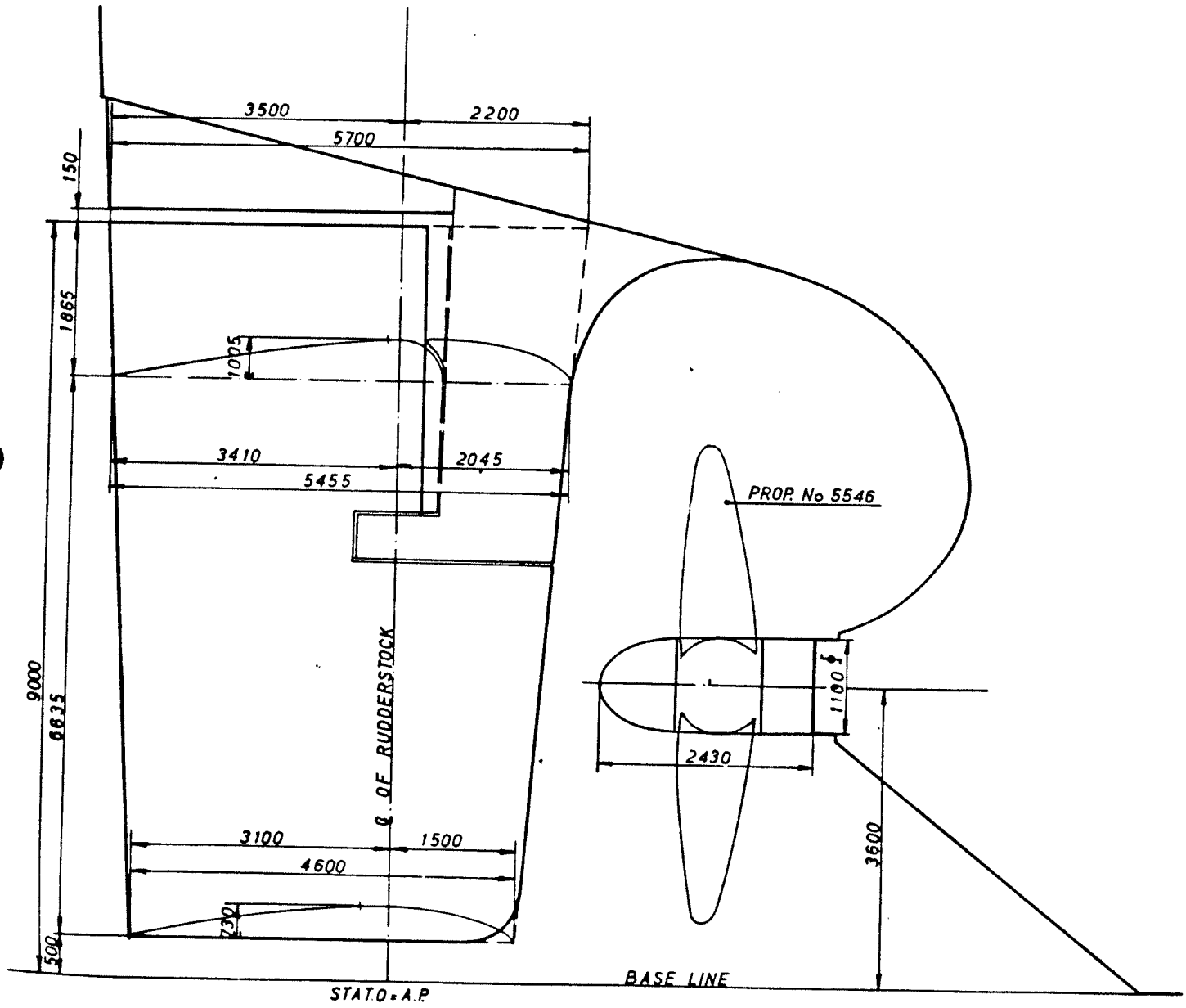


38000 DWT PRODUCT-TANKER "MAASSLUIS" V.D. GIESSEN D.N. 1981

Streamlines on afterbody.

GOOD VESSEL

SHIP MODEL No. 5734 B
RUDDER I AND SCREW APERTURE I
DIMENSIONS ARE GIVEN IN MM FOR SHIP



DRAWING No. M 5734 -66

HULL STRUCTURES

NEDLLOYD - BULK

B&W 6 cyl 67	By LIPS PROPELLER	Date
13100 BHP/123	Ø = 5800 4 BL. PITCH = 4161 - MEAN	21 - 3 - 83 TRIALS MAASSTAD

6-14

Subject **MAASSTAD - MAASSTROOM**

MAX IN SERVICE : 85% MCR

0,85 x 13100 = 11140 BHP

x 152½ g/HPH x 24 = 40.8 TON/DAY HFO

RPM - SPEED - SHP - TON/DAY - TON/1000 MILES

BALLAST

SERVICE 10% ON TRIALS + SHAFT GEN. à 500 KW

DRAFT FWD = 4,5 M

DRAFT AFT = 7.0 M

	RPM	KN
DEAD SLOW	35	4,9
SLOW	45	6,2
HALF	55	7.6
	CRITICAL 68/72	
	75	10,4
FULL	85	11,8
	86	11,9^{S}
	87	12,1
	88	12,2
	89	12,3^{S}
	90	12,5
	91	12,6
	92	12,8
	93	12,9
	94	13,0
	95	13,1
	96	13,2^{S}
	97	13,4
	98	13,5
	99	13,6
	100	13,7
	101	13,8
	102	13,9^{S}
	103	14,1
	104	14,2
	105	14,3
	106	14,4
	107	14,5
	108	14,6
	109	14,7
	110	14,8
	111	14,9
	112	15,0
	113	15,1
	114	15,2
	115	15,3
	116	15,4
	117	15,5
	118	15,6
	119	15,7
	120	15,8
	121	15,9
	122	16,0
	123	16,1
	124	16,2
	125	
	126	
	127	
	128	

PROPELLER SHP	SHAFT GEN	BHP	GR/HP.HR	TON/DAY	TON/1000
3555	+800 =	4355	162½	17,0	59½
3683	"	4483	162	17,4	60½
3813		4613	161½	17,8	61½
3946		4746	161	18,3	62½
4082		4882	160½	18,8	63½
4221		5021	160	19,3	64½
4363		5163	159½	19,8	65½
4509		5309	159	20,3	66½
4657		5457	158½	20,8	67½
4809		5609	158	21,3	68½
4964		5764	157½	21,8	69½
5123		5923	157	22,3	70½
5284		6084	157	22,9	71½
5450		6250	156½	23,5	72½
5618		6418	156½	24,1	73½
5790		6590	156	24,7	74½
5965		6765	156	25,3	75½
6114		6914	155½	25.9	76½
6327		7127	155½	26,5	78
6513		7313	155	27,1	79½
6703		7503	155	27,8	81
6896		7698	154½	28,5	82½
7093		7893	154½	29,2	84
7294		8094	154	29,9	85½
7498		8298	154	30,6	87
7706		8506	153½	31,3	88½
7917		8717	153½	32,0	90
8135		8935	153	32,8	91½
8354		9154	153	33,6	93
8578		9378	153	34,4	94½
8806		9606	152½	35,2	96
9038		9838	152½	36,0	97½
9273		10073	152½	36,8	99
9513		10313	152	37,6	100½
9757		10557	152	38,5	102
10005		10805	152	39,4	103½
10257		11057	152	40,3	105½
10514		11314	152½	41,3	107½
10774		11574	152½	42,3	109½
11039		11839	152½	43,3	111½
11309		12109			
11582		12382			
11860		12660			
12143		12943			

LOADED

SERVICE 10% ON TRIALS + SHAFT GEN à 500 KW

DRAFT = 11.3 M

	RPM	KN
	35	4.3
	45	5,6
	55	6,8
	CRITICAL 68/72	
	75	
	85	10,5
	86	10,6^{S}
	87	10,8
	88	10,9
	89	11,0
	90	11,1
	91	11,2^{S}
	92	11,4
	93	11,5
	94	11,6^{S}
	95	11,8
	96	11,9
	97	12,0^{S}
	98	12,2
	99	12,3
	100	12,4
	101	12,5^{S}
	102	12,7
	103	12,8
	104	12,9
	105	13,0
	106	13,1^{S}
	107	13,3
	108	13,4
	109	13,5
	110	13,6
	111	13,7
	112	13,8
	113	13,9
	114	14,0
	115	14,1^{S}
	116	14,3
	117	14,4
	118	14,5
	119	14,6
	120	14,7
	121	14,8
	122	14,9
	123	15,0
	124	15,1
	125	
	126	
	127	
	128	

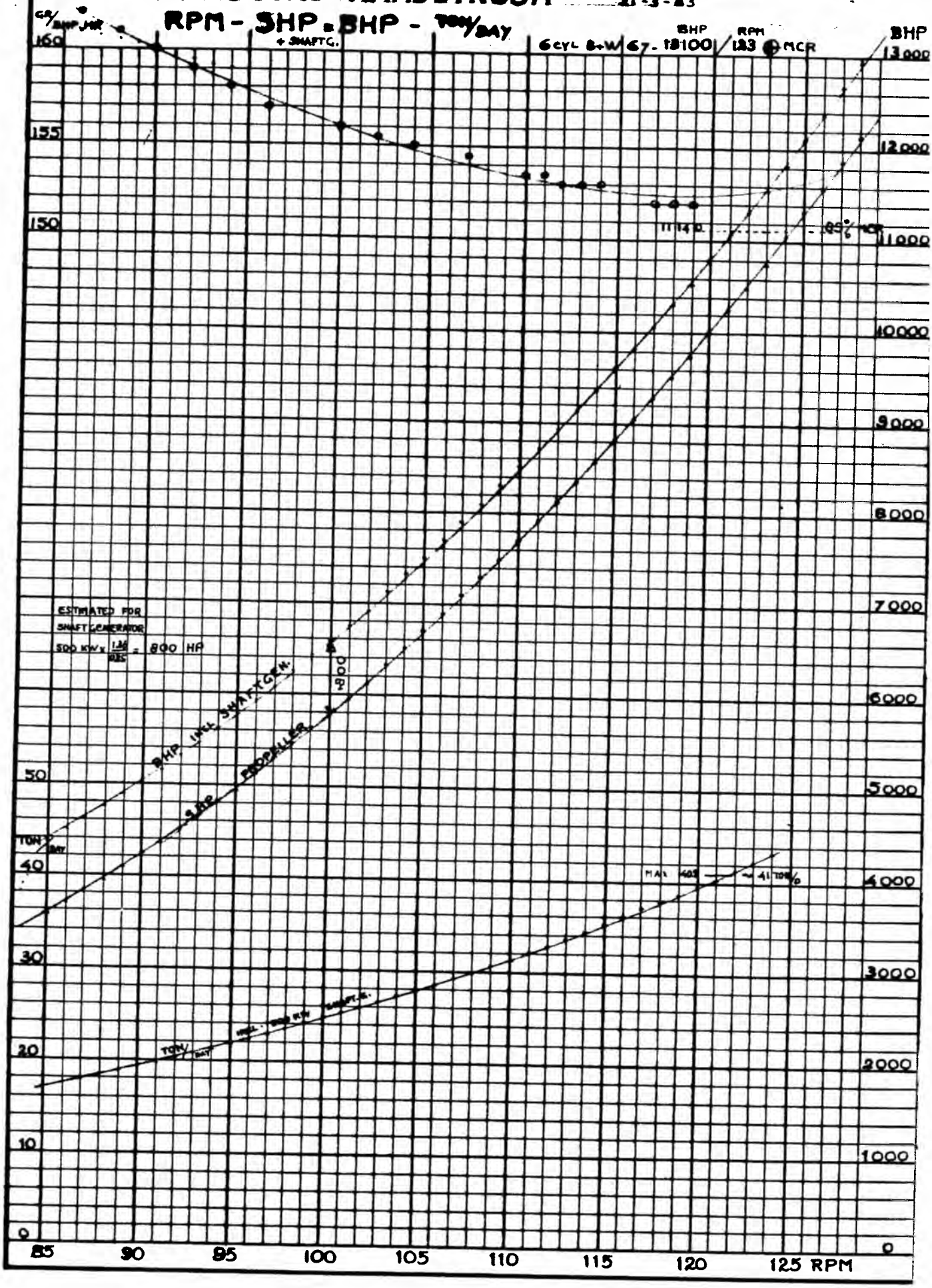
PROPELLER SHP	SHAFT GEN	BHP	GR/HP.HR	TON/DAY	TON/1000
3555	+800 =	4355	162½	17,0	67
3683	"	4483	162	17,4	68
3813		4613	161½	17,8	69
3946		4746	161	18,3	70
4082		4882	160½	18,8	71
4221		5021	160	19,3	72
4363		5163	159½	19,8	73
4509		5309	159	20,3	74
4657		5457	158½	20,8	75
4809		5609	158	21,3	76
4964		5764	157½	21,8	77
5123		5923	157	27,3	78
5284		6084	157	22,9	79
5450		6250	156½	23,5	80
5618		6418	156½	24,1	81
5790		6590	156	24,7	82
5965		6765	156	25,3	83
6114		6914	155½	25.9	84
6327		7127	155½	26,5	85½
6513		7313	155	27,1	87
6703		7503	155	27,8	88½
6896		7698	154½	28,5	90
7093		7893	154½	29,2	91½
7294		8094	154	29,9	93
7498		8298	154	30,6	94½
7706		8506	153½	31,3	96
7917		8717	153½	32,0	97½
8135		8935	153	32,8	99
8354		9154	153	33,6	100½
8578		9378	153	34,4	102
8806		9606	152½	35,2	103½
9038		9838	152½	36,0	105
9273		10073	152½	36,8	106½
9513		10313	152	37,6	108
9757		10557	152	38,5	109½
10005		10805	152	39,4	111½
10257		11057	152	40,3	113½
10514		11314	152½	41,3	115½
10774		11574	152½	42,3	117½
11039		11839	152½	43,3	119½
11309		12109			
11582		12382			
11860		12660			
12143		12943			

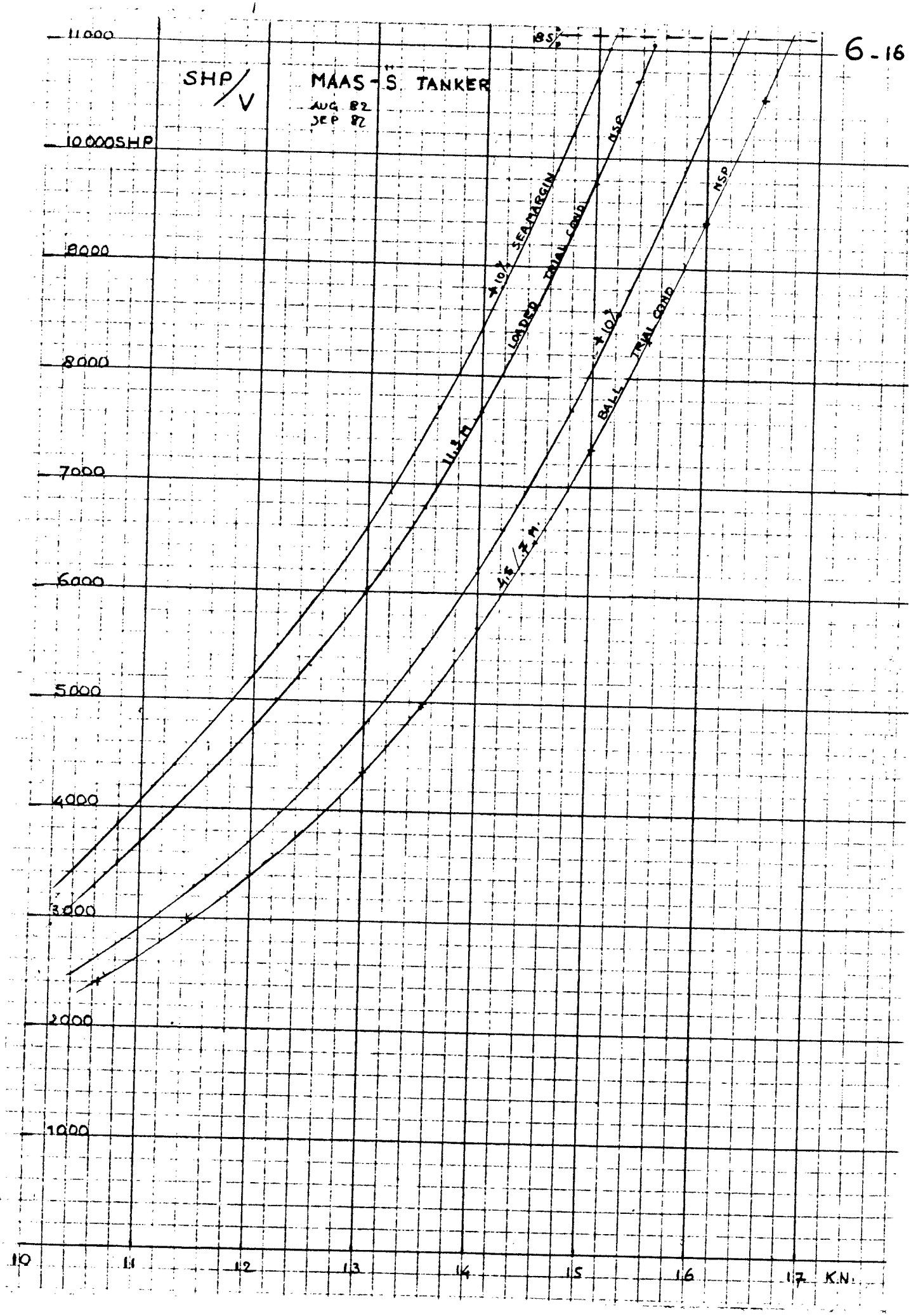
GV 21.3.83

Form 3806 (11/80)

"MAASSTAD - MAASSTROOM" 21-3-'83

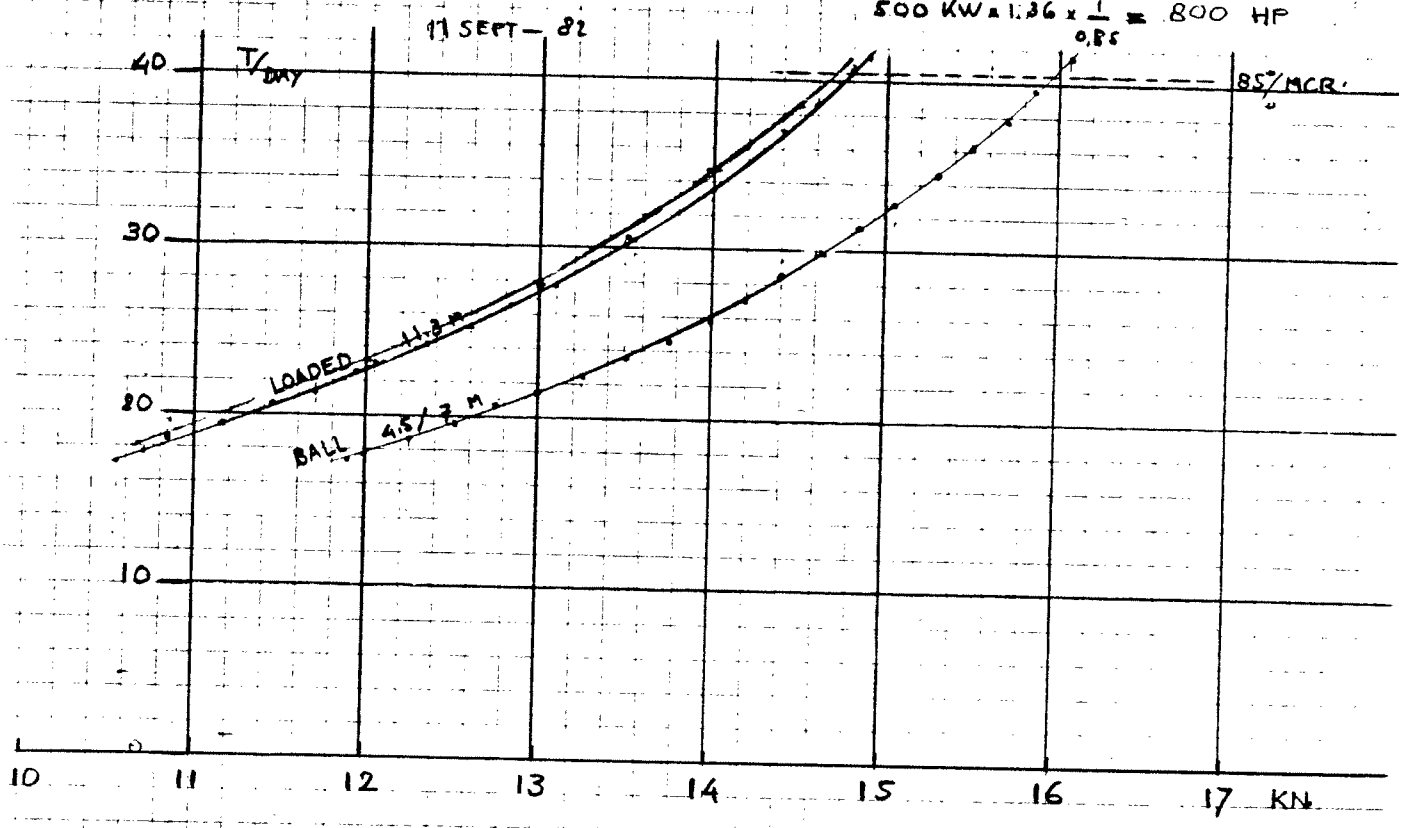
RPM - SHP - BHP - TON/DAY




SHP/V
MAAS - S. TANKER
AUG 82
SEP 82
11000
10000SHP
9000
8000
7000
6000
5000
4000
3000
2000
1000
10
11
12
13
14
15
16
17 K.N.
18%
+10% SEAMARGIN
LOADED TRIAL COND.
11.2 m
OSP
+10%
BALL. TRIAL COND.
4.5/7 m.
MSP

FUEL CONS 10% SEAMARGIN IN SERVICE + SHAFTGENERATOR

500 KW x 1.36 x $\frac{1}{0.85}$ = 800 HP



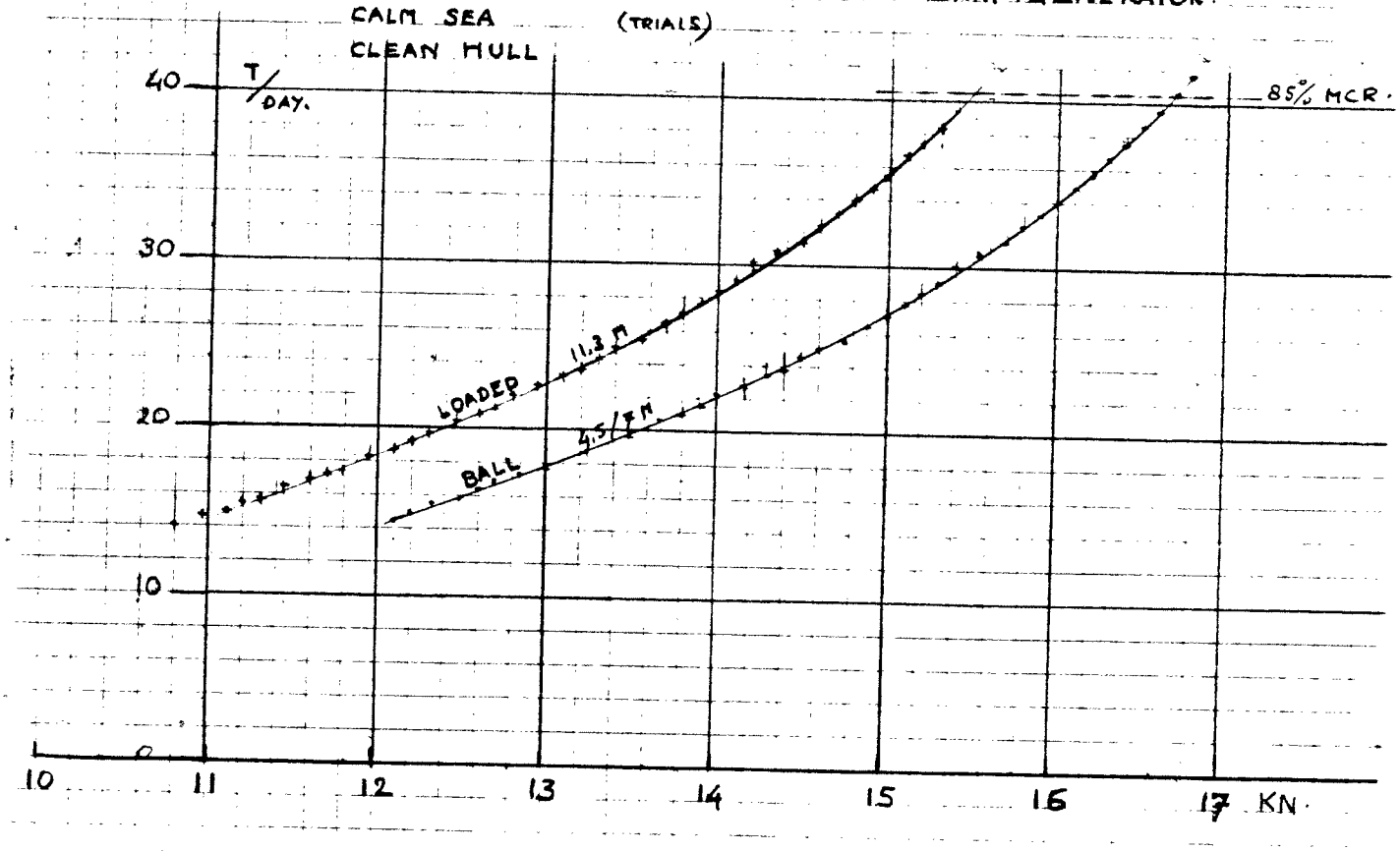
MAAS "S" TANKER

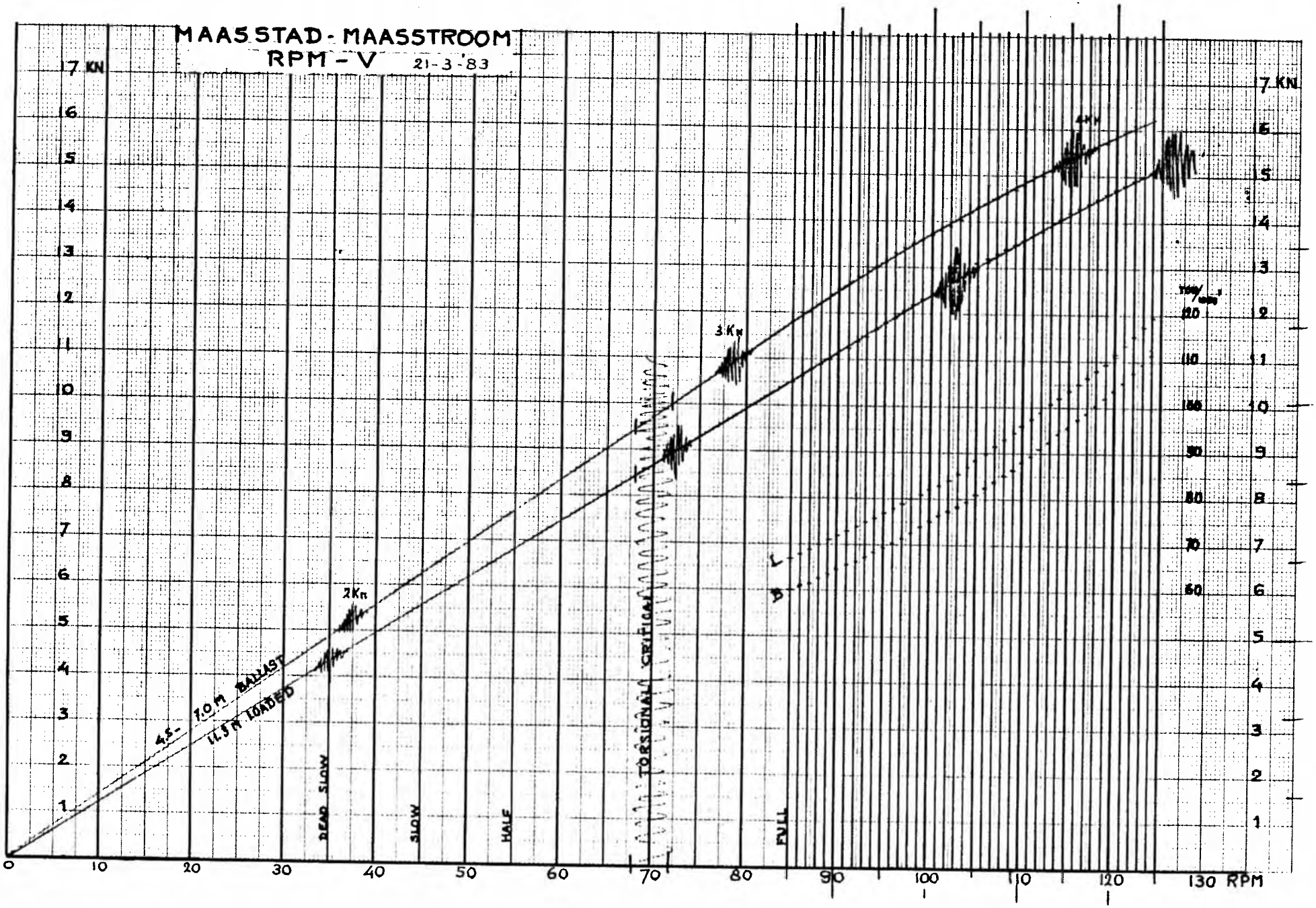
AUG 82 V/O G D N.

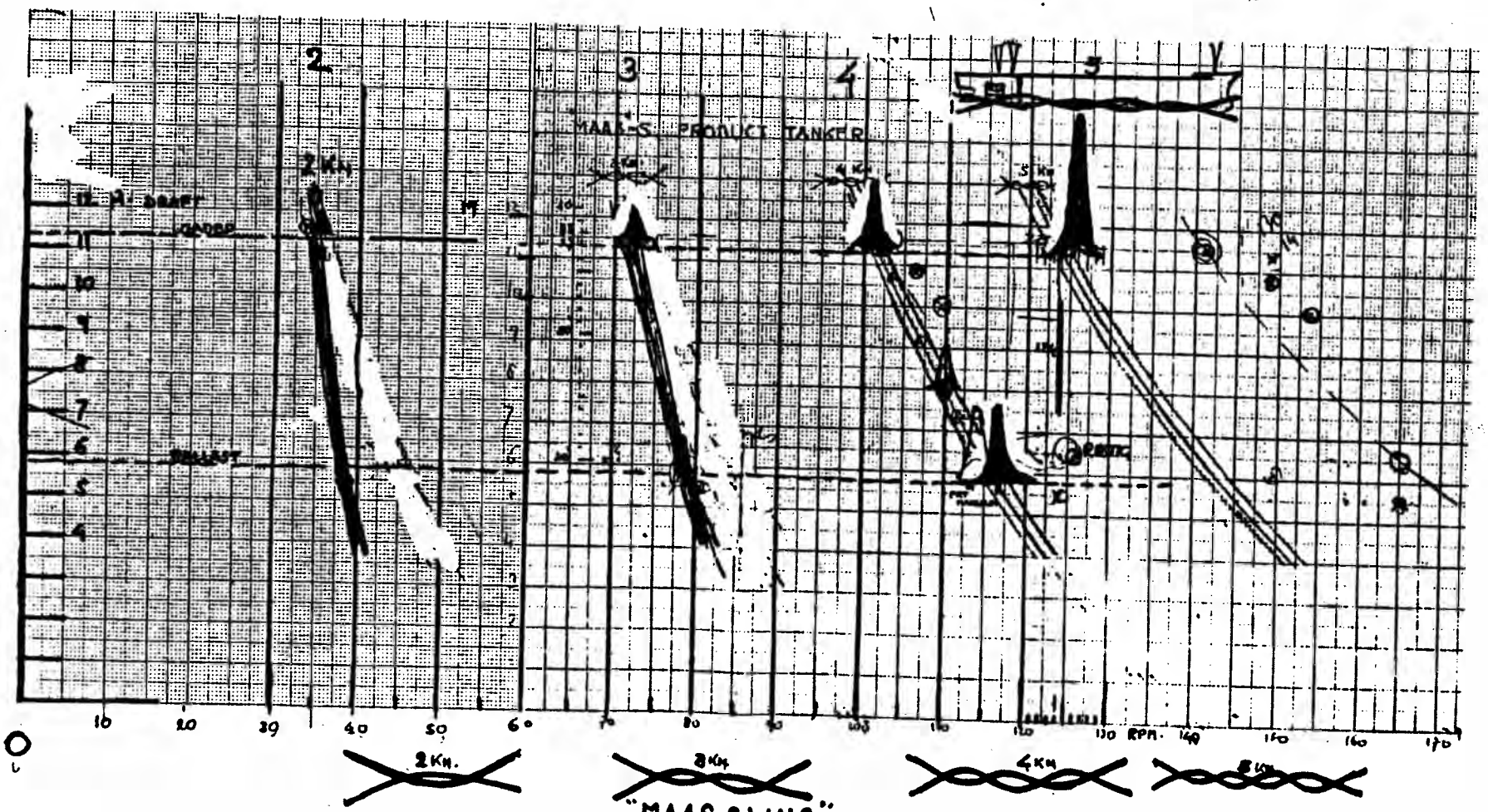
FUEL CONS. OPTIMAL CONDITIONS – WITHOUT SHAFTGENERATOR.

CALM SEA (TRIALS)

CLEAN HULL




MAASSTAD - MAASSTROOM
RPM - V
21-3-83
17 KN
4.5 - 7.0 M BALLAST
11.5 M LOADED
2 Kn
3 Kn
4 Kn
DEAD SLOW
SLOW
HALF
FULL
TORSIONAL CRITICAL
0
10
20
30
40
50
60
70
80
90
100
110
120
130 RPM

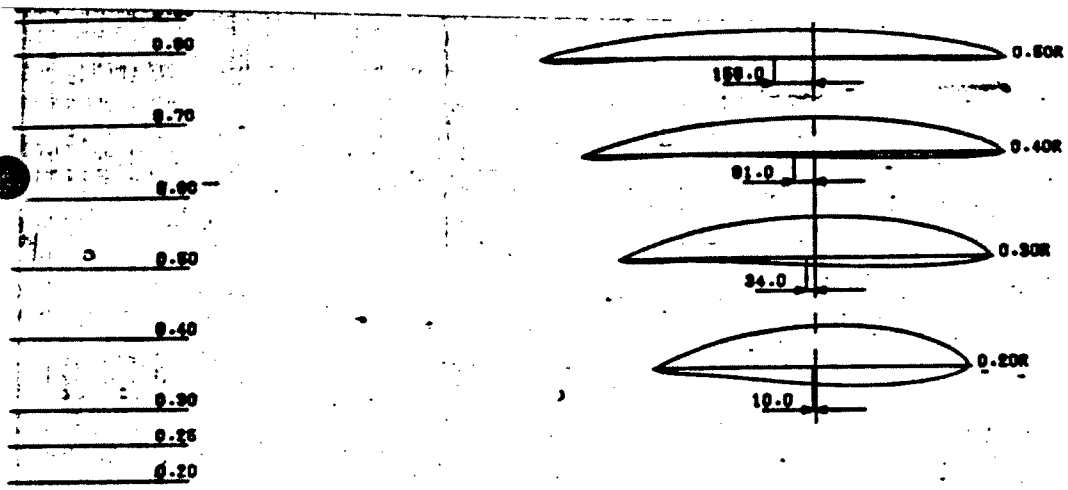


"MAAS SLUIS" – RESONANCE VERTIC. VIBRATION

11-11-82

EXCITATION BY FREE LONGIT MOMENT SEC. ORDER
OF 6 CYL DIESELENGINE

2x RPM ↓ ↑ NOT HAVING LANCASTER BALANCERS



PROPELLER DATA

DIAMETER	5800 MM
PITCH.	4161 MM
NUMBER OF BLADES	4
DIRECTION OF ROTATION	RIGHT-HANDED
EXPANDED BLADE AREA RATIO	0.627
MATERIAL	CUNIAL
MINIMUM TENSILE STRENGTH	647 N/MM²
MINIMUM ELONGATION	20 %
MASS	≈ 16000 KG
MASS MOM. OF INERTIA PROP. IN AIR	≈ 23400 KGM²(J-ENTRAINED WATER=6556 kgm²)
ENGINE OUTPUT	9635 KW
PROPELLER REVOLUTIONS	120.00 RPM
E FACTOR 0.25	1.0000
E FACTOR 0.6	1.2500
CLASSIFICATION	LLOYDS REG. OF SHIPPING.

PITCH SELECTED FOR 8190 KW 124.50 RPM . TRIAL CONDITION.

ALL BLADE FINISHING TOLERANCES IN ACCORDANCE WITH I.S.O. CLASS I.

Schroef 7759 Maasstroom
7760 Maasstad.

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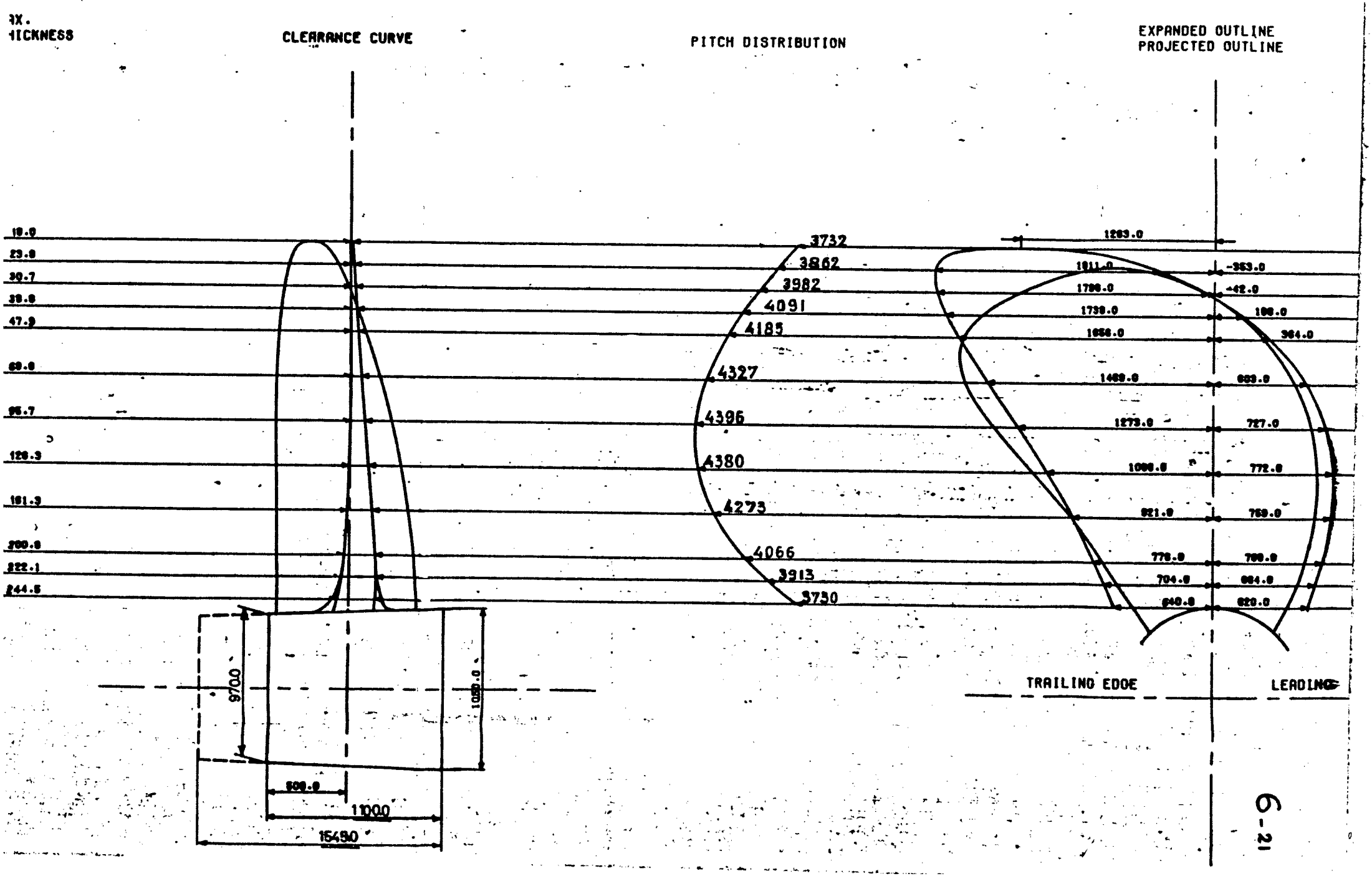
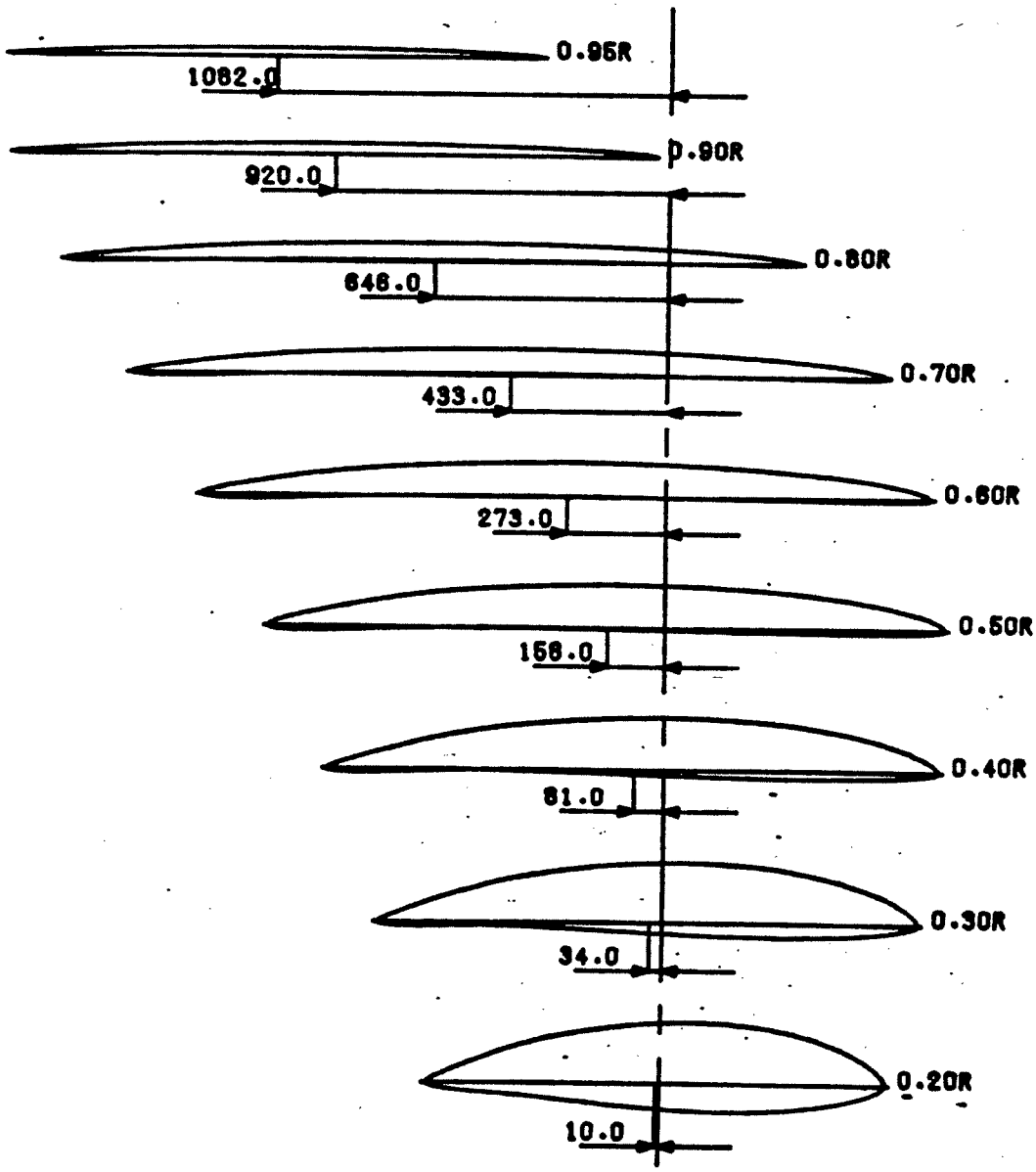
LIPS PROPELLER WORKS,
DRUNEN, NETHERLANDS

F.P.PROPELLER FOR NB 928		82/12/09
VD GIESSEN DE NOORD F7759/60		1/20
LIPS B.V. DRUNEN-HOLLAND	E7759-01	A 1


CLEARANCE CURVE
PITCH DISTRIBUTION
EXPANDED OUTLINE
PROJECTED OUTLINE
19.0
23.0
30.7
39.8
47.9
69.0
95.7
120.3
161.3
200.0
222.1
244.5
3732
3862
3982
4091
4185
4327
4396
4380
4273
4066
3913
3730
1263.0
1811.0
1796.0
1739.0
1656.0
1469.0
1273.0
1080.0
921.0
776.0
704.0
640.0
-363.0
-42.0
100.0
364.0
603.0
727.0
772.0
769.0
700.0
684.0
620.0
TRAILING EDGE
LEADING
9700
1050.0
500.0
11000
15450

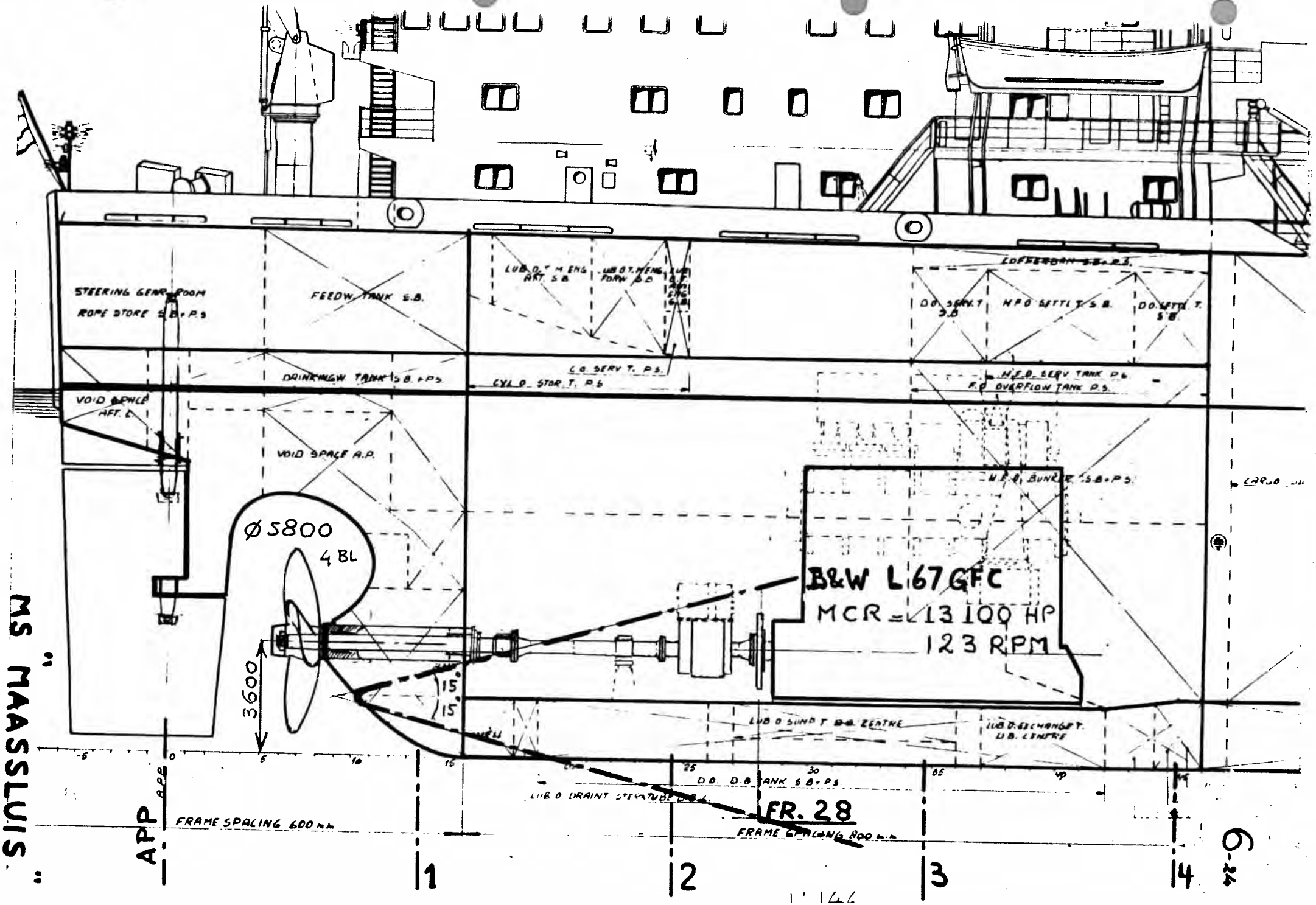
BLADE SECTIONS



PROPELLER DATA

DIAMETER 5800 MM

Ø 5800 MS. MAASSLUIS, 6 L 67 GFC
13100 HP/123 RPM


"MS MAASSLUIS"
STEERING GEAR ROOM
ROPE STORE S.B. + P.S.
FEEDW. TANK S.B.
DRINKING W. TANK S.B. + P.S.
VOID SPACE AFT.
VOID SPACE A.P.
Ø 5800
4 BL
3600
15°
15°
B&W L67GFC
MCR = 13100 HP
123 RPM
FR. 28
FRAME SPACING 600 mm
APP
1
2
3
4
6-24

Ø 6300

1A

4 L 80 GBE
12240 HP / 106 RPM.

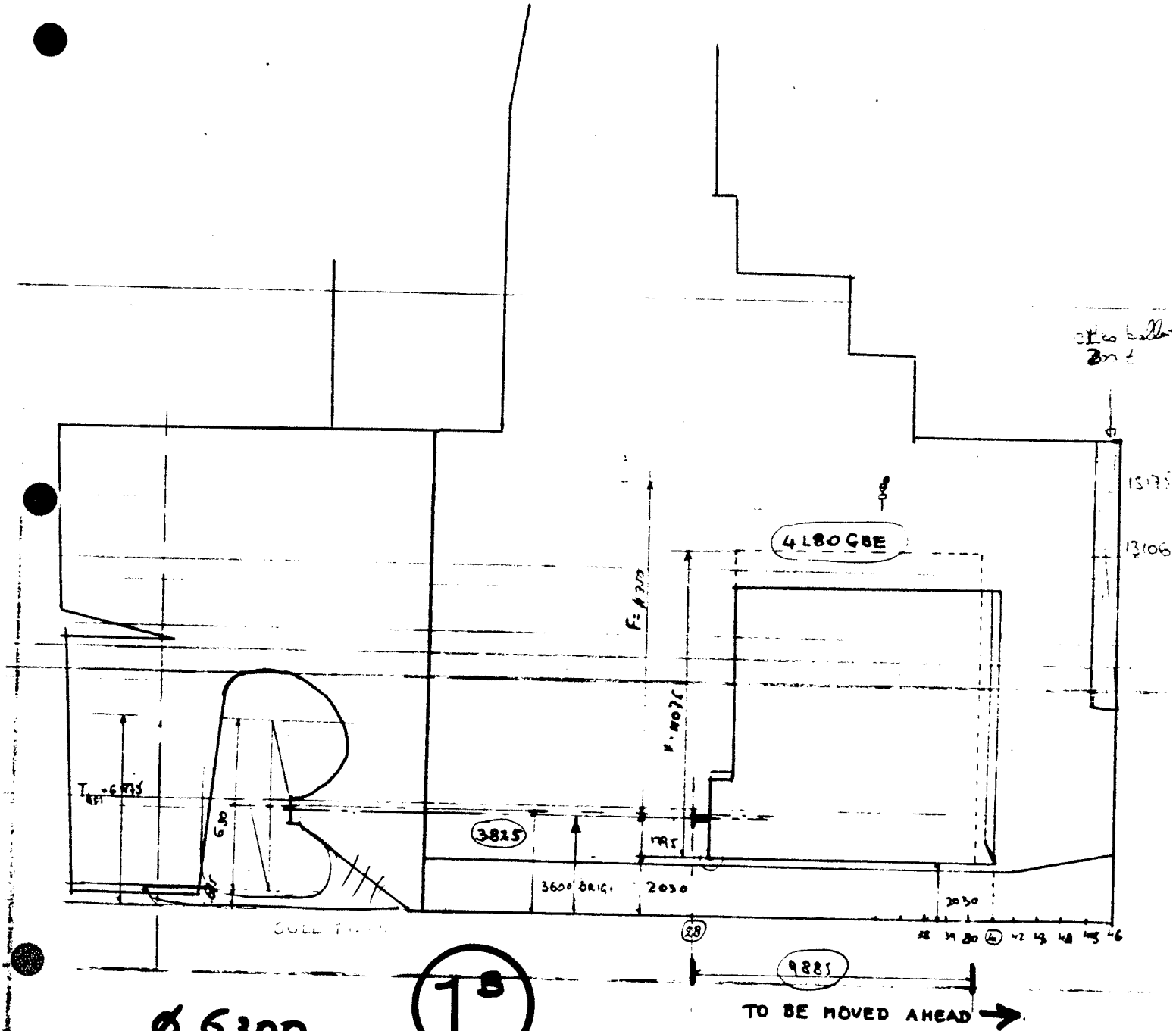
TYPE	: 4 L80GBE
MCR	: 12240 BHP × 106 RPM

D_P : 6.30 m

necessary T_{APP} : 6.978 m (low 6.70 m)

T_{APP} without Extra ballast : 6.81 m.

for indompelling Extra ballast : 200 t about.



Ø 6300

1B

4 L 80 GBE
12240 HP / 106 RPM.

Φ 6500

2

5 RTA 68
13250 HP / 105 RPM.

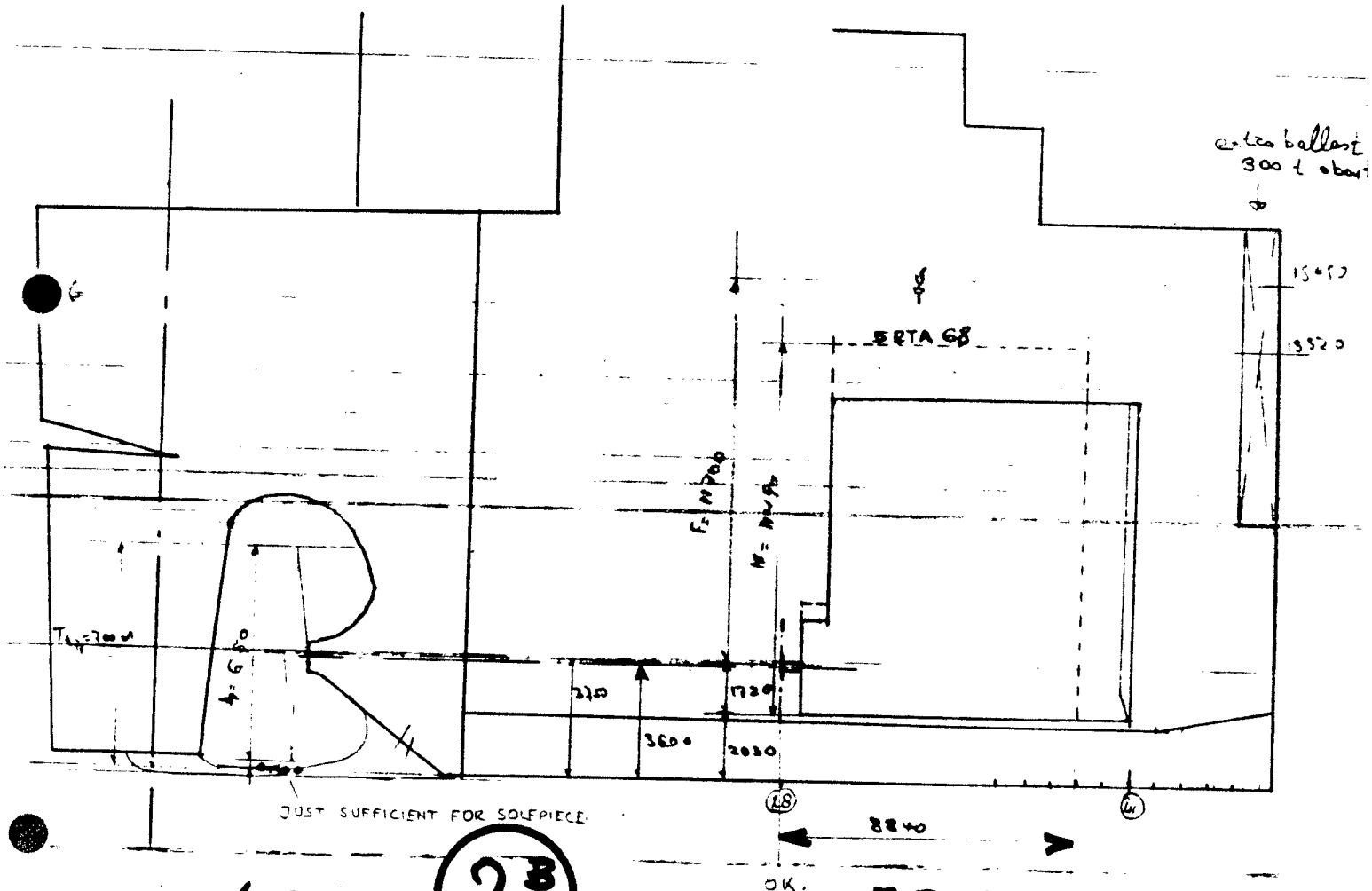
Type: 5 RTA 68
MCR : 13250 BHP × 105 RPM
Working MCR : 12500 × 100

Dp : 6.50 m.

necessary $T_{APP_{ballast}}$: 7.00 m. (to = 6.73 m)

T_{APP} without extra ballast : 6.78 m

For improvement Extra ballast : ≈300 t about



Ø 6500

2B

5 RTA 68
13250 HP / 105 RPM.

Ø 6700

3 A

5 L 70 MCE
12250 HP/95 RPM.

Type :	SL70 MCE
MCR :	12250 BHP x 95 RPM.
Working MCR :	Maximum

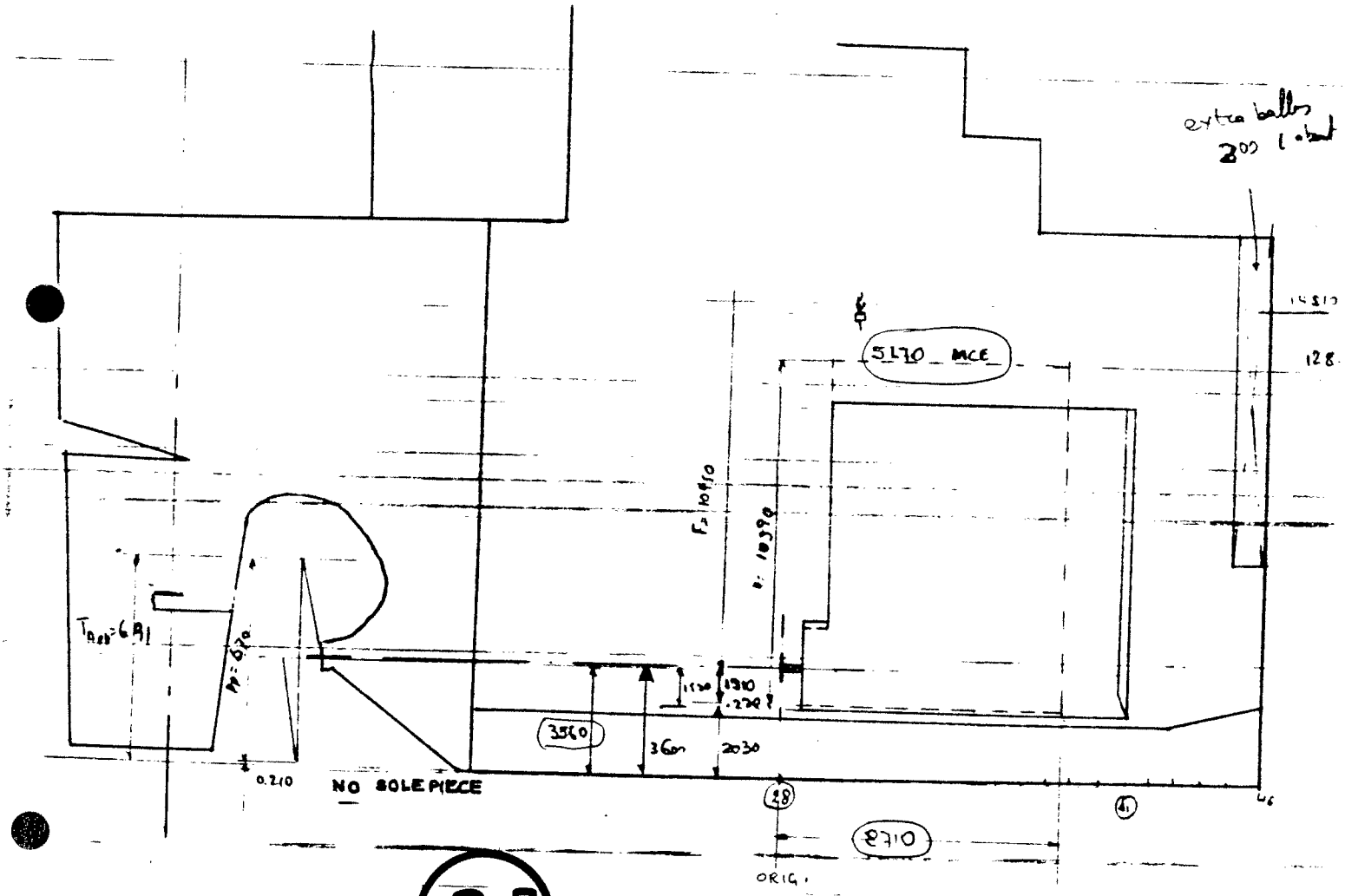
Dp : 6.70 m

Necessary T_{App} bal : 6.91 m (i.e. 6.70 m)

With extra ballast T_{App} : 6.82 m

For adequate extra ballast : 200 t about.

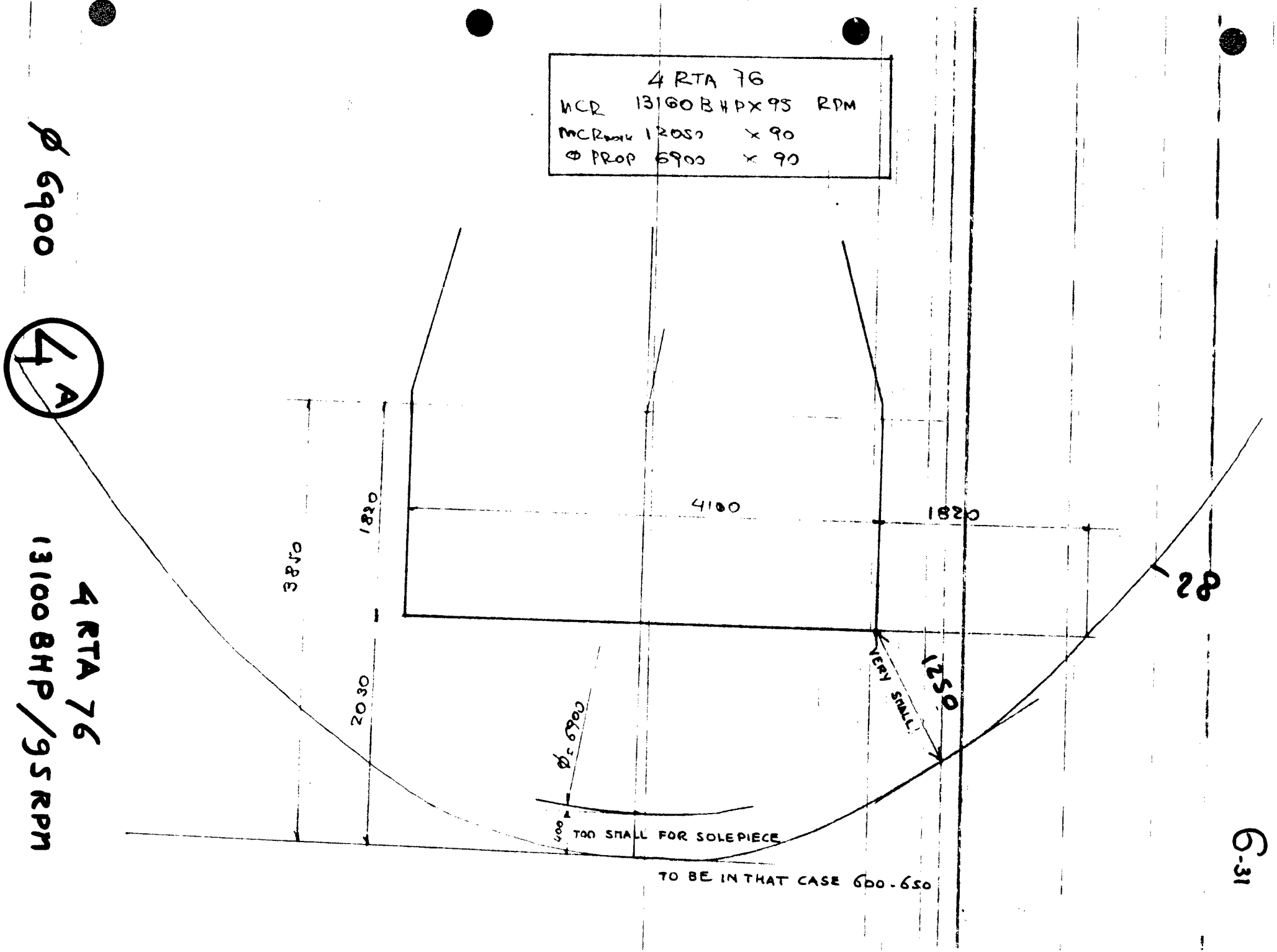
With 350 t between 45-46 T_{ap} : 7.16 m



Ø 6700

3B

SL 70 MCE
12250 HP/95 RPM.


4 RTA 76
MCR 13160 BHP x 95 RPM
MCR 12050 x 90
Φ PROP 6900 x 90
Φ 6900
4 A
4 RTA 76
13100 BHP / 95 RPM
3850
1820
2030
4100
1820
1250
VERY SMALL
Φ 6900
400
TOO SMALL FOR SOLEPIECE
TO BE IN THAT CASE 600 - 650
28

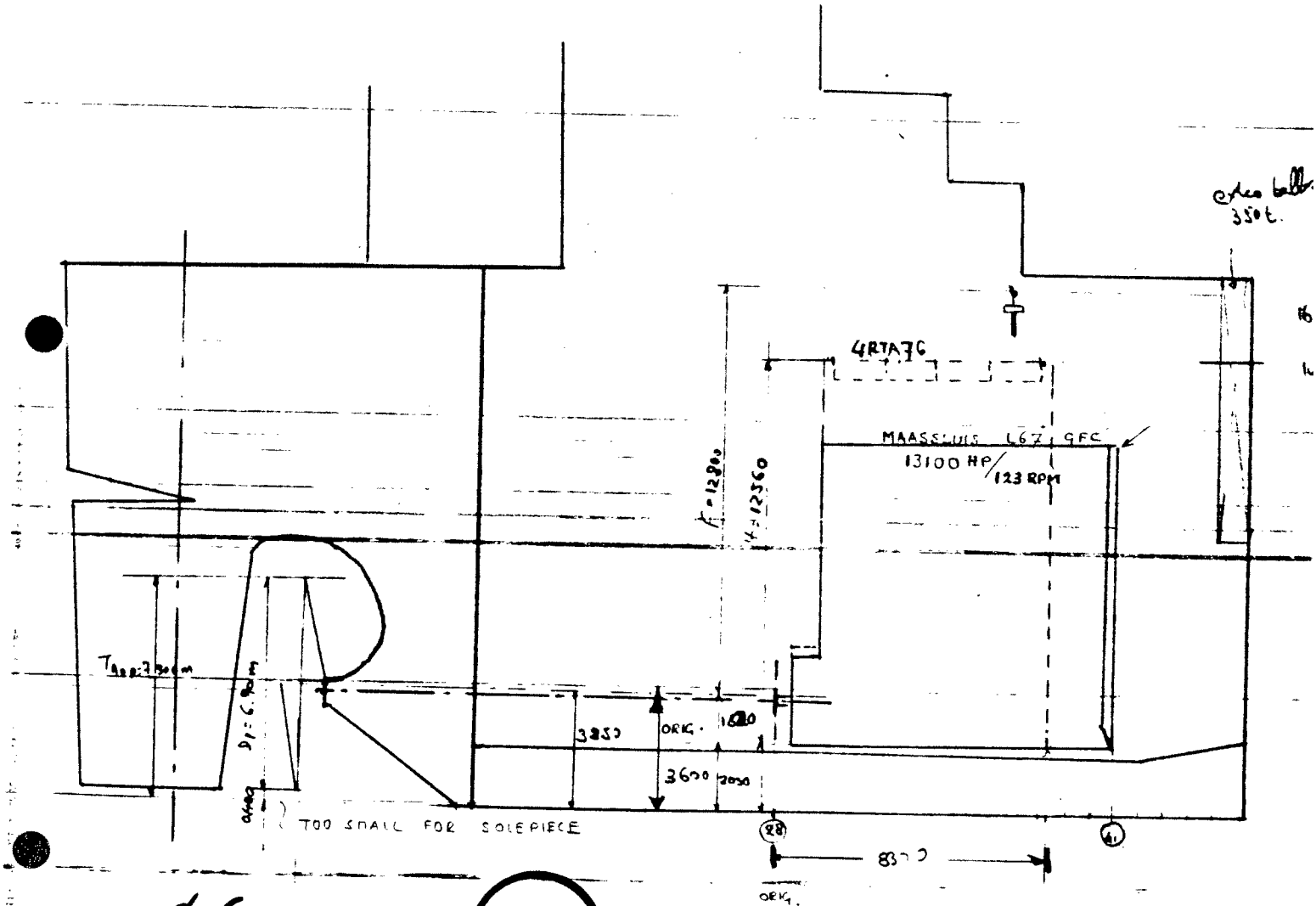
TYPE : 4RTA76
MCR : 13160 BHP × 95 RPM
Working MCR : 12050 × 90

Dp : 6.90m

necessary $T_{App_{ball}}$: 7.30m (> 6.90m)

T_{App} without extra ball : 6.82m

in ER for the indomitable extra bal 350 t. about.
T : 7.86 m
t : 2.59 m > 2.40m → more ballast ahead of ⌀ 4pr



⌀ 6900

4B

4 RTA 76
13100 BHP/95 RPM.

Ø 7300

5 A

4 L80 MCE
12800 HP / 83 RPM.

Type	:	4 L80MCE
MCR	:	12800 BHP × 83 RPM
Working MCR	:	11900 × 79

Dp : 7.30 m

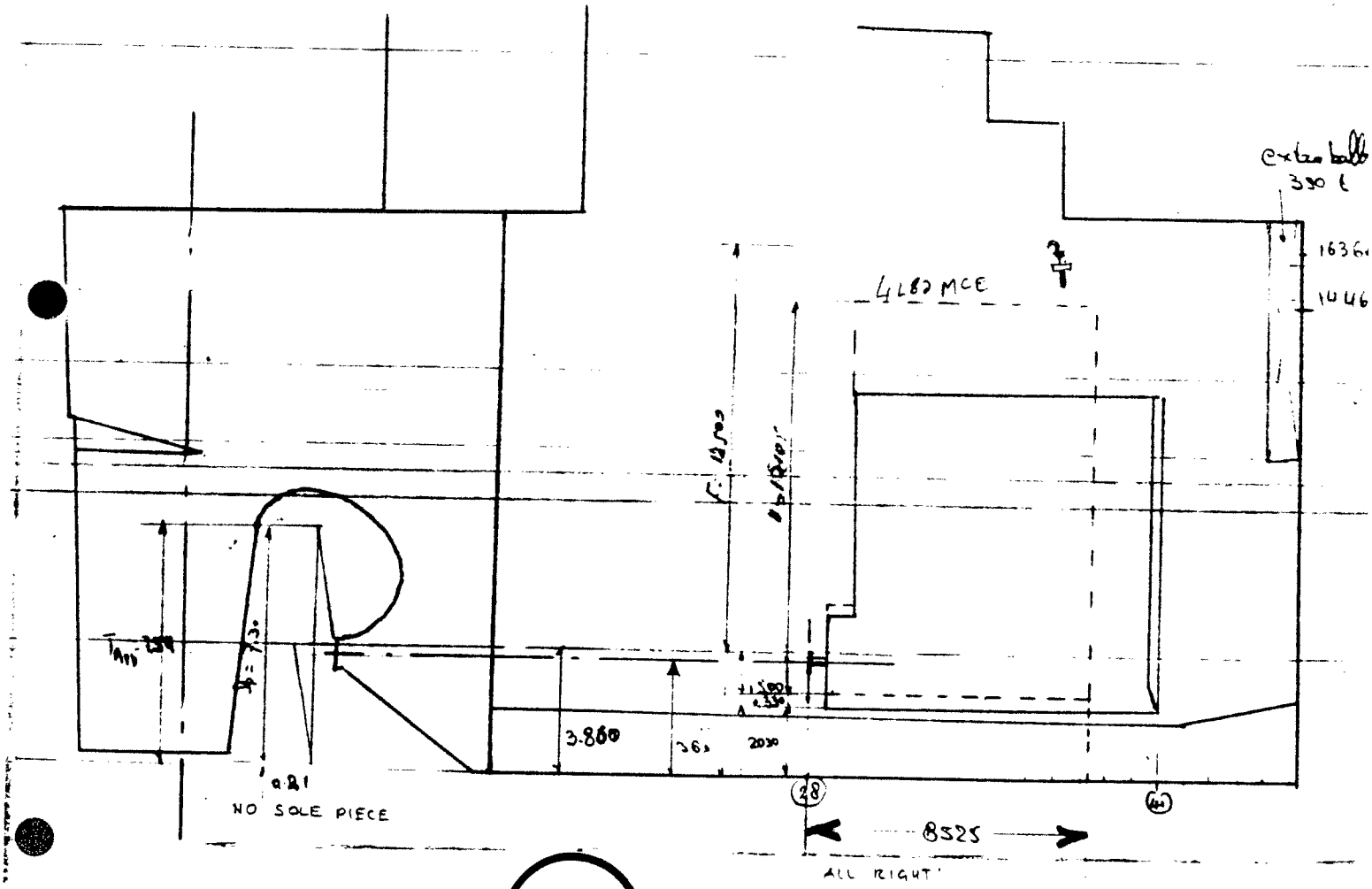
necessary T_{APP} : 7.51 m (> 6.90 m)

without ballast extra T_{APP} : 6.94 m

in ER extra ballast for trim down : 350 t about

T_{FP} : 7.27 m

t : 2.75 m > 2.40 m ⇒ more ballast in ER and ahead of ½ Lpp



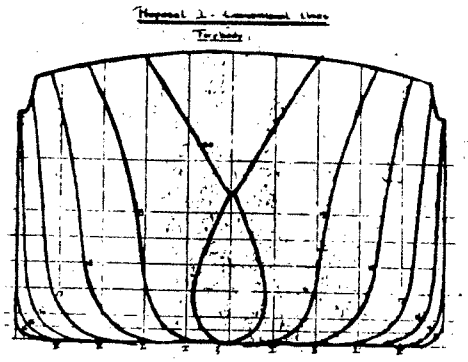
Ø7300

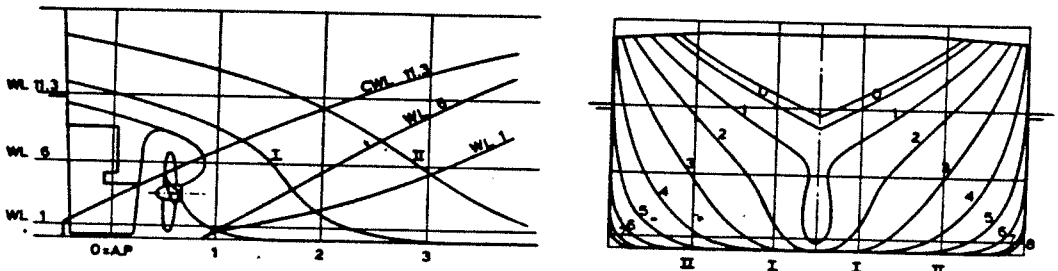
5B

4L80 MCE
12800 HP/83 RPM.

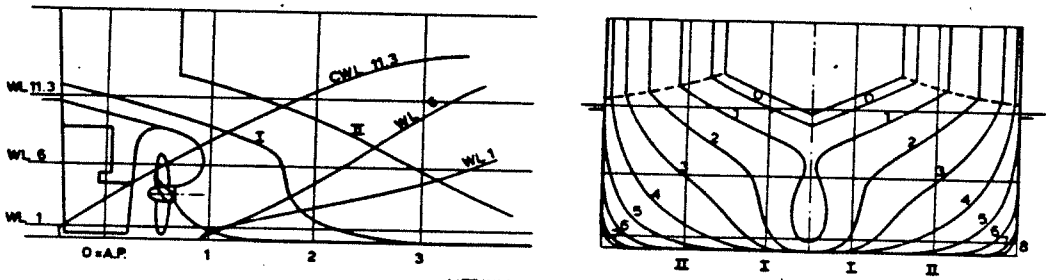
PROPOSED AFTERBODY SHAPES FOR S.S. PARCEL TANKER

L_{PP}	= 164.00	m	L_{pp}/B	= 5.09	
B mld	= 32.24	m	B / T	= 2.85	
T design	= 11.30	m	C_B	= 0.78	
V mld	= 46437	m^3	L.C.B.	= + 2.8 %	

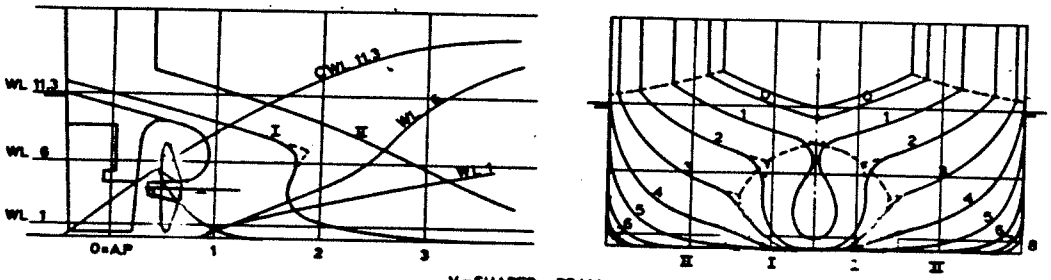




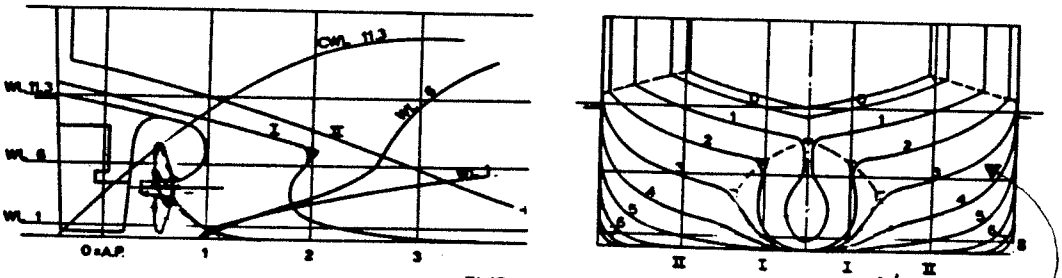
CONVENTIONAL SHAPE



MEDIUM SHAPE



V-SHAPED PRAM



FLAT PRAM

DECREASE OF TOWING HORSE POWER P_E.

DECREASE OF RPM.

INCREASE OF PROPELLER Ø

DISPLACEMENT.

PRAM WILL FLOAT DEEPER IN BALLAST

LARGER PROPELLER WILL REMAIN IMMERSED.

N.S.M.B. ref. ORDER No. P 03724
DRAWING No. P03724 -1

TRY TO REDUCE FUEL CONSUMPTION → PRAM + LARGE SLOW

6.36

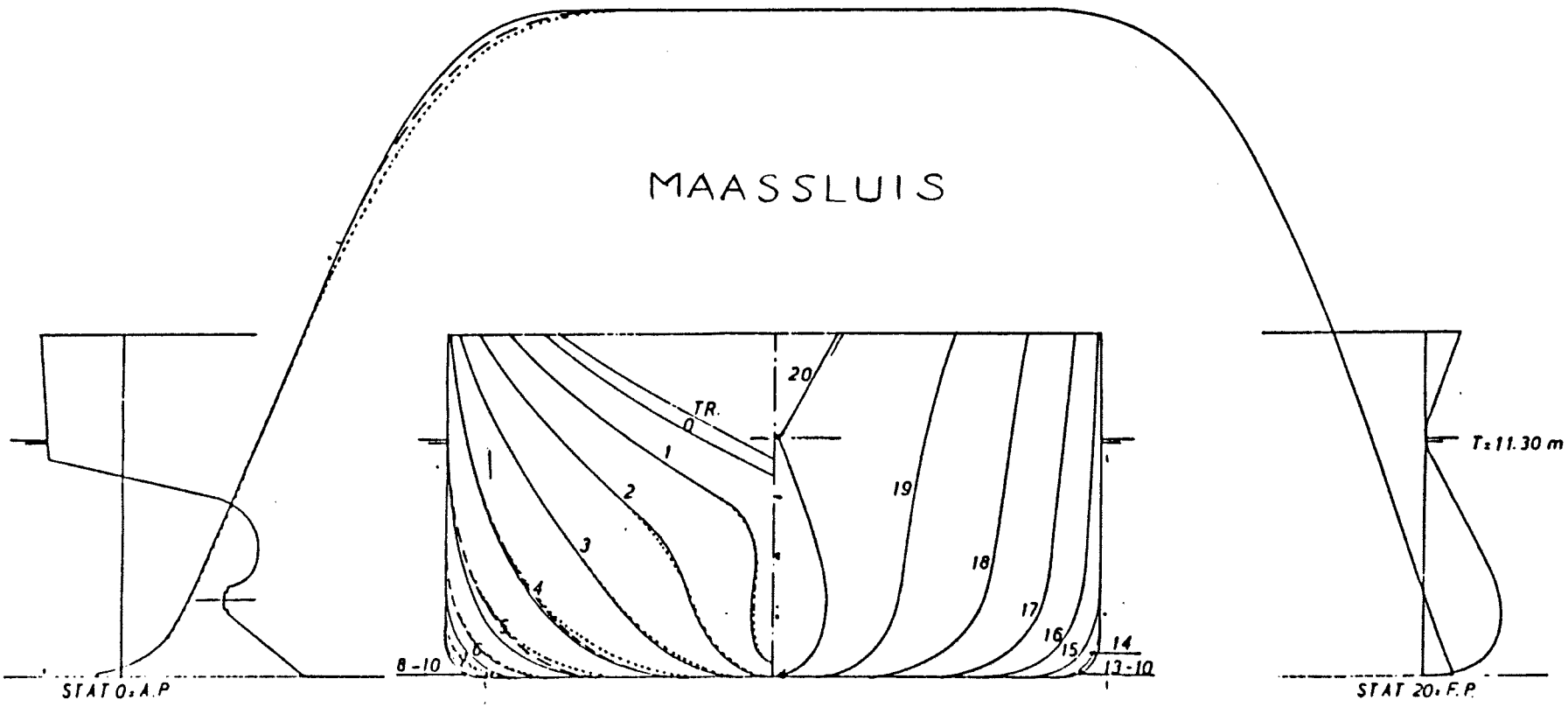
P_E	SHAFT HP: 11000 SHP 15.5 Kn SEA SPEED η_P PROP EFF.	η_{HULL} HULL EFF.	η_{RR}. REL. ROT	η_{TOT}. $\eta_P \cdot \eta_{HULL} \cdot \eta_{RR}$	P_D	FUEL CONSUMP
100%	WAKE $\psi = 0{,}42$ RPM = 123 PROPELLER $\phi = 5{,}90$ m. $\eta_{PROP} = 0{,}53$	THRUST DEDUCT. $t = 0{,}22$ $\frac{1-t}{1-\psi}$ $\frac{0{,}78}{0{,}58} = 1{,}34$	1.03	0,73	$\frac{100}{0{,}73} = 137$	100%
98%						
95%						
92%	WAKE $\psi = 0.30$ RPM = 90 $\phi = 7{,}10$ m. $\eta_{PROP} = 0{,}62^5$	$t = 0{,}22$ $\frac{1-t}{1-\psi}$ $\frac{0{,}78}{0{,}70} = 1{,}11$	1,03.	$0{,}71^5$	$\frac{92}{0{,}715} = 129$.	94%.
ESTIM BY NSP + NEDLL	CALCUL. BY LIPS	ESTIM BY NSP.				MIGHT BE EXPECTED. 26-3-80.

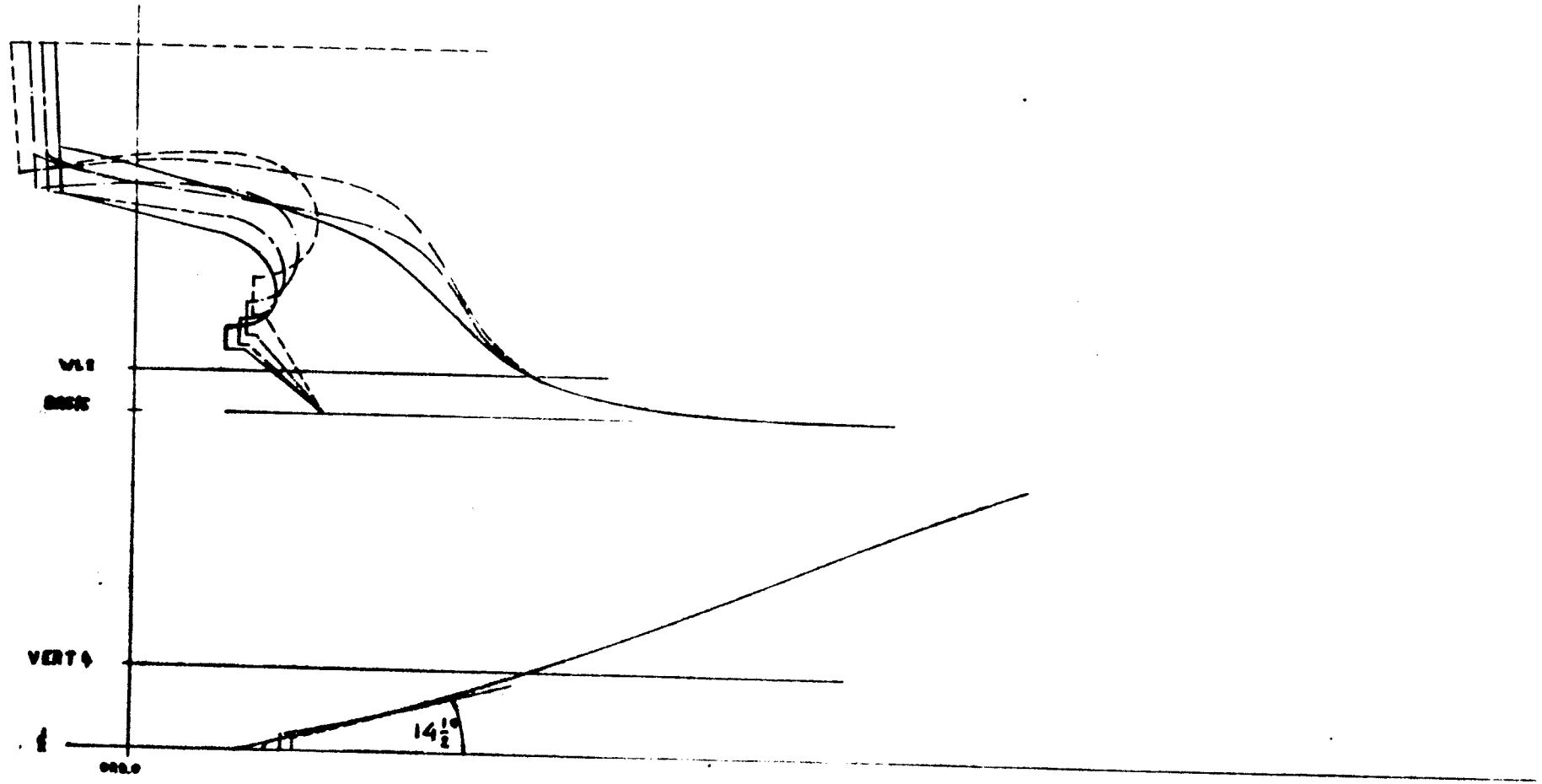
Figuur 1.: *BODY PLAN, STEM AND STERN OUTLINES AND CURVE OF SECTIONAL AREAS*

SHIP MODEL No. 5734 ————

SHIP MODEL No. 5734 A — — — —

SHIP MODEL No. 5734 B · · · · · · · · · · ·





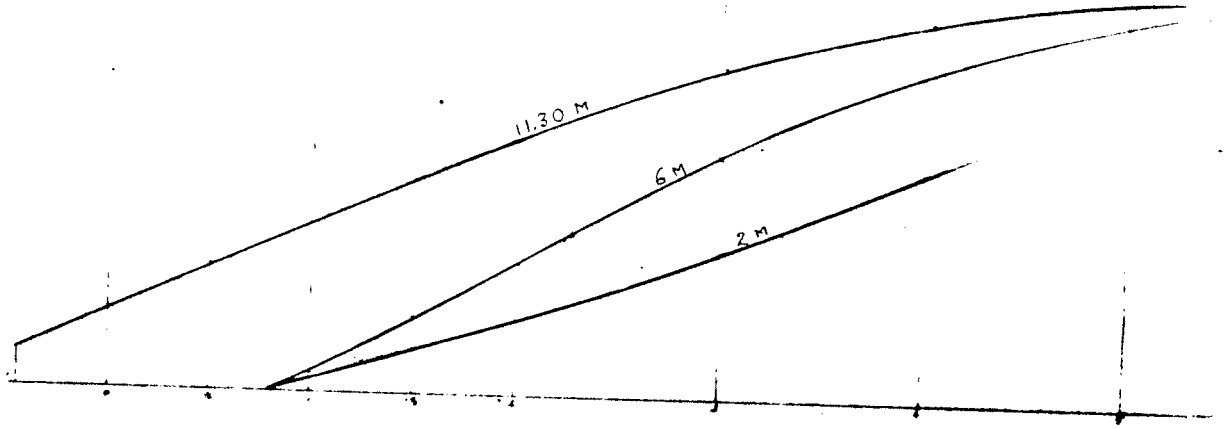
Maassluis
D = 5800 ; N = 122
D = 7100 ; N = 90
D = 8800 ; N = 60
D = 10800 ; N = 40

SEE P. 14

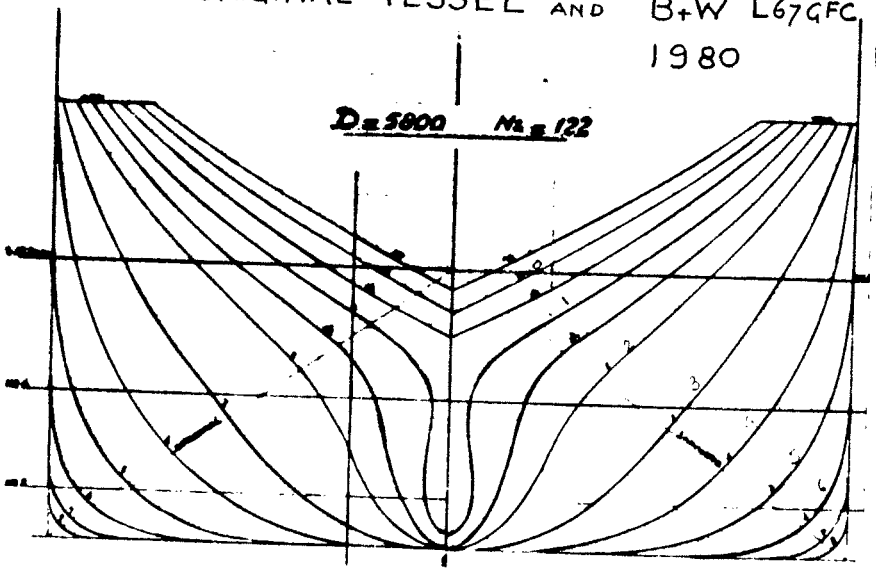
"MAASSLUIS" – INCREASE PROP ø

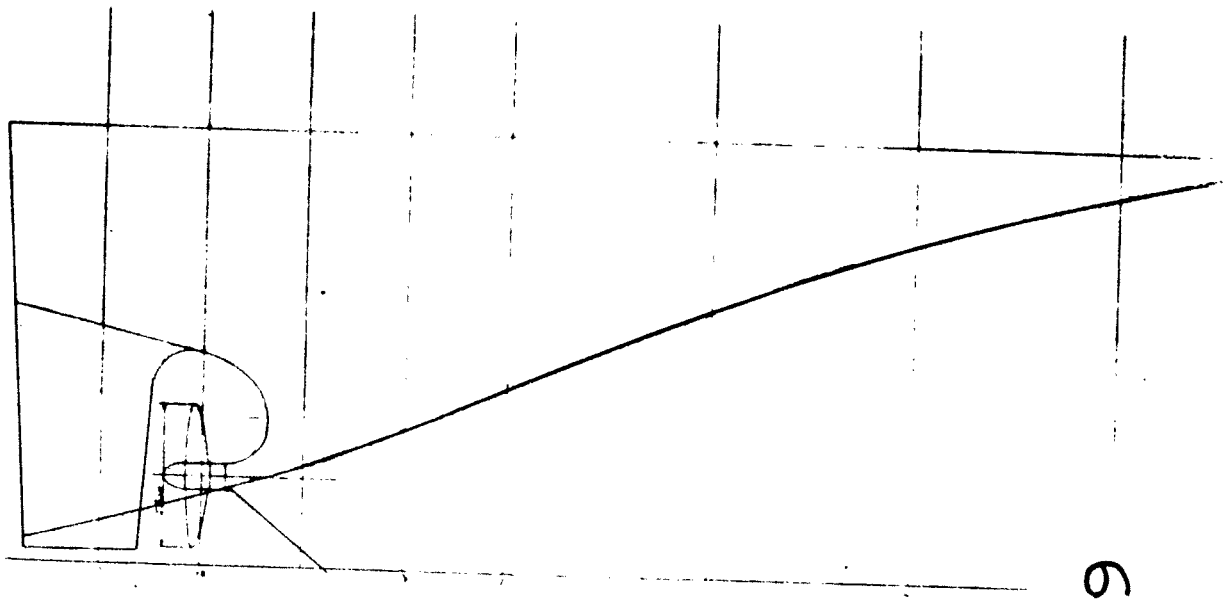
KEEP WATERLINE 2 M IDENTICAL

IN VIEW OF ENGINE FOUNDATION

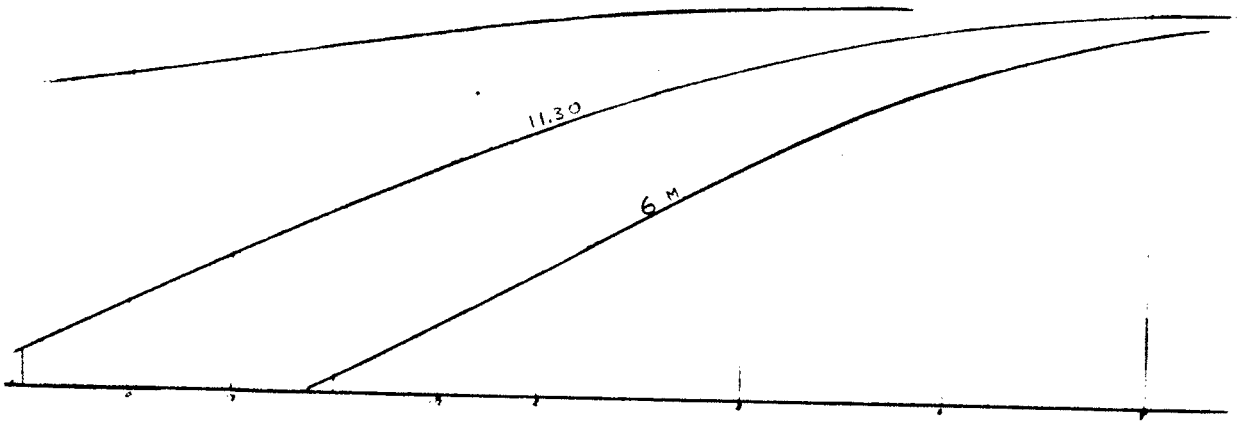


ORIGINAL VESSEL AND B+W L67GFC
1980

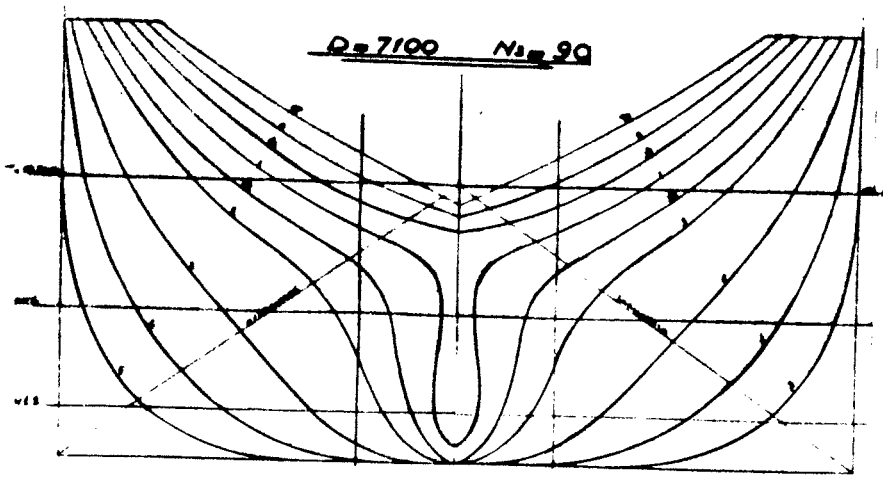


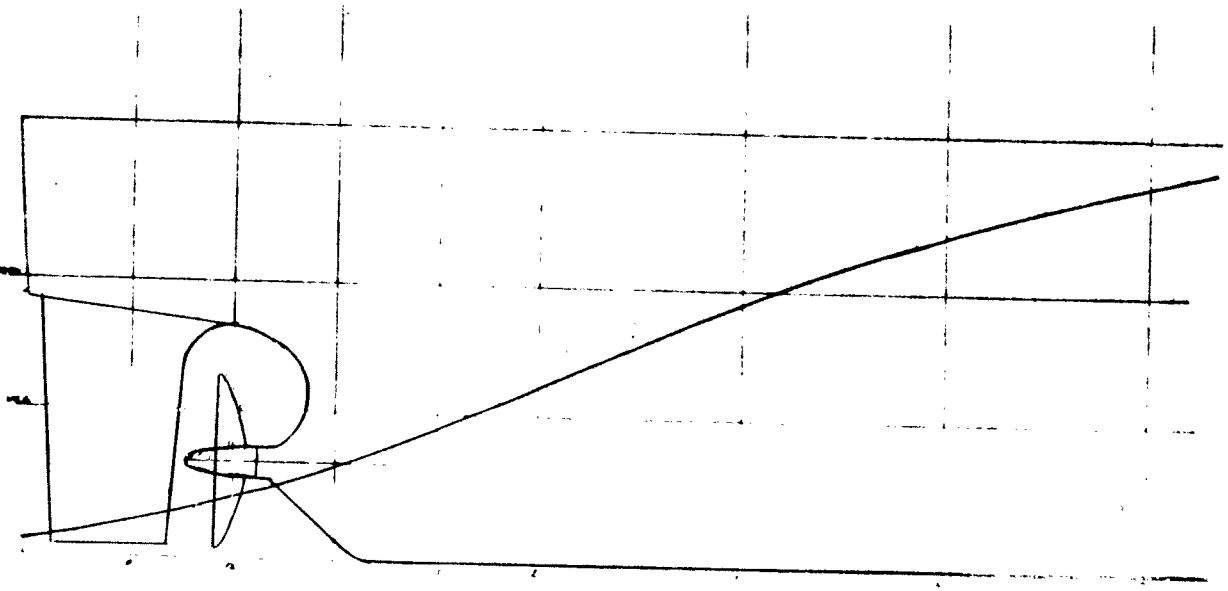


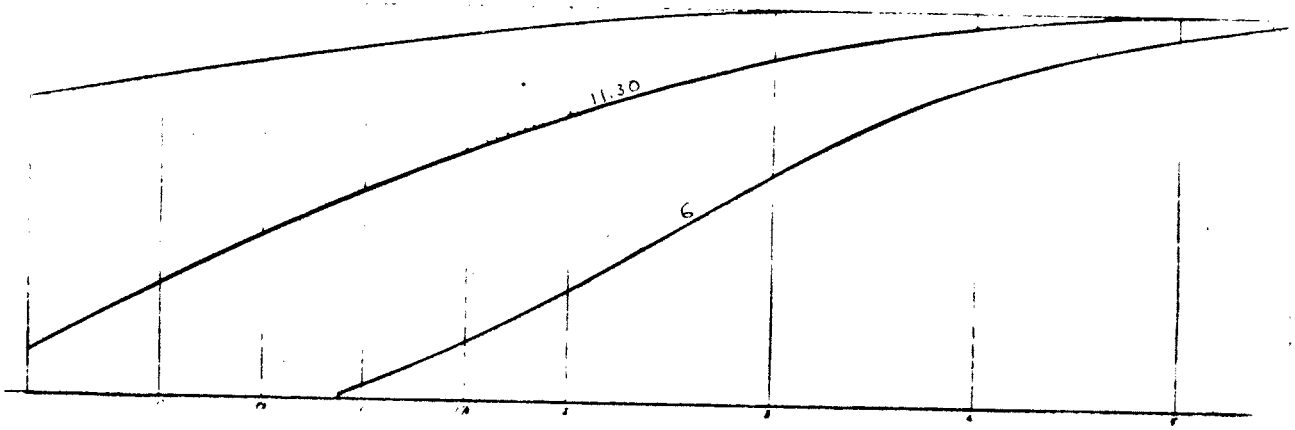
MAASSLUIS



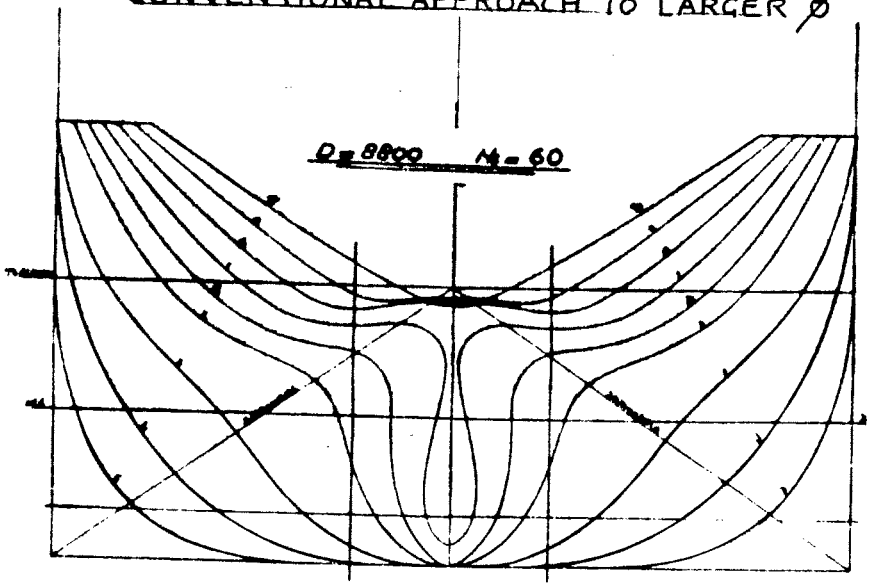
CONVENTIONAL APPROACH
TO LARGER PROP.

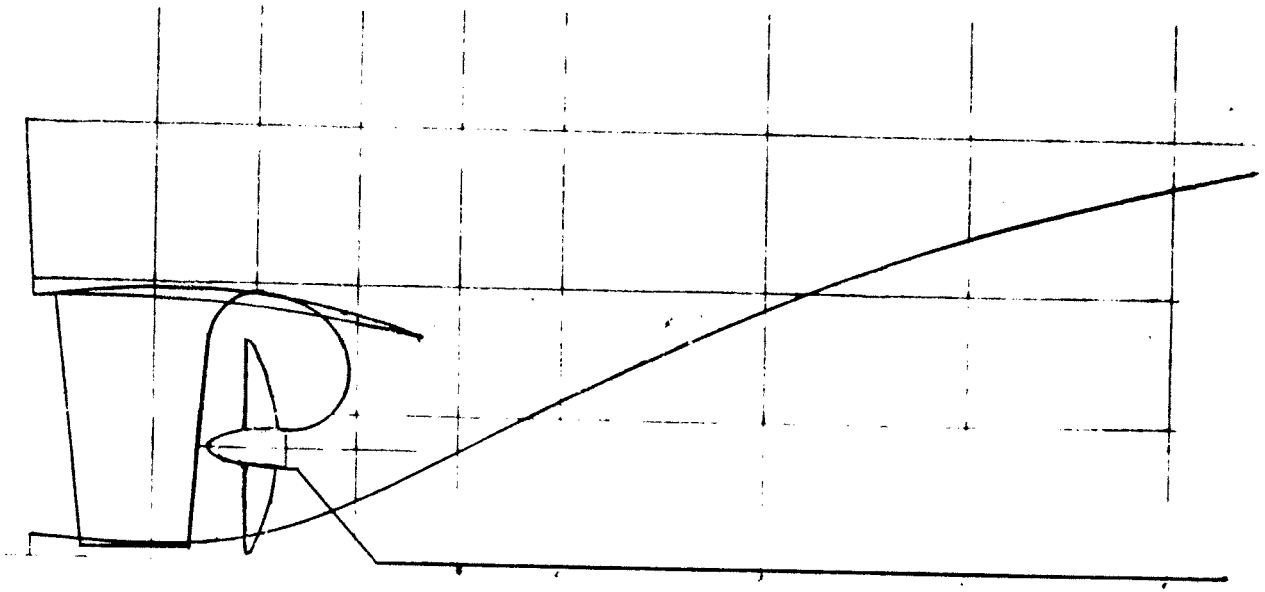




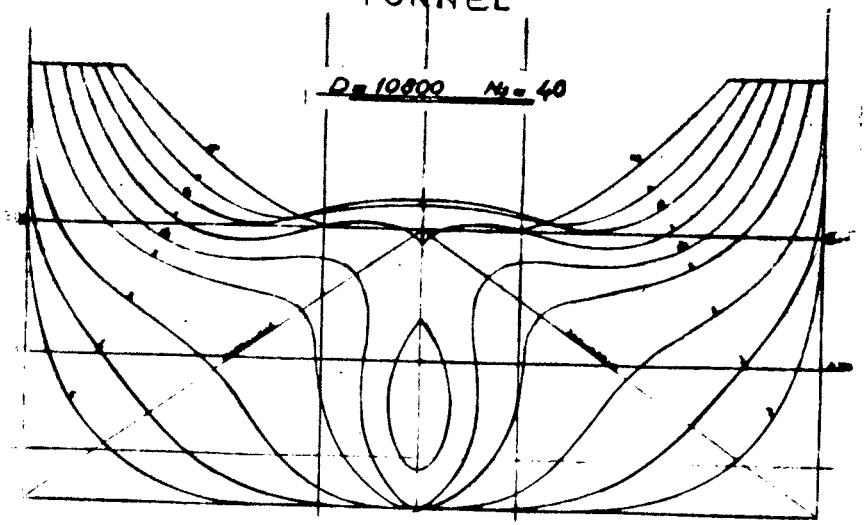


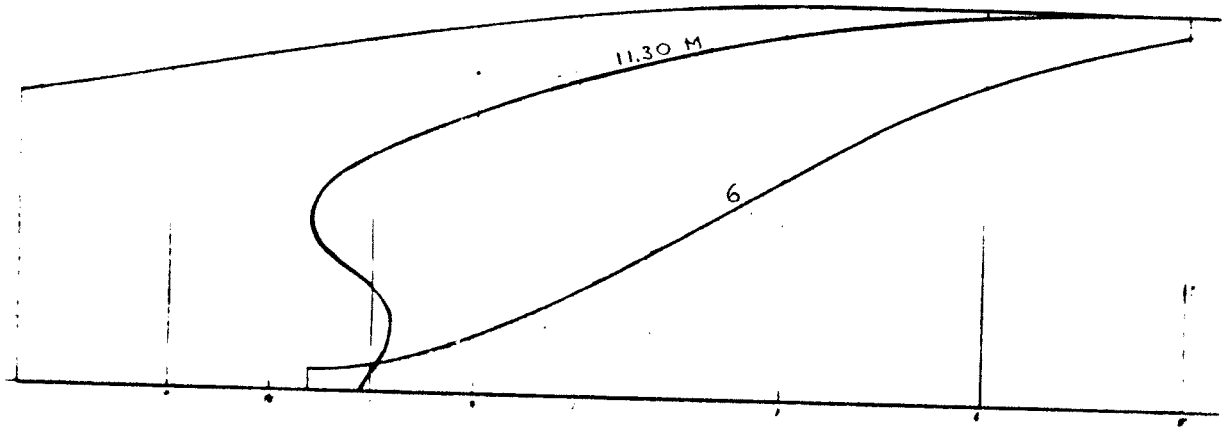
CONVENTIONAL APPROACH TO LARGER Ø

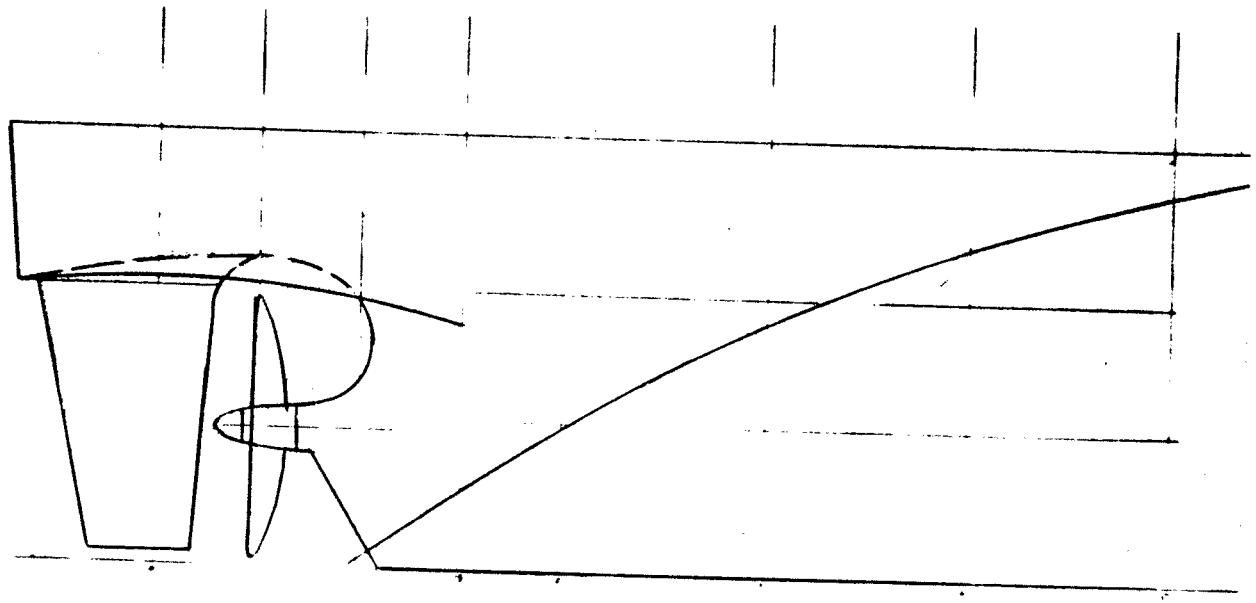




CONVENTIONAL APPROACH TO LARGER Ø TUNNEL







Example of research into R.P.M.

For a Product Tanker type ship.

Dimensions	Loa	Lpp	B(mld)	Depth	Draft (T) (all in metres)	d
aproximate	183	175.6	32.24	17.0	11.58 (38')	0.792

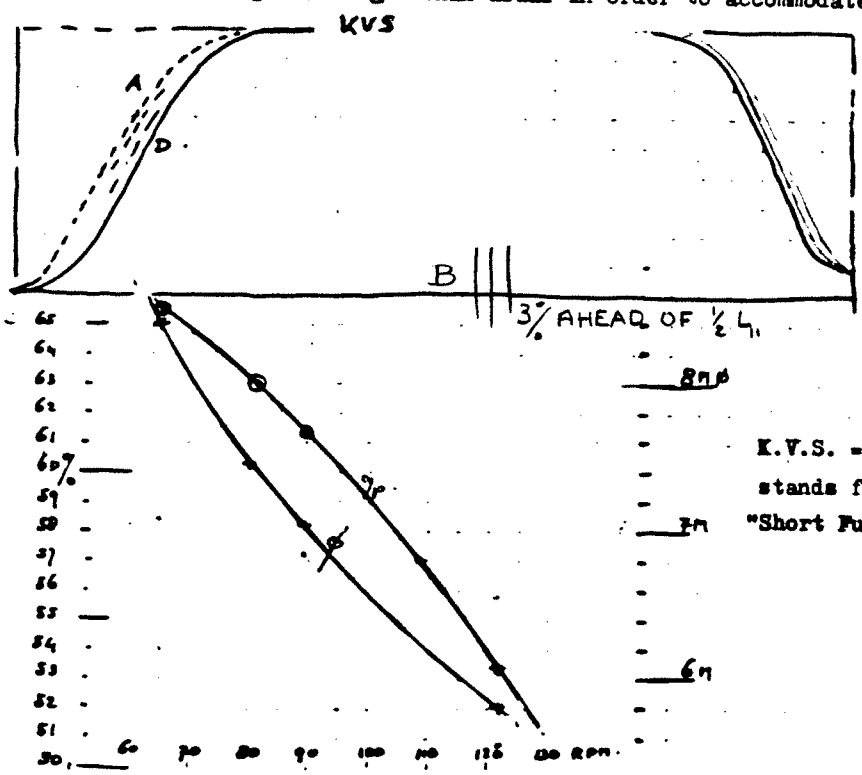
Light ship weight	10,400	Tonnes
Displacement (Lpp x B x T x d x 1.029)	53,400	Tonnes
∴ Deadweight "all told"	43,000	Tonnes

Prismatic coefficient (phi) taken = 0.796

Midship coefficient (betha) taken = 0.995

Ship-alt^ve	Engine(s)	Power (PK) installed	R.P.M.	Load (svce)	Service power	Prop. ø (4 bl)	Prop. effic^y	
(AA)	6 cyl x 67 ø B+W	11,135 /	127	x 0.85	9,450/122	5.80	53.4 %	x (1.80 ?)
(BB)	5 cyl x 68 ø Slzr	11,200 /	89	x 0.85 =	9,500/ 89	7.05	61.3 %	x (1.22 ?)
	4 cyl x 80 ø B+W	11,000 /	95	x 0.85 =				
(CC)	4 cyl x 76 ø SlzrRTA	11,160 /	81	x 0.85 =	9,470/ 81	7.45	63.—%	x (1.14 ?)
(DD)	2 x 8 MSpd SWD TM 410	11,000 /	65	x 0.85 =	9,340/ 65	8.40	65.6 %	x (1.06 ?)
(=pram-type)	1 x 6 MSpd SWD 620							

If proper, a ship due to get a propeller with a larger-than-usual diameter will have to be slightly longer-than-usual in order to accommodate such a propeller . So, (AA), (BB), (CC), (DD) being equal as to displacement, their bodies will differ in fulness and length.



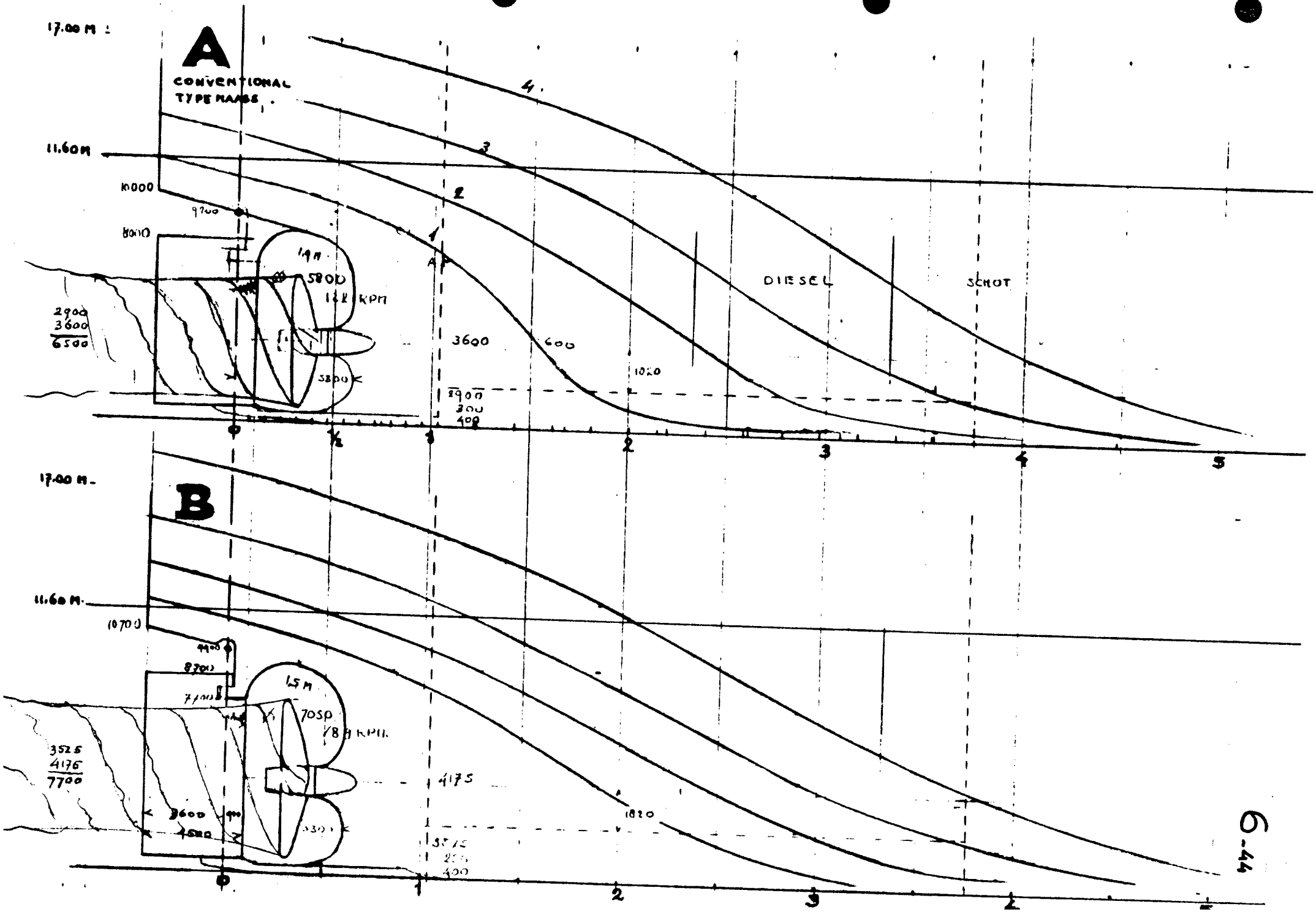
IT IS BETTER TO MOVE THE CENTRE OF BOYANCY IN FORWARD DIRECTION

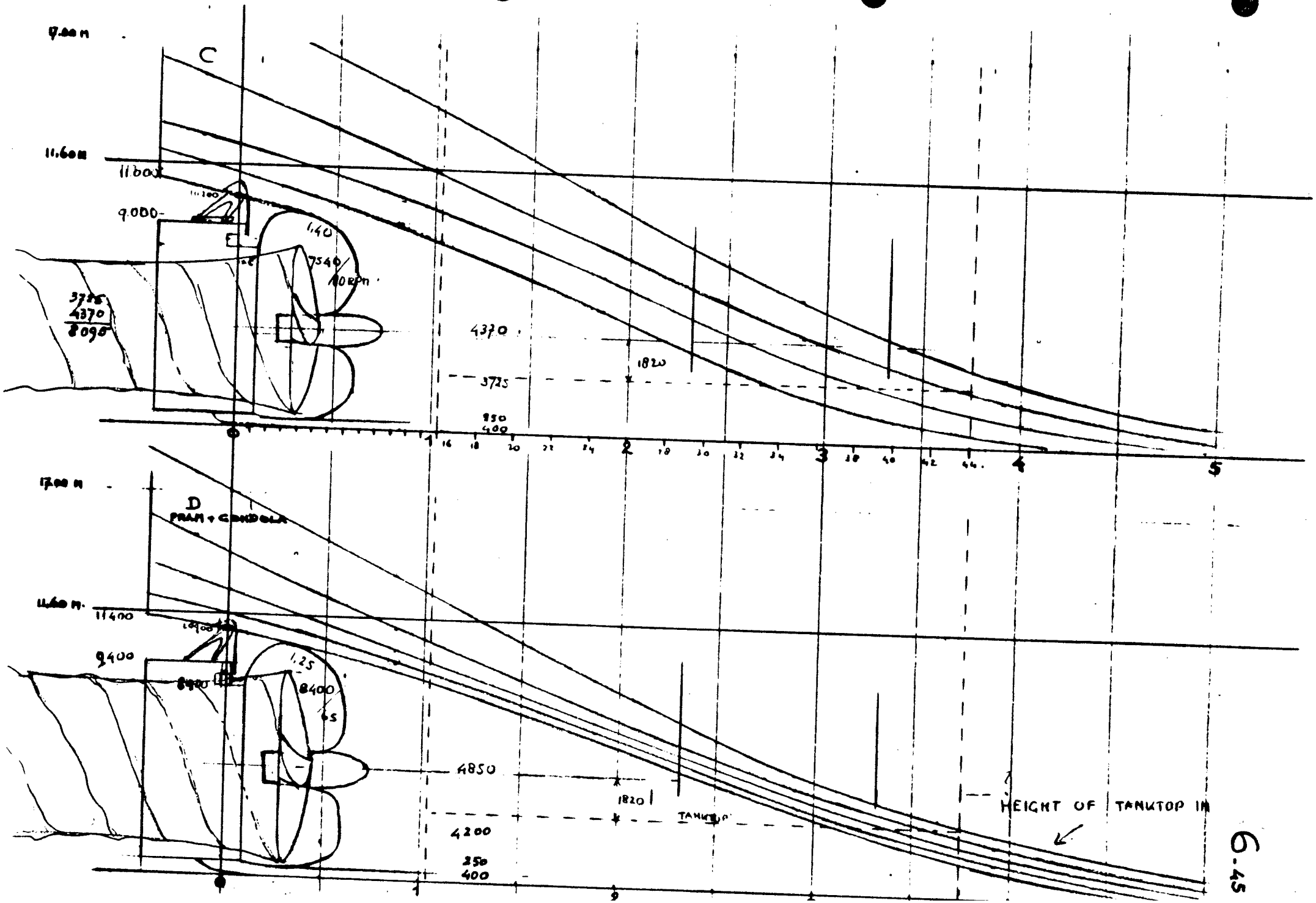
1983

K.V.S. = Korte Volle Schepen stands for "Short Fulbodied Ships"

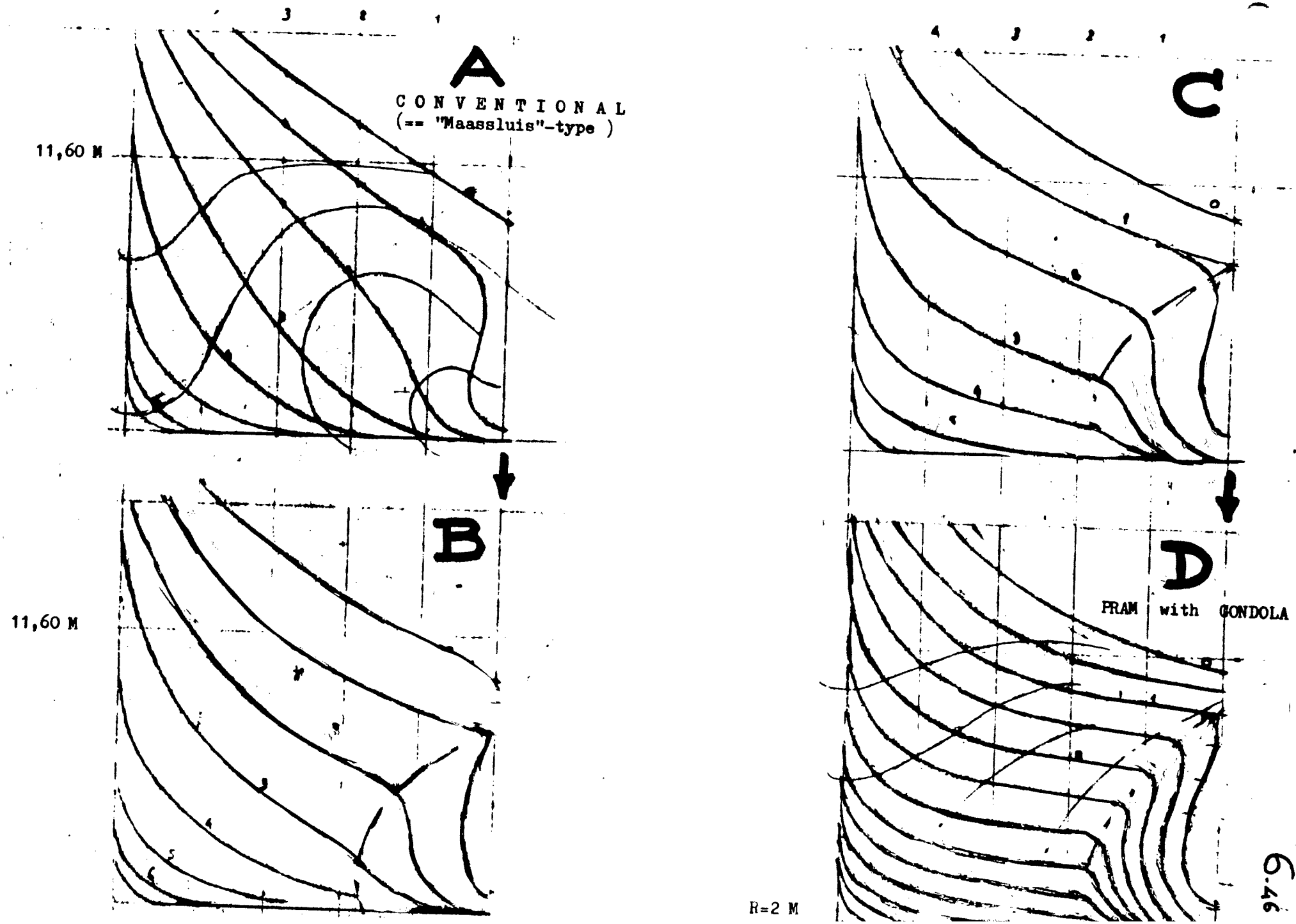
26-10-82.

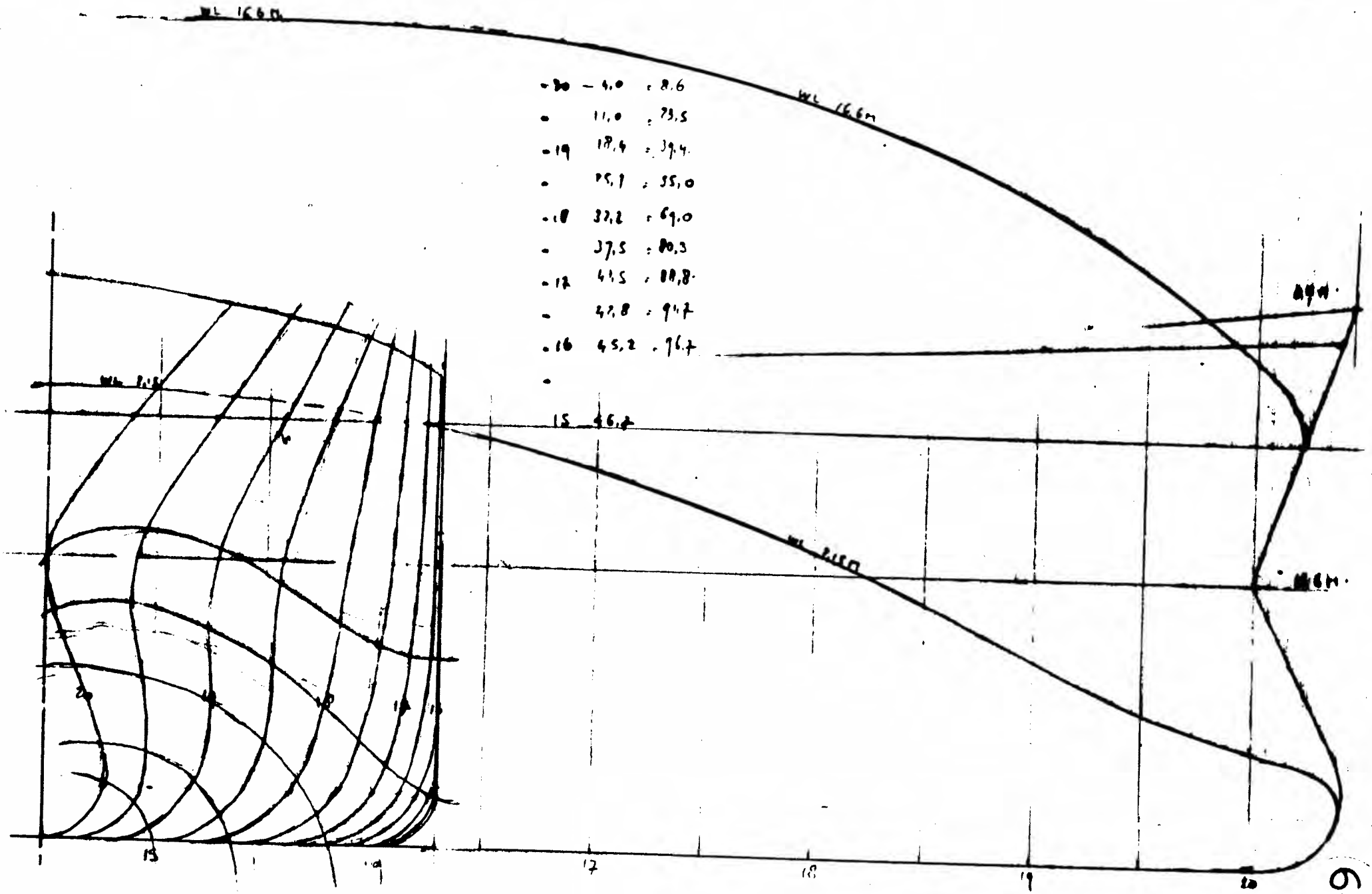
coe 7-11-82


A
CONVENTIONAL TYPE MASS
17.00 M
11.60 M
10000
9700
8000
1.9 M
5800
122 KPH
2900
3600
6500
3600
600
1020
2900
300
400
DIESEL
SCHOT
4
3
2
1
AP
0
1/2
1
2
3
4
5
B
17.00 M
11.60 M
10700
9900
8700
7700
1.5 M
7050
178 KPH
3525
4175
7700
3600
900
4500
4175
1820
3525
250
400
0
1
2
3
4


C
11.60m
11.600
9.000
1.40
7540
100 RPM
3725
4370
8090
4370
1820
3725
950
400
D
PRAM + GONDOLA
11.60 m.
11400
9400
1.25
8400
65
4850
1820
TANKTOP
4200
250
400
HEIGHT OF TANKTOP IN

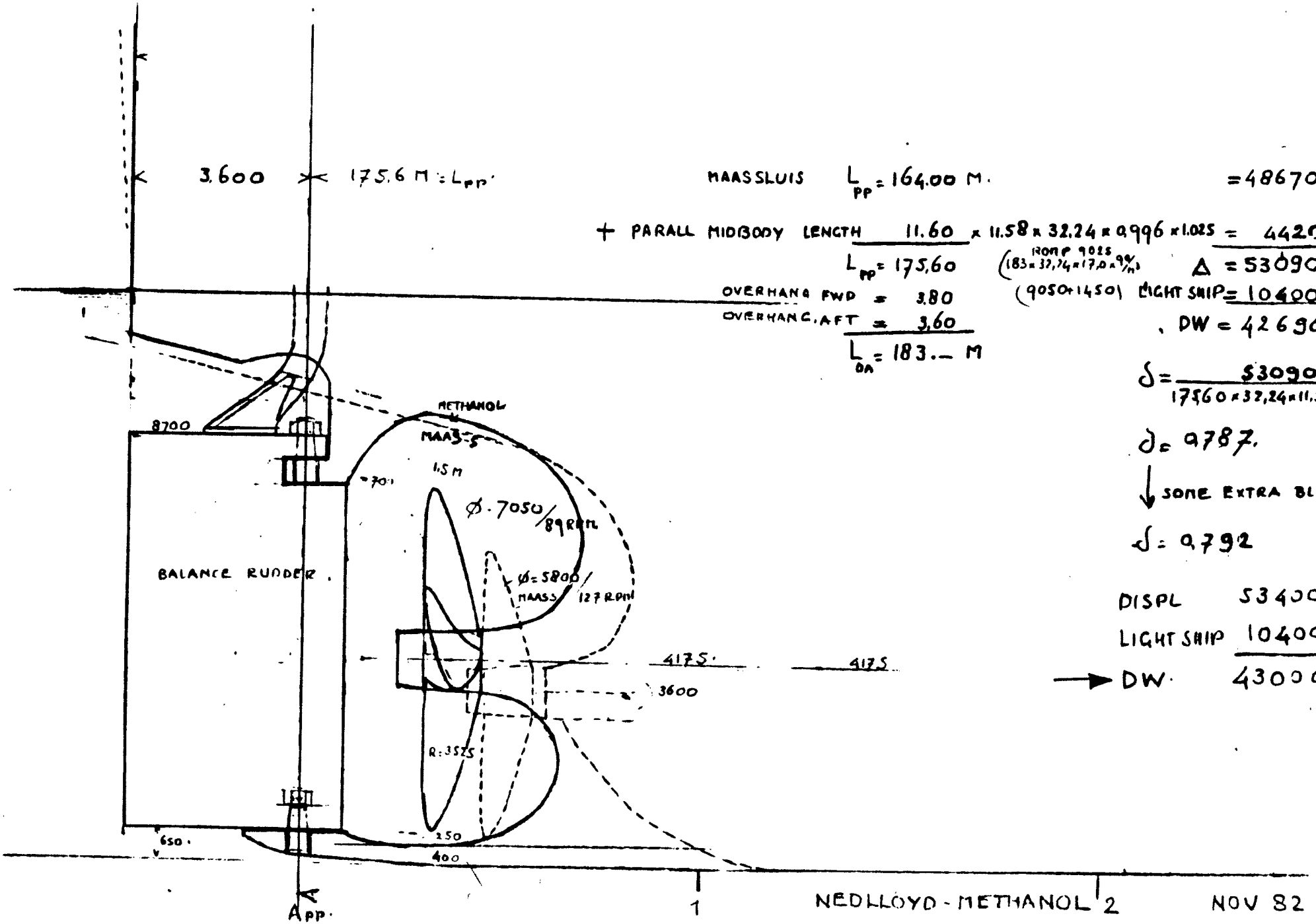
6.45


A
CONVENTIONAL
(== "Maassluis"-type)
11,60 M
B
11,60 M
C
D
PRAM with GONDOLA
R=2 M



-20	4.0	8.6
-	11.0	23.5
-19	18.4	39.4
-	25.1	55.0
-18	32.2	69.0
-	37.5	80.3
-17	41.5	88.8
-	42.8	91.7
-16	45.2	96.7
-		

23/10-'82



MAASSLUIS $L_{pp} = 164.00$ M. = 48670 TON

+ PARALL MIDBODY LENGTH 11.60 x 11.58 x 32.24 x 0.996 x 1.025 = 4420 TON

$L_{pp} = 175.60$ (183 x 32.24 x 17.0 x 9%) IRON P 9025 Δ = 53090 TON

OVERHANG FWD = 3.80 (9050+1450) LIGHT SHIP = 10400 TON

OVERHANG AFT = 3.60 . DW = 42690 TON

$L_{OA} = 183.-$ M

$$\delta = \frac{53090 \text{ TON}}{17560 \times 32.24 \times 11.58 \times 1.029}$$

$\delta = 0.787$

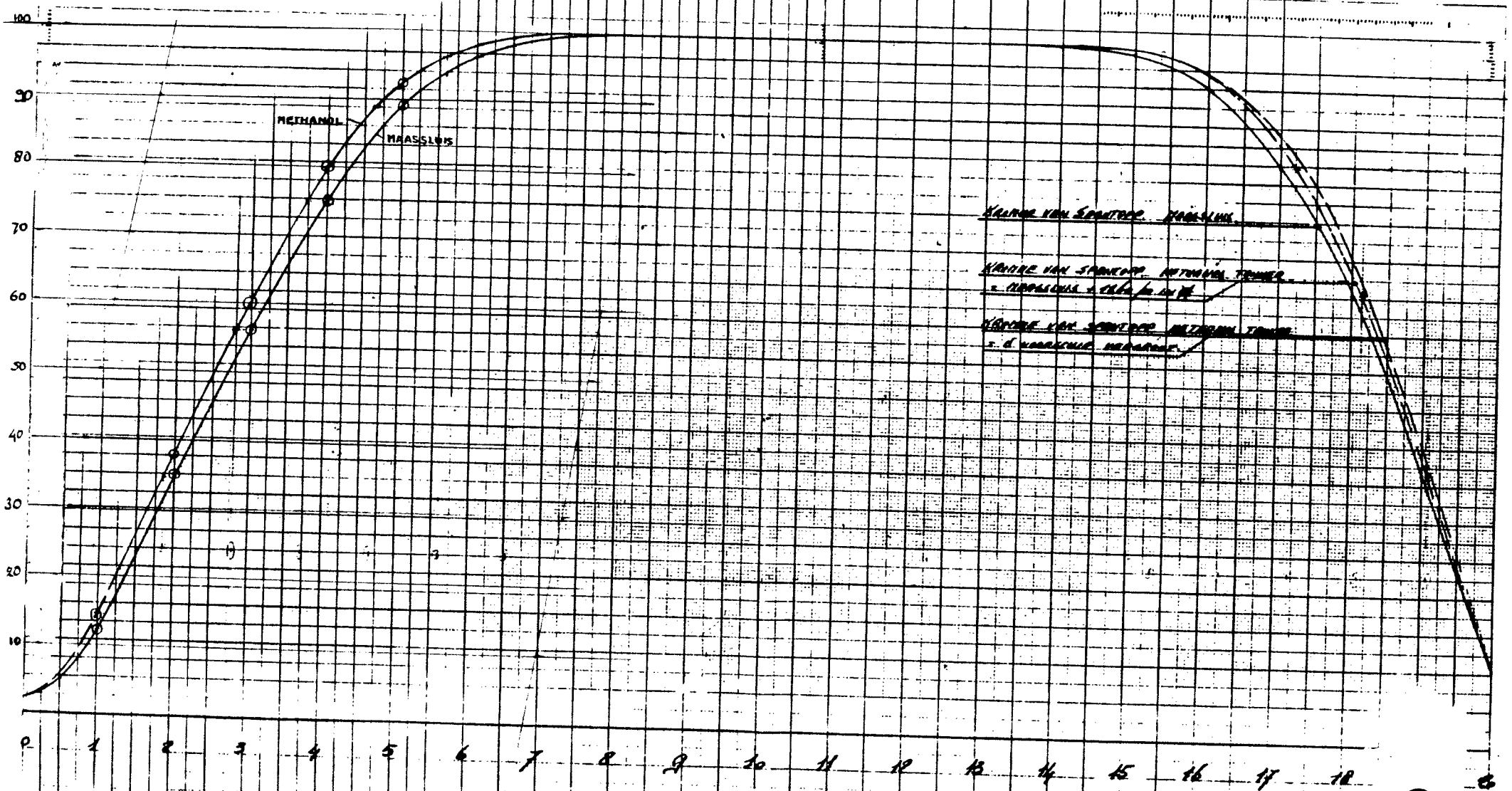
↓ SOME EXTRA BLOCK.

$\delta = 0.792$

DISPL	53400 TON
LIGHT SHIP	10400 TON
→ DW.	43000 TON

NEDLLOYD - METHANOL NOV 82

6-48



CURVE OF DISPL. NEDLLOYD METHANOL

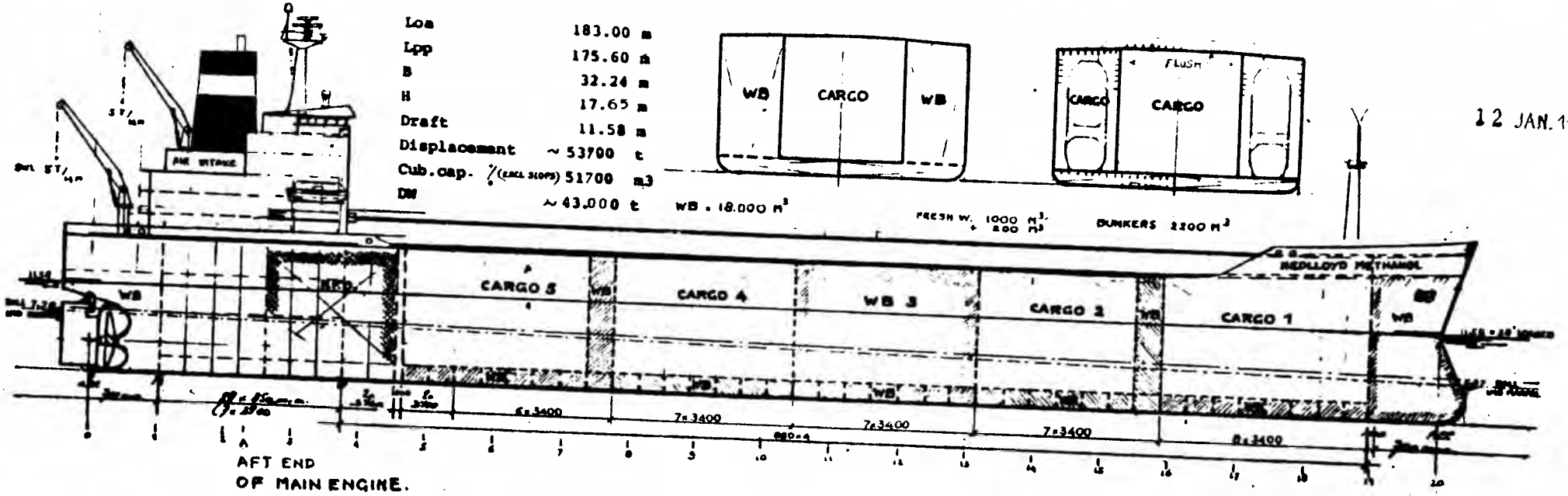
12 JAN. 1983

Loa	183.00 m
Lpp	175.60 m
B	32.24 m
H	17.65 m
Draft	11.58 m
Displacement	~ 53700 t
Cub.cap. % (incl. slops)	51700 m3
DW	~ 43.000 t

WB = 18.000 m³

FRESH W. 1000 m³ + 200 m³

BUNKERS 2200 m³



"NEDLLOYD METHANOL" (-DESIGN)

85% MCR = 11.500 HP

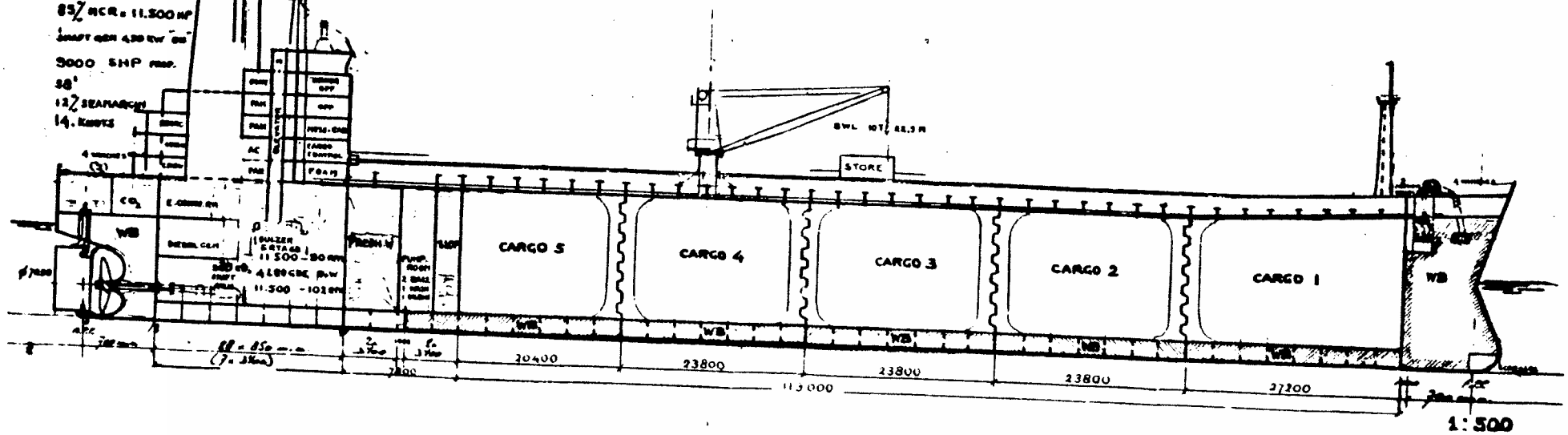
SHAFT GEN 430 kW

5000 SHP PROP.

58'

12% SEAMARGIN

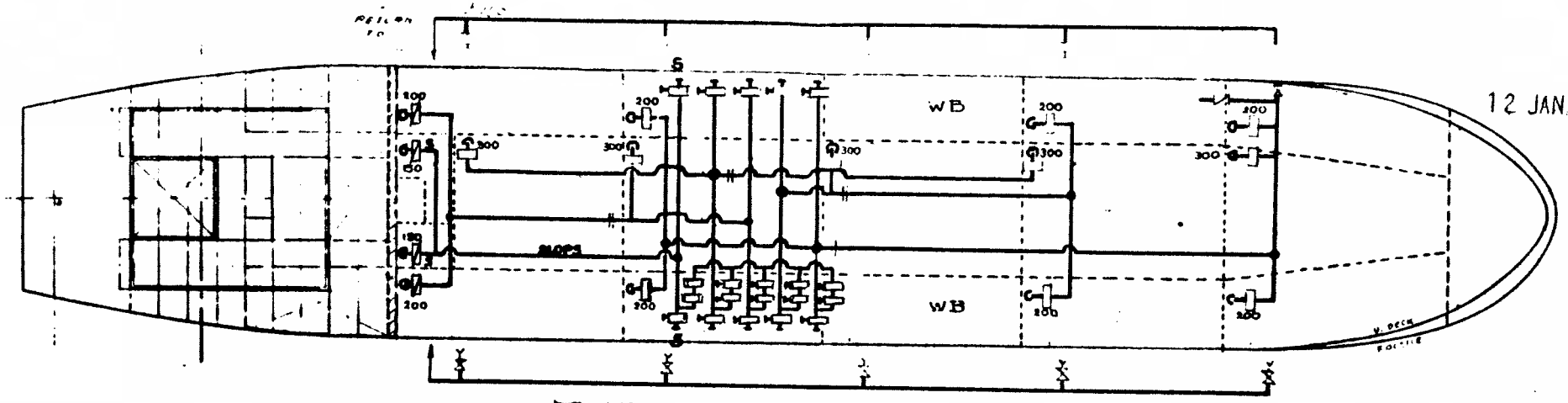
14. KNOTS



6.50

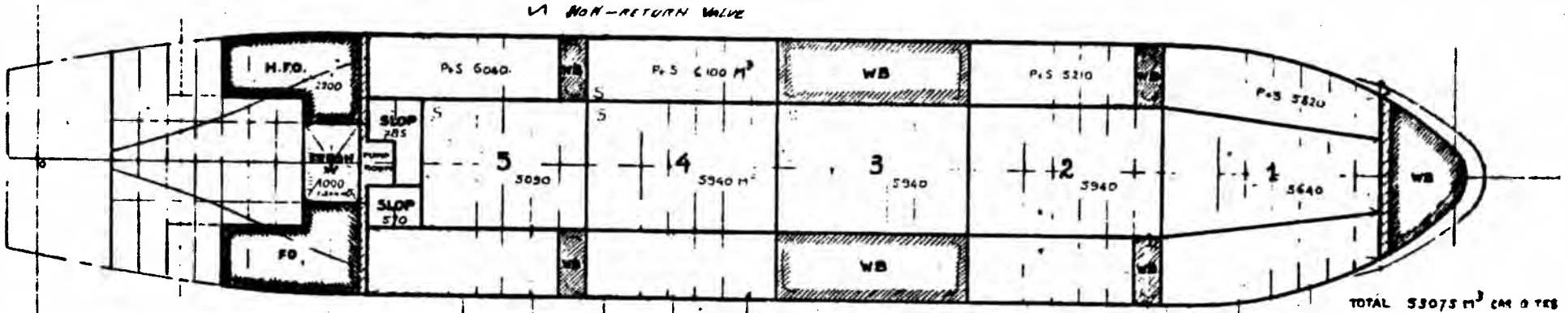
1:500

12 JAN. 1983

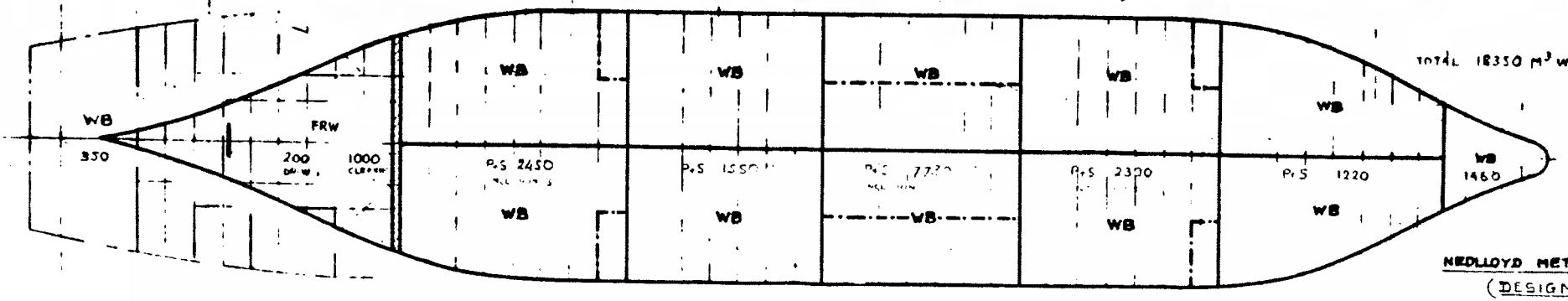


HYDRAULICALLY – REMOTELY CONTROLLED BUTTERFLY VALVE
HAND-OPERATED BUTTERFLY VALVE
HAND-OPERATED SHUTDOWN VALVE
NON-RETURN VALVE

4 SEGREGATIONS



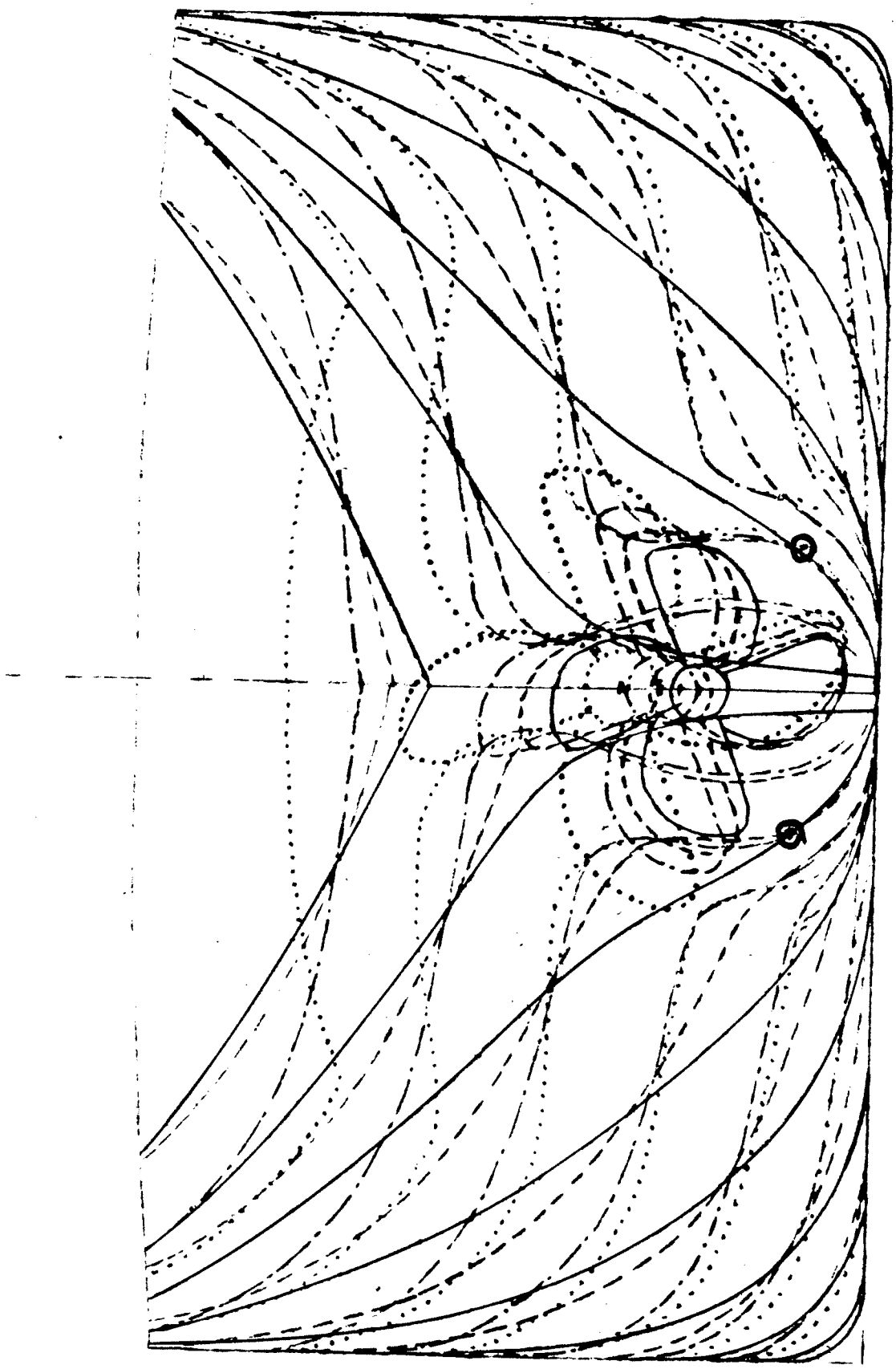
TOTAL 53075 M³ CAR & TKS

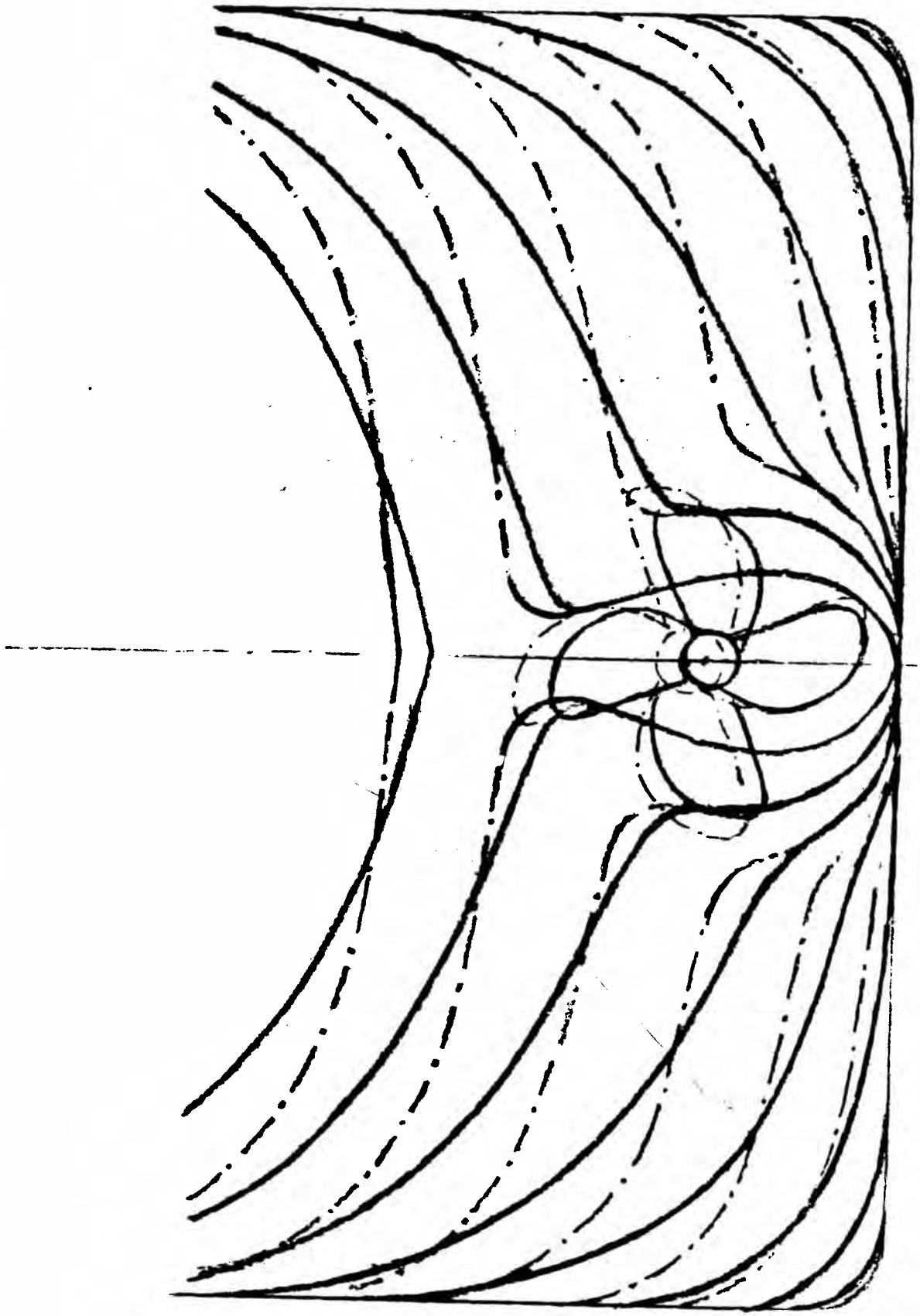


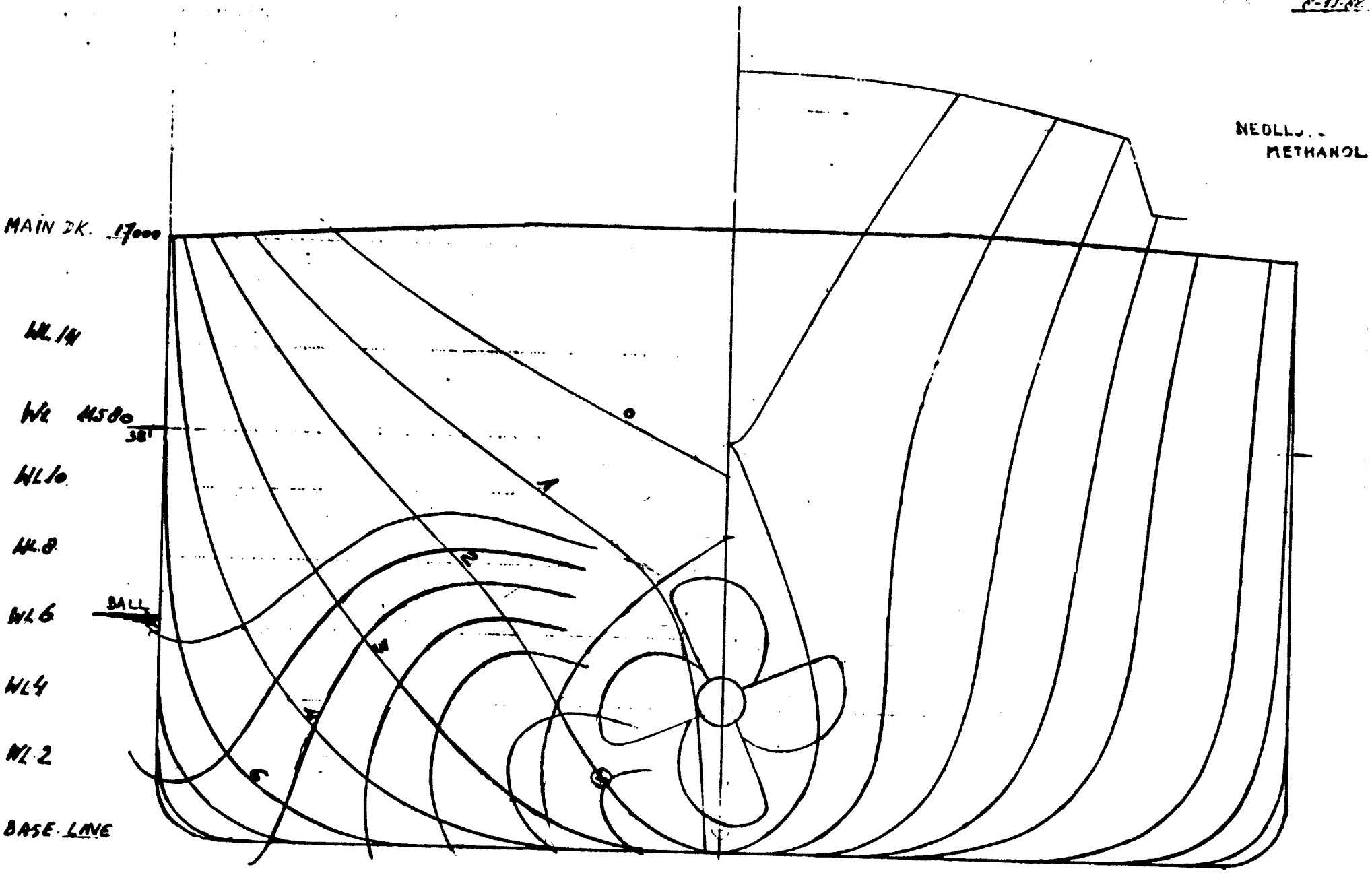
TOTAL 18350 M³ WA

6-51

NEDLLOYD METHANOL
(DESIGN)

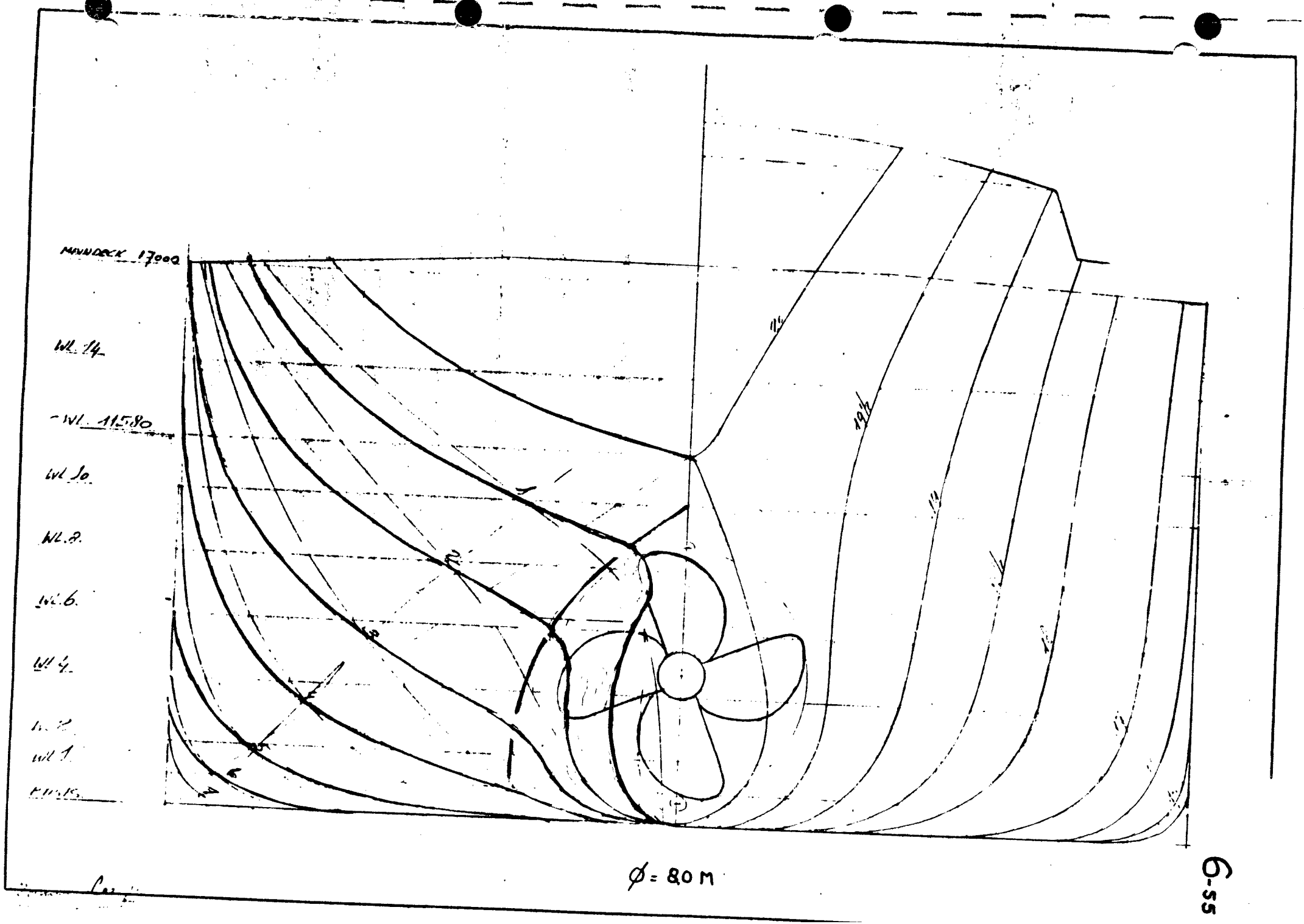


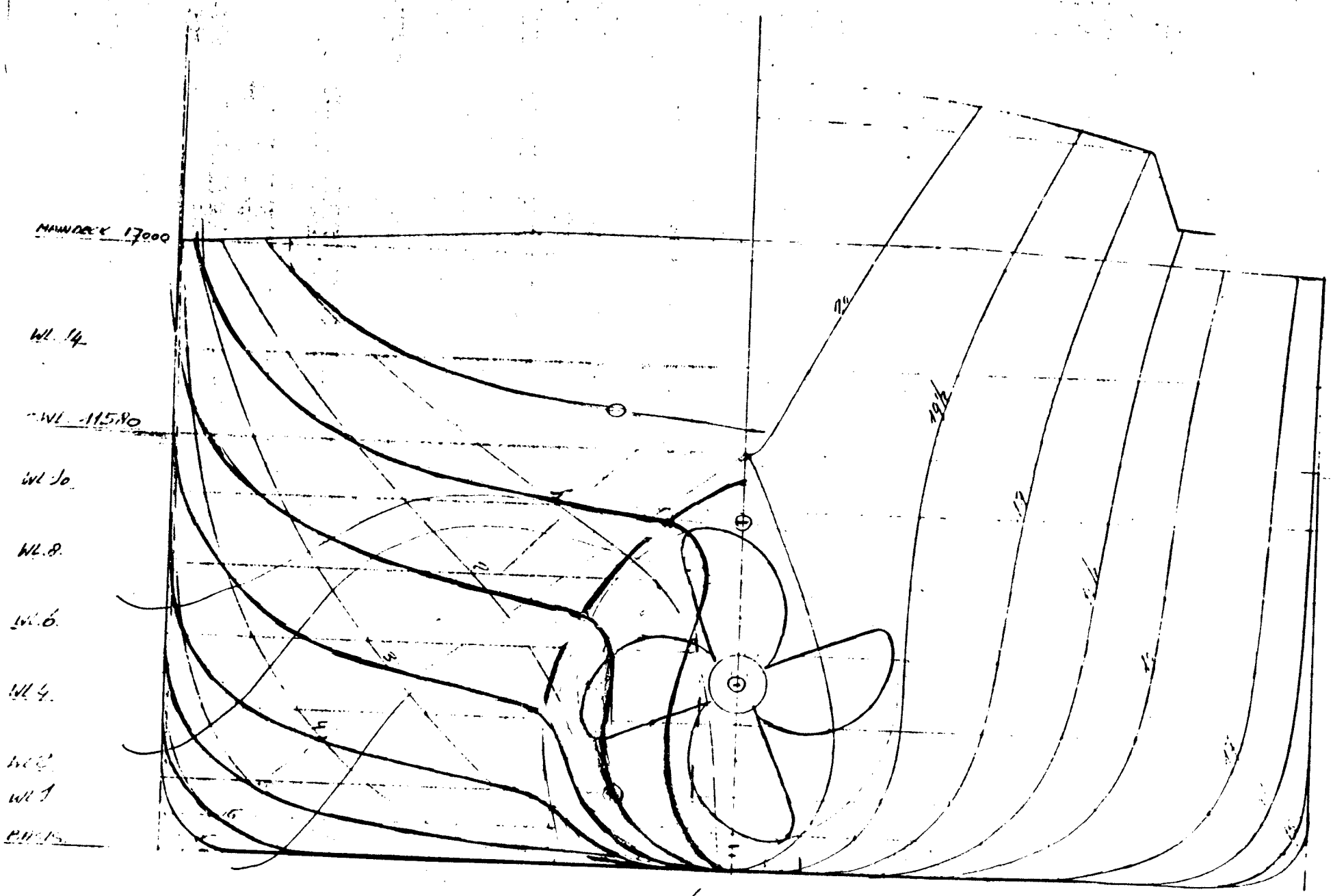



NEDLLO..
METHANOL
MAIN DK. 17000
WL 14
WL 12 11580
38'
WL 10
WL 8
WL 6
BALL
WL 4
WL 2
BASE LINE
NEDLLOYD METHANOL

$\phi = 7.2$ M.

6-54


MAINDECK 17000
WL 14
WL 11580
WL 10
WL 8
WL 6
WL 4
WL 2
WL 1
BASE
Ø = 8,0 M

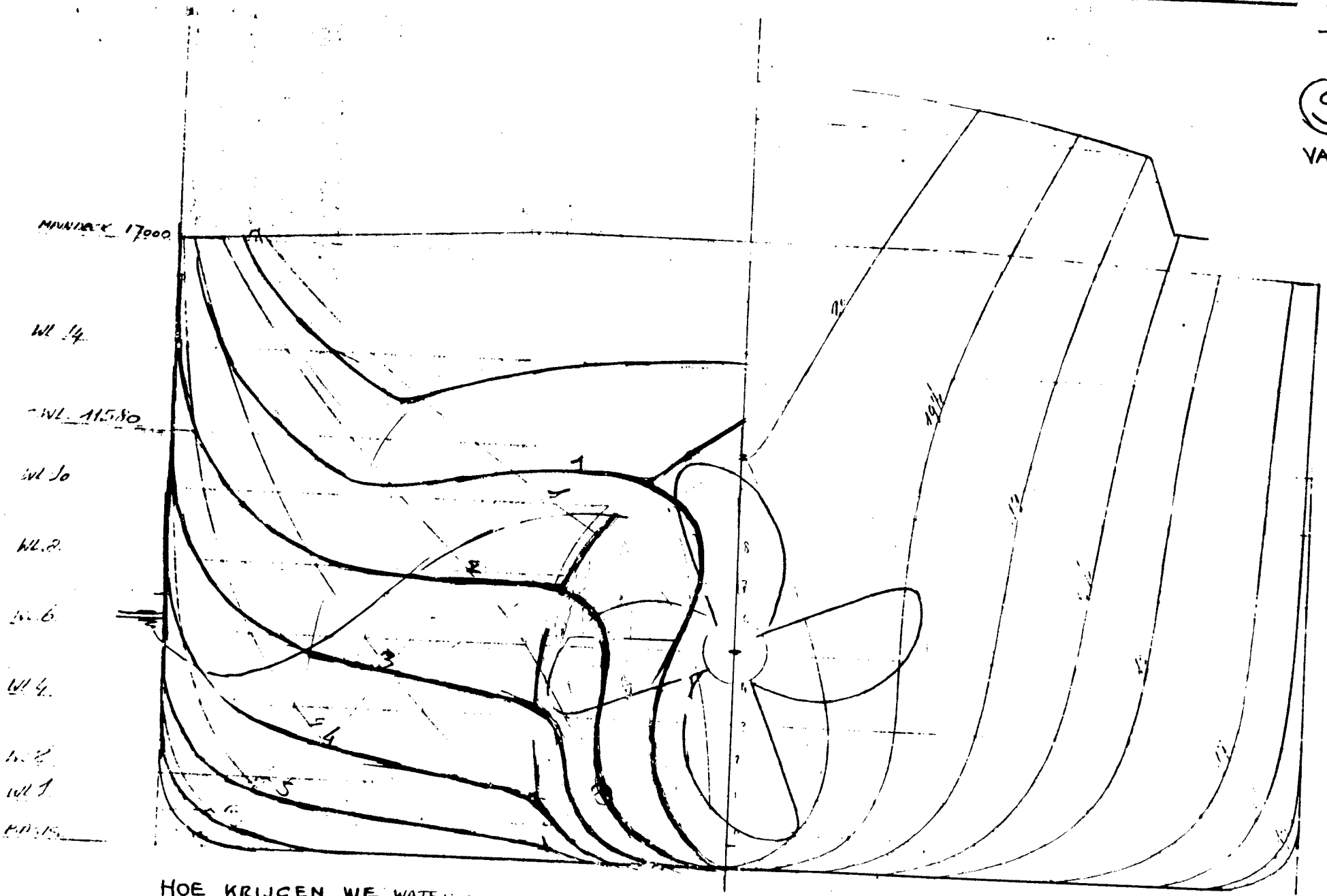

MAIN DECK 17000
WL 14
WL 11.580
WL 10
WL 8
WL 6
WL 4
WL 2
WL 1
BHD 15
19½
19
18½
18
17½
17

Ø = 9.0 M

8-11-82

S

VARIANT



7

Research into **SLOW-TURNING PROPELLERS ON SLOW/FULL-BODIED VESSELS.**

This would seem profitable for certain types of ships, e.g.:-

Ship type	Particulars								
	d	DW a/t (tonnes)	Loa (metres)	Lpp (metres)	Br. (metres)	Depth (metres)	Draught	Installed power (PK)	Multiplying factor
Coaster/Chem. tanker	.75	2,800	83	79.6	14.6	7.7	5.28 (17')	3,000	0.453 x
Great Lakes trader	...	...	..	...	22.8	...	...	...	.
Product tanker / Bulker (14 kn)	.792	43,000	183	175.6	32.24	17.0	11.58 (38')	11,000	1.- x
Bulker + 10 M	...	46,000	193	185.6	,,	,,	,,	..	
Bulker + 20 M	...	49,000	203	195.6	,,	,,	,,	. .	
Panmax bulker	.82	80,000	256	245	,,	19.3	14.3 (47')	18,000	.
Suez bulker	...	130,000	260	249.5	45.8	24.1	16.45 (54')	22,000	1.42 x
VLCC	.83 .85	375,000	361	364	63.5	33.5	22.8 (75')	40,000	1.97 x

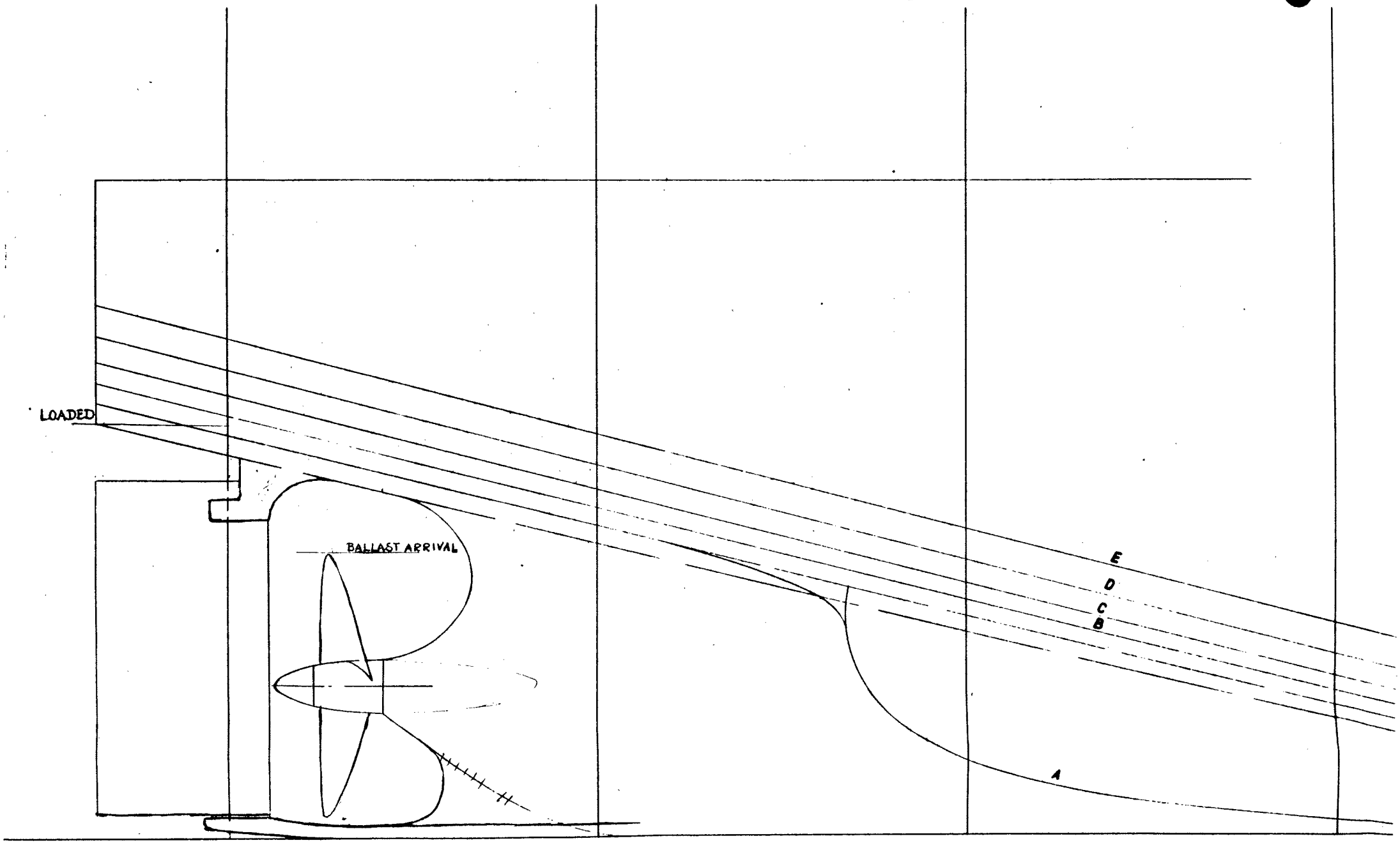
Conclusion is: "Pick Your Choice" from any of the w e i g h t carriers for improving their power efficiency by Greater Propeller ø coupled with Lower R.P.M.

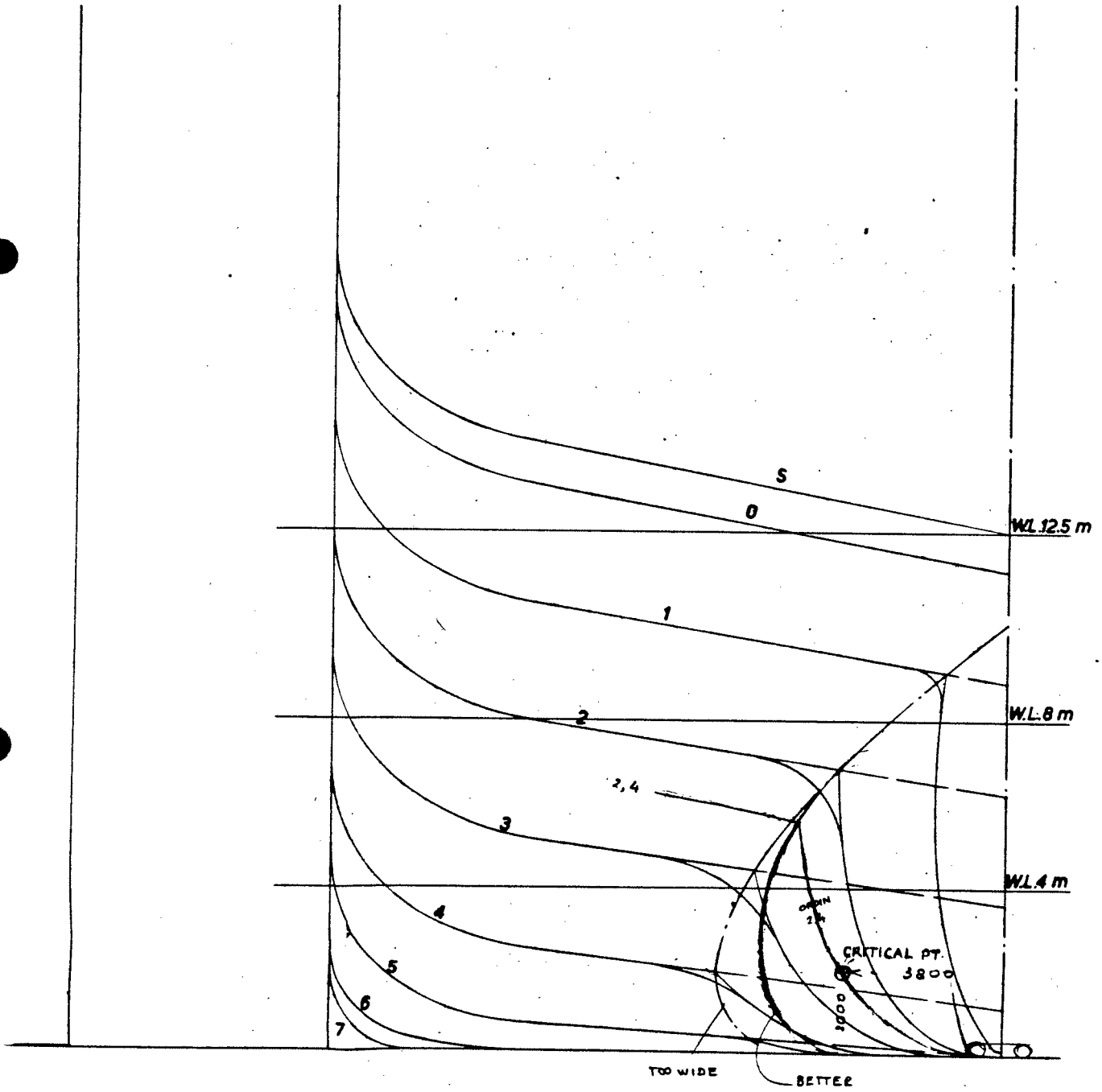
Note.

Nedlloyd - in line with any prospective needs - has proposed to have the following research carried out in the Netherlands for (⊗) a Product tanker/Bulker (14 kn.) type, as follows:-

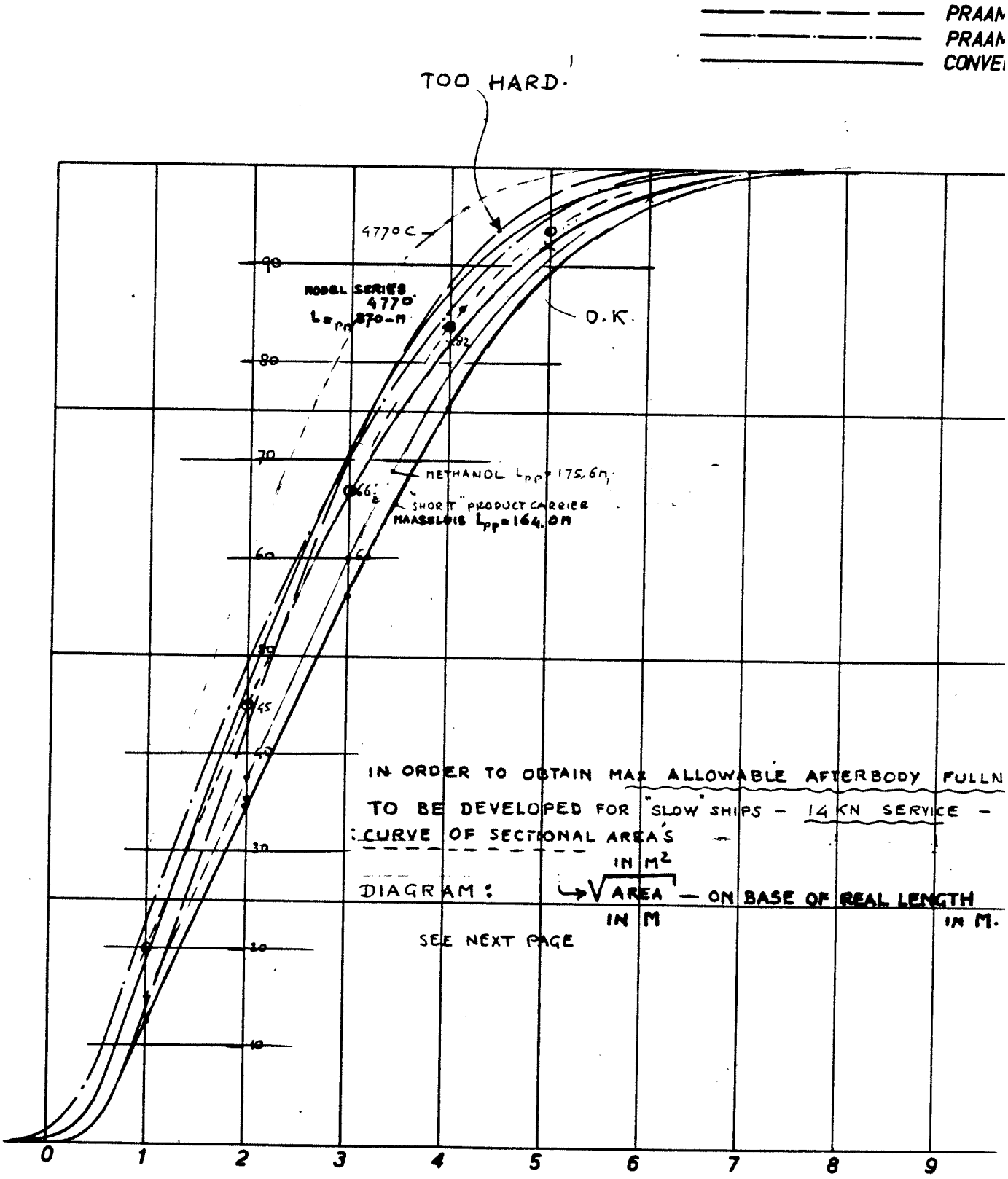
Take 4 or 5 alternatives, e.g.:-

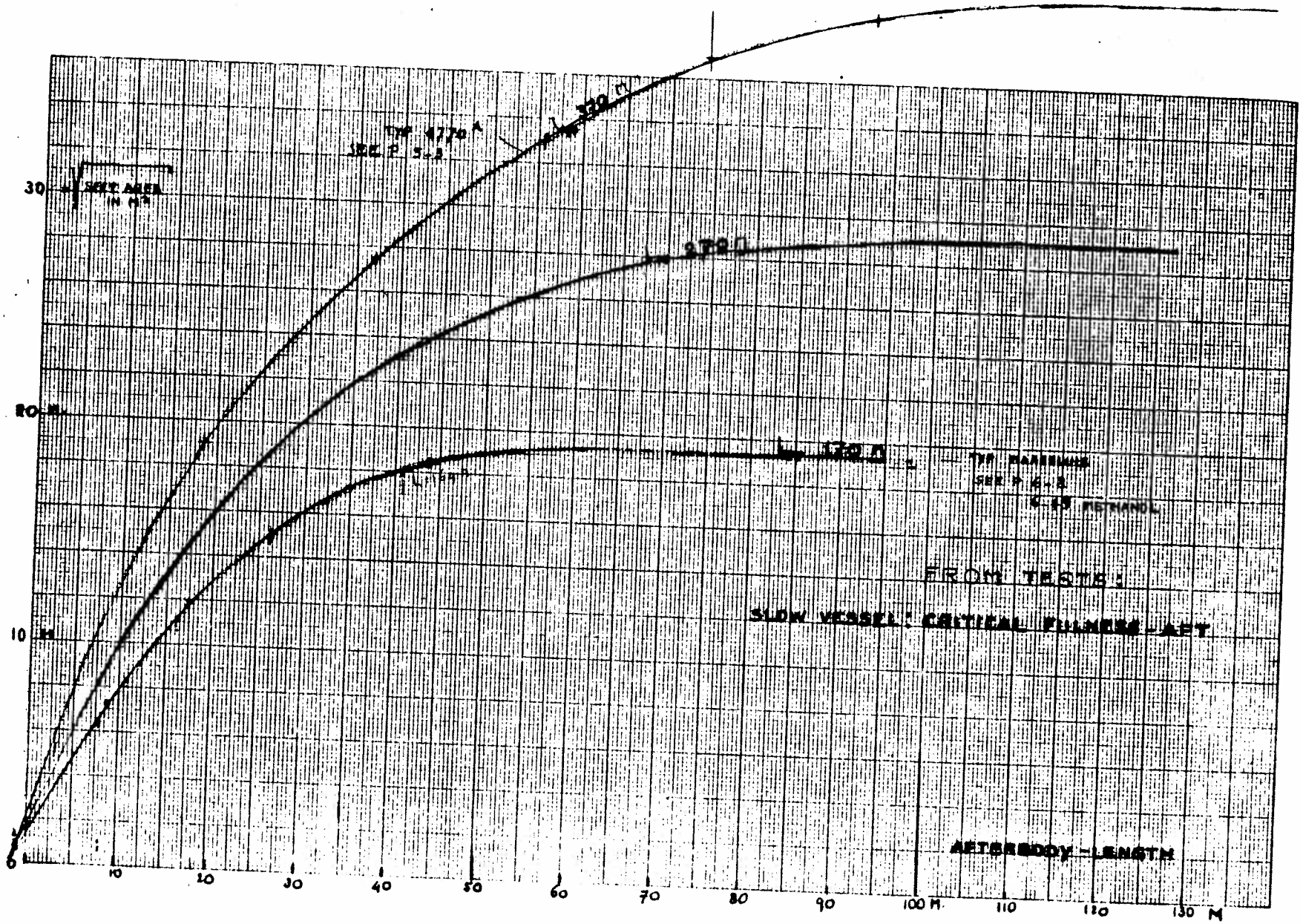
Alternative	Speed	Installed power	R.P.M.	Propeller ø	$\eta_{PROPELLER}$
(A)	14 kn.	11,000 pk	122	5.8 M	53.4 %
(B)			89	7.05 M	61.3 %
(C)			81	7.45 M	63.- %
(D)			65	8.40 M	65.5 %
(E)			45	?	?


LOADED
BALLAST ARRIVAL
E
D
C
B
A


S
0
1
2
2,4
3
4
5
6
7
W.L.12.5 m
W.L.8 m
W.L.4 m
CRITICAL PT.
3800
TOO WIDE
BETTER

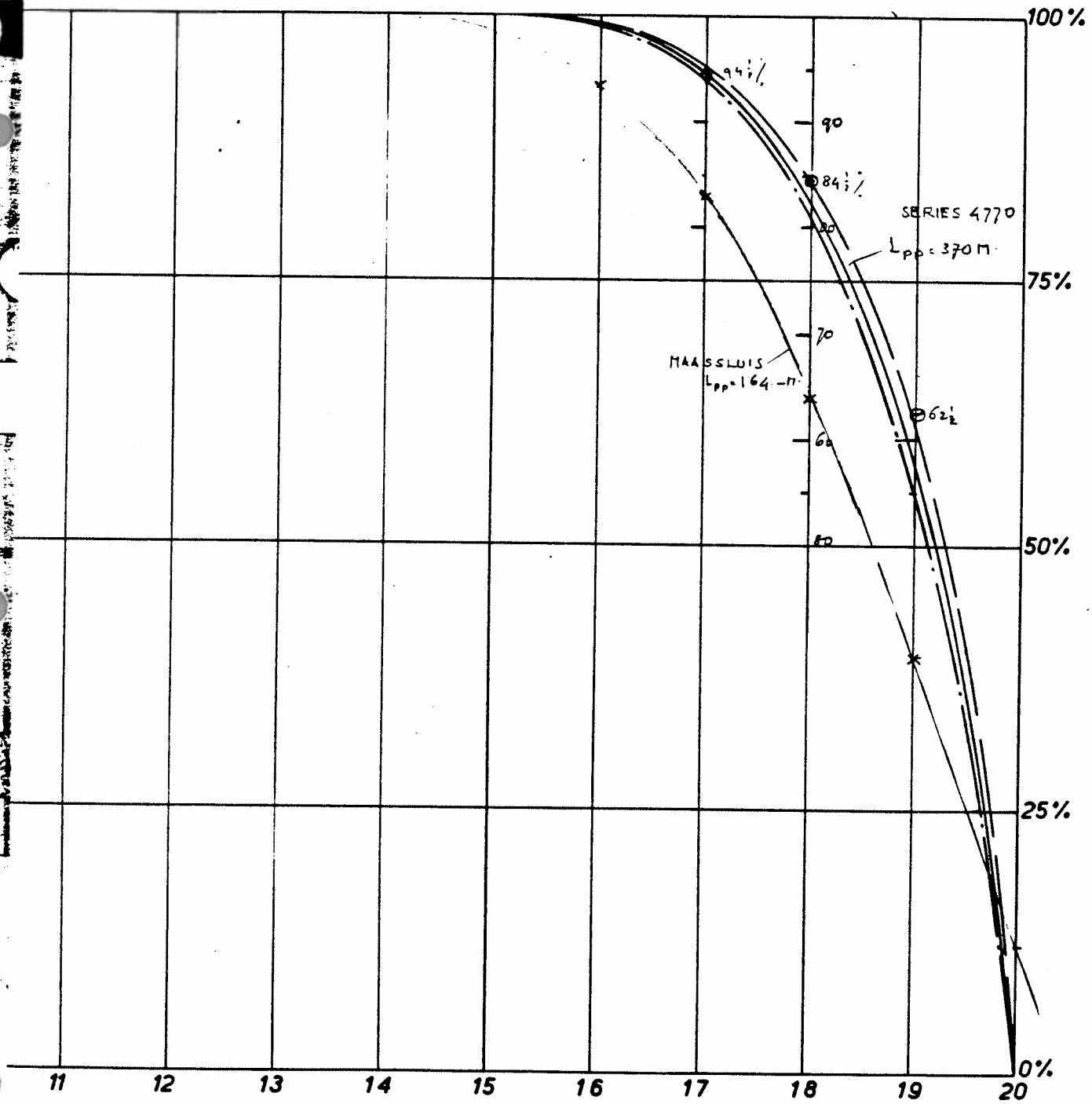
KROMMEN VAN S




30
20 M
10 M
6
10
20
30
40
50
60
70
80
90
100 M
110
120
130 M
320 M
270 M
170 M
FROM TESTS:
SLOW VESSEL: CRITICAL FULLNESS - AFT
AFTERBODY - LENGTH

PERVLAKKEN

Ns 40 RPM-D=11000 mm
Ns 60 RPM-D=8750 mm
Ns 60 RPM-D=8750 mm



TEK. No. P 45461-6

Ultra-long Stroke Diesel Engines

7-

HITACHI B&W

L-MC MCE

Main Engine Data

L90 MC/MCE Stroke 3940 mm; Bore 900 mm

	Speed rev/min	Pb (bar)	Power	Number of cylinders 4	5	6	7	8	9	10	11	12
Nominal MCR												
MC	74	15.0	kW	13 760	17 200	20 640	24 080	27 520	30 960	34 400	37 840	41 280
			BHP	18 720	23 400	28 080	32 760	37 440	42 120	46 800	51 480	56 160
MCE	74	13.0	kW	11 920	14 900	17 880	20 860	23 840	26 820	29 800	32 780	35 760
			BHP	16 240	20 300	24 360	28 420	32 480	36 540	40 600	44 660	48 720
Dry mass (tons)				775	890	1 025	1 165	1 305	1 440	1 550	1 700	1 835
Dimension (mm)	Length Flywheel-Bed edge			9 800	11 200	12 800	14 400	17 360	18 950	20 550	22 150	23 750
	Dismantling Height			14 050 (Normal)				12 950 (Minimum)				

L80 MC/MCE Stroke 2592 mm; Bore 800 mm

	Speed rev/min	Pb (bar)	Power	Number of cylinders 4	5	6	7	8	9	10	11	12
Nominal MCR												
MC	83	15.0	kW	10 840	13 550	16 260	18 970	21 680	24 390	27 100	29 810	32 520
			BHP	14 760	18 450	22 140	25 830	29 520	33 210	36 900	40 590	44 280
MCE	83	13.0	kW	9 400	11 750	14 100	16 450	18 800	21 150	23 500	25 850	28 200
			BHP	12 800	16 000	19 200	22 400	25 600	28 800	32 000	35 200	38 400
Dry mass (tons)				550	630	725	825	925	1 020	1 100	1 205	1 300
Dimension (mm)	Length Flywheel-Bed edge			8 525	9 945	11 365	12 785	14 205	16 825	18 245	19 665	21 085
	Dismantling Height			12 500 (Normal)				11 500 (Minimum)				

L70 MC/MCE Stroke 2268 mm; Bore 700 mm

	Speed rev/min	Pb (bar)	Power	Number of cylinders 4	5	6	7	8
Nominal MCR								
MC	95	15.0	kW	8 320	10 400	12 480	14 560	16 640
			BHP	11 320	14 150	16 980	19 810	22 640
MCE	95	13.0	kW	7 200	9 000	10 800	12 600	14 400
			BHP	9 800	12 250	14 700	17 150	19 600
Dry mass (tons)				390	445	515	585	655
Dimension (mm)	Length Flywheel-Bed edge			7 465	8 710	9 955	11 200	12 445
	Dismantling Height			10 950 (Normal)			10 100 (Minimum)	

L60 MC/MCE Stroke 1944 mm; Bore 600 mm

	Speed rev/min	Pb (bar)	Power	Number of cylinders 4	5	6	7	8
Nominal MCR								
MC	111	15.0	kW	6 120	7 650	9 180	10 710	12 240
			BHP	8 320	10 400	12 480	14 560	16 640
MCE	111	13.0	kW	5 320	6 650	7 980	9 310	10 640
			BHP	7 200	9 000	10 800	12 600	14 400
Dry mass (tons)				260	295	340	390	435
Dimension (mm)	Length Flywheel-Bed edge			6 005	7 070	8 135	9 200	10 265
	Dismantling Height			9 400 (Normal)			8 700 (Minimum)	

L50 MC/MCE Stroke 1620 mm; Bore 500 mm

	Speed rev/min	Pb (bar)	Power	Number of cylinders 4	5	6	7	8
Nominal MCR								
MC	133	15.0	kW	4 240	5 300	6 360	7 420	8 480
			BHP	5 760	7 200	8 640	10 080	11 520
MCE	133	13.0	kW	3 680	4 600	5 520	6 440	7 360
			BHP	5 000	6 250	7 500	8 750	10 000
Dry mass (tons)				160	185	210	240	270
Dimension (mm)	Length Flywheel-Bed edge			4 940	5 830	6 720	7 610	8 500
	Dismantling Height			7 850 (Normal)			7 200 (Minimum)	

L35 MC/MCE Stroke 1050 mm; Bore 350 mm

	Speed rev/min	Pb (bar)	Power	Number of cylinders 4	5	6	7	8	9
Nominal MCR									
MC	200	14.8	kW	2 000	2 500	3 000	3 500	4 000	4 500
			BHP	2 720	3 400	4 080	4 760	5 440	6 120
MCE	200	12.8	kW	1 720	2 150	2 580	3 010	3 440	3 870
			BHP	2 340	2 925	3 510	4 095	4 680	5 265
Dry mass (tons)				51	58	66	76	84	93
Dimension (mm)	Length Flywheel-Bed edge			4 130	4 730	5 330	5 930	6 530	7 130
	Dismantling Height			5 480 (Normal)			5 180 (Minimum)		

Specific Fuel Oil Consumption (g/BHP

Engine Type	Layout MCR	r/min	SFOC Layout MCR P and F	Minimum Specific Fuel Consumption at Part Load
L 90 MC	P1) F2)	74 60–74	126 123	124 121
L 90 MCE	P1) F2)	74 60–74	123 120	121 118

1) P: Designates Power-maximized MCR lay-out point.
2) F: Designates Fuel-minimized MCR lay-out point.

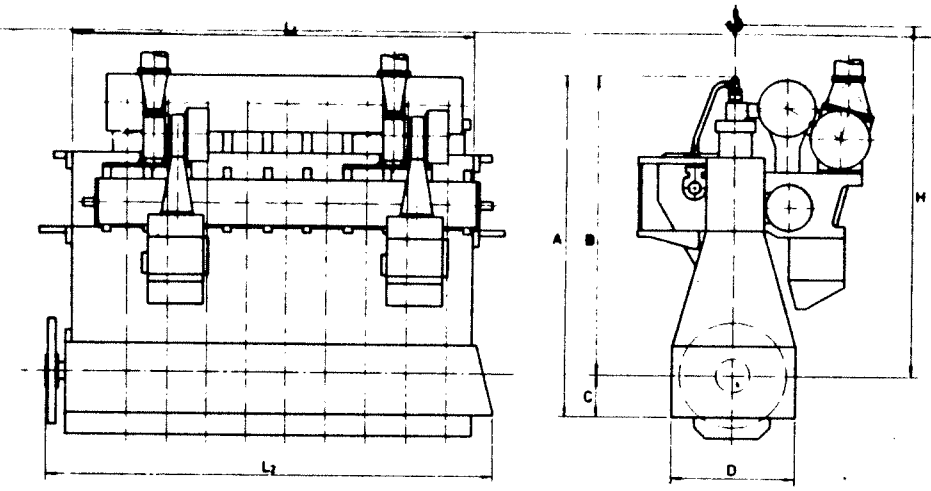
Engine Type	Layout MCR	r/min	SFOC Layout MCR P and F	Minimum Specific Fuel Consumption at Part Load
L 80 MC	P1) F2)	83 68–83	127 124	125 122
L 80 MCE	P1) F2)	83 68–83	124 121	122 119

Engine Type	Layout MCR	r/min	SFOC Layout MCR P and F	Minimum Specific Fuel Consumption at Part Load
L 70 MC	P1) F2)	95 77–95	127 124	125 122
L 70 MCE	P1) F2)	95 77–95	124 121	122 119

Engine Type	Layout MCR	r/min	SFOC Layout MCR P and F	Minimum Specific Fuel Consumption at Part Load
L 60 MC	P1) F2)	111 90–111	128 125	126 123
L 60 MCE	P1) F2)	111 90–111	125 122	123 120

Engine Type	Layout MCR	r/min	SFOC Layout MCR P and F	Minimum Specific Fuel Consumption at Part Load
L 50 MC	P1) F2)	133 108–133	129 126	127 124
L 50 MCE	P1) F2)	133 108–133	126 123	124 121

Engine Type	Layout MCR	r/min	SFOC Layout MCR P and F	Minimum Specific Fuel Consumption at Part Load
L 35 MC	P1) F2)	200 163–200	132 130	131 129
L 35 MCE	P1) F2)	200 163–200	130 128	129 127



L 90 MC/MCE

MCR: Maximum Continuous Rating, nominal

OR: Overload Rating corresponding to 110% of the power at MCR.
This may be permitted for a limited period of one hour every 12 hours.

The below engine powers remain valid up to tropical conditions at sea level, i.e. ambient air temp. at 45°C, ambient air pressure 1000 mbar, charge air coolant temp. at 32°C.

No. of cyl.	Nominal MCR				Dry mass tons	Dimension mm	
	MC		MCE				
	kW	BHP	kW	BHP		L_1	L_2
4	13760	18720	11920	16240	775	7910	9600
5	17200	23400	14900	20300	890	9510	11200
6	20640	28080	17880	24360	1025	11110	12800
7	24080	32760	20860	28420	1165	11520	14400
8	27520	37440	23840	32480	1305	14470	17350
9	30960	42120	26820	36540	1440	16070	18950
10	34400	46800	29800	40600	1550	17670	20550
11	37840	51480	32780	44660	1700	19270	22150
12	41280	56160	35760	48720	1835	20870	23750

Irrespective of number of cylinders			
	r/min	p_b (bar)	
		MC	MCE
MCR	74	15.0	13.0
OR	77	16.0	13.8
Bore × Stroke		900 × 2925 mm	
A mm	B mm	C mm	D mm
13620	11930	1690	4840

Dismantling height H = 14050 mm (Normal), 12950 mm (Minimum). When using a B&W double-jib-crane H = 12200 mm.

Specific Fuel Oil Consumption 3)						Minimum Specific Fuel Consumption at Part Load
Engine Type	Layout MCR	r/min	Output kW/cyl.	Output BHP/cyl.	SFOC layout MCR P and F	
L 90 MC	P1) F2)	74	3440 2370-2930	4680 3230-3980	126 123	124 121
L 90 MCE	P1) F2)	74 60-74	2980 2060-2540	4060 2800-3450	123 120	121 118

1) P: Designates Power-maximized MCR lay-out point.
2) F: Designates Fuel-minimized MCR lay-out point.
3) SFOC: Specific Fuel Oil Consumption at ISO ambient reference conditions and a fuel with a lower calorific value of 10,200 kcal/kg (42,700 kJ/kg).

Lubricating Oil consumption

Lub oil 6-7 kg/cyl. per 24 h
Cyl. lub oil 0.5 g/BHPh

Reference Conditions (ISO ambient)

Values mentioned apply to brake effect and for:
Blower inlet temp. 27°C
Blower inlet press. 1000 mbar
Charge air coolant temp. 27°C

Fuel oil lower calorific value 42 700 kJ/kg.

Guarantee

To arrive at B&W guaranteed specific fuel consumption at any specified MCR within the limits of the layout diagram (for any reference condition chosen): Add 3% to the expected specific fuel oil consumption figures either stated or calculated according to B&W Conversion Formulas.

Super-long Stroke Diesel Engines

HITACHI-SULZER

RTA

Main Engine Data

RTA84 Stroke 3400 mm; Bore 840 mm

	Speed rev/min	Cm m/s	m.e.p bar	m.e.p. kg/cm²	Power	Number of cylinders 4	5	6	7	8	9	10	12
ENGINE MAXIMUM CONTINUOUS RATING													
R1	87	6.96	15.36	15.67	kW	11 840	14 800	17 760	20 720	23 680	26 640	29 600	35 520
					BHP	16 120	20 150	24 180	28 210	32 240	36 270	40 300	48 360
ECONOMY RATINGS													
R2	82	6.56	13.64	13.90	kW	9 920	12 400	14 880	17 360	19 840	22 320	24 800	29 760
					BHP	13 480	16 850	20 220	23 580	26 950	30.330	33 700	40 440
R3	74	5.92	15.30	15.59	kW	10 040	12 550	15 060	17 570	20 080	22 590	25 100	30 120
					BHP	13 640	17 050	20 460	23 870	27 280	30 690	34 100	40 920
R4	70	5.60	13.53	13.78	kW	8 400	10 500	12 600	14 700	16 800	18 900	21 000	25 200
					BHP	11 400	14 250	17 100	19 950	22 800	25 650	28 500	34 200

RTA76 Stroke 2200 mm; Bore 760 mm

	Speed rev/min	Cm m/s	m.e.p. bar	m.e.p. kg/cm²	Power	Number of cylinders 4	5	6	7	8	9	10	12
ENGINE MAXIMUM CONTINUOUS RATING													
R1	95	6.97	15.31	15.62	kW	9 680	12 100	14 520	16 940	19 360	21 780	24 200	29 040
					BHP	13 160	16 450	19 740	23 030	26 320	29 610	32 900	39 480
ECONOMY RATINGS													
R2	90	6.60	13.56	13.83	kW	8 120	10 150	12 180	14 210	16 240	18 270	20 300	24 360
					BHP	11 040	13 800	16 560	19 320	22 080	24 840	27 600	33 120
R3	81	5.94	15.22	15.53	kW	8 200	10 250	12 300	14 350	16 400	18 450	20 500	24 600
					BHP	11 160	13 950	16 740	19 530	22 320	25 110	27 900	33 480
R4	76	5.57	13.61	13.88	kW	6 880	8 600	10 320	12 040	13 760	15 480	17 200	20 640
					BHP	9 360	11 700	14 040	16 380	18 720	21 060	23 400	28 080

RTA68 Stroke 2000 mm; Bore 680 mm

	Speed rev/min	Cm m/s	m.e.p. bar	m.e.p. kg/cm²	Power	Number of cylinders 4	5	6	7	8			
ENGINE MAXIMUM CONTINUOUS RATING													
R1	105	7.00	15.34	15.64	kW	7 800	9 750	11 700	13 650	15 600			
					BHP	10 600	13 250	15 900	18 550	21 200			
ECONOMY RATINGS													
R2	99	6.60	13.60	13.83	kW	6 520	8 150	9 780	11 410	13 040			
					BHP	8 840	11 050	13 260	15 470	17 680			
R3	89	5.92	15.31	15.59	kW	6 600	8 250	9 900	11 550	13 200			
					BHP	8 960	11 200	13 440	15 680	17 920			
R4	84	5.60	13.67	13.94	kW	5 560	6 950	8 340	9 730	11 120			
					BHP	7 560	9 450	11 340	13 230	15 120			

RTA58 Stroke 1760 mm; Bore 580 mm

	Speed rev/min	Cm m/s	m.e.p. bar	m.e.p. kg/cm²	Power	Number of cylinders 4	5	6	7	8	9		
ENGINE MAXIMUM CONTINUOUS RATING													
R1	123	6.97	15.31	15.64	kW	5 640	7 050	8 460	9 870	11 280	12 690		
					BHP	7 680	9 600	11 520	13 440	15 360	17 280		
ECONOMY RATINGS													
R2	116	6.57	13.59	13.82	kW	4 720	5 900	7 080	8 260	9 440	10 620		
					BHP	6 400	8 000	9 600	11 200	12 800	14 400		
R3	105	5.95	15.27	15.55	kW	4 800	6 000	7 200	8 400	9 600	10 800		
					BHP	6 520	8 150	9 780	11 410	13 040	14 670		
R4	98	5.55	13.63	13.90	kW	4 000	5 000	6 000	7 000	8 000	9 000		
					BHP	5 440	6 800	8 160	9 520	10 880	12 240		

RTA48 Stroke 1400 mm; Bore 480 mm

	Speed rev/min	Cm m/s	m.e.p bar	m.e.p. kg/cm²	Power	Number of cylinders 4	5	6	7	8	9		
ENGINE MAXIMUM CONTINUOUS RATING													
R1	150	7.00	15.32	15.63	kW	3 880	4 850	5 820	6 790	7 760	8 730		
					BHP	5 280	6 600	7 920	9 240	10 560	11 880		
ECONOMY RATINGS													
R2	141	6.58	13.61	13.86	kW	3 240	4 050	4 860	5 670	6 480	7 290		
					BHP	4 400	5 500	6 600	7 700	8 800	9 900		
R3	127	5.93	15.29	15.52	kW	3 280	4 100	4 920	5 740	6 560	7 380		
					BHP	4 440	5 550	6 660	7 770	8 880	9 990		
R4	120	5.60	13.62	13.91	kW	2 760	3 450	4 140	4 830	5 520	6 210		
					BHP	3 760	4 700	5 640	6 580	7 520	8 460		

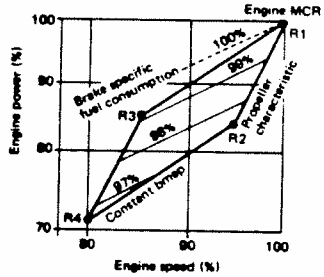
RTA38 Stroke 1100 mm; Bore 380 mm

	Speed rev/min	Cm m/s	m.e.p. bar	m.e.p. kg/cm²	Power	Number of cylinders 4	5	6	7	8	9		
ENGINE MAXIMUM CONTINUOUS RATING													
R1	190	6.97	15.44	15.76	kW	2 440	3 050	3 660	4 270	4 880	5 490		
					BHP	3 320	4 150	4 980	5 810	6 640	7 470		
ECONOMY RATINGS													
R2	179	6.56	13.70	13.90	kW	2 040	2 550	3 060	3 570	4 080	4 590		
					BHP	2 760	3 450	4 140	4 830	5 520	6 210		
R3	162	5.94	15.44	15.81	kW	2 080	2 600	3 120	3 640	4 160	4 680		
					BHP	2 840	3 550	4 260	4 970	5 680	6 390		
R4	152	5.57	13.81	13.76	kW	1 720	2 150	2 580	3 010	3 440	3 870		
					BHP	2 320	2 900	3 480	4 060	4 640	5 220		

Specific Fuel Oil Consumption

Engine Type	Standard Ratings	Rating Point	Specific Fuel Consumption +3% 100% P g/kWh	g/BHPh	90% P g/kWh	g/BHPh	85% P g/kWh	g/BHPh
RTA84	Engine Maximum Continuous Rating	R1	173	127	170	125	170	125
	Economy Ratings	R2	169	124	167	123	167	123
		R3	171	126	169	124	169	124
		R4	167	123	166	122	166	122
RTA76	Engine Maximum Continuous Rating	R1	173	127	170	125	170	125
	Economy Ratings	R2	169	124	167	123	167	123
		R3	171	126	169	124	169	124
		R4	167	123	166	122	166	122
RTA68	Engine Maximum Continuous Rating	R1	174	128	171	126	171	126
	Economy Ratings	R2	170	125	169	124	169	124
		R3	173	127	170	125	170	125
		R4	169	124	167	123	167	123
RTA58	Engine Maximum Continuous Rating	R1	175	129	173	127	173	127
	Economy Ratings	R2	171	126	170	125	170	125
		R3	174	128	171	126	171	126
		R4	170	125	169	124	169	124
RTA48	Engine Maximum Continuous Rating	R1	178	131	175	129	175	129
	Economy Ratings	R2	174	128	173	127	173	127
		R3	177	130	174	128	174	128
		R4	173	127	171	126	171	126
RTA38	Engine Maximum Continuous Rating	R1	181	133	178	131	178	131
	Economy Ratings	R2	177	130	175	129	175	129
		R3	179	132	177	130	177	130
		R4	175	129	174	128	174	128

RTA engine layout range
and respective brake specific fuel consumption.



Dimesions of Engines

RTA84 B_1 = 13 900 B_2 = 13 300 B_3 = 12 000

Number of cylinders		4	5	6	7	8	9	10	12
Length	mm	9 080	10 880	12 280	13 880	16 440	18 040	19 640	22 840
Weight	tonnes	600	720	830	960	1 080	1 210	1 340	1 600

RTA76 B_1 = 12 800 B_2 = 12 100 B_3 = 11 100

Number of cylinders		4	5	6	7	8	9	10	12
Length	mm	8 320	9 770	11 220	12 670	14 120	16 450	17 900	20 800
Weight	tonnes	450	550	640	740	830	930	1 030	1 230

RTA68 B_1 = 11 700 B_2 = 11 100 B_3 = 10 300

Number of cylinders		4	5	6	7	8			
Length	mm	7 510	8 800	10 090	11 380	12 670			
Weight	tonnes	340	410	480	550	620			

RTA58 B_1 = 10 180 B_2 = 9 600 B_3 = 8 900

Number of cylinders		4	5	6	7	8	9		
Length	mm	6 570	7 670	8 770	9 870	10 970	12 070		
Weight	tonnes	235	280	320	365	405	450		

RTA48 B_2 = 9 600 B_3 = 8 900

Number of cylinders		4	5	6	7	8	9		
Length	mm	5 620	6 540	7 460	8 380	9 300	10 220		
Weight	tonnes	150	185	215	250	285	315		

RTA38 B_2 = 5 700 B_3 = 5 300

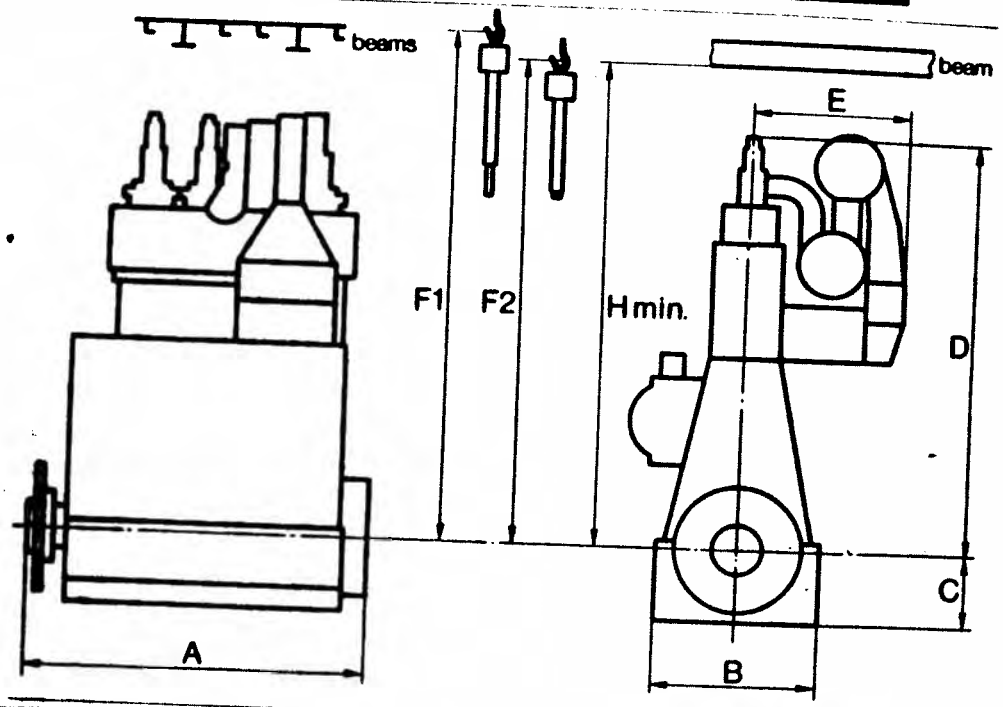
Number of cylinders		4	5	6	7	8	9		
Length	mm	4 280	4 980	5 640	6 320	7 000	7 680		
Weight	tonnes	80	97	112	126	141	156		

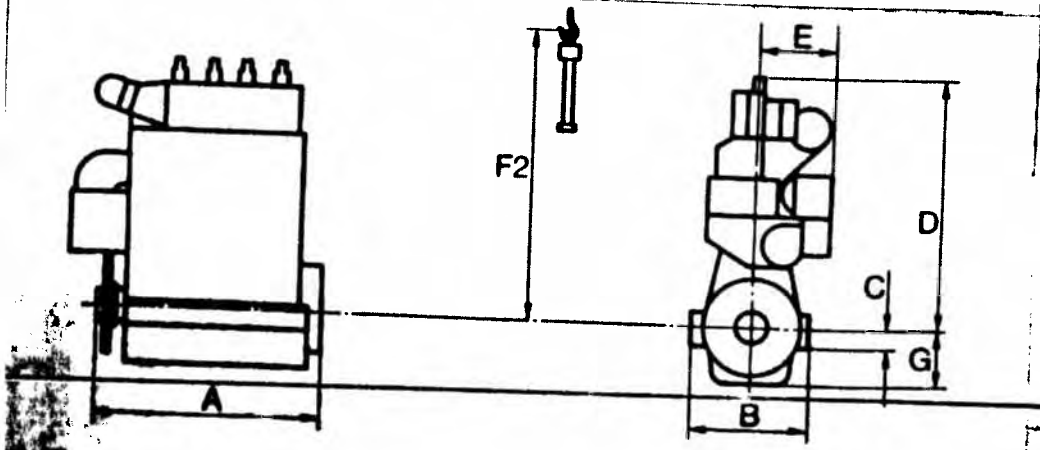
Length : Flywheel – Bed edge
Weight : Without water and oil

Dismantling height { B_1 = normal piston rod; B_2 = short piston rod; B_3 = special crane }

Printed in Japan

RTA dimensions





RTA 84

	Number of cylinders							
	4	6	8	7	8	9	10	12
A	9200	10800	12400	14000	16600	18200	19800	23000
B	4500	4500	4500	4500	4500	4500	4500	4500
C	2050	2050	2050	2050	2050	2050	2050	2050
D	11600	11600	11600	11600	11600	11600	11600	11600
E	4400	4400	4280	4400	4400	4400	4400	4280
F1	13900	13900	13900	13900	13900	13900	13900	13900
F2	13200	13200	13200	13200	13200	13200	13200	13200
H min								
Net engine mass in t without oil/water	590	710	830	950	1110	1235	1350	1590

RTA 76

	Number of cylinders							
	4	6	8	7	8	9	10	12
A	8370	9820	11270	12720	14170	16560	18010	20910
B	4100	4100	4100	4100	4100	4100	4100	4100
C	1820	1820	1820	1820	1820	1820	1820	1820
D	10740	10740	10740	10740	10740	10740	10740	10740
E	4000	4120	4120	4000	4000	4120	4120	4120
F1	12800	12800	12800	12800	12800	12800	12800	12800
F2	12100	12100	12100	12100	12100	12100	12100	12100
H min	12250	12250	12250	12250	12250	12250	12250	12250
Net engine mass in t without oil/water	450	545	640	730	825	950	1045	1230

RTA 68

	Number of cylinders							
	4	6	8	7	8	9	10	12
A	7550	8840	10130	11420	12710			
B	3780	3780	3780	3780	3780			
C	1720	1720	1720	1720	1720			
D	9770	9770	9770	9770	9770			
E	3600	3600	3400	3600	3600			
F1	11700	11700	11700	11700	11700			
F2	11100	11100	11100	11100	11100			
H min.								
Net engine mass in t without oil/water	335	400	470	535	605			

RTA 58

	Number of cylinders							
	4	6	8	7	8	9	10	12
A	6610	7710	8756	9810	11010	12110		
B	3280	3280	3280	3280	3280	3280		
C	1480	1480	1480	1480	1480	1480		
D	8430	8430	8430	8430	8430	8430		
E	3110	3300	3300	3300	3110	3300		
F1	10150	10150	10150	10150	10150	10150		
F2	9600	9600	9600	9600	9600	9600		
H min	9600	9600	9600	9600	9600	9600		
Net engine mass in t without oil/water	220	265	305	350	395	440		

RTA 48

	Number of cylinders							
	4	6	8	7	8	9	10	12
A	5045	5855	7705	8675	9605	10405		
B	2700	2700	2700	2700	2700	2700		
C	860	860	860	860	860	860		
D	6500	6500	6500	6500	6500	6500		
E	1900	1900	1900	1900	1900	1900		
F2	7360	7360	7360	7360	7360	7360		
G	1500	1500	1500	1500	1500	1500		
H min	6000	6000	6000	6000	6000	6000		
Net engine mass in t without oil/water	154	180	206	240	266	290		

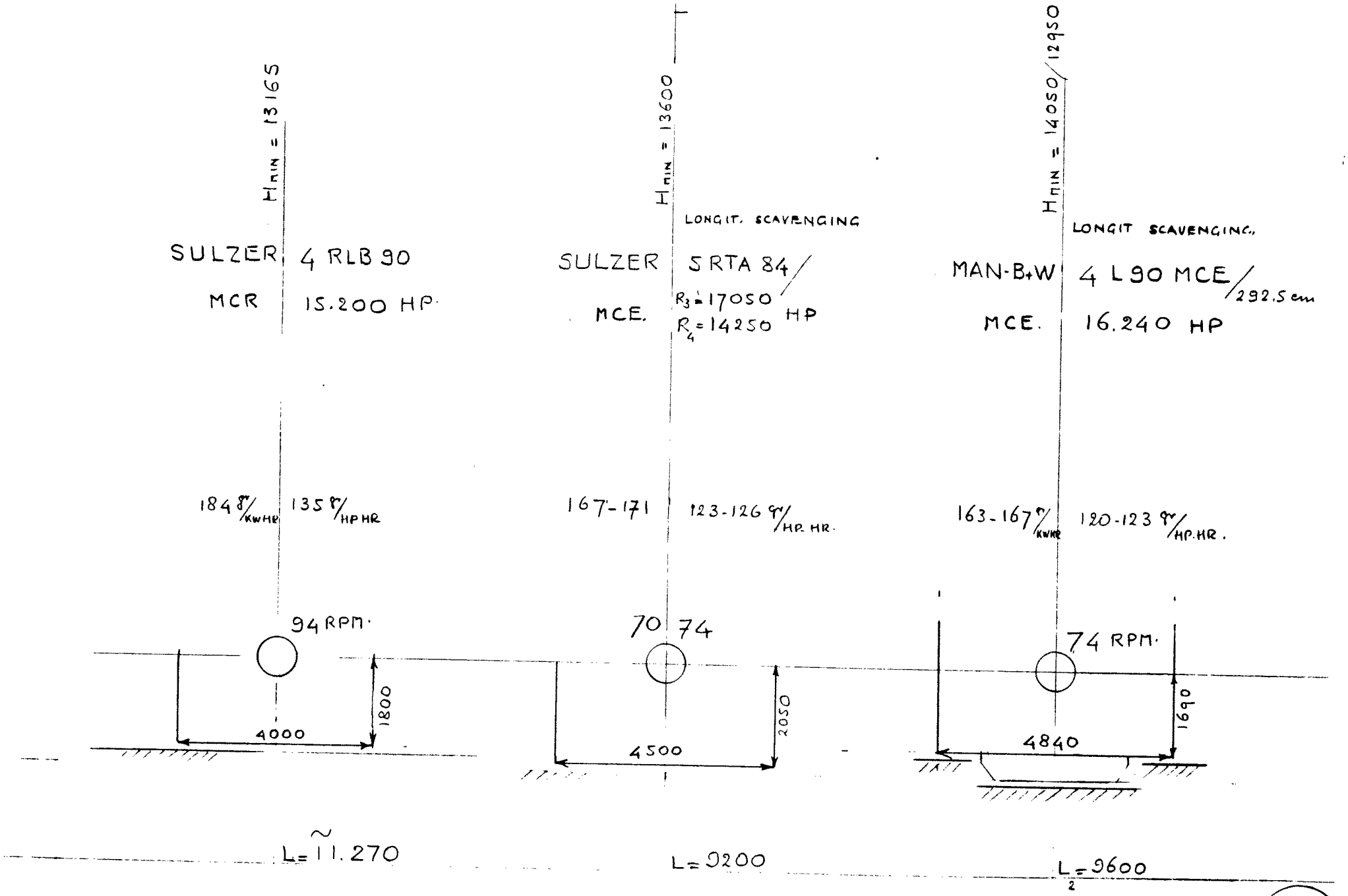
RTA 38

	Number of cylinders							
	4	6	8	7	8	9	10	12
A	4305	4995	5685	6345	7035	7705		
B	2300	2300	2300	2300	2300	2300		
C	400	400	400	400	400	400		
D	5600	5600	5600	5600	5600	5020		
E	1600	1600	1600	1600	1600	1600		
F2	5700	5700	5700	5700	5700	5700		
G	1100	1100	1100	1100	1100	1100		
H min	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
Net engine mass in t without oil/water	72	93	99	110	128	134		

Engine dimensions in mm, not binding

H min. requires special crane and devices, turbocharger module

7-9


H MIN = 13165
SULZER 4 RLB 90
MCR 15.200 HP
184 g/KWHR 135 g/HP HR
94 RPM
4000
1800
L = ~11.270
H MIN = 13600
LONGIT. SCAVENGING
SULZER 5 RTA 84/
R3 = 17050
MCE. R4 = 14250 HP
167-171 123-126 g/HP. HR.
70 74
4500
2050
L = 9200
H MIN = 14050/12950
LONGIT SCAVENGING.
MAN-B+W 4 L 90 MCE/292.5 cm
MCE. 16.240 HP
163-167 g/KWHR 120-123 g/HP.HR.
74 RPM
4840
1690
L2 = 9600

7-11

8

IN CASE MORE D.W. IS NEEDED, EXTERNAL INFLUENCES (PORTS) FORCE US TO DESIGN SHORT BEAMY SHIPS

L/B FROM 6 → 5 → 4

AS BASE:

PRODUCT TANKER "MAASSLUIS" 38000 DWT $L/B = 5.09$

REDUCTION OF LENGTH, INCREASE OF BEAM

DRAFT AND DISPLACEMENT KEPT CONSTANT

L/B 5.09 ···→ 4.0 IN STEPS

THIS PROGRAM FILLS A GAP IN OUR DESIGN-KNOWLEDGE

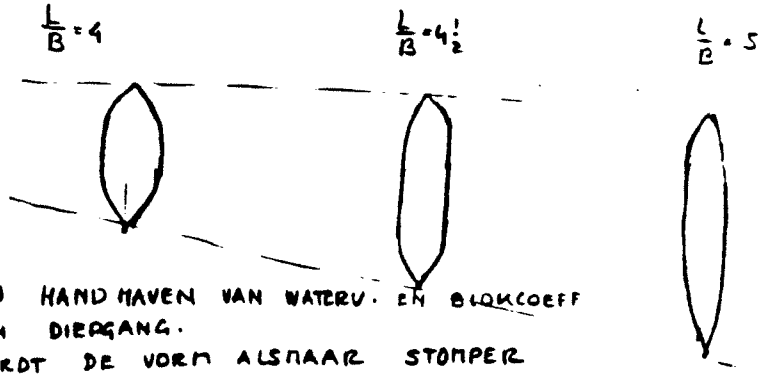
(COASTERS!)

KEPT IDENTICAL:

SHIPS PROPULSION-TAIL, INCL. ENGINE FOUNDATION

KORTE BREDE SCHEPEN.

— VORMONTWIKKELING GAAT DEZE KANT OP, GEDWONGEN DOOR BOUWKOSTEN EN HAVENBEPERKING.
— ONBEKEND TERREIN
— GROTE BEHOEFTE AAN EEN SYSTEMATISCHE BENADERING OP WEERST EN VOORTSTUWING VAN EEN SERIE DIKKE SCHEPEN.



BIJ HAND HAVEN VAN WATERV. EN BLOKCOEFF EN DIEPGANG.
WORDT DE VORM ALSMAAR STOMPER

INSTROOM EN AFSTROOMHOEKEN WORDEN ALSMAAR GROTER.

— LANGS HET ACHTERSCHIP GAAT BOVEN EEN ZEKERE KROMMING VAN DE VORM AFSCHEURING OPTREDEN V/D WATERSTROOM.

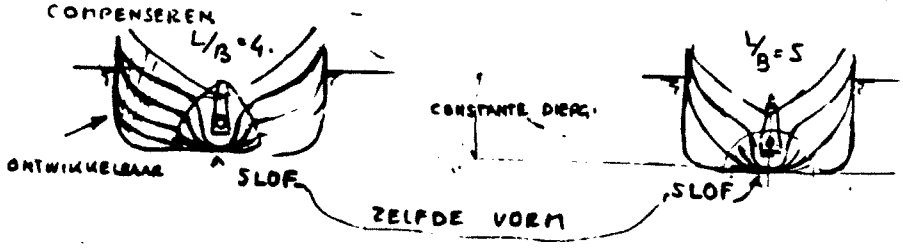
— MET DE NORMALE "STRYKYZER" VORM LOPEN WE VAST MET DRUKKINGSPUNT MOET NAAR VOREN TENEINDE DE KONT VOLDOENDE "SCHERP" TE HOUDEN.

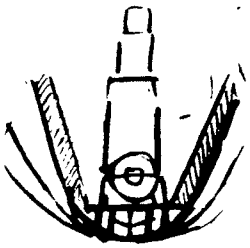
HET ALGEMEEN PLAN VERDRAAGT DIT VOORLIJKE DRUKKPUNT NIET DE BOOT WIL ALSMAAR ACHTEROVER

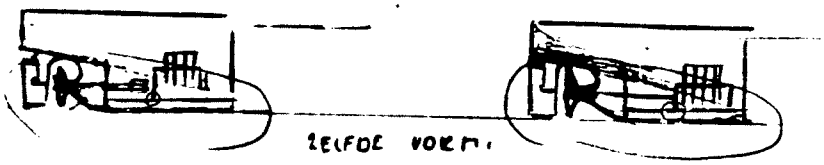
— WAT TE DOEN?

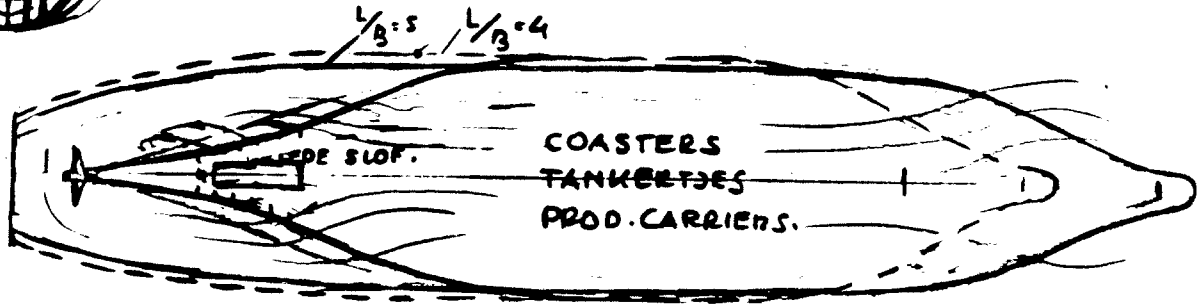
— HOUDT DE "SLOF" VAST, WAARIN DE MOTOR STAAT EN WAARACHTER DE SCHROEF DRAAIT. VAN DAT SCHIP WAARBIJ NOG DUIST GEEN AFSCHEURINGSVERSCHIJNSELEN OPTREDEN

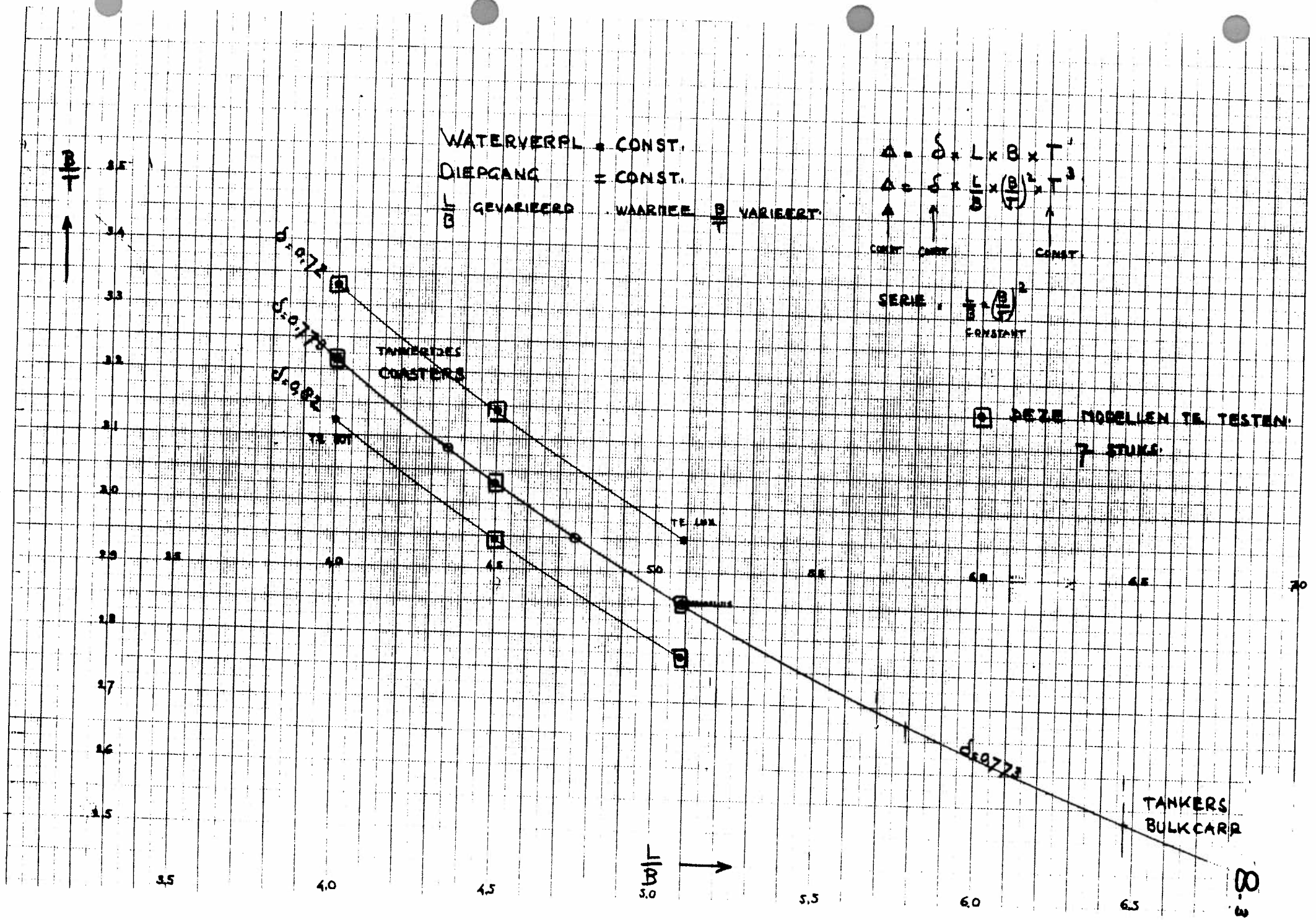
— BIJ DE BREDERE KORTERE SCHEPEN MOET DE ACHTERSCHEPS-VORM BOVEN DEZE SLOF HET VERLIES AAN WATERVERPLAATSING COMPENSEREN.

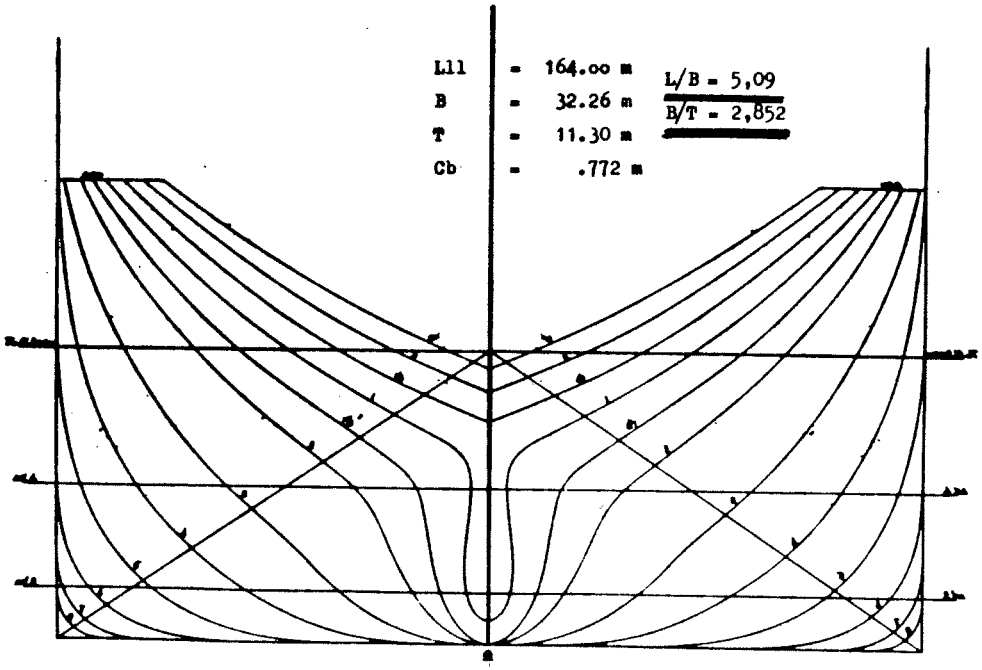









WATERVERPL. = CONST.
DIEPGANG = CONST.
L/B GEVARIEERD WAARMEE B/T VARIEERT.
Δ = δ x L x B x T
Δ = δ x L/B x (B/T)² x T³
CONST CONST CONST
SERIE : L/B x (B/T)² CONSTANT
DEZE MODELLEN TE TESTEN:
7 STUKS.
B/T
L/B
δ=0,72
δ=0,775
δ=0,82
TANKERTJES
COASTERS
TE LAX
TE LAX
δ=0,773
TANKERS
BULKCARR
3,5
4,0
4,5
5,0
5,5
6,0
6,5
2,5
2,6
2,7
2,8
2,9
3,0
3,1
3,2
3,3
3,4
3,5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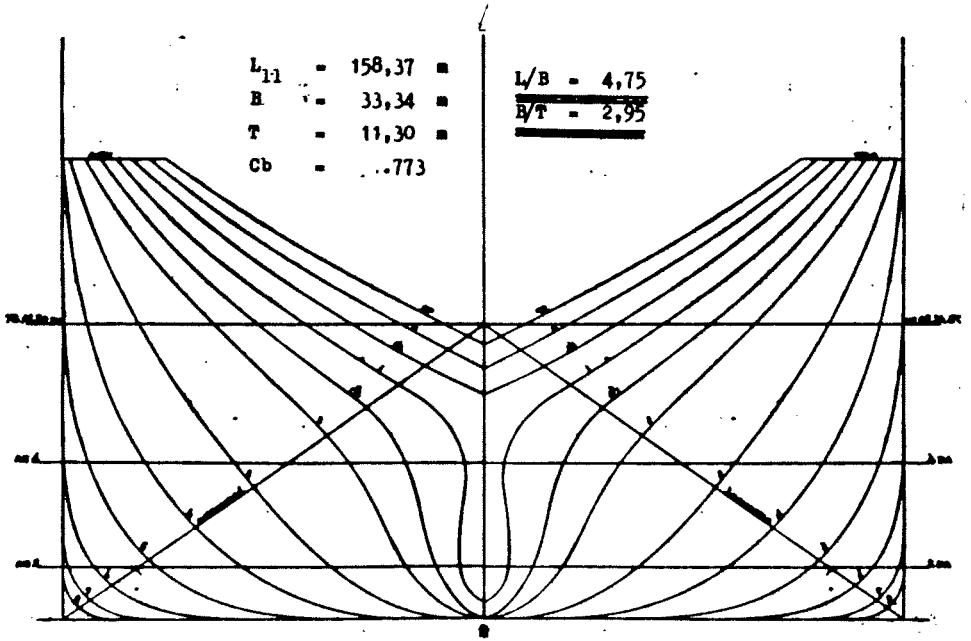
B = 32.26 m
T = 11.30 m
Cb = .772 m
L/B = 5,09
B/T = 2,852


L11 = 158,37 m
B = 33,34 m
T = 11,30 m
Cb = ..773
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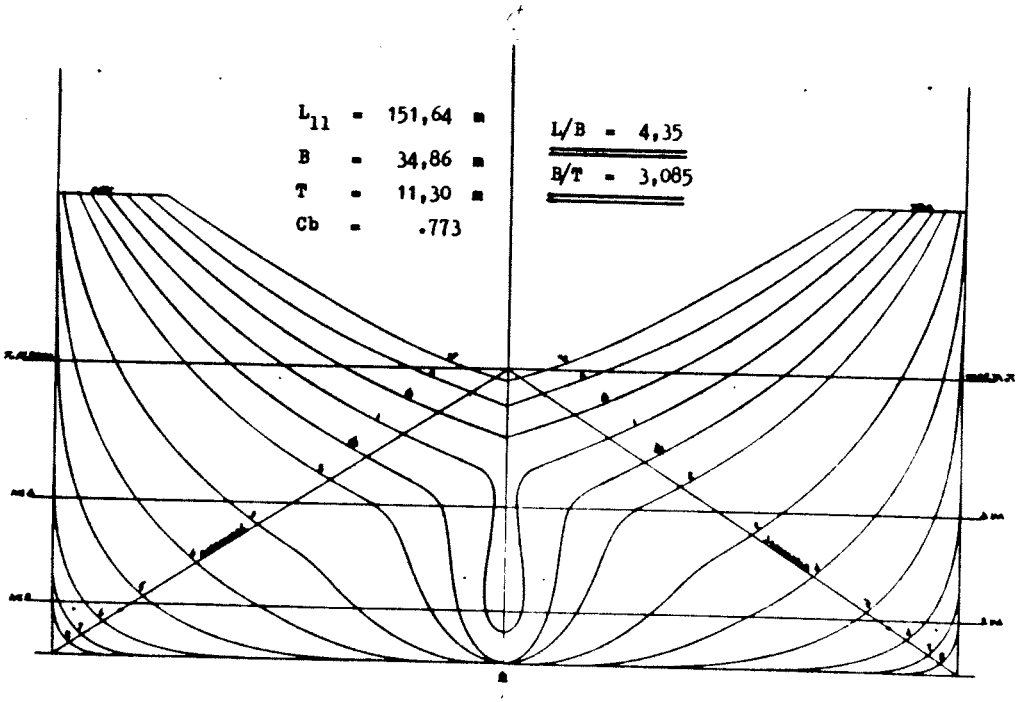
B/T = 2,95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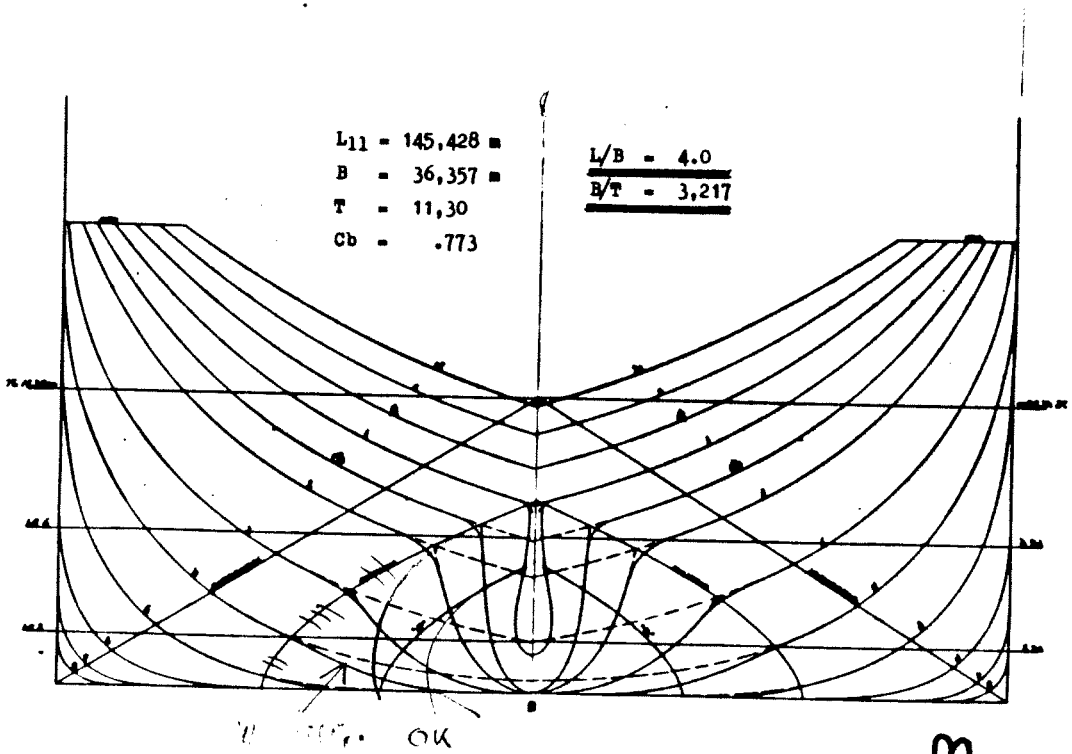
B/T = 3,085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B = 36,357 m
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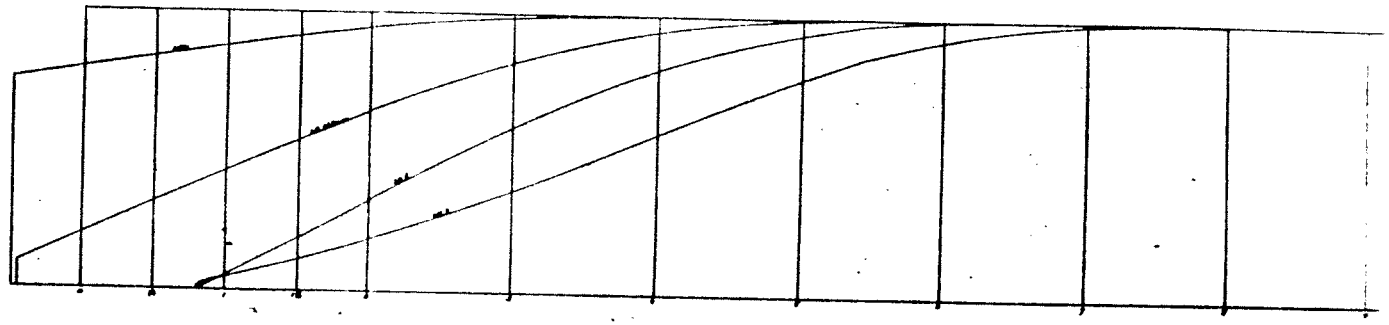
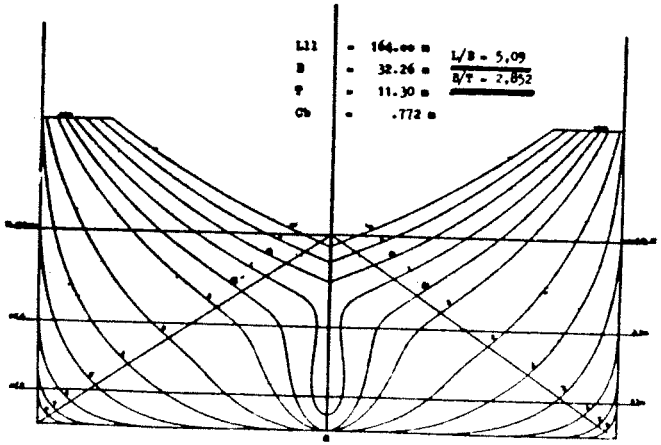
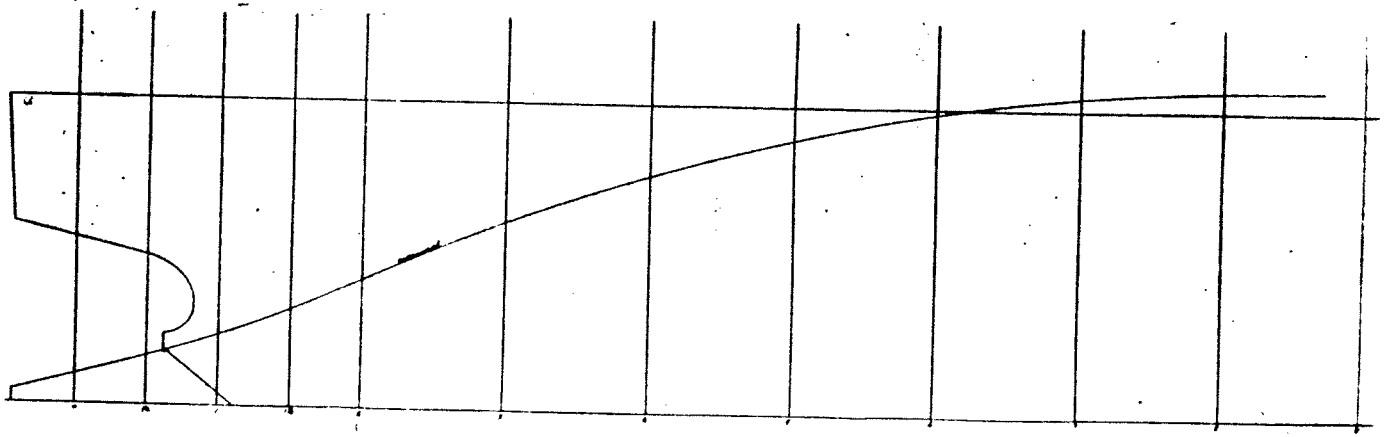
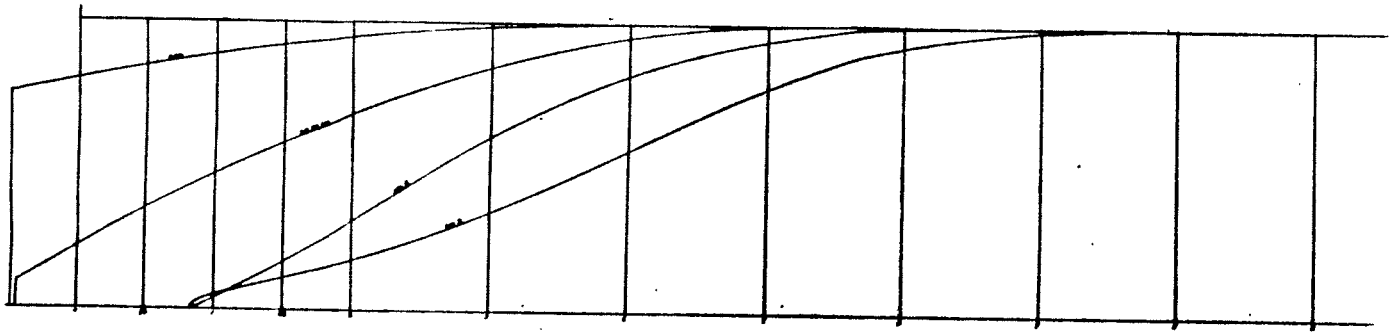
L/B = 4.0
B/T = 3,217
OK



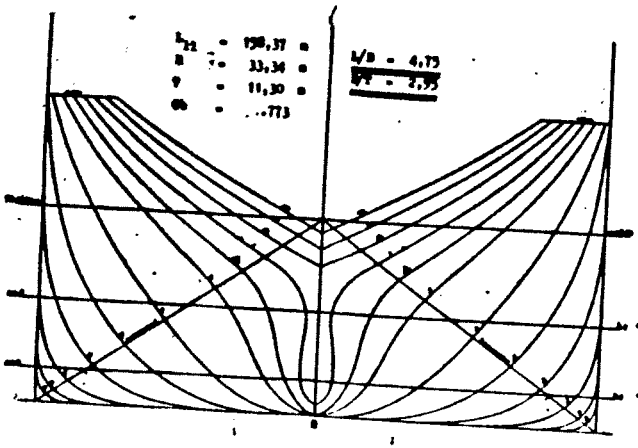
L_{II} /B = 5,09

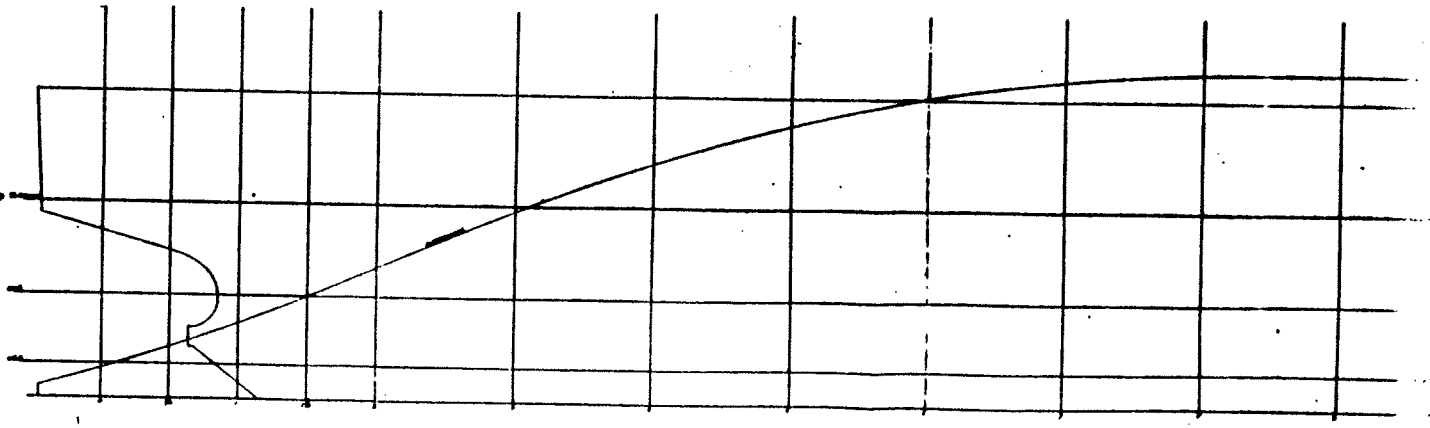


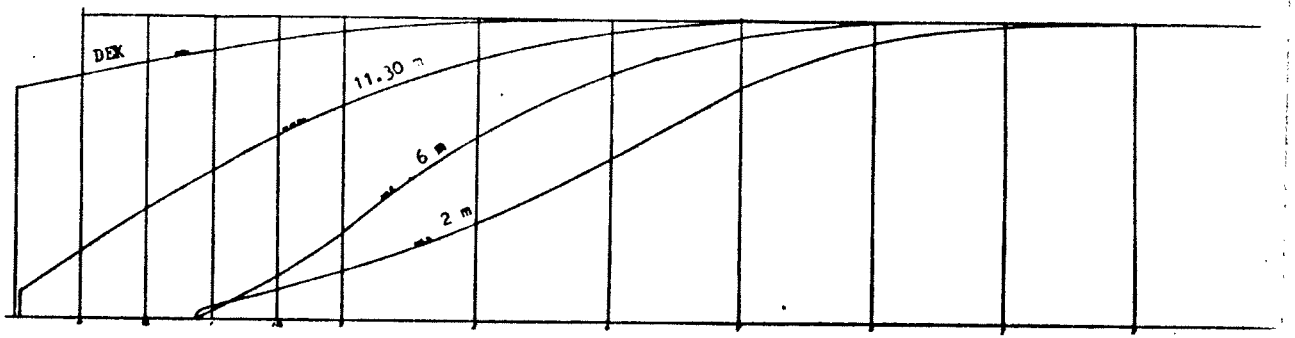




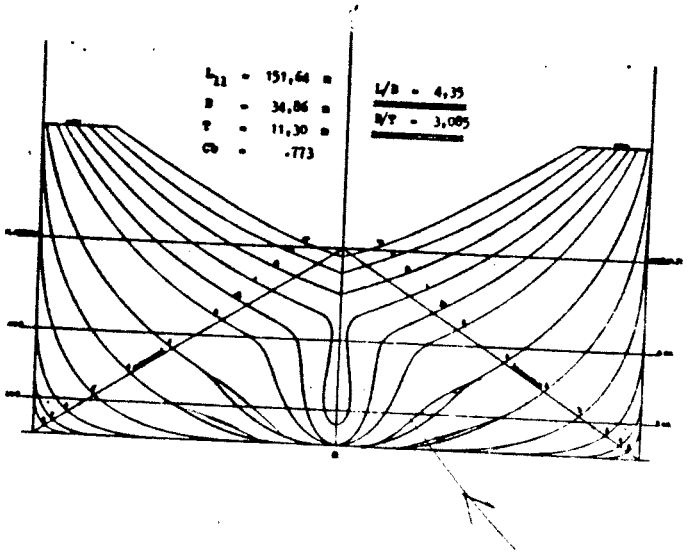
$L_{II}/B = 4,75$

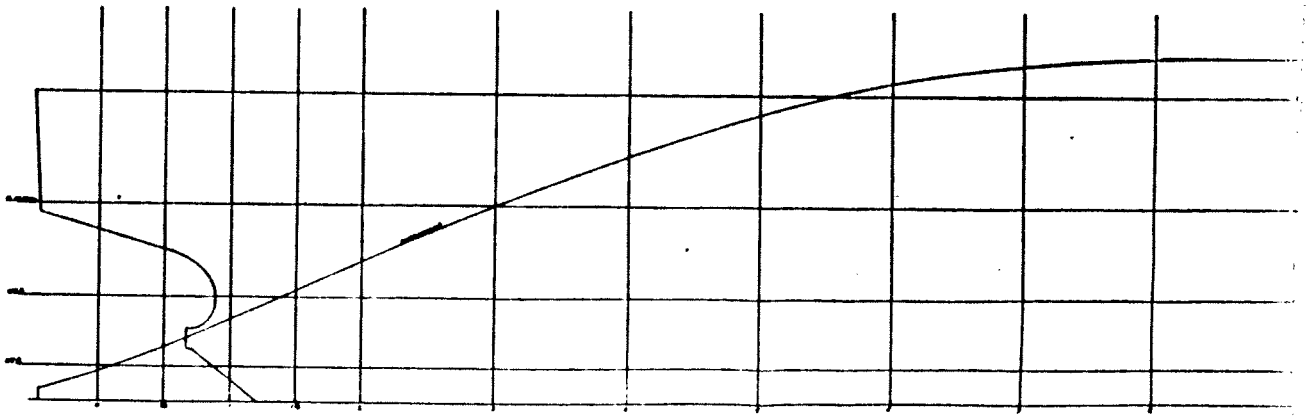






L /B = 4,35





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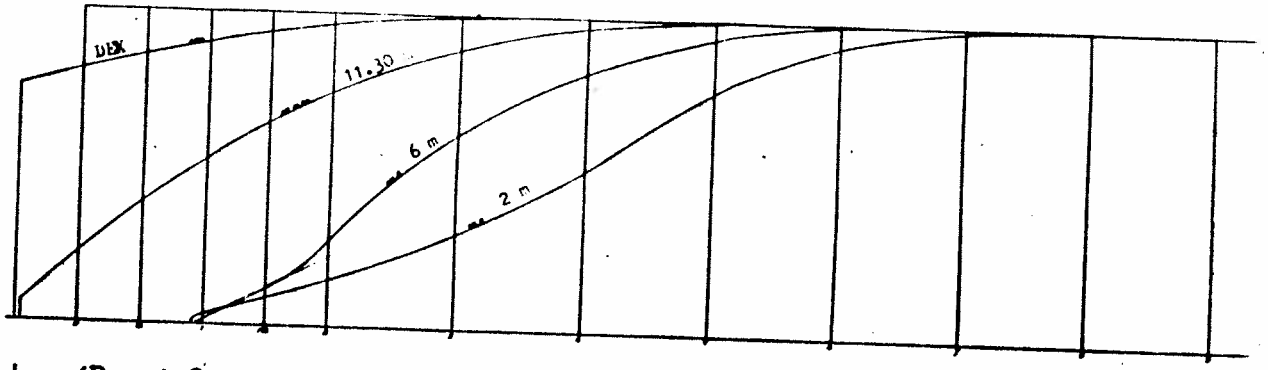
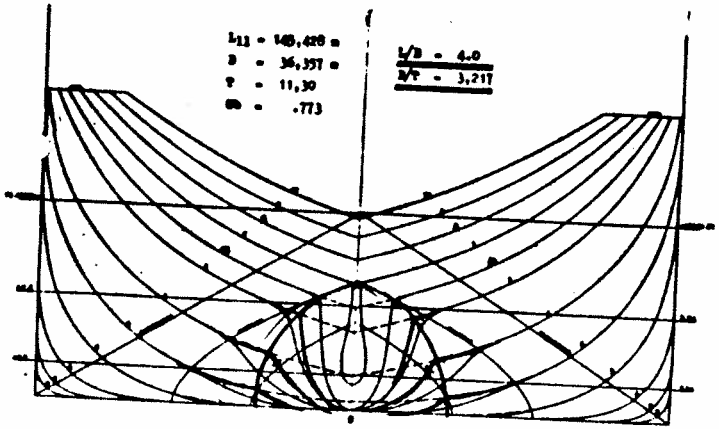
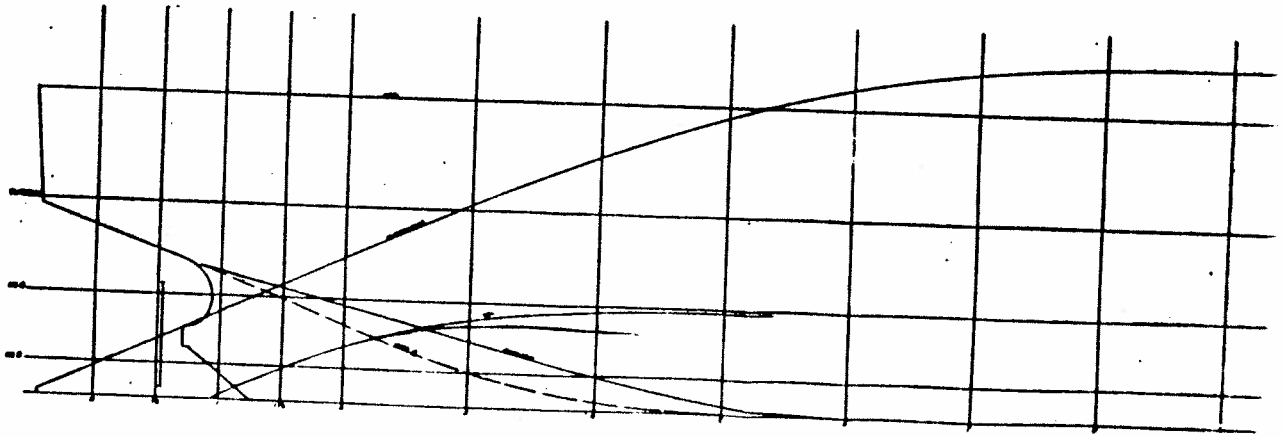


$L_{\parallel}/B = 4.0$





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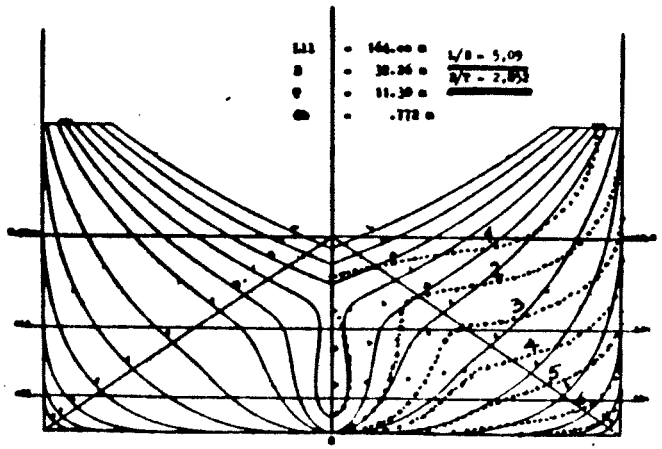
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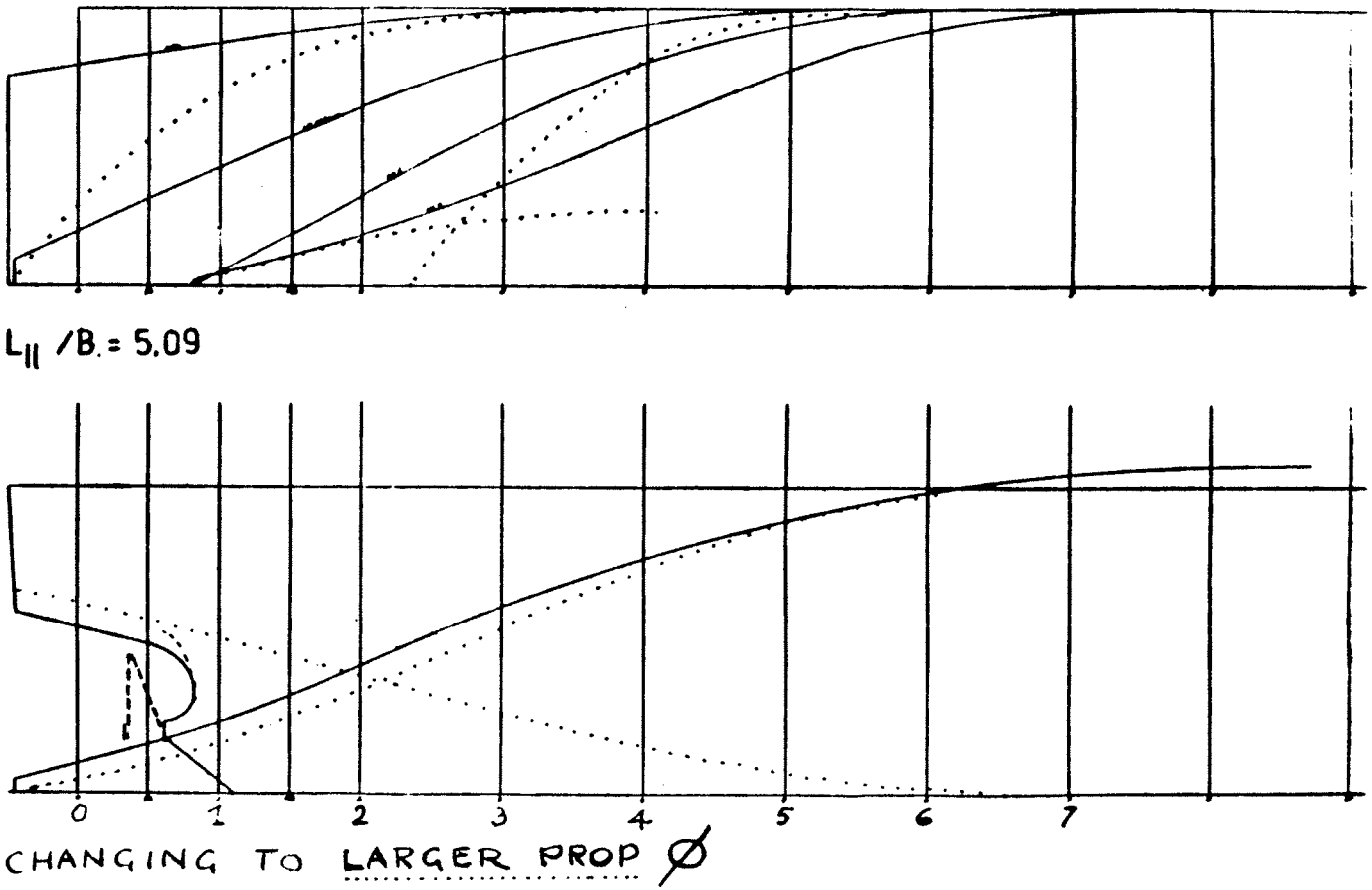
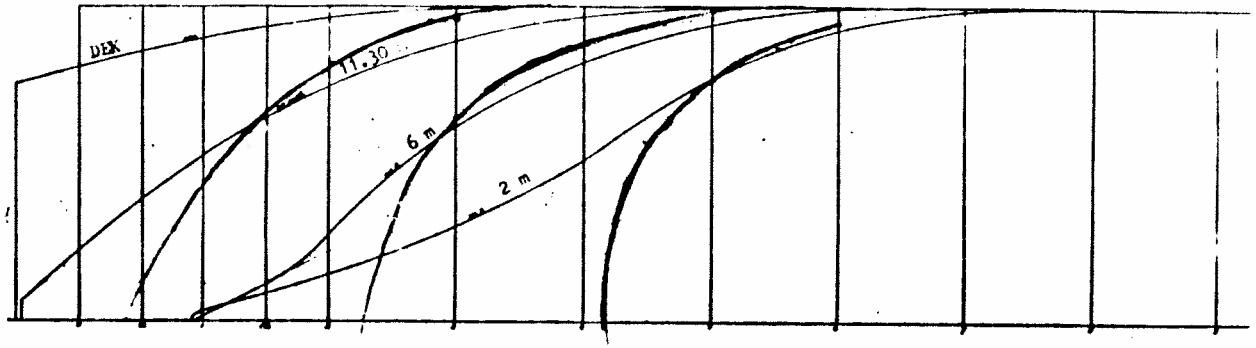
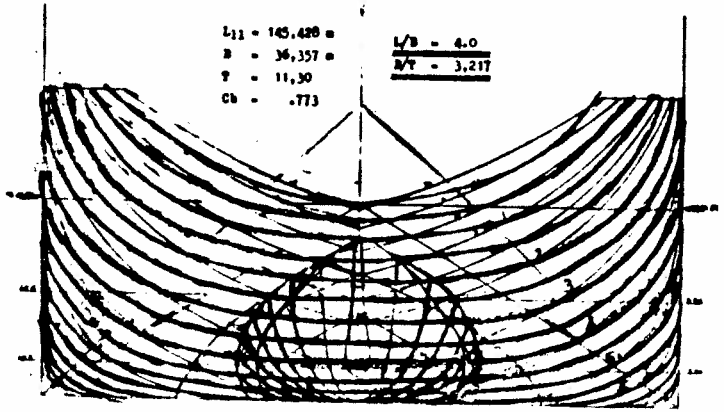
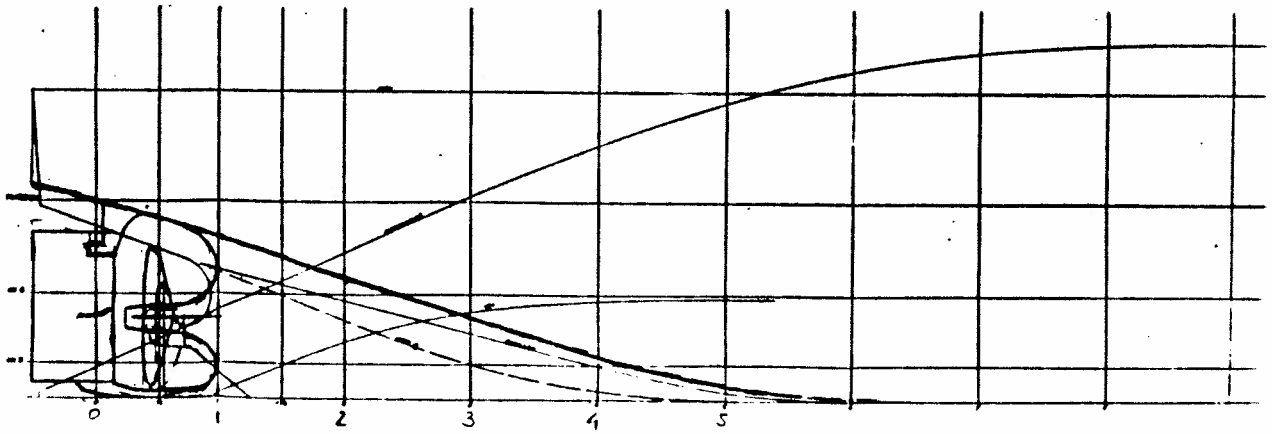
B/T = 2.853


L_{II} /B = 5.09
0
1
2
3
4
5
6
7
CHANGING TO LARGER PROP Ø



L_{II} /B = 4.0 MET GROTERE SCHROEF





B

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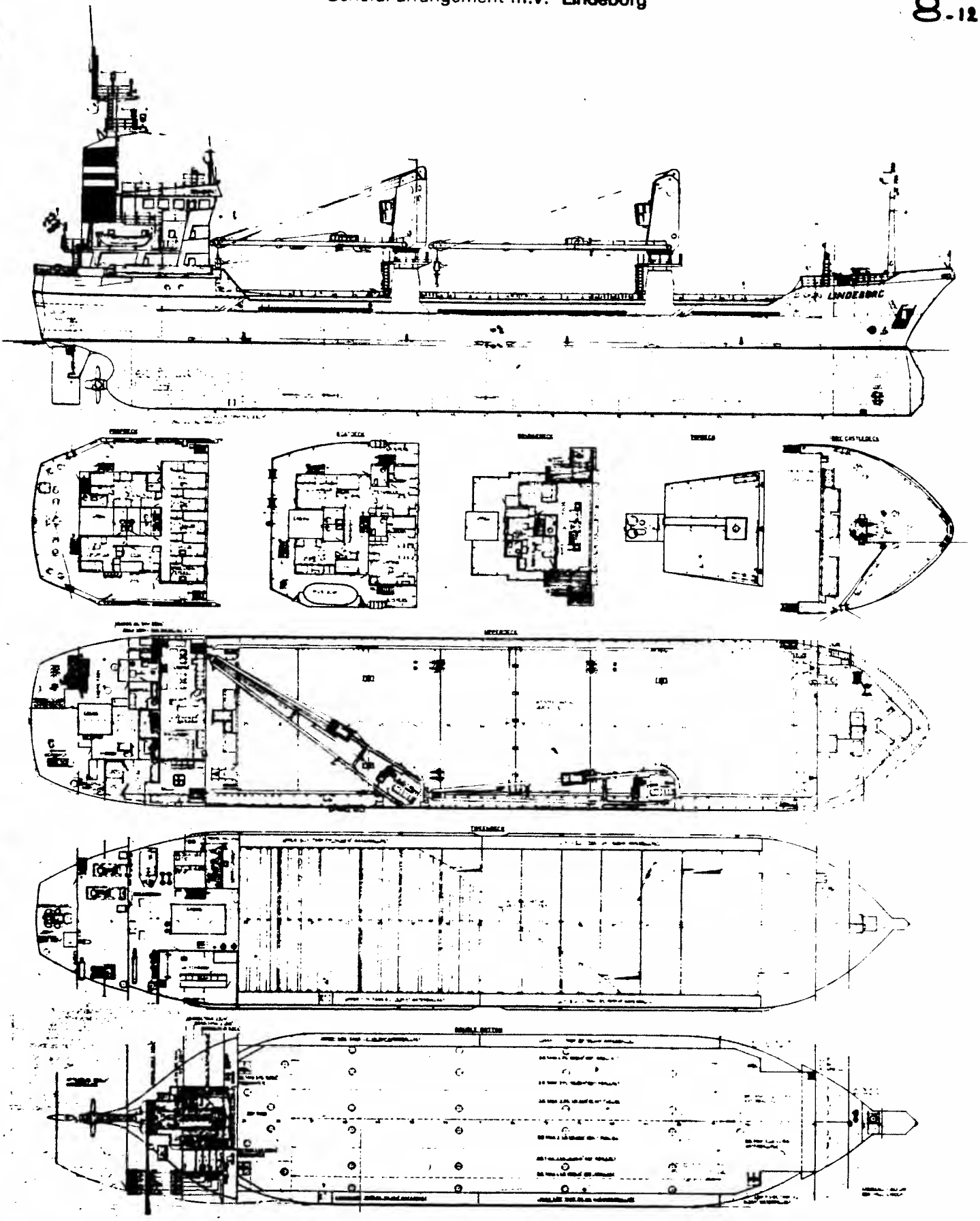
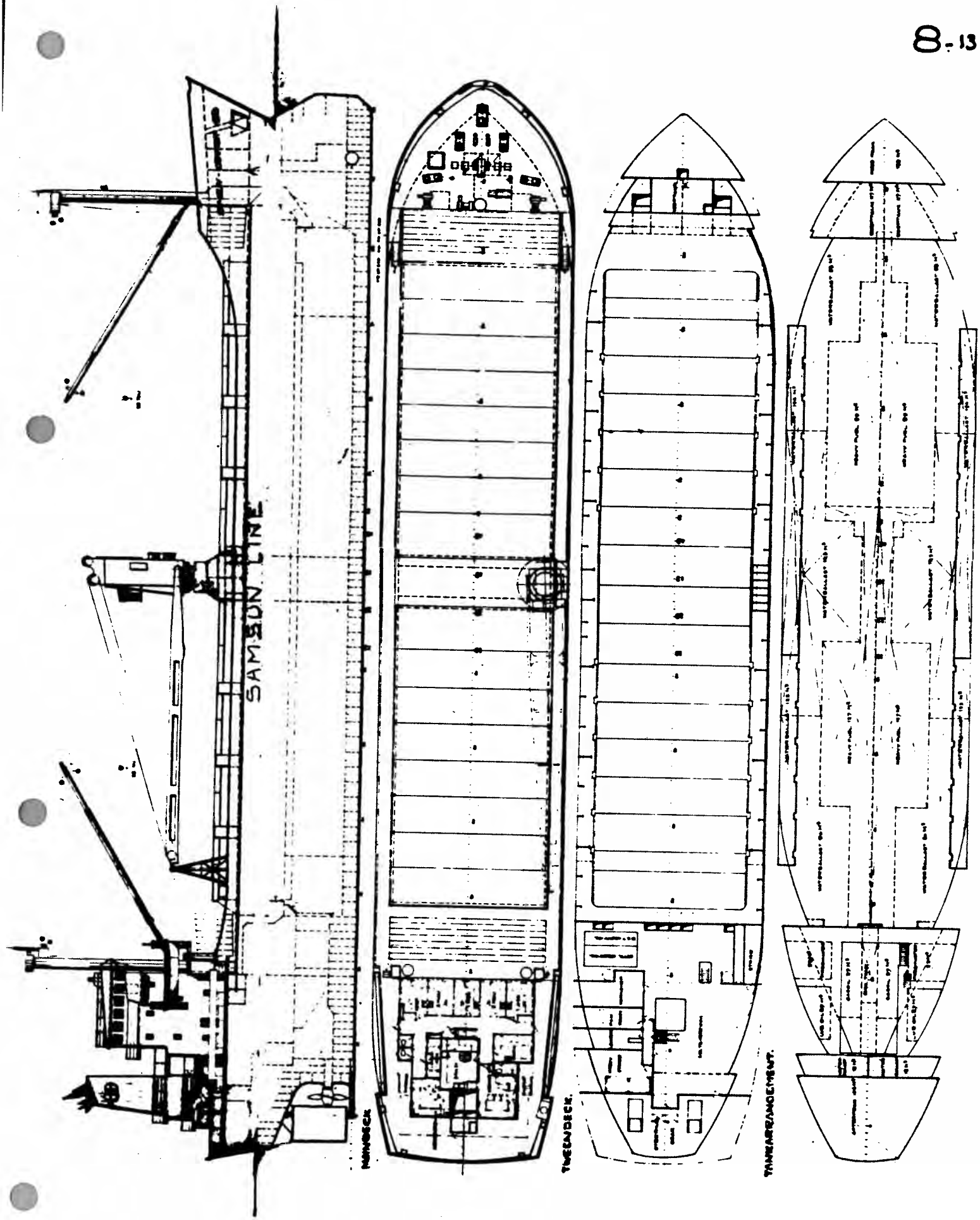
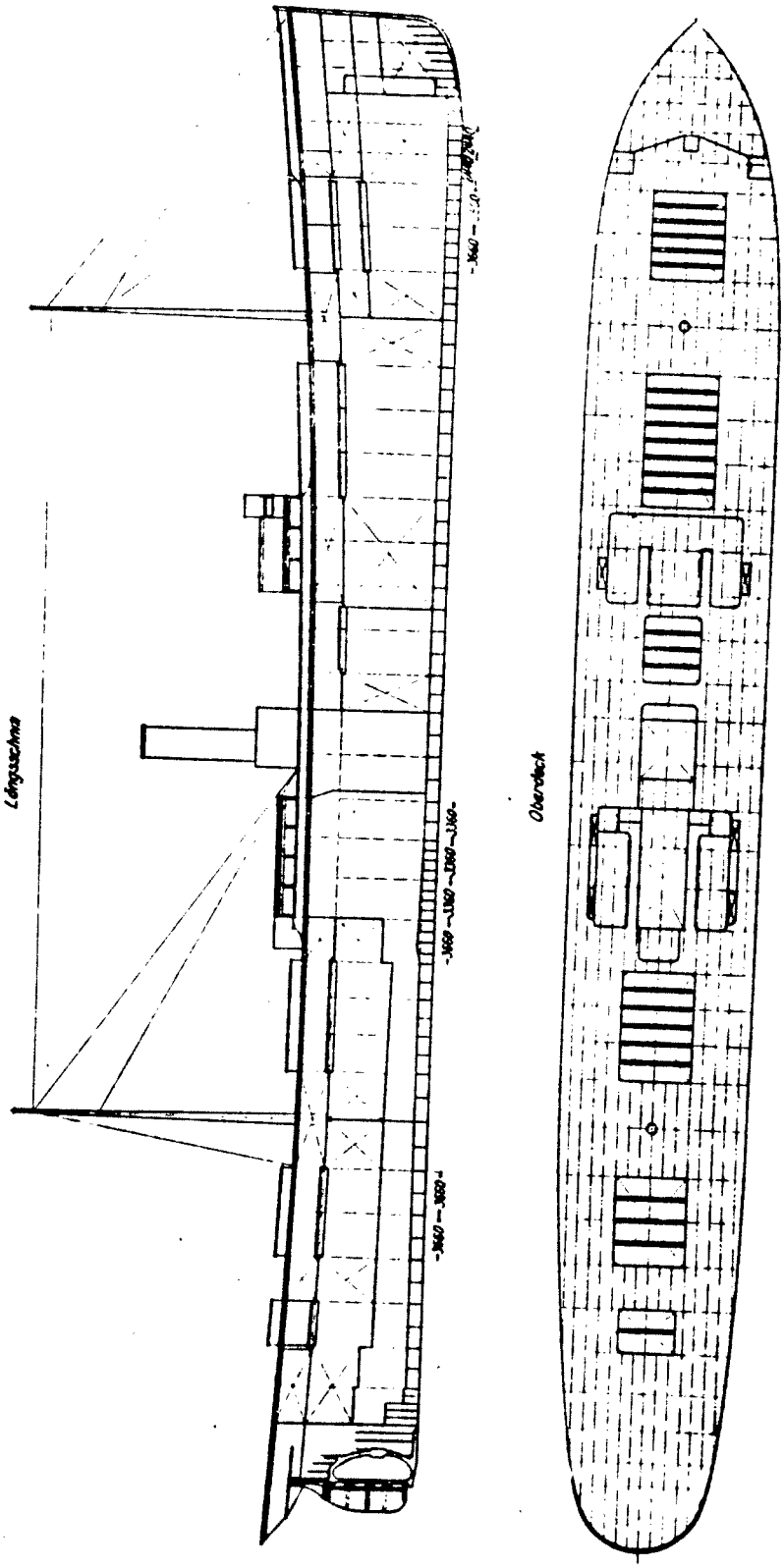
General arrangement m.v. 'Lindeborg'




SAMSON LINE
MAINDECK
TWEENDECK
TANKERAREA/MOEMENT



Längsschnitt und Oberdeck des Schiffes bei Längsspantenbauart (Isherwood).

Fig. 2.

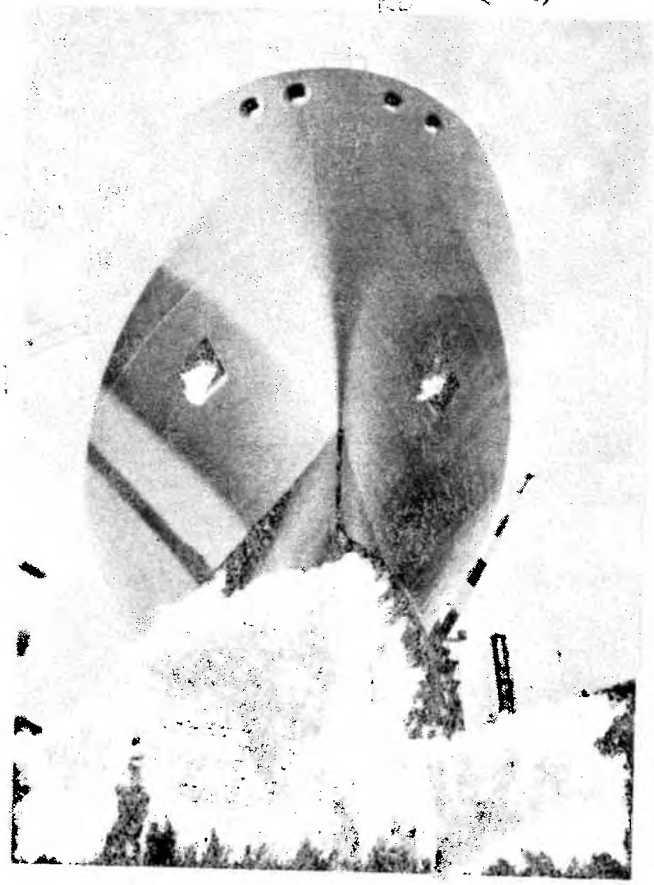
STG 1913

70 YEARS BACK

LINERFLEET DEVELOPMENT

DESIGN CHOICE FROM A LINER SHIPOWNER'S POINT OF VIEW.

NEDLLOYD ROSARIO (NKK)



Nedlloyd Lines

1983

E. Vossnack

C.T.Buys,

S.G.Vriend,

B.Bosschaart,

J.Onassis.

M.Leis

M.v.d. Ree.

Contents of this paper represent the personal views of the authors only.

DAVO 14 029

1 S

FINER HULL

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1.3	Economy of Scale ... limits ?
2.0	SYNOPSIS OF A TYPICAL LINER OPERATOR'S SERVICES - AND VESSEL TYPES USED THEREON (NEDLLOYD - 1982).
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(date 13-10-1983)

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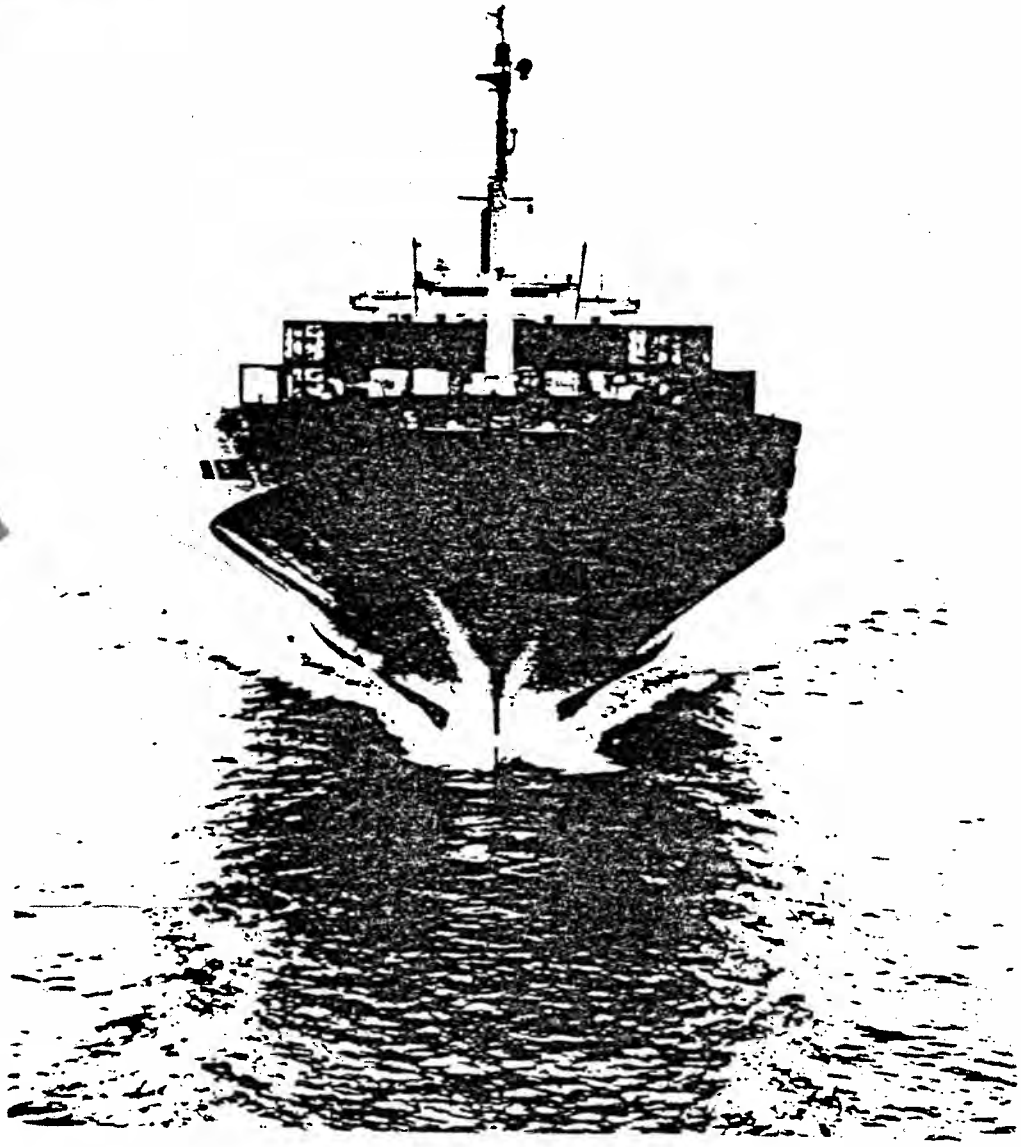
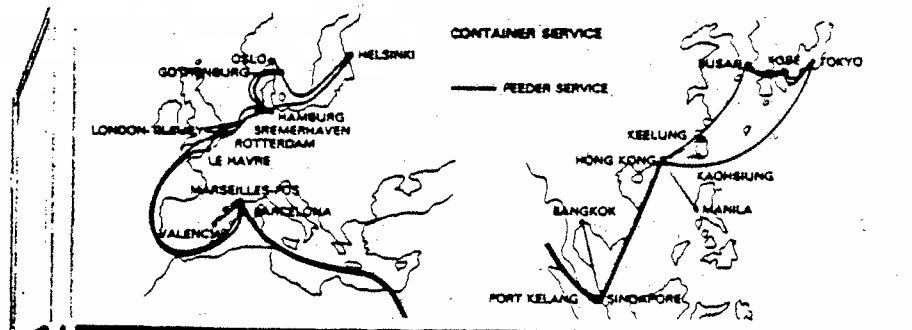
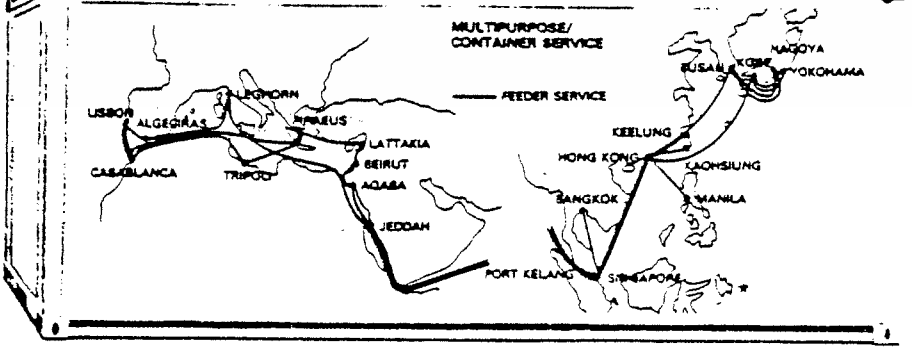



CONTAINER SERVICE
FEEDER SERVICE
OSLO
HELSINKI
GOTHENBURG
HAMBURG
BREMERHAVEN
LONDON
ROTTERDAM
LE HAVRE
MARSEILLES-FOS
BARCELONA
VALENCIA
BUSAN
KOBE
TOKYO
KEELUNG
HONG KONG
KAOHSIUNG
BANGKOK
MANILA
PORT KELANG
SINGAPORE


MULTIPURPOSE/
CONTAINER SERVICE
FEEDER SERVICE
LISBON
ALGECIRAS
LEGHORN
PIRAEUS
LATTAKIA
BEIRUT
AQABA
CASABLANCA
TRIPOLI
JEDDAH
NAGOYA
BUSAN
KOBE
YOKOHAMA
KEELUNG
HONG KONG
KAOHSIUNG
BANGKOK
MANILA
PORT KELANG
SINGAPORE

LINER FLEET DEVELOPMENT

(Subtitle: VESSEL DESIGN TOPICS)

PREAMBLE.

For many years until around 1968 most ocean liner shipping cargo vessels were of rather similar basic design, having tweendecks and cargo gear, the latter consisting in the main of masts and derricks, so as to accommodate as many working gangs as feasible.

Manual labour not being too expensive - at least at one end of the line - this system was workable very well. Trades were rather steady or developed rather steadily.

Then, after the second world war a number of causes - rising standards of living and social awareness, a.o. - brought much pressure to bear on shipowners to look for and find ways to counter strangling labour costs in developed countries and ever-increasing delays in a number of ports in under-developed areas.

In the labour-expensive countries mechanization of cargo handling was developed by utilization of forklift trucks and unit forming (palletisation) of the piece-good cargoes. This was no easy process as stevedoring firms had to invest in forklift trucks and had to pay for keeping these in repair.Also the formation of palletloads was labour-intensive, not to speak yet of the costs of pallets and retrieving same in case of permanent ones.

Effectual consequences of the unitload system for the ships were the introduction of wider hatches ("open" ships, necessitating heavier scantlings) and a shift away from masts and derricks towards luffing cranes suitable for spot-loading in the open ship.

Alternative trends were those of driving forktrucks with their loads over a ramp onto and off the ship in ports where circumstances allowed so, i.e. where tidal ranges were small at the loading <u>and</u> discharging ports and only few ports per voyage. Where circumstances were at variance with those described here, owners tried to apply the fork-to-fork method. Both these methods needed sideports in the

ship's shell and many other adaptations in the vessel (Strengthened tweendeck plating, sunken hatches, elevators, bulkhead ports, forced ventilation, stanchionless spaces requiring heavier beams, etc.).

All those specials were cost-enhancing and the gain in tons per manhour handled - although appreciable in short trades - was far from sufficient for the trunkline trades. Turn-round of those ships in port remained slow, too slow for the expensive ship.

Then at around the year 1968 after rumbling for some time, containerization explosively entered the ocean trades that were prone to the system, i.e. the high-volume, high-value, high-speed, two-way liner trades. Ship's design then was to change quickly - and radically in places. The horizontal tweendeck partitioning made way for a vertical system of cells, enabling complete spotloading the standard-sized I.S.O. boxes without any horizontal moves in the ship..

Tweendecks, where still applied on non-cellular ships, raised their free heights from a man's to a gargantuan's in order to accommodate large forklift trucks bearing or working I.S.O. boxes. Alternative cargoes like motorcars promoted the installation by some owners of hoistable platforms in the high tweendeck spaces.

CONTAINERIZATION.

Since 1968 the container system forced its way into many liner markets. It estimately already covers more than half of the suitable trades and within a few years will embrace three-quarters of the liner trade, also encompassing low-value cargoes as well as the more high-valued ones.

We may safely assume that all major developing countries will have implemented the container system by around 1990 - the conventional type of vessel gradually having been phased out, 20 years after "go" - not only between industrialized areas but also on the less suitable trades where cargoes are not that container-friendly. (Tropical produce,

ores, metals, for instance). This is because the shrinking piecemeal fashion of handling ('breakbulk' - handling) becomes more and more expensive per ton by the law of economy of scale, the most so at the "developed" end of trades, but perceivable too in developing countries.

Thus, the transport share by conventional liner vessel will dwindle ever more and may just subsist at the peripheries of the major markets.

Until recently, the early' eighties, transformation towards the container system in trades on developing countries such as those of South America, Mexico, India, S.E.Asia, China Mainland and Africa was but extremely slow. Here the retarding factors were a.o.:

- Political objections (employment aspects).
- Physical limitations in ports, roads and infrastructure.
- Lack of financial funds for investments into the physical handling and conveyance gear which are very expensive.
- Imbalance of trade, i.e. unsuitability of cargoes for containerization on one leg of the trade.

The conveyance - in an articulated transport chain - of cargoes in the form of small loose packages (bags, bales, cartons, etc.) becoming ever more costly in proportion to same being packed in ISO containers, plain economic and competitive forces will compell export shippers in developing countries to meet their overseas' consignees' wishes to fall into line with the new system, i.e. "containerize or suffer". Here, until now another big impediment has been - and still is - the risk (and fear for risk) of deterioration of quality of the goods, alternatively of the containers. At the moment, however, we are on the verge of tropical produce and primary products being taken into ISO containers as well, be it that these boxes (and the ships) need certain special qualities and adaptations such as ventilation, easy cleaning, movable cells, draining facilities or be of special shape or construction. The crux of the matter from the

operator's standpoint is that the implement (box, flat-rack etc.) must be utilizable properly in both directions, for different kinds of cargoes. In prospective trades it often is extremely difficult to arrive at a solution on on this point.

CONTAINER SYSTEM IMPLICATIONS:

- THE CONTAINER is a large supporting or enclosing means of certain standard sizes and strength which aids speedy handling of the cargo it contains, thus resulting in short port stays for the carrying vessel - provided that terminals can cope with the system.

- Alternatively, THE CONTAINER is a large standard UNIT giving a vessel, adapted for its carriage, the feasibility of greatly enhanced throughput of payload because of higher sailings frequency. Again, if terminals are equipped properly thereto.

- Once established, THE CONTAINER allows for a better service to the client where - under normal conditions of use - full container loads (FCL) can be taken undisturbed "from door to door".

Small consignments (Less than Container Load = LCL) can by consolidation (kind of bulking) into the big units of the ISO-box also be spared a number of labour-intensive piecemeal manhandlings, at least the four times piecemeal slinging and unslinging alongside and inside the ship (2 x "in" and "out").

- THE CONTAINER improves tractability of merchandise in a transport chain once the way thereto has been prepared as it makes easy transfer feasible from trunklines to branch services - if so necessary.

- THE CONTAINER promotes shorter transit times, viz.: Port stay time shrunk considerably relative to time spent at sea (from a 50-50 division to around 30-70 division). Loading/unloading speeds at well equipped/well run terminals allowed containervessels to reduce port stay to less than 24 hrs. even with large vessels. This fact is one of the keys to efficiency. On the other hand - it is almost impossible to earn money anymore with cargoliners which stay

normally 3 - 4 days in port.

Note: Now that fuel prices soared, ship's speed had to be taken back appreciably (but still lies near to 20 knots or over), so $\frac{\text{PORT}}{\text{time}} : \frac{\text{SEA}}{\text{time}}$ now even differs more"favourable" $\sim 30^{-} : 70^{+}$ (but yearly throughput suffers).

- In the main, THE CONTAINER lessens pilferage and contact damage to the merchandise.

- Under certain circumstances THE CONTAINER lessens susceptibility to labour disturbances.

- For the bulk of high-valued piecegood cargoes the total costs for moving these in the operator's container transport chain are lower than the total costs for moving the same quantity of piecegoods in the conventional chain over the same distance.

However, 'CONTAINER' - costs must be met out of the difference between the two, of course. At the start of a new container venture this usually works out negatively for some years because of the high initial investment costs.

The spectacular difference between the transport costs per cu. ft. 'Between Piers' in a door-to-door containership service in comparison with the costs 'From Pier To Pier' by conventional ships is shown in graphs on page 19. The overland transport costs by the CONTAINER are not incorporated in the graphs so as to give a pure comparison.

LARGE VESSELS - BUT NOT MANY.

- As a result of the progressive switch to the container system, the number of ships will expand much slower than expected formerly.In fact the number may shrink temporarily. This is caused by the much higher productivity of the vessel in the system.

- The average ship size will increase by the Economy of Scale effect. However, in a number of secondary trades,limits in volume and the physical restrictions dictated by lack of depths in canals, ports as well as by quaylengths will cause many a containership's size to remain limited - if not small - as yet.

- After owners' decision towards mutual co-operation in a consortium or otherwise, the carrying capacity per ship is governed by a number of considerations,often of political background as well, e.g.:-

  Shares in the trade to be 'converted' into (number of) ships?

  Interests of overprovider(s) and underprovider(s) to be interpreted.

  How many ships in total (can the volume bear)?

  How many ships per owner? (financial situation)

  What type of ship, capacity, speed? (Compatibility items).

- Smaller sized operators should realize the reduction in transport costs by employing large ships (but less in number) and highly developed logistic aids .
- Developing nations could take part in advanced technology by sharing the group.

CREATION OF THE LINER VESSEL DESIGN BY A CONSORTIUM.

- As to the ships, after being informed about capacity, dimensions and speed, the technicians should sit down together and try to create one ship design, if possible.
- A good co-operating team might possibly arrive at ONE General Arrangement, ONE type of propulsion and sharing for model-testing at ONE basin. Meanwhile, a building specification is produced and agreed upon. After the compatibility items have been finalized, each owner invites tenders for the agreed specified and designed vessel and the ideal situation for the co-operating owners is that identical ships will be built by the yard(s) and that owners will join in one integrated container operation.

PRODUCTIVITY.

- In order to keep the "Transport Production" of a vessel as high as feasible she should sail - and not wait..... "Keep the engine warm".

- A fast turnaround is the first requirement in attaining efficiency. (see table I)

---- Many conventional liners stay in port about 3 days per call on average. Altogether port stay takes 50 percent of roundtrip time. During this time only 20% is used for cargo handling and 30% waiting. Cranes, hatchcovers, etc. have hardly any effect in speeding up cargo handling as long as many small loads are handled in many successive steps. Also, labour problems might arise easily , they are very common in the conventional handling sector.

---- Cellular container vessels and roro vessels stay in port less than 1 day per call on average, altogether their port stay takes just 22%-28% of roundtrip time. Being less vulnerable to labour problems because of the high degree of mechanization, one might speak of a reliable "Round-The-Clock" service.

table I:

ROUND VOYAGE TIME ANALYSIS - MODERN VERSUS CONVENTIONAL

Vessel type & NAME		AT SEA Roundvoy miles/Sea days	IN PORT Port days/Number of ports
Cellular container NEDLLOYD DELFT	Europe/Far East & v.v. 70 days (100 % = Round)	24.859'/54 days = 460 miles/day(19.2 km) (...77 % at sea)	16 days/16 ports = 1 day/port (... 23 % in port)
Roro/container NEDLLOYD ROTTERDAM	U.S.A./N.East Gulf & v.v. 67 days (100 % = Round)	22.715'/49 days = 463 miles/day(19.3 km) (...73 % at sea)	18 days/18 ports = 1 day/port (...27 % in port)
Conventional Cargo-Liner (1972-built) NEDLLOYD NAGOYA	Far East/Southern Africa /South America E.C. & v.v. 121 days (= 100 %)	27.139'/63 days = 430 miles/day(17.9 km) (...52 % at sea)	58 days/22 ports = 2.6 days/port (...48 % in port)
Conventional Cargo-Liner (1968-built) NEDLLOYD AUCKLAND	Australia/Southern Africa & v.v....76 days Round (= 100%)	15.417'/37 days = 416 miles/day(17.3 km) (...49 % at sea)	39 days/9 ports = 4.3 days/port (. . 51 % in port)

MODERN -- CONVENTIONAL

We see that port stays for conventional systems are 2½ - 3 times larger than for cellular/roro systems.

AIM: ONE DAY PER PORT - MAX. FOR CARGO HANDLING

DEVELOPMENTS REGARDING TRUNKLINE CONTAINER VESSELS.

In general, container vessels need a great initial stability for reasons of their deck container loads.

Stability can be gained by taking ballast or by giving the ship a wider beam.

As to ballast, this is a very uneconomic 'cargo', so not suitable. It would cost a lot of extra fuel to carry it around the world for the lifetime of a ship and moreover it decreases the deadweight available for cargo. Cargo deadweight nowadays is an element of importance in container carriage, the density of cargoes stuffed in the containers now having much greater density than at the outset of the container 'adventure'.

As to wider beam, up till recently large container vessels were held restricted in their beam width because of narrow locks, e.g. those of

Manchester Ship Canal .. 63'7" == 19.4 M
St. Laurence Seaway .... 75'0" == 22.86 M (')
Panama Canal ..... 106'0" == 32.24 M

Panama Canal width ('Panmax') had to be sustained because of the unreliable political situation in the Middle East regions since 1956 when the Suez Canal passage got blocked more than once and vessels had to sail around the Cape of Good Hope and/or through the Panama Canal. As long as the Panama Canal was under USA control it was worthwhile to stick to this maximum beam of 32.24 M.

Economy of Scale, however, demands ships of carrying capacity appreciably larger than the present biggest vessels have. Length and width of the biggest vessels reached already the Panama locks limit (for 'Regular Passage'). Depth also reached an ultimate dimension. Experience showed that the Panama canal may cause delays. Moreover, control over the canal passes to Panama with all its political implications. As a result, time is ripe now that ship's beam grows beyond the Panmax limit.

Increased beam is no problem for the passage of the Suez Canal (as long as it is open). So, a breaktrough is eminent where container vessels will get beams wider than 32.24 Metres (moulded).

Note (')... from 'Fairplay', 3/3/1983, p.28.

Consequences will be, o.a.:-

(I) Services will avoid the Panama Canal. They might use a landbridge or Panmax-sized feeder vessels for transferring any containers between East and West coasts of the American Continent. So trunklines will shuttle back-and-forth between East and West through the Suez Canal, alternatively around Cape Good Hope.

(II) Crane problems ? The bulk of shore cranes for containers at present will probably be of a size sufficiently large to serve Panmax vessels. Possibly there are cranes of bigger size - be it few - already. However, terminal cranes on certain terminals (at the biggest trunkline ports) would have to be enlarged or some new ones of greater dimensions beinstalled, at key points. Besides the outreach, boom clearance may possibly have to be increased too. It may be expected that wider beamed ships will want to carry higher deck stacks, because they have plenty stability, so existing cranes should be jacked up 1 or 2 boxes.

(Note. In case new cranes are to be ordered, a decision should be made to choose from the following two alternatives:-

(A) for serving new cellular ships with beam increased from 32.24 to 35.5 M.
On deck bay-width increased from 13 to 14 boxes across;
In hull increased from 10 to 11 boxes across (4 + 3 + 4);
Tiers in hull 9 . On covers 3, max. 4.
Hull form stability is better when compared to ships with Panama beam.
This means that less waterballast needs to be carried.
Depth of ship should remain the same.
Outreach of crane from fender .. 35.5 M - 1.2 M = 34.3 M.
Working height of crane above quay 24.5 + 2.6 = 27.1 M.

(B) for serving new cellular ships with a beam increased to 39.6 M.
On deck bay-width increased from 13 across to 16 boxes across.
In hull increased from 10 to 12 boxes across (4 + 4 + 4)
Tiers in hull 9. Tiers on covers 4 (max. 5).
Hull form stability is sufficient to carry a full fourth tier on deck and a fifth tier over a certain length.
Depth of ship should remain the same.
Outreach of crane from fender ... 39.6 M - 1.2 M = 38.4 M.
Clear working height crane over quay 24.5 + 5.2 = 29.7 M.

It is quite well possible that for the next 20 years the alternative annotated (A) will be adequate.)

As far as the presently existing cranes are concerned, the mayority are suitable for servicing vessels of 13 boxes "across" on deck. Their outreach then from offshore side of quayfender is 106' - 4' = 102" == 31.1 Metres. Their above-quay free working height is 80' == 24.5 M. Such vessel has 12 tiers "high" in total, i.e.:-

either 9 in hull plus 3 tiers on covers,
or 8 in hull plus 4 tiers on covers,
or even 7 in hull and 5 on covers if stability permits.

The beam of these vessels usually is the Panmax 32.24 M. limit.

SURVIVAL.

At times of economic recession and under the pressure of the nearly always present competition threats one handhold to survival is to have the right ships and an efficient organization. This may imply, amongst other, to provide for the following measures:-

- Increase of ship's size (In the range of from 1,400 to 4,500 TEU)
- High sailing frequency (half-monthly, weekly or even less interspacing). This means that ship operators/owners have to co-operate and to act as one body viz-a-viz the total package.
- Specialization within the cellular fleet, e.g. to provide reefer cells, vented cells. Even breakbulk spaces and roro spaces could be necessary, but be provided only if at all unavoidable, however.
- First class care in case of flaws occuring, proper software service (documentary paperwork; see also note (1) hereafter)
- Transit time from door-to-door to be kept as small as possible, especially for urgent cargoes. This, by the way, may also generate extra cargoes of the perishable kind such as plants and vegetables that formerly would have deteriorated on account of long transit time.
- Trunk services between few ports (this, however, remains a questionable point in view of "feeder costs")
- In principle, few ports of call and short port stays, if necessary by intensifying the mutual co-operation in order to get full ships (see also notes (2) and (3) hereafter).
- Feeder services in order to minimize trunkline ports, but only if unavoidable as feeder services waste away the freight.
- Diversification of container equipment - adjusting to trade demands.

Notes.

(1) Liner shipping will develop further into the role of Transport Operators who manage, organize and control the c o m p l e t e transport chain.

(2) Scattered traditional market shares will be consolidated often with an intruder's into consortia for some length of time at least.

(3) Established consortia may be reshaped into:-

(a) Shipowning beside ship-operating function.

(b) Marketing and transport organization (that may charter extraneous hardware capacity in order to cope with peak requirements).

CONTAINERS IN BULKCARRIERS - ARE THEY THE LINER OPERATORS' WORRY ?

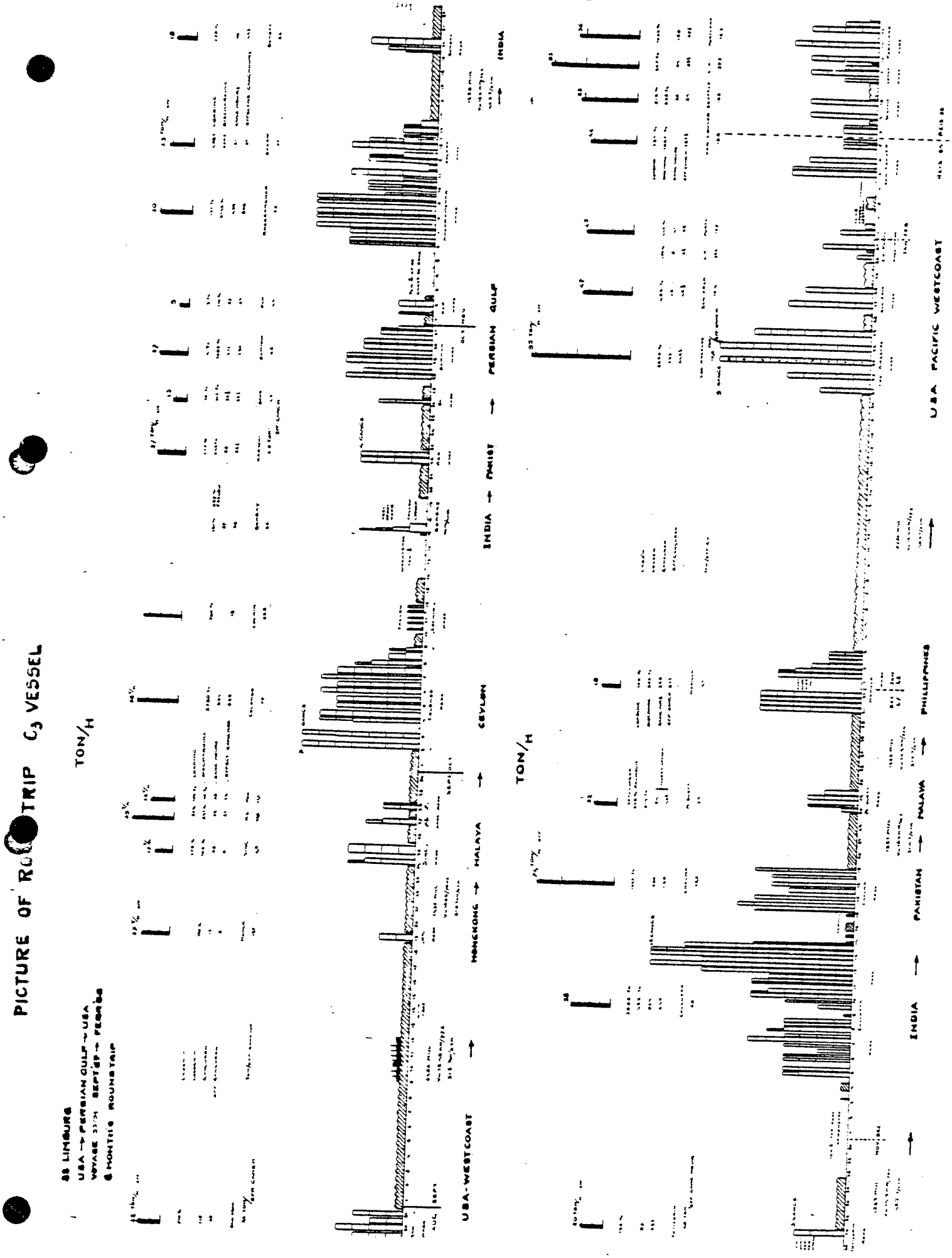
One consequence of the bulking of liner piecegood cargoes into sturdy standardized boxes will be an increased competition by bulkcarriers of the 'open' type, in particular on trades where regular/frequent traffic of dry bulk cargoes exists. In building up a trade, return trips "in ballast' are offered for boxes at a rather low box rate. (Subsequently container ships are chartered to fill-in schedule gaps.)

However, a fast and regular door-to-door service would not seem readily feasible with just bulkers. Box stowing may be hampered by lack of guides, shifting of boxes from covers, extra lashings, etc. . At sea, this kind of vessel is more liable to rolling and pitching heavily and to slamming in heavy seas on account of its fuller form. Also, sailing speed of such kind of vessel usually is appreciably lower than that of the containership proper, resulting in longer transit time. (With present secondary class container cargoes like wool etc. the transit time works in favour of the competing bulkcarrier, however, making it a medium to intrude in proper container services)

Remains the fact that bulkcarriers will transfer the effects of slumps in other markets into liner business much more than trampers did in former days.

Finally one cannot deny that a cellular containervessel is a "bulkcarrier for boxes" after all.

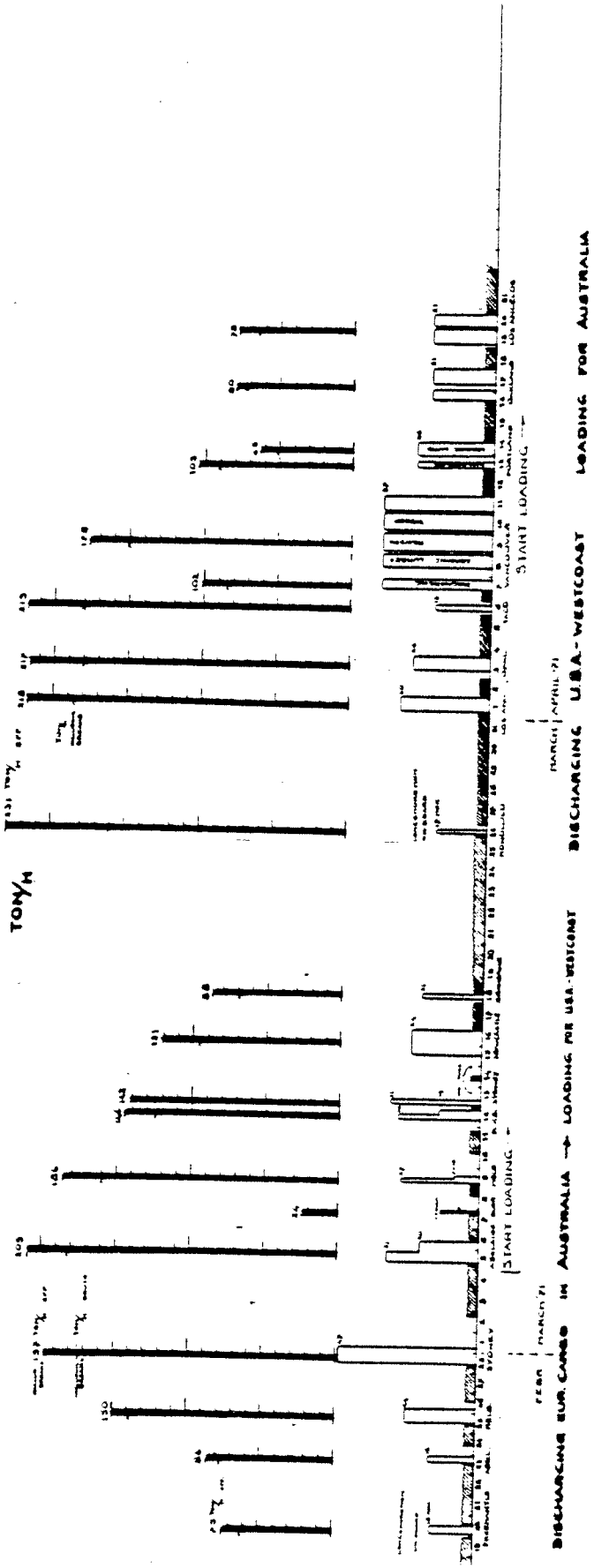
1.0.12 Round-Voyage Conventional Vessel



Round-Voyage RoRo Vessel

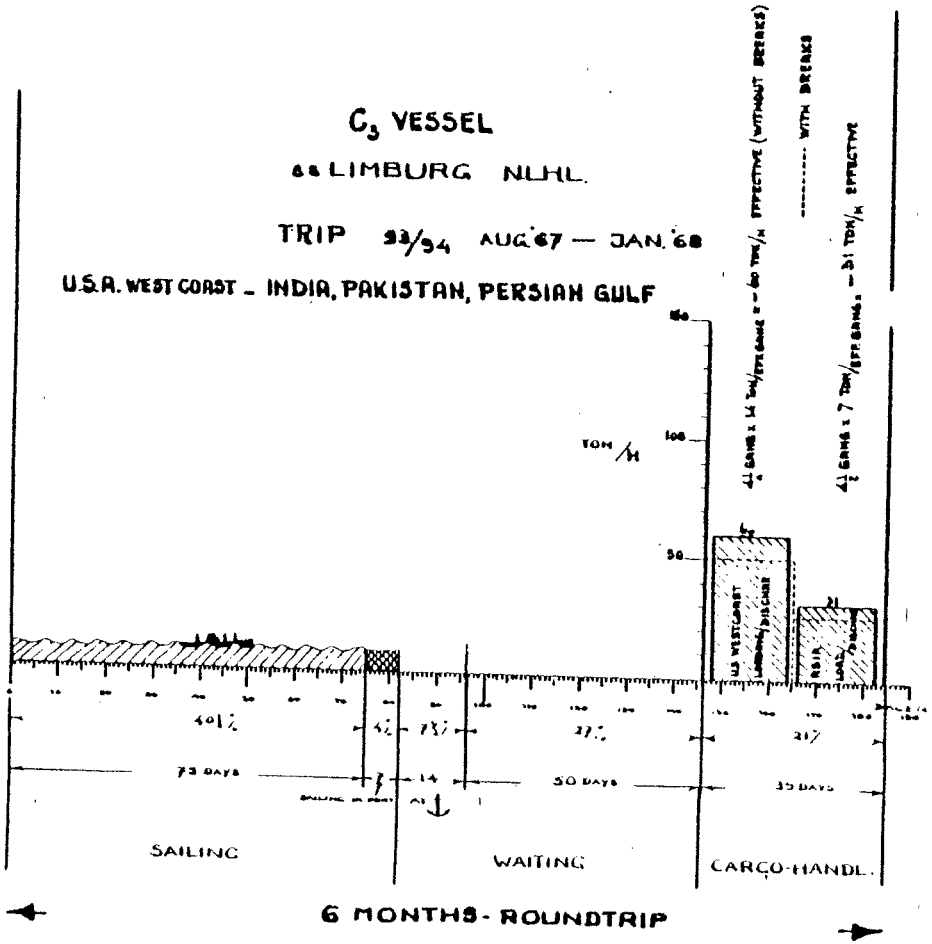
PICTURE OF ROUNDTRIP M.S. PARALLA

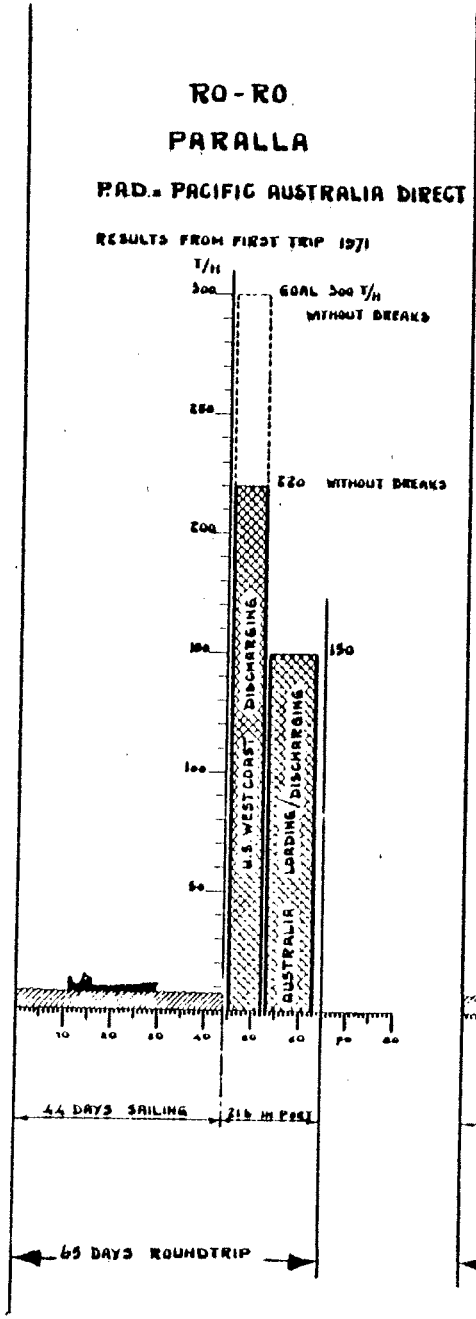
ROLL-ON ROLL-OFF PAD TYPE

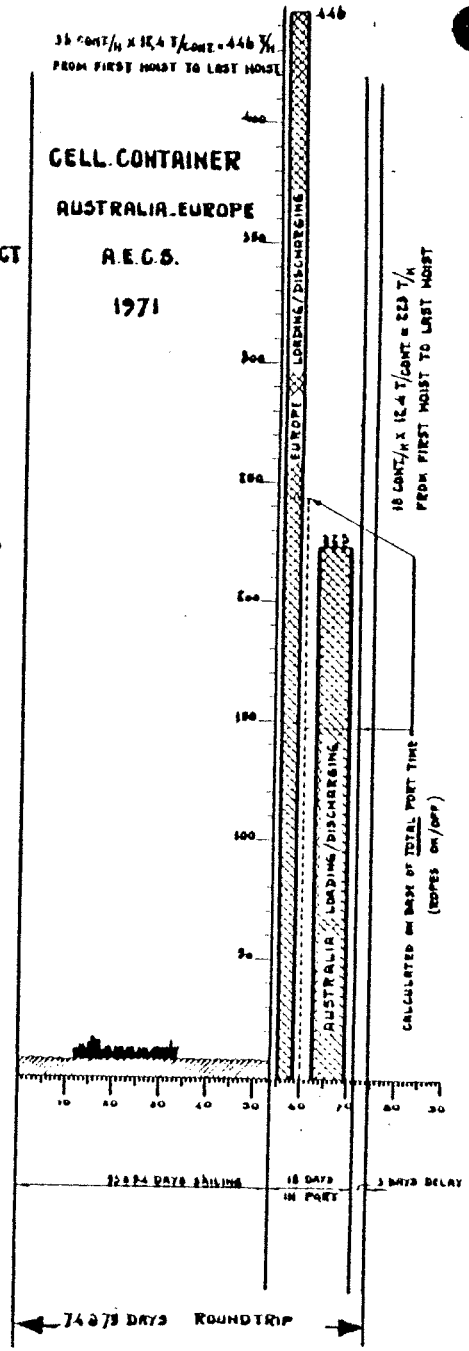


REDUCTION OF ROUNDTRIP

SPEED UP CARGO HANDLING

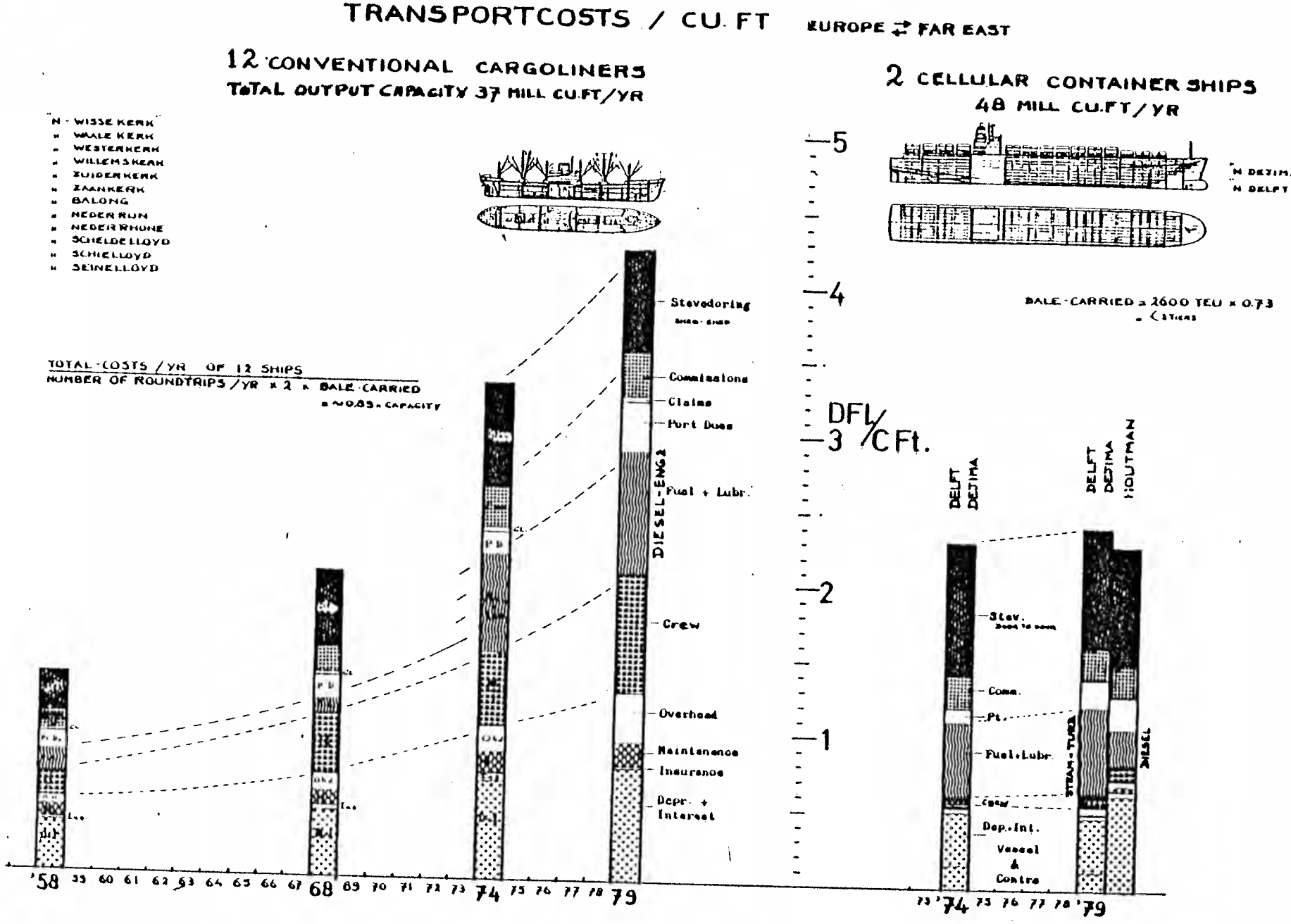






1.0.14

Comparison of Transport Costs
Pier - to - Pier
Conventional : Cellular Ctr.



(A) CRITERIA OF IMPORTANCE FOR THE SHIPOWNER WHEN CHOOSING A VESSEL.

As a matter of fact there are numerous criteria and we can for shortness sake only mention a few of the most weighty ones here.

For the private or incorporate Western shipowner/ operator we can mention the following long-term criteria:-

GENERAL CRITERIA.

Commercial	- Is there sufficient trading potential?
Financial	- Is there a possibility of making a profit with the ship at all? Is there sufficient capital to be found for investment, either risk-bearing or against reasonable interest? The profit - in case of the latter - should at least earn the interest and expenses made.
Political	- Is the Owner allowed to participate in the trade envisioned? (Coastal trade in foreign countries is often prohibited by law; Crosstrade between foreign countries is rapidly losing its potential, a.o. by UNCTAD decree influence; Home-trades of small countries without overseas provinces anymore have declined to a trickle).
Competition	- Is there a hard core of competitors to be met in that particular trade or not (yet)?
Ship-technical/ cargo-allied.	- Is there a chance to get cargo by offering a specialized vessel or a more efficient cargo handling method?

SPECIFIC CRITERIA.

Cargo-technical	- Which kind of cargo the Owner/operator wants to concentrate his efforts on?

What are the usual (trade's) parcel sizes to base his ship's capacity on? Will two ships be sufficient for such trade or are more ships necessary to keep the trade going (frequency of sailings: $\frac{\text{roundtrip days}}{\text{number of ships}}$ to be $\leqslant$ 2 weeks).

Nautical and climatological

- How are the ambient conditions? Will the ship have to sail ice-bound waters? Quiet but tropical conditions, typhoon risks, shallow waters, confined ports (thrusters). Relative time used in manoeuvring v.a.v. plain sailing and v.a.v. port stays "c.p.p." i.s.o. "f.p."propeller ;length of voyages i.c.w. fuel supplies/capacity; speed vessel has to be able to maintain on average and max. for making up towards scheduled arrival time after delays and set-backs.
  Single-screw, or twin-screws "safety"? The proper economical shape of the ship in connection with the parameters arrived at by the commercial men: carrying capacity, speed, bunkering intervals, type of fuel, behaviour (roll-suppression, vibration-avoidance, heel-control) And a lot more.

Social

- Will Owner be compelled to sail expensive and extensive number of crew members? Do they necessitate expensive/ extensive accommodation?

Safety

- Besides the usual ones, what rules and regulations the ship and Owner have to comply with in connection with type and quantity of (dangerous/hazardous) cargo(es) and waters to be sailed in/

ports to call at? These rules are ever-more and rapidly increasing in number and severity, making investments heavier and can the value of the cargo bear such costs?
&c, &c.

For the State-run fleets and (so-called) National shipping companies of a number of countries at present - mostly socialistic countries or developing countries, in that order - the same criteria are valid, be it that their sequence and specific weights may be at variance. For State-run fleets of socialistic countries e.g. the acquisition of foreign currency is a very heavy criterion, so much that they often compete heavily by quoting freight rates well below the commercial break-even levels. Also military latent potential may play a role for these 'owners' in building up a large fleet.

The 'national' shipping companies' procedure usually is to penetrate into existing shipping business (without giving heed to economic standards that govern free trades) by compelling importers to make use of their national ships for their goods on penalty of withholding import-licences or the like.
It goes without saying that such criterion tends to keep living standards in those countries but poor for the masses.

LINER SERVICES - LONG TERM CRITERIA.

Potential	- How big is the total cargo package in the trade for which the ship is intended to sail?
Shares	- How big is the share in that particular package for the Owner traditionally- or attainable by providing a better service by a better ship? How long may this share last with a view to the ship's expected life-

time - usually some 20 years, or 10+ 'good' years economically.

Nature of the goods -How is the composition of the cargo package? That is to say, how is the diversity of the package as to nature of the goods, what are the properties of the goods as to dangerousness, their values, their quantities, perishableness?
Can he expect seasonal fluctuations? (Fruits) Can he expect decline in parcels offered (air competition) of valuable types of goods? Or may he expect increasing trend of such cargoes when their value cannot bear rising air freight rates?
Can he generate new traffic?
This criterion - nature of the goods - is of great importance for the ship's arrangement (although with former conventional liners more so than with the present container liners) and for the ship's equipment and appliances. Specific stowage arrangements e.g. reefer cargo need refrigeration and insulation; edible oils and other harmless liquids are often offered for carriage in bulk in parcels of some hundreds of tons, requiring special tank compartments and pumping/heating machinery and apparatus.Ore parcels need strengthened floors against grab-damage; motor cars need fluctuating lots of space at precise heights.

Nature of packing and unitisation	- Are the goods offered for carriage at ship's side unpacked, in bulk? Packed in separate packages? Are they sturdily bundled? Palletized? Stuffed in ISO containers? Or does owner have to bundle, palletize, stuff them so? (Extra cost!) Can it be moved on its own wheels? These criteria are of extreme weight in arriving at the lay-out of the ship, the general arrangement and cargo handling gear - if so required.
Ports' Technical conditions -(and labour supply)	- Very closely tied to the foregoing are the port facility conditions. If all the ports to be served by the vessel do have the proper gear, the ship - in principle - would not need it. The Owner may, however, still want to provide the ship with its own gear for the sake of versatility of the ship: should she have to be switched into another line, she may have to have gear - a company like Nedlloyd must keep this fact very much in mind - so it may save the vessel's "life"- and the company's investments.
Ports' nautical conditions	- Attainability: water depths, river bars, swell occurrence, entrance width and turning feasibility (East London, Chilean ports, Middle East ports). For manoeuvring around in confined ports and windy places, bow-thrusters may need to be complemented by stern thrusters as time became a costly criterion in ship's exploita-

tion and many ports' nautical boundaries did not expand with the speed the ship's dimensions did of late.
Many ports became "narrow" in recent years and at the moment their boundaries often form crucial criteria in deciding ship's overall length - and economy, the latter being bound to ship's size.

LINER VESSELS - SHORT TERM CRITERIA.

Ship design from cargohandling point of view

- Put a completely different design vessel into a suddenly changed market- OIL CRISES:
THE example is the building of big RoRo vessels for serving the Middle East Gulf ports. As a result of explosively risen oil prices those countries experienced a tremendous rise of import of merchandise. The conventional cargo could not quickly enough be removed inland and so wharfs became glutted, ships had to wait and ports got 'congested' for months at the time.
By 'putting the cargo on wheels' , ship-operators could have their RoRo vessels emptied quickly by driving that cargo some miles into the desert where make-shift stacking terminals were made. That way the trade could be coped with normally again.

1.2 (B) MAY WE EXPECT A CHANGE IN CRITERIA IN THE FORTHCOMING DECENNIAL PERIOD?

The answer is Y E S.
There is a c o n t i n u o u s shifting in the criteria. We dare to say that there never is a dull moment, there never is a standstill in criteria.

Even the w e a t h e r has a big influence on trading volumes and we all know how difficult it is to forecast the weather in short term, let alone for longer periods ahead. So there are:-

Poor crops ..... more import of foodstuff, less merchandise;

Good crops ..... less import of foodstuff, possibly even exports, relatively more import allowed for merchandise.

Threat of war ..... high gold prices result in rising import in gold producing country.

Recession ..... lower gold prices may cause a halt to import licences.

High(er) oil prices .... rising imports of merchandise into oil-producing area's. (E.g. Middle East)

Lower income from oil .. import-stops on merchandise in one or more of the area's.

Autarchy-minded government ... little imports; usually also little exports (e.g. People's Republic of China after 1949 until recently).

Political system influences ... Eastern-European trading system is rather rigid by lack of flexibility in financial sphere, resulting in restraint in trade (barter system).

Political influences ... since OPEC raised oil prices manifold, oil-offtake by importing countries decreased considerably and this caused many tankers to be laid-up, especially the VLCC's. Moreover it caused the worldwide rise in economy to slow down, resulting in decreasing dry cargo trades as well.

Health conditions ..... since discovery of peniciline etc. the world population grew explosively which resulted in increased trade (bulk).

1.3 (C) IS THERE A LIMIT TO THE "ECONOMY OF SCALE" IN OPERATING/ BUILDING SHIPS?

The answer is: In theory ... YES - and in practice ... still earlier!
The better the mutual co-operation between operators will become, the more the practical limit will approach the theoretical one.
The application of the Economy of Scale by a single operator is a very risky business. This could be undertaken only with government backing.

SYNOPSIS OF NEDLLOYD LINES' SERVICES(:)- AND VESSEL TYPES USED THEREON.

2.0.01

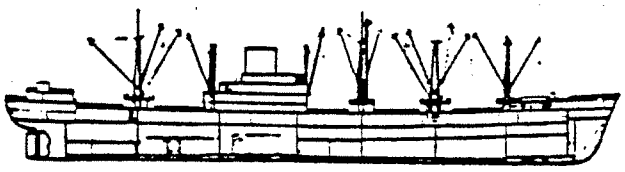
NEDLLOYD SERVICES

(Note. (:)

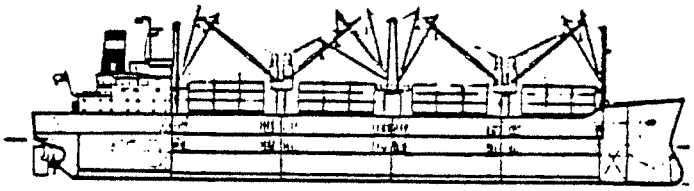
Services are in part independent and in part in mutual co-operation with other shipping companies.)

Selected typical liner vessel kinds of Nedlloyd.

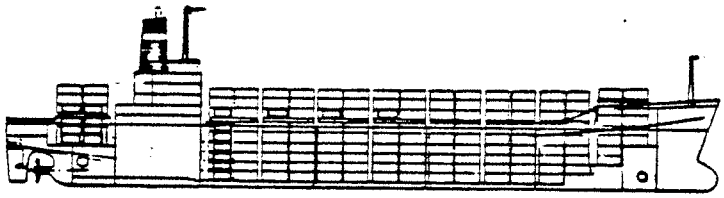
Type annotation used on next pages.



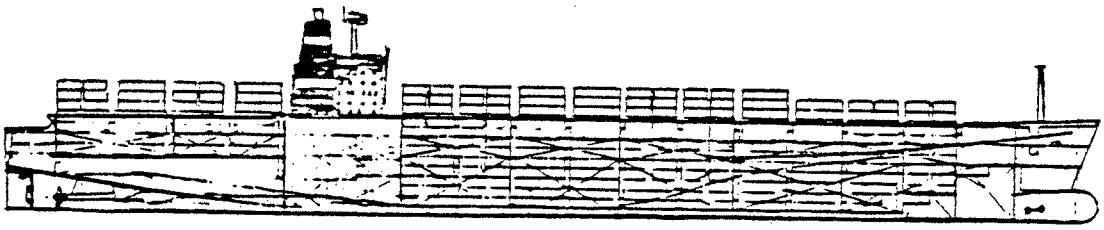
Conventional G/C liner



Container-friendly General Cargo liner (M.P.C.) — M

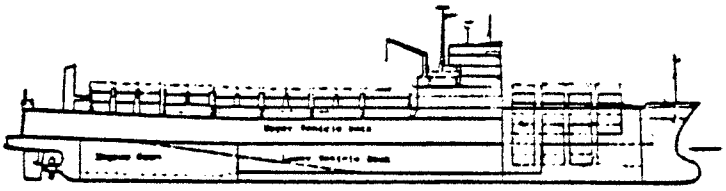


Cellular Containership for area trade

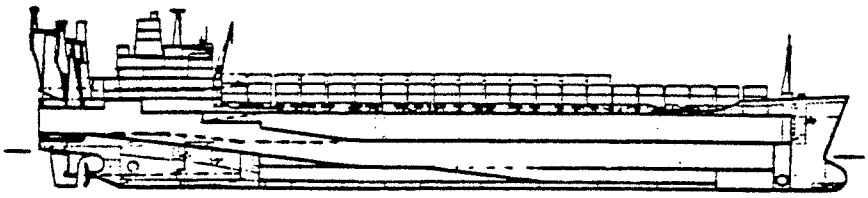


Cellular Containership for trunkline trade

C



Combined cellular container & Roro carrier

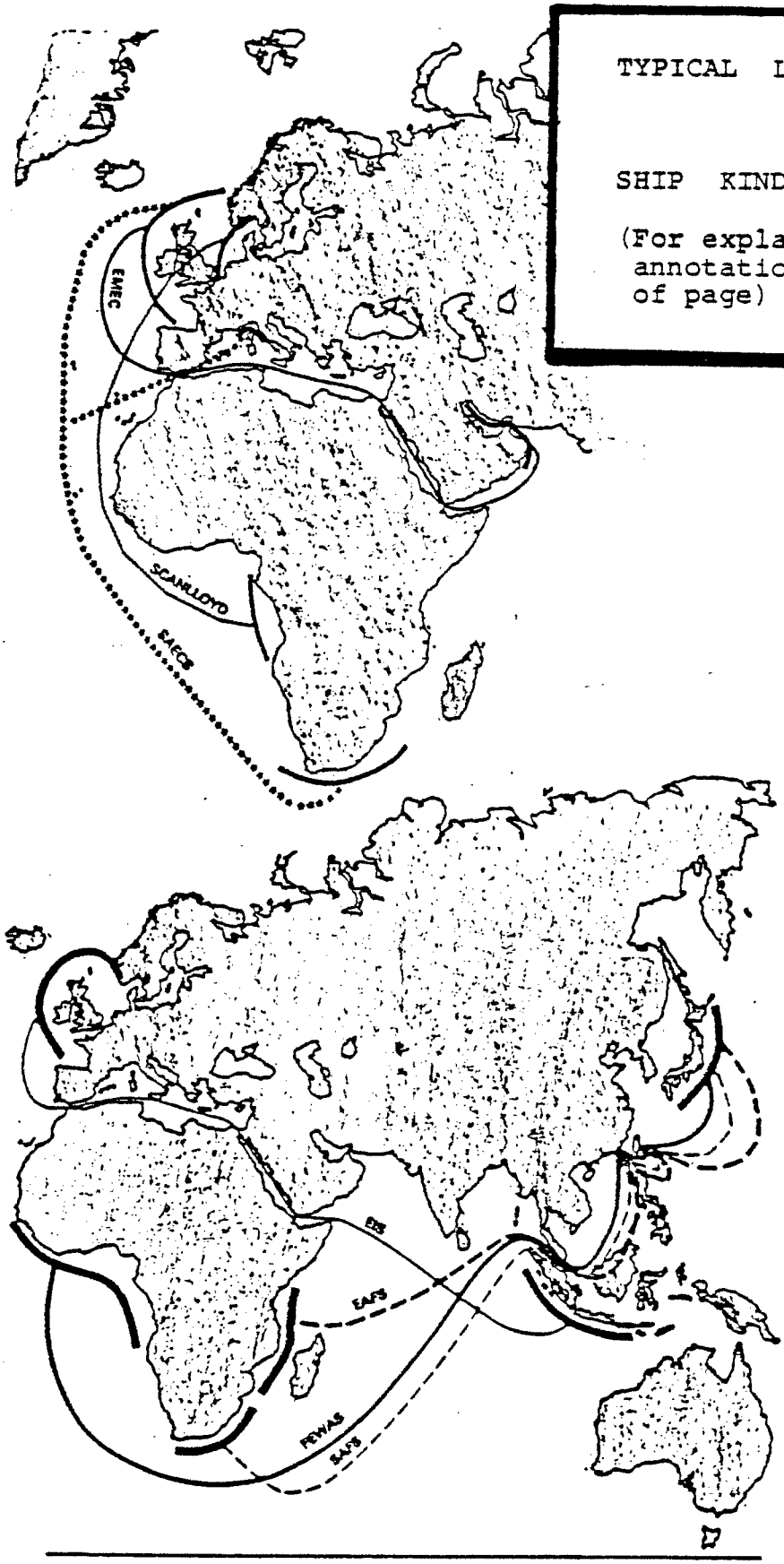


Roro & container carrier

R

TYPICAL LINER TRADEROUTES AND SHIP KINDS USED THEREON.

(For explanation of ship kind annotations see at bottom of page)



EMEC C + M
Europa/Midden-Oosten v.v.
1 × per week
In samenwerking met:
- Hapag-Lloyd en
- Compagnie Maritime Belge

Voor het containervervoer bovendien samen met:
- United Arab Shipping Company
- OCL (Engeland)

Nedlloyd van Diemen (gecharterd)
Nedlloyd Caledonia (gecharterd)
Nedlloyd Bahrain (1978) 22.500 ton
Nedlloyd Barcelona (1979) 22.500 ton
Nedlloyd Moji (1977) 16.186 ton

SCANLLOYD M
Europa/Zaire Angola v.v.
In samenwerking met:
- Scandinavian West Africa Line

Nedlloyd Tauranga (1967) 14.988 ton
Nedlloyd Talbot (1967) 14.988 ton

SAECS C + R
Europa/Zuidelijk Afrika v.v. (container- en roro-schepen)
1 × per week
In samenwerking met:
- Chargeurs Réunis (Frankrijk)
- Comp. Maritime Belge
- Comp. Générale Maritime (Frankrijk)
- Deutsche Afrika-Linien
- Safmarine (Zuid-Afrika)
- Lloyd Triestino (Italië)
- Ellerman (Engeland)
- Harrison Container Line (Engeland)

op het ogenblik geen eigen schip.
Nedlloyd Hoorn tijdelijk door SAECS verchartard aan ScanDutch

EIS M
Europa/Indonesië v.v.
samenwerking en frequentie in onderhandeling
Nedlloyd Wisselkerk (1967) 13.282 ton

EAFS M
Verre Oosten/Oost-Afrika v.v.
1 × per maand
Nedlloyd Hobart (1968) 12.771 ton
Nedlloyd Hongkong (1968) 12.836 ton
Nedlloyd Honshu (1968) 12.771 ton

FEWAS C / M
Verre Oosten/West-Afrika v.v.
1 × per maand
Nedlloyd Madras (1979) 16.927 ton
Nedlloyd Manila (1979) 16.927 ton
Nedlloyd Marseilles (1979) 16.927 ton

SAFS C / M
Verre Oosten/Zuid-Afrika v.v.
1 × per maand
2 gecharterde containerschepen
Nedlloyd Steenkerk (1961) 12.190 ton
Nedlloyd Sinoutskerk (1962) 12.285 ton
Nedlloyd Spaarnekerk (1962) 12.214 ton

C = Cellular container vessel

M = Multipurpose vessel

R = Roll-on roll-off vessel

EWAS M, future C
Europa/West Afrika v.v.
1 x per 14 dagen
Nedlloyd Nile (1972) 12.402 ton
Nedlloyd Niger (1971) 12.402 ton
Nedlloyd Rhone (1963) 12.047 ton
Waterland (1974) 15.180 ton

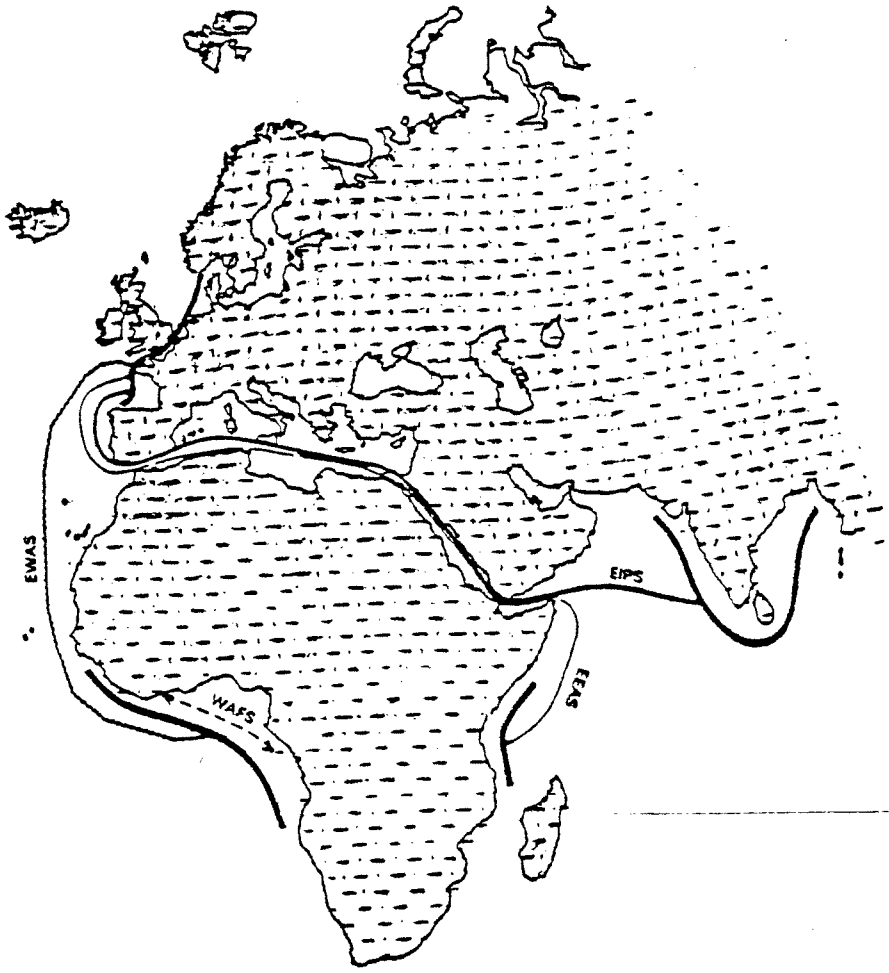
WAFS
Feederdienst voor FEWAS (tussen Afrikaanse havens)
Nedlloyd Sprinter (gecharterd)
M

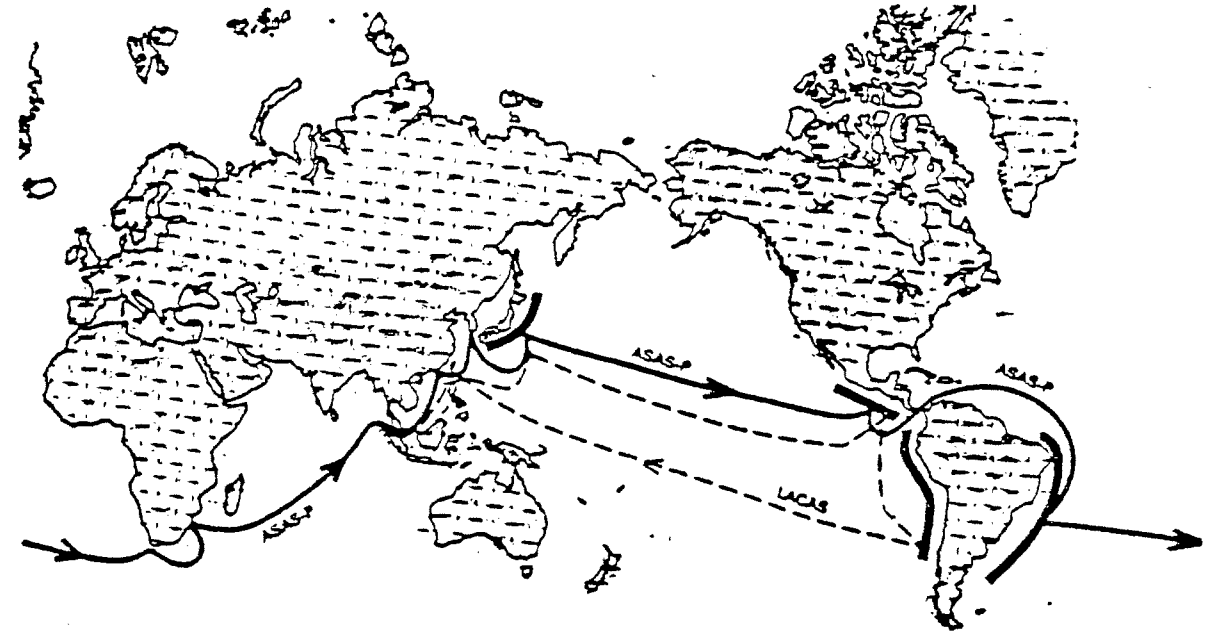
EIPS
Europa/India en Pakistan v.v.
1 x per maand
Nedlloyd Willemskerk (1967) 13.281 ton
Nedlloyd Waalekerk (1968) 13.281 ton

EEAS M
Europa/Oost-Afrika v.v.
1 x per 6 weken
Nedlloyd Katwijk (1971) 17.097 ton
Nedlloyd Westerkerk (1967) 13.281 ton

ASAS-P M, future C
Verre Oosten/Zuid-Amerika v.v. via Panama-kanaal
1 x per maand
Nedlloyd Nassau (1972) 15.936 ton
Nedlloyd Nagoya (1971) 15.936 ton
Nedlloyd Nagasaki (1972) 15.936 ton
Nedlloyd Napier (1972) 15.936 ton

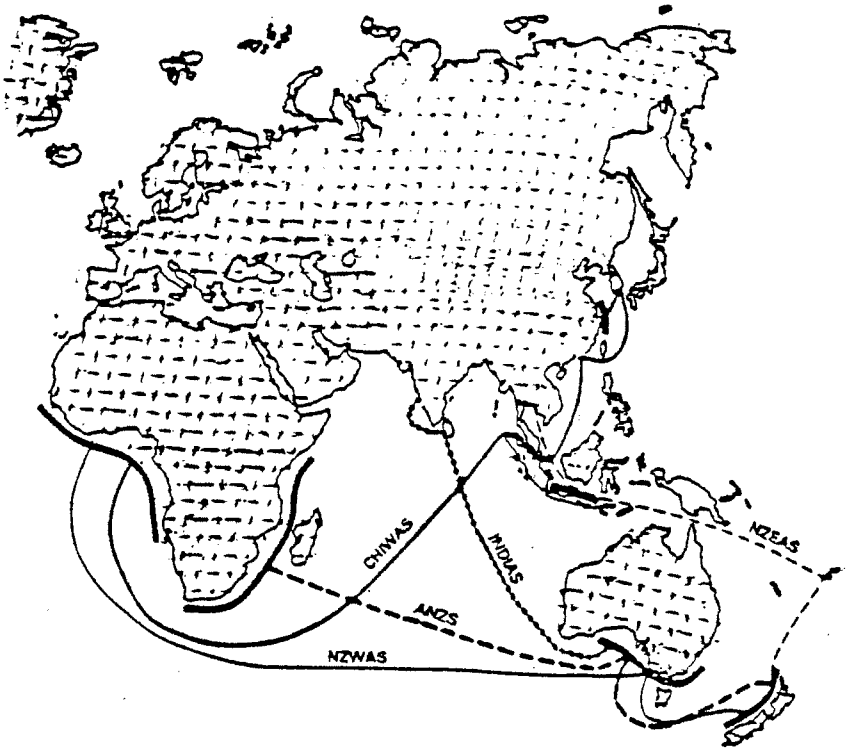
LACAS M, future C
Verre Oosten/Westkust Zuid-Amerika v.v.
1 x per maand
Nedlloyd Bangkok (1978) 22.500 ton
Nedlloyd Baltimore (1978) 22.500 ton
Nedlloyd Forcados (1962) 12.310 ton
Nedlloyd Streefkerk (1961) 12.255 ton





C = Cellular container vessels

M = Multipurpose vessels



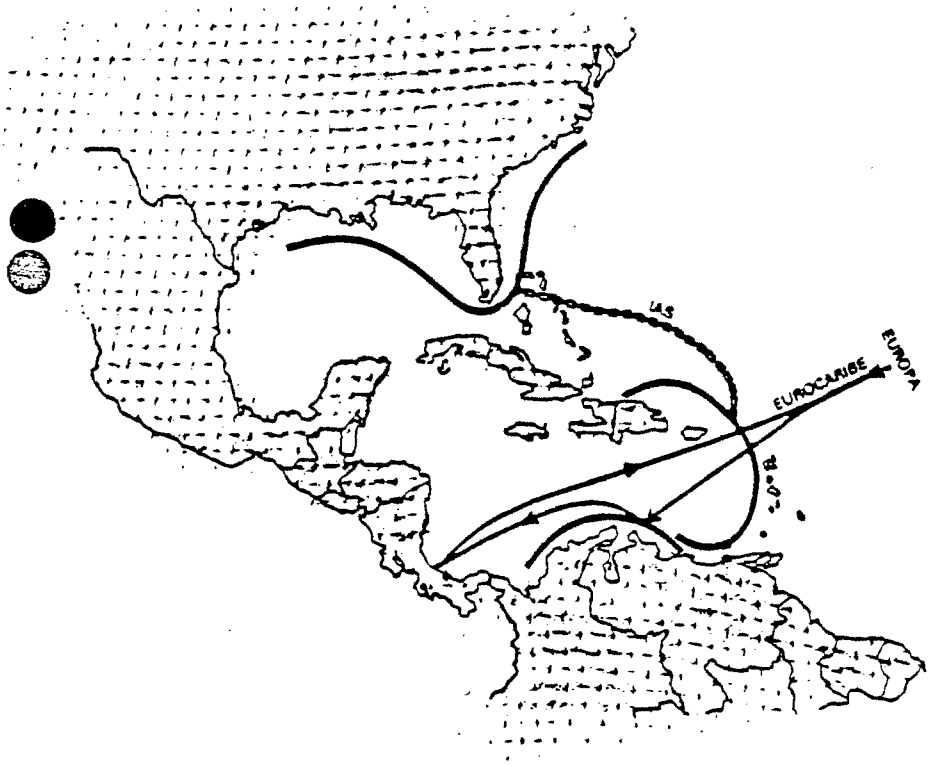
CHIWAS M
Volksrepubliek China/West-Afrika v.v.
1 × per maand
Nedlloyd Fukuoka (1961) 12.254 ton
Nedlloyd Fresco (1962) 12.335 ton
Nedlloyd Schie (1959) 12.873 ton
Nedlloyd Scheide (1958) 12.873 ton
Nedlloyd Weser (1960) 11.725 ton

INDIAS M
India/Australië v.v.
1 × per 3 weken
Nedlloyd Kyoto (1970) 16.967 ton
plus 1 gecharterd schip

ANZS M
Zuid- en Oost-Afrika/Nieuw-Zeeland v.v. (geünitiseerd)
1 × per 6 weken
Nedlloyd Fremantle (1964) 15.277 ton
Nedlloyd Frazer (1963) 15.206 ton

NZEAS M / future C
Nieuw-Zeeland/Oost-Azië (geünitiseerd) v.v.
1 × per 10 dagen
Nedlloyd Adelaide (1968) 14.045 ton
Nedlloyd Auckland (1968) 14.116 ton
Nedlloyd Amsterdam (1968) 14.116 ton
Nedlloyd Albany (1968) 14.005 ton
Nedlloyd Holland (1967) 12.775 ton

IAS
US-Oostkust en Golf van Mexico/Caribisch gebied v.v.
± wekelijks
Hector (1979) 3.455 ton
Mentor (1979) 3.455 ton
Stentor (1979) 3.455 ton
plus 3 veste gecharterde schepen



EUROCARIBE M
Europa/Venezuela, Colombia, Costa Rica v.v.
1 × per week
In samenwerking met:
- Compagnie Générale Maritime
- Hom-Linie Duitsland)
- Hapag-Lloyd

Alkmaar (1971) 11.465 ton

MEDUSA C + R
Middellandse Zee/Verenigde Staten (containers en roro)
1 × per 17 dagen
Nedlloyd Rouen (1978) 22.564 ton
Nedlloyd Rosario (1979) 22.564 ton
Nedlloyd Rotterdam (1978) 29.218 ton
Nedlloyd Rochester (1979) 29.218 ton

MCS M
Middellandse Zee/Caribisch gebied (éénrichting)
1 × per maand
Nedlloyd Main (1962) 11.735 ton
plus gecharterde schepen

PAS M
Westkust V.S./Zuidelijk Afrika v.v.
1 × per maand
Nedlloyd Kambla (1971) 17.097 ton
Nedlloyd Kingston (1971) 16.920 ton
Nedlloyd Kimberley (1971) 16.920 ton

EECSA M
Europa/Oostkust Zuid-Amerika
1 × per 14 dagen
Flevoland (1973) 13.945 ton
Gooiland (1969) 8.475 ton
Westland (1975) 15.123 ton
Amstelland (1962) 9.585 ton
Sailand (1974) 13.945 ton

C = Cellular vessel
M = Multipurpose vsl
R = Roll-on roll-off or partly roro vsl.

ANZECS C
Europa/Australië/Nieuw-Zeeland v.v.
1 x per week
in samenwerking met:
- Overseas Container Ltd
- Hapag-Lloyd (Duitsland)
- Compagnie Générale Maritime (Frankrijk)
- Lloyd Triestino (Italië)
- Shipping Corp. of New Zealand

Op het ogenblik geen eigen schepen tot voor kort:
Nedlloyd Tasman
Nedlloyd Houtman
die nu als Mounts Bay en Largs Bay in charter varen

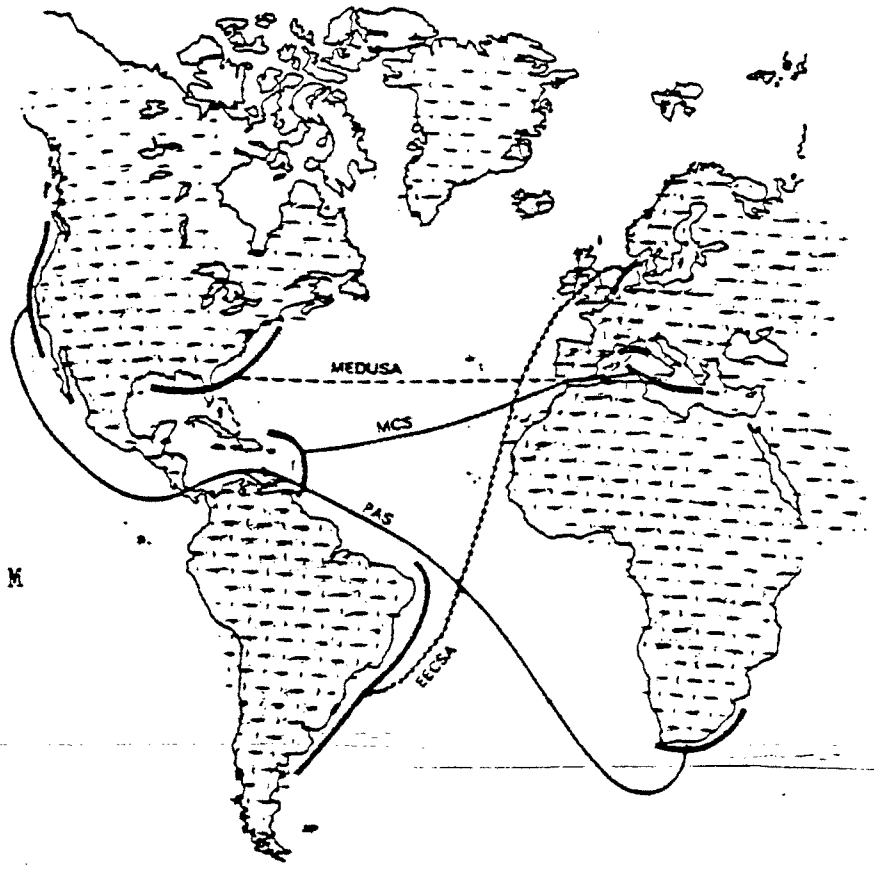
EPS/NZWAS M
Europa/Pacific-eilanden/Nieuw-Zeeland/-West-Afrika/Europa
(éénrichtingdienst rond de wereld)
1 x per maand
Amersfoort (1970) 11.465 ton
2 gecharterde schepen

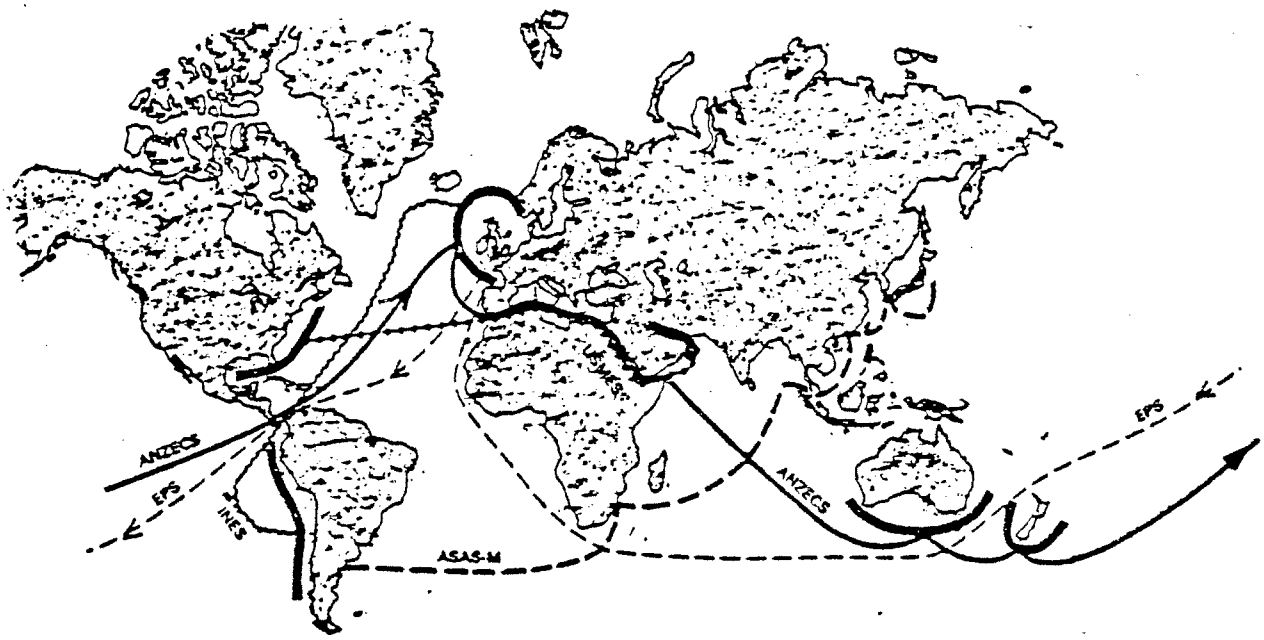
INES M
Europa/Westkust Zuid-Amerika
1 x per week
in samenwerking met:
- Hapag-Lloyd
- Sud Americana de Vapores (Chili)
- Transportes Navieros Ecuatorianos
- Lineas Navieras Bolivianas

Oranjestad (1971) 9.235 ton
Willemstad (1970) 9.165 ton

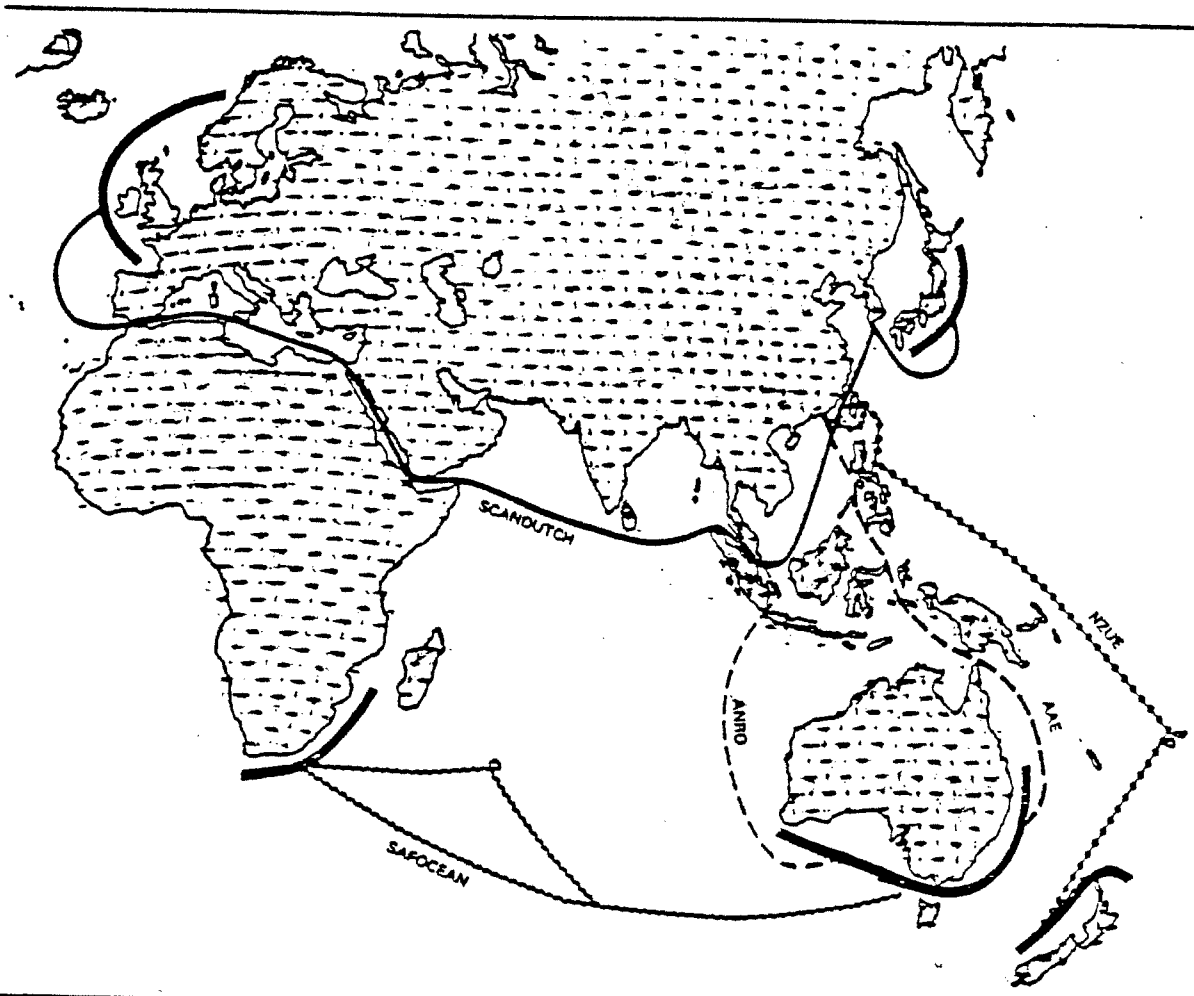
ASAS-M M, future C + M
Verre Oosten/Zuid-Amerika v.v. via Zuid-Afrika
1 x per maand
Nedlloyd Lauve (1966) 12.278 ton
Nedlloyd Linge (1967) 12.304 ton
Nedlloyd Loire (1967) 12.324 ton

USMES C + R
Verenigde Staten/Midden-Oosten v.v.
(containers en roro)
1 x per 17 dagen
Nedlloyd Rouen (1978) 22.564 ton
Nedlloyd Rosario (1979) 22.564 ton
Nedlloyd Rotterdam (1978) 29.218 ton
Nedlloyd Rochester (1979) 29.218 ton





C = Cellular vessel
M = Multipurpose vessel
R = Roll-on roll-off (or partly roro) vessel.



SCANDUTCH C
Europa/Verre Oosten v.v.
1 × per week
In samenwerking met:
- East Asiatic Comp. (Denemarken)
- Broström Shipping Comp. (Zweden)
- Wilh. Wilhelmsen (Noorwegen)
- Comp. Générale Maritime (Frankrijk)

Nedlloyd Delft (1973) 48.146 ton
Nedlloyd Dejima (1973) 48.146 ton
Bovendien is voor periode van inbouw nieuwe motoren in Delft en Dejima de Nedlloyd Hoorn gecharterd van SAECS

SAFOCEAN C
Zuidelijk Afrika/Australië v.v.
1 × per maand
In samenwerking met:
- Safmarine (Zuid-Afrika)

Safocean Mildura (ex Nedlloyd Lek) (1968) 15.390 ton

ANRO R + C
Australië/Indonesië/Singapore/Maleisië/-Thailand v.v. (containerdienst)
1 × per 3 weken
In samenwerking met:
- Australia Straits Container Line.
- Neptune Orient Lines (Singapore).
- Australian National Line.

Anro Asia (voor de helft eigendom Nedlloyd, andere helft Australia Straits Container Line)

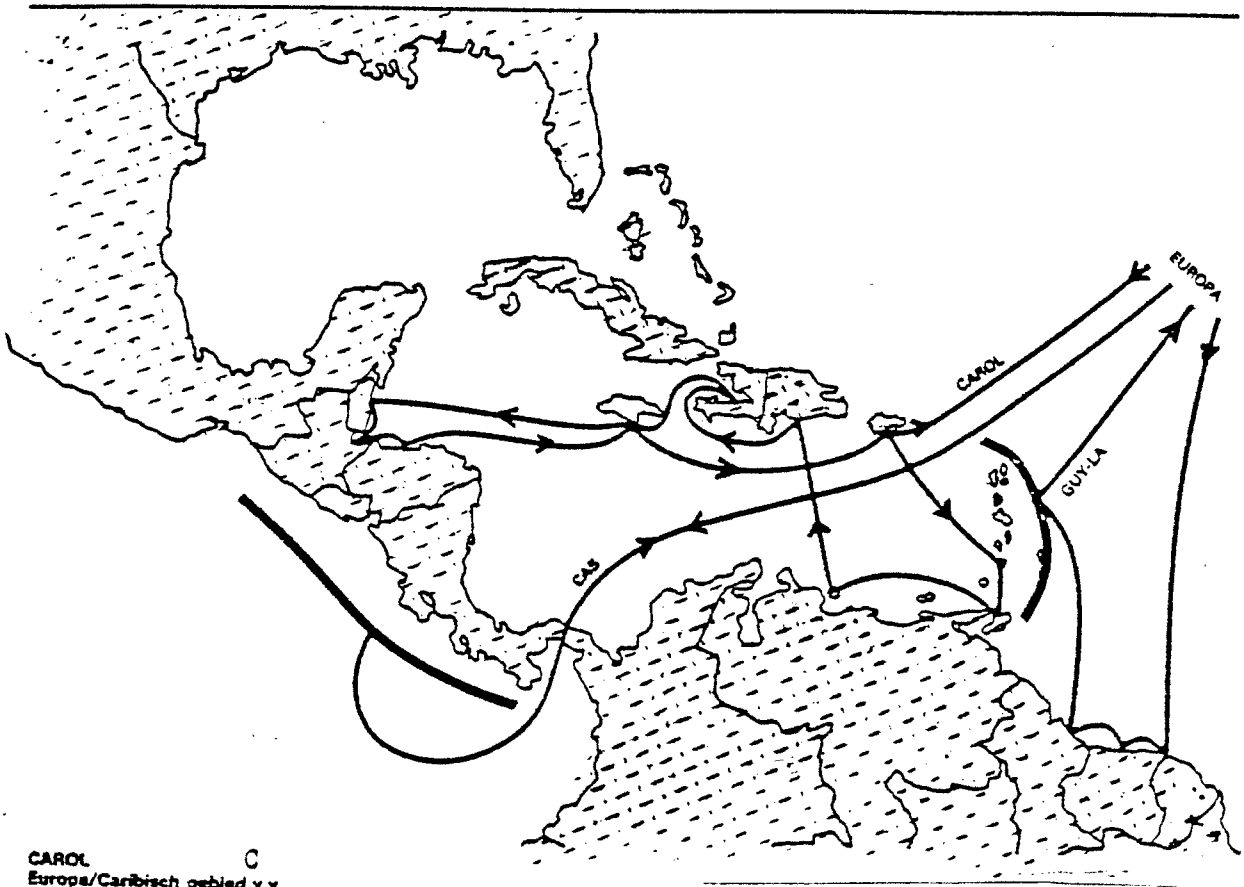
AAE C
Australië/Hongkong/Philippijnen/Taiwan v.v. (containerdienst)
In samenwerking met:
- Overseas Container Pacific Ltd. (Engeland)

slot-charters; geen eigen schip

NZUE M / future C + R
Nieuw-Zeeland/Verre Oosten v.v.
1 × per 10 dagen
In samenwerking met:
- Mitsui OSK Lines (Japan)
- China Navigation Co (Hongkong)

Nedlloyd Franklin (1963) 15.208 ton
Nedlloyd Freetown (1963) 15.177 ton

C = Cellular vessel
M = Multipurpose vessel
R = Roll-on roll-off vessel (or partly roro)



CAROL C
Europa/Caribisch gebied v.v.
in samenwerking met:
- Hapag Lloyd (Duitsland)
- Harrison Line (Engeland)
- Compagnie Générale Maritime (Frankrijk)
- Hollandia (1977) 23.230 ton

CAS C
Europa/Westkust Centraal-Amerika v.v.
in samenwerking met:
- Compagnie Générale Maritime (Frankrijk)
- Royal Mail Lines (Engeland)
- Transportación Maritima Mexicana

geen eigen schip
de 'Zeelandia', nu varend als 'Benattow',
was voor deze dienst gepland

GUY-LA M
Europa/Suriname-Guyana-Kleine Antillen
1 × per 3 weken
Nedlloyd Kobe (1964) 11.935 ton
Nedlloyd Korea (1964) 11.925 ton

GUY-S M
Europa/Suriname
in samenwerking met:
- Scheepvaartmaatschappij Suriname

dienst is verweven met de GUY-LA waarin
de Nedlloyd Kobe en Nedlloyd Korea varen.

C = cellular
M = Multipurpose

3.0

PROSPECTIVE CONTAINER PORTS.

WAITING FOR CONTAINER TERMINALS

3

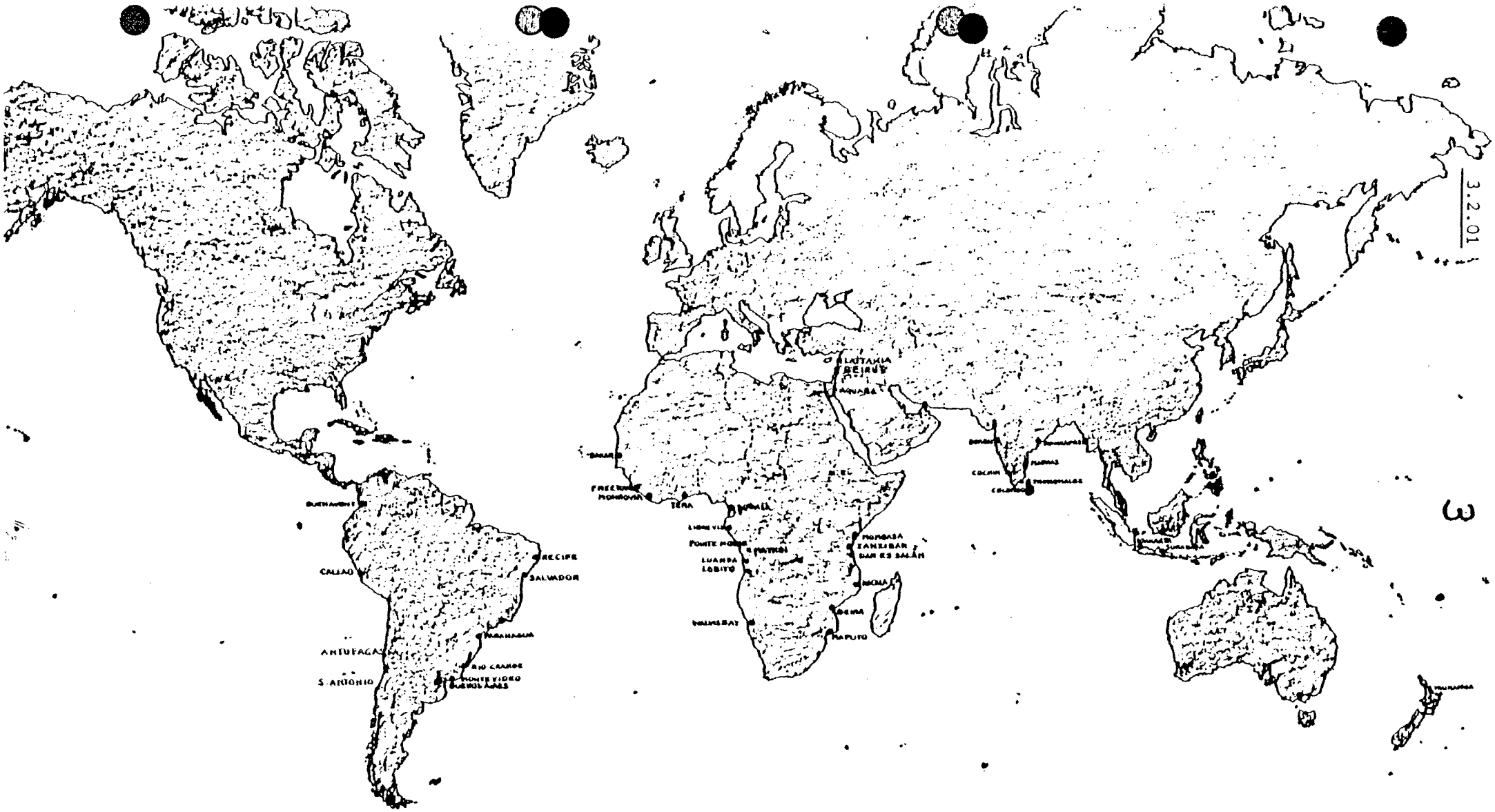
PORTS WITH RESTRICTIVE FACILITIES

	Restrictions to the ship: LOA	Draft	Crane
SOUTH-AMERICA WESTCOAST			
BUENAVENTURA	210 M	30'→35'	?
GUAYAQUIL	210	31'→35'	
CALLAO	210	34'	?
ANTOFAGASTA SWELL	210	30'	FUTURE
SAN ANTONIO SWELL	210	30'	FUTURE
MIDDLE EAST			
AQUABA			NO CRANE
BEIRUT			NO CRANE
LATTAKIA			NO CRANE
WEST AFRICA			
DAKAR	NO RESTR	33'	
FREETOWN	NO RESTR	32'	25 T MOBILE
MONROVIA	200	30'	40 T MOBILE
ABIDJAN	>200	37½'	
TEMA	205	32½'	
LOMÉ	200	31'	
APAPA	200	34'	2 NOT ACTIVE
PORT HARCOURT	>210	>30'	FUTURE
DOUALA	200	28'	
LIBRE VILLE	190	30'	
P. NOIRE	200	34'	35 T MOBILE
MATADI	182	25'	
LUANDA			
LOBITO			
WALVISBAY	192	31'	
MID AMERICA			
CALDERA COSTA RICA		36'	-
P. CORTES-HONDURAS		33'	45 T CONT
KINGSTON-JAMAICA			-3
PORTO PRINCE			
CURACAO			-
TRINIDAD			-
PORTO RICO			-
BARBADOS			NOT YET

	Restrictions to the ship: LOA	Draft	Crane
SOUTH AMERICA EAST-COAST			
RECIFE	220 M	31'	-
SALVADOR	NO RESTR	30'	-
RIO DE JANEIRO	NO RESTR	31'	-
SANTOS	272	32'	1 (2nd u/const:
PARANAGUA	NO RESTR	37'	-
SAO FRANCISCO	230	28'	-
RIO GRANDE DO SUL	NO RESTR	37'	2 (for 20' only)
MONTEVIDEO	NO RESTR	28' F&W	-
BUENOS AIRES	270	27' F&W	1 (35T, 20'/40')
EAST AFRICA			
MOMBASA	190	36'	40 T CONT
ZANZIBAR			×
DAR ES SALAM	192	33'	
NACALA	190	45'	
BEIRA	~180	32'	
MAPUTO	NO RESTR	40'	20 T CRANES
INDIA-CEYLON			
BOMBAY	BEAM<33'	29-30'	
COCHIN	168	30'	
TRINCOMALEE	180	32'	
COLOMBO	244	36'	
MADRAS	229	42'	
VISAKAPHATNAN	168	33'	50 T MOBILE
INDONESIA			
SURABAJA	210	31'	
DJAKARTA	NO RESTR	32'-33'	50 T MOBILE
NEW ZEALAND			
TAURANGA	225	30'-35'	40 T CRANE

Position 1982

(Note: Actual conditions may differ from the above given particulars !)



