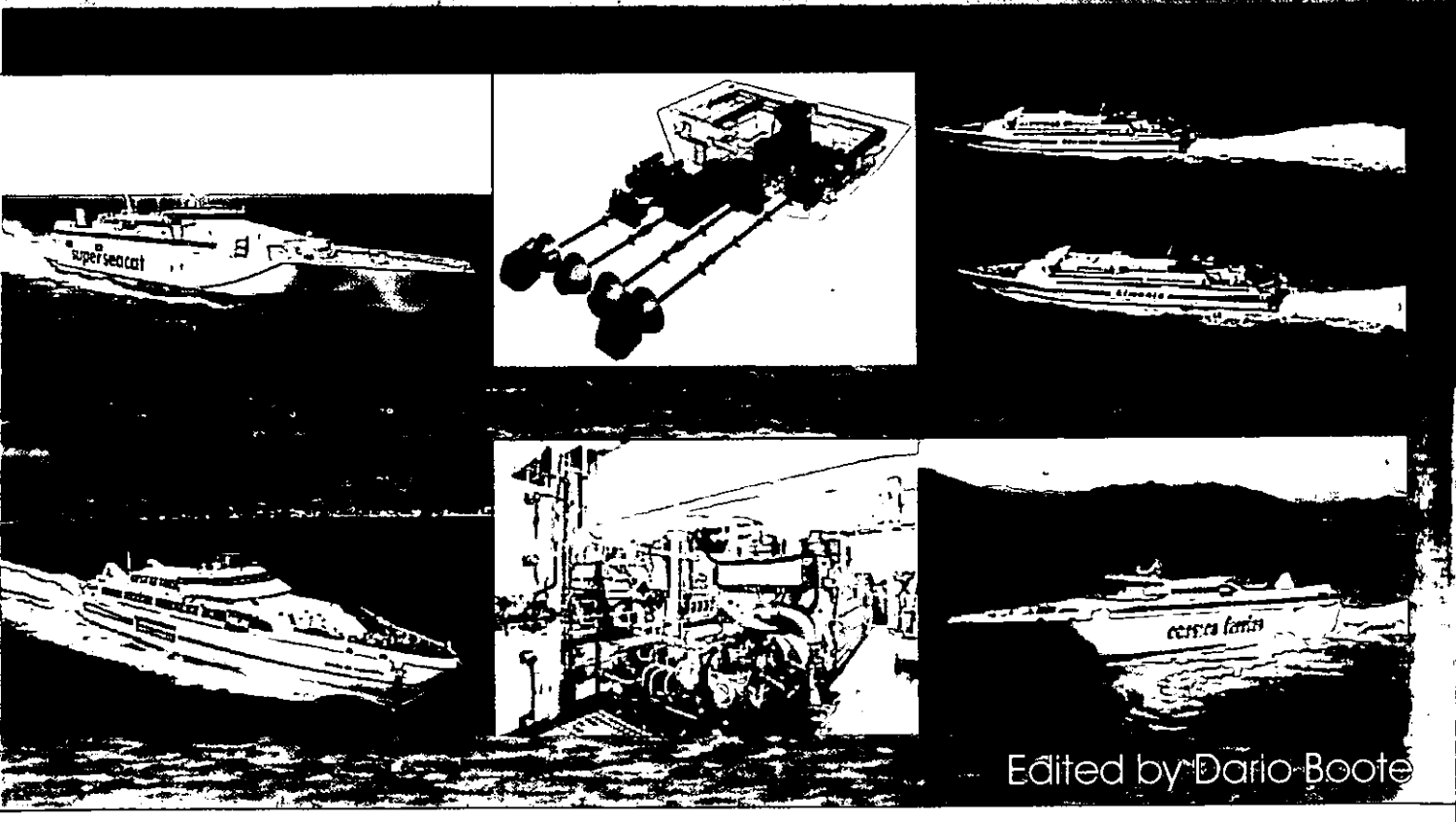
33rd WEGEMT SCHOOL

UNIVERSITA' DI GENOVA

D I N A V

WEGEMT

FAST SHIP PROPULSION PLANTS



Edited by Dario Boote

Monday 22 - Friday 26 May 2000
Venue

Villa Cambiaso-Genova

Hosted by
DINAV - Università di Genova
Dipartimento di Ingegneria Navale e Tecnologie Marine

33rd WEGEMT School

Fast Ship Propulsion Plants

Genova, May 22nd – 26th , 2000

Venue: Università degli Studi di Genova
Facoltà di Ingegneria

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ABOUT THE PUBLICATION

This publication represents a series of lecturers' papers, which were presented at a five-day event entitled *Fast Ship Propulsion Plants.* The event was hosted by the University of Genoa, Italy from 22nd to 26th May 2000. The event was partly sponsored by the European Commission through the Training and Mobility of Researchers Programme under a project submitted by WEGEMT entitled "Supporting Initiatives for Modern Marine Industries (SIMMS)".

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WEGEMT achieves this aim by encouraging universities to be associated with it, to operate as a network and therefore actively collaborate in initiatives relevant to this aim.

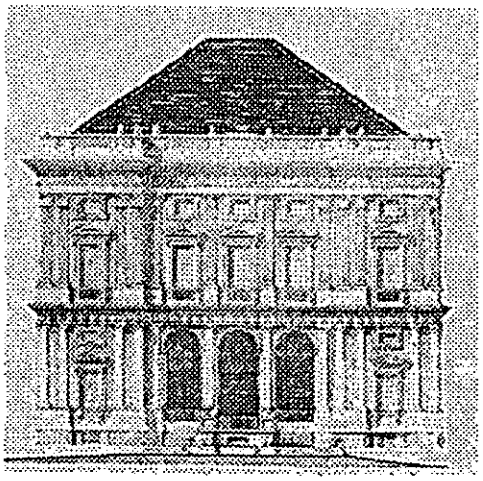
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For marine technology and related sciences, WEGEMT includes all aspects of offshore oil and gas exploration and production, shipping and shipbuilding, underwater technologies and other interdisciplinary areas concerned with the oceans and seas.

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The Regia Scuola Superiore Navale (Royal Naval High School) was founded in Genova in 1871 to form "engineers for the construction of ships and steam engines for the Merchant Fleet". In the late Twenties courses of civil, mechanic and electrical engineering were added, creating the nucleus of the Faculty of Engineering, which was formally established in 1935 within the University of Genoa.

From staff and resources of the former Istituto Policattedra di Ingegneria Navale, in 1993 the Dipartimento di Ingegneria Navale e Tecnologie Marine (D.I.NAV.) was established. It carries on the tradition of education and research in the field of naval architecture and marine engineering. DINAV is located in Albaro district and the headquarter is located into Villa Cambiaso, which hosts the Faculty of Engineering.



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PROGRAMME OF THE SCHOOL

Monday, 22nd May 2000

9:00 – 9:50	Registration and Welcome
9:50 – 11:40	Fast Ferries: Summary Of Concepts – *V. Farinetti* Fincantieri (Italy)
11:40 – 12:30	Hydrodynamic Requirements for Design I- *L. Grossi* Fincantieri (Italy)
12:30 – 14:00	Lunch
14:00 – 14:50	Hydrodynamic Requirements for Design II- *L. Grossi* Fincantieri (Italy)
14:50 – 16:40	Propellers and Water Jets - *G. Caprino* CETENA (Italy)
16:40 – 17:30	Gear Units for Fast Ships - *K.H. Merk* RENK (Germany)

Tuesday, 23rd May 2000

9:00 – 10:40	Propulsion Plant Configurations – *A. Ginnante* Fincantieri (Italy)
10:50 – 12:30	Diesel Engines for Fast Ships - *D. Stapersma* TU Delft (The Netherlands)
12:30 – 14:00	Lunch
14:00 – 14:50	MTU's New Engine Series 2000/4000 In Fast Ferry Application *G. Haussmann* MTU (Germany)
14:50 – 15:40	Experiences with Propulsion Systems in the Large Fast Ferry Business *G. Haussmann* MTU (Germany)
15:50 – 16:40	ZF Marine Transmission for Fast Ferry Applications *G.. E. Marsland, F. M. Bennati* – ZF Marine (Germany)
16:40 – 17:30	GEAR DESIGN - *J. Franz, H. Klemensi* – ZF Marine (Germany)

Wednesday, 24th May 2000

9:00 – 10:40	Gas Turbines for Fast Ships - *D. Stapersma* TU Delft (The Netherlands)
10:50 – 11:40	Use Of Gas Turbines For Marine Applications *G. Calabrese* FIAT Avio (Italy)
11:40 – 12:30	Gas Turbines for Marine Electric Propulsion Marine Gas Turbine Control System *F. Tortarolo* FIAT Avio (Italy)
12:30 – 14:00	Lunch
14:00 – 15:40	FMEA of Fast Ship Propulsion Plant *S. Capizzi* – CETENA Palermo
15:50 – 16:40	General Rina Requirements for the Automation of Propulsion Plants Aboard Fast Ships - *M. Bottino, W. Codeluppi* RINA (Italy)
16:40 – 17:30	Air Pollution Prevention In Propulsion and Auxiliary Machinery *C. Aiachini* - RINA (Italy) Auxiliary Systems of High Speed Crafts – *S. Padroni* RINA (Italy)

Thursday, 25th May 2000

9:00 – 10:40	Matching Ship, Propulsor and Prime Mover *J. Klein Woud* TU Delft (The Netherlands)
10:50 – 11:40	Mathematical Modelling and Simulation of Propulsion System *H. Engja* – NTNU Trondheim (Norway)
12:30 – 14:00	Lunch
14:00 – 14:50	Matching Applications - *J. Klein Woud* TU Delft (The Netherlands)
14:50 – 15:40	Clutching Applications – *H. Engja* NTNU Trondheim (Norway)
15:50 – 17:30	Mathematical Modelling and Simulation of a Waterjet Propulsion Plant *M. Figari* – DINAV (Italy)

Thursday, 26th May 2000

8:30 – 17:00	Technical visit to: - RODRIQUEZ Fast Ferry, c/o Corsica Terminal (Vado Ligure - Savona) - FINCANTIERI Fast Ferry, c/o Fincantieri Shipyards Muggiano (La Spezia)

Table of Contents

1) FAST FERRIES: SUMMARY OF CONCEPTS
Vincenzo Farinetti – FINCANTIERI Direzione Navi Militari, Genova (Italy)

2) HYDRODYNAMIC REQUIREMENTS FOR DESIGN
Luigi Grossi - FINCANTIERI Direzione Navi Militari, Genova (Italy)

3) PROPELLERS AND WATER JETS
Giovanni Caprino – CETENA, Genova (Italy)

4) DIESEL ENGINES FOR FAST SHIPS
Douwe Stapersma – Delt University of Technology (The Netherlands)

5) MTU's NEW ENGINE SERIES 2000/4000 IN FAST FERRY APPLICATION
Gerhard Haussmann – MTU Friedrichshafen (Germany)

6) EXPERIENCES WITH PROPULSION SYSTEMS IN THE LARGE FAST FERRY BUSINESS
Gerhard Haussmann – MTU Friedrichshafen (Germany)

7) GEAR UNITS FOR FAST SHIPS
K. H. Merk – RENK AG Augsburg (Germany)

8) ZF MARINE TRANSMISSION TECHNOLOGY
G. E. Marsland, F. M. Bennati – ZF MARINE, Friedrichshafen (Germany)

9) ZF MARINE TRANSMISSION FOR FAST FERRY APPLICATIONS
G. E. Marsland, F. M. Bennati – ZF MARINE, Friedrichshafen (Germany)

10) GEAR DESIGN
J. Franz, H. Klemensi – ZF MARINE, Friedrichshafen (Germany)

11) GAS TURBINES FOR FAST SHIPS
Douwe Stapersma – Delt University of Technology (The Netherlands)

12) USE OF GAS TURBINES FOR MARINE APPLICATIONS
Gennaro Calabrese – FIAT Avio, Torino (Italy)

13) GAS TURBINES FOR MARINE ELECTRIC PROPULSION
Franco Tortarolo – FIAT Avio, Torino (Italy)

14) MARINE GAS TURBINE CONTROL SYSTEM
Franco Tortarolo – FIAT Avio, Torino (Italy)

15) FMEA OF FAST SHIP PROPULSION PLANT
Salvatore Capizzi – CETENA, Palermo (Italy)

16) GENERAL RINA REQUIREMENTS FOR THE AUTOMATION OF PROPULSION PLANTS ABOARD FAST SHIPS
M. Bottino, W. Codeluppi – RINA, Genova (Italy)

17) AIR POLLUTION PREVENTION IN PROPULSION AND AUXILIARY MACHINERY
Carlo Aiachini – RINA, Genova (Italy)

18) AUXILIARY SYSTEMS OF HIGH SPEED CRAFTS
Sergio Padroni – RINA, Genova (Italy)

19) MATCHING SHIP, PROPULSOR AND PRIME MOVER
J. Klein Woud - Delt University of Technology (The Netherlands)

20) MATHEMATICAL MODELLING AND SIMULATION OF PROPULSION SYSTEM
Hallvard Engja – NTNU Trondheim (Norway)

21) MATHEMATICAL MODELLING AND SIMULATION OF A WATERJET PROPULSION PLANT
Massimo Figari – DINAV, Genova (Italy)

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

FAST FERRIES: SUMMARY OF CONCEPTS

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

33rd WEGEMT School
Fast Ship Propulsion Plants
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FAST FERRIES: SUMMARY OF CONCEPTS

Vincenzo Farinetti
Fincantieri C.N.I. – Naval Vessel Business Unit, Genova (Italy)

ABSTRACT

The major criteria to be adopted in the development of new designs of fast ferries basically refer to three main items: safety, freight transport capacity and speed.

Fast ferries already operating with success (HSS 1500, MDV 3000 Jupiter) have demonstrated the technical feasibility of high speed craft with payloads exceeding 1000 tons and their economical viability on selected lines. How many are the possible lines for high speed freight transport? The answer stays with the operators, not the builders.

The existing displacement hulls (both mono or multi) are approaching their upper limits in terms of speed, even if "Destriero" has shown that a monohull of several hundreds ton of displacement can reach sustained speeds far in excess of 60 knots in open ocean, with an adequate installed power. New hull typologies should be explored.

The most important threats to safety belong to three main categories: hull damage, fire and major failure of on-board systems. Most of these occurrences lead to ship evacuation and therefore escape paths and evacuation means must be always available in any "reasonable" case.
This rather simple requirement asks for the adoption of design criteria that must be defined well at the beginning of the basic design phase, which span from hull form selection, to garage and passenger area arrangement; from assessment of reliability of systems and equipment to material selection for the outfitting.

1 - INTRODUCTION

The effort to reach always higher speeds is not new: think of the clippers of the 19th Century.

At the beginning, the mechanical propulsion with reciprocating engines and paddle wheels was just able to free the ship from winds, not to increase the speed. A real quantum step was generated by the combination steam turbine - screw propeller and for a very long period nothing really new happened in the world of shipping in terms of speed.

The history of fastest Atlantic Ocean crossing is very significant:

- in 1909 "Mauritania" made the voyage at an average speed of 25 knots;
- slowly the speed increased and in the early fifties the USS "United States" broke the previous record with a speed of 35 knots, some 10 knots in more than forty years;
- then we had to wait about another forty years to see "Hoverspeed Great Britain", in 1990, winning the Hales Trophy with an average speed close to 37 knots; less than a mere two knots more;
- suddenly two years later, it was 1992, a real breakthrough: Destriero made the crossing at an average speed of more than 53 knots with a top speed of 67 knots.

What was not possible in eighty years , happened in two years time.

What kind of revolution did cause this "miracle"?

It was another new combination: gas turbine and waterjet, not to mention other innovative, even if not really new, features like deep-Vee hull and aluminium alloy structure.

Although Destriero was not a real commercial vessel, she was, anyhow, a perfect test bed, or re-test, for new technologies which are now adopted in Fincantieri Fast Ferries. "Re-test" because they are originating from decades of experience with high performing naval vessels.

Just to mention, the first gas turbine was installed by Fincantieri on a naval vessel in the early sixties and the first combination gas turbine/waterjet on an aluminium hull (a hydrofoil) backs to the seventies.

In the sixties fast passenger transportation appeared as surface piercing hydrofoils and air cushion vehicles (hovercraft) which were basically adopting ideas conceived during WW2.

These vessels, that could be named as "first generation" fast ferries, introduced speeds in excess of 30 knots (much more in case of hovercraft), but on short routes and with a deadweight of some tens of tonnes and hence capable of transporting only passengers.

In the late seventies, through its Resolution A.373(X) "Code of Safety for Dynamically Supported Craft", IMO established the legal framework allowing to develop the idea of fast transportation at sea. The possible deadweight entered the three-digit values, giving real industrial evidence to fast ferries.

Now a new legislation is available (HSC Code) and, provided that very strict safety criteria are observed, there is no limit to the number of passengers and vehicles that can be transported.

The existing code defines very clearly what can be considered as a "High Speed Craft": the vessel must have a continuous speed, at maximum operative displacement, not lower than

$$V > 3.7\ \nabla^{1/6}\ [m/s]$$

The present scenario of fast ferries is in very rapid evolution and the process is not over:

- typologies substantially different among them still co-exist (round bilge monohulls, deep-Vee monohulls, wave piercing catamarans, conventional catamarans, swath or semi-swath vessels, surface effect ships, etc.);
- the propulsion is mainly based on waterjets, thanks to their higher efficiency for speeds above 30 knots in respect to screw propellers;
- engines are either high speed Diesels, for unit power up to 7÷8 MW, or gas turbines, for unit power up to about 25 MW;
- hull material is aluminium alloy or high tensile steel or, in case of small vehicles, composite material (FRP).

For its high-speed ferries Fincantieri adopted a hull form, the monohull deep-Vee type, for several reasons, connected both to performances and safety.

By now, ten vessels have been built, belonging to three different classes: four of them are completely made of aluminium alloy, the others have steel hull (apart from an aluminium intermediate car deck) and aluminum superstructures.

The annexed tables show the main characteristics of these three families:

- Table 1: MDV 1200 Pegasus (H.T.S. hull)
- Table 2: MDV 1200 SuperSeaCat (Al alloy hull)
- Table 3: MDV 3000 Jupiter (H.T.S. hull)

2 - SAFETY

Safety must be the major driving factor when designing a vessel carrying thousands of passengers, and it cannot be compromised. However realism should not be neglected in order to avoid constraints that could impair the future of the industry.

The "Formal Safety Assessment" (FSA) approach seems to offer good safety margins coupled to reasonable measures to cope with hazards, by offering proper tools to evaluate risks.

By quoting IMO, "Formal Safety Assessment" means "a structured and systematic methodology, aimed at enhancing maritime safety including protection of life, health, the marine environment and property by using risk and cost/ benefit assessments".

Risk evaluation basically means to compare the likelihood of an event with its consequences, as shown by the so-called risk matrix in Fig. 1.

The risk connected to events of major consequences but of marginal probability is by far more acceptable than the one deriving from occurrences of lesser consequences but much higher probability.

The process associated to FSA is structured on five major steps: identification of hazard, risk assessment, risk control options, cost/ benefit assessment and decision making.

On the other hand, absolute and mere deterministic considerations and subsequent rules may lead to design implications and constraints not compatible with a product fit for the industry in terms of cost, performances and operability.

It is now on the table the revision of the existing High Speed Craft Code of Safety (HSC Code) and several proposals are in discussion.

Some initial deterministic formulas related to the extension of raking damage implied longitudinal extensions well exceeding the ship length, while other further proposals simply reduced it to 100% the ship length.

It may be interesting to analyse the causes and consequences of a raking damage. The only cause can be a collision with a solid or floating (or semi-floating) object.

Solid and fixed objects are named rocks and are supposed to be marked on charts and hence reasonably avoidable if the craft can correctly steer, its course keeping is acceptable and the systems on board are working properly.

Semi-floating objects are in principle not easily traceable and could represent a real danger, but if they are hardly visible very likely they are small and hence with a mass far lower than the one of the ship itself. Therefore an object of this type can severely damage the direct impact area and a reduced extension aft of it, but not the whole length of the vessel, also because it would be most probably displaced by the ship.

Collision with icebergs could be properly addressed and analysed according to the area of operation of the craft.

Rather than forcing absolute rules valid for all vessels and hence directly impacting on their characteristics and performances, it may appear more realistic to:

- in case of a ship operating in shallow water with a real risk of grounding, impose corridors, speed restrictions, Voyage Data Recorders (VDR) easily inspectable by Authorities, suitable redundancy in navigation equipment, good visibility from the bridge, more frequent evacuation drills. Said VDR's should record at least: speed, position, depth of water.
- in the other cases, require damage stability assured with an agreed portion the ship length damaged starting from the bow, and require small final heel angle after damage to allow the evacuation of passengers by means of the onboard escape systems (Fig. 2).

An aspect related to safety not often discussed is the influence of hull form and of general arrangement.

Vessels with a small waterplane area offer good comfort to passengers due to low heave forces, but this characteristics may turn dangerous in very rough sea when the ship does not ride the wave and hence can be hit by the wave and get damaged (Fig. 3).

The already mentioned visibility from the bridge should be such to allow to see the waters close to the ship not only to reduce grounding possibility, but, above all, to reduce the risk of collision with smaller boats.

Environmental impact can be seen, in a broader sense, as a component of safety.

The major components of environmental impact are three: air pollution, radiated noise, wave generation.

As regards air pollution the main agents are nitrogen oxides, carbon dioxide (CO_2), sulphur and particulate. The best way to reduce sulphur and particulate is the quality of fuel.

Its cost is higher, but it pays. Gas turbines and light fuels are a good solution.

Green house affect gases, like carbon dioxide, are closely deriving from the carbon based nature of today fuels.

While waiting for hydrogen based fuels or other zero-impact propulsion systems, the only way to reduce the quantity of CO_2 is to reduce the fuel consumption by increasing the efficiency of propulsion trains and hulls.