PORTS AWAITING THE INSTALLATION OF CONTAINER CRANES (SPRING 1982)

4

4.0

CONTAINER TERMINAL PERFORMANCES.

CONTR. TERMINAL

CONTAINER TERMINAL PERFORMANCES

PERFORMANCES ON MAIN VESSELS - PRESENT AND FUTURE.

INTRODUCTION.

In order to respond to future requirements of shipping lines serving a continuously expanding container trade, E.C.T.(') has investigated some characteristics of a number of container liner operators of importance (hereafter annotated with SD, SA and AN).

CALL CHARACTERISTICS.

Tables 1 and 2 show figures concerning main vessel calls of the three operators SD, SA and AN. For comparison's sake and in order to be able to estimate trends, the data of three other services are shown as well.
The figures of operator SD's vessels are also presented per main main vessel (see table 2). Explanatory notes are given on the two pages following next to the tables.

CONCLUSIONS.

Analysis of the operation characteristics has the following outcome:-

- The average performance in Moves Per Hour increases with a rising number of Moves Per Call (see table 1).
- An increasing number of Moves Per Call results in a decrease of the ratio Berth Time/Operation Time and draws near to 1.1 for calls realizing over 1,000 moves (see table 2).
- For some shipping lines there is a performance rate at present of more than 50 moves/hour. An average rate of 60 moves/hour is expected to be reached within the near future.

As is the case with E.C.T., it must be a container terminal's objective to be service- orientated with respect to the shipping lines' requirements regarding performance rates. So far, the performance rate to be realized has been based upon the philosophy that a vessel's call must be completely finished within 24 hours.
For a new container handling facility at present (1983) being built it is E.C.T.'s intention to improve performance rates to 20% over today's by applying even better handling techniques and information control. They intend to reach a rate of 50 moves/hr and more for not only the 1,000-up but also the 500-1,000 moves' calls.

(')... E.C.T. = Europe Container Terminus, at Rotterdam.

TAB. I

CHARACTERISTICS OF A TYPICAL MAIN VESSEL OPERATION AT ECT - ROTTERDAM (1980)

Vessel	2) Number of calls	3) Number of moves (total)	4) Number of moves per call	5) Mean berth-time	6) Mean operation time	7) Ratio berth-time/operation time	8) Average performance in number of moves/berthhour	9) Average performance in number of moves/oper. hour
BP	10	8320	832	23.0	18.7	1.23	36.2	44.4
BS	10	7919	792	22.0	18.3	1.20	35.9	43.3
NDA	10	8283	828	22.2	18.5	1.20	37.3	44.7
NDT	11	8561	778	23.8	18.0	1.32	32.7	43.3
J	10	7429	743	20.4	18.0	1.13	36.5	41.2
K	10	8919	892	29.0	22.3	1.30	30.8	40.0
ME	2	1197	599	21.2	17.7	1.20	28.2	33.8
NHN	10	7636	764	20.7	18.3	1.13	36.9	41.6
N	12	8602	717	18.8	16.2	1.15	38.2	44.2
S	10	7325	733	18.7	16.2	1.15	39.2	45.1
T	10	7031	703	23.7	19.1	1.24	29.6	36.9
Total 11 vessels	105	81222	774	22.2	18.3	1.21	34.9	42.2

See explanatory notes hereafter.

TAB. 2

CHARACTERISTICS OF MAIN VESSEL OPERATION OF MAJOR SHIPPING LINES AT ECT-ROTTERDAM (1980)

Shipping line 1)		2) number of calls	3) number of moves (total)	3) number of moves/call	4) percentage non standard moves	5) mean berthtime	6) mean operation time	7) ratio berth/ operation-time	8) Average performance in number of moves/ berthhour	Average performance in number of moves/ oper. hour
Line X	load	50	12.166	243	15.9	12.6	10.0	1.26	19.4	24.4
	discharge	49	15.939	325	9.6	14.1	9.4	1.50	23.1	34.6
	total	99	28.105	284	12.4	13.3	9.7	1.38	21.3	29.3
Line AN	load	49	17.337	354	9.5	15.0	11.5	1.30	23.7	30.7
	discharge	--	--	--	--	--	--	--	--	--
	total	50	17.773	355	9.5	15.1	11.6	1.30	23.6	30.6
Line SA	load	1	212	212	20.3	20.4	6.3	3.33	10.4	33.9
	discharge	1	162	162	11.1	5.8	4.2	1.50	27.9	39.0
	total	51	25.011	490	11.7	19.1	14.8	1.29	25.7	33.1
Line SD	load	53	36.538	689	8.6	23.5	19.3	1.21	29.4	35.7
	discharge	52	44.684	859	7.0	20.8	17.3	1.20	41.3	49.7
	total	105	81.222	774	7.7	22.2	18.3	1.21	34.9	42.2
Line Y1		Signif.	Signif.	1340	8.4	28.1	25.1	1.12	47.7	53.3
Line Y2		Signif.	Signif.	1675	10.3	31.9	28.5	1.12	52.5	58.7

See explanatory notes (page 4)

EXPLANATORY NOTES TO THE TABLES 1 AND 2

1. Lines X and Y are both shipping lines, served at ECT, Y1 and Y2 refer to different services.

2. These figures give the number of calls of all vessels exceeding 200 metres in length.

3. The number of moves (total and per call) refer to all moves (i.e. including reloads for discharge, shifters, hatchcovers, etc.).

4. This figure represents the amount of shifters, discharge for reload moves, hatchcover and stackingframes and is expressed as a percentage of the total number of moves.

5. Berthtime is defined as the time expressed in hours between arrival and sailing.

6. Operation time refers to the total time that cranes have been allocated to the vessel, regardless of the number of cranes.

7. The ratio berthtime/operationtime is an indication of the time spent in the port for additional activities, such as fuel bunkering, provisioning waiting for tugboats, customs, etc.

8. Load call has been defined as calls with a number of loadmoves, exceeding 80% of the total number of moves.

9. Discharge calls show a number of discharge moves, exceeding 80% of the total number of moves.

10. The total number of calls include discharge, load as well as mixed operation calls.

11. Obviously, SA-Line calls are mixed operations (discharging and loading during one call).

4

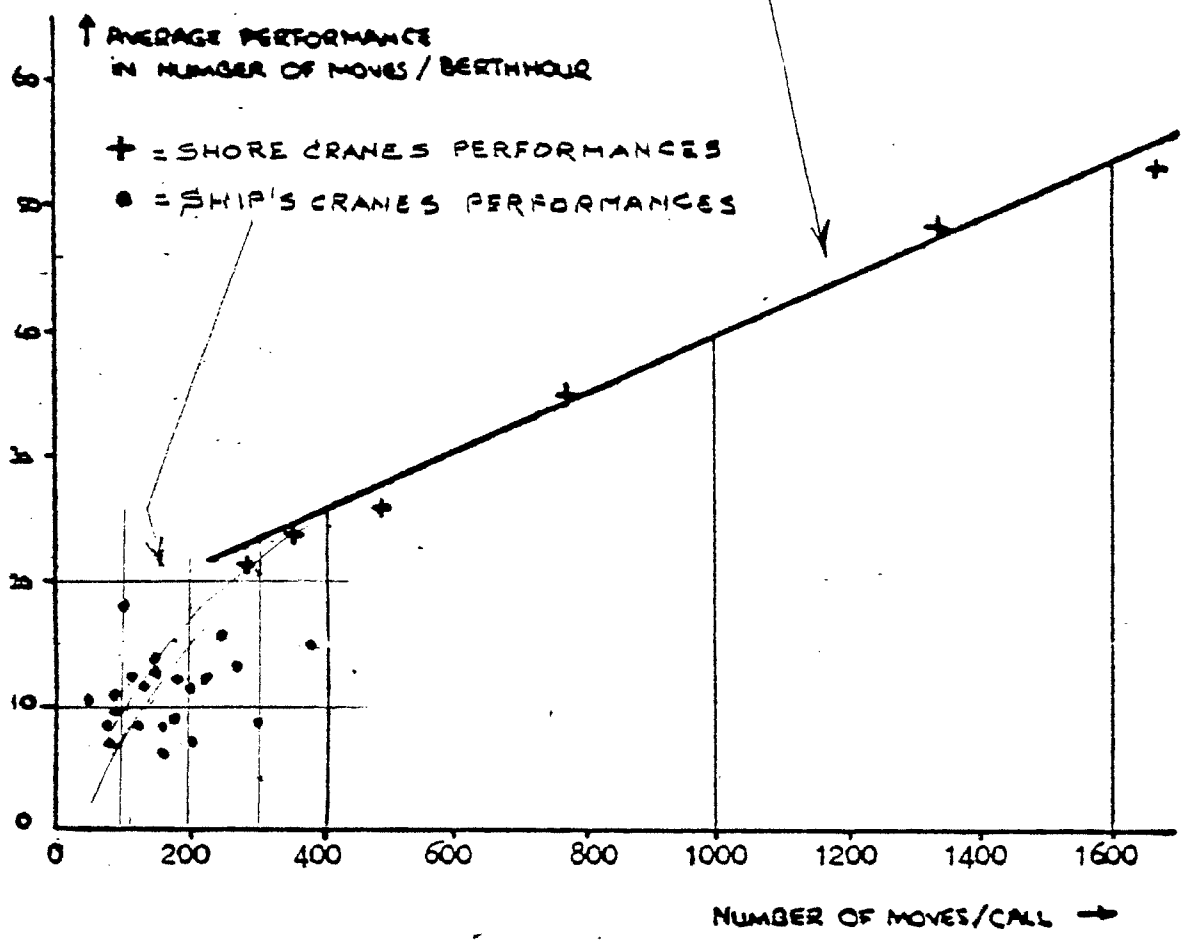
FACTORS INFLUENCING THE LENGTH OF BERTHTIME

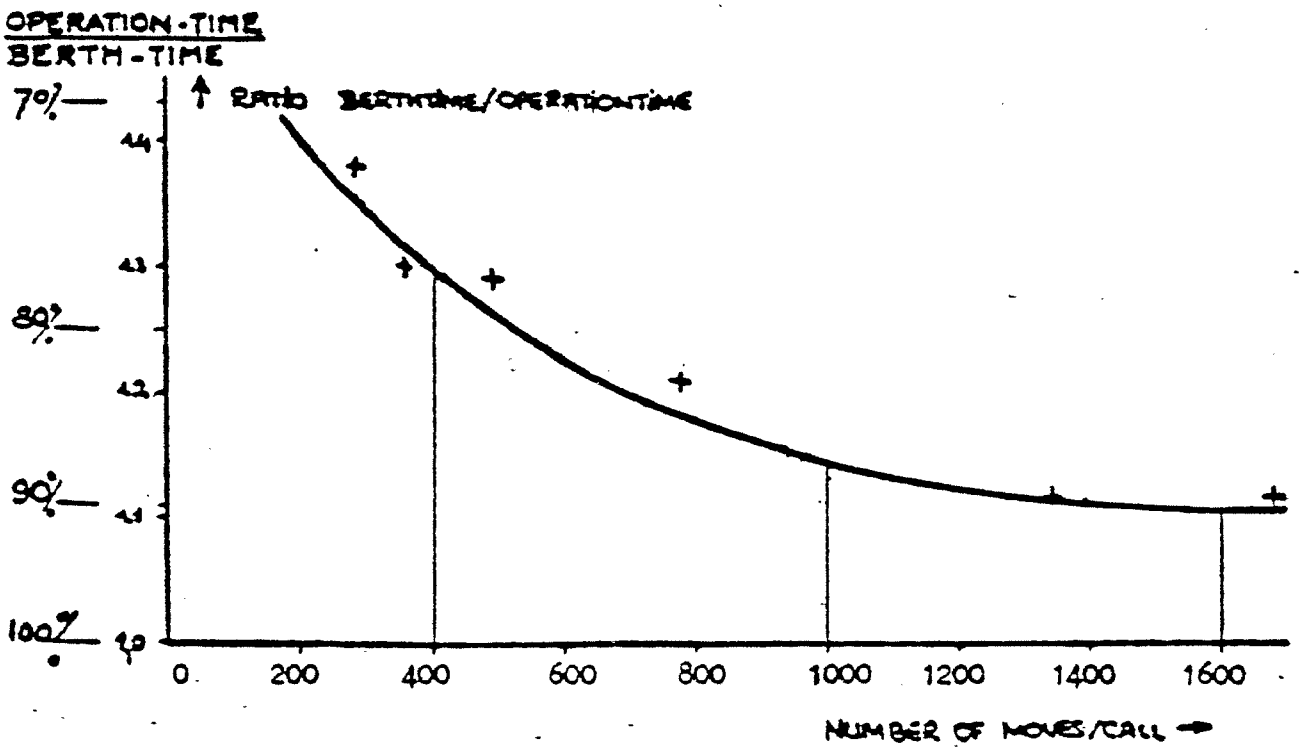
Both shipping lines and terminal operators are interested in the lowest possible berthtime in ports.
The terminal operator is basically responsible for a part of the total berthtime : the operation time.
All kinds of elements may affect the total berthtime and some of such aspects are given below:

- The ships operation time is shorter than estimated.Crew, pilots, tugboats, etc. must be ordered amply in advance and in some cases cannot be reordered.
- During berthtime in the port, the ship's engine room might be serviced. If the operation is faster than estimated, the vessel has to wait for completion of this maintenance service.
- The administrative activities have not been completed by the end of operation. All paperwork, i.e. stowage pan, customspapers, etc. must be ready before sailing.
- After completion of the operation, the ship has to wait for lashing activities, especially in case of labour intensive lashing systems.
- After completion of the operation, the ship has to wait for provisioning, supply of spare parts, fuel bunkering, etc,
- Due to tidal conditions sometimes a vessel cannot leave the harbour immediately after the end of operation.
- Tugboats and pilots are not available (during rushhours).
- A ship has to be in the next port at a fixed time and selects a later time of departure.
- The terminal has no cranes available when entering the port (rush hours).
- Waiting for cargo during operation might stop the operations completely (last minute container, late arrival of feeders).

CONTAINER MOVES PER HOUR

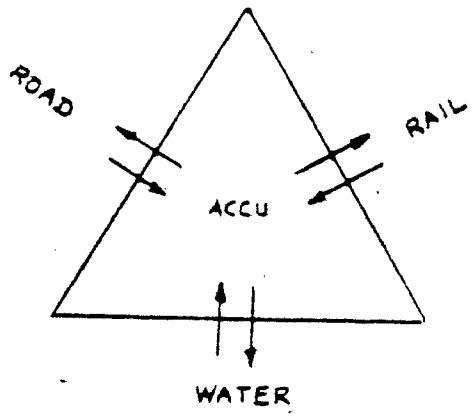
EUROPE CONTAINER TERMINUS - ROTTERDAM.

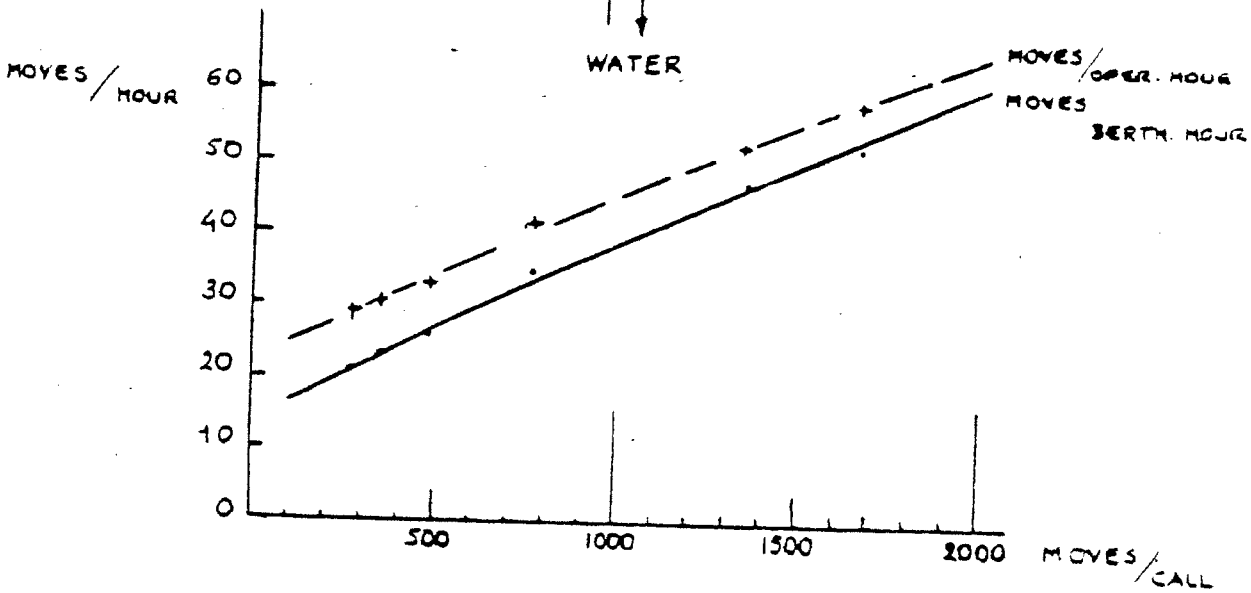


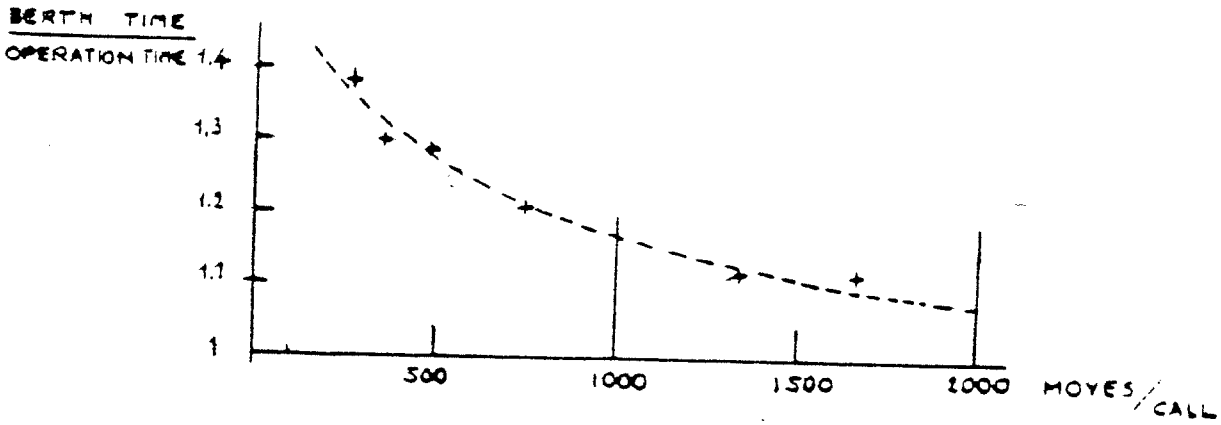


CONTAINER TERMINAL

CONNECTING TRANSPORT CHAIN







GOAL : PORT STAY $\leq$ 24 HRS

$$\frac{\text{BERTH-TIME}}{\text{OPERATION-TIME}} > 1$$

INFLUENCING ELEMENTS

- SAFETY MARGIN BY ORDERING IN ADVANCE : PILOTS TUGS
- SHIP TO BE IN NEXT PORT AT FIXED TIME
- TIDAL CONDITIONS
- NO CRANES AVAILABLE AT ARRIVAL (RUSH HOURS)
- WAITING FOR CARGO LAST MINUTE CONTAINER (LATE ARRIVAL OF FEEDER)
- LASHING
- "OFF-STANDARD" CARGO
  - HEAVY LIFT
  - REEFER
  - DANGEROUS
  - "COPPER IN BULK"
- REPAIR - MAINTENANCE TO SHIPS ENGINES + EQUIPM.
- BUNKERS PROVISIONS SPARES

5.0

MULTI-PURPOSE VESSELS.

MULTI-PURPOSE

(C) THE MULTI-PURPOSE CARGO SHIP.

For the trades between industrialized areas and areas bringing forth a great deal of raw materials and produce where vessels still have to call at ports having conventional wharves only and lacking shore cranes, we still need the (so called) multi-purpose cargo vessel.

This type vessel is characterized by her self-supporting qualities and capability to load a - be it limited - number of ISO boxes which are to be transported along with general breakbulk cargo and the larger kind loose pieces. The shipowner/operator has no choice, because if he does not want to carry the boxes (on account of the ship not being properly suitable for these loads) he then will loose the cargo.

It is our conviction that the combination of general piece-goods cargo and containers on the same ship is wrong by principle because - amongst other reasons - the efficiency of the expensive container system can attain a sufficient level only in case of a speedy turnaround of the vessel and boxes.

A speedy (i.e. short) turnaround time is in principle impossible in case of the old-fashioned cargo ship which handles small quantities of piece goods per each cranecycle. Stay-time in port will normally be some 2 to 5 days whereas in case of container and roro ships this normally should not exceed 1 day per port.

Meanwhile, the breakbulk operation is strongly hampered by the boxes on hatchcovers and in the holds, reason why many containers have to be shifted back and forth on the ship or in the holds, resulting in huge costs.

Also, a small-numbered integrated type crew is hardly feasible because of the many complicated and often clumsy handlings required (with the hatchcovers for instance). The crew costs per cubic foot of cargo transported during the year will therefore be much higher than those of the more productive container/roro ships.

However, the multi-purpose ship will still be needed for years to come in trades where a full container service or roro cannot be applied yet. Actually, this is the most complicated vessel because of hatchcovers and cargo gear.

- Per kilogram lightweight, the multipurpose hull, with all her fittings is much more expensive than a cellular containership hull.
- Vibration and noise problems normally are not so severe, notwithstanding the accommodation being all-aft. Reason for this is the low-level horsepower and the more V-shaped stern above the propeller.

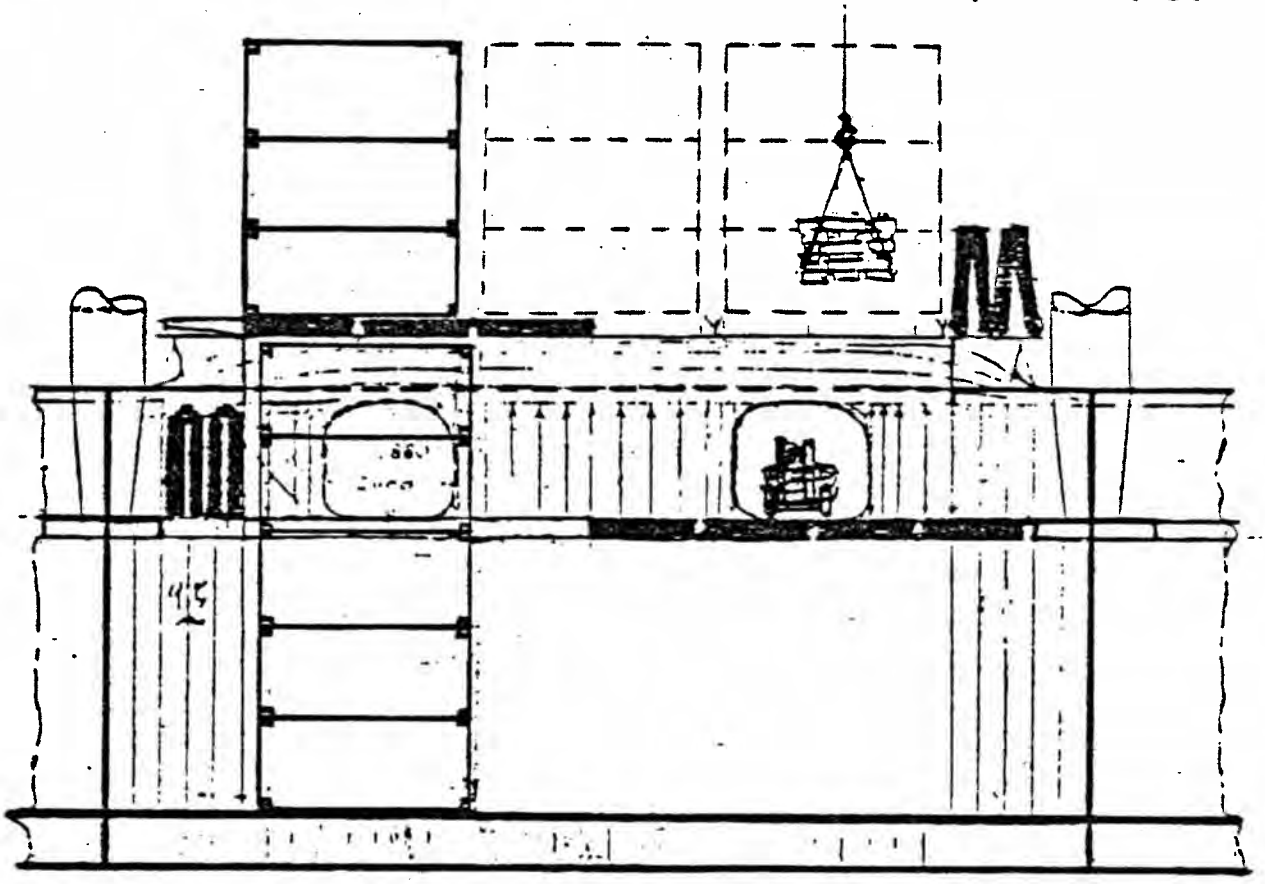
Example : IMPROVED 'NEDLLOYD BAHRAIN' TYPE (reference scheme on next page)

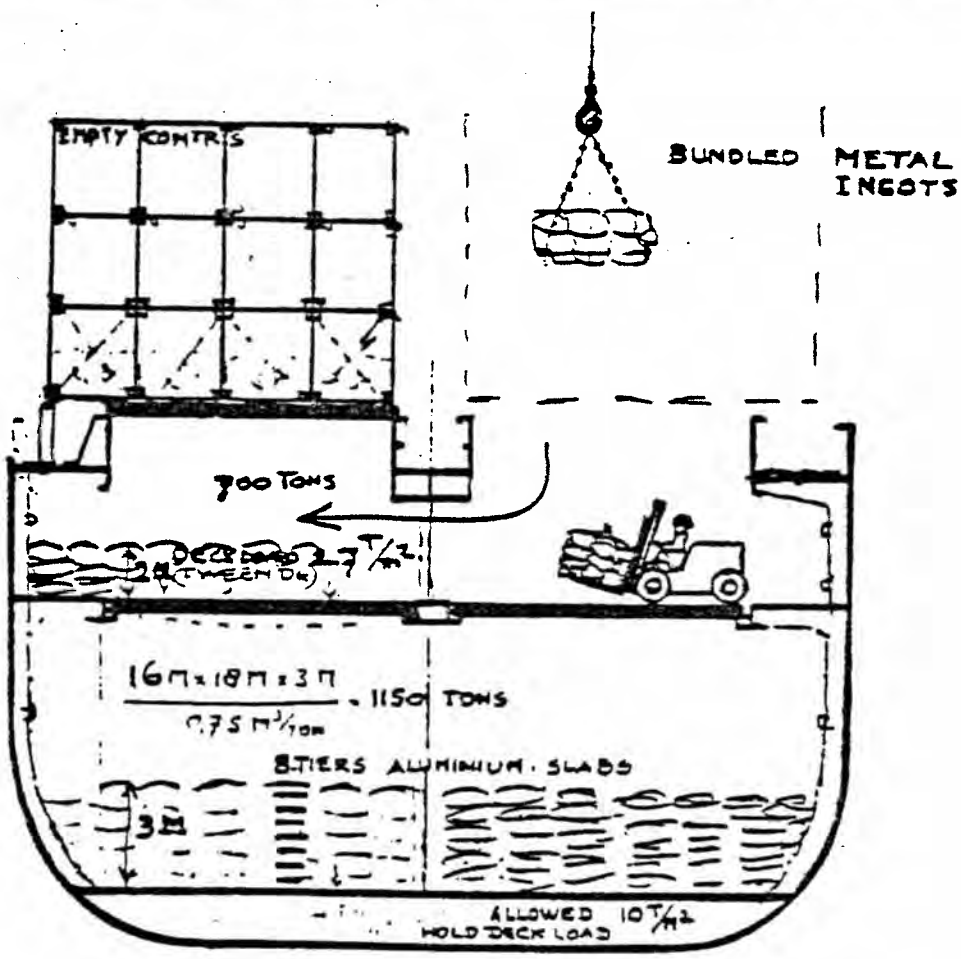
- L x B x D ... 180 x 28 x 16 M. (from 173 x 27.10 x 16 M)
- "Open" hatch type, with square holds, double hull,
- cells in holds numbers I and IV (with reefercells in hold nr IV)
- Two hydraulic folding tweendecks in holds nrs II and III.
- Strong hydraulic folding upperdeck covers, suitable for 50-tons
- stackloads.
- 2 or 3 twincranes of 25 tons capacity each. (altern. one 40-T-cr)

- Speed 16-18 knots.

- For carrying the third tier of containers a large amount of waterballast would be necessary for improving stability. Therefore it is advisable to increase the beam (in our case from 27,1 to 28 or even 30M) and to reduce the block coefficient by cutting off the bottom bilges and make more V-shape sections. "It is better to keep the waterballast in the double bottom outside the steel shell" in order to save horsepower !
- Blockcoefficient from 0.70 to 0.65 .

MULTI PURPOSE_CLUMSY ING





IN HOLD 4

TD 700

LH 1150 +

TOTAL = 1850 TONS ALUMINIUM

CASE HISTORY OF MULTIPURPOSE

(1 9 7 9 through 1 9 8 0.)

EXAMPLE ... JAPAN/KOREA/TAIWAN/HONGK/SING ⟶ W. AFRICA AND V.V.

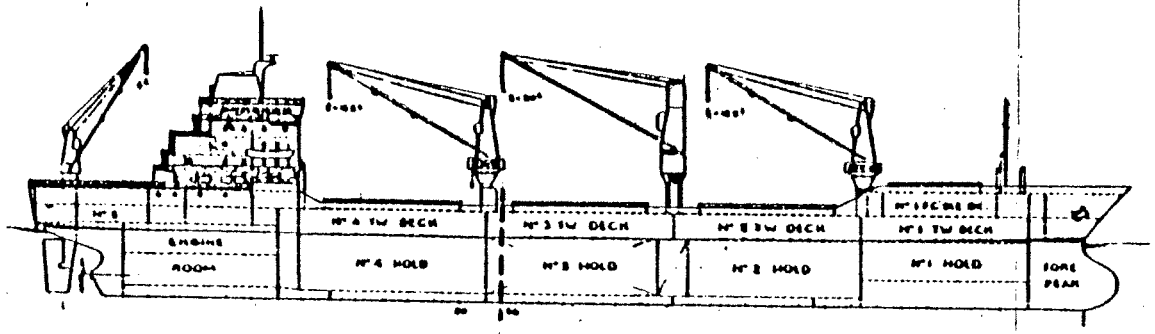
STEEP INCREASE OF CARGO IN CONTAINERS.
BECAUSE THE SHIPPER WANTS THE CARGO IN "BOXES" !

PROBLEM ... HOW TO CARRY BACK THE "EMPTIES", IN COMBINATION WITH TROPICAL-BREAKBULK WHICH IS THE PAYING CARGO

(IN PRINCIPLE.. THE "BOXES" BELONG TO THE VESSEL)

HOWEVER ... LACKING SPACE ON THE SHIP : EMPTIES REMAIN IN W. AFRICA
SAME IN S. AMERICA, ETC.

NEED FOR SPACE ASHORE + STUFFING POSSIBILITIES



Vessel to be cut here

NOV 1979

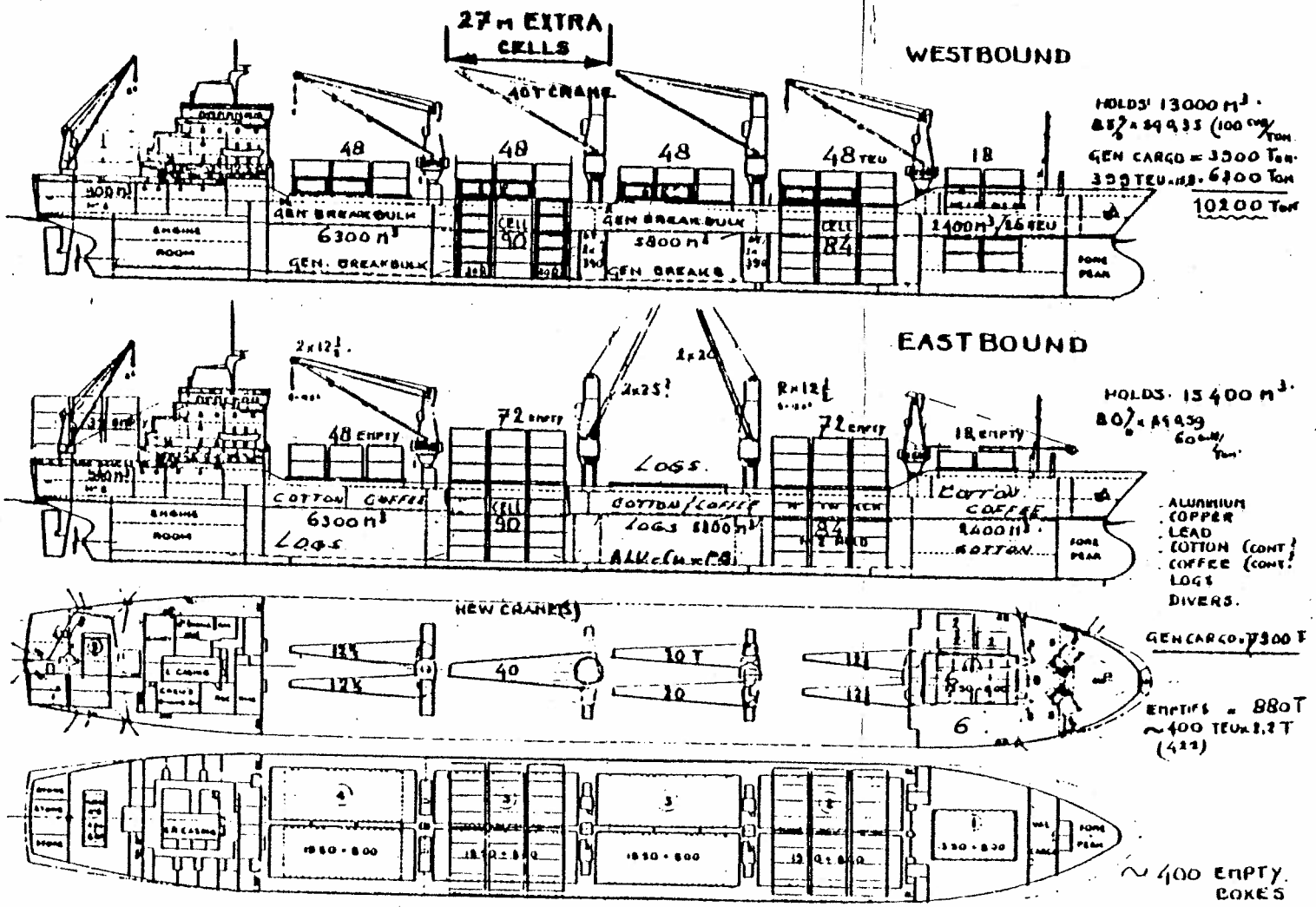
"STANDARD VESSEL": TOO SMALL CUBIC

So, to be jumboised
with cellular hold.

CASE HISTORY OF MULTIPURPOSE

(Continued)

New insert.



AS REQUIRED DEC 1979:

FAR EAST → W-AFRICA

400 TEU + BREAKBULK

W.AFRICA → FAR EAST

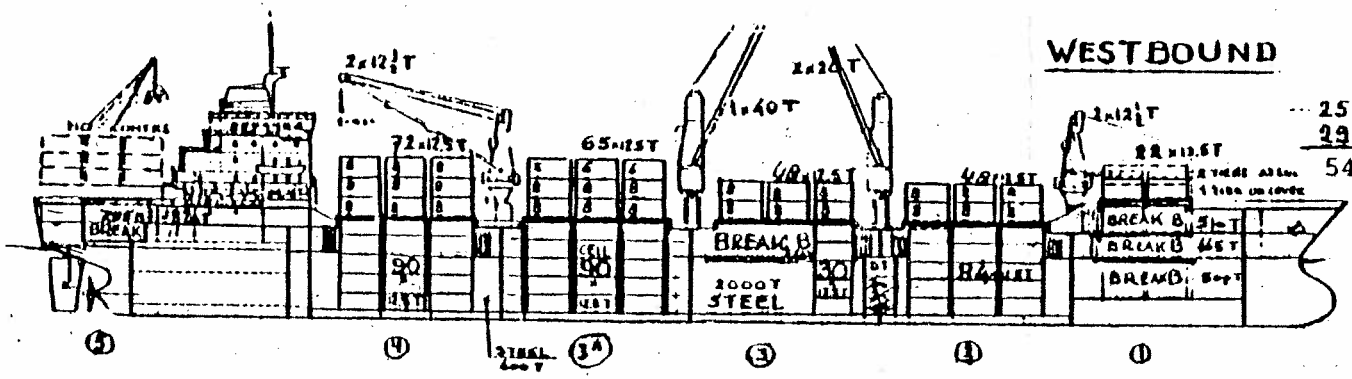
400 EMPTIES + BREAKBULK

5

2

CASE HISTORY OF MULTIPURPOSE

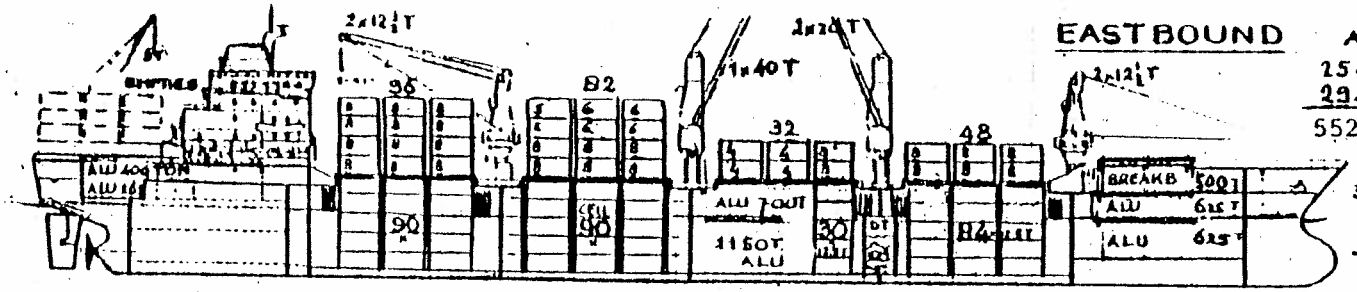
(Continued)



AS REQUIRED AUG 1980:

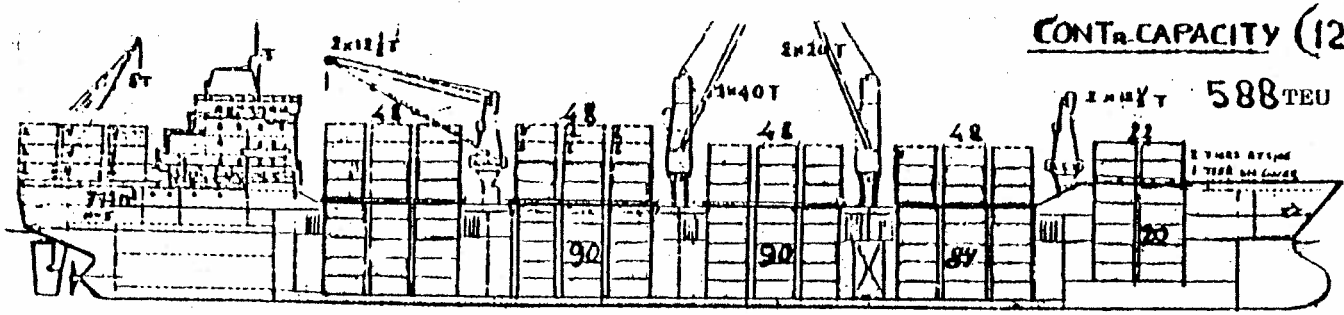
FAR EAST → W. AFRICA

550 TEU + STEEL + BR.BULK



W. AFRICA → FAR EAST

550 EMPTIES + ALUMINIUM INGOT.



"NO Breakbulk anymore"

AS REQUIRED NOV 1980:

588 TEU - 12½ Ton "BOXES" ONLY

5.1.04

5

FAR EAST - W. AFR 3 MONTHS' ROUNDTRIP

PROPOSED CONTAINER SERV

	MILES	PORT DAYS
YOKOHAMA		1
	208	
NAGOYA		1
	234	
KOBE		1
	162	
BUSAN		1
	719	
KEELUNG		1
	471	
HONGKONG		1
	1.459	
SINGAPORE		1
	5 610	
CAPE		
	2 575	
- APAPA/LAGOS		8
	477	
- ABIDJAN		5
	258	
- TEMA		5
	2600	
CAPE		
	5 610	
SINGAPORE		1
	1 459	
HONGKONG		1
	471	
KEELUNG		1
	925	
YOKOHAMA		
	23,438	30

15 PORTS × 2 = 30 DAYS IN PORT

$\frac{23438}{24 \times 16\frac{1}{2} \text{ kn}} = \frac{23438}{396'}$ = 60 DAYS AT SEA

30 DAYS

ROUNDTRIP

3 SHIPS NEEDED

SERVICE MONTHLY

30 SEPT 80

4 MONTHS' R/T.

(M/S "STR. AGULHAS" 1980)

	MILES	PORT DAYS
YOKOHAMA		1
	208	
NAGOYA		1
	234	
KOBE		1
	925	
KEELUNG		1
	471	
HONGKONG		1
	1459	
SINGAPORE		1
	5610	
CAPE		
	1830	
MATADI		1
	774	
DOUALA		2
	211	
P NOIRE		4
	321	
APAPA		6
	146	
LOMÉ		3
	203	
TEMA		7
	157	
ABIDJAN		3
	470	
MONROVIA		6
	258	
FREETOWN		2
	461	
BANJUL		2
	104	
DAKAR		3
	2300	
FETOWE P.N RANGE		2
	1136	
WALVISBAY		2
	6322	
SINGAPORE		1
	1459	
HONGKONG		1
	471	
KEELUNG		1
	925	
YOKOHAMA		
	26455	52
	982	

22 PORTS 52 DAYS IN PORT

$\frac{26455'}{24 \times 16\frac{1}{4}} = \frac{26455}{390'/\text{DAY}}$ 68 DAYS AT SEA

120 DAYS

ROUNDTRIP

4 SHIPS

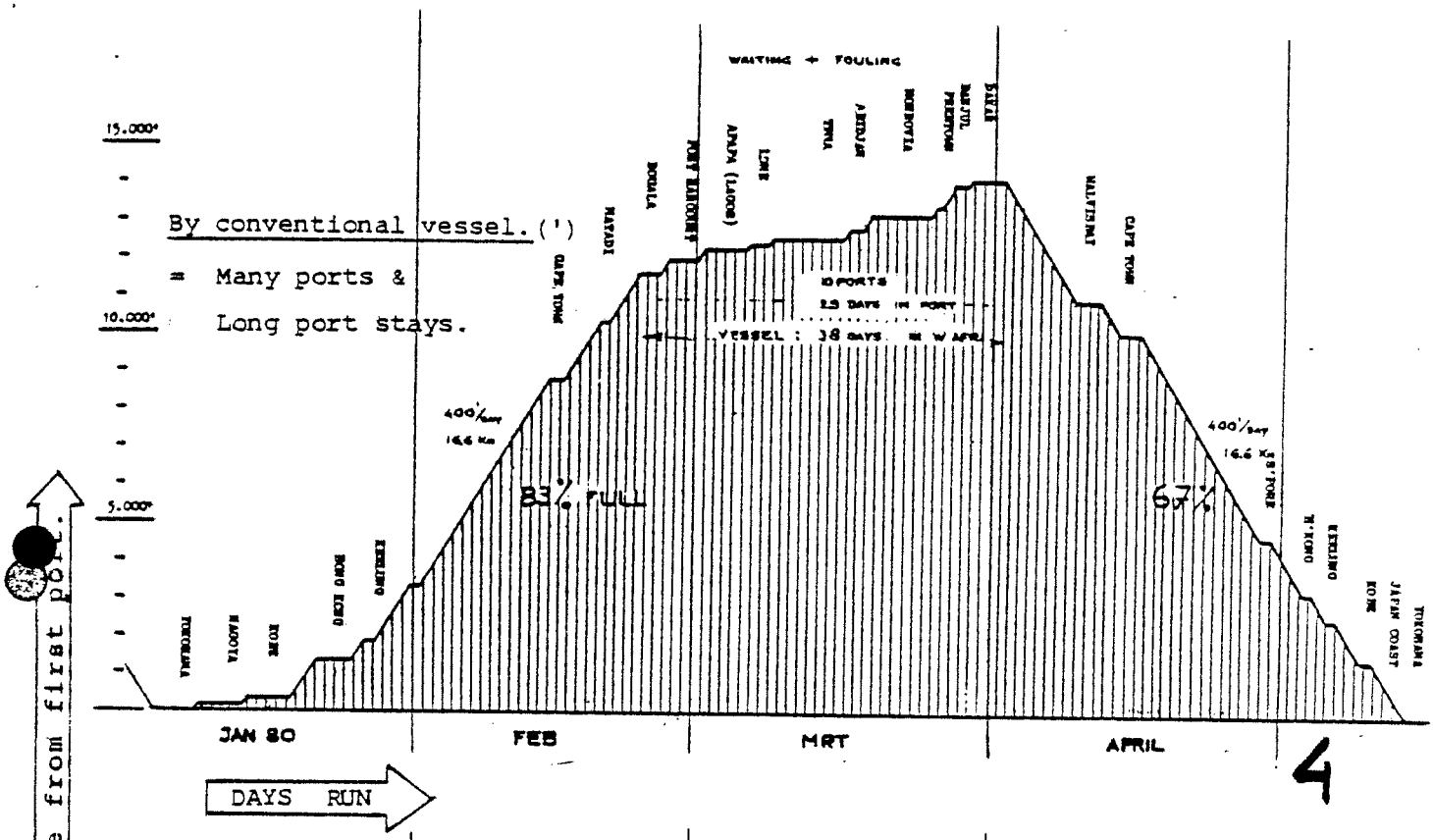
SERVICE MONTHLY

5.1.05

5

REDUCTION OF ROUNDTRIP TIME ----- EXAMPLE .

(As an example the Far East - West Africa &vv run is taken)



(Top figure) Monthly calls by CONVENTIONAL liners ... 4 ships x 120 days each.

(Bottom fig.) Monthly calls by CONTAINER liners ..... 3 ships x 90 days each.

(') m.s. Nedlloyd Agulhas. (") Project.

Some essential factors derived from analysing ten voyages in the Far East - West Africa service by conventional vessels.

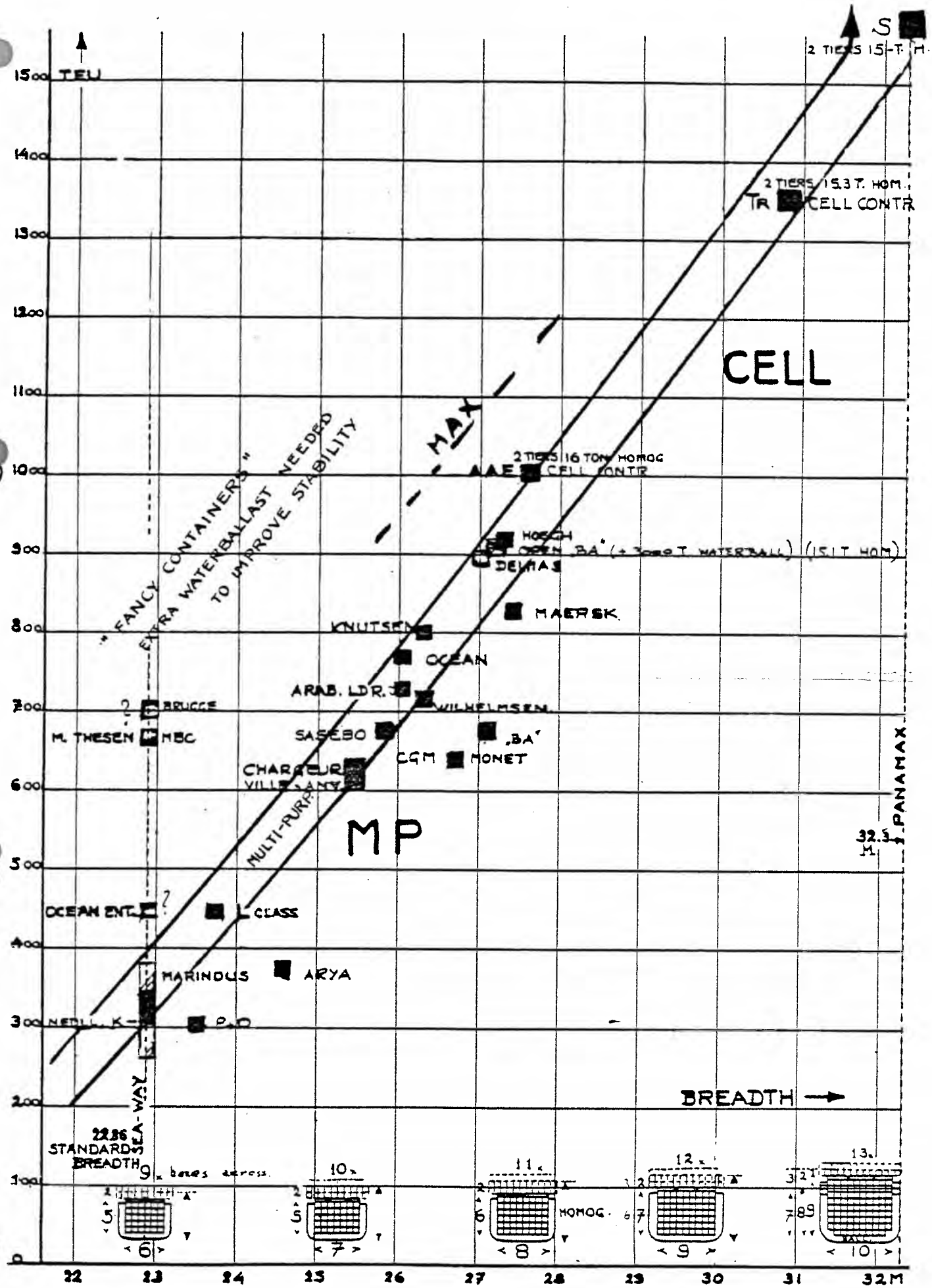
- The service (until the time of analysing) is being run on the basis of a 120-days roundtrip duration (by 4 ships). This way, fixed sailingdates from main loading areas can be maintained in the most appropriate periods for shippers.

- Nevertheless, the actual results of the service are less than those considered feasible.
  Most likely causes of this lag are as follows:-

  (a) Voyage duration is too long.
  This is caused by the low cargo handling output in West African ports resulting in too long portstays.

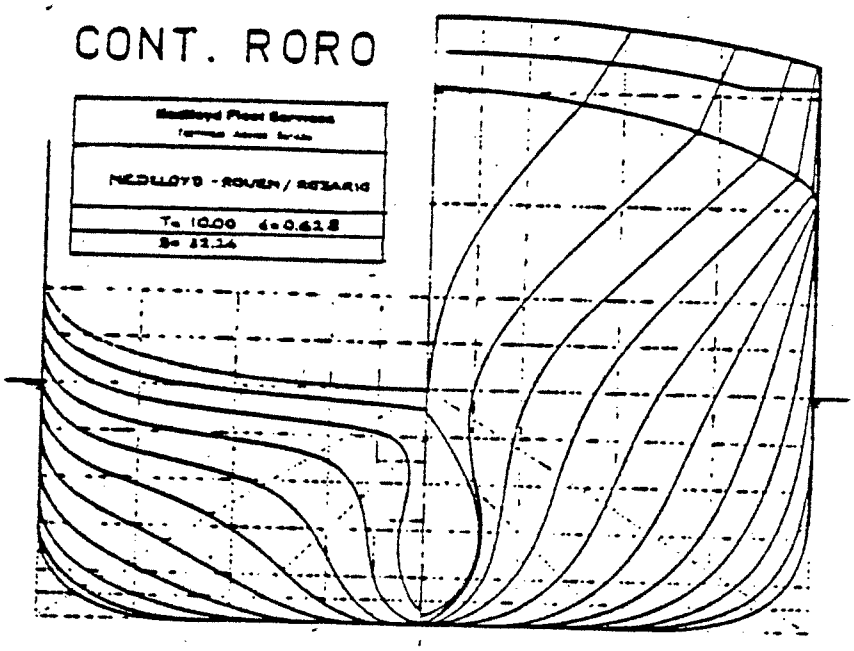
  (b) Too many ports are called at where freight of tonnage landed and/or taken in is insufficient to cover the calling costs - direct and/or indirect.

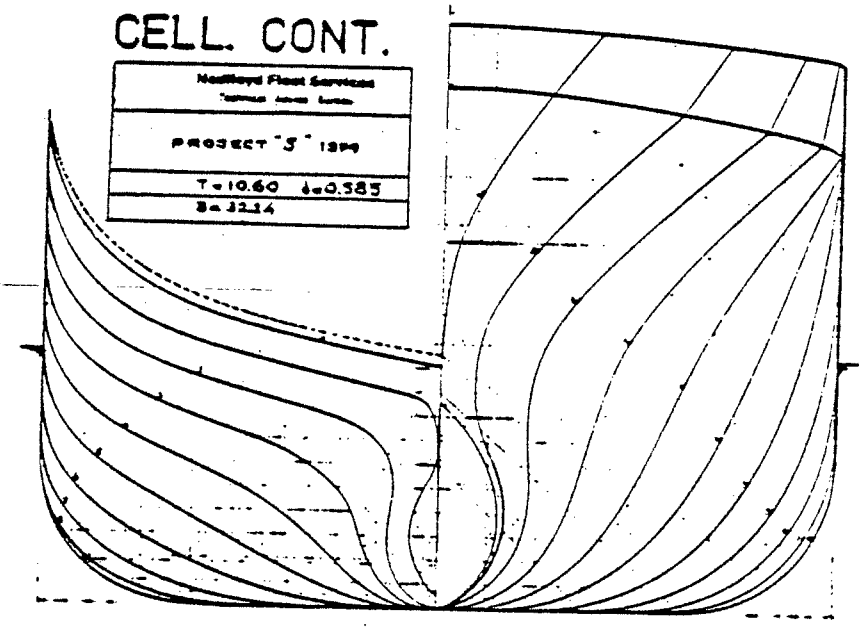
(v.E., Oct. 1979)

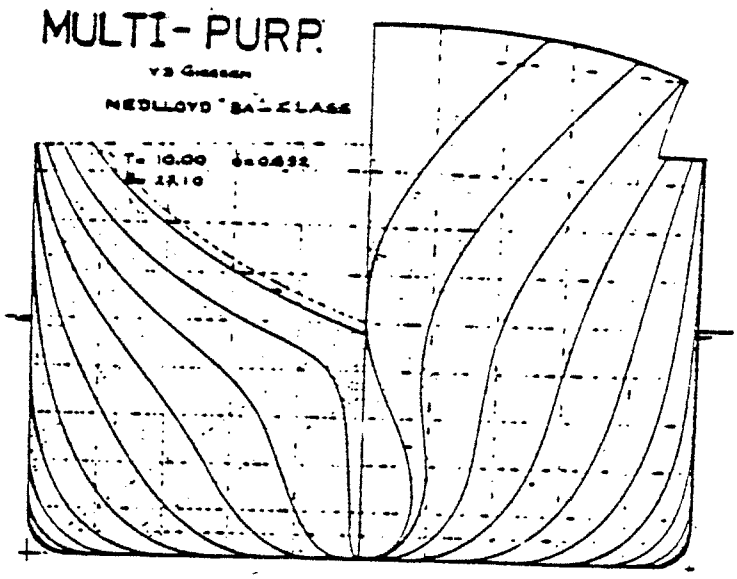


REALISTIC CONTAINER CAPACITY - AS FUNCTION OF SHIP'S BREADTH.

(Selected ships - with "normal" length, depth, δ)





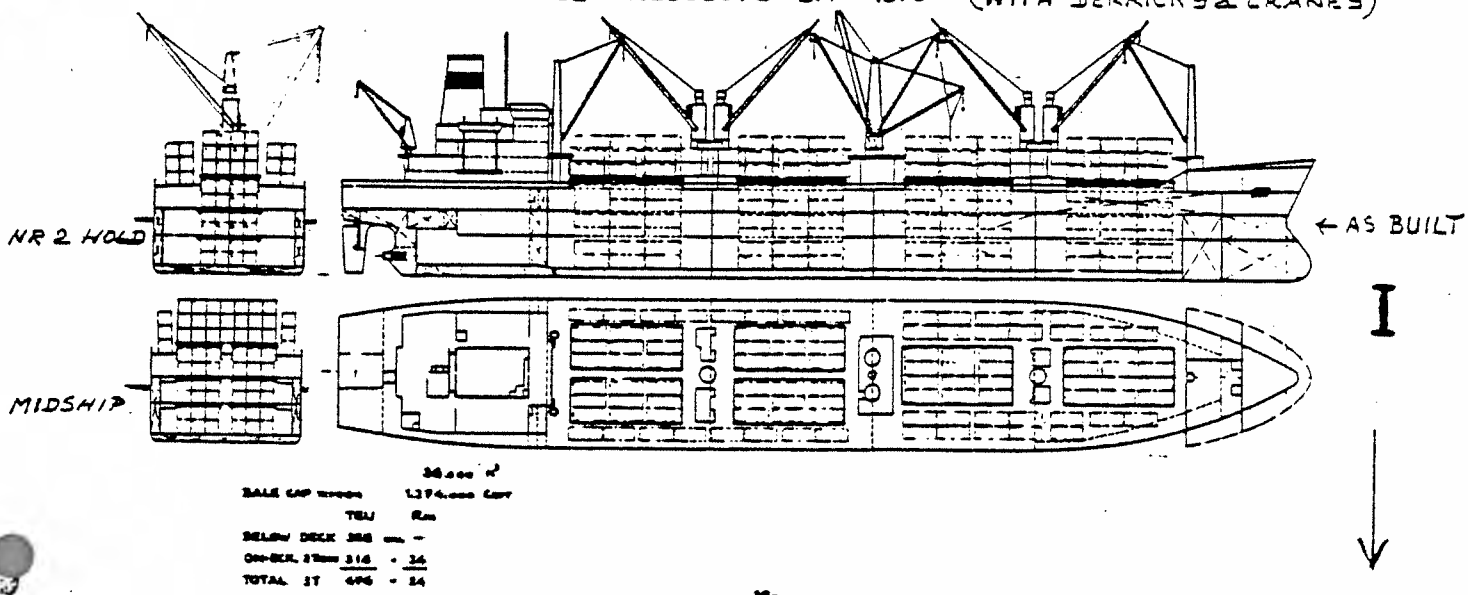


DEVELOPMENT OF NARROW FULL BODY TO WIDER V PRAM
TO CREATE MORE STABILITY FOR DECK-CARGO

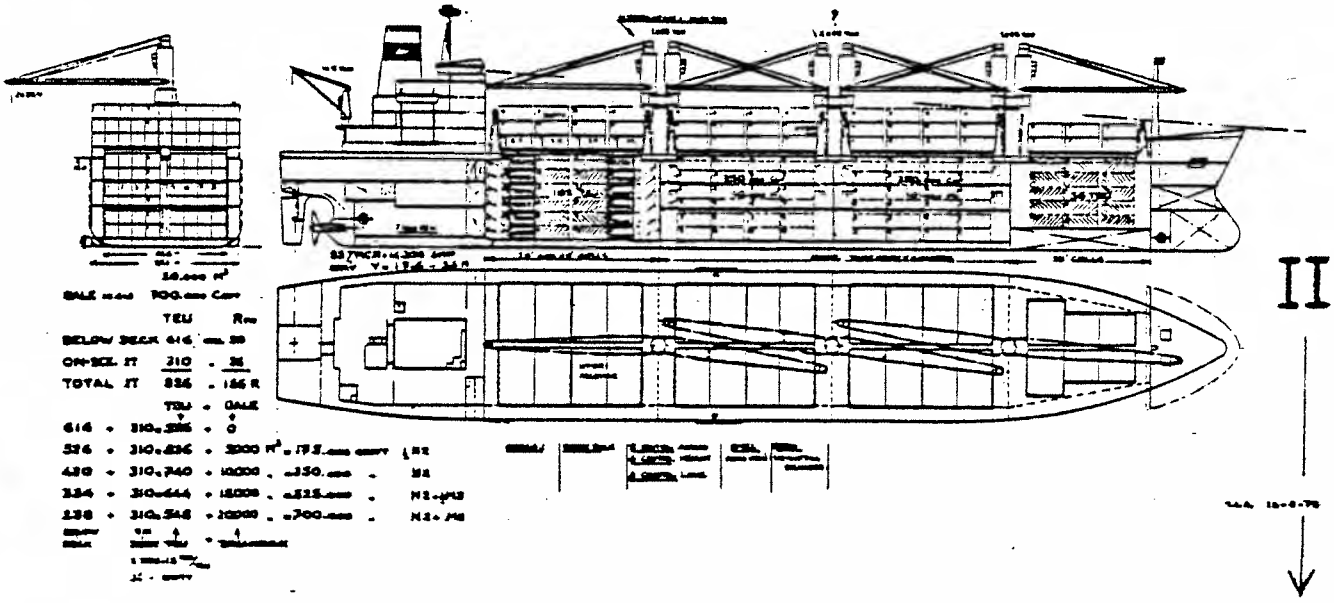
CONSIDERATIONS 5

Same hull - Alternative arrangements - From CONVENTIONAL to CONTAINER type.

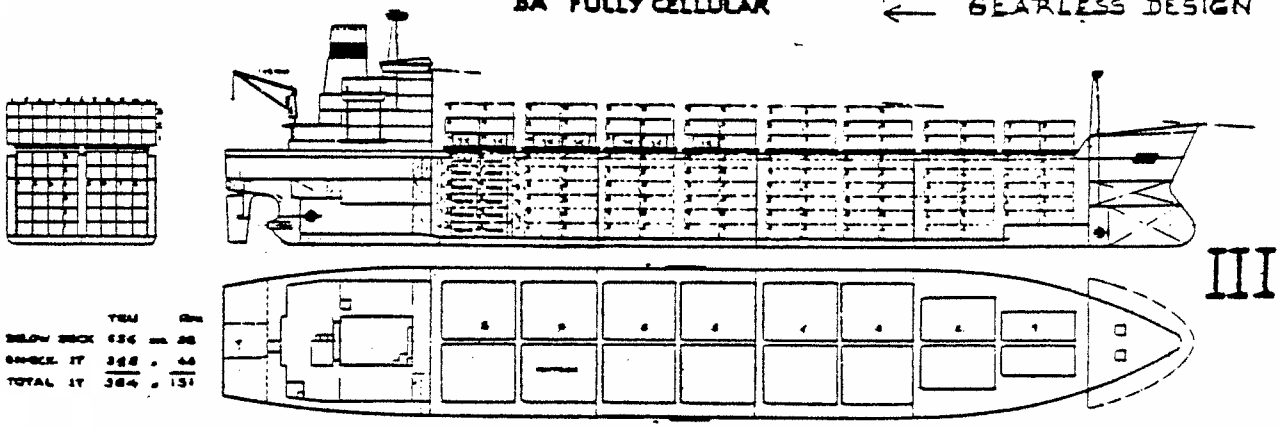
MULTI-PURPOSE "NEDLLOYD-BA" 1976 (WITH DERRICKS & CRANES)



"BA" OPEN HATCH + CRANES ← RE-DESIGNED



"BA" FULLY CELLULAR ← "GEARLESS" DESIGN

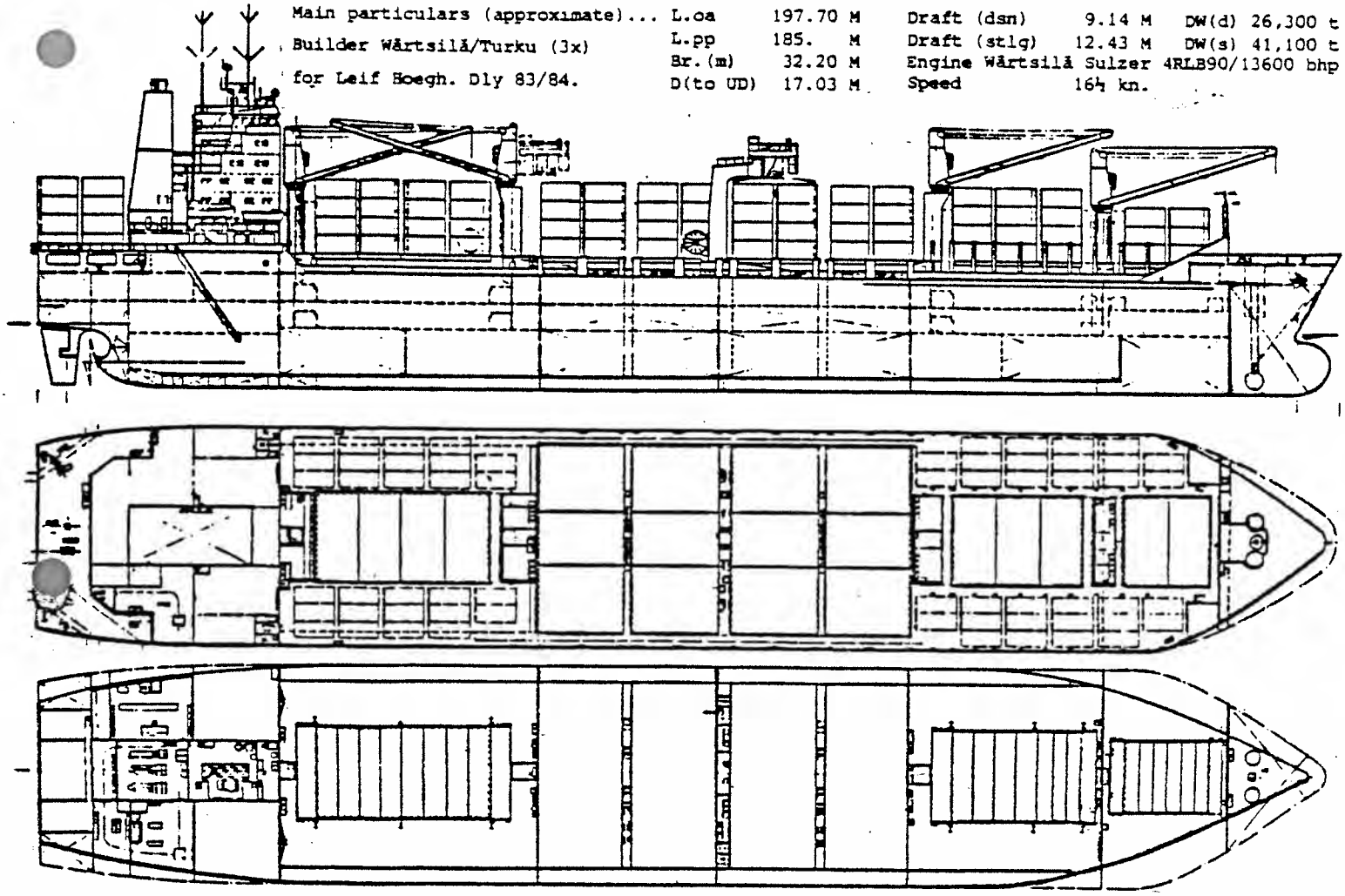


		Utilisation max. Breakbulk	Utilisation max. Containers
(I)	As built, 1979 (greatly CONVENTIONAL)	36,000 M3 + 318 TEU	676 TEU.
(II)	Possible alternative (MORE BOX cap.)	20,000 M3 + 492 TEU	926 TEU.
(III)	As FULL BOX carrier (but GEARLESS)	→	984 TEU, and HEAVIER BOXES.

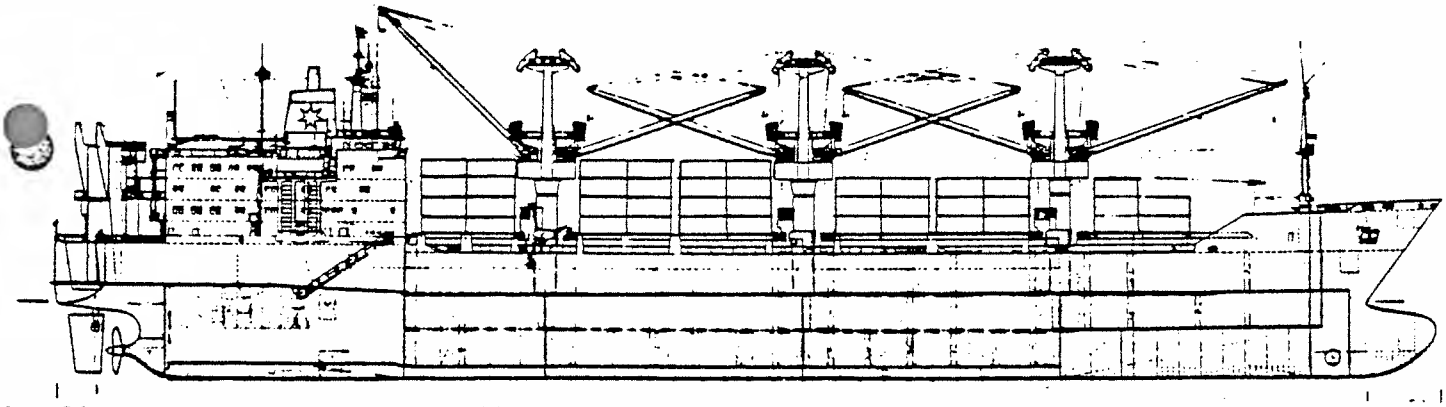
(Self-sustained) PART-CONTAINER / 5 BREAKBULK CARRIER (Newbuilding)

Main particulars (approximate)... Builder Wärtsilä/Turku (3x) for Leif Boegh. Dly 83/84.

L.oa	197.70 M	Draft (dsn)	9.14 M	DW(d)	26,300 t
L.pp	185. M	Draft (stlg)	12.43 M	DW(s)	41,100 t
Br. (m)	32.20 M	Engine Wärtsilä Sulzer 4RLB90/13600 bhp			
D(to UD)	17.03 M	Speed	16¼ kn.		



PART RORO / PART CONTAINER-FRIENDLY BREAKBULK VESSEL (mv Eleo Maersk)



uilders: Odense Staalskibsvaerft A/S, Odense-Lindø
wner: A. P. Møller, Copenhagen
ard number: L 73
ype: "Caroliner"
elivery: April 16th, 1979

onnage:

freedecker	13 706 GRT
	6 856 NRT
full scantling vessel	21 799 GRT
	13 995 NRT

eadweight:

freedecker	18 500 t
full scantling vessel	29 750 t
eng. o. a.	182 25 m
ength b. p.	167.00 m

Moulded breadth	27.43 m
Depth to upperdeck	17.50 m
Depth to 2nd deck	11:30 m
Depth to 3rd deck	6.49 m
Draught:	
freedecker	9.76 m
full scantling vessel	11.85 m

Speed (10 565 kW): abt. 18.5 kn

Classification:
L.R. + 100 A 1 + LMC, UMS

Propelling machinery:
Sulzer-Diesel engine
type: 7 RND 76 M
11 739 kW (15 960 BHP) at 112 rpm
Fuel consumption: 152 g/BHP/h
bow thruster 736 kW/1000 HP (KaMeWa)

Crew: 35 persons
4 twin hatches upperdeck
Hatch covers: system MacGREGOR
stern ramp S. (MacGREGOR)
Loading gear: 1 twin crane 2×16 t and 2 twin cranes of 2×30 t each (Hägglund)
4 mooring winches of 16 t and 2 comb. windlasses/mooring winches of 16 t (Brattvaag)

Total cargo capacity:

grain	1 340 000 cub. feet
bale	1 200 000 cub. feet
Trailer length	900 m
Container capacity:	TEU
in holds	404
on deck	426
Total	830

COMBINATION CARRIERS, OF VARIOUS DESIGN, IN AN ENDEAVOUR TO SUIT THEIR TRADE(S).

6

6.0

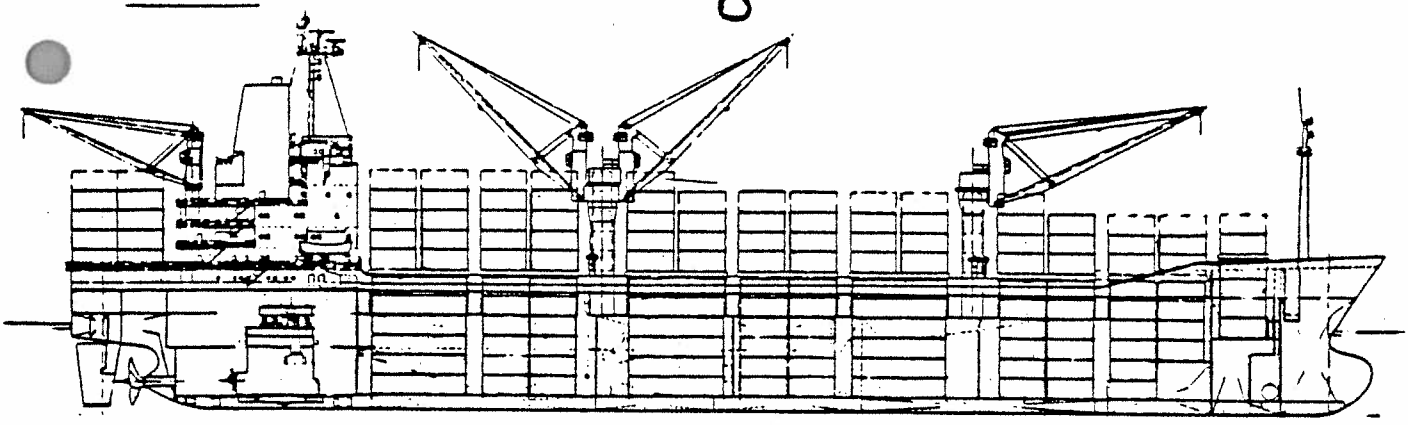
M E D I U M S I Z E CELLULAR CONTAINER-SHIPS

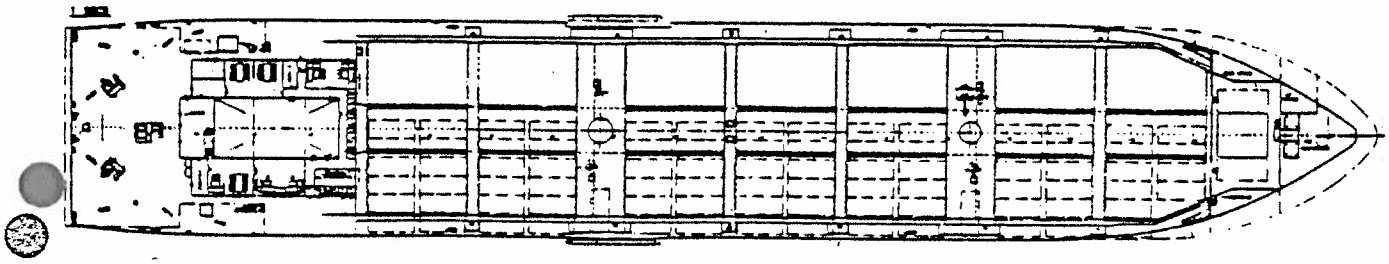
1,100 - 1,600 TEU

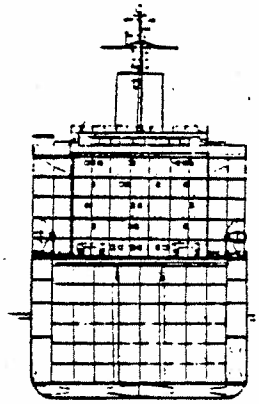
(TEMPORARY) GEARED

BREAKBULK SPACE CAPACITY

MEDIUM-SIZE CELLULAR, GEARED







L.o.a.	173.98 m		
L.p.p.	161.44 m		
Br.(m)	28.40 m		
Depth (1st d)	15.45 m		
Depth (2nd d)	12.45 m		
Draught (svc)	9.32 m		
Draught (S)	11.223 m		
Deadweight	t		
Deadweight (S)	28,149 t		

Cargo gear

aft (jib)	1 x 20 t/22m	
fwd (jib/twin)	2 x 40 t/30m	
fwd (jib)	1 x 40 t/30m	

Main engine	B & W 5-cyl, 2-stroke	
type	5L80GFCA supercharg	
power	11,300 kW(15,400 bhp)/106	
Aux. engs.		
1 st.t.gen.	670 kW	
2 diesel gen.	2 x 936 kW	
1 diesel gen.	690 kW	
1 em. del gen.	78 kW	
Bow thruster 11 t - 736 kW/1000 HP		
Speed(9.32m, 10,170 kW) 18.7 kn.		
G.R.T.	20,360	
Anti-heeling system.		

Hold cubic cap.		
hold 1	894	m3
2	8,546	m3
3	10,939	m3
4	5,860	m3
5	10,673	m3
Total bale	36,911	m3

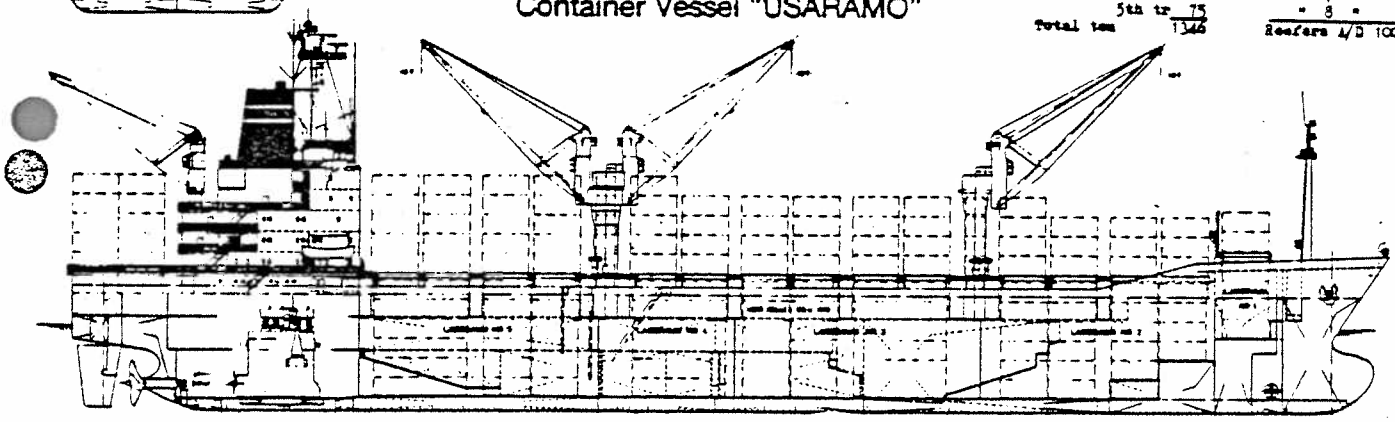
Container cap.(spacewise)

In holds total	610 teu,	
of which 138 x 40'	(45%)	
On deck, 1st tr	168	
2nd tr	176	
3rd tr	176	
subtotal 3 trs	1130 teu	
4th tr	141	
subtotal 4 trs	1271	
5th tr	75	
Total teu	1346	

"USARAMO" type
D.A.L.
Blt A.G.W. '12/82

of which:		
In nr 1 hatch		12
" 2 "		52
" 3 "		80
" 4 "		96
" 5 "		96
" 6 "		96
" 7 "		94
" 8 "		84
Reefers A/B	100 sockets	

Container Vessel "USARAMO"



Builders: Aktien-Gesellschaft "Weser" Werft Bremen
Owner: Partenreederei MS "Usaramo"
Managing owner: DAL Deutsche Afrika-Linien GmbH & Co., Hamburg
Yard number: 1415
Delivery: December 6th, 1982

Tonnage	20 360 GRT
	10 220 NRT
Deadweight	28 149 t
Length o. a.	173.98 m
Length b. p.	161.44 m
Moulded breadth	28.40 m
Depth to 1st deck	15.45 m
Depth to 2nd deck	12.45 m
Draught	11.223 m

Speed (draught 9.32 m, 10 170 kW): 18.7 kn

Classification:
G.L. + 100 A 4 Containerschiff + MC AUT

Propelling machinery:
2-stroke 5-cyl. B & W-Diesel engine
type: 5L80GFCA supercharged
11 300 kW (15 400 BHP) at 106 rpm
propeller (Ostermann).

Auxiliary engines:
1 turbo generator 670 kW driven by A.G. "Weser"-steam turbine
2 Wärtsilä-Diesel engines
type: 8 R 22 HF
1060 kW at 900 rpm each
2 generators 936 kW each
1 Wärtsilä-Diesel engine
type: 6 R 22 HF — 790 kW at 900 rpm
1 generator 690 kW
1 Deutz-Diesel engine
type: F5L 413 FR — 85 kW at 1800 rpm
1 emergency generator 78 kW

separators (Westfalia)
1 CJC-finefilter (Karberg & Hennemann)

rotary vane steering gear type HS 361 D. (Frydenbø)
bow thruster type 100 F — thrust 11 t
736 kW/1000 HP (Jastram)
anti-heeling system

2 radars and echo sounder (Atlas)
gyro compass and automatic pilot (Anschütz)
direction finder, telex, satellite navigator, wireless station and 2 VHF-equipments (Debeg)

Hatches:

hatch 1	6.47 × 11.98 m
hatch 2–8 C. each	12.95 × 5.44 m
hatch 2 P. and S. each	12.95 × 8.20/5.49 m
hatch 3–8 P. and S. each	12.95 × 8.20 m

Hatch covers: system MacGREGOR
Loading gear:
1 crane 20 t/22 m, 1 crane 40 t/30.5 m.
1 twin crane 2 × 40 t/30.5 m (O & K)
4 electric mooring winches of 15/5 t
2 electric comb. windlasses/mooring winches 68 mm K3 chain, 22 t — 15/5 t

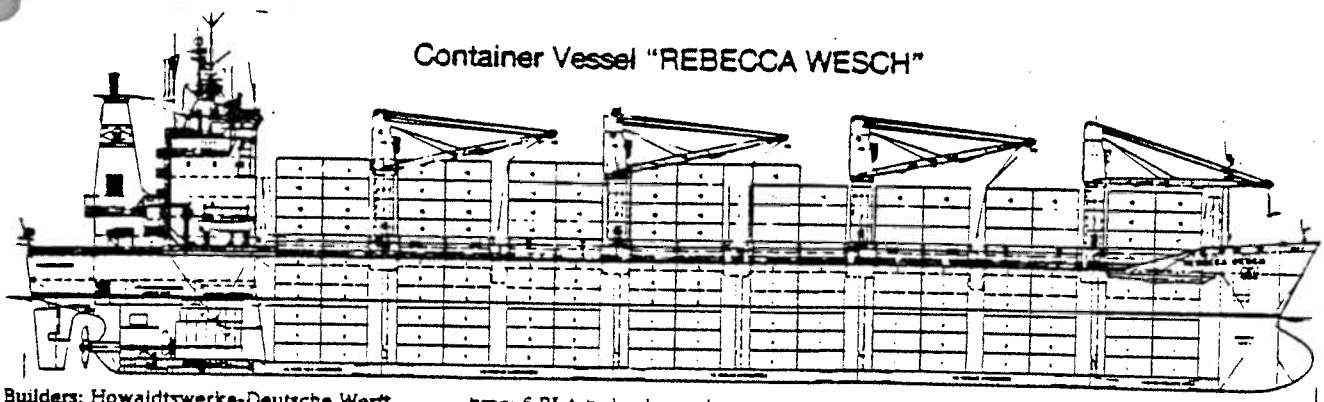
Hold capacity:

hold 1	894.5 m³
hold 2	8 545.6 m³
hold 3	10 939.3 m³
hold 4	5 859.2 m³
hold 5	10 672.9 m³
Total cargo capacity	36 911.5 m³

Container capacity:	TEU
holds	610
on deck	736
Total	1346

100 sockets for refrigerated containers

Container Vessel "REBECCA WESCH"



Builders: Howaldtswerke-Deutsche Werft Aktiengesellschaft Hamburg und Kiel, Werk Ross, Hamburg
Owner: Kommanditgesellschaft MS "Rebecca Wesch"
Managing owner: Reederei Jonny Wesch KG, Jork
Yard number: 167
Type: ore/bulk/container carrier
Delivery: October 2nd, 1982

Tonnage	18 535.24	GRT
	13 046.93	NRT
Deadweight	25 085	t
Length o. a.	169.15	m
Length b. p.	160.00	m
Moulded breadth	25.40	m
Depth to maindeck	15.50	m
Draught	9.964	m

Speed: 14.1 kn

Classification:
G.L. + 100 A 4 E mit Freibord 5.58 m
Containerschiff + MC E AUT

Propelling machinery:
single acting 2-stroke Sulzer-Diesel engine, direct reversible
type: 6 RLA turbocharged
4400 kW at 101 rpm
heavy oil operation 380 cst
propeller (Ostermann)

Auxiliary engines:
3 Yanmar-Diesel engines
type: 6 GL-DT
596 kW at 720 rpm each
3 generators 550 kW, 450 V, 60 Hz each
1 Caterpillar-Diesel engine
type: DE 3304 NTA — 91 kW at 1800 rpm
1 generator 80 kW, 450 V, 60 Hz
separators (Westfalia)
2 radars and echo sounder (Atlas)
gyro compass and automatic pilot (Anschütz)
satellite communication system, Simrad-doppler speed log, wireless station and 2 VHF equipments (Debeg)
satellite navigator
air conditioning plant according to the Duovent-system (Noske-Kaeser GmbH/ Svenska Fläktfabriken)

8 hatches

hatch 1	12.79 × 10.37	m
hatch 2 and 8 each	12.99 × 20.22	m
hatch 3–7 each	12.79 × 20.22	m

Hatch covers: system MacGREGOR

Loading gear: 4 electric hydraulic cranes of 36 t/22 m (Hägglunds)

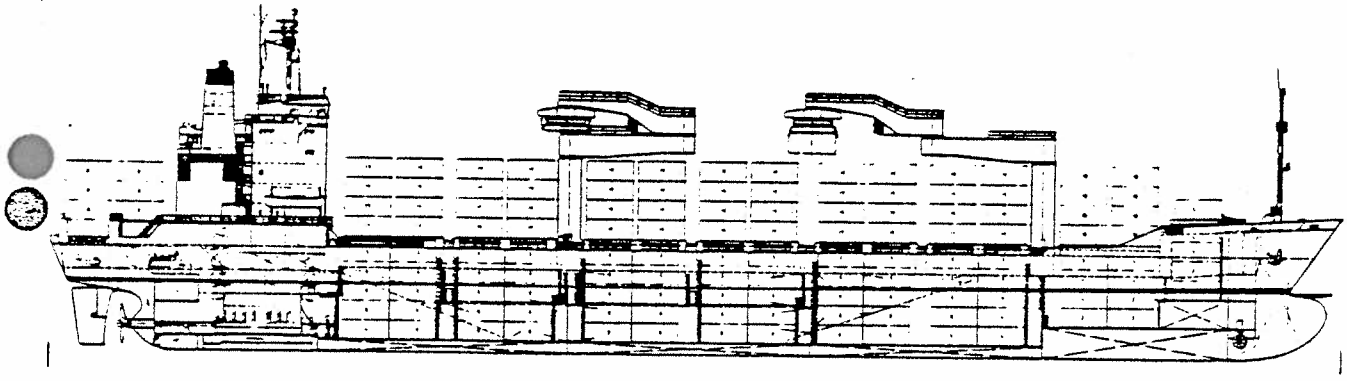
2 electric automatic mooring winches of 15/5 t
2 electric comb. windlasses/automatic mooring winches 62 mm K3 chain, 20/40 t — 15/5 t

Hold capacity:	grain	
hold 1	3 832.4	m³
hold 2/3	9 331.3	m³
hold 4	4 860.0	m³
hold 5	4 860.0	m³
hold 6/7	9 331.2	m³
hold 8	4 967.9	m³
Total cargo capacity	37 182.8	m³

Container capacity:	TEU	or 40'	+ 20'
holds	712	354	4
on deck	517	238	
on deck — empty containers	91	36	
Total	1 320	628	4

32 reefer plugs on deck

Container Vessel "GOWA"



Builders: Flensburger Schiffbau-Gesellschaft, Flensburg
Owner: P. T. Djakarta Lloyd, Jakarta
Yard number: 660/1450
Delivery: July 29th, 1982

Tonnage	15 454	GRT
	10 278	NRT
Deadweight	20 758	t
Length o. a.	170.15	m
Length b. p.	157.00	m
Moulded breadth	28.40	m
Depth to maindeck	14.00	m
Depth to 2nd deck	11.30	m
Draught	9.681	m

Speed: 18.5 kn

Classification:
B.K.I. + 100 I ES + SM ES 0-24/24
Kapal Peti Kemas

G.L. + 100 A 4 E + MC E AUT
Containerschiff

Propelling machinery:
MAN-Diesel engine
type: K 7 SZ 70/125 BL
10 640 kW (14 460 HP) at 130 rpm

Auxiliary engines:
3 Stork-Werkspoor-Diesel engines
type: 6 FEHD 240
770 kW (1050 HP) at 900 rpm each
3 generators 900 kVA each

silencer type ES (Huss)
boiler AQ 10/5 (Aalborg)
2 separators OSA 35 and 2 separators OSA 7 (Westfalia)
compressors (Hatlapa)

bow thruster — thrust 92 kN, 588 kW (Hatlapa)
electric hydraulic rotary vane steering gear HS 280 D (Prydenbø)

Speich-clear view screens (W. G. Schulz)

Crew: 34 persons

Hatches:

hatch 1	7.00 × 10.80	m
hatch 2 P. and S. each	12.80 × 8.10	m
hatch 3–8 P. and S. each	13.26 × 10.80	m

Hatch covers: system MacGREGOR pontoon type

2 cranes of 35 t/30.6 m (Orenstein & Koppel)

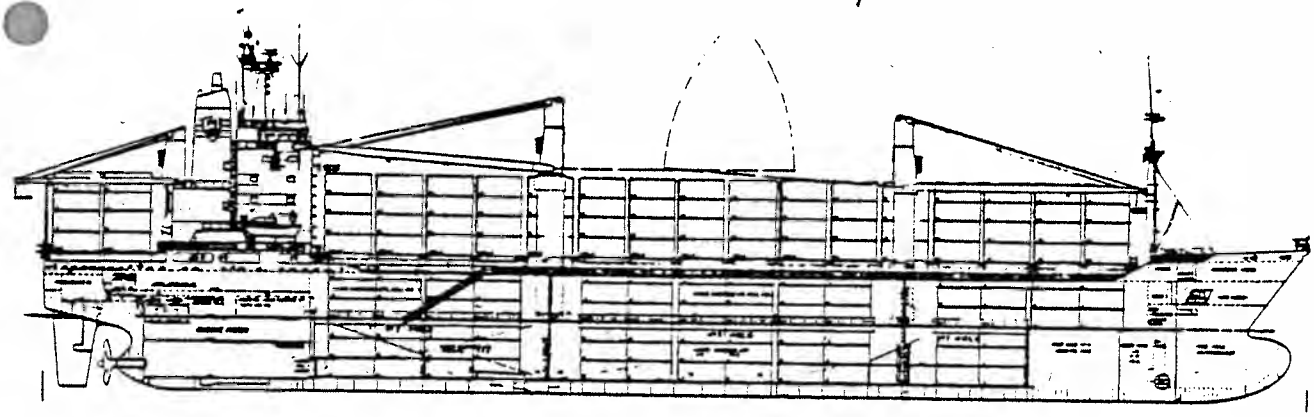
1 electric hydraulic provisions crane 3 t/3.5–18 m

Hydraulic mooring winches (Brattvaag)

Container capacity:	TEU	or FEU
holds	490	234
on deck	662	328
Total	1 152	562

100 sockets for refrigerated containers
Container securing equipment and cell-guides (CONVER)

Container Vessel "CONCORDIA"/"CORONA"



Builders: Thyssen Nordseewerke GmbH, Emden
Owner: Concordia Schiffahrtsgesellschaft mbH & Co. KG, Elsfleth
Managing owner: Containerschiffsreederei H. W. Janssen GmbH, Elsfleth
Yard number: 474
Delivery: December 30th, 1982

Tonnage	17 750.52	GRT
	10 834.21	NRT
Deadweight	25 412	t
Length o. a.	166.04	m
Length b. p.	152.60	m
Moulded breadth	28.50	m
Depth to maindeck	15.80	m
Draught	11.606	m

Trial speed (draught 10.80 m, 9142 kW): 18.1 kn

Classification:
G. L. + 100 A 4 E Containerschiff + MC AUT

Propelling machinery:
single acting 2-stroke B & W-Diesel engine, direct reversible
type: 5L80GFCA supercharged
11 300 kW (15 400 BHP) at 106 rpm
heavy oil operation
Fuel consumption: abt. 50.6 t per day
propeller (Ostermann)
1 shaft driven generator

Auxiliary engines:
3 Wärtsilä Vasa-Diesel engines
type: 624 TS
821 kW (1116 BHP) at 720 rpm each
1 Deutz-Diesel engine
type: F6L 413 FR
96 kW (130 BHP) at 1800 rpm
boiler AQ 10/5 (Aalborg)
separators (Westfalia)
1 CJC-finefilter (Karberg & Hennemann)
bow thruster type 100 F – 882 kW (Jastram)
steering gear (Hatlapa)

gyro compass and automatic pilot (Anschütz)
visual direction finder, weather chart facsimile recorder, wireless station and 2 VHF equipments (Hagenuk)

3 hatches
hatch 1 P. and S. each 25.50 × 10.50/8.12 m
hatch 2 P. and S. each 39.00 × 10.50 m
hatch 3 P. and S. each 25.50 × 10.50 m
Hatch covers: system KVAERNER

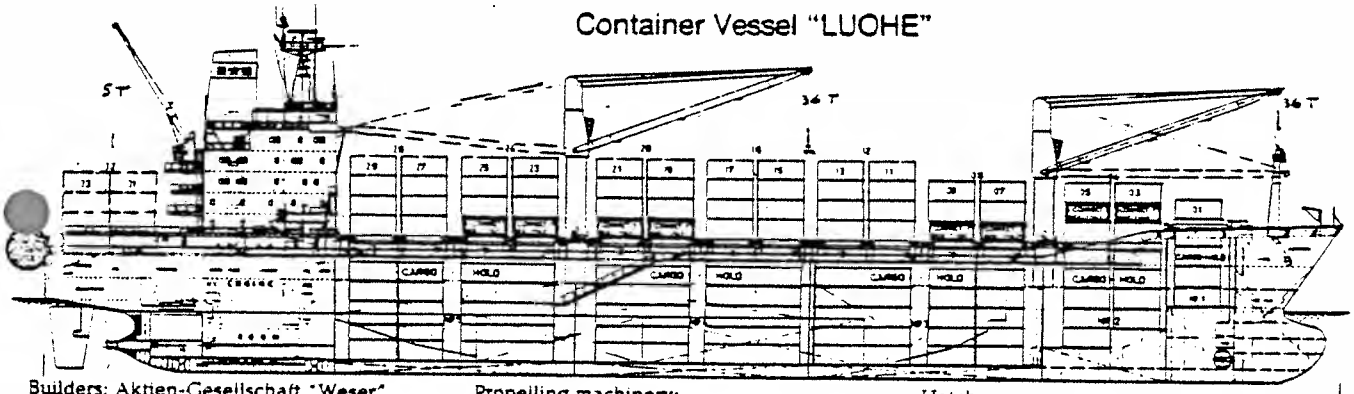
Loading gear: 1 single crane 36/40 t (between hatch 1 and 2), 1 twin crane 2 × 36/40 t (between hatch 2 and 3), 1 single crane 25 t (aft)
4 automatic constant tension mooring winches of 120 kN and 2 comb. windlasses/mooring winches of 194.5/389 kN — 120 kN (TNSW)

Hold capacity:	grain	
hold 1	7 578.4	m³
hold 2	15 755.8	m³
hold 3	10 584.0	m³
Total cargo capacity	33 918.2	m³

Container capacity:	TEU	or 40'	+ 20'
holds	556	272	12
on deck	625	291	43
on deck refr. cont.	101	41	19
Total	1282	604	74

Container lashing and securing material (CONVER)

Container Vessel "LUOHE"



Builders: Aktien-Gesellschaft "Weser" Seebeckwerft, Bremerhaven
Owner: China Ocean Shipping Company (COSCO), Peking
Managing owner: Shanghai Ocean Shipping Company, Shanghai
Yard number: 1032
Delivery: January 17th, 1983

Tonnage	19 915	GRT
	9 263	NRT
Deadweight	26 025	t
Length o. a.	170.02	m
Length b. p.	158.20	m
Moulded breadth	28.40	m
Depth to 1st deck	15.45	m
Depth to 2nd deck	12.45	m
Draught	10.728	m

Trial speed (draught 9.50 m, 8550 kW): abt. 17.7 kn

Classification:
G.L. + 100 A 4 E mit Freibord Containerschiff + MC E AUT

Propelling machinery:
single acting 2-stroke MAN-Diesel engine
type: K7SZ 70/150 CL
11 550 kW (15 715 HP) at 120 rpm
propeller (Ostermann)

Auxiliary engines:
3 Yanmar-Diesel engines
type: 6 GL-UT
736 kW (1000 HP) at 720 rpm each
3 generators 862.5 kVA, 440 V, 60 Hz each
1 MWM-Diesel engine
type: D 226-6 — 56 kW at 1800 rpm
1 emergency generator 70 kVA, 450 V, 60 Hz

compressors (Hatlapa)
bow thruster — thrust 110 kN, 736 kW (Lips)
rotary vane steering gear type HS 280 D. (Frydenbø)
satellite Omega navigator
air conditioning plant according to the Regovent-system (Noske-Kaeser GmbH/Svenska Fläktfabriken)
Crew: 33 persons + 6 cadets

Hatches:
hatch 1 6.45 × 11.98 m
hatch 2—8 C. each 12.93 × 5.44 m
hatch 2 P. and S. each 12.93 × 8.20/5.44 m
hatch 3—8 P. and S. each 12.93 × 8.20 m
Hatch covers: system MacGREGOR pontoon type

Loading gear: 2 electric hydraulic cranes of 36 t, 30.5 m (Hägglunds)
1 electric hydraulic provisions crane 5 t/18 m (NMF)
2 automatic mooring winches of 150/50 kN,
2 comb. windlasses/automatic mooring winches 68 mm K3 chain, 150/50 kN

Container capacity:	TEU	+	FEU
holds	230		190
on hatches	624		
Total	854	+	190 = 1234 TEU

60 sockets for refrigerated containers on hatch covers
Container lashing material and construction of cellguides (CONVER)

CONCORDIA
CORONA

CONCORDIA

VESSEL: Scandinavian Concordia

VOYAGE: 628 westbound

628 WESTB

CONCORDIA
APRIL MAY JUNE 83

PORT Port of Dep.	Port of Arr. (1.)	TIME of DEPARTURE/ARRIVAL From Date	From GMT	To Date	To GMT (2.)	TIME AT SEA Steaming Pilot – Pilot: Hrs	Min (3a.)	Stopped or Anchored: Hrs	Min (-b.)	Man. & pilotage: Hrs	Min (-c.)	obs. DISTANCE Pilot – Pilot: (4a.)	Man. & pilotage (-b.)	Av. SPEED Pilot – Pilot (5)	DRAUGHT Average Mean Draft (6a)	TRIM Average during Passage (-b.)	OIL CONSUMPTION (metric tons) Main Engines Pilot – Pilot Fuel (7a.)	Main Engines MAN. pilotage Fuel	Auxiliary Engines at sea: Fuel (-b.)	Auxiliary Engines at sea: Diesel	Auxiliary Engines In Port of Dep.: Fuel (-c.)	Auxiliary Engines In Port of Dep.: Diesel	Boilers Diesel (-d.)	Total Consumption Fuel (-c.)	Total Consumption Diesel	PER FORM. Tons Fuel per 24 hours (8.)
KOY	NGO	28/4	04.00	29/4	06.30	11	30	5	00	5	00	192	38	16.7	7.2	+1.1	19.6	3.6		1.7				23.2	1.7	40.9
NGO	YON	29/4	15.00	30/4	05.00	8	00			6	00	148	54	18.5	7.1	-0.5	13.4	4.1		1.4		1.1		17.5	2.5	40.2
YON	KOY	2/5	21.00	3/5	23.00	17	00	4	00	5	00	294	52	17.3	7.8	+0.3	24.3	5.1		1.4		7.9		28.0	9.3	34.3
KOY	HAG	5/5	07.00	8/5	20.00	81	00			4	00	1375	23	17.0	8.5	+1.0	119.1	2.1		1.0		4.7		121.2	5.7	35.2
HAG	SIN	9/5	19.00	13/5	07.00	79	00			5	00	1412	41	17.9	9.8	+1.2	134.2	2.9		1.1		3.1		137.1	4.2	40.8
SIN	JED	13/5	15.00	24/5	03.00	247	30			3	30	4331	10	17.5	10.5	+1.0	392.3	1.9		1.1		1.3		394.2	2.4	38.5
JED	AQI	24/5	16.30	27/5	04.00	31	30	21	00	7	00	553	13	17.6	9.8	+0.9	50.6	2.4		3.6		2.2		53.0	5.3	38.6
AQI	SUE	28/5	04.30	28/5	21.30	15	00			2	00	277	13	18.5	9.6	+1.1	25.3	1.6		0.6		4.1		26.9	4.7	40.5
SUE	PSD	29/5	06.00	29/5	17.30					11	30		97		9.6	+1.1		4.0		1.4		1.4		4.0	2.8	
PSD	BEY	29/5	17.30	30/5	07.00	11	30			2	00	208	9	18.1	9.6	+1.1	19.5	1.1		0.6				20.6	0.6	40.7
BEY	LAT	1/6	04.00	2/6	12.30	5	00	24	00	3	30	94	11	18.8	8.6	+0.6	7.5	1.4		3.7		6.2		8.9	9.9	36.0
LAT	LHS	3/6	16.30	4/6	04.00	7	30			4	00	136	11	18.1	8.5	+0.3	12.0	0.6		0.8		4.3		12.6	5.1	38.4
LHS	PAS	4/6	15.30	5/6	21.00	27	30			2	00	514	8	18.7	8.5	+0.7	46.5	1.0		0.7		1.6		47.5	2.3	40.6
PAS	FOS	7/6	08.00	9/6	20.00	57	00			3	00	1061	9	18.6	8.4	+0.3	100.8	1.8		1.6		4.0		102.6	5.6	42.4
FOS	CAS	10/6	17.30	12/6	18.00	46	00	1		2	30	872	9	19.0	7.8	+0.2	77.3	1.8		0.7		2.7		79.1	3.4	40.3
CAS	LIS	13/6	18.00	14/6	16.30	20	00			2	30	315	14	15.8	7.9	+0.4	32.3	2.0		1.0		3.2		34.3	4.2	38.8
MARCH APRIL EB Total						645	30	7	18	95	06	11378	579	17.6			1053.3	35.75		29.6		43.56		1082.05	73.11	39.16
→ WB Total						665	00	54	00	68	30	11782	412	17.7			1079.7	36.0		22.4		47.8		1115.7	70.2	38.96
Grand Total						1310	30	61	18	163	36	23160	991	17.67			2133.0	71.75		52.0		91.36		2204.75	143.11	39.06

11.67

9. REMARKS:

WESTBOUND FAR EAST → MIDIT

1282 TEU × 9.5 TON/TEU = 12180 TON

	Fuel	cSt at 50°C	Diesel
Stock at beginning of voyage:	1035.8		270.2
Bunkered, port: SIN date: 13/5	697.9	180	
Bunkered, port: JED date: 24/5	905.0	140	
Bunkered, port: date:			
Consumption during voyage	1115.7		70.2
Stock at end of voyage	1523.0		200.0

C1 WESTB

6.3.01

Date 24/6/83

Master

VESSEL: „ScanDutch Concordia"

VOYAGE: 657 eastbound

657 EASTB.

CONCORDIA

JUNE JULY 83

PORT		TIME of DEPARTURE/ARRIVAL				TIME AT SEA						obs. DISTANCE		Av. SPEED	DRAUGHT		TRIM	OIL CONSUMPTION (metric tons)										PER-FORM.
Port of Dep.	Port of Arr.	From Date	From GMT	To Date	To GMT	Steaming Pilot – Pilot Hrs	Steaming Pilot – Pilot Min	Stopped or Anchored Hrs	Stopped or Anchored Min	Man. & pilotage Hrs	Man. & pilotage Min	Pilot – Pilot	Man. & pilotage	Pilot – Pilot	Average Mean Draft	Average during Passage		Main Engines Fuel	Main Engines Diesel	Auxiliary Engines at sea Fuel	Auxiliary Engines at sea Diesel	Auxiliary Engines in Port of Dep. Fuel	Auxiliary Engines in Port of Dep. Diesel	Boilers Diesel	Total Consumption Fuel	Total Consumption Diesel	Tons Fuel per 24 hours
1		2.				3a.		-b.		-c.		4a.	-b.	5.	6a.	-b.		7a		-b.		-c.		-d.	-c		8
LIS	ROC	17/6	21 30	18/6	[illegible]	15	30			2	30	284	18	18.3	8.52	+0.46		27.8	1.3		0.5		11.6		39.1	12.1	43.0
ROC	LIV	19/6	21 00	21/6	05 30	47	30	6	00	3	00	577	10	18.3	8.46	+0.13		53.5	1.2		1.7		0.7		55.0	2.4	42.9
LIV	FOS	21/6	20.55	22/6	17.50	43	30	5	00	2	30	230	16	17.0	9.83	+0.31		20.5	1.5		1.4		2.0		22.0	3.4	36.4
FOS	PSO	3/6	10.00	27/6	05.00	81	30			1	30	1505	24	16.8	10.84	+0.57		113.2	1.6		0.6		2.2		116.0	2.3	[illegible]
PSO	SUE	27/6	22.30	28/6	16.30			5	00	13	00		97		10.88	+0.37			5.0		3.1		2.0		5.1	5.1	
SUE	JED	28/6	16.30	30/6	05.30	35	30			1	30	612	11	17.2	10.88	+0.37		55.5	1.0		0.3		-		56.5	1.3	37.5
JED	SIN	30/6	23.00	14/7	23.30	20	00			4	30	4343	16	16.7	10.92	+0.24		344.5	3.1		1.4		2.6		347.6	4.0	30.7
SIN	TAT	13/7	07.00	14/7	07.00	21	00	5	00	4	00	549	10	16.6	9.21	+0.34		54.0	2.7		1.7		1.2		56.7	2.9	34.2
TAT	SGB	15/7	06.30	16/7	10.00	21	00	2	00	4	30	331	31	17.7	8.30	+0.72		34.4	3.8		1.1		4.1		38.5	5.2	34.3
SGB	HKG	17/7	00.00	21/7	12.00	69	00			4	00	1364	45	17.1	8.14	+0.7		168.8	4.3		6.9		2.1		178.1	5.0	37.1
HKG	NGO	21/7	22.00	26/7	03.30	77	00	47	00	7	30	1435	52	18.6	7.91	+1.57		115.5	7.4		4.4		0.7		122.9	5.1	30.1
				JUNE JULY EB Total		101-30		46	00	48-30		12 068	341	17.2				1048.0	33.0		12.1		24.2		1131.0	46.3	37.5
				WB Total																					1115.7	70.2	
				Grand Total																					2246.7	116.5	

9. REMARKS:

EASTBOUND MEDIT → FAR EAST

1125 TEU x 14,2 TON/TEU = 16000 TON

	Fuel	cSt at 50°C	Diesel
Stock at beginning of voyage: 14/6/83	1523		200.0
Bunkered, port: FOS date: 27/6/83			295.0
Bunkered, port: JED date: 30/6/83	100	150	
Bunkered, port: date:			
Consumption during voyage	[illegible]		[illegible]
Stock at end of voyage			

C2 EASTB

25.07.83

Date

Master

ScanDutch Concordia

6.3.02

2

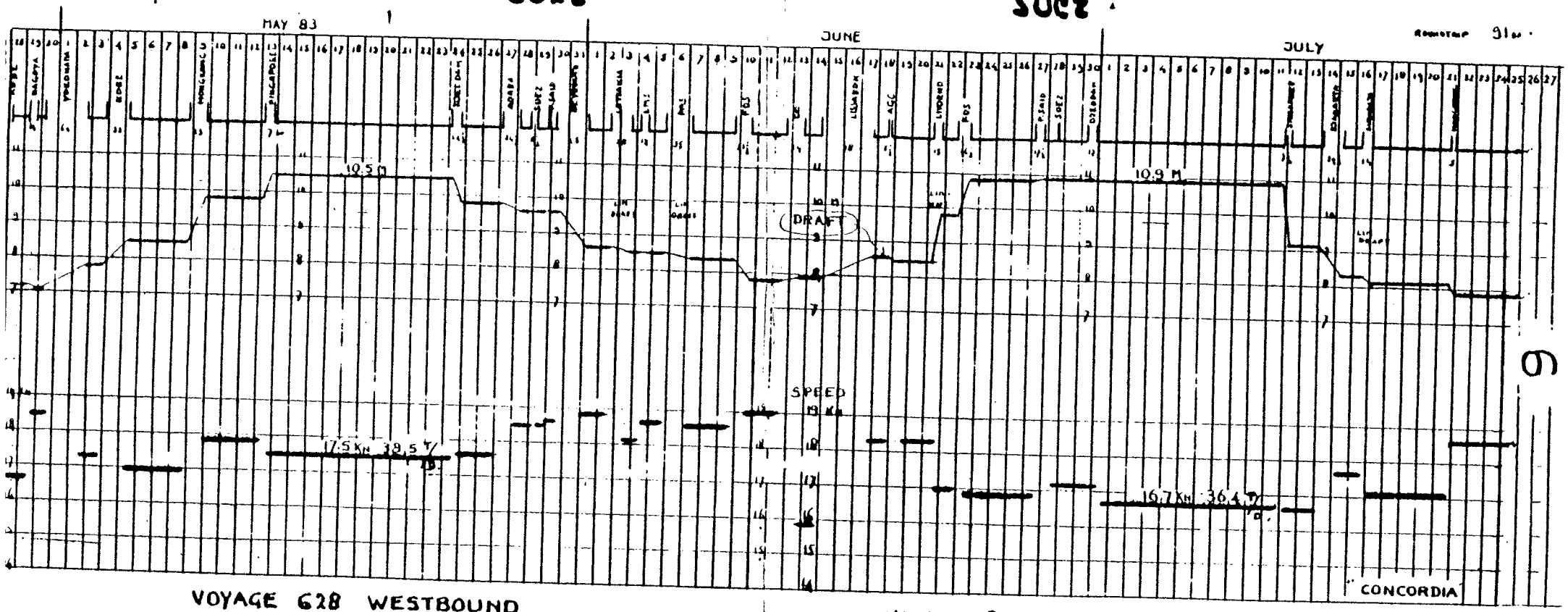
FAR EAST

MEDITERRANÉE

FAR EAST

SUEZ

SUEZ



VOYAGE 628 WESTBOUND

VOYAGE 657 EASTBOUND

6

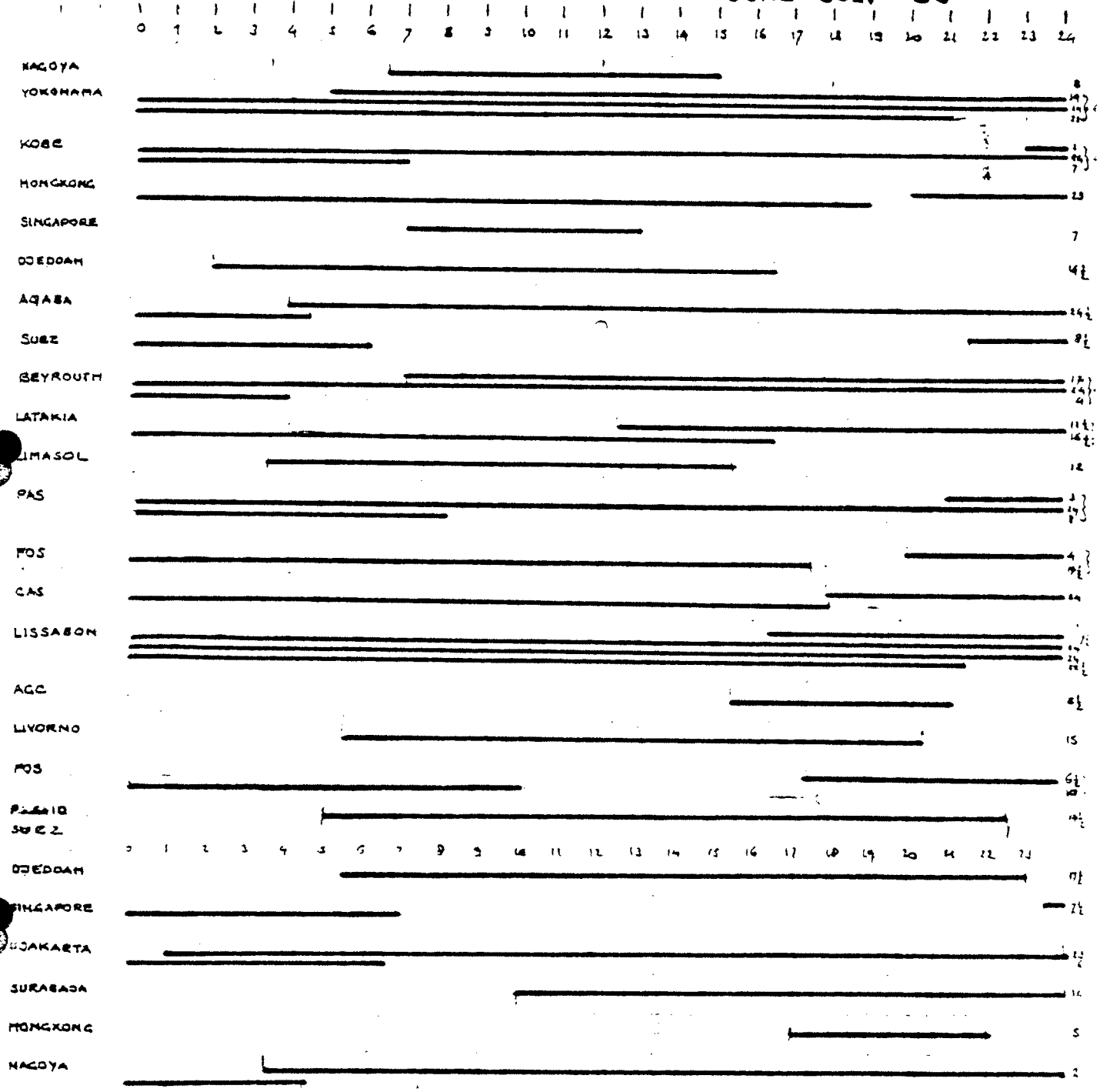
6.3.03

"CONCORDIA"

6.3.04

6

"CONCORDIA" PORT STAY MAY JUNE JULY '83



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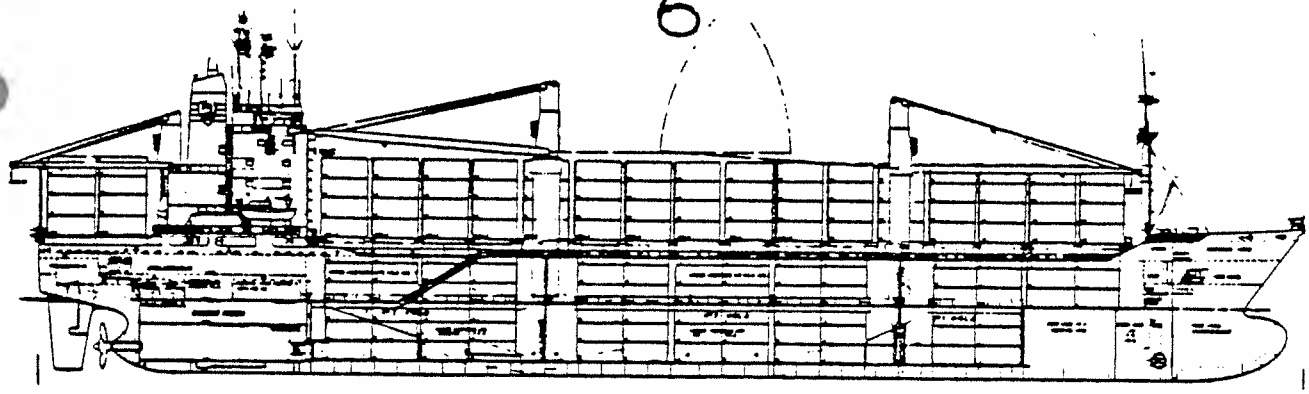
Container Vessel "CONCORDIA"/"CORONA"

6



Builders: Thyssen Nordseewerke GmbH, Emden
Owner: Concordia Schiffahrtsgesellschaft mbH & Co. KG, Elsfleth
Managing owner: Containerschiffsreederei H. W. Janssen GmbH, Elsfleth
Yard number: 474
Delivery: December 30th, 1982

Tonnage	17 750.52	GRT
	10 834.21	NRT
Deadweight	25 412	t
Length o. a.	166.04	m
Length b. p.	152.60	m
Moulded breadth	28.50	m
Depth to maindeck	15.80	m
Draught	11.606	m

Trial speed (draught 10.80 m, 9142 kW): 18.1 kn

Classification:
G. L. + 100 A 4 E Containerschiff + MC AUT

Propelling machinery:
single acting 2-stroke B & W-Diesel engine, direct reversible
type: 5L80GFCA supercharged
11 300 kW (15 400 BHP) at 106 rpm
heavy oil operation
Fuel consumption: abt. 50.6 t per day
propeller (Ostermann)
1 shaft driven generator 790 kW

Auxiliary engines:
3 Wärtsila Vasa-Diesel engines
type: 624 TS
821 kW (1116 BHP) at 720 rpm each
1 Deutz-Diesel engine
type: F6L 413 FR
96 kW (130 BHP) at 1800 rpm
boiler AQ 10/5 (Aalborg)
separators (Westfalia)
1 CJC-finefilter (Karberg & Hennemann)
bow thruster type 100 F – 882 kW (Jastram)
steering gear (Hatlapa)

gyro compass and automatic pilot (Anschutz)
visual direction finder, weather chart facsimile recorder, wireless station and 2 VHF equipments (Hagenuk)

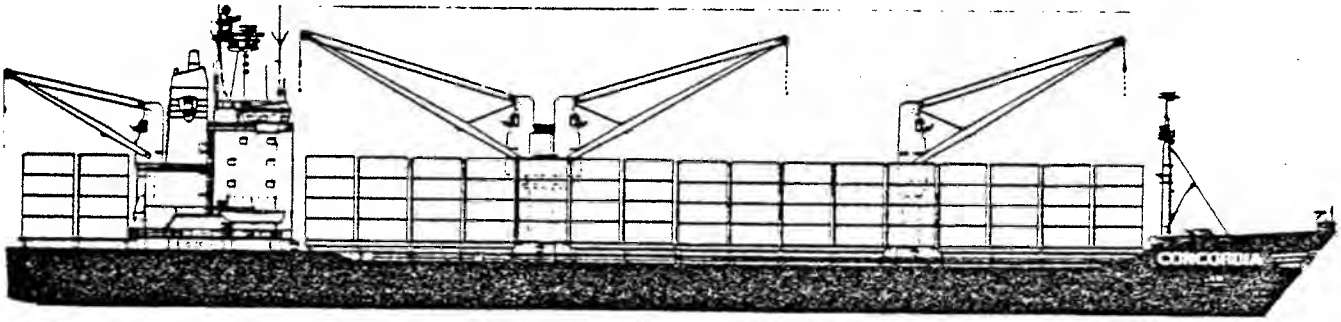
3 hatches
hatch 1 P. and S. each 25.50 × 10.50/8.12 m
hatch 2 P. and S. each 39.00 × 10.50 m
hatch 3 P. and S. each 25.50 × 10.50 m
Hatch covers: system KVAERNER

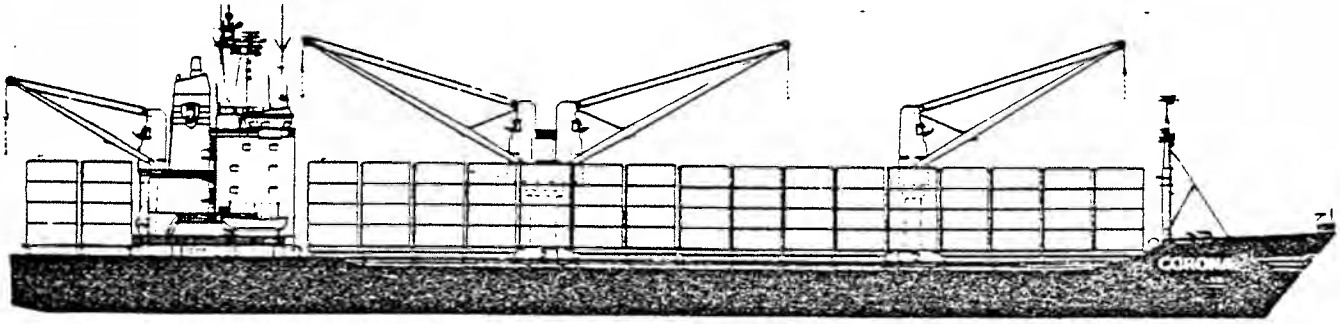
Loading gear: 1 single crane 36/40 t (between hatch 1 and 2), 1 twin crane 2 × 36/40 t (between hatch 2 and 3), 1 single crane 25 t (aft)
4 automatic constant tension mooring winches of 120 kN and 2 comb. windlasses/mooring winches of 194.5/389 kN — 120 kN (TNSW)

Hold capacity:	grain	
hold 1	7 578.4	m³
hold 2	15 755.8	m³
hold 3	10 584.0	m³
Total cargo capacity	33 918.2	m³

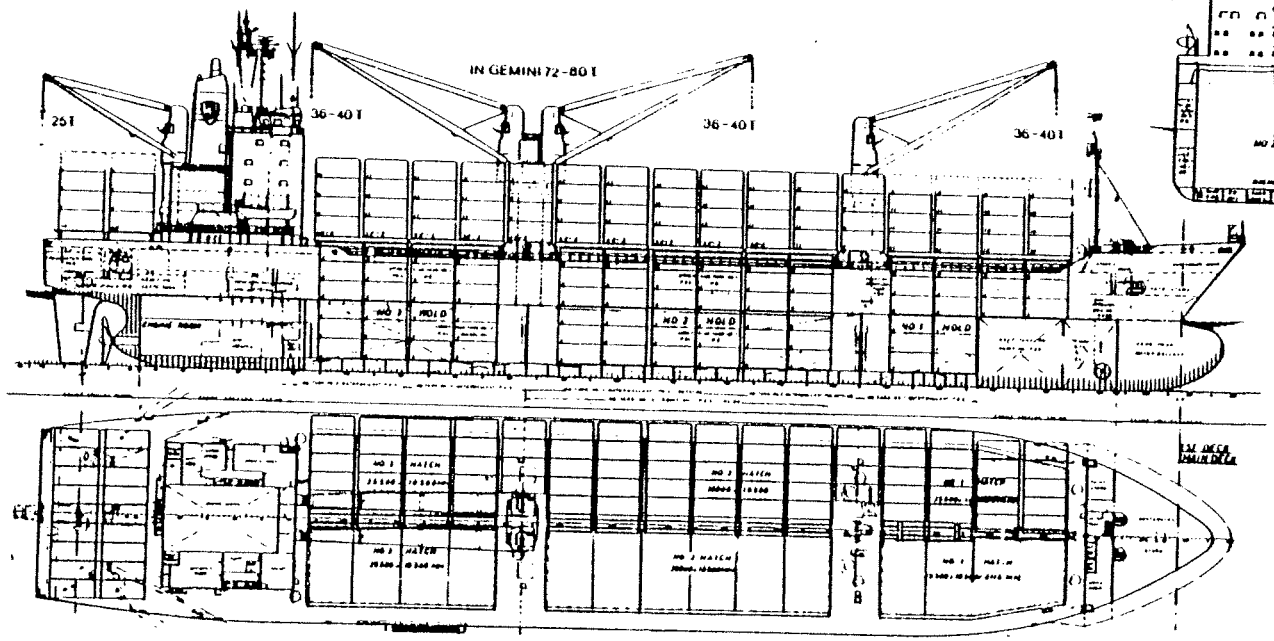
Container capacity:	TEU	or 40'	+ 20'
holds	556	272	12
on deck	625	291	43
on deck refr. cont.	101	41	19
Total	1282	604	74

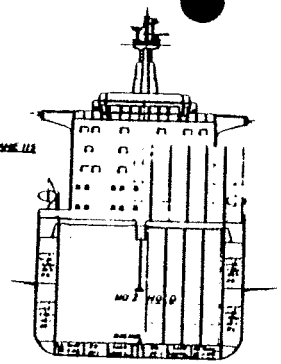
Container lashing and securing material (CONVER)





Two Sistervessels from
"Thyssen Nordseewerke GmbH" Emden West-Germany.
Hull nos. 474 (MV "CONCORDIA"), 479 (MV "CORONA").
Delivery dates December 1982 + June 1983





Breakdown of Containerstowage (1 FEU ≙ 2 TEU, nos. in brackets are single TEU) 40' x 8' x 8'6" and 20' x 8' x 8'6"

Hatch	Poop-Deck	III			II						Intake	
40' Bays	34	30	26		20	16	12		06	02		
20' Bays	35 33	31 29	27 25	23	21 19	17 15	13 11	09	07 05	03 01	40' + 20' or	20'
On Deck 4	6	11	11	(6)	11	11	11	(8)	8	8	77 + 14	168
3	9	11	11	(8)	11	11	11	(8)	11	10	85 + 16	186
2	9	11	11	(8)	11	11	11	(8)	11	10	85 + 16	186
1	9	11	11	(8)	11	11	11	(8)	11	10	85 + 16	186
total	33	44	44	(30)	44	44	44	(32)	41	38	332 + 62	726
Under Deck 6		8	8		8	8	8		8	6	54	108
5		8	8		8	8	8		8	(2) + 4	52 + 2	106
4		8	8		8	8	8		(2) + 6	4	50 + 2	102
3		6 + (2)	8		8	8	8		(2) + 4		42 + 4	88
2		4 + (2)	8		8	8	8		4		40 + 2	82
1		4	6 + (2)		8	8	6		2		34 + 2	70
total		38 + (4)	46 + (2)		48	48	46		(4) + 32	(2) + 14	272 + 12	556
Grand total	33	82 + (4)	90 + (2)	(30)	92	92	90	(32)	(4) + 73	(2) + 52	604 + 74	1282

geared containerships with cellguides, 20' convertible to 40' without need of shore-appliances (alternatively vessels can be delivered with tweendeck without cellguides subject to T/C duration/rate and provided change is required early enough view building-stage and time needed for supply of tweendeck pontoons).

4 Haegglund-deckcranes:
between hatches
1+2: **single 36 t x 30 m** (or 40 t x 27 m each)
2+3: **twin 36 t x 30 m each**
(or 40 t x 27 m each)
or 72 t in gemini x 30 m
(or 80 t x 27 m)
(all fitted with power swivels (cargo-spotting-devices) equipped with 4 liftingeyes each and otherwise according Haegglund specification no. 488 5325 a including hydraulic stabilizing winch and electronic lining-up device)
crane no. 4 behind superstructure:
single 25 t x 20 m
(on the whole fitted as cranes 1 to 3 but according Haegglund specification no. 488 5321 a which a. o. means no electronic lining-up equipment)

Containerintake subject to vessel's stability/trim/permissible weights:
1.282 units of 20 x 8 x 8' 6" whereof 556 in 6 tiers under deck = 70/82/88/102/106/108 and 726 in 4 tiers on deck = 186/186/186/168
alternatively
604 units of 40 x 8 x 8' 6" whereof 272 under deck and 332 on deck **plus 12 resp. 62 units of 20 x 8 x 8' 6"** under resp. on deck

Containerfittings:
Cellular under deck, twistlocks and rigid lashing bars / bottle screws for full load incl. 4th tier on deck.
101 reefersockets CEE model supplying 440 volt 60 cycles 3 phases earthing pin in 3 o'clock position (connection cables not on board)

Good stability due to high water ballast capacity of abt 7.975 cbm and 28,50 m beam of vessel, stability-cases can be made available immediately by telex upon request.

Pointload strengthenings:
under deck = 144 t / 210 t per 20' / 40' stack
on deck = 60 t / 80 t per 20' / 40' stack
D/B strengthened for grab discharge.

German flag
class: G. L. + 100 A4 (E), "Containerschiff", + MC (E) aut 24/24
dwat abt 24.950 mt on a draft of 11,50 m summer freeboard
main engine: B + W 5 L 80 GFCA 15.400 hp

p.t.o.

6.3.06

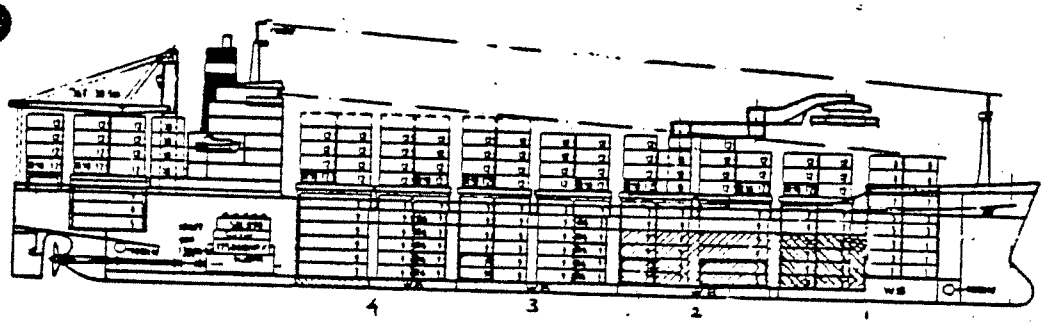
7-o-o

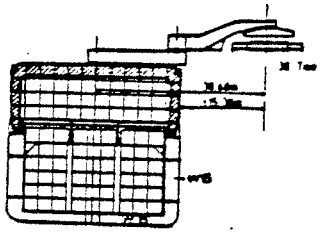
NEDLLOYD v NECK
v NOORT
v DIEMEN

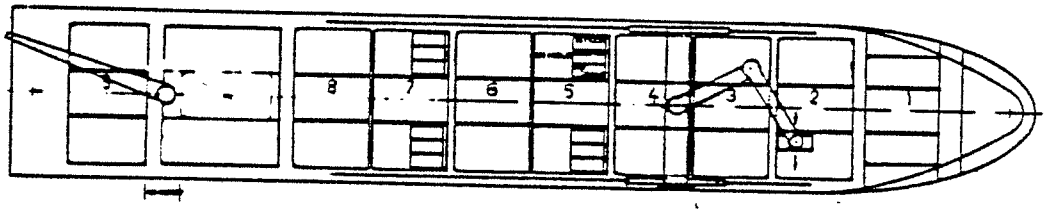
DEC '83

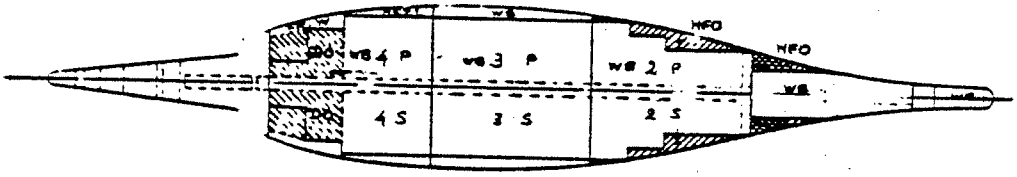
FNB

Three NEDLLOYD "KLECO"-class cellular containerships.









Length over all	182.50M+4.23
Length betw. perpend.	174.00M
Breadth mld	30.50M
Depth mld	16.20M
Draft design mld	10.00M
Displacement	34852TON
Light ship	11030TON
Deadweight 1	23822TON
Water ballast	7874 m3
Fresh water	411 m3
Heavy Fuel Oil	2883m3
Diesel oil	366m3
Lubricating oil	281m3

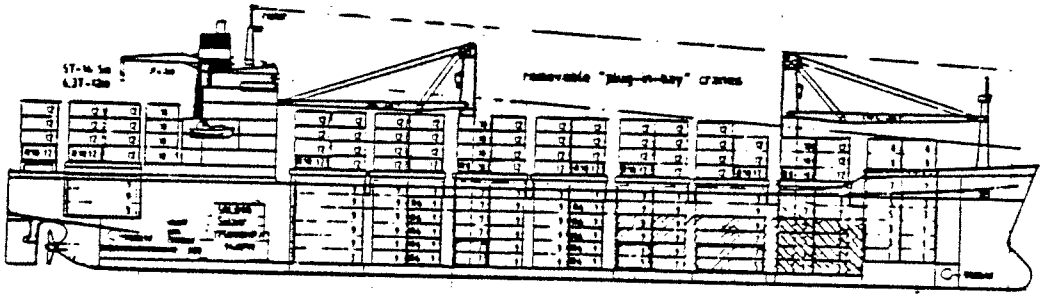
Cont. below deck	10376T. 14x784	
Cont. on deck	9674T. 14x691	
Homog. loaded	20050T. 14x1475	G'M = 0.41 M
Centre gravity box = 45%/41'		

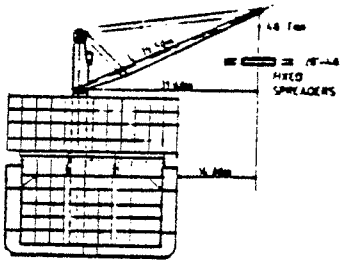
EVEN KEEL: 10 M

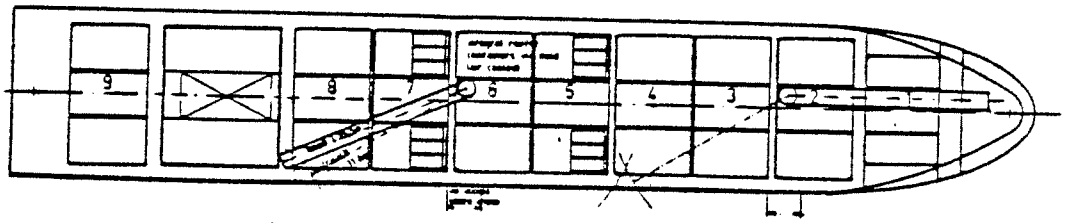
NEDLLOYD VAN NECK 929

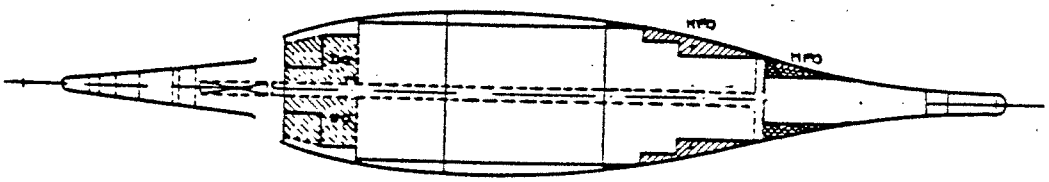
Delivery date: 2 December 1983
Yard v/d Giessen de Noord

1475 TEU x 14 TON









Length over all	182.50M+4.23
Length betw. perpend.	174.00M
Breadth mld	30.50M
Depth mld	16.20M
Draft design mld	10.00M
Displacement 9.94 M	34687 TOM
Light ship	11000 "
Deadweight	23687 TOM

Water ballast	7874m3
Fresh water	411 m3
Heavy Fuel Oil	2883m3
Diesel oil	366m3
Lubricating oil	281m3

NEDLLOYD VAN NOORT 930
" **VAN DIEMEN** 931

Delivery April 1984 / July 1984
Yard v/d Giessen de Noord
Class Bureau Veritas

1426 TEU x 14.2 TON

	TEU	
Cont. below deck	10678T. 142 x 752	
Cont. on deck	9571T. 142 x 674	
Homog. loaded	20249. 142x1426	G'M = 0.46 M
Centre gravity box = 43%		

EVEN KEEL 9.94 M

BV 2-12-83

NEDLLOYD VAN NECK VAN NOORT VAN DIEMEN

DESIGN PHILOSOPHY 7-0-2 "SMALL-SIZE" CONTAINERSHIP.

23-6-'83

Capacity and speed.

The commercial departments required a vessel of about 1400 TEU of 14 tonnes per TEU and 17-18 knots service speed and being as flexible as possible for interchange between the company's various services. Contrary to earlier periods, this vessel was not to be designed for a certain trade but was to be utilisable for worldwide container transport. Many times Nedlloyd experienced reshuffling her ships in co-operative services.

Length Over All.

Almost all container ports of some importance can accommodate ships of 183 M. length over all.

- Good manoeuvring properties, however, are required, especially in the ports on the West African coast where the vessel has to go up-river with very sharp bends and strong currents (Matadi: chaudron d'enfer) or has to turn around within 300 metres length by her own means.
- Also for this reason a rather large rudder is fitted which can be laid at 45° helm, a bowthruster and a sternthruster were fitted which also have the advantage of shortening the roundtrip time and reduction of deck-crew.

Breadth.

30,5 M. beam has been selected in order to obtain on the worldwide routes the sufficient stability which will allow for maximum container carrying capacity.
Panama beam should have resulted in a too blunt vessel having generally too much stability.

- This "Small-Size" container is able to carry 6 tiers containers in the holds plus 4 tiers on deck, resulting in a carrying capacity of 1600 TEU at the maximum, at an average container-weight of 12½ tonnes per TEU. The usual requirements of ahead view and non-homogeneous stow have been taken into account.
- Normally, the ship can carry at 10.00 M draft a number of 1444 TEU (14 t. homogeneous stow).
Herewith the stability is about 0.4 M metacentric height, being an acceptable value.

Hull form and propulsion.

A V-shaped forebody and a Pram-type afterbody (like those of old sailing ships and yachts) will result in an easily driven hull with maximum form stability and maximum deck space.

- The flow of water towards the propeller has been smoothened as much as possible - by scraping off some displacement - in order to obtain a good propulsion and a minimum of propeller-excited vibrations.
- The 'large' propeller at rather low revolutions will give a good propulsive efficiency.
- A fine underwater-afterbody, however, will automatically ask that engine room plus deckhouse are to be moved somewhat foreward.
- The reason for this is the space required for a sufficiently stiff engine foundation and also the trim-balance of the ship.
- Moreover, the fuel oil is to be carried in the forepart of the hull in order to sail on even keel. (Sailing with - say - about 3 feet astern trim on these vessels is very uneconomical, because ship's resistance increases enormously by the combined action of emerging bulb and immersing transom stern).
- For this reason - and also for keeping the hogging moments in the hull at a low(er) value - Nedlloyd decided on a general arrangement with engine room not too far aft (i.e. one hatch-lengths before the stern instead of none as is the case on many a similar containership). With such arrangement the balance of the ship is better and she needs less water-ballasting in the forebody for compensating the astern trim.
- Meanwhile, by placing the deckhouse more forward, the vibration hindrance caused by the propeller will estimately become less.
- A small disadvantage is the loss of a few containers in holds. This loss was accepted by us because a very great part of the containerload is carried on deck.

Draft.

The scantlings are allowing for 11.25 metres draft.
However, this (= 37 feet) draft is too deep for an economical service condition, especially in view of immersion of the transom.

- The hull form will be optimal between 7 M and 9 M drafts. This mainly decided the transom depth and bulb diameter.

- Too many ships are designed with too big bulbs. Big bulbs will function as water'turbulating' brakes at the smaller service-draft ranges which quite often occur on coastal legs between the many ports.
- West African ports allow for 25 - 32 ft drafts which figures are limiting the cargo deadweight of the vessel.

Block coefficient

In order to have sufficient cargo deadweight at the smaller drafts, the block coefficient should not be too fine.

- Remunerativeness calculations in 1981 - at which time fuel prices and ships' prices were rather high - the optimum blockcoefficient for an 18½ knots service speed varied between 0.625 and 0.65.
- At the moment (June 1983) with fuel prices at somewhat lower level as well as ship's prices the optimum varies between 0.635 and 0.66 .
- Many containerships nowadays have too fine lines and (consequently) cannot carry weight from shoal ports !

Cubic capacity.

The containerslot capacity within the hull is fully utilized up-to the last centimetre.

Experience with containership-designs of the latest 10 years showed the optimum length of a cellular container hold to be 28.70 metres (== 35 frame spacings of 820 MM each).

The minimum required length of engine room and length of mooring space of the fore and aft decks are to be added to a number of cellular blocks and the overall-length is born. This L.o.a. in our opinion is a muchbetter value for defining a vessel than the old-fashioned L.p.p. (Length between the load-lines).

1444 TEU boxes - with 3 tiers above deck - on a 183-metres ship of 30,5 M beam is a good capacity. However, it has to be realised that the boxes on the rear end of the ship (above the mooring winches) are NOT to be assessed as normal loaded containers. This is because hogging moment in the hull and astern trim of the vessel will limit the weight of these boxes to be very light ones only.

Seaworthiness.

The vessel accommodating 6 tiers containers within the hull has sufficient freeboard at service draft of 10 M and the range of stability is rather large. Thus, a low metacentric height (0.4M) is allowable and will give a gently moving ship.

- However, if only a small amount of containers is to be transported, the vessel is too stable because metacentric height is very great and as a result jerking rolling movements will occur at sea. The only answer to that problem is to pile the boxes up as high as possible and sail them mainly on the hatchcovers.
- Securing of the containers on deck should in general be done by means of twistlocks only (except when the topweight on the lowermost box of a stack exceeds 30 tons).
- Application of fin -stabilizers is considered to be not economical anymore. At the low(er) speed level the fins' surface would have to be large, the necessary larger size of the aparatus would be (more) expensive and the fuel bill considerably higher than formerly .
- At the other hand, 'steered antirolling tanks' would have to be very large sized and this would incur corrosion problems in the course of the 20 years' service.

Strength.

The structural 'sideboxes' of 2.35M width have already proved to be sufficiently stiff and strong on the larger sisterships and moreover the Length/Depth ratio of the 183M by 16.2M vessel is quite reasonable.

In this respect also the knuckle line at the forebody is to be mentioned as it accentuates the box-girder surrounding the hatch openings in the foreship.

- The sidebox creates axial stiffness against headseas and also will throw the water away from the vessel when the ship is pitching in headseas.
- We prefer this knuckle line more than breakwaters on deck for this reason. Also, it looks nice because this feature is a 'last rudiment' of the ship's sheer.

Hatch configuration. Reefers.

- With a view to flexibility in container stowage, the triple-hatch arrangement is to be preferred above the twinhatch type.
- When many ports are to be called at in one voyage, the twin-hatch vessel will incur much more reshuffling of deck-containers than the triple-hatch configuration. This will cost much more than moving of covers only.
- Structural-wise, the triple-hatch arrangement creates two 'violin strings' over the whole length of the vessel, acting as longitudinal strength members (and may fall less heavy than the single-string of a twin-hatch arrangement vessel).
- Contrary-wise, the hatchcoaming structure above the deck should be of the interrupted type in order to avoid high stresses.
- Transverse members consist of wide and slender structures (cross-overs). The widest ones are accommodating ventilation ducts and stairways.
- By adopting this configuration it is always possible to provide the ship with 'integral' type reeferbox capacity underdeck. Cooling these containers - or rather their engines - can be executed by means of aircooling (pressure ventilation) only or by means of airblowers and water-piping (connected b.m.o. flexible hoses to a shipboard system).
- Nedlloyd Newbuilding department has found from experience that almost every newbuilding vessel will have to be provided - once she is in service - with a number of some 20 to 100 reefers slots extra at a later date. For such an exercise the Small Size containervessel stands wide open.
- The 'integral reefer' system applied on this vessel in general has much more flexibility than the ducted (Porthole) system. With the latter, the cooling installation is shipborne - which makes the ship more expensive as to investment costs .
  At the other hand, the flexible worldwide applicable 'integral' reefersystem means a higher investment cost in containers.
  'Ducted' systems are more economical where a high-volume reefer trade exists and ships are to remain in the same service for much of their lifetime, so there where flexibility counts little.

General arrangement. (Fuel and waterballast location).

- As has already been explained, location of the engineroom and deckhouse is 'semi-aft' with the fuel oil storage 'semi-forward'. As high-viscosity fuel oil is used, there exists the problem of pumping the fuel from tanks forward to the engineroom aft, when needed.
  For this reasons, the fueloil pipelines run through pipe tunnels located in the double bottom.
  The lines can be heated by means of steam lines lying next to the fuel lines and having been wrapped around by insulating material.
- The weight of the consumed fuel can be compensated by taking waterballast in doublebottom and side tanks in the vicinity of the fuel oil tanks.
  Waterballast tanks in the wings of the upper part of the foreship will make it possible to compensate for trim astern without increasing the transverse stability of the vessel too much.
  (On many a containervessel the stability increases too excessively when - in certain conditions - forepeak and deeptank Nr. 1 are used for this purpose.)

Engine installation philosophy.

- Maintenance costs are directly related to the amount of bolts and nuts utilized and for this reason a slow-turning 5-cylinder diesel engine, has been installed together with a shaft-generator, thyristor-controlled.
- Maximum reliability has been strived after in utmost simplicity.
- The aim is not to use separate diesel-engine generators at sea. However, during manoeuvring with the bow and sternthruster at work or standing-by, all three diesel-engined generators should be running.
- It is the opinion of Nedlloyd Newbuilding department that it is not a paying proposition to have an exhaustgas turbo-generator installed on a containervessel as such a ship has to serve many ports (with short sea-stretches in-between) and has often to sail many coastal stretches at reduced speed.
  (In this respect we are in favour, however, of an exhaustgas generator installation on a tanker and bulkcarrier which ships - normally - sail on long stretches and at full power.)

- Centralized control has been adopted in the wheelhouse and is arranged in such a way that the nautical duty officer is not disturbed in the exercise of his duty by less-urgent engineroom alarms. During critical periods, the Chief Engineer is (to be) in the wheelhouse.
- The engineroom control station is minimized in layout and is situated near the workshop where engineers do their maintenance jobs. The space is air-conditioned and kept at low noise level.

Mooring arrangement.

- All wires are to be fed directly from the large-diameter winch-drums via fairleads of the large-radius solid cast steel Panama type towards the quay bollards.
- Four (4) mooring winches forward and four (4) mooring winches aft are to be handled by two men forward and two men aft. Bow- and sternthrusters are used in combination with the mooring cables. The vessel can in principle move transversely on its own power and could press herself against the quaywall whilst meantime the wires can be made fast'. After having been made fast, the winches are set to 'self-tensioning'.

  Two ways of mooring are feasible, i.e.:-

  Alternative (1)... 3 wires plus 1 spring, forward as well as aft.

  Alternative (2)... 2 wires plus 2 springs, forward and aft and remotely controlled from bulwark.

  The steel wires system is adopted because of low maintenance load on the wires and on the fixed fairlead-chocks.
- The anchor type is of Spek design. It is flush mounted in an anchor pocket and always fits well when 'coming in'. It never slams by the action of headseas.

Deck layout in view of maintenance .

... as simple as possible...

Smooth flat coamings and closed box-foundations. Whatever labour-requiring features could be omitted should be omitted (i.e. no frills and freaks - geen overbodige Toeters en Bellen)

Grit-blasting of weather-exposed surfaces after finishing construction and application of zinc-epoxy plus chlorinated rubber paints is a 'must' for the shipowner if he intends to sail with a minimum number of men (18) .

(Design background) 23/6-'83

Deck layout in view of maintenance.

- As simple as possible...

Smooth flat coamings and closed box-foundations. Whatever labour-requiring can be omitted should be omitted.
Grit-blasting of weather-exposed surfaces after finishing construction and application of zinc-epoxy plus chlorinated rubber paints is a 'must' for the shipowner if he intends to sail with a minimum number of crew (18).
Van der Giessen shipyard managed so far to create a 'well painted ship' during the building period inside shed, notwithstanding the problems which had to be solved regarding dust hindrance of blasting.

Double bottom waterballast tanks. Preparation.

After our experience of troubles recently met, Nedlloyd nowadays is in favour of tank conservation in the final stage to be carried out as follows:

- Gritblasting of welds and sweepblast of remaining surfaces of finished tanks.
- Application of 1 coat of 350 micron thickness of non-solvent coaltar-epoxy type paint by means of high-pressure spraying apparatuses.

This system saves manhours, has no intercoat adhesion problems and does not create an explosion hazard.
The grit can be removed via temporary cut holes in the shell-plating.
Dust problems during the building of the ship in the shed do not exist.

Underwater.

Application of self-eroding (SPC of International) acrylic resin copolymer pigmented with organic tin antifouling paint, 3 coats each 100 micron.
This has proven to keep the vessel free from algae and barnacles.

(Design background)

SPECIAL FEATURES OF FIRST VESSEL:

- The first vessel, to be named Nedlloyd van Neck, (Yard nr. 929) is intended for the Europa-W.Afrika service, carrying general cargo in containers Southbound from western Europe to Dakar-Matadi Range.
- Northbound, the vessel has to transport coffee, cacao, cotton, sawn timber and other tropical produce, although many containers will have to be carried empty at the start.
- At this moment, only Apapa is fitted with a container gantry crane and Port Harcourt is in development stage (Port Onne). Because of this situation it was decided to provide this containervessel with a shipboard container gantry crane. It is an advantage of the gantry system that ships can be geared or de-geared within say 1 day - in case rails are fitted - which is an interesting feature with a view to interchangeability between services.

- Below hatches no. 3 , no. 4 and no. 6 break bulk cargo can be carried on tanktop like tractors - busses - p.cars - trafo's - and other awkward cargo which cannot be stowed into containers.

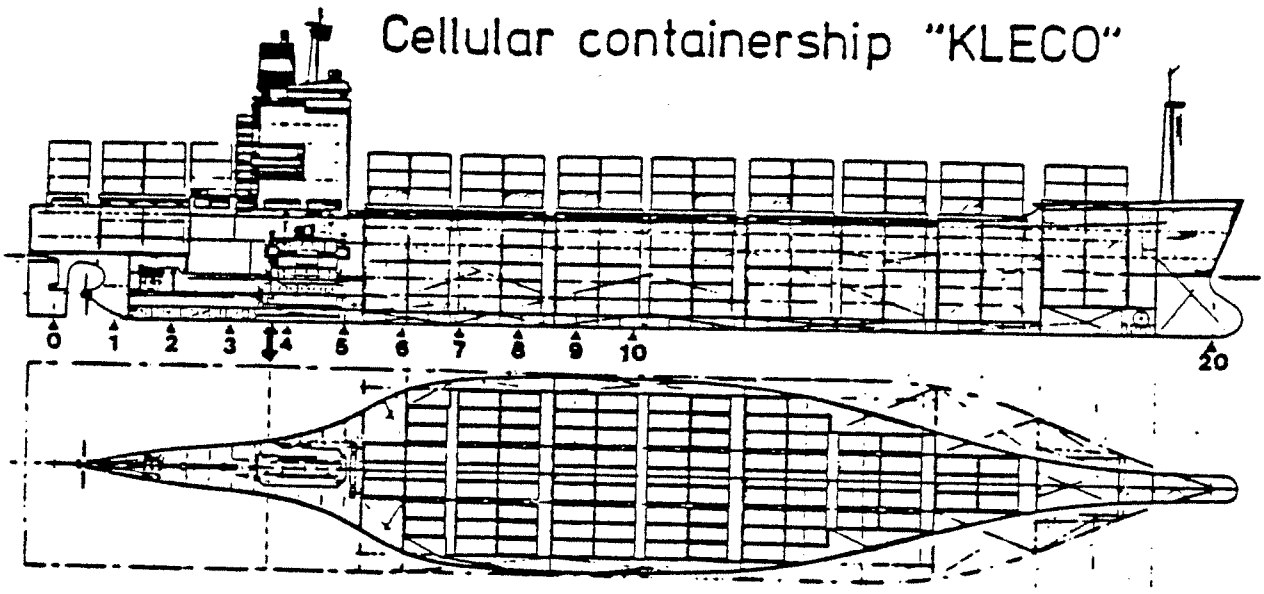
- Above this general cargo the containers can be stowed in the cells and supported by removable chocks inside the anglebar guides at second tier level.

- The underdeck capacity allows for 70% in 20' units and 30% in 40' units.

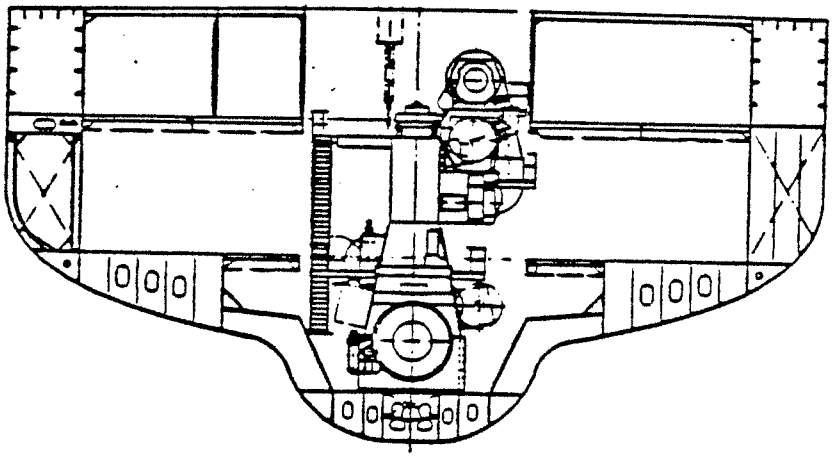
  Moreover the 40' cells in hatch no. 8 and nr. 9 can carry 20' units without adaptions.

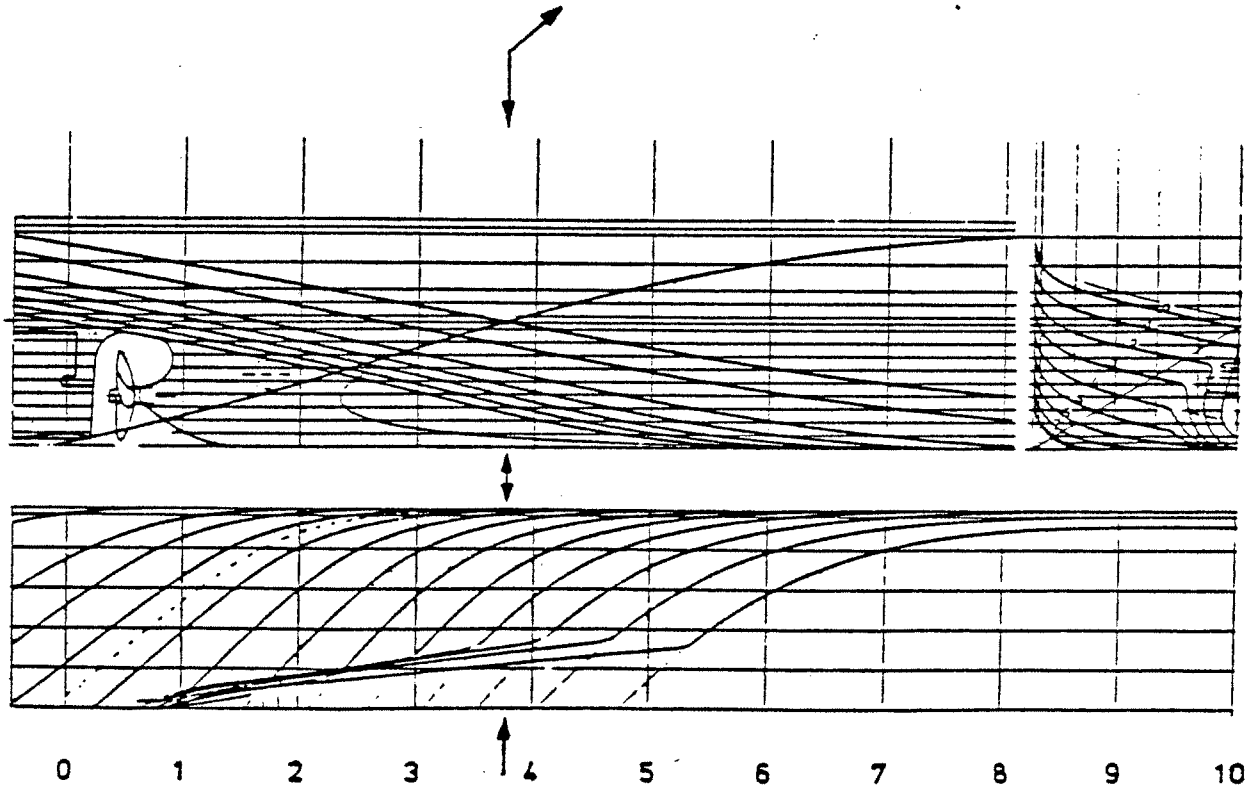
  This results in a 40-Ft flexibility of the whole vessel ranging from 3% to 58%, taking into account that hatchcovers have fittings for both: 20 Ft and 40 Ft.

Cellular containership "KLECO"



FRAME 42 AT ORDINATE 3.75

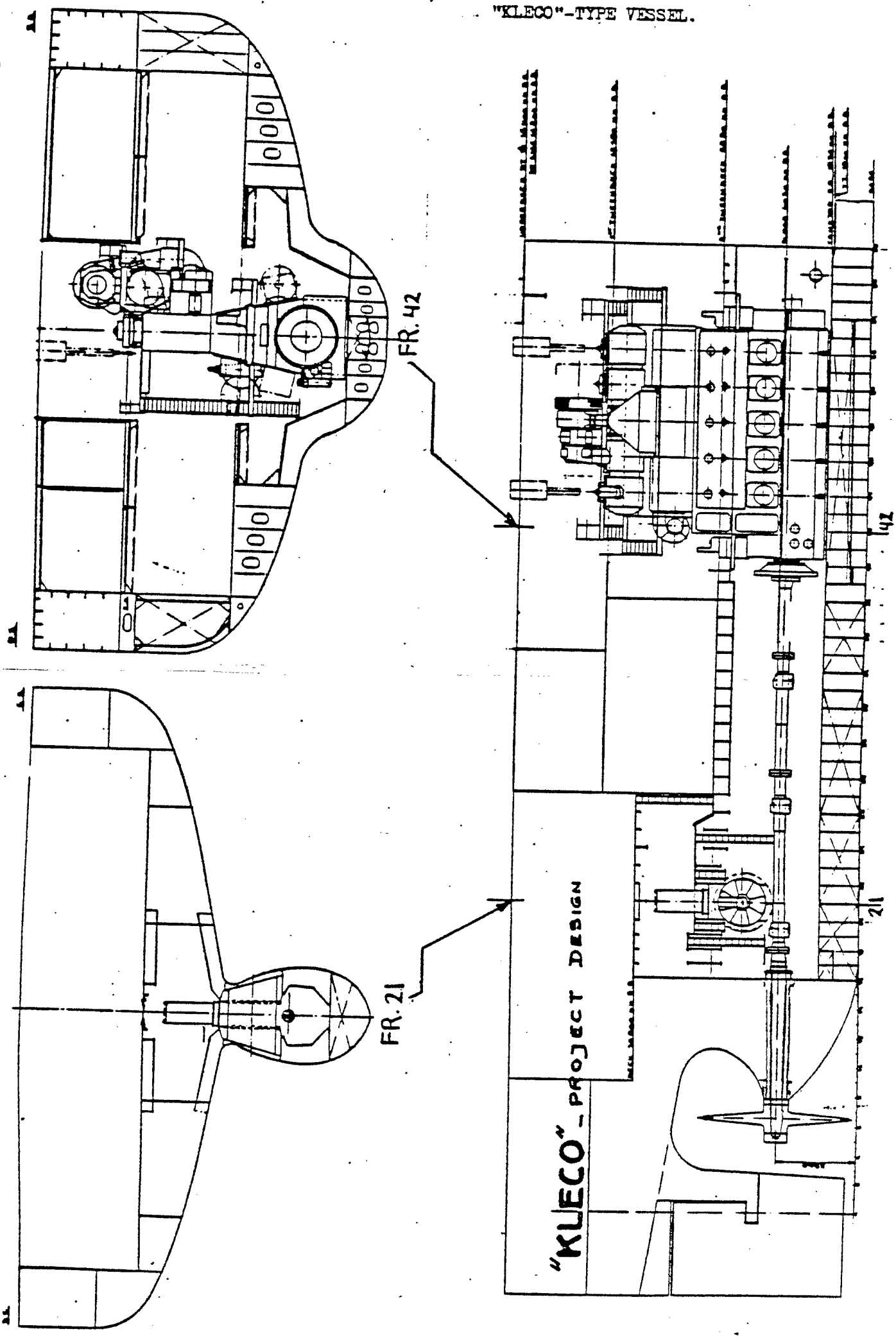


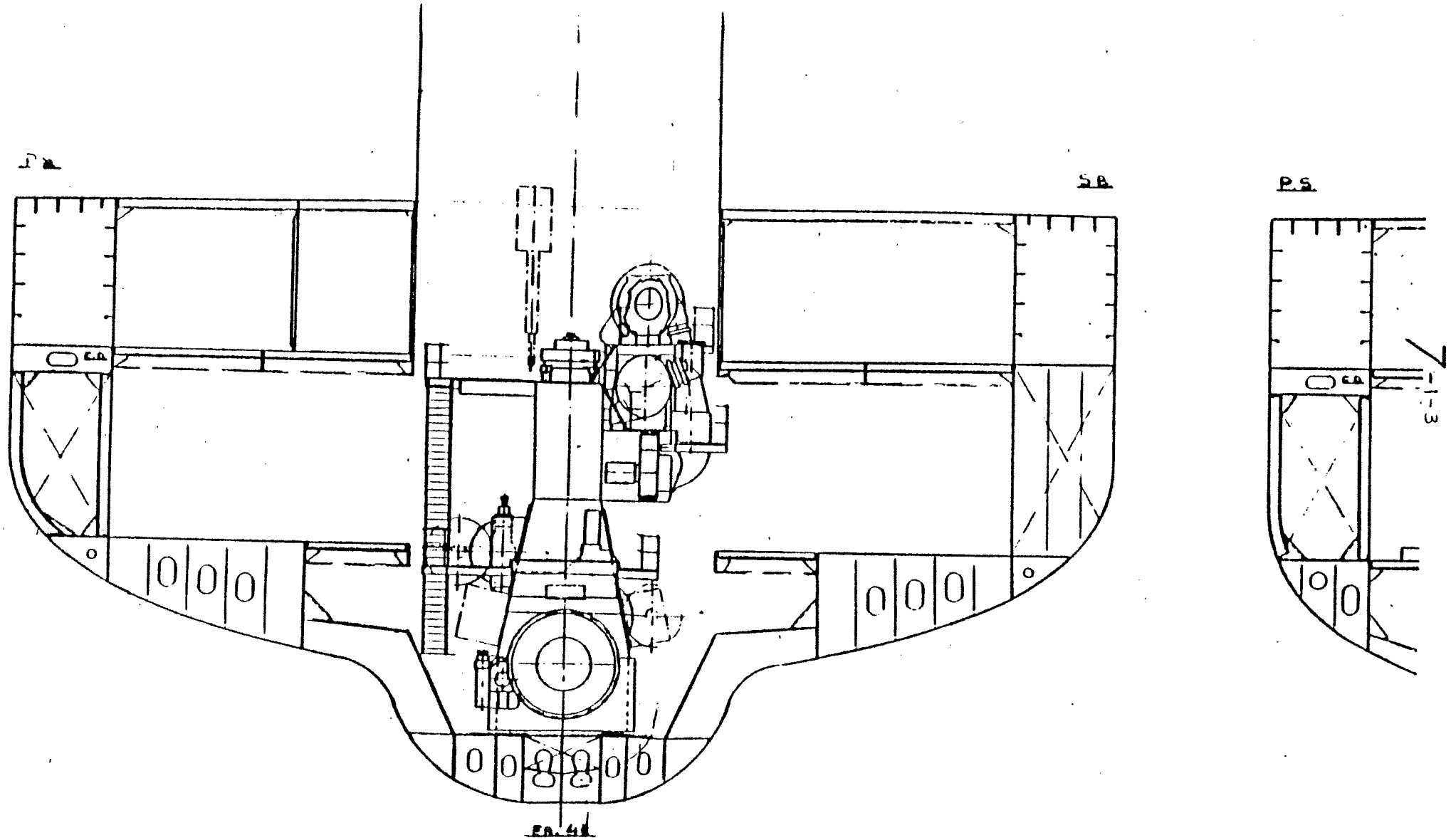


7-1-2

CRITICAL FRAMES

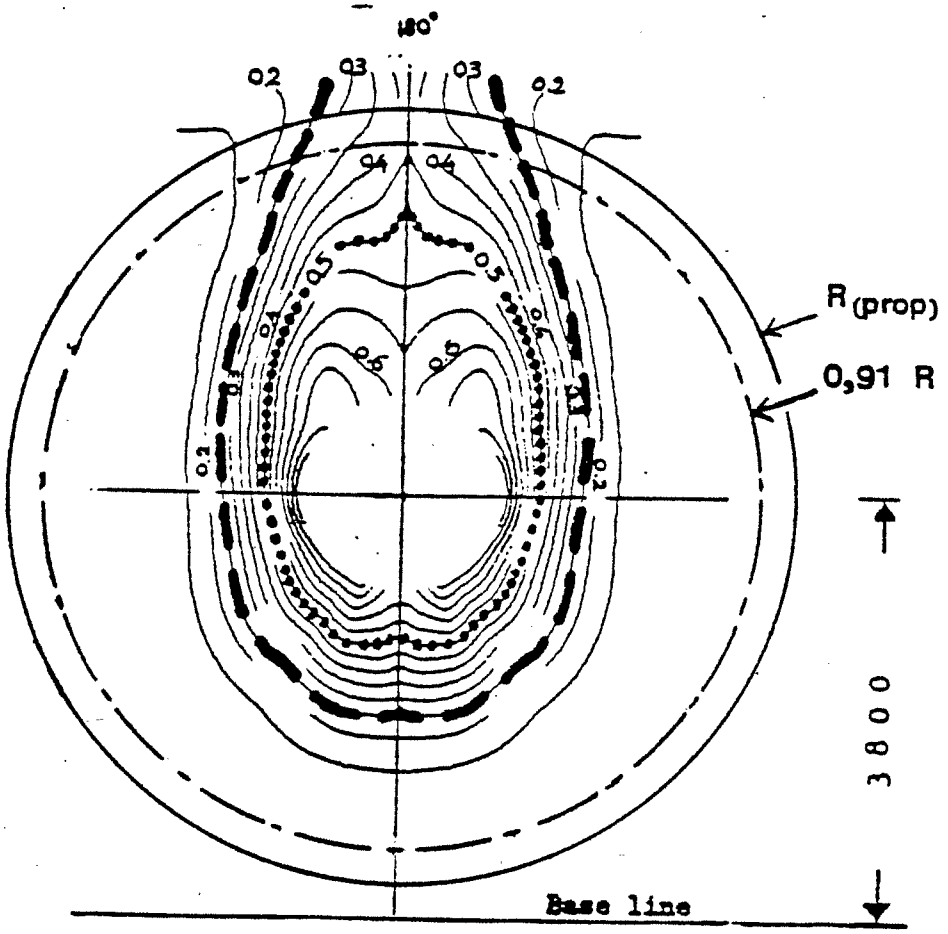
"KLECO"-TYPE VESSEL.




S.B.
P.S.
FR. 48

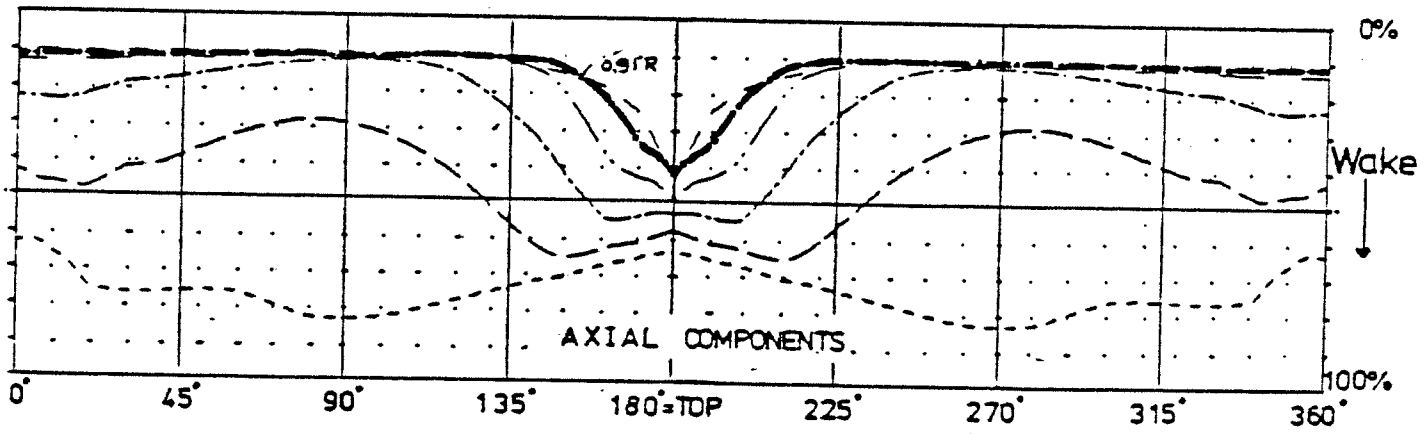
7-1-4

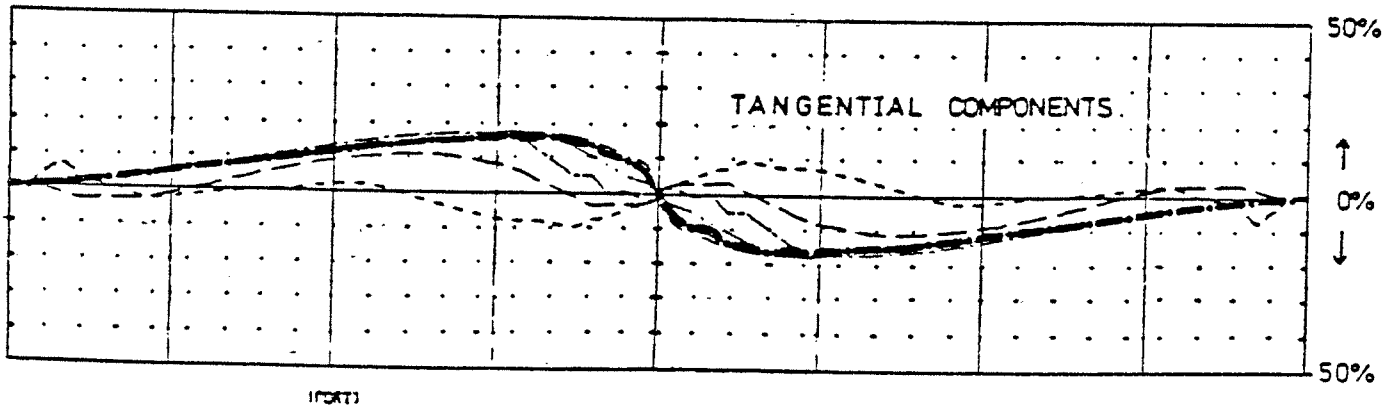
WAKE FIELD MEASUREMENTS
"Klein" containervessel ["KLECO"]
AT 10,00 m DRAFT.



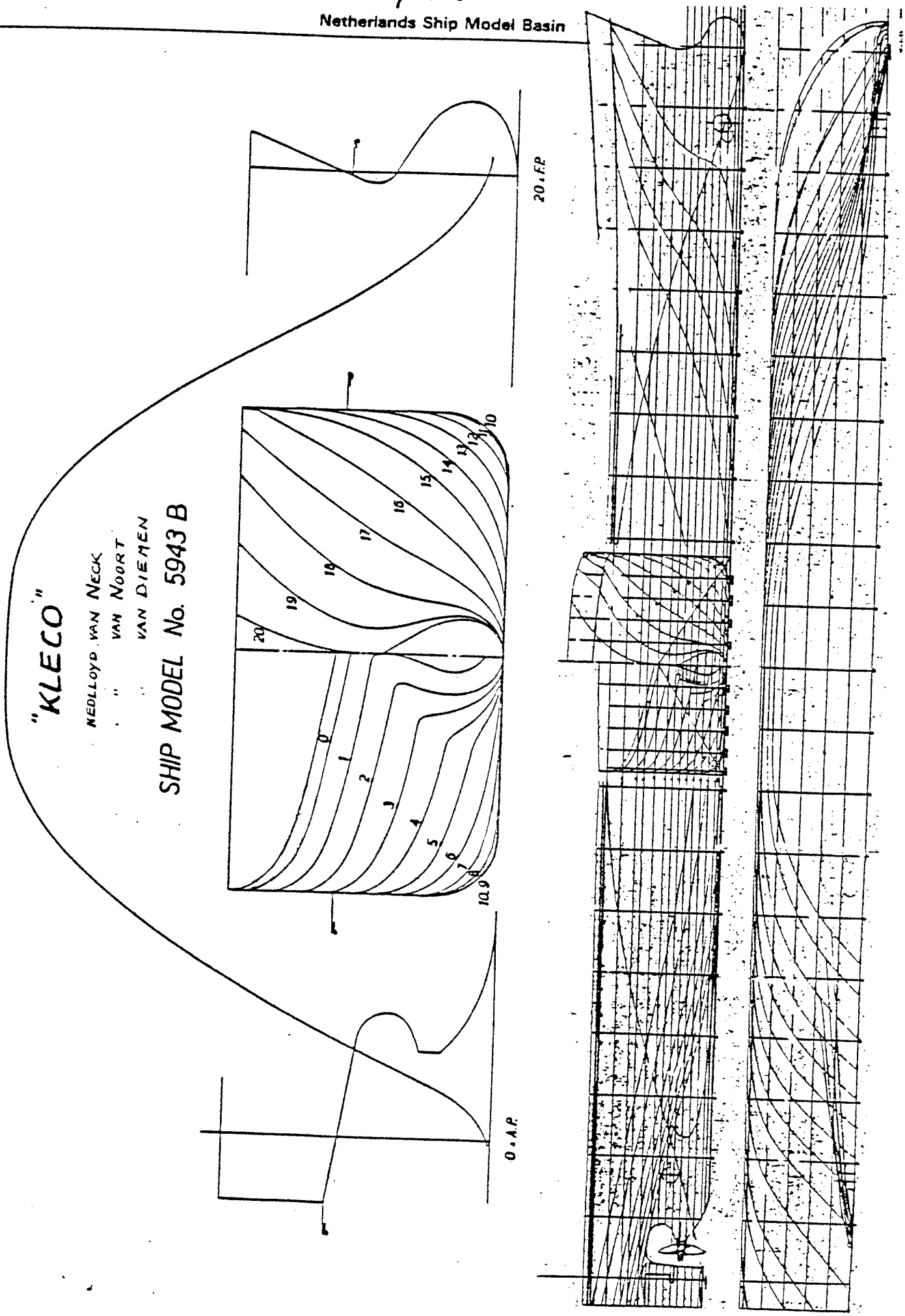
RADIUS IN mm	TYPE OF LINE
870	
1450	
2030	
2610	
3190 (0.91 R)	
3770	

LINES OF EQUAL AXIAL VELOCITY COMPONENTS





Netherlands Ship Model Basin



DRAWING No M 5943 - 49

Particulars of ship:

Ship model No.		5943 B	5943 B
Length between perpendiculars	L_{PP}	174.00	174.00 m
Length on waterline	L_{WL}	177.58	169.33 m
Breadth moulded	B	30.50	30.50 m
Draft moulded on A.P.	T_A	10.00	7.20 m
Draft moulded on F.P.	T_F	10.00	5.50 m
Displacement volume moulded	∇	33926	19398 m^3
Displacement weight in sea water	Δ	34774	19883 m.t.
Wetted surface, bare hull	S_1	6706	- m^2
Wetted surface, rudder incl.	S_2	6815	5148 m^2
L.C.B. position aft of ½ L	θ	0.97	- $\%L_{PP}$
Block coefficient	C_B	0.639	-
Midship section coefficient	C_M	0.980	-
Prismatic coefficient	C_P	0.652	-
Length-Breadth ratio	L_{PP}/B	5.70	-
Breadth-Draft ratio	B/T	3.05	-

Ship model material: wood.
Linear scale ratio λ = 29.
Turbulence induced by studs.

Propeller characteristics:

Propeller model		3937 (stock prop.)	5711 (design pitch)	5711 A (final pitch)
Diameter	D	6983	6800	6800 mm
Pitch at 0.7R	$P_{0.7}$	6244	6189	5880 mm
Pitch ratio	$P_{0.7}/D$	0.894	0.910	0.865
Expanded blade area ratio	A_E/A_O	0.619	0.556	0.556
Boss-Diameter ratio	d/D	0.162	0.169	0.169
Number of blades	Z	4	4	4

NSMB
Netherlands Ship Model Basin

η_D

CONTROL TEST

PROPULSION TEST RESULTS

Test No. 42234

Model condition: SEE BELOW

Ship model No. 5943B
Propeller model No. 3937
Scale ratio λ = 29.00000

Draft moulded: on FP = 10.000 m; on AP = 10.000 m; mean = 10.000 m

Displacement volume ∇ = 33926 m³

Skin friction correction E_D from model to ship is according to ITTC

model-ship correlation allowance coefficient CA = .000100

Ship Speed in knots V_S	P_D	P_E x1.06	$\eta_D = \frac{P_E}{P_D}$	C_A	Revs/min n_S	Resistance R_T in tons	Thrust T in tons	Res $t = \frac{T - R_T}{T}$
14.50	6490	6880		500	74.5		58.247	
15.00	7198	7630		500	76.9		62.747	
15.50	7965	8440		498	79.5		66.997	
16.00	8729	9260		500	81.9		71.246	
16.50	9605	10180		498	84.5		75.996	
17.00	10544	11165		496	87.1		80.996	
17.50	11642	12340		490	90.1		86.496	
18.00	12763	13530		487	92.8		92.495	
18.50	14007	14850		482	95.6		98.995	
19.00	15421	16350		474	98.6		105.995	

MODEL CONDITION:
BULBOUS BOW III
SCREW APERTURE II
RUDDER II
2 BOW THRUSTER TUNNELS II
STERN THRUSTER TUNNEL I AND FAIRINGS I

VALUES ARE GIVEN IN METRIC UNITS

P_D = delivered HP at propeller.
P_E = effective HP.
n_S = RPM of the ship's screw under the below conditions, not corrected for difference between the wake of ship and model, without any allowance

$C_A = \frac{\Delta_1^{2/3} V_S^3}{P_D \text{ metric}}$ Δ_1 = Displacement weight in salt water, in metric tons.

British HP = $\frac{75}{76}$ metric HP

Note: Results of self-propulsion tests as stated above refer to the self propulsion point of ship.
The results are calculated direct from measured model values without any allowance for appendages not present in the model, wind and sea; so values for ship are for tank condition corrected for salt water (γ_1 = 102⁵) and 15° centigrade.
Model and ship dimensions are moulded.

7-1-8

SHIP MODEL 5943 B

PROPELLER 5711 A

VALUES ARE DERIVED FROM RESULTS OF PROPULSION TEST No. 41909

DRAFT 10.00 m.

voor vlagenkopje: $\left(\frac{95.4}{10}\right)^3 \times 14.290 = 12230$ FACTOR.

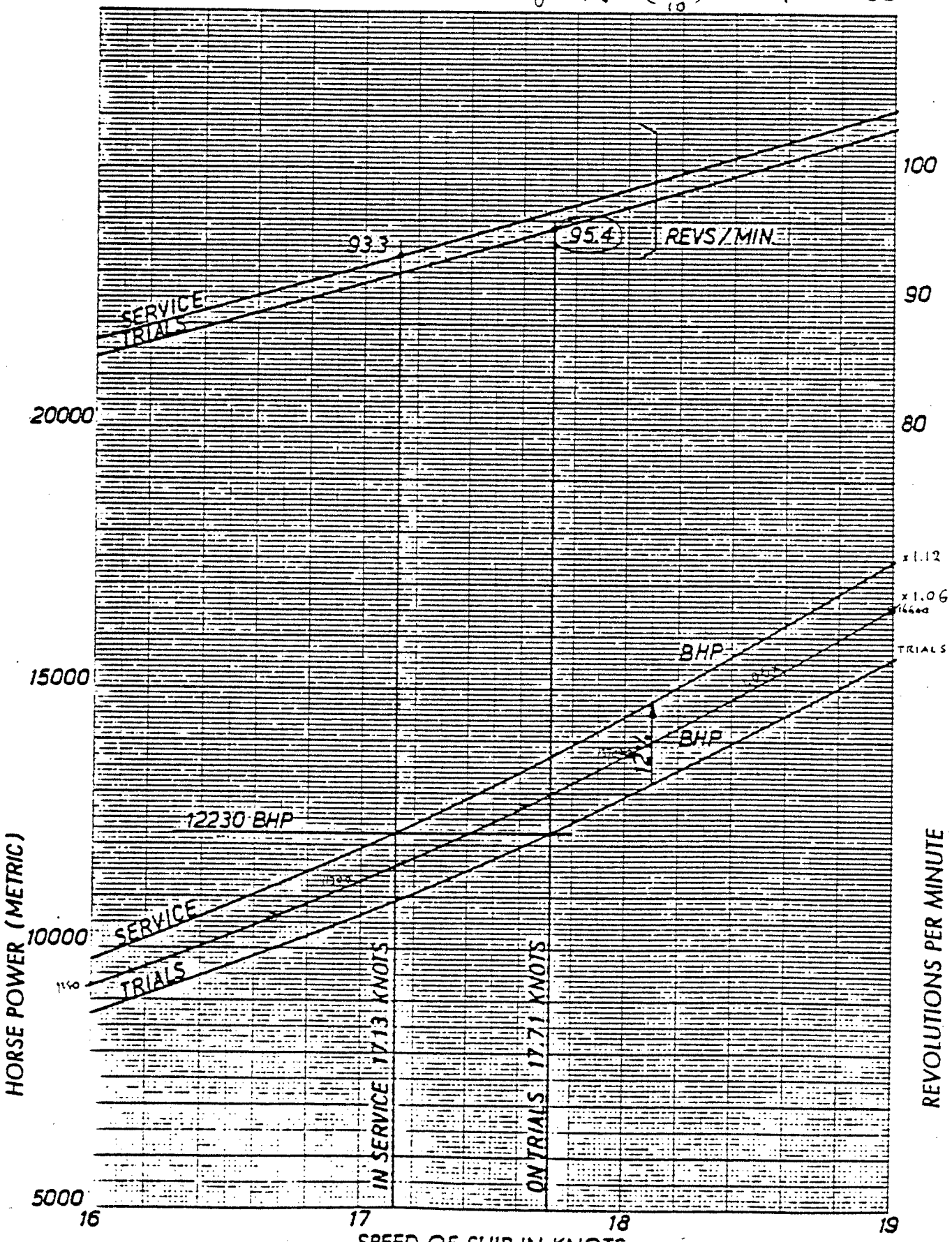


DIAGRAM No. M5943—60A

7-1-9

SHIP MODEL 5943 B

PROPELLER 5711 A

VALUES ARE DERIVED FROM RESULTS OF PROPULSION TEST No. 41910

DRAFT 5.50 / 7.20 m.

MEAN 6.35

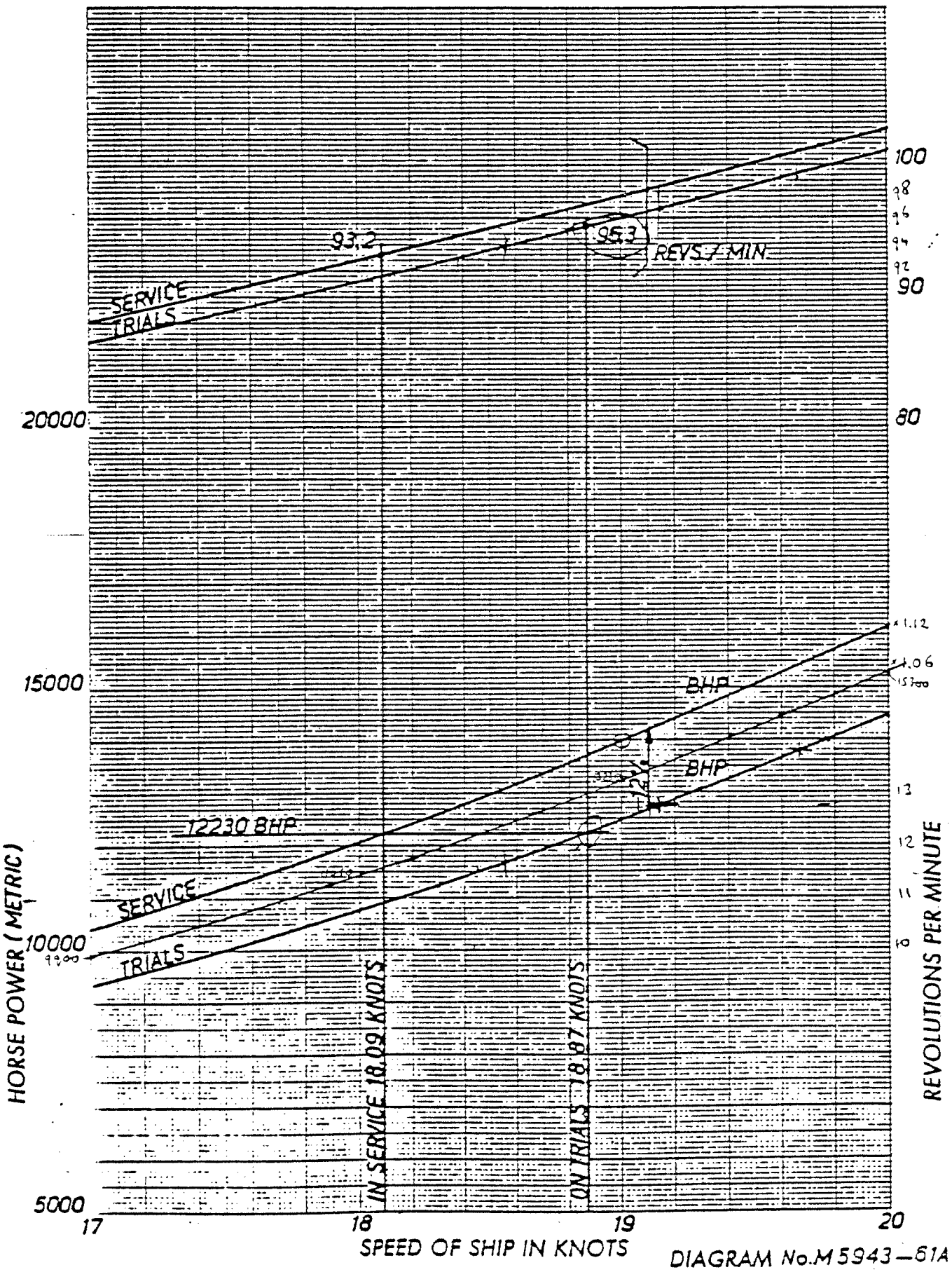


DIAGRAM No. M 5943 — 61A

7-1-10

PROPELLER DATA

DIAMETER	6800 MM
PITCH.	5723 MM
NUMBER OF BLADES	4
DIRECTION OF ROTATION	RIGHT-HANDED
EXPANDED BLADE AREA RATIO	0.547
MATERIAL	CUNIAL
MINIMUM TENSILE STRENGTH	647 N/MM²
MINIMUM ELONGATION	20 %
MASS	23000 KG
MASS MOM. OF INERTIA PROP. IN AIR	46575 KGM²
MASS MOM. OF INERTIA OF ENTR. WATER	16940 KGM²
ENGINE OUTPUT	13970 KW
PROPELLER REVOLUTIONS	94.00 RPM
CLASSIFICATION	BUR.VERITAS

22-7-83

Meelgewend

Maart 1983

PITCH SELECTED FOR 9000 KW 94.00 RPM,SERVICE CONDITION.

FOR TANKTESTING ONLY

1		INITIALS	THE DESIGN SHOWN ON THIS DRAWING IS THE ABSOLUTE PROPERTY OF -LIPS PROPELLER WORKS- IT MAY NOT BE COPIED OR REPRODUCED. IT MAY NOT BE EXECUTED BY,USED FOR OR SHOWN TO THIRD PARTIES WITHOUT THE WRITTEN CONSENT OF THE OWNER.
		DATE	
	PITCH	DESCRIPTION	HYDRODYNAMIC DESIGN DEPARTMENT
012		REVISION	LIPS PROPELLER WORKS DRUNEN,THE NETHERLANDS

F.P.PROPELLER FOR NEWBUILDING NRS 929-31

CLIENT V.D.GIESSEN-DE NOORD

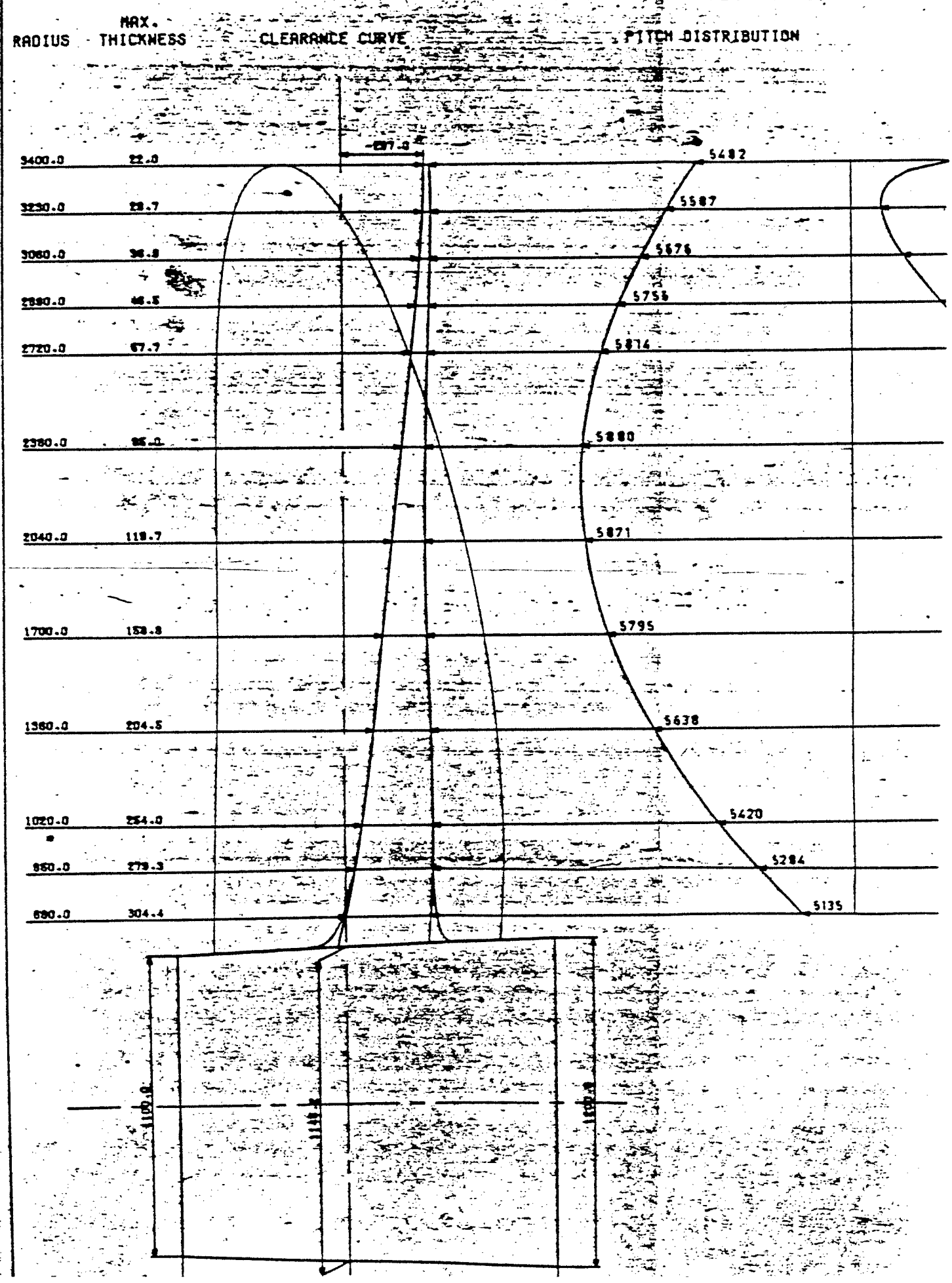
F7856-859

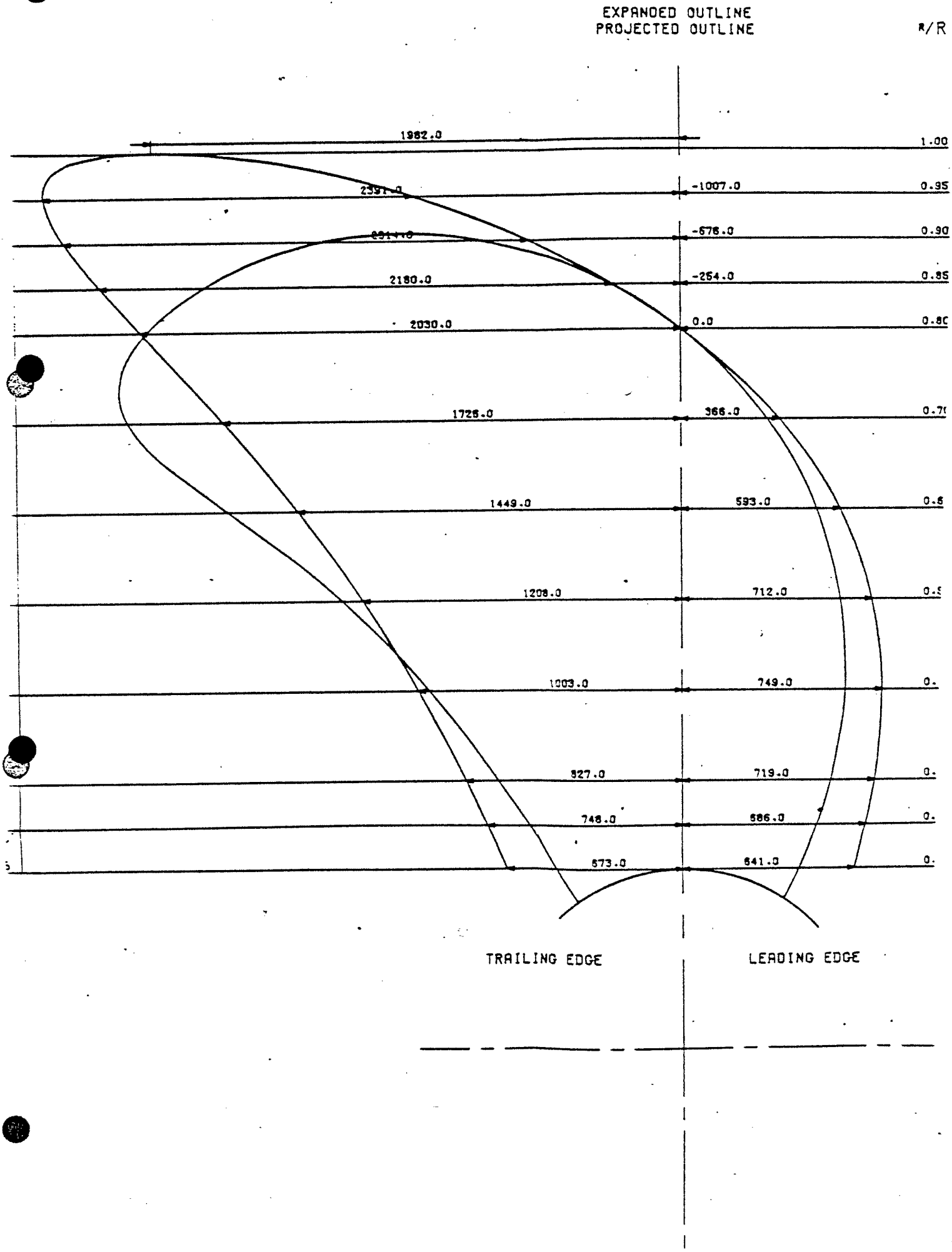
DATE 82/09/08

SCALE 1:12.5

DRAWING F7856-012

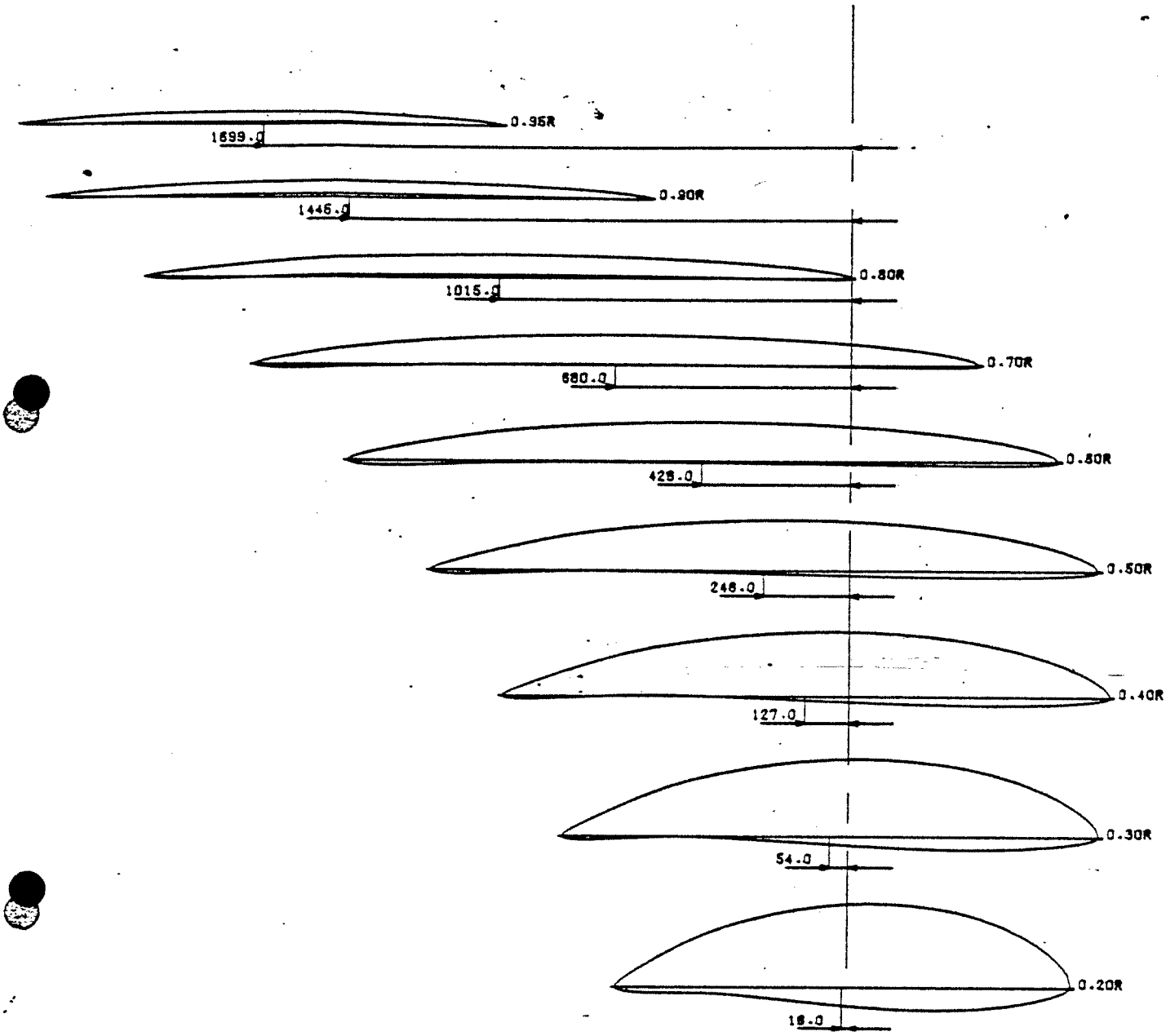
SIZE A1


RADIUS
MAX. THICKNESS
CLEARANCE CURVE
PITCH DISTRIBUTION
-697.0
3400.0 22.0 5482
3230.0 28.7 5587
3060.0 36.8 5676
2890.0 46.5 5756
2720.0 67.7 5874
2380.0 95.0 5880
2040.0 119.7 5871
1700.0 153.8 5795
1360.0 204.5 5638
1020.0 254.0 5420
850.0 279.3 5284
680.0 304.4 5135


EXPANDED OUTLINE
PROJECTED OUTLINE
R/R
1982.0
1.00
2391.0
-1007.0
0.95
2314.0
-676.0
0.90
2180.0
-254.0
0.85
2030.0
0.0
0.80
1726.0
366.0
1449.0
593.0
1208.0
712.0
1003.0
749.0
827.0
719.0
748.0
686.0
673.0
641.0
TRAILING EDGE
LEADING EDGE

7-1-13

BLADE SECTIONS



PROPELLER DATA

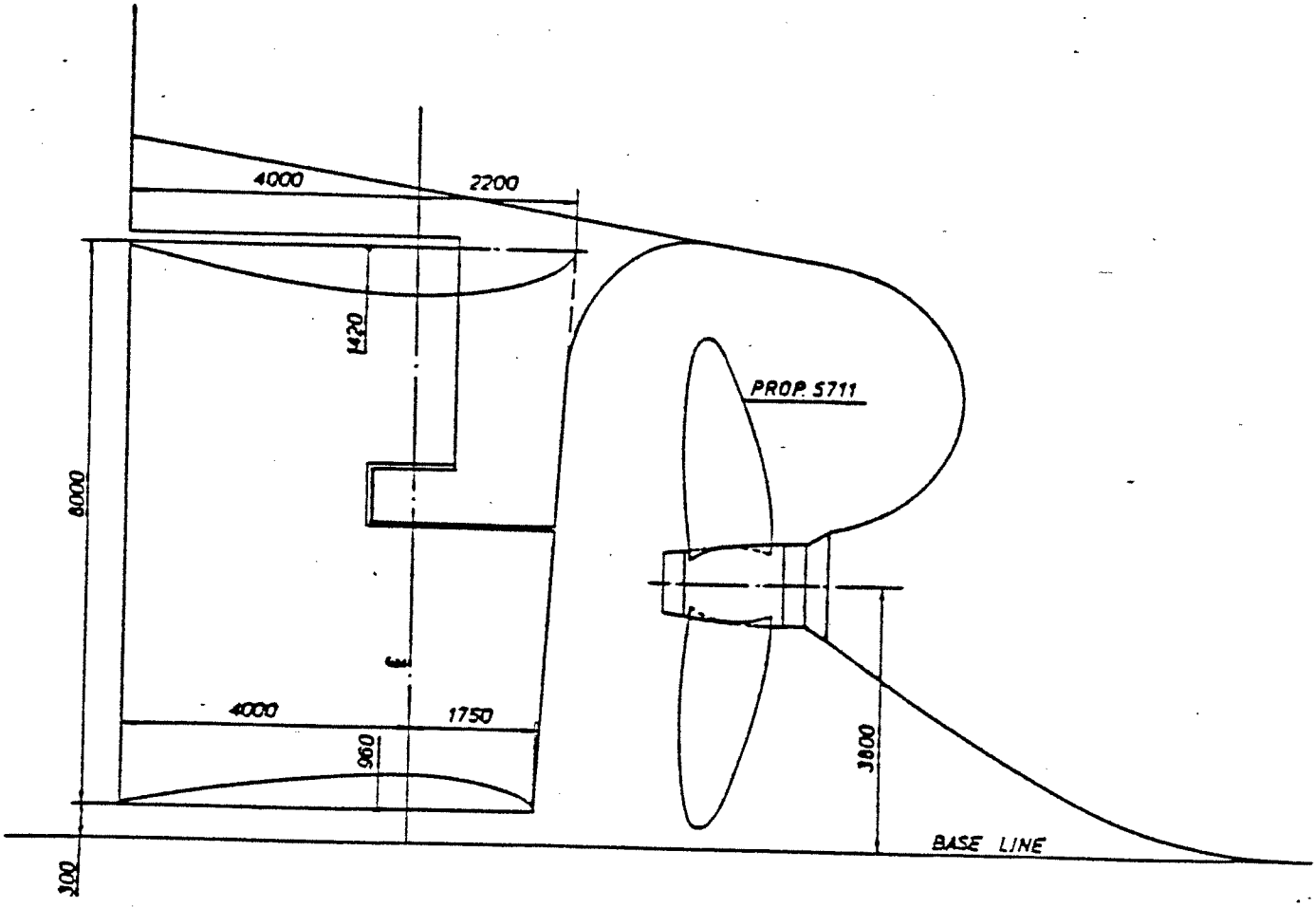
DIAMETER	6800 MM
PITCH.	5723 MM
NUMBER OF BLADES	4
DIRECTION OF ROTATION	RIGHT-HANDED
EXPANDED BLADE AREA RATIO	0.547
MATERIAL	CUNIAL
MINIMUM TENSILE STRENGTH	647 N/MM²

22-7-83

NSMB
Netherlands Ship Model Basin

SHIP MODEL No. 5943B

RUDDER II AND SCREW APERTURE II



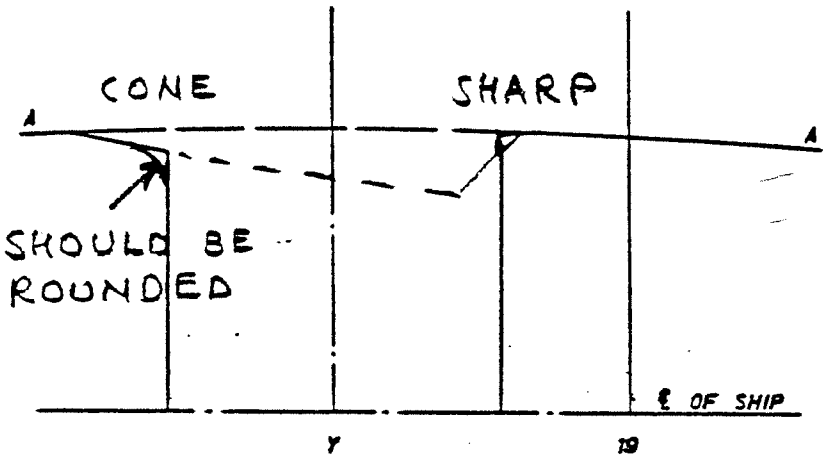
DRAWING No M 5943.33

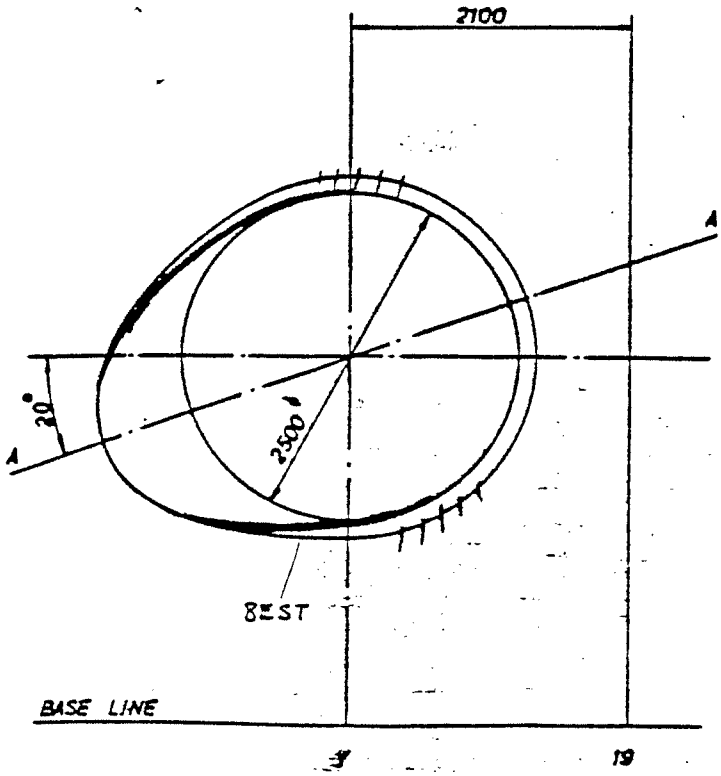
NSMB

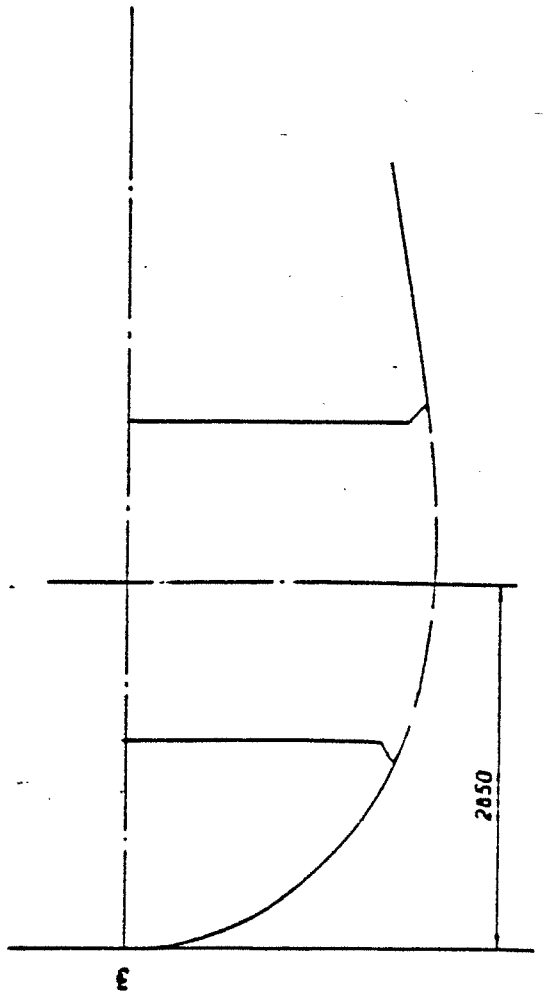
Netherlands Ship Model Basin

SHIP MODEL No. 5943 B

BOW THRUSTER TUNNEL II AND FAIRINGS II







DRAWING No. M 5943_52

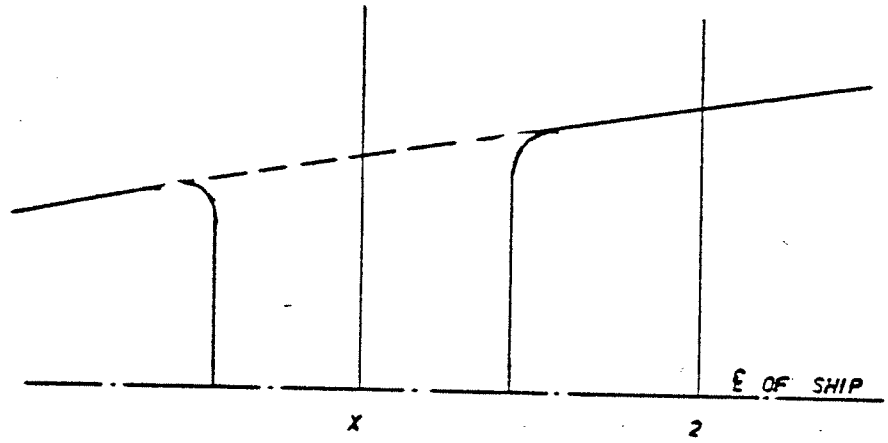
7-1-16

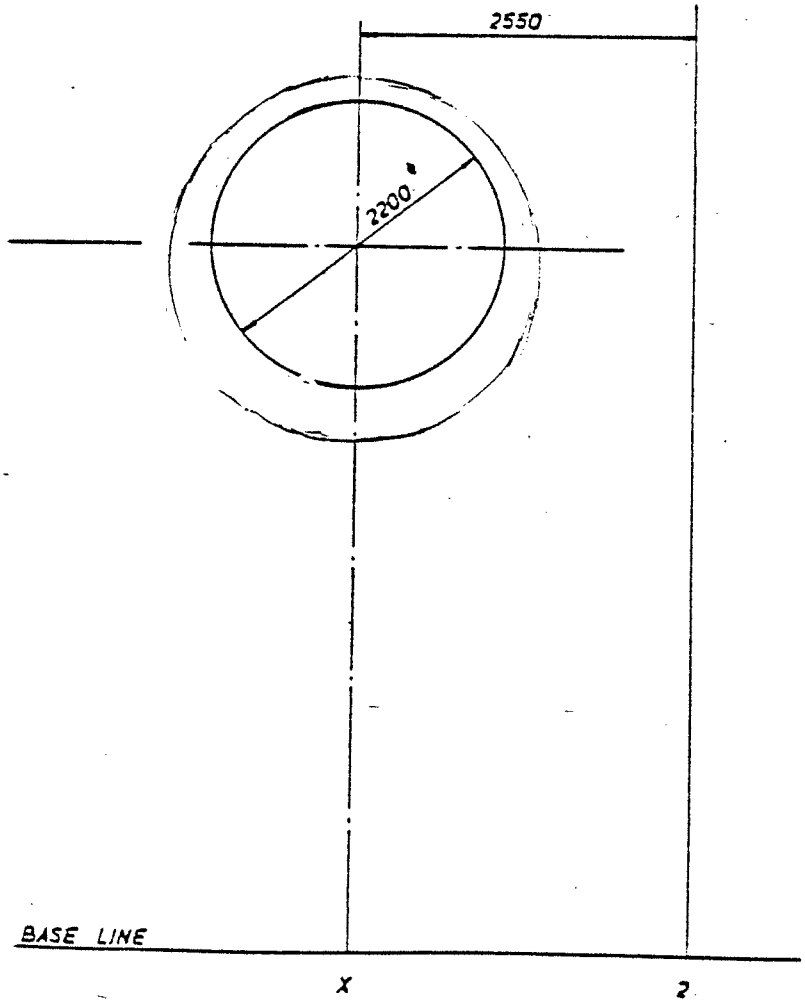
NSMB

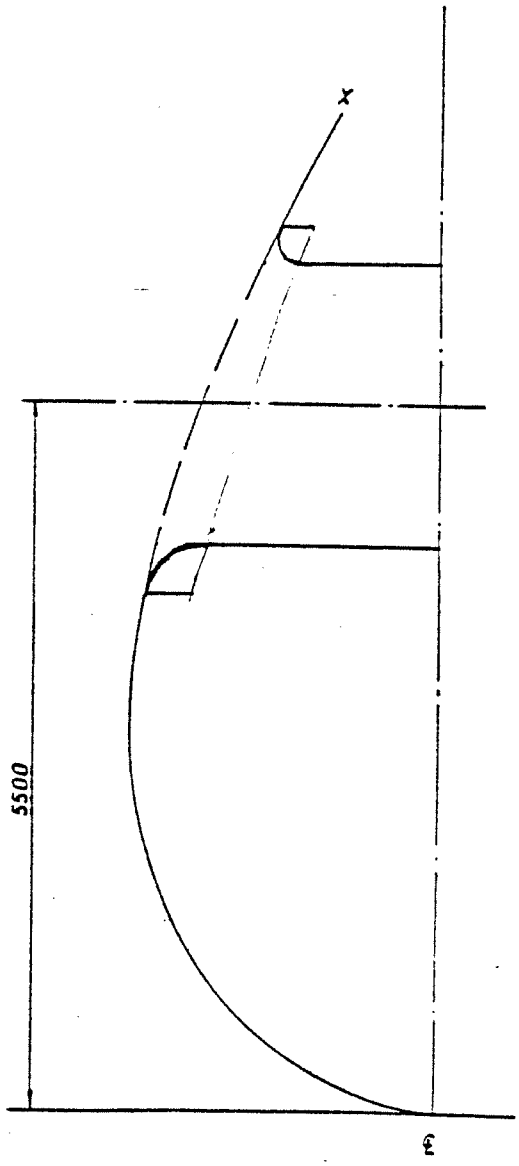
Netherlands Ship Model Basin

SHIP MODEL No 5943 B

STERN THRUSTER TUNNEL I AND FAIRINGS I







DRAWING No M 5943_36

7-2-1

MAIN ENGINE:	SULZER 5RLB90
MCR	17500 HP at 94 RPM
service rating	14875 HP at 94 RPM
(85%) = ± 51 ton/day.	
Fuel for main eng.:	HFO max.600 cSt at 50°
Fuel for aux.eng.:	IFO = 50%HFO/50%MDO
Heavy Fuel Oil consumption excl.	
shaft generator in tons/day	
SHP service = $SHP_{tank} \times C_1 \times C_2$	
C_1 = margin for trials = 1,06	
C_2 = sea margin = 1,12 (service allowance)	
Spec. fuel consumption = 143 gr/SHP.h	

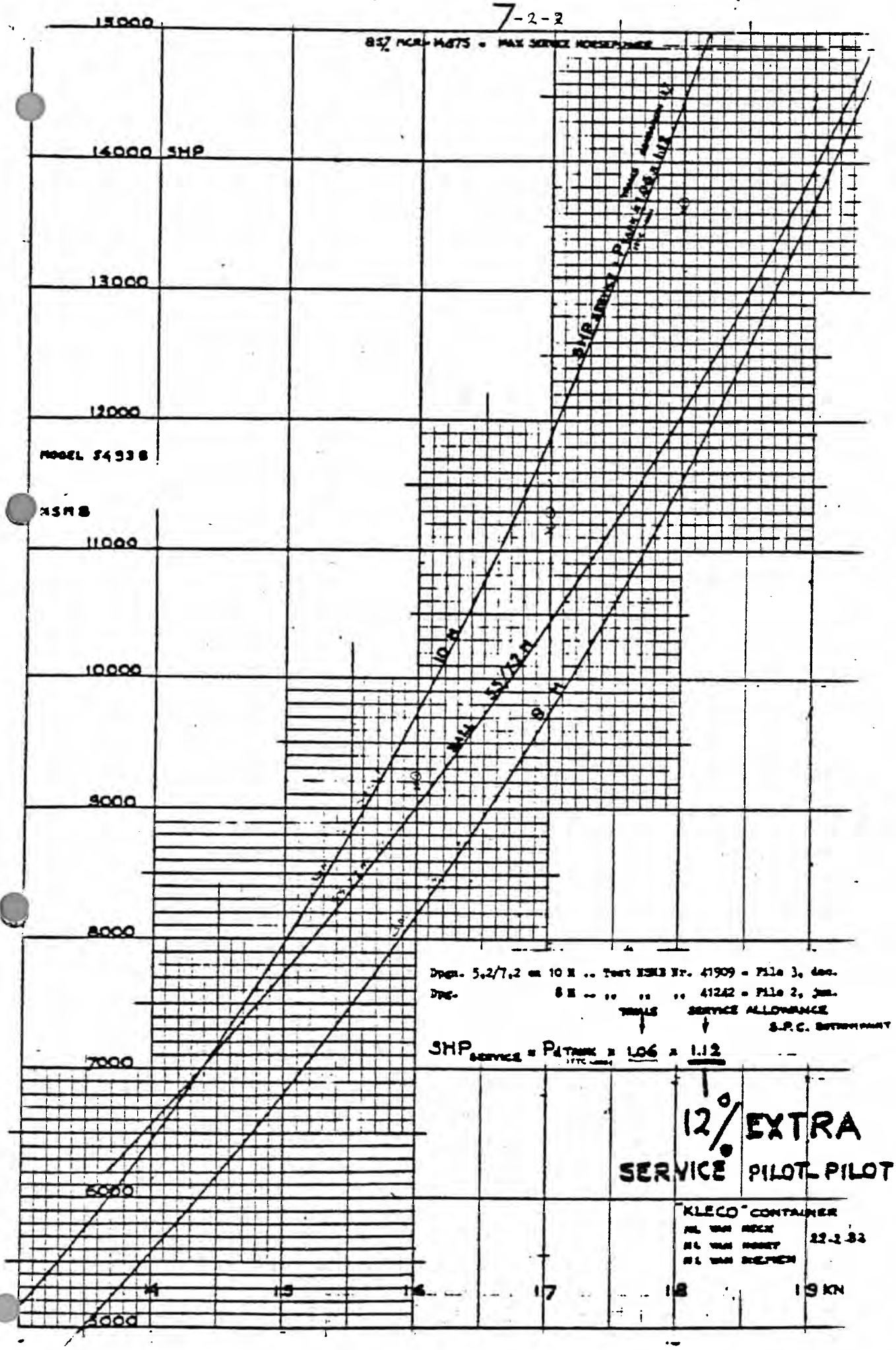
Draft	15.5	16	16.5	17	17.5	18	18.5	19kn
7M	26.5	29.0	31.4	34.1	36.9	39.9	43.1	46.5
8M	25.5	28.1	30.7	33.4	36.4	39.6	43.1	46.7
9M	26.7	29.4	32.2	35.1	38.2	41.5	45.3	49.1
10M	30.1	33.4	37.1	40.9	45.1	49.7	55.0	ton/day

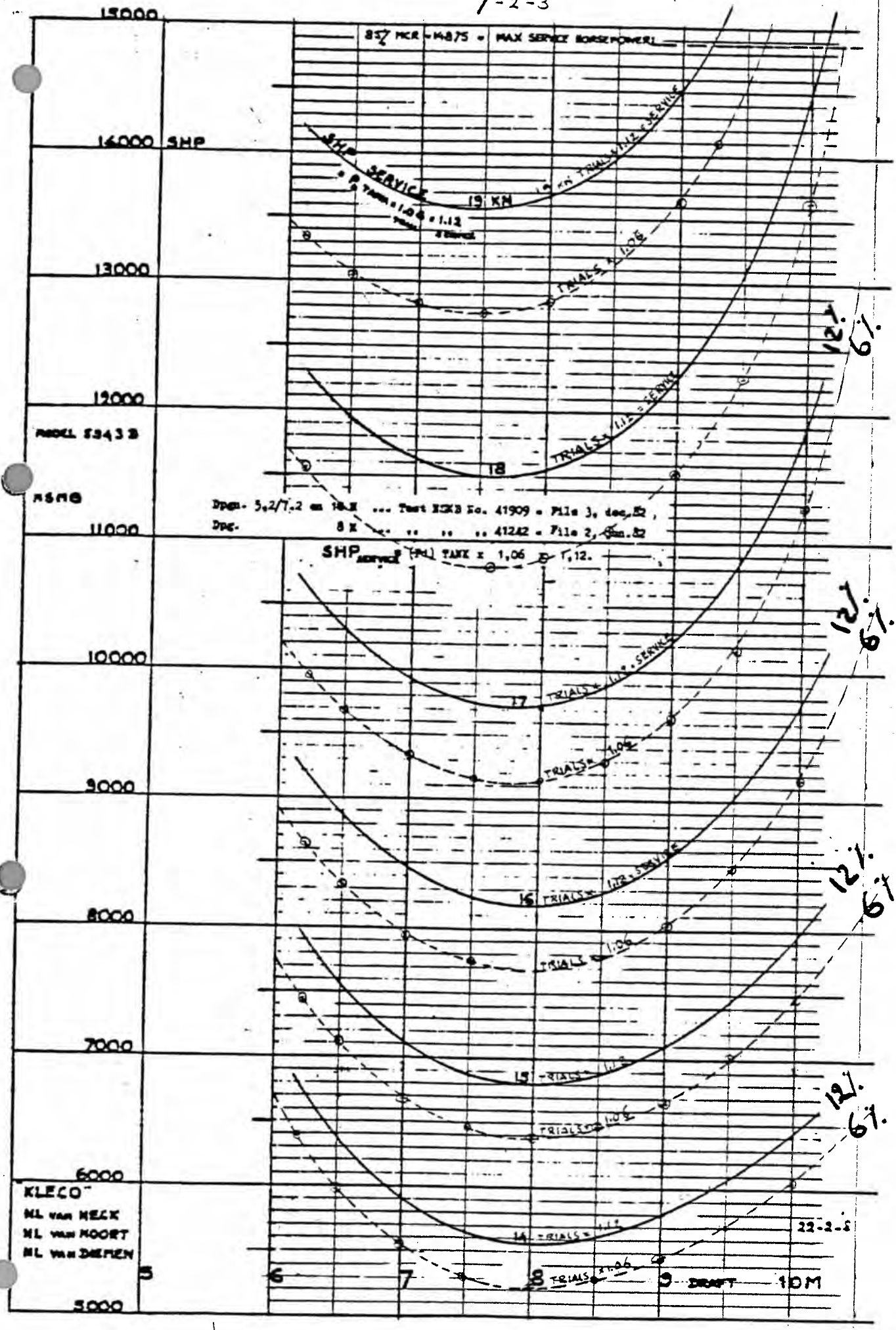
FUEL CONSUMPTION EXCL SHAFTGE
SEA SERVICE PILOT-PILOT
12% EXTRA TO TRIALS
(SPC BOTTOM PAIN

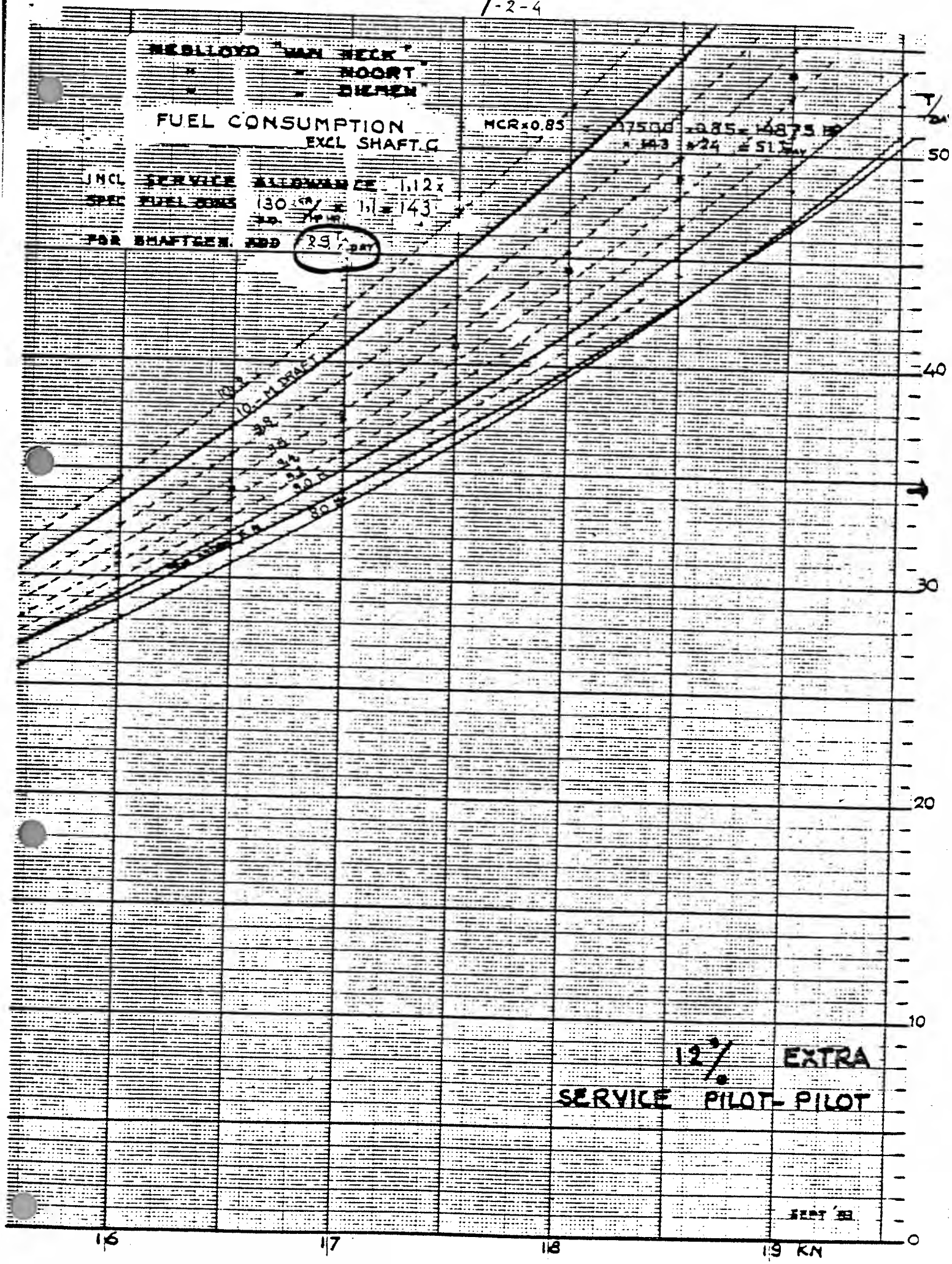
Shaft generator max.	1300 kW
Normal: 540 kW x $\frac{1,36}{.85}$ =	860 HP
x 143 gr/SHP.hr x 24 =	2,9 T/day
Diesel generators	
Diesel gen. I:	1500kW-720RPM
Diesel gen. II:	1500kW-720RPM
Diesel gen. III:	700kW-720RPM
Emergency gen.:	160kW-1800RPM
Normal electr. load at sea	540kW
-IFO diesel gen.	2,9T/day
-HFO shaft gen.	2,9T/day
Idle in port: 400kW IFO	2,3T/day
Boiler: HFO	1,0T/day
Extra for reefers (100 integr. 20')	
box x kW/box x HP/kW x	
gr/HP.hr x hr =	
100 x 5,0 x 1.5 x 0.143x24 =	2,6T/day

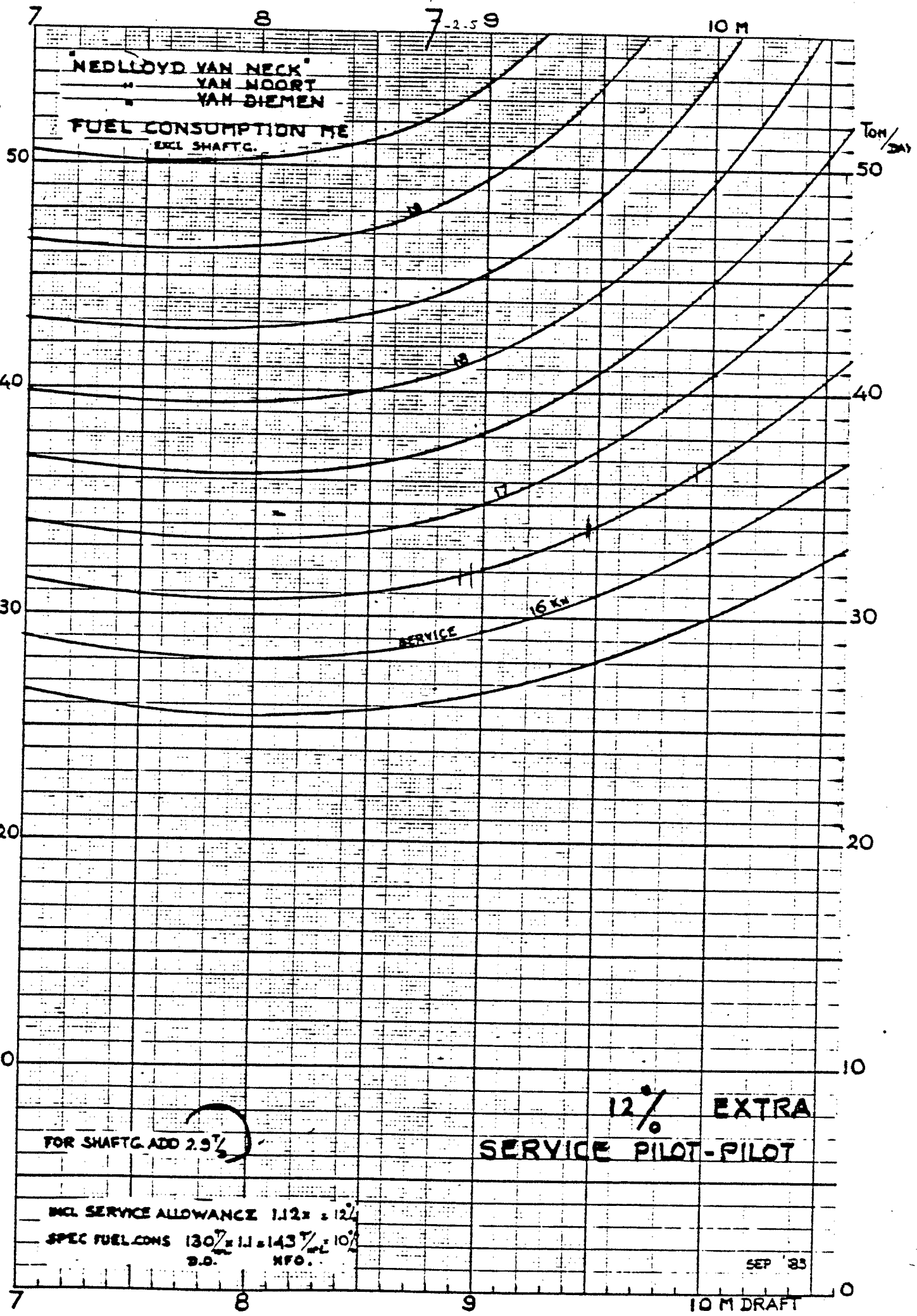
THRUSTERS (var. pitch)
Bowthruster 900 kW, 1220 HP
Sternthruster 900 kW, 1220 HP
Rudder max. angle 45°

INTEGRAL REEFERS
Reeferpoints on deck 20'/40':
BOX = 90 à 5,5 kW 20'/7,5 kW 40'
Reeferpoints in hold 20' BOX = 60
ventilation 1 m3/sec per box
Receptacles 440V, CEE 17, 3h, 32 AMP


85% MCR-14875 = MAX SERVICE HORSEPOWER
15000
14000 SHP
13000
12000
MODEL 5493 B
XSMB
11000
10000
9000
8000
7000
6000
5000
10 ft
8 ft
14
15
16
17
18
19 KN
Dpgn. 5,2/7,2 m 10 ft .. Test ESRA Nr. 41909 = File 3, dec.
Dpg. 8 ft .. " " " 41242 = File 2, jan.
TRIALS
SERVICE ALLOWANCE
S.P.C. STANDARD
SHP SERVICE = P4 TRIAL x 1.06 x 1.12
12% EXTRA
SERVICE PILOT-PILOT
"KLECO" CONTAINER
22-2-82


85% MCR = 14875 = MAX SERVICE HORSEPOWER
15000
16000 SHP
13000
19000
MODEL 5343 B
ASMB
11000
10000
9000
8000
7000
6000
5000
SHP SERVICE
19 KM
TRIALS x 1.12 = SERVICE
TRIALS x 1.06
18
17
16
15
14
12%
6%
Dpm. 5,2/7,2 m 18 K ... Test ZKB No. 41909 = File 3, dec.82
Dpc. 8 K .. .. 41242 = File 2, Jan.82
SHP SERVICE (PM) TANK x 1,06 x 1,12.
KLECO
NL van HECK
NL van MOORT
NL van DIEMEN
5
6
7
8
9
DRAFT
10M
22-2-5


NEDLLOYD "VAN NECK"
" " NOORT"
" " BREMEN"
FUEL CONSUMPTION
EXCL SHAFT.G
MCR×0.85
17500 ×0.85 = 14875 HP
× 143 ÷24 = 51 T/DAY
INCL SERVICE ALLOWANCE 1.12 ×
SPEC FUEL CONS 130 GM/HP HR × 1.1 = 143
FOR SHAFTGEN. ADD 29 T/DAY
10.3 M DRAFT
9.8
9.4
9.0
8.6
8.0
T/DAY
50
40
30
20
10
0
12% EXTRA
SERVICE PILOT-PILOT
SEPT '83
16
17
18
19 KN


7
8
7-2.5 9
10 M
"NEDLLOYD VAN NECK"
" VAN NOORT
" VAN DIEMEN
FUEL CONSUMPTION ME
EXCL SHAFTG.
TON/DAY
50
40
30
20
10
0
16 KN
SERVICE
12% EXTRA
SERVICE PILOT-PILOT
FOR SHAFTG. ADD 2.9 T/D
INCL SERVICE ALLOWANCE 1.12 x = 12%
SPEC FUEL CONS 130 T/DAY x 1.1 = 143 T/DAY = 10%
D.O. HFO.
SEP '83
7
8
9
10 M DRAFT

NEDLLOYD "VAN NOORT" 7-2-6

OPTIMAL EASTB 1426 TEU x 14,2 T/TEU 9,94 m DRAFT
" WESTB 1590 " x 9,5 " 8,95 m "

ROUNDTRIP SCANDUTCH "S"

AT SEA	HFO	IFO	DAYS
23.370 MILES PILOT-PILOT AT 16,5 KN 1416 NRS = 59 DAYS DRAFT MEAN 9,45 m 59 DAYS x (34 T/D MAIN E + 2,9 T/D) x 59 x 36,9 = SHAFT G 540 KW.	2180		59 = 65%
IN PORT			
23 PORTS IN 23 DAYS 23 DAYS (AUXIL 400 KW 2,3 T/D) = 23 x 2,3 = BOILER 23 x 1 =	23	53	23 = 25%
MANOEUVRING			
9 DAYS x (AUXIL 2,9 + 2,3 T/D) = 9 x 5,2 = BOILER 9 x 1 =	9	47	9 = 10%
FUEL CONSUMPTION P. ROUNDTRIP EXCL REEFERCONTRS.	2212 T	100 T	91 = 100%

Electrische belasting op zee 540 kW

$$\frac{540 \times 1,36}{0,85\ (AEZ)} = 860 \text{ pk}$$

860 x 142 x 24 = 2,9 Ton/dag H.F.O.

Electrische belasting in haven 400 kW

$$\frac{400 \times 1,36 \times 24}{0,92} = 2,3 \text{ Ton/dag I.F.O.}$$

15/9/83

SLAGEN LIJSTJE - BRANDST. VERBR

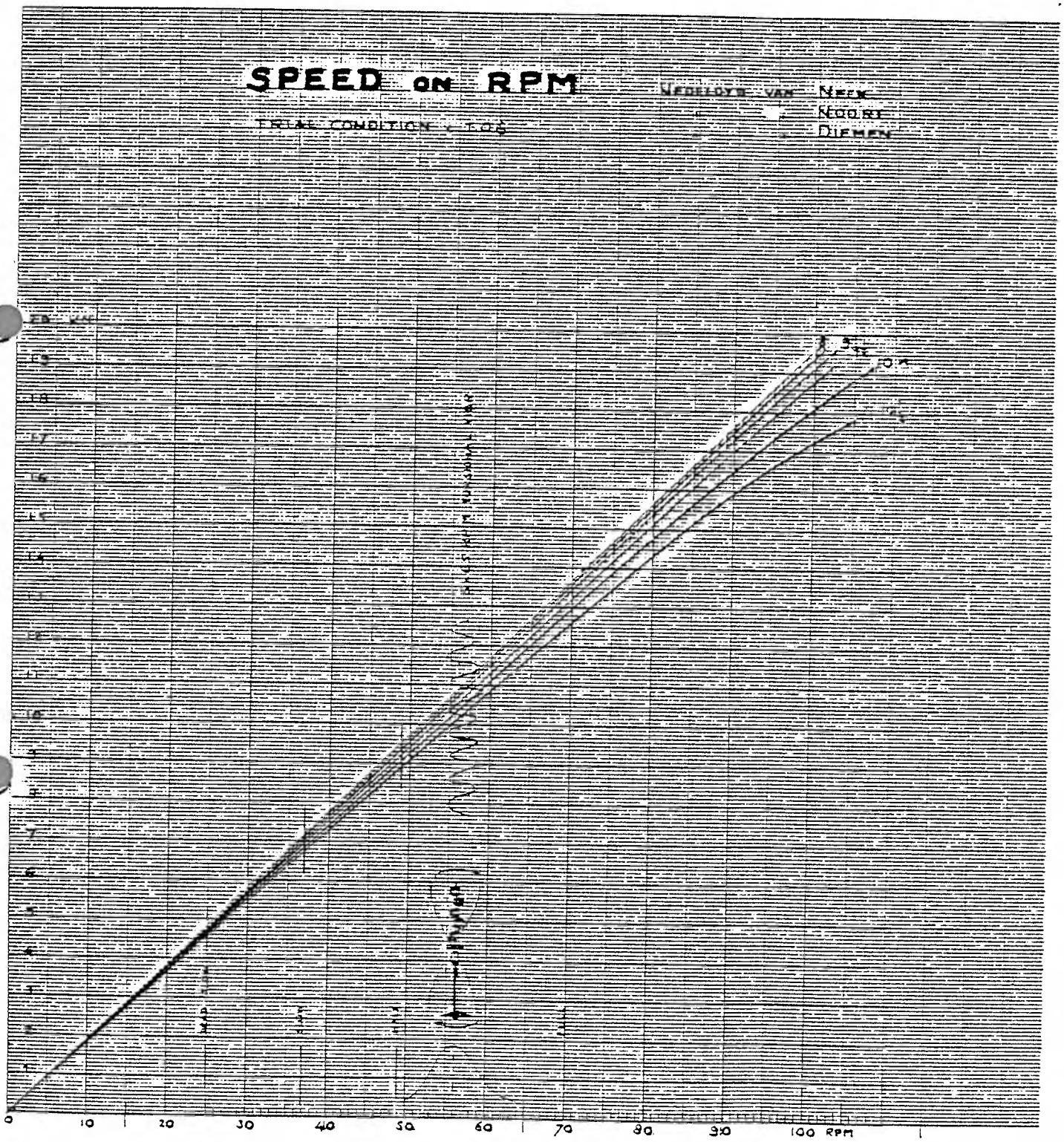
NEDLLOYD VAN NECK — NSMB MODEL S343 B PROP 5711A
" NOORT — PROPELLER LIPS 4 BL.
" VAN DIEMEN — ϕ = 6800 H_{mean} = 5723 $H_{4/5E}$ 5980

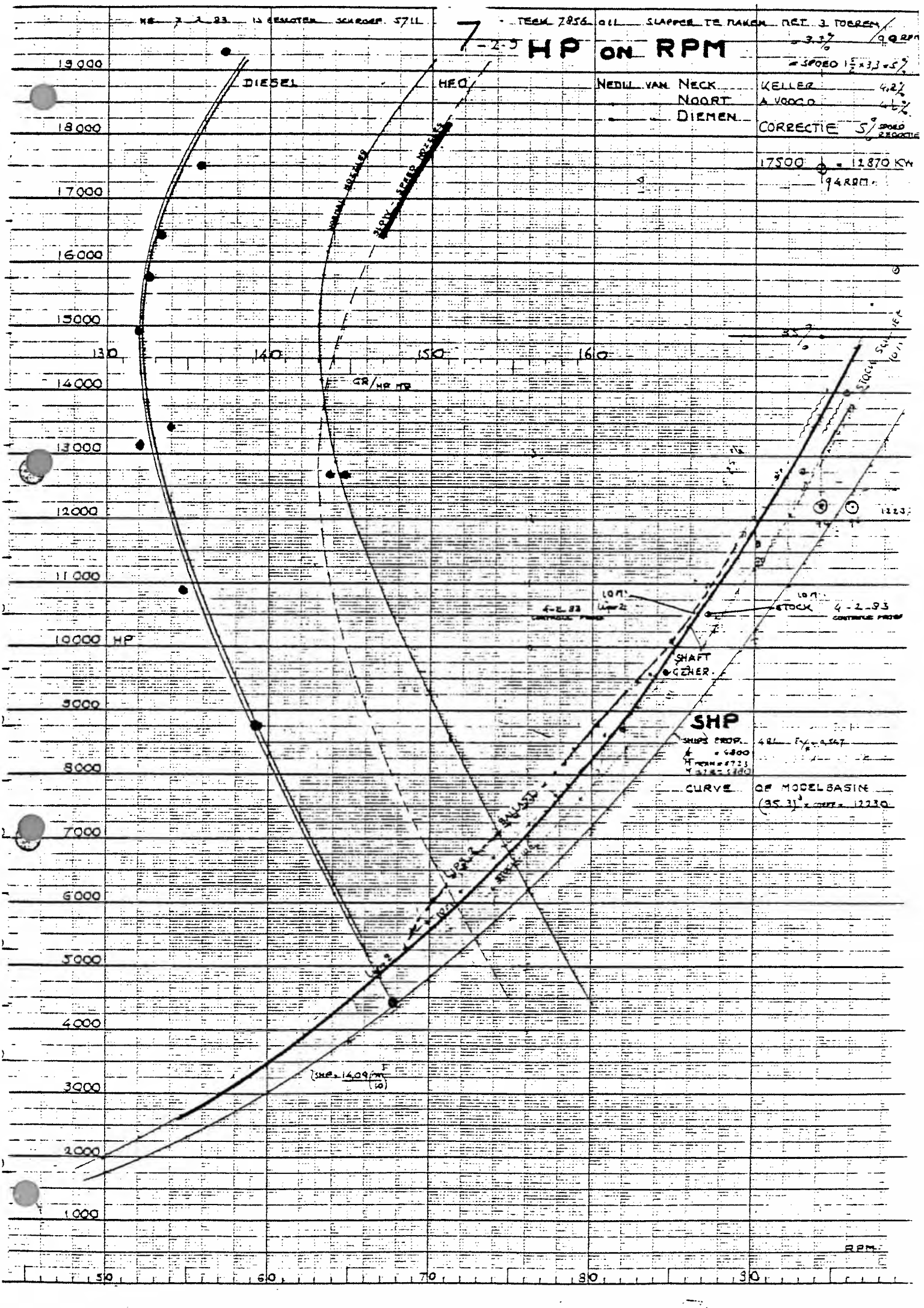
NORMAL FUEL NOZZLES

6% EXTRA HP ABOVE TRIALS

	RPM	SHP PROPELL	+860 =	BHP ENGINE	% HFO HP-HR	KG/HR	TON/DAY	8.-M KN	TON/1000'	8.5 KN	TON/1000'	9.- KN	TON/1000'	9.5 KN	TON/1000'	10.- KN	TON/1000'	10.5 KN	TON/1000'	RPM
DEAD SLOW	25							4.8				4.7				4.6				
SLOW	37							7.2				7.0				6.8				
HALF	49							9.5				9.3				9.1				
CRITICAL! TORSION	56																			
FULL	70	4830	+860 =	5690	157.2	895	21.5	13.7 KN	65	13.6	66	13.4	67	13.2	68	13.0	69	12.6	71	70
	71	5040		5900	156.8	925	22.2	13.9	66	13.8	67	13.6	68	13.4	69	13.2	70	12.8	72	71
	72	5260		6120	156.3	956	22.9	14.1	68	14.0	68½	13.8	69	13.6	70	13.4	71	13.0	73	72
	73	5480		6340	155.8	988	23.7	14.2	69	14.1	70	14.0	71	13.8	72	13.5	73	13.2	75	73
	74	5710		6570	155.3	1020	24.5	14.4	71	14.3	71½	14.2	72	14.0	73	13.7	74	13.3	77	74
	75	5940		6800	154.8	1053	25.3	14.6	72	14.5	72½	14.4	73	14.2	74	13.9	76	13.5	78	75
	76	6190		7040	154.4	1086	26.1	14.8	73	14.7	73½	14.6	74	14.4	75	14.1	77	13.6	80	76
	77	6430		7290	153.8	1121	26.9	15.0	74	14.9	75	14.8	76	14.6	77	14.3	79	13.8	81	77
	78	6690		7550	153.3	1157	27.8	15.2	76	15.1	76½	15.0	77	14.8	78	14.5	80	14.0	83	78
	79	6950		7810	152.8	1193	28.7	15.4	77	15.3	77½	15.2	78	15.0	79	14.6	81	14.2	84	79
	80	7210		8070	152.2	1228	29.5	15.6	78	15.5	79	15.4	80	15.2	81	14.8	83	14.4	85	80
	81	7490		8350	151.6	1265	30.4	15.8	80	15.7	80½	15.6	81	15.3	82	15.0	84	14.6	87	81
	82	7770		8630	151.1	1304	31.3	16.0	81	15.9	81½	15.8	82	15.5	83	15.2	86	14.7	89	82
	83	8050		8910	150.6	1341	32.2	16.2	83	16.1	83	16.0	84	15.7	85	15.4	87	14.9	90	83
	84	8350		9210	150.0	1381	33.2	16.4	84	16.3	84½	16.2	85	15.9	86	15.5	88	15.1	92	84
	85	8650		9510	149.4	1420	34.1	16.6	85	16.5	86	16.4	87	16.1	88	15.7	91	15.2	93	85
	86	8960		9820	148.8	1461	35.0	16.8	87	16.7	87½	16.6	88	16.3	89	15.9	92	15.4	95	86
	87	9280		10140	148.2	1503	36.0	17.0	88	16.9	88½	16.8	89	16.5	90	16.0	94	15.5	97	87
	88	9600		10460	147.7	1544	37.0	17.2	89	17.1	90	17.0	91	16.7	92	16.2	95	15.7	99	88
	89	9930		10790	147.1	1588	38.0	17.4	91	17.3	92	17.2	93	16.9	94	16.4	97	15.8	100	89
	90	10270		11130	146.5	1630	39.1	17.6	92	17.5	93	17.3	94	17.0	95	16.5	98	16.0	102	90
	91	10620		11480	146.0	1676	40.2	17.8	94	17.7	95	17.5	96	17.2	97	16.7	100	16.1	104	91
	92	10970		11830	145.4	1720	41.3	18.0	95	17.9	96	17.7	97	17.4	98	16.8	102	16.2	106	92
	93	11330		12190	144.9	1765	42.4	18.2	97	18.1	98	17.9	99	17.6	100	17.0	104	16.4	108	93
	94	11700		12560	144.4	1813	43.5	18.4	98	18.3	99	18.1	100	17.8	102	17.2	106	16.5	110	94
	95	12080		12940	144.0	1864	44.7	18.6	100	18.5	101	18.3	102	17.9	104	17.4	108	16.6	112	95
	96	12460		13320	143.7	1914	45.9	18.8	102	18.7	103	18.4	104	18.1	106	17.5	110	16.8	114	96
	97	12860		13720	143.4	1968	47.2	19.0	103	18.9	104	18.6	106	18.3	108	17.7	112	16.9	116	97
	98	13260		14120	143.2	2024	48.5	19.2	105	19.1	106	18.8	108	18.4	110	17.8	114	17.0	119	98
	99	13670		14530	143.1	2080	49.9	19.4	107	19.3	108	19.0	110	18.6	112	18.0	116	17.1	121	99
	100	14090		14950	143.0	2138	51.3	19.6	109	19.4	110	19.1	112	18.8	114	18.1	118	17.3	124	100
	101	14510		15370	143.1	2200	52.8	19.8	111	19.6	112	19.3	114	19.0	116	18.3	120	17.4	126	101
85%	102	14950		15810	143.3	2266	54.4	19.9	113	19.8	114	19.5	116	19.2	118	18.4	122½	17.5	129	102
24-11-83.	103	15390		16250	143.7	2336	56.0	20.1	115	20.0	116	19.7	118	19.3	120	18.6	125½	17.6	132	103

7-2-7


SPEED ON RPM
TRIAL CONDITION
NEDERLOYD VAN NEEK
NOORD
DIEMEN
0
10
20
30
40
50
60
70
80
90
100 RPM


7-2-93 HP ON RPM
DIESEL
HFO
NEDW. VAN NECK
NOORT
DIEMEN
KELLER
A. VOOGD
CORRECTIE
17500 = 12870 KW
94 RPM
GR/HP HR
HP
SHAFT
GEMER.
SHP
CURVE OF MODELBASIN
RPM

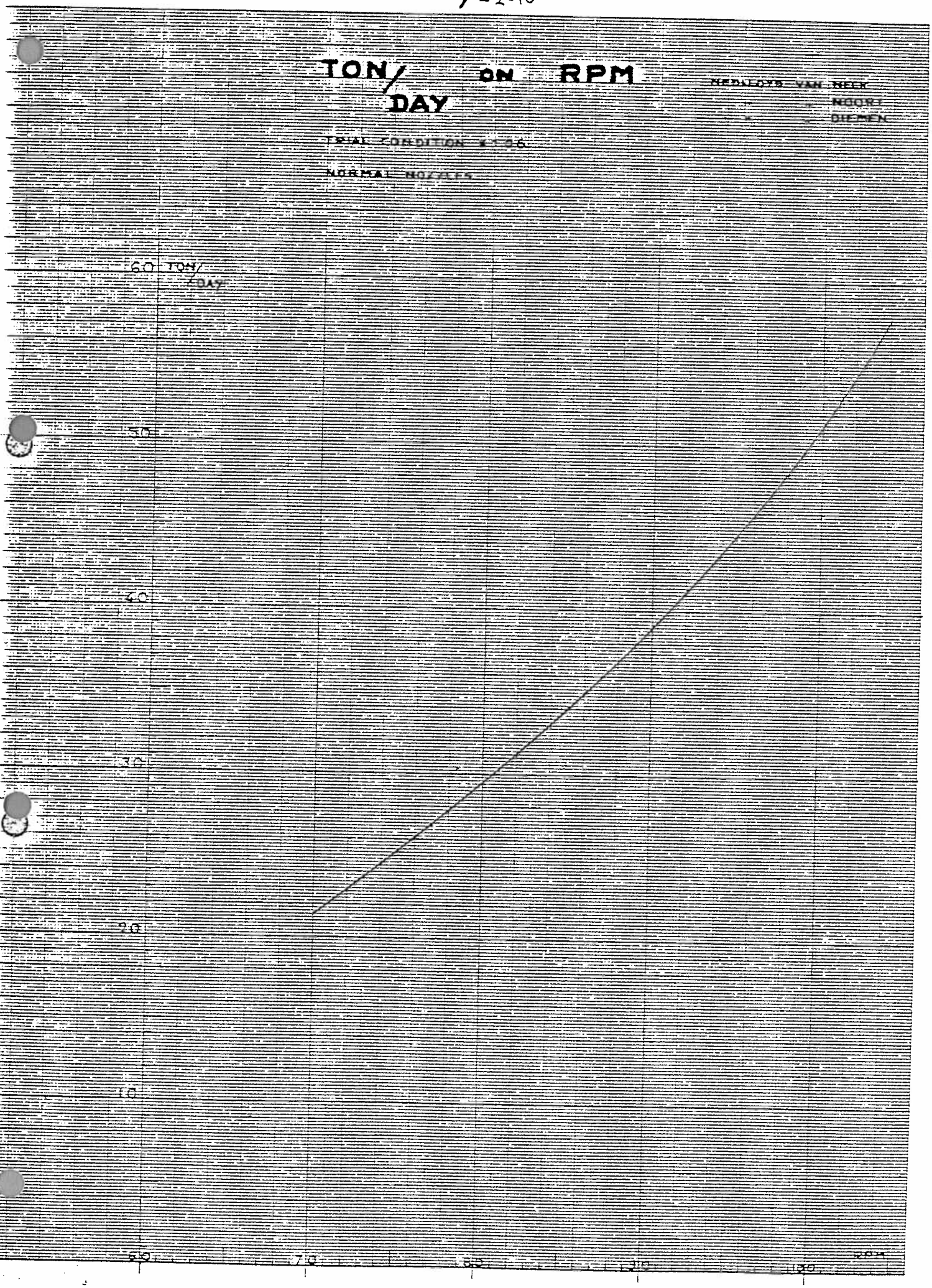
7-2-10

TON/DAY ON RPM

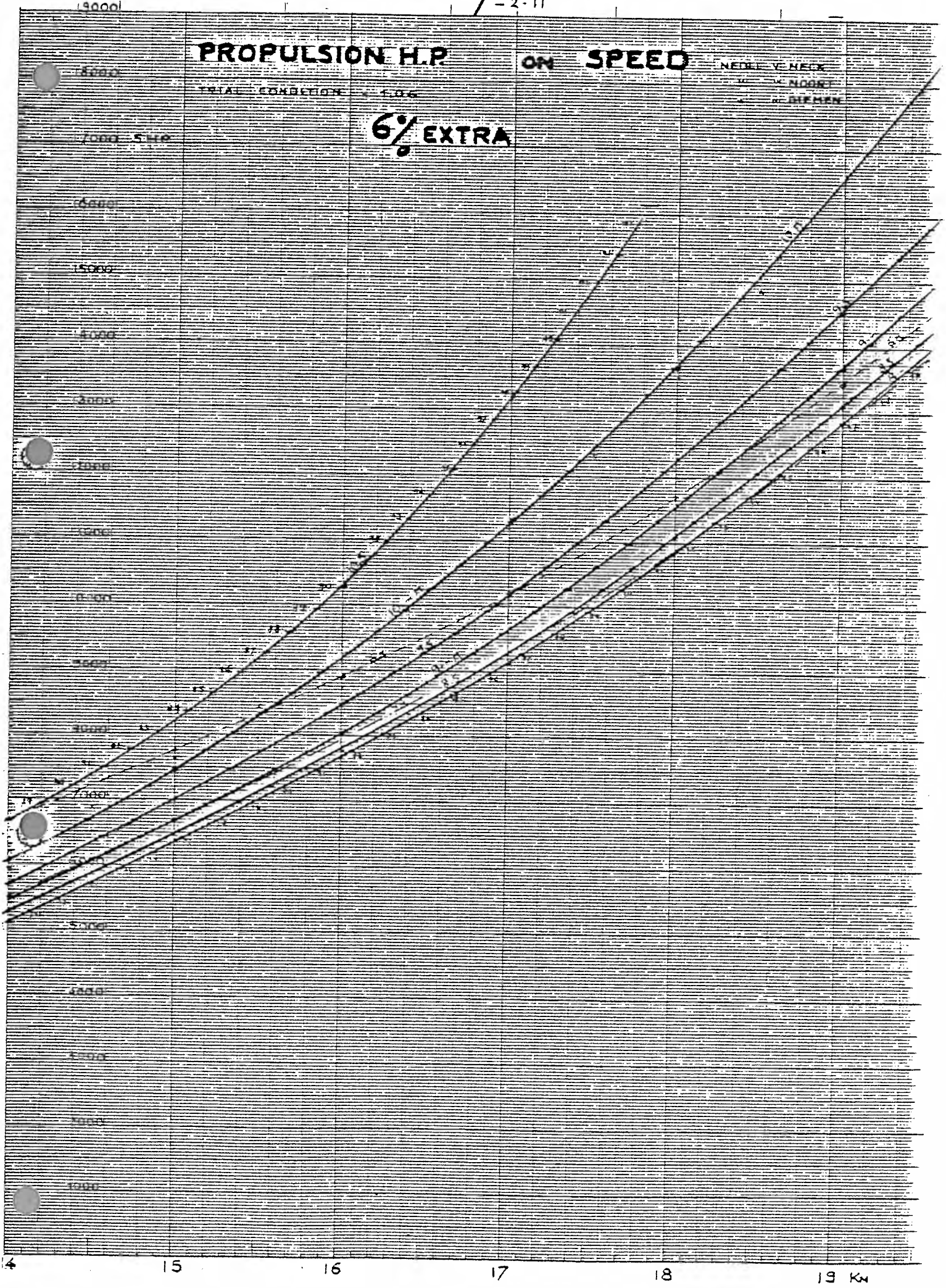
NEDAMOYB VAN HECK
" NOORT
" DIEMEN

TRIAL CONDITION 8-10-66

NORMAL NOZZLES



7-2-11


PROPULSION H.P. ON SPEED
TRIAL CONDITION = 1.06
6% EXTRA
NEDLLYNECK
NOORT
BREMEN
9000
8000
7000 S.H.P.
6000
5000
4000
3000
2000
1000
14
15
16
17
18
19 Kn

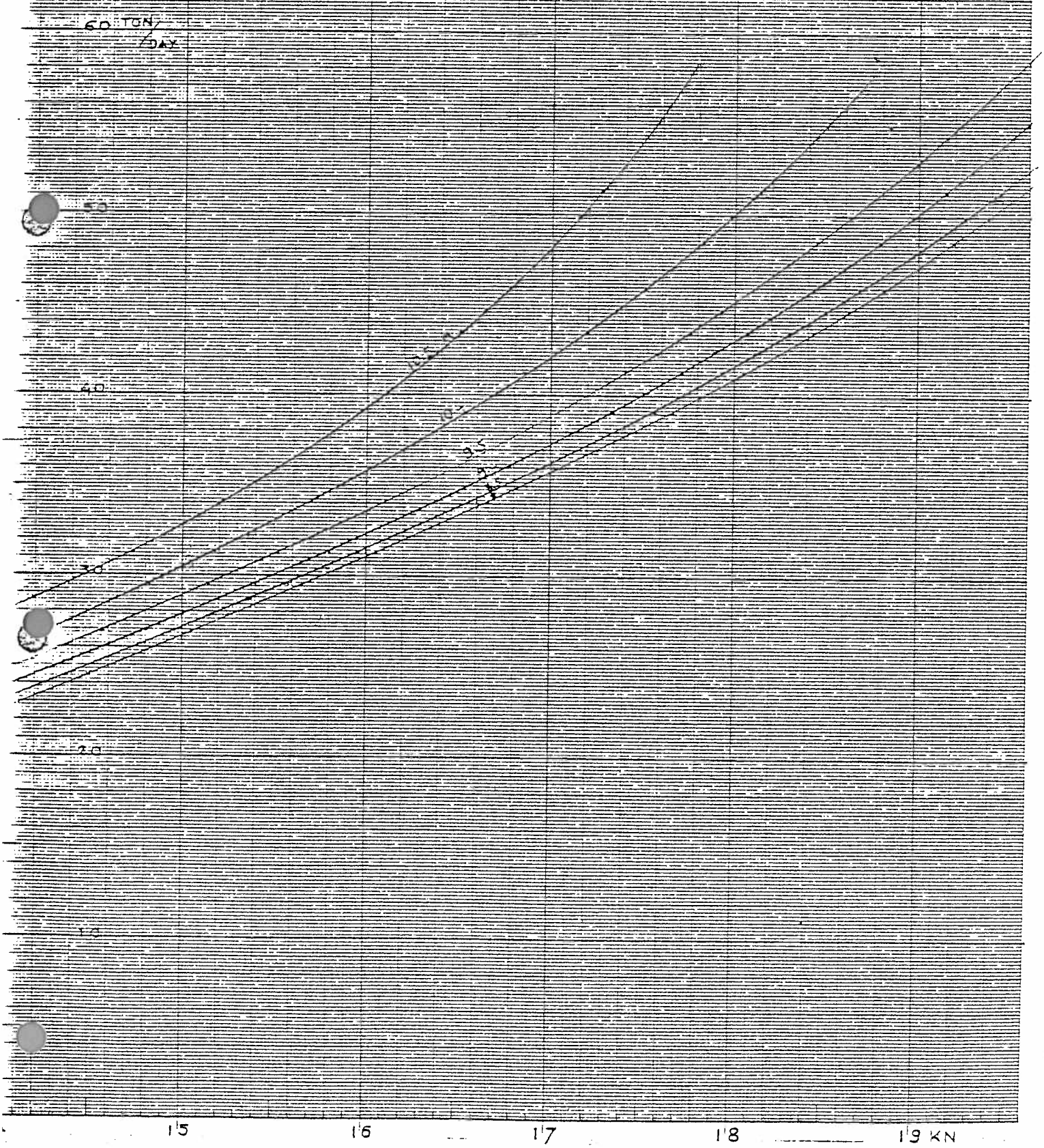
7-2-12

TON/DAY ON SPEED

NEDLLOYD VAN NECK
NOORT
BIEMEN

TRIAL CONDITION 4.1.06

NORMAL NOZZLES



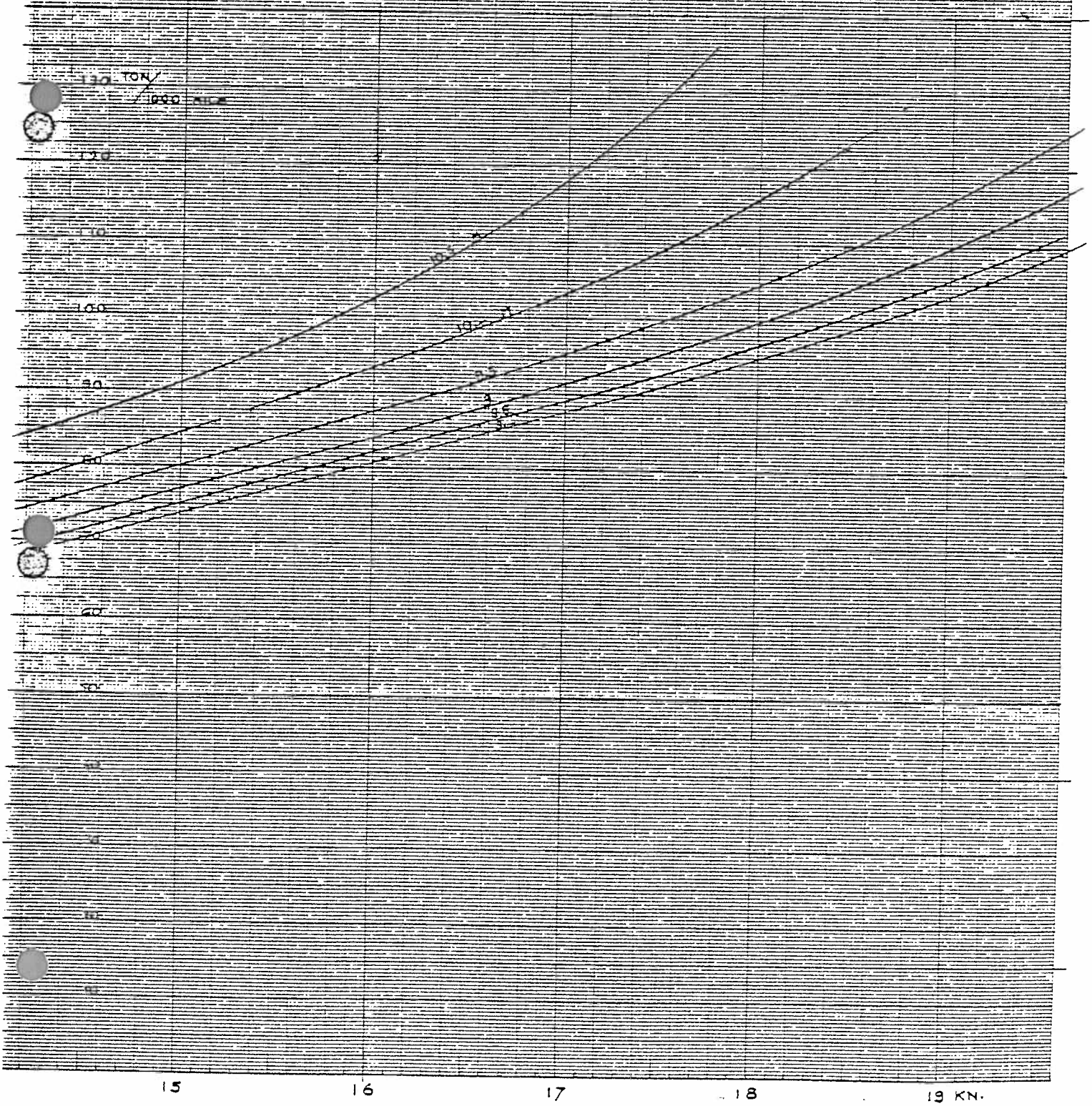
7-2-13

TON/1000 MILE on SPEED

NEDELOYD VAN NECK
NOORT
DIEKEN

TRIAL CONDITION 1.06

NORMAL NOZZLES



ABOUT CARGO HANDLING

SHORE CONTAINERGANTRY

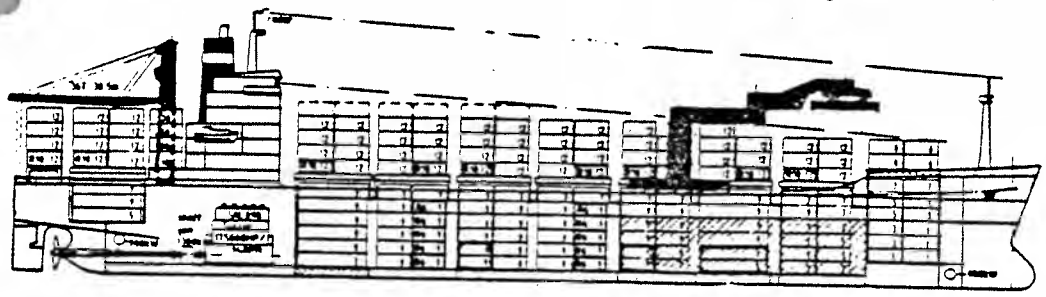
SHIPS' JIB CRANES

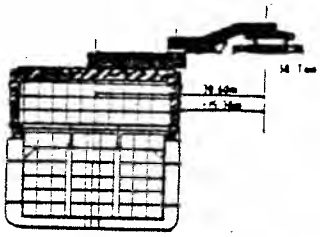
" POCKET-KNIFE CRANE : FIXED

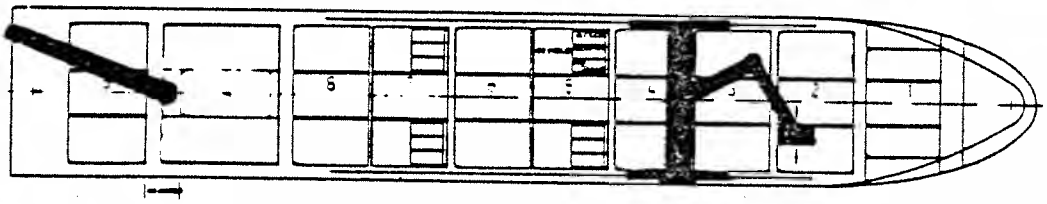
ALTERN: ON GANTRY

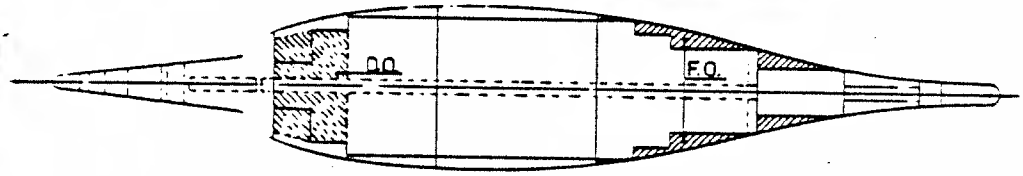
Three NEDLLOYD "KLECO"-class cellular containerships.

(Preliminary particulars as per Nov. 2, 1983.)









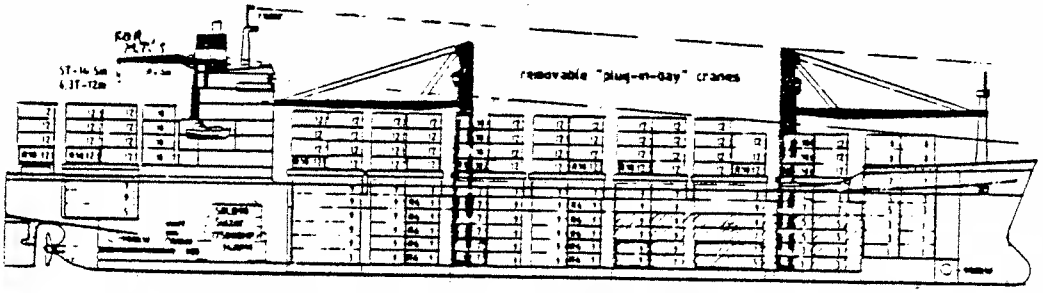
Length over all	182 50M
Length betw perpend	174 00M
Breadth mld	30 50M
Depth mld	16 20M
Draft scantling	11 27M

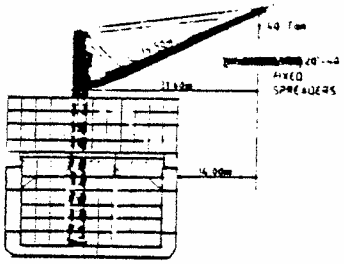
NEDLLOYD van NECK 929

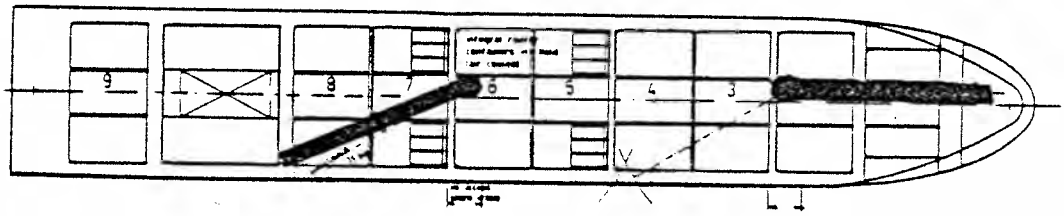
Delivery date : 2 December 1983
Yard v/d Giessen de Noord

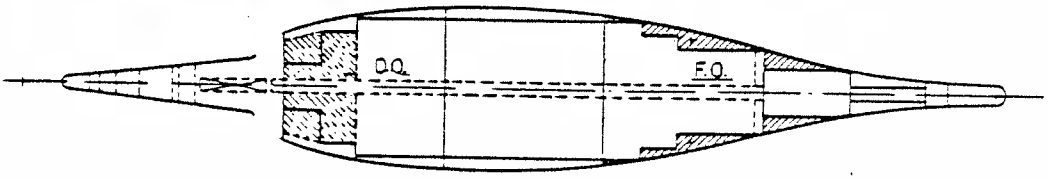
FIRST vessel.

Cont below deck	10976 T = 14 x 784
Cont on deck	9674 = 14 x 691
Homog LOADED	20650 T = 14 x 1475 TEU
CENTR. GRAVITY BOX = 65%	









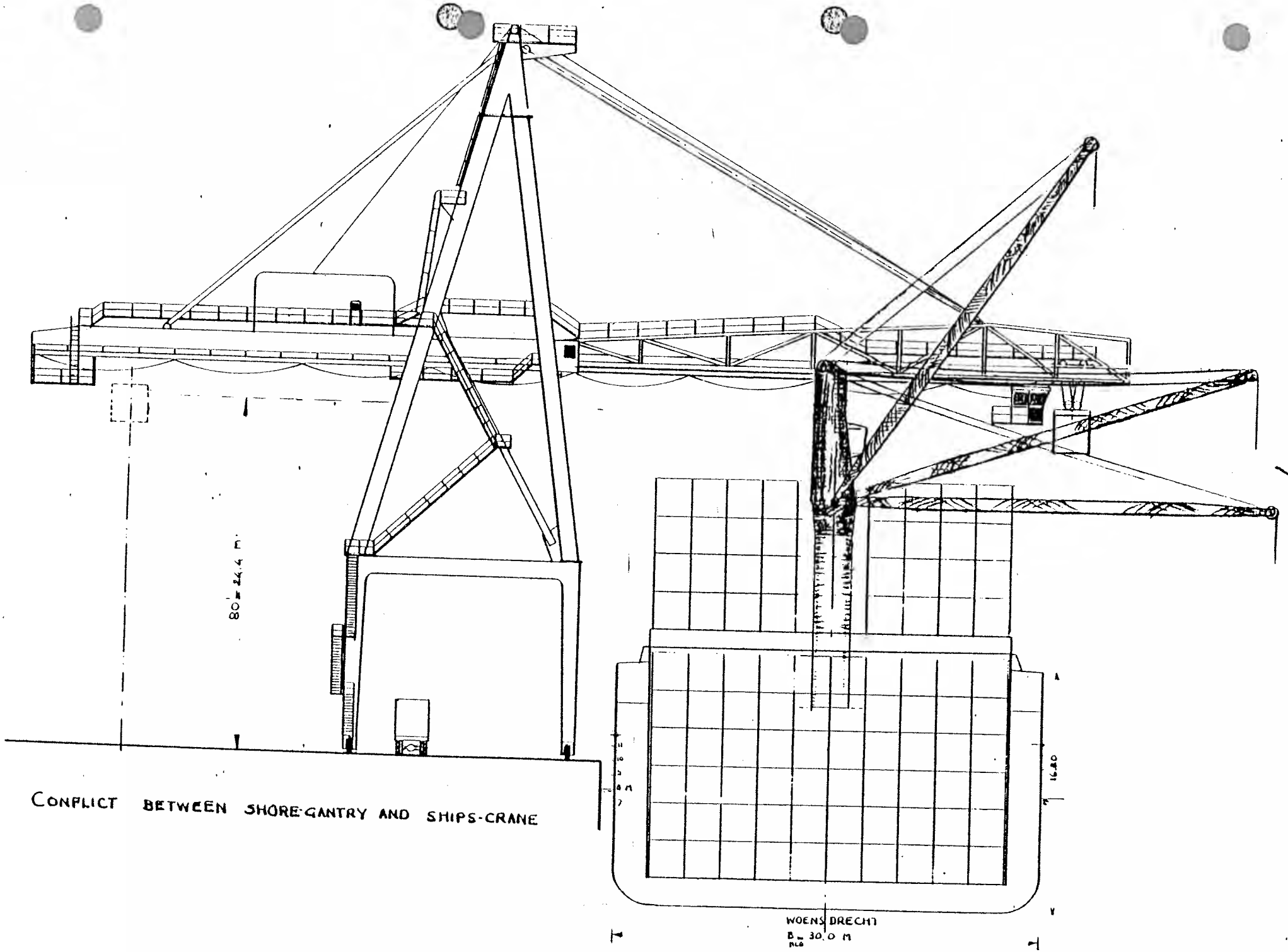
Length over all	182 50M
Length betw perpend	174 00M
Breadth mld	30 50M
Depth mld	16 20M
Draft scantling mld	11 27M

NEDLLOYD van NOORT 930
van DIEMEN 931

Delivery April 1984 / July 1984
Yard v/d Giessen de Noord
Class Bureau Veritas

SECOND & THIRD vessel

Cont below deck	10678 T = 142 x 752
Cont on deck	9571 T = 142 x 674
Homog LOADED	20249 T = 142 x 1426 TEU
CENTR. GRAVITY BOX = 45%	



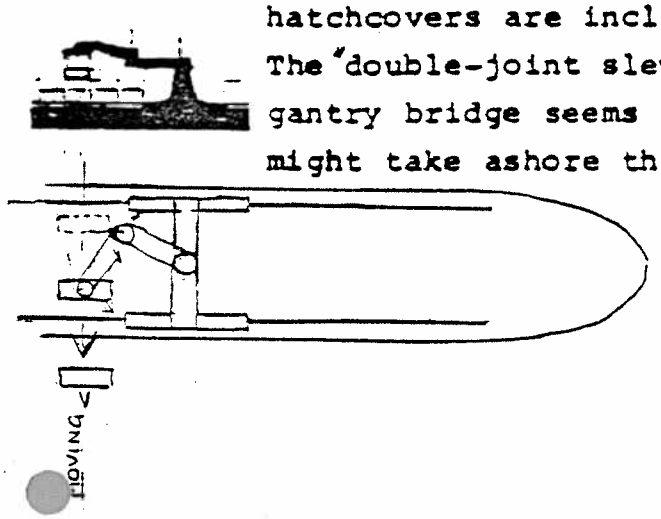
CONFLICT BETWEEN SHORE-GANTRY AND SHIPS-CRANE

GEAR

In case the container vessel has to serve ports which do not have a terminal container crane, the vessel should be geared by 40-ton or 2 x 25 ton cranes.
Following conditions to be fulfilled:

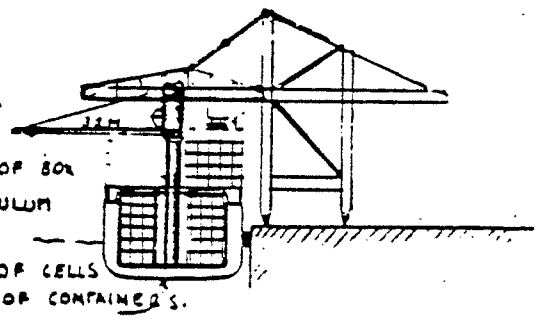
- Ships gear should not give hindrance to container handling by terminal cranes.
- Ships gear should steer the container parallel to the vessel.
- Ships gear should be able to handle the pontoon covers and lay them down ashore far away from vessel.
- Ships gear should be fast and flexible for small quantities.
- Too many cranes to be avoided.
- Turnable spreader gear in conjunction with Jib crane may be helpful - although asking for skilful crane drivers.
- Double-joint slewing crane with turntable and spreader more suitable (provided hung by spread multiple point).
- In case of fixed cranes these DJS1 cranes should have their foundations preferably in the ship's sides, i.c.w.:-
  - minimum of hindrance for shore crane.
  - acceptable view from wheelhouse.

The "flexible" containership as far as gear is concerned.
If the Owner foresees that switching from one service - having shore cranes - to another service - lacking shore cranes - in a very short period would probably be required, it is worthwhile to consider a gantry crane.Such type of crane can be taken off/put on the ship at short notice and without structural conversion if the tracks, foundations and suitable hatchcovers are included in the newbuilding design.
The "double-joint slewing arm" type of crane mounted on the gantry bridge seems a suitable one. In 10 years time we might take ashore the gantry crane, if not needed anymore.



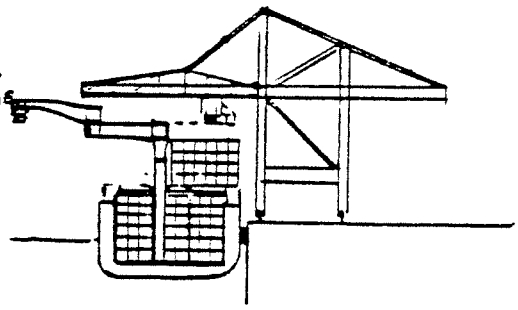
SHIPBORNE CONTAINER-CRANES.

3 x JIBCRANE AT CENTRELINE



- HINDRANCE FOR SHORE GANTRY
- HANDLING CONTAINERS BY JIB IS VERY SLOW
  AUTOMATIC ROTATION/EQUALIZING SPREADER
  IS NOT VERY EFFICIENT : BAD "STEERING" OF BOX
- LONG JIB IN VERTICAL POSITION CAUSES LONG PENDULUM
  MUCH DAMAGE TO CONTAINERS
- TEMPORARY SOLUTION ONLY.

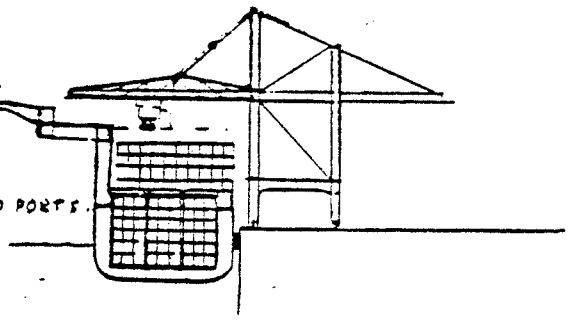
2 x SLEWING "POCKETKNIFE" AT CENTRELINE



- SOME HINDRANCE FOR SHORE GANTRY
- HOWEVER THE CAT CAN WORK OVER SHIPSCRANE
- CONTAINER IS "STEERED" BY SPREADER
  ALONG COORDINATES TRANSVERSELY
  AND LONGITUDINALLY
- HATCH COVERS TO BE CORRECTED IN CASE
  CRANES ARE REMOVED

LOSS OF CELLS / LOSS OF CONTAINERS

2 x SLEWING "POCKETKNIFE" AT SHIPS SIDE

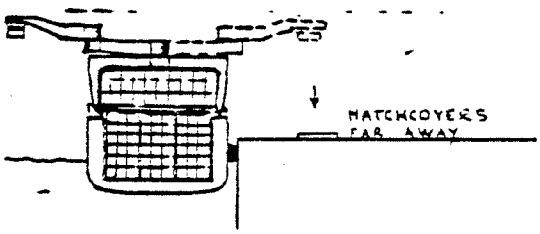


- NO HINDRANCE FOR SHORE GANTRY
- VERY SMALL LOSS OF CONTAINERS
- NO DEFORMED HATCHCOVERS
- SIMPLE CRANE FOUNDATION
- EXCELLENT VIEW FROM WHEELHOUSE EXCEPT TO PORTS.
- HATCHCOVERS CAN BE DEPOSITED ASHORE
  FAR AWAY FROM VESSEL.

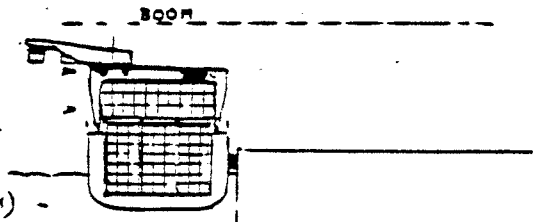
GENERALLY POCKETKNIFE CRANES CAN SPOT EASILY BY CONTAINERSTEERING ALONG COORDINATES

- HOWEVER LARGE HEELING MOMENTS ARE CAUSED BY HEAVY ARMS
- TWO SLEWING POCKETKNIFES WILL INFLUENCE EACHOTHER DURING SPOTTING!
  IN CASE WORKING SIMULTANEOUSLY

1 SHIPSGANTRY

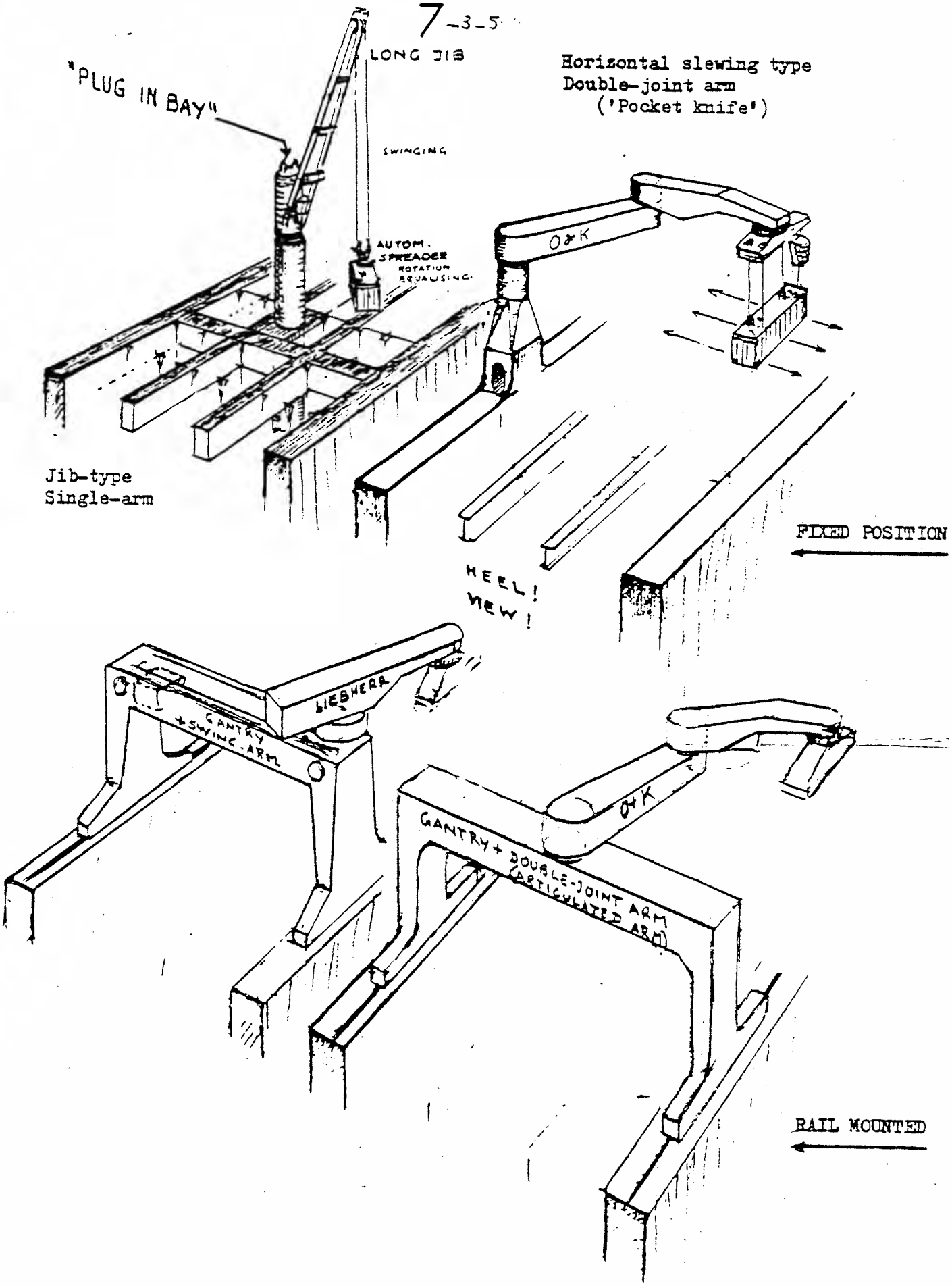


- ONE GANTRY COULD BE VERY FLEXIBLE
  FOR A SERVICE WITH MANY PORTS
  AND SMALL CONTAINER QUANTITIES.
- THE VESSEL CAN BE CONVERTED VERY EASILY TO ANOTHER SERVICE
  CRANE CAN BY TAKEN ON / OFF
  - NO CONVERSION OF HATCHCOVERS
  - NO CUTTING OFF/ CRANE FOUNDATIONS
  - JUMBOISING IS POSSIBLE



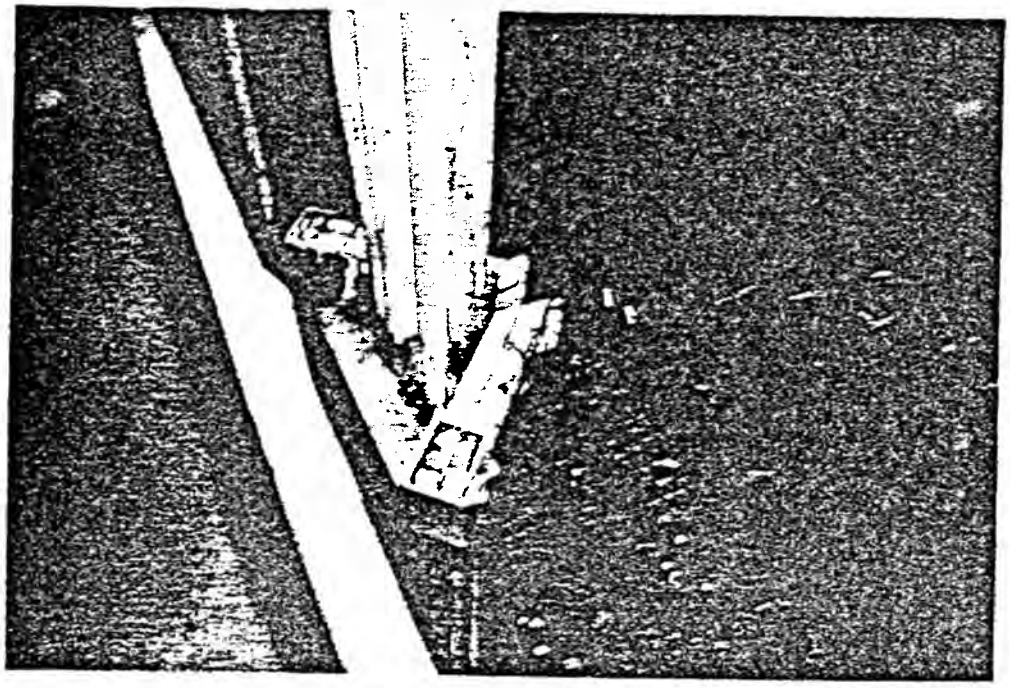
- LOSS OF CONTAINERS AT FOURTH TIER ON EACH SIDE
- COSTLY CRANE TRACK ON DECK (GEARTEETH)
- RELIABILITY DEPENDS MAINLY ON : LIMIT SWITCHES IN SEAWATER AND VIBRATIONS
  RELAIS " "
  HYDRAULIC /ELECTRIC LAYOUT
- GANTRY IS THE MOST EXPENSIVE GEAR
  IT SHOULD BE SIMPLE WITH ENCLOSED MACHINERY PREFERABLY ELECTRICALLY

SOME CONSIDERATIONS (MERITS AND DEMERITS) OF VARIOUS CRANE TYPES.



TYPES OF SHIPBORNE CRANES CONSIDERED FOR CONTAINER HANDLING.

Forklift truck working copper ingots in cellular hold.



Latch -type chocks at 2nd tier level of cellguides leaving space below containerstack for copper ingots.

Breakbulk? RoRo?

- Breakbulk handling in cellular ship is to be avoided.
- Breakbulk handling at container terminal is to be avoided.

Transportation of ingots (Copper, tin, zinc, aluminium, etc.), drums and paper reels could be done on flatracks or in bins if to be carried by cellular ships. However, it is difficult - if not impossible - for a cellular flatrack/bin system to compete effectively with the breakbulk system, because:
Who will pay for the flatracks?
Who will pay for the extra money on/off the flatracks or bin-boxes?

Example:

- Copper ingots in Chili, destined for export, inland are carried by old-fashioned railway cars.
- Transfer in port is carried out by 12-ton cranes, which move the bundles directly from waggon into hold (or tweendeck) of the multipurpose vessel.
- Upon arrival in Europe the copper is very often discharged into big lighters or onto the copper terminal by Jib cranes.

As to carriage, it would seem that Ro-Ro vessels are more suitable for this type of cargo.
The limitation, however , of the use of Ro-Ro ships lies in the swell condition in Chilean loading ports which makes the vessel ride and causes damage by interaction of ramp and wharf.

Nedlloyd's opinion is that we - after all - end up with the fastest turnaround system!

- Cellular vessel : lift on/off of containers, bins, flatracks.
- RoRo Container : lift on/off of containers on upperdeck, by means of suitable shorecrane. Roll on/off of wheeled cargo in holds.

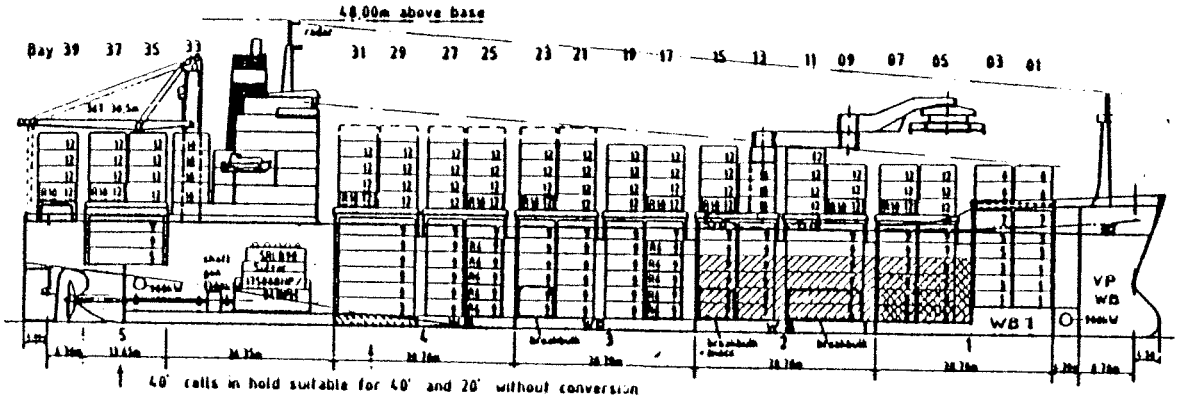
Shifting of ships from one berth to another costs time and is old fashioned now. So, either a container vessel goes cellular only - and is handled at such a wharf or she goes part-roro and for cargo handling goes to a "mixed" wharf.

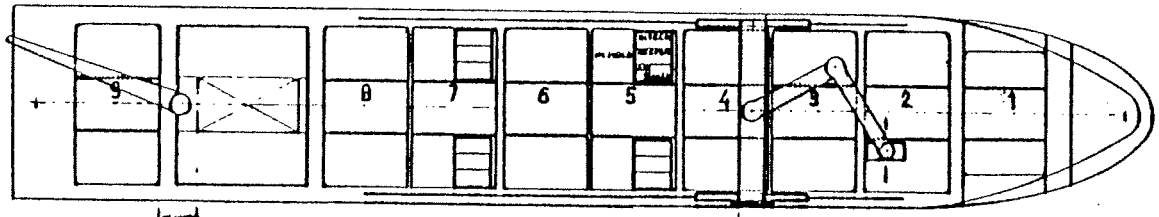
Conclusion all in one:

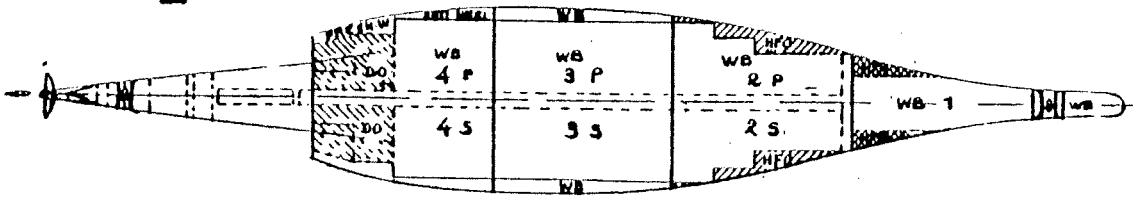
- Terminals should cope with the cargo systems of both ship types, cellular and RoRo, bearing in mind also the feeder aspect.

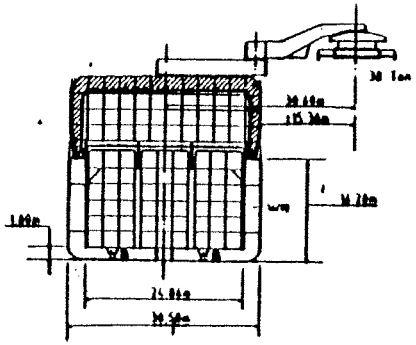
NEDLLOYD VAN NECK

(Van Der Giessen-De Noord yardnr. 929)









TYPICAL LOADING CONDITION DESIGN

Draft design mld	10 - M
Displacement	36852T
Light ship	11030
Deadweight 10M	23822
FO	1200
DO+Lub+Fr w+stores	512
Waterballast	1460
Cargo containers:	20650=14Tx1475 TEU
" in hold	10976=14Tx 784
3 tiers on deck	8638=14Tx 617
4th tier on deck	1036=14Tx 74
Homogeneous contr box=	0.45x0.5FT
KM	14.37 M
KG	13.90
GM	0.47
free surface corr.	0.06
G'M'	0.41 M
BRT	23930T
NRT	10756T

MAIN ENGINE	SULZER 5RLB90
MCR/RPM	17500 BHP/94
85%	14875 BHP/94 x24x142
	= 151 Ton/day
Fuel n.e. :	HFO 600 cSt - mt
AUXIL:	50% HFO + 50% MDO

Heavy Fuel Oil consumption excl shaft generator in tons/day

SHP service = $SHP_{tank} \times C_1 \times C_2$

C_1 = margin for trials = 1.06

C_2 = sea margin = 1.12 (service allowance)

Spec fuel consumption = 143 gr/SHP h

Draft	15.5	16	16.5	17	17.5	18	18.5	19kn
7M	26.5	29.0	31.4	34.1	36.9	39.9	43.1	46.5
8M	25.5	28.1	30.7	33.4	36.6	39.6	43.1	46.7
9M	26.7	29.4	32.2	35.1	38.2	41.5	45.3	49.1
10M	30.1	33.5	37.1	40.9	45.1	49.7	55-	Ton/day

Shaft generator max 1300 kW

Normal 540 kW x $\frac{1.36}{0.88}$ = 860 HP

x 143 gr/SHP hr x 24 = 2.9 T/day

Diesel generators

Diesel gen I	1500kW-720RPM
Diesel gen II	1500kW-720RPM
Diesel gen III	700kW-720RPM
Emergency gen :	160kW-1800RPM

Normal electr load at sea

600kW -MDO diesel gen 2.9T/day

-HFO shaft gen 2.9T/day

Idle in port 450kW MDO 2.3T/day

Boiler HFO 1.0T/day

Extra for reefers (100 integr 20')

box x kW/box x HP/kW x

gr/HP hr x hr =

100 x 5.5 x 1.5 x 143 x 24 = 2.6 T/day

Length over all	182.50M+0.25
Length betw perpend	174.00M
Breadth mld	30.50M
Depth mld	16.20M
Draft scantling-exm.	11.27M
Displacement	40760 TON
Light ship	11030TON
Deadweight	29730TON

Water ballast	7874 m3
Fresh water	411 m3
Heavy Fuel Oil	2883m3
Diesel oil	366 m3
Lubricating oil	281m3

MAX TEU CAPACITY SPACE WISE		
Hold	776	of which 117 - 40'=30%
Deck	212	1st tier
	228	2nd tier
	228	3rd tier
	1444	
	166	4th tier
	1610	

Space on tanktop 2 tiers high
In guides chocks
Suitable for BREAKBULK
Load on tanktop 12 ton/M2

Hatchrows 3 + 3 + 3 (cover weight 18T and 13T)
Number of hatches: 8 fwd of deckh
1 aft of deckhouse
40' cells in hatch no 3 - 8 - 9
40' box below deck 117 = 46+48+23
In no 8 and 9 also 20' boxes can be stowed on cones
Max stackload on covers
20' stack 60 Ton
40' stack 90 Ton
Max stackload if twistlocks only 40 T
If stackload exceeds 40T lash by rods + turnbuckles

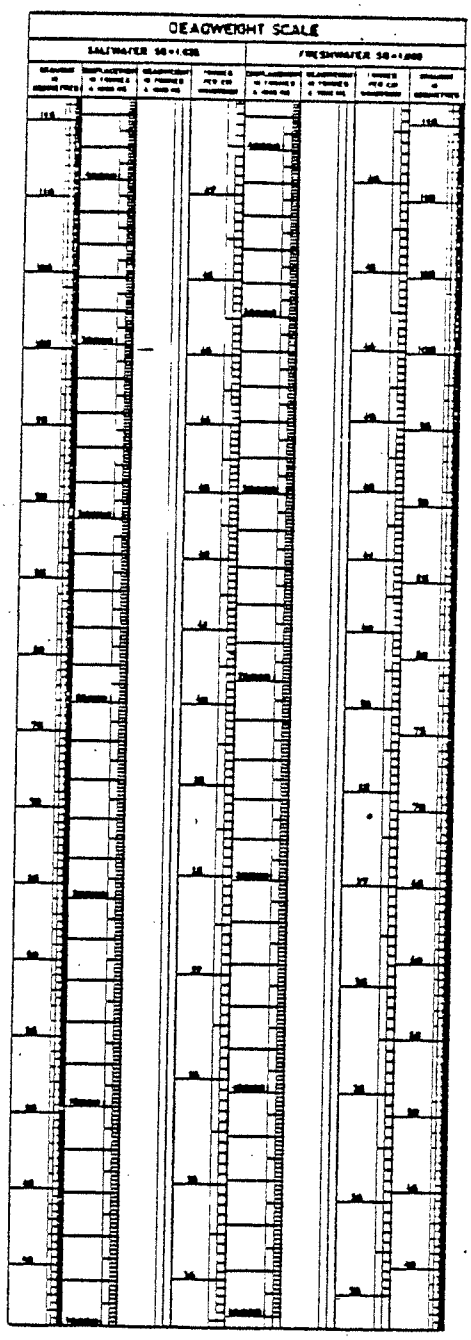
THRUSTERS (var pitch)
Bowthruster 900 kW, 1220 HP
Sternthruster 900 kW, 1220 HP
Rudder max angle 45°

INTEGRAL REEFERS	
Reeferpoints on deck 20'/40'	90
5 kW 20'/7.5kW 40'	
Reeferpoints in hold 20'	60
ventilation 1 m3/sec per box	NO WATER AIR ONLY
Receptacles 440V, CEE 17, 3h, 32 Amp	
Hold ventilation H 1, 2 & 5	2.5
H 3 & 4	9

NEDLLOYD van NECK 929
Delivery date: 2 December 1983
Yard v/d Giessen de Noord
Class Bureau Veritas

8-1-1

COMM: 61' 2-12-83



GENERAL PARTICULARS:

LENGTH OVERALL	183.731 m
LENGTH BETWEEN PERPENDICULARS	174.000 m
BREADTH MOULDED	30.500 m
DEPTH MOULDED	16.200 m
DRAUGHT ON SUMMER FREEBOARD S.W.	11.274 m

TONNAGE			
	INTERNATIONAL	SUEZ	PANAMA
GROSS	13950		
NET	10756		

MAIN PROPULSION MACHINERY:

1 SCHELDE/MAN/B&W 5 K 45 GF
MAXIMUM CONTINUOUS RATING [illegible] AT 56 R.P.M.

AUXILIARY MACHINERY:

1 MAN ENGINE TYPE 5 SW 180, 1050 KW EACH AT 720 R.P.M. COUPLED WITH 2 HOLEC ALTERNATORS TYPE DP 173/36/4B, 1500 KW EACH.
1 MAN ENGINE TYPE 4 SW 180, 775 KW AT 720 R.P.M. COUPLED WITH 1 HOLEC ALTERNATOR TYPE DS 1104-4A 16, 750 KW.
1 MAIN ENGINE DRIVEN 440 SHAFT ALTERNATOR TYPE DP 173/4/4 HOLEC, [illegible]
2 STARTING AIR COMPRESSORS 150 m³/h, 30 BAR.
1 TOPPING UP AIR COMPRESSOR 150 m³/h AT 30 BAR.
1 GENERAL SERVICE AIR COMPRESSOR 150 m³/h AT 7 BAR.

PORT OF REGISTRY: ROTTERDAM
OFFICIAL NUMBER:
SIGNAL LETTERS: P G E B
CLASS: BUREAU VERITAS ✠ 3/3 E ✠ [illegible]
OWNERS: NEDLLOYD LINIEN B.V. ROTTERDAM

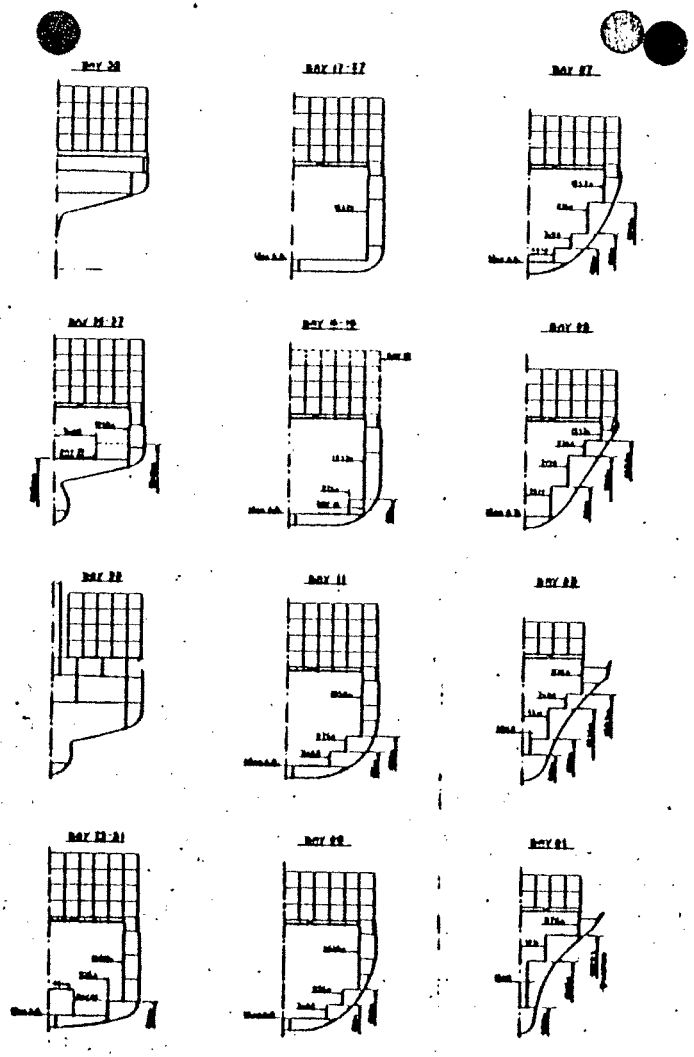
CAPACITY PLAN 1:400

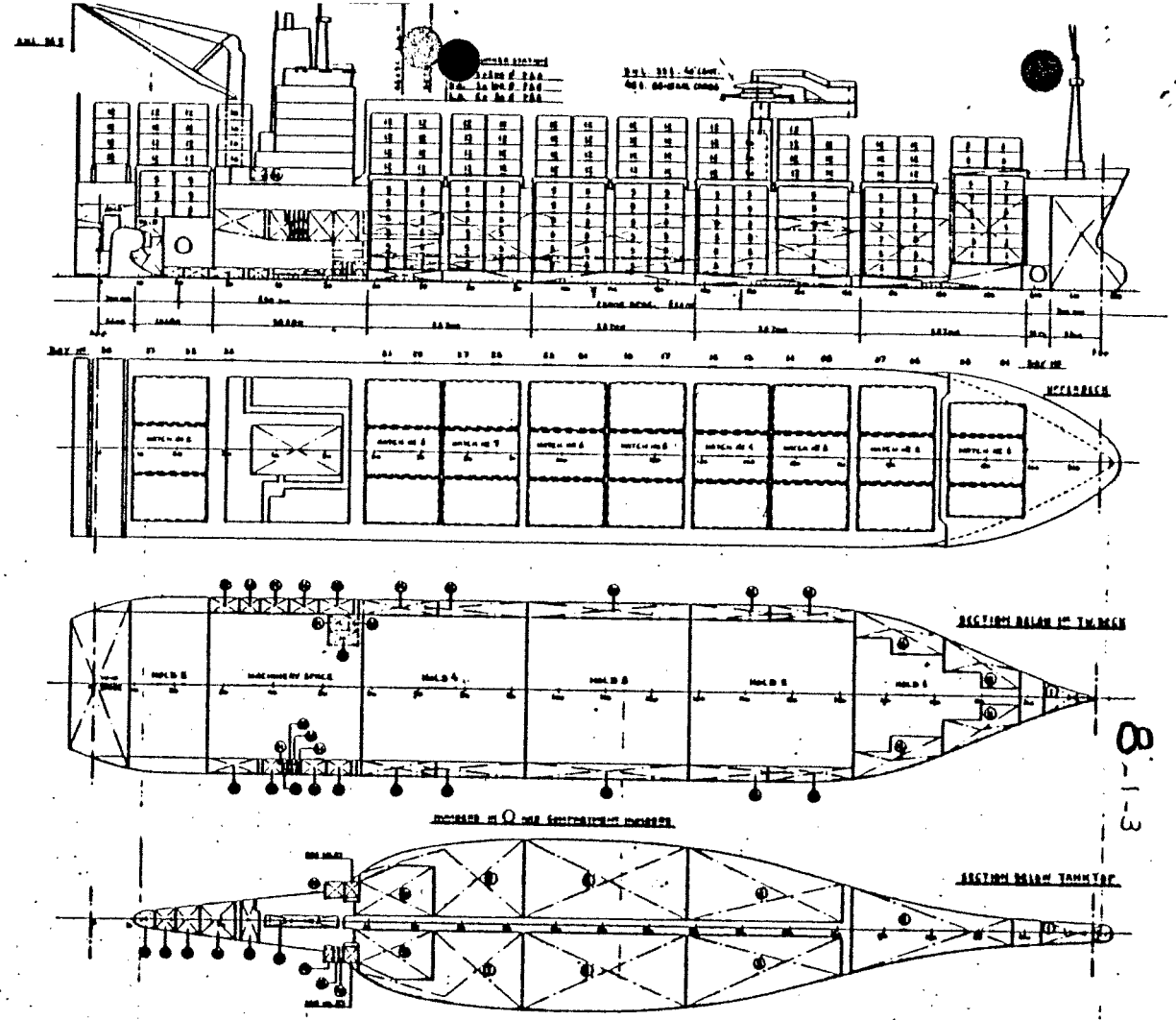
M.S. "NEDLLOYD VAN HECK"
YARD NR. 329

van der Giessen-de Noord
shipbuilding division B.V.
Krimpen aan den IJssel - Holland

FREEBOARD PARTICULARS					
LOAD LINE MARKS	DRAUGHT m	FREEBOARD m	DISPLACEMENT TONNES	DEADWEIGHT TONNES	
S.W.	11.469	4.769	40769		
TOPMAST	11.274	4.964	40769	[illegible]	

DATE: DEC. 1983 DRAW. NR. S.80.0929.0.015.17





CAPACITIES AND CENTRES OF TANKS

No.	Tank	Frames	Vol.	V.C.G.	L.C.G.	Moment
	HEAVY FUEL OIL TANKS: 96.5 LBS					
	[illegible]					
	TOTAL HEAVY FUEL OIL					
	DIESEL OIL TANKS: 96.5 LBS					
	[illegible]					
	TOTAL DIESEL OIL					
	LUBRICATING OIL TANKS: 96.5 LBS					
	[illegible]					
	TOTAL LUBRICATING OIL					

CAPACITIES AND CENTRES OF TANKS

No.	Tank	Frames	Vol.	V.C.G.	L.C.G.	Moment
	FRESHWATER TANKS: 96.5 LBS					
	[illegible]					
	TOTAL FRESHWATER					
	WATER BALLAST TANKS: 96.5 LBS					
	[illegible]					
	TOTAL WATER BALLAST					
	MISCELLANEOUS TANKS					
	[illegible]					

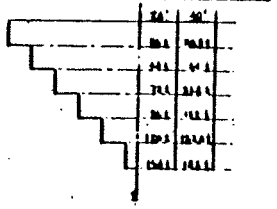
PROVISION STORES

COMPARTMENT	FRAMES	VOLUME	POSITION
LARDER	[illegible]	[illegible]	[illegible] DECK
POTATOES	[illegible]	[illegible]	[illegible]
VEGETABLES	[illegible]	[illegible]	[illegible]
MEAT	[illegible]	[illegible]	[illegible]
FISH	[illegible]	[illegible]	[illegible]
TOTAL		[illegible]	
DRY PROVISION	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]
TOTAL		[illegible]	

WEIGHT & CENTRES LIGHT SHIP

	WEIGHT	V.C.G.	L.C.G.
[illegible]			

MAX. STACKLOADS CONTAINERS IN HOLDS



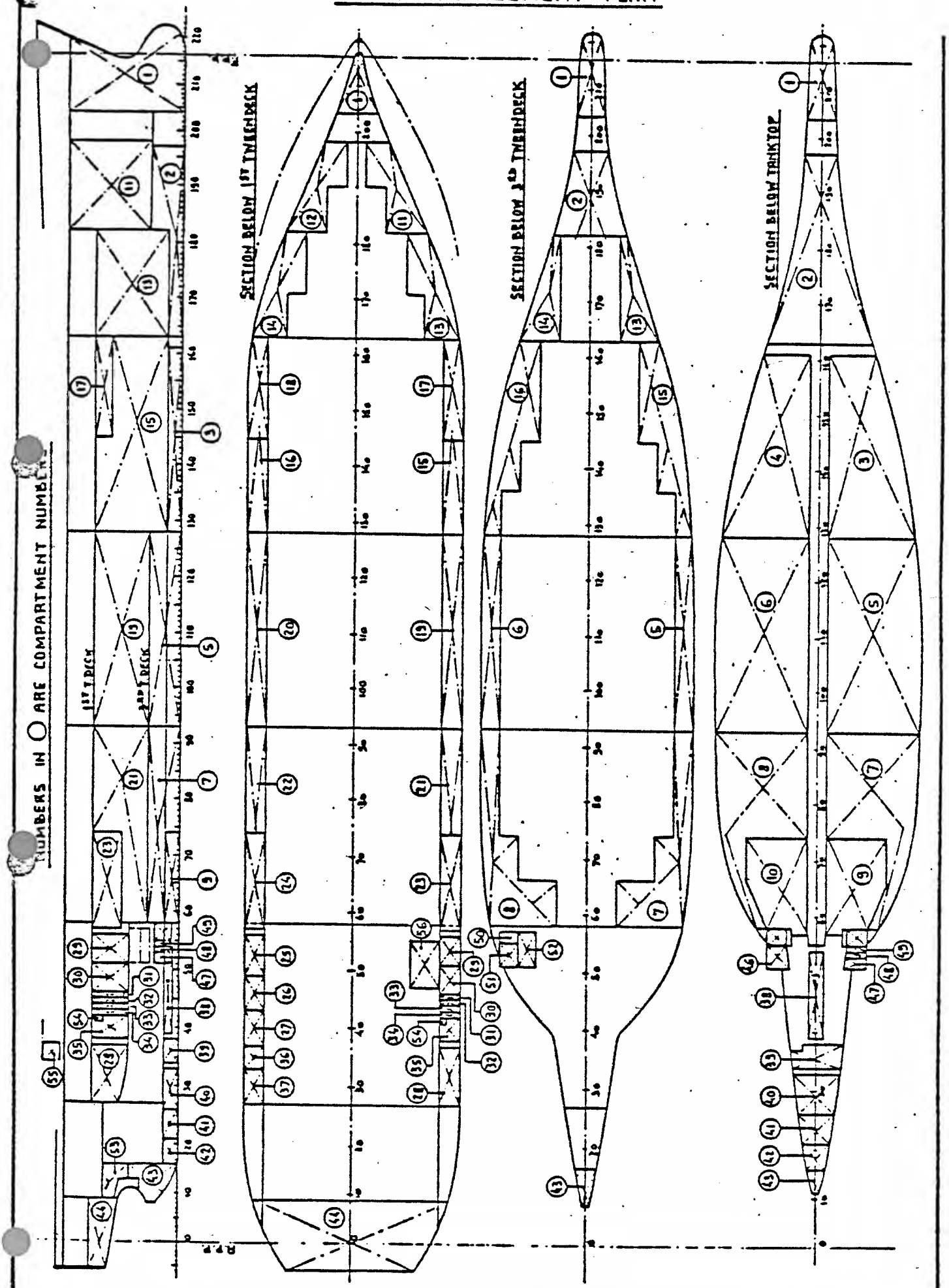
MAX. STACKLOADS CONTAINERS ON DECK

20' CONTAINERS = [illegible]

40' CONTAINERS = [illegible]

(MAX. STACK. WEIGHT)

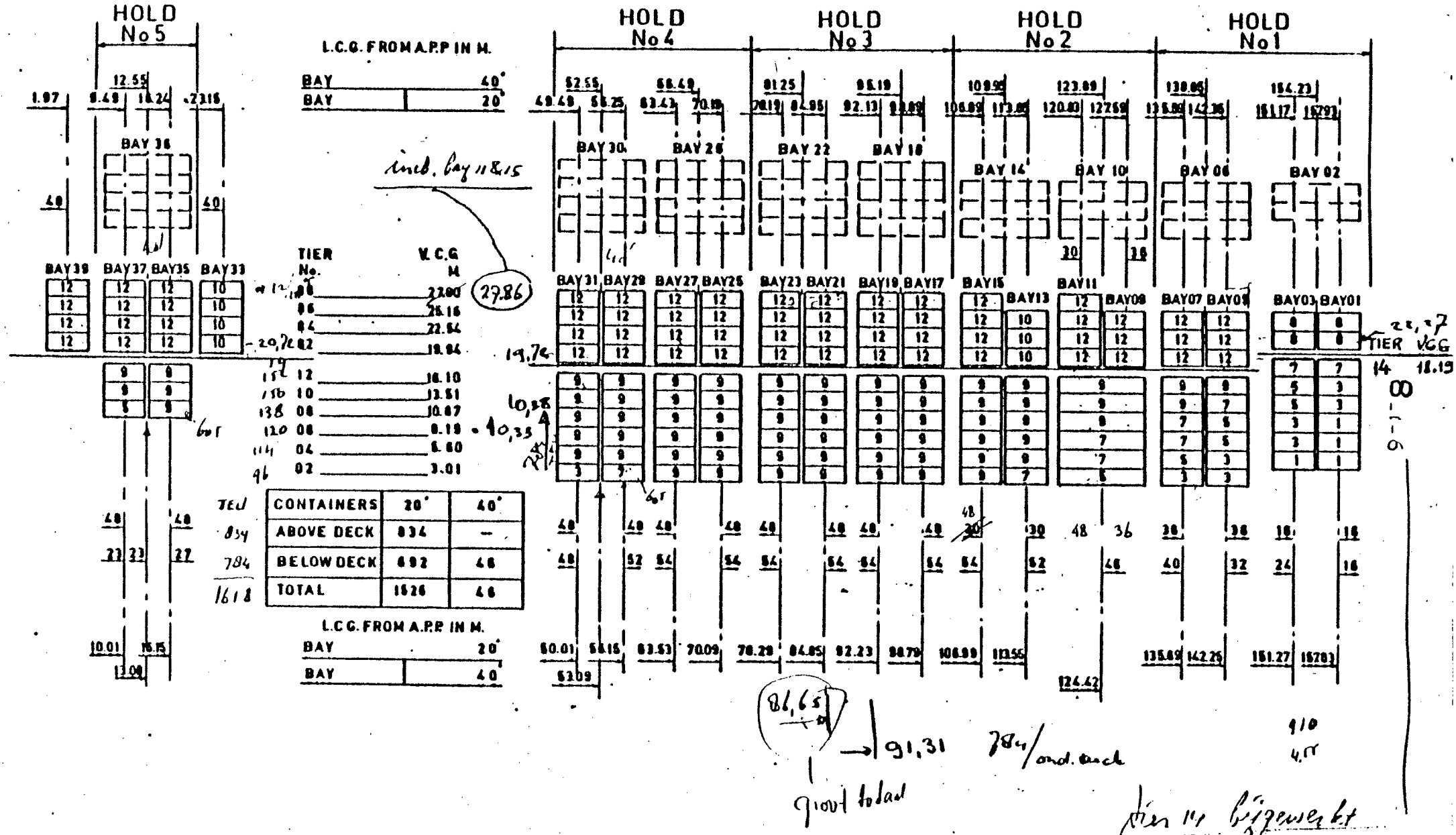
TANK ARRANGEMENT PLAN



INDEX

COMPARTMENT	COMP. Nº	FRAME	CAPACITY 100% FULL m³	CENTRE OF GRAVITY V.C.G. AB. BASE m	L.C.G. FROM AP m	FROM C.L. m	MAX. FREE SURF. MOMENT I IN m⁴	PAGE Nº
W.B. FORE PEAK	1	204 - FWD	545.02	8.67	169.69	0.00	540.3	4
W.B. D.B. TANK 1	2	163 - 197	532.66	1.87	147.32	0.00	2195.6	5
W.B. D.B. TANK 2 S.B.	3	128 - 161	402.54	0.97	115.43	5.86	2551.2	6
W.B. D.B. TANK 2 P.S.	4	128 - 161	402.54	0.97	115.43	-5.86	2551.2	7
W.B. D.B. TANK 3 S.B.	5	93 - 128	804.82	1.44	89.17	8.93	5327.0	8
W.B. D.B. TANK 3 P.S.	6	93 - 128	804.82	1.44	89.17	-8.93	5327.0	9
W.B. D.B. TANK 4 S.B.	7	58 - 93	650.80	2.07	61.87	9.13	2894.0	10
W.B. D.B. TANK 4 P.S.	8	58 - 93	650.80	2.07	61.87	-9.13	2894.0	11
D.O. D.B. TANK 4 S.B.	9	57 - 74	141.69	1.06	53.54	4.79	560.1	12
D.O. D.B. TANK 4 P.S.	10	57 - 74	141.69	1.06	53.54	-4.79	560.1	13
W.B. WINGT. 1 S.B. FORW.	11	182 - 198	465.85	10.82	154.19	5.33	322.9	14
W.B. WINGT. 1 P.S. FORW.	12	182 - 198	465.85	10.82	154.19	-5.33	322.9	15
H.F.O. WINGT. 1 S.B. AFT	13	163 - 182	486.56	7.36	139.47	8.26	225.0	16
H.F.O. WINGT. 1 P.S. AFT	14	163 - 182	486.56	7.36	139.47	-8.26	225.0	17
H.F.O. WINGT. 2 S.B.	15	128 - 163	767.65	6.37	117.70	12.16	502.4	18
H.F.O. WINGT. 2 P.S.	16	128 - 163	767.65	6.37	117.70	-12.16	502.4	19
H.F.O. OVERFLOW TANK S.B.	17	145 - 163	92.83	10.91	124.53	13.73	22.5	20
H.F.O. OVERFLOW TANK P.S.	18	145 - 163	92.83	10.91	124.53	-13.73	22.5	21
W.B. WINGT. 3 S.B.	19	93 - 128	612.27	2.34	89.34	13.86	51.4	22
W.B. WINGT. 3 P.S.	20	93 - 128	612.27	2.34	89.34	-13.86	51.4	23
W.B. WINGT. 4 S.B.	21	58 - 93	461.96	7.73	63.22	13.85	51.4	24
W.B. WINGT. 4 P.S.	22	58 - 93	461.96	7.73	63.22	-13.85	51.4	25
DRINKING W. TANK S.B.	23	57 - 74	155.75	10.14	52.43	13.86	25.0	26
DRINKING W. TANK P.S.	24	57 - 74	155.75	10.14	52.43	-13.86	25.0	27
H.F.O. SETTLING TANK P.S.	25	49 - 56	72.57	9.23	41.63	-13.86	10.6	28
H.F.O. SERVICE TANK P.S.	26	43 - 49	60.48	9.27	36.12	-13.82	9.1	29
H.F.O. SETTLING TANK P.S.	27	37 - 43	55.91	9.39	31.05	-13.74	9.1	30
FEEDWATER TANK S.B.	28	27 - 37	83.19	10.27	24.53	13.70	15.2	31
L.O. STOR. T. M.ENG. S.B.	29	51 - 56	60.37	9.58	42.48	13.86	7.6	32
L.O. STOR. T. M.ENG. S.B.	30	46 - 51	59.73	9.60	38.23	13.84	7.6	33
DIRTY L.O. STOR. T. AUX. ENG. S.B.	31	45 - 46	11.72	9.64	35.68	13.82	1.5	34
CLEAN L.O. STOR. T. AUX. ENG. S.B.	32	44 - 45	11.62	9.65	34.83	13.81	1.5	35
L.O. STORAGE T. AUX. ENG. S.B.	33	43 - 44	11.50	9.67	33.98	13.80	1.5	36
L.O. STORAGE T. AUX. ENG. S.B.	34	42 - 43	11.37	9.70	33.13	13.79	1.5	37
CYL. OIL STORAGE TANK S.B.	35	38 - 42	42.72	9.70	30.97	13.79	6.0	38
D.O. SERVICE TANK P.S.	36	33 - 37	37.90	10.05	26.80	-13.71	6.1	39
D.O. SETTLING TANK P.S.	37	27 - 33	45.27	10.44	22.63	-13.68	9.1	40
L.O. SUMP TANK	38	39 - 54	26.60	1.35	36.48	0.00	9.4	41
L.O. EXCHANGE TANK	39	34 - 38	31.43	1.06	27.57	0.13	85.9	42
H.F.O.-D.O. TR. OVERFLOW & DRAINT.	40	26 - 33	46.72	1.08	22.20	0.00	96.5	43
DIRTY WATER TANK	41	21 - 26	25.96	1.10	17.06	0.00	39.7	44
L.O. SUMPT. STERNTUBE	42	16 - 21	12.58	1.08	13.48	0.00	10.3	45
FRESH WATER T. STERNTUBE	43	9 - 16	37.38	3.52	9.68	0.00	6.4	46
VOID SPACE A.P.	44	AFT - 9	49.24	11.34	2.07	0.00	15320.6	47
PISTON COOLW. TANK P.S.	46	51 - 55	16.22	2.45	42.08	-5.36	8.2	48
HOTWATER PREP. TANK S.B.	47	51 - 53	7.90	2.48	41.21	5.26	4.5	49
RESIDUE TANK S.B.	48	53 - 54	4.10	2.44	42.48	5.41	1.9	50
AIR COOLER CLEANING W.T. S.B.	49	54 - 55	4.20	2.41	43.33	5.49	1.8	51
SLUDGE MIXING TANK P.S.	50	55 - 57	8.21	6.05	44.60	-11.02	2.7	52
L.O. SLUDGE TANK P.S.	51	51 - 55	16.43	6.05	42.05	-11.02	5.5	53
H.F.O.-D.O. SLUDGE TANK P.S.	52	51 - 57	34.44	5.68	42.90	-8.36	8.0	54
VOID SPACE	53	9 - 16	143.01	9.48	8.90	0.00	1166.7	55
CYL. OIL TANK S.B.	54	41 - 42	1.42	11.25	32.27	12.93	0.1	56
D.O. TANK S.B.	55	31ᴮ - 35ᶜ	3.97	18.15	27.43	8.04	0.0	57
SWIMMING POOL S.B.	56	47 - 55	59.72	11.09	40.35	10.40	39.5	58

M.S. "NEDLLOYD VAN NECK" CONTAINER STOWAGE PLAN



	CONTAINERS	20'	40'
834	ABOVE DECK	834	—
784	BELOW DECK	692	46
1618	TOTAL	1526	46

TEU

784/ond. dek

910

4.55

9-1-6

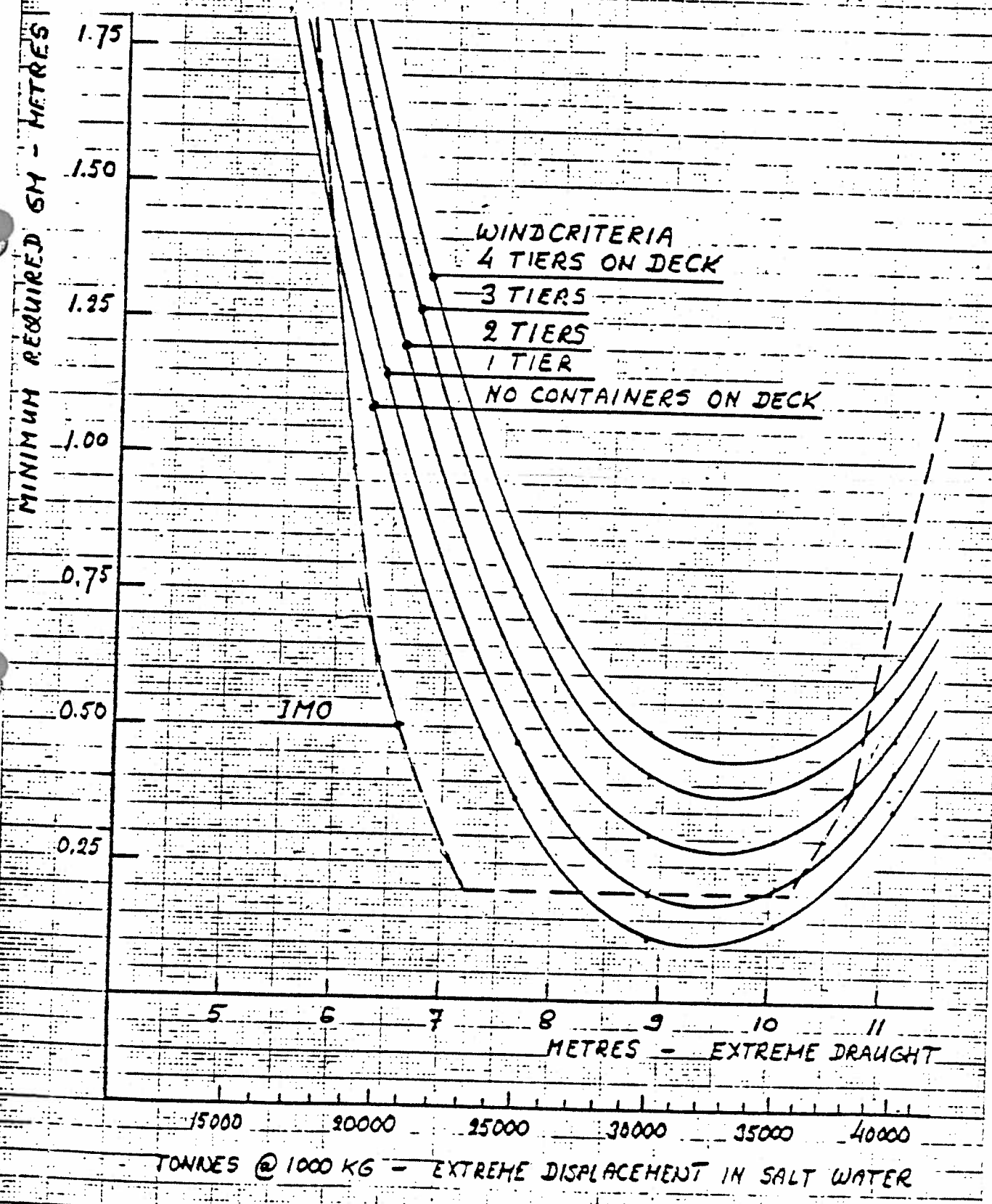
Gross total

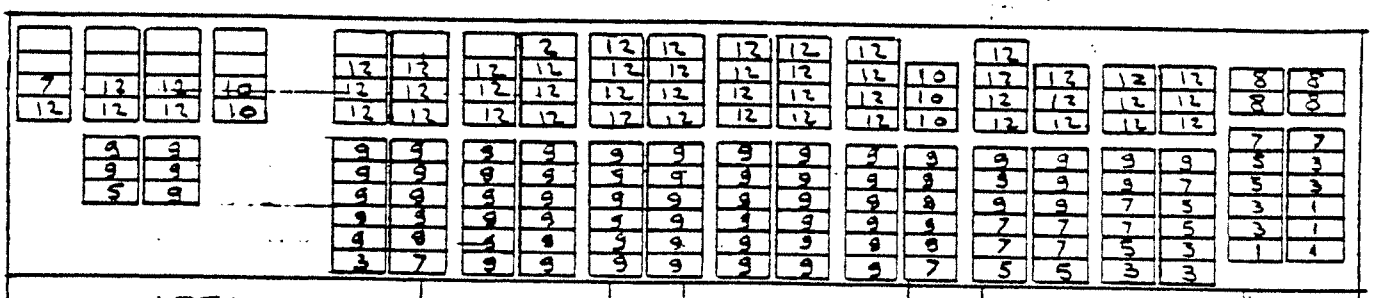
Dien 1/1 bijgewerkt

31-8-83

VAN DER GIESSEN - DE NOORD YARD NAS 929/930/931

MINIMUM REQUIRED GM-VALUES





ITEM		WEIGHT	VCG	VERT MOM	LCG	LONG MOM	FREE S.M.
LIGHT SHIP		11030	11.71		70,64		
STORES		200	13.20		47.63		26
SHIP READY FOR SEA		11230	11.74	131 801	70.23	788 685	26
FO WINGT. 1 P+S	100	924	7.36		139.47		
FO WINGT. 2 P+S	10.4	151	2.60		117.70		955
FO SETTL. T. P	98.6	68	9.23		41.63		19
FO SERV. T. P	99.2	57	9.27		36.12		
FO TOTAL		1200	6.96	8 349	126.28	151 533	974
DO SERV. T.	65.2	21	10.05		26.80		13
DO SETTL. T	98.8	24	10.44		22.63		
DO DB. T. SB	45.7	55	0.62		53.54		430
DO TOTAL		100	4.96	496	40.51	4 051	443
LO STORAGE T. MAIN E.	99.4	54	9.58		42.48		9
LO CYL. OIL STORAGE T.	98.8	38	9.70		30.97		
LO STORAGE T. AUX. E	97.2	20	9.69		33 55		
LO TOTAL		112	9.64	1 080	36.98	4 142	9
FEED WATER	24.4	20	10.05		24.53		15
DRINKING WATER	25.7	80	10.14		52.43		25
FRESH W. TOTAL		100	10.12	1 012	46.85	4 685	40
WB FORE PEAK	100	559	8.67		169.69		
WB DB T 2 P+S	100	825	0.97		115.43		662
WB WINGT 1 P	15.9	76	5.60		154.19		
WB TOTAL		1460	4.16	6 072	138.22	201 805	662
BELOW DECK	784	10976	10.38		91.31		
3 TIERS ON DECK	617	8638	22.35		82.37		
4TH TIER	74	1036	27.52		96.25		
CONT. TOTAL	1475	20650	16.25	335500	87.82	1 813 448	
		34852	13.90	484 310	85.17	2 968 349	2154

F.S. 0.06

kG' 13.96

DRAUGHT MEAN	10,00 M	KM	14.37 M	LCB	85.17 M	LCF	78.32 M
" FORW	10.00 "	KG'	13.96 "	ARM	- "	TRIM AFT	"
" AFT	10.00 "	GM	0.41 "	TRIMFACT.	47428	" FORW	"
TPC	45.03			TRIM	"		

LOADING CONDITION

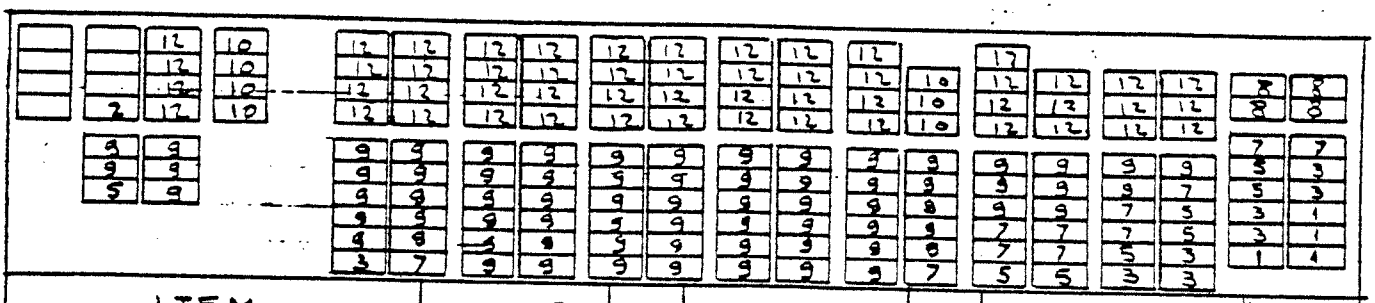
ARRIVAL DAKAR. 1475 TEU @ 14T

DRAUGHT 10 M EVEN KEEL

NEDLLOYD - NEWB. DEP.

NL VAN NECK

DATE DEC / 83 SH. 1:2



ITEM	WEIGHT	VCG	VERT MOM	LCG	LONG MOM	FREE S.M.
SHIP READY FOR SEA	11 230	11.74	131 801	70.23	788 685	26
F.O TOTAL	150	8.12	1 218	55.43	8 315	279
D O TOTAL	50	9.25	463	27.56	1 378	121
L O TOTAL	112	9.64	1 080	36.98	4 142	9
F W TOTAL	100	10.12	1 012	46.85	4 685	40
WB FORE PEAK 100%	559	8.67		169.69		
WB D.B.T. 1 100%	546	1.87		147.32		
WB D.B.T 2 100%	825	0.97		115.43		
WB WING T. 1 100%	955	10.82		154.19		
WB TOTAL	2885	5.89	17 000	144.81	417 775	
1524 TEU @ 12 T	18288	16.76	306 507	86.57	1 583 192	
	32 815	13.99	459 081	85.57	2 808 172	475

F.S 0.01

KG' 14.00

DRAUGHT MEAN	9.54 M	KM	14.41 M	LCB	85.57 M	LCF	M
" FORW	9.54 "	KG'	14.00 "	ARM	"	TRIM AFT	"
" AFT	9.54 "	G'M	0.41 "	TRIMFACT.		" FORW	"
				TRIM	"		

LOADING CONDITION

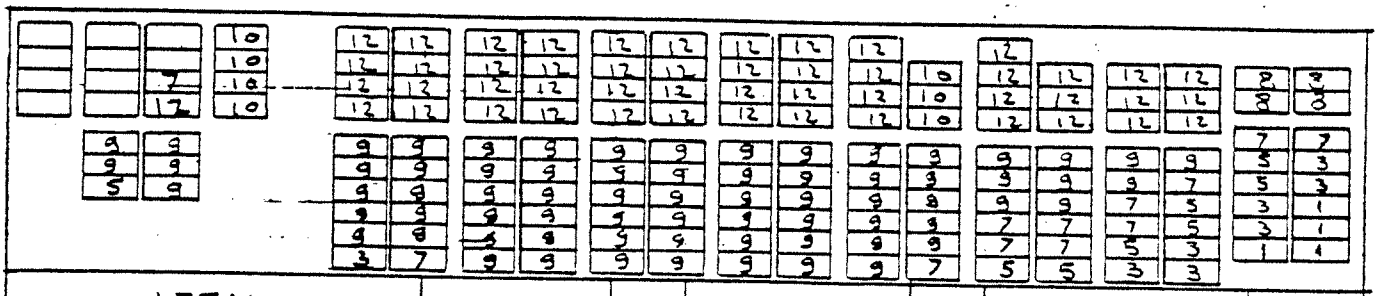
ARRIVAL 1524 TEU @ 12 T HOMOG.

DRAUGHT 9.54 M EVEN KEEL

NEDLLOYD - NEWB. DEP.

NL VAN NECK

DATE DEC/83 SH. No



ITEM	WEIGHT	VCG	VERT MOM	LCG	LONG MOM	FREE S.M.
SHIP READY FOR SEA	11 230	11.74	131 801	70.23	788 685	26
F.O TOTAL	150	8.12	1 218	55.43	8 315	279
DO TOTAL	50	9.25	463	27.56	1 378	121
LO TOTAL	112	9.64	1 080	36.98	4 142	9
FW TOTAL	100	10.12	1 012	46.85	4 685	40
WB Fore peak 100%	559	8.67		169.64		
WB D.B.T. 1 100%	546	1.87		147.32		
WB D.B.T. 2 100%	825	0.97		115.43		
WB WING.T. 1 100%	955	10.82		154.19		
WB TOTAL	2 885	5.89	17 000	144.81	417 775	
1493 TEU @ 10T	14 930	16.56	247 286	88.04	1 314 446	
	29 457	13.57	399 860	86.21	2 539 426	475

F.S. 0.01

kG' 13.58

DRAUGHT MEAN	8.77 M	KM	14.53 M	LCB	86.21 M	LCF	M
" FORW	8.77 "	KG'	13.58 "	ARM	- "	TRIM AFT	"
" AFT	8.77 "	GM	0.95 "	TRIMFACT.		" FORW	"
				TRIM	"		

LOADING CONDITION

ARRIVAL 1493 TEU @ 10T HOMOG.

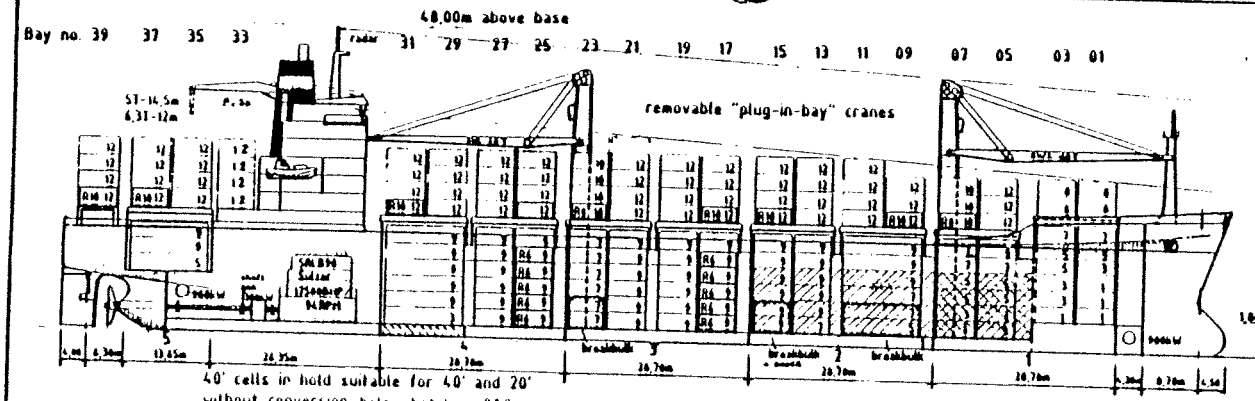
DRAUGHT 8.77 EVEN KEEL

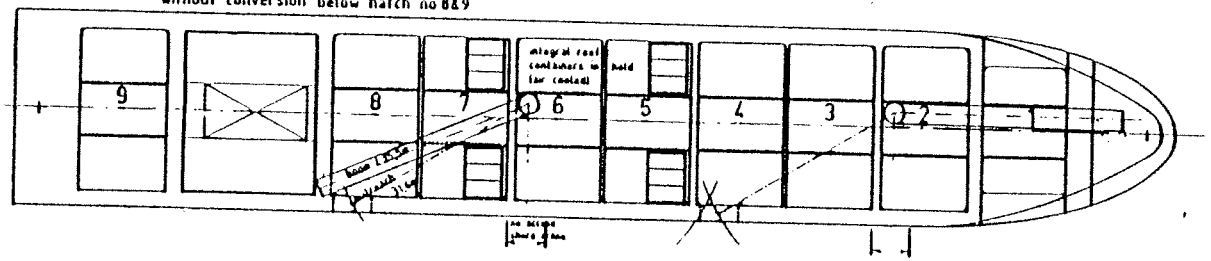
NEDLLOYD - NEWB. DEP.

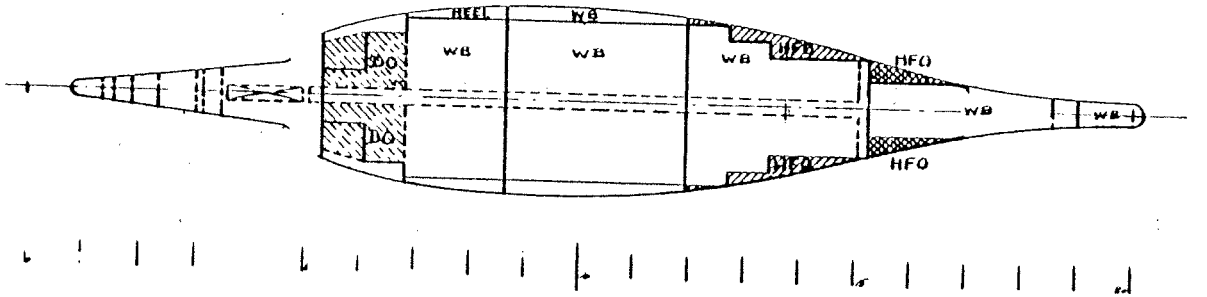
NL VAN NECK

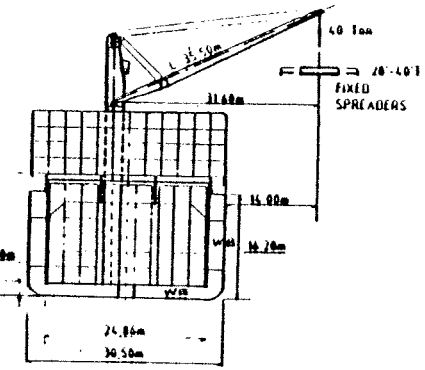
DATE DEC/83 SH. 112

NEDLLOYD VAN NOORT
" VAN DIEMEN









TYPICAL LOADING CONDITION-DESIGN

Draft-design mld	9.24 M ~10 M
Displacement	34 687 Ton
Light ship	11000
Deadweight	23687 Ton
FO+DO+Lub+Dr w +prov+stores	2349
Waterballast	1089
Cargo containers	20249 Ton
14.2Ton per TEU = 14.2x 1426 TEU homogeneous load	
KM	14.37 M
KG	13.85
GM	0.52
Free surface correction	0.06
GM'	0.46 M
FO	1429 Ton
DO	250
Lub oil	100
Fresh water	250
Sev tanks	95
Crew+prov+stores	225
	2349 Ton
Waterballast	1089
	3438 Ton
Cont below deck	1067 8T.142 x 752 TEU
Cont on deck	957 1T.142 x 674
HOMOG LOADED	
CENTER GRAVITY BOX - 43'/8'	20249T.142 x 1426

Length over all	182.50M+0.23
Length betw perpend	174.00M
Breadth mld	30.50M
Depth mld	16.20M
Draft scantling	11.27M
Displacement	40760TON
Light ship estim.	11000 -
Deadweight 11M	29760TON

Water ballast	7874 m3
Fresh water	411 m3
Heavy Fuel Oil	2883 m3
Diesel oil	366 m3
Lubricating oil	281 m3

MAIN ENGINE SULZER 5RLB90							
MCR	17500 HP at 94 RPM						
service rating 14875 HP at 94 RPM							
(85%) = : 51 ton/day							
Fuel for main eng HFO max 600 cSt at 50°							
Fuel for aux eng HFO = 50%HFO/50%MDO							
Heavy Fuel Oil consumption excl shaft generator in tons/day							
SHP service = SHPtank x C_1 x C_2							
C_1 = margin for trials = 1.06							
C_2 = sea margin = 1.12 (service allowance)							
Spec fuel consumption = 143 gr/SHP h							
Draft	15.5	16	16.5	17	17.5	18	18.5	19kn
7M	26.5	29.0	31.4	34.1	36.9	39.9	43.1	46.5
8M	25.5	28.1	30.7	33.4	36.4	39.6	43.1	46.7
9M	26.7	29.4	32.2	35.1	38.2	41.5	45.3	49.1
10M	30.1	33.4	37.1	40.9	45.1	49.7	55.0	tons/day

Shaft generator max 1300kW	
Normal 540kW x 1.16/0.75 = 860 HP	
x 143 gr/SHP hr x 24 = 29 T/day	
Diesel generators	
Diesel gen I	1500kW-720RPM
Diesel gen II	1500kW-720RPM
Diesel gen III	700kW-720RPM
Emergency gen	160kW-1800RPM
Normal electr load at sea	540kW
-IFO diesel gen	2.9T/day
-HFO shaft gen	2.9T/day
Idle in port 400kW IFO	2.3T/day
Boiler HFO	1.0T/day
Extra for reefers (100 integr 20') box x kW/box x HP/kW x gr/HP hr x hr =	
100 x 5.0 x 1.5 x 0.143x24 =	2.6T/day

Preliminary!

BRT 23930

NRT 10756

MAX. TEU CAPACITY SPACE WISE	
Hold	752 of which 117 - 40'=31%
Deck	210 1st tier
	226 2nd tier
	226 3rd tier
	1414
	176 4th tier
	1590

Space on tanktop 2 tiers high
In guides chocks
Suitable for breakbulk
Load on tanktop 12 ton/M2

Hatchrows 3 + 3 + 3 WEIGHT 18T-19T
Number of hatches 8 fwd of deckh
1 aft of deckhouse
40' cells in hatch no 3 - 8 - 9
40' box below deck 117 = 46.48.23
In no 8 and 9 also 20' boxes can be stowed on cones
Max stackload on covers
20' stack 60 Ton
40' stack 90 Ton
Max stackload if twistlocks only 40 T
If stackload exceeds 40T lash by rods + turnbuckles

THRUSTERS (var pitch)
Bowthruster 900 kW, 1220 HP
Sternthruster 900 kW, 1220 HP
Rudder max angle 45°

INTEGRAL REEFERS	
Reeferpoints on deck 20'/40'	: 86
5. kW 20'/7.5kW 40'	
Reeferpoints in hold 20'	: 60
ventilation 1 m3/sec per box	
Receptacles 440V CEE 17, 3h, 32 Amp	
Hold ventilation H1, 2 & 5 2.5x/hr	
H3 & 4 9x/hr	

NEDLLOYD VAN NOORT 930
" VAN DIEMEN 931

Delivery April 1984 / July 1984
Yard v/d Giessen de Noord
Class Bureau Veritas
(Corr/BV/2-11-83) corr. 6/4-11-84

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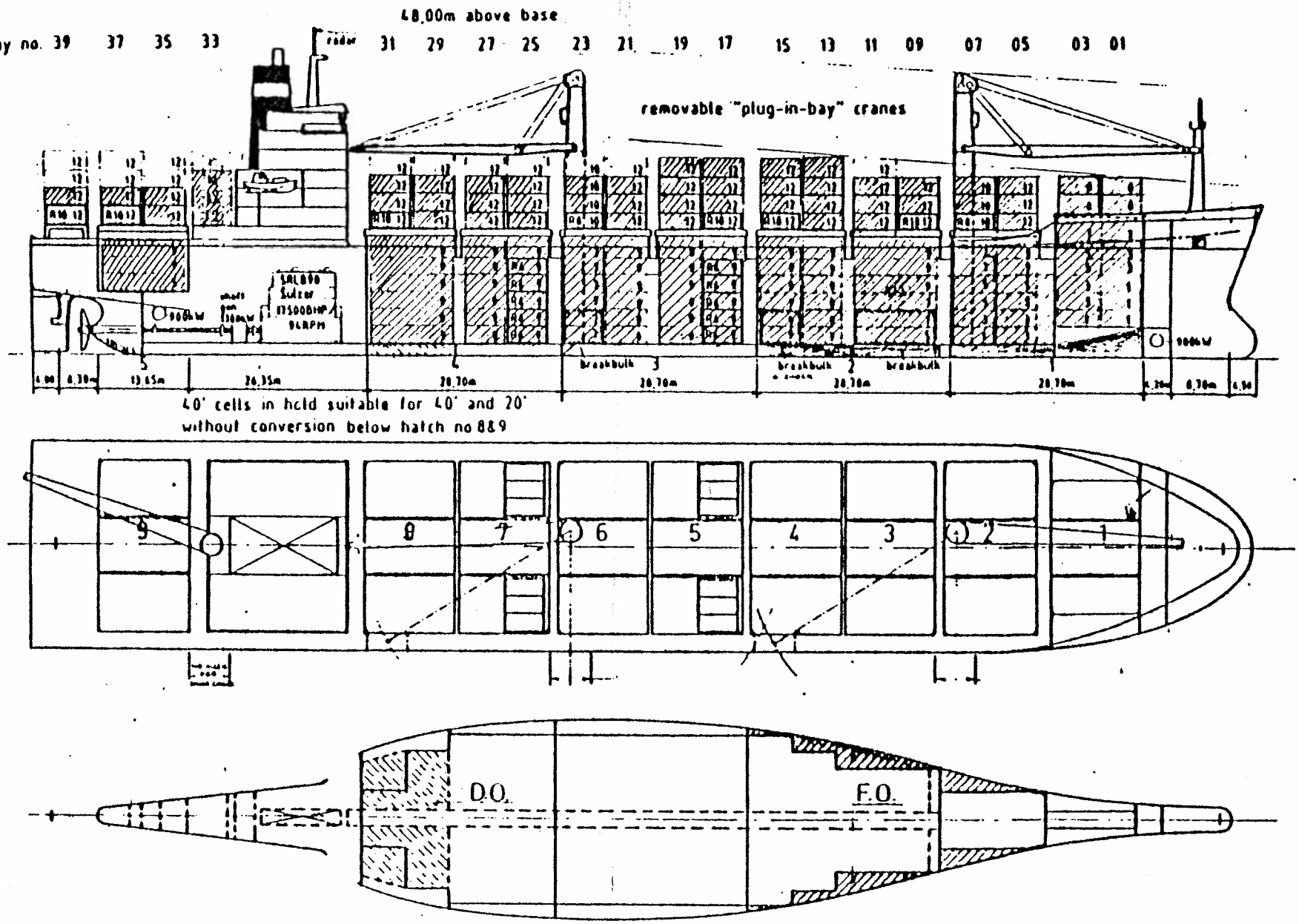


9.94 M

1426 TEU x 14.2 T/TEU HOM

" NEDLLOYD VAN NOORT "

OPTIMAL EASTBOUND

9-1-2

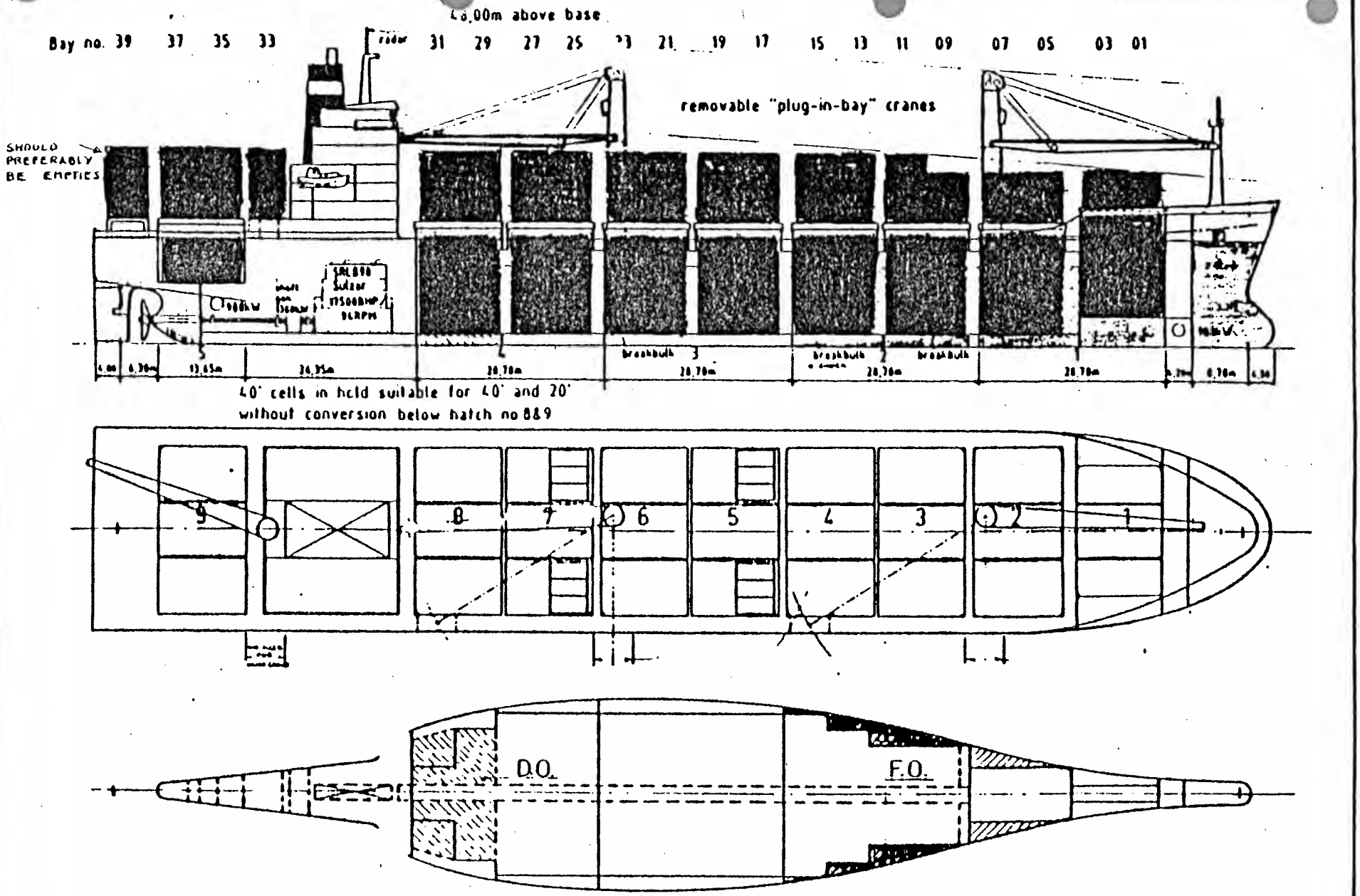
OPTIMAL EASTBOUND 1426 TEU x 14.2 TON/TEU

NEDLLOYD VAN NOORT " " DIEMEN + 3 JIBCRANES	WEIGHT IN TON. OF 1000 K.G.	V. C.O.G. ABOVE BASE m.	MOMENT ABOVE BASE m.t.	L. C.O.G. FROM A.P.P. m.	MOMENT FROM A.P.P. m.t.
LIGHT SHIP	11000	11.95	131450	7340	807400
CREW AND PROV.	25	21.80	545	3450	863
STORES FORW.	40	18.60	744	16530	6612
STORES AFT.	160	11.00	1760	3310	5296
HEAVY FUEL OIL	1429	6.25	8931	11780	1683336
DIESEL OIL	250	2.49	623	4871	1218
LUBR.OIL	100	8.55	855	3629	3629
FRESHWATER	250	7.69	1924	4346	1086 4
SEVERAL T.ENG.ROOM	95	1.67	159	1966	1868
CONTAINERS: 1426 TEU					
HOLDS 752	10678	10.88	112973	9133	975222
TIER 7 (onhatch) 210	2982	2.000	5964	7714	23002
TIER 8 226	3209	2.61	7238	8362	2651288
TIER 9 190	2698	2.823	6807	9562	2600681
TIER 10 48	682	2.792	1904	16288	7016 4
TIER 11					
TOTAL CONTAINERS	20249	16.41	332280	8895	1601226
WATERBALLAST:					
FOREPEAK 1					
DB TANK 1 2	300	0.97	291	14329	4298 7
DB TANKS 2 S.+P. 3+4	789	0.95	750	11360	9103 1
DB TANKS 3 S.+P. 5+6					
DB TANKS 4 S.+P. 7+8					
WINGT. 1 FORW. S.+P. 11+12					
WINGTANKS 3 S.+P. 19+20					
WINGTANKS 4 S.+P. 21+22					
TOTAL WATERBALLAST	1089	0.96	1041	12308	1340 38
DISPLACEMENT	34687	13.85	480312	8511	2952310

CONDITION: NR. HOM – 1426 TEU @ 14.2 TON DEPARTURE

Bendingmoment = mt. Shearforce = t. 57 % Bunkers

LCB from APP	= 85.20 m.	Trim.moment	= 3122 mt.	KM	= 14.37 m.
LCG from APP	= 85.11 m.	LCF from APP	= 78.39 m.	KG	= 13.85 m.
Draught mean	= 9.94 m.	Total trim	= 0.07 m.	GM	= 0.52 m.
Draught APP	= 9.97 m.	Trim APP	= 0.03 m.	GG'	= 0.06 m.
Draught FPP	= 9.90 m.	Trim FPP	= 0.04 m.	GM	= 0.46 m.



8.95 M

1590 TEU x 9.5 T/TEU HOM

"NEDLLOYD VAN NOORT" : OPTIMAL WESTBOUND

4 bay

BAY 11-21, 25-31, 35-39 — 882,690 : 13 = 67,90 X 156 = 10592
" — — 23, 33 — 101,241 : 2 = 50.62 X 20 = 1012
65.93 176 = 11604

NEDLLOYD VAN NOORT — OPTIMAL WESTBOUND
" " DIEMEN
+ 3 JIBCRANES — 1590 TEU x 9.5 TON/TEU — 8,95 M.

ITEM	WEIGHT	VCG	VERT MOM	LCG	LONG MOM	FREE S.M.
SHIP	11000		131450		807400	
CREW - STORES	225		3049		12771	
D.O/LO/FW/SEVERAL T	695		3561		28539	
FO	1429		8931		168336	
	2349		15541		209646	
CARGO, HOLDS 9.5 x 752 =	7144	10.58	75584	91.33	652,462	
1: 9.5 x 210	1995	20.-	39900	77.14	153894	
2: 9.5 x 226	2147	22.61	48544	82.62	177385	
3: 9.5 x 226	2147	25.23	54169	82.62	177385	
4: 9.5 x 176	1672	27.92	46682	65.93	110235	
HOMOG. 9.5 x 1590 TEU Ton/TEU	15105	17.54	264973	84.17	1271361	
TOTAL WITHOUT WB	28454	14.47	411870	80.42	2283407	
WB FOREPEAK	558	8,67	4838	169.69	94687	
TANK 1	546	1,87	1021	147.32	80437	
TANK 2 P+S	825	0,97	800	115.43	95230	
	1929		6659		270354	
DISPL	30383	13.78	418529	84.22	2558761	

LIGHT SHIP 11000
DW 19383

DRAUGHT MEAN	8.95 M	KM ~ 14.48	M	LCB	85.95 M	LCF	81.02 M
" FORW	8.25 "	KG - 13.78	"	ARM	- 1.73 "	TRIM AFT	.61 "
" AFT	9.56 "	0.70	"			" FORW	.70 "
		0.12	"				
		MG' 0.58	"	TRIM ASTERN	7.31 "		

CONCLUSION: STABILITY IS NO PROBLEM. HEAVIER CONTS TO BE F.WD. AND EMPTIES AFT. TO COMPENSATE FOR STERN-TRIM

LOADING CONDITION WESTBOUND

NEDLLOYD - NEWB. DEP.
NEDLL. VAN NOORT/V DIEMEN
DATE 16-9-83 | SH. No

"O P E N" TYPE BULKCARRIER

G E A R E D

SUITABLE FOR CONTAINERS
LUMBER / TIMBER
PIPES
PAPER
MOBILE HOMES
LARGE MODULES
GRAIN, COAL, ORES:

"OPEN" BULKCARRIER

CONTAINER-BULKCARRIER.

Starting points.

- It seems to be very difficult to estimate the demand for containervessels and it became clear that containerisation in the various trades is developing faster than expected.
- Some standardisation between 1000 and 1600 TEU is anticipated whereas a high speed is not anymore required.
- In future containertransportation might go into the direction of "slot chartering".

Development in shipdesign.

- Bulker/container with Loa up to 200 M.
- Draft 38 ft.
- Deadweight abt. 40,000 tonnes.
- Speed about 15-16 knots.
- Double-hulled vessel, rectangular holds and hatchcover pontoons from board to board.
- Suitable for grain 42-50 cft/t - coal - ore(altern.holds empty) (stability sufficient with two holds slack in case of heavy grain)
- Suitable for lumber, packaged timber, pipes, mobile homes, large modules
- suitable for abt 1,600 TEU.
- Conversion by installing guide structures should NOT be necessary.
- Hatchcover dimensions suitable for containers below and on top.
- NO 'piggyback' hatchcover handling, because this causes (involves) shifting of the containers on top of both covers.
- Folding covers (Hydraulic) are preferable, abt. 22' width.
- In athwartship direction blockstowage only (25mm) - on deck and in holds and with minimum required 'support' against ship's sides.
- In longitudinal direction abt 2' space between container fronts in order to make twistlock securing or rodlashing feasible.
- Ship's jib cranes or gantries should cause a minimum hindrance to shore container cranes when handling containers or timber.
- Ship's cranes suitable for a load of 38 ton (40' box + spreader)
- Maximum deadweight at minimum draft.
- Large cubic - well accessible to grabs - smooth walls - no pockets - no overhangs.
- Long and short holds in order to cope with (heavy) grain cargoes and heavy ores (with alternate holds empty).
- Seaworthy vessel loaded / in ballast / loaded with containers. Sufficient ballastwater capacity in ship's bottom, in double hull and between double bulkheads to give her sufficient draft forward against slamming and keeping bulb immersed and sufficient draft aft for propeller immersion.
- Nevertheless the vessel will behave poorly in heavy headseas if insufficiently loaded and will blow off her course by a strong side wind when loaded with 4 tiers of containers above deck.

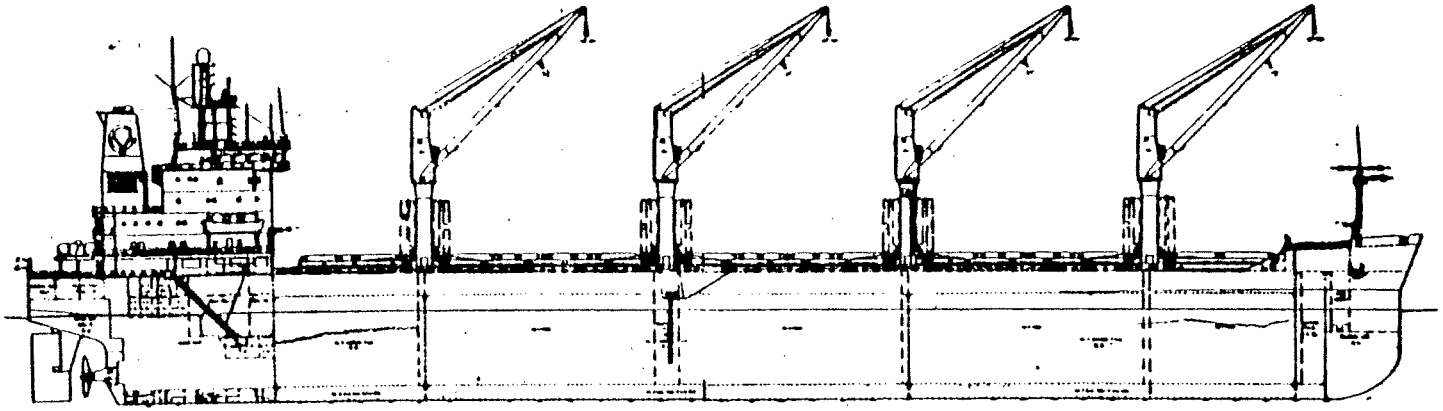
Conclusive note.

For a liner company it would seem worthwhile to have a sister company at hand with such 'liner-type' bulkers in order to be able to draw a number from that pool in case of 'emergencies' (e.g. when being squeezed out of her conference trades threatens her existence - or in order to fight herself into new trade(s)).

On the other hand, a bulkship operator should be able in times of low bulk-market conditions to let her ships to liner operator(s), which is near-impossible with diamond-shaped-section bulkcarriers but only likely for square-hold types.

t.S., 1979

Bulkcarrier "WIELDRECHT"



Builders: Öresundsvarvet AB, Landskrona
Owner: Phs. van Ommeren (Nederland) B.V., Rotterdam
Yard number: 281
Type: bulk/ore/container carrier
Delivery: May 1982

Tonnage	26 171.09	GRT
	16 913.70	NRT
Deadweight	41 800	t
Length o.a.	186.54	m
Length b.p.	180.00	m
Moulded breadth	30.00	m
Depth to maindeck	16.80	m
Draught	11.856	m

Speed: 16 kn
Classification:
B.V. I 3/3 E + Bulk-Ore-Carrier, Deep Sea, Glace 1, AUT-MS
Propelling machinery:
single acting 2-stroke B & W-Diesel engine, direct reversible
type: 4L90GFCA turbocharged
10 187 kW (13 850 BHP) at 96 rpm

propeller (Lips)

Auxiliary engines:
3 Wärtsilä Vasa-Diesel engines
type: 624 TS
724 kW (985 BHP) at 720 rpm each
3 generators 850 kVA, 450 V, 60 Hz each

1 Scania-Diesel engine
type: DS 1101 A 11
162 kW (220 BHP) at 1800 rpm
1 emergency generator 150 kVA, 450 V, 60 Hz

steering gear (Hatlapa)

gyro compass and automatic pilot (Anschütz)

5 hatches

hatch 1	12.90 × 15.40	m
hatch 2, 3 and 4 each	25.80 × 25.30	m
hatch 5	12.90 × 25.30	m

Hatch covers: system NAVIRE CARGO GEAR folding type

Loading gear: 4 cranes of 36 t/30 m

7 self-tensioning mooring winches of 16 t
2 comb. windlasses/mooring winches of 20 t

Hold capacity:	grain m^3	bale m^3
hold 1	6 188	5 990
hold 2	13 209	12 869
hold 3	12 465	12 288
hold 4	12 732	12 550
hold 5	7 086	6 901
Total cargo capacity	51 680	50 598

Container capacity:	TEU
holds	892
on hatch covers	508
on deck	102
Total	1502

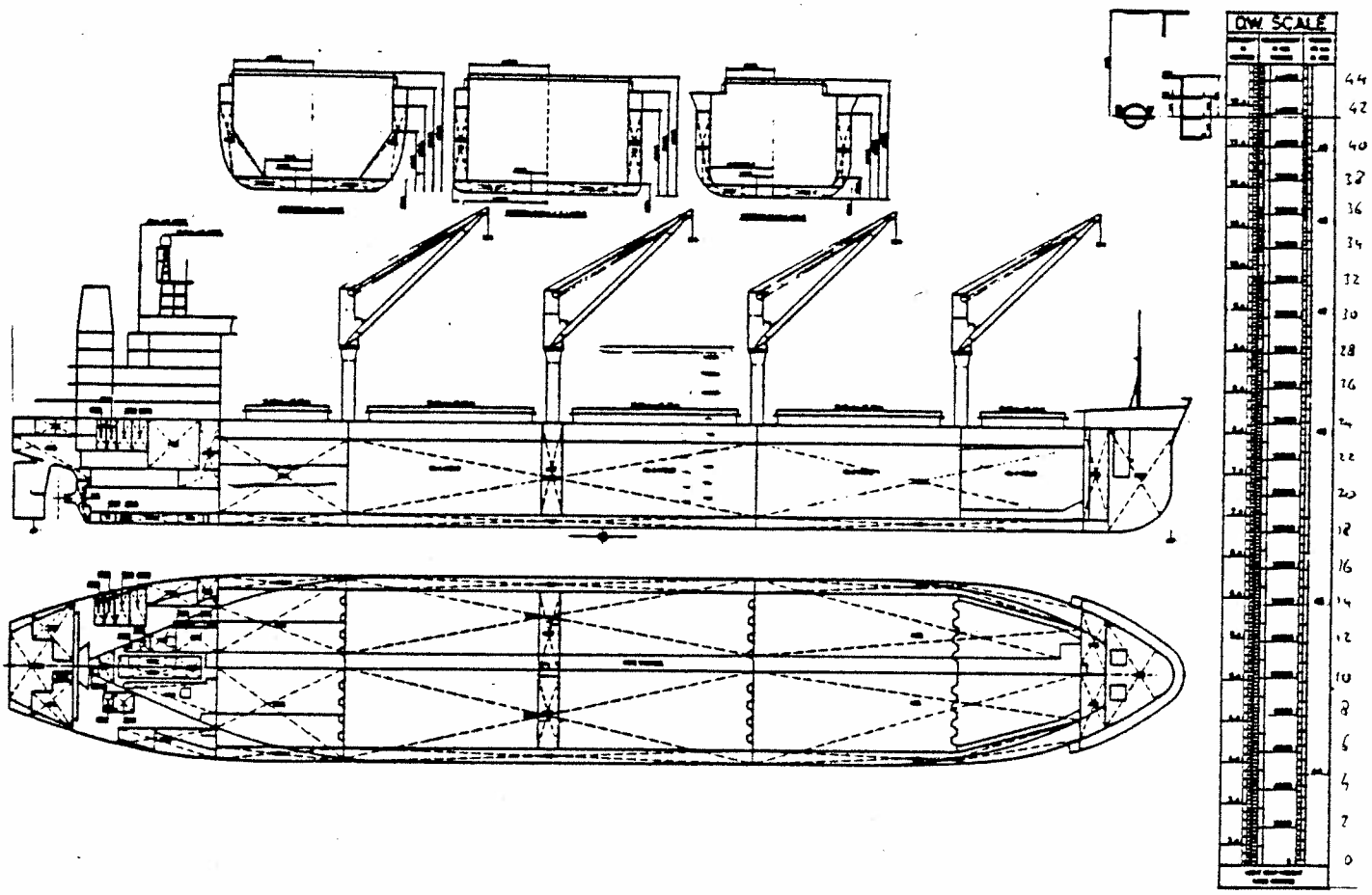
48 cft/t
↔
42 cft/t

10-0-2

m.s. "WAARDRECHT", "WIELDRECHT", "WOENSDRECHT"

built 1982 by :

Öresundsvarvet AB, Landskrona, Sweden



PARTICULARS

summer deadweight	41.800 mt	41.142 lt
summer draught	11.856 m	38'11"
immersion	49.46 tpcm	123.6 tpi
length overall	186.54 m	612'
length between p.p. ...	180.00 m	590'7"
breadth	30.00 m	98'5"
depth	16.80 m	55'1"
total grain capacity ..	51.680 m3	1.825.060 cuft

Lightship weight 11,200 tonnes

Class

Bureau Veritas I 3/3 + Bulk-Ore-Carrier, Deep Sea Aut.-MS
Glace 1 (equiv. Lloyds' Ice Class 1)

Speed and consumption

service speed at container draught	16 kn
average service speed fully loaded	15 kn
m.e. B&W 4L90 GFCA MCR 13.860 hp,	96 rpm
daily consumption HFO	40 t
daily consumption auxiliaries 1,5 t HFO + 0.5 t DO	

Cargo gear

Four Hägglunds 36 tons electro-hydraulic deckcranes with a slewing radius of 30 m, combinable to 72 t. Fitted with grabs for a theoretical selfunloading speed of 25.000 t / 24 hours.

Hatches

Five Navire folding type hatches, dimensions:

hatch 1	12.05 x 15.44 m
hatches 2, 3, 4	25.91 x 25.36 m
hatch 5	12.05 x 25.36 m

Maximum loads

tanktop strength hold 2, 4	19.44 t/m2
tanktop strength hold 1, 3, 5	17.50 t/m2
hatches	2.60 t/m2

Hold capacities

	length	grain m3	grain cuft.
no. 1	19.44 m	6.188	218.530
no. 2	32.40 m	13.209	466.470
no. 3	30.78 m	12.465	440.200
no. 4	30.78 m	12.732	449.620
no. 5	20.25	7.086	250.240
		51.680	1.825.060

Containers

Fully self-sustaining with a capacity of up to 1.664 TEU with 120 reefer plugs.

Stackloads

tanktop	120 tons (20') ..	180 tons (40')
hatches	45 tons (20') ..	68 tons (40')

Container cargo

	20' units	40' units
in holds	892	446
on deck	656	276
	1.548	722
optional extra ... (on deck)	116	46
	1.664	768

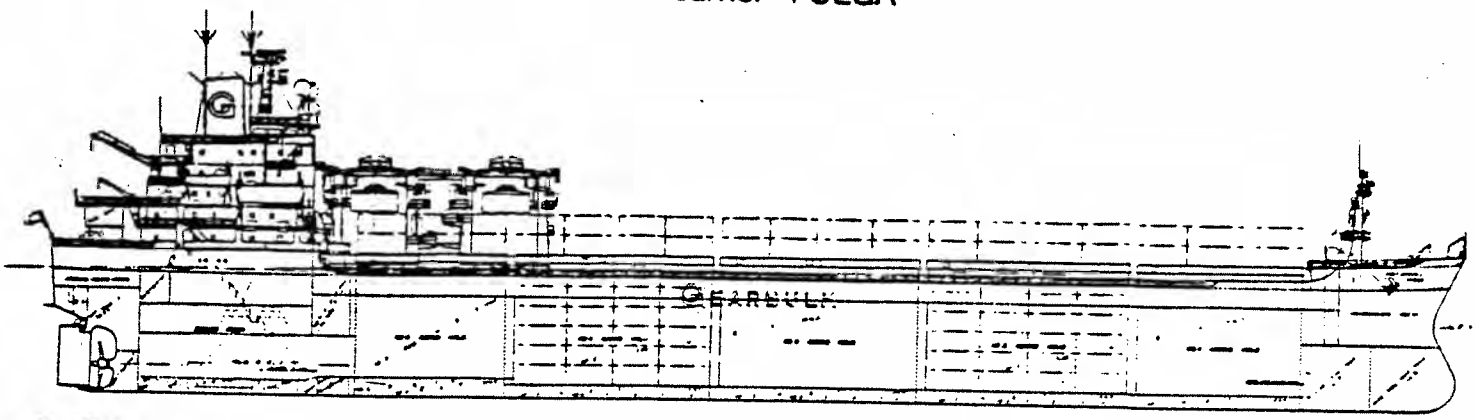
Pipe cargo

length 12.0 m	diam. 1.0 m	diam. 36"
in holds	3.270	3.924
on deck	1.414	1.591
total	4.684	5.515

Timber cargo

	m3	cuft
in holds	50.600	1.787.000
on deck	10.500	371.000
total	61.100	2.158.000

Bulkcarrier "FOLGA"



Builders: Sanoyasu Dockyard Co., Ltd., Mizushima Shipyard
Owner: A/S J. Ludwig Mowinckels Rederi, Bergen
Yard number: 1042
Delivery: February 25th, 1982

Tonnage	24 999.99 GRT 13 794.20 NRT
Deadweight	38 806 t
Length o. a.	182.00 m
Length b. p.	174.00 m
Moulded breadth	29.00 m
Depth to maindeck	16.10 m
Draught max.	11.613 m

Speed: 13.8 kn

Classification:
N.V. + 1 A 1 HC (holds Nos. 2 and 4 or hold No. 3 may be empty) + MV, EO

Propelling machinery:
Mitsui/B & W-Diesel engine
type: 6L67GFC
8238 kW (11 200 HP) at 119 rpm
Fuel consumption (7502 kW):
35.6 t per day
3 Diesel generators 812.5 kVA, 450 V, 60 Hz each

Crew: 31 persons

5 hatches

hatch 1	21.00 × 16.50 m
hatch 2, 3 and 4 each	25.60 × 23.00 m
hatch 5	19.20 × 23.00 m

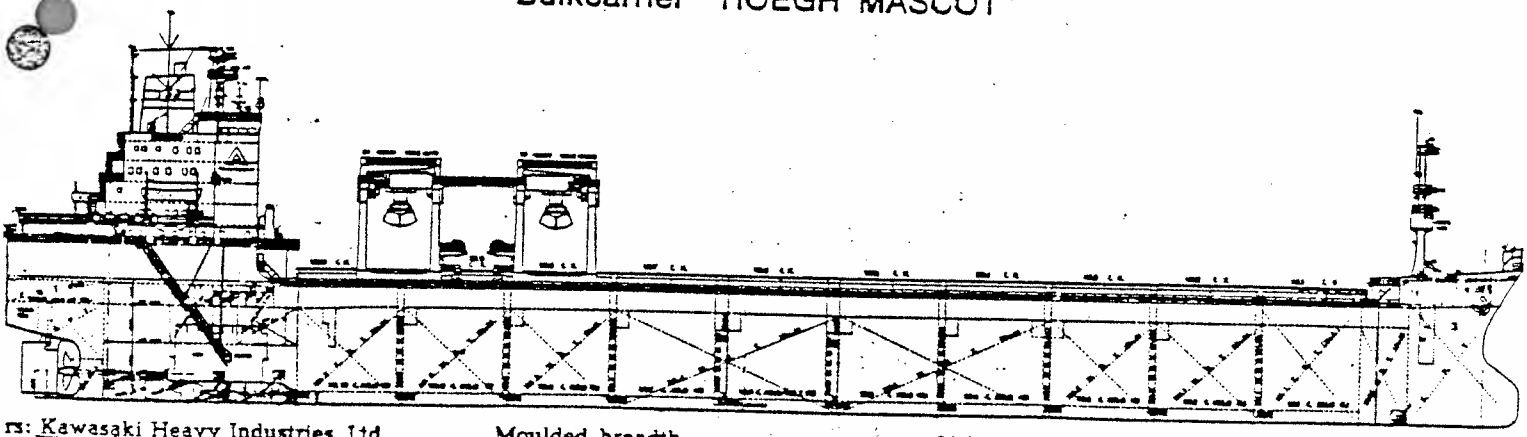
Hatch covers: system MacGREGOR piggy-back type

Loading gear: 2 electric Gantry-cranes of 30 t/7 m — 25 t/10 m
2 electric hydraulic mooring winches of 15 t
2 electric hydraulic windlasses of 27 t

Total cargo capacity:
grain 42 190.9 m³

Container capacity:	TEU
holds	432
on deck	306
Total	738

Bulkcarrier "HÖEGH MASCOT"



rs: Kawasaki Heavy Industries, Ltd., aide Works
: A/S Alliance
ring owner: Leif Höegh & Co. A/S, Oslo
number: 1252
ry: July 1977

ge	29 212.63 GRT 17 018.33 NRT
ei	45 063 t
o. a.	200.50 m
b. p.	190.00 m

Moulded breadth	30.80 m
Depth to maindeck	15.70 m
Draught	11.565 m

Speed: 15.3 kn

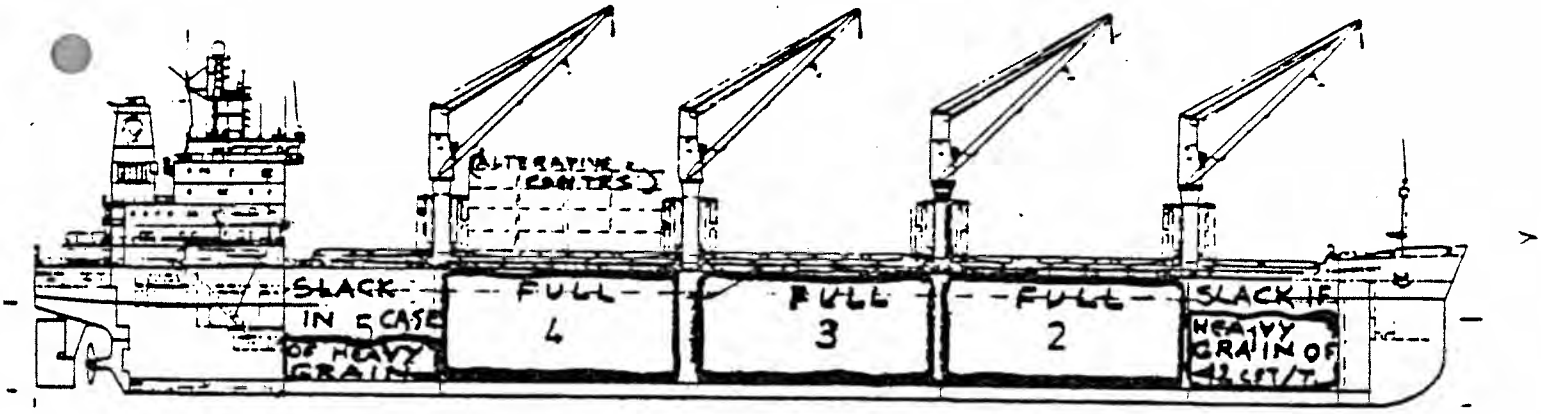
Classification:
N.V. + 1 A 1 + MV, HC (holds Nos. 3, 5, 7 and 9 may be empty), EO, ib(+)

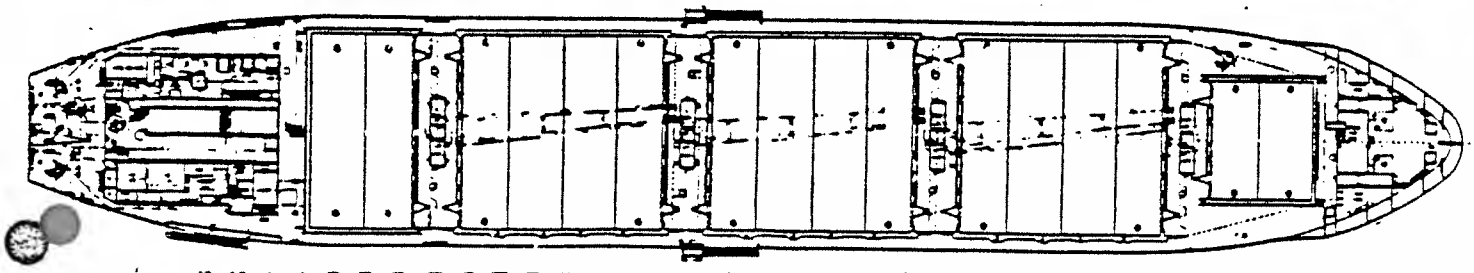
Propelling machinery:
single acting 2-stroke Kawasaki/MAN-Diesel engine
type: K8SZ 70/125 supercharged
15 200 HP at 145 rpm

Crew: max. 32 persons
10 hatches
Hatch covers: pontoon type
Loading gear: 2 gantry cranes of 30 t
4 mooring winches of 16 t, 2 comb. windlasse mooring winches of 27/16 t

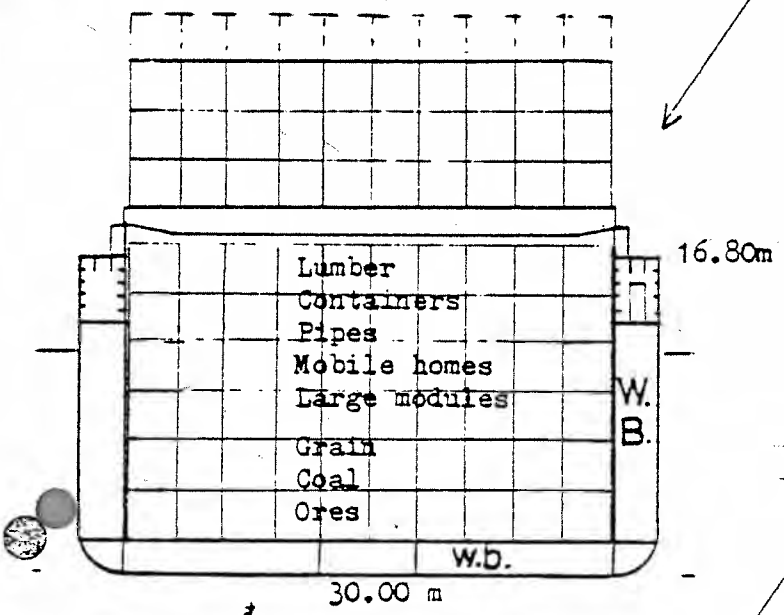
Total cargo capacity:

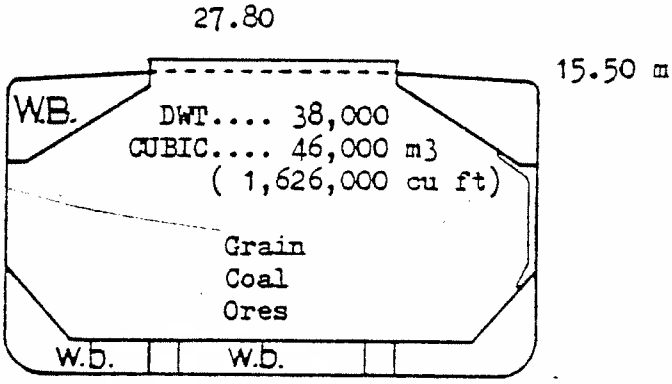
grain	1 770 582 cub.
bale	1 764 385 cub.





"W A A R D R E C H T" = 'O p e n' bulkcarrier.

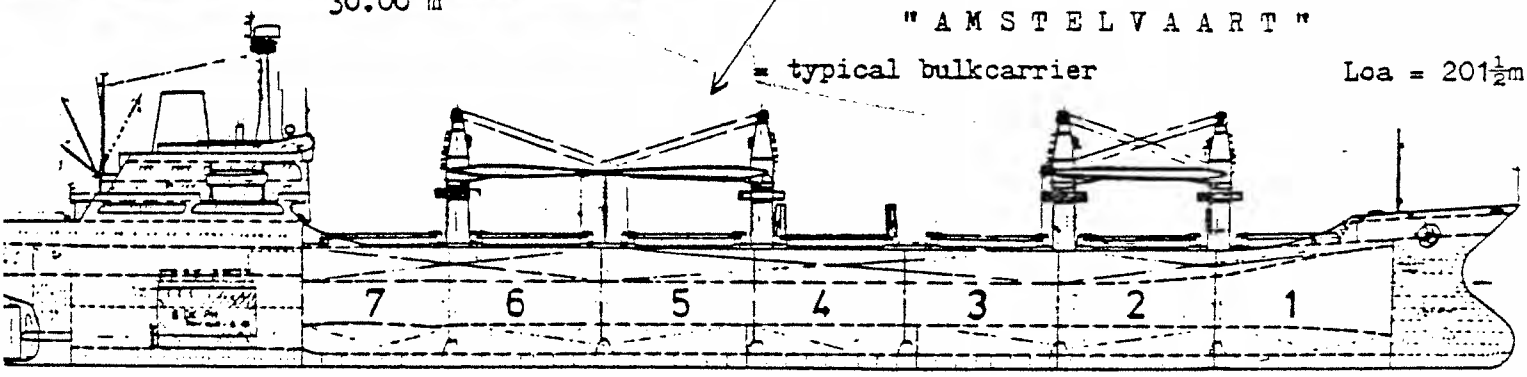


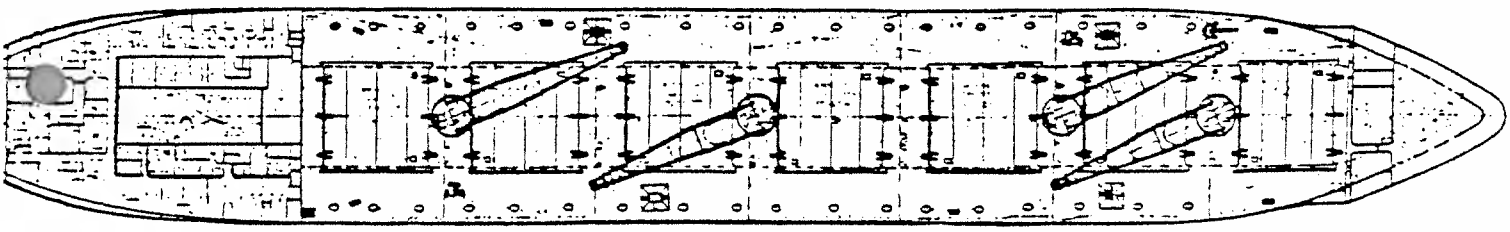


"A M S T E L V A A R T"

= typical bulkcarrier

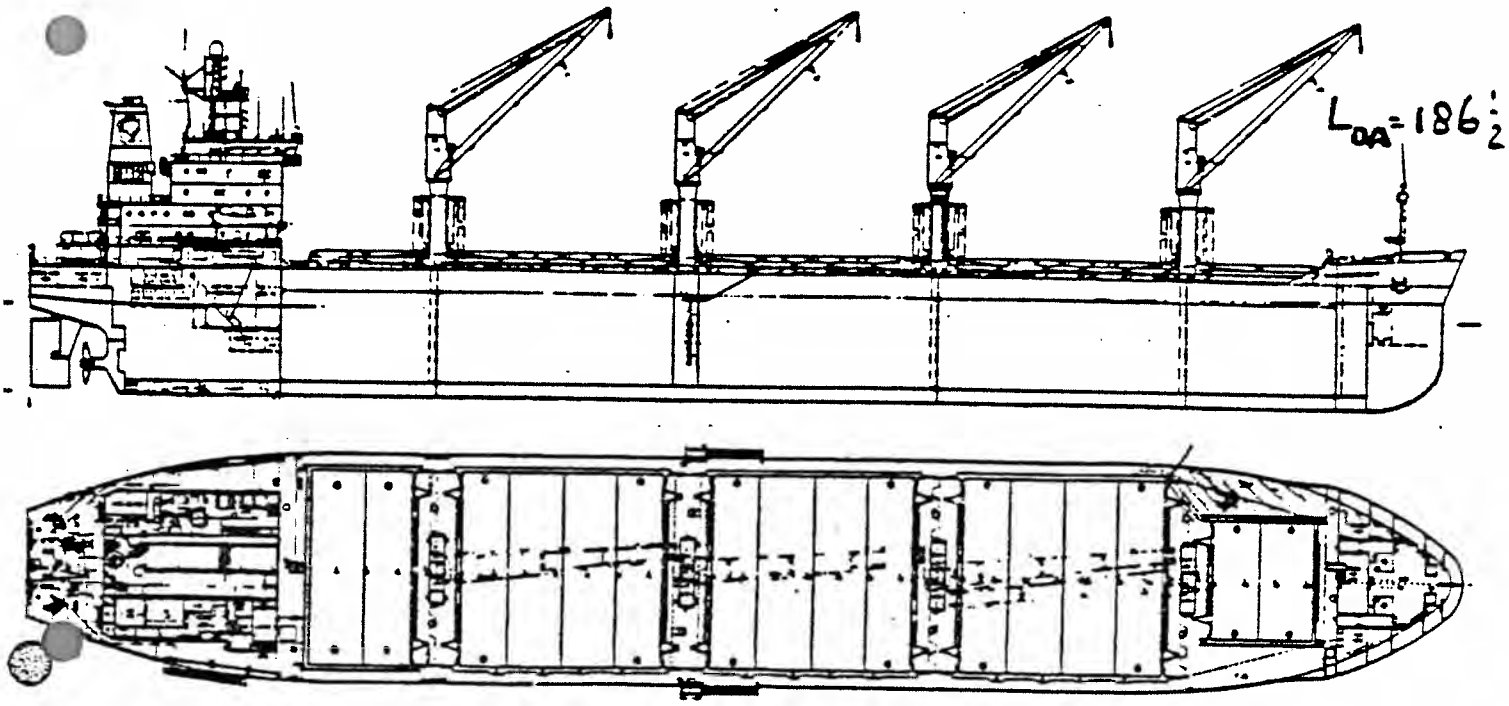
Loa = 201½m





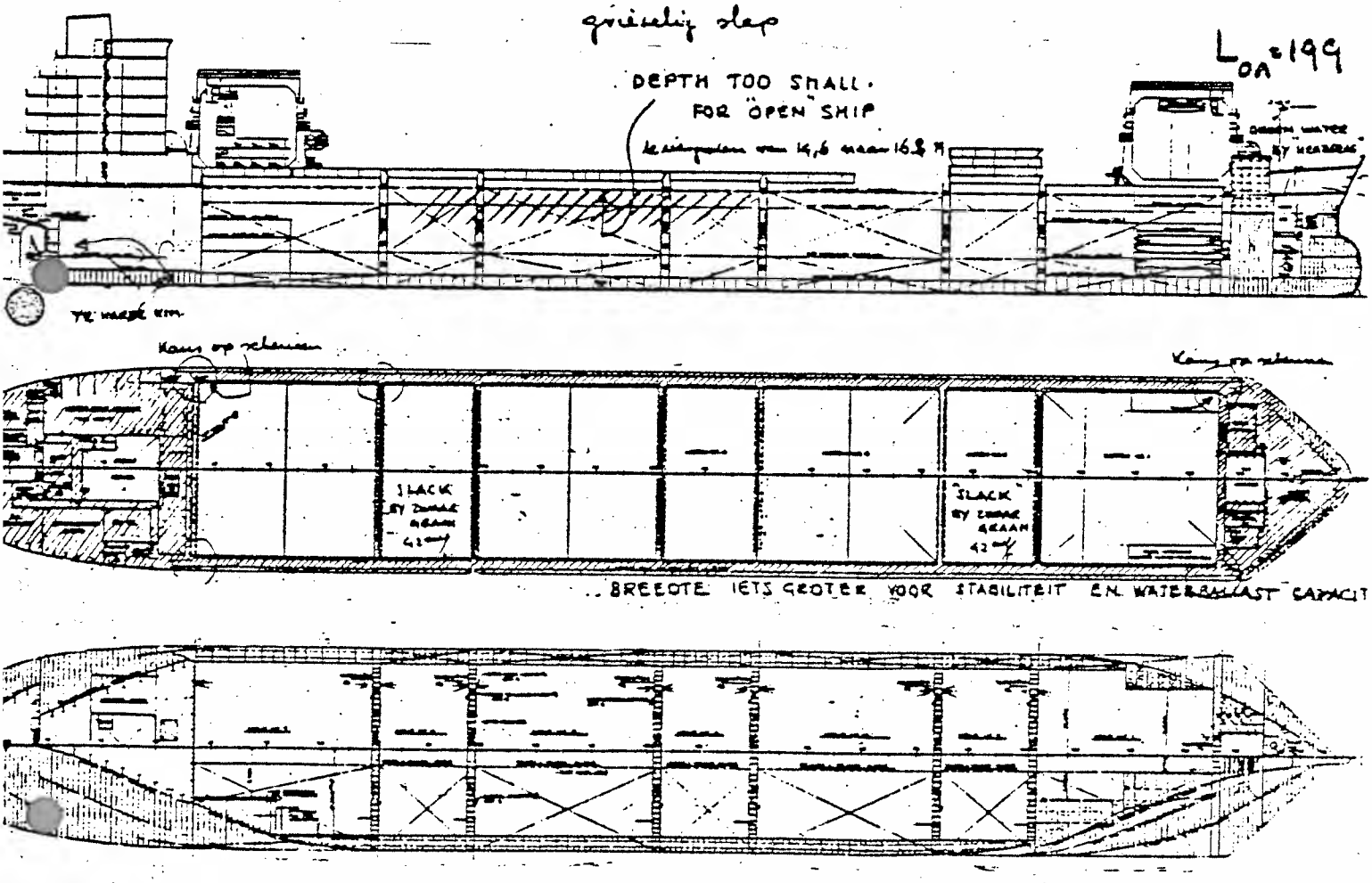
10-0-5

WAARDRECHT V. OMMEREN.

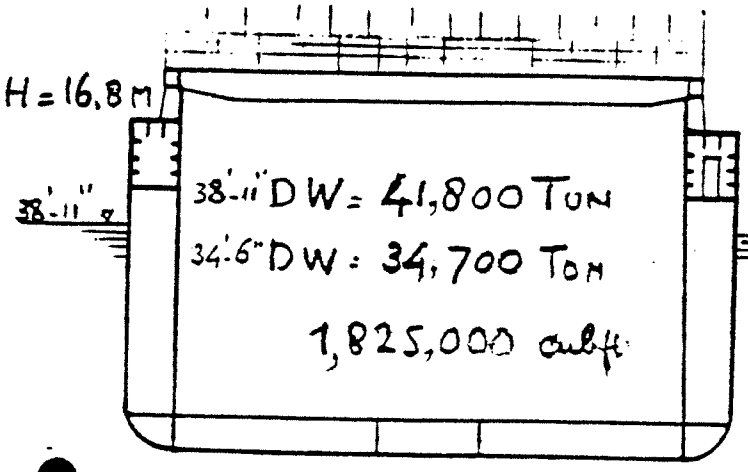


CANADIAN TRANSPORT DESIGN HERBERT

grieselig slap

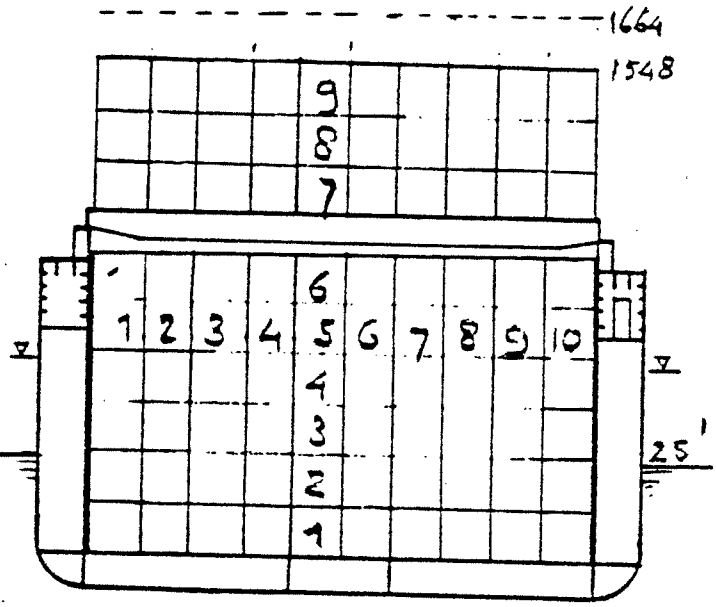


"WAARDRECHT"



" Timber cargo "

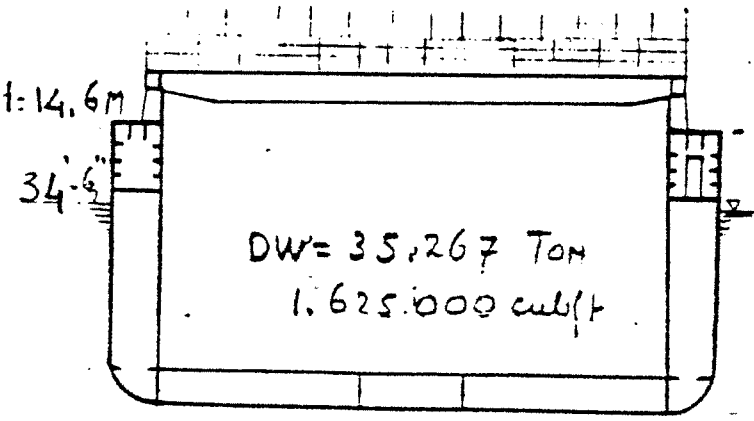
B = 30,0 M



"Container cargo"

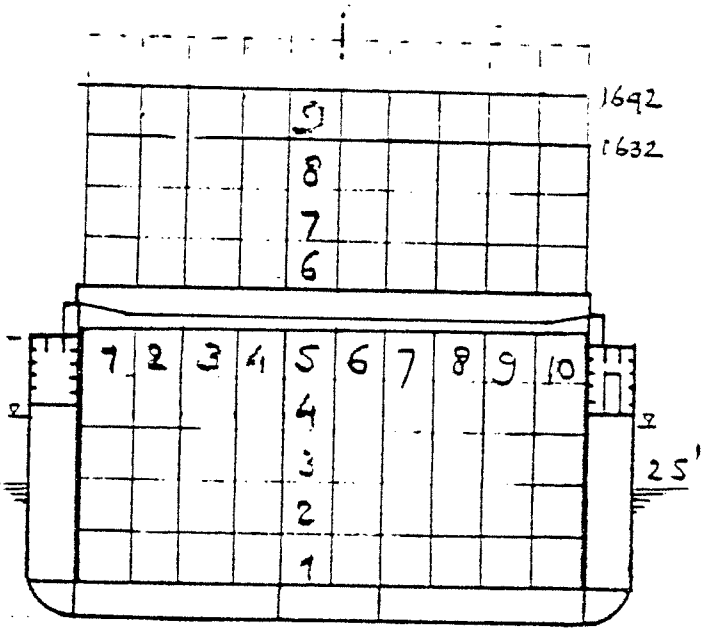
IN HOLD 892
ON DECK 656
1548
2 x 6 in 4/D 116
1664 TEU.

"CAN. TRANSPORT"



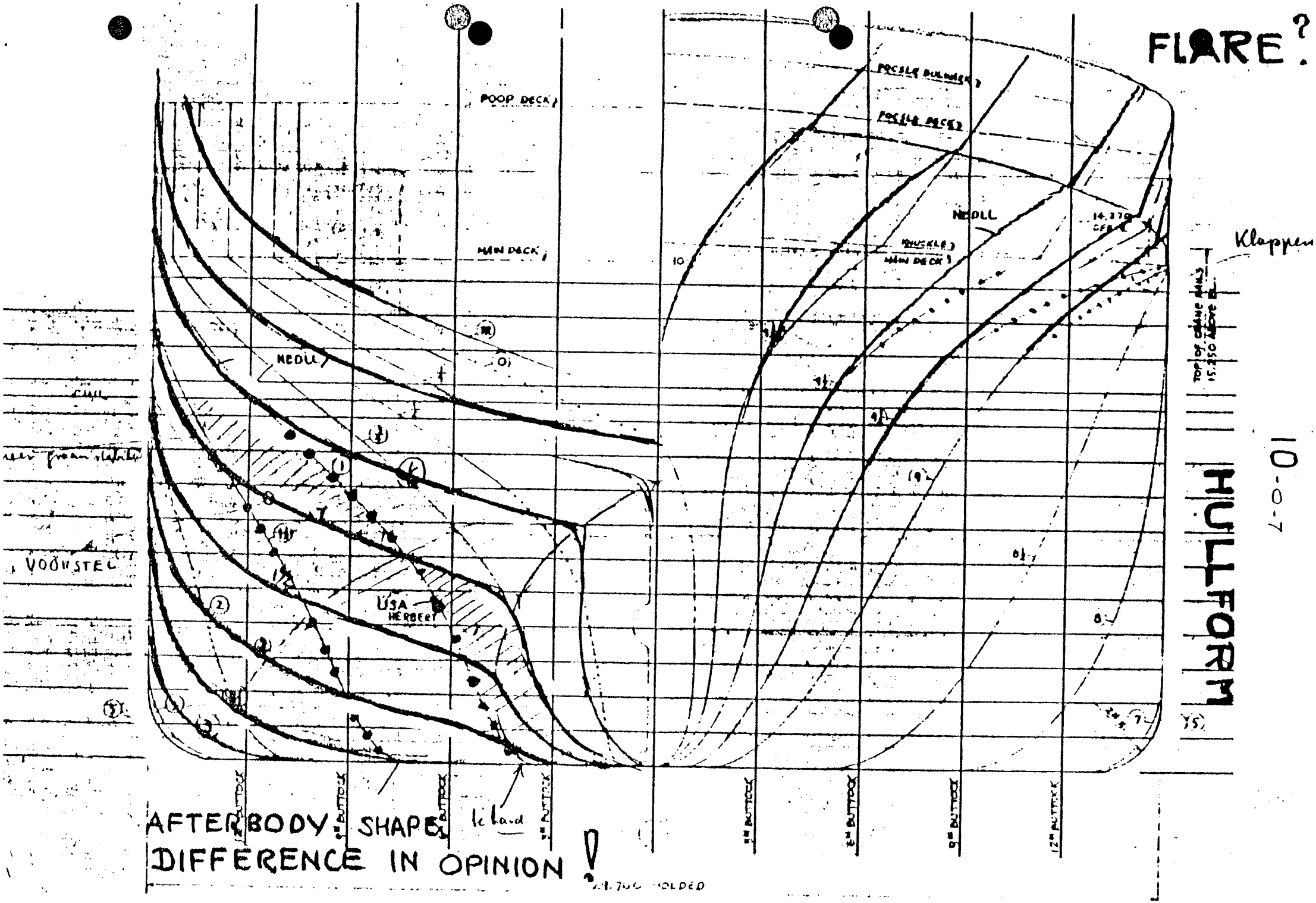
" Timber cargo "

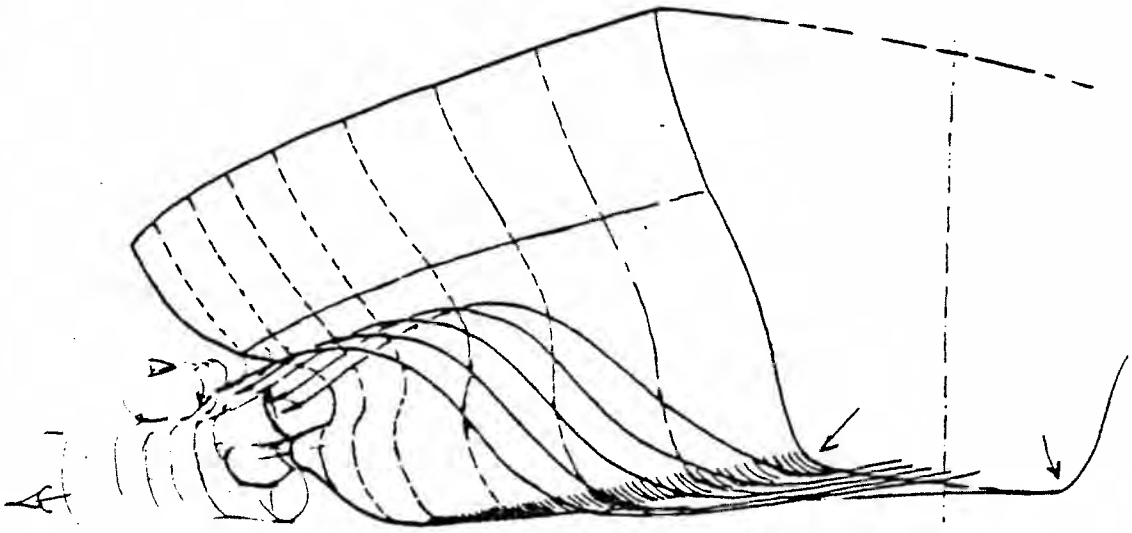
B = 29,76 M.



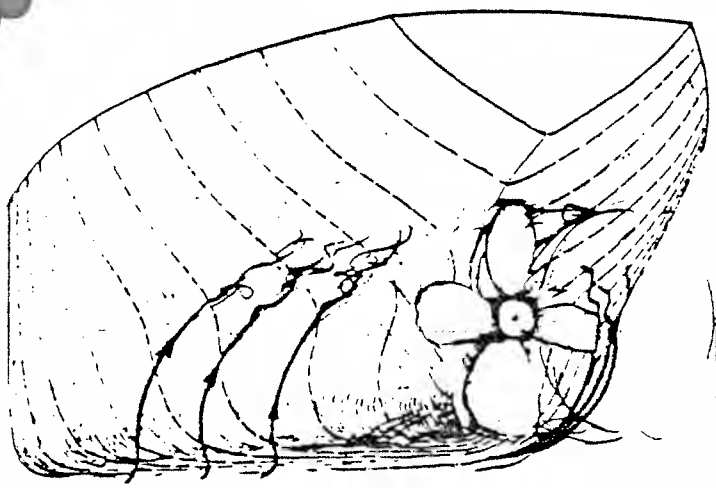
"Container cargo"

IN HOLD 1012
AD, incl. 60 in 4 tier 680
1692 TEU

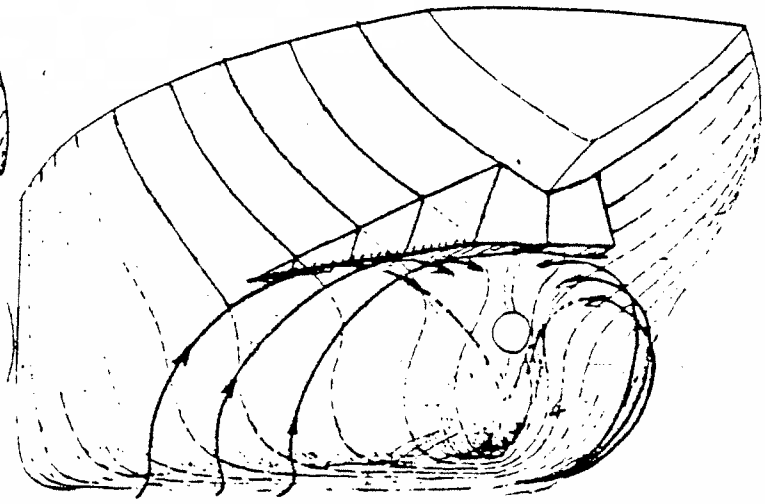

FLARE?
POOP DECK
MAIN DECK
NEDLL
USA
HERBERT
Klappen
HULLFORM
10-0-7
AFTERBODY SHAPE
DIFFERENCE IN OPINION!



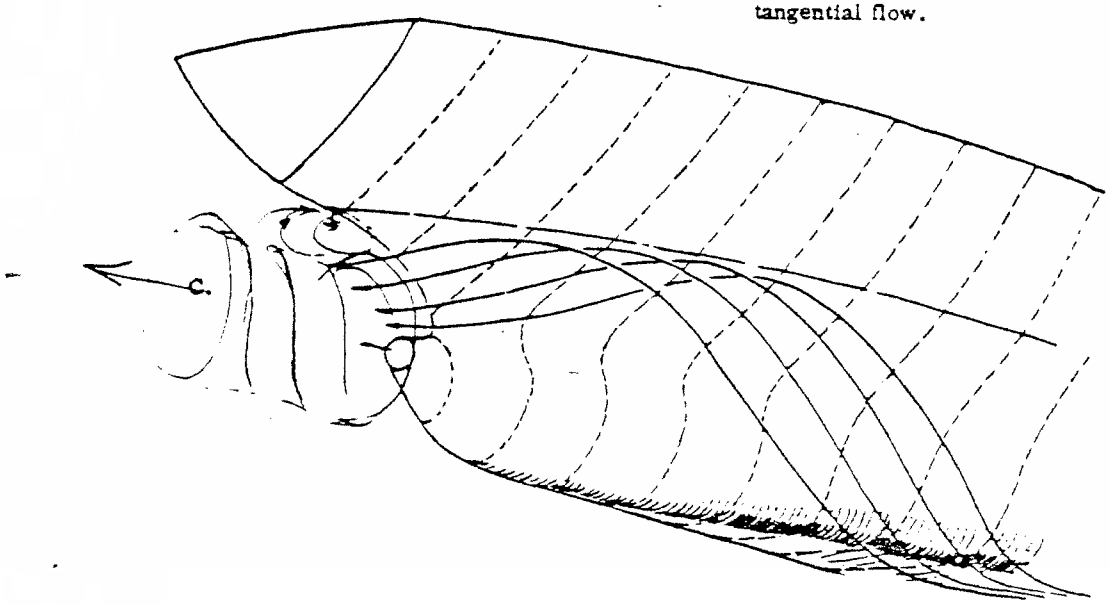
A. Excited vortex by small bilge roundings.



B. Excited vortices strongly influenced by trim and shallow water.



D. Function of fin or tunnel. Directing flow to propeller equalizing wake by reducing the wake peak in top position. However not influencing tangential flow.



" WOENSDRECHT " - CLASS Bulkcarrier

PERFORMANCE FIGURES (CASE HISTORY).

VESSEL: Woensdrecht

VOYAGE: 638 WESTB

WOENSDRECHT
MAY - JUNE - JULY '83

m.s. "WOENSDRECHT" 638 ANCHORED/STOPPED AWAITING: B=Berth. C=Convoy. P=pilot. T=Tugs. Q=Quarantine

PORT	TIME of DEPARTURE/ARRIVAL		TIME AT SEA			obs. DISTANCE		Av. SPEED	DRAUGHT	TRIM	OIL CONSUMPTION (metric tons)									PER-FORM.
											Main Engines		Auxiliary Engines				Boilers	Total Consumption		
Port of Dep. / Port of Arr.	From Date GMT	To Date GMT	Steaming Pilot – Pilot Hrs Min	Stopped or Anchored Hrs Min	Man. & pilotage Hrs Min	Pilot – Pilot	Man. & pilotage	Pilot – Pilot	Average Mean Draft	Average during Passage	Fuel	Diesel	at sea: Fuel	Diesel	In Port of Dep.: Fuel	Diesel	Fuel Diesel	Fuel	Diesel	Tons Fuel per 24 hours
1	2.		3a.	-b.	-c.	4a.	-b.	5.	6a.	-b.	7a.		-b.		-c.		-d.	-c.		8.
YON-KOY	27/5 06.55	27/5 23.15	11.30	P01.00	3.50	189	25	16.43	07.10	+2.30	21.2	-	0.6	0.6	0.6	0.6	0.3	22.7	1.2	44.24
KOY-HWG	30/5 07.24	31/5 08.50	22.06	-	3.20	331	22	14.98	07.40	+1.80	30.7	-	1.4	1.4	3.7	3.7	2.4	38.2	5.1	33.74
HWG-SIN	1/6 19.55	5/6 15.40	89.30	-	2.15	1372	15	15.33	07.20	+2.00	144.5	-	5.0	5.3	2.6	2.6	1.2	153.3	7.9	38.75
SIN-SIN	6/6 03.00	10/6 01.40	91.36	0.45 & 0.30	1.49	1440	20	15.72	07.40	+2.00	160.4	-	5.3	5.4	0.7	0.8	0.4	166.8	6.2	43.7
SIN-JED	10/6 07.42	22/6 19.40	294.00	P3.45	2.13	4307	14	14.65	07.50	+2.10	506.9	-	17.2	18.6 (1.4 K)	0.6	0.6	2.3	527.0	19.2	43.0
JED-AQJ	23/6 11.00	25/6 10.30	28.54	B7.36	1.00	566	8	14.55	07.85	+1.90	54.7	-	2.1	2.3	1.2	1.5	0.5	58.5	3.8	47.6
AQJ-SUE	26/6 15.00	27/6 12.00	19.00	-	2.00	288	12	15.16	07.60	+1.60	28.5	-	0.3	0.4	2.2	2.6	0.7	31.7	3.0	36.0
SUE-CAS							94		07.80	+0.50	-	8.7	1.9	2.1	-	-	1.2	3.1	10.8	-
CAS-BEY	28/6 0.00	29/6 09.10	14.18	-	1.52	219	9	15.31	07.60	+1.60	25.9	0.2	0.8	0.9	-	-	0.2	26.9	1.1	43.3
BEY-LAT	30/6 08.40	30/6 16.30	6.12	-	1.38	100	9	16.12	07.10	+2.00	10.9	0.2	0.5	0.5	1.6	1.7	0.9	13.9	2.4	43.3
LAT-LHS	1/7 05.33	2/7 04.00	8.42	B12.15	1.40	135	10	15.51	07.00	+2.00	16.6	0.3	0.5	0.6	0.9	1.4	1.1	20.1	2.3	46.6
LHS-PAS	2/7 14.42	4/7 06.40	33.24	P4.14	2.20	510	11	15.27	06.90	+1.80	56.9	0.3	1.7	1.9	1.4	2.2	1.1	61.1	4.4	37.7
PAS-CAS	4/7 21.56	9/7 09.10	105.36	-	1.38	1657	11	15.69	07.10	+1.95	187	0.2	3.7	9.0	0.6	2.5	1.0	194.3	11.7	44.7
CAS-CAS	9/7 17.44	10/7 13.20	21.00	-	2.36	314	14	14.95	06.85	+2.20	26.2	7.1	0.5	2.1	0.5	0.9	0.3	34.6	10.1	39.5
											diesel 0.5	13.0	fuel							
		EB Total	775.42	73.26	36.20	12110	296	15.61			1371.1	12.1	56.7	31.5	21.63	11.51	12.7	1449.23	56.6	41.49
		WB Total	745.48	30.05	28.11	11428	274	15.32			1272.4	17.0	41.5	51.1	16.6	21.1	14.6	1345.1	89.2	40.95
		Grand Total	1521.30	103.31	64.31	23538	570	15.47			2613.5	30.1	98.2	82.6	38.23	32.61	32.3	2794.33	145.8	41.23
											0.5 bo	12.1 fo								

9 REMARKS: YON-KOY: economic speed.
JED-AQJ: —— " ——
AQJ-SUE: —— " ——
CAS-CAS: —— " ——

	Fuel	cSt at 50°C / 100°F	Diesel
Stock at beginning of voyage:	992.0		199.8
Bunkered, port: SIN date: 10-6-83	6000	347	
Bunkered, port: JED date: 23-6-83	7500	140	
Bunkered, port: date:			
Consumption during voyage	1345.1		89.2
Stock at end of voyage	994.9		110.6

WESTBOUND 971 TEU x 9.9 T/TEU = 9640 T
BREAKBULK 240 T
9889 T

FAR EAST — MEDIT

W1
WESTBOUND

Date 11-07-1983

Master

10-0-10

VESSEL: m.s. "WOENSDRECHT"

VOYAGE: 667

667 EASTB.

WOENSDRECHT

JULY – AUG '83

Awaiting B = Berth P = pilot C = convoy

PORT		TIME of DEPARTURE/ARRIVAL				TIME AT SEA			obs. DISTANCE		Av. SPEED	DRAUGHT	TRIM	OIL CONSUMPTION (metric tons)										PER-FORM.
														Main Engines		Auxiliary Engines				Boilers		Total Consumption		
Port of Dep.	Port of Arr.	From Date	From GMT	To Date	To GMT	Steaming Pilot – Pilot Hrs	Min	Stopped or Anchored: Hrs	Min	Man. & pilotage: Hrs	Min	Pilot – Pilot	Man. & pilotage	Fuel	Diesel	at sea: Fuel	Diesel	In Port of Dep.: Fuel	Diesel	Diesel	Fuel	Fuel	Diesel	Tons Fuel per 24 hours
1.		2.				3a.		–b.		–c.		4a.	–b.	5.	6a.	–b.		7a.		–b.		–c.		8.
LIS	AGC	15/7	21.47	15/7	19.10	18.12		–		03.11		300	24	16.48	6.90	+1.30		34.9	–	0.8	2.1	3.6	9.6	46.0
AGC	FOS	16/7	12.57	18/7	16.30	45.30		B 3.05		2.58		682	23	14.99	7.15	+0.70		57.7	0.9	1.0	4.3	0.4	2.0	60.4
FOS	LIV	19/7	11.06	20/7	05.20	15.06		P 1.35		1.33		242	15	16.03	7.50	+1.00		27.4	0.4	0.6	1.2	0.9	1.9	29.8	3.7	43.5
LIV	PAS	21/7	22.40	24/7	09.10	56.24		–		2.06		892	8	15.82	8.70	+0.10		108.0	–	1.7	4.8	1.8	4.7	109.4	11.1	43.8
PAS	BSD	24/7	23.24	26/7	12.40	36.06		–		1.10		585	7	16.20	8.85	+0.90		65.3	–	1.2	2.9	0.8	2.6	67.9	5.5	43.4
BSD	SUE	26/7	12.42	27/7	16.18	–		C 09.50		13.28		–	100	–	8.85	+0.90		10.3	–	2.0	3.5	–	–	13.7	3.5	–
SUE	JED	27/7	16.18	29/7	08.00	38.24		–		1.18		622	8	16.20	8.85	+0.90		67.8	–	1.4	2.8	–	–	69.2	2.8	47.4
JED	SIN	29/7	14.10	10/8	00.36	272.30		–		1.56		4342	11	15.93	9.20	+0.60		492.5	–	(14) 1.2	(1.7) 1.5	0.3	0.5	512.1	17.2	43.8
SIN	JKT	10/8	12.30	12/8	02.42	35.18		–		3.04		540	10	15.30	7.85	+1.70		65.5	–	2.7	1.4	0.9	0.6	69.5	2.5	44.5
JKT	SUB	13/8	10.56	14/8	16.30	24.00		P 00.41		4.53		367	40	15.29	7.85	+1.70		44.8	–	1.5	1.1	3.1	1.6	50.7	2.7	44.8
SUB	HKG	15/8	04.55	20/8	15.00	118.36		B 06.01		5.38		1859	48	15.67	7.55	+1.20		127.7	–	9.1	4.2	2.3	1.1	139.7	5.3	35.9
HKG	KEE	20/8	22.31	22/8	04.30	27.30		–		2.29		466	19	16.95	6.70	+1.00		43.3	–	2.1	1.0	1.5	0.8	47.3	1.8	37.8
KEE	AGO	22/8	23.12	25/8	22.30	57.18		P 09.13		4.47		983	48	17.16	6.90	+1.40		108.5	–	4.5	1.7	2.0	0.8	115.5	2.5	45.4
					EB Total	744.54		35.45		47.31		11880	368	15.95				1303.7	1.3	42.6	42.7	9.6	26.2	3.5 / 11.7	1376.6	78.7	42.00
					WB Total																				1345.1	89.2	
					Grand Total																				2721.7	167.9	

9. REMARKS: AGC – FOS economic Speed

SUB – HKG ——"——

EASTBOUND 11.63 TEU x 15.7 TON/TEU = 182.93 T

MEDIT → FAR EAST

	Fuel	cSt at 50°C	Diesel
Stock at beginning of voyage:	996.9		110.6
Bunkered, port: LIS date: 13/7			100.–
Bunkered, port: Jed date: 29/7	800		
Bunkered, port: date:			
Consumption during voyage	1376.6		78.7
Stock at end of voyage	420.3		131.9

W2

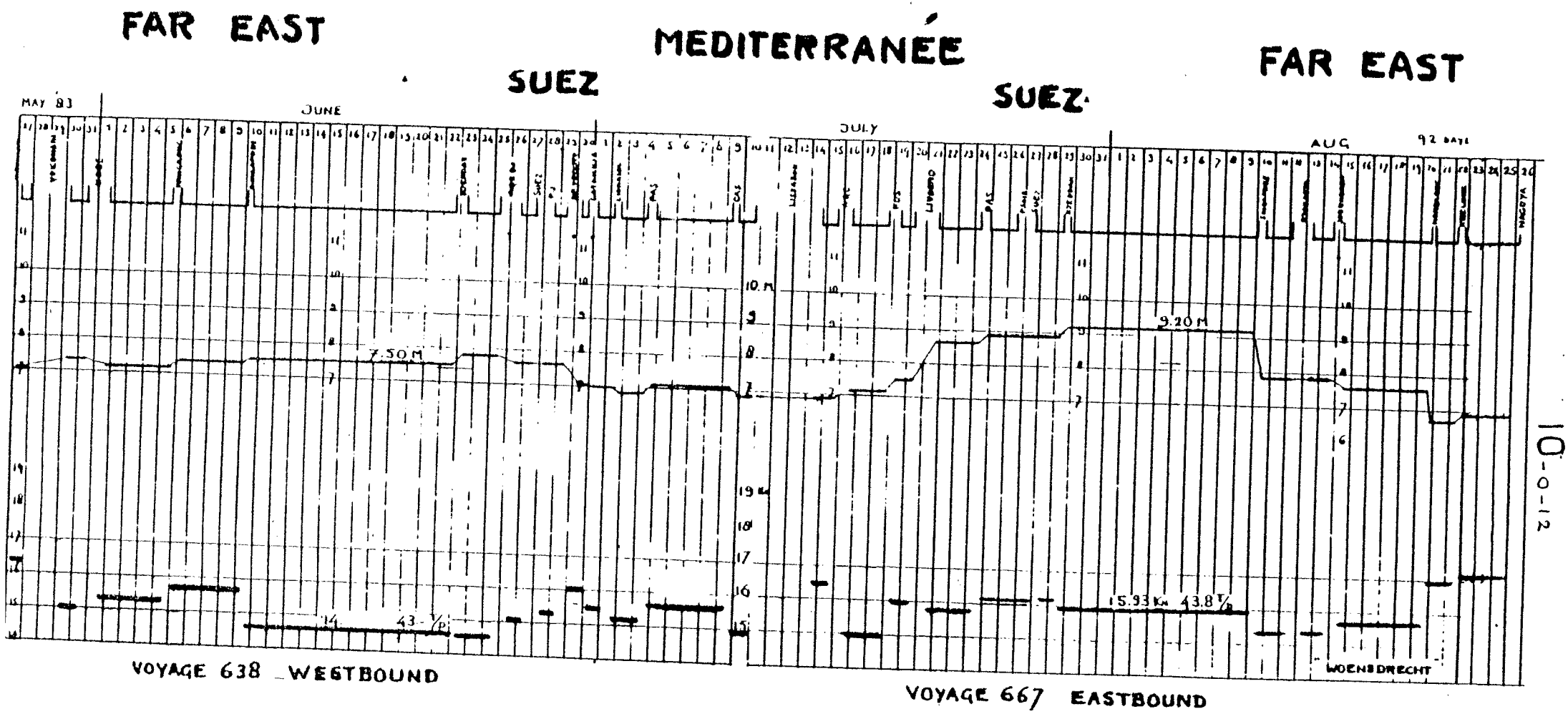
EASTBOUND.

28-8-83

Date

Master

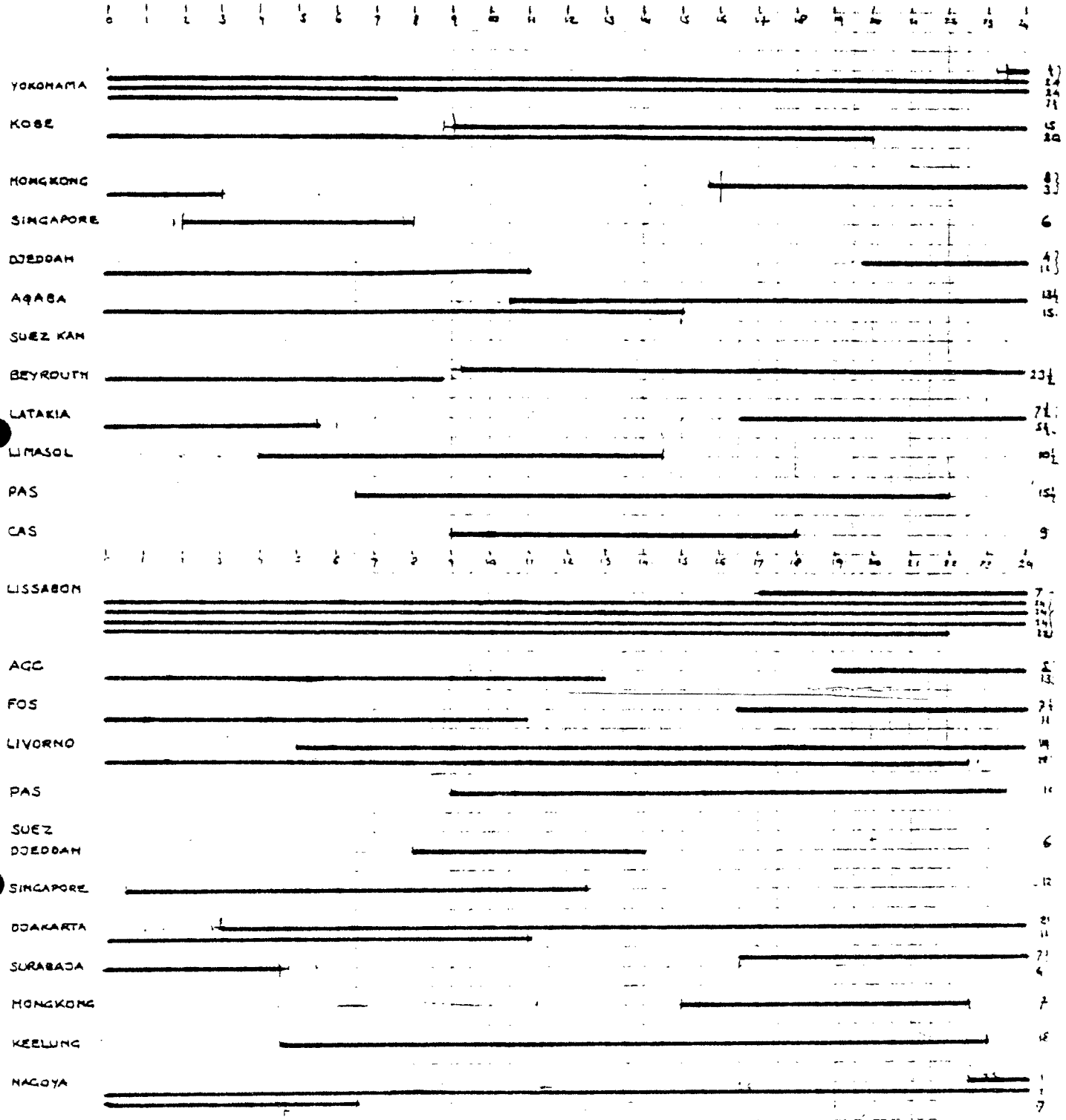
10-0-11



10-0-12

"WOENSDRECHT"

WOENSDRECHT PORT STAY JUNE JULY AUG '83



MV "WAARDRECHT"
MV "WIELDRECHT"
MV "WOENSDRECHT"

42.000 TDW

SELF-SUSTAINED SEMI-CONTAINERSHIP

CONTAINER STOWAGE PLAN

HOLDS

	TEU		40 FT.	+	TEU
no. 1	72	no. 1	36		
2	240	2	120		
3	240	3	120		
4	240	4	120		
5	100	5	50		
	892		446		

HATCHES

2 tiers	362	2 tiers	154	+	54
total	1254		600	+	54
3rd tier	181	3rd tier	77	+	27
total	1435		677	+	81
4th tier	113	4th tier	46	+	21
total	1548		723	+	102

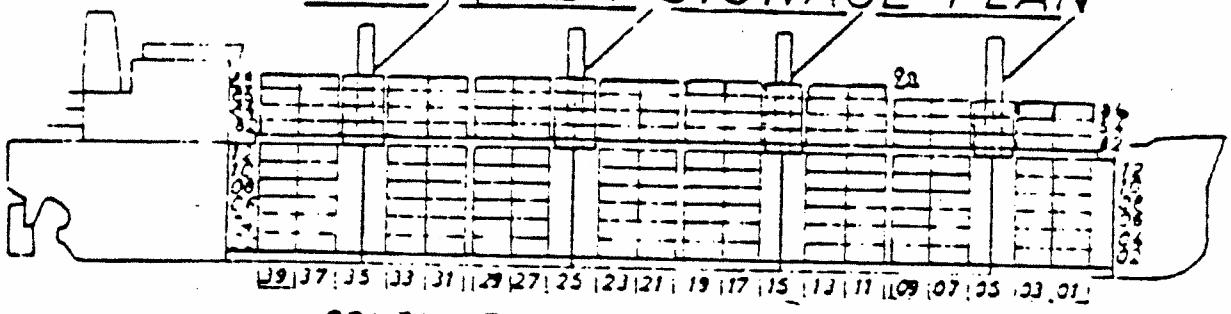
Stackload in holds 120 ton for 20'
180 ton for 40'

Stackload on hatchcovers 44 ton for 20'
66 ton for 40'

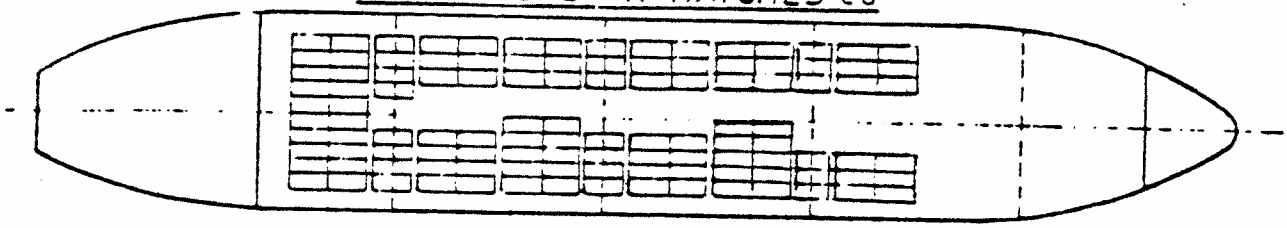
4th tier subject to weight and stability

Reefercontainers (with connection for 11 kw. el.motor)
120 of 20' in first tier on hatch 2, 3, 4 and 5.

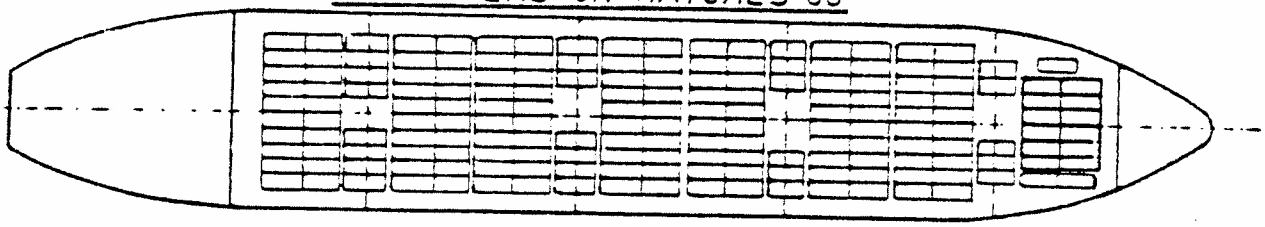
CONTAINER STOWAGE PLAN



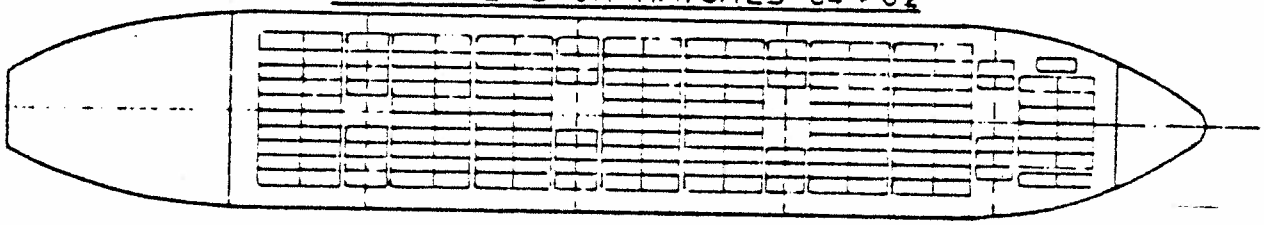
CONTAINERS ON HATCHES 88



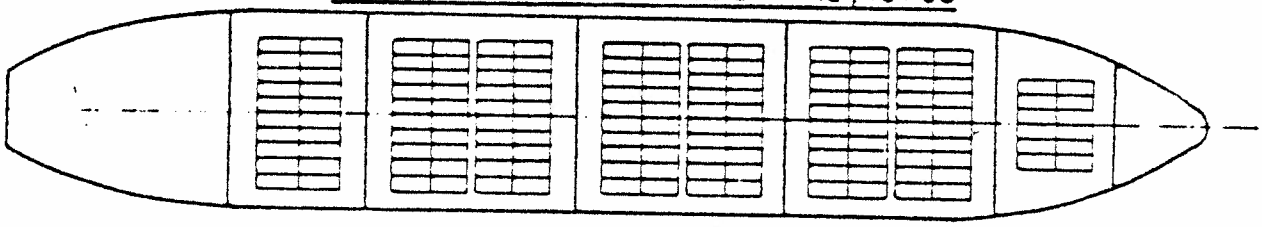
CONTAINERS ON HATCHES 86



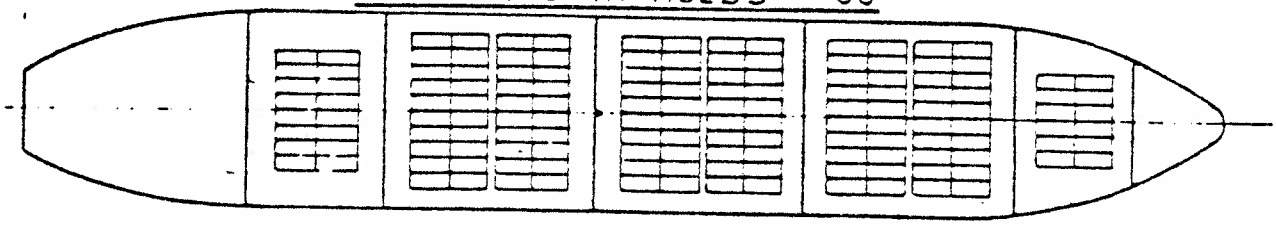
CONTAINERS ON HATCHES 84 + 82



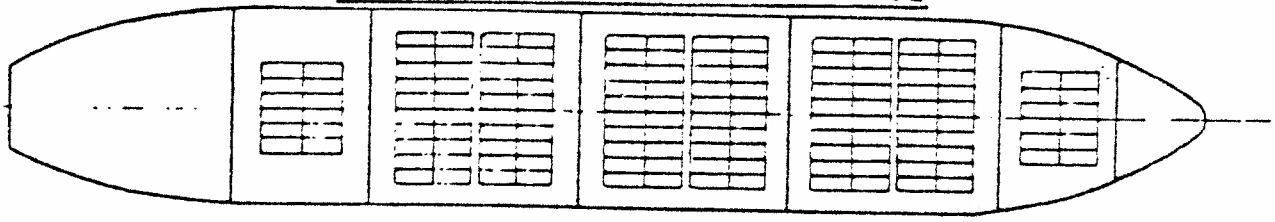
CONTAINERS IN HOLDS 12, 10 + 08



CONTAINERS IN HOLDS 06



CONTAINERS IN HOLDS 04 + 02



MV "WAARDRECHT"
MV "WIELDRECHT"
MV "WOENSDRECHT"
CONTAINER STOWAGE PLAN

NAME	AMSTELVAART	MUDRECHT	WAARDRECHT	SWAN ARROW	HOECH-MASLOT	PROJECT
COMPANY	NEDLL BULK	N. ONNEREN	V. ONNEREN	JEBSEN	HÖEGH	CANADIAN TT.
YARD	YARNA	GÖTAWERKEN	GÖTAWERKEN	MITSUI	KAWASAKI	DESIGN HERBERT
DELIVERY	'81	'78	'82	77	77	83
CLASS	LR	BV ICE	BV ICE	NV	NV	
L_{OA} M	201.56	191.38	186.54	182.00	200.50	199.00
L_{PP}	187.70	185.00	180.00	174.00	190.00	192.00
B MLD	27.80	PAM 39.23	30.00	29.00	30.80	29.76
H. MLD	15.50	16.05	16.80	16.10	15.70	→ 14.80
T SCANTL	11.18	11.33	11.86	11.59	11.57	→ 10.50
δ	0.806	0.783	0.807	0.824	0.826	0.776
DISPL TON	48.200	54.250	53.000	49.407	57.363	47.683
LIGHT SHIP	10.200 (17½)	9.650 (17½)	11.200 (19)	10.794 (17)	12.300 (17)	12.416 (14)
DEADWEIGHT	38.000	44.600	41.800	38.613	45.063	35.267
SUM	1.500	1.500	1.500	1.500	1.500	1.500
CARGO	36.500	43.100	40.300	37.113	43.563	33.767
CUBIC GRAIN	1.626.000 ft³	2.114.380 ft³	1.825.060	1.472.000	1.770.552	1.625.000
	46.000 m³	58.873	51.680	41.800	50.000	46.050
cubft/TON	44½	49	45½	39½	40½	48
CONTES IN HOLD		772	892			1012
" ON DECK		3T 586	656			620
		1358	1548			1632
			116			60
			1664			1692
W.B. CAPACITY	19.000 m³	18.737 m³	16.713 m³			14.000 m³
H.F. OIL	2074 "	2.087 m³	1.685 m³			2050 T
MAIN ENGINE	B+W 8K74	B+W 7K67GA	B+W 4L90 GFCA	B+W 7K74EF	MAN 8KSZ70/125	B+W 5L70MC
H.P./RPM	13700/120	11.900/140	13850/96	13100/124	15200/145	12.250/95
FUEL/DAY	48 T/DAY		40 T/DAY	46.1 T/DAY		39 T/DAY
V	15.1 Kn		15 Kn	14.5 Kn	15.3	15.-
PROP		ø 5400	ø 6800			ø 6800
DIESELGEN		680 KW	725 KW 720 RPM	1000 KW 1000 RPM		830 KW
		680 "	725 "	1000		830
		908 "	775 "	1000		1000
		4 JIBCRANES	4 JIBCRANES	2 GANTRIES	2 GANTRIES	2 GANTRIES
		25T/30M	36T/30M	25T	30T	40T
						EXCL LOWER P.

← OPEN BULK CARRIERS →

FUEL CONSUMPTION IN SERVICE PILOT-PILOT.

STR. BRANDSTOFVERBRUIK VERGELIJKING VAN PRESTATIE GETAL = $\frac{\text{WATERVPL}^{2/3} \times \text{SNELH}^{3}}{\text{BRANDST/ETM.}}$

"AMSTEL WAL"	"MAAS-SLUIS"	WOENSDRECHT	CAN. TRANSPORT LUMBER PROJECT HERBERT DESIGN
VD G. 110	VD G. 120	KOCKUMS 90 RPM	
V = 16.5 KM.	V = 15.4 Km	V = 15,0 Km	V = 15,0 Km
D = 45.000 T	D = 46.400 T	D = 53.000 T	D = 47.683 T.
Ton/DAY = 49.9	Ton/DAY = 41 –	Ton/DAY = 40. –	Ton/DAY = 31,3 ? OPGAVE IN BESTEK
$\frac{D^{2/3} \cdot V^3}{Ton/DAY} = \frac{(45.000)^{2/3} \times 16.5^3}{49.9}$	$\frac{D^{2/3} \cdot V^3}{Ton/DAY} = \frac{(46.400)^{2/3} \times 15.4^3}{41.}$	$\frac{D^{2/3} \cdot V^3}{Ton/DAY} = \frac{(\ldots)^{2/3} \cdot 15^3}{40}$	$\frac{D^{2/3} \cdot V^3}{Ton/DAY} = \frac{(47.683)^{2/3} \cdot 15^3}{31.3}$
= 116½	= 115	= 119 OPGAVE ONBEKEND	= 142 ? too high !
V = 16,1	V = 14,8 Km		realistic = 114 à 118
DEP = 45000.	D = 46.400 T		$\frac{(47.683)^{2/3} \times 15^3}{39 \text{ à } 38}$ = 114 à 119
Ton/DAY = 46,4.	Ton/DAY = 38,2.		
$\frac{D^{2/3} \cdot V^3}{Ton/DAY}$ = 114.	$\frac{D^{2/3} \cdot V^3}{Ton/DAY}$ = 110		~~31,3 Ton/DAY~~ → 39 à 38 Ton/DAY

"AMSTELVAART"	"MAAS KANT"
VARNA 116	BOEL 120
V = 15,1 Km	V = 15,1 Km
D = 48.200 T	D = 38.000 T
Ton/DAY = 48. –	Ton/DAY = 40,6
$\frac{D^{2/3} \cdot V^3}{Ton/DAY}$ = 94	$\frac{D^{2/3} \cdot V^3}{Ton/DAY}$ = 96

SERVICE COEFF (ADMIRALITY) = $\frac{(\text{DISPL IN TONS})^{2/3} \times V^3_{\text{PILOT-PILOT}}}{\text{TON/DAY (HFO MAIN ENG)}}$

10-1-2

70 YEARS DEVELOPMENT 1913-1983

STG 1913

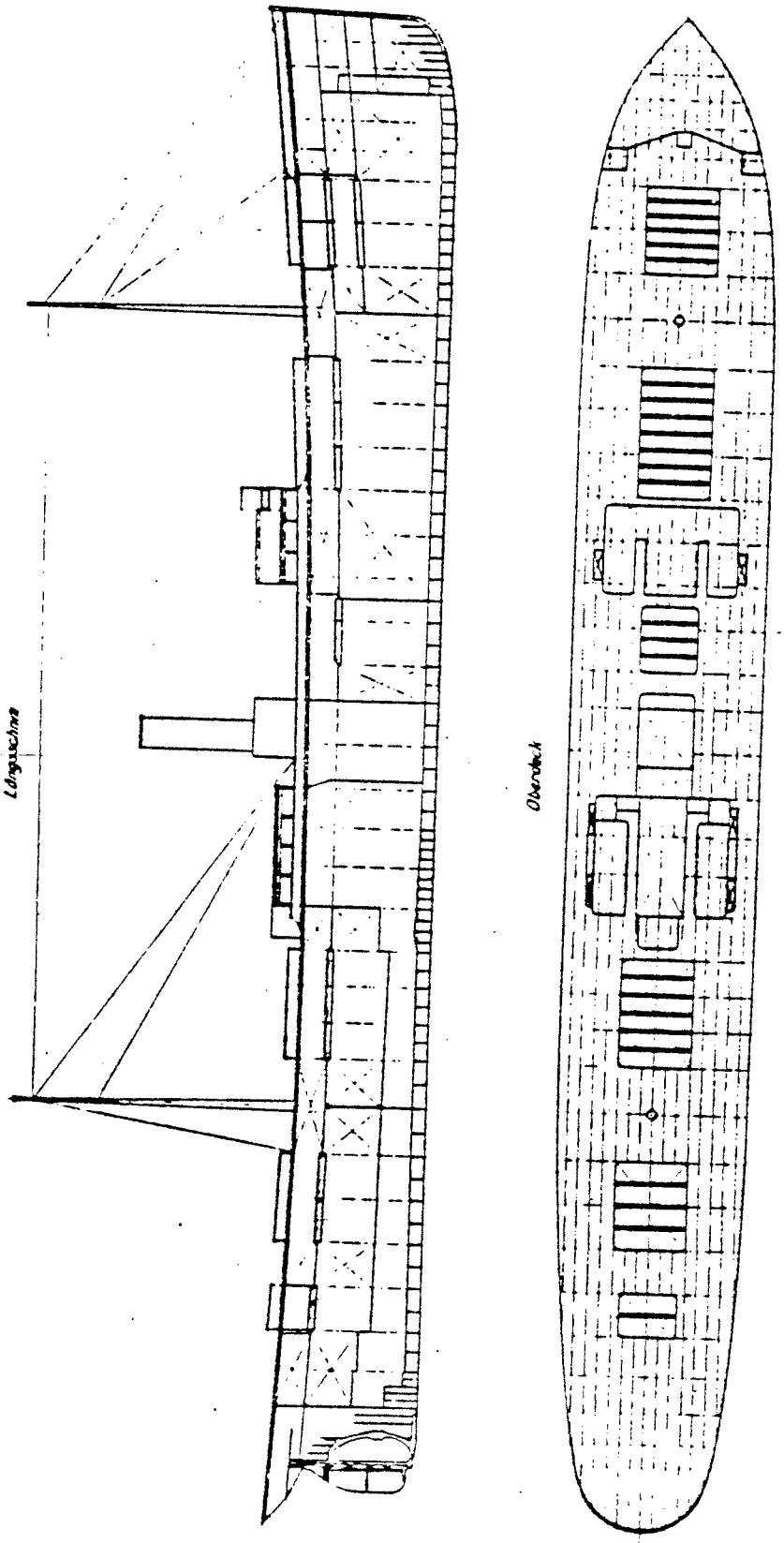
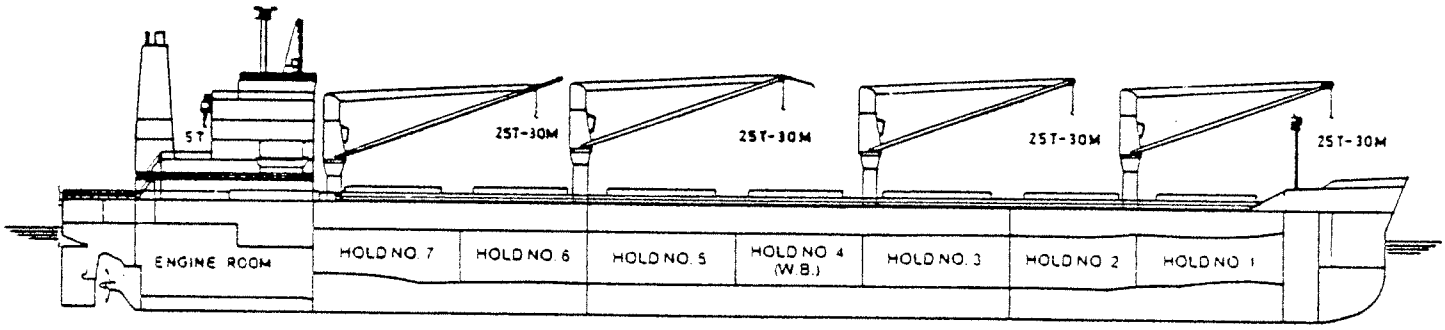


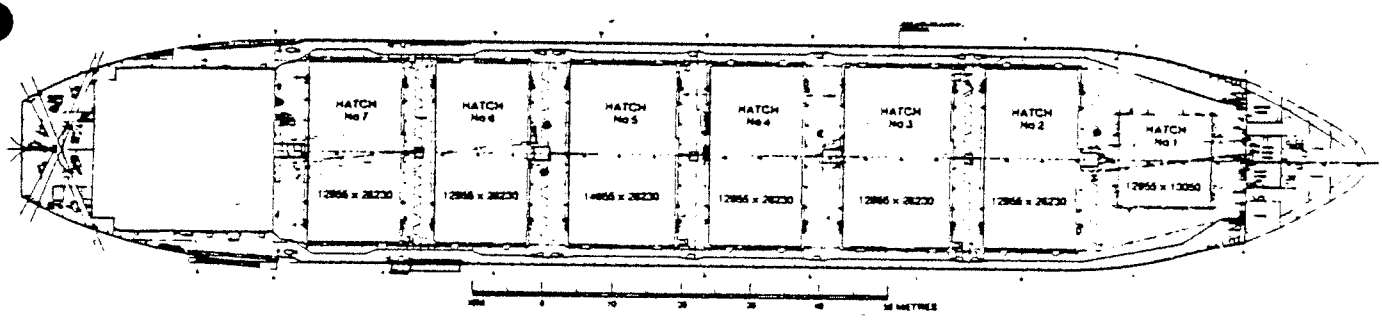
Fig. 2.

10-1-3

COLUMBUS 44 -- CLASS

COntainer LUMber BUlk Ship -- Builder's views.





Particulars

summer deadweight	44,600 mt	43,800 lt
summer draft	11.32 m	37'2"
immersion	53.20 tpcu.m	133 tpi
length overall	191.38 m	627'10"
extreme breadth	32.23 m	105' 9"
total grain capacity	60,000 cu.m	2,155,000 cu.ft
ore cargo in holds 1, 3, 5, 7	35,160 cu.m	1,241,665 cu.ft

speed and consumption

average service speed	15 knots
average service speed as container vessel	16 knots
daily consumption fuel oil 180 cSt	45 tons
daily consumption marine diesel	2.5 tons

cargo gear

Four Hägglund 25 tons electro-hydraulic deck cranes slewing radius 30 metres combinable up to 45 tons.
(For long term contracts lifting capacity could be increased up to 36 tons, in combination up to 65 tons).

hatches

Seven MacGregor wide opening cargo hatches of which six are of one-panel "piggy back" type and one of three-panel folding type.

hatch dimensions (length x width)

	meters	feet
no. 1	12,955 x 13,050	42' 6" x 44' 3"
no. 2, 4, 6, 7	12,955 x 26,230	42' 6" x 86'07"
no. 3, 5	14,955 x 26,230	49'08" x 86'07"

maximum loads

tanktop strength oreholds	19,44 t/m²
tanktop holds 2/6	11,00 t/m²
hatches	2,6 t/m²

stackload containers

tanktop 120 tons (20')	180 tons (40')
hatches 40 tons (20')	60 tons (40')

containers

Fully self-sustaining with a capacity of over 1,350 t units (20' x 8' x 8'6"), twenty and forty feet container spreaders and 36 reefer plugs.

fire-fighting equipment

Cargo holds are CO_2-fitted.

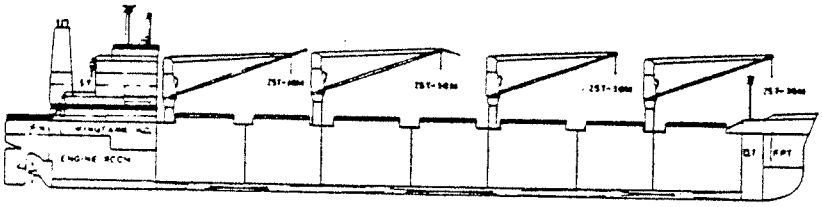
Main features

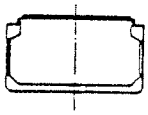
1. Open hatch bulk carrier with virtually box-shaped holds.
2. Low fuel consumption because of B&W concept. Slow running large propeller and cylindrical bow.
3. Very high cubic capacity. 50,5 cu.ft/ton.
4. Very low summer draft allows 44,600 ton at 11.32M.
5. Long and wide hatches suitable for containers, mobile homes, long pipes and structures with large dimensions.
6. Fast 25 tonnes cranes, having an impressive outreach.
7. Length over all and draft allow greater flexibility in trading to restricted ports.

10-1-4

Conventional bulk cargoes

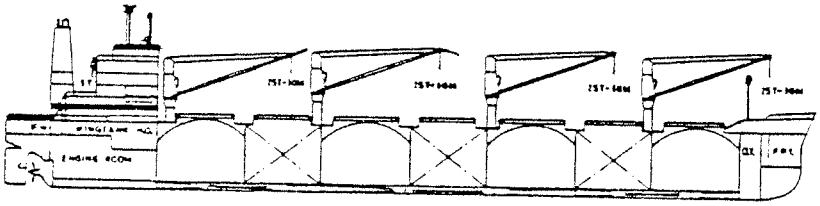
Homogeneous cargo





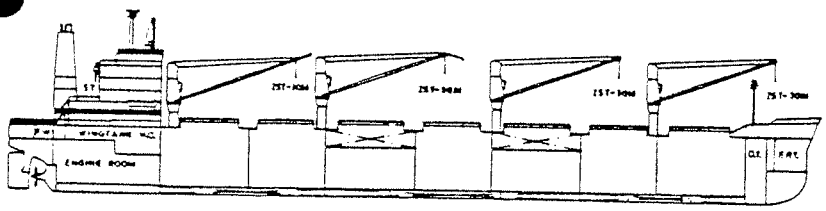
☐ **stowage factor**
with full deadweight
and full bunkers
50.5 cu.ft/ton

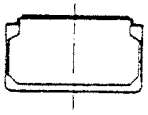
Ore cargo. Alternate holds



☐ **stowage factor**
with full deadweight
and full bunkers
29.0 cu.ft/ton

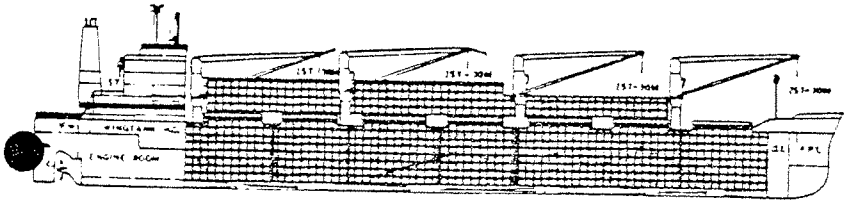
Grain cargo. 2 holds slack

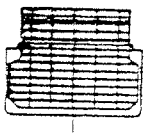




☐ **stowage factor**
with full deadweight
and full bunkers
50.5 cu.ft/ton

Packaged lumber

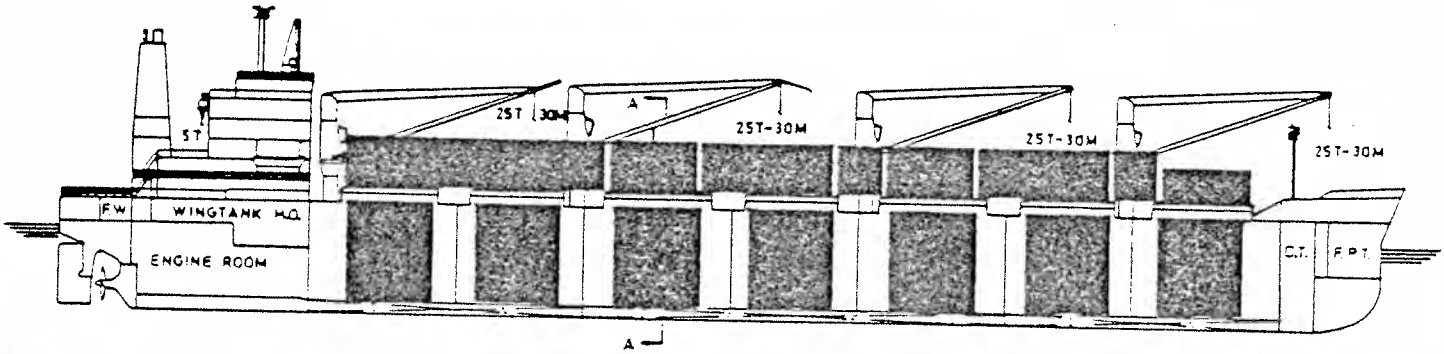


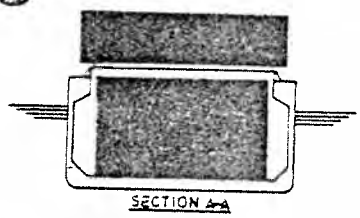


☐ **cubic capacity**
in holds 60,000 cubics metres
on deck 12,500 cubics metres

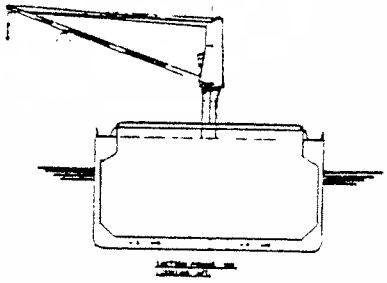
Cargo flexibility

Containers

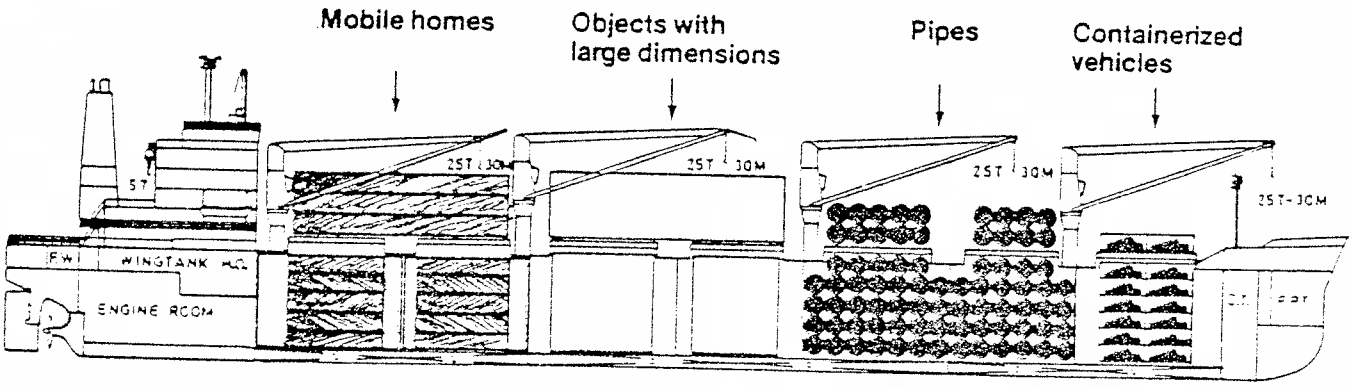




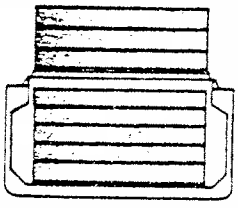
Containers	20′ units	max. 40′ units + balance 20′ units
In holds	772	384 + 4
On deck	586	203 + 170
Total	1358	587 + 174

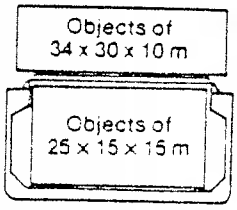


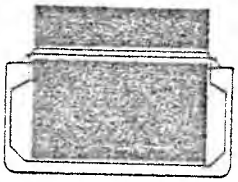
Neo-bulk cargoes



Mobile homes



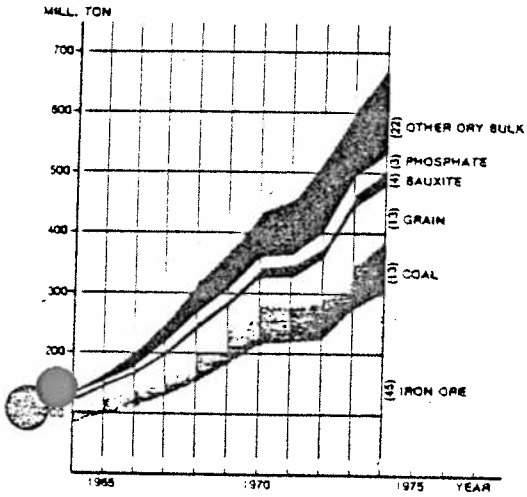




Double-joint pipes up to 25 m

DESIGN PHILOSOPHY

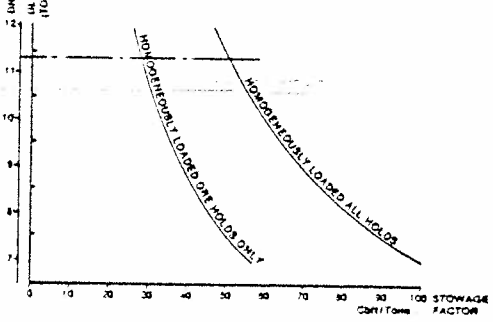
SHIPMENTS BY BULK CARRIERS AND COMBINED CARRIERS 1964—74



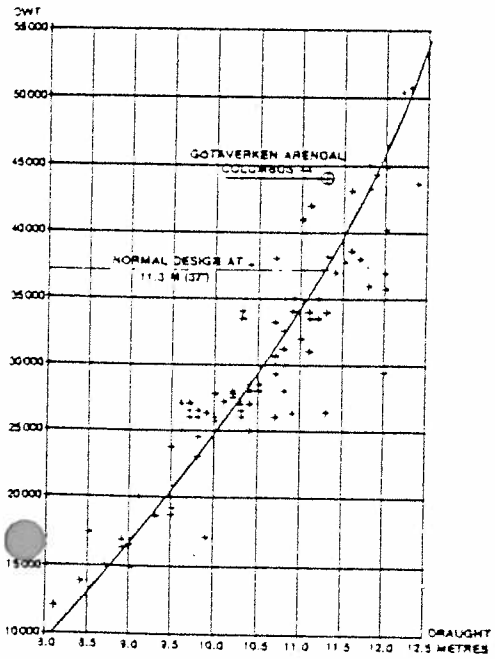
In recent years there has been a marked tendency to introduce bulk carriers into new trades — the minor bulks representing a variety of raw materials and semi-manufactured goods are increasingly shipped in small and medium sized bulk carriers. Of course the reasoning behind this lies in the economy which can be achieved by rationalizing and simplifying the sea-borne transportation of these various bulk cargoes.

In 1974 the minor bulks amounted to 22% of the total bulk shipment at sea. The need for handy-sized tonnage specially designed to increase the transport economy in the minor bulk trades is evident.

DEADWEIGHT AND DRAUGHT WHEN THE SHIP IS FULLY LOADED WITH CARGOES WITH DIFFERENT STOWAGE FACTORS



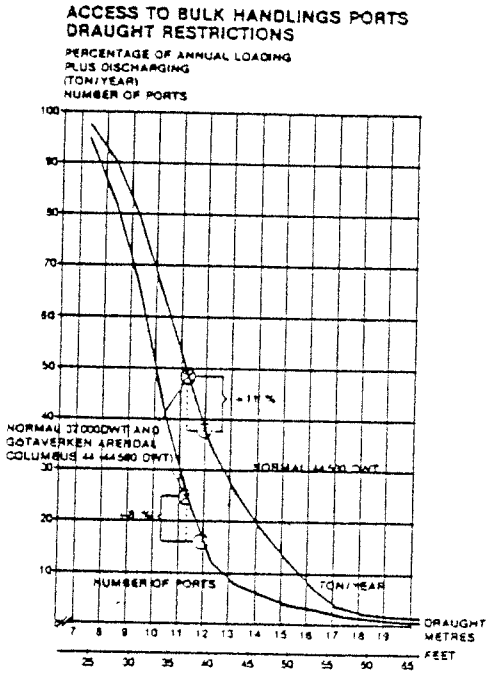
W./DRAUGHT FOR BULK CARRIERS ON ORDER
NMAX AND ST LAWRENCE EXCLUDED



Götaverken Arendals new bulk carrier was designed taking into consideration the emerging need for minor bulk freights. As a very versatile vessel incorporating all standard features of a multipurpose bulk carrier this 44 500 dwt vessel is ideally suited to carry a wide spectrum of dry cargoes like iron ore, coal, grain, unitized cargo, iron slabs, pipes and various other types of raw materials and semi-manufactured goods. Special cargoes like containers and lumber can be carried with the instalment of minor additional stowing equipment. The necessary stability and suitable dimensions are already incorporated in the basic design of the vessel. The very generous cargo volume gives the ship ability to carry very light cargoes at full deadweight. At a draught corresponding to the summer freeboard the stowage factor for a homogeneous cargo is as high as 50 cubic feet per long ton.

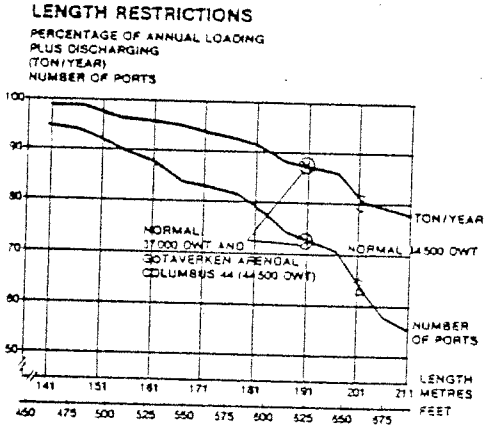
Attention has been given to obtain a high deadweight relatively to the draught. From the deadweight against draught curve shown opposite, it is obvious that Columbus 44 has a considerable deadweight.
The curve is based upon statistical evaluations from existing bulk carrier designs. Hence, with a normal design the Columbus 44 would on its restricted draught have had a deadweight of 37 000 tons instead of its actual 44 500 tons.

Extensive analyses of maximum permissible ship dimensions in bulk ports have been carried out at Götaverken Arendal. These included about 1 400 different berths covering more than 90% of the world bulk cargo turnover. The result combined with series of model tests at the model basin of SSPA have given the background for the decision of the main dimensions of Götaverken Arendals new multipurpose bulk carrier.

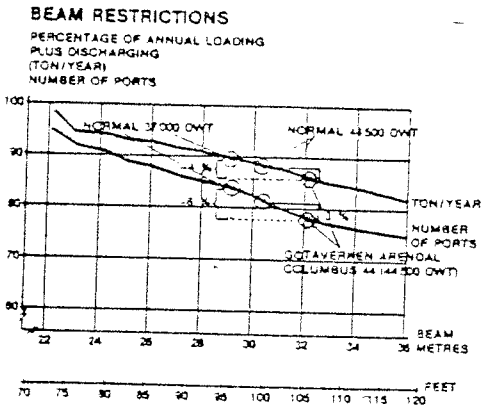


The intention with the analyses was to find dimensions which would permit the ship to call at more ports than the conventional designs with the same carrying capacity. When comparing the dimensions with the conventional 44 500 dwt ship you will find that Columbus 44 is shorter, wider and has a shallower draught.

The effect of length, beam and draught restrictions in the bulk ports of the world is illustrated in the diagrams. Columbus 44 and conventional bulk designs are indicated in the diagrams for comparison.
For example Columbus 44 can due to the shallow draught call at 8% more ports than the conventional 44 500 dwt bulk carrier. This corresponds to 11% of total turnover of the world bulk trade equal to 100 million tons of cargo or about the same as the total seaborne coal trade 1973.

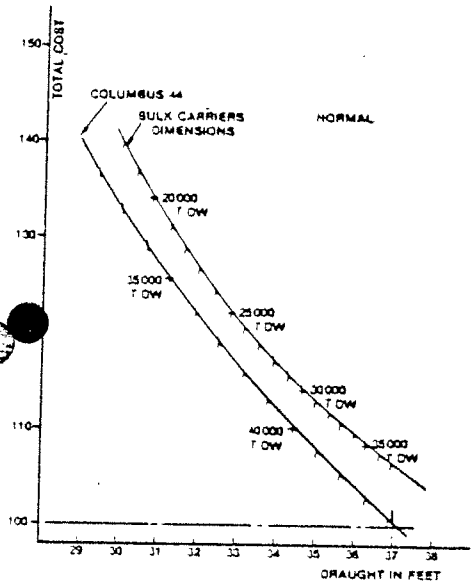


Compared to a conventional 37 000 dwt bulk carrier Columbus 44 has the same length and draught, but a wider beam. This results in a 20% increase in pay load when trading on a draught of 37'. The wide beam of the vessel means however according to the beam restrictions in the diagram a cut of 4% of the port accessibility. This drawback is unimportant when compared to the economic benefits from the extra cargo carrying capacity of the Götaverken Arendal ship.



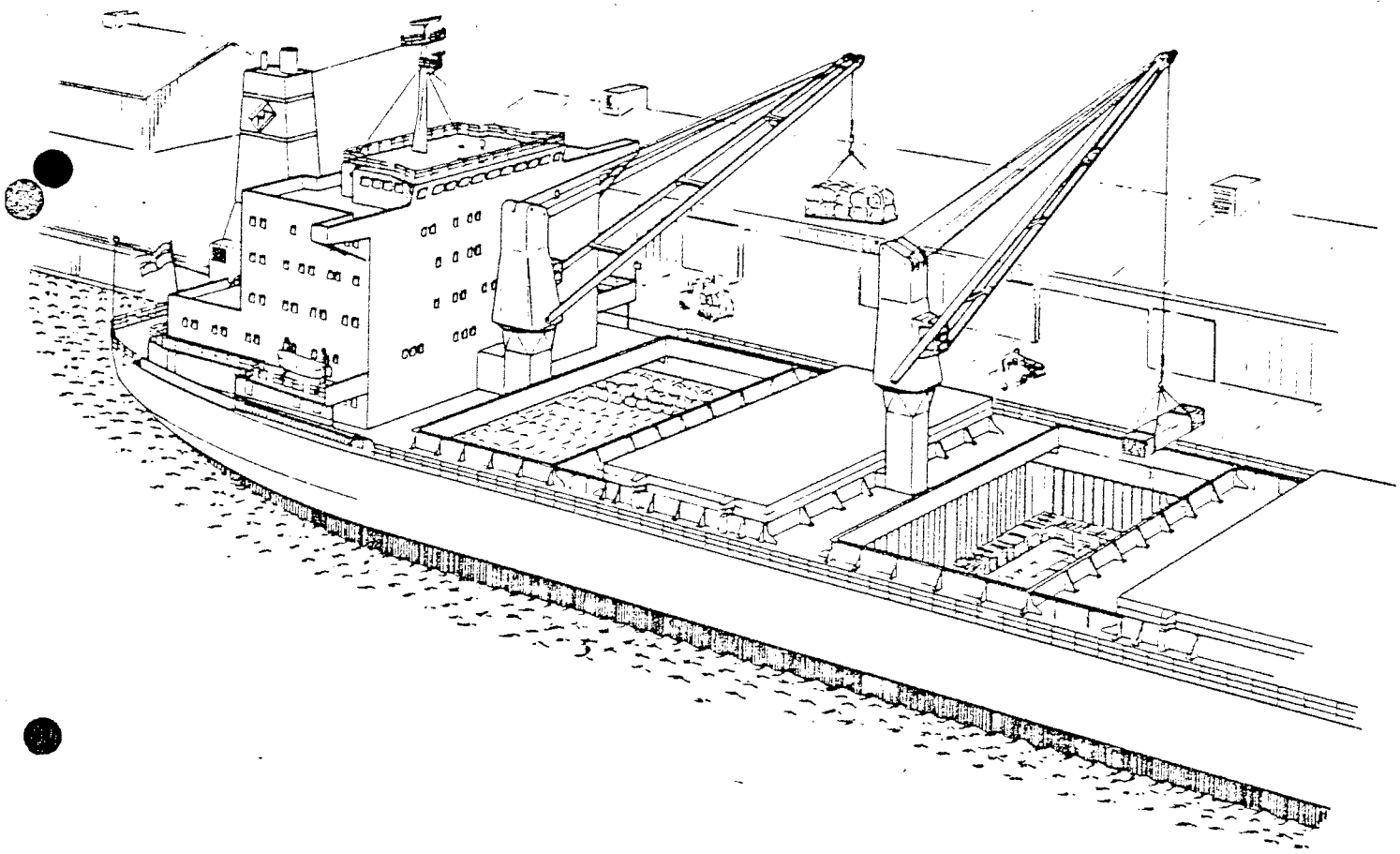
Freight cost analyses have been made for Columbus 44 at "full deadweight" and part loaded. These are

TRANSPORT COST AT DIFFERENT DRAUGHTS FOR GÖTAVERKEN ARENDAL COLUMBUS 44 COMPARED W. CONVENTIONAL BULK CARRIERS ON FULL DRAUGHTS



compared with cost analyses for normal designs at full deadweight and at corresponding draughts. The graph shows that as long as the draught restrictions are the factors that limit the cargo intake at least a 5% saving of the freight cost per carried ton is achieved with the Götaverken Arendal multipurpose bulk design of 44 500 dwt.

Thanks to the thoroughly balanced main dimensions this bulk carrier has access to a greater part of the bulk turnover than conventional designs of the same size giving her owner a competitive tool in his hand.



DATA

Metric units, unless otherwise stated

DIMENSIONS

Length o.a.	191.40 m (627'—11 7/16")
Length p.p.	185.00 m (606'—11 7/16")
Breadth, moulded	32.23 m (105'—8 7/8")
Depth, moulded	16.05 m (52'—7 7/8")
Draught, on summer freeboard	11.33 m (37'—1 15/16")
Ton per cm immersion in SW at 11.33 m draught	53.2
International tonnage	abt. 25 000 GRT

RULES AND REGULATIONS

Lloyd's Register of Shipping ✠ 100 A1, ✠ LMC, UMS, Strengtened for heavy cargoes. Holds 2, 4, 6 may be empty, additional notation i b (+).
British Department of Trade (DOT).
The Norwegian Maritime Directorat.
International Convention for the safety of Life at Sea 1960 (Solas 1960).
International Load Line Convention 1966.
International Regulations for Preventing Collision at Sea 1960.
International Telecommunications Union — radio regulations, 1967 (ITU — Geneva).
Suez Canal Navigation and Tonnage, Rules and regulations as in force 1967.
Panama Canal Navigation, Rules and Regulations in force 1967.

APPROXIMATE CAPACITIES

Grain cargo in all holds	60 000 m^3 (2 115 000 ft^3)
Ore cargo in holds Nos. 1, 3, 5 and 7	35 160 m^3 (1 241 665 ft^3)
Heavy fuel oil tanks	1 900 tonnes at 0.94 tonnes/m^3
Diesel oil tanks	290 tonnes at 0.86 tonnes/m^3
Lubrication oil tanks	39 tonnes at 0.90 tonnes/m^3
Cylinder oil tanks	19 tonnes at 0.90 tonnes/m^3
(All oil tanks 98% full)	
Fresh water tanks	220 tonnes at 1.00 tonnes/m^3
Permanent water ballast tanks	10 700 tonnes at 1.025 tonnes/m^3
Water ballast in No. 4 hold	8 460 tonnes at 1.025 tonnes/m^3

Stowage factor for homogeneously distributed grain cargo based upon actual deadweight with 1 800 tonnes allowances for bunkers, stores etc. is 1.41 m^3/tonnes (50.5 ft^3/tons).
Stowage factor for ore cargo in alternate holds is correspondingly 0.82 m^3/tonnes (29.0 ft^3/1 tons).

SPEED, FUEL CONSUMPTION, CRUISING RANGE

Trial speed, fully loaded	15.7 knots at MCR
Average service speed	abt. 15.0 knots at CSR
Heavy fuel oil consumption of main engine	abt. 46 tonnes/day at CSR*
Cruising range	abt. 13 800 nautical miles (3 days consumption in reserve)

*Low heating value for heavy fuel oil 40 600 kj/kg (9 700 kcal/kg)

DECK EQUIPMENT

Seven wide opening hydraulically operated cargo hatches of which six are of one panel "Piggy-back" type and one of 3 panel folding type.
Four 25 tons electro-hydraulic deck cranes.
One 5 tons trolley crane for handling provision and machinery parts.
Four 20 tons and one 16 tons electro-hydraulic mooring winches with automation for self-tensioning and self-rendering.
Two of the 20 tons winches to be combined with windlasses.
Life saving equipment according to the requirements of the specified authorities.

ACCOMMODATION

Single-berth cabins with private shower and WC for:
Master, three officers, radio officer, chief engineer, three engineers, steward, electrician and 17 other members of the crew including petty officers, cook and boys.
Double-berth cabins to accommodate four repairers.
Cabin for Suez crew.

Cabins for owner and pilot and one as spare.
Total 33 cabins.
Other rooms as follows:
Messroom, day- and recreation rooms, hobby-rooms, sauna and offices.
Swimming-pool and recreation area.
All bulkheads are of sound insulated type. Ceilings are of sound insulated and sound absorbing type. Decks in accommodation are sound insulated by using Arendal's "floating" type cover.
Air conditioning by a high pressure double pipe system.
Separate single pipe system for the galley, sanitary spares, and provision store.

PAINTING

Shotblasting to Sa 2—2.5 (Swedish Standard) of all steel plates thicker than -5.5 mm and all steel profiles.
Ferric oxide shopprimer treatment on all steel surfaces before coating with paint (excluding floatcoat, tectyl or equal).
Summary of painting plan:
Bottom shell and boottop area: CR* primer to 225 microns, CR antifouling to 50 microns on flat bottom, CR antifouling longlife to 50 microns on sides.
Topsides and deck: CR primer to 150 microns, CR finish to 50 microns.
Cargo holds, inside, hatches and hatch coamings, under deck and down to a level at underside top side tanks: Vinyl-tar to 200 microns.
Permanent water ballast tanks: Special tar epoxy with high contents of coal tar 200 microns.

*CR = Chlorinated rubber

MACHINERY

One Burmeister & Wain 7 cylinder K67GF crosshead, direct reversible, two-stroke, turbo-charged diesel engine with a continuous service rating (CSR) of 8.75 MW brake (11 900 BHP) at 140 r/m and a maximum continuous rating (MCR) of 9.65 MW brake (13 000 BHP) at 145 r/m.
Propeller of fixed pitch type in Ni Al-Bronze. One combined exhaust gas/oil fired donkey boiler. Capacity exhaust gas fired: 2 600 kg/h. Capacity oil fired 1 500 kg/h. Saturated steam at 0.7 MPa (1 ata). Three diesel engines for main generators, two rated 680 kW and one 900 kW at 720 r/m.
One diesel engine for emergency generator rated 80 kW at 1 800 r/m.
Two sea cooling water systems, one for main engine lubricating oil coolers, air coolers and fresh water coolers and one for cooling of auxiliaries.
One fresh water generator with a capacity of 25 000 kg/24 h.
One sewage treatment plant with a capacity of 20 m^3/24 h corresponding to 50 persons.
One incinerator for burning of sludge and refuse.
Two water ballast pumps, each with a capacity of 1 500 m^3/h at 20 m LC.

ELECTRICAL SYSTEM

One main switchboard and three diesel engines for electric generating of a total power of 2.2 MW.
Each diesel is provided with automatic stand-by starting and connection to the main switchboard in case of black-out.

CONTROL SYSTEM

A central control station is provided in an air conditioned control room in the engine room.
The control room also comprises the main switchboard and a control console.
From the control station it is possible to operate the main engine and the auxiliary machinery as well as having complete supervision of the machinery as required for unmanned engine room spaces. Essential parameters are digitally displayed.

COMPARISON BETWEEN

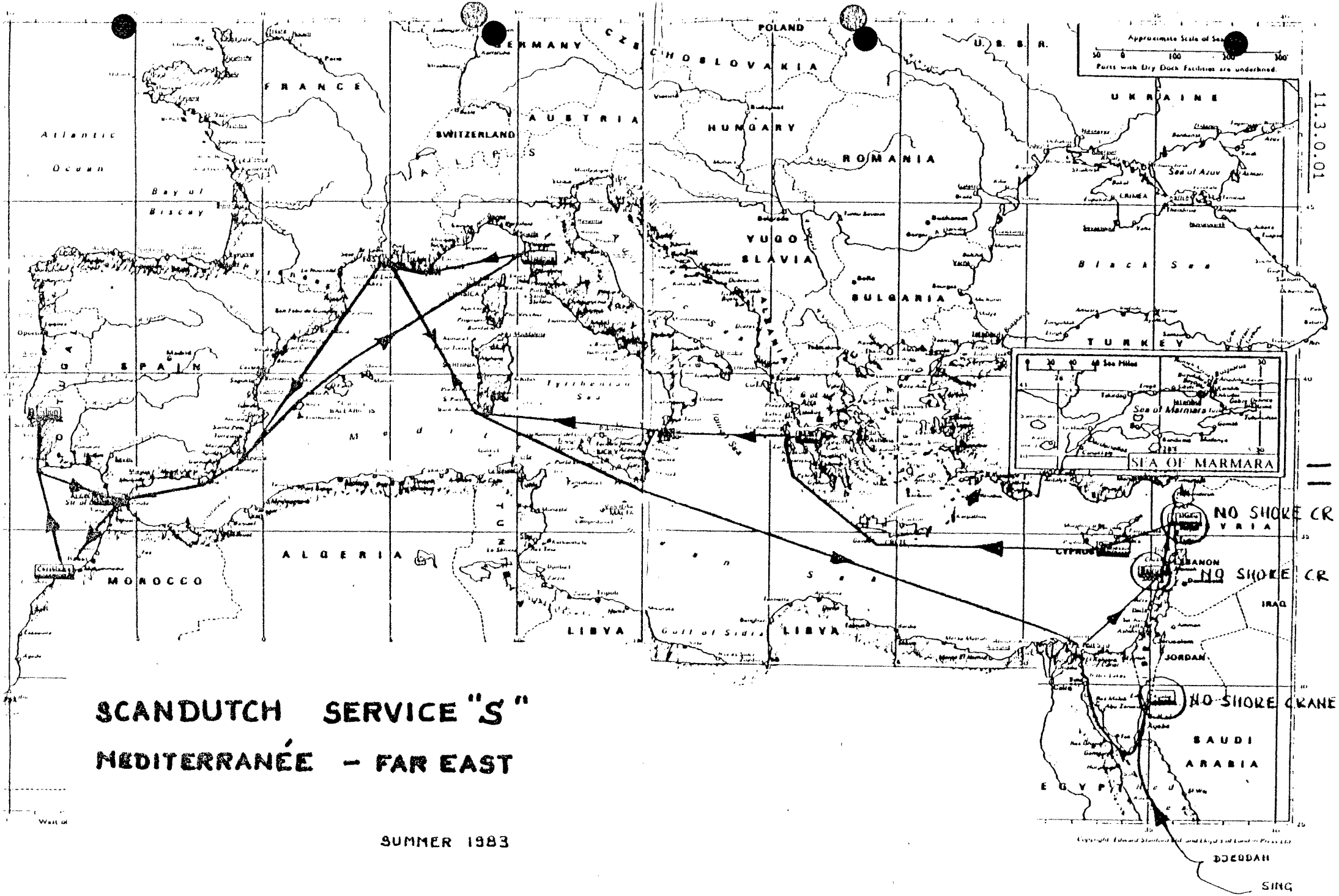
CONCORDIA
NEDLL. v. NOORT
WOENSDRECHT

SCANDUTCH SERVICE "S"

MEDITERRANÉE — FAR EAST

SEP '83

64



SCANDUTCH SERVICE "S"

MEDITERRANÉE – FAR EAST

SUMMER 1983

"CONCORDIA" MAY JUNE JULY '83

	KOBE			
	NAGOYA	8 HR		
	YOKOHAMA	64		
	KOBE	32		
	HONGKONG	23		
	SINGAPORE	7		
	DJEDDAH	14½		MOVES
NO GANTRY	AQABA	24½		2x 150 BOX
	SUEZ	8½		
NO —	BEYROUTH	45		2x 150 BOX
NO —	LATTAKIA	28		2x 200 BOX
	LIMASOL	12		
	PAS. PIRAEUS	35		
	FOS. MARS.	21½		
	CASABLANCA	24		
	LISSABON	78		
	ALGECIRAS	5½		
	LIVORNO (LEGHORN)	15		
	FOS	16½		
	P. SAID	17½		
	DJEDDAH	17½		
	SINGAPORE	7½		
	DJAKARTA	29½		
NO —	SURABAJA	14		2x 100
	HONGKONG	5		
	NAGOYA	25		

IN PORT —— $\frac{552 \text{ HR}}{24} = 23$ DAYS / 24 PORTS

AT SEA WB. 665 HRS — 11.378
EB 701½ — 12.069
1366½ = 1366½ = 57 DAYS AT SEA — 23.447 ≈ 17,15 KN

MAN. ANCHOR 94 / MAN. 117 = 211 = 9 DAYS MAN.
1577½

LOST HOURS ? = 56½ = 2 ?

ROUNDTRIP 91 x 24 = 2186 = 91 DAYS

"WOENSDRECHT" JUNE JULY AUG '83

	YOKOHAMA	56 HR		YON	YES
	KOBE	35		KOY	YES
	HONGKONG	11		HKG	YES
	SINGAPORE	6		SIN	YES
	DJEDDAH	15		JED	YES
—	AQABA	28½	$\frac{300}{24 - 28 \text{ HR}} = 12 \div 10$ MOVES/HR	AQJ	NO MOBILE CRANES 3x45T 3x20T 1x70T. PERMANENT CONTR. BERTH UNDER CONSTR. 2 CONTR GANTRIES 1985
—	BEYROUTH	23½	$\frac{300}{45 - 73\frac{1}{2} \text{ HR}} = 7 - 13$ MOVES/HR	BEY	NO MOBILE CRANE OF INSUFFICIENT OUTREACH CONTR. GANTRIES PLANNED ONLY
—	LATTAKIA	13	$\frac{400}{20 - 1\frac{1}{2} \text{ HR}} = 14 - 31$ MOVES/HR	LAT	NO FLOATING CRANE 32T
	LIMASOL	10		LMS	(YES) 4x45160T 20x 3-16T 1x35T CONSTRUCTION OF SPEC. CONTR. BERTH IS UNDERWAY NEW CONTR. TERMINAL READY 1983
	PAS	15		PAS	YES
				FOS	YES
	CASABLANCA	9		CAS	YES — CONTR GANTRY SOMETIMES NOT AVAILABLE
	LISSABON	101		LIS	YES — POOR PERFORMANCE
	ALGECIRAS	18		AGC	YES
	FOS.	18½			
	LIVORNO	41½		LIV	YES — CONTAINER GANTRY SOMETIMES NOT AVAILABLE
	PAS. PIRAEUS	14½		PAS	YES
	DJEDDAH	6			
	SINGAPORE	12		SIN	YES
	DJAKARTA	32		DKT	YES? MOBILE CRANES 1x16T 1x 2x10T 8x 2x6T
—	SURABAJA	12	$\frac{200}{14} = 14 - 15$ MOVES/HR	SUB	NO FLOATING CRANE 50T
	HONGKONG	7½		HKG	YES
	KEELUNG	18½		KEE	YES
	NAGOYA	32½		NGO	YES

$\frac{539}{24} = 22\frac{1}{2}$ DAYS / 23 PORTS ~ 1 DAY PER PORT!

AT SEA WB 746 HRS — 11.428
EB 745 — 11.880
1491 = 1491 = 62 DAYS AT SEA — 23.308 ≈ 15,65 KN

ANCHOR 66 / MAN. 75 = 141 = 6 DAYS
1632

LOST HOURS = 15 = ½

91 x 24 = 2186 = 91 DAYS

SHORE CONTAINER CRANE GANTRY

— 4 PORTS LACKING SHORE C. CRANE ON 22 CALLS

1. ROUNDTRIP
MEDIT — FAR EAST
3 MONTHS

SEE CI

Comparing Ship Designs.

Concordia :

- Well designed containervessel, used to her maximum. Good hull form, rather fine : blockcoefficient 062-063. Low fuel consumption : 17,15 kn - 39 T/day incl. shaft-generator.

- However, beam is too small for larger carrying capacity.

- Sailing with heavy containers (14,2 ton p.teu) Eastbound, homogeneously loaded.
  The vessel needs a large amount of waterballast to give her sufficient stability (KM = abt 12,5 m)
  1125 Teu x 14,2 Ton is about her maximum Eastbound
  1228 Teu x 9,5 Ton is about het maximum Westbound.

- The cellguides are rather weak because of the removability and sometimes deformation and "cell-jamming" did occur.

- The Concordia and Corona got a forecastle which has improved seaworthiness in comparison to Balandra, Barbarossa (Nedlloyd Van Diemen) which ships did not have the fo'cstle.

Woensdrecht:

- This ship is designed as bulkcarrier and therefore her blockcoëfficient is rather large : 0,78 - 0,80. Because of her fat underwaterbody she is floating as a bubble on the water if loaded with a small cargoweight.

- To improve seaworthiness and avoid propeller racing and slamming in a rough sea, she needs large amounts of waterballast to get the hull sufficiently deep in the water... (Westbound condition)

- Moreover she also needs waterballast for improving stability because of the fact that the metacenter of the fullbody vessel is rather low; KM = 12,5 m. (Eastbound condition).

- Analysing the service results the conclusion is that for the relatively small container weight and large amount of waterballast Woensdrecht has a rather high fuel consumption, 15,65 kn - 44 T/day, excl. auxiliaries WHICH RUN ON D.O.

- Eastbound : light ship = 11200 T
  cons + WB = 10807 T
  cargo = 18293 T = 1163 Teu x 15,7 Ton
  Draft = 9,2 m - displ. = 40300 T

  Westbound : light ship = 11200 T
  cons + WB = 11311 T
  cargo = 9889 T = 971 Teu x 9,9 T + 240 T BREAKB.
  Draft = 7,5 m - displ. = 32400 T

  Conclusion : The vessel is too blunt for the container service.

NI Van Noort : - This vessel is designed with a wide beam (30,5 m) and a rather fine hull: BLOCKCOEFF. = 0.62-0.64

- The metacenter above keel KM = 14,3 - 14,4 m is rather high and allows for relatively large deckloads.

- Her most economical draft lies between 8,5 and 10 meter. Above 10 meter (> 20.000 ton cargo) she needs extra waterballast and will require an increased amount of fuel oil (V shaped hull).

- For a larger amount of heavy containers, homogeneously loaded, a larger vessel is more suitable. ("Nedlloyd-Clarence" type): B=32.24 M

NEDLLOYD V. NOORT ROUNDTRIP

16.5 KN 34,0 + 2.9 = 36.9 T/DAY
M.E SHAFTG.

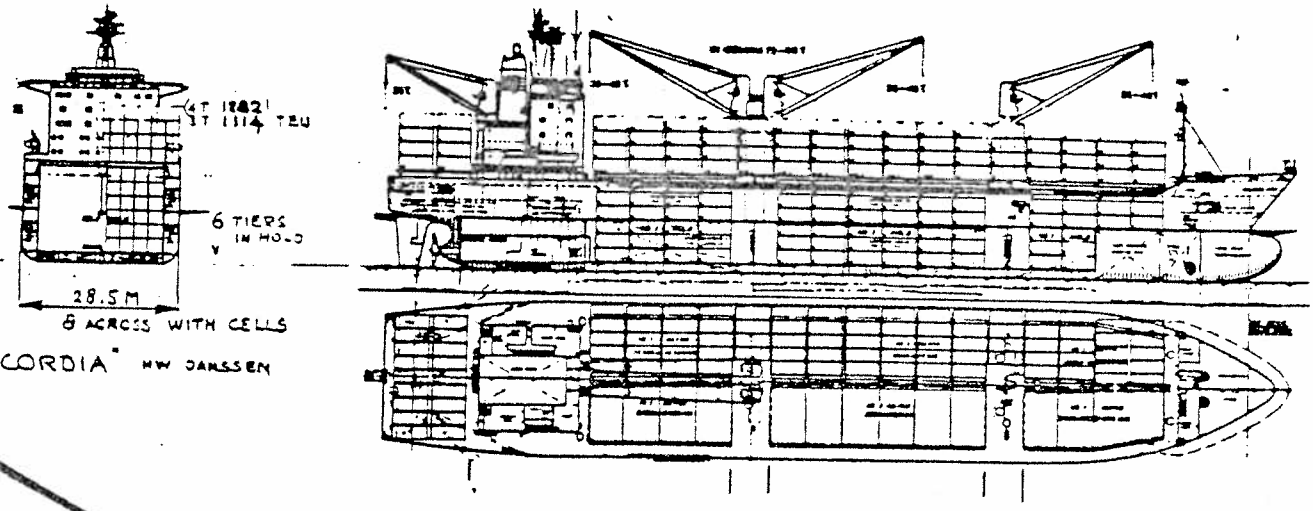
Optimal loading conditions:

Eastbound	Light ship	= 11000 T	
	Cons + WB	= 3438	
	Cargo	= 20249	= 1426 Teu x 14,2 T/Teu
Draft = 9,94 m -	Displ.	34687 T	GM = 0,46 m.

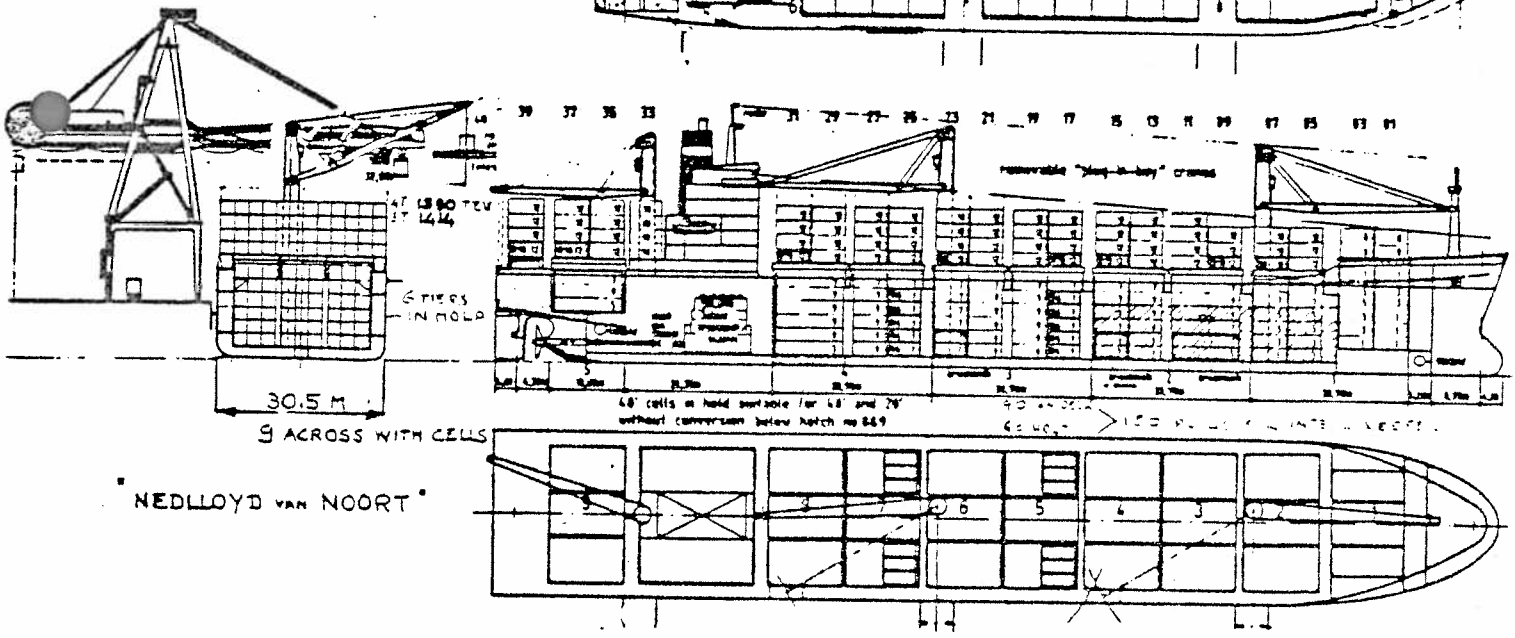
Westbound	Light Ship	= 11000 T	
	Cons + WB	= 4278	
	Cargo	= 15105	= 1590 Teu x 9,5 T/Teu
Draft = 8,95 m -	Displ.	30383 T	GM = 0,58 m.

GV 20-9-83

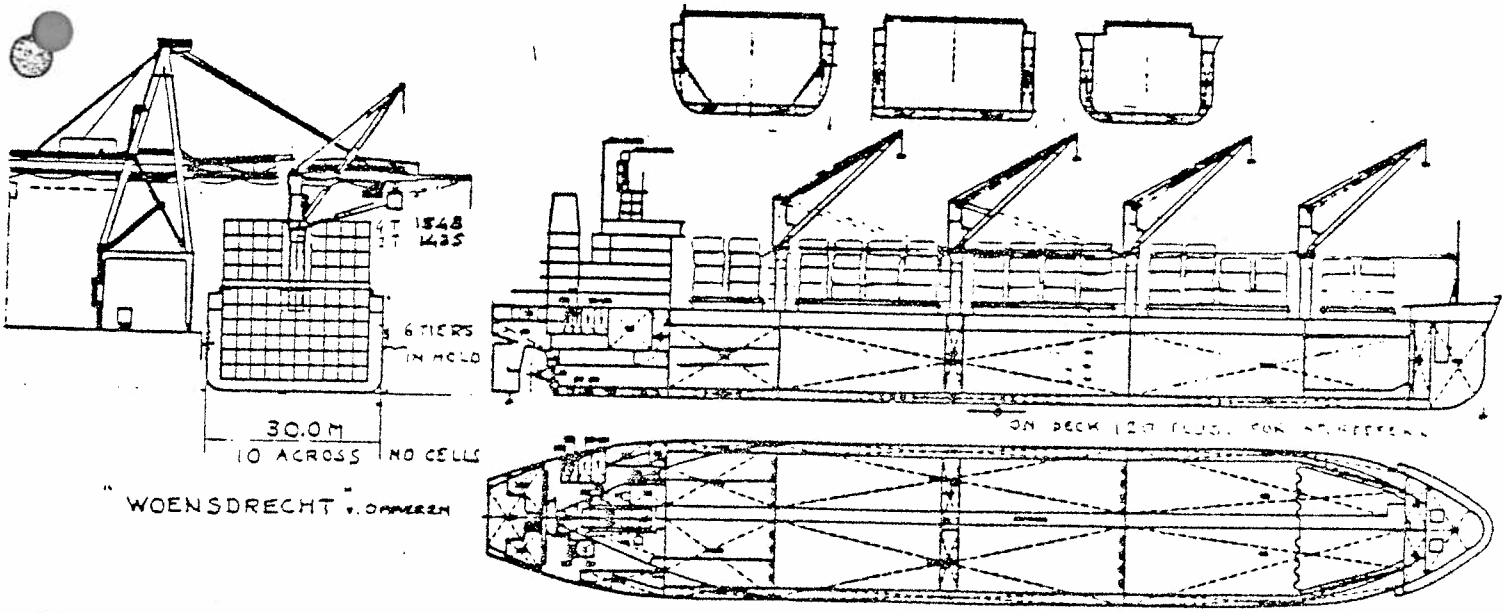
SCANDUTCH MEDIT-FAR EAST SERVICE 3 MONTHS ROUNDTRIP



"CONCORDIA" HW DANSSEN



"NEDLLOYD VAN NOORT"

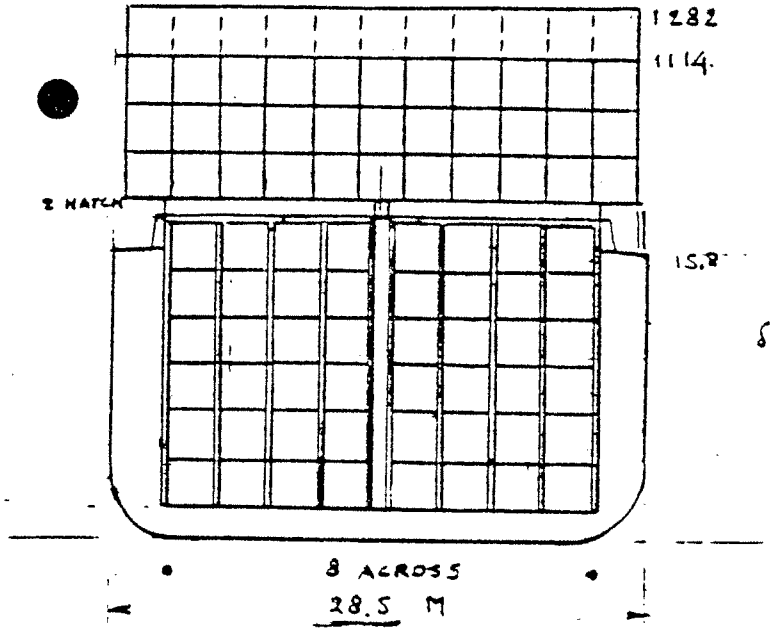


"WOENSDRECHT" V. DAMMEN

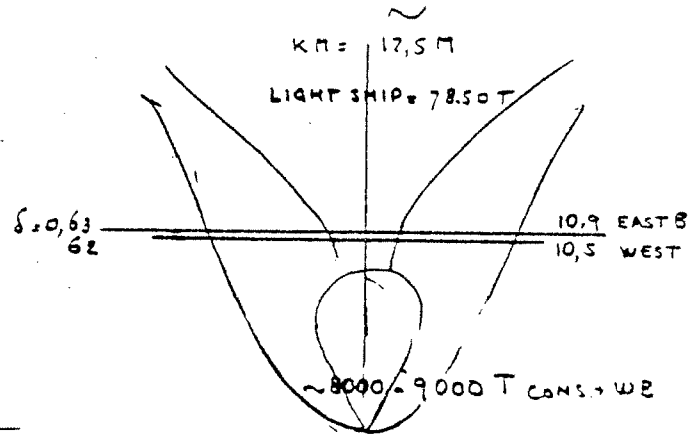
3 TYPES OF VESSELS IN COMPETITION

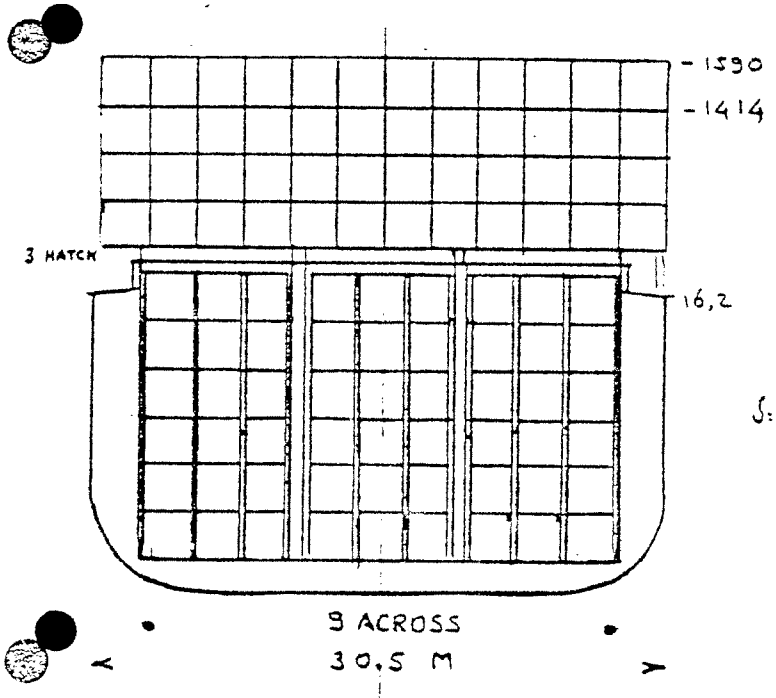
COMPARISON ON SHIPS' MAIN PARTICULARS

NAME	CONCORDIA CORONA	NEDLLOYD v NOORT " v DIEMEN	WOENSDRECHT
OWNER	JANSSEN GERMANY	NEDLLOYD	V OMMEREN
YARD	THYSSEN NORSEE EMDEN	VD GIESSEN KR.ºº YSSEL	GÖTAWERKEN LANDSKR.
DELIVERY	DEC 82	APRIL 84 JULY 84	82
CLASS	G.L. 100A 4 E + MC AUT CONTAINERVESSEL	BV +I-3/3 E HAUTE MER CONTAINERVESSEL AUT	BV +I-3/3 E BULK ORE DEEP SEA-CLACE I- AUT-MS
BRT / NRT	17750. - 10834	ESTIM. 22700 - 13500	26171 - 16914.
LOA	166,04	182,50	186,54
LPP	152,60	174,00	180,00
B	28,50	30,50	30,00
H	15,80	16,20	16,80
T SCANTL	11,61	11,00	11,86
DW	MAX 25,412 T	28,407 T	41.800 T
LIGHT SHIP	7850 T	11000 T	11 200 T
DISPL.	33262 T	38407 T	53,000 T
BLOCK	.642	.642	.806
3 TIERS	1125 TEU	1414 TEU	1435 TEU
4 TIERS	1282	1590	1548
REEFER P.	DECK 101	DECK 90 HOLD 60	DECK 150
		SPACE FOR BREAKBULK IN HOLDS	
M.E.	B+W 5 L 80 GFCA 15400 BHP / 106 RPM	SULZER 5 RLB 90 17500 BHP / DERATED 19000 " / 94 MAXIMUM	B+W 4 L 90 GFCA 13850 BHP / 96 RPM
SHAFTGEN	790 KW	SIEM 1300 KW MAX	—
AUXIL.	WÄRTS 821 KW / 720 RPM	SWD 1500 KW / 720 RPM	WÄRTS 724 KW / 720 RPM
	" 821	1500	724
	" 821	700	724
	DEUTZ 96 KW / 1800 "	160 KW / 1800 "	SCANIA 162 KW / 1800
$\frac{D^{2/3} \cdot V^3}{TON/DAY}$	120 - 125	120 - 125	AS BULKCARRIER 119 11,8M AS CONTR 100 9,2M " 80 7,5M

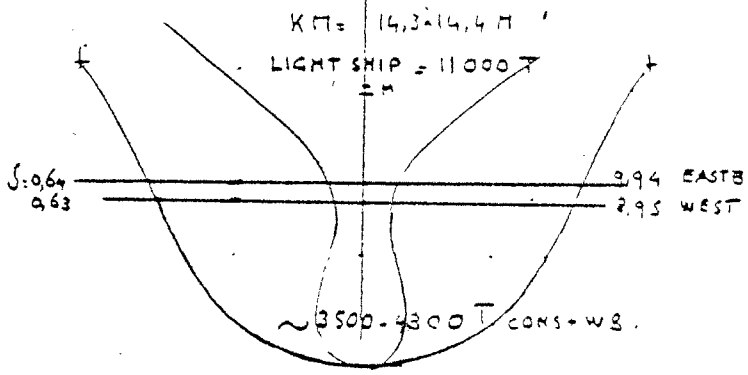


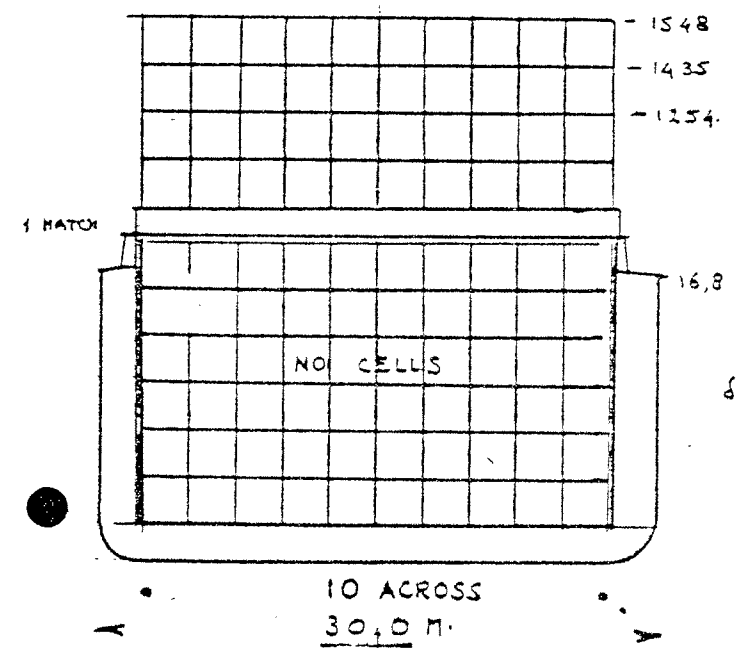
"CONCORDIA"



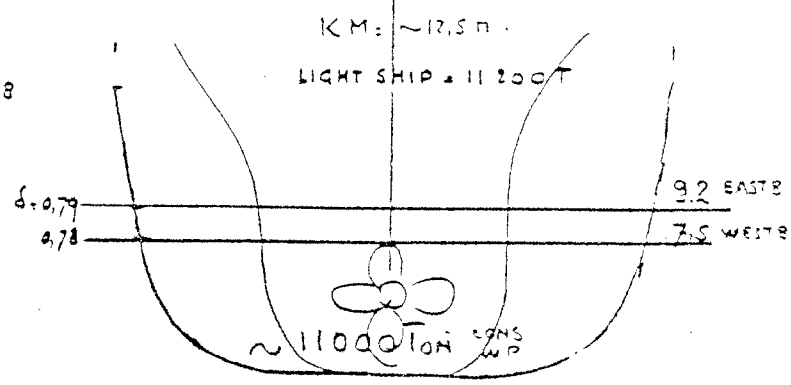


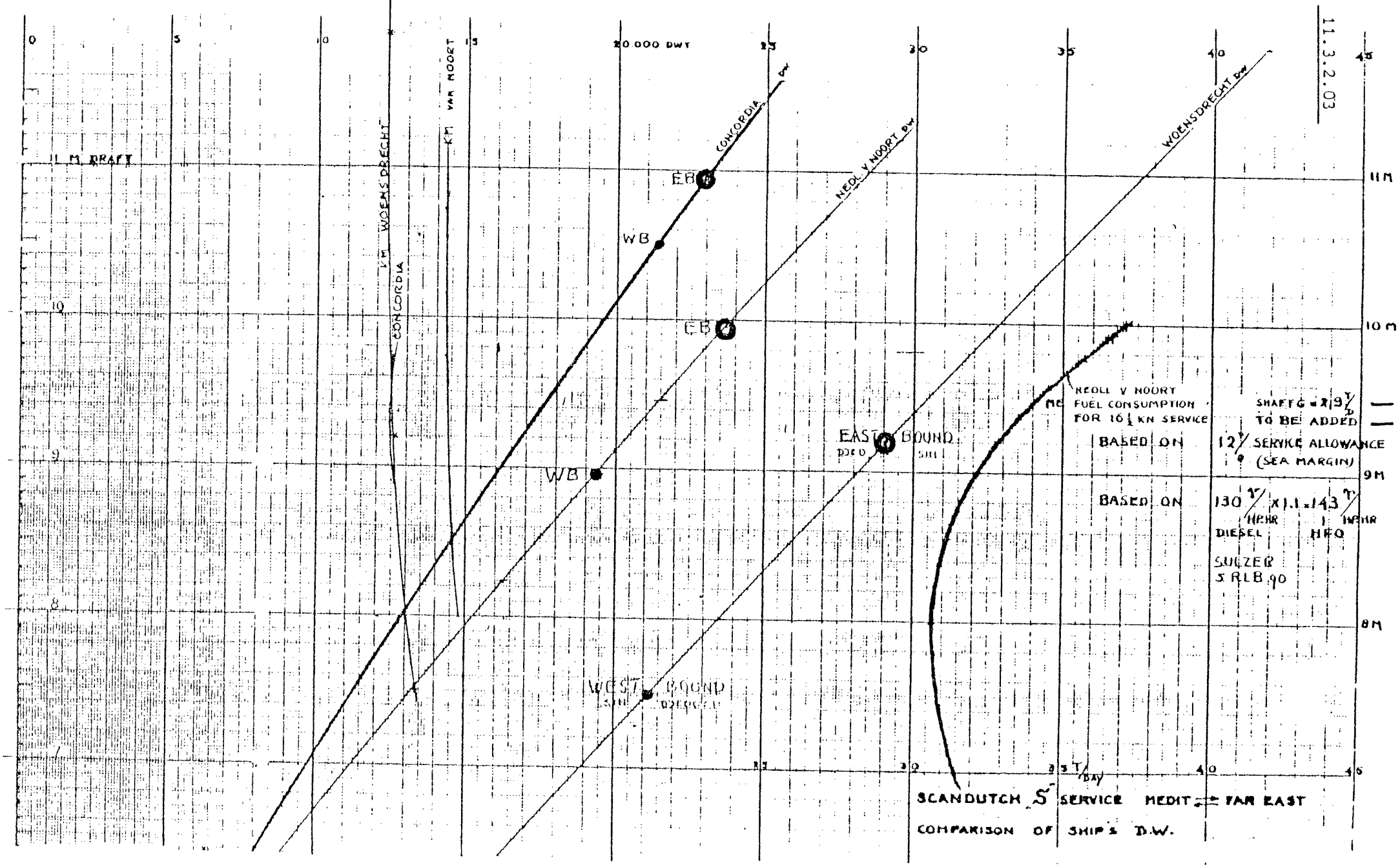
"NEDLLOYD v NOORT"





"WOENS DRECHT"

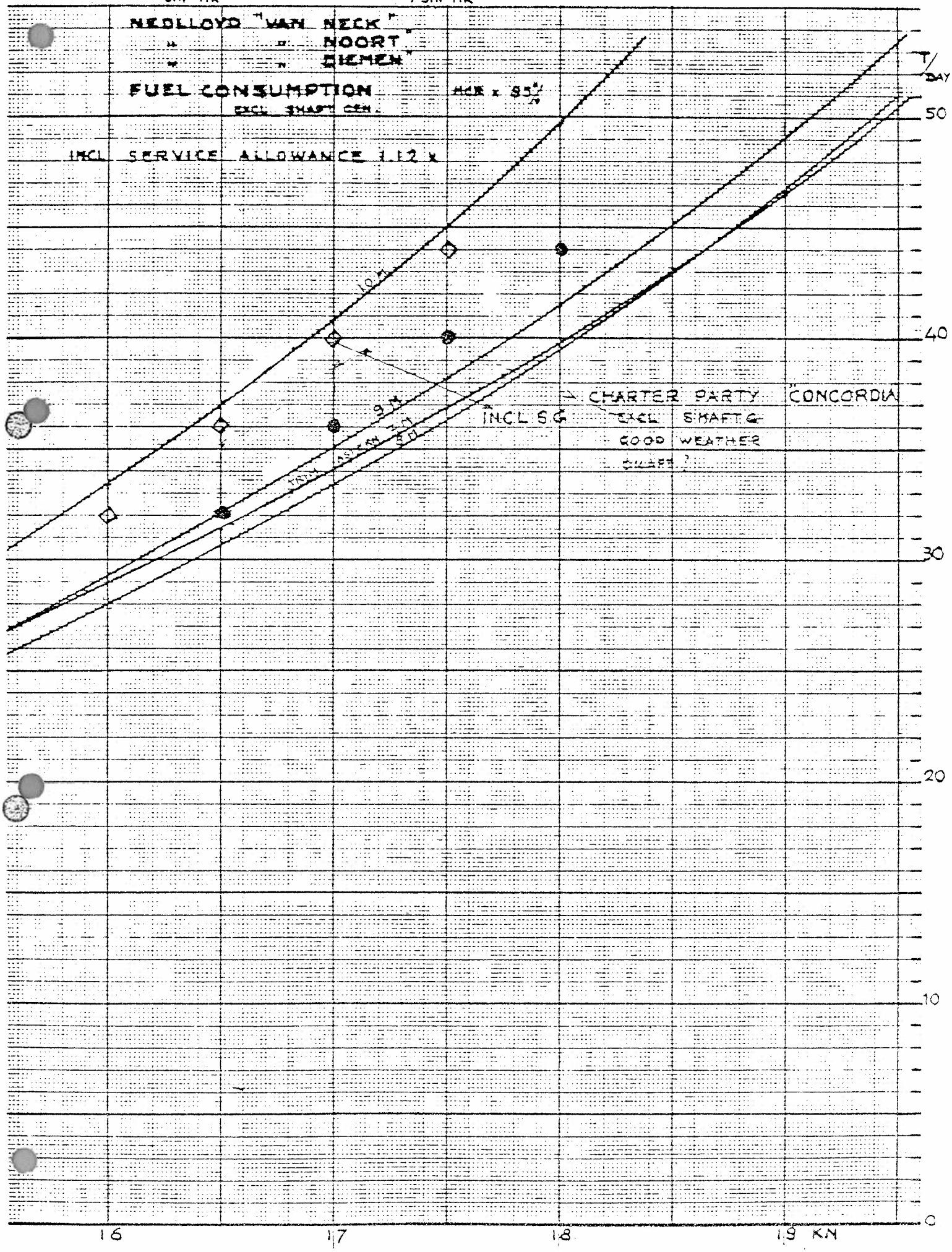



11.3.2.03
0
5
10
15
20.000 DWT
25
30
35
40
45
11 M DRAFT
11 M
10
10 M
9
9 M
8
8 M
KM WOENSDRECHT
KM VAN NOORT
CONCORDIA
CONCORDIA DW
NEDL V NOORT DW
WOENSDRECHT DW
EB
WB
EB
WB
EAST BOUND
LOADED
BALLAST
WEST BOUND
BALLAST
LOADED
NEDL V NOORT
FUEL CONSUMPTION
FOR 16½ KN SERVICE
SHAFT G = 2.9 T/D
TO BE ADDED
BASED ON 12% SERVICE ALLOWANCE
(SEA MARGIN)
BASED ON 130 T/HR x 1.1 = 143 T/HR
DIESEL HFO
SULZER
5 RLB 90
35 T/DAY
SCANDUTCH 5 SERVICE MEDIT ⇄ FAR EAST
COMPARISON OF SHIP'S D.W.

11.3.2.04

BASED ON SERVICE ALLOWANCE OF 12% ABOVE TRIAL COND. (SMOOTH HULL, CALM SEA, BF 2) { WIND-WAVES-PITCHING ROUGHNESS (FOG)

130 g/SHP HR DIESEL × 1.1 = 143 g/SHP HR HEAVY FUEL OIL



COMPARISON ON FUEL CONSUMPTION

	ACTUAL CONCORDIA	PROGNOSIS NEDLL. VAN. NOORT	ACTUAL WOENSDRECHT
AT SEA	57 DAYS	59 DAYS	62
PILOT-PILOT	23447 MILES	23370 MILES	23308 MILES
KN – TON/D	17.15 KN 39.4 T/DAY INCL SHAFTG	(16,50) KN 30,7 + 2,9 = 33,6 T/DAY INCL SHAFTG. MEAN VALUE	15.65 KN 44 – T/DAY EXCL AUXIL
FUEL ROUNDTRIP	2247 TON HFO + 116,5 TON D.O	59 x 33,6 = 2040 = 2072 HFO + 100 HFO 32 x 1 SOLKB 32 2072	2722 T HFO + 168. T.D.O
WESTB	1282 TEU x 9,5 T/TEU = 12.180 T	1282 TEU x 9,5 = 12 180	971 TEU x 9.9 T/TEU + 240 = 9889 T
EASTB	1125 TEU x 14,2 = 16.000 T	1125 TEU x 14,2 = 16.000	1163 TEU x 13. = 18293 T
	2407 TEU 28 180 T	2407 28,180 T	2134 28182 T 20 – 2154 TEU
		12180 T CARGO WESTBOUND 4000 CONS+WB 16180 DW → T = 8,24 M.	
TON FUEL / 1000' x TEU	2247 / 23447 x 2407 = 0,0398 ~ 0,040	30,7 + 2,9 = 33,6 T/DAY INCL SHAFTG	
	59 DAYS 23370 MILES (16,5) KN 35 – T/DAY INCL. SHAFTG. 59 x 35 = 2064 HFO + 116,5 TON DO	16000 T CARGO EASTBOUND 4000 CONS+WB 20.000 DW → T = 9,14 M 32,7 + 2,9 = 35,6 T/D.	
WESTB	1282 TEU x 9,5 = 12180 T		
EAST B	1125 TEU x 14,2 = 16.000 T		
	2407 TEU 28 180 T		
TON FUEL / 1000' x TEU	2064 / 23.370 x 2407 = 0,0367 ~	SAME 2040 / 23.370 x 2407 = 0,0363	2722 / 23.370 x 2407 = 0,0484
		59 DAYS 23370 MILES (16,50) KN 34,0 + 2,9 = 36,9 T/DAY INCL SHAFTG MEAN 59 x 36,9 = 2180 = 2212 HFO + 100 HFO 32 x 1 32 2212	
WESTB.		1590 TEU x 9,5 = 15105 T	
EASTB		1426 " x 14,2 = 20249 T	
		3016 TEU 35354 T OPTIMAL	
		15105 T CARGO WESTB. (SEE M 12) 4278 CONS + WB 19383 DW. → T = 8.95 M.	
		20249 T CARGO EASTB (SEE M 10) 3438 CONS + WB. 23687 DW → T = 9.94 M	
TON FUEL / 1000' x TEU		2180 / 23.370 x 3016 = 0,031	

PERFORMANCE COËFF SINGAPORE⇄DJEDDAH

WESTBOUND

	CONCORDIA	NEDLLOYD v NOORT	WOENSDRECHT
TRAJECT	SINGAPORE – DJEDDAH		SINGAPORE–DJEDDAH
DATE	13-5-83 24-5-83	PROGNOSIS	10-6-83 22-6-83
DRAFT/DISPL	10.50 – 29.010 TON	8.95 / – 30.383 TON	7.50 – 32.400 TON
TON P. DAY	38.5 INCL SHAFTGEN.	32.0 + 2.9 SHAFTG = 34.9	43.0 HFO + 2.9 D.O.
SPEED	17.5 KN	16.5 KN	14.65 KN
CONTR. WEIGHT	~ 12180		~
T WESTB.	10.50	8.95	7.50
δ	0.62	0.624	0.78
DISPL	29.010	30383	32400
LIGHT SHIP	7850	11000	11200
DEADW	21160	19383	21200
CONS + WB	8980	4278	11311
CONTR. WEIGHT	12.180 = 1282 x 9.5 T/TEU	15105 = 1590 x 9.5 T/TEU	9889 < 9649 = 971 x 9.9 T/TEU; 240 BREAKB
$D^{2/3} \cdot V^3$ / TON/DAY	$\frac{(29.010)^{2/3} \cdot (17.5)^3}{38.5} = \frac{945 \cdot 5350}{38.5}$	$\frac{(30383)^{2/3} \cdot (16.5)^3}{32.0 + 2.9} = \frac{974 \times 4490}{34.9}$	$\frac{(32400)^{2/3} \cdot (14.65)^3}{43.0 + 2.9} = \frac{1016 \cdot 3140}{45.9}$
	INCL. SHAFTG (= 131 1/2)	INCL SHAFTG. (= 125 1/2)	INCL AUXIL = 70

EASTBOUND

TRAJECT	DJEDDAH – SINGAPORE		DJEDDAH – SINGAPORE
DATE	30-6-83 11-7-83 (C_2)		29-7-83 10-8-83 (W_2)
DRAFT / DISPL	10.92 – 30610 T	9.94 – 34.687	9.20 – 40.300
TON P. DAY	36.4 INCL SHAFTG.	36.7 + 2.9 SHAFTG = 39.8	43.8 HFO + 2.7 D.O.
SPEED PILOT/P	16.7 KN	16.5 KN	15.93 KN
CONTR WEIGHT	16.000	20.249	18.293
T EASTB	10.92	9.94	9.20
δ	0.629	0.642	0.792
DISPL	30.610	34.687	40.300
LIGHT SHIP	7850 = 1.8H x 105 K/m³	11.000 122 K/m³	11.200 119 K/m³
DEADW	22.760	23.687	29.100
CONS + WB	6.760	LOADING CONO. 1A: 1429 F.O.; 250 D.O.; 100 LUB.; 250 FRESHW; 95 SMALL TKS; 25 CREW; 200 STORES; 7089 / 3438; 3.438	1565 F.O.; 9.242; 10.807
CONTR. WEIGHT	16.000 = 1125 x 14.2 T/TEU	20.249 = 1426 x 14.2 T/TEU	18.293 = 1163 x 15.7 T/TEU
$D^{2/3} \cdot V^3$ / TON/DAY	$\frac{(30610)^{2/3} \cdot 16.7^3}{36.4} = \frac{980 \times 4460}{36.4}$ (106 RPM)	$\frac{(34.687)^{2/3} \cdot 16.5^3}{36.7 + 2.9} = \frac{1063 \times 4490}{39.6}$ ~(94 RPM)	$\frac{(40.300)^{2/3} \times 15.93^3}{43.8 + 2.7} = \frac{1176 \times 4040}{46.5}$ (96 RPM)
	INCL. SHAFTG. (= 125 1/2)	INCL. SHAFTG = 120 1/2	INCL AUXIL (= 102)

11.3.3.02

PERFORMANCE COËFF

WESTBOUND
EAST "

	CONCORDIA	NEDLLOYD VAN NOORT	WOENSDRECHT
TOTAL VOYAGE WESTBOUND	665.— HRS PILOT-PILOT 11782 MILES. 17,7 KN. 1079,7 TON HFO. 38,96 TON/DAY		745,48 HRS PILOT-PILOT 11428 MILES " " 15,32 KN 1272,4 TON HFO 40,95 TON/DAY
MEAN DRAFT DISPL LIGHT SHIP D.W.	9,4 M 25,260 7,845 17,415	~ 8,55 M 28,515 11,100 17,415	7,4 M. 31,900 T 11,200 20,700
$\frac{DISPL^{2/3} \cdot V^3}{TON/DAY}$	$\frac{(25260)^{2/3} \cdot 17,7^3}{38,96} = \frac{861 \times 5540}{38,96}$ INCL SHAFTGEN = $12\frac{1}{2}$	$\frac{(28515)^{2/3} \cdot 16,5^3}{31,2+2,9} = \frac{933 \times 4490}{34,1}$ INCL. SHAFTGEN 123 CONCLUSION WESTBOUND. $16\frac{1}{2}$ Kn — 34,1 T/DAY — DW = 17415	$\frac{(31.900)^{2/3} \cdot 15.32^3}{40.95+2,85} = \frac{1007 \times 3590}{43,8}$ INCL. AUXIL = $82\frac{1}{2}$

TOTAL VOYAGE EASTBOUND.	701.30 HRS. 12069 MILES. 17,2 KN. 1098.0 TON HFO 37,56. TON/DAY		744,54. HRS 11880 MILES. 15,95 KN 1303,7 TON HFO 42,00 TON/DAY
MEAN DRAFT DISPL LIGHT SHIP D.W.	9,7 M. 26.310 7845 18465	8,75 M. 29565 11100 18465	8,2 M. 35.600 11.200 24.400
$\frac{DISPL^{2/3} \cdot V^3}{TON/DAY}$	$\frac{(26.310)^{2/3} \times (17,2)^3}{37,56} = \frac{885 \times 5080}{37,56}$ INCL SHAFTGEN = 120	$\frac{(29.565)^{2/3} \times (16,5)^3}{31,5+2,9} = \frac{956 \times 4490}{34,4}$ 125. CONCLUSION EASTBOUND $16\frac{1}{2}$ KN — 34,4 T/DAY — DW = 18465	$\frac{(35600)^{2/3} \times 15,95^3}{42,0+2,9} = \frac{1082 \times 4050}{44,9}$ INCL AUXIL = $97\frac{1}{2}$

NEDLLOYD "VAN NOORT"

OPTIMAL EASTB 1426 TEU x 14,2 T/TEU — 9,94 M DRAFT
" WESTB 1590 " x 9,5 — 8,95 M " ROUNDTRIP SCANDUTCH "S"

	HFO	IFO	DAYS
AT SEA			
23.370 MILES PILOT-PILOT AT 16,5 KN			
1416 HRS = 59 DAYS DRAFT MEAN 9,45 M			
59 DAYS x (34 T/D MAIN E + 2,9 T/D SHAFT G 540 KW) = 59 x 36,9 =	2180		59 = 65%
IN PORT			
23 PORTS IN 23 DAYS			
23 DAYS (AUXIL 400 KW 2,3 T/D) = 23 x 2,3 =		53	23 = 25%
BOILER 23 x 1 =	23		
MANOEUVRING			
9 DAYS x (AUXIL 2,9 + 2,3 T/D) = 9 x 5,2 =		47	9 = 10%
BOILER 9 x 1 =	9		
FUEL CONSUMPTION P. ROUNDTRIP EXCL REEFERCONTRS.	2212 T	100 T	91 = 100%

Electrische belasting op zee 540 kW

$$\frac{540 \times 1,36}{0,85\ (ACZ)} = 860 \text{ pk}$$

860 x 142 x 24 = 2,9 Ton/dag H.F.O.

Electrische belasting in haven 400 kW

$$\frac{400 \times 1,36 \times 24}{0,92} = 2,3 \text{ Ton/dag I.F.O.}$$

15/9/83

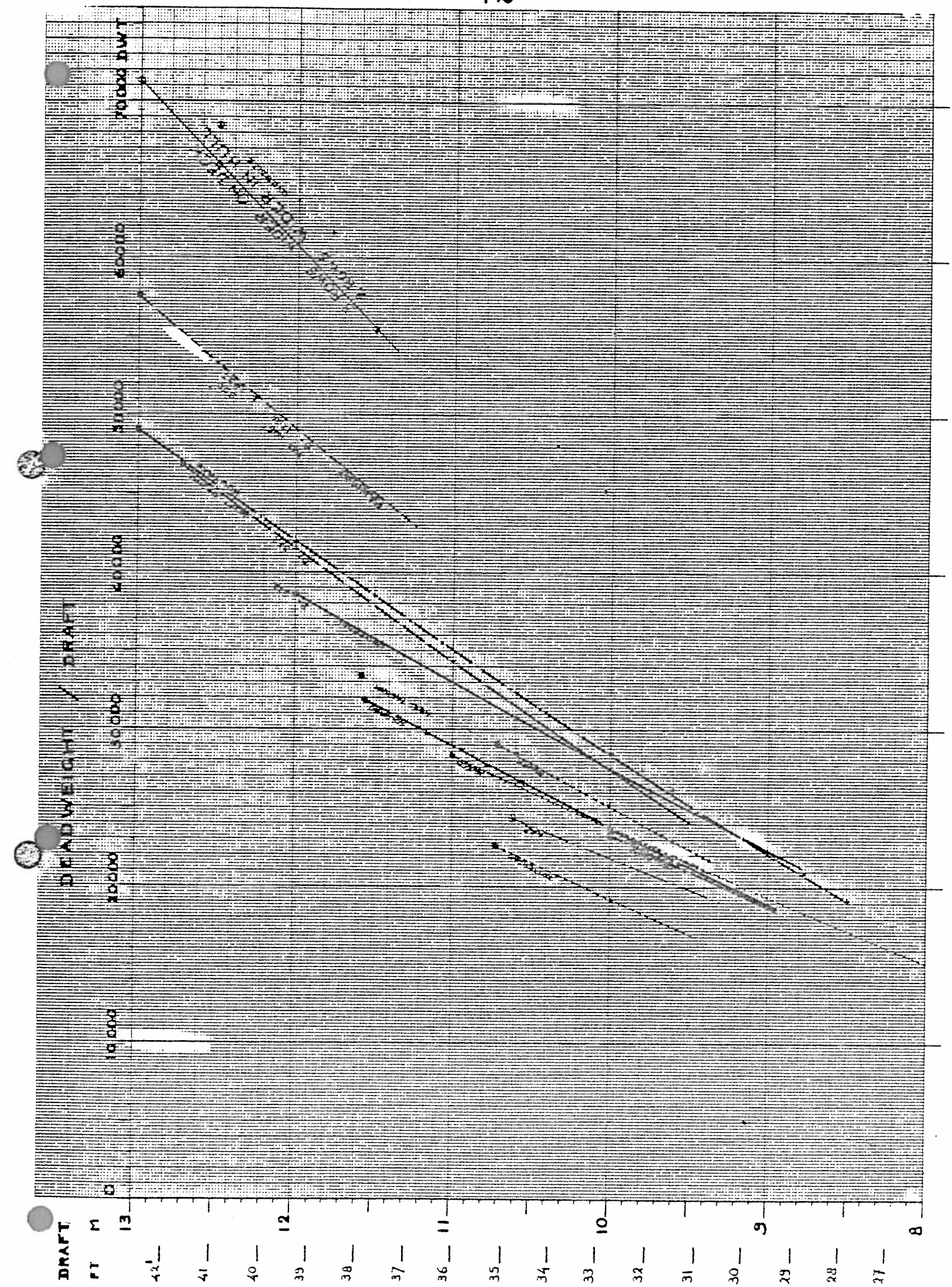
12.0
LARGE CELLULAR CONTAINER VESSELS

4,600 TEU - 1,200 TEU

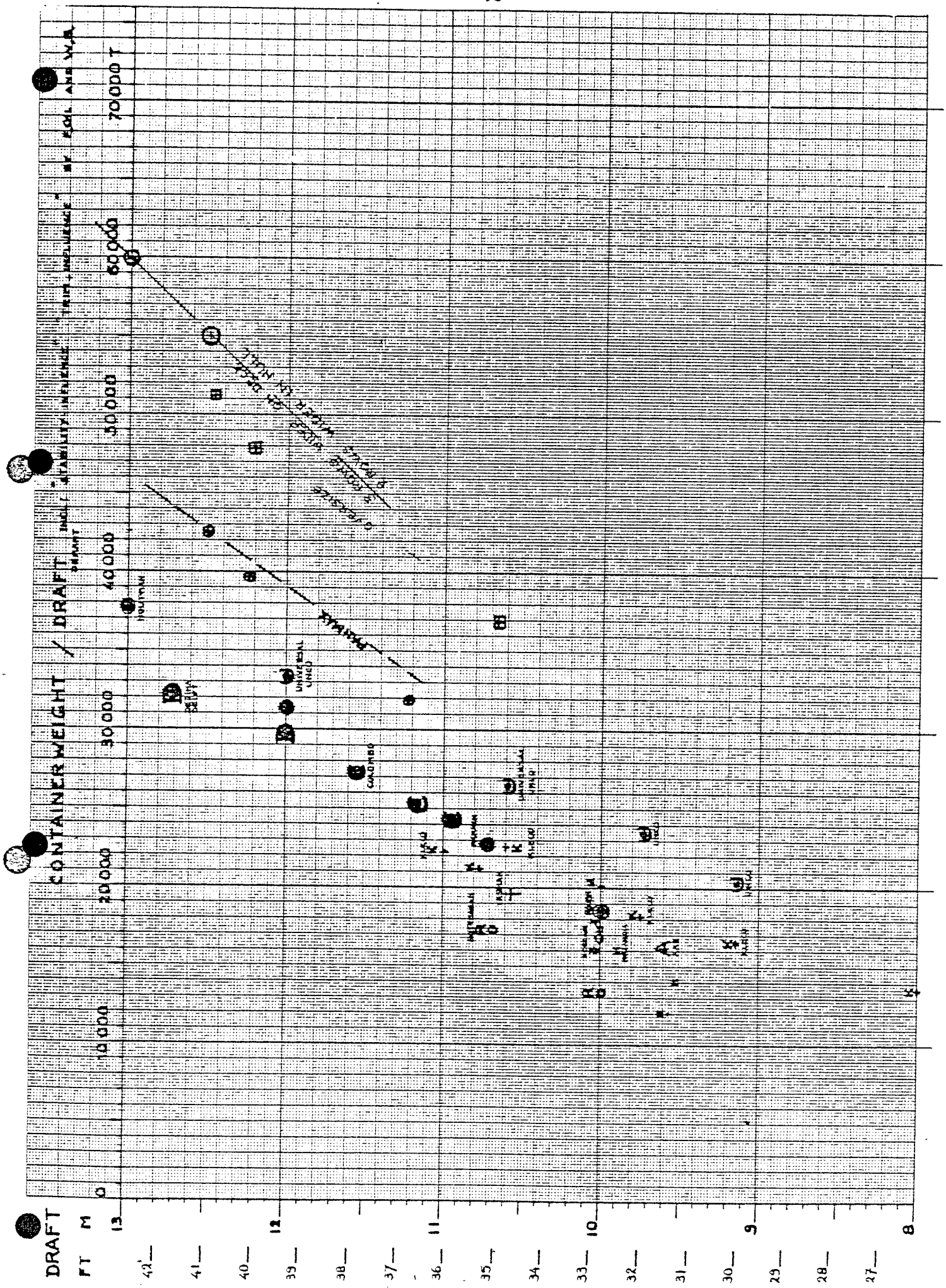
GEARLESS

GEARED

CELLULAR VESSELS



DEADWEIGHT - TO - DRAFT RATIO.



CONTAINERWEIGHT - TO - DRAFT RATIO.

12.1.03

12

Ro Ro CONTR | CELL CONTR

LINE	ANRO ASIA	ROTTERDAM-ROCH.R.	ROUEN - ROSARIO	ASIAN JADE	VAN HECK	VAN NOORT	HOLLANDIA	ZEELANDIA
CONTR.	AUSTR-MALAYA	USA-MIDDLEAST	USA-MIDDLEAST	AUSTR-MAL-HONGKONG	KLUE-WAFR	N. ZEALAND-JAPAN-MALAYA	EUROPA-CARIB.	EUROPA-NORTH AMERICA
	1976	1977	1977	1977	1982	1982	1975	1978
L_{oa}	213.17	196.50	212.10	186.00	182.50	182.50	204.00	205.56
L_{pp}	199.16	183.20	198.80	177.40	174.00	174.00	193.10	192.00
B	27.50	32.24	32.24	27.60	30.50	30.50	30.80	30.80
H	16.75	23.25	20.40	16.20	16.20	16.20	18.80	18.75
T	8.91	10.30	10.70	10.62	11.00	11.00	10.02	10.02
LIGHT SHIP	12,100	17,249	16,170	9,228	11,030	10,600	13,740	13,750
	18.02 M	14.55	12.30	11.43	11.71		12.48	12.65
	LLOYDS	LLOYDS	LLOYDS	LLOYDS	B. VERITAS	B. VERITAS	LLOYDS	LLOYDS
TEU	634	746	811	611	786	780	878	920
	562 3 TR	394 3 TR	717 3 TR	344 3 TR	658 3 TR	680 3 TR	528 3 TR	514 3 TR
	422	177 3°	185 4°	186	36	120 4°	66 4°	66 4°
TEU	1136	1317 3 TR	1713 4 TR	1150 3 TR	1550	1580	1472	1500
	134 R			88 R	60 R	60 R	120 R	180 R
	66 R 20/40	100 R 20/40	100 R 20/40	54 R 20/40	90 R 20/40	90 R 20/40	126 R 20/40	126 R 20/40
M. ENGINE	MAN	SULZER 8RND50 M	SULZER 8RND50 M	SULZER 8 RND76M	SULZER 5 RL850	IDEA	SULZER 10 RND 50	SULZER 9 RND50M
MCR/RPM	14,990 - 430	16,800 - 122	16,800 - 122	18,380 - 122	17,500 - 94		15,000 - 122	30,000 - 122
85%	16,140 - 110 RPM	13,800 - 122 RPM	13,800 - 122 RPM	16,200 - 122 RPM	14,870 - 94 RPM		24,650 - 122 RPM	25,600 - 122 RPM
	2000 HORSEPOWER				1200 HP SHAFT GEN			
	ø 6000 - 4 BL - P/D 1.35 - 0.715	ø 6500 - 4 BL - P/D .715	ø 6500 - 4 BL - P/D .727	ø 6120 - 4 BL - P/D .65	ø 7000 - 4 BL - P/D .55		?	ø 6500 - 4 BL - P/D .874
	3464 8 3520 PROP	5150 6 5142 PROP	5274 5285 PROP	5367 4.08 5016 PROP	5344 5 PROP	IDEA		
	8.91 32,610 T	10.00 36,180 T	10.00 41,070 T	9.60 29,420 T	10.00 34,780 T		HSVA TESTS	
	4.178 32,200 M	4.417 35,289 M	4.625 48,042 M	4.461 28,658 M	4.639 33,526 M		10.00 36,352 T	10.00 37,328
V-IMP	18.— KN 12,380 HP	19.— KN 16,350 HP	19.— KN 17,380 HP	19.— KN 13,050 HP	18.— KN 13,250 HP		4.405 36,000 M	4.418 36,430 M
							19.— KN 18,600 HP	19.— KN 17,550 HP
	14,150	15,500	15,500	15,000	15,240		21,400	20,630
TEU/SHIP	165 x 24 x 1.84 x 52.6	162 x 24 x 753	162 x 24 x 774	162 x 24 x 58.4	150 x 24 x 55.—		162 x 24 x 88.3	162 x 24 x 80.3
		VESSEL TOO SHORT			EFFICIENT ENGINE		BAD HULL: BAD PROP	BAD PROP
	1042 x 5832 / 12300 = 475	1093 x 6859 / 16350 = 443	1191 x 6859 / 17300 = 475	954 x 6859 / 13050 = 502	1062 x 5832 / 12250 = 469		1110 x 6859 / 18600 = 409	1120 x 6859 / 17950 = 428
	1042 x 5832 / 596 = 102	1093 x 6859 / 753 = 99	1191 x 6859 / 77.4 = 106	954 x 5832 / 58.4 = 112	1062 x 5832 / 55.— = 113		1110 x 6859 / 81.3 = 91	1120 x 6859 / 80.3 = 96

	ANRO ASIA	ROTTERDAM-ROCH.	ROUEN - ROSARIO	ASIAN JADE	KLECO		HOLLANDIA	ZEELANDIA
	8.91 33,610	10.72 35,809	10.72 45,153	10.62 33,610	11.00 39,407	11.00 39,407	10.00 36,352	10.02 37,428
	12,100	17,249	16,170	9,228	11,100	10,600	13,740	13,750
	21,510	22,560	28,023	24,382	28,307	28,807	22,612	23,678
	4,570	5,154	6,303	7,888	6,112	6,112	5,864	5,658
	896 x 17 = 16,940	17,406	1530 x 4.6 = 22,770	1176 x 14.0 = 16,464	1465 x 15.13 = 22,195	22,635	1240 x 14 = 17,348	1283 x 14 = 17,980
	634	1,877		660 x 16 = 10,560	785 x 15.13 = 11,895		878 x 14 = 12,280	910 x 14 = 12,850
	362	5,529		516 x 4.2 = 2,934	680 x 15.33 = 8,300		362 x 14 = 5,068	363 x 14 = 5,050
				KM = 12.61	KM = 14.33		KM = 13.12	KM = 14.01
				KG = 11.55	KG = 13.96		KG = 12.64	KG = 13.91
				MG = 1.06 - .006 = 1.00	MG = 1.33 - .038 = 1.21		MG = 2.63 - .05 = 2.58	MG = 1.10 - .012 = 0.98
		9.97 36,315	10.02 41,325	9.62 29,583	10.77 38,350	38,350	9.89 36,538	10.02 37,428
		17,249	16,170	9,228	11,100	10,600	13,740	13,750
		19,066	25,155	20,355	27,250	27,750	22,798	23,678
		5,927	6,664	7,888	6,011	6,011	6,394	7,678
		13,753	18,491	1176 x 18.6 = 21,866	1585 x 18.6 = 21,339	21,739	1200 x 13.3 = 16,000	1250 x 13.8 = 16,000
		7,817		660 x 10.6 = 6990	785 x 18.6 = 10,520		878 x 13.3 = 11,700	910 x 13.8 = 11,780
		5,936		516 x 10.6 = 5,476	800 x 18.6 = 10,719		322 x 13.3 = 4,300	330 x 13.8 = 4,220
				KM = 12.51	KM = 14.34		KM = 13.11	KM = 14.01
				KG = 11.32	KG = 13.47		KG = 12.00	KG = 13.13
				MG = 0.39 - .057 = 0.33	MG = 0.87 - .087 = 0.80		MG = 1.11 - .209 = 1.06	MG = 1.66 - .115 = 1.54
			10.02 41,325	9.62 29,583	9.75 33,700	33,700	9.53 34,758	9.57 35,151
			16,170	9,228	11,100	10,600	13,740	13,750
			25,155	20,355	22,600	23,100	21,053	21,501
			8,295	6,482	3,420	3,420	7,053	9,501
			16,860	1004 x 13.3 = 13,863	1056 x 17.5 = 18,488	19,680	1200 x 17.7 = 16,440	1250 x 15.6 = 19,000
				660 x 13.3 = 10,420	785 x 17.5 = 13,740		878 x 17.7 = 10,250	910 x 15.6 = 8,550
				344 x 13.3 = 5,443	311 x 17.5 = 5,440		322 x 17.7 = 3,750	330 x 15.6 = 3,170
				KM = 12.51	KM = 14.37		KM = 13.05	KM = 13.56
				KG = 11.65	KG = 13.13		KG = 11.89	KG = 11.73
				MG = 0.26 - .008 = 0.18	MG = 1.24 - .009 = 1.12		MG = 1.16 - .025 = 1.11	MG = 1.13 - .032 = 1.11
				9.62 29,583	9.15 31,070	31,070	9.15 33,080	9.15 33,382
				9,228	11,100	10,600	13,740	13,750
				20,355	19,970	20,470	19,340	19,632
				3,488	3,420	3,420	~ 3,000	3,000
				1004 x 16.3 = 16,367	1356 x 12.5 = 16,950	17,050	16,340	16,632
				660 x 16.3 = 11,990	785 x 12.5 = 9,810			
				344 x 16.8 = 5,777	339 x 12.5 = 6,740			
				KM = 12.51	KM = 14.45			
				KG = 12.00	KG = 13.37			
				MG = 0.43 - .009 = 0.34	MG = 1.48 - .08 = 1.40			

CORR 27 NOV 83
GV

12.1.04

12

CELL. CONTR.

	ABEL TASMAN	COLOMBO	CLEMENT	CLARENCE	HOUTMAN + HOORN	DEJIMA + DELFT	PANMAX	3 ROWS WIDER
Loa	225.87	211.19	210.00	210.00	258.00	287.02	289.50	289.50
Lpp	210.00	195.00	202.00	202.00	247.00	273.00	277.00	277.00
B	30.50	32.20	32.24	32.24	32.26	32.24	32.24	39.60
H	16.40	19.00	18.80	18.80	24.15	25.00	24.15	24.35
T	11.58	11.56	12.00	12.00	13.00	12.72	13.00	13.00
	14 200	13 000	14 998	14 543	22 281	24 359	24 700	29 800
	30.76	79.3	24.23	24.31	34.00	113.34	106	
	11.78	12.81	12.50	12.39	13.58	13.50		
	CERR/LLOYD	LLOYDS	LLOYDS		LLOYDS	LLOYDS		
	861	1014	1150	1150	1794	2020	2264	2720
	558	511	558	562	660	668	842	1036
	218	252	285	231	260	264	364	448
	1677	1777	1397	2003	2714	2952	3470	4204
	100	0	116	0	352	0		448
	0	130 R	64 R	88 R	886	70 R		4652
	20 R	88 R	99/88 R	99/88 R	55 R	50 R	121 R	
M. ENGINE	JUREL STAL LAVAL	SULZER 6RLA 50	B&W 6L50GBE	IDEM	SULZER 2x8RND90M	SULZER 2x8RND90M	(2x7 CYL L 50 GBE)	(2x7 CYL RTA 84)
MCR/RPM	32.450 - 1161	20.400 - 50	22.800 - 97		2x 26.800 - 122	2x 26.800 - 112	2x 18.300 - 97	2x 28.210 - 87
85%	17.600 - 110 RPM	17.240 - 50 RPM	20.230 - 97 RPM		2x 22.800 - 102 RPM	2x 22.800 - 112 RPM	2x 24.050 - 97 RPM	2x 24.000 - 87 RPM
							2x 1800 KW	2x 1800 KW
	7400	7400	7200		6450	6618	7000	7350
	10.67	10.60	10.60		11.50	11.50	11.50	11.50
	42.870 T	40.430 T	44.800 T		60.800 T	62.600 T	70.100 T	86.200 T
	41.805 M	39.434 M	42.774 M		59.188 M	61100	68.475 M	84.100 M
	19.600 HP	16.300 HP	17.250 HP		28.700	28.200	29.220	?
	22.550	18.700	19.830		33.000	33.400	33.600	?
TON/DAY	225 x 24 = 112	150 x 24 = 67.5	147 x 24 = 70.0		168 x 24 = 122	157 x 24 = 122	147 x 24 = 119	147 x 24 =
	TO BE IMPROVED							
D.V.	1313 x 4800 / 19600 = 500	1177 x 6859 / 16300 = 495	1261 x 6859 / 17250 = 502		1547 x 3261 / 18700 = 500	1578 x 3261 / 18300 = 518	1702 x 3261 / 19320 = 535	1952 x 3261 = ?
D.V.	1323 x 4000 / 122 = 81	1177 x 6859 / 67.5 = 120	1261 x 6859 / 70.0 = 124		1547 x 3261 / 128 = 112	1578 x 3261 / 123 = 120	1702 x 3261 / 119 = 132	1952 x 3261 = ?
	ABEL TASMAN	COLOMBO	CLEMENT	CLARENCE	HOUTMAN + HOORN	DEJIMA + DELFT	PANMAX	3 ROWS WIDER
	11.23	11.56	11.58	11.58	13.03	12.72	12.46	13.00
	45.317	45.816	52.332	52.332	71.343	71.683	77.635	101400
	14.200	13.000	14.998	14.545	22.281	24.359	24.700	29.800
	31.717	32.816	37.334	39.447	49.062	46.784	52.935	71.100
	8.163	4.456	6.334	6.347	11.386	14.306	10.455	11400
	NOT MEN	NOT MEN	2000 x 15.8 = 31 600	2000 x 16 = 32.100	2714 x 13.5 = 37.676	2588 x 13.5 = 32.278	43.540	4284 x 14.5 = 53800
	23.554	1777 x 16 = 28 360	1150 x 15.8 = 18.170		1794 x 4.2 = 3 ...	2 x 4.3 = 26 394		4652 x 11.8
	13.84.2		836 x 15.8 = 13.440		920 x 8.6 = 7994	332 x 6.3 = 2884		
	9.618		KM = 15.07		KM = 15.14	KM = 14.95		KM =
	KM = 13.09	KM = 15.38	KG = 14.81		KG = 14.84	KG = 14.35		KG =
	KG = 12.35	KG = 14.95	MG = 0.16 - 0.08 = 0.26		MG = 0.39 - 0.12 = 0.38	MG = 0.78 - 0.06 = 0.66	0.40	MG =
	MG = 0.67 - 0.05 = 0.62	MG = 0.43 - 0.07 = 0.36						
	10.60	11.17	10.37	10.57	12.00	12.00	12.24	12.50
	41.373	43.776	45.139	45.139	64.097	66.303	75.845	95.800
	14.200	13.000	14.998	14.545	22.281	24.359	24.700	29.800
	28.418	30.776	30.141	30.594	41.716	41.944	51.145	66.000
	8.213	4.456	4.741	4.694	10.400	11.598	11.365	~11.000
	1388 x 13/12 = 15.600	1316 x 16 = 25.328	1000 x 13 = 13.000	2005 x 13 = 25.300	1454 x 13.5 = 21.316	2417 x 12.5 = 29.806	39.780	55000
	346 x 13 = 10.250		1150 x 13 = 14.600		1794 x 13.5 = 23.922	2 x 13 = 26 394	3106 x 1.6	4652 x 11.8
	379 x 13 = 9.350		836 x 13 = 10.800		660 x 11.2 = 7 394	337 x 8.6 = 3412	346 x 2 = 78	
	KM = 12.40	KM = 15.36	KM = 15.05		KM = 13.10	KM = 14.82	KM =	KM
	KG = 12.39	KG = 15.11	KG = 14.86		KG = 14.58	KG = 14.16	KG =	KG
	MG = 0.49 - 0.05 = 0.44	MG = 1.22 - 0.07 = 1.16	MG = 0.19 - 0.04 = 0.15		MG = 0.32 - 0.12 = 0.20	MG = 0.68 - 0.04 = 0.64	448	MG
		10.95	9.75	9.75			11.24	
		42.744	40.440	40.440			62.655	
		13 000	14.998	14.545			24 700	
		29.744	25.442	25.895			42 955	
		4.456	2.002	1.955			9.045	
		1525 x 16 = 25.328	1804 x 13 = 23.440	1840 x 13 = 23.940			43 x 16 x 1.67 = 33.910	
							3750 x 2 = 31.470	
							1260 x 1 = 460	
		KM = 15.32	KM =					
		KG = 14.10	KG =					
		MG = 1.22 - 0.07 = 1.15	MG =					
		7.88	9.15	9.15				
		27.868	37.340	37.360				
		13.000	14.998	14.545				
		14.865	22.342	22.795				
		11.711	2.002	1.955				
		1853 x 9 = 3154	1630 x 13.5 = 20.340	1670 x 13.5 = 20.840				
		KM = 13.46	KM =					
		KG = 14.10	KG =					
		MG = 1.36 - 0.31 = 1.05	MG =					

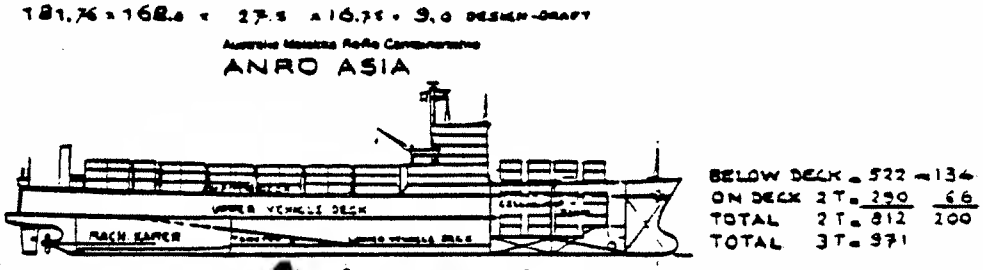
NOTE: CLEMENT WITH BULBS

NOTE: HOUTMAN 10' 353 / HOORN 8' 886

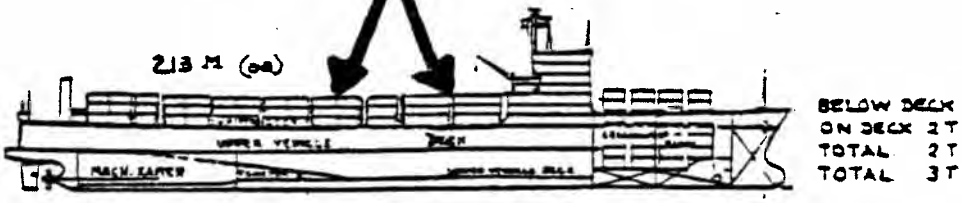
LIGHT SHIP L 22.706

CORR 27 NOV 83

CY



As built, 1976 for the Malaysia-Australia service, acc. prevalent needs.



January 1981... Lengthened! (as predicted at time of ordering) (but denounced by principals then)

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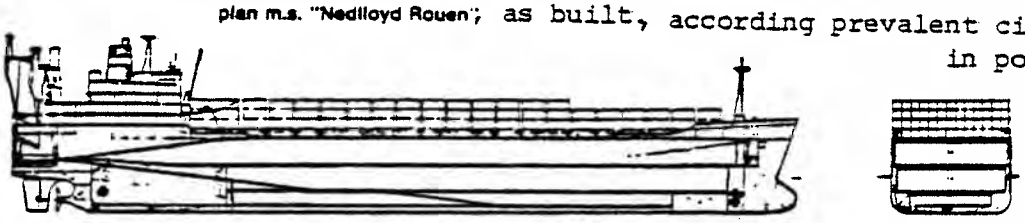
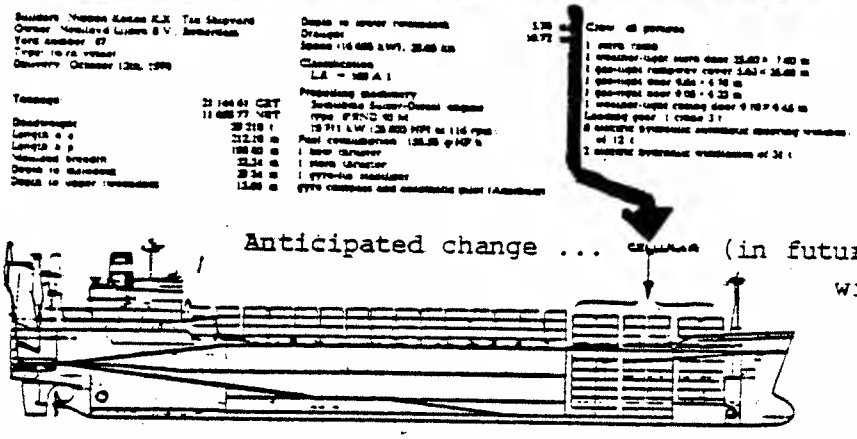
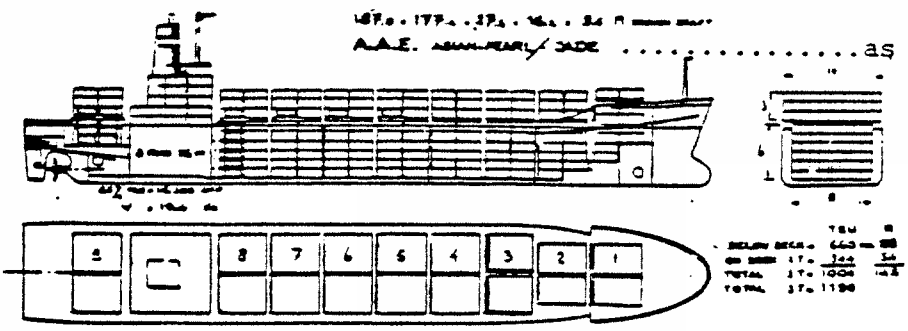
plan m.s. "Nedlloyd Rouen"; as built, according prevalent circumstances in ports of call.



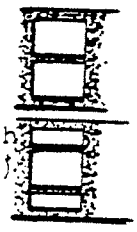


Anticipated change ... (in future) as port cranes will be installed.



.....as built (acc. yard's standard design).

'Twin hatch'

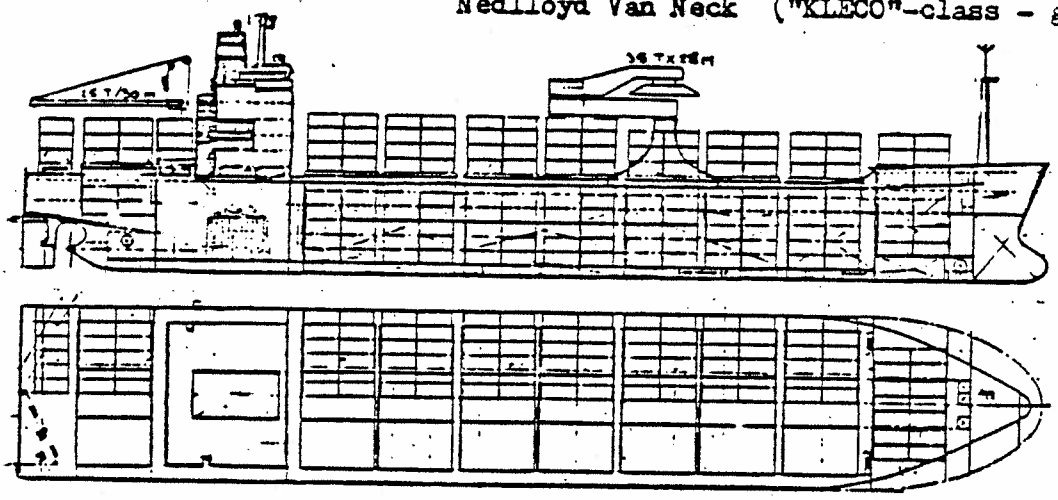


(Triple hatch would be preferable.)

Three examples of "Good design but could still have been better"

182.5 x 174.00 x 30.50 x 16.20 x 11.00 (SCANTL DRAFT)

Nedlloyd Van Neck ("KLECO"-class - geared version)

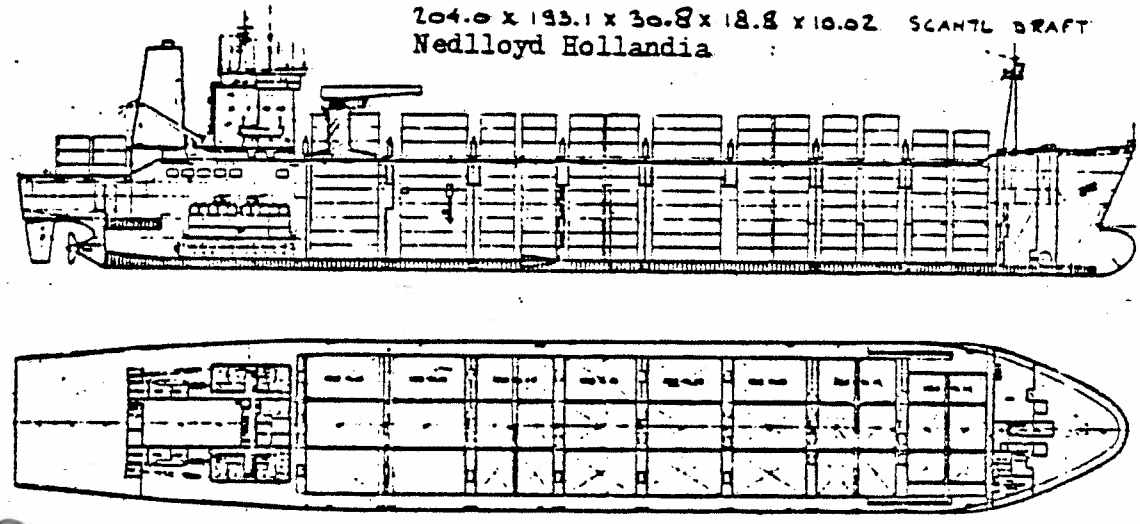


CAPACITY (APPROXIMATE)

BELOW DECK = 785 TEU
ON DECK 3T = 674 "
TOTAL 3T = 1459 "
TOTAL 4T = 1579 "

204.0 x 193.1 x 30.8 x 18.8 x 10.02 SCANTL DRAFT

Nedlloyd Hollandia

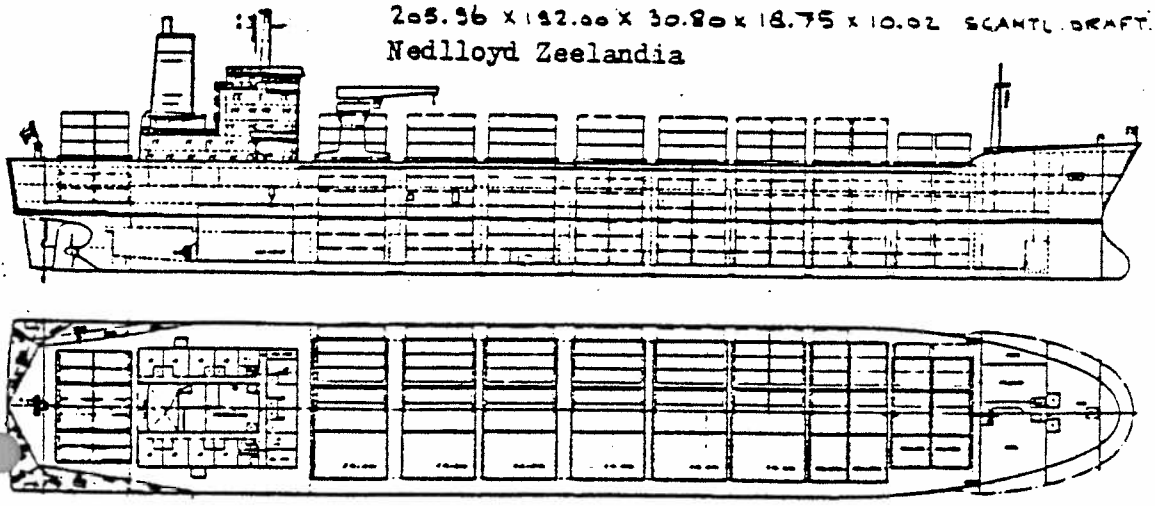


CAPACITY (EXCL CRANE)

BELOW DECK = 878 TEU
ON DECK 3T = 514 "
TOTAL 3T = 1392 TEU
TOTAL 4T = 1472 "

205.96 x 192.00 x 30.80 x 18.75 x 10.02 SCANTL. DRAFT.

Nedlloyd Zeelandia



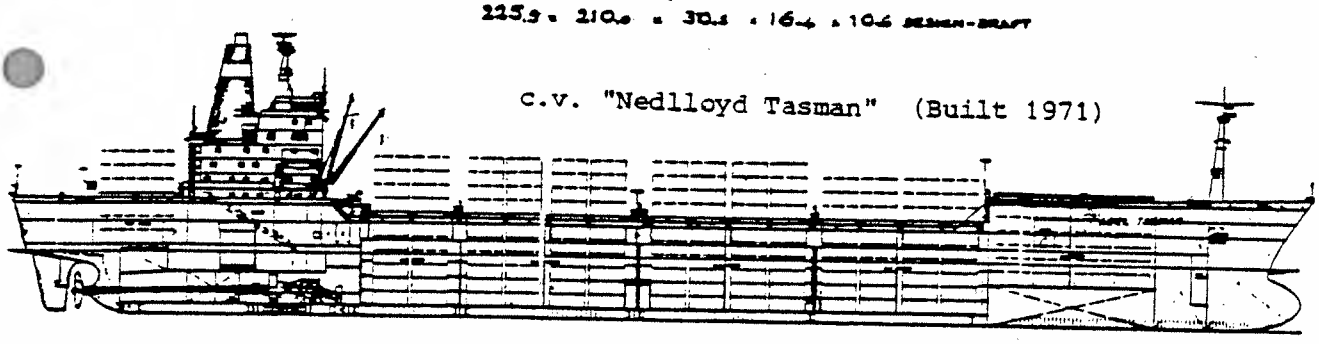
CAPACITY (EXCL. CRANE)

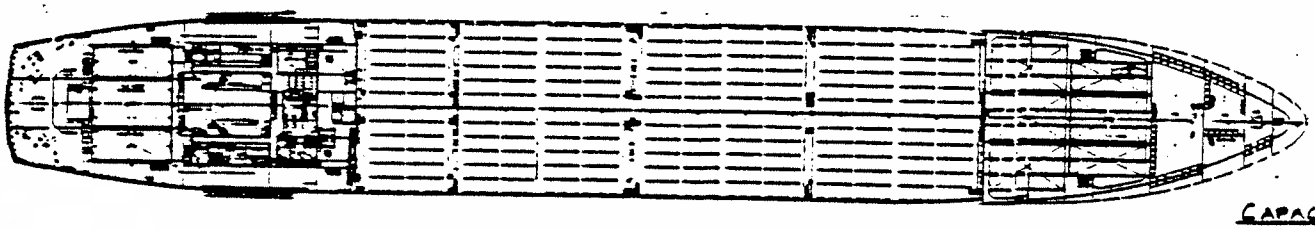
BELOW DECK: 920 TEU
ON DECK 3T = 514 "
TOTAL 3T = 1434 TEU
TOTAL 4T = 1500 "

Examples of medium-sized geared cellular container vessels.

225.9 x 210.0 x 30.1 x 16.4 x 10.6 SCANTL-DRAFT

C.V. "Nedlloyd Tasman" (Built 1971)



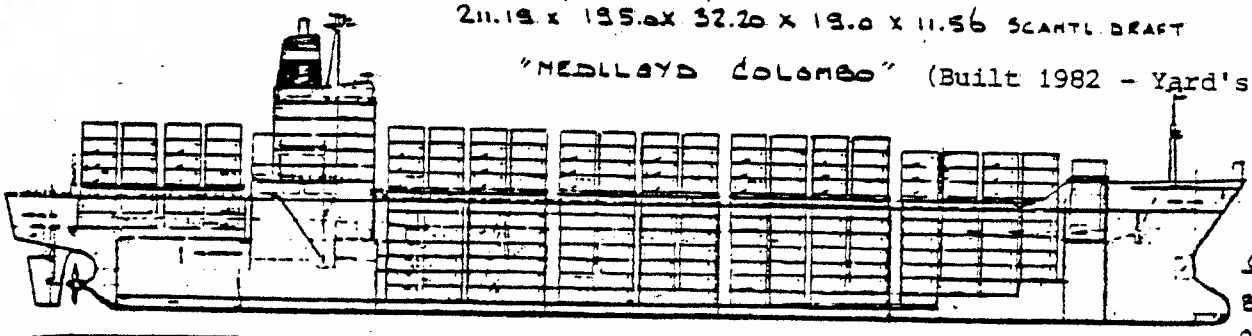


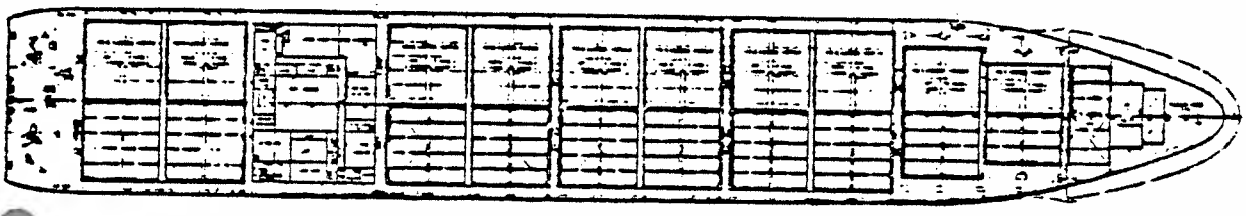
CAPACITY

	TEU
BELOW DECK =	861
ON DECK 3T =	598
TOTAL 3T =	1459
TOTAL 4T =	1677

211.19 x 195.0 x 32.20 x 19.0 x 11.56 SCANTL. DRAFT

"NEDLLOYD COLOMBO" (Built 1982 - Yard's design)





CAPACITY

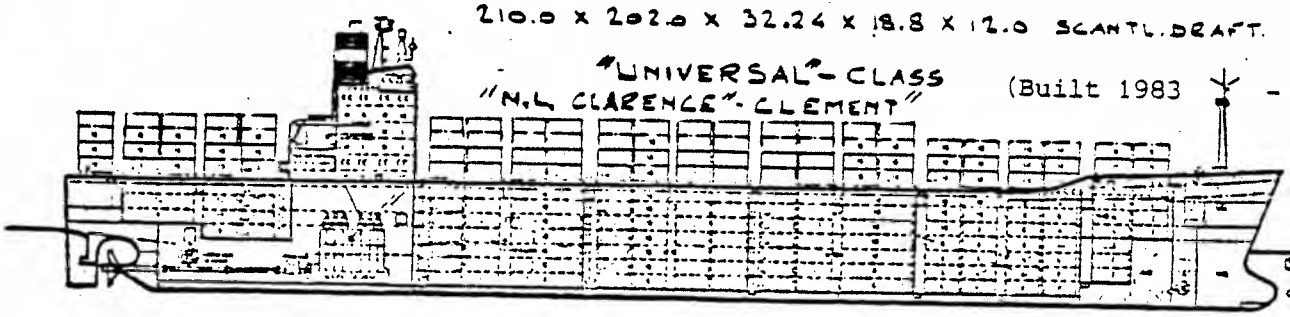
	TEU
BELOW DECK =	1014
ON DECK 2T =	511
TOTAL 2T =	1525

Total 3T=1777

210.0 x 202.0 x 32.24 x 18.8 x 12.0 SCANTL. DRAFT

"UNIVERSAL"-CLASS
"N.L. CLARENCE"-CLEMENT

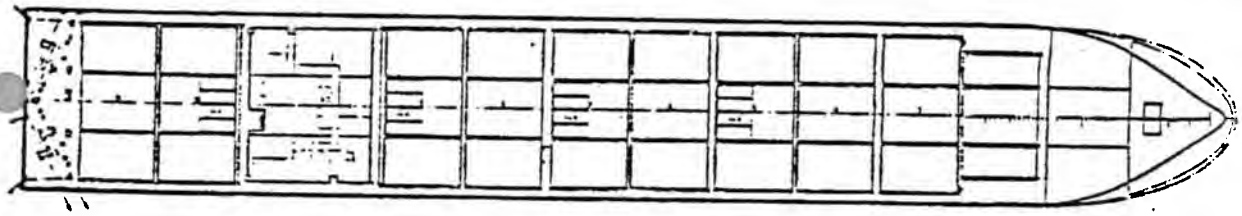
(Built 1983 - Owners design)



CAPACITY

	TEU
BELOW DECK =	1150
ON DECK 2T =	562
TOTAL 2T =	1712

Total 3T=2003



MEDIUM-SIZED GEARLESS CONTAINERVESSELS OF VARIOUS VINTAGE AND DESIGNERS.

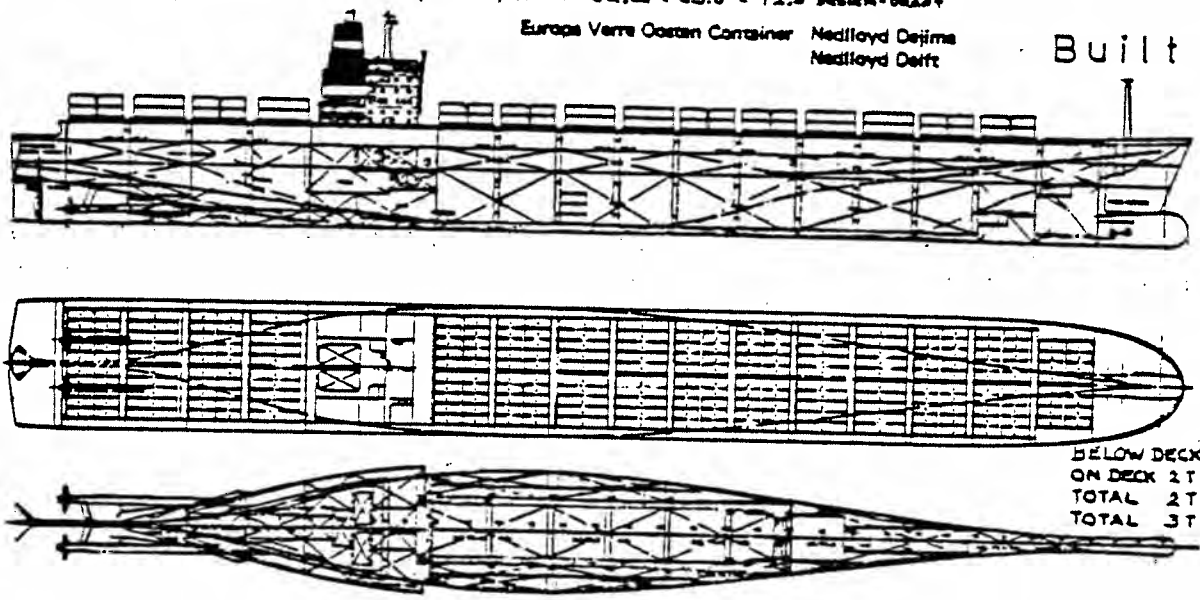
12.1.09

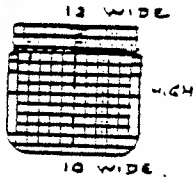
12

287.0 x 273.0 x 32.24 x 25.0 x 12.0 DESIGN-DRAFT

Europa Verre Oostan Container Nedlloyd Dejima
Nedlloyd Delft

Built 1973



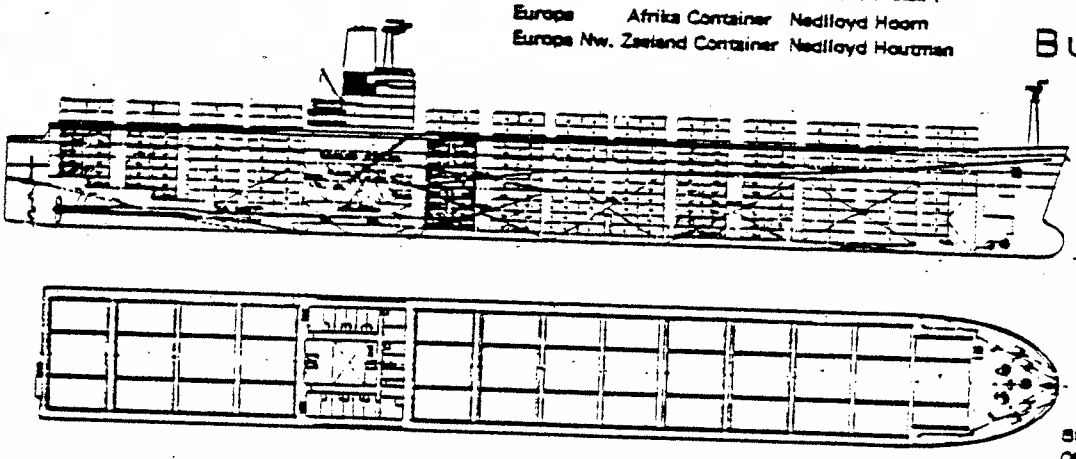


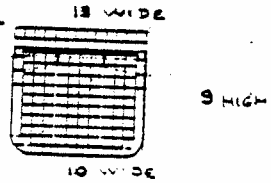
BELOW DECK = 2020 = 70
ON DECK 2T = 668 50
TOTAL 2T = 2688 120
TOTAL 3T = 2952

258.5 x 247.0 x 32.26 x 24.15 x 12.0 DESIGN-DRAFT

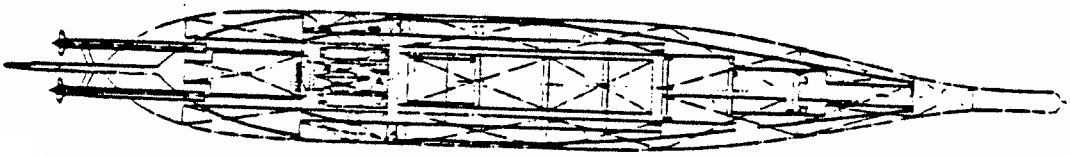
Europa Afrika Container Nedlloyd Hoorn
Europa Nw. Zeeland Container Nedlloyd Houtman

Built 1978





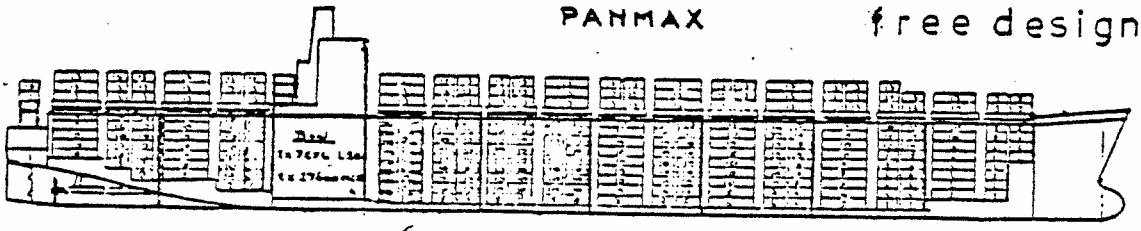
BELOW DECK = 1788 = 886 D
ON DECK 2T = 660 77
TOTAL 2T = 2448
TOTAL 3T = 2708 HOORN

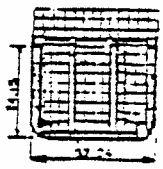


BELOW DECK = 1794 = 352 D
ON DECK 2T = 660 53
TOTAL 2T = 2454
TOTAL 3T = 2714 HOUTM

283.5 x 277.0 x 32.24 x 25.0 x 12.72 SCANTL. DRAFT

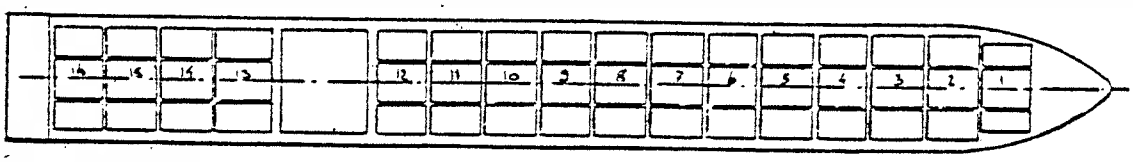
PANMAX free design 1983





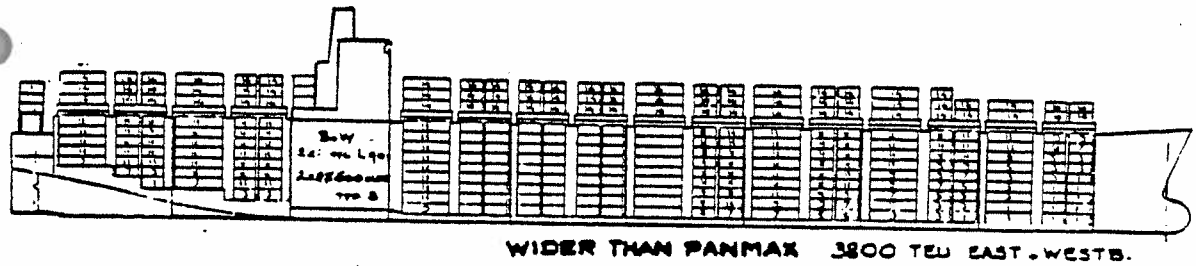
(FIGURE NOT ON SCALE)

BELOW DECK = 2264
ON DECK 2T = 842
TOTAL 2T = 3106
TOTAL 3T = 3470

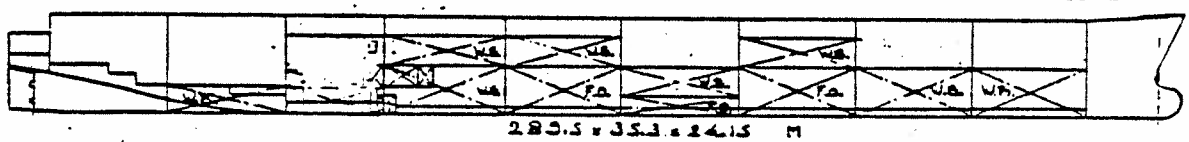


TRUNKLINE CELLULAR CONTAINERVESSELS OF VARIOUS VINTAGE -

SAME DESIGNER.



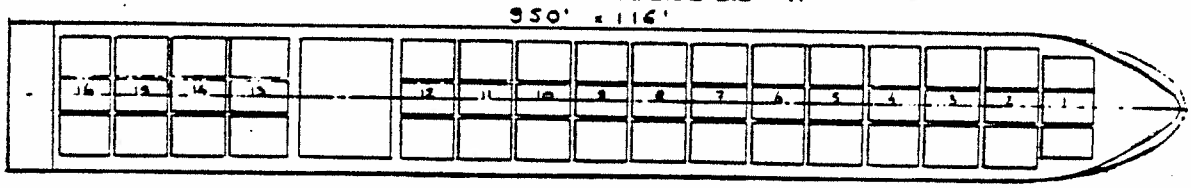
WIDER THAN PANMAX 3800 TEU EAST-WESTB.

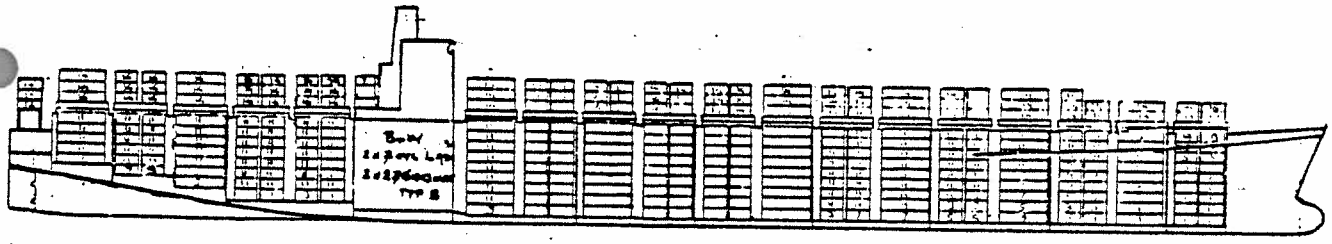


CAPACITY

IN HOLDS = 2499 TEU
ON DECK ST: 1300 "
TOTAL ST: 3799 "

285.5 x 35.3 x 24.15 M
950' x 116'

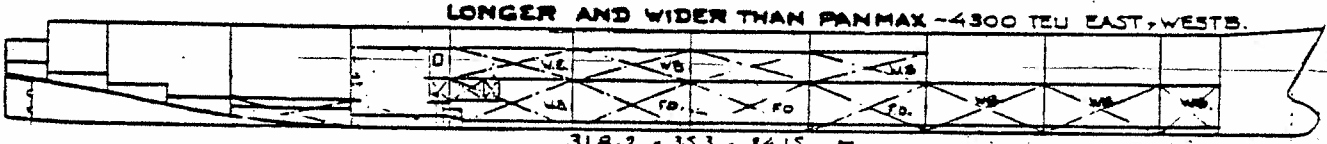




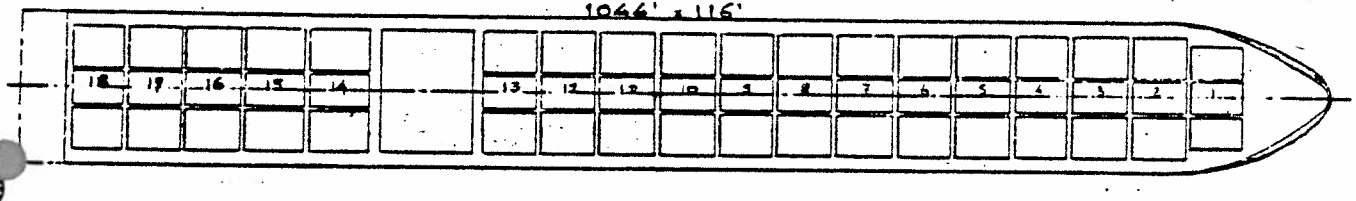
CAPACITY

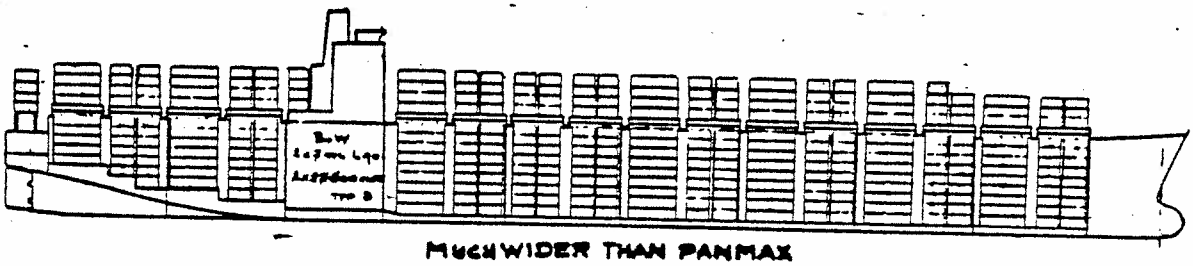
IN HOLDS = 2845 TEU
ON DECK ST = 1468 "
TOTAL ST: 4313 "

LONGER AND WIDER THAN PANMAX ~4300 TEU EAST-WESTB.

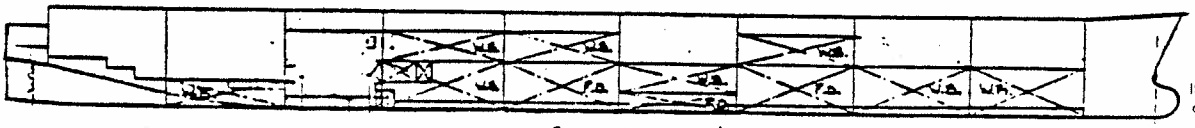


318.2 x 35.3 x 24.15 M
1046' x 116'





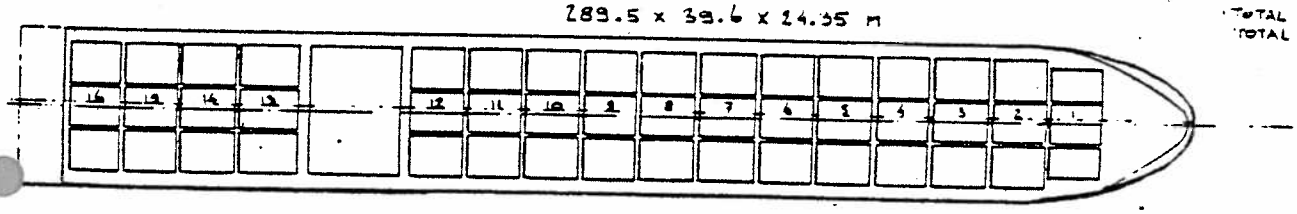
MUCH WIDER THAN PANMAX



CAPACITY

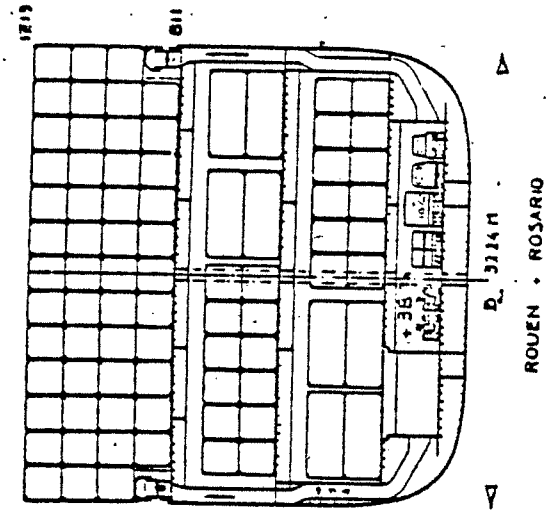
IN HOLDS = 2720 TEU
ON DECK ST = 1484 "
TOTAL ST = 4204 "
TOTAL ST = 4652 "

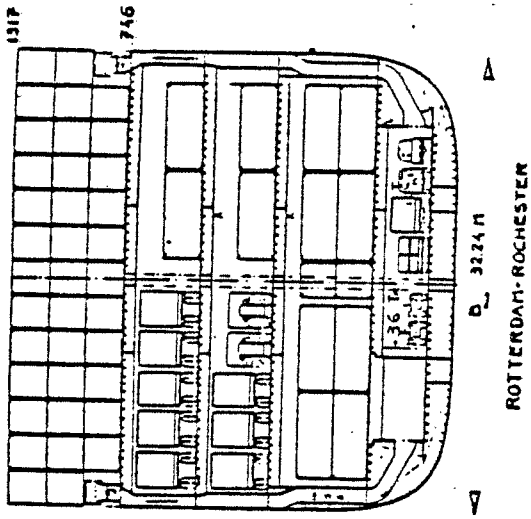
289.5 x 39.6 x 24.35 M



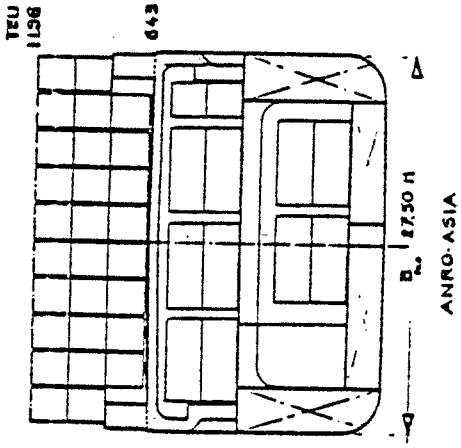
THREE "OVERSIZE" (') CELLULAR CONTAINERVESSEL DESIGNS.

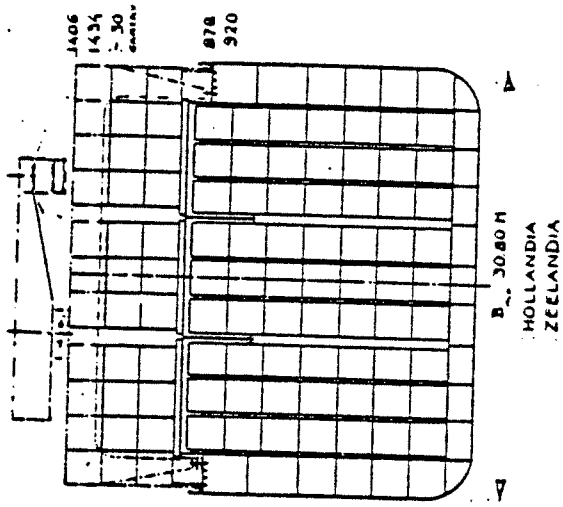
(') = I.E. OF A SIZE TOO BIG TO PASS THE PANAMA CANAL.

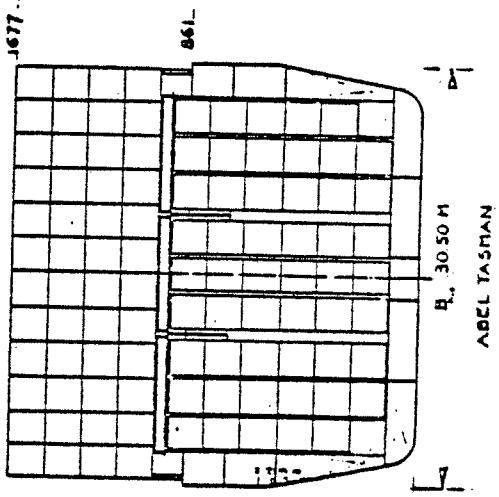


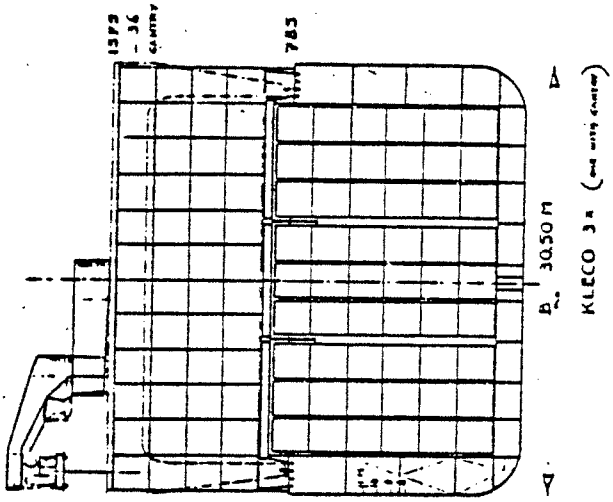


(PART) RORO-TYPE

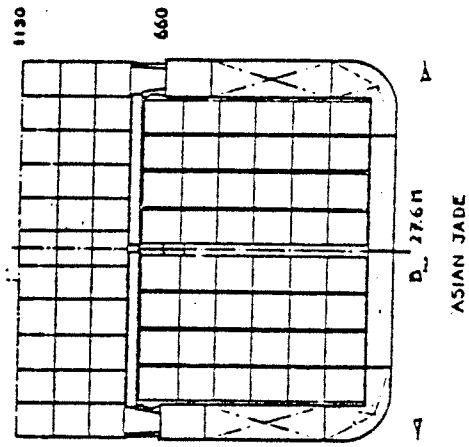




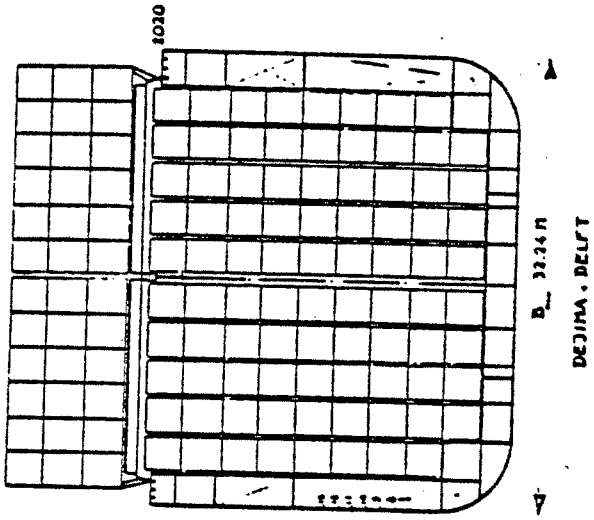


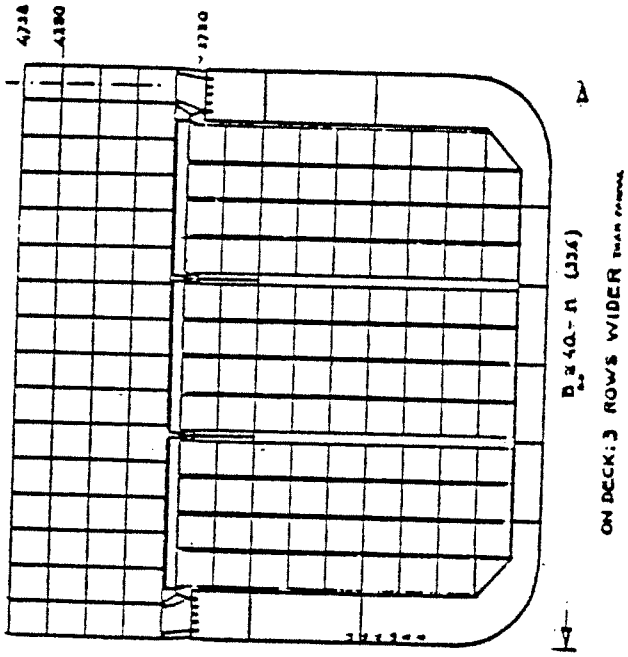


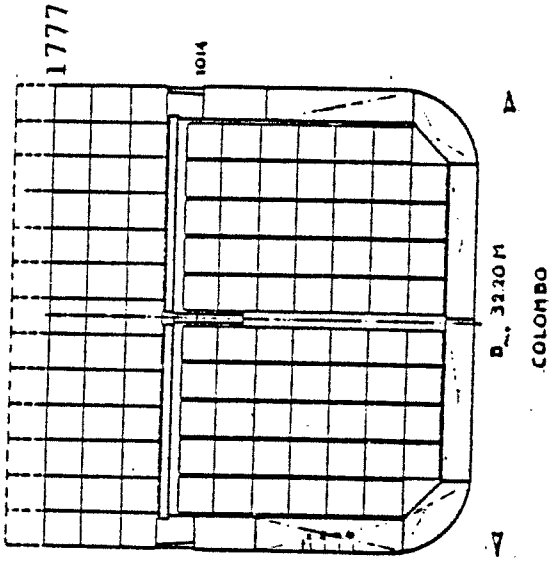
CELLULAR TYPE

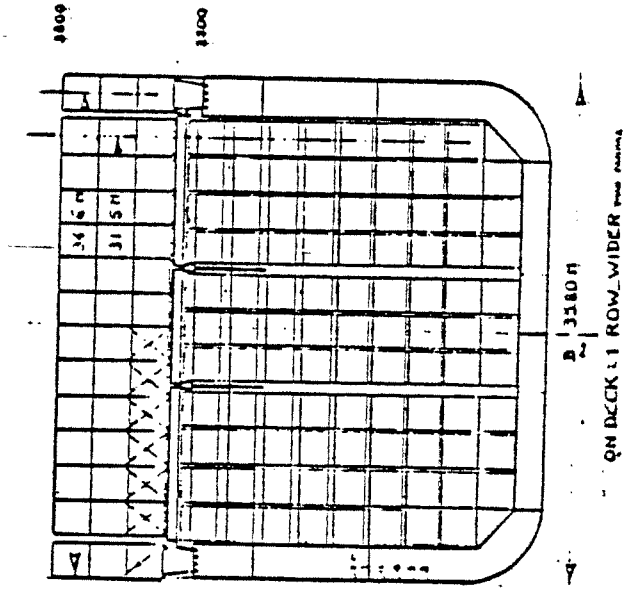


COMPARATIVE MIDSHIP CROSS-SECTIONS OF SOME MEDIUM-SIZED CONTAINER CARRIERS.









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-36
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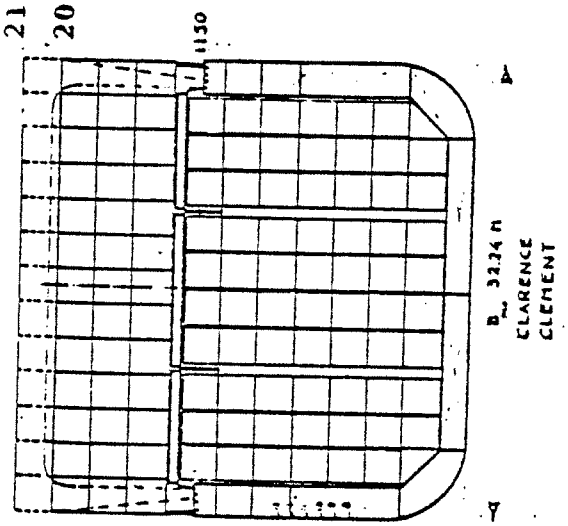
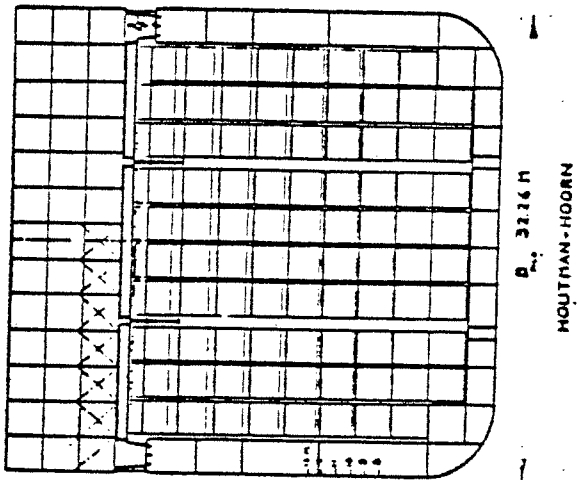
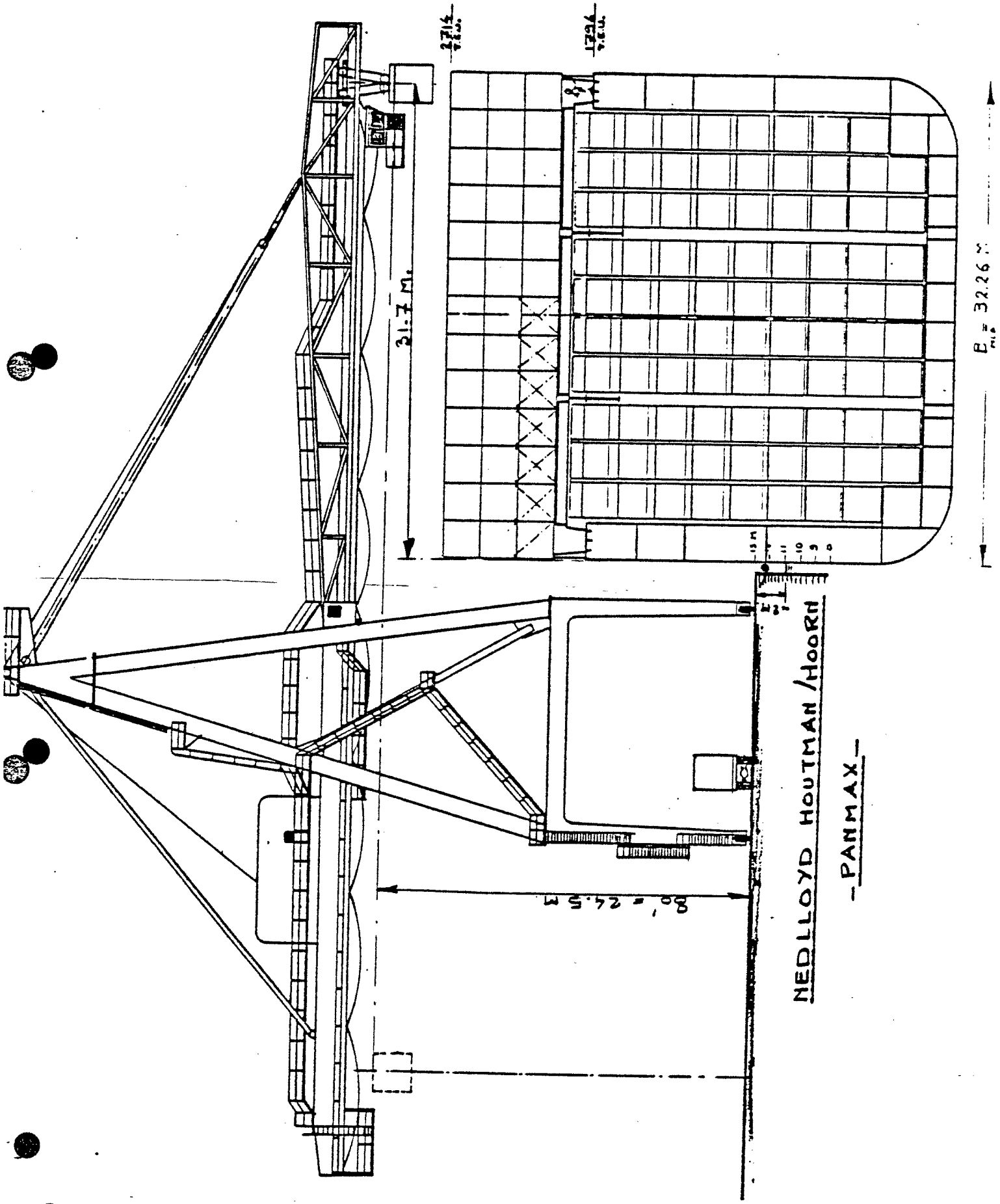
2003

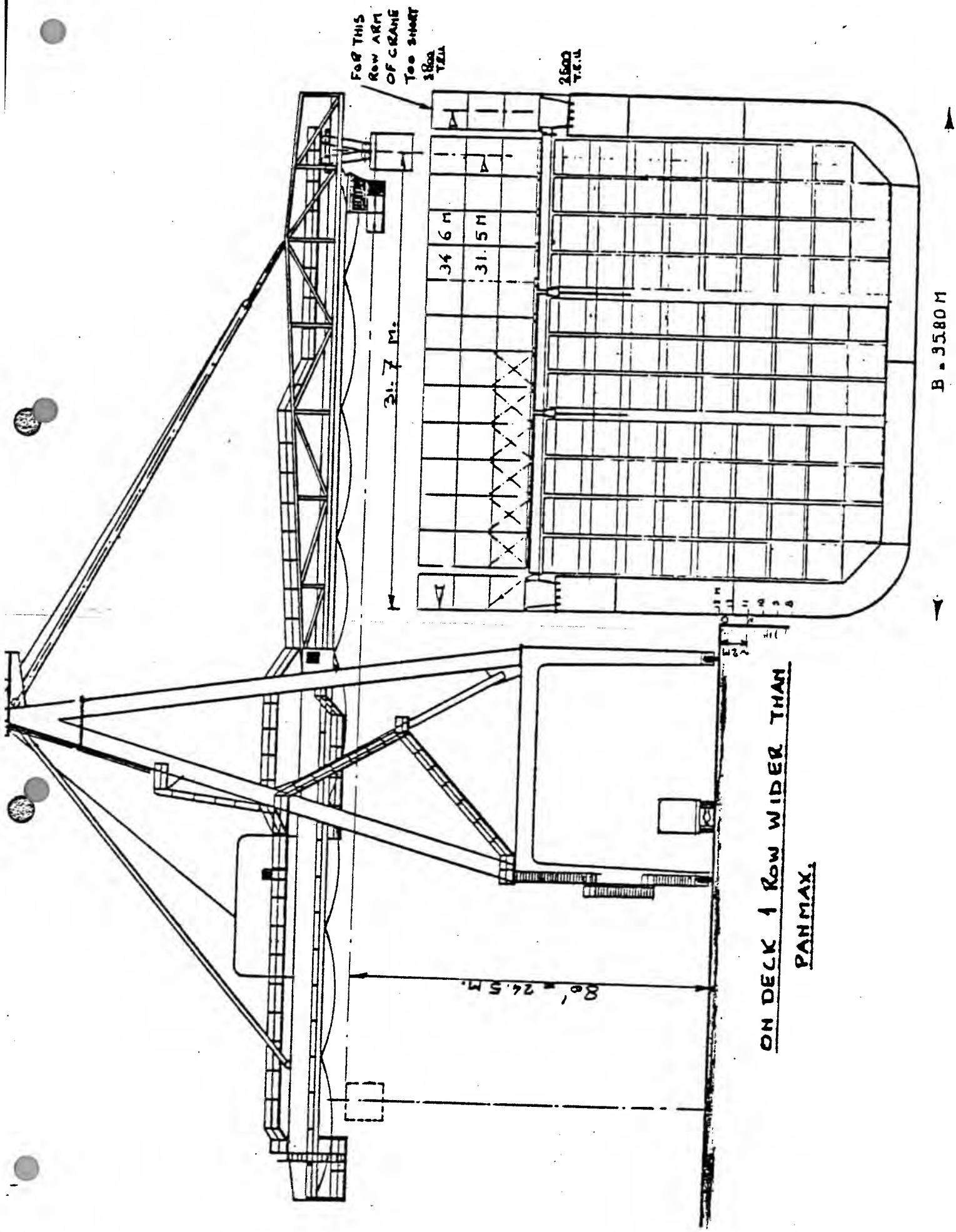




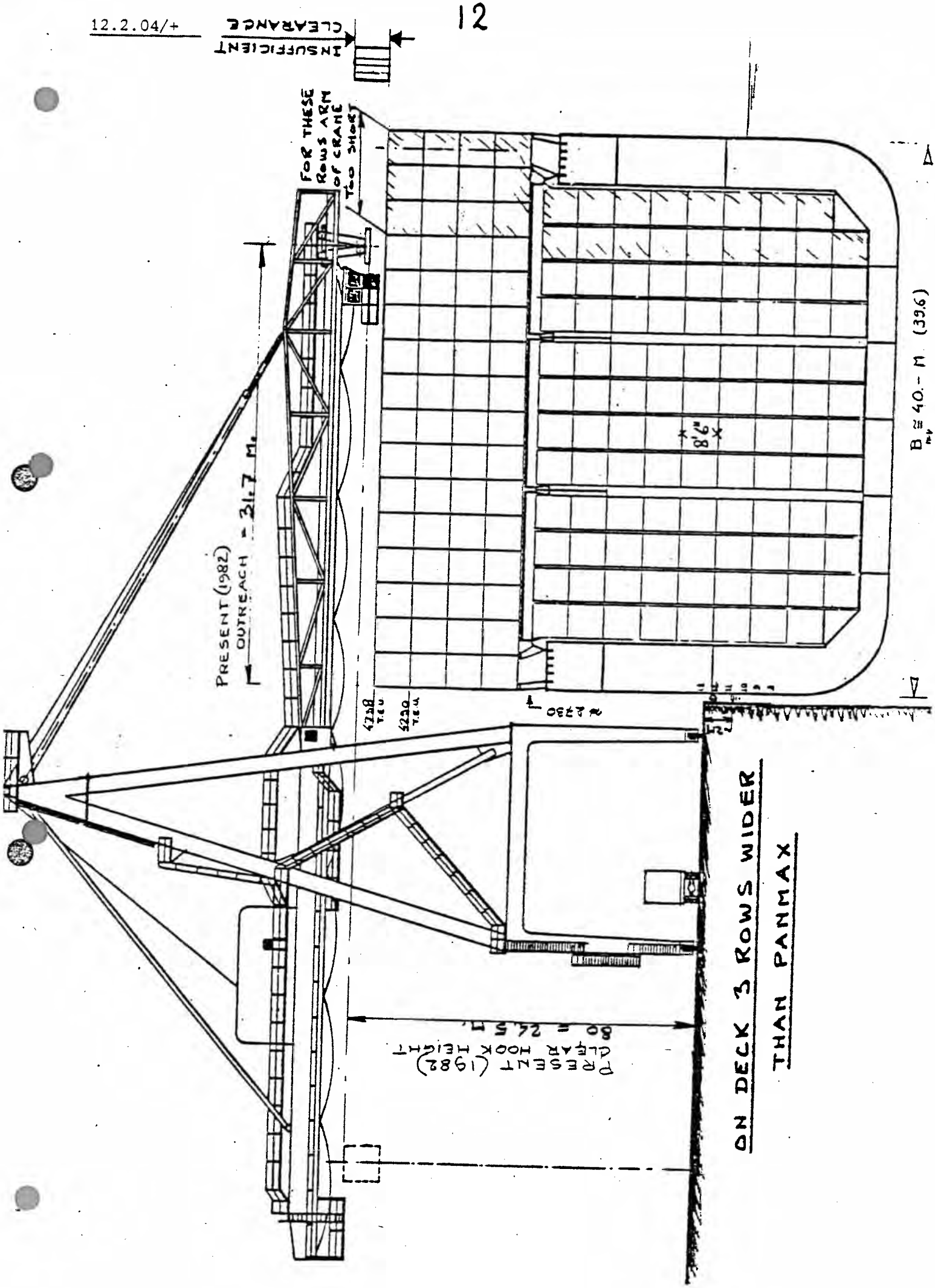
COMPARATIVE MIDSHIP SECTIONS OF SOME LARGER-SIZE CELLULAR CONTR.VSLS.



PRESENT-DAY SHORE CRANE IN RELATION TO PRESENT DAY "PANMAX" C.V.



PRESENT-DAY SHORE CRANE IN RELATION TO SHIP (1 BOX) WIDER THAN PANMAX.



PRESENT-DAY SHORE CRANE IN RELATION TO A SHIP OF 3 ROWS WIDER THAN PANMAX WIDTH

Capacity limited by beam

About 10 years ago, the Panama-size "Far East"-trade vessel (hull : 10 across/9 high + 3 tiers on covers) sailed with 10500 tons fuel oil in ship's bottom which kept the ship well upright.

- During consumption of the oil the weights were replaced by W.B. to maintain stability.
- Since oil crises speed dropped from 27 to 20,5 km.
- Steam turbines were replaced by diesels.Nowadays the necessary fuel oil for the trip is about 3500 ton.
- To keep the vessel well upright 10500 - 3500 = 7000 ton waterballast is necessary right from the start of the trip.
- Is it right to design ships which carry waterballast around the world?

13.0

CONTAINER-RORO VESSELS

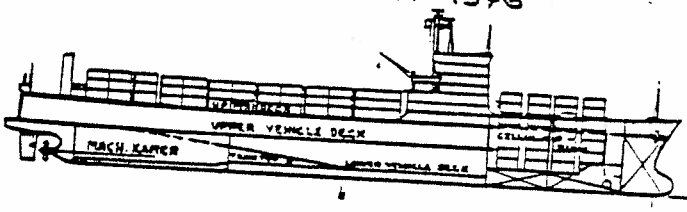
CONTAINER-RORO

13

181,76 x 168,0 x 27,5 x 16,75 x 9,0 DESIGN-DRAFT

Australia Malaisia RoRo Containerschip

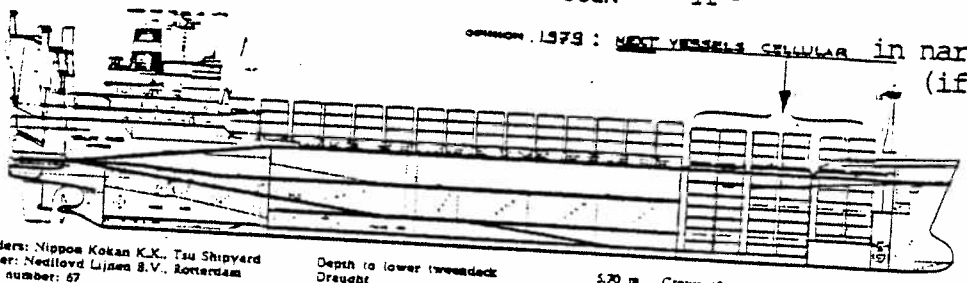
ANRO ASIA 1976



BELOW DECK = 522 = 134
ON DECK 2 T = 290 66
TOTAL 2 T = 812 200
TOTAL 3 T = 971

M/V. "NEDLLOYD ROUEN" – type

OPINION 1979 : NEXT VESSELS CELLULAR in narrow forepart. (if vessel is long enough).



Builders: Nippon Kokan K.K., Tsu Shipyard
Owner: Nedlloyd Lijnen B.V., Rotterdam
Yard number: 67
Type: ro ro vessel
Delivery: October 12th, 1978

Tonnage:

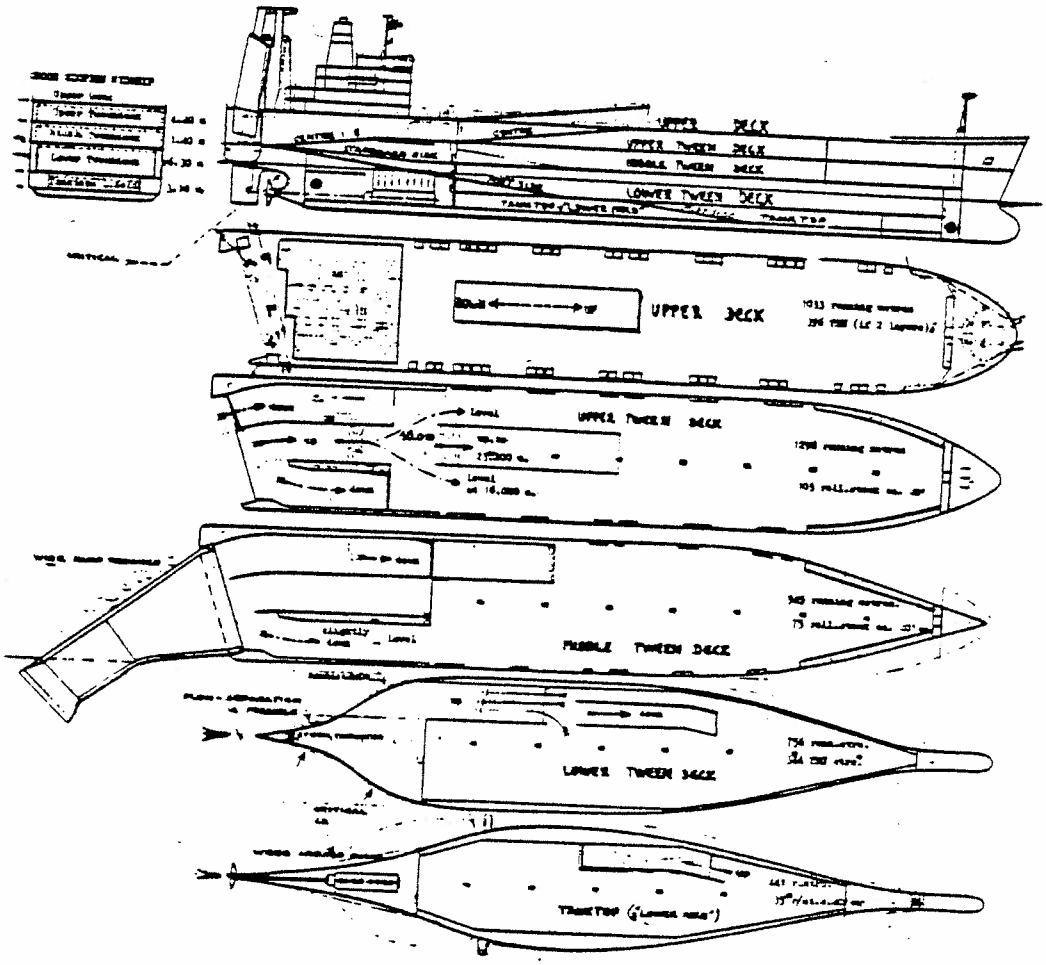
	21 144.61 GRT
	11 698.77 NRT
Deadweight	29 218 t
Length o. a.	212.10 m
Length b. p.	198.83 m
Moulded breadth	32.24 m
Depth to maindeck	20.24 m
Depth to upper tweendeck	13.05 m
Depth to lower tweendeck	5.70 m
Draught	10.72 m

Speed (16 686 kW): 20.66 kn

Classification:
L.R. – 100 A 1

Propelling machinery:
Sumitomo/Sulzer-Diesel engine
type: 8 RND 90 M
19 711 kW (28 800 HP) at 116 rpm
Fuel consumption: 156.56 g/HP/h
1 bow thruster
1 stern thruster
1 gyro-fin stabilizer
gyro compass and automatic pilot (Anschütz)

Crew: 40 persons.
1 stern ramp
1 weather-tight stern door 25.00 × 7.00 m
1 gas-tight rampway cover 5.60 × 35.00 m
1 gas-tight door 9.04 × 6.70 m
1 gas-tight door 9.05 × 6.33 m
1 weather-tight casing door 9.10 × 6.45 m
Loading gear: 1 crane 3 t
8 electric hydraulic automatic mooring winches of 12 t
2 electric hydraulic windlasses of 31 t



USA Arabische Golf RoRo Trailerschip

PROJECT 1976

NEDLLOYD ROTTERDAM

(B) THE CONTAINER-RO-RO VESSEL

To Middle East countries with quickly expanding trades RoRo-cum-Container vessels have proved a necessity. (')

- These ships are to carry containers, rolling stock and project cargoes, comprising vulnerable (assembly-type) pieces of cargo necessary for building factory halls, housing, manufacturing plants, bridges, ports and the like.
- Therefore, the roro part of such vessel is to be suitable for many kinds of wheeled cargoes, such as roadmasters, tractors, wheelloaders, cement trucks, cranes, movable houses, boats, a.s.o.
- Pipes, odd boxes, palletized cargo, CKD's are to be mounted on trailers.
- At the developed side of the line it is possible to handle the upperdeck containers by the shore crane, simultaneously with the fast roro operation via the ramp.

- An integrated crew is necessary because of the many improvisations during the cargo handling process. However, in comparison with the cellular type of ship, much extra work is to be done by the men, such as for instance:-
  - Complicated cargo stowage planning / Handling of the roro cargo by forklifts and tugmasters sometimes by ships crew. RoRo shipboard equipment to be repaired on board.

- For ISO-box cargo, this type of vessel is like a "deck-barge" for the boxes (no cells). For rollable cargo - either with own wheels or "borrowed" ones - the vessel underdecks works like a "pencil-box".

- The roro cargoes are handled inside the holds partly by means of forklift trucks which load the chassis/lorries/or "cont-rollers". These chassis are pulled out of the ship by tug-masters via internal rampways, ending in a mouth at the stern where a wide quarterramp leads to the quay.

(') i.e. all cargo had to be put on wheels for bypassing the congested wharves.

- Nedlloyd's roro vessels have a rampway on P.S. leading to L.T.D. and tanktop, another rampway, at centreline, leading towards upperdeck and a third rampway - on S.B. - almost horizontally leading towards an uppertweendeck (on the 3-decker) recpectively towards a middletweendeck (on the 4-decker vessel).
- Width of the internal rampways is 9.1 metres, width of the "hall aft" is 24 m., length is 30 m., height of the "hall aft" is 7 m.
- Heights of tweendecks should be 6.35 metres.(Altern.4.4m for trailers) ,
- Ramps should be simple in construction and operation! (not too many gadgets)

EXAMPLES.

Nedlloyd Rouen/Rosario (1978-1979) 212 x 32.24 x 20. 4 m (3 decks)
Nedlloyd R'dam/Rochester(1978-1979) 196 x 32.24 x 23.25 m (4 decks)

Development of this type container-RoRo vessel (4-deck trailerminded) (3-deck containerminded)might be in making the forepart of the vessel cellular. This trend was shown already in 1968 by A.C.L. (Atlantic Crown type) and by the '75-built - Nedlloyd-affiliated - ANRO ASIA vessel.

- There is a stronger urge to carry ISO boxes in comparison to proper cargo for RoRo.
- Also, the utilisation of space in the forebody is better by vertical stowage of containers between guides, backed up with trimtanks in the ship's sides. Where holdspace gets narrow the usually rapid roro stowage becomes more difficult.
- The safety of the roro ship would be improved by the watertight subdivision of the forepart - this being the most-often hit part in case of collision.
- Integral reefercontainers could be stowed in these underdeck cells, if sufficient aircooling is applied.

(1) The heavy construction of the decks and complicated ramp makes the vessel more expensive in comparison with the cellular container vessel.

(2) The large span of the deckbeams between the vessel's side and the centreline pillar, results in a heavy steel construction.

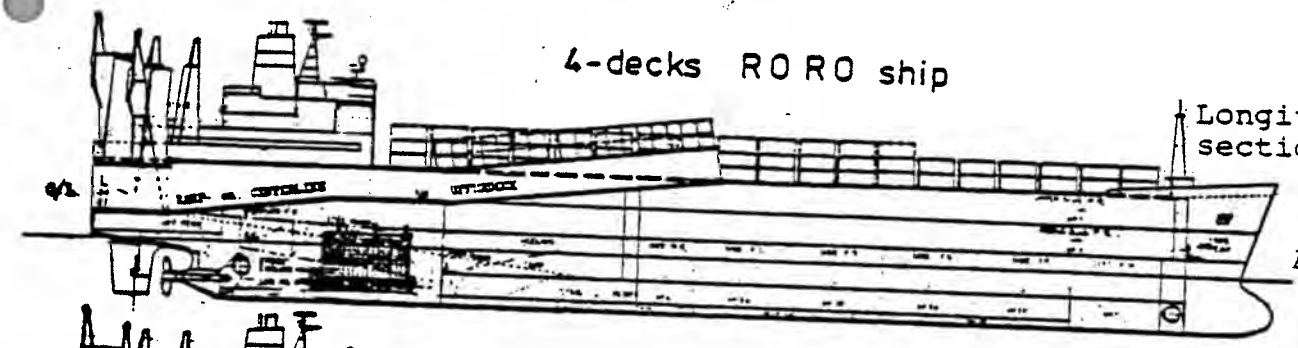
(3) Critical are stress concentrations at beam/web and beam/pillar connections because of deletion of rounded corners (knees). The horizontal movement of cargoes, trailers, containers on chassis as well as stacking space requirements, should not be hampered by protruding knees!

The ships should be single screw or twin-screw diesel-propelled, speed 18-21 knot range. Blockcoefficient 0.60 - 0.68.

- In case of single screw, the main-engine is well arranged underneath the centre ramp - which leads upward towards the upper deck, and a key in the design is that taking out pistons should just be possible.
- Vibration and noise, caused by the propeller, are much more difficult to combat because of the fact that the accommodation is situated right over the propeller and moreover the flat stern above the propeller enables a strong transfer of pressure fluctuations caused by the cavitating propeller blades in topposition
- The hull design of the single screw vessel is difficult in its compromise because the underwater hull ahead of the propeller should be very fine in order to avoid vibrations on the one hand - whilst on the other hand the heavy stern-part of the vessel asks for buoyancy-aft.
- The single-screw vessel will have a strong tendency to trim astern - which has to be compensated by extra waterballast in the ship's forepart.
- Extra waterballast forward, however, increases the hogging moment.
- The 3-lane entrance aft is asking for a wide transom stern which has to be combined with the fine underwater hull resulting in flat stern-sections.
- The flat wide stern does significantly increase stability.
- On the other hand, slamming will be excited by a heavy swell at zero speed or at low speed - a rather strange experience. Up till now, no damage to the structure occurred.

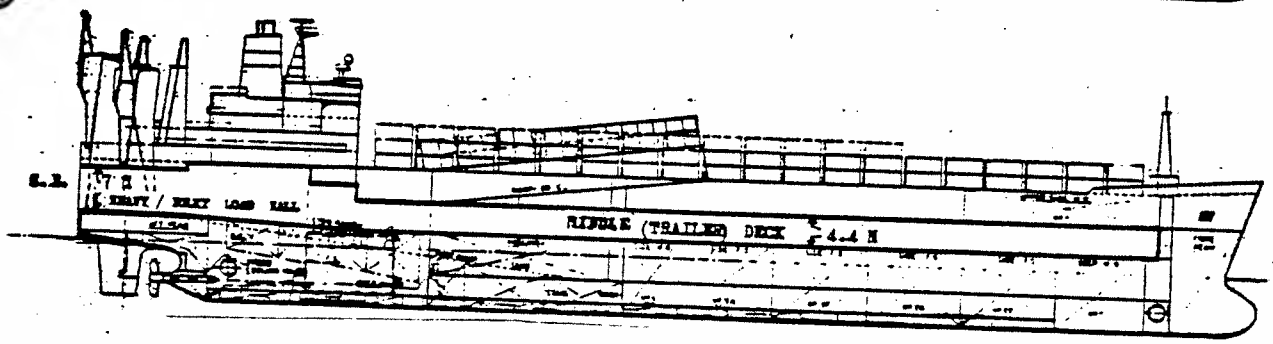
4-decks RORO ship

Longitudinal sections:



At C/L

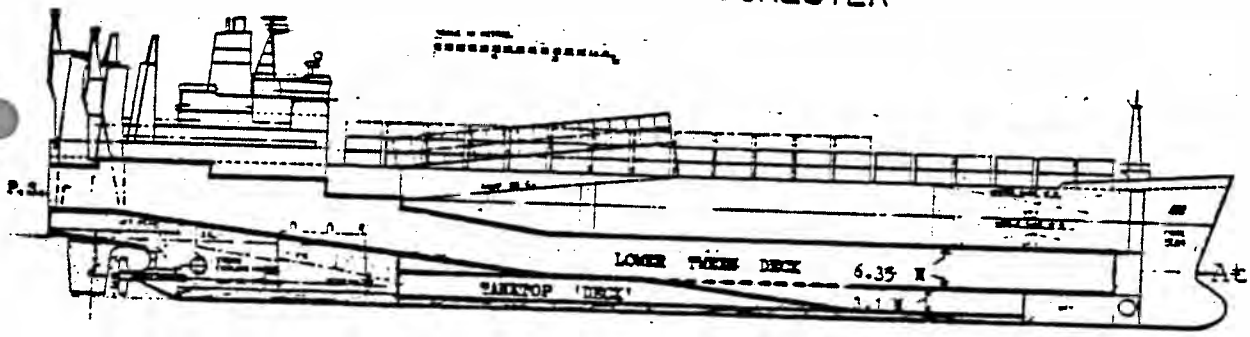
Next to C/L



At SB side

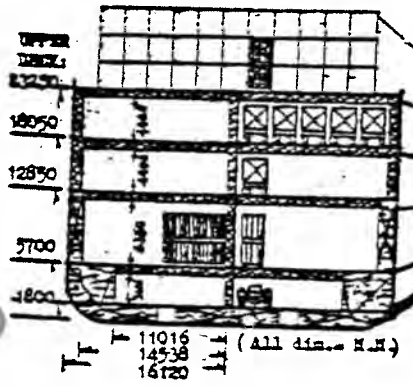
196.5 × 183.1 × 32.24 × 23.15 × 10.0 DESIGN DRAFT

NEDLLOYD ROTTERDAM
NEDLLOYD ROCHESTER



At Port Side

Midship's Cross Section ← 4 - DECKS RORO →



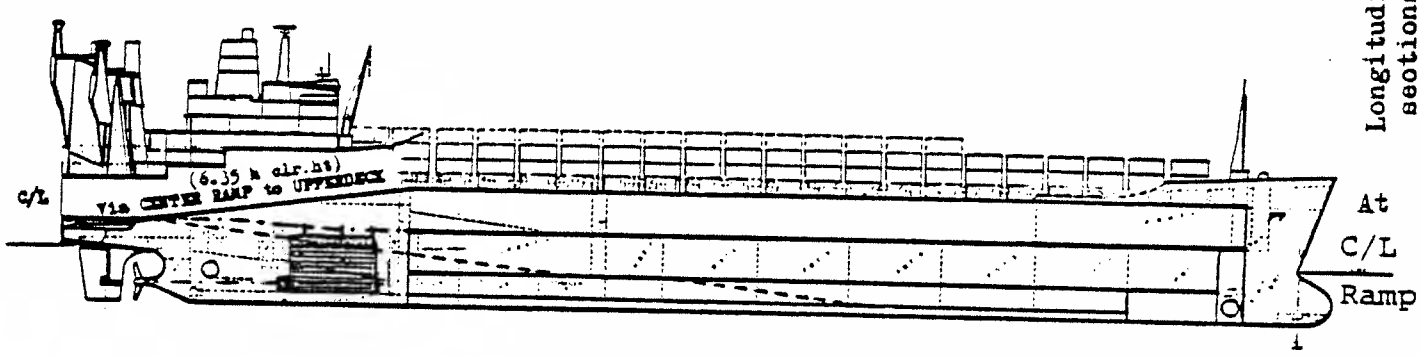
CARGO CAPACITIES

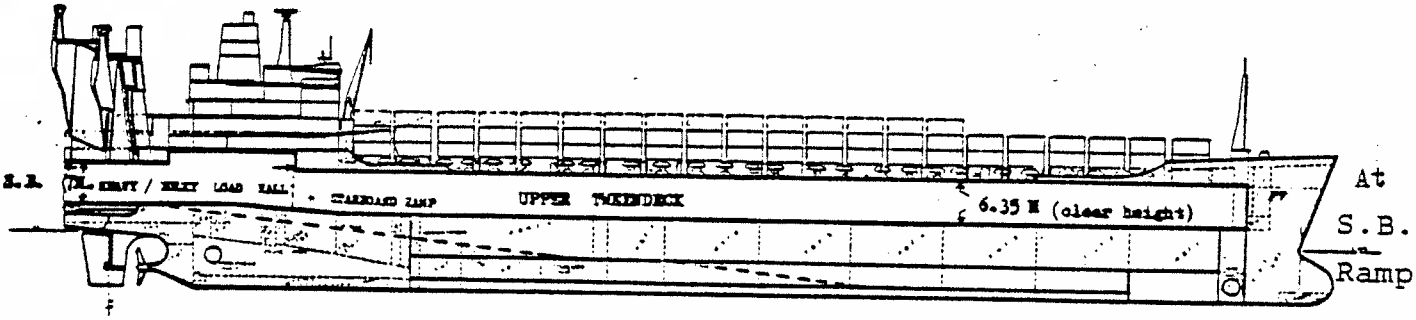
4 - DECKS RORO	CARGO CONDITION ALTERNATIVES 1	2	3	4
Positions in ship	[illegible]	[illegible]	[illegible]	[illegible]
Upper Lower Deck Comp	1.833 × [illegible]	- 177 [illegible] 704 [illegible]	- 177 [illegible] 704 [illegible]	- 177 [illegible] 704 [illegible]
Upper Tweendeck	1.298 × 183	183 -	183 -	- 228 [illegible]
Middle Tweendeck	945 × 75	75 -	75 -	- 168 [illegible]
Lower Tweendeck Lower Deck Comp	756 × 60	60 -	- 132	- 132
TankTop Deck	441 × 35	35 -	35 -	35 -
Total	4.473 × 356 [illegible]	272 × 171 [illegible]	213 × 693 [illegible]	35 × 1317 [illegible]

[illegible]
[illegible]
[illegible]

3-decks R O R O ship

Longitudinal sections

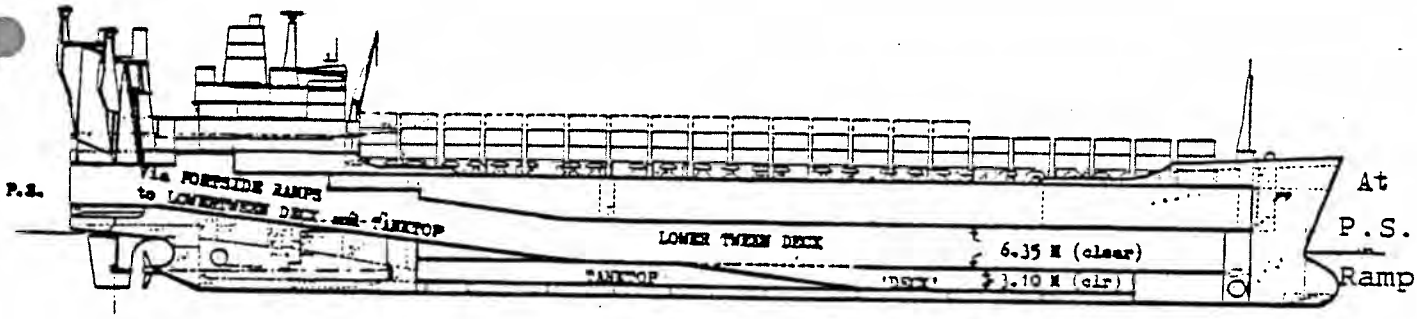




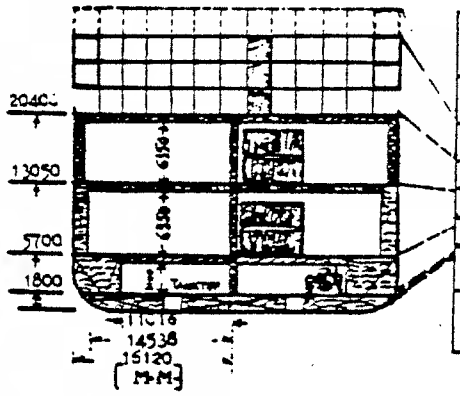
212.1 x 198.8 x 32.24 x 20.4 x 10.0 DESIGN-DRAFT

NEDLLOYD ROUEN

NEDLLOYD ROSARIO



Cross section midship ← 3 - DECKS RORO → CARGO CAPACITIES



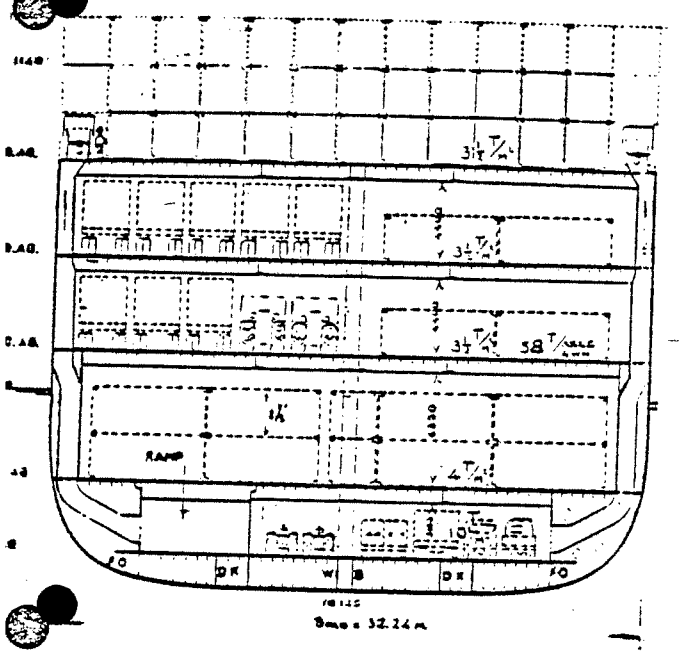
3 - DECKS RORO	CARGO CAPACITIES ALTERNATIVE											
	1		2		3		4		5			
Position in ship	Standard sections	Walks & 12.6 m	Walks	TEU	Walks	TEU	Walks	TEU	Walks	TEU	Walks	TEU
Upperdeck Lowl. ramp	1.176	+ 91	-	185 (1) 717 (2)	-	185 (1) 717 (2)	-	185 (1) 717 (2)	-	185 (1) 717 (2)	-	
Upper/Tweendeck Lowl. ramp S.B.	1.232	+ 180	180	-	180	-	-	453	-	453		
Lower/Tweendeck Lowl. ramp P.S.	540	+ 71	71	-	-	358	71	-	-	358		
Tweetop hold	540	+ 30	30	-	30	-	30	-	30	-		
TOTAL	3.488	+ 300	309	+ 902	130	+ 1260	109	+ 1356	30	+ 1713		
	300	Walks	Walks	TEU	Walks	TEU	Walks	TEU	Walks	TEU		

(1) one tier
(2) 2 tiers

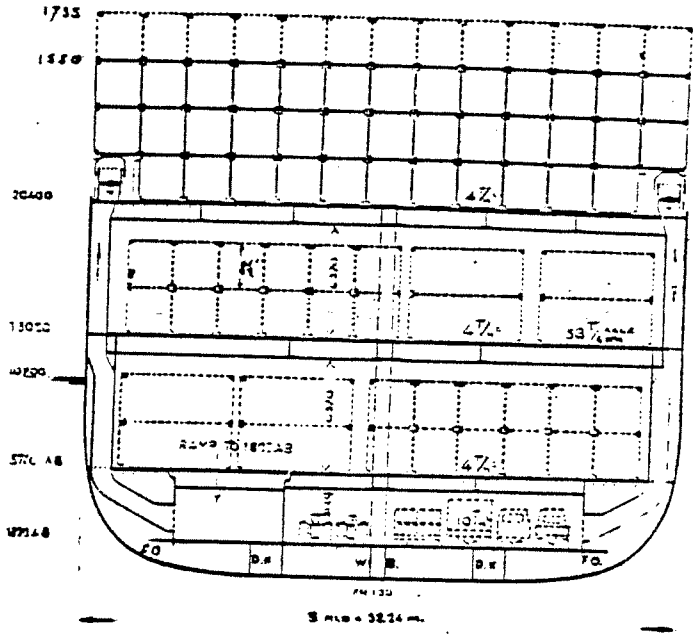
Main particulars - and differences.

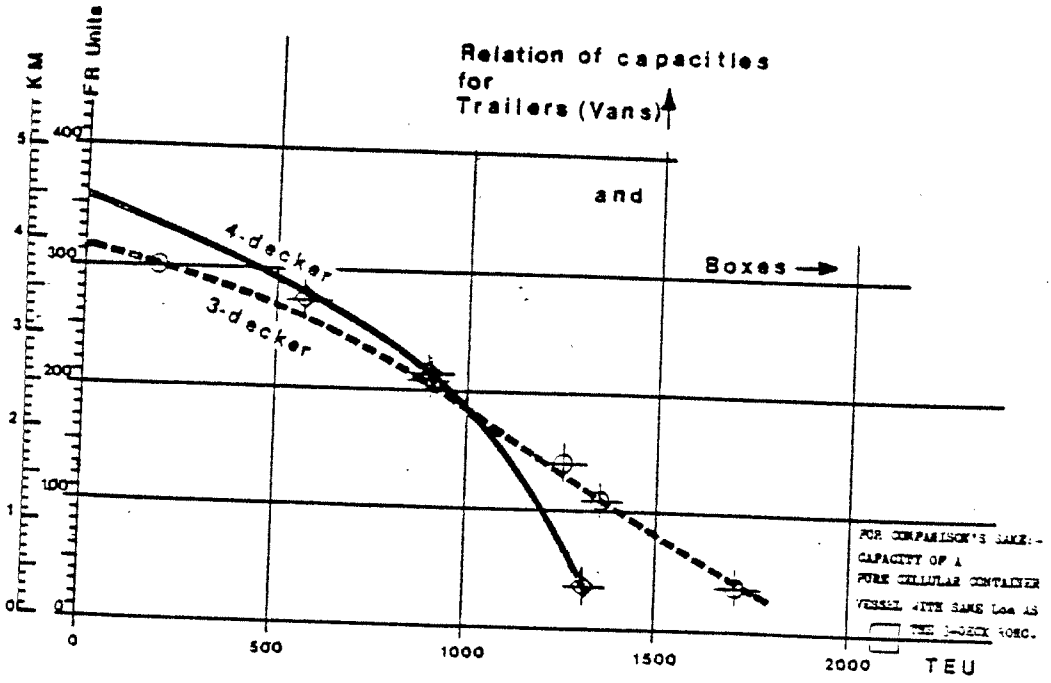
Vessels' names,	Nedlloyd Rotterdam Nedlloyd Rochester			Nedlloyd Rouen Nedlloyd Rosario		
Number of decks	4 decks			3 decks		
Permissible deck loads and clear heights						
Upperdeck : containers stackloads	Max. 3 tiers 20' = 40 T/stack 40' = 80 T/stack			Max. 4 tiers 20' = 60 T/stack 40' = 80 T/stack		
specific load	Ton/4W axle 58—66 T	Ton/m² 3½ T		Ton/4W axle 58—66 T	Ton/m² 4 T	
	Ton/axle	Ton/m²	Height (M.M.)	Ton/axle	Ton/m²	Height (M.M.)
Uppertween dk (15½ mm)	58—66 T	3½ T	4400 (cl)			
Webs-girders (each deck)			780 ÷ 25			950 ÷ 25
Freeboard deck (15½ mm)	58—66 T	3½ T	4400 (cl)	58—66 T	4 T	6350 (clr)
Lowest tweendeck (15½ mm)	58—66 T	4 T	5350 (cl)	58—66 T	4 T	6350 (clr)
Tanktop deck	58—66 T	10 T	3100 (cl)	58—66 T	10 T	3100 (clr)
Spacing longitudinals	586 mm					
Spacing of webs	3 x 780 = 2340 mm					
Width of ventilation ducts	a number of 1560 mm and a number of 2340 mm					

4 DECK RORO



3-DECK RORO





13.3.04

13

RPM - SPEED - SHP - FUEL UNDER OPTIMAL CONDITIONS

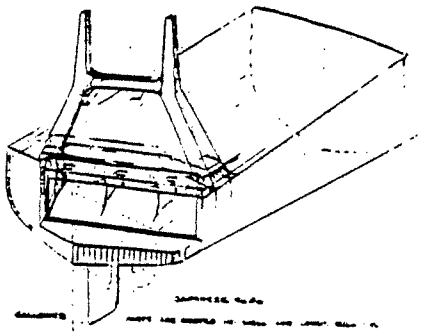
NEDLLOYD ROUEN NKK No. 57 DEL 12 OCT 78

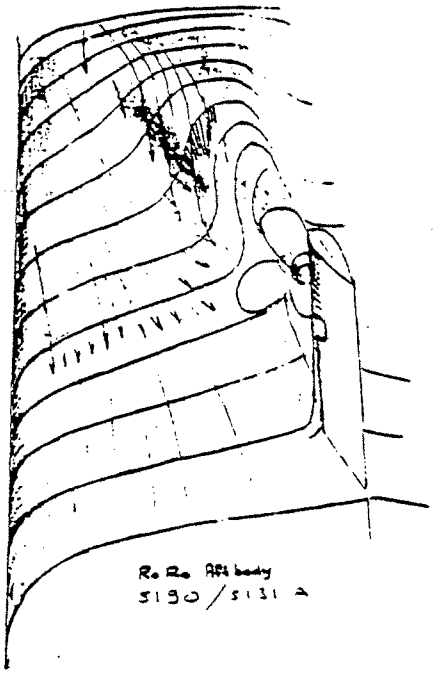
NEDL[illegible]D ROSARIO NKK No. 58 DEL 19 JAN 79

L OA = 212.10 M
L OP = 198.88 M
B MLD = 32.34 M
H MLD = 20.46 M

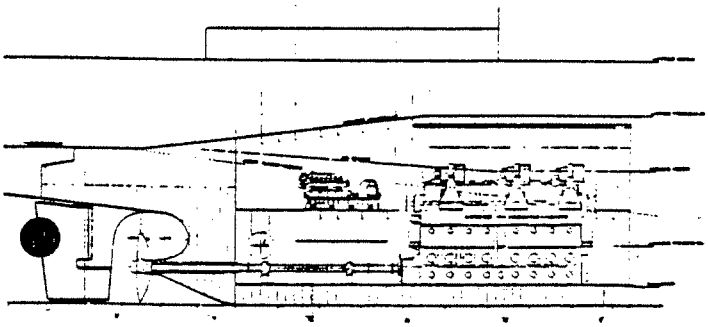
	RPM	TRIALS 27 SEPT. 78 BALLAST + DRAFT 6.84 F = 6.46				6.4 M				DESIGN DRAFT 10.0 M				MAX. DRAFT 10.7 M + EXTRA SEA MARGIN 8% ON TRIALS			
		KN				KN	SHP	tons/day	tons/1000'	KN	SHP	tons/day	tons/1000'	KN	SHP	tons/day	tons/1000'
DEAD SLOW	30	5.1	0.1 HOGG A = 7.43							MODEL BASIN WAGENINGEN TRIAL CONDITIONS				ESTIMATED			
TORSION	33	6.0															
CRITICAL	34																
	40	6.8	SMOOTH HULL			SMOOTH HULL CALM SEA				SMOOTH HULL CALM SEA							
SLOW	45	7.7	CALM SEA														
	50	8.6															
	55	9.4															
HALF	60	10.3															
	65	11.2															
	70	12.0															
	75	12.9															
	80	13.7															
FULL	85	14.6															
	90	15.8															
	95	16.3	SHP	tons/day	tons/1000'												
	100	16.8	11,800	46	114	17.0	11,900	47	115	17.2	12,200	48	117	17.1	12,500	50	121
	101	17.1	12,000	47½	116	17.2	12,300	48	118	17.3	12,600	50	120	17.2	12,900	51	123
	102	17.3	12,400	49	118	17.5	12,700	50	120	17.5	13,000	51½	122	17.3	13,300	53	126
	103	17.6	12,700	50	119	17.7	13,000	51	121	17.6	13,300	52½	124	17.5	13,700	54	128
	104	17.8	13,100	51½	121	17.9	13,400	53	123	17.8	13,700	54	126	17.6	14,100	55½	130
	105	18.1	13,500	53	122	18.1	13,800	54½	125	17.9	14,100	55½	129	17.7	14,600	57	133
	106	18.3	13,900	54½	124	18.2	14,200	56½	127	18.1	14,500	57	131	17.9	14,900	58	136
	107	18.6	14,300	56	126	18.4	14,600	57	129	18.2	15,000	58½	134	18.0	15,400	60	139
	108	18.8	14,700	57½	128	18.6	15,000	59	131	18.4	15,400	60½	136	18.1	15,800	62	142
	109	19.0	15,100	59	130	18.8	15,400	60	133	18.5	15,800	62	139	18.2	16,200	63½	146
	110	19.2	15,500	60½	132	19.0	15,900	62	136	18.7	16,300	64	142	18.4	16,700	65	149
	111	19.4	15,900	62	134	19.1	16,300	64	139	18.8	16,700	65½	146	18.5	17,100	67	151
	112	19.6	16,400	64	136	19.3	16,800	65½	141	19.0	17,200	67	147	18.6	17,600	68½	154
	113	19.8	16,800	65½	138	19.5	17,200	67	143	19.1	17,600	68½	149	18.7	18,100	70½	157
	114	19.9	17,200	67	140	19.6	17,600	68½	146	19.3	18,100	70½	152	18.9	18,600	72½	160
	115	20.1	17,700	69	143	19.7	18,100	70½	149	19.4	18,600	72½	155	19.0	19,100	74	163
	116	20.3	18,100	70½	146	19.9	18,600	72½	152	19.5	19,100	74	158	19.1	19,600	76	166
	117	20.5	18,600	72½	147	20.1	19,100	74	155	19.7	19,600	76	161	19.2	20,100	78	169
	118	20.6	19,100	74	150	20.3	19,600	76	157	19.8	20,100	78	164	19.3	20,600	80	173
	119	20.8	19,600	76	153	20.4	20,100	78	160	19.9	20,600	80	167	19.4	21,100	82	176
	120	21.0	20,100	78	156	20.5	20,600	80	163	20.1	21,100	82	170	19.5	21,700	84½	180
	121	21.2	20,700	80½	158	20.7	21,200	82½	166	20.2	21,700	84½	174	19.6	22,200	86½	184
	122	21.3	21,300	82½	161	20.9	21,700	84½	169	20.3	22,200	86½	177	19.8	22,800	88½	187
	123	21.5	21,700	85	165	21.0	22,200	87	172	20.6	22,700	89	181	19.9	23,300	91	191
	124	21.6	22,200	87	168	21.1	22,700	89	175	20.6	23,300	91	186	20.0	23,900	93½	196
85%	125	21.8	22,800	89½	171	21.3	23,300	91	179	20.7	23,900	93½	188	20.1	24,500	96	199
	126	21.9	23,300	92	174	21.4	23,900	94	183	20.9	24,500	96½	192	20.2	25,100	98½	203
	127	22.0	23,900	94	178	21.5	24,400	96	186	21.0	25,000	98½	195	20.3	25,600	101	207
	128	22.1	24,400	96	181	21.7	25,000	98½	189	21.1	25,600	101	199	20.4	26,200	103½	211
	129	22.2	25,000	98½	186	21.8	25,600	101	192	21.2	26,200	103½	203				

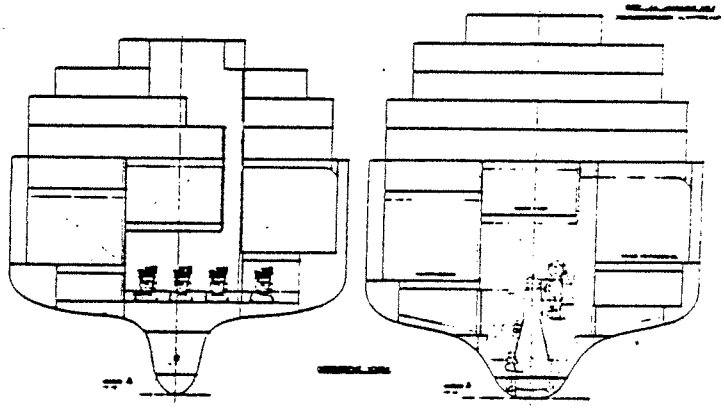
e-propeller e = 6500, 4 BL — Curved — 26.2 T
pitch mean = 5886, Fo/F = 0.728



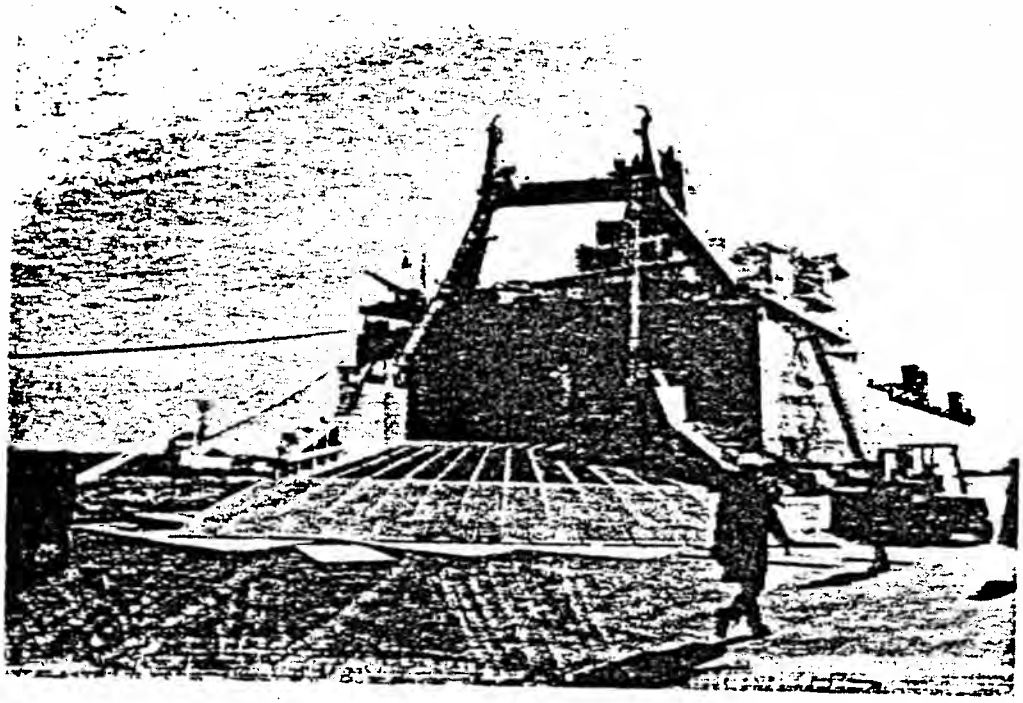


he rudder is 3% of Lpp and the rudder area is 2.8 per
ent of Lpp x 10 m.

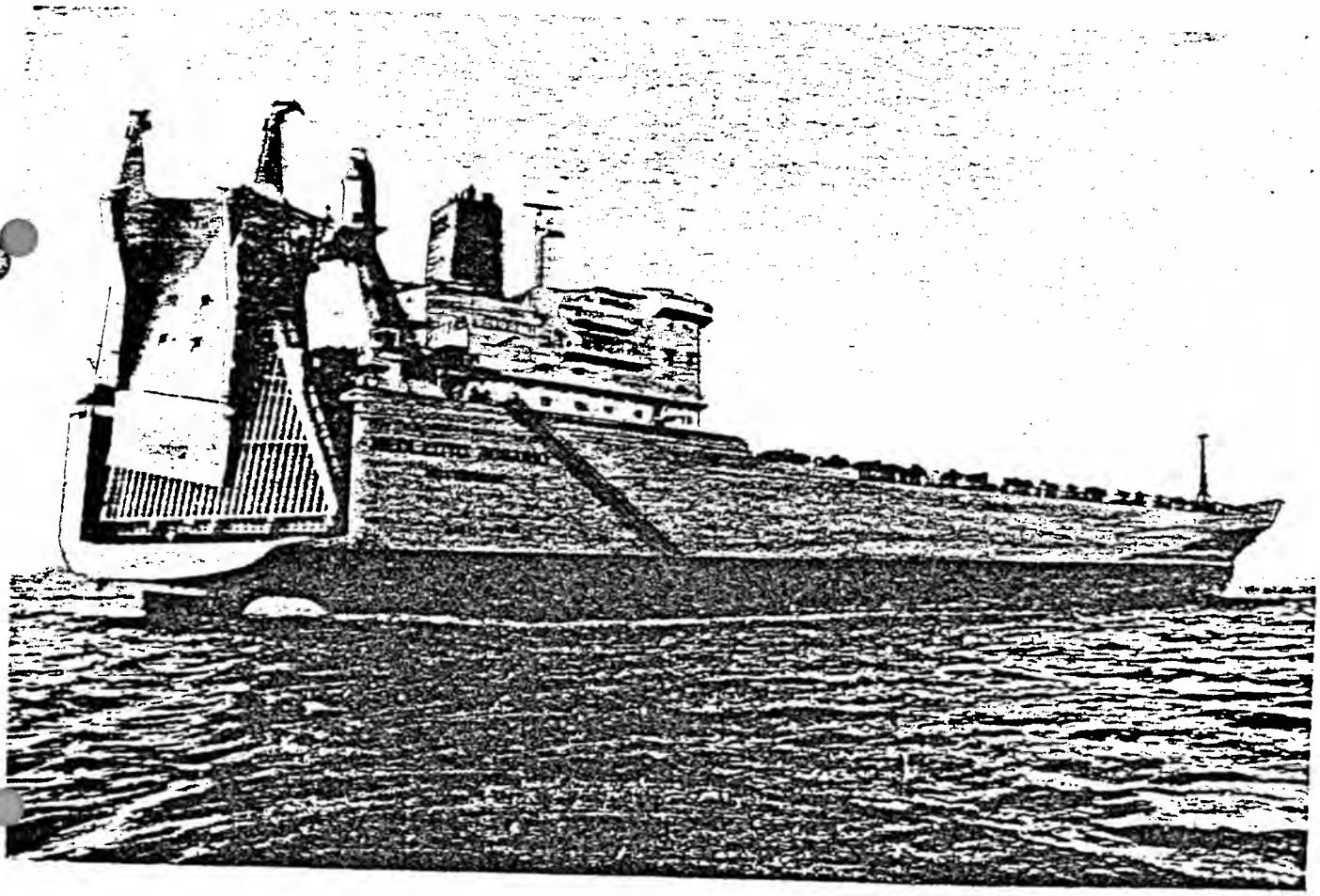




RORO / CONTAINER VESSEL WITH QUARTER RAMP .



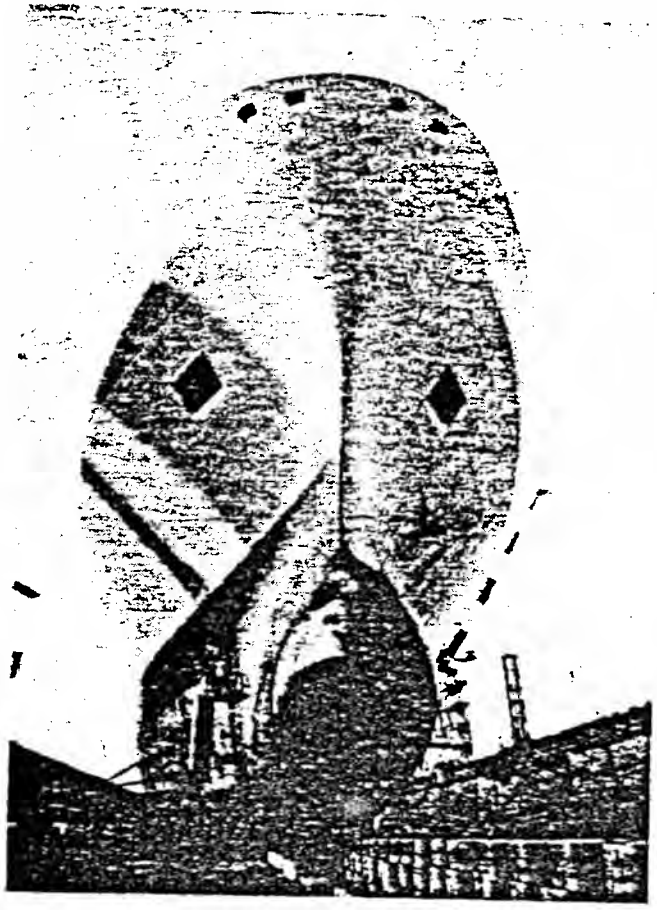
The ramp, lowered onto the wharf, is the (only) entry to and exit from the vessel's holds.



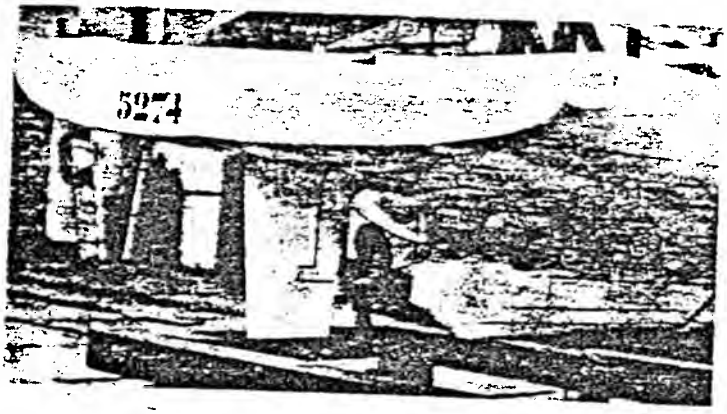
Vessel ready for sea, with ramp in stowed position. Note the great number of ventilation fans on the upper deck leading from/to roro holds.

1979 PROPELLER OBSERVATION NEDLLOYD ROCHESTER - (RSV)
N.B. ZINC ANODES AROUND STERNTHRUSTER : WRONG

NEDLLOYD ROSARIO - NKK 1978

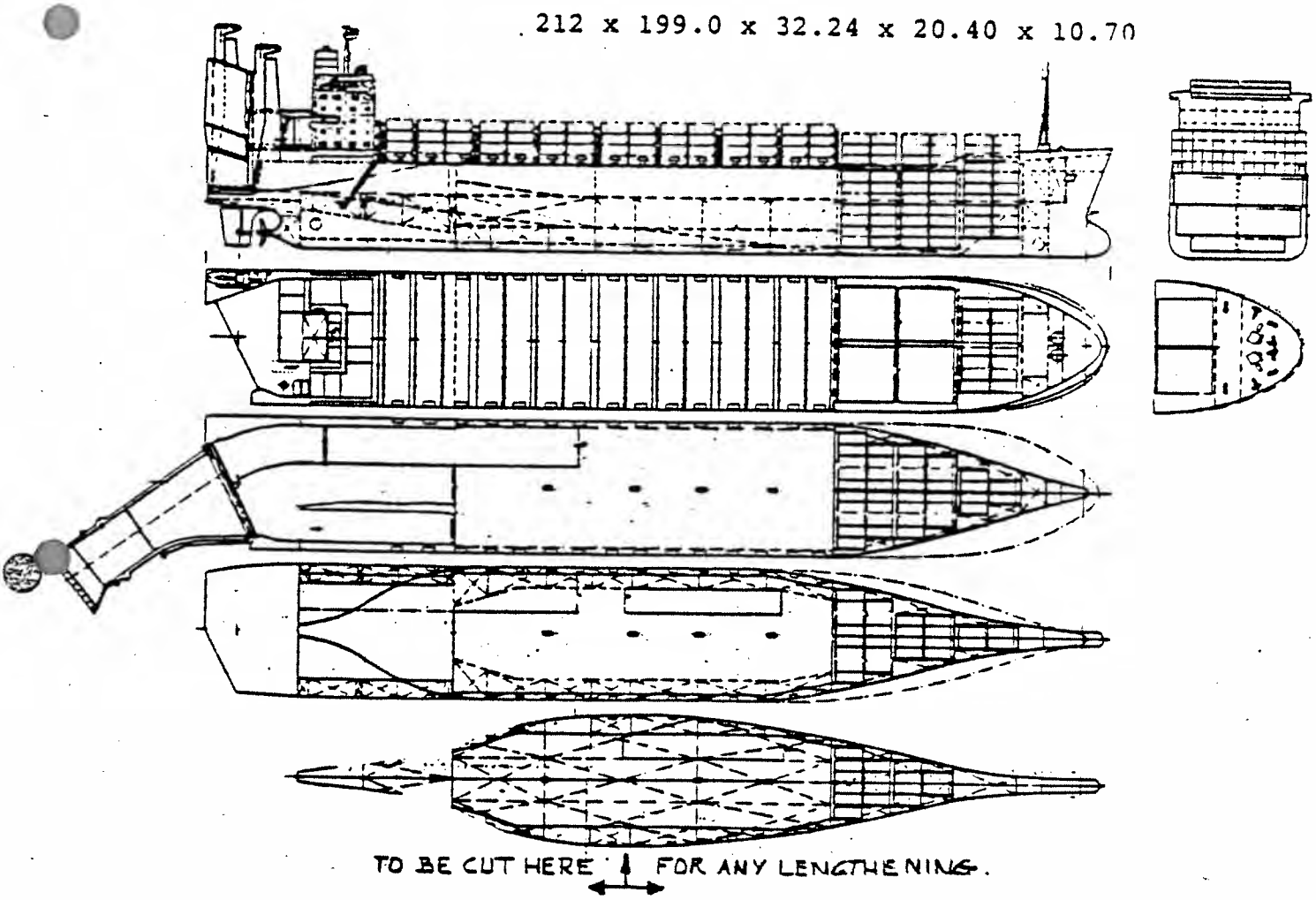


BULB SUITABLE: T_{max} 7.5 → 10.7 M
19 - 21 Kn SEA SPEED.



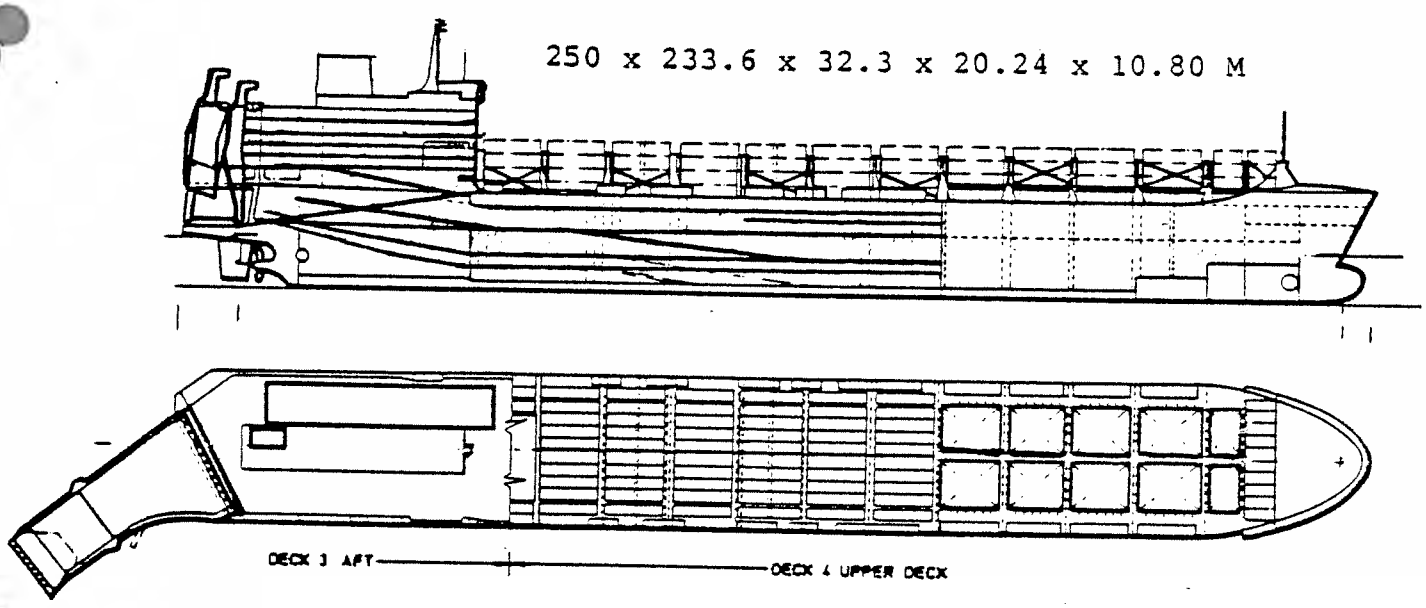
MODEL TESTS AT NSMB WAGENINGEN + EDE
FLOW VISUALISATION + WAKE (3 DIM)
CAVITATION
HULL-PRESSURES

212 x 199.0 x 32.24 x 20.40 x 10.70



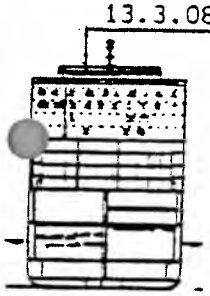
Ro/Lo design. Nedlloyd Rouen/Rosario type, if provided with cells in the foreship.

250 x 233.6 x 32.3 x 20.24 x 10.80 M



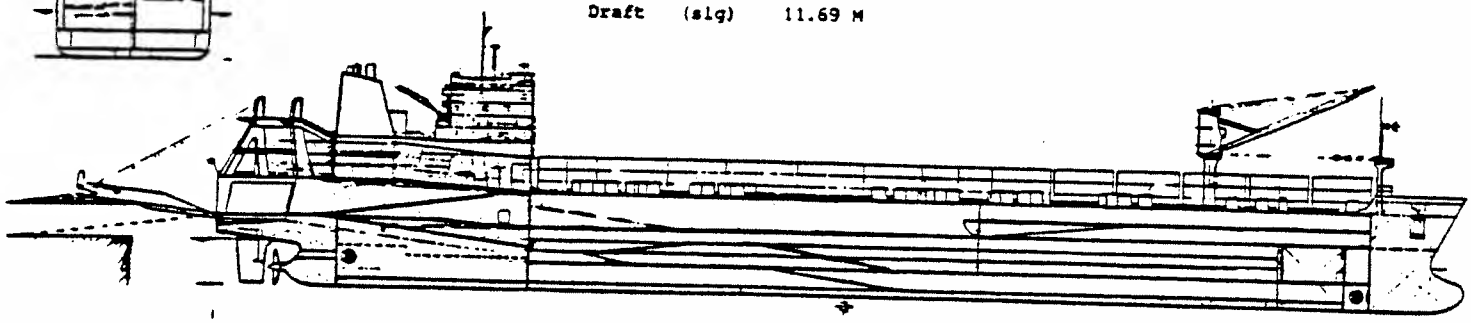
Ro/Lo newbuilding. General arrangement of the ACL third generation hybrid ro-ro.

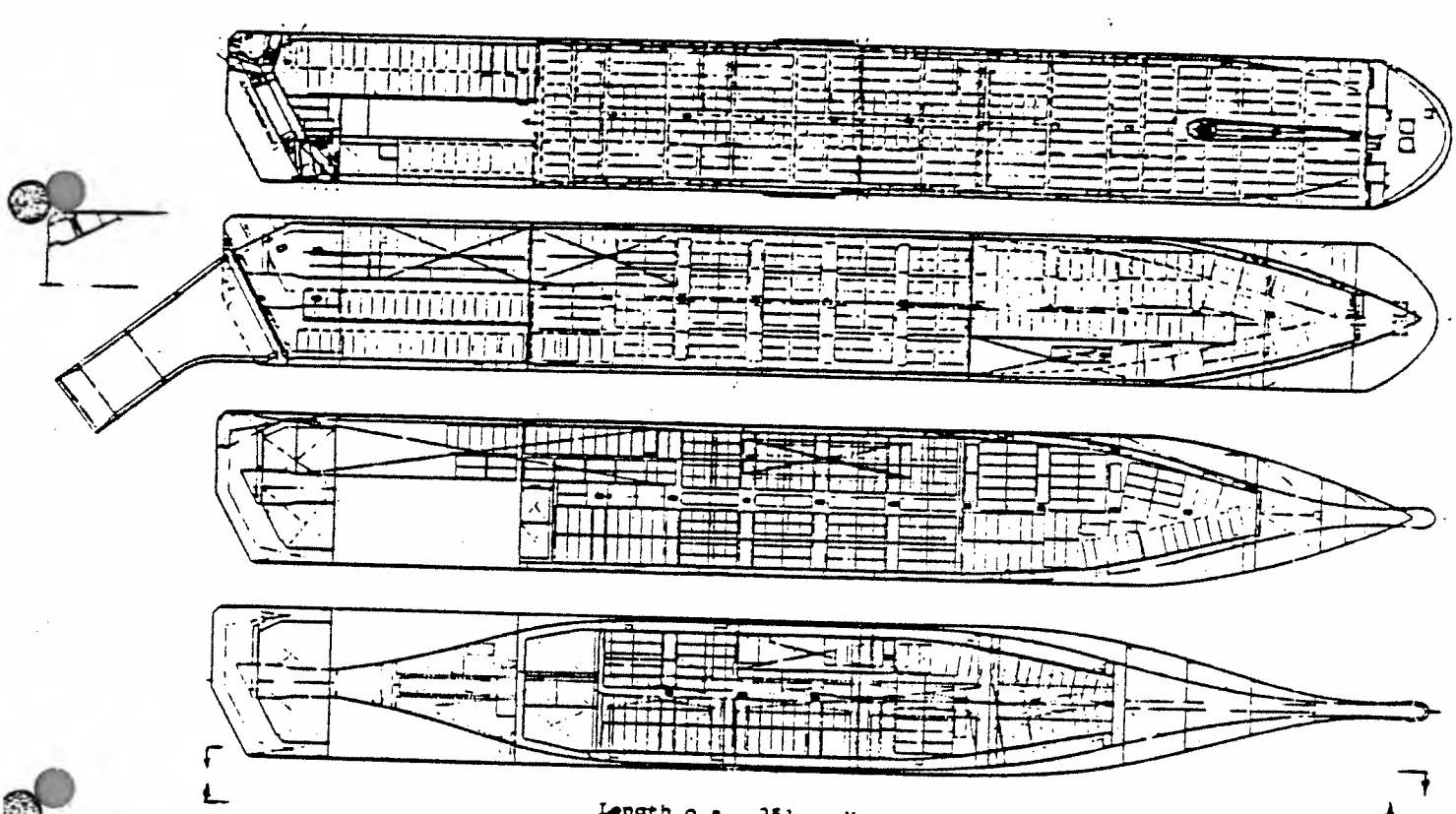
Provided with cells in the foreship underdeck and with cell guide structures above deck and - what's new - above the cellular hatches.



Length o.a.	262	M
Length p.p.	246.40	M
Breadth	32.26	M
Depth (m)	21	M
Draft (dsn)	9.75	M
Draft (slg)	11.69	M

Newbuilding
(W.W.-1983)



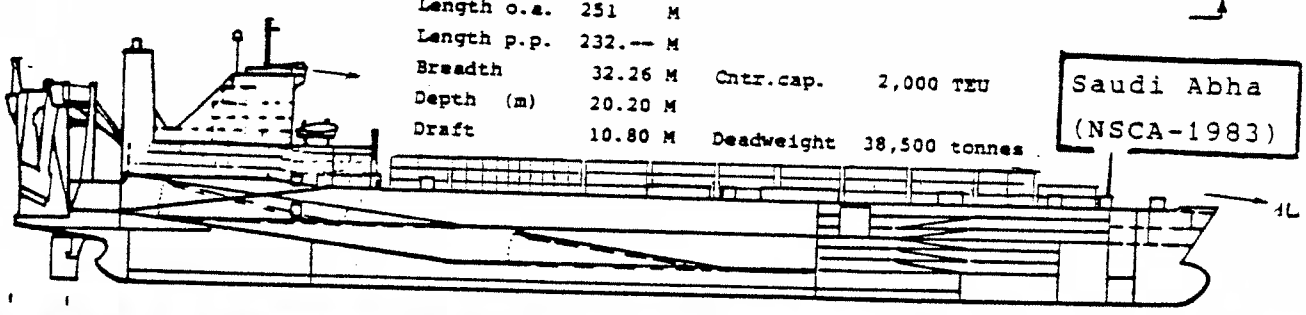


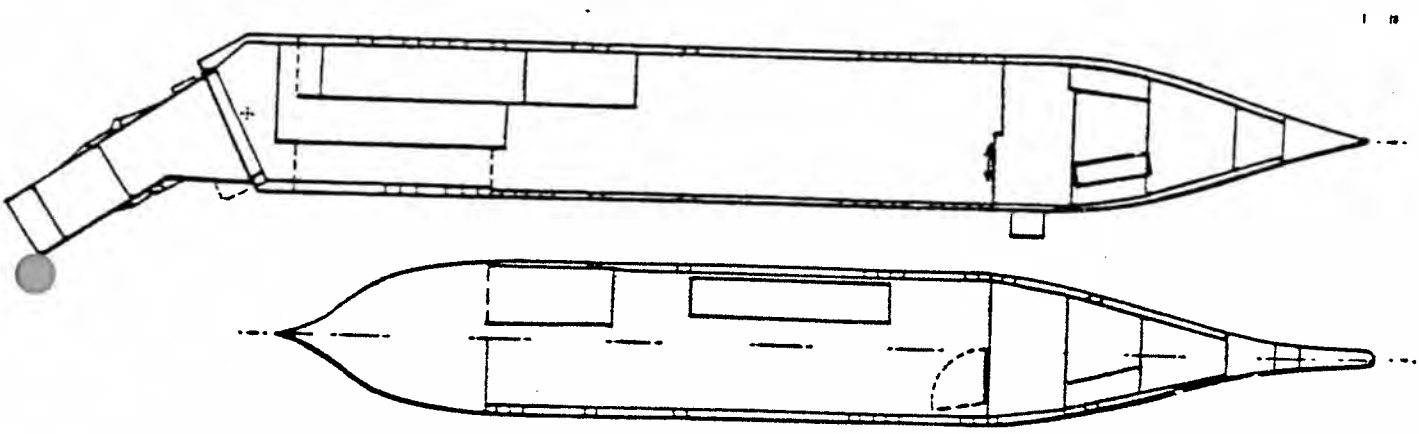
Length o.a.	251	M
Length p.p.	232.--	M
Breadth	32.26	M
Depth (m)	20.20	M
Draft	10.80	M

Cntr.cap. 2,000 TEU

Deadweight 38,500 tonnes

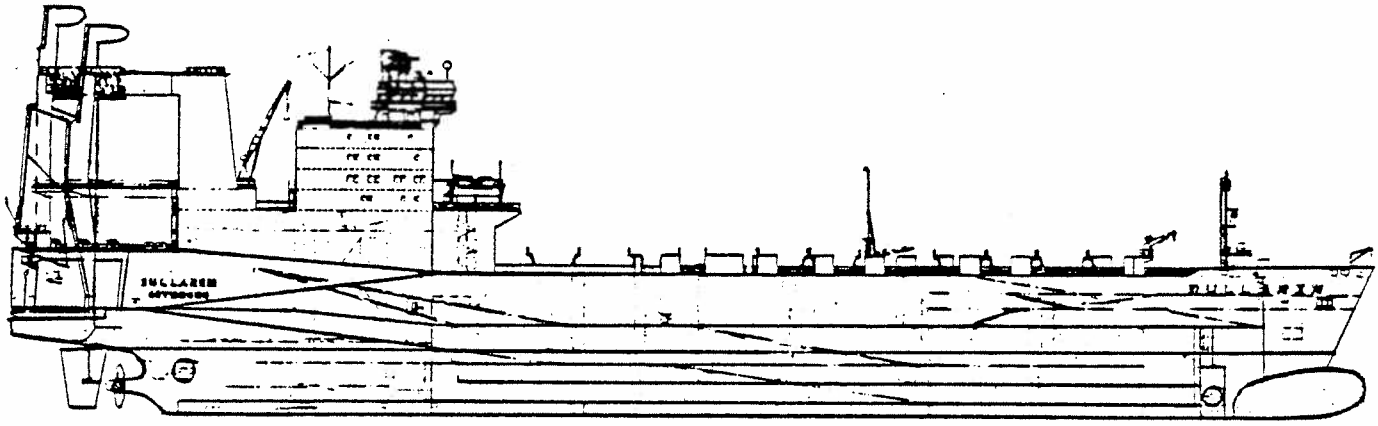
Saudi Abha
(NSCA-1983)





Two examples of modern roro carriers

M/V. "BULLAREN"



Builders: Götaverken Arendal AB, Gothenburg
Owner: Rederi AB Transatlantic, Gothenburg
Yard number: 898
Type: ro/ro vessel
Delivery: June 1st, 1979

Tonnage	18 288 GRT
	8 733 NRT
Deadweight	28 600 t
Length o. a.	187.60 m
Length b. p.	171.80 m
Moulded breadth	32.26 m
Depth to upperdeck	19.85 m
Depth to maindeck	12.60 m
Depth to tweendeck	7.20 m
Draught	10.80 m

Speed: 18 kn

Classification:
L.R. + 100 A 1 + LMC, UMS

Propelling machinery:
B & W-Diesel engine
type: 7L80GF
12 400 kW at 100 rpm
1 distributor gear GJZ 7845, 1 friction clutch PNEUMAFLEX® KAE 410, 1 radial bearing RADILUS® LRL 35/335 and 1 tooth coupling DENTILUS® VZ 5.6 (Lohmann & Stolterfoht)
controllable pitch propeller (KaMeWa)
1 shaft driven generator 1800 kVA, 450 V, 60 Hz
bow thruster 1500 HP
stern thruster 1500 HP
Crew: 30 persons
1 stern ramp, Jumbo type,
1 ramp cover deck No. 2, 37.60×7.50 m.
1 ramp cover deck No. 3, 44.40×11.68 m.
1 stern door on deck No. 3, 26.40×6.70 m.
1 deckhouse door on deck No. 4, 8.30×6.50 m and
5 watertight bulkhead doors on deck No. 1, 2 and 3 separating the cargo room on each deck in two sections (Navire)
2 hoistable cardecks

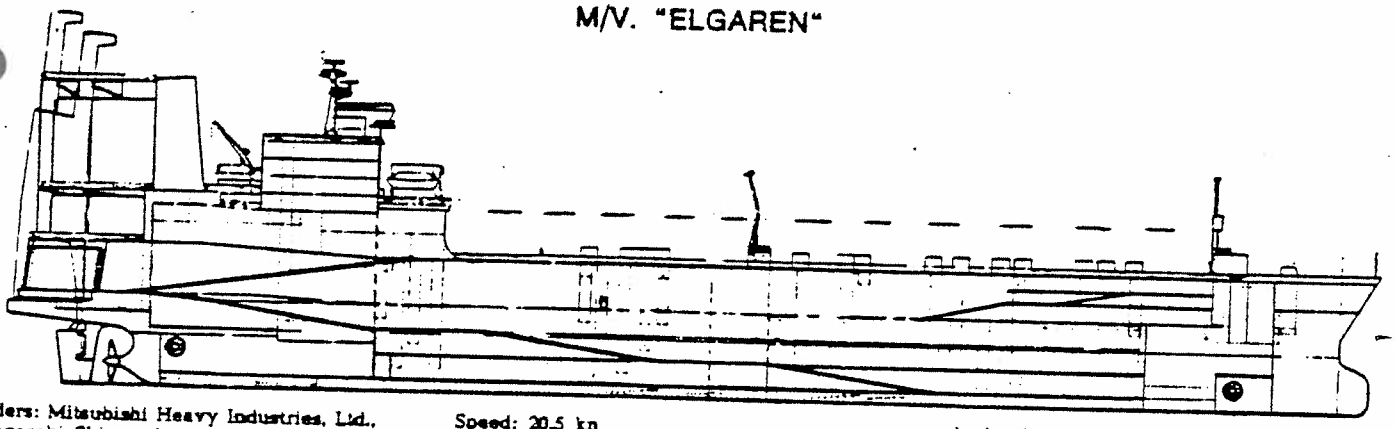
Electric winches:
5 of 10 t, 4 of 20 t (Brissonneau & Lotz)

Total cargo capacity:

bale	50 450 m³
Deck area	14 950 m²
Trailer lane	3 590 m

1218 containers 20'
50 sockets for refrigerated containers

M/V. "ELGAREN"

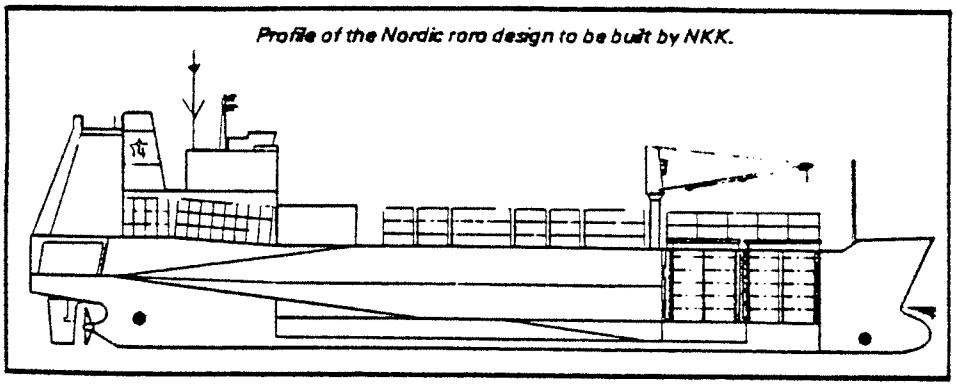


Builders: Mitsubishi Heavy Industries, Ltd., Nagasaki Shipyard
Owner: Rederi AB Transatlantic, Gothenburg
Yard number: 1834
Type: ro/ro vessel
Delivery: February 1979

Tonnage	21 721 GRT
	11 752 NRT
Deadweight	28 098 t
Length o. a.	216.50 m
Length b. p.	198.00 m
Moulded breadth	32.26 m
Depth to upperdeck (5th deck)	20.65 m
Depth to 4th deck	13.50 m
Draught	10.80 m

Speed: 20.5 kn
Trial speed: 22.7 kn
Classification:
L.R. + 100 A 1 UMS + LMC, + RMC
Propelling machinery:
Mitsubishi/Sulzer-Diesel engine
type: 9 RND 90 M turbocharged
22 175 kW (30 150 BHP) at 122 rpm
Fuel consumption: 99 t per day
bow thruster 1324 kW/1800 HP (Mitsubishi/KaMeWa)
stern thruster 1324 kW/1800 HP (Mitsubishi/KaMeWa)
Crew: 31 persons
hydraulic quarter ramp 50.00×12.00 m, max. load 350 t
hydraulic stern door 26.40×6.70 m
hydraulic casing door
4 internal ramps
2 cardecks in the fore part of deck 4
deck area 17 500 m²
fork lift trucks: 1 of 3 t, 2 of 12 t, 2 of 25 t
4 electric hydraulic mooring winches of 10 t,
5 electric hydraulic mooring winches of 20 t,
2 electric hydraulic windlasses of 36 t
Hold capacity:
bale 1 986 454 cub. feet
incl. refr. cargo space (+ 7° to — 25° C)
2 cargo oil tanks 25 179 cub. feet
1546 containers 20'
100 connections for refr. containers
125 cars

TRUNKLINE RORO VESSELS - SAME OWNER.



Profile of the Nordic roro design to be built by NKK.

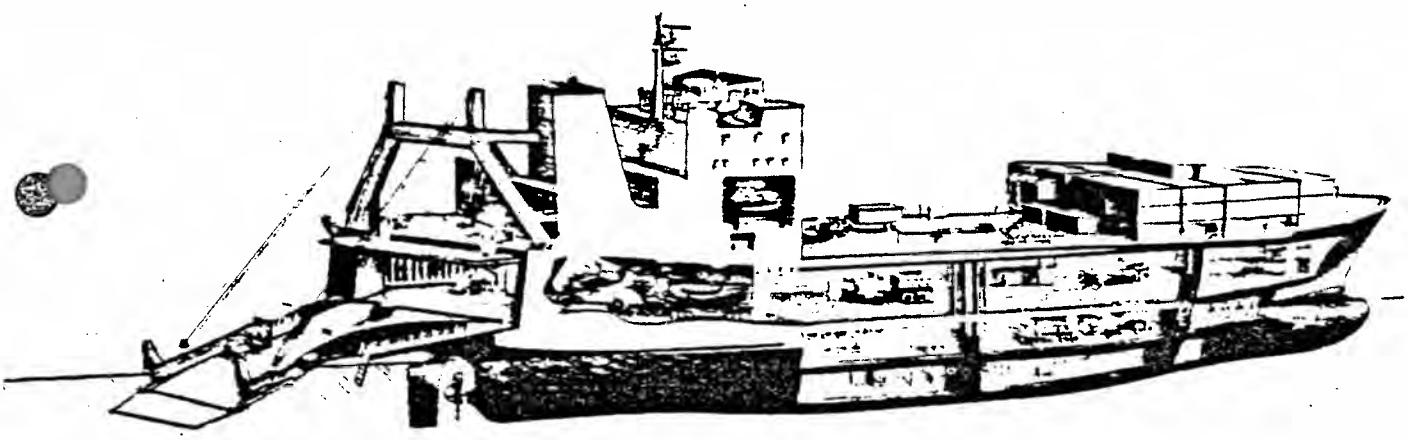
Nordic consortium buys roros from Japan

Two roro/container vessels, worth about US$100 million, have been ordered by the Nordic consortium from Japan's Nippon Kokan yard. The consortium, consisting of Helsinki's EFFOA, Stockholm's Johnson Line and a Wilh Wilhelmsen subsidiary, The Norwegian South-America Line of Oslo, is to extend its liner service to south America. The two 1 400 teu ships will operate between northern Europe and Brazil/River Plate. Much of their capacity will be for refrigerated containers, and their wide quarter ramp will contribute to efficient handling of all types of cargo. Johnson Line and Wilh Wilhelmsen will each manage one vessel, and there is an option for a third, which would come under EFFOA's jurisdiction. Deliveries are expected in early 1985.

The Motor Ship September, 1983

I.H.I. standard roro design **"FREEWAY"** -900

(March 1983)



Living up to its reputation as the pioneer of standardized ships in Japan's shipbuilding industry, Ishikawajima-Harima Heavy Ind. has again taken the lead by adding the "Freeway" type Ro-Ro ship to its "F" series of standardized ships. The new Ro-Ro ship was developed by combining the knowhow of a leading ocean-going Ro-Ro designer, Transconsultants of Sweden, and the shipbuilding expertise of IHI.

The history of "F" series of standardized ships by IHI goes back nearly two decades, when in May 1966, the company received an order from a Greek shipowner for 13 standardized cargo vessels, which it had developed to replace the standardized "Liberty" type freighter, developed by the U.S.A. during World War II.

Named the "Freedom" ship, the first of the "F" series of standardized ships developed by IHI was a multi-purpose two-deck cargo carrier of around 14,900 dwt. Since the delivery of the first ship in 1967 to the end of 1979, a total of 124 "Freedom" ships were built by IHI for both domestic and overseas owners, attesting to their great popularity.

Following in the wake of the highly successful "Freedom" ships, came the second of the "F" series, the "Fortune" type, which was a multi-purpose single-decker cargo vessel of the 22,000 dwt class, designed to carry bulk and general cargoes, as well as containers and automobiles. The first of the "Fortune" type ships was completed in 1971, and 24 more of them were built up to the end of 1981.

The next type developed in the "F" series was the bulk carrier type named the "Future" ship, available in three different classes of 32,000 dwt, 45,000 dwt and 60,000 dwt. To denote their classes, they were referred to as "Future 32", "Future 45" and "Future 60". The first of the "Future" type built in 1971 was of the 32 class, and 61 more "Future" ships including a "Future 60", were built up to the end of 1981.

In 1978, another type in the "F" series, the single-open-decker of the 23,000 dwt class, multi-purpose cargo vessel called the "Friendship" made its appearance on the market. Depending on the type of cargo-handling facilities provided, the "Friendship" vessels were available in the "G" and "H" types. Up to the end of 1981, the number of "Friendship" ships of the "H" and "G" types together totaled 11.

(Continued on page 2)

"FREEWAY"

-page2- *Continued*

Also in 1978, an additional "Freedom" type, named the "Freedom Mark II" of the 16,000 dwt class was put on the market. By the end of 1981, a total of 25 of these vessels were built by the company.

And now....as the latest addition to its "F" series of standardized ships, IHI is ready to put the "Freeway" type Ro-Ro ship on the market.

Anticipating the varied requirements of owners and conditions under which the new type Ro-Ro ships will be used, the "Freeway" vessels will be available in 4 classes. Designated by the approximate average number of containers in terms of TEU, accommodable on board, they are classified as "Freeway 900", "Freeway 1100" "Freeway 1300" and "Freeway 1500". The principal particulars of the ships of each class are shown in the table.

Besides offering the advantages to be gained from standardization of design, principal particulars and from the shortened construction period etc. common to standardized ships, the "Freeway" Ro-Ro ship boasts a number of features including the following.

Quarter Ramp

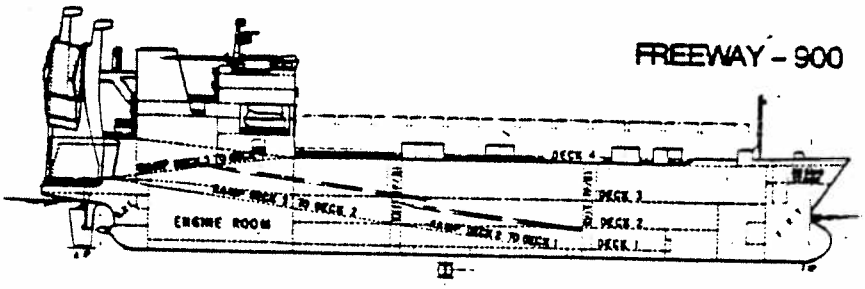
- Its generous dimensions and geometry of the quarter ramp allow true two-way traffic for ordinary cargo handling vehicles, and maneuvering space for heavy-lift trailers.
- Strong enough to permit passage of loads of over 400 tons.
- Eliminates the need for special Ro-Ro berth.
- Offers wider and better access to a ship than a straight stern ramp.
- Being tension-controlled, it automatically adjusts to tide, trim and draft changes, permitting the ship to work continuously even at large tidal range ports.
- Supported by tension winches, it creates practically no pressure on the pier.

Decks and Lashing Systems

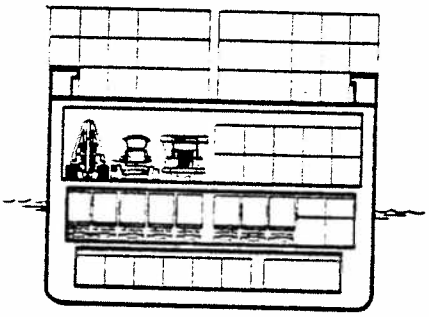
- Adequate clearance and strength of the decks permit most types of cargo handling vehicles to operate on board.
- Flush-mounted container sockets on all decks permit firm securing of containers.
- Recessed fitting for lashings provided in the inner skin, all bulk heads and rampway, combined with lashing lugs and container fittings, serve as terminals for lashings of non-containerized cargoes.

Stability

- The ship is designed to carry approximately 4,800 tons of containerized cargoes on the weather deck at the scantling draft.
- The ballast tank system which includes an anti-heeling arrangement that automatically compensates for asymmetric loading or handling.



		FREEWAY - 900	FREEWAY - 1100	FREEWAY - 1300	FREEWAY - 1500
LENGTH OA		150.80 M	166.53 M	182.26 M	198.00 M
LENGTH BP		135.80 M	151.53 M	167.26 M	183.00 M
BREADTH MLD		28.00 M	28.00 M	28.00 M	28.00 M
DEPTH MLD		17.70 M	17.70 M	17.70 M	17.70 M
DRAUGHT, DESIGN		8.70 M	8.70 M	8.70 M	8.70 M
DRAUGHT, SCANTLING		9.50 M	9.50 M	9.40 M	9.30 M
DEADWEIGHT, DESIGN		12200 T	15300 T	18300 T	21200 T
DEADWEIGHT, SCANTLING		14800 T	18200 T	21100 T	23900 T
CARGO SPACE	AREA	7650 SQM	9170 SQM	10690 SQM	12210 SQM
	VOLUME	28000 CBM	33400 CBM	38800 CBM	44200 CBM
	LANE LENGTH	2370 M	2880 M	3390 M	3890 M
CONTAINERS	DESIGN	840 TEU	999 TEU	1161 TEU	1340 TEU
	MAX	1022 TEU	1216 TEU	1414 TEU	1635 TEU
MAIN ENGINE	TYPE	6RTA58	6RTA58	7RTA58	7RTA58
	MCR	8460 kW	8460 kW	9870 kW	9870 kW
SERVICE SPEED AT DESIGN DRAUGHT		17.1 KNOTS	16.6 KNOTS	16.8 KNOTS	16.3 KNOTS



Working Environment

- Spacious unobstructed cargo decks with vertical, smooth bulk heads, combined with a system of wide ramp make cargo handling fast, safe and easy.
- Weather-protected, well-illuminated working environment is provided for stevedores and crew.
- Effective, low-noise ventilation system supplies adequate amount of fresh air to cargo spaces.

Fuel Economy & Ease of Maintenance

- To achieve fuel economy, the ship is equipped with a two-cycle low-speed RTA type main diesel engine, which offers a high fuel economy of over 10,000 TEU-miles per ton of fuel.
- The main engine is fitted neatly into the stern, designed to permit the use of the dead space under the ramp, while offering easy access to all engine components for ease of maintenance.
- A central fresh water cooling system is arranged in the engine room to reduce maintenance work. In addition, the "Freeway" Ro-Ro ships will offer a wide range of optional items, as shown in the table of "Optional items". Other items not listed in the table may be made available to meet the owner's requirements.

—Optional Items—

1. Classification (NK, AB, NV, BV, GL)
2. Bow thruster (700 kw)
3. Stern thruster (700 kw)
4. Plugs for reefer containers (upto 50 sets of 15kw)
5. Deck crane (40 t, electro-hydraulic) on DECK 4
6. Hoistable car deck
7. Watertight bulkhead door

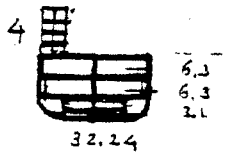
(From: SEA JAPAN, March 1983)

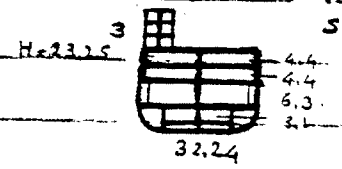
13

RoRo REMARKS, DEC 83

1. 3-DECK RoRo TO BE PREFERRED ABOVE 4-DECK RoRo

MORE FLEXIBLE (TWEEN DECK. HEIGHT)





MORE SUITABLE FOR TRAILERS SHORT DISTANCE

EXTRA TWEENDECK ATTRACTIVE FOR PAPER DRUMS. INGOTS STEEL

SCANDINAVIAN TYPE CAPT STENBERG.

VESSEL IS MORE HEAVY — LESS D.W.
MORE EXPENSIVE
" HAS LESS STABILITY FOR DECKLOAD

2. IF POSSIBLE ONE SHOULD NOT BUILD SHORTER THAN 212 m WITH PAN. BEAM.
212 m ALLOWS FOR "WORLD-WIDE" PORTS (ROUEN/ROSARIO CLARENCE/CLEMENT/COLOMBO)

CONDITION: VESSEL TO BE FITTED WITH BOW- + STERN THRUSTER AND LARGE RUDDER

— IN CASE VESSELS ARE SHORTER THAN 190 m, DIFFICULTIES WITH DEADWEIGHT AND A BAD SEAGOING PERFORMANCE IN W. N. ATLANTIC. "PITCHING" WILL OCCUR.

— IN CASE PORTCONDITIONS ALLOW, THEN A LONGER VESSEL WITH THE ORIGINAL CROSS-SECTION UP TO 260 m IS MORE ECONOMICAL: 3 USDCT/MILE/TEU → 2½ USDCT/MILE-TEU / SAUDI NAT. / ACL / BARBER

(3 PILLAR SPACINGS TO BE ADDED TO THE 212 m VESSEL)

3. FOREBODY TO HAVE CELLULAR HOLDS FOR CONTAINERS BELOW DECK

REASONS: THERE IS ALWAYS A NEED FOR CELLS BELOW DECK.

EXTRA CARGO WEIGHT IN THE FOREBODY BECAUSE OF BETTER STOWAGE RESULTS IN MORE FAVOURABLE TRIM CONDITIONS AND LESS WB IN F.P.

RoRo SPACE TO BE OVER THE PARALLEL PART OF THE VESSEL ONLY

UNECONOMICAL RoRo STOWAGE IN CONVERGING FOREBODY

EXTRA SAFETY BECAUSE OF EXTRA TRANSVERSE BULKHEADS (ESPECIALLY IN FOREBODY)

TRAILER VESSELS

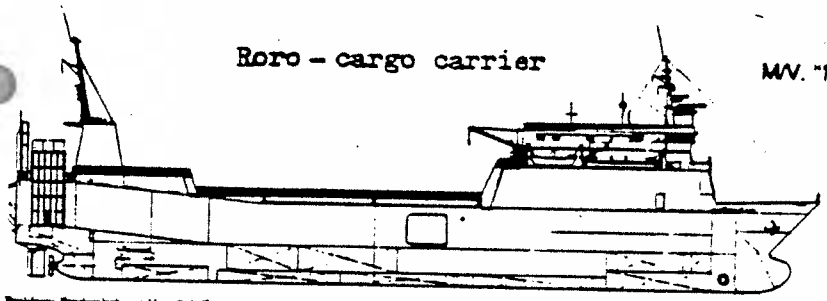
ROLLING STOCK

14.0

TRAILER VESSEL

Roro – cargo carrier

M/V. "MERCANDIAN IMPORTER III"



Builders: Frederikshavn Vaerft A/S, Frederikshavn
Owner: Mercandia Rederiene Per Henriksen, Copenhagen
Yard number: 383
Type: roro vessel
Delivery: April 23rd, 1982

Tonnage	1 589 GRT 835 NRT
Deadweight	3 575 t
Length o. a.	105.60 m
Length b. p.	96.00 m
Moulded breadth	18.00 m
Depth to 1st deck	10.55 m
Depth to 2nd deck	5.00 m
Draught	4.97 m
Speed: abt. 16 kn	

Classification:
N.V. + 1 A 1 Car Ferry A

Propelling machinery:
MaK-Diesel engine
type: 12 MK 453 AK turbocharged
3310 kW (4500 HP) at 440/220 rpm
1 Simplex-compact-seal size 300 P (HDW)

Auxiliary engines:
3 Mercedes-Benz-Diesel engines
type: OM 404 A
304 kW (414 HP) at 1800 rpm each
3 generators 345 kVA, 3 × 440 V, 60 Hz each
anti-heeling system (Intertug)
bow thruster 265 kW (Lastrum)
hydraulic stern ramp — length 15.4 m, breadth 4.6 m incl. flaps, width of traffic lane 4.0 m, axle load 2 × 17 t,
hydraulic side ramp off S. — length 13.35 m, breadth 4.6 m incl. flaps, width of traffic lane abt. 4.3 m, axle load 17 t,
side door with cargo platform breadth 4.7 m, height 3.7 m and pallet lift: outside the side door are two separately operated pallet lifts with steps at maindeck and cargo platform, lifting capacity 2 × 3 t
1 lift 16.0 × 3.2 m, steps at maindeck, 2nd deck and 1st deck, lifting capacity 35 t
cargo handling equipment (MacGREGOR)
3 automatic mooring winches, 2 comb. windlasses/automatic mooring winches

Head capacity:

gross	19 371 m²
lane	18 313 m²

Car capacity:

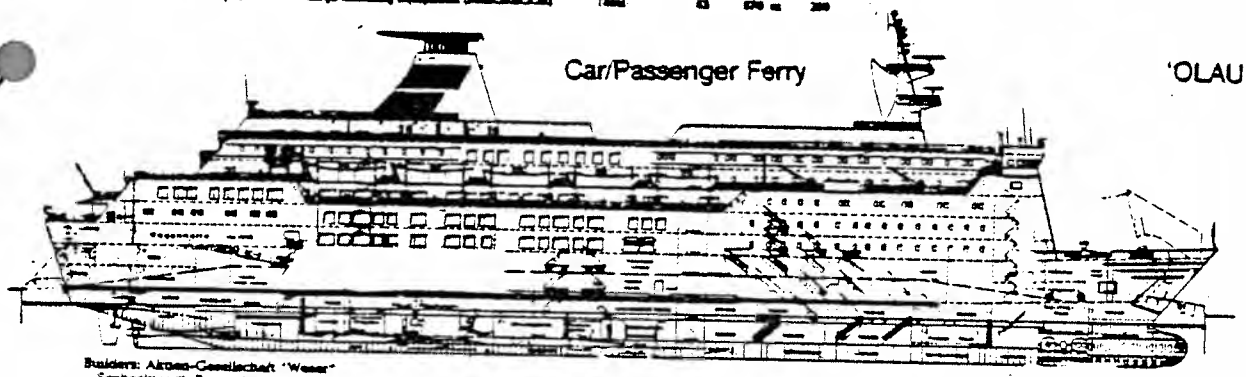
	length of car tracks breadth	number of cars length
mainlanding	1.70 m	4.25 m
2nd deck	255 m	30
pilot. deck	660 m	126
1st deck	140 m	28
	700 m	144
Total	1700 m	356

Trailer-container capacity:

	trailer	length of trailer tracks	container 20'
2nd deck	28	400 m	58
1st deck	35	470 m	142
Total	63	870 m	200

Car/Passenger Ferry

'OLAU BRITANNIA'



Builders: Aktien-Gesellschaft "Weser" Seebeckwerft, Bremerhaven
Owner: Olau-Line (U. K.) Ltd. & Co., Hamburg
Yard number: 1 031
Delivery: May 5th, 1982

Tonnage	14 985 GRT 8 990 NRT
Deadweight	2 880 t
Length o. a.	153.40 m
Length b. p.	136.00 m
Moulded breadth	24.28 m
Depth to E-deck (cabindeck)	13.60 m
Depth to F-deck (cardeck)	8.00 m
Draught	5.82 m
Speed: 21 kn	

Classification:
G.L. + 100 A 4 M E 2 mit Freibord 2.21 m Fahrschiff-Fahrgastschiff + MC E AUT-Z

Propelling machinery:
4 B + V SEMT-Pielstick-Diesel engines non reversible
type: 8PC 2 – 5L
4 × 3825 kW (4 × 5200 HP) at 520/220 rpm
2 twin input — single output marine reduction gears NAVILUS® GVA 1250 So. (Lohmann & Stolterfoht),
2 highly elastic friction clutches PNEUMAFLEX® KAA 340 (Lohmann & Stolterfoht)
2 controllable pitch propellers (Escher-Wyss)
2 Simplex-compact-seals size 420 (HDW)

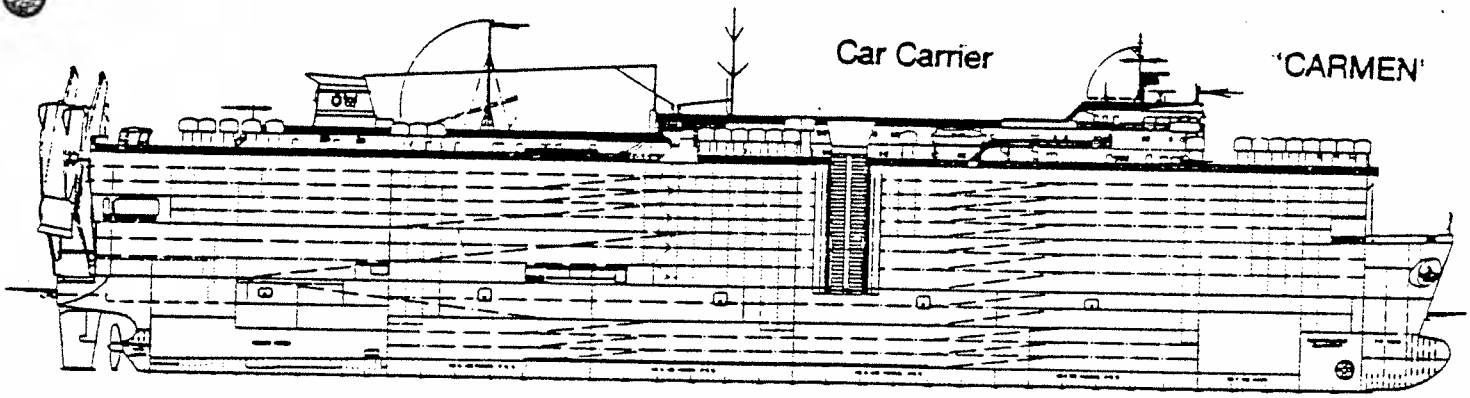
Auxiliary engines:
2 MaK-Diesel engines
type: 6 M 332 AK
852 kW (1160 HP) at 720 rpm each
2 generators 1000 kVA, 450 V, 60 Hz each
2 MaK-Diesel engines
type: 8 M 332 AK
1200 kW (1630 HP) at 720 rpm each
2 generators 1400 kVA, 450 V, 60 Hz each
1 Deutz-Diesel engine
type: BA8M 816 oLLK —
256 kW at 1800 rpm
1 generator 255 kVA, 450 V, 60 Hz.

separators (Westfalia)
2 fresh water evaporators (Atlas-Denmark)
2 bow thrusters type 100 F — thrust 11 t, 736 kW each (Lastrum)
stabilizers (Denny Brown/AEG)
anti-heeling system (Intertug)
steering gear type HS 90 × 2 D (Frydenbø)
2 radars (Atlas)
gyro compass and automatic pilot (Anschütz)
Speech-marine wipers (W.G. Schulz)
Crew: 156 persons

1600 passengers
410 passengers in 2-berth cabins
144 passengers in 3-berth cabins
292 passengers in 4-berth cabins
213 aircraft-type seats
bow ramp/door — width 5.60 m, length abt. 10.00 m excl. flaps,
2 stern ramps/doors — width 5.50 m length 5.60 m excl. flaps,
hoistable cardeck,
hoistable cardeck ramps fore and aft and
internal ramp from cardeck to E-deck on S.-side — length 30.00 m, width 2.60 m (MacGREGOR)
2 automatic mooring winches of 100/35 kN
3 automatic mooring winches of 160/50 kN
2 comb. windlasses/automatic mooring winches 60 mm KI chain
550 cars or 65 trailers + 60 cars

Car Carrier

'CARMEN'



Builders: Kockums AB, Malmö
Owner: Reden AB Soya, Stockholm
Yard number: 583
Type: pure car/truck carrier
Delivery: March 23rd, 1982

Tonnage	18 725 GRT 9 975 NRT
Deadweight on design draught	28 223 t 17 558 t
Length o. a.	197.91 m
Length b. p.	190.00 m
Moulded breadth	32.25 m
Depth to upperdeck	31.20 m
Depth to 4th deck	13.64 m

Draught:
Designed draught 11.61 m
9.50 m

Speed: 20.3 kn

Classification:
L.R. + 100 A 1 Vehicle Carrier Ice Class 3 + LMC, UMS

Propelling machinery:
B & W-Diesel engine, direct reversible
type: 7L80GFCA supercharged
13 533 kW (18 400 BHP) at 106 rpm
propeller (Lips)
Fuel consumption: abt. 65 t per day

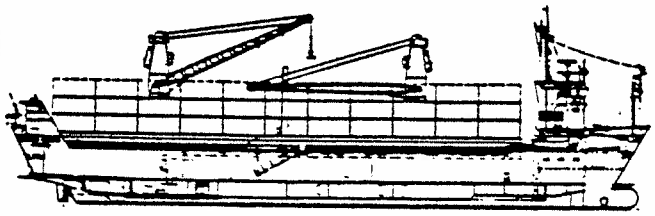
Auxiliary engines:
3 Wärtsilä Vasa-Diesel engines
type: 6 R 32 — 1700 kW at 720 rpm each
3 generators 2000 kVA, 440 V, 60 Hz each
1 emergency Diesel generator 165 kW
bow thruster 1103 kW — thrust 18 t (Lips)
doppler log (Atlas)
Crew: 27 persons

6 hydraulic mooring winches of 25 t and 2 hydraulic comb. windlasses/mooring winches of 25 t (Norwinch)

1 stern quarter ramp/door — length 42 m, width 7 m — 200 t,
1 side ramp/door on S.-side — length 25 m, width 4.5 m — 34 t,
6 internal ramps and hoistable cardecks (MacGREGOR)
(the 5th, 7th and 10th deck and partly the 8th are divided into hoistable sections, which allow vehicle heights up to 6.2 m)
heavy vehicles on 4th, 6th and 9th deck
13 cardecks
parking area abt. 52 000 m² (truck deck 11 000 m²)
6120 passenger cars or
2900 cars and 520 heavy vehicles

15.0
CONTAINER/RORO VESSELS FOR PROTECTED SEA AND LIMITED WATERDEPTH.

CONTR. RORO "SMALL DRAFT"

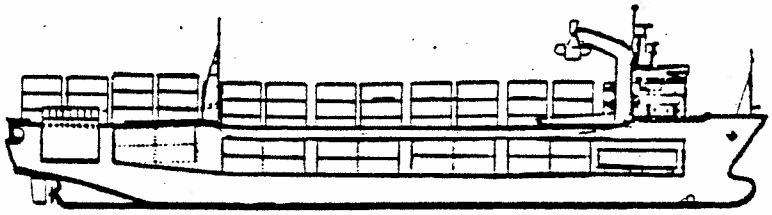


SLOMAN RANGER

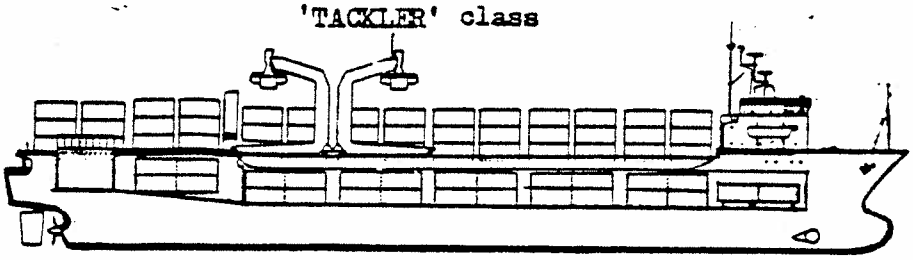
('Coaster' size)

m.v. SLOMAN RANGER, blt EDW, 1979

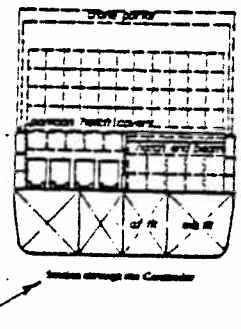
Length oa	92.07 m
Length bp	79.20
Breadth, moulded	18.0
Depth to upper deck	7.90
Depth to main deck	3.68
Summer load draft	3.65
DWt. total at Summer dft	2 570 tonnes
Propulsion	2 x 1 440 bhp
Speed, trial	12.5 knots
Capacity	319 teu
Trailer cap. 25x40' + 5 x 20'	
Complement	12

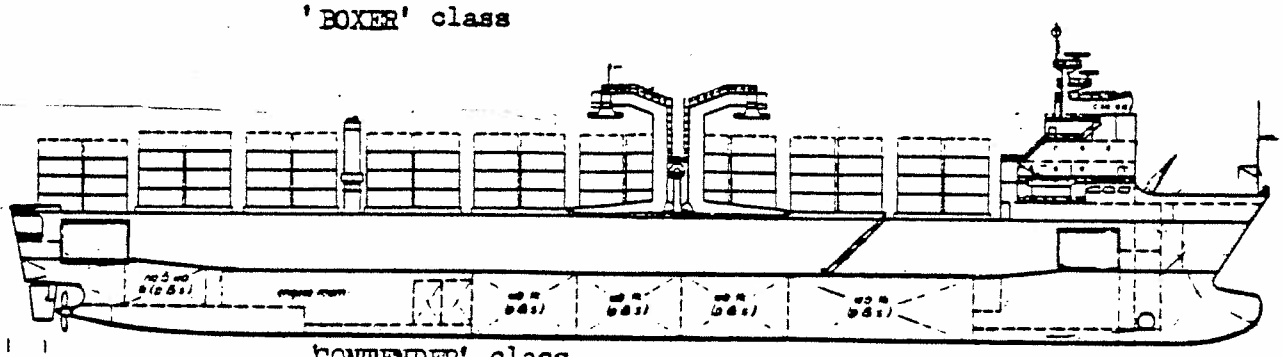


'TACKLER' class

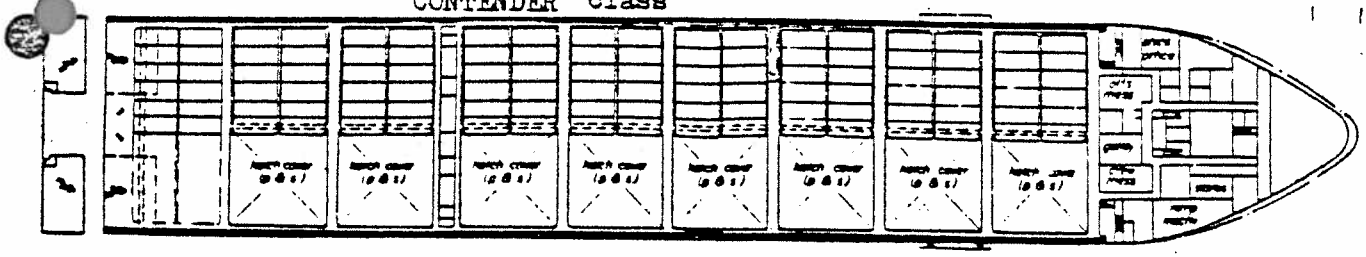


'BOXER' class





'CONTENDER' class



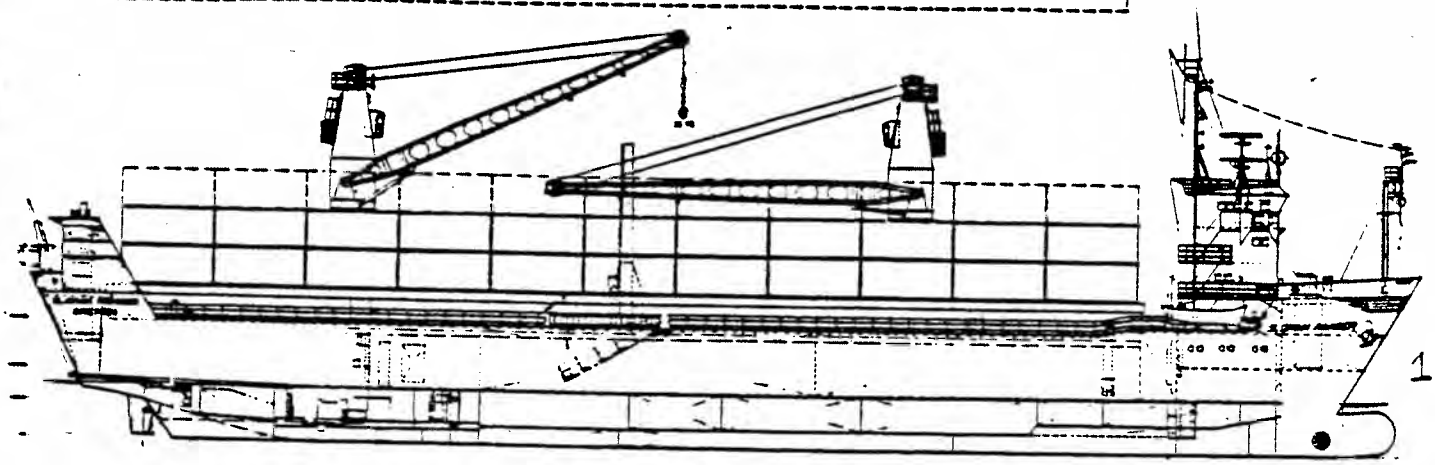
Sea Container International's

	TACKLER	BOXER	CONTENDER
Length oa	114.8m	133.5m	173.0m
Length bp	102.0	122.61	160.0
Breadth, moulded	19.5	25.0	30.4
Depth to upper deck	10.6	12.13	17.0
Depth to main deck	5.3	6.63	9.0
Summer load draught	5.25	6.5	8.2
Dwt. total at summer draught	4 760 tonnes	9 000 tonnes	18 500 tonnes
Propulsion	MaK 9Mu552AK	MAN 14V32/55A	Twin Pielstick 18PC2.5V
Speed, trial	15.85 knots	17.8 knots	19.2 knots
Capacity	378teu	576teu	1 306teu
Lane length, gross	500m	761m	920m
Complement	18	19	26

EXAMPLES OF ROLL-ON/LIFT-ON COMBINATION CARRIERS DESIGNED TO SUIT CONTAINERS.

H.D.W.-built RoLo/MPC- 15 **"Sloman Ranger"** class, 1979

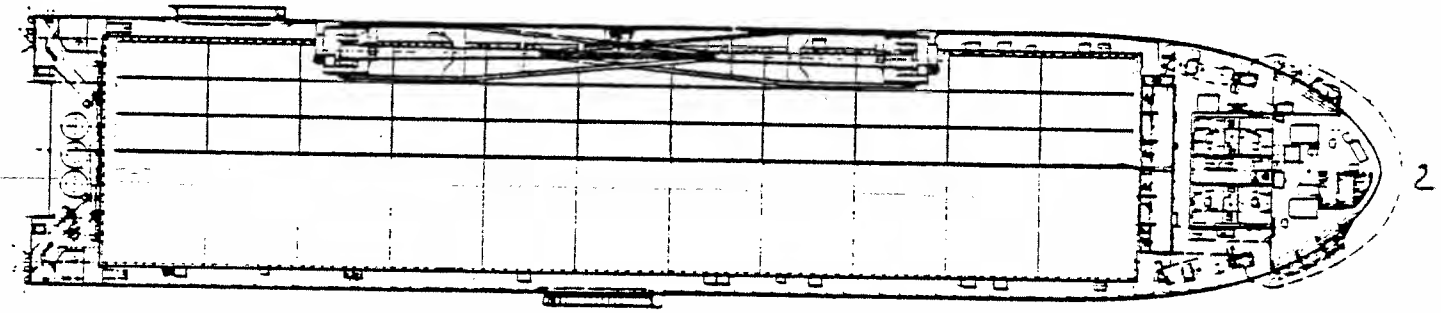
15.1.02

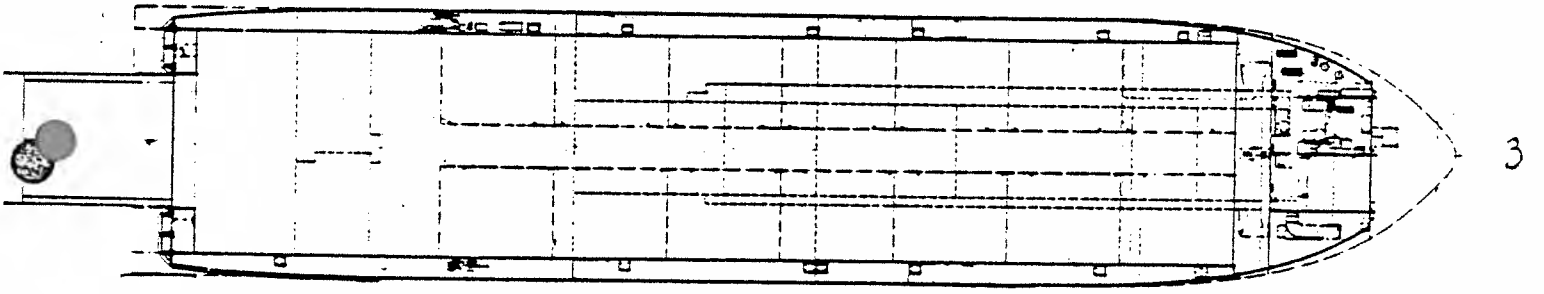


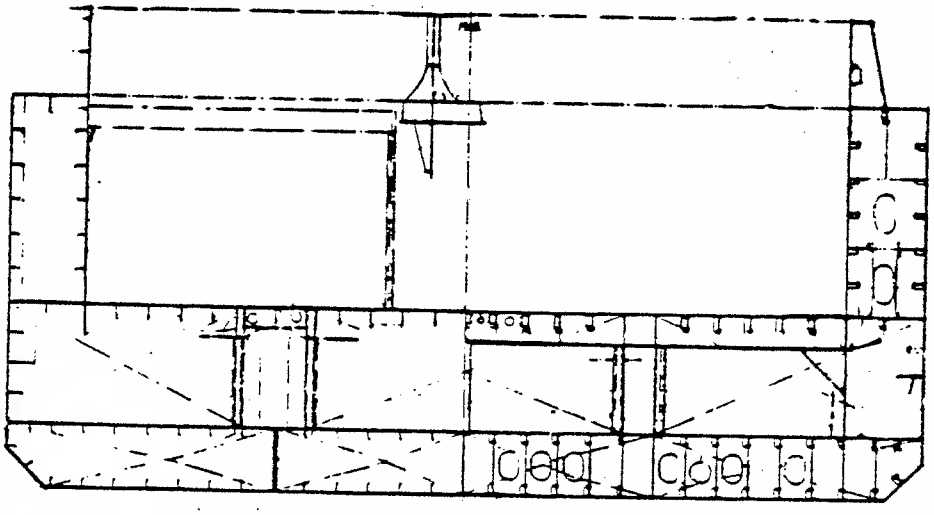
Length over all	92.07	m
Length between perpendiculars	79.20	m
Breadth, moulded	18.00	m
Depth, moulded to upper deck	7.90	m
Depth, moulded to main deck	3.68	m
Draught on summer freeboard	3.65	m
Corresponding deadweight	2,570.00	t
Light ship	2,010.00	t
Gross tonnage, International	999.96	
Suez	3,793.01	
Net tonnage, International	477.25	
Suez	3,074.75	

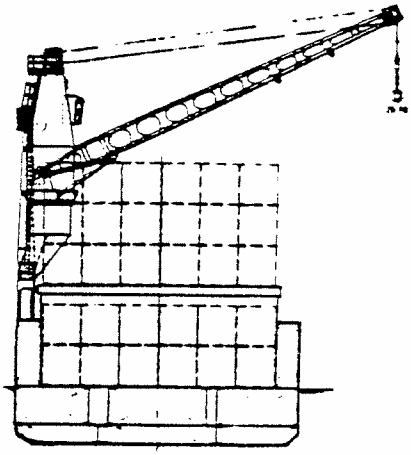
Container capacity, 20-ft units	319
Trailer capacity, 40-ft + 20-ft units	25 + 5
Complement	12
Propulsion power (90 % mcr)	2 × 1,440 bhp at 750 rev/min
Speed at 90 % mcr and draught 3.65 m	12.5 knots

Hold capacity, without portable tweendeck	6,182.00	m³
portable tweendeck fitted	5,910.00	m³
Tank capacity, wine	484.16	m³
chemical products	903.95	m³
fuel oil	350.20	m³
diesel oil	89.30	m³
lubricating oil	8.00	m³
fresh water	86.60	m³
water ballast	1,543.60	m³



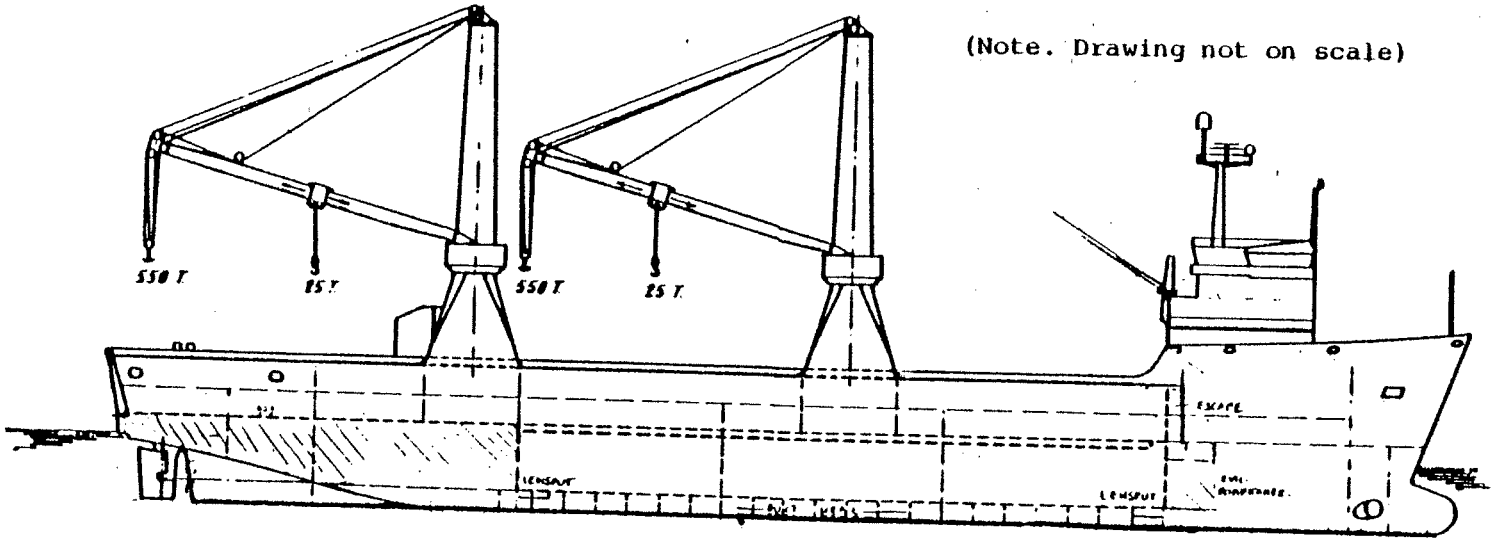


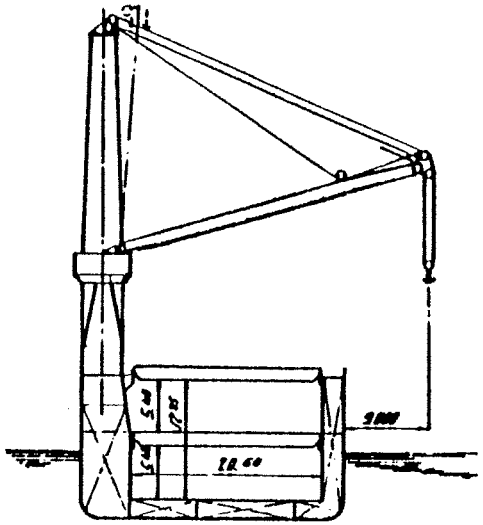


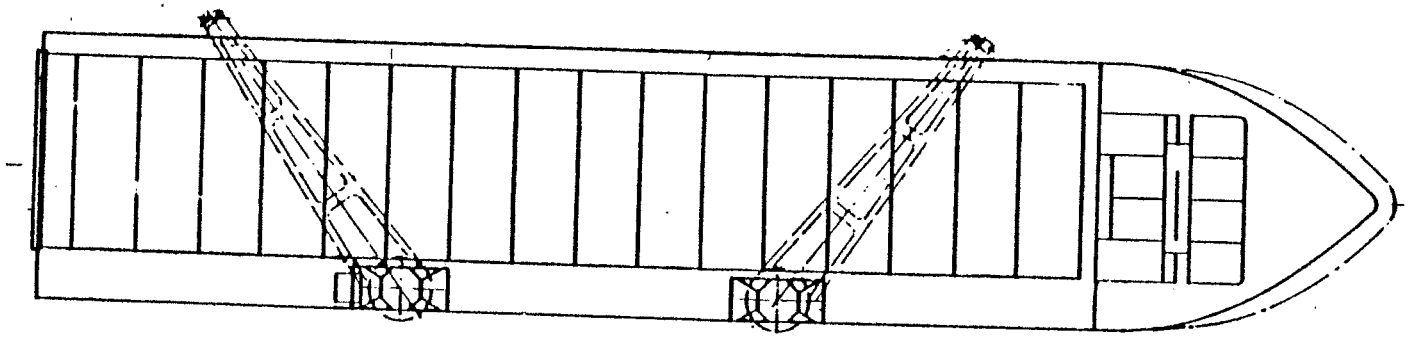


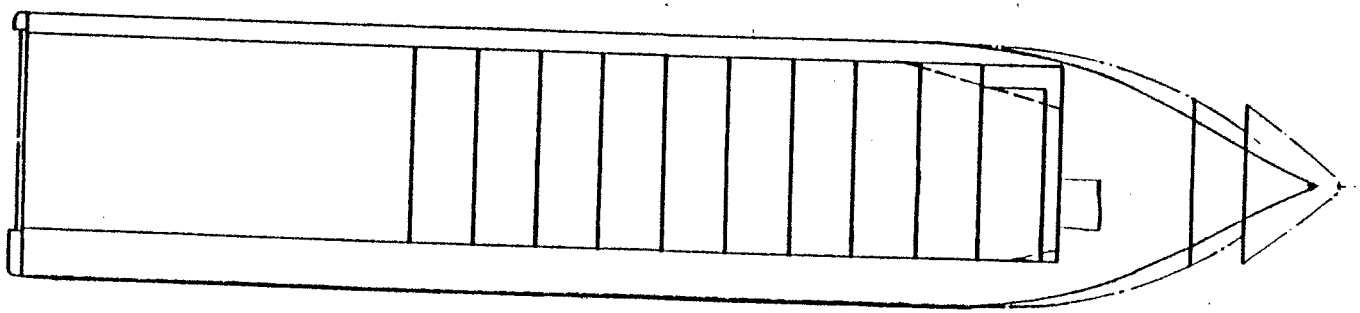
HEAVY LIFT / ROLO / CONTAINER CARRIER - NEWBUILDING PROJECT FOR MAMMOET SHIPPING, 1983.

(Note. Drawing not on scale)









Length o.a.	145.50	M
Length p.p.	134.00	M
Breadth	28.3	M
Depth	14.80	M
Draught (dsn)	7.00	M
Deadweight (")	9000	T
Speed (svc)	14	Kn.
Engine output	2 x 3800	KW
Lifting capacity	2 x 550	T
Aux. hoists	2 x 25	T

	Sq.M	Cub.M	Contnr.Cap.
L.Hold	1320	7100	155 TEU
Twn.deck	2200	11900	256 TEU
Weath.dk	3180	(4 trs)	656 TEU
Totals	6700	19000	1076
	Sq.M	Cub.M	TEUS

Stern ramp LxW = 7½ x 20 M
Cap. 2500 T.

15.1.03

15

16.0 CONTAINER FEEDER
AND GEN CARGO SHIPS.

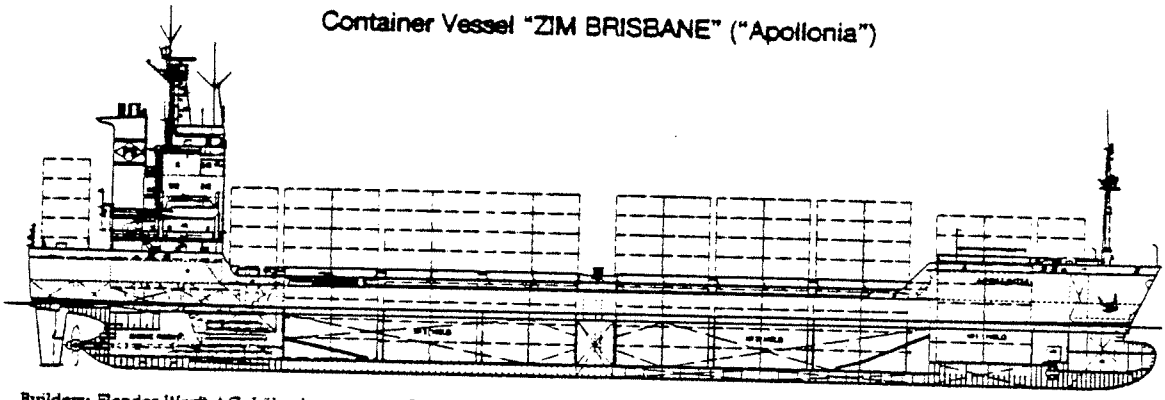
GEARED AND GEARLESS

SEA SERVICE

RIVER-CANAL SERVICE

CONTR. FEEDER

Container Vessel "ZIM BRISBANE" ("Apollonia")



Builders: Flender Werft AG, Lübeck
Owner: Peter Döhle Schiffahrts-KG (GmbH + Co.), Hamburg
Yard number: 631
Delivery: April 29th, 1983

Tonnage	10 500	GRT
Deadweight	14 400	t
Length o. a.	147.50	m
Length b. p.	140.00	m
Moulded breadth	22.80	m
Depth	11.20	m
Draught	8.43	m

Speed: 17.4 kn
Classification:
G.L. + 100 A4 E + MC E AUT
eingerichtet für Containertransport, verstärkt für Schwergutladung

Propelling machinery:
MaK-Diesel engine, type 8 M 601, 8088 kW at 425 rpm
reduction gear (425 : 132), HSN 1000.V (Tacke)
1 coupling EZR 3522 (Vulkan)
controllable pitch propeller
1 gear generator 1000 kW
Fuel consumption: 31 t/day (at 15.8 kn)

Auxiliary engines:
3 Diesel engines, 486 kW (660 PS) at 720 rpm each
3 generators 450 kW each
bow thruster BU 80 F (Jastram)

Crew: 26 persons
gyro compass and automatic pilot (Anschütz)

2 VHF equipments, weather chart facsimile recorder, satellite navigator (Debeg)

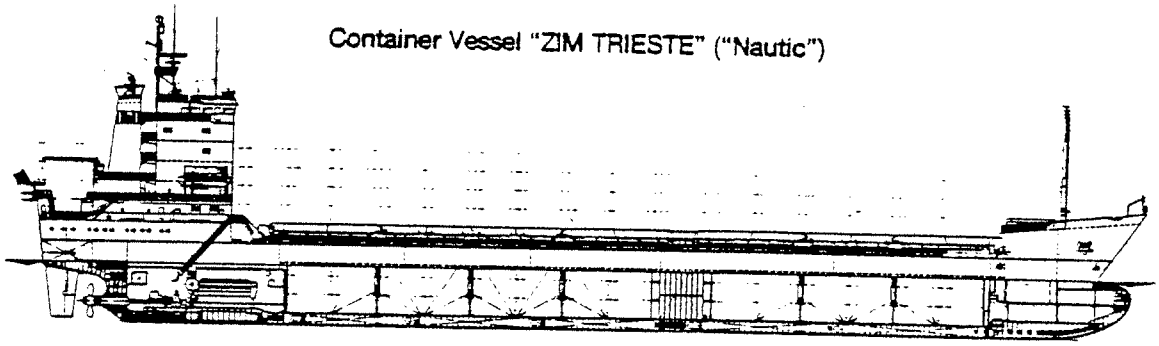
Hatch covers: system Kvaerner, folding/rolling type

Loading gear:
1 provisions crane
Winches:
2 comb. anchor/mooring winches 150 kN (forward),
1 comb. anchor/mooring winch and 1 mooring winch (after deck)

Container capacity:

holds	368	TEU
on deck	546	TEU
Total	914	TEU

Container Vessel "ZIM TRIESTE" ("Nautic")



Builders: O & K Orenstein & Koppel Aktiengesellschaft, Werk Lübeck
Owner: Peter Dörfler Schiffahrts KG, Brake
Yard number: 768
Delivery: April 15th, 1983

Tonnage	abt.	10 400	GRT
		7 000	NRT
Deadweight		14 033	t
Length o. a.		153.62	m
Length b. p.		142.50	m
Moulded breadth		22.80	m
Depth to maindeck		11.10	m
Draught		8.25	m

Speed (90% MCR): 16.8 kn

Classification:
G.L. + 100 A 4 E G eingerichtet für Containertransport + MC AUT C

Propelling machinery:
MaK-Diesel engine
type: 8 M 601 supercharged
7350 kW at 425/150 rpm
reduction gear HSN 950.A (Tacke)
1 coupling EZR 3522 (Vulkan)
Fuel consumption: 31 t/day

Auxiliary engines:
2 Yanmar-Diesel engines
type: 6 UAL-UT — 558 kW at 900 rpm each
2 generators 635 kVA, 450 V, 60 Hz each
1 shaft-driven generator 550 kW
compressors (Hatlapa)
2 CJC-finefilters HDU 38/100 (Karberg & Hennemann)
"Tornado"-bow thruster — thrust 84 kN, 585 kW (O & K)

2 Raytheon-radars, Koden-visual direction finder, satellite navigator, weather chart facsimile recorder, wireless station and 2 VHF equipments
gyro compass and automatic pilot (Anschütz)

3 hatches

hatch 1	6.60 × 8.00 m
hatch 2	38.25 × 18.50 m
hatch 3	51.00 × 18.50 m

Hatch covers: system NAVIRE CARGO GEAR folding type

1 electric hydraulic mooring winch 80 kN,
1 electric hydraulic mooring winch 80 kN comb. with stern anchor drum and
2 electric hydraulic comb. windlasses/mooring winches 58 mm K3 chain, 80 kN (Hatlapa)

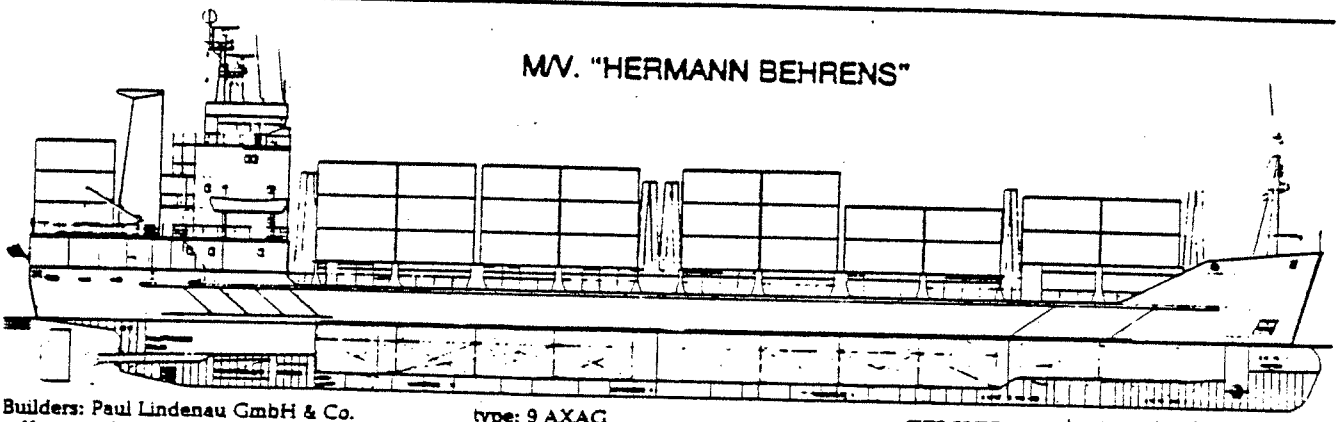
Hold capacity:

	grain
hold 1	463.5 m³
hold 2	7 582.0 m³
hold 3	10 686.6 m³
Total cargo capacity	18 732.1 m³

Container capacity:	TEU	or 40'	+ 20'
holds	370	176	18
on deck	563	234	95
Total	933	410	113

85 sockets for refrigerated containers on deck

M/V. "HERMANN BEHRENS"



Builders: Paul Lindenau GmbH & Co. Kommanditgesellschaft Schiffswerft und Maschinenfabrik, Kiel-Friedrichsort
Owner: Herbert Behrens & Sohn KG, Jork-Estebrügge
Type: full scantling vessel
Yard number: S 205
Delivery: December 30th, 1982

Tonnage	1 599 GRT
	1 190 NRT
Deadweight	4 150 t
Length o. a.	99.95 m
Length b. p.	95.29 m
Moulded breadth	15.70 m
Depth to maindeck	8.00 m
Depth to tweendeck	5.29 m
Draught	5.10 m

Speed: max. 14.5 kn

Classification:
G.L. + 100 A 4 E G "KORR" verstärkt für Schwergutladung, eingerichtet für Containertransport + MC E AUT

Propelling machinery:
Wichmann-Diesel engine
type: 9 AXAG
1460 kW at 475/149 rpm
(max. 2647 kW)
heavy oil operation IF 180
reduction gear (Reintjes)
EZR-coupling (Vulkan)
controllable pitch propeller
1 shaft driven generator 265 kVA

Auxiliary engines:
2 Cummins-Diesel engines
type: NT 855 G (M)
233 kW at 1800 rpm each
2 generators 244 kVA, 440 V, 60 Hz each
1 Cummins-Diesel engine
type: N 855 G (M) — 139 kW at 1800 rpm
1 generator 160 kVA, 440 V, 60 Hz

2 separators OSA 7 and 1 separator OTB 2 (Westfalia)
bow thruster 184 kW (Hatlapa)
rotary vane steering gear type HS 90 D. (Frydenbø)
2 Decca-radars, direction finder, echo sounder, weather chart facsimile recorder, satellite navigator, wireless station and 2 VHF equipments (Debeg)
gyro compass and automatic pilot (Anschütz)

3 hatches
hatch 1 — 12.38 × 10.12 m
hatch 2 and 3 each — 24.64 × 12.58 m
Hatch covers: system NAVIRE CARGO GEAR folding type

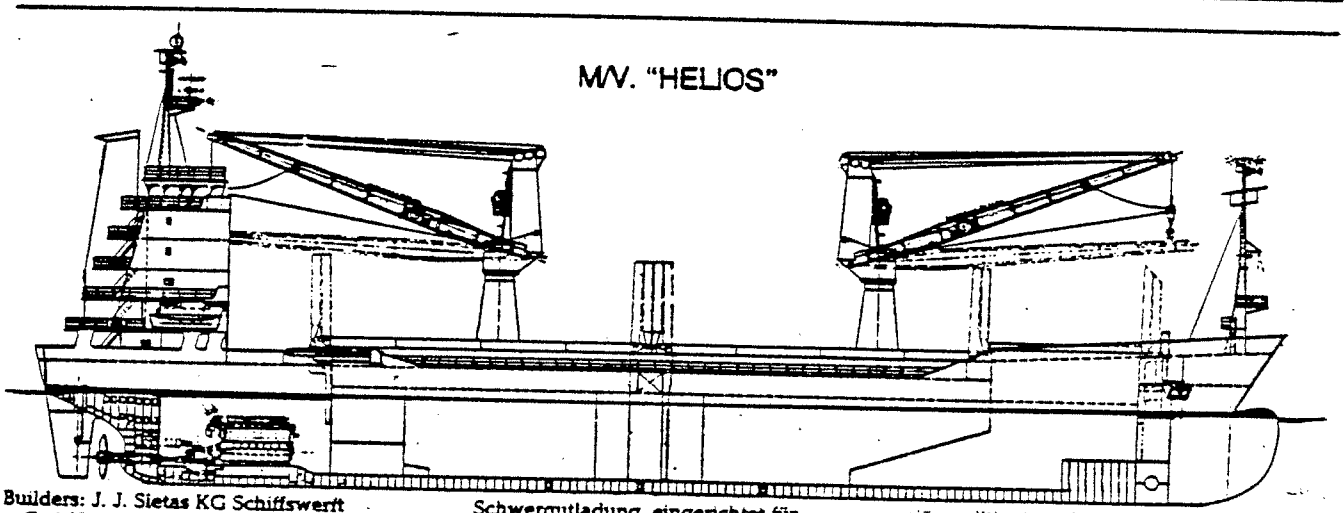
2 electric automatic mooring winches and 2 electric comb. windlasses/automatic mooring winches (Steen)

Hold capacity:	grain m³	bale m³
hold 1	1 397	1 250
hold 2	2 953	2 900
hold 3	2 924	2 892
Total cargo capacity	7 274	7 042

Container capacity:	TEU	or 40'	+ 20'
holds	140	68	4
on deck	165	72	9
Total	305	140	13

30 sockets for refrigerated containers
Container lashing and securing material (CONVER)

M/V. "HELIOS"



Builders: J. J. Sietas KG Schiffswerft GmbH & Co., Hamburg-Neuenfelde
Owner: Partenreederei MS "Helios", Bad Schwartau
Managing owner: Hans-Hermann Knüppel, Hamburg-Finkenwerder
Yard number: 903
Delivery: September 3rd, 1982

Tonnage	abt. 3 890 GRT
	2 470 NRT
Deadweight	7 785 t
Length o. a.	106.46 m
Length b. p.	98.60 m
Moulded breadth	19.00 m
Depth to maindeck	10.40 m
Depth to 2nd deck	8.60 m
Draught	7.72 m

Speed: abt. 15 kn
Classification:
G.L. + 100 A 4 E G verstärkt für Schwergutladung, eingerichtet für Containertransport + MC E AUT

Propelling machinery:
MaK-Diesel engine
type: 8 M 551 AK supercharged
2942 kW (4000 HP)
heavy oil operation up to IF 380
reduction gear HSU 800.D (TACKE-SCHIFFSGETRIEBE®)
1 coupling EZS 3601 (Vulkan)
controllable pitch propeller (Lips)
1 shaft driven generator 560 kVA

silencer type ES (Huss)
3 separators OSA 20 and 2 separators OTA 7 (Westfalia)
boiler AQ 5 (Aalborg)
fresh water evaporator (Atlas-Denmark)
bow thruster 60 F — 316 kW (Jastram)
2 radars (Atlas)
gyro compass and automatic pilot (Anschütz)
air conditioning plant according to the Duovent-system (Noske-Kaeser GmbH/ Svenska Fläktfabriken)

3 hatches
hatch 1 — 12.60 × 10.20 m
hatch 2 and 3 each — 25.10 × 15.30 m
Hatch covers: system NAVIRE CARGO GEAR folding type

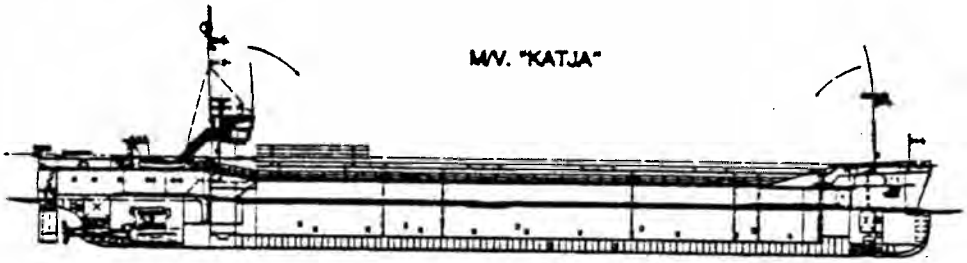
Loading gear: 2 electric hydraulic cranes of 35 t/24.5 m (NMF)

Total cargo capacity:		
grain	abt. 10 492	m³
bale	10 341	m³

Container capacity:	TEU	or 40'	+ 20'
holds	210	100	10
on deck	203	85	32
Total	413	185	42

20 reefer plugs

M/V. "KATJA"



Builders: J. J. Sietas KG Schiffswerft GmbH & Co., Hamburg-Neuenfelde
Owner: Jürgen Stahmer KG, Jork
Yard number: 920
Type: full scantling vessel
Delivery: December 11th, 1982

Tonnage	998.34	GRT
	676.67	NRT
Deadweight	2 880	t
Length o. a.	87.95	m
Length b. p.	85.30	m
Moulded breadth	11.30	m
Depth to maindeck	6.75	m
Depth to tweendeck	4.88	m
Draught	4.68	m

Speed: 11.5 kn
Classification:
G.L. + 100 A 4 M verstärkt für Schwergutladung G, eingerichtet für Containertransport + MC AUT

Propelling machinery:
Deutz-Diesel engine
type: SBV 8 M 628
735 kW (999 HP) at 684 rpm
Fuel consumption: abt. 5 t IF 30 per day
marine reverse reduction gear NAVILUS® GWC 42.45 (Lohmann & Stolterfoht)
1 shaft shrink disc flange coupling (Stihwe)
2 Diesel generators 190 kVA each
silencer type ES (Huss)
separators (Westfalia)
bow thruster type 20 F — 147 kW (Jastram)
gyro compass and automatic pilot (Anschütz)
2 radars, vessel direction finder, weather chart facsimile recorder, satellite navigator, wireless station and 2 VHF equipments (Hagenuk)
air conditioning plant according to the Duovent-system (Noske-Kaeser GmbH/ Svenska Fläktfabriken)
1 hatch 55.90 × 9.30 m

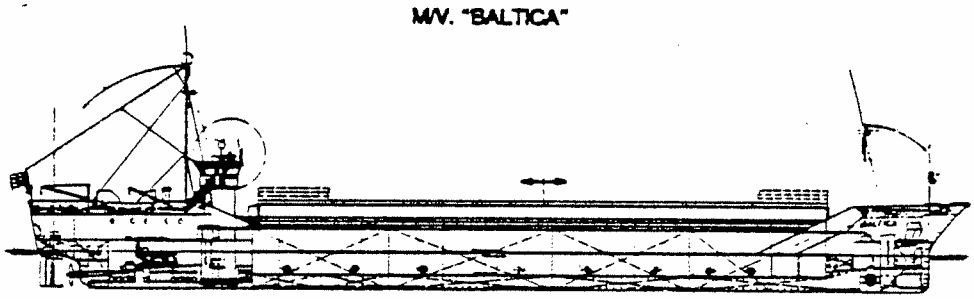
Hatch covers: system NAVIRE CARGO GEAR piggy-back type
1 electric bow windlass (Steen)
1 electric stern rope anchor windlass (Steen)
Total cargo capacity:
grain 3 777 m³
bale 3 752 m³
90 TEU
Container lashing and securing material (CONVER)

3752 m3
90 teu
1000 GRT
T 4.68 m
Loa 88 m

Sistership M/V. "ILKA"

Builders: J. J. Sietas KG Schiffswerft GmbH & Co., Hamburg-Neuenfelde
Owner: MS "Ilka" Hartmann Schiffahrts-KG Leer
Yard number: 907
Delivery: December 30th, 1982

M/V. "BALTICA"



Builders: Schiffswerft Hugo Peters Weweisfleth, Weweisfleth
Owner: MS "Baltica" Beutler Schiffahrts-KG, Lübeck-Travemünde
Yard number: 592
Type: tweedecker
Delivery: February 15th, 1983

Tonnage	499	GRT
	315	NRT
Deadweight	1761	t
Length o. a.	82.45	m
Length b. p.	76.80	m
Moulded breadth	11.30	m
Depth to maindeck	5.40	m
Depth to tweendeck	3.31	m
Draught	3.55	m
Draught ballast	abt. 2.65	m

Trial speed: 10.6 kn
Classification:
G. L. + 100 A 4 M E G mit Freibord 1.87 m, verstärkt für Schwergutladung, eingerichtet für Containertransport, + MC E AUT

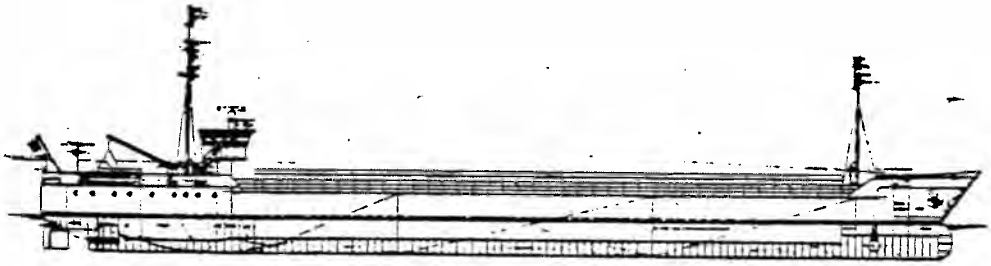
Propelling machinery:
Deutz-Diesel engine
type: SBA 8 M 528
441 kW (600 HP) at 620 rpm
(max. 783 kW)
reduction gear i = 2.542:1 (Reintjes)
Hinze-flaprudder R 2040.K
1 shaft driven generator 57 kVA
Auxiliary engines:
1 Deutz-Diesel engine
type: BA6M 816 aLLK
184 kW at 1500 rpm
1 generator 210 kVA, 380 V, 50 Hz
1 Deutz-Diesel engine
type: F5L 413 FR — 68 kW at 1500 rpm
1 generator 75 kVA, 380 V, 50 Hz
compressors (Hatlapa)

1 CJC-filger-separator FWFU 27/108 and 1 CJC-finefilter HDU 38/100 (Karberg & Hennemann)
bow thruster type 20 F — thrust 2 t, 132 kW (Jastram)
steering gear type HS 30 D. (Frydenbø)
2 Decca-radars, wireless station and 2 VHF equipments (Debeg)
Sirius-gyro compass and Decca-automatic pilot (W.G. Schutz)
Crew: 6 persons
1 hatch 49.80 × 9.00 m
Hatch covers: system MacGREGOR multi-stacking piggy back type
1 hydraulic bow windlass and 1 electric stern rope anchor windlass (Steen)

Total cargo capacity:
grain 2914 m³
bale 2909 m³
48 TEU

2809 m3
48 teu
500 GRT
T 3.55 m
Loa 8.25 m

M/V. "GERHARD PRAHM"



Builders: Schiffswerft Gebr. Köther, Haren/Ems
Owner: Wilfried Hammann, Wiechbaren and Günther Prahm, Leer
Yard number: 73
Delivery: November 24th, 1982

Tonnage	299	GRT
Deadweight	1 089	t
Length o. a.	75.00	m
Length b. p.	70.50	m
Moulded breadth	9.46	m
Depth to maindeck	4.90	m
Depth to tweendeck	3.05	m
Draught	2.88	m
Draught ballast	2.60	m

Speed: 10 kn
Classification:
G.L. + 100 A 4 M
Propelling machinery:
Deutz-Diesel engine
type: SBA 6 M 528
441 kW (600 HP) at 661 rpm
propeller (Ostermann)
1 CJC-filter separator FWFU 27/54 (Karberg & Hennemann)
bow thruster type S 100/103 ZK — 150 kW (Schottel)

gyro compass and automatic pilot (Anschütz)
1 hatch 46.80 × 7.34 m
Hatch covers: system MacGREGOR multi-stacking piggy-back type
1 hydraulic bow windlass (Steen)
1 electric stern rope anchor windlass (Steen)
Total cargo capacity:
grain 1 882.70 m³
bale 1 866.20 m³
Constructed by: Ingenieurbüro Franz Steinkopf, Leer-Loga

1866 m3
300 GRT
T 2.88 m
Loa 75 m

17.0

O P E N "C O A S T E R"

GEARED WITH 35-TON CRANES.

Suitable for

Steel plate, sections, pipes, reels

Bulk grain, maize, nuts, china clay, baryte.

Drums chemicals, oils,

Prefab concrete sections for houses,

Containers loaded with machine parts; empties (up to 3 tiers)

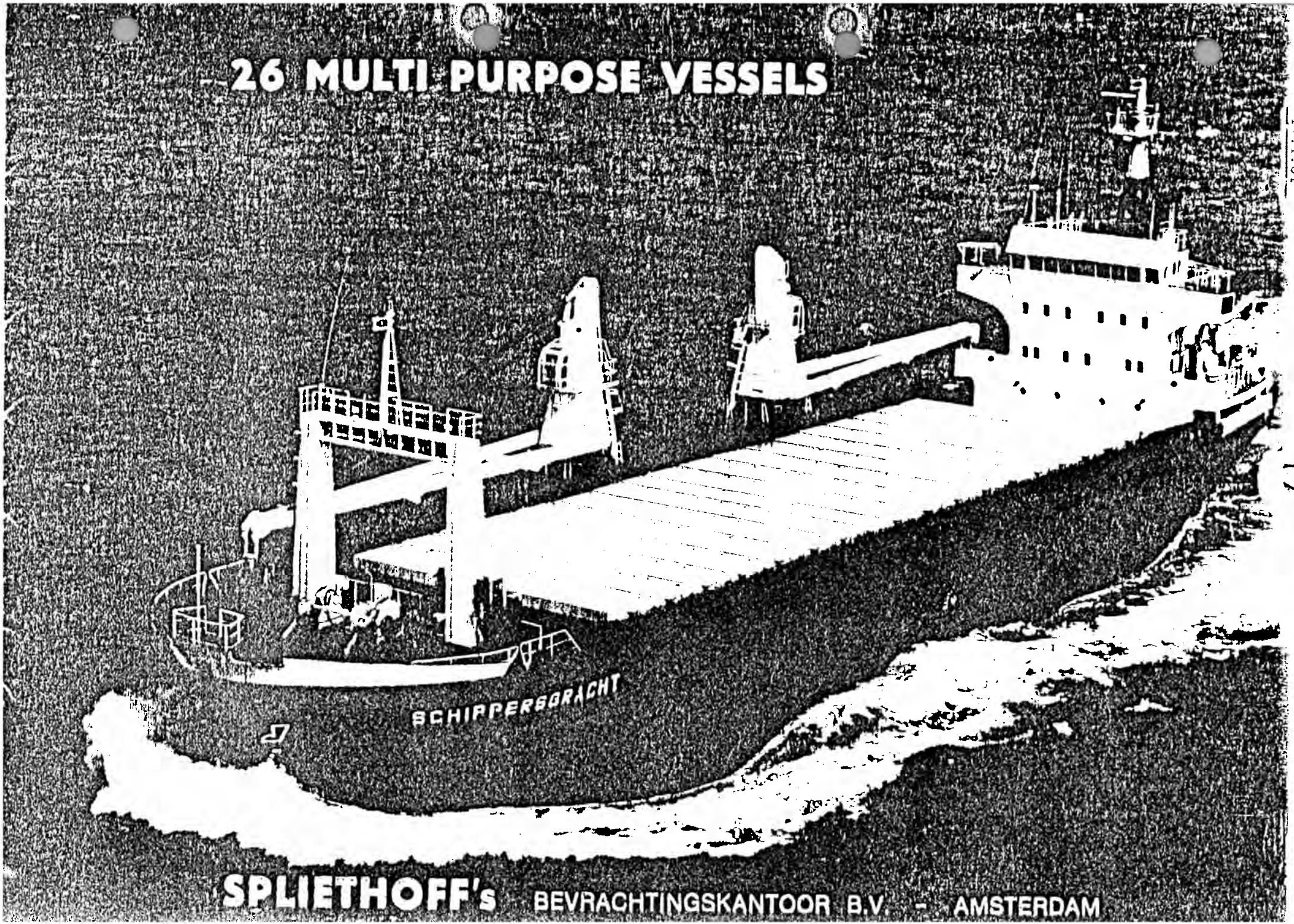
Waggons, busses, tractors.

Tugs, launches, fishing vessels, small cutters.

Dredging equipment.

Logs, lumber, packaged timber, paper.

OPEN COASTER


26 MULTI PURPOSE VESSELS
SCHIPPERSGRACHT
SPLIETHOFF's BEVRACHTINGSKANTOOR B.V. - AMSTERDAM

26 OPEN/CLOSED SHELTERDECK VESSELS — CONVERTIBLE SINGLEDECKERS

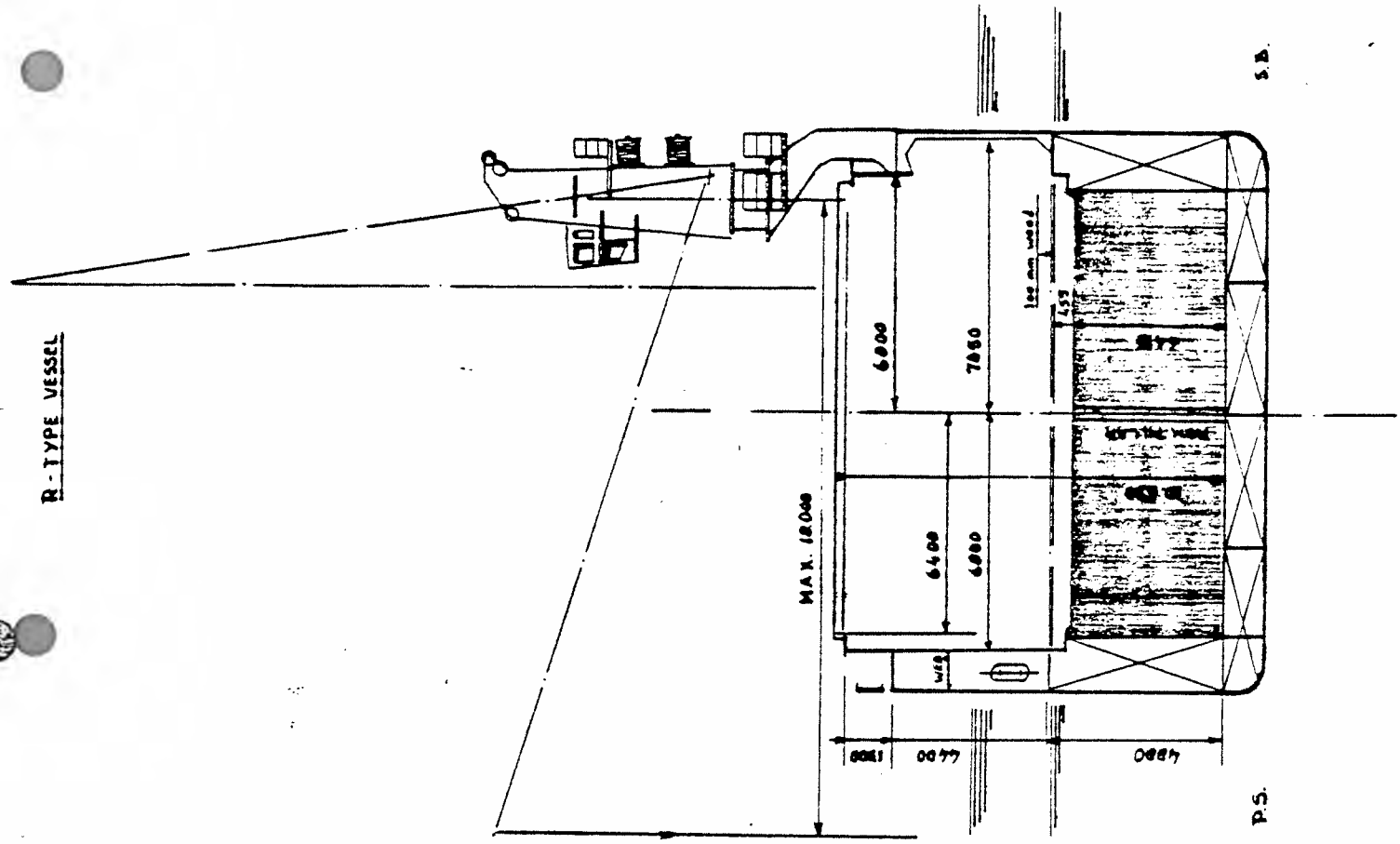
for GENERAL CARGO/PACKAGED TIMBER/PAPER/CONTAINERS/BULK/GRAIN/FRUIT/PIPES ETC.