Gas turbine exhaust gases have a NO_X contents easily within the limits required by IMO (Fig. 4) in the near future, and even lower levels can be reached by means of the so-called DLN (Dry Low NO_X) combustion systems, which have been already tested in stationary installations, but are yet to be seen on board vessels.

Diesel engines, due to the higher combustion temperature reached in the cylinder, generate larger quantities of nitrogen oxides, but they can be fitted with catalytic reactors able to reduce the NO_X contents of the exhaust gases down to about 1 gr/ kWh.

A complete system is basically composed by an urea tank and a catalytic reactor, where the ammonia of urea and NO_X react to generate nitrogen (N_2) and steam (H_2O).

SuperSeaCat Three and SuperSeaCat Four, recently delivered by Fincantieri to Messrs. Sea Containers, are among the first vessels in the world fitted for the future installation of catalytic reactors, without the need of further platform modifications.

Also in case of radiated noise gas turbines are better positioned than Diesel engines because they generate high frequency noise rather than low frequency noise and high frequency noise is more easily reduced by silencers and by propagation.

Excessive wave generation may represent a danger, may cause damage and is a waste of energy. Its reduction has hence a threefold benefit.

Proper hull forms must be studied and tank and full scale tests have to be performed. It is easier to reduce wash in case of hulls designed for calm water operations, like rivers or lakes, of small vessels, while it is much more complex in case of real ships operating in open waters.

Full scale tests have been performed by SuperSeaCat One to determine the best angle between ship course and coast line to reduce wave effect on shore when approaching land.

3 - FREIGHT TRANSPORT CAPACITY

Ships already in service like the Stena's HSS 1500 and the Tirrenia's MDV 3000 Jupiter have demonstrated the technical feasibility and economical viability of fast ferries having a deadweight far in excess of 1000 tonnes. Both ship types can load freight vehicles as a complement to private cars which are, for sure, the best high value cargo for a fast ferry.

In this way freight vehicles can compensate the reduction of private traffic in low tourist season, offering a good income to operators throughout the year. To be meaningful, this capacity should be sized for tens of heavy vehicles rather than few units, hence the ships have to be big and therefore operate on lines of high volume of traffic.

In the opposite direction it can be interesting a smaller ship, with limited freight transport capacity, but suitable to serve inter-island routes in archipelagos, or from islands to the main land, carrying, for instance, in one direction fuel and industrial products and in the other one fresh produce and dairy or fishing products. In this way it will be assured a service having, in addition to an economical value, also a high social value.

Freight capable fast ferries must be conceived bearing in mind the characteristics of conventional road vehicles and not "special" or purpose built means. These common trucks present a harbour axle load of 10-12 tonnes. This value must be then multiplied by the vertical acceleration due to rough sea and it can easily overtake 20 tonnes. Lower values are unrealistic or refer to small freight vans not real lorries, or imply heavy operational restrictions for sea conditions.

Moreover, ships serving several ports must be fitted with onboard facilities, like ramps, to load/ unload their cargo, otherwise the operator should provide suitable facilities in every port or pay higher harbour-fees for their use.

A self-supporting vessel can be readily moved from one line to another one.

All these aspects (strong garage deck and strong ramps) have important weight implications and hence a "multipurpose" ship is heavier than a "car only" one. It means that a ship like this, with the same power, is slower than the "car only" version, but in the long run may be more profitable, thanks to its service flexibility.

The economical feasibility of containerised goods transport on short to medium haul lines is to be demonstrated yet, even if the qualified shipbuilding industry has already shown to be ready to propose technical solutions.

In case of non-self-moving freight units, like containers, it is very important for the whole logistic chain to be very fast in an integrated door-to-door intermodal system. It would be almost useless to have a high speed waterborne transport, by definition rather expensive, coupled to an inefficient loading/ unloading and distribution system. In this case the problem cannot be solved by the ship designer/ builder alone.

The ship is only a link in a more complex logistic chain that has to be planned and designed as an integrated system.

4 - SERVICE FLEXIBILITY

The major driving factor when designing the MDV 3000 JUPITER was to offer the operator the highest flexibility in terms of speed, payload mix and independence from major shore facilities, with the ultimate goal of maximising the earning capacity.

A ship design well suited for industry has to be conceived around the basic idea of profit. It simply means that the ship should offer the maximum income capacity with the minimum total cost.

The earning capacity of a vessel is exploited to its best when the average loading factor throughout the service period is high, and the service period itself is stretching on an annual basis. The vessel has, therefore, to match the traffic demand, in terms of transport capacity and transport mix expected by the market, and, possibly, to generate a new market herself.

Generation, or potential generation, of a new market is one of the fundamental ideas of fast transport at sea, by offering quicker and more frequent sailings.

The major components of the cost are: capital cost, i.e. the price of the vessel and its financing, the operating costs (crew, fuel, port fees, insurance) and those costs associated to possible investments for shore facilities.

Flexibility means easy adaptation to needs. The needs may be very different and the major ones are related to:

- traffic mix variations (payload flexibility)
- alternate line of service (harbour flexibility)
- operating speed variations (speed flexibility)

4.1 - Payload Flexibility

The garage volume (See Fig. 5) is arranged on three decks:

- Lower Garage Deck. It is for cars only. It has an axle load capacity of 1 ton and a clear height of 2 m. Total car capacity of this deck is 75 cars.
- Main Garage Deck. Its axle load capacity varies from 15 to 12 ton. The clear height is related to the position of the upper garage deck and could be from a minimum of 2.3 m up to 4.4 m. In case of car load its capacity is 195 cars, while in "freight" mode its capacity is 30 trailers and 15 cars.
- Upper Garage Deck. It is almost for its interity hoistable; apart from a small portion in the fore part. It is composed by thirty hydraulic operated modular sections plus four ramps, which are tilting and hoistable. Its axle load capacity is 1 ton and the clear height, when deployed is 2.2 m. Its car capacity is 190 cars.

All the above axle loads are harbour axle loads, i.e. the deck structures are calculated considering the load increasing factor due to acceleration at sea.

The vehicle capacity is therefore ranging between 460 cars, with no freight vehicles, and 30 trailers (of average length of 13.6 m and average harbour weight of 30 ton) plus some 100 cars.

The high modularity of the upper garage deck permits any loading combination between the two above limits, for instance 14 trailers and about 300 cars.

This flexible transport capacity allows to maximise the passenger and car service during the peak tourist season, while permitting seasonable revenue in low season and at night, deriving from the freight service and the high standard of comfort offered to night passengers that can use highly reclining seats with head and leg rest.

4.2 - Route/ Harbour Flexibility

A ship needing major shore facilities to operate may not result profitable to her possible best both for direct economical reasons (operating cost increase) and for the not possible exploitation of better routes.

In case of private harbour facilities it simply means that the operator should provide the requested facilities and hence he should bear the burden of their cost, which has an impact on capital costs. In case of public facilities, not to mention the real availability of said required facilities, the harbour fees would certainly be higher, with a direct influence on the running costs of the vessel.

Moreover, should the ship require tailor made shore installation, it would mean that in case of unavailability of said facilities, either for meteorological or technical reasons, the service could not be performed because there would be no alternate port of call.

On the other hand a ship requiring major shore facilities like ramps or gangways should be lighter and less expensive than a ship having these facilities on board.

The MDV 3000 JUPITER has been conceived under the assumption of availability of basic facilities only, like a proper interface for loading/ unloading of operating or waste fluids and catering goods.

4.2.1 - Vehicles Loading/ Unloading

The ship is fitted with two stern ramp/ doors having each a total length (including flaps) of abt 10 m and a width of abt 8 meters (see Fig. 6). The ramps are calculated for a harbour axle load of 15 tonnes, like the aft part of the main garage deck. The slope of the Garage Deck and the length of ramp allow the ship to operate with a wharf average height above tide level between 1.5 and 3 meters (see Fig. 7).

The width of the ramps is sufficient to embark two parallel rows of cars.

The internal arrangement of the garage decks allows to load/ unload all three decks at the same time.

4.2.2. - Passenger Embarkation

Foot passengers board the vessel by means of a dedicated door/ ramp leading to a fire proof trunk fitted with a reversible escalator paralleled by a conventional stairway. A second escalator connects the main passenger deck with the upper passenger deck.

Drivers can use two 12-persons elevators or two stairs. Stairs and elevators are enclosed in SFP60 trunks (see Fig. 8).

Should a proper gangway be available, foot passenger could board directly at main passenger deck level through side doors leading to the reception lobby.

4.3 - Speed Flexibility

The propulsion system is based on four shafts, acting four waterjets. The wing ones, which have steering and reversing capacity, are each powered by a pair of Diesel engines through a double input/ single output reduction gear. The central waterjets, having booster function, are driven by a gas turbine each and the high speed shaft of the two GT reduction gears can be cross connected, in order to allow to power the two boosters by one GT only. The schematic arrangement of the propulsion system is shown at Fig. 9.

4.3.1 - Diesel Shafts

The selected waterjets (Kamewa 180 SII), which are the largest steering/ reversing units ever installed, are bigger than what would be theoretically necessary for forty knots continuous operations, in fact a size 160 would be more than sufficient.

This dimension was selected to allow navigation at lower speeds without running the risk of cavitation.

Moreover, the Diesel engines that have been selected (MTU 20V 1163 TB73L) are featuring sequential turbo charging, which means that, in case of failure of one of the two engines powering the same shaft, the remaining engine would not be overloaded.

Should the engines not have the torque margin provided by sequential turbo charging, than the reduction gears should be fitted with a kind of shift, which would make them much more complicated and "weak".

4.3.2 - GT Shafts

The high speed (or input) shafts of the two reduction gears can be cross connected in way to power the two booster waterjet by one GT only.

This fact allows to exploit the power of one GT only also at lower speeds without cavitation, thanks to the splitting of the power on two impellers.

To obtain the same result with completely independent shafts, much bigger waterjets would be necessary.

A similar possibility may be got by adopting an unique reduction gear with double input/ double output, but it would be excessively heavy and its possible failure would prevent the use of both gas turbines. In the adopted configuration the two GT trains are completely independent and the failure of one of the two GT reduction gears would not hinder the use of the remaining GT shaft, although it would operate with a certain power penalty to avoid cavitation.

Therefore, with the adopted propulsion configuration, it is possible to operate the vessel, in continuous mode, at three different speed, all of them close to the MCR of each engine and hence with the lowest specific fuel consumption and higher efficiency.

Said speeds for continuous operations are:

- Conventional speed (23 knots): four Diesels
- Intermediate speed (31 knots): four Diesels and one G.T.
- High speed (40 knots): four Diesels and two G.T.s

This speed flexibility allows the Operator to select the speed better suited for the market needs and expectations that may vary not only according to the season but also within the same week and even within the same day.

5 - SPEED

So far, all fast ferries having a key role in the industry are practically of the pure displacement type, regardless if they are mono or multi hulls.

In terms of speed these vessels are approaching their upper limits and new avenues should be explored.

The key point is the reduction of drag that for displacement hulls means further enhancing the hull forms also by increasing the L/B ratio or reducing the displacement by adding, for instance, another lifting force to help Archimedes in his job. This one could be a new path to be explored, or better a non-new-path to be re-explored to obtain something fit for the industry.

Surface Effect Ships (SES) have been a dream in the seventies, a not completely happy reality in the eighties and something close to nothing in the nineties.

Small dimensions, not optimised ride control systems and maintenance problems seriously damaged the image of SES. But the idea is still valid, only its "industrialisation" has to be re-thought.

Fincantieri has continued investing resources in this field, with the fundamental support of the European Community through its Brite-Euram research programme, and with the co-operation of several enthusiastic European partners.

The characteristic of a so-called "Target Vessel" were identified and four ad hoc research teams were put together by merging the expertise of shipbuilders and designers, research organisations, model tank test institutes, classification societies, system suppliers and operators. These four research themes were related to: hydrodynamics of a large SES, structural options, experimental test procedures and optimisation of the main systems, including the ride control system.

The Target Vessel (see Fig. 10) has the following main characteristics:

• Length o.a.	abt	160	m
• Beam moulded	abt	35	m
• Dead-weight		2300	t
• Service speed		55	knots

These results can be reached with reasonable installed power and affordable running costs.

Big emphasis was put on the comfort level onboard with particular attention to noise and accelerations reduction.

Tank tests performed with a model fitted with a ride control system gave really encouraging results.

We may affirm to be ready to pass to the real thing, because the required know how and technology are available.

6 - HULL MATERIALS FOR HSC

One of fundamental stages in the design of high-speed vessels is the selection of material. The choice has to be made taking into account several factors, which play important roles not only in the design/construction phase, but also during the operative life of the vessel. Strength, density, corrosion resistance and cost of wrought material, as well as design and production capabilities, are the main factors, which can influence the final decision. Provided that design and production capabilities are not a constraint, the other aspects have to be deeply considered. The use of lighter materials can lead to significant saving in structural weight and consequently a valuable addition to the earning capacity of the vessel. On the other hand, lighter materials are generally much more expensive and need proper

design and production quality. That is why the choice should be made considering global costs, as a more expensive but lighter vessel could lead to higher profit. Corrosion resistance is directly connected with cost of maintenance.

In case of high-speed vessels of limited length (less then 50 metres), when a considerable number of identical vessels is often built, large-scale production indicates composite materials as the best compromise between low costs and lightness. In case of single units, aluminium alloys are certainly a cheaper solution, as the cost of mould would excessively affect the global cost. Steel is not suitable for high-speed craft of limited dimensions, as minimum thickness required for welding and corrosion, together with the high density of the material, would lead to unacceptable structural weight.

When ship length increases from 50 to 100 metres, FRP construction is technically feasible, but costs involved would be hardly affordable due to the need for expensive tooling and high performance fibres. Aluminium alloys are probably the best solution for vessels up to about 70÷80 metres in length, taking into account that longitudinal strength behaviour and hull deflection are not so critical. When LBP increases to more than 80 metres, mentioned problems become extremely serious and quite difficult to cope with in case of aluminium construction. When ship length exceeds 100 metres, the response of the aluminium structure to global loading conditions has to be carefully evaluated as the risk of structural failures is in a way proportional to overall dimensions. High tensile steel is then the preferred, if not obliged, choice.

A synoptic table comparing aluminium alloys with steel and FRP is proposed (Table 4), taking into account cost of wrought materials and production processes, density, mechanical properties, fire resistance, ease of repair, corrosion resistance and need for maintenance.

7 - WORKMANSHIP PROCEDURES FOR ALUMINIUM STRUCTURES

Proper care and skill are imperative when dealing with aluminium structures.

7.1 - Quality of workmanship

Quality of workmanship is to be higher, as specific qualification of welders and NDT operators is needed. Furthermore all production personnel must be aware that increased attention to production processes is essential in order to avoid all those problems (misalignments, heat-induced buckling, etc.) which can become critical in a highly stressed aluminium structure.

7.2 - Handling

All materials (plates, pre-fabricated extruded panels and single profiles) should be stored in a dry and clean environment, avoiding possible contamination by water, oil, fumes and solid material particles, specially copper and its alloys.

Welding material should be kept in its packaging at room temperature to prevent condensation of moistures. When withdrawn from storing, it should be used within hours or returned to store.

In general, the manufacturing of aluminium alloys should be totally separated from other metal working to avoid possible contamination.

7.3 - Cutting

Aluminium can be cut basically by four methods: machining, shearing, plasma arc and laser.

Shearing of Al alloys containing more than 3,5% of magnesium is not advisable since edges would get sensitive to stress corrosion.

Edges deriving from shearing or plasma arc cutting should be smothered before welding.

7.4 - Welding

Cleaning before welding is of vital importance for aluminium, to avoid inclusions in the seam. Between cleaning and welding only very few hours may pass. The most common welding processes are based on inert gas protection of the melted metal. The welding consumable can be either the electrode (MIG) or not (TIG).

Specially when making primary welds run on - run off tabs should be adopted to avoid crater defects. Tabs have to be made of material compatible with the parent metal. The weld must be protected against weather conditions to avoid porous or oxidised seams.

To reduce distortions, proper welding sequences must be adopted. Excessive distortion may be corrected by heat settings, which have to be adjacent to already heat affected areas.

The acceptance criteria of weld imperfections are based on the ISO 10042 standard and they differ according to the importance of the structural element where welds are applied.

7.5 - Detail design

From a design point of view, aluminium alloys mean special consideration of all structures, which may easily run into damages especially when subject to vibrations or other cyclic loading (fatigue). This implies the study of more sophisticated details so to keep stress concentration factors as low as possible. Details must ensure good continuity, avoid generation of hard points and reduce possible

notches (for instance, mouse-holes should be avoided in all areas subject to significant vibrations, like water-jet room). At the same time, they should be as easy as possible to carry out, as good quality can be reached only ensuring that every manufacturing step is done at best.

TABLE 1

MDV 1200 PEGASUS

MAIN CHARACTERISTICS

Length o.a.		95.00	m
Beam		16.00	m
Max Deadweight	over	400	t

Hull in high tensile steel and aluminium superstructures

TRANSPORT CAPACITY

Passengers		600
Cars	over	170
Buses		6

PERFORMANCES AND PROPULSION

Service Speed	over	37	knots
Main Engines		4 x MTU 20V 1163 TB 73L	
Propulsion Power		4 x 6500 KW	
Propulsion Units		4 KAMEWA 112 Waterjets	

CLASSIFICATION SOCIETIES

RINA	★ 100-A1.1 - HSC (B);TP, Tr (RO-RO); Nav S - IAQ - 2
Germanischer Lloyds	✠ 100-A5"RO-RO/PASSENGER SHIP"; HSC-B, ✠ MC AUT

TABLE 2

MDV 1200 SUPERSEACAT

MAIN CHARACTERISTICS

Length o.a.	100.00 m
Beam	17.10 m
Max Deadweight	340 t
Hull and superstructures in aluminium alloy	

TRANSPORT CAPACITY

Passengers	800
Cars	175

PERFORMANCES AND PROPULSION

Service Speed	38 knots
Maximum Continuous Speed	40 knots
Main Engines	4 x RUSTON 20RK270 diesel engines
Propulsion Power	4 x 6875 kW
Propulsion Units	4 Waterjets

CLASSIFICATION SOCIETIES

RINA	★ 100-A1.1 - HSC (B);TP, Tr (RO-RO); Nav S - IAQ - 2
Det Norske Veritas	✠ 1A1 HSLC PASSENGER CAR FERRY A R1 EO

TABLE 3

MDV 3000 JUPITER

MAIN CHARACTERISTICS

Length o.a.	146.00	m
Beam	22	m
Max Deadweight	1200	t

Hull in high tensile steel and aluminium superstructures

TRANSPORT CAPACITY

Passengers		1800
Cars		460
Trucks (30 t)	up to	30

PERFORMANCES AND PROPULSION

Service Speed	40 knots
Main Engines	2 x Gas Turbines LM 2500
	4 x MTU 20V 1163TB73L Diesel Engines
Propulsion Power	2 x 22200 kW (GT)
	4 x 6500 kW (DE)
Propulsion Units	4 KAMEWA Waterjets

CLASSIFICATION SOCIETIES

RINA	★ 100-A1.1 - HSC (B);TP, Tr (RO-RO); Nav S - IAQ - 2
ABS	A1E Med. Sea Ferry Service - Annual Survey ★ AMS ★ ABCU

TABLE 4: HULL MATERIAL FOR HSC

	MATERIAL		
	Steel	Aluminium alloy	FRP
CHARACTERISTIC			
Density	about 7850 kg/t	about 2700 kg/t	To be determined. Generally low.
Young modulus	about 206000 MPa	about 70000 MPa	To be determined. Low stiffness.
Mechanical properties	High. Successfully used on HSVs up to 460 MPa yield.	Generally good. Properties are reduced in welded zone, leading also to fatigue problems. Structural optimisation through the use of purpose designed extrusions.	Very high specific strength. Possible structural optimisation by selection of fibre type, quantity and direction. Risk of sudden failure due to lack of ductility.
Fire resistance	Non-combustible material.	Low softening and melting points. Need for suitable insulation in moderate or major hazard areas.	Combustible. Fire resisting divisions can be obtained by adding inhibitors. Smoke production can be kept below rule limits, but toxicity could remain a problem.
Corrosion resistance	Low. Need for protective coatings.	Generally good. Some alloys are not suitable for immersed structures (i.e. 6xxx or 7xxx series).	Excellent, provided care is taken in the selection of resin and fibre coating.
Cost of wrought material	Low.	Medium. Delivery time can be considerable in case of extrusions.	High, especially for high performance fibres.
Production costs	Low. Problems arise in case of low thickness (distortions).	Medium. Distortion from welding. Extruded structures are cost-effective.	High, due to need for specific tooling.
Ease of repair	High. Steel technology is wide-spread	Medium. Re-welds are to be avoided. Problems arise in case of extrusions (especially hollow ones). Repairing procedures should be studied and approved together with the overall design.	Medium. Repairing procedures should be studied and approved together with the overall design.