	L-TYPE	S-TYPE	B-TYPE	R-TYPE
CLASS ICECLASS Finnish/Swedish	LLOYDS' 100 A1 ✚ LMC 1A upto 5.97 m	do. 1A upto 5.97 m	do. 1A upto 7.25 m 1B upto 8.48 m	do. 1A upto 5.97 m —
LENGTH o.a.	80.20 m	do.	do.	do.
LENGTH b.p.	74.60 m	do.	do.	do.
BREADTH moulded	16.00 m	do.	do.	do.
DEPTH moulded to upperdeck	10.50 m	do.	do.	do.
DEPTH moulded to tweendeck	6.10 m	do.	do.	do.
SUMMERDRAFT open/closed	5.98/8.47 m	do.	do.	do.
G.R.T. open/closed	1599/3500	do.	do.	do.
N.R.T. open/closed	1122/2295	do.	do.	do.
DEADWEIGHT open/closed	3480/6175 MT	3480/6175 MT	3435/6130 MT	3605/6301 MT
CAPACITY grain	L.H. 106.000 + U.H. 152.000 = 258.000 cft	L.H. 106.000 + U.H. 152.000 = 258.000 cft	L.H. 101.500 + U.H. 152.000 = 253.500 cft	L.H. 108.000 + U.H. 152.000 = 260.000 cft
bale	L.H. 105.000 + U.H. 148.000 = 253.000 cft	L.H. 105.000 + U.H. 148.000 = 253.000 cft	L.H. 100.700 + U.H. 148.000 = 248.700 cft	L.H. 107.000 + U.H. 149.000 = 256.000 cft
BALLASTCAPACITY f.w.	1175 MT	1220 MT	1129 MT	1175 MT
TIME BALLASTING/UNBALL.	5 hrs	do.	do.	do.
AIRCHANGES (empty holds)	30 x	do.	do.	do.
FRESHWATER	33 MT	do.	do.	do.
FRESHWATER EVAPORATOR	yes	do.	do.	do.
CONTAINER INTAKE	ISO 20' / ISO 40' + 20'	ISO 20' / ISO 40' + 20' / ISO 35'	ISO 20' / ISO 40' + 20'	ISO 20' / ISO 40' + 20' / ISO 35'
HOLD units	112 / 40 + 32	136 / 58 + 20 / 76	136 / 58 + 20	136 / 58 + 20 / 76
ON DECK units	29 + 29* / 15	29 + 29* / 17 / 21	29 + 29* / 17	29 + 29* / 17 / 21
total units	141 (170*) / 55 + 32	165 (194*) / 75 + 20 / 97	165 (194*) / 75 + 20	165 (194*) / 75 + 20 / 97
*) 2nd tier subject to weight & stability				
HATCHES weatherdeck:	44.40 x 12.80 m Steel, end rolling type	do. do.	do. do.	do. do.
tweendeck:	44.40 x 12.80 m Steel, side folding type beams & pillars removable	do. do. do.	do. do. do.	46.80 x 12.80 wood beams & pillars removable
MAXIMUM LOAD weatherdeck hatchcovers	1.5 t/M²	do.	do.	1.5 t/M²
tweendeck hatchcovers	3.7 t/M²	do.	do.	1.93 t/M²
tanktop	15.0 t/M²	do.	do.	15.0 t/M²
DECKCRANES situated on starboardside	2 of 10 MT	2 of 15 MT	2 of 15 MT	2 of 10 MT
reach	all 10 t/18 m + some 8 t/21 m	22 m	22 m	all 10 t/18 m + some 8 t/21 m
MAINENGINE	3500 HP	3000 HP	3000 HP	3000 HP
speed (draft 5,97 m)	abt 13 Kn.	do.	do.	do.
speed (draft 8,47 m)	abt 11 Kn.	do.	do.	do.
fuel consumption p/day	abt 10.5 MT	do.	reduced MDO consumption owing to shaftgenerator and reversibel pitch propeller	abt 10.5 MT
BUNKERCAPACITY Int. Fuel Oil max. 30/60 cSt	350 MT	350 MT	435 MT	350 MT
Marine Diesel Oil	84 MT	58 MT	58 MT	84 MT

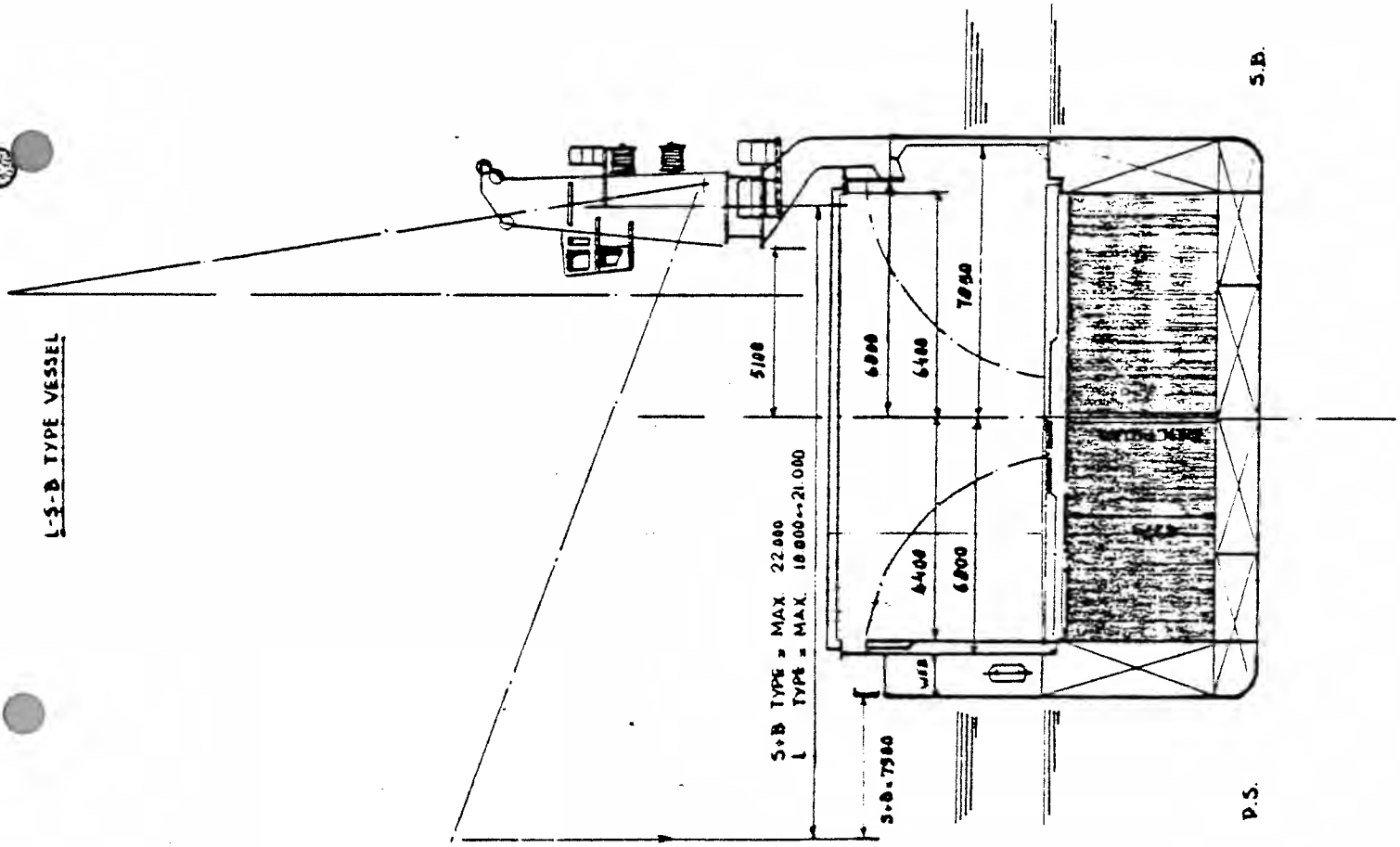
These particulars are believed to be correct, but without guarantee, and they must not be used as basis for charterparties or contracts without owners' explicit authority.

SPLIETHOFF's BEVRACHTINGSKANTOOR B.V. – AMSTERDAM – P.O. BOX 409 – DANZIGERKADE 8-10 – TELEPHONE 828905 – TELEX 12065

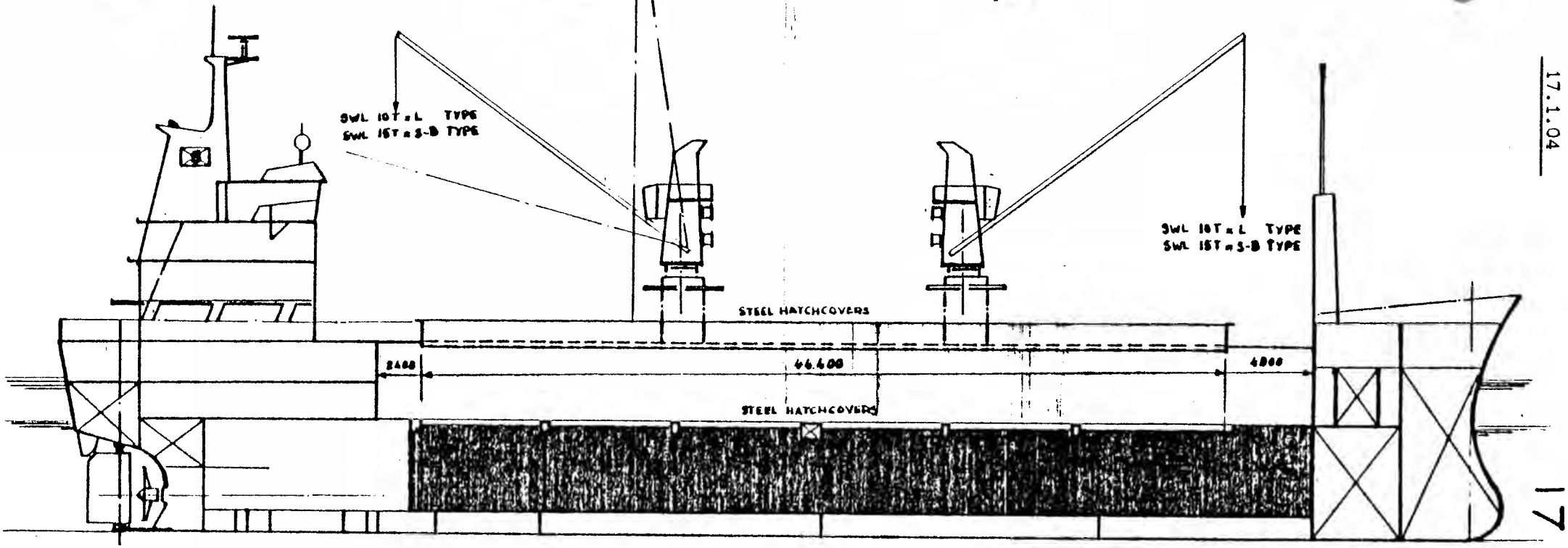
R-TYPE VESSEL



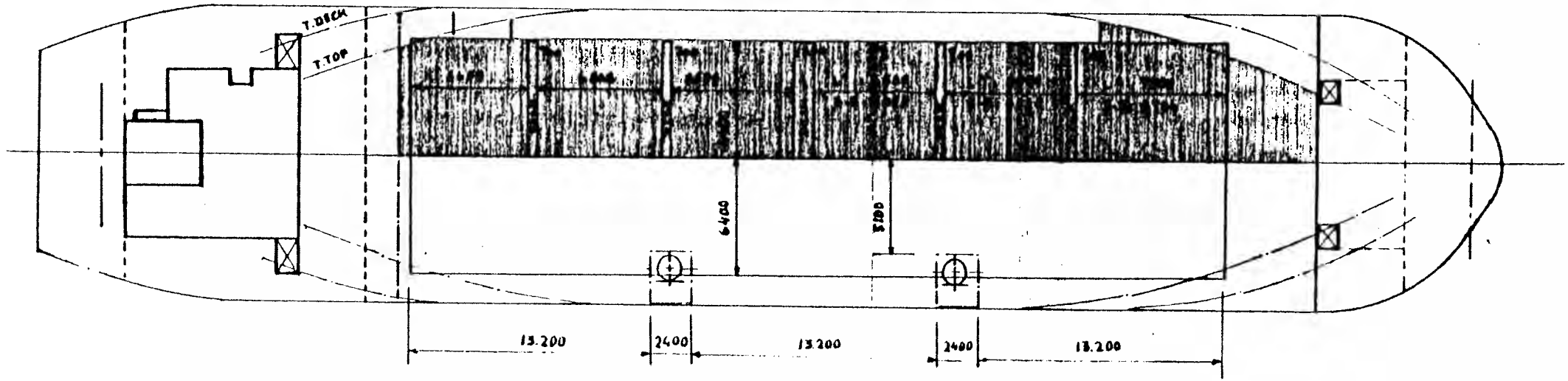
L-S-B TYPE VESSEL

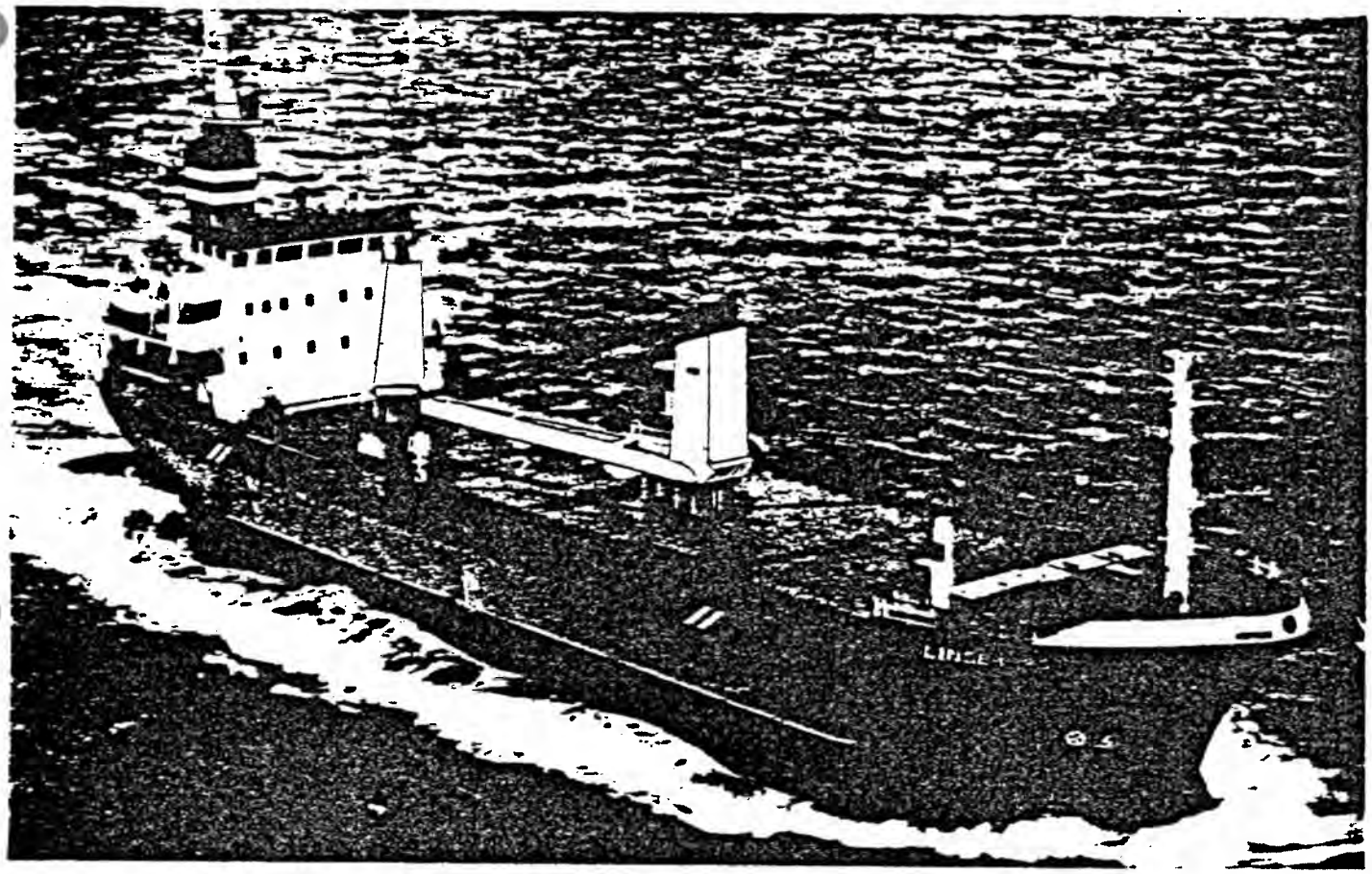


L-S-B TYPE VESSEL



BEAMS AND PILLARS REMOVABLE





m.v. 'Lindeborg'

new 3700 dwt coaster delivered by Nieuwe Noord Nederlandse Scheepswerven

B.V. Nieuwe Noord Nederlandse Scheepswerven at Groningen, Holland, has delivered the new 3,700 dwt coaster, m.v. 'Lindeborg', to the owners Linde Lloyd Zeeschepen II B.V. at Ursem.

The ship has one big cargohold with a total capacity of about 208,000 cbft (5,890 m³).

This hold is almost rectangular and provided with a tweendeck. It is served by two deck cranes of make N.M.F. The capacity of each crane is 16 tons at a spread of 20 m.

Propulsion is by a Wärtsilä main diesel engine which drives a KaMeWa propeller through a Tacke reduction gearbox and Vulkan highly elastic coupling.

The principal dimensions of m.v. 'Lindeborg' are:

length o.a.	82.37 m
length b.p.	74.53 m
breadth o.a.	15.83 m
breadth mld.	15.40 m
depth to upper deck	9.00 m
depth to tween deck	6.38 m
draught	5.98 m
deadweight	3,712 tonnes
gross tonnage	1,599 tons
net tonnage	1,217 tons
speed (service)	11.5 knots

The various tank and hold capacities are:

heavy fuel oil	272.7 m³
diesel oil	121.9 m³
water ballast	1,941.9 m³
freshwater	39.1 m³
lower hold	3,380.8
upper hold	1,815.3
hatch	693.8
total hold capacity	5,889.7 m³

m.v. 'Lindeborg' meets the classification requirements of Bureau Veritas I 3/3 E Cargo-Haute Mer and Iceclass IA.

Machinery

m.v. 'Lindeborg' is propelled by a Wärtsilä main diesel engine, type 6R32, developing 2,048 kW at 750 rpm. The speed of the main engine is reduced by a Tacke reduction gearbox, type HSN 800 C, with an incoming speed of 750 rpm and outgoing speed of 170 rpm (reduction ration 4.42 : 1).

The reduction gearbox is provided with a PTO.

Through the gearbox and Vulkan highly elastic couplings, type EZR 1723 the main engine drives a KaMeWa propeller of the controllable pitch type and with a diameter of 3,500 mm. The propeller is made of niel bronze.

The main engine and propeller are remotely controlled from the wheelhouse with a pneumatic control system which consists of a desk mounted main stand and engine room stand for remote control (also desk mounted), an automatic electronic load control system and an alarm transmitter. The arrangement meets the requirements for unmanned operation (O-man) according to the Netherlands Shipping Inspectorate. To facilitate manoeuvring the vessel is equipped with an electric bow thruster.

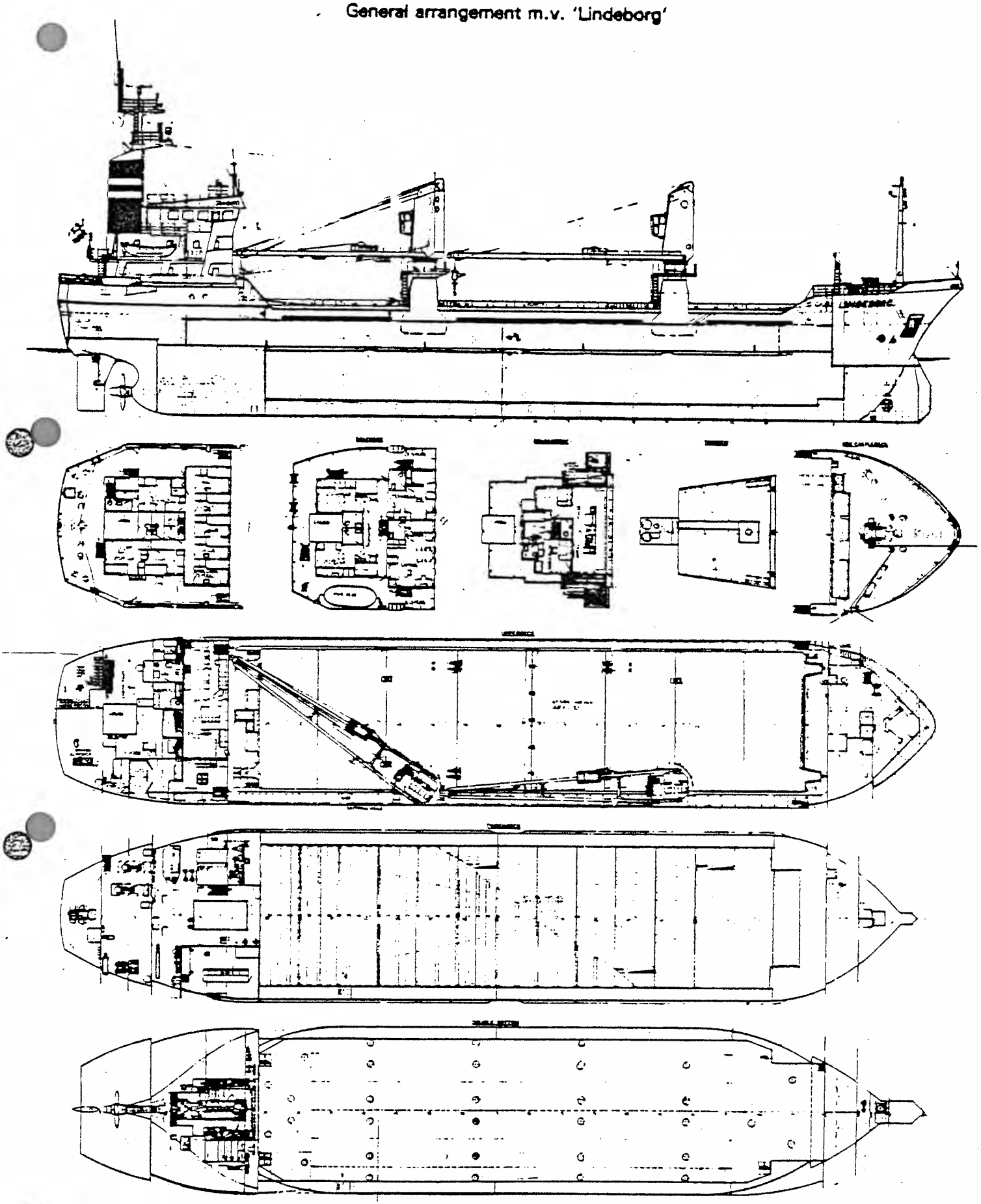
Through the step-up PTO and Vulkan highly elastic coupling type K3910, the Wärtsilä main engine also drives an Indar shaft generator at 1,500 rpm, 3-phase, 380 Volts, 50 Hz.

Other electric requirements are met by two auxiliary generator sets, each consisting of a Scania diesel engine, type DSI-1101 M, with a continuous output of 264 hp at 1,500 rpm, a flexible coupling and an Indar 180 kW generator at 1,500 rpm. The generators and auxiliary engines are placed in sound insulated casings to reduce noise levels.

The vessel is also equipped with a 63 Amp shore connection.

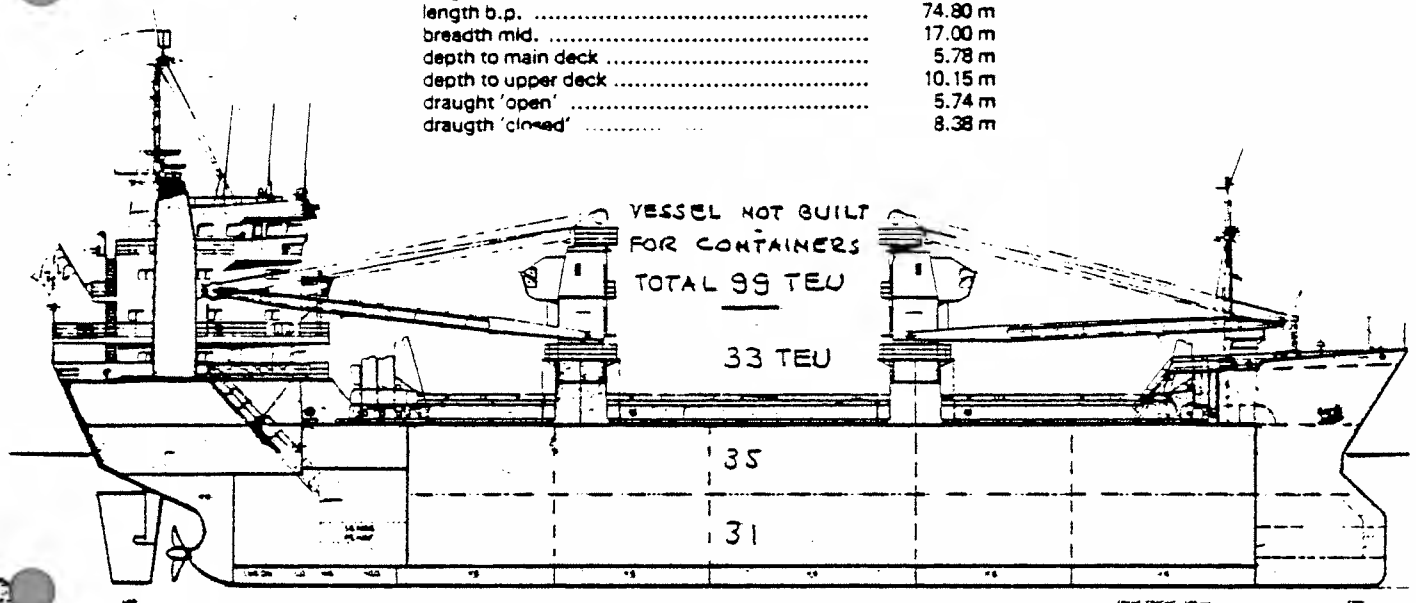
Other auxiliary machinery includes Nijhuis cooling water pumps and general service pumps, Leistritz fuel transfer pump, two Megator heavy

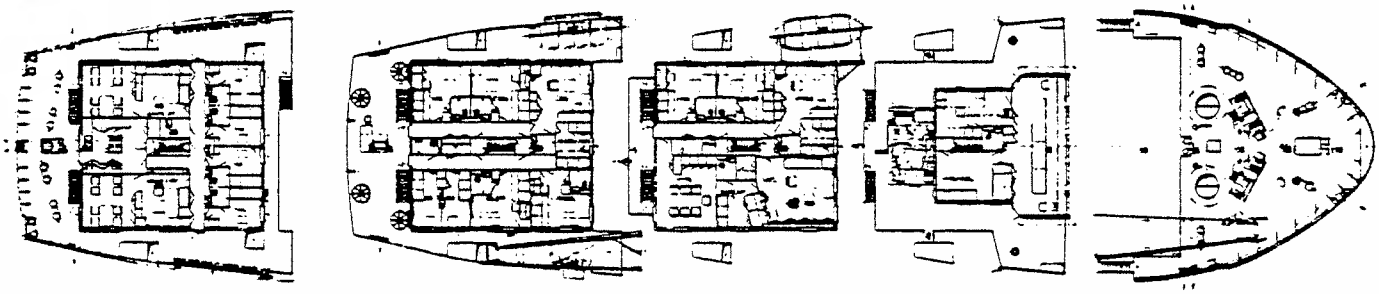
General arrangement m.v. 'Lindeborg'

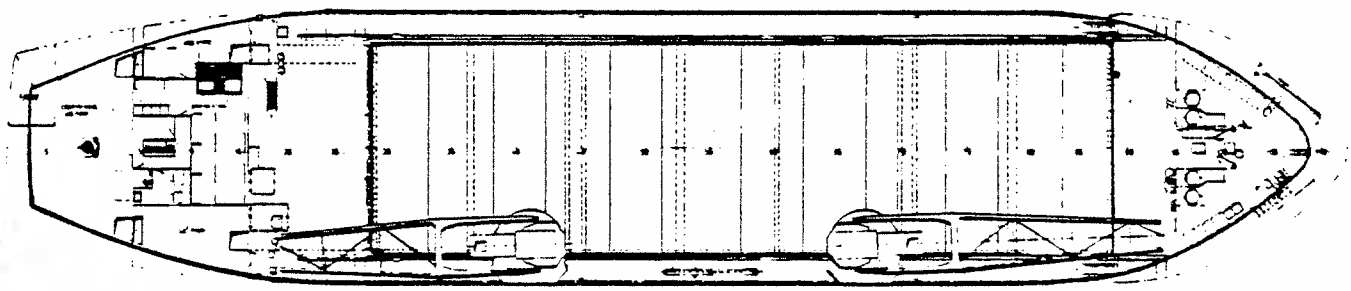


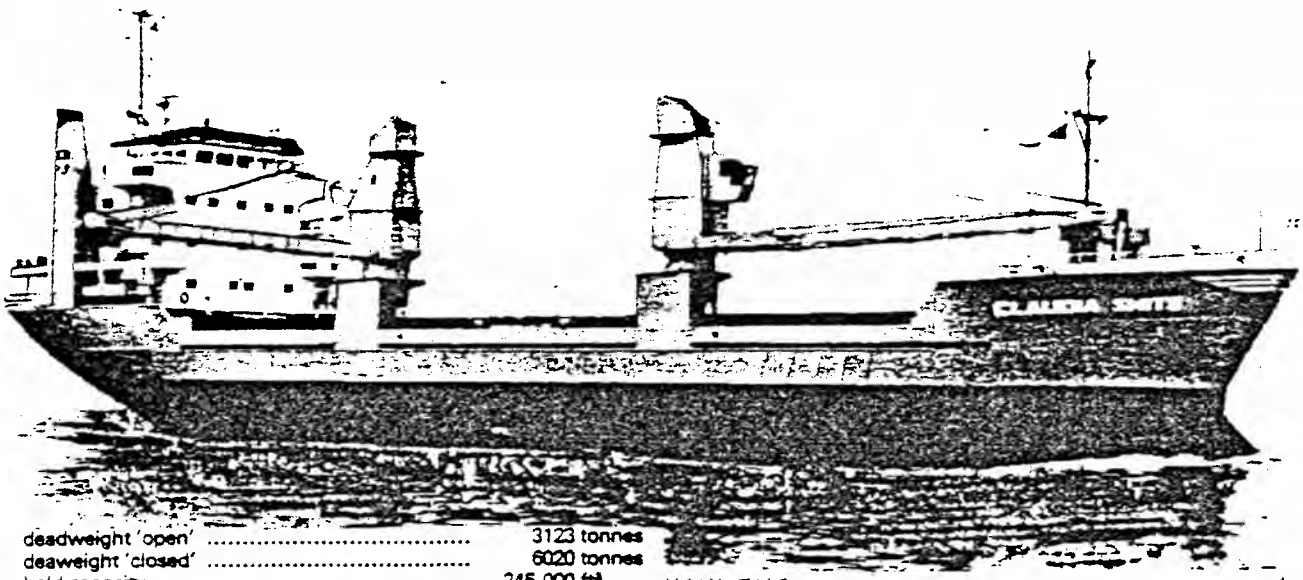
6,020 tonnes coaster "Claudia Smits"

length o.a.	84.18 m
length b.p.	74.80 m
breadth mld.	17.00 m
depth to main deck	5.78 m
depth to upper deck	10.15 m
draught 'open'	5.74 m
draught 'closed'	8.38 m



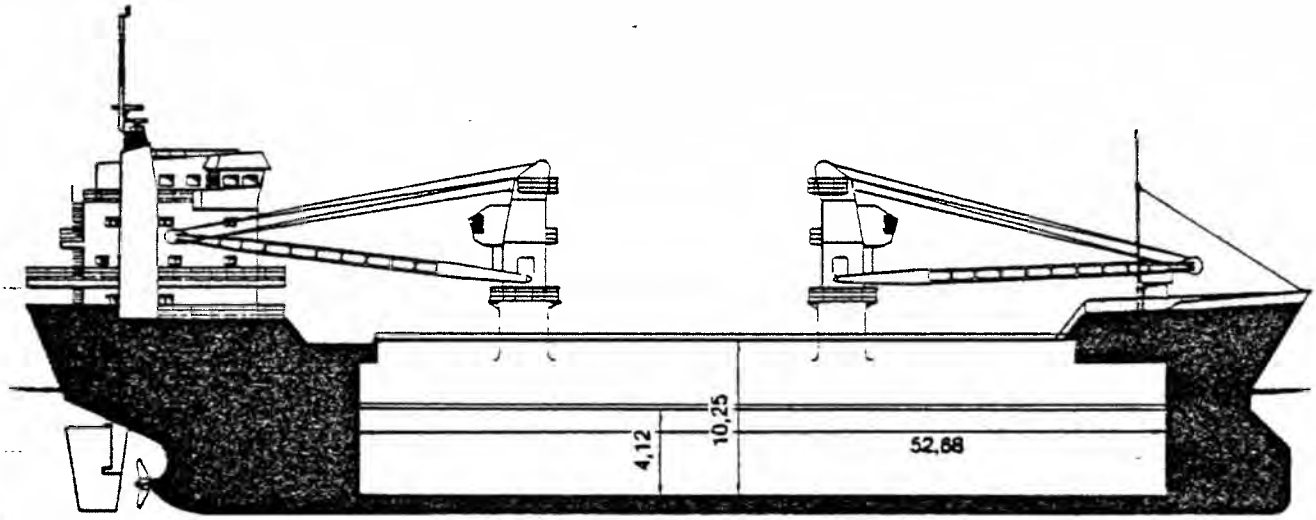




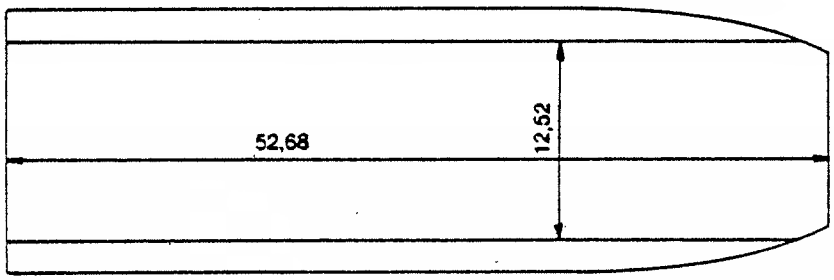


deadweight 'open'	3123 tonnes
deaweight 'closed'	6020 tonnes
hold capacity	245,000 ft³
fuel capacity	593 tons
water ballast capacity	1971 tons

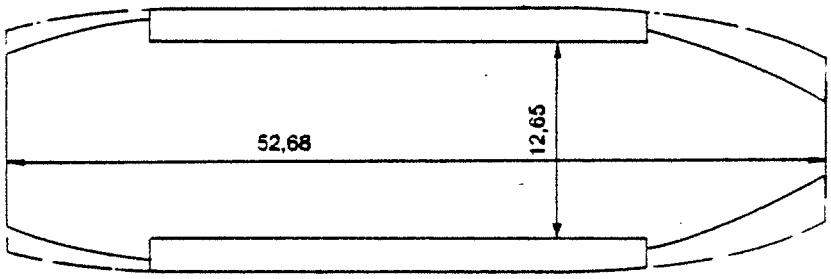
MAIN ENG SWD TM 410 6CYL 3100 BHP/580 RPM
SHAFT GEN 600 1:4.76
SHP 2500



Upper hold

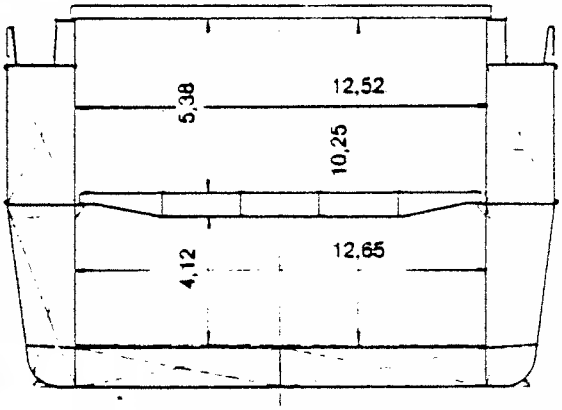


Lower hold



Hatch on upperdeck 45,58 x 12,52

Main section



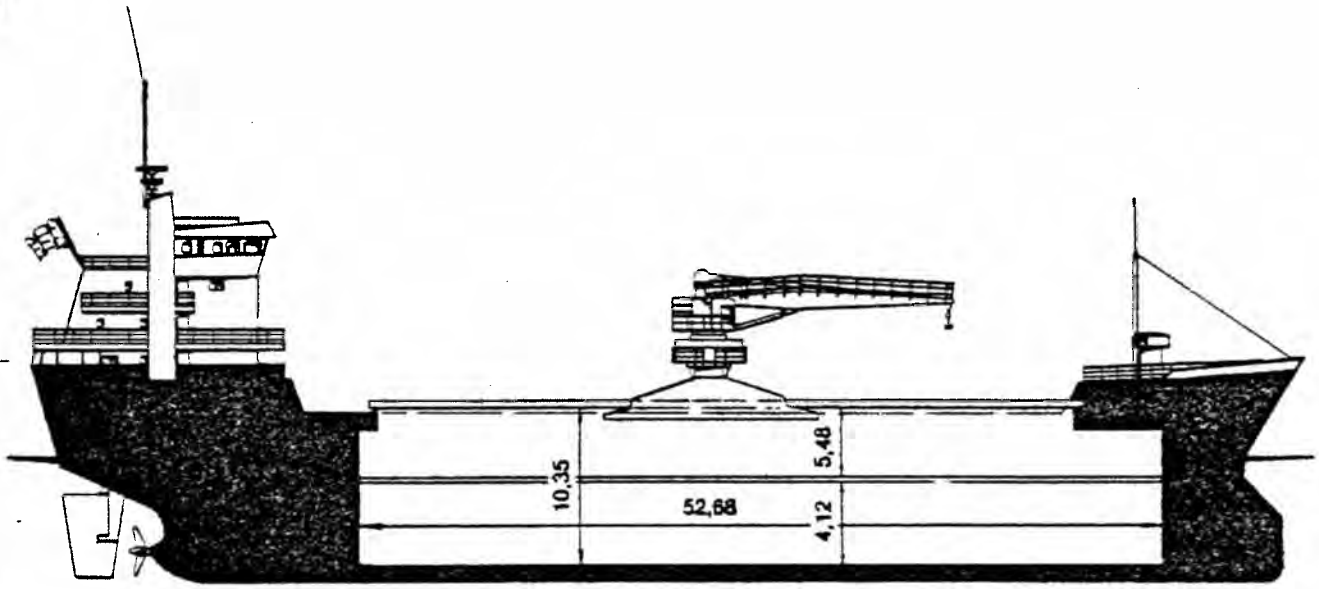
6020 dwat/230.000 cft bale.

Claudia Smits

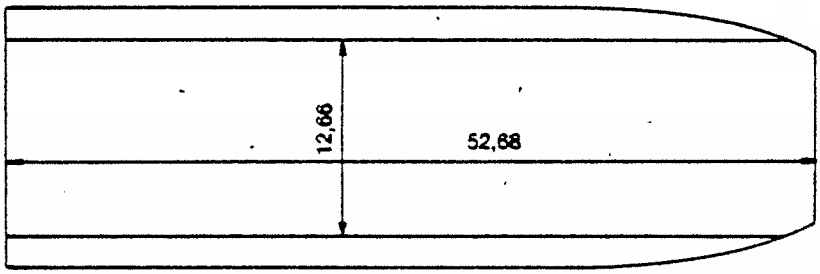
Cecilia Smits

Christina Smits

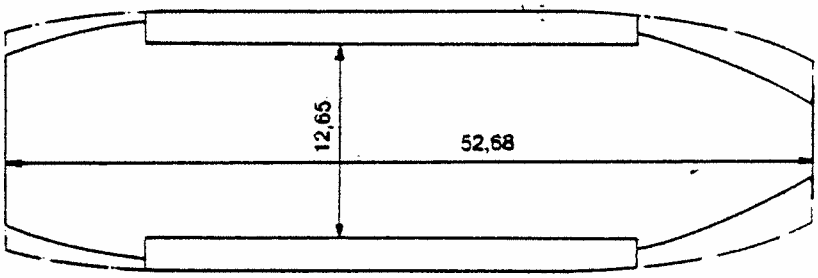
Crane capacity (fixed cranes)
2 cranes each 32 ton
combined 64 ton



Upper hold

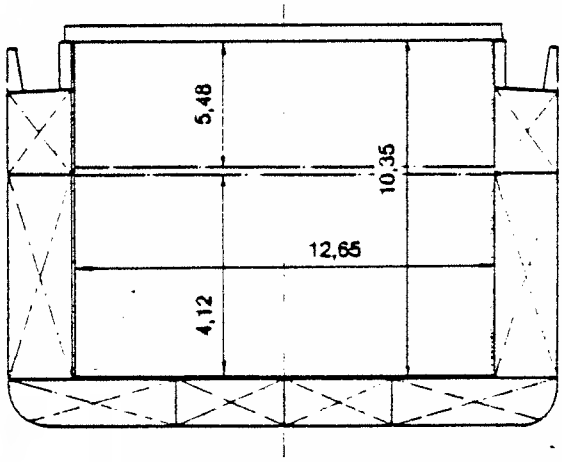


Lower hold



Hatch on upperdeck 45,58 x 12,66 –

Main section



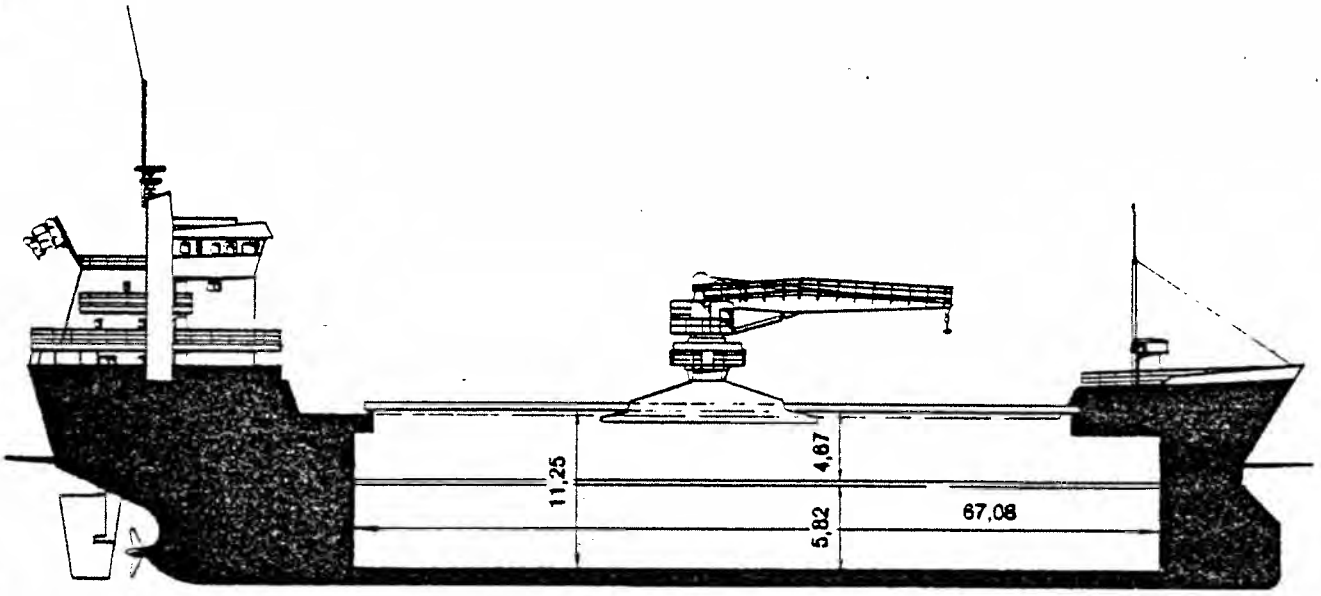
6110 dwat/230.000 cft bale.

Alida Smits

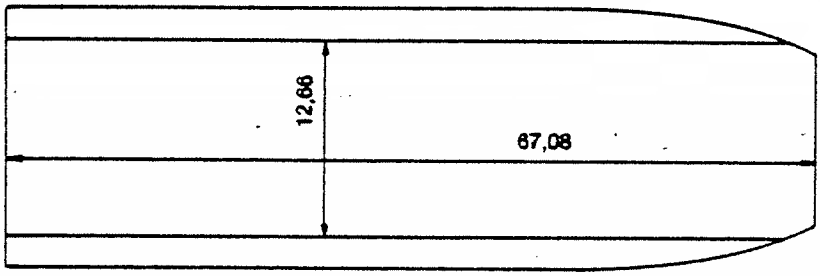
Andrea Smits

Alsyta Smits

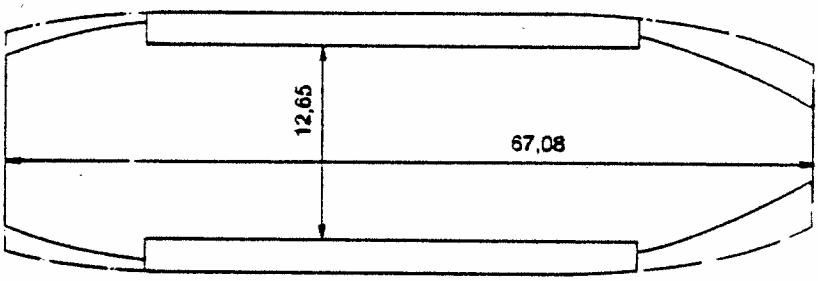
Crane capacity (Gantry cranes)
2 cranes each 40 ton
combined 75 ton



Upper hold

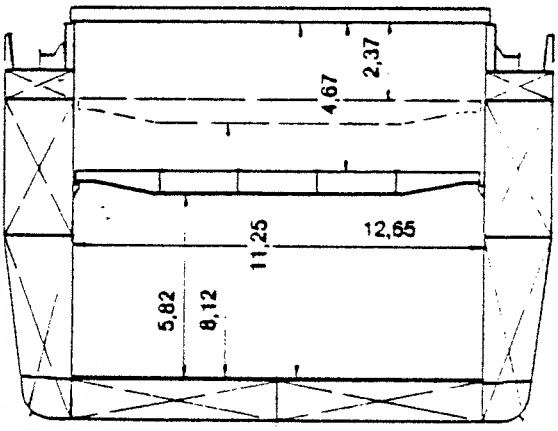


Lower hold



Hatch on upperdeck 59,98 x 12,66

Main section



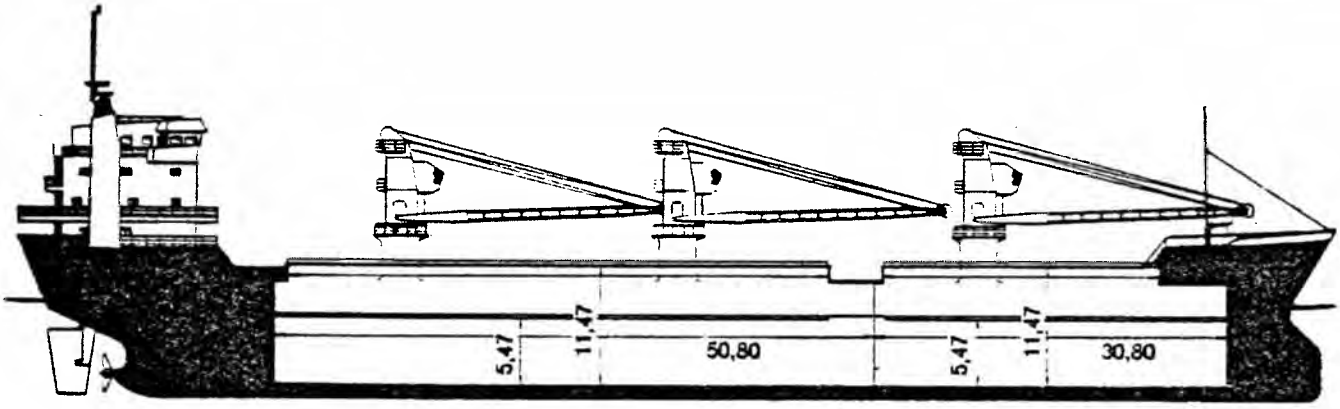
7840 dwat/320.000 cft bale.

Anita Smits

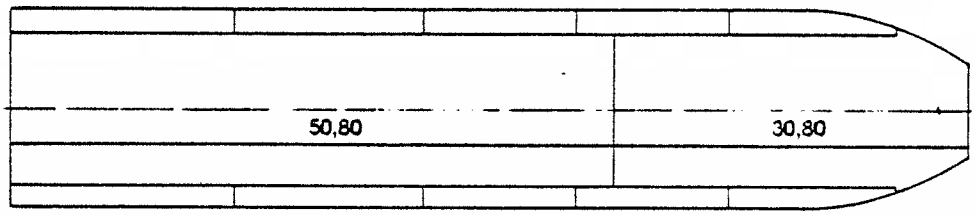
Amanda Smits

Angela Smits

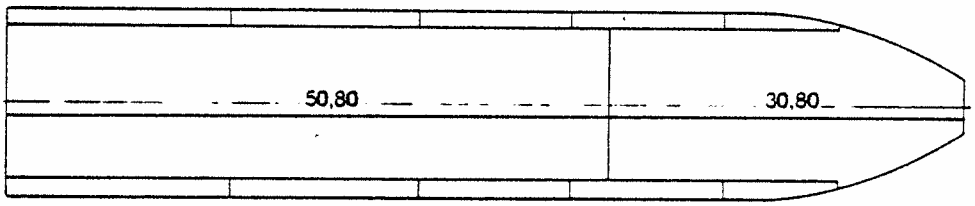
Crane capacity (Gantry crane)
2 cranes each 40 ton
combined 75 ton



Upper hold

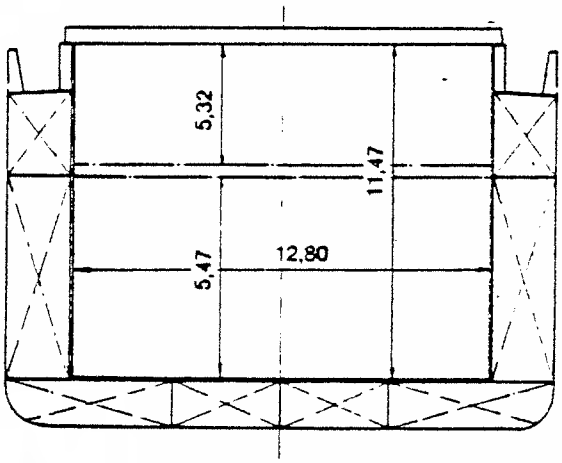


Lower hold



Hatches on upperdeck: 45,60 × 12,80
22,80 × 12,80

Main section



9200 dwat/425.000 cft bale.

Carola Smits
Catalina Smits
Carina Smits

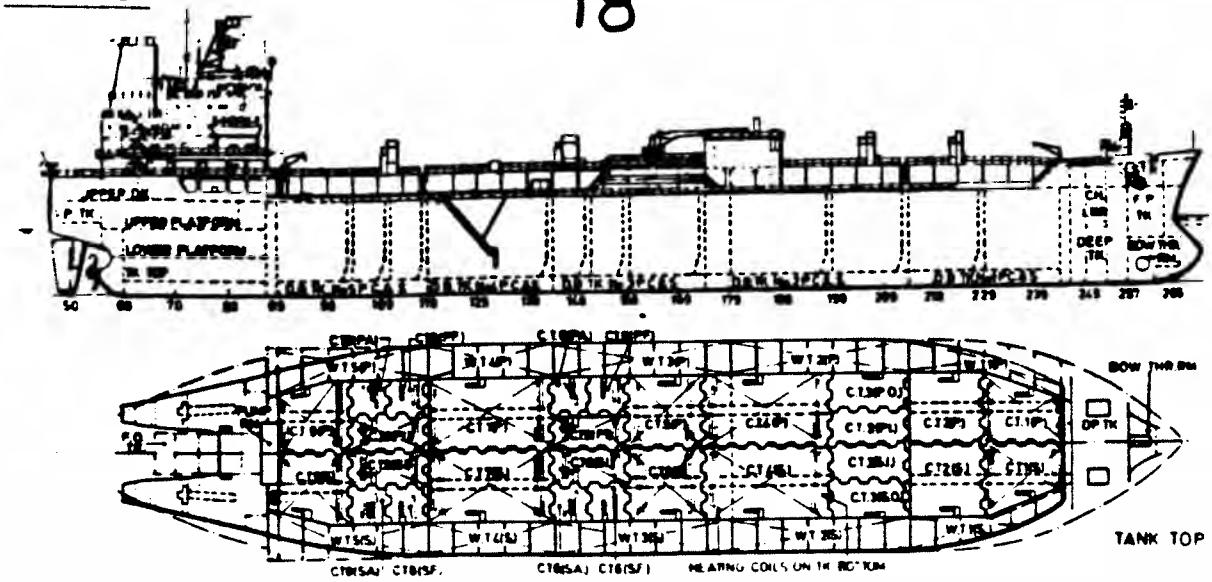
Crane capacity (fixed cranes)
3 cranes each 32 ton
combined 64 ton

It is obvious that apart from the cargoes shown on the previous pages, the nearby boxtype holds of all "SMITS" vessels are ideal for commodities such as: woodpulp in units, palletised goods, but also bulky goods like ores, clays, cereals, etc.

18.0

PRODUCT- CHEMICAL- TANKERS

PROD.-CHEM TANKER



PROFILE, TANK TOP, AND SECTIONS OF THE 38 000 TONNE D.W CHEMICAL TANKERS "JOHNSON CHEMSTAR" AND "JOHNSON CHEMSUN"

chemical tankers

$L_{OA} = 175$ — M

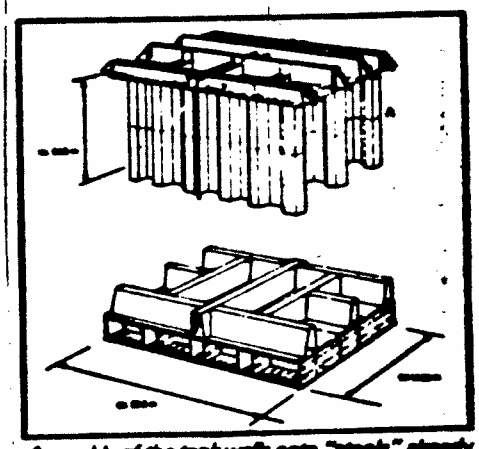
$L_{pp} = 168$ —

$B = 32$ —

$H = 14.25$

$T_{SC} = 10.70$ — DW = 38000 T

CENTRE TKS	100%	= 30160 m³	
SIDE "	100%	= 14220	INCL 2+4
SIDE "	100%	= 8640	EXCL 2+4



Assembly of the tank walls onto "stools" already in position on the tank floor. Dotted line, "A", shows the position of manual welding of the corrugated tank walls, performed during pre-fabrication with the walls horizontal.

Examples of the above types are shown in Fig. 1, which, for comparison, also illustrates the type used before MARPOL was established.

These types have been compared in every respect: safety, convenience of operation as well as economy, as shown in Table 1. As a result, of these comparison, Hitachi Zosen selected the conventional type for all sizes of tankers and double-sided hull type for small tankers.

A. Pre-MARPOL ships

B. Conventional type

C. Double-sided hull type

D. Double-bottom type

E. Partial double-bottom type

F. Double-hull type

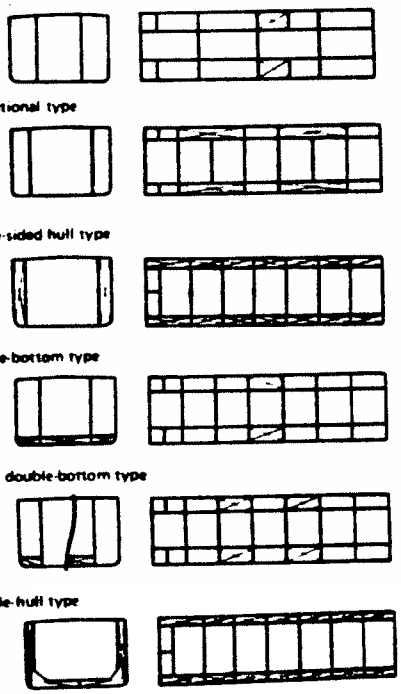
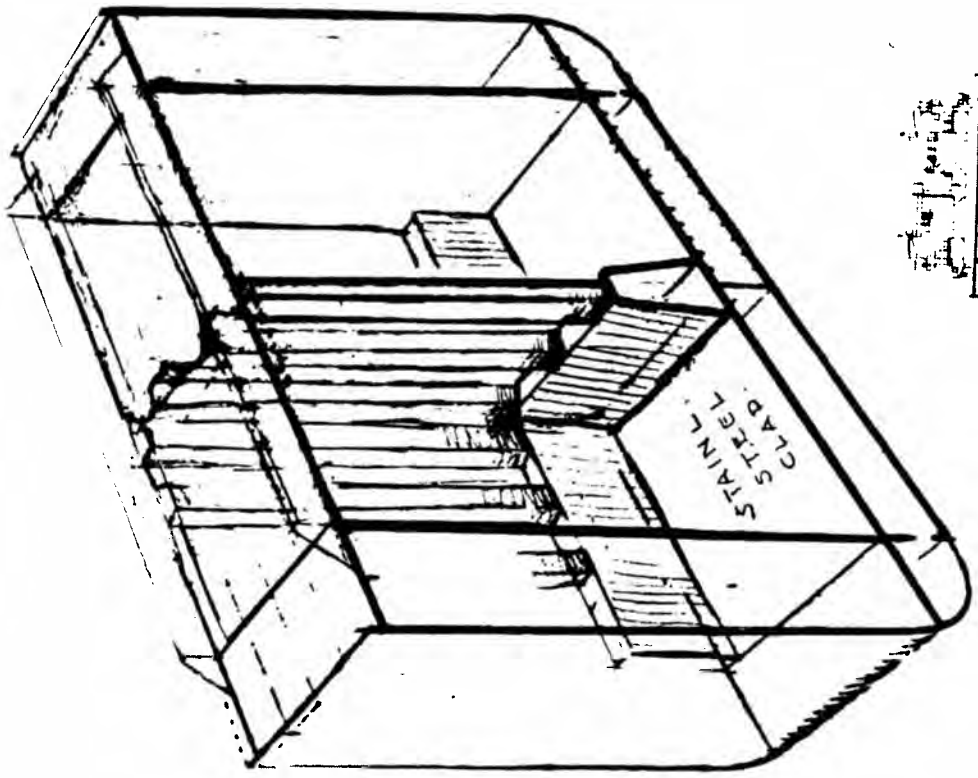


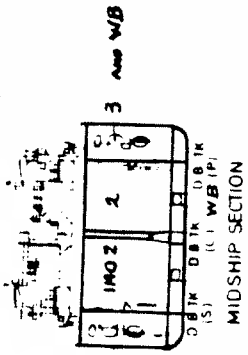
Fig. 1 Tank Arrangements

Table 1 Comparison of Tank Arrangements Meeting SBT-PL Specifications

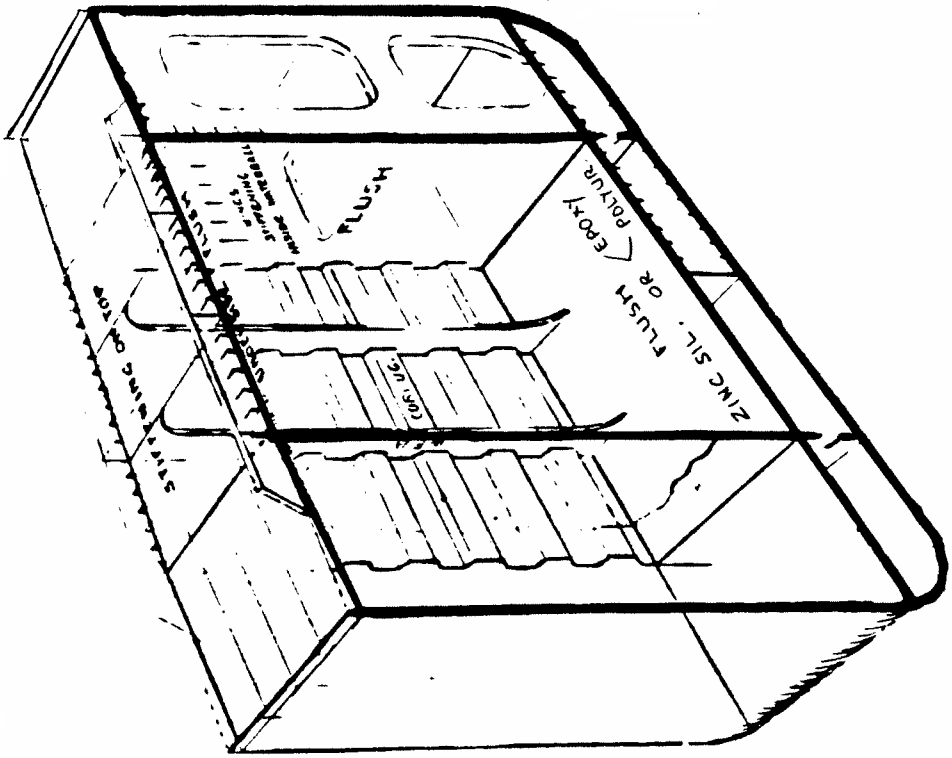
Item \ Type	Conventional Type	Double-Sided Hull Type	Double-Bottom Type	Partial Double-Bottom Type	Double-Hull Type
1. Protection against oil outflow in the event of;					
Grounding	B	B	A	B	A
Collision	B	A	B	B	A
Leakage through clean ballast line in COT	C	A	A	C	A
2. Easiness of cargo handling	C	B (Because of few cargo oil tanks)	A (Because of easy stripping)	B (Same as left)	AA (Because of easy stripping and few cargo oil tanks)
3. Easiness of COW	C	B	A	B	AA
4. Heat loss of tank heating system	B	A	B	B	A
5. Flexibility of cargo segregation	B	A	A	B	A
6. Ship price					
Hull weight	1	2	4	3	5
Coating of WBT	1	2	4	3	5
Whole evaluation	1	2	4	3	5

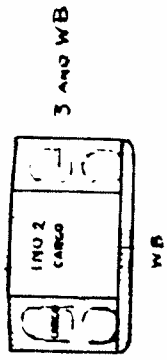
Remarks) 1. Grade AA, A, B, C in the table stand for the following:
C: Same as Pre-MARPOL vessels, B: Better than Pre-MARPOL vessels,
A: Considerably better than Pre-MARPOL vessels, AA: Excellent
2. Numbers in the table give order of economy, larger numbers being given to more expensive ships.
3. The comparison is made on tankers around 100,000 DWT.

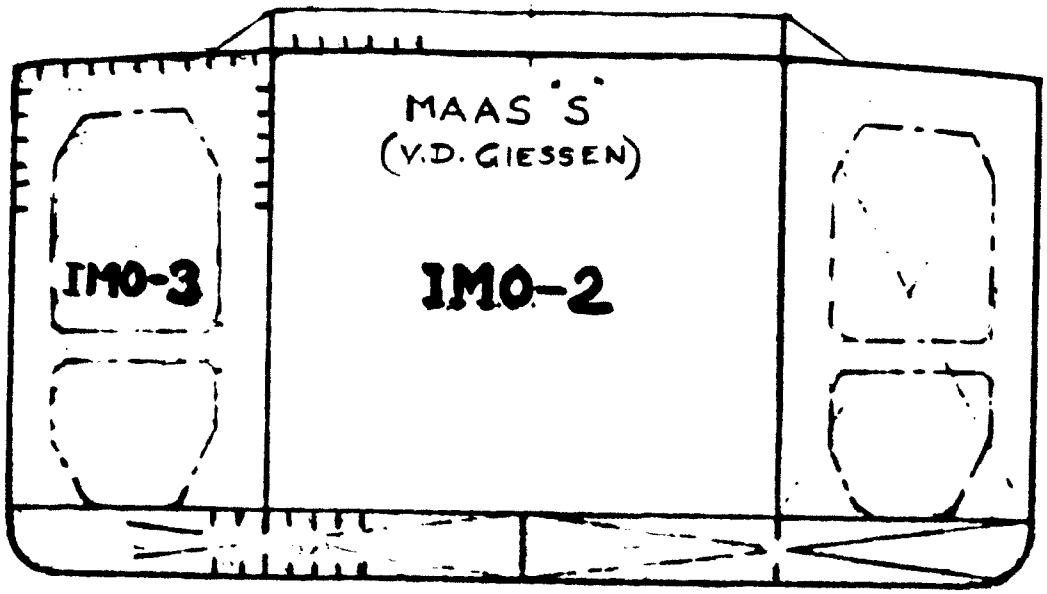




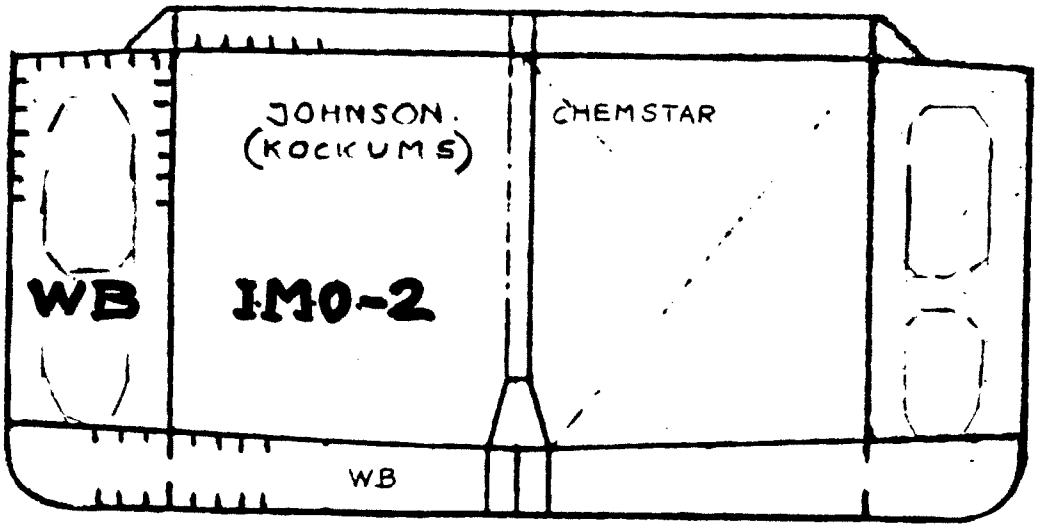
PRINCIPAL PARTICULARS
JOHNSON CHEMSTAR and
JOHNSON CHEMSUN

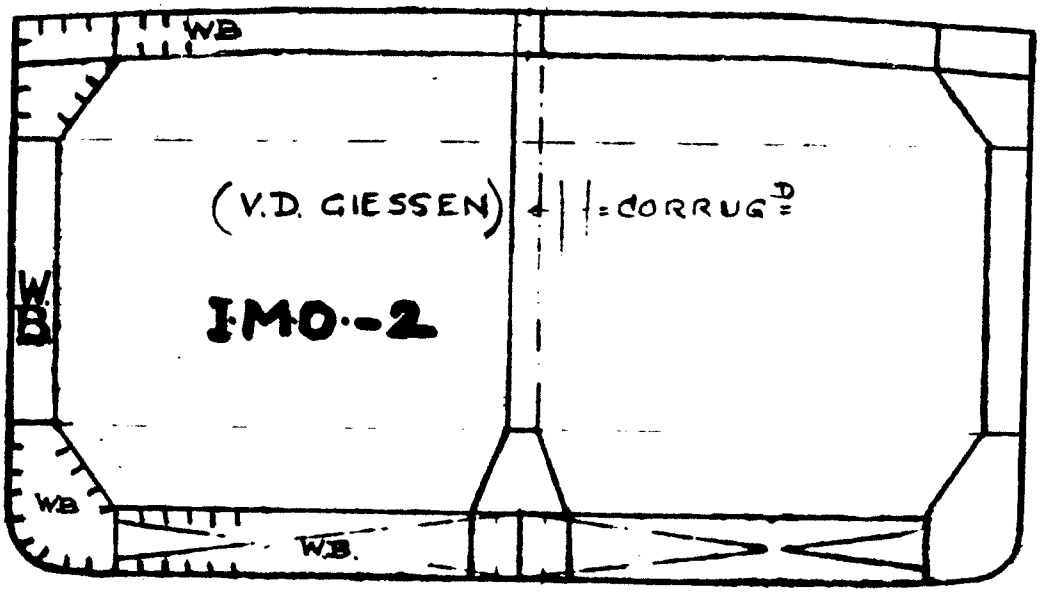


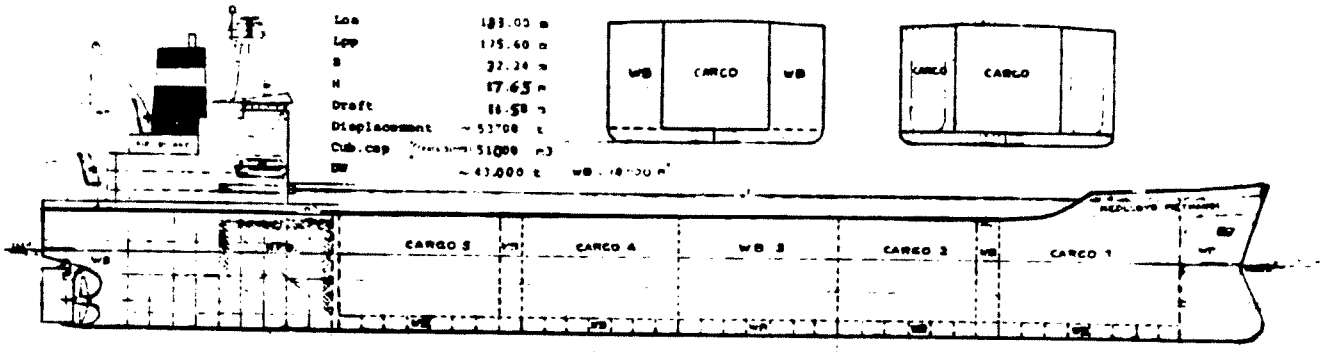


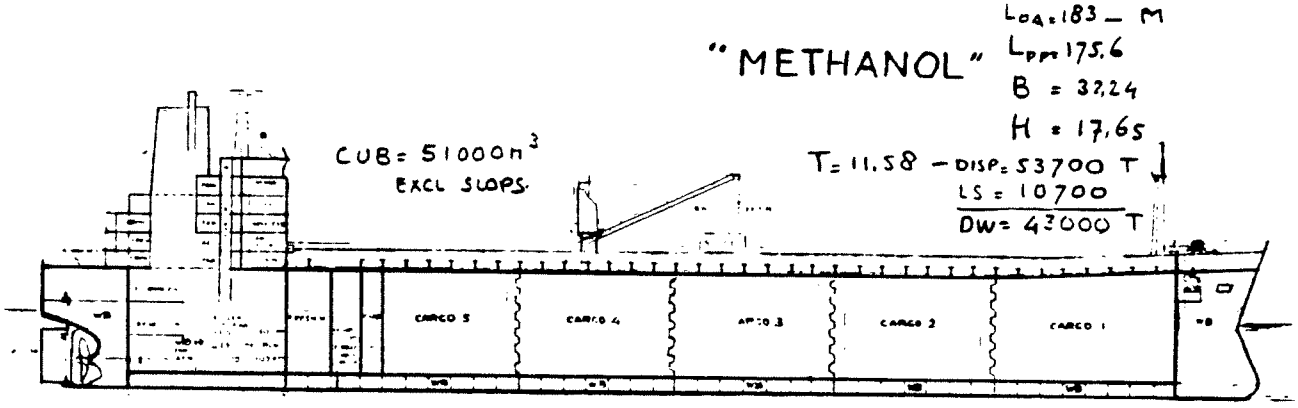


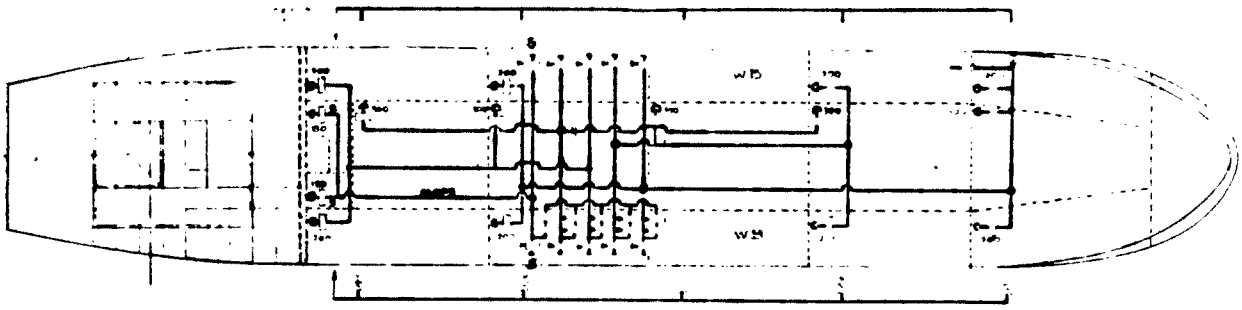
PRODUCT - CHEMICAL TANKER



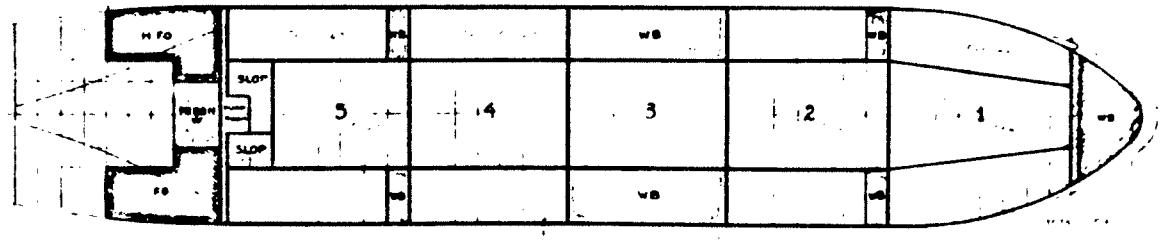


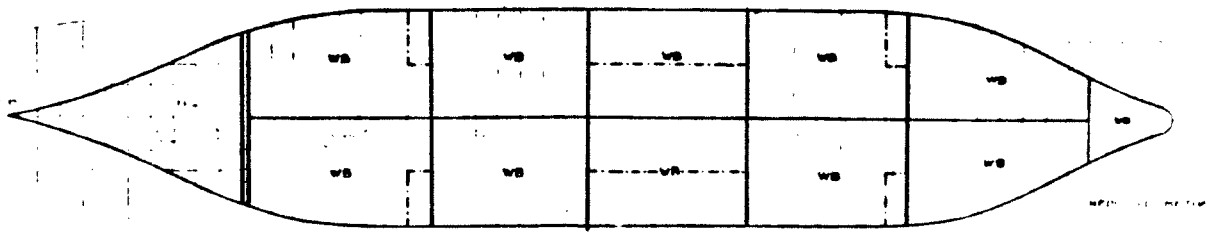




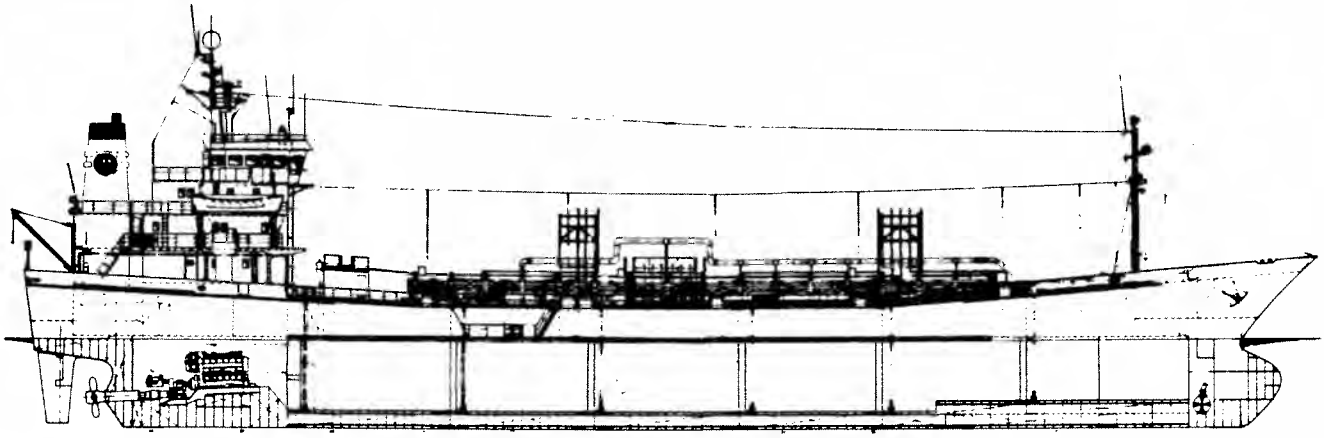


4 SEGREGATIONS





M/T. "CREST ISLAND"



Builders: Büsumer-Werft GmbH, Büsum
Owner: P/F Global Chemical Tankers, Ltd., Thorshavn
Yard number: 1444
Type: chemical tanker
Delivery: March 17th, 1983

Tonnage	1599.76 GRT
	1188.94 NRT
Deadweight	3900 t
Length o. a.	91.70 m
Length b. p.	83.80 m
Moulded breadth	13.60 m
Depth to maindeck	8.64 m
Draught	6.23 m

Speed: 12.5 kn

Classification:
G. L. + 100 A 4 M E Chemikalientankschiff Typ 2 (Mitteltanks), Typ 3 (Seitentanks) Öltankschiff "Korr", Inert, + MC E AUT

Propelling machinery:
4-stroke MaK-Diesel engine
type: 6 M 453 AK turbocharged
1800 kW (2445 HP) at 600/144 rpm
1 coupling EZS 235-1 (Vulkan)
reduction gear (Ulstein)
controllable pitch propeller (Ulstein)
Fuel consumption:
8.77 t heavy oil IF 180 per day
1 shaft driven generator 300 kVA, 450 V, 60 Hz
1 coupling EZS 95 (Vulkan)
Auxiliary engines:
1 Diesel generator set
engine Deutz BA6M 816 LLK "U" — 274 kW at 1800 rpm
generator 300 kVA, 450 V, 60 Hz
1 Diesel generator/hydraulic pump set
engine Deutz BA6M 816 LLK — generator 245 kVA, 450 V, 60 Hz
hydraulic pump 485 1/min.
1 Diesel hydraulic pump set
engine Deutz BA6M 816 LKK — 214 kW at 1800 rpm
hydraulic pump 485 1/min.
spark arrestor type AF and soot separator for flue gas boiler (Huss)
compressors (Hatlapa)
4 separators OSA 7 (Westfalia)
bow thruster type 50 F — 294 kW/400 HP (Jastram)
2 radars, visual direction finder, wireless station and 2 VHF equipments (Hagenuk)
gyro compass and automatic pilot (Anschütz)
satellite navigator
engine room plant according to the Dirivent-system (Noske-Kaeser GmbH/ Svenska Fläktfabriken)
1 hose handling derrick 990 kg
2 hydraulic automatic mooring winches of 30 kN (Hatlapa)
2 hydraulic comb. windlasses/automatic mooring winches 42 mm K2 chain, 30 kN (Hatlapa)
3 hydraulic deepwell pumps 100 m^3/h each
11 hydraulic deepwell pumps 80 m^3/h each
1 hydraulic booster pump 175 m^3/h
Refrigeration plant:
2 compressors 1850 kcal/h and 1 compressor 63 000 kcal/h (Sabroe)

8 stainless steel centre tanks for IMO type 2 products	2183.03 m^3
12 coated wing tanks for IMO type 3 products	1622.61 m^3
Total capacity of cargo tanks (98 % full)	3805.64 m^3

B O W R A M P - LANDING CRAFT

FOR INTERISLAND TRAFFIC

AND DEVELOPMENT.

19.0

LANDING CRAFT

'Ibn Ouf'—beaching to discharge tanks

The very first vessel to be considered a Ro-Ro ship in the modern sense was the US Navy's *Comet*, a revolutionary ship conceived 25 years ago and developed hand in hand with MacGregor experts to enter service in January 1958. If there were forerunners of this Ro-Ro vessel, they were the landing craft of the Second World War which showed what advantages there were to the military of vessels enabling tanks and other weapons of war to roll on and off.

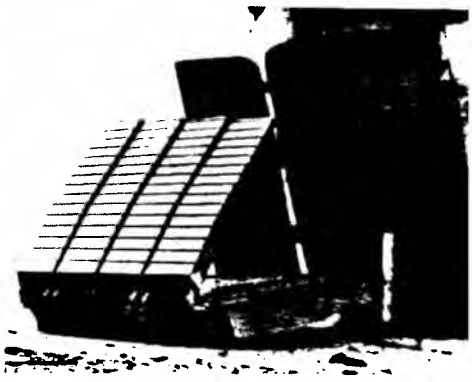
MacGregor is still in tune with the military world and one recent example is *Ibn Ouf*, the first of two similar vessels ordered from Constructions Navales et Industrielles de la Méditerranée by the ibyan Navy.

Ibn Ouf, is a tank-landing craft that is capable of 'ertaking ocean voyages. With a length overall of 99.5m and a loaded displacement of 2,800 tons she is quite a sizeable craft. Allowance is made for a considerable self-defence capability with guns and space for a helicopter landing pad.

MacGregor has provided *Ibn Ouf* with the ability to discharge her cargo either onto a Ro-Ro berth or onto a beach. Bearing a load of up to 570 tonnes, and given a slope of some 2deg she is able to run her bow aground in some 50cm of water. Bow doors open to allow a three-section 17m long ramp to unfold onto the sand. Vehicles can then be driven either from below deck stowage positions or from the weatherdeck via a movable MacGregor ramp.

Ibn Ouf has a maximum service speed of 15.2 knots while her range at 14 knots is 4,000 miles. The necessary power is provided by two Pielstick diesels each driving a c.p. propeller.

Ibn Ouf *is able to beach for discharge*



RORO LANDING CRAFT - FOR INTER-ISLAND SERVICE.

(July 1980) FAIRPLAY

Now on order in Singapore for delivery next year is a unique type of general purpose freighter that could well be the prototype of the standard ship that is needed in huge numbers to modernise the decrepit fleet of inter-island traders operating around the archipelago of South East Asia. Designed by Burness Corlett & Partners Australia, which has a headquarters in Sydney, this 80m vessel has been ordered from the yard of Selco in Singapore by the Australian operator V. B. Perkins for operation between Singapore, other Asean ports and the Northern Territory capital of Darwin. The craft will also provide a much needed service to the mining companies around the remote Northern coast of Australia, where there is an increasing amount of mineral activity, much requiring the shipment of heavy or awkward plant, to sites where there are no port facilities.

The Australian company, which has operated a variety of ex-Naval landing craft tonnage, first was established to provide support to the various agencies assisting the aboriginal tribes in the North West coast of the country, but the considerable investment in mineral exploitation, and the development of the North-West shelf oilfields have led to a great demand for specialised tonnage. The specification has called for a ship of unusual flexibility, able to operate off remote beaches, take the ground, operate with ro-ro operations and carry large quantities of cattle, including the large Brahmin cattle that thrive in the hot climate. There is also a need for the ship to carry large quantities of fuel in bulk form, high grade ores, and general cargo, while trailers and containers are offering.

Of about 2,000 d.w.t., the craft that has been developed to fulfil this demanding specification provides for a bow loading arrangement, with the lengthy ramp being capable of landing either on wharf or beach. Cargo drives straight up the ramp, which will be of Navire manufacture, onto the main deck which is open to the sky for half its length. Overheight cargo can be carried in this area. The after end of the main deck is fully enclosed, with the watertightness being assured by a wide door across the forepart. This enclosed space is divided into two by an athwartships bulkhead. Two telescopic cranes are to be fitted on top of the deck erection for cargo handling when the ship is lying alongside, or for working into craft. Thus, with the exception of the central exhaust casing, and a small space right aft where steering gear and stores are situated, the whole of the main deck, for its full height, is available for the stowage of cargo. Other cargo gear includes the provision of portable racks for the stowage of up to 600 large cattle. Cargo fuel oil tanks are situated beneath the main deck port and starboard and on the centreline. Two small freezer lockers are carried port and starboard in the forecastle, to cope with the quantities of refrigerated stores that require transport.

Main engines specified for the V. B. Perkins vessel will be two Deutz SBV6 628 engines driving the twin screws, and each providing 1500 h.p. at 1,000 rev/min. Three Deutz generators, each producing 253 kW are also to be supplied. A comprehensive alarm and control system will be provided to monitor the machinery and tanks, the vessel being outfitted for UMS running. She is to be delivered in March 1981.

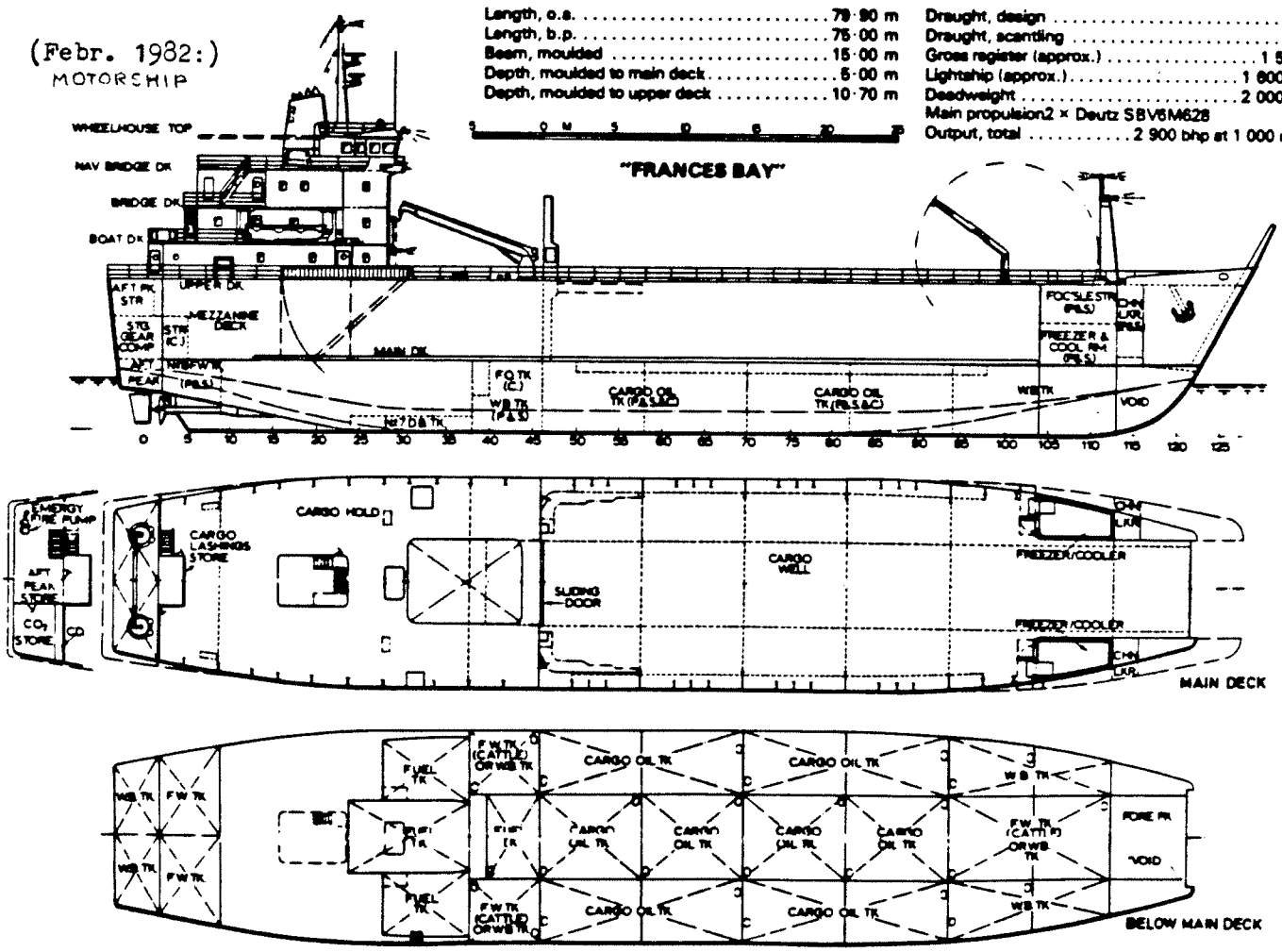
A four decked accommodation will be provided on the ship as designed, a large crew being provided for under Australian manning arrangements. It is however clear that this arrangement would be easily altered to provide limited passenger accommodation, or a much smaller accommodation block.

The design is thought to have much merit for fulfilling the need for an economical and efficient vessel that capital starved owners in the Philippines or Indonesia could use to replace their elderly tonnage, a design that would improve port handling times in largely inadequate facilities. It has been calculated that there is a need for more than one hundred vessels of such type over the next ten years, but finance, in the form of aid packages remains the key.

(Febr. 1982:)
MOTORSHIP

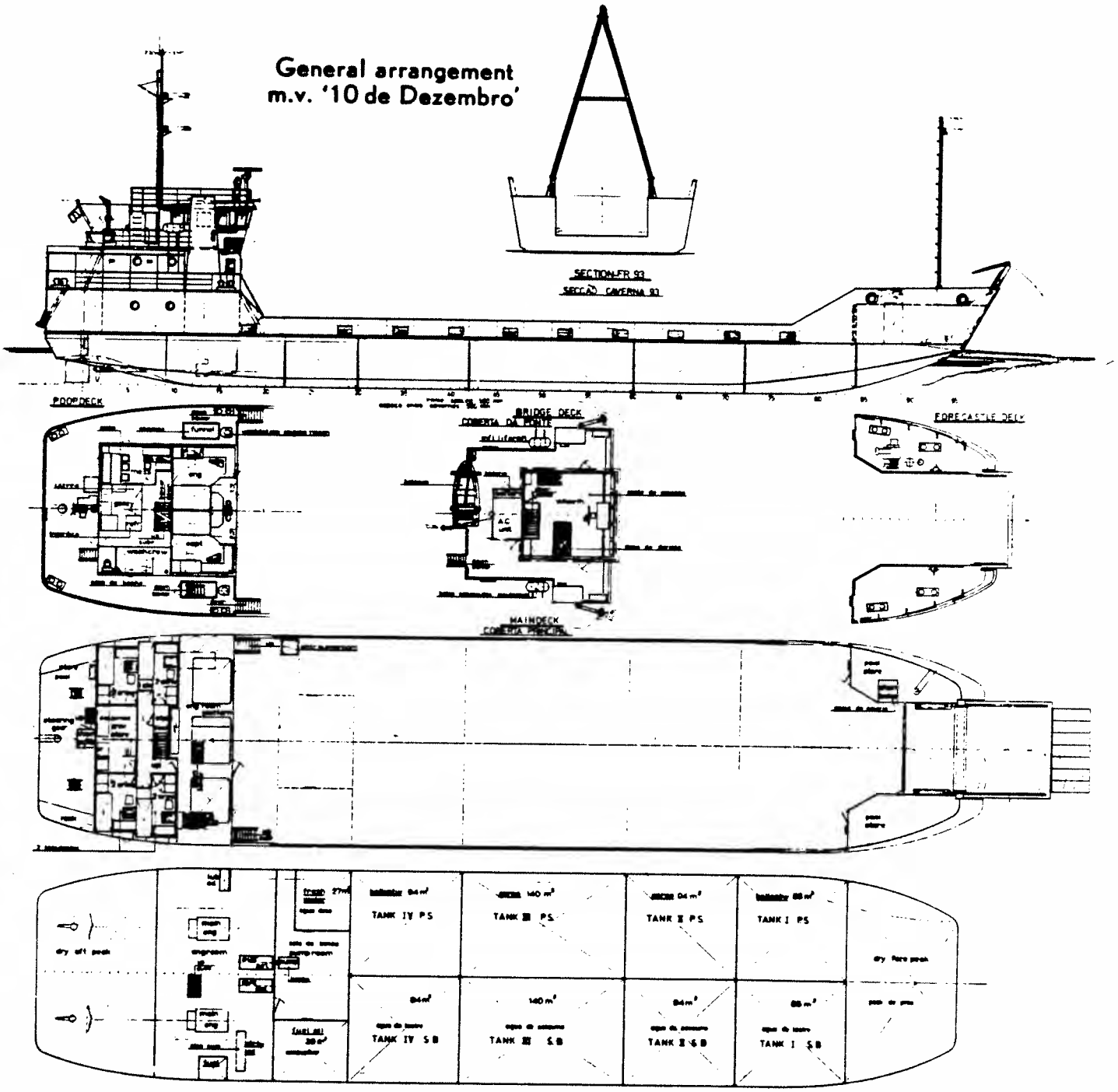
Length, o.a.	79·90 m	Draught, design	3·65 m
Length, b.p.	75·00 m	Draught, scantling	4·00 m
Beam, moulded	15·00 m	Gross register (approx.)	1 595 tons
Depth, moulded to main deck	5·00 m	Lightship (approx.)	1 600 tonnes
Depth, moulded to upper deck	10·70 m	Deadweight	2 000 tonnes
		Main propulsion 2 × Deutz SBV6M628	
		Output, total	2 900 bhp at 1 000 rev/min

"FRANCES BAY"



M.V. '10 de DEZEMBRO'
M.V. '11 de NOVEMBRO'

two 53 m trailer carriers built by Shipyard Ton Bodewes B.V.



twin-screw/twin rudder (hand/hydr.electr.)
2x Caterpillar 3408PCjwac, 4-cycle, 8-cyl.
cont.serv.rtg each eng. 365 hp at 1,800 rpm
red.gear ratio 4½:1. Prop 3-bl, ø 1.35m.
lightweight ..... 400 t.

length o.a.	53.00 m
length b.p.	46.65 m
breadth mld.	11.50 m
depth	2.80 m
draught	2.00 m
deadweight	575 t
gross tonnage	500 grt
net tonnage	410 nrt
displacement light	370 t
cargo deck	33.70 x 11.25 m
ramp	4.00 x 5.00 m

deck load	7 t/m²
axle load	10 t
freshwater	440 m³
freshwater discharge rate	100 m³/hr
ballast water	410 m³
fuel oil	35 m³
lub oil	2.5 m³
speed	10 knots
crew	10

RoRo Landing Craft - For inter-island Services

The *Martaban 3* is one of four sister vessels which were built simultaneously by Selco Shipyard Pte Ltd at Singapore for Myanma Oil Corporation of Burma. The twin-screw landing craft was designed by Kenton Marine Pte Ltd to meet the Owner's specified requirements of carrying 230 t, of which 200 t are to be deck or liquid cargo, on a draught of 1.22 m and at a speed of 10 knots when the main engines are developing not more than 90 per cent of their continuous rated power. All performances and stability standards had to be guaranteed by the designers and shipbuilder.

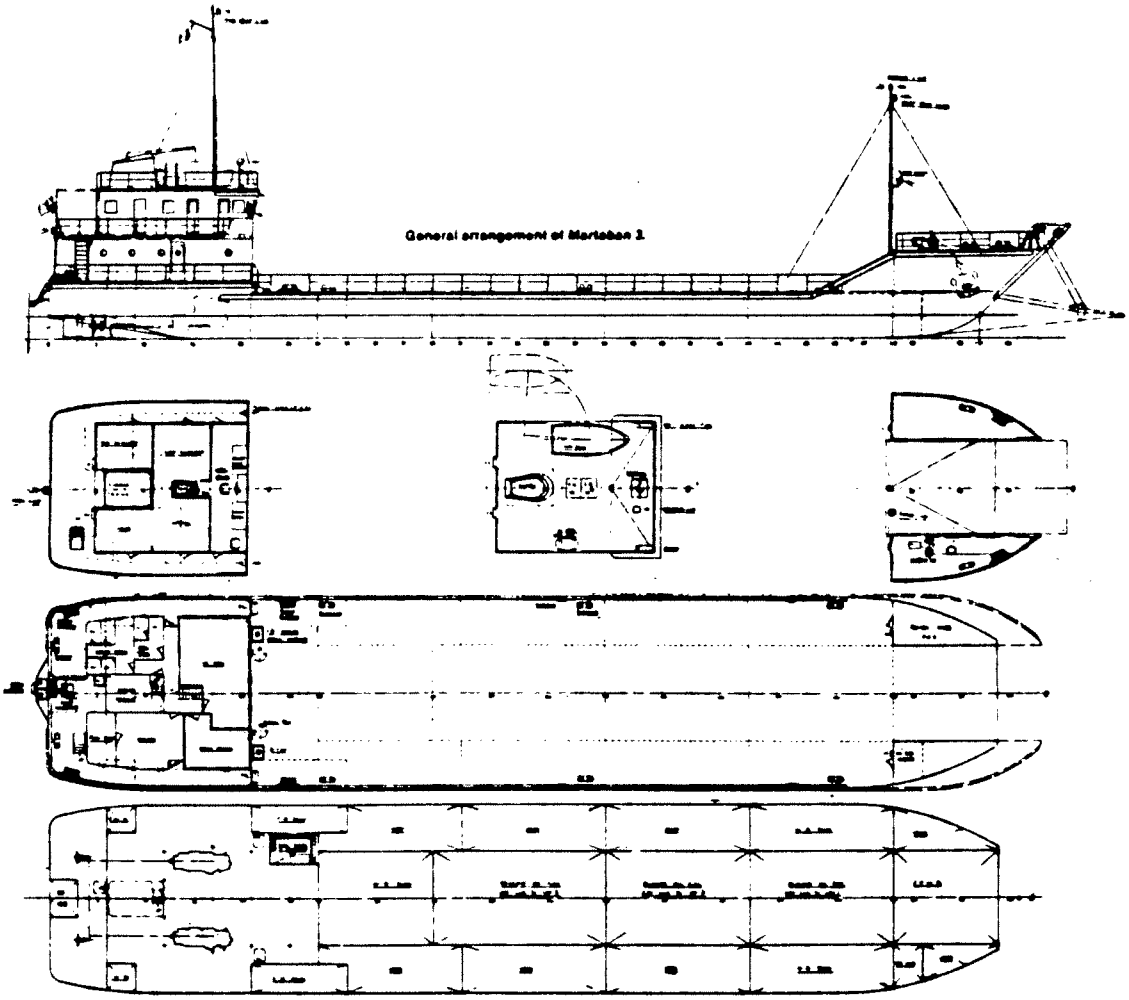
Intended mainly for the transportation of oil well drilling equipment, heavy trucks and tracked vehicles, the vessel has a clear deck area of 335 m² and is specially strengthened for vehicles of 60 t gross weight and maximum point loadings of 6.5 t on two tyres spaced at least 1,900 mm apart or 13.0 t on four tyres, at least 1,000 mm apart; also for tracked vehicles with loading per track of 20 t, or uniformly distributed deck load of 10t/m².

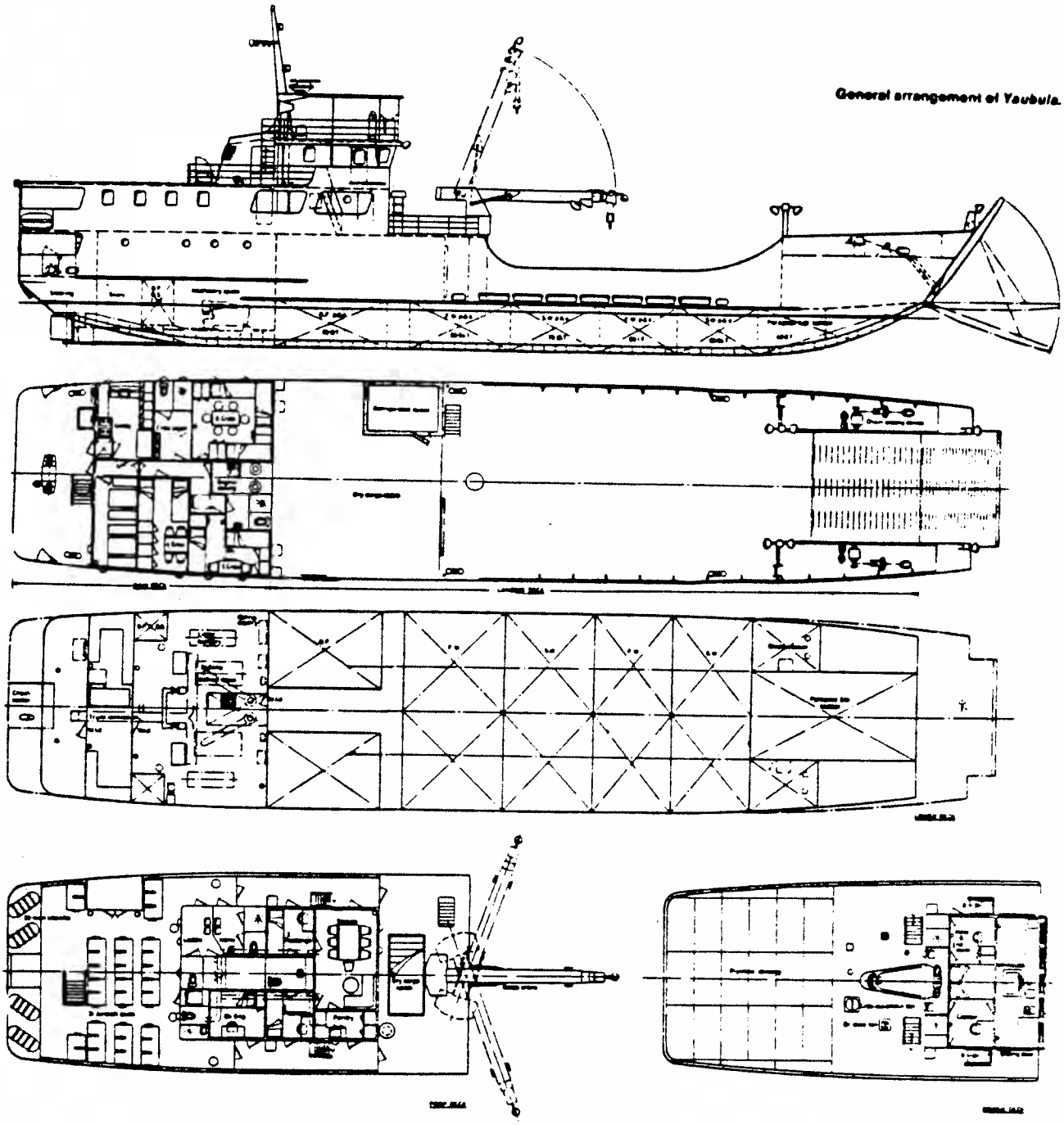
The 5 m wide double hinging bow ramp facilites operations in the estuaries and swift shallow rivers of Burma which generally have no prepared landing areas. Accordingly, the stern is tunnelled and the bottom forward is strengthened for beach landings.

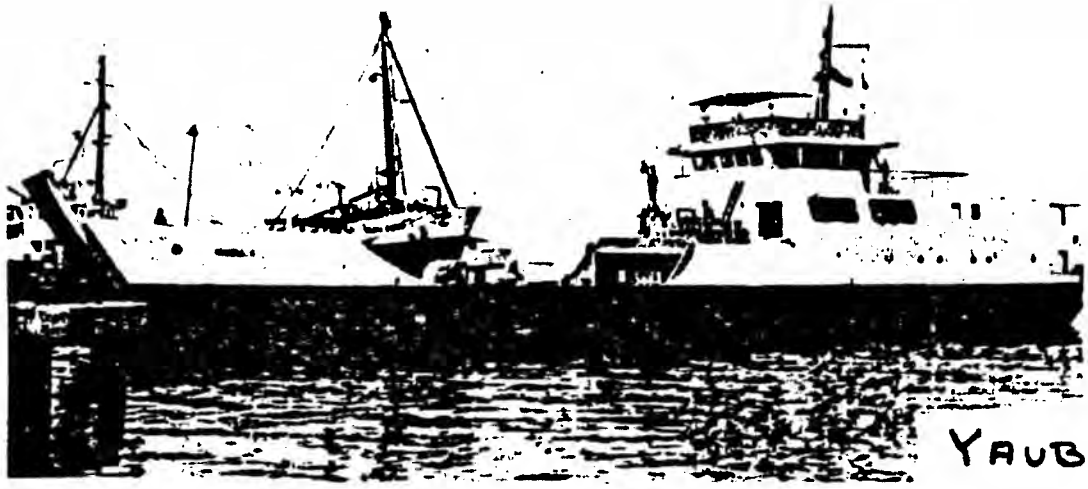
Framing is on the 'combination' system with scantlings selected to be the lightest possible. This is reflected in the hull structural steel weight of only 155 t in spite of the two longitudinal full height internal bulkheads which were considered essential to avoid longitudinal flexing of the overall hull. The hull form is of multi-chine type which together with building on the 'block' principle adopted by the Builder, make for ease and speed of construction and better quality control. The vessel meets a 'one compartment' floodability standard specified by Owners.

Shipbuilder	**Selco Shipyard Pte Ltd, Singapore**
Owner	**Myanma Oil Corporation, Burma**
Design & Drawings	**Kenton Marine Pte Ltd, Singapore**
Classification	**Bureau Veritas I 3/3 E + NI, 0**

Length o.a.	**51.98 m**	Engines	**Caterpillar D353TA**
Breadth mld	**10.0 m**	Bhp	**2 × 425, Twin Screw**
Depth mld	**2.4 m**	Rpm	**1,225**
Draught (max.)	**1.4 m**	Gear ratio	**3.03:1**
Gross tonnage	**408.97 t**		
Net tonnage	**118.94 t**	Service speed	**10.0 knots**
Deadweight	**230 t**	Maximum speed	**10.3 knots**
F.O. capacity	**29.5 t**		
Reserve F.O. capacity	**243.9 t**	Auxiliaries	**MWM-Diter D226-4**
Portable water capacity	**8.3 t**	Bhp	**2 × 44**
		Rpm	**1,500**
Complement	**14**	Alternators	**Leroy Sommer 2 × 35 KVA**



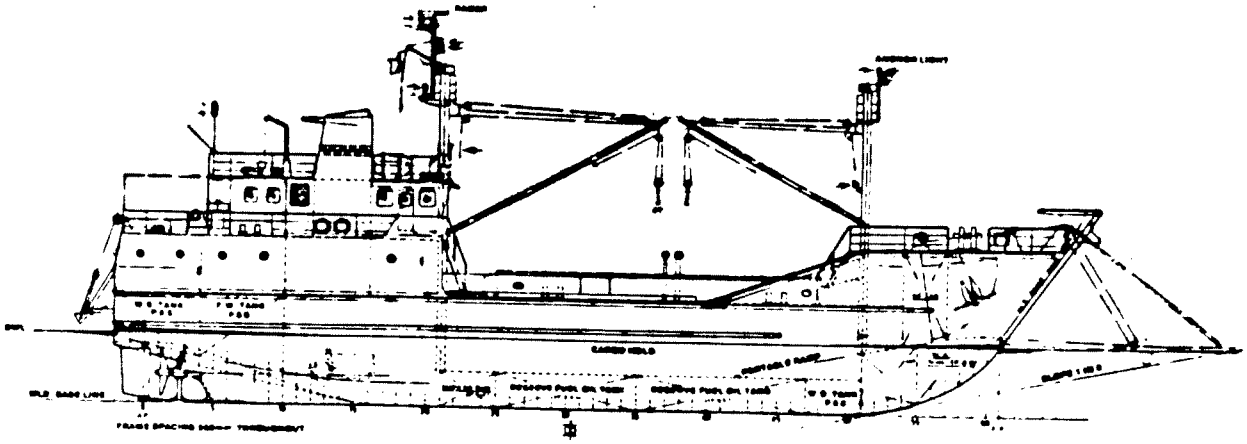




YAUBULA

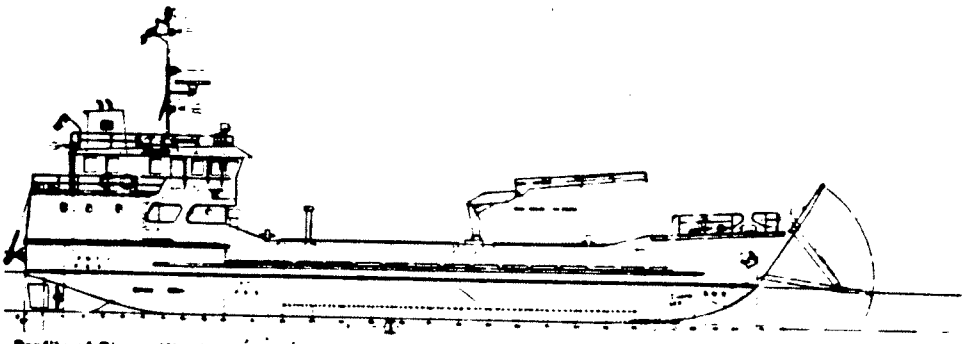
42,5 × 9,1 × 2,15

Four mini-cargo ships for Papua New Guinea



During 1981 Sing Koon Seng Pte. Ltd., at Jurong, delivered four 350 tonne d.w. mini-cargo ships under a Sing.$12 million contract from Burns Philp, of Papua New Guinea, as part of a coastal fleet replacement programme. These 37·25 m long twin-screw vessels are specially designed by Kenton Marine Pte. Ltd. to carry mixed general cargo such as bulk groceries, palletised goods, and engineering products, also trucks up to 6 tonnes, all in one hold and on deck. Special tanks are also available for cargo fuel. A Navire bow ramp and door are fitted for vehicle loading, and 60 deck passengers can be sheltered under a portable awning, together with 18 crew. Computer techniques were used to produce an optimum hull form for a full-load 9-knot service speed. Two 4-tonne cargo derricks are worked by Norwinch hydraulic winches.

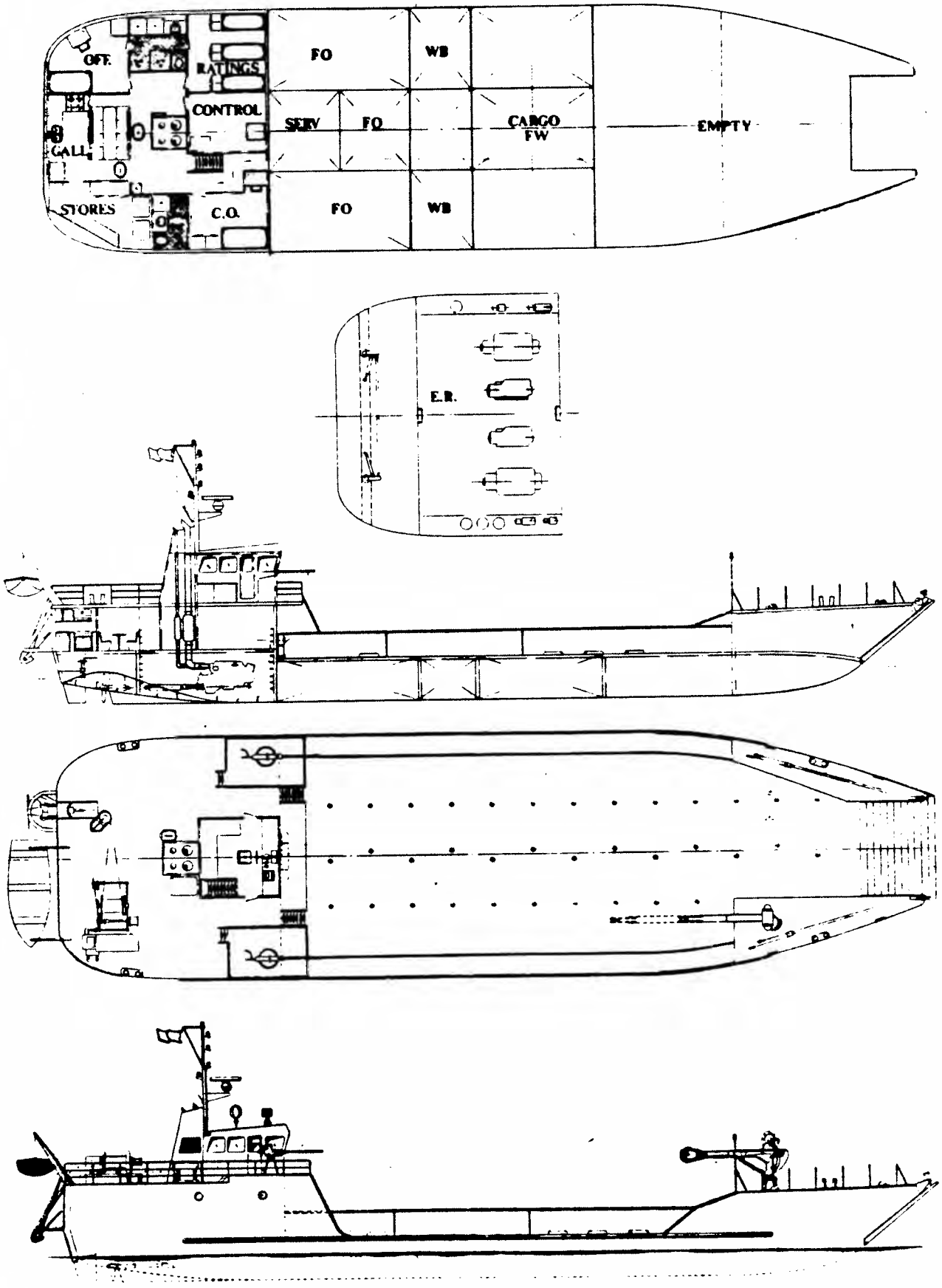
Twin Deutz SBA6M816 engines, each of 313 bhp, drive fixed-pitch propellers through Reintjes BGA3008 reverse/reduction (4·45:1 ratio) gearboxes. Deutz also supplied the two F5L413FR air-cooled engines for 52 kW service alternators and an F3L912 engine for a 2 kW harbour set. A Furuno radar and echo sounder are also fitted in the wheelhouse.



Profile of Conan Wu designed cargo ship for PNG.

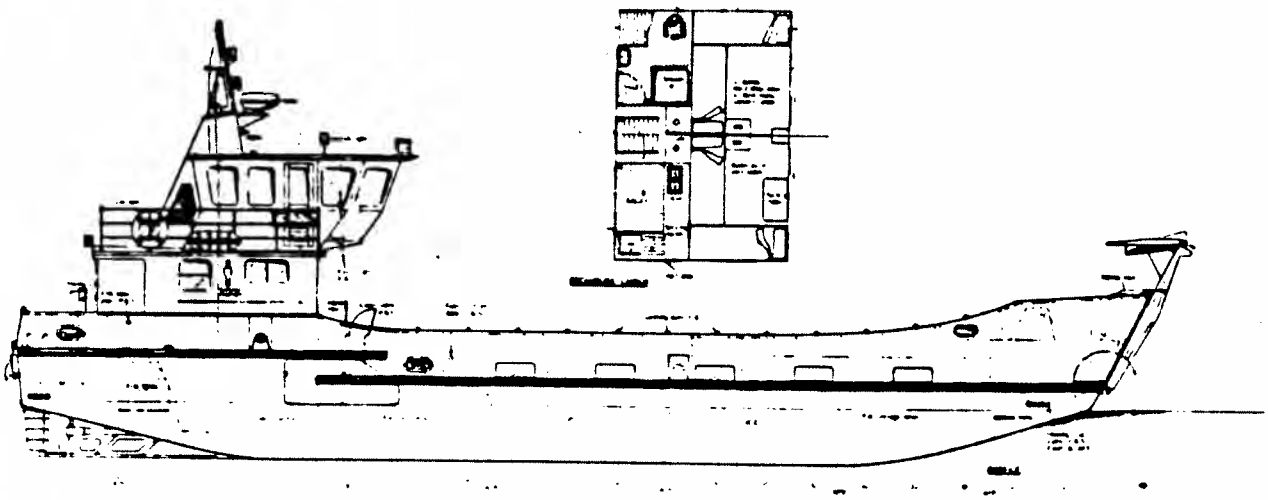
MOTUAN CHIEF

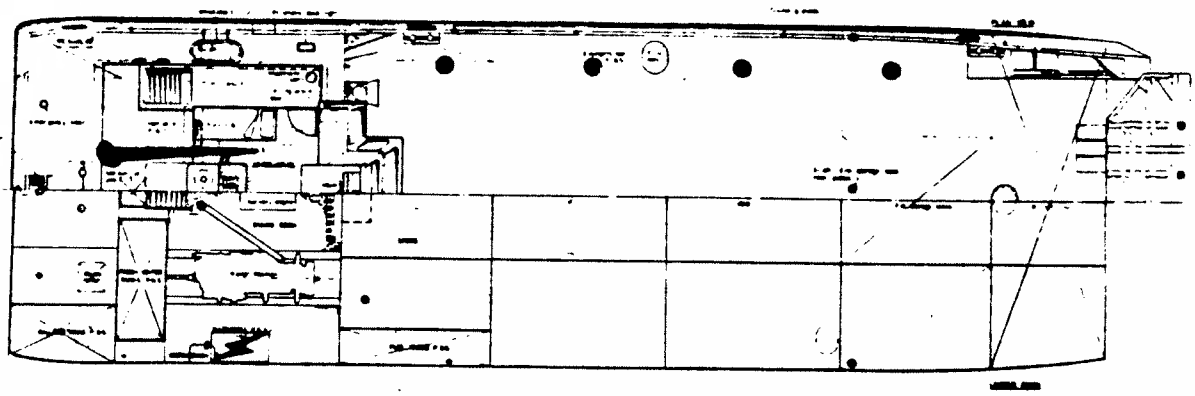
37.25 x x

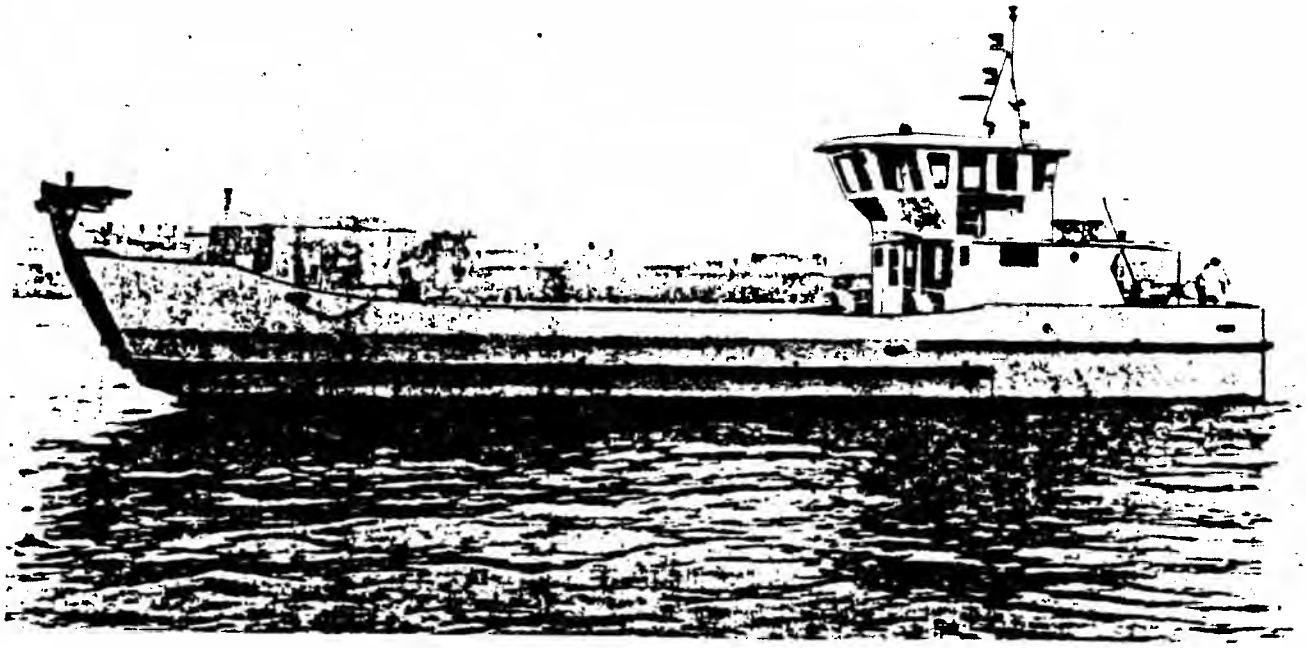


General arrangement of Swiftships' landing craft

SWIFTSHIP
39.63 x 11 x 2.9







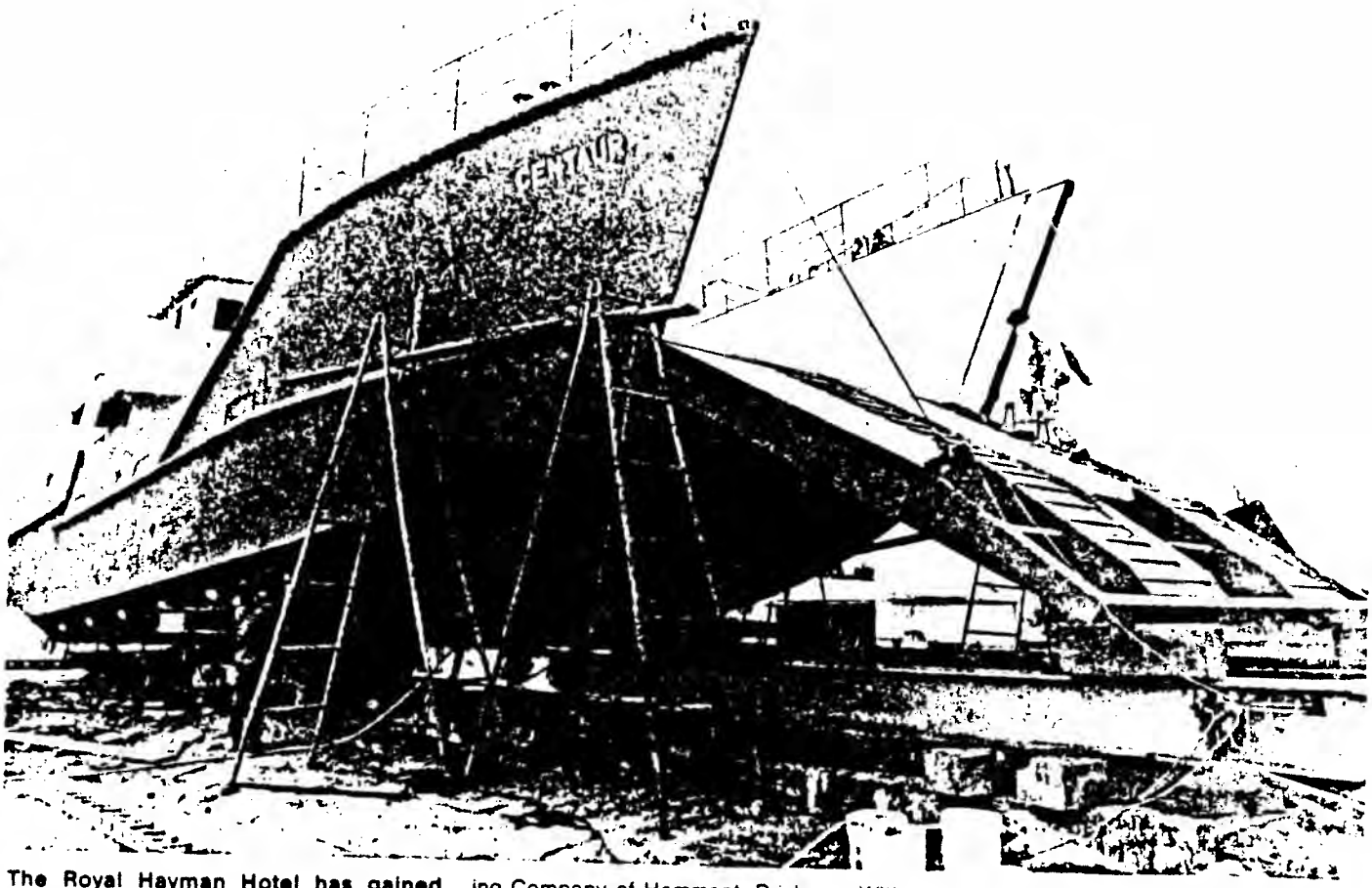
IMPALA MARINE

24.4 x 7.3 x 1.8

When Ansett Transport Industries decided to expand their resort facilities on Hayman Island in the inner Great Barrier Reef, the time had come to augment their boat fleet with a bulk carrier.

"CENTAUR"

By Peter Tuck



On the slips "Centaur's" shape is evident.

The Royal Hayman Hotel has gained considerably from a big jump in tourist traffic to the Whitsunday Passage islands off the central Queensland coast.

Rather than lease a suitable vessel for the duration of their building program, the Australian conglomerate simply turned to it's subsidiary Bulkships Limited of Sydney and had them provide a long term solution.

As shipowners and operators in all aspects of sea transport — and a subsidiary of that other big arm, Thomas Nationwide Transport — Bulkships worked on behalf of Ansett to produce the "Centaur".

This 26 metre landing craft vessel recently joined the Ansett fleet and brought its complement to nine vessels. These include a pair of 100' passenger boats; a sailing catamaran, three game boats and a couple of motor launches. Certainly everything necessary for an island resort, but nothing capable of "Centaur's" chores which included lugging a Caterpillar D-9 bulldozer across the water for Hayman's expansion program.

Australia wide tenders were called for the construction of "Centaur" before the contract was awarded to Morton Engineering Company of Hemmant, Brisbane. With experience in the design and construction of tugs, barges and dredges, Morton's were able to offer a completed vessel from their own shipyard on the banks of the lower Brisbane River at a cost of A$500,000.

"Centaur" is a landing barge with a relatively shallow draught considering its big loading capacity of 90 tonnes and relatively thick steel plate construction in the hull of 8-10 mm. An ability to drop and pick up cargo from any beach area demands a tough skin and the considerable tidal fall in this region of the Queensland coast requires such versatility.

Twin screws ensure each forward demi-hull can back off the various landing points "Centaur" is likely to face in its field of operation — a region subject to cyclones and also demanding caution in chart reliability. Recent hydrographic work revealed an uncharted deep water channel for ships on the run to Mackay and Gladstone which offers a 500 kilometre saving in transit time to the north out of the Great Barrier Reef.

Expansion has outstripped the fresh water capacities of Hayman Island and one of the tasks of the "Centaur" will be the transport of 22,000 gallons per shipment to supplement the Island's existing water supply. As befitting a landing barge work boat, "Centaur" has limited passenger capacity in comparison with other vessels in the Ansett fleet. Six passengers in addition to a crew of two are catered for in the aft deckhouse.

Twin CAT 3306 diesels give "Centaur" a cruising speed of 8.5 knots with a 9.5 knot maximum. This makes the 12 mile crossing to the nearest mainland landing and road-head at Airlie Beach in good time with Bowen (30 miles) and Mackay (50 miles) well developed ports to the north and south.

Her survey to the Queensland Department of Harbour and Marine (Class 2B) enables island cargo duties around the Great Barrier Reef on an 8,000 litre fuel load.

In appearance and name "Centaur" looks similar to the huge fleet of landing barges that service Greece's Ionian and Aegean islands Functional utility wins out against nearly all attempts at building a beautiful sea craft.

The mounting of a Hiab 650 deck crane on

the port side further accentuates "Centaur's" working role. An extra high prow gives her a couple of extra feet to allow the bow ramp to reach dry land. Bow winches, rather than hydraulics, raise and lower the high ramp.

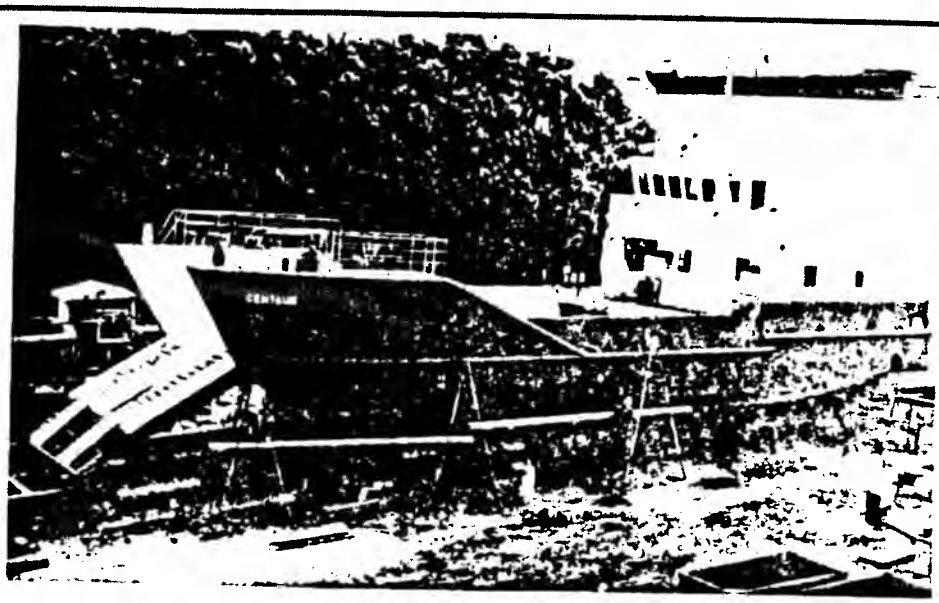
A commanding wheelhouse contains a good range of electronic aids, including T.M.Q. autopilot and Daiwa echo-sounder.

An attractive passenger deck immediately behind the bridge looks out over a square stern with a single anchor well and winch ready to secure the boat when required. Steering gear is from Palmarine and the 8.2 metre beam leaves a wide wake from big 36" x 26" props.

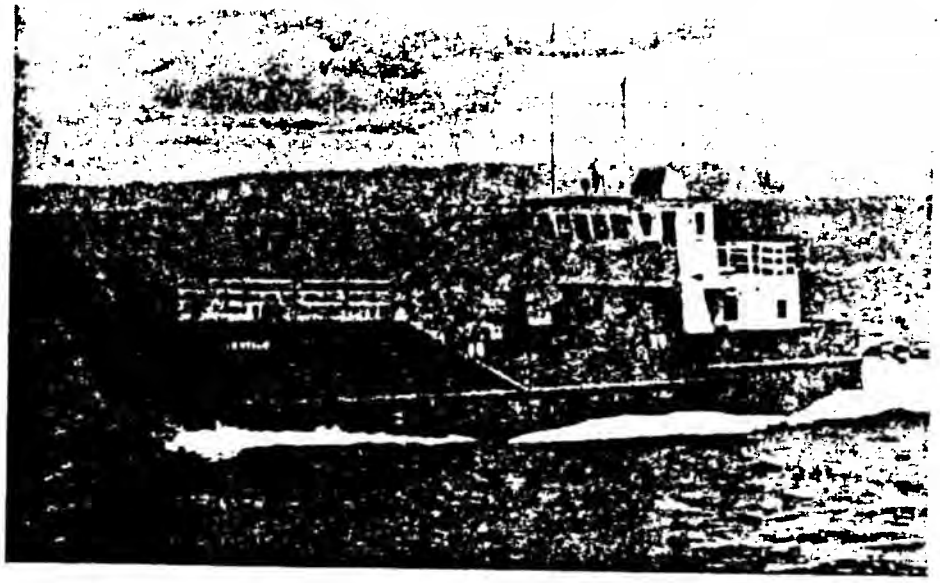
A second anchor installation on the starboard bow, some distance from the ramp, allows extra stability without risk of any snarl-ups from nearby gear. The hull lines toward the bow are sheer but become increasingly angled toward the stern, leaving little doubt where the big 100 tonne capacity fresh water tank is located.

"CENTAUR" SPECIFICATIONS:

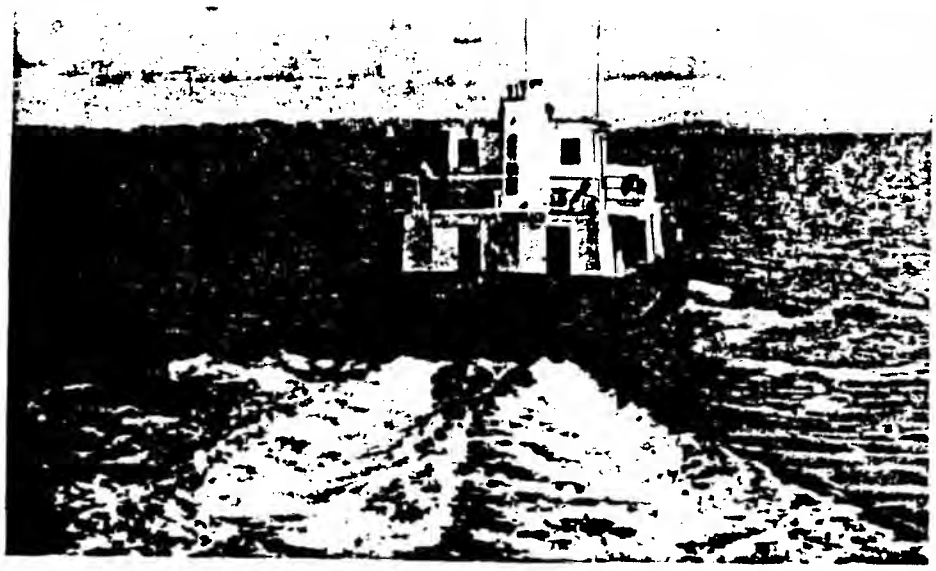
ype of vessel: Cargo barge, resh water lighter.
In survey to: Department of Harbour and Marine, Queensland. Class 2B.
Name: "Centaur".
Owner: Ansett Transport Industries Pty. Ltd.
Builder: Morton Engineering Co., Brisbane, Qld.
Designer: F.C. Borlase, Dip. N.A. Buderim, Qld.
L.O.A.: 25.85 m.
L.W.L.: 24.00 m.
Beam: 8.2 m.
Draught: 1.4 m.
Hull Construction Material: 8 mm and 10 mm steel.
Superstructure: 5 mm and 6 mm steel.
Main Engines: 2 x Cat 3306.
Gear Box: Twin disc.
Auxiliary: V.M. Driving 12.5 kw alternator.
Propeller/s: 2 x 4 blade 36 x 26.
Electrical system installation: 24V and 240V.
Depth Sounder: Daiwa 250D.
Radar: Furuno FR-701.
Radio: Codan 6801-S.
Autopilot: T.M.Q.
Compass: Eiwa Seiko.
Steering Gear: Palmarine 16AX.
Fuel Capacity: 8,000 litres.
Maximum Speed: 9.5 knots.
Cruising Speed: 8.5 knots.
Deck Carrying Capacity: 90 tonnes general cargo. Deck will take D9 dozer with blade. Hiab 650 crane mounted on port side.
Fresh Water Capacity: 22,000 gallons in main tank. 80 gallons in domestic tank.
Remarks: Extensive life saving and fire fighting equipment. Carries 2 crew + 6 passengers between Hayman Island and mainland. Excellent vessel that will adopt to many cargo duties around the Great Barrier Reef. Another fully designed and manufactured Queensland product.



"Centaur" could easily be plying the Aegean and not Australia's Whitsunday coast.

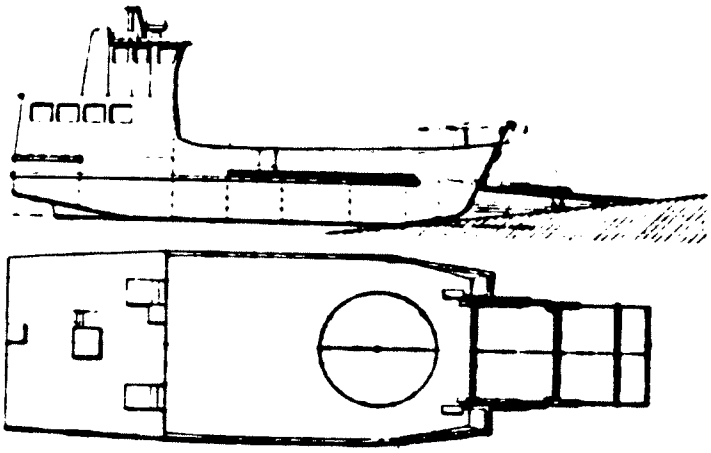


The single stern anchor arrangement seems inadequate for such a massive craft.



EILEAN NA-H-OIGE

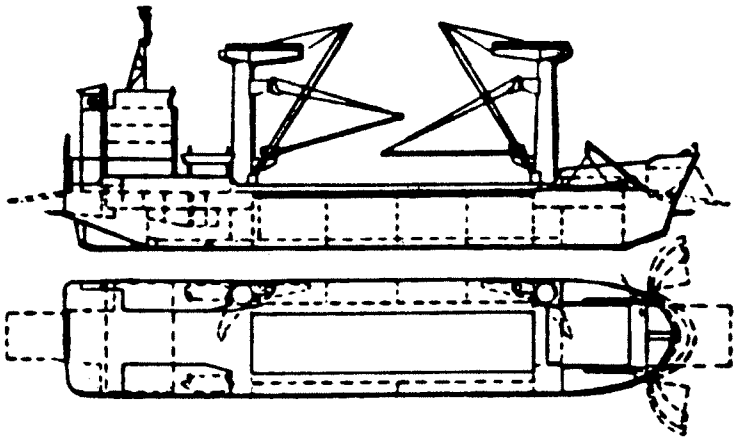
18.5 x 7 x 1.9



PRINCIPAL PARTICULARS

Eilean Na-H-Oige

Length (o.a.)	18.5m
Length at waterline	16.8m
Breadth (mld.)	7.0m
Depth (mld.)	1.9m
Draught, loaded	1.2m
Deadweight	22 tonnes
Displacement	108 tonnes
Speed	6½ knots



JOHN HENRY

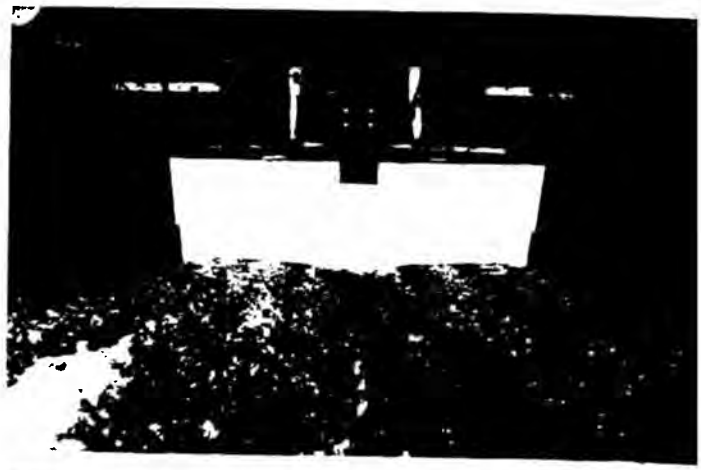
L B D

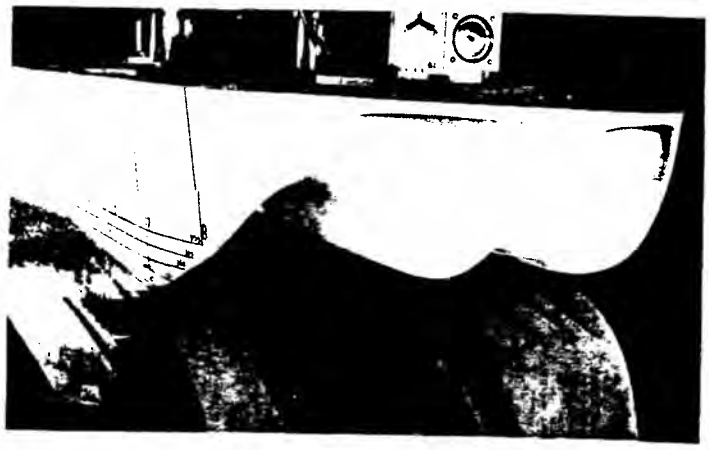
91.5 x 16.8 x 6.45

Model and Full-Scale Evaluation of the Stolkraft

19

The Stolkraft designed and manufactured by Stolkraft Pty. Ltd. of Sydney, Australia, is a novel high-speed craft making use of hydrodynamic and aerodynamic lift. The hull is designed to plane at moderate and high speeds without significant wash or waves. Air is fed through twin bow intakes and vented from a transverse step beneath the hull aft. As speed increases the importance of the aerodynamic ground effect increases, resulting in the absence of large trim variations once the so-called hump speed is passed. Model and full-scale measurements were recently carried out at NSMB.





Stolkraft undergoing resistance tests in the High-speed towing tank

The model experiments were carried out with a 2.5 m long model in the High Speed Basin with model speeds in excess of 13 m/sec. Further measurements were carried out in Sydney, Australia, on 5 m and 8.5 m long prototype hulls. The aims of the experiments were to ascertain the relative importance of the various components of resistance and to determine the performance as related to the size (or scale) of the hull.

Decreasing resistance

Once scaling relations have been established it is possible to determine the speed power performance of relatively large-sized craft on the basis of results from tests with models and smaller hulls. The results of many resistance, thrust, trim, speed, air-flow rate and cushion pressure measurements revealed that the performance of the Stolkraft improves with increase in size due to an increase in air-flow rate through the air-intakes and tunnels as size increases. The subsequent increase in ground effect results in a decrease of the bow-up-trim (at a constant LCG position), thereby decreasing resistance.

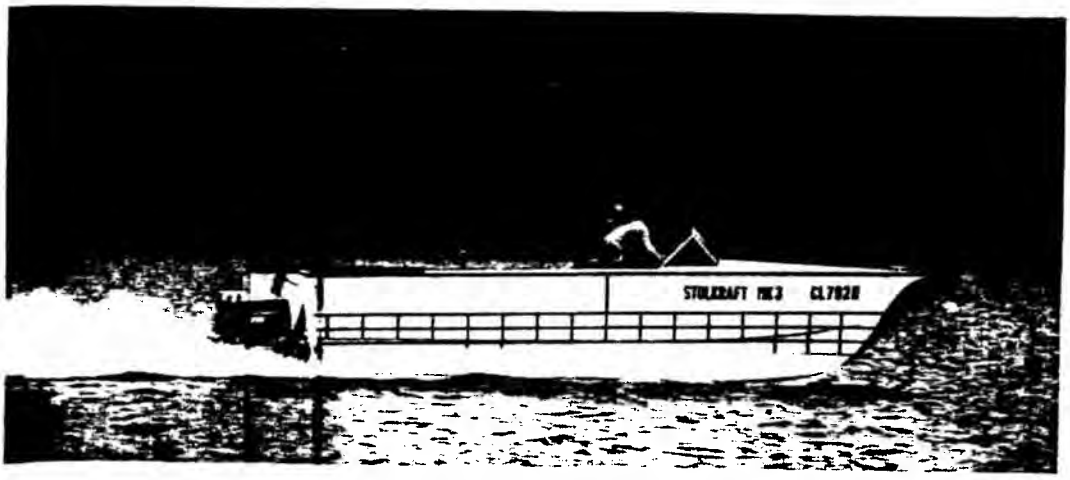
Reynold's number

It was found that the wave-making resistance is not a function of Froude number alone but also a function of the Reynold's number which governs properties of the air-flow. On the basis of the results of tests with the 3 Stolkraft hulls (comprising a so-called geosim), approximate scaling relations for frictional and residual resistance were established.

The influence of the air cushion, or ground effect, on the performance is significant, giving the craft favourable properties. At high speeds ($F_n\nabla$ values in excess of four) about 85% of the weight of the craft is borne by dynamic forces.

At high speeds the performance of the Stolkraft compares favourable with other high speed craft such as hydrofoils, air-cushion vehicles and surface effect ships.

■



The 8.5 m Stolkraft during testing in Sydney, Australia.

20

M A I N F I G U R E S NECESSARY FOR A FEASIBILITY STUDY ON LINER TRADES

- What ship is most suitable for the trade ?

(1) Amount of cargo weight and cubic.

(2) Number of ports - Distance(s) - Routeing.

(3) Frequency of service ... Fortnightly ? 10-dais' ? Weekly ?

(4) Size of ships - Capacity and number - Service speed - Port time - Duration of roundtrip.

(5) Price of ship - Financing - Depreciation + Interest

(6) Fuel consumption (Main & Auxiliaries) Fuel costs
Smoothnes and efficiency of 1-screw, 2-screw - Slow turning
Shafting HFO

(7) Port dues.

(8) Daily costs
- Manning
- Radio
- Insurance
- Repairs & Maint.
- Lubrication

(9) Transport costs/teu

(10) Draft restriction .... Cargoweight limitation.

(11) Stability criterion... ,, ,, .

(12) View restriction ..... Cargo volume limitation.

PRICE AND RUNNING COSTS

	ACTUAL CONCORDIA	PROGNOSIS NEDLLOYD VAN NOORT	ACTUAL WOENSDRECHT
NAME		" VAN DIEMEN	VAN OMMEREN
COMPANY	JANSSEN	VD GIESSEN DE NOORD	
YARD	THYSSEN/NORDSEE	APRIL '84 JULY '84	
DELIVERY	DEC '82	B.V.	B.V. I 3/3 CLACE 1
CLASS	G.L + 100A4E + MC AUT		
LOA	166.04 M 4T-1282 TEU (1125)	182.50 M 4T-1590 TEU	186.54 4T-1548 TEU
LPP	152.60	174.00	180.00
BMLD	28.50	30.50	30.00
HMLD	15.80	16.20	16.80
LIGHT SHIP	7850 T LBP B H x 105 KG/m³	11000 T 122 KG/m³	11200 T 119 KG/m³
KOREA USD/KG	2.30	2.23	2.21
	USD 18.— MILL	USD 25.6 MILL	USD 25.6 MILL
CRANES	2.4	1.8	2.4
BOWTHR.	0.4	B+ST 0.6	
SHAFTG.	0.3	0.4	
REEFER PLUGS	101 P 0.4	150 P 0.6	120 P 0.5
SHIPS PRICE	USD 21.5 MILL = 16800 USD/TEU (19100)	USD 29.0 MILL = 18200 USD/TEU	USD 28.5 MILL = 18400
CAPITAL COST 8% 20YR	0.10 x 2.15 MILL USD/YR	0.10 x 2.90 MILL USD/YR	0.10 x 2.90 MILL USD/YR
DAILY CAP COST	5890 USD/DAY	7940 USD/DAY	7810 USD/DAY
RUNNIN EXP.			
MANNING	3240.	3240	3240
RADIO	140	140	140
INSURANCE	500	621	611
REPAIRS	910	910	910
LUB.	256	256	256
	5046 5046 USD/DAY	5167 5167 USD/DAY	5157 5157 USD/DAY
CAPITAL + RUNNING	10936 USD/DAY	13107 USD/DAY	12967 USD/DAY
P. ROUNDTRIP 91 DAYS	995.000 USD	1.193.000 USD	1.180.000 USD
FUEL COSTS 91D ROUNDTRIP	2247 T x 168 USD/T HFO 378.000 — 116 T x 240 DO 28.000 406.000 USD	2212 T x 168 USD/T HFO 372.000 — 100 T x 240 DO 24.000 396.000 USD	2722 T x 168 USD/T HFO 457.000 — 168 T x 240 DO 40.000 497.000 USD
PORT DUES 91D ROUNDTRIP	17750 GRT ~12.5 USD/GRT 221.500 USD	22700 GRT ~11.8 USD/GRT 267.800 USD	26171 GRT ~11.1 USD/GRT 289.600 USD
TOTAL COST/ROUNTR.	1.622.500 USD	1.856.800 USD	1.966.600 USD
CONTRS/ROUNDTRIP	TEU	TEU	TEU
	WESTBOUND 1282	WESTB 1590	WESTB ~991
	EAST 1125	EASTB 1426	EASTB 1163
COST/TEU	ACTUAL ALMOST OPTIMAL 2407 → 674 USD/TEU	OPTIMAL 3016 → 616 USD/TEU	ACTUAL 2154 → 912 USD/TEU
		609 CONTRS, LESS 4 × 1685 T/TEU = 7200 TON / 2	2920 ← 674 " CONCORDIA
		3600 TON LESS CARGO → 9.450 ≈ 8.610	
		2,7 TON/DAY × 59 DAYS = 160 TON × 148 = 23.000 USD	CONCORDIA ACTUAL 2407 → 818 "
		TOTAL COST 1.829.800 USD	
		CONCORDIA 2407 → 760 USD/TEU	

20.1.18

COMPARISON OF COSTS - FOR VESSELS OF DIFFERENT SIZES (CAPACITIES)

Vessels under consideration here

1,598-TEU ship (mv NEDLLOYD VAN NECK)

398-TEU ship (CONOSHIP-vessel)

Costs under consideration here

(a) Running costs /per TEU /per Mile,

(b) Annual running costs - same service.

Notes.

- Costs for the containers proper (e.g. running costs, capital costs) are not shown in the comparison as the total yearly number of TEUs involved is assumed equal for each of the two vessel types here considered.

- Also, it has been assumed that cargo handling costs - i.e. per TEU - are equal for each of the two vessel types.

(a.) COSTS /per TEU /per MILE.

	mv NEDLLOYD VAN NECK	CONOSHIP vessel
Number of TEUs capacity	1,598	389
Price (DFls x mln)	110	25
Running costs per day.		
Capital charges	30,849	7,600
Manning	10,092	3,629
Bunkers (18' - 45t)	19,125	3,400
Insurance	2,228	862
ROU	2,100	825
Lub.oil	725	260
Electr. equipment	400	225
	65,519	16,700
Costs p.TEU p.Mile (DFls)	0.095	0.138

Note.
A graph showing COSTS/TEU/MILE for a range of vessel capacities from 400 TEUs to 4,000 TEUs each is given separately.

fta - Oct. '83.

20.1.19

(b.) COMPARISON OF COSTS - ON ANNUAL BASIS

(excluding container costs and handling costs which assumed equal for both ships)

Ships compared:		mv NEDLLOYD VAN NECK		CONOSHIP vessel
Containers p.voy.	(2x 1598) =	3196	(2x 389) =	778
Round voyage	(40 days) =	9 /year.	(45 days) =	8 /year.

SEA-DAYS	DFl per roundvoyage	DFl per roundvoy.
24 days x DFl 65,519	1,572,456	
33 days x DFl 16,700		551,100
PORT-DAYS		
16 days x DFl 48,094	769,504	
12 days x DFl 14,151		169,812
PORT DUES per roundvoyage	386,400	134,400
TOTAL COSTS PER ROUNDVOYAGE	2,728,360	855,312
Distance covered per r.voy.	10,322 Naut.miles.	10,322 Naut.miles.
Costs/TEU/Mile	$\frac{2,728,360}{3196 \times 10,322} = 0.0827$	$\frac{855,312}{778 \times 10,322} = 0.1065$
Number of CONTAINERS/YEAR/SHIP	28,764	6,224
Number of ships required	1 :	5 (viz. $\frac{28,764}{6,224} = 4.6$)
TOTAL COSTS/YEAR	9 x 2,728,360 = 24,555,240	4.6 x 8 x 855,312 = 31,475,481

fta/Oct. '83.

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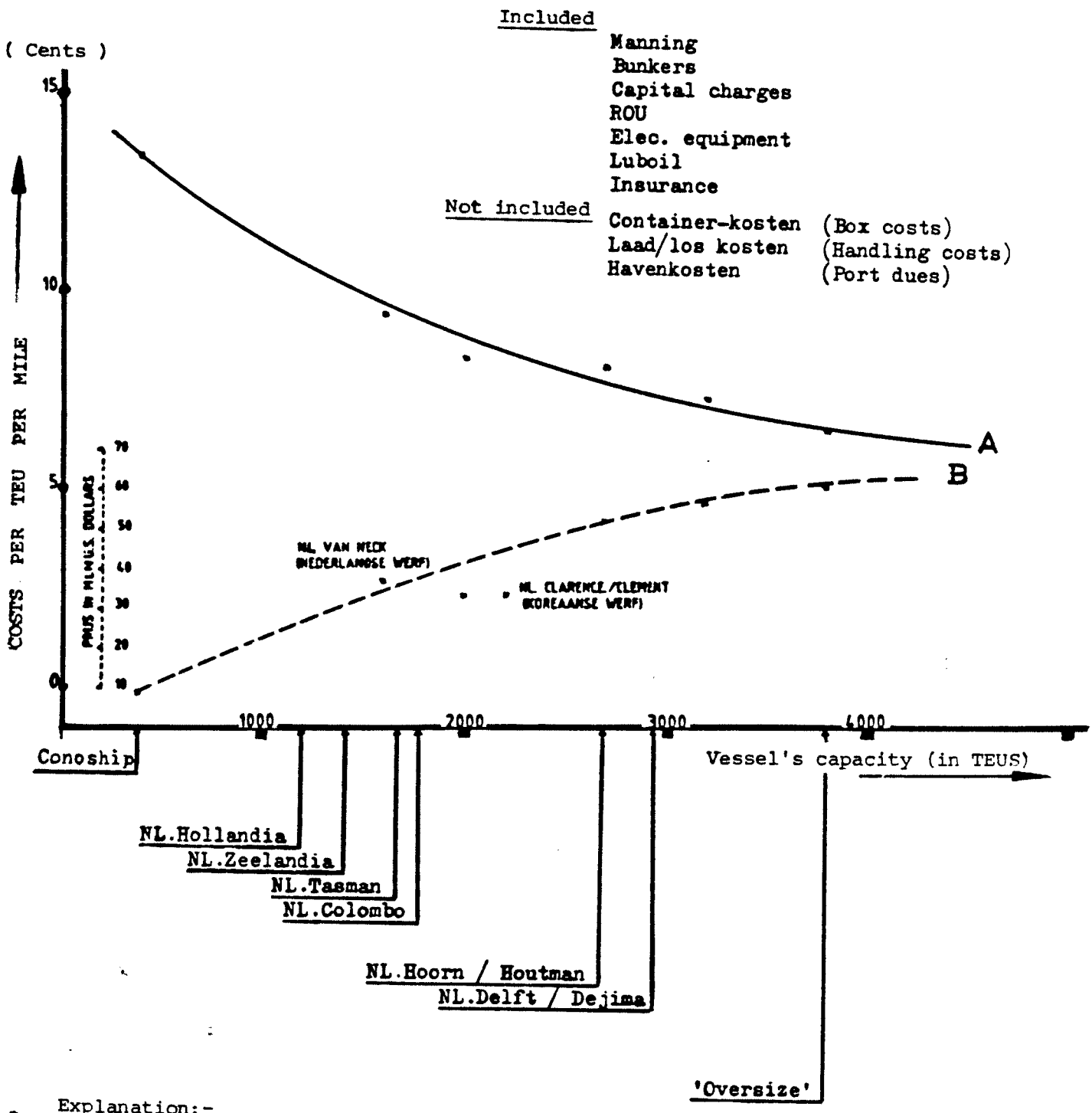
BOX CARRYING COSTS — BY SIZE OF SHIP (Graph "A")

SHIP'S BUILDING COSTS — BY SIZE OF SHIP (Graph "B")



Explanation:-

T E U costs per mile - by vessel's size (capacity).

Vessel building costs - by vessel's size (capacity).

(PRELIMINARY) CALCULATION OF TRANSPORT COSTS PER TEU

FOR A 210m CONTAINER SHIP FOR DIFFERENT BLOCKCOEFFICIENTS - CONDITIONS AS PER MAY 1981

210 m. UNIV. CONTAINER	V SERVICE = 18½ KN.	ROUNDTRIP = 45½ DAYS AT SEA + 25½ DAYS IN PORT = 71 DAYS × 5.07 ROUNDTRIP/YR = 360 DAYS (5 ROUTES)				
MAY '81	1 USD = 2.50 GLD	CB = .59	CB = .62	CB = .635	CB = .65	CB = .665	CB = .68
BHP incl. gen. = 17470	motor: B&W 6L90GBE MCR = 23000 PK	17700	17850	18080	18730	20040
SHAFT. GEN. = 1500		1500	1500	1500	1500	1500
18970 = 80% MCR.	FUEL PRICE 20% UP	19200 = 81%	19350 = 81.6%	19580 = 82.6%	20230 = 85.4%	21540 = 90.9%
F.O. CONSUMP. sea = 18970 × 150 × 24 × 10⁻⁶ = 68.3 T/day		69.1 T/day	69.7 T/day	70.5 T/day	72.8 T/day	77.5 T/day
D.O. CONSUMP. port = 720 kW × 1½ × 164 × 24 × 10⁻⁶ = 4.25 T/day						
F.O. sea = 5.07 × 45½ × 68.3 × 282 USD/T =	4.443.000	4.497.000	4.532.000	4.586.000	4.738.000	5.043.000
F.O. port = 5.07 × 25½ × 1.5 × 282 =	55.000	55.000	55.000	55.000	55.000	55.000
D.O. port = 5.07 × 25½ × 4.25 × 420 USD/T =	231.000	231.000	231.000	231.000	231.000	231.000
FUEL COSTS	4.729.000	4.783.000	4.818.000	4.872.000	5.024.000	5.331.000
MANNING 3040 USD/day × 365 =	1.109.600					
RADIO 144 × 365 =	52.600					
INSURANCE 680 × 365 =	248.200					
REPAIRS 1800 × 365 =	657.000					
LUB 340 × 365 =	124.100					
RUNNING EXP.	2.192.000	2.192.000	2.192.000	2.192.000	2.192.000	2.192.000
LIGHT SHIP = 13800 T × 3.15 USD/kg × 1000 = 43.5 MILL. USD		13900 × 3.15 = 43.8 MILL.	13950 × 3.15 = 43.95 MILL	14000 × 3.15 = 44.1 MILL	14050 × 3.15 = 44.25 MILL	14100 × 3.15 = 44.4 MILL
ANNUITY 12% - 20 YR: FACTOR 0.1339						
CAPITAL COSTS	5.821.000	5.863.000	5.884.000	5.905.000	5.926.000	5.947.000
DISPL. = 42000 T × 55 USD/T =		44150 × 55 =	45200 × 55 =	46250 × 55 =	47350 × 55 =	48400 × 55 =
PORT DUES	2.310.000	2.428.000	2.486.000	2.544.000	2.604.000	2.662.000
TRANSPORT COSTS	15.052.000	15.266.000	15.380.000	15.513.000	15.746.000	16.132.000
10.60 M Δ = 42000 T.		44.150 T.	45.200 T.	46.250 T	47.350 T	48.400 T
LS = 13.800		13.900	13.950	14.000	14.050	14.100
DW = 28200		30.250	31250	32250	33300	34300
CONS + WB = 4500		4500	4500	4500	4500	4500
CARGO = 23700		25750	26750	27750	28800	29800
14 TON/TEU → 1692 TEU		→ 1839 TEU	→ 1910 TEU	→ 1982 TEU	→ 2057 TEU	→ 2128 TEU
PHYSICAL → 1881 TEU 3T.		PH → 1936 TEU 3T.	PH → 1955 TEU 3T.	PH → 1972 TEU 3T.	PH → 1986 TEU 3T.	PH → ~2000 TEU 3T
TR. COSTS/PH. TEU (2 × 5.07 TRIPS) = **789** USD/TEU		**777½** USD/TEU	**776** USD/TEU	**776** USD/TEU	**782** USD/TEU	**795½** USD/TEU
9.15 M Δ = 34300 T. (C_B 0.558)		36550 T. (C_B .595)	37450 T (C_B .60)	38350 T (C_B .624)	39200 T. (C_B .648)	40100 T. (C_B .653)
LIMITED DRAFT LS = 13800		13900	13950	14000	14050	14100
DW = 20500		22650	23500	24350	25150	26000
CONS + WB = 2900		2900	2900	2900	2900	2900
CARGO = 17600		19750	20600	21450	22250	23100
12.5 TON/TEU → 1408 TEU		→ 1580 TEU	→ 1648 TEU	→ 1716 TEU	→ 1780 TEU	→ 1848 TEU
TR. COSTS/TEU = **1054** USD/TEU		**953** USD/TEU	**920** USD/TEU	**892** USD/TEU	**872** USD/TEU	**861** USD/TEU

(PRELIMINARY) CALCULATION OF TRANSPORT COSTS PER TEU

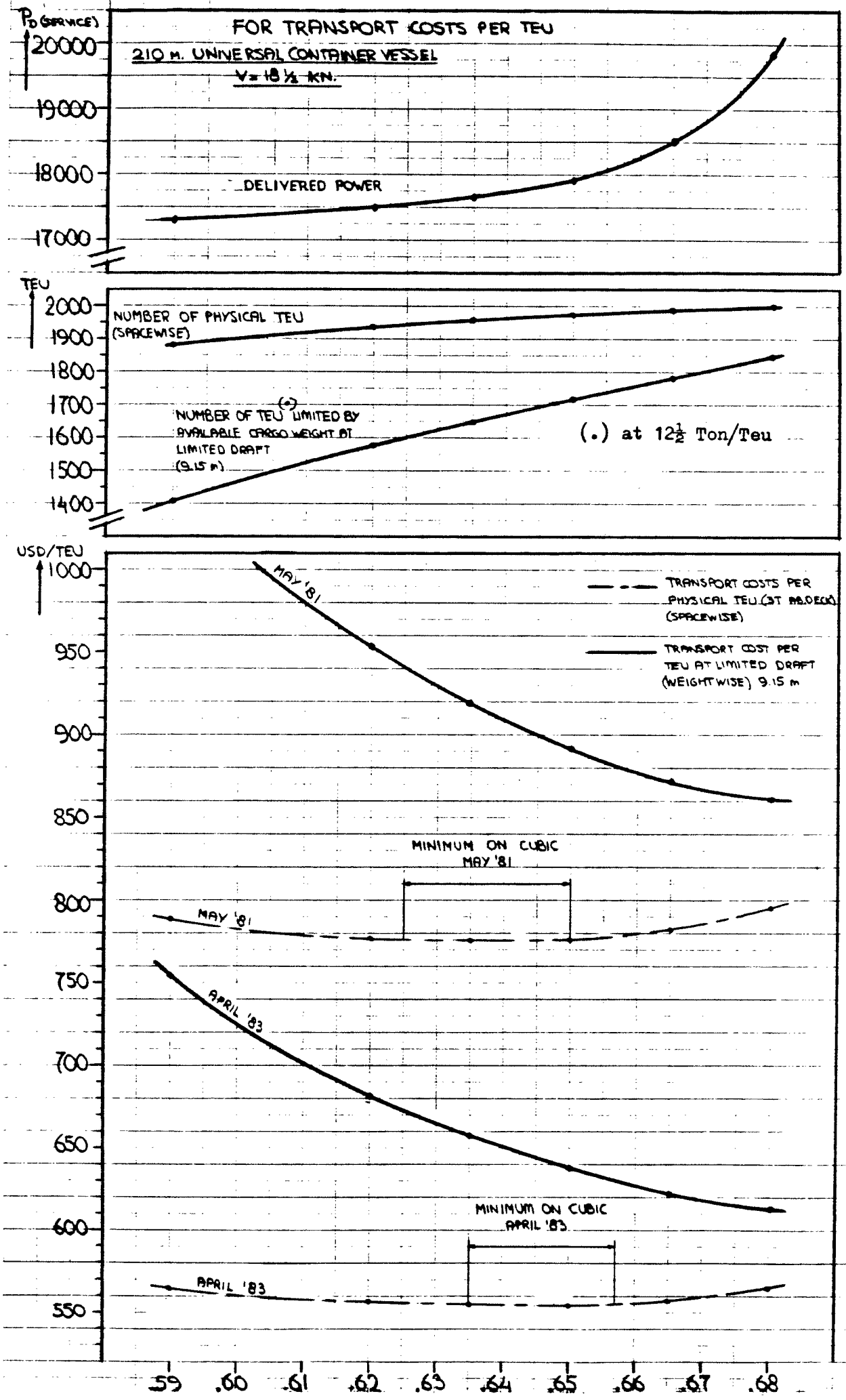
FOR A 210m CONTAINER SHIP FOR A RANGE OF BLOCKCOEFFICIENTS

CONDITIONS AS PER APRIL 1983

210 m UNIV. CONTAINER	V SERVICE = 18½ kn.	ROUNDTRIP = 45½ DAYS AT SEA + 25½ DAYS IN PORT = 71 DAYS x 5,07 ROUNDTR/YR = 360 D (+5 DAYS)				
APRIL '83 / 1 USD = 2,75 DM	CB = .59	CB = .62	CB = .635	CB = .65	CB = .665	CB = .68
BHP INCLUDEN = 17470	motor: B&W 6L80GFB MCR = 23200/07	17700	17850	18080	18730	20040
SHAFT.GEN. = 1500		1500	1500	1500	1500	1500
18970 = 80% M.C.R.		19200 = 81%	19350 = 81.6%	19580 = 82.6%	20230 = 85.4%	21540 = 90.9%
F.O. CONSUMP. SEA = 18970 x 141 x 24 x 10⁻⁶	= 64.2 T/DAY	65.0 T/DAY	65.5 T/DAY	66.3 T/DAY	68.5 T/DAY	72.9 T/DAY
D.O. CONSUMP. PORT = 720 KW x 1½ x 141 x 24 x 10⁻⁶	= 4.25 T/DAY					
F.O. SEA = 5.07 x 45½ x 64.2 x 168 USD/T	2.488.000	2.518.000	2.538.000	2.568.000	2.635.000	2.825.000
F.O. PORT = 5.07 x 25½ x 1.5 x 168	33.000	33.000	33.000	33.000	33.000	33.000
D.O. PORT = 5.07 x 25½ x 4.25 x 238.5 USD/T	131.000	131.000	131.000	131.000	131.000	131.000
FUEL COSTS	2.652.000	2.682.000	2.702.000	2.732.000	2.817.000	2.989.000
MANNING 3670 USD/DAY x 365 =	1.339.600					
RADIO 147 x 365 =	53.700					
INSURANCE 1448 x 365 =	528.500					
REPAIRS 909 x 365 =	331.800					
W.B. 345 x 365 =	125.900					
RUNNING EXP.	2.380.000	2.380.000	2.380.000	2.380.000	2.380.000	2.380.000
LIGHT SHIP = 13800 T x 2.2 USD/kg x 1000 = 30.4 MILL. USD		13900 x 2200 = 30.6 MILL	13950 x 2200 = 30.7 MILL	14000 x 2200 = 30.8 MILL.	14050 x 2200 = 30.9 MILL	14100 x 2200 = 31.0 MILL
ANNUITY 12% - 20 YR : FACTOR .1339						
CAPITAL COSTS	4.065.000	4.095.000	4.109.000	4.124.000	4.139.000	4.155.000
DISPL. 42000 T x 40 USD/T =		44150 x 40 =	45200 x 40 =	46250 x 40 =	47350 x 40 =	48400 x 40 =
PORT DUES	1.680.000	1.766.000	1.808.000	1.850.000	1.894.000	1.936.000
TRANSPORT COSTS	10.777.000	10.923.000	10.999.000	11.086.000	11.230.000	11.460.000
10,60 M Δ = 42000 T.		44150 T	45200 T	46250 T	47350 T.	48400 T.
LS = 13800		13900	13950	14000	14050	14100
DW = 28200		30250	31250	32250	33300	34300
CONST + WB = 4500		4500	4500	4500	4500	4500
CARGO = 23700		25750	26750	27750	28800	29800
14 TON/TEU → 1692 TEU		→ 1839 TEU	→ 1910 TEU	→ 1982 TEU	→ 2057 TEU	→ 2128 TEU
PHYSICAL → 1881 TEU 3T.		PH → 1936 TEU 3T.	PH → 1955 TEU 3T.	PH → 1972 TEU 3T.	PH → 1986 TEU 3T.	PH → ~2000 TEU 3T.
TR.COSTS/PH.TEU (2x5,07 TR/PS) =	565 USD/TEU	556½ USD/TEU	555 USD/TEU	554½ USD/TEU	557½ USD/TEU	565 USD/TEU
9,15 M Δ = 34300 T. (CB .558)		36550 T (CB .595)	37450 T. (CB .610)	38350 T. (CB .624)	39200 T. (CB .638)	40100 T. (CB .653)
LIMITED DRAFT LS = 13800		13900	13950	14000	14050	14100
DW = 20500		22650	23500	24350	25150	26000
CONST + WB = 2900		2900	2900	2900	2900	2900
CARGO = 17600		19750	20600	21450	22250	23100
12.5 TON/TEU → 1408 TEU		→ 1580 TEU	→ 1648 TEU	→ 1716 TEU	→ 1780 TEU	→ 1848 TEU
TR.COSTS/TEU =	755 USD/TEU	682 USD/TEU	658 USD/TEU	637 USD/TEU	622 USD/TEU	612 USD/TEU

20.1.17

OPTIMUM BLOCKCOEFFICIENT



Ro Ro CONTR | CELL CONTR

LINE	ANRO ASIA	ROTTERDAM-ROCH.R.	ROUEN - ROSARIO	ASIAN JADE	VAN NECK	VAN NOORT	HOLLANDIA	ZEELANDIA
	AUSTR-MALAYA	USA-MIDDLEAST	USA-MIDD-EAST	AUSTR-PHI-MIDDLE EAST	EUR-WAFR	N ZEALAND-IND MALAYA	EUROPA-CARIB	EUROPA-MID AMERICA W C
CONTR. IND	1976	1977	1977	1977	1981	1982	1973	1978
L_{oa}	213.17	196.50	212.10	186.00	182.50	182.50	204.00	205.96
L_{pp}	199.16	183.20	198.80	177.40	174.00	174.00	193.10	192.00
B	27.50	32.24	32.24	27.60	30.50	30.50	30.80	30.80
H	16.75	23.25	20.40	16.20	16.20	16.20	18.80	18.75
T	8.91	10.30	10.70	10.62	11.00	11.00	10.02	10.02
LIGHT SHIP	12.100	17.249	16.170	9.228	11.030	10.600	13.740	13.750
DW	21.510	15.080	25.155	20.355	28.307	28.807	21.212	21.678
CONTR. IND	18.02	14.55	13.30	11.43	11.71	120	12.49	12.65
	LLOYDS	LLOYDS	LLOYDS	LLOYDS	B. VERITAS	B. VERITAS	LLOYDS	LLOYDS
TEU-HOLD	634	946	811	611	788	780	878	920
TEU on DECK	362	494	717	344	668	680	328	514
TEU	4	172	185	186	36	120	66	66
TEU	1136	1317	1713	1190	1550	1580	1472	1500
	134 R 20/40	100 R 20/40	100 R 20/40	88 R 20/40	60 R 20/40	60 R 20/40	120-40	180-40
	66 R 20/40			54 R 20/40	90 R 20/40	90 R 20/40	50 R 20/40	50 R 20/40
							126 R 20/40	126 R 20/40
M. ENGINE	MAN	SULZER 8 RND 30 M	SULZER 8 RND 30 M	SULZER 8 RND 76/M	SULZER 5 RLB 90	IDEM	SULZER 10 RND 90	SULZER 9 RND 90
MCR/RPM	18.990-430	26.800-122	26.800-122	19.200-122	"17.500"-84		28.000-122	30.000-122
85%	16.140-110 RPM	22.800-122 RPM	22.800-122 RPM	16.300-122 RPM	14.870-94 RPM		24.650-122 RPM	25.600-122 RPM
	2000				1200			
LLER	ø 6000-4 BL-% 59 V.C.	ø 6500-4 BL-% .715	ø 6500-4 BL-% .727	ø 6320-4 BL-% .65	ø 7000-4 BL-% .55		?	ø 6500-4 BL-% .874
	5101 B 3520 PROP	51908 3420 PROP	5274 3300 PROP	5367 AOD 3015 PROP	5343 B 3930 PROP	IDEM	HSVA TESTS	
SR/DISPL	8.91 32.610 T	10.00 36.200 T	10.00 41.070 T	9.60 29.420 T	10.00 34.780 T		10.00 36.352 T	10.00 37.828
BLOCK	4.692 32.800 m³	4.517 35.189 m³	4.615 40.042 m³	4.4 28.638 m³	4.639 33.926 m³		4.605 36.000 m³	4.618 36.430 m³
V-SHIP	18.-KN-12.300 HP	19.-KN-16.950 HP	19.-KN-17.300 HP	19.-KN 13.050 HP	18.-KN 13.250 HP		19.-KN-18.600 HP	19.-KN 17.930 HP
	14.150	19.500	19.900	15.000	15.240		21.400	20.630
TEN/SHIP	165×24×1.84×8.2.6	161×24×7.29	161×24×7.74	162×24×8.8	150×24×5.5		162×24×8.3	162×24×8.3
		VESSEL TOO SHORT			EFFICIENT ENGINE		BAD HULL: BAD PROP	BAD PROP
D^2V^3	1042×5832 / 11300 = 475	1095×6859 / 16350 = 448	1191×6859 / 17400 = 475	954×6859 / 13050 = 502	1068×5832 / 13250 = 469		1110×6859 / 18600 = 409	1120×6859 / 17950 = 438
D^2V^3 / TEU	1042×5832 / 596 = 102	1095×6859 / 739 = 99	1191×6859 / 77.4 = 106	954×6859 / 58.4 = 112	1068×5832 / 55.- = 113		1110×6859 / 83.3 = 91	1120×6859 / 80.3 = 96

	ANRO ASIA	ROTTERDAM-ROCH.	ROUEN-ROSARIO	ASIAN JADE	KLECO		HOLLANDIA	ZEELANDIA
DR/DISPL	8.91 33.610	10.72 35.869	10.72 43.153	10.62 33.610	11.00 39.407	11.00 39.407	10.00 36.352	10.02 37.428
LIGHT SHIP	12.100	17.249	16.170	9.228	11.100	10.600	13.740	13.750
DW	21.510	22.560	29.023	24.387	28.307	28.807	23.212	23.678
CONS+WB	4.570	5.154	6.303	7.888	6.112	6.112	5.864	5.698
TEU	896×17×16.940	17.406	1530×14×22.720	1176×14×16.494	1465×13.5×22.195	22.695	1240×14=17.348	1283×14=17.980
IN HULL	634	11.877	660×16=10.560	660×16=10.560	785×15.5=11.895		878×14=12.280	910×14=12.850
ON DECK	362	5.529	516×18=9.934	516×18=9.934	680×15.5=10.500		362×14=5.068	363×14=5.050
STABILITY			KM=13.41 KG=14.62 MG=1.14-0.08=1.06	KM=12.61 KG=11.55 MG=1.06-0.06=1.00	KM=14.35 KG=13.06 MG=1.29-0.08=1.21		KM=13.18 KG=12.44 MG=0.63-0.05=0.68	KM=14.01 KG=12.91 MG=1.10-0.12=0.98
DR/DISPL		9.97 36.319	10.02 41.325	9.62 29.583	10.77 38.350	38.350	9.89 36.538	10.02 37.428
LIGHT SHIP		17.249	16.170	9.228	11.100	10.600	13.740	13.750
DW		15.080	25.155	20.355	27.250	27.750	22.798	23.678
CONS+WB		5.827	6.666	7.888	6.011	6.011	6.298	7.678
TEU		13.753	18.491	1176×10.6=12.466	1283×13.5=21.239	21.739	1200×13.3=16.000	1250×13.8=16.000
IN HULL		7.817	660×10.6=6990	660×10.6=6990	785×13.5=10.520		878×13.3=11.700	910×12.8=11.780
ON DECK		5.436	516×10.6=5.476	516×10.6=5.476	500×13.5=10.719		322×13.3=4.300	330×12.8=4.220
STAB			KM=12.33 KG=14.16 MG=1.97-0.12=0.85	KM=12.31 KG=11.32 MG=0.99-0.07=0.92	KM=14.34 KG=13.47 MG=0.87-0.07=0.80		KM=13.11 KG=12.00 MG=1.11-0.05=1.06	KM=14.01 KG=12.35 MG=1.66-0.12=1.54
DR/DISPL			10.02 41.325	9.62 29.583	9.75 32.700	33.700	9.53 34.758	9.57 35.251
LIGHT SHIP			16.170	9.228	11.100	10.600	13.740	13.750
DW			25.155	20.355	22.600	23.100	21.053	21.501
CONS+WB			8.295	4.492	3.420	3.420	7.053	9.501
TEU			16.860	1004×13.5=13.863	1035×12.5=19.188	19.680	1200×11.7=14.000	1250×11.6=12.000
IN HULL			660×13.5=10.420	660×13.5=10.420	785×12.5=12.748		878×11.7=10.250	910×11.6=8.830
ON DECK			344×13.5=5.443	344×13.5=5.443	311×12.5=3.449		322×11.7=3.750	330×11.6=3.170
STAB			KM=12.49 KG=14.35 MG=1.86-0.13=1.73	KM=12.31 KG=11.65 MG=0.66-0.08=0.58	KM=14.37 KG=12.13 MG=2.24-0.09=2.15		KM=13.05 KG=11.89 MG=1.16-0.05=1.11	KM=13.96 KG=11.73 MG=2.23-0.12=2.11
DR/DISPL	M			9.62 29.583	9.15 31.070	31.070	9.15 33.080	9.15 33.382
LIGHT SHIP	THE			9.228	11.100	10.600	13.740	13.750
DW	"			20.355	19.970	20.470	19.340	19.632
CONS+WB	"			3.488	3.420	3.420	~3.000	3.000
TEU	"			1004×16.8=16.867	1036×15.5=16.350	17.050	16.340	16.632
IN HULL	"			660×16.8=11.090	785×12.5=9.810			
ON DECK	"			344×16.8=5.777	339×12.5=6.740			
				KM=12.51 KG=12.00 MG=0.43-0.09=0.34	KM=14.49 KG=12.92 MG=1.40-0.08=1.40			
	M							CORR 27 NOV 83 GV

CELL. CONTR.

LINE	ABEL TASMAN	COLOMBO	CLEMENT	CLARENCE	HOUTMAN + HOORN	DE JIMA + DELFT	PANMAX	4 TIER EXTRA 3 ROWS WIDER
Loa	225.87	211.15	210.00	210.00	258.00	287.02	289.50	289.50
Lpp	210.00	195.00	202.00	202.00	247.00	273.00	277.00	277.00
B	30.50	32.20	32.24	32.24	32.26	32.24	32.24	39.60
H	16.40	19.00	18.80	18.80	24.15	25.00	24.15	24.35
T	11.58	11.56	12.00 (EXCL CRANE INCL RUDDER)	12.00 (GEARLESS)	13.00	12.72	13.00	13.00
LIGHT SHIP	14 200	13 000	14 998	14 545	22 281	24 899	24 700	29 800
Cb	30.76	79.3	84.25	84.31	102.09	112.34	112.94	115.34
% LOA	11.78	12.81	12.50	12.39	13.58	13.50		
CLASS/SYSTEM	GERM LLOYD	LLOYDS	LLOYDS		LLOYDS	LLOYDS		
TEU-HOLD	861	1014	1130	1130	1794	2020	2264	2720
TEU-DECK	598	511	658	562	660	668	842	1036
TEU	218	252	285	291	260	264	364	448
TEU	1677	1777	1997	2003	2714	2852	3470	4204
REEFER SLOTS	100 20 & 8' CRANE	0	116 40 & 8' CRANE	0	352 20 & 8' CRANE	0		448
DRY CYL. HOLD	0	130 R 20' MIXED	64 R 20' MIXED	88 R 20' MIXED	886 20 & 8'	70 R 40' MIXED	< 88 R 40' MIXED FOR 20'	4652
DRY CYL. DECK	20 R 20/40'	88 R 20/40'	99/68 R 20/40'	99/88 R 20/40'	53 R 20/40'	50 R 20/40'	121 R 20/40'	
M ENGINE	TURB. STAL LAVAL AP 32	SULZER 6RLA 90	B+W 6L90GBE	IDEM	SULZER 2 x 8 RND 90 M	SULZER 2 x 8 RND 90 M	(2 x 7 CYL L 90 GBE)	(2 x 7 CYL RTA 84)
MCR/RPM	32 450 - 1181	20 400 - 50	23 800 - 97		2 x 26 800 - 122	2 x 26 800 - 112	2 x 28 300 - 97	2 x 28 210 - 87
85%	SHP 27 600 - 110 RPM	SHP 17 340 - 50 RPM	SHP 20 230 - 97 RPM		SHP 2 x 22 800 - 102 RPM	SHP 2 x 22 800 - 112 RPM	SHP 2 x 24 050 - 97 RPM	SHP 2 x 24 000 - 87 RPM
AUX. GEN.			2 x 1200 KW / 2 x 1000 KW				2 x 1800 KW	2 x 1800 KW
TUNNEL	Ø 7400 - 5.8 m - % = 0.646	Ø 7400 - 5.8 m - % = 0.781	Ø 7200 - 4.8 m - % = 0.63		Ø 6850 - 4.8 m - % = 0.558	Ø 6618 - 4.8 m - % = 0.558	Ø 7000 - 4.8 m - % = 0.592	Ø 7250 - 4.8 m - % = 0.56
L BP	189.4	172.9	189.0		238.0	264.5	272.5	272.5
B/DEPTH	10.67	10.60	10.60		11.50	11.50	11.50	11.50
BLOCK	41.870 T	40.430 T	44.800 T		60.800 T	62.600 T	70.200 T	84.200 T
MODEL	41.805 M	35.434 M	43.774 M		55.288 M	61.100	68.475 M	84.100 M
V	19.600 KT	16.300 KT	17.250 KT		20.700	20.200	20.220	?
TEUS	22.558	18.750	19.030		33.000	32.400	33.600	?
TON/DAY	225 x 24 = 122 TO BE IMPROVED	150 x 24 = 67.5	147 x 24 = 70.0		168 x 24 = 128	157 x 24 = 122	147 x 24 = 119	147 x 24 =
D/V	122 x 6000 / 19600 = 500	177 x 6059 / 16300 = 495	126 x 6059 / 17350 = 502		1547 x 9261 / 28700 = 500	1578 x 9261 / 28300 = 518	1702 x 9261 / 29210 = 533	1931 x 9261 = ?
D/V'	122 x 6000 / 122 = 81	177 x 6059 / 67.5 = 120	126 x 6059 / 70.0 = 124		1547 x 9261 / 128 = 112	1578 x 9261 / 122 = 120	1702 x 9261 / 119 = 132	1931 x 9261 = ?

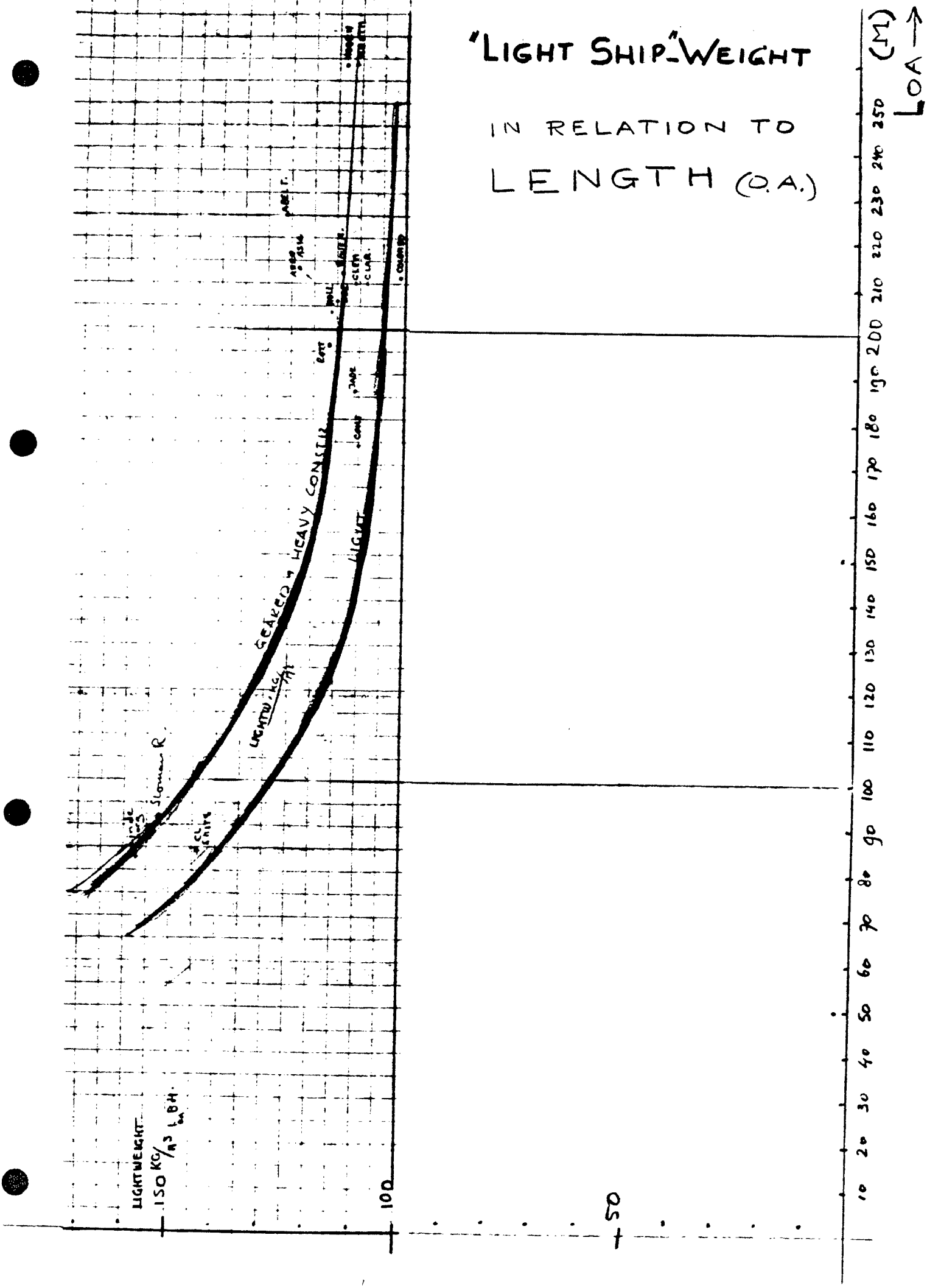
	ABEL TASMAN	COLOMBO	CLEMENT	CLARENCE	HOUTMAN + HOORN	DE JIMA + DELFT	PANMAX	3 ROWS WIDER
DW/DISPL	11.23 45.917	11.36 45.816	11.58 52.992	11.58 52.992	13.03 71.343	12.72 71.683	12.46 77.695	13.80 101.400
LIGHT SHIP	14.200	13.000	14.998	14.545	HOUTMAN 22.281	24.899	24.700	29.800
D.W.	31.717	32.816	37.994	38.447	49.062	46.784	52.995	71.200
CONS + WB	NOT MEN 8.163	NOT MEN 4.456	6.354	6.347	11.386	14.386	10.455	11.400
TEU/TEU	LT 1189 x 14.8 = 22.454	LT 1777 x 16 = 28.360	2000 x 15.8 = 31.600	2000 x 16 = 32.100	2714 x 13.9 = 37.676	2888 x 11.9 = 32.278	42.540	LT 4284 x 14.5 = 59.800
IN HULL	908 x 13.7 = 13.842	1130 x 13.8 = 15.670	1130 x 13.8 = 15.670		1794 x 14.8 = 26.394	2020 x 13.7 = 26.394		LT 4652 x 11.8 =
ON DECK	LT 861 x 10.6 = 9.613	830 x 13.8 = 11.460	830 x 13.8 = 11.460		920 x 8.6 = 7.994	832 x 6.3 = 5.884		
STABILITY	KM = 13.69 KG = 13.25 MG = 0.67 - 0.05 = 0.62	KM = 13.80 KG = 14.95 MG = 0.43 - 0.07 = 0.36	KM = 13.87 KG = 14.01 MG = 0.16 - 0.02 = 0.14		KM = 13.14 KG = 14.04 MG = 0.30 - 0.12 = 0.18	KM = 14.93 KG = 14.25 MG = 0.70 - 0.04 = 0.66	KM = KG = MG = 0.40	KM = KG = MG =
DW/DISPL	10.60 41.377	11.17 42.776	10.57 45.139	10.57 45.139	12.00 64.097	12.00 66.303	12.24 75.845	12.50 95.800
LIGHT SHIP	14.200	13.000	14.998	14.545	22.281	24.899	24.700	29.800
D.W.	28.418	30.776	30.141	30.594	41.716	41.404	51.145	66.000
CONS + WB	8.918	4.456	4.741	4.694	10.400	11.598	11.365	~11.000
TEU/TEU	1389 x 13/1.6 = 15.600	LT 1316 x 16 = 26.320	2000 x 15.2 = 31.600	2003 x 12.9 = 25.900	2454 x 12.7 = 31.316	2417 x 12.9 = 29.806	39.780	LT 4284 x 12.1 = 51.800
IN HULL	908 x 13 = 10.250		1130 x 13.7 = 14.600		1794 x 13.9 = 23.922	2020 x 13.7 = 26.394	3106 x 12.6 = 39.08	LT 4652 x 11.8 =
ON DECK	319 x 13 = 9.350		830 x 13.7 = 11.800		660 x 11.2 = 7.394	597 x 8.6 = 5.412	3/4 x 2 = 78	
STABILITY	KM = 13.86 KG = 13.39 MG = 0.49 - 0.05 = 0.44	KM = 13.36 KG = 13.11 MG = 1.22 - 0.07 = 1.16	KM = 13.05 KG = 14.06 MG = 0.15 - 0.06 = 0.15		KM = 13.10 KG = 14.38 MG = 0.32 - 0.12 = 0.40	KM = 14.82 KG = 14.16 MG = 0.66 - 0.04 = 0.64	KM = KG = MG = 0.48	KM = KG = MG =
DW/DISPL		10.95 42.744	9.75 40.440	9.75 40.440			11.24 62.655	
LIGHT SHIP		13.000	14.998	14.545			24.700	
D.W.		29.744	25.442	25.895			42.955	
CONS + WB		4.456	2.002	1.955			WBTS DBL 9.045	
TEU/TEU		LT 1525 x 16.5 = 25.186	1804 x 12.9/1.6 = 23.440	1840 x 13 = 23.940			43% 20' 1157 40' 1451 = 33.910	
IN HULL							3239 x 2.7 = 31.470	
ON DECK							230 x 2 = 460	
STABILITY		KM = 13.32 KG = 14.10 MG = 1.22 - 0.07 = 1.15	KM = KG = MG =					
DW/DISPL		7.88 27.865	9.15 37.340	9.15 37.340				
LIGHT SHIP		13.000	14.998	14.545				
D.W.		14.865	22.342	22.795				
CONS + WB		BUNKERS = 11.711	2.002	1.955				
TEU/TEU		1853 x 1.7 = 3.154	1630 x 12.7/1.6 = 20.340	1670 x 12.5 = 20.840				
IN HULL								
ON DECK								
STABILITY		KM = 13.46 KG = 10.10 MG = 3.36 - 0.11 = 3.25	KM = KG = MG =		LIGHT SHIP 22 706			CORR 27 NOV 83 GV

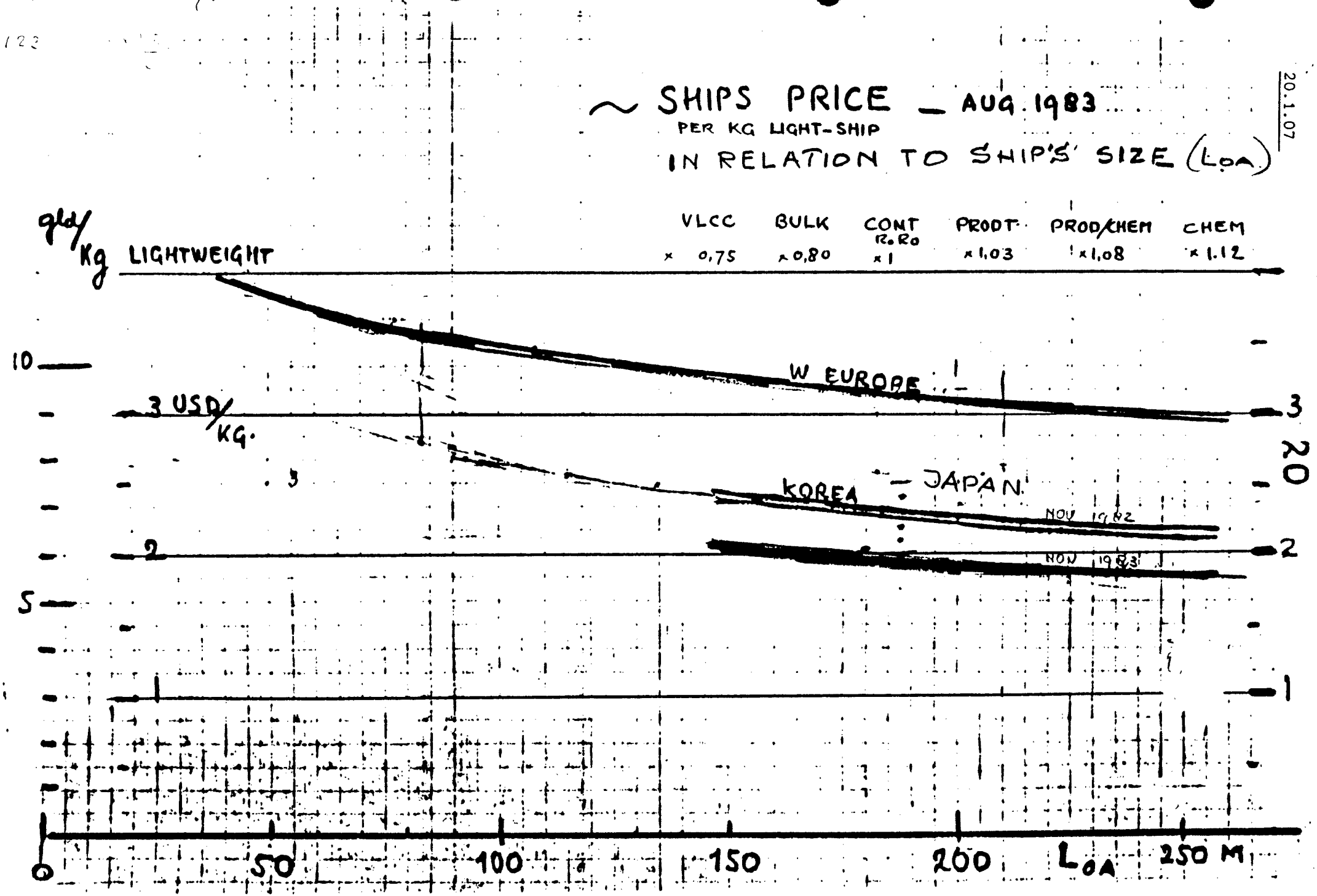
20.1.06

"LIGHT SHIP" WEIGHT

IN RELATION TO

LENGTH (O.A.)




~ SHIPS PRICE _ AUG. 1983
PER KG LIGHT-SHIP
IN RELATION TO SHIP'S SIZE (LOA)
20.1.07
VLCC × 0,75
BULK × 0,80
CONT R.Ro × 1
PRODT × 1,03
PROD/CHEM × 1,08
CHEM × 1.12
gld/Kg LIGHTWEIGHT
3 USD/KG.
W EUROPE
KOREA
JAPAN
NOV 1982
NOV 1983
10
5
0
3
2
1
20
50
100
150
200
LOA
250 M

PRICE OF NEWBUILDING AT DATE OF CONTRACT

KOREA / JAPAN.

LIGHTWEIGHT x MANHR/KG x DFL/MANHR = COST

1 USD = 2,62 GLD.

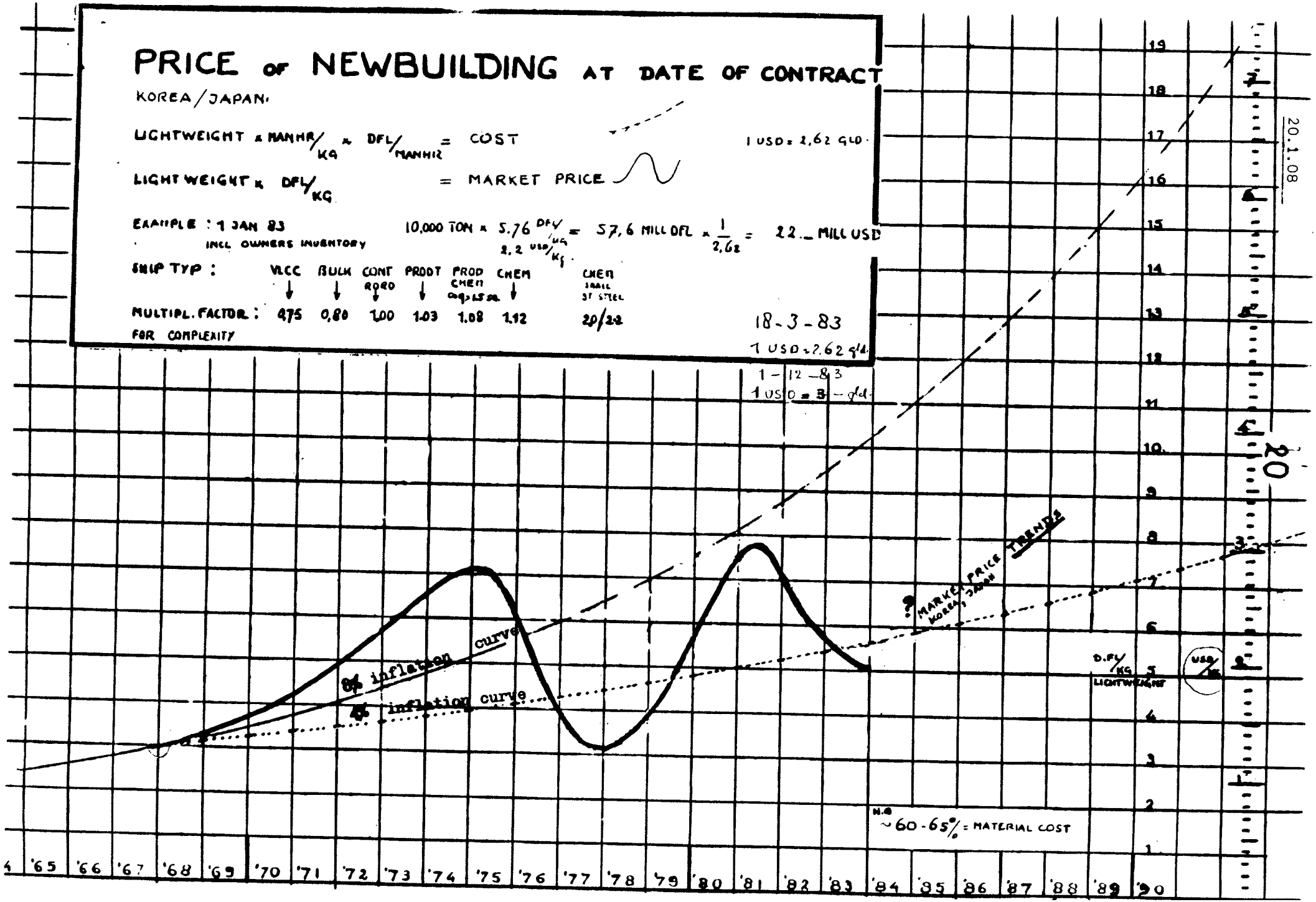
LIGHTWEIGHT x DFL/KG = MARKET PRICE

EXAMPLE : 1 JAN 83 — 10,000 TON x 5,76 DFL/KG = 57,6 MILL DFL x 1/2,62 = 22.– MILL USD

INCL OWNERS INVENTORY — 2,2 USD/KG

SHIP TYP :	VLCC	BULK	CONT ROROD	PRODT	PROD CHEM	CHEM	CHEM SMALL ST STEEL
MULTIPL. FACTOR : FOR COMPLEXITY	0,75	0,80	1,00	1,03	1,08	1,12	2,0/2,2

18-3-83
1 USD = 2,62 gld.
1-12-83
1 USD = 3 – gld.



20.1.01

PRICE AND FUELCONSUMPTION PER MILE PER TEU

LARGE CELLULAR CONTAINERVESSELS

	UNIVERSAL CONTR	S.A. CONTR	FAR EAST CONTR.	PAN CANAL	PAN CANAL
	N. CLARENCE	N HOUTMAN	PANMAX	OVERSIZE	OVERSIZE
	1S	2S ECL REF	2S	2S	1S
	1 x 6 CYL L90	2 x 7 CYL L90	2 x 7 CYL L90	2 x 7 CYL L90	1 x 9 CYL L90
	3 TIERS 2003 TEU				
	4: 221				
	2224 TEU	2775	3238	3892	3886
L_{OA}	210	258	289,5	290,7	290,7
L_{PP}	202	247	277	279.-	279.-
B	32,24	32,26	32,24	→ 35,3	35,3
H	18,8	24,15	24,15	24,35	24,35
T_{SCANTL}	12 m	13 m	13 m	13 m.	13 m.
LBH_{OA}	127300	201000	225500	250000	
K_g/m^3	112	109	109½	106	102½
LIGHT SHIP	14300 T	21873 T	24700 T	26500 T.	25600 T
USD/kg ↓ PRICE	x 2,2 = 31,5	x 2,2 = 48-	x 2,2 = 54,4	x 2,2 = 57,3	x 2,2 = 56,4
1S		2S 1.2	2S 1,2	2S 1.2	1S - 0
BOW + STERN THK 0,7		Bow 2x 0.7	Bow 2x 0,7	Bow thk 2x 0,7	0,7 T
SHAFT GEN 0,5		SHAFT GEN 0,8	SHAFT G. 0,8	SHAFT GEN 0,8	0,5
INTEGRALS 187 P 0,7		— 0,7	— 0,7	— 0,7	0,2
PRICE MILL USD	33,4	51,4	57,8	60,7	58,3
PRICE/TEU	15000	18500	17900	15600	15000
19.- KN		20,5	20,5	20,5	20,5
10,0 M		11,5 M	11,5 M	11,0 M	11,0 M
SERVICE : 1,12 x TRIALS					
↳ 20300 HP		28600 HP	29300 HP	32700 HP	31700 HP
x 125 gr/HPh x 1,1 x 24		IN CASE FITTED WITH MODERN DIESEL ENG			
TON/DAY	61	94,4	96,7	108 -	104,5
TON/1000'	134	192	197	219	213
TON/TEU x 1000'	0.060	0,069	0,061	0,056	0,055

SMALLER CELLULAR CONTAINERVESSELS WITH CRANES

	GOWA	CORONA	USAMBARA	N. v NOORT / v DIEMEN	
	DJAKARTA LLOYD	MARTIN JANSEN	D.O.A.L	NEDLLOYD	
HOLD	490 TEU	556	610	752	
DECK	4T 662 "	4T 726	3T 520 4T 191	3T 662 4T 176	
	1152 TEU	1282 TEU	1271 TEU	1590 TEU	
L_{OA}	170.2 M	166.04	174.00	182,50	
L_{PP}	157.-	152.60	161.44	174.—	
B	28.40	28.50	28.40	30,50	
H	14.00	15,80	15.45	16.20	
T_{SCANTL}	9.68	11.60	11.20.	11.20	
$L_{OA}BH$	67600	74800	76400	90200	
K_g/m^3	105	103	115½	117½.	
LIGHT SHIP	6610 500 7110 T	7210 500 7710 T	8810 T	10600 T	
USD/kg × PRICE	2.30 = 16.4	× 2,30 = 17,8	× 2,27 = 20.—	× 2,23 = 23,6.	
CRANES	2,2	2,4	2,4	2,4	
BOWTHR	0,4	0,4	0,5	Bow + St 0,6.	
SHAFTG.	0,4	0,4	TURBO GEN 0,5	SHAFTG. 0,4	
REEFERPLUGS		100 P 0.4	100 P 0,4	150 P 0,6	
PRICE MILL USD	19.4	21.4	23,7	27,6	MILL
PRICE/TEU	16 800	16700	18600	17300	
	18 KN 9 M SERVICE 1.12 x TRIALS MAN 7 SZ 70/125 BL 130 RPM	18 KN 9 M 106 RPM	18 KN 9 M 106 RPM	18 KN. 9 M. 94 RPM	
TON/DAY (ECL CHG.)	45 —	44 —	42,5	41.5	
SHAFTG. +	3,8	SHAFTG + 3,8	TURBO G + AUXIL + 2,4	+ 3,4.	
	48,8	47,8	44.9	44.9	
TON/1000'	113	111	104.	104.	
TON/TEU × 1000'	0,098	0,087	0,082	0,065	

NOV 1982

CONTAINER RO-RO VESSEL WITH QUARTERRAMP

	BULLAREN TRANSATLANTIC	N. ROTTERDAM NEDLLOYD	N. ROUEN NEDLLOYD	N. ROUEN IF LENGTHENED BY 3 PILLAR SPACES
	~1500 TEUS + LANES	~1600 TEUS + LANES	1735 (TEUS) + LANES	2397 (TEUS) + LANES
L_{OA}	187,6	196.5	212,10	258,9
L_{PP}	171.8	183.2	195,80	245,6
B	32.26	32,24	32.24	32.24
H	19.85	23.25	20,40	20,40
T_{SCANTL}	10,8	10,72	10,72	10,72 ?
LBH	12000	14730	13950	17050
t/m^3	117	117	116	122
LIGHT SHIP	~ 14100	17249	16170	20830
USD/kg ? PRICE	2,2 = 31,0	x 2,2 37,9	x 2,2 = 35,6	x 2,2 = 45,9
RAMP	2,0	2,0	2,0	2,0
BOW + STERN	0,7	0,7	0,7	0,7
SHAFT GEN	0,5	0,5	0,5	0,5
REEFER P. 100 (IN CASE)	0,4	0,4	0,4	0,4
PRICE MILL USD	34,6	41,5	39.2	49.5
PRICE/TEU	23000	26000	22600	20600
	19 KN 10 M SERVICE = 1,12 x TRIALS ~17300 x 1,12 19000 x 125 g/BHE x 1.1 x 24 DO → HFO	19 KN 10 M 17300 x 1,12 = 19000	19 KN 10 M 17300 x 1.12 = 19000	19 KN 10 M. ~21000 x 1,12 = 23500
TON/DAY	62,7 + 4,3 = 67.0	62,7 + 4,3 = 67,0	62,7 + 4,3 = 67.0	~~77,7 + 4,3 = 82,0~~
TON/1000'	147	147	147	180
TON/TEU x 1000'	0,098	0,092	0,085	0,075

ESTIM. 20 **NOV 1983**

20.1.03 A

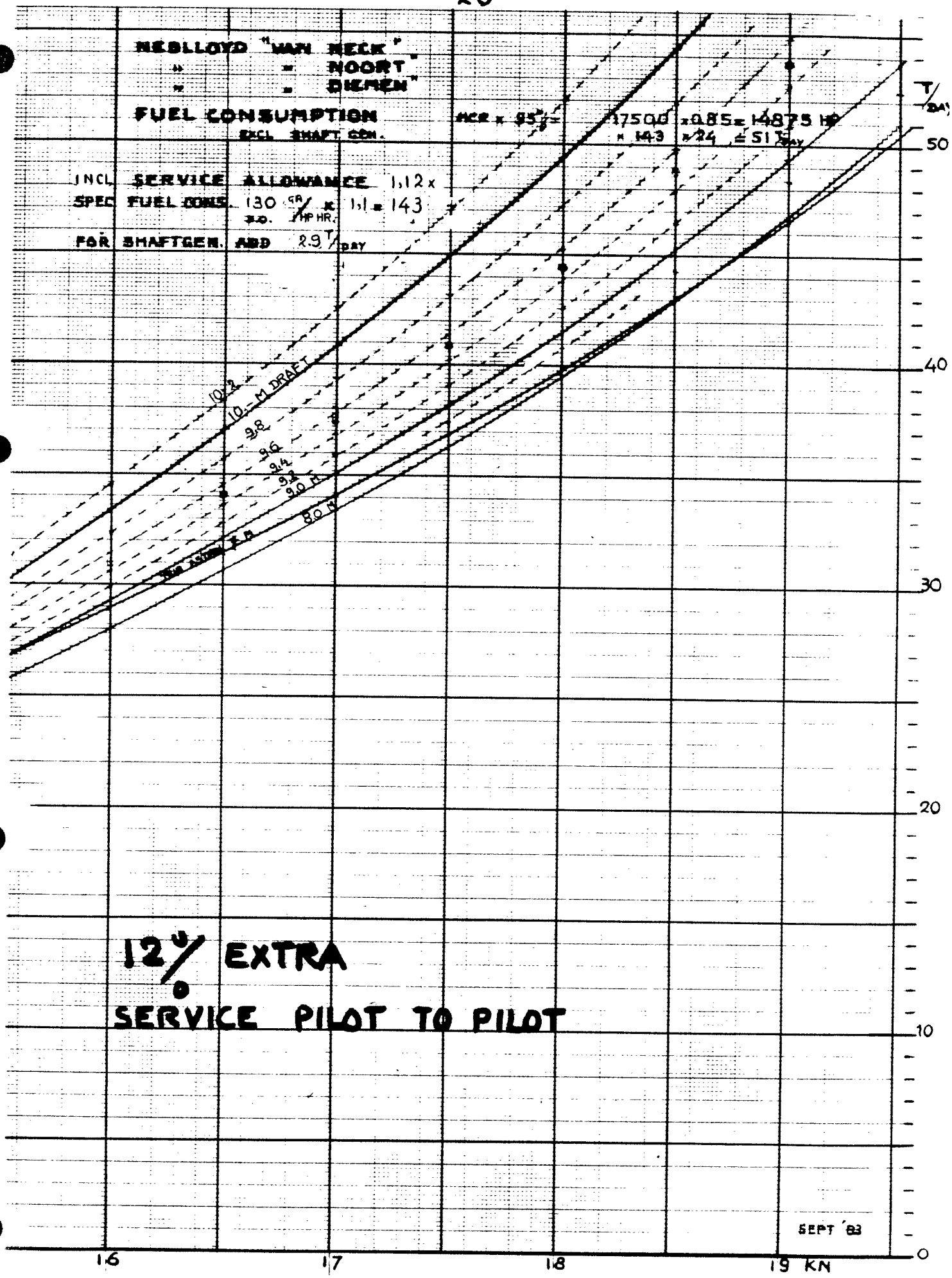
CONTAINER **RO-RO** VESSEL.

	BULLAREN NEWBUILD	N. ROTTERD. NEWBUILD	N. ROUEN NEWBUILD.	N. ROUEN LONGER VERSION NEWBUILDING	N. ROUEN ORIG. LENGTHENED.
TEU	1500 + LANES ROLLING ST	1600 + LANES ROLLING ST	1735 + LANES ROLLING ST	2397 LANES ON ST	2397 + LANES ON STOCK.
LOA	187,6	196,5	212,10	258.9	258.9.
B	32,26	32,24	32,24	32,24.	
H.	19.85	23.25	20. 40	20.40	
HT S. TON.	14100	17249	16170	ESTIM. 20830	
USD/kg EXTRA	x 1.93 = 27.2	1.92 = 33.1	1.9 = 30,7	1,85 = 38 6	
	RAMP = 2.0	2,0	2,0	2,0	
	BOW+STERN = 0.7	0,7	0,7	0,7	
	SHAFT GEN = 0,5	0,5	0,5	0,5	
	REEFER P. (100) = 0,4	0,4	94	0,4.	
	PRICE MILL USD = 30,8	36.7	34,3	42,2	ROUEN CONVERTE. YEN BILL 34,3
	MILL DEL. (92,4)	(110,1)	(102.9)	(126 6)	1.43/NKK x 4.2 = 6 — 40,3 MILL US
USD/TEU.	30,8/1500 = 20,600	36,7/1600 = 22.900	34,3/1735 = 19.800	42,2/2397 = 17.600	40,3/2397 = 16.800.
19 kn TON/DAY	10 M. DRAFT FUEL CONSUMP.		ME 62.7 SHAFTG. 4.3 67,0	ME 72,7 SHAFTG. 4,3 77,0	
ie FTA	CAPITAL CHARGES.		USD/DAY. 10.000	12.300.	
	MANNING.		3 300	3 300.	
	BUNKERS. 19'-67 TON.		9.500	10,900	
	INSURANCE		700	860	
	LUB.		250.	290	
	ELEC.		150	150	
			23.900	27,800	
	MILE DAY = 19 x 24 = 456'		1/456' x 1735 =	1/456 x 2397 =	
	COSTS P. TEU P. RUNNING MILE (EXCL PORT DUES)		USD/TEU/MILE = 0,0302 3ct	0,0254 2½ct.	

BV

SMALLER RO-RO CONTAINER FOR LIMITED WATERDEPTH AND PROTECTED SEA

	SLOMAN RANGER	TACKLER	BOXER	CONTENDER.
	2S	7S	7S	2S.
				HOLD 500
				3TRS 594
				4S 198
	319 TEU	378 TEU	576 TEU 3TRS	1292 TEU
L_{OA}	98.1 M	114,8	133.5	173.0
L_{PP}	79.2	102,0	122,6	160.0
B	18,0	19,5	25,0	30,4
H	7,9	10,6	12,1	17,0
T_{SCANTL}	3.65.	5,25.	6,5.	8,2.
LBH OA K_{t/m^3}	à 154	à 120	à 115	à 110.
LIGHT SHIP	2010	2840	4640	9835 T.
USD/kg PRICE	2,7 = 5,4	x 2,6. = 7,4.	x 2,5 = 11,6	x 2,27 = 22,3
2 CRANES + RAMP	1,6	1 GANTRY + RAMP 1,0	2 GANTRIES + RAMP 2,0	2 GANTRIES + RAMP 3,0
TWIN SCREW	0,6			TWIN 2,3.
BOWTHR.	0,2	BOW TH 0,2	BOW TH 0,3	BOW TH. 0,6
REEFER P	0,2	50 P 0,2	75P 0,3	100P 0,4.
PRICE MILL USD	8,0	8,8	14,2	28,4.
PRICE/TEU	25000	23.200	24 700	28 000
	12,5 KN 3,65 M.	15 KN HIGH. 5,25 M.	16½ KN 6,5 M.	18 KN 8,2 M.
TON/DAY	10,4	27	40	58
TON/1000'	35	75	101	134
TON/TEU x 1000'	0,110	0.198	0,175	0,104

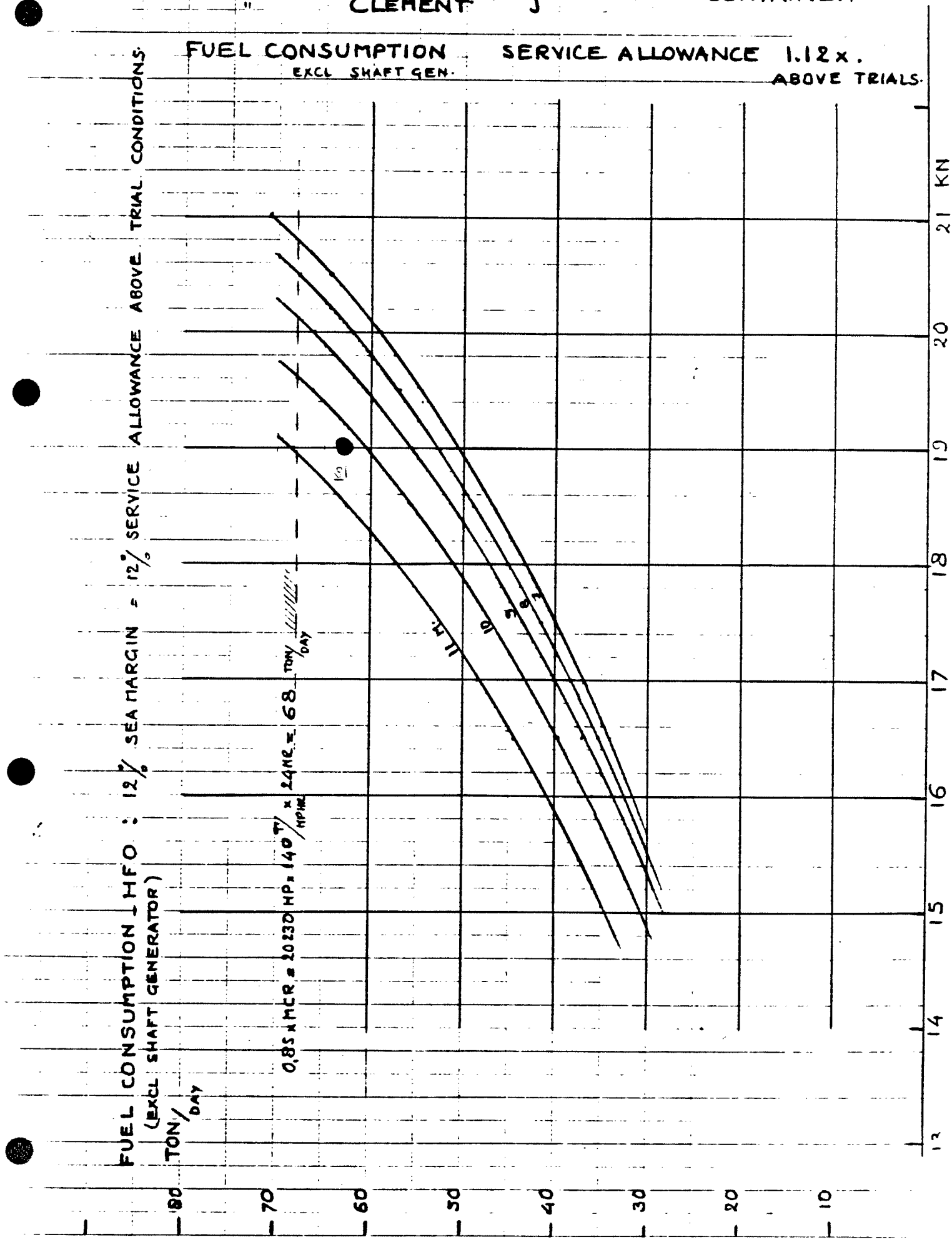


12% EXTRA

SERVICE PILOT TO PILOT

"NEDLLOYD" "CLARENCE" } = UNIVERSAL CONTAINER
"" "CLEMENT" "

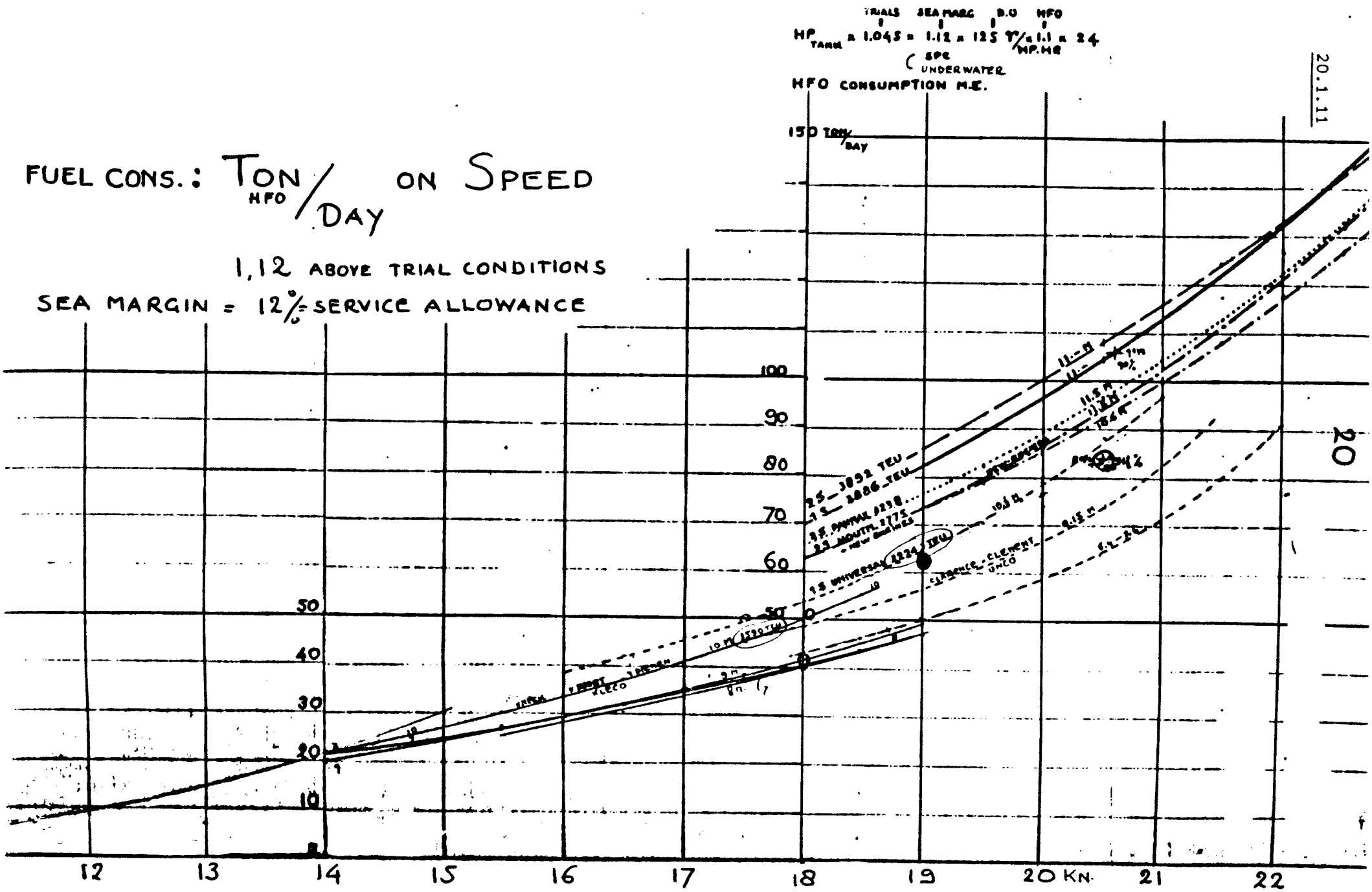
FUEL CONSUMPTION EXCL SHAFT GEN. SERVICE ALLOWANCE 1.12 x. ABOVE TRIALS.



FUEL CONS.: TON HFO / DAY ON SPEED

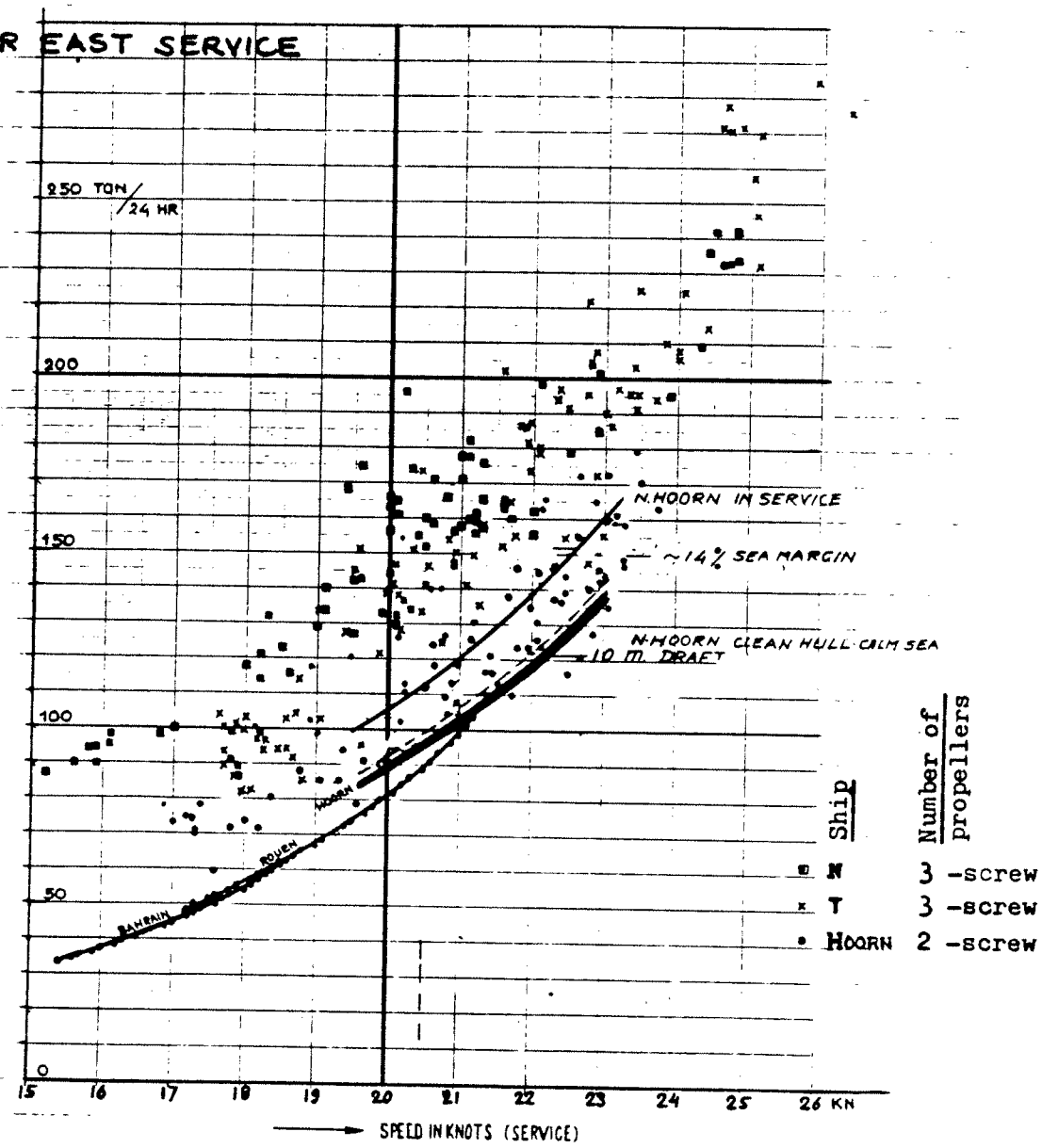
1,12 ABOVE TRIAL CONDITIONS

SEA MARGIN = 12% = SERVICE ALLOWANCE



COMPARATIVE FUEL CONSUMPTION FIGURES OF SELECTED CONTAINERVESSELS UNDER VARYING PROPULSIVE CONFIGURATIONS .

EUROPE - FAR EAST SERVICE



DAILY FUEL CONSUMPTION IN TON/24 HR

		3 S-DIESEL (CENTRE PROP + C.P.)				2 S-STEAM				2 S-DIESEL (SHORTER VESSELS)			
		< SISTERS >				DELFT DEJIMA < SISTERS >		KOREKUNI		HOORN HOUTMAN < SISTERS >			
	Speed	J	SL	T	N	DL	DM	K	aver.	H	P	S	aver.
	23	173	180	185	202	251	261	300		160	185	185	
	22,5	161	169	172	188	239	247	286		149	174	175	
	22	151	159	160	176	227	235	273		139	164	165	
	21,5	142	149	149	165	216	223	260		129	154	156	
	21	134	139	138	155	205	212	248		120	145	147	
SERVICE :	20,5 Kn	124	130	128	146	196	202	237		112	136	139	
	20	117	121	118	137	187	192	226		105	128	131	
		B&W		MITSUI	GÖTA	DR. VULKAN		HDW		RSV	SUMITOMO		
		B+W ENG			GÖTA	STAL LAVAL TURB.		AAE TURB.		SULZ	SULZ + TURBO-CON		
	Const. cons. /day	10 Ts DO				in port: 80 Ts FO -80				10 Ts DO prt: 7 Ts FO	At sea: 2,5 Ts DO in port: 10 Ts DO		

TRIPLE-SCREW : TWIN-SCREW ..... STEAM V.A.V. DIESEL

Performance "Nedlloyd Delft" als motorschip

Hieronder volgen uitsluitend de trajecten Jedda/Port Said-Singapore via versa.

Datum	---) (---	Displ. TON	Speed V	F.O. TON	Prest.getal $\frac{DISPL^{2/3} \cdot V^3}{TON/DAY}$	F.O. als turbine	Besparing
28-1-82	---)	53937	21,8	135	110	210	75
26-2-82	(---	55118	21,7	116	128	210	94
6-4-82	---)	62746	21,6	134	119	222	88
7-5-82	(---	59646	21,1	121	119	205	84
17-6-82	---)	55856	21,2	113	123	203	90
16-7-82	(---	56742	21,0	128	107	202	74
25-8-82	---)	57480	20,5	106	121	193	87
24-9-82	(---	59843	21,4	125	120	214	89
3-11-82	---)	60000	22,1	150	110	229	79
2-12-82	(---	58000	21,6	125	121	215	90
14-1-83	---)	68000	22,2	167	109	247	80
10-2-83	(---	60100	22,5	144	121	238	94
23-3-83	---)	67100	20,9	125	121	213	88
22-4-83	(---	60300	20,7	113	120	198	85
1-6-83	---)	68200	20,3	127	110	200	73
30-6-83	(---	56990	22,4	166	100	228	62
Gemiddeld		**60000**	**21,4**	**131**	**115**	**214**	**83**

FTA

1-9-83

Performance "Nedlloyd Dejima" als motorschip

Hieronder volgen uitsluitend de trajecten Jedda/Port Said-Singapore via versa.

Datum	---) (---	Displ. TON	Speed V	F.O. TON/DAY	Prest.getal $\frac{DISPL^{2/3} . V^3}{TON/DAY}$	F.O. als turbine	Besparing
21-5-82	(---	60827	21,0	115	125	205	90
29-6-82	---)	62992	22,4	144	124	242	98
29-7-82	(---	60600	21,8	147	109	223	76
08-9-82	---)	55413	20,8	119	110	195	76
7-10-82	(---	56397	21,9	126	122	220	94
17-11-82	---)	58464	20,8	113	119	199	86
7-12-82	(---	55118	21,0	105	127	198	93
26-1-83	---)	51870	20,0	112	99	177	65
24-2-83	(---	63090	22,3	138	127	240	102
6-4-83	---)	65000	21,6	146	112	226	80
5-5-83	(---	63693	22,7	154	121	253	99
15-6-83	---)	65800	21,2	123	126	217	94
14-7-83	(---	62500	22,3	152	116	239	87
Gemiddeld		**60136**	**21,5**	**130**	**117**	**218**	**88**

FTA

1-9-83

PHYSICAL ASPECTS OF THE LINER VESSEL

AND IMPROVEMENTS IN ITS ECONOMY.

21.0

ECONOMY OF VESSEL ITSELF

IMPROVEMENTS IN ECONOMY OF THE LINER VESSEL

Modern trends in cargo liner ship operation call for changes in ship design in various aspects, almost all aiming towards improving their economy. Differently from ten years ago, changes are aiming mainly towards savings in propulsion fuel expenses. This is because where fuel costs in the late sixties were just 'part of total' costs, they nowadays are dominant.
Therefore emphasis in changes is laid very much on improving the

SPEED versus FUEL CONSUMPTION.

Consequently, whilst adding improvements in other fields as well, we arrive at a number of changes in design, as follows:-

- Slow-turning large-diameter propeller(s).
- Slow-turning long-stroke diesel engine main mover with a minimum of cylinders.
- Continuous optimalisation of hull form, especially at stern - and bow.
- Excellent surface preparation and painting.
- "Self-eroding" underwater antifouling giving minimum friction resistance.
- Shaft generator.
- Thrusters at bow and stern - for ease, safety and rapidity of manoeuvring around of the (relatively) long ships in confined spaces.
- 4 to 5 mooring winches fore and aft.
- Utmost simpicity in lay-out and in construction.
- Minimum number of crewmembers.
- Sophisticated and reliable equipment for control and communication.
- Reefer containers to be of integral type if their number is relatively low - say below (about) 400 TEU - per vessel.

PHYSICAL ASPECTS OF THE LINER SHIP.

SPEED

- As soon as requirements in "Transit time" and "Regular Call" allow for it, the ship's speed should be lowered because of propelling fuel costs being appreciably high.
- Speed - in case of non-perishable cargoes - is often a matter of competition. Lower speed needs less than proportional horsepower and therefore saves fuel per mile.

- A service speed lowered to around 19 knots, or even 17 knots on certain stretches, from the preceding years' high level seems reasonable and has hardly any influence on the transit time.
- However, a regular call frequency - of say, every 7 days - should be maintained in the schedule of the operating vessels.

STABILITY

- Maximum breadth - up to Panama-beam (32.3 m/106 feet)
- Pram-type hull with V-shaped cross sections offers great stability which allows the operator to "make the most" of space and weight above deck. Fig. 15.
- Depth to be decided in view of ship's weight, strength ($\frac{L}{H}$) and stability($\frac{B}{H}$)

- The cheaper vessel will tend to a small depth, which ends up with higher lashing costs for the owner if stability allows for "4-tiers high" The right depth should incorporate 3 tiers maximum.

COURSE KEEPING.

- The coefficient $\frac{Loa}{B}$ can be reduced to 6.1 without sacrificing course keeping ability if the skeg-aft and rudder present sufficient surface.
- The block-coefficient can have a deteriorating effect on course keeping capability if increased too much (we refer to next page), especially in lightly-draughted conditions during bad weather.
  (Note. This problem will be the more serious when adapting a slow-turning large-diameter propeller system so as is being promoted for bulkers and tankers in general for the sake of fuel economy.)

B L O C K .

- According to the old law of naval architecture, the block coefficient can be increased when speed/length ratio ($\frac{V}{\sqrt{L}}$) of the ship is decreased. However, a bigger block results in a higher fuel consumption and too big a block slows the vessel down in bad weather with headseas. Keeping up to schedule will become more difficult. It is a matter of weighing fuel costs against revenue - and because cargo weight on deck is now an important part part of Deadweight, the spacious hull therefore should be kept rather fine in order to raise the metacentre.
- A 'big-block' vessel - like a bulk carrier - will float like a foambubble on the surface in a loading condition with light to medium weight boxes and the ship will be difficult to handle in bad weather from ahead which may cause slamming. Loaded with lightweight boxes this ship will be rather stable and inclined to roll heavily.
Propeller immersion might then be insufficient for good manoeuvring and for preventing cavitation as well.
Extra weight below by means of waterballasting could better this poor situation but although this would make the ship more seaworthy, she in fact would - in the long run - be too large and too expensive for the cargo she carries.
- If loaded with heavy and medium weight boxes at full draft, the big-block vessel will lack stability and a large amount of waterballast is necessary to keep her upright.
Waterballast means extra fuel consumption. This should be avoided.
- The finer wider V-form with soft bilges will float deeper and has a better stability margin than the narrow U-form with hard bilges.
In service, the V-Pram form will need less power and is more seaworhty.

However, if a vessel has a fine block coefficient and she has to carry cargo from a port with draft limitation, her Cargo Deadweight cannot be utilized in full at that port.

SMOOTHNESS OF HULL - (FRICTIONAL RESISTANCE).

- Improving smoothness is a most promising way to save on fuel.
- Frictional resistance is 70-75% of the total resistance of the vessel ! People have not always been sufficiently aware of the magnitude and importance of this figure.
- The treatment of BLASTING + ZINCEPOXY + 250 MICRON CHLORINATED RUBBER AC + 100 MICRON A.F.

results in an excellent smooth surface which can be mended and recoated easily at drydocking, every 18 to 24 months.(Roughness = 100 mu (90-120))

Note. From recent experience it appeared that in this respect the so-called SPC-type of coating is even better in smoothness performance.

- Container and roro vessels will hardly suffer from fouling because of their high speed and short port stays. It is therefore that the contact-type anti fouling gives satisfactory results. The latest development of the 'self-polishing' acrilic-polymer antifouling spends its poison (tributyl tin oxide) during the seatrip and arrives in port with a fresh poisonous surface. This self-polishing kind a.f. should in first instance be applied to ships with longer portstays (e.g. multi-purpose type ship) but it has become clear that even ships staying in port not longer than one day are able to save on fuel by the application of the SPC-type of anti fouling paints. The reason why is that algae-fouling on the ship's sides is prevented by the sunlight.

- It is very important to have keen supervisors in the drydock during cleaning and painting. Poor attention will end up in paint being sprayed over weeds and mud. The way of cleaning and paint spraying can strongly influence the surface roughness. From tests it has been found that the first extra 100 microns roughness will cause the larger percentage increase in resistance (5-7%) which cannot be remedied.

  In the newbuilding stage too the utmost attention is to be paid to the smoothness below water surface... butts are to be ground level to the plate surface. The shipyard has to accept this.

- Zinc blocks / b ilge keels / openings of thrusters / fins / E.R. intakes and propeller lifting eyes are all to be well arranged according the streamline flow along the hull.

Influence of hull-roughness on power and speed.

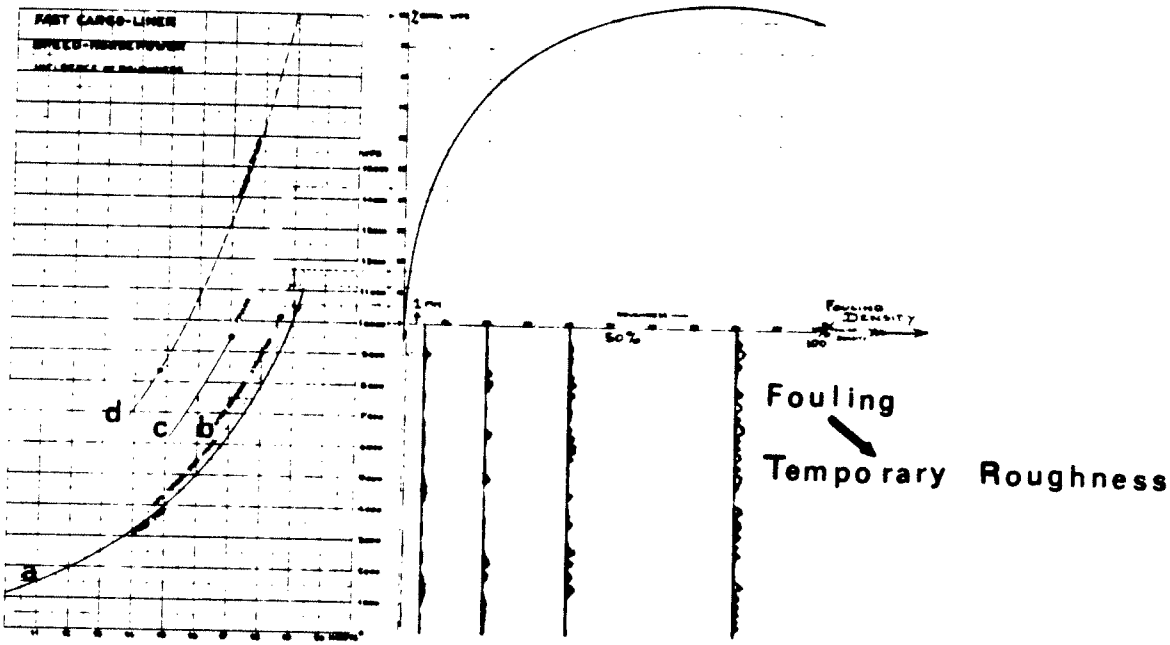
Diagram shows the speed/hp curve of a fast cargo liner under various conditions:

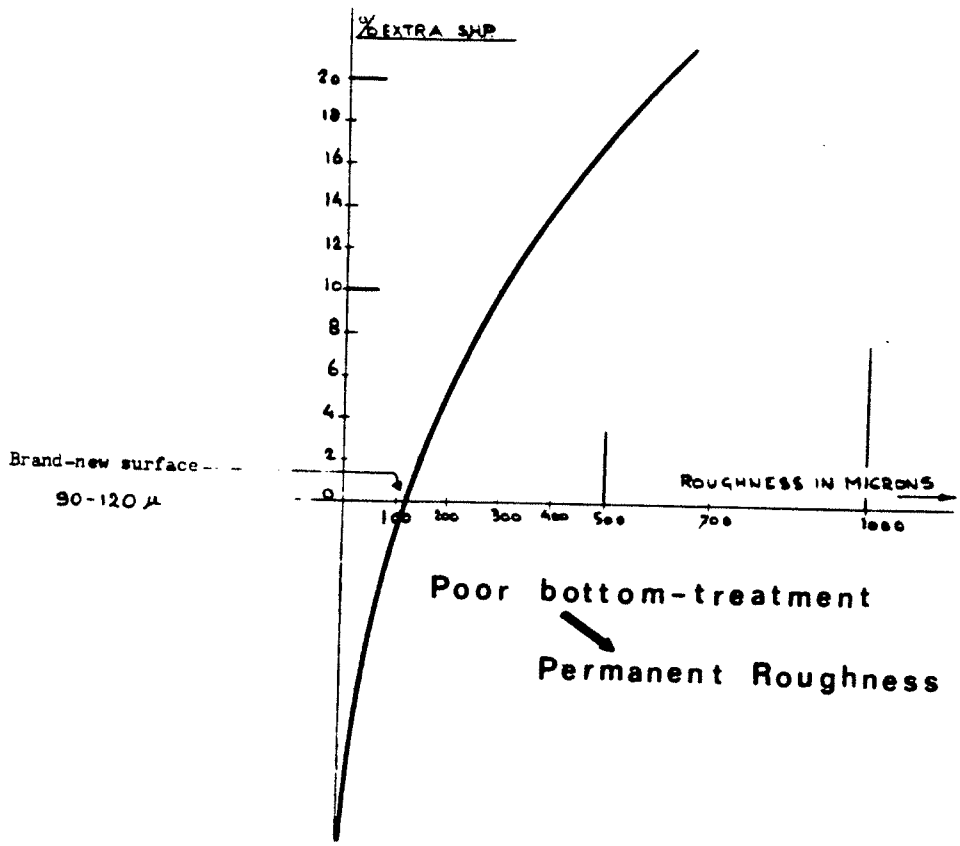
a) Brand new on first trip (SHP often below tank testing prognosis) ROUGHNESS : 30 - 120 MICRON

b) 3-5years old, after leaving drydock – freshly painted: 12-17% more SHP (½ knot less speed, if torque of diesel engine is maintained) CONVENTIONAL SYSTEM

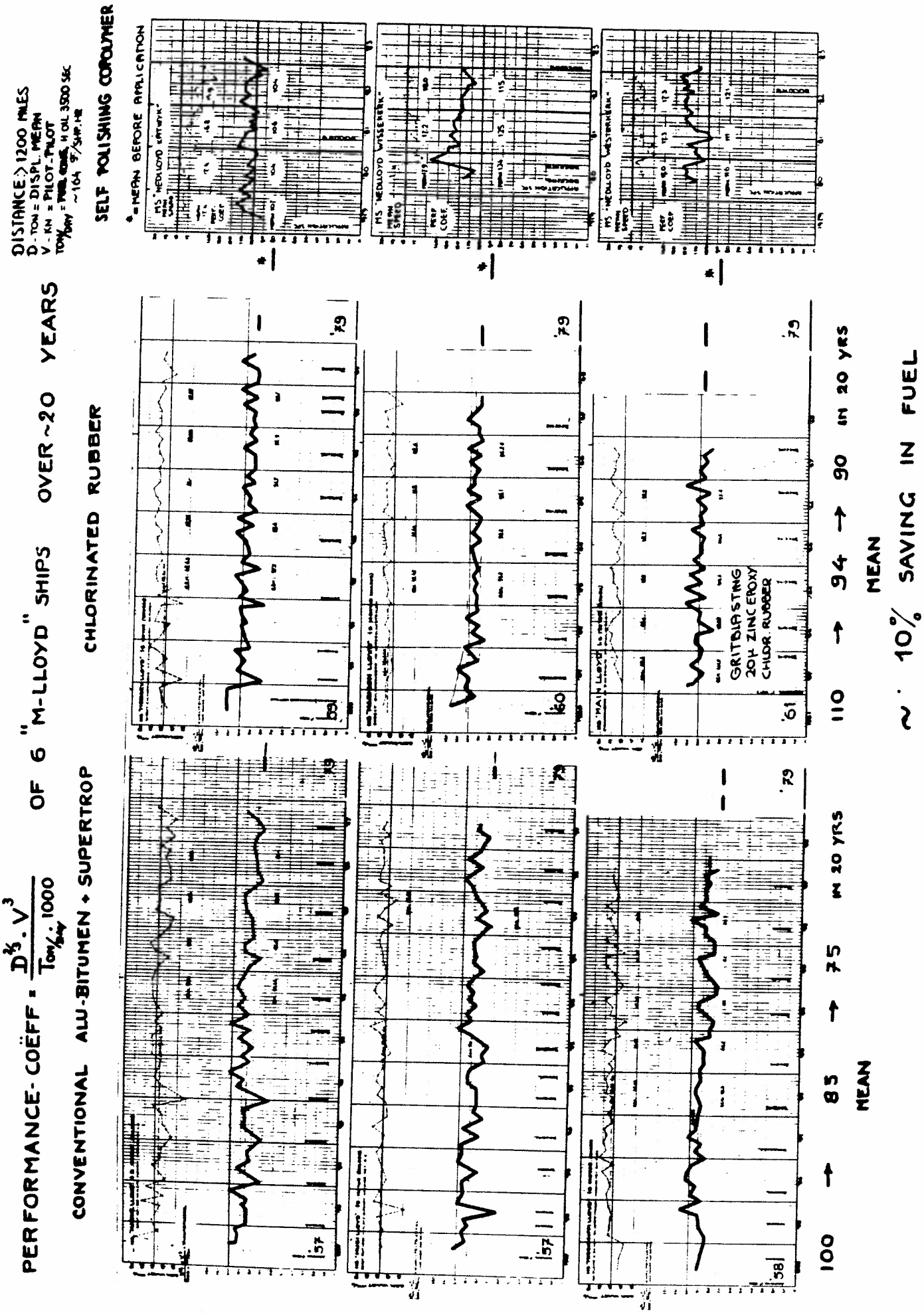
c) With 5% fouling density of barnacles: 40-45% more SHP (2 knots less speed if torque of diesel engine is maintained) (EXH TEMP)

d) With 50% fouling density of barnacles: 90-100% more SHP (4 knots less speed if torque of diesel engine is maintained).





INFLUENCE OF SHELL EXTERIOR (ANTIFOULING PAINT) CONDITION ON A SHIP'S SPEED PERFORMANCE.



HULL FORM (WAVE AND FORM RESISTANCE)

- Ship's body lines should be basin-tested and retested up till the optimum form has been attained (30%-25% of the total resistance consists of wave + form resistance).
- Normally the shipyard's Standard Vessel will be the shortest/fullest possible vessel (cheapest - per ton deadweight offered). Designed for the required deadweight at required trial speed. Afterwards the fuel consumption turns out to be high - and stern-vibration might occur.
- For obtaining maximum speed at full draft, some shipyards will propose a very big bulb, being advised so by certain modelbasins. Afterwards, the ship in service has to cleave the waves at medium draft with "big" bulb half immersed, accompanied by wild rotating vortexes, and the owner will be confronted with a high fuel consumption per mile.
- Normally, testing is stopped by the shipyard at the very moment that contract requirements at full load and ballast are fulfilled - and further research is forsaken in order to save money.
- Therefore, actual service conditions are to be given by owner and tanktests continued until finally a complete net of Horsepower/ Speed/Consumption curves under a variety of draft and trim conditions has been analized. Fig. 5.
  Many large containervessels with bulbous bow and transom stern save 3-6% fuel when sailing with 1 metre trim-in-the-head in comparison with Even Keel condition - which clearly shows that tanktesting has sense.
- Because of the fact that a wide aperture is asking ship's length - which means money ! - many Standard Vessels have small apertures. A small aperture especially behind the Standard Vessels' often blunt afterbody might excite heavy propeller cavitation and the ensueing hull pressure fluctuations will cause a vibrating afterbody. Only a few good shipyards design a well-bred afterbody!

- Therefore, it would be wise to optimalize the afterbody of the single-screw ship by carrying out extensive testing in deepwater tank, cavitation tunnel and vacuumtank analysing flow visualisation and wake, cavitation and hull pressures. Fig. 13.

- Research costs should be shared between Owner and Yard and Government-sponsored/ subsidized research programs are to be interconnected and co-ordinated.
- Technical failures in this respect are almost non-correctable and of no use to anybody. In too many cases research money is spent after the "fiasco" ("After the steed is stolen, the stable door is locked")
- It is preferable to locate the superstructure not "all-aft" over the propeller, but to move it forward from aft by ¼L in order to create better living conditions for the crew and also to save vibration-sensitive instruments. For containervessels this would mean sacrificing a little of the earning capacity of the design. However, for low speed multipurpose vessels it is more economical to keep the engineroom and accommodation "all aft".

R U D D E R

- A large semi-balance rudder of 2,2 - 3% percent ship's length will give master and pilot confidence in the ever-obeying vessel, even under conditions of manoeuvring in confined water with current and strong wind during low-speed sailing,
  Waiting outside harbour, because of entering in rough weather might be too risky, should occur rarely only.
- Such confidence will not be built up by the arrangement of a scull-type semi-balance rudder of 1,7 percent ship's length, working in the wake behind a full-bodied Standard Vessel.

T H R U S T E R S

- Master and pilot should be given the feasibility to manoeuvre the ship alongside and from the quay without tug assistance under normal wind and water conditions, not so much only for saving costs for tug-assistance and ship's timesaving, but also - and to a great extent- the independency from tugs in case of strike. It has become clear that installing bowthrust and sternthrust of adequate power is a paying proposition. Controllable-pitch propellers, coupled to 6000-Volts motors started directly via trafo's are to be preferred above low-voltage installations with switchgear in the high-Amp-circuit.

- Capacities: - Ship's length 185 M .... 2 x 1,000 KW
  If " " 250 M .... 2 x 1,200 KW
- The electrical power required when manoeuvring the ship is to be supplied by three diesel generators of 1,200 - 1,500 KW each, a fourth generator may possibly be in overhaul condition.

- Container vessels in a regular service normally may carry 100 to 200 integral reefer containers, asking 700-1,400 KW.
  This power can be supplied by one or 2 diesel generators. During manoeuvring and Canal passages the electric current supply to those reefers may be shut off during a short period.
- Openings of bowthruster and of sternthruster tunnels are to be carefully streamlined.-

  At the BOWthruster, poor shaping may cost some loss of speed, alternatively 1% to 3% extra fuel. At the STERN-thruster, poor shaping may cause cavitation-erosion to the main propeller and might excite extra vibration and noise. However, speed loss is not to be expected here.

F I N - S T A B I L I Z E R S

- These will cost fuel if used too often. Some guidance to masters as to the use of the fins is that fins should be "out" if the ship's angle of roll gets 8 degrees and over, where-as fins can again be taken "in" after conditions have improved so that the fin-angle does not exceed the 15 degrees' limit.
- Nautical staff is very much in favour of fins, mainly with a view to avoiding cargo damage.
- Technical superintendents do not like fins, as these bring about extra work and costs when ship is in drydock.
- Shipowners have to weigh the extra costs of fin installation against safer container depatch (cargo care) which often is a difficult choice
- Less rolling - and consequently a more "confident" ship's staff - may result in less lashing ... but fins might not work sometimes ! (low speed)
- On roro cargo carrying vessels it is wise to have and use fins as lashings sometimes may be doubtful and metacentric height may vary over a wide range, usually between 0.3 and 3.0 M.
- In case of pure cellular containervessels sailing with metacentric heights between 0.3 and 0.9 M, the fins are less urgent, but useful.
- Also here, the recess-openings in the shell at the location of the fins should have well-streamlined edges, in order to keep extra drag to a minimum. From this viewpoint, fins with self-closing flaps are preferable.

P R O P E L L E R

- The most simple way to increase propulsive efficiency is to adopt a slow(er) turning propeller (with large(r) diameter) where applicable.
- This now has been made possible by the development of the slow-turning long-stroke diesel engine (down to 70 r.p.m.).
- Lowering revolutions from 122 to some 90, enlarging propeller diameter by 11 percent is claimed to offer an increase in efficiency by 6 % to 7 %. When reducing further to about 70 rpm then more profit is attainable even.
- Of course it is possible to install a gearbox instead, coupled to medium speed engine(s) but 'we' consider this solution not yet reliable, with an eye to worsening of the quality of fuel oil. Also, price difference between 1,500 sec and 3,500 sec fuel oils, together with the questionable future availability of low viscosity oils on worldwide basis refrain us from adoption of medium speed diesels.
- It is difficult to accommodate the larger "wheel" underneath the transom stern while simultaneously maintaining sufficient tip clearance. Steeper buttocks of the afterbody will cause slowing-down the water flow with an increase in risk of separation below the stern. It is clear that a flat stern over a cavitating propeller will transfer the propeller-induced pressure fluctuations more intensely into the structure than a V-shaped stern.
- Because of the difficulties mentioned before, it would be wise to design the lower hull ahead of the propeller as slender as feasible in order to minimize wake-peak and to try and maximize the inflow into the uppper part of the propeller disc as much as possible.
- More homogeneous inflow, in combination with the lower tip speed of the slow-turning propeller, will decrease the cavitation and accessory vibrations.
- Under these circumstances the tip clearance might be chosen to be somewhat less in relation to the necessary tip clearance of the fast-turning propeller.

(Note. In an endeavour to minimize propeller-induced vibration, we at the moment are studying the application of a shock-absorbing layer applied to the hull above the propeller.)

- The problem of propeller immersion with large "wheels" is even more difficult on (product) tankers and bulkers. In the case of the product tankers, tip immersion is a 'must' during ballast voyages .

Extra waterballast capacity should be accommodated which then makes the vessel larger (more depth) and somewhat more expensive. Meanwhile, the seaworthyness is improved also.

BRIDGE - WHEELHOUSE

- Free cross-over passage near front windows.
- Simple concentrated manoeuvring console in centre. Two identical consoles on bridgewings (1PS, 1 SB).
- Daylight radar screens, Anti-collision radar - near centre (SB).
- Time / Course / RPM / Speed / Torque / Horsepower / Wind / Depth / etc., recorders all to be visible from everywhere.
- Television display of engineroom.
- Alarms and lights of secondary importance - to be placed in the afterpart of the wheelhouse so as not to disturb the officer on watch unduly.
- Podium with working desk and seat allowing work to be done during watch.

CREW.

- Integrated shipboard organisation is - a.o. - made possible by rationalized systems. Crew complement and crew qualification therefore are to be closer to the standards of the air transport's. Repair squads and specialists can temperarily board for the necessary upkeep during coastal runs.
- Integrated complement estimated to be 18 on cellular containership.
  22 on roro-containership.
- Number of cabins to be about 28, in order to accommodate extra persons on coatal trips.
- Lifeboat capacity for 40 persons.
- Each cabin to have sanitary unit and separated sleeping department.
- Galley, messrooms, recreation rooms to be located on one deck.
- Provisions ... to be loaded as units in an (open-top) container by terminal crane and to be relocated by shipboard crane via a trunk to provision room.
- Swimming pool / sports room / sauna ... to be accommodated in the uppertween deck space.

D E C K L A Y - O U T is to be as simple as possible in every detail

- The rule for the naval architect should here be :
  "What can be omitted should be omitted !", and therefore,
- Generally ...
  - The least possible number of movable parts.
  - Closed-box constructions for easy maintenance.
  - alternatively: wide-open structures - for proper accessibility.
- Hatch covers ... the least possible amount of hand-operated fittings.
- Guides/chocks ... of robust construction and well-guiding.
- Mooring ... - All wires/ropes on drums.
  - Winches : 4 forward and 4 aft / Constant tension / steel wire with fore runner (altern.synthetic fibre)
  - Remote control from ship's bulwark.

  (Note: - Rope-handling by means of warping-heads as well as fastening around bitts should be methods of the past.
  - However, some bitts to be kept for Panama passage and emergency towing/fastening or coupled mooring.)
- Fairleads ... of enlarged Panama-type without moving parts and with soft bends.
  - Airguns, for shooting heaving-line ashore, especially when berthing by means of thrusters only, 1 gun fwd, and 1 gun aft.
- Anchors ... To be stowed in recessed pockets. Winches for both of the anchors to be separate from C.T.winches in order to attain good utilisation of the mooring deck space, (which is often quite narrow) and to allow application of one winch type.
  - Chainlockers to have sufficient height to be self-stowing.

  (Note: - Simplicity in operation allows for a quicker action in case of danger/emergency. Cable indicator is useful.
- Pilot door in ship's sides Port & Starboard to allow for safe embarkation of pilot via a short-length normal pilot ladder.

(Note: - No sophisticated electric ladders, crawlers and that kind of gear!)

- Accommodation ladder ... of simple swing-out rigid-bridge construction is to be preferred.
  Telescopic outreach.

CORROSION PREVENTION

(A) Weather-exposed surfaced,

(B) Waterballast tanks.

Weather-exposed surfaces.

- After construction is finished, all weather-exposed surfaces are to be blasted and immediately thereafter sprayed with zinc-epoxy.
  For the shipowner, this treatment is the sole and only way to get real corrosion prevention. Steel surface preparation is even more important than paint quality !
  For the yard, this requirement usually creates a big problem because of pollution by the grit used in the blasting process.
  The problem could be solved by completely blasting & coating of the blocks in the blasting shed, followed by local blasting of connection welds and local damages during the fitting-out stage.

  (Note: - Standard Vessels lack such treatment, or - at least - we can surely assume that for the usual "Standard Vessel" this treatment is "a far cry off". Consequently, within one year the rust will appear through the paint coat.)

Waterballast tanks.

- One of the most important items in the building process is the way of preparation of the inner surfaces of the w.b. tanks, the application of the coatings requiring the utmost attention of the inspectors. Should corrosion-prevention fail after a few years, the vessel cannot be saved and very expensive steel replacements will be necessary at the 3rd and/or 4th Survey.
- Big failures have teached owners bitter lessons and made us assume that the most safe coating is application of the modified coaltar products in a two-coat coating or three-coat coating system, derived from the "old-fashioned" coaltar pitch method.
  These coatings are always repairable because of the one-pack system

which attaches well to the underlayer - in contrast to the chemically two-pack epoxy-coaltar system.

- Shipyard-offered "Standard Vessels" normally have their w.b. tanks coated by a single layer of coaltar-epoxy which will normally be penetrated by pinpoint holidays from which corrosion starts. Deterioration of the coating breaking up gradually faster is to be expected. The coaltar is NOT repairable - unless blasting is applied - which is difficult in an extended double bottom structure.
- In this respect there is a development during the last 4 years:- After finishing welding of the complete w.b.bottom, welds are blasted and surfaces sweep-blasted (a temporary hole is made in the bottom plating of the vessel for removal of the grit). Afterwards, solvent<u>free</u> epoxy is applied in one coat of 350 microns thickness. No pinholes, no explosion hazard.

DIFFERENCE IN THE PROCEDURES OF ORDERING AN OWNER-SPECIFIED SHIP VIZ-A-VIZ A STANDARD VESSEL.

I CONTAINER LINER FOR A GROUP OF OWNERS, reguires the following procedures :

- Owners to agree to one design, i.e. General Arrangement, Lines, Propulsion, Specification.
- Preliminary tanktesting by owners.
- Tenders to several yards ... Owners being in a negotiating position.
- After discussion about price and specification the contract is signed with a certain yard.
- Design to be taken over by yard - who from there on is responsible for the design.
- Continued research on 50-50 basis.
- Building identical ships.

II A STANDARD VESSEL (BULKER / TANKER / MULTIPURPOSE) tender involves the following procedures :-

- Owner stipulates "outline" requirements, together with simple sketch arrangement drawing(s) and asks for "standard Vessel" at several yards, supplementing his inquiry with a list of preferences.
- Negotiations with yards about application of many alternative specification details follow, mainly concerning quality of materials and surface preparation.
- A definite conception and price are agreed next.
- In this way, the yard may go for h e r preference in many details and way of construction.
- One important thing to be kept in mind is that :

STANDARDISATION TENDS TO A CHEAP SHIP - HOWEVER, BLOCKS THE FUTURE DEVELOPMENTS.

Designs often are more than five years old. Improvement in dimensions or operation-efficiency is not anymore possible. As usually a ship is to last some 20 years in service, it is clear that competitors or even a daring outside entrepreneur may get a leading edge, by introducing more advanced material.

(Note 1. An outstanding example is the Sea-Land case, who intruded into the throat-cutting North Atlantic conventional trade ... and did win gloriously with her fresh advanced container system over the thirteen-odd established lines !)

(Note 2. The 'open ship' was to be initiated by a shipowner, because of the fact that a yard would never be the first in offering a more expensive ship.

(Note 3. The Quarter Ramp, making the RoRo vessel more flexible, came forth from shipowners brains and did break "standards".

- The builder is interested in selling and therefore offers a cheap ship on the verge of being "just acceptable" ! The builder is interested in first place to improve his yard and not the ship.
- The owner is interested in offering a low transport tariff and therefore tries buying a cheap ship. And an owner is - if proper - interested in improving the vessel's design.

In short: -

A SHIP, OPTIMUM TO THE YARD IS NOT AN OPTIMUM ONE TO THE OWNER - ESPECIALLY A LINER OWNER.

The investment for a 20 year period is too large to start with "old designs".

- The best solution, as we see it, is to develop a design between Owner's and Yard's ideas for a ship that is suitable for the service(s) it is intended for, with an eye on the future.

  Such a vessel may be good for a series of units for theservice concerned and - with minor corrections - for services of comparable ambient circumstances for other owners. (Nedlloyd Houtman type : 10 vessels).
- Once committed to a yard, the owner is not anymore in a favourable negotiating position and he will have to pay much more for his requirements. This is the background for the reasoning that an owner should first make his own design b e f o r e he 'goes to the yards'.

C O N C L U S I O N CONCERNING STANDARD VESSELS AS SEEN BY LINER OWNER

- Cheap - but not always reliable in service.
- Very seldom an advanced vessel as regard principal dimensions, cargo handling features and propulsion, because of the earlier date of design. (For instance, a Seaway-beam pf 22,86 m limits development) fig. 18.
- Possibility of vibration troubles, because of bluntness of afterbody(fig.16) (too short length)(too big block)(too small aperature)(too small rudder)
- Possibility of breakdown of pipe systems - caused by seawater corrosion,generally by galvanic action and high velocity in small diameter thin-walled steel pipes.
- Possibility of breakdown of electrical systems - caused by vibrations, sea atmosphere, high temperature. Instruments being not shipshape.
- Paint and corrosion troubles on ship's underwater part.
- Rust formation on weather-exposed surfaces.
- Corrosion in tanks.
- Many 'cheap' construction details which are difficult to maintain.Fig.17.

Generally speaking ... TOO MUCH WORK FOR A REDUCED CREW !

R E M E D Y

- Yard should give the owner the possibility to selection of high quality products (for example: application of stainless steel and copper alloys)
- Allowance by yard to apply sound preparations methods (for example: blasting)
- A wise owner will pay for quality - but doen not like to get 'skinned' !
- Like-minded shipowners should co-operate in grading-up shipyard-offered yard-quality. (Meanwhile, yard should not yield to owner-representatives' 'hobby-horses'- i.e. no "hootersand bells)
- A good co-operation between yard and owner can result in a useful vessel of simple, sound and clean construction without being expensive. Fig. 17.
- At the best invention nothing can be omitted anymore.

GENERAL DESCRIPTION

OF A MEDIUM-SIZE CLASS

CELLULAR CONTAINERSHIP.

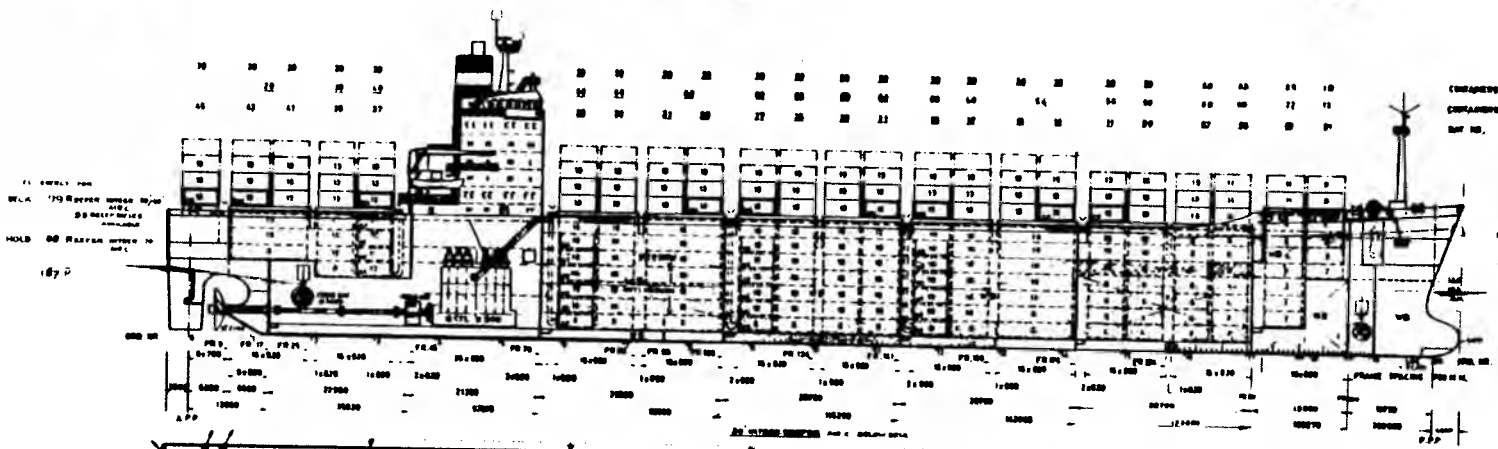
--oOo--

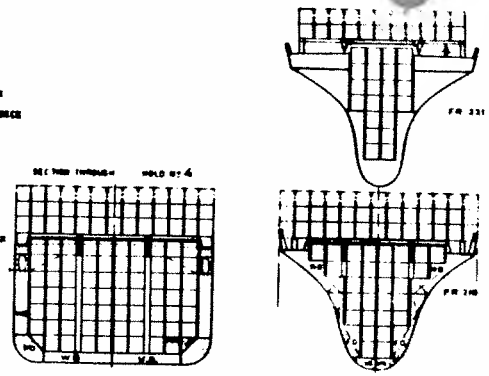
"NEDLLOYD CLARENCE"/CLEMENT",

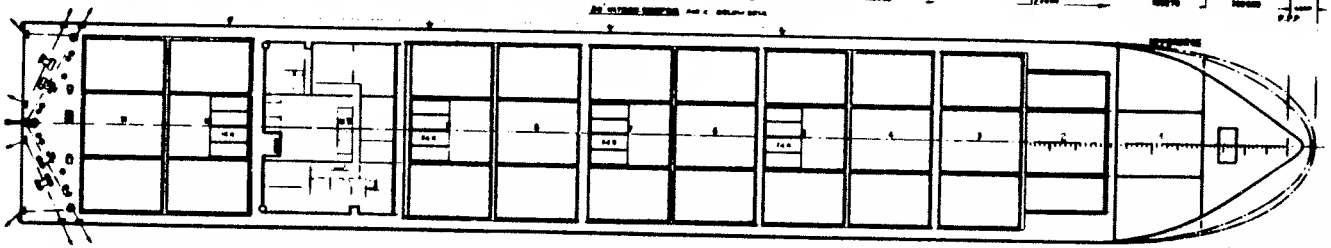
(NEWBUILDINGS HYUNDAI 225/6.)

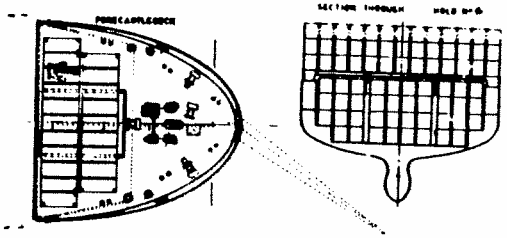
1983

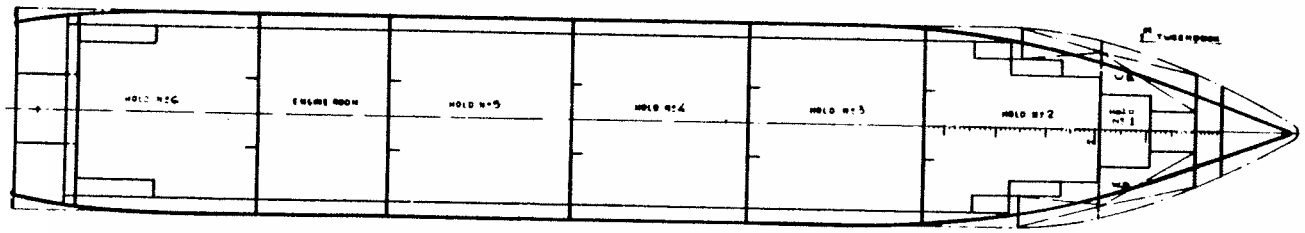
UNIVERSAL CONTAINER

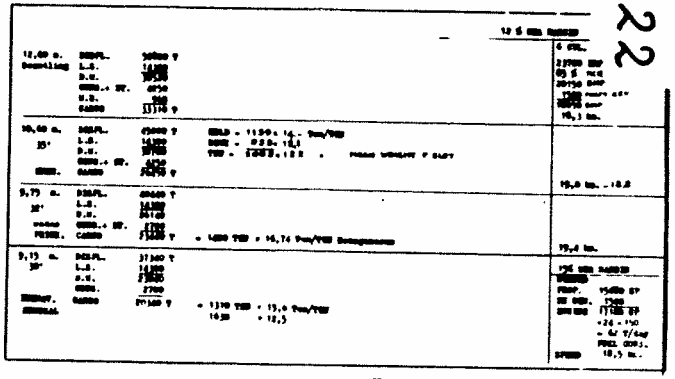


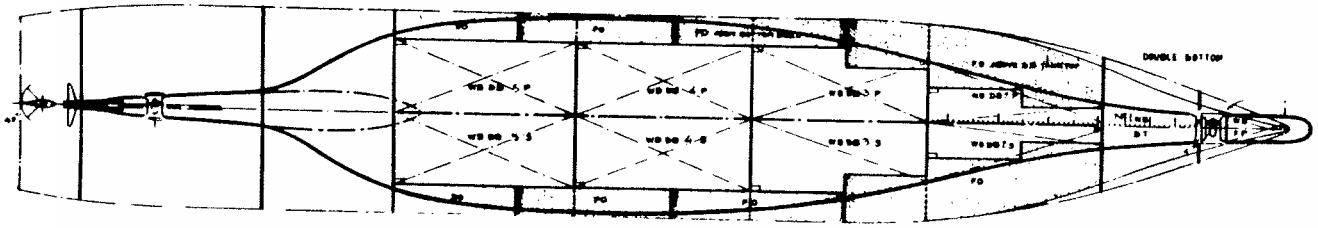












Container capacity below deck

40' x 160 confrs = 320 TEU = 28%
20' x 830 TEU = 72%

below deck 1150 TEU 100%

CONTAINER CAP BELOW DECK	1150 TEU
CONTAINER CAP ON DECK 2 TIERS	562 TEU
GRAND TOTAL 2 TIERS	1712 TEU
3e TIER	291 TEU
GRAND TOTAL 3 TIERS	2003 TEU
4e TIER	221 TEU
IN CASE STABILITY ALLOWS	2224 TEU

Revised 16-8-83

Nedlloyd Fleet Services
Newbuilding Department

"UNIVERSAL" CONTAINERSHIP

NEDLLOYD CLARENCE

1 : 500 EMEC

22.1.01/+ (=vp 02)

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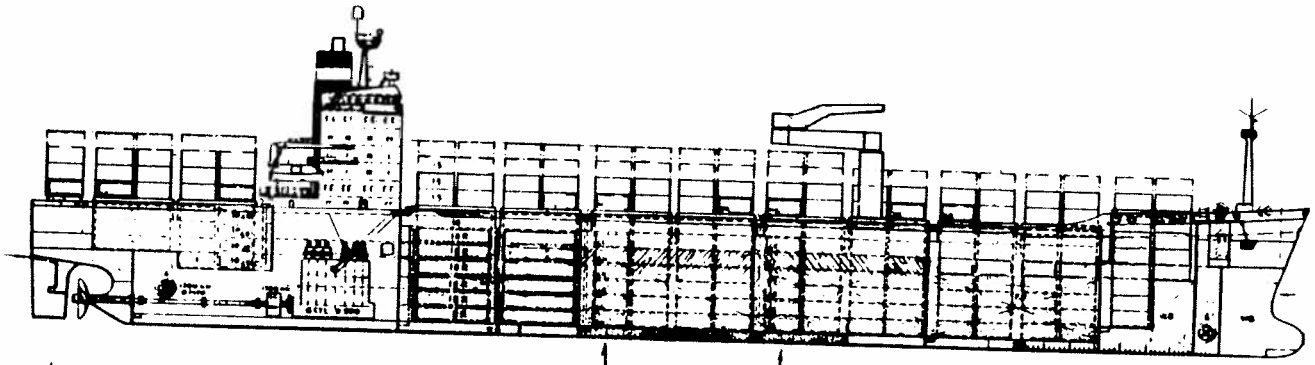
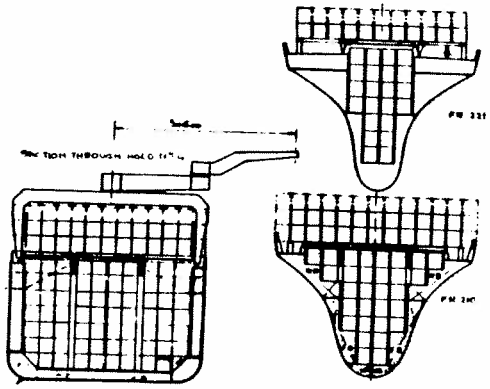
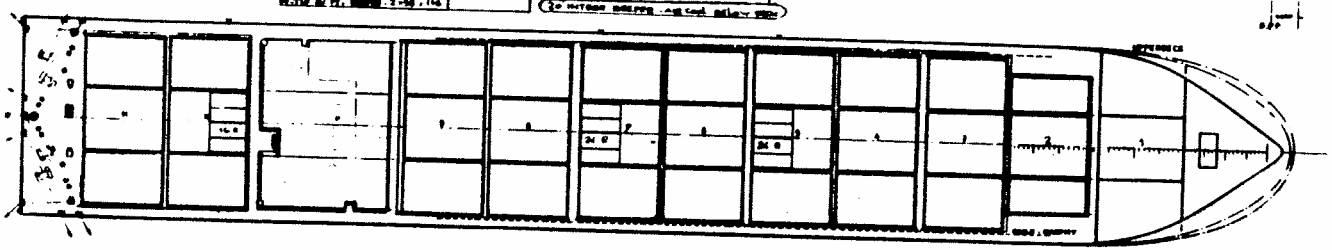
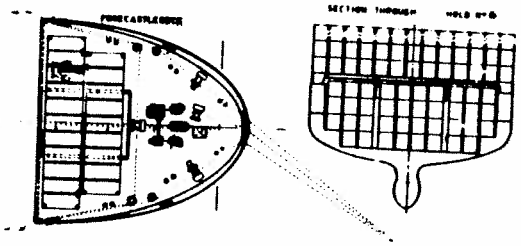
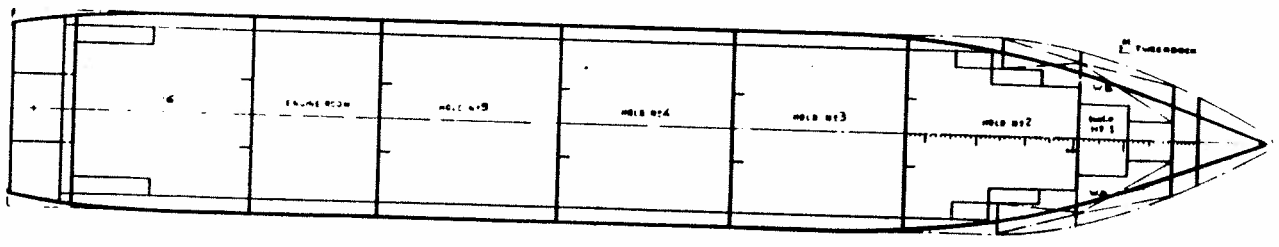
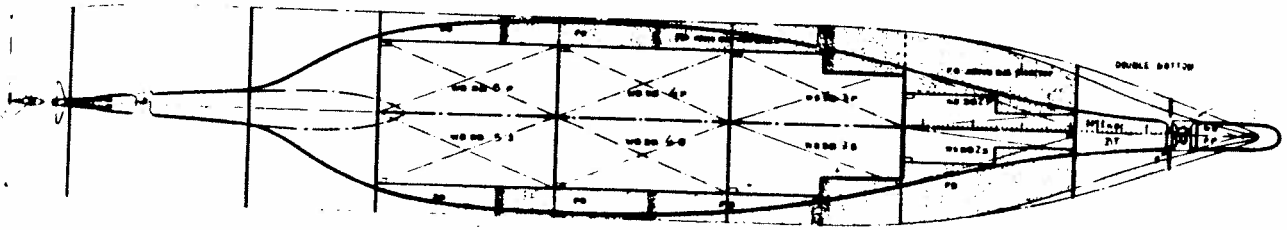












NEDLLOYD ELEMENT (1000-class, with Gantry crane fwd)

12,00 m (Seasailing)	DISPL.	52,000 T	
	L.B.	[illegible]	
	D.W.	[illegible]	
	CONS.+ DT.	4,050	
	W.B.	[illegible]	
	CARGO	33,[illegible] T	
10,60 m (35')	DISPL.	45,000 T	1150 x 14 V/ton
	L.B.	[illegible]	
	D.W.	[illegible]	
	CONS. + DT.	4,250	
(BUNC)	CARGO	30,[illegible] T	(Data: 6 ton for gantry-leg positioning.)
9,75 m (32')	DISPL.	40,440 T	
	L.B.	[illegible]	
	D.W.	[illegible]	
	CONS. + DT.	[illegible]	
(PRIME)	CARGO	[illegible]	Homogeneous 1400 x 16,6 V/ton
9,15 m (30')	DISPL.	37,340 T	
	L.B.	[illegible]	
	D.W.	[illegible]	
	CONS. + DT.	[illegible]	
(ALLEGAT.)	CARGO	[illegible]	1310 x 15,4 V/ton 1630 x 12,4

Container capacity below deck

40' = 164 contrs = 378 TEU = 28.9%
20' = = 822 TEU = 71.9%

below deck 1160 TEU 100%

PRINCIPAL DIMENSIONS

LENGTH OVER ALL	210.00 m
LENGTH BETWEEN PERP	200.00 m
BREADTH MOULDED	32.24 m
DEPTH TO UPPERMOST HOLD	16.00 m
DEPTH TO 1st TWEENDECK	14.77 m
DRAUGHT DESIGN	9.15 m
DRAUGHT CONSTRUCTION	9.15 m

CONTAINER CAP BELOW DECK	1150 TEU
CONTAINER CAP ON DECK 2 TIERS	558 TEU
GRAND TOTAL 2 TIERS	1708 TEU
3e TIER	289 TEU
GRAND TOTAL 3 TIERS	1997 TEU
4e TIER	195 TEU
IN CASE STABILITY ALLOWS	2192 TEU

Revised 16-8-83

Nedlloyd Fleet Services
Newbuilding Department

"UNIVERSAL" CONTAINERSHIP

** NEDLLOYD ELEMENT **

226

L OA = 210,00 M LR - 100 A1 - LMC-UMS

L PP = 202,00 M

B = 32,24 M

H = 18,80 M

DRAFT = 12,00 M SCANTLING

DRAFT = 10,60 M DESIGN.

WATERBALLAST	10.484	m^3
FRESH W. SP	150	m^3
" W P	119	m^3
FEEDWATER P	30	m^3
HEAVY F.O	4088	m^3
LUB. OIL	287	m^3.

TYPICAL LOADING CONDITION

DRAFT = 10,60 M.

DISPL = 45000 T

L SHIP = 14550

30450 T

CONS + WB ~ 4000

CARGO 26450 — HOLD = 1150 TEU × 14 T/TEU ⊙ 45% HEIGHT OF CONTR.

DECK = 853 " × 12,1

MEAN = 2003 TEU × 13.2 T/TEU GM ~ 0,4 m

MAX TEU CAPACITY SPACE-WISE : NEDLL - CLARENCE

IN HOLDS = 1150 TEU

ON DECK 2 TRS = 562

3° TIER = 291

2003 TEU

4° TH TIER 221

2224 TEU.

REEFERS. IN HOLDS {24, 24, 24, 16} = 88 TEU - 20' INTEGRAL AIR COOLED : EXTRA HOLD VENTIL = 1 m^3/TEU.SEC.

ON DECK 11 × 9 = 99 TEU - 20' à 5 KW OR 40' à 7½ KW — RECEPTACLES : 440V CEE 17, 3 HR. 32 AM

HOLD VENTILATION : N° 1 = 2,5 × HR. EMPTY HOLD

HOLD N° 2 = 2,5 × " EXHAUST ONLY : DANGEROUS CARGO

HOLD N. 3-4-5.6 = ~7 × " BECAUSE OF EXTRA FANS FOR REEFERS

SPACE ON TANKTOP FOR BREAKBULK (2 TIERS HIGH) — NOT ON "N. CLARENCE"

HATCHROWS : 3 - 4 - 3

N° OF HATCHES : 2 - DECKHOUSE - 9

40' CELLS BAY 14 = HATCH N° 4 = 28 : 40'

BAY 30 = " 8 = 68 "

BAY 42 = " 11 = 64 "

160 = 320 TEU / 1150 HOLD TEU = 28% BELOW DECK

MAX STACKLOAD ON COVERS : 20' STACK = 60 TON

40' " = 90 TON.

LASHING BY "TWISTLOCKS-ONLY" : UP TO 45 TON / STACK. ; ABOVE 45 T : RODS + TURNBUCKLE

MAIN ENGINE B+W HYUNDAI 6L 90 GBE

MCR : 23.800 HP – 97 RPM
85% MCR = SERVICE 20.230 HP – 97 RPM

$\times 127^{GR}/_{HPHR} \times 1.1$ (DO → HFO) $= 140^{gr}/_{HPR} \times 24 \times 20.230$ (85%) $= 68$ TON/DAY.

FUEL FOR ME HFO MAX 600 cSt AT 50°C

18 TON - BOWTHRUSTER = KAMEWA 1200KW
- STERN " = " "

STEERING GEAR 4 RAM TYPE - KAWASAKI
35° – TORQUE 150 TON M MAX 155 9
45° – RUDDERANGLE UP TO 12 KN

FUEL CONSUMPTION (EXCL. SHAFTGEN)

KN.	7	8	9	10	11 M	DRAFT
16.5	34.0	35.4	37.2	39.8	44.5	TON/DAY
17.–	37.0	38.4	40.3	43.2	48.3	
17.5	40.0	41.8	43.6	47.0	52.5	
18.–	43.5	45.3	47.7	51.3	57.1	
18.5	46.6	48.9	51.1	55.6	62.5	MAX 68
19.–	50.5	52.6	55.5	60.5	68.5	
19.5	54.5	57.2	60.6	66.3		
20.–	59.0	62.0	66.2			
20.5	64.4	67.6				
21.–	71.0					

MEAN SPEED FROM PILOT TO PILOT
12% SERVICE ALLOWANCE
FOR ROUGHNESS
WIND
SEA
↓
$HP_{SERVICE} = 1.12$ HP
TRIAL CONDITIONS
SMOOTH HULL
BF 2
CALM SEA.

SHAFT GENERATOR MAX 1272 KW – 73 → 97 – MAX RPM.

NORMAL KW EXCL REEFER ~600 KW $\times \frac{1.36}{0.815} = 1000$ HP
1000 HP $\times 140^{gr}/_{HP\,HR} \times 24 = 3.4$ TON/DAY.

FUEL CONSUMPTION INCL. SHAFTGENERATOR 3.4 T/DAY EXTRA ABOVE TABLE

EXAMPLE : 9.5 M DRAFT / 19.5 KN 63.4 T/DAY + 3.4 T/DAY = 66.8 T/DAY.
PROP + S GEN = 68.– MAX

DIESEL GENERATORS

N: 1.	1700 KW – 600 RPM	DAIHATSU	8.D.S.32	IFO = 55% HFO + 45% MDO
N: 2	800 KW – 720 RPM	"	6.D.S 26 A	" "
N: 3	1700 KW – 600 RPM	"	8.D.S.32	" "
EMERG. GEN	180 KW – 1800 RPM	D.A.F.	6 cyl D.K.S 1160	= LIGHT D.O.

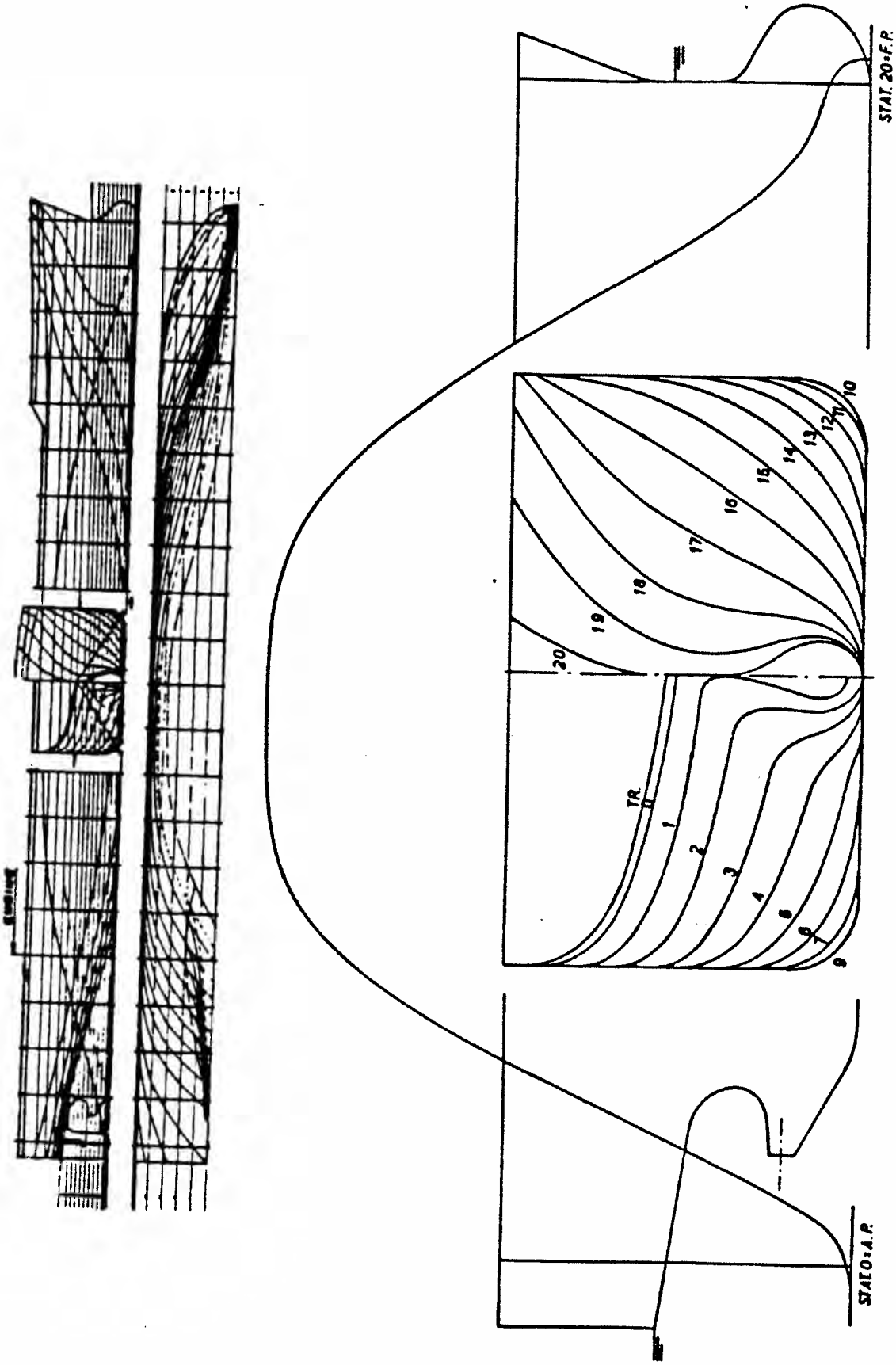
ELECTRIC LOAD AT SEA : MEAN 600 KW (EXCL. REEFER CONTRS) BY DIESEL GEN : 3.4 TON/DAY IFO
BY SHAFT GEN : 3.4 TON/DAY H.F.O

IDLE IN PORT : 450 KW BY DIESEL GEN : 2.6 TON/DAY IFO.
+ AUXIL BOILER (MAX 2.5 T. STEAM/HR) MEAN : 1.0 " H.F.O

EXTRA KW FOR REEFER CONTAINERS (INTEGRALS) : N: BOX × KW/BOX × HP/KW × GR/HPHR × 24

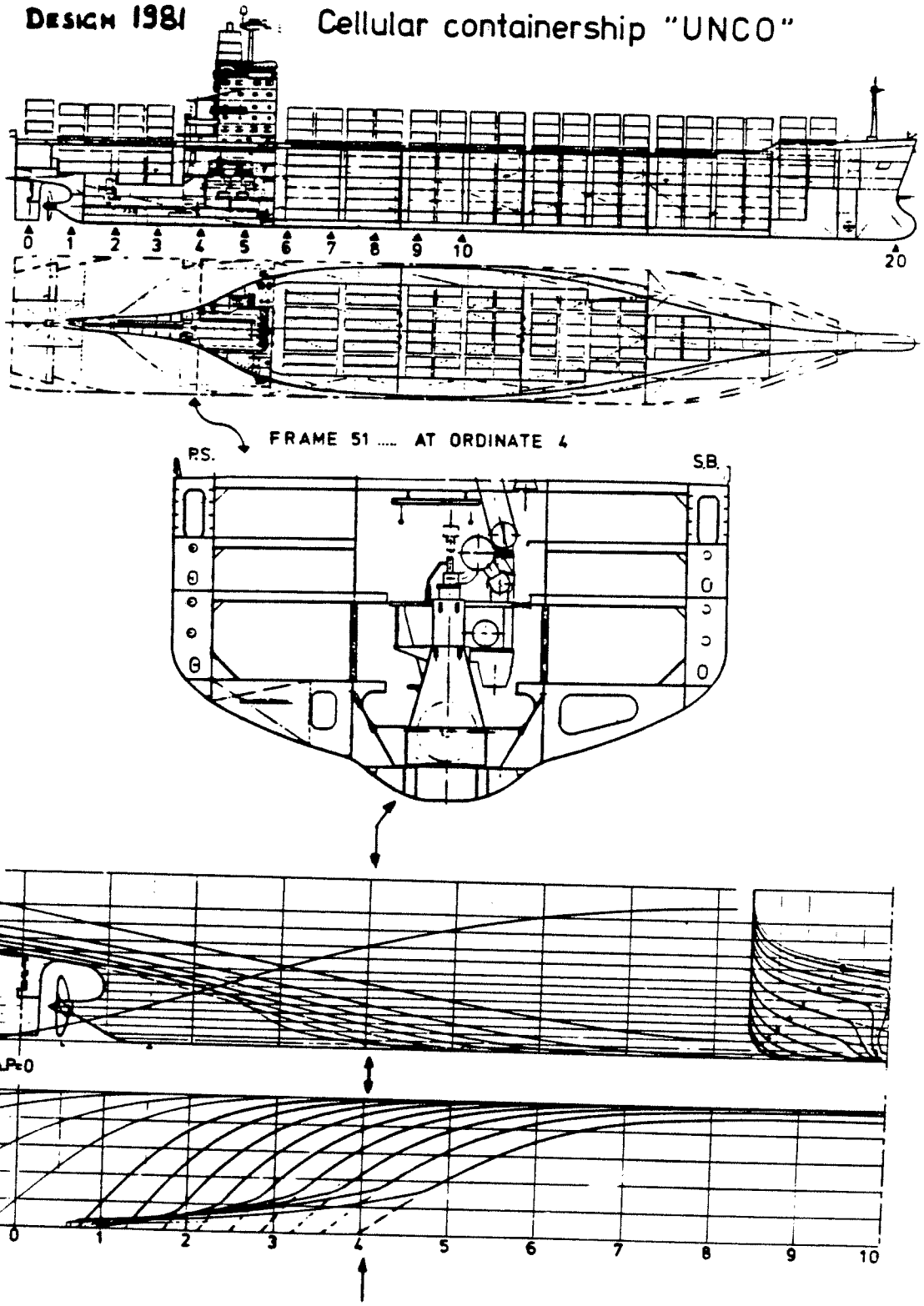
EXAMPLE 100 × 5 FOR 20' × 1.5 × 160 × 24 = 2.9 TON/DAY
7½ FOR 40'

SHIP MODEL No. 5901 C BULB IV



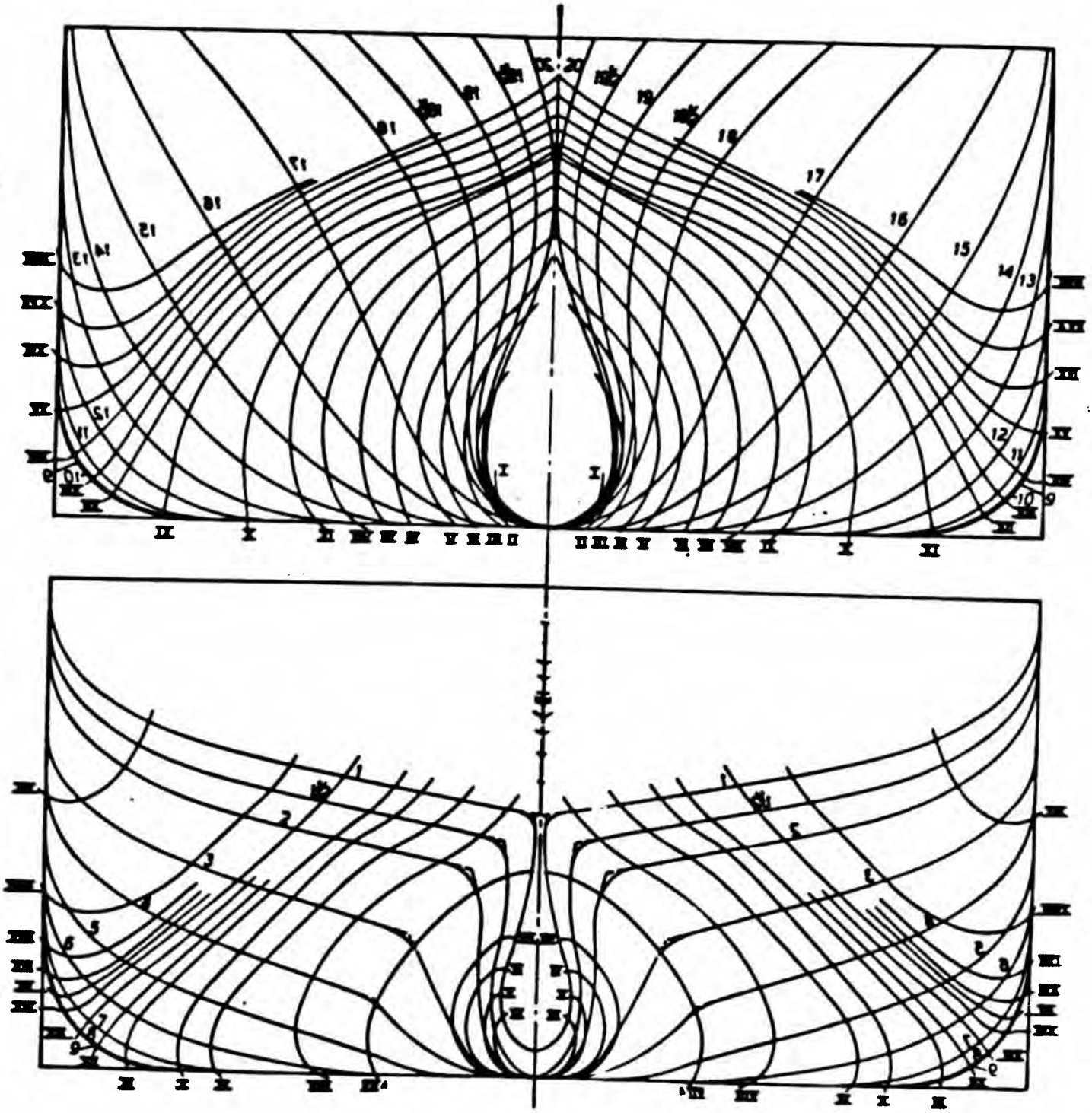
DRAWING No. M 5901-36

DESIGN 1981 Cellular containership "UNCO"



SHIP MODEL Nº 5901 ("UNCO" PROJECT)
STREAMLINES.

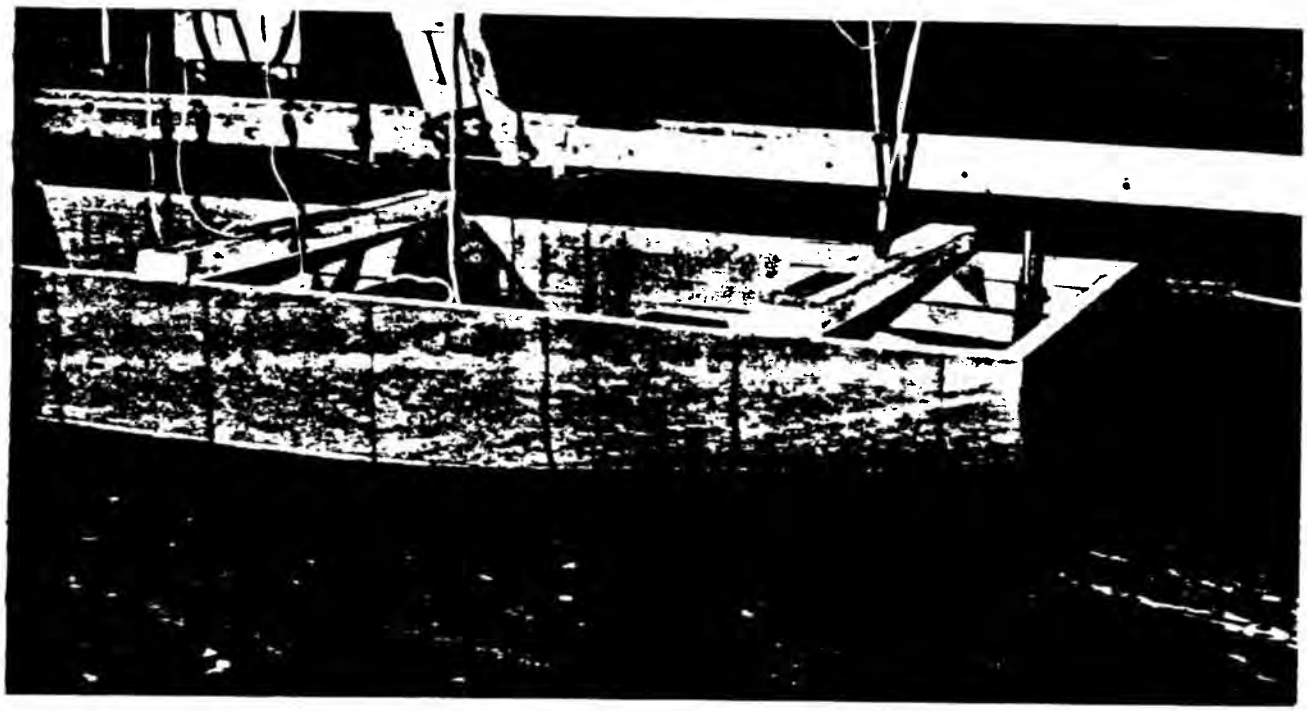
DRAFT 10.60 m SPEED 19 KNOTS TEST Nº 39891



SHIP MODEL No. 5901

BULBOUS BOW I (BIG BULB)

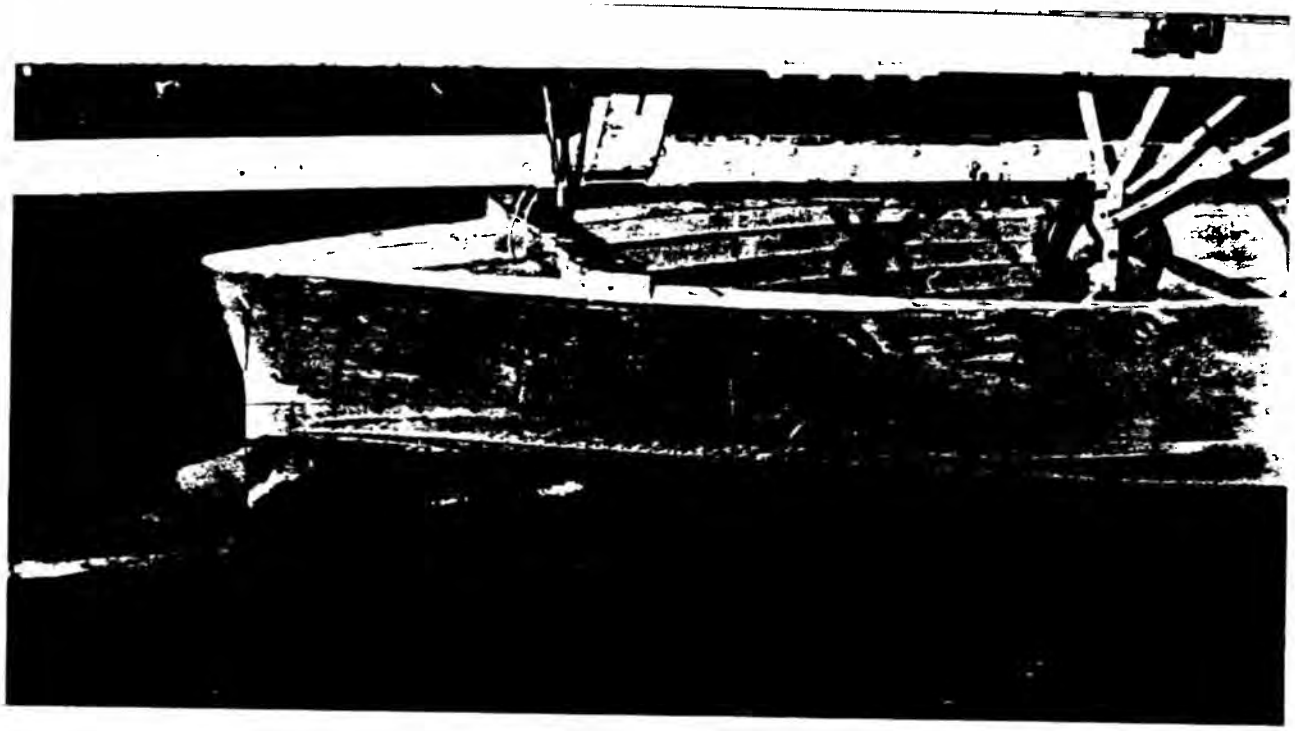
DRAFT 10.60m SPEED 19.0 KNOTS TEST No. 39841



DRAFT 6.40 / 7.60m SPEED 20.0 KNOTS TEST No. 40907

SHIP MODEL No. 5901 C

BULBOUS BOW IV (BALLAST BULB)

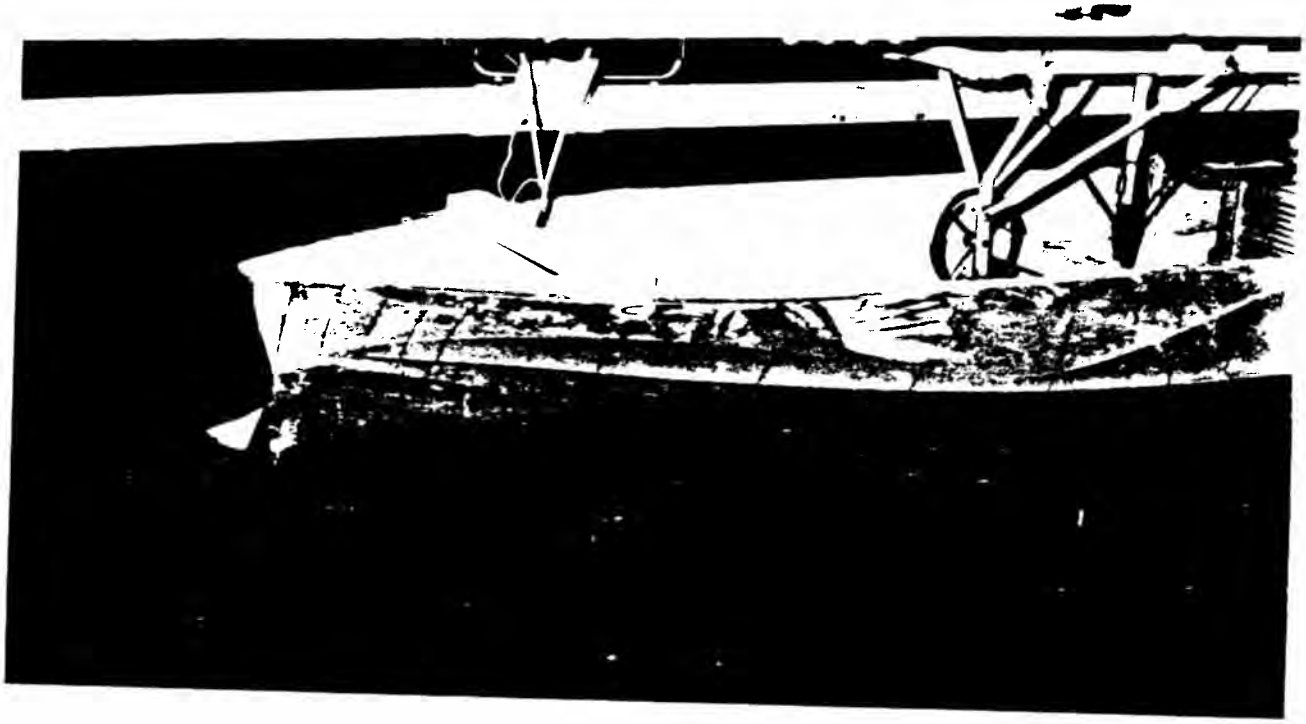


DRAFT 7.00 m SPEED 20.5 KNOTS TEST No. 40957

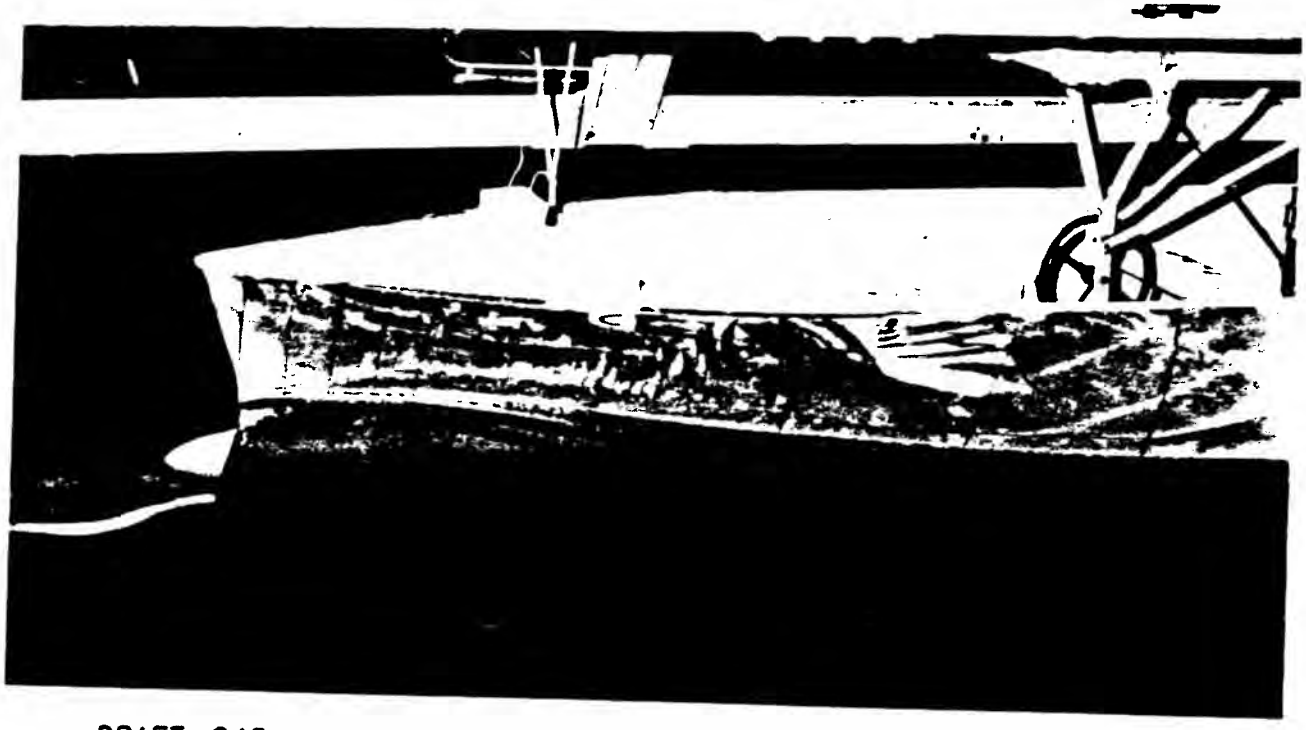
DRAFT 6.40 7.60m SPEED 20.5 KNOTS TEST No. 40956

SHIP MODEL No. 5901 C

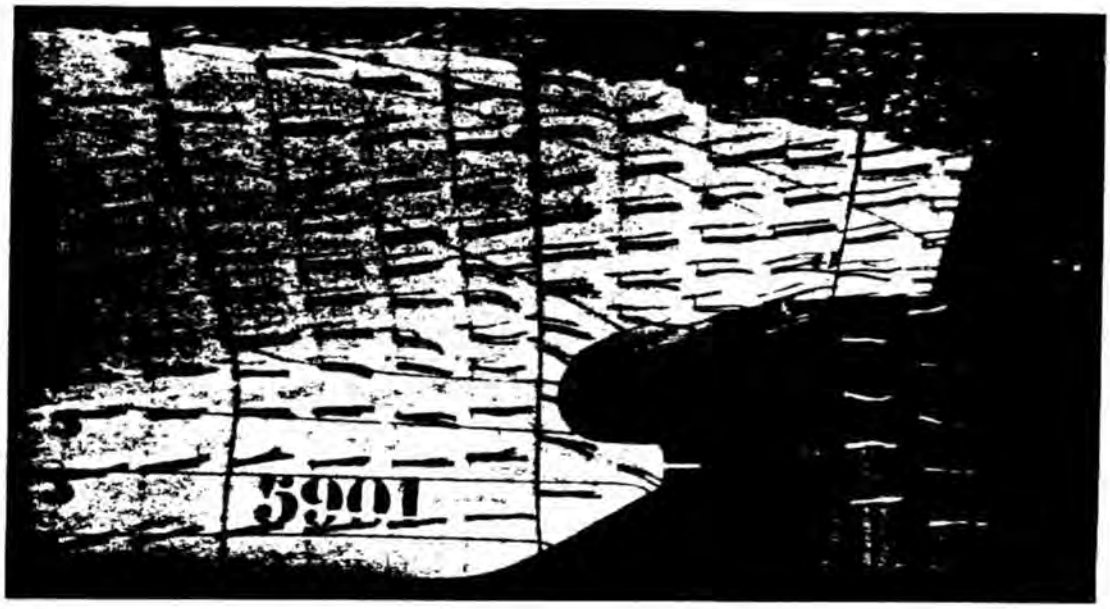
BULBOUS BOW IV (BALLAST BULB)



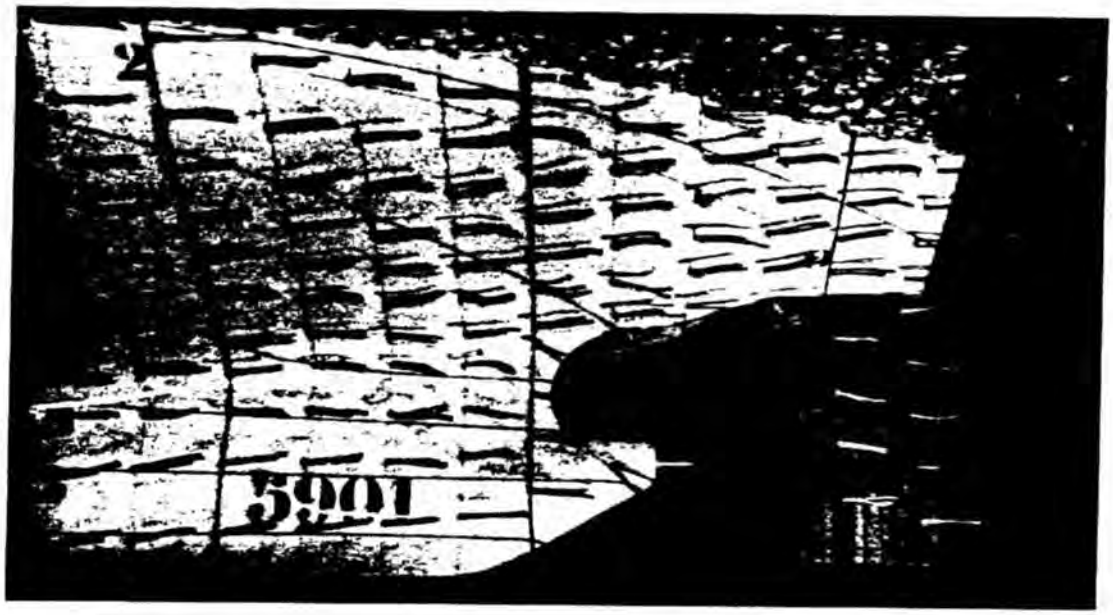
DRAFT 10.60 m SPEED 19.0 KNOTS TEST No. 40954

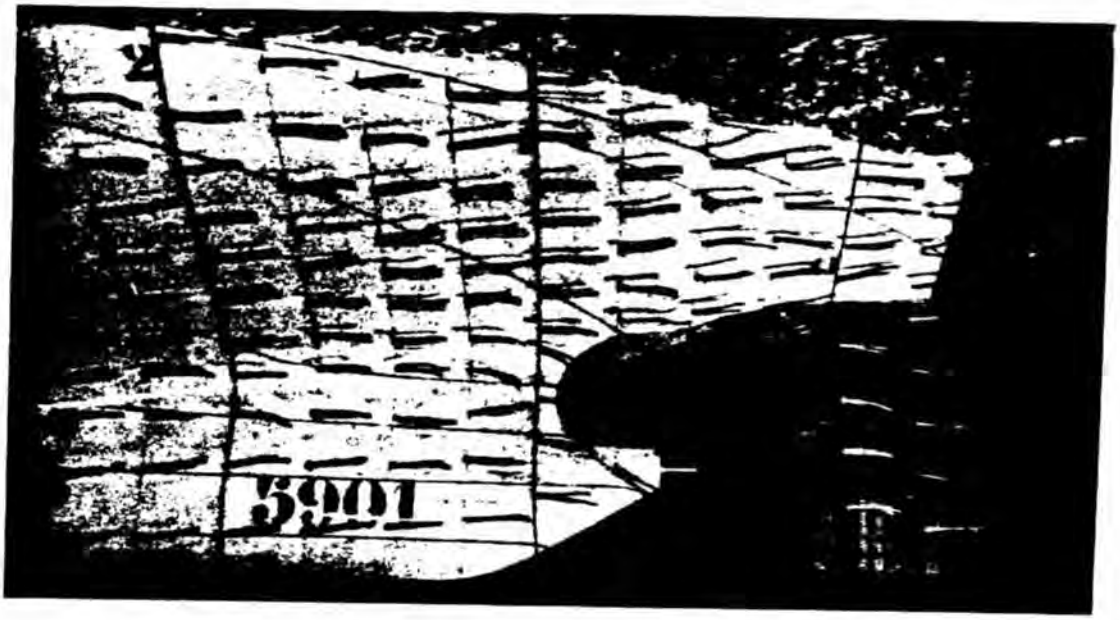


DRAFT 9.15 m SPEED 19.5 KNOTS TEST No. 40955



WITHOUT PROPELLER





WITH PROPELLER

NOTE DISTURBANCE ABOVE PROP

CAVITATION OBSERVATIONS

TEST No. 82-110
SHIP MODEL No. 5901C
PROPELLER MODEL No. 5631
SHIP DRAUGHT ON F.P. 10.60 m
SHIP DRAUGHT ON A.P. 10.60 m

V_s = 18.80 KNOTS
N_s = 1.583 Hz
P_S = 13713 kW

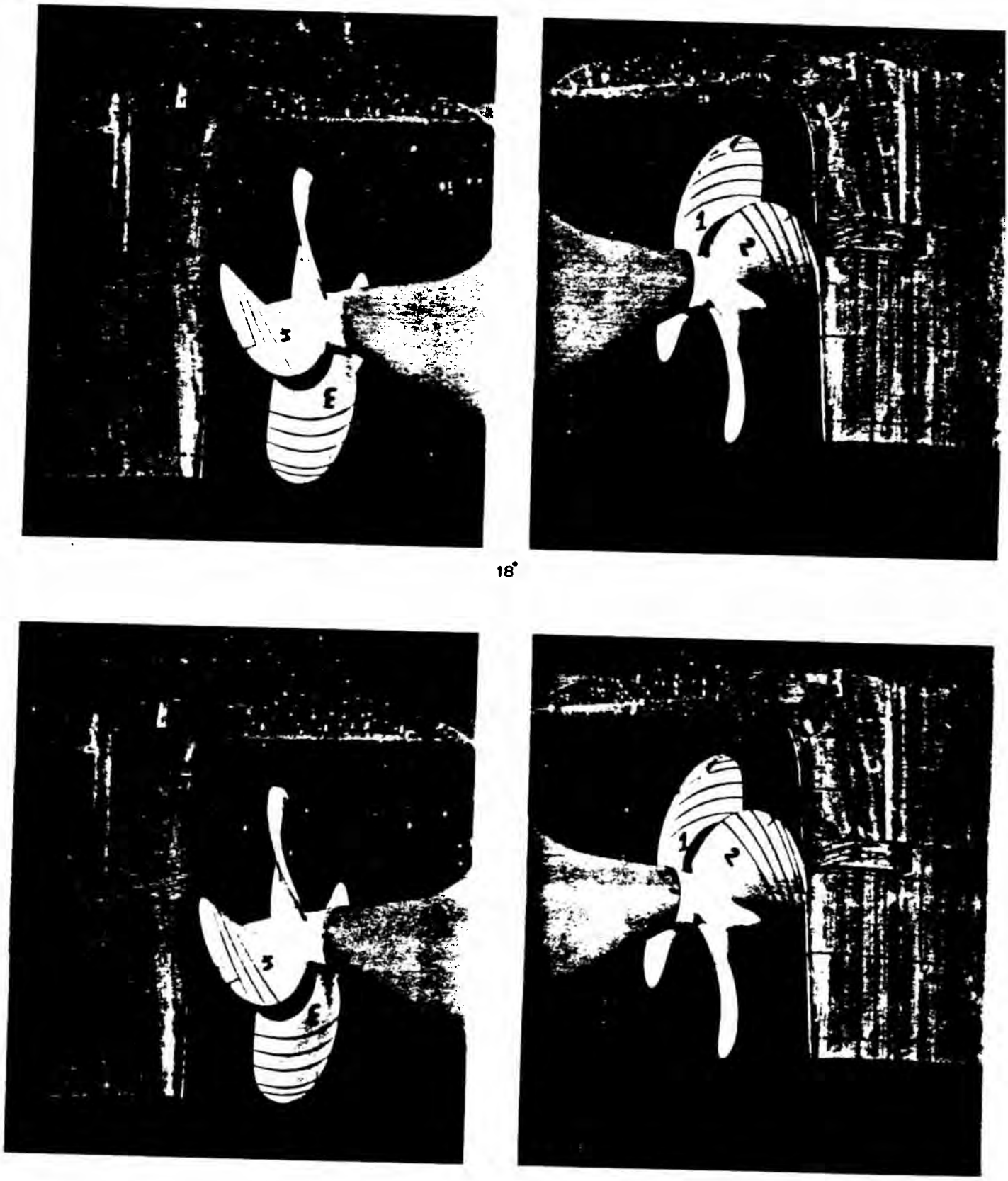


FIG. 2.3. SUCTION-SIDE CAVITATION OBSERVATIONS
ANGLES INDICATE POSITION OF BLADE No. 1 RELATIVE TO TOP POSITION

CAVITATION OBSERVATIONS

TEST No. 82-107
SHIP MODEL No. 5901C
PROPELLER MODEL No. 5631
SHIP DRAUGHT ON F.P. 6.40 m
SHIP DRAUGHT ON A.P. 7.60 m

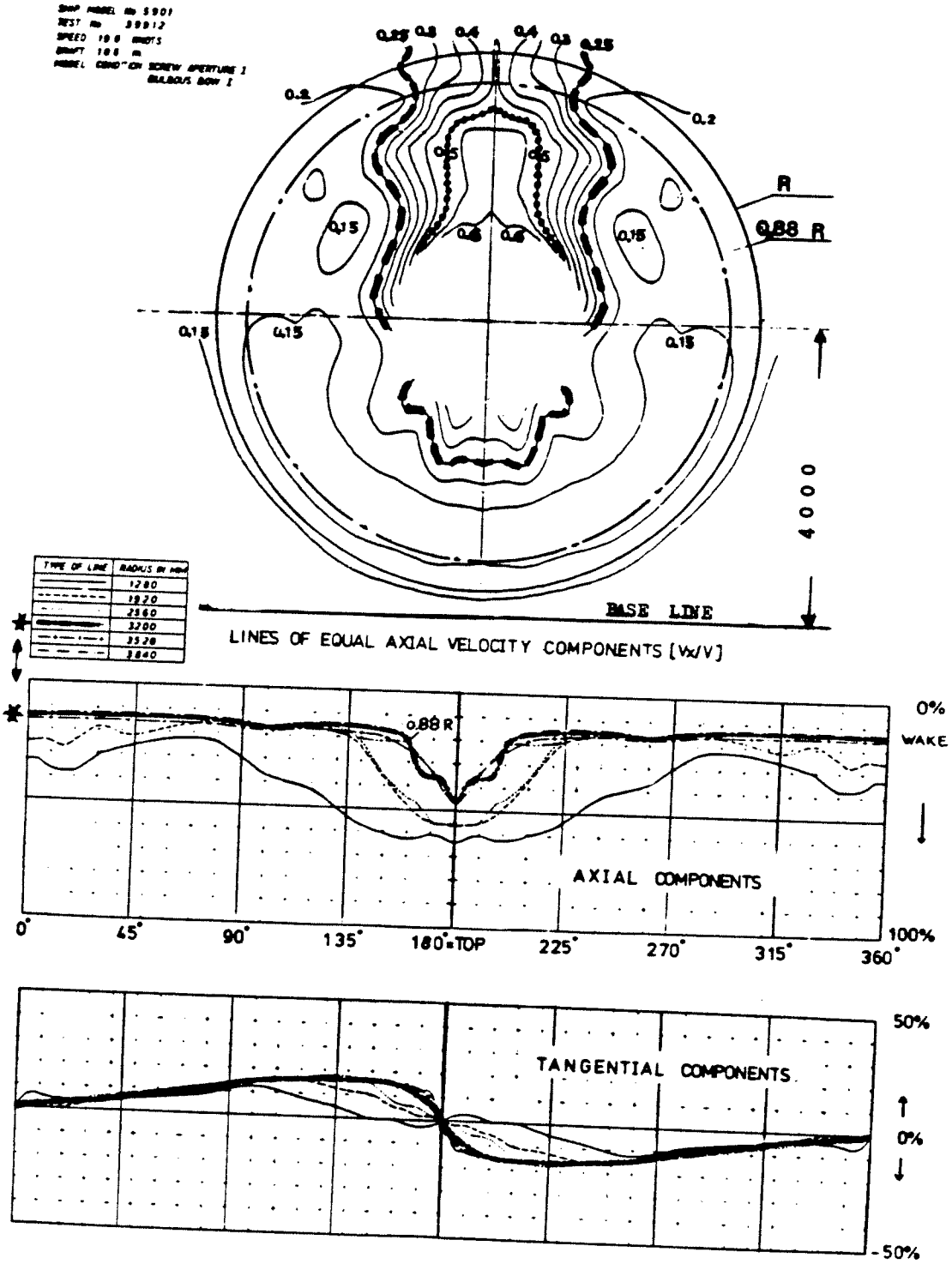
V_s = 20.32 KNOTS
N_s = 1.575 Hz
P_S = 13713 kW

18°

24°

FIG. 2.7. SUCTION-SIDE CAVITATION OBSERVATIONS.
ANGLES INDICATE POSITION OF BLADE No 1 RELATIVE TO TOP POSITIO

WAKE FIELD MEASUREMENTS
"Universal" containervessel ("UNCO"),
AT 10,60 m DRAFT.



22.3.12 (=VP 17)

22 CRITICAL FRAMES ON "UNCO"-TYPE VESSEL

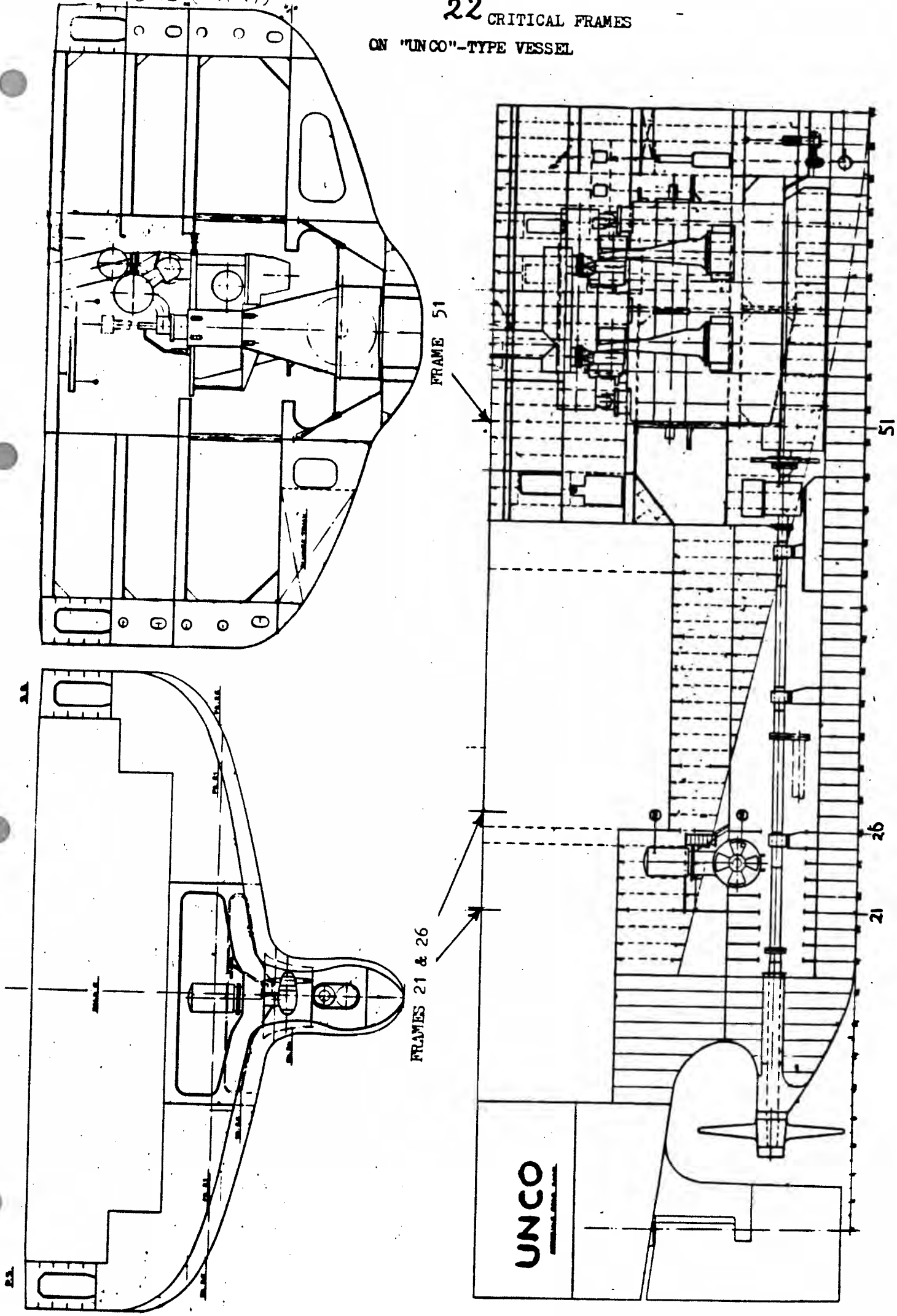


Table I

Particulars of ship

Ship represented by ship model 5901C with bulbous bow IV

Length between perpendiculars	L_{PP}	202.00	202.00	202.00	202.00 m
Length on waterline	L_{WL}	205.57	205.57	192.86	193.96 m
Breadth moulded	B	32.24	32.24	32.24	32.24 m
Draft moulded on A.P.	T_A	10.60	9.15	7.00	6.40 m
Draft moulded on F.P.	T_F	10.60	9.15	7.00	7.60 m ← TR
Displacement volume mld (without tunnels)	∇	43810	36359	--	-- m^3
Displacement volume mld (with tunnels I)	∇	--	36319	26150	-- m^3
(with tunnels II)	Δ	43774	36323	--	26154
Displacement weight (without tunnels)	Δ	44905	37268	--	-- mt
Displacement weight (with tunnels I)	Δ	--	37227	26804	-- mt
(with tunnels II)	Δ	44868	37231	--	26808
Wetted surface, bare hull	S_1	8139	7331	6241	6241 m^2
Wetted surface, rudder incl.	S_2	8250	7431	6314	6314 m^2
L.C.B. position aft of ½ Lpp in % of Lpp	θ	1.71	--	--	--

Ship model details:

Ship model material: wood
Linear scale ratio λ = 32
Turbulence induced by studs

Note:

The presented values valid for draft 6.40/7.60 m are approximately: the trim of the model has been obtained by shifting the ballast weights.

NSMB Netherlands Ship Model Basin

Table II

Characteristics of propellers:

Propeller No.		3754[1)]	5631[2)]	5631A[3)]
Diameter	D	7259	7200	7200
Pitch at 0.7R	$P_{0.7}$	6010	6635	6436
Pitch ratio	$P_{0.7}/D$	0.828	0.922	0.894
Expanded blade area ratio	A_E/A_O	0.568	0.637	0.637
Boss-Diameter ratio	d/D	0.172	0.179	0.179
Number of blades	Z	4	4	4

MEAN P 6297

1) stock propeller
2) final Lips propeller
3) final Lips propeller with corrected pitch for required power - RPM relationship.

22.3.15 (=vp 20)

22

WORKING PROPELLER

BASE : ENGINE MCR×0,85 = 20 230
~ 900 KW - SHAFT GEN - 1 500
(N.B ESTIM 600 KW - 1000 HP) MEAN 18730
SHAFTLINE FRICTION - 1.50
SHP PROPELLER = 18 580
10,6 M - V = 19,6 kn
TRIAL CONDITION

PROPELLER DATA

DIAMETER		7200 MM
PITCH, MEAN	= NSMB MODEL PROP N: 5631 A	→ MEAN = 6297 MM
PITCH 0.7 R		0.7 R = 6436 MM.
NUMBER OF BLADES		4
DIRECTION OF ROTATION		RIGHT-HANDED
EXPANDED BLADE AREA RATIO		0.630
MATERIAL		CUNIAL
MINIMUM TENSILE STRENGTH		647 N/MM²
MINIMUM ELONGATION		20 %
MASS		ABT 29100 KG
MASS MOM. OF INERTIA PROP. IN AIR		ABT 69260 KGM²
MASS MOM. OF INERTIA OF ENTR. WATER		ABT 28425 KGM²
ENGINE OUTPUT		17500 KW
PROPELLER REVOLUTIONS		97.00 RPM
E FACTOR 0.26		1.0000
E FACTOR 0.6		1.2500
CLASSIFICATION		LLOYDS REG.

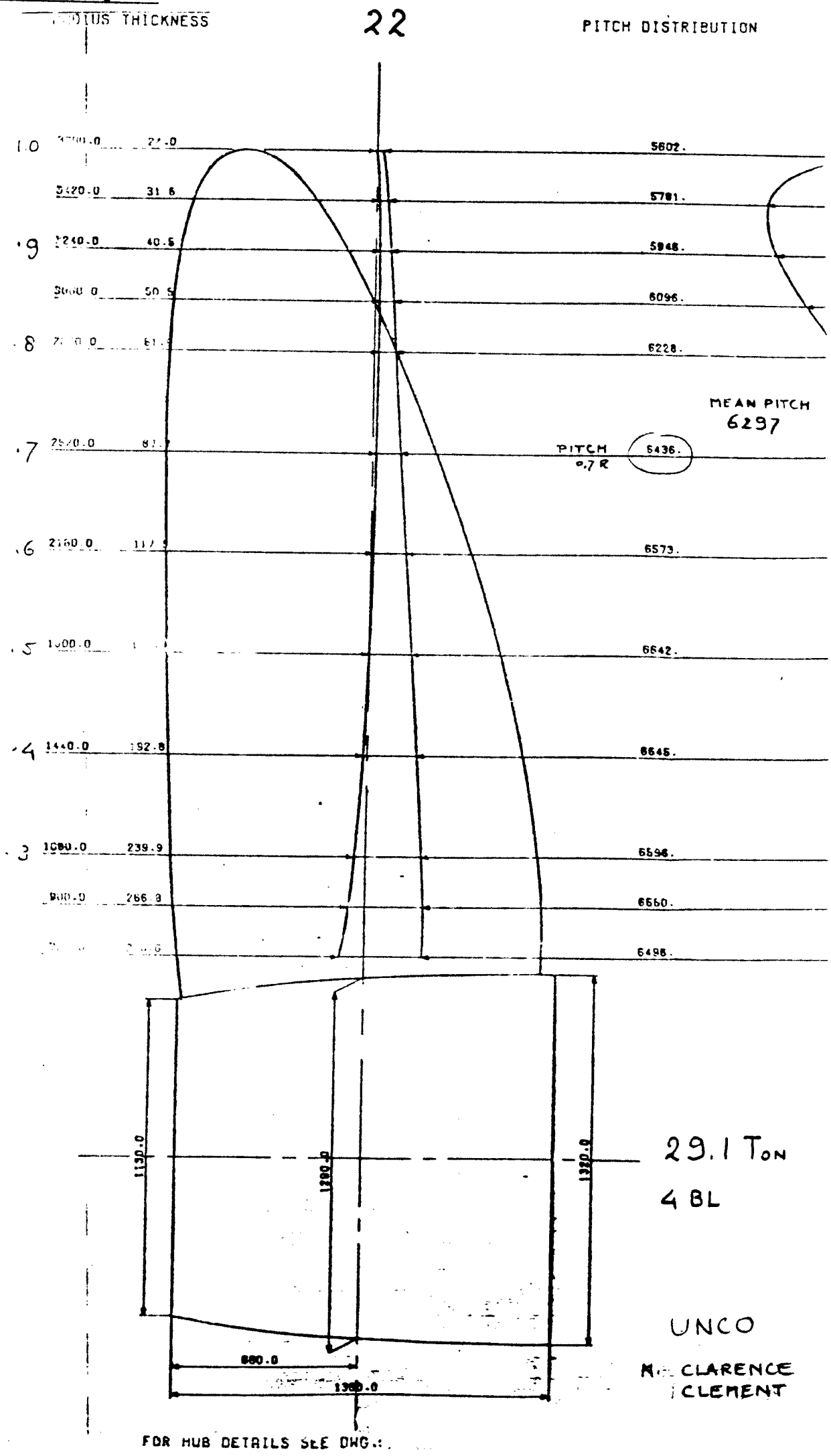
PITCH SELECTED FOR 13775 KW 97.00 RPM, SERVICE CONDITION.

BLADE FINISHING TOLERANCES IN ACCORDANCE WITH ISO CLASS I

INITIALS	THE DESIGN SHOWN ON THIS DRAWING IS THE ABSOLUTE PROPERTY OF -LIPS PROPELLER WORKS- IT MAY NOT BE COPIED OR REPRODUCED, IT MAY NOT BE EXECUTED BY, USED FOR OR SHOWN TO THIRD PARTIES WITHOUT THE WRITTEN CONSENT OF THE OWNER.	F.P. PROPELLER FOR NB NOS 225-226 UNCO		
DATE		CLIENT HYUNDAI HEAVY INDUSTRIES		
DESCRIPTION	HYDRODYNAMIC DESIGN DEPARTMENT	NO. F7846-848 NEDLLOYD CLARENCE " CLEMENT		
		DRAWN		DATE Y/M/D 82/03/18
		CHECKED		
		APPROVED		SCALE 1:12.5
MODIFICATION	LIPS PROPELLER WORKS DRUNEN, THE NETHERLANDS	DRAWING F7846-01	MODIFICATION	SIZE A1

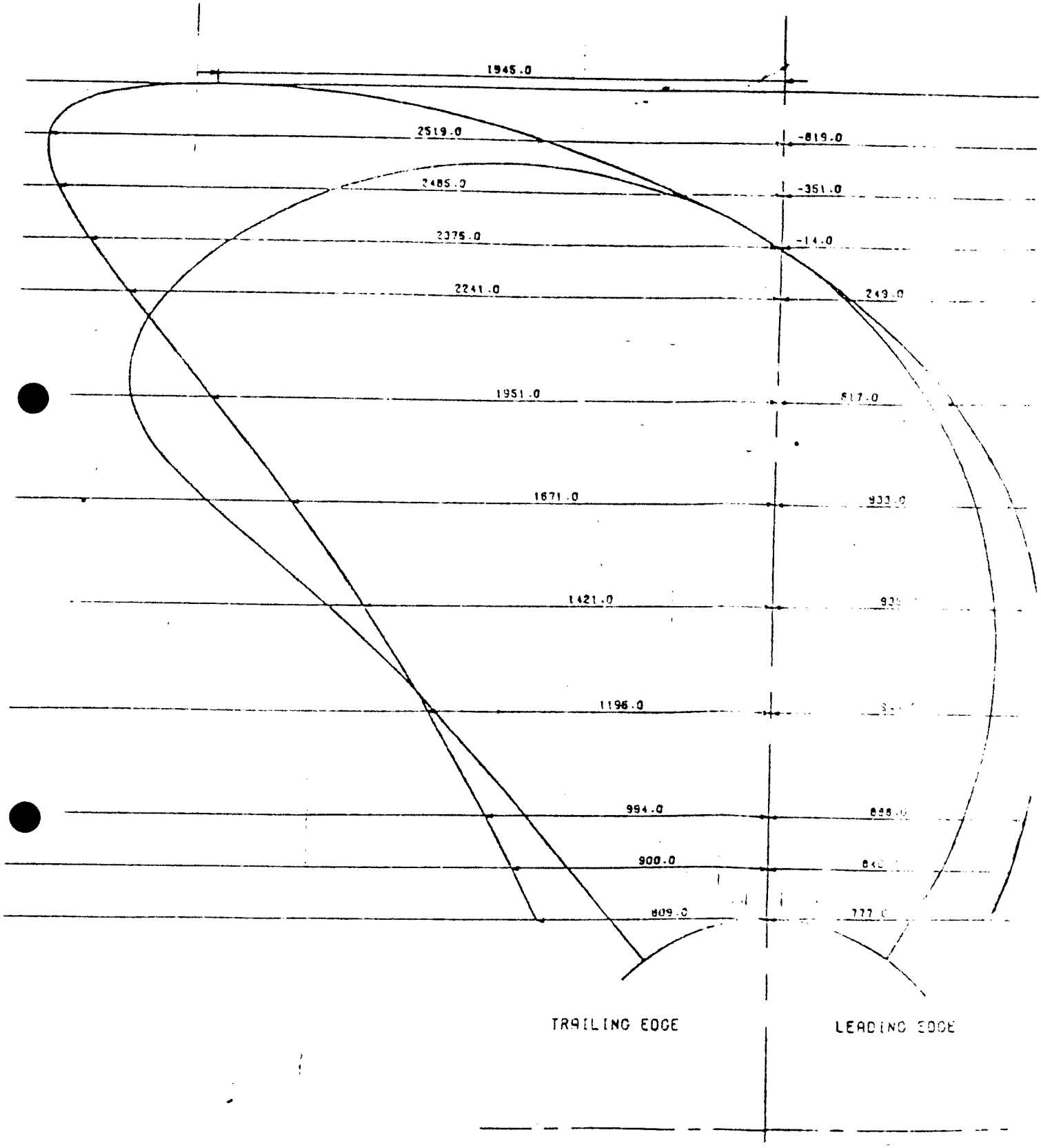
22.3.16 (=vp 21)

22



22.3.17 (=vp 22)

22



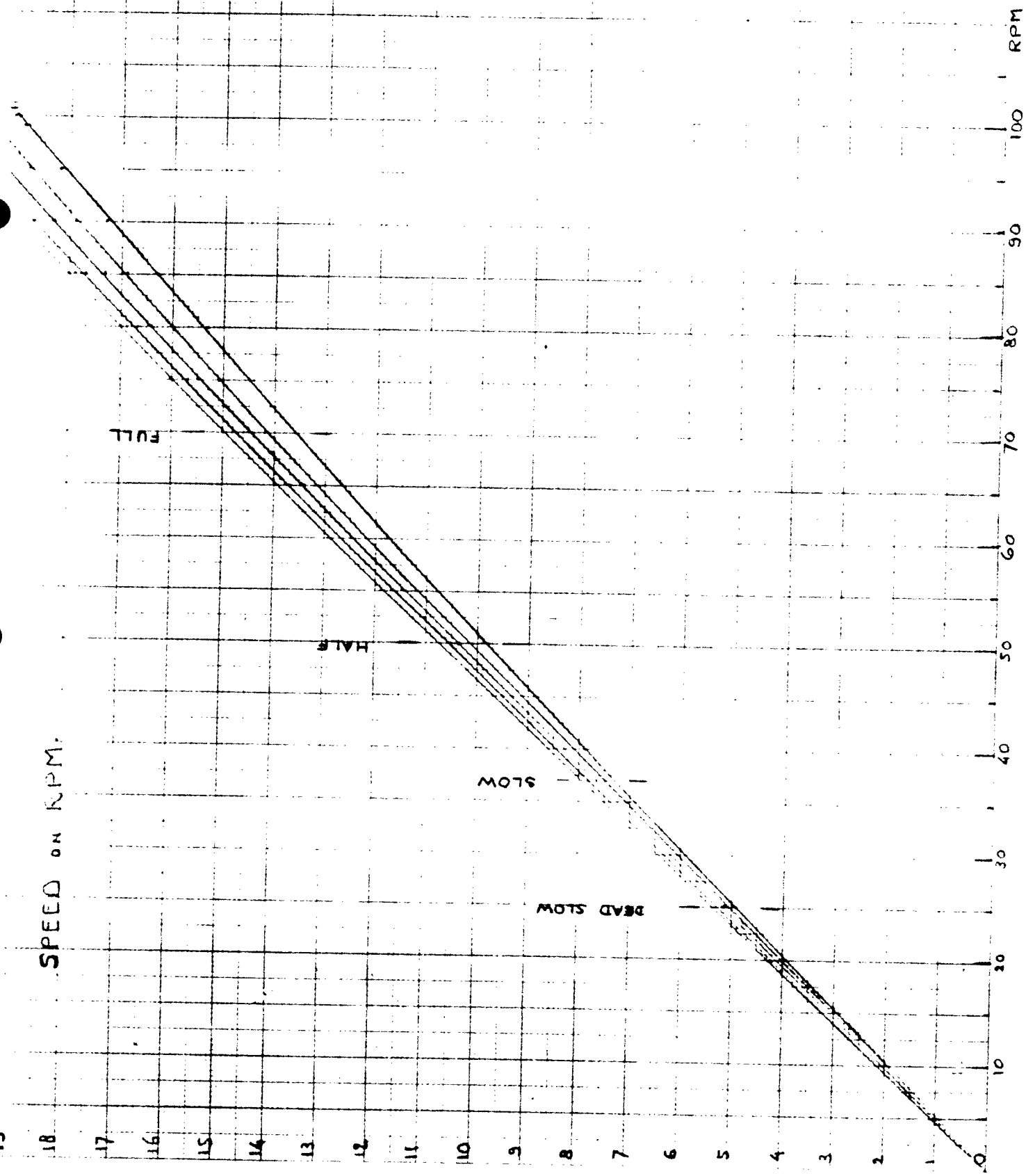
HHI 225/6 - 1983

22.3.18 (=vp 23)

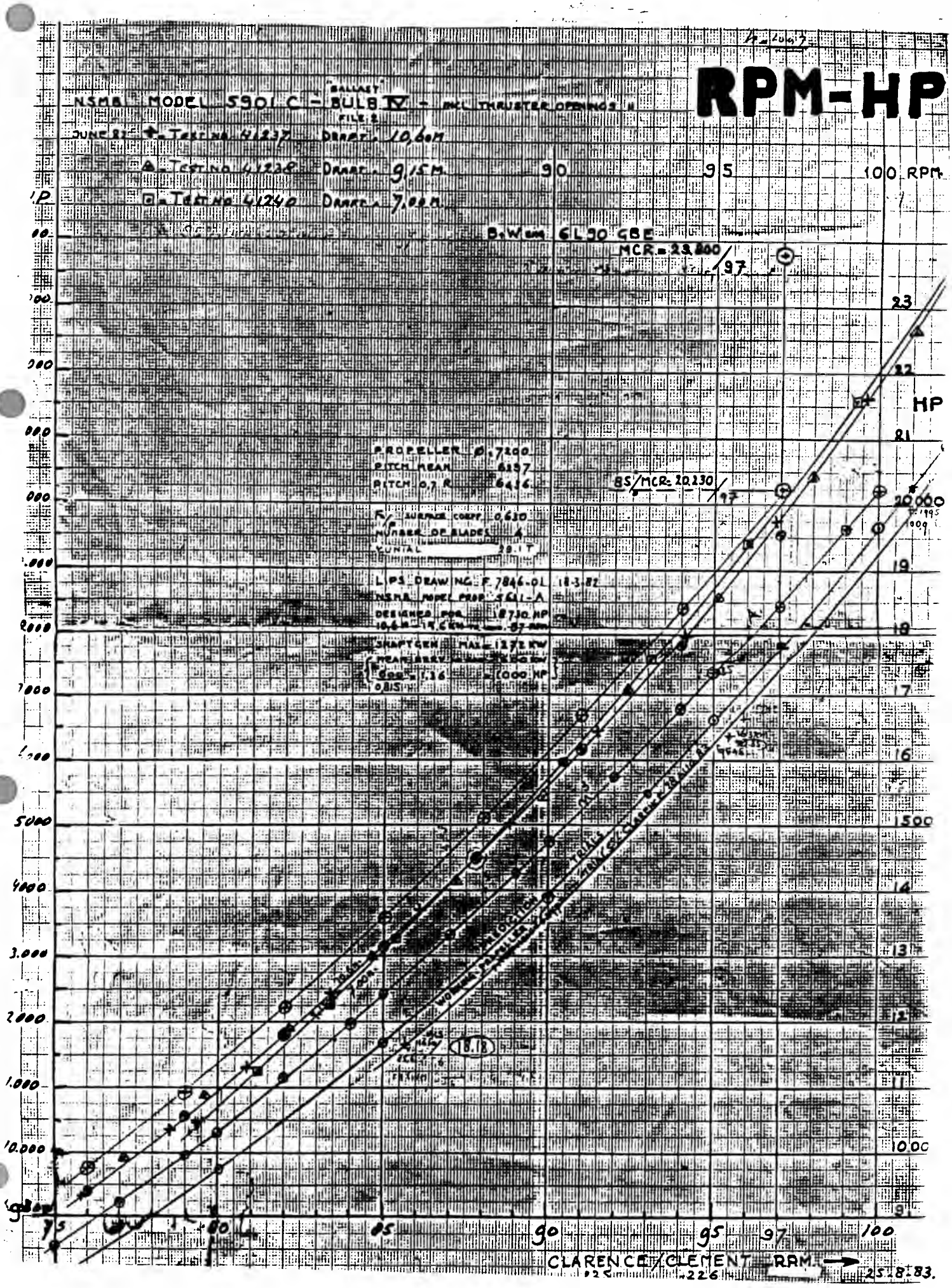
22

SPEED ON RPM.

6% SERVICE ALLOWANCE ON HP TRIALS



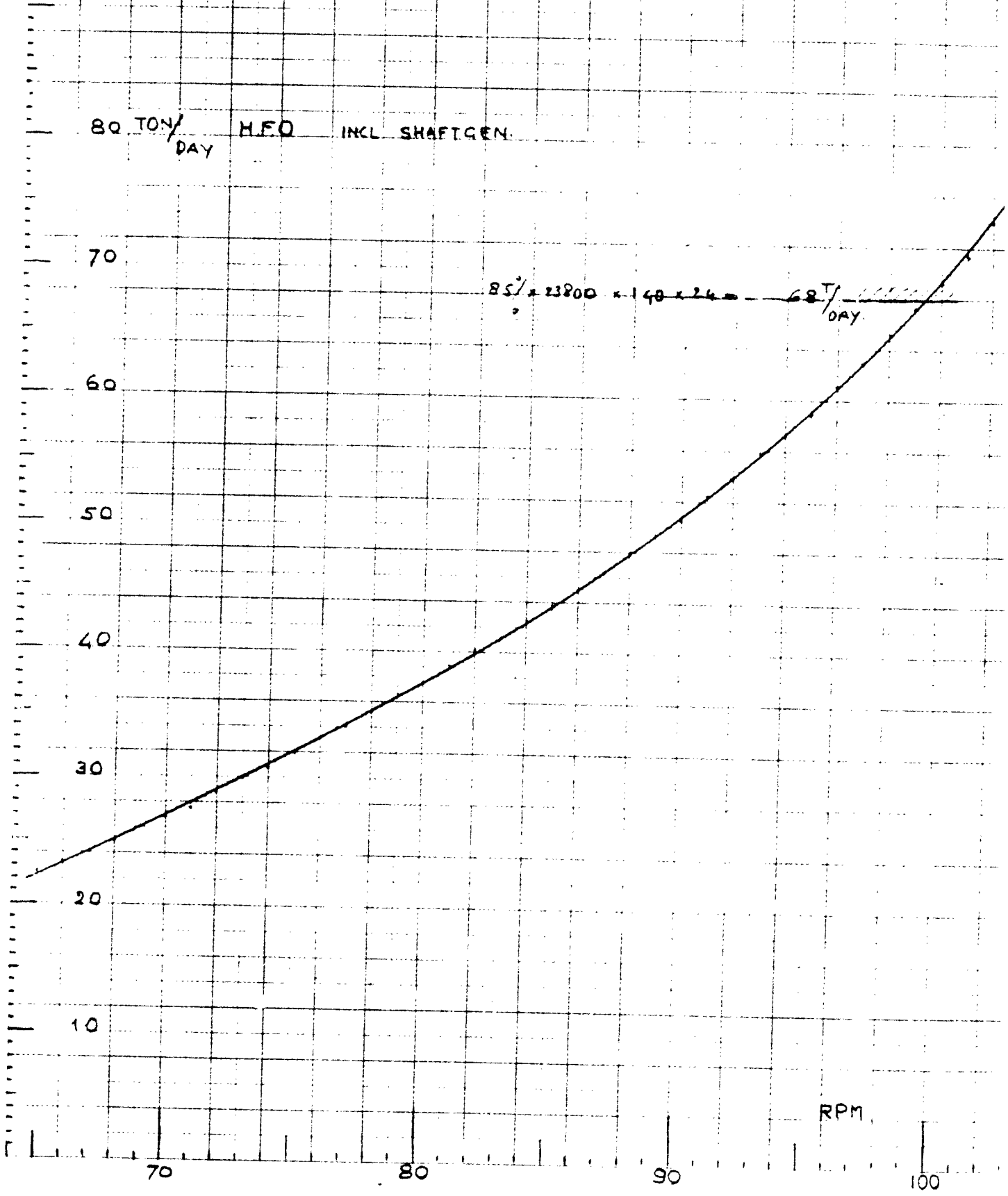
HH 25/6 -1983

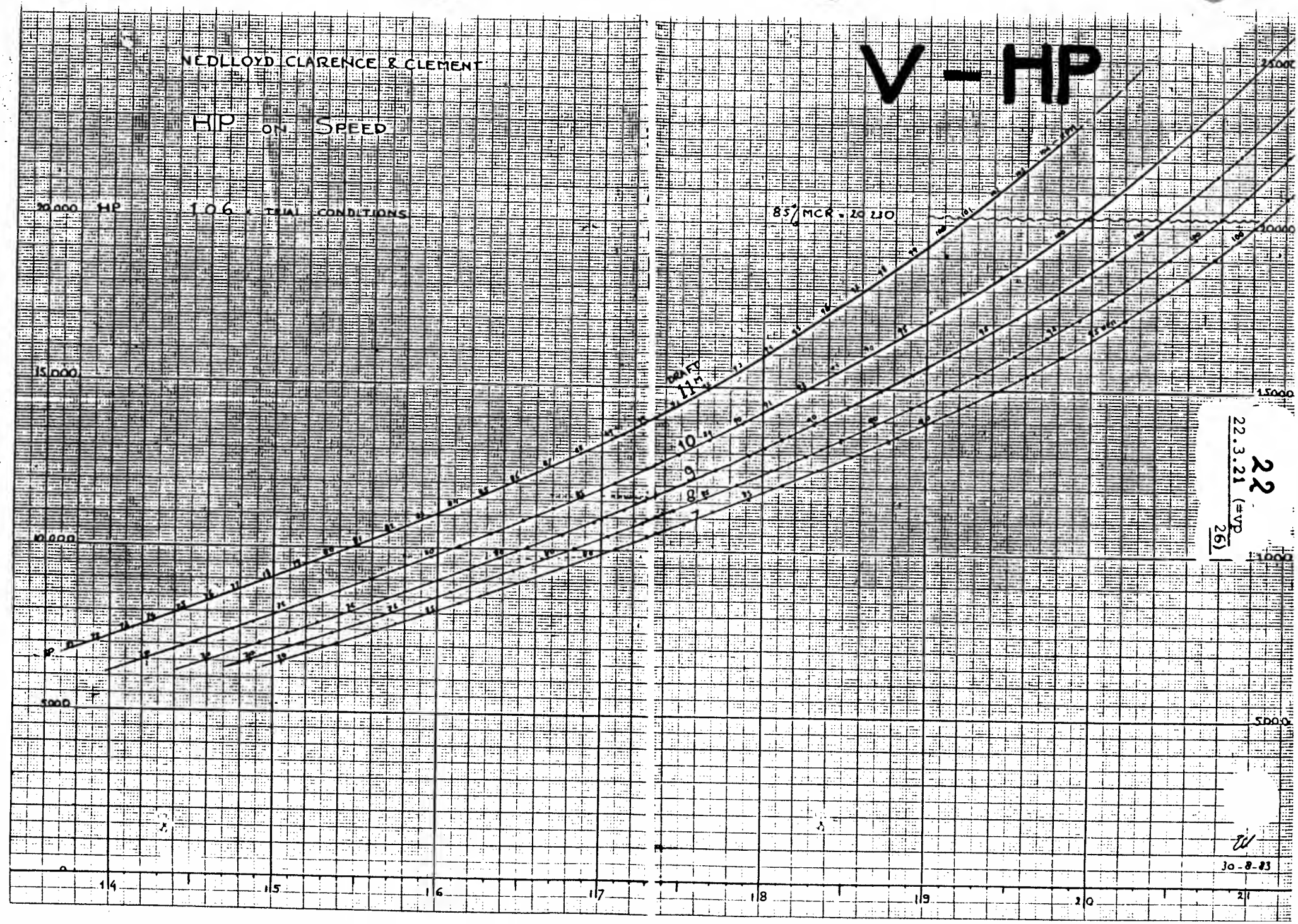

RPM-HP
NSMB MODEL 5901 C - BALLAST BULB IV - INCL THRUSTER OPENINGS
FILE 2
JUNE 82 - TEST NO 41237 DRAFT 10.60M
TEST NO 41238 DRAFT 9.15M
TEST NO 41240 DRAFT 7.00M
B&W GEN 6L30 GBE
MCR = 23,800
85 MCR = 20,230
PROPELLER Ø 7200
PITCH MEAN 6257
PITCH 0.7 R 6436
NUMBER OF BLADES 4
LPS DRAWING F 7846-01 18.3.82
DESIGNED FOR 18730 HP
SHAFT GEN MAX 1272 KW
90 95 100 RPM
HP
CLARENCE/CLEMENT RAM
25.8.83.

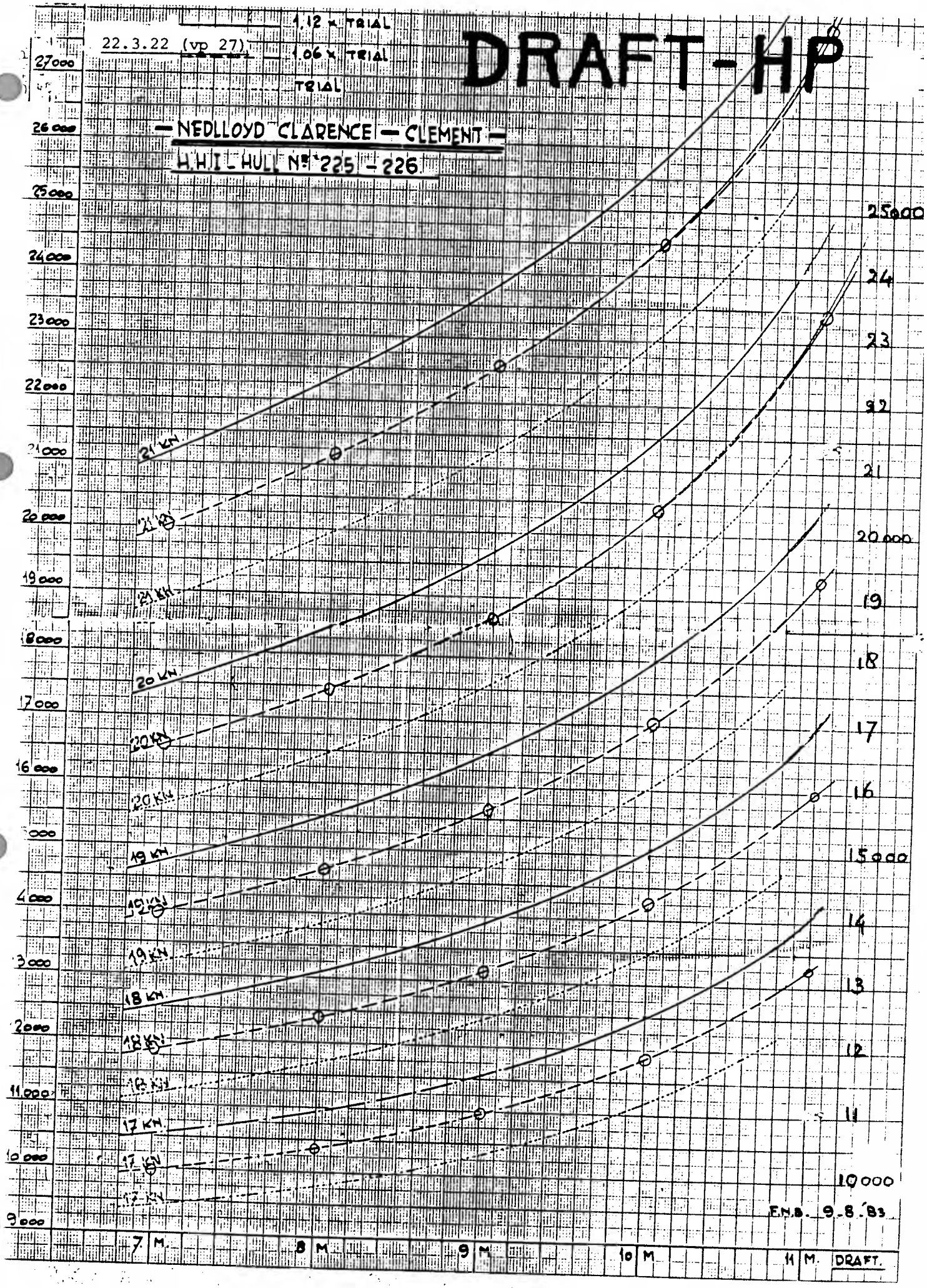
N. CLARENCE & CLEMENT

FUEL CONSUMPTION ON R.P.M.

6% ALLOWANCE ABOVE TRIALS FOR SERVICE




NEDLLOYD CLARENCE & CLEMENT
HP ON SPEED
V - HP
20000 HP
15000
10000
5000
0
106 % TRIAL CONDITIONS
85% MCR = 20 210
DRAFT
11
10
9
8
7
14
15
16
17
18
19
20
21
25000
20000
15000
10000
5000
22
22.3.21 (= vp 26)
30-8-83


DRAFT - HP
22.3.22 (vp 27)
1.12 × TRIAL
1.06 × TRIAL
TRIAL
— NEDLLOYD CLARENCE — CLEMENT —
H.H.I. HULL Nr 225 - 226
21 KN
21 KN
21 KN
20 KN
20 KN
20 KN
19 KN
19 KN
19 KN
18 KN
18 KN
18 KN
17 KN
17 KN
17 KN
27000
26000
25000
24000
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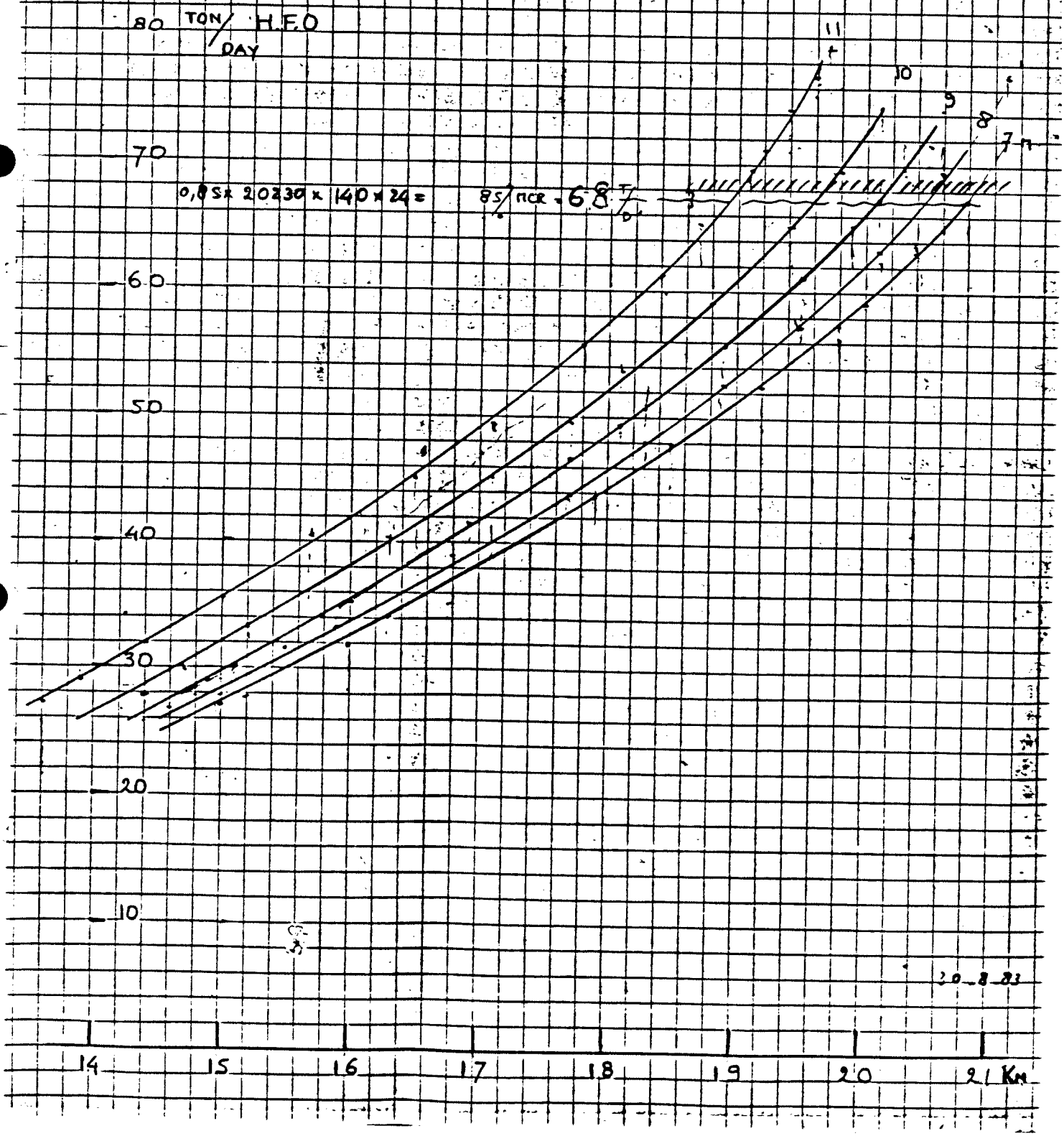
20000
19000
18000
17000
16000
15000
14000
13000
12000
11000
10000
9000
25000
24
23
22
21
20000
19
18
17
16
15000
14
13
12
11
10000
7 M.
8 M
9 M
10 M
11 M.
DRAFT.
E.N.B. 9.8.'83

NEDLLOYD CLARENCE & CLEMENT

FUEL CONSUMPTION PER DAY (INCL 600 KW SHAFTGENERATOR)

1.06 x TRIAL CONDITIONS = 6% SERVICE ALLOWANCE

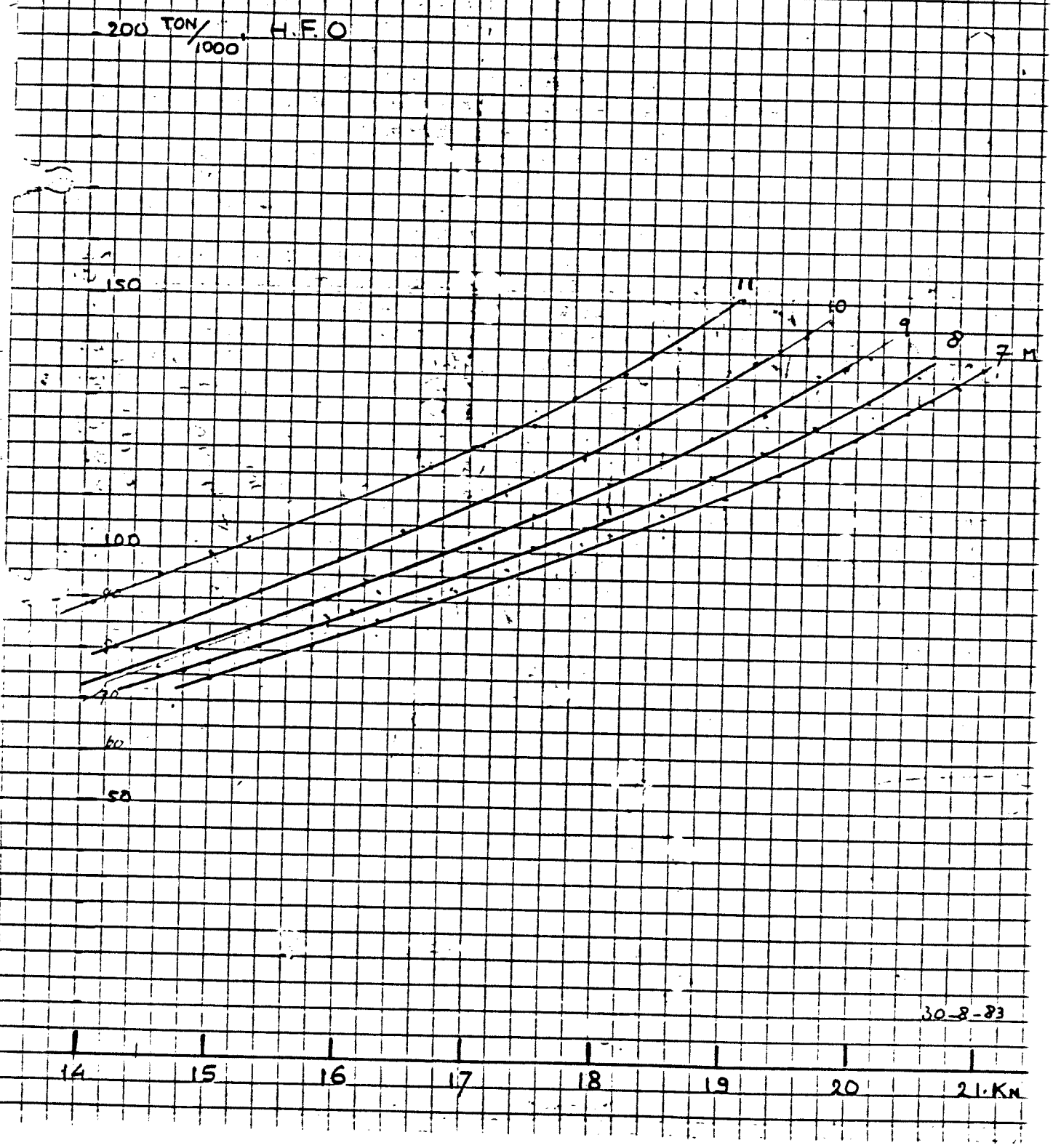


22.2.24/+ (=vp 29)

NEDLLOYD, CLARENCE & CLEMENT

FUEL CONSUMPTION PER 1000 MILES (INCL 600KW SHAFTGEN)

1.06 x TRIAL CONDITIONS ! = 6% SERVICE ALLOWANCE



"NEDLLOYD CLARENCE" – HHI HULL Nº 225 | B

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 7 M

R.P.M	KN.	A 600 KW (= 1000 PK) + SHAFTGENERATOR – EXCLUDING REEFERS.				
DEADSL. 25	5,4	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW. 37	8,0					
HALF. 50	10,8	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
65	14.0	—	—	—	—	
66	2	5220	6220	151	22.5	
67	4	5460	6460	151	23.4	
68	6	5710	6710	150	24.2	
69	8	5970	6970	150	25.1	
FULL 70	15.0	6240	7240	150	26.1	
71	2	6520	7520	149	26.9	75
72	4	6800	7800	149	27.9	76
73	6	7090	8090	149	28.9	77
74	8	7390	8390	149	30.0	79½
75	16.0	7700	8700	148	30.9	81
76	4	8020	9020	148	32.0	83½
77	3	8340	9340	148	33.2	85
78	5	8670	9670	147	34.1	86½
79	7	9020	10020	147	35.4	88
80	9	9370	10370	147	36.6	90
81	17.1	9730	10730	146	37.6	92
82	3	10100	11100	146	38.9	94
83	5	10480	11480	146	40.2	96
84	7	10860	11860	145	41.3	98
85	9	11260	12260	145	42.7	100
86	18.1	11670	12670	144	43.8	102
87	3	12090	13090	144	45.2	104
88	5	12510	13510	143	46.4	106
89	8	12950	13950	143	47.9	108
90	19.0	13400	14400	143	49.4	110
91	2	13890	14890	142	50.7	112
92	4	14400	15400	142	52.5	114
93	6	14930	15930	141	53.9	116
94	8	15480	16480	141	55.8	118
95	20.0	16050	17050	140	57.3	120
96	2	16620	17620	140	59.2	123
97	4	17210	18210	140	61.2	126
98	6	17810	18810	140	63.2	129
99	8	18420	19420	140	65.3	132
100	21.0	19040	20040	140	67.3	135
101	2	19660	20660	140	69.4	138
102	4	20290	21290	140	71.5	
103	5	20920	21920	141	74.2	
104		21560	22560	142	76.9	
105		22200	23200	142	79.1	

"NEDLLOYD CLARENCE" – H.H.I HULL Nº 225 — B

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 8 m

R.P.M	KN.	A 600 KW (= 1000 PK) + SHAFTGENERATOR – EXCLUDING REEFERS.				
DEADSL. 25	5.3	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW. 37	7.8					
HALF. 50	10.6	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
		—	—	—	—	
65	13.7	5220	6220	151	22.5	
66	9	5460	6460	151	23.4	
67	14.1	5710	6710	150	24.2	
68	3	5970	6970	150	25.1	
69	5	6240	7240	150	26.1	
FULL 70	14.7	6520	7520	149	26.9	76
71	14.9	6800	7800	149	27.9	77
72	15.1	7090	8090	149	28.9	78
73	3	7390	8390	149	30.0	80
74	5	7700	8700	148	30.9	82
75	7	8020	9020	148	32.0	84
76	9	8340	9340	148	33.2	86
77	16.1	8670	9670	147	34.1	88
78	3	9020	10020	147	35.4	90
79	5	9370	10370	147	36.6	92
80	7	9730	10730	146	37.6	94
81	9	10100	11100	146	38.9	96
82	17.1	10480	11480	146	40.2	98
83	3	10860	11860	145	41.3	100
84	5	11260	12260	145	42.7	102
85	7	11670	12670	144	43.8	104
86	9	12090	13090	144	45.2	106
87	18.1	12510	13560	143	46.4	108
88	3	12950	13950	143	47.9	110
89	5	13400	14400	143	49.4	112
90	7	13890	14890	142	50.7	114
91	9	14400	15400	142	52.5	116
92	19.1	14930	15930	141	53.9	118
93	3	15480	16480	141	55.8	120
94	5	16050	17050	140	57.3	122
95	7	16620	17620	140	59.2	125
96	9	17210	18210	140	61.2	128
97	20.1	17810	18810	140	63.2	131
98	3	18420	19420	140	65.3	134
99	5	19040	20040	140	67.3	137
100	6	19660	20660	140	69.4	140
101	8	20290	21290	140	71.5	
102	9	20920	21920	141	74.2	
103	21.1	21560	22560	142	76.9	
104	2	22200	23200	142	79.1	
105		22850	23850	143	81.9	

"NEDLLOYD CLARENCE" – H.H.I HULL Nr 225 — B

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 9 M

R.P.M	KN.	+ SHAFTGENERATOR A 600 KW (= 1000 PK) – EXCLUDING REEFERS.				
DEADSL .25	5.2	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW .37	7.7					
HALF .50	10.4	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000M
65	13.5	5220	6220	151	22.5	
66	7	5460	6460	151	23.4	
67	9	5710	6710	150	24.2	
68	1	5970	6970	150	25.1	
69	3	6240	7240	150	26.1	
FULL 70	14.5	6520	7520	149	26.9	76
71	7	6800	7800	149	27.9	78
72	9	7090	8090	149	28.9	80
73	15.1	7390	8390	149	30.0	82
74	3	7700	8700	148	30.9	84
75	5	8020	9020	148	32.0	86
76	6	8340	9340	148	33.2	88
77	8	8670	9670	147	34.1	90
78	16.0	9020	10020	147	35.4	92
79	2	9370	10370	147	36.6	94
80	4	9730	10730	146	37.6	96
81	6	10100	11100	146	38.9	98
82	8	10480	11480	146	40.2	100
83	9	10860	11860	145	41.3	102
84	17.1	11260	12260	145	42.7	104
85	3	11670	12670	144	43.8	106
86	5	12090	13090	144	45.2	108
87	7	12510	13560	143	46.4	110
88	9	12950	13950	143	47.9	112
89	18.1	13400	14400	143	49.4	114
90	3	13890	14890	142	50.7	116
91	5	14400	15400	142	52.5	118
92	7	14930	15930	141	53.9	120
93	9	15480	16480	141	55.8	122
94	19.1	16050	17050	140	57.3	125
95	3	16620	17620	140	59.2	128
96	5	17210	18210	140	61.2	131
97	7	17810	18810	140	63.2	134
98	9	18420	19420	140	65.3	137
99	20.1	19040	20040	140	67.3	140
100	3	19660	20660	140	69.4	143
101	5	20290	21290	140	71.5	
102	6	20920	21920	141	74.2	
103	8	21560	22560	142	76.9	
104	9	22200	23200	142	79.1	
105		22850	23850	143	81.9	

"NEDLLOYD CLARENCE" — H.H.I HULL Nr 225 — B

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 10 M

R.P.M	KN	A 600 KW (= 1000 PK) + SHAFTGENERATOR - EXCLUDING REEFERS.				
DEADSL. 25	5.1	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW. 37	7.5					
HALF. 50	10.1	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
		—	—	—	—	
65	13.2	5220	6220	151	22.5	
66	4	5460	6460	151	23.4	
67	6	5710	6710	150	24.2	
68	8	5970	6970	150	25.1	
69	14.0	6240	7240	150	26.1	
FULL 70	14.2	6520	7520	149	26.9	79
71	4	6800	7800	149	27.9	81
72	5	7090	8090	149	28.9	83
73	7	7390	8390	149	30.0	85
74	9	7700	8700	148	30.9	87
75	15.1	8020	9020	148	32.0	89
76	2	8340	9340	148	33.2	91
77	4	8670	9670	147	34.1	93
78	6	9020	10020	147	35.4	95
79	8	9370	10370	147	36.6	97
80	16.0	9730	10730	146	37.6	99
81	1	10100	11100	146	38.9	101
82	3	10480	11480	146	40.2	103
83	5	10860	11860	145	41.3	105
84	7	11260	12260	145	42.7	107
85	9	11670	12670	144	43.8	109
86	17.1	12090	13090	144	45.2	111
87	3	12510	13560	143	46.4	113
88	5	12950	13950	143	47.9	115
89	7	13400	14400	143	49.4	117
90	9	13890	14890	142	50.7	119
91	18.1	14400	15400	142	52.5	121
92	3	14930	15930	141	53.9	123
93	5	15480	16480	141	55.8	125
94	7	16050	17050	140	57.3	128
95	8	16620	17620	140	59.2	131
96	19.0	17210	18210	140	61.2	134
97	2	17810	18810	140	63.2	137
98	4	18420	19420	140	65.3	140
99	6	19040	85% 20040	140	67.3	143
100	8	19660	20660	140	69.4	146
101	20.0	20290	21290	140	71.5	
102	1	20920	21920	141	74.2	
103		21560	22560	142	76.9	
104		22200	23200	142	79.1	
105		22850	23850	143	81.9	

"NEDLLOYD CLARENCE" – H.H.I HULL Nº 225 | B

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT **11** M

R.P.M	KN.	A 600 KW (= 1000 PK) + SHAFTGENERATOR – EXCLUDING REEFERS.				
DEADSL .25	4.9	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW .37	7.3					
HALF .50	9.8	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000M
65	12.6	5220	6220	151	22.5	
66	8	5460	6460	151	23.4	
67	13.0	5710	6710	150	24.2	
68	2	5970	6970	150	25.1	
69	4	6240	7240	150	26.1	
FULL 70	13.6	6520	7520	149	26.9	82
71	8	6800	7800	149	27.9	84
72	9	7090	8090	149	28.9	86
73	14.1	7390	8390	149	30.0	88
74	3	7700	8700	148	30.9	90
75	4	8020	9020	148	32.0	92
76	6	8340	9340	148	33.2	94
77	8	8670	9670	147	34.1	96
78	15.0	9020	10020	147	35.4	98
79	1	9370	10370	147	36.6	100
80	3	9730	10730	146	37.6	102
81	5	10100	11100	146	38.9	104
82	7	10480	11480	146	40.2	106
83	9	10860	11860	145	41.3	108
84	16.1	11260	12260	145	42.7	110
85	3	11670	12670	144	43.8	112
86	5	12090	13090	144	45.2	114
87	7	12510	13510	143	46.4	116
88	9	12950	13950	143	47.9	118
89	17.1	13400	14400	143	49.4	120
90	3	13890	14890	142	50.7	122
91	5	14400	15400	142	52.5	124
92	7	14930	15930	141	53.9	127
93	8	15480	16480	141	55.8	130
94	18.0	16050	17050	140	57.3	133
95	2	16620	17620	140	59.2	136
96	4	17210	18210	140	61.2	139
97	6	17810	18810	140	63.2	142
98	7	18420	19420	140	65.3	145
99	9	19040	20040	140	67.3	148
100	19.1	19660	20660	140	69.4	151
101	2	20290	21290	140	71.5	
102	4	20920	21920	141	74.2	
103	6	21560	22560	142	76.9	
104	7	22200	23200	142	79.1	
105		22850	23850	143	81.9	

GENERAL DESCRIPTION

"Nedlloyd Clarence" - "Nedlloyd Clement"

At the shipyard of Hyundai Heavy Industries in Ulsan, South Korea, two cellular container vessels are in the building stage, which are expected to be delivered end of August and end of November respectively.

Newbuilding 225 - "Nedlloyd Clarence"
Newbuilding 226 - "Nedlloyd Clement"

Main Dimensions

Length over all	-	210,00 m
Length between perp.	-	202,00 m
Breadth	-	32,24 m
Depth to upperdeck	-	18,80 m
Design draught	-	10,60 m
Max. draught	-	12,00 m

Deadweight

At the design draught of 10,60 m the deadweight will be abt. 30.500 tons.

Container capacities

The holds are equipped with container guides for carrying 10 containers in breadth and 7 in height.
On deck containers can be carried 13 in breadth and 3 to 4 in height.

	"Nedlloyd Clarence"	"Nedlloyd Clement"
Under deck	1150 teu	1150 teu
On deck	853 teu	847 teu
Total	2003 teu	1997 teu

The hold capacity is divided into :

160 - 40' cntrs = 320 teu = 28%	164 - 40' cntrs = 328 teu = 28½%
830 - 20' cntrs = 830 teu = 72%	822 - 20' cntrs = 822 teu = 71½%

On deck the container positions are interchangeable between 20 ft and 40 ft, except for the most aftermost bay which is for 20 ft only.

The vessels are based on containers with a height of 8'-6".
Both vessels have positions and electrical connections for 187 resp. 163 integral reefer containers, while "Nedlloyd Clement" will also be fitted with a ducted air cooling and distribution system for refrigerating of 116 porthole containers of 40'

Loading gear

Each vessel will be equipped with two 8 tons provision cranes, while on "Nedlloyd Clement" a 45 tons travelling gantry crane will be fitted forward of the deckhouse, for handling containers.

Main engine

The vessels will have a main diesel engine of the type B & W 6L 90 GBE, developing a maximum continuous output of 23.800 HP. The normal output (85%) is 20.230 HP at 97 RPM.
A shaft generator with an output of 1200 kW will be fitted.

Speed

The service speed at design draught of 10,60 m, at normal output, with shaft generator in operation, will be abt. 18,7 knots.

The vessels will be equipped with a bow thruster and a stern thruster with an output of 1200 kW each for easier manoeuvring.

Accommodation

Accommodation is available for a crew of 26 persons, while extra cabins are available for pilot, repair crew and Suez crew.
All crew members are situated in single cabins, each with private sanitary equipment.
Cabins and public spaces are of modern design and provided with airconditioning.
A covered swimming pool is available.

Miscellaneous

Sophisticated nautical equipment will be fitted.

DESIGN PHILOSOPHY REGARDING NEDLLOYD 'UNIVERSAL' CONTAINERSHIP.

Capacity and speed.

The commercial departments required a vessel of about 1800 TEU of 14 tonnes per TEU and 18-18.5 knots service speed and being as flexible as possible for interchange between the company's various services. Contrary to earlier periods, this vessel was not to be designed for a certain trade but was to be utilisable for worldwide container transport. Many times Nedlloyd experienced reshuffling her ships in co-operative services.

Length Over All.

Almost all container terminals can accommodate ships of 210 M length over all - except some West and east African ports.(The latter are to be served by a smaller ship type such as "Nedlloyd Van Neck type of 182.50 M overall and 1500 TEU capacity.)

- Good manoeuvring properties, however, are required, especially in the ports on the South American West coast where the vessel has to turn around within 300 metres length by her own means.
- For this reason a bowthruster and sternthruster were fitted which also have the advantage of shortening the roundtrip time and reduction of deck-crew.

Breadth.

Max. Panama beam has been selected in order to obtain on the worldwide routes the maximum attainable stability which will allow for maximum container carrying capacity.

- This 'Universal' container is able to carry 7 tiers containers in the holds plus 4 tiers on deck, resulting in a carrying capacity of 2224 TEU at the maximum, at an average container-weight of 12 tonnes per TEU. Normal in-homogeneous stow and ahead-view requirements have been taken into account.
- Normally, the ship can carry at 10.60 M draft a number of 2000 TEU at 14 t.av.wt. in the hold and 12.1 t. on deck, which results in 13.2 t mean weight per slot. Herewith the stability is about 0.4 M metacentric height, being an acceptable value.

(Design background)

Hull form and propulsion.

A V-shaped forebody and a Pram-type afterbody (like those of old sailing ships and yachts) will result in an easily driven hull with maximum form stability and maximum deck space.

- The flow of water towards the propeller has been smoothened as much as possible - by scraping off some displacement - in order to obtain a good propulsion and a minimum of propeller-excited vibrations.
- The 'large' propeller at rather low revolutions will give a good propulsive efficiency.
- A fine underwater-afterbody, however, will automatically ask that engine room plus deckhouse are to be moved somewhat foreward.
- The reason for this is the space required for a sufficiently stiff engine foundation and also the trim-balance of the ship.
- Moreover, the fuel oil is to be carried in the forepart of the hull in order to sail on even keel. (Sailing with - say - about 3 feet astern trim on these vessels is very uneconomical, because ship's resistance increases enormously by the combined action of emerging bulb and immersing transom stern).
- For this reason - and also for keeping the hogging moments in the hull at a low(er) value - Nedlloyd decided on a general arrangement with engine room not too far aft (i.e. two hatch-lengths before the stern instead of only one as is the case on many a similar containership). With such arrangement the balance of the ship is better and she needs less water-ballasting in the forebody for compensating the astern trim.
- Meanwhile, by placing the deckhouse more forward, the vibration hindrance caused by the propeller will estimately become less.
- A small disadvantage is the loss of a few containers in holds. This loss was accepted by us because a very great part of the containerload is carried on deck.

Draft.

The scantlings are allowing for 12 metres draft.
However, this (= 40 feet) draft is too deep for an economical service condition, especially in view of immersion of the transom.

- The hull form will be optimal between 8.5 M and 10.5 M drafts. This mainly decided the transom depth and bulb diameter.

(Design background)

- Too many ships are designed with too big bulbs. Big bulbs will function as water'turbulating' brakes at the smaller service-draft ranges which quite often occur on coastal legs between the many ports.
- South American ports and Indonesian ports allow for 9.15M (30') Certain and 9.75M (32') which figures are limiting the cargo deadweight of the vessel.

Block coefficient

In order to have sufficient cargo deadweight at the smaller drafts, the block coefficient should not be too fine.