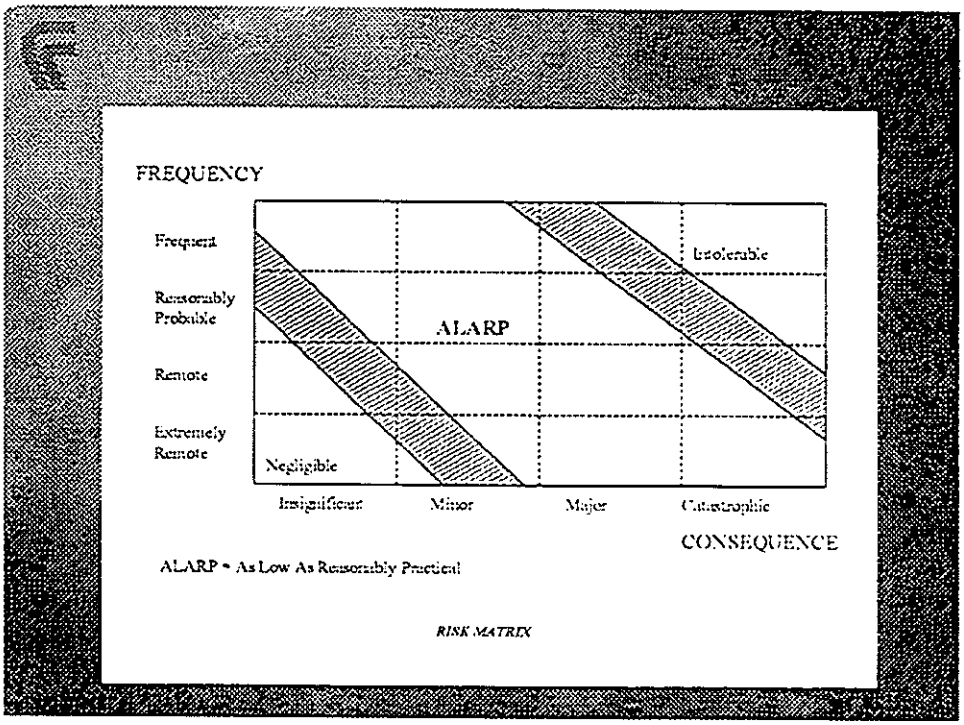

FREQUENCY
Frequent
Reasonably Probable
Remote
Extremely Remote
Intolerable
ALARP
Negligible
Insignificant
Minor
Major
Catastrophic
CONSEQUENCE
ALARP = As Low As Reasonably Practical
RISK MATRIX

FIG. 1

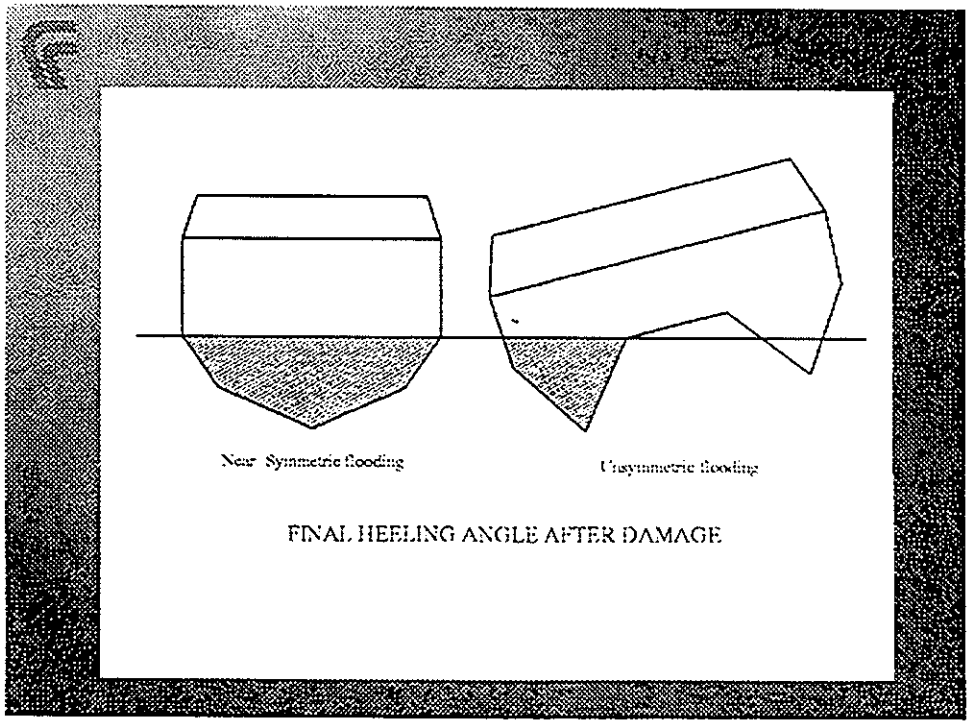

Near Symmetric flooding
Unsymmetric flooding
FINAL HEELING ANGLE AFTER DAMAGE

FIG. 2

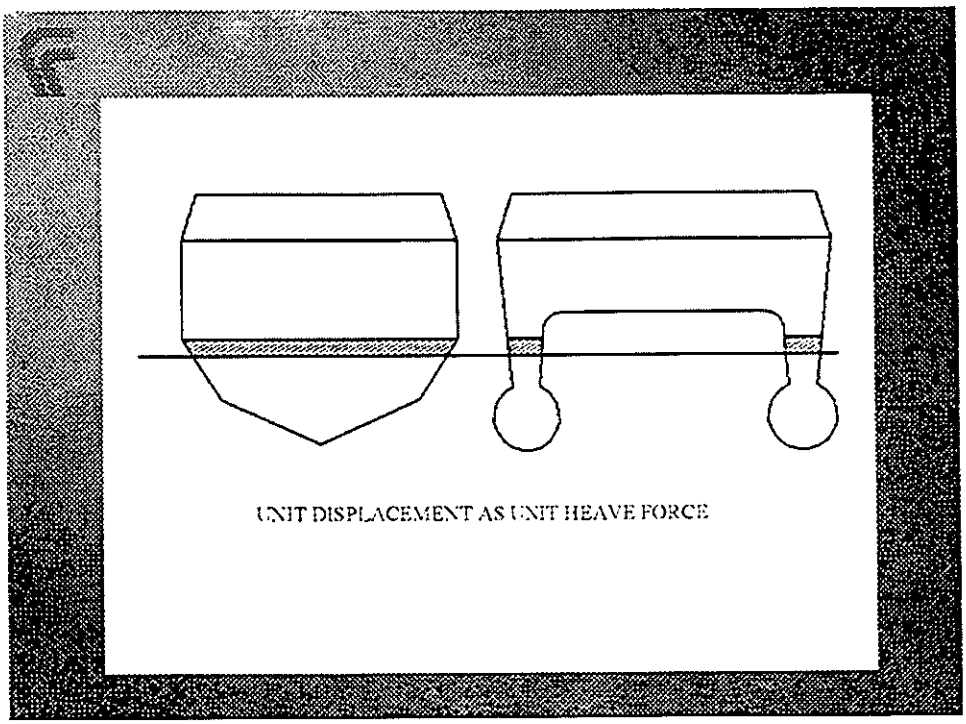

UNIT DISPLACEMENT AS UNIT HEAVE FORCE

FIG. 3

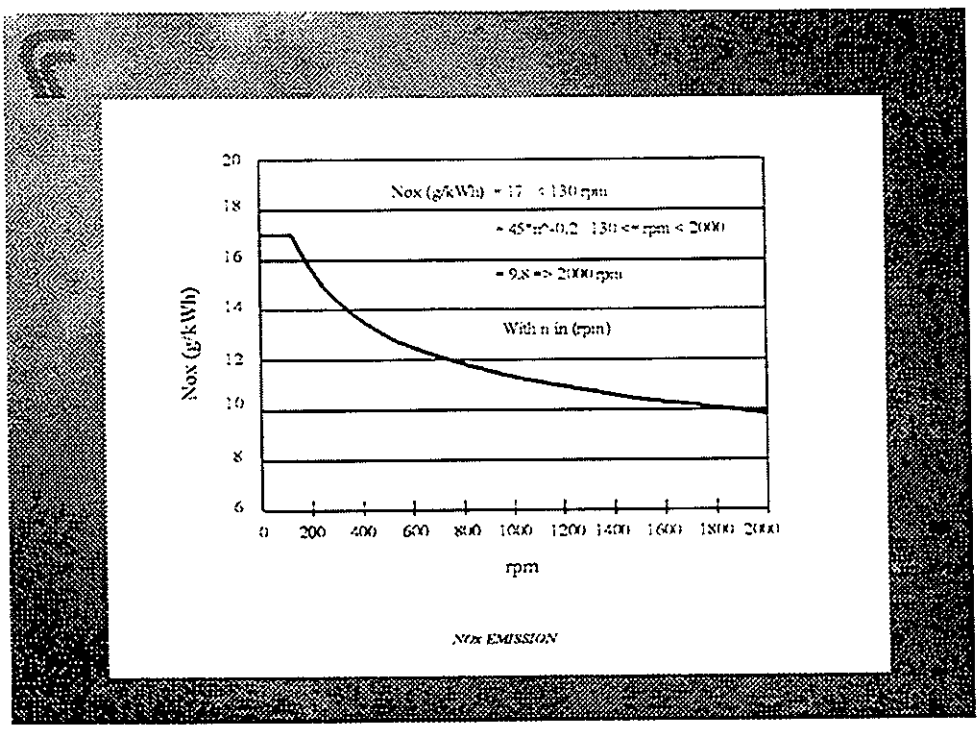

Nox (g/kWh) = 17 < 130 rpm
= 45*n^-0,2 130 <= rpm < 2000
= 9,8 => 2000 rpm
With n in (rpm)
Nox (g/kWh)
20
18
16
14
12
10
8
6
0
200
400
600
800
1000
1200
1400
1600
1800
2000
rpm
NOx EMISSION

FIG. 4

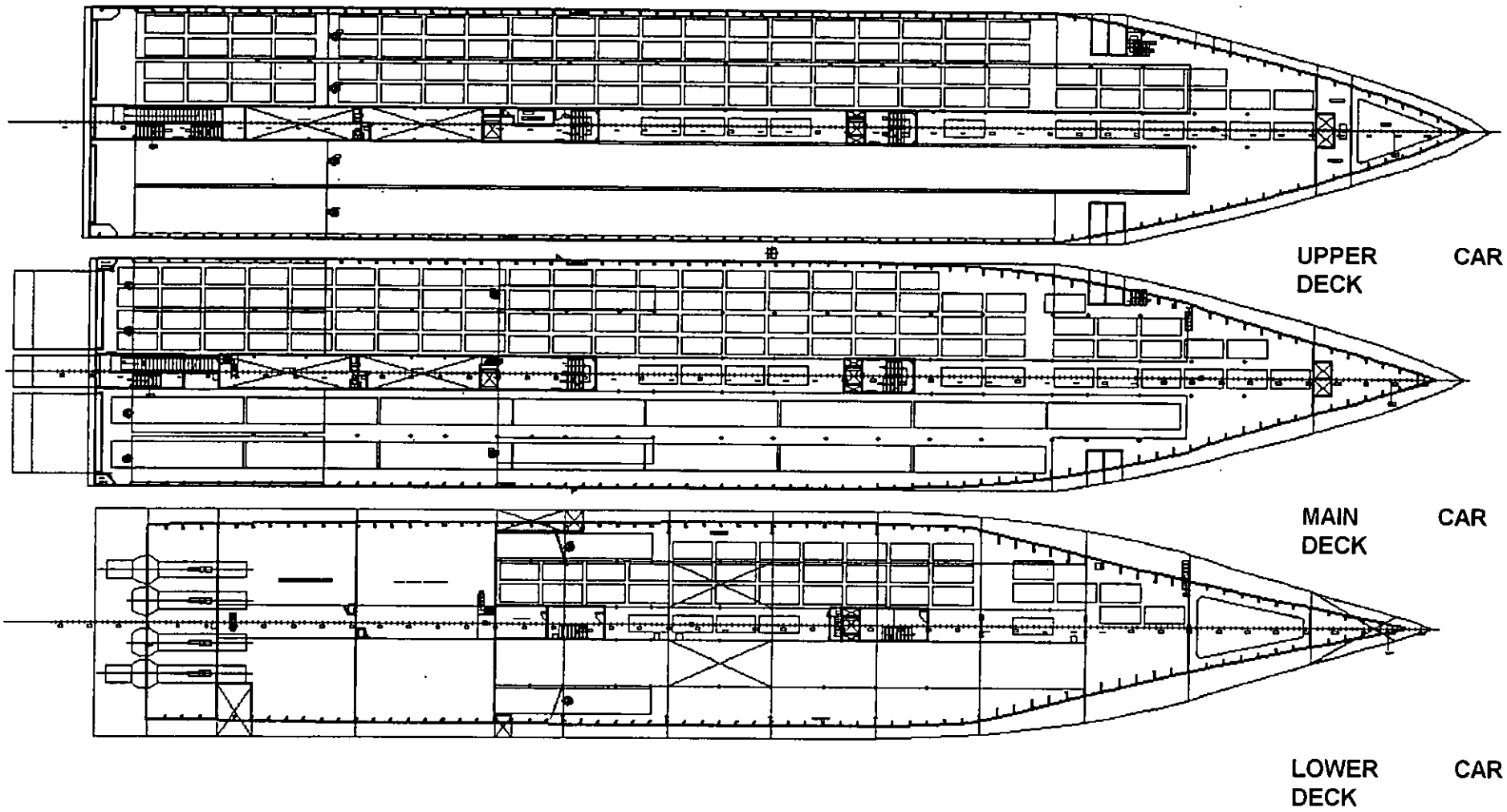


FIG. 5

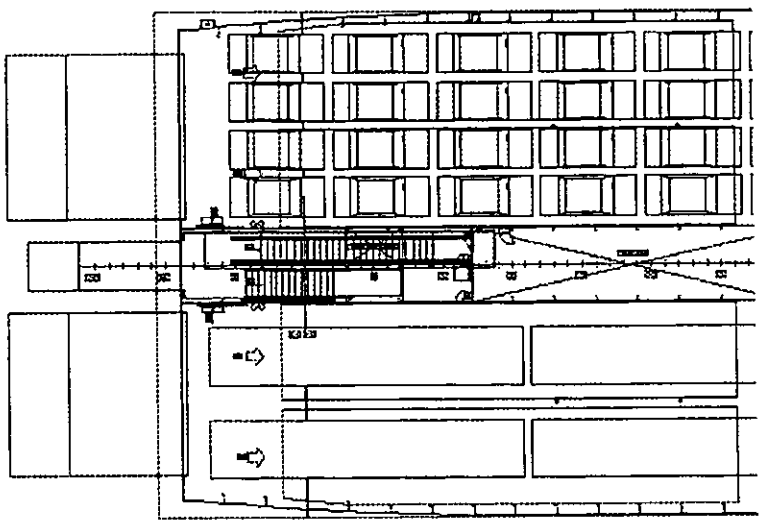


FIG. 6

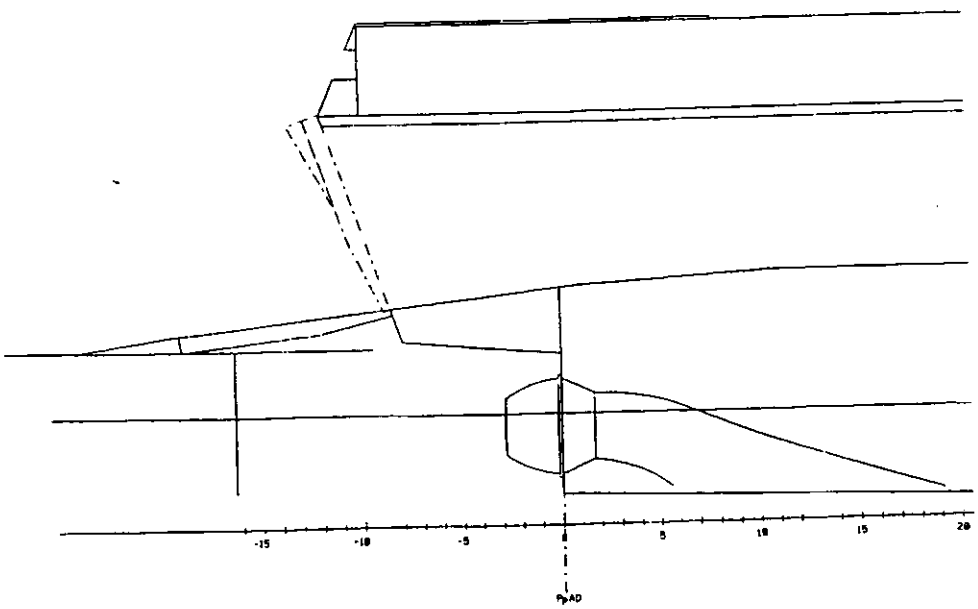


FIG. 7

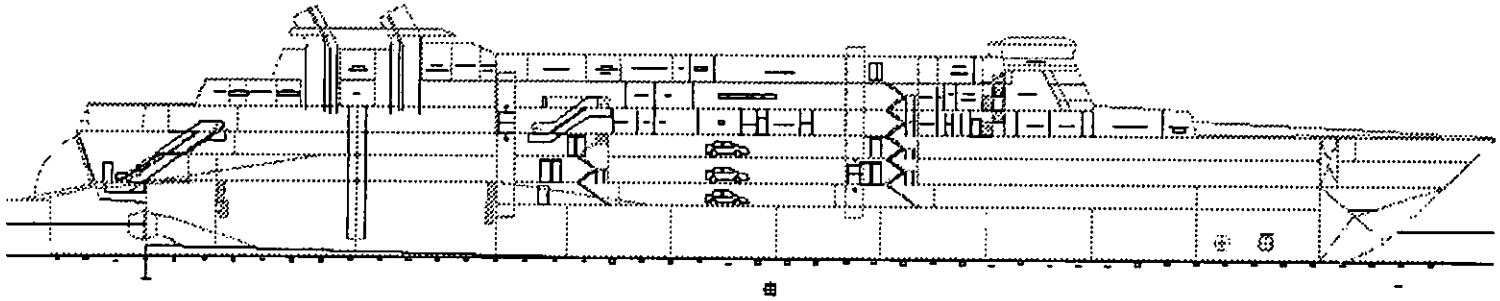


FIG. 8

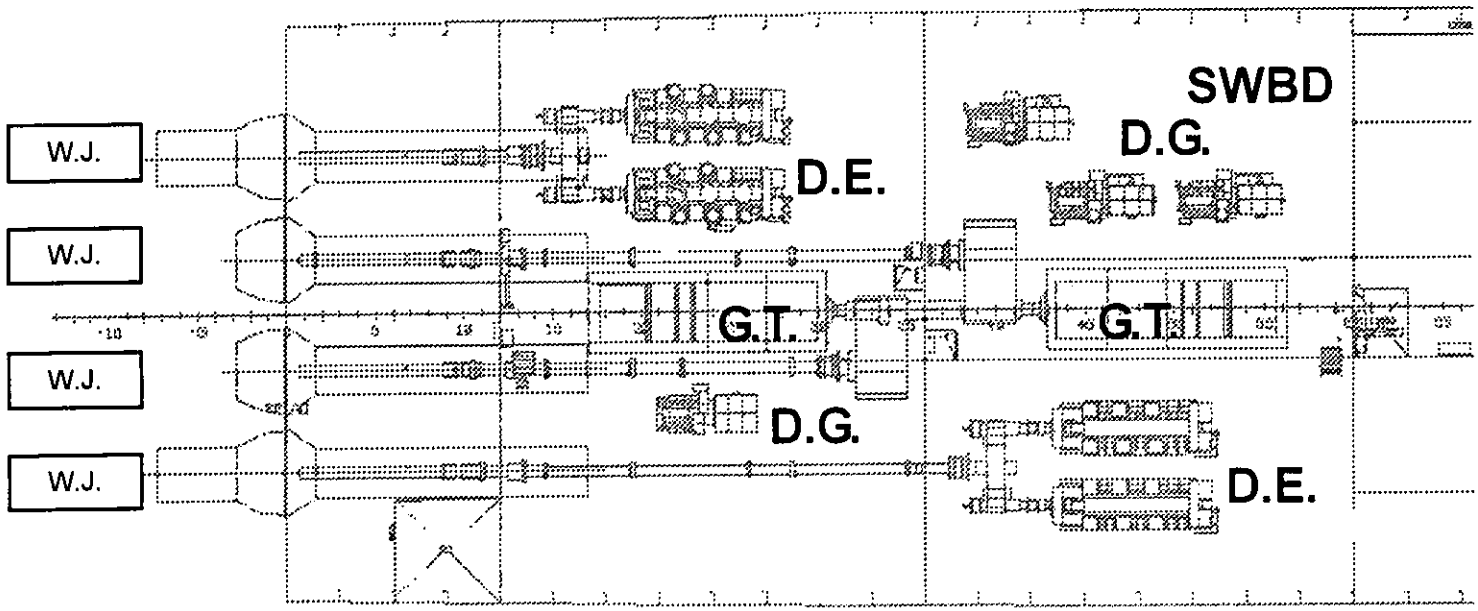


FIG. 9

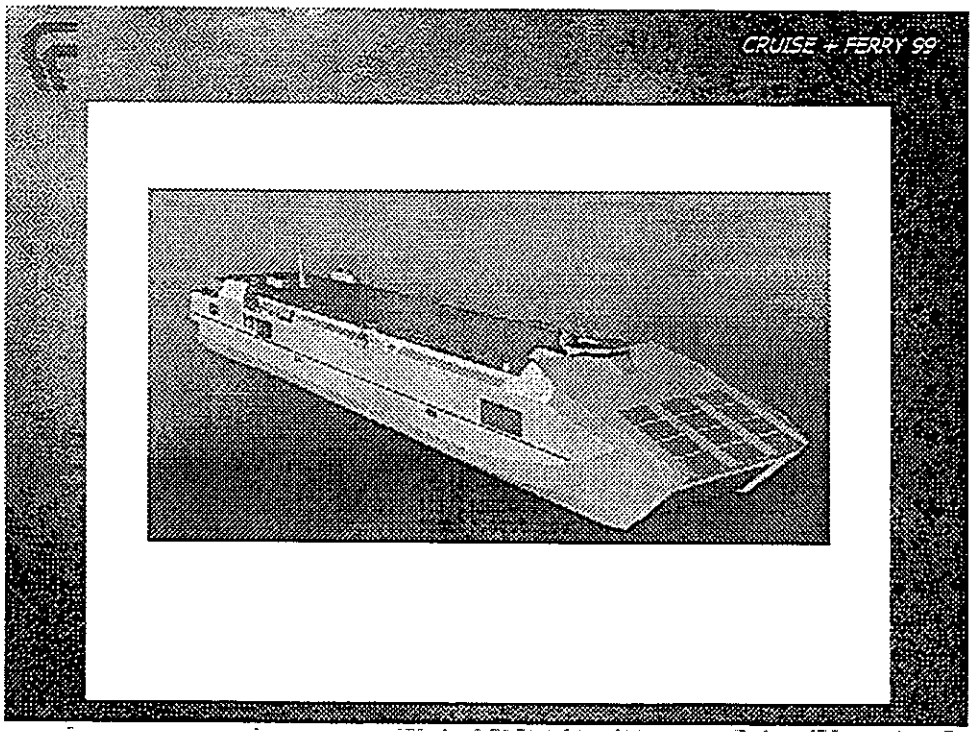


FIG. 10

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

HYDRODYNAMIC REQUIREMENTS FOR DESIGN

Luigi Grossi

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

33 rd WEGEMT School
Fast Ship Propulsion Plants
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HYDRODYNAMIC REQUIREMENTS FOR DESIGN

Luigi Grossi
Hydrodynamic and Acoustic Department
FINCANTIERI Naval Shipbuilding Division, Genova (Italy)

1. Introduction

The recent increase of the demand for high speed, which is being experienced especially in the market of commercial ships, has given most shipyards the opportunity to improve and employ their know-how in the design of fast vessels. The combined low powering and good seaworthiness requirements, which are of vital importance in fast transportation, have stressed the research and development of unconventional hull forms and determined the growth of new ships typologies. Nowadays talking about vessels of more than 4000 tons of displacement which can sail in calm seas at more than 40 knots and grant good seakeeping and speed performances also for sea states higher that 4, is common. This is the field where hard-chine monohulls and multihulls of various types have found their specific range of applicability. Looking into the future, speed limits greater than 50 knots at the same cargo capacity are awaiting the market maturity and extensive research activities are being carried out at present by shipbuilders. For these requirements other types of ships, such as SES, seems to be more suitable.