- Remunerativeness calculations in 1981 - at which time fuel prices and ships' prices were rather high - the optimum blockcoefficient for an 18½ knots service speed varied between 0.625 and 0.65.
- At the moment (June 1983) with fuel prices at somewhat lower level as well as ship's prices the optimum varies between 0.635 and 0.66 .
- Many containerships nowadays have too fine lines and (consequently) cannot carry weight from shoal ports !

Cubic capacity.

The containerslot capacity within the hull is fully utilized up-to the last centimetre.

Experience with containership-designs of the latest 10 years showed the optimum length of a cellular container hold to be 28.70 metres (== 35 frame spacings of 820 MM each).

The minimum required length of engine room and length of mooring space of the fore and aft decks are to be added to a number of cellular blocks and the overall-length is born. This L.o.a. in our opinion is a muchbetter value for defining a vessel than the old-fashioned L.p.p. (Length between the load-lines).

2000 TEU boxes - with 3 tiers above deck - on a 210-metres ship of Panama beam is a good capacity. However, it has to be realised that the boxes on the rear end of the ship (above the mooring winches) are NOT to be assessed as normal loaded containers. This is because hogging moment in the hull and astern trim of the vessel will limit the weight of these boxes to be very light ones only.

(Design background)

Seaworthiness.

The vessel accommodating 7 tiers containers within the hull has sufficient freeboard at service draft of 10.6 M and the range of stability is rather large. Thus, a low metacentric height (0.4M) is allowable and will give a gently moving ship.

- However, if only a small amount of containers is to be transported, the vessel is too stable because metacentric height is very great and as a result jerking rolling movements will occur at sea. The only answer to that problem is to pile the boxes up as high as possible and sail them mainly on the hatchcovers.
- Securing of the containers on deck should in general be done by means of twistlocks only (except when the topweight on the lowermost box of a stack exceeds 30 tons).
- Application of fin -stabilizers is considered to be not economical anymore. At the low(er) speed level the fins' surface would have to be large, the necessary larger size of the aparatus would be (more) expensive and the fuel bill considerably higher than formerly .
- At the other hand, 'steered antirolling tanks' would have to be very large sized and this would incur corrosion problems in the course of the 20 years' service.

Strength.

The structural 'sideboxes' of 2.35M width have already proved to be sufficiently stiff and strong on the larger sisterships and moreover the Length/Depth ratio of the 210M by 18.8M vessel is quite reasonable.

In this respect also the knuckle line at the forebody is to be mentioned as it accentuates the box-girder surrounding the hatch openings in the foreship.

- The sidebox creates axial stiffness against headseas and also will throw the water away from the vessel when the ship is pitching in headseas.
- We prefer this knuckle line more than breakwaters on deck for this reason. Also, it looks nice because this feature is a 'last rudiment' of the ship's sheer.

(Design background)

Hatch configuration. Reefers.

- With a view to flexibility in container stowage, the triple-hatch arrangement is to be preferred above the twinhatch type.
- When many ports are to be called at in one voyage, the twin-hatch vessel will incur much more reshuffling of deck-containers than the triple-hatch configuration. This will cost much more than moving of covers only.
- Structural-wise, the triple-hatch arrangement creates two 'violin strings' over the whole length of the vessel, acting as longitudinal strength members (and may fall less heavy than the single-string of a twin-hatch arrangement vessel).
- Contrary-wise, the hatchcoaming structure above the deck should be of the interrupted type in order to avoid high stresses.
- Transverse members consist of wide and slender structures (cross-overs). The widest ones are accommodating ventilation ducts and stairways.
- By adopting this configuration it is always possible to provide the ship with 'integral' type reeferbox capacity underdeck. Cooling these containers - or rather their engines - can be executed by means of aircooling (pressure ventilation) only or by means of airblowers and water-piping (connected b.m.o. flexible hoses to a shipboard system).
- Nedlloyd Newbuilding department has found from experience that almost every newbuilding vessel will have to be provided - once she is in service - with a number of some 20 to 100 reefers slots extra at a later date. For such an exercise the Universal containervessel stands wide open.
- The 'integral reefer' system applied on this vessel in general has much more flexibility than the ducted (Porthole) system. With the latter, the cooling installation is shipborne - which makes the ship more expensive as to investment costs .
At the other hand, the flexible worldwide applicable 'integral' reefersystem means a higher investment cost in containers.
'Ducted' systems are more economical where a high-volume reefer trade exists and ships are to remain in the same service for much of their lifetime, so there where flexibility counts little.

(Design background)

General arrangement. (Fuel and waterballast location).

- As has already been explained, location of the engineroom and deckhouse is 'semi-aft' with the fuel oil storage 'semi-forward'. As high-viscosity fuel oil is used, there exists the problem of pumping the fuel from tanks forward to the engineroom aft, when needed.
  For this reasons, the fueloil pipelines run through pipe tunnels located in the Port and Starboard corners of holds, between sidewall and tanktop. The lines can be heated by means of steam lines lying next to the fuel lines and having been wrapped around by insulating material.
- The weight of the consumed fuel can be compensated by taking waterballast in doublebottom and side tanks in the vicinity of the fuel oil tanks.
  Waterballast tanks in the wings of the upper part of the foreship will make it possible to compensate for trim astern without increasing the transverse stability of the vessel too much.
  (On many a containervessel the stability increases too excessively when - in certain conditions - forepeak and deeptank Nr. 1 are used for this purpose.)

Engine installation philosophy.

- Maintenance costs are directly related to the amount of bolts and nuts utilized and for this reason a slow-turning 6-cylinder diesel engine, longitudinal scavenging type, has been installed together with a shaft-generator, thyristor-controlled.
- Maximum reliability has been strived after in utmost simplicity.
- The aim is not to use separate diesel-engine generators at sea. However, during manoeuvring with the bow and sternthruster at work or standing-by, all three diesel-engined generators should be running.
- It is the opinion of Nedlloyd Newbuilding department that it is not a paying proposition to have an exhaustgas turbo-generator installed on a containervessel as such a ship has to serve many ports (with short sea-stretches in-between) and has often to sail many coastal stretches at reduced speed.
  (In this respect we are in favour, however, of an exhaustgas generator installation on a tanker and bulkcarrier which ships - normally - sail on long stretches and at full power.)

(Design background)

- Centralized control has been adopted in the wheelhouse and is arranged in such a way that the nautical duty officer is not disturbed in the exercise of his duty by less-urgent engineroom alarms. During critical periods, the Chief Engineer is (to be) in the wheelhouse.
- The engineroom control station is minimized in layout and is situated near the workshop where engineers do their maintenance jobs. The space is air-conditioned and kept at low noise level.

Mooring arrangement.

- All wires are to be fed directly from the large-diameter winch-drums via fairleads of the large-radius solid cast steel Panama type towards the quay bollards.
- Four (4) mooring winches forward and four (4) mooring winches aft are to be handled by two men forward and two men aft. Bow- and sternthrusters are used in combination with the mooring cables. The vessel can in principle move transversely on its own power and could press herself against the quaywall whilst meantime the wires can be 'made fast'. After having been made fast, the winches are set to 'self-tensioning'.

  Two ways of mooring are feasible, i.e.:-

  Alternative (1)... 3 wires plus 1 spring, forward as well as aft.

  Alternative (2)... 2 wires plus 2 springs, forward and aft and remotely controlled from bulwark.

  The steel wires system is adopted because of low maintenance load on the wires and on the fixed fairlead-chocks.
- The anchor type is of Spek design. It is flush mounted in an anchor pocket and always fits well when 'coming in'. It never slams by the action of headseas.

(Design background)

Deck layout in view of maintenance.

- As simple as possible...

Smooth flat coamings and closed box-foundations. Whatever labour-requiring can be omitted should be omitted .

Grit-blasting of weather-exposed surfaces after finishing construction and application of zinc-epoxy plus chlorinated rubber paints is a 'must' for the shipowner if he intends to sail with a minimum number of crew (18).

Double bottom waterballast tanks. Preparation.

Grit-blasting of welds and sweep-blasting of the remaining surfaces has been carried out in the block-stage and

2 coats of solvent type coaltar epoxy paint, i.e. 2 x 175 micron layers were applied.

After our experience of troubles recently met, Nedlloyd nowadays is in favour of tank conservation in the final stage to be carried out as follows:-

- Application of 1 coat of 350 micron thickness of non-solvent coaltar-epoxy type paint by means of high-pressure spraying apparatuses.

This system saves manhours, has no intercoat adhesion problems and does not create an explosion hazard.

Underwater.

Application of self-eroding (SPC of International) acrylic resin copolymer pigmented with organic tin antifouling paint, 3 coats each 100 micron. This has proven to keep the vessel free from algae and barnacles.

SPECIAL FEATURES OF THE SECOND VESSEL.

- The second vessel, to be named Nedlloyd Clement, (Yard nr. 226) is intended for the EUROSAL service, carrying general cargo in containers Southbound from western Europe via the Panama Canal to Colombia, Equador, Peru and Chile.
- Northbound, the vessel has to transport copper and fruit from Chile and coffee from Colombia - and eventually bananas from Equador.
- At this moment, only the Equadorian port of Quayaquil is fitted with a container gantry crane. Probably San Antonio in Chile will have a container shorecrane at the beginning of 1984.

Because of this situation it was decided by the EUROSAL principals to provide each containervessel with a shipboard container gantry crane. Should - in future - all ports be equipped with cranes, then the shipboard cranes could be done away with.

It is an advantage of the gantry system that ships can be geared or de-geared within say one day- which is an interesting feature with a view to interchangeability between services.

At the other hand, the reliability of shipboard gantries 10 years ago was not very good. However, we trust that this has been improved meantime.

Copper ingots to be carried from Chile.

- This type of cargo will give a lot of problems on the Eurosal cellular containervessels.
- Logically, carriage of this cargo should also be done in containers.

However, extra stowage costs and damage feared to be caused by the ingots in the cargo containers stand in the way of containerizing this cargo as this all will/might add too much to the transportation costs.

Carriage of copper ingots by means of Flatracks or Hälfheight ISO-bins should technically be the best way out. However, commercial departments involved are afraid for deadheading of the 'special' implements as yet.

Nedlloyd's opinion tends towards the utilisation of the flatracks Southbound for drums and steel plates.

Up to Writingdate (June '83) it is the intention to carry 75% of the copper ingots 'breakbulk'-fashion on the tanktopdeck of the ships. The copper ingots are to be loaded onto the ship - by means of shore-based cranes (12 t. cap.) - while bundled in chain slings.

Forklift trucks in the ship must stow the bundles crossover from Portside to Starboard-side between longitudinal bulkheads whilst manoeuvring between the bottom ends of the container cellguides.

Lashing/shoring of the bundles might be necessary and there is some risk of damage to the cell guides.

- Acceptance of copper in bundles on a container terminal in Europe is a (near) impossibility and a special copper receiving terminal has to be found nearby.

Reefer cargo.

With a view to the foreseeable carriage of bananas from Equador and of other fruit kinds from Chile in 40-ft insulated porthole-type boxes, this ship has been provided with a ducted cooling air system as well. Freon 22 will be used as activator through its direct expansion.

Possibly these well ventilated containers can be utilized also for the carriage of coffee beans.

23.0.0.

CELLULAR CONTAINER VESSELS .

- GENERAL REMARKS.

CELLULAR CONTAINER

(A) THE CELLULAR VESSEL.

For trunk services we need large cellular vessels:-

- Ships calling main ports only around the world.
- Feeder services to these main ports are to be carried out by smaller cellular vessels, local roro, or local general cargo ships.
- A short port time is feasible because of the big unit load operation by the shore equipment.
- A small integrated crew is feasible because of no commitment with cargo operations and not much maintenance thanks to the simplicity of the vessel.
- The smaller ships to be of Single-Screw/Diesel propulsion
- The larger ships Single/Twin-screw :Diesel propulsion
- Speed 18-21 knots
- Block coefficient between 0.59 and 0.68
- Pontoon covers / Triple hatch preferable.
- Tiers on covers, 3-high to 4-high / 60 tons per 20' stack. For stacks of up to 45 tons load, twistlock securing only should be feasible, (in that case the maximum load onto the lowermost box should not be over 30 ton at present.)
- Reefer cells in holds: As regards the type of the system to be applied on board in view of cost and maintenance we suggest
  Below 200 TEU ... Integral reefers, air cooled or water cooled.
  Over 200 TEU .... Insulated boxes connected to ductsystem with cooled air of the vessel (Conair/Stal).
- Actually a "container ship"is an "open" bulkcarrier with cells for boxes. The hull, with guides and simple pontooncovers is relatively cheap. The double hull construction is simple and straight-forward. Critical are stress concentrations at hatch-corners by bending and torsion.
- The engine installation might be expensive if a high speed is required and a large amount of K.W. is wanted to supply reefers/ bow & sternthrusters.
- Vibration and noise, caused by the propeller, are quite small if the superstructure is in a location about ¼L (i.e. 2 hatch-covers' length) from aft.
- From vibrations point-of view twinscrew is preferable to single screw.

Twin-screw moreover allows for the engine room to fall shorter, resulting in a larger payload space.

ITEMS OF INTEREST IN DEVELOPING CONTAINER CARRYING MODERN VESSELS.

Triple hatch superior to twin hatch for ship operation:
- less shiftings of deck containers in Port (ship -ship or ship - shore - ship).
- less torsional problems.
- smaller covers (mind obstruction if laid down ashore).

N.B. Shipyards prefer twinhatch in view of costs.

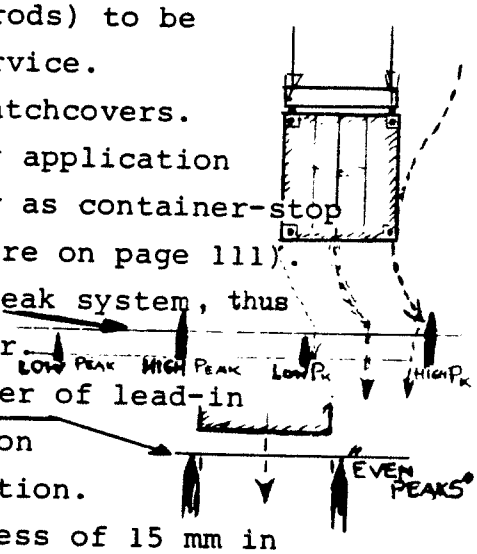
Cover sufficiently strong: - 60 ton per 20 ft stack.
- 90 ton per 40 ft stack.

Lashings: - twistlock only up to 45 ton per 20 ft stack, deck box stacking sequence 20-15-10 t/box preferable above 15-15-15 t/box upwards.
- single rod/turnbuckles 45-ton stack (20').
- high tensile rods (no twin rods) to be thoroughly tested before service.

Provisions for lashing heavy loads on hatchcovers.

Possibility of heavy loads on tanktop by application of sturdy latches in guides serving as container-stop at 2nd tier level (see bottom-picture on page 111).

Guide intakes preferably of "high-low" peak system, thus enabling crane driver to spot easier.
- in case of "even peak" system taper of lead-in to be 95 mm in transverse direction
  150 mm in longitudinal direction.
- guide profile to have min. thickness of 15 mm in upper part.



Guides for hatchcovers to be sufficiently high in order to prevent damage by swinging covers to railings, lampposts etc.

Supports of cover on coaming to be of sound construction! Many older container ships are suffering problems on this point.

Clearance between container and bridge front/funnel back to be at least 2400 mm at level of boom of terminal container crane and for high water springtide.

Clearance between first container and foremast at least 2400 mm - or/more in case of helicopter landing place.

Provisions, engine parts, spares, garbage to be in containers and handled by terminal container crane in order to prevent traffic jam on the quay beside the vessel.

- deckhouse and funnel to be one squeezed block (not 2 separate blocks) if container number capacity is paramount.

Aft deck, to have stowage arrangement for containers above mooring winches suitable for dangerous cargo and reefer containers so as to avoid these being shifted.

Reefer containers on covers: first tier only - excluding in outer row. Good access by platforms for repair/ maintenance to the boxes' individual cooling apparatus.

Reefer containers below deck (which is preferable to ship-planners) using Integrals:

- freshwater cooled + 15 m^3/min air per 20' box.
- aircooled only: 60 " " " " " " .

  This is possible but difficult to get down that amount of air).

When using porthole boxes which are connected to ship's cold air ducts by rubbercoupling:

- unflexible, either 20' x 8'
  or 20' x 8½'
  or 40' x 8½'

  attractive if ship is tied to the trade and large quantities of frozen or fruit are to be transported.

In case of tropical fruit it is imperative:

a. to cut fruit - say - one day before loading. Cooling down to be carried out on board immediately.

b. after discharge, to transport fruit (in moderate climate) without delay to customer.

In case a. and b. are not feasible, then cold stores ashore to be utilized to prevent heating-up.

Note: Porthole box cannot be used in sub-zero (Canadian , e.g.) ambient conditions when containing fruit, unless provided with a suitable clip-on conditioning apparatus. However, clip-on units - in practice - are rather a nuisance.

23.1.00

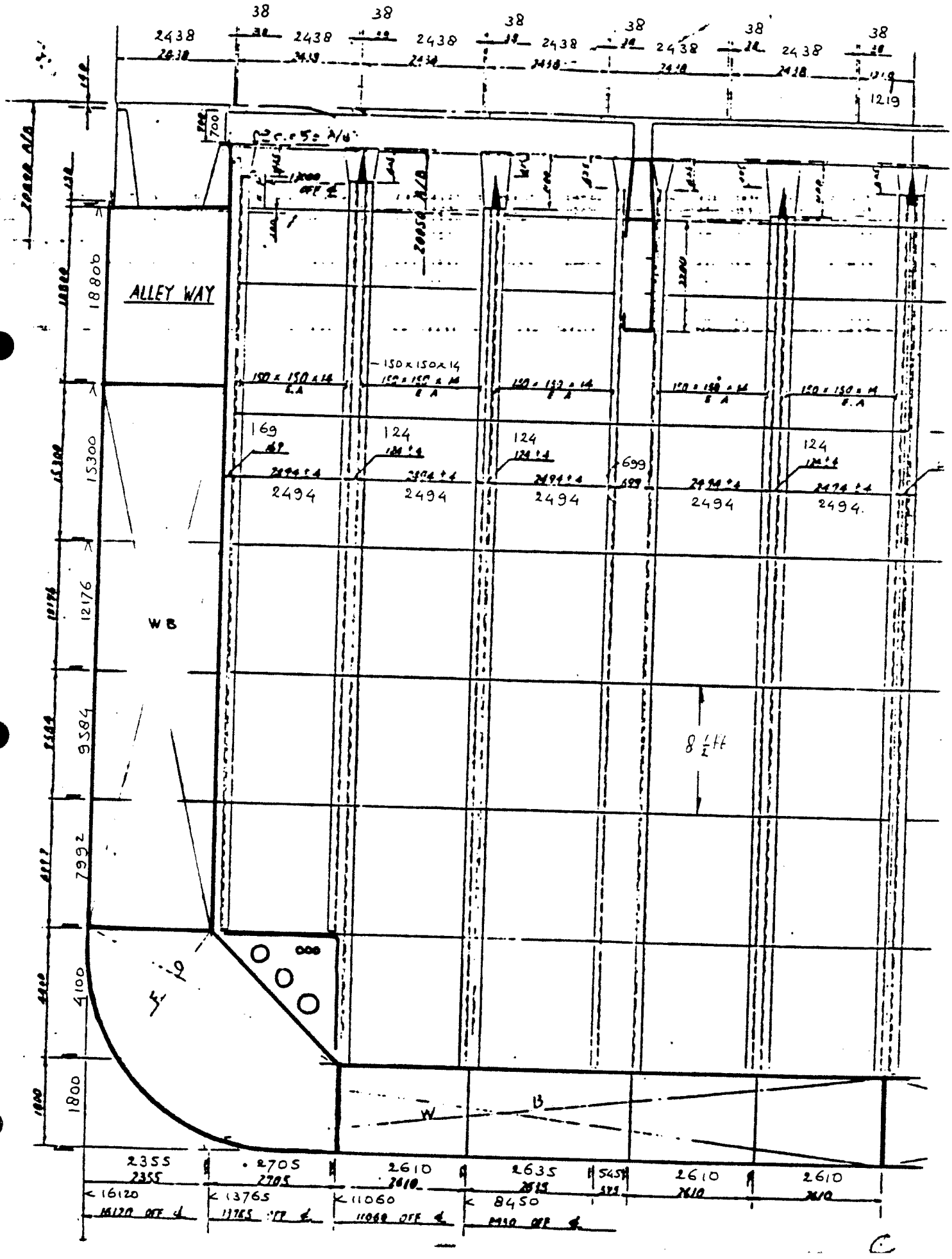
CELLULAR CONTAINER VESSELS -

MODULE OF CELLULAR HOLD.

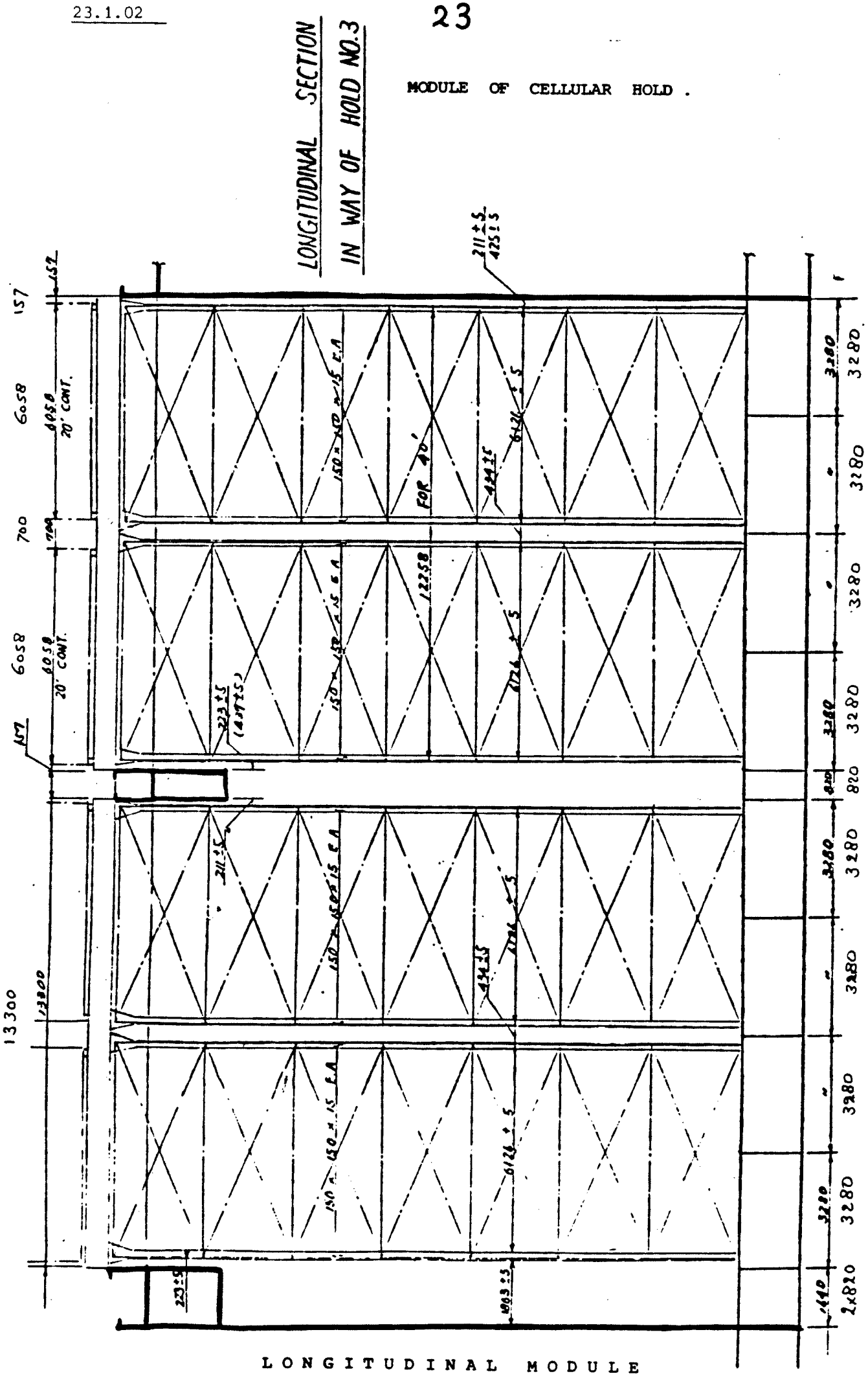
23.1.01

23
CELLULAR HOLD

MIDSHIP SECTION

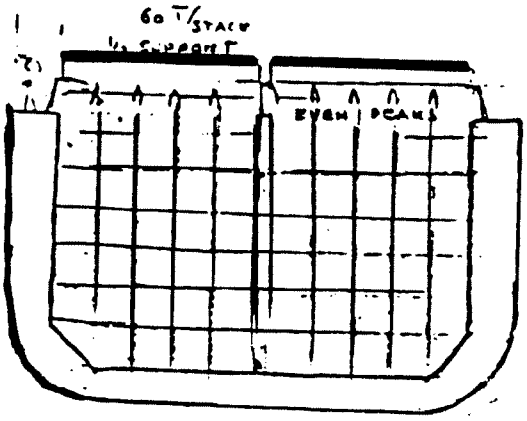


MODULE OF CELLULAR HOLD .



LONGITUDINAL MODULE

2 (TWIN) - HATCH



IN CASE OF MANY PORTS

EXTRA SHIFTING (MIND REEFERS, DANGEROUS CARGO)

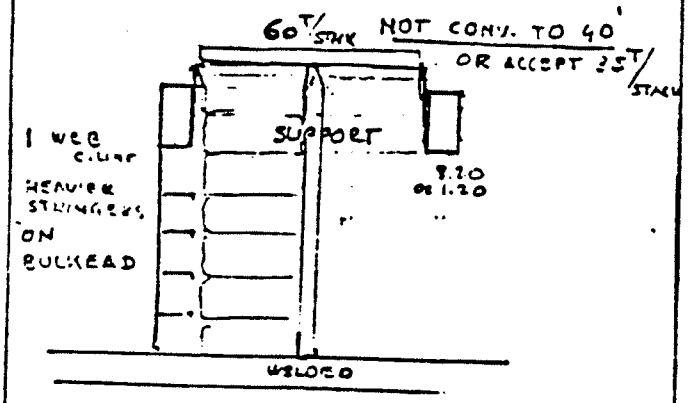
40' ~ 200 DOLLAR/2 MOVES

20' ~ 120 " /2 MOVES.

IN CASE OF 50 EXTRA SHIFTINGS/PORT

50 x ~160 DOLL = 8000 DOLL/PORT

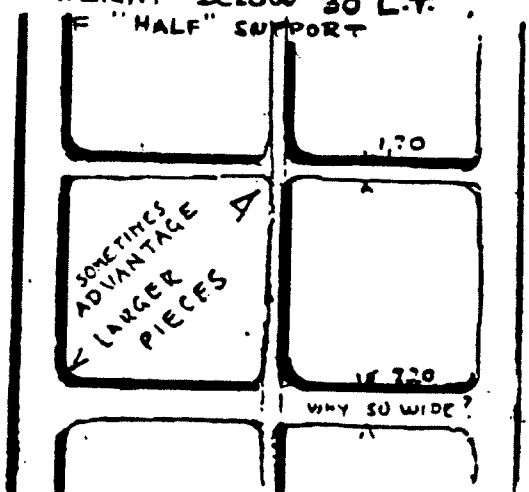
STACKLOAD



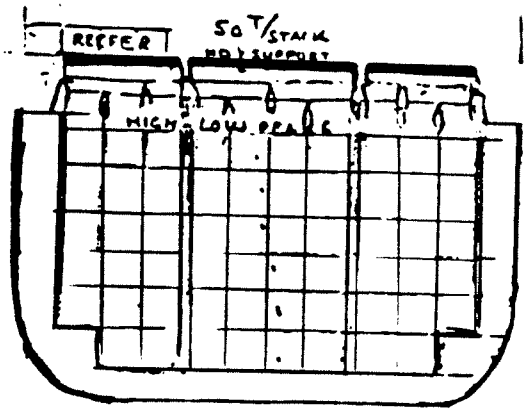
WEAK COVER (BUCKLING STRENGTH)

WEIGHT BELOW 30 L.T.

IF "HALF" SUPPORT

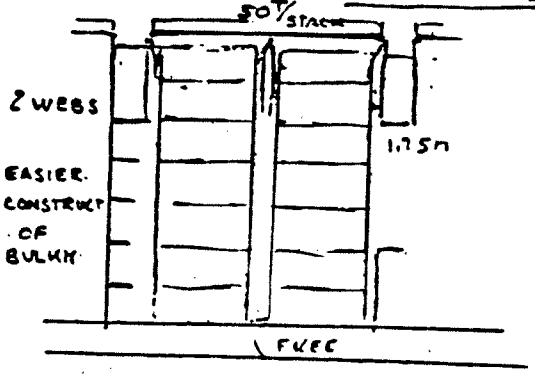


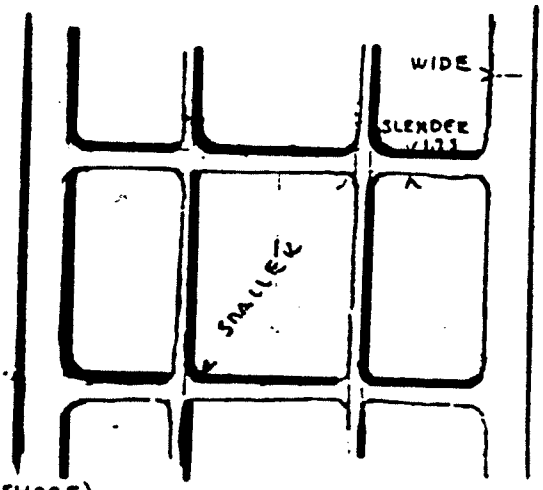
3 (TRIPLE) - HATCH BETTER



- BETTER STOWAGE : 1 PORT/PER COVER
  MORE FLEXIBLE
  IN CASE OF 1500 TEU AND 10. PORTS P. ROUNDTRIP
- LESS TORSION PROBLEMS
- STACKLOAD CAN BE HIGHER.
  MORE EXPENSIVE CONSTRUCTION BUT NOT SO MUCH EXTRA STWEIGHT

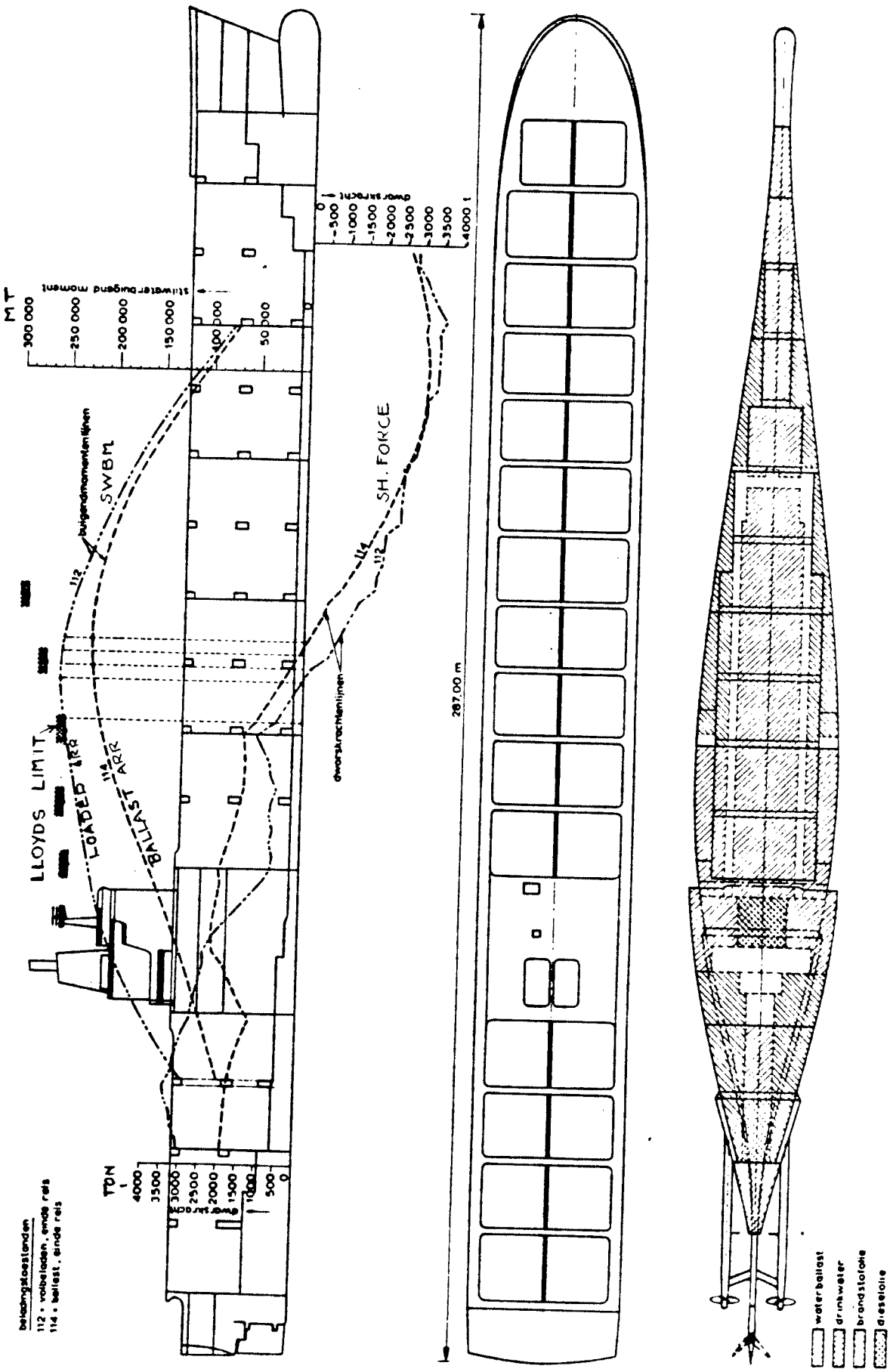
HOLDS TO BE CONVERTABLE TO 40'





COVERS SOMEWHAT MORE OBSTRUCTIONS AT TERMINALS (ASHORE)

STILL WATER BENDING MOMENTS AND SHEARING FORCES.



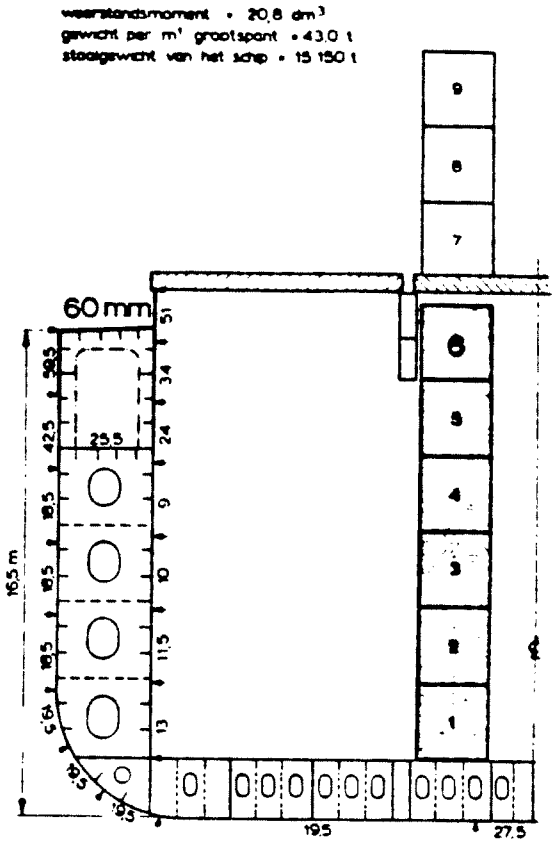
- 3 MIDSHIP SECTIONS HAVING SAME SECTION-MODULE
- SAME STRESS BY SAME BENDING-MOMENT

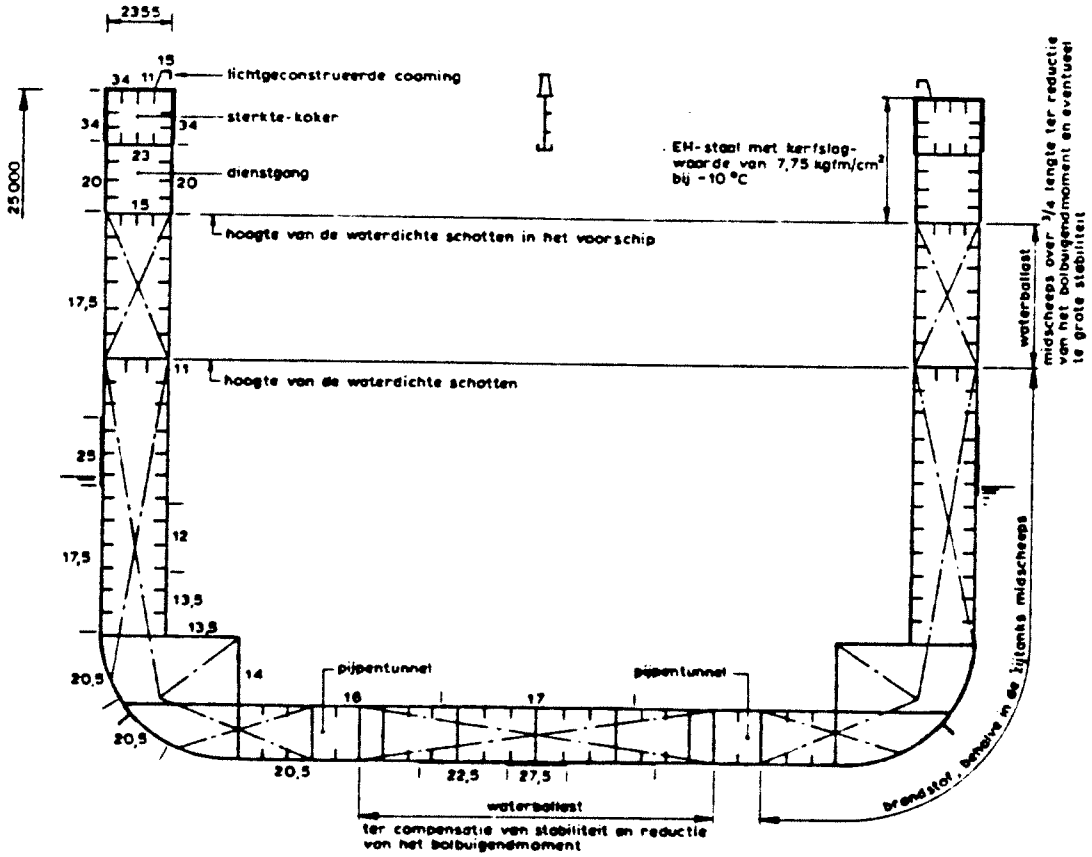
⇩

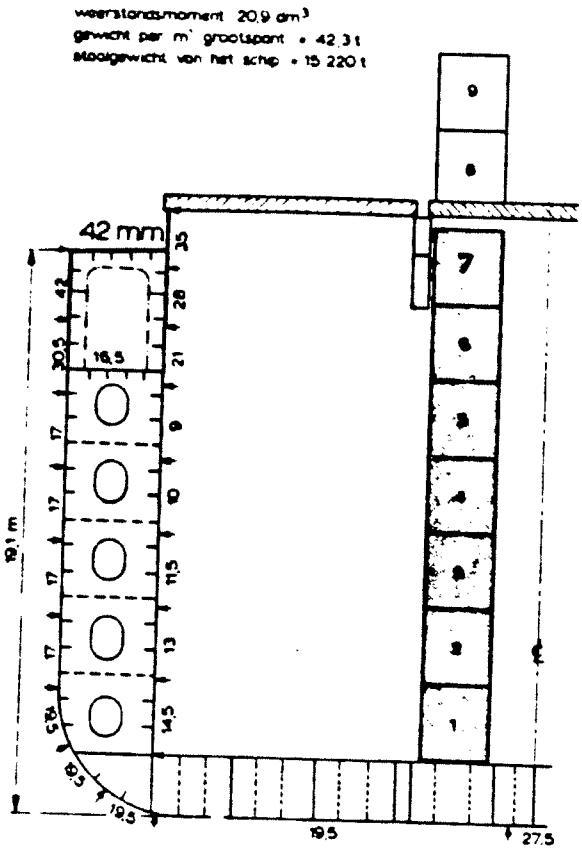
- SMALL-DEPTH VESSEL WILL HAVE:
- LARGEST DEFLECTION
- HEAVIEST PLATE THICKNESS

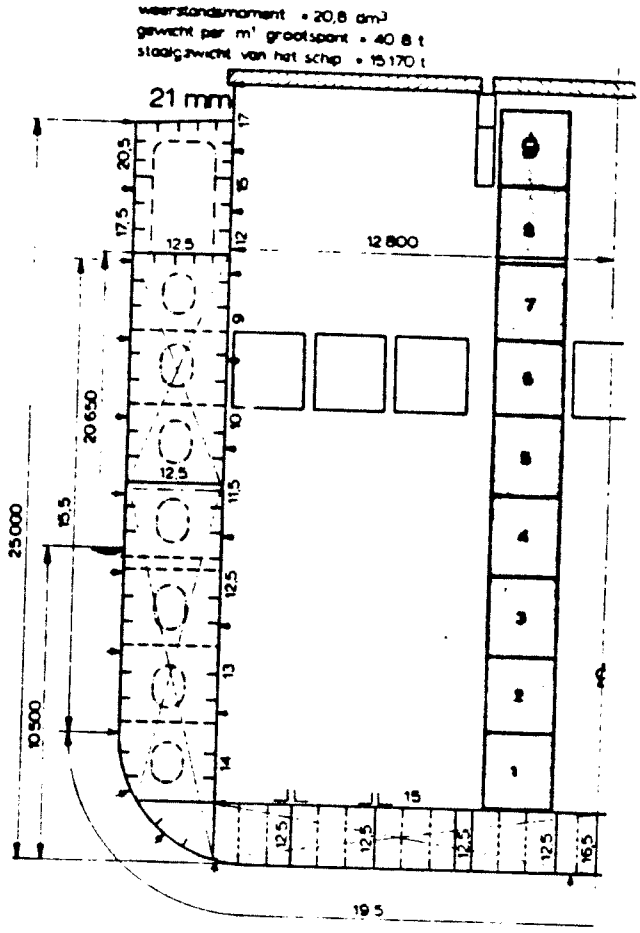
60 mm IS TOO HEAVY

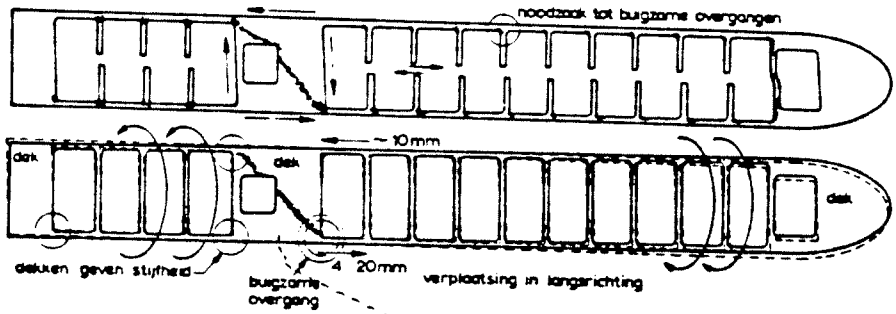
42 „ IS PRACTICAL LIMIT.



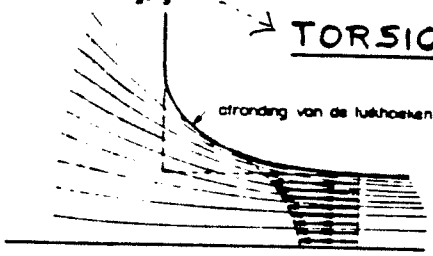




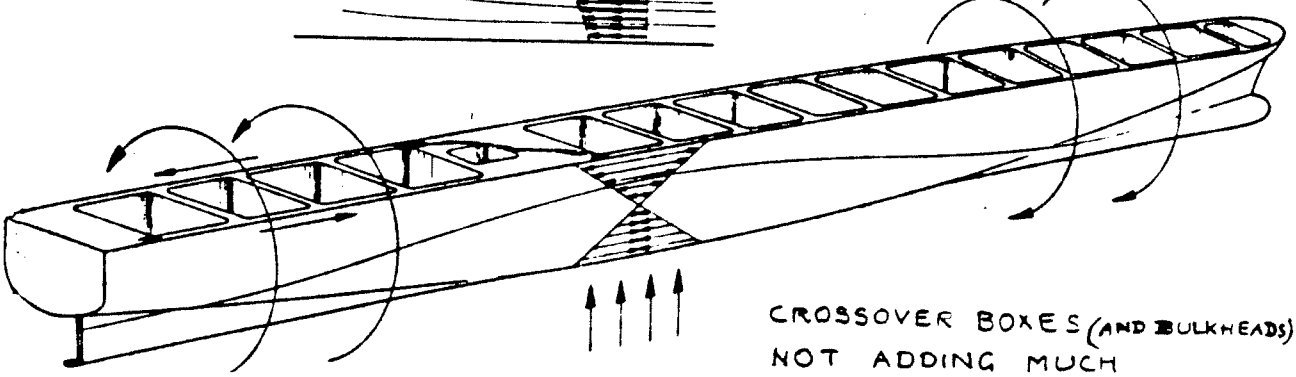




23.2.3



TORSIONAL STIFFNESS MAINLY BY UPPER DECK AT SUPERSTR./ENG.R.

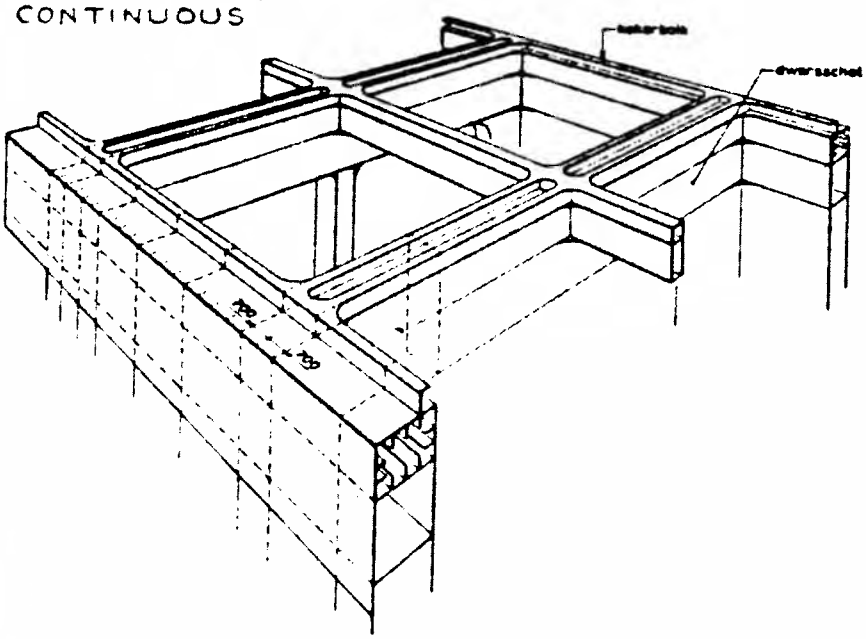


CROSSOVER BOXES (AND BULKHEADS) NOT ADDING MUCH

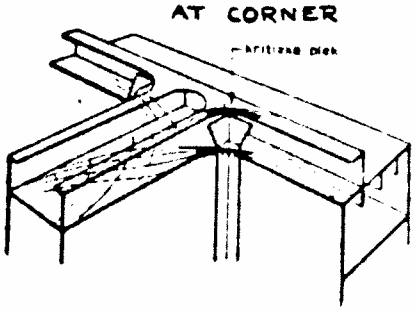
De constructie van de doorlopende coamings en die van de niet-doorlopende coamings.

Doorlopende coamings

CONTINUOUS

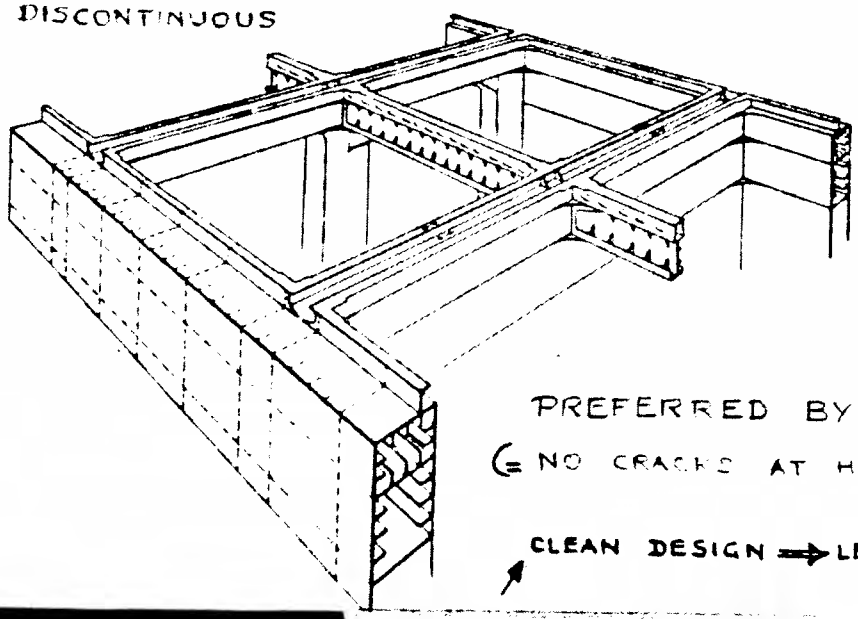


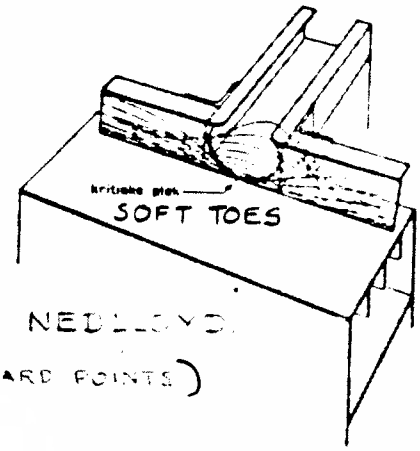
CRITICAL AT CORNER



Niet-doorlopende coamings

DISCONTINUOUS

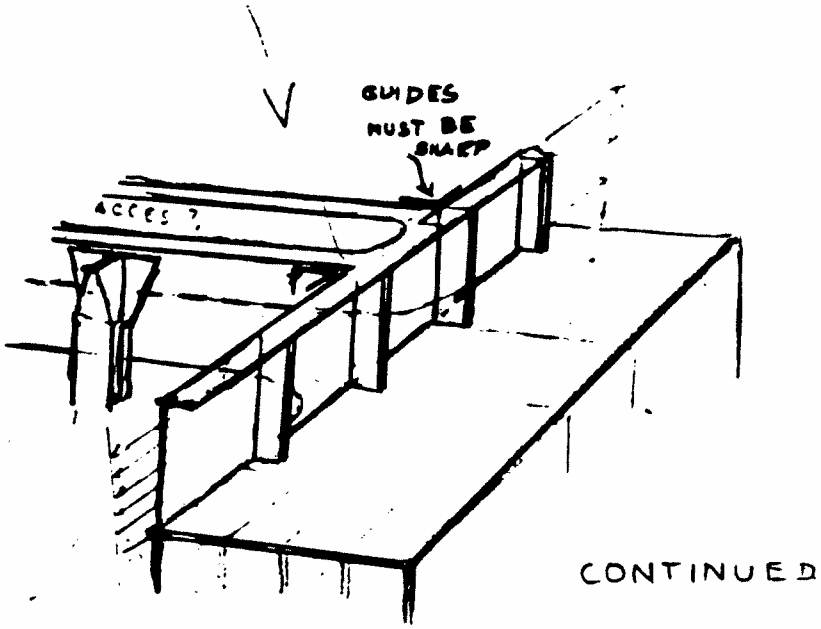


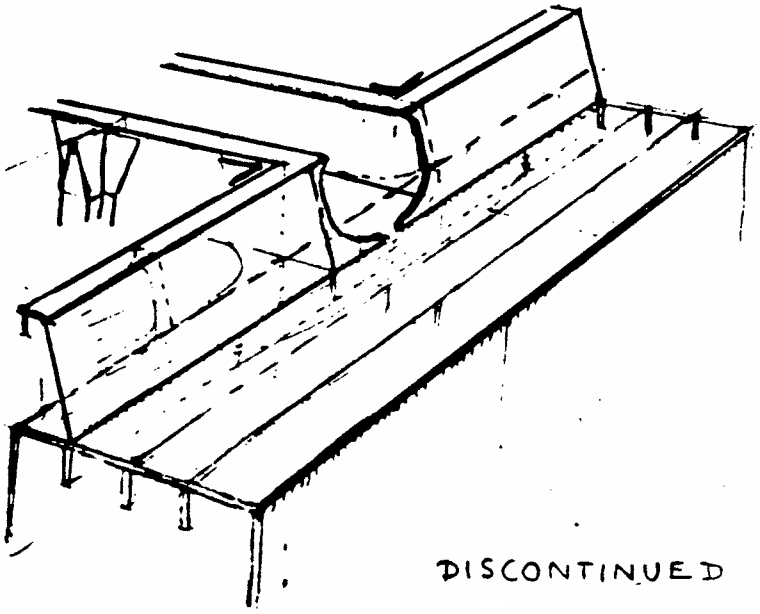


PREFERRED BY NEDLLOYD

(= NO CRACKS AT HARD POINTS)

CLEAN DESIGN ⟹ LESS MAINTENANCE


GUIDES
MUST BE
SHARP
ACCES ?
CONTINUED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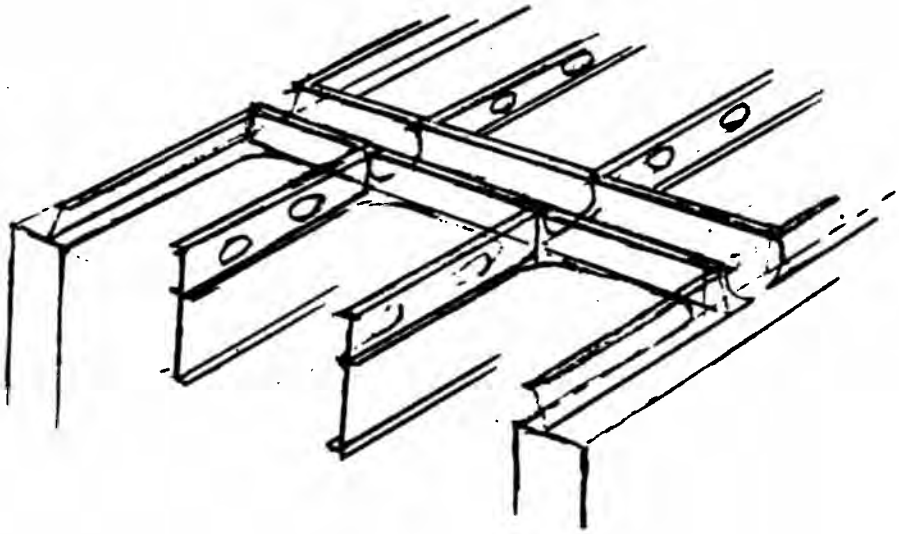
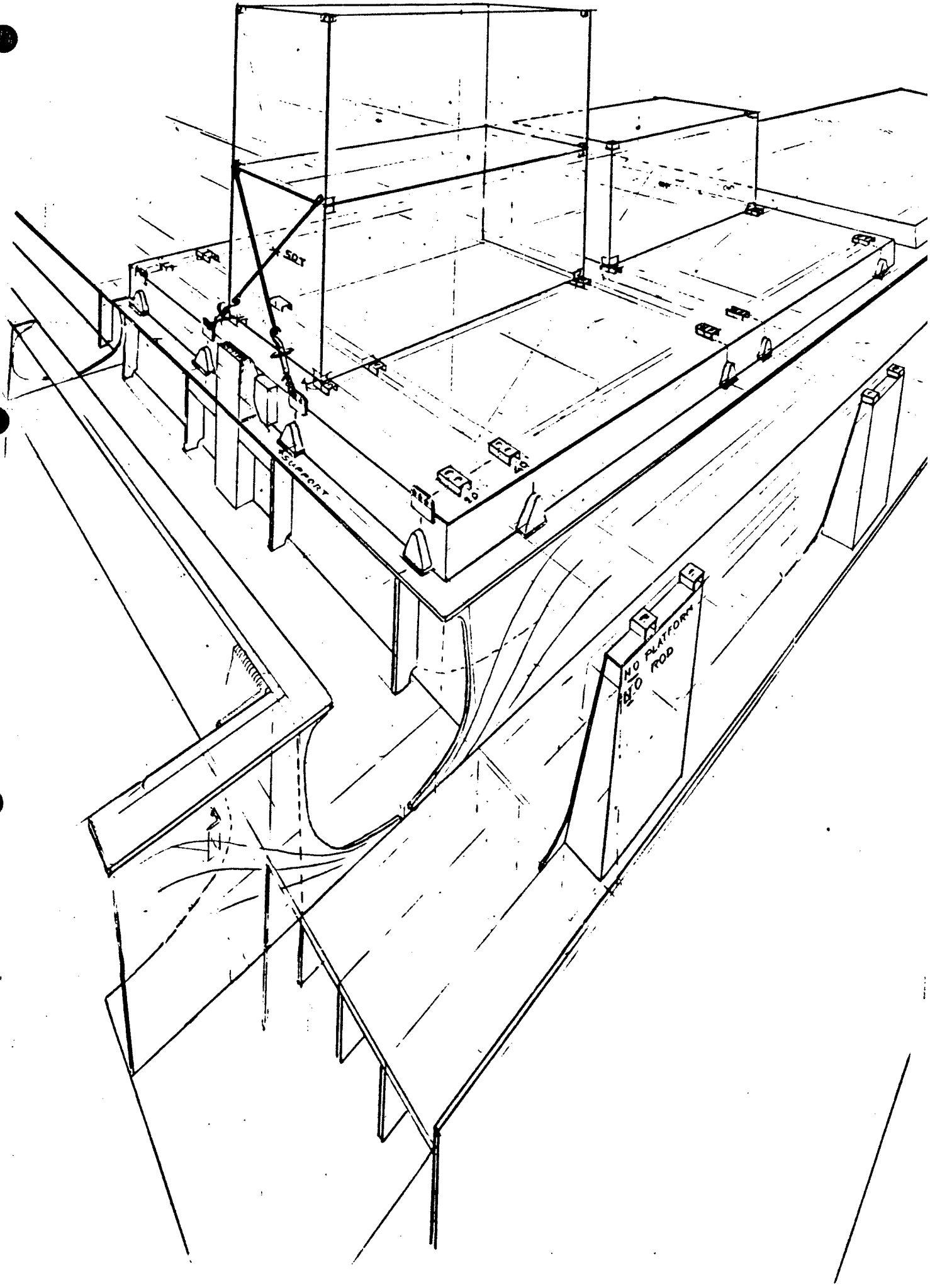
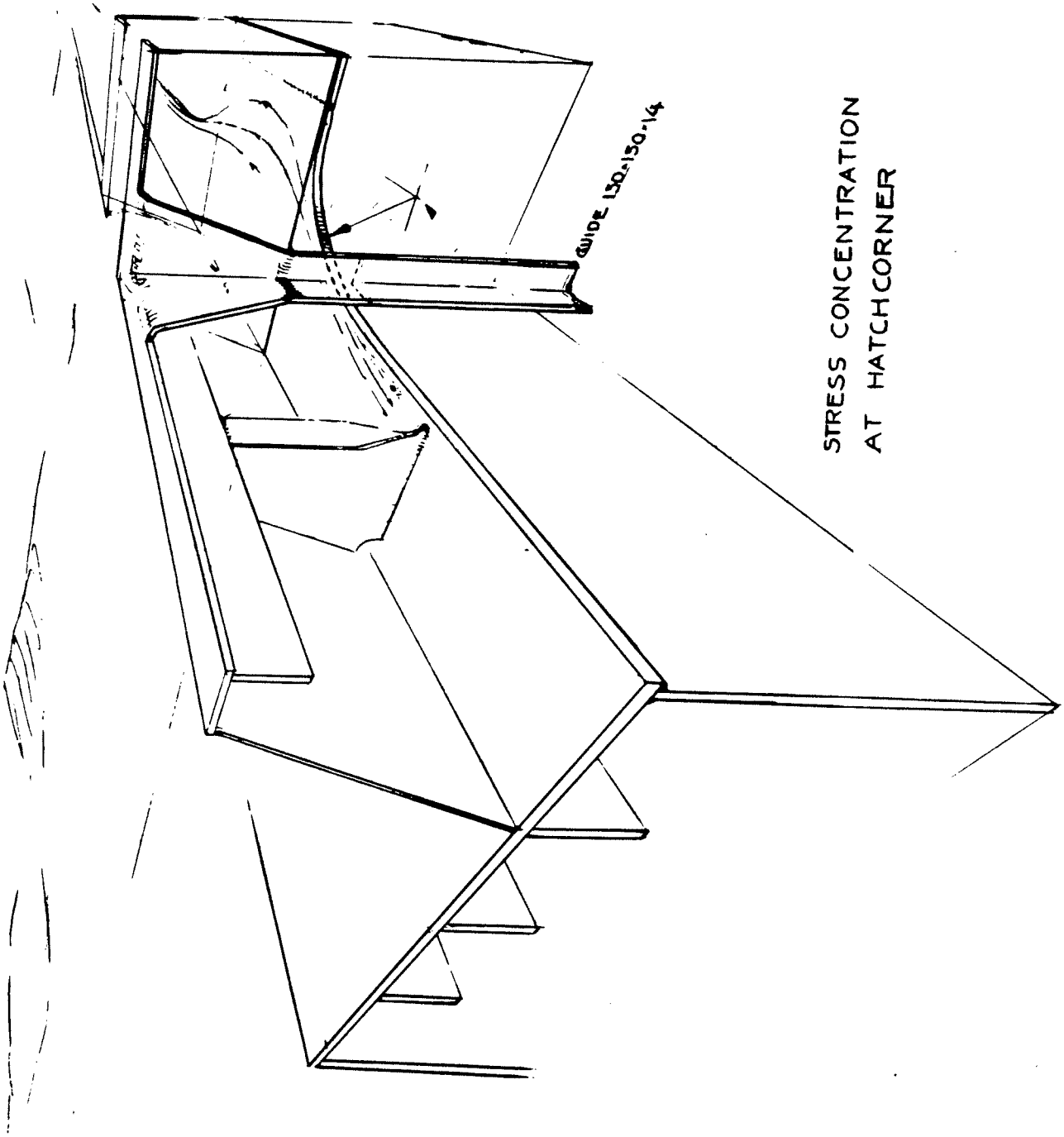



SUPPORT
NO PLATFORM
NO ROD



STRESS CONCENTRATION
AT HATCHCORNER

24

CELLULAR CONTAINER VESSELS -

SECURING OF DECK-
CONTAINERS.

LASHING

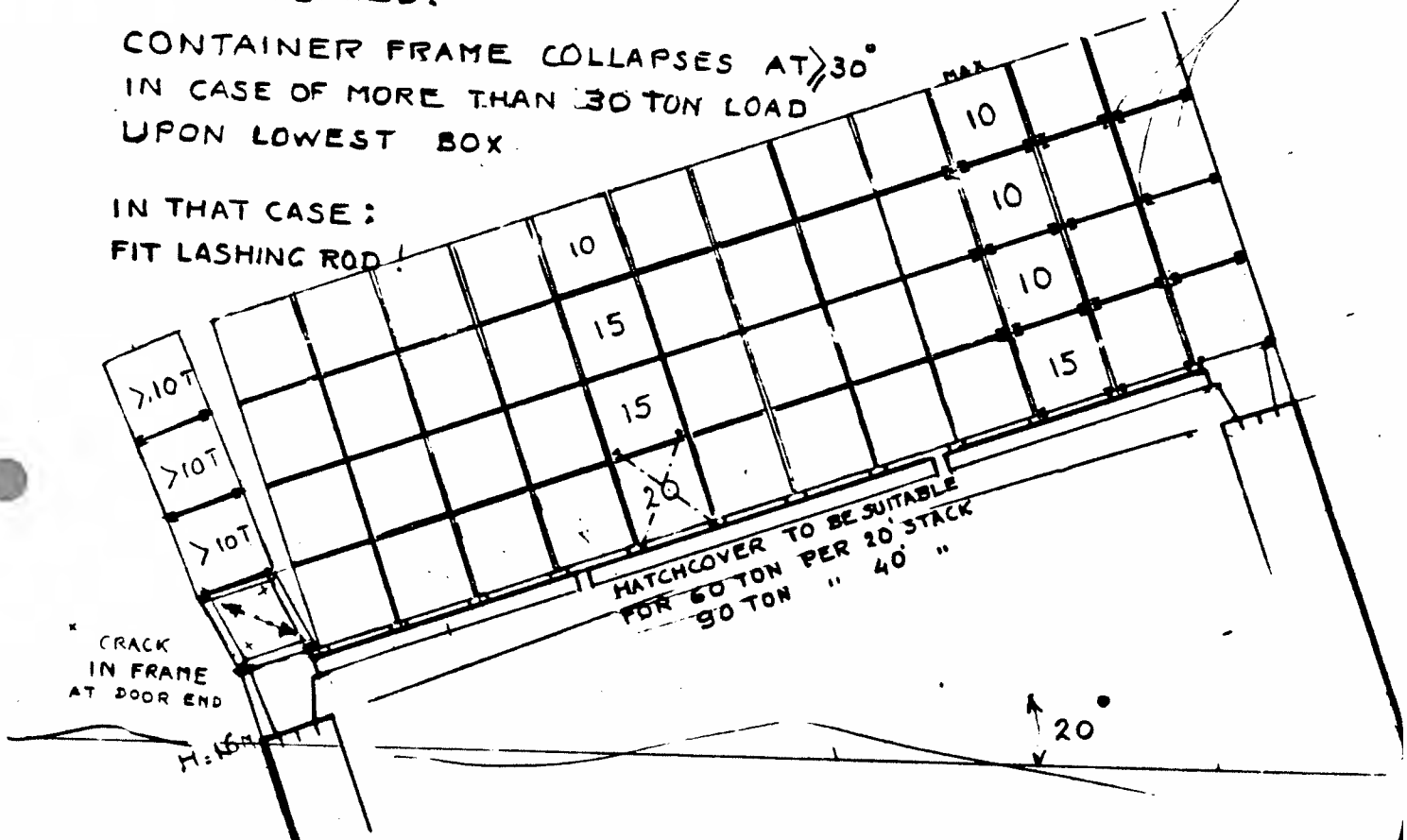
SECURING

NORMALLY: TWISTLOCKS ONLY
IN CASE STACKLOAD ≤ 45 TON

TO BE FEARED:

CONTAINER FRAME COLLAPSES AT ≥ 30°
IN CASE OF MORE THAN 30 TON LOAD
UPON LOWEST BOX

IN THAT CASE:
FIT LASHING ROD!

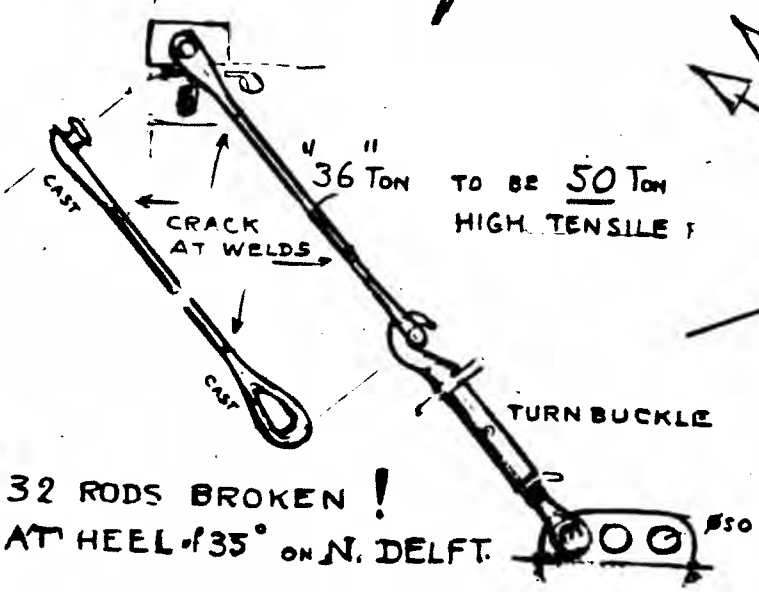


PRACTICAL HEELING TEST WITH 4 HIGH STACK
IS BARELY NEEDED, TO SEE WHAT WILL HAPPEN

T=10M

IF GM REASONABLE: ABOUT 6-8% EXTRA BY DYNAMIC EFF
AVOID ROLLING RESONANCE!
GM TO BE < 6'
ANTI ROLLING FINS NOT EFFICIENT
IN HEAVY GALE
AT SLOW SPEED

HANDY RODS

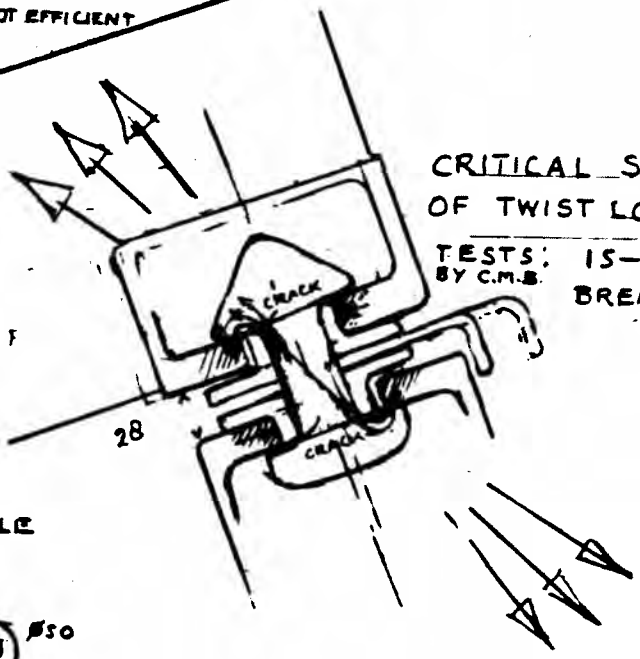


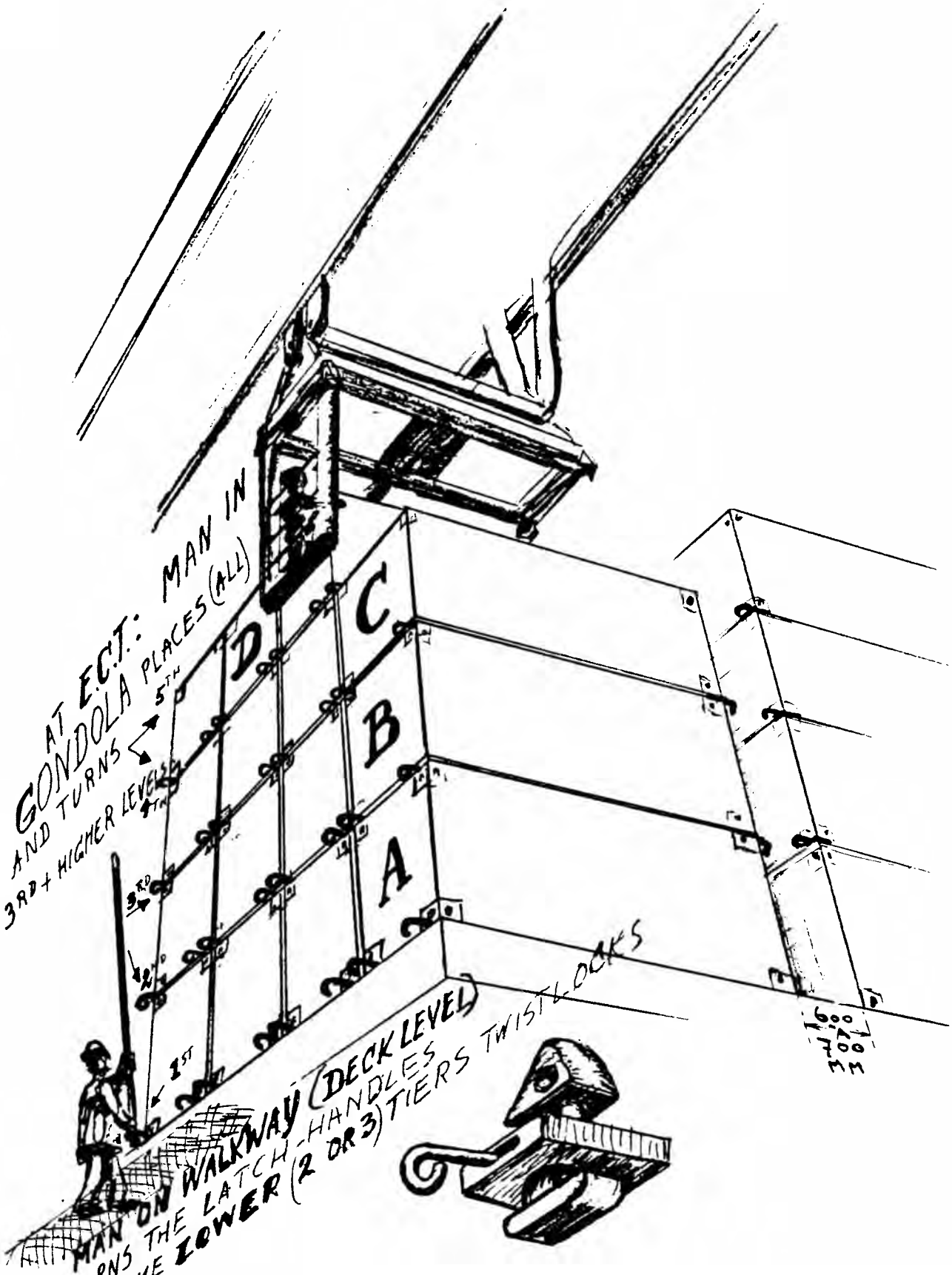
" " 36 TON TO BE 50 TON
HIGH TENSILE!

32 RODS BROKEN!
AT HEEL of 35° ON N. DELFT

CRITICAL SPOTS
OF TWIST LOCKS

TESTS: 15–50 T
BY C.M.B. BREAKLOAD



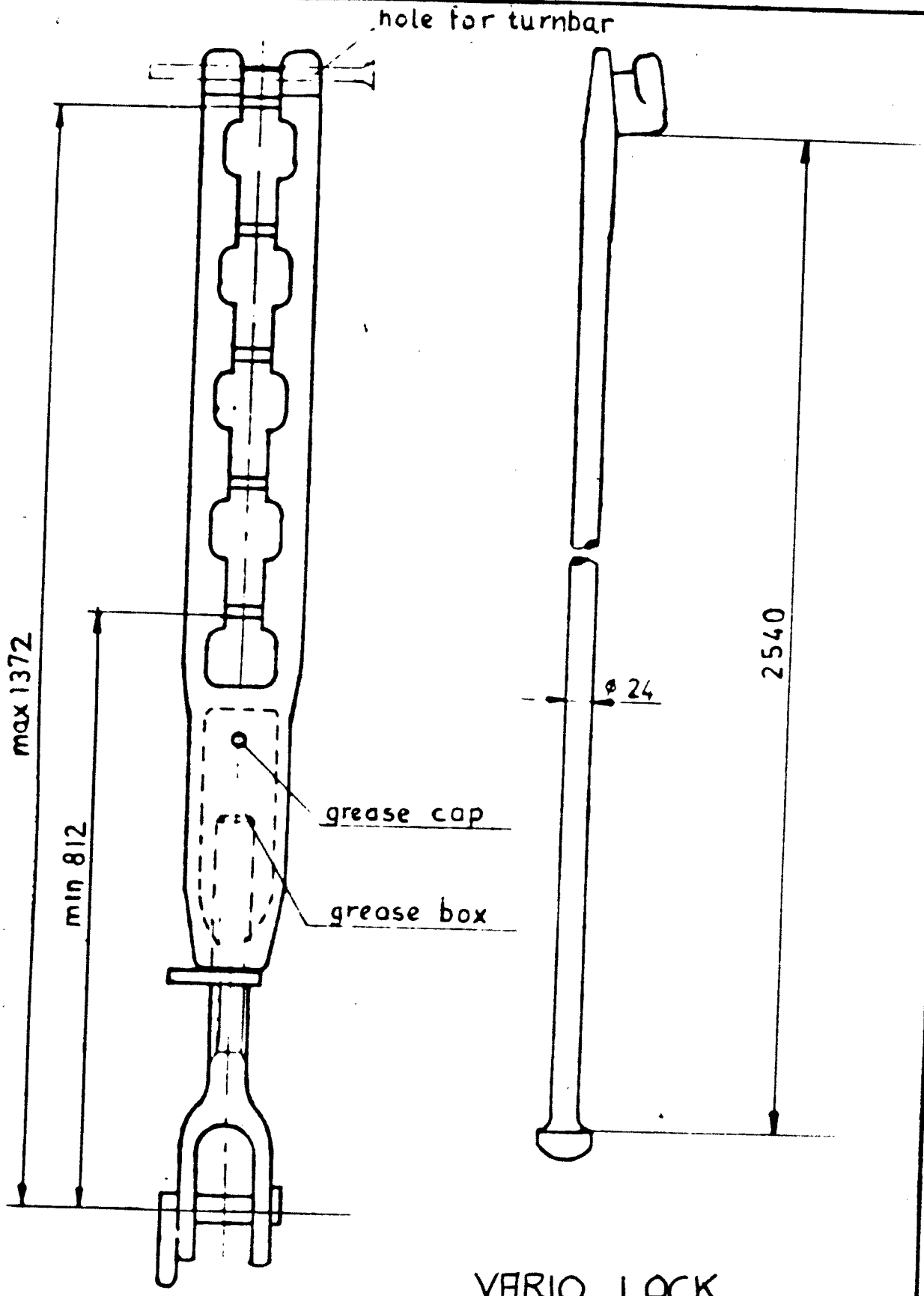


Example of SAFE WAYS OF PLACING TWIST-LOCKS AND OF TURNING THEIR LATCH-HANDLES.

IV -82

NORMALLY:

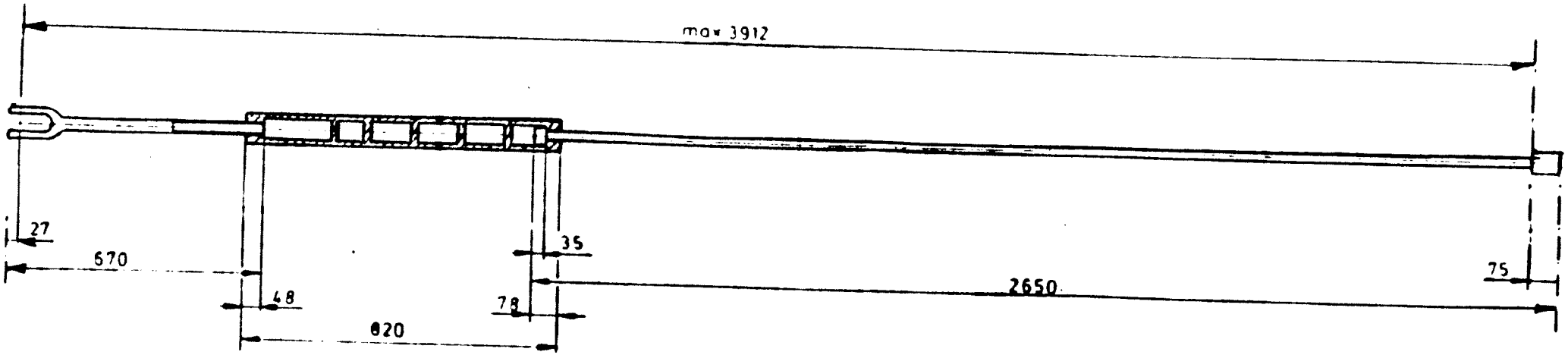
TWISTLOCKS ONLY IN CASE STACKLOAD < 45TON

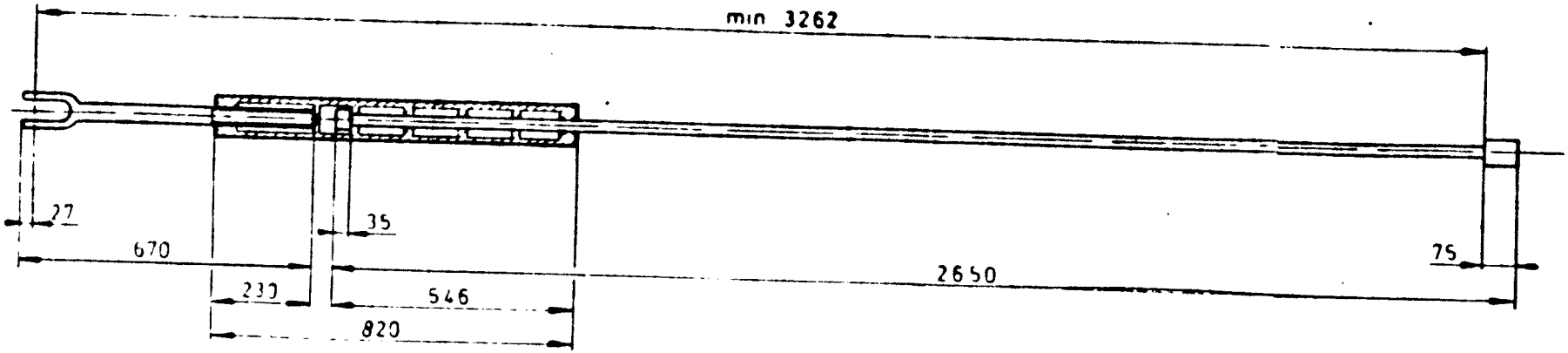


VARIO LOCK

AND

LASHING BAR



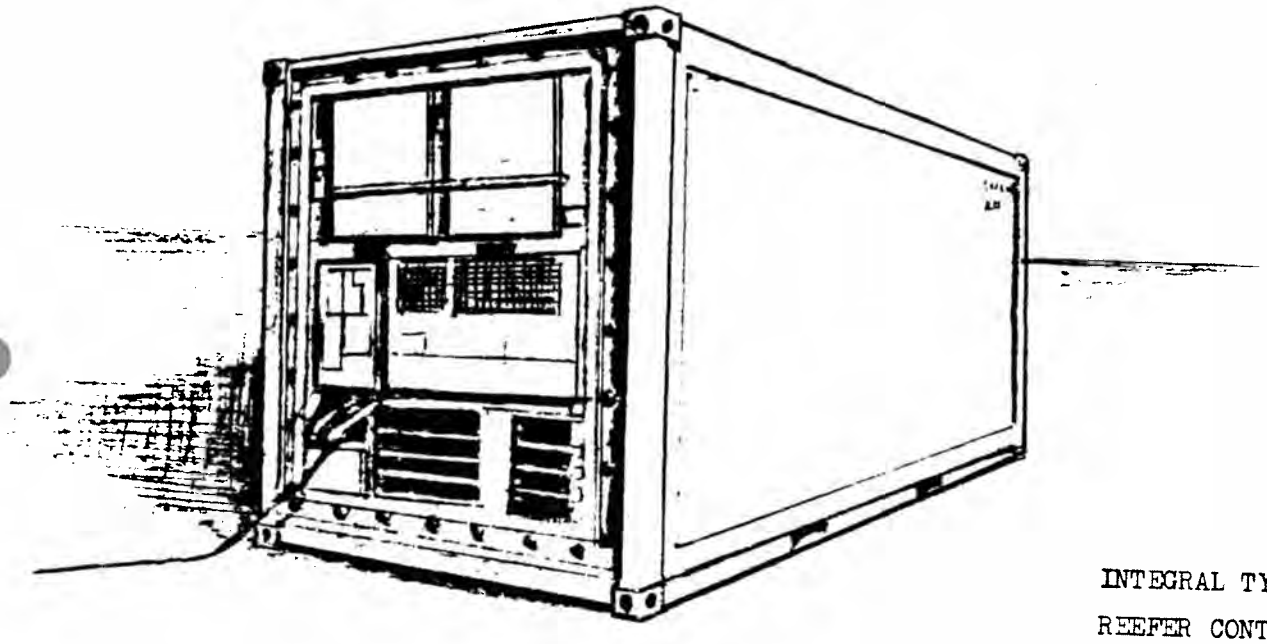


VARIO LOCK SYSTEM.

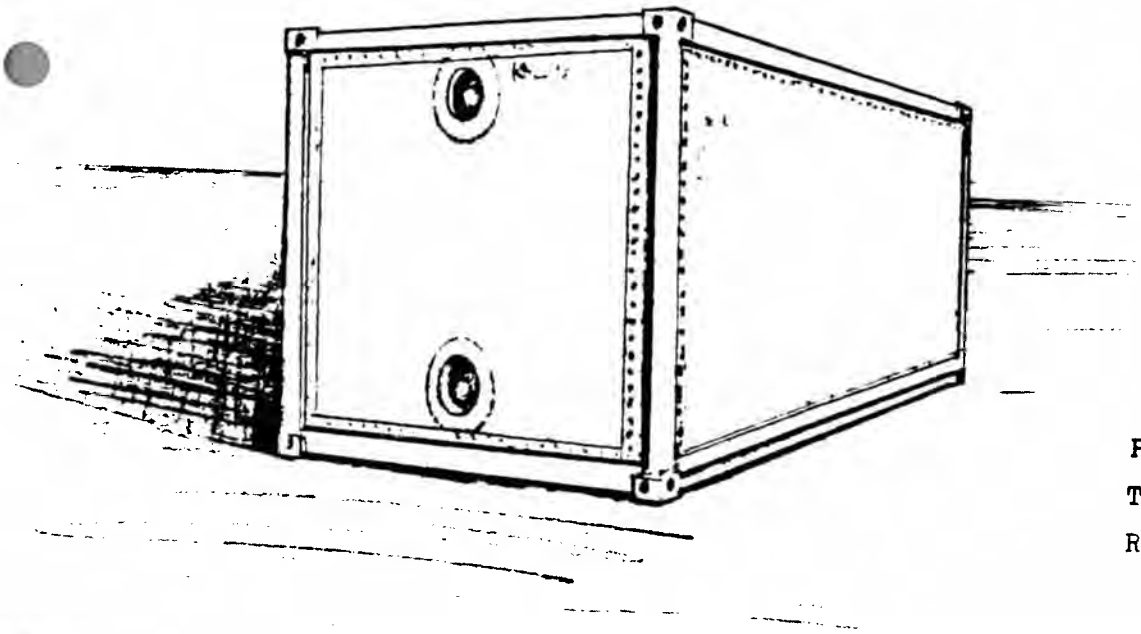
25.0

R E E F E R S .

REEFERS



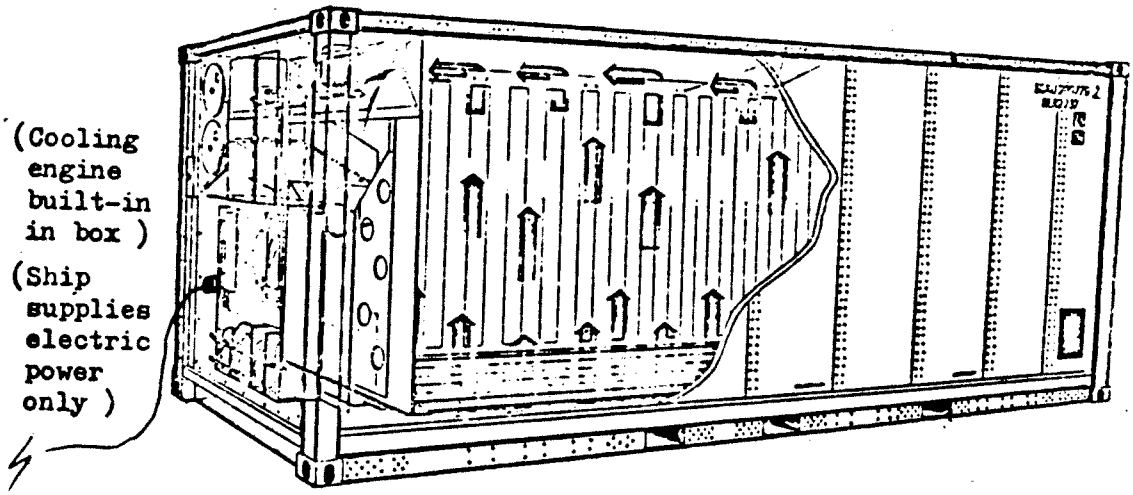
INTEGRAL TYPE
REEFER CONTAINER



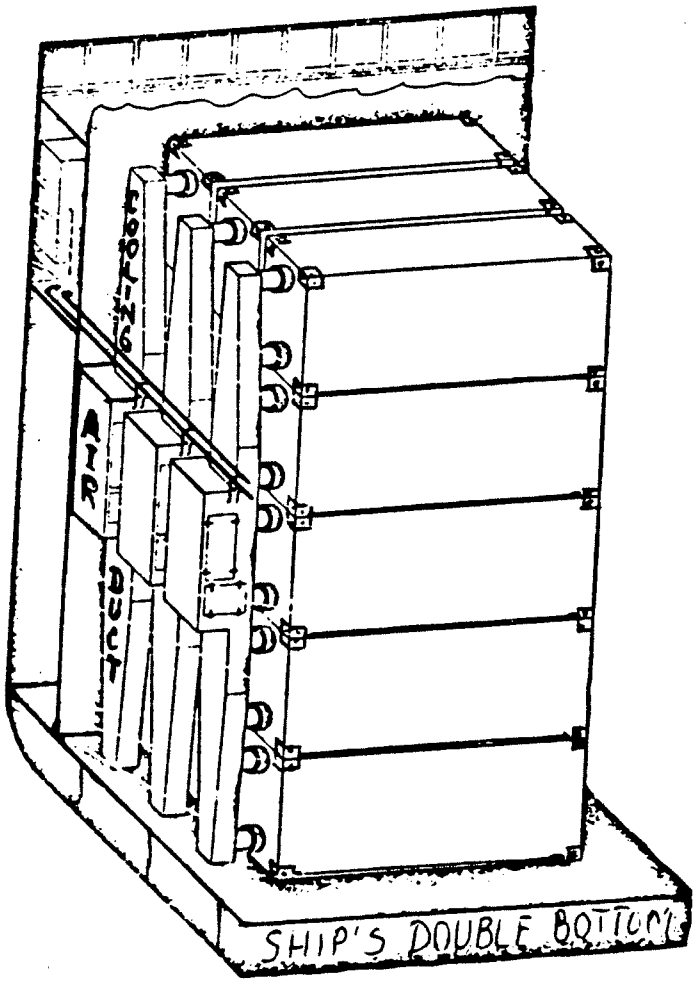
PORTHOLE / DUCTED
TYPE
REEFER CONTAINER

25.1.02 (= bottom figure)
25.2.02 (= upper figure)

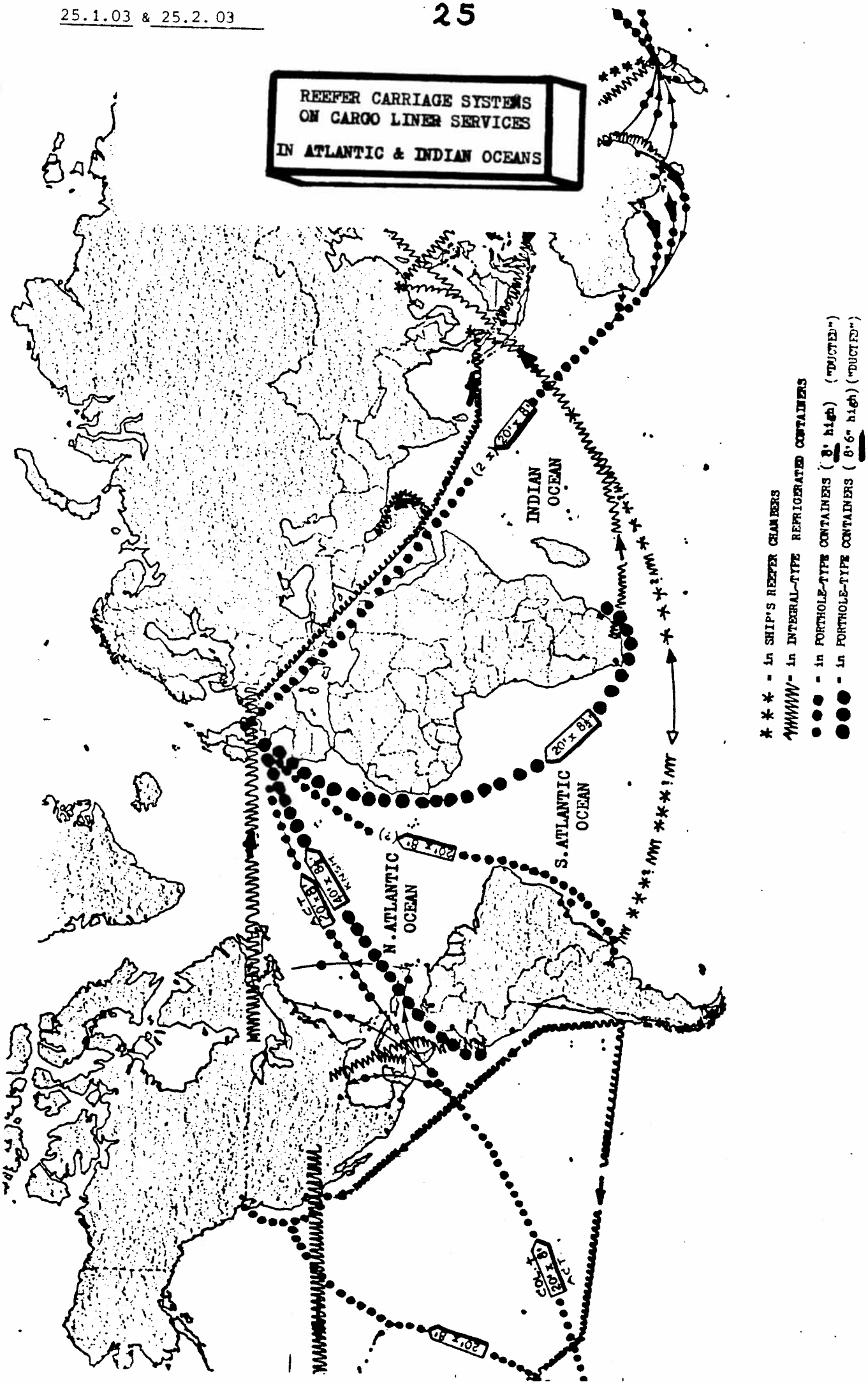
"INTEGRAL" SYSTEM



DUCTED SYSTEM
(Also called "PORTHOLE" system)

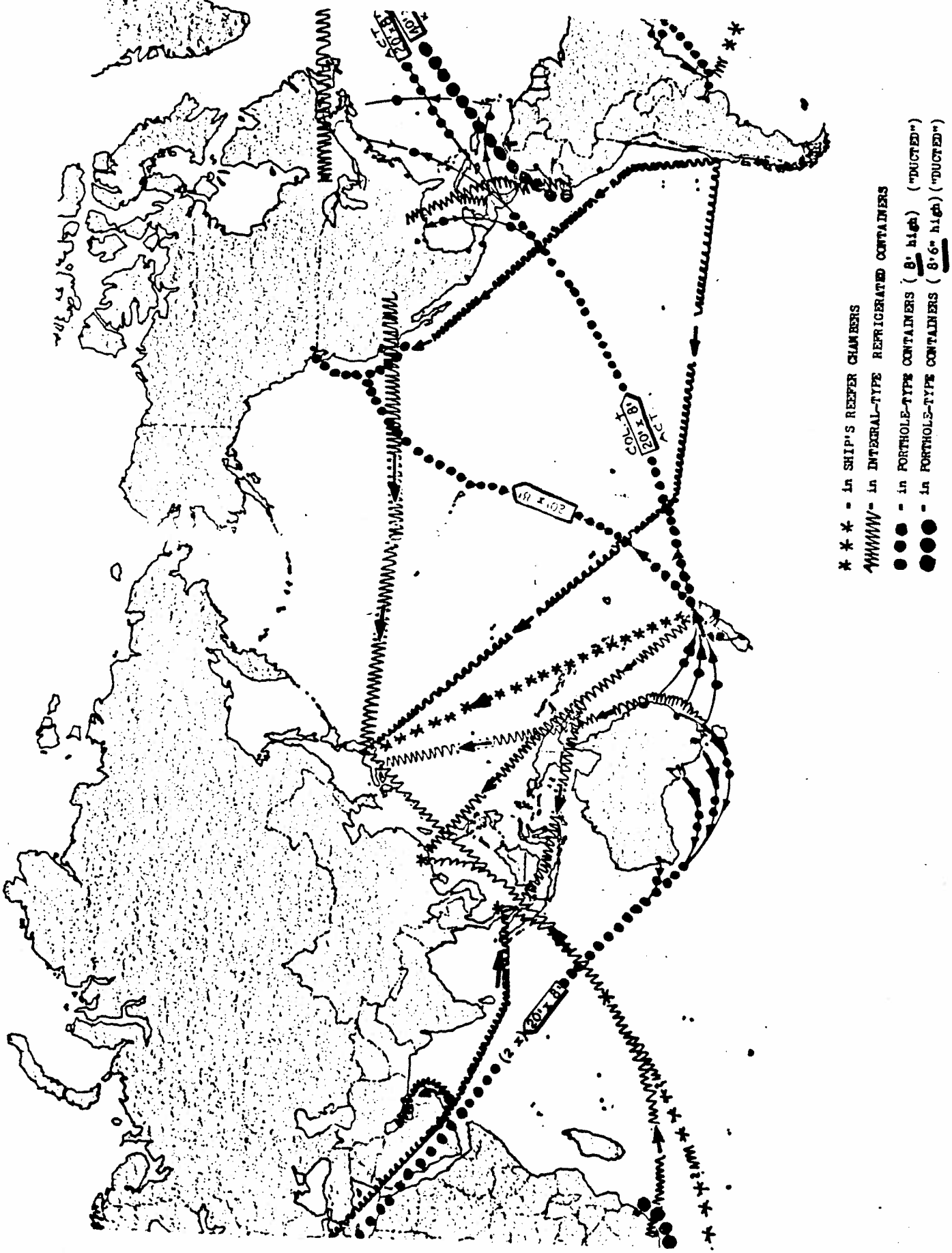


(Ship's cooling system)


REEFER CARRIAGE SYSTEMS
ON CARGO LINER SERVICES
IN ATLANTIC & INDIAN OCEANS
INDIAN OCEAN
S.ATLANTIC OCEAN
N.ATLANTIC OCEAN
20' x 8'
20' x 8½'
40' x 8'
40' x 8½'
KNSTT
CT
COW
PCT
* * * - in SHIP'S REEFER CHAMBERS
- in INTEGRAL-TYPE REFRIGERATED CONTAINERS
- in PORTHOLE-TYPE CONTAINERS (8' high) ("DUCTED")
- in PORTHOLE-TYPE CONTAINERS (8'6" high) ("DUCTED")

25.1.04 & 25.2.04

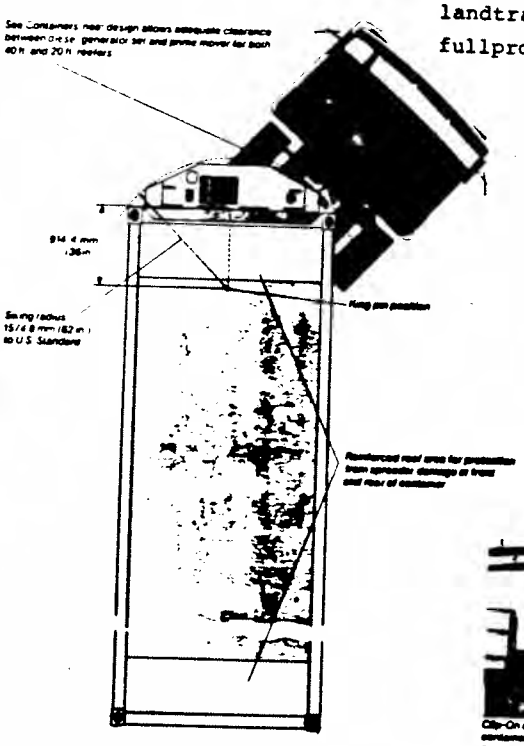
REEFER CARRIAGE SYSTEMS ON CARGO LINER SERVICES IN PACIFIC OCEAN BASIN

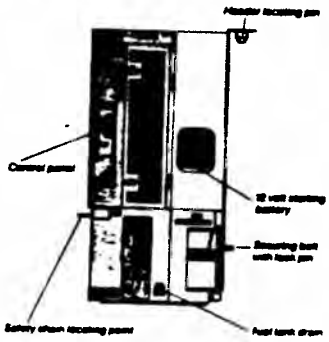


REEFER BOX SYSTEMS COMPARISON.

INTEGRAL type v.a.v. PORTHOLE type

	INTEGRAL REEFER BOX system :	DUCTED PORTHOLE BOX system
INVESTMENT	Independent of vessel Plugs on deck....Air cooling Plugs in holds Air cooling Water cooling	Vessel to have refrigeration plant (Freon R 22 - brine system) Ducts in hold to be suitable for 40' x 8½' or for 20' x 8½' or for 20' x 8 ' (It is difficult to convert ducting)
INTERCHANGEABILITY OF BOXES AND FLEXIBILITY OF VESSELS FOR OTHER TRADES	Very flexible	Not flexible. Vessel tied to one trade.
LEASING OF BOX	Integral reeferbox is always leasible(') This might be important in case of small fluctuations in reefer transport. (') FOR REEFERS ONLY.	Impossible to lease porthole box. Sometimes it is difficult to offer an insulated porthole box for general cargo at the normal rate because space is 1½ m^3 less than normal G/C-box.
HOT CLIMATE OPERATION COOLING DURING FEEDERING WAITING LANDTRANSPORT-	Usually rather easily feasible to keep cargo cooled ashore by - providing plugs at terminals - " plugs on feedership - " generator set to the integral container during landtransport seems to be fullproof.	Usually almost infeasible to keep cargo cooled down ashore without the use of - special nitrogen cooling plant at terminals - or terminal clip-ons. Special space is needed. - Special clip-on refrigeration units during land transport are necessary which are cumbersome/unreliable (Now liquid-nitrogen is used as replacement of Clip-On Frig., but...) - On feederships it is impossible to cool by clip-on frig. units.
		Only suitable if waiting time in tropic areas without cooling is very short: - Boxes to be sent to plantations where cooled-down fruit is loaded. - Alternatively the boxes are to be loaded with the fruit (bananas eg.) near to the vessel alongside the quay - and immediately after loading the bananas, the box is to be taken into the vessel and connected onto the cooled-air ducting for cooling-down.
		- In case of availability of cold stores at loading ports or at the plantations the ducted box type may be applicable (higher cube for bananas Northbound and utilizable for general cargo Southbound). - Land legs in tropics to be short. - Land legs in moderate climate (NW Europe) to be covered within 1-2 days' time if cooling is not provided).





INTEGRAL type : PORTHOLE type

SYSTEM'S SUITABILITY

INTEGRAL:
Easily interchangeable.
Suitable for most trades, especially in irregular trades:
- FAR EAST & PACIFIC BASIN
- JAPAN - USA & AUSTRALIA trfc.
- JAPAN - AFRICA trades

(so: Low volume/High value cargo)
Often practical limit yet as to quantity acceptable (due to labour-intensive attending) aboard trunkline vessels: up to 600 boxes/vessel, or less. (proportionally small quantity per vessel)

PORTHOLE:
Very poorly interchangeable
Suitable for regular trades with high volume (Box = cheaper/better volume):

NW EUROPE - AUSTRALIA/N.Z.	20' x 8'
NW EUROPE - SOUTHERN AFR.	20' x 8½'
NW EUROPE - CARIBBEAN	40' x 8½'

(so: High volume/Incl. low value cargo)
Proportionally large quantities/vessel acceptable (and wanted) aboard the trunkline vessels.

EARNING CAPACITY

	INTEGRAL 20'	INTEGRAL 40'	PORTHOLE 20'	PORTHOLE 40'
Weight &	3200 kg	6000 (± 5%) kg	2600 kg	5000 kg
Volume	27 m^3		29 m^3	
Floor cap.	9 pallets	19 pallets	10 plts	20 pallets

(so: larger earning capacity/box)

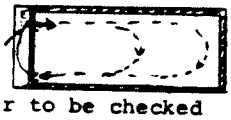
TEMPERATURE REGULATION

INTEGRAL:
Requirements to be fulfilled, more flexible in adaptation

PORTHOLE:
- 4 temperatures of brine which is circulating in the vessel.
- Same temperature per stack (≈ 7)
- Problems may arise if a stack is only partly filled with reefer boxes:
  - Compressors cannot work below a certain minimum load level.
  - Problems with crosstainting might arise (fruit/fish e.g.)
  - Fruit Board requirements to be fulfilled, (if applicable).
- Less problems for engineers on board vessel.
- More stable conditions at sea.

PROBLEMS DURING SEA TRANSPORT

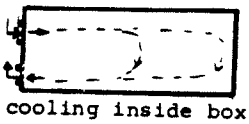
INTEGRAL:
- More problems for ships' engineers: Knowledge, spare-parts available? Instruction books! Against 'reduced crew' concept line.



PTI
- Every container to be checked before being taken aboard.
- Good organisation ashore (for looking after the integrals) is an absolute requirement.
- During initial period there may be teething troubles, however, on the long run things usually run smoothly.

PORTHOLE:
- Less problems for ships' engineers
- More stable conditions at sea.

However:



- Non-homogeneous cooling inside box might occur (40' box with bananas).
- Differences in air temperature between highest and lowest boxes in same stack may occur due to ambient conditions and surroundings (e.g. near hatchcovers or tanktop).
- During initial period there may be teething troubles such as:
- Air leakage of rubber couplings,
- Freezing of box to coupling and damage to rubber 'tires'.

INTEGRAL type	:	PORTHOLE type
- Integral reefer must be of first class quality, e.g.: Carrier, Daikin, Thermoking, Mitsubishi.		- Damage to fruit by water particles which circulate in the airflow (water produced by defrosting) might occur if supervision should lack.
STOWAGE FLEXIBILITY ON BOARD SHIP		
Integrals can be stowed on deck and/or underdeck (if air-blower or preferably water cooling is provided underdeck which is not that expensive as ducting system)		As (usually) not ducting provided above deck, stowage must be under deck and into certain stacks which limits flexibility of stow on board.
(- More flexible / less investment on shipboard).		(-Less flexible / more investment on shipboard).
MAINTENANCE COSTS / BOX		
Per box (solely)... 240 USD/BxYr		Per box (solely) ..... 60 USD/BxYr.
INVESTMENT IN SHIP + BOX (PER SLOT PER YEAR OF A SHIP'S LIFETIME)		
PLUGS & TRAFO'S 4000 $/Bx, Over 20 Yrs at 8% .....400 USD/BxYr		SHIP'S PLANT 25000 $/Bx (Compr./Ductg) Over 20 Yrs at 8% .....2600 USD/BxYr
2x INTEGRAL = 2x 18000 $/Bx (Each 10 Yrs 8%) x 2 .....=5365 USD/Yr. slot		2x PORTHOLER = 2x 8000 $/Bx (Each 10 Yrs 8%) x 2 = 2400 USD/YrSlot
Total, incl. mainten^{ce} 6005 USD/Yr. Slot		Total, incl. maint^{ce} 5060 USD/YrSlot
FURTHER COSTS:		
Plug-in points on terminals Generator sets for land hauls		Cooling facilities ashore Clip-on/Gen. sets for long land hauls.

PART I

DESIGN CHOICE FROM SHIPOWNERS POINT OF VIEW.

How to cope with the demands of transportation ?

Liner Fleet Retonnaging.

ABSTRACT

The container is dominating

- The container, whether door to door, or pier to pier, is now generally accepted and in many instances demanded by the customer. Even for trades which in fact are not yet ready for this concept.

Dying cargoliner

- The sophisticated high-speed-cargoliner has phased out; a small amount of boxes and a high fuel consumption, insufficient stability, no containergear, no space on deck, too weak hatchcovers, no square space in holds.
  Union purchase derrick gear, heavy derrick, deeptanks, reeferspaces are of no use anymore.
  3 to 4 Days average stay in port is too long for efficient transportation.

Large cell. Container vessels

- Introduction of the large cellular containervessels operating on trunklines between terminals equipped with container gantries, resulted in a far higher transport production per year and consequently a lower transportation cost per cubft.

Coöperation

- In order to offer a regular pattern of sailings with rationalised frequency to cope with the cargo flow, shipping companies have to co-operate closely which may lead to formation of consortia by which the results can be maximized. Shipowners individuality is, however, more or less absorbed by such consortia.

Weekly service

- In case the speed is sufficient to call 6 ports in Asia and 6 ports in Europe in a 70 days roundtrip, than a weekly service is offered to the customer. By 10 vessels
- Only this system allows for economy of scale and reduction of costs.
- Feederlines connect wayports with the mainports of the trunklines.

Large RoRo container vessel

- Meanwhile large RoRo containervessels have been introduced as well, which can cope also with cargo commodities like steelplates, pipes, ingots, paper, drums, pallets, rolling stock, which cargo is less suitable for containerisation but can easily be carried on platform or trailers.
  These ships are more flexible in the trade but require more skill and ingenuity of the crew to obtain the maximum benefit.

M.P.vessels

- Since in many developing countries the infrastructure is not (yet) geared to the container era, this type of vessel - suitable to carry considerable numbers of container and/or bulk (gen.cargo as well as loose bulk commodities) - is probably for the transitional period the best answer for

  trades between industrial countries on one side and under-developed countries on the other side of the world (for example Japan - W.Africa). One way traffic with loaded containers Westbound to W.Africa will accummulate empty boxes in W.Africa.
  Eastbound - carriage of tropical products will probably go in the oldfashioned way.

Problems with stowage

- The empty containers should be returned to Japan, but helas, there is on a multipurpose vessel hardly any space for stowing empties if the vessel is filled with logs and cotton. Adding an extra midbody to the vessel cou solve the problem for a limited period of time.

Too long in port

- Breakbulk and boxes are in conflict with each other on board and the quay, which after all causes extra costs and many delays, results in a long portstay and a long turnaround time.

Expensive vessel

- The production of the multipurpose vessel will always be less in comparison to fully cellular and RoRo, meanwhile the m.p.vessel is complicated and relatively expensive (per Teu).
  The m.p. vessel will therefore loose on the long run.

Waiting for port development

- The first requirement which has to be fulfilled is the development of ports in West and East Africa - India, Indonesia - West and East America.
  We are awaiting for dredging in ports lacking depth and length for turning the vessel.
  We are waiting for suitable quays, containercranes and parking areas.

Switch to another trade

- If these facilities or part thereof have been provided for the trade will switch over from conventional to container transport very quickly. Considering the severe competition between liner operators it is for the shipowner of enormous importance that he can change one vessel from one trade into another in a very short time.

Flexibility of vessel

- Vessels features should be such that the ship can serve almost all over the world and cope with the normal liner trade requirements.
- The smaller type cellular vessel for the future should be very flexible and try to match or supplement her RoRo sister.
- This means that at a maximum capacity a small draft and a short length overall should be persued.
- A maximum beam will allow for a huge deckload containers because of vessels stability.
- Maximum manoeuvrability should be obtaind by bow- and sternthrusters plus a large rudder.

Gear on/off

- Installation of a gantry crane or alternatively 2 or 3 "plug in bay" jibcranes will give the ship her own gear and independance from underdeveloped port equipment.
- After a certain period of time the port might be equipped with a portainer or the vessel should shift to another trade and can be converted to gearless within one to altern, 3 days.

Reefer containers

- The "worldwide" containervessel should have special facilities for containers loaded with frozen cargo; chilled cargo, or cargo which needs to be ventilated, temperature and moisture controlled.
- The vessel should have space below deck to carry these reefers undisturbed as the alternative of stowage on hatchcovers will cause many shiftings of the container during loading/discharging operations in the intermediate ports.

- The integral reefer container is to be preferred over the ducted porthole box since the integral box is independent of the vessel and needs only to be plugged in. Provided sufficient aircooling is available in the hold (altern. watercooling), the integral reefer can be kept cold during ports stay and feeder-transport by plugging in only. Meanwhile the capital investment in the reeferbox is independant from the capital investment in the ships which makes the use of such equipment more flexible.
  Condition is however, that before the loaded integral reefer container goes on board, her machinery should be carefully checked (PTI)
- To the contrary, when a large quantity of fruit or frozen cargo is to be carried regularly and no terminal stay, and/or feeder transport in a hot climate will occur, then the system of insulated porthole boxes, connected to the ships cooling ducts is to be preferred, because of the simple reliability.
  In that case, however, we design a special ship suited to a special trade which is contrary to the required flexibility by management.

Breakbulk

- Special facilities for breakbulk below deck on tanktop can be considered underneath the container stacks to improve flexibility.

Deckload

- Hatchcovers to be strong and rigid enough to carry modules and long and heavy lifts which could be loaded by floating cranes.

Lashing

- Container stacking on hatchcovers up to 4 (5) tiers high. and preferably lashing by "twistlocks" only if the stackload upon the lowest box does not exeed 30 Ton. Over 30 Ton diagonal rodlashings are necessary to prevent racking on the lowest box.
- It must be realised that on many ships 50% of the cargo is carried upon hatchcovers as "deckload" and lashings are of enormous importance to safety of cargo, crew and vessel.

Rolling

- In this respect rolling should be avoided which should be done as far as possible by loading the vessel up to the right metacentric height (0,6 - 0,8m).

Range of stability

- Sufficient freeboard (upto 35° heeling angle) is to be strived after in order to give the vessel a sufficient range of dynamic stability.
  Too many smaller type container vessels do not comply with these basic requirements.

Special shiptypes.

From page 46 - 150 many special shiptypes are described, which have to take over the transportation which was carried out upto 1970 by the conventional cargoliner.

- container RoRo for protected sea and limited waterdepth.
- open coaster.
- heavy lift vessel.
- bow ramp landing craft.

- container feeder for coastal trade.
- container transporter over rivers and canals.
- small chemical tanker.
- 38000 product chemical tanker.

Ship Dedicated to the trade.

From page 150 - 163.

- Main figures necessary for a feasibility study what type of ship is most suitable for the trade.

Improving economy of the ship itself.

From page 167 - 230.

- Physical aspects of the liner vessel and improvements on economy,
  - speed
  - stability
  - course keeping
  - block coefficient
  - smoothness of hull
  - hullform
  - rudder
  - thrusters
  - fin stabilizers
  - propeller
  - shaftgenerator.

Example of Universal Containerschip "Nedlloyd Clarence".

PART II

AFTERBODY DEVELOPMENT.

Will regard to propulsion, fuel consumption and vibration + noise problems.

- Many single screw vessels suffer from propeller exited vibration, causing cracks, damage to instruments and hindrance to the crew (in particular with the accommodatio deckhouse "All-Aft").

- By application of a larger propeller diameter in combination with lower r.p.m., a higher propulsive efficiency can be obtained, however, what about the vibration troubles?
  Generally these troubles will become smaller when lowering the propeller tipspeed and tipload.
  It is a must to adapt the afterbody to the larger "wheel".
  How to shape the hull,however, how to keep the propeller immersed in ballast condition?
  How much tip clearence should be given in relation to wakepeak and tiploadpeak of the propeller ??

- Systematic step-by-step research should be carried out to obtain the influence on hull resistance, propeller efficiency, hull efficiency, cavitation pattern and hull presure fluctuations.

- The flow pattern around the afterbody is dominant to the problem and in our opinion flow separation should be avoided by all means (which is very difficult on full body ships).

- The application of pram shaped hulls with a slender gondola underneath, accommodating the dieselengine is promising.
  The weakpeak at 12 o'clock should be minimal.
  The propeller should turn in the least disturbed flow.
  On the real ship, (scale 1:1) this arrangement will require minimum horsepower and excite minimum vibrations generally speaking.

- Good and bad shapes are laid down in figures :
  Single screw, full body, slow ship (bulkcarrier - tanker - coaster).
  " " finer body (cargo liner) (tunnel - fins).
  " " container vessel.
  " " roro vessel.
  Twin screw , ferry (single skeg versus twin skeg).

- Fuel consumptions figures "an service" of container ships fitted with 2-3 propeller - diesel and steam are given.

- Single screw and twin screw are compared for the future.

PART III

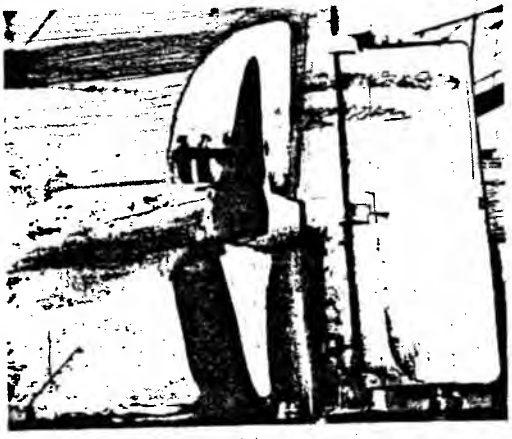
ABOUT STEERING.

Described are steering problems with twin screw ships having differences in arrangements :

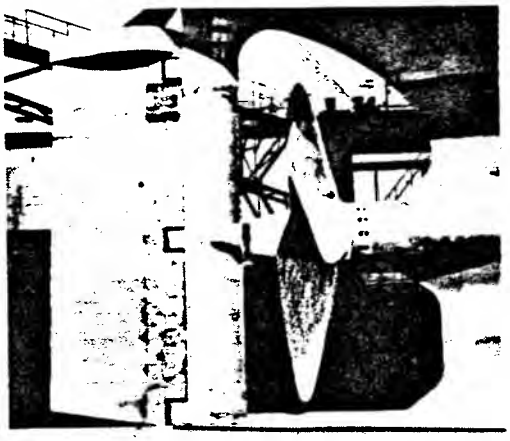
- Twin screw, single balance rudder.
  Twin screw, single skeg rudder.
  Twin screw, twin rudder.
  Single screw, single rudder, open skeg, propeller on pip
  Triple screw, single rudder.
  Outward/inward turning propellers : swinging with stern.
  C.P. propellers.
  Bow- and stern thrusters.

FNB/E. Vossnack

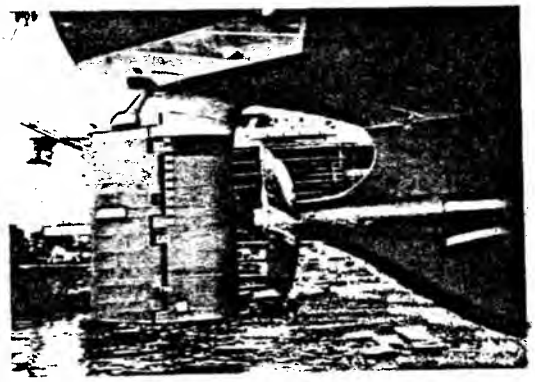
23rd August 1983.



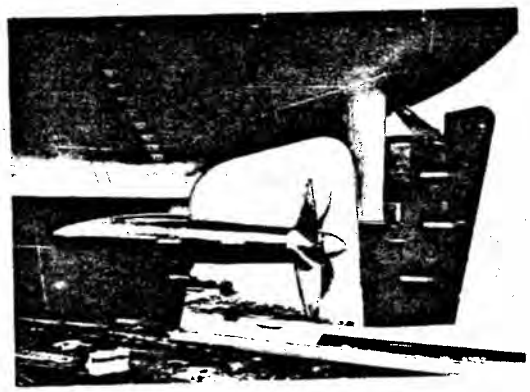
C3 vessel 1942. 8500 s.h.p./85 r.p.m.



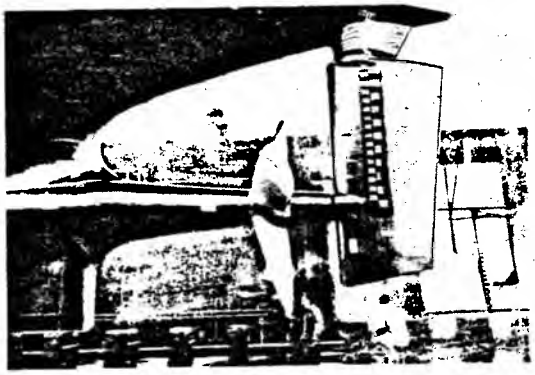
'S.-Lloyd' 1949. 10500 s.h.p./115 r.p.m.

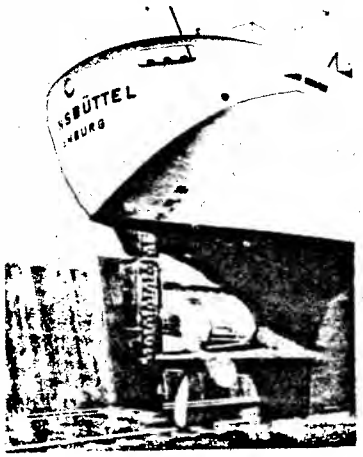


'Leuve Lloyd'. 17000 s.h.p./115 r.p.m.



'Koudekerk'. 14200 s.h.p./115 r.p.m.



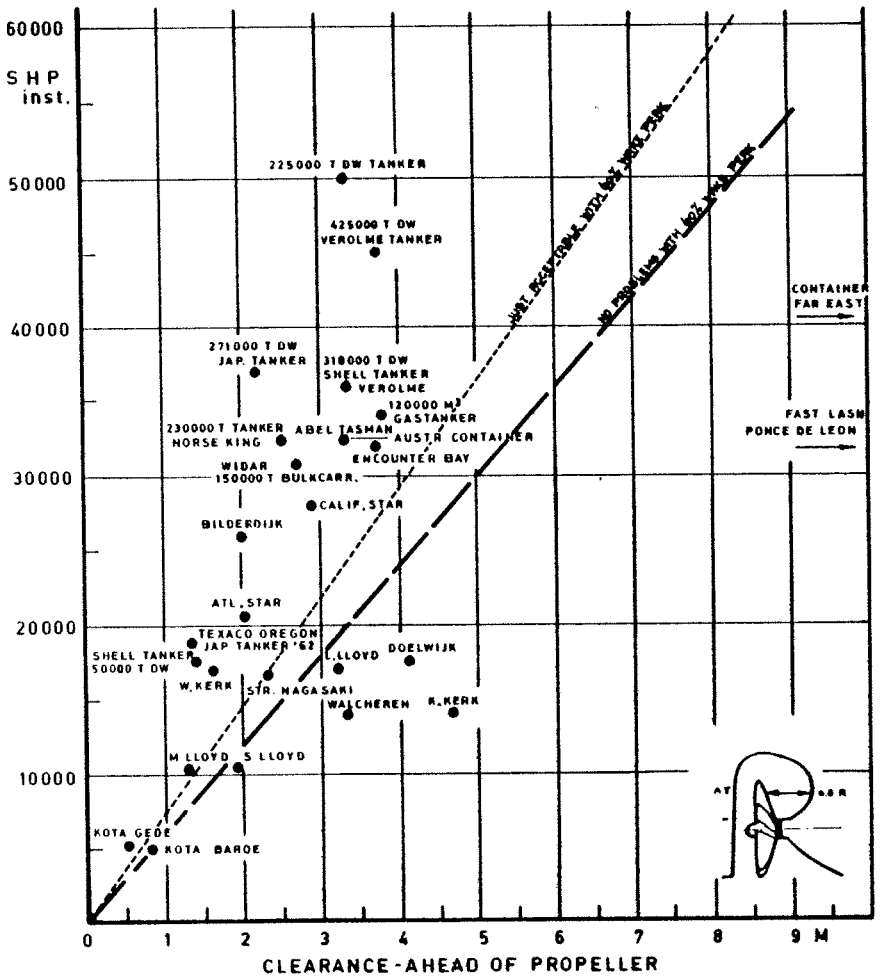


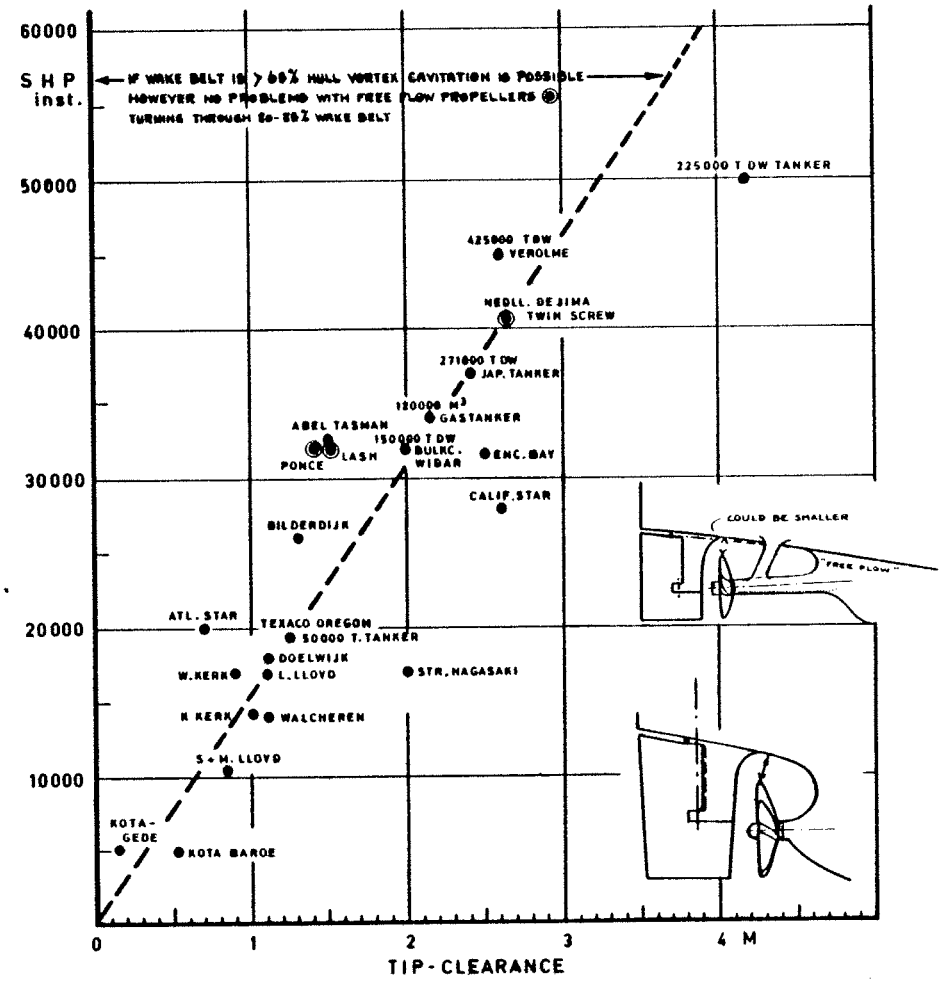
'Brunsbüttel'. Ir. E. van Dieren.

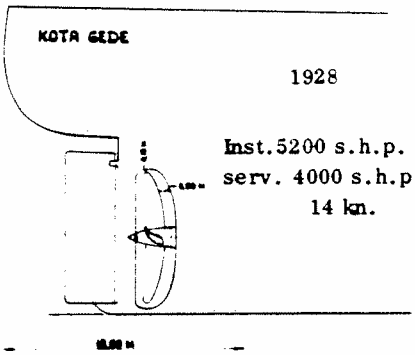
CARGOLINERS 16-19 kn. DEVELOPMENT TO FREE FLOW PROPELLER

TO RELATE CLEARENCE OF PROPELLER IN APERTURE TO % OF DIAM ONLY, IS FULLY INADEQUATE!

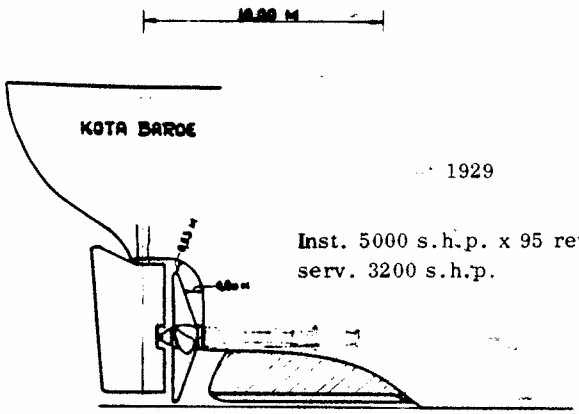
HORSEPOWER - PROPELLER LOADING - WAKE FIELD AND TIPSPEED TO BE TAKEN INTO ACCOUNT



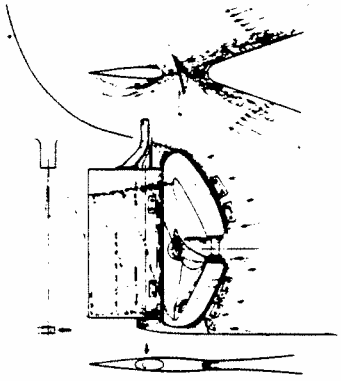




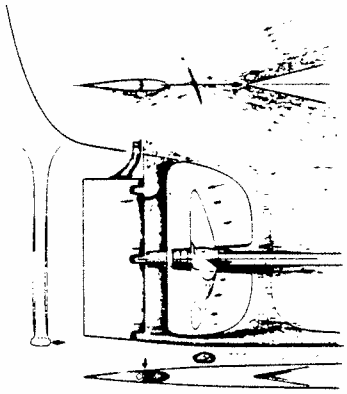
. Good arrangement



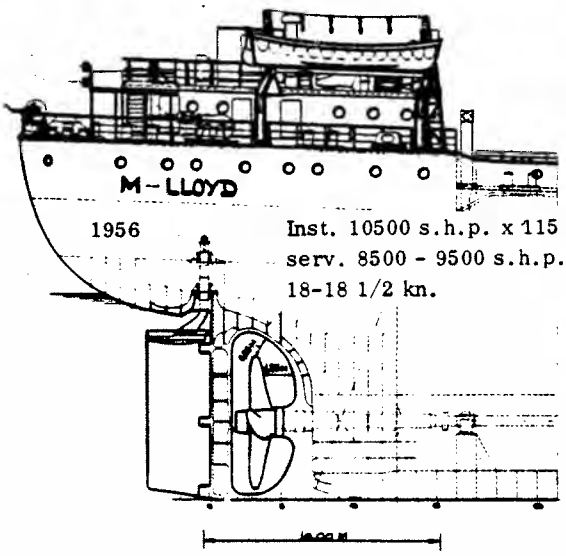
. Bad arrangement. Rudder was a cudgel, being too fat. Bad course keeping. Later on skeg fitted.



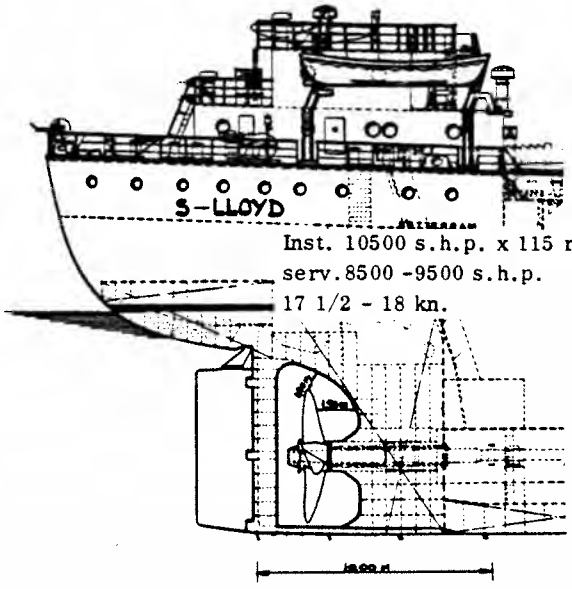
Normal bad construction 1950-1960. High maintenance costs. Strong thrust and torque variations. Wear of sterntube and simplex rudder bearings. Loosening of cone in sole piece. Cracks in sole piece at hull connection. Cavitation at propeller tips.



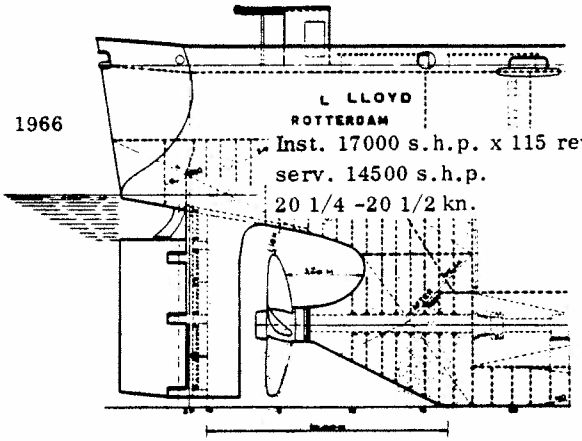
Improved construction 1955. Low maintenance on rudderbearings. No cavitation. However still high wake peaks in top and bottom position.



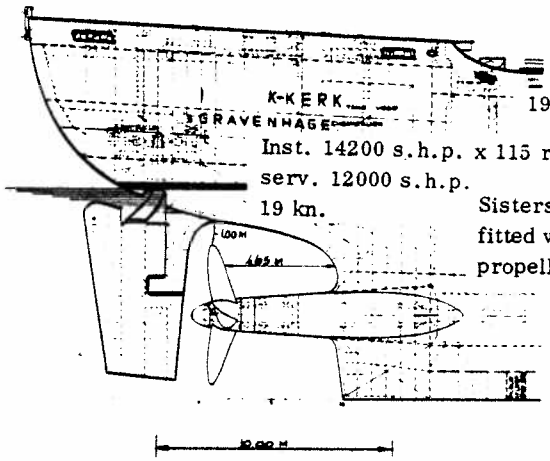
Cast steel. Vibration level in accommodation just acceptable.



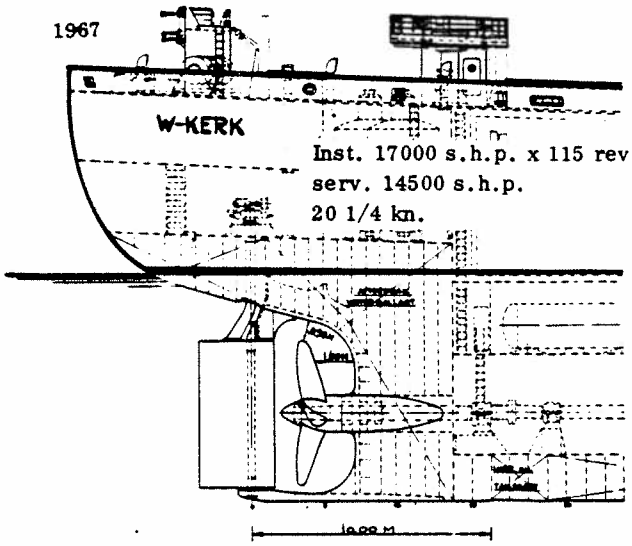
. Wide sternframe (all welded construction). Improvement in relation to 1E.



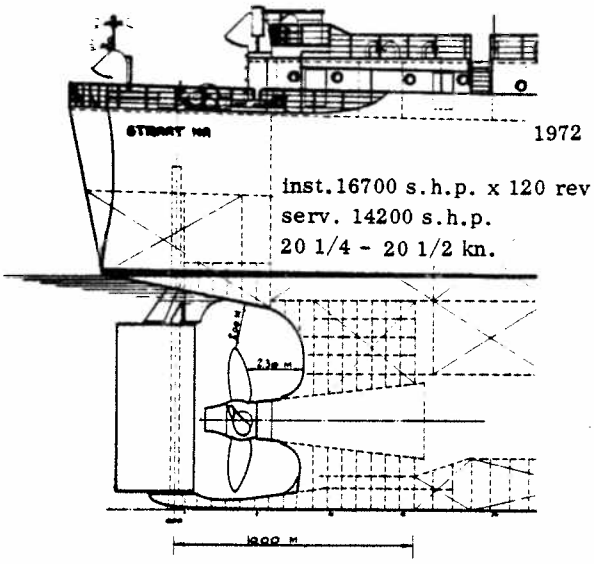
Good construction. No vibrations.
Just acceptable rumbling in stearinggear-flat because of small tipclearance.



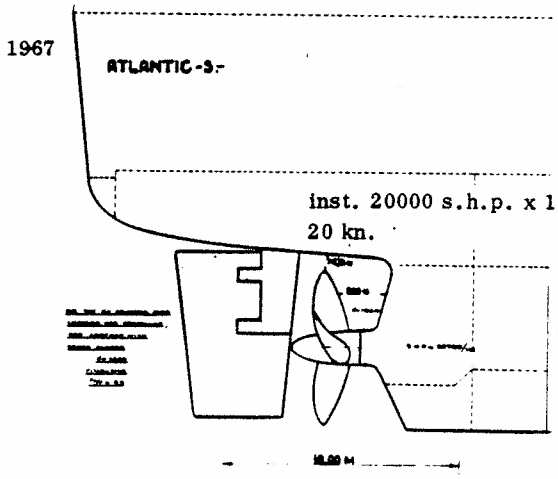
Good construction. No vibrations.
Rudderhorn too small for hard-over at full speed.
Heavy load on rudderpintle.



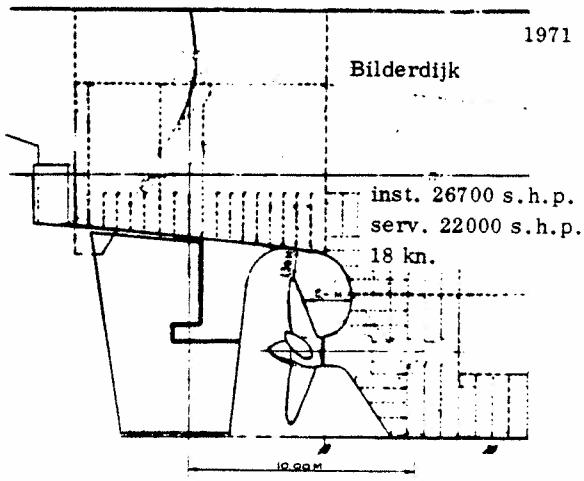
Suffering from vibrations. Too small clearance before propeller. Heavy solepiece ahead of propellertip. Crack below sterntube. High maintenance on cone of rudder.



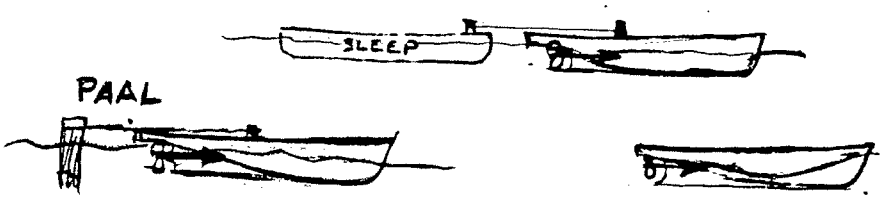
Suffering from vibrations. Too small clearance before propeller. Heavy solepiece ahead of propellertip.

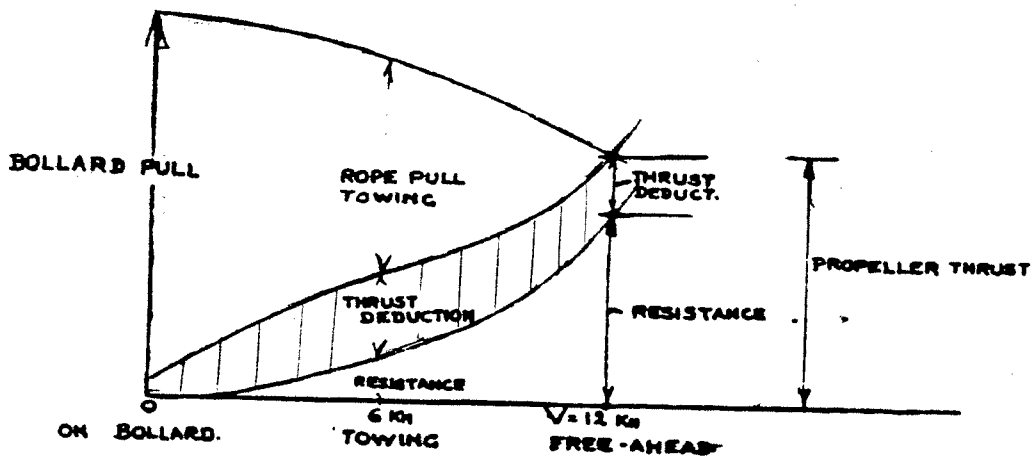


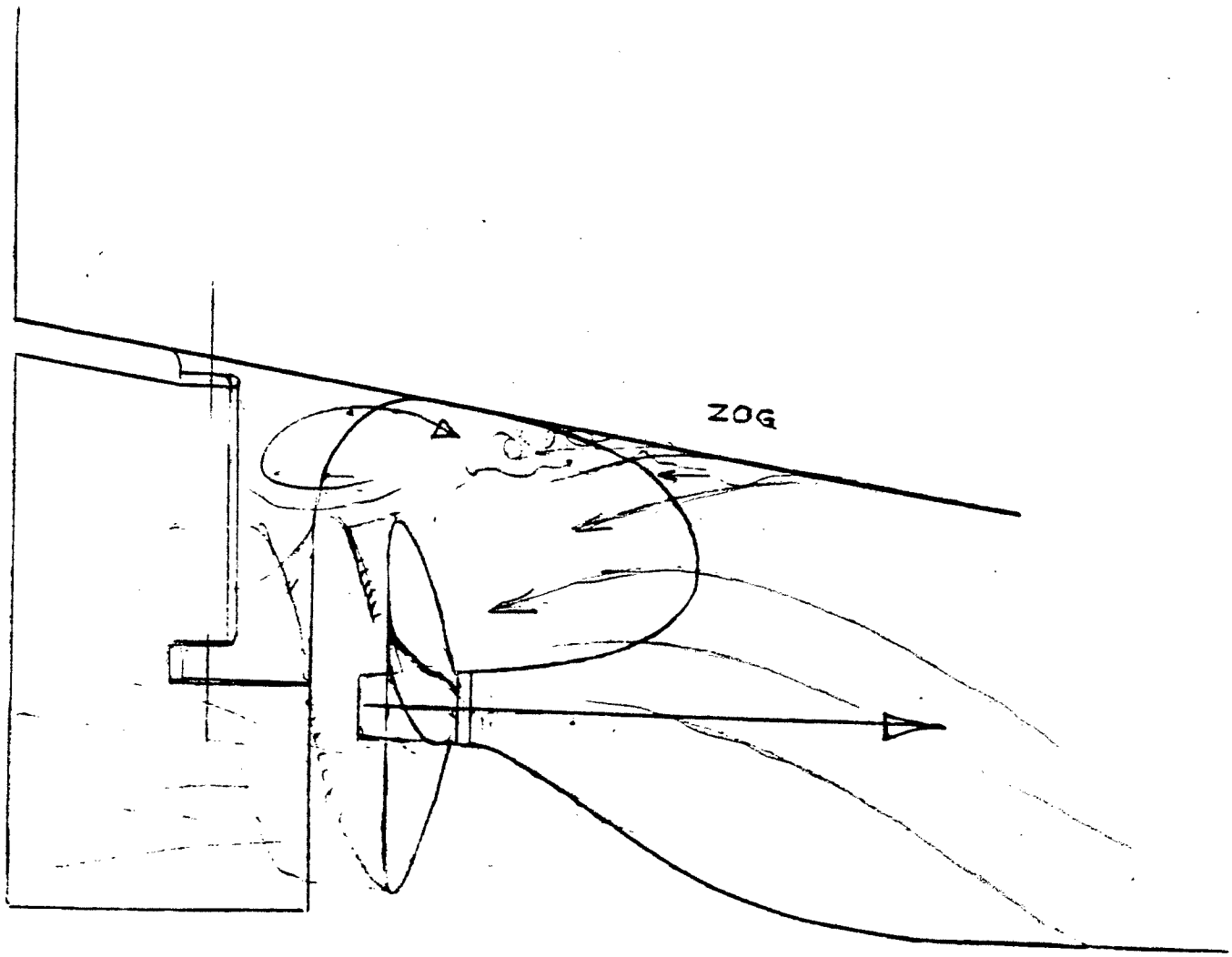
Too small tip-clearance. Hull vortex cavitation. Changed to smaller prop.dia., 7 blades. (on sistership fins above propeller were fitted).



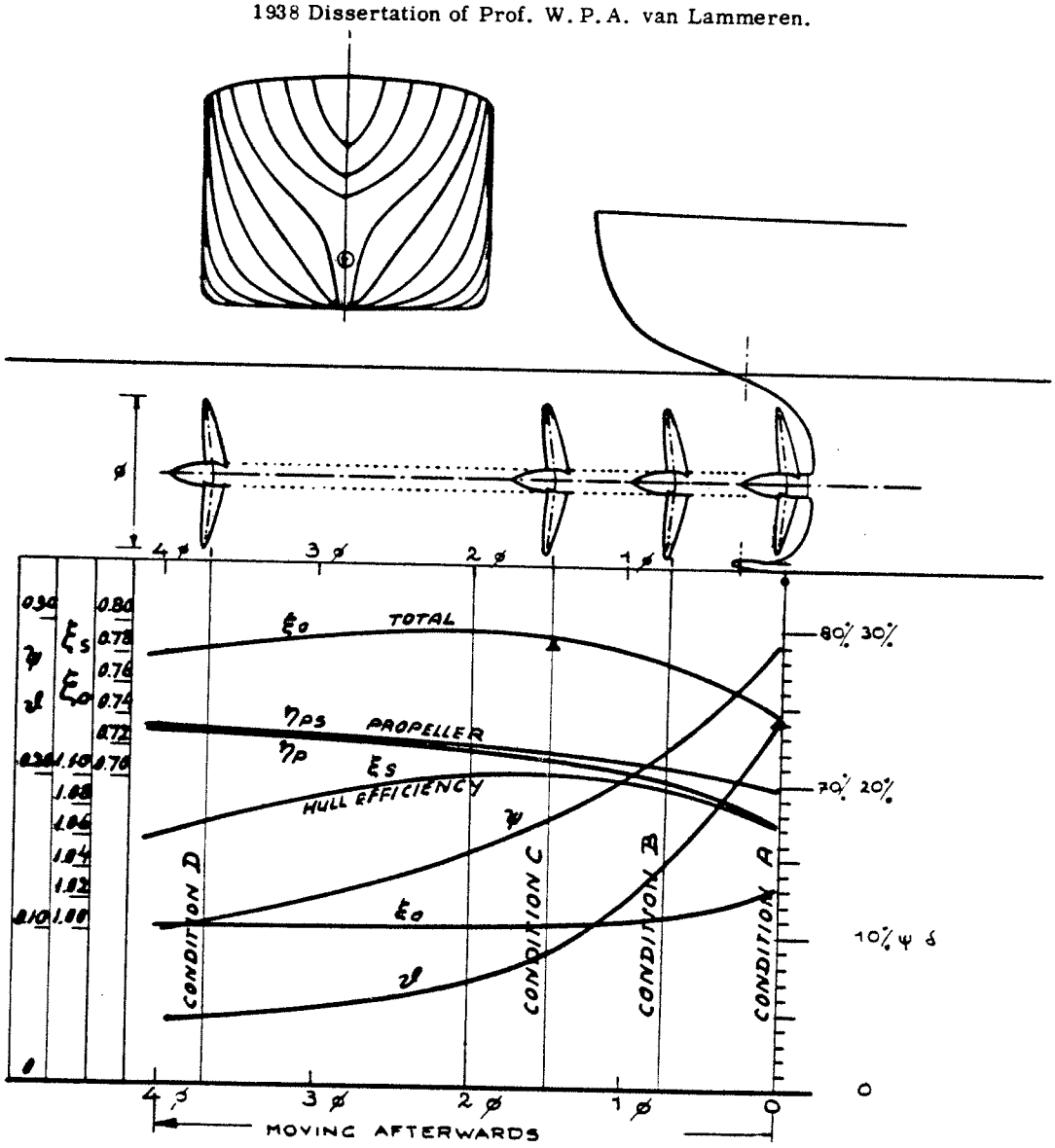
. Slow barge carrier 'Lash'.
Design Goldman-Somitono.


SLEEP
PAAL


BOLLARD PULL
ROPE PULL
TOWING
THRUST
DEDUCT.
PROPELLER THRUST
THRUST
DEDUCTION
RESISTANCE
RESISTANCE
O
ON BOLLARD.
6 Kn
TOWING
V = 12 Kn
FREE-AHEAD


ZOG

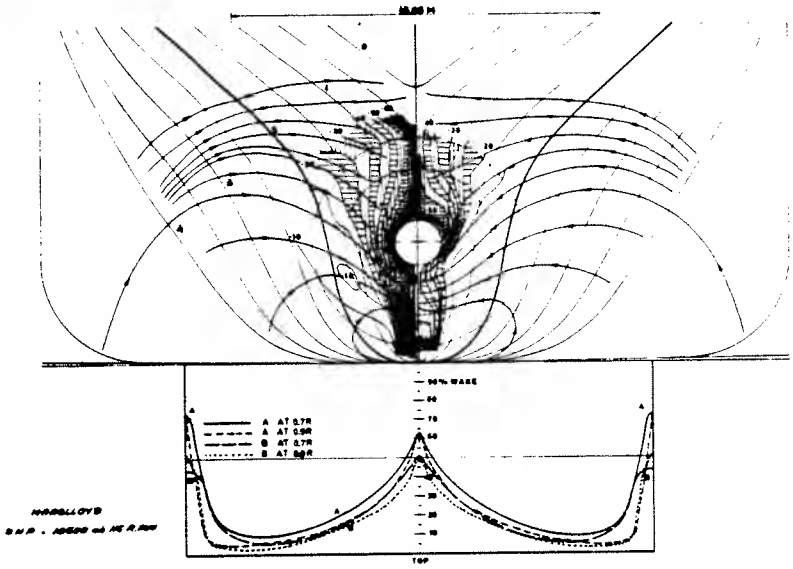
1938 Dissertation of Prof. W. P. A. van Lammeren.



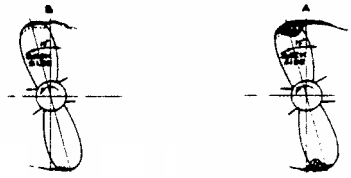
Toe-stand	$\xi_o = \frac{N_o}{N_w}$	ψ	θ	$\xi_s = \frac{1-\theta}{1-\psi}$	η_{ps}	η_p	$\xi_a = \frac{\eta_{ps}}{\eta_p}$	N/min
A	0,745	0,287	0,236	1,072	0,695	0,674	1,031	75,0
B	0,778	0,218	0,144	1,096	0,710	0,702	1,012	75,9
C	0,793	0,173	0,087	1,105	0,718	0,715	1,005	77,1
D	0,783	0,107	0,044	1,070	0,732	0,728	1,006	79,8
∽	0,734	0	0	1,000	0,734	0,734	1,000	84,4

Analysing propulsive components. s.s. 'Simon Bolivar', model without rudder.

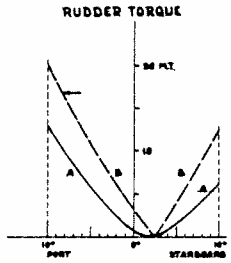
'M. -Lloyd'



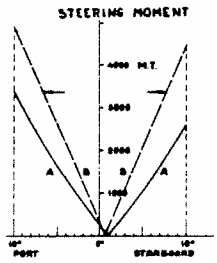
Effect of moving. The propeller aft. Smaller wake peak.



Less cavitation.
Less vibration.



Increased rudder torque.



Improved steering.

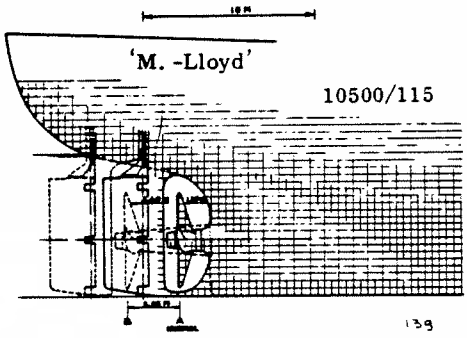


Figure 33C.

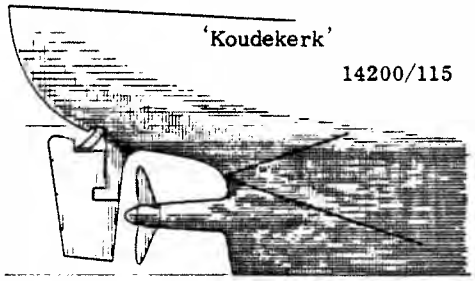
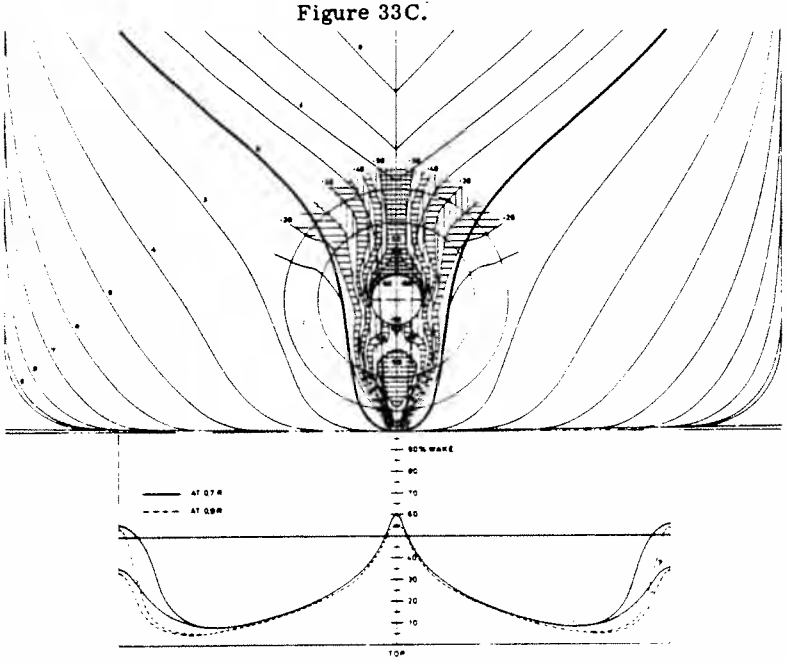


Figure 33D

1-2% less h.p. required.
Dependent on thickness of rudder.



From the model test wake field it could be concluded that the wake peak evoked by the skeg is noticeable far behind it, but this effect is reduced on full scale.

Fast cargoliners.
'Leuve Lloyd'

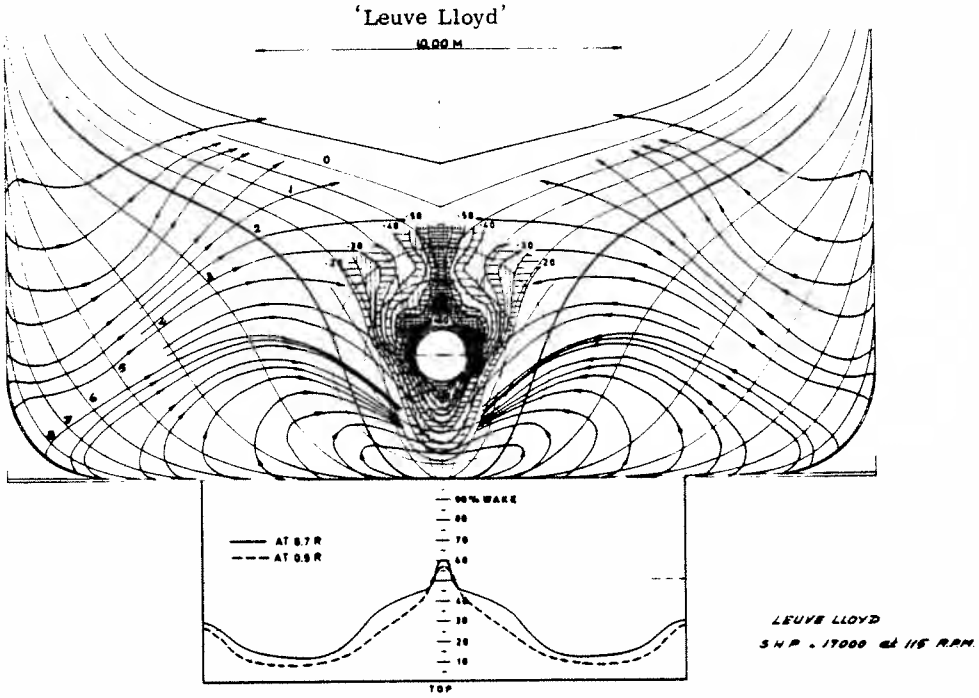
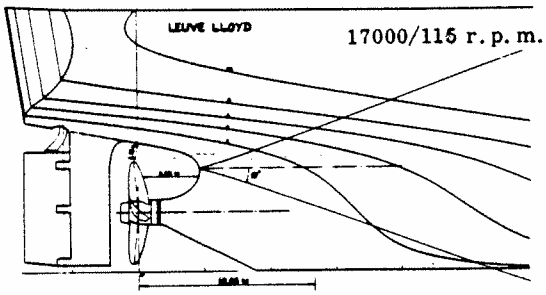
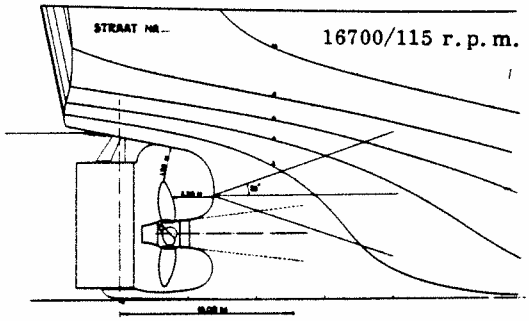


Figure 33A.

Conventional underwater bodies with transom stern.



Small tip-clearance. Rumbling of steering. Gear flat.



Small clearance before propeller. Big sole piece. Suffering from vibrations.

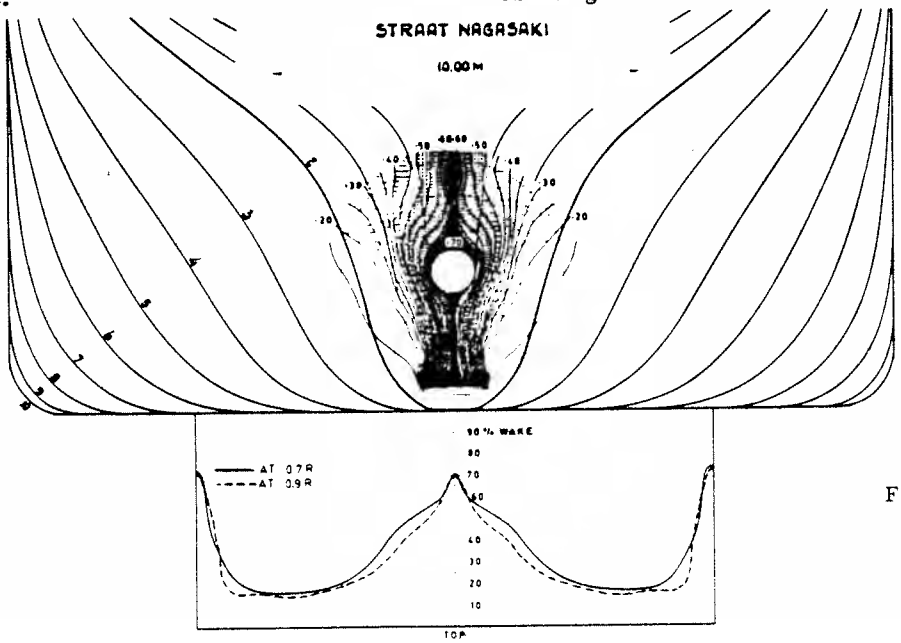
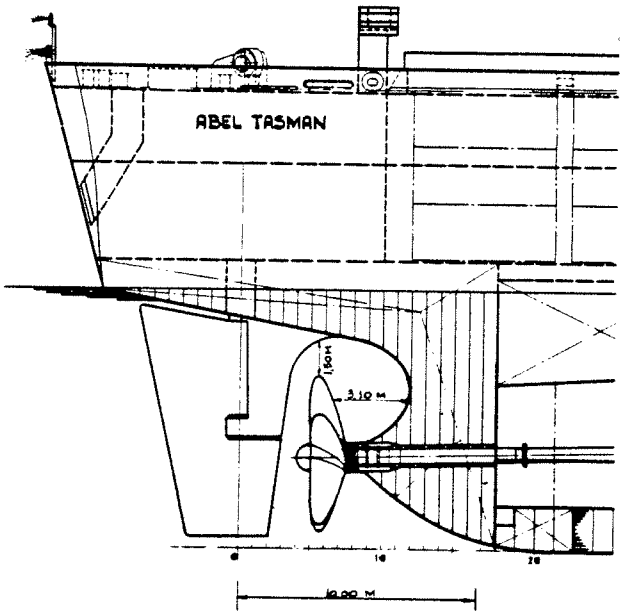


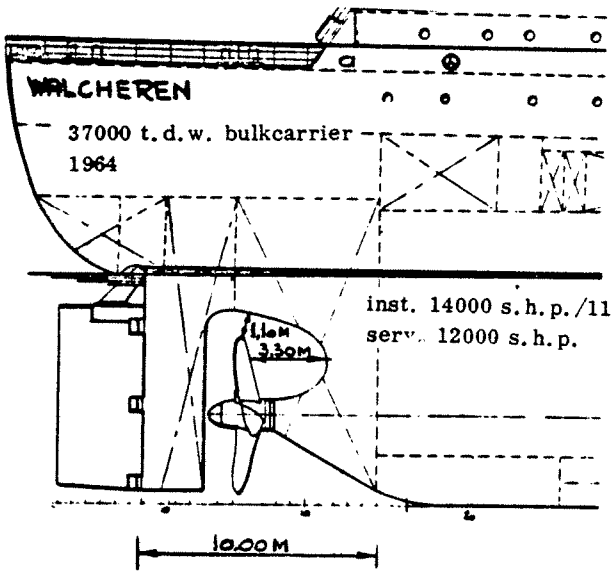
Figure 33 B.



Sidney Express 1970, Containership
Abel Tasman 1971

inst. 32450 s.h.p. x 110 rev
serv. 26600 s.h.p. = 82%
21-21 1/2 kn.

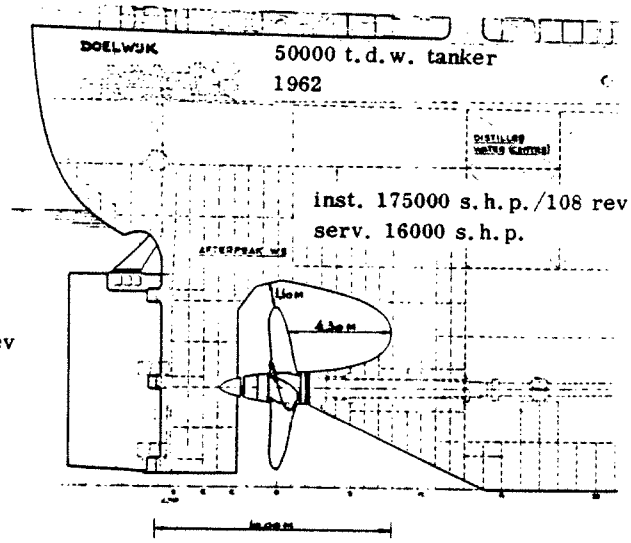
. Suffering from vibrations in afterbody above 26000 s.h.p. Too small clearance of aperture above propellers. Rudder and propeller to be located more aft. (Fortunately the accomodation is located away from stern.



37000 t.d.w. bulkcarrier
1964

inst. 14000 s.h.p./115 rev
serv. 12000 s.h.p.

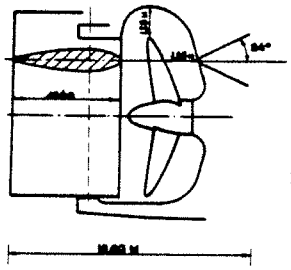
Good construction.
No propeller excited vibrations.



50000 t.d.w. tanker
1962

inst. 175000 s.h.p./108 rev
serv. 16000 s.h.p.

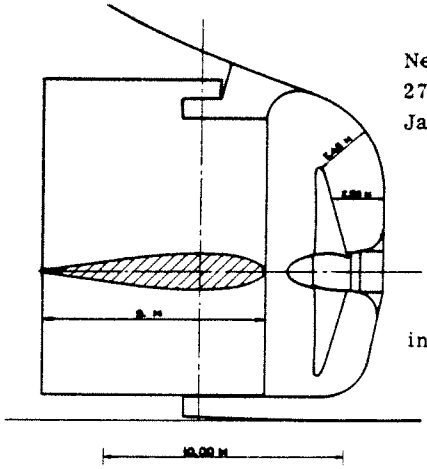
. Good construction.
No propeller excited vibrations. Good steering at low speed.
No maintenance on rudder.



Texaco Oregon
Japan 47000 ts tanker
1962

inst. 19000 s.h.p./105 rev
serv. 17100 s.h.p.

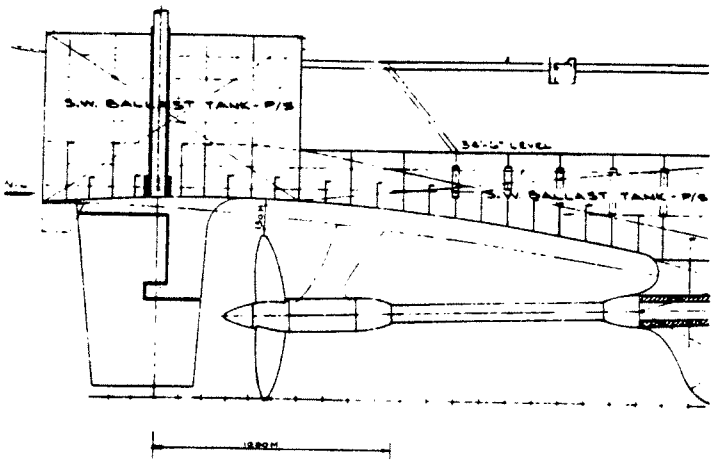
Propeller excited vibrations.
Too small aperture. Big sole piece. High maintenance on rudder.



Newbuilding
271000 ts tanker
Japan 1973

inst. 37000 s.h.p.
/90 rev

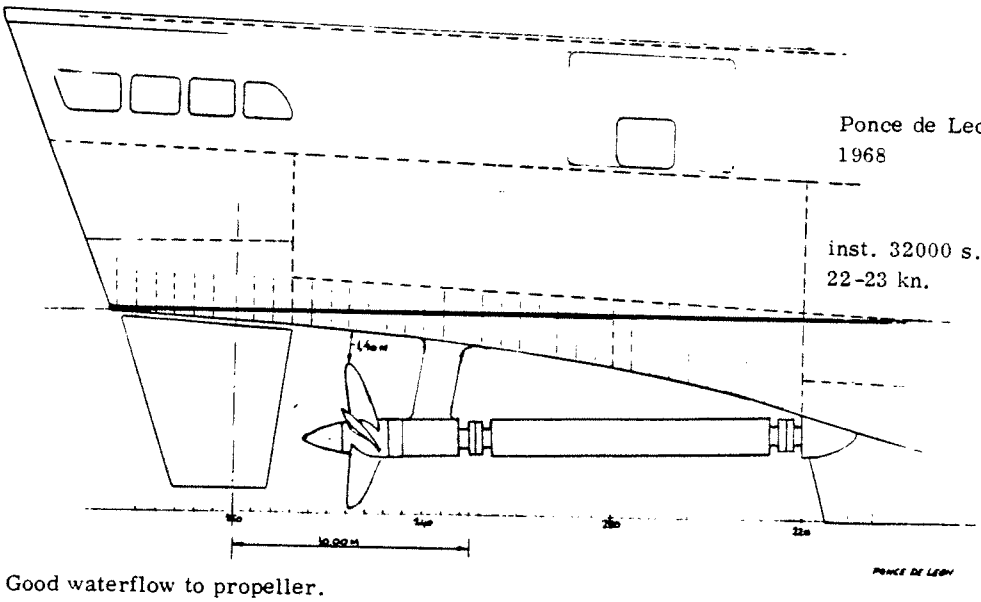
. Cavitation and heavy propeller excited vibrations. Too small apertures. Big sole piece. High maintenance.



Thomas e Cuffe
1970

fast barge carrier
Lash type
design Goldman
inst. 32000 s.h.p. x 105 rev
21-21 1/2 kn.

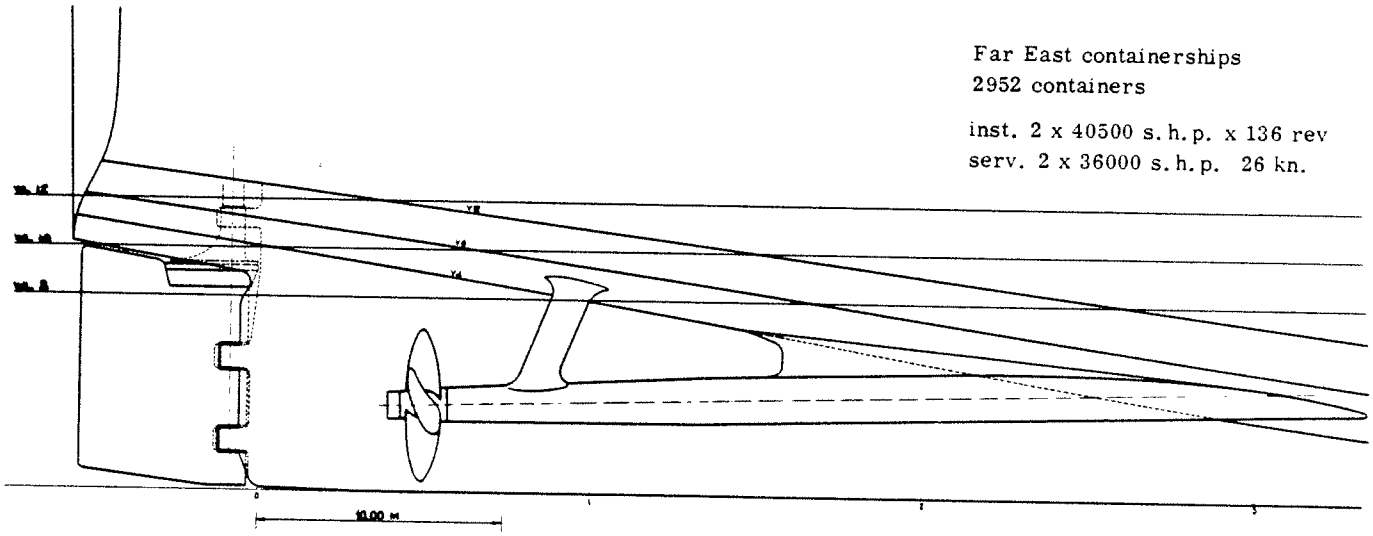
Optimum waterflow to propeller.
Bare shaft between tailshaft and sterntube. Fairing caps should be 100% fixed. Difficult to obtain reliable corrosion protection of shafting.



Ponce de Leon, Trailership
1968

inst. 32000 s.h.p. x 128 rev
22-23 kn.

. Good waterflow to propeller.
A hollow torque tube between tailshaft and sterntube. Difficulties with corrosion protection.

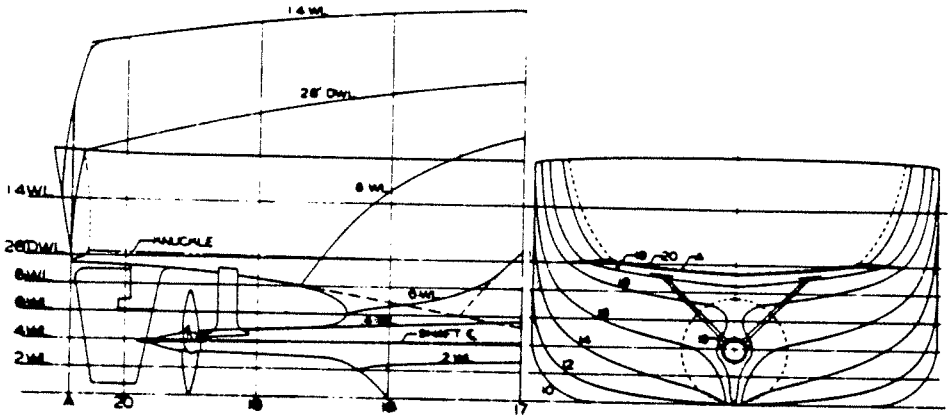


Far East containerships
2952 containers

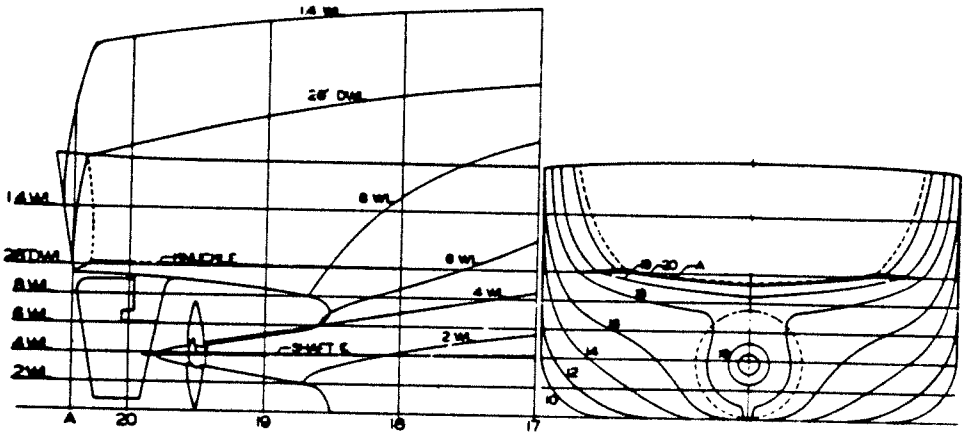
inst. 2 x 40500 s.h.p. x 136 rev
serv. 2 x 36000 s.h.p. 26 kn.

. Good waterflow to propeller - no vibration troubles. Large tip clearance. Propeller extended on Mannesmann tube supported by brackets. Covered shafting is always accessible. Bearings could eventually be adjusted if necessary.

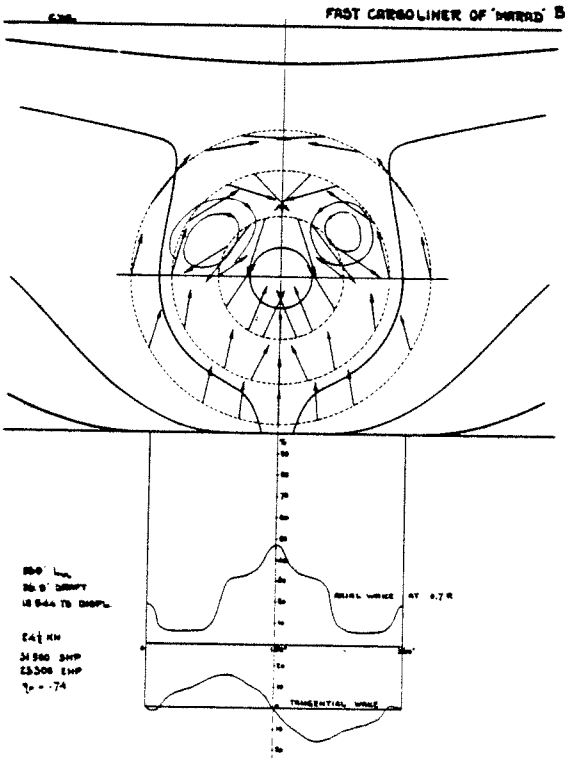
'Marad'-investigations on fast ships. SNAME. Transactions 1962.



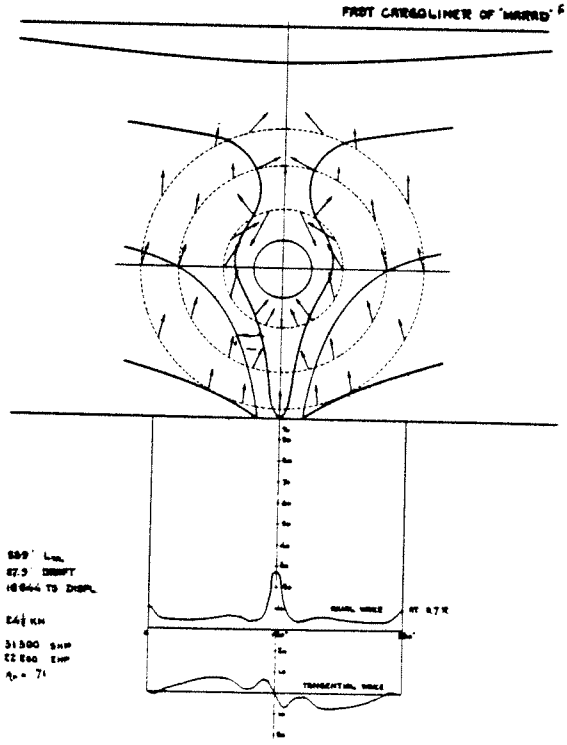
Shaft supported by brackets.



Hogner cigar.

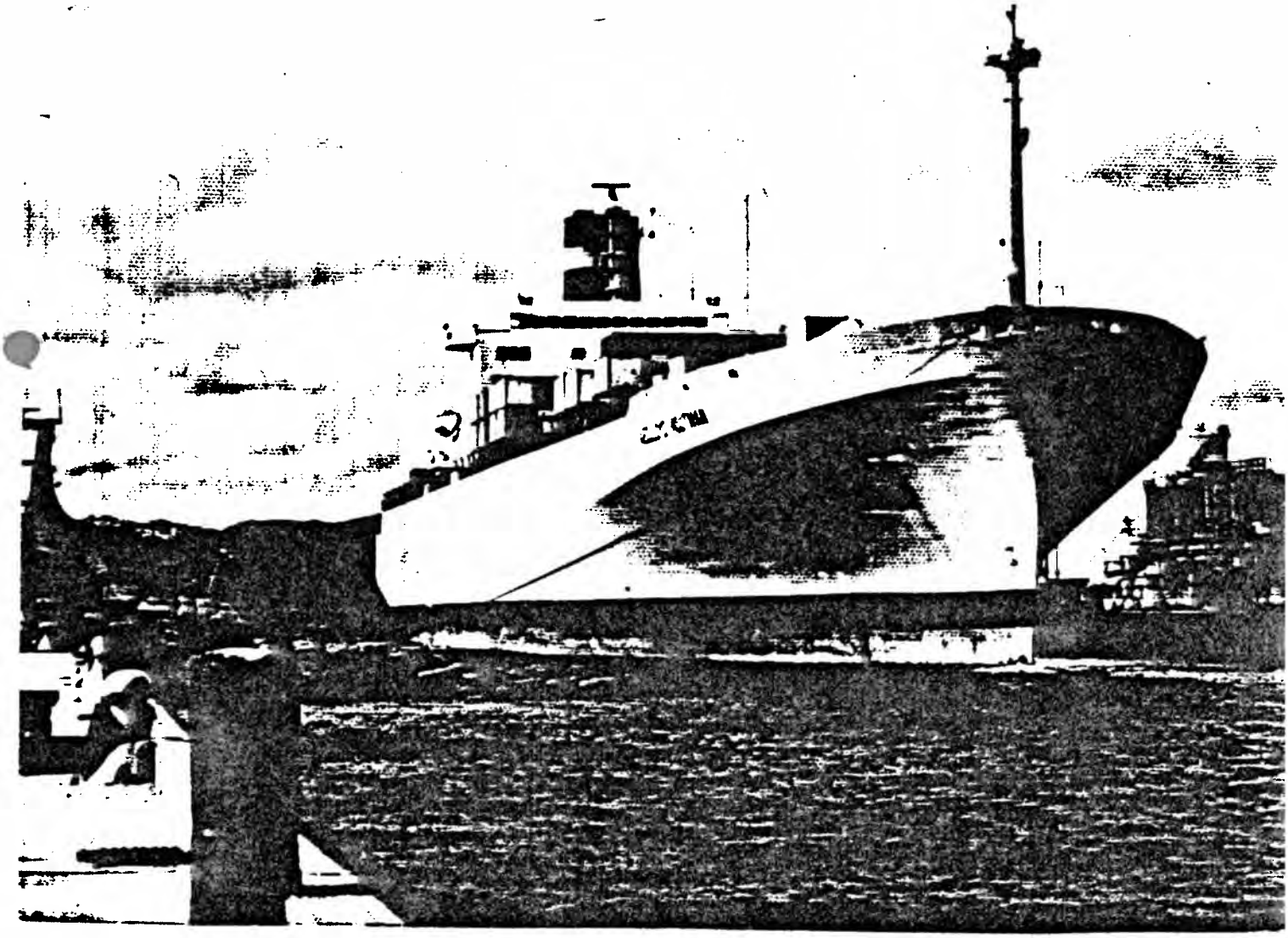


Increased:
- Axial wake peak
- Tangential wake



- Narrow sector of moderate axial wake
- Small tangential wake. Due to slender shaft tube.

1 S - 2 S CONTR.



NEDLLOYD HOUTMAN

CELL. CONTR.

LINE	ABEL TASMAN	COLOMBO	CLEMENT	CLARENCE	HOUTMAN + HOORN	DEJIMA + DELFT	PANMAX	4 TIER EXTRA 3 ROWS WIDER
CONTR. YARD	1968	1980	1981	IDEM	1974	1979		
L_{OA}	225.87	211.19	210.00	210.00	258.00	287.02	289.50 350'	289.50
L_{pp}	210.00	195.00	202.00	202.00	247.00	273.00	277.00	277.00
B	30.50	32.20	32.24	32.24	32.26	32.24	32.24 106'	39.60
H	16.40	19.00	18.80	18.80	24.15	25.00	24.15	24.35
T_{MAX}	11.58	11.56	12.00 EXCL CRANE INCL REEFER	12.00 GEARLESS	13.00	12.72	13.00	13.00
LIGHT SHIP	14 200	13 000	14.998	14.543	22.281	24.899	24 700	29.800
Q AND APP % IN LOA	30.76	79.3	24.23	24.31	109.00	113.34		
KG	11.78	12.81	12.50	12.39	13.58	13.50		
CLASS/SYSTEM	GERM. LLOYD	LLOYDS	LLOYDS		LLOYDS	LLOYDS		
TEU - HOLD	861	1014	1150	1150	1794	2020	2264	2720
TEU ON DECK	598 2 TR	511 2 TR	558 2 TR	562 2 TR	660 2 TR	668 2 TR	842 2 TR	1036
TEU	218 4°	252 3°	285 3° + GANTRY	291 3°	260 3°	264 3°	364 3°	448
TEU	1677 4 TR	1777 3 TR	1997 3 TR	2003 3 TR	2714 3 TR	2952 3 TR	3470 3 TR	4204 3 TR
REEFER HOLD	100 20'/8' CONAIR TYP	0	116 40'/8' CONAIR	0	352 20'/8' CONAIR	0		448 40'
REEFER DECK	0	130 R 20' WATERC	64 R 20' WATERC 15'	88 R 20' AIRC.	886 20'/8'	70 R 40' WATERC	80 R 40' WATERC 70 R 20'	4 652 4 TR
REEFER DECK	20 R 20'/40'	88 R 20/40'	99/68 R 20'/40'	99/88 R 20'/40'	53 R 20/40'	50 R 20/40'	121 R 20'/40'	
M. ENGINE	TURB. STAL LAVAL AP 32	SULZER 6RLA 90	B+W 6L90GBE	IDEM	SULZER 2 x 8 RND 90 M	SULZER 2 x 8 RND 90 M	(2 x 7 CYL L 90 GBE)	(2 x 7 CYL RTA 84)
MCR/RPM	32.450 - 1101	20.400 - 90	23.800 - 97		2 x 26.800 - 122	2 x 26.800 - 112	2 x 28.300 - 97	2 x 28.210 - 87
85%	SHP 27.600 - 110 RPM	SHP 17.340 - 90 RPM	SHP 20.230 - 97 RPM		SHP 2 x 22.800 - 122 RPM	SHP 2 x 22.800 - 112 RPM	SHP 2 x 24.050 - 97 RPM	2 x 24.000 - 87 RPM
GEN.			2 x 1200 KW / 2 x 1800 KW				2 x 1800 KW	2 x 1800 KW
	Ø 7400 - 5 BL - P/D = 0.636	Ø 7400 - 5 BL - P/D = 0.581	Ø 7200 - 4 BL - P/D = 0.63		Ø 6850 - 4 BL - P/D = 0.558	Ø 6618 - 4 BL - P/D = 0.558	Ø 7000 - 4 BL - P/D = 0.592	Ø 7350 - 4 BL - P/D = 0.66
MODEL SHIP	HSVA / NEW UPS	5736 B / 3323	5901 C / 5681	IDEM	4751 OTB / 5011	5802 DMB / 4602	5782 / 4449	
DR/DISPL.	10.67 / 42.870 T	10.60 / 40.430 T	10.60 / 44.800 T		11.50 / 60.800 T	11.50 / 62.600 T	11.50 / 70.200 T	11.50 / 84.200 T
BLOCK / MODEL	6..616 / 41.805 M³	6..592 / 39.434 M³	6..614 / 43.774 M³		6..647 / 59.288 M³	6..601 / 61.100	6..666 / 68.475 M³	6..666 / 84.100 M³
V SHIP / TRIALS	20.. KN / 19.600 HP	19.. KN / 16.300 HP	19.. KN / 17.250 HP		21.. KN / 28.700	21.. KN / 28.200	21.. KN / 29.220	21.. KN / ?
115 SERVICE	22.550	18.750	19.830		23.000	32.400	33.600	?
TON/DAY	225 x 24 = 122..	150 x 24 = 67.5	147 x 24 = 70.0		162 x 24 = 128..	157 x 24 = 122..	147 x 24 = 113..	147 x 24 =
	TO BE IMPROVED							
$D^{2/3}V$ / SHP	1223 x 8000 / 19600 = 500	1177 x 6859 / 16300 = 495	1261 x 6859 / 17250 = 502		1547 x 9261 / 28700 = 500	1578 x 9261 / 28200 = 518	1702 x 9261 / 29220 = 539	1951 x 9261 = ?
D^3V^3 / TON/DAY	1223 x 8000 / 122 = 81	1177 x 6859 / 67.5 = 120	1261 x 6859 / 70.0 = 124		1547 x 9261 / 128 = 112	1578 x 9261 / 122 = 120	1702 x 9261 / 113 = 132	1951 x 9261 = ?

	ABEL TASMAN	COLOMBO	CLEMENT	CLARENCE	HOUTMAN + HOORN	DEJIMA + DELFT	PANMAX	3 ROWS WIDER
DR/DISPL.	11.23 45.917	11.56 45.816	11.98 52.992	11.98 52.992	13.03 71.343	12.72 71.683	12.46 77.635	13.00 101400
LIGHT SHIP	14.200	13.000	14.998	14.545	HOUTMAN 22.281	24.899	24.700	29.800
D.W.	31.717	32.816	37.994	38.447	49.062	46.786	52.995	71.200
CONS + WB	NOT MEN 8.163	NOT MEN 4.456	6.394	6.347	11.386	14.506	10.455	11400
TEU = T/TEU	AT 1589 x 14.8 = 23.454	AT 1777 x 16 = 28 360	2000 x 15.8 = 31 600	2000 x 16 = 32.100	2714 x 13.9 = 37.676	2988 x 10.9 = 32.278	42.540	AT 4284 x 14.2 = 59800
IN HULL	988 x 15.7 = 13.842		1150 x 15.8 = 18.170		1794 x 4.2 = 3 ...	total = 1.07 = 26 394		AT 4652 x 11.8
ON DECK	AT 86 x 12.. = 2.612	AT	850 x 15.8 = 13.480		920 x 8.2 = 7594	832 x 6.31 = 5884		TEU
STABILITY	KM = 13.09 KG = 12.25 MG = 0.67 - 0.05 = 0.62	KM = 15.88 KG = 14.95 MG = 0.43 - 0.07 = 0.36	KM = 15.07 KG = 14.01 MG = 0.26 - 0.02 = 0.24		KM = 13.14 KG = 14.64 MG = 0.50 - 0.12 = 0.38	KM = 14.35 KG = 14.15 MG = 0.70 - 0.04 = 0.66	KM = KG = MG = 0.40	KM = KG = MG =
DR/DISPL.	10.60 41.377	11.17 42.776	10.57 45.139	10.57 45.139	12.00 64.097	12.00 66.303	12.24 75.845	12.50 95.800
LIGHT SHIP	14.200	13 000	14.998	14.545	22.281	24.899	24 700	29.800
[illegible]	28 418	30.776	30.141	30.594	41.716	41.404	51.145	66.000
CONS + WB	8.958	4.456	4.741	4.694	10.400	11.588	11.365	~ 11.000
TEU = T/TEU	1389 x 13 = 19.600	AT 1316 x 20 = 26.320	2000 x 12.7 = 25.400	2003 x 12.5 = 25.300	2454 x 12.9 = 31.316	2417 x 12.9 = 29.806	2.. x 10.7 = 29.780	AT 4284 x 12.1 = 55000
IN HULL	986 x 13 = 10.250		1150 x 12.7 = 14.600		1794 x 13.9 = 23.922	2.. x 12.9 = 26.394	3106 x 12.6 = 39.06	AT 4652 x 11.8
ON DECK	719 x 13 = 9.350		850 x 12.7 = 10.800		660 x 11.2 = 7 394	597 x 8.6 = 5412	6.. x 1.. = 78	
STABILITY	KM = 12.88 KG = 12.39 MG = 0.49 - 0.05 = 0.44	KM = 13.36 KG = 13.11 MG = 1.22 - 0.07 = 1.16	KM = 13.05 KG = 14.86 MG = 0.19 - 0.04 = 0.15		KM = 13.10 KG = 14.58 MG = 0.52 - 0.12 = 0.40	KM = 14.16 KG = 14.16 MG = 0.68 - 0.04 = 0.64	KM = KG = MG = 0.48	KM KG MG
DR/DISPL.		10.95 42.744	9.75 40.440	9.75 40.440			11.24 67.655	
LIGHT SHIP		13 000	14.998	14.545			24 700	
D.W.		29.744	25.442	25.895			42 955	
CONS + WB		4.456	2.002	1.955			WASTE DEP 9.045	
TEU = T/TEU		AT 1525.166 = 25.288	1804 x 13 = 23.440	1840 x 13 = 23 940			43' 20' x 11.5T 40' x 14.5T 33.910	
IN HULL							3930 x 8 = 31.470	
ON DECK		AT					220 x 2 460	
STABILITY		KM = 13.33 KG = 14.10 MG = 1.22 - 0.07 = 1.15	KM = KG = MG =					
DR./DISPL.		7.88 27.865	9.15 37.340	9.15 37.340				
LIGHT SHIP		13.000	14.998	14.545				
D.W.		14.865	22.342	22.795				
CONS + WB		EMPTIES = 11.711	2.002	1.955				
TEU = T/TEU		1853 x 1.7 = 3.154	1630 x 12.5/10.340	1670 x 12.5 = 20.840				
IN HULL								
ON DECK								
STABILITY		KM = 13.46 KG = 10.10 MG = 3.36 - 0.11 = 3.25	KM = KG = MG =		DOCKED R = 20' HOUTMAN 333 HOORN 886 LIGHT SHIP L = 22.706			CORR 27 NOV 83 GV

INTRODUCTION.

Modern trends in cargo liner ship operation call for changes in ship design in various aspects, almost all aiming towards improving their economy. Differently from ten years ago, changes are aiming mainly towards savings in propulsion fuel expenses. This is because where fuel costs in the late sixties were just 'part of total' costs, they nowadays are dominant.
Therefore emphasis in changes is laid very much on improving the

SPEED versus FUEL CONSUMPTION

Consequently, whilst adding improvements in other fields as well, we arrive at a number of changes in design,as follows:-

- Slow-turning large-diameter propeller(s).
- Slow-turning long-stroke diesel engine main mover with a minimum number of cylinders.
- Continuous optimalisation of hull form, especially at stern - and bow.
- Excellent surface preparation and painting.
- "Self-eroding" underwater antifouling giving minimum friction resistance.
- Shaft generator.
- Thrusters at bow and stern - for ease, safety and rapidity of manoeuvring around of (relatively) long ships in confined spaces.
- 4 to 5 mooring winches fore and aft.
- Utmost simplicity in lay-out and in construction.
- Minimum number of crewmembers.
- Sophisticated and reliable equipment for control and communication.
- Reefer containers to be of integral type if their number is relatively low, say below (abt) 400 teu, per vessel.

HULL FORM

The hull of the fast sailing ship of 1850, having a "cods-head" and a "mackerel-tail" has developed into a seagoing motorlaunch having a V-shaped forebody and a pram type transom stern.
This will provide sufficient seaworthness and stability.

- Prismatic coefficient
  The optimum prismatic coefficient as function of speed-length ratio is given in a graph (:) . The coefficient normally lies between 0,61 and 0,68. Higher fuel prices will tend to finer ships and in each case a rentability calculation should be made. Fuel costs are about 1/3 of the total transportation costs as far as the ship is concerned. In case the vessel has to bear a certain cargoweight at a threshold draft, it pays to keep the fuller prismatic coefficient.

- Afterbody
  From 1850 the transom stern developed into an elliptical stern , since 1920 the elliptical stern became a cruiser stern. From 1968 the cruiser stern is cut off to a transom stern and in the year 1970 the transom is widened to full-ships-breadth.
  - The new afterbody is in point of fact a well approved old one on fishing vessels, Buttocks steepness to be 9-11° (max. 12,5^{o}) should not be exceeded in order to prevent flow separation.
  - One of the main advantages of the wide - pramtype - stern is the large stability, hold space, and deck-surface within a given over-all-length.

  Also rolling angles are smaller in comparison to fine-lined cruiser-stern vessels under same conditions at sea.
  - The disadvantages of the pram type transom stern are:
    1. the larger frictional surface in comparison to a cruiser stern hull.
    2. the slamming, which sometimes occurs during heavy weather or laying on the roads in a large "Atlantic" swell.

3. Flat sterns above the propeller are also more prone to transmit the propeller-excited-pressure-fluctuations into the hull structure.

- Midship
  - A large bilge radius is favourable for obtaining low resistance.
    Nedlloyd prefers a soft bilge above inclined shipsides with a hard bilge.
  - Possibility of rolling should in first place be counteracted by loading the vessel to the right metacentric height; preferably GM=0,4 to 1,0 m (Shipsplanners' duty.)

  (Fins have their drawback nowadays by the high fuel price and the lower service speed which asks for larger fins).
- Antirolling tanks should be very large to have effect. Adequate corrosion protection against a heavy waterwash is not solved yet.

- Forebody

  Originally "Bremen/Hongkong express" and "Delft/Dejima" were designed in 1969 for 27 knot sea speed and many tanktests were carried out by owners in cooperation with Bremer Vulkan to obtain a soft-increasing power-speed curve.
  - A cilindrical bulb combined with a shoulderless forebody (section 11-13), (prof Inui, Tokyo, idea) was adopted and indeed the vessels-hull showed full cancellation of bow wave at 25-28 knot in a calm sea. Lowest ship values at top speed were obtained with this hull
    However, after the fuel crisis 1974, the service speed dropped to 20-21 knots (max 23) which condition does require a much smaller bulb.
  - Tanktesting in 1980 learned us that abt. 3% fuel saving could be obtained when replacing the original bulb by a smaller one and we decided to do so.
  - We expect that speed loss due to bulb action, when pitching in heavy weather, will also have become smaller (too many modelbasins design a bulb for full

draft, calm sea and top speed only). (These bulbs are a "brake" in service)

- The U-shaped forebody, which is favourable at the high speed length ratio (25-27 kn), should nowadays be redesigned into a more V-shape because the V forebody gives more form stability (KM value is considerably increased).
- V-shape should not require more horsepower at the lower speed length ratio's in comparison to the "U"-body vessel (at high speed the U-shape is better).
- Extra containerscan be accommodated keeping the original metacentric height, which means that the earning capacity of the V-body-ship is larger.
- A fine entrance angle at the loadline is strongly advisable in order to enable the ship to penetrate head seas in rough weather, easily.
- Sufficient freeboard and reserve Buoyancy-forward should be designed in the forebody to keep the foredeck free from green water. Smaller vessels should have a long foc'sle. Yard should not save steelweight on this item !
- Large amounts of WB in the ships-end are to be omitted in order to reduce the moment of inertia. Fuel oil tanks to have the centre of gravity 0,4 L from FPP.
- It has been proven to Nedlloyd, that a flared forebody is advantageous in headseas
- A knuckle sheerline in the higher part of the forebody will act as a wavebreaker and meanwhile creates a box-type hull girder of sufficient compressive strength, on both sides of hatch No. 1 and 2, which is important when diving into a wave. This knuckle is in our opinion preferable to a wavebreaker on deck mounted on a slim forebody.
- Last but not least, the knuckle sheerline is perhaps the last rudiment of ships beauty; a sheerless container vessel blocked up with boxes is an awful sight.

SINGLE SCREW HULL FORM

The development 1950-1965 on fast cargo liners from a narrow propeller aperture to a wide aperture is shown.
After that period the shaft horsepower was nearly doubled.
The preference of Nedlloyd is for the propeller on pipe and not for the stern bulb.
It ought to be mentioned that the 22-knot container vessels (1.600 teu, with block coefficients of about 0.60-0.64 and with 27.000 shp to 30.000 shp on one propeller) are all suffering from aft-end vibration to a greater or lesser degree. (Abel Tasman-Hollandia-Zeelandia) the design of such high-powered container vessels is very difficult and even more so in the case of ro-ro ships where the hull is more flat. The trim balance of the vessels will require the centre of buoyancy as far aft as possible, which means a full afterbody, whilst the free flow into the propeller requires at the same time an afterbody as slender as possible.
As always in naval architecture, a compromise had to be reached.
The afterbody should be slender and meanwhile fuel or waterballast should be ahead of ½ L.
Some vessels having a blunt afterbody immediately ahead of the propeller and a rather sharp turn to the engine-room bilge suffer from separation of the waterflow towards the propeller. Other vessels have been designed with a (too) small clearance in the upper part of the stern frame, ahead of and just above the propeller. This causes a wake peak of 65 to 75 per cent in the "12-0'clock" position.
In case the flowlines towards the propeller have to make a sharp bent, separation occurs and a stagnated flow above the propeller can activate bursts of hull vortex cavitation.
On certain vessels a tunnelplate has been mounted above and ahead of the propeller which directs the inflow of water more accurately, shielding off the separation.
Results were often very good (on "Sea Land Economy" - "Melbourne Express") enabling the use of more horsepower at reduced vibration levels. There is also a case where a misdesigned leading edge of some kind of tunnel in the shell at an angle to the flowlines is acting as a water-stumbling barrage ("Encounter Bay" class).

From experience, it has been found that ships with a propeller blade frequency of 5x130-136 rev/min and 6x106-110 rev/min suffer from full resonance of bulkhead panel vibrations in the afterbody. The worst vessels for vibration have been found to be those with the accommodation far-aft and with propeller tip speeds in the higher ranges (e.g. 130 rev/min in comparison with 110 rev/min). (over 40 m/sec)
The higher harmonics: 3: and 4: x blade frequency are causing most troubles regarding noise and vibration.
The result is vibration inconvenience to the crew and even damage to the structure in the after peak area and damage to piping, bearings and instruments. These troubles have frequently led to the practice of sailing in service at reduced power. Vibrations in the first-generation ships are acceptable below 27.000 shp which resulted in a reduced service speed of 21 knots on the Europe-Australia run, a reduction of one knot of the capability.
For the naval architect it is a relieve now to have the possibility to apply lower propeller revolutions : 90-100 RPM instead of 122 RPM, which allows for lower tipspeeds.
The flow of water should be kept "undisturbed" as far as feasible for (both twin- and) single screw vessels. If a wake peak of 55 per cent is to be accepted or is unavoidable, thus creating cavitation and hull pressure fluctuations a shock absorber above the propeller,mounted on the flat afterbody, might be tried. (Tests at TPD-Delft are proceeding).

IMPROVEMENTS 1971-1981

In 1971 we experienced the vibration free propulsion with the twin screw vessels, fitted with propellers on shaftings covered by slender pipes and supported by streamlined brackets.

- In accordance with this good experience we designed the same arrangement for the South Africa service:
  Safcon . one fitted with twin screws on pipes
  . and one fitted with one screw on one pipe

- The wake belt on both vessels was very low (abt 25-30%) and troubles by vibrations should not be excepted

- The single screw vessel needed more horsepower for the same speed which was caused by the propeller of limited diameter nad a larger blade area, and so a lower efficiency.
- The single screw vessel was missing the wake and therefore also the hull efficiency was low .

"(No wake - no vibrations - no hull efficiency" consequently!).
It must be mentioned that all conclusions were derived without taking into account the scale effect (from model to ship) on wake and thrust deduction. (Reality might give a slightly different conclusion).
This disappointing result on propulsion performance for the vibration free single screw pram with propeller on pipe lead us to the latest developments in single screw hull form: figures:
We try to accommodate the diesel engine which should be as far aft as possible in a stre lined bossing ("Gondola") underneath the pram-hull.

- The bossing to be as slender as possible in order to reduce the wake peak

Therefore the web frames on the tank top sho uld enclose the aft end of the diesel engine including the thrustblock. The engine is hanging in a sling of shell plating. Fig
Marine engineer should help the naval architects to realise this engine seating to be stiff meanwhile allowing for filling pieces after lining up.

- Our latest single screw design 1981 is shown

The speed power performance is very good according to tests at Wageningen. The 12-0'clock wake peak is less than 50%. Counter rotating vortexes are relatively small

RO-RO DESIGN AND FULL SCALE

Nedlloyd "Rotterdam-Rochester" hull form (ro-ro) 1976-'77 is a compromise between following requirements:

- horsepower to be minimum
- centre of buoyancy as far aft as possible to counteract stern trim and hogging moment
- engine as far aft as possible to obtain max. cargo-space and still sufficient stiffnes of engine foundation
- flow lines to propeller as smooth as possible
- minimum wake peak at 12-0'clock position
- minimum counter rotating vortexes
- minimum vibrations
- maximum stability : wide stern
- maximum acces at stern ramp : wide stern

The resulting compromise was quite reasonable, however, the higher propeller excited frequencies are dominant in the top of the superstructure.

- Full scale vibration and noise measurements were carried out and also propeller observation was made in order to analyse phenomena which are present on almost every single screw vessel of fine lines and moderate power.

Full scale tests conclusions by "Techn. Physical Dienst" (Ir. A. de Bruyn)

1. The acoustic source strength of the propeller cavitation of m.v. "Nedlloyd Rochester" is acceptable in view of data from other propellers
2. The 4th harmonic of the blade-rate frequency (ca 32 Hz) (4x4x120 RPM) is dominant in all kind of spectra viz hull pressure, hull acceleration, volume velocity and accelerations in the superstructure
3. The representation of the acoustic source strength by an equivalent monopole leads to reasonable agreement regarding hull pressure and velocities amplitudes.

- The ship is acceptable in view of vibration and noise level, however, we think to improve on new designs:

- Propeller, which was quite well designed by Mr. Voogd of Lips, and free from cavitation erosion, should have more skew-back in future ships in order to reduce the excitation of the higher frequencies.
- The higher frequencies could be reduced by about 10 DB by application of a "rubber-air-cushion" between water and the steel structure above the propeller. (Thickness: 0,5 m, extension of surface 15x15 m) For newbuildings it might be possible to mount the elastic layer into a recess in the hull in order to keep the shell surface flush.

- The influence of the fulness of sections 3 and 4 on the flowlines into the propeller plane at 0,8 R is enormous.
  - It seems rather impossible to make lines finer ahead of the propeller than Rotterdam-Rochester and keeping the engine at the same place
  - The centre of gravity of light ship is 0,38 L forward of APP and fuel tanks respectively waterballast tanks are to be positione in the forebody.

TWIN SCREW HULL FORM

The attractive feature in the lines of twin screw ships (block coefficient : 0,60-0,68) is the "straight-forward" character of the design.
Free-flow propellers mounted on slender shafting having a high entrance velocity will provide a high-propeller efficiency and because of the low wake, there will be less vibration in comparison to single screw ships.
A "fishtail" deadwood, with skeg-rudder designed to give good course-keeping (even when sailing in following seas) also gives shallow water-and slow-speed directional stability.
A wide pram-shaped afterbody having a soft run gives maximum stability and cargo space-aft, which is much needed in container and ro-ro ships. A flat afterbody can excite slamming, but only when the vessel is lying still in a sea swell. The immersion of the transom should not exceed 0,6 m in order to limit extra fuel consumption by "Borrelzog".(Froth-wak
Contrary to the vibration troubles experienced on the single screw Australia run vessels of the first generation, it is noteworthy that the twin screw far east container vessels, which used up to 2 x 40.000 shp for attaining a 27-knot sea speed, are quite free from vibration. This is greatly attributable to the slender pipe or bossing around the wingshafts, resulting in an axial wake of not more than 25 to 30 percent.
Shafts should be covered by pipes in order to prevent corrosion.

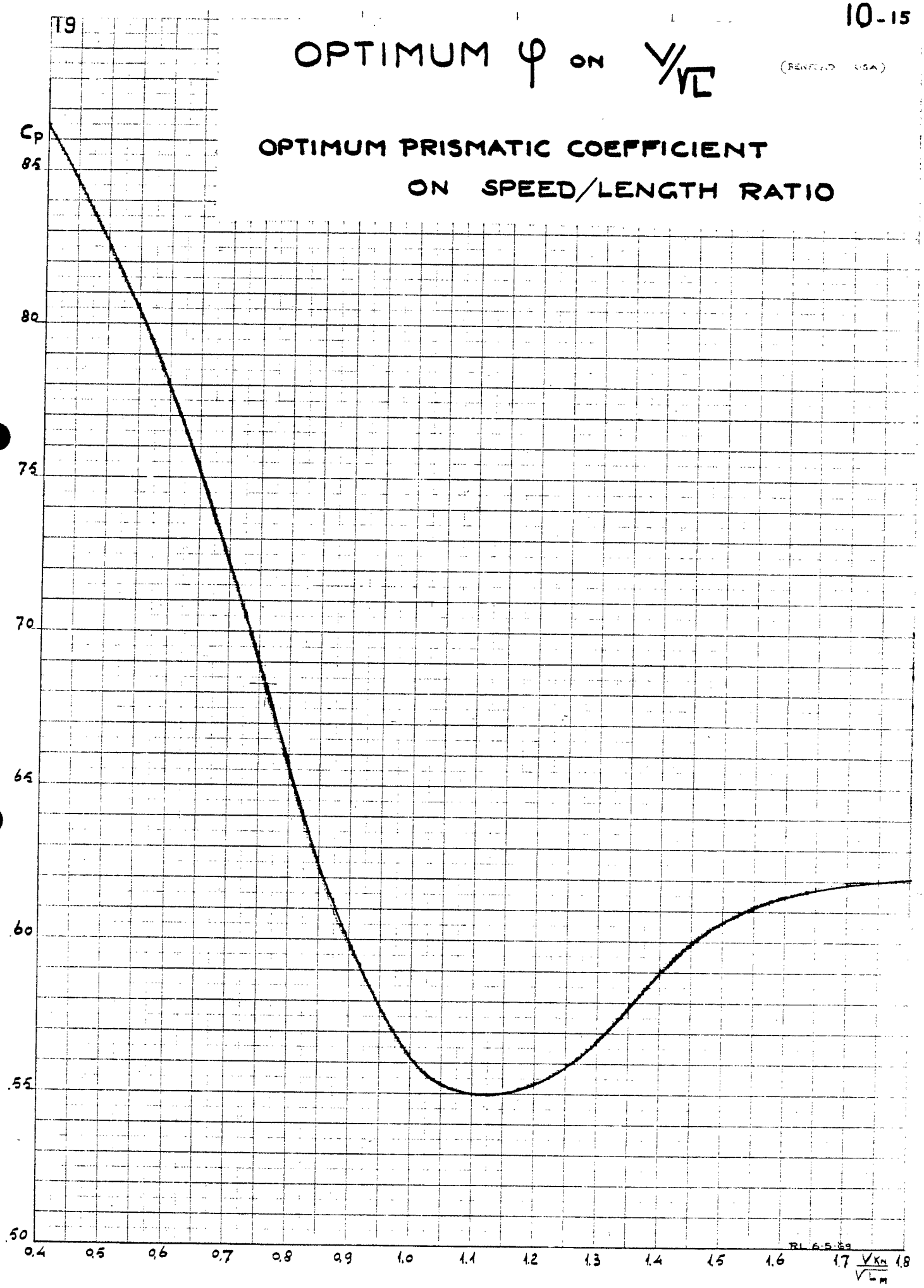
Aircraft-design brackets supporting the slender pipes seem to have no harmful effect to the wake-field and do not create propeller-excited vibration (it should be noted that bossings designed to create contro-rotating flow in order to the increase propulsive efficiency, should be avoided because the larger wake peak will excite cavitation and vibration). Sufficient tip clearance between the propeller and the ship's hull, to keep hull pressure fluctuations at a low value is also a must for a good design. This requirement will tend to arrange the propellers far out and well aft. Unfortunately, this will reduce the flow of water along the single rudder, resulting in a poor steering capability.

Therefore, from a manoeuvring point of view, the propellers should preferably be close to the skeg and well forward. This, of course, will result in strong hull pressure fluctuations. Once again a compromise has to be found. Experience up till now, obtained with Nedlloyd's "Dejima/Delft" and also "Houtman/Hoorn" shows their stern arrangement to be a good compromise.

T9

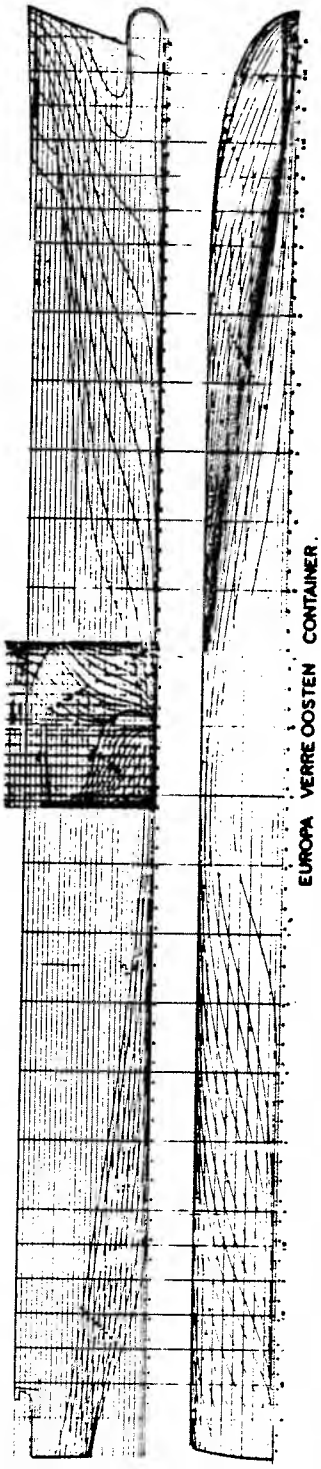
OPTIMUM φ ON $V/\sqrt{L}$

(RENFRED USA)



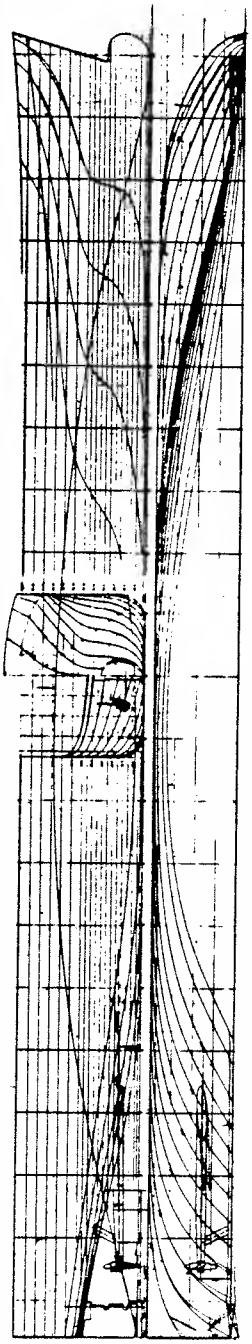
TWIN-SCREW HULL

A



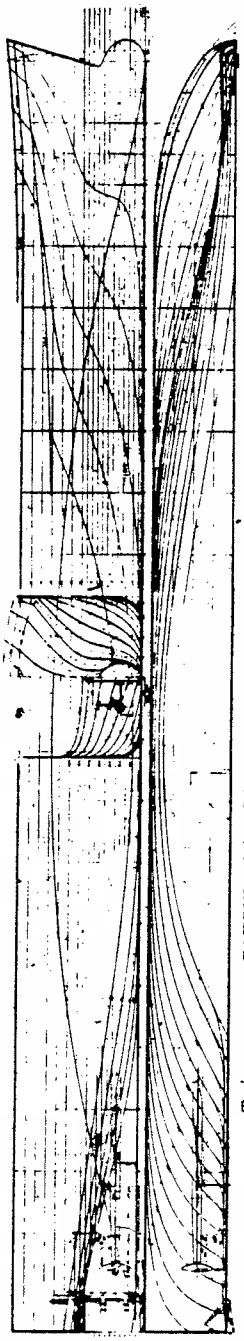
EUROPA VERRE OOSTEN CONTAINER.
MODEL 3956
"NEDLLOYD DEJIMA" "NEDLLOYD DELFT"

B



Twin-screw SAFCON ship lines (development stage) $L_{b.p.}$=247 M.

C



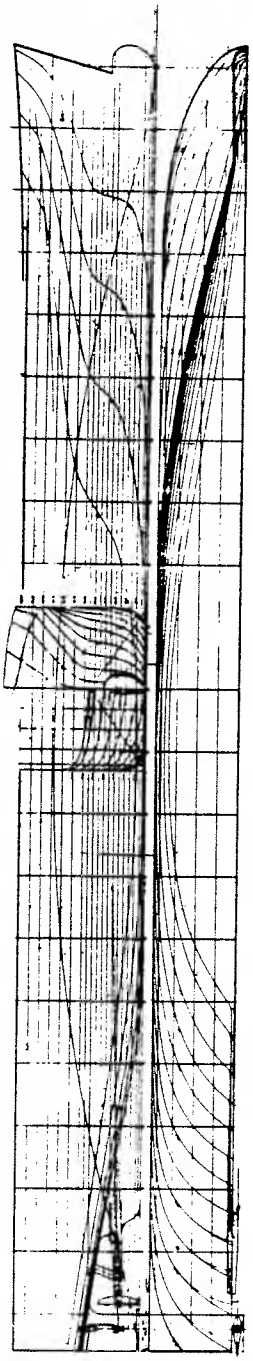
Twin-screw SAFCON ship lines(as built) $L_{b.p.}$247 M. MODEL 47510

"NEDLLOYD HOORN" EUR.-Z.AFRIKA

"NEDLLOYD HOUTMAN" EUR.-NW.ZEELAND

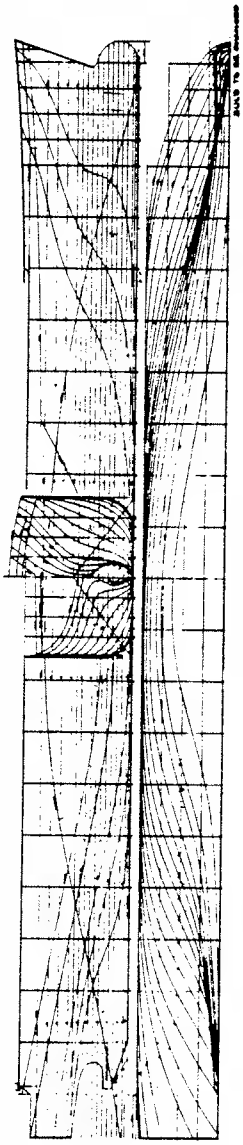
SINGLE SCREW HULL

A



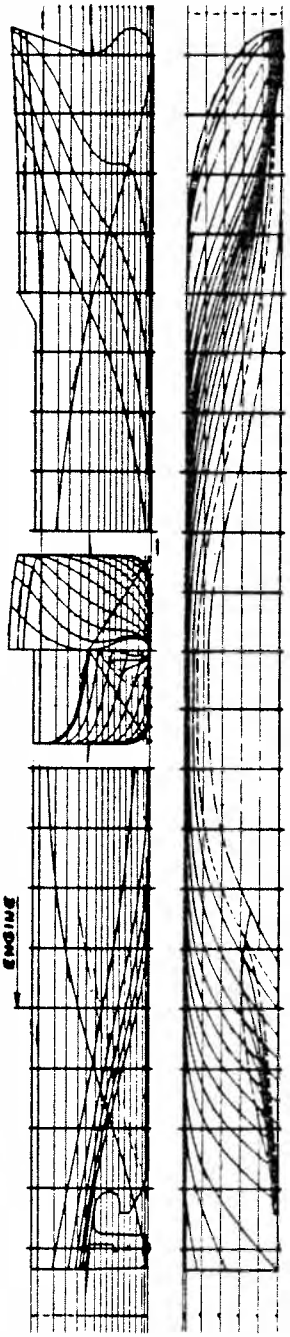
Single-screw SAFCON project in development stage $L_{b.p.}$=247 M. propeller on pipe

B

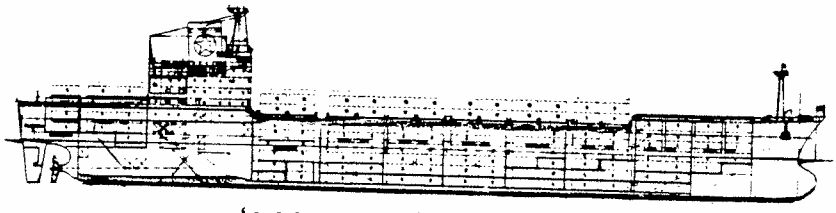


single screw ZUIDAFRIKA - VERRE OOSTEN CONTAINER PROJECT 1975 $L_{b.p.}$=207 M.

C

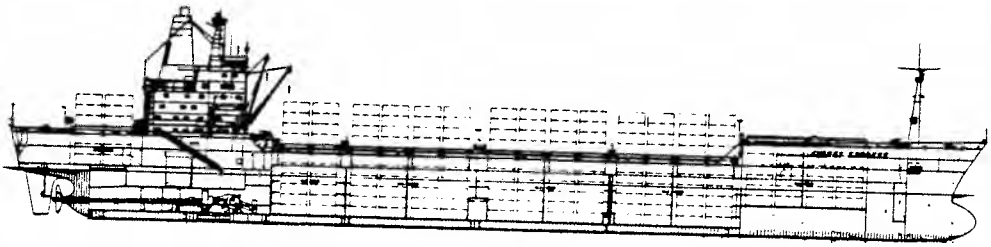


PLAN WITH GONDOLA SINGLE SCREW UNIVERSAL CONT. 1971 $L_{b.p.}$ = 202 M.

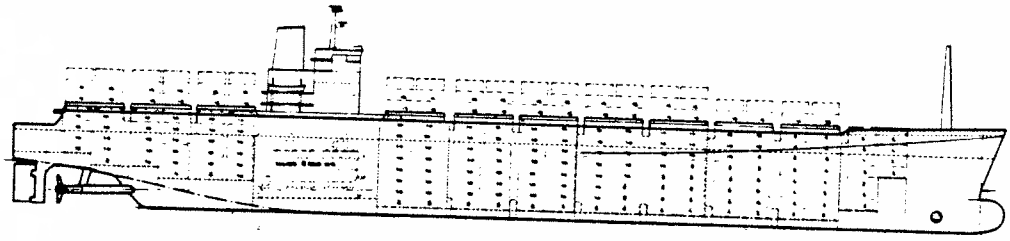


'California Star' Bremen. 28000.

'Encounter Bay' h.d.w., 32000 s.h.p./136.



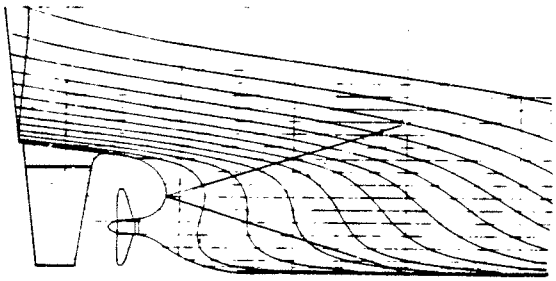
'Sydney'/'Abel Tasman'. 32450 s.h.p./110.

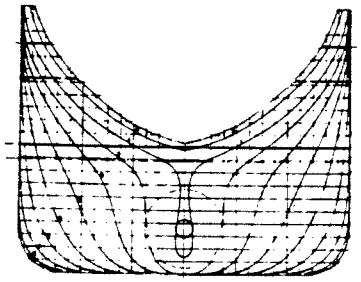


Project South Africa. 36000 s.h.p./121.

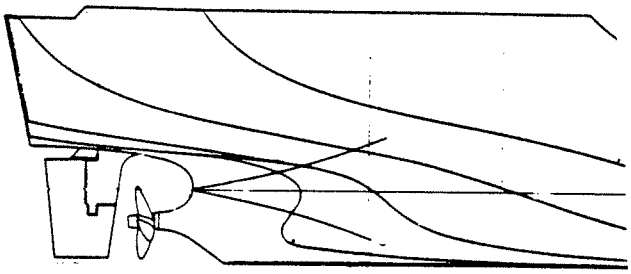
'Bremen Express'/NedLloyd 'Dejima'. 2 x 40500/136. Development of container ships trend towards:
- Free flow propeller.
- Superstructure away from stern.

California Star

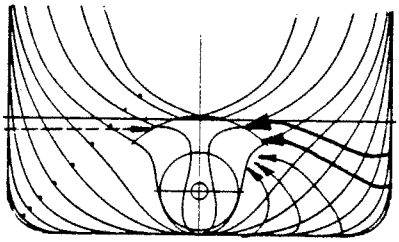




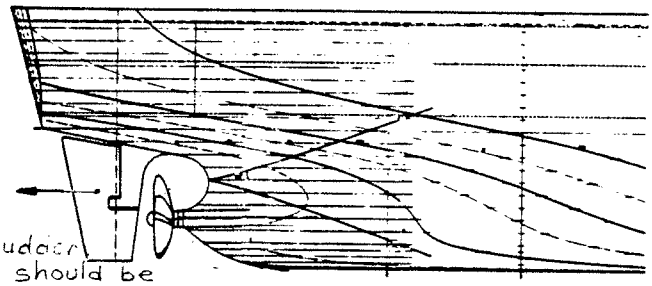
Encounter Bay



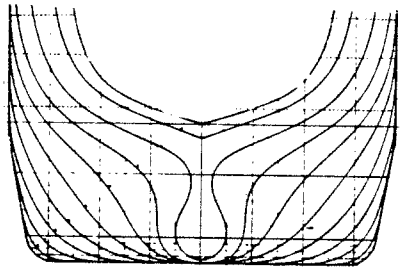
tunnel causes flow-separation



Abel Tasman

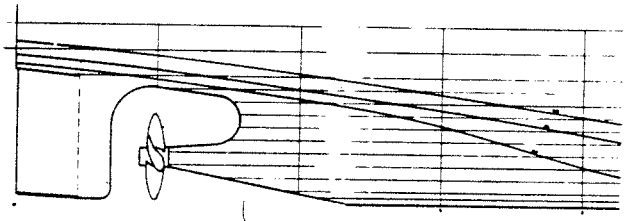


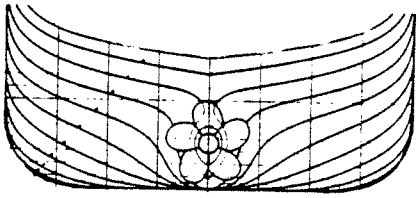
rudder should be more aft positioned



sloped?

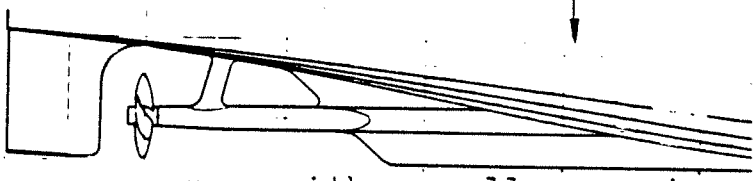
Project'73

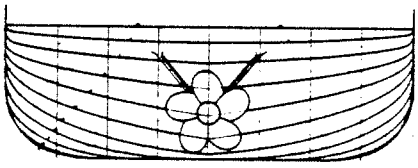




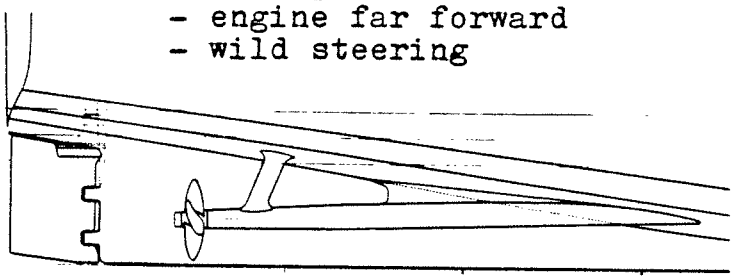
better than

Project'73



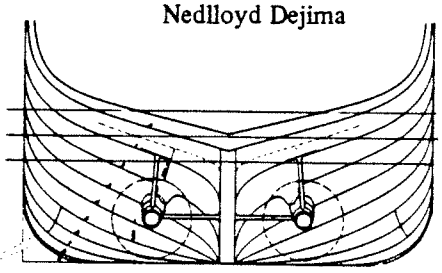


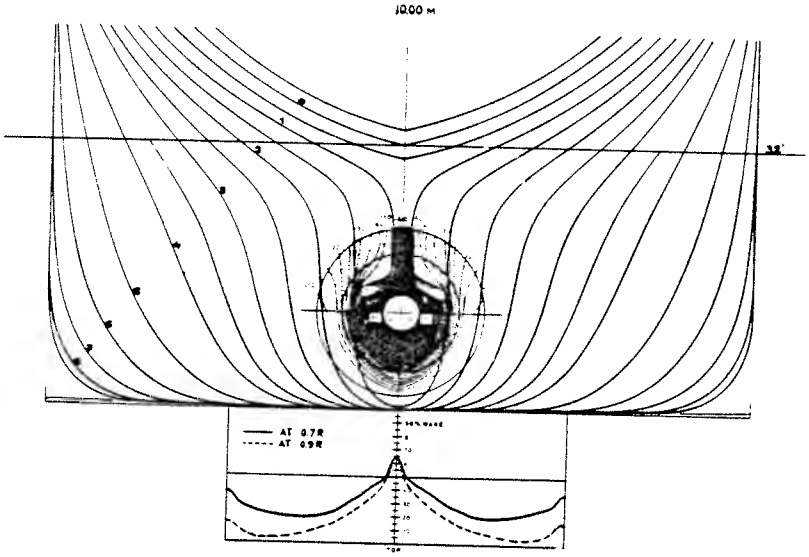
pram with propeller on pipe
- not optimal in HP
- engine far forward
- wild steering



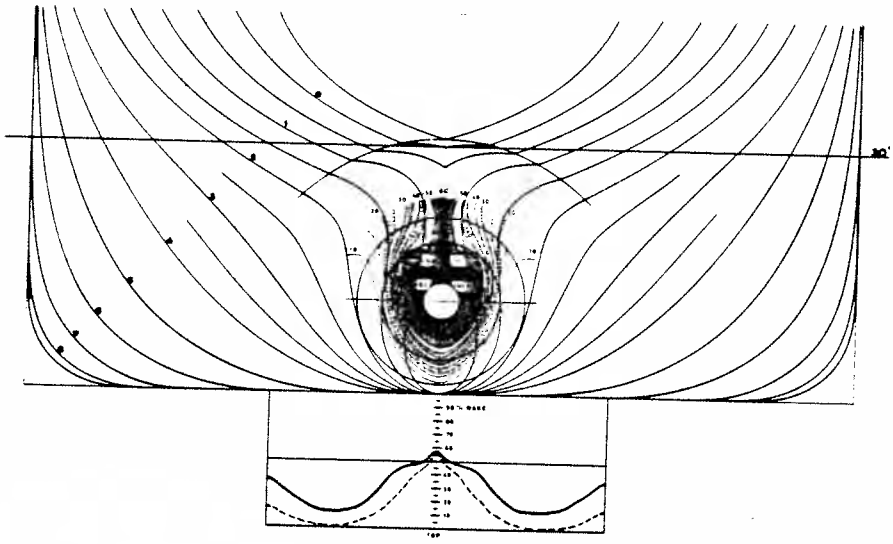
good

Nedlloyd Dejima



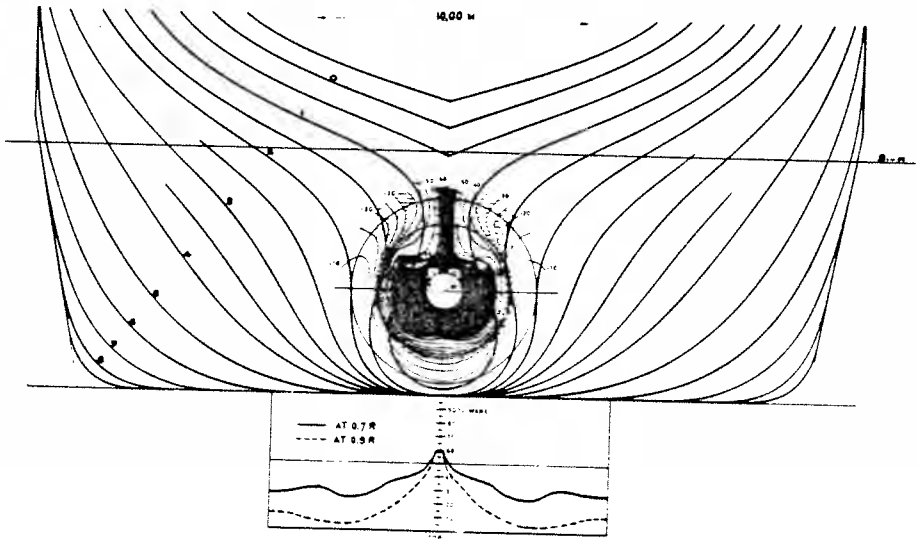


California Star



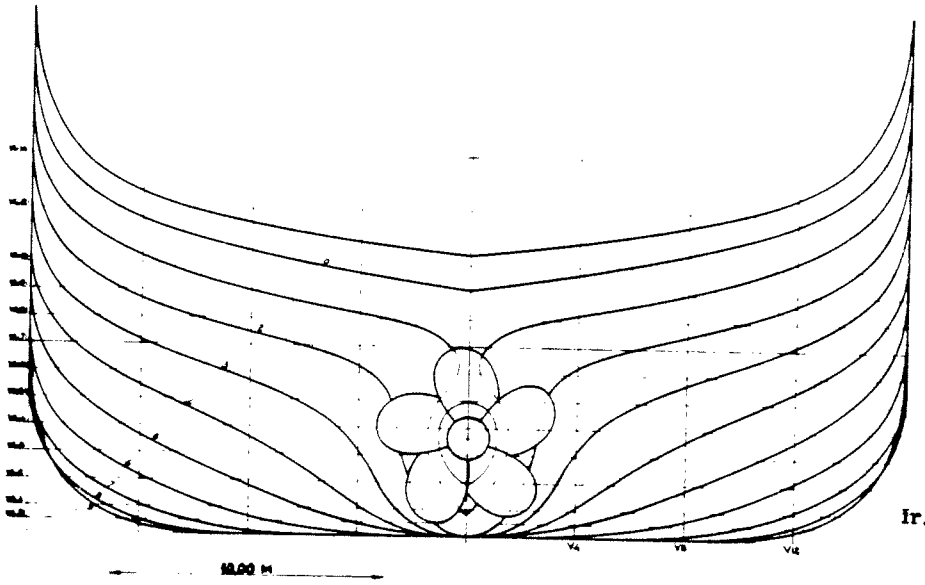
Encounter Bay

Clearance between propeller and tunnel too large.



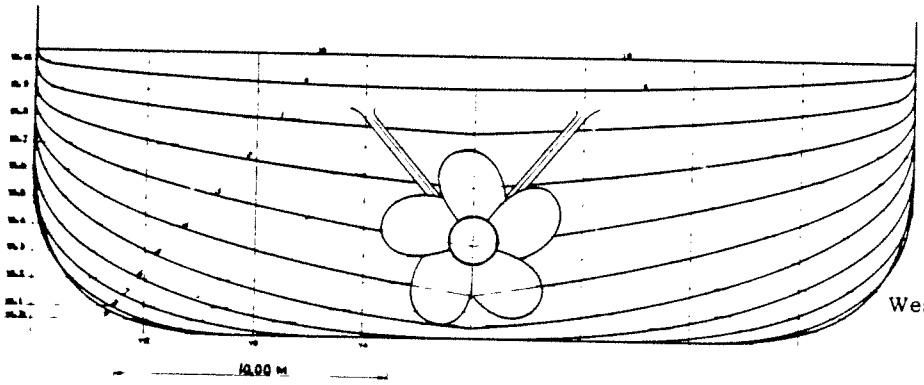
Abel Tasman

A.G. Weser stern bulb not effective in top quadrant.



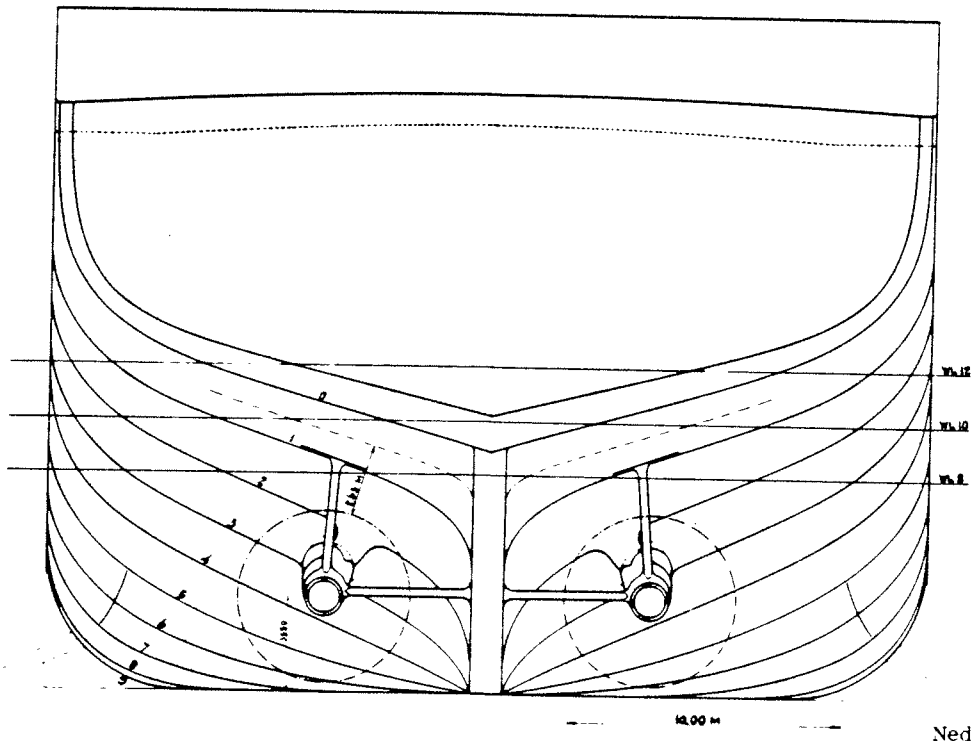
Ir. E. van Dieren

Wake field tending to pattern of a twin screw bossing configuration.



Westlander Project

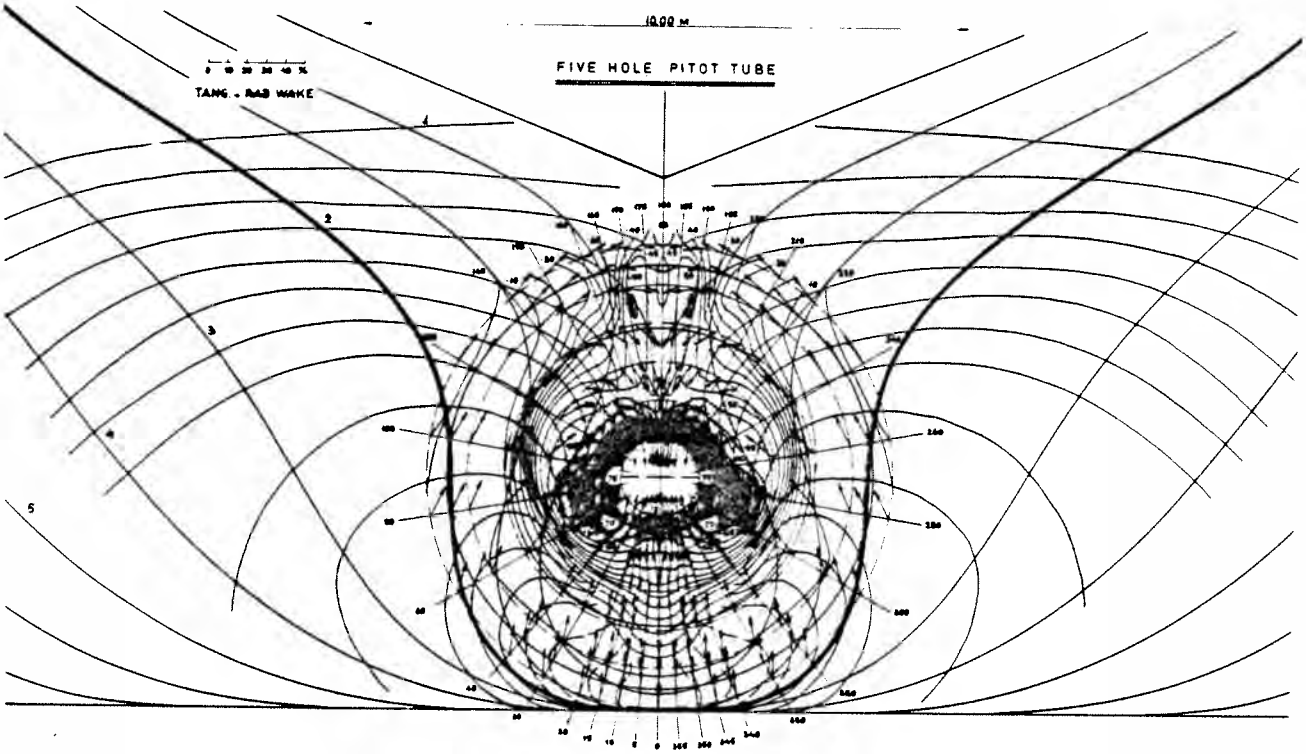
Wake field tending to pattern of a twin screw bracket configuration.



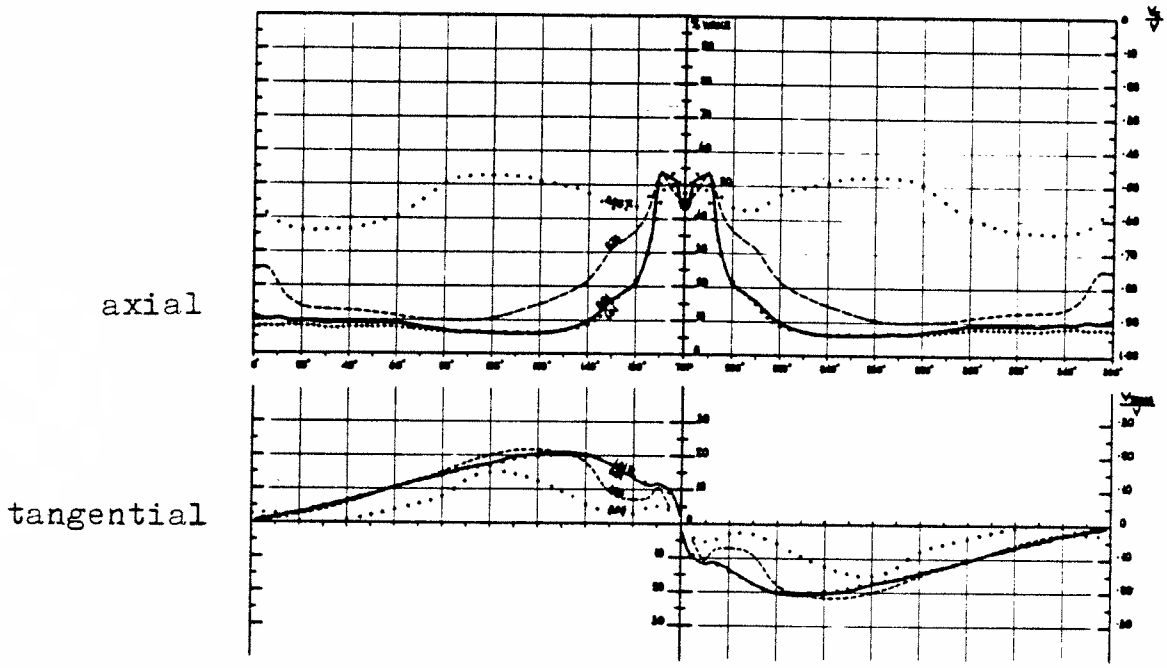
NedLloyd Dejima

Far East container vessel 2 x 40000 s.h.p./136 r.p.m. Bracket configuration.

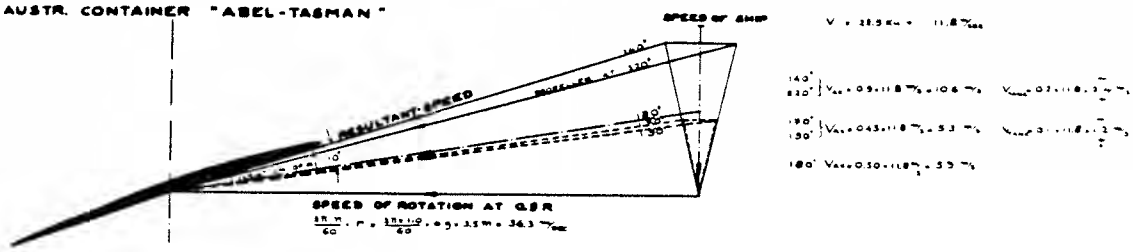
WAKE ABEL TASMAN



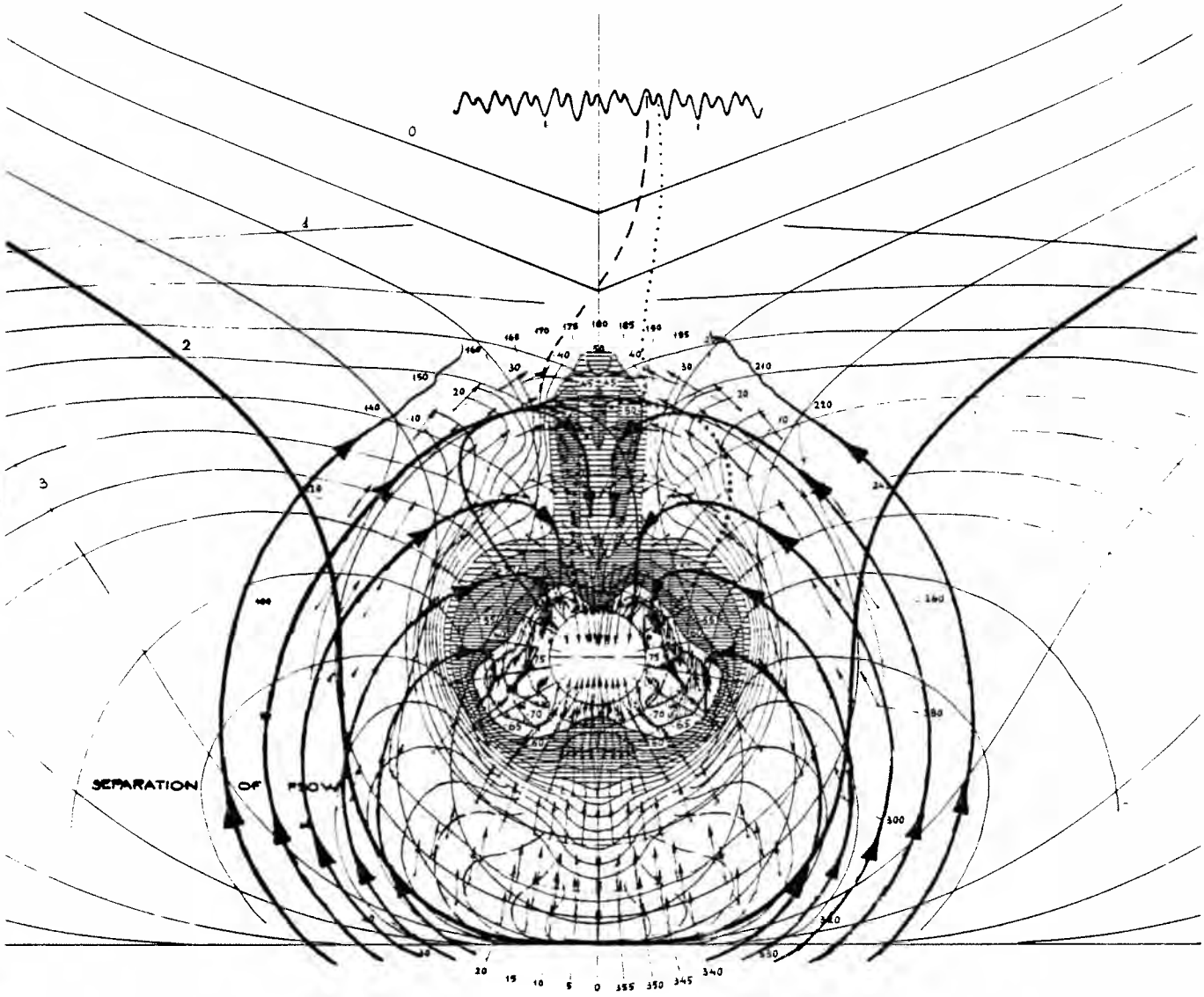
s.s. Abel Tasman containership 32450 s.h.p. x 110 revs $\delta = 0.62$.



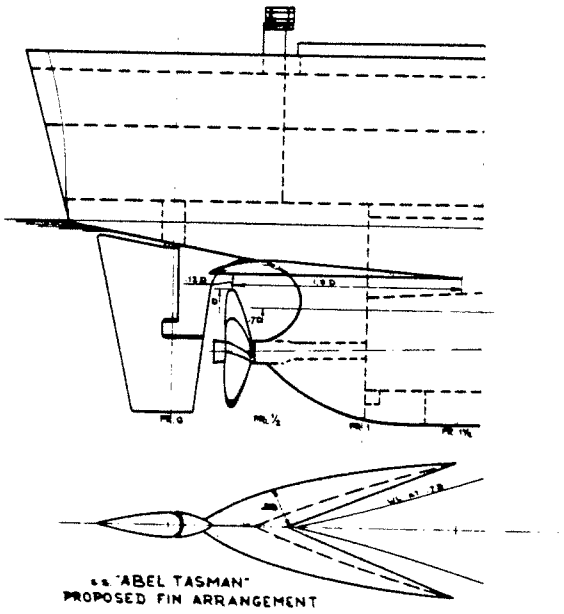
Results of 3-dimensional wake measurements on the Abel Tasman model

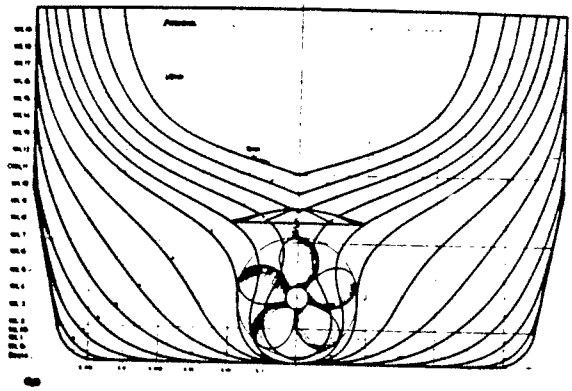


Variations of resultant speed. Strong variations of angle of attack.



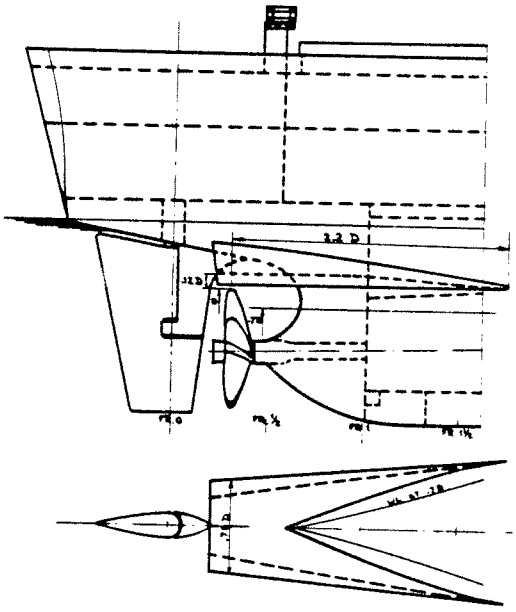
Two contra rotating vortexes on ss 'Abel Tasman'
Fin only can reduce wake peak in top position.

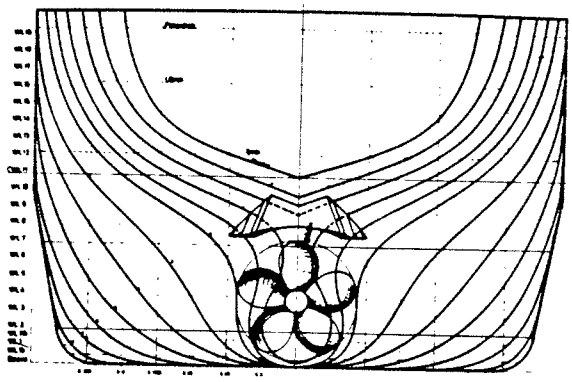




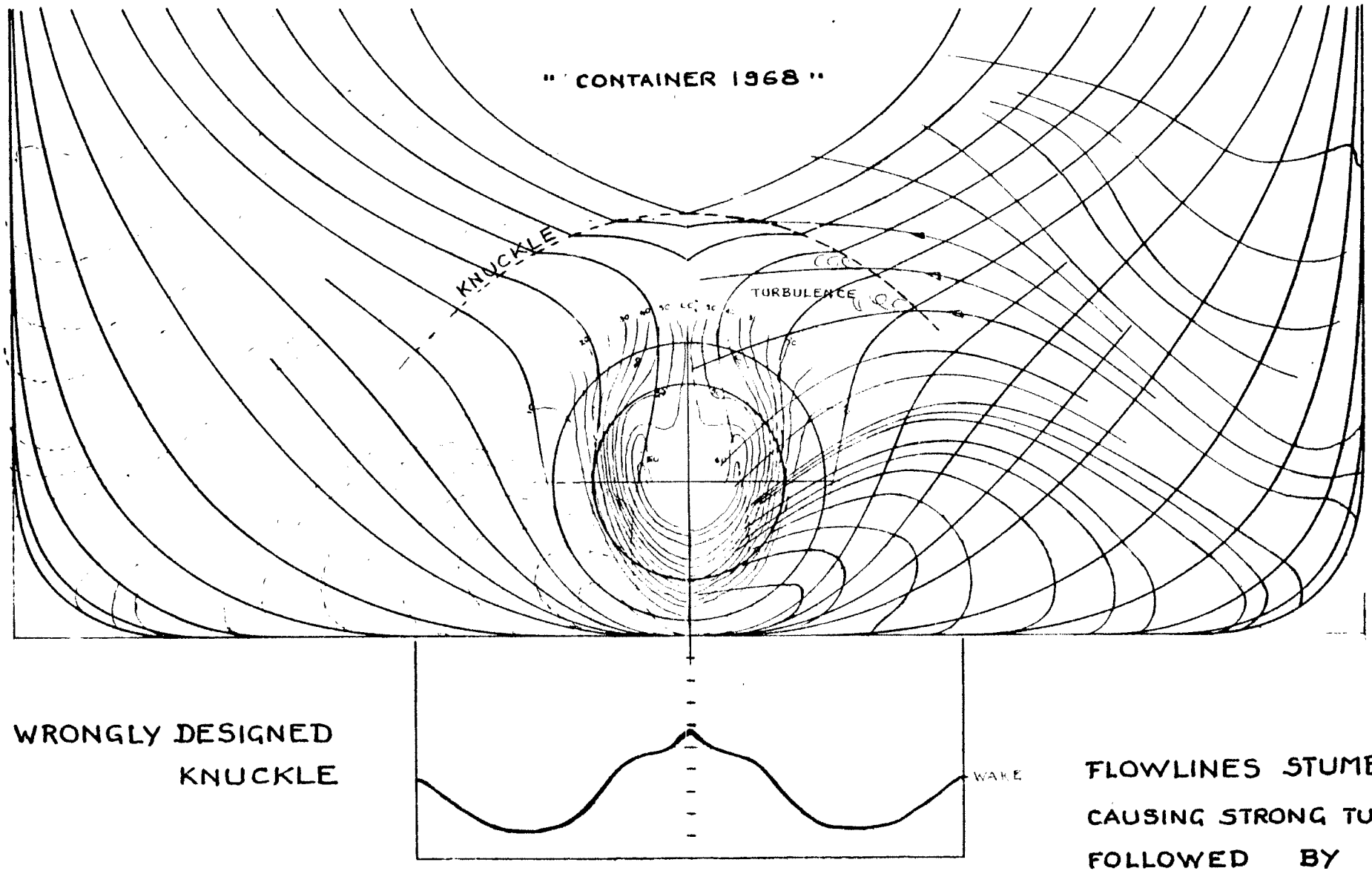
Installed 32450 s.h.p./110 revs. Normally in service, without fin or tunnel: 26600 s.h.p. = 82% can be used without suffering from heavy fantail vibration.

. s.s. 'Abel Tasman'. Fin-arrangement proposed by S.M.T., Trondheim.



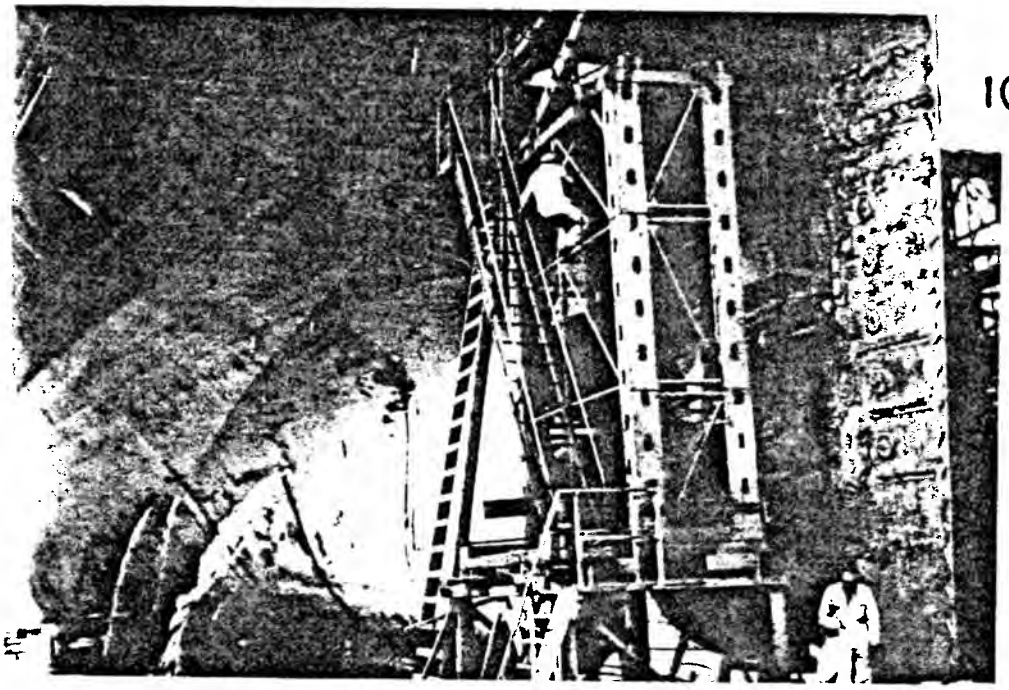


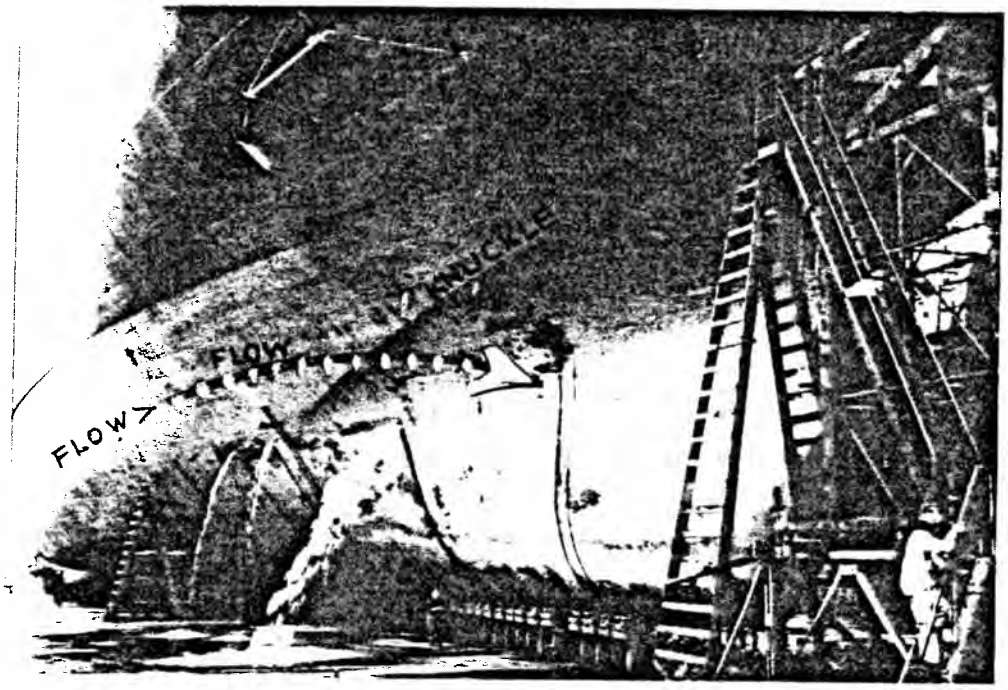
Proposed tunnel arrangement (it is important to extend leading edge far ahead).



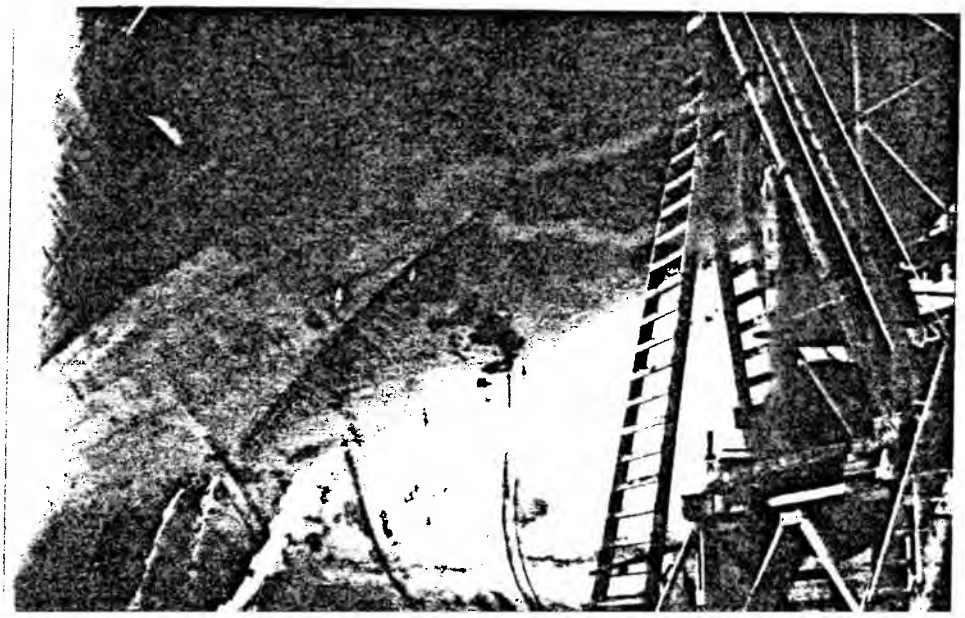
WRONGLY DESIGNED KNUCKLE

FLOWLINES STUMBLING CAUSING STRONG TURBULENCE FOLLOWED BY HEAVY CAVITATION EROSION IN PROPELLER AND RUDDER

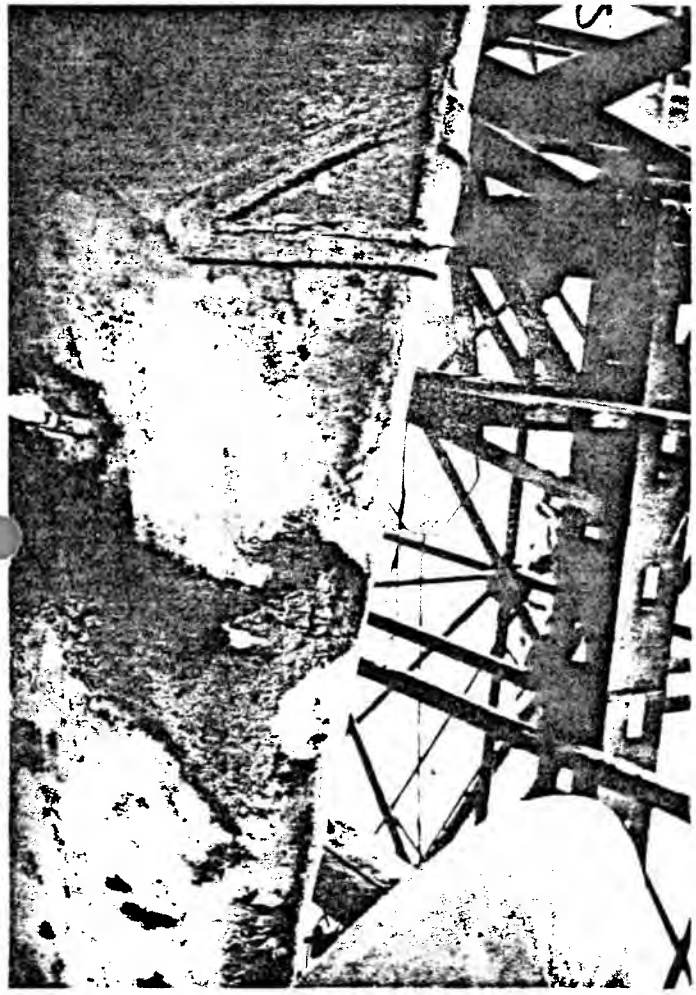




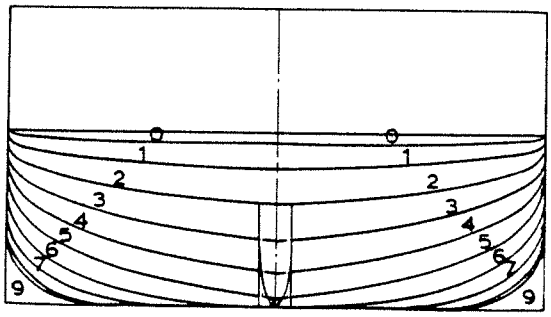
FLOW CROSSES "TUNNEL-EDGE"



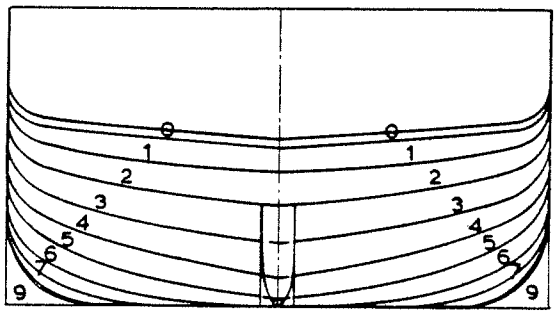
WITHOUT RESEARCH ON FLOWLINES
BY PAINT TEST AND TUFTTEST,
IT IS VERY RISKY TO DESIGN KNUCKLES



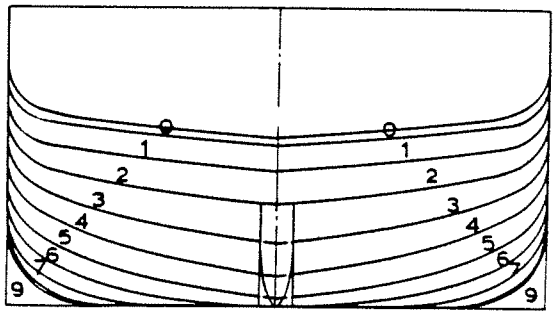
Heavy cavitation erosion on rudderpost and propeller due to strong turbulence in water afflux at that level .



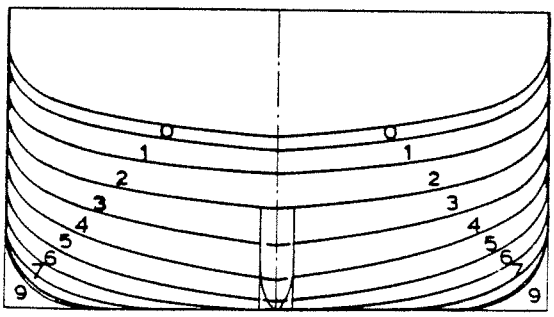
MODEL 4632
Kist vorm ('Box' shape)



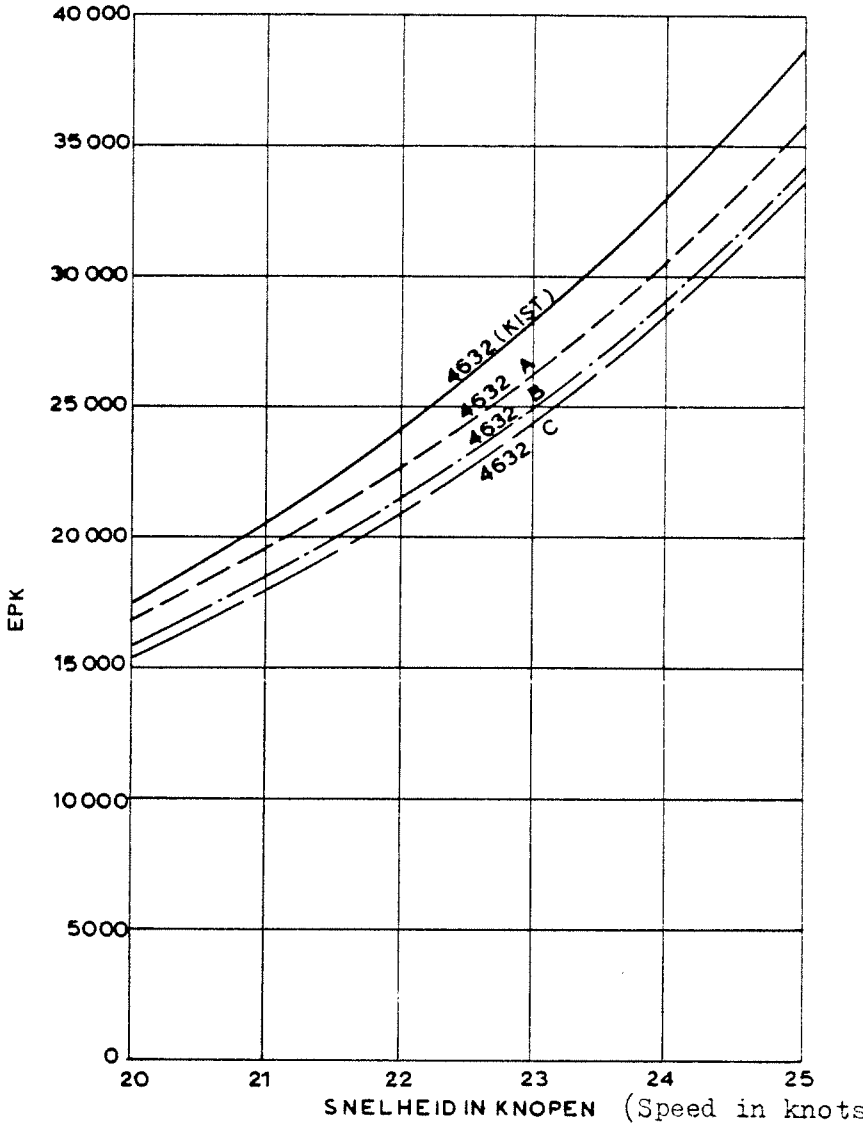
MODEL 4632 A



MODEL 4632 B



MODEL 4632 C



SCHEEPSWEERSTAND ALS FUNKTIE VAN SNELHEID

(Ship's resistance as a function of speed)

EUROPA-ZUID AFRIKA CONTAINERSCHIP

ONDERZOEK VORM VAN E.S. ACHTERSCHIP

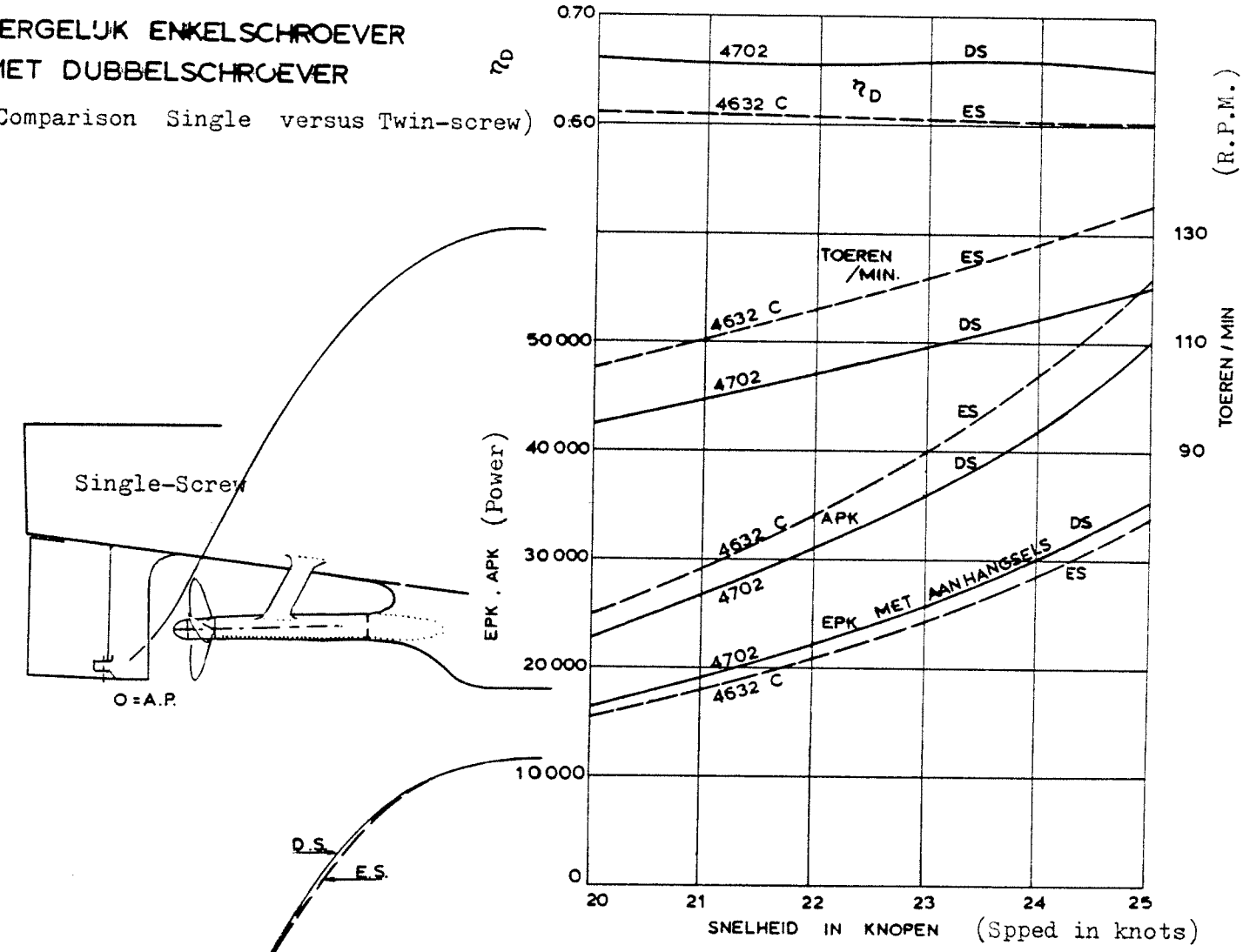
(Research on form of single-screw afterbody)

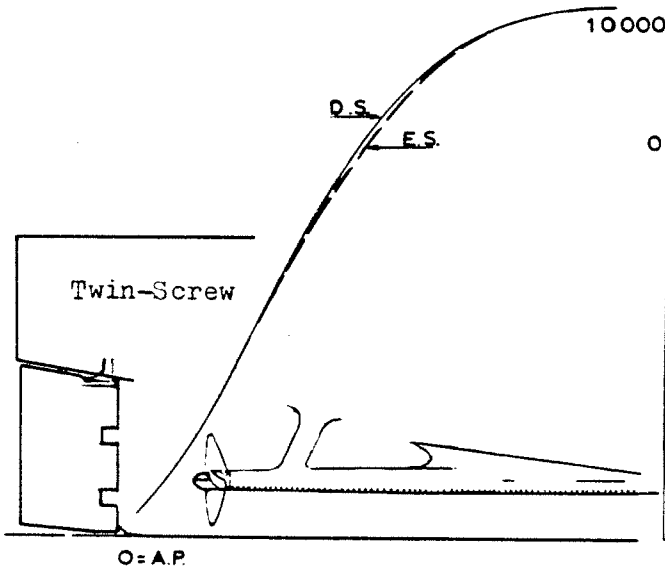
VAN KIST TOT SCHIP

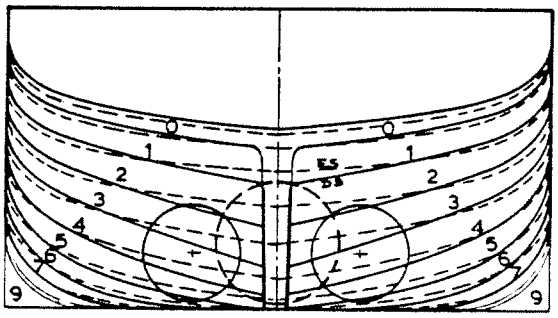
(From 'Box' to 'Ship')

VERGELIJK ENKELSCHROEVER
MET DUBBELSCHROEVER

(Comparison Single versus Twin-screw)







DUBBELSCHROEFSCHIP MODEL 4702

ENKELSCHROEFSCHIP MODEL 4632 C

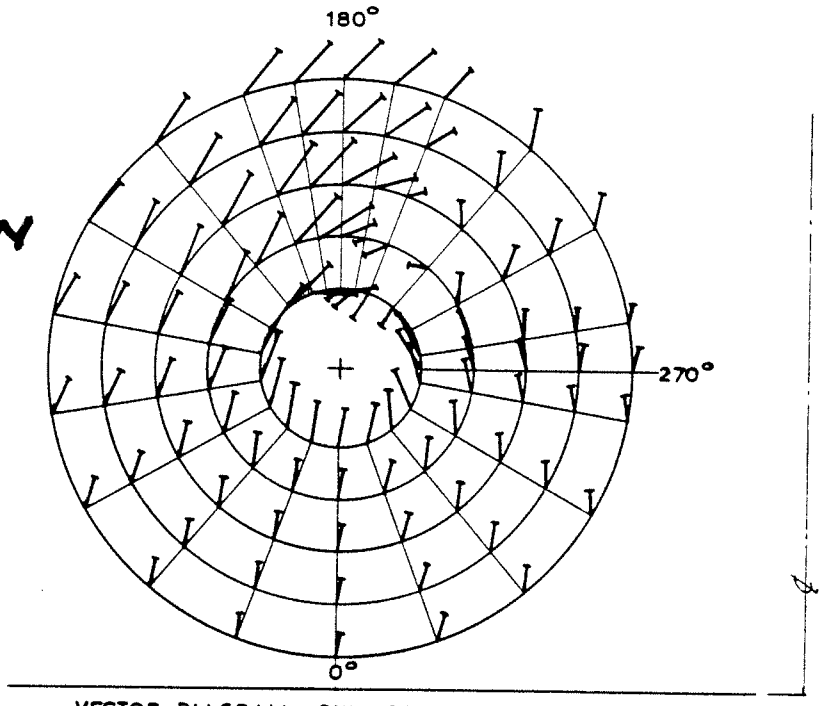
EUROPA-ZUID AFRIKA CONTAINERSCHIP

SAFCON PROJECT

SAFCON PROJECT

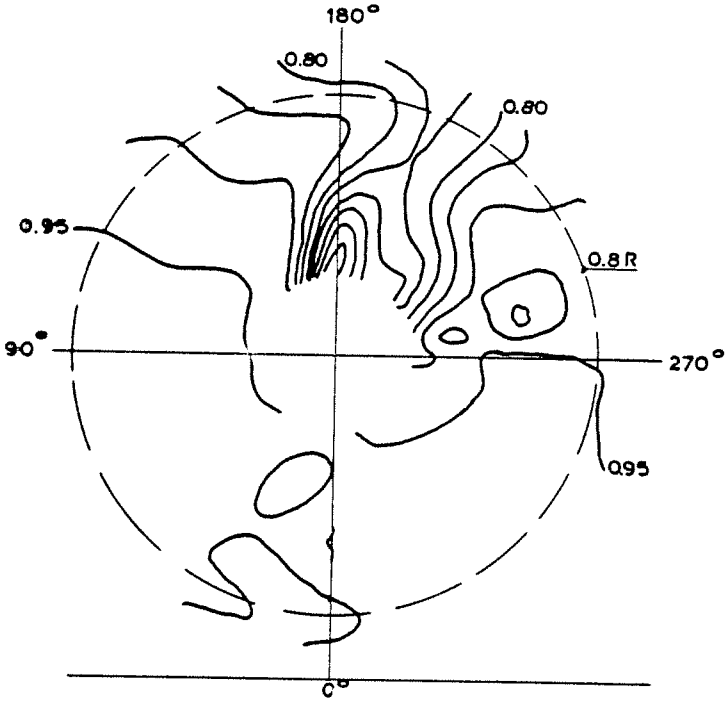
D.S.

TWIN SCREW



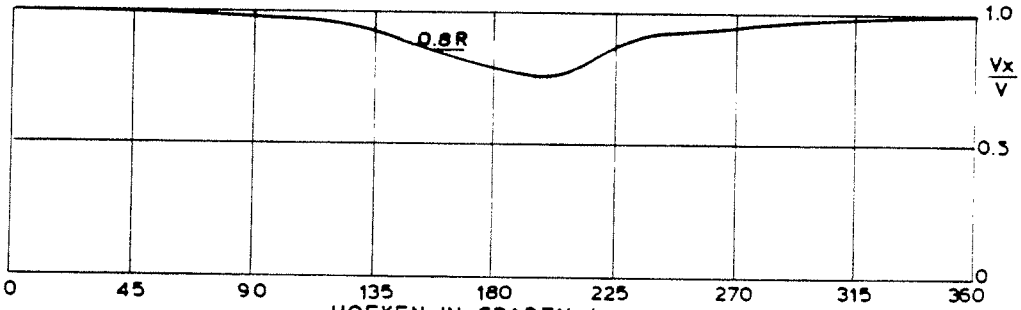
VECTOR DIAGRAM DWARSCOMPONENTEN

(Transverse components)



LIJNEN VAN CONSTANTE SNELHEID

(Curves of Constant Speed)



HOEKEN IN GRADEN (Angles in degrees)

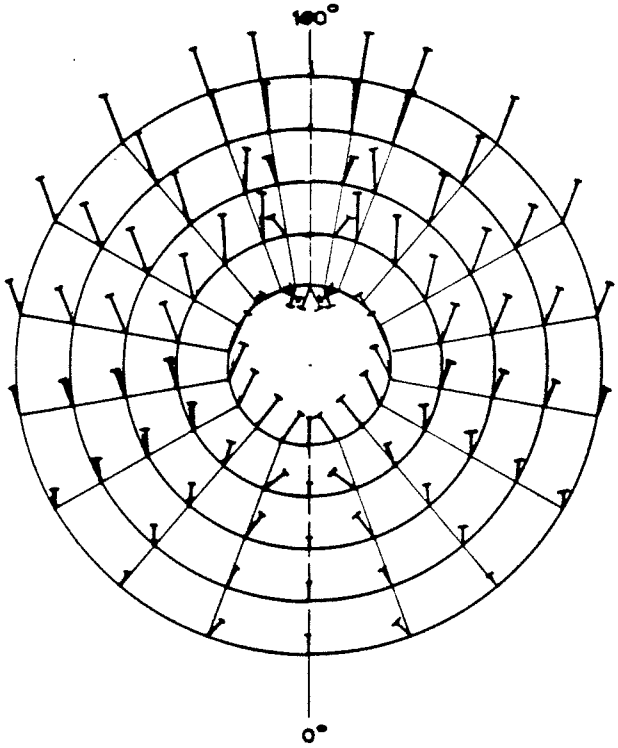
VOLGSTROOM KARAKTERISTIEK D.S. MODEL No. 4751 O

(Wake characteristics twin-screw model 4751-0)

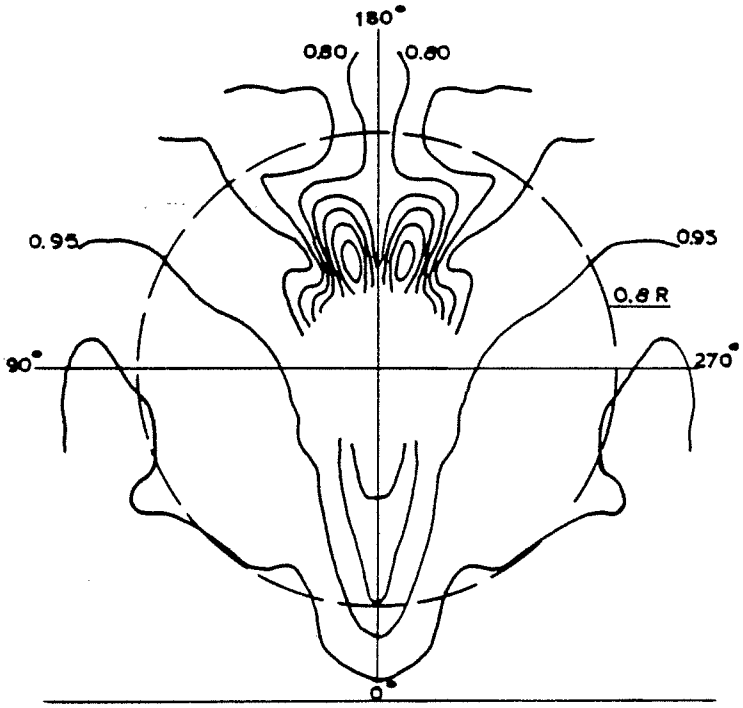
SAFCON PROJECT

E.S.

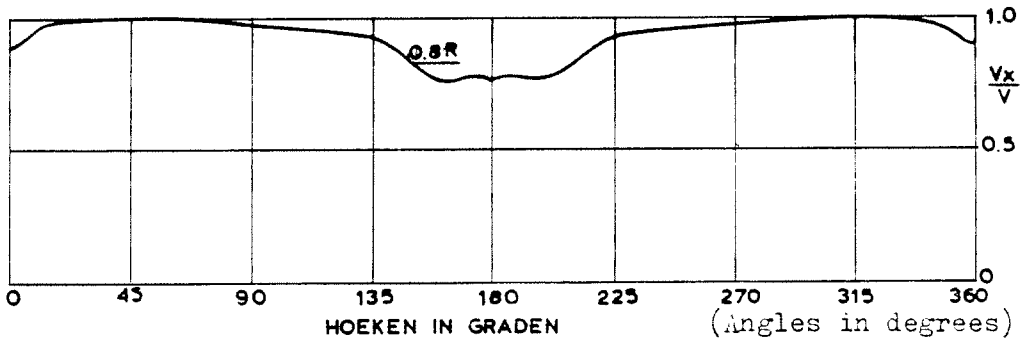
SINGLE SCREW



VECTOR DIAGRAM DWARSCOMPONENTEN (transverse components)



LIJNEN VAN CONSTANTE SNELHEID (Curves of constant velocity)

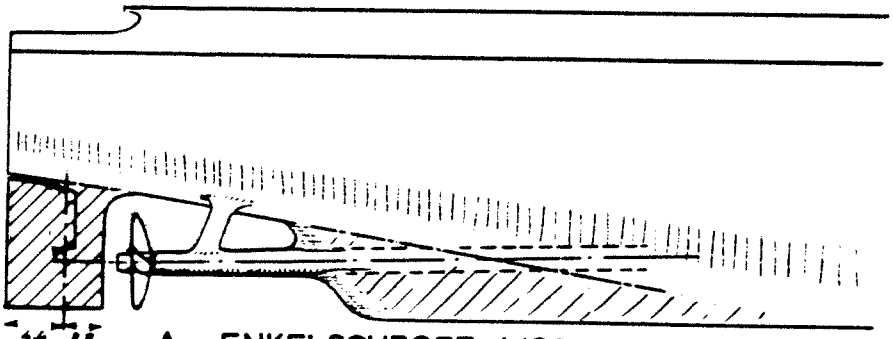


(Angles in degrees)

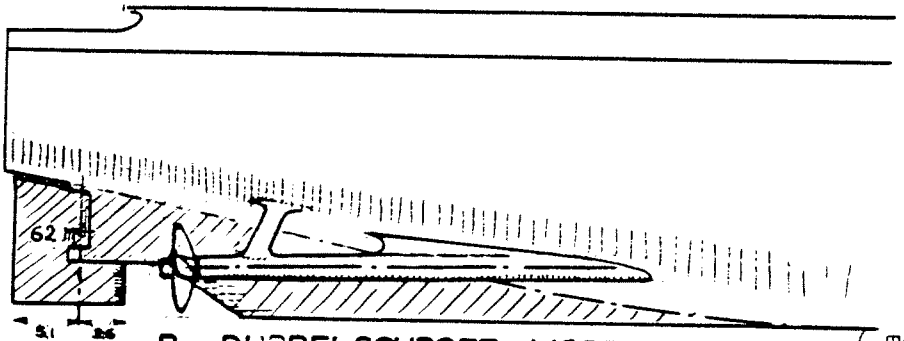
VOLGSTROOM KARAKTERISTIEK E.S. MODEL No. 4736 A

(Wake characterisitics Single-Screw, model 4736-A)

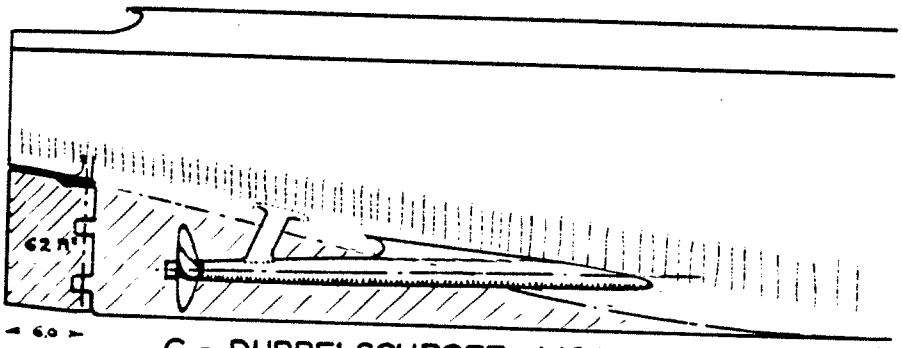
SÀFCON PROJECT



A = ENKELSCHROEF-MODEL No. 4736A MET SEMI-BALANS ROER. (Single-Screw with Semi-balanced rudder)



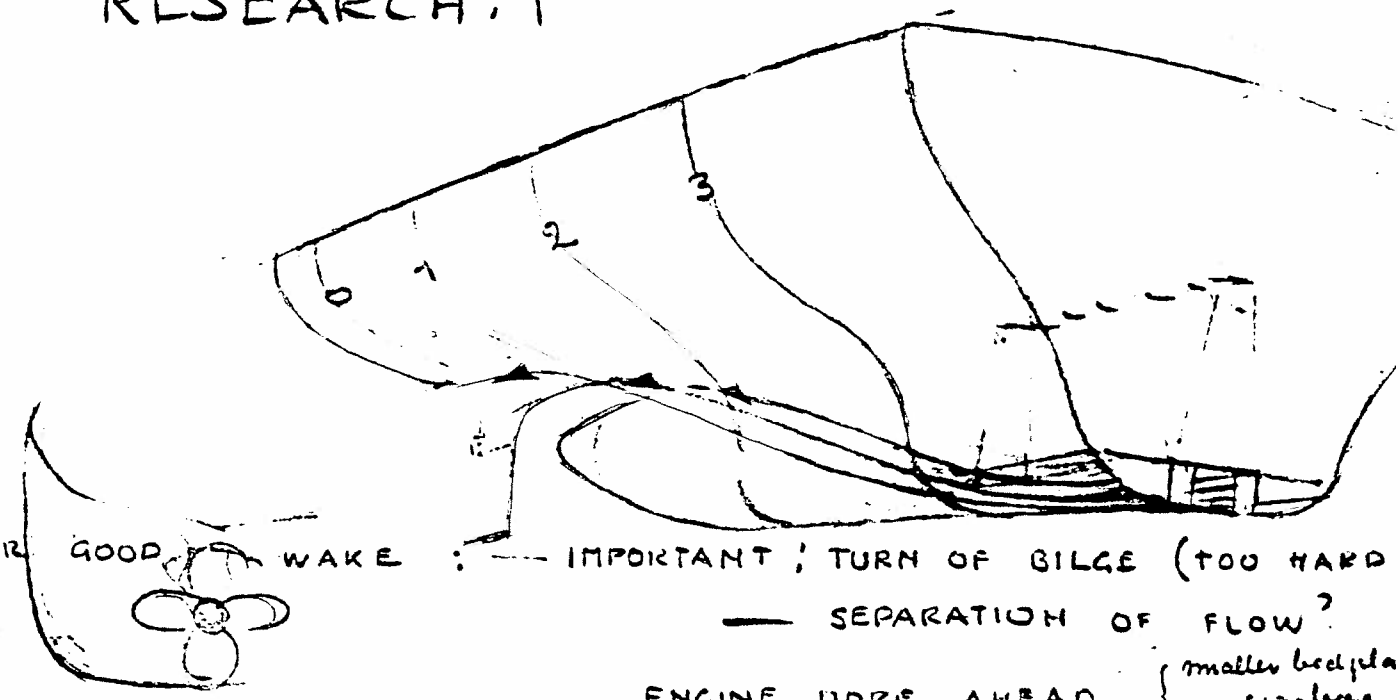
B = DUBBELSCHROEF-MODEL No. 4751 B MET SEMI-BALANS ROER. (Twin-screw with Semi-balanced rudder)



C = DUBBELSCHROEF-MODEL No. 4751 J MET SKEG ROER. (Twin-Screw with 'Barndoor'-type skeg-rudder)

FLOW IMPROVE AROUND BILGE

RESEARCH. 1

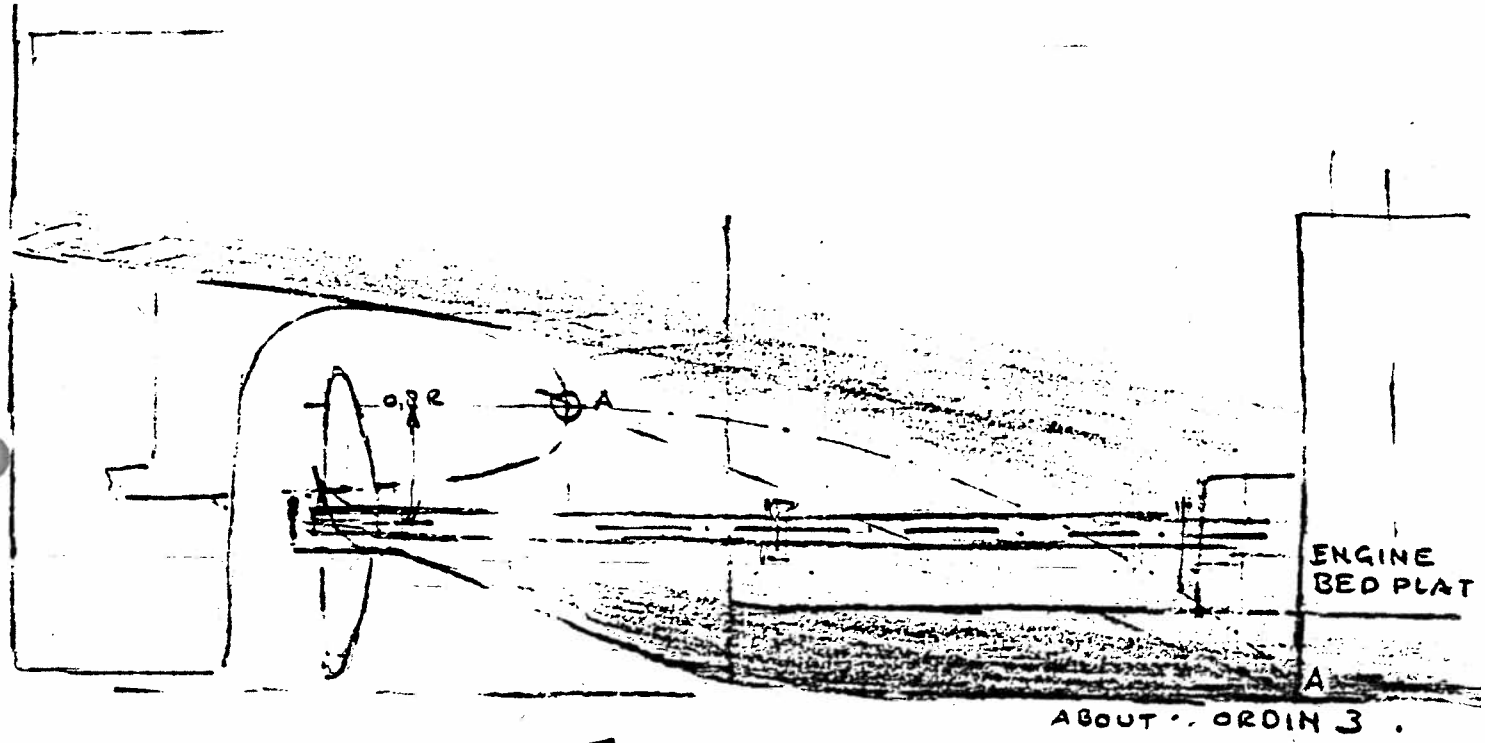


FOR GOOD WAKE : — IMPORTANT ; TURN OF BILGE (TOO HARD

— SEPARATION OF FLOW ?

ENGINE MORE AHEAD { smaller bedpla on sinkage with 2 medium

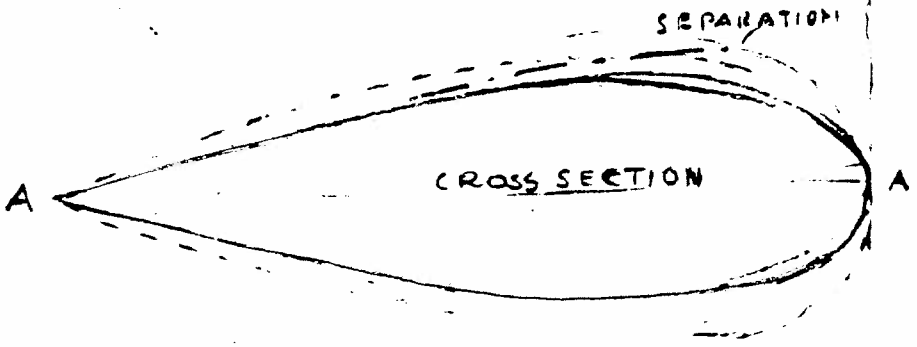
LOSS OF CARGO SPACE.

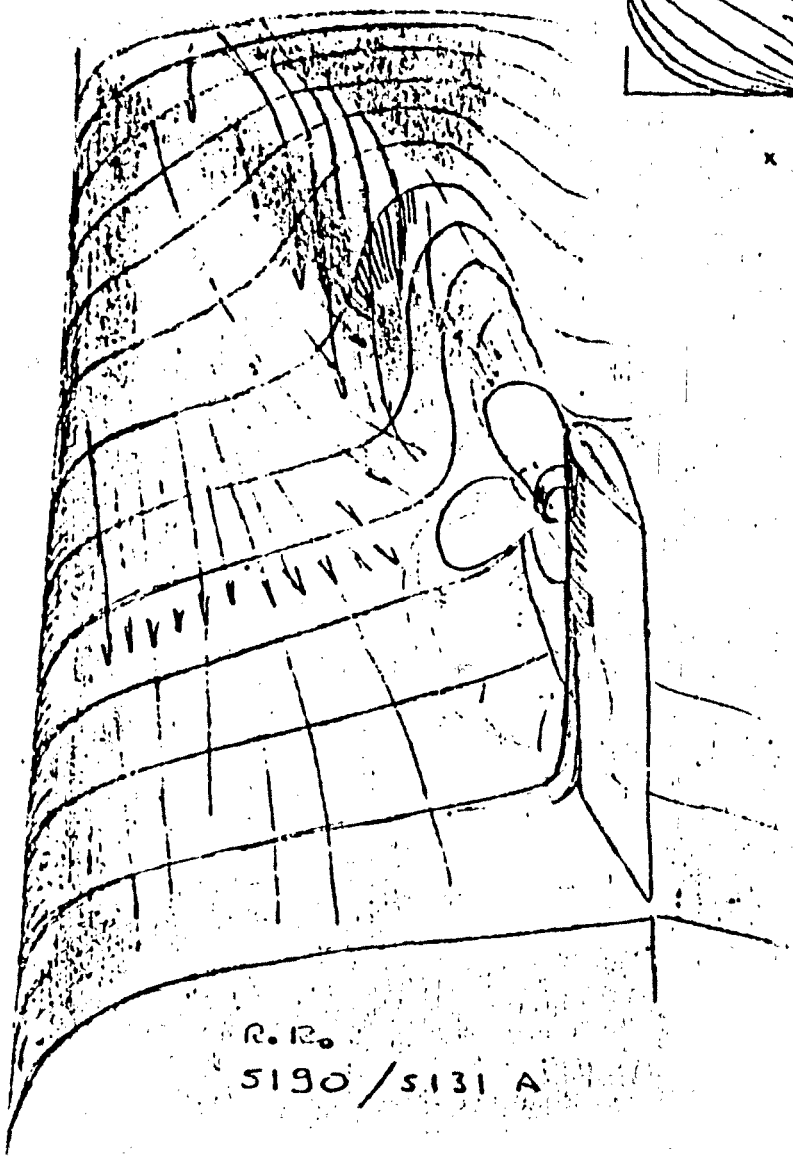


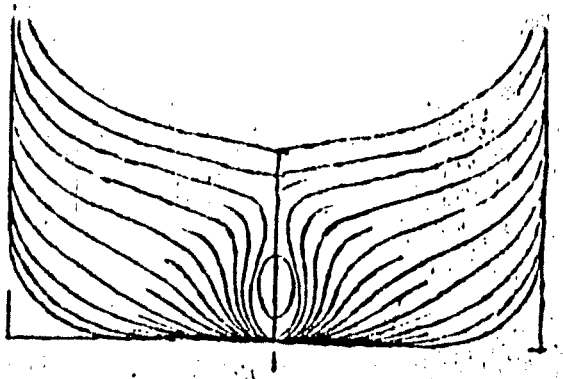
MY PROPOSAL is

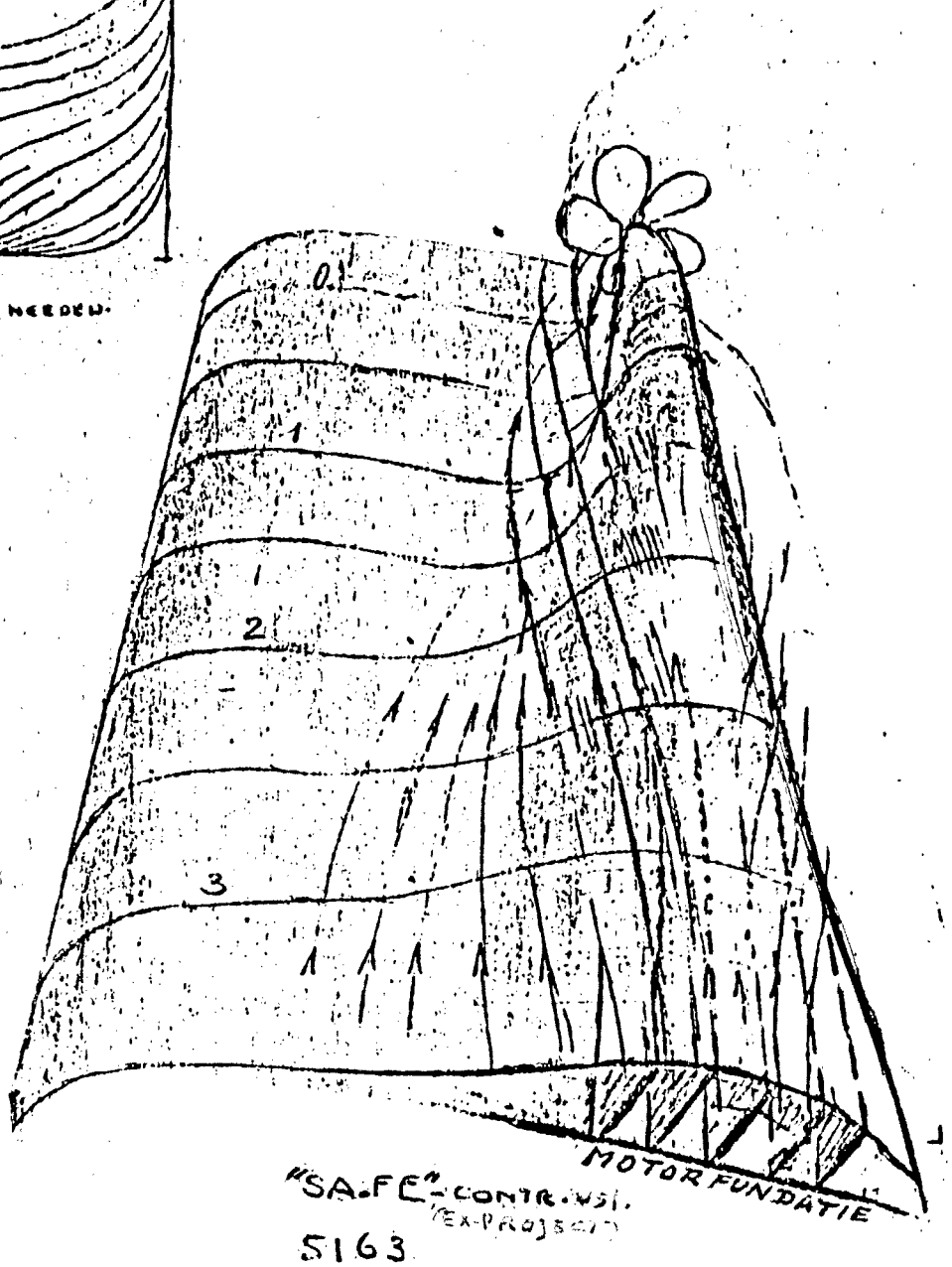
STANDARD LENGTH

TO ANALYSE SEVERAL VESSELS.



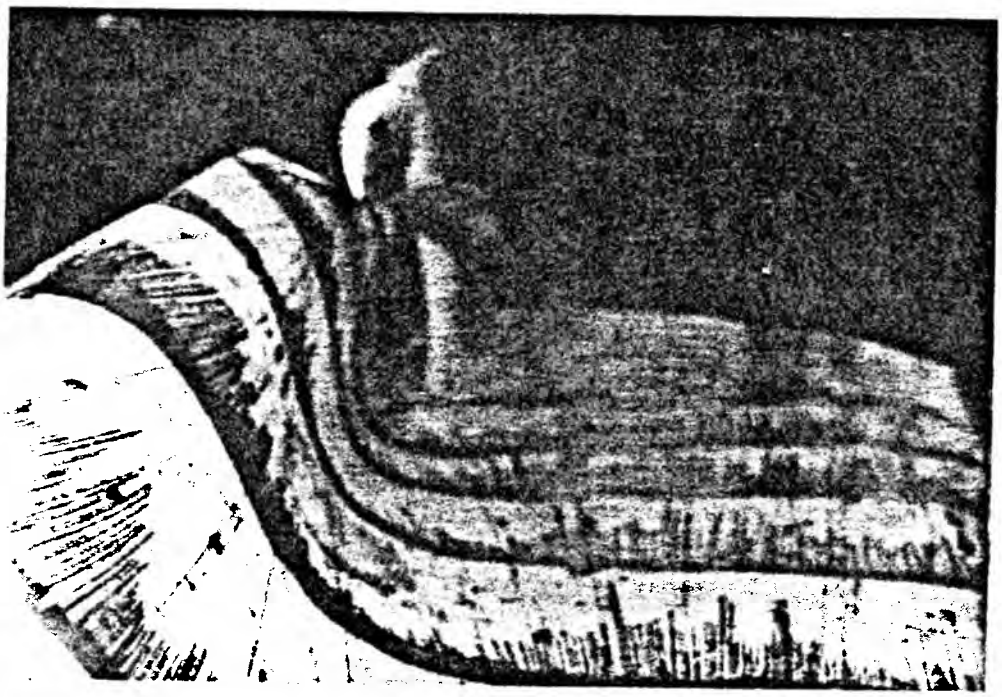


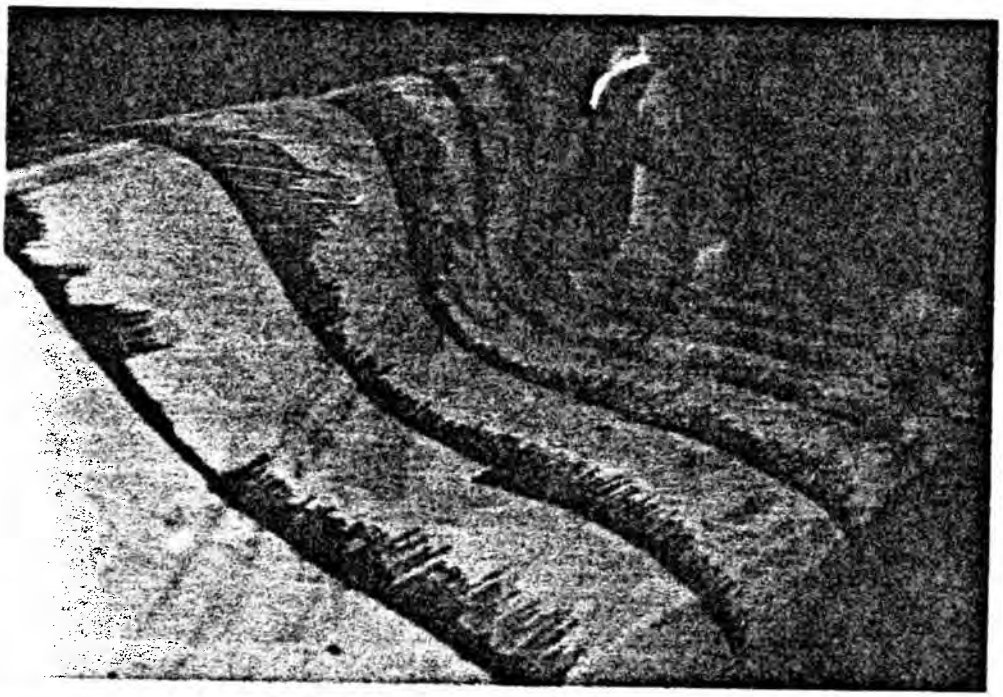


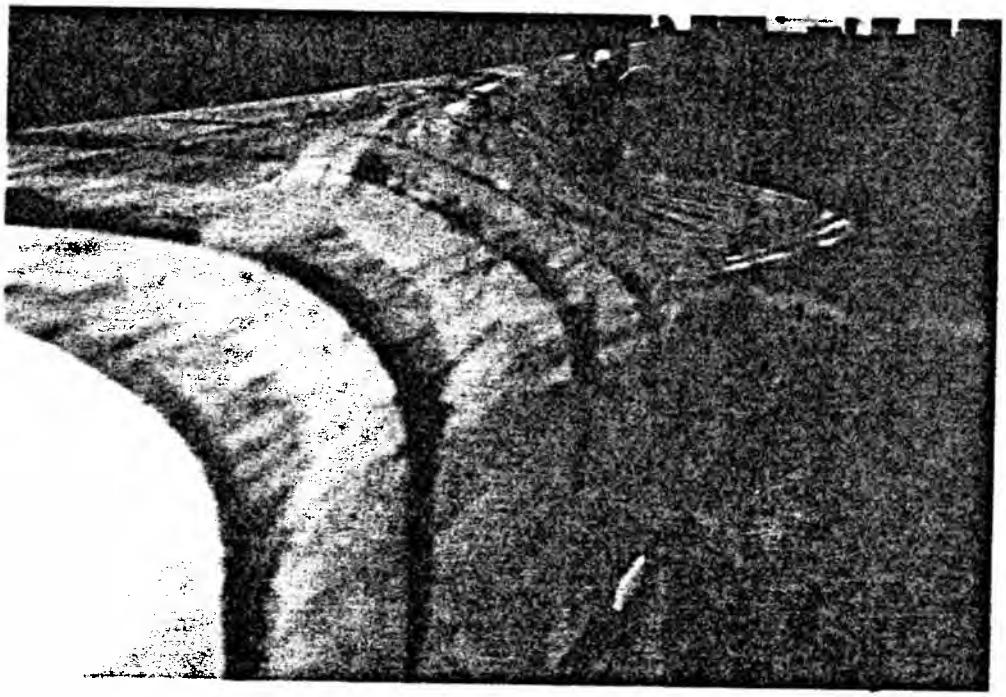


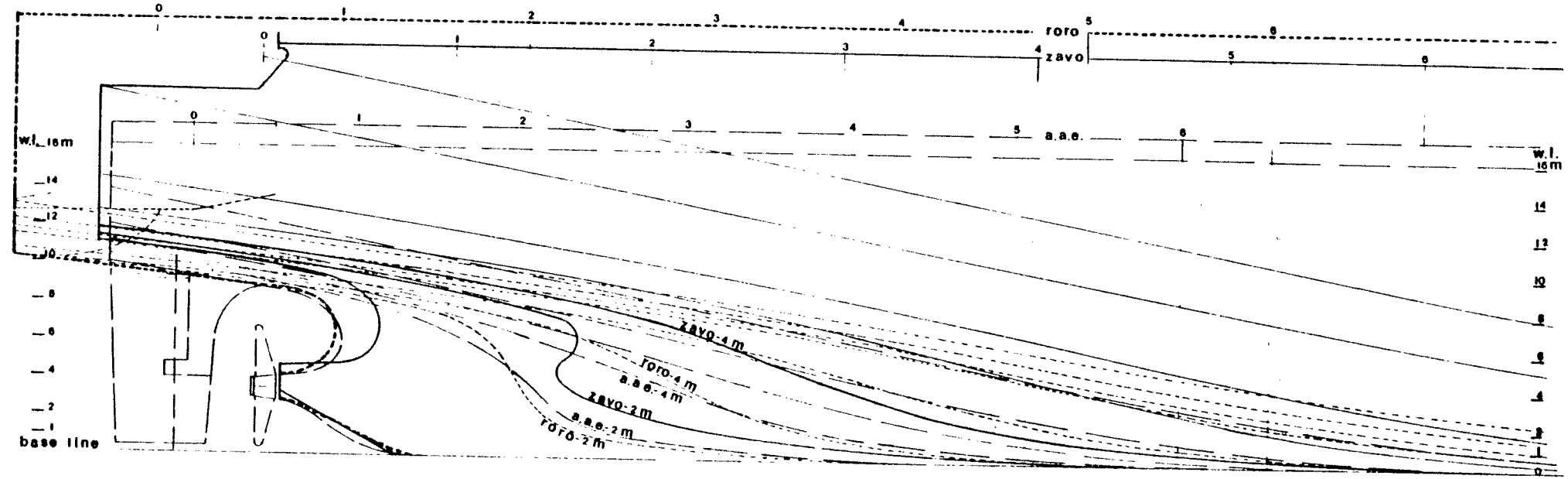
GONDEL

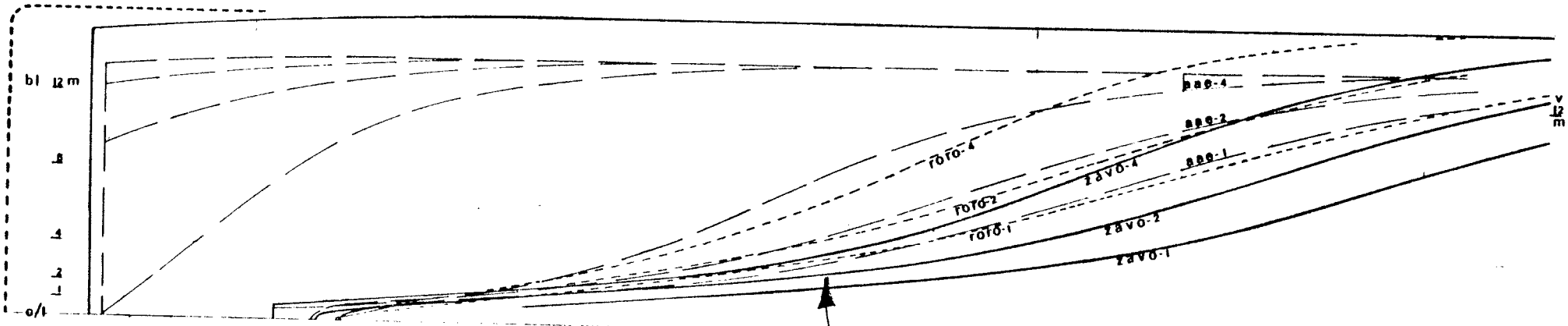
('Gondola' engine support)







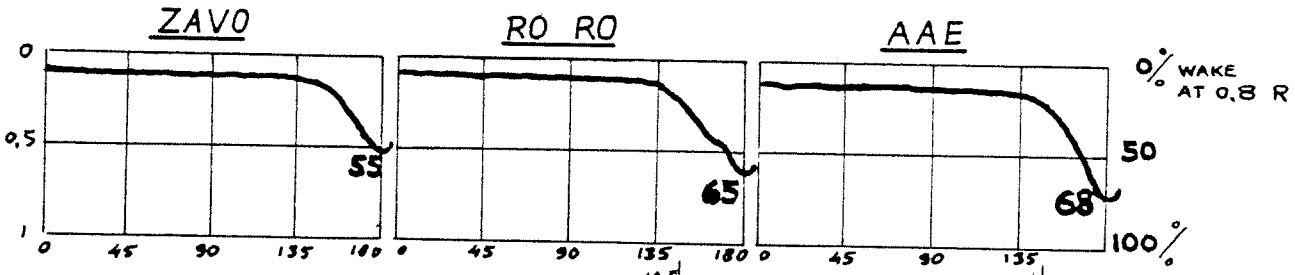


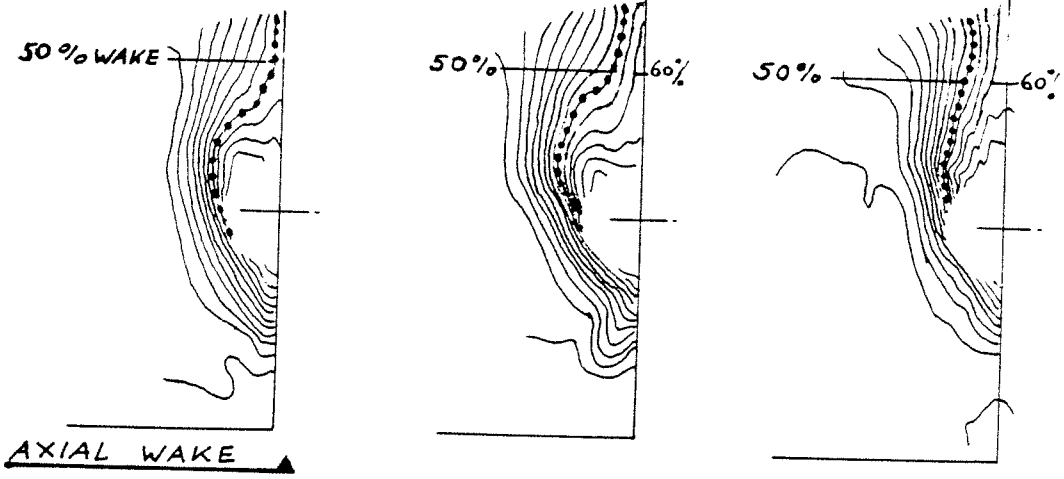


HERE VESSEL TO BE SLENDER AND SOFTLY ROUNDED

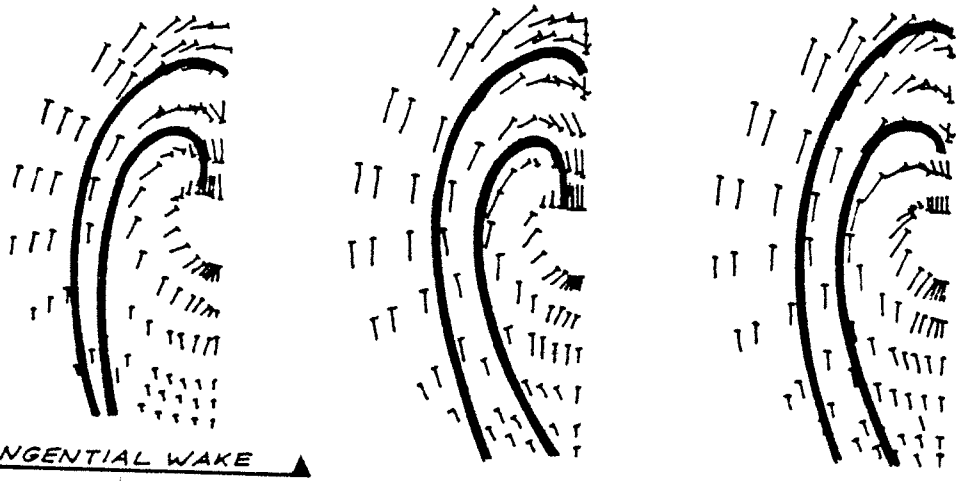
HERE THE WAKE IN UPPER-SCREWDISK IS CREATED!

STRONG INFLUENCE OF DOUBLE BOTTOM LINE ON WAKE PEAK





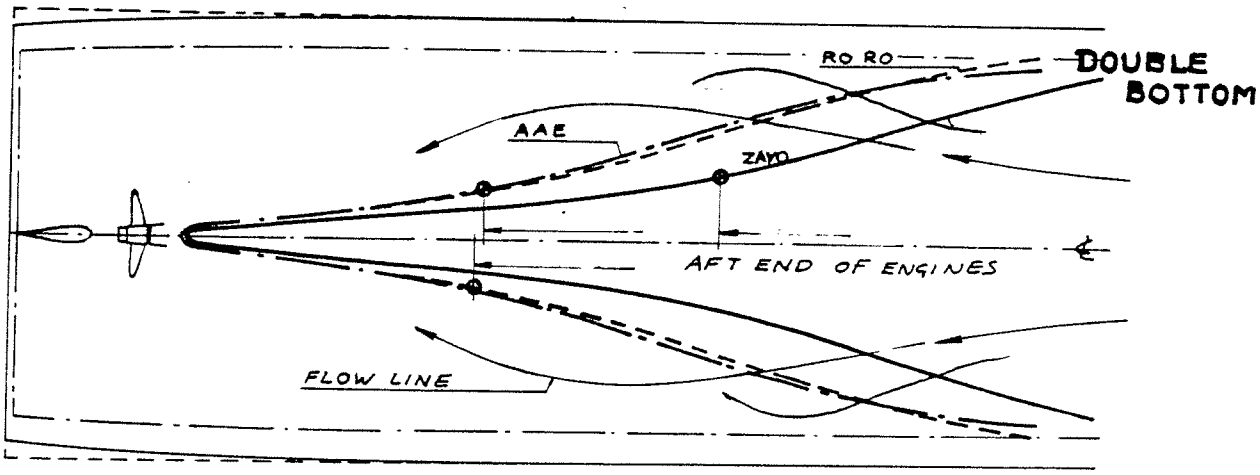
AXIAL WAKE



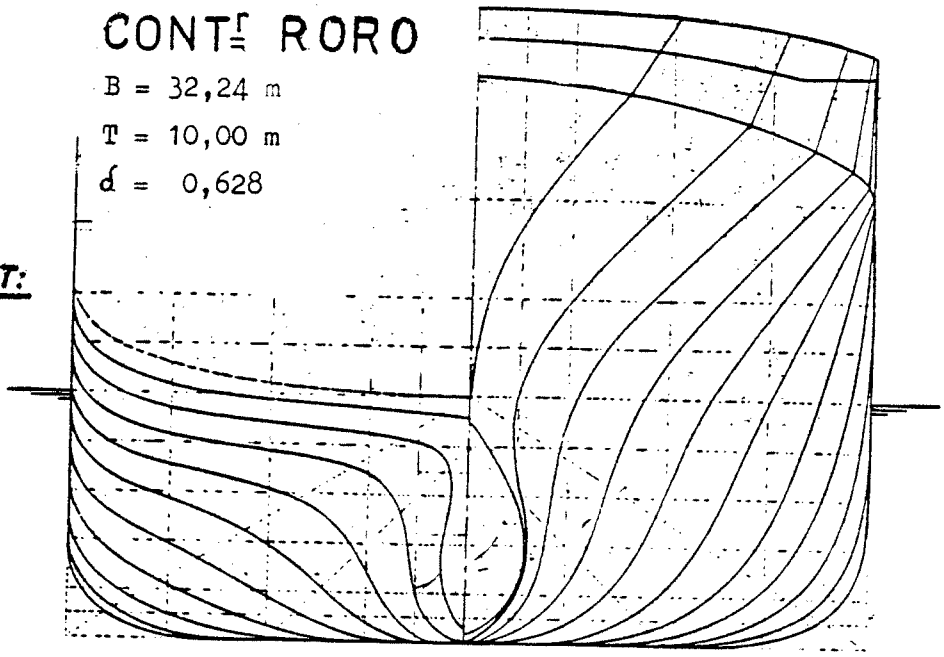
TANGENTIAL WAKE

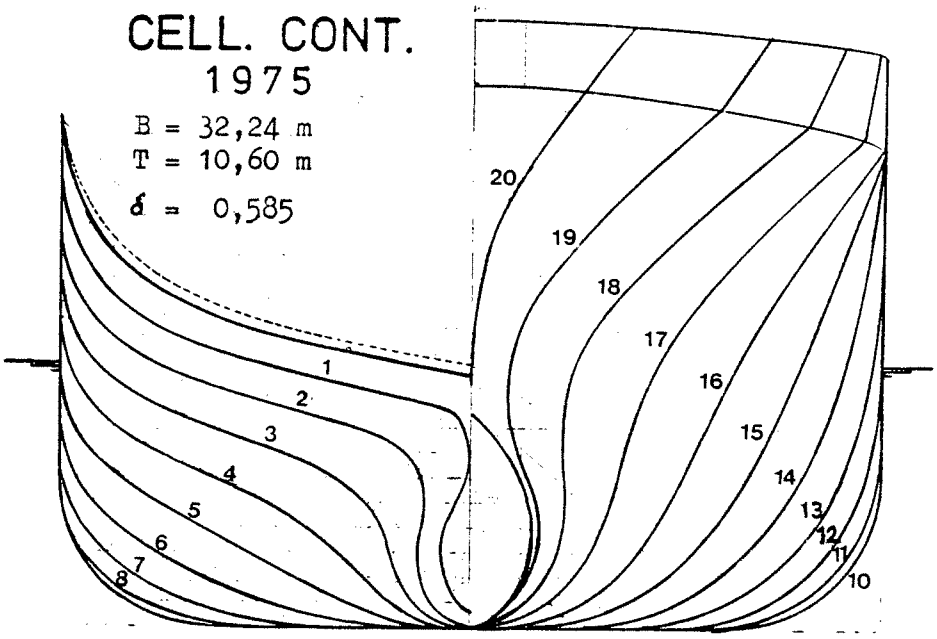
INFLUENCE OF FULLNESS ON WAKE PEAK

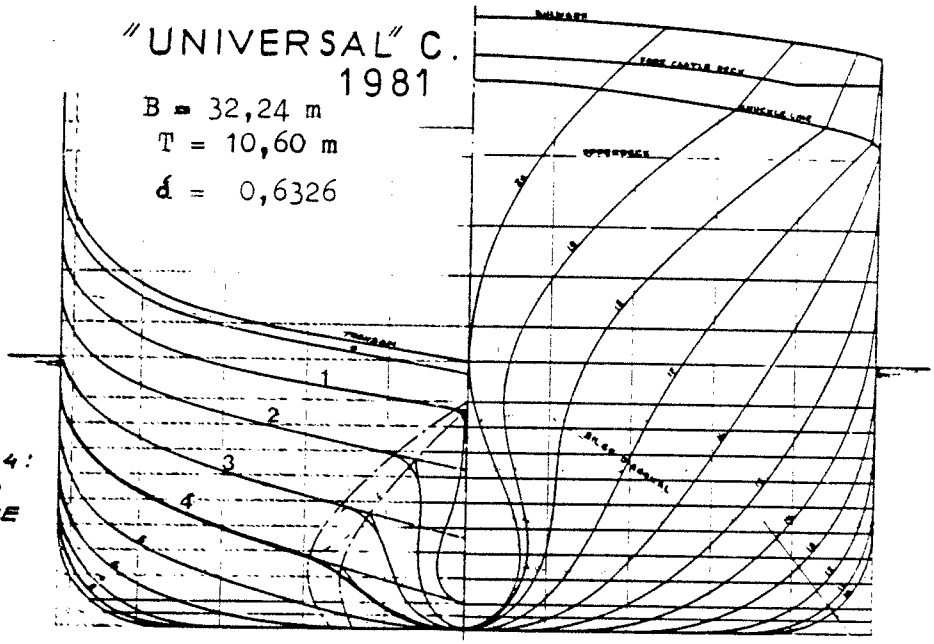
WATERLINES 2m. ABOVE BASELINE



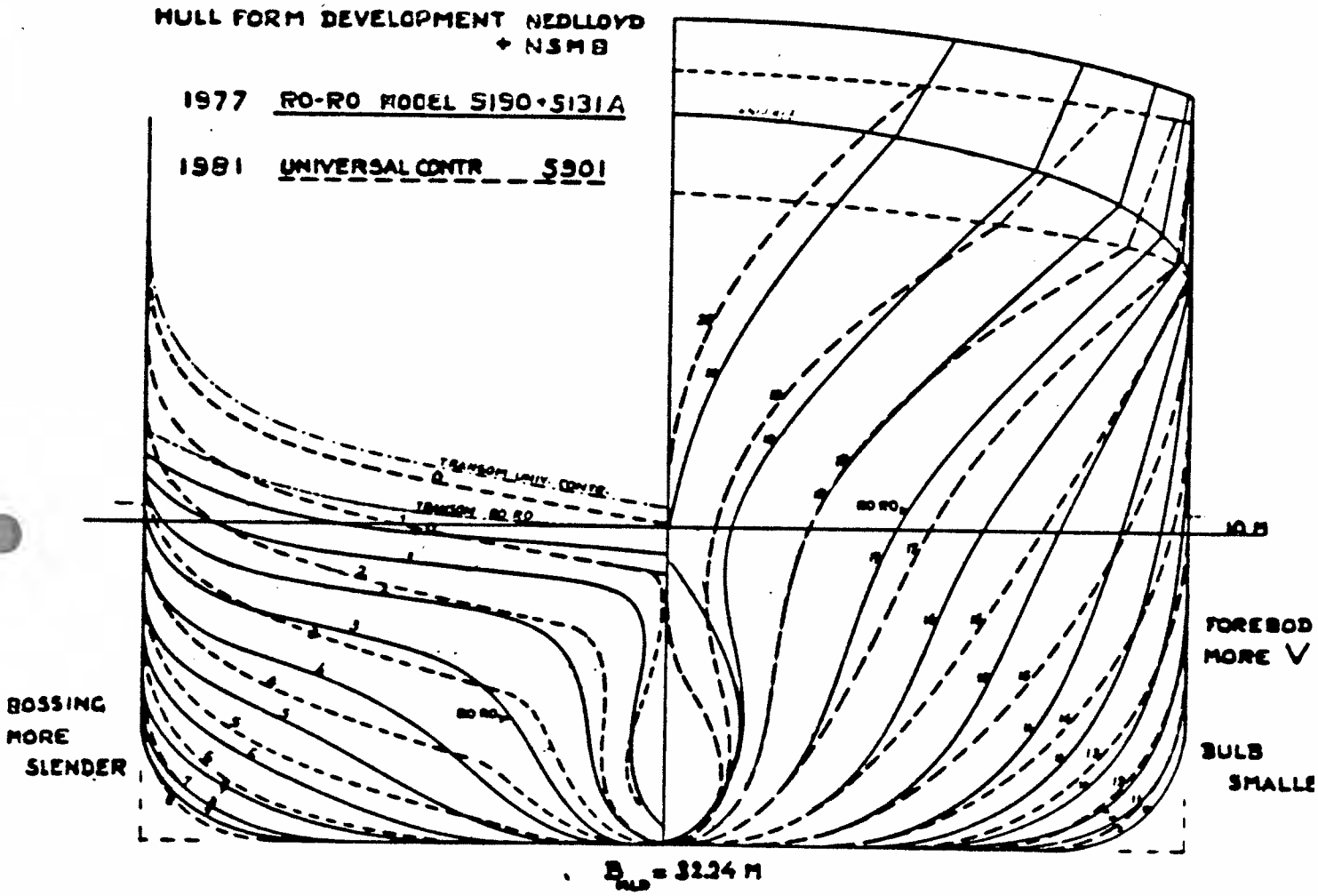
HULL FORM DEVELOPMENT:

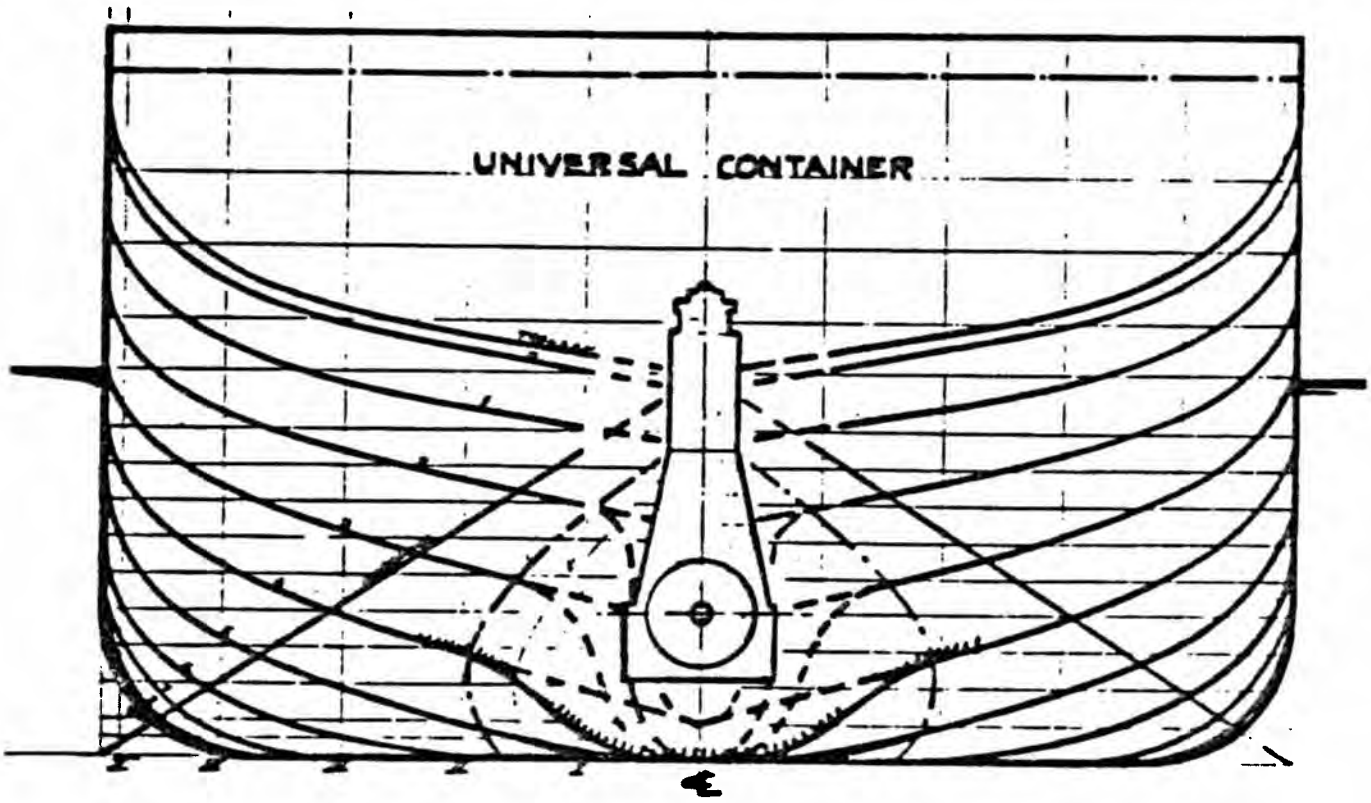






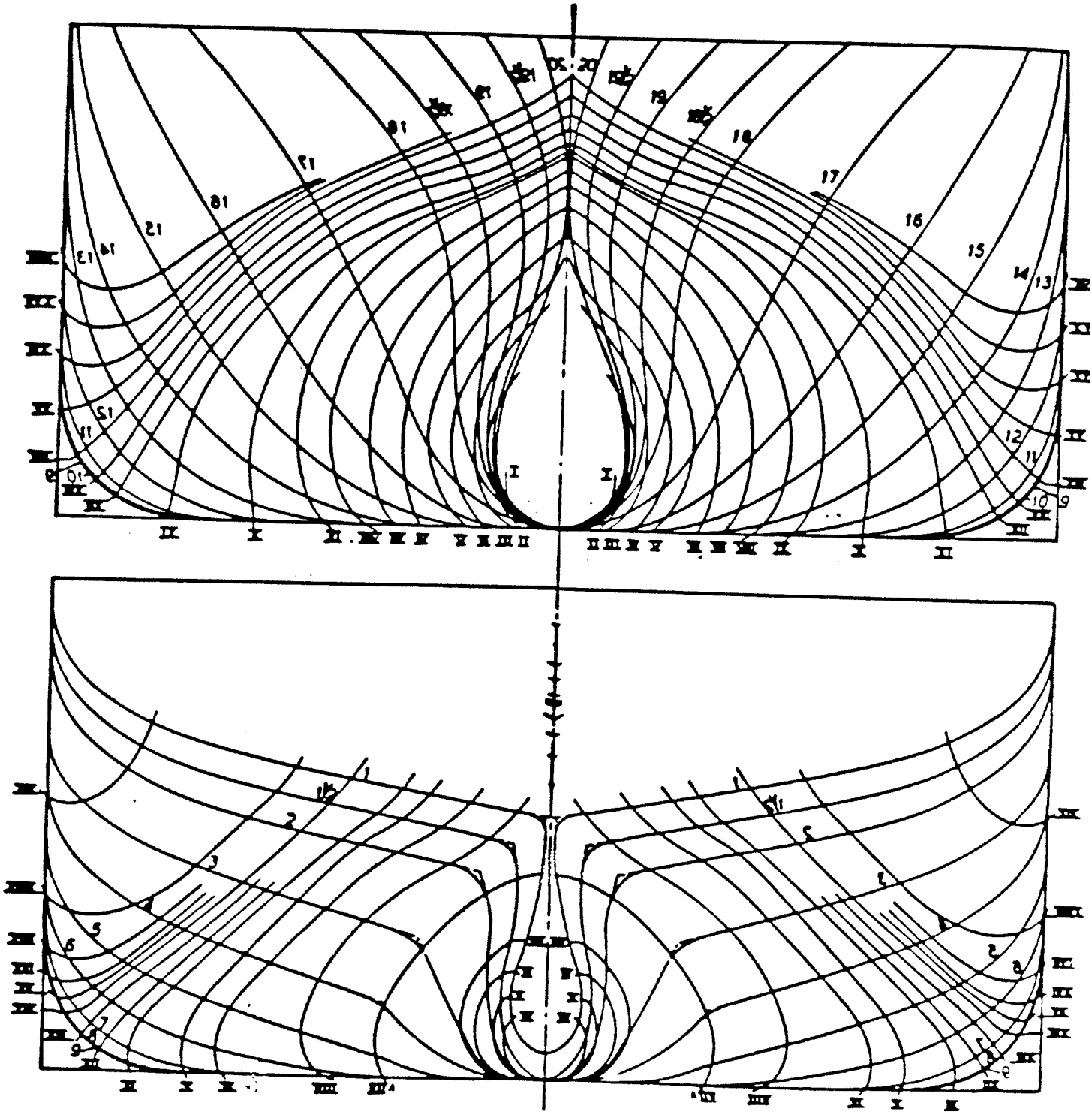
SECTION 4: AFT END OF ENGINE


HULL FORM DEVELOPMENT NEDLLOYD
+ NSMB
1977 RO-RO MODEL 5190+5131A
1981 UNIVERSAL CONTR 5901
10 M
FOREBOD
MORE V
BOSSING
MORE
SLENDER
BULB
SMALLE
B = 32.24 M

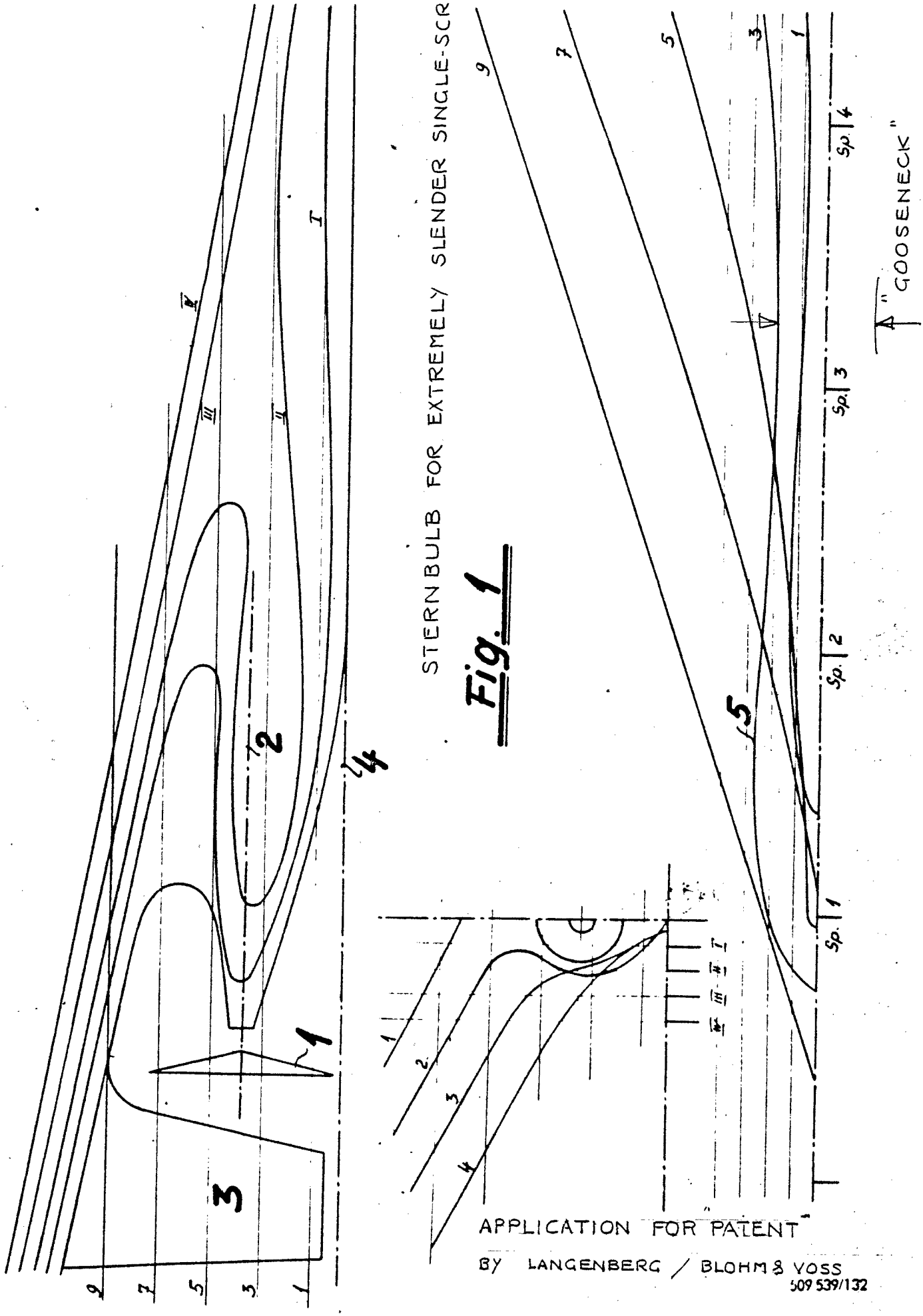

UNIVERSAL CONTAINER

SHIP MODEL Nº 5901 ("UNCO" PROJECT)
STREAMLINES.

DRAFT 10.60 m SPEED 19 KNOTS TEST Nº 39891



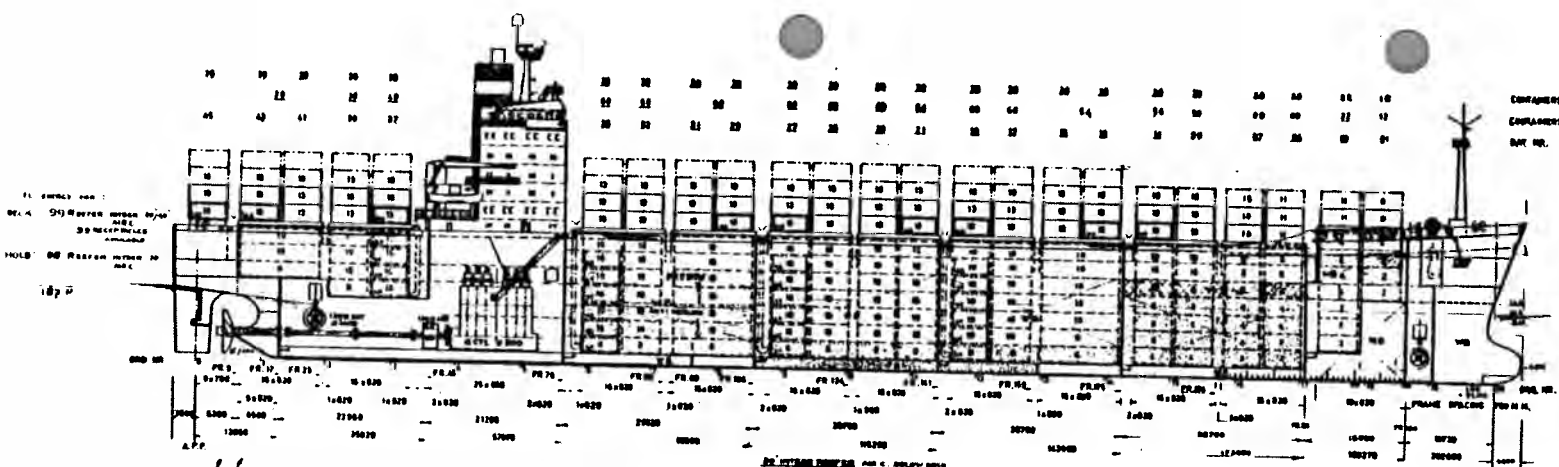
Nummer: 24 41 556
Int. Cl.²: B 63 B 1-08
Bekanntmachungstag: 25. September 1975

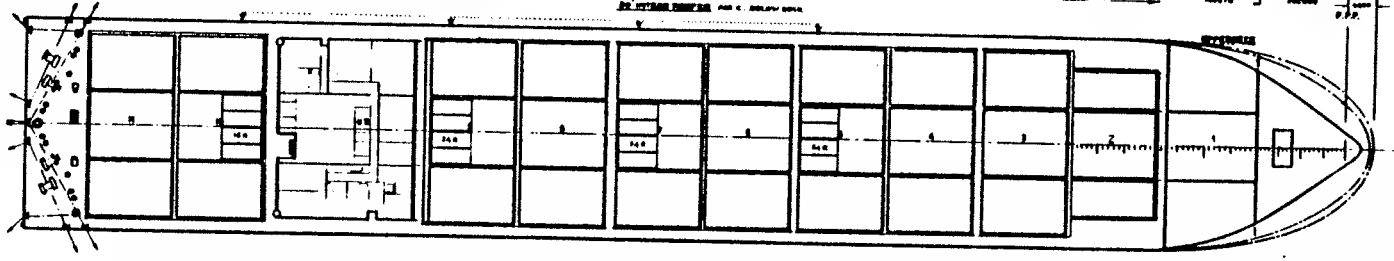


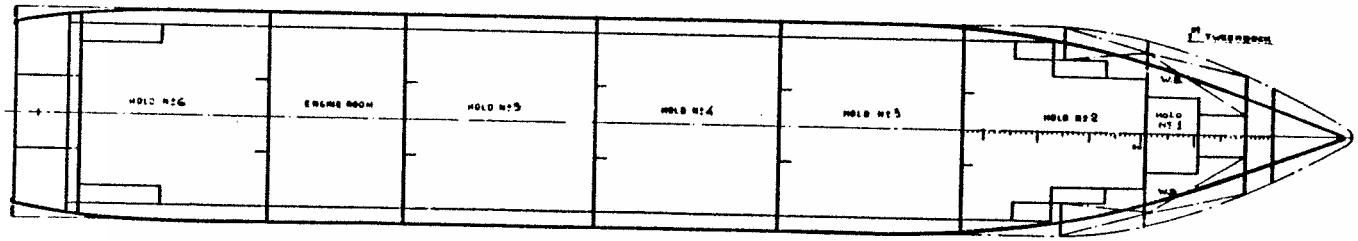
NEDLLOYD CLARENCE
" CLEMENT

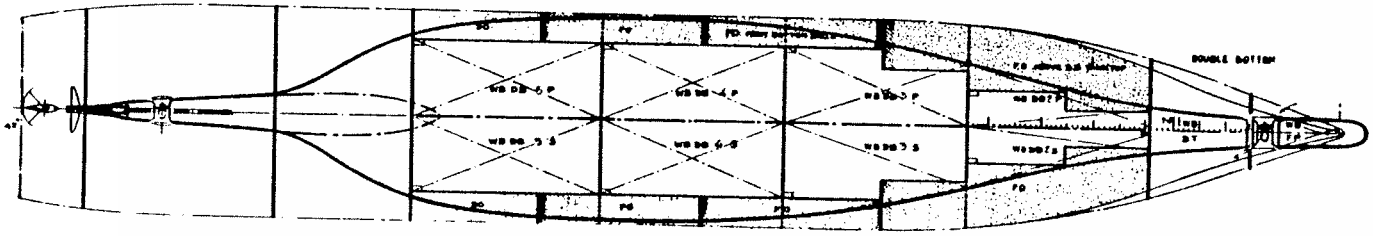
SEPT '83

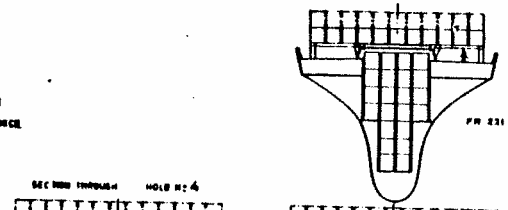
R'dam/NRD/FNB,

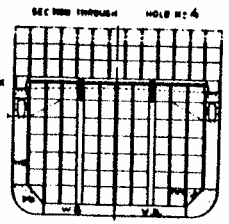


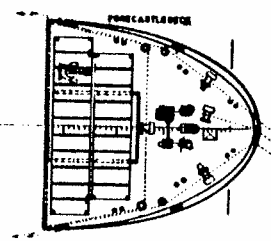


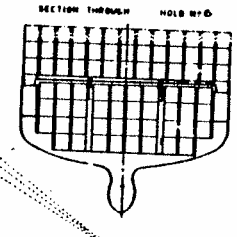












18,00 m. Deadwing	DISPL.	54800 T		6 CYL.
	L.B.	14300		23900 BHP
	B.W.	26100		89.5 RPM
	CONS.+ DT.	4250		20150 BHP
	W.B.	900		1500
	CARGO	33150 T		18650 BHP
				18,3 kn.
10,00 m. 35'	DISPL.	45800 T	HOLD = 1180 = 14,- Ton/TEU	
	L.B.	14300	DECK = 623 = 18,1	
	B.W.	30700	TEU = 2083 = 19,8 = MEAN WEIGHT P. SLOT	
	CONS.+ DT.	4250		
DEEC.	CARGO	26450 T		19,0 kn. - 18,8
9,75 m. 32'	DISPL.	40440 T		
	L.B.	14300		
	B.W.	26140		
	CONS.+ DT.	2700		
PRIDE.	CARGO	23140 T	= 1400 TEU = 16,74 Ton/TEU Dangerous	19,4 kn.
9,15 m. 30'	DISPL.	37340 T		19% SEA MARGIN
	L.B.	14300		
	B.W.	23040		PROP. 15480 BP
	CONS.	2700		SB GEN. 1400
SUMMER.	CARGO	20340 T	= 1310 TEU = 15,6 Ton/TEU	SB GEN. 1180 BP
MERCHAL			1630 = 12,5	= 24 = 150
				= 42 T/day
				FUEL CONS.
				SPEED 18,5 kn.

Container capacity below deck

40' = 160 contrs = 320 TEU = 28%
20' = 830 TEU = 72%

below deck 1150 TEU 100%

CONTAINER CAP. BELOW DECK	1150 TEU
CONTAINER CAP. ON DECK 2 TIERS	562 TEU
GRAND TOTAL 2 TIERS	1712 TEU
3e TIER	291 TEU
GRAND TOTAL 3 TIERS	2003 TEU
4e TIER	221 TEU
IN CASE STABILITY ALLOWS	2224 TEU

PRINCIPAL DIMENSIONS

LENGTH OVER ALL	210.96 M.
LENGTH BETWEEN PERP.	202.00 M.
BREADTH MOULDED	32.26 M.
DEPTH TO UPPERDECK MLD	18.00 M.
DEPTH TO 1st TWEENDECK	14.77 M.
DRAUGHT DESIGN	10.66 M.
DRAUGHT CONSTRUCTION	11.00 M.

A
10-45

Revised 16-8-83

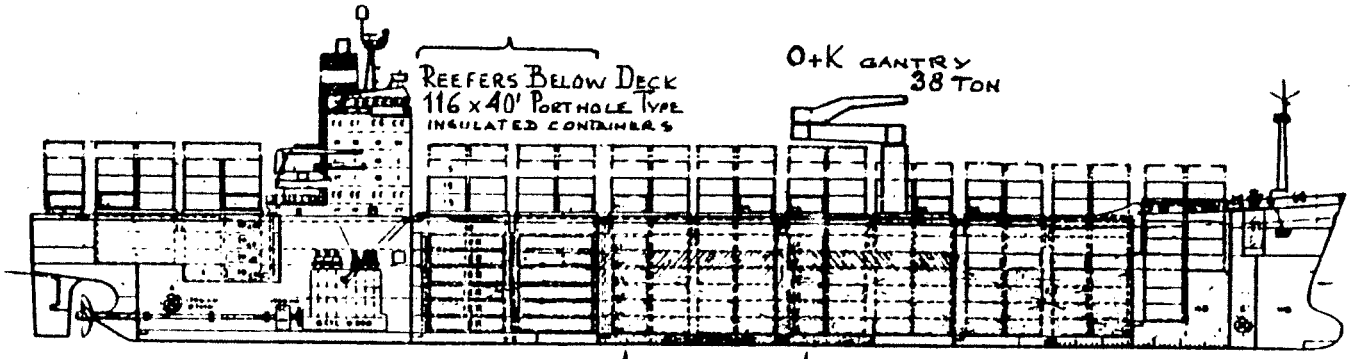
Nedlloyd Fleet Services
Newbuilding Department

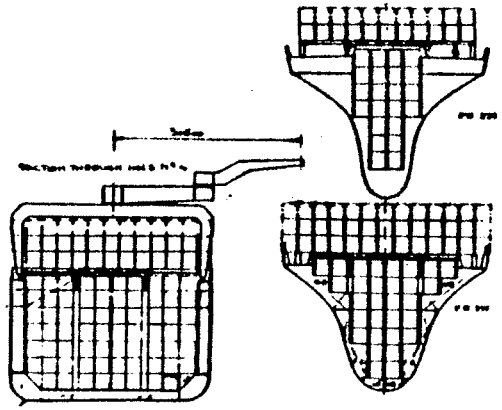
"UNIVERSAL" CONTAINERSHIP

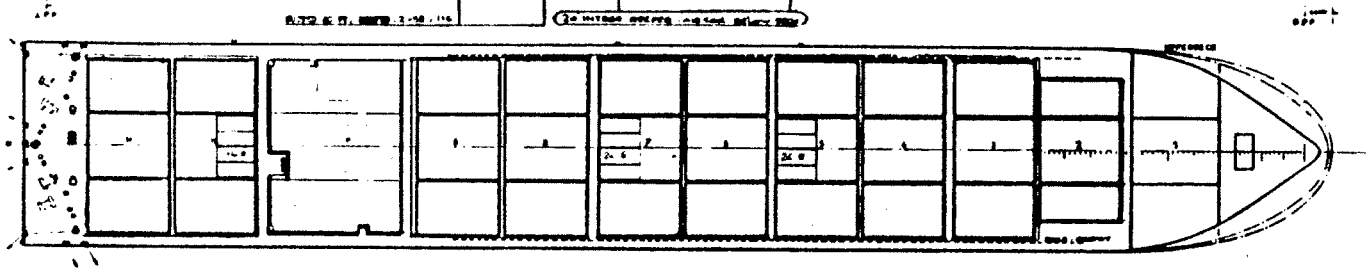
NEDLLOYD CLARENCE

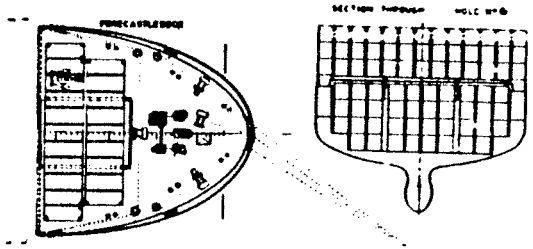
Scale 1 : 500 Row EMEC

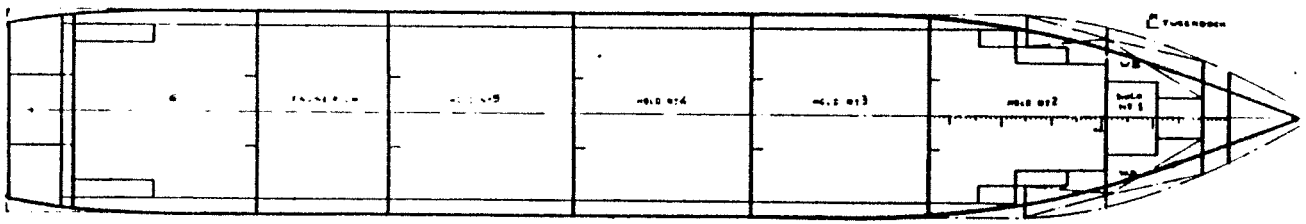
HYUNDAI Nr 225

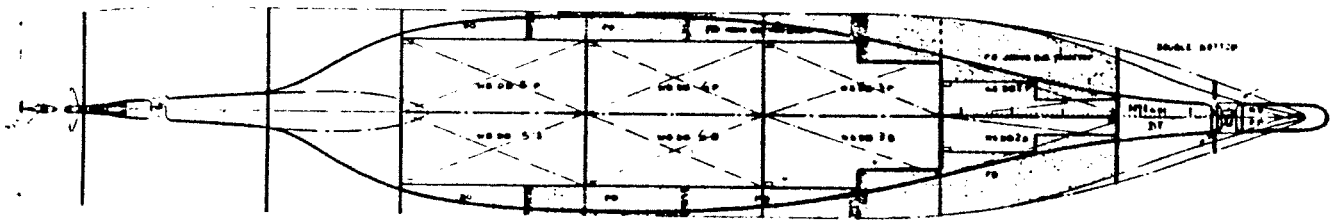












NEDLLOYD CLEMENT (DNB-class, Hole Gantry Crane fitted)

12,00 m (39'3")	DISPL.	34,890 T	
	L.S.	14,500	
	D.W.	20,390	
	OOBS. + BF.	6,750	
	W.B.	940	
	CARGO	13,110 T	
10,60 m (35')	DISPL.	45,000 T	1950 x 14 t/tee
	L.S.	14,500	1997 x 13,1 mean weight per slot
	D.W.	30,500	(Beta 6 tee for gantry-leg positioning.)
	OOBS. + BF.	6,750	
(DWL)	CARGO	23,750 T	
9,15 m (30')	DISPL.	40,440 T	
	L.S.	14,500	
	D.W.	25,940	
	OOBS. + BF.	1,700	
(FT IOK)	CARGO	23,240	Homogeneous 1600 x 16,6 t/tee
9,15 m (30')	DISPL.	37,340 T	
	L.S.	14,500	
	D.W.	22,840	
	OOBS. + BF.	1,700	1310 x 15,4 t/tee
(D.W.A.T.)	CARGO	20,140	1630 x 12,4
Revised			

Container capacity below deck

40	144 containers	328 TEU	28,5%
20	"	822 TEU	71,5%
below deck 1150 TEU			100%

CONTAINER CAP BELOW DECK	1150 TEU
CONTAINER CAP ON DECK 2 TIERS	558 TEU
GRAND TOTAL 2 TIERS	1708 TEU
3e TIER	289 TEU
GRAND TOTAL 3 TIERS	1997 TEU
4e TIER	195 TEU
IN CASE STABILITY ALLOWS	2192 TEU

PRINCIPAL DIMENSIONS

LENGTH OVER ALL	146,00 m
LENGTH BETWEEN PERP.	140,00 m
BREADTH MOULDED	22,86 m
DEPTH TO UPPERMOST DECK	16,00 m
DEPTH TO 2nd TWEENDECK	14,77 m
DRAUGHT DESIGN	10,04 m
DRAUGHT CONSTRUCTION	11,00 m

Revised 16-8-83

Nedlloyd Fleet Services
Newbuilding Department

"UNIVERSAL" CONTAINERSHIP

" NEDLLOYD CLEMENT "

10-43

L_{OA} =	210,00	M	LR-100 A1-LMC-UMS	WATERBALLAST	10.484	m^3
L_{PP} =	202,00	M		FRESH W. SP	150	m^3
B =	32,24	M		" W P }	119	m^3
H =	18,80	M		FEEDWATER P }	30	m^3
				HEAVY F.O	4088	m^3
DRAFT =	12,00	M	SCANTLING	LUB. OIL	287	m^3.
DRAFT =	10,60	M	DESIGN.			

TYPICAL LOADING CONDITION

DRAFT = 10,60 M.

DISPL = 45000 T

L SHIP = 14550

30450 T

CONS + WB ~ 4000

CARGO 26450 — HOLD = 1150 TEU x 14 T/TEU · ⊙ 45% HEIGHT OF CONTR.

DECK = 853 " x 12,1

MEAN = 2003 TEU x 13.2 T/TEU GM ~ 0.4 M

MAX TEU CAPACITY SPACE-WISE : NEDLL - CLARENCE

IN HOLDS = 1150 TEU

ON DECK 2 TRS = 562

3.° TIER = 291

2003 TEU

4.TH TIER 221

2224 TEU.

REEFERS. IN HOLDS {24, 24, 24, 16} = 88 TEU - 20' INTEGRALS AIR COOLED : EXTRA HOLD VENTIL = 1 m^3/TEU.SEC.

ON DECK 11x9 = 99 TEU - 20' à 5KW OR 40' à 7½ KW — RECEPTACLES : 440V CEE 17, 3 HR. 32A

HOLD VENTILATION : N.° 1 = 2,5x HR. EMPTY HOLD -

HOLD N.° 2 = 2,5 x " EXHAUST ONLY : DANGEROUS CARGO

HOLD N.° 3-4-5-6 = ~7 x " BECAUSE OF EXTRA FANS FOR REEFER

SPACE ON TANKTOP FOR BREAKBULK (2 TIERS HIGH) - NOT ON "N. CLARENCE"

HATCHROWS : 3 - 4 - 3

N.° OF HATCHES : 2 - DECKHOUSE - 9

40' CELLS BAY 14 = HATCH N.° 4 = 28 à 40'

BAY 30 = " 8 = 68 "

BAY 42 = " 11 = 64 "

160 = 320 TEU / 1150 HOLD TEU = 28% BELOW DECK

MAX STACKLOAD ON COVERS : 20' STACK = 60 TON

40' " = 90 TON.

LASHING BY "TWISTLOCKS-ONLY" : UP TO 45 TON / STACK. ; ABOVE 45 T : RODS + TURNBUCKLE

MAIN EGINE B+W HYUNDAI 6L 90 GBE

MCR : 23.800 HP – 97 RPM
85% MCR = SERVICE 20.230 HP – 97 RPM

$\times 127^{gr}/_{HPHR} \times 1.1$ (DO → HFO) $= 140^{gr}/_{HPHR} \times 24 \times 20.230$ (85%) $= 68$ TON/DAY.

FUEL FOR ME HFO MAX 600 cSt AT 50°C

18 TON - BOWTHRUSTER = KAMEWA 1200
- STERN " = " "

STEERING GEAR 4 RAM TYPE - KAWASA
35° – TORQUE 150 TON M MAX 155
45° – RUDDERANGLE UP TO 12 KN

FUEL CONSUMPTION (EXCL. SHAFTGEN)

KN.	7	8	9	10	11 M	DRAFT
16.5	34.0	35.4	37.2	39.8	44.5	TON/DAY
17.–	37.0	38.4	40.3	43.2	48.3	
17.5	40.0	41.8	43.6	47.0	52.5	
18.–	43.5	45.3	47.7	51.3	57.1	
18.5	46.6	48.9	51.1	55.6	62.5	MAX 68
19.–	50.5	52.6	55.5	60.5	68.5	
19.5	54.5	57.2	60.6	66.3		
20.–	59.0	62.0	66.2			
20.5	64.4	67.6				
21.–	71.0					

MEAN SPEED FROM PILOT TO PILOT
12% SERVICE ALLOWANCE
FOR ROUGHNESS
WIND
SEA
↓
$HP_{SERVICE} = 1.12\ HP$

TRIAL CONDITION
SMOOTH HULL
BF 2
CALM SEA.

SHAFT GENERATOR MAX 1272 KW – 73 → 97 - MAX RPM.

NORMAL KW EXCL REEFER ~600 KW $\times \frac{1.36}{0.815} = 1000$ HP
1000 HP $\times 140^{gr}/_{HP\,HR} \times 24 = 3.4$ TON/DAY.

FUEL CONSUMPTION INCL. SHAFTGENERATOR 3.4 T/DAY EXTRA ABOVE TABLE

EXAMPLE : 9.5 M DRAFT / 19.5 KN 63.4 T/DAY + 3.4 T/DAY = 66.8 T/DAY.
PROP + S GEN = 68.– MAX

DIESEL GENERATORS

N: 1. 1700 KW – 600 RPM DAIHATSU 8.D.S.32 IFO = 55% HFO + 45% MDO

N: 2 800 KW – 720 RPM " 6. D.S 26 A " "

N: 3 1700 KW – 600 RPM " 8.D.S.32 " "

EMERG. GEN 180 KW – 1800 RPM D.A.F. 6 CYL D.K.S 1160 = LIGHT D.O.

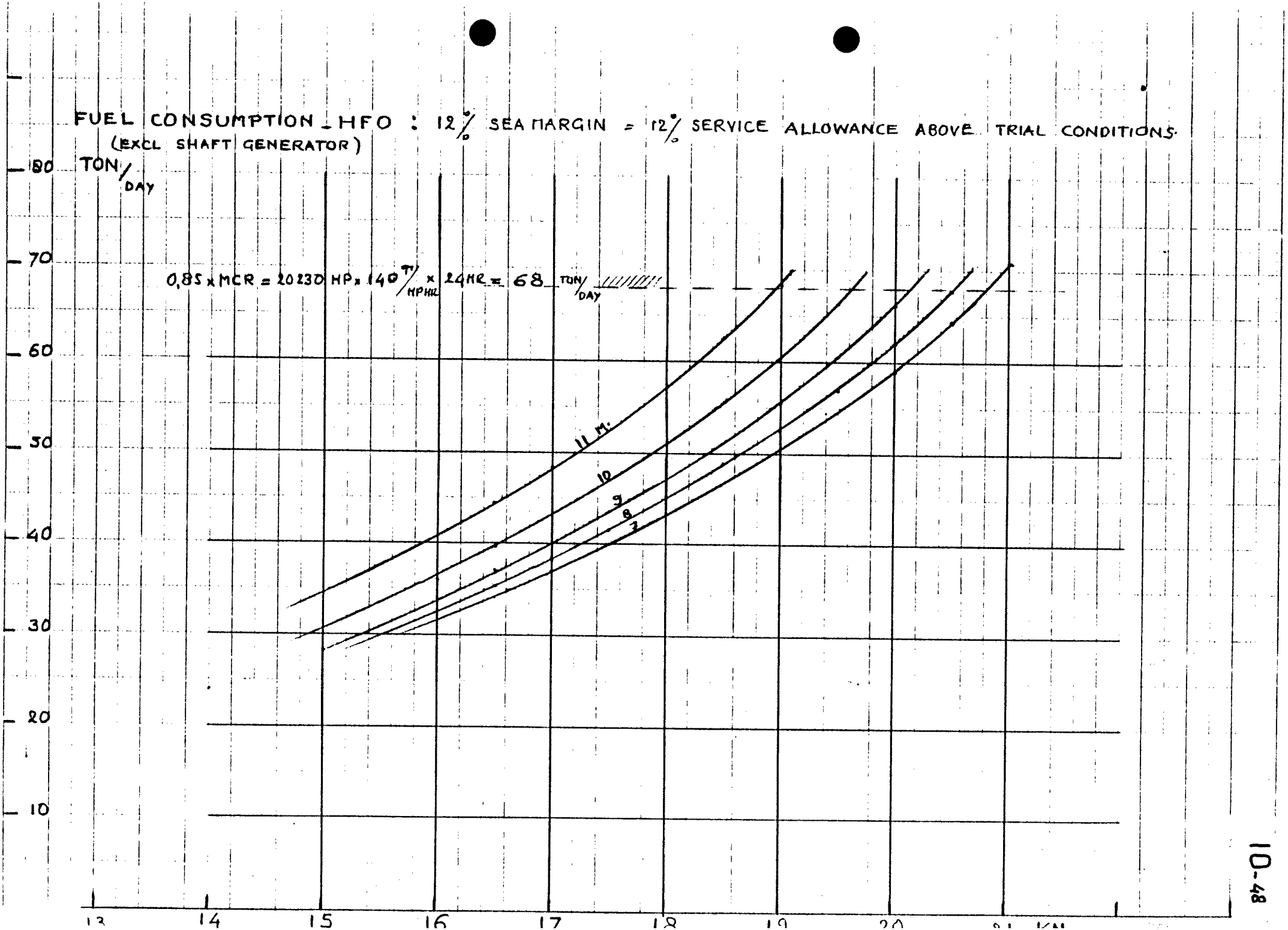
ELECTRIC LOAD AT SEA : MEAN 600 KW BY DIESEL GEN : 3.4 TON/DAY IFO
EXCL. REEFER CONTRS

BY SHAFT GEN : 3.4 TON/DAY H.F.O

IDLE IN PORT 450 KW BY DIESEL GEN : 2.6 TON/DAY IFO.
+ AUXIL. BOILER (MAX 2.5 T STEAM/HR) MEAN : 1.0 " H.F.O

EXTRA KW FOR REEFER CONTAINERS (INTEGRALS) : N: BOX × KW/BOX × HP/KW × GR/HPHR × 24

EXAMPLE 100 × 5 FOR 20' / 7½ FOR 40' × 1.5 × 160 × 24 = 2.9 TON/DA


FUEL CONSUMPTION _ HFO : 12% SEA MARGIN = 12% SERVICE ALLOWANCE ABOVE TRIAL CONDITIONS.
(EXCL SHAFT GENERATOR)
TON/DAY
80
70
60
50
40
30
20
10
0,85 x MCR = 20230 HP x 140 T/HPHR x 24HR = 68 TON/DAY
11 M.
10
9
8
7
13
14
15
16
17
18
19
20
21 KN

GENERAL DESCRIPTION

"Nedlloyd Clarence" - "Nedlloyd Clement"

At the shipyard of Hyundai Heavy Industries in Ulsan, South Korea, two cellular container vessels are in the building stage, which are expected to be delivered end of August and end of November respectively.

Newbuilding 225 - "Nedlloyd Clarence"
Newbuilding 226 - "Nedlloyd Clement"

Main Dimensions

Length over all	-	210,00 m
Length between perp.	-	202,00 m
Breadth	-	32,24 m
Depth to upperdeck	-	18,80 m
Design draught	-	10,60 m
Max. draught	-	12,00 m

Deadweight

At the design draught of 10,60 m the deadweight will be abt. 30.500 tons.

Container capacities

The holds are equipped with container guides for carrying 10 containers in breadth and 7 in height.
On deck containers can be carried 13 in breadth and 3 to 4 in height.

	"Nedlloyd Clarence"	"Nedlloyd Clement"
Under deck	1150 teu	1150 teu
On deck	853 teu	847 teu
Total	2003 teu	1997 teu

The hold capacity is divided into :

160 - 40' cntrs = 320 teu = 28%	164 - 40' cntrs = 328 teu = 28½%
830 - 20' cntrs = 830 teu = 72%	822 - 20' cntrs = 822 teu = 71½%

On deck the container positions are interchangeable between 20 ft and 40 ft, except for the most aftermost bay which is for 20 ft only.

The vessels are based on containers with a height of 8'-6".
Both vessels have positions and electrical connections for 187 resp. 163 integral reefer containers, while "Nedlloyd Clement" will also be fitted with a ducted air cooling and distribution system for refrigerating of 116 porthole containers of 40'

Loading gear

Each vessel will be equipped with two 8 tons provision cranes, while on "Nedlloyd Clement" a 45 tons travelling gantry crane will be fitted forward of the deckhouse, for handling containers.

Main engine

The vessels will have a main diesel engine of the type B & W 6L 90 GBE, developing a maximum continuous output of 23.800 HP. The normal output (85%) is 20.230 HP at 97 RPM.
A shaft generator with an output of 1200 kW will be fitted.

Speed

The service speed at design draught of 10,60 m, at normal output, with shaft generator in operation, will be abt. 18,7 knots.

The vessels will be equipped with a bow thruster and a stern thruster with an output of 1200 kW each for easier manoeuvring.

Accommodation

Accommodation is available for a crew of 26 persons, while extra cabins are available for pilot, repair crew and Suez crew.
All crew members are situated in single cabins, each with private sanitary equipment.
Cabins and public spaces are of modern design and provided with airconditioning.
A covered swimming pool is available.

Miscellaneous

Sophisticated nautical equipment will be fitted.

FNB/JHdJ.

DESIGN PHILOSOPHY REGARDING NEDLLOYD 'UNIVERSAL' CONTAINERSHIP.

Capacity and speed.

The commercial departments required a vessel of about 1800 TEU of 14 tonnes per TEU and 18-18.5 knots service speed and being as flexible as possible for interchange between the company's various services. Contrary to earlier periods, this vessel was not to be designed for a certain trade but was to be utilisable for worldwide container transport. Many times Nedlloyd experienced reshuffling her ships in co-operative services.

Length Over All.

Almost all container terminals can accommodate ships of 210 M length over all - except some West and east African ports.(The latter are to be served by a smaller ship type such as "Nedlloyd Van Neck type of 182.50 M overall and 1500 TEU capacity.)

- Good manoeuvring properties, however, are required, especially in the ports on the South American West coast where the vessel has to turn around within 300 metres length by her own means.
- For this reason a bowthruster and sternthruster were fitted which also have the advantage of shortening the roundtrip time and reduction of deck-crew.

Breadth.

Max. Panama beam has been selected in order to obtain on the worldwide routes the maximum attainable stability which will allow for maximum container carrying capacity.

- This 'Universal' container is able to carry 7 tiers containers in the holds plus 4 tiers on deck, resulting in a carrying capacity of 2224 TEU at the maximum, at an average container-weight of 12 tonnes per TEU. Normal in-homogeneous stow and ahead-view requirements have been taken into account.
- Normally, the ship can carry at 10.60 M draft a number of 2000 TEU at 14 t.av.wt. in the hold and 12.1 t. on deck, which results in 13.2 t mean weight per slot. Herewith the stability is about 0.4 M metacentric height, being an acceptable value.

Hull form and propulsion.

A V-shaped forebody and a Pram-type afterbody (like those of old sailing ships and yachts) will result in an easily driven hull with maximum form stability and maximum deck space.

- The flow of water towards the propeller has been smoothened as much as possible - by scraping off some displacement - in order to obtain a good propulsion and a minimum of propeller-excited vibrations.
- The 'large' propeller at rather low revolutions will give a good propulsive efficiency.
- A fine underwater-afterbody, however, will automatically ask that engine room plus deckhouse are to be moved somewhat foreward.
- The reason for this is the space required for a sufficiently stiff engine foundation and also the trim-balance of the ship.
- Moreover, the fuel oil is to be carried in the forepart of the hull in order to sail on even keel. (Sailing with - say - about 3 feet astern trim on these vessels is very uneconomical, because ship's resistance increases enormously by the combined action of emerging bulb and immersing transom stern).
- For this reason - and also for keeping the hogging moments in the hull at a low(er) value - Nedlloyd decided on a general arrangement with engine room not too far aft (i.e. two hatch-lengths before the stern instead of only one as is the case on many a similar containership). With such arrangement the balance of the ship is better and she needs less water-ballasting in the forebody for compensating the astern trim.
- Meanwhile, by placing the deckhouse more forward, the vibration hindrance caused by the propeller will estimately become less.
- A small disadvantage is the loss of a few containers in holds. This loss was accepted by us because a very great part of the containerload is carried on deck.

Draft.

The scantlings are allowing for 12 metres draft.
However, this (= 40 feet) draft is too deep for an economical service condition, especially in view of immersion of the transom.

- The hull form will be optimal between 8.5 M and 10.5 M drafts. This mainly decided the transom depth and bulb diameter.

- Too many ships are designed with too big bulbs. Big bulbs will function as water'turbulating' brakes at the smaller service-draft ranges which quite often occur on coastal legs between the many ports.
- Certain South American ports and Indonesian ports allow for 9.15M (30') and 9.75M (32') which figures are limiting the cargo deadweight of the vessel.

Block coefficient

In order to have sufficient cargo deadweight at the smaller drafts, the block coefficient should not be too fine.

- Remunerativeness calculations in 1981 - at which time fuel prices and ships' prices were rather high - the optimum blockcoefficient for an 18½ knots service speed varied between 0.625 and 0.65.
- At the moment (June 1983) with fuel prices at somewhat lower level as well as ship's prices the optimum varies between 0.635 and 0.66 .
- Many containerships nowadays have too fine lines and (consequently) cannot carry weight from shoal ports !

Cubic capacity.

The containerslot capacity within the hull is fully utilized up-to the last centimetre.

Experience with containership-designs of the latest 10 years showed the optimum length of a cellular container hold to be 28.70 metres (== 35 frame spacings of 820 MM each).

The minimum required length of engine room and length of mooring space of the fore and aft decks are to be added to a number of cellular blocks and the overall-length is born. This L.o.a. in our opinion is a muchbetter value for defining a vessel than the old-fashioned L.p.p. (Length between the load-lines).

2000 TEU boxes - with 3 tiers above deck - on a 210-metres ship of Panama beam is a good capacity. However, it has to be realised that the boxes on the rear end of the ship (above the mooring winches) are NOT to be assessed as normal loaded containers. This is because hogging moment in the hull and astern trim of the vessel will limit the weight of these boxes to be very light ones only.

Seaworthiness.

The vessel accommodating 7 tiers containers within the hull has sufficient freeboard at service draft of 10.6 M and the range of stability is rather large. Thus, a low metacentric height (0.4M) is allowable and will give a gently moving ship.

- However, if only a small amount of containers is to be transported, the vessel is too stable because metacentric height is very great and as a result jerking rolling movements will occur at sea. The only answer to that problem is to pile the boxes up as high as possible and sail them mainly on the hatchcovers.
- Securing of the containers on deck should in general be done by means of twistlocks only (except when the topweight on the lowermost box of a stack exceeds 30 tons).
- Application of fin -stabilizers is considered to be not economical anymore. At the low(er) speed level the fins' surface would have to be large, the necessary larger size of the aparatus would be (more) expensive and the fuel bill considerably higher than formerly .
- At the other hand, 'steered antirolling tanks' would have to be very large sized and this would incur corrosion problems in the course of the 20 years' service.

Strength.

The structural 'sideboxes' of 2.35M width have already proved to be sufficiently stiff and strong on the larger sistership and moreover the Length/Depth ratio of the 210M by 18.8M vessel is quite reasonable.

In this respect also the knuckle line at the forebody is to be mentioned as it accentuates the box-girder surrounding the hatch openings in the foreship.

- The sidebox creates axial stiffness against headseas and also will throw the water away from the vessel when the ship is pitching in headseas.
- We prefer this knuckle line more than breakwaters on deck for this reason. Also, it looks nice because this feature is a 'last rudiment' of the ship's sheer.

Hatch configuration. Reefers.

- With a view to flexibility in container stowage, the triple-hatch arrangement is to be preferred above the twinhatch type.
- When many ports are to be called at in one voyage, the twin-hatch vessel will incur much more reshuffling of deck-containers than the triple-hatch configuration. This will cost much more than moving of covers only.
- Structural-wise, the triple-hatch arrangement creates two 'violin strings' over the whole length of the vessel, acting as longitudinal strength members (and may fall less heavy than the single-string of a twin-hatch arrangement vessel).
- Contrary-wise, the hatchcoaming structure above the deck should be of the interrupted type in order to avoid high stresses.
- Transverse members consist of wide and slender structures (cross-overs). The widest ones are accommodating ventilation ducts and stairways.
- By adopting this configuration it is always possible to provide the ship with 'integral' type reeferbox capacity underdeck. Cooling these containers - or rather their engines - can be executed by means of aircooling (pressure ventilation) only or by means of airblowers and water-piping (connected b.m.o. flexible hoses to a shipboard system).
- Nedlloyd Newbuilding department has found from experience that almost every newbuilding vessel will have to be provided - once she is in service - with a number of some 20 to 100 reefers slots extra at a later date. For such an exercise the Universal containervessel stands wide open.
- The 'integral reefer' system applied on this vessel in general has much more flexibility than the ducted (Porthole) system. With the latter, the cooling installation is shipborne - which makes the ship more expensive as to investment costs .
  At the other hand, the flexible worldwide applicable 'integral' reefersystem means a higher investment cost in containers.
  'Ducted' systems are more economical where a high-volume reefer trade exists and ships are to remain in the same service for much of their lifetime, so there where flexibility counts little.

General arrangement. (Fuel and waterballast location).

- As has already been explained, location of the engineroom and deckhouse is 'semi-aft' with the fuel oil storage 'semi-forward'. As high-viscosity fuel oil is used, there exists the problem of pumping the fuel from tanks forward to the engineroom aft, when needed.
  For this reasons, the fueloil pipelines run through pipe tunnels located in the Port and Starboard corners of holds, between sidewall and tanktop. The lines can be heated by means of steam lines lying next to the fuel lines and having been wrapped around by insulating material.
- The weight of the consumed fuel can be compensated by taking waterballast in doublebottom and side tanks in the vicinity of the fuel oil tanks.
  Waterballast tanks in the wings of the upper part of the foreship will make it possible to compensate for trim astern without increasing the transverse stability of the vessel too much.
  (On many a containervessel the stability increases too excessively when - in certain conditions - forepeak and deeptank Nr. 1 are used for this purpose.)

Engine installation philosophy.

- Maintenance costs are directly related to the amount of bolts and nuts utilized and for this reason a slow-turning 6-cylinder diesel engine, longitudinal scavenging type, has been installed together with a shaft-generator, thyristor-controlled.
- Maximum reliability has been strived after in utmost simplicity.
- The aim is not to use separate diesel-engine generators at sea. However, during manoeuvring with the bow and sternthruster at work or standing-by, all three diesel-engined generators should be running.
- It is the opinion of Nedlloyd Newbuilding department that it is not a paying proposition to have an exhaustgas turbo-generator installed on a containervessel as such a ship has to serve many ports (with short sea-stretches in-between) and has often to sail many coastal stretches at reduced speed.
  (In this respect we are in favour, however, of an exhaustgas generator installation on a tanker and bulkcarrier which ships - normally - sail on long stretches and at full power.)

- Centralized control has been adopted in the wheelhouse and is arranged in such a way that the nautical duty officer is not disturbed in the exercise of his duty by less-urgent engineroom alarms. During critical periods, the Chief Engineer is (to be) in the wheelhouse.
- The engineroom control station is minimized in layout and is situated near the workshop where engineers do their maintenance jobs. The space is air-conditioned and kept at low noise level.

Mooring arrangement.

- All wires are to be fed directly from the large-diameter winch-drums via fairleads of the large-radius solid cast steel Panama type towards the quay bollards.
- Four (4) mooring winches forward and four (4) mooring winches aft are to be handled by two men forward and two men aft. Bow- and sternthrusters are used in combination with the mooring cables. The vessel can in principle move transversely on its own power and could press herself against the quaywall whilst meantime the wires can be made fast'. After having been made fast, the winches are set to 'self-tensioning'.

  Two ways of mooring are feasible, i.e.:-

  Alternative (1)... 3 wires plus 1 spring, forward as well as aft.

  Alternative (2)... 2 wires plus 2 springs, forward and aft and remotely controlled from bulwark.

  The steel wires system is adopted because of low maintenance load on the wires and on the fixed fairlead-chocks.
- The anchor type is of Spek design. It is flush mounted in an anchor pocket and always fits well when 'coming in'. It never slams by the action of headseas.

Deck layout in view of maintenance .

... as simple as possible...

Smooth flat coamings and closed box-foundations. Whatever labour-requiring features could be omitted should be omitted (i.e. no frills and freaks - geen overbodige Toeters en Bellen)

Grit-blasting of weather-exposed surfaces after finishing construction and application of zinc-epoxy plus chlorinated rubber paints is a 'must' for the shipowner if he intends to sail with a minimum number of men (18) .

(Design background)

Deck layout in view of maintenance.

- As simple as possible...

Smooth flat coamings and closed box-foundations. Whatever labour-requiring can be omitted should be omitted .

Grit-blasting of weather-exposed surfaces after finishing construction and application of zinc-epoxy plus chlorinated rubber paints is a 'must' for the shipowner if he intends to sail with a minimum number of crew (18).

Double bottom waterballast tanks. Preparation.

Grit-blasting of welds and sweep-blasting of the remaining surfaces has been carried out in the block-stage and

2 coats of solvent type coaltar epoxy paint, i.e. 2 x 175 micron layers were applied.

After our experience of troubles recently met, Nedlloyd nowadays is in favour of tank conservation in the final stage to be carried out as follows:-

- Application of 1 coat of 350 micron thickness of non-solvent coaltar-epoxy type paint by means of high-pressure spraying apparatuses.

This system saves manhours, has no intercoat adhesion problems and does not create an explosion hazard.

Underwater.

Application of self-eroding (SPC of International) acrylic resin copolymer pigmented with organic tin antifouling paint, 3 coats each 100 micron. This has proven to keep the vessel free from alg and barnacles.

SPECIAL FEATURES OF THE SECOND VESSEL.

- The second vessel, to be named Nedlloyd Clement, (Yard nr. 226) is intended for the EUROSAL service, carrying general cargo in containers Southbound from western Europe via the Panama Canal to Col mbia, Equador, Peru and Chile.
- Northbound, the vessel has to transport copper and fruit from Chile and coffee from Col mbia - and eventually bananas from Equador.
- At this moment, only the Equadorian port of Quayaquil is fitted with a container gantry crane. Probably San Antonio in Chile will have a container shorecrane at the beginning of 1984.

Because of this situation it was decided by the EUROSAL principals to provide each containervessel with a shipboard container gantry crane.
Should - in future - all ports be equipped with cranes, then the shipboard cranes could be
It is an advantage of the gantry system that ships can be geared or de-geared within say - which is an interesting feature with a view to interchangeability between services.
At the other hand, the reliability of

Copper ingots to be carried from Chile.

- This type of cargo will give a lot of problems on the Eurosal cellular containervessels.
- Logically, carriage of this cargo should also be done in containers.

However, extra stowage costs and damage feared to be
cargo containers stand in the way of containerizing this cargo as this all will/might add too much to the transportation costs.
Carriage of copper ingots by means of Flatracks ('bedsteads' of 20x8x4 Ft) should technically be the best way out. However, commercial departments involved are afraid for deadheading of the 'special' flatracks as yet.
Nedlloyd's opinion tends towards the utilisation of the flatracks Southbound for drums and steel plates.
Up to this moment (June '83) it is the intention to carry 75% of the copper ingots 'breakbulk' fashion on the tanktopdeck of the ships. The copper ingots are to be loaded onto the ship by means of shore-based 12-tons capacity cranes while bundled in chain slings.
Forklift trucks in the ship must stow the bundles crossover from Portside to Starboard-side between longitudinal bulkheads whilst manoeuvring between the bottom ends of the container cellguides.