An assessment of the newest unconventional hull types which are currently under construction or object of research projects, is reviewed, focusing the attention either on the methodology and tools employed in the design phase, either on the peculiar hydrodynamic characteristics of each hull type. The use of the modern computational fluodynamic tools (CFD) in the preliminary stage of the project is useful to reach a satisfying solution, which can fulfill the specifications. The validity and effectiveness of these types of computer simulation programs are discussed on the base of the results achieved in the application on the latest designed ships, which were tested in model scale and/or built. Some criteria on the evaluation of the data, which such programs can provide, are also included.

2. Hydrodynamic Design Aspects and their application to fast ferries

The main factors encountered in hydrodynamic design of fast craft can be listed as follows:

- hull form design;
- resistance;
- seakeeping;
- course stability and manoeuvrability;
- propulsion system layout;
- propulsors;
- hull appendages;
- internal volume, deck area, accommodations
- structure

In the next paragraphs, the various aspects listed above will be discussed with special attention to the peculiar hydrodynamic related activities.

3 Hull Forms

The trend to design always faster and comfortable craft has induced Naval Architects to design hull shapes very different from one other. Each type corresponds to a particular requirement like speed, seakeeping, space availability and economy. The various hulls can be displacement, hydrodynamic or aerostatic sustained. (See *figures a, b, c*)

3.1 Displacement hulls

The most diffused hulls are the displacement ones; these hulls can be mono or multi-hulls crafts. From a hydrodynamic point of view, the common characteristics of these crafts is the high resistance to swell, making them not economic for very high speed.

During the last years, mono-hull vessels have been subjected to technical refining in the hull shapes, propulsion systems and employed materials to improve hydrodynamic efficiency at high speeds. Therefore, various hull configurations have been studied, tested and realized to obtain high speeds maintaining good seakeeping characteristics.

These monohulls derive from round bottoms or hard chine, with more or less deep-V hull. They have low building costs and limited technological risks.

To better meet the requirements of on-board space availability, designers have developed crafts characterized by twin-hulls and a connecting platform (Catamarans). This catamaran hull shape has been particularly improved to meet the passenger fast transport requirements. These crafts are used in calm seas.

A catamaran type optimized for seakeeping is the SWATH. This craft is characterized by a reduced floating area, which allows the reduction of swell effects. However this craft is penalized for high-speed employment, considering submerged surface and therefore is mostly used for ocean research craft.

3.2 Hydrodynamic lifting hulls

The lift, originated by water speed on foil and hull surface, allows the hydrodynamic sustaining of the craft. To this hull category belong hydrofoils with fully submerged or piercing foils and planing hulls. The latter are characterized by mono-hull with hydrodynamic sustaining shapes. They are usually of small dimensions (pleasure craft) and are subject to Sea State.

Hydrofoils are an evolution compared to planing hulls in respect to seakeeping capabilities. The lift acting on the foils makes them the only part in contact with the water. They represent a technological innovation but are limited to small dimensions and very high speeds.

The piercing hydrofoils, technically the simplest, have good seakeeping and speed characteristics, limited only by the foil interference with sea surface.

The fully submerged foil gives to the craft good seakeeping qualities to the detriment of a very sophisticated and expensive control system.

3.3 Aerostatic lifting hulls

The aerostatic lift hulls realize the lift through a fan system, which sends pressurized air in a cushion limited by flexible structures named skirts. In the hovercraft the skirts are extended to the entire craft perimeter and represent the only contact with the sea surface. In this case the propulsion will be aerial, with a considerable decrease in propulsion efficiency. In Surface Effect Ship the air cushion delimitation is made forward and afterwards by skirts and laterally by rigid walls formed by catamaran-type hulls. In these crafts are combined both the positive aspects of catamarans and air cushion craft, particularly the limited resistance to high speeds and the great hydrodynamic performances of the marine propulsion.

3.4 Deep-V monohulls

For fast ships with a Fn between 0.5 and 0.7 a typical hull form is the deep-V. *Figure d* shows a typical deep-V hull[1], which constituted the very early prototype of the modern and reviewed hull forms utilised today in Fincantieri.

Main concepts of these kind of hulls are the triangular midship section with chine in way of the waterline and the practically cylindrical after-body where the main frame section remains almost constant for at least 50% of the ship length.

An extensive campaign of model tests aided by computer simulations on different hull form solutions has been conducted making various modifications to the original model to improve, mainly the seakeeping and resistance characteristics of the MDV (Monohull Deep-V) family of hulls. In the 147 m overall length version, it is the world biggest monohull fast ferry to overcome the 40 knots of speed. Some versions of MDV hulls in the fast ferry application are listed in *Table a.*

Having these hulls commercially found immediate application in the market of fast ferries - transportation at speeds of over 40 knots of cars/buses and passengers which are not usually used to high accelerations and motions- a special care, during the body plan definition, has to be taken to improve the seakeeping quality of the vessels.

	MDV 1200 "SUPERSEACAT"	MDV 1200 "PEGASUS"	MDV 2000 "NEPTUNE"	MDV 3000 "JUPITER"
Hull construction material	alluminium alloy	high tensile steel	high tensile steel	high tensile steel
MAIN CHARACTERISTICS				
Lenght overall	100 m	95 m	128 m	147.7m
Breadth moulded	17.1 m	16 m	19.4 m	22 m
Depth	10.7 m	10.5 m	12.2 m	12.6 m
TRANSPORT CAPACITY				
Passengers	800	600	1000	up to 1800
Cars	175	172	up to 300	up to 450
Coaches		5	up to 8	
Trucks (13.6 m long)				up to 30
deadweight	370 t	400 t	600 t	1400 t
PERFORMANCE & PROP.				
Maximum speed	40	38.5	42	42
Service speed	38 knots	37 knots	40 knots	40 knots
Main engines	4 Diesel	4 x Diesel	CODAG	CODAG
Propulsion power	4 x 6875 kW	4 x 6500 kW	49500 kW	2 x 21 MW + 4 x 6.5 MW
Propulsion units	4 steering WJ	4 WJs (2+2steering)	4 WJs (2+2steering)	4 WJs (2+2steering)
Range	abt 250 miles	abt 250 miles	abt 250 miles	abt 300 miles

Table A - Main characteristics of some MDV family ships.

Main concepts in the design of a deep-V hull, indicated also by various authors[2, 3, 4], are: maintaining high deadrise angles at midship and as far as possible down to the stern; preferring bow convex or mixed (convex-concave) lines together with a certain height of the chine on the waterline at bow to avoid probability of slamming and green water or spray on the deck.

This concept of hull is convenient also for optimizing internal volumes deck area and accommodations. Moreover it is not to neglect the simplicity of construction and the consequent reduction of costs and delivery times.

The displacement/length ratio is of primary importance and has to be accurately chosen in the preliminary phase of the project, and usually, for these Froude numbers, as higher as possible, either for benefits on resistance and on seakeeping.

Other fundamental parameters as length beam ratio and longitudinal center of gravity position, are, in effect, mainly dominated by the transport capacity which usually is given and constrains the breadth of the hull and its length to grant the desired deck area or equivalent car row length.

At the end, for the improvement of the resistance characteristic of the hull, the designer can play, mainly, with small modifications to the hull, as a proper keel-rise and accurate volume distribution to calibrate the LCB location to the LCG estimate position and, as is the case, the introduction of a stern wedge, which will be discussed in the next paragraph.

Special care has to be taken for the afterbody of the ship (*figure e*) which sometimes is designed to install waterjets at the transom. The length and volumes of main engines compartments, for example, has to be chosen accurately, in order to correctly install the main engines, without affecting hydrodynamic performances and stability requirements.

It is not to be neglected the intact and damaged stability, which is one of the main advantages of these types of hulls. The peculiar almost triangular immersed sections continued from the bow down to the stern, in fact, make the waterline transversal moment of inertia very large respect to the volume immersed, with a subsequent high position of the metacentre over the keel. For having an idea on MDV1200LL fast ferry (*figure f*) the lowest metacentric height over the keel is not less

than 8.5m, respect to a depth of 10.5m. Such good intact stability gives a large degree of freedom in the definition of the watertight bulkheads positions and enlarges the survivability of the damaged ship in rough seas. *Ad hoc* towing tank tests of stability for the vessel with two damaged compartments in waves (it was the first example of these tests for fast crafts) has demonstrated[5] the survivability of the MDV1200LL up to irregular waves of significant height of Hs=5m (Sea State 6÷7).

4. Resistance

In order to show up the differences in terms of performances among different types of vessels, identical reference ship displacements and operating speeds are used, as an example, in *figure g*. In this analysis a conventional monohull, a deep-V hull, a SES and a catamaran, all with 1000 tons of displacement, are compared. The table below shows the total installed power versus speed. It can be seen that in the speed range between 30 and 40 knots the SES requires 30% less power than the other ships. At speeds over 40 knots this gap increases to 50%, including also the power needed for lift. The following table contains the most significant data.

Ship type	Speed (kn)	Displacement (t)	Installed Power (kW)
Catamaran	40	1000	25000
Monohull (Round bilge)	40	1000	27000
Monohull (Deep-V)	40	1000	24000
SES	40	1000	16000

The deep-V hulls in table A are to be considered displacement hulls for their operational speeds and semi-planing hulls only at very high speeds. It results also from other authors, that the resistance of deep-V hulls can be comparable at high speed to that of a reference round bilge. *Figure h* represents a comparison made in the case of a Fincantieri MDV hull against a reference fast round bilge brought at the same length and the same displacement. It is visible that for high Froude numbers the resistance of the deep-V hull is lower than the RB one, while at low speeds it suffers of a peculiar increment respect to the RB. The involuntary speed loss in rough sea for a deep-V, in addition, is considerably less than that for a RB. For the MDV1200 "Superseacat" at sea, a reduction of 1.7 knots from the maximum speed of 40 knots has been measured in head sea with Hs=3.0m (and contrary wind). The involuntary reduction for a round bilge of the same displacement should have been nearly doubled. The values measured at full scale agreed well with the towing tank tests which predicted a reduction of less than 1.5 knots for Hs=3.0m (no adverse wind) and 0.2 knots for Hs=2.0m. This fact is essentially due to the good seakeeping characteristics of these hulls, which will be discussed later on.

As for displacement fast round bilge hulls, the resistance in still water is very sensible to the dynamic trim and sinkage assumed at high speed by the hull.

A known device to play with for obtaining the desired trim and reducing the resistance, is a stern wedge (*figure i*). In the case of for the MDV1200, once optimized, it was able to reduce the still water resistance of the hull of about the 5% in the high range of Froude numbers.

Series of towing tests have been made varying the inclination of the wedge and its shape. During the preliminary design phase a number of CFD simulations were performed, using a linear potential

flow panel method already successfully applied to fast round bilge monohulls and catamarans[6], to estimate the wave resistance of the hull with the various solutions, and to orientate the choice towards a restricted number of configurations. Considering potential flow, only, it was found that inclination of 9-10 deg of the chosen configuration offered the lowest wave resistance. Towing tank tests confirmed partially the trends of the results predicted by the programs. It was found, in fact, that increasing the wedge angle over the optimum experimental entity, causes a decrease in the running trim predicted by the CFD program, but leaves the resistance almost unchanged.

Other devices, like interceptors, may be utilized in order to obtain the correct trim of the vessel.

Waterjets were chosen as propulsive devices for the MDV. In fact, for speed higher than 35 knots, in general, propulsion systems based on flush inlet waterjets, up to now are able to grant higher overall efficiencies, than propeller various alternatives. The typical transom stern of the deep-V offers, in addition, very easy fitting of the waterjet nozzles. At low speeds (below 25 knots for our case), instead, the efficiency of waterjets systems passes under the propeller one and dramatically increase the negative gap which has been noted in *figure h* on the effective power.

Then, on the whole, it follows that only for ships, whose operational profiles are centered mainly towards high speeds, it is convenient to adopt a deep-V hull con-figuration, unless other strong sea-keeping operativity requirements tend in the opposite sense.

This is the case, for instance, of FAC or of fast ferries too, which spend most of their life sailing at full speed.

5. Seakeeping

A number of operability criteria, nowadays, exist to define the seakeeping quality of a ship, which, in general, derive from research made in the navies world. Together with the requested operability performance of the ship, one or more "mission" profiles and the external environment reference conditions are, usually, given to the designer, already in the early stages of the project.

Besides the common operability criteria which involve the ship as a system (i.e. hull damage, deck equipment damage, systems-structures damage, propulsion plant and system efficiency), there are others which are connected specifically to the mission profile and regard for instance ship's payloads (helicopter/aircraft landing, cargo shifting...) and personnel/passengers (comfort, motion sickness, fatigue, task proficiency and safety).

A comprehensive seakeeping performance criteria is given in reference[7, 8]. Example in the field of fast passenger transportation is motion sickness incidence (MSI).

This subject was first studied by O'Hanlon & MacCauley[9], which statistically related the motion sickness incidence of a group of non-acclimatized people to the amplitude of a regular vertical sinusoidal oscillation of a given frequency and the exposure time. These results combined with more recent requirements of ISO Standards[10] in the manner explained in reference[7, 8], brings to the definition of some curves that superimposed to the energy spectra of the vertical acceleration, limit the punctual value of each frequency component of the vertical acceleration spectra at a given ship's point (example in *figure l*).

The table below shows a comparison of the vertical acceleration, as a function of g, at the center of gravity of different types of vessels. This parameter is significant of the level of comfort aboard. The data refer to a speed of 45 knots and a displacement of 1000 tons. In these conditions it can be seen, from *figure k*, that with a SES low vertical acceleration values are possible even in Seastate 5 and 6. The following table shows the most significant terms of comparison:

Ship type	Speed (kn)	Displacement (t)	Acceleration (g)		
			Seastate 3	Seastate 4	Seastate 5
Deep-V monohull	45	1000	0.04	0.08	0.18
Conv monohull	45	1000	0.07	0.12	0.23
Catamaran	45	1000	0.07	0.12	0.20
SES	45	1000	0.07	0.10	0.15

Since only the heave, pitch and roll motions are responsible of the vertical acceleration at a point of the ship, for example, the peculiar high damping characteristics of the deep-V to these motions make these types of hulls preferable to normal round bilge ones. A series of model forced roll tests performed to determine the roll damping characteristics of MDV hulls at various speeds, confirmed, as expected, is notably higher that that of a correspondent round bilge hull. The roll damping predicted by strip theory seakeeping programs which uses empirical formulas valid mainly for RB hulls, need to be corrected to adequately reproduce the experimental results.

For similarly good designs, pitch and heave motions of deep-V hull result a bit weaker than those of the corresponding RB, as well. In agreement with general indications of seakeeping optimization practice for fast slender ships, in fact, Deep-V hulls offer high C_{WL} coefficients, in addition to relatively low vertical prismatic coefficients C_{PV}; characteristics which are usually deemed to reduce vertical motions responses in rough seas. In *figure j*, the pitch motion RMS amplitude of MDV1200 "Superseacat" (L_{BP}=88m) is maintaining under the 0.6 deg (RMS) values for sea state 5.

In the design of the MDV hulls, then, a strong effort to increase the passengers' comfort on board has been made in the above-explained sense of MSI. Extensive use of computer simulation programs has been made from the first hull lines definitions to address the towing tanks tests optimization activity towards the most effective direction. Many different hull configurations were simulated and tested in model scale with irregular waves. Different bow forms, chine profiles and stern configurations were tested. Being the operational area of these fast ferries most of the time restricted and not typically open, Jonswap spectrum was preferred to define the various sea state conditions.

The present achieved results are presented in *figure l*, where the energy spectra of the vertical accelerations at LCG in head sea is superimposed on the MSI curves for sea state 4 (Hs=2 m and Tm=6.5 s) and sea state 5 (Hs=2 m and Tm=7.5 s), for the "Superseacat" fast ferry; the considered exposure time is 3 hours, which is thought to represent the typical voyage length of a fast ferry of this kind.

Excellent threshold values of MSI=1% and MSI=5% (i.e. 1% and 5% of unused people is probable to be sick after a 3 hours voyage in sea states 4 and 5) for considered sea states conditions SS4 (Hs=2m Tm=6.5s) and SS5 (Hs=3m Tm=7.5s) are accomplished.

At present a deeper investigation on the influence of a systematic parametric variation on seakeeping and resistance performances is conducting by use of different computer codes and will be finalized with seakeeping in irregular waves coming from different direction and resistance tests on the two most promising hulls. This research project should lead to the definition of the medium-small deep-V hull of the next generation.

6. Manoeuvrability and Coursekeeping

Manoeuvrability and coursekeeping are two important aspects in the design of fast ships. A fast ship is requested to have stopping distances and turning diameters as small as possible, since at high speeds, keeping constant the human reaction times, the distances with unexpected obstacles become shorter.

Generally the absence of appendages in the stern part of the hull as shaft brackets and rudders, due to the typically installed waterjet propulsion systems instead of conventional propeller arrangements, which is an advantage for the resistance point of view, obliges the designer to deeply study the coursekeeping characteristics of the vessels.

Theoretic evaluations of the dynamic course stability of the bare hull is quite difficult being the database of hydrodynamic derivatives scarce for hull forms like MDV, catamaran and SES. So, in general, PMM tests with fully appended and propelled models or mixed combined theoretical and experimental approaches are necessary.

In addition, some peculiar problems such as the evaluation of the influence of the waterjets inlets flow, which sometimes can develop stabilizing yaw moments, or the evaluation of the full scale transversal component of steering waterjets as a function of the bucket angle, are relatively new to naval architect's experience, but are obviously essential to determine the course stability and the manoeuvring characteristics of fast crafts.

The approach followed by Fincantieri[11] for deep-V hulls was of the mixed theoretical and experimental type. The hull was tested at the PMM (at Danish Maritime Institute) without and with a series of appendages, towed. The influence of the waterjet steering forces and waterjet inlets were theoretically evaluated and added to the derivatives of the experimental hydrodynamic measured forces.

So it was possible (***Errore. L'origine riferimento non è stata trovata.m***) to define the proper stabilizers fixed fins longitudinal position (for MDV1200LL and MDV3000) or fixed stern fins (for the MDV1200 which has retractable stabilizers fins) to grant the desired dynamic course stability.

In *figure m*, in fact, the various segments represent the derivatives of the ship turning rate respect to the steering angle in the origin ($d\psi/d\beta$ for $\beta=0$) of the various configuration either tested or desk evaluated.

The curve represents the result of a simulated bare hull Dieudonnè spiral manoeuvre. The instability hysteresis loop is not visible, since the simulation is theoretic, so it is possible to observe the derivative in the origin of the axis. In this representation positive derivatives (segments 1st-3rd quadrant) imply dynamic course unstable ship's configuration, though negative (segments 2nd-4th quadrant) mean course stable ship's configurations.

To assure the desired course stability, a couple of fixed stabilizers fin were placed at after midship for the MDV1200LL and for the MDV3000, while for the MDV1200, which is equipped with a pair of retractable antiroll fins, a couple small fixed fins were placed at the stern just in front of the transom. Besides a sufficient hull dynamic course stability there must exist an appropriate autopilot interfaced with the stabilizer fins control system, to grant integrated yaw motion control.

Manoeuvring characteristics of MDV hulls very much depend on the waterjet systems configuration.

The whole MDV family of vessels, in fact, is provided with four stern waterjets, which can be all steerable, or two boosters and two steering, and bow thrusters. The steering waterjets have also

the ability to inverse the thrust by means of a reversing bucket. The combination between waterjets' directional and reversible thrust and bow transverse thrusters, give the vessels an excellent harbour manoeuvring ability, also in case of strong wind.

Sea trials tests registered, at full speed, final diameter varying from 4.5 to 6.5 ship's length and crash stopping distances of 4 to 5 ship's length in relation to the divers propulsive arrangements.

7. Conclusions

Main concepts in the choice of appropriate hull forms have been analyzed for application to displacement / high speed semi-planing vessels. For a deep-V hull, can be summarized in the following topics:

- triangular midship section with chine in way of the waterline and high deadrise angles (25÷30 degree);
- the main frame section remains as much as possible triangular in the almost cylindrical after-body for more than 50% of the ship length;
- bow convex or mixed (convex-concave) lines together with a certain height of the chine on the waterline at bow to avoid probability of slamming and green water or spray on the deck;
- Length displacement ratio as high as possible to reduce wave resistance;
- Proper LCB positioning in relation to LCG location, to obtain favorable initial trims.

The simplicity of construction which derive by the adoption of hard chines hulls respect to RB ones and the good intact and damaged stability properties are point of success of these family of hulls.

Other advantages and design aspects in terms of resistance, seakeeping and manoeuvrability have been analyzed.

Still water resistance of deep-V hulls result slightly lower for high Froude numbers, though higher for slow speeds and make these vessels convenient over the RB for high speed centered operating profiles. Indications about the effectiveness of a stern wedge to reduce wave resistance on MDV hulls has been given, resulting in a total effective power reduction of more than 5%.

Nevertheless, the added resistance in rough sea for the MDV hulls is exceptionally lower than a corresponding RB one. The sea trials in rough seas confirmed the towing tank predicted values of 1.5 knots involuntary speed loss for the 88m long fast ferry, in sea state 5 (3m significant wave height). This value is estimated to range around half the corresponding RB one.

Seakeeping superior characteristics of MDV hulls were also presented and can be summarized as follows:

- higher roll damping of the bare hull (due to the vortex shed by the immersed chines) than a reference RB hull;
- low pitch and heave responses due to high C_{WL} coefficients and low C_{PV} coefficients typical of deep-V hulls;
- the first two statements resulting in low vertical accelerations along ship which can grant for the 88m fast ferry motion sickness incidence less than 1% in sea state 4 (Hs=2m) and less than 5% in sea state 5 (Hs=3m), head sea.

Coursekeeping ability of the vessels derive from both the combination of a sufficiently stable hull and an adequate integrated control system of stabilizer fins and waterjets (and other control

surfaces) to grant the best synergetic yaw motion control. Some results of PMM tests made on a MDV hull have been presented together with the device adopted to grant the dynamic course stability of the hull.

Manoeuvring characteristics at high speeds depends very much on the propulsion arrangement installed and can give exceptional performances in the best cases.

Restricted water manoeuvrability is excellent and assures satisfactory ability to combat strong winds, due to the combined actions of waterjets' directional and reversible thrust and bow thrusters.

DISPLACEMENT HULLS

TRANSVERSAL SECTIONS

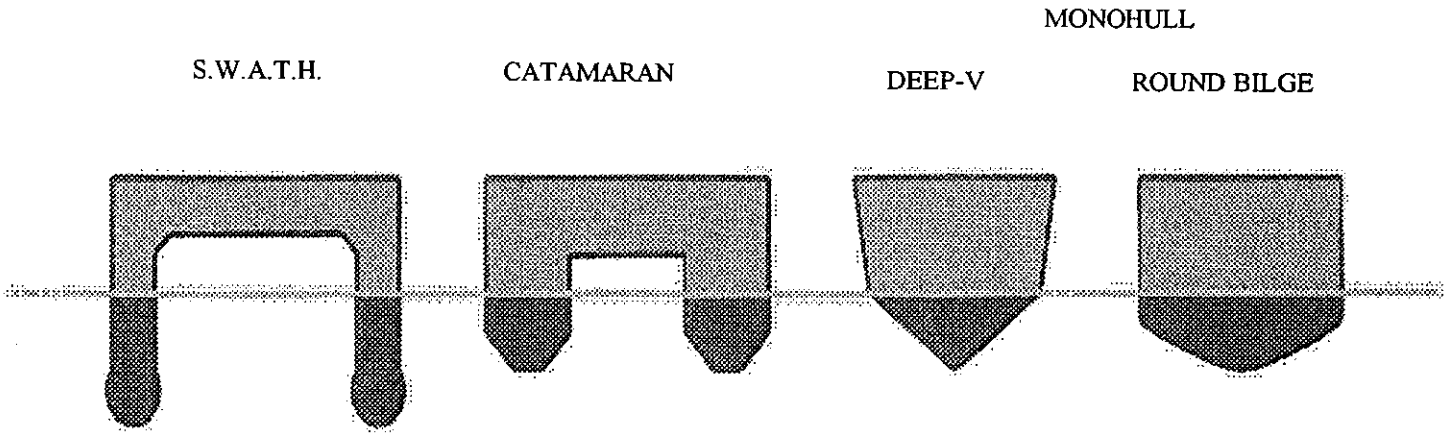


Figure a

HYDRODYNAMIC LIFTING HULLS

TRANSVERSAL SECTIONS

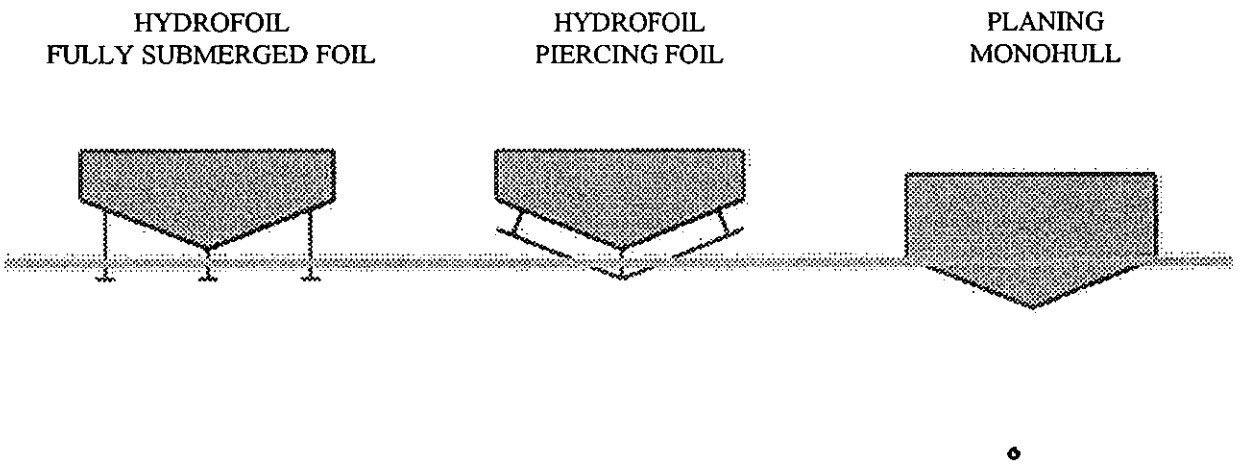


Figure b

AEROSTATIC LIFTING HULLS

TRANSVERSAL SECTIONS

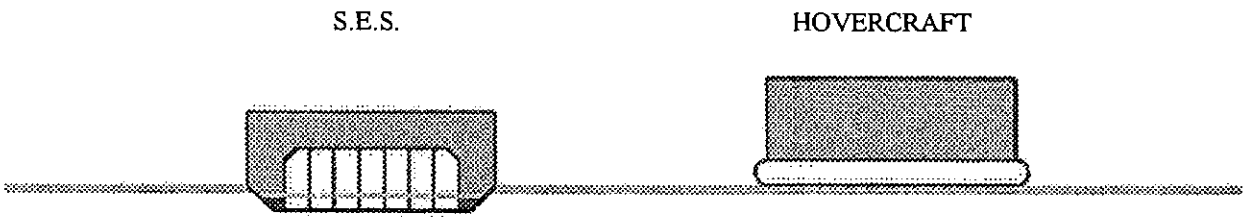


Figure c

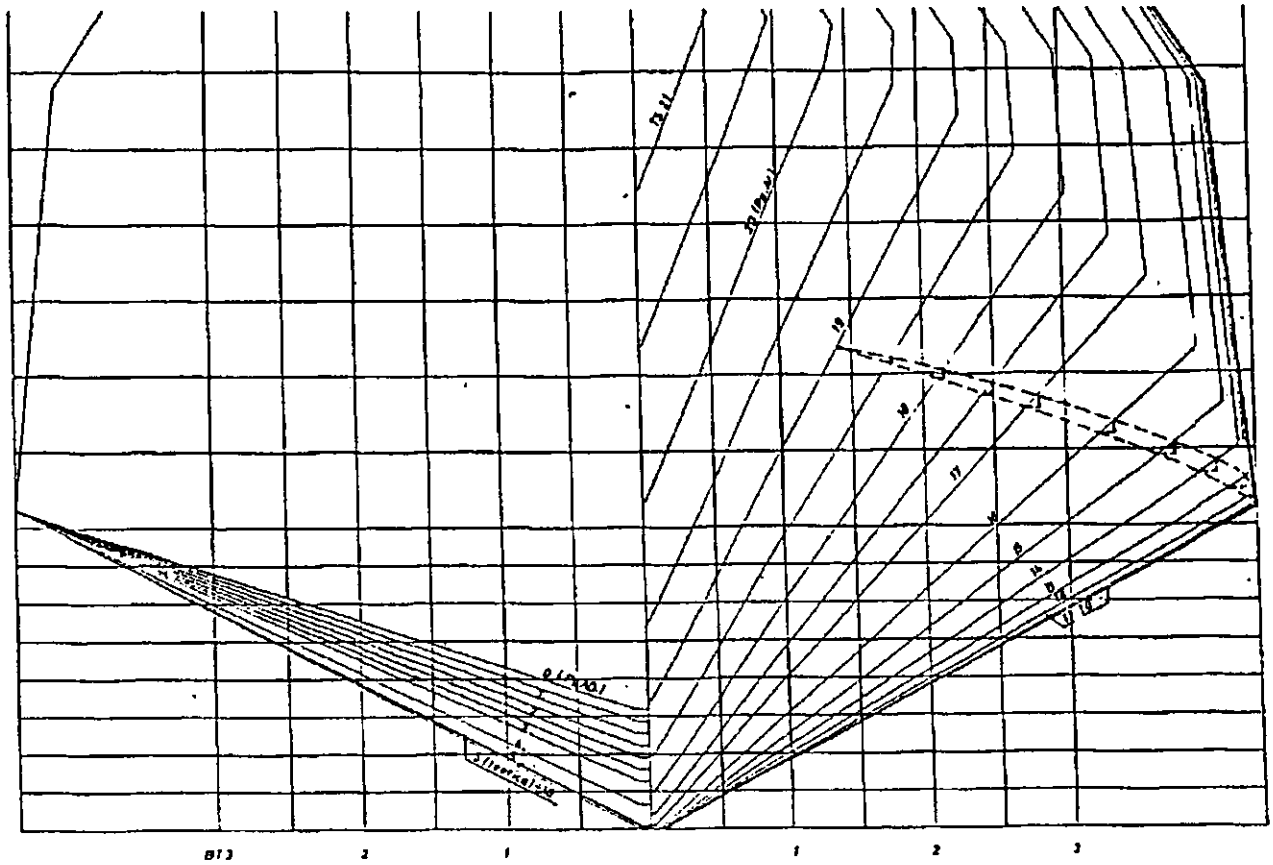


Figure d - First version of deep-V for small FAC application (1981)

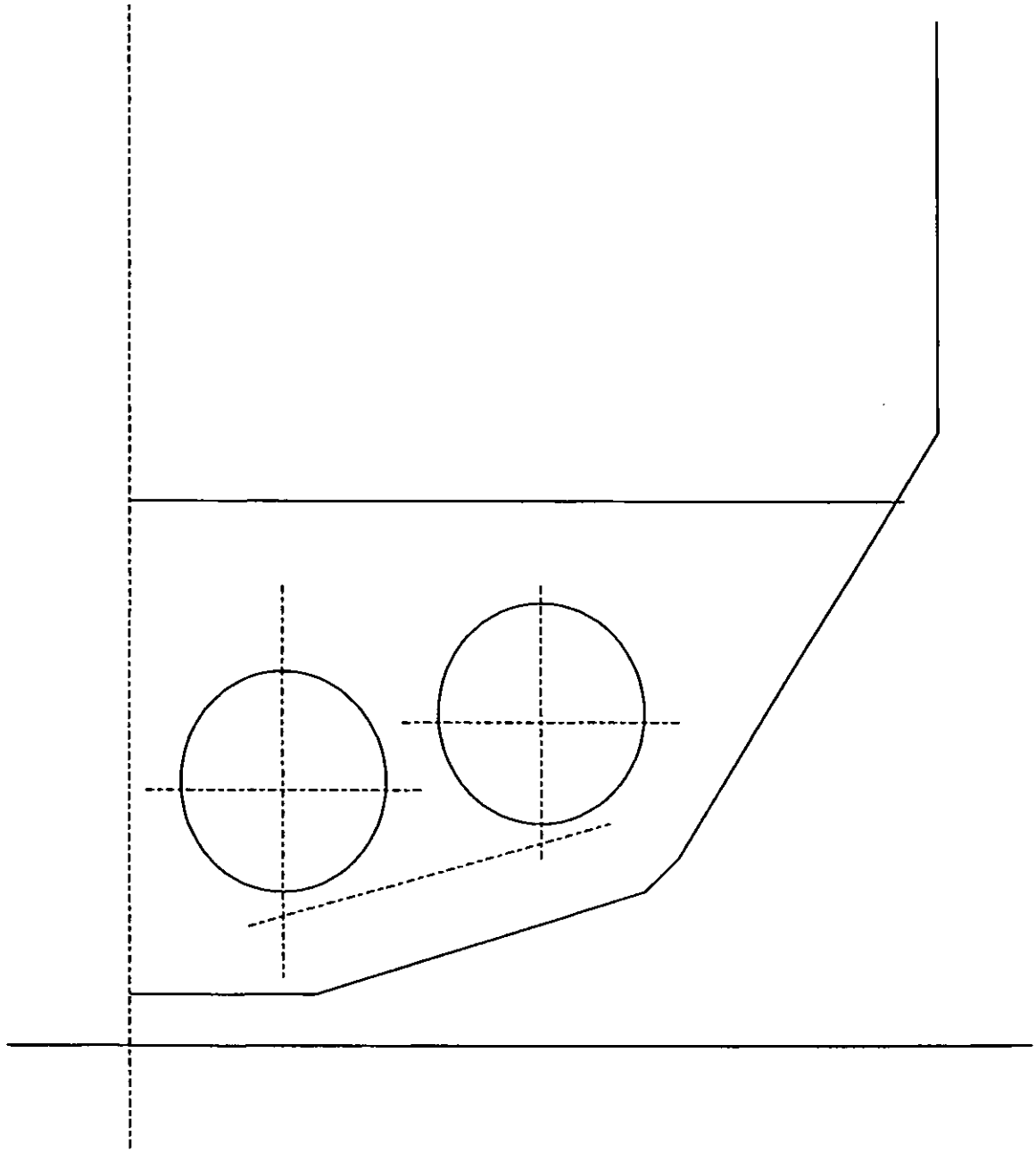


Figura e Example of stern transom with 4 waterjets

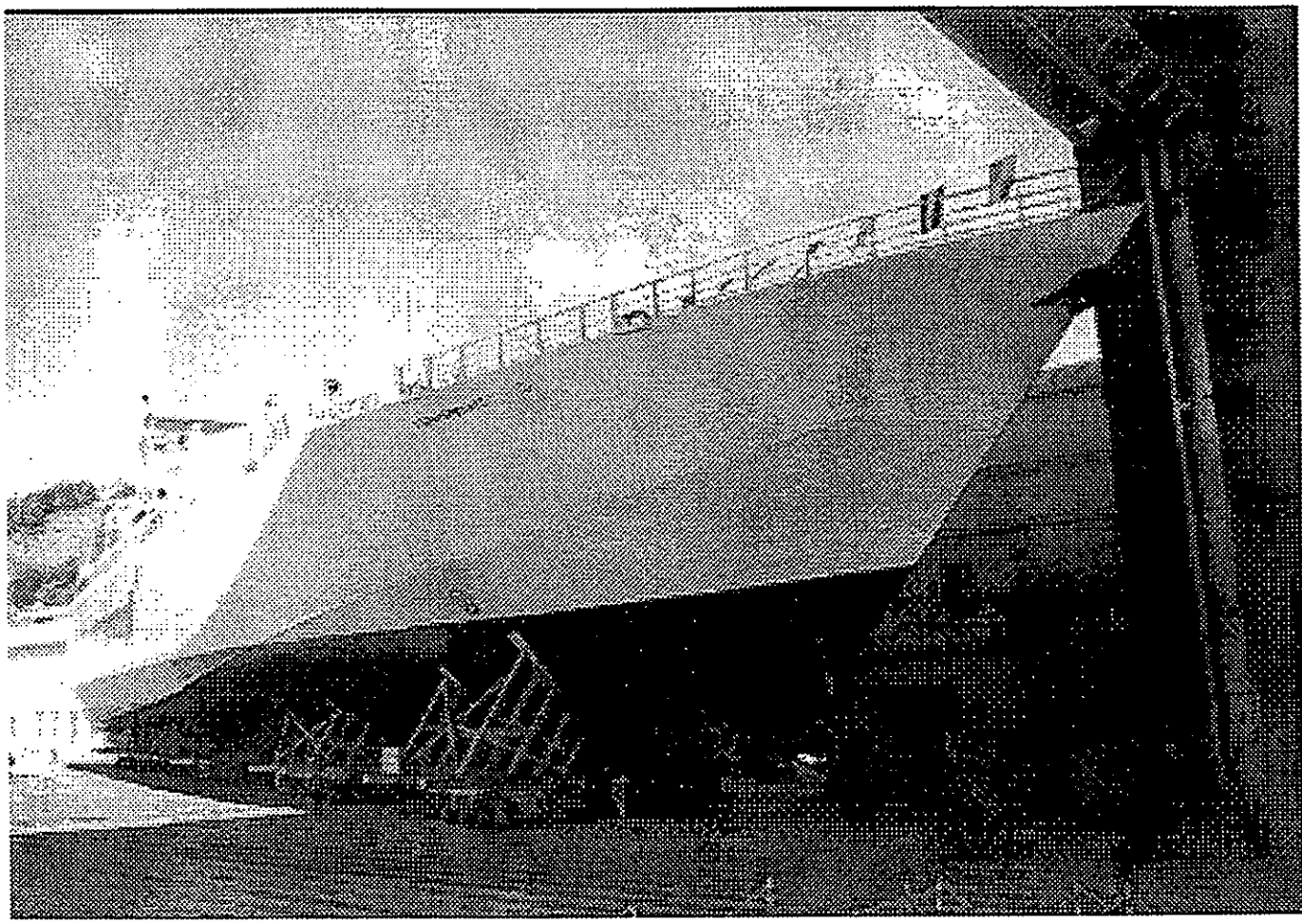


Figure f - Prospective view of the MDV-1200LL "Superseacat" hull (March 1997)

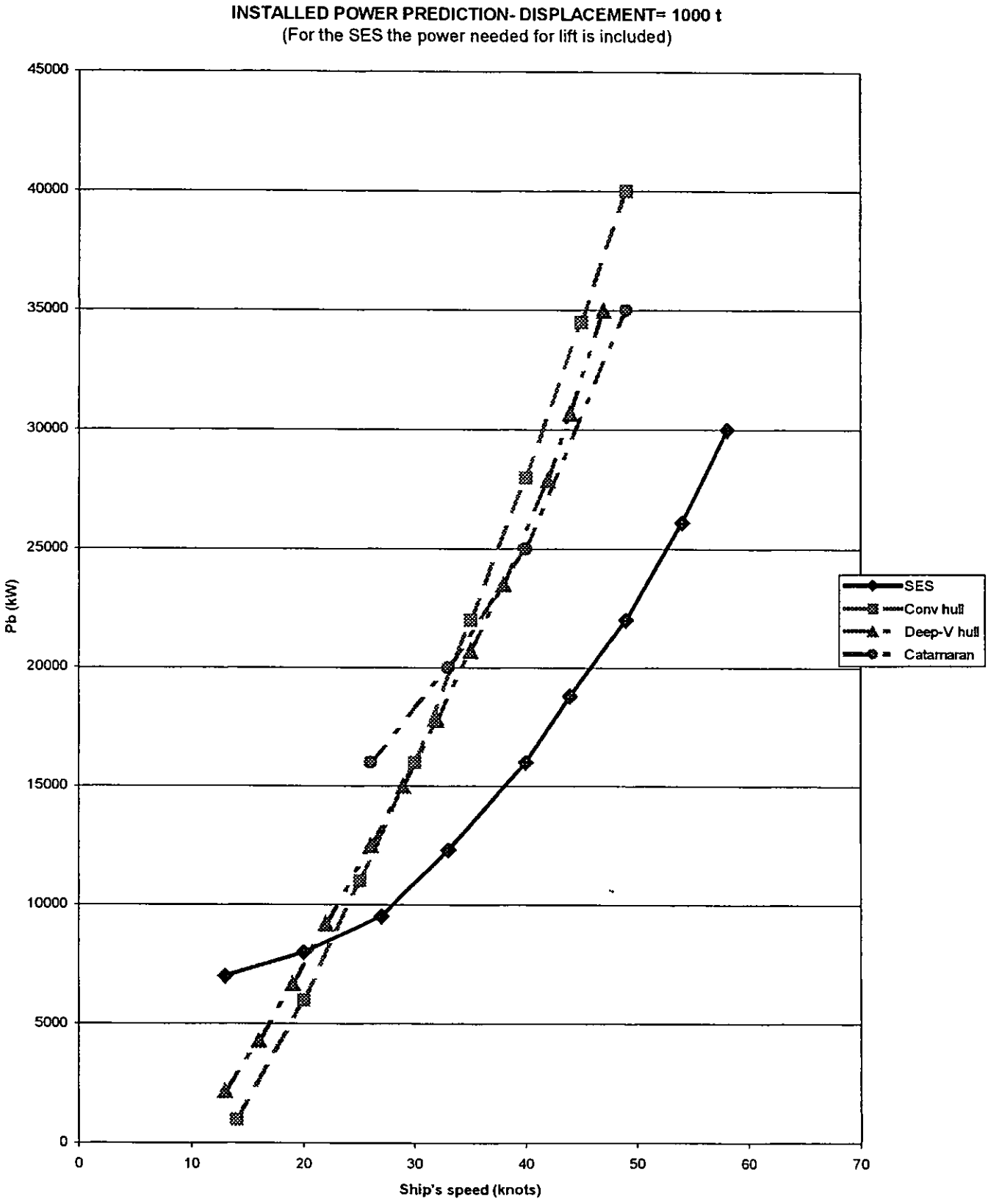

INSTALLED POWER PREDICTION- DISPLACEMENT= 1000 t
(For the SES the power needed for lift is included)
Pb (kW)
45000
40000
35000
30000
25000
20000
15000
10000
5000
0
0
10
20
30
40
50
60
70
Ship's speed (knots)
SES
Conv hull
Deep-V hull
Catamaran