(Design background)

Lashing/shoring of the bundles might be necessary and damage to the cell guides is likely to occur

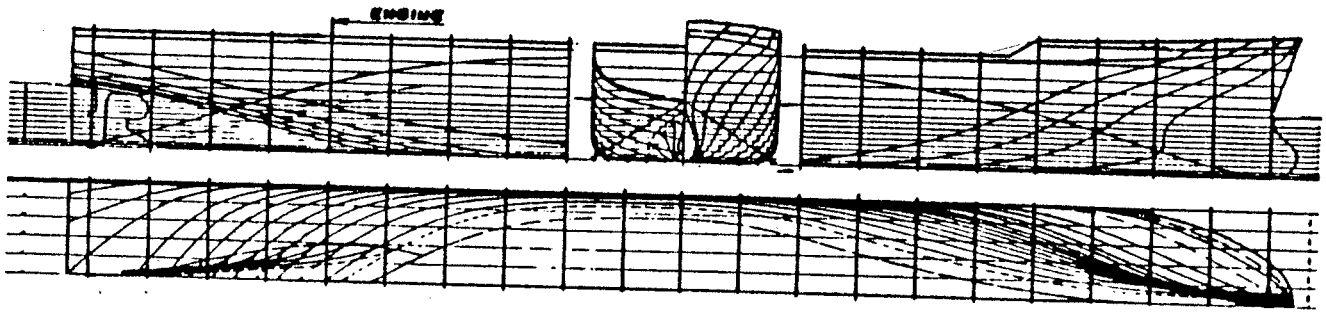
- Acceptance of copper in bundles on a container terminal in Europe is a (near) impossibility and a special copper receiving terminal has to be found nearby.

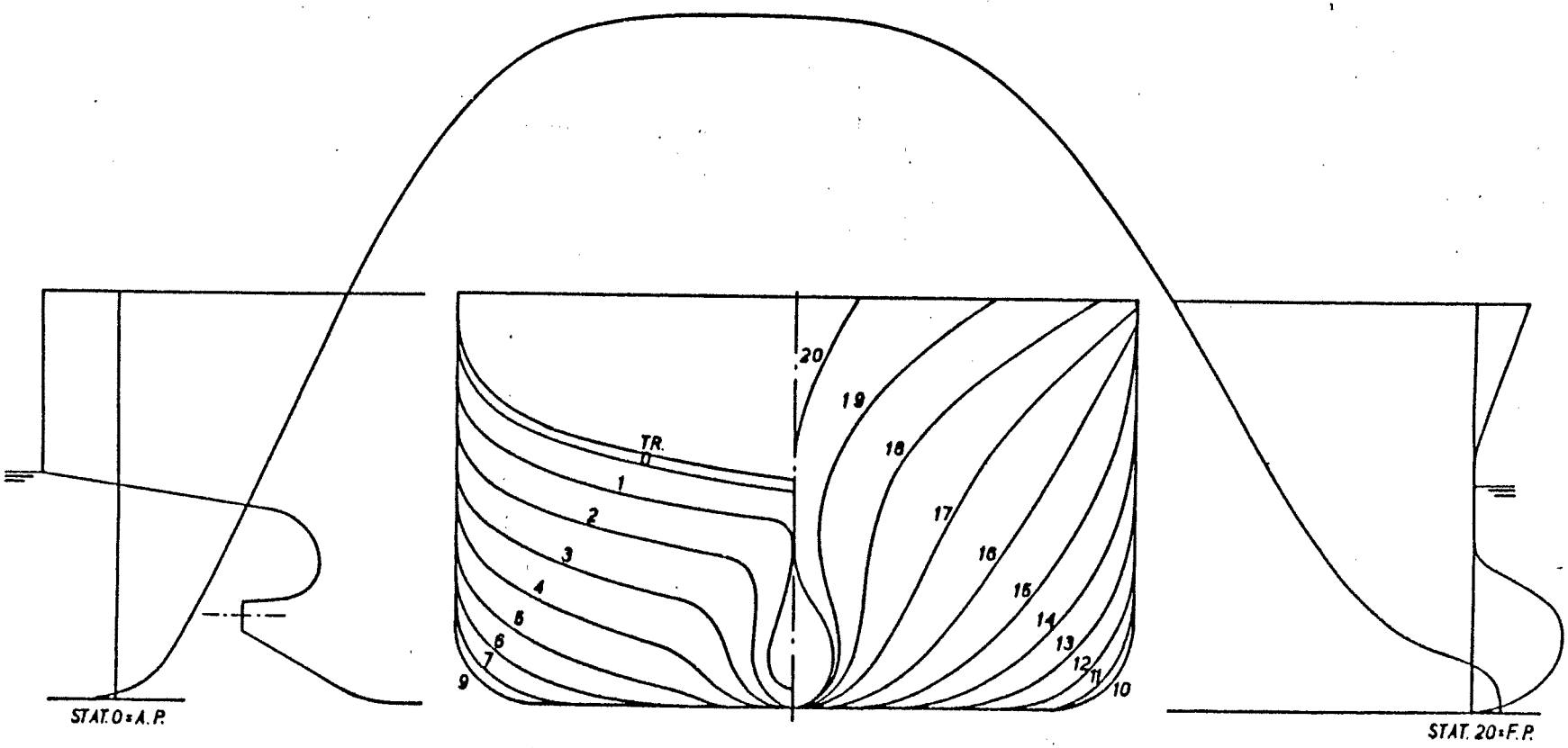
Reefer cargo.

With a view to the foreseeable carriage of bananas from Equador and of other fruit kinds from Chile in 40-ft insulated porthole-type boxes, the ship has been provided with a ducted cooling air system as well. Freon 22 will be used as activator through its direct expansion.

Possibly these well ventilated containers can be utilized also for the carriage of coffee beans.

SHIP MODEL No. 5901 C BULB IV

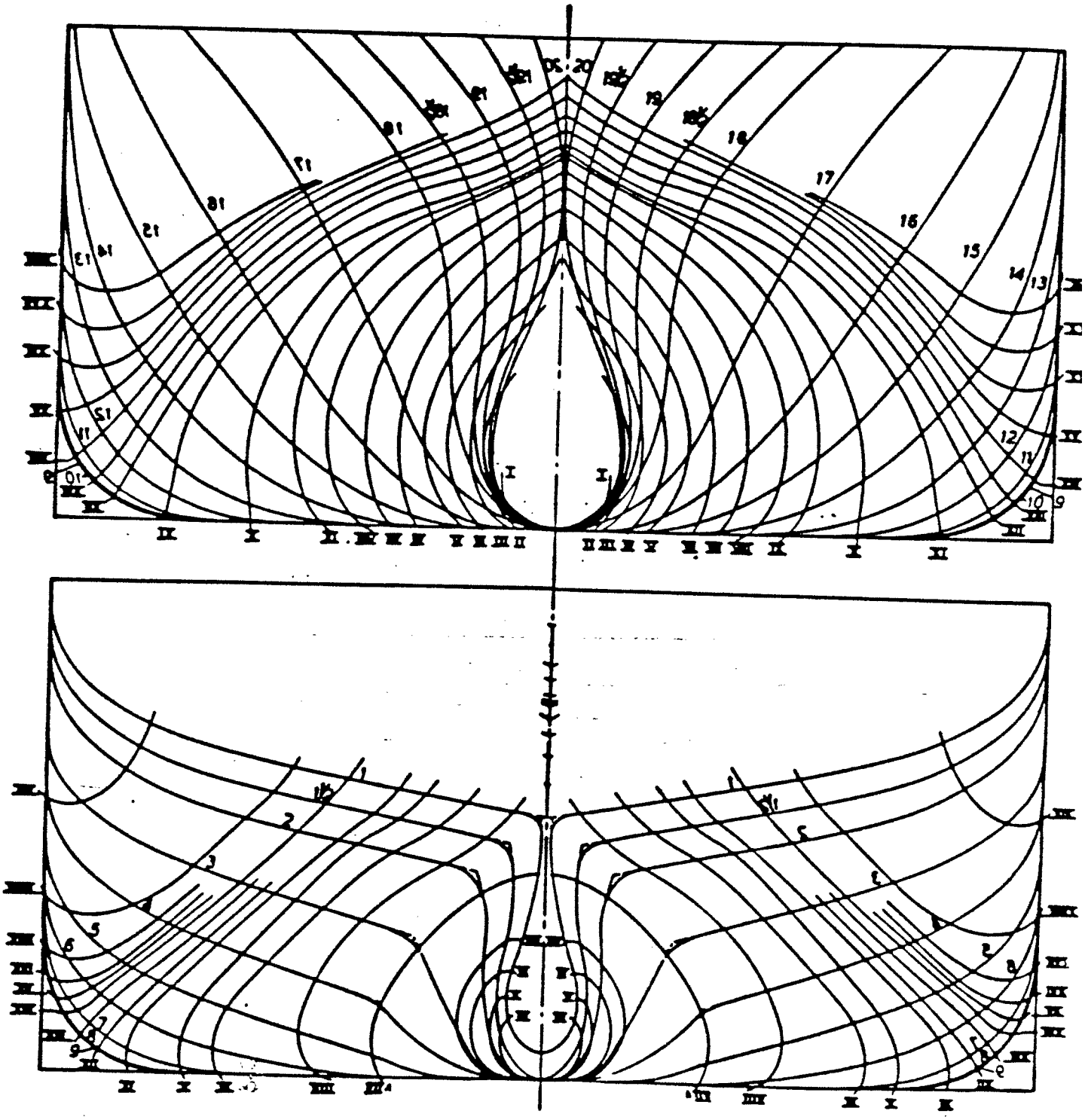




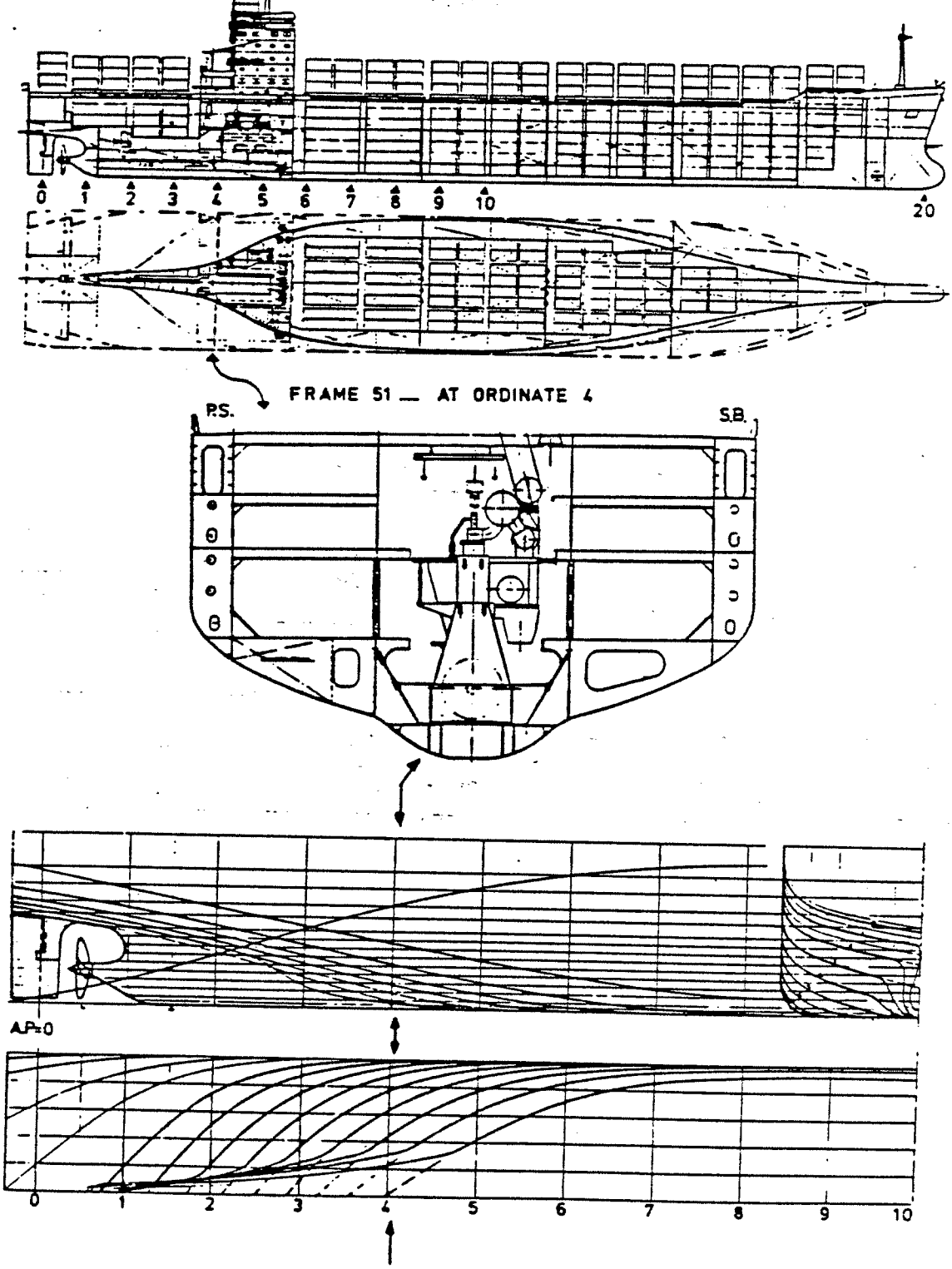
DRAWING No. M 5901-36

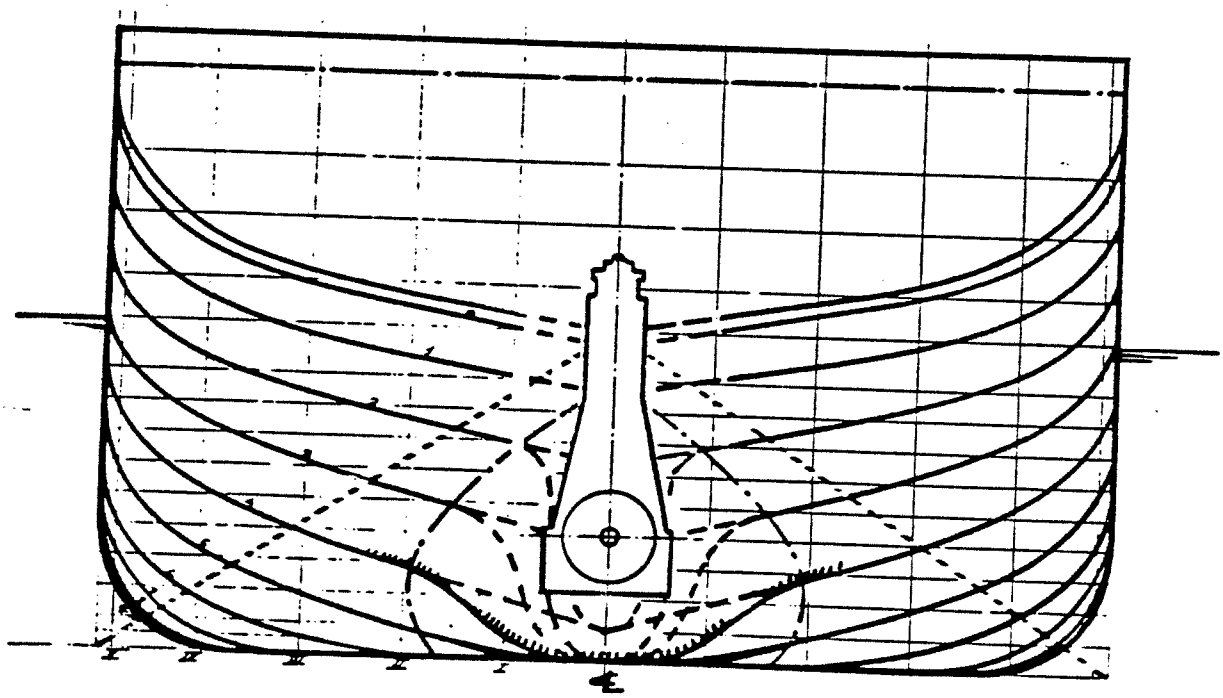
SHIP MODEL Nº 5901 ("UNCO" PROJECT)
STREAMLINES.

DRAFT 10.60 m SPEED 19 KNOTS TEST Nº 39891



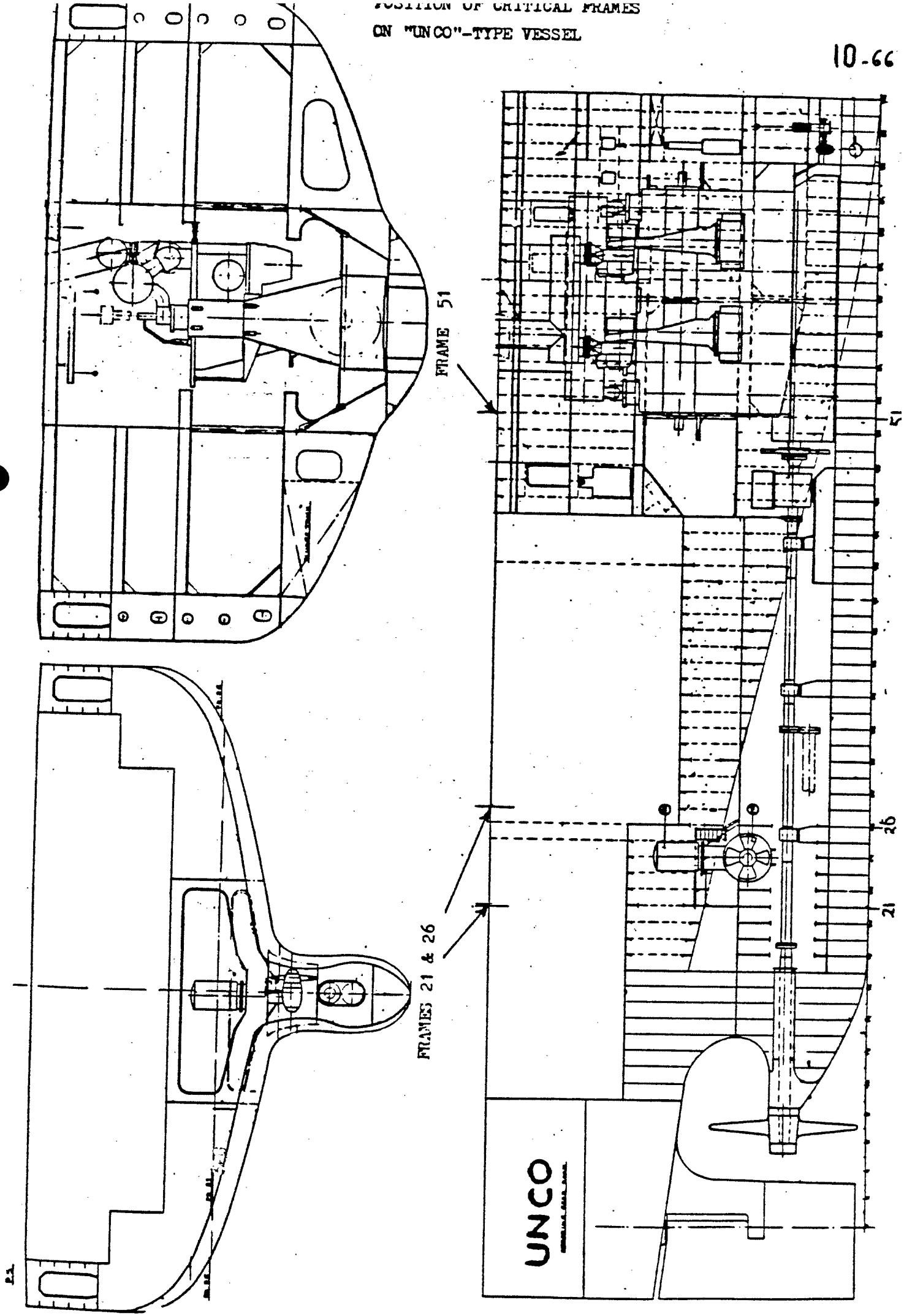
DESIGN 1981 Cellular containership "UNCO"

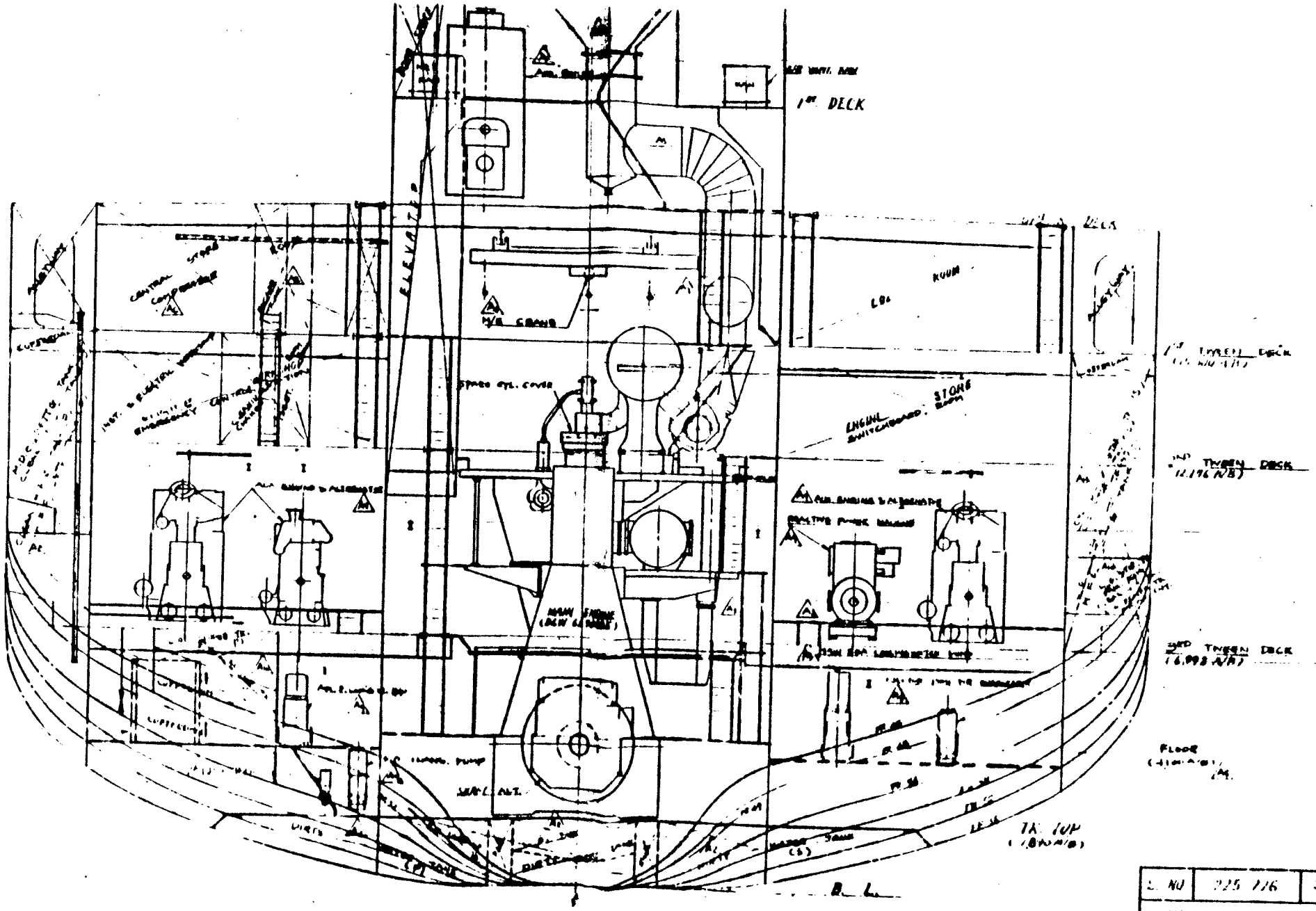




POSITION OF CRITICAL FRAMES
ON "UNCO"-TYPE VESSEL

10-66





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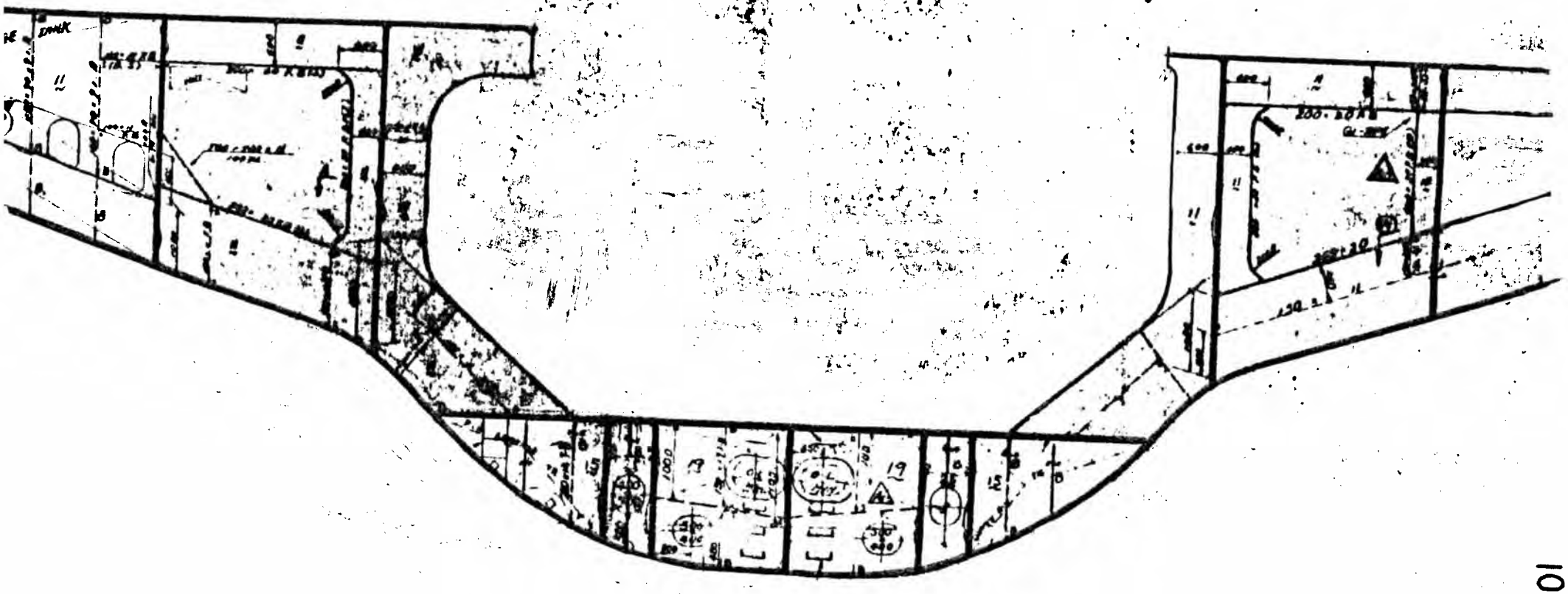
FR 45 SECTION
(LOOKING FORE)

10-67

UNCO
FR. 50

N. - CLARENCE
N. - CLEMENT

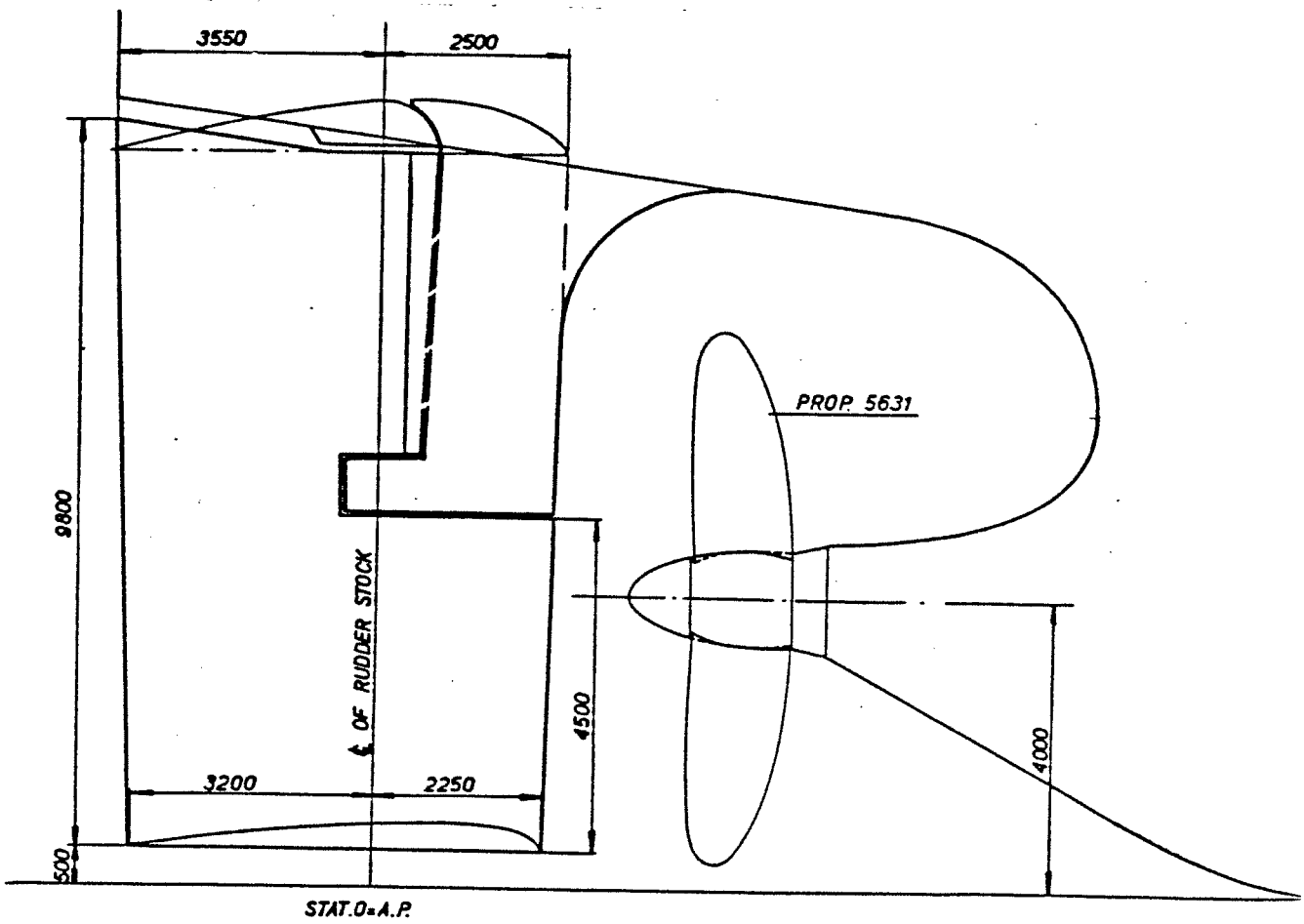
HYUNDAI



10-68

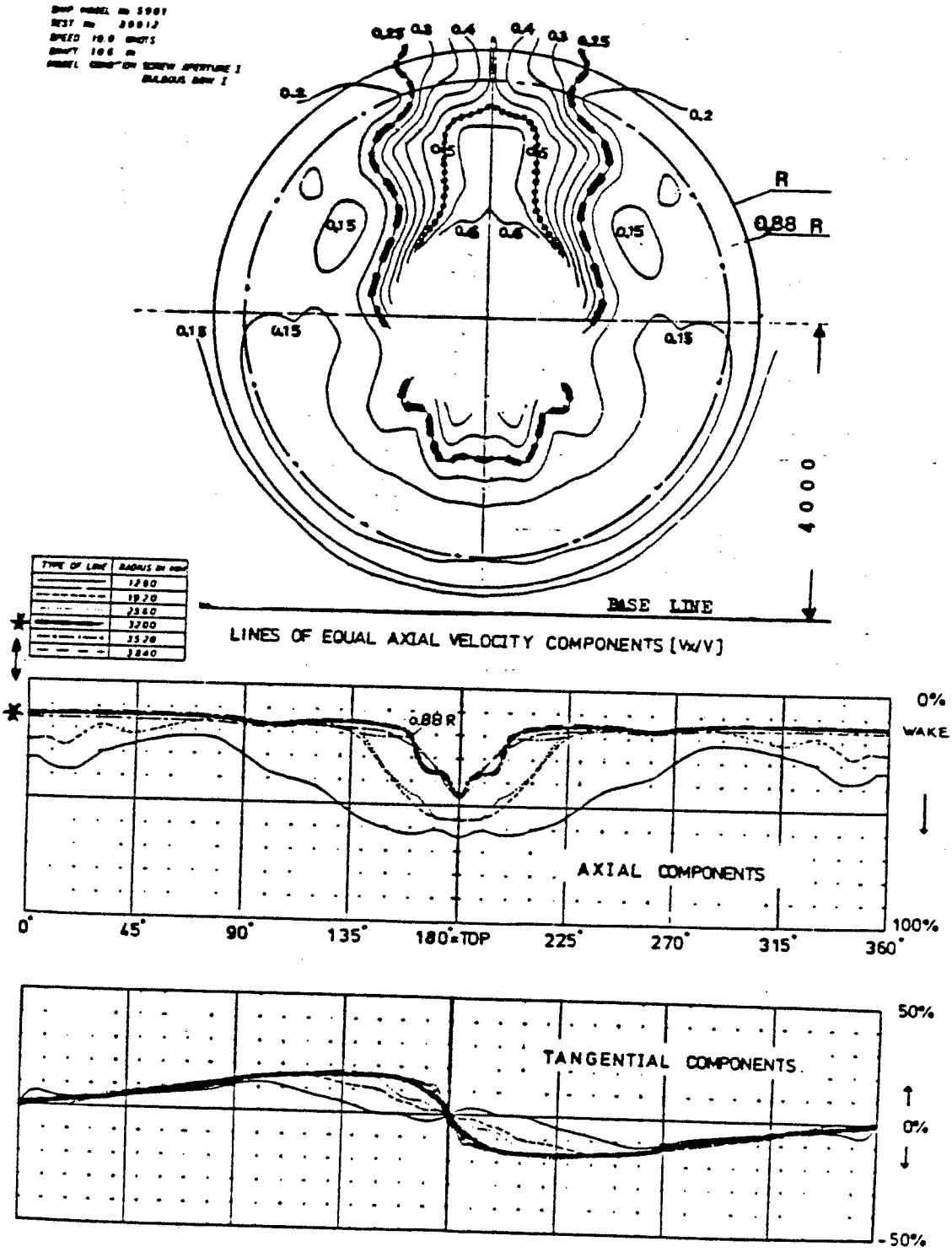
SHIP MODEL No. 5901 C

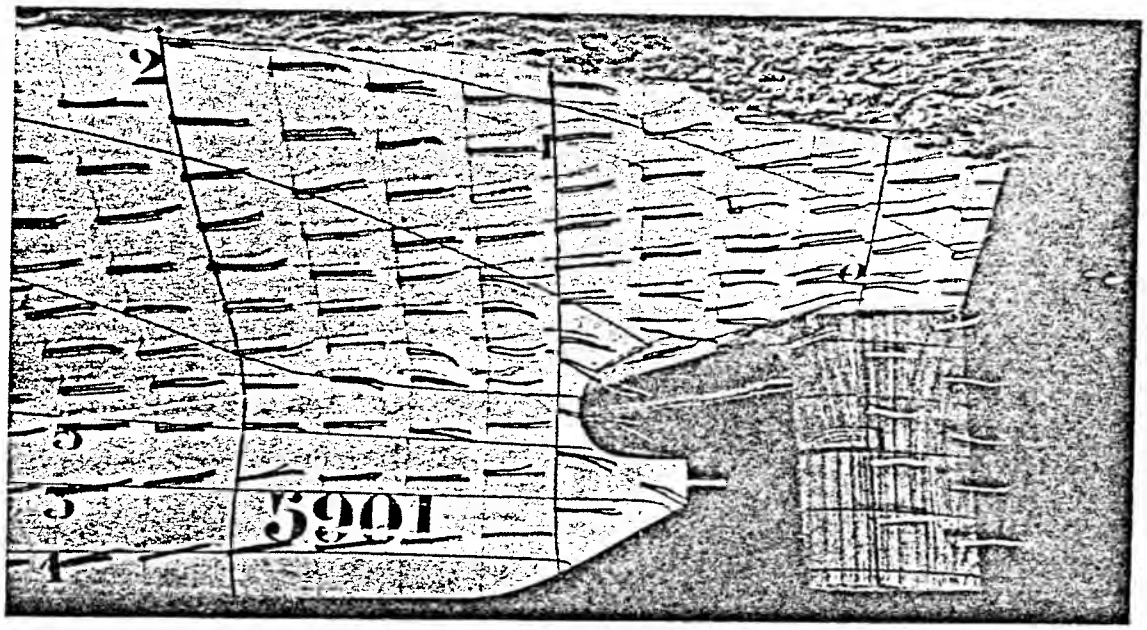
RUDDER I AND SCREW APERTURE I



DRAWING No. M 5901-52

WAKE FIELD MEASUREMENTS
"Universal" containervessel ("UNCO"), AT 10,60 m DRAFT.





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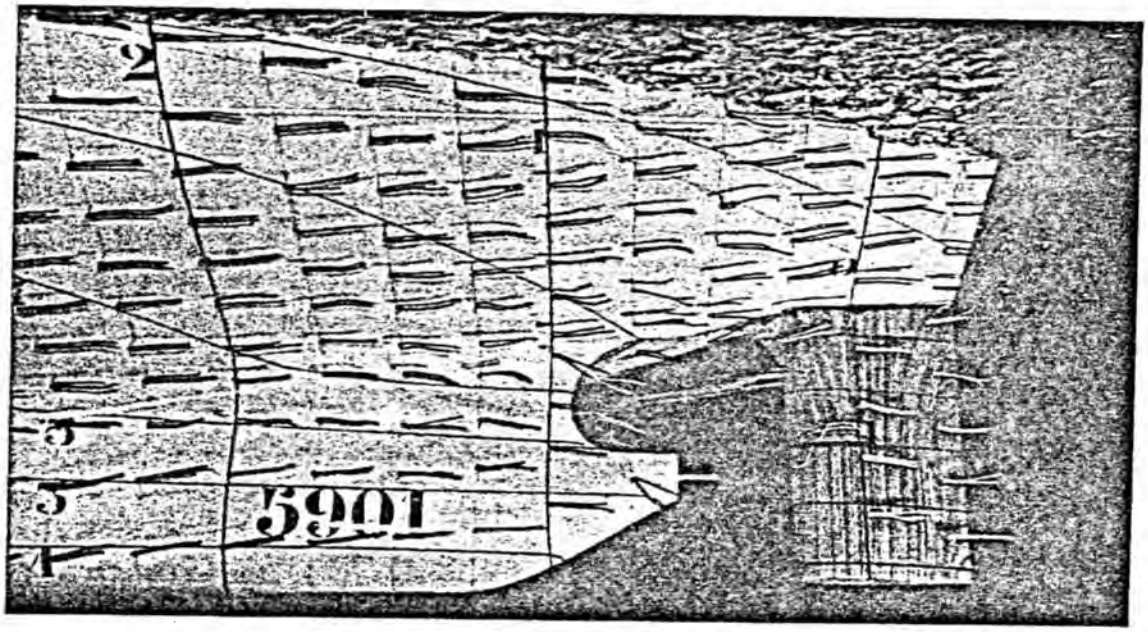

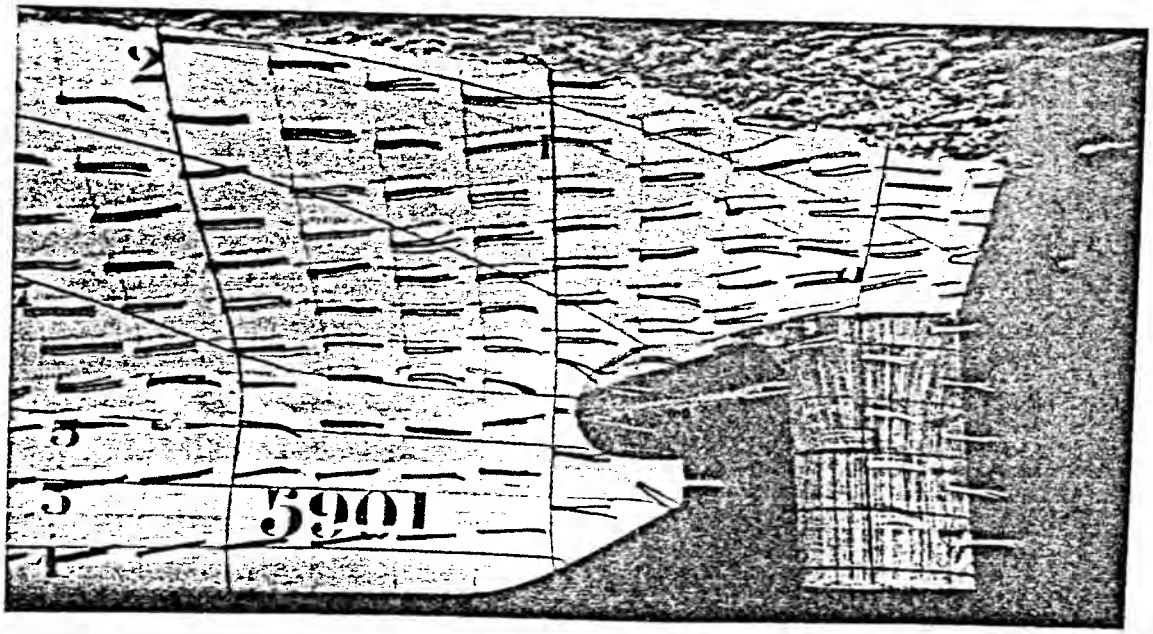

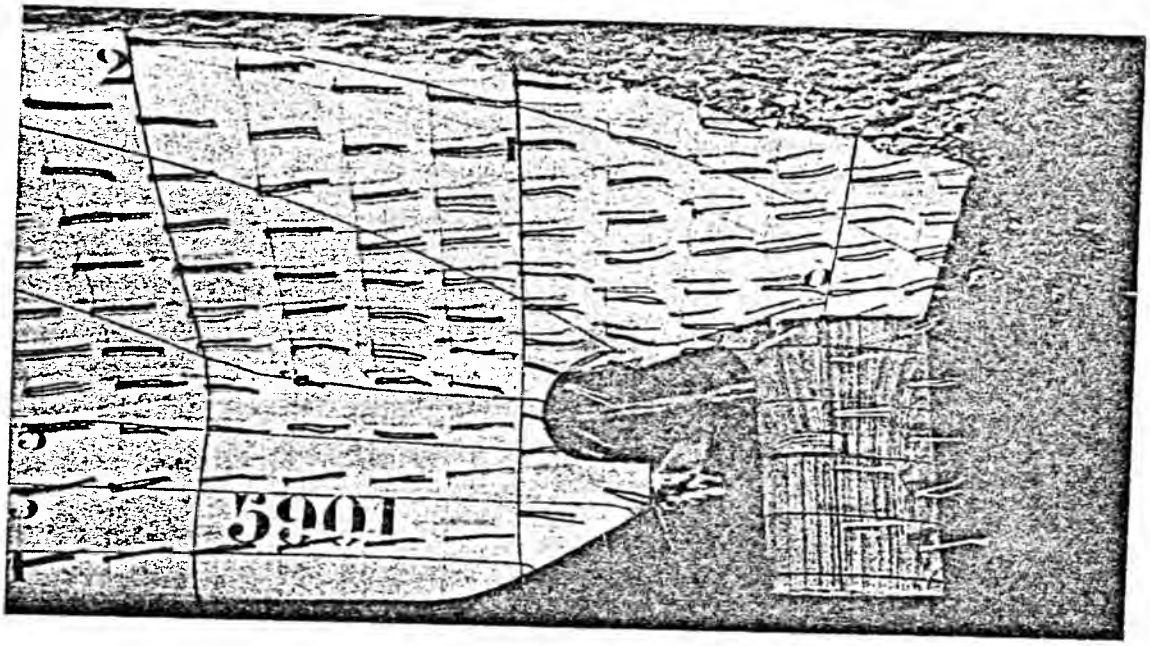

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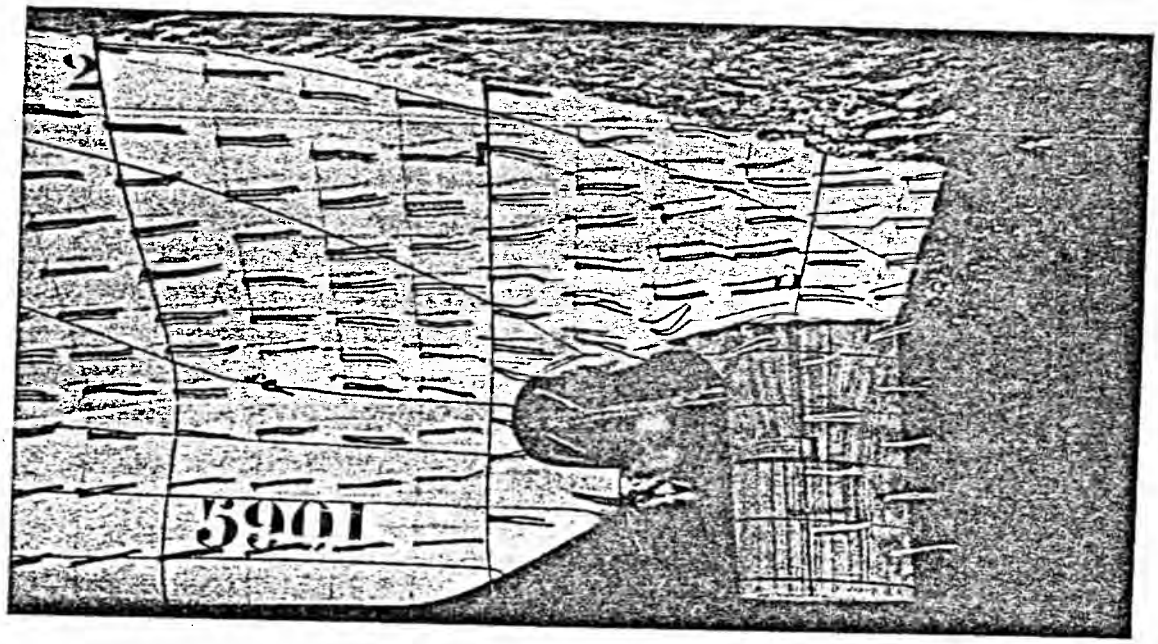
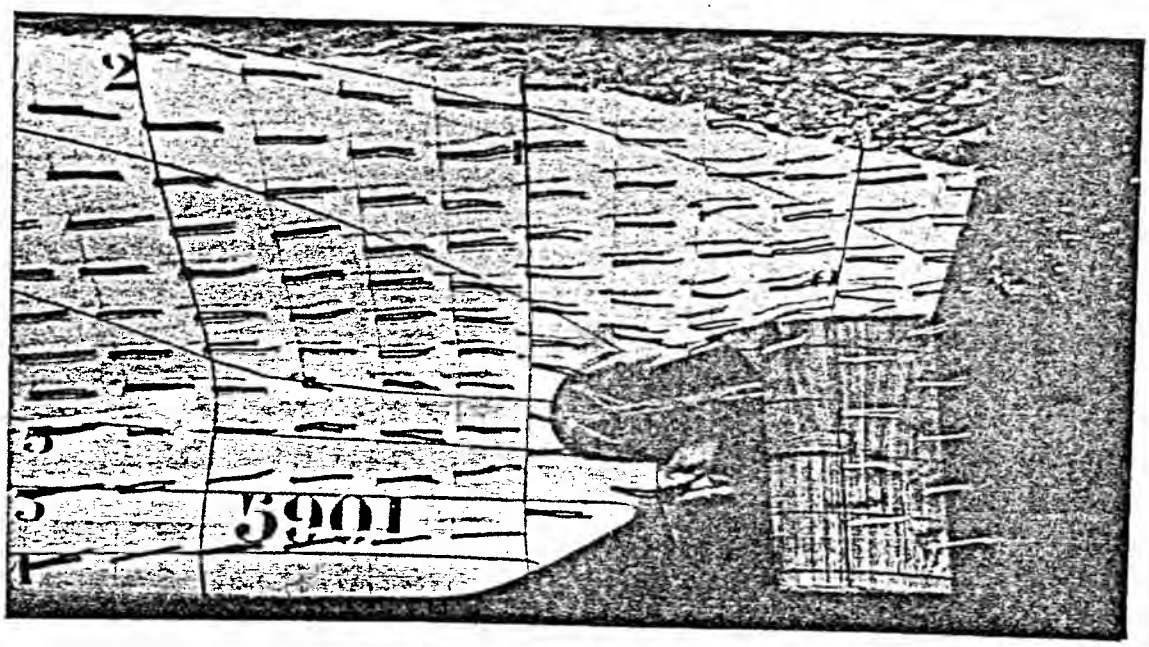


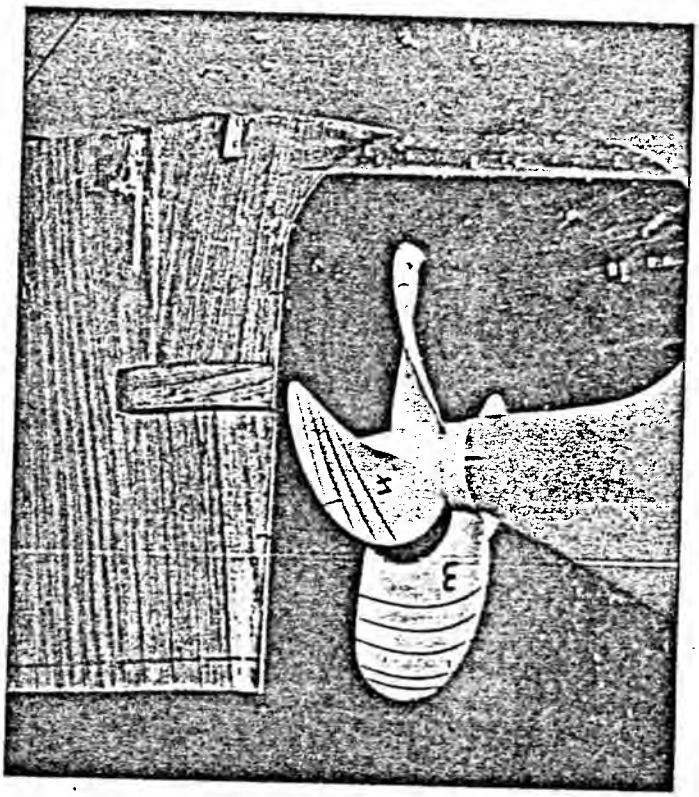


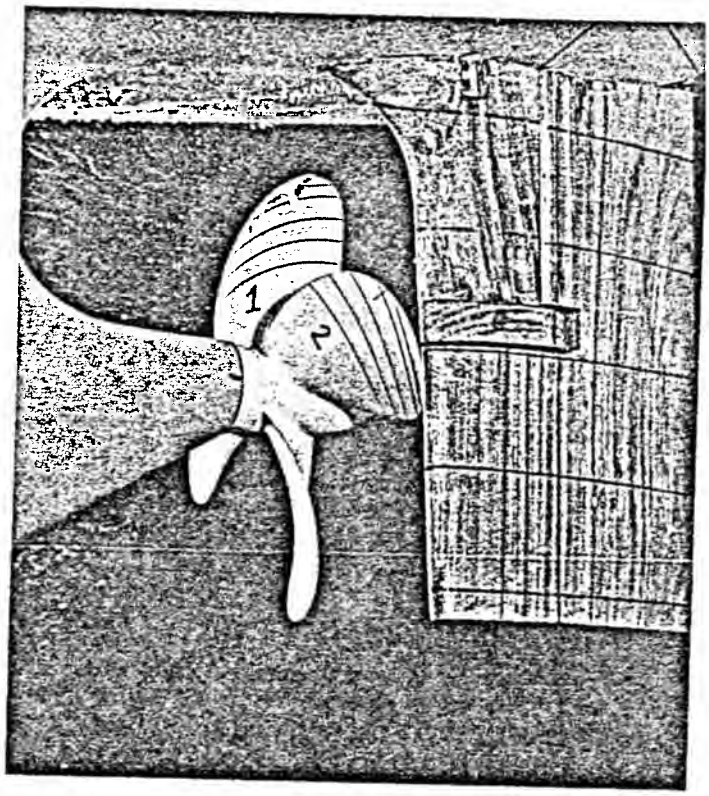
CAVITATION OBSERVATIONS

TEST No. 82-110
SHIP MODEL No. 5901C
PROPELLER MODEL No. 5631
SHIP DRAUGHT ON F.P. 10.60 m
SHIP DRAUGHT ON A.P. 10.60 m

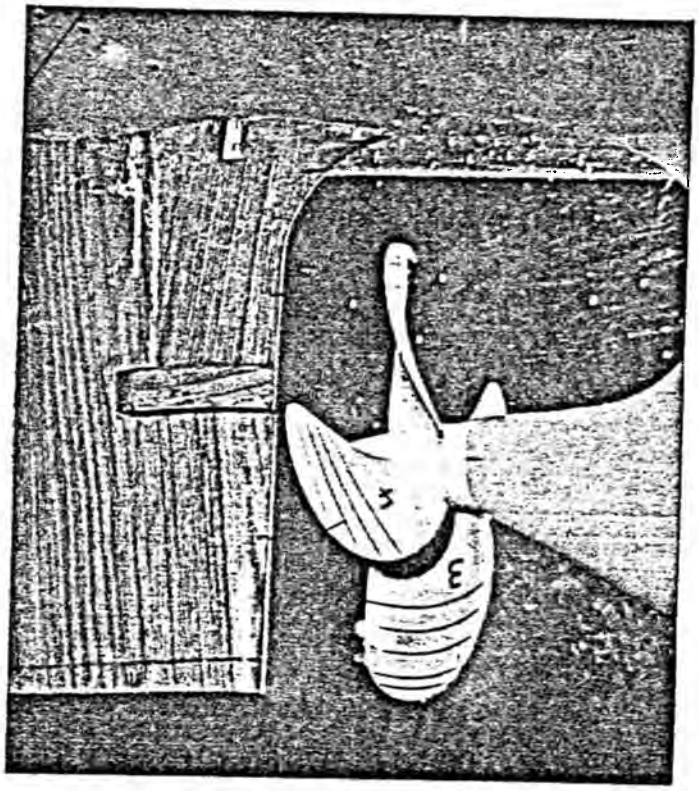
V_s = 18.80 KNOTS
N_s = 1.583 Hz
P_S = 13713 kW

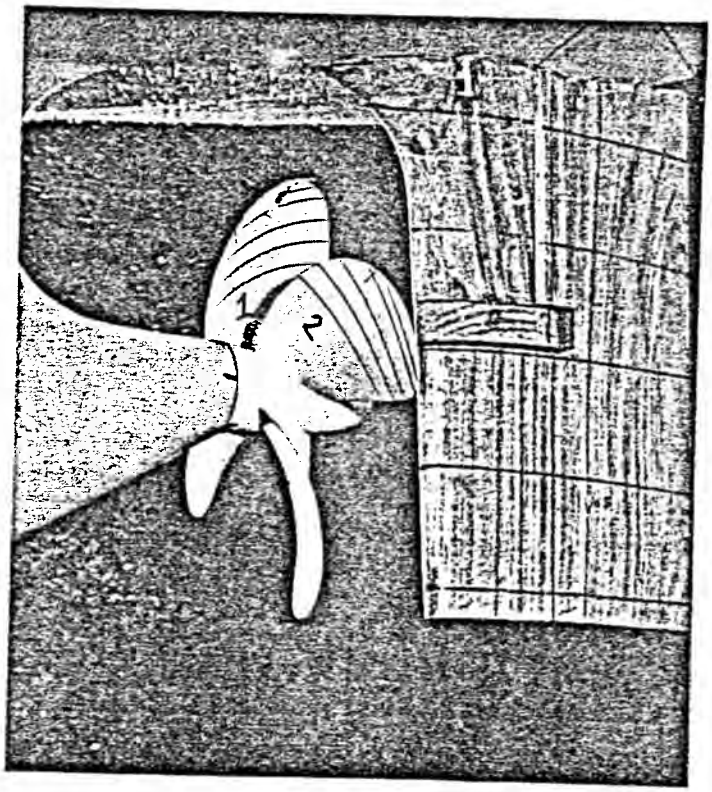
10-73 A





18°





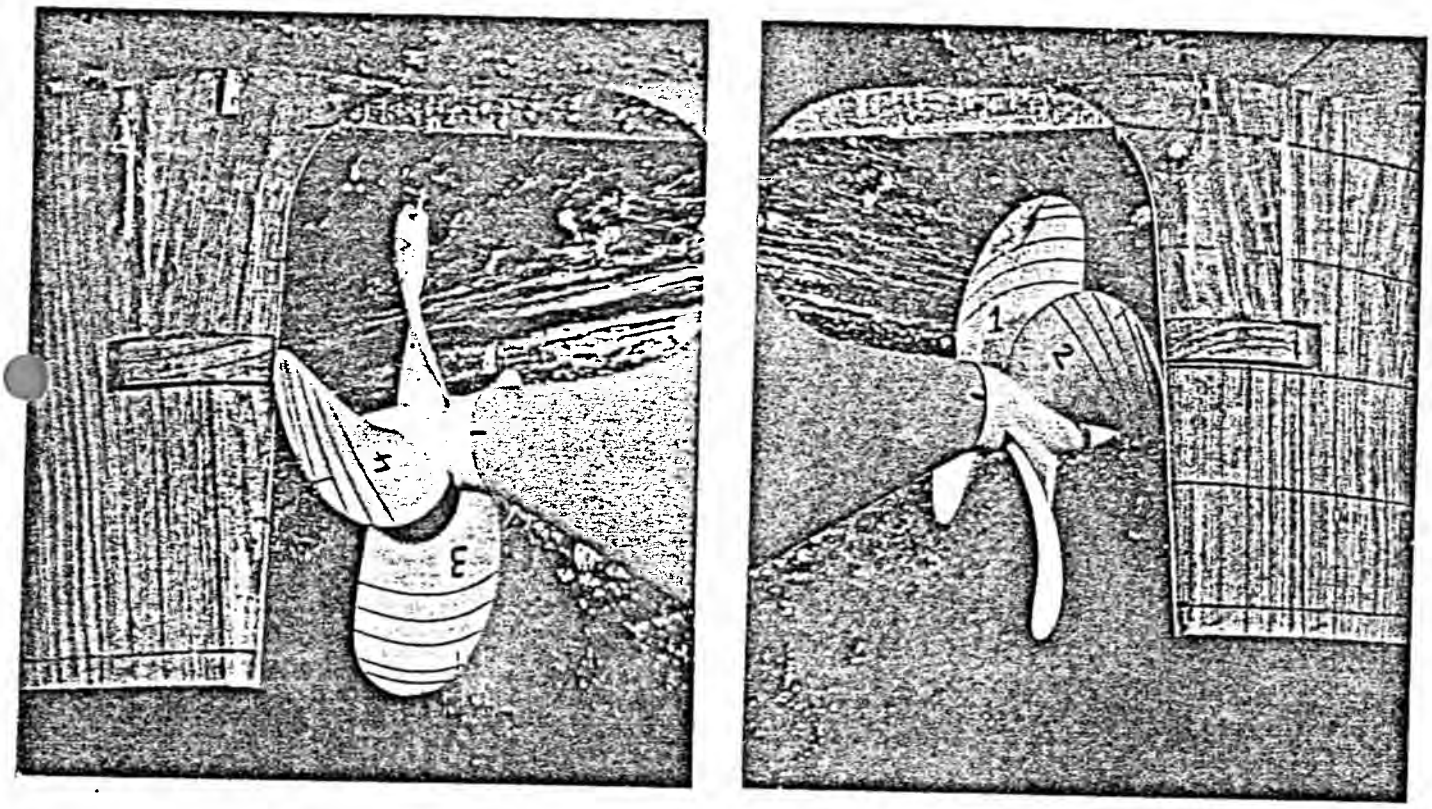
24°

FIG. 2.3. SUCTION-SIDE CAVITATION OBSERVATIONS
ANGLES INDICATE POSITION OF BLADE No. 1 RELATIVE TO TOP POSITION

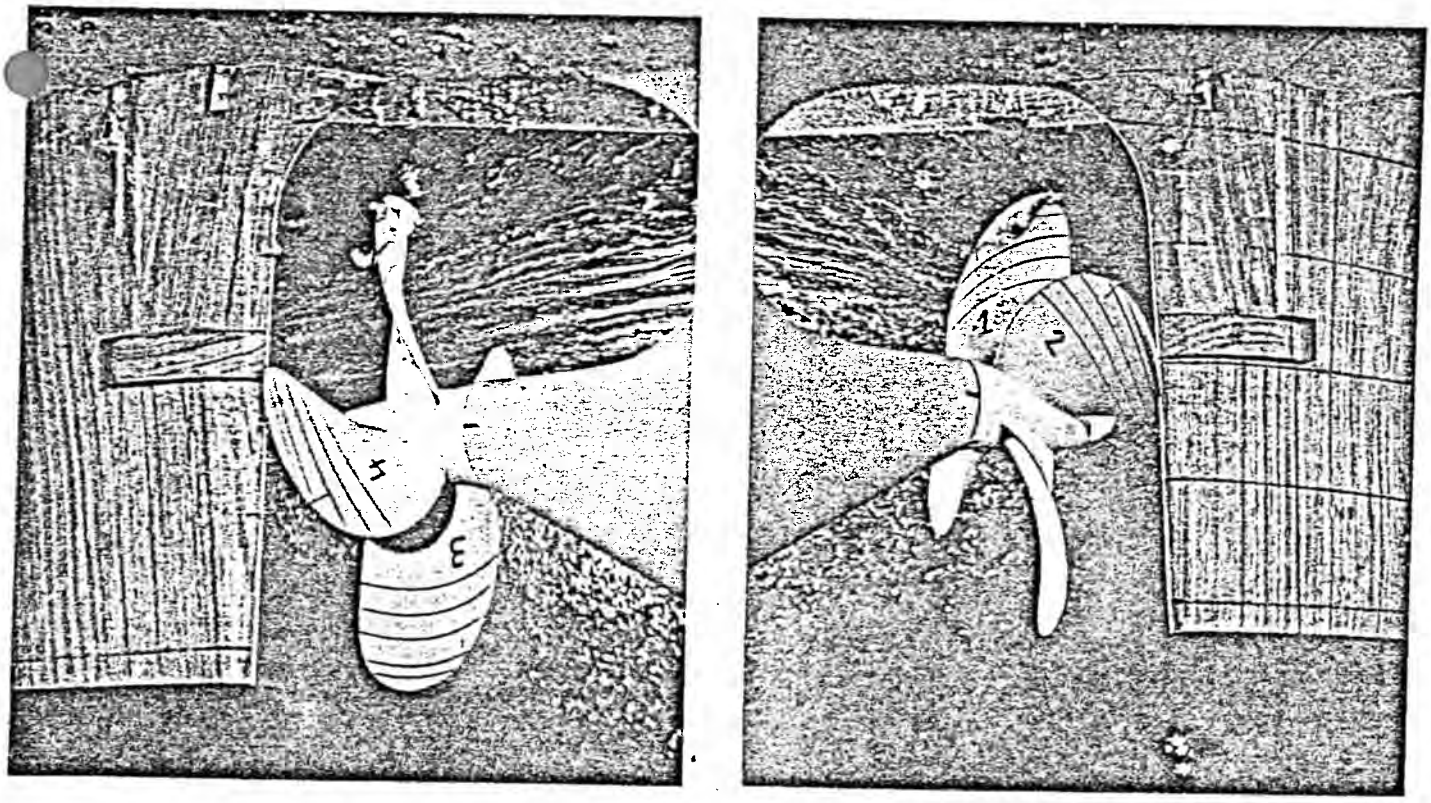
CAVITATION OBSERVATIONS

TEST No. 82-107
SHIP MODEL No. 5901C
PROPELLER MODEL No. 5631
SHIP DRAUGHT ON F.P. 6.40 m
SHIP DRAUGHT ON A.P. 7.60 m

V_s = 20.32 KNOTS
N_s = 1.575 Hz
P_S = 13713 kW



18°



24°

FIG. 2.7. SUCTION-SIDE CAVITATION OBSERVATIONS.
ANGLES INDICATE POSITION OF BLADE No. 1 RELATIVE TO TOP POSITION

WORKING PROPELLER

BASE : ENGINE MCR x 0,85 = 20 230 HP
~ 900 KW - SHAFT GEN - - 1 500 HP
(N.B ESTIM 600 KW - (1000 HP) MEAN) 18730 HP
SHAFTLINE FRICTION - 150
SHP PROPELLER = 18580 HP
10,6 M - V = 19,6 KN
TRIAL CONDITION

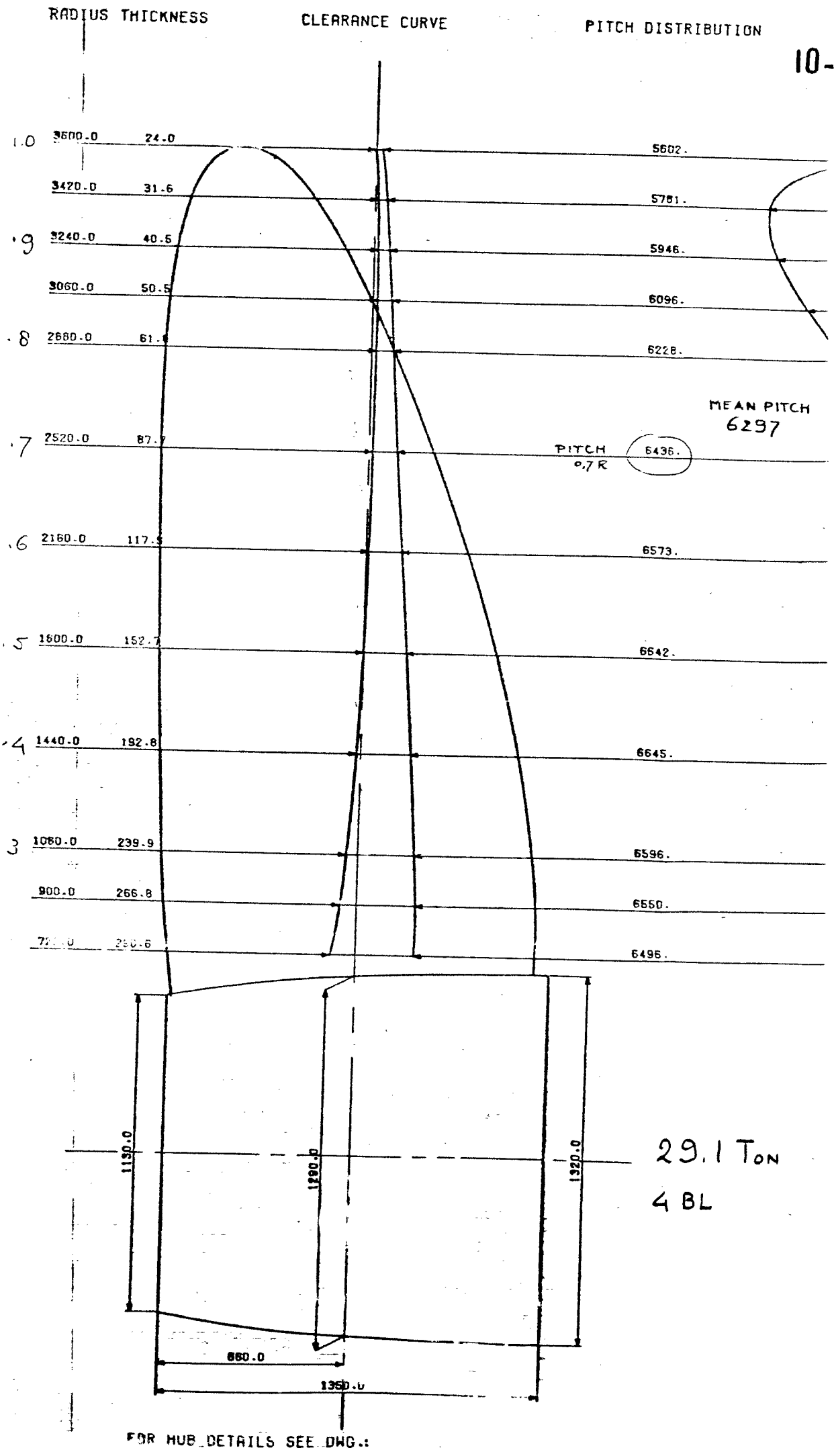
PROPELLER DATA

DIAMETER		7200 MM
PITCH, MEAN	= NSMB MODEL PROP N: 5631 A	→ MEAN = 6297 MM
PITCH 0.7 R		0.7 R = 6436 MM
NUMBER OF BLADES		4
DIRECTION OF ROTATION		RIGHT-HANDED
EXPANDED BLADE AREA RATIO		0.630
MATERIAL		CUNIAL
MINIMUM TENSILE STRENGTH		647 N/MM²
MINIMUM ELONGATION		20 %
MASS	RST	29100 KG
MASS MOM. OF INERTIA PROP. IN AIR	RST	69260 KGM²
MASS MOM. OF INERTIA OF ENTR. WATER	RST	28425 KGM²
ENGINE OUTPUT		17500 KW
PROPELLER REVOLUTIONS		97.00 RPM
E FACTOR 0.25		1.0000
E FACTOR 0.6		1.2500
CLASSIFICATION		LLOYDS REG.

PITCH SELECTED FOR 13775 KW 97.00 RPM, SERVICE CONDITION.

BLADE FINISHING TOLERANCES IN ACCORDANCE WITH ISO CLASS I

INITIALS	THE DESIGN SHOWN ON THIS DRAWING IS THE ABSOLUTE PROPERTY OF -LIPS PROPELLER WORKS- IT MAY NOT BE COPIED OR REPRODUCED. IT MAY NOT BE EXECUTED BY, USED FOR OR SHOWN TO THIRD PARTIES WITHOUT THE WRITTEN CONSENT OF THE OWNER.	F.P.PROPELLER FOR NB NOS 225-226 UNCO		
DATE		CLIENT HYUNDAI HEAVY INDUSTRIES ORDER F7846-848		
DESCRIPTION	HYDRODYNAMIC DESIGN DEPARTMENT	DRAWN		DATE Y / M / D 82/03/18
		CHECKED		
		APPROVED		SCALE 1:12.5
MODIFICATION	LIPS PROPELLER WORKS DRUNEN, THE NETHERLANDS	DRAWING F7846-01	MODIFICATION	SIZE A1


RADIUS THICKNESS
CLEARANCE CURVE
PITCH DISTRIBUTION
1.0 3600.0 24.0 5602.
3420.0 31.6 5781.
.9 3240.0 40.5 5946.
3060.0 50.5 6096.
.8 2880.0 61.8 6228.
MEAN PITCH
6297
.7 2520.0 87.7
PITCH
0.7 R
6436.
.6 2160.0 117.5 6573.
.5 1800.0 152.7 6642.
.4 1440.0 192.8 6645.
.3 1080.0 239.9 6596.
900.0 266.8 6550.
6496.
1130.0
1320.0
660.0
1360.0
29.1 TON
4 BL
FOR HUB DETAILS SEE DWG.:

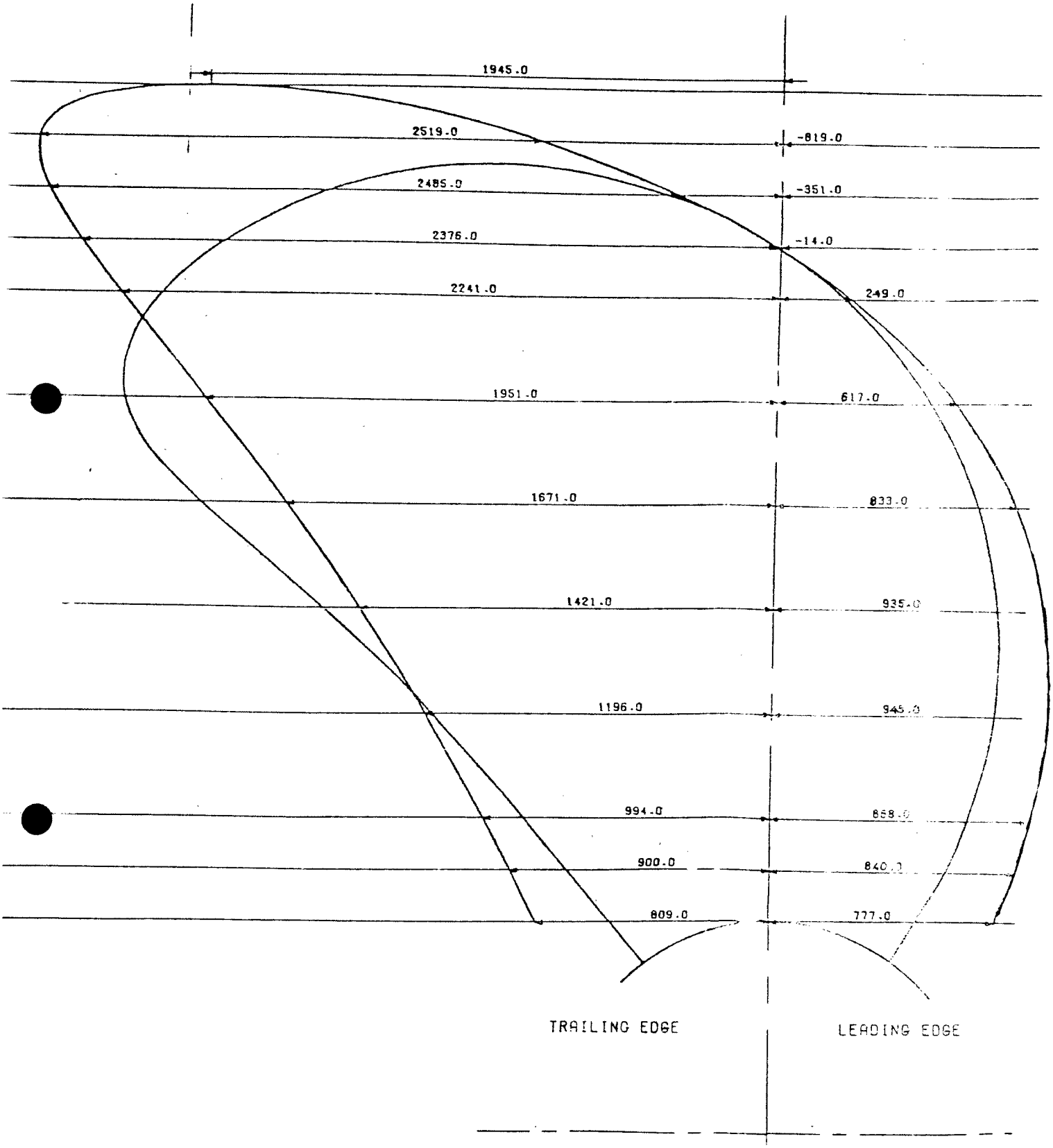

1945.0
2519.0
-819.0
2485.0
-351.0
2376.0
-14.0
2241.0
249.0
1951.0
617.0
1671.0
833.0
1421.0
935.0
1196.0
945.0
994.0
858.0
900.0
840.0
809.0
777.0
TRAILING EDGE
LEADING EDGE

Table I

Particulars of ship

Ship represented by ship model 5901C with bulbous bow IV

Length between perpendiculars	L_{PP}	202.00	202.00	202.00	202.00 m
Length on waterline	L_{WL}	205.57	205.57	192.86	193.96 m
Breadth moulded	B	32.24	32.24	32.24	32.24 m
Draft moulded on A.P.	T_A	10.60	9.15	7.00	6.40 m
Draft moulded on F.P.	T_F	10.60	9.15	7.00	7.60 m TR
Displacement volume mld (without tunnels)	∇	43810	36359	--	-- m^3
Displacement volume mld (with tunnels I)	∇	--	36319	26150	-- m^3
(with tunnels II)	Δ	43774	36323	--	26154
Displacement weight (without tunnels)	Δ	44905	37268	--	-- mt
Displacement weight (with tunnels I)	Δ	--	37227	26804	-- mt
(with tunnels II)	Δ	44868	37231	--	26808
Wetted surface, bare hull	S_1	8139	7331	6241	6241 m^2
Wetted surface, rudder incl.	S_2	8250	7431	6314	6314 m^2
L.C.B. position aft of ½ Lpp in % of Lpp	Θ	1.71	--	--	--

Ship model details:

Ship model material: wood

Linear scale ratio $\lambda = 32$

Turbulence induced by studs

Note:

The presented values valid for draft 6.40/7.60 m are approximately: the trim of the model has been obtained by shifting the ballast weights.

Table II

Characteristics of propellers:

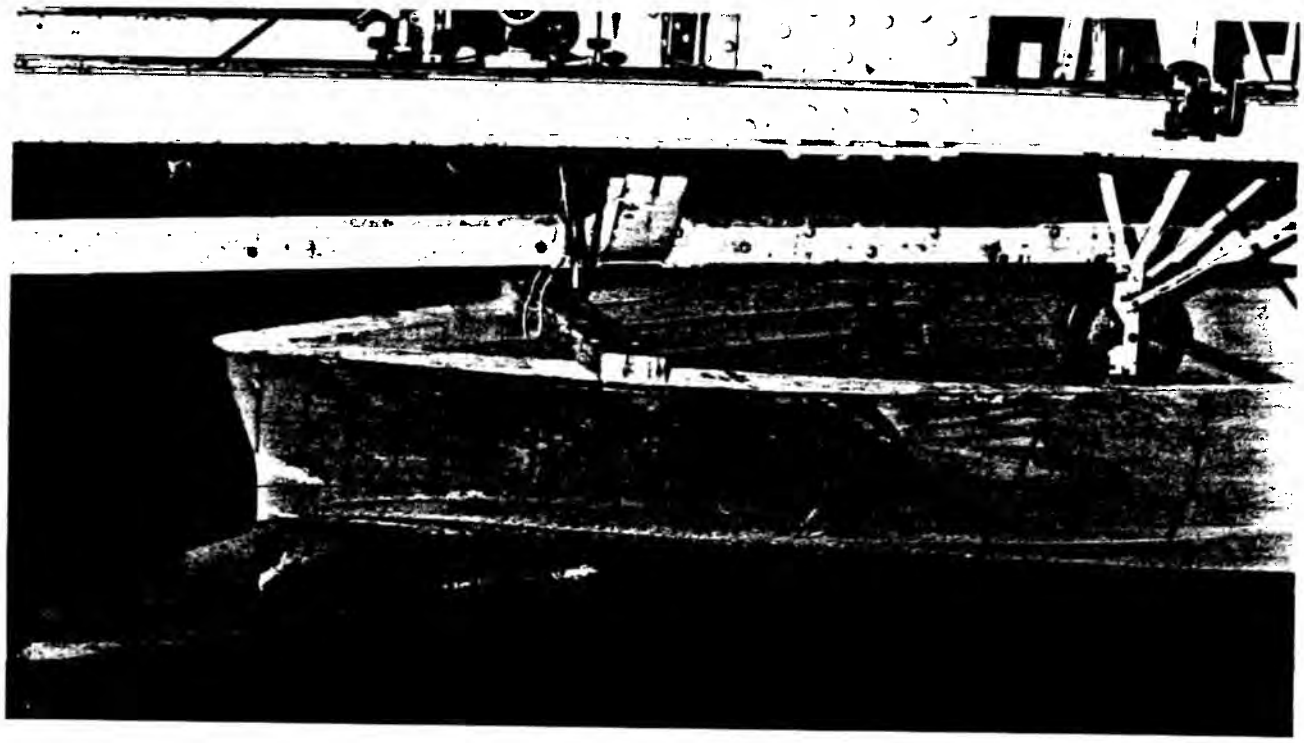
Propeller No.		3754[1)]	5631[2)]	5631A[3)]
Diameter	D	7259	7200	7200
Pitch at 0.7R	$P_{0.7}$	6010	6635	6436
Pitch ratio	$P_{0.7}/D$	0.828	0.922	0.894
Expanded blade area ratio	A_E/A_O	0.568	0.637	0.637
Boss-Diameter ratio	d/D	0.172	0.179	0.179
Number of blades	Z	4	4	4

MEAN PITCH 6297

1) stock propeller
2) final Lips propeller
3) final Lips propeller with corrected pitch for required power - RPM relationship.

SHIP MODEL No. 5901 C

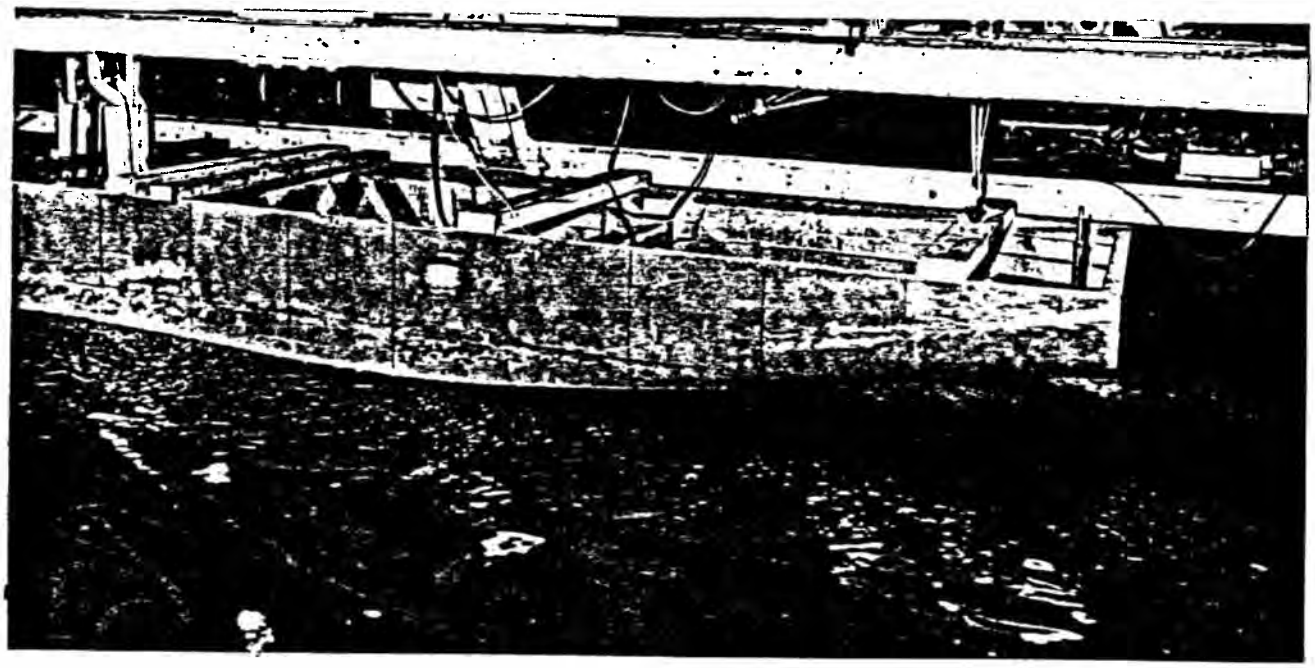
BULBOUS BOW IV (BALLAST BULB)



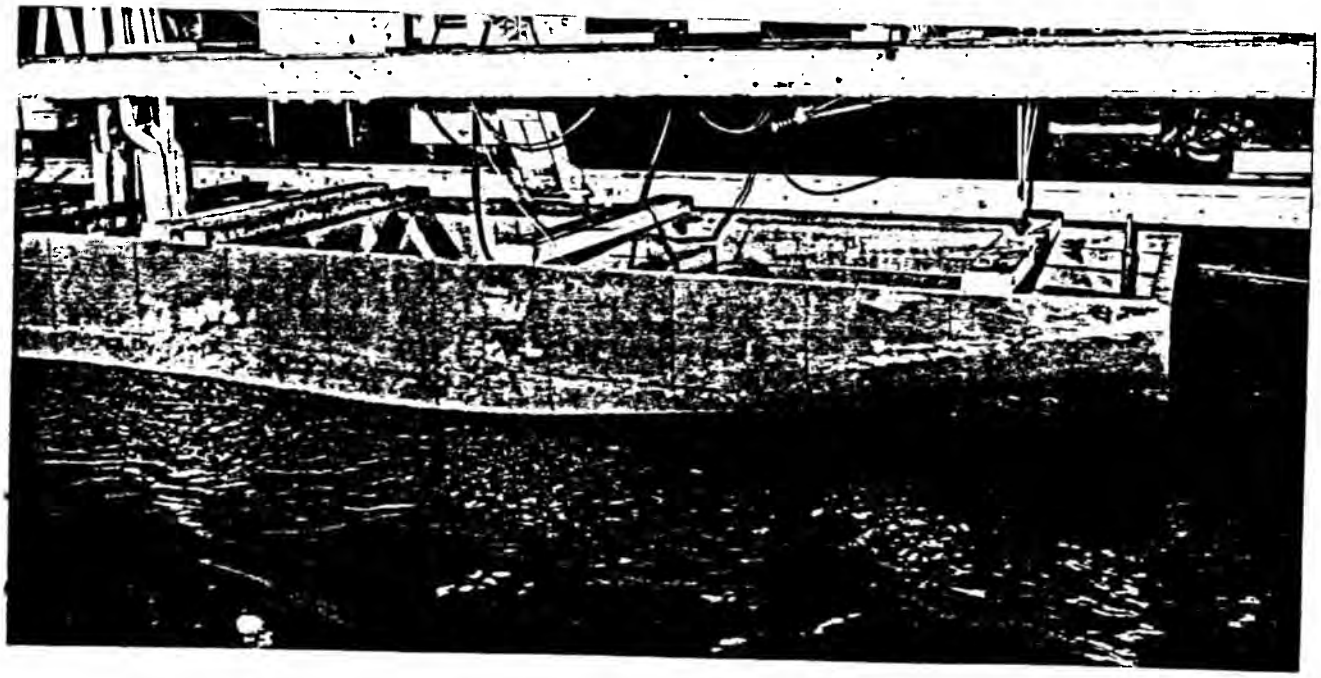
DRAFT 7.00 m SPEED 20.5 KNOTS TEST No. 40957

DRAFT 6.40 7.60m SPEED 20.5 KNOTS TEST No. 40956

DRAFT 7.0 m SPEED 20.0 KNOTS TEST No. 40908

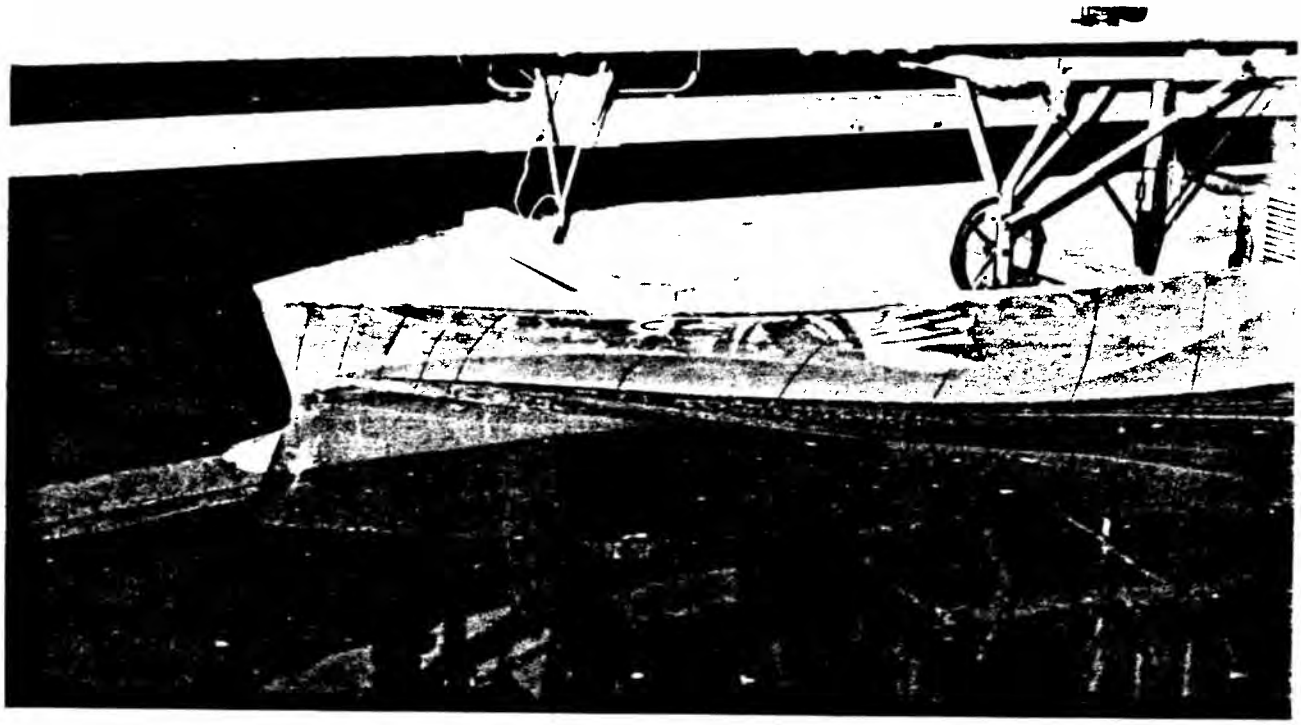


DRAFT 6.40 / 7.60 m SPEED 20.0 KNOTS TEST No. 40907

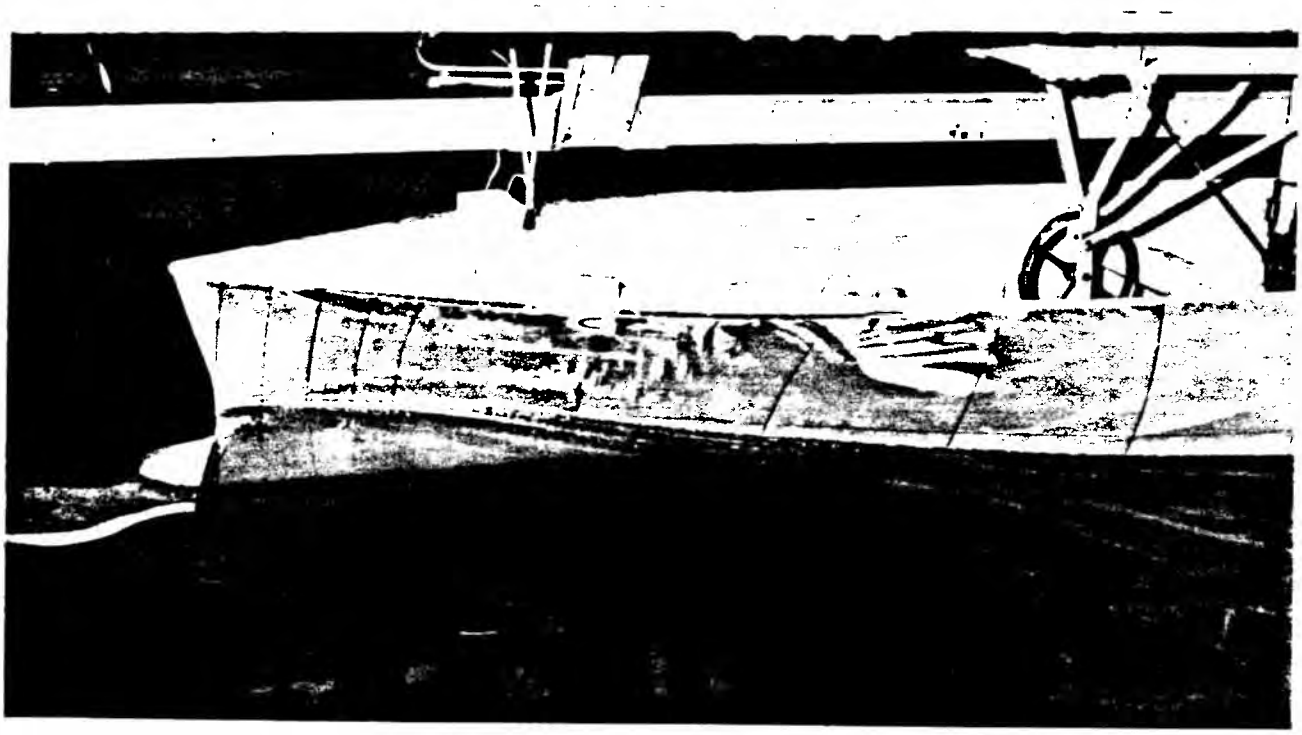


SHIP MODEL No. 5901 C

BULBOUS BOW IV (BALLAST BULB)



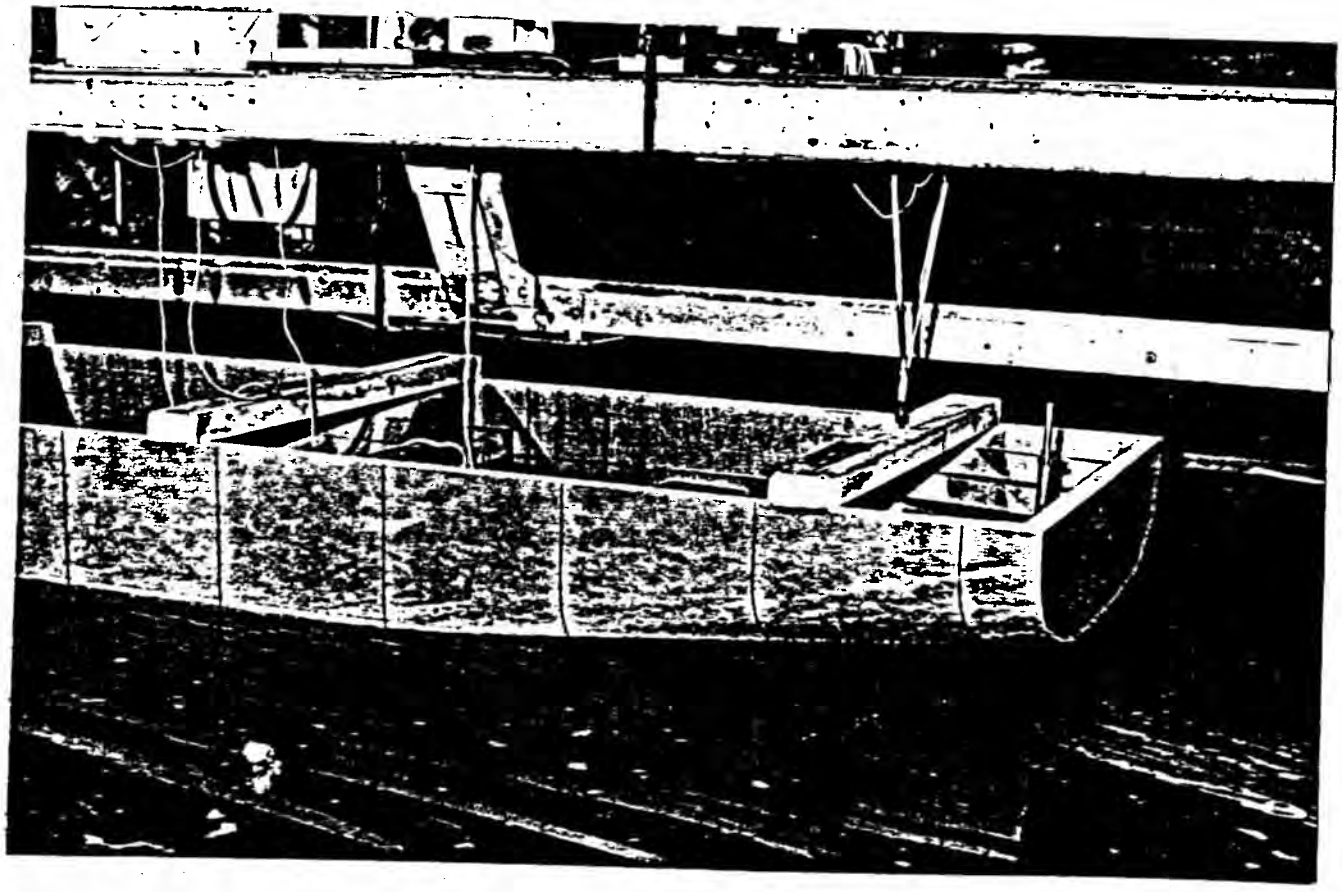
DRAFT 10.60 m SPEED 19.0 KNOTS TEST No. 40954



DRAFT 9.15 m SPEED 19.5 KNOTS TEST No. 40955

DRAFT 9.15 m SPEED 19.5 KNOTS TEST No. 40906 10-8

DRAFT 10.60 m SPEED 19.0 KNOTS TEST No. 39841



PREDICTION

SHIP MODEL 5901 C BULBOUS BOW IV

PROPELLER 5631 A

TANK VALUES ARE BASED ON RESULTS OF PROPULSION TEST No 41237

DRAFT 10.60m.

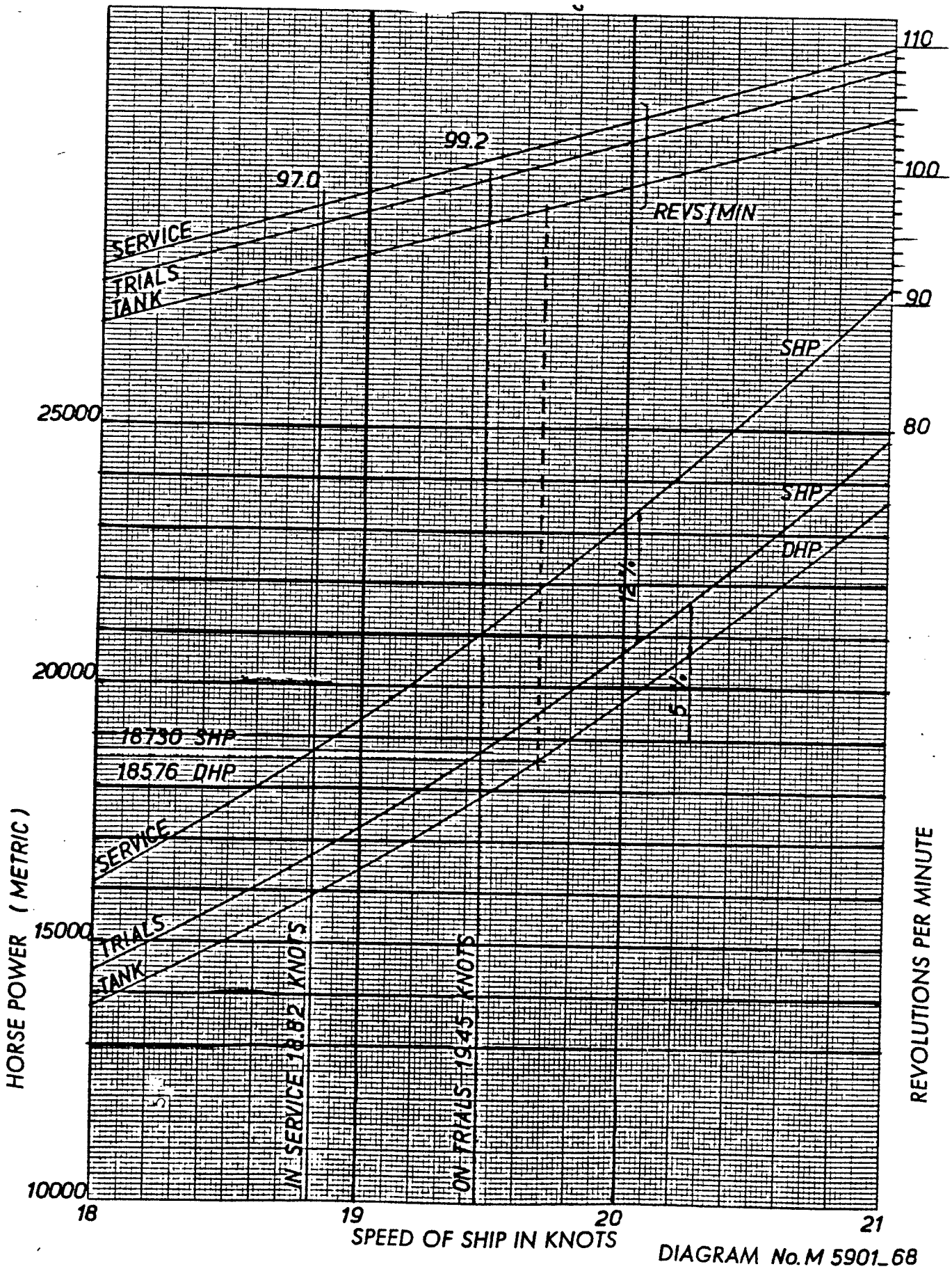


DIAGRAM No. M 5901_68

PREDICTION

SHIP MODEL 5901C BULBOUS BOW IV

PROPELLER 5631 A

TANK VALUES ARE BASED ON RESULTS OF PROPULSION TEST No 41238

DRAFT 9.15 m.

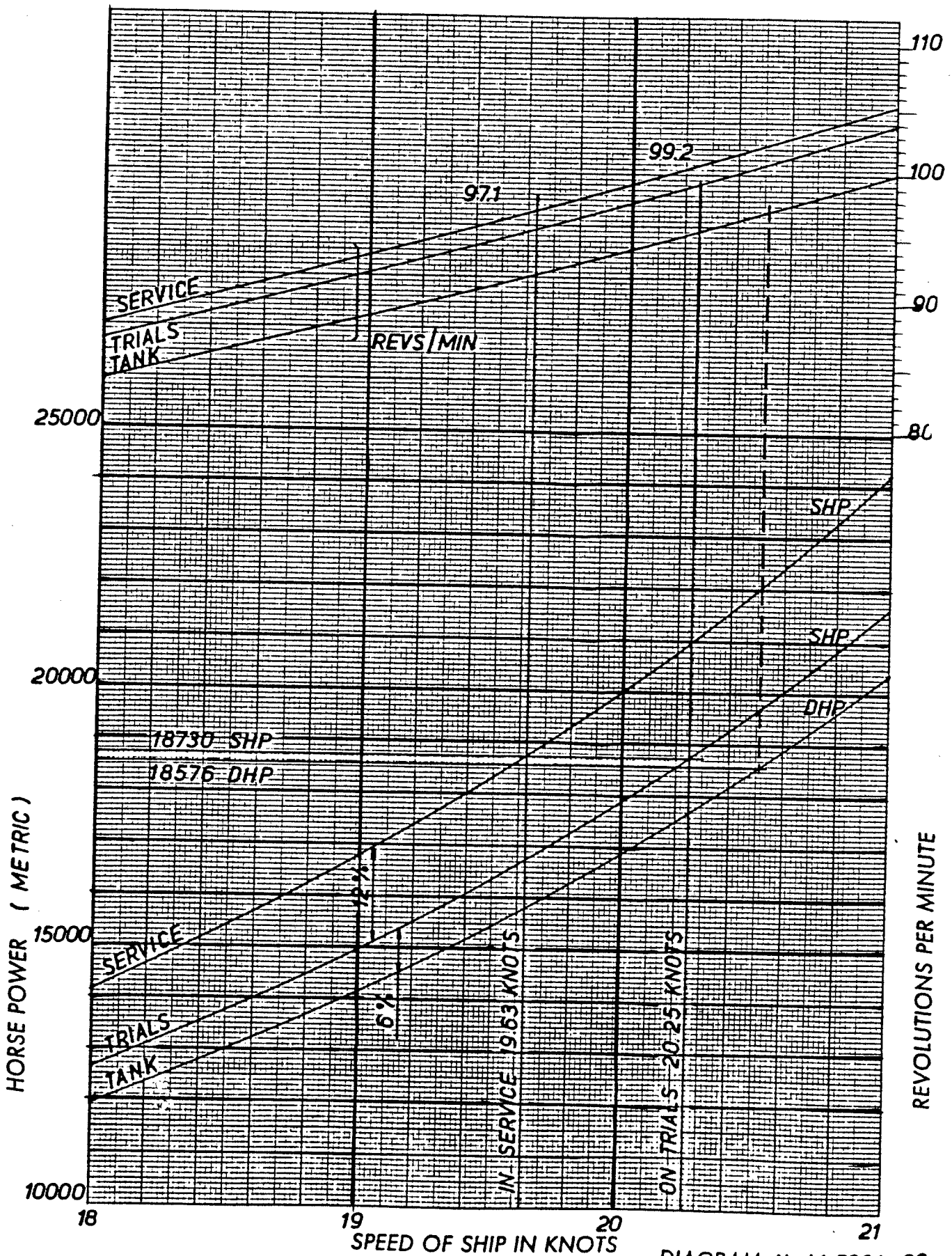


DIAGRAM No. M 5901_ 69

PREDICTION

SHIP MODEL 5901 C BULBOUS BOW IV

10-83

PROPELLER 5631 A

TANK VALUES ARE BASED ON RESULTS OF PROPULSION TEST No 41240

DRAFT 6.40 / 7.60 m.

TRIALS.

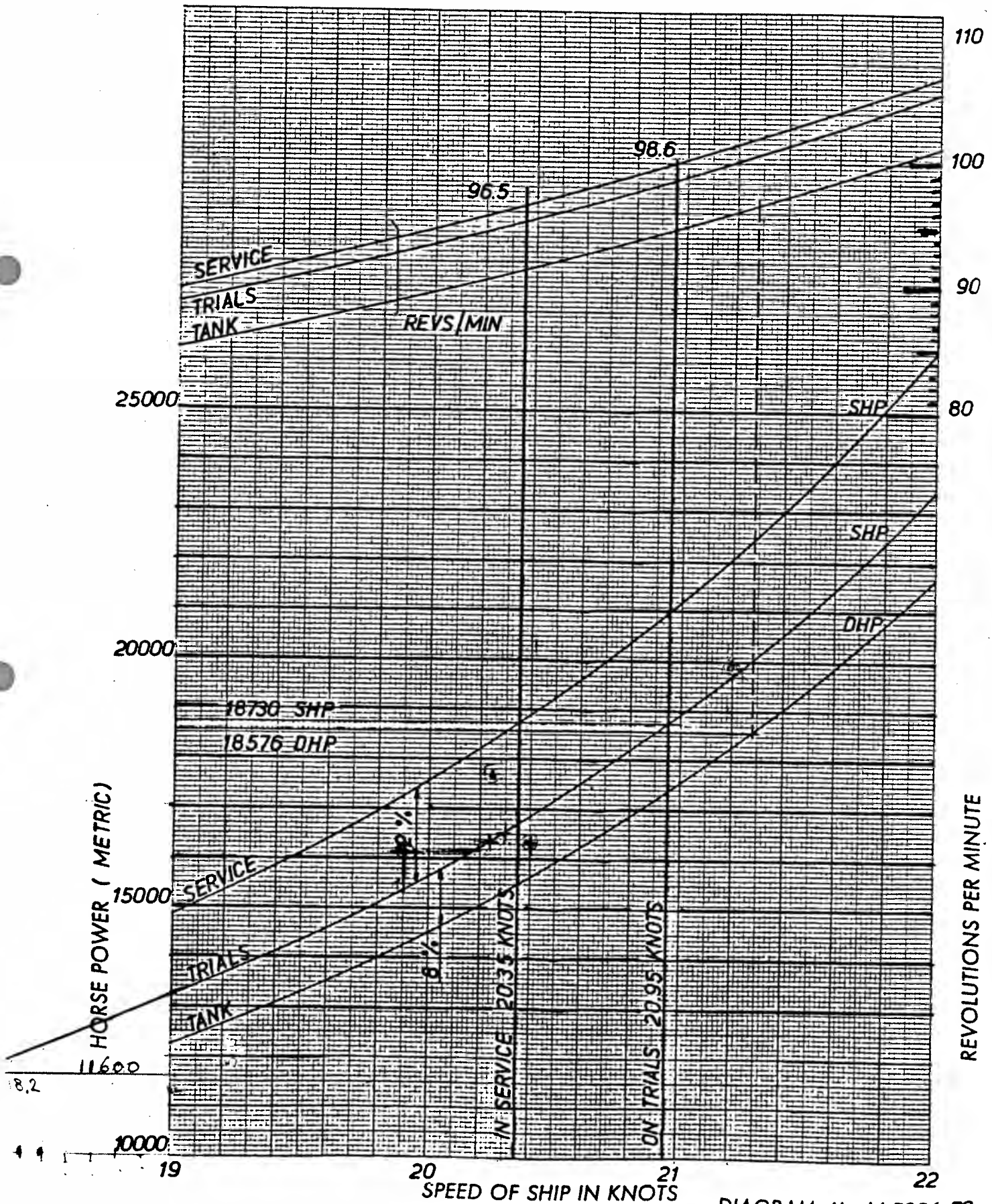
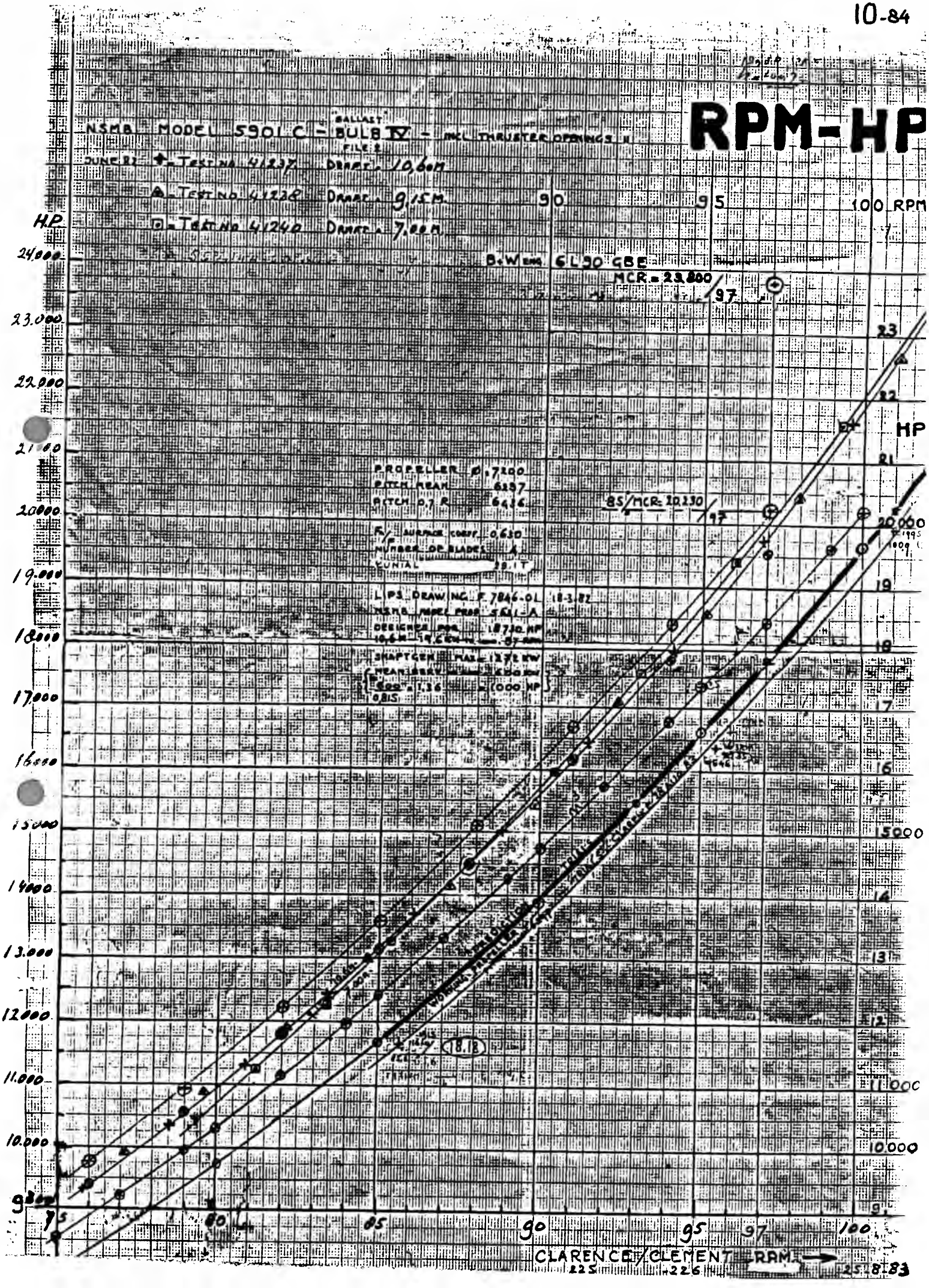
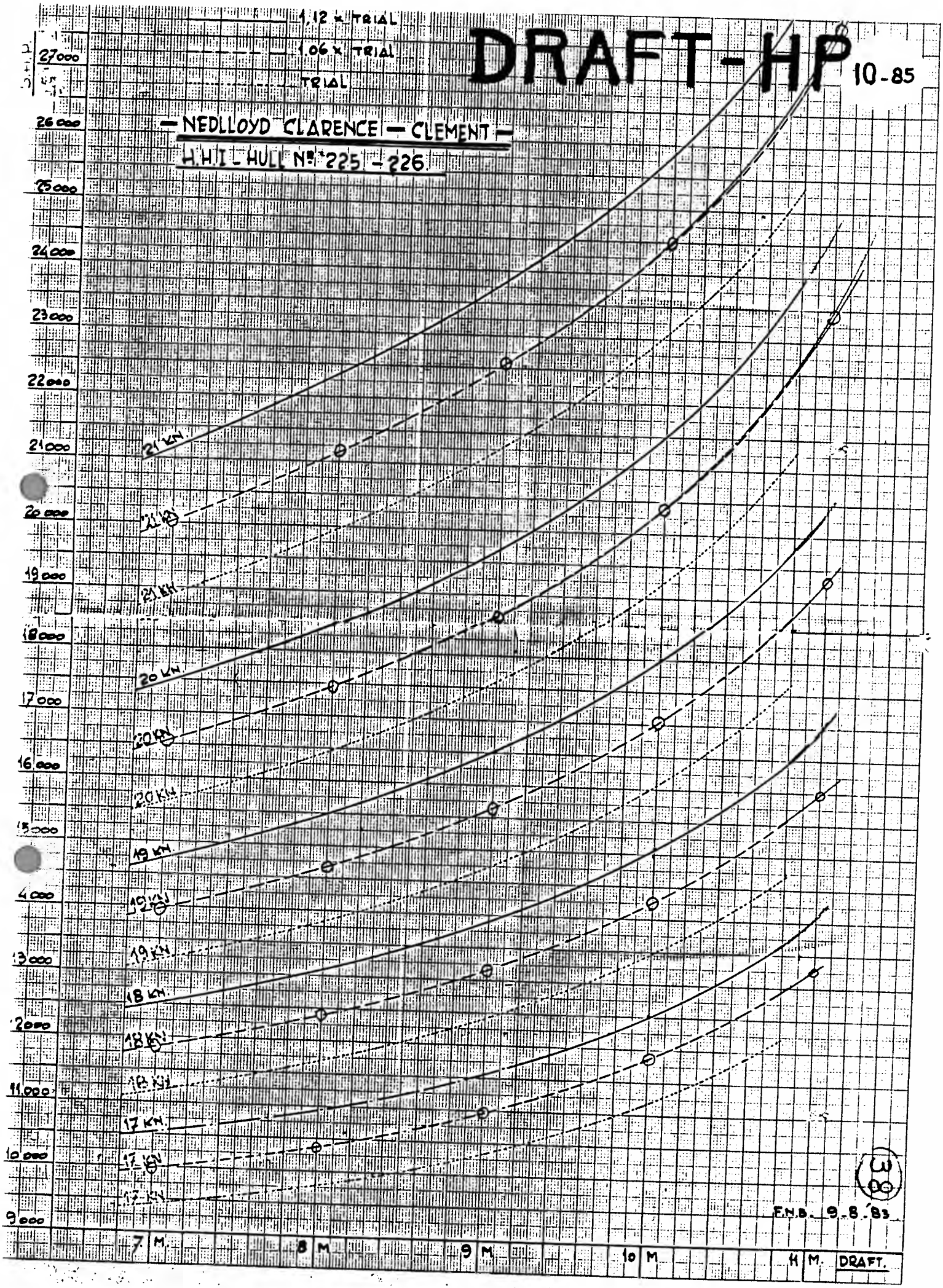
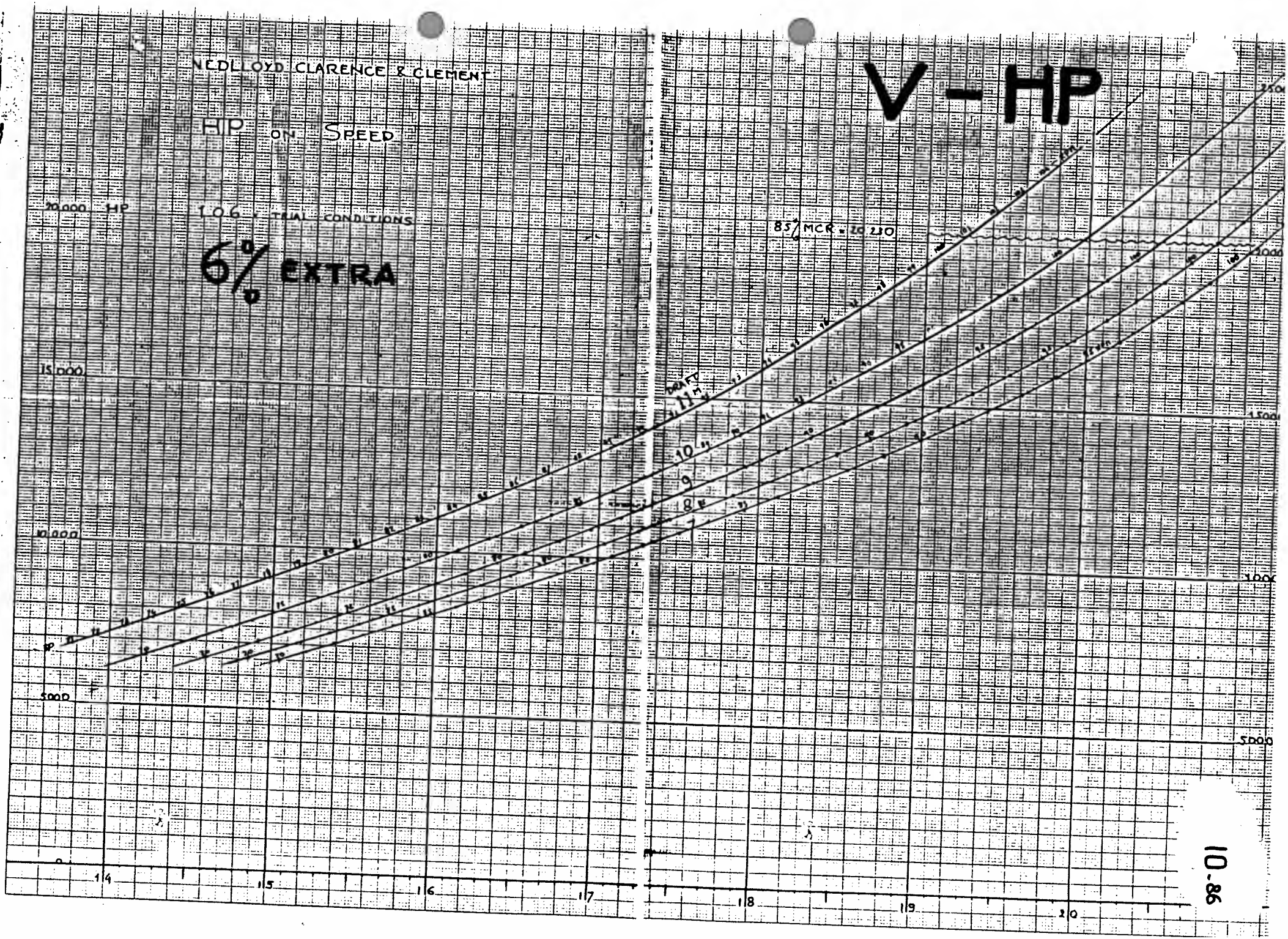


DIAGRAM No. M 5901_70

RPM-HP


NSMB MODEL 5901 C - BALLAST BULB IV - INCL THRUSTER OPENINGS
FILE 2
JUNE 82 ✚ = TEST NO 41237 DRAFT = 10.60M
▲ = TEST NO 41238 DRAFT = 9.15M
□ = TEST NO 41240 DRAFT = 7.00M
B+W EMS 6L90 GBE
MCR = 23,800
85/MCR = 20,230
PROPELLER Ø 7200
PITCH MEAN 6237
PITCH 0.7 R 6436
F/F SURFACE COEFF 0.630
NUMBER OF BLADES 4
CUNIAL 28.1 T
LPS DRAWING F 7846-01 18-3-82
NSMB MODEL PROP 5901-A
DESIGNED FOR 18730 HP
SHAFTGEN MAX 1272 KW
MEAN 1 REV
"600" = 1.36
0.815
(1000 HP)
18.18
HP
RPM
CLARENCE/CLEMENT RAM
225
226
25.8.83

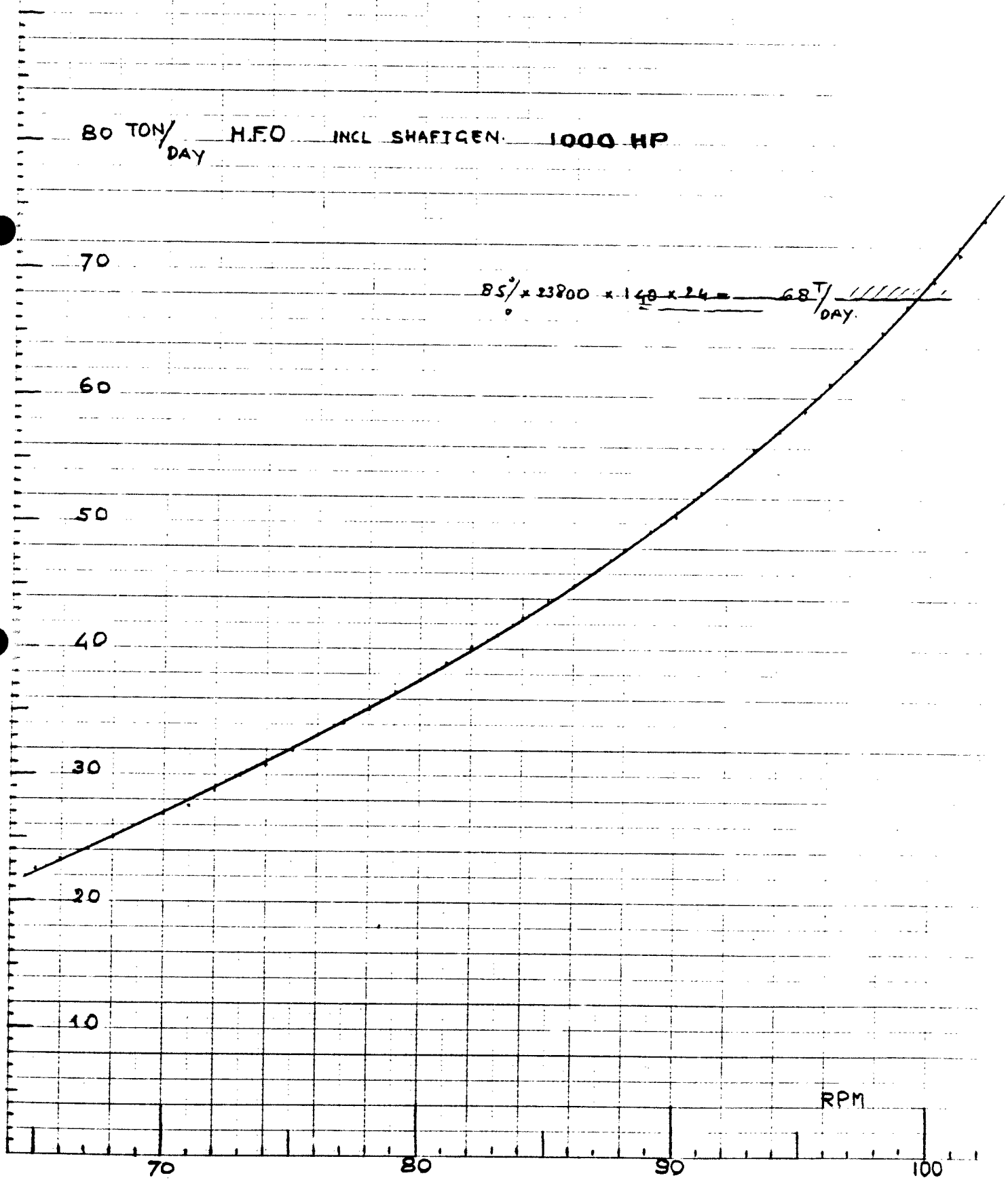

DRAFT - HP
10-85
1.12 x TRIAL
1.06 x TRIAL
TRIAL
NEDLLOYD CLARENCE — CLEMENT
H.H.I. HULL Nº 225 - 226
27000
26000
25000
24000
23000
22000
21000
20000
19000
18000
17000
16000
15000
14000
13000
12000
11000
10000
9000
21 KN
21 KN
21 KN
20 KN
20 KN
20 KN
19 KN
19 KN
19 KN
18 KN
18 KN
18 KN
17 KN
17 KN
17 KN
7 M
8 M
9 M
10 M
11 M
DRAFT
F.N.B. 9.8.83

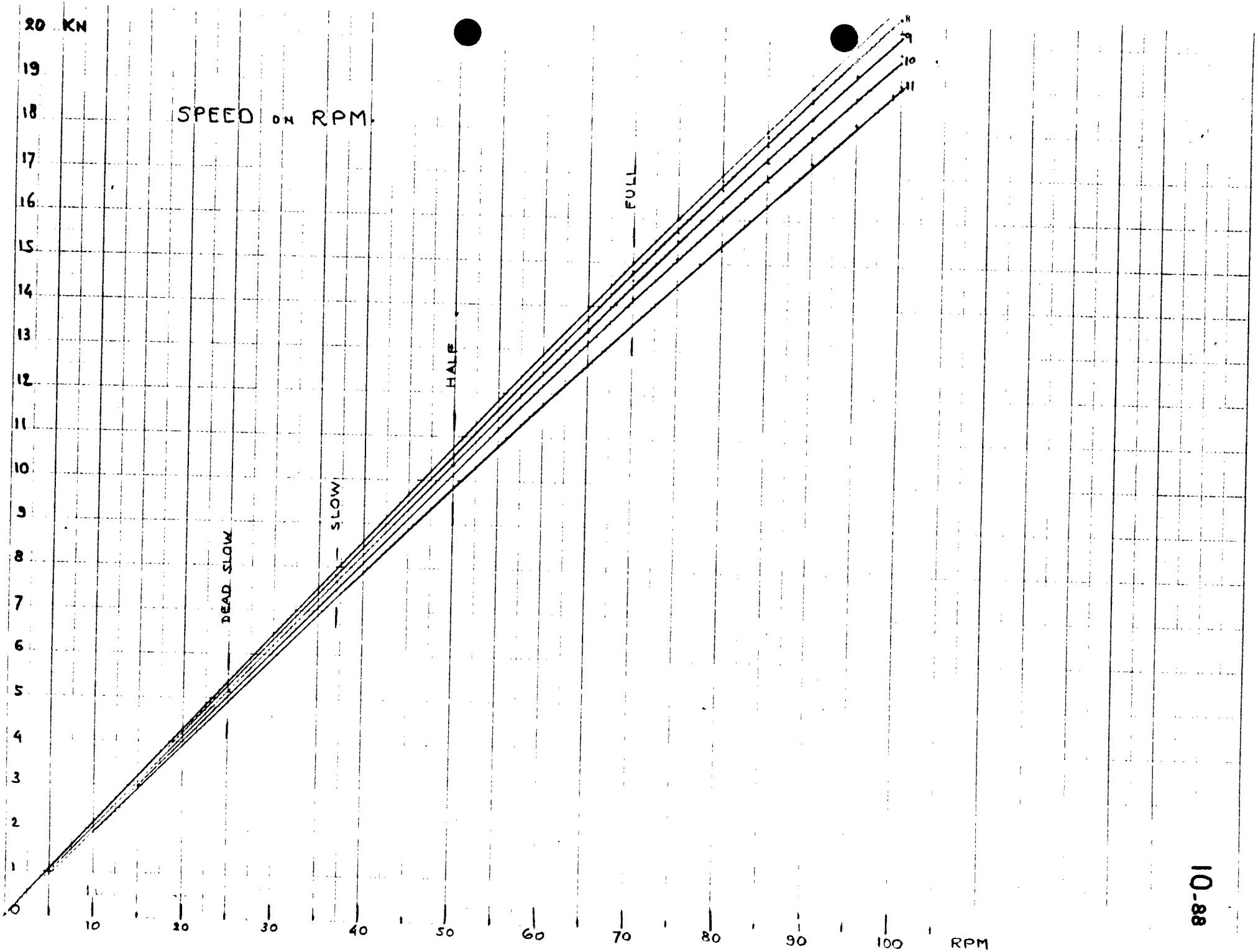

NEDLLOYD CLARENCE & CLEMENT
HP ON SPEED
V-HP
20,000 HP
1.06 TRIAL CONDITIONS
6% EXTRA
85% MCR = 20,230
15,000
10,000
5000
2500
2000
1500
1000
500
11
10
9
8
14
15
16
17
18
19
20

10-86

N. CLARENCE & CLEMENT

FUEL CONSUMPTION ON R.P.M.



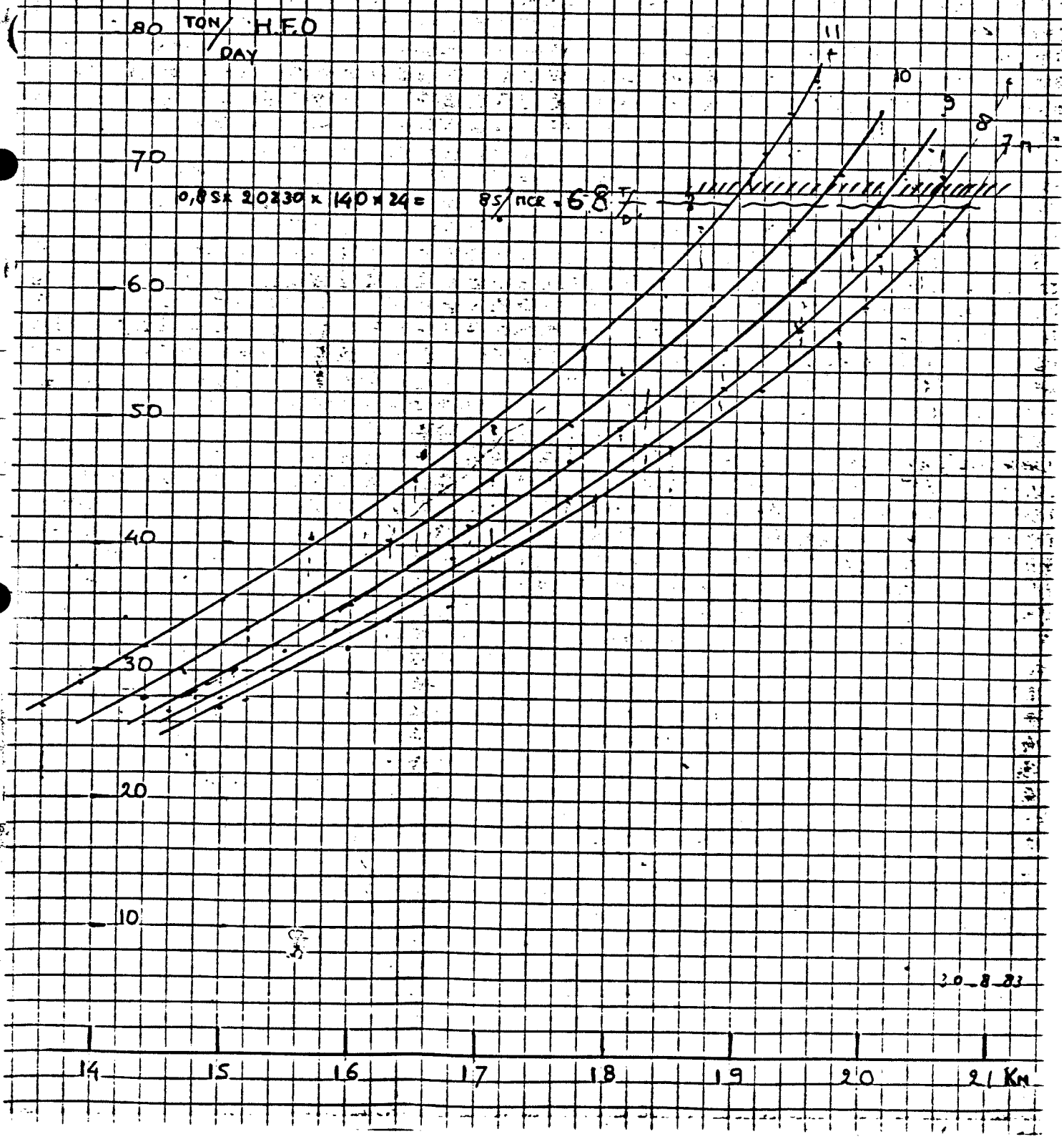

SPEED ON RPM.
KN
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
10
20
30
40
50
60
70
80
90
100
RPM
DEAD SLOW
SLOW
HALF
FULL
8
9
10
11

10.89

NEDLLOYD CLARENCE & CLEMENT

FUEL CONSUMPTION PER DAY (INCL 600 KW SHAFTGENERATOR)

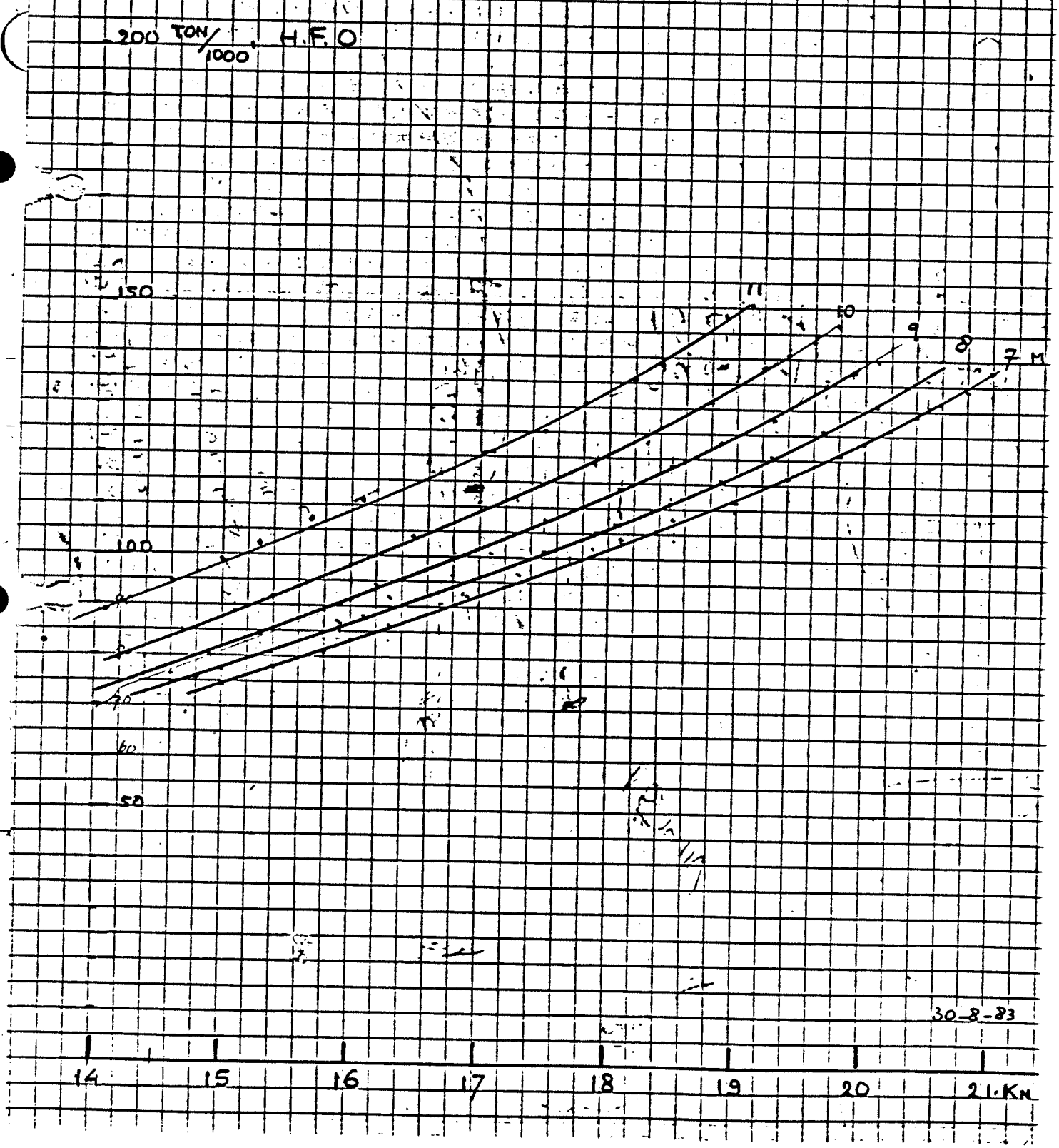
1.06 x TRIAL CONDITIONS = 6% SERVICE ALLOWANCE



NEDLLOYD CLARENCE & CLEMENT

FUEL CONSUMPTION PER 1000 MILES (INCL 600KW SHAFTGEN)

1.06 x TRIAL CONDITIONS = 6% SERVICE ALLOWANCE



NEDLLOYD CLARENCE — H.H.I HULL Nº 225

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 7 M

R.P.M	KN.	A 600 KW (= 1000 PK) + SHAFTGENERATOR - EXCLUDING REEFERS.				
DEADSL. 25	5,4	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW. 37	8,0					
HALF. 50	10,8	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
65	14.0	—	—	—	—	
66	2	5220	6220	151	22.5	
67	4	5460	6460	151	23.4	
68	6	5710	6710	150	24.2	
69	8	5970	6970	150	25.1	
FULL 70	15.0	6240	7240	150	26.1	
71	2	6520	7520	149	26.9	75
72	4	6800	7800	149	27.5	76
73	6	7090	8090	149	28.9	77
74	8	7390	8390	149	30.0	79½
75	16.0	7700	8700	148	30.9	81
76	4	8020	9020	148	32.0	83½
77	3	8340	9340	148	33.2	85
78	5	8670	9670	147	34.1	86½
79	7	9020	10020	147	35.4	88
80	9	9370	10370	147	36.6	90
81	17.1	9730	10730	146	37.6	92
82	3	10100	11100	146	38.9	94
83	5	10480	11480	146	40.2	96
84	7	10860	11860	145	41.3	98
85	9	11260	12260	145	42.7	100
86	18.1	11670	12670	144	43.8	102
87	3	12090	13090	144	45.2	104
88	5	12510	13560	143	46.4	106
89	8	12950	13950	143	47.9	108
90	19.0	13400	14400	143	49.4	110
91	2	13890	14890	142	50.7	112
92	4	14400	15400	142	52.5	114
93	6	14930	15930	141	53.9	116
94	8	15480	16480	141	55.8	118
95	20.0	16050	17050	140	57.3	120
96	2	16620	17620	140	59.2	123
97	4	17210	18210	140	61.2	126
98	6	17810	18810	140	63.2	129
99	8	18420	19420	140	65.3	132
100	21.0	19040	20040	140	67.3	135
101	2	19660	20660	140	69.4	138
102	4	20290	21290	140	71.5	
103	5	20920	21920	141	74.2	
104		21560	22560	142	76.9	
105		22200	23200	142	79.1	

NEDLLOYD CLARENCE - H.H.I HULL Nr 225

10-92

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0,7R 6436

DRAFT 8 M

R.P.M	KN.	A 600 KW (= 1000 PK) + SHAFTGENERATOR - EXCLUDING REEFERS.				
DEADSL. 25	5,3	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW. 37	7,8					
HALF. 50	10,6	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
65	13,7	5220	6220	151	22,5	
66	9	5460	6460	151	23,4	
67	14,1	5710	6710	150	24,2	
68	3	5970	6970	150	25,1	
69	5	6240	7240	150	26,1	
FULL 70	14,7	6520	7520	149	26,9	76
71	14,9	6800	7800	149	27,9	77
72	15,1	7090	8090	149	28,9	78
73	3	7390	8390	149	30,0	80
74	5	7700	8700	148	30,9	82
75	7	8020	9020	148	32,0	84
76	9	8340	9340	148	33,2	86
77	16,1	8670	9670	147	34,1	88
78	3	9020	10020	147	35,4	90
79	5	9370	10370	147	36,6	92
80	7	9730	10730	146	37,6	94
81	9	10100	11100	146	38,9	96
82	17,1	10480	11480	146	40,2	98
83	3	10860	11860	145	41,3	100
84	5	11260	12260	145	42,7	102
85	7	11670	12670	144	43,8	104
86	9	12090	13090	144	45,2	106
87	18,1	12510	13510	143	46,4	108
88	3	12950	13950	143	47,9	110
89	5	13400	14400	143	49,4	112
90	7	13890	14890	142	50,7	114
91	9	14400	15400	142	52,5	116
92	19,1	14930	15930	141	53,9	118
93	3	15480	16480	141	55,8	120
94	5	16050	17050	140	57,3	122
95	7	16620	17620	140	59,2	125
96	9	17210	18210	140	61,2	128
97	20,1	17810	18810	140	63,2	131
98	3	18420	19420	140	65,3	134
99	5	19040	20040	140	67,3	137
100	6	19660	20660	140	69,4	140
101	8	20290	21290	140	71,5	
102	9	20920	21920	141	74,2	
103	21,1	21560	22560	142	76,9	
104	2	22200	23200	142	79,1	
105		22850	23850	143	81,9	

NEDLLOYD CLARENCE — HHI HULL Nr 225 — B

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 9 M

46

R.P.M.	KN.	A 600 KW (= 1000 PK) + SHAFTGENERATOR - EXCLUDING REEFERS.				
DEADSL .25	5.2	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW .37	7.7					
HALF .50	10.4	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000M
		—	—	—	—	
65	13.5	5220	6220	151	22.5	
66	7	5460	6460	151	23.4	
67	9	5710	6710	150	24.2	
68	1	5970	6970	150	25.1	
69	3	6240	7240	150	26.1	
FULL 70	14.5	6520	7520	149	26.9	76
71	7	6800	7800	149	27.5	78
72	9	7090	8090	149	28.9	80
73	15.1	7390	8390	149	30.0	82
74	3	7700	8700	148	30.9	84
75	5	8020	9020	148	32.0	86
76	6	8340	9340	148	33.2	88
77	8	8670	9670	147	34.1	90
78	16.0	9020	10020	147	35.4	92
79	2	9370	10370	147	36.6	94
80	4	9730	10730	146	37.6	96
81	6	10100	11100	146	38.9	98
82	8	10480	11480	146	40.2	100
83	9	10860	11860	145	41.3	102
84	17.1	11260	12260	145	42.7	104
85	3	11670	12670	144	43.8	106
86	5	12090	13090	144	45.2	108
87	7	12510	13510	143	46.4	110
88	9	12950	13950	143	47.9	112
89	18.1	13400	14400	143	49.4	114
90	3	13890	14890	142	50.7	116
91	5	14400	15400	142	52.5	118
92	7	14930	15930	141	53.9	120
93	9	15480	16480	141	55.8	122
94	19.1	16050	17050	140	57.3	125
95	3	16620	17620	140	59.2	128
96	5	17210	18210	140	61.2	131
97	7	17810	18810	140	63.2	134
98	9	18420	19420	140	65.3	137
99	20.1	19040	20040	140	67.3	140
100	3	19660	20660	140	69.4	143
101	5	20290	21290	140	71.5	
102	6	20920	21920	141	74.2	
103	8	21560	22560	142	76.9	
104	9	22200	23200	142	79.1	
105		22850	23850	143	81.9	

NEDLLOYD CLARENCE — H.H.I HULL Nº 225 — B

PROP: 4 BL — Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 10 M

47

R.P.M	KN	A 600 KW (= 1000 PK) + SHAFTGENERATOR - EXCLUDING REEFERS.				
DEADSL-25	5.1	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW-37	7.5					
HALF-50	10.1	PROPELLER S.H.P.	+ GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
		—	—	—	—	
65	13.2	5220	6220	151	22.5	
66	4	5460	6460	151	23.4	
67	6	5710	6710	150	24.2	
68	8	5970	6970	150	25.1	
69	14.0	6240	7240	150	26.1	
FULL 70	142	6520	7520	149	26.9	79
71	4	6800	7800	149	27.9	81
72	5	7090	8090	149	28.9	83
73	7	7390	8390	149	30.0	85
74	9	7700	8700	148	30.9	87
75	15.1	8020	9020	148	32.0	89
76	2	8340	9340	148	33.2	91
77	4	8670	9670	147	34.1	93
78	6	9020	10020	147	35.4	95
79	8	9370	10370	147	36.6	97
80	16.0	9730	10730	146	37.6	99
81	1	10100	11100	146	38.9	101
82	3	10480	11480	146	40.2	103
83	5	10860	11860	145	41.3	105
84	7	11260	12260	145	42.7	107
85	9	11670	12670	144	43.8	109
86	17.1	12090	13090	144	45.2	111
87	3	12510	13510	143	46.4	113
88	5	12950	13950	143	47.9	115
89	7	13400	14400	143	49.4	117
90	9	13890	14890	142	50.7	119
91	18.1	14400	15400	142	52.5	121
92	3	14930	15930	141	53.9	123
93	5	15480	16480	141	55.8	125
94	7	16050	17050	140	57.3	128
95	8	16620	17620	140	59.2	131
96	19.0	17210	18210	140	61.2	134
97	2	17810	18810	140	63.2	137
98	4	18420	19420	140	65.3	140
99	6	19040	20040	140	67.3	143
100	8	19660	20660	140	69.4	146
101	20.0	20290	21290	140	71.5	
102	1	20920	21920	141	74.2	
103		21560	22560	142	76.9	
104		22200	23200	142	79.1	
105		22850	23850	143	81.0	

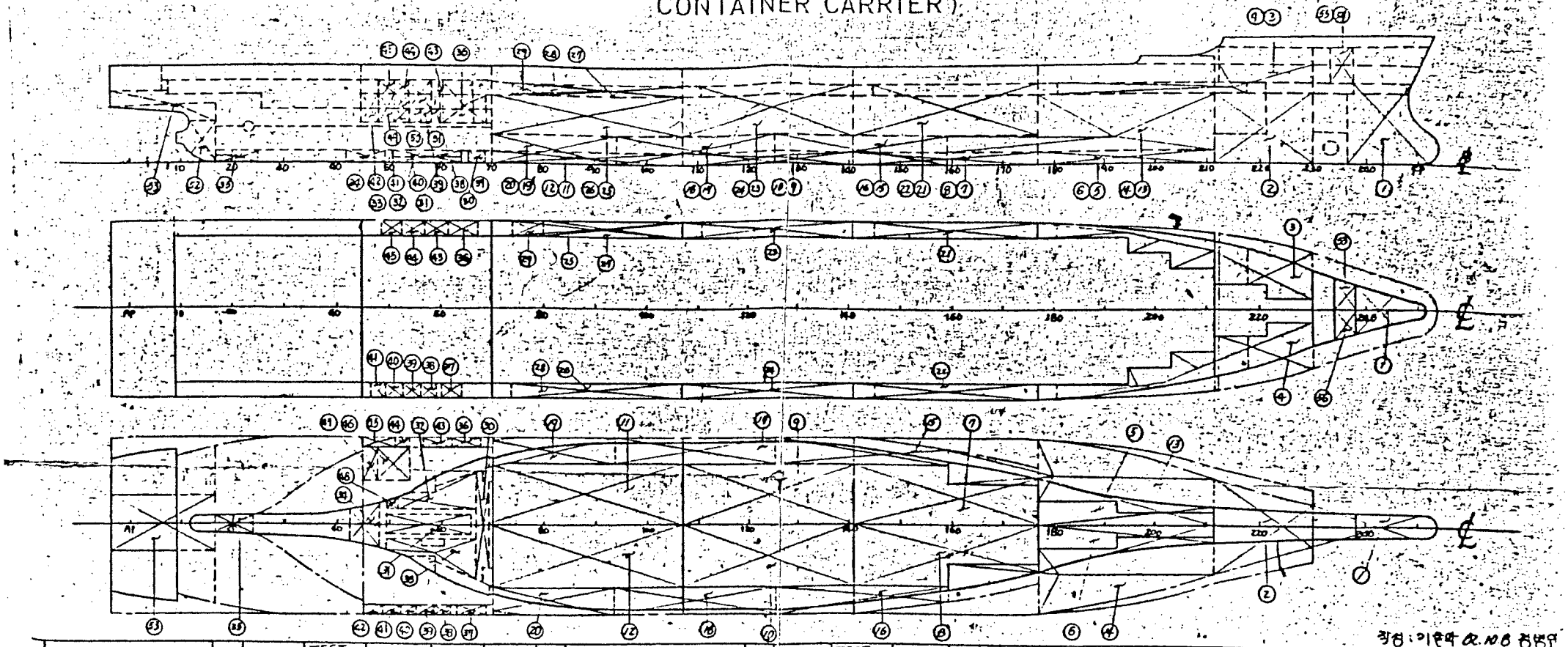
NEDLLOYD CLARENCE – H.H.I HULL Nº 225

PROP: 4 BL Ø 7200
PITCH MEAN 6297
0.7R 6436

DRAFT 11 M

R.P.M	KN.	À 600 KW (= 1000 PK) + SHAFTGENERATOR – EXCLUDING REEFERS.				
DEADSL.-25	4.9	6% SERVICE ALLOWANCE OUTPUT		FUEL CONSUMPTION.		
SLOW.-37	7.3					
HALF.50	9.8	PROPELLER S.H.P.	+GENER. B.H.P.	GR/HP.HR	TON/DAY	TON/1000 M
		—	—	—	—	
65	12.6	5220	6220	151	22.5	
66	8	5460	6460	151	23.4	
67	13.0	5710	6710	150	24.2	
68	2	5970	6970	150	25.1	
69	4	6240	7240	150	26.1	
FULL 70	13.6	6520	7520	149	26.9	82
71	8	6800	7800	149	27.9	84
72	9	7090	8090	149	28.9	86
73	14.1	7390	8390	149	30.0	88
74	3	7700	8700	148	30.9	90
75	4	8020	9020	148	32.0	92
76	6	8340	9340	148	33.2	94
77	8	8670	9670	147	34.1	96
78	15.0	9020	10020	147	35.4	98
79	1	9370	10370	147	36.6	100
80	3	9730	10730	146	37.6	102
81	5	10100	11100	146	38.9	104
82	7	10480	11480	146	40.2	106
83	9	10860	11860	145	41.3	108
84	16.1	11260	12260	145	42.7	110
85	3	11670	12670	144	43.8	112
86	5	12090	13090	144	45.2	114
87	7	12510	13510	143	46.4	116
88	9	12950	13950	143	47.9	118
89	17.1	13400	14400	143	49.4	120
90	3	13890	14890	142	50.7	122
91	5	14400	15400	142	52.5	124
92	7	14930	15930	141	53.9	127
93	8	15480	16480	141	55.8	130
94	18.0	16050	17050	140	57.3	133
95	2	16620	17620	140	59.2	136
96	4	17210	18210	140	61.2	139
97	6	17810	18810	140	63.2	142
98	7	18420	19420	140	65.3	145
99	9	19040	20040	140	67.3	148
100	19.1	19660	20660	140	69.4	151
101	2	20290	21290	140	71.5	
102	4	20920	21920	141	74.2	
103	6	21560	22560	142	76.9	
104	7	22200	23200	142	79.1	
105		22850	23850	143	81.9	

225-226 TANK TESTING PLAN
(CONTAINER CARRIER)



TANK NAME	P.S	FR NO	TEST TYPE	CAPACITY	INT	TEST
W B FORE PEAK TANK	C	236-F.E	AIR/STRE WATR	419.0		
NO1 W B DEEP TANK	C	211-230	"	814.5		
NO1 W B UPPERSIDE TANK	P	180-230	AIR	690.9		
NO1 "	S	"	"	703.4		
NO2 F.P.E TANK	P	176-211	AIR/STRE WATR	236.4		
NO2 "	S	"	AIR	236.4		
NO3 "	P	121-176	"	505.3		
NO3 "	S	"	"	505.3		
NO4 "	P	106-141	"	557.05		
NO4 "	S	"	"	557.05		
NO5 "	P	69-106	"	559.35		
NO5 "	S	"	AIR/STRE WATR	559.35		
NO2 H F.O SIDE TANK	P	159-221	AIR	1187.45		
NO2 "	S	"	AIR/STRE WATR	1187.45		
NO3 H.F.O LOWERSIDE TANK	P	125-159	AIR	299.2		
NO3 "	S	"	"	299.2		
NO4 "	P	94-125	AIR/STRE WATR	372.05		
NO4 H F O LOWERSIDE TANK	S	"	AIR	372.05		
NO5 H.F.O "	P	69-94	"	199.15		

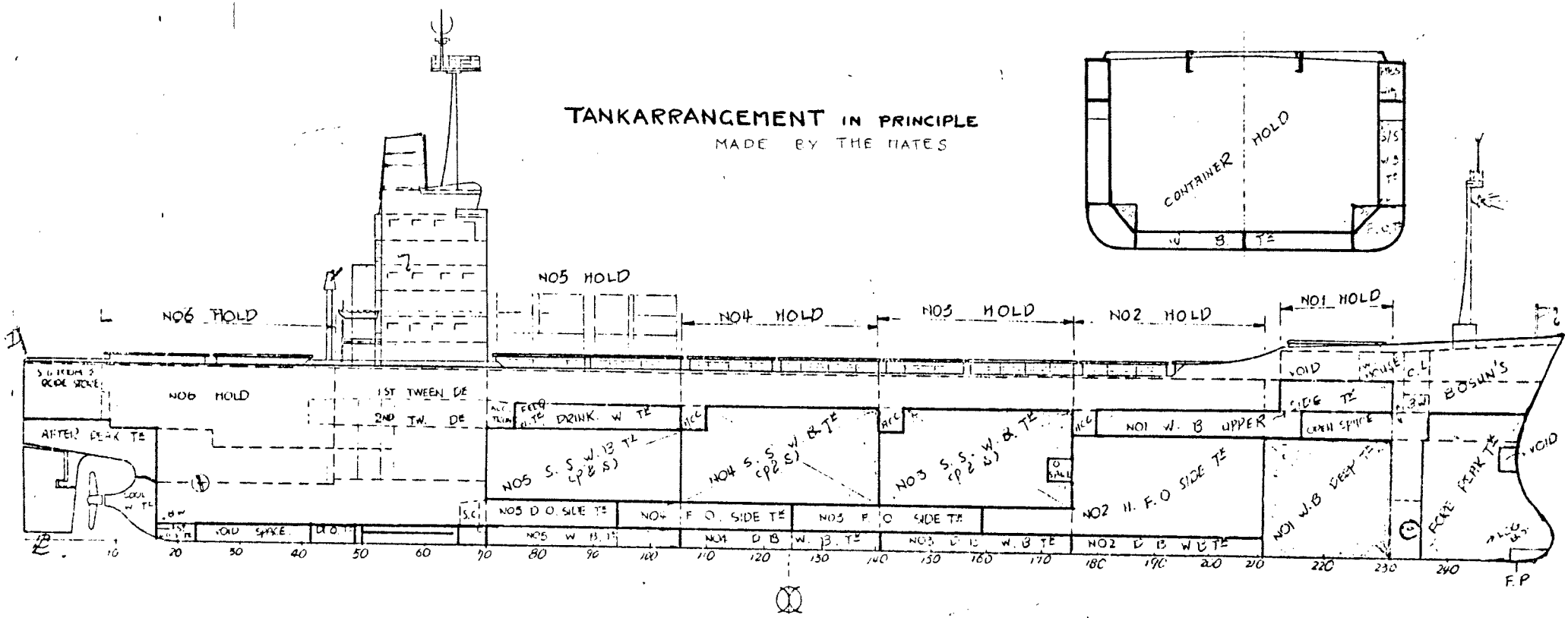
NO.	TANK NAME	P.S	FR.NO	TEST TYPE	CAPACITY	INT	TEST
20	NO5 M.D.O LOWERSIDE TANK	S	69-94	AIR	199.15		
21	NO3 W B UPPERSIDE TANK	P	161-176	"	668.5		
22	NO3 "	S	"	AIR/STRE WATR	668.5		
23	NO4 "	P	106-141	AIR	652.4		
24	NO4 "	S	"	"	652.4		
25	NO5 "	P	70-106	"	524.0		
26	NO5 W B UPPERSIDE TANK	S	"	"	524.0		
27	DRINKING WATER TANK	P	61-106	"	119.3		
28	"	S	95-106	"	180.5		
29	FEED WATER TANK	P	75-81	"	29.2		
30	F.O OVERFLOW TANK	C	66-69	"	74.0		
31	L.O SUMP TANK	C	50-65	"	44.1		
32	DIRTY WATER TANK	P	52-66	"	62.5		
33	"	S	50-66	HYDRO	55.9		
34	DIRTY OIL TANK	C	42-49	"	57.1		
35	STREN TUBE L.O SUMP TANK	C	19-23	AIR	12.3		
36	NO1 M F O SETTLING TANK	P	62-68	"	93.0		
37	NO2 "	S	"	HYDRO	93.0		
38	L.O PURIFYING TANK M E	S	58-61	AIR	48.4		

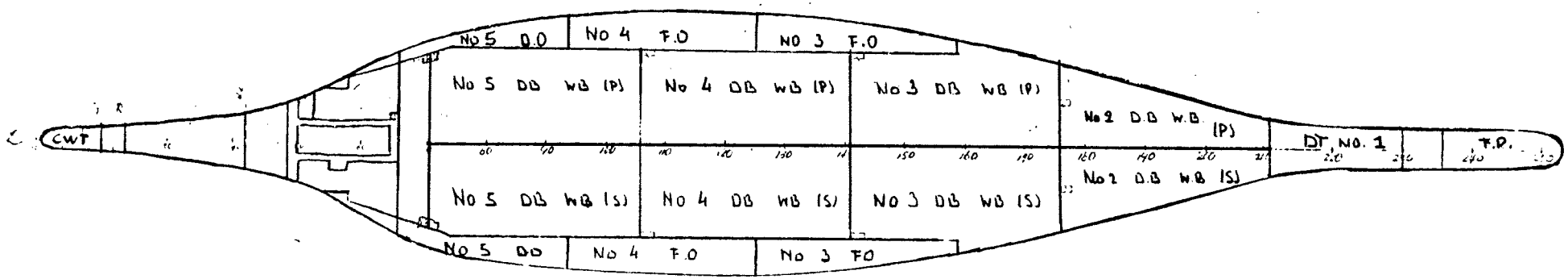
NO	TANK NAME	P.S	FR.NO	TEST TYPE	CAPACITY	INT	TEST
39	NO1 L.O STOR TANK ME	S	54-58	HYDRO	64.0		
40	NO2 "	S	51-54	AIR	28.9		
41	NO1 CYL OIL STOR TANK	S	48-51	HYDRO	27.9		
42	NO2 "	S	45-48	AIR	27.9		
43	CLEAN H F.O TANK	P	56-62	HYDRO	64.2		
44	CLEAN M D O TANK	P	51-54	AIR	30.3		
45	M.D.O SETTLING TANK	P	45-51	"	60.6		
46	L.O SLUDGE TANK	P	45-48	HYDRO			
47	F.O "	P	48-54	AIR			
48	F.O LEAK TANK	P	50-52	"	3.1		
49	L.O STOR TANK	S	51-54	"	9.2		
50	CLEAN L O TANK A E	S	54-56	"	6.0		
51	L O RENOV TANK A E	S	56-58	"	6.0		
52	COOLING WATER TANK	C	9-17	HYDRO			
53	W B AFT PEAK TANK	C	AE-17	"	540.6		
54	CHAIN LOCKER	P	233-237	FLOODING	55.2		
55	"	S	"	"	55.2		

10-96

TANKARRANGEMENT IN PRINCIPLE

MADE BY THE MATES





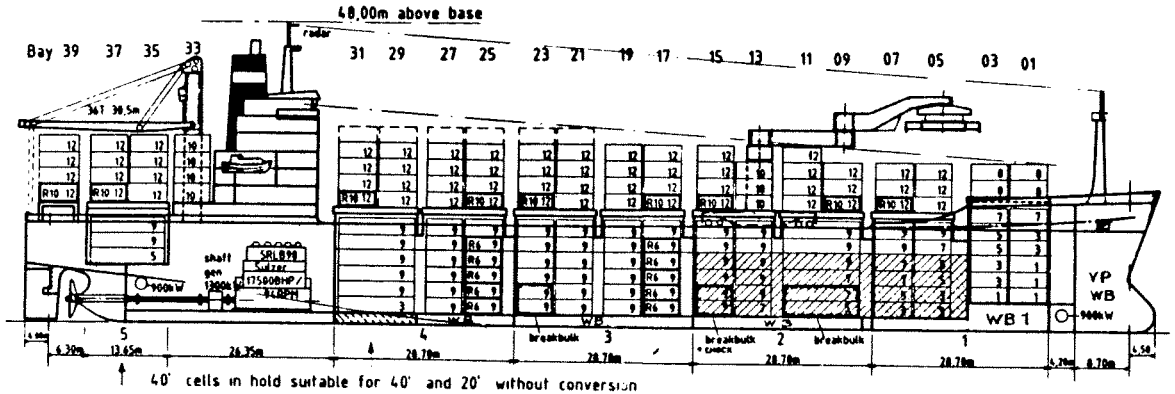
TANK TOP PLAN

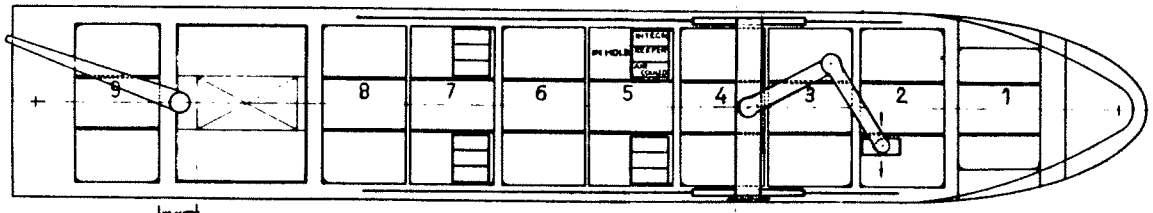
10-97

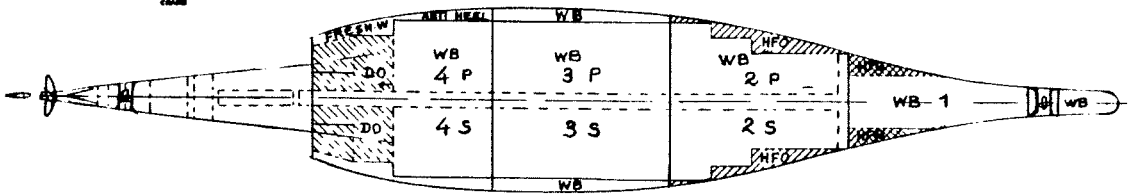
10-98

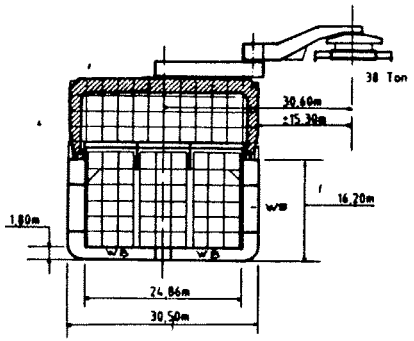
NEDLLOYD v NECK
v NOORT
v DIEMEN

DEC '83









TYPICAL LOADING CONDITION DESIGN

Draft design mld	10 - M
Displacement	36852T
Light ship	11030
Deadweight 10M	23822
FO 25 DAYS - 48 %/day - 17½ x 24 hr	1200
DO+Lub+Fr.w+stores	512
Waterballast	1460
Cargo containers: TOTAL	20650=14Tx1475 TEU
" in hold	10976=14Tx 784
3 tiers on deck	8638=14Tx 617
4th tier on deck	1036=14Tx 74
Homog.loaded-centr.gr/box=	0,45x8,5FT
KM	14,37 M
KG	13,90
GM	0,47
free surface corr.	0,06
G'M'	0,41 M
BRT	23930T
NRT	10756T

Length over all	182,50M + 0,23
Length betw. perpend	174,00M
Breadth mld	30,50M
Depth mld	16,20M
Draft scantling-EXTR.	11,27M
Displacement	40760 TON
Light ship	11030 TON
Deadweight	29730 TON

Water ballast	7874 m3
Fresh water	411 m3
Heavy Fuel Oil	2883 m3
Diesel oil	366 m3
Lubricating oil	281 m3

MAIN ENGINE	SULZER 5RLB90
MCR/RPM	17500 BHP/94
85%	14875 BHP/94 x24x142
	= ±51 Ton/day
Fuel M.E. :	HFO 600 cst - 50°C
AUXIL:	50% HFO + 50% MDO

Heavy Fuel Oil consumption excl shaft generator in tons/day

SHP service = $SHP_{tank} \times C_1 \times C_2$

C_1 = margin for trials = 1,06

C_2 = sea margin = 1,12 (service allowance)

Spec fuel consumption = 143 gr/SHP h

Draft	15,5	16	16,5	17	17,5	18	18,5	19kn
7M	26,5	29,0	31,4	34,1	36,9	39,9	43,1	46,5
8M	25,5	28,1	30,7	33,4	36,4	39,6	43,1	46,7
9M	26,7	29,4	32,2	35,1	38,2	41,5	45,3	49,1
10M	30,1	33,5	37,1	40,9	45,1	49,7	55-	ton/day

Shaft generator max	1300 kW
Normal 540 kW x 1,36/0,85	= 860 HP
x 143 gr/SHP hr x 24	= 2,9 T/day
Diesel generators	
Diesel gen I	1500kW - 720RPM
Diesel gen II	1500kW - 720RPM
Diesel gen III	700kW - 720RPM
Emergency gen	160kW - 1800RPM

Normal electr load at sea	
600kW - MDO diesel gen	2,9T/day
- HFO shaft gen	2,9T/day
Idle in port 450kW MDO	2,3T/day
Boiler HFO	1,0T/day

Extra for reefers (100 integr 20')	
box x kW/box x HP/kW x	
gr/HP hr x hr =	
100 x 5,5 x 1,5 x 143 x 24	= 2,6 T/day

MAX. TEU CAPACITY SPACE WISE		
Hold	776	of which 117 - 40'=30%
Deck	212	1st tier
	228	2nd tier
	228	3rd tier
	1444	
	166	4th tier
	1610	

Space on tanktop 2 tiers high
In guides chocks
Suitable for BREAKBULK
Load on tanktop 12 ton/M2

Hatchrows 3 + 3 + 3 cover weight 18T and 19T
Number of hatches 8 fwd of deckh
1 aft of deckhouse
40' cells in hatch no 3 - 8 - 9
40' box below deck 117 = 46+48+23
In no 8 and 9 also 20' boxes can be stowed on cones
Max stackload on covers
20' stack 60 Ton
40' stack 90 Ton
Max stackload if twistlocks only 40 T
If stackload exceeds 40T lash by rods + turnbuckles

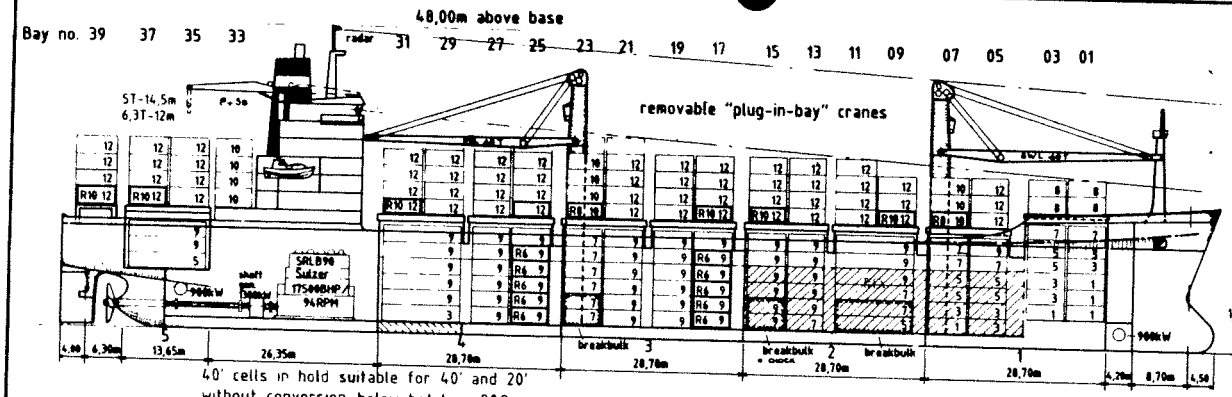
THRUSTERS (var pitch)	
Bowthruster 900 kW, 1220 HP	
Sternthruster 900 kW, 1220 HP	
Rudder max angle 45°	

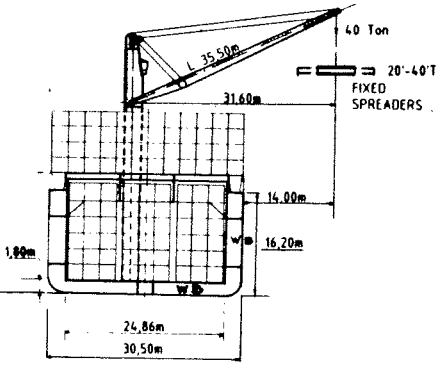
INTEGRAL REEFERS	
Reeferpoints on deck 20'/40'	90
5 kW 20'/7,5kW 40'	
Reeferpoints in hold 20'	60
ventilation 1 m3/sec per box NO WATER AIR ONLY	
Receptacles 440V, CEE 17, 3h, 32 Amp	
Hold ventilation H 1, 2 & 5	2,5 cfm
H 3 & 4	9 cfm

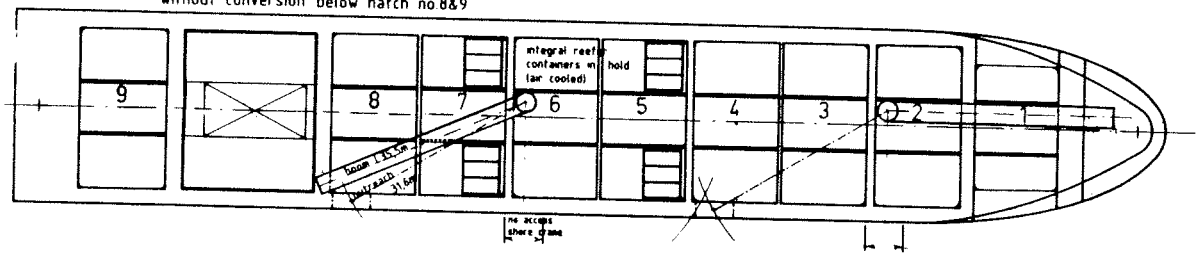
NEDLLOYD VAN NECK 929 VDG

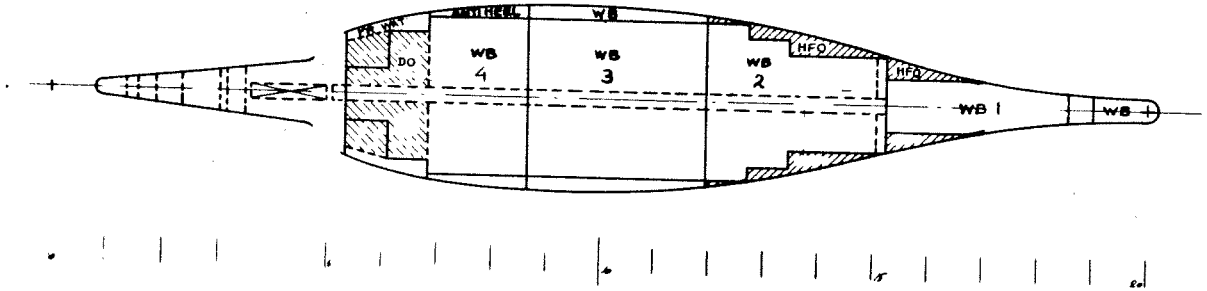
Delivery date: 2 December 1983
Yard v/d Giessen de Noord
Class Bureau Veritas

10-99









estimated BRT 23930
NRT 10756

MAX. TEU CAPACITY SPACE WISE		
Hold	752	of which 117 - 40' = 31%
Deck	210	1st tier
	226	2nd tier
	226	3rd tier
	1414	
	176	4th tier
	1590	

Space on tanktop 2 tiers high
In guides chocks
Suitable for breakbulk
Load on tanktop 12 ton/M2

Hatchrows 3 + 3 + 3	COVER: 18-19T
Number of hatches 8 fwd of deckh	
1 aft of deckhouse	
40' cells in hatch no 3 - 8 - 9	
40' box below deck 117 = 46+48+23	
In no 8 and 9 also 20' boxes can be	
stowed on cones	
Max stackload on covers	
20' stack 60 Ton	
40' stack 90 Ton	
Max stackload if twistlocks only 40 T	
If stackload exceeds 40T lash by	
rods + turnbuckles	

THRUSTERS (var pitch)
Bowthruster 900 kW, 1220 HP
Sternthruster 900 kW, 1220 HP
Rudder max angle 45°

INTEGRAL REEFERS	
Reeferpoints on deck 20'/40'	90
5,- kW 20'/7,5kW 40'	
Reeferpoints in hold 20'	60
ventilation 1 m3/sec per box	
Receptacles 440V, CEE 17, 3h, 32 Amp	
Hold ventilation H 1, 2 & 5	
H 3 & 4	

MAIN ENGINE SULZER 5RLB90	
MCR	17500 HP at 94 RPM
service rating	14875 HP at 94 RPM
(85%) = ± 51 ton/day	
Fuel for main eng HFO max 600 cSt at 50°	
Fuel for aux.eng IFO = 50%HFO/50%MDO	
Heavy Fuel Oil consumption excl	
shaft generator in tons/day	
SHP service = $SHP_{tank} \times C_1 \times C_2$	
C_1 = margin for trials = 1,06	
C_2 = sea margin = 1,12 (service allowance)	
Spec fuel consumption = 143 gr/SHP h	

Draft	15.5	16	16.5	17	17.5	18	18.5	19kn
7M	26.5	29.0	31.4	34.1	36.9	39.9	43.1	46.5
8M	25.5	28.1	30.7	33.4	36.4	39.6	43.1	46.7
9M	26.7	29.4	32.2	35.1	38.2	41.5	45.3	49.1
10M	30.1	33.4	37.1	40.9	45.1	49.7	55.0	ton/day

Shaft generator max	1300 kW
Normal 540 kW x $\frac{1,36}{,85}$ =	860 HP
x 143 gr/SHP hr x 24 =	2,9 T/day
Diesel generators	
Diesel gen I	1500kW - 720RPM
Diesel gen II	1500kW - 720RPM
Diesel gen III	700kW - 720RPM
Emergency gen	160kW - 1800RPM
Normal electr load at sea	540kW
-IFO diesel gen	2,9T/day
-HFO shaft gen	2,9T/day
Idle in port 400kW IFO	2,3T/day
Boiler HFO	1.0T/day
Extra for reefers (100 integr 20')	
box x kW/box x HP/kW x	
gr/HP hr x hr =	
100 x 5,0 x 1,5 x 0,143 x 24 =	2,6T/day

TYPICAL LOADING CONDITION - DESIGN

Draft-design mld	9,94 M ~10 M
Displacement	34 687 Ton
Light ship	11000
Deadweight	23687 Ton
FO+DO+Lub +Dr w	
+prov +stores	2349
Waterballast	1089
Cargo containers	20249 Ton
14,2Ton per TEU = 14.2x	1426 TEU
homogeneous load	
KM	14,37 M
KG	13,85
GM	0,52
Free surface correction	0,06
GM'	0,46 M
FO	1429 Ton
DO	250
Lub oil	100
Fresh water	250
Sev tanks	95
Crew+prov +stores	225
	2349 Ton
Waterballast	1089
	3438 Ton
Cont below deck	10678T-14,2 x 752 TEU
Cont on deck	9571T-14.2 x 674
HOMOG LOADED	
CENTR. GRAVITY BOX - 45% 8'	20249T-14.2 x 1426

Length over all	182,50M
Length betw perpend.	174,00M
Breadth mld	30,50M
Depth mld	16,20M
Draft scantling EXTR	11,27M
Displacement	40760TON
Light ship ESTIM.	11000 -
Deadweight 11M	29760TON

Water ballast	7874 m3
Fresh water	411 m3
Heavy Fuel Oil	2883 m3
Diesel oil	366 m3
Lubricating oil	281 m3

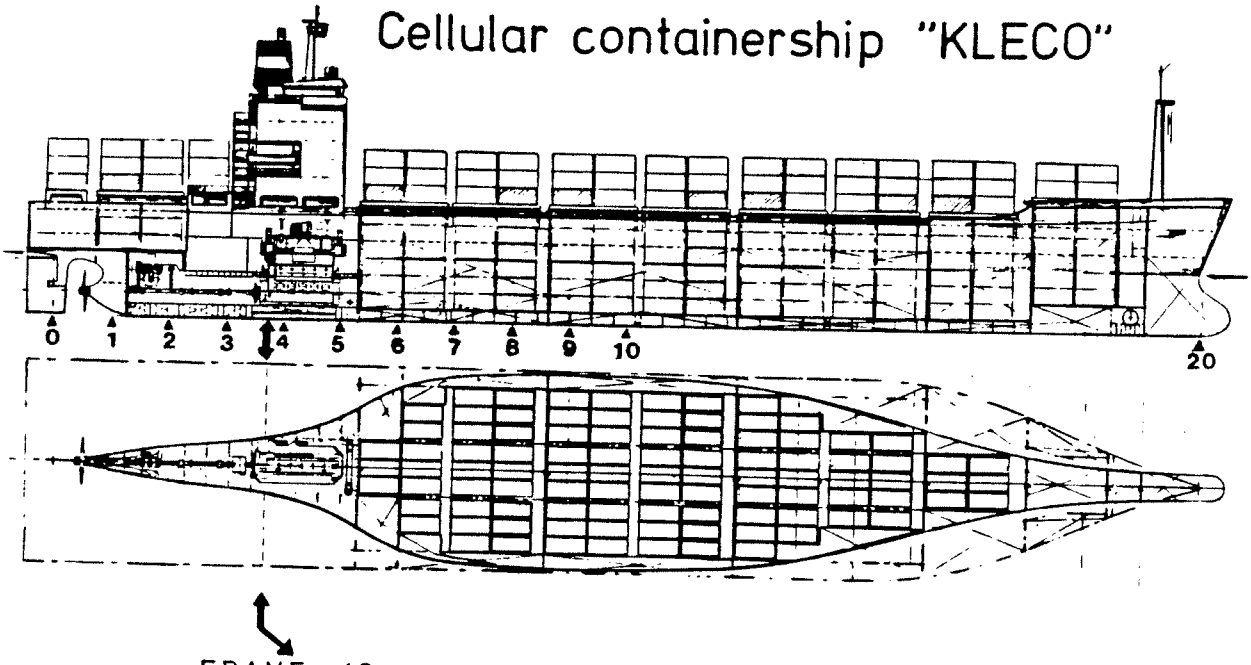
NEDLLOYD VAN NOORT 930
" **VAN DIEMEN** 931

Delivery date April 1984 / July 1984
Yard v/d Giessen de Noord
Class Bureau Veritas

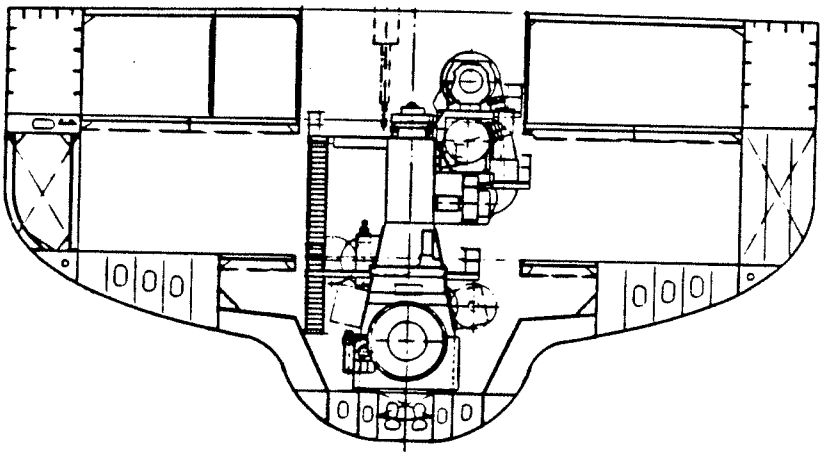
LORR 2-11-83

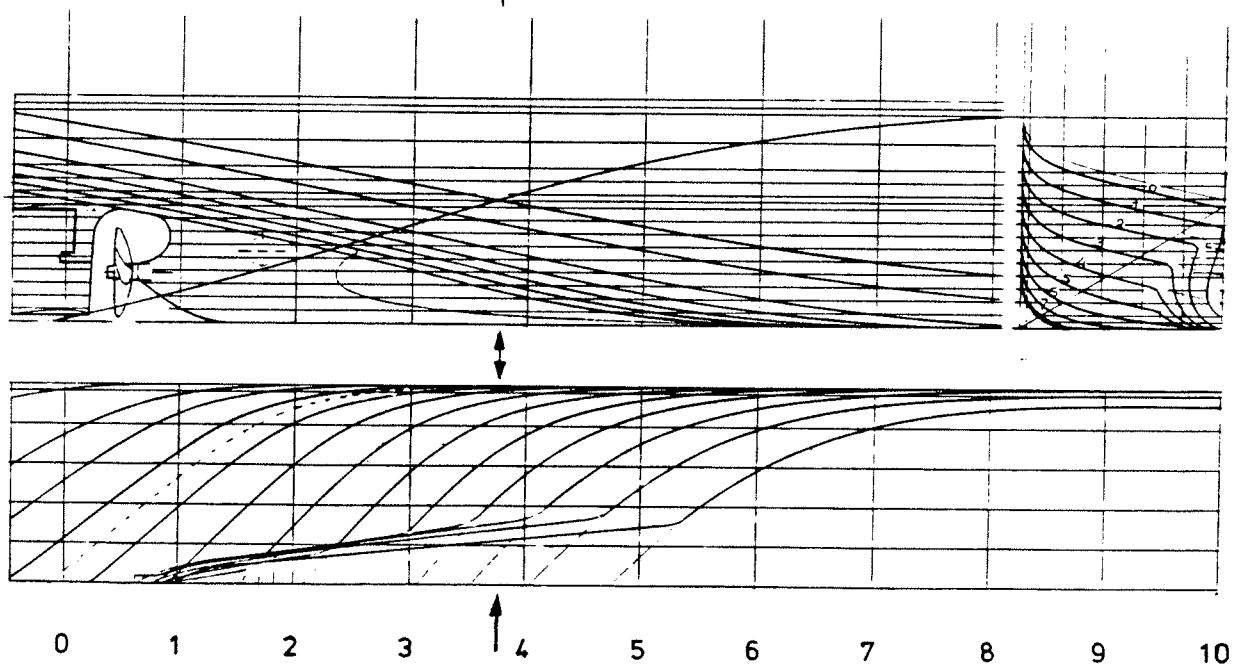
10-100

Cellular containership "KLECO"

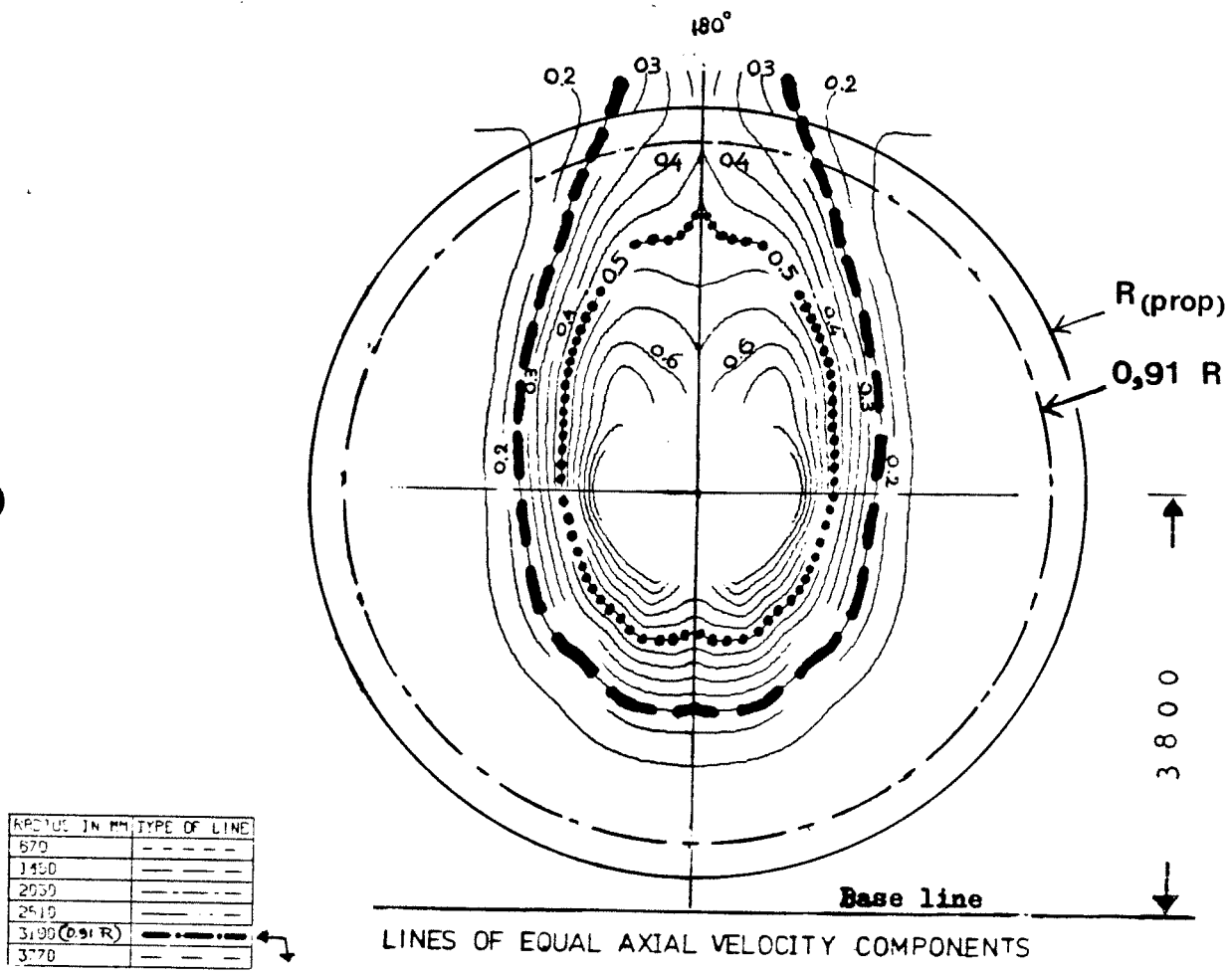


FRAME 42 AT ORDINATE 3,75

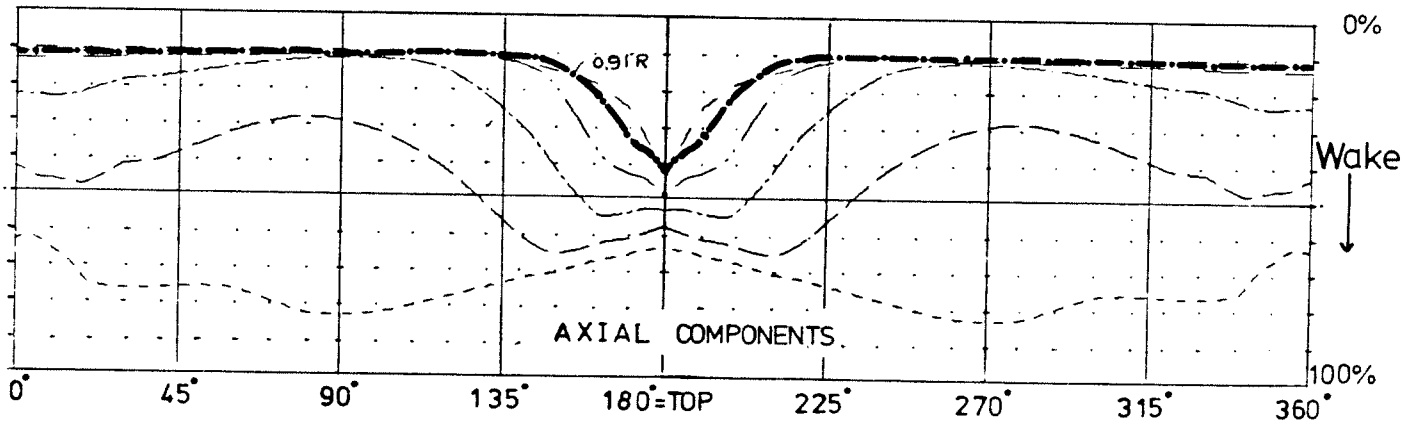


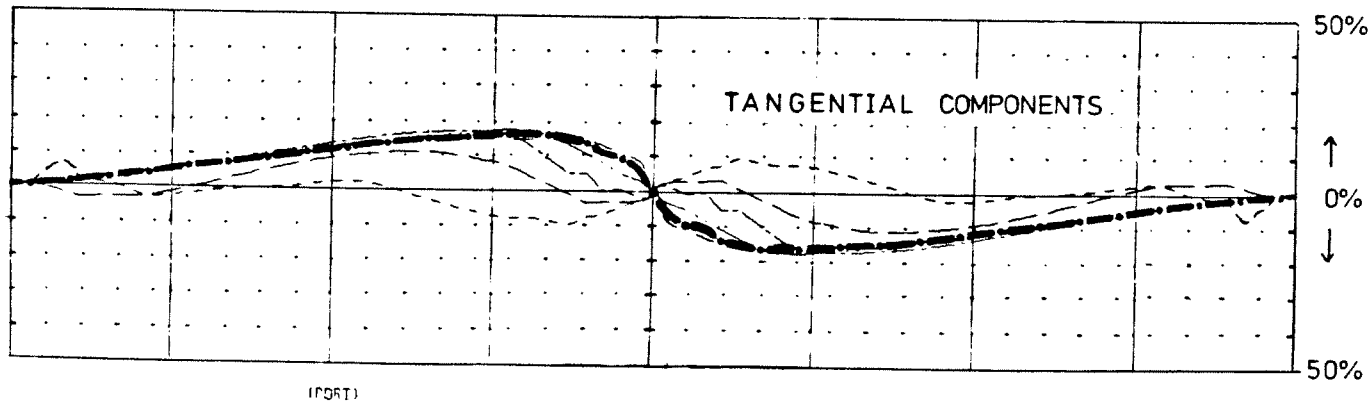


WAKE FIELD MEASUREMENTS
"Klein" containervessel ["KLECO"]
AT 10,00 m DRAFT.



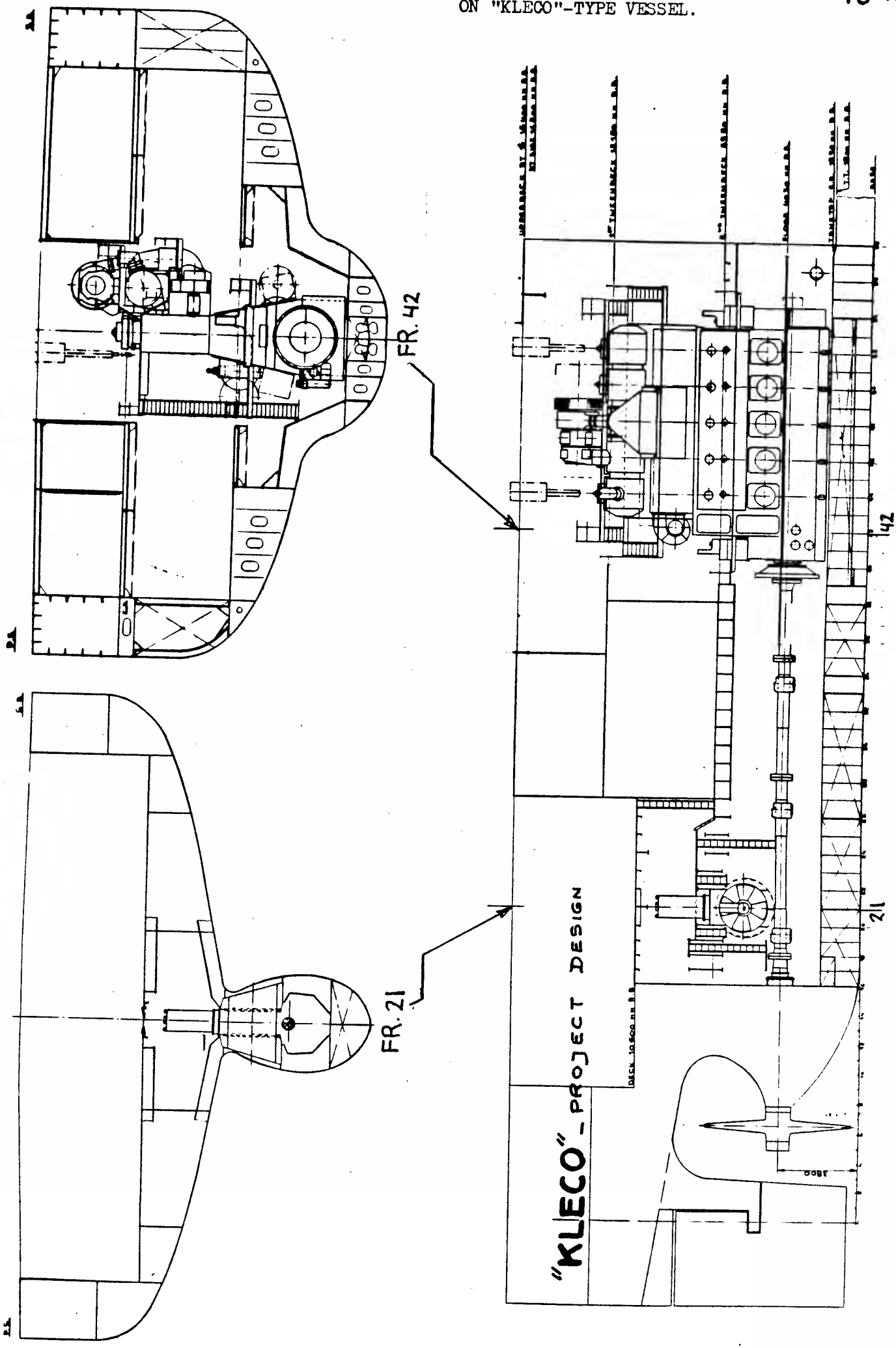
LINES OF EQUAL AXIAL VELOCITY COMPONENTS



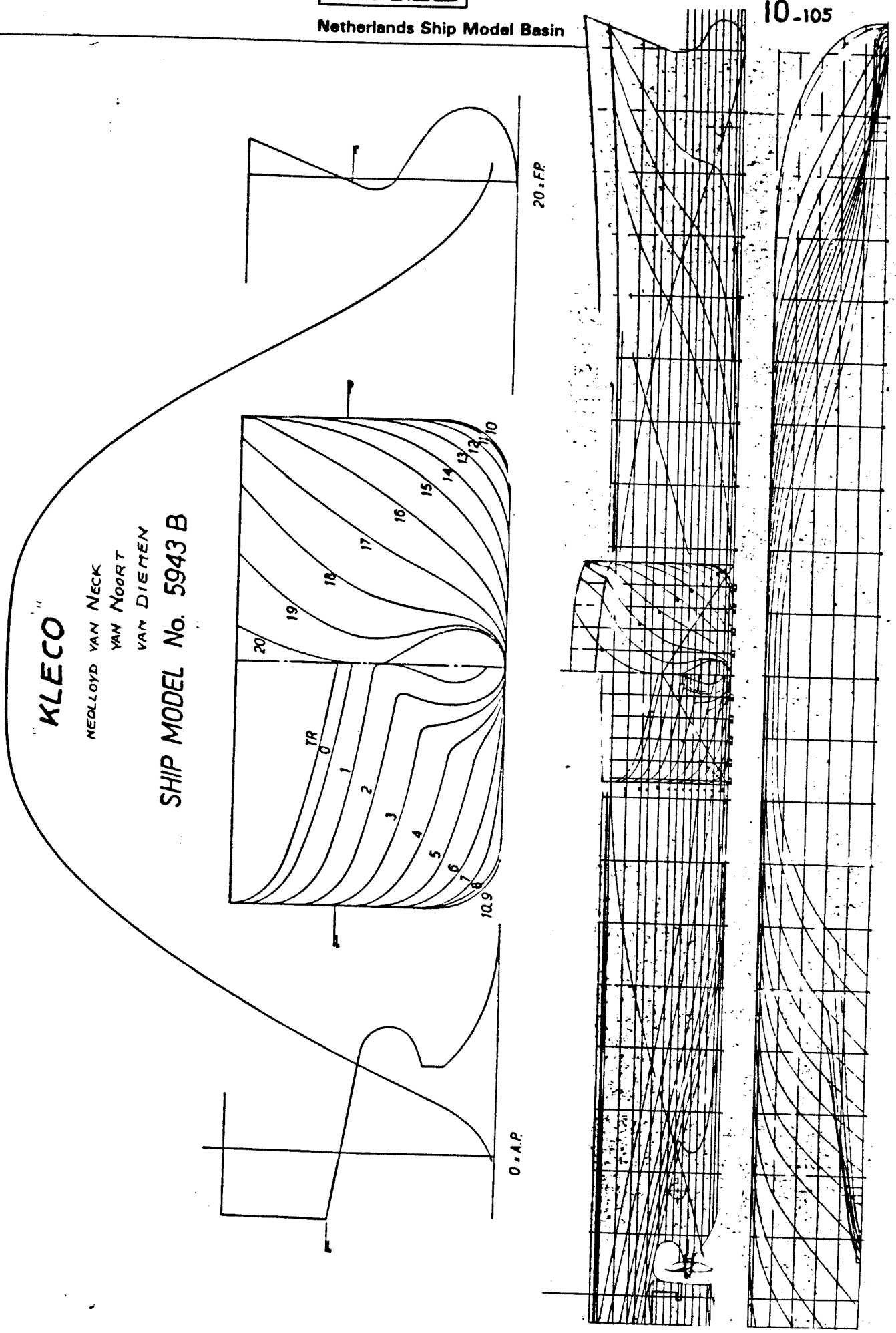


(DGRI)

POSITION OF CRITICAL FRAMES
ON "KLECO"-TYPE VESSEL.



P.S.
S.B.
FR. 41



DRAWING No M 5943_49

Particulars of ship:

Ship model No.		5943 B	5943 B
Length between perpendiculars	L_{PP}	174.00	174.00 m
Length on waterline	L_{WL}	177.58	169.33 m
Breadth moulded	B	30.50	30.50 m
Draft moulded on A.P.	T_A	10.00	7.20 m
Draft moulded on F.P.	T_F	10.00	5.50 m
Displacement volume moulded	∇	33926	19398 m^3
Displacement weight in sea water	Δ	34774	19883 m.t.
Wetted surface, bare hull	S_1	6706	- m^2
Wetted surface, rudder incl.	S_2	6815	5148 m^2
L.C.B. position aft of ½ L	Θ	0.97	- $\%L_{PP}$
Block coefficient	C_B	0.639	-
Midship section coefficient	C_M	0.980	-
Prismatic coefficient	C_P	0.652	-
Length-Breadth ratio	L_{PP}/B	5.70	-
Breadth-Draft ratio	B/T	3.05	-

Ship model material: wood.
Linear scale ratio λ = 29.
Turbulence induced by studs.

Propeller characteristics:

Propeller model		3937 (stock prop.)	5711 (design pitch)	5711 A (final pitch)
Diameter	D	6983	6800	6800 mm
Pitch at 0.7R	$P_{0.7}$	6244	6189	5880 mm
Pitch ratio	$P_{0.7}/D$	0.894	0.910	0.865
Expanded blade area ratio	A_E/A_O	0.619	0.556	0.556
Boss-Diameter ratio	d/D	0.162	0.169	0.169
Number of blades	Z	4	4	4

h_p

CONTROL TEST

PROPULSION TEST RESULTS

Test No. 42234

Model condition: SEE BELOW

Ship model No. 5943B
Propeller model No. 3937
Scale ratio λ = 29.00000

Draft moulded: on FP = 10.000 m; on AP = 10.000 m; mean = 10.000 m

Displacement volume ∇ = 33926 m³

Skin friction correction F_D from model to ship is according to ITTC

model-ship correlation allowance coefficient CA = .000100

Ship Speed in knots V_S	P_D	P_E x106	$\eta_D = \frac{P_E}{P_D}$	C_A	Revs/min n_S	Resistance R_T in tons	Thrust T in tons	$t = \frac{T-R_T}{T}$
14.50	6490	6880		500	74.5		58.247	
15.00	7198	7630		500	76.9		62.747	
15.50	7965	8440		498	79.5		66.997	
16.00	8729	9260		500	81.9		71.246	
16.50	9605	[illegible]		498	84.5		75.996	
17.00	10544	11165		496	87.1		80.996	
17.50	11642	12370		490	90.1		86.496	
18.00	12763	13530		487	92.8		92.495	
18.50	14007	14850		482	95.6		98.995	
19.00	15421	16350		474	98.6		105.995	

MODEL CONDITION:
BULBOUS BOW III
SCREW APERTURE II
RUDDER II
2 BOW THRUSTER TUNNELS II
STERN THRUSTER TUNNEL I AND FAIRINGS I

VALUES ARE GIVEN IN METRIC UNITS

P_D = delivered HP at propeller.
P_E = effective HP.
n_S = RPM of the ship's screw under the below conditions, not corrected for difference between the wake of ship and model, without any allowance

$C_A = \frac{\Delta_1^{2/3} V_S^3}{P_D \text{ metric}}$ Δ_1 = Displacement weight in salt water, in metric tons.

British HP = $\frac{75}{76}$ metric HP

Note: Results of self-propulsion tests as stated above refer to the self propulsion point of ship.
The results are calculated direct from measured model values without any allowance for appendages not present in the model, wind and sea; so values for ship are for tank condition corrected for salt water (γ_1 = 1.025) and 15° centigrade.
Model and ship dimensions are moulded.

PREDICTION

SHIP MODEL 5943B

PROPELLER 5711A

VALUES ARE DERIVED FROM RESULTS OF PROPULSION TEST No. 41909

DRAFT 10.00 m.

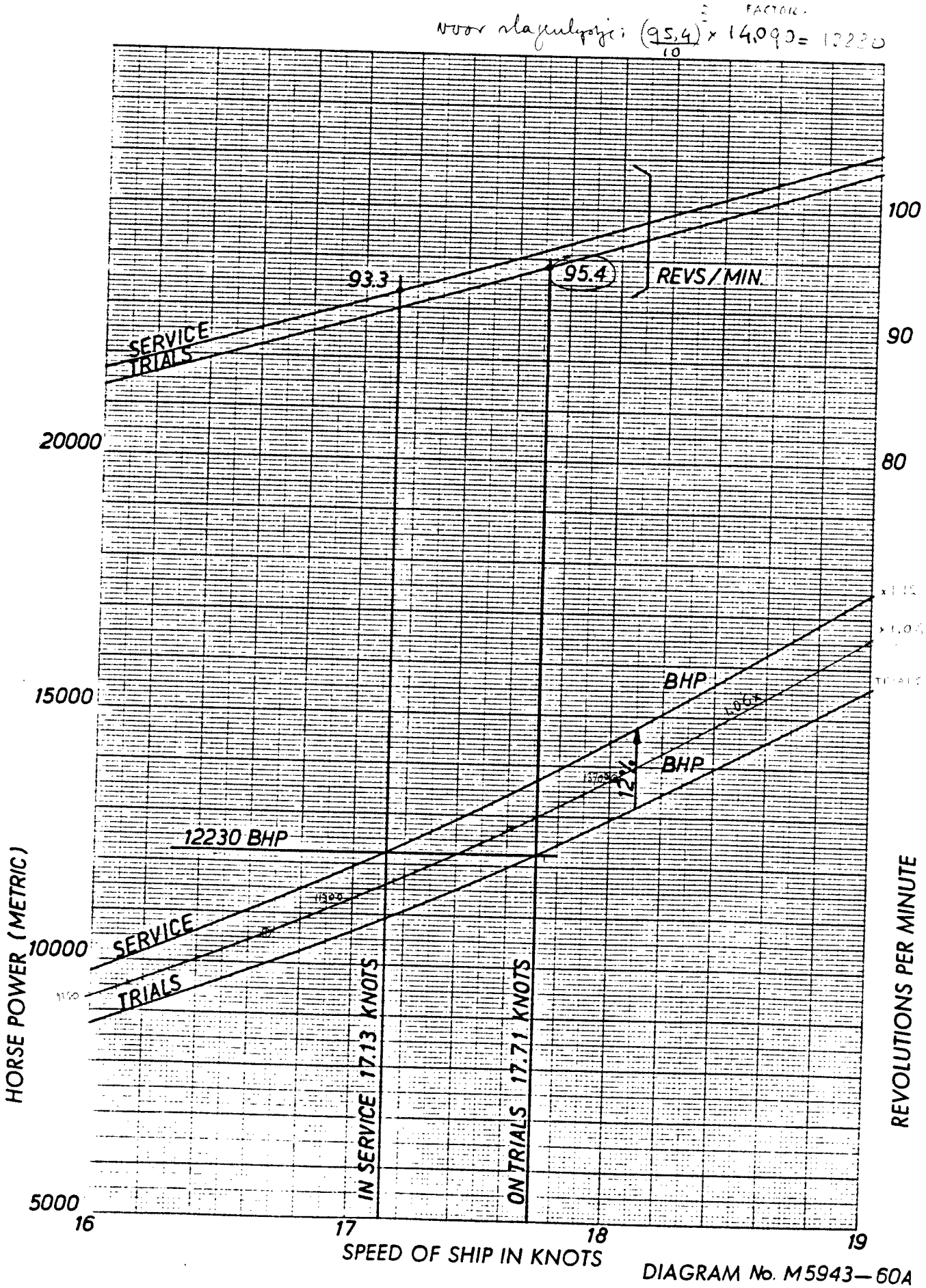


DIAGRAM No. M5943—60A

PREDICTION

SHIP MODEL 5943 B

PROPELLER 5711 A

VALUES ARE DERIVED FROM RESULTS OF PROPULSION TEST No. 41910

DRAFT 5.50 / 7.20 m.

MEAN 6.35

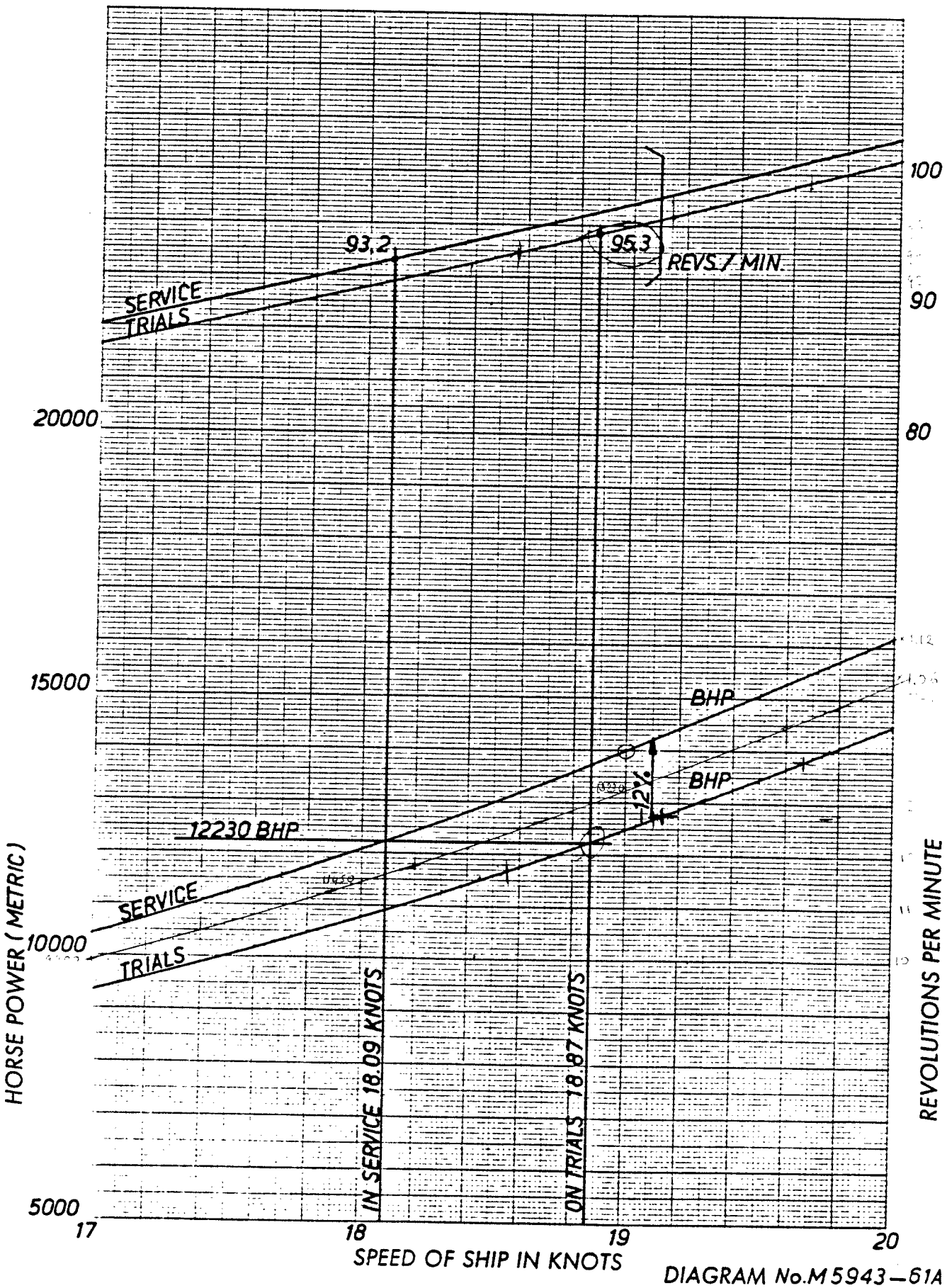


DIAGRAM No.M 5943—61A

PROPELLER DATA

DIAMETER		6800 MM
PITCH, MEAN		5723 MM
NUMBER OF BLADES		4
DIRECTION OF ROTATION		RIGHT-HANDED
EXPANDED BLADE AREA RATIO		0.547
MATERIAL		CUNIAL
MINIMUM TENSILE STRENGTH		647 N/MM²
MINIMUM ELONGATION		20 %
MASS	DRY	23000 KG
MASS MOM. OF INERTIA PROP. IN AIR	DRY	46575 KGM²
MASS MOM. OF INERTIA OF ENTR. WATER	DRY	16940 KGM²
ENGINE OUTPUT		13970 KW
PROPELLER REVOLUTIONS		94.00 RPM
CLASSIFICATION		BUR.VERITAS

22-7-83 Maalgwend Maart 1983

12240 SHP

PITCH SELECTED FOR 9000 KW 94.00 RPM, SERVICE CONDITION.

FOR TANKTESTING ONLY

012	PITCH	22-8-83	INITIALS
MODIFICATION	DESCRIPTION	DATE	

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HYDRODYNAMIC DESIGN DEPARTMENT

LIPS PROPELLER WORKS
DRUNEN, THE NETHERLANDS

F.P.PROPELLER FOR NEWBUILDING NRS 929-31

CLIENT V.D.GIESSEN-DE NOORD
ORDER F7856-859

DRAWN
CHECKED
APPROVED

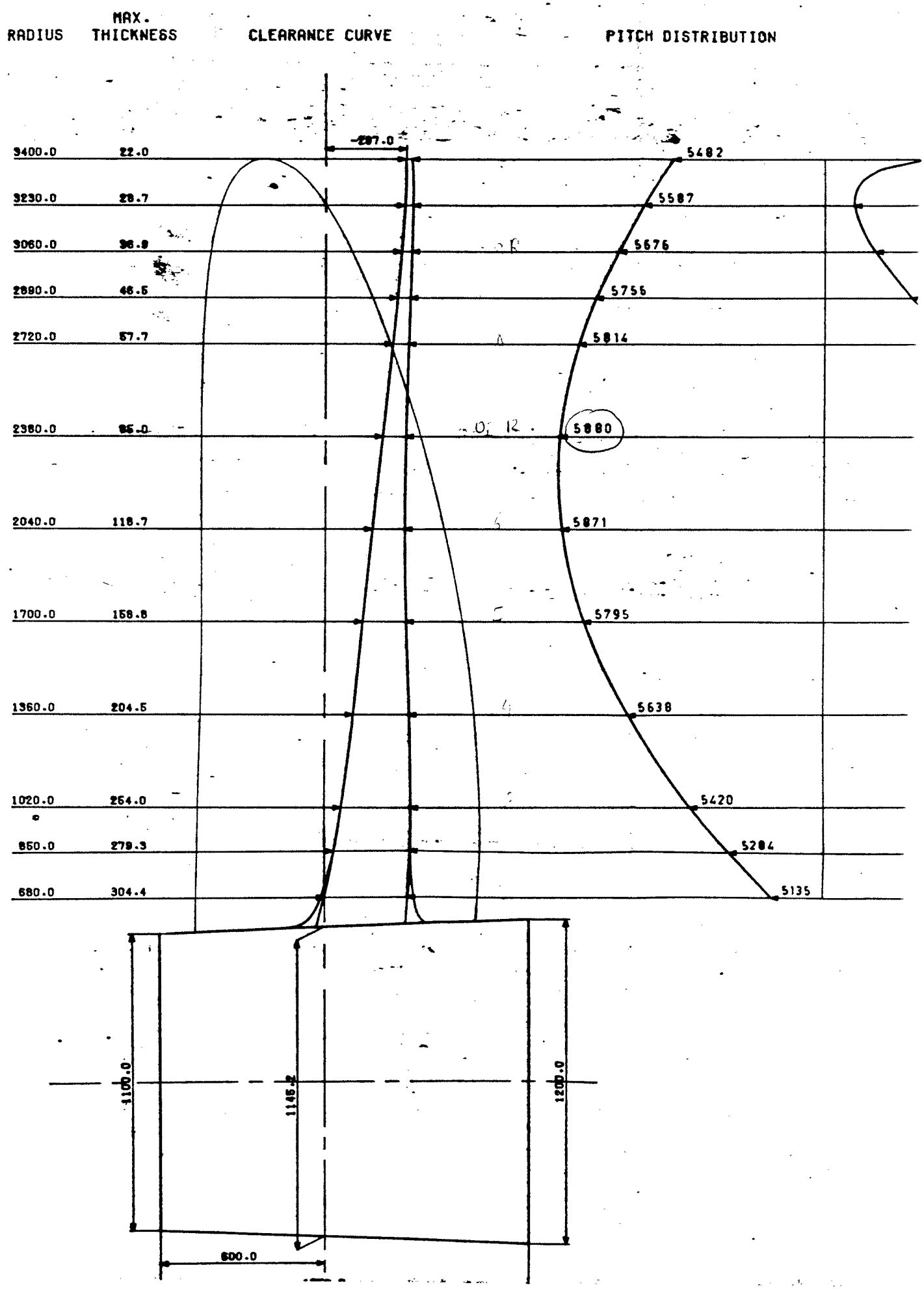
DATE Y/M/D 82/09/08

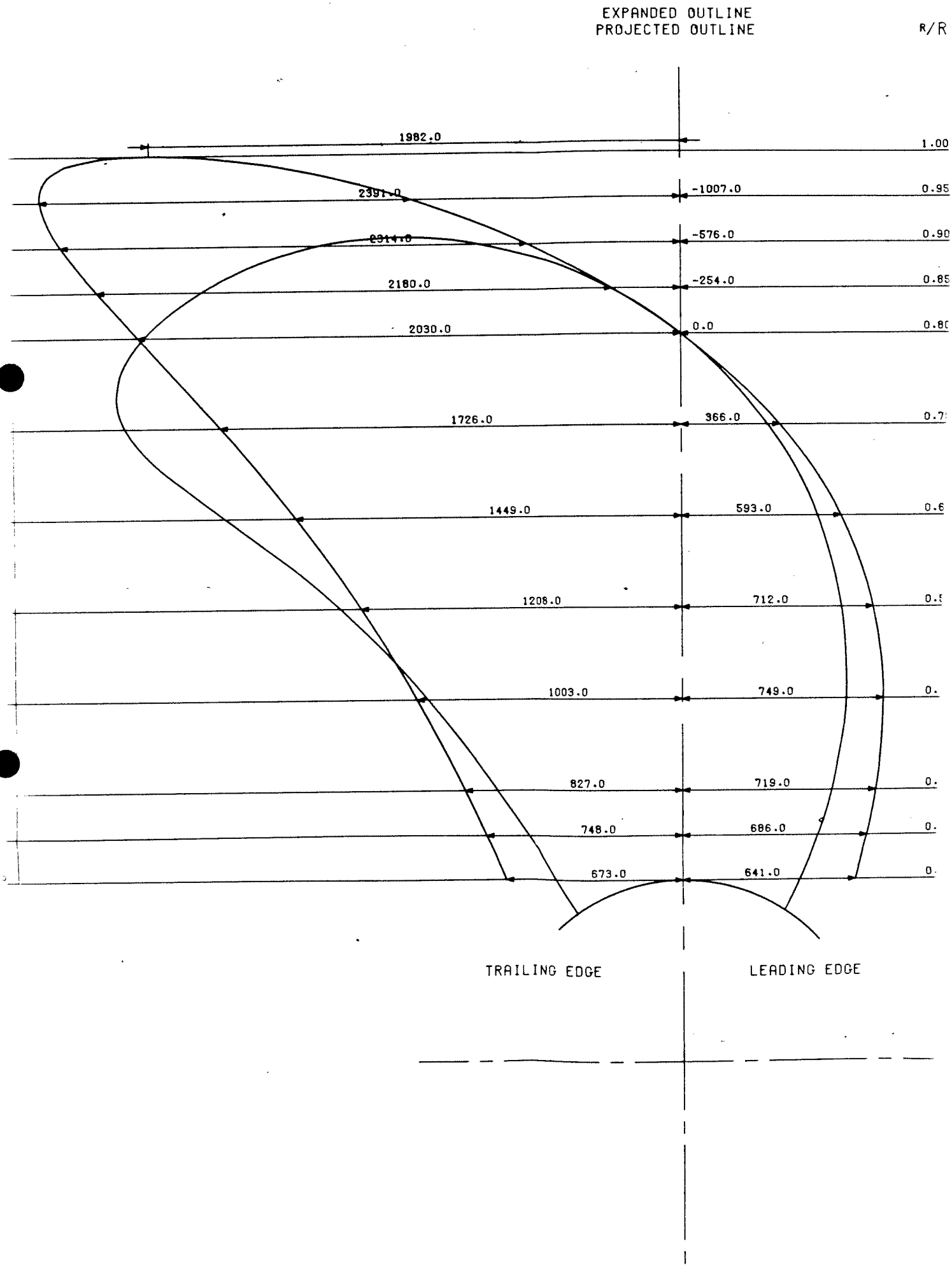
SCALE 1:12.5

DRAWING F7856-012

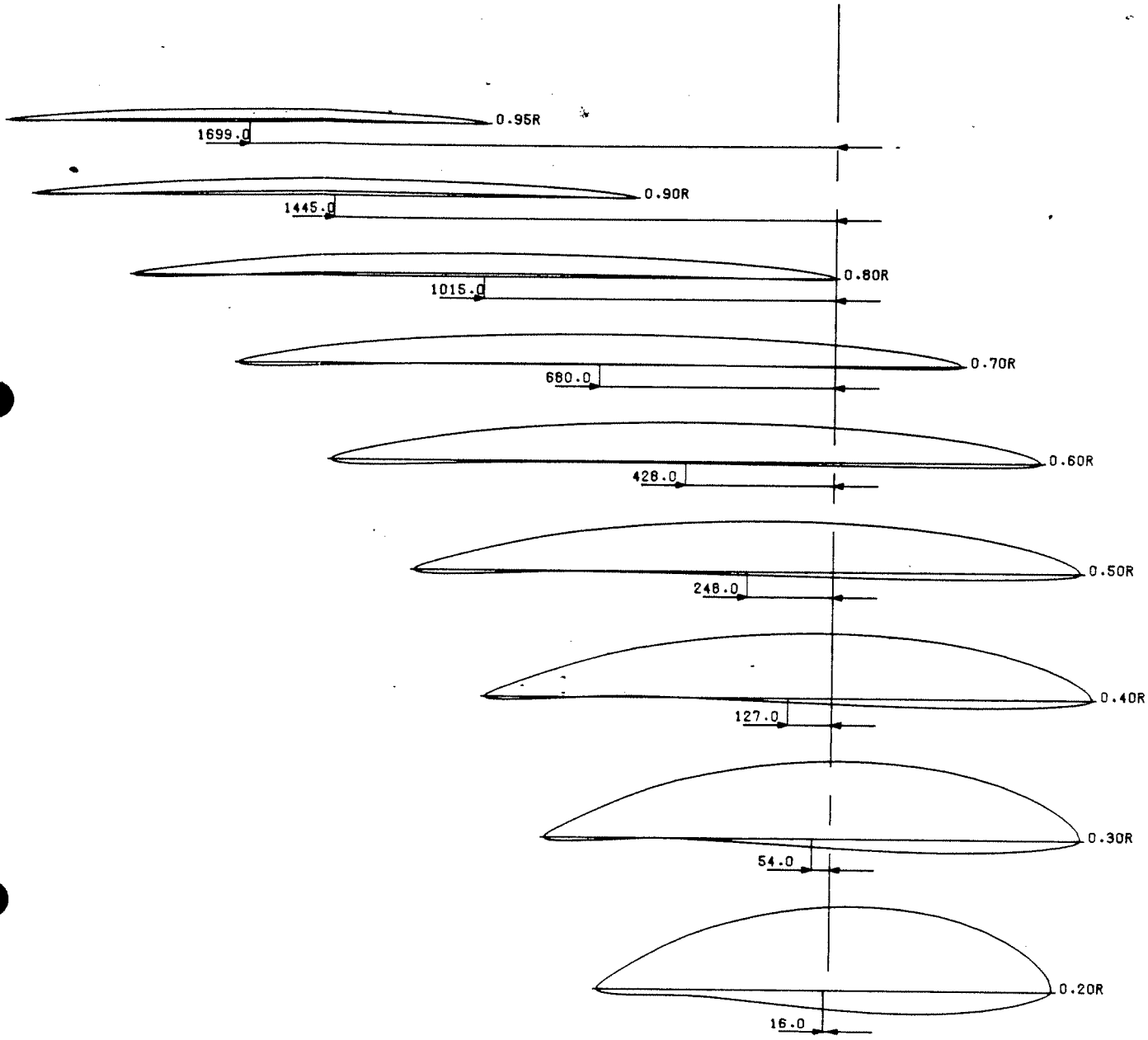
MODIFICATION

SIZE A1


RADIUS
MAX. THICKNESS
CLEARANCE CURVE
PITCH DISTRIBUTION
-287.0
3400.0 22.0 5482
3230.0 28.7 5587
3060.0 36.9 5676
2890.0 46.5 5756
2720.0 57.7 5814
2380.0 85.0 5880
2040.0 118.7 5871
1700.0 158.8 5795
1360.0 204.5 5638
1020.0 254.0 5420
850.0 279.3 5284
680.0 304.4 5135
1100.0
1145.2
1200.0
600.0


EXPANDED OUTLINE
PROJECTED OUTLINE
R/R
1982.0
1.00
2391.0
-1007.0
0.95
2314.0
-576.0
0.90
2180.0
-254.0
0.85
2030.0
0.0
0.80
1726.0
366.0
1449.0
593.0
1208.0
712.0
1003.0
749.0
827.0
719.0
748.0
686.0
673.0
641.0
TRAILING EDGE
LEADING EDGE

BLADE SECTIONS



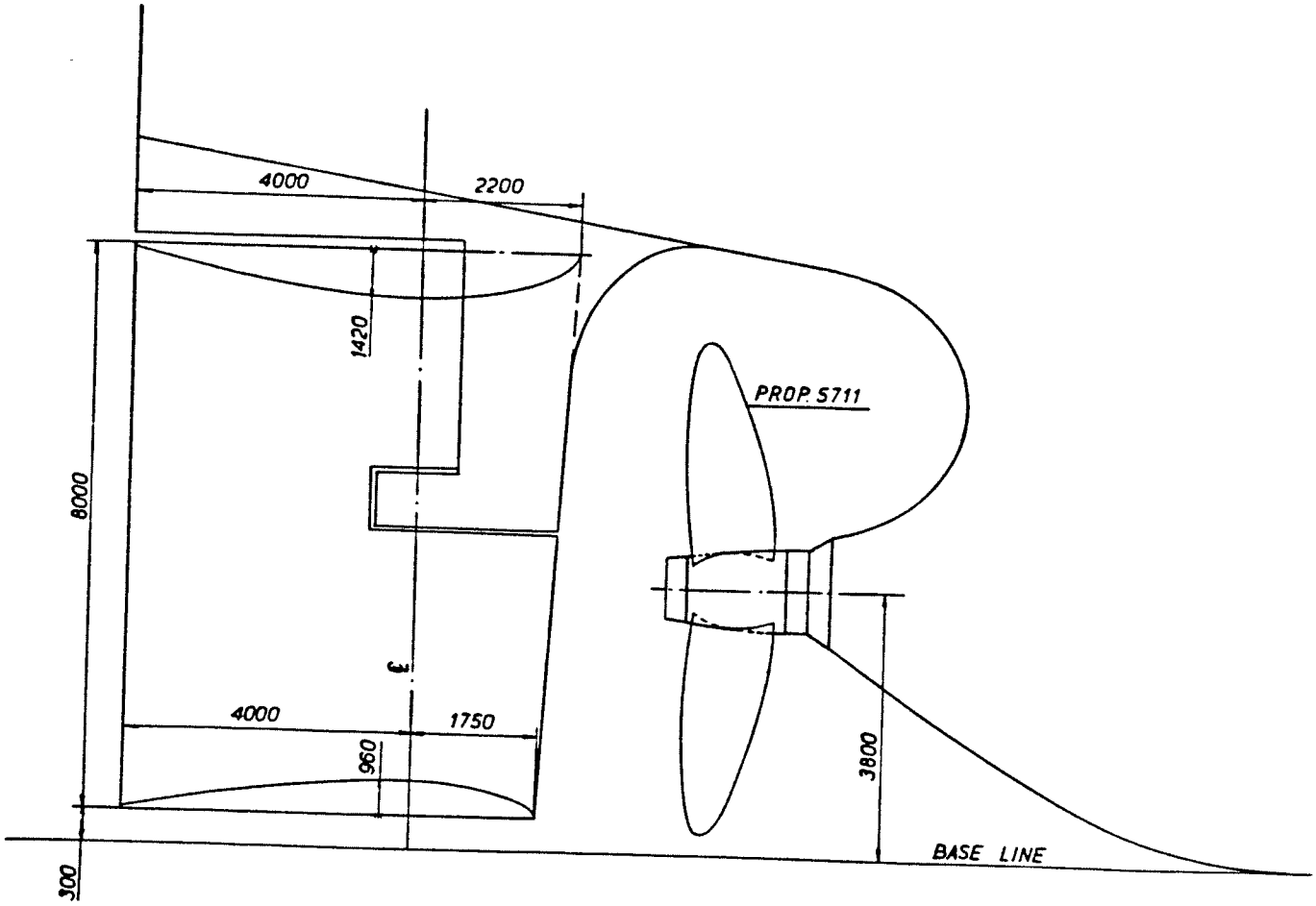
PROPELLER DATA

DIAMETER	6800 MM
PITCH, MEAN	5723 MM
NUMBER OF BLADES	4
DIRECTION OF ROTATION	RIGHT-HANDED
EXPANDED BLADE AREA RATIO	0.547
MATERIAL	CUNIAL
MINIMUM TENSILE STRENGTH	647 N/MM²

22-7-83

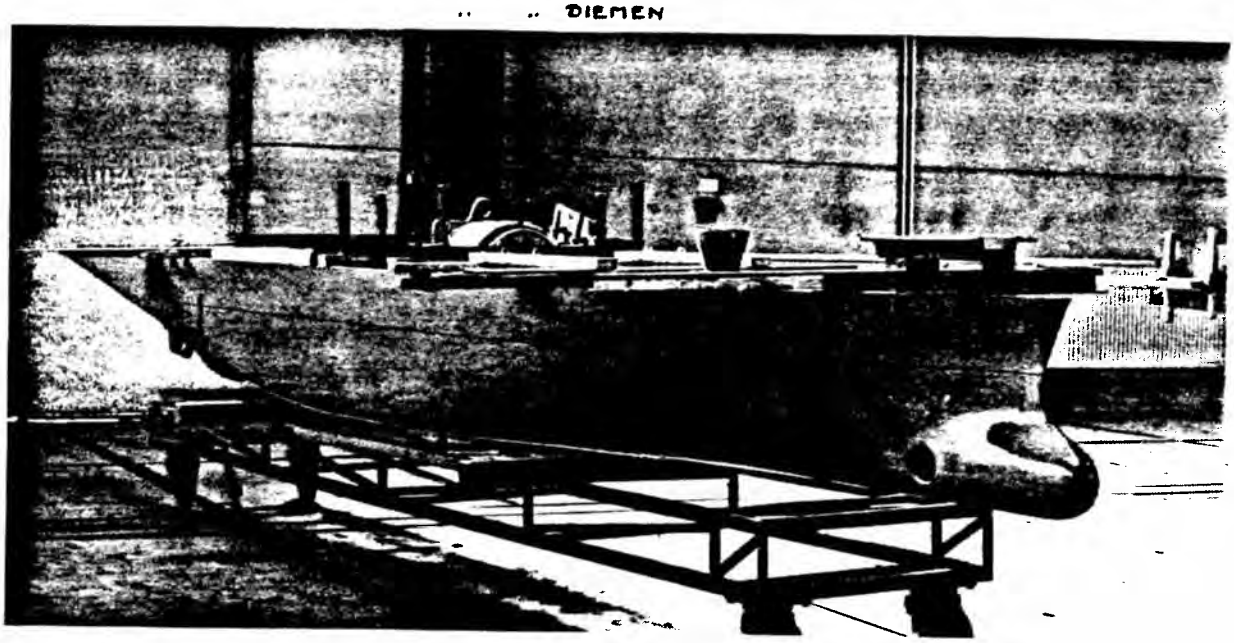
SHIP MODEL No. 5943B

RUDDER II AND SCREW APERTURE II



DRAWING No M 5943_33

"KLECO"
SHIP MODEL
NEDLL V NECK
" " NOORT
" " DIEMEN



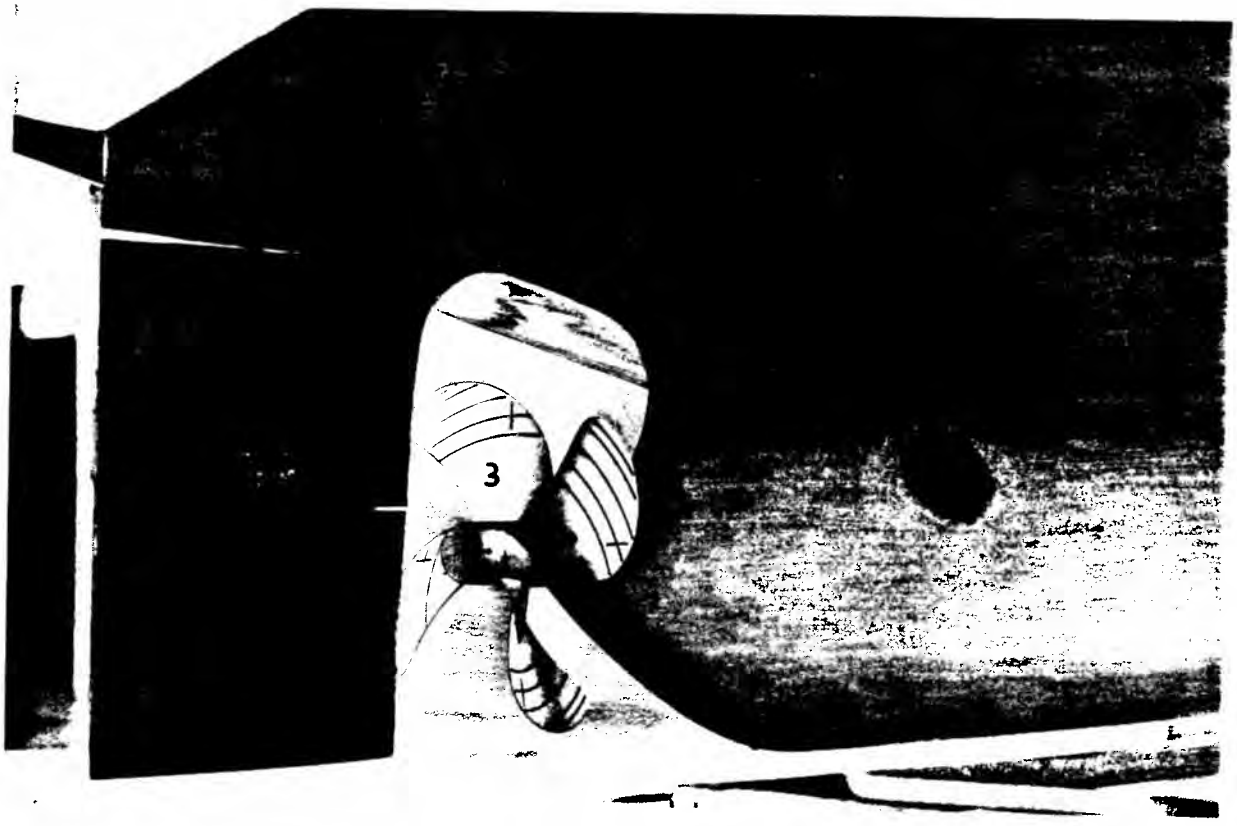
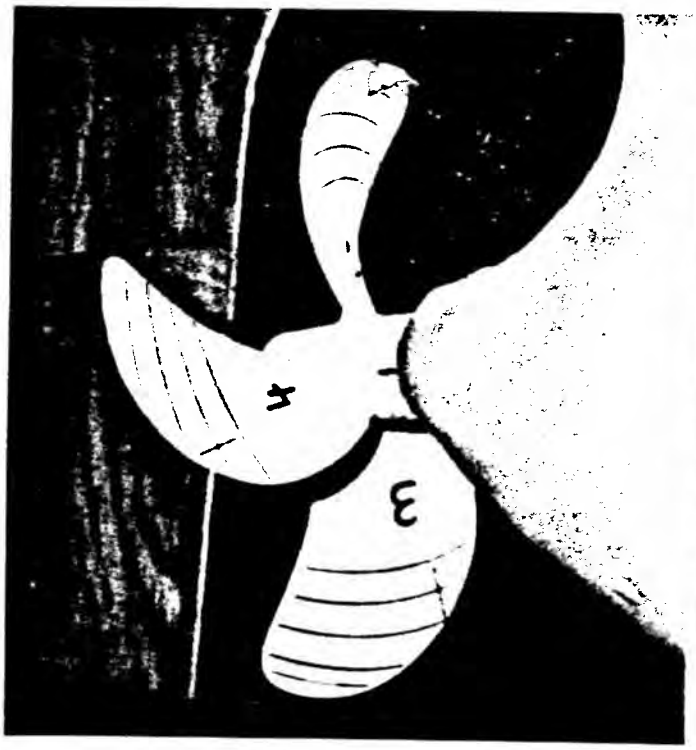


FIG 2.2 SHIP MODEL No. 5943B WITH PROPELLER MODEL No. 5711.

CAVITATION OBSERVATIONS

TEST No. 82-661
SHIP MODEL No. 5943 B
PROPELLER MODEL No. 5711
SHIP DRAUGHT ON F.P. 10.00 m } LOADED
SHIP DRAUGHT ON A.P. 10.00 m

V_s = 17.43 KNOTS
N_s = 1.532 Hz
P_s = 9525 kW



24°

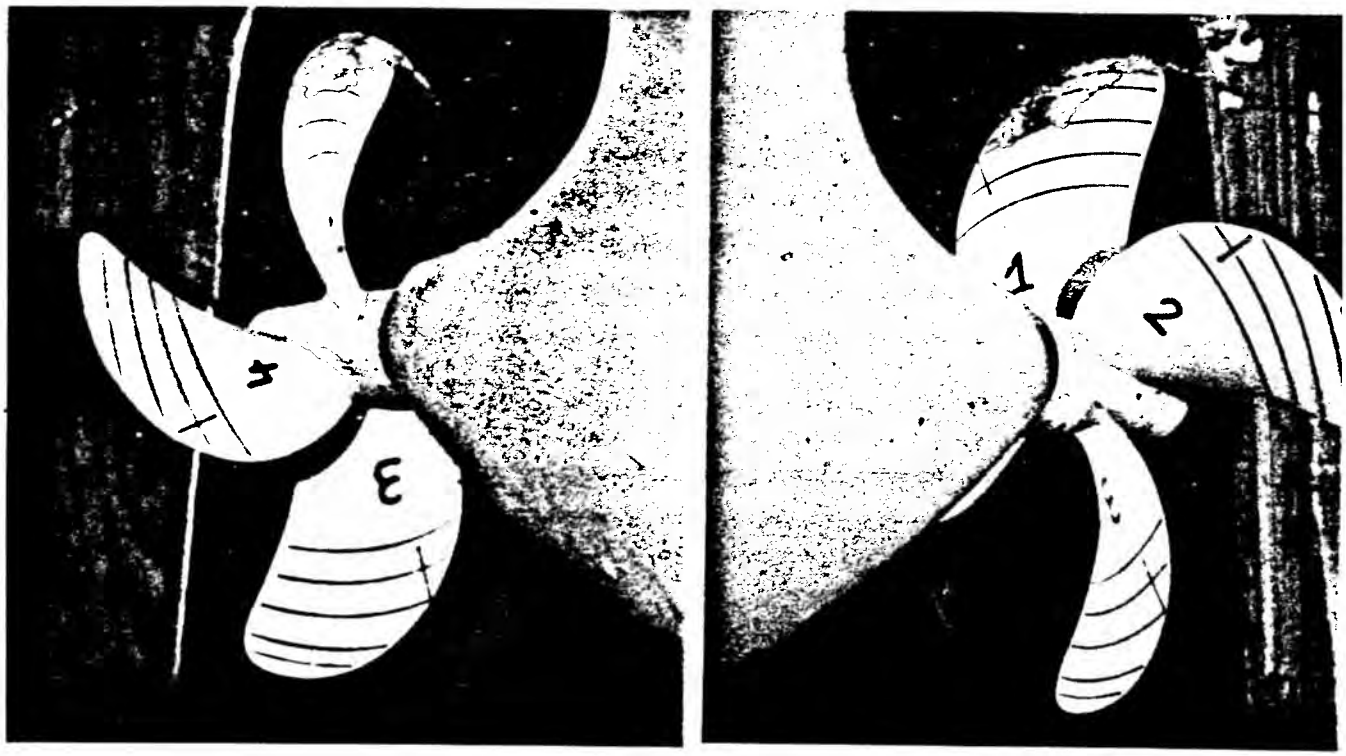
30°

FIG. 3.3 SUCTION-SIDE CAVITATION OBSERVATIONS.
ANGLES INDICATE POSITION OF BLADE No.1 RELATIVE TO TOP POSITION

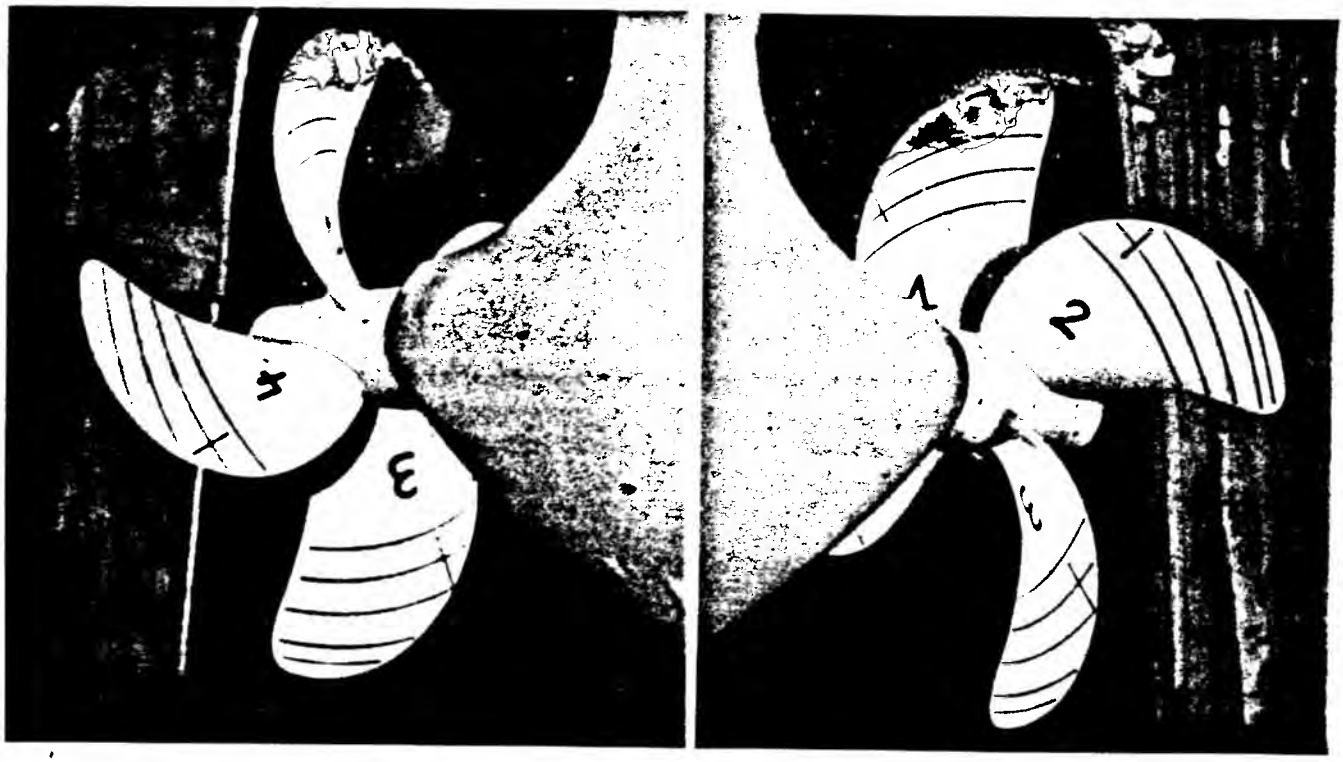
CAVITATION OBSERVATIONS

TEST No. 82-660
SHIP MODEL No. 5943B
PROPELLER MODEL No. 5711
SHIP DRAUGHT ON F.P. 5.50 m } BALLAST
SHIP DRAUGHT ON A.P. 7.20 m

V_s = 18.48 KNOTS
N_s = 1.535 Hz
P_S = 9525 kW



24°

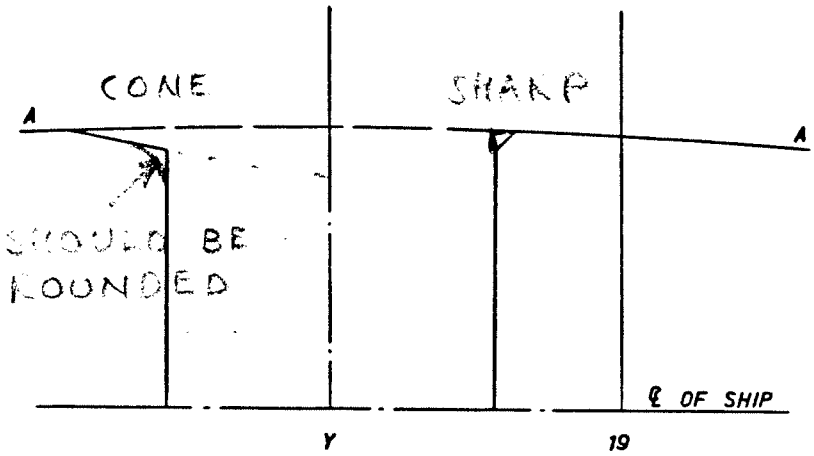


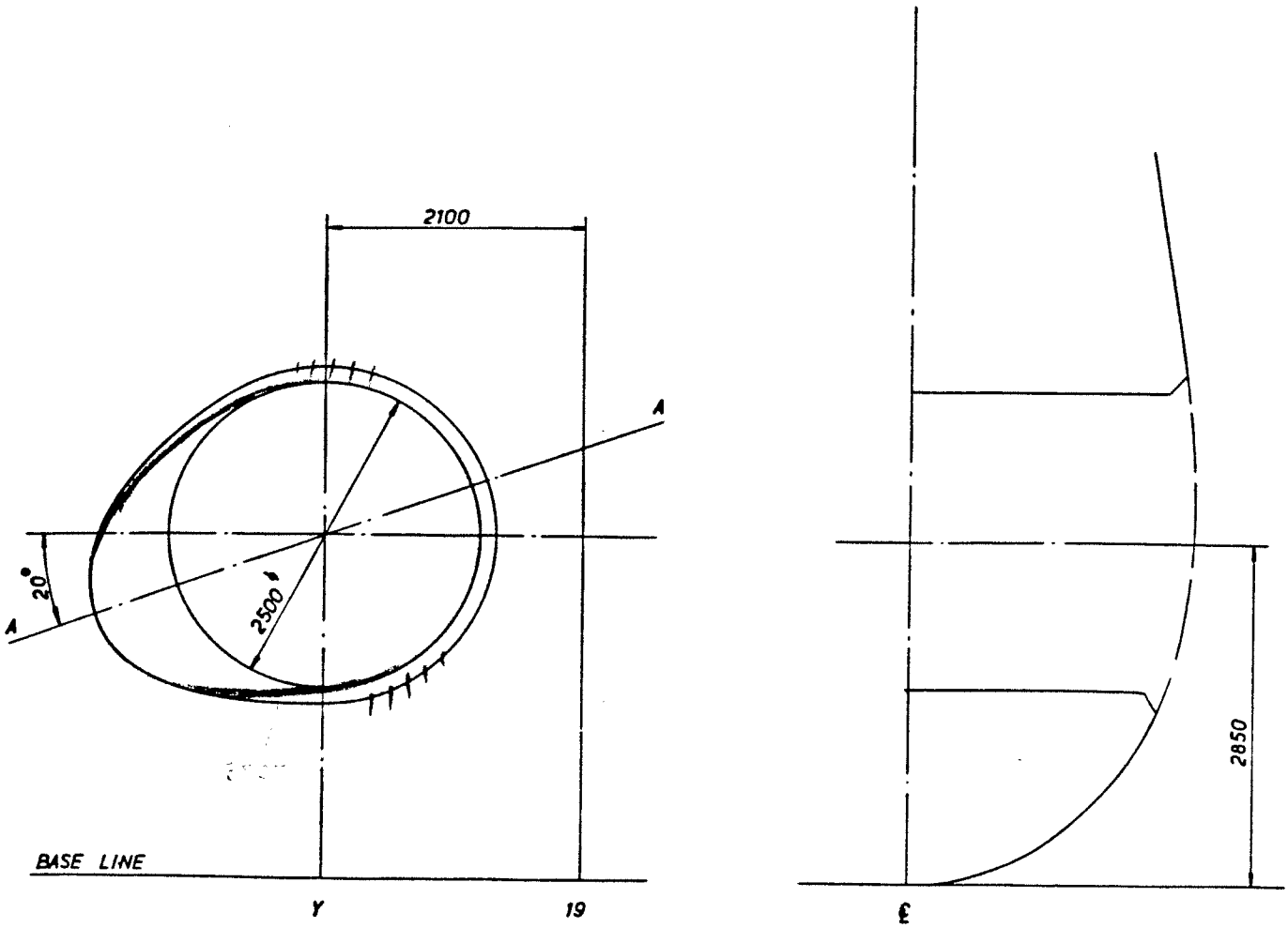
30°

FIG. 3.8 SUCTION-SIDE CAVITATION OBSERVATIONS.

SHIP MODEL No. 5943 B

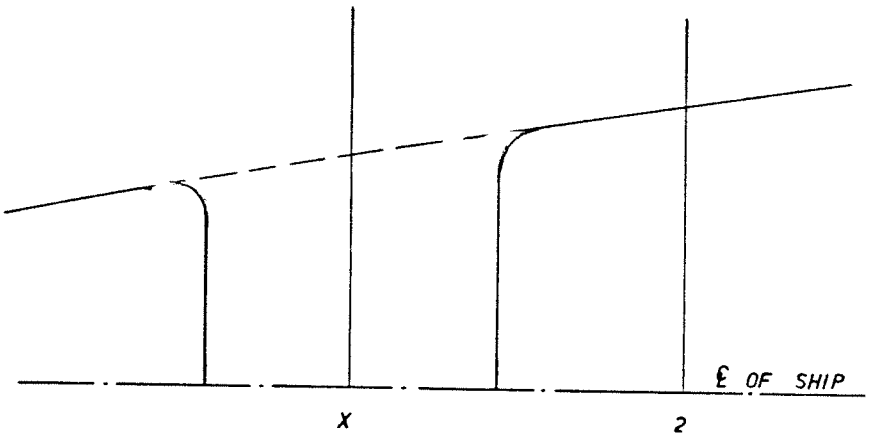
BOW THRUSTER TUNNEL II AND FAIRINGS II

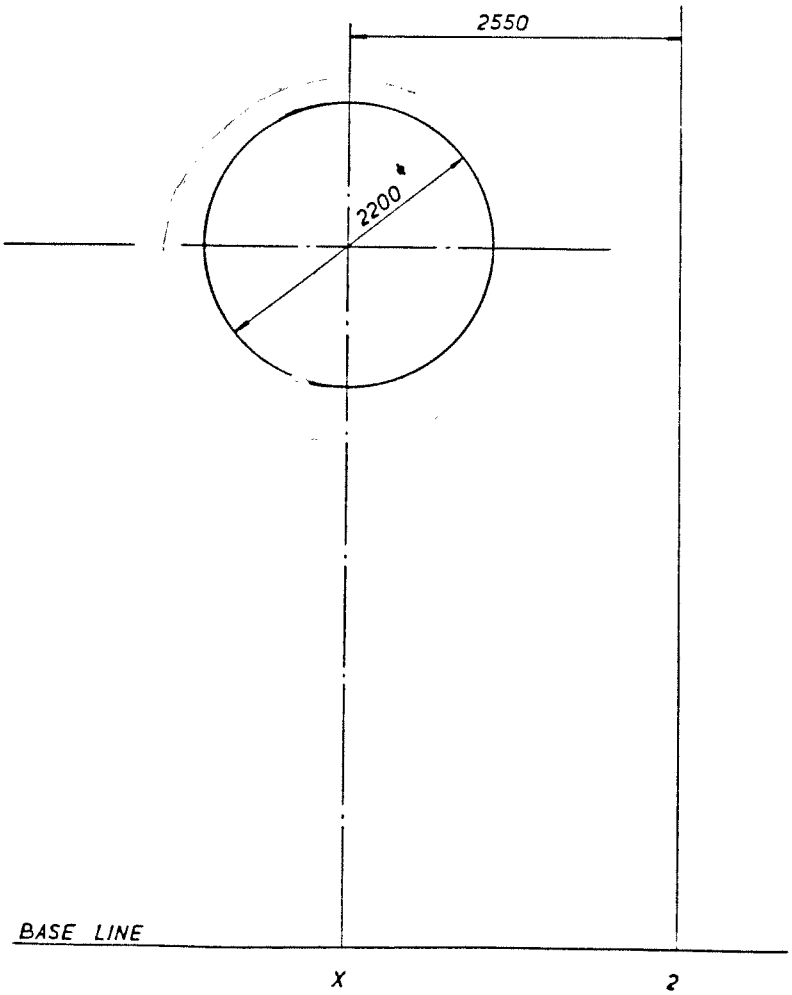


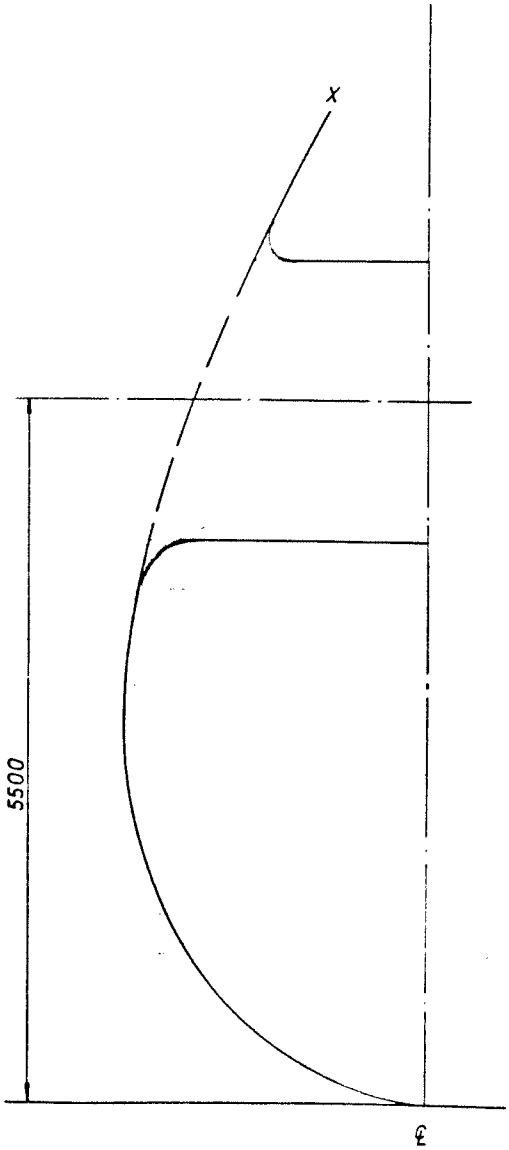


SHIP MODEL No. 5943 B

STERN THRUSTER TUNNEL I AND FAIRINGS I





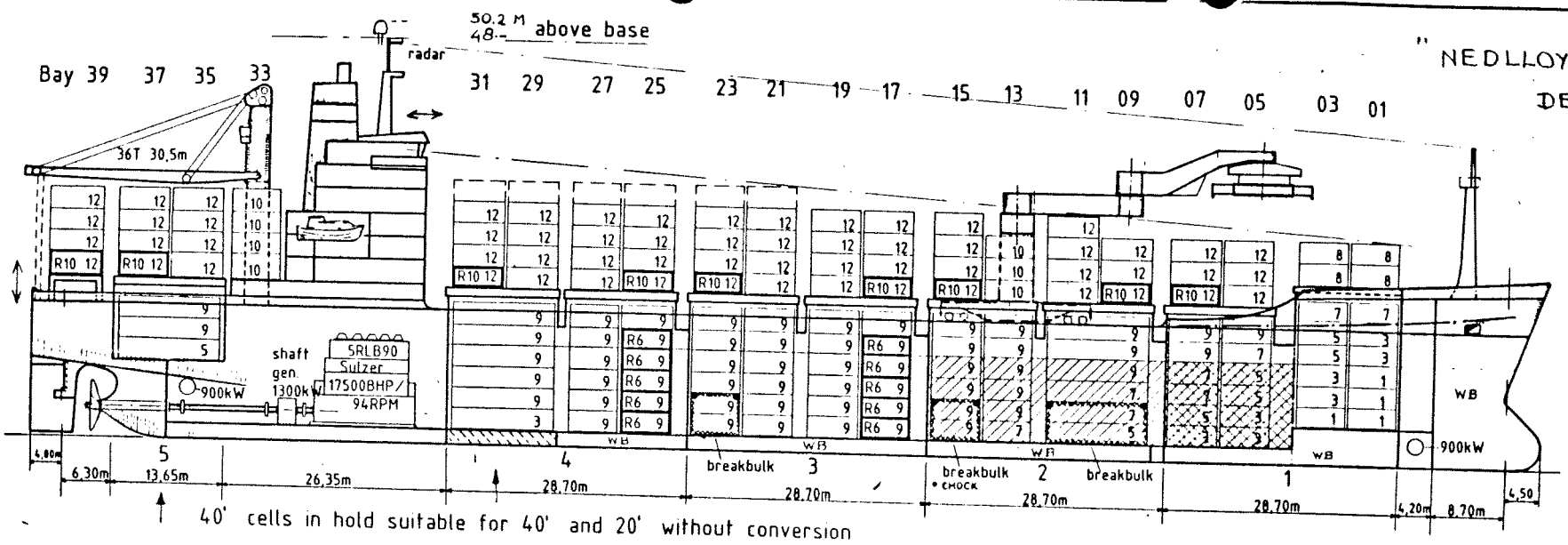


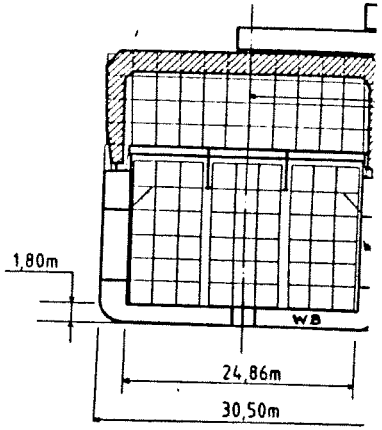
DRAWING No. M 5943_36

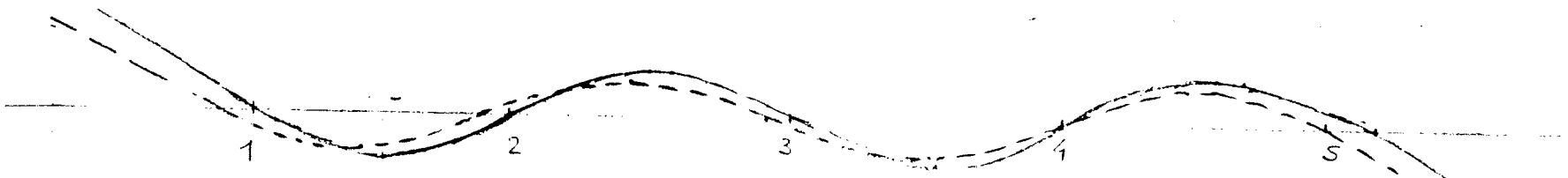
"NEDLLOYD"

DEC 1983

(1)







—— BALLAST 5.4 Hz

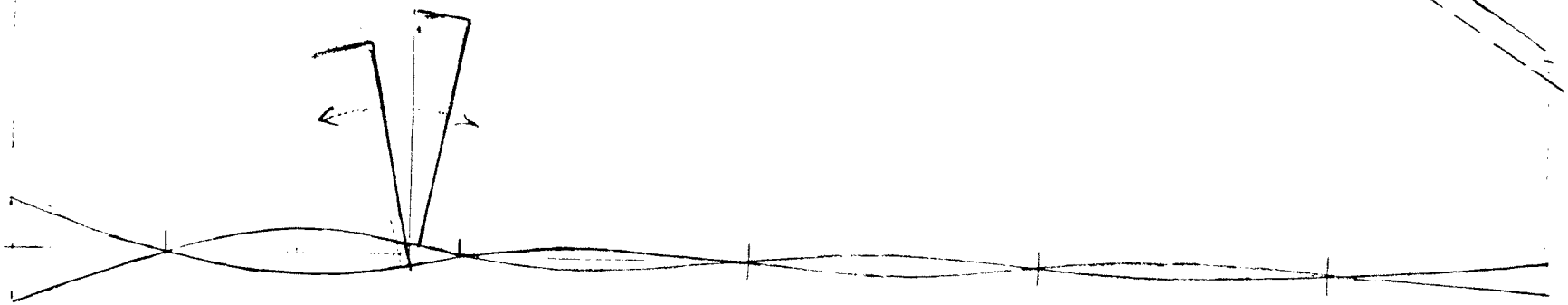
- - - LOADED 4.7 Hz

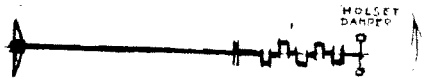
CALCULATION BY B.V.

5 x 57-62 RPM

VIBRATION OF HULL

5 NODE VERTICAL.





TORSIONAL VIBR : 4.7 HZ

AXIAL VIBRATION : 5.2 HZ

AXIAL DAMPER TO BE INSTALLED.

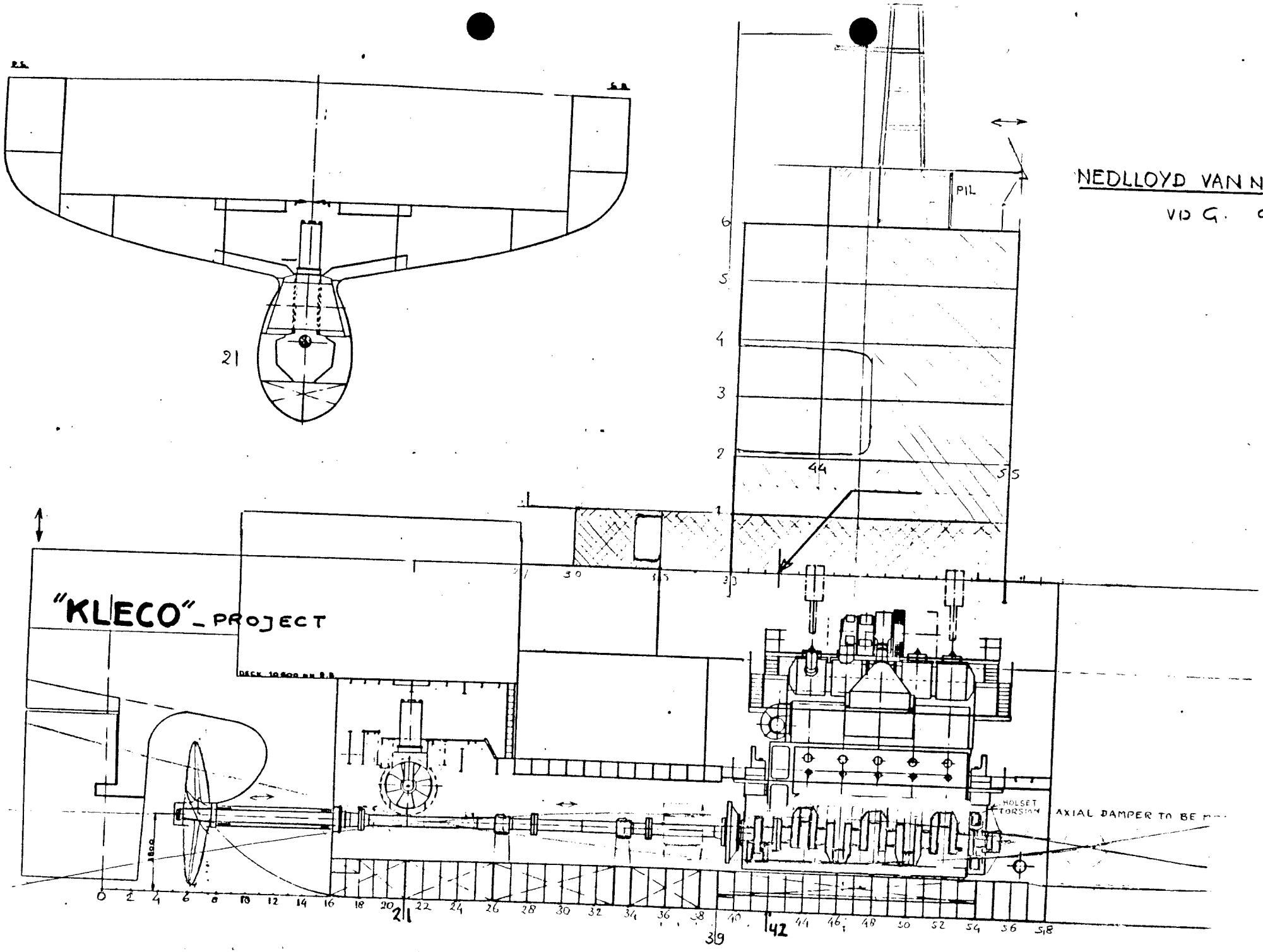
"NEDLLOYD VAN NECK"

10-12

(2)

NEDLLOYD VAN NECK

VID G. 929.

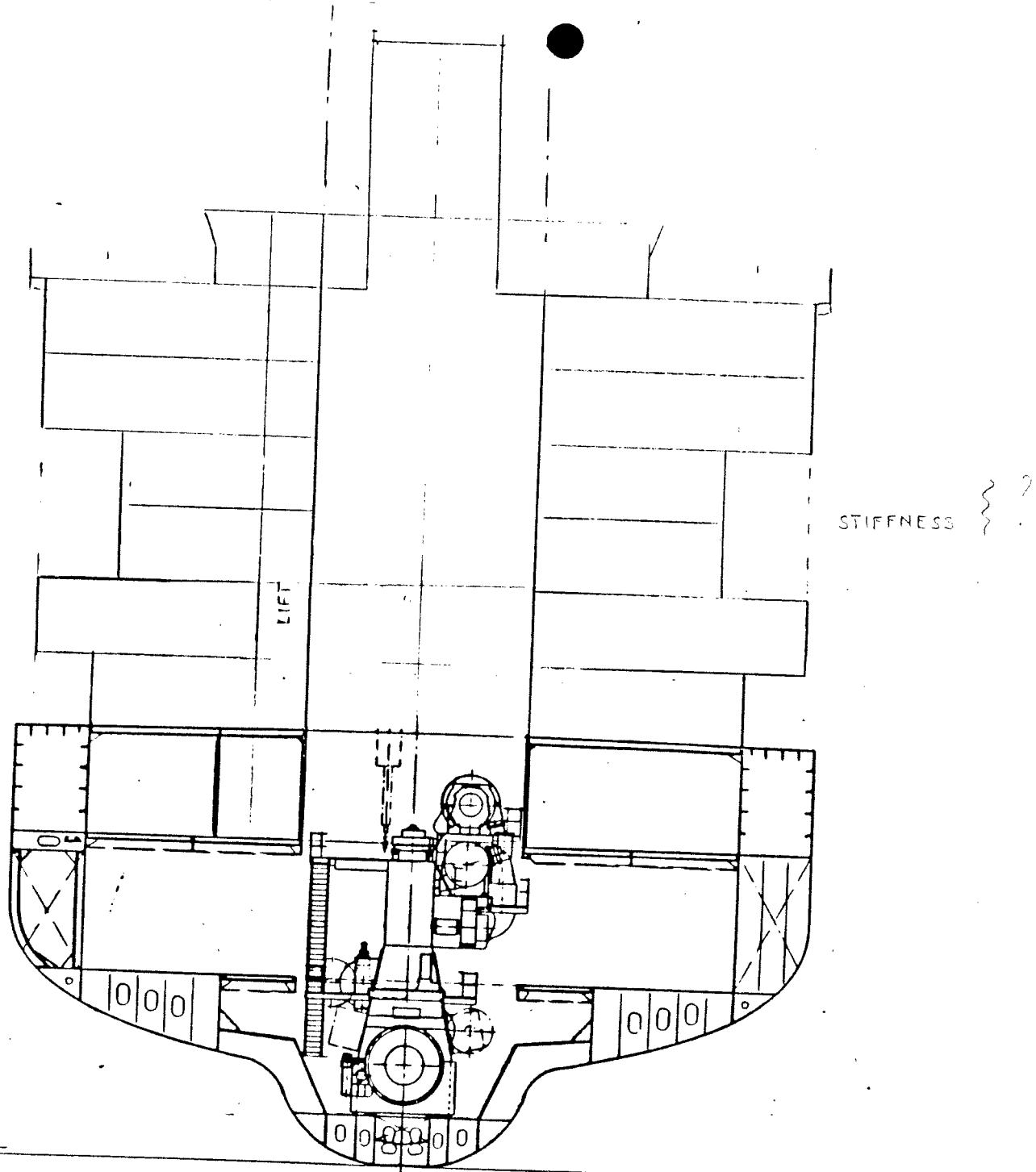


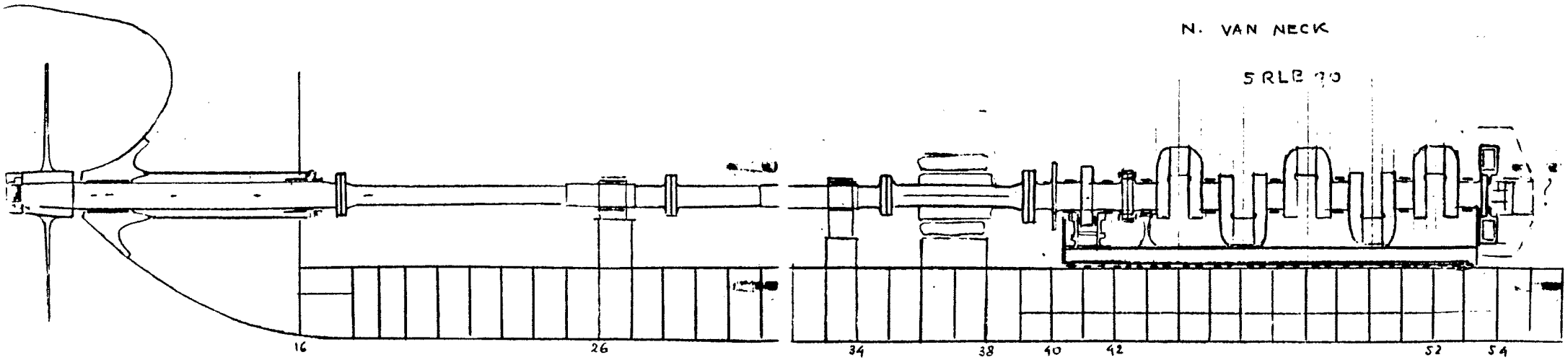
POSITION OF CRITICAL FRAMES ON "KLECO"-TYPE VESSEL.

10-121

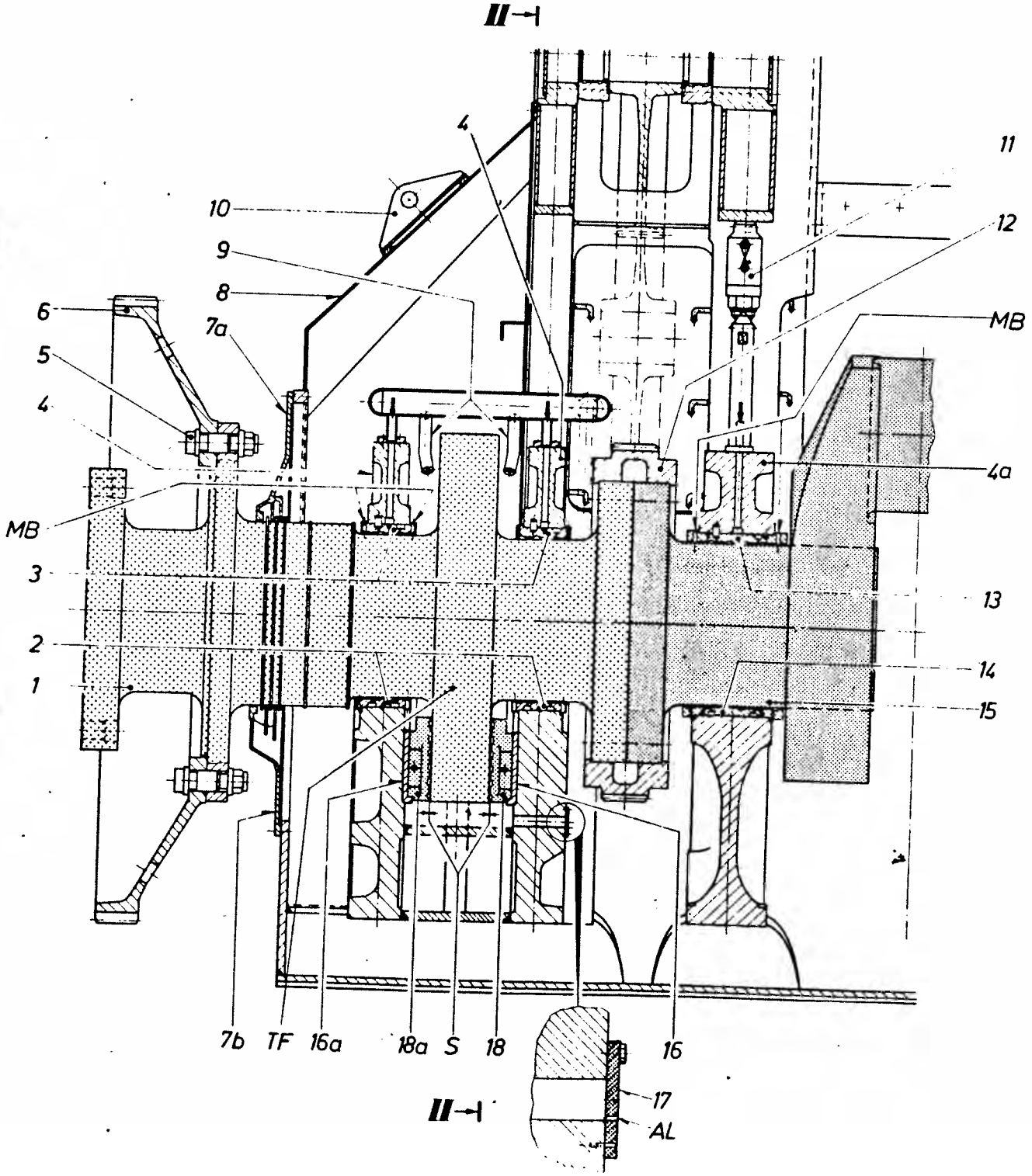
13 DEC 83

(3)




N. VAN NECK
5 RLB 70
16
26
34
38
40
42
52
54

4÷6 RL 90


I-I
II-I
4
11
10
12
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8
6
7a
MB
5
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4a
MB
3
13
2
14
1
15
7b
TF
16a
18a
S
18
16
17
AL
II-I

mmss NEDLLOYD VAN NECK/VAN NOORT/VAN DIEMEN ..... FUEL CONSUMPTION

at various R.P.M. & DRAUGHTS

(NORMAL FUEL NOZZLES)

10-1

SLAGEN LIJSTJE - BRANDST. VERBR

NEDLLOYD VAN NECK
" " NOORT
" " VAN DIEMEN

NSMB MODEL 5343 B PROP 5711A
PROPELLER LIPS 4BL
ϕ = 6800 H_{mean} = 5723 $H_{a/2}$ 5880

NORMAL FUELNOZZLES

6% EXTRA HP ABOVE TRIALS

BV 17500 x 85%
24-11-83.

	RPM	SHP PROPELL + 860 = BHP ENGINE		g/HP-HR HFO	KG/HR	TON/DAY ↓	→ 8.-M KN	TON/1000'	8.5 KN	TON/1000'	9.- KN	TON/1000'	9.5 KN	TON/1000'	10.- KN	TON/1000'	10.5 KN	TON/1000'	DRAFTS RPM ↑
DEAD SLOW	25						4.8				4.7				4.6				
SLOW	37						7.2				7.0				6.8				
HALF	49						9.5				9.3				9.1				
CRITICAL! TORSION	56																		
FULL	70	4830+860 =	5690	157.2	895	21.5	13.7 KN	65	13.6	66	13.4	67	13.2	68	13.0	69	12.6	71	70
	71	5040	5900	156.8	925	22.2	13.9	66	13.8	67	13.6	68	13.4	69	13.2	70	12.8	72	71
	72	5260	6120	156.3	956	22.9	14.1	68	14.0	68½	13.8	69	13.6	70	13.4	71	13.0	73	72
	73	5480	6340	155.8	988	23.7	14.2	69	14.1	70	14.0	71	13.8	72	13.5	73	13.2	75	73
	74	5710	6570	155.3	1020	24.5	14.4	71	14.3	71½	14.2	72	14.0	73	13.7	74	13.3	77	74
	75	5940	6800	154.8	1053	25.3	14.6	72	14.5	72½	14.4	73	14.2	74	13.9	76	13.5	78	75
	76	6190	7040	154.4	1086	26.1	14.8	73	14.7	73½	14.6	74	14.4	75	14.1	77	13.6	80	76
	77	6430	7290	153.8	1121	26.9	15.0	74	14.9	75	14.8	76	14.6	77	14.3	79	13.8	81	77
	78	6690	7550	153.3	1157	27.8	15.2	76	15.1	76½	15.0	77	14.8	78	14.5	80	14.0	83	78
	79	6950	7810	152.8	1193	28.7	15.4	77	15.3	77½	15.2	78	15.0	79	14.6	81	14.2	84	79
	80	7210	8070	152.2	1228	29.5	15.6	78	15.5	79	15.4	80	15.2	81	14.8	83	14.4	85	80
	81	7490	8350	151.6	1265	30.4	15.8	80	15.7	80½	15.6	81	15.3	82	15.0	84	14.6	87	81
	82	7770	8630	151.1	1304	31.3	16.0	81	15.9	81½	15.8	82	15.5	83	15.2	86	14.7	89	82
	83	8050	8910	150.6	1341	32.2	16.2	83	16.1	83½	16.0	84	15.7	85	15.4	87	14.9	90	83
	84	8350	9210	150.0	1381	33.2	16.4	84	16.3	84½	16.2	85	15.9	86	15.5	89	15.1	92	84
	85	8650	9510	149.4	1420	34.1	16.6	85	16.5	86	16.4	87	16.1	88	15.7	91	15.2	93	85
	86	8960	9820	148.8	1461	35.0	16.8	87	16.7	87½	16.6	88	16.3	89	15.9	92	15.4	95	86
	87	9280	10140	148.2	1503	36.0	17.0	88	16.9	88½	16.8	89	16.5	90	16.0	93	15.5	97	87
	88	9600	10460	147.7	1544	37.0	17.2	89	17.1	90	17.0	91	16.7	92	16.2	95	15.7	99	88
	89	9930	10790	147.1	1588	38.0	17.4	91	17.3	92	17.2	93	16.9	94	16.4	97	15.8	100	89
	90	10270	11130	146.5	1630	39.1	17.6	92	17.5	93	17.3	94	17.0	95	16.5	98	16.0	102	90
	91	10620	11480	146.0	1676	40.2	17.8	94	17.7	95	17.5	96	17.2	97	16.7	100	16.1	104	91
	92	10970	11830	145.4	1720	41.3	18.0	95	17.9	96	17.7	97	17.4	98	16.8	102	16.2	106	92
	93	11330	12190	144.9	1765	42.4	18.2	97	18.1	98	17.9	99	17.6	100	17.0	104	16.4	108	93
	94	11700	12560	144.4	1813	43.5	18.4	98	18.3	99	18.1	100	17.8	102	17.2	106	16.5	110	94
	95	12080	12940	144.0	1864	44.7	18.6	100	18.5	101	18.3	102	17.9	104	17.4	108	16.6	112	95
	96	12460	13320	143.7	1914	45.9	18.8	102	18.7	103	18.4	104	18.1	106	17.5	110	16.8	114	96
	97	12860	13720	143.4	1968	47.2	19.0	103	18.9	104	18.6	106	18.3	108	17.7	112	16.9	116	97
	98	13260	14120	143.2	2024	48.5	19.2	105	19.1	106	18.8	108	18.4	110	17.8	114	17.0	119	98
	99	13670	14530	143.1	2080	49.9	19.4	107	19.3	108	19.0	110	18.6	112	18.0	116	17.1	121	99
	100	14090	14950	143.0	2138	51.3	19.6	109	19.4	110	19.1	112	18.8	114	18.1	118	17.3	124	100
	101	14510	15370	143.1	2200	52.8	19.8	111	19.6	112	19.3	114	19.0	116	18.3	120	17.4	126	101
	102	14950	15810	143.3	2266	54.4	19.9	113	19.8	114	19.5	116	19.2	118	18.4	122½	17.5	129	102
	103	15390	16250	143.7	2336	56.0	20.1	115	20.0	116	19.7	118	19.3	120	18.6	125½	17.6	132	103

mmss NEDLLOYD VAN NECK/VAN NOORT/VAN DIEMEN ..... FUEL CONSUMPTION

at various R.P.M. & DRAUGHTS

(SLOW-SPEED FUEL NOZZLES)

10-12

SLAGEN LIJSTJE - BRANDST. VERBR.

NEDLLOYD VAN NECK — NSMB MODEL 5343 B — PROP. 5711 A
" NOORT — PROPELLER LIPS 48
" DIEMEN — ϕ: 6800, H_{MEAN} = 5723, H 0,72 = 5880

SLOW-SPEED FUEL NOZZLES

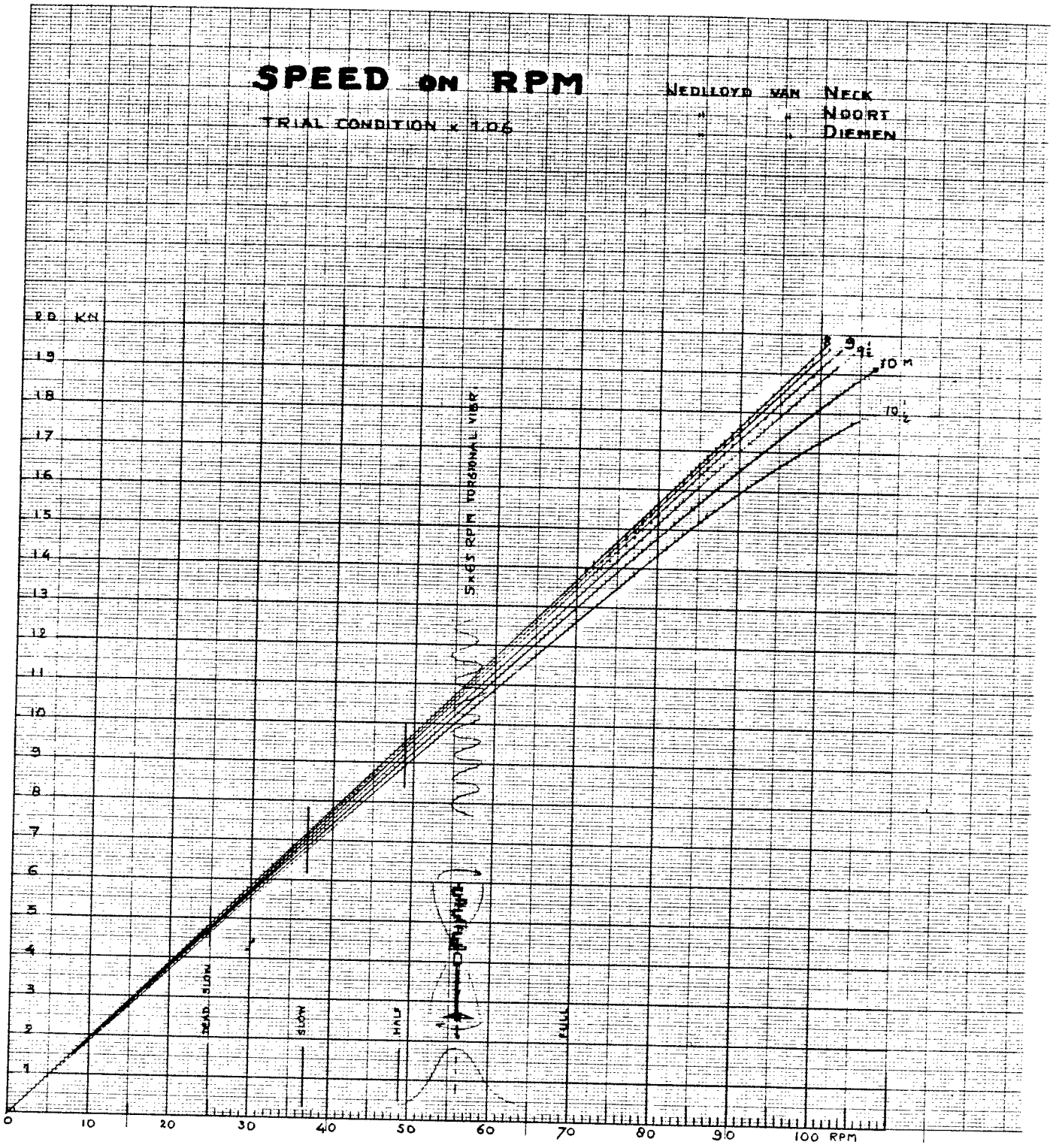
DEAD SLOW 25
SLOW 37
HALF 49
TORSION ~ 56
FULL

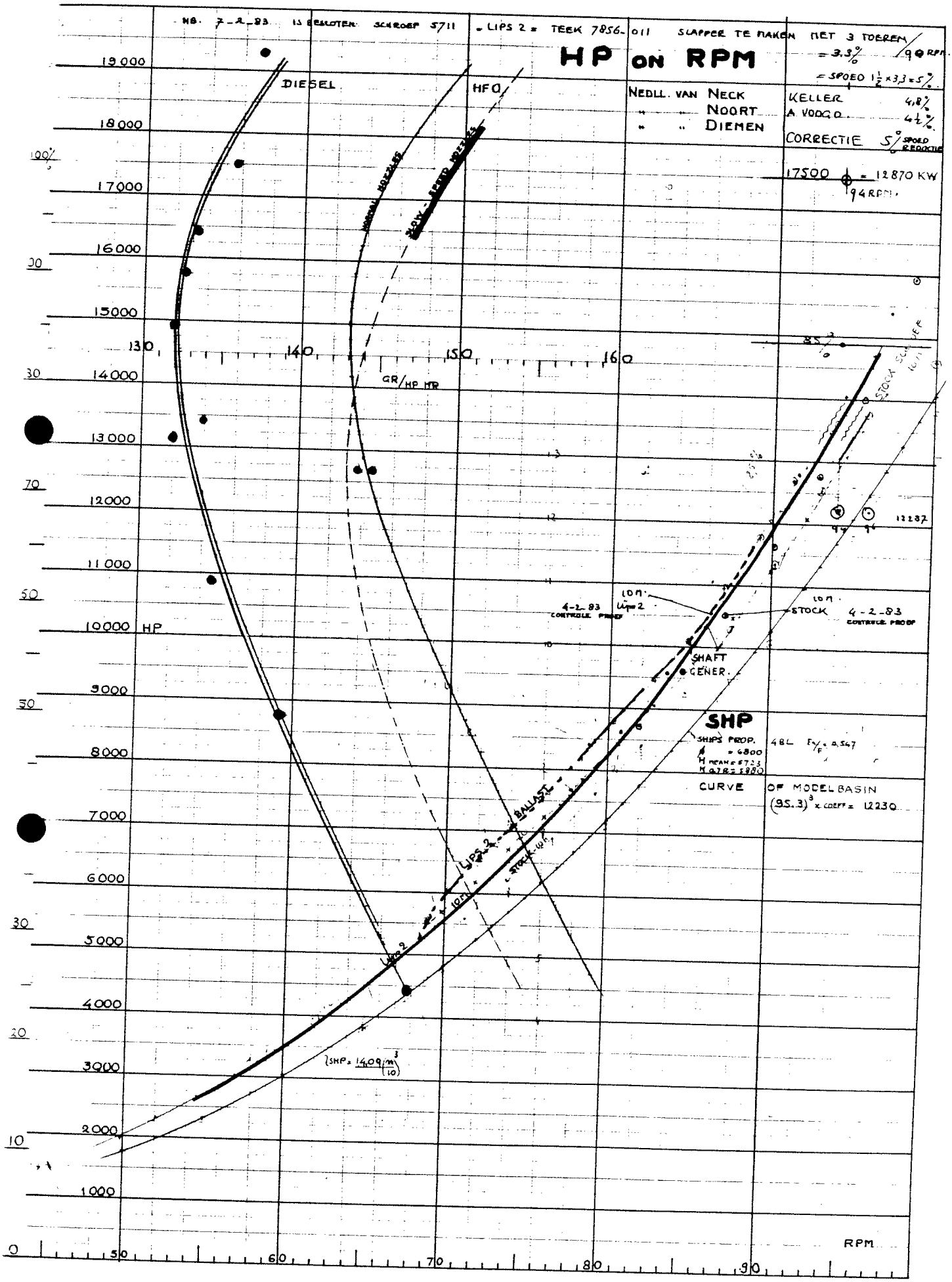
6% EXTRA HP ABOVE TRIALS

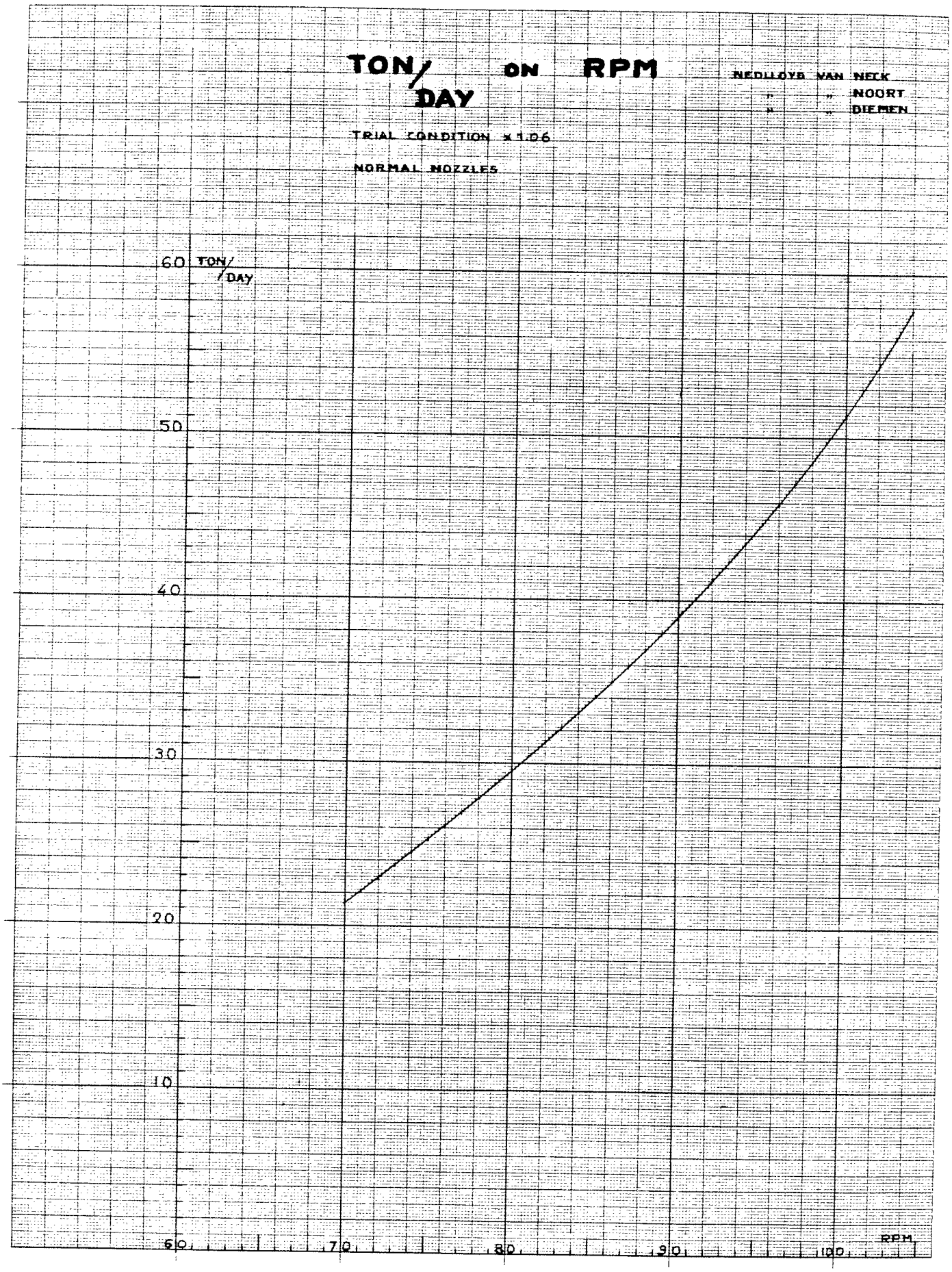
RPM	SHP + 860. BHP PROPELL	BHP ENG	GR/HP HR HFO	KG/HR	TON/DAY	8.- M KN	8.- M TON/1000	8.5 KN	8.5 TON/1000	9.- KN	9.- TON/1000	9.5 KN	9.5 TON/1000	10.- KN	10.- TON/1000	10.5 KN	10.5 TON/1000	Drafts R.P.M.
						4.8				4.7				4.6				25
						7.2				7.0				6.8				37
						9.5				9.3				9.1				49
																		56
70	4830	5690	152.2	8.66	20.8	13.7	63	13.6	63½	13.4	64½	13.2	65½	13.0	66½	12.6	68½	70
71	5040	5900	151.8	8.96	21.5	13.9	64	13.8	65	13.6	66	13.4	67	13.2	68	12.8	70	71
72	5260	6120	151.3	9.26	22.2	14.1	65½	14.0	66	13.8	67	13.6	68	13.4	69	13.0	71	72
73	5480	6340	150.8	9.56	22.9	14.2	67	14.1	67½	14.0	68½	13.8	69½	13.5	71	13.2	72½	73
74	5710	6570	150.4	9.88	23.7	14.4	68	14.3	68½	14.2	69½	14.0	70½	13.7	72	13.3	74	74
75	5940	6800	150.0	10.20	24.5	14.6	69½	14.5	70	14.4	71	14.2	72	13.9	73½	13.5	75½	75
76	6190	7040	149.4	10.52	25.2	14.8	70½	14.7	71	14.6	72	14.4	73	14.1	74½	13.6	77½	76
77	6430	7290	148.9	10.86	26.0	15.0	72	14.9	72½	14.8	73½	14.6	74½	14.3	76	13.8	78½	77
78	6690	7550	148.4	11.20	26.9	15.2	73½	15.1	74	15.0	75	14.8	76	14.5	77½	14.0	80	78
79	6950	7810	148.0	11.56	27.7	15.4	75	15.3	75	15.2	76	15.0	77½	14.6	79	14.2	81½	79
80	7210	8070	147.5	11.90	28.6	15.6	76	15.5	76½	15.4	77	15.2	78½	14.8	80½	14.4	83	80
81	7490	8350	147.0	12.27	29.5	15.8	77	15.7	78	15.6	79	15.3	80	15.0	81½	14.6	84½	81
82	7770	8630	146.6	12.65	30.4	16.0	78½	15.9	79	15.8	80	15.5	81½	15.2	83½	14.7	86	82
83	8050	8910	146.2	13.03	31.3	16.2	80	16.1	80½	16.0	81½	15.7	83	15.4	85	14.9	87½	83
84	8350	9210	145.7	13.42	32.2	16.4	81½	16.3	82	16.2	83	15.9	84½	15.5	86½	15.1	89	84
85	8650	9510	145.3	13.82	33.2	16.6	83	16.5	83½	16.4	84½	16.1	86	15.7	88	15.2	91	85
86	8960	9820	144.9	14.23	34.2	16.8	84½	16.7	85	16.6	86	16.3	87½	15.9	89½	15.4	92½	86
87	9280	10140	144.6	14.66	35.2	17.0	86	16.9	86½	16.8	87½	16.5	89	16.0	91	15.5	94½	87
88	9600	10460	144.2	15.08	36.2	17.2	87½	17.1	88	17.0	89	16.7	90½	16.2	93	15.7	96½	88
89	9930	10790	144.0	15.54	37.3	17.4	89	17.3	89½	17.2	90½	16.9	92	16.4	95	15.8	98	89
90	10270	11130	143.7	15.99	38.4	17.6	90½	17.5	91	17.3	92	17.0	94	16.5	97	16.0	100	90
91	10620	11480	143.4	16.46	39.5	17.8	92	17.7	93	17.5	94	17.2	95½	16.7	98½	16.1	102	91
92	10970	11830	143.2	16.94	40.7	18.0	94	17.9	94½	17.7	95½	17.4	97½	16.8	100½	16.2	104½	92
93	11330	12190	143.1	17.44	41.9	18.2	95½	18.1	96	17.9	97½	17.6	99	17.0	102½	16.4	106½	93
94	11700	12560	143.0	17.96	43.1	18.4	97½	18.3	98	18.1	99	17.8	101	17.2	104½	16.5	109	94
95	12080	12940	143.0	18.50	44.4	18.6	99½	18.5	100	18.3	101½	17.9	103	17.4	106½	16.6	111½	95
96	12460	13320	143.1	19.06	45.7	18.8	101	18.7	102	18.4	103½	18.1	105	17.5	109	16.8	114	96
97	12860	13720	143.3	19.66	47.2	19.0	103½	18.9	104	18.6	105½	18.3	107½	17.7	111	16.9	116	97
98	13260	14120	143.5	20.26	48.6	19.2	105½	19.1	106	18.8	107½	18.4	110	17.8	113½	17.0	119	98
99	13670	14530	143.9	20.91	50.2	19.4	108	19.3	108½	19.0	110	18.6	112	18.0	116½	17.1	122	99
100	14090	14950	144.4	21.53	51.8	19.6	110	19.4	111	19.1	112½	18.8	115	18.1	119	17.3	125	100
101	14510	15370	145.0	22.23	53.5	19.8	112½	19.6	113½	19.3	115½	19.0	117½	18.3	122	17.4	128	101
102	14950	15810																102
103	15390	16250																103

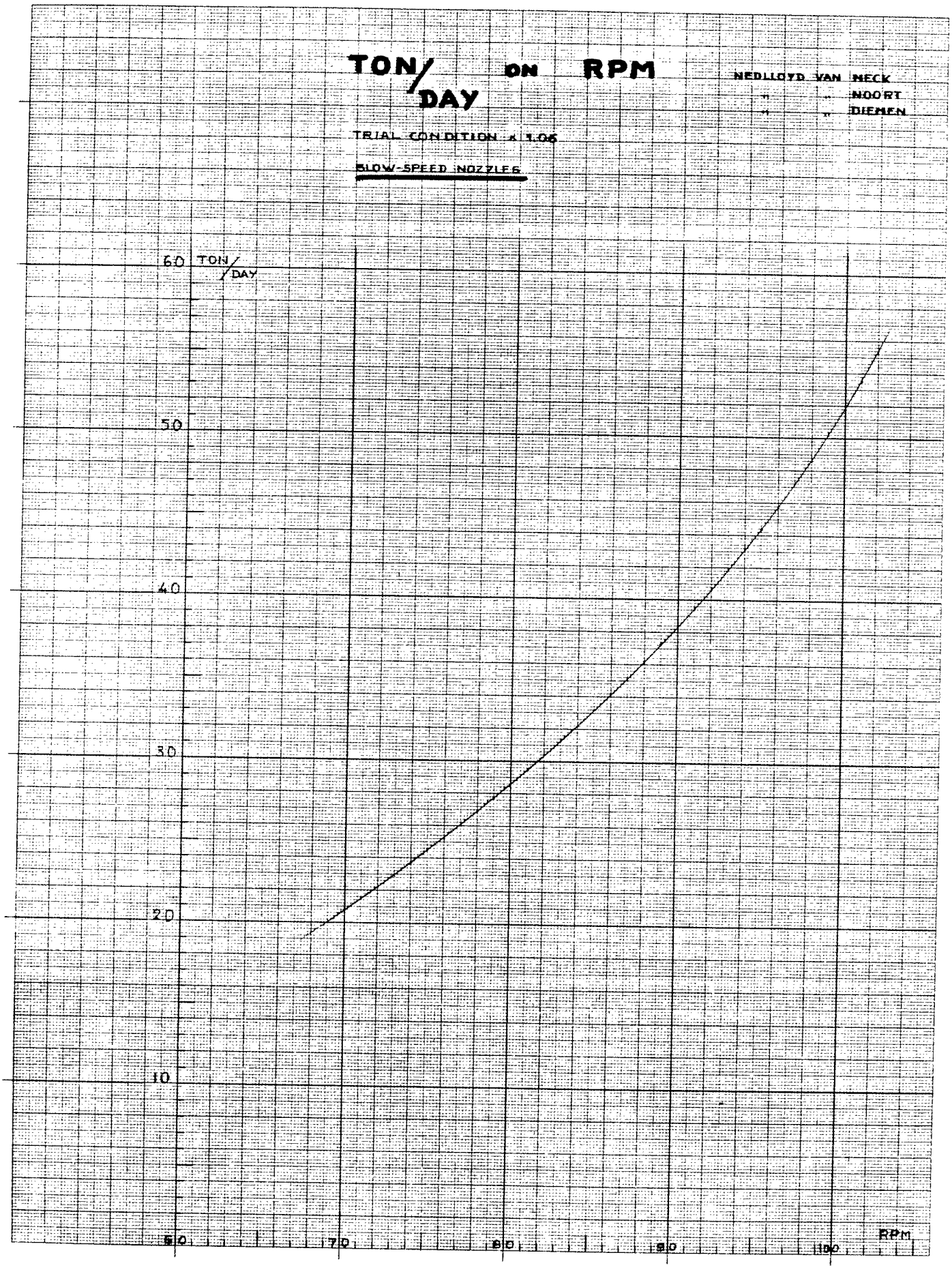
17500 × 85%

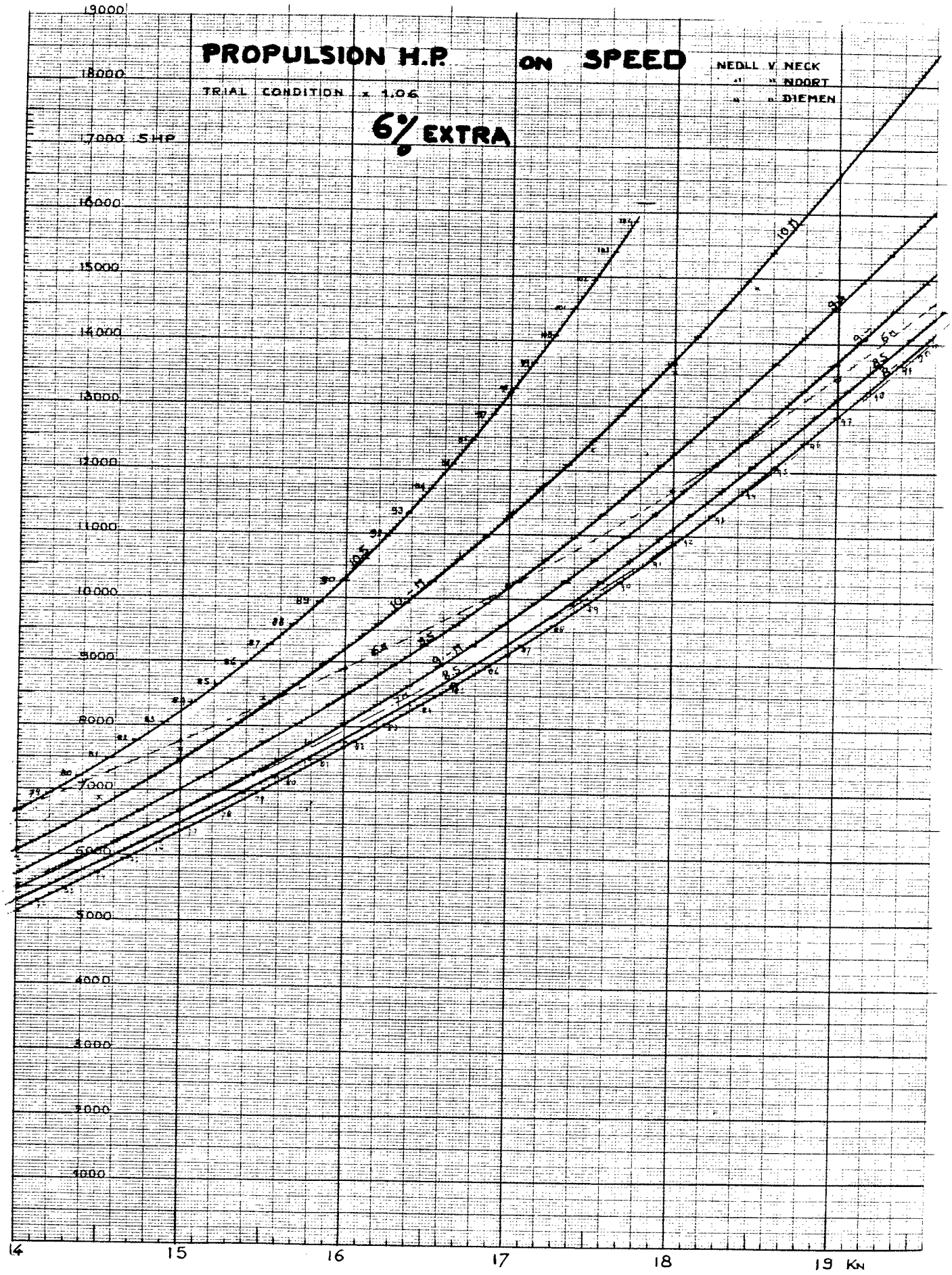
61
24-11-83

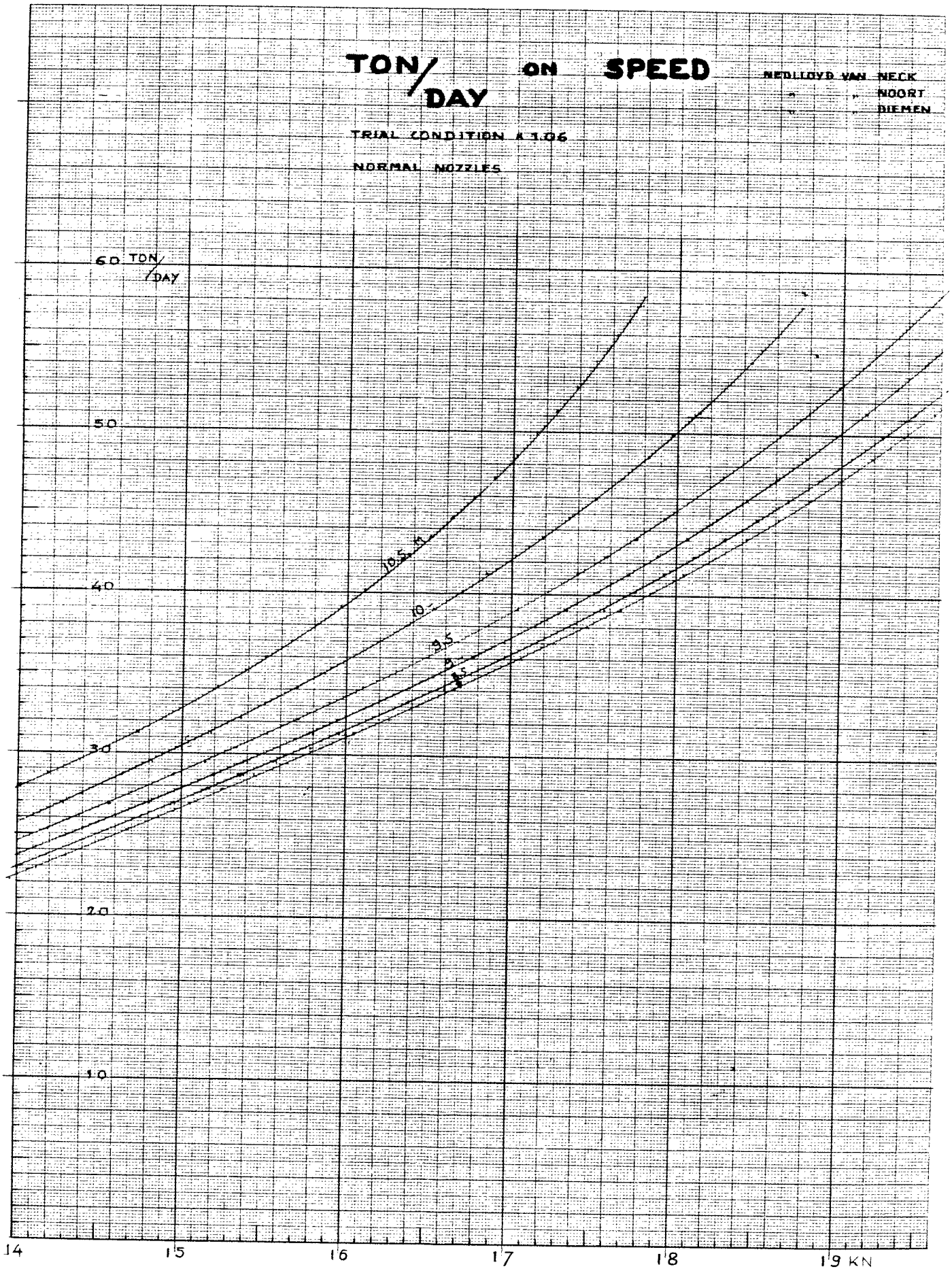

SPEED ON RPM
TRIAL CONDITION x 1.06
NEDLLOYD VAN NECK
" " NOORT
" " DIEMEN
20 KN
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
10
20
30
40
50
60
70
80
90
100 RPM
9 9½
10 M
10½
SKES RPM TORSIONAL VIBR
DEAD SLOW
SLOW
HALF
FULL

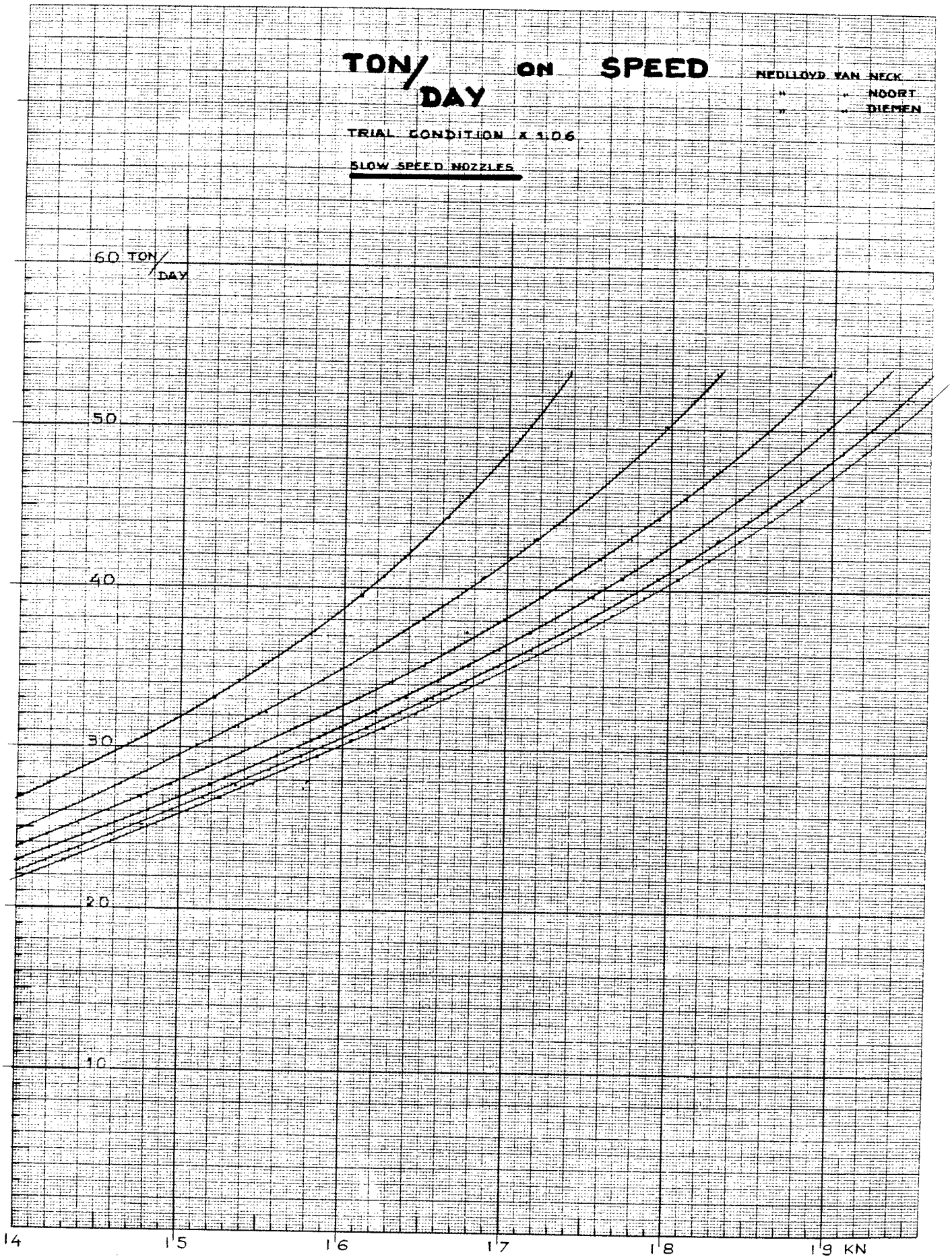

HP ON RPM
DIESEL
HFO
NEDLL. VAN NECK
NOORT
DIEMEN
KELLER 4,8%
A. VOOGD 4½%
CORRECTIE 5% SPOED REDUCTIE
17500 = 12870 KW
94 RPM
GR/HP HR
HP
SHP
SHAFT
GENER.
STOCK
4-2-83
LIPS 2
BALLAST
SHIPS PROP. 48L
CURVE OF MODEL BASIN
RPM

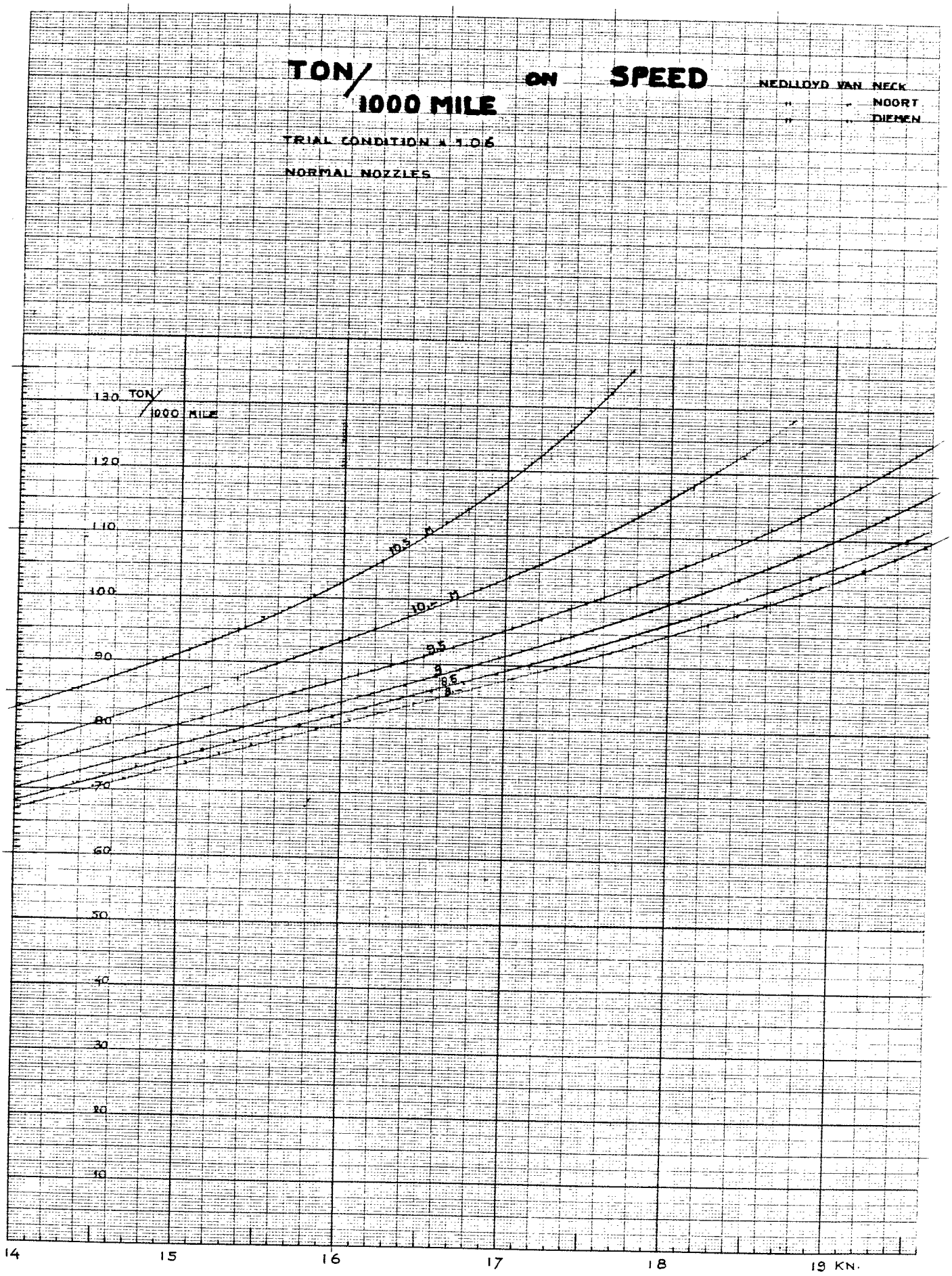

TON/DAY ON RPM
NEDILLOYD VAN NECK
" " NOORT
" " DIEMEN
TRIAL CONDITION x 1.06
NORMAL NOZZLES
60 TON/DAY
50
40
30
20
10
60
70
80
90
100
RPM

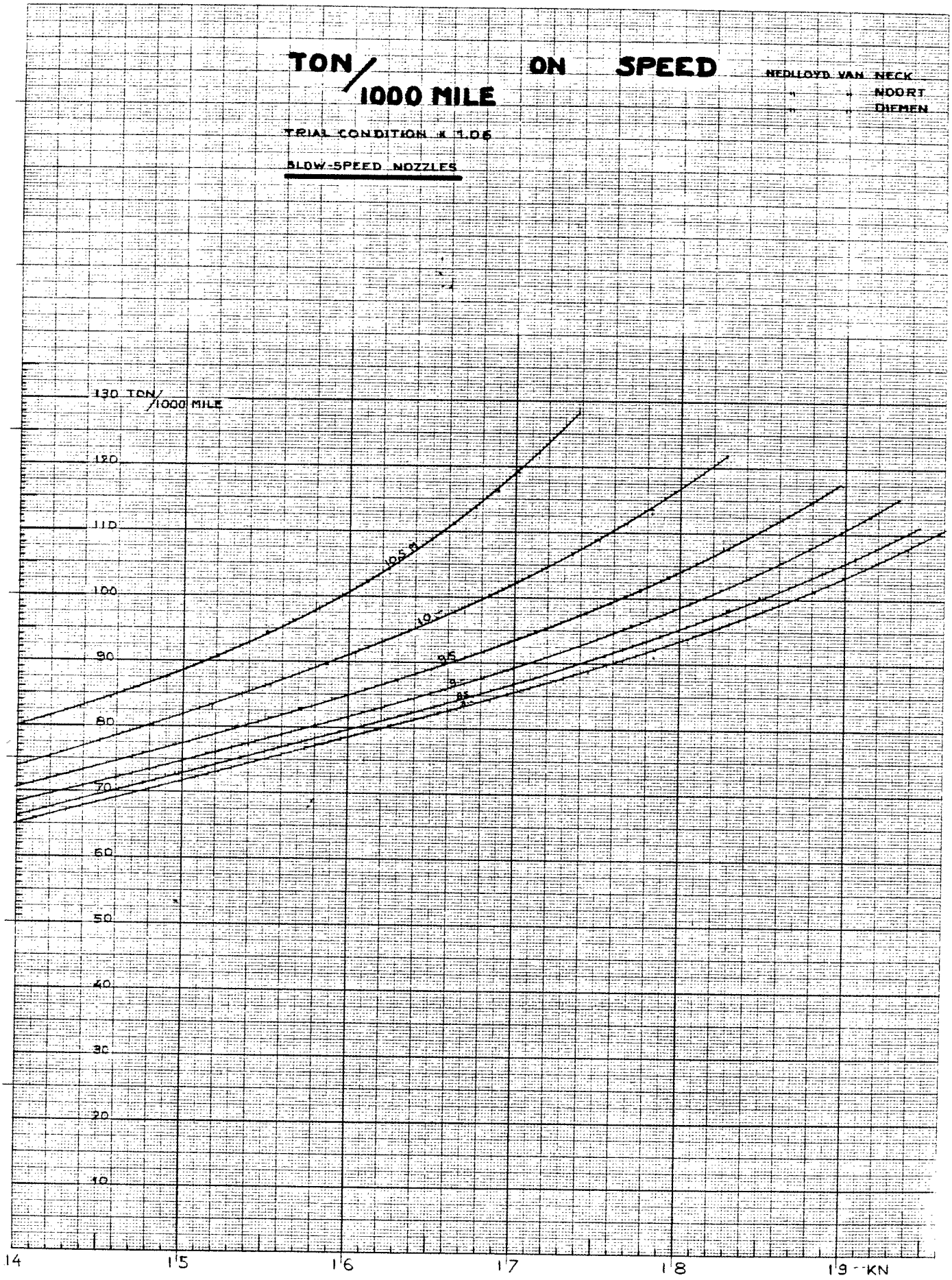

TON/DAY ON RPM
NEDLLOYD VAN NECK
" " NOORT
" " DIEMEN
TRIAL CONDITION x 1.05
SLOW-SPEED NOZZLES
60 TON/DAY
50
40
30
20
10
50
70
80
90
100
RPM


PROPULSION H.P. ON SPEED
TRIAL CONDITION x 1.06
6% EXTRA
NEDLL V. NECK
NOORT
DIEMEN
19000
18000
17000 SHP
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
14
15
16
17
18
19 KN


TON/DAY ON SPEED
NEDLLOYD VAN NECK
" " NOORT
" " DIEMEN
TRIAL CONDITION A 1.06
NORMAL NOZZLES
60 TON/DAY
50
40
30
20
10
10.5 KN
10
9.5
9
8.5
14
15
16
17
18
19 KN


TON/DAY ON SPEED
NEDLLOYD VAN NECK
" " NOORT
" " DIEMEN
TRIAL CONDITION x 1.06
SLOW SPEED NOZZLES
60 TON/DAY
50
40
30
20
10
14
15
16
17
18
19 KN


TON/1000 MILE ON SPEED
TRIAL CONDITION x 1.06
NORMAL NOZZLES
NEDLLOYD VAN NECK
" " NOORT
" " DIEMEN
130 TON/1000 MILE
120
110
100
90
80
70
60
50
40
30
20
10
10.5
10
9.5
9
8.5
8
14
15
16
17
18
19 KN.


TON/1000 MILE ON SPEED
NEDLLOYD VAN NECK
" " NOORT
" " DIEMEN
TRIAL CONDITION × 1.06
SLOW-SPEED NOZZLES
130 TON/1000 MILE
120
110
100
90
80
70
60
50
40
30
20
10
10.5 m
10
9.5
9
8.5
8
14
15
16
17
18
19 KN

MAIN ENGINE: SULZER 5RLB90
MCR 17500 HP at 94 RPM
service rating 14875 HP at 94 RPM
(85%) = ± 51 ton/day
Fuel for main eng: HFO max 600 cSt at 50°
Fuel for aux.eng.: IFO = 50%HFO/50%MDO
Heavy Fuel Oil consumption excl.
shaft generator in tons/day
SHP service = $SHP_{tank} \times C_1 \times C_2$
C_1 = margin for trials = 1,06
C_2 = sea margin = 1,12 (service allowance)
Spec. fuel consumption = 143 gr/SHP.h

Draft	15.5	16	16.5	17	17.5	18	18.5	19kn
7M	26.5	29.0	31.4	34.1	36.9	39.9	43.1	46.5
8M	25.5	28.1	30.7	33.4	36.4	39.6	43.1	46.7
9M	26.7	29.4	32.2	35.1	38.2	41.5	45.3	49.1
10M	30.1	33.4	37.1	40.9	45.1	49.7	55.0	ton/day

FUEL CONSUMPTION EXCL SHAFTGE
SEA SERVICE PILOT-PILOT
12% EXTRA TO TRIALS
(SPC BOTTOM PAIN

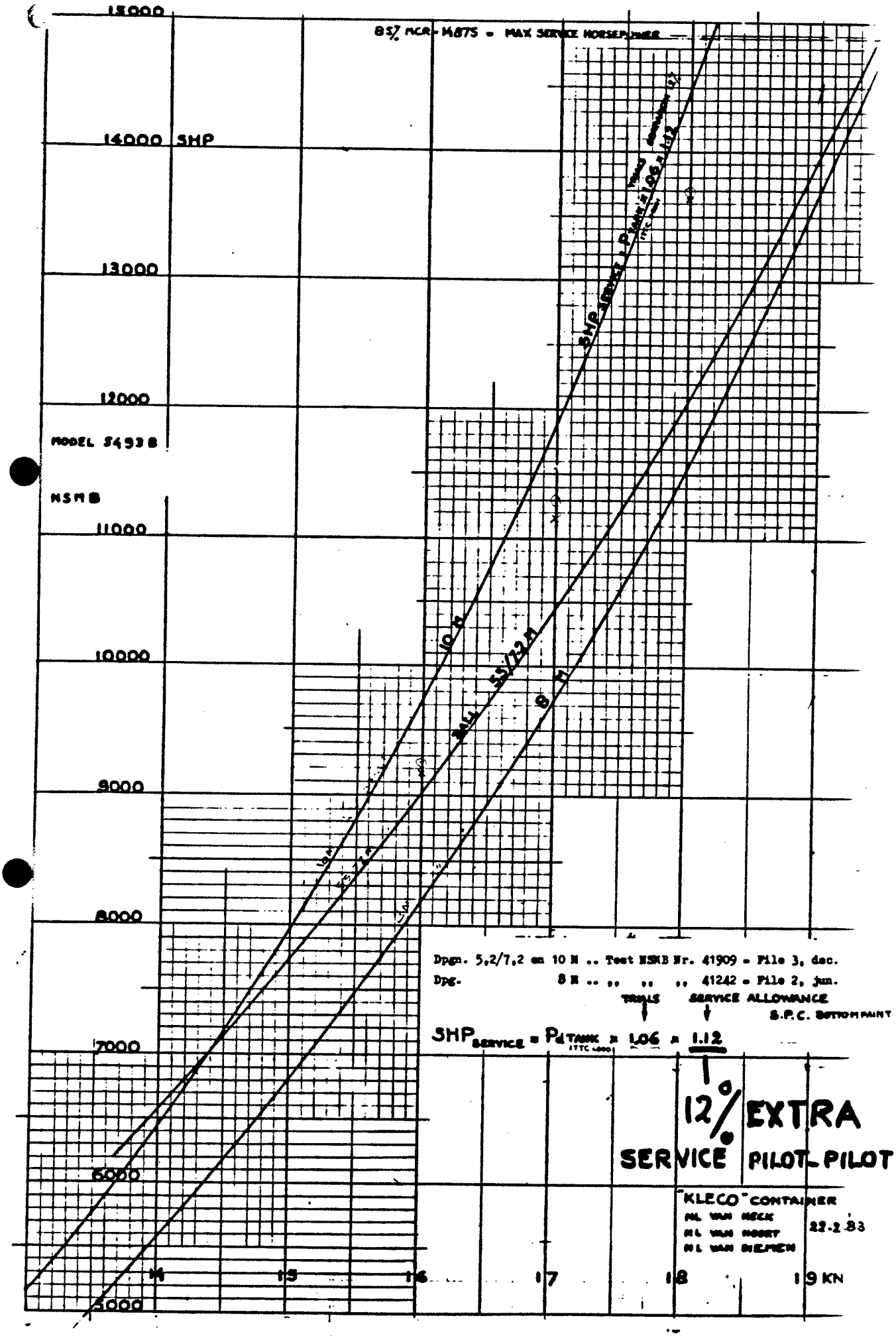
Shaft generator max 1300 kW
Normal: 540 kW x $\frac{1,36}{.85}$ = 860 HP
x 143 gr/SHP.hr x 24 = 2,9 T/day
Diesel generators
Diesel gen. I: 1500kW-720RPM
Diesel gen. II: 1500kW-720RPM
Diesel gen. III: 700kW-720RPM
Emergency gen.: 160kW-1800RPM

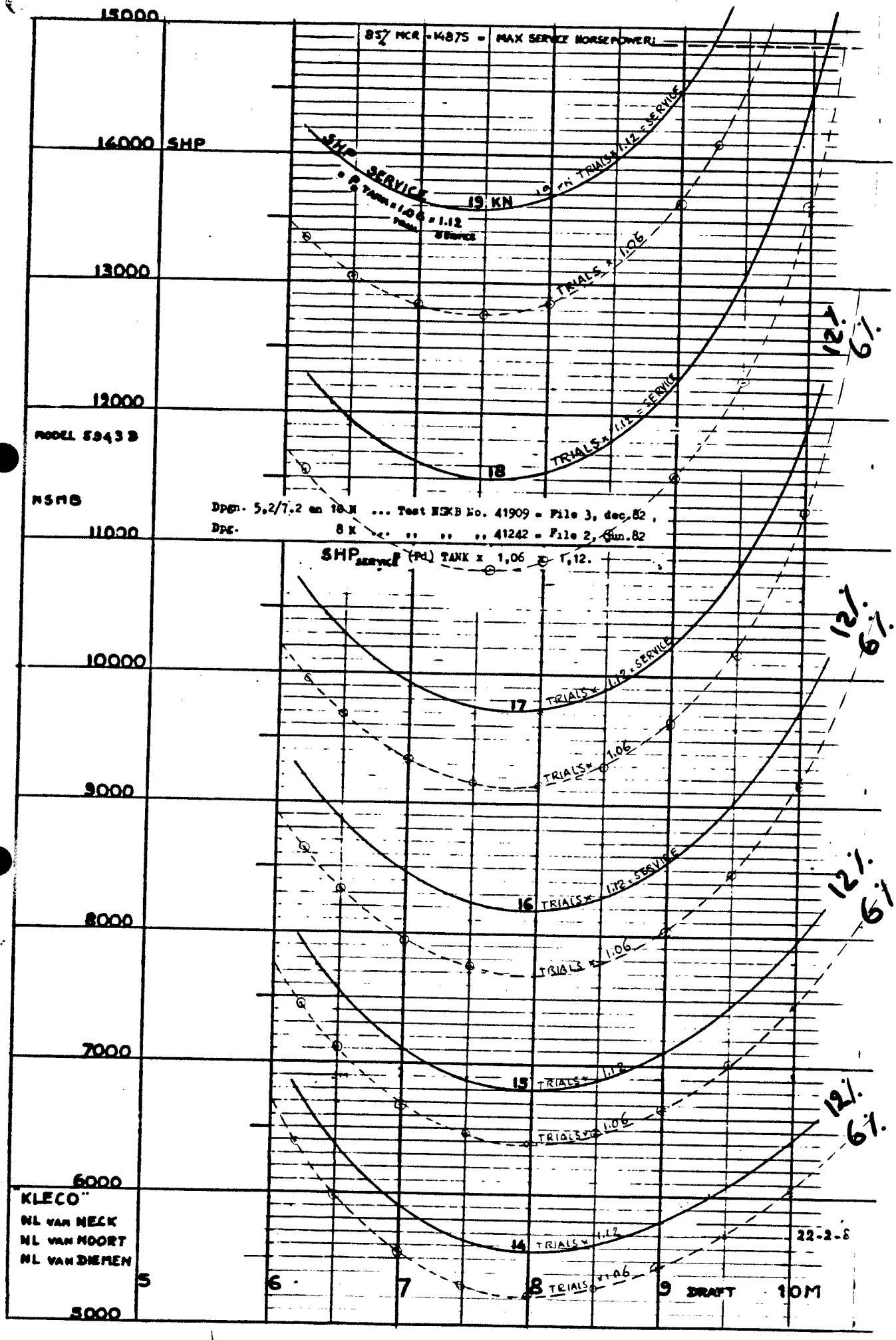
Normal electr. load at sea 540kW
-IFO diesel gen. 2,9T/day
-HFO shaft gen. 2,9T/day
Idle in port: 400kW IFO 2,3T/day
Boiler HFO 1.0T/day

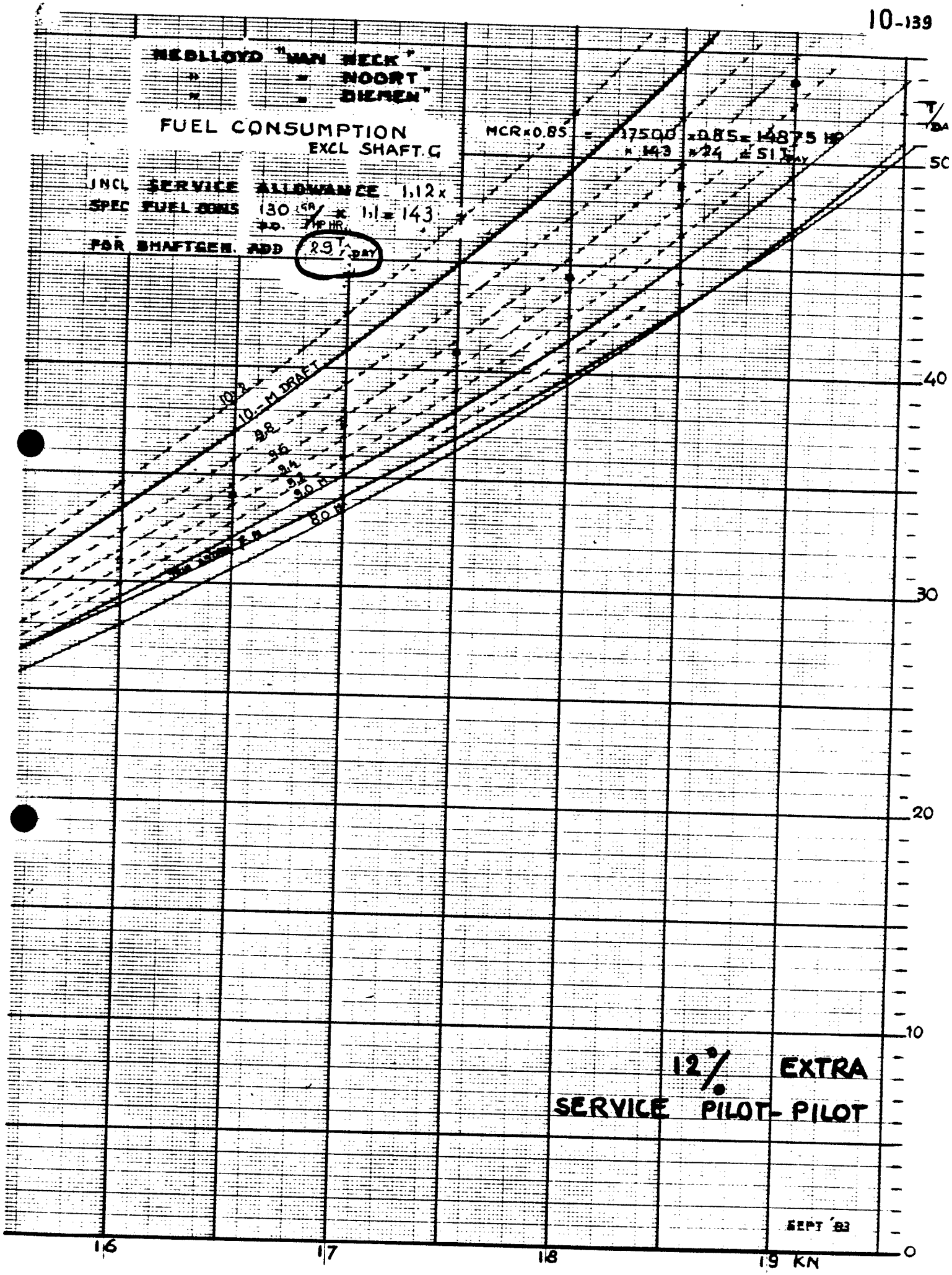
Extra for reefers (100 integr. 20')
box x kW/box x HP/kW x
gr/HP.hr x hr =
100 x 5.0 x 1.5 x 0.143 x 24 = 2,6 T/day

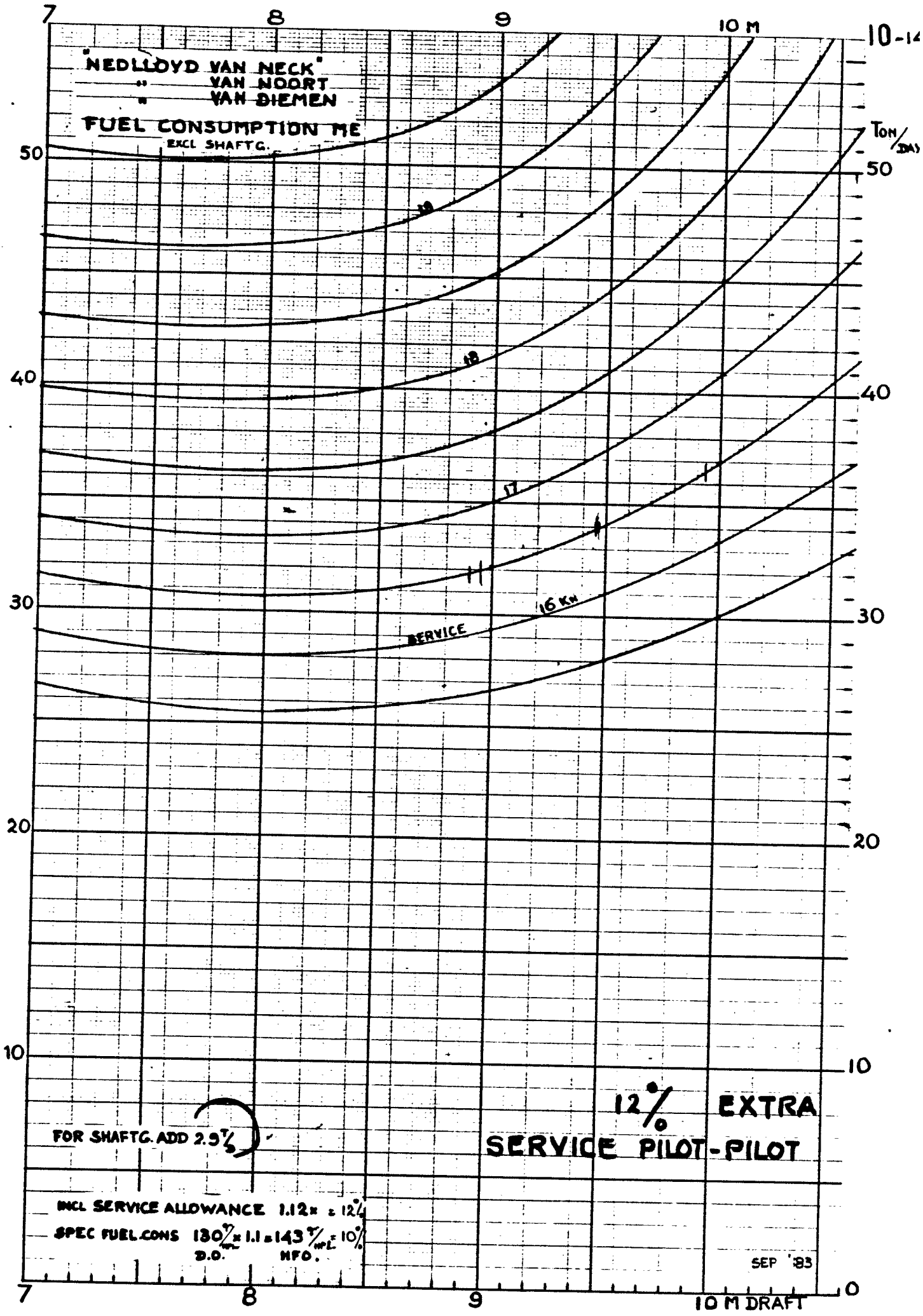
THRUSTERS (var. pitch)
Bowthruster 900 kW, 1220 HP
Sternthruster 900 kW, 1220 HP
Rudder max. angle 45°

INTEGRAL REEFERS
Reeferpoints on deck 20'/40':
BOX = 90 à 5,5 kW 20'/7,5 kW 40'
Reeferpoints in hold 20' BOX = 60
ventilation 1 m3/sec per box
Receptacles 440V, CEE 17, 3h, 32 AMP


85% MCR = 14875 = MAX SERVICE HORSEPOWER
15000
14000 SHP
13000
12000
MODEL 5493 B
NSMB
11000
10000
9000
8000
7000
6000
5000
14
15
16
17
18
19 KN
10 M
8 M
Dpgn. 5,2/7,2 m 10 M .. Test NSMB Nr. 41909 = File 3, dec.
Dpg. 8 M .. ,, ,, ,, 41242 = File 2, jun.
TRIALS
SERVICE ALLOWANCE
S.P.C. BOTTOM PAINT
SHP SERVICE = PdTANK (ITTC 1978) x 1.06 x 1.12
12% EXTRA
SERVICE PILOT-PILOT
"KLECO" CONTAINER
NL VAN HECK
NL VAN HOORT
NL VAN DIEPEN
22.2.83


85% MCR = 14875 = MAX SERVICE HORSEPOWER
15000
16000 SHP
13000
12000
11000
10000
9000
8000
7000
6000
5000
MODEL 5343 B
NSMB
SHP SERVICE
19 KN
18
17
16
15
14
TRIALS x 1.12 = SERVICE
TRIALS x 1.06
12%
6%
Dpgn. 5,2/7,2 en 18 N ... Test NSMB No. 41909 = File 3, dec.82
Dpg. 8 K " " " " 41242 = File 2, Jun.82
SHP service (PL) TANK x 1,06 = 1,12.
KLECO
NL VAN HECK
NL VAN MOORT
NL VAN DIEMEN
22-2-8
5
6
7
8
9
DRAFT
10M


NEDLLOYD "VAN NECK"
" = NOORT"
" = DIEMEN"
FUEL CONSUMPTION
EXCL SHAFT.G
INCL SERVICE ALLOWANCE 1.12 x
SPEC FUEL CONS 130 GR/HP HR x 1.1 = 143
FOR SHAFTGEN. ADD 1.29 T/DAY
MCR x 0.85 = 17500 x 0.85 = 14875 HP
x 143 x 24 = 51 T/DAY
10.2
10.-M DRAFT
9.8
9.6
9.4
9.2
9.0 M
8.0 M
T/DAY
50
40
30
20
10
0
16
17
18
19 KN
12% EXTRA
SERVICE PILOT-PILOT
SEPT '83


"NEDLLOYD VAN NECK"
" VAN NOORT
" VAN DIEMEN
FUEL CONSUMPTION ME
EXCL SHAFTG.
TON/DAY
19
18
17
16 KN
SERVICE
12% EXTRA
SERVICE PILOT-PILOT
FOR SHAFTG. ADD 2.9%
INCL SERVICE ALLOWANCE 1.12x = 12%
SPEC FUEL CONS 130 T/MPL x 1.1 = 143 T/MPL = 10%
D.O. HFO.
SEP '83
10 M DRAFT
7
8
9
10 M
10-14
50
40
30
20
10
0

NEDLLOYD "VAN NOORT"

OPTIMAL EASTB 1426 TEU = 14,2 T/TEU 9.94 M DRAFT
" WESTB 1590 " = 9,5 " 8.95 M "

ROUNDTRIP SCANDUTCH "S"

	HFO	IFO	DAYS
AT SEA			
23.870 MILES PILOT-PILOT AT 16,5 KN			
1416 HRS = 59 DAYS DRAFT MEAN 9.45M			
59 DAYS x (34 T/D MAIN E + 2,9 T/D SHAFT G 540KW) = 59 x 36,9 =	2180		59 = 65%
IN PORT			
23 PORTS IN 23 DAYS			
23 DAYS (AUXIL 400KW 2,3 T/D) = 23 x 2,3 =		53	23 = 25%
BOILER 23 x 1 =	23		
MANOEUVRING			
9 DAYS x (AUXIL 2,9 + 2,3 T/D) = 9 x 5,2 =		47	9 = 10%
BOILER 9 x 1 =	9		
FUEL CONSUMPTION P. ROUNDTRIP EXCL REEFERCONTRS.	2212 T	100 T	91 = 100%

Electrische belasting op zee 540 kW

$$\frac{540 \times 1,36}{0,85\ (Acc)} = 860 \text{ pk}$$

860 x 142 x 24 = 2,9 Ton/dag H.F.O.

Electrische belasting in haven 400 kW

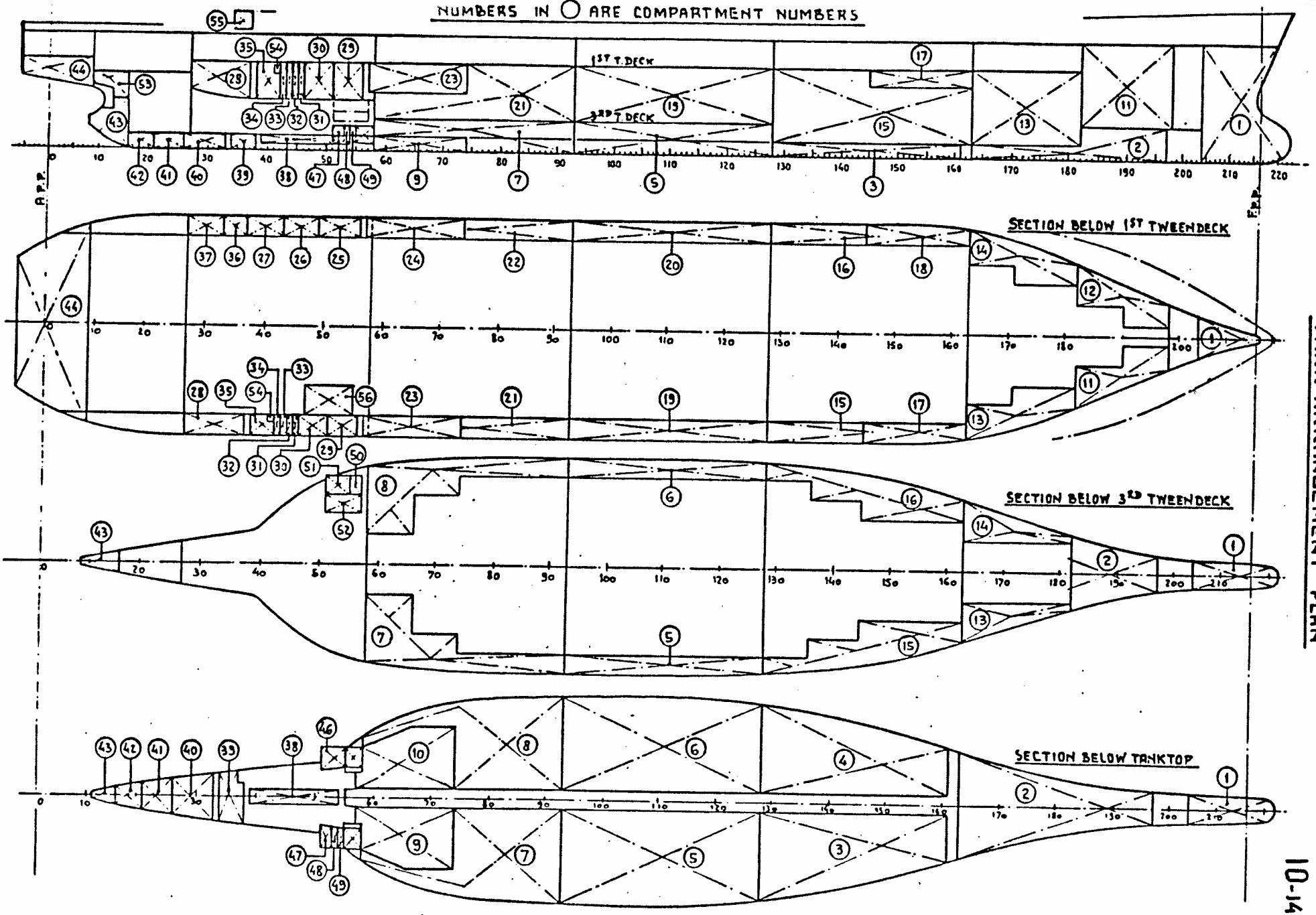
$$\frac{400 \times 1,36 \times 24}{0,92} = 2,3 \text{ Ton/dag I.F.O.}$$

15/9/83

R.

TANK ARRANGEMENT PLAN

NUMBERS IN ○ ARE COMPARTMENT NUMBERS



10-142

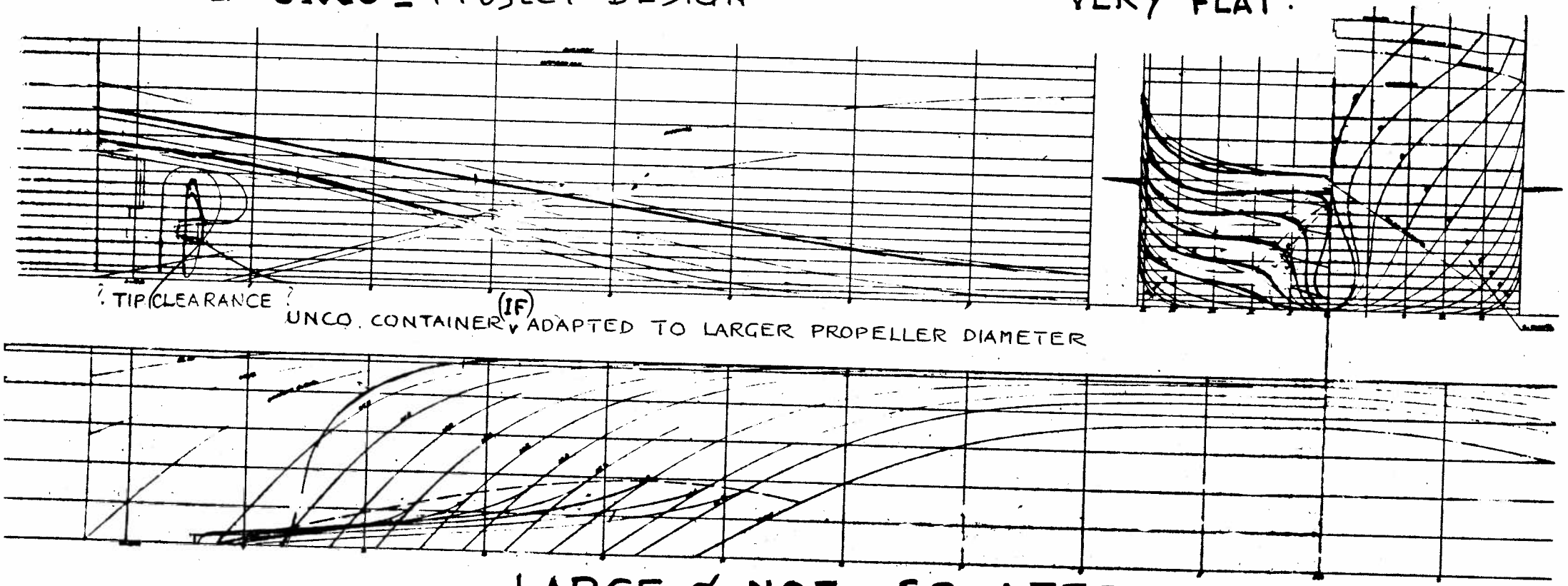
INDEX

COMPARTMENT	COMP. Nº	FRAME	CAPACITY 100% FULL m^3	CENTRE OF GRAVITY V.C.G. AB. BASE m	L.C.G. FROM APP m	FROM C.L. m	MAX. FREE SURF. MOMENT I IN m^4	PAGE Nº
W.B. FORE PEAK	1	204 - FWD.	545.02	8.67	169.69	0.00	540.3	4
W.B. D.B. TANK 1	2	163 - 197	532.66	1.87	147.32	0.00	2195.6	5
W.B. D.B. TANK 2 S.B.	3	128 - 161	402.54	0.97	115.43	5.86	2551.2	6
W.B. D.B. TANK 2 P.S.	4	128 - 161	402.54	0.97	115.43	-5.86	2551.2	7
W.B. D.B. TANK 3 S.B.	5	93 - 128	804.82	1.44	89.17	8.93	5327.0	8
W.B. D.B. TANK 3 P.S.	6	93 - 128	804.82	1.44	89.17	-8.93	5327.0	9
W.B. D.B. TANK 4 S.B.	7	58 - 93	650.80	2.07	61.87	9.13	2894.0	10
W.B. D.B. TANK 4 P.S.	8	58 - 93	650.80	2.07	61.87	-9.13	2894.0	11
D.O. D.B. TANK 4 S.B.	9	57 - 74	141.69	1.06	53.54	4.79	560.1	12
D.O. D.B. TANK 4 P.S.	10	57 - 74	141.69	1.06	53.54	-4.79	560.1	13
W.B. WINGT. 1 S.B. FORW.	11	182 - 198	465.85	10.82	154.19	5.33	322.9	14
W.B. WINGT. 1 P.S. FORW.	12	182 - 198	465.85	10.82	154.19	-5.33	322.9	15
H.F.O. WINGT. 1 S.B. AFT	13	163 - 182	486.56	7.36	139.47	8.26	225.0	16
H.F.O. WINGT. 1 P.S. AFT	14	163 - 182	486.56	7.36	139.47	-8.26	225.0	17
H.F.O. WINGT. 2 S.B.	15	128 - 163	767.65	6.37	117.70	12.16	502.4	18
H.F.O. WINGT. 2 P.S.	16	128 - 163	767.65	6.37	117.70	-12.16	502.4	19
H.F.O. OVERFLOW TANK S.B.	17	145 - 163	92.83	10.91	124.53	13.73	22.5	20
H.F.O. OVERFLOW TANK P.S.	18	145 - 163	92.83	10.91	124.53	-13.73	22.5	21
W.B. WINGT. 3 S.B.	19	93 - 128	612.27	2.34	89.34	13.86	51.4	22
W.B. WINGT. 3 P.S.	20	93 - 128	612.27	2.34	89.34	-13.86	51.4	23
W.B. WINGT. 4 S.B.	21	58 - 93	461.96	7.73	63.22	13.85	51.4	24
W.B. WINGT. 4 P.S.	22	58 - 93	461.96	7.73	63.22	-13.85	51.4	25
DRINKING W. TANK S.B.	23	57 - 74	155.75	10.14	52.43	13.86	25.0	26
DRINKING W. TANK P.S.	24	57 - 74	155.75	10.14	52.43	-13.86	25.0	27
H.F.O. SETTLING TANK P.S.	25	49 - 56	72.57	9.23	41.63	-13.86	10.6	28
H.F.O. SERVICE TANK P.S.	26	43 - 49	60.48	9.27	36.12	-13.82	9.1	29
H.F.O. SETTLING TANK P.S.	27	37 - 43	55.91	9.39	31.05	-13.74	9.1	30
FEEDWATER TANK S.B.	28	27 - 37	83.19	10.27	24.53	13.70	15.2	31
L.O. STOR. T. M.ENG. S.B.	29	51 - 56	60.37	9.58	42.48	13.86	7.6	32
L.O. STOR. T. M.ENG. S.B.	30	46 - 51	59.73	9.60	38.23	13.84	7.6	33
DIRTY L.O. STOR. T. AUX.ENG. S.B.	31	45 - 46	11.72	9.64	35.68	13.82	1.5	34
CLEAN L.O. STOR. T. AUX. ENG. S.B.	32	44 - 45	11.62	9.65	34.83	13.81	1.5	35
L.O. STORAGE T. AUX. ENG. S.B.	33	43 - 44	11.50	9.67	33.98	13.80	1.5	36
L.O. STORAGE T. AUX. ENG. S.B.	34	42 - 43	11.37	9.70	33.13	13.79	1.5	37
CYL. OIL STORAGE TANK S.B.	35	38 - 42	42.72	9.70	30.97	13.79	6.0	38
D.O. SERVICE TANK P.S.	36	33 - 37	37.90	10.05	26.80	-13.71	6.1	39
D.O. SETTLING TANK P.S.	37	27 - 33	45.27	10.44	22.63	-13.68	9.1	40
L.O. SUMP TANK	38	39 - 54	26.60	1.35	36.48	0.00	9.4	41
L.O. EXCHANGE TANK	39	34 - 38	31.43	1.06	27.57	0.13	85.9	42
H.F.O.-D.O. TR. OVERFLOW & DRAIN T.	40	26 - 33	46.72	1.08	22.20	0.00	96.5	43
DIRTY WATER TANK	41	21 - 26	25.96	1.10	17.06	0.00	34.7	44
L.O. SUMPT. STERNTUBE	42	16 - 21	12.58	1.08	13.48	0.00	10.3	45
FRESH WATER T. STERNTUBE	43	9 - 16	37.38	3.52	9.68	0.00	6.4	46
VOID SPACE A.P.	44	AFT - 9	494.24	11.34	2.07	0.00	15320.6	47
PISTON COOLW. TANK P.S.	46	51 - 55	16.22	2.45	42.08	-5.36	8.2	48
HOTWATER PREP. TANK S.B.	47	51 - 53	7.90	2.48	41.21	5.26	4.5	49
RESIDUE TANK S.B.	48	53 - 54	4.10	2.44	42.48	5.41	1.9	50
AIR COOLER CLEANING W.T. S.B.	49	54 - 55	4.20	2.41	43.33	5.49	1.8	51
SLUDGE MIXING TANK P.S.	50	55 - 57	8.21	6.05	44.60	-11.02	2.7	52
L.O. SLUDGE TANK P.S.	51	51 - 55	16.43	6.05	42.05	-11.02	5.5	53
H.F.O.-D.O. SLUDGE TANK P.S.	52	51 - 57	34.44	5.68	42.90	-8.36	8.0	54
VOID SPACE	53	9 - 16	143.01	9.48	8.90	0.00	1166.7	55
CYL. OIL TANK S.B.	54	41 - 42	1.42	11.25	32.27	12.93	0.1	56
D.O. TANK S.B.	55	31D - 35C	3.97	18.15	27.43	8.04	0.0	57
SWIMMING POOL S.B.	56	47 - 55	59.72	11.09	40.35	10.40	39.5	58

LARGER PROPØ FOR CONTAINER AND RO-RO

EX "UNCO" PROJECT DESIGN

VERY FLAT!

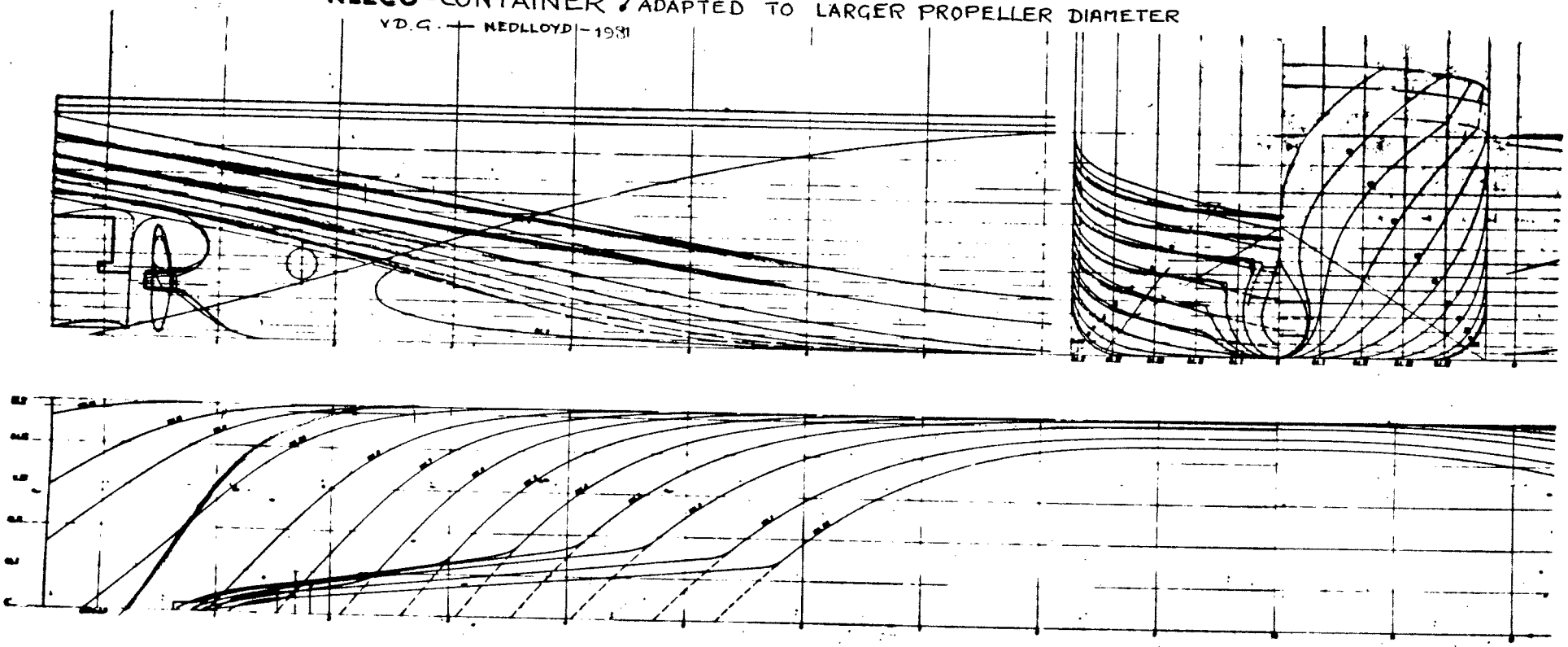


LARGE Ø, NOT SO ATTRACTIVE

The larger the propeller diameter, the higher and more flat the aftship. (' Beaver Tail ')
At a certain stage we arrive at a 'tunnel' shape.
The rise of the water onto the propeller should not be too steep so as to avoid separation of flow.
Loss of displacement by the tunnel has to be compensated by more massive buttocks.
Of great importance is the flow along the bottom to lead the water smoothly along the gondola towards the top position (12 o'clock) of the propeller disc in order to provide sufficient water there.
Still requiring research are the emergence of pressure fluctuations - and how much clearance needed ?

"KLECO-CONTAINER" (IF) ADAPTED TO LARGER PROPELLER DIAMETER

V.D.G. — NEDLLOYD — 1981



GERMAN PROPOSAL 1982

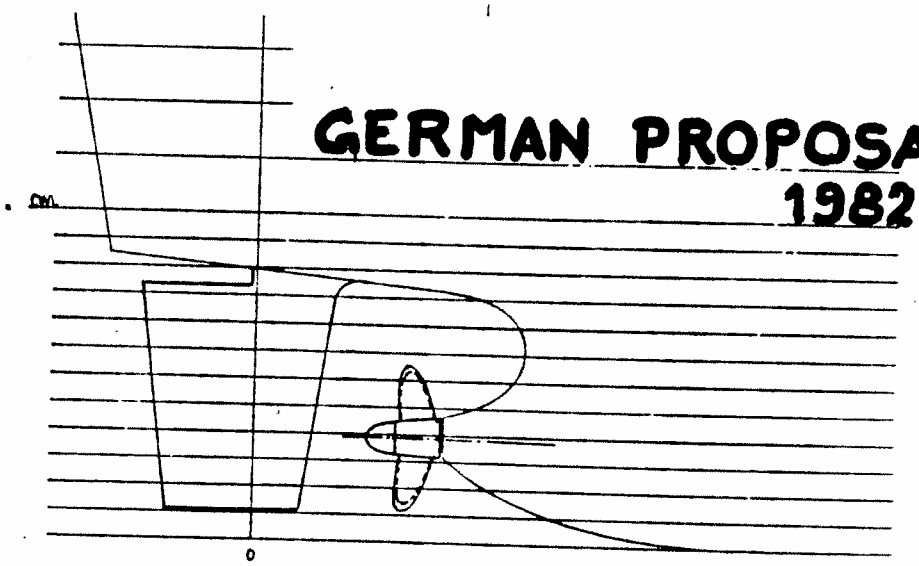


Bild 2 Hintersteven-Kontur mit Anordnung von Ruder und Propeller

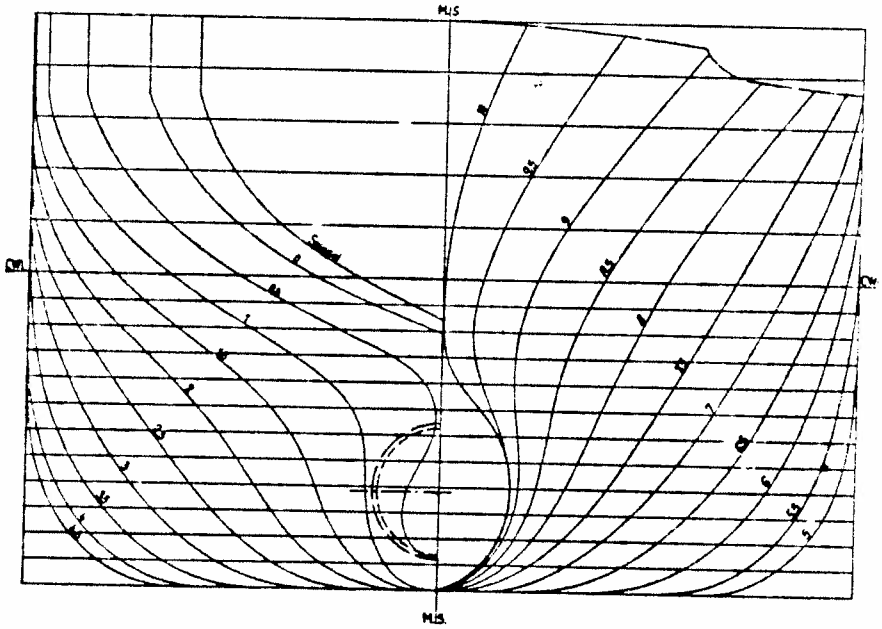
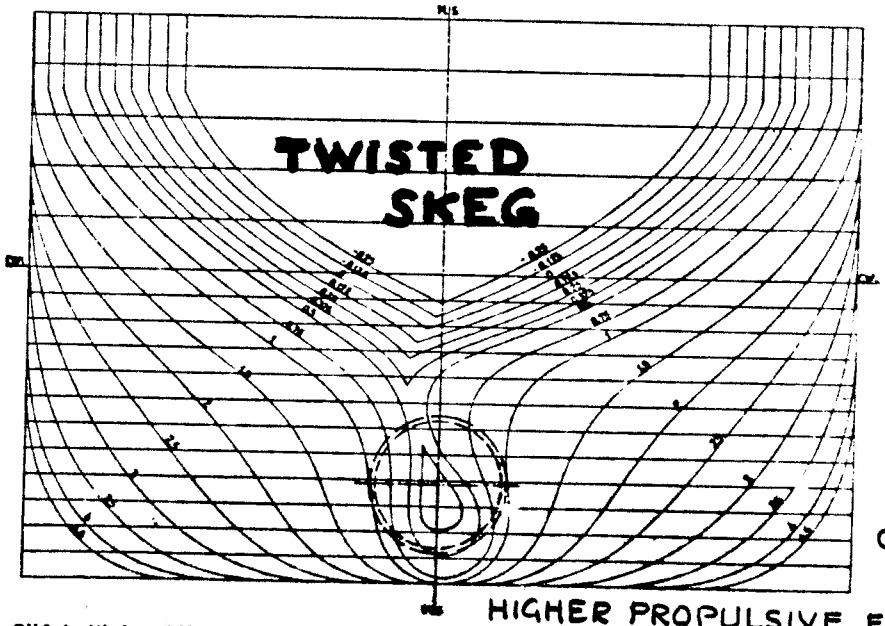


Bild 1 Vor- und Hinterschiffsspanten des Ausgangsmodells



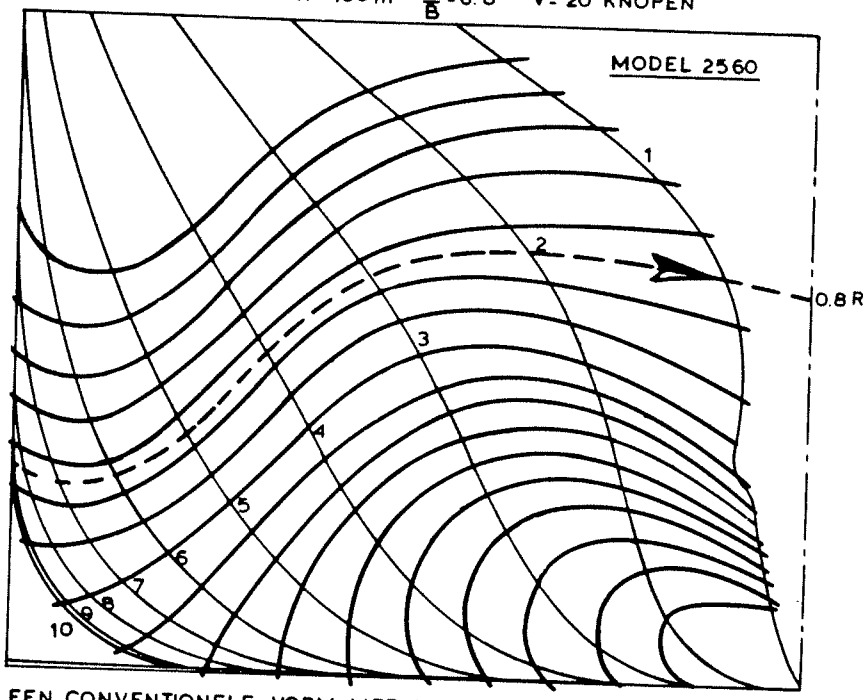
CONTRA "DRALL"

HIGHER PROPULSIVE EFFICIENCY

Bild 4 Hinterschiffsspanten, Variante 3

STROOMLIJNEN

VRACHTSCHIP 1963 LII = 150 m $\frac{L}{B}$ = 6.8 V = 20 KNOPEN

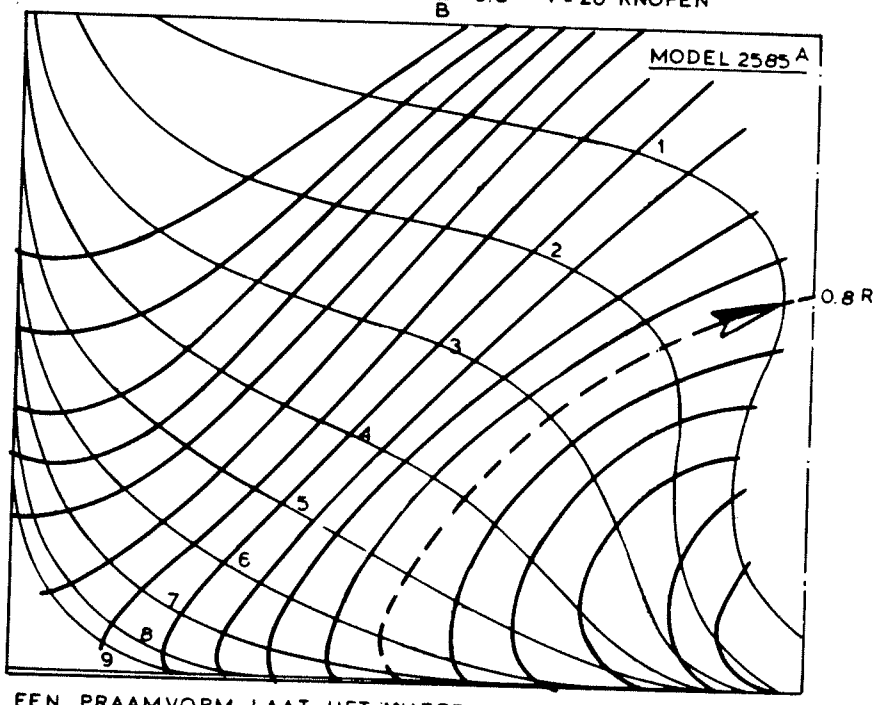


EEN CONVENTIONELE VORM MET MATIGE U SPANTEN LAAT HET WATER VAN DE ZIJDEN NAAR DE SCHROEF TOESTROMEN.

„ WATERLIJN STROMING"

(Shape of a 'conventional' cargo ship (1963) where moderately U-shaped frames allow the water to flow towards the propeller from the sides, so for the better part horizontally = 'Waterline Flow'.)

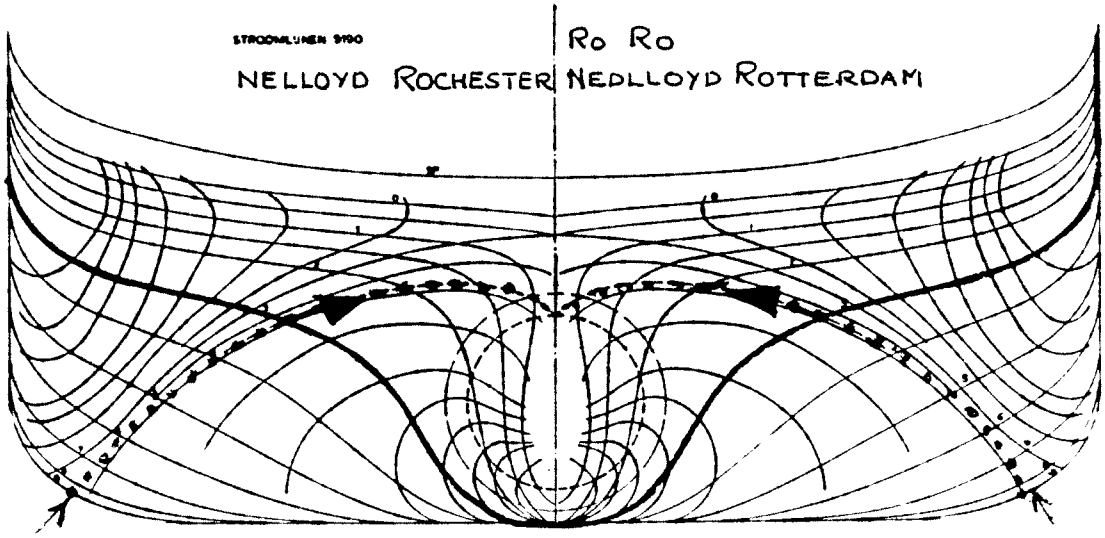
VRACHTSCHIP 1963 LII = 150 m $\frac{L}{B}$ = 6.8 V = 20 KNOPEN

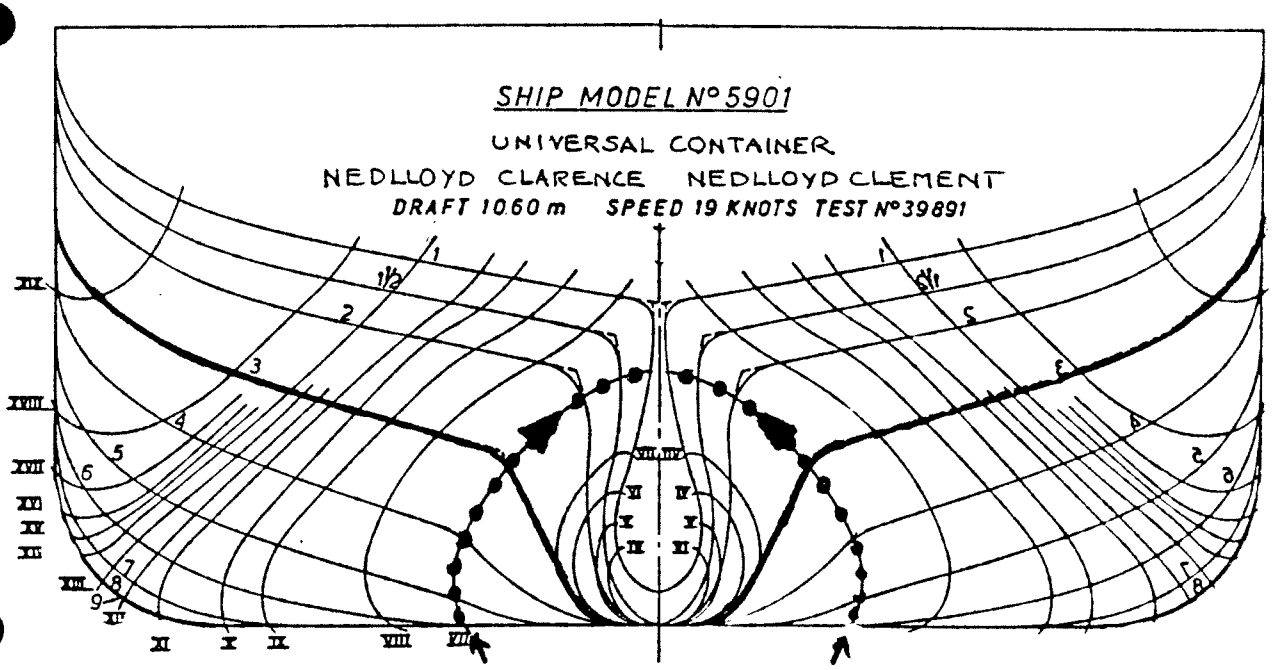


EEN PRAAMVORM LAAT HET WATER MEER VAN ONDER NAAR DE SCHROEF TOESTROMEN. "VERTIKALE STROMING"

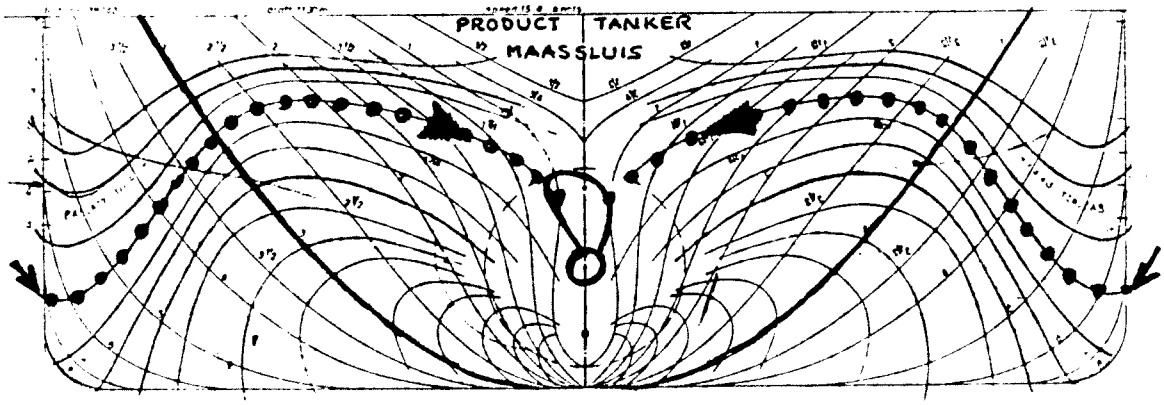
(Pram shape allows the water to approach the propeller more from below = 'Vertical Flow'.)

MODEL 5190



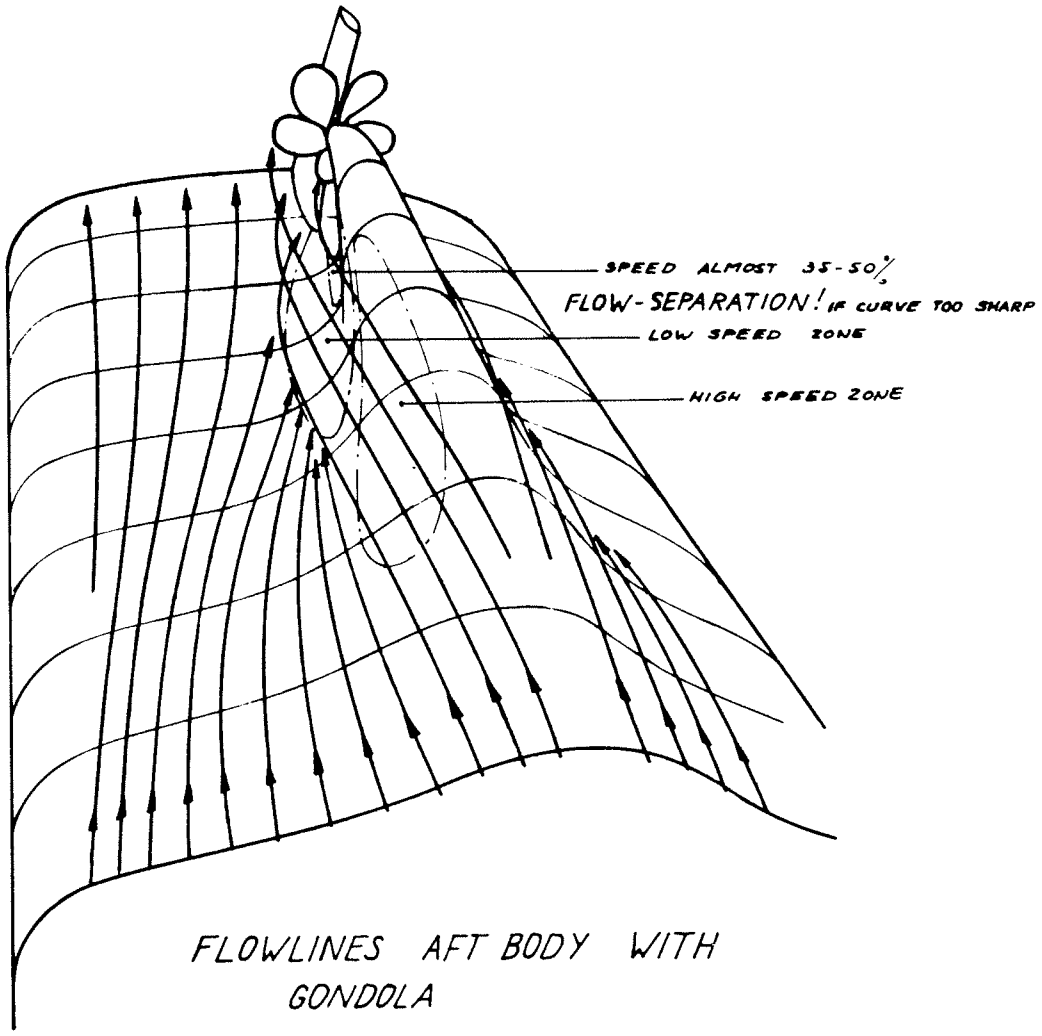


MODEL 5734B

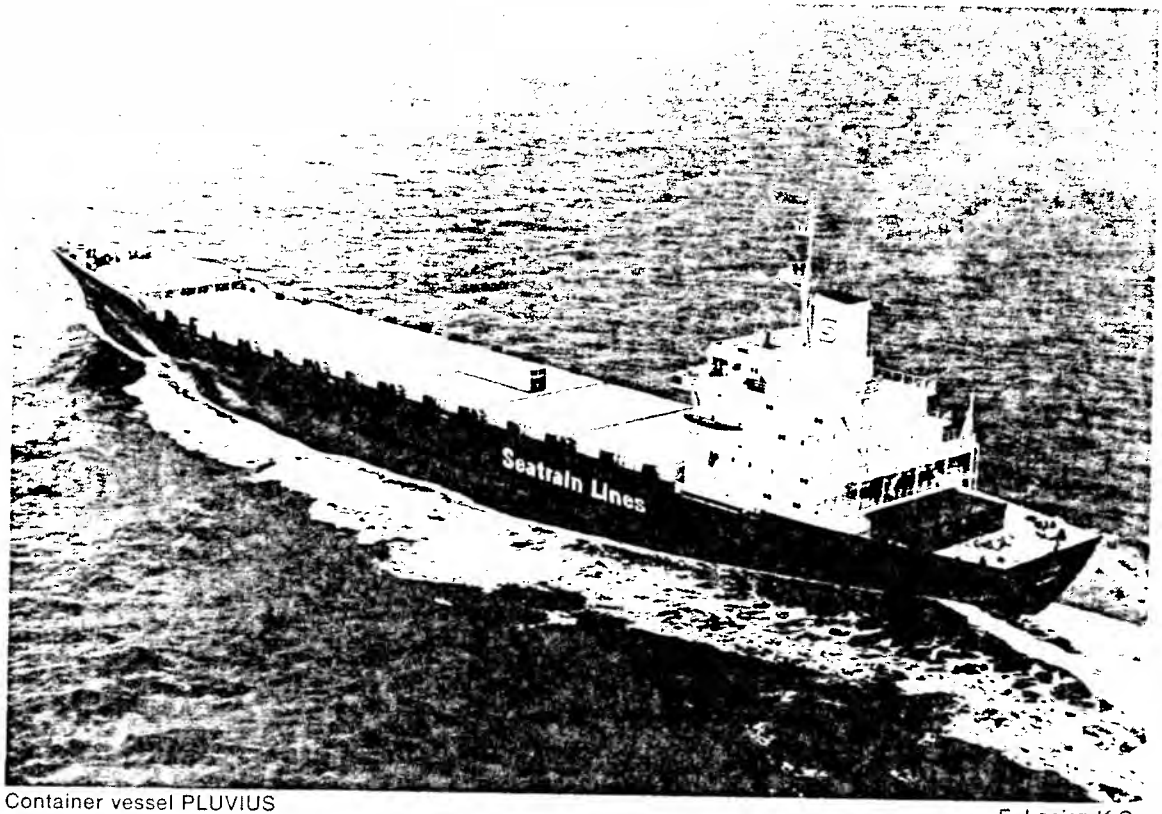


38 000 DWT PRODUCT TANKER MAASSLUIS

WATER FLOW LINES TOWARDS PROPELLER TIP IN TOP POSITION .



FLOWLINES AFT BODY WITH GONDOLA



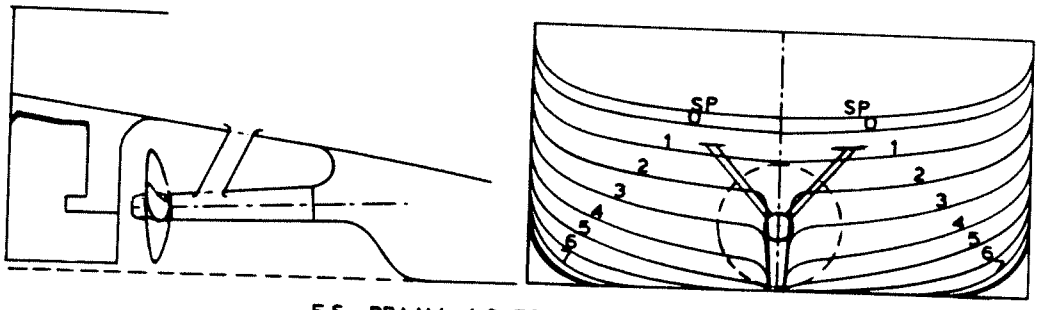
Container vessel PLUVIUS
13,400 tons d.w.