Figure g

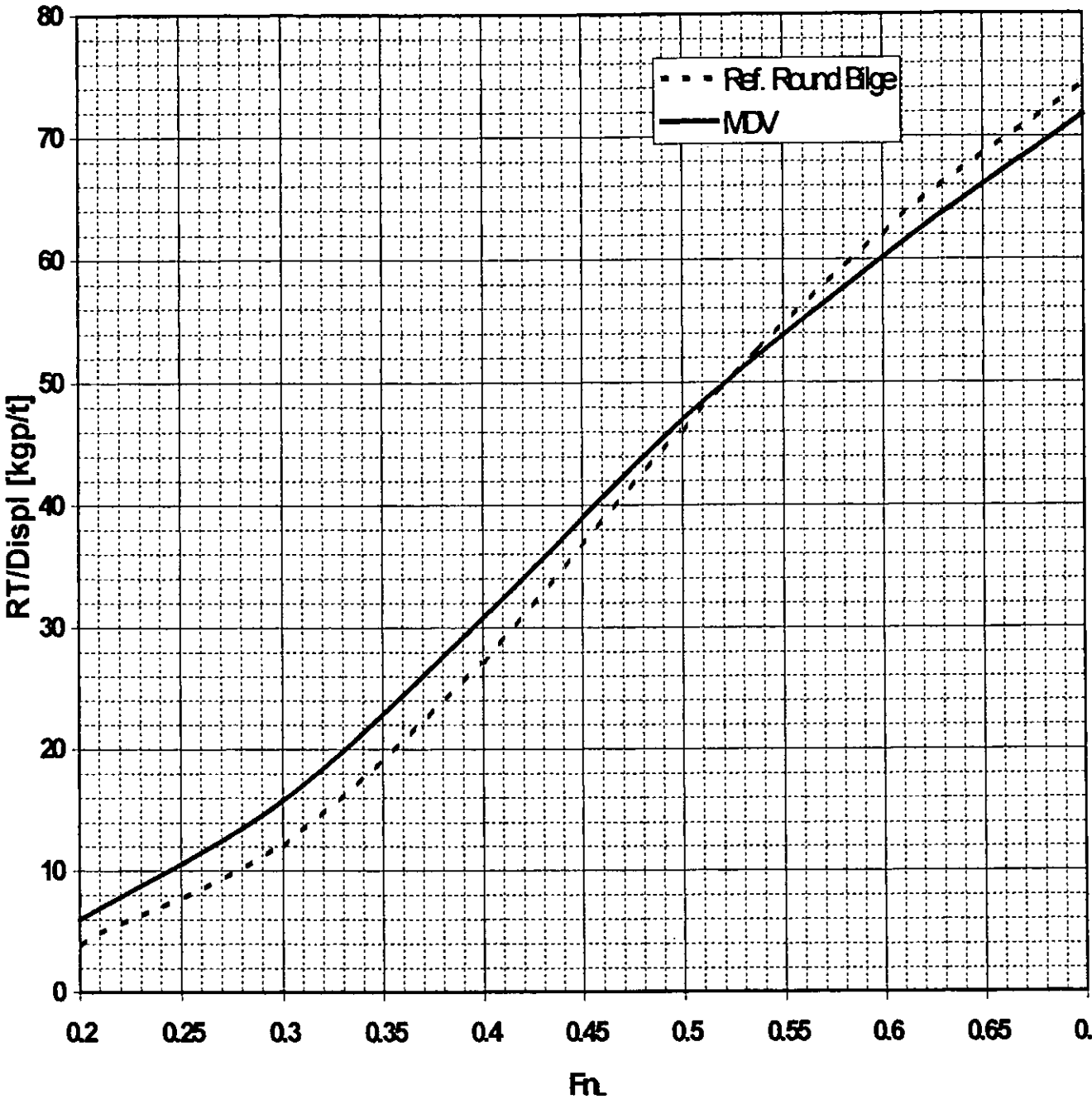


Figure h- Still water resistance comparison Round Bilge / Deep-V

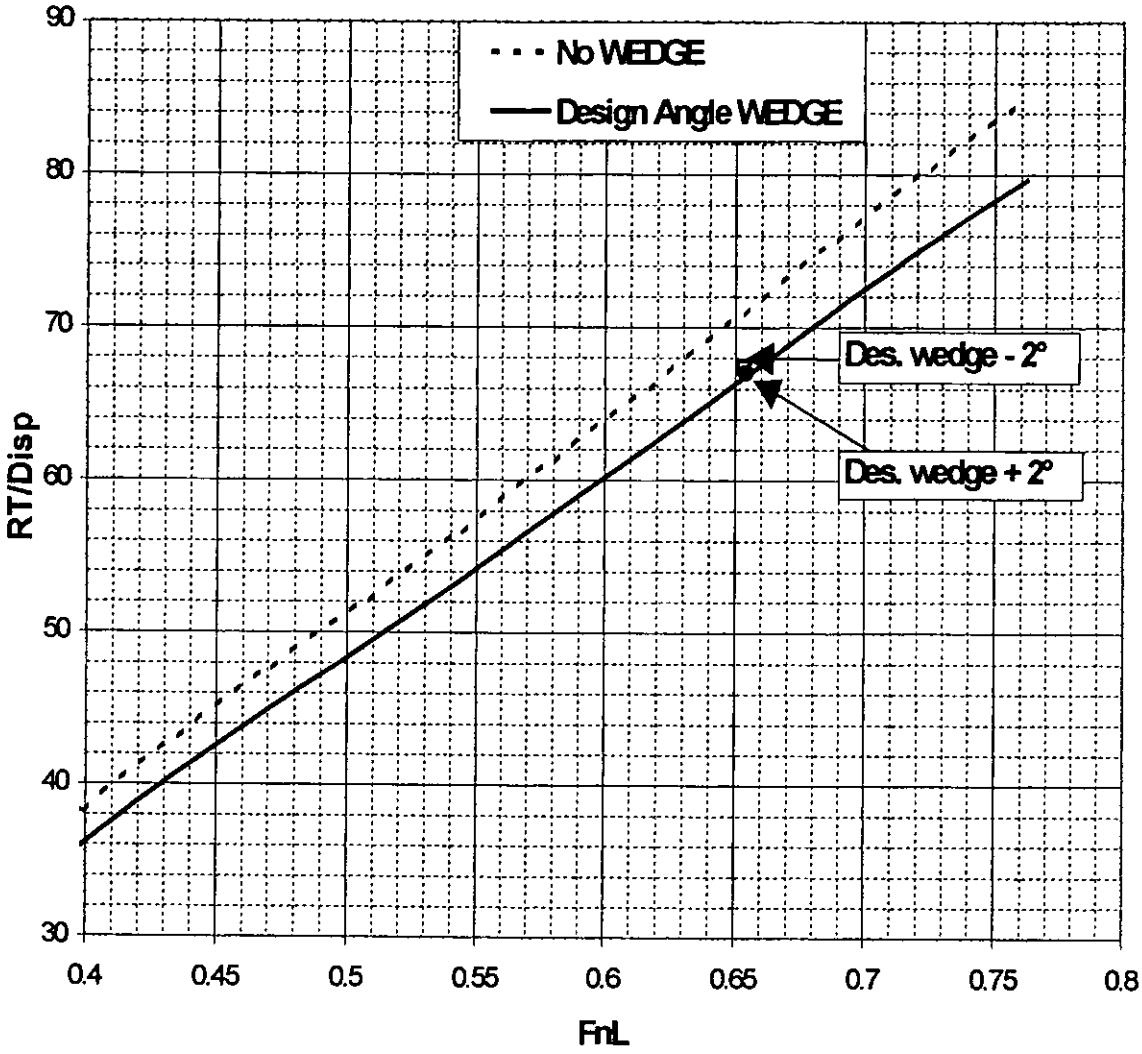


Figure i- Wedge influence on still water resistance for the MDV1200

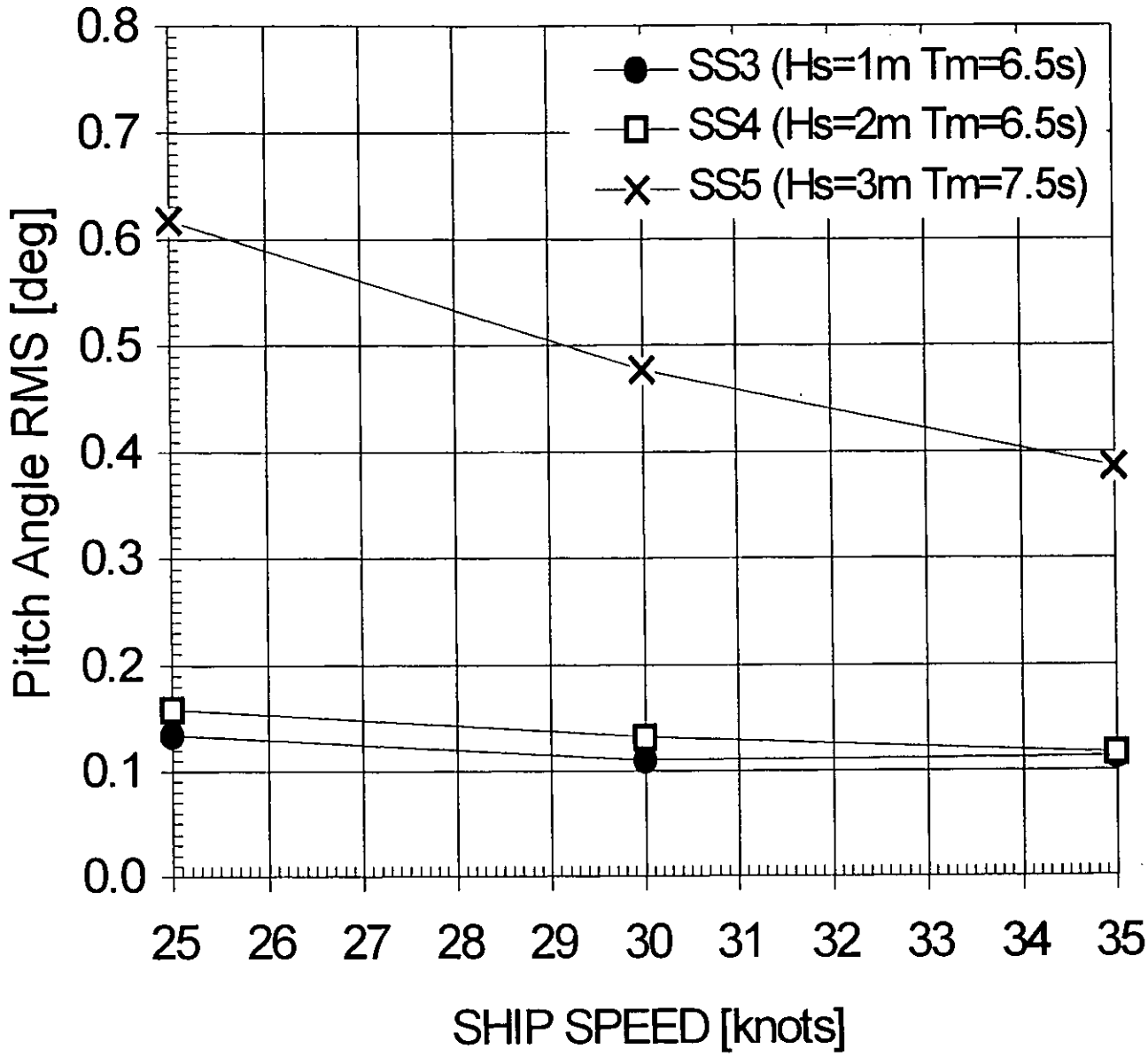


Figure j- Pitch motion t.t. tests for MDV1200LL

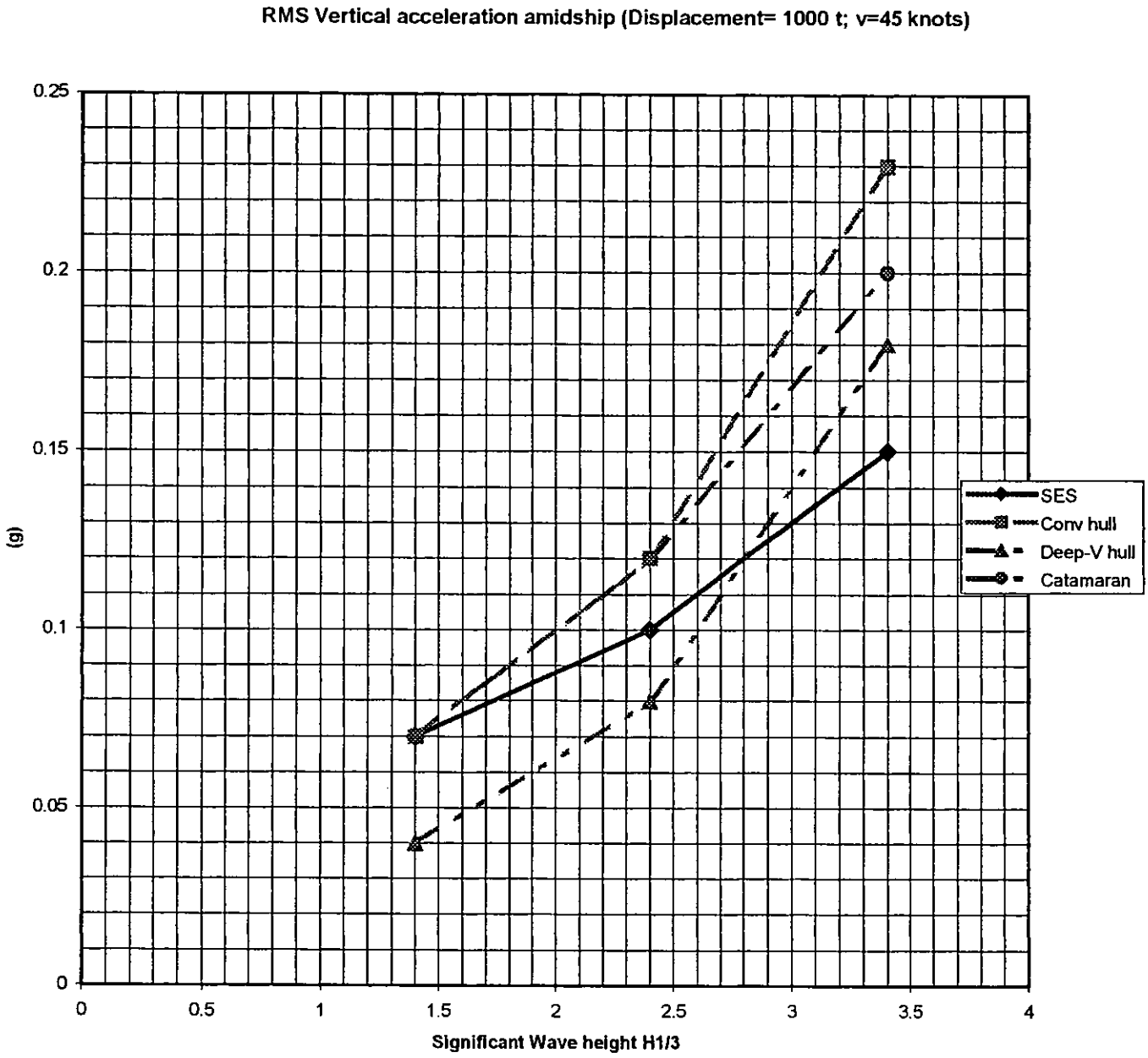

RMS Vertical acceleration amidship (Displacement= 1000 t; v=45 knots)
0.25
0.2
0.15
0.1
0.05
0
(g)
0
0.5
1
1.5
2
2.5
3
3.5
4
Significant Wave height H1/3
SES
Conv hull
Deep-V hull
Catamaran

Figure k

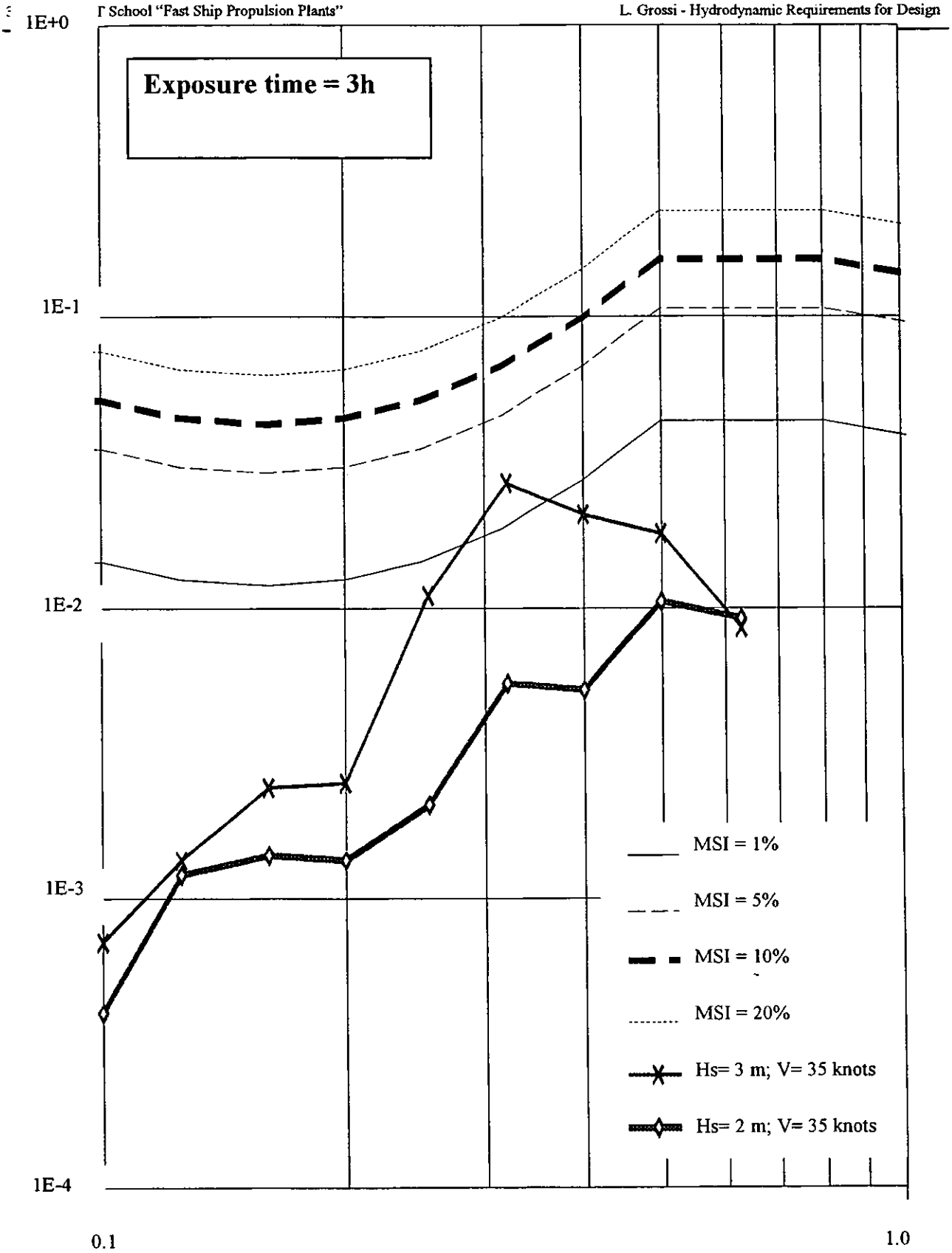


Figure l- MDV1200LL: Seasickness analysis at LCG, head sea SS4 ans SS5

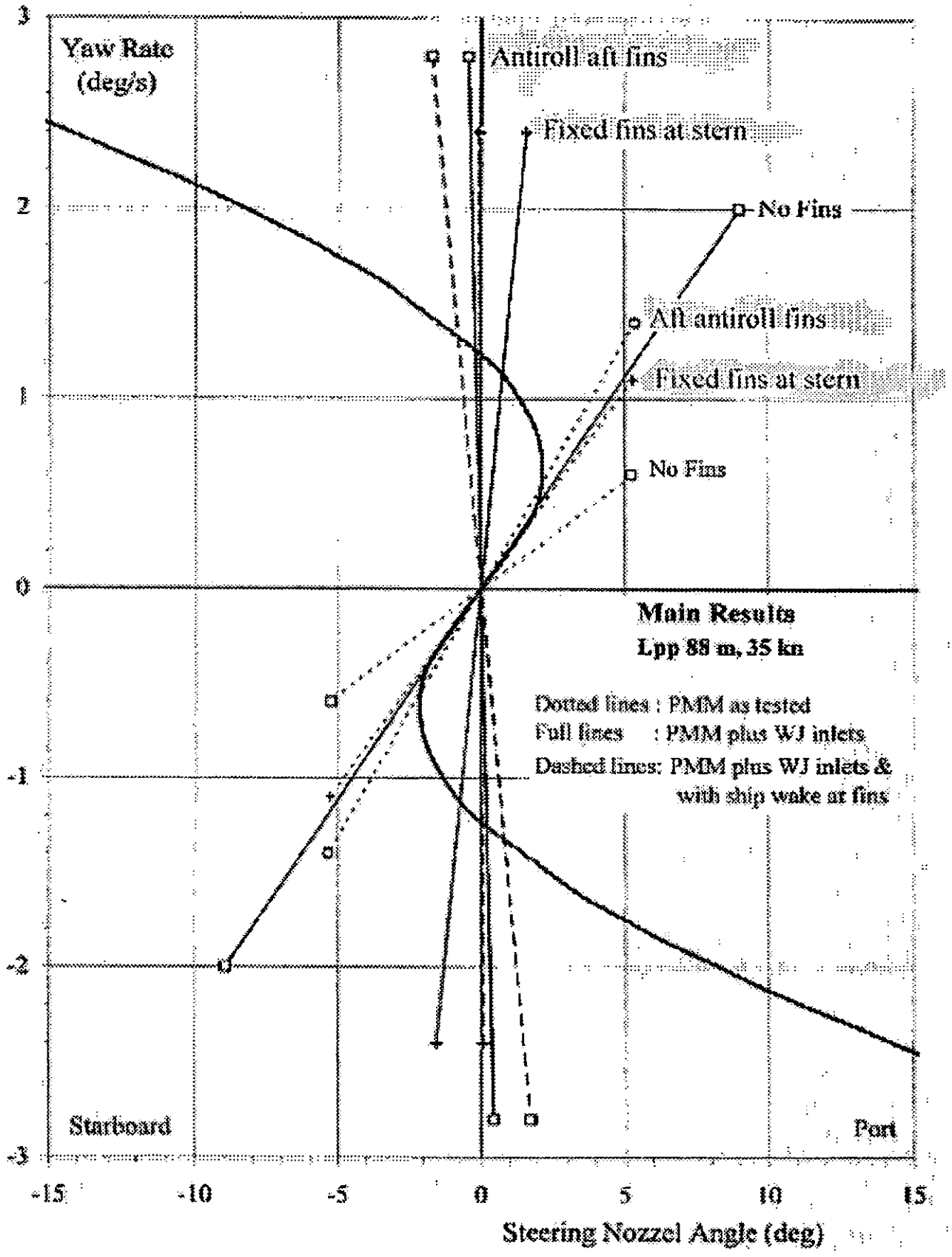


Figure m - Influence on the dynamic course stability of various appendages configurations

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WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
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PROPELLERS AND WATERJETS

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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PROPELLERS AND WATERJETS

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ABSTRACT

This lesson presents a review of existing propulsion systems, with special attention for high-speed advanced marine vehicles which are nowadays a market, and addresses the problem of how to select the best propulsion system for a specific ship configuration / mission.