F. Laeisz K.G.,
Hamburg, Germany

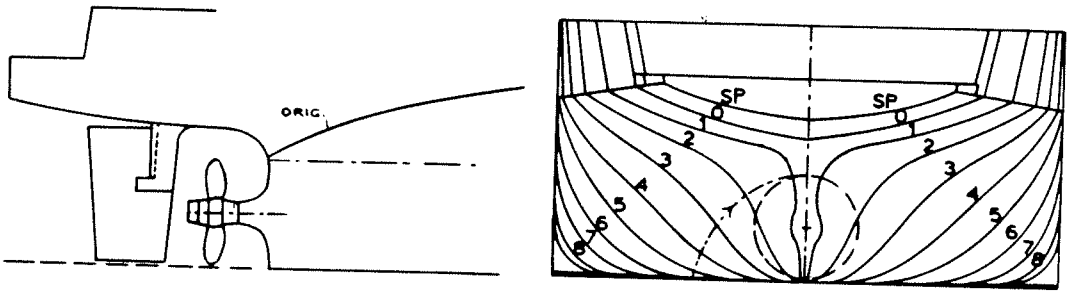
FLOW SEPARATION AT STERN

11

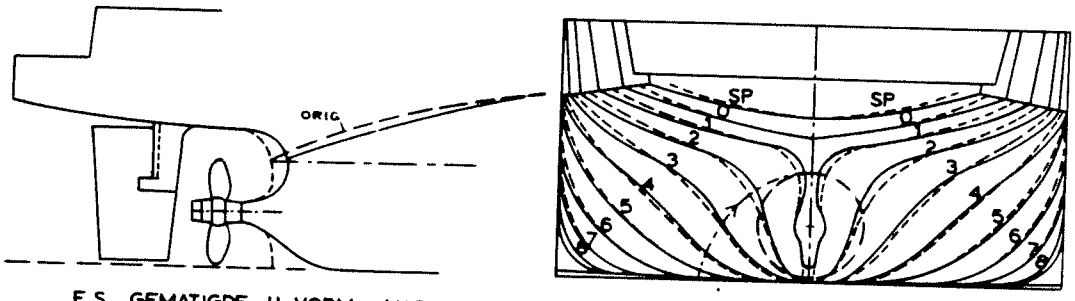
1S
RO-RO



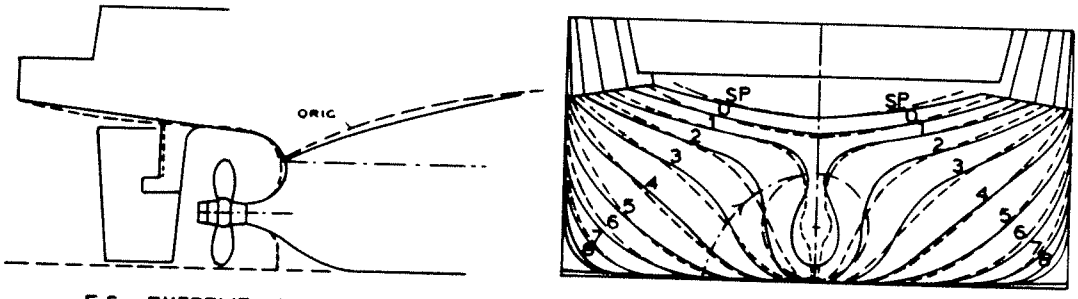
E.S. PRAAM (SAFCON) MODEL 4736A



E.S. V-VORM (WERF ONTWERP) MODEL 5100



E.S. GEMATIGDE U-VORM (N.S.U.) MODEL 5101 ——— . E.S. V-VORM MODEL 5100 ----



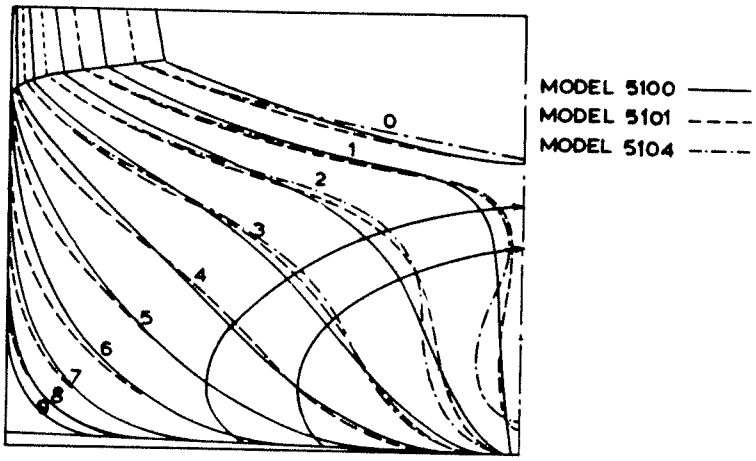
E.S. EXTREME U-VORM (N.S.P.) MODEL 5104 ——— . E.S. V-VORM MODEL 5100 ------

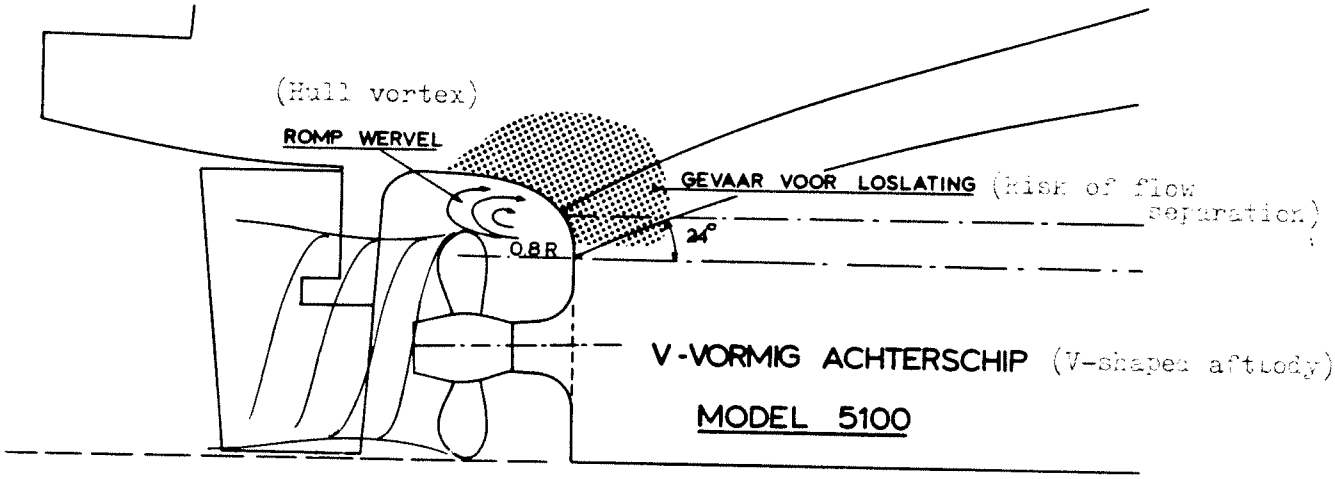
OPTIMALISERING ACHTERSCHIP

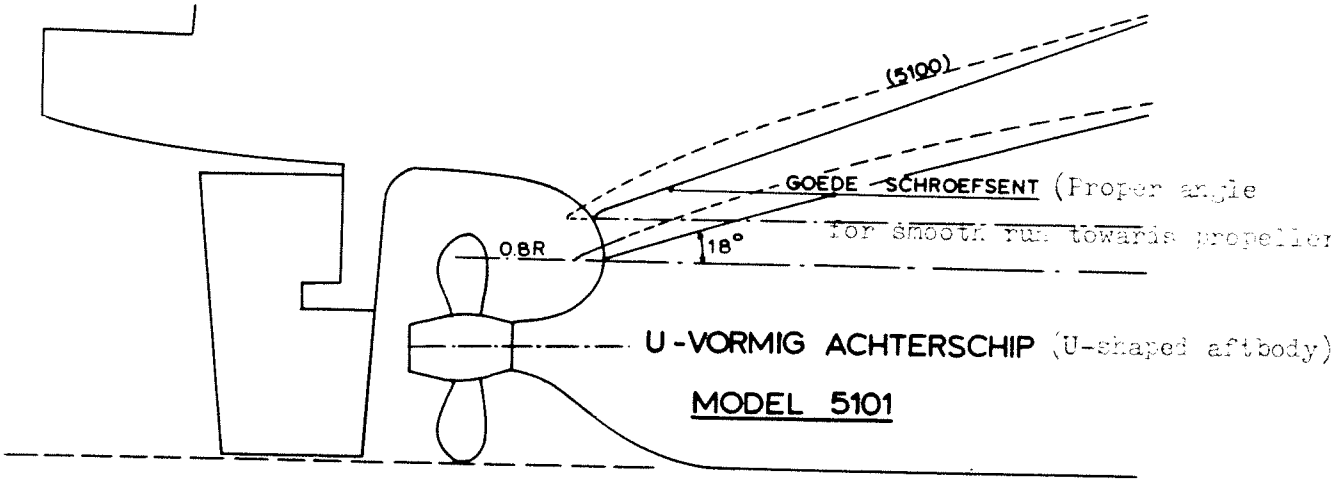
AUSTRALIË - MALAKKA RORO - CONTAINERSCHIP

— STREKKEN VAN HET SCHROEFSENT —

(Smoothening waterflow towards propeller
by sharpening ship's stern in front of propeller top)





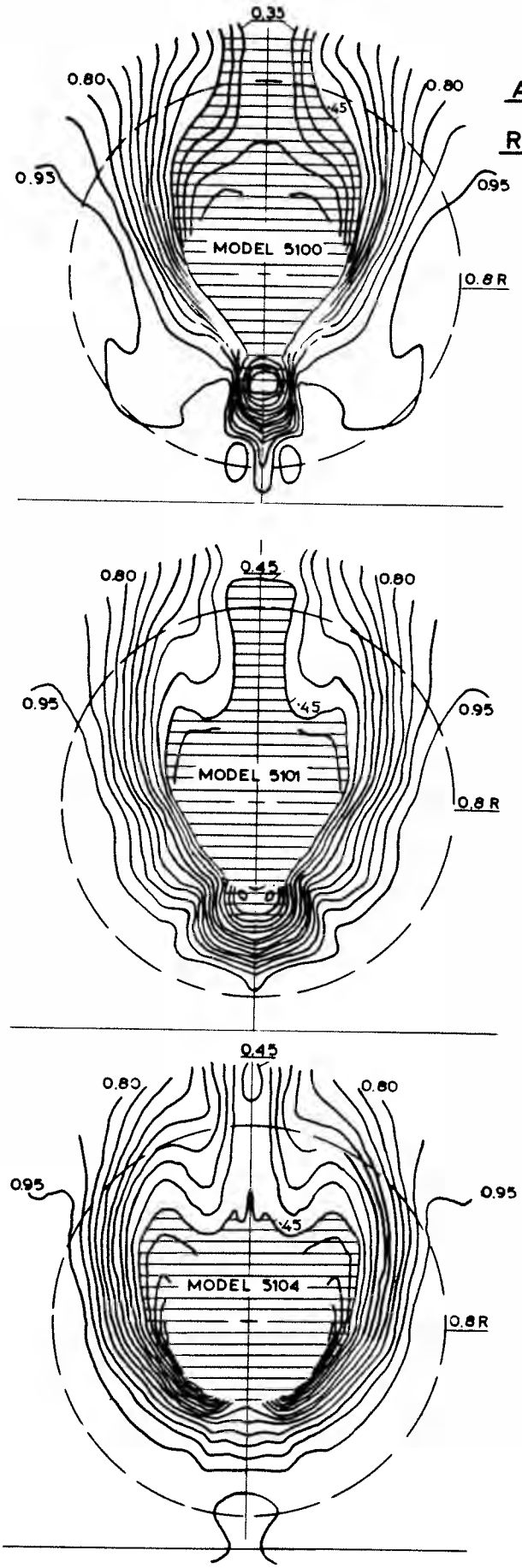


ONDERZOEK VAN HET STROOMBEELD BOVEN IN HET SCHROEFRAAM (Investigation into waterflow characteristics at top of the propeller aperture)

AUSTRALIË - MALAKKA RORO - CONTAINERSCHIP

AUSTRALIE-MALAKKA

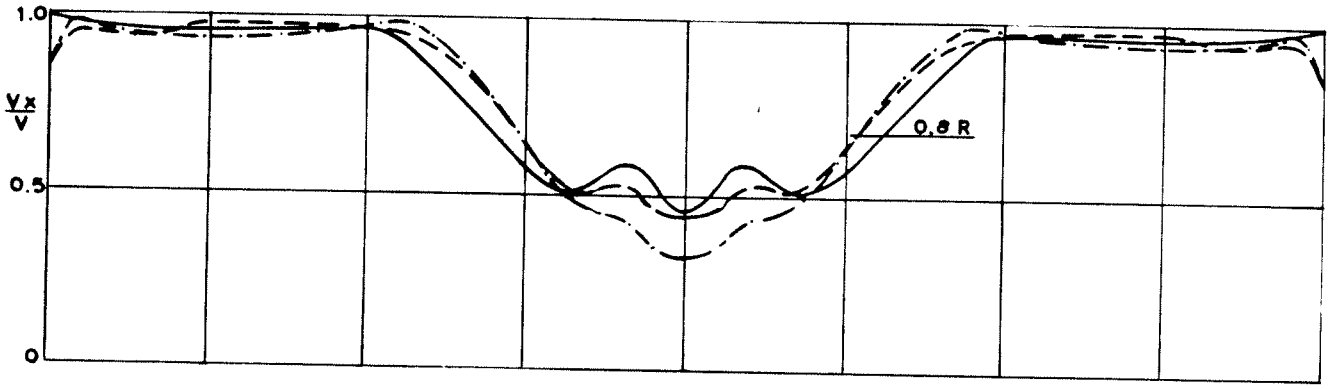
RO-RO CONTAINERSCHIP



KROMMEN VAN GELIJKE SNELHEID

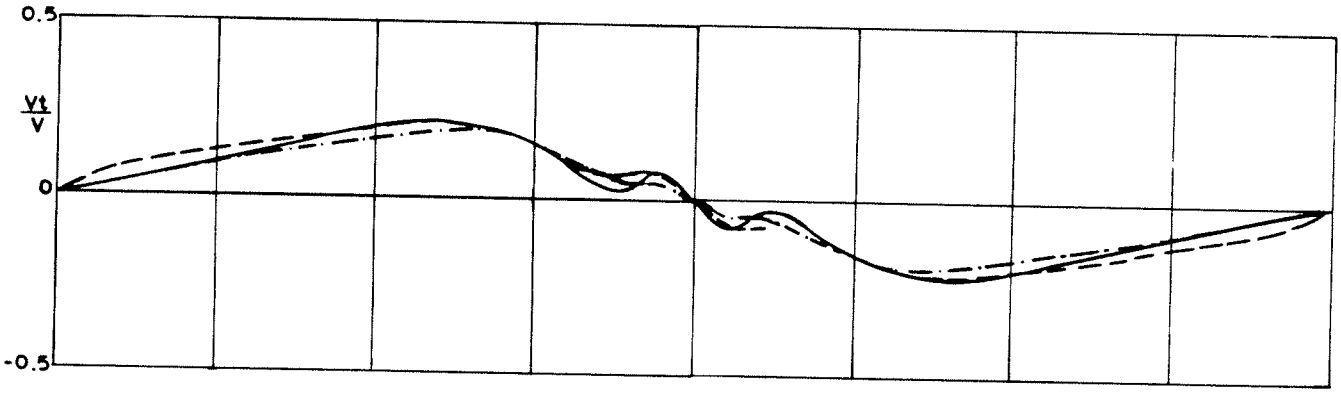
(Constant velocity curves)

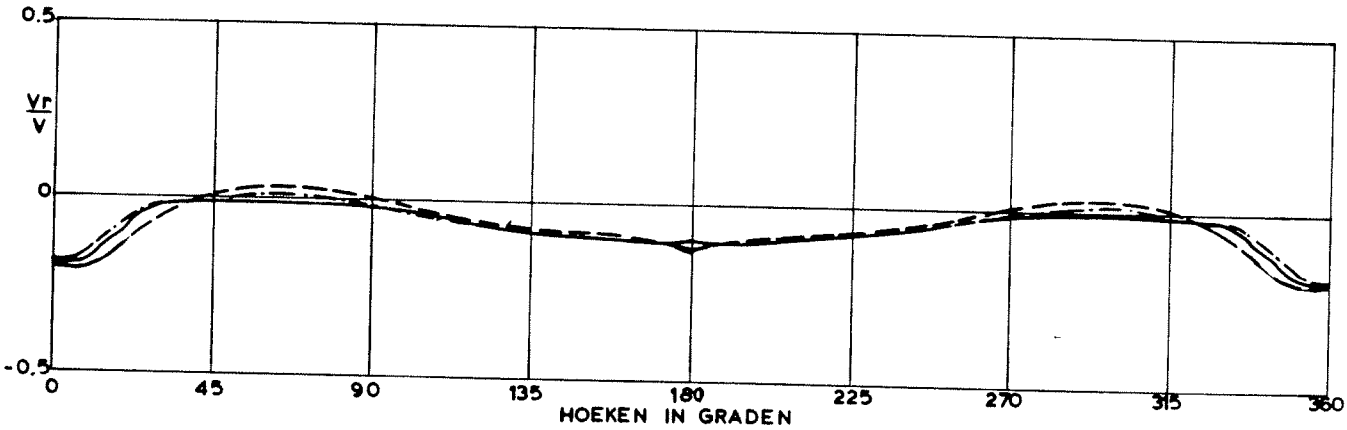
AUSTRALIE _ MALAKKA
RORO _ CONTAINERSCHIP



REDUCTIE VOLGSTROOMPIEK T.G.V. STREKKEN VAN HET SCHROEFSENT

(Reduction of wake peak by sharpening ship's trailing edge ahead of propeller top)

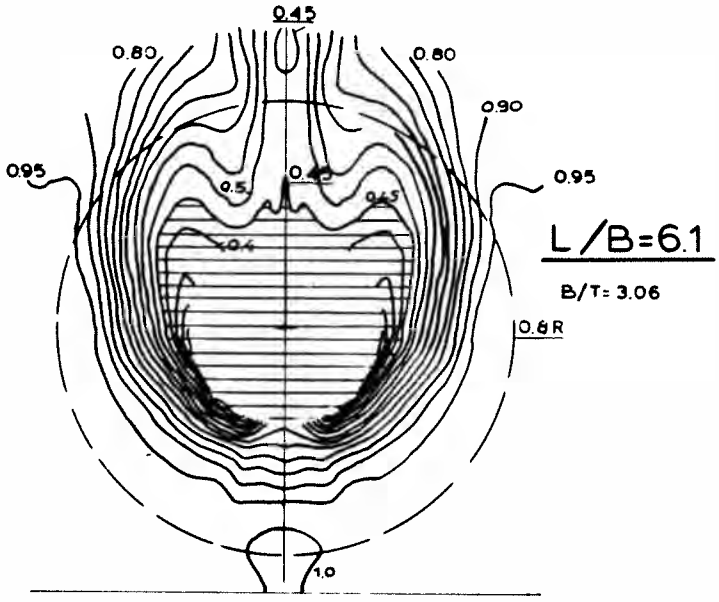




PERIFERIALE VERDELING VAN DE DRIE SNELHEIDSCOMPONENTEN

—·—·—·— = MODEL 5100 — — — — — = MODEL 5101 ——— = MODEL 5104

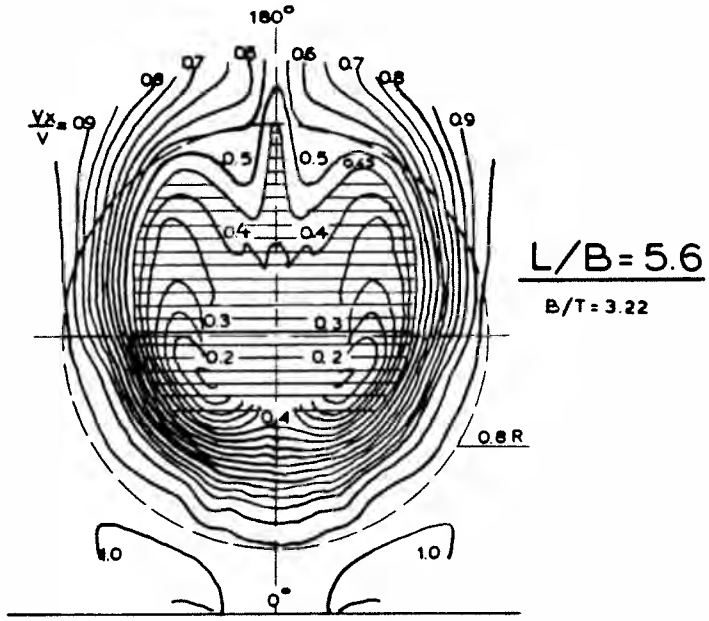
(Peripheral distribution of the three velocity components)



LUNEN VAN CONSTANTE SNELHEID

MODEL 5104

PROJECT AUSTRALIE -MALAKKA RORO CONTAINERSCHIP



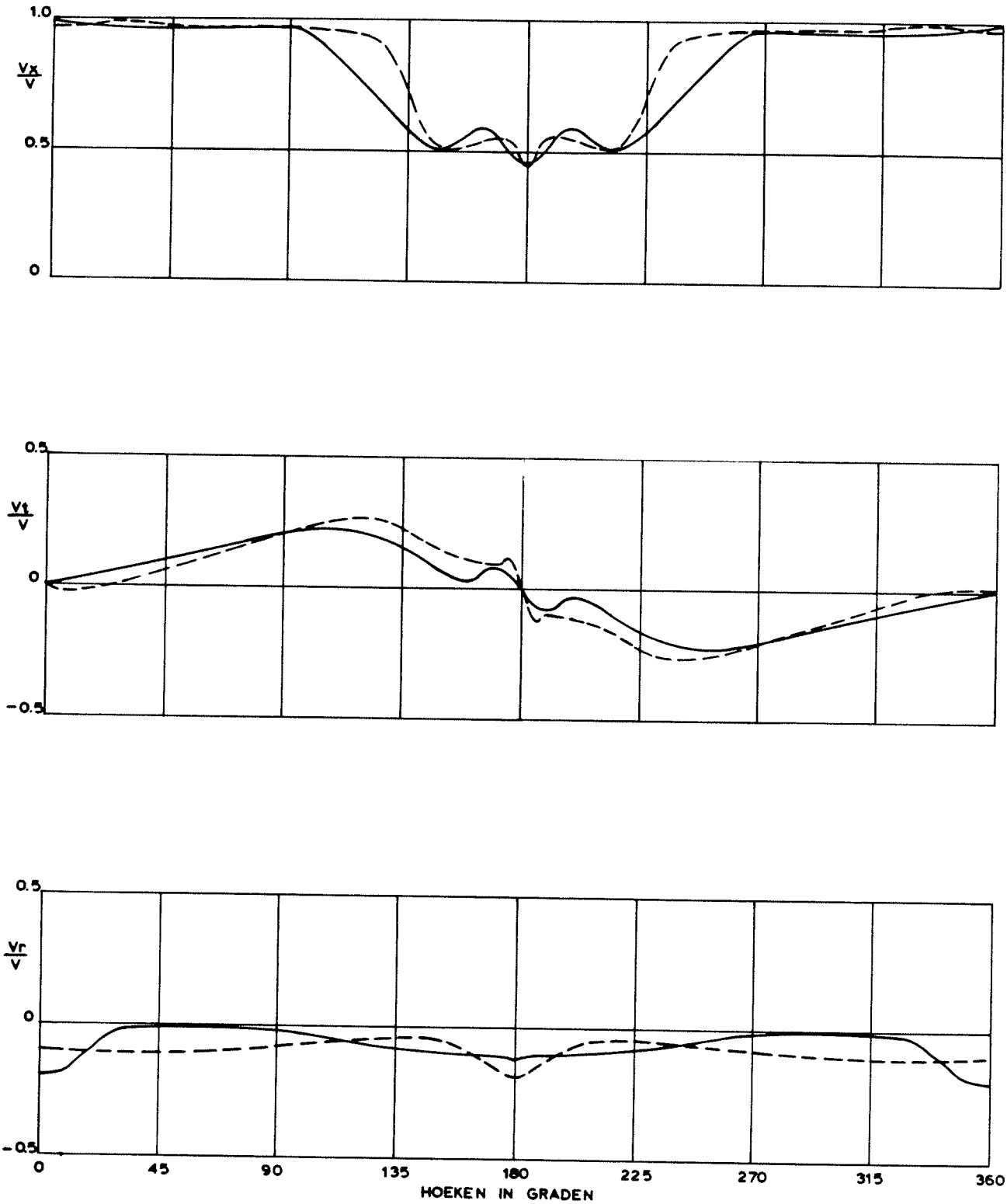
LUNEN VAN CONSTANTE SNELHEID

MODEL 5120

BASISVORM USA-ARABISCHE GOLF RORO TRAILERSCHIP

—— INVLOED L/B OP AXIALE VOLGSTROOM ——

(Influence of L/B factor on axial wake)



PERIFERIALE VERDELING VAN DE DRIE SNELHEIDSCOMPONENTEN

MODEL 5104 ———
L / B = 6.1
B / T = 3.1

MODEL 5120 - - - -
L / B = 5.6
B / T = 3.2

(Peripheral distribution of the three velocity components at S.& R)

— INVLOED L/B OP VOLGSTROOM —

(Influence of L/B factor on wake)

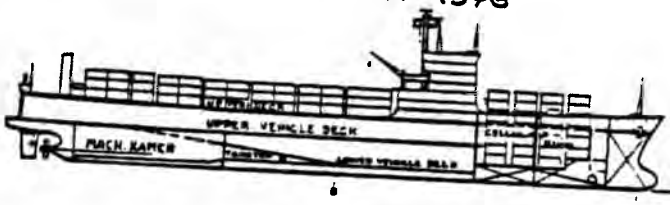
ROTTERDAM
ROCHESTER

LARGE SIZE SINGLE SCREW RO-RO'S

181,76 x 168,0 x 27,5 x 16,75 x 9,0 DESIGN-DRAFT

Australia Malaisia RoRo Containership

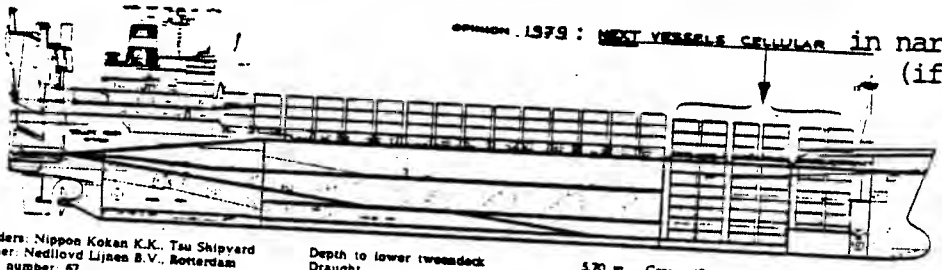
ANRO ASIA 1976



BELOW DECK = 522 = 134
ON DECK 2 T = 290 66
TOTAL 2 T = 812 200
TOTAL 3 T = 971

M/V. "NEDLLOYD ROUEN" - type

OPINION 1979 : NEXT VESSELS CELLULAR in narrow forepart. (if vessel is long enough).



Builders: Nippon Kokan K.K., Tsu Shipyard
Owner: Nedlloyd Lijnen B.V., Rotterdam
Yard number: 67
Type: ro ro vessel
Delivery: October 12th, 1978

Tonnage

	21 144.61 GRT
	11 698.77 NRT
Deadweight	29 218 t
Length o. a.	212.10 m
Length b. p.	198.83 m
Moulded breadth	32.24 m
Depth to maindeck	20.24 m
Depth to upper tweendeck	13.05 m
Depth to lower tweendeck	5.70 m
Draught	10.72 m

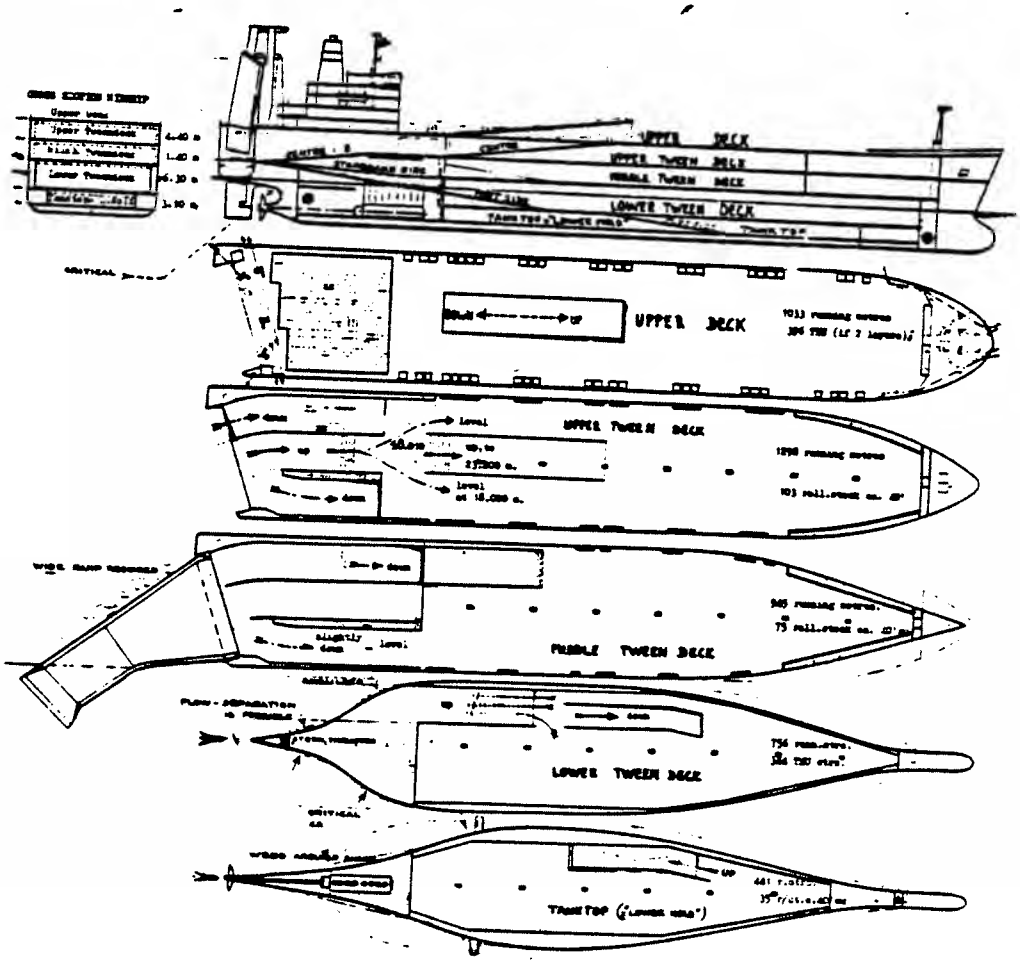
Speed (16 600 kW): 20.66 kn

Classification:
L.R. ✠ 100 A 1

Propelling machinery:
Sumitomo/Sulzer-Diesel engine
type: 8 RND 90 M
19 711 kW (26 800 HP) at 116 rpm
Fuel consumption: 156.56 g/HPh
1 bow thruster
1 stern thruster
1 gyro-fin stabilizer
gyro compass and automatic pilot (Anschütz)

Crew: 40 persons

1 stern ramp
1 weather-tight stern door 25.00 × 7.60 m
1 gas-tight rampway cover 5.60 × 35.00 m
1 gas-tight door 9.04 × 6.70 m
1 gas-tight door 9.05 × 6.33 m
1 weather-tight casing door 9.10 × 6.45 m
Loading gear: 1 crane 3 t
8 electric hydraulic automatic mooring winches of 12 t
2 electric hydraulic windlasses of 31 t



USA Arabische Golf RoRo Trailership

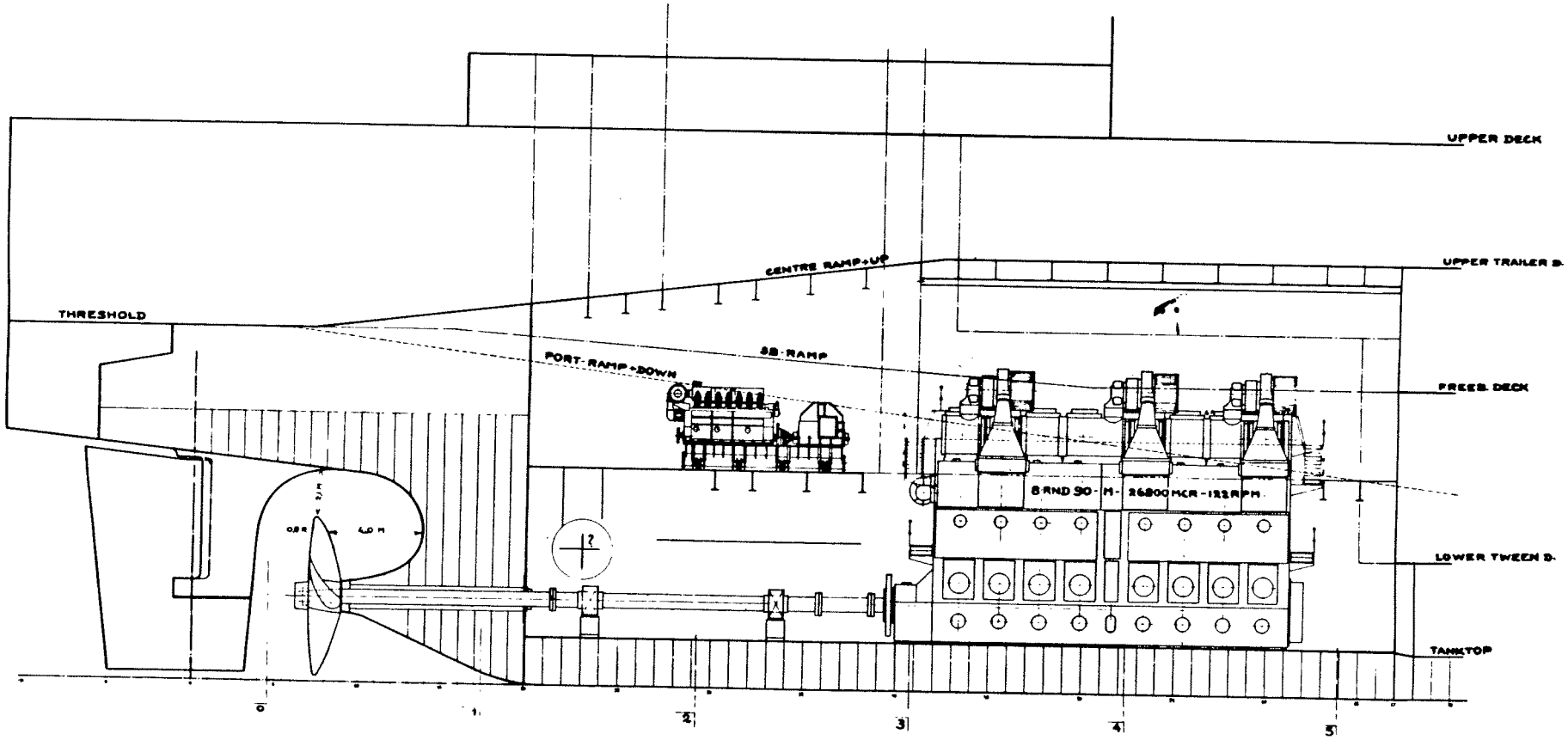
PROJECT 1976

NEDLLOYD ROTTERDAM

RORO USA - ARABISCHE GOLF

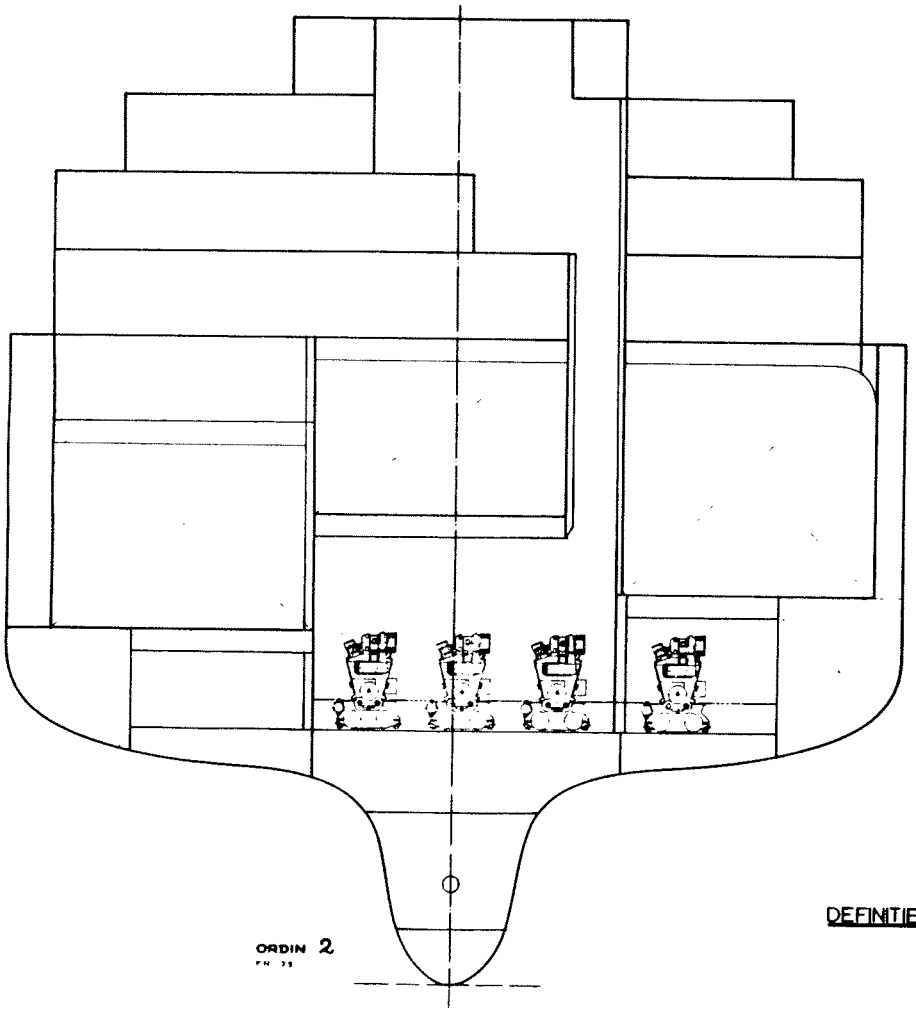
LANGSDOORSNEDE ACHTERSCHIP

DEFINITIEVE VORM

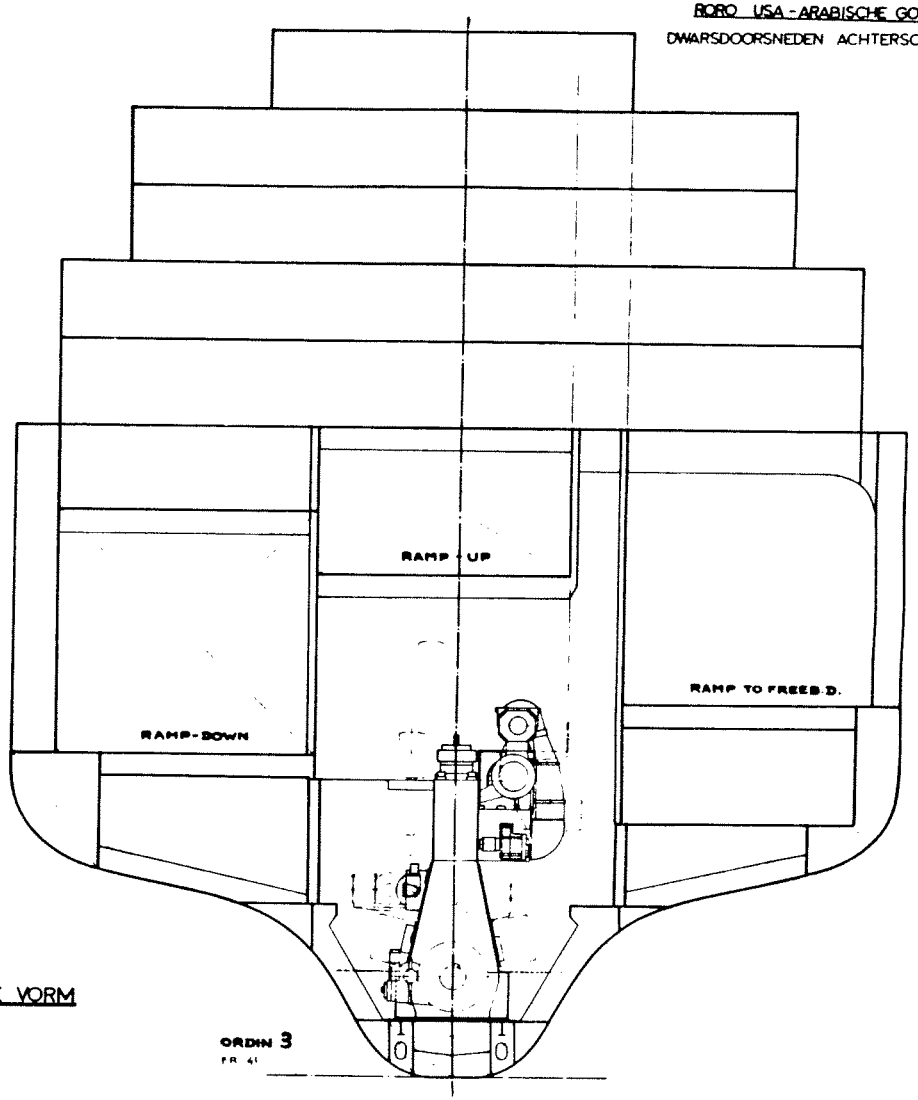


RORO USA - ARABISCHE GOLF

DWARSDOORSNEDEN ACHTERSCHIP



DEFINITIEVE VORM



(Summary and analysis of powering results)

Overzicht en analyse van de vermogensresultaten.

	Europa - Zuid Afrika Containerschip				Australië - Malakka Ro-Ro - Containerschip			
	L_{oa} = 258,80 m L_{WL} = 252,16 m L_{11} = 247,00 m B = 32,26 m				L_{oa} = 182,00 m L_{WL} = 175,50 m L_{11} = 168,00 m B = 27,50 m			
Model No.	4736A	4751C	4751D	4751D	5100	5101	5101B	5104
E.S./D.S.	E.S.	D.S.	D.S.	D.S.	E.S.	E.S.	E.S.	E.S.
Achterschip	praam	praam	praam	praam	V-vorm	U-vorm	U-vorm	Hogner
Voorschip	11,7% bulb				4% bulb	4% bulb	7½% bulb	4% bulb
Vermogen	40000 APK	2x26800 APK	2x26800 APK	2x26800 APK	16400 APK	16400 APK	16400 APK	16400 APK
RPM	100	122	122	122	110	110	110	110
⊕	-2,29%	-2,41%	-2,51%	-3,11%	-1,4%	-1,4%	-1,4%	-1,4%
T	11,0	11,0	11,0	12,0	9,0	9,0	9,0	9,0
C_B	0,628	0,636	0,639	0,656	0,614	0,615	0,617	0,615
∇ (m^3)	55179	55780	55805	62696	25523	25570	25647	25553
S (m^2)	10468	10883	10561	11186	5573	5620	5665	5607
V_s (kn.)	22½	22½	22½	22½	19	19	19	19
P_E wrijv.	16264	17288	16800	17758	5301	5345	5376	5333
P_E rest.	5965	6010	4709	6270	3502	3340	2865	3368
P_E	22229	23298	21509	24028	8803	8685	8241	8701
η_D	0,648	0,716	0,725	0,709	0,703	0,700	0,726	0,714
P_D	34303	32526	32298	36596	12495	12409	11345	12193
P_D (%)	-	-	-	-	100	99	91(•)	97,5
RPM	103,5	106,7	106,0	110,4	107,4	107,5	105,7	106,9
S	173,79	167,05	167,82	185,66	80,11	81,60	75,40	82,87
W	144,03	150,96	139,37	169,54	67,54	66,64	63,24	66,77
S-W	29,76	16,09	28,45	16,12	12,57	14,96	12,16	16,10
t	0,171	0,086	0,085	0,087	0,157	0,183	0,161	0,194
w	0,142	0,113	0,111	0,113	0,247	0,247	0,239	0,258
$\frac{1-t}{1-w}$	0,966	1,019	1,037	1,030	1,119	1,085	1,103	1,087
η_B	0,671	0,693	0,694	0,694	0,629	0,645	0,659	0,657
η_D	0,672	0,704	0,703	0,704	0,630	0,629	0,629	0,626
η_B/η_O	1,000	0,986	0,985	0,986	0,999	1,023	1,046	1,049
EXTRAPOLATIEMETHODE	I.T.T.C. MET ACF = 0.0001				I.T.T.C. MET ACF = NUL			

	U.S.A. - Arabische Golf Ro-Ro - Trailerschip																	
	L_{oa} = 194,00 m L_{WL} = 188,50 m L_{11} = 180,00 m B = 32,24 m																	
Model No.	5120	5121	5122	5131	5141	5150	5122	5141A	5141B	5141C	5141D	5141E	5190		
E.S./D.S.	E.S.	E.S.	E.S.	D.S.	E.S.	E.S.	E.S.	E.S.	E.S.	E.S.	E.S.	E.S.	E.S.		
Achterschip	Hogner	praam	praam	praam	praam	praam	praam	praam	praam	praam	praam	praam	praam		
Voorschip	vol voorschip, 9% bulb			slank voorschip met 9% bulb											
Vermogen	23450 APK	23450 APK	23450 APK	2x12145 APK	23450 APK	23450 APK	23450 APK	23450 APK	23450 APK	23450 APK	23450 APK	23450 APK	23450 APK	(*)	
RPM	122	122	122	122	122	122	122	122	122	122	122	122	122	(* *)	
⊕	-1,33%	-1,37%	-1,34%	-2,29%	-2,23%	-2,63%	-1,85%	-2,33%	-2,37%	-2,44%	-2,23%	-2,06%	-2,14%		
T	10,0	10,0	10,0	9,82	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0	10,0		
C_B	0,615	0,614	0,616	0,626	0,610	0,616	0,604	0,612	0,612	0,613	0,610	0,607	0,609		
∇ (m^3)	35704	35766	35655	35770	35405	35775	35060	35489	35513	35563	35425	35229	35348		
S (m^2)	6886	7020	6963	7211	7161	7236	6896	7164	7164	7168	7115	7110	7130		
V_s (kn.)	20	20	20	20	20	20	20	20	20	20	20	20	20		
P_E wrijv.	8030	8186	8120	8723	8350	8438	8042	-	8354	8359	8297	8291	8315		
P_E rest.	4743	4858	4720	4566	5018	5355	4650	-	5108	5245	4920	4610	4402		
P_E	12773	13044	12840	13289	13368	13793	12692	-	13462	13604	13217	12901	12717		
η_D	0,693	0,715	0,697	0,695	0,722	0,710	0,686	-	0,709	0,706	0,713	0,686	0,689		
P_D	18430	18232	18432	19117	18514	19434	18503	18959	18975	19259	18530	18802	18467		
P_D (%)	100	99	100	104	101	105	101	102	102,5	104	101	102	100		
RPM	117,8	117,1	118,0	145,8	118,0	120,9	117,8	119,0	118,6	118,3	119,4	121,6	122,1		
S	124,33	125,26	123,52	108,94	123,52	127,76	122,76	127,76	128,41	129,47	121,31	120,52	117,89		
W	93,11	95,08	93,52	96,87	97,44	100,54	92,52	-	98,13	99,17	96,34	94,04	92,70		
S-W	31,22	30,18	30,00	12,07	28,08	27,22	30,24	-	30,28	30,30	24,97	26,48	25,19		
t	0,251	0,241	0,242	0,111	0,224	0,213	0,246	-	0,236	0,234	0,206	0,220	0,214		
w	0,336	0,347	0,331	0,106	0,337	0,306	0,334	0,331	0,339	0,350	0,306	0,272	0,254		
$\frac{1-t}{1-w}$	1,128	1,162	1,132	0,994	1,170	1,134	1,133	-	1,155	1,179	1,144	1,071	1,054		
η_B	0,614	0,616	0,615	0,699	0,617	0,626	0,606	0,619	0,614	0,599	0,624	0,640	0,653		
η_D	0,600	0,596	0,603	0,663	0,599	0,608	0,601	0,600	0,596	0,590	0,612	0,623	0,630		
η_B/η_O	1,023	1,032	1,021	1,055	1,029	1,030	1,007	1,032	1,030	1,015	1,018	1,028	1,037		
EXTRAPOLATIEMETHODE	I.T.T.C. MET ACF = 0.0001														

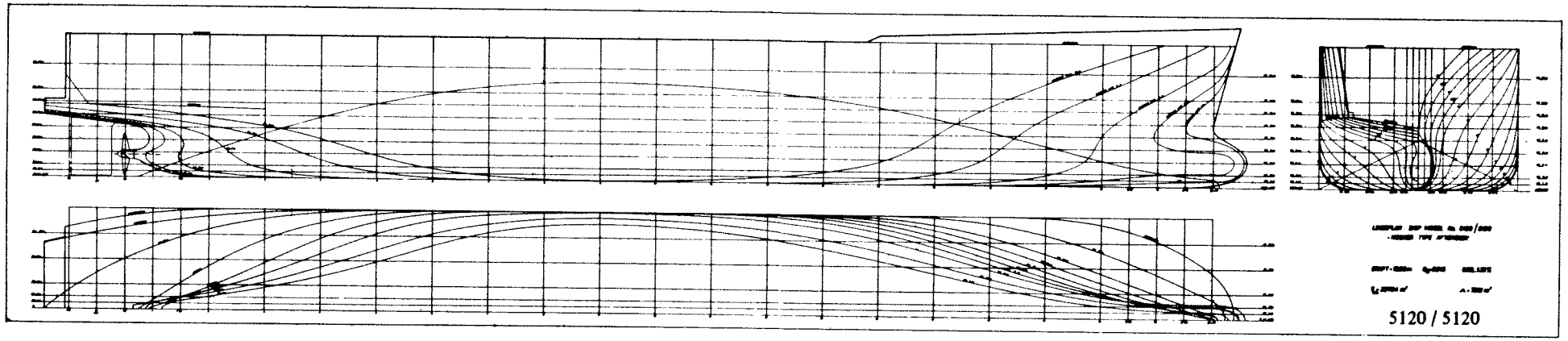
waarin:

- t = zoggetal
- w = volgstroomgetal
- $\frac{1-t}{1-w}$ = "hull efficiency"
- η_B = schroefrendement achter model
- η_O = schroefrendement vrijvarend
- η_B/η_O = overgangscoefficient.

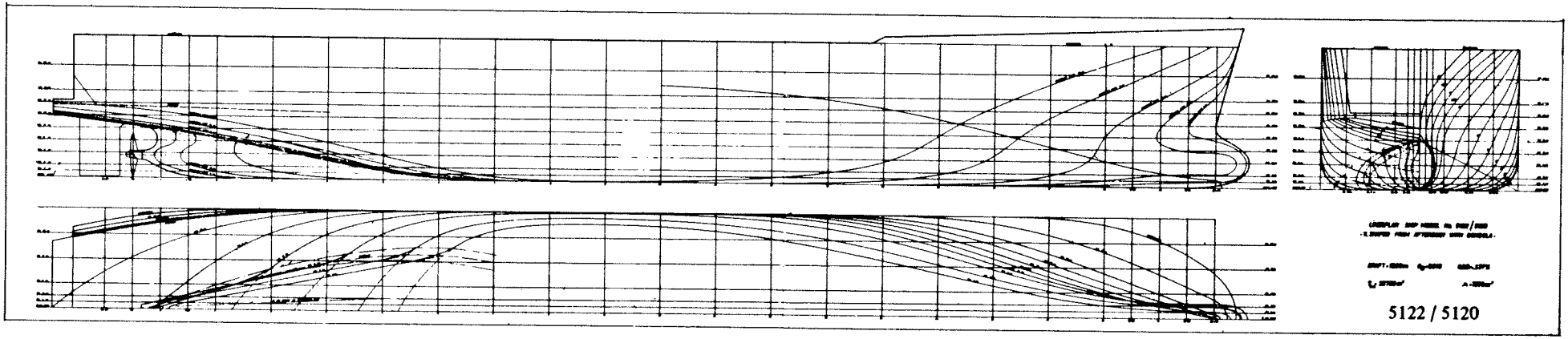
(•) Opmerking: Dit gunstige resultaat is het gevolg van de grote bulb. (= This good result is due to the large forward bulb.)

(*) E.S. = Single screw
D.S. = Twin screw
Achterschip = Aftbody
Voorschip = Fore body
(* *) Vermogen = Power
A.P.K. = S.H.P.

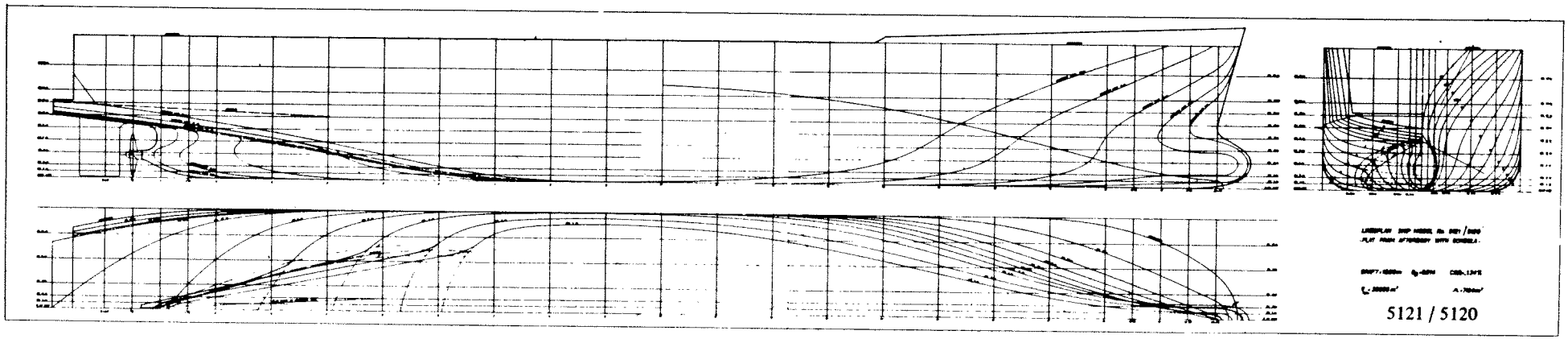
t = thrust deduction factor
w = wake
η_B = propeller efficiency behind model
η_O = propeller efficiency in open water
η_B/η_O = efficiency ratio



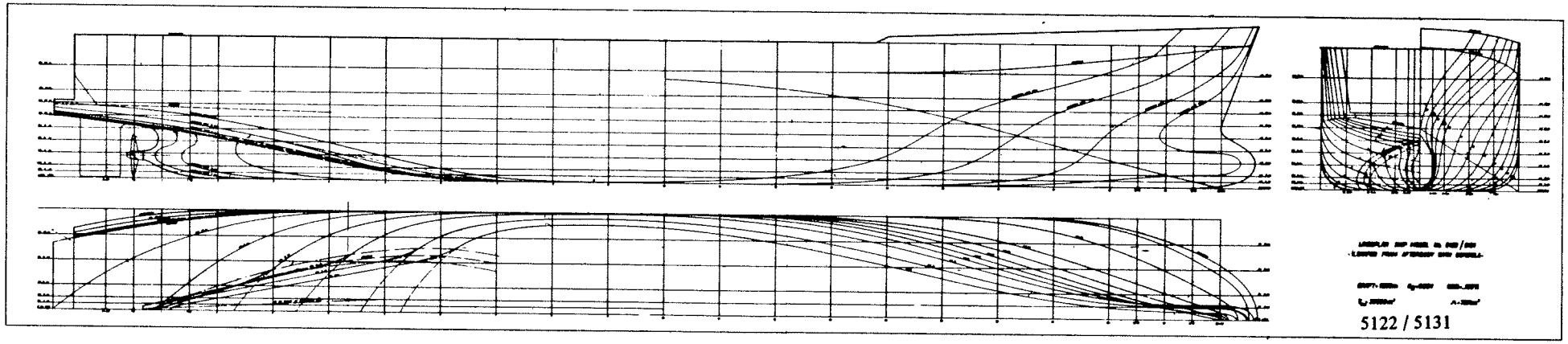
5120 / 5120



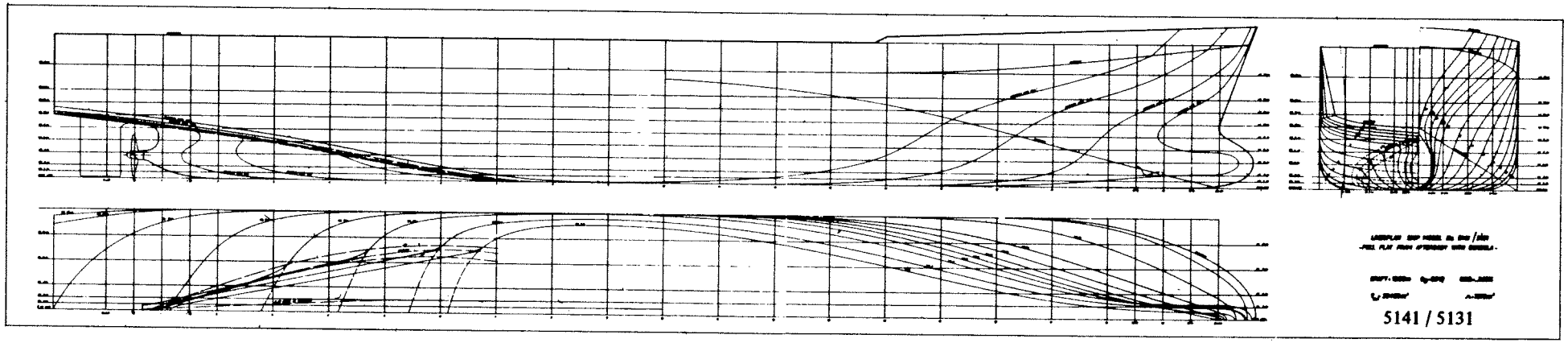
5122 / 5120



5121 / 5120

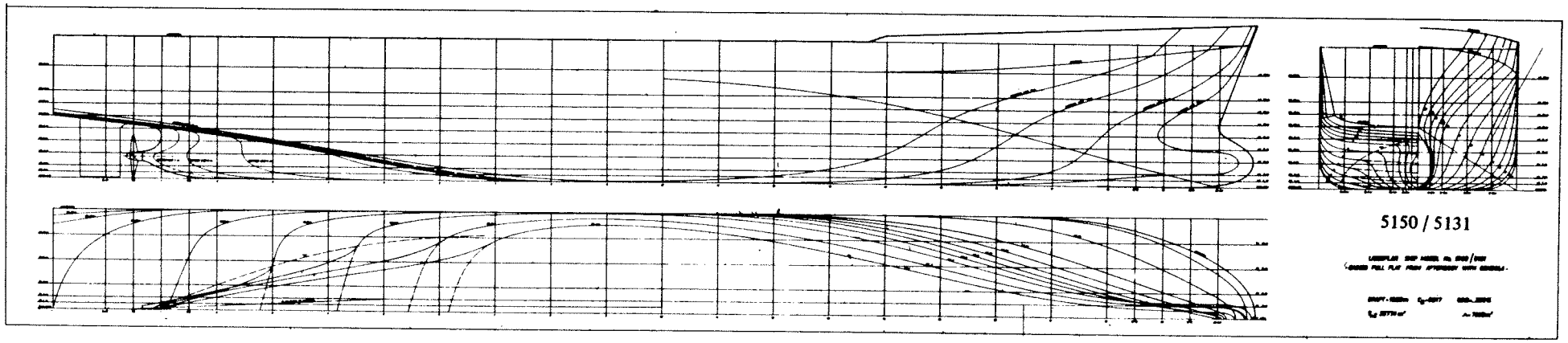


5122 / 5131

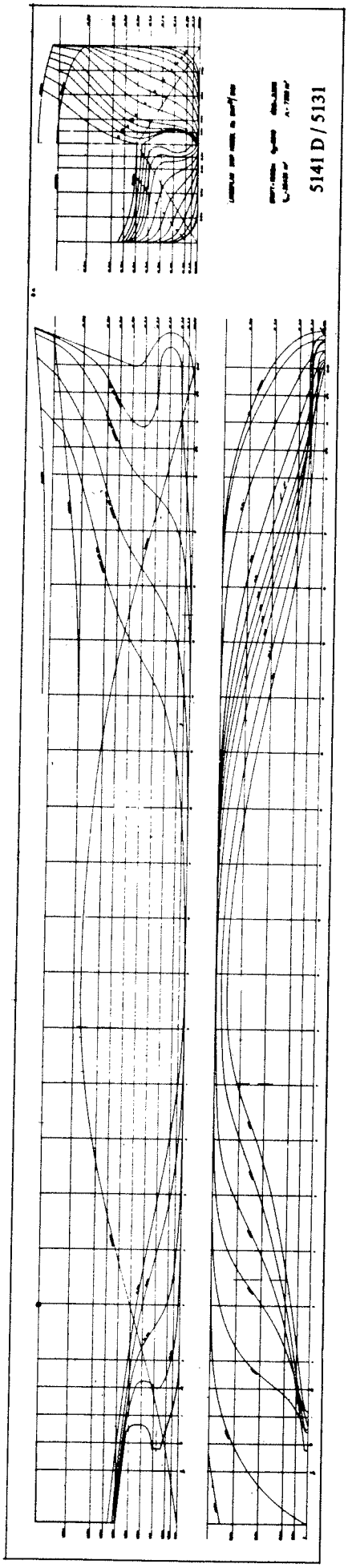


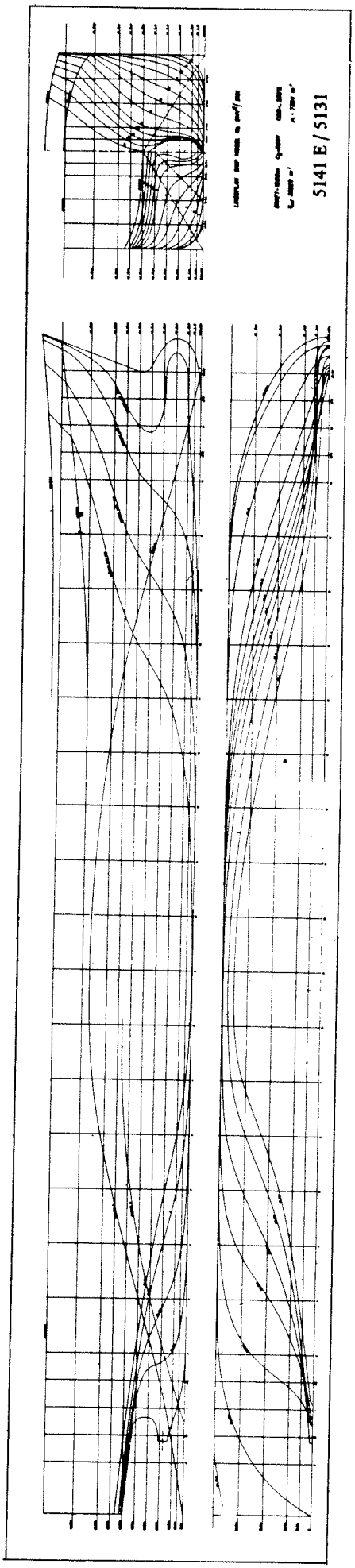
5141 / 5131

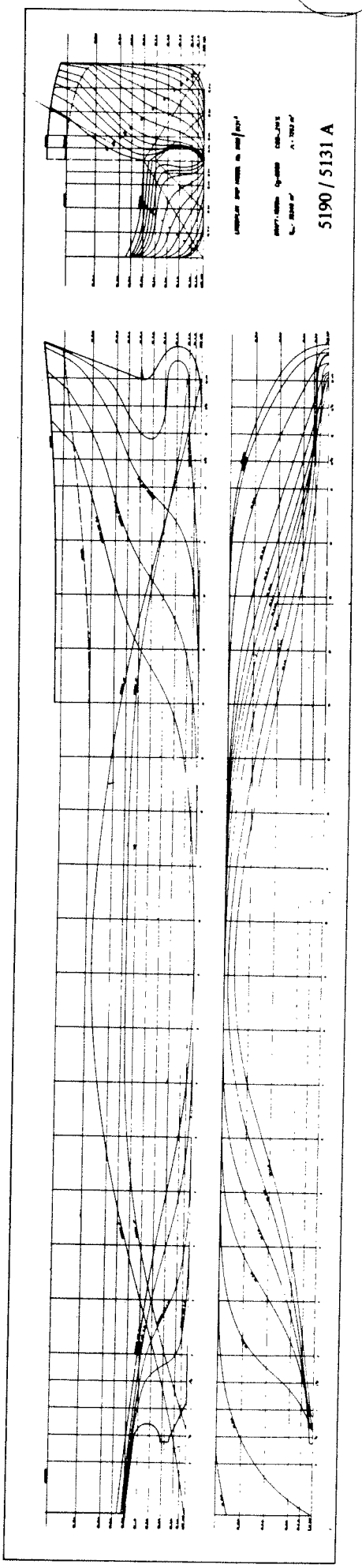
39
BOVEN

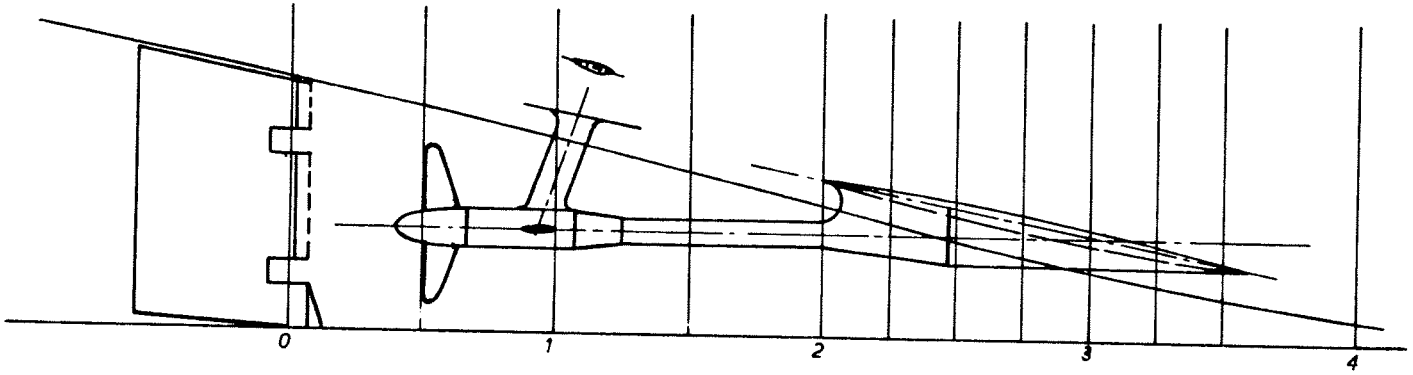


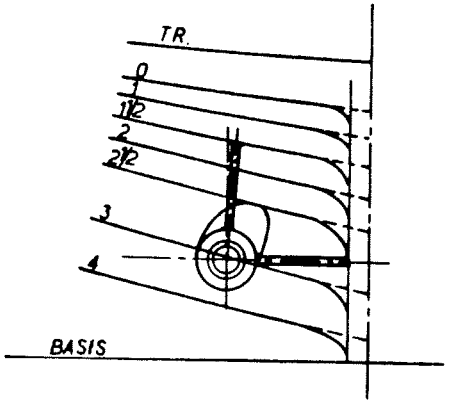
5150 / 5131


5141 D / 5131


5141 E / 5131

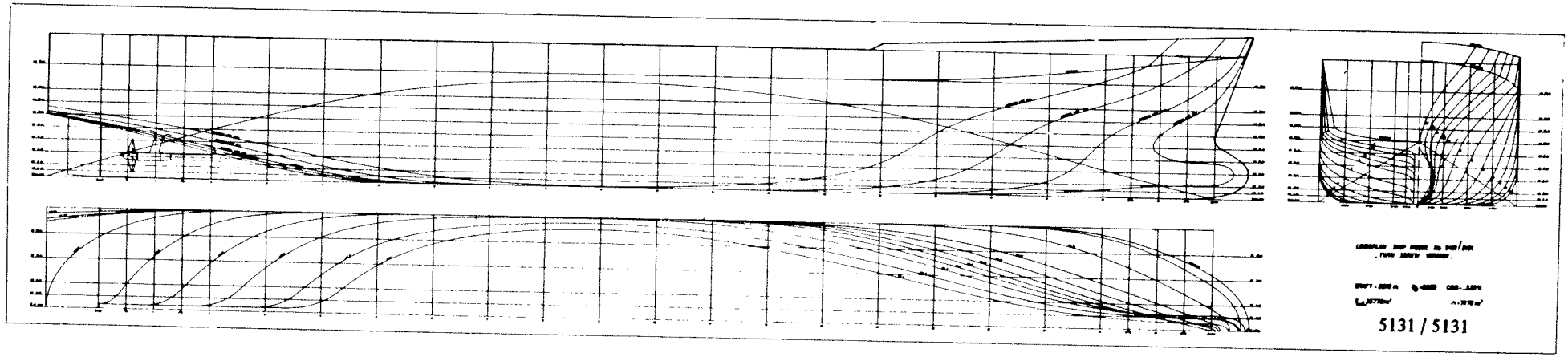

5190 / 5131 A





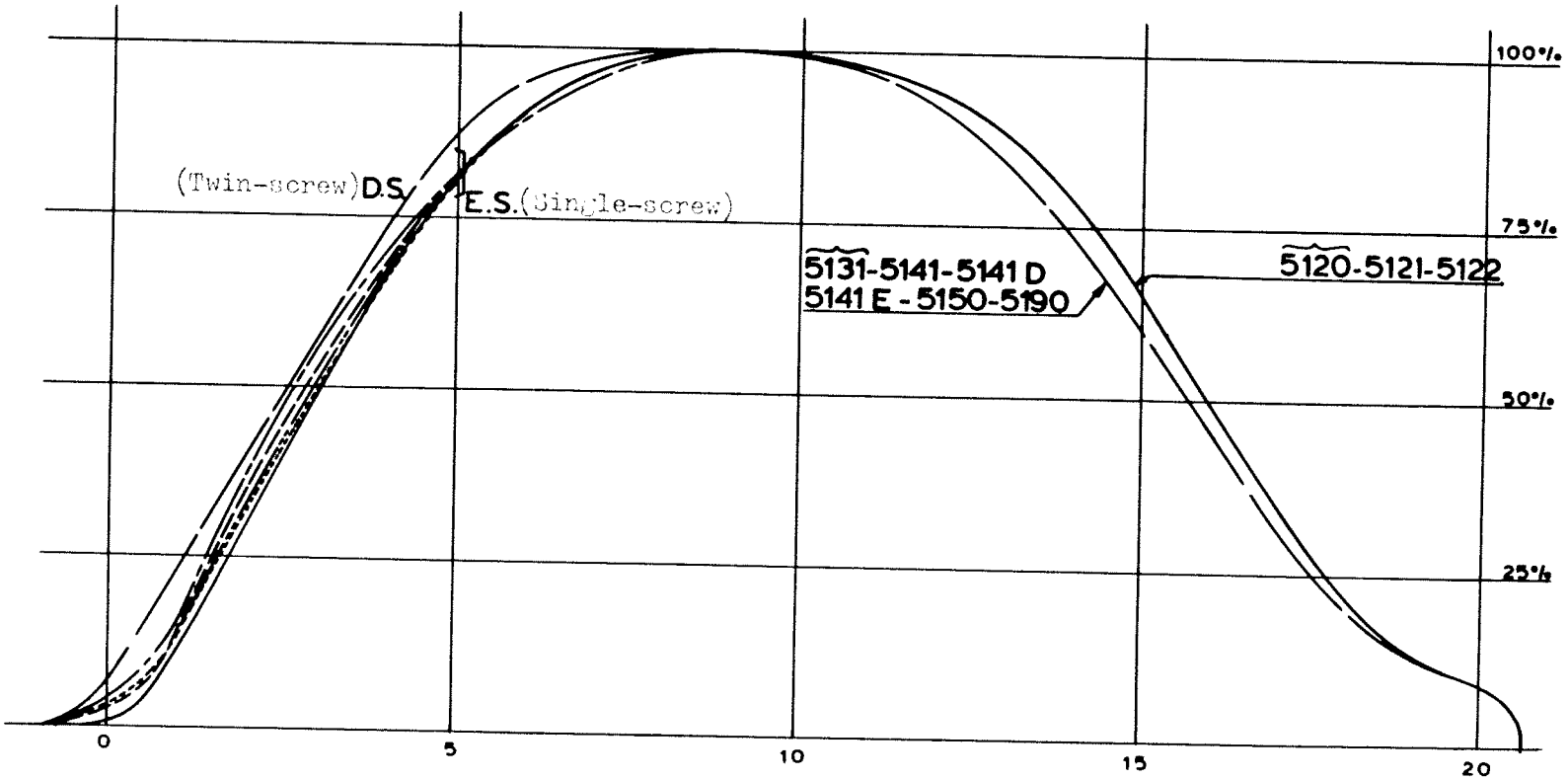
DETAIL ASUITHOUDERS

(Shaft brackets)

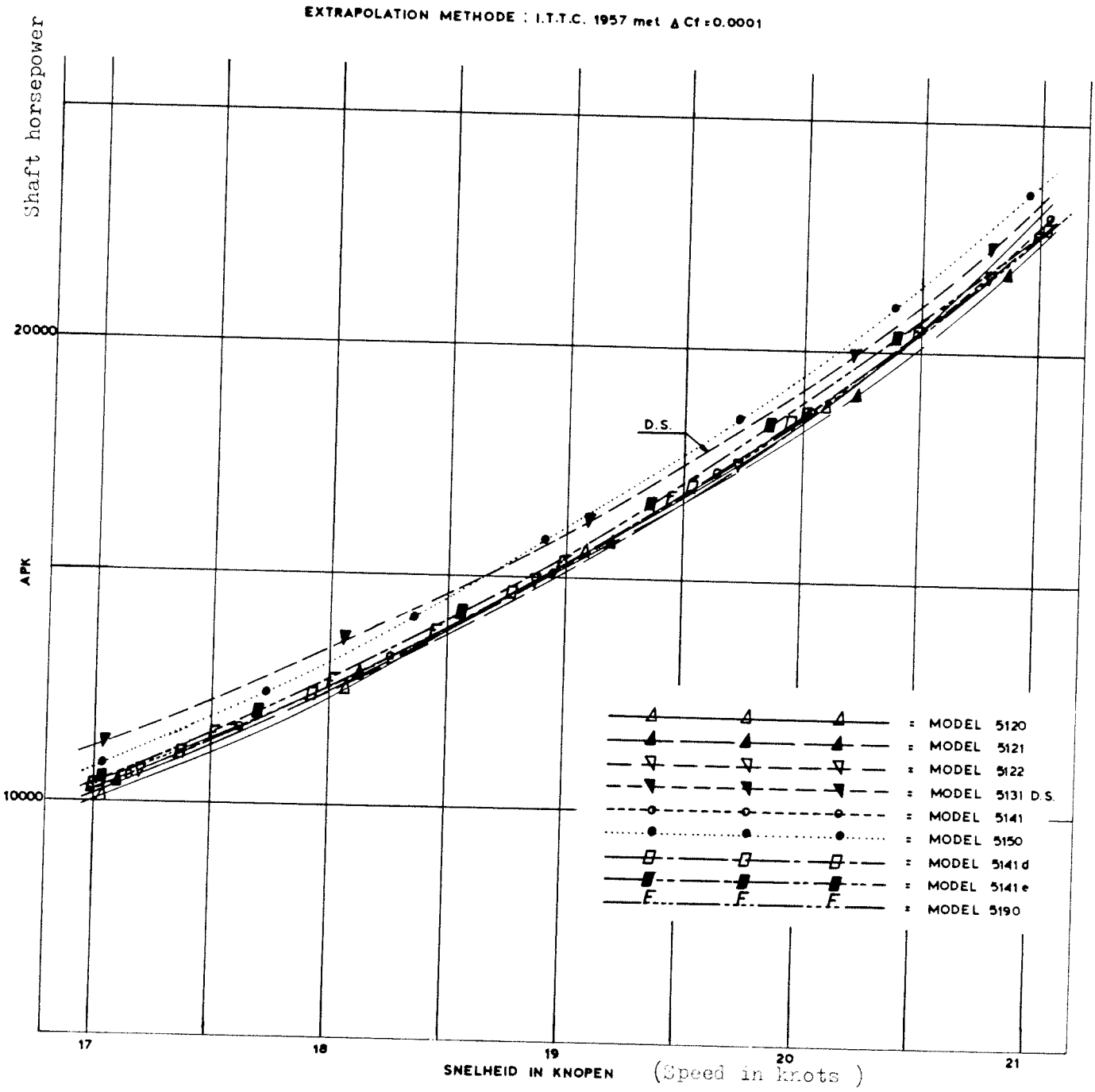


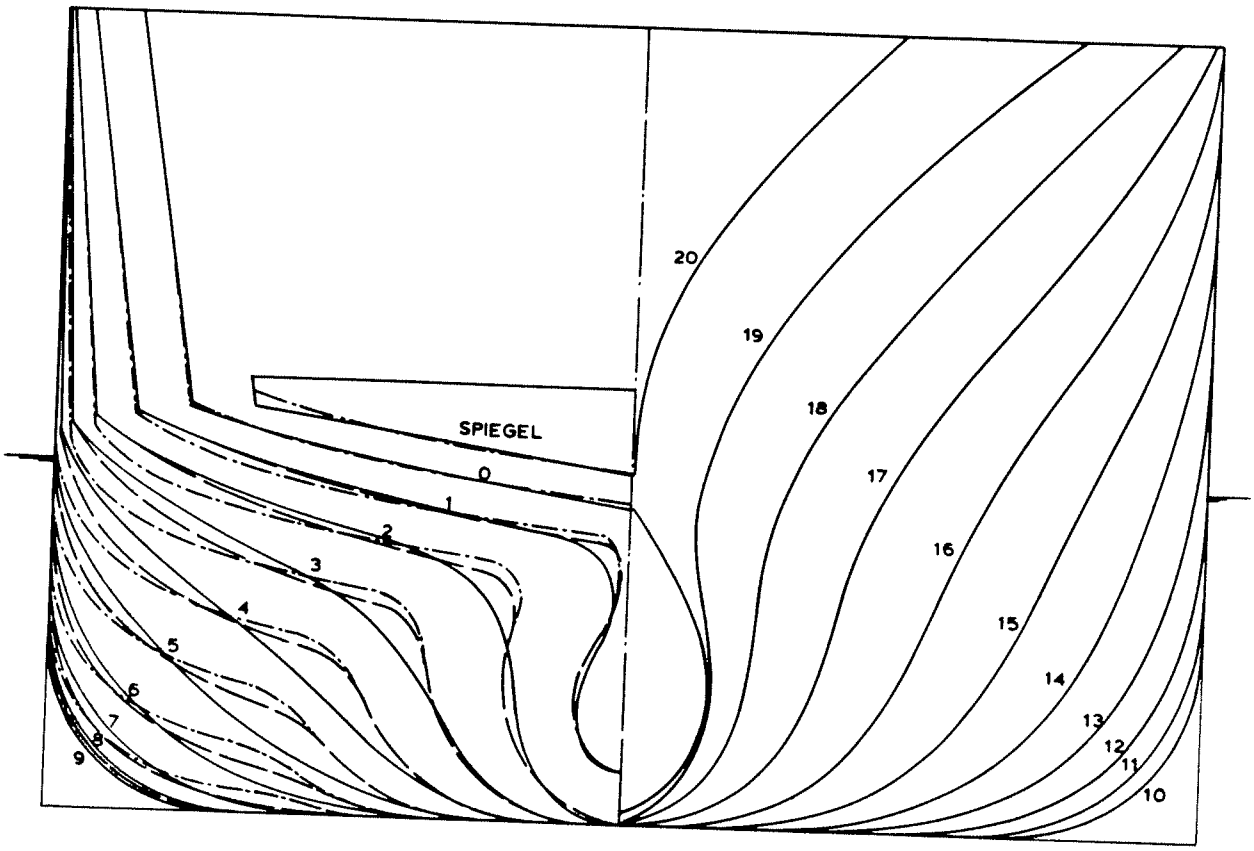
KROMMEN VAN SPANTOPPERVLAKKEN (Curves of transverse sectional areas)

Model Nr. MODEL No.	Centre of Buoyancy DRUKKINGSPUNT	Line type LIJNTYPE	Draught DIEPGANG	Block C_B	Forebody (Model Nr.) VOORSCHIP
5120	-1.33 %		10.00 m	0.615	51 20
5121	-1.34 %		10.00 m	.616	20
5122	-1.37 %		10.00 m	.614	20
DS 5131	-3.37 %		9.82 m	.626	51 31
5141	-2.23 %		10.00 m	.610	31
5141 D	-2.23 %		10.00 m	.610	31
5141 E	-2.06 %		10.00 m	.607	31
5150	-2.63 %		10.00 m	.616	31
5190	-2.15 %		10.00 m	.609	31



APK TANK RESULTATEN





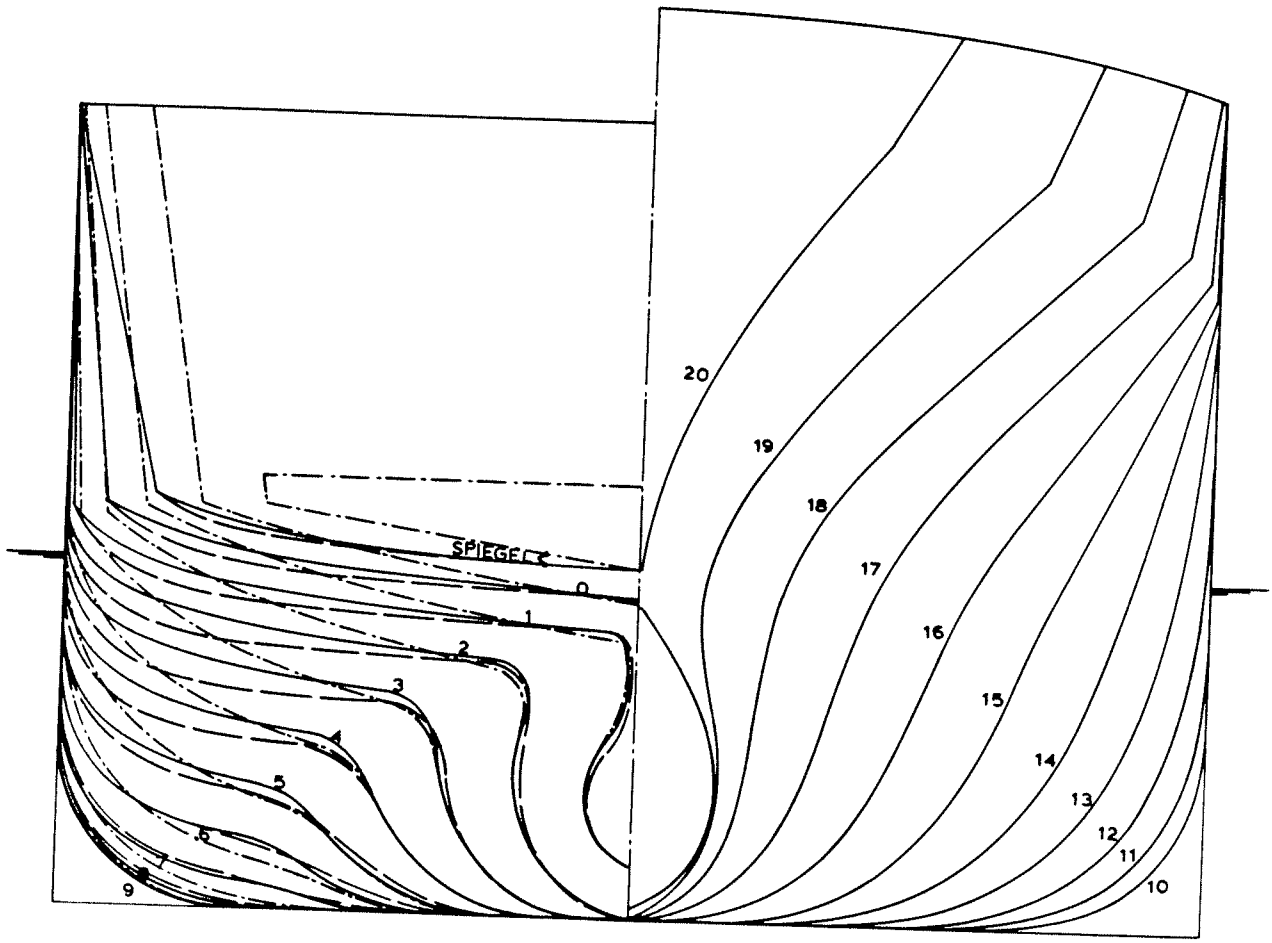
MODEL No. 5120 ———— BASIS VORM , C_B = 0,615 .

MODEL No. 5122 — — — V PRAAM , C_B = 0,616 .

MODEL No. 5121 —·—·— PLATTE PRAAM . C_B = 0,614 .

} VOL VOORSCHIP

L.C.B. = ~ – 1,4 %

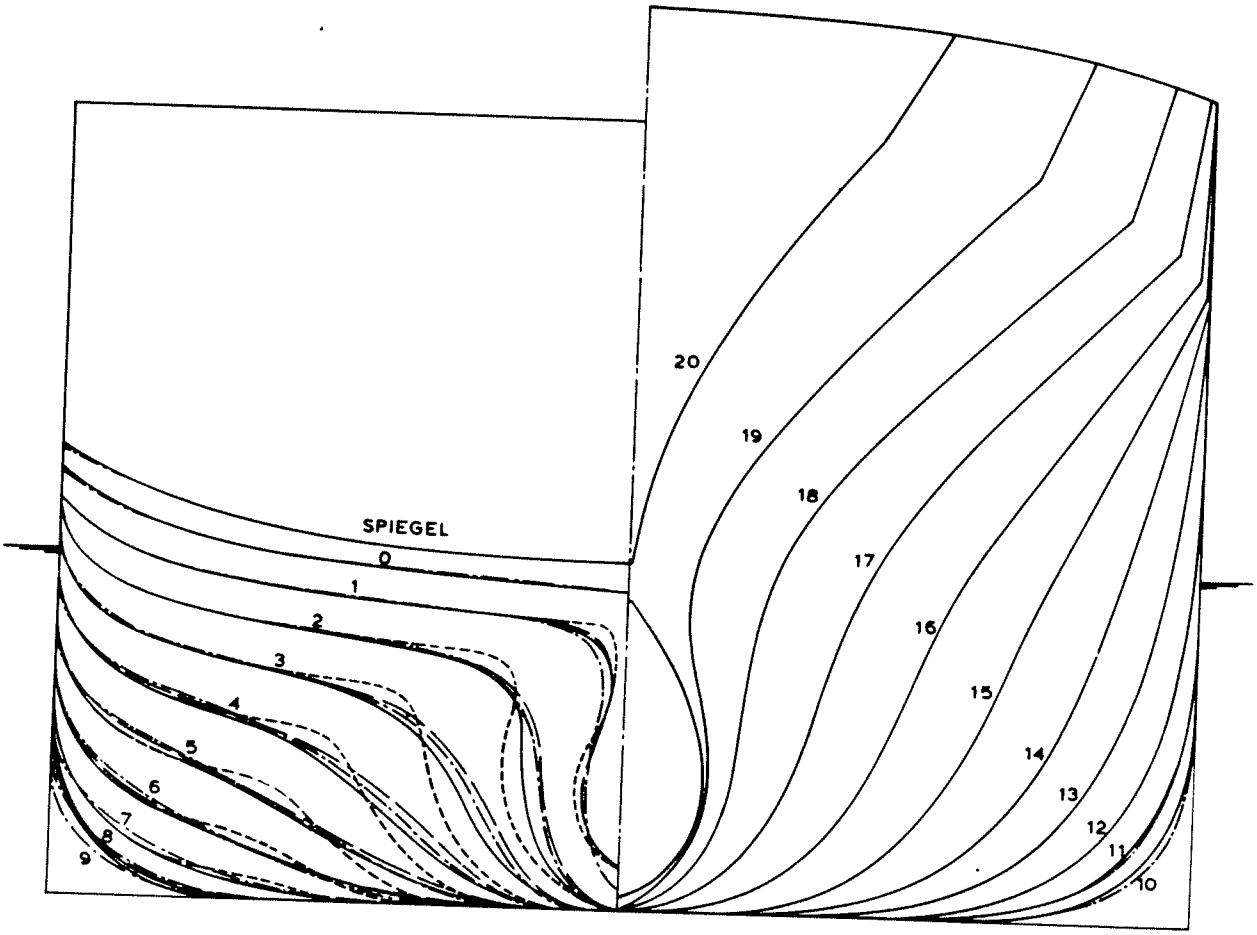


MODEL No. 5141 ——— VOLLE VLAKKE PRAAM, L.C.B. = −2,23% , C_B = 0,610

MODEL No. 5150 — — GESTREKTE VOLLE PRAAM, L.C.B. = −2,63% , C_B = 0,617

MODEL No. 5122 —·— V PRAAM , L.C.B. = −1,85% , C_B = 0,604

SLANK VOORSCHIP



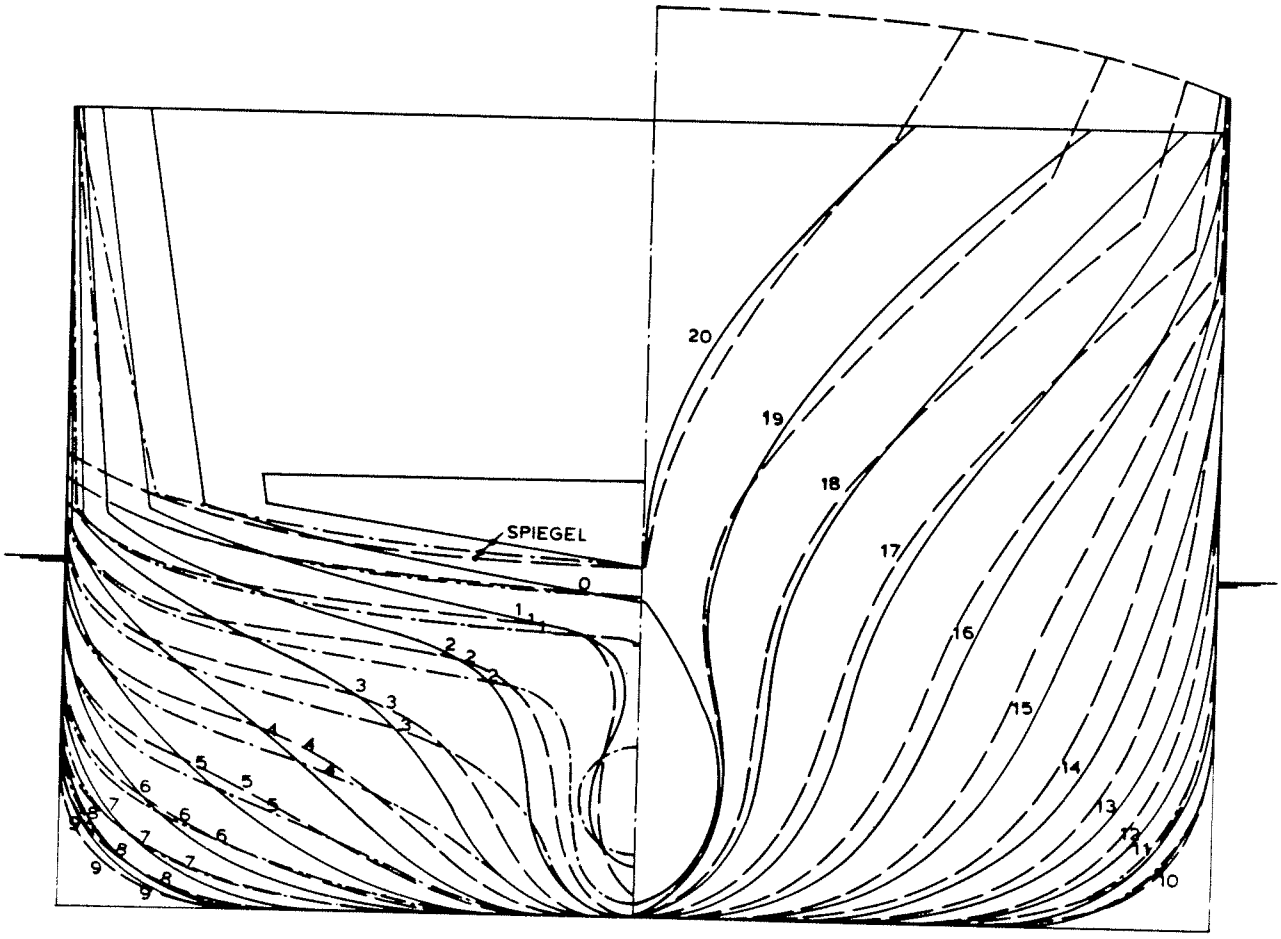
MODEL No. 5141 D ——— , L.C.B. = −2,23% , C_B = 0,610 .

MODEL No. 5141 E — — , L.C.B. = −2,06% , C_B = 0,607 .

MODEL No. 5190 —·— , L.C.B. = −2,14 % , C_B = 0.609 .

MODEL No 5141 --------- , L.C.B. = −2.23 % , C_B 0.610

5131 - SLANK VOORSCHIP

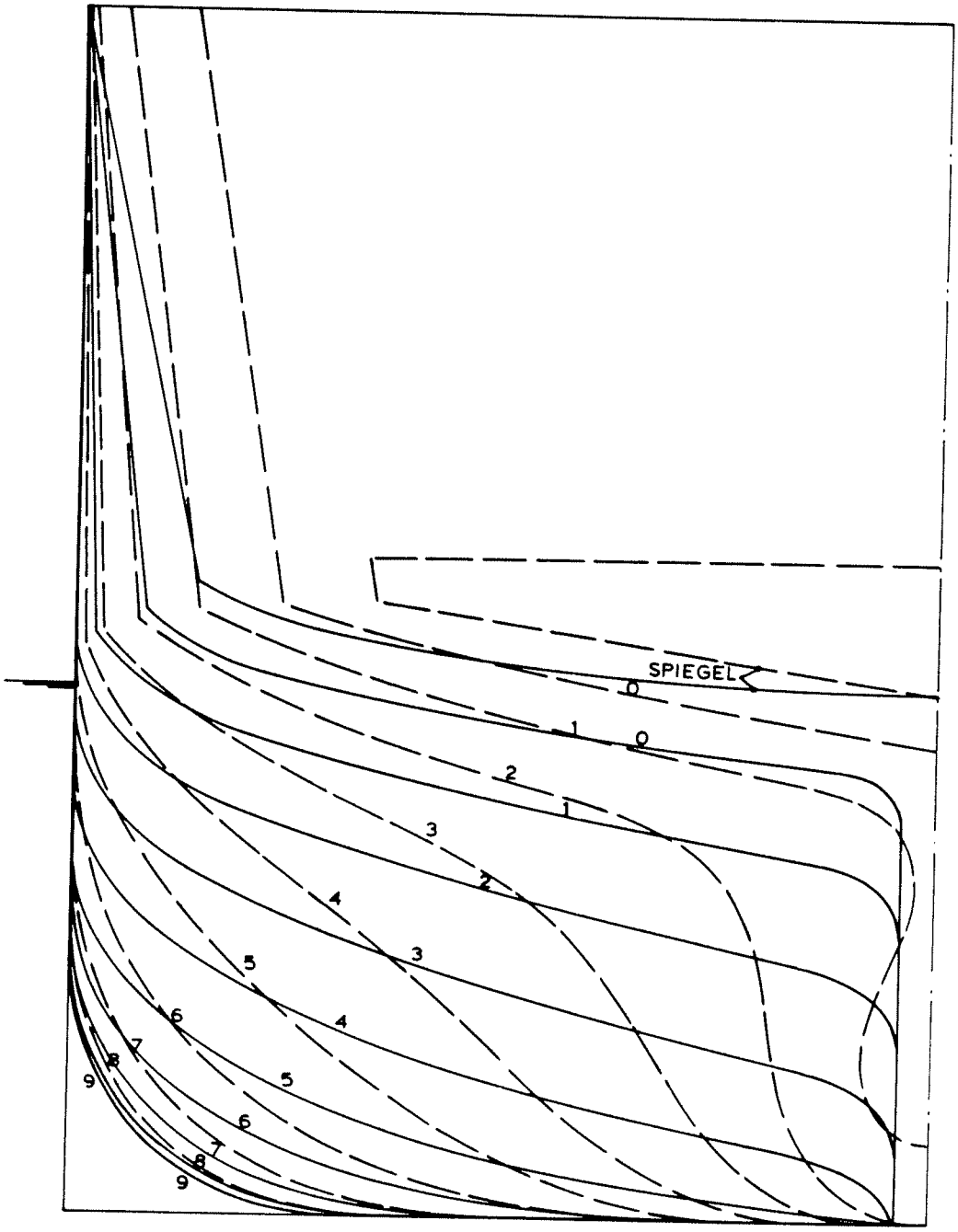


MODEL No. 5120 ———, BASISVORM. VOL VOORSCHIP

MODEL No. 5190 — —, DEFINITIEVE VORM

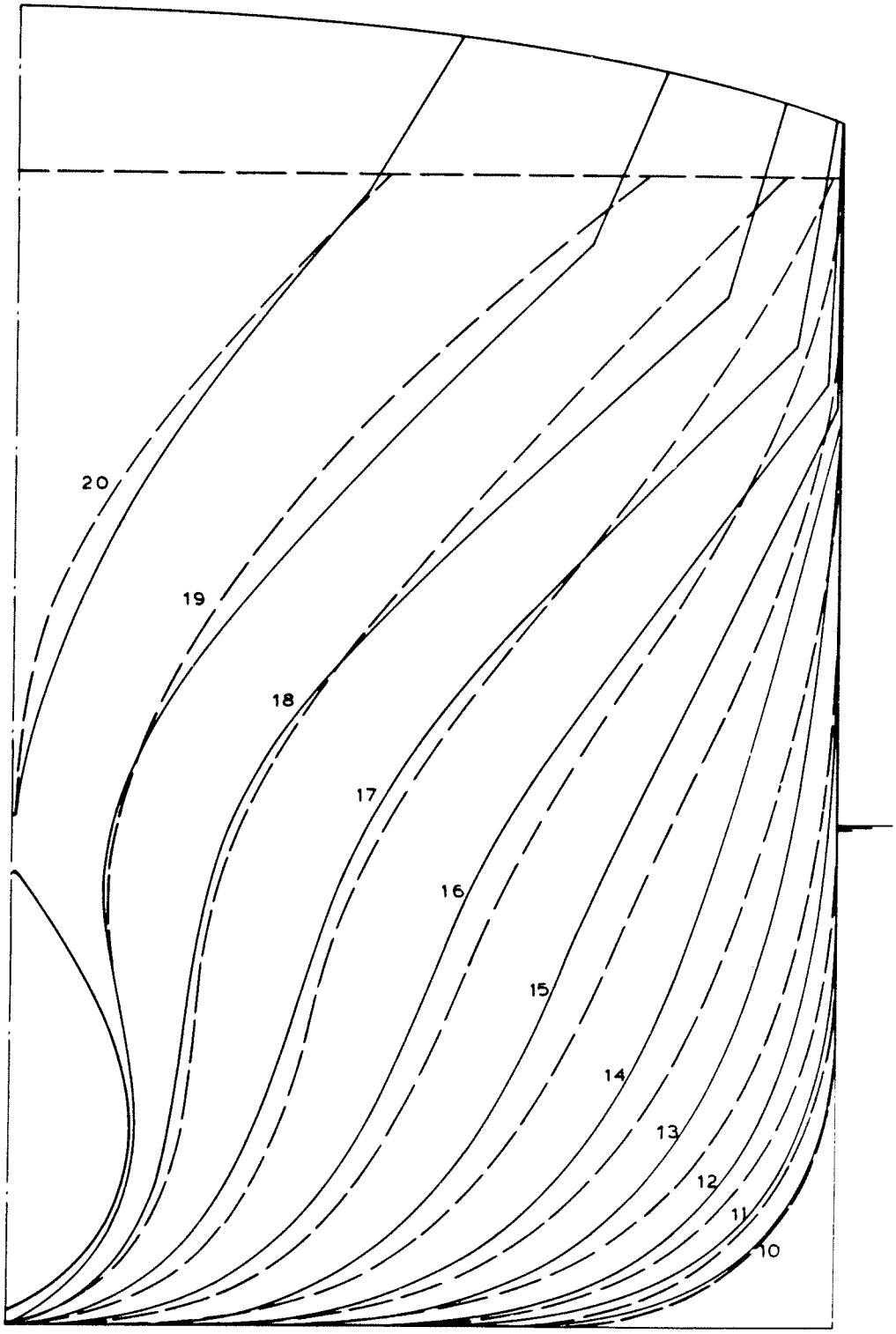
"GOOSENECK" —·—, VOORSTEL (proposal)

} SLANK VOORSCHIP



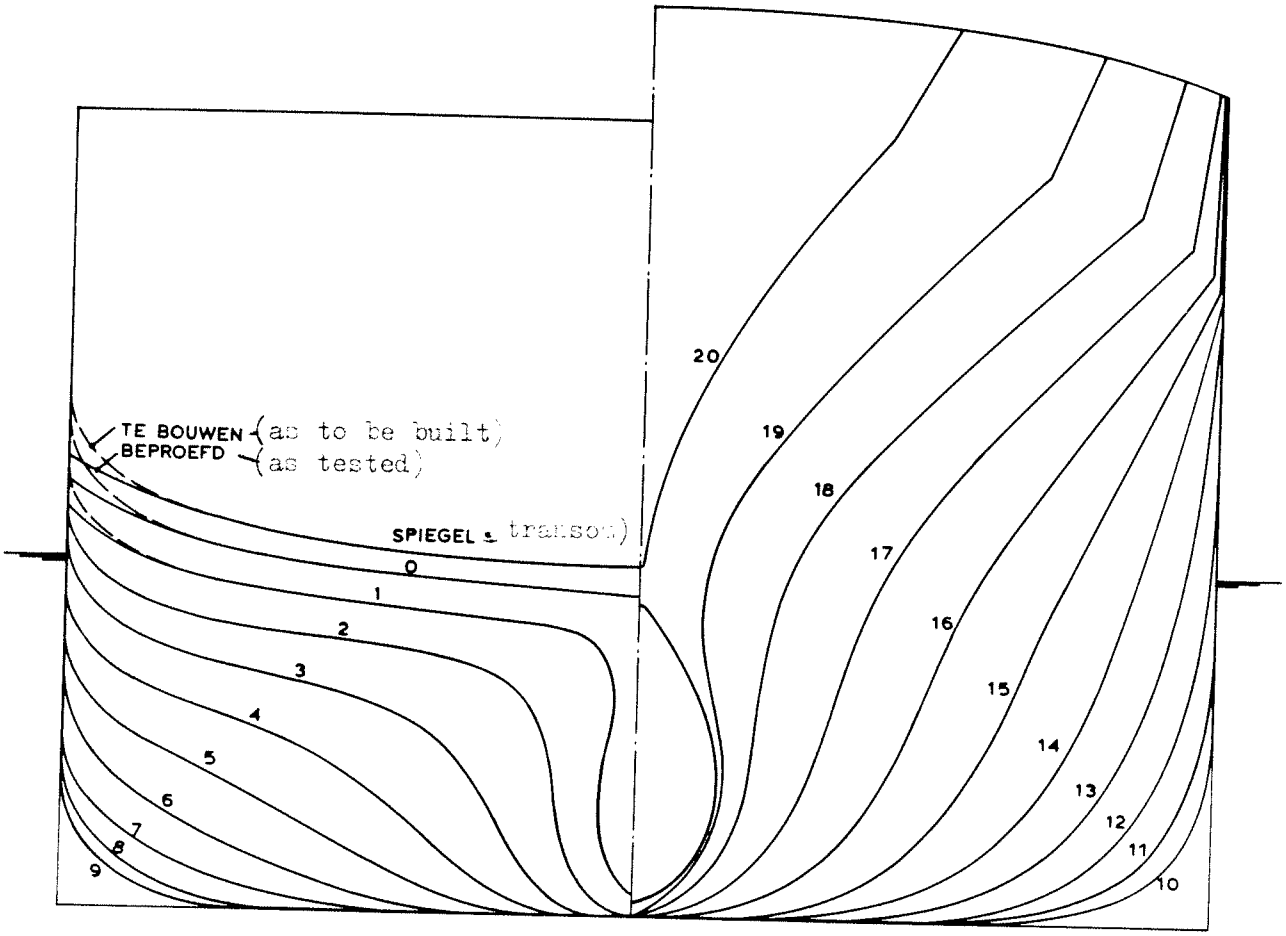
MODEL No. 5131 ———, D.S. (Twinscrew)

MODEL No. 5120 — —, E.S. (BASISVORM).
(Single-screw)

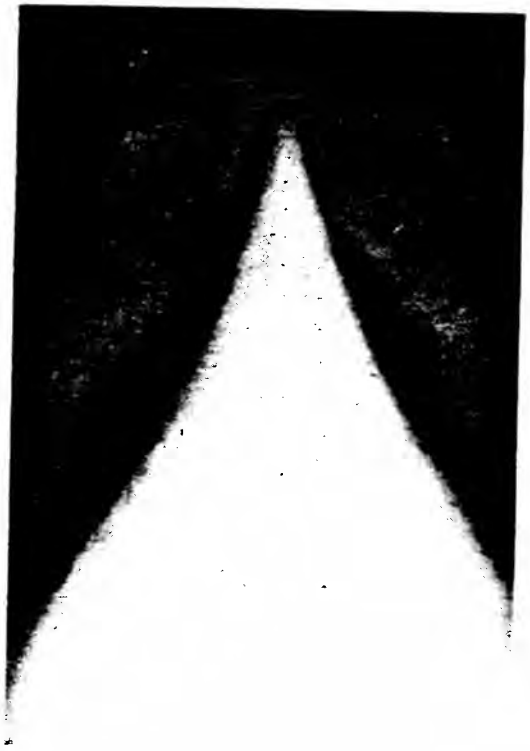


MODEL No. 5131 ———, SLANK VOORSCHIP. (Slender forebo

MODEL No. 5120 — —, VOL VOORSCHIP. (Full forebody)



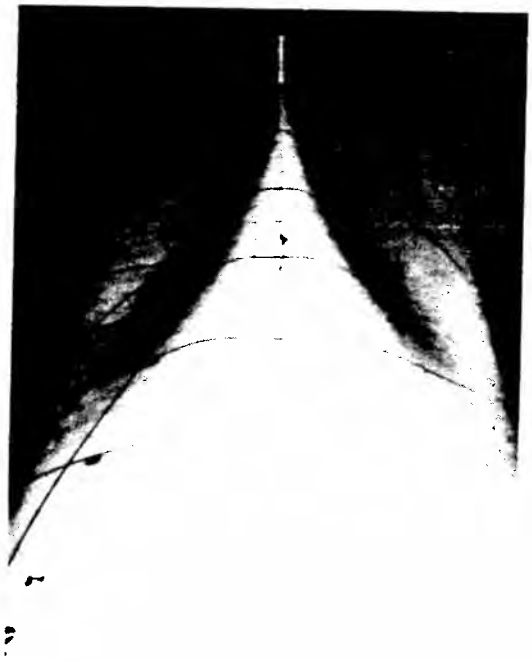
MODEL No. 5190 — DEFINITIEVE VORM .
(Ultimate shape)



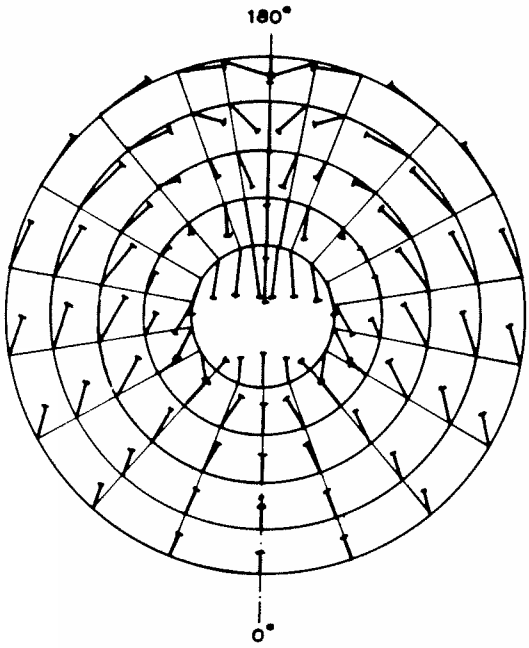
MODEL 5120
BASIS VORM
(Basic shape)

MODEL 5150
GONDEL
(Gondola chiselled out)

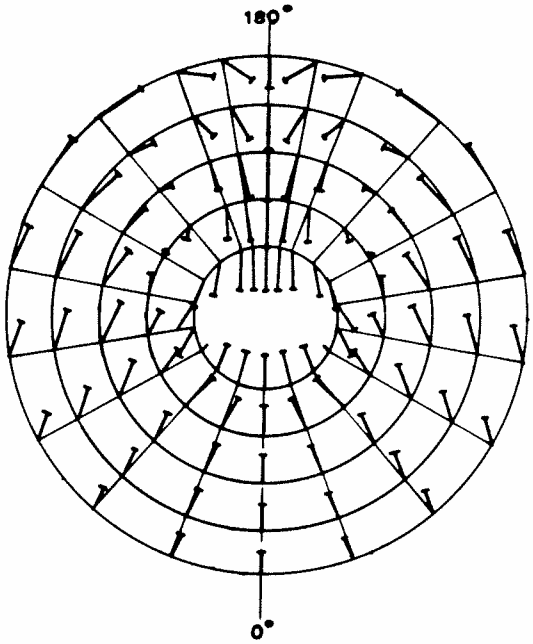
MODEL 5131
DUBBEL SCHROEVER
(Twin-screw shape)



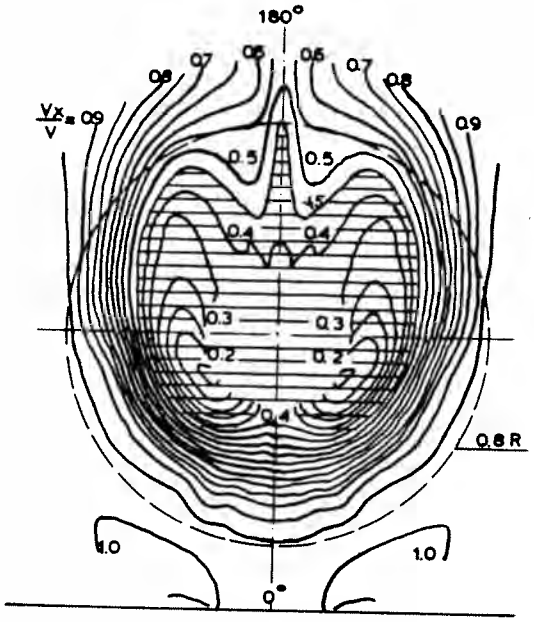
MODEL 5190
STERK VERSLANKTE GONDEL
OM LOSLATING TE VOORKOMEN
(Starkly slendered gondola
for preventing separation of flow)



VECTOR DIAGRAM DWARSCOMPONENTEN

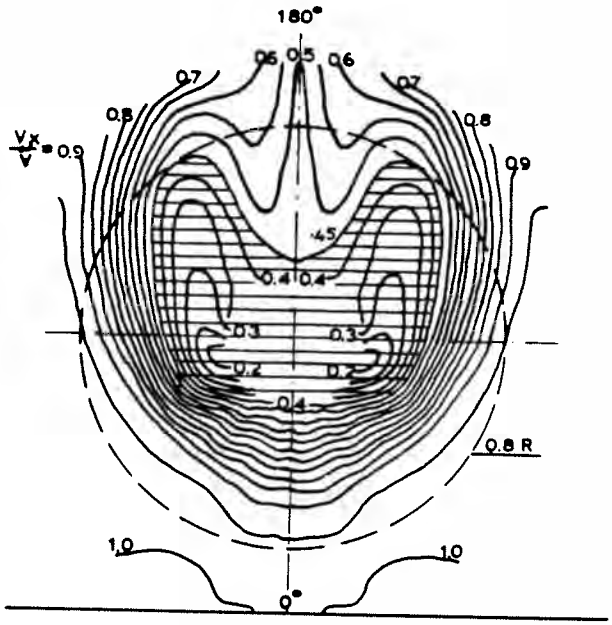


VECTOR DIAGRAM DWARSCOMPONENTEN



LUNEN VAN CONSTANTE SNELHEID

MODEL 5120

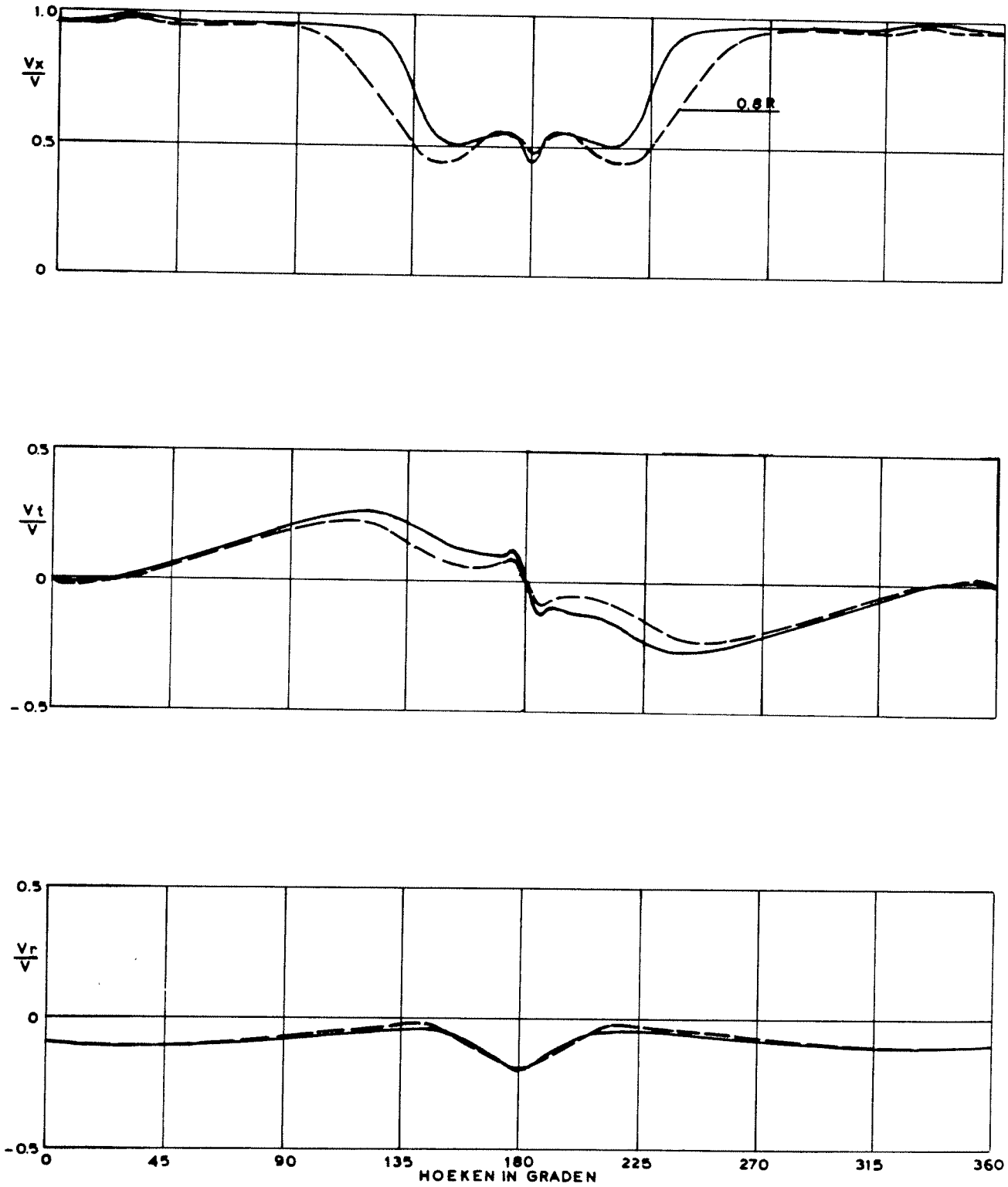


LUNEN VAN CONSTANTE SNELHEID

MODEL 5141

RESULTATEN 3 DIMENSIONALE PITOTBUISMETINGEN

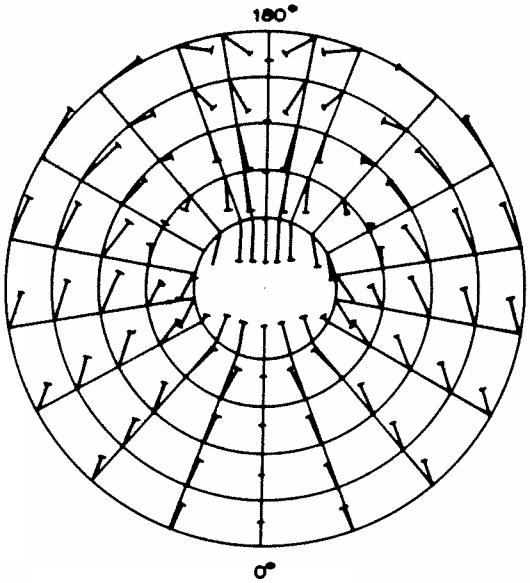
USA-ARABISCHE GOLF
RORO-TRAILERSCHIP



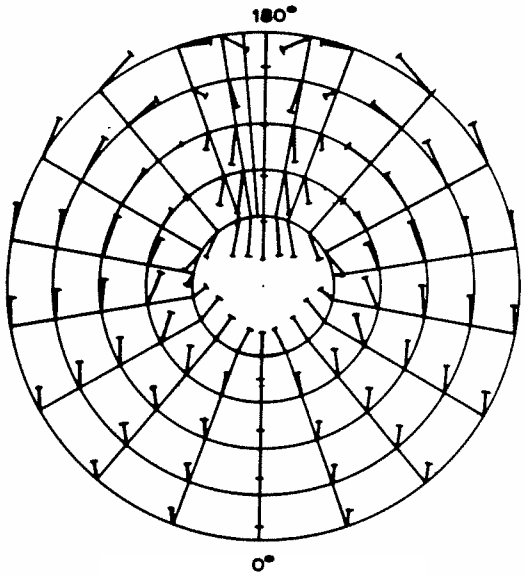
PERIFERIALE VERDELING VAN DE DRIE SNELHEIDSCOMPONENTEN

MODEL 5120 = ———— MODEL 5141 = — — —

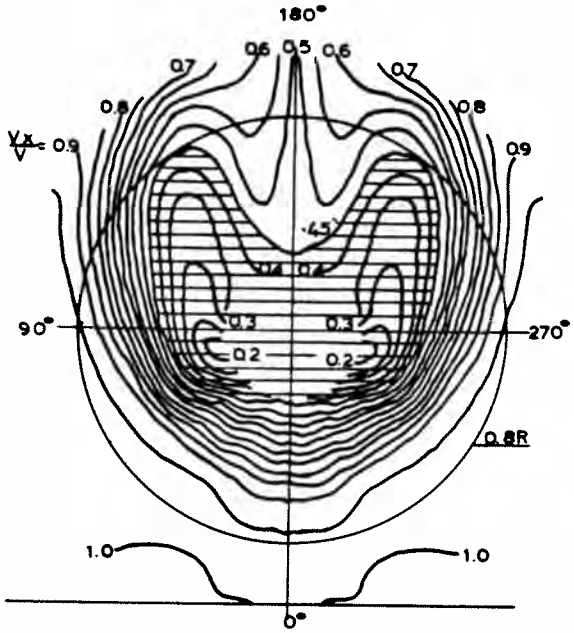
U.S.A.-ARABISCHE GOLF RO-RO TRAILERSCHIP.



VECTOR DIAGRAM DWARSCOMPONENTEN

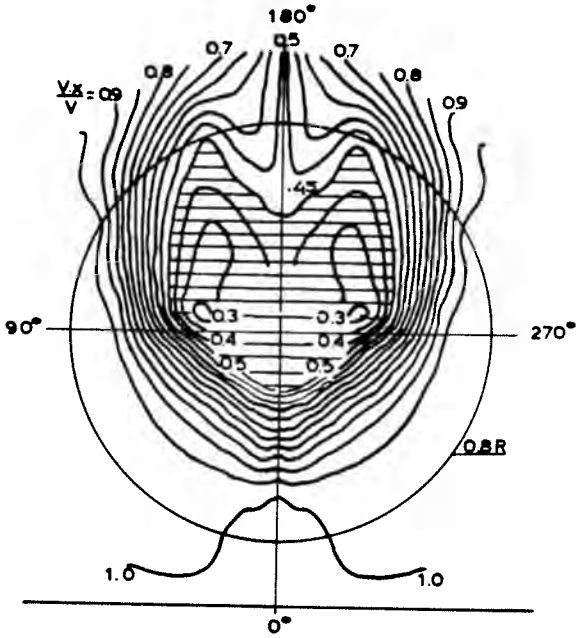


VECTOR DIAGRAM DWARSCOMPONENTEN



LIJNEN VAN CONSTANTE SNELHEID

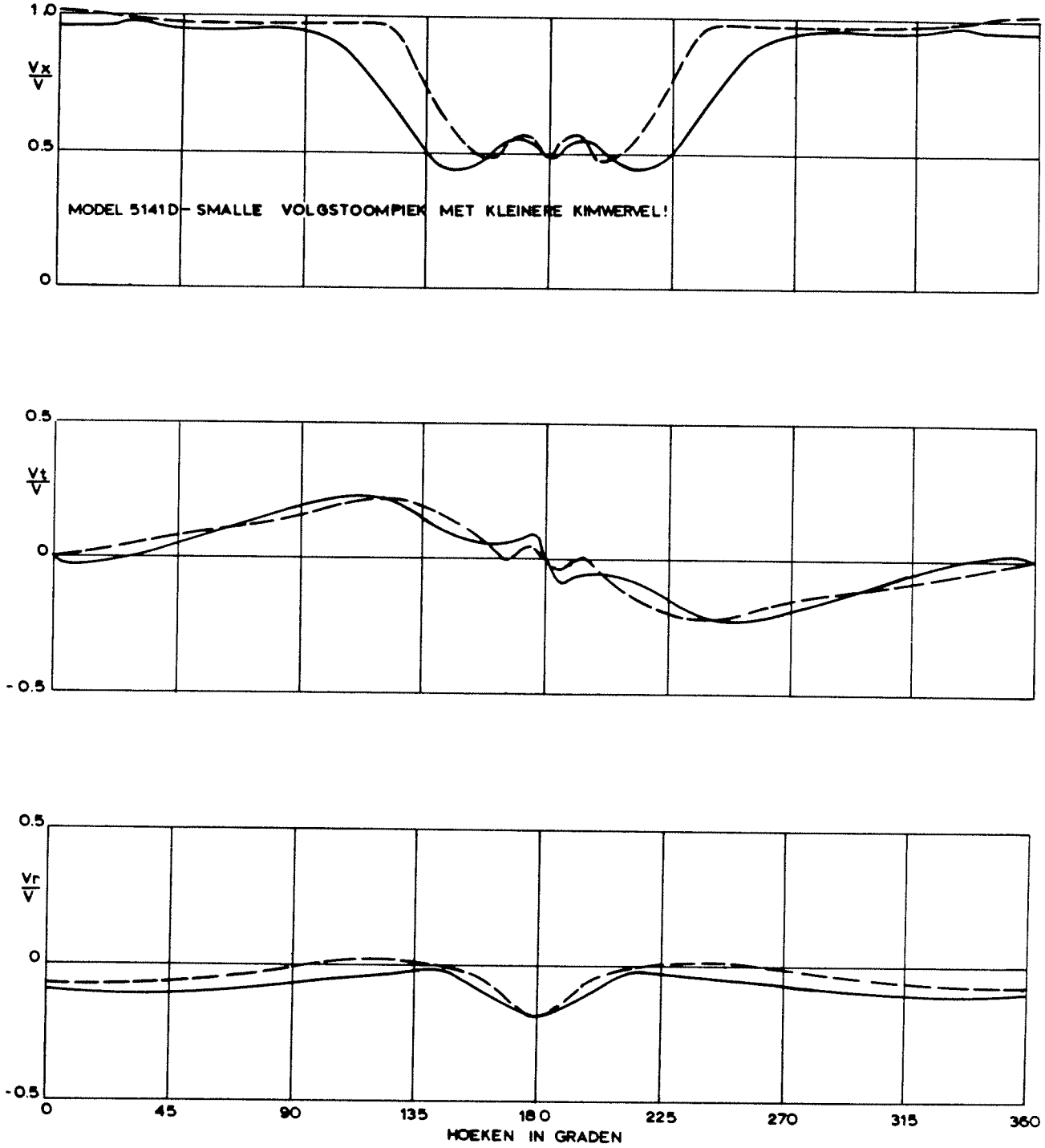
MODEL 5141



LIJNEN VAN CONSTANTE SNELHEID

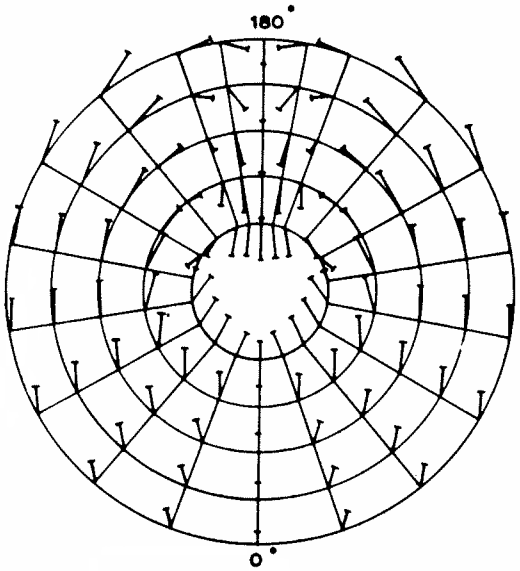
MODEL 5141D

RESULTATEN 3 DIMENSIONALE PITOTBUISMETINGEN

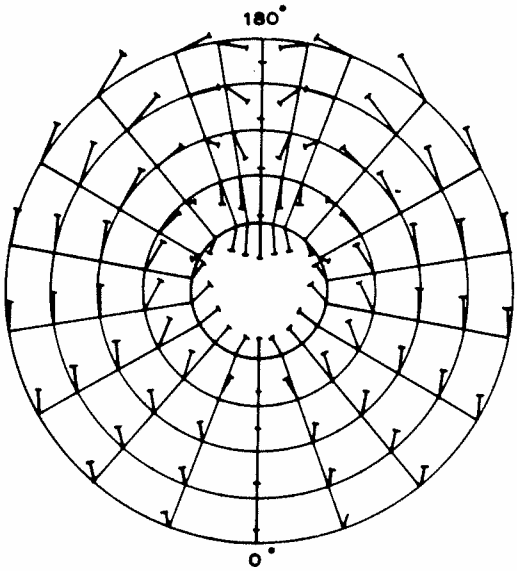


PERIFERIALE VERDELING VAN DE DRIE SNELHEIDCOMPONENTEN.
MODEL 5141 _____ MODEL 5141 D _ _ _ _

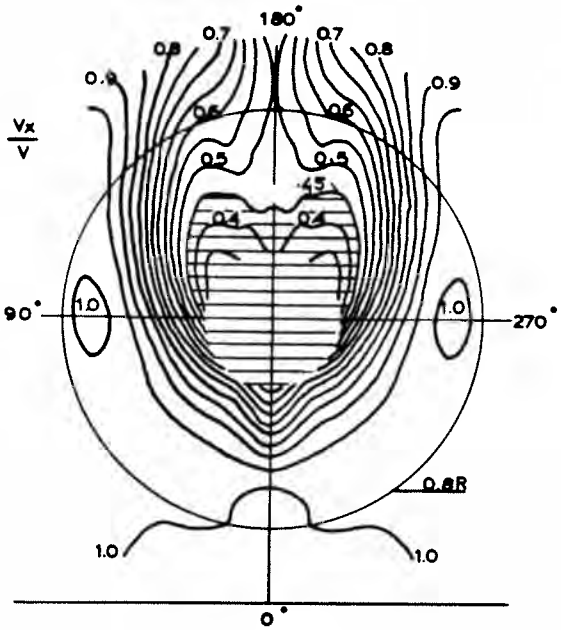
U.S.A - ARABISCHE GOLF RO-RO TRAILERSCHIP



VECTOR DIAGRAM DWARSCOMPONENTEN

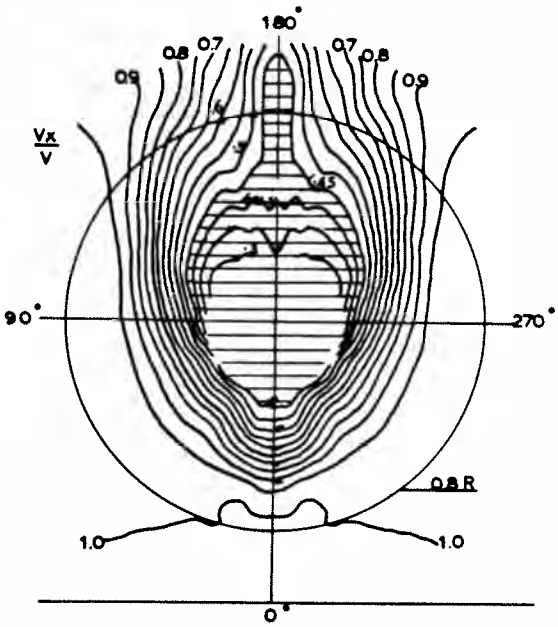


VECTOR DIAGRAM DWARSCOMPONENTEN



LIJNEN VAN CONSTANTE SNELHEID

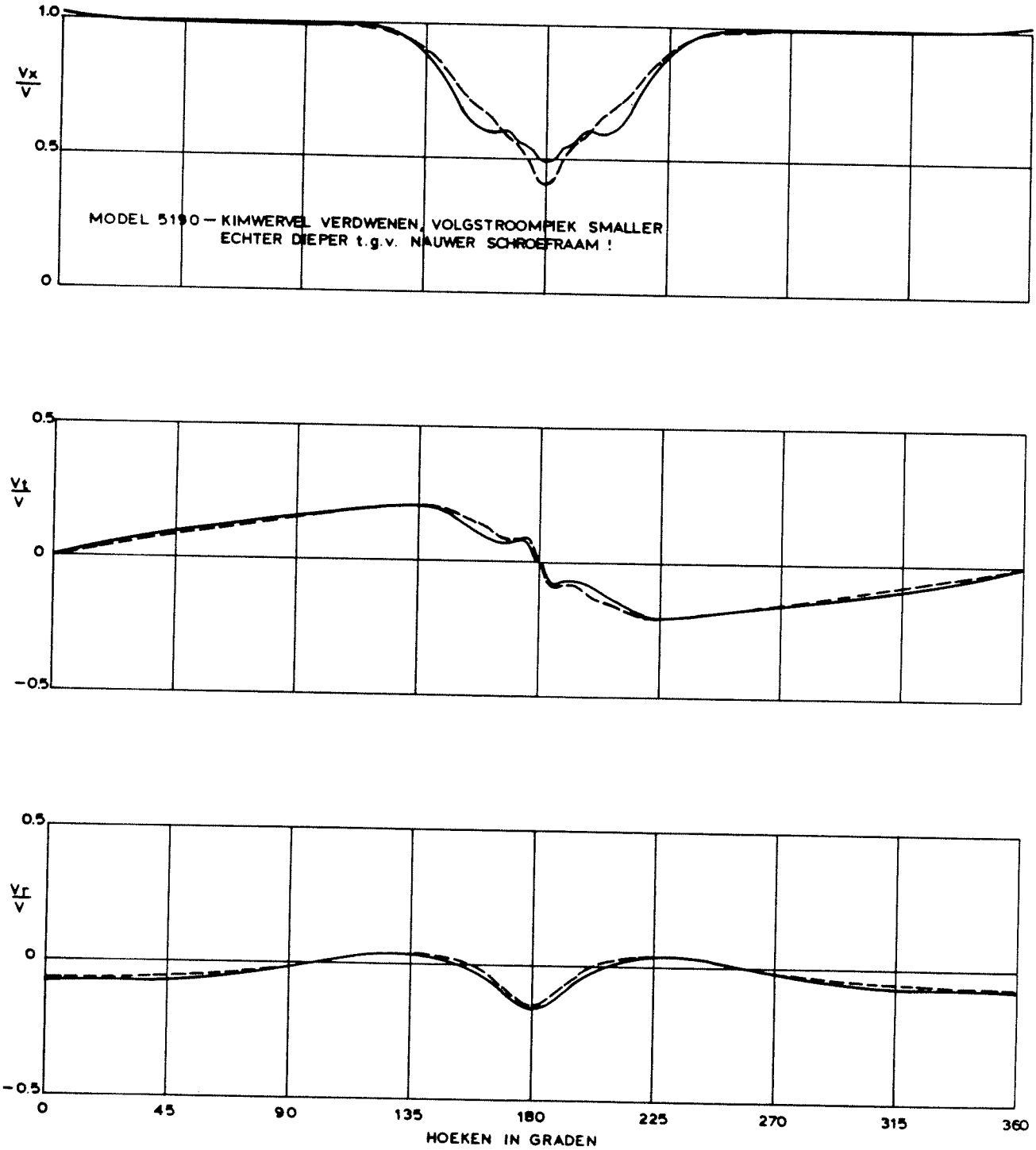
MODEL 5141E



LIJNEN VAN CONSTANTE SNELHEID

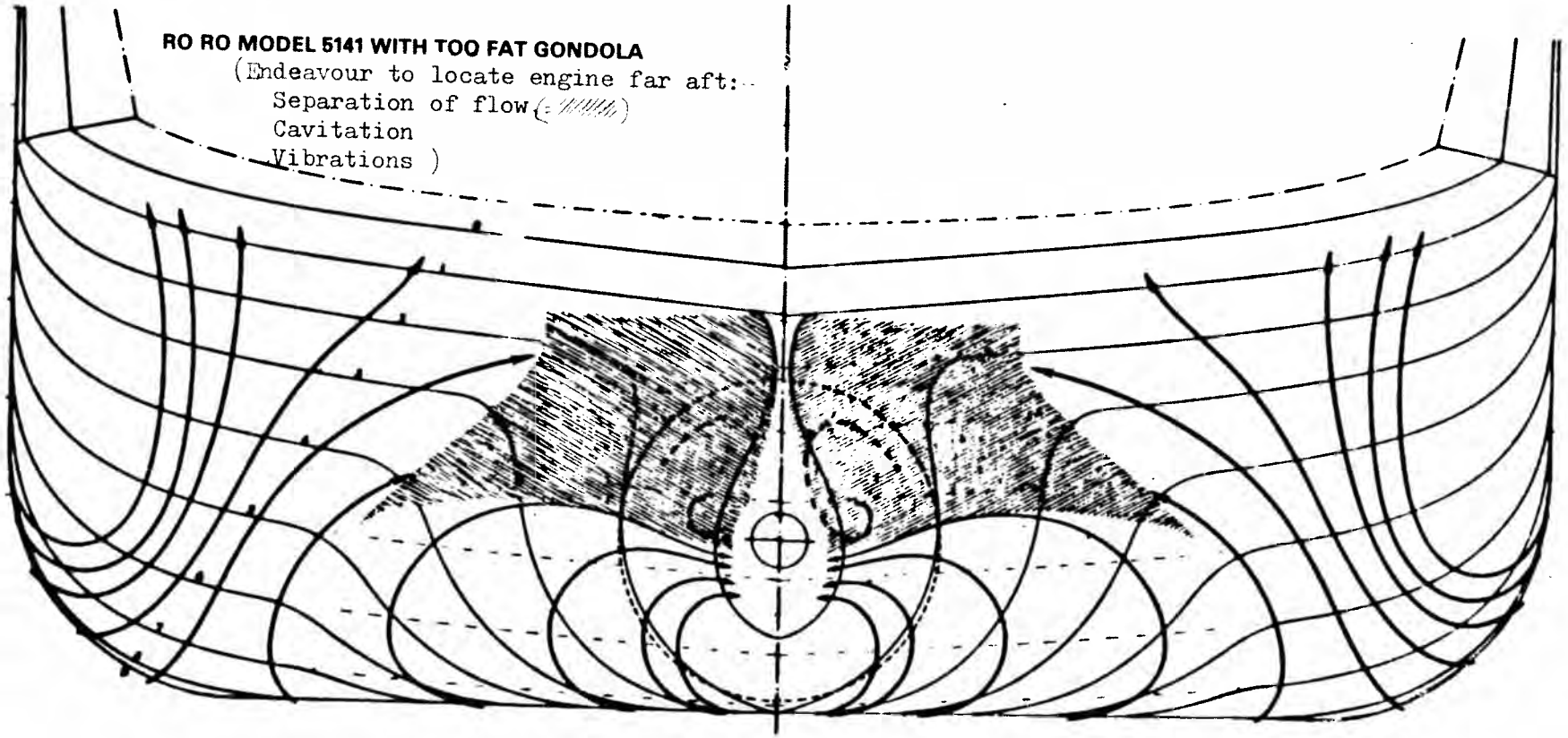
MODEL 5190

RESULTATEN 3 DIMENSIONALE PITOTBUISMETINGEN

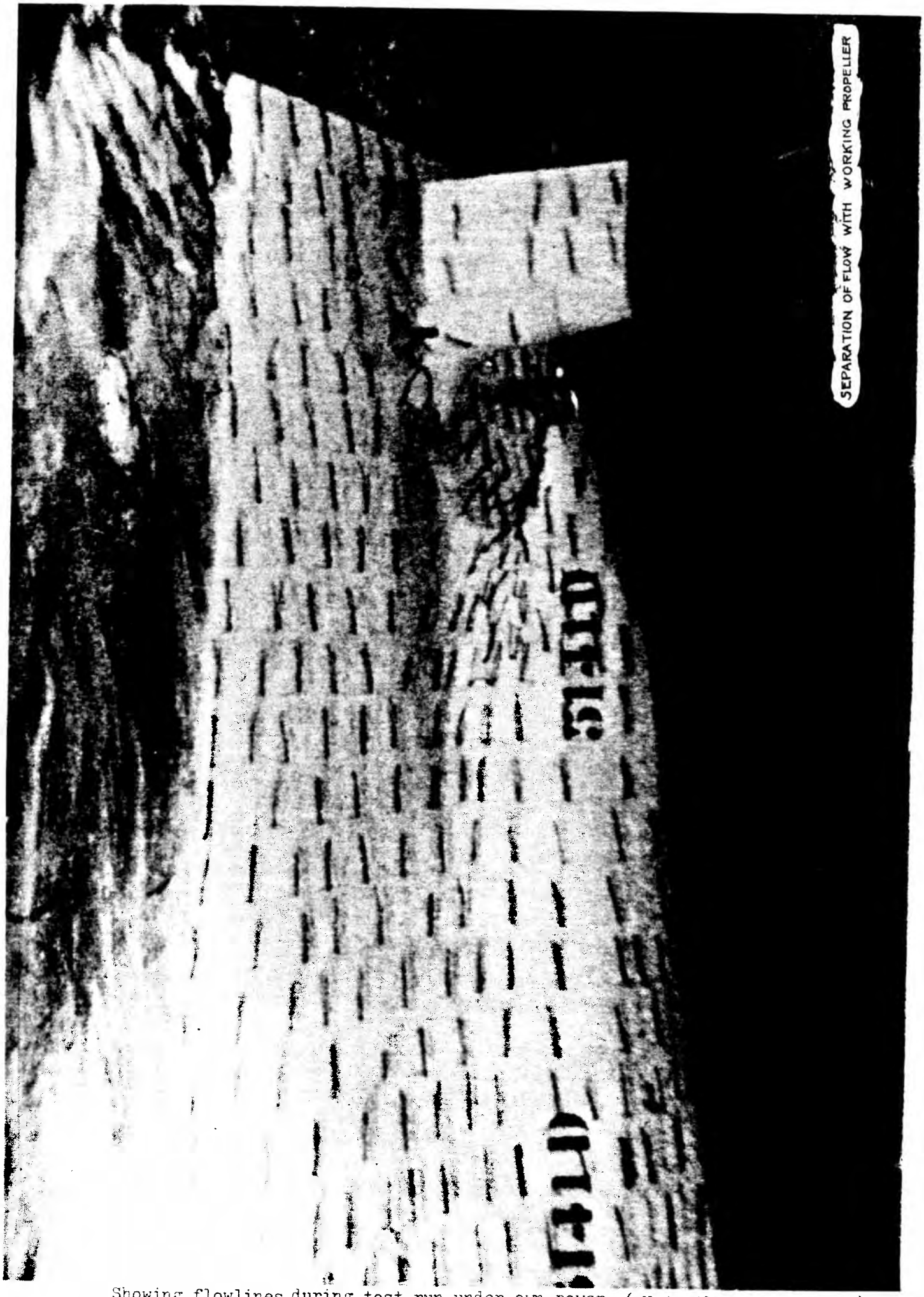


PERIFERIALE VERDELING VAN DE DRIE SNELHEIDCOMPONENTEN.

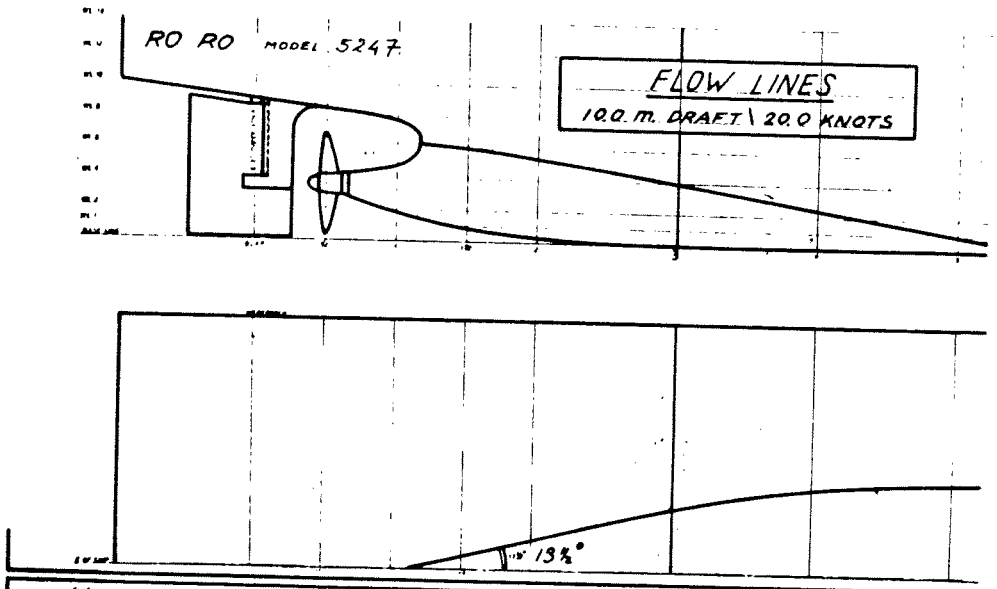
MODEL 5141 E _____ MODEL 5190 _ _ _ _

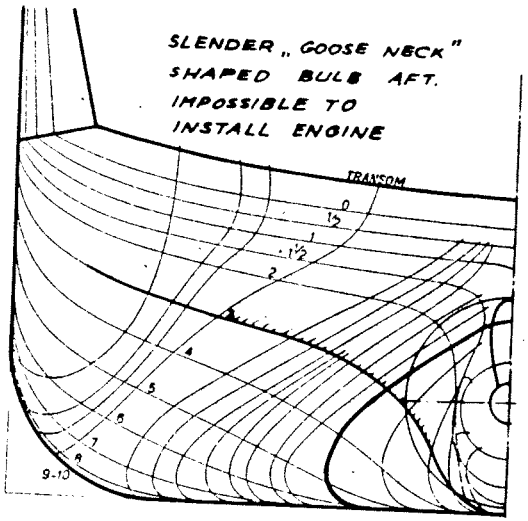

RO RO MODEL 5141 WITH TOO FAT GONDOLA
(Endeavour to locate engine far aft:
Separation of flow
Cavitation
Vibrations)

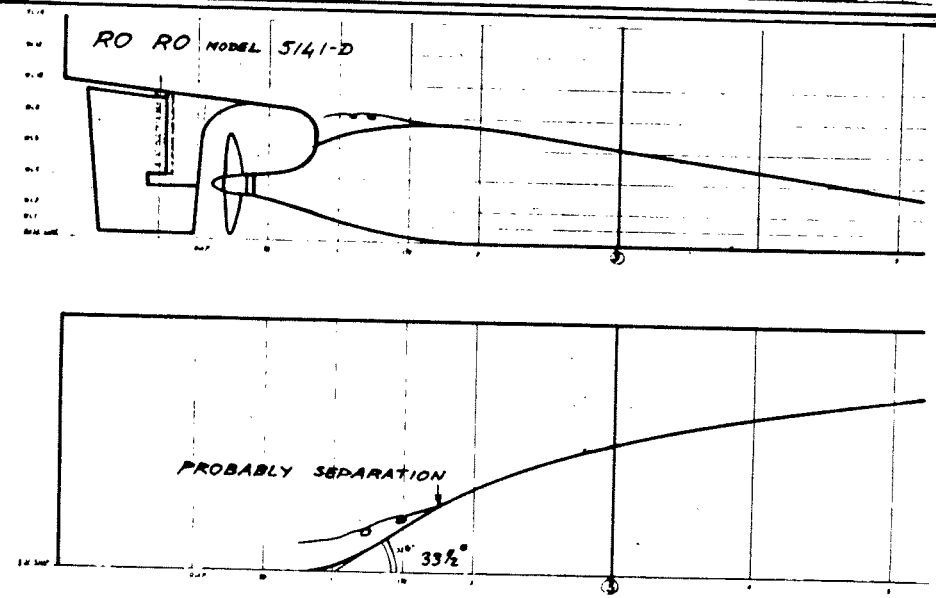
Separation of flow - as towed without propeller.

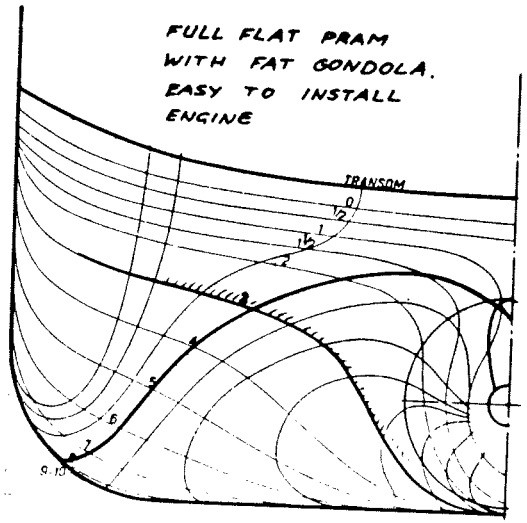


Showing flowlines during test run under own power (Note the propeller !)

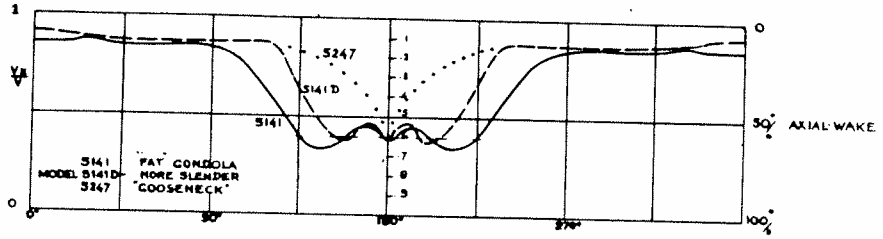

RO RO MODEL 5247
FLOW LINES
10.0 m. DRAFT \ 20.0 KNOTS
13½°

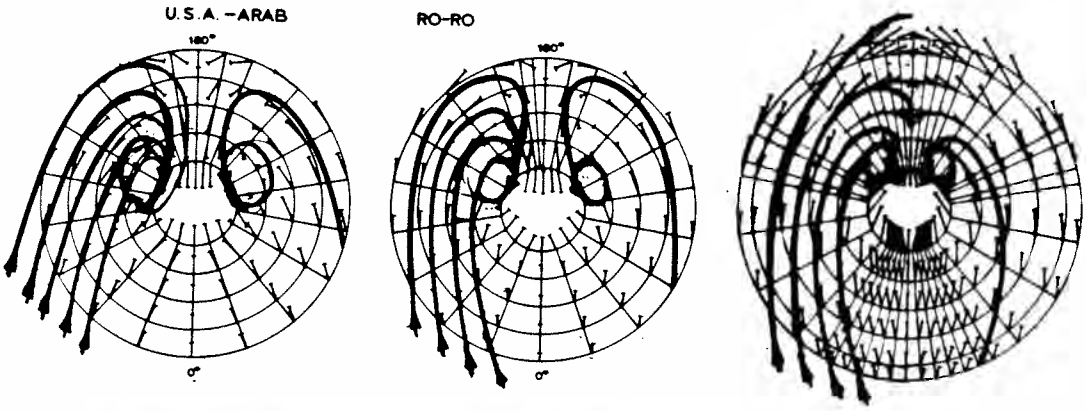

SLENDER „GOOSE NECK"
SHAPED BULB AFT.
IMPOSSIBLE TO
INSTALL ENGINE
TRANSOM

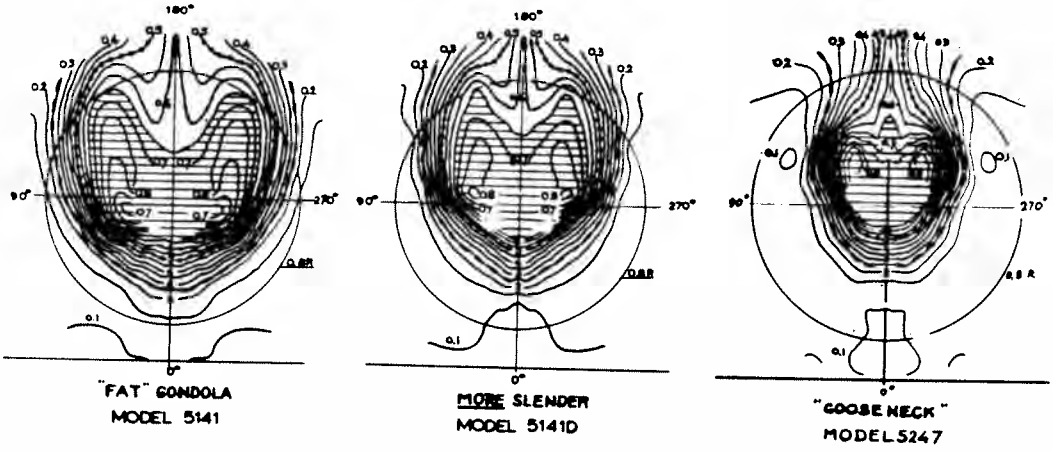

RO RO MODEL 5141-D
PROBABLY SEPARATION
33½°


FULL FLAT PRAM
WITH FAT GONDOLA.
EASY TO INSTALL
ENGINE
TRANSOM

SINGLE SCREW : ENGINE "FAR-AFT" CAUSES VIBRATION



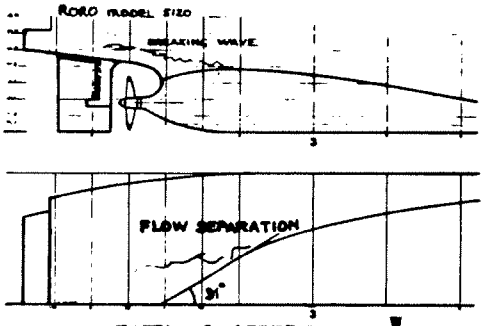


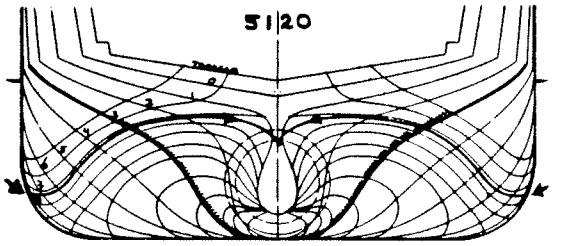


RESULTS OF 3-DIMENSIONAL WAKE-MEASUREMENTS*
* BY MEANS OF PITOT TUBE

Ro Ro RESEARCH 20 KN.

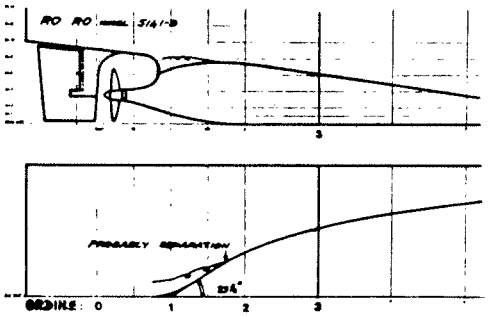
NSMB JONK VD BEEK Vo + Bu NEDILL

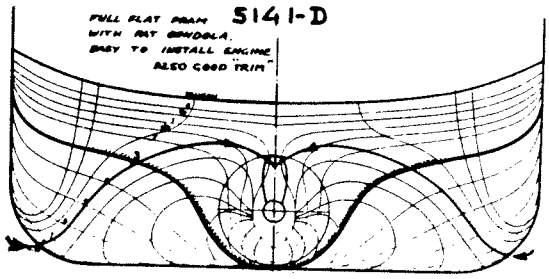


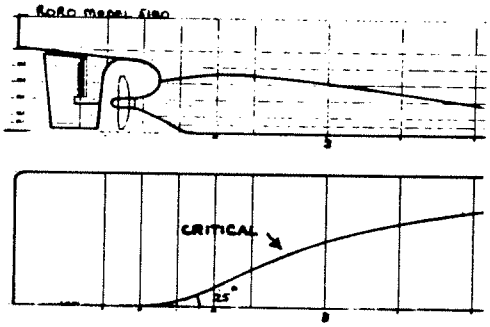


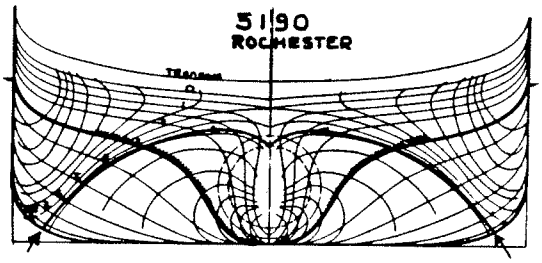
ENGINE FAR AFT

RATTLING AFTER BODIES !

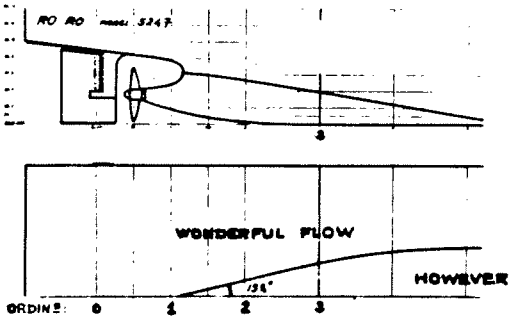


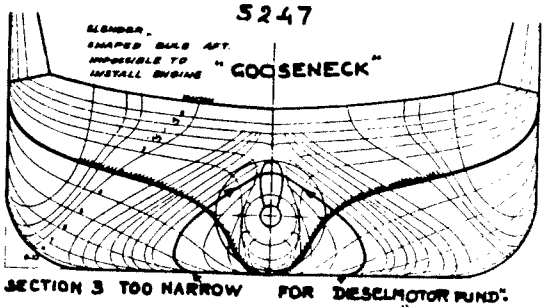






FLOW LINE TOWARDS PROPELLER TIP IN 12 O'CLOCK POSITION IS STRONGLY INFLUENCED BY THE FULLNESS OF THE FOOT OF SECTIONS 3 AND 4, JUST AT AFT-END OF ENGINE!

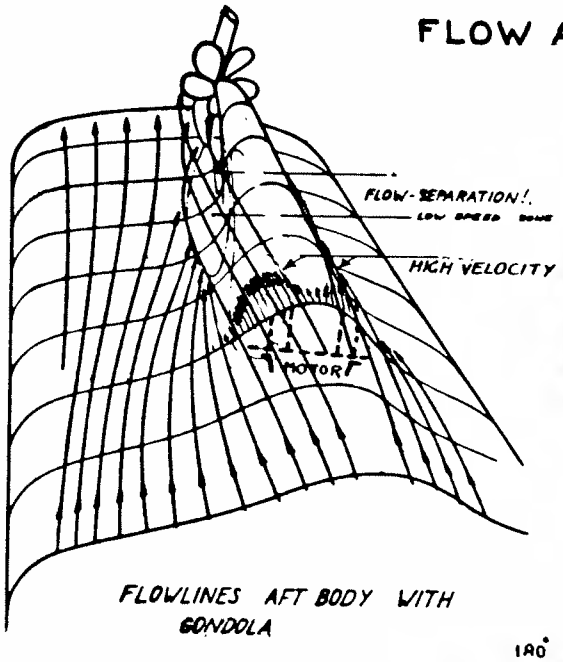


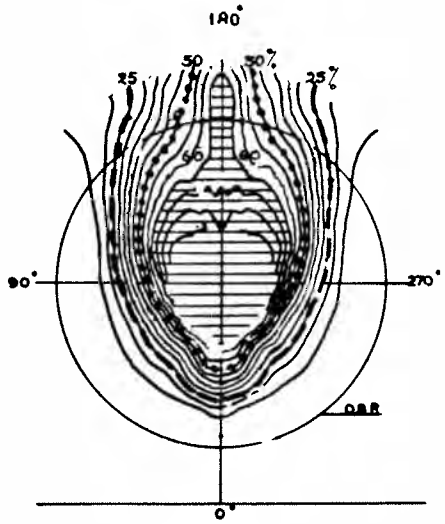


SECTION 3 TOO NARROW FOR DIESELMOTOR FUND.

MOTOR TO MOVE MORE "FORWARD"
...CAUSING LOSS OF CARGO SPACE !

FLOW AROUND ENGINE FOUNDATION





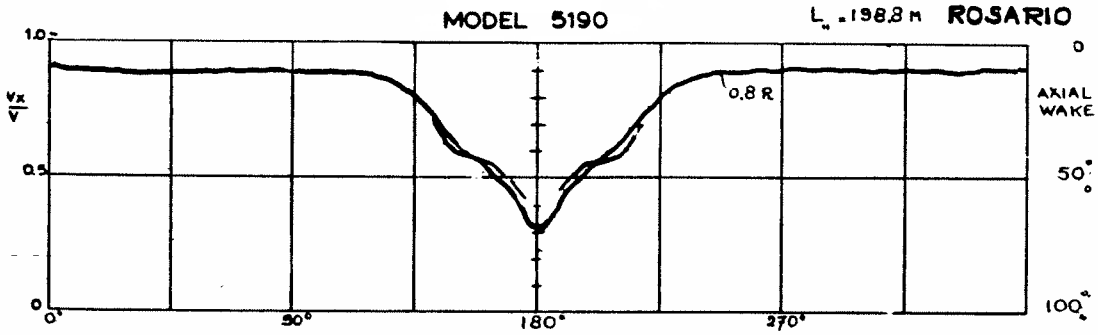
Curves of Constant Velocity

MODEL 5190

1977

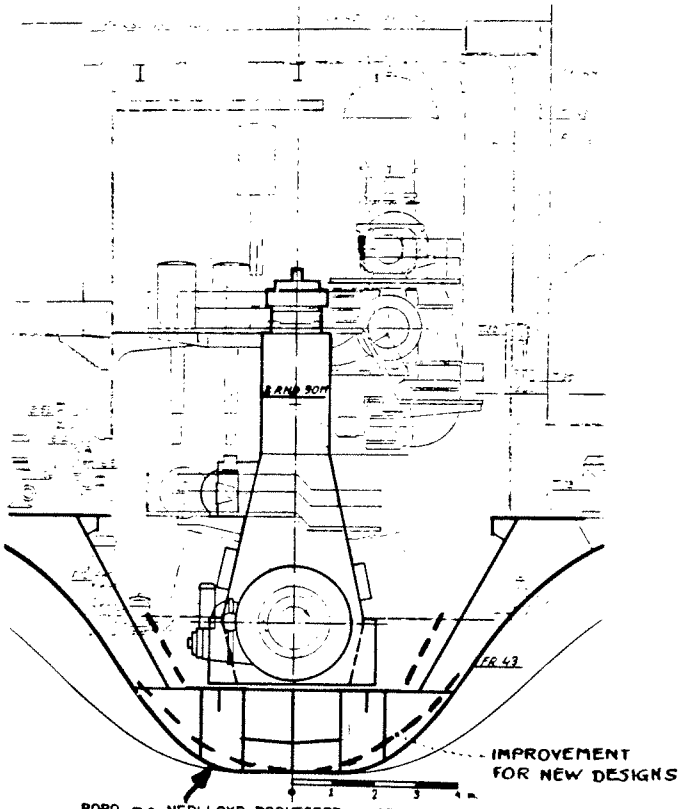
NEDLLOYD "ROTTERDAM"
L_{pp} = 183,2 m ROCHESTER

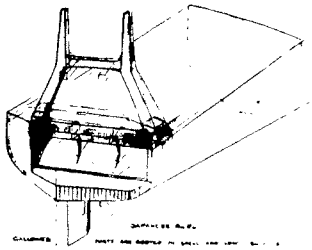
NEDLLOYD ROUEN
L_{pp} = 198,8 m ROSARIO

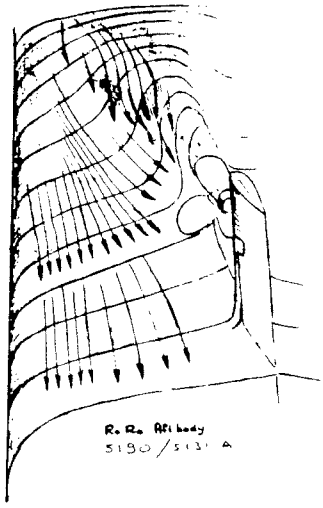


Results of three-dimensional Pitot tube measurements.

LOCATION AND FOUNDATION OF ENGINE

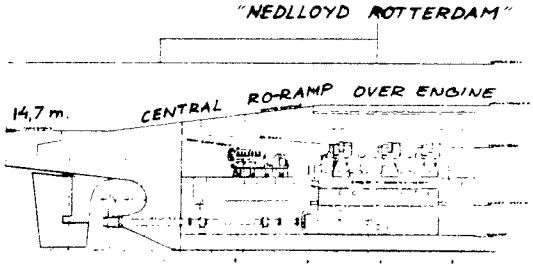


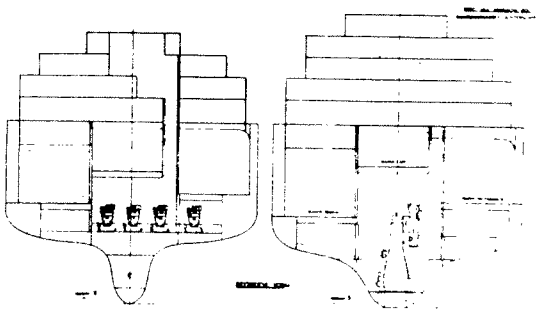




TOO HARD CURVE : FLOW RETARDING

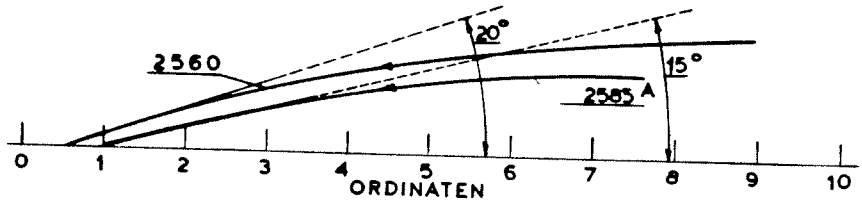
the rudder is 3% of Lpp and the rudder area is 2.8 per cent of Lpp x 10 m.

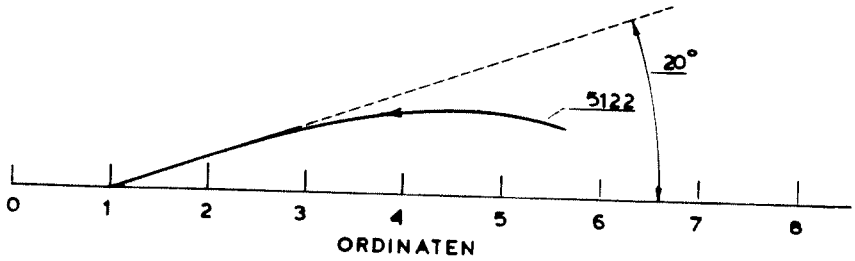




DOORSNEDEN OVER STROOMLIJNEN NAAR SCHROEF OP 0.8 R
(SCHROEFBLAD IN TOPSTAND)

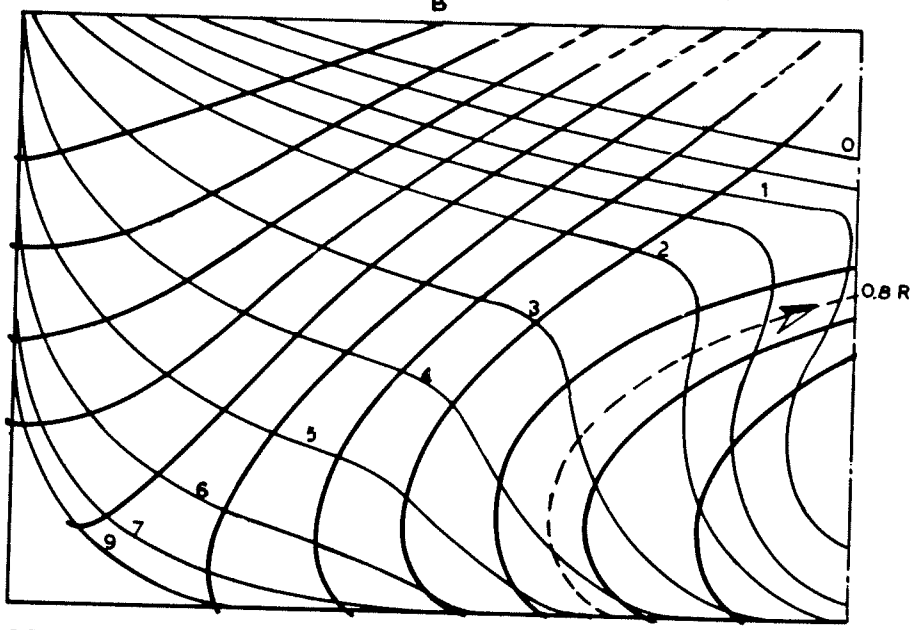
Longitudinal direction of waterflow towards propeller at 0.8 R.
(Propeller blade in top position)





STROOMLIJNEN (Flow lines)

MODEL 5122 LII = 180m $\frac{L}{B}$ = 5.6 V = 20 KNOPEN (at 20 knots)



DE STROMING HEEFT KANS OP LOSLATING T.P.V. DE NEK BOVEN DE BULB (BULB BLOKKEERT)

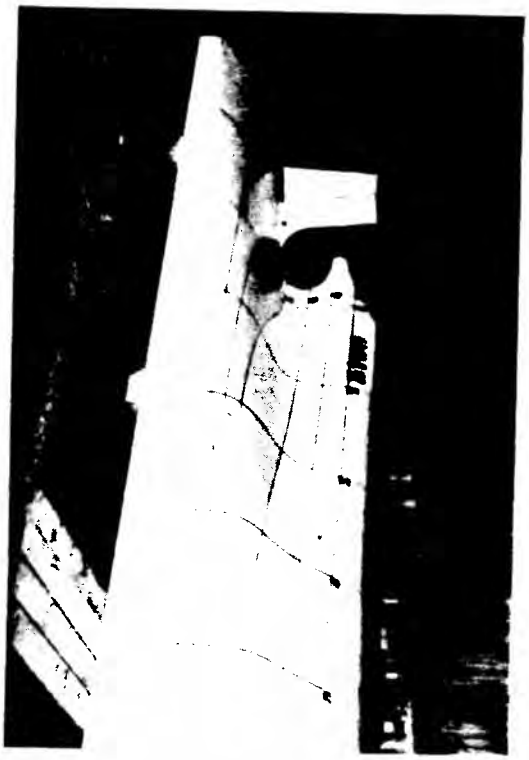
The flow may show separation in way of the neck above the gondola bulb. The bulb obstructs the flow somewhat.

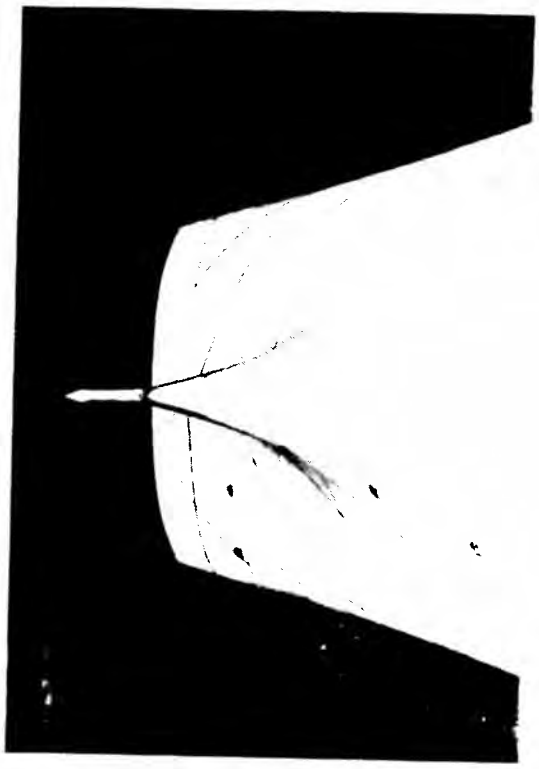
RO RO SHORTBODY MODEL 5190

NSMB
RSY 964/965

11-42

NEDLL. ROTTERDAM-ROCHESTER 1977





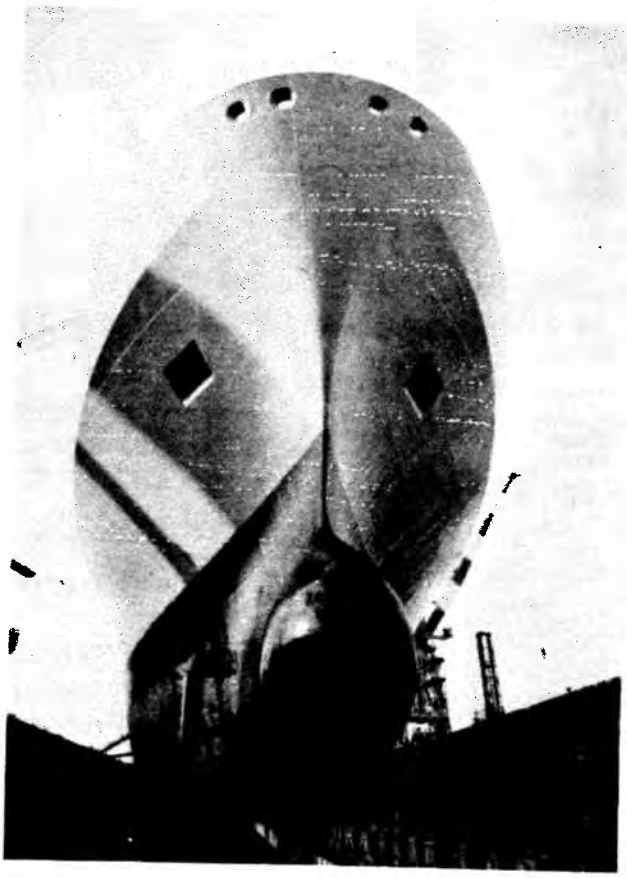
MODEL 5190

SCHIP MET EXTREEM PLATTE KONT

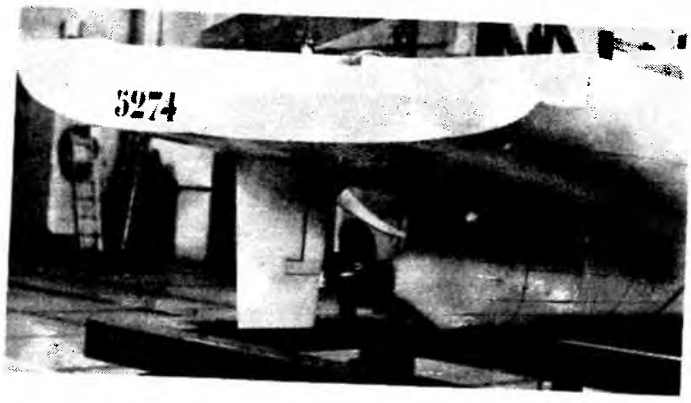
ship with very flat aftship shape (lower hull)

1979 PROPELLER OBSERVATION NEDLLOYD ROCHESTER - (RSV)
N.B. ZINC ANODES AROUND STERNTHRUSTER : WRONG

NEDLLOYD ROSARIO - NKK 1978



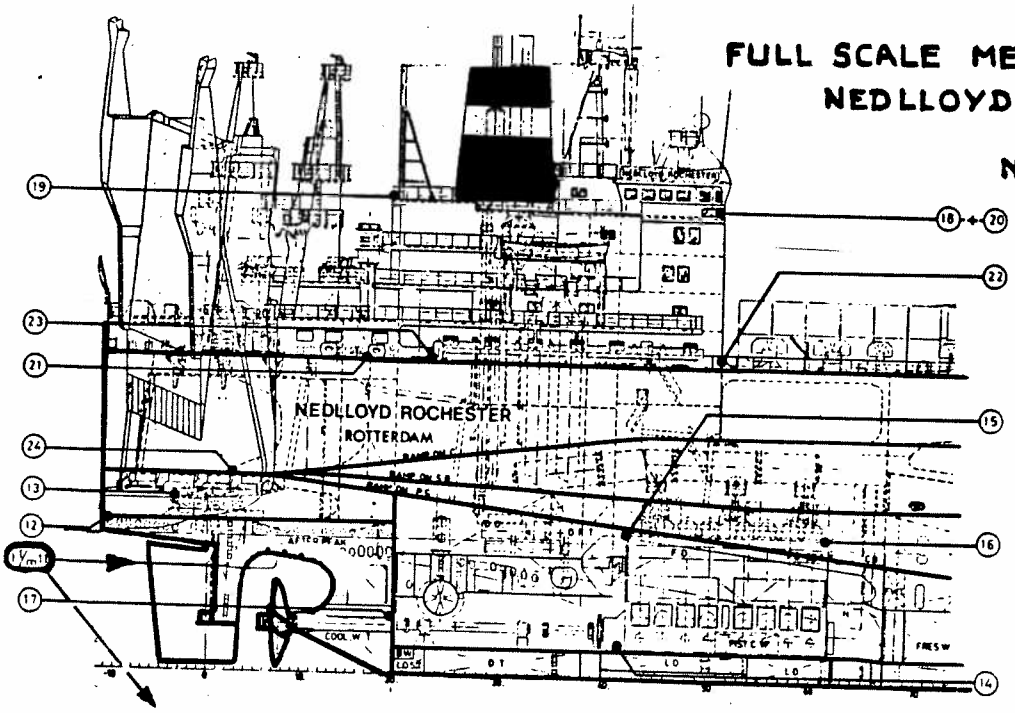
BULB SUITABLE: $T_{FWD} = 7.5 \rightarrow 10.7$ M
19 - 21 KN SEA SPEED.



MODEL TESTS AT NSMB WAGENINGEN + EDE
FLOW VISUALISATION + WAKE (3 DIM)
CAVITATION
HULL-PRESSURES

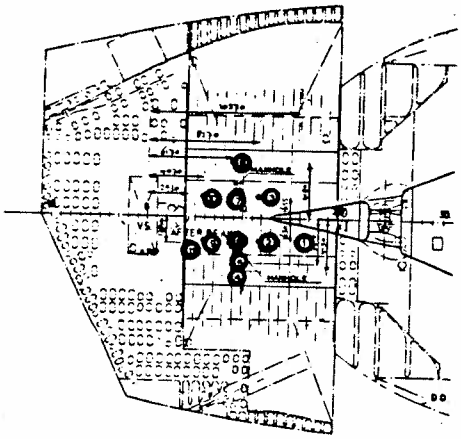
FULL SCALE MEASUREMENTS
NEDLLOYD ROCHESTER

NSMB & CRUYFF



LOCATION OF ACCELEROMETERS

(= Nos 12 & upw.)



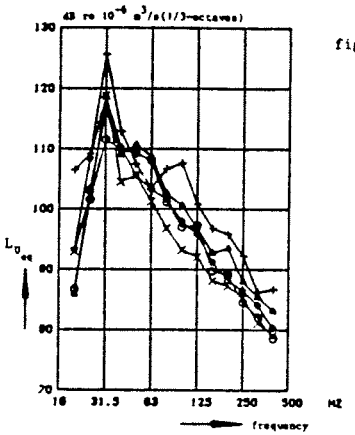
LOCATION OF PRESSURE PICK - UPS (= Nos. 1 u/11)

DIMENSIONS GIVEN IN mm

Notes: 1. Amplitudes (A) given in mbar, phase angles (α) in degrees.
2. Values are not corrected for contribution due to accelerations.
3. Harmonic components based on blade frequency.

Run No.	Number of Revolutions (RPM)	Torque (tm)	Power (HP)	Thrust (t)	Ship speed (knots)
39	120.7	128.44	21,658	114.81	21.4

Pick-up No.	1st.harmonic		2nd.harmonic		3rd.harmonic		4th.harmonic		5th.harmonic	
	A1	α1	A2	α2	A3	α3	A4	α4	A5	α5
	RESULTS OF HULL-PRESSURE MEASUREMENTS									
	PROPELLER-INDUCED HULL-PRESSURES									
6	23.901	-153	20.727	151	26.862	0	39 299	125	13.783	-114
	RESULTS OF ACCELERATION MEASUREMENTS									
	PROPELLER-INDUCED ACCELERATIONS									
6	.055	117	.213	-134	.182	10	.458	-153	.181	-55



measured with the aid of the loudspeaker

measured with the aid of the exciter

fig. 5: Volume velocity of propeller cavitation as a function of frequency. Sailing conditions: 100% propulsion power (14100 kW); ship's speed: 20,8 knots; propeller rotation speed: 116,5 rpm.

FULL SCALE MEASUREMENTS
"NEDLLOYD - ROCHESTER" 79

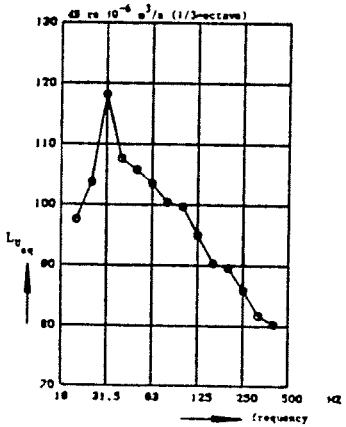


fig. 6: Volume velocity of propeller cavitation averaged with respect to transducer position. Sailing conditions: 100% propulsion power (14100 kW); ship's speed: 20,8 knots; propeller rotation speed: 116,5 rpm

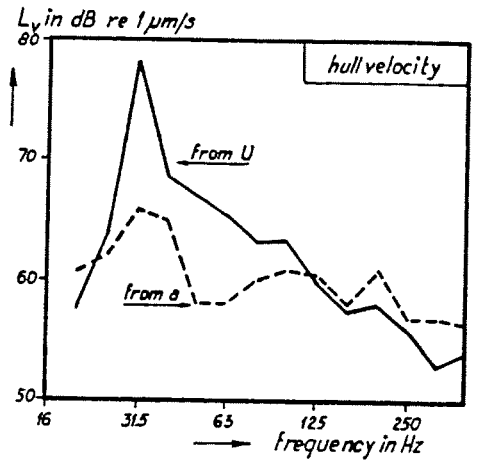


fig. 10:
Velocity level of the hull above the propeller calculated from the propeller cavitation source strength U and from the acceleration a, respectively.
Condition: 100% propulsion power

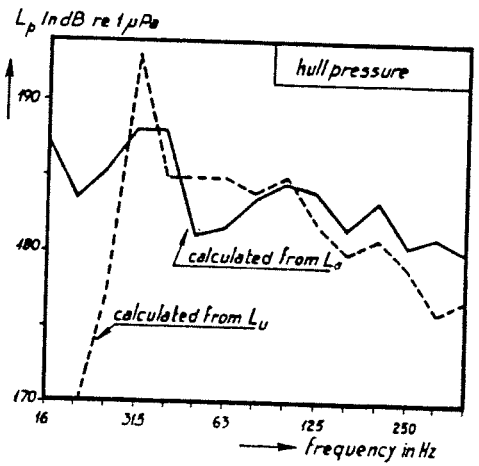


fig. 11:
Pressure underwater in front of the hull above the propeller calculated from the propeller cavitation source strength and from the acceleration level of the hull, respectively.
Condition: 100% propulsion power

8 CYL ENGINE
4 BLADED PROP

HIGHER HARMONICS RATHER STRONG!
PROPELLER SHOULD HAVE
LESS TIPLOAD AND MORE "SKEW"

BLADE FREQUENCY 1. 2. 3. 4. 5. HARMONIC.

~120 RPM
= 2 Hz

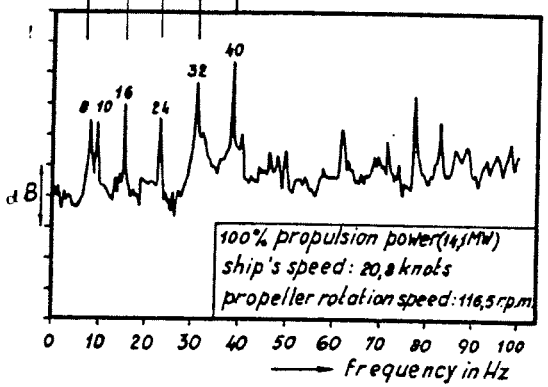


fig. 12: Relative acceleration level of the hull above the propeller, analysed in narrow frequency bands (Δf=1/4 Hz). The amplitude at blade-rate frequencies are clearly visible (8,16,24,40 Hz). Moreover a pure tone due to the propulsion engine can be observed (10 Hz).

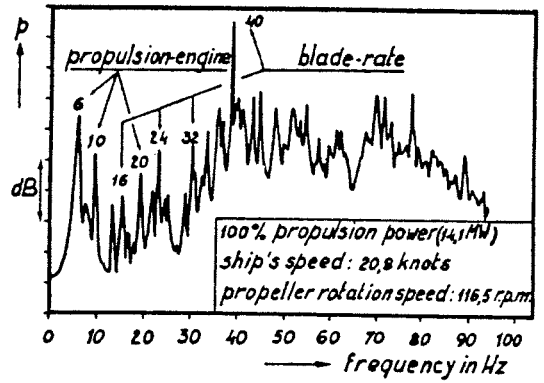


fig. 13: Relative sound pressure level in the after-peak space above the propeller, analysed in narrow frequency bands (Δf=1/4 Hz). Pure tones due to the propeller (blade-rate) and the propulsion engine are indicated.

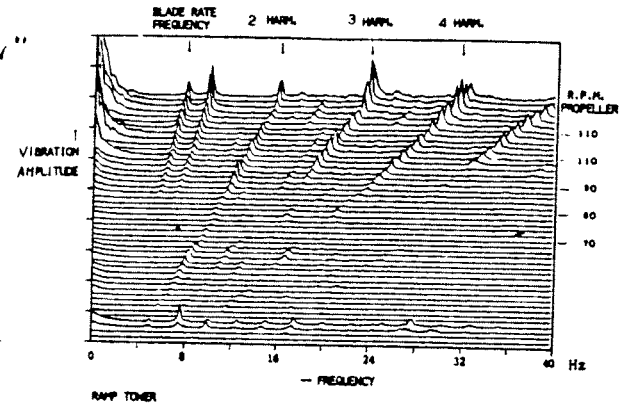


fig. 14: Vibration amplitudes as a function of frequency with the propeller rotation speed as parameter measured on the port ramp tower

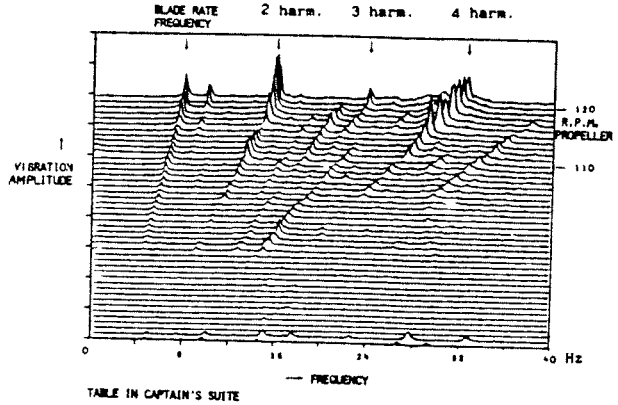
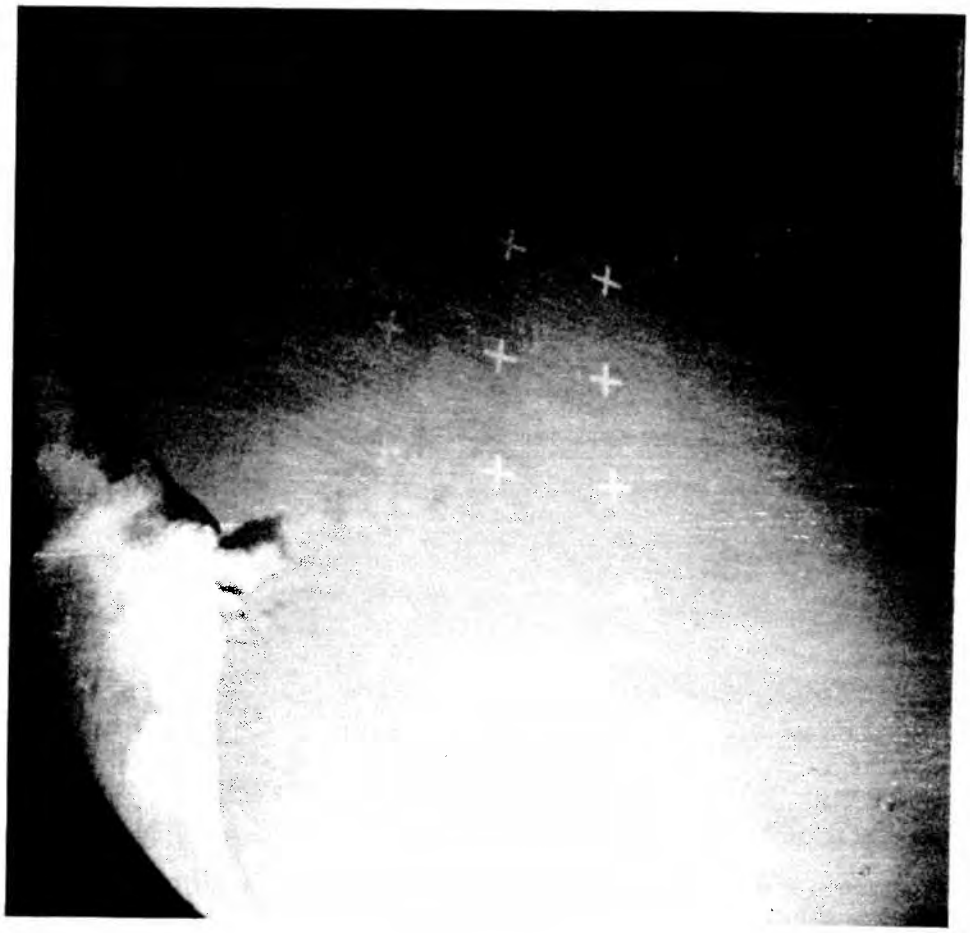
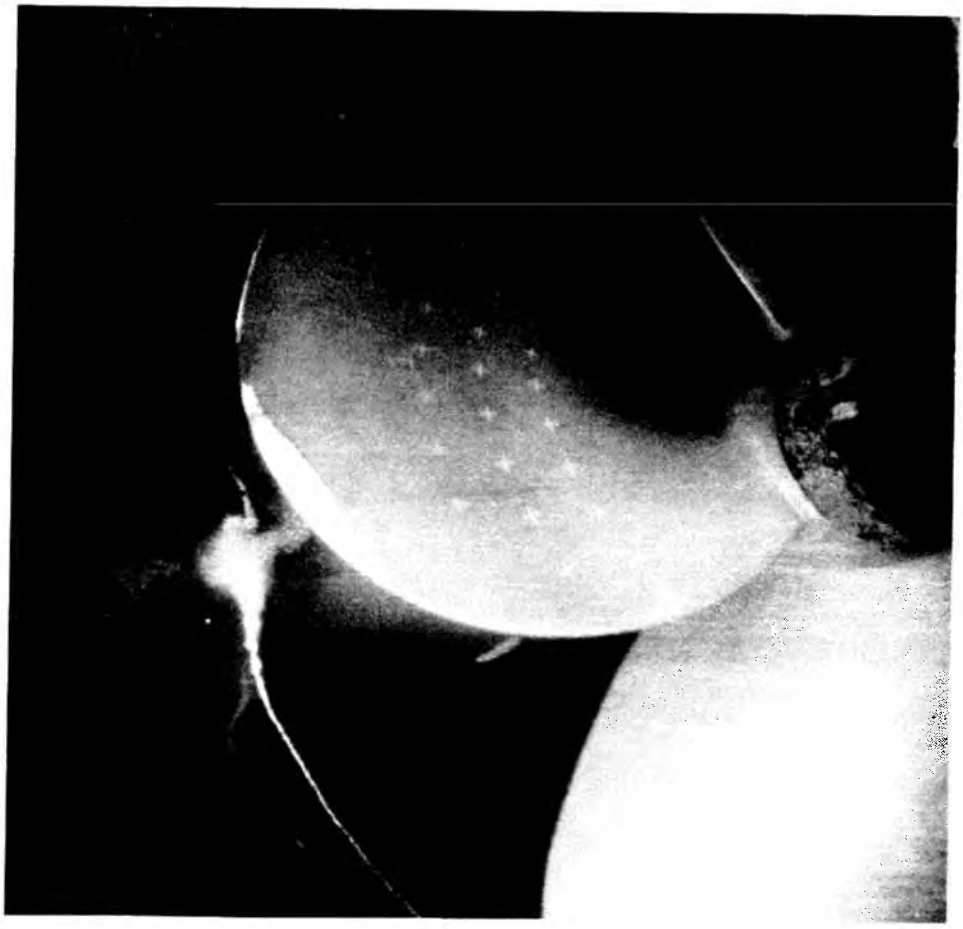
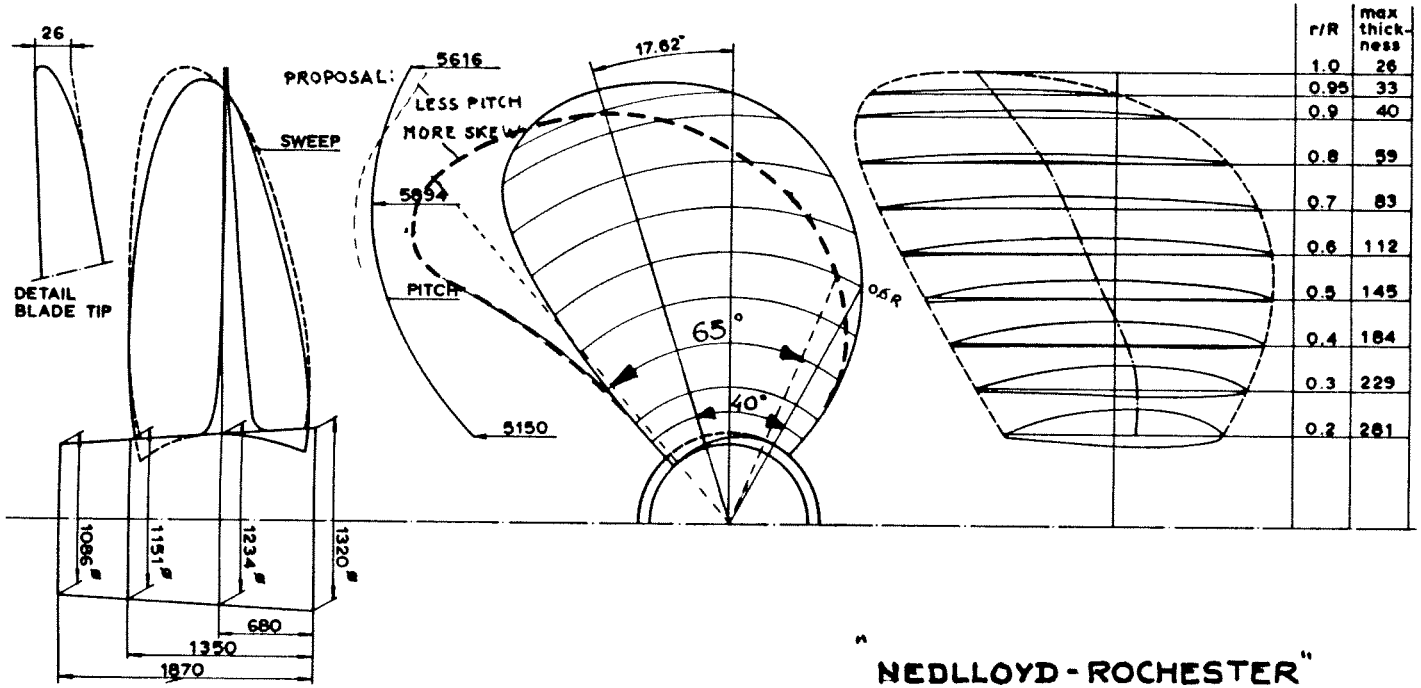


fig. 15: Vibration amplitudes as a function of frequency with the propeller rotation speed as parameter measured on the table in the captain's suite





INCREASE OF SKEW REDUCES "HIGHER-HARMONICS"-EXCITATION



r/R	max thickness
1.0	26
0.95	33
0.9	40
0.8	59
0.7	83
0.6	112
0.5	145
0.4	184
0.3	229
0.2	281

GEOMETRY OF PROPELLER

DIMENSIONS ARE GIVEN IN mm FOR SHIP

"NEDLLOYD-ROCHESTER"

PROPELLER DIAMETER	D =	6500
PITCH RATIO AT 0.7 R	$P_{0.7}/D$ =	0.907
EXPANDED BLADE AREA RATIO	A_E/A_O =	0.726
CHORD LENGTH$_{0.7}$ / DIAMETER	$c_{0.7}/D$ =	0.420
THICKNESS / CHORD LENGTH$_{0.7}$	$t/c_{0.7}$ =	0.0305

GEOMETRY OF 4-BLADED RIGHT-HANDED PROPELLER MODEL No. 5520.

11-49

SHOCKABSORBER ACCELERATOR

ABOVE

PROPELLER

PROPOSALS '82

HUMP
ABOVE PROPELLER

INCREASE OF FLOW
IN 12 OCLOCK POS.?

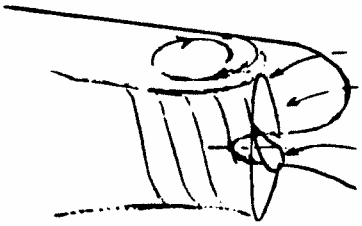
STROOMLIJNEN ACHTERSCHIP T.P.V. GONDEL EN „PUKKEL"

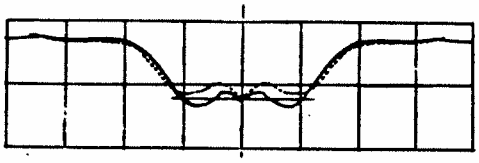
MODEL 5141A

LET OP LOSLATINGSWERVEL LANGS GONDEL!

PROPOSALS FOR "BULGE-UNDER-THE-STERN".

BULGE/LUMP IN SHELL, FOR WAKE REDUCTION PURPOSES.





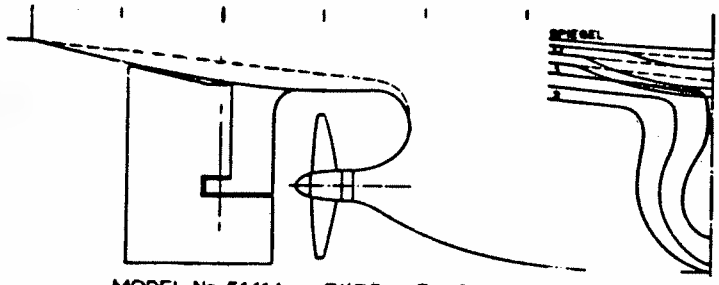
Wake graphs at 0.8R .

——— = Model 5141 (= without lump)

- - - = Model 5141-A (= with lump)

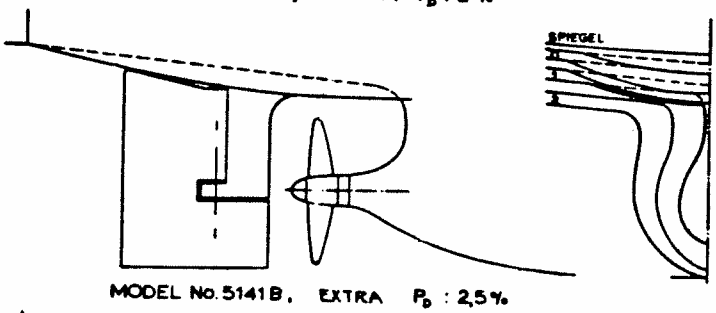
Wake reduction by speeding-up the water flow overhead of the propeller.

Ordin. $-\frac{1}{2}$ 0 $+\frac{1}{2}$ 1 $1\frac{1}{2}$



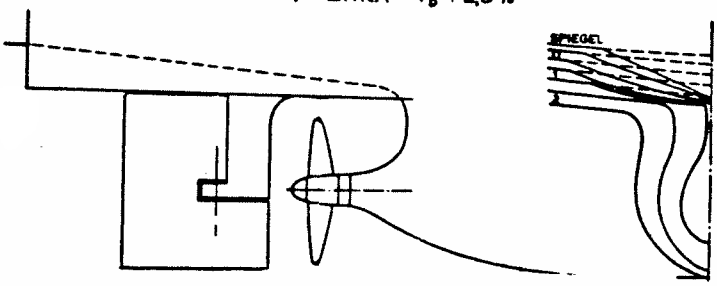
MODEL No. 5141A, EXTRA P_D : 2 %

Lump over propeller.



MODEL No. 5141B, EXTRA P_D : 2,5 %

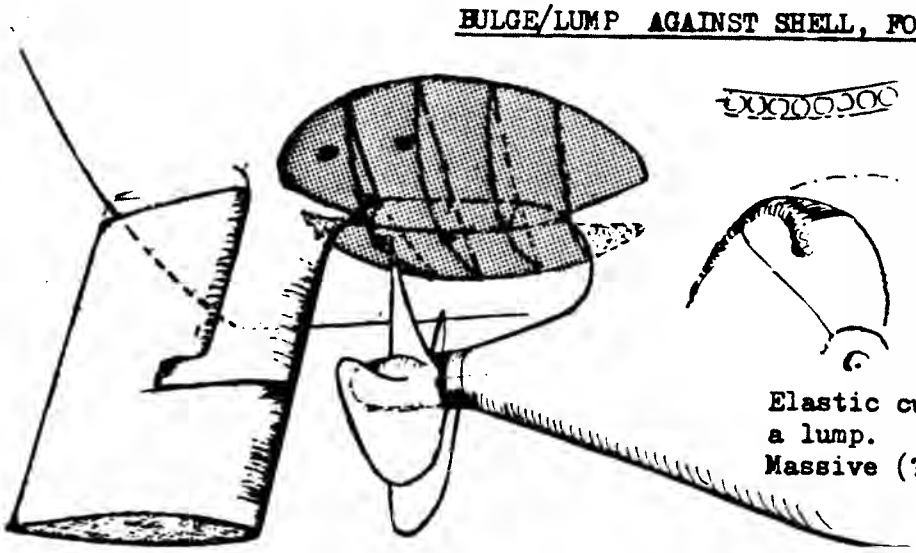
Longer and wider lump.



MODEL No. 5141C. EXTRA P_D : 4 %

Lump extended to transom.

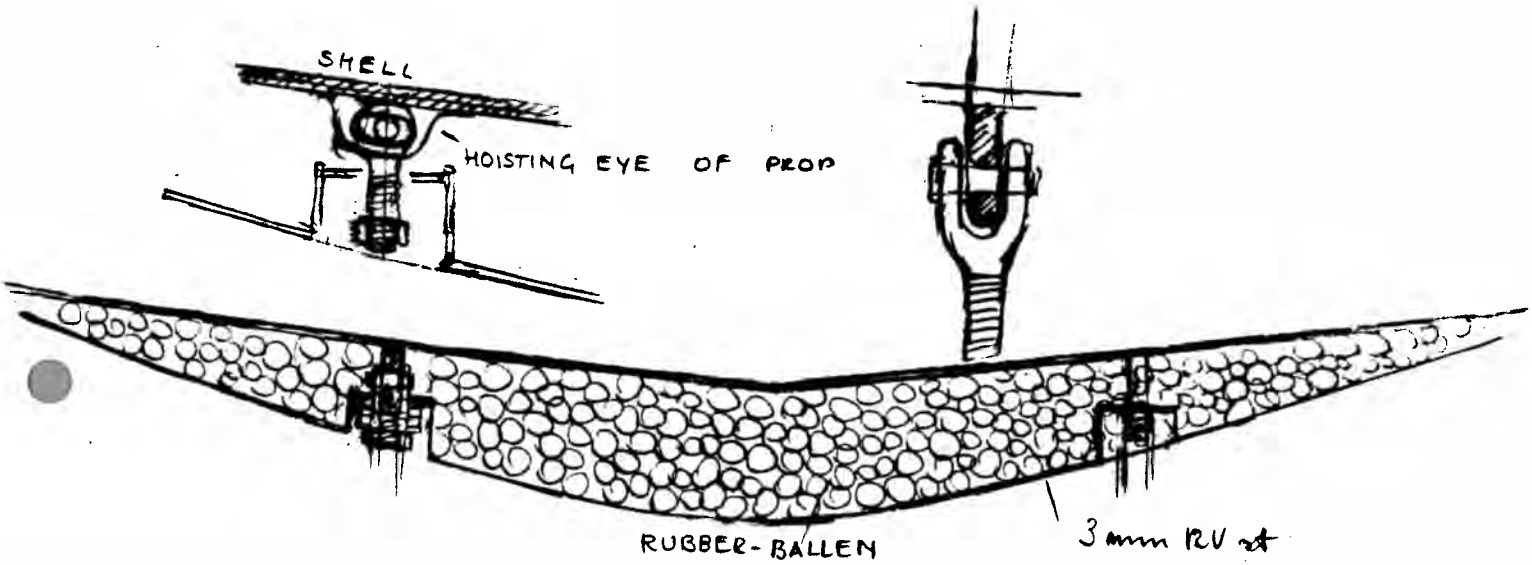
BULGE/LUMP AGAINST SHELL, FOR SHOCK-ABSORBING NOISE-SUBDUEING PURPOSE.



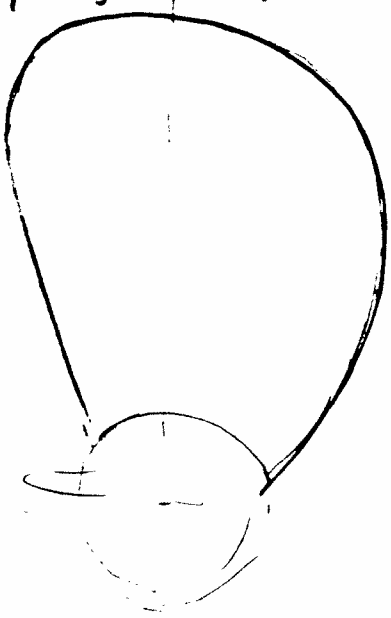
Elastic cushion, in the shape of a lump.
Massive (?) or air-containing (?).

"EI[illegible] COLUMBUS" ?

11-52



38 Tc 1' [illegible] . 8 Hz
20 T 1' [illegible] 16 Hz.
10 T 3 ~ | 24 Hz

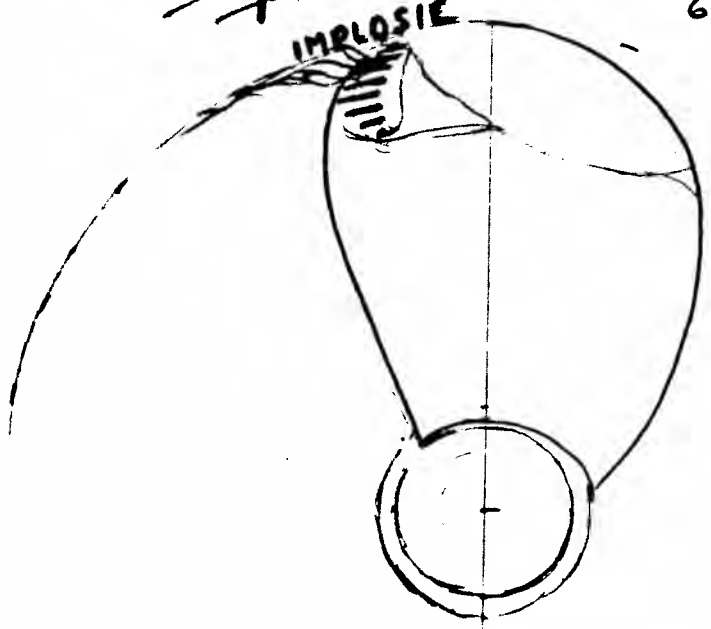


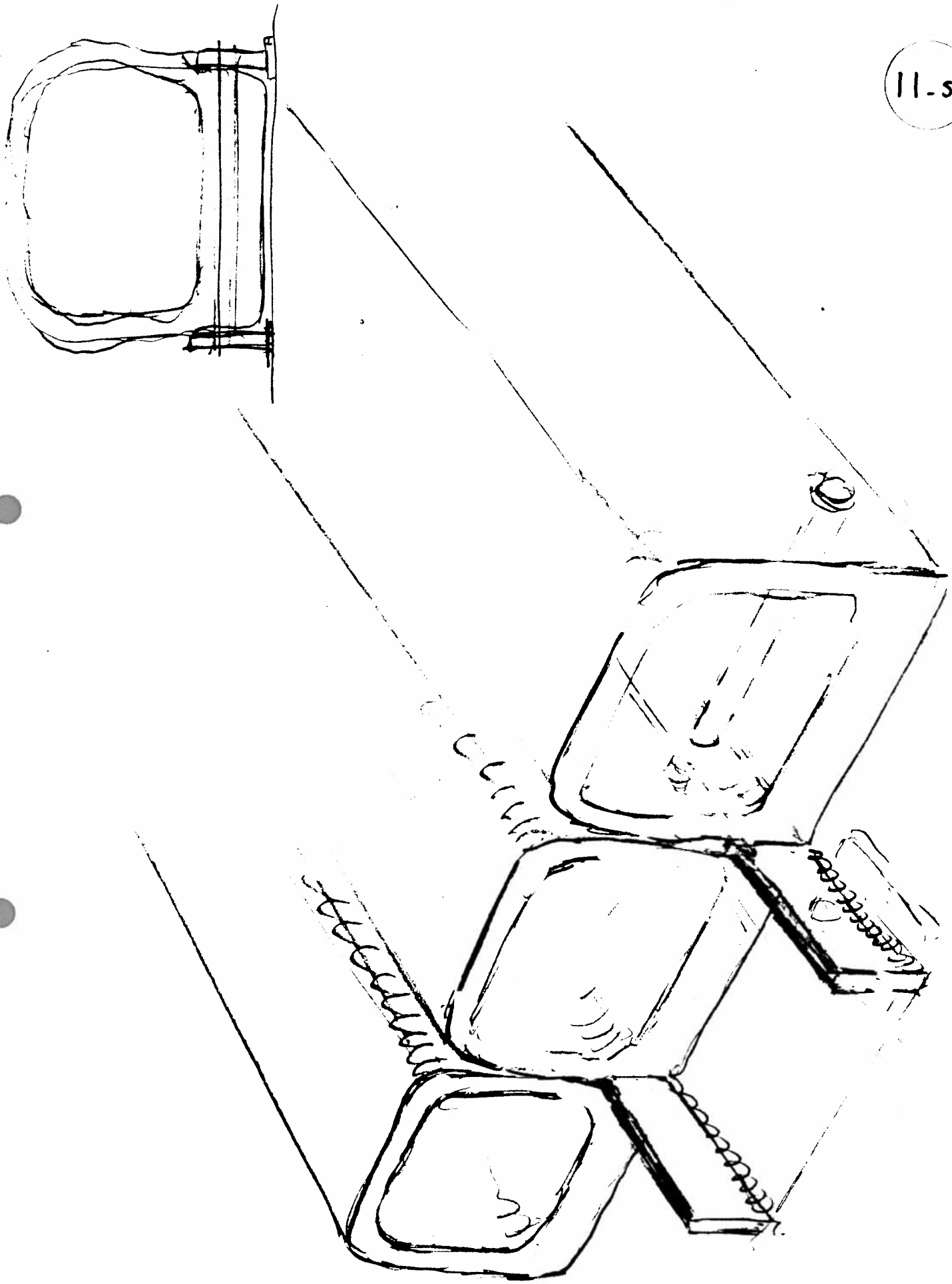
TO BE MOUNTED
IN DRYDOCK AS ONE UNIT

$$\frac{120\ \text{RPM}}{60} = 2\ \text{Hz} \times 4 = 8\ \text{Hz}$$

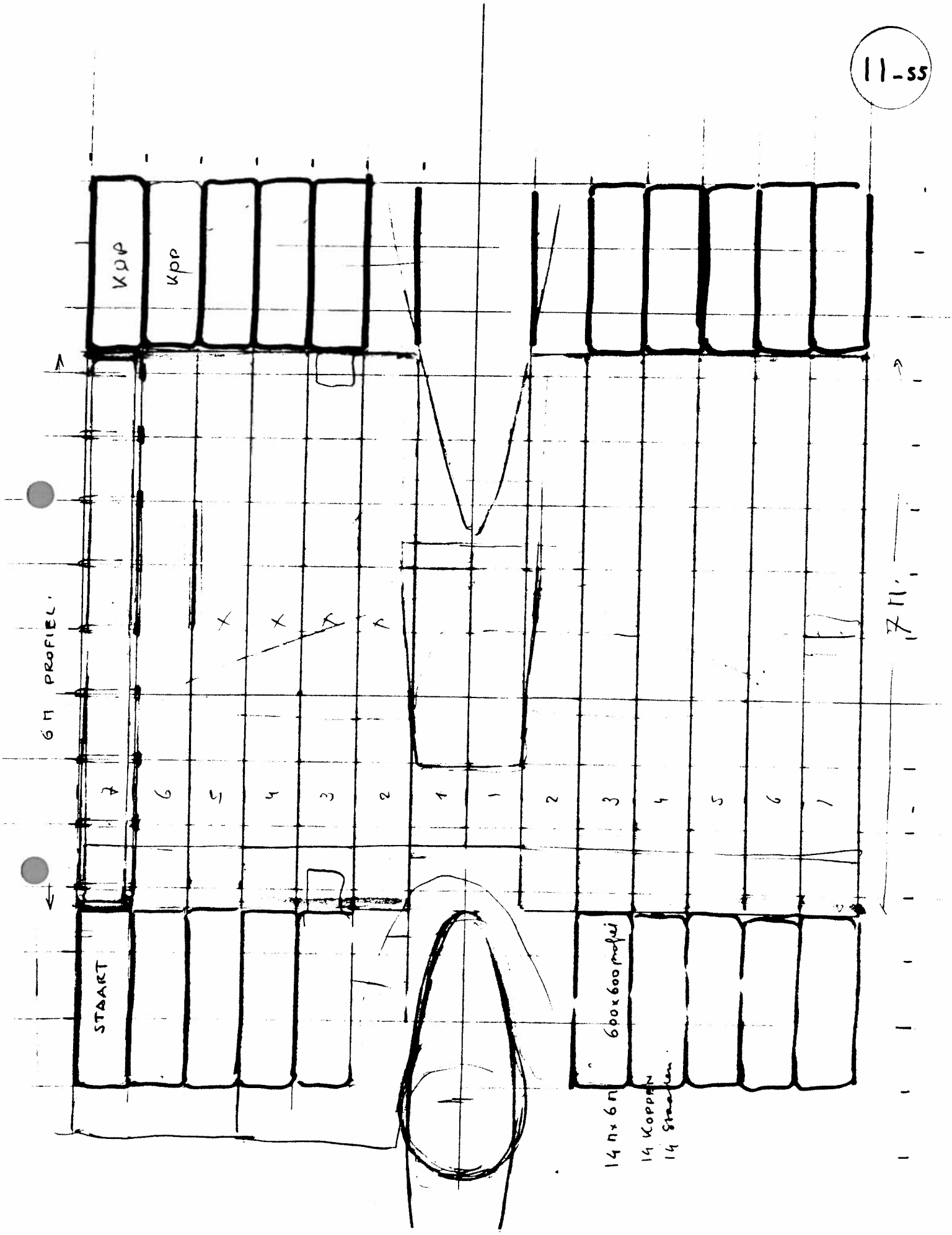
FENDER CONSTRUCTION

$\frac{120 \times 4}{60} = 8$ Hz - x2 = 16
3 = 24
4 = 32
5 = 40



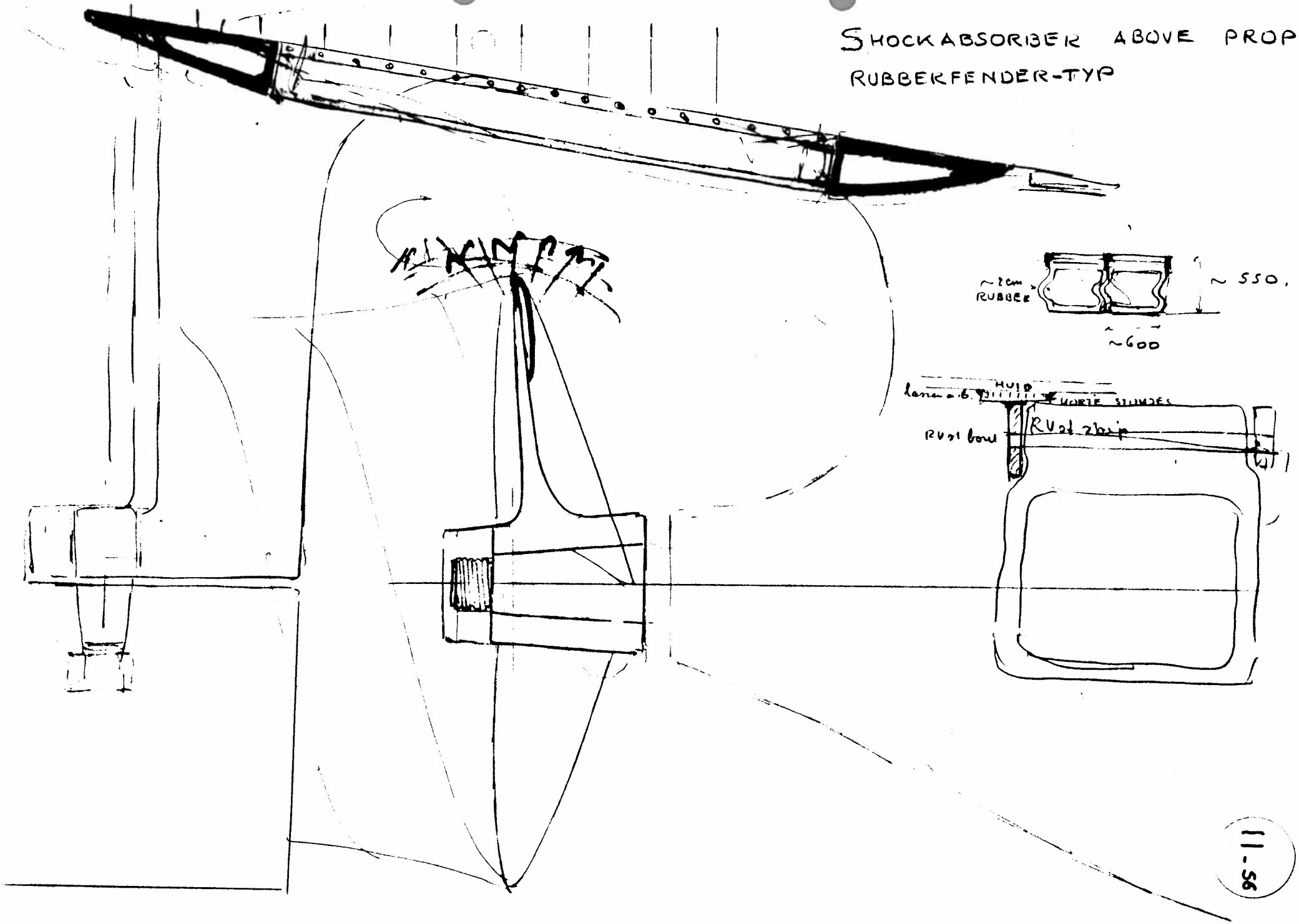


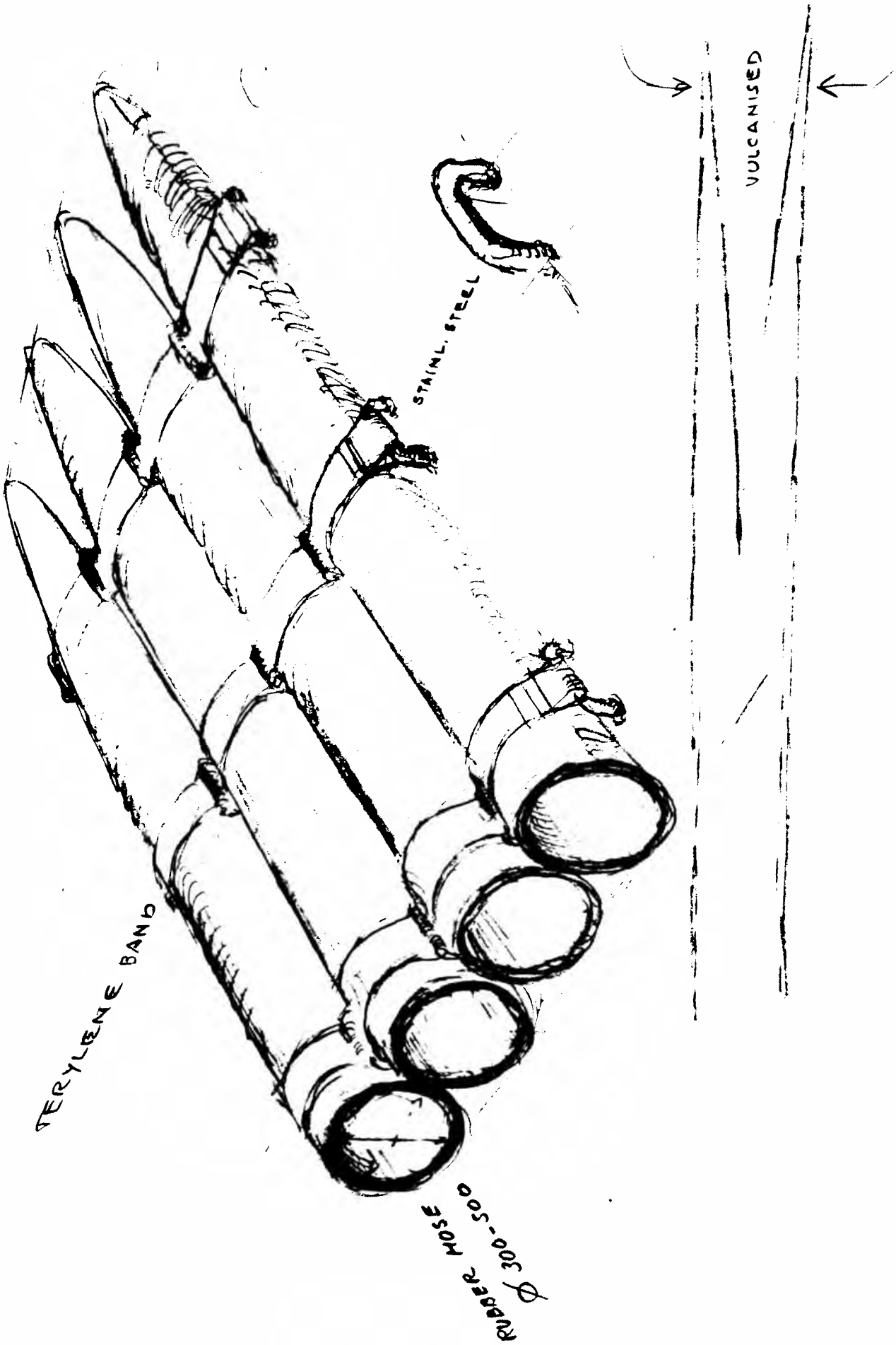
Proposal for shock absorbing arrangement



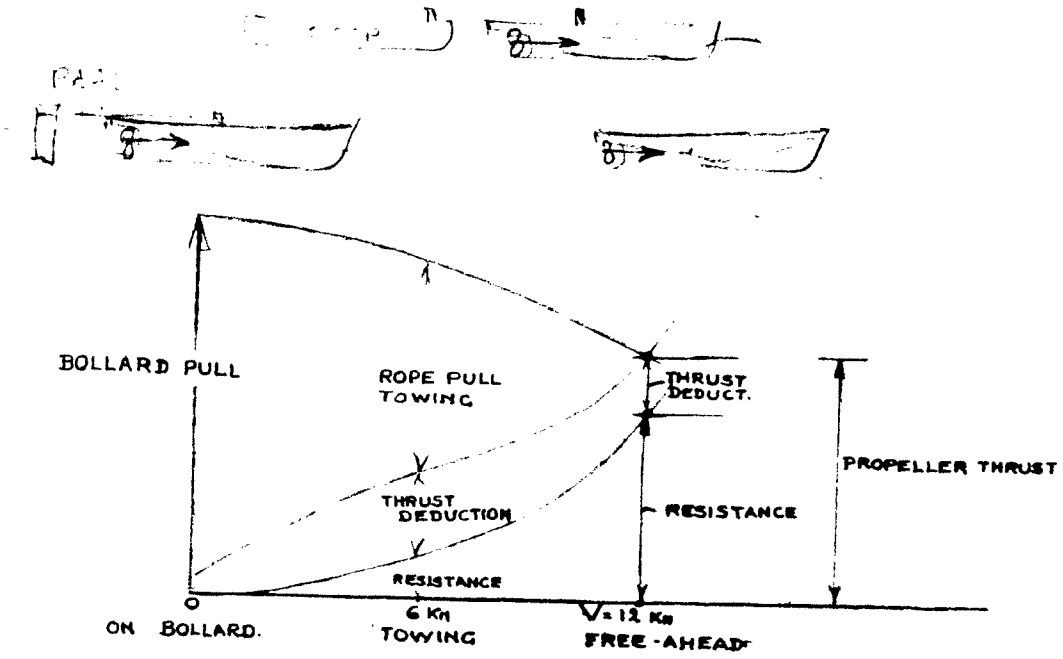
Proposal for shock absorbing arrangement

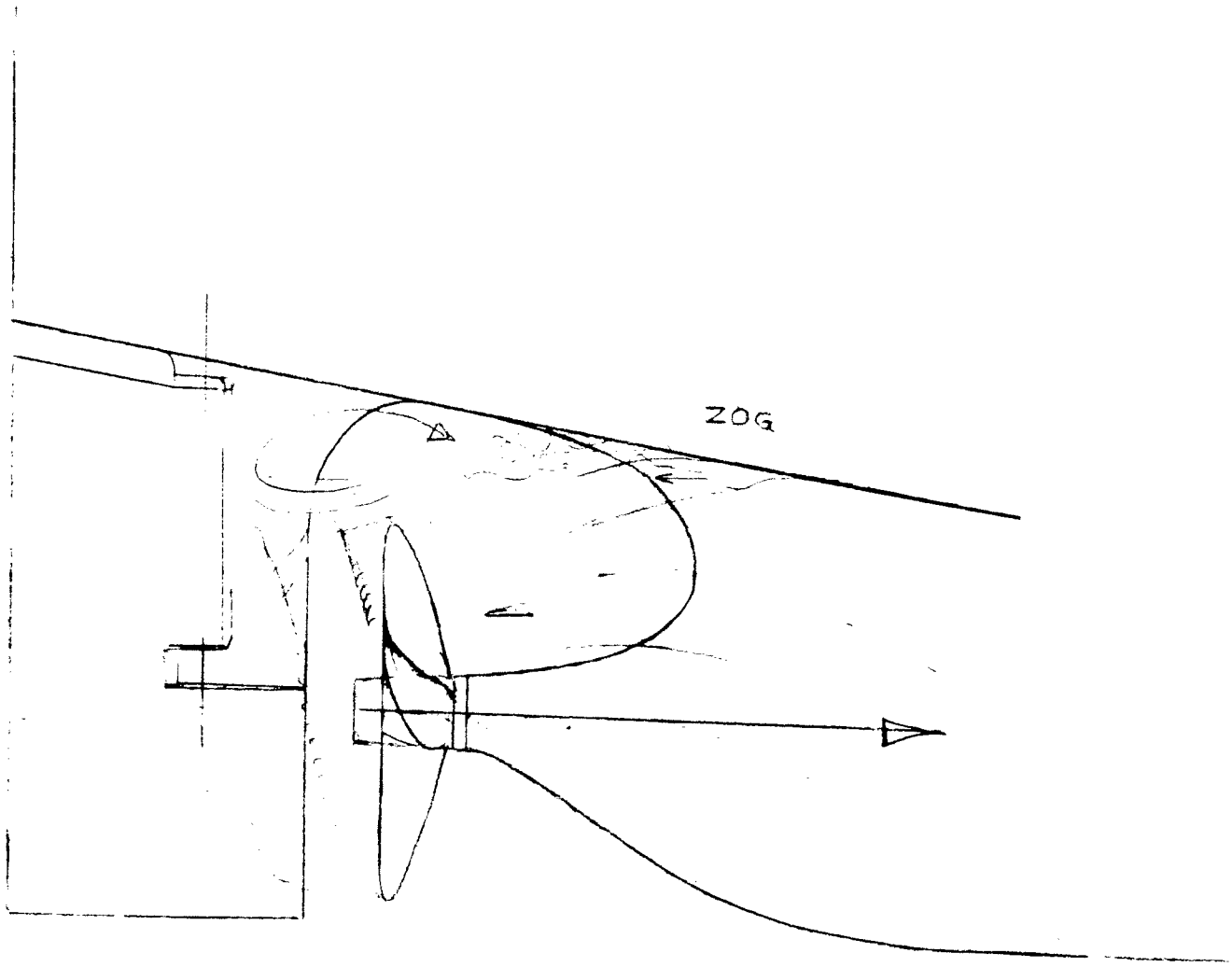
SHOCKABSORIBER ABOVE PROP.
RUBBERFENDER-TYP





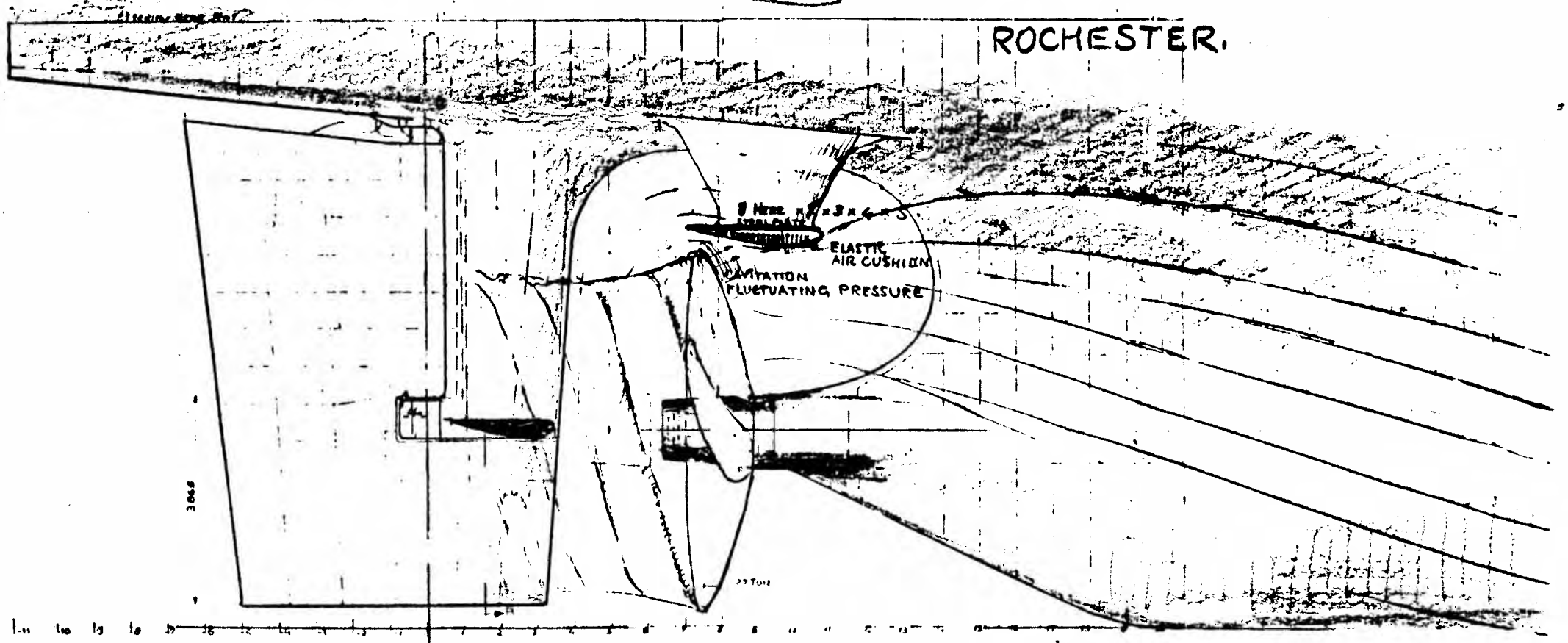
Proposal for shock absorbing arrangement


BOLLARD PULL
ROPE PULL
TOWING
THRUST
DEDUCT.
PROPELLER THRUST
THRUST
DEDUCTION
RESISTANCE
RESISTANCE
O
6 Kn
V = 12 Kn
ON BOLLARD.
TOWING
FREE-AHEAD


ZOG

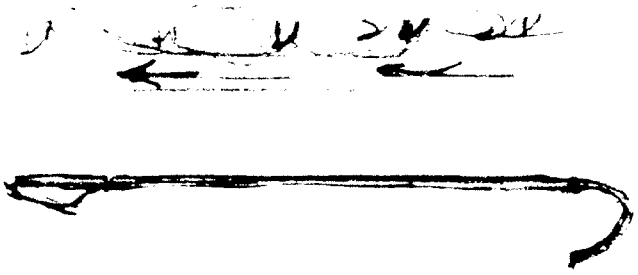
CURVED WING ABOVE PROPELLER

R. R. Neelloyd
VDSH 964-965

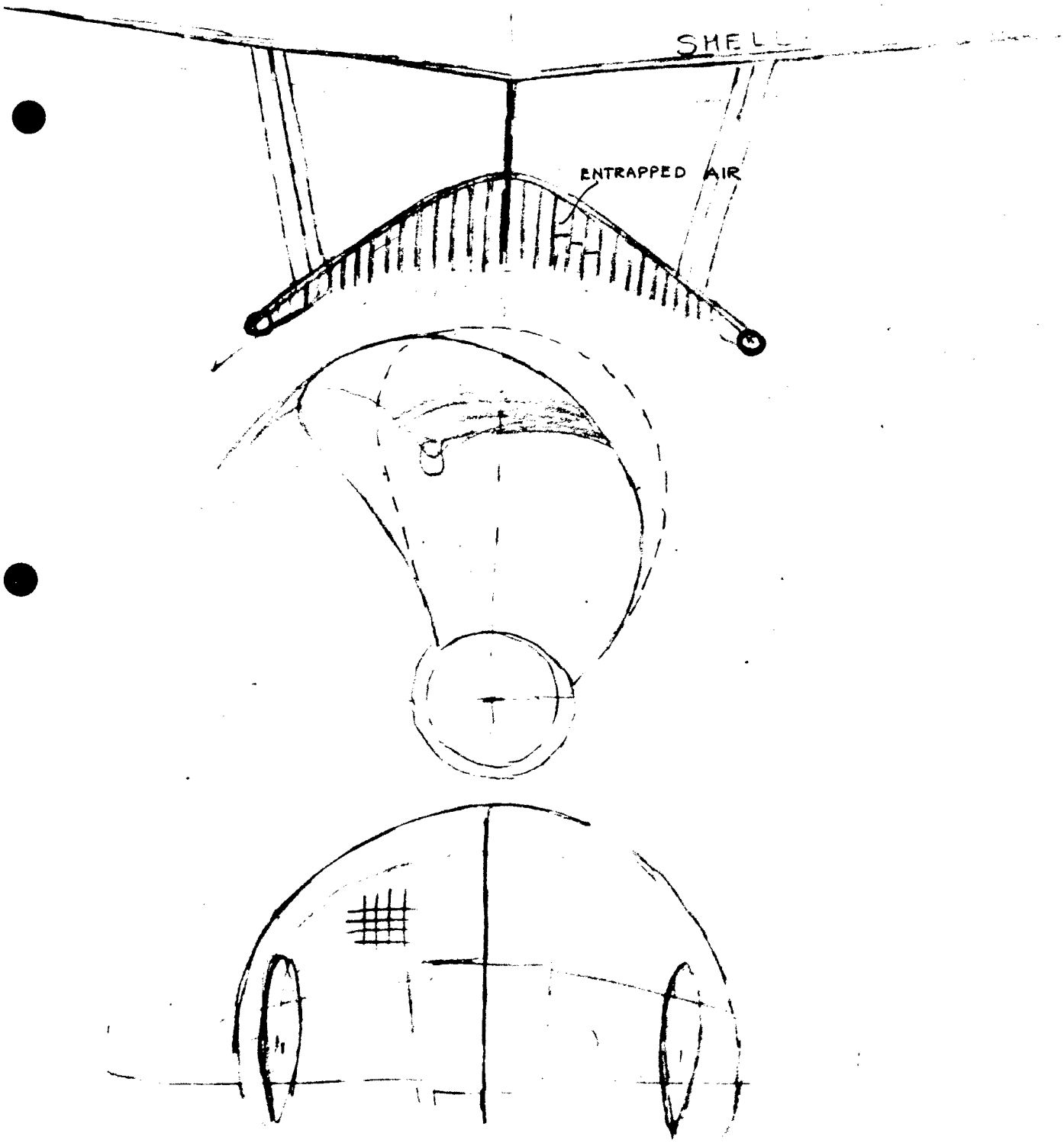


DOORSNLDE OP SPT. -2

DOORSNLDE OP SPT. 0



RED COPPER (?) MESH (OPEN AIR PIPES)
SHOCK ABSORBER
FLOW ACCELERATOR



FLOW ACCELERATING NOZZLE IN UPPER QUADRANT

PROF SCHNEEKLUTH AACHEN UNIVERSITY
MR STEINER DUISBURG MODELBASIN

REDUCTION OF WAKE PEAK AT 12 O'CLOCK.

LESS CAVITATION (EROSION)
LESS VIBRATIONS
LESS NOISE

SCHNEEKLUTH

"HALF" NOZZLE
IN UPPER QUADRANT

REDUCTION OF THRUST DEDUCTION : t

LESS SEPARATION OF FLOW
LESS TURBULENCE.

INCREASE OF HULL-EFFICIENCY $\frac{1-t}{1-\psi}$ ψ = WAKE

INCREASE OF PROPELLER EFFICIENCY BECAUSE OF INCREASED V_E

INCREASE OF TOTAL PROPULSIVE EFFICIENCY

$$\eta_{TOT} = \eta_P \cdot \frac{1-t}{1-\psi} \cdot \eta_{rel\,rot}$$

HOWEVER:

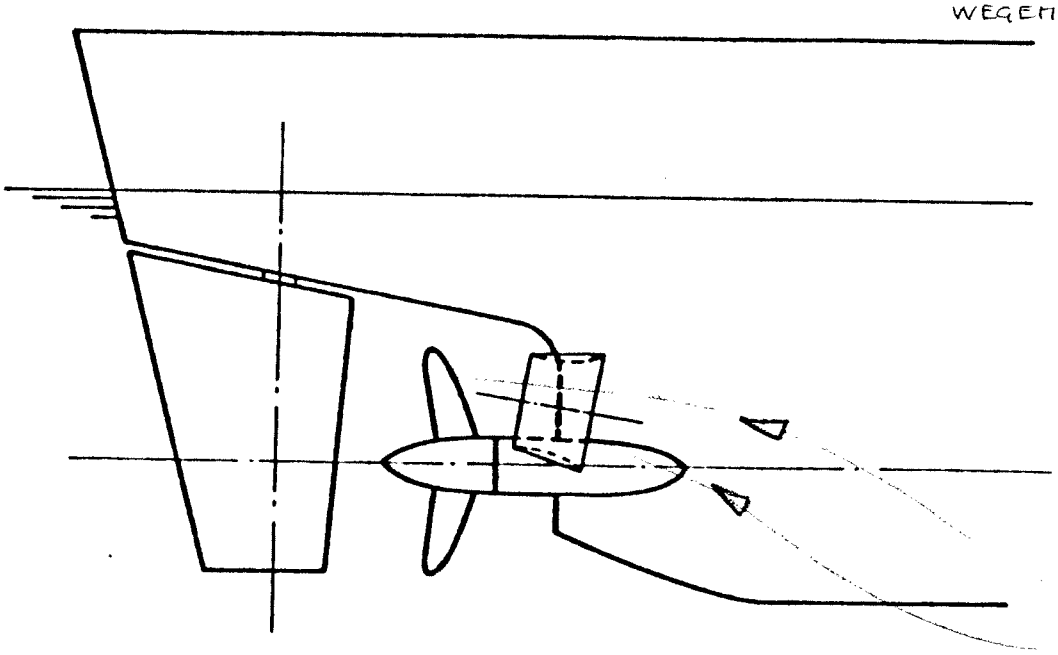
ALSO INCREASE OF SHIPS RESISTANCE E.H.P.

ACCORDING TO TESTS IN DUISBURG:

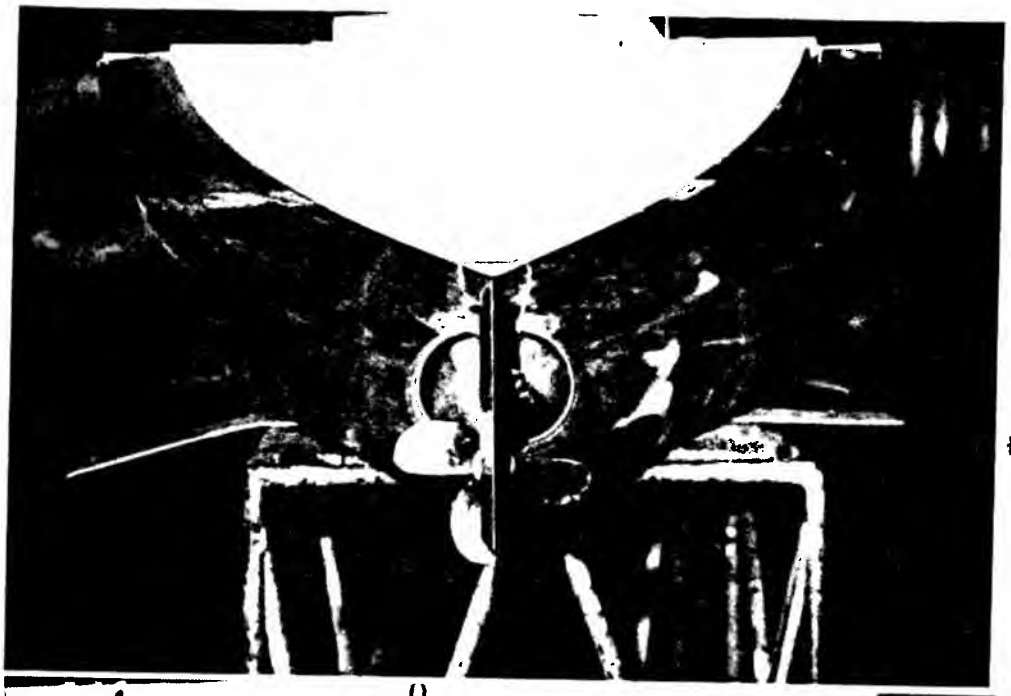
INCREASE OF SHIPS SPEED
ALTERN: DECREASE OF REQUIRED SHP FOR SAME SPEED.

FULLBODY 6-8%
FINER HULL LESS

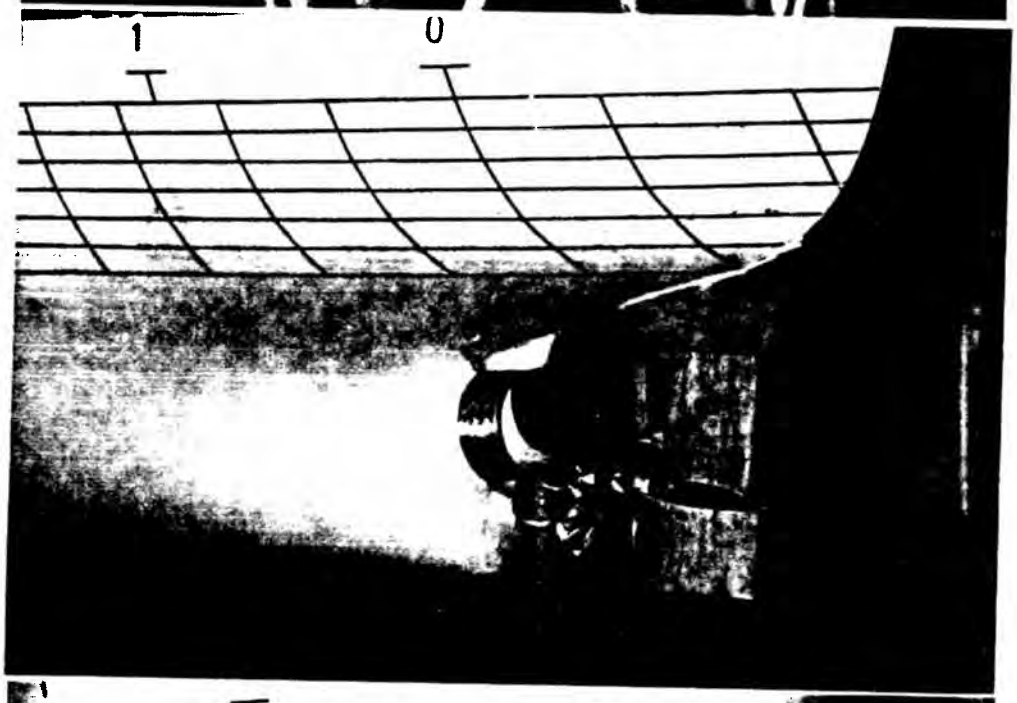
SEPT 1983
GÖTEBORG
WEGEMT

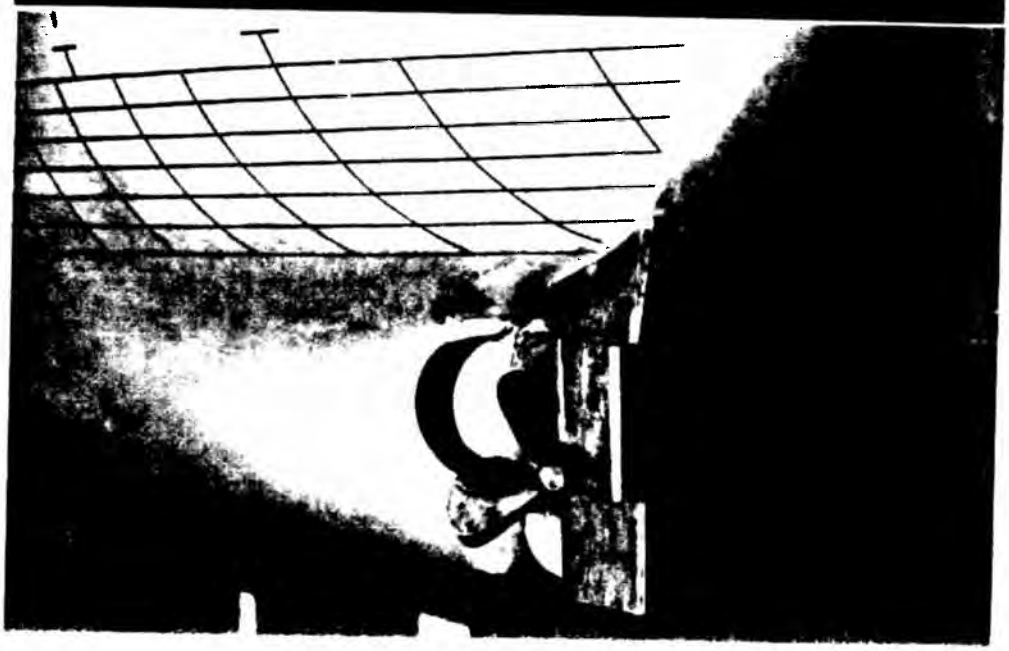


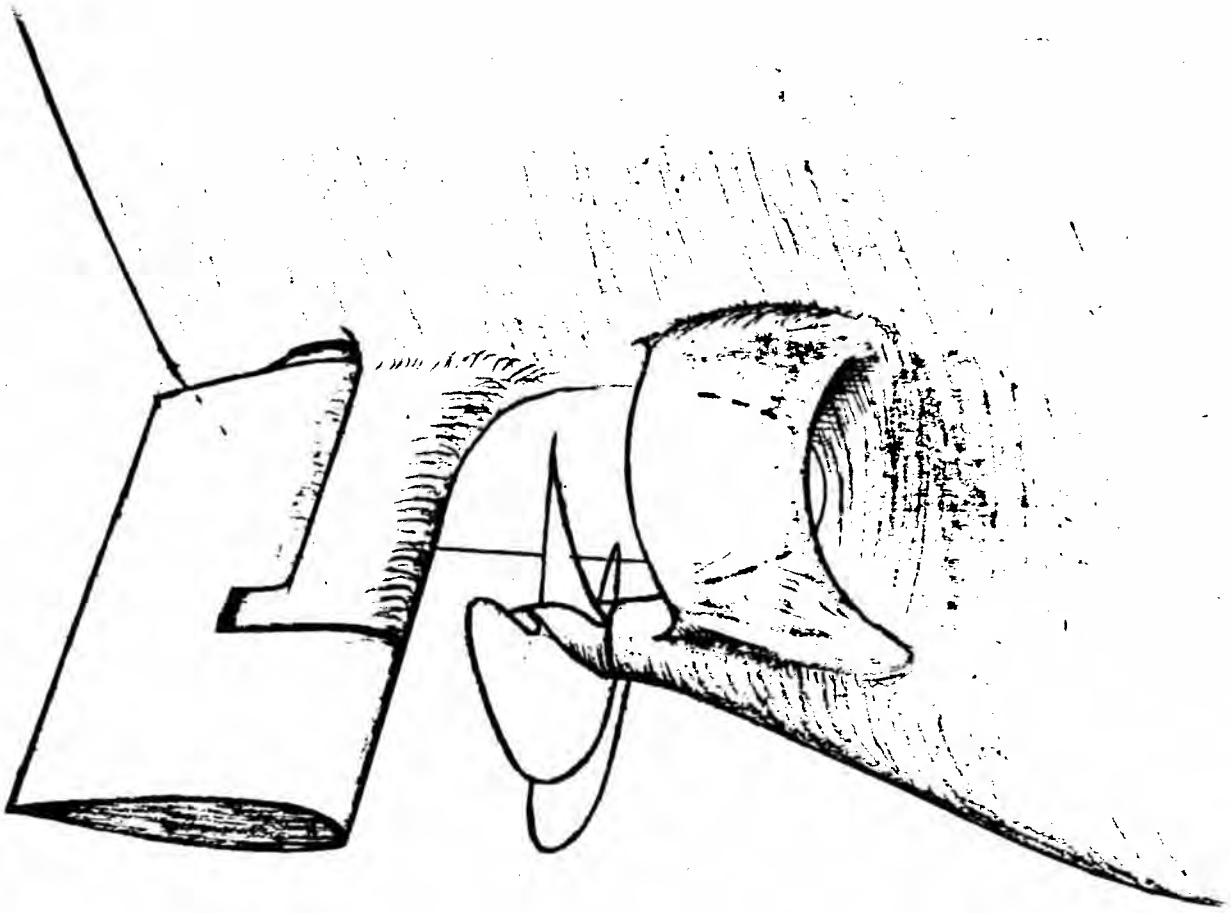
FRACHTSCHIFFSMODELLE MIT VORDÜSEN

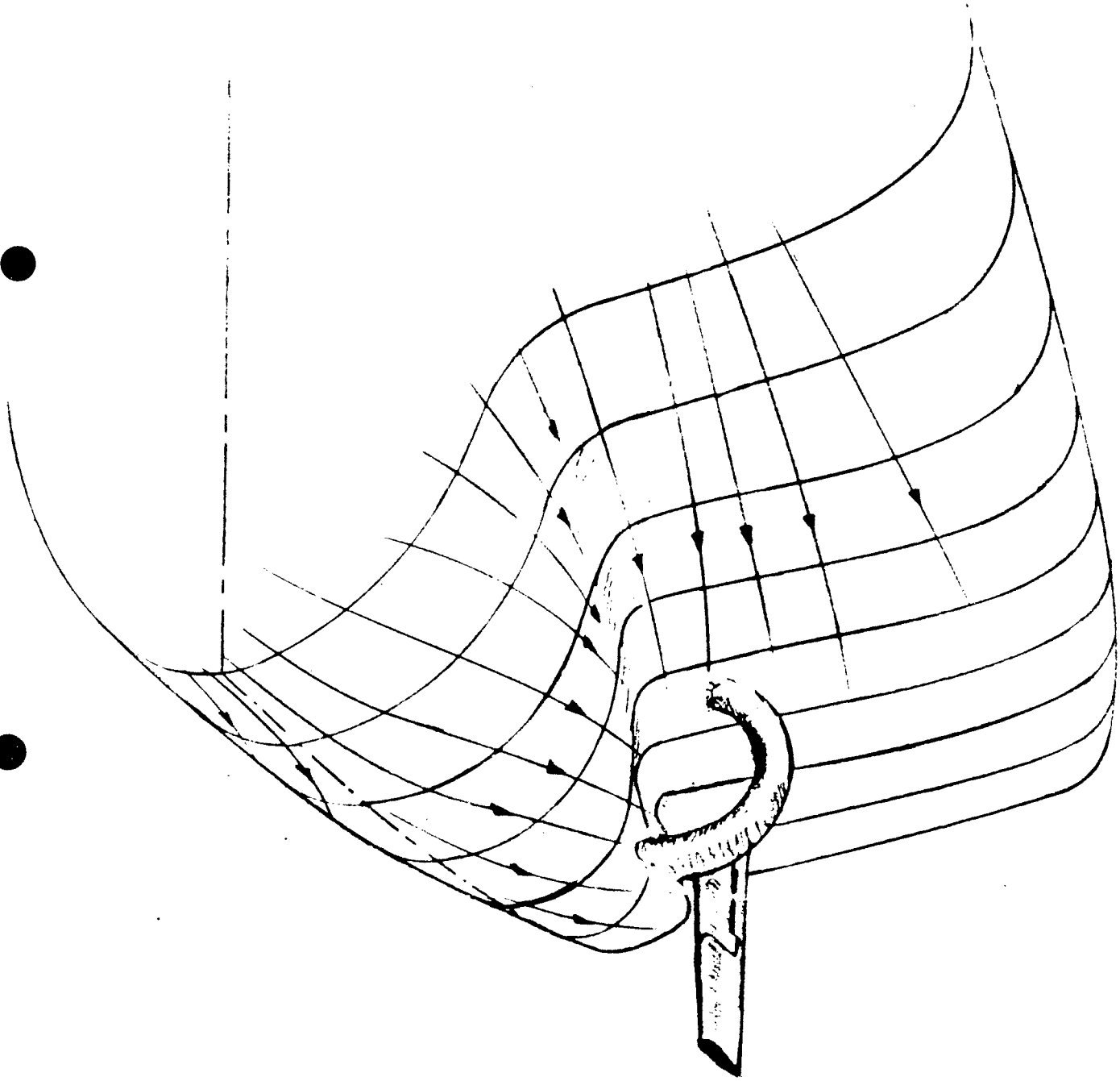


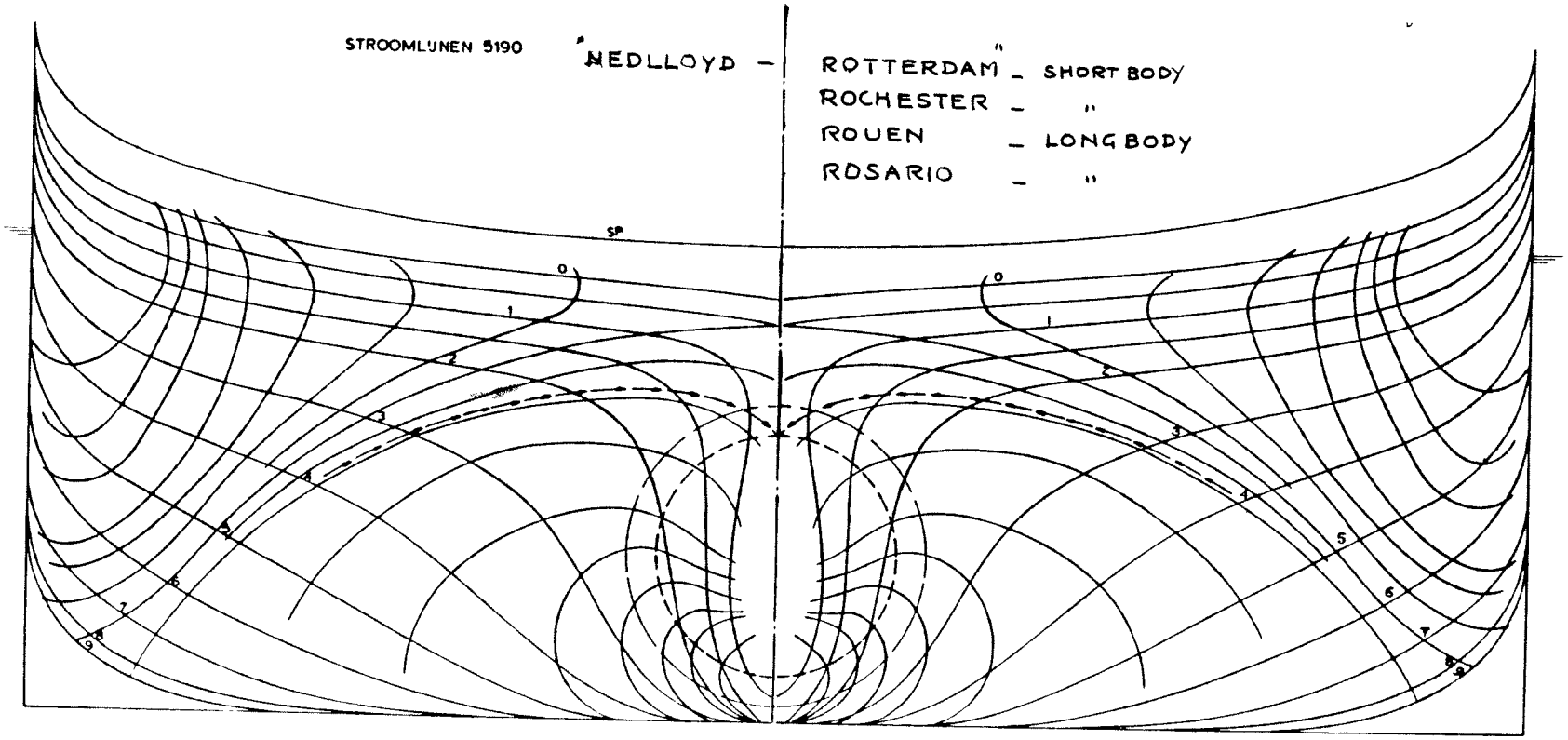
SCHNEEKLUTH
HOCHSCH.-AACHEN
DUISBURG
TANK











Flow lines of model 5190 aloft aftship

12

2S

Ro Ro FERRY

TWIN SCREW ON TWIN SKEGS/TWIN RUDDER Fig 45-46-47-48

This arrangement is to be considered as two single screw afterbodies parallel to each other, which means higher resistance in comparison with a normal twin screw ship, because of the more complicated waterflow along a larger hull surface.
The twin skeg ship has a higher propulsive coëfficient in comparison to the normal twin screw ship, because of the 50 per cent to 60 per cent wake instead of 25-30 per cent wake. The horsepower for a given speed, however, is about the same, however, due to the larger wake peaks there exists danger of hull pressure fluctuations.
A ferry with restricted draught, high horsepower and restricted propellers turning at high tip-speed is critical, especially, where controllable pitch propellers do not permit sufficient blade surface as would be required for the power available. The propeller blade-tip crosses the wake peak at high speed (which becomes critical at about 42 m/sec), cavitation will occur and strong hull pressure fluctuations will be excited.

The advantage of a twin-skeg arrangement is that the engines can be accommodated further aft and utilisation of space in the lower hull is superior to that in the normal twin screw hull. The centre of buoyancy is further aft, which is attractive to obtain a good trimming balance of the vessel; however, these advantages do not outweigh the higher risk of vibration troubles in the case of highly powered passenger ferries. Generally speaking, we would adopt this twin-skeg arrangement only on ships of sufficient draught to enable large diameter propellers which are less heavily loaded and run at low revolutions. The centre-line skeg-aft is still necessary for good coursekeeping at low speed in case c.p. propellers blank off the rudders. This skeg is also needed for drydocking purposes and the extra frictional resistance should be taken for granted as necessary disadvantage.

Ferryboat captains: (Nordsea)

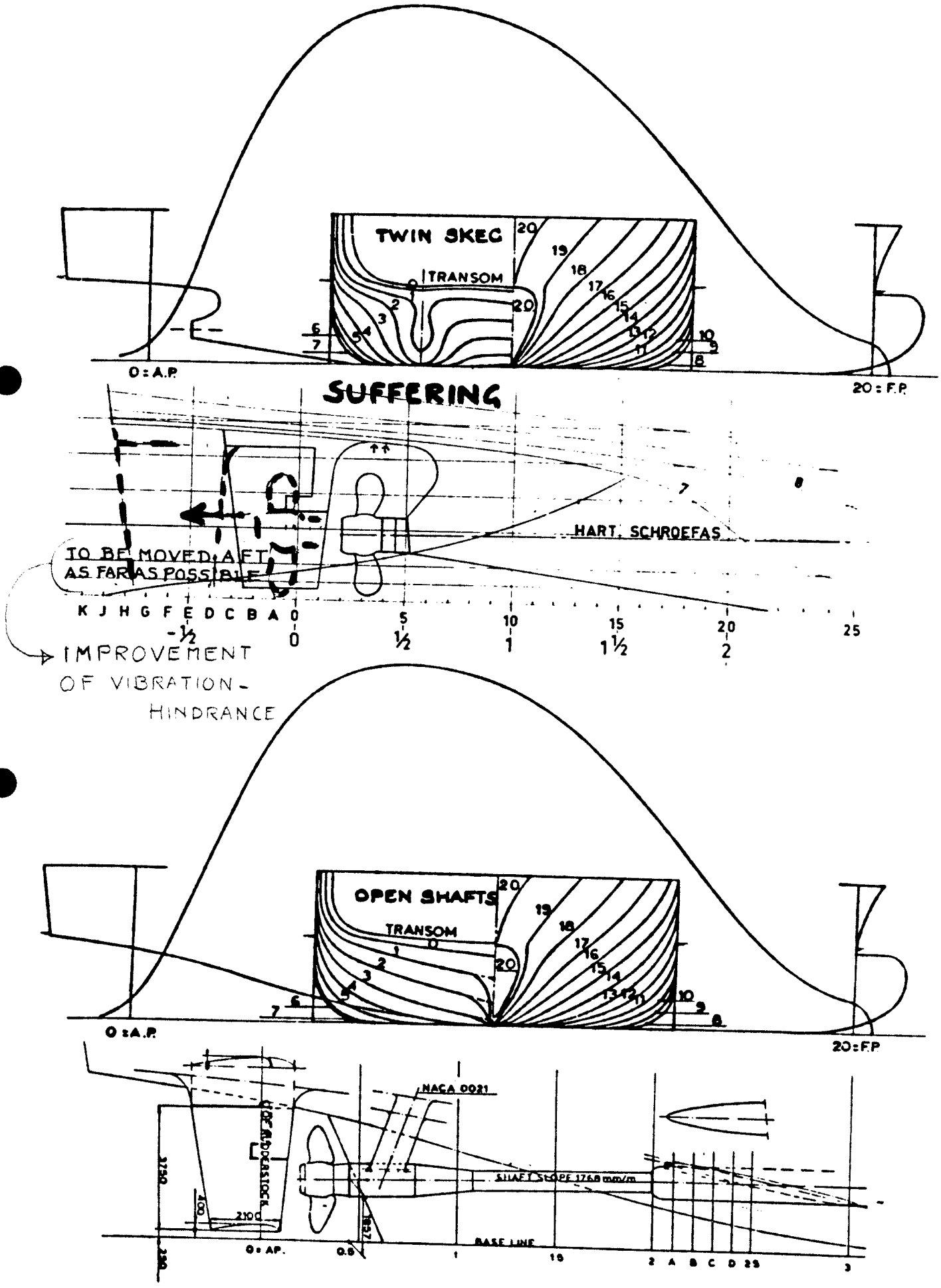
The twin rudder twin skeg ferry, fitted with c.p. propellers will steer badly when slowing down because the rudders have no flow caused by the wake of the twin hull configuration, together with the blanking off action of the c.p. propellers. A third rudder, as skegrudder added to the centre line skeg will improve greatly the steering while slowing down; three rudders to be moved by one steering gear.

Sofar Nedlloyd is in favour of the normal hull form having a normal curve of cross section areas and would prefer the free flow propellers.

In case of the twin skeg ferry boat, she would recommend to locate propellers and rudders as far aft as possible.

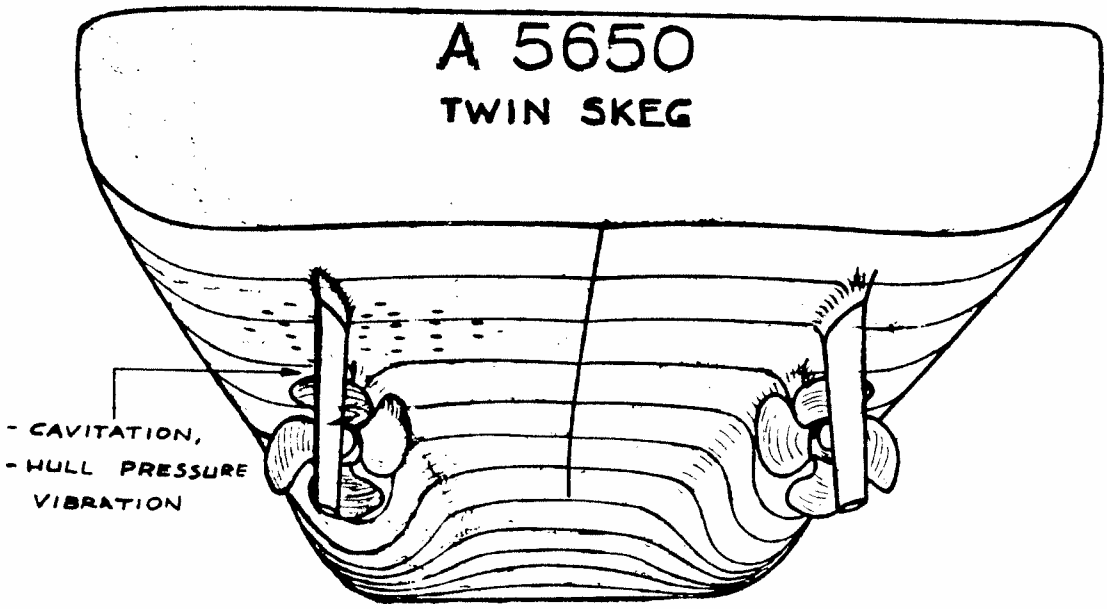
(Figure 45)

FERRY

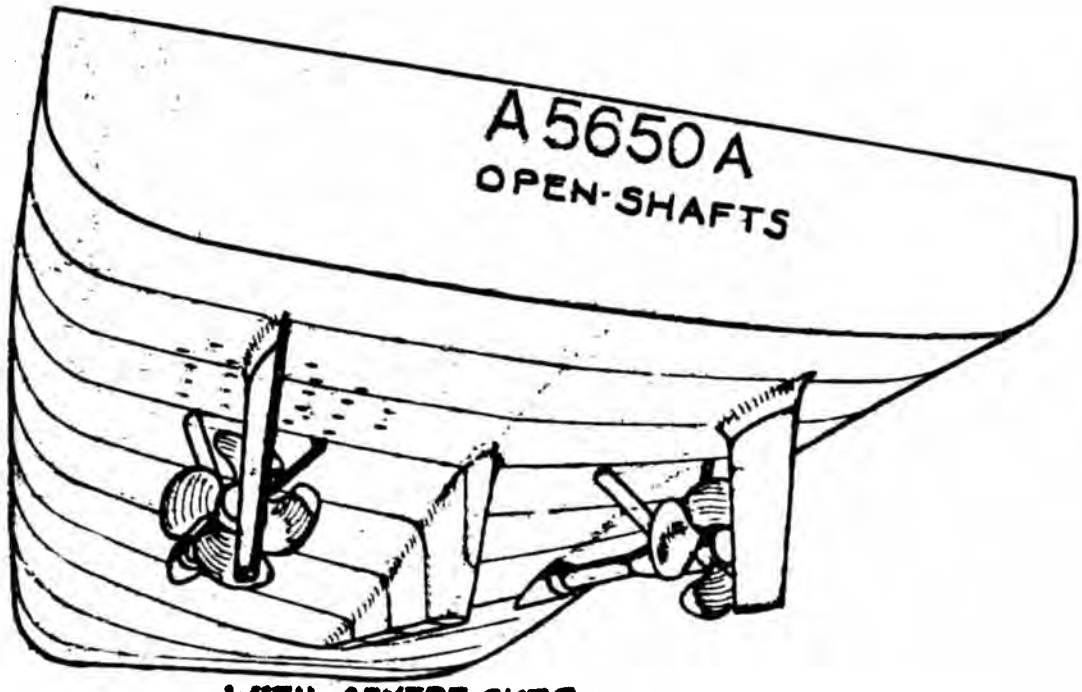


(Fig. 46)

FERRY



NO CENTRELINE SKEG



WITH CENTRE SKEG

[ABT. SAME HP FOR SPEED]

NETHERLANDS SHIP MODEL BASIN WAGENINGEN	REPORT No 03520-1-VT

RESULTS WAKE MEASUREMENTS

TEST No 80-041
SHIP MODEL No 5650
SHIP DRAUGHT ON FP 5.000 m
SHIP DRAUGHT ON AP 5.000 m

SHIP SPEED 22.00 KNOTS

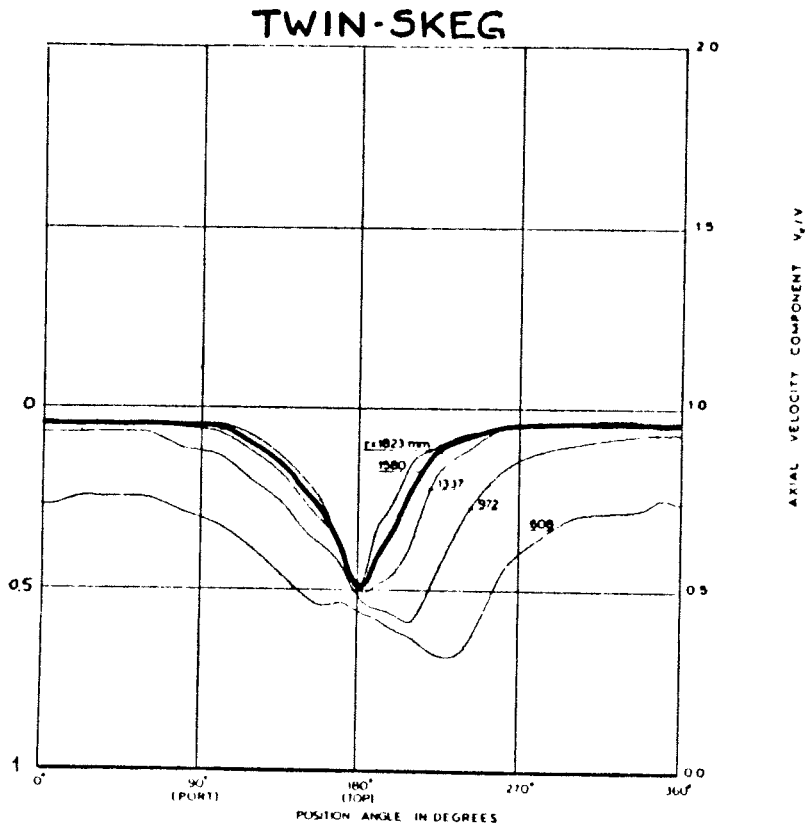


FIG 31 CIRCUMFERENTIAL DISTRIBUTION OF AXIAL VELOCITY COMPONENT

NETHERLANDS SHIP MODEL BASIN WAGENINGEN	REPORT No. 03520-3-VT

RESULTS WAKE MEASUREMENTS

TEST No 80-086
SHIP MODEL No 5650A
SHIP DRAUGHT ON FP 5.000 m
SHIP DRAUGHT ON AP 5.000 m

SHIP SPEED 22.00 KNOTS

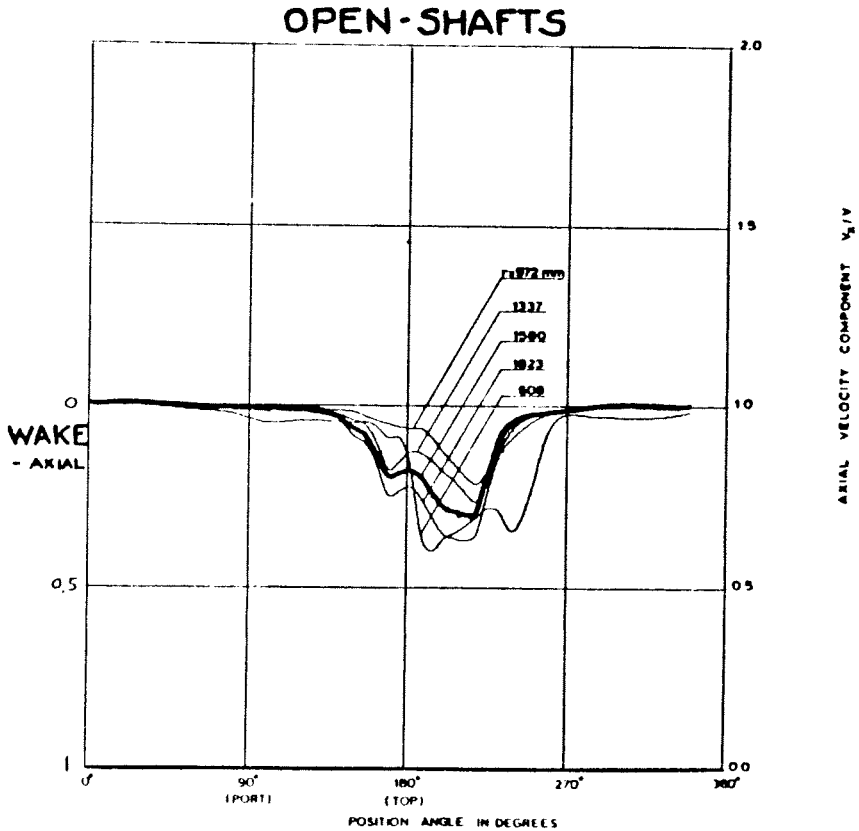


FIG 31 CIRCUMFERENTIAL DISTRIBUTION OF AXIAL VELOCITY COMPONENT

(Figure 47)

12-5

NETHERLANDS SHIP MODEL BASIN WAGENINGEN	REPORT No 03520-1-VT	PAGE 26

RESULTS WAKE MEASUREMENTS

TEST No 80-041
SHIP MODEL No. 5650
SHIP DRAUGHT ON F.P. 5.000 m
SHIP DRAUGHT ON A.P. 5.000 m

SHIP SPEED 22.00 KNOTS

TWIN - SKEG

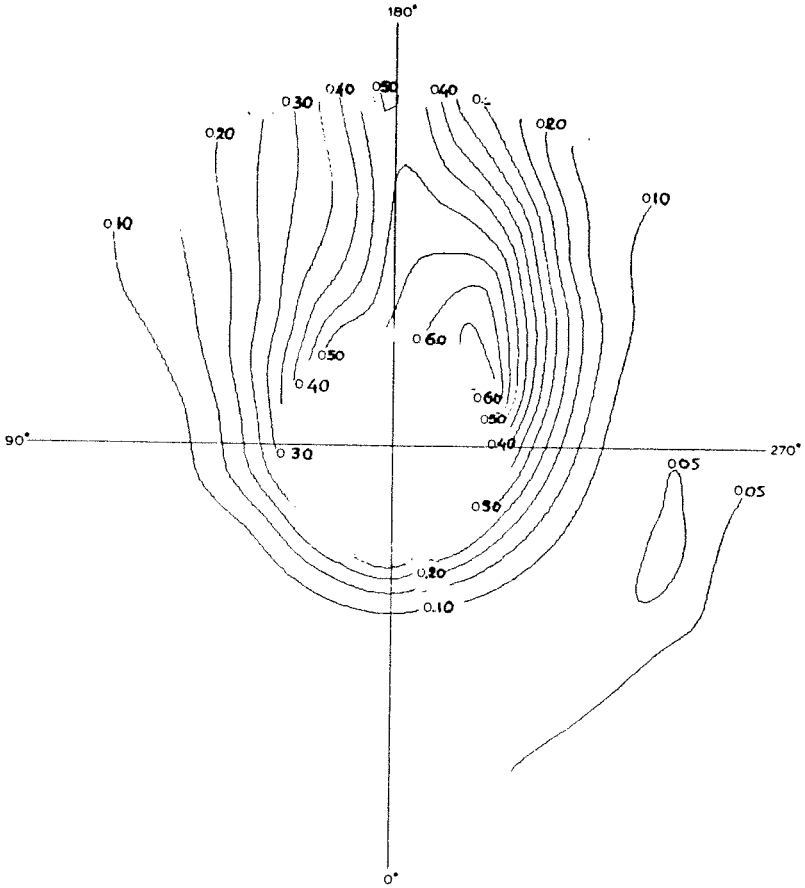


FIG 32. LINES OF EQUAL AXIAL VELOCITY.

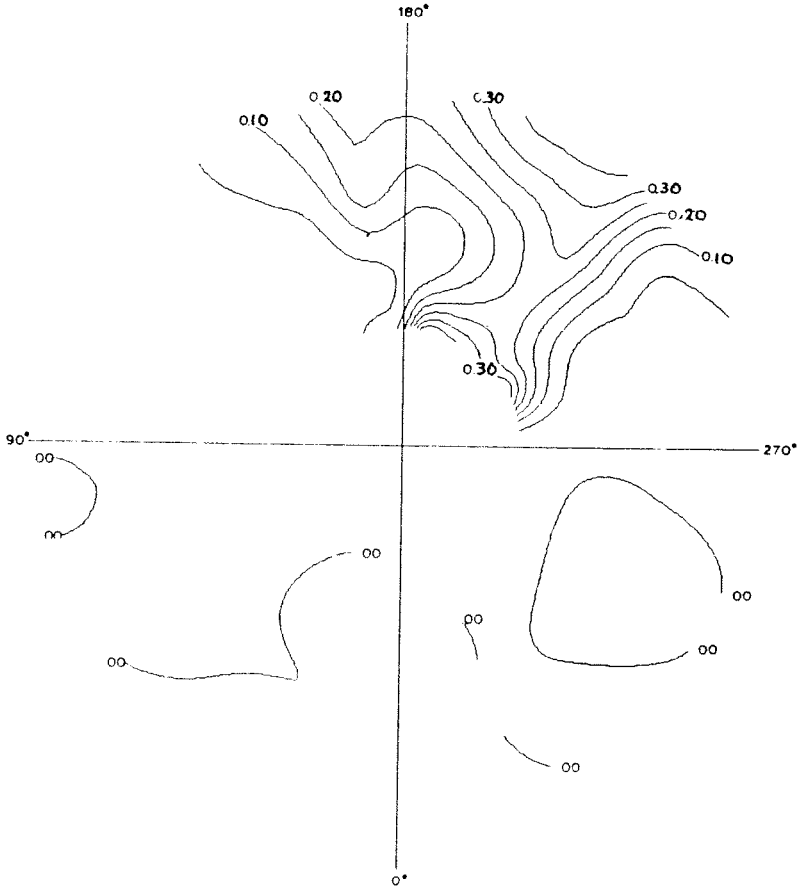
NETHERLANDS SHIP MODEL BASIN WAGENINGEN	REPORT No 03520-3-VT	PAGE 26

RESULTS WAKE MEASUREMENTS

TEST No. 80-086
SHIP MODEL No. 5650A
SHIP DRAUGHT ON F.P. 5.000 m
SHIP DRAUGHT ON A.P. 5.000 m

SHIP SPEED 22.00 KNOTS

OPEN-SHAFTS



(Figure 48)

12-6

FERRY

G = LARGE MODEL

	Scheepsmodel			
	ORIG TWIN - G SKEG		GOA SINGLE SKEG	
V_s (knoop)	P_E (kW)	P_E (%)	P_E (kW)	P_E (%)
18	4031	100	3893	96,6
19	5021	100	4982	99,2
20	6290	100	6369	101,3
21	7878	100	7972	101,2
22	9770	100	9721	99,5
23	11912	100	11642	97,7

K = SMALL MODEL

	Scheepsmodel TWIN SKEG					
	K		KA		KV	
V_s (knoop)	P_E (kW)	P_E (%)	P_E (kW)	P_E (%)	P_E (kW)	P_E (%)
18	-	-	3845	95,4	4099	101,7
19	-	-	4855	96,7	5204	103,6
20	6483	103,1	6097	96,9	6471	102,9
21	7996	101,5	7590	96,3	7984	101,3
22	9819	100,5	9349	95,7	9783	100,1
23	11846	99,4	11300	94,9	11771	98,8

TWIN SKEG

	Scheepsmodel			
	KAD		KAS	
V_s (knoop)	P_E (kW)	P_E (%)	P_E (kW)	P_E (%)
18	3793	94,1	3857	95,7
19	4772	95,0	4854	96,7
20	5982	95,1	6163	98,0
21	7497	95,2	7696	97,7
22	9251	94,7	9409	96,3
23	11084	93,0	11450	96,1

V_s (knoop)	G/5410L+R TWIN SKEG P_S (kW)	G/5410L+R TWIN SKEG P_S (%)	GOA/5410L+R P_S (kW)	GOA/5410L+R P_S (%)
18	-	-	-	-
19	7968	97,4	8397	102,6
20	10111	97,6	10810	104,4
21	12859	98,0	13622	103,8
22	16195	98,2	16704	101,3
23	20024	98,4	20102	98,8

SINGLE SKEG SHAFTS + BRACKETS

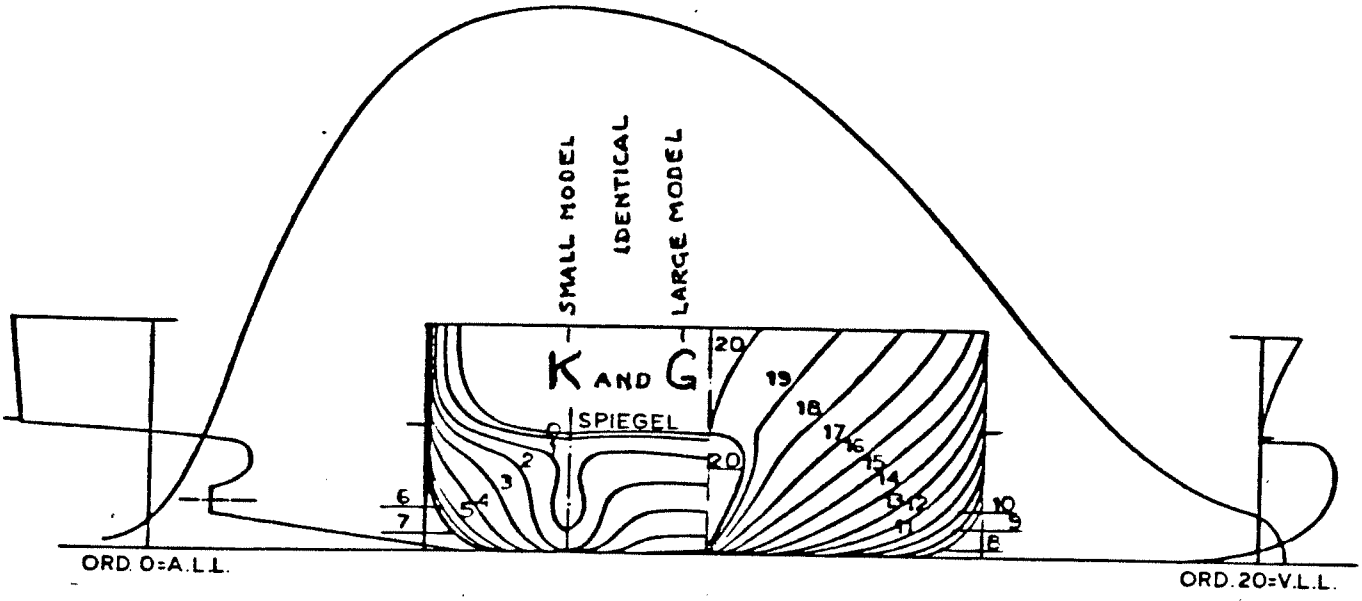
V_s (knoop)	G/5422L+R P_S (kW)	G/5422L+R P_S (%)	G/5423L+R P_S (kW)	G/5423L+R P_S (%)	G/5424L+R TWIN SKEG P_S (kW)	G/5424L+R TWIN SKEG P_S (%)
18	6520	100	6224	95,5	5973	91,6
19	8183	100	7818	95,5	7501	91,7
20	10355	100	9899	95,6	9489	91,6
21	13127	100	12555	95,6	12021	91,6
22	16485	100	15774	95,7	15088	91,5
23	20340	100	19473	95,7	18615	91,5

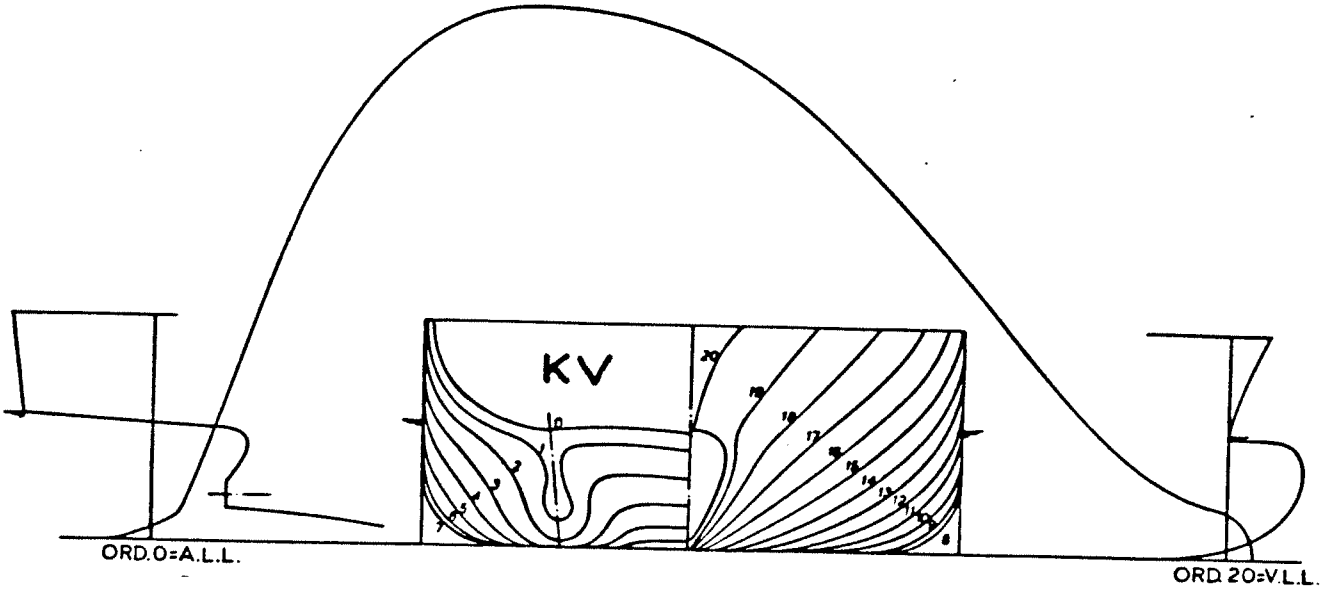
V_s (knoop)	GOA/5481L+R P_S (kW)	GOA/5481L+R P_S (%)	GOA/5482L+R P_S (kW)	GOA/5482L+R P_S (%)	GOA/5483L+R P_S (kW)	GOA/5483L+R P_S (%)
18	6556	100,6	6284	96,4	6075	93,2
19	8480	103,6	8126	99,3	7845	95,9
20	10982	106,1	10519	101,6	10140	97,9
21	13918	106,0	13330	101,5	12829	97,7
22	17155	104,0	16433	99,7	15790	95,8
23	20743	102,0	19879	97,7	19076	93,8

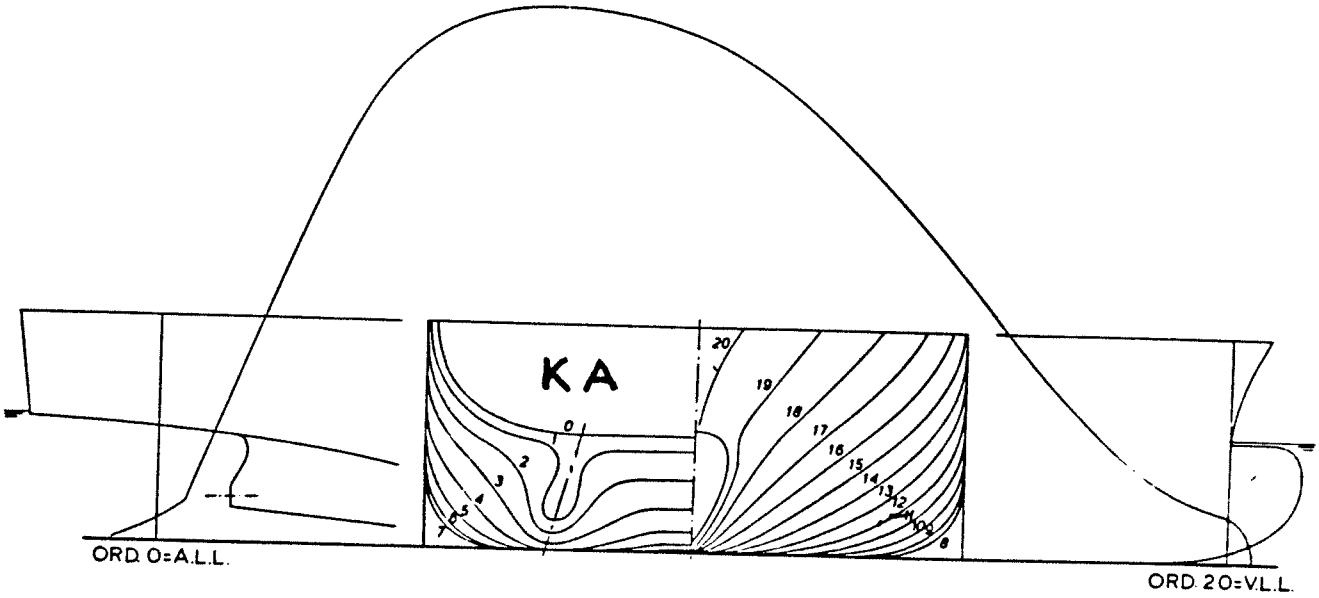
SINGLE SKEG SHAFTS + BRACKETS

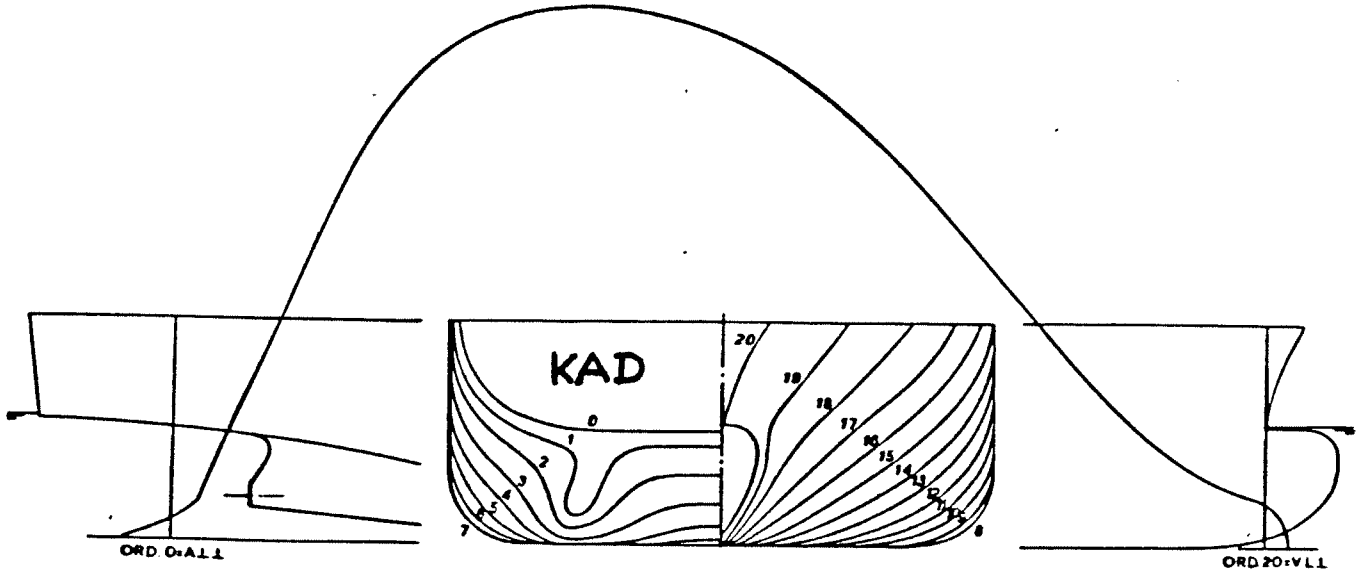
V_s (knoop)	K/5163L+R P_S (kW)	K/5163L+R P_S (%)	KA/5163L+R P_S (kW)	KA/5163L+R P_S (%)	KV/5163L+R TWIN SKEG P_S (kW)	KV/5163L+R TWIN SKEG P_S (%)
18	-	-	6668	102,3	7056	108,2
19	-	-	8484	103,7	9049	110,6
20	11504	111,1	10751	103,8	11359	109,7
21	14356	109,4	13516	103,0	14159	107,9
22	17858	108,3	16821	102,0	17538	106,4
23	21796	107,2	20516	100,9	21301	104,7

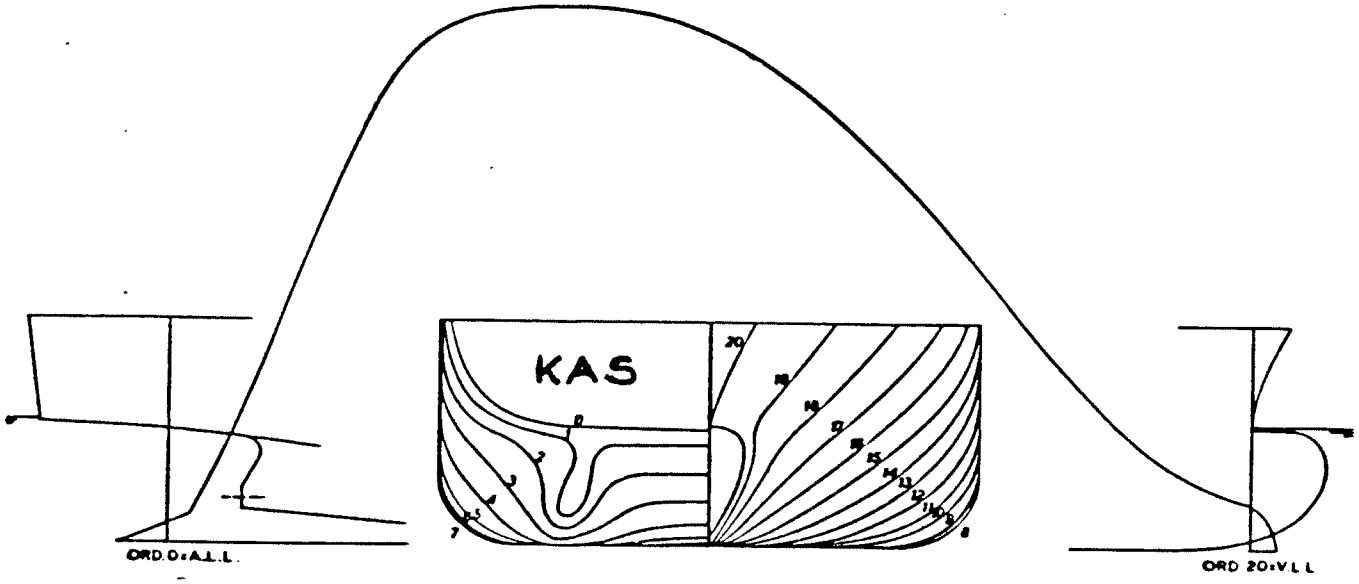
V_s (knoop	KAD/5163L+R P_S (kW)	KAD/5163L+R P_S (%)	KAD/5740L+R P_S (kW)	KAD/5740L+R P_S (%)	KAS/5163L+R TWIN SKEG P_S (kW)	KAS/5163L+R TWIN SKEG P_S (%)
18	6563	100,7	6771	103,8	6779	104,0
19	8323	101,7	8562	104,6	8598	105,1
20	10532	101,7	10806	104,4	11035	106,6
21	13350	101,7	13663	104,1	13930	106,1
22	16658	101,0	17008	103,2	17194	104,3
23	20122	98,9	20499	100,8	21146	104,0

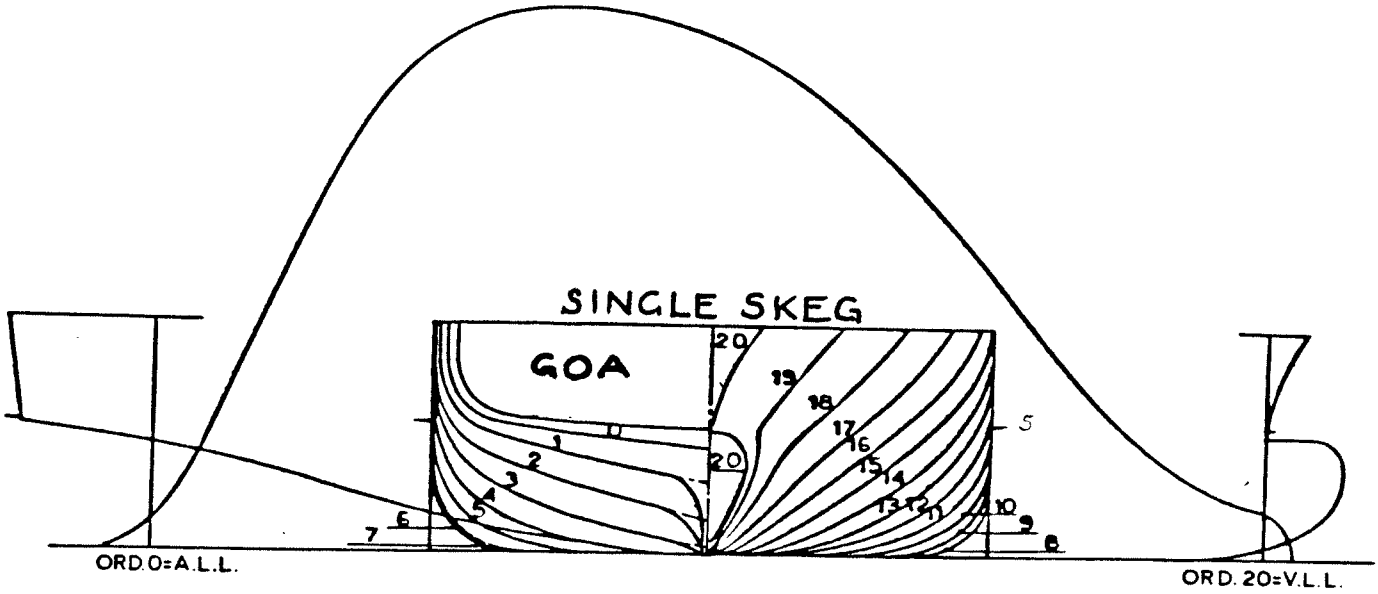

SMALL MODEL
IDENTICAL
LARGE MODEL
K AND G
SPIEGEL
ORD. 0=A.L.L.
ORD. 20=V.L.L.

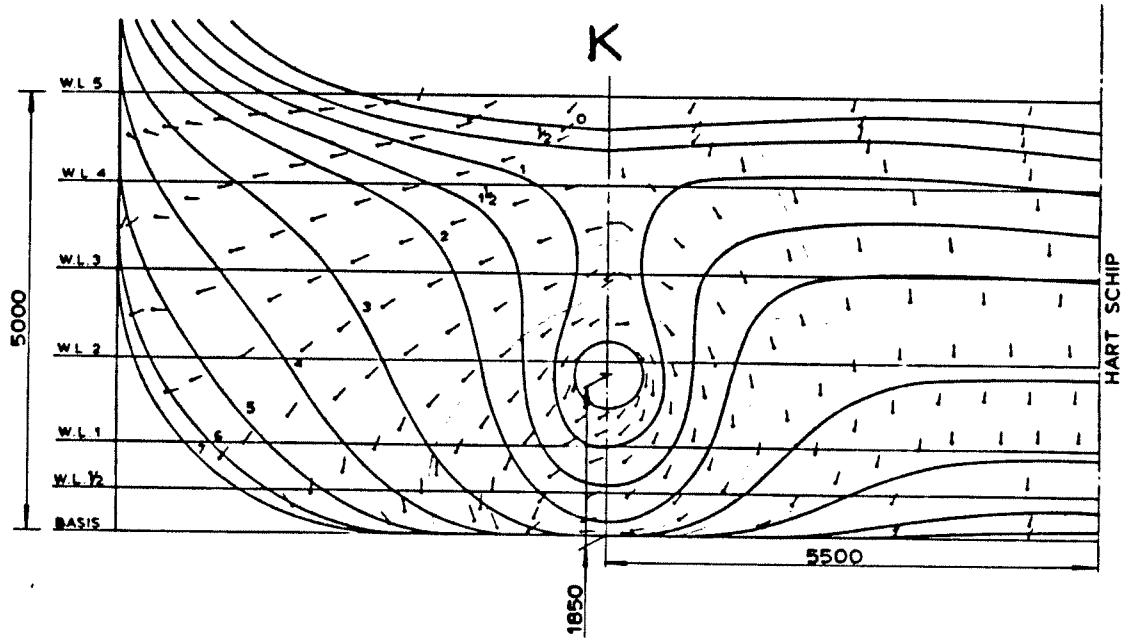

KV
ORD. 0=A.L.L.
ORD. 20=V.L.L.

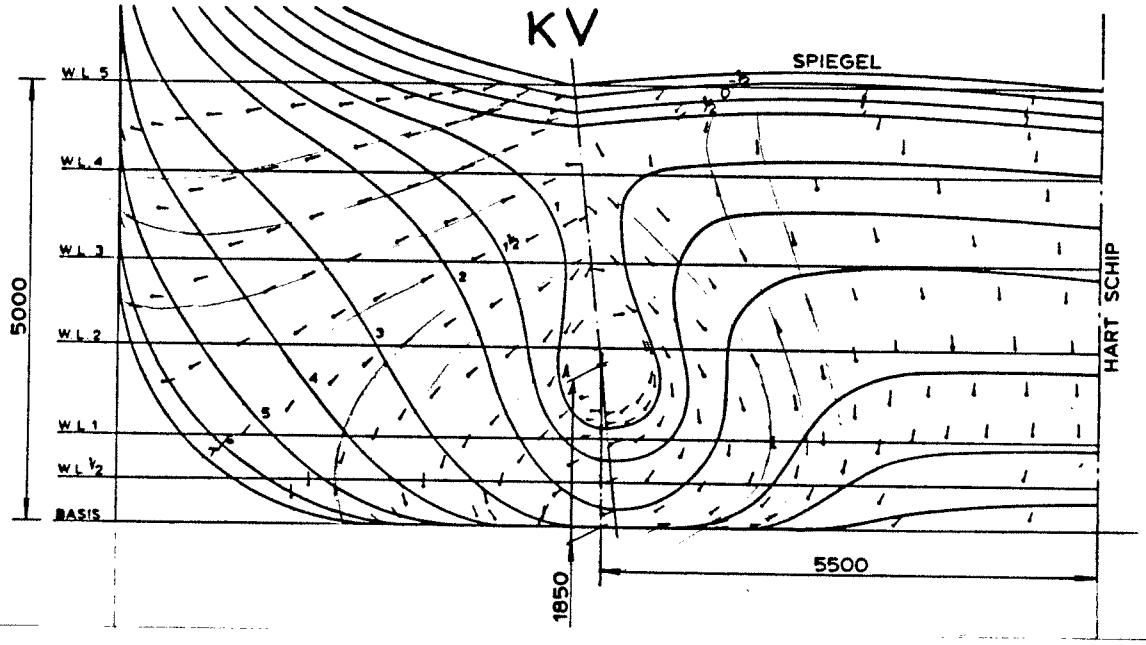

KA
ORD. 0=A.L.L.
ORD. 20=V.L.L.

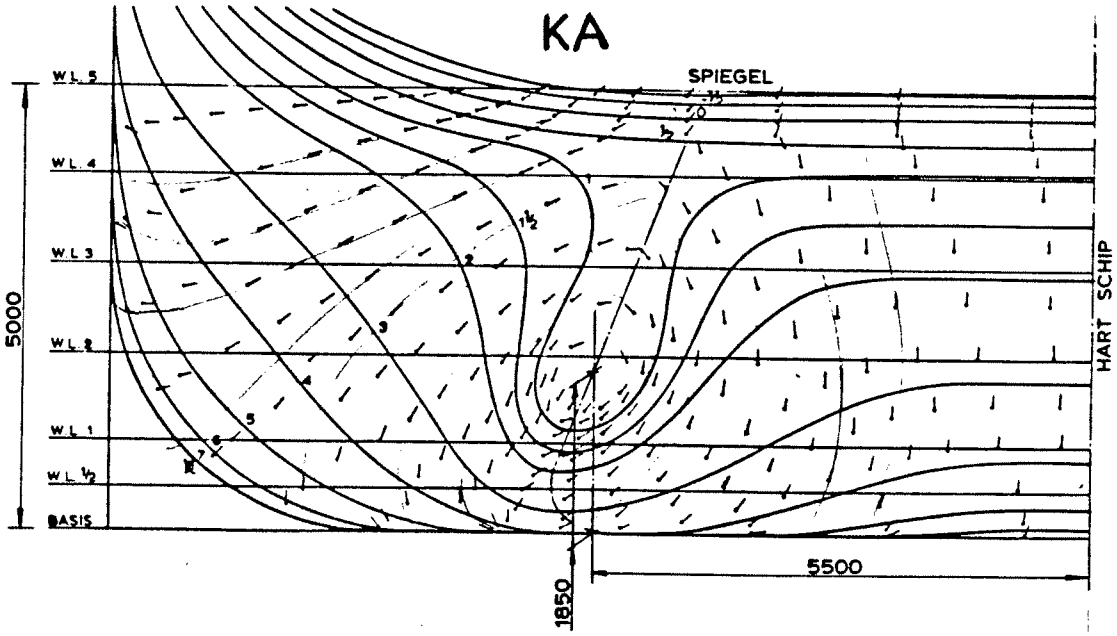

KAD
ORD. 0=A.L.L.
ORD. 20=V.L.L.


KAS
ORD. 0=A.L.L.
ORD. 20=V.L.L.


SINGLE SKEG
GOA
ORD. 0=A.L.L.
ORD. 20=V.L.L.


K
W.L. 5
W.L. 4
W.L. 3
W.L. 2
W.L. 1
W.L. ½
BASIS
5000
5500
1850
HART SCHIP

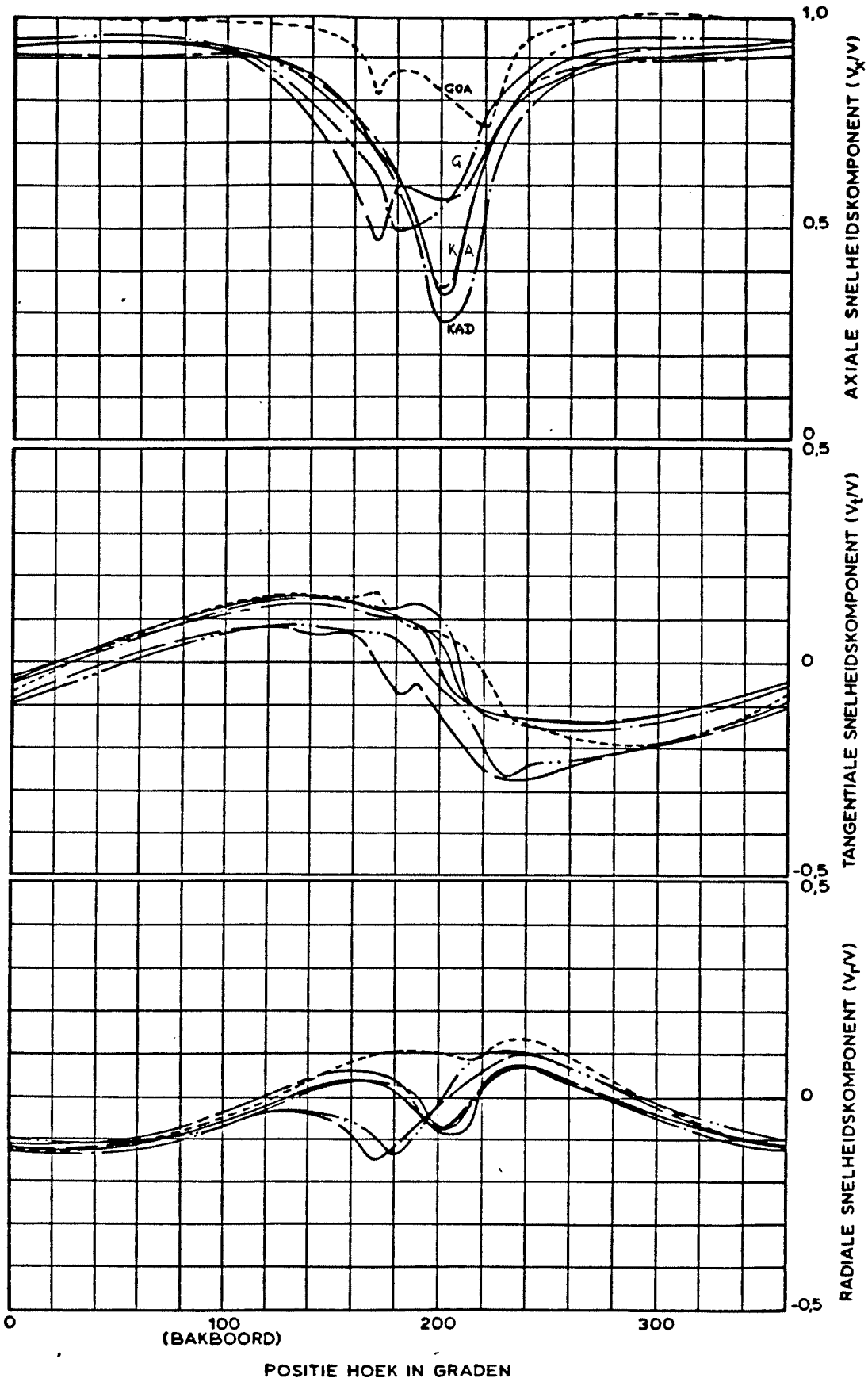

KV
SPIEGEL
W.L. 5
W.L. 4
W.L. 3
W.L. 2
W.L. 1
W.L. ½
BASIS
5000
5500
1850
HART SCHIP


KA
SPIEGEL
W.L. 5
W.L. 4
W.L. 3
W.L. 2
W.L. 1
W.L. ½
BASIS
5000
5500
1850
HART SCHIP

12-12

SCHEEPSMODEL	LIJNTYPE	
KA	———	TWIN
KV	— — — —	"
KAD	—·—·—	"

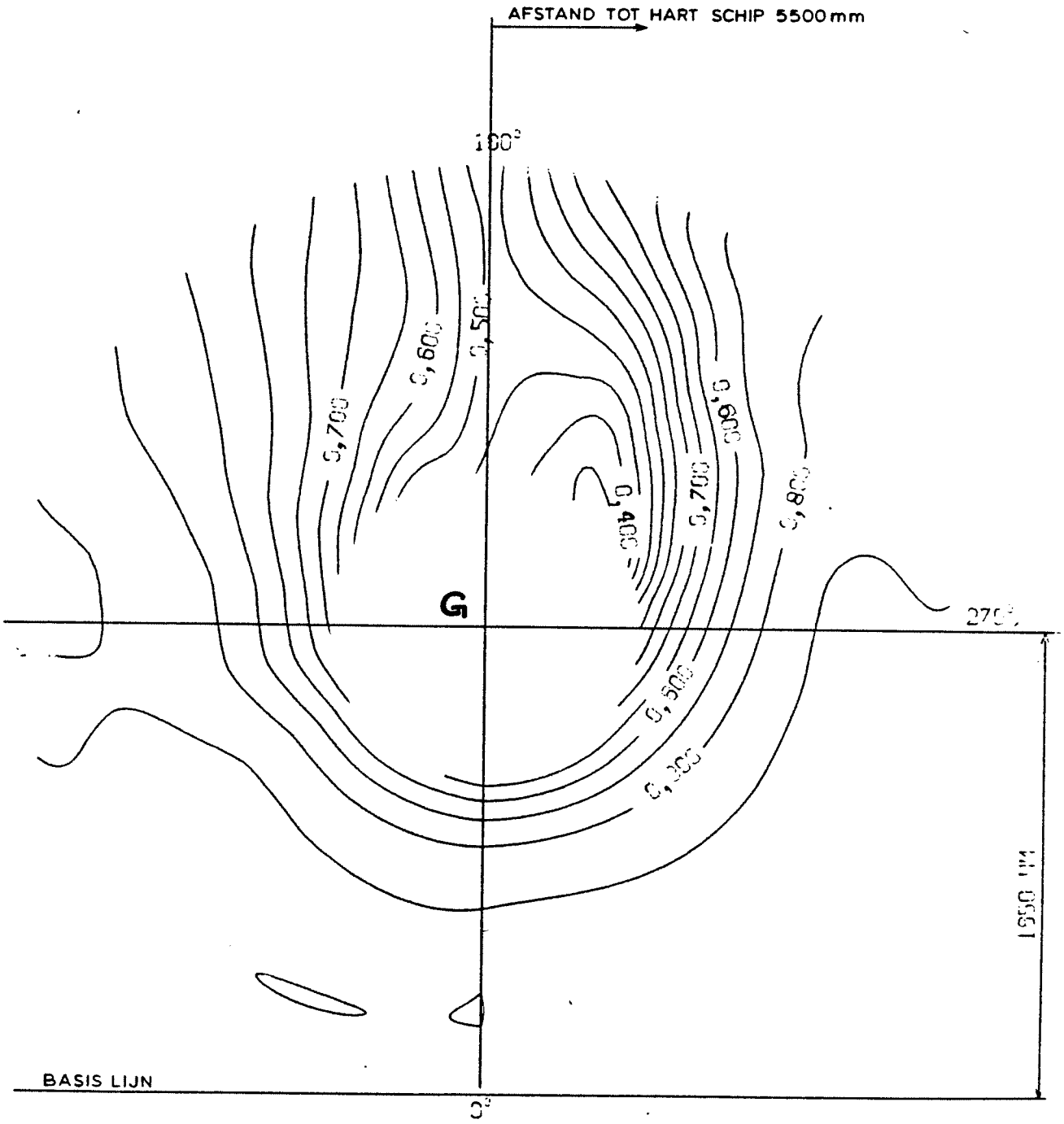
SCHEEPSMODEL	LIJNTYPE	
KAS	— — — —	TWIN
G	—··—··—	TWIN
GOA	- - - - - - - -	SINGLE SKEG



Vergelijking van de volgstroommetingen op 0,8R.

12.13

Proef No. 80-041
Scheepsdiepgang 5,00 m gelijklastig
Scheepssnelheid 22,00 knoop



Lijnen van gelijke axiale snelheid (V_x/V), gemeten in het bakboord schroefvlak van scheepsmodel G.

Proef No. 39398
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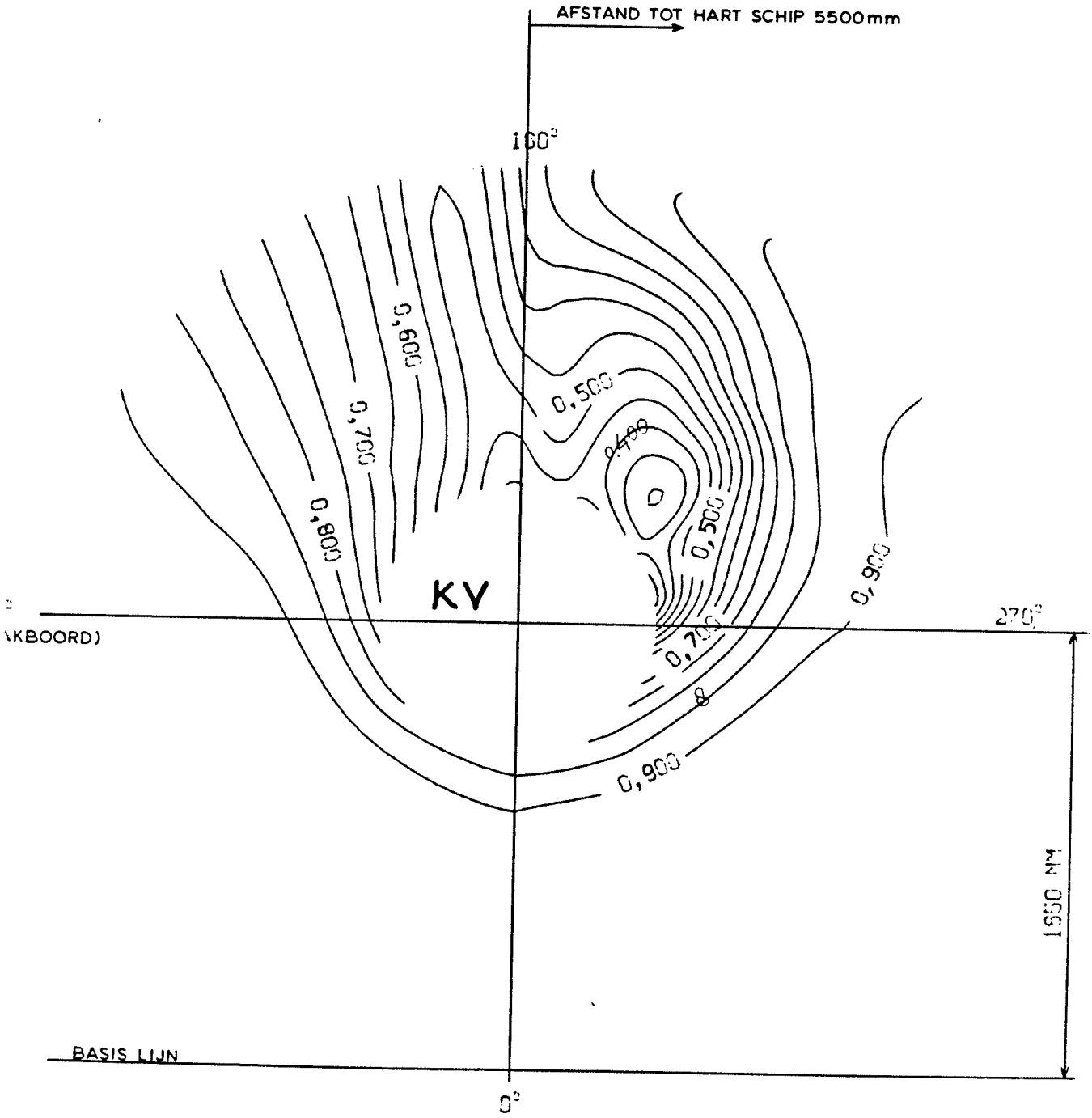
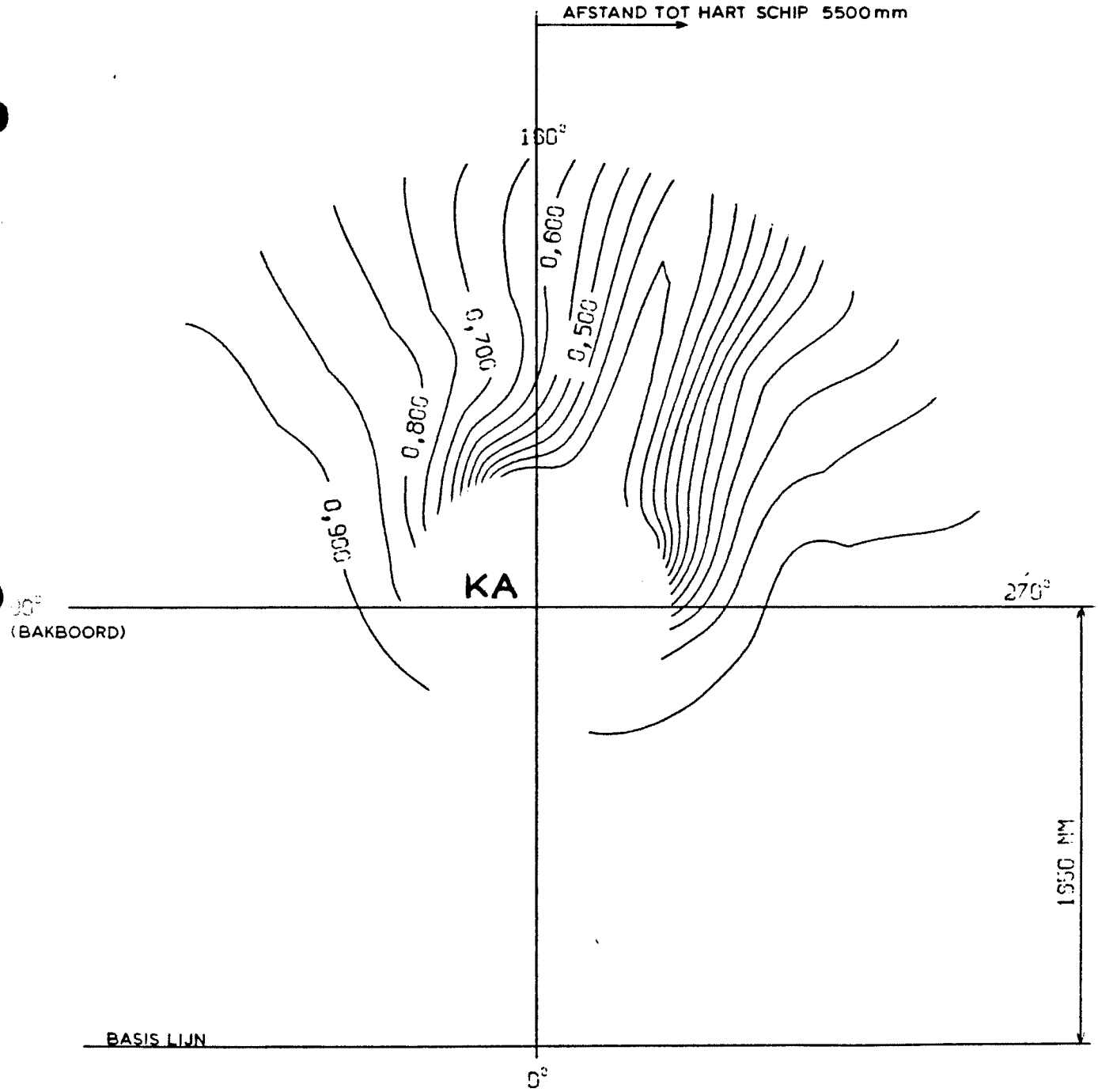
Scheepssnelheid 21,30 knoop



Lijnen van gelijke axiale snelheid (V_x/V), gemeten in het bakboord schroefvlak van scheepsmodel KV.

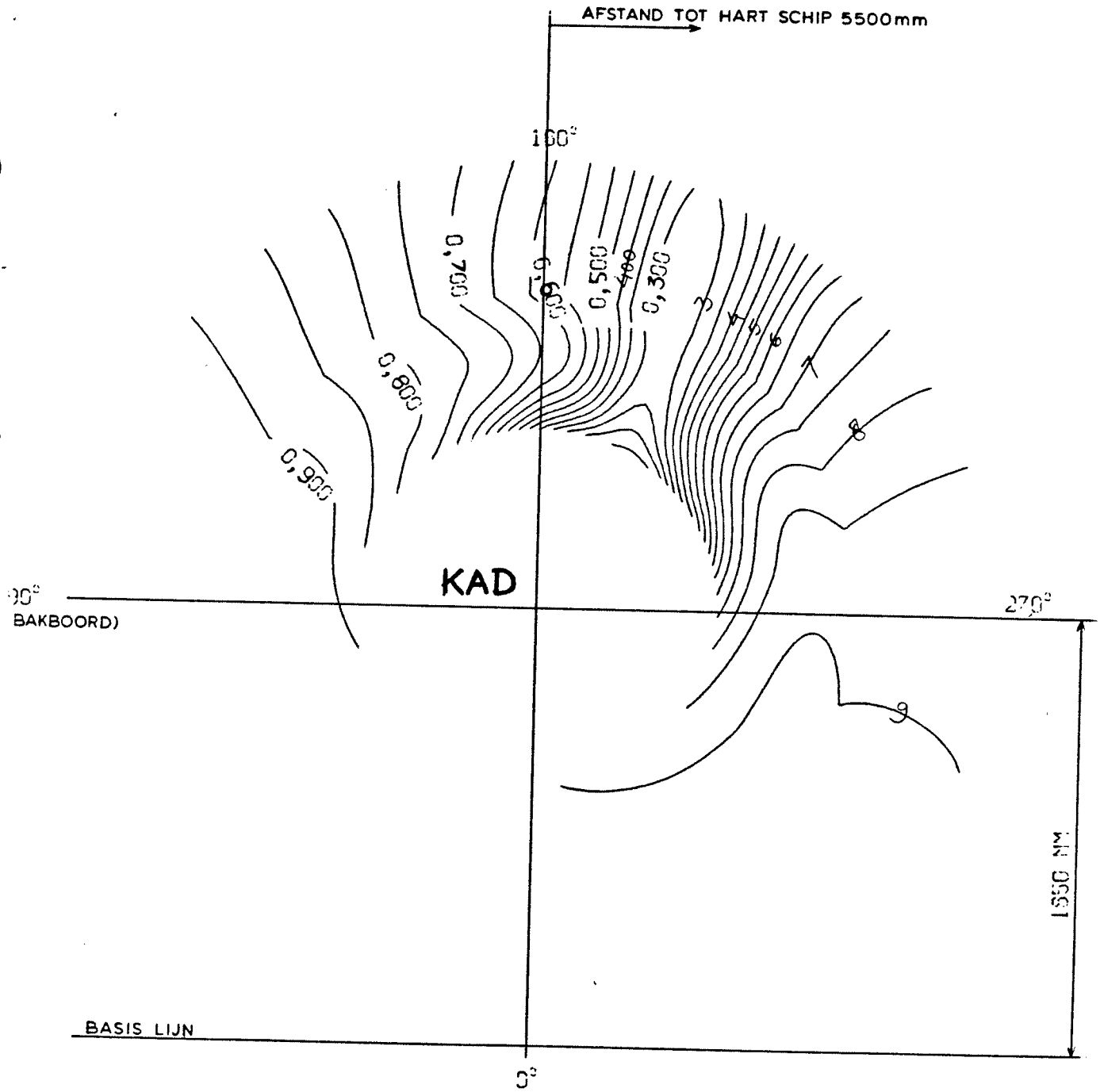
Proef No. 39395
Scheepsdiepgang 5,00 m gelijklastig
Scheepssnelheid 21,30 knoop



Lijnen van gelijke axiale snelheid (V_x/V), gemeten in het bakboord schroefvlak van scheepsmodel KA.

12-16

Proef No. 41756
Scheepsdiepgang 5,00 m gelijklastig
Scheepssnelheid 21,30 knoop

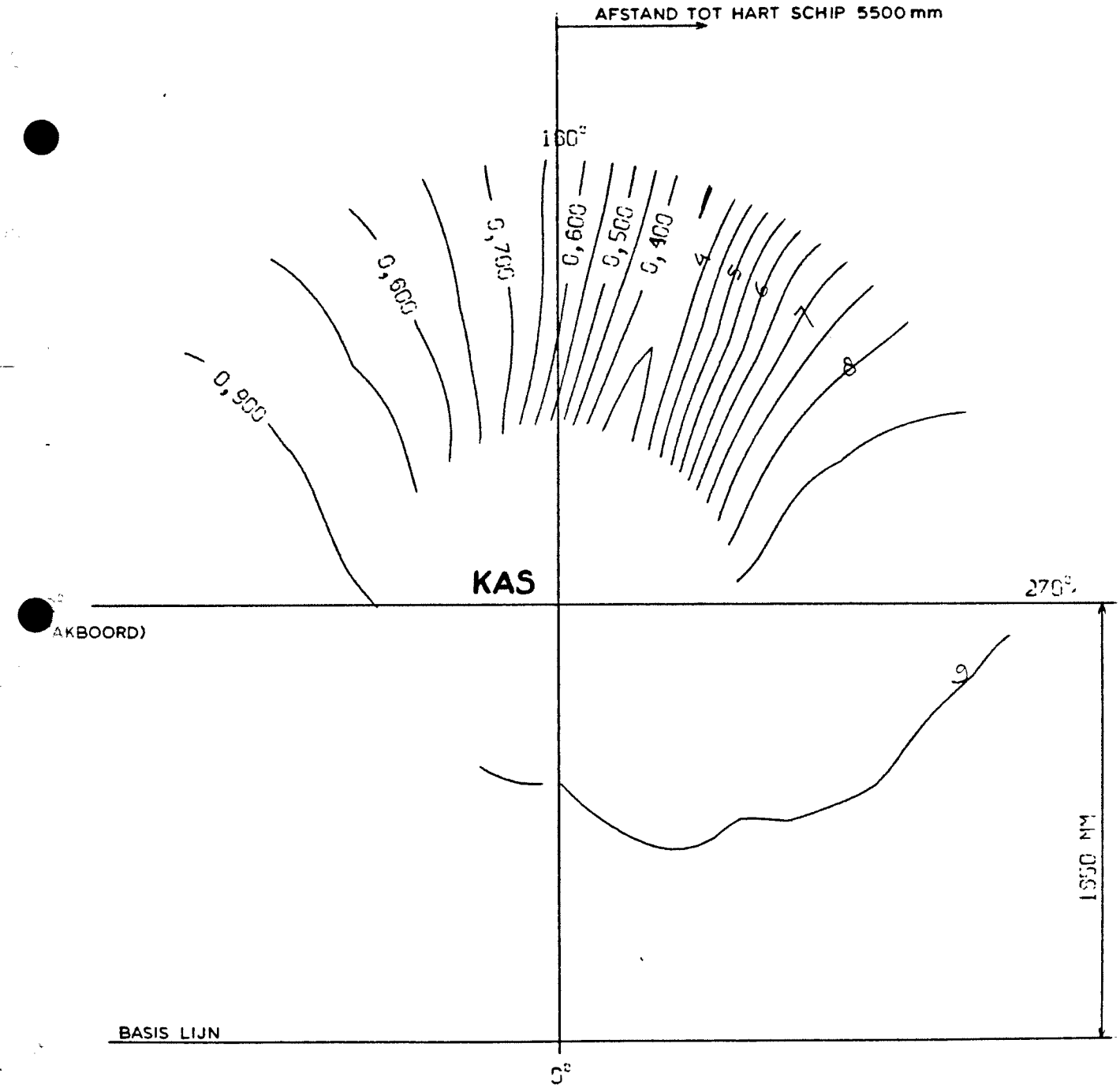


Lijnen van gelijke axiale snelheid (V_x/V), gemeten in het bakboord schroefvlak van scheepsmodel KAD.

MARIN
Maritiem Research Instituut Nederland

12-17

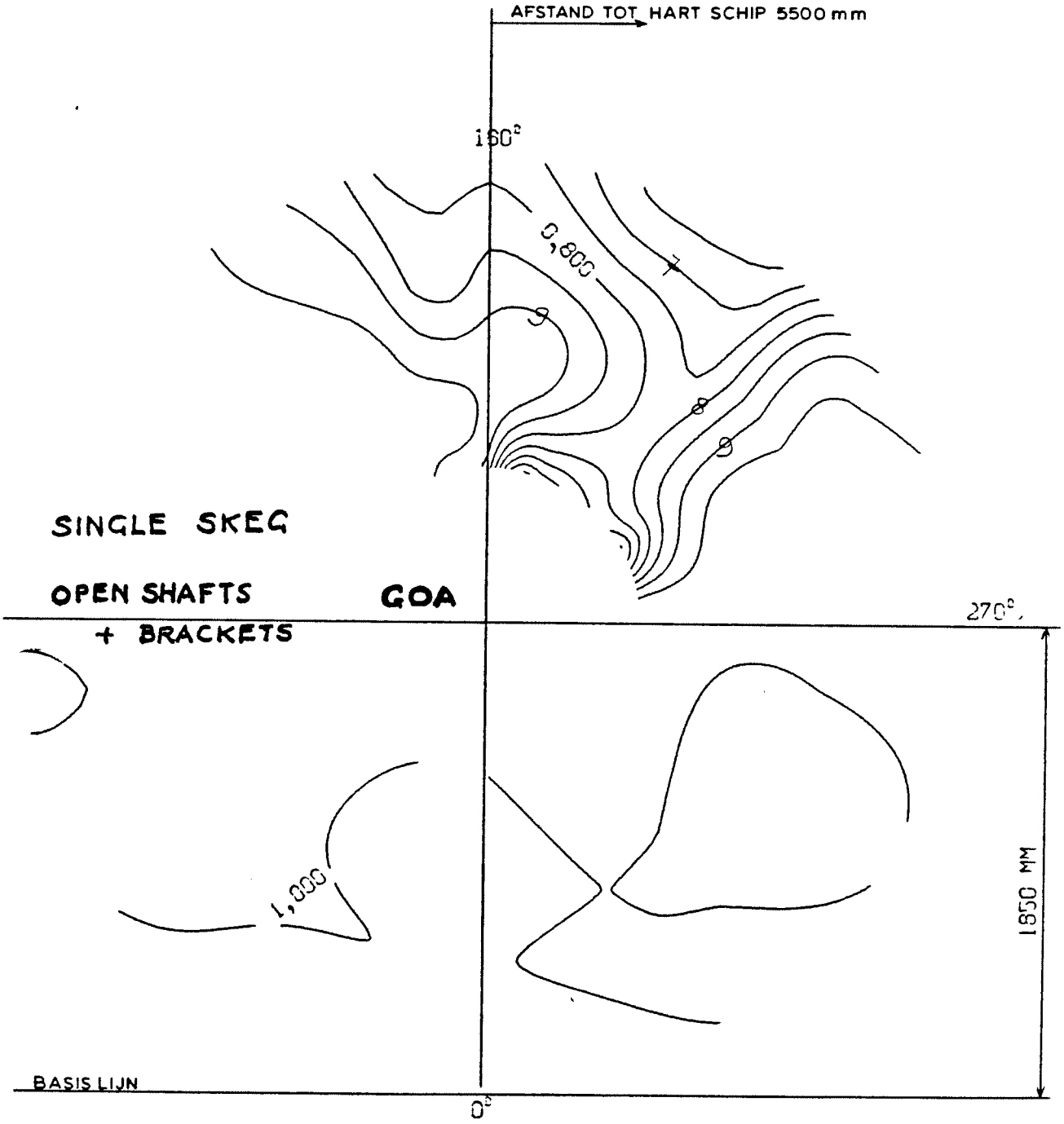
Proef No. 41805
Scheepsdiepgang 5,00 m gelijklastig
Scheepssnelheid 21,30 knoop



Lijnen van gelijke axiale snelheid (V_x/V), gemeten in het bakboord schroefvlak van scheepsmodel KAS.

12.18

Proef No. 80-086
Scheepsdiepgang 5,00 m gelijklastig
Scheepssnelheid 22,00 knoop



Lijnen van gelijke axiale snelheid (V_x/V), gemeten in het bakboord schroefvlak van scheepsmodel GOA.

DEC. 1983 **FERRY** REPORT 45477-1-VT MARIN

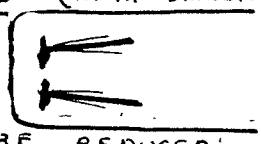
LIGTELIJN + JONK TWIN SKEG VERSUS SINGLE SKEG + OPEN SHAFTS IN BRACKETS

CONCLUSION :
- TWIN-SKEG-VESSELS ARE HAVING A LARGER AXIAL-WAKEPEAK, THEREFORE THEY MIGHT SUFFER FROM VIBRATIONS.
- MEANWHILE – BECAUSE OF THE WAKE, THE HULL-EFF. IS ALSO LARGER.
  THIS MEANS THAT PROP. PERFORMANCE IS RATHER GOOD.
- SHP FOR A GIVEN SPEED IS ABOUT THE SAME AS FOR THE NORMAL SINGLE SKEG FERRY WITH OPEN SHAFTS AND BRACKETS.
- SOMETIMES THE TWIN SKEG MODEL WILL BE SUPERIOR
- SOMETIMES THE SINGLE SKEG MODEL

MIND : SCALE EFFECT ON WAKE WILL REDUCE THE SUPERIORITY OF TWIN SKEG AGAINST SINGLE SKEG FOR THE SHIP 1:1

N.B. RECENT TESTS HAVE SHOWN THAT THE TWIN SKEG FERRY MODEL (OR SIMILAR FLAT PRAM HULLS HEAVY LIFT) CAN BE GREATLY IMPROVED BY APPLICATION OF :

CONVERGING SHAFTS (IN AFT DIRECTION)



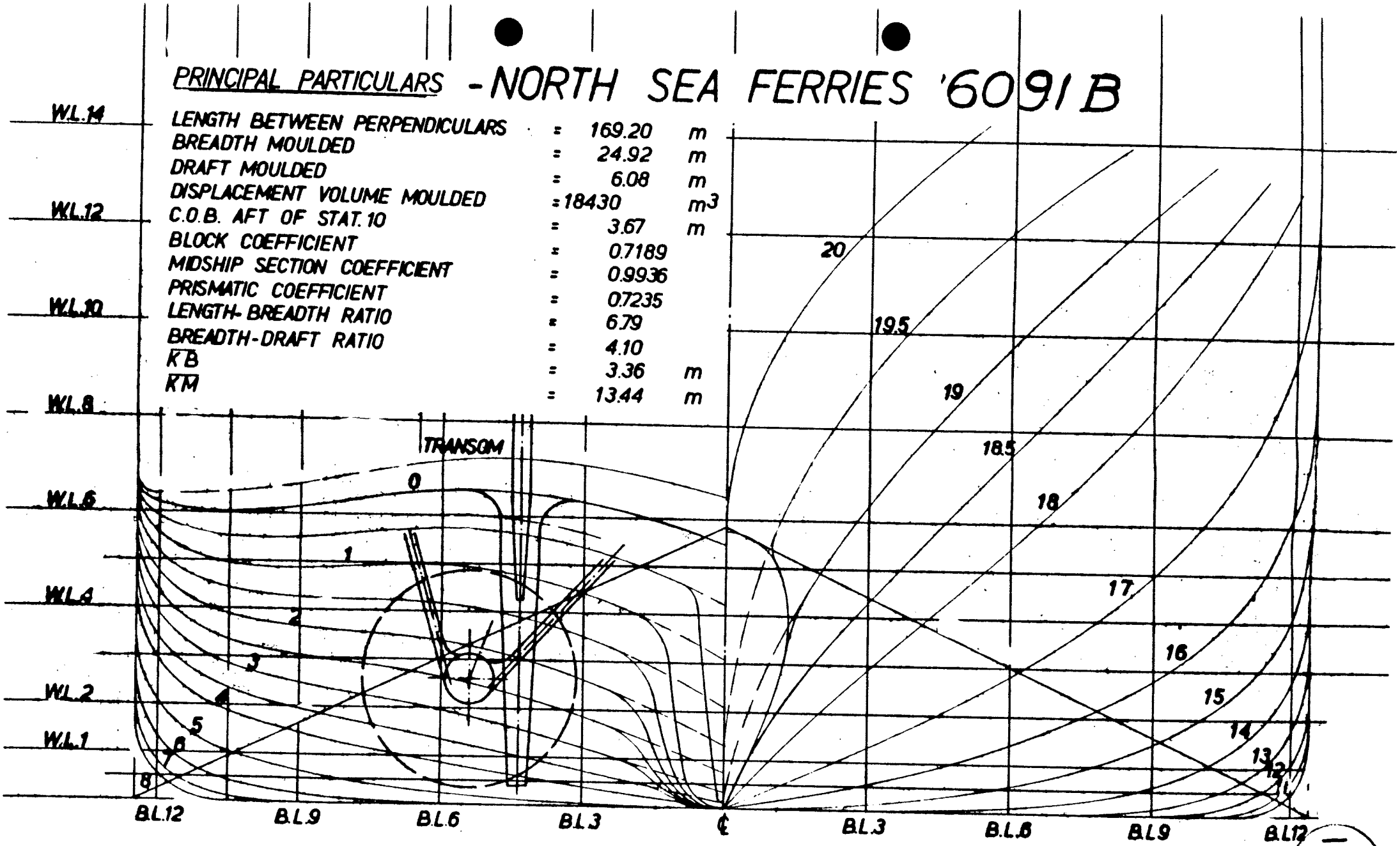
P_E AND P_D WILL BE SMALLER AND AXIAL WAKE PEAK WILL BE REDUCED; TANGENTIAL VELOCITY COMPONENTS ALSO WILL BE SMALLER

- THE "PROPELLER BOSSINGS" ARE MORE IN LINE WITH THE FLOW AROUND AFTERBODY.
  VIBRATION AND NOISE MIGHT COME DOWN TO THE LEVEL OF THE SINGLE SKEG VESSELS WITH OPEN SHAFTS AND BRACKETS WITH "FREE-FLOW" PROPELLERS
- CONTRA ROTATING BOSSINGS OFTEN GIVE IMPROVEMENT IN PROPULSIVE PERFORMANCE OF THE MODEL — HOWEVER — LATER ON THE SHIP, THE SHIPOWNER WILL BE CONFRONTED WITH STRONGER PROPELLER EXCITED VIBRATIONS

C.P. PROPELLERS TO HAVE STRONGLY REDUCED TIP LOAD (SMALLER PITCH AND CAMBER) AND TO "HIGLY-SKEWED" (ONE DIRECTION OF ROTATION)

PRINCIPAL PARTICULARS - NORTH SEA FERRIES '6091 B

LENGTH BETWEEN PERPENDICULARS	=	169.20	m
BREADTH MOULDED	=	24.92	m
DRAFT MOULDED	=	6.08	m
DISPLACEMENT VOLUME MOULDED	=	18430	m^3
C.O.B. AFT OF STAT. 10	=	3.67	m
BLOCK COEFFICIENT	=	0.7189	
MIDSHIP SECTION COEFFICIENT	=	0.9936	
PRISMATIC COEFFICIENT	=	0.7235	
LENGTH-BREADTH RATIO	=	6.79	
BREADTH-DRAFT RATIO	=	4.10	
KB	=	3.36	m
KM	=	13.44	m



"DECEMBER LINES"

"Tunnel" alternative

SCALE 1: 100

14 - 12 - 82

12-20

N.S.M.B. SHIP MODEL No. 6091 B

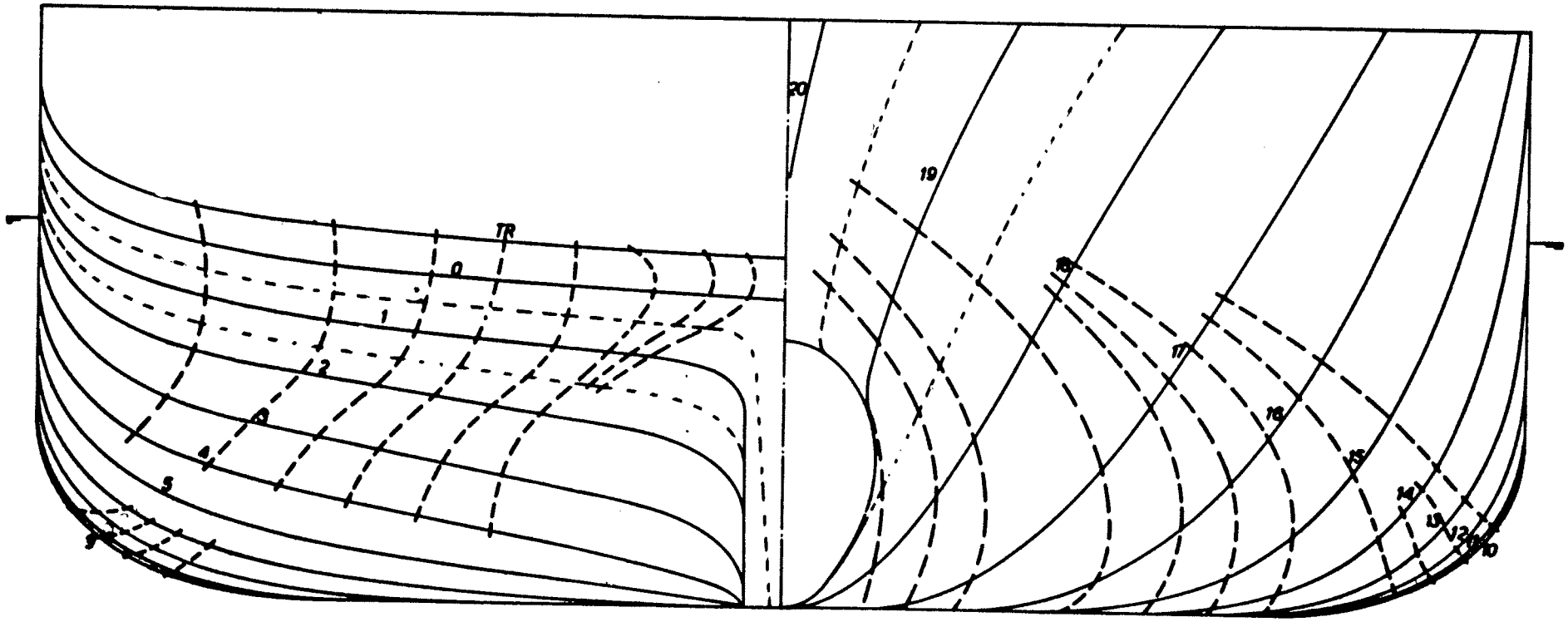
ORDER No P 04416

HEAVY-LIFT- SHIP MODEL No. 6061

STREAMLINES

DRAFT 7.00 m **SPEED 15.2 KNOTS**

LENGTH OVERALL	:	144.00 m
LENGTH BETWEEN PERPENDICULARS	:	134.00 m
BREADTH MOULDED	:	28.00 m
DRAFT DESIGN	:	7.00 m
DISPLACEMENT WEIGHT IN SEA WATER	:	19004 tons
C.O.B. AFT OF F.P.	:	70.19 m
BLOCK COEFFICIENT	:	0.7059
MIDSHIP SECTION COEFFICIENT (C_M)	:	0.9894
PRISMATIC COEFFICIENT (C_P)	:	0.7135
LENGTH-BREADTH RATIO	:	4.786
BREADTH-DRAFT RATIO	:	4.00



DRAWING No. M 6061_26

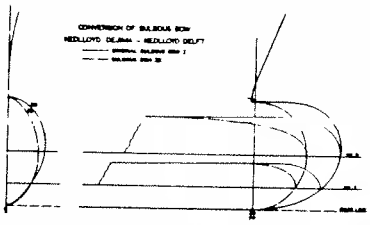
13

13-1

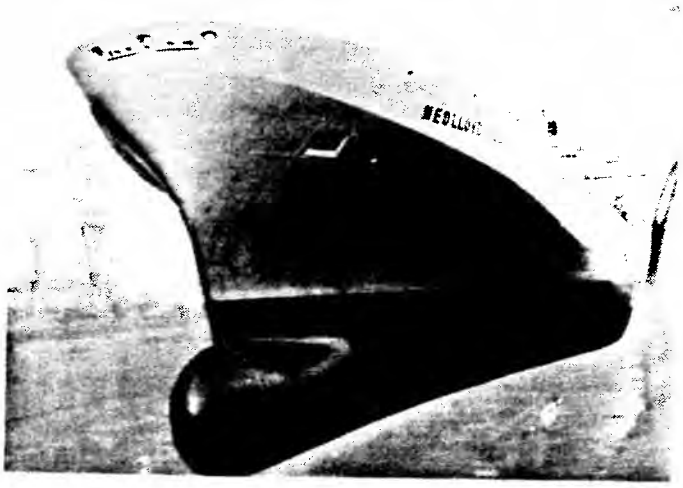
LARGE FAST CONTR 1-2-3 S

FUEL CONSUMPTION
STEAM / DIESEL

Bulbous bow of Nedlloyd Delft.



Original (1971)
27 knots bulb.



Reconverted bulb (1981)
for reduced service speed
of around 22 knots.

FUEL CONSUMPTION IN SERVICE.

For marine engineers and naval architects it is an exceptional and rare opportunity if results from real practice are available for comparison.

A number of some eleven ships in the same service could be compared, viz.:-

Ship designation	Propellers	Main mover types
- K	twin screw	steam turbine
- Nedlloyd Dejima/Delft	twin screw	steam turbine
- Nedlloyd Houtman/Hoorn	twin screw	diesel
- BP/BS	twin screw	diesel
- N T S J	triple screw	diesel

From compilation of a number of voyages it appeared that at 20½ knots service speed, average consumptions per day were as follows:-

Main mover type	Props.	Ship designations	Average cons.
Steam turbine	2-screw	Nedlloyd Delft/Dejima	196-202 tons/day
Diesel	2-screw	Nedlloyd Hoorn	112 tons/day
Diesel	3-screw	ships S, SL, T.	124-130 tons/day

(Tables/Figures : pp. 183, 184.

Conclusion (A)

The difference between steam and diesel was about 84-90 tons/day in fuel consumption at the speed of 20.5 knots.
This fact made it clear to a number of shipowners that it was necessary to convert their main movers from steam to diesel.

Note. Other owners may maintain their steam turbine installation, mainly with a view to possible future downward trend in fuel oil quality. As an example we assume that Hapag-Lloyd thus converted their steam turbine ships from twin-screw into single-screw with one slow-turning propeller of large diameter in their endeavour to save on fuel oil expenditure.

Conclusion (B)

Twin-screw diesel is better than triple-screw diesel propulsion as to fuel consumption. Figures : pp. 183,185,186

The centre-line propeller, working in the race of the side-propellers, does not take advantage of the wake belt which normally on single-screw vessels increases the propulsion efficiency ("Hull Efficiency").

Recent test in Lyngby-Hydro showed the optimum twin-screw to be about 6% better than the optimum triple-screw vessel. (SSPA already mentioned this phenomina earlier).

Figures : pages 212, 213.

From separate information, we observed that - as to fuel consumption - of rather comparable ships

- Twin-screw ships Hoorn Houtman are better than twin-screw BP and BS (Figures: pp 183-186.)

Possible reasons : Hull form quality (slender pipe with brackets giving less resistance than fat bossing around shafting)

: Propeller design

: Hull roughness

Most probably : Turbo generator might be driven at sea by auxil. boiler, during legs at reduced speed. In this case the exhaust gas temperature is too low for steamproduction by the exhaust gas boilers.

From practice figures we observed that

- Sea margin on the far East-Europe service seems to be 12% - 14% for Nedlloyd Hoorn (chlor. rubber paint system)

Main item to be improved is algae antifouling by selferosion a.f. paint of ships side, which might allow for 12% seamargin in future.

(Acrylic copolyner pigmented by organic tin = SPC antifouling).

- Service speed
  A reduction of service speed from 20.5 kn. to 19.7 knots will bring the fuel consumption from 112 tons/day down to 101 tons/day.
  - At 19.7 kn. service speed the mean level of shaft horsepower lies at about 30,000 HP on one screw - which should give minor trouble as to vibration, noise, cavitation-erosion to the screw and fractures in ship structure above the propeller. However, we strongly object against utilising more horsepower continously, based on experience with the first generation of container ships, especially those having accomodation aft.
  - The Hull-lines of single and twin screw models are according to the latest developments and much attention has been paid in keeping the 12 o'clock wake peak in the propeller influx. as small as possible.
  - The difference in horsepower between single screw and twin screw for obtaining the same speed is marginal according to the latest prognosis.
    The real modeltesting shows 2 - 4% in favour of single screw.

Figures:

pp 203/204

- Both models are fitted with large slow-turning (97 rpm) propellers of diameters 7500 mm for the twin screw type and 8400 mm for the single screw type.
- The single screw ship might lose her advantage, if her propeller is designed with a rather large blade surface. This surface, however, may be necessary in view of reduction of cavitation, hull pressure fluctuations, vibrations, noise, etc.
- Nedlloyd experience with twin screw ships is very favourable in view of low fuel consumption and low vibration levels and from that point of view the Nedlloyd Newbuilding dept. would go the safe way for twin screw, especially in case of the higher speeds.
- Also, the reliability in service of the twin screw installation is higher.
  Slow-speed legs could be sailed with only one engine working whilst the other propeller is windmilling being uncoupled. The working propeller (fixed pitch) should not overload the engine however.

The rudder will need about 1 à 2 degrees angle to sail a straight course. In this way, the other engine could have maintenance at sea.
A too low load on the main engine is prevented by this method and troubles with bad combustion (linerwaer-pistrings) will be less.
However, speed then will be limited to some 16 knots.
Means for easily mechanically declutching should be provided in such case.

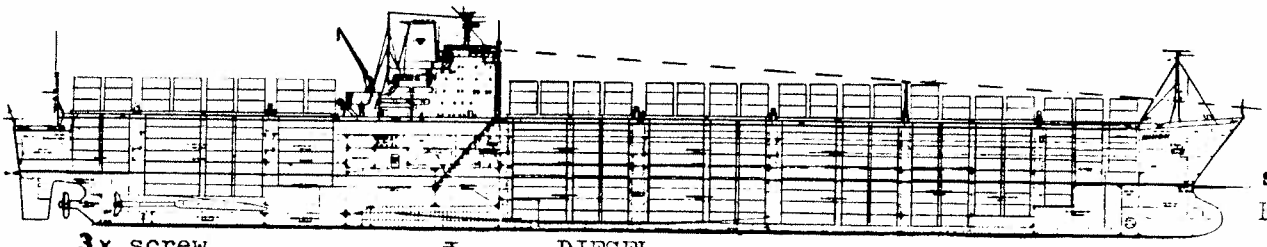
- Twin screw versus single screw
The twin screw ship will cost about 5% more and does require more maintenance in the E.R., thereby will offer the feasibility of reaching about 23 knots against 22 knots of the single screw vessel.
However important the possibility for keeping schedule may be, we like to emphasise the considerable reduction in fuel costs (and consequently slot-costs) when reducing the service speed from 20.5 knots twin screw to 19.5 knots - which would simultaneously enable the adoption of the cheaper single screw vessel.

PROPULSION ON FUTURE TONNAGE

Basic assumptions for the far east run : using 85-90% MCR, sea margin 12-14%, draft 12,5 m.,capacity over 3000 Teu.

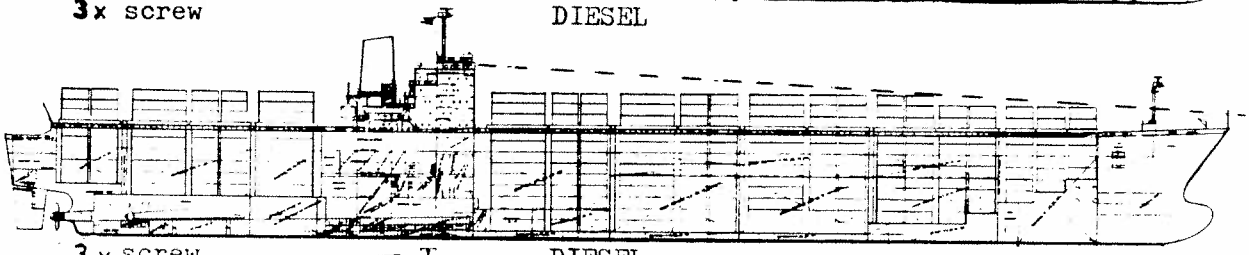
- Twin screw
In case 20.5 knots mean service is required with peaks of 23 knots max. then Nedlloyd promotes the twin screw vessel.
- Single screw
In case 19½ kn mean service speed is required with peaks of 22 knots max. then Nedlloyd has the courage to go for single screw.

THIRD-GENERATION CELLULAR CONTAINER VESSELS.



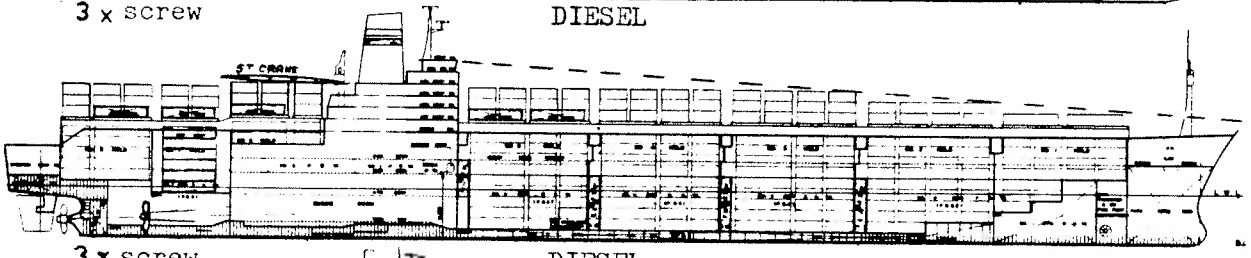
Ship "N"
BUILT 1972

3x screw DIESEL



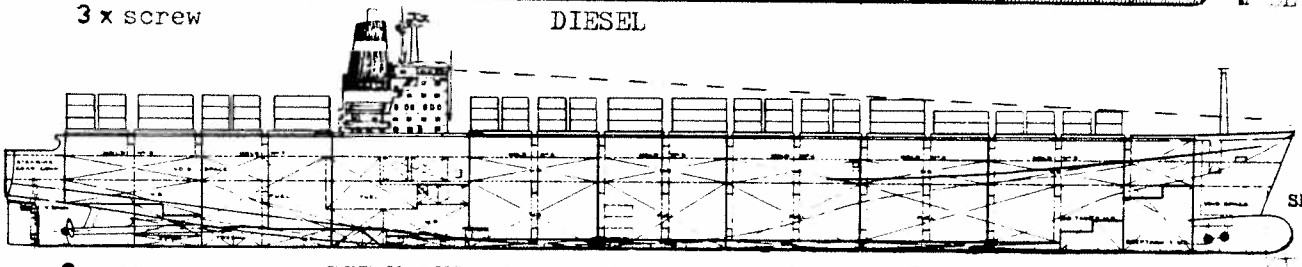
Ship "S"
BLT 1972

3 x screw DIESEL



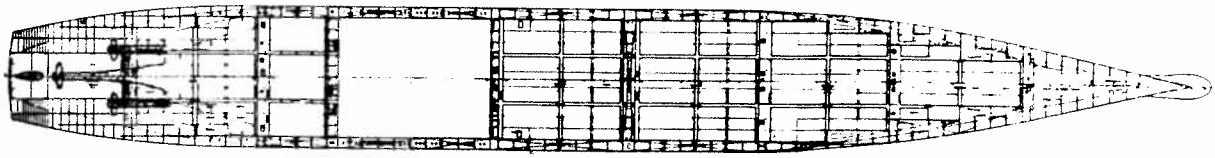
Ship "T"
BLT 1972

3 x screw DIESEL



Ship "D"
1973

2x screw STEAM TURBINE altered to DIESEL 1981



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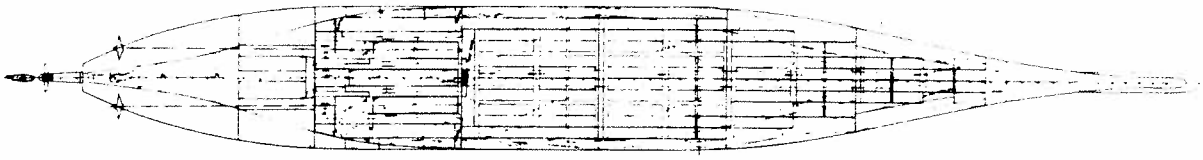
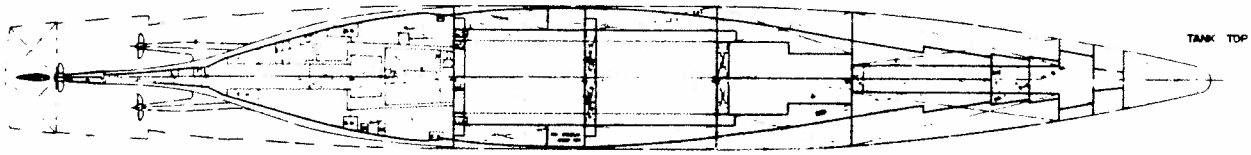
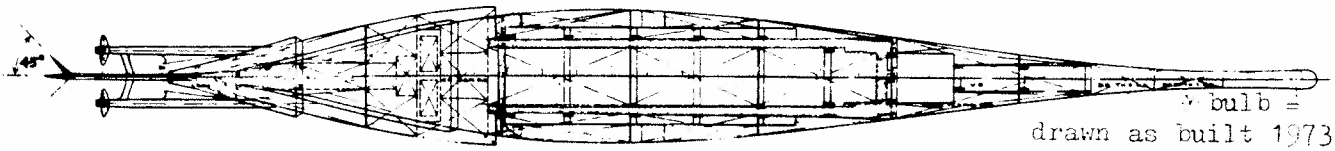


S



T



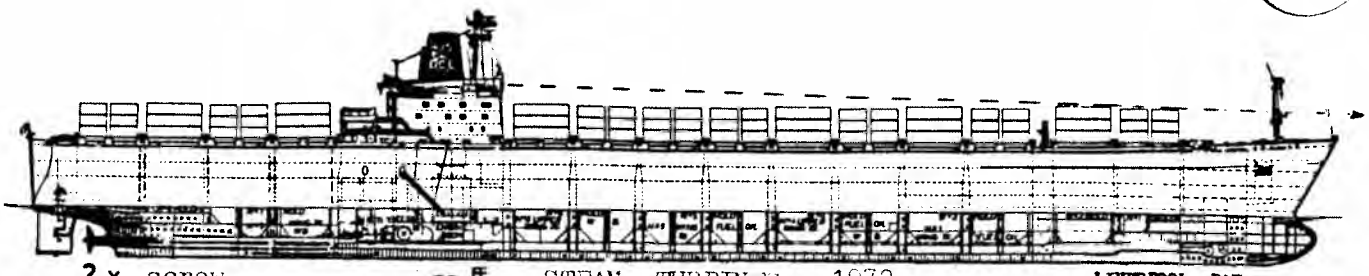
D

* bulb = drawn as built 1973

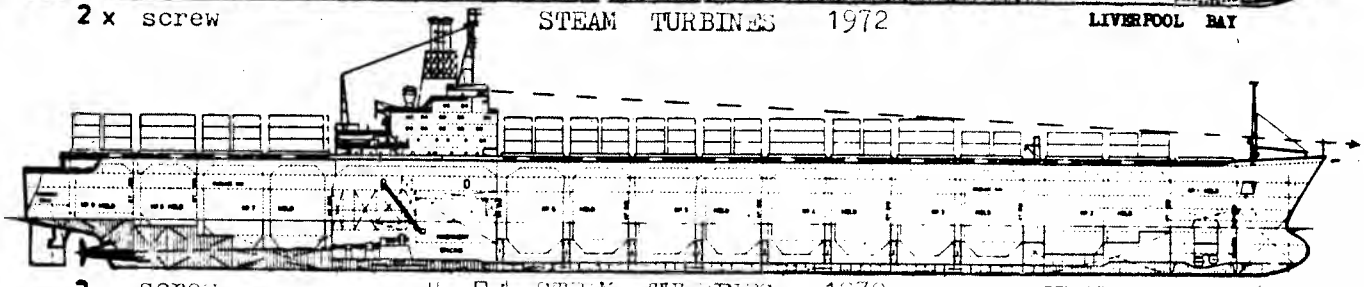
TANKTOP VIEWS

SHIPS / ITEMS	Ship "NIHON"	Ship "SELANDIA" "JUTLANDIA"	Ship "TOYAMA"	Ship "DEJIMA" "DELFT"	
	1972	1972	1972	1972 STEAM	1982 CONV DIESEL
DIMENSIONS					
Length o.a.	275,27 M	274,32 M	275,09 M	287,02 M	
Length b.p.	257,60 "	257,60 "	259,00 "	273,00 "	
Breadth, extreme	32,28 "	32,31 "	32,31 "	32,31 "	
Breadth, moulded	32,21 "	32,23 "	32,20 "	32,24 "	
Depth, moulded, to Upperdeck	23,90 "	23,90 "	24,00 "	25,00 "	
Depth to 2nd deck	20,96 "	14,39 "	abt. 17,70 "	20,44 "	
Draught, max.	11,61 M	11,60 M	11,61 M	... 12,72 M	
CAPACITIES					
Deadweight	35.000 t	34.730 t	34.005 t	48.150 t	47.078 t
	23.466	22.000	24.823	23.827 STEAM	24.899 DIESEL
Displacement	58.446 t	56.730 t	58.828 t	71.977 t	71.977 t
Containers					
Below deck (TEU)	1,692	1.662	1.608	2.020	
20'-boxes	964	950	860	1.152	
40'-boxes	364	356	374	434	
Above deck (3-h)(TEU)	850	874	838	932	
20'-boxes	850	874	810	932	
40'-boxes	416	430	386	466	
Grand total (TEU)	2.542	2.536	2.446	2.952	
Of which reefers	110	110	102	110	
(Above/Below)	(50/60)	(50/60)	(52/50)	(40/70)	
PROPULSION					
Kind	Diesel	Diesel	Diesel	Steam Turbine	DIESEL
PS and SB engines:					
- Make/type	G.V.-DM850/1700VGS 10 cyl.	B & W- 9K84 EF 9 cyl.	B&W-type 9K84EF 9 cyl.	Foster Wheeler Stal Laval/St.	SULZER MHI IHI 2x8 RND M
- BHP/Revs.	23.400/115	22.500/117	22.500/117	40.550/136	25.440/112
Center engine:					
- Make/type	G.V.-DM850/1700VGS 12 cyl.	B & W- 12K84EF 12 cyl.	B&W-type 12K84EF 12 cyl.	—	
- BHP/Revs.	28.200/115	30.000/119	30.000/119	—	
Total b.h.p.	75.000	75.000	75.000	81.100	50.880 BHP
Length of Eng.Room	34,20 M	34,17 M	(not opmparable)	30,60 M	
Frames / % of L.b.pp	85-128 / 13,3 %	83-127 / 13,3 %		88-128 / 11,2 %	
PROPELLERS					
Center	1 x Var. Pitch, 5-blades	1 x Var.Pitch, 4-bl.	1 x Var.Pitch, 4-bl.	(No center engine)	
ø and weight	6,25 M, 39,7 t	6,50 M, 40 t	... 38,6 t	—	
Sides	2 x fixed pitch, 6 blades	2 x fixed p., 6-bl.	2 x fixed.p., 6-bl.	2 x fixed p.,	
ø and weight	5,85 M, 19,6 t	5,85 M, 19,6 t	... 19,1 t	5 BL ø 6.30 29.5 T	4 BL ø 6.50 24.2 T
BOWTHRUSTER	1 x 1.200 hp, var.pitch	1 x 1.000 hp, V.P.	1 x 1.000 hp, V.P.	2 x 1.000 hp, V.P.	
STABILIZERS	Fins 6,7 m2 each	Fins 6,7 m2 each	Fins 6,7 m2 each	Fins 7 m2 each	
Location	Frames 136-140	Frames 136-140	Frames 150-155	Frames 165-168	
RUDDER					
- Type	Semi-spade	Semi-spade	Semi-spade	"Barn door"	
- Surface area	Abt. 46 m2	Abt. 46 m2	Abt. 46 m2	Abt. 58 m2	
- Weight	45 t	48 t	96 t	120 t	
AUXILIARY POWER plant h.p. (about)	4 x 1.300 KW (1742 hp)	5 x 1.008 KW (1350 hp)	5 x 980 KW (1313 hp)	3 x 1500 KW + 2 x 900 KW	3 x 1200 KW 2 x 900 KW
Voltage/Frequency	440/220 ac/ 60 hz	440/220 ac/ 60 hz	440/220 ac/ 60 hz	440/220 ac/ 60 hz	

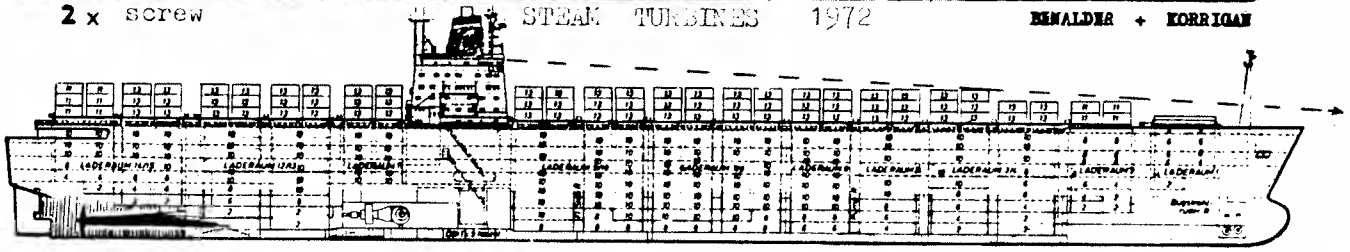
THIRD-GENERATION CELLULAR CONTAINER VESSELS.



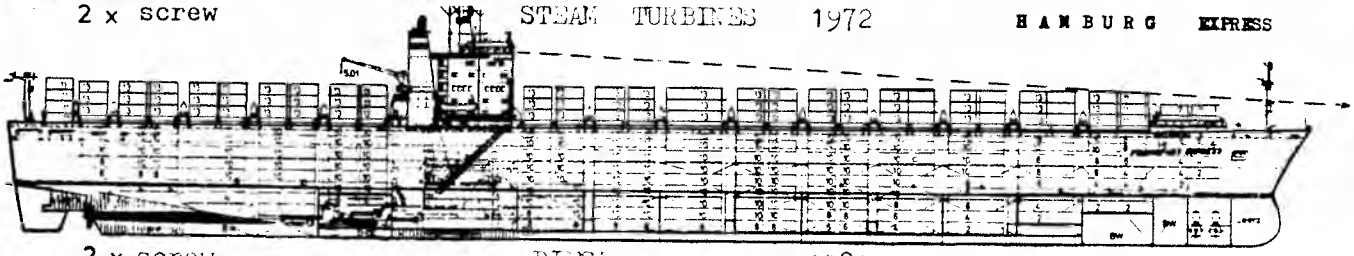
2 x screw STEAM TURBINES 1972 LIVERPOOL BAY



2 x screw STEAM TURBINES 1972 EMALDER + KORRIGAN

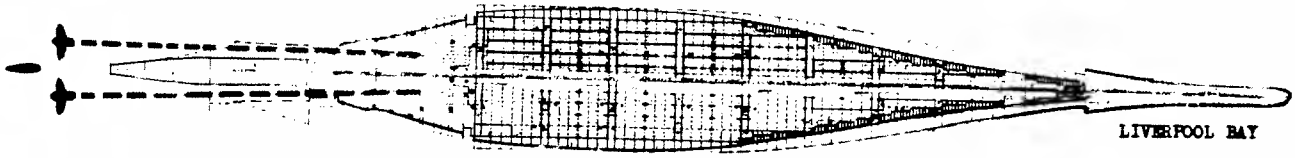


2 x screw STEAM TURBINES 1972 HAMBURG EXPRESS

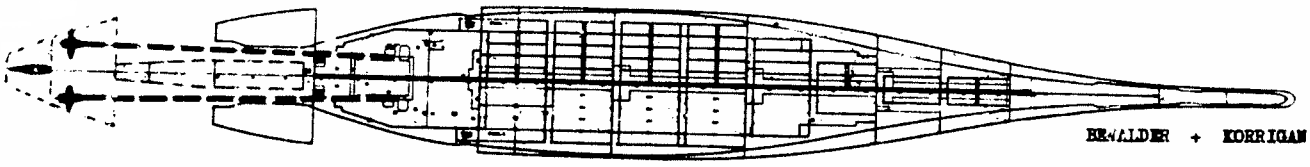


2 x screw DIESEL 1981 FRANKFURT EXPRESS ('81)

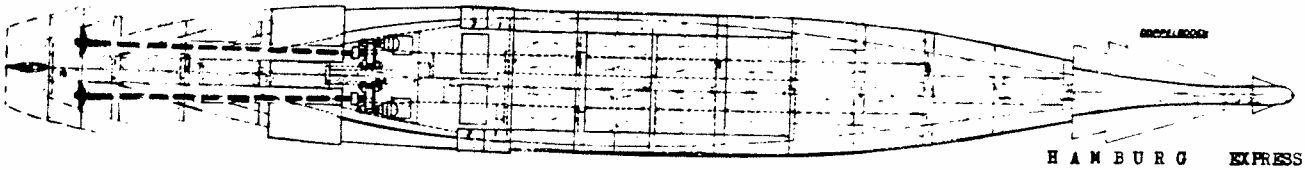
LONGITUDINAL VIEWS



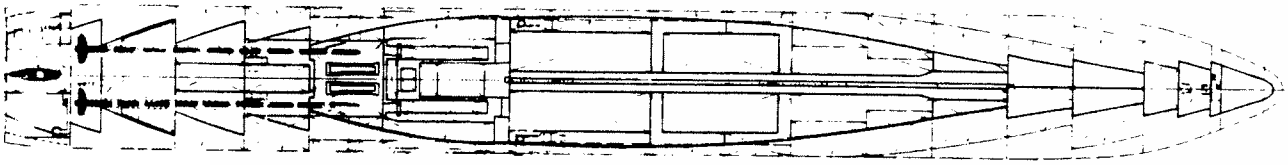
LIVERPOOL BAY



EMALDER + KORRIGAN



HAMBURG EXPRESS (1972)



FRANKFURT EXPRESS (1981)

SHIPS / ITEMS	LIVERPOOL BAY Tokyo Bay Kowloon Bay Cardigan Bay Osaka Bay	BERWALDER Benavon City of Edinburg	KORRIGAN	HAMBURG EXPRESS Tokyo Express	FRANKFURT EXPRESS
BUILT	1972 H.D.W., Kiel	1972 H.D.W., Kiel	1973 H.D.W., Kiel	1972 Hl.& Voss, Hambg.	1981 H.D.W., Kiel
DIMENSIONS					
Length o.a.	289,50 M	288,75 M	288,75 M	287,70 M	287,73 M
Length b.p.	274,32 "	274,32 "	274,32 "	273,00 "	271,00 "
Breadth, extreme	Panamax.	Panamax.	Panamax.	32,26 "	Panamax.
Breadth, moulded	32,26 "	32,26 "	32,26 "	32,19 "	32,20 "
Depth, moulded	24,60 "	24,60 "	24,60 "	25,00 "	24,00 "
Draught, max.	13,03 "	13,03 "	13,03 "	12,03 "	13,06 "
CAPACITIES					
Deadweight	48.542 t	49.600 t	49.742 t	42.470 t	51.540 t
Displacement	73.642 t	73.642 t	73.642 t	65.870 t	
Lpp x Bmld x D mld	217.800 m3	217.800 m3	217.800 m3	219.800 m3	209.500 m3
Containers					
Below deck (TEU)	1.944	1.948	1.944	1.990	1.940
20'-boxes	1.108	1.284	1.108	1.126	1.104
40'-boxes	418	332	418	432	418
Above deck (TEU)(3-h)	1.017	856	856	1.020	1.105
20'-boxes	717 and:	556 and:	856 or:	1.020	1.105
40'-boxes	150	150	428		
Grand total (TEU)	2.961	2.804	2.800	3.010	3.045
Of which reefers 40'		66 a/deck	66 a/deck		132 a/deck
TONNAGE					
Gross	58.889 R.T.	57.887 R.T.	57.249 R.T.	58.088 R.T.	58.385 R.T.
Nett	35.191 "	34.014 "	33.967 "	38.425 "	37.934 "
PROPULSION					
Kind/quantity	Steam turbine/2	Steam turbine/2	Steam turbine/2	Steam turbine/2	Diesel/2
Make/type	Foster-Wheeler/ES III	F-W / ES III	F-W / ES III	F-W / ES III	M.A.N./ K9SZ 90/160B
particulars	121/145 T/hr steam 65,6 atü, 513 C	118/138 T/hr 63,4 bar, 516 C	/132 T/hr 64,7 bar, 516 C	115/138 T/hr 64,5 atü, 515 C	
turbines	2 x Stal-Laval/AP 40	2x GEC/Engl.El.	2x GEC/Engl.El.	2x Stal-Laval/AP40	
s.h.p./ prop.revs	2 x 40.560/136	2 x 44.000/140	2 x 44.000/140	2 x 40.560/136	2 x 27.200 /122
S.H.P. total	81.120	88.000	88.000	81.120	54.400
PROPELLERS	2 x 6.150 ⌀	2 x 6.150 ⌀	2 x 6.150 ⌀	2 x 6.150 ⌀	2 x 6.100 ⌀, 6 bl.
BOWTHRUSTS	2 x 1.000 hp, Var.P. (2 x 11 T)	2 x 1.000 HP/VP (2 x 11 T)	2 x 1.000 HP/VP (2 x 11 T)	2 x 1.000 HP/VP (2 x 11 T)	2 x 1.000 KW(3.300 V)/ Fixed Pitch
STABILIZERS	Fins, Denny-Brown/AEG	Fins, D-B/AEG	Fins, D-B/AEG	Fins, Hl+Voss/Siem.	Fins, D-B/AEG/HDW 12 S - 9,1
AUXILIARY POWER PLANT					
Turbo-generators	2 x 2.500 KW	3 x 1.800 KW	3 x 1.800 KW	3 x 1.950 KVA	2 x 2.800 KW
Voltage/Frequency		440 V 60 Hz	440 V 60 Hz	440 V 60 Hz	450 V 60 Hz
Diesel generators	2 x 1.250 KW	2 x 900 KW	2 x 900 KW	1 x 1.650 KW	1 x 900 KW 450 V 60 Hz
Shaft generators	—	—	—	—	2 x 1.400 KW

HOORN

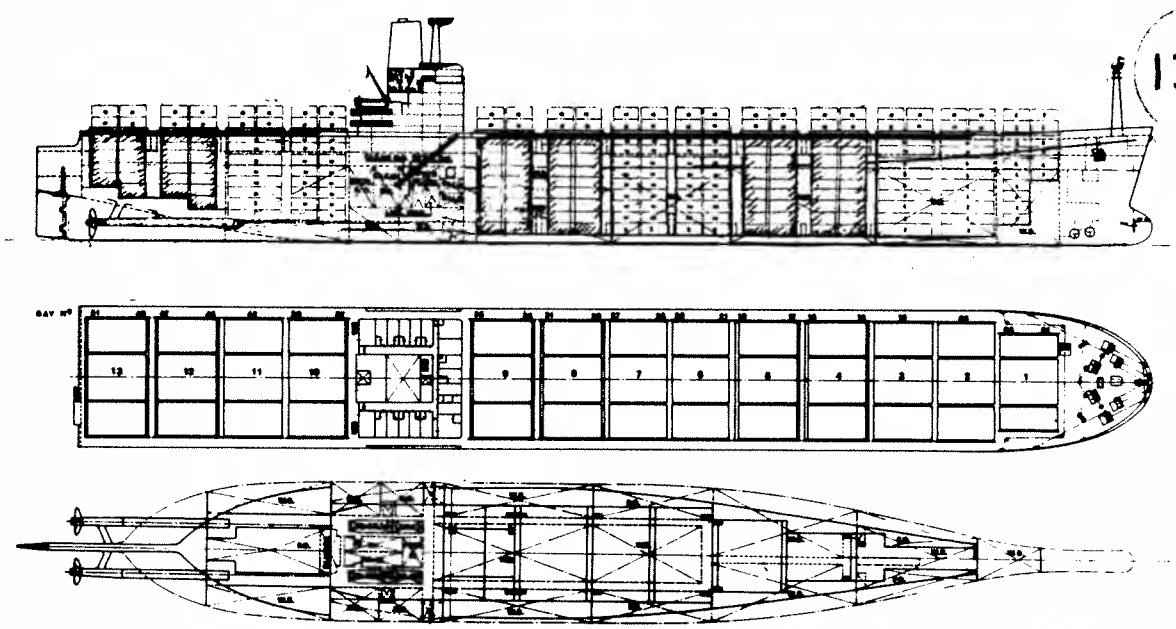
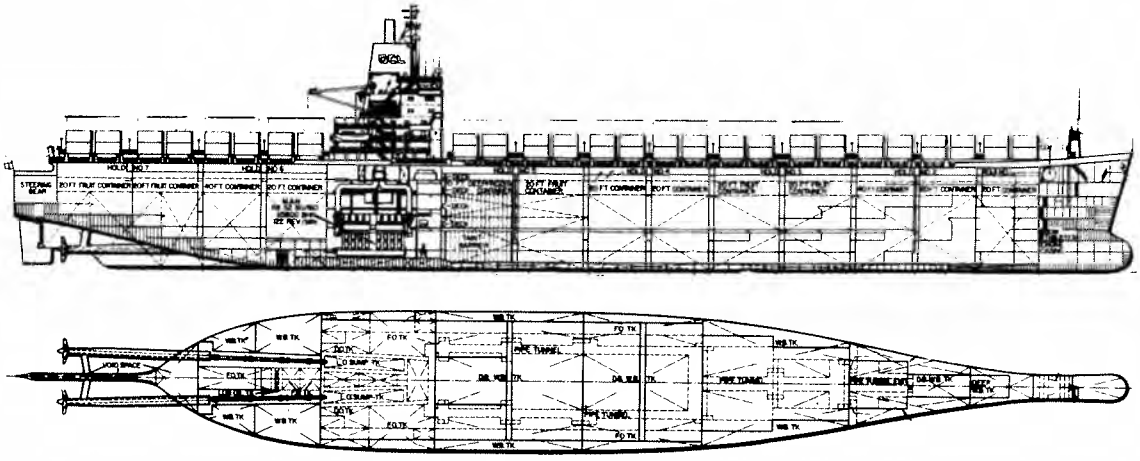
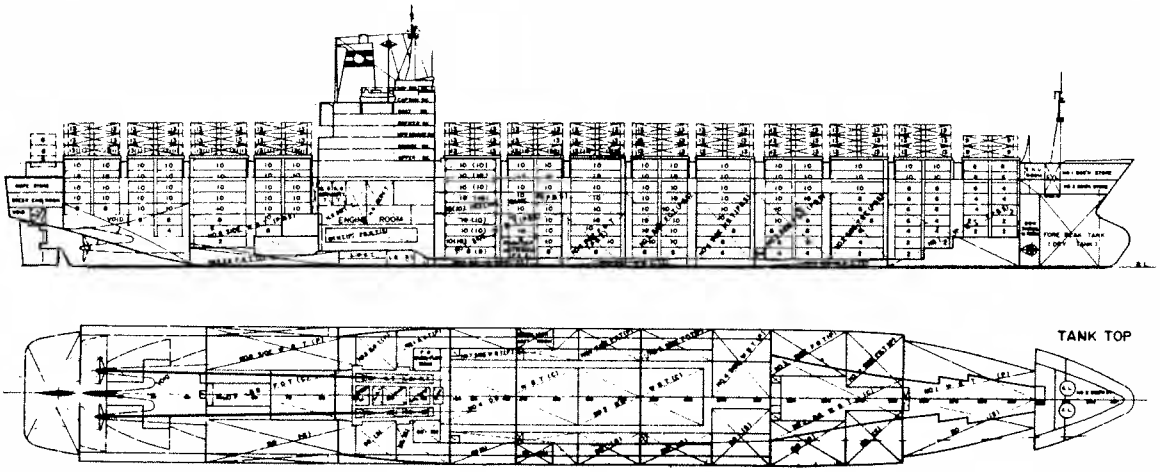


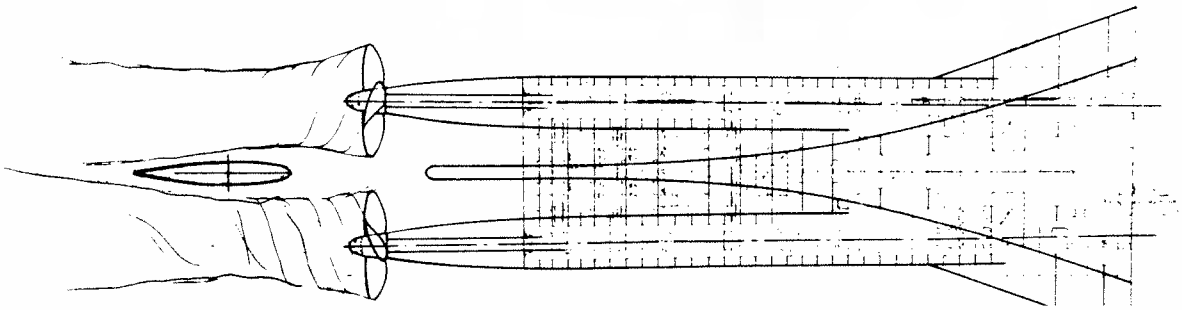
TABLE BAY



BP
BS



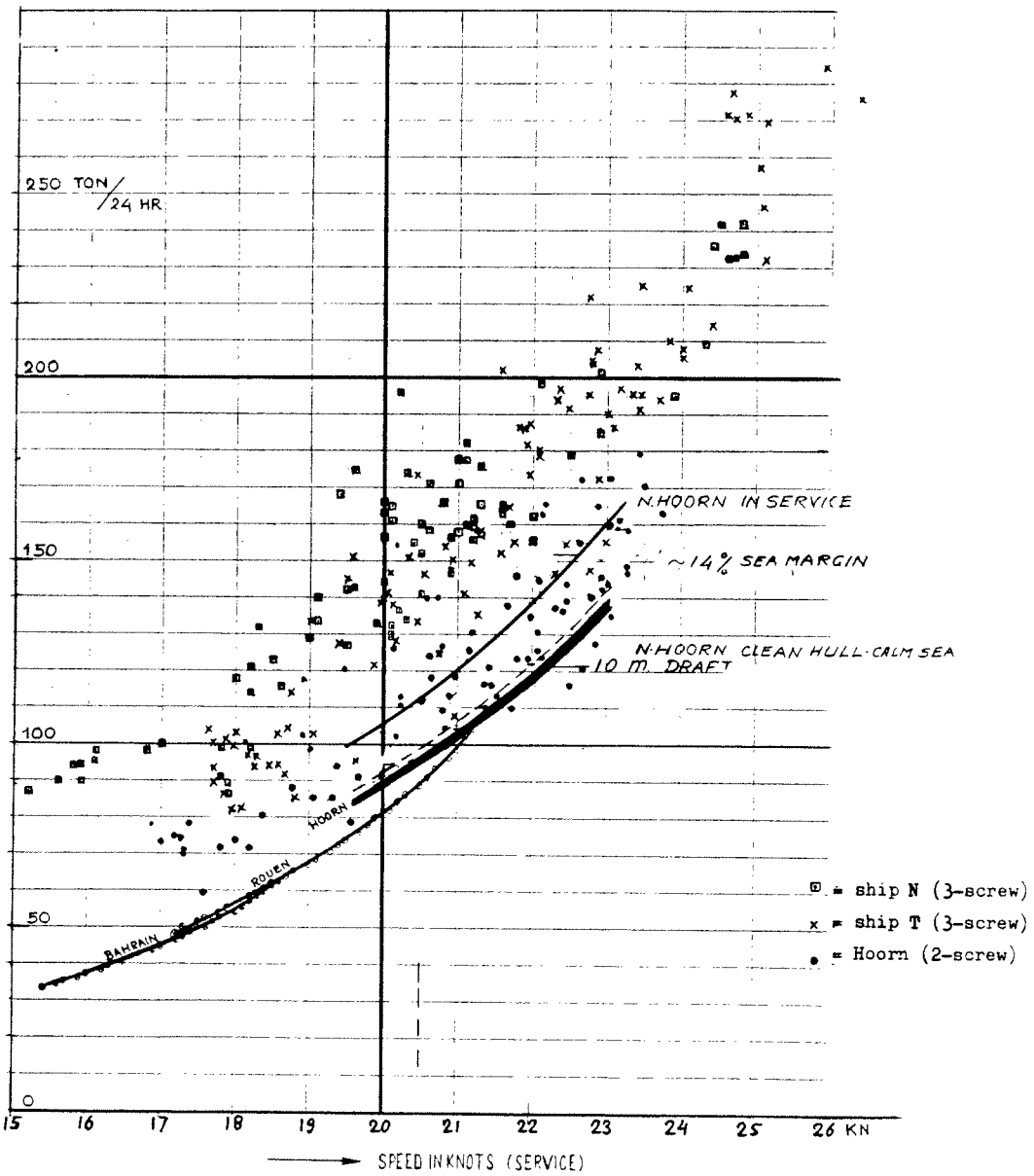
SVENDBORG
MAERSK



SHIPS / ITEMS	TABLE BAY	HUNGA PERMAI HUNGA SURIA	SVENDBORG MAERSK KISO MARU	NEDLLOYD HOORN
BUILT, year	1979	1979	1974	1979
yard	A.G.W., Bremen	Sumitomo H.I., Oppama	I.H.I., Aioi	(aft) V.D.S.M., Rozenbg. (fwd) N.D.S.M., Amsterdam
DIMENSIONS				
Length o.a.	258,50 M	267,00 M	261,20 M	258,50 M
Length b.p.	248,20 "	250,00 "	242,00 "	247,00 "
Breadth, extreme	Panamax.	Panamax.	Panamax.	Panamax.
Breadth, moulded	32,26 M	32,24 M	32,20 M	32,26 M
Depth, moulded	24,15 "	24,15 "	19,50 "	24,15 "
Draught, max.	13,03 "	13,03 "	11,70 "	13,03 "
CAPACITIES				
Deadweight (at max. drft)	47.197 t	49.149 t	31.771 t	48.437 t
Displacement	71.542 t			71.343 t
Lpp x B(m) x Depth(m)	193.367 m3			192.435 m3
Containers				
Below deck (TEU)	1.780	1.780	984	1.788
20'-boxes	1.036	1.004	388	1.032
40'-boxes	372	388	298	378
Above deck (TEU)	656	672	548	660
20'-boxes	656	672	548	660
40'-boxes	326	330	274	330
Grand total (TEU)(service)	2.436	2.452	1.532	2.448
" " (")(maximum)	2.696	2.770	2.020	2.708
of which Reefers	941	308 TEU (in 154 boxes 40')	151	943
(Above/Below dk)	55/886	132/176	/	55/886
PROPULSION				
Kind/quantity	Diesel, 2 x	Diesel, 2 x	Diesel, 2 x	Diesel, 2 x
Make/Type	AGW / M.A.N., K 8 SZ-90/160	Sumitomo/ Sulzer, 9 RND-90 M	IHI / Sulzer, 12 RND-90	De Schelde/ Sulzer, 8 RND-90 M
ea. B.H.P./R.P.M.	25.680 / 122	30.150 / 122	34.800 / 122	26.800 / 122
Total B.H.P.	51.360	60.300	69.600	53.600
PROPELLERS	2 x Fixed Pitch 6-blades	2 x F.P.	2 x F.P., ϕ 6,50 M 5-blades	2 x F.P., ϕ 6,35 M 4-blades
BOWTHRUSTERS	2 x 1.000 HP	1 x 1.500 HP	1 x 1.400 HP (V.P.)	2 x 1.000 HP
STABILIZERS				Fins
SKEG	1, Open to ½ height	1, Nearly Closed	1, Open	1, Closed
RUDDER	1, Semi-Balance	1, Semi-Balance	1, Semi-Balance	1, "Barndoor"-type
AUXILIARY POWER	6 x 1.500 KW	2 x 1.360 KW	2 x 1.360 KW 2 x 1.000 KW	6 x 1.100 KW
Voltage/Frequency		440 V / 60 Hz ac	440 V / 60 Hz ac	450 V / 60 Hz ac

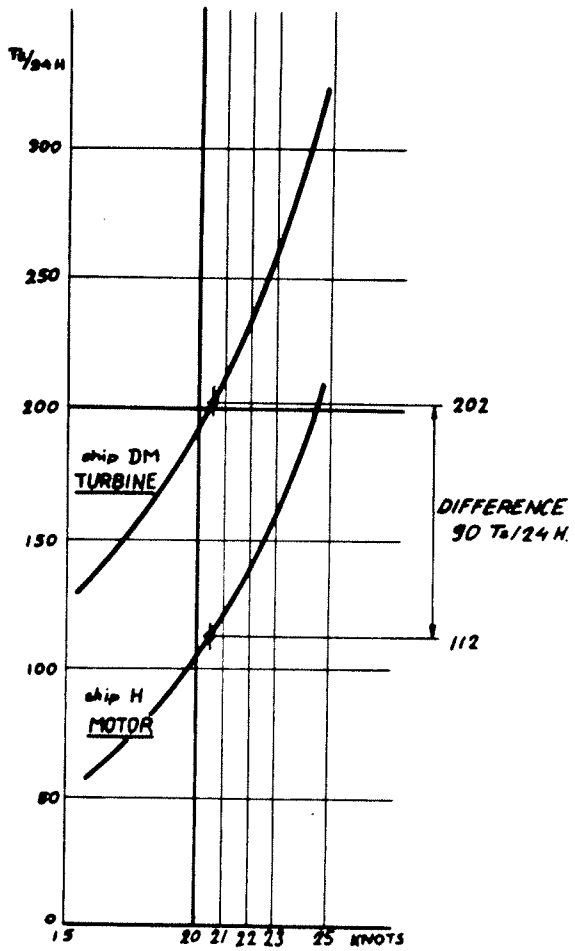
FUEL CONSUMPTION OF DIESEL-VESSELS (TRIPLE-SCREW : TWIN-SCREW)

13-13



DAILY FUEL CONSUMPTION IN TON/24 HR (STEAM : DIESEL)

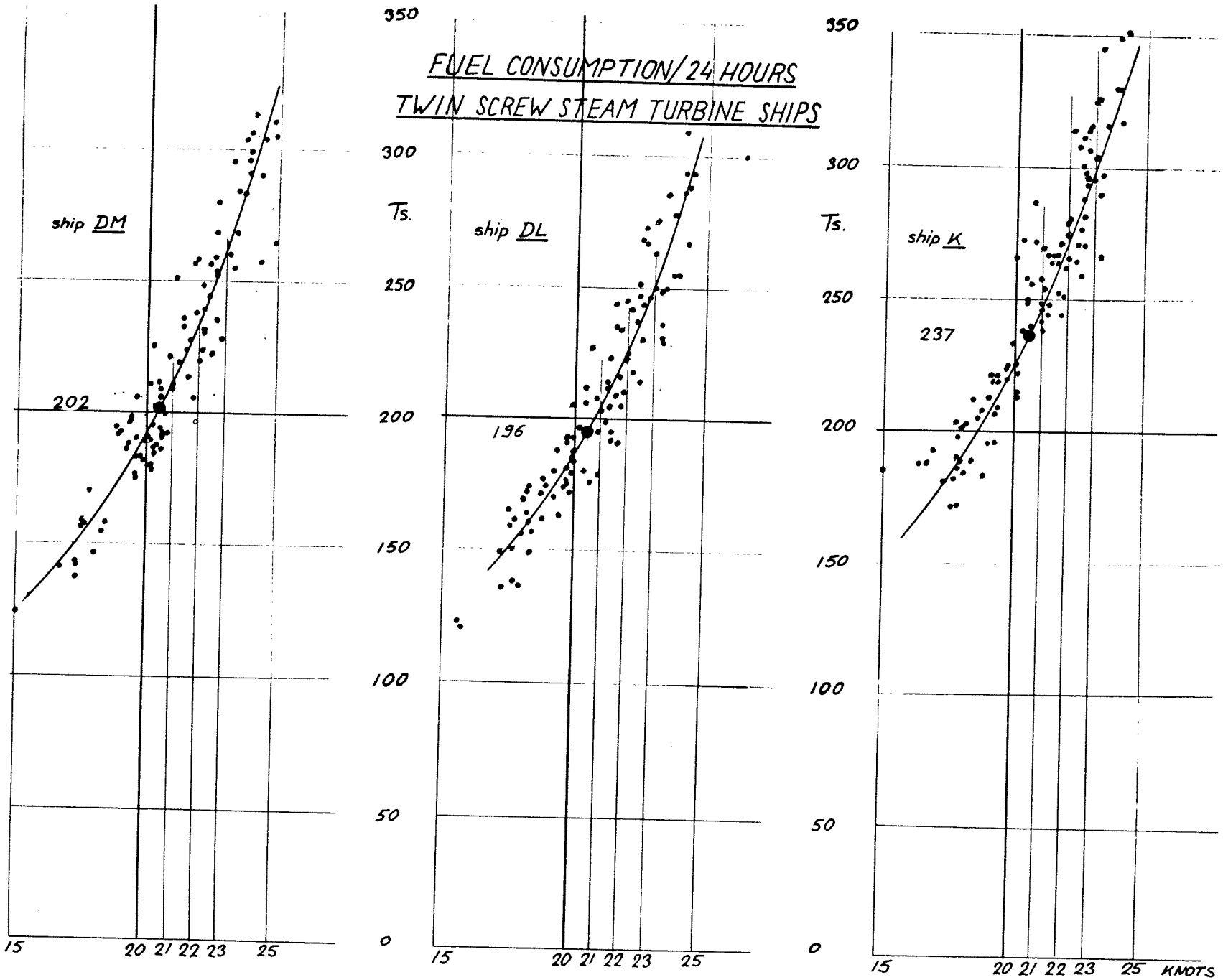
		3 S-DIESEL CENTRE PROP & C.P.				2S-STEAM				2S-DIESEL SHORTER VESSELS			2S-DIESEL (Converted from Steam)
		< SISTERS >				DELFT DEJIMA < SISTERS >				HOORN HOUTMAN	< SISTERS >		DELFT DEJIMA < SISTERS >
	Speed	J	SL	T	N	DL	DM	K	aver.	H	BP	BS	
	23	173	180	185	202	251	261	300		160	185	185	149
	22,5	161	169	172	188	239	247	286		149	174	175	141
	22	151	159	160	176	227	235	273		139	164	165	132
	21,5	142	149	149	165	216	223	260		129	154	156	125
	21	134	139	138	155	205	212	248		120	145	147	118
SERVICE :	20,5 Kn	124	130	128	146	196	202	237		112	136	139	112
	20	117	121	118	137	187	192	226		105	128	131	105
		B+W		MITSUI	GÖTA	BR. VULKAN		HDW		RSV	SUMITOMO		
		B+W ENG > GÖTA				< STAL LAVAL TURB. >		AAE TURB.		< SULZ >	SULZ+TURBO-GEN		
	Const. cons. /day	10 Ts DO				in port: 80 Ts FO -80				10 Ts DO prt 7 Ts FO	At sea: 2,5 Ts DO in port: 10 Ts DO		

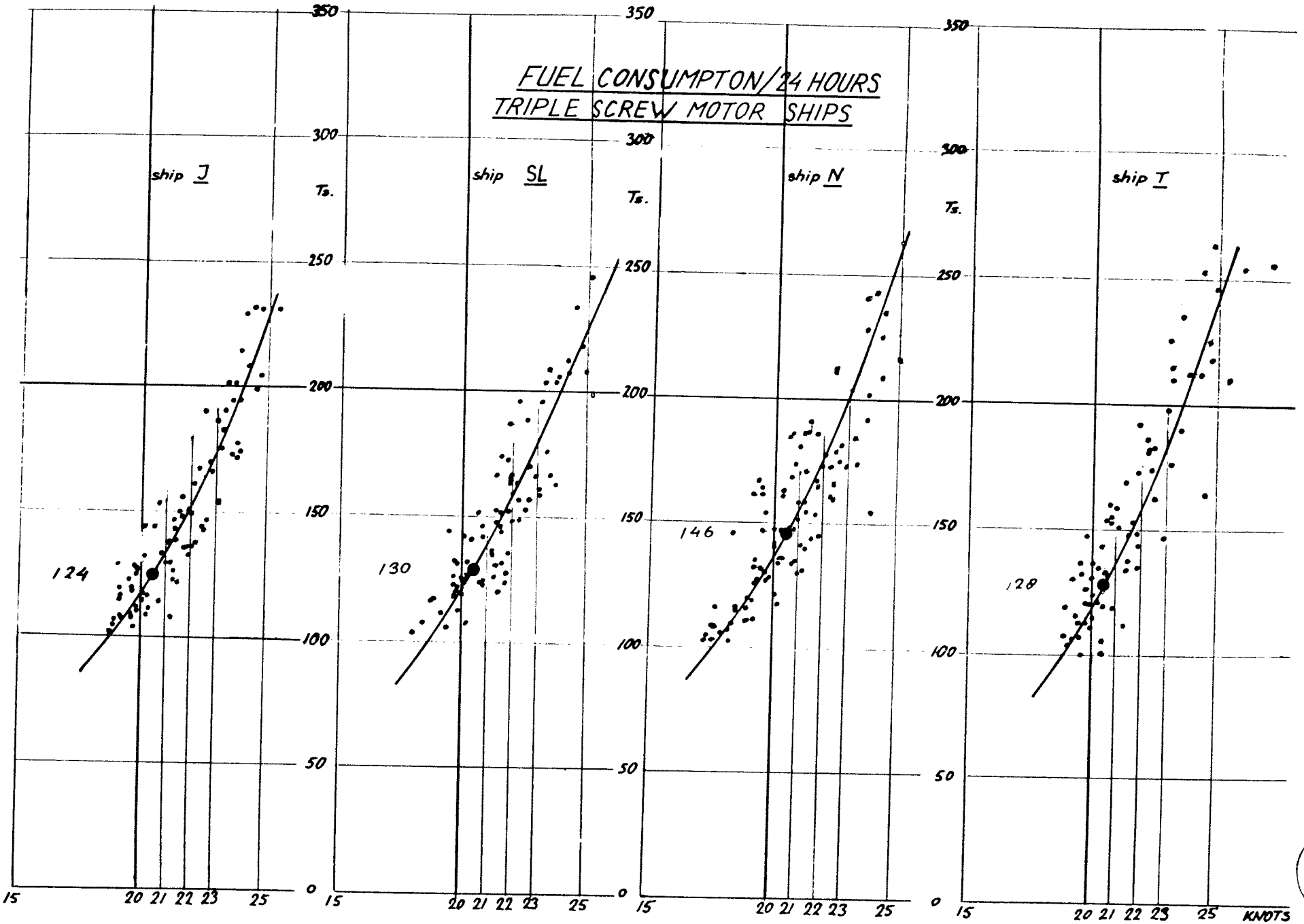


COMPARISON FUEL CONSUMPTION
STEAM TURBINE : DIESELMOTOR
FAR EAST SERVICE CONDITION

		mijlen	AT SEA Vaar-tijd hr	FO. sea	DO. sea	FO. port	DO. port	
2 S-DIESEL - N. HOUTMAN	1. kw. '79	30885	1568	6833	603,5	123,5	286,2	
N. "	2. kw. '79	33573	1616	8026	593,9	205,4	288,7	
N. HOORN	3. kw. '79	30219	1481	6804	563,8	157,4	308,4	
		94677	4665	21663	1761,2	486,3	883,3	
			KNOTS 20,3	111,5 T/ETM				229 Ton / 1000 MILES
				× 4/3= 28900 T P. YR				
2 S-STEAM - NEDL. DEJIMA	1. kw. '79	32447	1585	13371	0	1376	19	
"	2. kw. '79	29463	1402	13043	0	1960	23	
"	3. kw. '79	32407	1653	13768	2	1589	21	
		94317	4640	40182	2	4925	63	
			KNOTS 20,3	208 T/ETM				426 Ton / 1000 MILES
				× 4/3= 53600 T P. YR				
2 S-STEAM - NEDL. DELFT	1. kw. '79	28078	1435	10735		1838	15	
"	2. kw. '79	32114	1547	13999	10	1657	25	
"	3. kw. '79	33714	1585	14755	---	1443	24	
		93906	4567	39489	10	4938	64	
			KNOTS 20,5	208 T/ETM				420 Ton / 1000 MILES
				× 4/3= 52700 T P. YR				

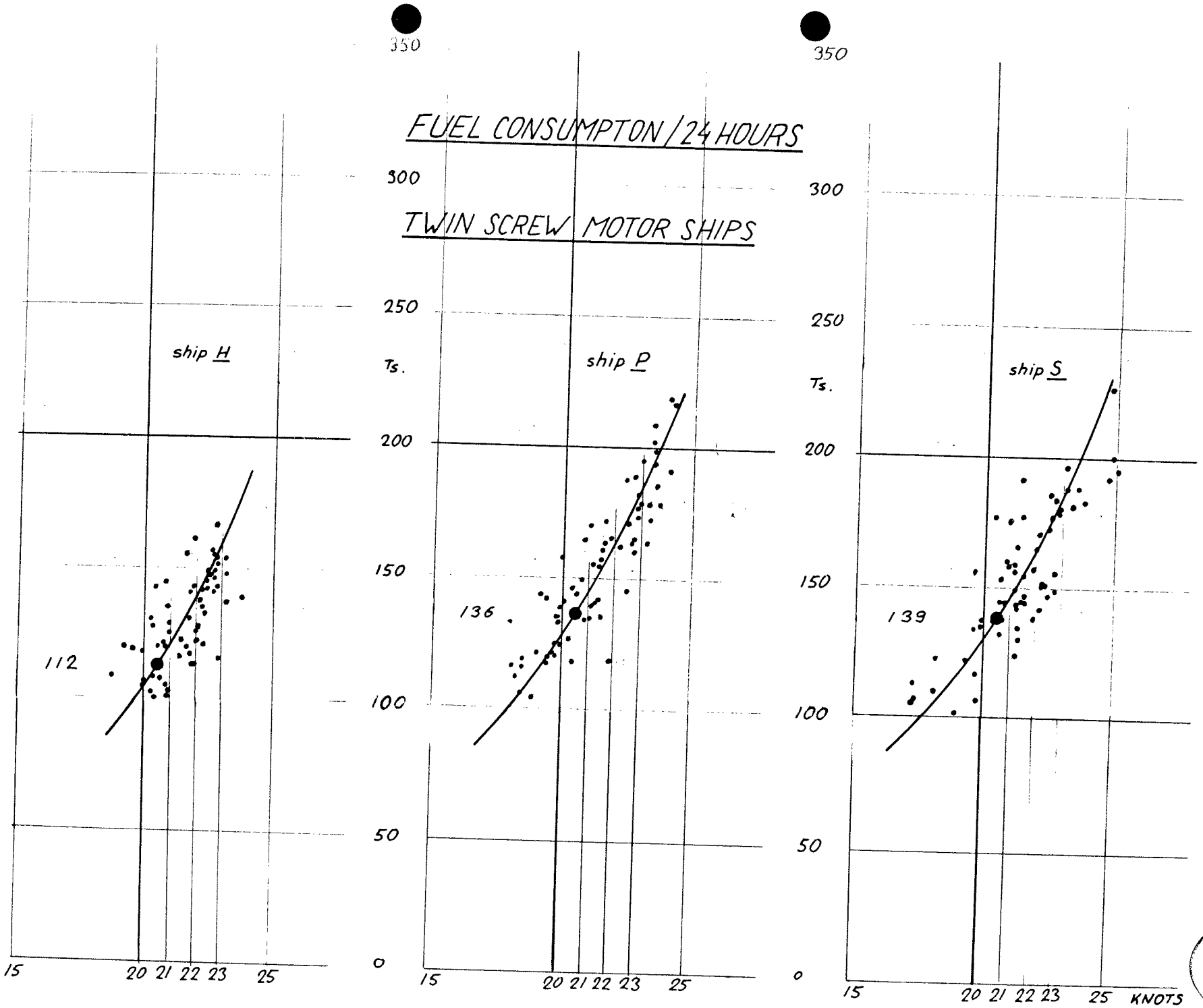
FUEL CONSUMPTION/24 HOURS
TWIN SCREW STEAM TURBINE SHIPS




FUEL CONSUMPTON/24 HOURS
TRIPLE SCREW MOTOR SHIPS
ship J
ship SL
ship N
ship T
Ts.
124
130
146
128
350
300
250
200
150
100
50
0
15
20 21 22 23 25
KNOTS

FUEL CONSUMPTON / 24 HOURS

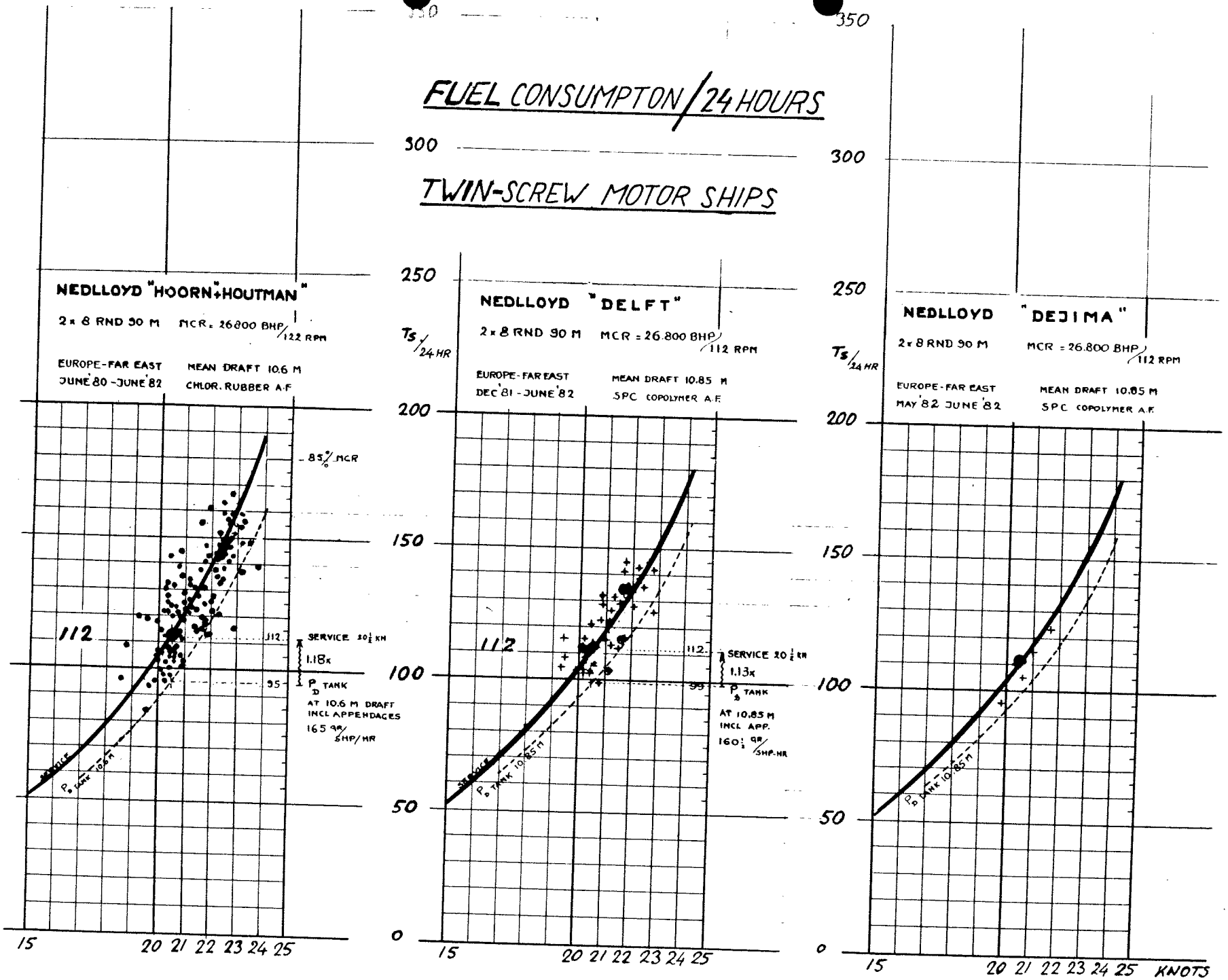
TWIN SCREW MOTOR SHIPS



13-17

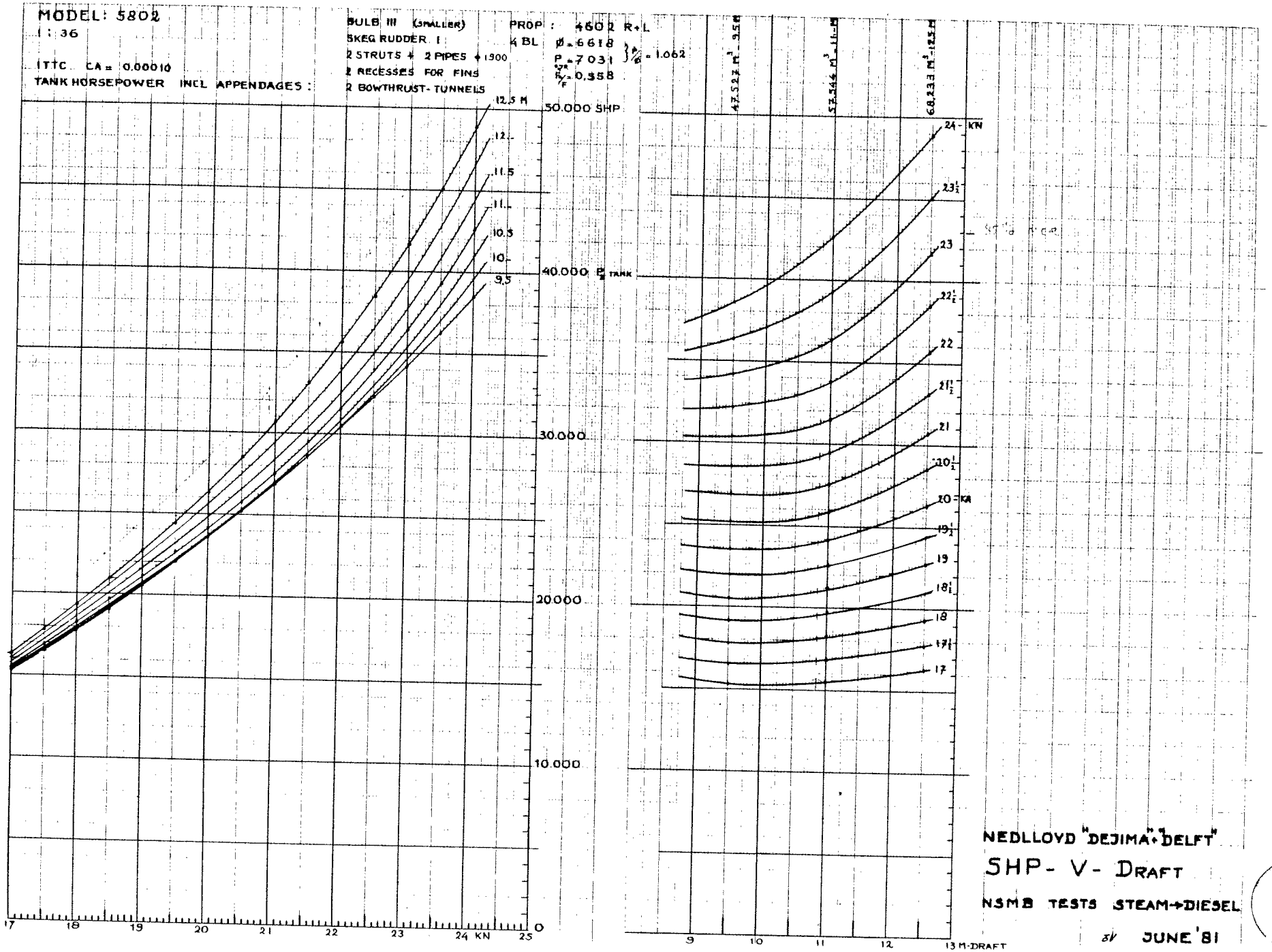
FUEL CONSUMPTON / 24 HOURS

TWIN-SCREW MOTOR SHIPS

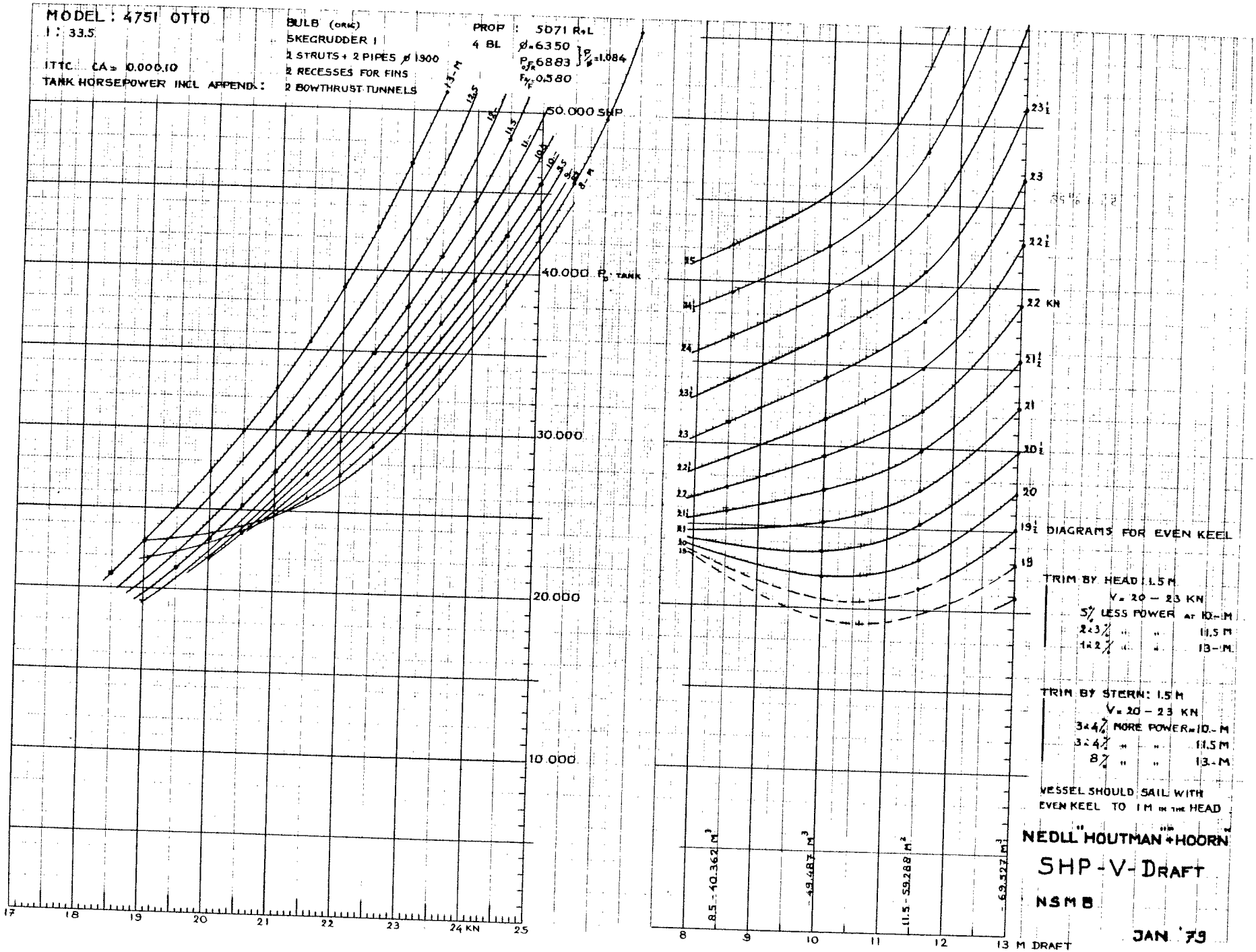


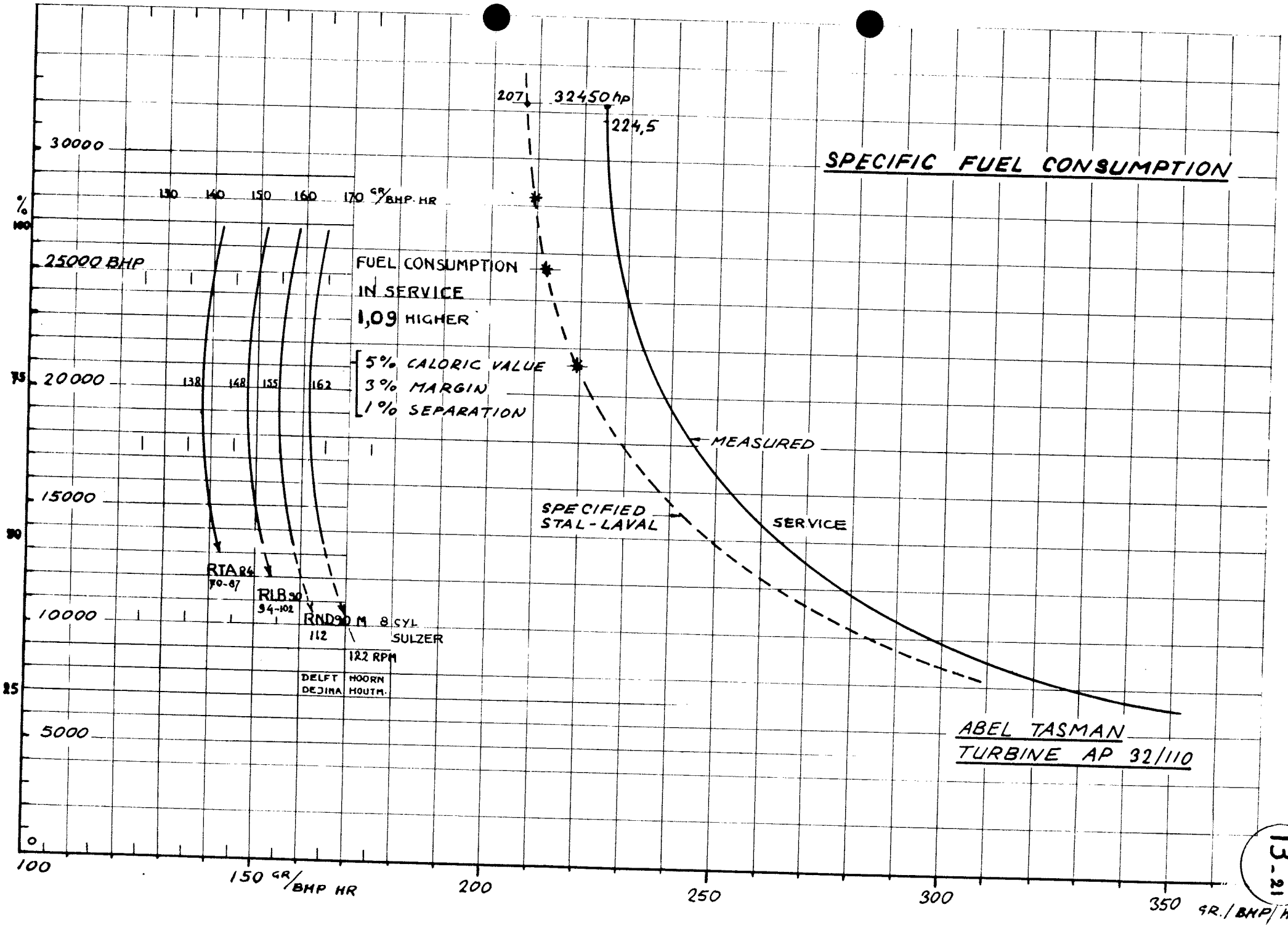
13-18

IV 4-7-82


MODEL: 5802
1 : 36
ITTC CA = 0.00010
TANK HORSEPOWER INCL APPENDAGES :
BULB III (SMALLER)
SKEG RUDDER I
2 STRUTS + 2 PIPES + 1900
2 RECESSES FOR FINS
2 BOWTHRUST- TUNNELS
PROP : 4602 R+L
4 BL
Ø = 6618
P = 7031
Ø = 1.062
12.5 M
12
11.5
11
10.5
10
9.5
50.000 SHP
40.000 P TANK
30.000
20.000
10.000
0
17
18
19
20
21
22
23
24 KN
25
24 - KN
23½
23
22½
22
21½
21
20½
20 - KN
19½
19
18½
18
17½
17
9
10
11
12
13 M-DRAFT
NEDLLOYD "DEJIMA" + "DELFT"
SHP - V - DRAFT
NSMB TESTS STEAM→DIESEL
JUNE '81

13-19


MODEL : 4751 OTTO
1 : 33.5
ITTC CA = 0.000.10
TANK HORSEPOWER INCL APPEND. :
BULB (ORIG)
SKEGRUDDER 1
2 STRUTS + 2 PIPES ø 1900
2 RECESSES FOR FINS
2 BOWTHRUST TUNNELS
PROP : 5071 R+L
4 BL ø=6350
P/D=1.084
50.000 SHP
40.000 P_D TANK
30.000
20.000
10.000
17 18 19 20 21 22 23 24 KN 25
13 M
12.5
12
11.5
11
10.5
10
9.5
9
8.5 M
23½
23
22½
22 KN
21½
21
20½
20
19½
19
DIAGRAMS FOR EVEN KEEL
TRIM BY HEAD 1.5 M
V = 20 – 23 KN
5% LESS POWER AT 10.- M
2 à 3% " " 11.5 M
1 à 2% " " 13.- M
TRIM BY STERN: 1.5 M
V = 20 – 23 KN
3 à 4% MORE POWER = 10.- M
3 à 4% " " 11.5 M
8% " " 13.- M
VESSEL SHOULD SAIL WITH
EVEN KEEL TO 1 M IN THE HEAD
8.5 - 40.362 M³
- 49.487 M³
11.5 - 59.288 M²
- 69.527 M³
8 9 10 11 12 13 M DRAFT
NEDLL "HOUTMAN" + "HOORN"
SHP - V - DRAFT
NSMB
JAN '79


SPECIFIC FUEL CONSUMPTION
207
32450 hp
224,5
30000
%
100
130
140
150
160
170 GR/BHP·HR
25000 BHP
FUEL CONSUMPTION
IN SERVICE
1,09 HIGHER
5% CALORIC VALUE
3% MARGIN
1% SEPARATION
75
20000
138
148
155
162
MEASURED
15000
SPECIFIED
STAL-LAVAL
SERVICE
50
RTA84
70-87
RLB90
94-102
RND90 M 8 CYL
112
SULZER
122 RPM
10000
DELFT
DEJINA
HOORN
HOUTM.
25
5000
0
ABEL TASMAN
TURBINE AP 32/110
100
150 GR/BMP HR
200
250
300
350 GR./BHP/h.

NEDLLOYD "HOUTMAN + HOORN"

V KN	P_D TANK 10.6M	× GR/SHP.HR	× 24 =	TON/24 HR	× 1.18 =	SERVICE TON/24 HR
19	19100	× 168	× 24 =	77	× 1.18 =	91
19½	20400	167		82		97
20	22100	166		88		104½
20½	23900	165		95		112
21	25800	164		102		120
21½	28000	164		110		130
22	30400	163		119		140
22½	32800	163		128		151
23	35500	162		138		163
23½	38300	162		149		176
24	41000	× 162	× 24 =	160	× 1.18 =	188

P_D × 1.045 = TRIAL CONDITION

(RND 90M 122 RPM)

TRIAL CONDITION × 1.13

P_D × 1.18 2 YRS = SERVICE

NEDLLOYD "DEJIMA + DELFT"

V KN	P_D TANK 10.85 M	× GR/SHP.HR	× 24 =	TON/24 HR	× 1.13 =	SERVICE TON/24 HR
17	15400	× 170	× 24 =	63	× 1.13 =	71
17½	16800	168		67½		76
18	18100	167		72½		82
18½	19600	165		77½		87½
19	21000	163½		82		93
19½	22500	162		87½		99
20	24000	161		93		105
20½	25700	160		98½		112
21	27400	158		104		117½
21½	29300	157		110½		125
22	31300	156		117½		132
22½	33500	155½		125		141
23	36000	155		134		151½
23½	38800	155		144		163
24	42100	× 155	× 24 =	157	× 1.13 =	177

P_D × 1.045 = TRIAL CONDITION

(RND 90M 112 RPM)

TRIAL CONDITION × 1.08

P_D × 1.13 ½ YR = SERVICE

RPM – SPEED – POWER – FUEL CONSUMPTION NEDLLOYD "D"

13-23

17 NOV 81

2 x 8 RND 90 M SULZER

2 x 25440 BHP / 112 RPM

2 x 0,85 x 25440 SHP / 112 RPM

PROPELLER DESIGN

Ø = 6500 – LIPS – WEIGHT = 24,2 TON

P = 7310 MEAN 4 BLADES CONIAL

$\frac{EA}{F}$ = 0,573

	RPM	KN
DEAD SLOW	30	6,2
SLOW	45	9,3
HALF	50	10,3
	55	11,4
	60	12,4
	65	13,5
	70	14,5
	75	15,5

P_a TANK x 1,065 = TRIAL CONDITIONS x 1.02

DRAFT = 10.85 M

	RPM	KN	SHP	GR/SHP/HR	TON/DAY (14)	TON/1000'
FULL	80	16.6	15100	170	61.6	155
	81	16.8	15720	170	64.1	159
	82	17.0	16340	170	66.7	163
	83	17.3	16960	169	68.8	166
	84	17.5	17580	168	70.9	169
	85	17.7	18110	168	73.0	172
	86	17.9	18790	167	75.3	175
	87	18.1	19470	166	77.6	179
	88	18.4	20150	166	80.3	182
	89	18.6	20825	165	82.5	185
	90	18.8	21500	164	84.6	188
	91	19.0	22250	164	87.6	192
	92	19.2	23000	163	90.0	195
	93	19.5	23700	163	92.7	198
	94	19.7	24400	162	94.9	201
50%	95	19.9	25200	161	97.4	204
	96	20.2	26000	160	99.8	206
	97	20.4	26800	159	102.3	209
	98	20.6	27600	159	105.3	213
	99	20.8	28400	158	107.7	216
	100	21.1	29200	158	111.0	219
	101	21.3	30000	158	114.0	223
	102	21.5	30900	158	117.–	227
	103	21.7	31900	157	120.–	231
	104	21.9	32900	157	123.5	235
	105	22.1	33800	157	127.–	239
	106	22.3	34800	156	130.5	243
	107	22.5	35800	156	134.0	248
	108	22.7	36700	156	137.5	252
	109	22.9	37700	155	141.–	255
	110	23.1	38800	155	144.5	260
	111	23.3	39900	155	148.4	265
	112	23.5	40900	155	152.1	270
	113	23.7	42000	155	156.2	275
85% MCR	114	23.9	43100	155	160.3	280
	115	24.1	44200	156	165.5	286
	116					

DRAFT = 12.65 M

RPM	KN	SHP	GR/SHP/HR	TON/DAY (14)	TON/1000'
80	16.4	15700	170	64.1	163
81	16.6	16340	170	66.7	167
82	16.8	16980	169	68.9	171
83	17.0	17620	168	71.0	174
84	17.2	18260	167	73.2	177
85	17.4	18900	166	75.3	180
86	17.6	19610	166	78.1	184
87	17.8	20320	165	80.5	188
88	18.0	21030	165	83.3	193
89	18.2	21740	164	85.6	196
90	18.4	22450	164	88.4	200
91	18.6	23240	163	90.9	204
92	18.8 S	24030	163	94.0	208
93	19.0 S	24820	162	96.5	211
94	19.2 S	25610	161	99.0	214
95	19.5	26400	160	101.4	217
96	19.7	27270	159	104.1	220
97	19.9	28140	158	106.7	223
98	20.1	29010	157	109.3	227
99	20.3	29880	157	112.6	231
100	20.5	30750	156	115.1	234
101	20.7	31720	156	118.8	238
102	20.9	32690	155	121.6	242
103	21.1	33660	155	125.2	247
104	21.3	34630	155	128.8	252
105	21.4 S	35600	155	132.4	257
106	21.6 S	36700	155	136.5	263
107	21.8 S	37800	155	140.6	268
108	22.0	38900	155	144.7	274
109	22.2	40000	155	148.8	279
110	22.4	41100	155	152.9	284
111	22.5 S	42260	155	157.2	290
112	22.7 S	43420	155	161.5	296
113	22.9	44580	156	166.9	304
114					
115					
116					

mmss "NEDLLOYD HOUTMAN+HOORN"

SLAGEN LIJSTJE = PROPULSION PERFORMANCE

P_D TANK incl.app. x 1.045 = TRIAL CONDITIONS

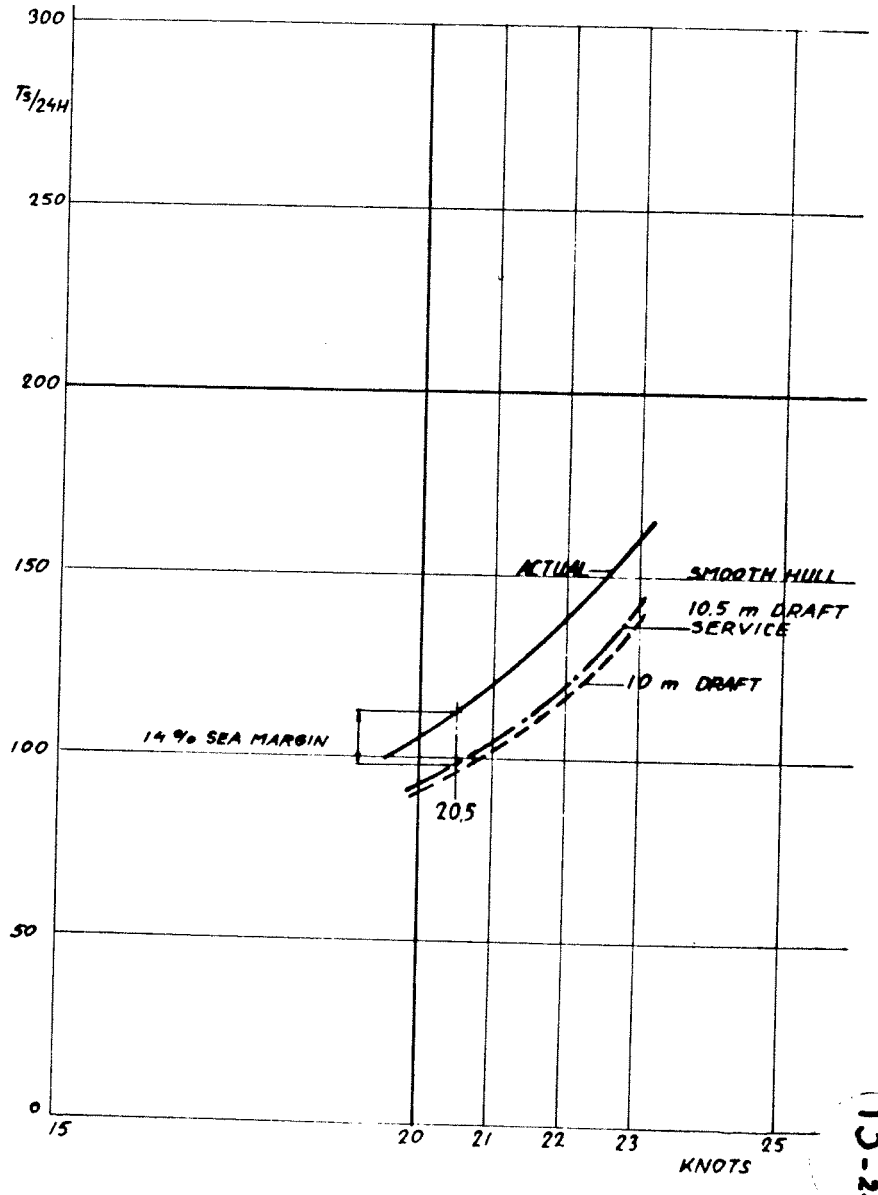
27/9 1977

Sulzer 8RND90

R.P.M.	SPEED, in knots, with new ship, clean hull, smooth seas at draughts:- 8.5 M	10 M	11.5 M	13 M	S.H.P., total, at draughts:- 8.5 M	10 M	11.5 M	13 M	HEAVY FUEL CONSUMPTION Gr./SHP/Hr.	Tons / 24 hours at draughts: 8.5 M	10 M	11.5 M	13 M
DEAD SLOW 30	5.9	5.9	5.8	5.7									
35	6.9	6.85	6.75	6.6									
40	7.9	7.82	7.74	7.55									
SLOW .. 45	8.9	8.8	8.7	8.5									
50	9.85	9.72	9.6	9.42									
HALF .. 55	10.85	10.7	10.55	10.35									
60	11.85	11.7	11.55	11.32									
65	12.82	12.66	12.5	12.22									
70	13.86	13.7	13.53	13.24									
75	14.85	14.65	14.46	14.13									
80	15.83	15.67	15.46	15.13									
FULL .. 85	16.84	16.64	16.42	16.06									
90	17.84	17.64	16.41	17.00	15500	15500	15700	16700					
91	18.04	17.8	17.56	17.14									
92	18.26	18.00	17.79	17.35									
93	18.45	18.20	17.95	17.52									
94	18.65	18.42	18.16	17.73									
95	18.87	18.62	18.36	17.92	18200	18200	18600	19800					
96	19.04	18.80	18.54	18.07									
97	19.25	19.04	18.74	18.24									
98	19.45	19.22	18.92	18.42									
99	19.63	19.42	19.10	18.60									
100	19.9	19.65	19.34	18.80	21300	21300	21900	23200	165	84.4	84.4	86.7	92.6
101	20.08	19.8	19.5	18.95	22000	22000	22500	23900	165	87.2	87.2	88.2	94.7
102	20.25	20.0	19.65	19.10	22700	22700	23200	24500	165	90.0	90.0	92.0	97.0
103	20.45	20.2	19.85	19.30	23300	23300	23900	25300	164	91.7	91.7	94.0	99.5
104	20.65	20.4	20.03	19.50	24000	24000	24600	26100	164	94.5	94.5	96.8	102.7
105	20.86	20.6	20.22	19.65	24700	24700	25300	26800	164	97.2	97.2	99.5	105.5
106	21.03	20.8	20.41	19.82	25500	25500	26100	27600	163	99.7	99.7	102.0	108.0
107	21.23	20.97	20.58	19.98	26100	26200	26900	28500	163	102.0	102.4	105.2	111.5
108	21.45	21.2	20.8	20.2	26800	26900	27700	29300	163	104.8	105.2	108.2	114.5
109	21.62	21.34	20.93	20.3	27600	27800	28500	30100	163	108.0	108.8	111.4	117.7
110	21.90	21.60	21.15	20.52	28400	28600	29400	31100	163	111.0	111.8	114.9	121.6
111	22.05	21.74	21.3	20.63	29100	29300	30300	32000	163	113.8	114.1	118.4	125.2
112	22.27	21.94	21.48	20.85	30000	30200	31200	33000	162	116.7	117.5	121.2	128.2
113	22.46	22.08	21.65	21.0	30800	31100	32200	34000	162	119.8	121.0	125.2	132.2
114	22.67	22.3	21.84	21.15	31600	31900	33000	34900	162	123.0	124.0	128.2	135.7
115	22.90	22.48	22.01	21.34	32500	32800	33900	36000	162	126.4	127.5	131.9	140.0
116	23.1	22.67	22.19	21.5	33400	33[illegible]0	34900	37100	162	129.8	131.5	135.8	141.2
117	23.27	22.85	22.34	21.66	34300	34700	35900	38100	162	133.3	135.0	139.6	148.0
118	23.48	23.04	22.52	21.83	35200	35600	36800	39200	162	137.0	138.5	143.0	152.3
119	23.67	23.22	22.69	21.97	36[illegible]0	36600	37900	40300	162	140.8	142.3	147.4	156.8
120	23.90	23.41	22.89	22.14	37200	37700	39000	41400	162	144.7	146.6	151.2	161.0
121	24.03	23.55	23.04	22.28	38200	38800	40100	42600	162	148.5	150.8	156.0	165.8
122	24.22	23.73	23.21	22.43	39200	39900	41100	43700	162	152.5	155.2	159.8	170.0
123	24.41	23.89	23.36	22.57	40300	40900	42200	44900	163	157.7	160.0	165.0	175.4
124	24.6				41300								

SEA MARGIN

("NEDLLOYD HOORN"— IN FAR EAST SVCE.)



"NEDLLOYD ROUEN" + "ROSARIO"

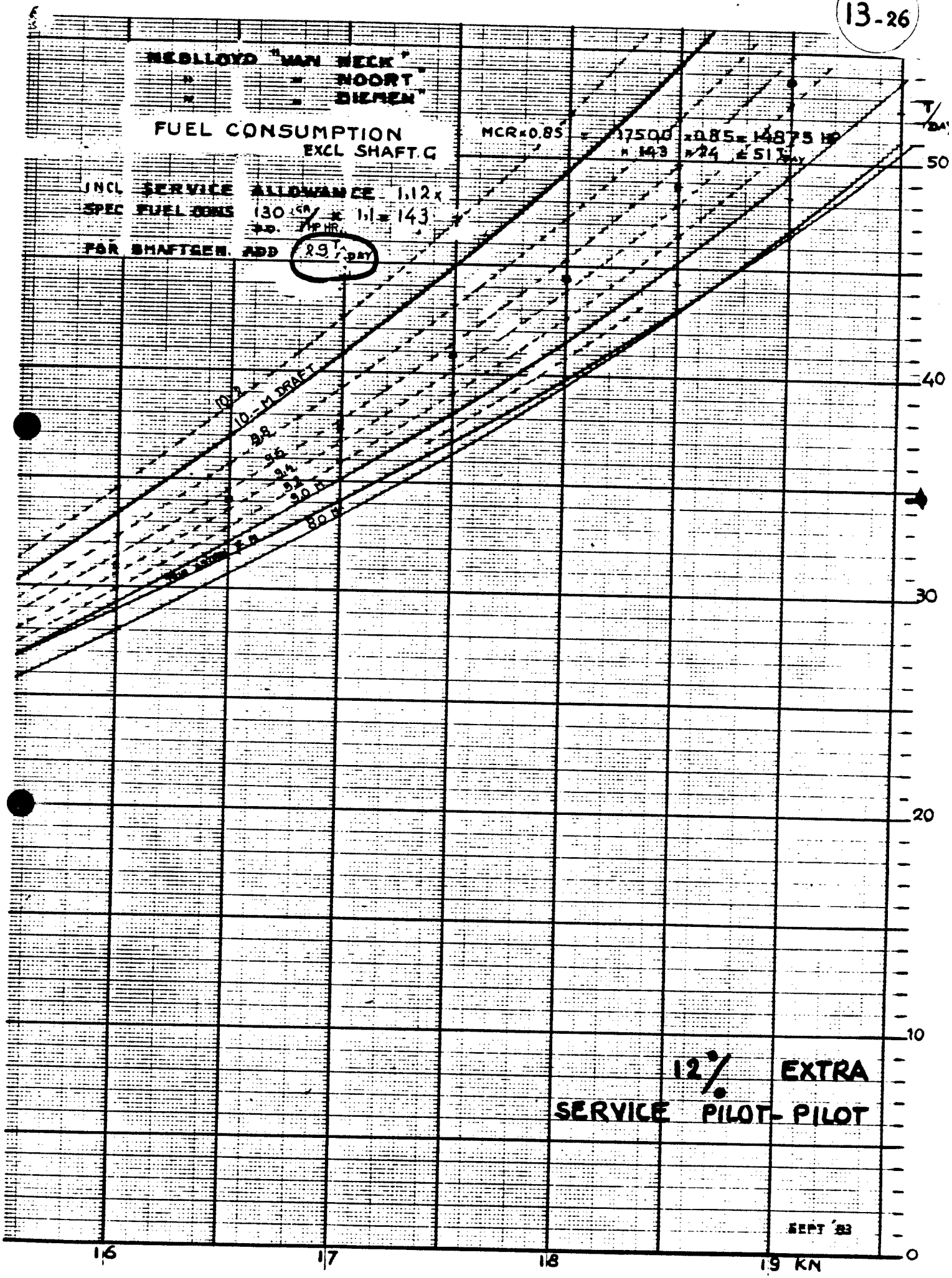
RPM - SPEED - SHP - FUEL UNDER OPTIMAL CONDITIONS

		TRIALS 27 SEPT. '78 BALLAST ↓ DRAFT 6.84 F = 6.46 0.1 HOGG A = 7.43				8.4 M				DESIGN DRAFT 10.0 M MODEL BASIN WAGENINGEN TRIAL CONDITIONS				MAX. DRAFT 10.7 M + EXTRA SEA MARGIN				
	RPM	KN				SMOOTH HULL CALM SEA				SMOOTH HULL CALM SEA				ESTIMATED				
DEAD SLOW	30	5.1																
TORSION	33	6.0																
CRITICAL	34																	
	40	6.8	SMOOTH HULL															
SLOW	45	7.7	CALM SEA															
	50	8.5																
	55	9.4																
HALF	60	10.3																
	65	11.2																
	70	12.0																
	75	12.9																
	80	13.7																
FULL	85	14.6																
	90	15.5																
	95	16.3	SHP	ton/day	ton/1000'	KN	SHP	ton/day	ton/1000'	KN	SHP	ton/day	ton/1000'	KN	SHP	ton/day	ton/1000'	
	100	16.8	11,600	46	114	17.0	11,900	47	115	17.2	12,200	48	117	17.1	12,500	50	121	
	101	17.1	12,000	47½	116	17.2	12,300	49	118	17.3	12,600	50	120	17.2	12,900	51	123	
	102	17.3	12,400	49	118	17.5	12,700	50	120	17.5	13,000	51½	122	17.3	13,300	53	126	
	103	17.6	12,700	50	119	17.7	13,000	51	121	17.6	13,300	52½	124	17.5	13,700	54	128	
	104	17.8	13,100	51½	121	17.9	13,400	53	123	17.8	13,700	54	126	17.6	14,100	55½	130	
	105	18.1	13,500	53	122	18.1	13,800	54½	125	17.9	14,100	55½	129	17.7	14,500	57	133	
	106	18.3	13,900	54½	124	18.2	14,200	55½	127	18.1	14,500	57	131	17.9	14,900	58	136	
	107	18.6	14,300	56	126	18.4	14,600	57	129	18.2	15,000	58½	134	18.0	15,400	60	139	
	108	18.8	14,700	57½	128	18.6	15,000	59	131	18.4	15,400	60½	136	18.1	15,800	62	142	
	109	19.0	15,100	59	130	18.8	15,400	60	133	18.5	15,800	62	139	18.2	16,200	63½	145	
	110	19.2	15,500	60½	132	19.0	15,900	62	136	18.7	16,300	64	142	18.4	16,700	65	148	
	111	19.4	15,900	62	134	19.1	16,300	64	139	18.8	16,700	65½	145	18.5	17,100	67	151	
	112	19.6	16,400	64	136	19.3	16,800	65½	141	19.0	17,200	67	147	18.6	17,600	68½	154	
	113	19.8	16,800	65½	138	19.5	17,200	67	143	19.1	17,600	68½	149	18.7	18,100	70½	157	
	114	19.9	17,200	67	140	19.6	17,600	68½	146	19.3	18,100	70½	152	18.9	18,600	72½	160	
	115	20.1	17,700	69	143	19.7	18,100	70½	149	19.4	18,600	72½	155	19.0	19,100	74	163	
	116	20.3	18,100	70½	145	19.9	18,600	72½	152	19.5	19,100	74	158	19.1	19,600	76	166	
	117	20.5	18,600	72½	147	20.1	19,100	74	155	19.7	19,600	76	161	19.2	20,100	78	169	
	118	20.6	19,100	74	150	20.3	19,600	76	157	19.8	20,100	78	164	19.3	20,600	80	173	
	119	20.8	19,600	76	153	20.4	20,100	78	160	19.9	20,600	80	167	19.4	21,100	82	176	
	120	21.0	20,100	78	155	20.5	20,600	80	163	20.1	21,100	82	170	19.5	21,700	84½	180	
	121	21.2	20,700	80½	158	20.7	21,200	82½	166	20.2	21,700	84½	174	19.6	22,200	86½	184	
	122	21.3	21,200	82½	161	20.9	21,700	84½	169	20.3	22,200	86½	177	19.8	22,800	88½	187	
	123	21.5	21,700	85	165	21.0	22,200	87	172	20.5	22,700	89	181	19.9	23,300	91	191	85%
	124	21.6	22,200	87	168	21.1	22,700	89	175	20.6	23,300	91	185	20.0	23,900	93½	195	
85%	125	21.8	22,800	89½	171	21.3	23,300	91	179	20.7	23,900	93½	189	20.1	24,500	96	199	
	126	21.9	23,300	92	174	21.4	23,900	94	183	20.9	24,500	96½	192	20.2	25,100	98½	203	
	127	22.0	23,900	94	178	21.5	24,400	96	186	21.0	25,000	98½	195	20.3	25,600	101	207	
	128	22.1	24,400	96	181	21.7	25,000	98½	189	21.1	25,600	101	199	20.4	26,200	103½	211	
	129	22.2	25,000	98½	185	21.8	25,600	101	192	21.2	26,200	103½	203					

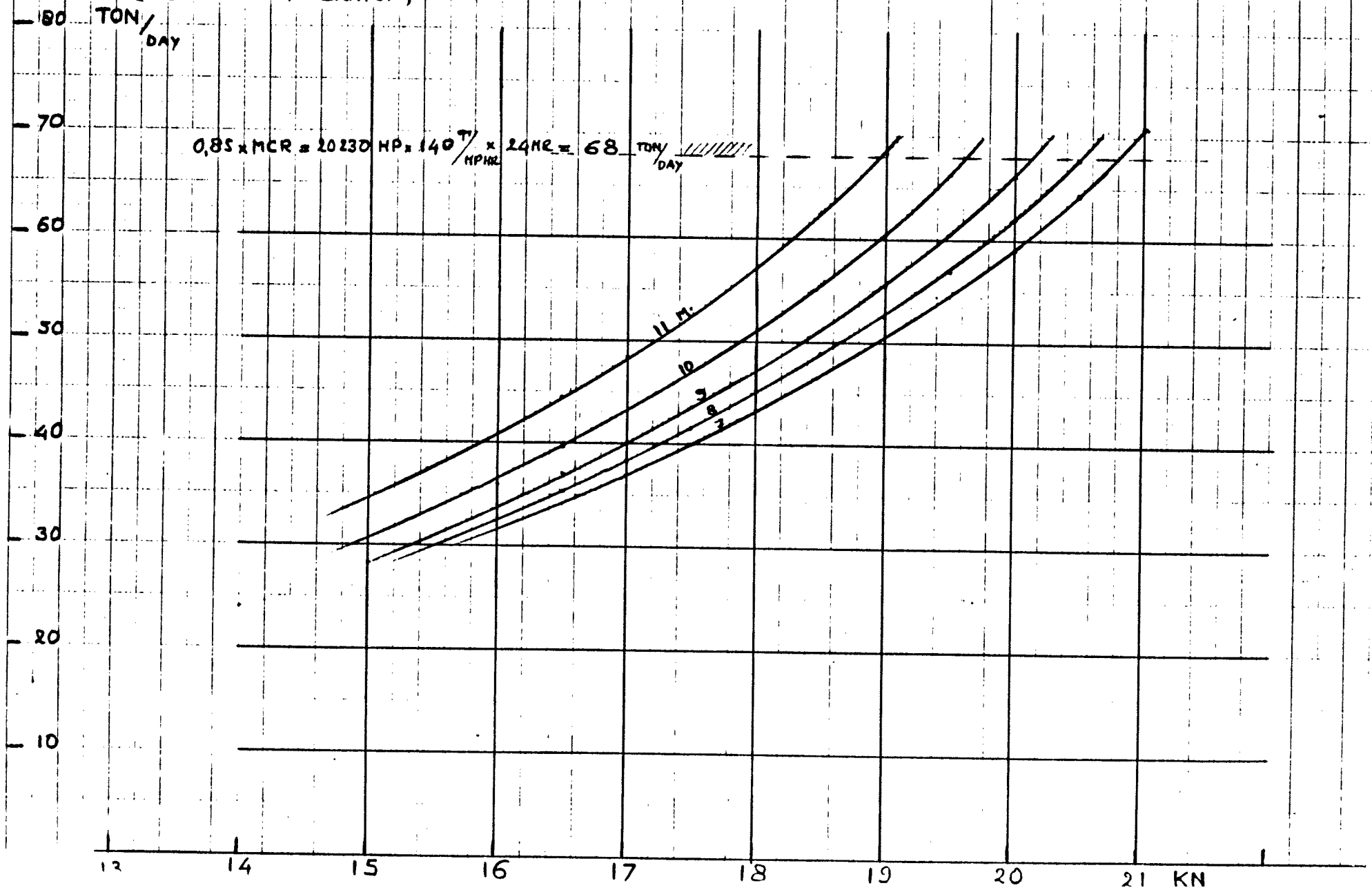
Lips-propeller ø = 6500, 4 BL — Cunial — 26.2 T
pitch mean = 5695, Fo/F = 0.728

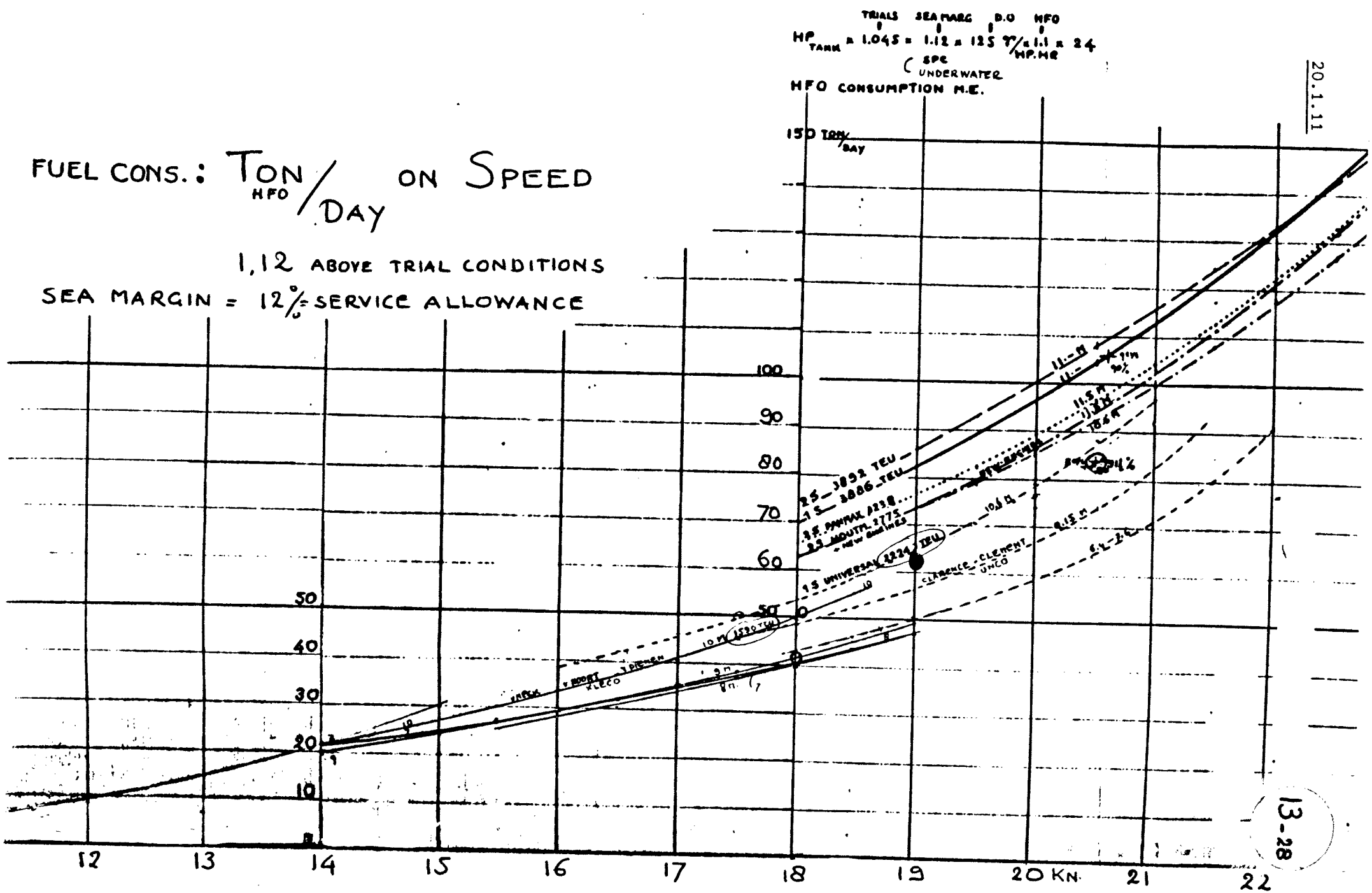
NEDLLOYD ROUEN NKK No. 67 DEL 12 OCT 78
NEDLLOYD ROSARIO NKK No. 68 DEL 19 JAN 79

L-OA = 212.10 M
L-PP = 198.80 M
B-MLD = 32.24 M
H-MLD = 20.40 M

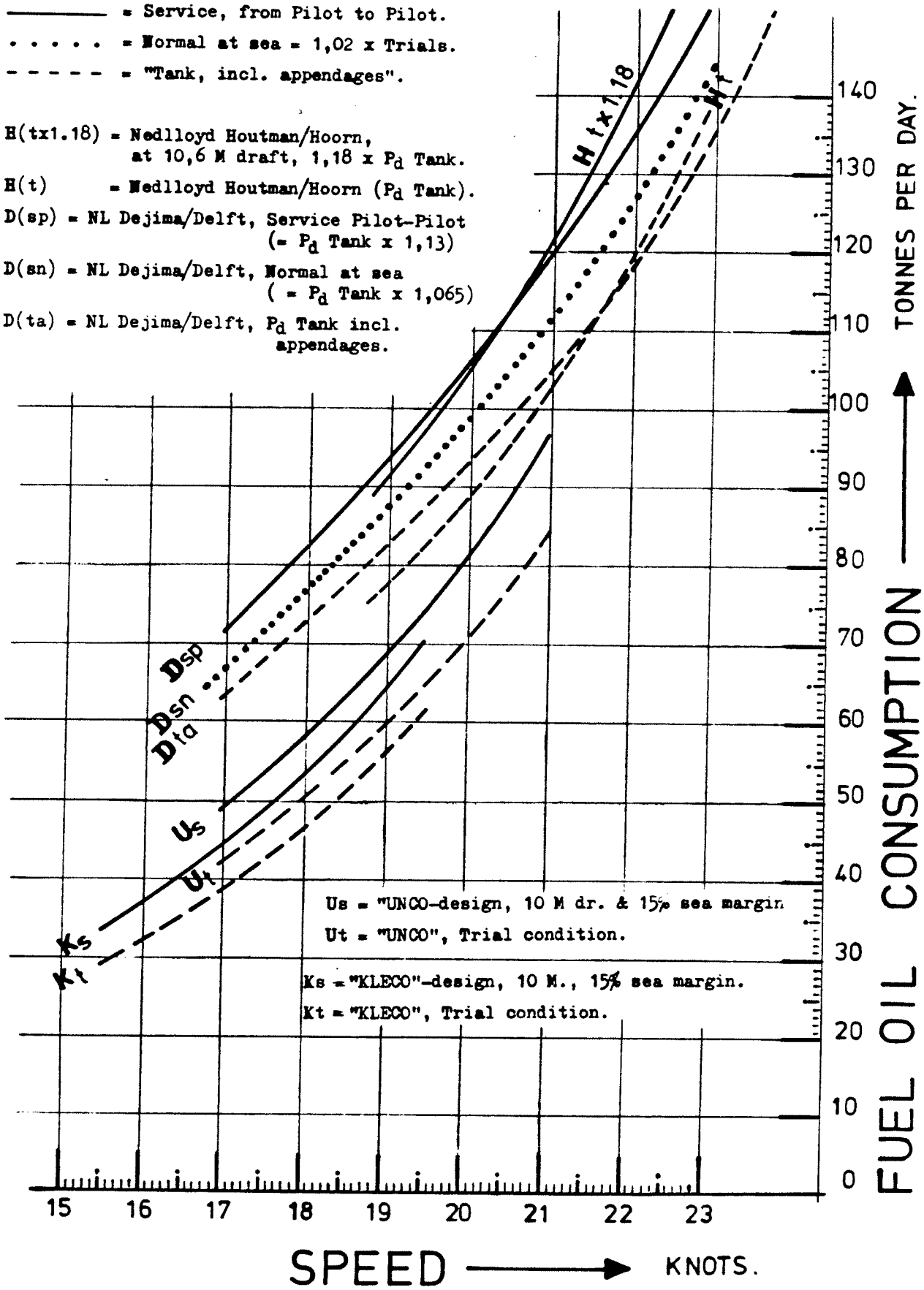

NEDLLOYD "VAN NECK"
" " NOORT"
" " DIEMEN"
FUEL CONSUMPTION
EXCL SHAFT.G
MCR×0.85 = 17500 ×0.85 = 14875 HP
" 143 " 24 = 51 T/DAY
INCL SERVICE ALLOWANCE 1.12×
SPEC FUEL CONS 130 GM/HP HR. × 1.1 = 143
FOR SHAFTGEN. ADD 1.9 T/DAY
10.2
10.-M DRAFT
9.8
9.6
9.4
9.2
9.0 M
8.0 M
T/DAY
50
40
30
20
10
0
12% EXTRA
SERVICE PILOT-PILOT
SEPT '83
1.6
1.7
1.8
1.9 KN

FUEL CONSUMPTION - HFO : 12% SEA MARGIN = 12% SERVICE ALLOWANCE ABOVE TRIAL CONDITIONS.
(EXCL SHAFT GENERATOR)



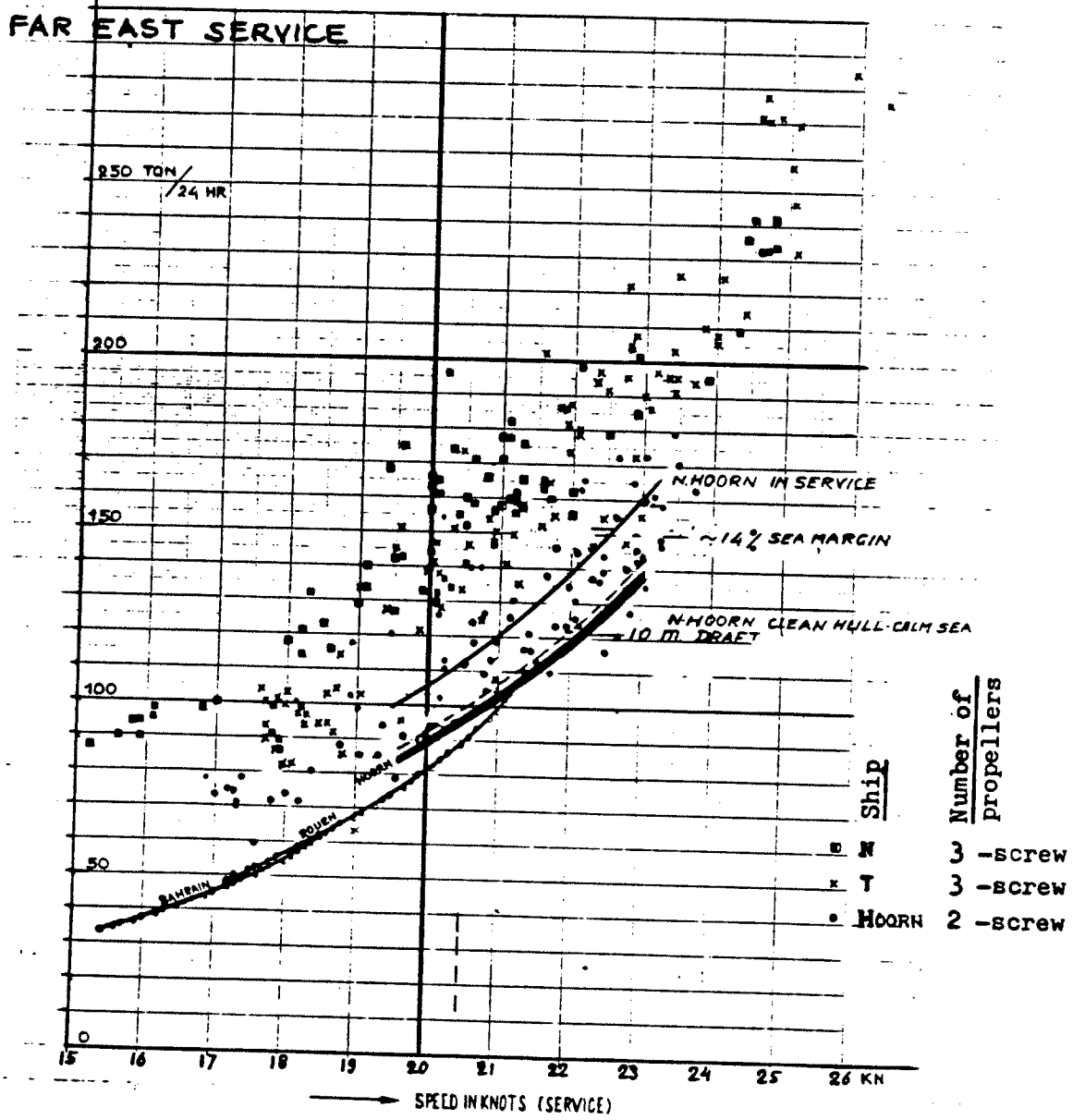

FUEL CONS.: TON HFO / DAY ON SPEED
1,12 ABOVE TRIAL CONDITIONS
SEA MARGIN = 12% = SERVICE ALLOWANCE
TRIALS SEA MARG D.O HFO
HP TANK = 1.045 = 1.12 x 125 % x 1.1 x 24
SPC UNDERWATER
HP.HR
HFO CONSUMPTION M.E.
150 TON/DAY
100
90
80
70
60
50
40
30
20
10
12
13
14
15
16
17
18
19
20 KN.
21
22
25 — 3892 TEU
1 S — 3886 TEU
3 S. PANAMAX 4238
3 S. MOUTH. 2775
NEW BUILDS
1 S UNIVERSAL 2224 TEU
CLARENCE - CLEMENT
UNCO
1520 TEU
11.5 M
10.5 M
9.5 M
8.5 M
7.5 M
7 M
6.5 M
6 M
10 M
9 M
8 M
4%
13-28

FUEL OIL CONSUMPTION - TO - SPEED RATIO



COMPARATIVE FUEL CONSUMPTION FIGURES OF SELECTED CONTAINERVESSELS UNDER VARYING PROPULSIVE CONFIGURATIONS .

EUROPE - FAR EAST SERVICE



DAILY FUEL CONSUMPTION IN TON/24 HR

		3 S-DIESEL (CENTRE PROP + C.P.)				2 S-STEAM				2 S-DIESEL (SHORTER VESSELS)			
		< SISTERS >				DELFT SEDIMA < SISTERS >		KORREAN		HOORN NOUTHAM < SISTERS >			
	Speed	J	SL	T	N	DL	DM	K	aver.	H	P	S	aver.
	23	173	180	185	202	251	261	300		160	185	185	
	22,5	161	169	172	188	239	247	286		149	174	175	
	22	151	159	160	176	227	235	273		139	164	165	
	21,5	142	149	149	165	216	223	260		129	154	156	
	21	134	139	138	155	205	212	248		120	145	147	
SERVICE :	20,5 Kn	124	130	128	146	196	202	237		112	136	139	
	20	117	121	118	137	187	192	226		105	128	131	
		B+W		MITSUI	GÖTA	BR. VULKAN		NDW		RSV	SUMITOMO		
		B+W ENG			GÖTA	STAL LAVAL TURB.		AAE TURB.		SULZ	SULZ + TURBOCHR.		
	Const. cons. /day	10 Ts DO				in port: 80 Ts FO		80		10 Ts DO prt: 7 Ts FO	At sea: 2,5 Ts DO in port: 10 Ts DO		

TRIPLE-SCREW : TWIN-SCREW ..... STEAM V.A.V. DIESEL

Performance "Nedlloyd Delft" als motorschip

Hieronder volgen uitsluitend de trajecten Jedda/Port Said-Singapore via versa.

Datum	---) (---	Displ. TON	Speed V	F.O. TON/DAY	Prest.getal $\frac{DISPL^{2/3} . V^3}{TON/DAY}$	F.O. als turbine	Besparing
28-1-82	---)	53937	21,8	135	110	210	75
26-2-82	(---	55118	21,7	116	128	210	94
6-4-82	---)	62746	21,6	134	119	222	88
7-5-82	(---	59646	21,1	121	119	205	84
17-6-82	---)	55856	21,2	113	123	203	90
16-7-82	(---	56742	21,0	128	107	202	74
25-8-82	---)	57480	20,5	106	121	193	87
24-9-82	(---	59843	21,4	125	120	214	89
3-11-82	---)	60000	22,1	150	110	229	79
2-12-82	(---	58000	21,6	125	121	215	90
14-1-83	---)	68000	22,2	167	109	247	80
10-2-83	(---	60100	22,5	144	121	238	94
23-3-83	---)	67100	20,9	125	121	213	88
22-4-83	(---	60300	20,7	113	120	198	85
1-6-83	---)	68200	20,3	127	110	200	73
30-6-83	(---	56990	22,4	166	100	228	62
Gemiddeld		60000	21,4	131	115	214	83

FTA
1-9-83

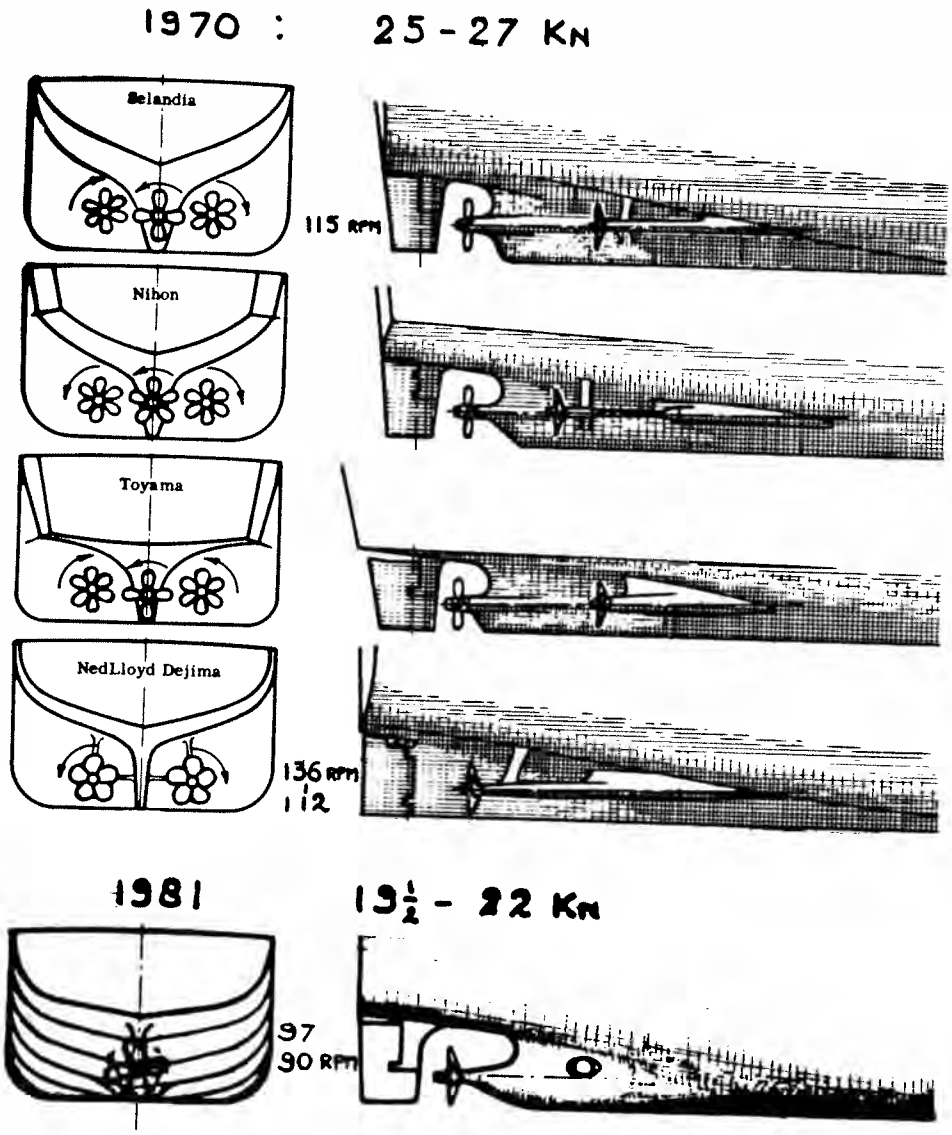
Performance "Nedlloyd Dejima" als motorschip

Hieronder volgen uitsluitend de trajecten Jedda/Port Said-Singapore via versa.

Datum	---) (---	Displ. TON	Speed V	F.O. TON/DAY	Prest.getal $\frac{DISPL^{2/3} . V^3}{TON/DAY}$	F.O. als turbine	Besparing
21-5-82	(---	60827	21,0	115	125	205	90
29-6-82	---)	62992	22,4	144	124	242	98
29-7-82	(---	60600	21,8	147	109	223	76
08-9-82	---)	55413	20,8	119	110	195	76
7-10-82	(---	56397	21,9	126	122	220	94
17-11-82	---)	58464	20,8	113	119	199	86
7-12-82	(---	55118	21,0	105	127	198	93
26-1-83	---)	51870	20,0	112	99	177	65
24-2-83	(---	63090	22,3	138	127	240	102
6-4-83	---)	65000	21,6	146	112	226	80
5-5-83	(---	63693	22,7	154	121	253	99
15-6-83	---)	65800	21,2	123	126	217	94
14-7-83	(---	62500	22,3	152	116	239	87
Gemiddeld		60136	21,5	130	117	218	88

FTA

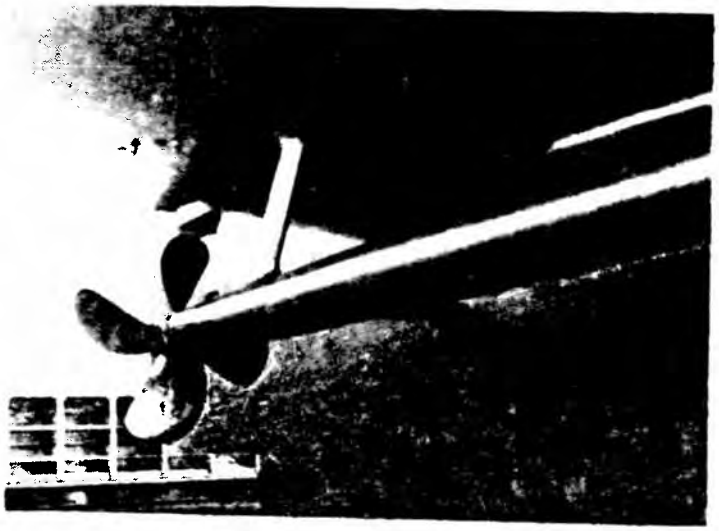
1-9-83



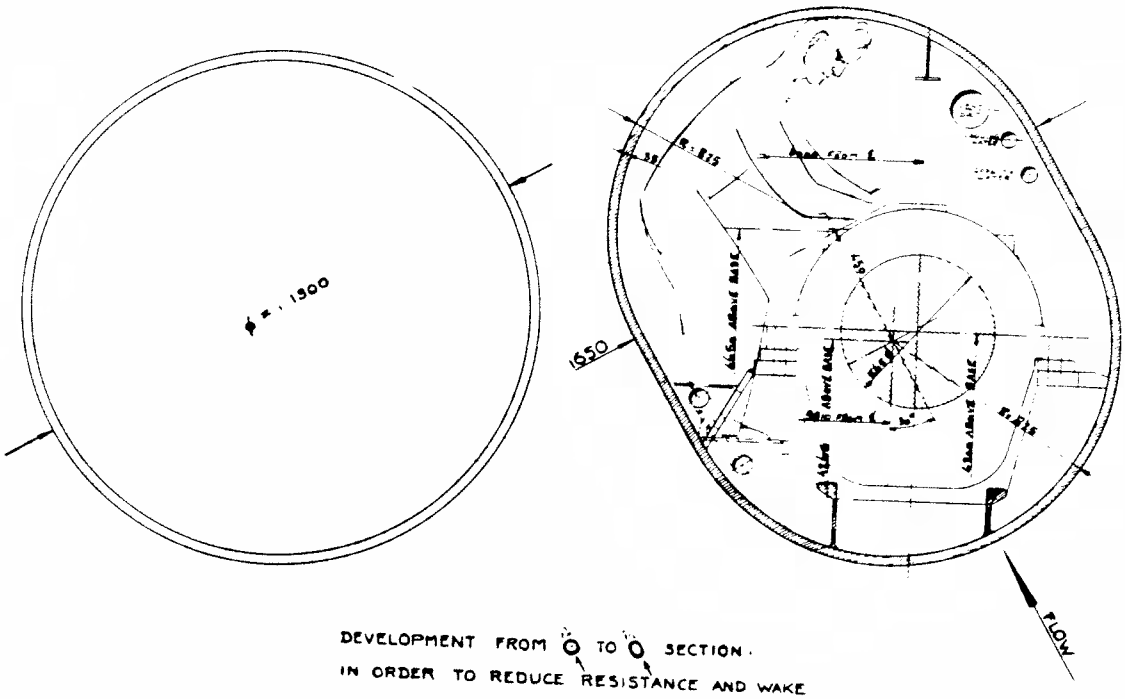
TREND IN PROPULSION OF LARGE CONTAINERSHIPS

Nice triple screw arrangement of 'Selandia'- Far East container.

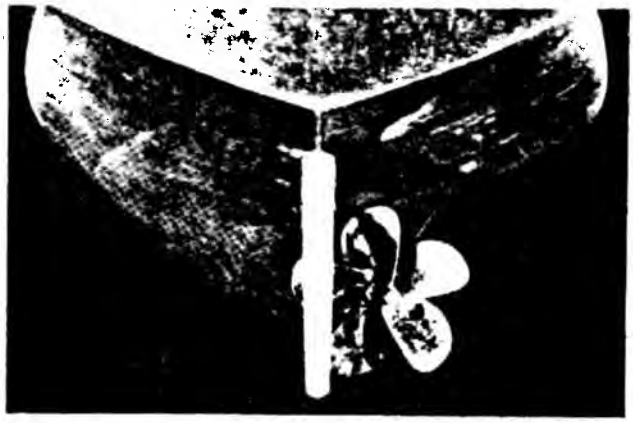
Fat bossings of twin screw passenger-liner.

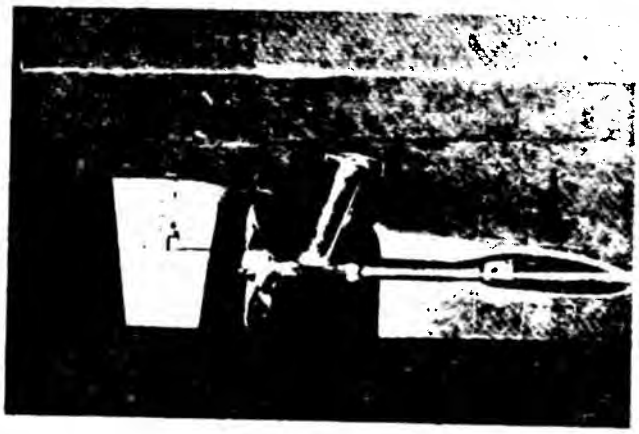


Slender pipe with struts.
Pipe without internal stiffening !
Internal access up to the sterntube seal.

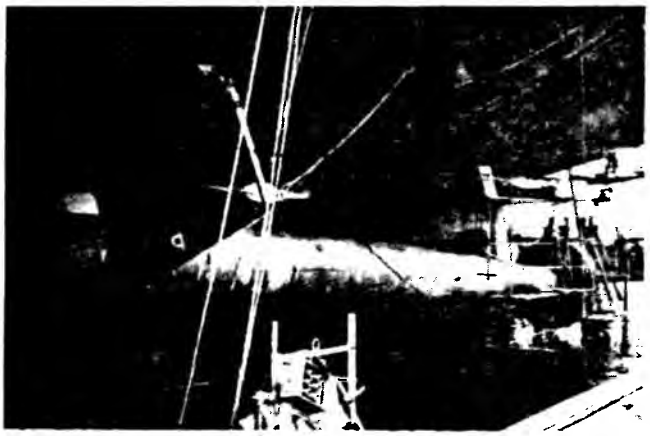


DEVELOPMENT FROM O TO O SECTION.
IN ORDER TO REDUCE RESISTANCE AND WAKE

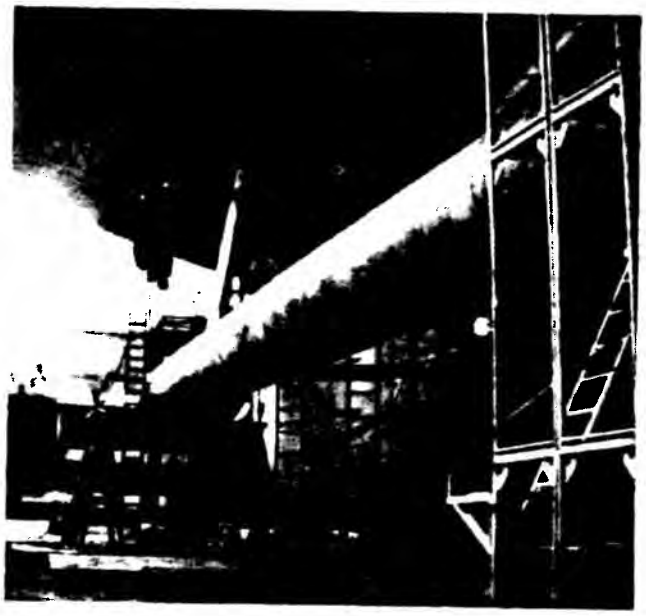




Free flow propeller on single screw Ro-Ro vessel. Designed by Transatlantic in combination with SSPA. Slow turning diesel engine out of centreline because of 3 Ro-Ro lanes in cross section.



Twin screw tube in brackets. Far East containerships.



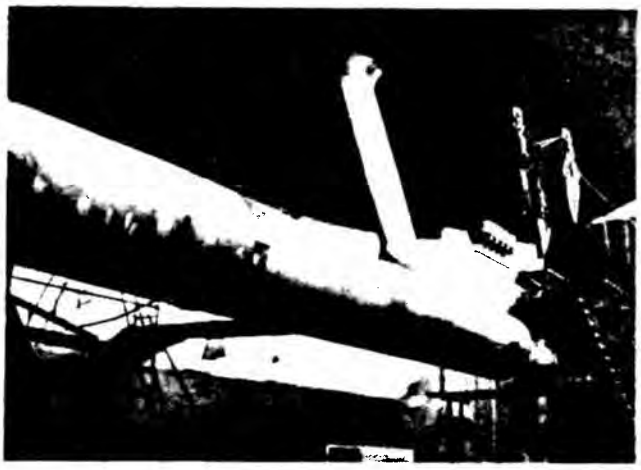
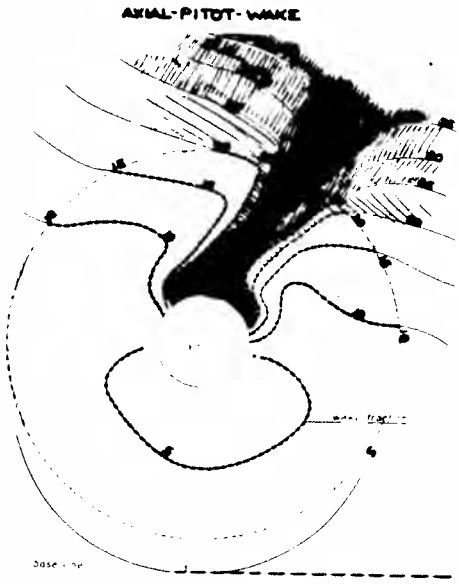
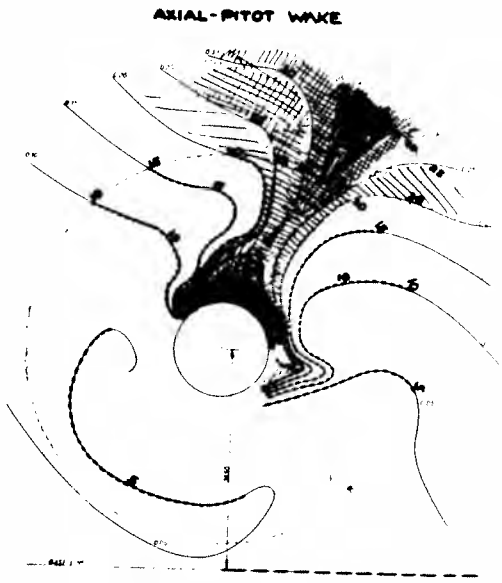


Figure 38. Some typical shafting arrangements.

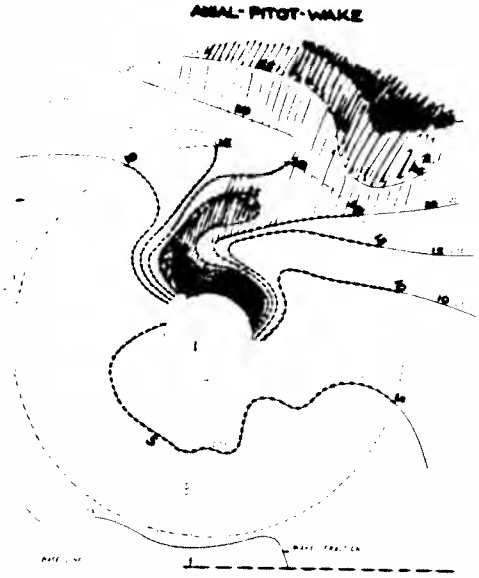
Sea-Land twin screw container.
- U.S construction of open shafts,
- Careful attention should be paid to fairing-caps.



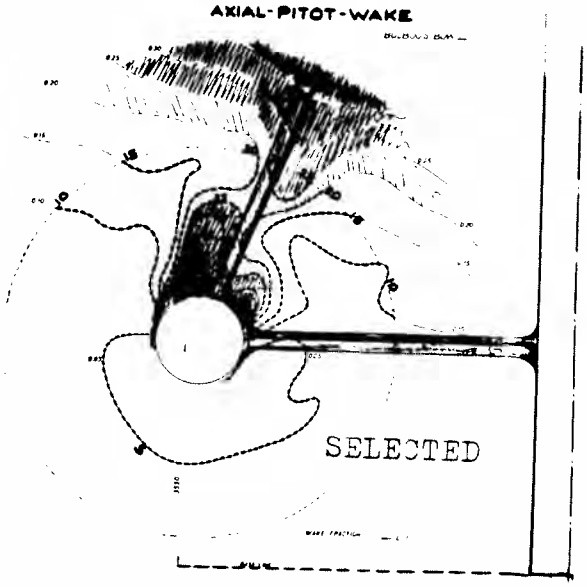
Bossing with small clearance ahead of propeller.



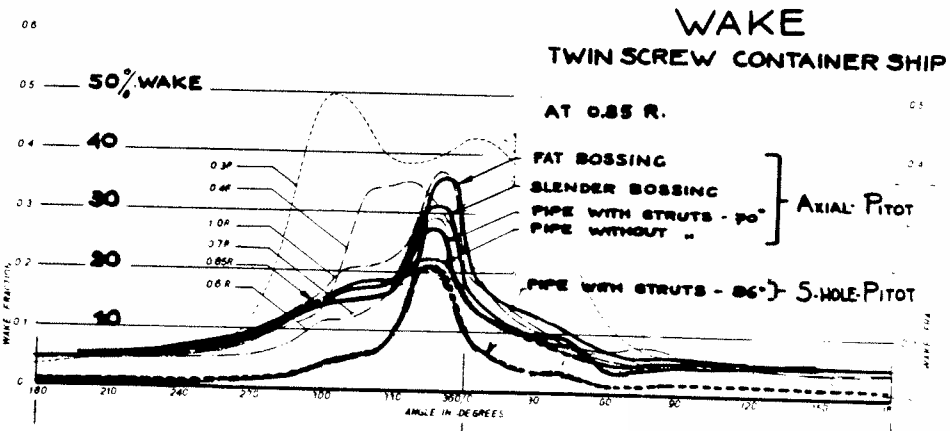
Bossing with increased clearance.



Pipe without struts.



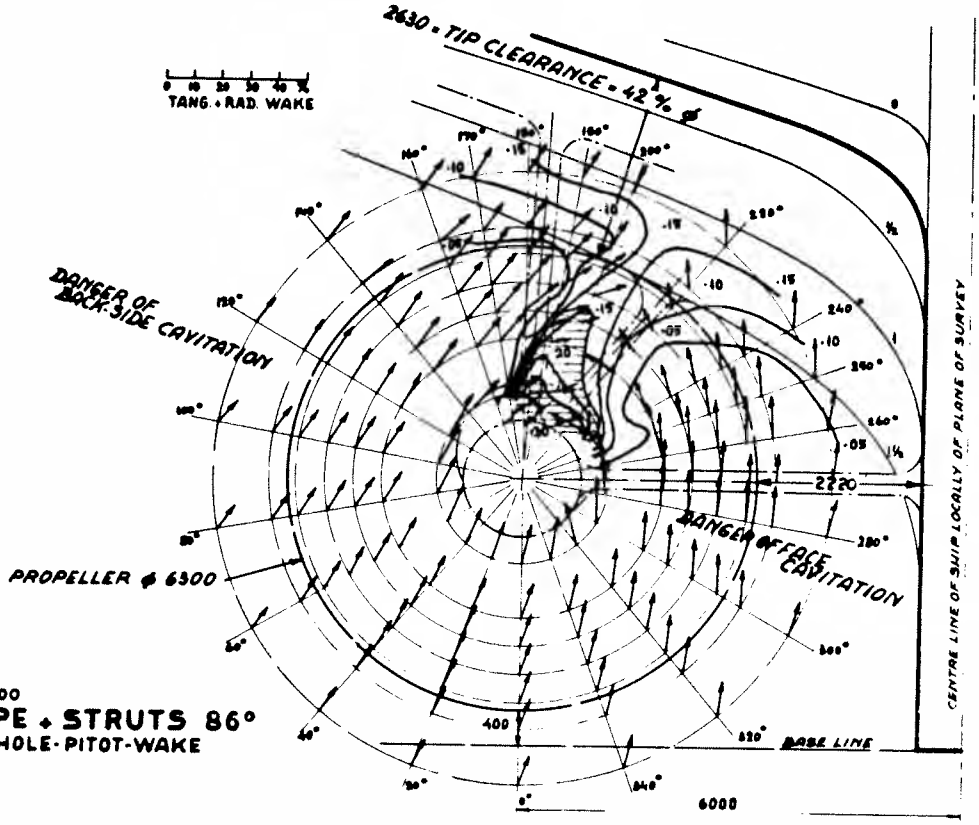
Pipe with struts 70°.



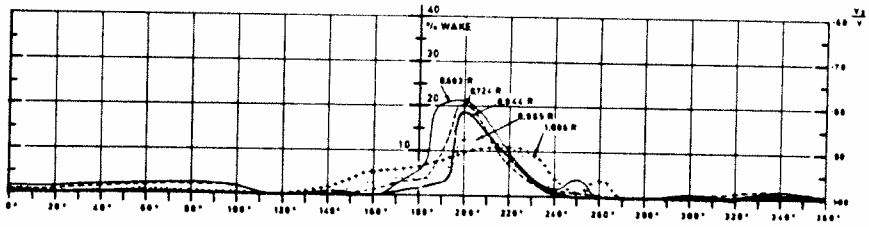
Note small influence of struts on wake.

TESTS WITH BOSSINGS AND STRUTS

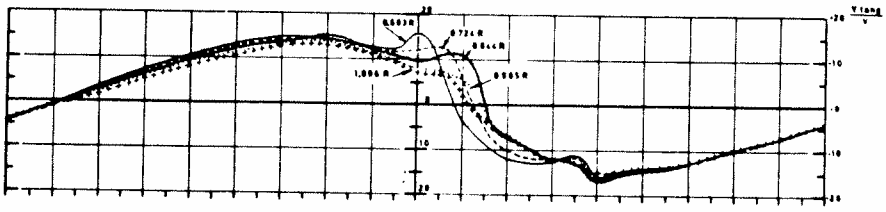
WAKE "NEDLLOYD DEJIMA" MODEL .



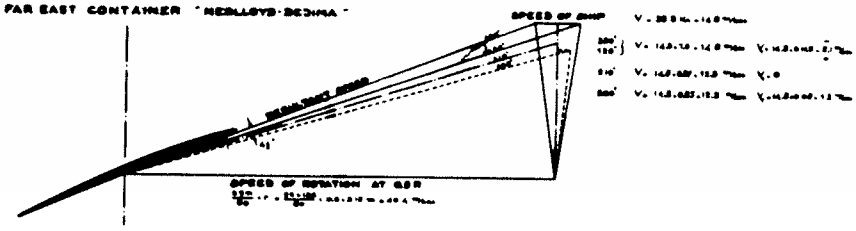
axial



tangential

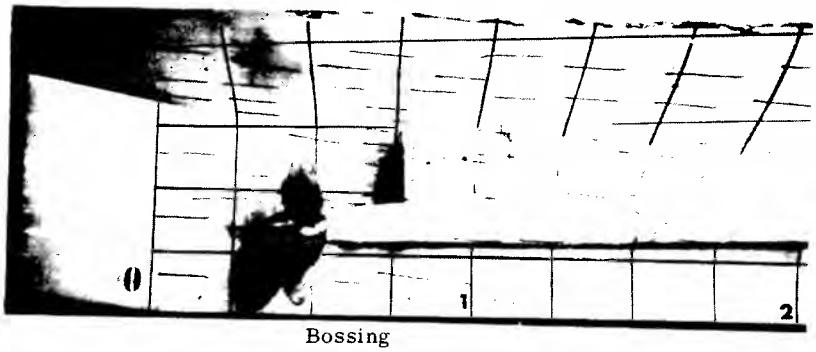


RESULTATEN VAN 3-DIMENSIONALE VOLGSTROOMMETINGEN AAN HET "NEDLLOYD DEJIMA" MODEL

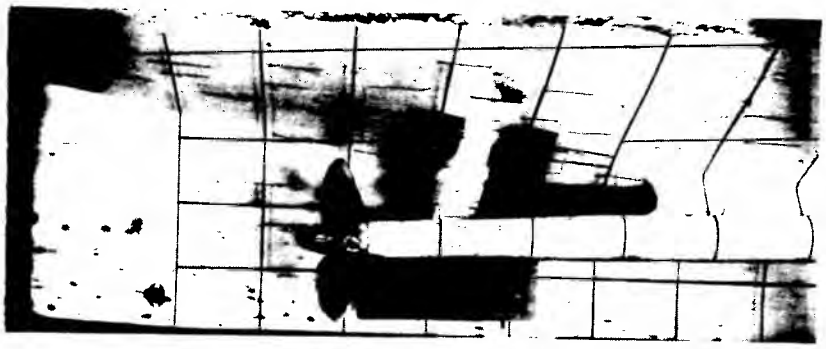


Far East container ship NedLloyd 'Dejima'. Results of three dimensional wake measurements.

Streamline tests at Berlin Model Basin.



Bossing



Pipe with struts. 70° angle.

Pipe with struts. 86° angle.

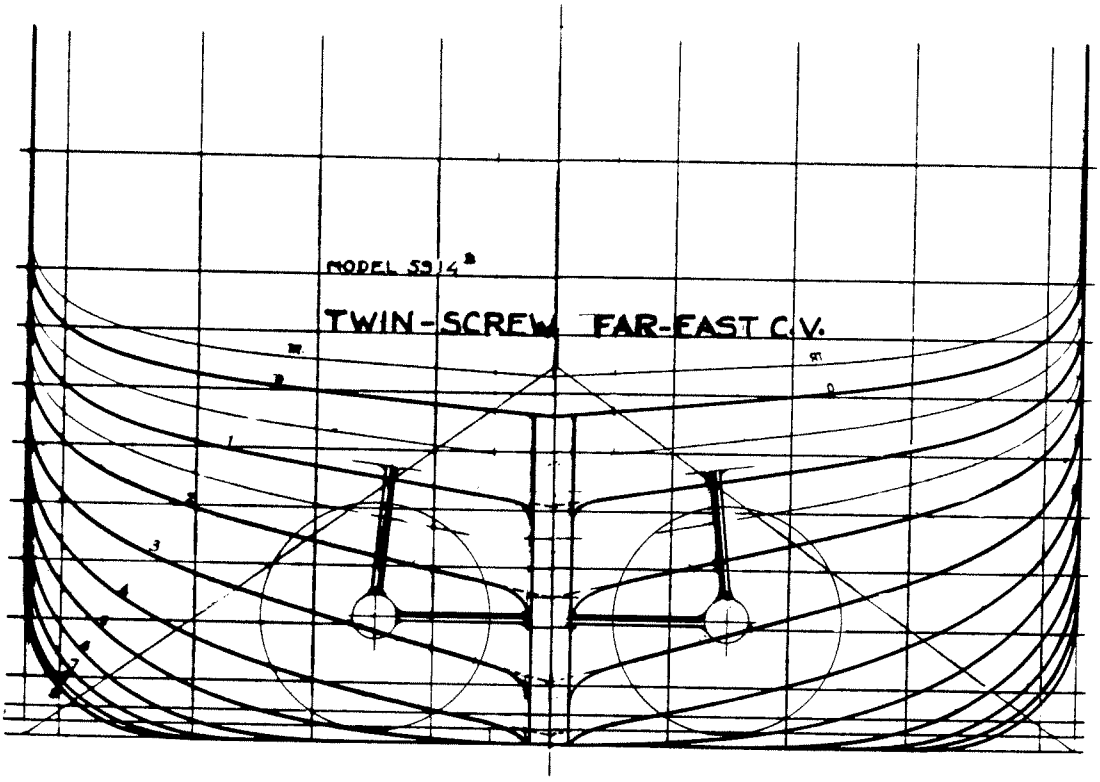
Paint test at N. S. M. B.

Figure 36A. Far East container ship NedLloyd 'Dejima'/'Bremen Express'.

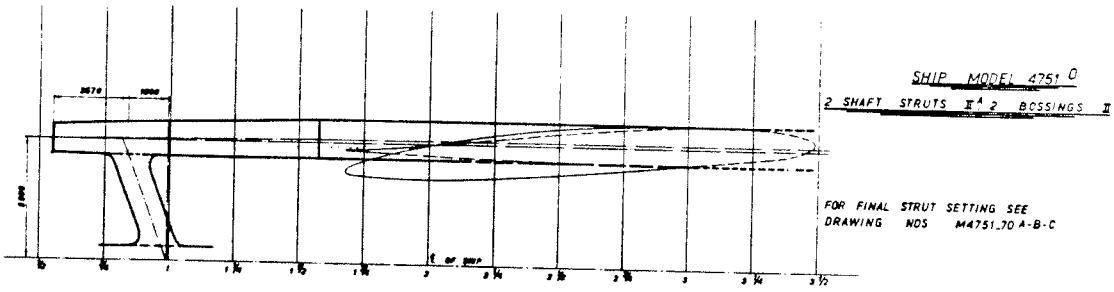
RELATIVE COMPARISON BETWEEN HULL-RESISTANCE AND PROPULSIVE EFFICIENCY 1S–2S–9SCREW

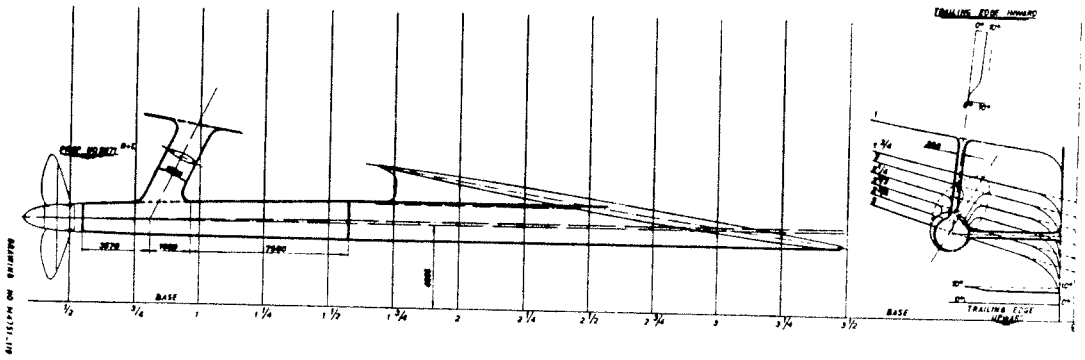
22 KN

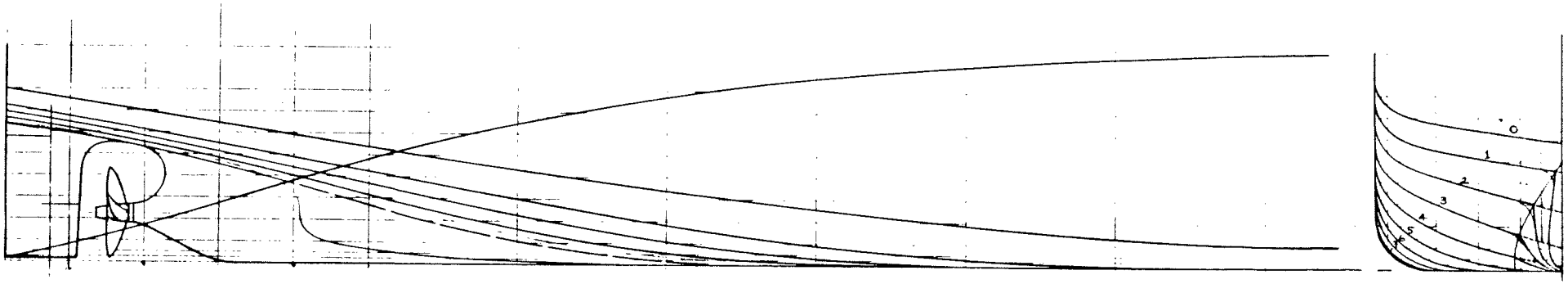
SEP 81 WAGENINGEN + LYNGBY TANK	NAKED RESISTANCE	+ APPEND.	EHP (TOWING)	η OPEN WATER EFFIC PROPELLER	HULL EFFICIENCY $\frac{1-t}{1-w} = \eta_{HULL}$	$\eta_{REL.\ ROTATIVE}$	η_{TOTAL}	EHP / η_{TOTAL}	SHP TANK	
1S — WAG 12.5 M / 11.-M — MODEL 5915 B — Ø 8.325 0.80 · n_s	100%	× 1.00	= 1.00	0.652 / 93 rpm	$\frac{1-0.178}{1-0.194} = 1.019$	1.022	0.680	$\frac{100\%}{.680}$	= 147.0%	
	100%	× 1.00	= 1.00	0.649 / 90	$\frac{1-0.193}{1-0.214} \sim 1.027$	1.032	0.697	$\frac{100\%}{.697}$	= 143.½%	GOOD
					WAKE PEAK 45-50%				SOME VIBRATIONS	COMPARABLE
2S — WAG 12.5 M / 11.-M — MODEL 5914 A — STRUT WRONG DIRECTION — Ø 7.270 0.60 · n_s	99.0	× 1.090	= 1.08	0.710 / 86	$\frac{1-0.079}{1-0.100} \sim 1.023$	1.034	0.751	$\frac{108}{0.751}$	= 144.-%	
	98.8	× 1.093	= 1.08	0.705 / 84	$\frac{1-0.102}{1-0.120} \sim 1.020$	1.035	0.744	$\frac{108}{.744}$	= 145.-%	
					WAKE PEAK 25%				NO VIBRATIONS	
2S — WAG — MODEL DIRK 5783 D — Lwl = 209.5 — Δ = 4674 — R = 27600 — PROP Ø = 7000 0.89 · n_s — MCR/	97% CLEAR PRAM	× 1.078	= 1.045 GOOD!	0.705 / 89 rpm — STOCK PROP	$\frac{1-0.061}{1-0.102} = 1.046$	1.032	0.762 COULD BE IMPROVED BY LARGER Ø	$\frac{104.5\%}{0.762}$	PANMAX = 137.-%	BEST
					WAKE PEAK 6.25%				NO VIBRATIONS	
3S — LYNGBY-DENMARK — MODEL 80120-20 — Ø 9.200 6.700	100%	× 1.085	= 1.083	0.63			$\eta_{RR} \cdot \eta_o \cdot \eta_{PROP}$ 0.730	$\frac{108.5\%}{0.730}$	= 148½%	4% WORSE
2S — SAFCON — WAG — MODEL OTTO 4751 B — Lwl = 198.5 — Δ = 4644 — INSTALLED 2 × 26900 HP — Ø 8.180 0.88 · n_s	97%	× 1.078	= 1.045 GOOD!	0.704 / 104 rpm	$\frac{1-0.077}{1-0.101} = 1.027$	0.991	0.730	$\frac{104.5\%}{0.730}$	HOORN-HOUTMAN = 143%	GOOD
					WAKE PEAK 6.25%				NO VIBRATIONS	COMPARABLE
1S — SAFCON — WAG — MODEL 4736 A — Lwl = 258.5 — Δ = 6644 — 85000 HP — Ø 6.579 0.80 · n_s	97%	× 1.03	= 1.000	0.672 / 101 rpm	CLEAR WATER - STERN $\frac{1-0.171}{1-0.142} = 0.966$	1.00	0.649	$\frac{100\%}{0.649}$	= 154%	8% WORSE
					WAKE PEAK = 25-30%				NO VIBRATIONS	
1S — ZAVO — WAG — MODEL 5163 — Lwl = 220 — Δ = 4644 — 37.600 — Ø 7.117 0.72 · n_s	100%	× 1.00	= 1.000	0.648 / 119 rpm	MORE WAKE MORE EFFICIENCY MORE VIBRATIONS $\frac{1-0.303}{1-0.217} = 1.019$	1.02	0.674	$\frac{100\%}{0.674}$	= 149%	
					WAKE PEAK ~ = 50%				SOME VIBRATIONS	



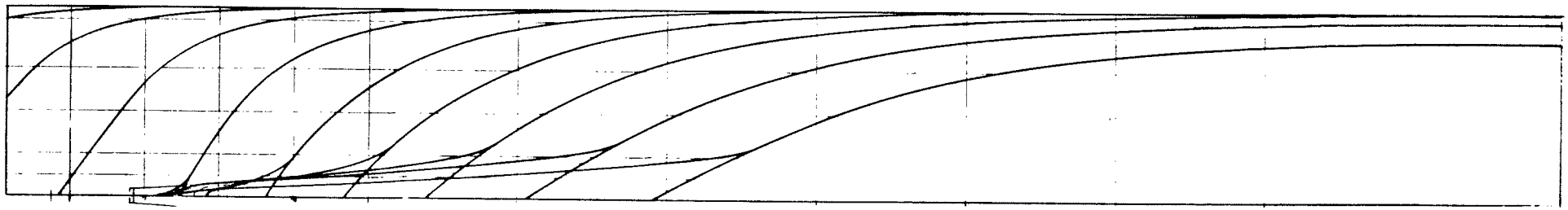
WAKE PEAK AST 25 %



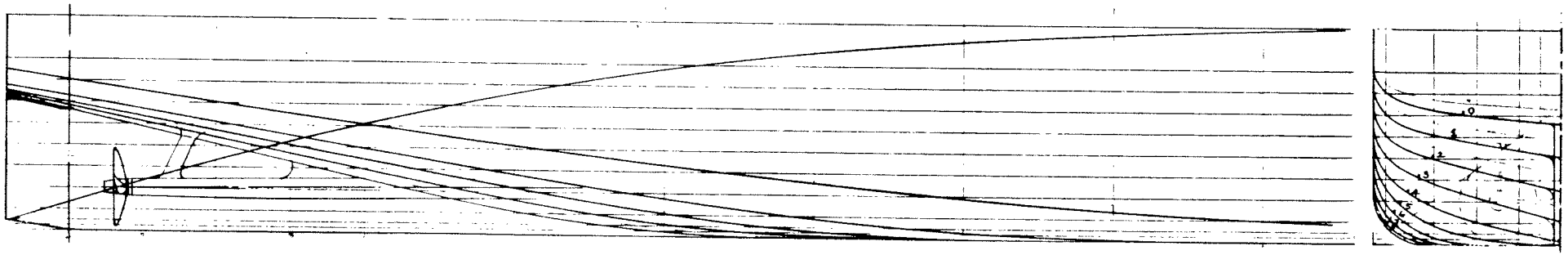




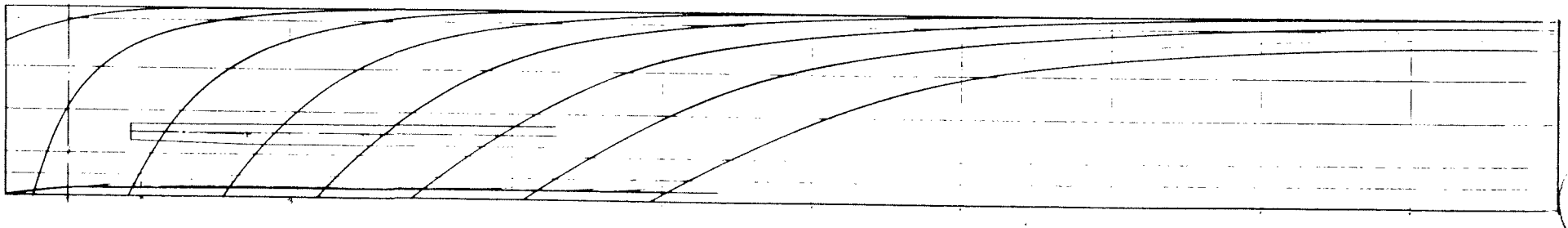
5915 A SINGLE SCREW FAR EAST CONTR.

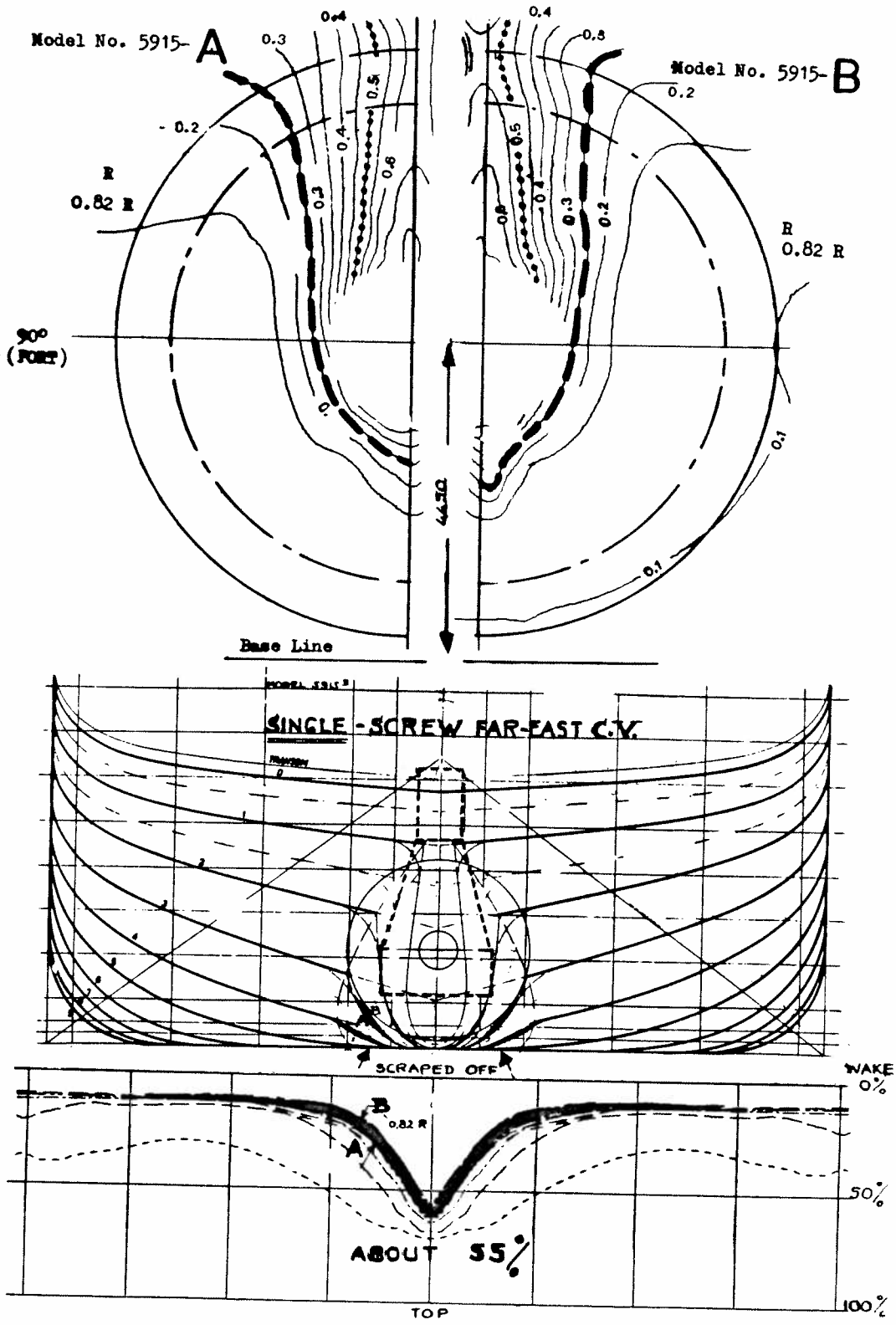


ORDINATES 0 1 2 3 4 5 6 7 8 9 10

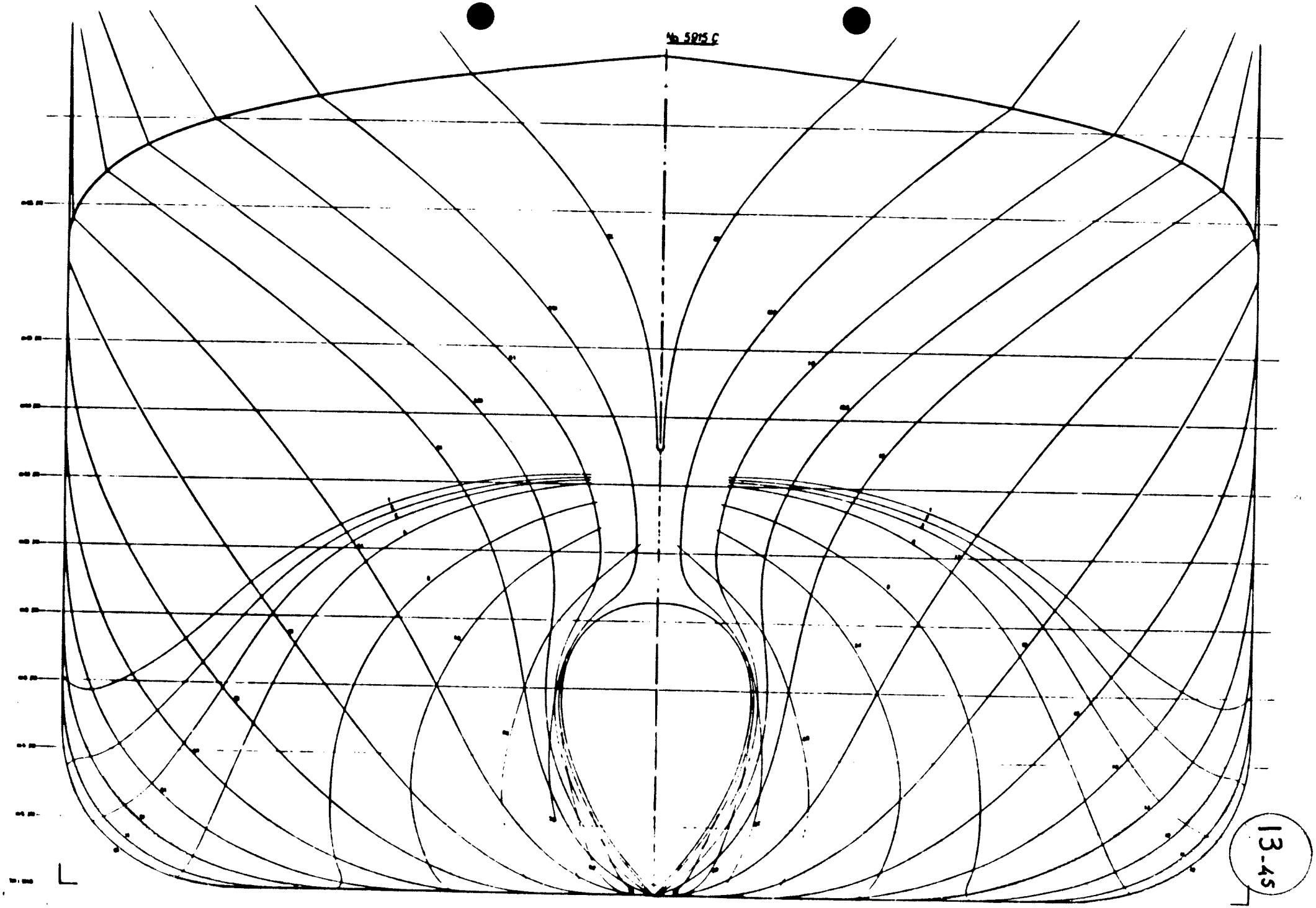


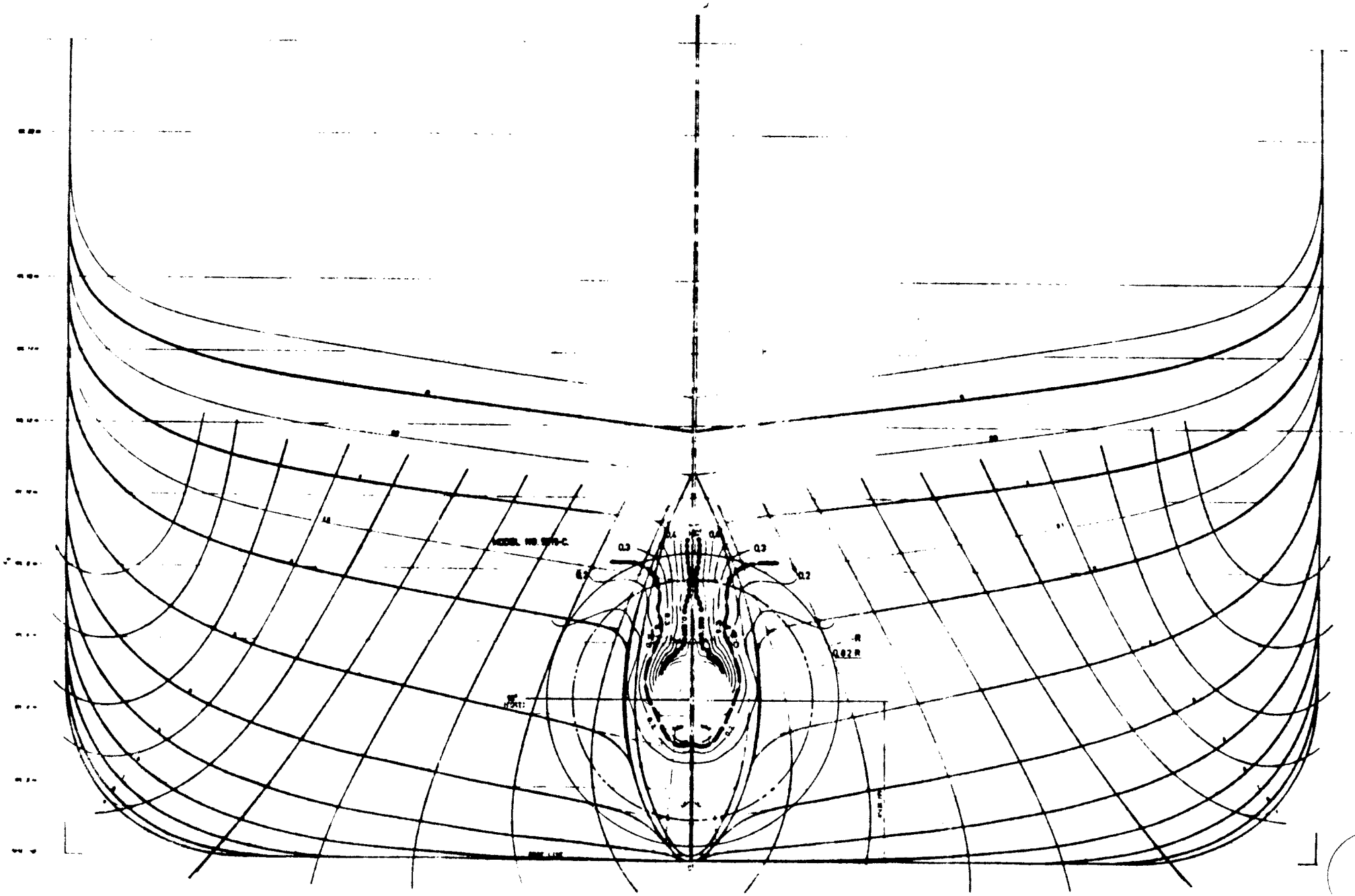
5914 A TWIN SCREW FAR EAST CONTR

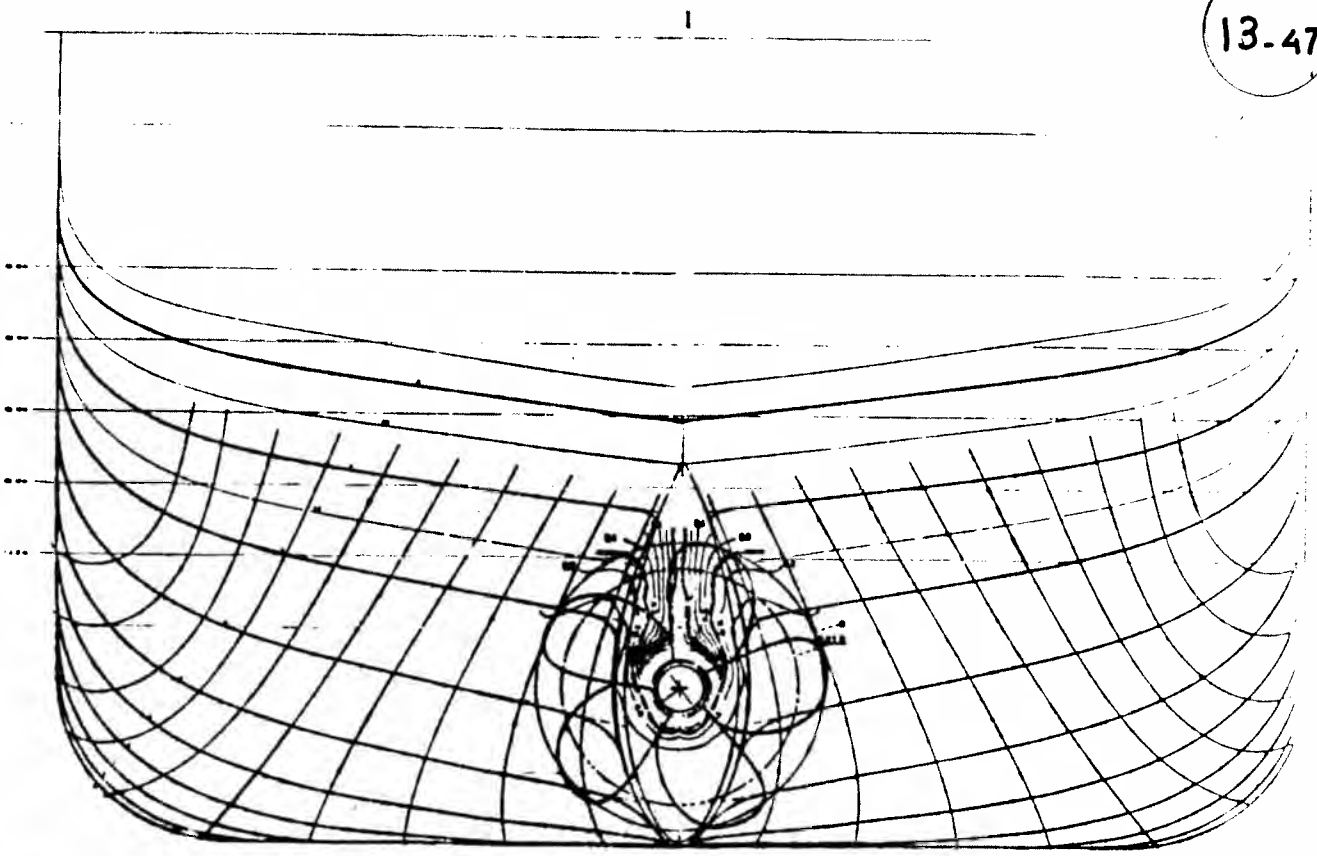




REDUCTION OF WAKE PEAK BY SOFTENING CURVATURE ORD.4-3

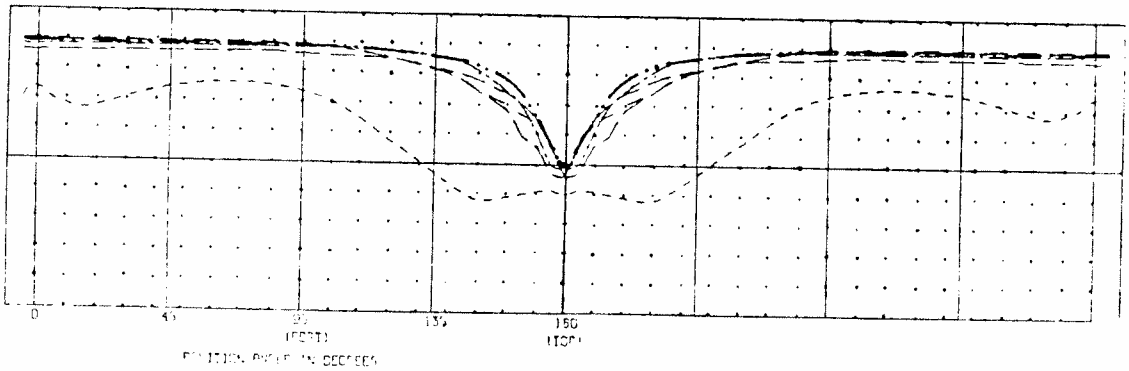

No 5915 C
13-45


0.3
0.3
0.2
0.02 R
R

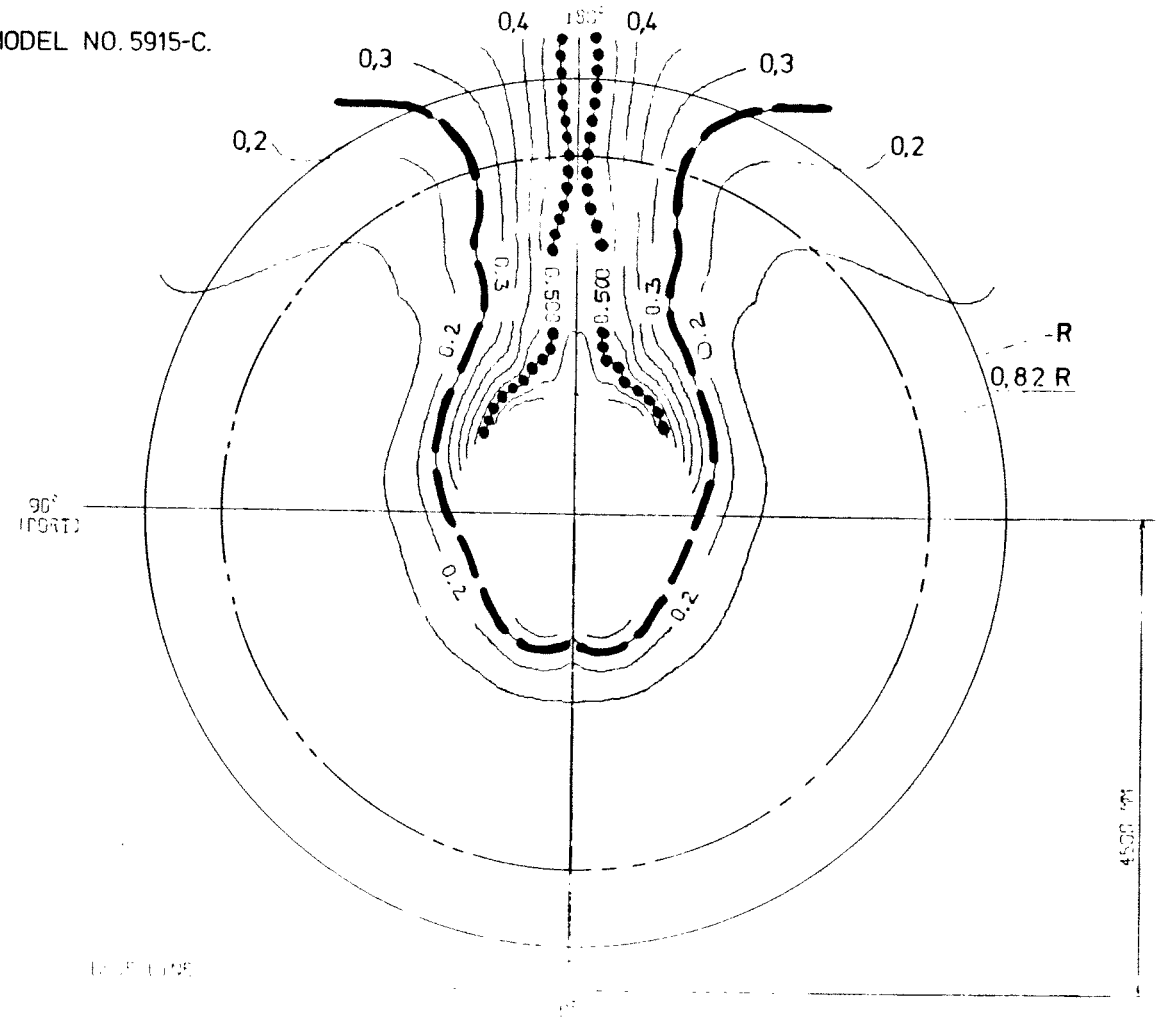


TEST NO. 4356D 5915 C SPEED 22.12 KNOTS
SHIP MODEL NO. [illegible] BULBOUS BOW I
SHIP DRAUGHT ON P.P. 12.562 ft SCREW APERTURE IB
SHIP DRAUGHT ON A.P. 12.562 ft WITHOUT RUDDER IB

FREQ. IN [illegible]	TYPE OF LINE
1.128	
1.503	
2.630	
3.394	
4.135	
4.512	



MODEL NO. 5915-C.



14

STRAAT "H"

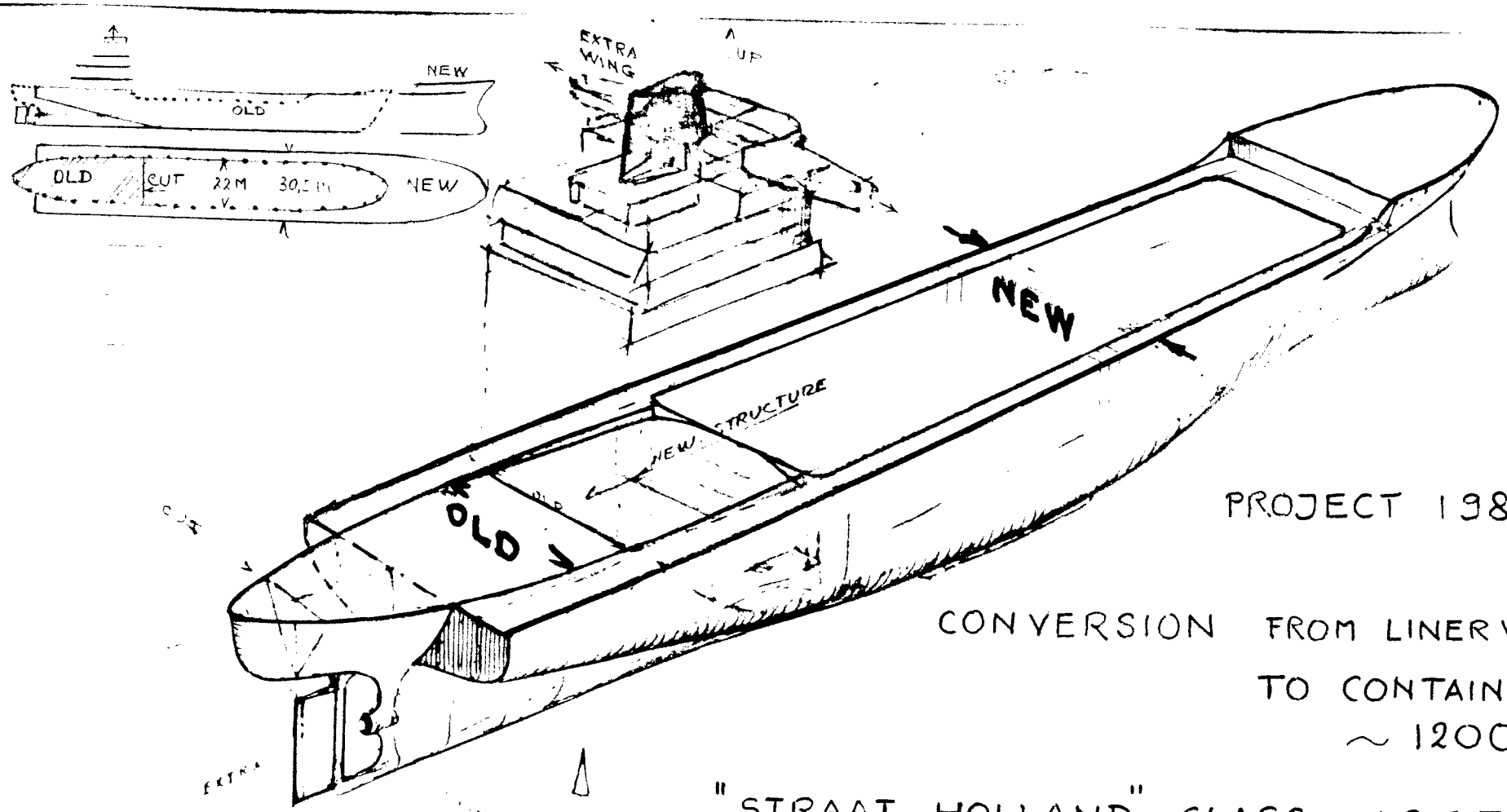
CONVERSION

▼

FORMER AFTERBODY
OF CARGOLINER

+

NEW WIDER FOREBODY
OF CELL.B CONTAINER



PROJECT 1981

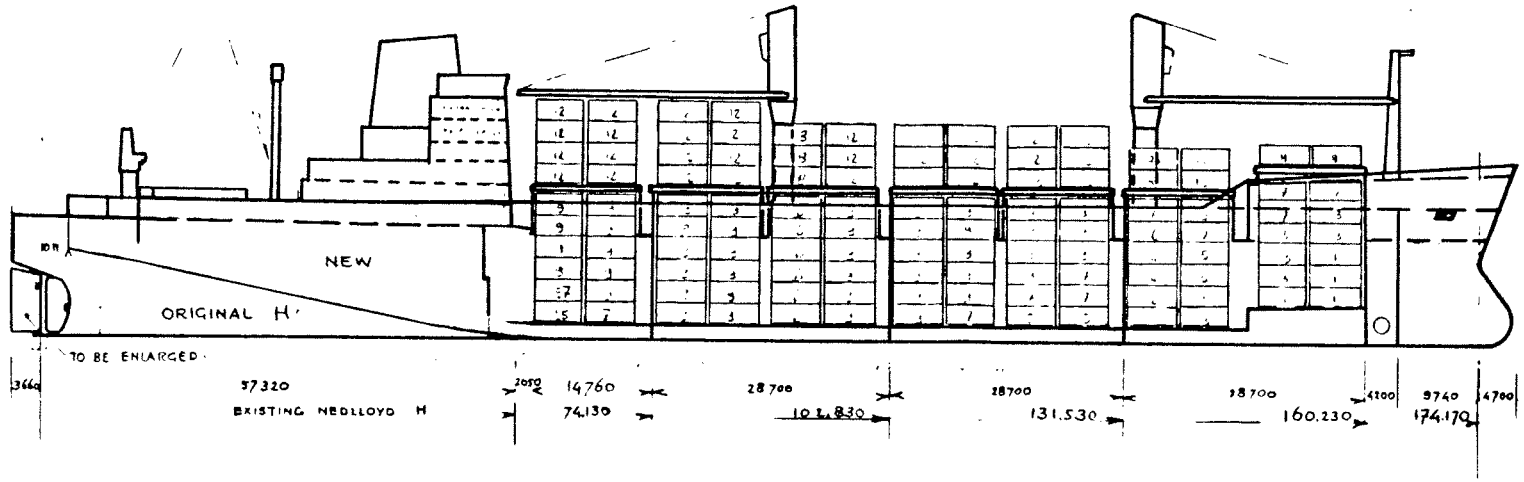
CONVERSION FROM LINER VESSEL
TO CONTAINER.
~ 1200 TEU

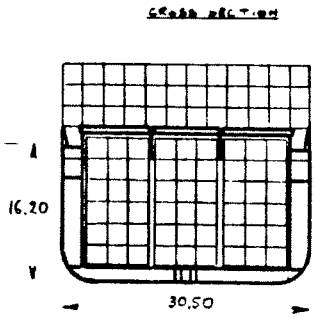
"STRAAT-HOLLAND"-CLASS 1965
{ ENGINE STILL ALL RIGHT (B+W)
{ ACCOMODATION ALL RIGHT

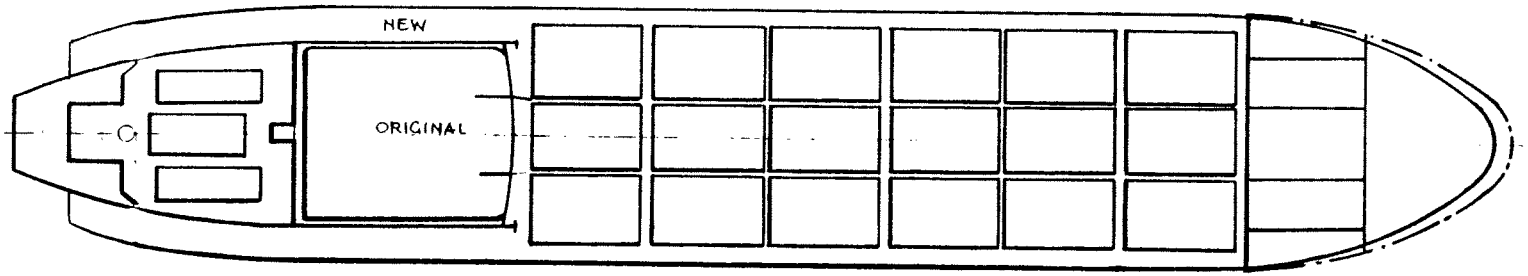
USE ORIG. AFTERBODY + NEW CELLULAR FOREBODY (TYP "KLECO").

14-1

48	48	49	48	24	36	36	30	36	36	24	24	8	8	T.E.U. ON DECK	452 TEU.
48	52	54	54	36	54	54	[illegible]	[illegible]	46	22	32	24	[illegible]	TEU BELOW DECK	592 TEU
27	25	23	21	19	17	15	13	11	09	07	05	03	01	BAY Nº	TOTAL 1044 T.E.U.







PRIOK						
9.75 M DRAFT δ = 0.630	15.9 KN	L_{OA} = 182.53 M	B+W OLD	10 M DRAFT δ = 0.634	15.8 KN SERV	8 M DRAFT δ = 0.594	16.6 KN SERV
DISPL = 33.578 T	15% SEA MARG.	L_{PP} = 174.17	6 CYL	DISPL = 34.678 T	15% SEA MARG.	DISPL = 25.946 T.	15% SEA MARG.
L.SHIP = 11.400		B = 30.50	MCR = 12000	L.SHIP = 11.400		L.SHIP = 11.400	
DW = 22.178	10.200 SHP	H = 16.20	RPM = ~110	DW = 23.278	10.200 SHP	DW = 14.546	10.200 SHP
CONS+WB = 4.678		T DESIGN = 9.50		CONS+WB = 8.662		CONS+WB = 1.596	
CARGO = 17.500	40 TON/DAY.	T SCANTL = 10.00	164 gr/HP.HR.	CARGO = 14.616	40 TON/DAY	CARGO = 12.950	40 TON/DAY.
17½ TON/TEU × 1000 TEU				14 TON/TEU × 1044 TEU		14 TON/TEU × 925 TEU	

CONV. SPACE AFT = 2286 m³
" REFR = 1006 m³
3292 m³

CONTAINER SPACE IN HOLD = 592 TEU
ON DECK 3T = 404
996 TEU
4T = 48
1044 TEU

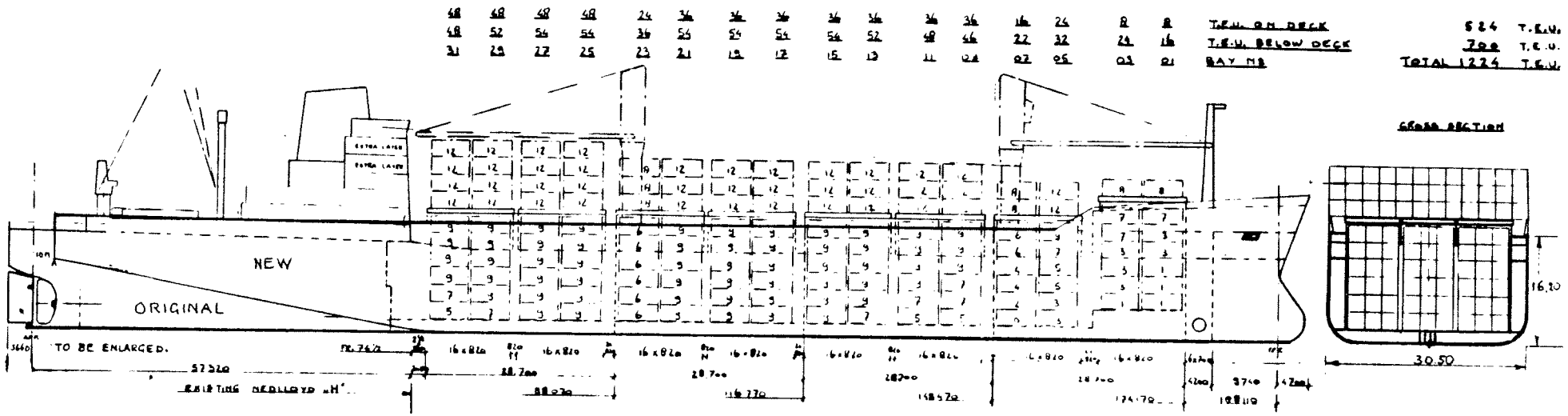
SHORT

KORTE - H

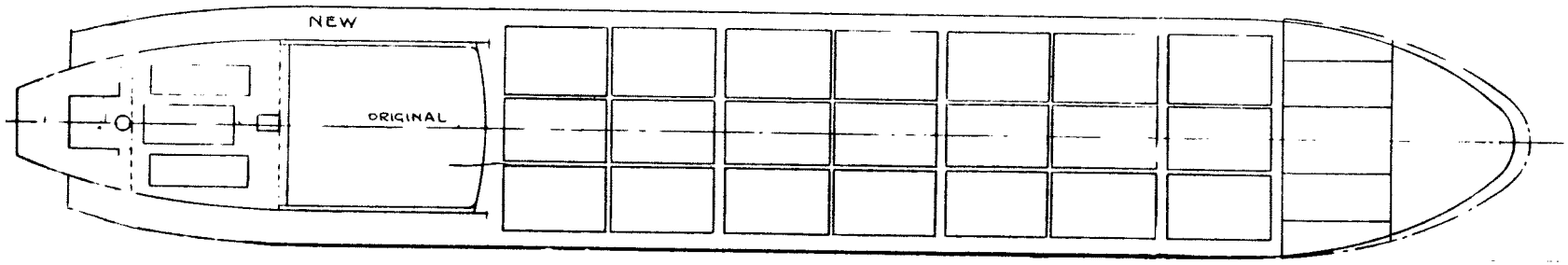
REDESIGNED NEDLLOYD NOLLAND type
S[illegible] (7 HOLDS)

28 OKT. 1981

14-2A



1:581



9.75 M DRAFT δ=0,656	15.9 KN	L OA = 196.47 M	B+W OLD	10 M DRAFT δ=0,659	15.8 KN SERV	8 M DRAFT δ=0,621	16,6 KN SERV
DISPL = 37.764 T	15% SEA MARGIN	L PP = 188.11	6 cyl	DISPL = 38.964 T	15% SEA MARG	DISPL = 29.357 T	15% SEA-MARG
L SHIP = 12.500		B = 30.50	MCR = 12.000	L.SHIP = 12.500		L.SHIP = 12.500	
D.W. = 25.264		H = 16.20	RPM = ~110	D.W. = 26.464		D.W. = 16.857	
CONS+WB = 5.524	10.200 SHP=85%	T DESIGN = 9.50	85% = 10.200	CONS+WB = 9.328	85% = 10.200 SHP	CONS+WB = 1.596	85% = 10.200 SHP
CARGO = 19.740		T SCANTL = 10.00	164 cm	CARGO = 17.136		CARGO = 15.261	
17½ TON/TEU × 1128 TEU	40 TON/DAY.		40 TON/HP HR. DAY.	14 TON/TEU × 1224 TEU	40 TON/DAY	14 TON/TEU × 1090 TEU	40 TON/DAY

CONV. SPACE APPR. = 2286 m³
" REFR. = 1006 m³
3292 m³

CONTAINER SPACE IN HOLD = 700 TEU
ON DECK 3T = 476
1176 TEU.
4T = 48
1224 TEU.

LONG

LANGE-H

(RESHAPED "NEDLLOYD HOLLAND" TYPE LONG-VERSION) (8 HOLDS)

28 OKT. 1981

14-28

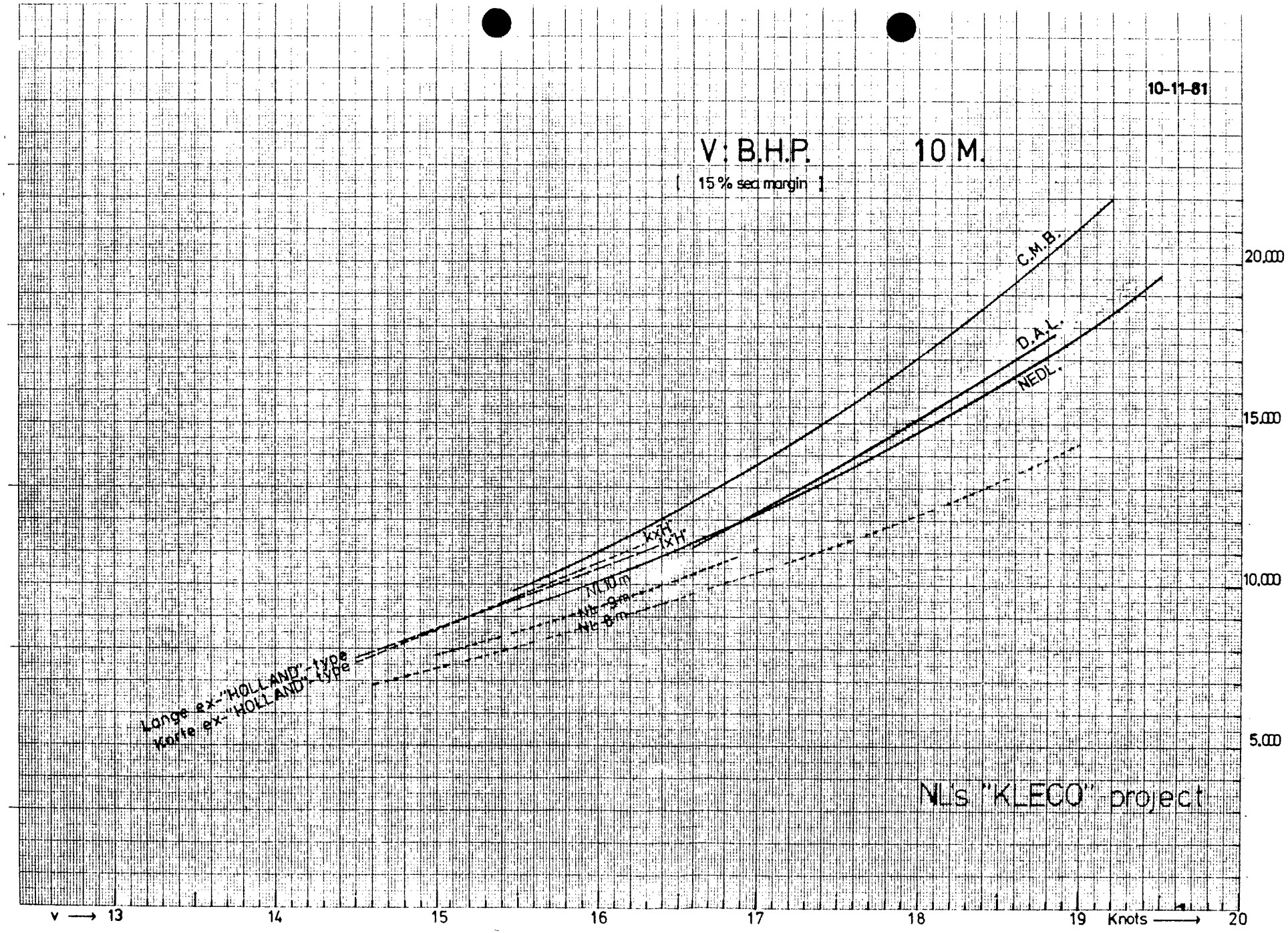
28-10-81	D.A.L. (Woermann) PROJ STAGE	C.N.R. PROJECT	NEDLL. KLECO (Kleine C.V.) VNECK VNOOT TYP	HERSCHAPPEN NEDLL.HOLLAND KORTE VERSIE SHORT	LANGE VERSIE LONG
L.o.a.	169,80 M	175,00 M	182,50 M	182,53 M	196,47 M
L.p.p.	158,20 "	167,— "	174,— "	174,17 "	188,11 "
Br.	28,40 "	30,50 "	30,50 "	30,50 "	30,50 "
Depth	15,45 "	16,55 "	16,20 "	16,20 "	16,20 "
T(dsgn)	9,50 "	10,00 "	9,50 "	9,50 "	9,50 "
T(sclg)	11,20 "	10,65 "	11,00 "	10,00 "	10,00 "
Engine	B & W, L80	B & W, L80	Sulzer	B & W ('old' = existing one)	
Type	5 cyl GFCA	6 cyl GFCA	5 RLA 90	6 cyl.	
M.C.R.	15400	18.400	17.000	12.000	12.000
R.P.M.	106	106	90	— 110	10.200 (85%) — 110
Shft gen.			1.500 (svc)		
FO gr/HpHr	150 g/HH	150 g/HH	153 g/HH	164 g/HH	164 g/HH
At draft	**10 Metrs**	**10 Metrs**	**10 Metrs**	**10 Metrs**	**10 Metrs**
d	0,671	0,682	0,641	0,634	0,659
Displ.	31.060 t	35.865 t	34.966 t	34.678 t	38.964 t
L.S.	8.600 "	11.190 "	11.150 "	11.400 "	12.500 "
D.W.(a/t)	22.460 "	24.675 "	23.816 "	23.278 "	26.464 "
Cons+WB	6.400 "	5.100 "	4.020 "	8.662 "	9.328 "
Cargo DW	16.060 t	19.575 t	19.796 t	14.616 t	17.136 "
à 14 ts/TEU =	1.147 TEU	1.398 TEU	1.414 TEU	1.044 TEU	1.224 TEU
Speed (svc)	17½ kn.	17½ kn.	17½ kn.	15,8 kn(svc)	15,8 kn(svc)
at Seamgn	15 %	15 %	15 %	15 %	15 %
SHP (MCR)	13.700 (89%)	15.350 ( ) (Tank)	13.660 (est)	10.200	10.200 (85 %)
FO Tons/day	49,3 t/d	55,3 t/d	50. t/d	40 t/d	40 t/d
At draft	**8 Mtrs**	**8 Mtrs**	**8 Mtrs**	**8 Mtrs**	**8 Mtrs**
d	0,645		0,606	0,594	0,621
Displ.	23.825 t	27.513 t	26.450 t	25.946 t	29.357 t
L.S.	8.600 "	11.190 "	11.150 "	11.400 "	12.500 "
D.W.(a/t)	15.225 "	16.323 "	15.300 "	14.546 "	16.857 "
Cons.+ WB	2.780 "	2.100 "	1.300 "	1.596 "	1.596 "
Cargo DW	12.445 t	14.223 t	14.000 t	12.950 t	15.261 t
à 14 t/TEU =	889 TEU	1.016 TEU	1.000 TEU	925 TEU	1.090 TEU
Speed (svc)	17½ kn	17½ kn.	17½ kn.	16,6 kn(svc)	16,6 kn(svc)
at Seamgn	15 %	15 %	15 %	15 %	15 %
SHP (MCR)	11.650 (76%)	13.400 ( )	11.900 (est)	10.200	10.200 (85 %)
FO ts/day	40,8 t/d	46,9 t/d	43,7 t/d	40 t/d	40 t/d
				"K"	"L"
in HOLD	624 TEU	714 TEU	746 TEU	592 TEU 2286 m3 Aft	700 TEU
on DECK(3-h)	516 "	596 "	628 "	404 " 1006 m3 Rfr	476 "
	1.140 TEU	1.310 TEU	1.374 TEU	996 TEU 3292 m3 brb	1.176 TEU
incl 4.tr	1.250 "	1.450 "	1.500 "	1.044 " = ALTERNATIVE IF DW+TRIM !	1.224 "
(if stability permits !)		(stability ?)	(stability ?)		

	HERSCHAPPEN "NEDL. HOLLAND" KORTE VERSIE	LANGE VERSIE
WEIGHT-WISE:		
At draft	**9,75 Mts**	**9,75 Mts**
d	0,630	0,656
Displ.	33.578 t	37.764 t
L.S.	11.400 "	12.500 "
D.W.(a/t)	22.178 "	25.264 "
Cons+WB	4.678 "	5.524 "
Cargo DW	17.500 t	19.740 t
à 17½ ts/TEU =	1.000 TEU (+?)	1.128 TEU (+?)
Speed (svc)	15,9 kn	15,9 kn
at Sea-agn	15 %	15 %
SHP	10.200	10.200 (=85%)
FO Tons/day	40 t/d	40 t/d
SPACE-WISE:		
in HOLD	592 TEU	700 TEU
on DECK(3-h)	404 "	476 "
	992 TEU	1.176 TEU
incl. 4th tr	1.044 "	1.224 "

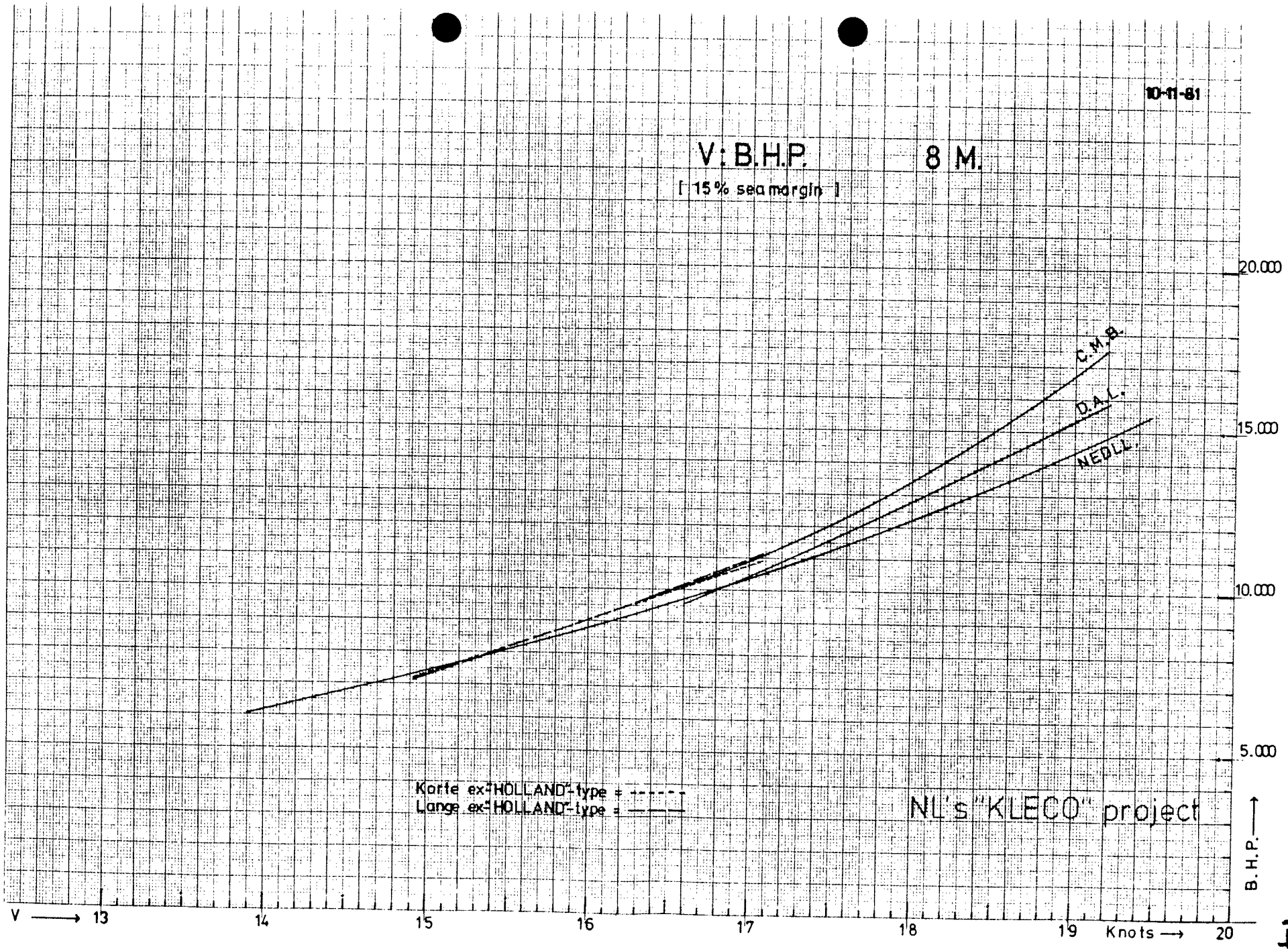
(+?): Breakbulk in AFTship: 2286 m3 Gen. + 1006 m3 Rfr. — 3292 m3 BrBk (= Alternativ utilisable if there is D.W. to spare and if trim does allow so !!)

14-3

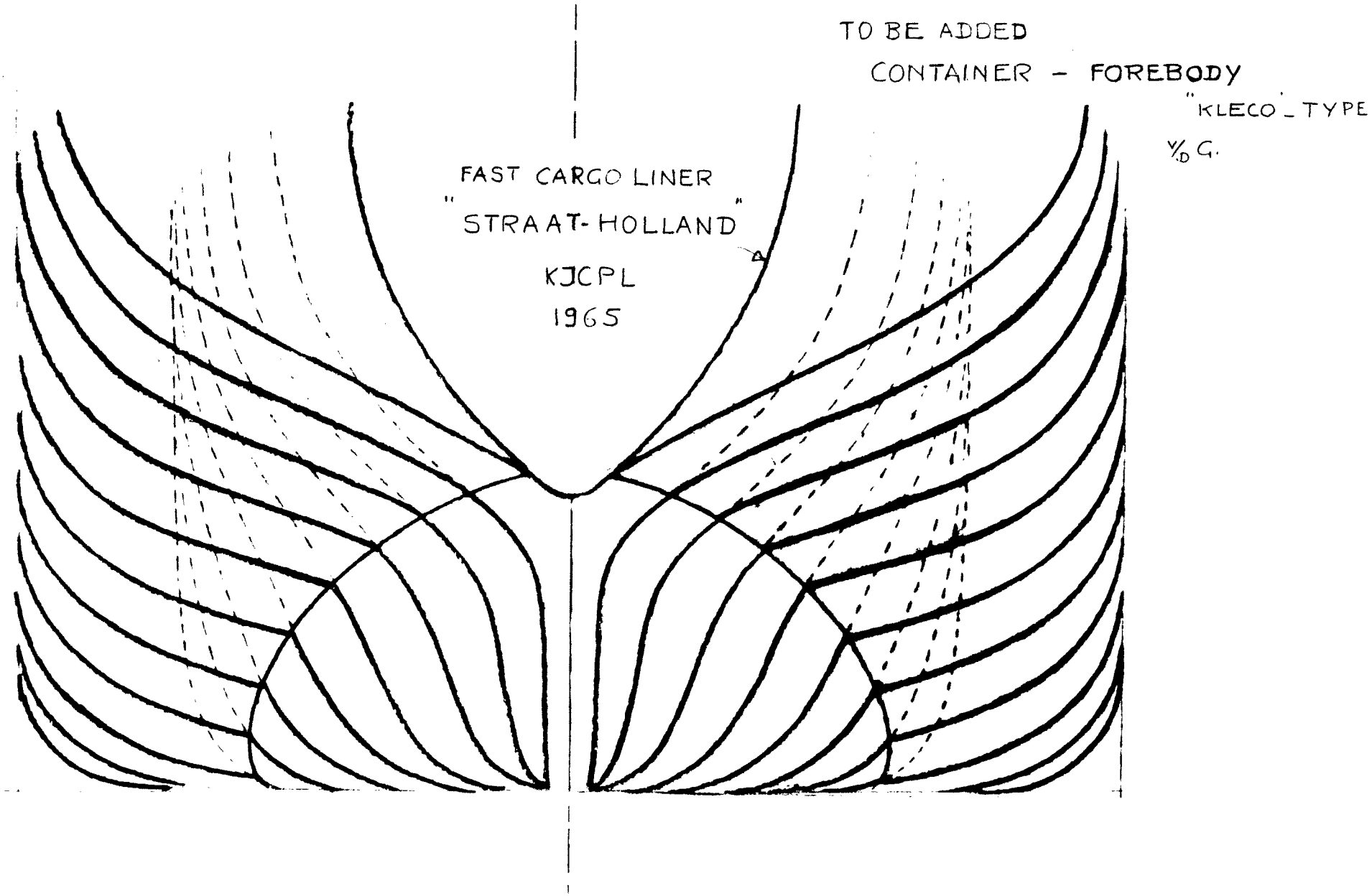
28 OKT. 1981

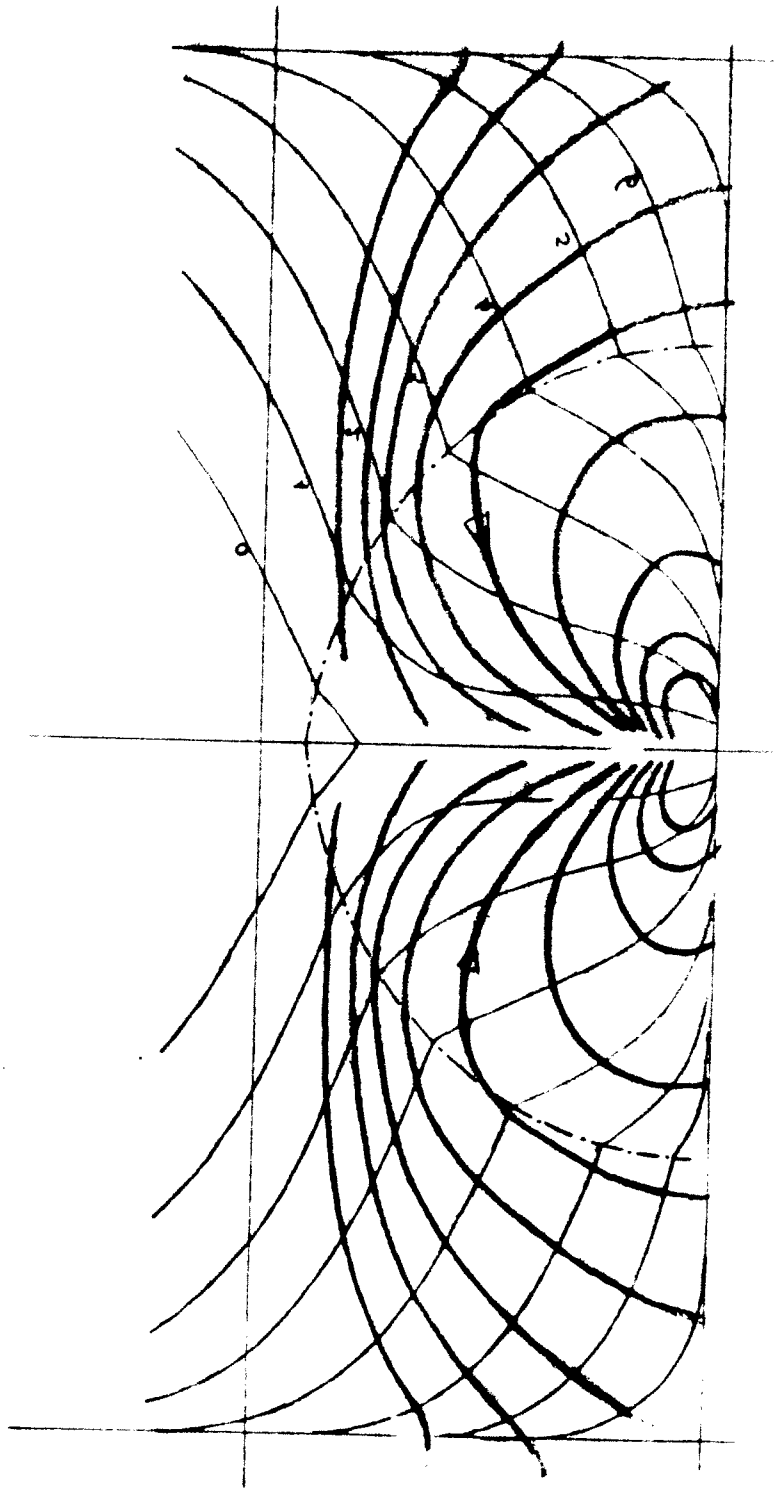

10-11-81
V: B.H.P. 10 M.
[15% sea margin]
C.M.B.
D.A.L.
NEDL.
kx"H"
lx"H"
N.10m
Nc 9m
Nc 8m
Lange ex-"HOLLAND"-type
Korte ex-"HOLLAND"-type
20.000
15.000
10.000
5.000
v → 13
14
15
16
17
18
19
Knots → 20
NLs "KLECO" project

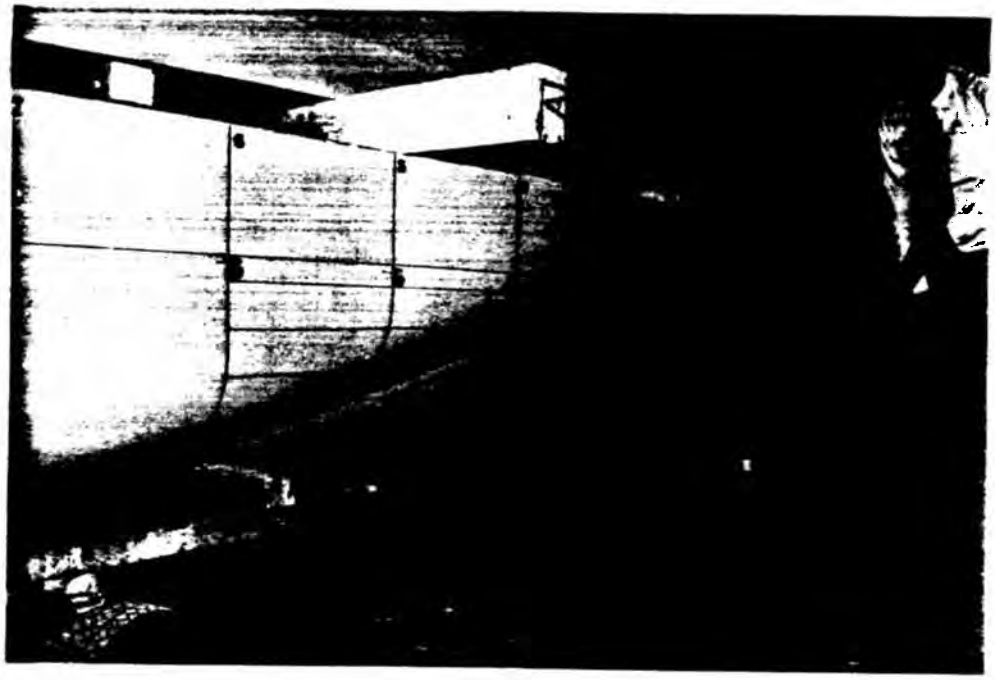
14-4


10-11-81
V: B.H.P. 8 M.
[15% sea margin]
20.000
15.000
10.000
5.000
C.M.S.
D.A.L.
NEDLL.
Korte ex "HOLLAND"-type = - - - - -
Lange ex "HOLLAND"-type = — — —
NL's "KLECO" project
B.H.P. ⟶
V ⟶ 13
14
15
16
17
18
19
Knots ⟶
20

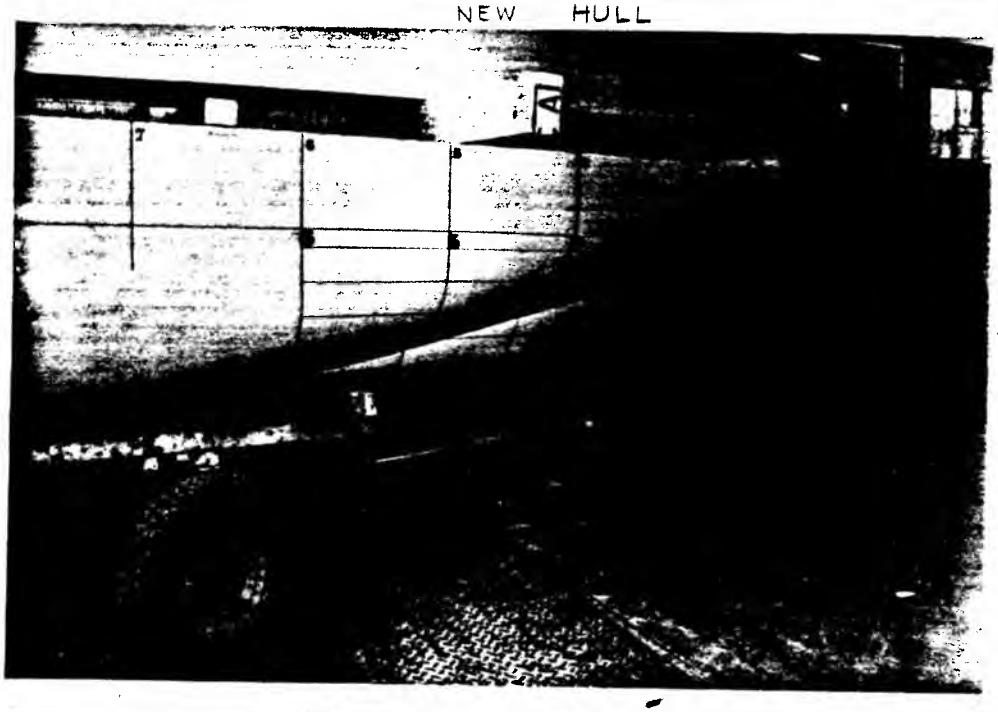
14-4B

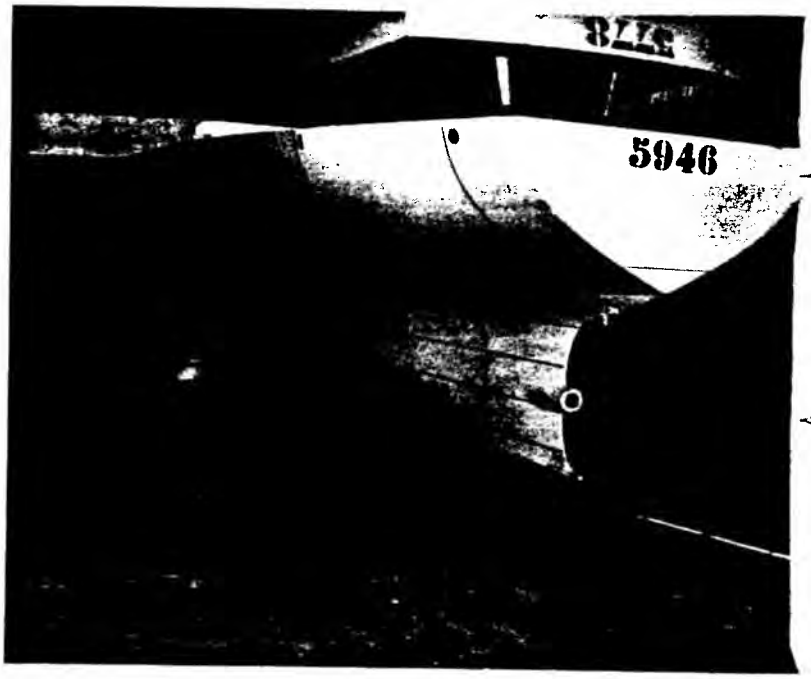

TO BE ADDED
CONTAINER - FOREBODY
"KLECO" - TYPE
½ G.
FAST CARGO LINER
"STRAAT-HOLLAND"
KJCPL
1965





NEW HULL





CONVERSION

STRAAT-HOLLAND

NEW

ORIGINAL AFTERBODY

CONVERSION

Main particulars of ship: "STR. H." AFTERBODY + "KLECO" FOREBODY
B=22,0 M. B=20,5 M

Ship lines represented by ship model 5946:

Length between perpendiculars	L_{PP}	188.11 m
Length on waterline	L_{WL}	191.96 m
Breadth moulded at CWL	B	30.50 m
Draft moulded on F.P.	T_F	10.00 m
Draft moulded on A.P.	T_A	10.00 m
Displacement volume moulded	∇	37704 m^3
Displacement weight in sea water	Δ_1	38647 m.t.
Wetted surface bare hull	S_1	7221 m^2
Wetted surface with rudder	S_2	7283 m^2
L.C.B. aft of ½ Lpp	Θ	1.08 m
Length-Breadth ratio	L_{PP}/B	6.17
Breadth-Draft ratio	B/T	3.05
Block coefficient	C_B	0.657
Midship section coefficient	C_M	0.980
Prismatic coefficient	C_P	0.670

Model details:

Ship model material: wood.
Linear scale ratio λ = 29.
Turbulence induced by studs.

Characteristics of stock propeller:

Stock propeller model 4485		
Diameter	D	5926 mm
Pitch at 0.7R	$P_{0.7}$	5557 mm
Pitch ratio	$P_{0.7}/D$	0.938
Expanded blade area ratio	A_E/A_O	0.651
Boss-Diameter ratio	d/D	0.200
Number of blades	Z	4

ITTC-57

CALCULATION OF EFFECTIVE POWER TO I.T.T.C. CONSTANTS

Test No. 41081
Ship model No. 5946

Scale ratio λ = 29.00000

Model condition:

Bow configuration: BULBOUS BOW I

Stern outline: SCREW APERTURE I

Appendages: RUDDER I

Draft moulded
- on F.P. = 10.000 m
- on A.P. = 10.000 m
- mean = 10.000 m

Displacement volume ∇ = 37704 m^3
Wetted surface ship S_s = 7221 m^2
Wetted surface model S_m = 8.5858 m^2
Submerged ship length L_s = 196.13 m
Submerged model length L_m = 6.763 m

Symbols

V_S	Ship speed
V_M	Model speed
R_{TM}	Measured model resistance
C_{TM}	Specific total model resistance
R_n	Reynolds number
C_{FM}	Specific frictional model resistance coefficient
C_{FS}	Specific frictional ship resistance coefficient
C_A	Incremental resistance coefficient for model-ship correlation
C_{RM}	Specific residuary model resistance coefficient
C_{TS}	Specific total ship resistance coefficient
R_{TS}	Total ship resistance
P_E	Effective power
C_e	Admiralty coefficient
©	R.E. Froude's resistance coefficient

Introduced values and formulas

Temperature tank water = 14.4 °C

Standard temperature = 15 °C

Coefficient of kinematic viscosity for 15 °C
ν_s = .11888E-05 $m^2.sec^{-1}$ (sea water)

Coefficient of kinematic viscosity for 14.4 °C
ν_M = .11590E-05 $m^2.sec^{-1}$ (tank water)

Mass density of tank water ρ' = 101.94 $kg.sec^2m^{-4}$

Mass density of sea water ρ = 104.5 $kg.sec^2m^{-4}$

$R_n\,(\text{model}) = \frac{V_M \times L_M}{\nu_M}$ $\qquad R_n\,(\text{ship}) = \frac{V_S \times L_S}{\nu_S}$

$C_{TM} = C_{FM} + C_{RM}$ $\qquad C_{TS} = C_{FS} + C_A + C_{RM}$

Frictional resistance coefficient according to the formula

$C_F = \frac{0.075}{(\log R_n - 2)^2}$ $\qquad$ © $= 425.455 \times \frac{P_E\,\text{metric}}{\Delta_1^{2/3} . V_S^3}$

C_A = .000100

$C_e = \frac{\Delta_1^{2/3} . V_S^3}{P_E\,\text{metric}}$ $\qquad \Delta_1 = 1.025\nabla$

V_S in knots	V_M in $m.sec^{-1}$	R_{TM} in g	C_{TM} x 10^3	R_n (model)	C_{FM} x10^3	R_n (ship)	$C_{FS}+C_A$ x10^3	C_{RM} x10^3	C_{TS} x10^3	R_{TS}	P_E	P_E frictional	P_E residuary	C_e	©
12.00	1.1464	2156	3.749	.6689E+07	3.221	.1018E+10	1.627	.528	2.155	30.99	2550	1926	624	774	.549
12.50	1.1941	2319	3.716	.6968E+07	3.198	.1061E+10	1.619	.519	2.138	33.36	2860	2166	694	781	.545
13.00	1.2419	2497	3.700	.7247E+07	3.175	.1103E+10	1.612	.524	2.137	36.05	3215	2426	789	781	.545
13.50	1.2897	2685	3.689	.7525E+07	3.154	.1146E+10	1.605	.535	2.140	38.95	3606	2705	901	780	.546
14.00	1.3374	2878	3.677	.7804E+07	3.134	.1188E+10	1.598	.543	2.142	41.91	4025	3004	1021	779	.546
14.50	1.3852	3088	3.678	.8083E+07	3.114	.1231E+10	1.592	.564	2.155	45.25	4501	3324	1177	774	.549
15.00	1.4330	3307	3.680	.8362E+07	3.095	.1273E+10	1.586	.585	2.171	48.76	5017	3666	1351	769	.553
15.50	1.4807	3546	3.696	.8640E+07	3.078	.1316E+10	1.580	.618	2.198	52.72	5606	4029	1577	759	.560
16.00	1.5285	3796	3.713	.8919E+07	3.061	.1358E+10	1.574	.652	2.226	56.91	6246	4416	1830	750	.568
16.50	1.5763	4062	3.736	.9198E+07	3.044	.1400E+10	1.569	.692	2.260	61.45	6954	4826	2128	738	.576
17.00	1.6240	4347	3.766	.9476E+07	3.028	.1443E+10	1.563	.738	2.301	66.41	7744	5260	2484	725	.587
17.50	1.6718	4643	3.796	.9755E+07	3.013	.1485E+10	1.558	.783	2.341	71.60	8594	5719	2875	713	.597

SHIP RESISTANCES ARE GIVEN IN METRIC TONS -
SHIP POWERS ARE GIVEN IN METRIC HP -

h_p

PROPULSION TEST RESULTS

Test No. 41082

Model condition: BULBOUS BOW I
SCREW APERTURE I
RUDDER I

Draft moulded: on FP = 10.000 m; on AP = 10.000 m; mean = 10.000 m

Displacement volume: ∇ = 37704 m^3

Ship model No. 5946
Propeller model No. 4485
Scale ratio λ = 29.00000

Skin friction correction F_D from model to ship is according to ITTC
INCREMENTAL RESISTANCE COEFFICIENT CA = .000100

Ship Speed in knots V_S	P_D	P_E	$\eta_D = \frac{P_E}{P_D}$	C_A	Revs/min n_S	Resistance R_T in tons	Thrust T in tons	$t = \frac{T-R_T}{T}$
12.00	3557	2551	.717	555	73.1	30.99	37.35	.170
12.50	3968	2860	.721	563	76.0	33.36	40.30	.172
13.00	4417	3215	.728	569	78.8	36.05	43.22	.166
13.50	4897	3606	.736	574	81.8	38.95	46.40	.161
14.00	5409	4025	.744	580	84.8	41.91	49.97	.161
14.50	6022	4501	.747	579	88.0	45.25	53.67	.157
15.00	6724	5017	.746	574	91.3	48.76	57.62	.154
15.50	7530	5606	.744	565	94.7	52.72	62.10	.151
16.00	8421	6246	.742	556	98.3	56.91	66.80	.148
16.50	9401	6954	.740	546	101.8	61.45	72.05	.147
17.00	10499	7744	.738	535	105.6	66.41	77.65	.145
17.50	11665	8594	.737	525	109.2	71.60	83.37	.141

VALUES ARE GIVEN IN METRIC UNITS

P_D = delivered HP at propeller.
P_E = effective HP.
n_S = RPM of the ship's screw under the below conditions, not corrected for difference between the wake of ship and model, without any allowance

$C_A = \frac{\Delta_1^{2/3} V_S^3}{P_D \text{ metric}}$ Δ_1 = Displacement weight in salt water, in metric tons.

British HP = $\frac{75}{76}$ metric HP

Note: Results of self-propulsion tests as stated above refer to the self propulsion point of ship.
The results are calculated direct from measured model values without any allowance for appendages not present in the model, wind and sea; so values for ship are for tank condition corrected for salt water (γ_1 = 1.025) and 15° centigrade.
Model and ship dimensions are moulded.

hp

SLIP, WAKE, THRUST DEDUCTION VALUES, ETC

Propulsion test No. 41082
Open water test No. 30038
Model condition: BULBOUS BOW I
SCREW APERTURE I
RUDDER I

Ship model No. 5946
Propeller model No 4485

Draft moulded { on F.P.= 10.000 m
on A.P.= 10.000 m
mean = 10.000 m

Scale ratio= 29.00000

Ship speed in knots V_S	Thrust HP P_T	True slip S_R	Apparent slip S_A	Wake fraction (Taylor) w	Thrust deduction fraction t	Hull efficiency η_H	Efficiency behind ship η_B	Open water efficiency η_O	Relative rotative eff $\frac{\eta_B}{\eta_O}$	Advance coefficient $J=\frac{V_A}{nD}$
12.0	2206	.346	.089	.282	.170	1.157	.620	.631	.983	.613
12.5	2476	.345	.086	.283	.172	1.155	.624	.632	.988	.614
13.0	2759	.344	.083	.284	.166	1.165	.625	.632	.988	.615
13.5	3091	.340	.083	.281	.161	1.166	.631	.634	.995	.619
14.0	3460	.339	.083	.279	.161	1.164	.640	.636	1.006	.620
14.5	3865	.337	.084	.276	.157	1.164	.642	.637	1.008	.622
15.0	4320	.335	.088	.271	.154	1.161	.642	.638	1.008	.623
15.5	4815	.337	.091	.271	.151	1.164	.639	.637	1.005	.622
16.0	5372	.337	.096	.267	.148	1.163	.638	.636	1.002	.621
16.5	5990	.339	.100	.265	.147	1.161	.637	.635	1.003	.620
17.0	6681	.340	.106	.262	.145	1.159	.636	.635	1.003	.619
17.5	7386	.343	.109	.262	.141	1.164	.633	.633	1.000	.616

NOTE: THE VALUES HAVE BEEN CALCULATED WITH THE
MEAN J-VALUES OF THOSE BASED ON KT AND KQ
PT IN METRIC HP

PREDICTION

SHIP MODEL 5946

PROPELLER MODEL 4485 (STOCK PROPELLER)

TANK VALUES ARE BASED ON RESULTS OF PROPULSION TEST No 41082

DRAFT 10.00 m

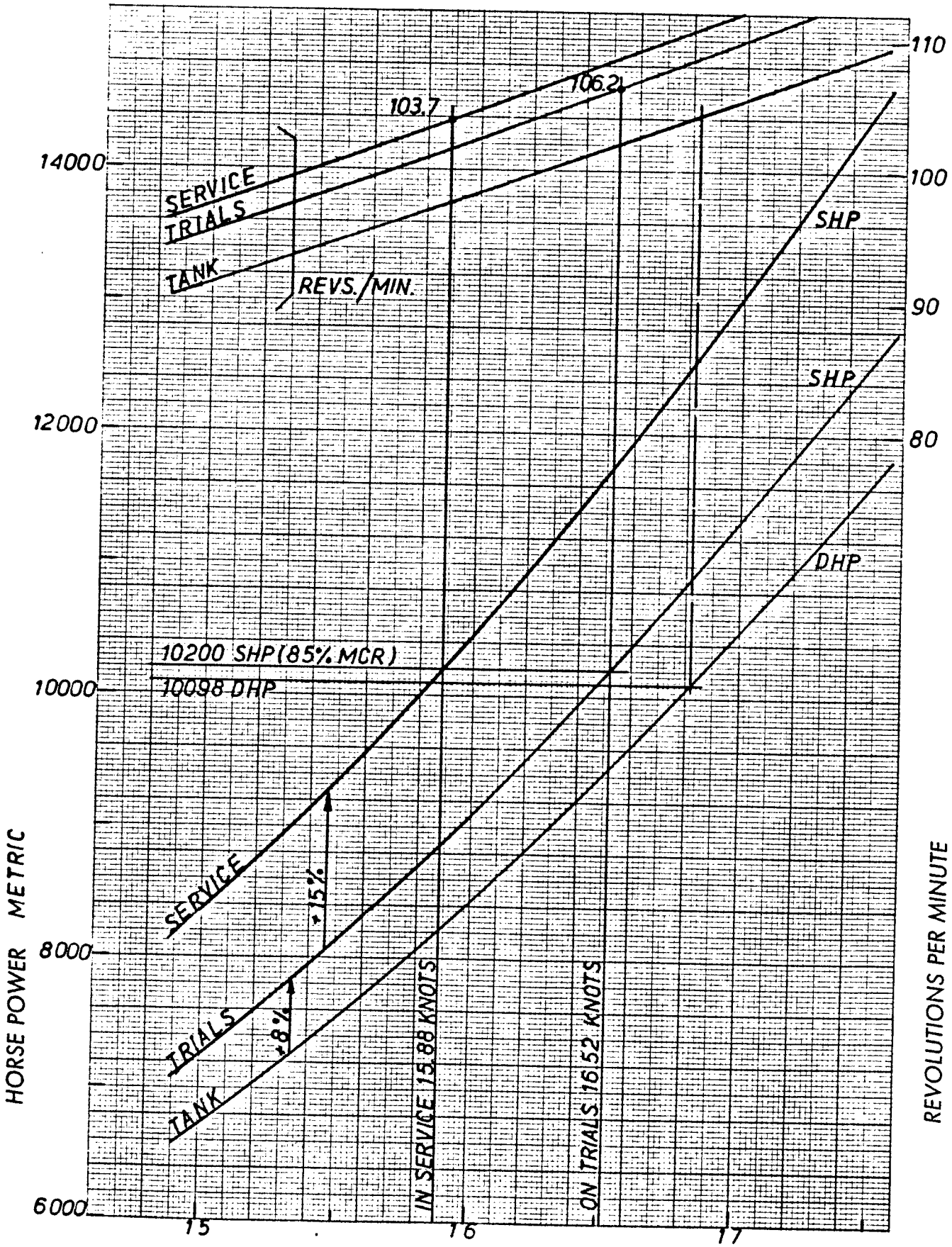
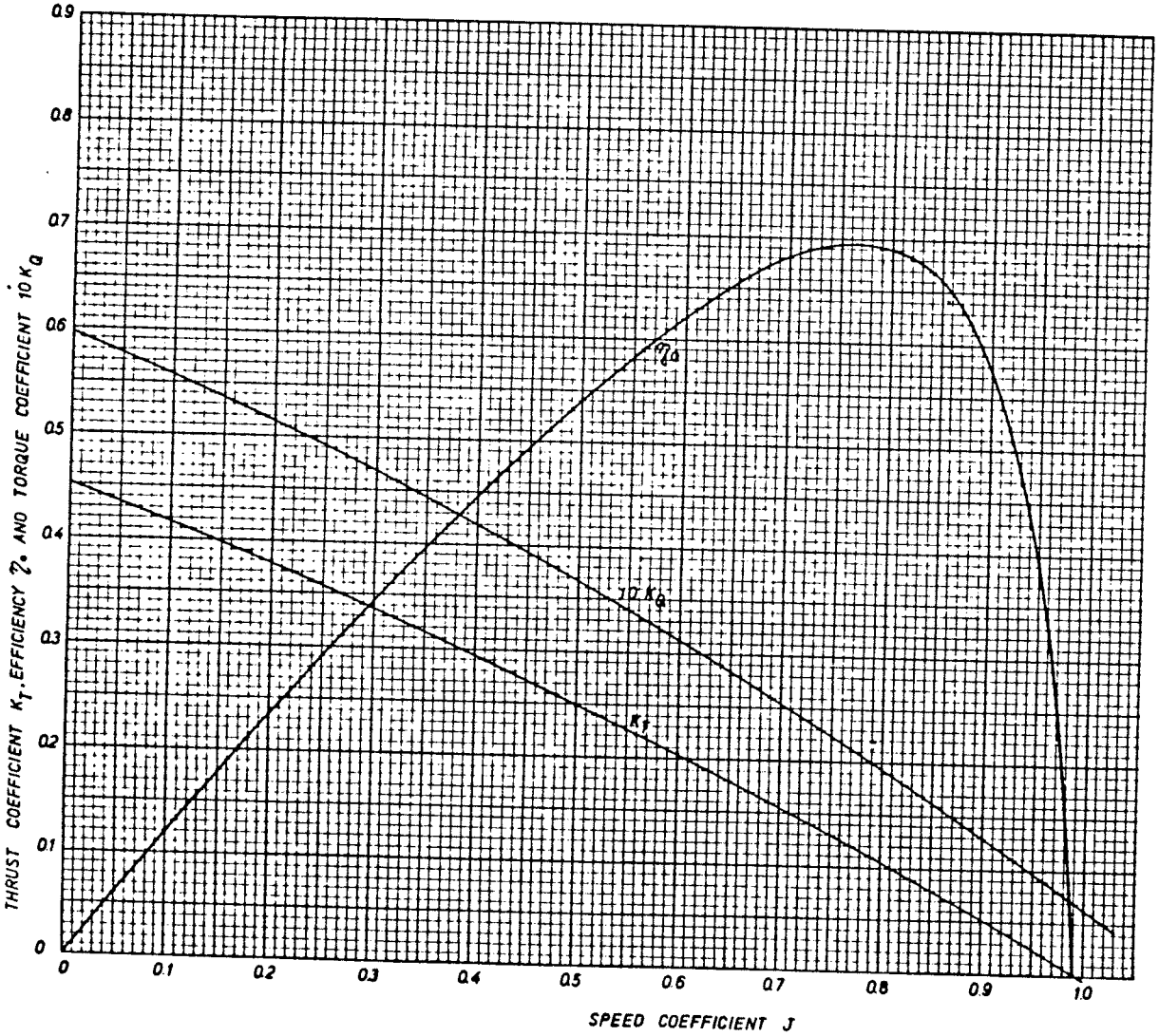


DIAGRAM M5946_15

NETHERLANDS SHIP MODEL BASIN
WAGENINGEN

OPEN WATER CHARACTERISTICS

PROPELLER Nº. 4485



PROPELLER CHARACTERISTICS

DIAMETER	= 5926	mm
PITCH AT BOSS	= 4859	mm
PITCH AT 0.7D	= 5557	mm
PITCH AT BLADE TIP	= 5129	mm
PITCH RATIO AT 0.7D	= 0.938	
BOSS DIAMETER RATIO	= 0.200	
EXPANDED BLADE AREA RATIO	= 0.651	
NUMBER OF BLADES	= 4	
NUMBER OF REVOLUTIONS (MODEL)	= 12.0	PER SECOND

THRUST COEFFICIENT $K_T = \dfrac{T}{\rho D^4 n^2}$

TORQUE COEFFICIENT $K_Q = \dfrac{Q}{\rho D^5 n^2}$

EFFICIENCY $\eta_o = \dfrac{K_T}{K_Q}\dfrac{J}{2\pi}$

SPEED COEFFICIENT $J = \dfrac{V_A}{nD}$

T = THRUST
Q = TORQUE
n = NUMBER OF REVOLUTIONS
V_A = SPEED OF ADVANCE
ρ = MASS DENSITY OF WATER
D = DIAMETER

TEST Nº 30038

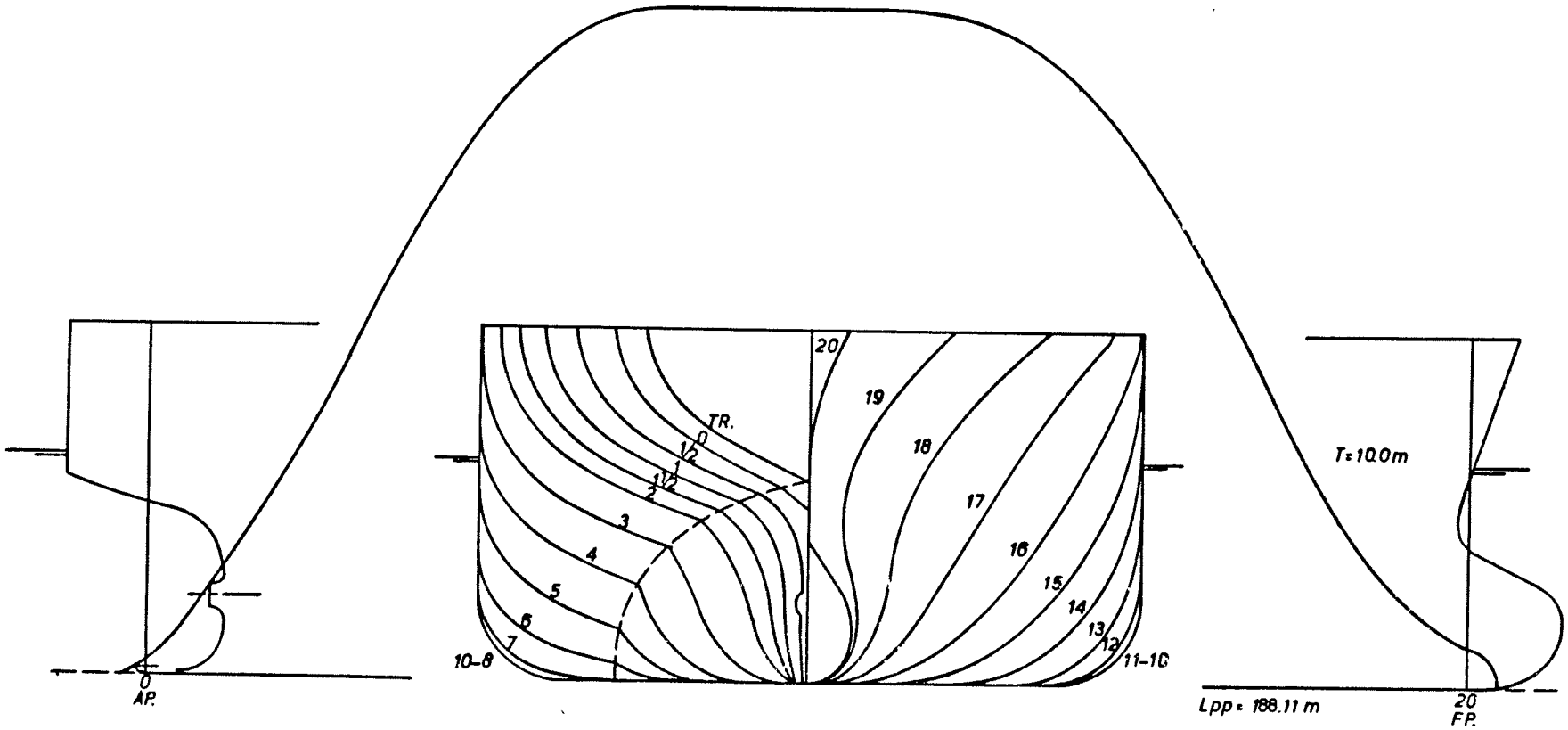
SCALE OF PROPELLER MODEL 1:29
PROPELLER MODEL RELATED TO SHIP MODEL 5946

DIAGRAM M5946 _16

14-13

SHIP MODEL No. 5946

BODY PLAN, STEM AND STERN OUTLINES AND CURVE OF SECTIONAL AREAS.

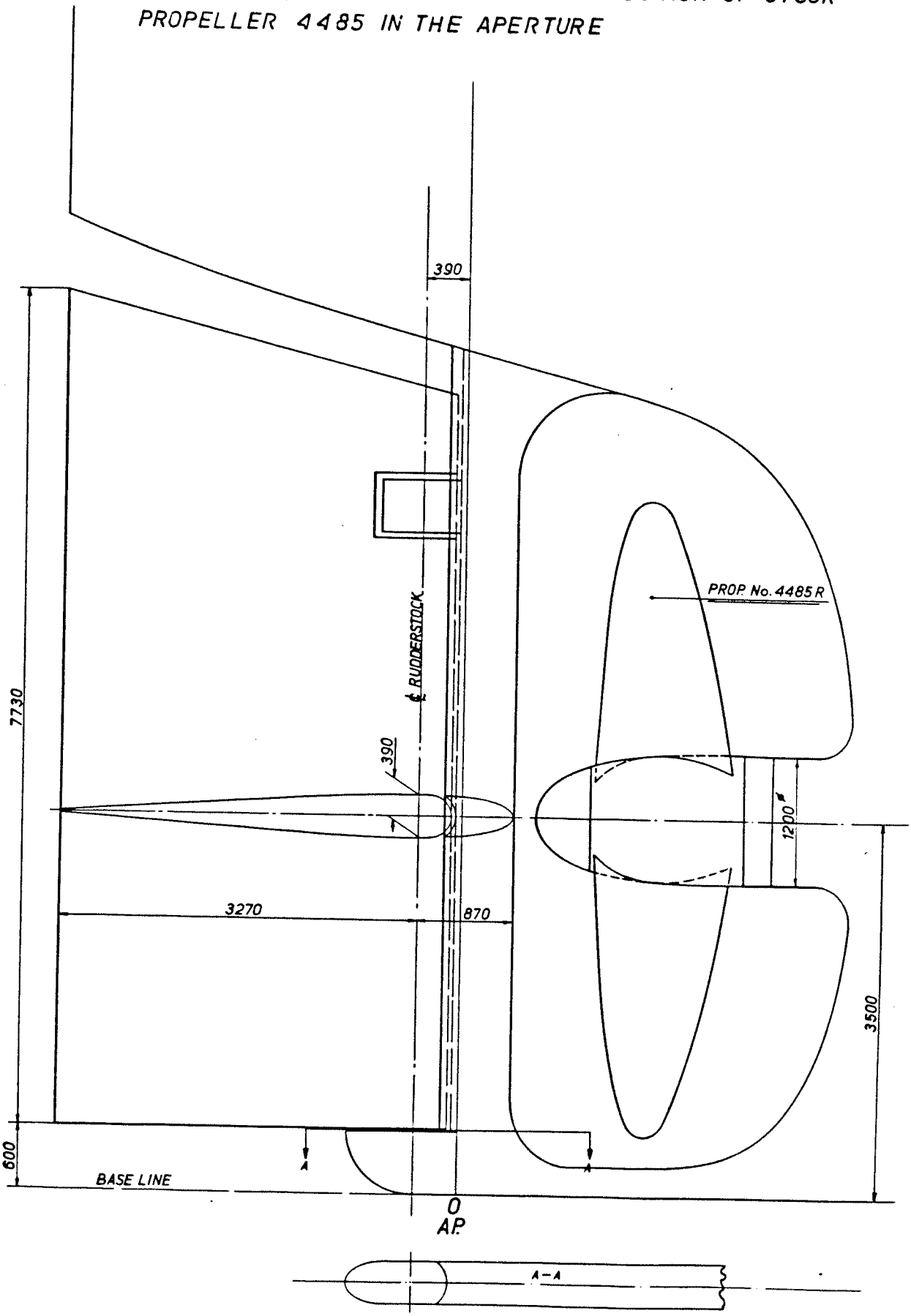


DRAWING No. M5946 — 14

14-14

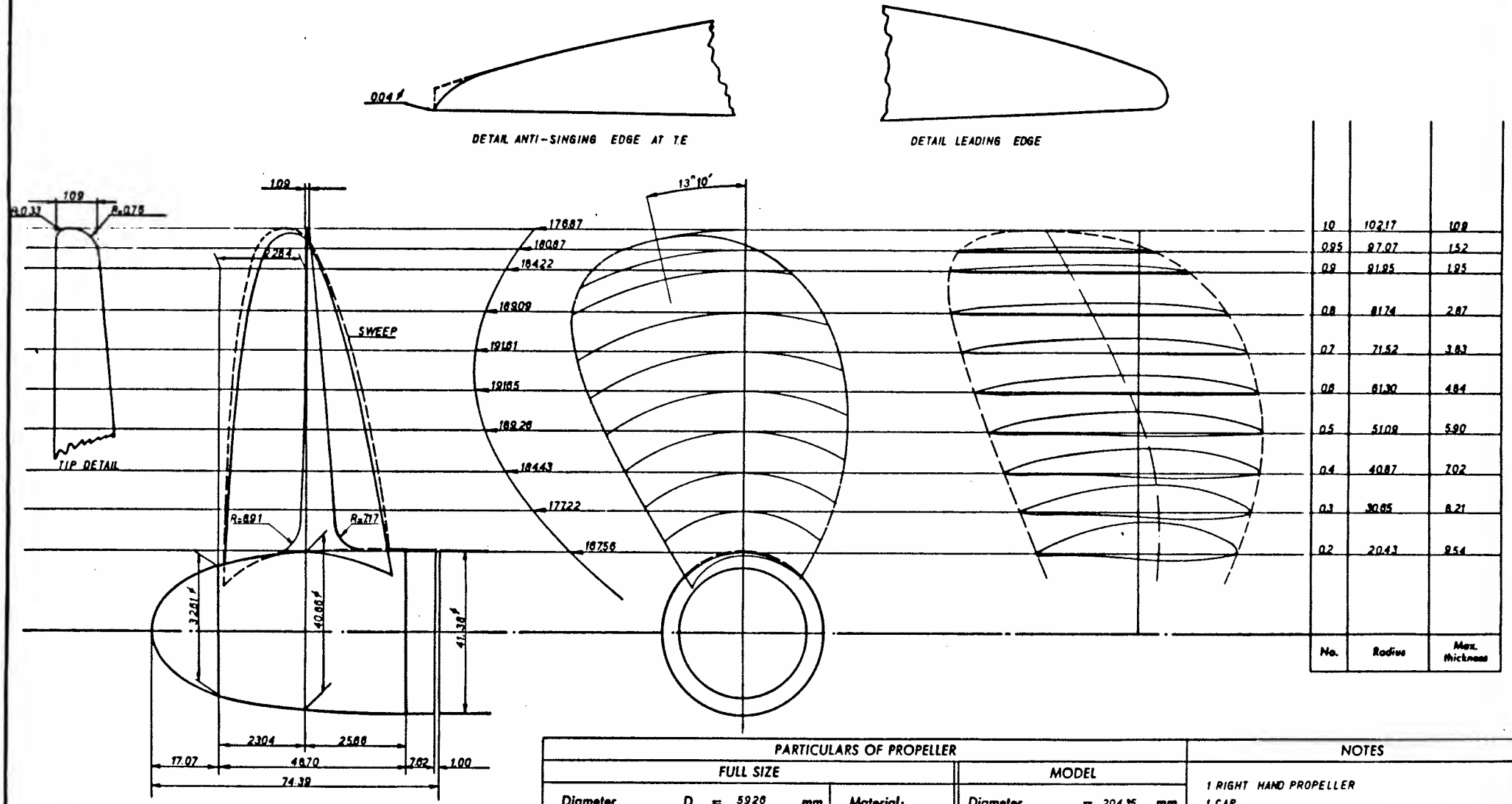
SHIP MODEL No. 5946

1 RUDDER I, SCREW APERTURE I AND POSITION OF STOCK PROPELLER 4485 IN THE APERTURE



DRAWING No. M5946–13

NETHERLANDS SHIP MODEL BASIN
WAGENINGEN



No.	Radius	Max. thickness
1.0	102.17	1.09
0.95	97.07	1.52
0.9	91.95	1.95
0.8	81.74	2.87
0.7	71.52	3.83
0.6	61.30	4.84
0.5	51.09	5.90
0.4	40.87	7.02
0.3	30.65	8.21
0.2	20.43	9.54

PARTICULARS OF PROPELLER										NOTES
FULL SIZE						MODEL				
Diameter	D	=	5926	mm	Material:	Diameter	=	204.35	mm	1 RIGHT HAND PROPELLER
Pitch at root		=	4859	mm	LIMA-BRONZE	Material:	BRONZE			1 CAP
Pitch at 0.7 R	$P_{0.7}$	=	5557	mm		Pitch at root	=	16756	mm	1 RING
Pitch at blade tip		=	5129	mm	d/D = 0.200	Pitch at 0.7 R	=	19161	mm	
Disc area	A_O	=	27.581	m^2	$P_{0.7}/D$ = 0.938	Pitch at blade tip	=	17687	mm	PROPELLER MODEL No. 4485
Exp. blade area	A_E	=	17.955	m^2						
Proj. blade area	A_P	=	15.749	m^2	A_E/A_O = 0.651	d = Boss diameter				Ship model No. 5946
Number of blades	Z	=	4		A_P/A_O = 0.571					Scale ratio λ = 29

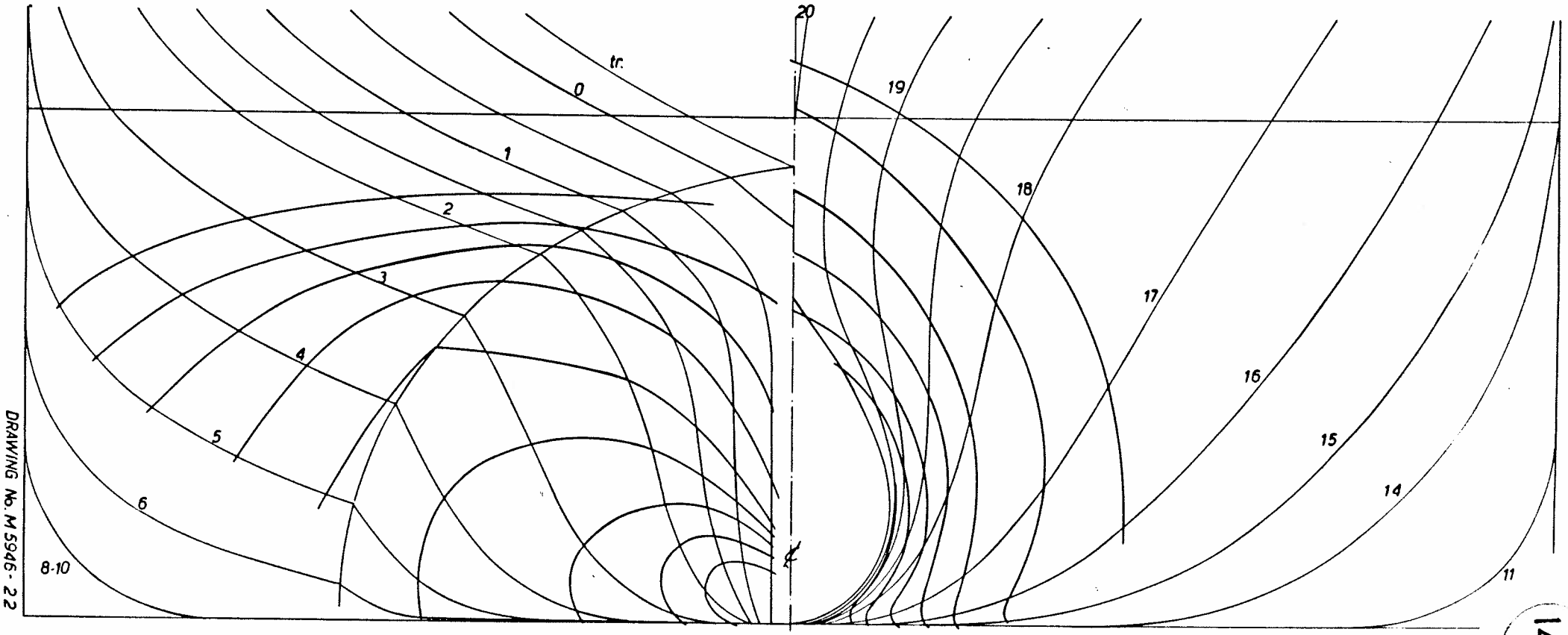
Drawing No. M 5946—12

14-16

SHIP MODEL 5946

STREAMLINES in way of FORE and AFTER BODY

TEST No. 41110 - DRAFT 10.0 m - SPEED = 16 KNOTS - Lpp = 188.11 m



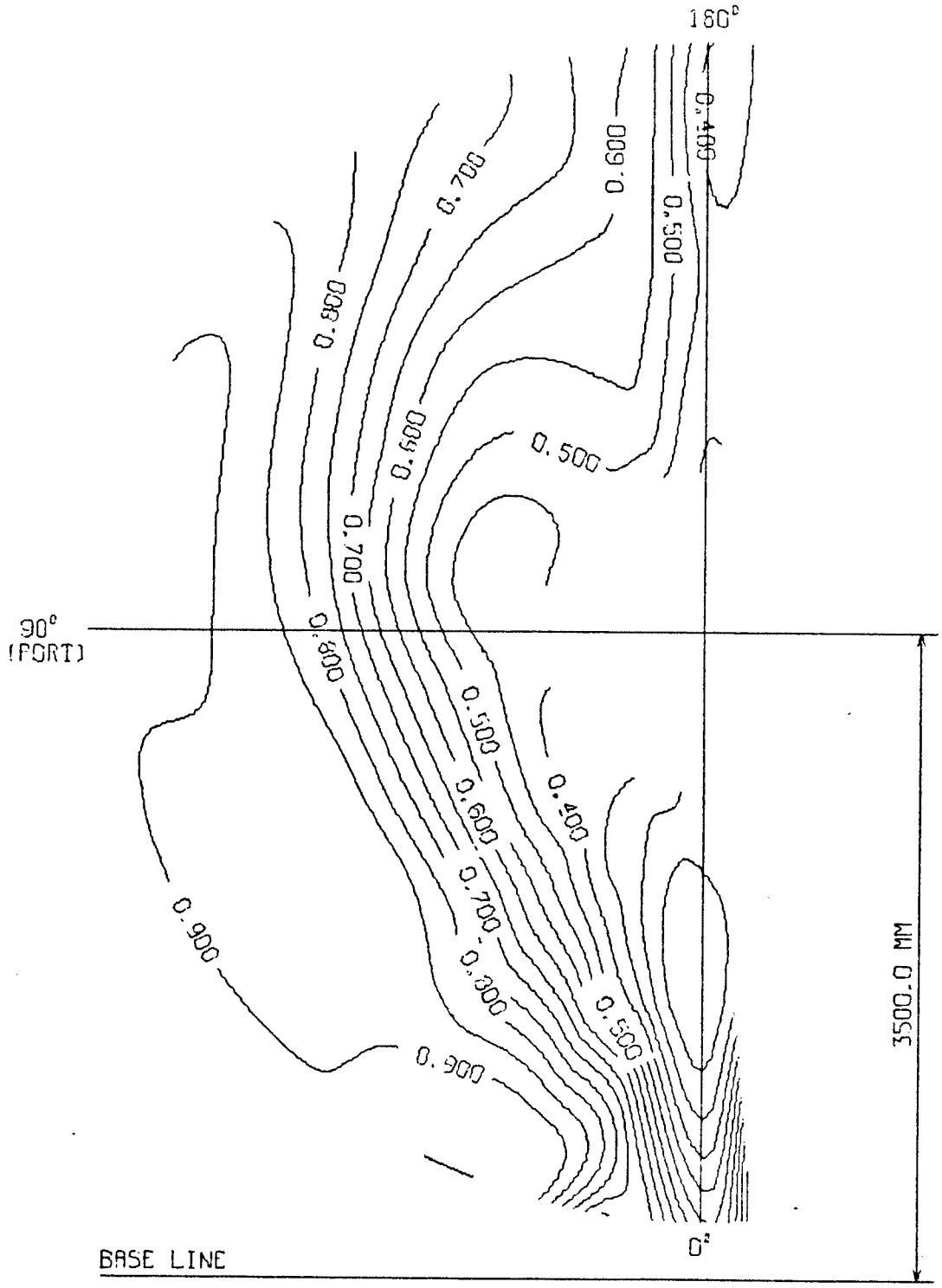
DRAWING No. M 5946 - 22

RESULTS WAKE MEASUREMENTS

14-18

TEST NO. 41157
SHIP MODEL NO. 5946
SHIP DRAUGHT ON F.P. 10.000 M
SHIP DRAUGHT ON A.P. 10.000 M

SPEED 16.00 KNOTS
BULBOUS BOW I
SCREW APERTURE I
WITHOUT RUDDER



LINES OF EQUAL AXIAL VELOCITY COMPONENT (VX/V)

DIAGRAM No M 5946_20

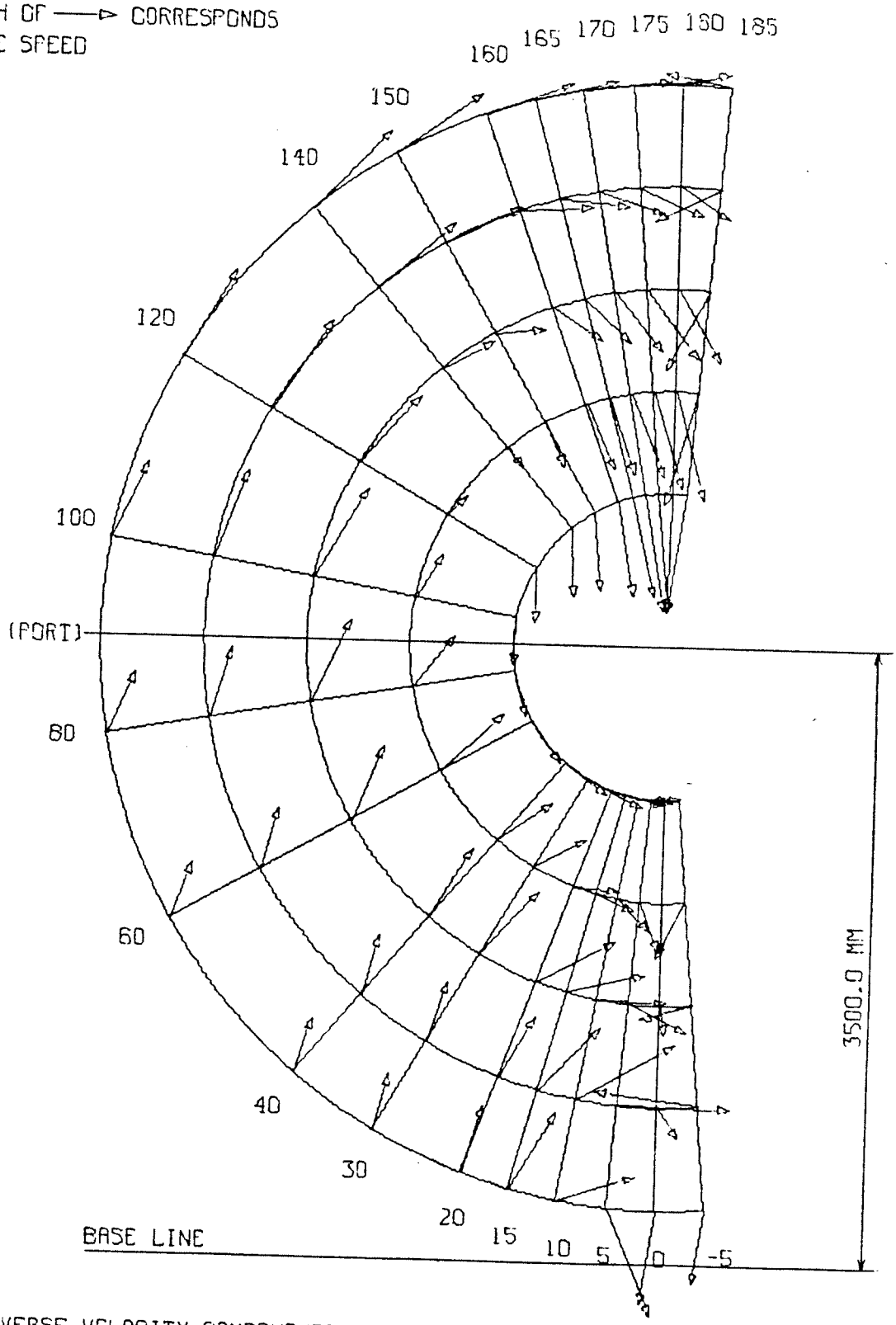
RESULTS WAKE MEASUREMENTS

14-19

TEST NO. 41157
SHIP MODEL NO. 5946
SHIP DRAUGHT ON F.P. 10.000 M
SHIP DRAUGHT ON A.P. 10.000 M

SPEED 16.00 KNOTS
BULBOUS BOW I
SCREW APERTURE I
WITHOUT RUDDER

VECTOR LENGTH OF ——▷ CORRESPONDS
TO 0.1 OF THE SPEED



TRANSVERSE VELOCITY COMPONENTS.

DIAGRAM No. M 5946_21

15

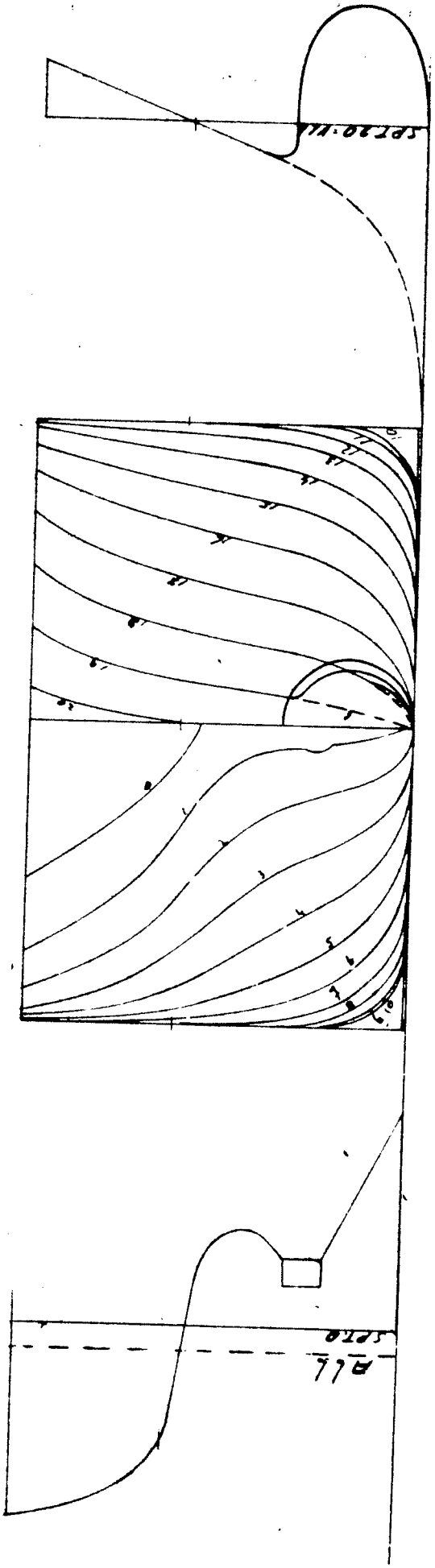
FAST CARGOLINER

1964 ("L" CLASS KRL)

- **CRUISER** STERN
  VERSUS
  TRANSOM STERN
  + INUI FOREBODY

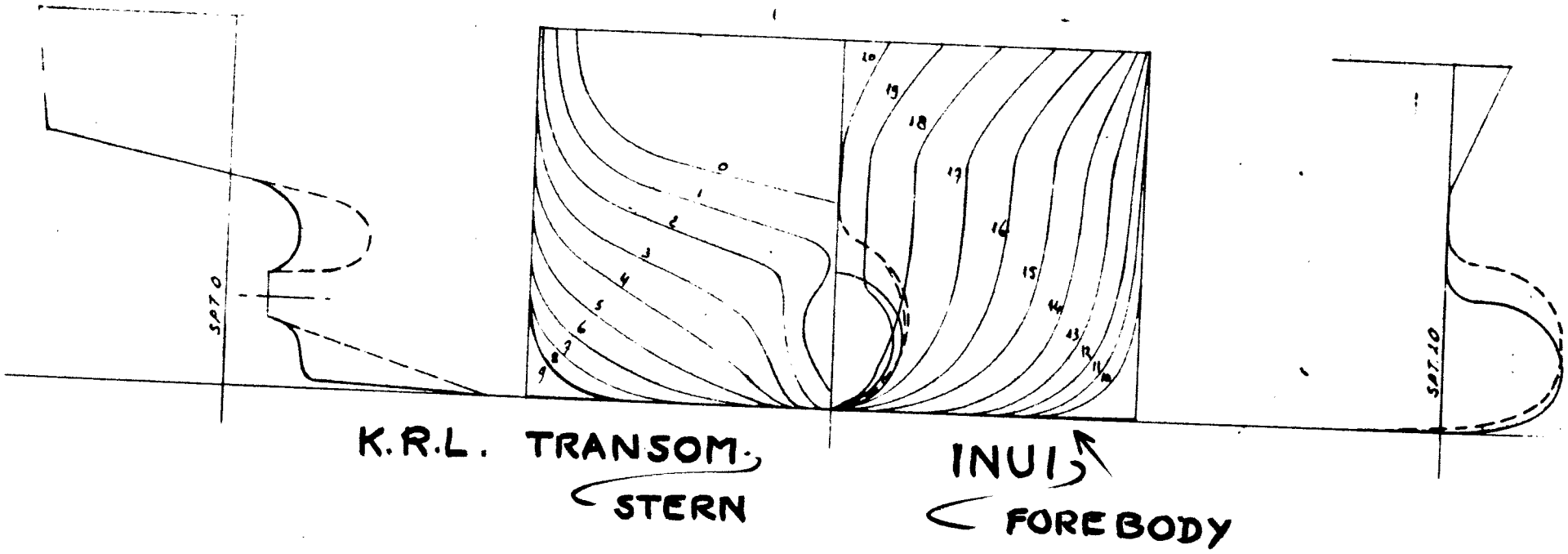
- INFLUENCE ON : STEERING
  ROLLING
  BALL - LOADED

Model no.	2560	2560G
Lengte van spt. 0 tot spt. 20	150,00 m	150,00 m
Breedte op buitenkant spant	22,00 m	22,00 m
Diepgang voor	7,163 m	7,163 m
Diepgang achter	8,077 m	8,077 m
Waterverplaatsing	15593 m^3	15792 m^3
Lijntype	------	------



CONVENTIONAL

Model no.	2745	2745A
Lengte van spt. 0 tot spt. 20	150,00 m	150,00 m
Breedte op buitenkant spant	22,80 m	22,80 m
Diepgang voor	7,163 m	7,163 m
Diepgang achter	8,077 m	8,077 m
Waterverplaatsing	15813 m^3	15620 m^3
Lijntype	— — — —	————



NEDERLANDSCH SCHEEPSBOUWKUNDIG PROEFSTATION WAGENINGEN

Stuurroefrap, ort
NO. 38

Modellen nos. 2560-2560G-2745-2745A.

STANDAARD STUTPROEVEN

Zwenksnelheden als functie van de roerhoek.

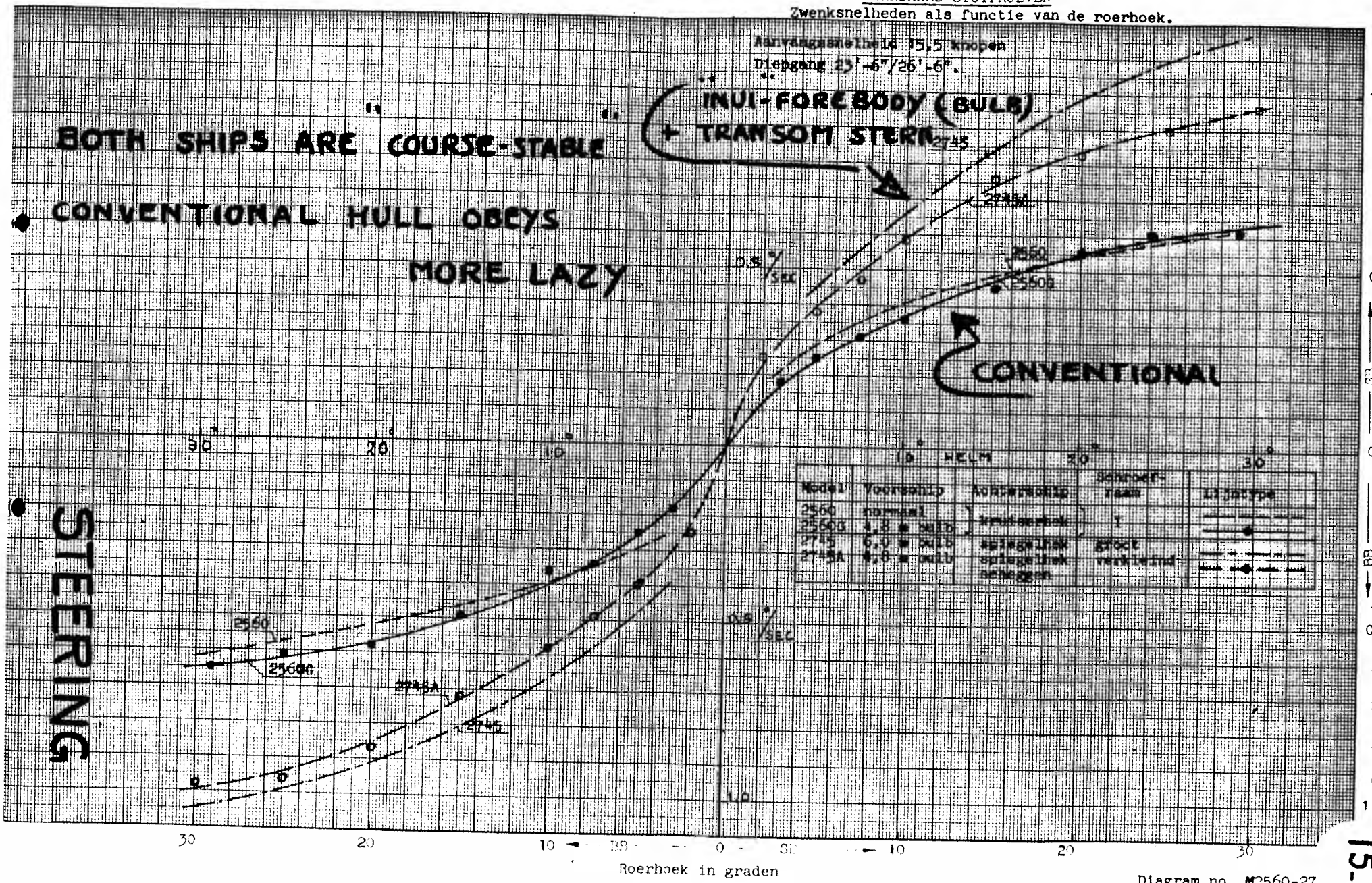
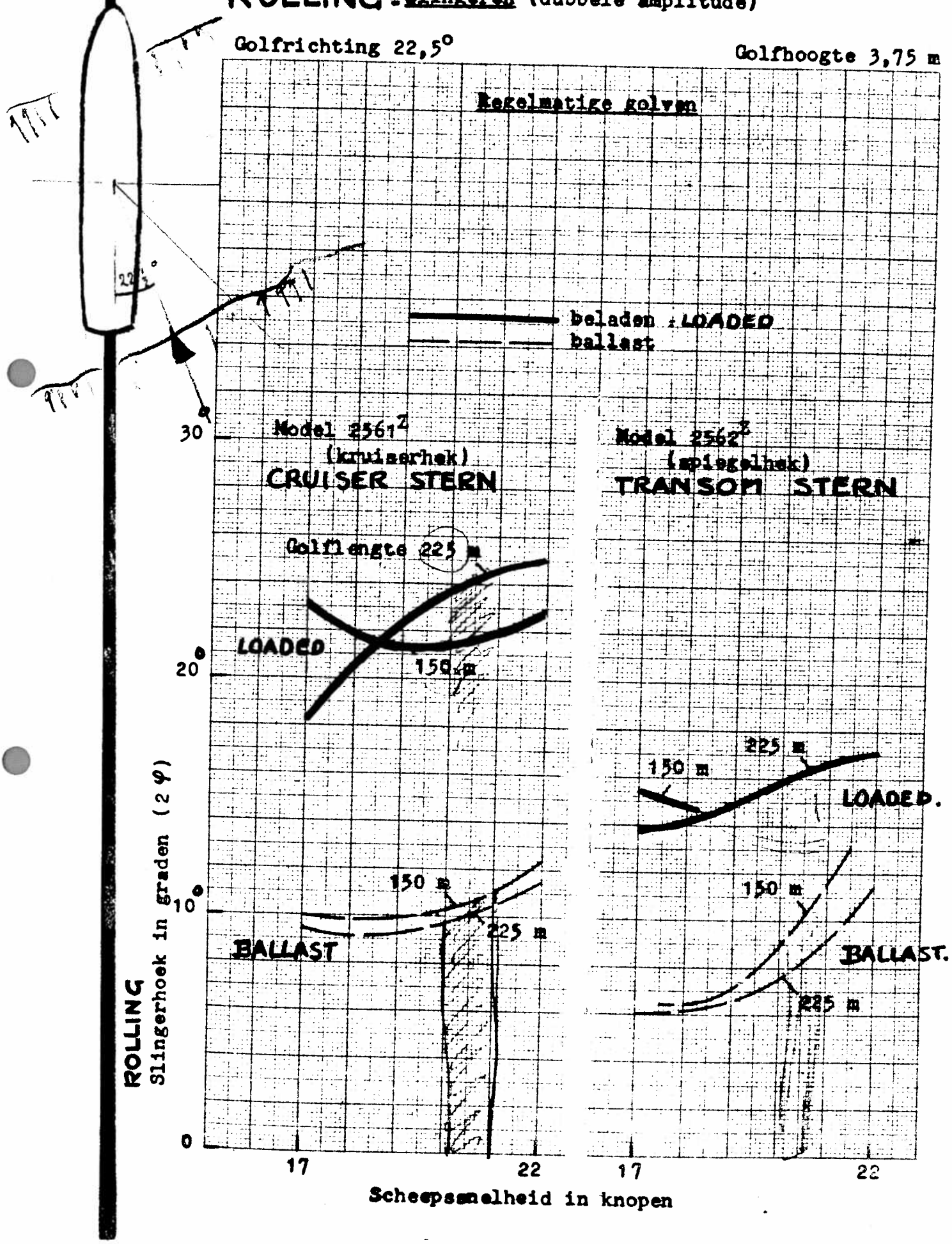
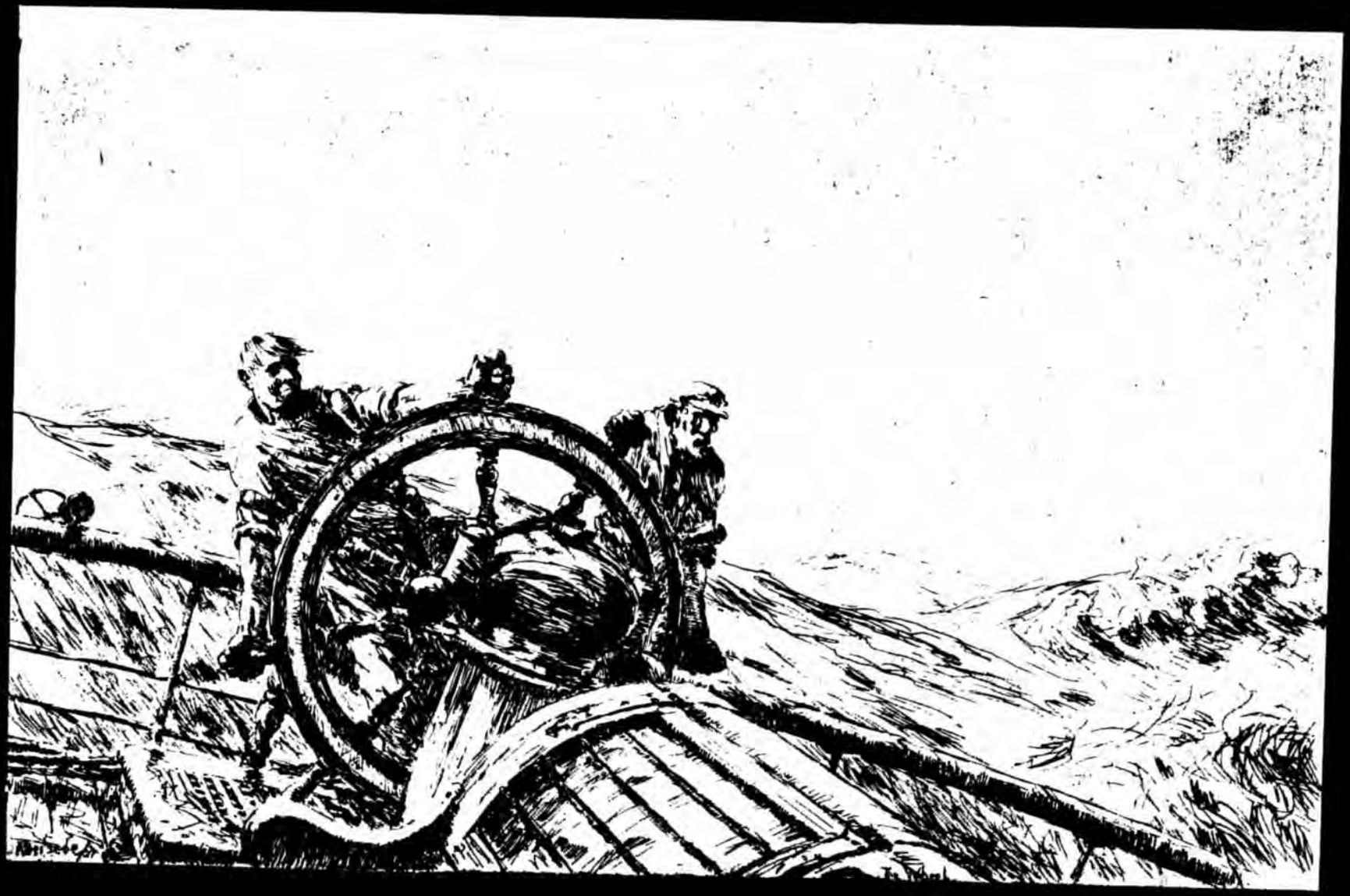


Diagram no. M2560-27.

ROLLING = Slingeren (dubbele amplitude)



16



STEERING

4. STEERING PROBLEMS

The first generation of large European container ships 1969 such as OCL's "Encounter Bay", the "Sydney Express" of Hapag Lloyd, and "Melbourne Express" of NDL and Nedlloyd's "Abel Tasman" were designed in the mid-sixties for the Europe to Australia run via Suez. With the unfortunate closing of Suez in 1967 these 1 600 TEU single-screw-container ships, each powered by a 32 000 shp steam turbine giving 22 knots service speed, could no longer use the route for which they had been designed and were compelled to sail via the Cape of Good Hope for many years.

As a result, the "economy of scale" law was applied and for the same reasons whereby tankers increased in size at this time, to improve their efficiency over the extended sailing distance, the container ships for the Europe/Far East run followed suit, firstly by increasing in size for up to 3000 TEU and secondly by being powered by twin-shaft steam or triple-shaft diesel engine installations totalling some 75000 to 84000 shp per ship, maintaining a service speed of 26 knots. To accommodate these changes, the second-generation design differs considerably from their predecessors.

By the mid 1970's after several years of successful operation, these new ships were badly hit by rising world inflation and particularly by the rapid rise in fuel prices resulting in operation at reduced speeds. Consequently, when it became necessary for the world's container ship operators to build new tonnage, a modern third generation of container ships evolved which are shorter and slower but of the same containercarrying capability as the second-generation vessels. Examples of these are OCL's "Table Bay" 1978 and "Nedlloyd Houtman" and "Nedlloyd Hoorn" of Nedlloyd Lines.

"Encounter Bay" and her sister first generation ships had single screw open-stern-frame arrangements and were fitted with single, centrally-positioned, semi-balanced

rudders. This layout was ideal for Suez Canal navigation, providing excellent manoeuvrability by having the rudder positioned directly in the race of the propeller, thus producing a maximum steering effect, even at low speeds. Combined with a gondola-shaped lower afterbody the ships achieved good course-keeping characteristics when sailing through the canal.
In contrast with these single-screw ships, the twin-screw second generation container ships were fitted with a single rudder, sited centrally between the propellers. This arrangement was well-suited for high-speed ocean cruising around the world, offering the least resistance and the best speed, but unfortunately it has poor low-speed steering capabilities, especially in shallow waters as the propeller races do not hit any part of the rudder. At the outset, sailing the route from Europe to Japan around the Cape of Good Hope and from Japan to Europe via the Panama Canal, the second-generation container ships operated without troubles.

In March 1977, following the succesful re-opening of the Suez Canal for some two years, it was decided to send the container ships via the shorter route; in consequence a number of accidents befell the twin-screw, single-balance ruddered ships, especially those with diverging shaft-lines. It is a challenge to find an explanation for these unfortunate phenomena and even more so to put forward suitable solutions, for new twin screw designs.

In general, all single rudder twin-screw vessels have poor steering capabilities in the speed range up to 8 knots, particularly when sailing in shallow waters with only 10 per cent draught or less water below the keel. Pilots used to say that "ships are hesitating".
Fig. 49-50-51-52-53-54.
The Scandinavian model basin has strongly advised designers not to apply single rudders on twin-screw

ships. Manoeuvring tests with 1:50 models under aforementioned unfavourable circumstances of draught, have indeed shown frightening results. Fig. 54. Between 6-8 knots and 1,2 m of water below the keel, a "dead-band" of some 10 degrees to port and to starboard was observed.

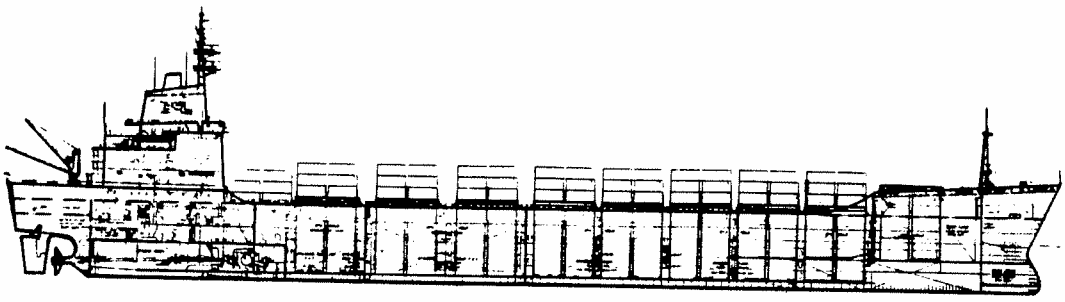
In actual practice the steering qualities are less unfavourable. Our reasoning for this is based on the influencing thickness of the boundary layer, which on a scaled-down model is too large in comparison with that of the actual vessel (e.g. on a 1:50 model, the boundary layer is relatively four times too large). Small scale maoeuvring tests with models of some 3.5 to 4 m length should in our opinion therefore be utilised mainly for comparison purposes between the various configurations and not for full-scale prediction.

POSSIBLE IMPROVEMENTS

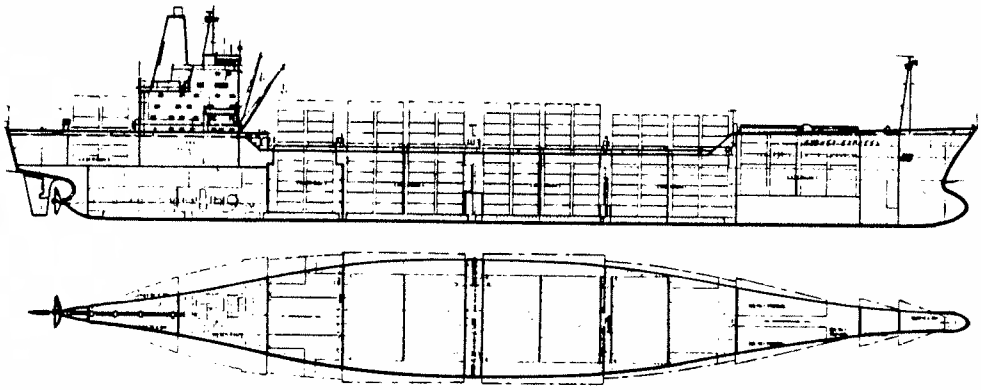
How can the effectiveness of an existing single semi-balanced rudder, twin-screw arrangement, which shows "confined water flaws" be improved? When enlarging the rudder, because of the fact that allowable torque is limited by the dimensions of the stock and steering gear, the rudder length can only be increased by extending the leading and trailing parts simultaneously to keep the balance intact. The resulting higher load on the lower bearing creates accelerated wearing of bush and sleeve.

Wing-rudders, in outrigger fashion as fitted to the Esso "Harper" class and connected by means of brackets to the main rudder and exposed to the propeller races, are a proven remedy to improve poor steering of twin-screw tankers and should also be a great improvement for container ships. Fig. 50.

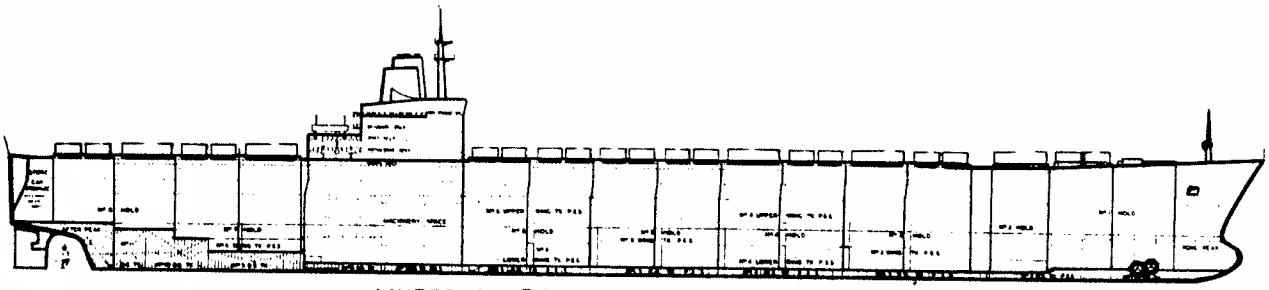
(Figure 49)



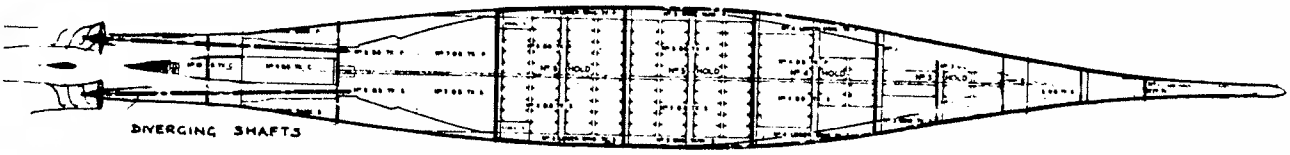
ENCOUNTER · BAY



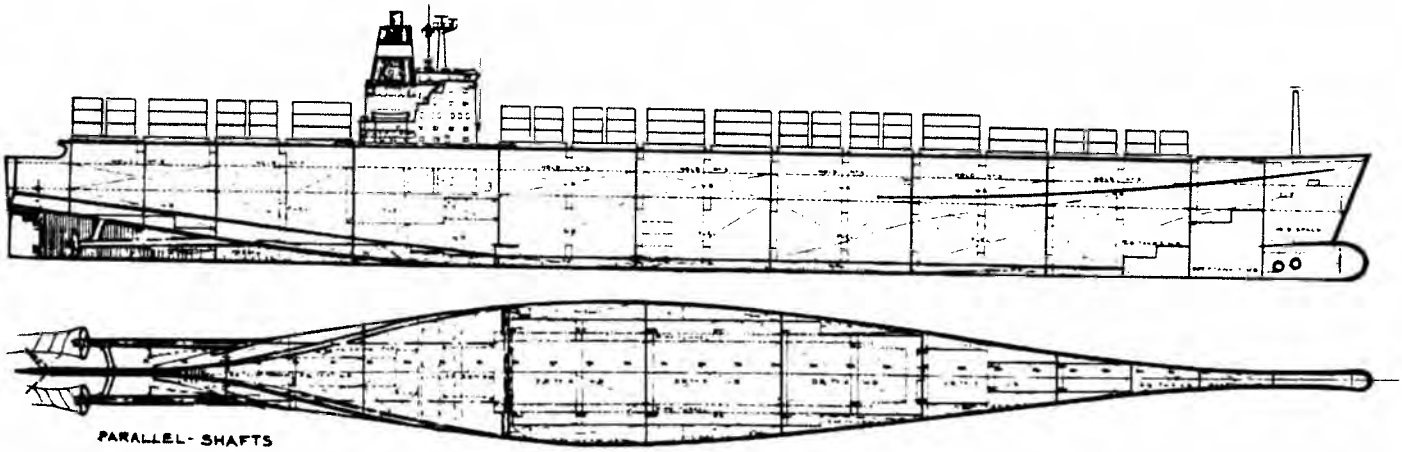
SYDNEY EXPR./ABEL TASMAN



LIVERPOOL · BAY



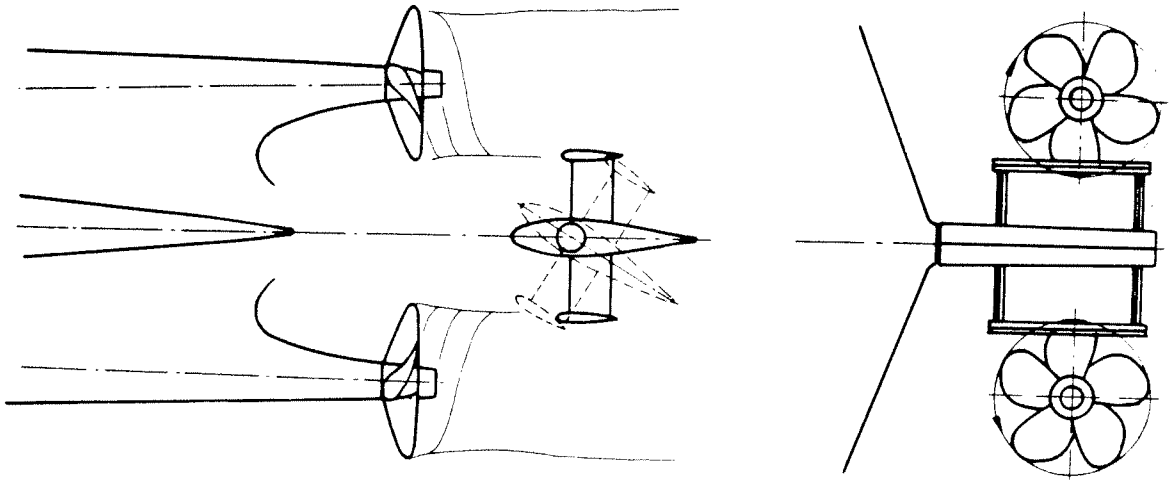
DIVERGING SHAFTS



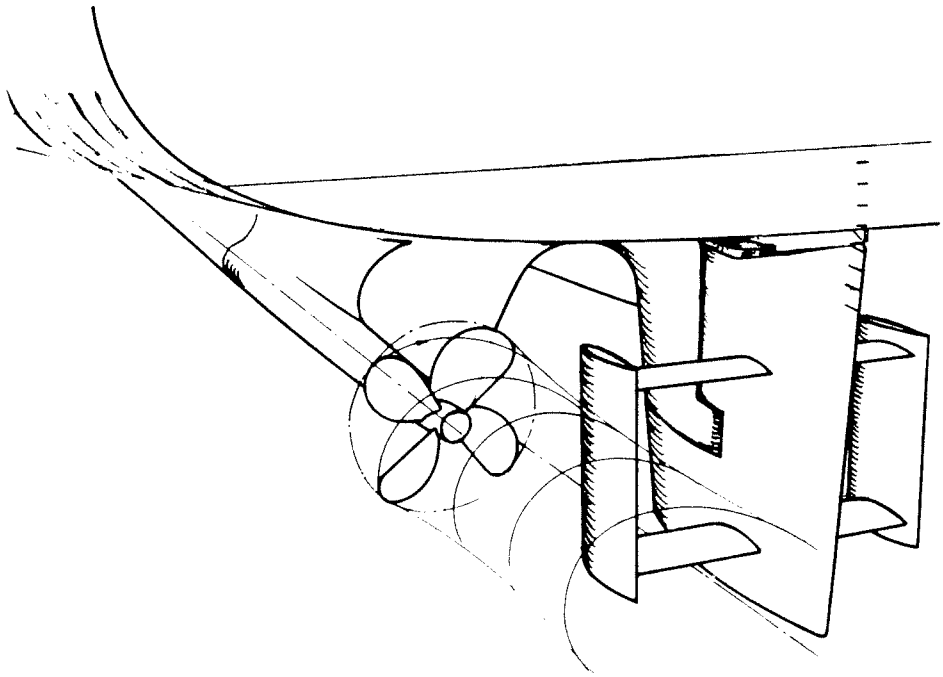
PARALLEL - SHAFTS

NEDLLOYD · DEJIMA

STEERING IMPROVEMENT



AUXILIARY "OUTRIGGER" RUDDERS



TWIN-SCREW SINGLE RUDDER

The twin-screw, single-rudder arrangement has three basic veriations: Fig. 39.

a) The semi-balanced rudder behind an open skeg, such as fitted to the "Liverpool Bay" class, "Table Bay" class, "Helderberg" class and "Sea-Land Market" class container ships, also the "QE2", "Remuera" and "Portelius".

b) The "fat-head" semi-balanced rudder behind a closed skeg fitted to "Hamburg Express" and "Tokyo Express".

c) The skeg-"barndoor"-rudder behind a closed skeg, fitted to "Bremen Express", "Hong Kong Express", Nedlloyd Dejima", "Nedlloyd Delft", "Nedlloyd Houtman", "Nedlloyd Hoorn", and "Transvaal". (NEDLLOYD - BREMERVULKAN - HDW - DAL).

After 10 years operational experience on the Europe: Far East services, where twin-screw single-rudder ships of the three mentioned types (a), (b) and (c) are sailing, the following conclusions can be drawn:

Resumé:

a) Semi balance with open skeg and diverging shafts.
   - bad steering in shoal water and canal at low speed;
   - good steering at full speed at sea;
   - maintenance on pintle bearing.

b) Fat had semi-balance rudder behind closed skeg.
   - good steering with small angles in shoal water;
   - bad steering if large rudderangles are needed;
   - large turning circle;
   - good steering at full speed at sea;
   - maintenance on pintle bearing.

c) Skeg-rudder (barn door) behind closed skeg.
- good steering in shoal water;
- at 40° helm in propeller race the ship will respond.
- good steering at full speed at sea;
- steering gear, tiller and stock more expensive to produce! (see flanged tiller); fig. 66.
- no maintenance on pintle bearings in case the right materials are applied) no lifting of rudder. Fig. 65-66.

a) <u>The semi-balanced rudder behind an open skeg (Fig. 51 Type a)</u>

This was a normal arrangement on former passenger vessels, giving good deep water steering performance at full speed and a relatively small turning circle. Passages through the Panama Canal were reported as trouble-free-there being ample water below the keel, but passages through the Suez Canal, however, met with many troubles. This is explained as a consequence of there being only shallow water below the ship's bottoms and the canal's relatively small cross section dimensions. Under these conditions water flow towards the rudder is hampered and the balanced-rudder blade find itself in a disturbed water mass between the propellers which are wide apart.
A ship sometimes passes close to one of the canal sides (e.g. by pressure of sidewind) and under these conditions the rudder, being surrounded by disturbed water, is unable to provide a proper steering force. In this way, several groundings came to pass and/or damage was sustained by ships touching the sloping side walls of the canal.

b) <u>The "fat-head" semi-balanced rudder behind a closed skeg (Fig. 51 Type b)</u>

This is a special arrangement which was developed at HSVA-Hamburg and Blohm & Voss.

It has a good steering performance at full speed on a straight course, but the turning circle diameter (Fig. 3) is about 1.5 times larger than that of the open skeg configuration (Fig. 57-58).

When the rudder is hard over, the small gap ahead of the rudder nivellates the difference in pressure between both sides of skeg + rudder thus reducing steering performance.

Difficulties were met at a sharp bend in the Panama Canal and one ship grounded. In contrast, the passage through the Suez Canal is without troubles because of the good steering performance of the rudder at small angles of helm (no gap between skeg and the fat rudder-head - Fig. 4) and only slightly curved bends are to be overcome in this confined canal.

c) <u>The "barndoor" rudder behind a closed skeg (Fig. 51 Type c)</u> (mr. Koning, Mr. Kamps, Mr.van der Klis,Wageningen)

This arrangement gives good steering performance at full speed in deep water. Course-keeping is excellent, especially in following seas and swell, an important fact with modern wide-transom-stern ships which are more prone to the broaching effect of stern seas than old finer shaped cruiser-stern ships.

The overshoot angle is some 40 per cent smaller that in the case of the semi-balanced rudder arrangement and the lapse of time needed in steadying the ship is some 20 per cent shorter. Both these features are advantageous but in contrast the diameter of the turning circle is some 8 to 10 per cent larger than that of the ship with semi-balanced rudder and an open skeg, when applying 35 degrees of helm in both cases.

Steering in the Suez Canal gives no difficulties. The good steering attributed to a "fishtail effect" created by the uninterrupted deadwood and skeg rudder surfaces on which a (port versus starboard) difference in water pressure is built up, extending far ahead of the actual rudder. It was found, however, that any opening ahead of the rudder reduced this effect. On particular vessels mentioned under this class the steering gear is capable of giving 45 degrees emergency helm at speeds below 16 knots. Moreover, the horizontal length of the skeg rudder (6 metres) is such that the trailling edge enters the propeller race at 40 degrees of helm and the ship then always responds. A sharp turn at low speed should be carried out by giving 45 degrees of helm and ordering full "ahead" for the propeller on the side one wishes to turn to, which gives a push on the rudder surface (equivalent to a single-screw ship effect). Under these conditions the turning circle diameter will be small.
The fact that a "barndoor"-rudder is not popular to the yard, is caused by the higher price for the stronger steering gear and rudderstock (Fig. 65-66). For the owner, this rudder is very reliable and can be left during 20 yrs without taking it out. The semi balance rudder needs lifting 1 à 2 times per 20 yrs in order to renew the liner of the heavy-loaded pintle.

PROPELLER STEERING ON TWIN SCREW/SINGLE RUDDER SHIPS BY PILOTS

Very often, pilots try to change heading by using the thrust and pulling forces exerted by the propellers. They expect to generate a ship-turning moment towards the desired new course with the least offset of the stern from a courseline. The method is effective only on relatively short ships, having their shafts arranged wide apart, as on supply boats and ferries which can

generate large turning moments. When ships are extremely long, like those of the Far East container service, with shafts arranged relatively close to the centreline, the steering effect by the turning moment of the propeller thrusts will be small.
If the shaft lines are not run parallel and their extended centrelines intersect close before the stern or-even worse-inside the ship near the forepeak (or just behind the forepeak bulkhead), the steering effect solely from the propeller thrusts is negligible as there will be no or hardly any moment (Fig. 61-62)

Converging : "shaft centrelines should be avoided when the intersection lies behind the forward perpendicular" is considered a law by the Wageningen Model Basin. A few large container ships, however, do have slightly converging shafts based on following reasons :

a) Fear of vibration from propeller-induced pressure fluctuations on the skeg. This fear tends, in the design stage, to move the propellers more outwards (the propeller race is then directed further from the centreline rudder which makes steering by means of a centreline rudder worse too).

b) The tendency to keep the engine-room as far aft as feasible which needs the engines to come nearer to each other. This also creates sufficient height of the main engine foundation in the afterpart of the engine-room double bottom. Closer main engines enables the diesel generators to be positioned on the engine-room tanktop, outside the main engines.
This diverging shafting arrangement has caused a lot of grounding because the ship did not obey to propeller steering and the semi balance rudder was moving in the disturbed mass of water between the propellers.
It is much better to design the ship with parallel

shafts :

- use the propeller on the side you want to turn.
- give 45° helm to the skeg rudder in order to get the propellerrace on its surface.
- the ship will obey immediately.

Fig. 62

Pictures in the wheelhouse must show pilots how to manoeuvre.

TWIN-SCREW-TWIN-RUDDER Fig. 52-53-54-55-56

This arrangement is far superior in low-speed shallow water steering to all single rudder arrangements and should always be adopted for short sea ships such as ferries, supply boats, and pilot vessels. Sometimes there is lack of course-keeping ability in following seas, in which case the centre skeg should be extended to the after perpendicular. The turning circle is very small and in shallow water, at low speed, the vessel responds immediately because of the influence of two propeller races directly and fully onto two rudders.

Ferryboat captains

a) The Twin-Screw-twin rudder ferry fitted with CP propellers will steer excellent when speeding up and having to make a sharp turn.
However, this vessel will obey badly when slowing down, before entering her berth or a lock because the CP propellers at almost zero pitch will blank of the rudders.

b) The Twin-Screw-single rudder ferry will steer badly when speeding up and having to make a sharp turn.

However, this vessel will obey excellently when slowing down because of the fact, the the centre-line rudder still has sufficient waterflow along and can act as fishtail.

A sacrifice is the extra resistance created by the freely exposed twin rudders when compared with the "shadowed" single rudder. This extra can be at least 3½ to 4 per cent more and works out as substantial amount on the fuel bills for a high-speed steamer in a worldwide service. For this reason, twin rudders were not adopted for the Far East service container vessels.

During the planning stage, extensive tank tests were undertaken with the outcome that the increment of power needed for twin-rudder sets, as compared with the single-rudder, was appreciable:

Initially measured at 7 per cent it was reduced to 3.5 per cent on conclusion but this was still considered too costly.

The steering gear of a twin-rudder, semi-balanced installation might not necessarily be more expensive in comparison with a single balanced rudder, since high torque steering gear is not needed, but the sternframe castings, however, will be somewhat more expensive.

TRIPLE-SCREW, SINGLE-RUDDER Fig. 1-14-15

Steering is superior to the twin-screw, single-rudder combination. It was first adopted in 1930 on the passenger vessels "Oranje" of the Netherlands Steamship Co. (the former S.M.N.), on the "Ruys", "Tegelberg" and "Boissevain" - (Royal Interocean Lines).

The arrangement is especially suitable in combination with direct-coupled low-speed diesel engines to modern high-speed ship propulsion. Thus the Scandinavians, in their early-1970's services to the Far East, turned up with such arrangements on their container vessels "Nihon" (Broström), EAC's "Selandia" and "Jutlandia" (EAC) and "Toyama" (Wilhelmsen).

Steering when sailing at full speed at sea is excellent. During the Suez Canal passage and when moving in port,

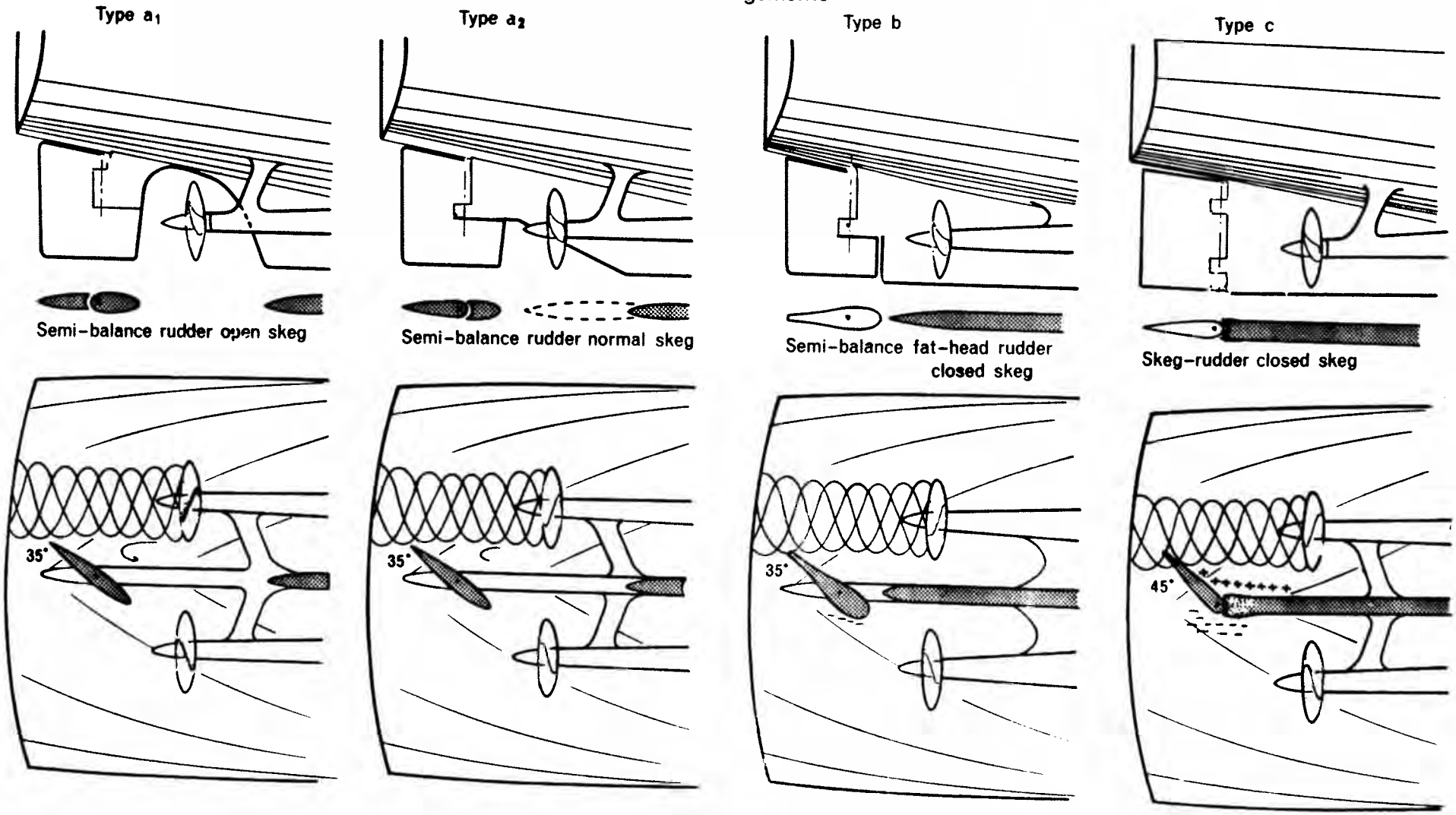
the wing propellers (called "boosters") may be disconnected from the engine and left "windmilling". The ship then behaves much as a single-screw ship, which we have already noted as being of excellent performance in confined waters and is also more familiar to pilots than a twin-screw single-rudder ship. Maoeuvring is realised by the centre propeller, positioned just ahead of the single-semi-balanced rudder which operates directly in the full propeller race.

On the above Scandinavia vessels, the centre propeller is of the controllable-pitch type. In this respect it must be mentioned that a c.p. propeller has serious effects when turning at "zero" pitch by shielding off the rudder, thus depriving it of all steering capacity. Experience, however, has built up a special way of sailing at low speed. A weak water race towards the rudder is developed by setting a small amount of pitch ahead and if speed ahead is building up too high, short jerks of astern-pitch are applied to keep a vessel's ahead motion within limits. The centre propeller turning in a single-screw wake field with 60 per cent peak value, will of course, excite stronger propeller vibratory forces than in the case of two wing propellers working in a wake field with 25 per cent peak value, so there is more reason to fear vibration inconveniences in comparison to the twin-screw ship.

Recent tanktests show that triple screw propulsion requires 6-6½% more horsepower for the same speed compared to twin screw.
Fuel consumption figures on the Far East service show same tendency where the effect of the centre propeller being CP has to be added.

Meanwhile test show that the twin screw ship needs about 0 - 4% more horsepower compared to single screw.

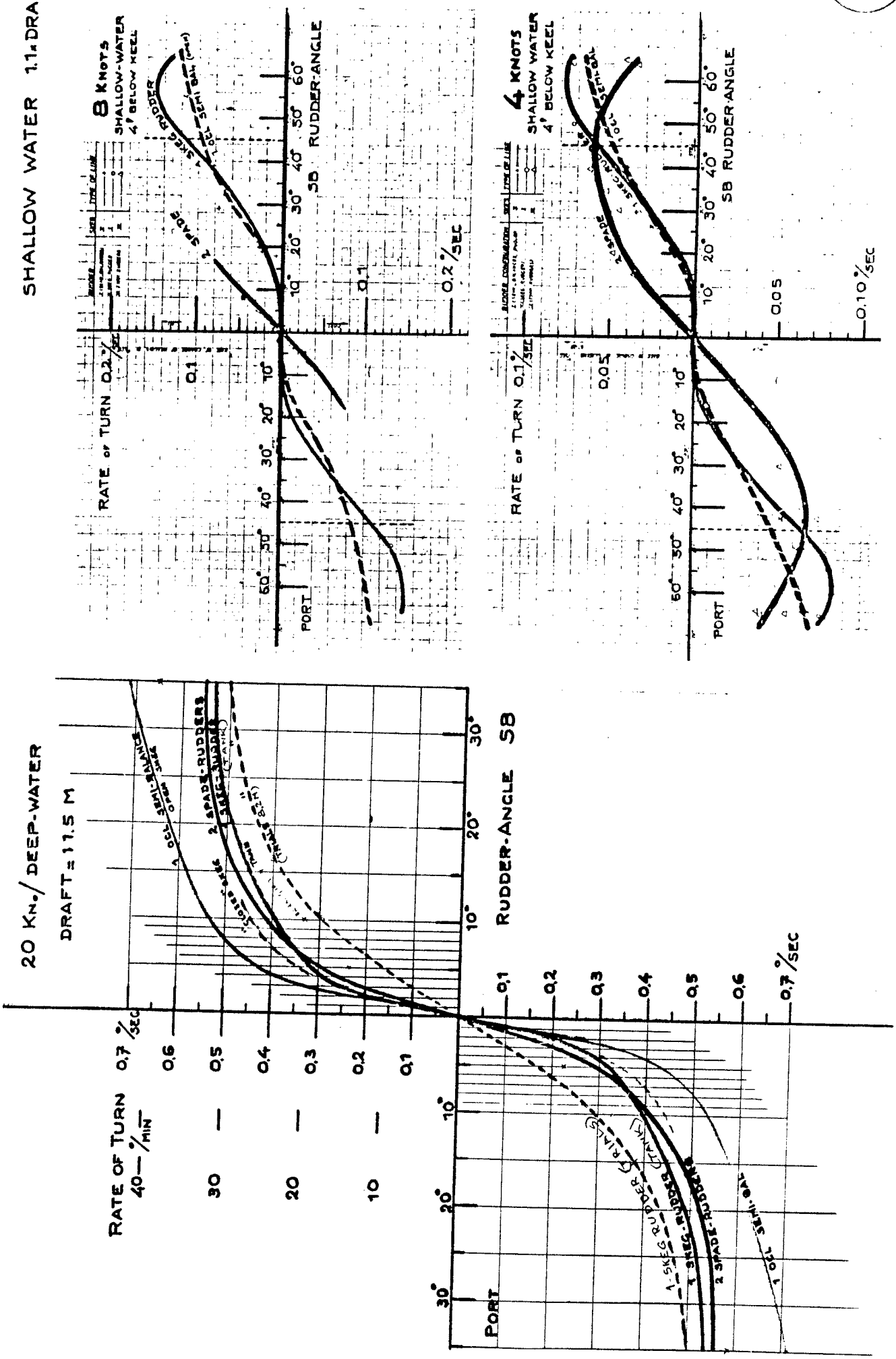
26 Knot Far-East containships-single rudder twin-screw arrangements



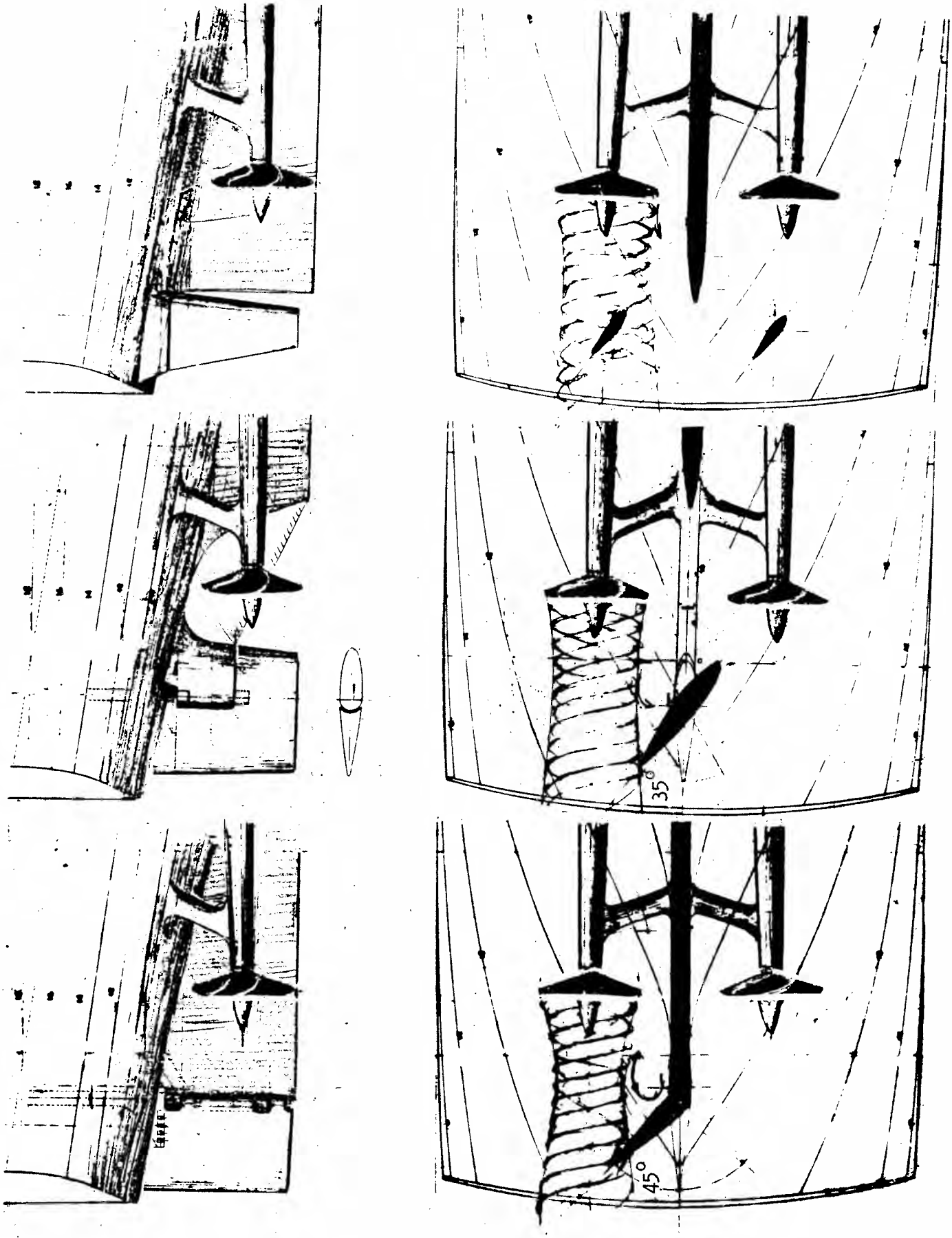
(Fig. 51)

(Figure 54)

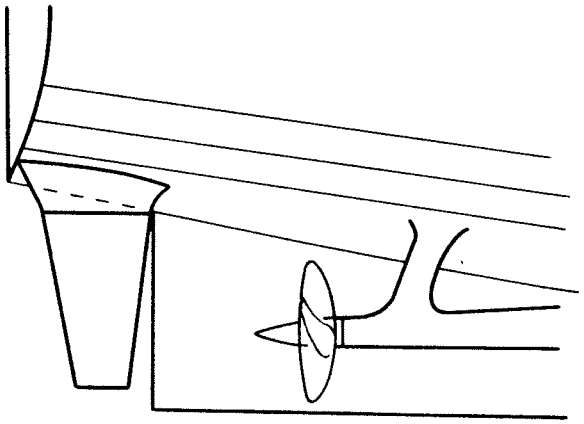
RUDDER - RESPONSE



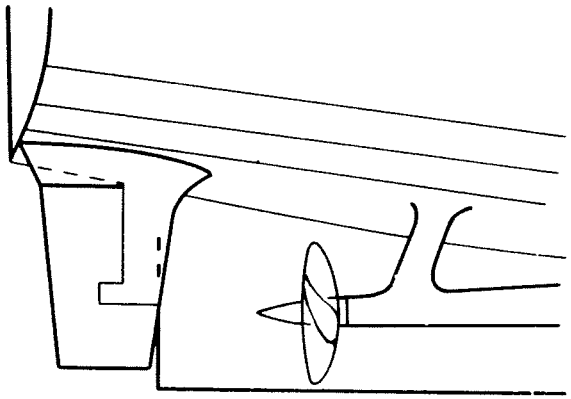
(Figure 53)



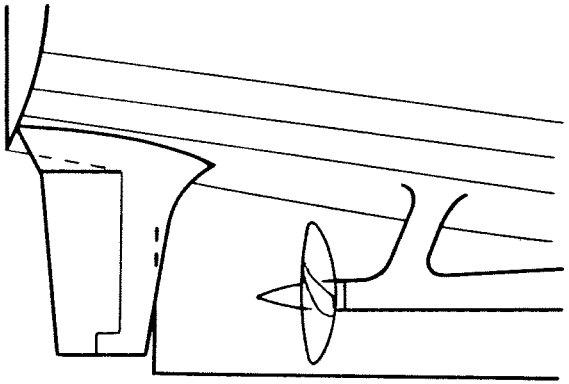
Twin-rudder twin screw arrangments



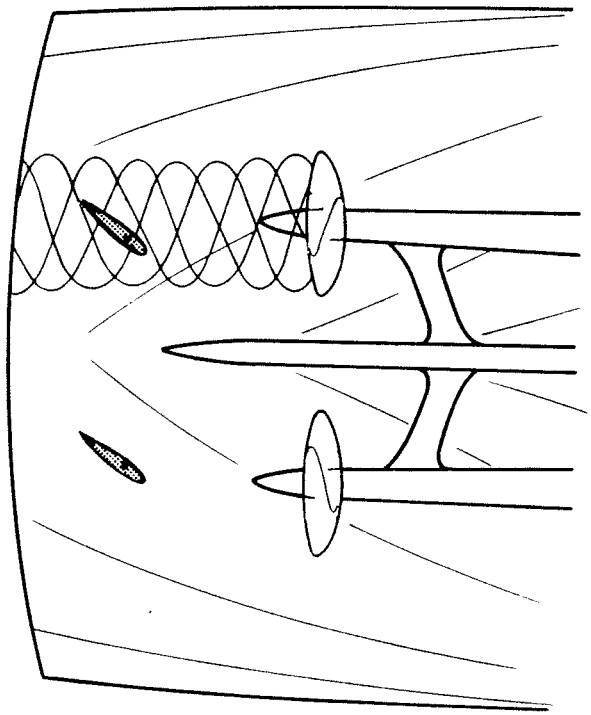
Twin spade rudders

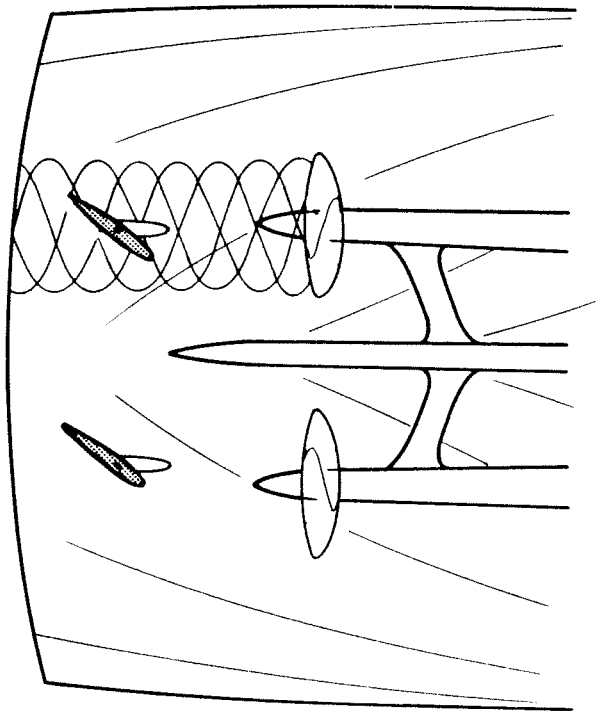


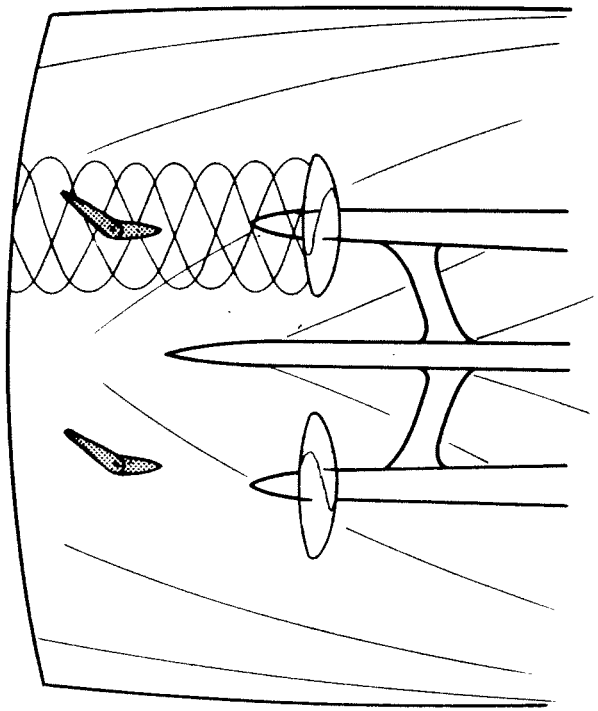
Twin semi-B rudders



Twin Oertz rudders







(Figure 52)

NETHERLANDS SHIP MODEL BASIN WAGENINGEN	REPORT No 70-140 S.B.T.	PAGE 1

INITIAL-STEERING AT 4 KNOTS IN SHOAL WATER

$H/T = 1.1$

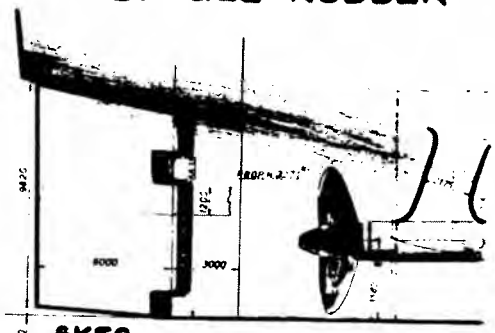
TRIO CONTAINER SHIP - MODEL 3978

INITIAL STEERING TEST

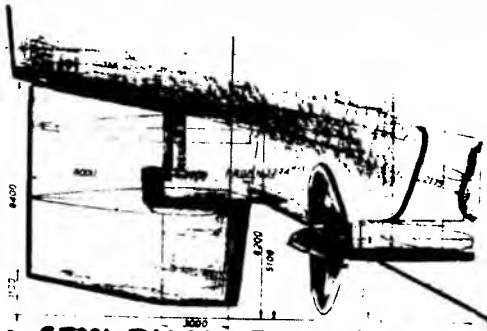
H/T = 1.10 SPEED 4 KNOTS ZERO DRIFT ANGLE

PROPELLER SPEED 53.9 RPM

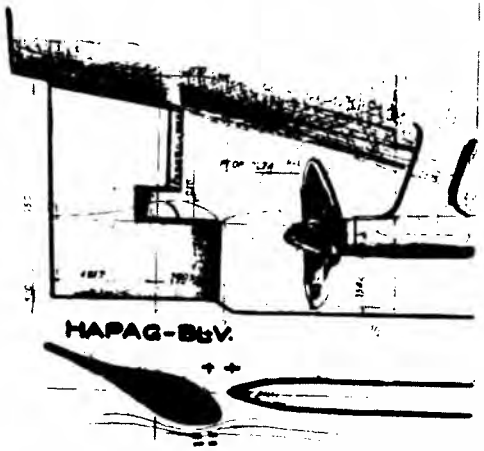
SINGLE RUDDER



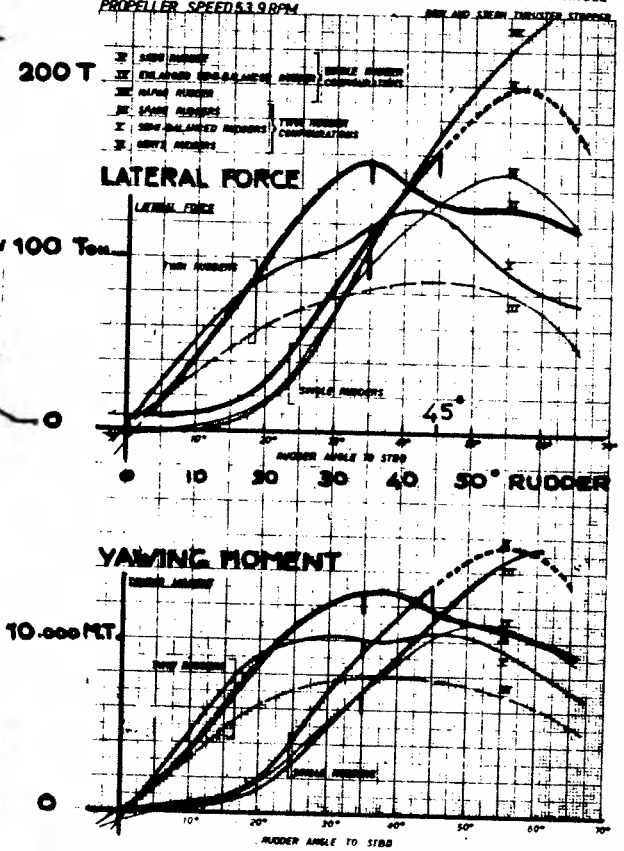
SKEG



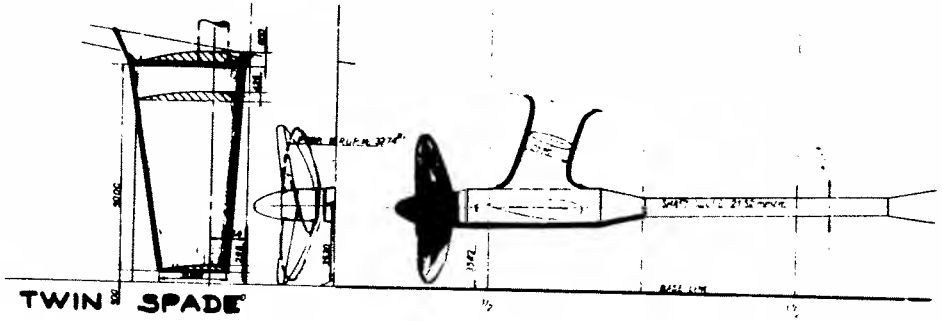
SEMI-BALANCE



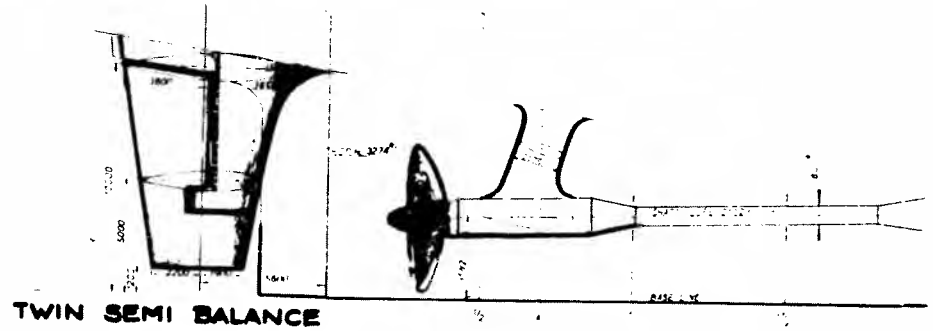
HAPAG-BbV.



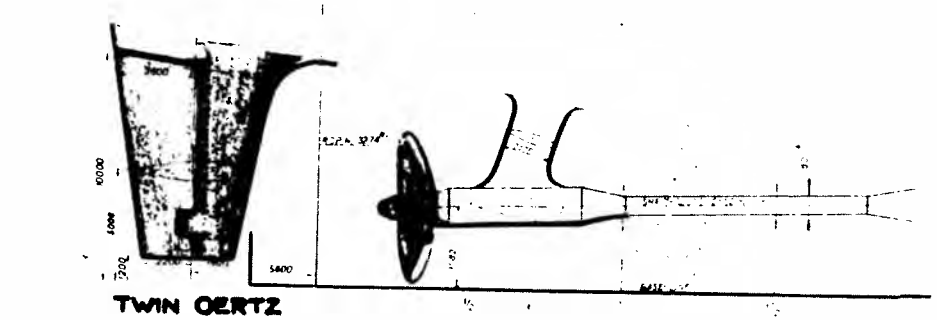
TWIN-RUDDER



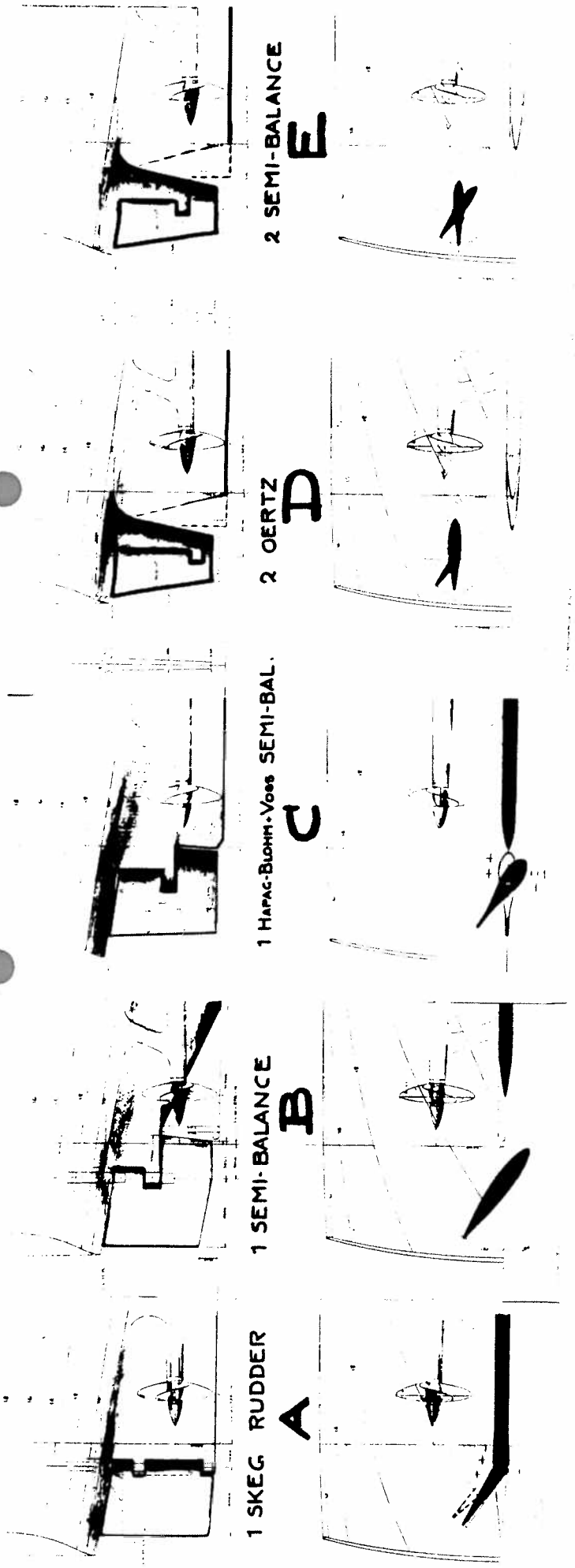
TWIN SPADE

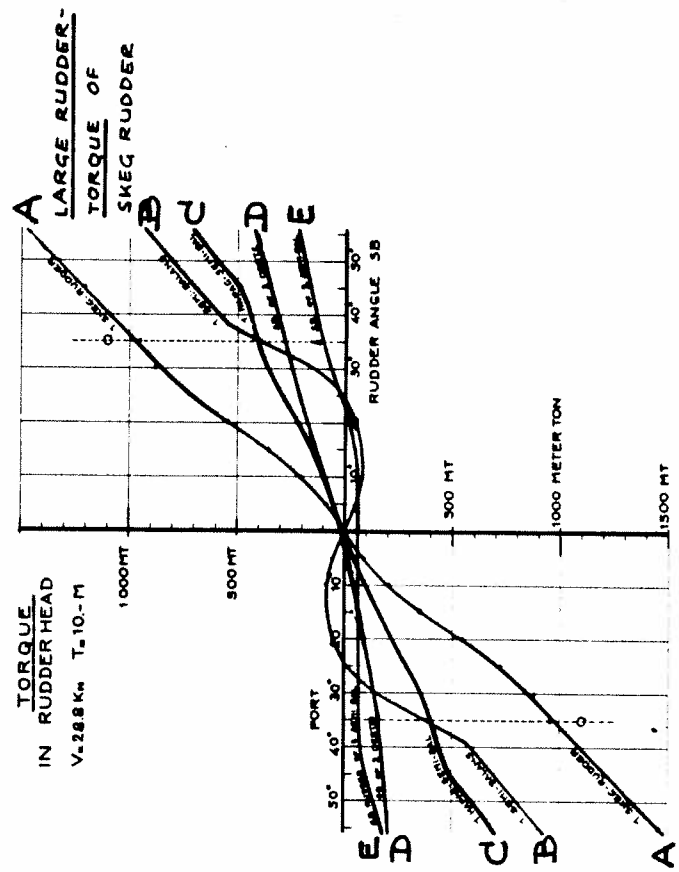


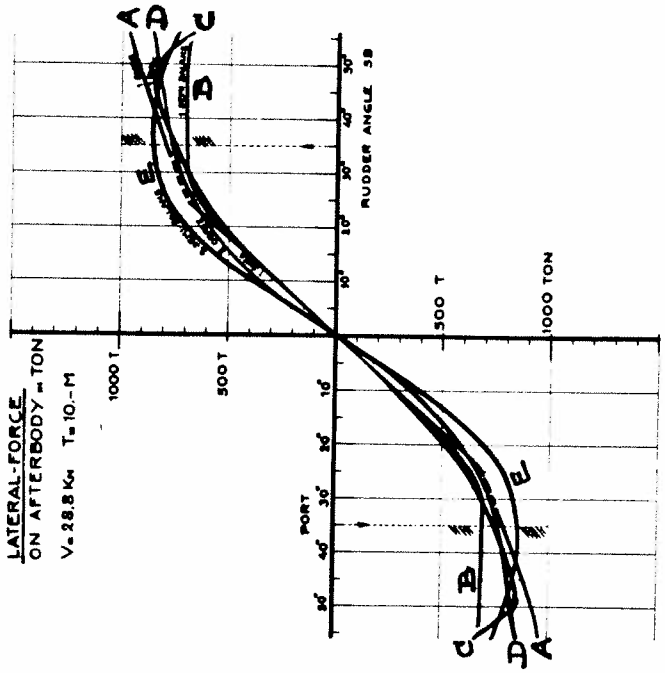
TWIN SEMI BALANCE



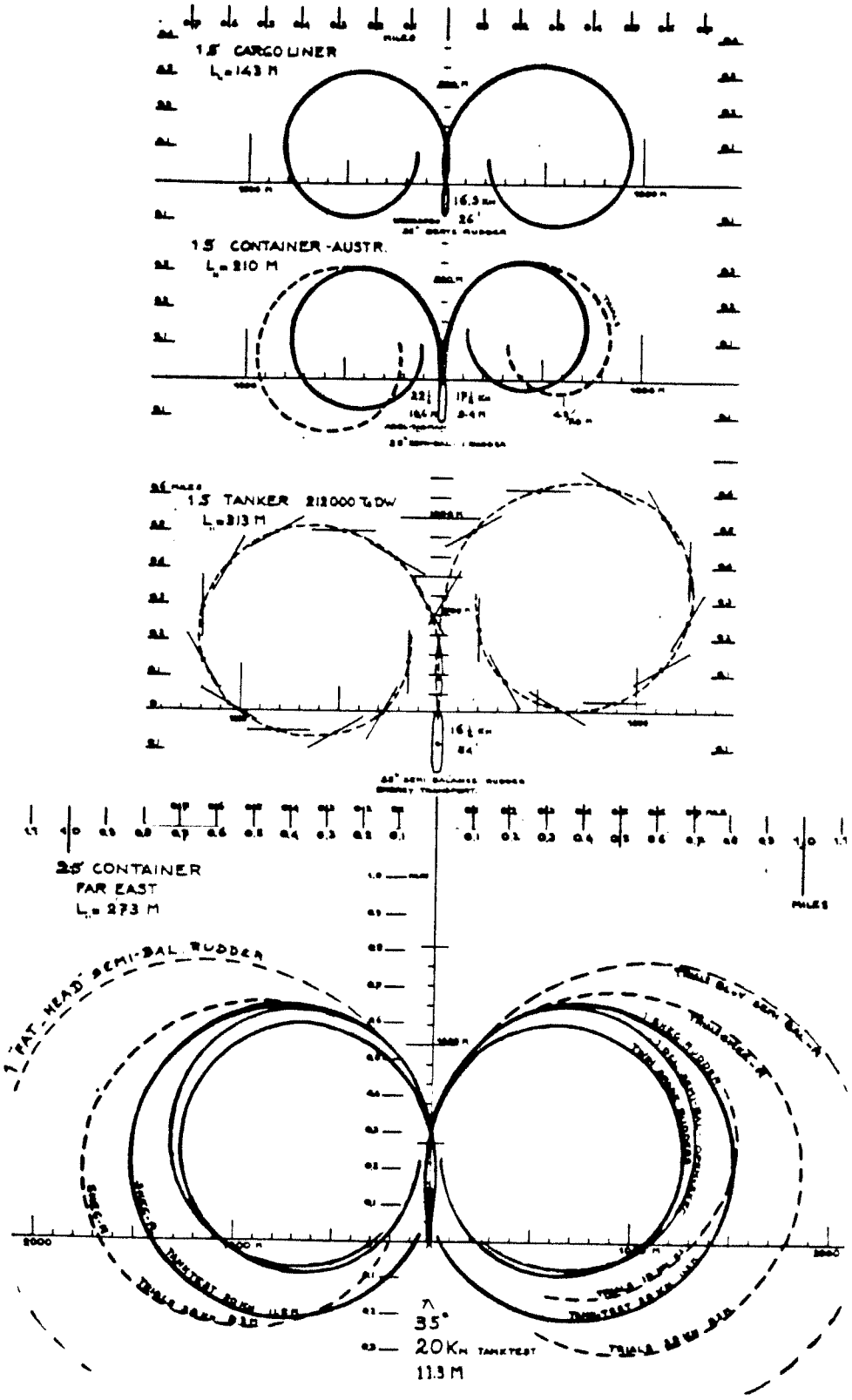
TWIN OERTZ


1 SKEG RUDDER
A
1 SEMI-BALANCE
B
1 HAPAG-BLOHM+VOSS SEMI-BAL.
C
2 OERTZ
D
2 SEMI-BALANCE
E

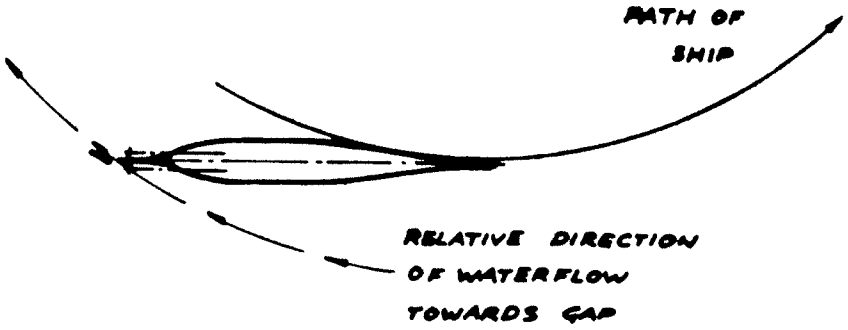

TORQUE
IN RUDDER HEAD
V=28.8 Kn T=10.-M
LARGE RUDDER-
TORQUE OF
SKEG RUDDER
1000 MT
500 MT
300 MT
1000 METER TON
1500 MT
PORT
RUDDER ANGLE SB
A
B
C
D
E


LATERAL-FORCE
ON AFTERBODY = TON
V=28.8 Kn T=10.-M
1000 T
500 T
300 T
1000 TON
PORT
RUDDER ANGLE SB
A
B
C
E

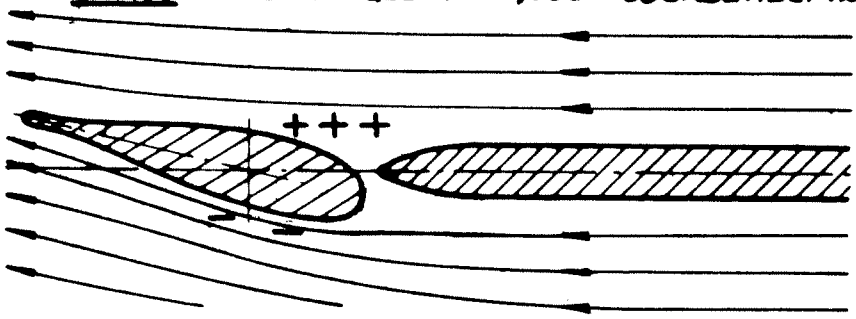
(Fig. 57)

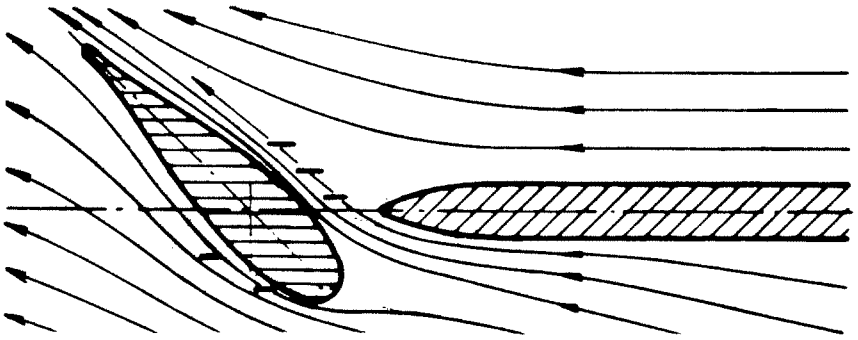

1.5' CARGO LINER
L=143 M
1.5' CONTAINER-AUSTR.
L=210 M
1.5' TANKER 212000 T DW
L=313 M
2.5' CONTAINER
FAR EAST
L=273 M
MILES
35°
20Kn TANKTEST
11.3 M

STEERING PROPERTIES OF THE "FAT HEAD" SEMI BALANCE RUDDER CLOSED DEADWOOD

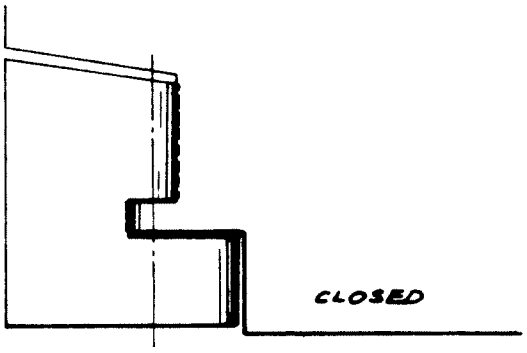


AT SMALL RUDDERANGLE : GOOD COURSEKEEPING





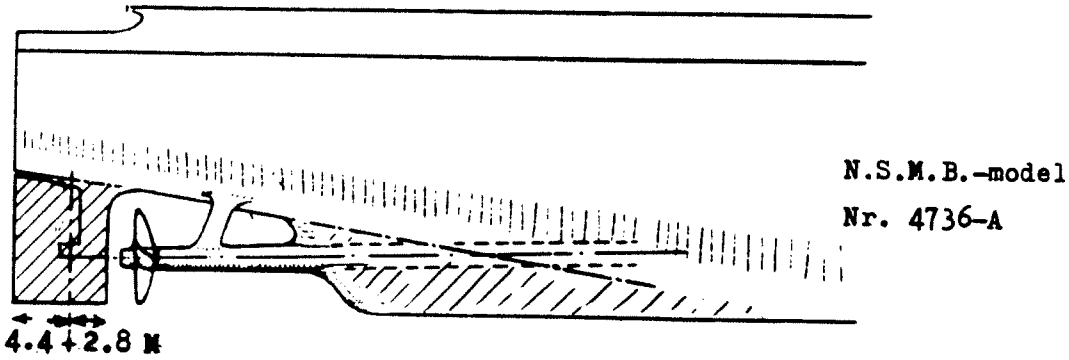
AT LARGE RUDDERANGLE : POOR TURNING



SAFCON PROJECT

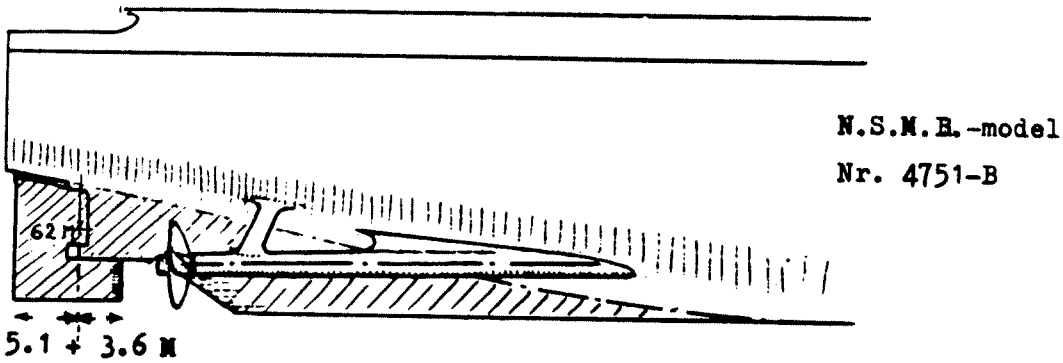
Tank testing models

A



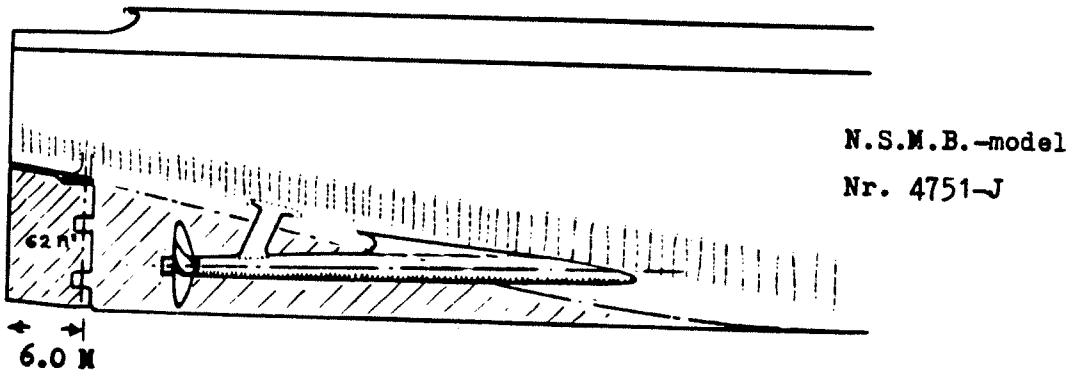
SINGLE-SCREW, SEMI-BALANCE RUDDER CONFIGURATION.

B



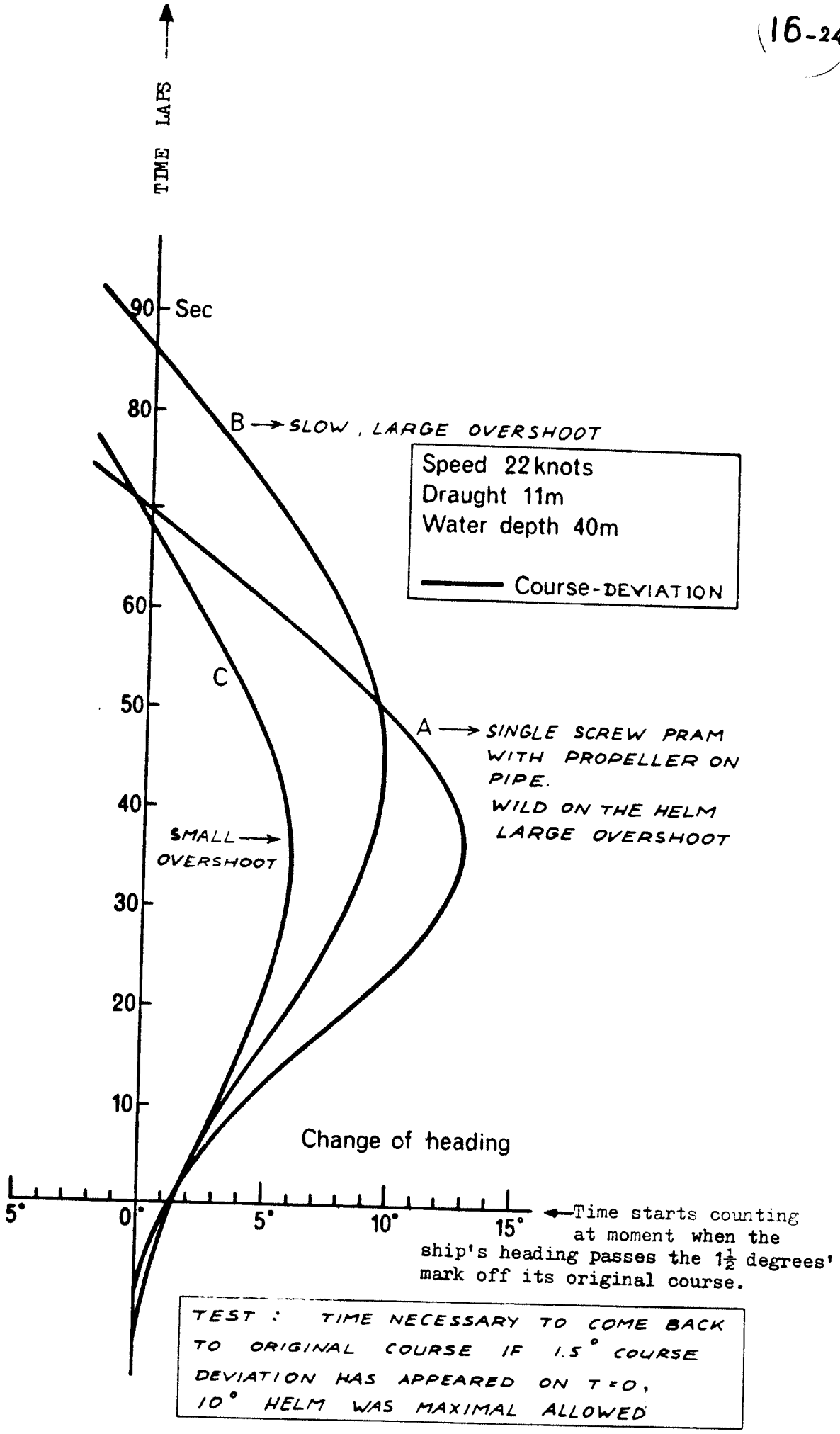
T W I N -SCREW, SEMI-BALANCE RUDDER CONFIGURATION .

C



TWIN-SCREW, S K E G -RUDDER CONFIGURATION.

(Figure 60)



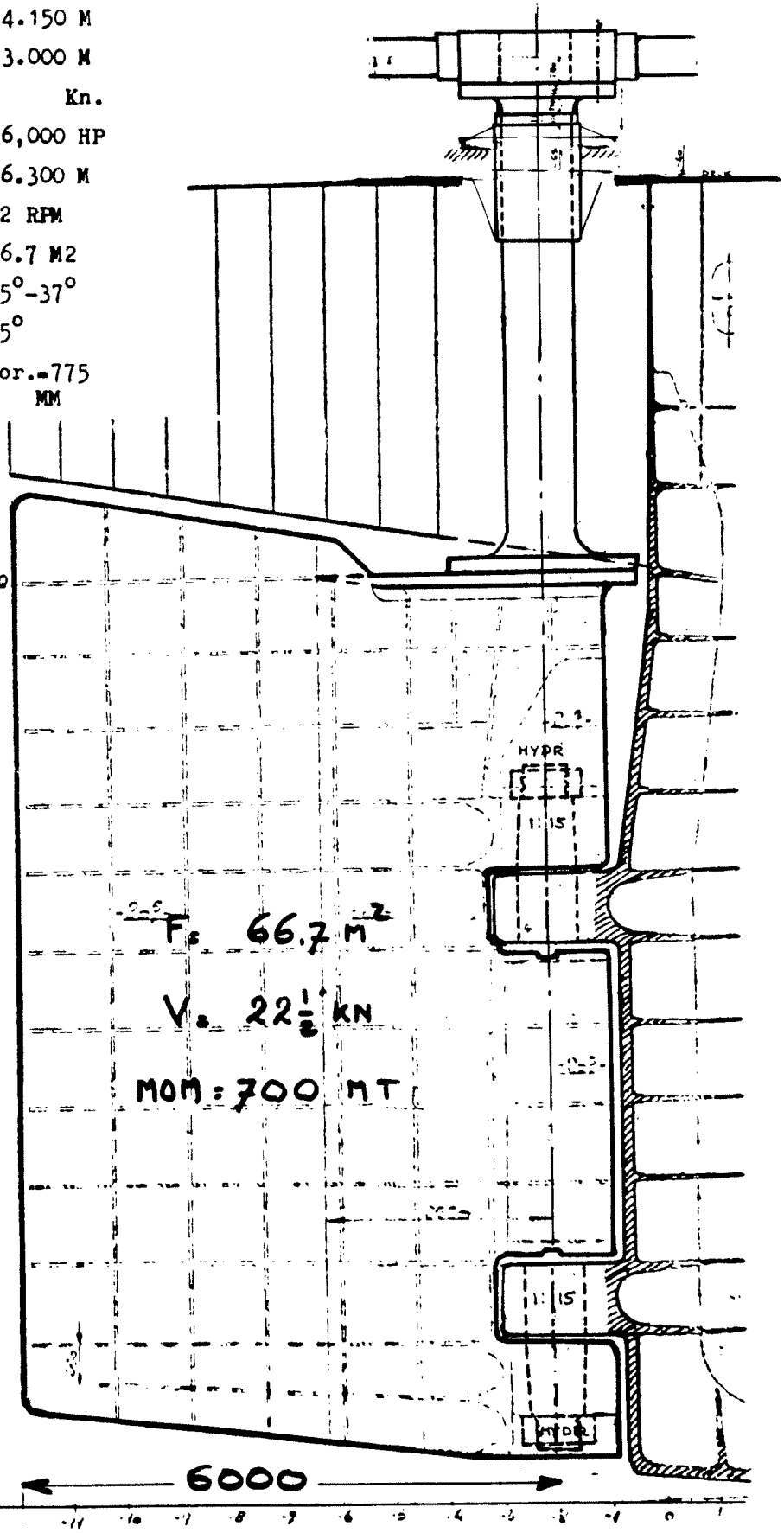
OVERSHOOT TEST.

(Fig. 65)

(KEYLESS) FLANGE-TYPE TILLER FIT
N. HOUTMAN - N. HOORN

Principal related particulars:-

Ship's Length p.p.	247.000 M
-,,- Length (sctlg)	246.574 M
-,,- Breadth, mld.	32.260 M
-,,- Depth, mld.	24.150 M
-,,- Draught	13.000 M
-,,- Speed, service	Kn.
-,,- Main Eng.Power 2 x	26,000 HP
Diameter of Propeller	6.300 M
Revolutions of Prop.	122 RPM
Total Area of Rudder	66.7 M2
Helm Angle (v=21 kn.)	35°-37°
Helm Angle (v=10.5 kn.)	45°
Diam. of Rudderstock 760+15 or.=775 MM	



BUSH IN STERNFRAME

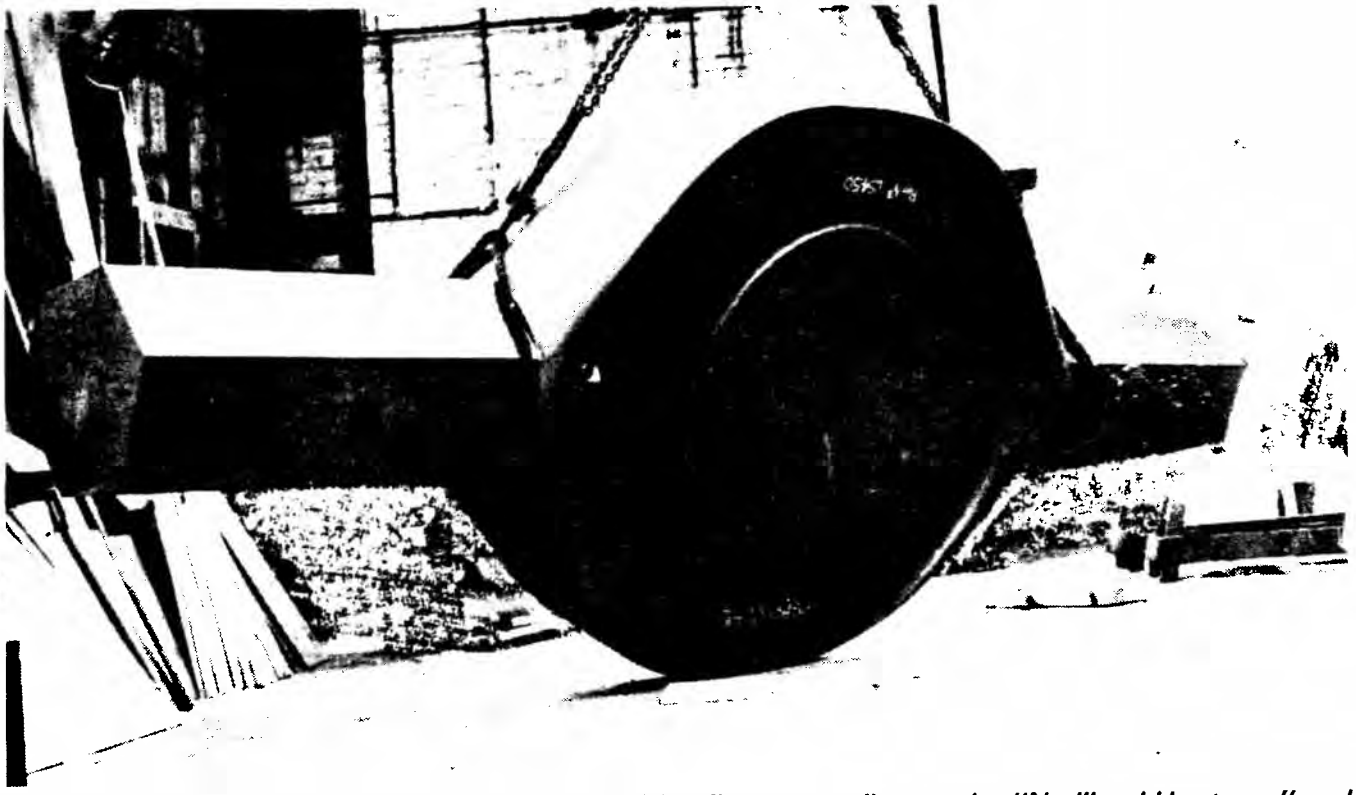
WERKSTOFF:
Nº 1.4086 Hb=260-330
GX 120 CR29
HARDENING 7-800°C
C = 0.9 - 1.3%
Si = 2.0 MAX
Mn = 1.0 MAX
P =
S =
Ni =
Cr = 27-30%
Mo =

SLEEVE ON PINTLE

WERKSTOFF:
Nº 1.4339 Hb = 210
GX 32 CrNi 2810

C = 0.25-0.40
Si = 2.0 MAX
Mn = 1.5 MAX
P 0.045
S 0.030
Ni 9-11%
Cr 27-29%
Mo

R.S.V.- CASE LINE.



The 700 tonne metre tiller connector as fitted with a flange coupling on the "Nedlloyd Houtman" and "Nedlloyd Hoorn".

SWINGING WITH THE STERN

OUTWARD- OR INWARD-TURNING PROPELLERS?

It is a well known fact that to swing the stern of twin-screw ships by means of fixed-pitch propellers (without utilising the rudder) it is necessary for the ahead-propeller to turn outward, and simultaneously for the astern-propeller to turn inward, thus creating a pressure difference across the skeg which will push the ship "away: from the astern propeller (Fig. 62-63-64).

Similarly to obtain this effect with controllable-pitch propellers, the astern-propeller should also turn inward. Consequently both propellers of a c.p. twin-engined installation should turn inward normally (ferries) (supply boats) (tugs).

Experience with supply vessels at Smit-Lloyd has shown that with c.p. propellers turning outward, no "swinging of ship's stern" is feasible at all. With c.p. propellers turning inward, reasonable "swinging of stern" is feasible, and with c.p. propellers working like fixed-pitch ones, the engines being reversible, it was found that by turning the astern propeller inward and the ahead propeller outward, excellent "swinging of the stern" was possible. Under similar conditions with the astern propeller turning outward and the ahead propeller inward, no "swinging of stern" was possible at all.

Ferry boat captains on the Harwich-Hook of Holland service have stated that they prefer fixed-pitch propellers, reversible by gearbox, rather than controllable-pitch propellers which are continuously rotating. Swinging of the stern by means of fixed-pitch propellers (with the effect of a stern "walking on her wheels") is about two times faster in comparison to the same manoeuvre with inward turning c.p. propellers as has been found by several captains.

However the fine regulation of speed by the c.p. propeller will still give her the preference.

MOVING SIDEWAYS Fig. 62-63-64

Moving sideways from the quay with a container ship (pram-type - twin-screw hull form)is possible if the ship has two f.p. propellers of the outward turning type and the rudder lying in the amidships position. A lateral movement is attained by a transverse force on the skeg aft and the transverse force of the bowthruster.

- Figures were obtained from model tank tests for moving sideways without headway parallel from the quay.
- In practice, however, this method is seldom used (too high RPM!)

A stern thruster is to be recommended to make the ship independant from tugs in combination with bowthruster.

PROPULSION EFFICIENCY TWIN-SCREW/FIXED PROP VERSUS C.P./SHAFT GENERATOR

From tests with models of large container ships of recent years with rather flat pram-type afterbodies and pipe bossings, conclusions have been drawn that outward-turning propellers require 2 – 7 per cent less power for the same speed in comparison with inward turning ones.

With other hull forms, like those of large twin-screw tankers, this percentage may differ and is mainly influenced by the bossings. Where bossings are forcing a counter-rotating flow the wake becomes more excited. Sometimes inward-turning propellers may need less horsepower because the propulsive efficiency is increased.

With regard to the propeller itself, the efficiency of c.p. screws, with their large bosses and adapted blade

sections, is generally some 2 to 4 per cent less than that of the fixed-pitch propellers. The general conclusion, therefore, is that for normal slender twin-screw hull forms two inward-turning c.p. propellers will need about $\overset{2}{7} + \overset{2}{2} = \overset{4}{9}$ per cent more shaft horse-power in comparison with two outward-turning f.p. propellers for the same speed. This should be kept in mind when having to make a choice on propulsion for ferries.

On many c.p. propeller vessels, the shaft revolutions are kept constant for ease of control and also because of the requirement for 60 Hz from any shaft generators. At reduced speed with reduced pitch and full rev/min, there is a danger of face cavitation under reduced pitch conditions. FUEL CONSUMPTION UNDER THIS REDUCED-PITCH CONDITION IS 10-15% HIGHER IN COMPARISON TO FIXED-PITCH AT LOWER R.P.M.

COMPARISON: CARGO LINER 19 KN/110 RPM AT 17 KN. REDUCED SP.

C.P. PROP 110 RPM SHAFT GEN 60~ "ON"	FIXED PITCH 98 RPM SH. GEN "OF" DIESELGEN "ON"
FUEL CONS. 100%	85% + 1½ TON D.OIL/DAY

CONCLUSION: GENERALLY IS A THYRISTER CONTROLLED SHAFT GENERATOR PREFERABLE IN CONNECTION WITH A FIXED PITCH PROPELLER

In high powered ships with restricted propeller dimensions it is necessary to reduce shaft revolutions during manoeuvring to reduce the risk of face-cavitation/erosion. (Australian Enterprise-Emblem).

C.P. when lowering service speed:

A comparison of the speed power curves of two ships, one with f.p. propellers and one with c.p. propellers, clearly shows that the c.p. propeller ship using constant revolutions over the total speed range needs more horsepower at reduced service speed due to the lower propulsive efficiency of the faster turning propeller. This should be taken into account if c.p. propelled ships have comparatively long distances at reduced speed in the schedule, and when the fuel bill is an important factor.

A shaft generator is a good thing to have: electricity on heavy fuel and less maintenance to auxiliaries, however, it is advisable to pay the costs for a thyrister installation + fixed pitch propeller in stead

of running the c.p. propeller at full revs (60Hz) which will cost much more fuel. In future, the main engine should have a built-on generator.

SPEED CONTROL AND STOPPING

Controllable-pitch propellers can offer excellent fine adjustment of maoeuvring speed.

Of ships fitted with fixed-pitch propellers, one can hardly, if at all, speak of "fine speed control" if the propeller is coupled to a highly supercharged large-bore, low-speed diesel engine. These ships are a nuisance indeed in the Suez Canal when sailing in the midst of a low-speed convoy, their lowest propeller speed being 35 rev/min and consequently lowest continuous speed of 7 knots. (With steam turbine installations, the propeller speed can go down to 2 rev/min, but here the strong argument of pilots against steam turbine-driven container vessels is their sluggish reaction to orders for changes in revolution, especially from ahead to astern and reverse.

The adaption of the electronic fuel injection on the slow turning diesel will allow for turning down to 18 rpm dead slow !

So the c.p. propeller is the right choice if vessels have to manoeuvre very often; reason why all ferries, supply boats, tugs etc. today are fitted with c.p. propellers.

However, care should be taken as to their direction of rotation. On single-screw ships the c.p. propeller should properly turn "anti clockwise when looking forward". This is with a view to the astern manoeuvre, in an attempt to keep the phenomena similar to those of a fixed-pitch propeller. For instance, the particular tendency of "the bow swinging to starboard when the

fixed propeller turns full astern" is common knowledge to all pilots, and to keep the tendency similar on a single c.p. propeller ship, its c.p. propeller should turn anti-clock wise always througout its operation.

We must annotate here, however, that if the propeller is arranged in a wide aperture the tendency of the ship's head swinging to starboard is far less pronounced and also wide apertures are necessary to reduce vibration.

BOW AND STERNTHRUSTERS

Nedlloyd's newbuildings will have bow and sternthrusters in future ships. (single screw and probably also twin screw).

- It makes the vessel independant, captain has the vessel more "in hand"
- It saves time in turnaround

- It saves tugboats: from 2 to 1, or from 1 to 0, if wind is below 6 BF.
- It saves crew fore and aft
- Damages to the hull are reduced entering locks and quays
- Port approach more safe (take over pilot more easy)
- Canal passage in convoy more safe.

It is just the combination of bow + sternthruster which will give the advantages.
Practical experience with bow + sternthruster has been gained by the Scandinavians since about 10 years.

- The large ro-ro vessels of Nedlloyd Lines were fitted with both. Uptill now, results are very satisfactory during 3 years of operation on 4 ships and captains are strongly in favour of it.
  The power might well be already available in their reefer container capacity. By shutting down reefer equipment during manoeuvring operations sufficient

power could then be supplied to a bow and stern-thruster.

- A bowthruster is nowadays generally accepted for ships which call many ports per year. (container, ro-ro chemical tanker). Extra resistance if mounted in a bulb is about 1%.
- A sternthruster mounted in the afterskeg is still considered superflues by many shipowners. Model tests showed that a sternthruster is very efficient and more effective than a bow thruster, since the bowthrust has its working limit at about 4 knots speed and a sternthruster is still able to pump more water through the tunnel, at 5 to 8 knots, because it is working in the wakebelt of the vessel.
  Extra resistance of a sternthrust opening is nil, however, the opening should be wel rounded to prevent cavitationerosion to the main propeller.
- Strong side winds acting on deck containers and superstructure-aft will blow the vessel to leeward and probably try to luff her.
  Bow and sternthruster can compensate this in combined action (taking over pilot in narrow entrance is far safer).
- For the naval architect it is very important to rely on thrusters because by these units he is able to design a ship with maximum length overall within the units of narrow ports. "The longest possible ship will give the maximum economy of scale".
- The necessary power to be 1200kW per thruster on a 210 M container. Installing too small power means money for nothing (a toy).
- The major problem of the sternthruster is the extra diesel generator to be installed, which means extra investment and maintenance costs.
- The way Nedlloyd tries to solve this, is to sail normally at sea with the shaft generator on only (thyristor + fixed p. propeller).

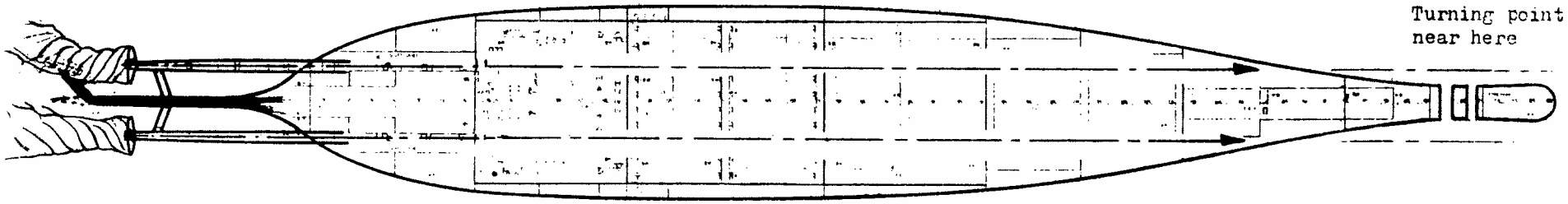
- During manoeuvring the 3 diesel generators installed are in action 2 x 1500 kW for the two thrusters:
  1 x 750 kW for general service.
  Reefer is "shut off" temporarily.

Conclusion:

In future most ships will be equipped with :
- one slow turning propeller.
- one slow turning longstroke diesel.
- "hang-on" shaftgenerator, thyrister controlled.
- electronic fuel injection which allows for dead slow turning 18 rpm.
- heating of turbocharged scavenging air at low rpm in cold climate to prevent condensation.
- turbogenerator in case of sailing long distances at full power only: bulkers - tankers.
- bow- and sternthrusters - C.P. - 6000 Volt on ships which have to call many ports per year.
- stability (metacentric height) and trim adjustment by the right way of distribution of cargo and fuel oil and waterballast.
  (this is more a question of design and cargoplanning, than of mounting stabiliser fins or antirolling tanks).

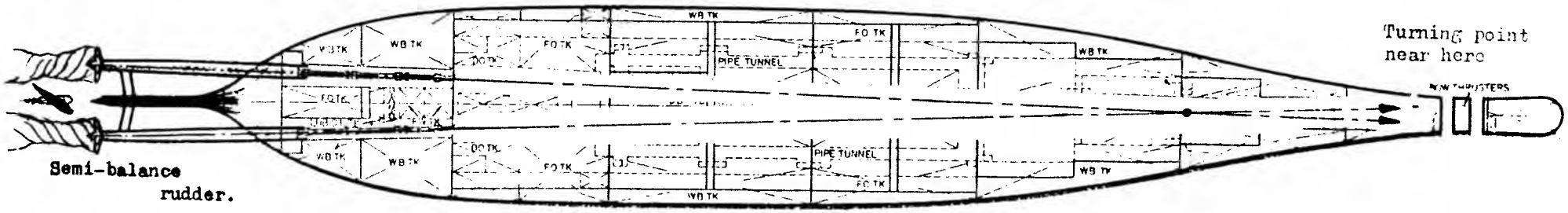
Steadying the vessel/course keeping - in shallow water.

A twin-screw vessel single SKEG rudder has better steadying abilities than a twin-screw vessel having a single rudder of SEMI-BALANCE type of equal movable surface. The long undisturbed fixed skeg with the rudder - like a fish's tail - reduces the overshoot angle. Also the lapse of time required for steadying the vessel is shorter because of the rapid pressure build-up along the skeg including rudder.



(Barndoor type) skeg-rudder.

Parallel shafts

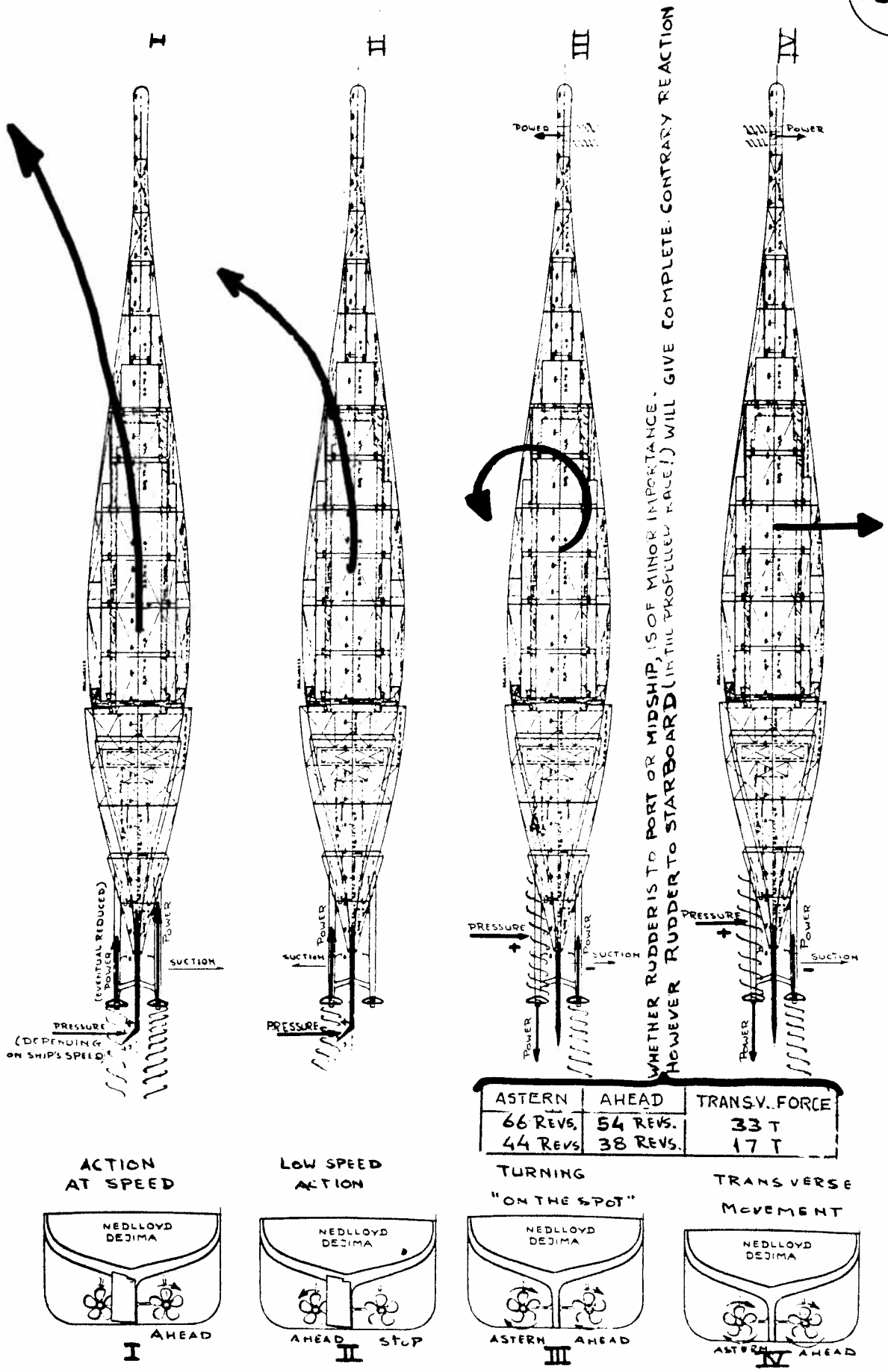


Semi-balance rudder.

Converging shafts

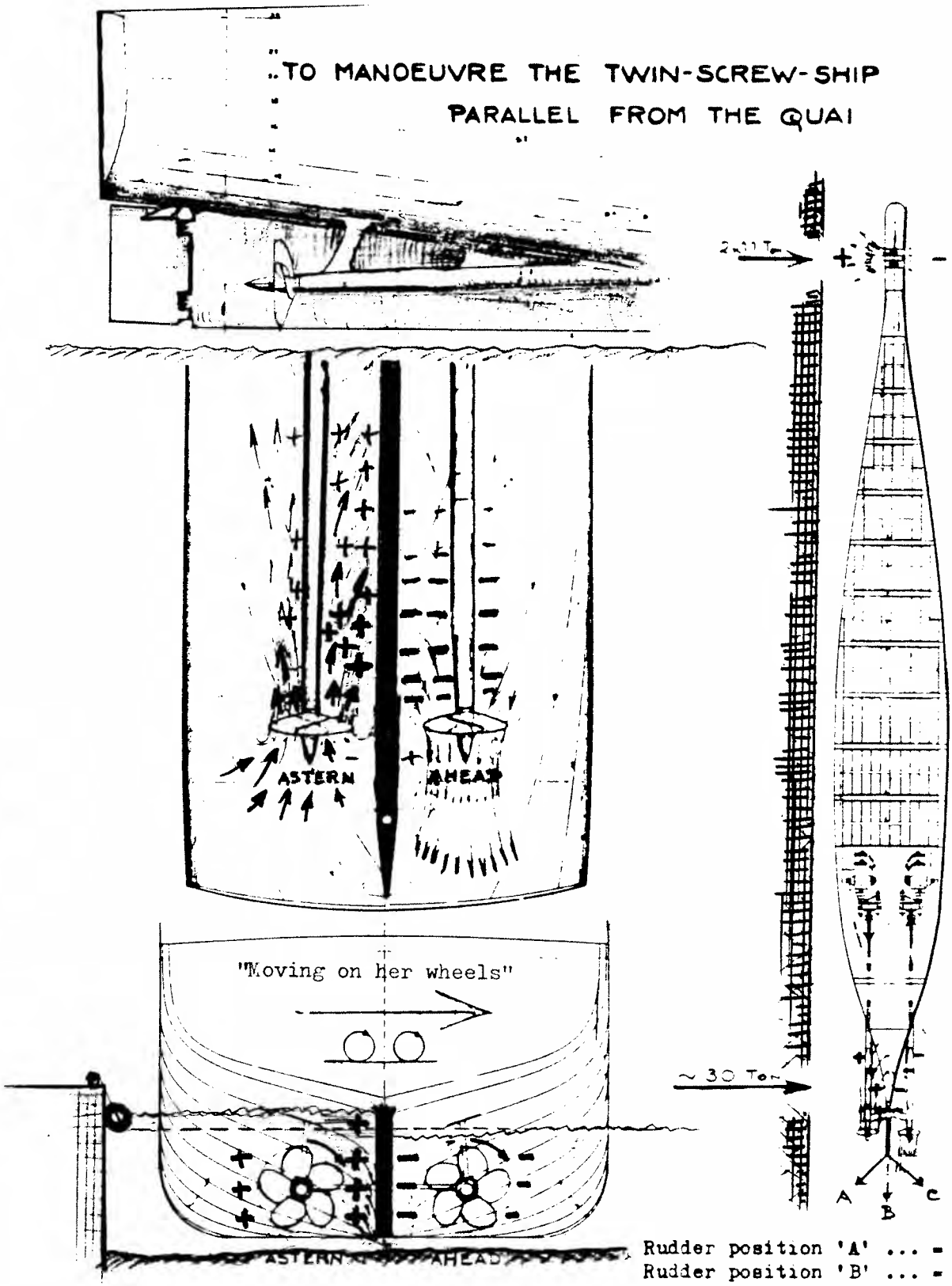
(Figure 61)

(Figure 62)



MANOEUVRING OF PANAMAX CONTAINER SHIP

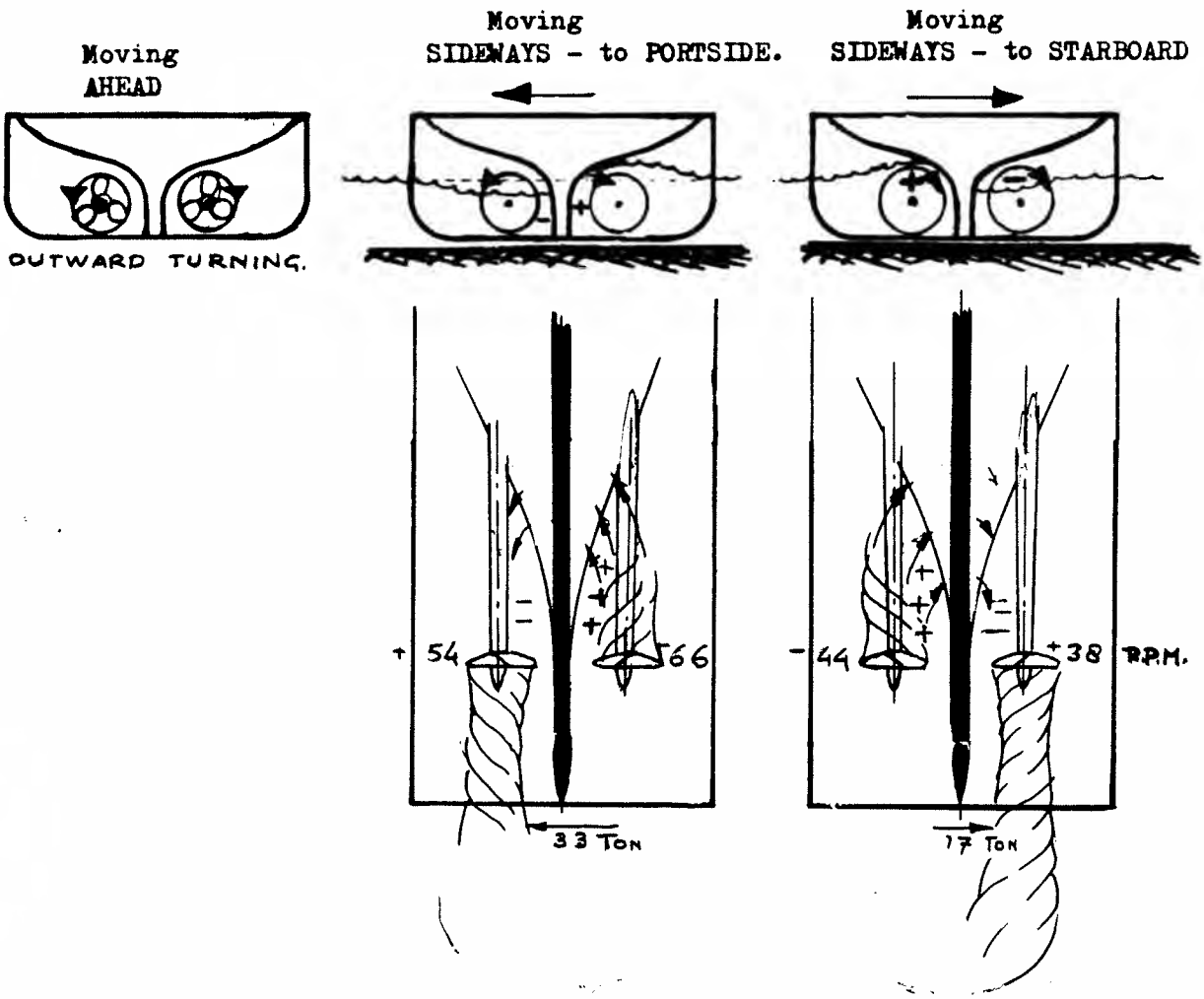
TWIN SCREW – SINGLE RUDDER ARRANGMENT



Rudder position 'A' ... = good
Rudder position 'B' ... = also good
Rudder position 'C' ... = NOT good

Moving sideways from quay or towards quay.

Because of the fact that the twin-screw ships have outward turning propellers when going ahead, it is possible to use the effect of "walking on her wheels" when lying stopped, having to move sideways as shown herebelow.



For causing the vessel to move sideways off the quay (whilst staying headed parallel to it) propeller speeds of 66 r.p.m. astern and 54 r.p.m. ahead are needed.

Note (1) - The helm should be kept amidships! This is because:-
rudder lying towards Astern-propeller has NO effect, whilst
rudder lying towards Ahead-propeller has a strong COUNTER-effect.

Note (2) - The depth of the water under the keel strongly influences the effect of "Walking on her wheels" in that :-
the more waterdepth, the lesser the sideways moving effect.

17

Schiffbautechnische Gesellschaft Hauptversammlung 19-11-1982, Berlin

Reaction on paper of J.J.Kappel, A.P.Møller, Copenhagen, bij E.Vossnack - Nedlloyd.

17-1

CAVITATION-EROSION ON RUDDERPOST AND RUDDER.

In our opinion the heavy cavitation is caused by A and B.

A. Tipvortex of propeller, ahead of sternpost, is very intense.

See Fig. 1

This is due to heavy load on propeller tip on the Maersk vessels.
- The tipvortex can be softened by reduction of pitch at tip and strong increase of skew (see fig. 2).

"MISSISSIPPI-LL" See fig. 3 - Mississippi Lloyd (PROP HAS max. pitch at tip and without skew)

"MADISON-LL" See fig. 4 - Madison Lloyd with damage to tip. (STRONGER EROSION)

B. Flow, ahead of propeller, is strongly disturbed, which might be due to:

"LAURA-MAERSK" 1. Sharp outlet of sternthruster (see fig. 5 Laura Maersk)

TANKER 2. Scoop of condensor inlet, which even might give strong cavitation erosion to propeller (Verolme AND HDW - Tankers)

"MORETON-BAY" 3. Flow lines towards propeller are crossing a knuckle being the edge of a tunnel which was designed in the wrong direction (fig. 6 and 7). See bodyplan of afterbody, Moreton Bay, flowlines crossing knuckle.
A disturbed flow enters propeller and rudderpost. Heavy cavitation erosion is the result (fig 8-9-10).
Cathodic protection by zincanodes could only suppress partly the erosion.

The remedy is by smoothing the shell, the tunnel recess has to be closed by a fairing plate.

Changing the propeller will not help.

"SEA-LAND" 4. Fairing caps ahead of shaftbearings went away on trials already after a few hrs at full speed.
The GRP material could not stand the forces by flow (see fig. 11 "Sea Land").

The hard edge front of the bearings and the rotating boltheads cause strong turbulence, and enormous erosion of propellers and rudder occured.

"HAMBURG-EXPR." 5. Bossing of Hamburg Express (twin screw) (fig. 12) had contra rotating fairing against shell in order to gain higher propulsive efficiency.

The strong tangential turbulence behind the pipe bossing caused propeller-blade-root-cavitation-erosion (fig. 13)

It turned out to be very difficult to correct the fairing. (Fig. 14) (Fig. 15).

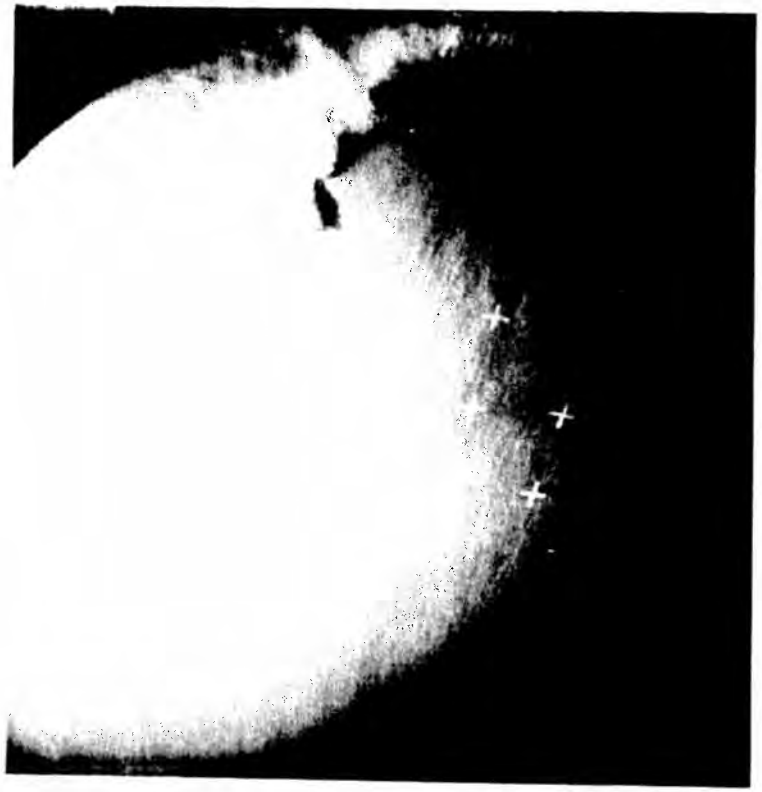
FNB

4th January 1983.

1

TIP VORTEX OF PROPELLER
HITS RUDDERPOST AND
CAUSES CAVITATION-EROSION.

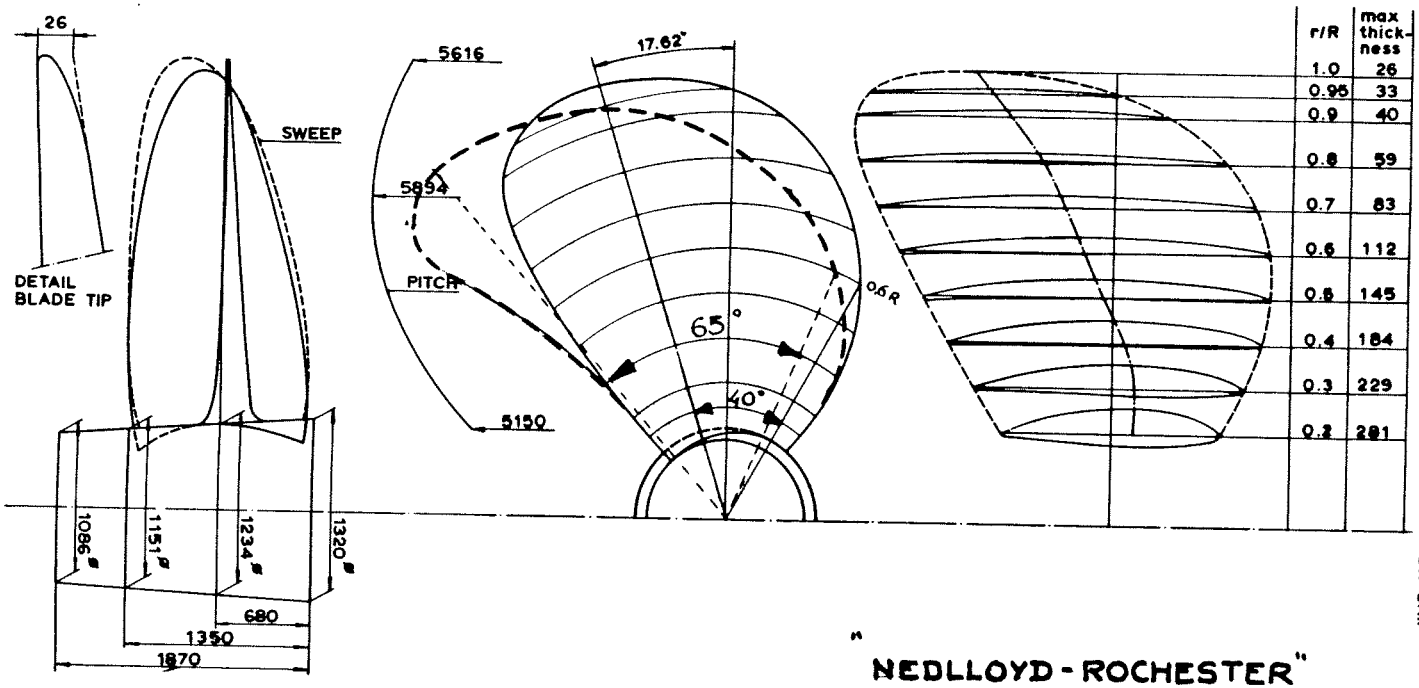
FULL SCALE "N. ROCHESTER"



2

INCREASE OF SKEW REDUCES "HIGHER-HARMONICS"-EXCITATION

AND REDUCES INTENSITY OF TIPVORTEX



r/R	max thickness
1.0	26
0.95	33
0.9	40
0.8	59
0.7	83
0.6	112
0.5	145
0.4	184
0.3	229
0.2	281

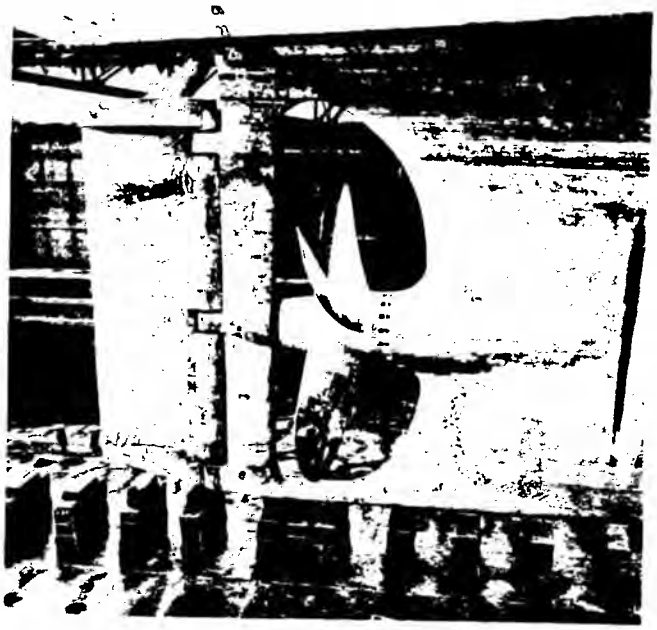
GEOMETRY OF PROPELLER

DIMENSIONS ARE GIVEN IN mm FOR SHIP

"NEDLLOYD - ROCHESTER"

PROPELLER DIAMETER	D =	6500
PITCH RATIO AT 0.7 R	$P_{0.7}/D$ =	0.907
EXPANDED BLADE AREA RATIO	A_E/A_O =	0.726
CHORD LENGTH$_{0.7}$/ DIAMETER	$c_{0.7}/D$ =	0.420
THICKNESS / CHORD LENGTH$_{0.7}$	$t/c_{0.7}$ =	0.0305

GEOMETRY OF 4 - BLADED RIGHT - HANDED PROPELLER MODEL No. 5520.



3

MISSISSIPPI LLOYD.

TIP VORTEX HITS RUDDER AND STERNPOST

4 A

4 B

MADISON LLOYD.

WITH DAMAGED PROPELLER TIP CAVITATION EROSION IS STRONGER.

MR. RAPO
LLOYDS REGISTER OF SHIPPING.
FENCHURCH STR. 31
LONDON E.C. 3.

-81.

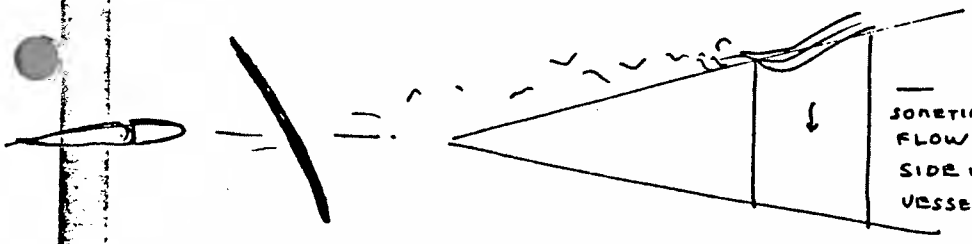
5

„Schroefpartij" van de „Laura Maersk".

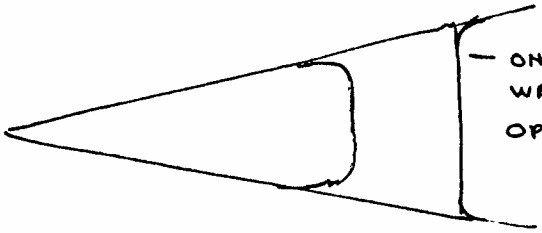
ZA MAERSK
91

:NTHRUSTER
E CAV. EROSION.
'ERDISTURBANCE
PART) OF

SHARP EDGE ↑



SOMETIMES THERE IS EVEN A TRANSVERSE FLOW IN THE CHANNEL BECAUSE OF SIDE WIND- AND WAVE ACTION ON THE VESSEL WHICH ~~MAKES~~ CREATES LATERAL MOVEMENT



ON NEDLLOYD RO RO VESSELS WE MADE VERY NICELY ROUNDED OPENINGS (STAINLESS STEEL CAST VERY EXPESIVE.
ON THESE SHIPS NO TROUBLES WITH PROP. CAV. AND RUDDER EROSION

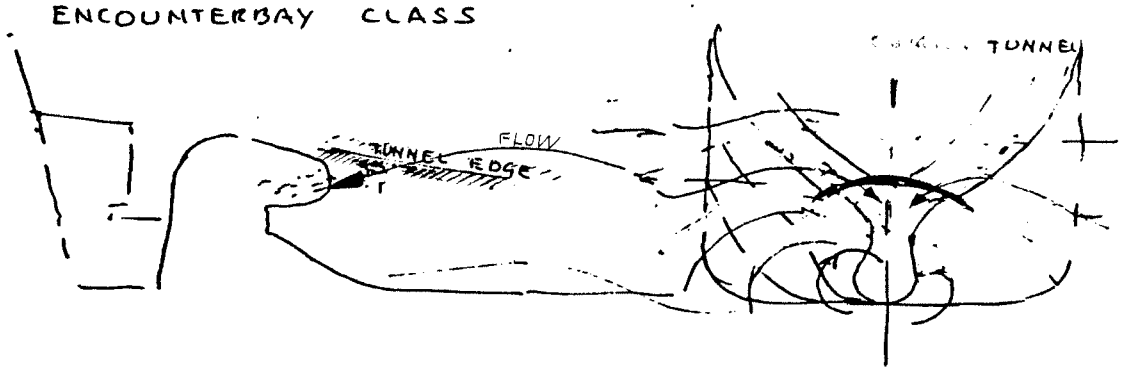
Nedlloyd Rederijdiensten B.V.
Houtlaan 21
3016 DA Rotterdam
P.O. Box 1468
3000 BL Rotterdam
General: 10-177933
Telex 24690
Telegrams Nedship
Chamber of Commerce
Rotterdam no. 121289

IT MIGHT WELL BE, THAT THE PROPELLER TROUBLES WITH THE MITSUBISHI RoRo ARE CAUSED BY THE STERNTHRUSTER OPENING (BARBER - BOOGABILLA - KOLNÄREY ~~etc.~~
after ½ YEAR A HOLE IN PROPELLER

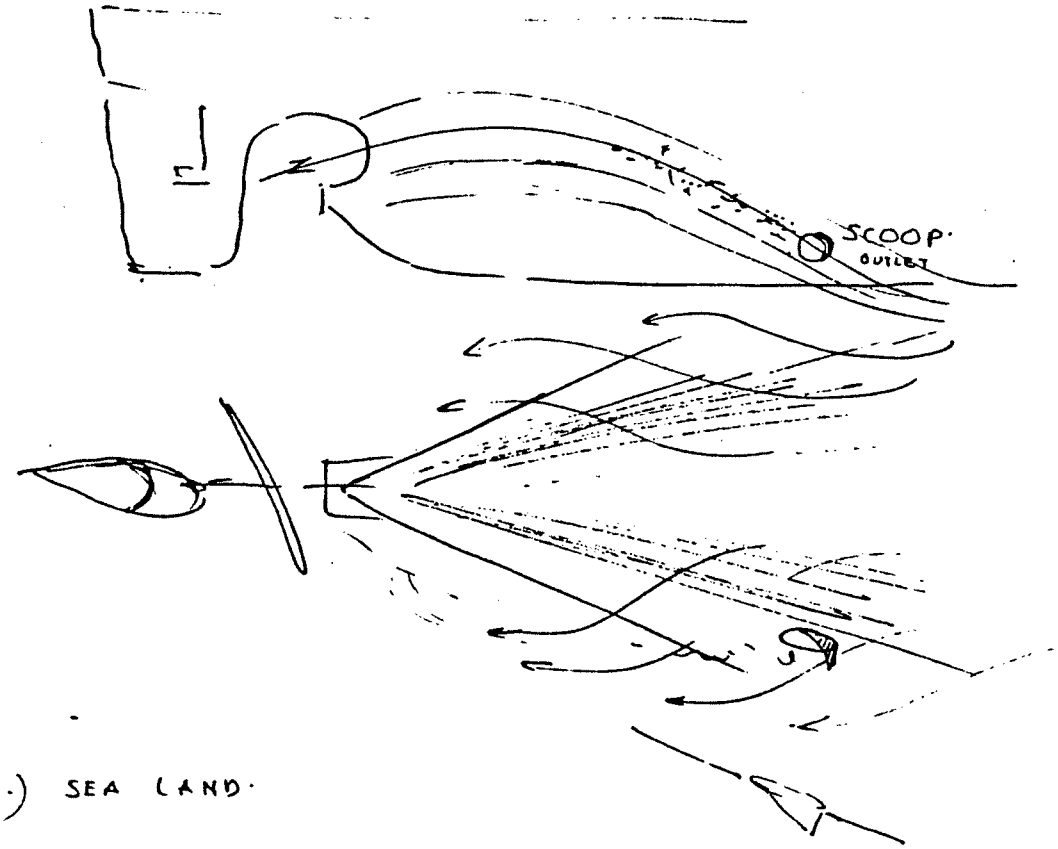
DISTURBANCE OF FLOW AHEAD OF PROPELLERS IS VERY DANGEROUS

) REMEMBER:

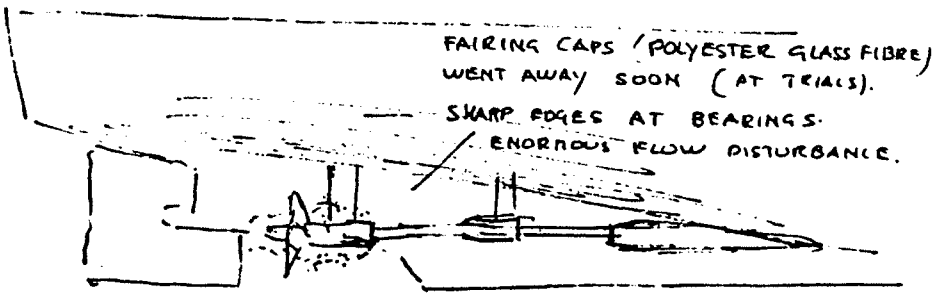
a) ENCOUNTERBAY CLASS



b) TANKER VEROLME

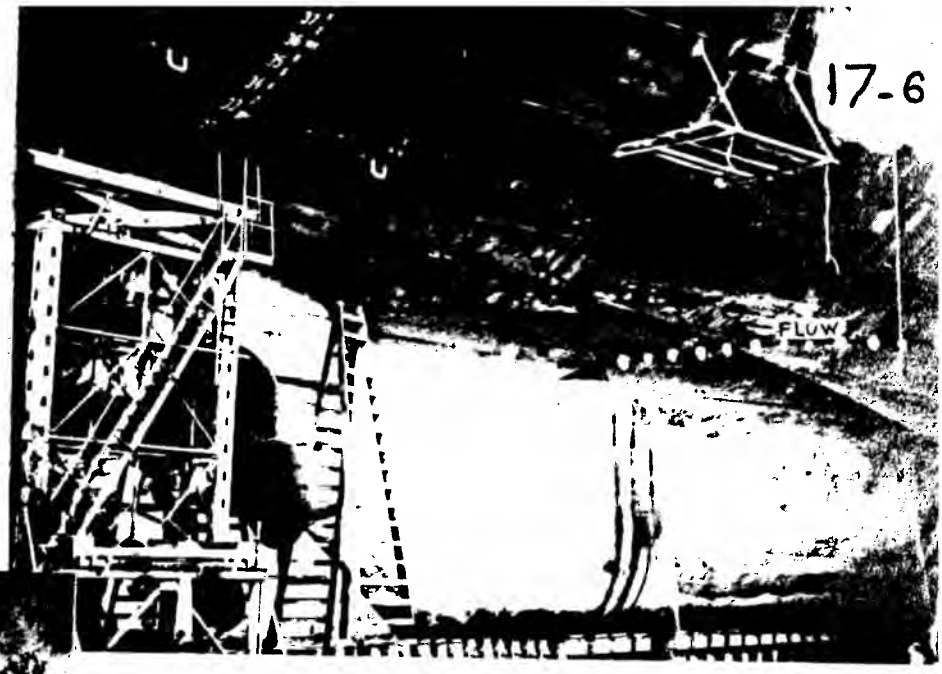


C.) SEA LAND.



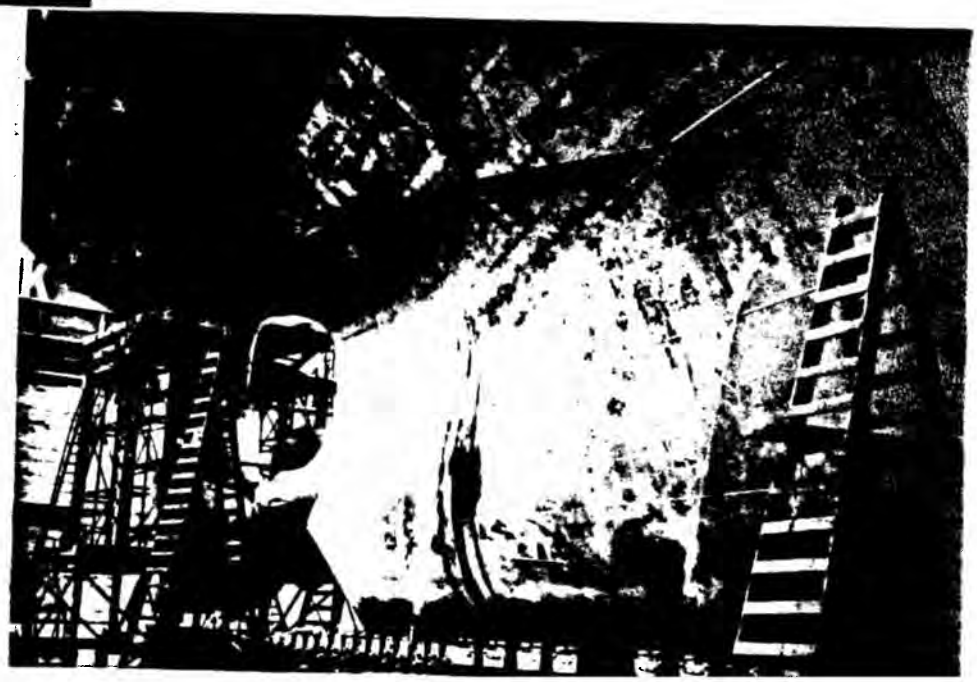
TERIRIBLE PROPELLER CAVITATION EROSION ON TOTAL SURFACE
(LATER THE COVERED THE OPEN SHAFTS BY
STEEL TUBES – AND EROSION WAS PREVENTED

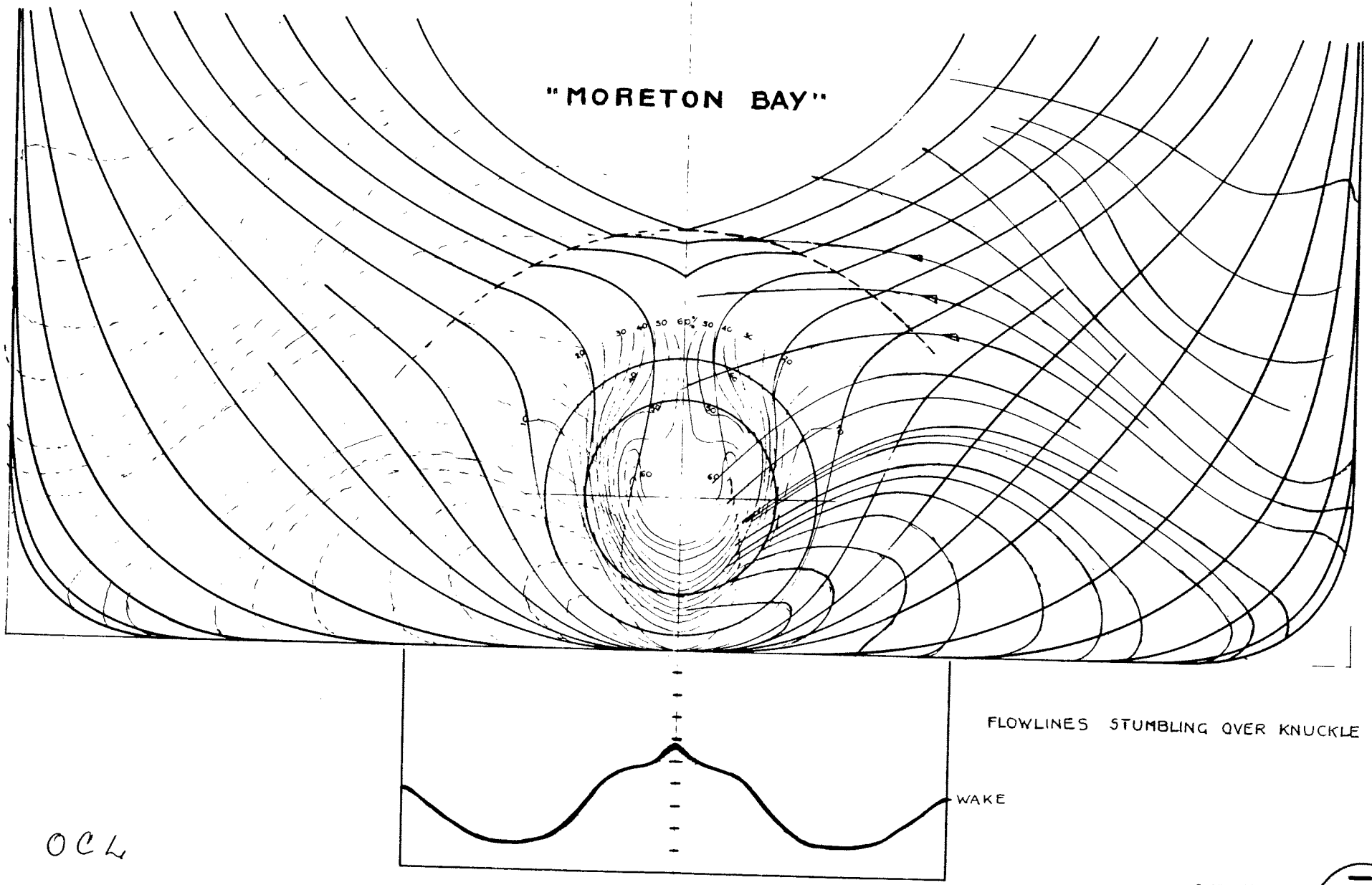
MORETON BAY



KNUCKLE NOT IN LINE WITH WATER FLOW.

STRONG TURBULENCE INTO PROPELLER AND RUDDER.





FLOWLINES STUMBLING OVER KNUCKLE

OCL

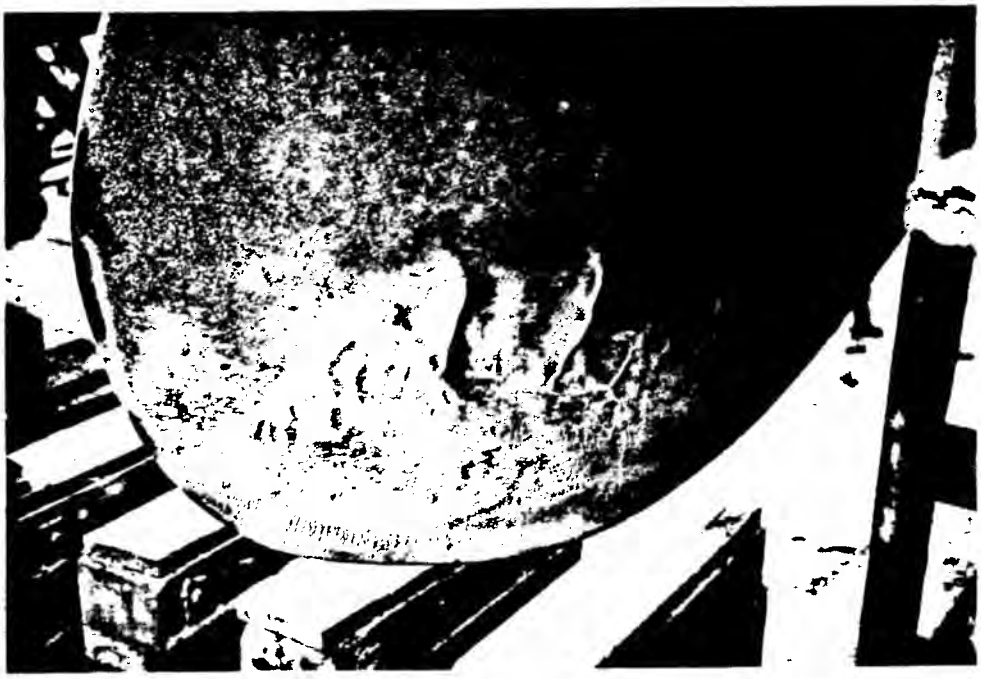
ENCOUNTER BAY CLASS

17-7

MORETON BAY.

17-8

RUDDER POST CAVITATION EROSION BY INTENSE WATERTURBULENCE



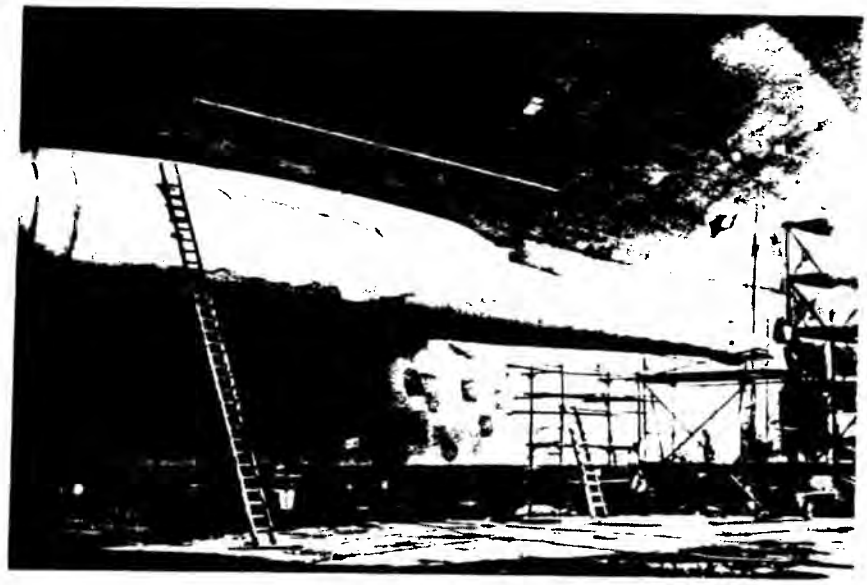
PROPELLER CAVITATION EROSION

FIG 12

CONTRA DRALL IN FAIRING:

TANGENTIAL FLOW AROUND ROOT OF FAIRING CAUSED TURBULENCE

FOLLOWED BY CAV. ER. IN BLADE ROOT



↗ PROPELLER FACE CAVITATION IN BLADE ROOT

18

Literature.

NSMB reports

Contr. model 4248 October '78 - Austr. contr."Abel Tasman", ANZECS
model 3865 March '70 - Far East contr. Delft/Dejima,
3956 January'72 - ScanDutch
model 5902 June '81
model 4751 December '76- South Afr. Europe Safcon contr. "Hoorn/Houtman" SEACS/ANZECS.
model 5163 March '77 - Container project - not built, ZAVO
model 5268-5307 March'77- Container Hongk.-Austr., AAE "Asian Jade"
model 5655 Sept.'80 - Cont. South Afr-Far East, not built
model 5736 Sept.'80 - Contr.South Afr-Far East, MHI design
model 5901 Aug.'81 - Contr. Universal typ - EMEC-Eurosal

Ro-Ro model 5100-5104 sept.'76- RoRo contr. Austr-Malaya, ANRO
model 5120-22 June '77 - Shortbody Arabian Gulf
model 5141-5131
model 5150
model 5190
model 5247 Febr. 79 - Gooseneck
model 5274 Nov. 77 - Longbody Arabian
Calculation boundary layer flow around 3 RoRo afterbodies - March 1979 - Raven.
Interim report simplified Hull forms August'78 -
model 5650 Nov. '80 - Pass/Car ferry twin skeg-twin screw, single skeg-single screw.

PUBLICATIONS (PRIVATE)

- Developments of ships afterbodies/ Propeller excited vibrations, May '73- Lips 2° Propellersymposium
- Trillingshinder bij schepen Febr.'77- KIVI - Int. Periodical Press

Developments of afterbovy contr.-RoRo March/78- BSRA Translation of KIVI

Costaspects in linershipping May'72-Oct.'73 - TH Delft Symposium

Efficiency on Linershipping Anno 1977 - Nedlloyd-Europoort Publication

Nedlloyd Lines RoRo-concept for the Middle East April '79 - Holland Shipbuilding

Ontwikkelingen in de lijnvrachtvaart 19 juni 1980 - Ingenieur

18.2

Delft-Dejima-Houtman-Hoorn Aug.'75 - Unischakel
Verre Oosten containerschepen
Aug.-Sept.'73 - Nautical/Technical tijdschrift
Containerschiff Bremen Express 1972- Hansa P.2043
Nedlloyd Houtman-Hoorn Okt. '77 - Holland Shipbuilding
1977 - Schip en Werf pag. 719
- Grenco Publication
Haven en Technical Planning Nov.'80- T.H.Afdeling Civiele Techniek
"Havens I".