INTRODUCTION

This lesson presents a review of existing propulsion systems, with special attention for high-speed advanced marine vehicles which are nowadays a market, and addresses the problem of how to select the best propulsion system for a specific ship configuration / mission.
Maximum speeds for this type of vehicles is, generally, more than 30 knots and displacement ranges from a few tons for planing craft to thousands of tons for the largest fast ferries.
Water-jets and marine propellers (and air propulsion systems for hovercrafts and SES) are actually considered the most appropriate propulsion devices. Other systems, such as magneto-hydrodynamic propulsion system, are still in phase of development and feasibility assessment.
Is important to note that in the definition of propulsion system is included the propeller, the mounting appendages, transmission gears, engine and propulsion fuel.

DESCRIPTION OF PROPULSION SYSTEMS

Water-jet systems

A water-jet propulsion system consists of engine, transmission, pump, nozzle, thrust reversing mechanisms, diffuser and ducting, inlet and appendages required for the mounting of the inlet.
Water-jet systems can be divided in two categories in the ground of inlet type used: pod-strut inlet and flush inlet (Fig. 1); either of this inlet types may be with fixed-area or variable-area in which case there is a control system for establishing the correct inlet opening as a function of operating conditions. The motivation for variable-area inlets is to provide the required flow at all the conditions with minimum inlet drag and with cavitation-free inlet operation; this is useful for vehicles operating at high speeds and requiring large thrust at both low and high forward speed.
Pod-strut inlets are usually characterized by the classic nacelle at the end of the strut with a circular inlet opening in the nose; in the hydrofoil craft the nacelle is generally at the strut-foil intersection.
For flush inlet an "inlet fairing", that is a modification to the hull ahead of the flush inlet, is required to reduce the dangers of ventilation or cavitation.
A single inlet may serve more than one pump; in this case split ducting will be required. In water-jet propulsion systems pumps may be axial flow, mixed flow or centrifugal.

Marine propeller systems

A propeller system consists of a fixed-pitch (FPP) or controllable reverse-pitch (CPP) propeller, propeller mounting appendage or fairing, transmission with reduction gears and one or more engines.
CPP propellers, which have slightly lower design point efficiency than FPP propellers, are often required for reversing thrust and for a better propeller-engine matching.
Propeller types include supercavitating and superventilated , transcavitating and partially submerged; subcavitating propellers are unsuited for high-speed because of blade cavitation erosion damage.
Supercavitating and superventilated propellers, which differ only in cavity gas content (water vapour, air, etc.), are suited to operation at all speeds above 40 knots; they have blade cavities which spring from the blade leading edge, fully cover the back side of the blade, and collapse downstream of the blade trailing edge. Superventilated condition is useful when it is necessary to reduce propeller-radiated noise.
Transcavitating propellers have modified blade sections to achieve subcavitating operation at low speeds.
Fully submerged propellers, which can operate either supercavitating or superventilated, if ventilating air is introduced, must have appropriate appendages to house the shafting; these appendages impose drag penalties which become quite severe at very high speeds. A possible solution to this problem may be the partially submerged propellers which are usually transom-mounted and have therefore low appendage drag; partially submerged propellers, which almost always are superventilated, have often vibration and strength problems arising from the cyclic loading and unloading of the blades. For twin-hull vessels, because of the comparatively narrower sidehull, a fairing ahead of the propeller may be required to avoid excessive partially submerged propeller force fluctuations.

Air propulsion systems

An air propulsion system consists of a propulsor, like fan or propeller, mounting appendages, transmission with reduction gears and engine.
The principal advantage of air propulsion systems over water-borne propulsors is freedom from cavitation due to the absence of submerged components, while the principal disadvantage is the exposure to spray impingement and salt-induced corrosion.
Air propulsor types for high-speed include free and shrouded air propellers, turbofans, cruise fan and Q-fan systems and centrifugal and axial-flow fan systems.
Free and shrouded air propellers, generally used for propelling hovercraft, have high potential efficiency, but unfeasible large diameters are often required to achieve efficiencies above 60 percent; shrouding usually permits the use of smaller propellers for a given efficiency. One important problem of air propellers is radiated noise.
Turbofans have some limits of application because of their high fuel consumption; as turbofan fuel consumption is nearly independent of ship speed (for fixed thrust), turbofans become more attractive as ship speed increases.
Cruise fan and Q-fan systems belong to a category of close-coupled turbojet, axial-flow fan systems and lie between turbofans and turbine-driven air propellers, representing a compromise between propulsor size and fuel consumption or efficiency; their radiated noise should be less than that of propeller or turbofan systems.
Centrifugal-flow and axial-flow fan systems, proposed primarily for SES and hovercraft, are analogous to water-jet propulsion systems, but use air instead of water. This type of propulsor system can have high efficiencies, but very large spaces are required for ducting and fans; radiated noise levels should be considerably lower than those with propeller or turbofan propulsion systems.

Magneto-hydrodynamic propulsion systems

A magneto-hydrodynamic propulsion system is based on the Lorentz principle according to which a point charge moving with constant speed in a magnetic field having induction is subject to a force (Lorentz force) with intensity proportional to the charge, the induction intensity and the motion velocity, perpendicular to the direction of the magnetic induction and to the direction of the charge motion.
If, in place of the point charge, one considers a rectilinear conductor of unitary length with a constant current intensity, the force acting on the conductor will be proportional to the magnetic induction and the current intensity.
In the marine field, for such type of propulsion, the conductor is represented by the sea water crossed by electric charges due to the polarising action of two electrodes immersed in the flow itself.
The magnetic field induced by stator coils consisting of superconductor fixed to the hull and the current generated in the sea, induce a Lorentz force directed along the ship's longitudinal axis which represents the propulsive thrust of the water jet through the appropriate duct (Fig. 2).
High density current will therefore be used for this type of propulsion and magnetic fields much more intense than the conventional ones will have to be realised. The negative aspect of the matter is the necessity to have big cryogenic plants to reach the required very low temperatures close to absolute zero.

The studies carried out till now have individuated the following effects:

- the thrust is generated directly in water;
- the thrust is directly proportional to the electric current in water and to the density of the magnetic flow.

The following advantages can be envisaged:

- noiseless and vibration-free propulsion system;
- easy operation and motion reversing;
- simple construction and no maintenance (excepting the cryogenic plant).

Ref. [9] is an interesting paper on magneto-hydrodynamic (MHD) pump-jet for ship propulsion, in which the solution of some important problems, such as the fabrication of better superconductors to generate very high magnetic fields, the development of a control system to insure efficient variable-speed operation, the development of an optimum cryostat to maintain the proper temperatures efficiently, is discussed.

PROPULSION SYSTEM DESIGN REQUIREMENTS AND CRITERIA FOR EVALUATING PROPULSION SYSTEM PERFORMANCE

The choice of propulsion system must be done taking into account specific design requirements such as range of operating speeds, corresponding drag and required thrust, endurance and limitations on propulsor or engine mounting locations. Every design imposes a maximum speed or maximum available power and the propulsor design will be the best compromise between conflicting requirements at maximum, cruise and hump speeds. There are, moreover, special requirements on the propulsion system, imposed by ship type due to the different geometries or performance requirements; great is, for example, the influence of ship geometry on suitable appendages and locations for mounting the propulsor.
For a twinhull, the two sidehulls provide a logical location for two water-jet inlets or two propellers, the size of which is closely related to sidehull beam; it may be difficult to efficiently use more than two propellers or two inlets. An hovercraft is best suited to air propulsion rather than to water-borne propulsion, because of its full flexible seals; deck space is generally available for mounting large air propulsion systems. For an hydrofoil ship, the pre-existence of struts provide a natural location for pod-strut mounted propellers and water-jet inlets, although propeller-strut-foil interactions can cause problems; hydrofoil ship hulls do not provide adequate deck area for air propulsion, particularly air propellers, because they are typically narrow to minimize hull impact loads. For planing craft all types of water-borne propulsors are attractive, particularly inclined shaft and transom mounted, partially submerged propellers and water-jet systems with flush inlets.
It is quite difficult to find a completely adequate criterion for evaluating and comparing the performance of all propulsion systems for all vehicle applications. We can say that the best criterion,

especially when endurance and useful load are important, is total propulsion system weight, including propulsion fuel weight, for a given endurance; unfortunately total system weight may be difficult to calculated accurately, in particular during preliminary design.
Another good criterion, particularly when endurance is not essential, is to consider the overall propulsive coefficient, opc, which is based on net thrust (propulsor thrust less added drag due to the propulsor) rather than the propulsor open-water efficiency.
In both cases we have to take into account that a ship can have different speeds operating conditions, so sometimes it is necessary a mean weighted evaluation over the ship operating profile to make considerations not only at a single speed . Other important factors in the choice of optimum propulsion system, in addition to propulsive performance, are economy, reliability and maintainability.

PERFORMANCE OF THE DESCRIBED PROPULSION SYSTEMS

Water-jet systems

In a water-jet propulsion system a great number of components are concerned, such as engine, transmission system, propulsion pump, nozzle, diffuser and ducting, inlet and appendages required for inlet mounting.
Some components, like engines and propulsion pumps, are complex and expensive so that the choice is very limited. Diffuser and ducting have usually configurations determined by the geometric requirements of the particular application. The water-jet inlet with its associated appendages is probably the most critical component of the water-jet system; it operates at ship speed, often in a nonuniform velocity field, and is easily susceptible to cavitation and ventilation.
We can say that the selection of the best water-jet system for a particular application is the best hydrodynamic, geometric and mechanical matching of available components.
Hereinafter the various components and their performances are described.

Propulsion appendages

Water-jet propulsion appendages are those structures or modifications to the basic craft geometry necessary for the installation of the water-jet inlet.
For strut-pod inlet, the strut section must be the best compromise of structural and hydrodynamic characteristics: it must give an adequate lateral strength and internal duct area and must permit cavitation-free operation at all operating speeds and yaw angles with minimum drag.
For flush inlet it is almost always necessary the inlet fairing, that is a modification of the sidehull geometry to permit inlet installation in a horizontal surface; it obviously must be designed to produce minimum drag, cavitation and ventilation.

Inlet

As we previously said either pod-strut or flush inlets may have fixed or variable area; variable area can be useful to have the correct inlet opening according to the different operating conditions, but generally it is not used to avoid complex mechanisms.

In the case of pod -strut inlets, water-jet efficiency is more influenced by inlet drag than by internal losses, so that pod-strut sizing is generally determined by the minimum size required to avoid both internal and external cavitation.
For flush inlets instead, the inlet drag is generally a small percentage of total ship drag.
The more important advantages of pod-strut inlets are: higher efficiency due to diffusing a more uniform flow and reduced probability of inlet ventilation in rough seas.
For flush inlets the more important advantages are: less added drag due to the inlet , less cavitation due to ship motions, simpler methods of inlet area change and less probability of inlet damage for collision with floating wreckages.

Diffuser

Diffuser-ducting is situated between inlet and pump, hence its geometry is mainly determined by inlet location and area and by pump location and characteristics.
While for flush inlet systems, diffuser geometry has little influence on external shape and drag, for pod-strut systems, diffuser geometry is closely correlated to external shape and drag.
Pod -strut diffusers ingest a very uniform flow so that they usually operate unstalled, that is without flow separation; in contrast, flush diffusers generally operate partially stalled at high speeds because they ingest a nonuniform velocity profile due to the hull boundary layer. The worse condition for danger of internal diffuser cavitation is hump condition because of the limited dynamic head and the higher ratio between average inlet velocity and ship speed.
Usually diffuser performances are valued through static pressure recovery and total energy recovery measurements.

Pump and nozzle

Water-jet pumps can be of the axial-flow, mixed-flow or centrifugal-flow type according to the design requirements. Generally with mixed- or centrifugal-flow pumps the water-jet system weight is reduced; however axial-flow pumps allow maximum compactness and best arrangements. Typical water -jet pumps have efficiencies of 0.88 to 0.90.
Water-jet nozzles have usually efficiencies of 98 percent. Sometimes variable-area nozzles, which are more complex and less efficient than fixed-area nozzles, are mounted to improve water-jet performance; these nozzles infact allow better pump-turbine matching and less momentum losses.

The design of an optimized water-jet propulsion system must have the purpose of reach the minimum propulsion system and fuel weight and the maximum propulsive efficiency, taking into account also material and structural problems.
Water-jet propulsive performance depends on the various components efficiencies, hence, as previously said, the overall propulsive coefficient based on net propulsor thrust is a better criterion than propulsor open-water efficiency.
Ref. [2] presents a chart (Fig. 3) that shows overall propulsive efficiency of vessels driven by KaMeWa water-jet; this chart is a summary of the results from a large number of trials with monohulls, catamarans and SES. As can be noted, the propulsive efficiency ranges from 50% at 15-20 knots and reaches 70% at about 40 knots. Installations with a high specific load at the speed in question, operating close to the recommended cavitation margins, usually follow the lower part of the dashed area; lightly loaded installations follow the upper part of the dashed area.

Marine propeller systems

Propeller system selection will be principally influenced by ship type and maximum speed but also propeller-appendages interactions, effect of lateral forces on ship control and stability and other factors must be considered. As well as for water-jet system, simplicity, reliability and cost are very important factors.
The best propeller system will have acceptable stresses, no vapour cavitation and will produce the required thrust at all speeds, with minimum system weight or maximum overall propulsive efficiency at the design speed.
Fig. 4 (ref. [1]) shows a table with a possible choice of appropriate types of propeller systems for typical ships; the indication of fixed or CPP propellers is based on performance and not on reverse thrust requirements.
As previously stated, marine propellers for high-speed craft and ships an be divided in a few categories.

Supercavitating and superventilated propellers

A supercavitating-type section under ventilated flow conditions can operate in three different flow regimes in function of the value of attack angle (ref. [4]): a base-vented regime, in which the cavity springs from the trailing edge of the section and the highest lift-to-drag ratio is reached; a partially cavitating regime, in which a vapour cavity of less than one chord length exists on the back side of the section; a fully vented region in which a cavity greater than one chord length exists on the back side starting from the leading edge and in which there is a drop in lift coefficient and in lift-to-drag ratio (Fig. 5 from ref. [4]).
Fig. 6 and Fig. 7 from ref. [1] show nondimensional performance curves of fully submerged supercavitating and superventilated propellers derived from model test data; superventilated propellers generally operate at lower cavitation numbers than supercavitating propellers and hence produce less thrust.
In ref. [3], the potentiality of supercavitating and superventilated propellers are discussed and, in particular, the effect of various design parameters on propeller performance is illustrated. Fig. 8 from this reference, shows the effect of thrust loading C_T and advance coefficient J on the efficiency of four-bladed, fully submerged propellers; it can be noted that the higher propeller efficiencies are obtained with low C_T, that is with large diameter, and high J, that is low rotative speed.
Ref. [5] analyzes the comparison between supercavitating and conventional propeller performance and discusses experimental performance of the supercavitating propeller during various operating conditions such as ahead, astern, crash back and crash ahead; Fig. 9 from this reference shows the practicability of supercavitating propellers for high-speed applications.

Partially submerged propellers

The principal advantage of the partially submerged propeller system is that it require little mounting appendages. One important disadvantage of partially submerged propellers is that they generate large transverse forces (with horizontal and vertical components) and thrust eccentricity; for this kind of problems heavier transmissions are usually required.

In ref. [4], the results of open-water experiments on three- and two-bladed supercavitating propellers operating in the partially submerged condition, have been compared with existing experiments made with propellers fully submerged both cavitating and noncavitating.
Fig. 10 and Fig. 11 from this reference, show some results of these comparisons; to make more significant the comparison between the performance of the propeller at different test conditions, the system of η_O and J versus C'_T, which is the thrust coefficient based on the submerged-disk area, have been used. As can be noted from these figures, the maximum efficiencies in the partially submerged condition are comparable to those in the fully wetted condition but the range of C'_T value over which a partially submerged propeller can operate efficiently is narrower than for fully submerged condition.
Ref. [6] presents the results of model scale tests of a limited series of surface piercing propellers tested at the high-speed free-surface cavitation tunnel of Technische Universitat Berlin. These results may be useful to estimate optimum diameters and associated pitch-diameter ratios of surface piercing propellers, for the design speed range $35 \leq V_S \leq 45$ knots; test results give also the magnitude of propeller vertical forces.
Ref.[7] is an interesting paper on surface piercing propellers, which discusses the past history, the recent successful applications and the possible future development of such propellers.

Inclined shaft and pod-strut systems.

Inclined shaft systems needs mounting appendages, such as shaft fairing to avoid excessive drag, shaft support strut and propeller hub fairing; the drag of these appendages with the nonuniform propeller inflow due to inclination and with shaft fairing wake, can cause a reduction of 10 to 20 percent in overall propulsive efficiency. In these systems the inclination of the shaft (typically of 8 to 12 deg) causes load variations and the shafting wake can generally cause a local increase in blade loading.
A pod-strut system must be the best compromise between mechanical and hydrodynamic requirements, in fact it must house the shaft, operate vapour cavitation free and have minimum drag.
One important problem of this kind of systems is propeller-pod-strut interaction, either in the case of pusher pod (propeller aft) or in that of tractor pod (propeller forward). If the propeller is pusher, the strut operates in a uniform inflow but its wake can cause a reduction in propeller efficiency because of unsteady propeller forces; a tractor propeller has usually higher efficiencies due to its uniform inflow but the propeller wake can produce significant lateral forces and induced drag on the strut; for this reason pusher propeller system is on the whole more efficient than tractor propeller system.

Air propulsion systems

As we previously said, air propulsion systems are usually used for hovercraft, because of their full flexible seals, and it is the only possible choice when these vehicles must operate also over land or ice.
One important problem of air propulsors is the airborne noise, that can reach near field levels of 100 dB for propellers and fans and that can be partially reduced only by decreasing blade tip speeds and by shrouding.

Free and shrouded air propellers can usually reach efficiencies not more than 65 percent, because of limitations on allowable diameter, blade tip speed, blade strength etc; as noted earlier, shrouded propeller efficiencies are higher than those of free propellers with equal diameter.
Turbofan systems may be very attractive, because of their light weight and compactness, for applications where endurance and airborne noise are not of prime concern.
Also cruise fan and Q-fan systems have low weight and compactness and they generally have less fuel consumption and radiated noise than turbofans.
Centrifugal-flow and axial flow fan systems are analogous to water-jet propulsion systems, except for the absence of cavitation because they use air instead of water; as previously said to reach high efficiencies very large spaces for ducting and fans are required.

OVERALL PROPULSIVE EFFICIENCY AND HYDROACOUSTIC NOISE - COMPARISON OF SEVERAL PROPULSION SYSTEMS

As previoulsy said, the overall propulsive efficiency is a good criterion to compare the performances of different propulsion systems.
To estimate the overall propulsive efficiency for water-borne propulsors, it is important to know the correct value of some propulsive coefficients such as thrust wake fraction w_T, thrust deduction fraction t, relative rotative efficiency; Fig.12 from ref. [8] presents a table in which bands of typical values of propulsive coefficients for high speed craft are shown in function of dimensionless speed and of propulsor type.
The author of ref. [8], moreover, presents a very interesting chart (Fig. 13) in which he establishes reasonable bandwidths of achievable overall propulsive coefficients for flush inlet water-jets, surface piercing propellers and submerged propellers, derived from his experience of instrumented full-scale trials. As can be noted from this figure, from the only point of view of propulsive performance, submerged propellers are the best choice for a range of speeds $\geq$ 25 knots and water-jets the best choice for a range of speeds $\geq$ 40 knots; for the intermediate range of speeds the water-jet system could be still a good choice.
One important factor that can influence the selection of propulsion system, especially for naval vessels, is the hydro-acoustic noise. Fig. 14 from ref. [2] presents a comparison of hydro-acoustic noise for three different navy craft with three different propulsion systems, based on full scale measurements; as can be noted the lower levels of hydro-acoustic noise correspond to water-jet propulsion system.
In addition to overall propulsive efficiency and hydro-acoustic noise, however, others important factors, such as light weight, cost, maintainability and reliability, must be considered to select optimum propulsion system.

CONCLUSIONS

As a conclusion of this discussion we can say that a great deal of experience is necessary in the selection and design of high-speed propulsion systems and that model tests may be very useful to verify performance predictions.
For all types of high-speed propulsion systems, machinery components are similar: for example, engines must be selected from a limited number of marinized gas turbines or lightweight diesels; lightweight diesels may be very competitive with gas turbines at low powers, especially when endurance is of prime concern.
In the case of fast ferries a water-jet propulsion system with flush inlets seems a logical choice. Partially submerged propeller systems, which at high speeds are more efficient than fully submerged propeller systems, offer good performances and low system weight but they have difficulty in providing the large thrusts required and, moreover, they produce large transversal forces which generally cause problems of ship control and manoeuvrability.
On the contrary, a water-jet propulsion system can offer some additional advantages such as: steering, reversing and good manoeuvrability over the whole speed range; protected installation and shallow draught; lower hydroacoustic and internal noise and vibration levels compared to propeller propulsion; good fuel economy.

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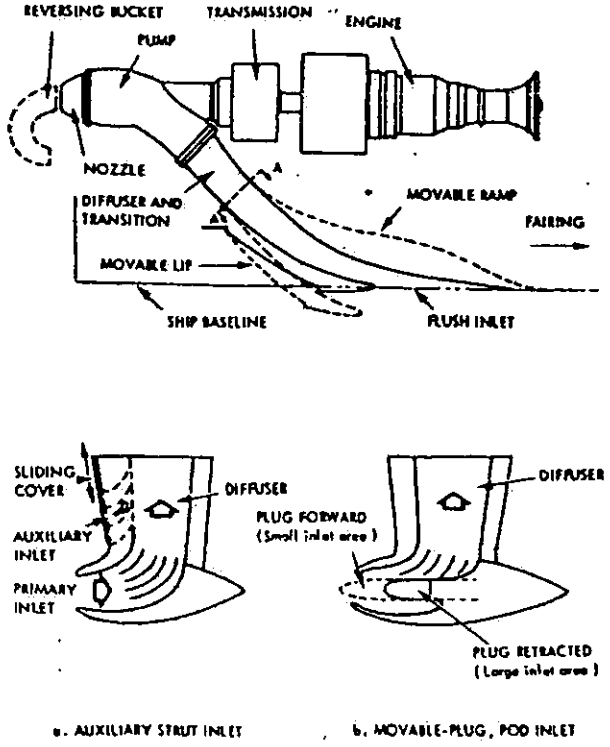


Fig. 1. Typical water-jet system - flush-inlet and pod-strut inlet schemes

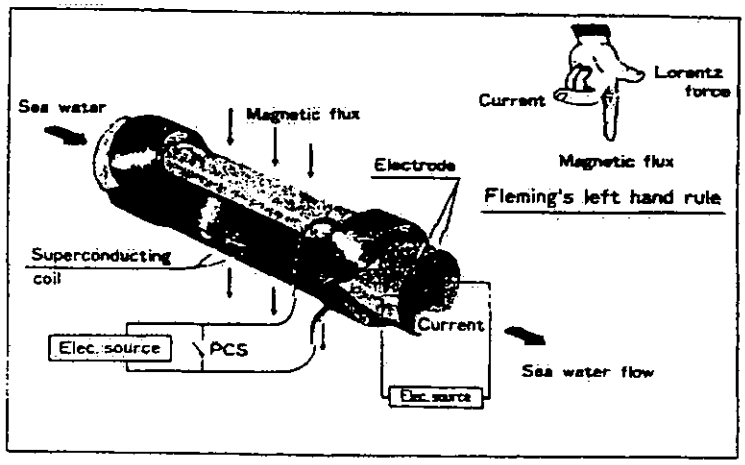
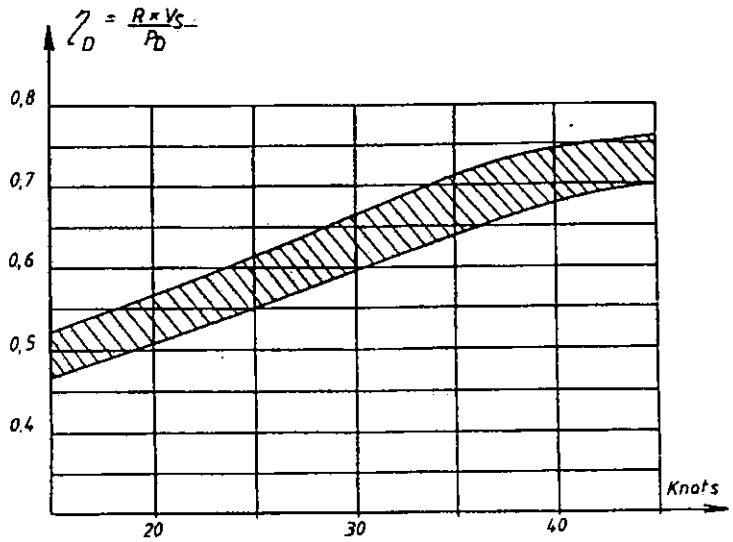


Fig. 2. Magneto-hydrodynamic propulsion system



R = Hull drag based on model tests at a displacement including water entrained in jet units

V_S = Ship speed

P_D = Input power to pumpshafts measured during trials

Based on trials with monohulls, catamarans and SESs in the displacement range 85-400 t

Power range up to 14000 HP/shaft

Fig. 3. Water-jet overall propulsive efficiency

Ship Type	Maximum Speed-knots	Mounting Type			Pitch	
		Inclined Shaft	Pod-Strut Submerged	Partially Submerged	Fixed	CRP
Planing	< 60	x		x	x	
Hydrofoil	< 60		x		x	x
Hydrofoil	> 60		x			x
SES	< 60	x	x	x	x	x
SES	> 60			x		x

Fig. 4. Suitable marine propeller systems for several ship type

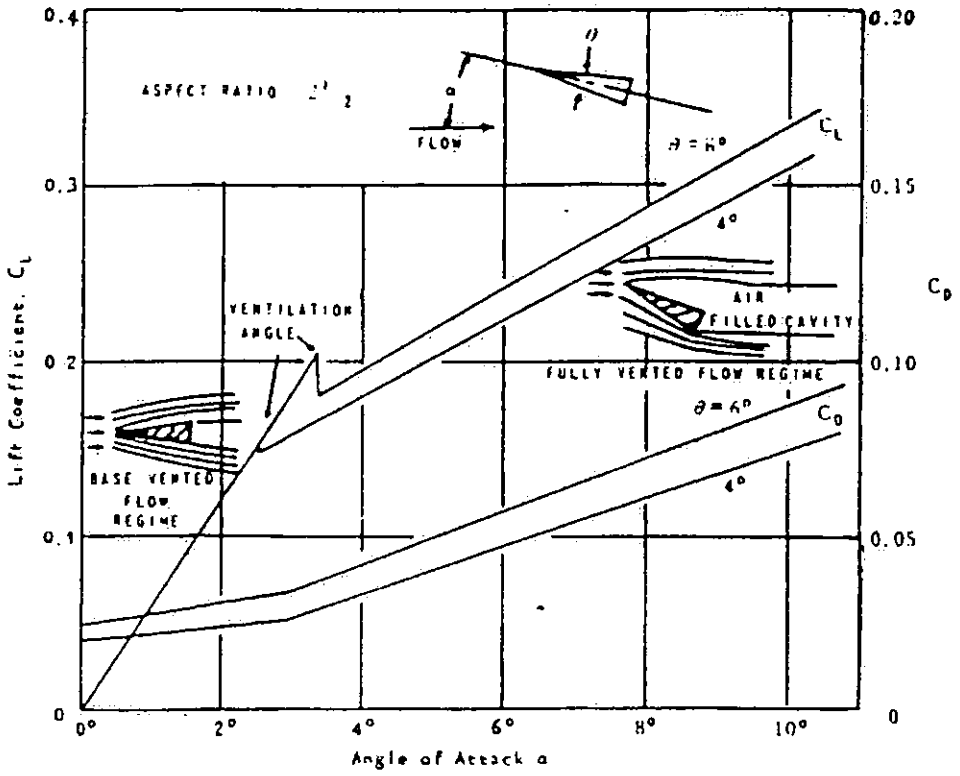


Fig. 5. Lift and drag coefficient of a supercavitating-type section in various flow regimes

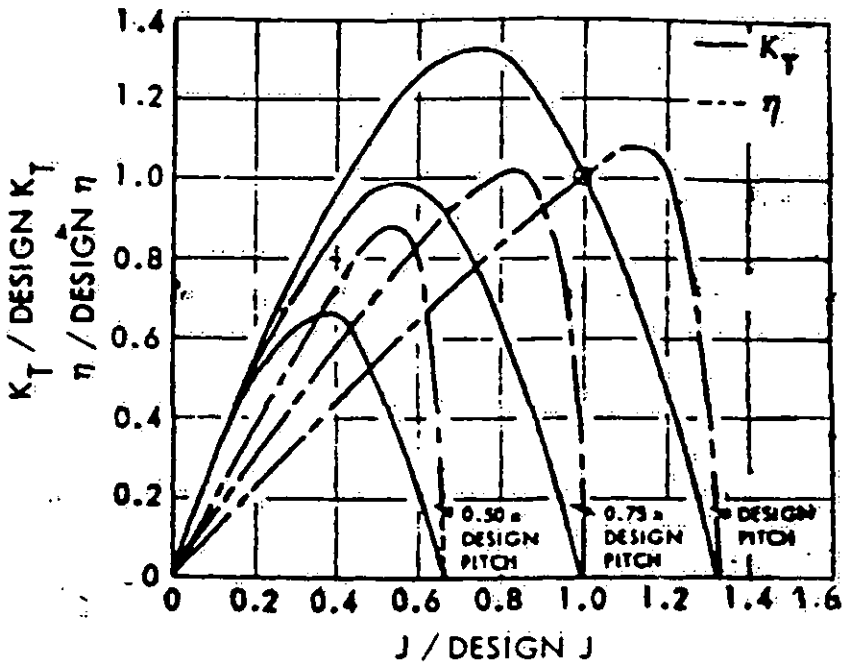


Fig. 6. Non-dimensional performance curves for supercavitating propellers

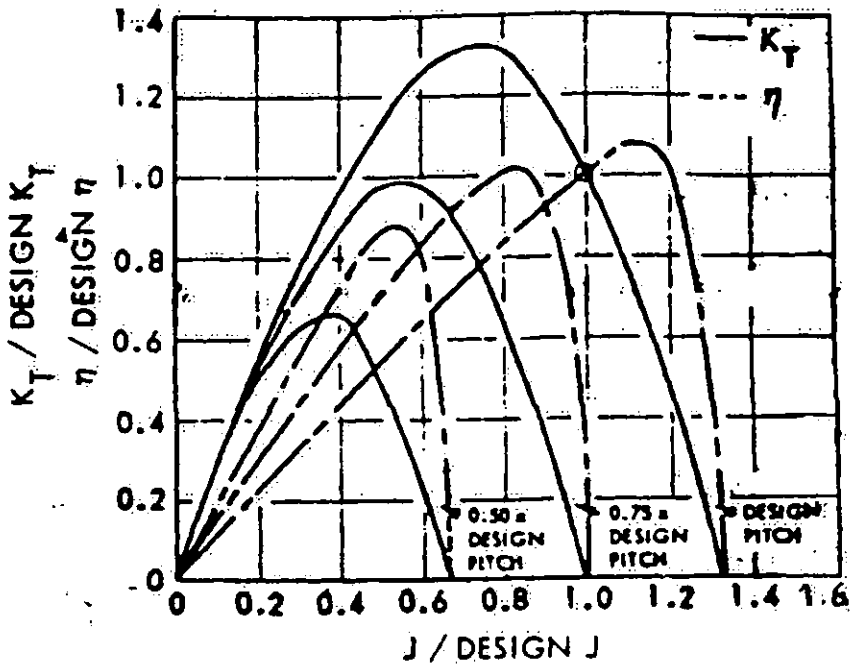


Fig. 7. Non-dimensional performance curves for superventilated propellers

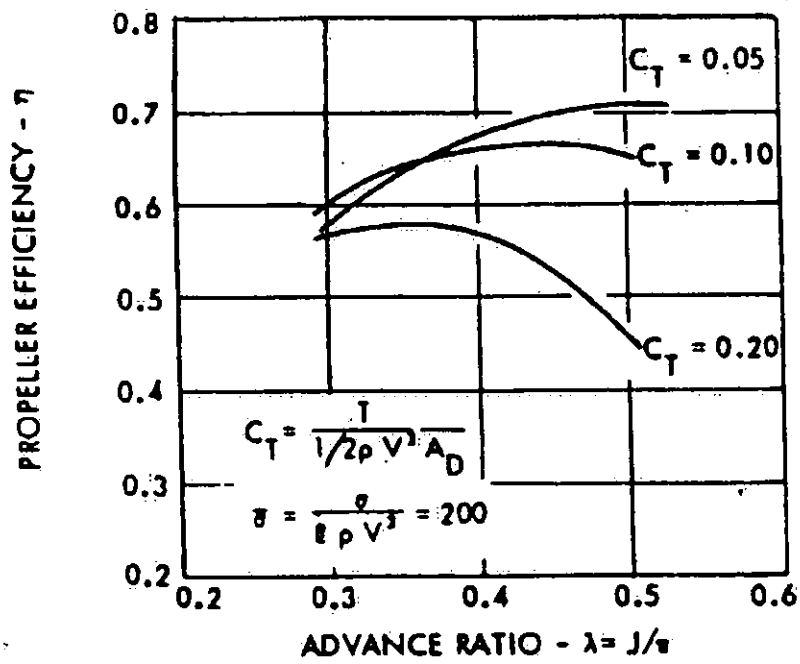


Fig. 8. Calculated efficiency of four-bladed, fully submerged, supercavitating propellers

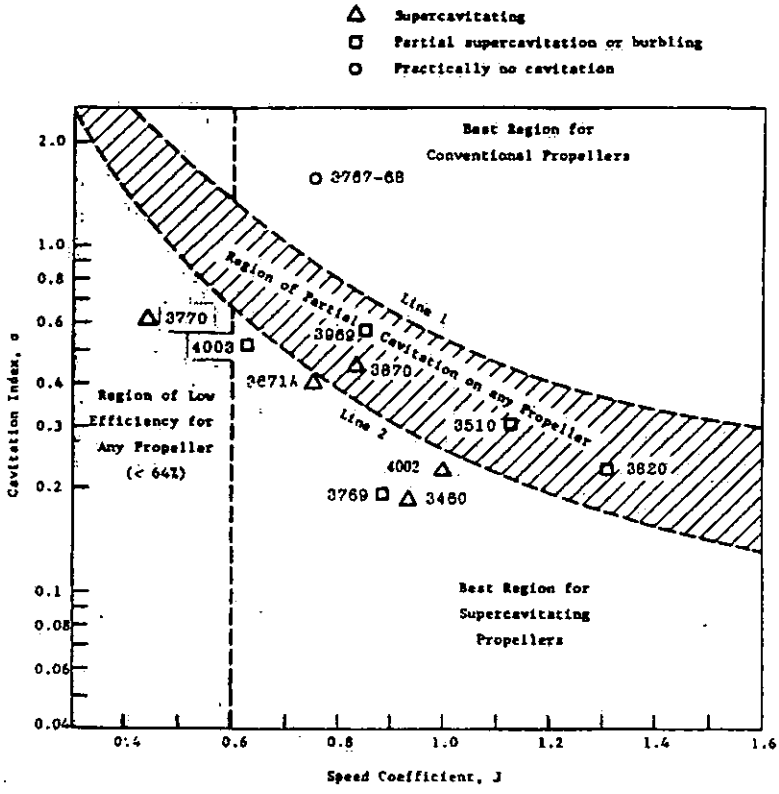


Fig. 9. Practicability of supercavitating propellers

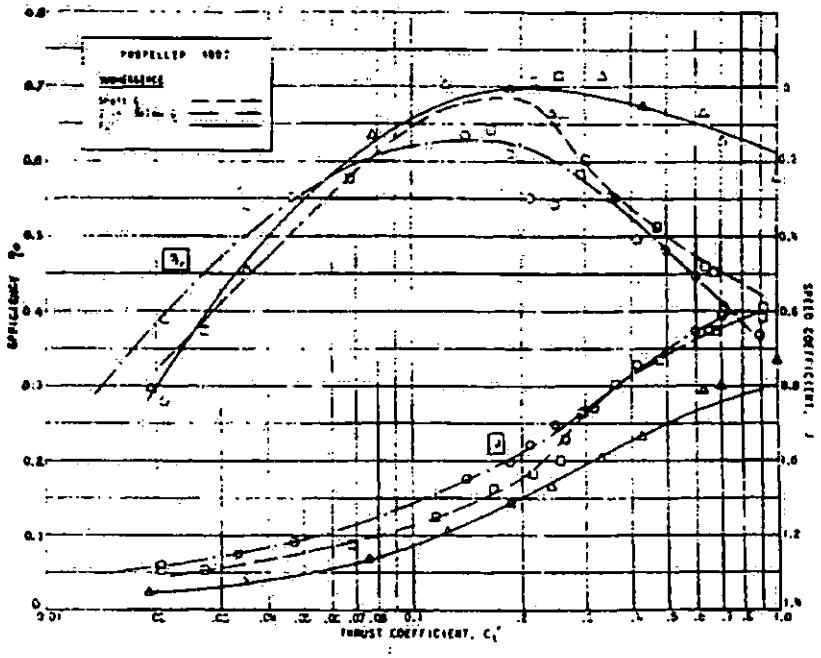


Fig. 10. C_T' - η_O - J diagram of propeller 4002 partially submerged

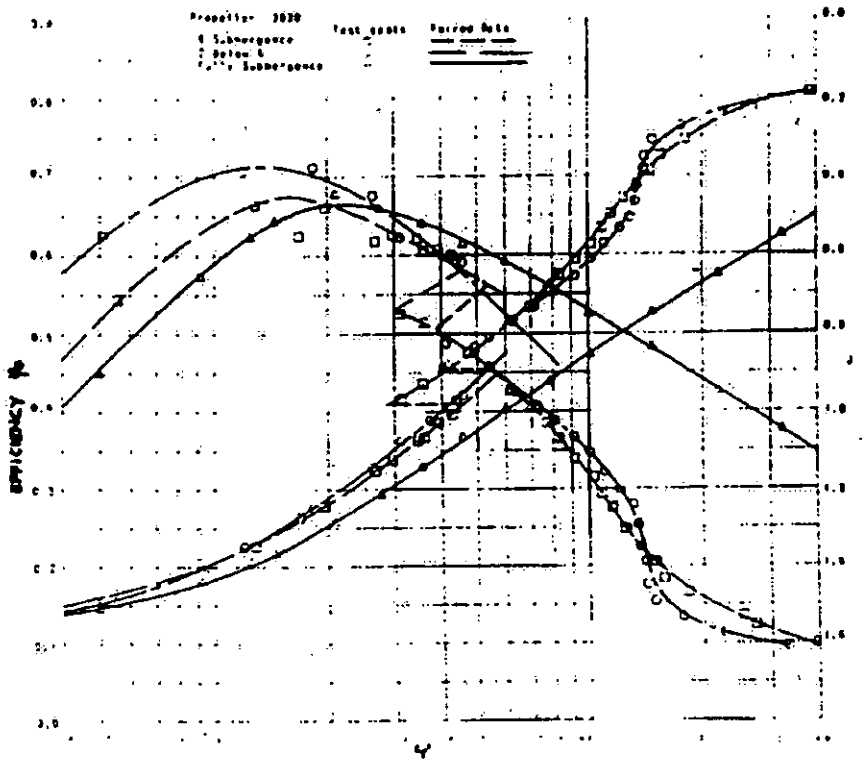


Fig. 11. C_T' - η_O - J diagram of propeller 3820 partially submerged

PROPULSION CONCEPT		SPEED RANGE: DISPLACEMENT $F_{\nabla}<1$			SEMI-PLANING $1 \le F_{\nabla} \le 2.5$			PLANING $F_{\nabla}>2.5$		
		ψ_T	t	η_R	ψ_T	t	η_R	ψ_T	t	η_R
PROPELLER ON INCLINED SHAFT	6 DEG SHAFT	0.01 TO -0.02	0.01	0.97 TO 1.01	0 TO 0.04	0.01 TO 0.02	0.97 TO 1.01	0 TO -0.10	0.03	0.97 TO 1.01
	12 DEG SHAFT	0.02 TO -0.02	0.05	0.97 TO 1.01	0.04 TO -0.05	0.05 TO 0.07	0.97 TO 1.01	0.03 TO -0.05	0.07 TO 0.11	0.97 TO 1.01
PROPELLER IN TUNNEL	40% D	-0.03	0.10	0.92	0.02 TO -0.03	0.07 TO 0.10	0.93 TO 0.90	0.03	0.03 TO 0.07	0.88 TO 0.90
	65% D	-0.03	0.12	0.92	0 TO 0.05	0.10 TO 0.12	0.93 TO 0.90	0.04 TO 0.05	0.08 TO 0.10	0.88 TO 0.90
OUTBOARD & OUTDRIVE PROPELLER		0.03	0	0.97 TO 1.01	0.03	0	0.97 TO 1.01	0.03	0	0.97 TO 1.01
PARTIALLY SUBMERGED PROPELLER		0	0	0.97 TO 1.01	0	0	0.97 TO 0.98	0	0	0.97 TO 1.01
FLUSH INLET WATERJET		0 TO 0.02	0.05 TO 0.08		0.02 TO 0.04	0.05		0.05	-0.02 TO -0.07	
TRACTOR PROPELLER		0	0 TO 0.05	1.00	0	0 TO 0.05	1.00	0	0 TO 0.05	1.00
PUSHER PROPELLER (UNDER HULL)		0.05 TO 0.07	0.05 TO 0.07	0.97 TO 1.01	0.05 TO 0.07	0.05 TO 0.07	0.97 TO 1.01	0.05 TO 0.07	0.05	0.97 TO 1.01

Fig. 12. Typical propulsive coefficients for high speed craft

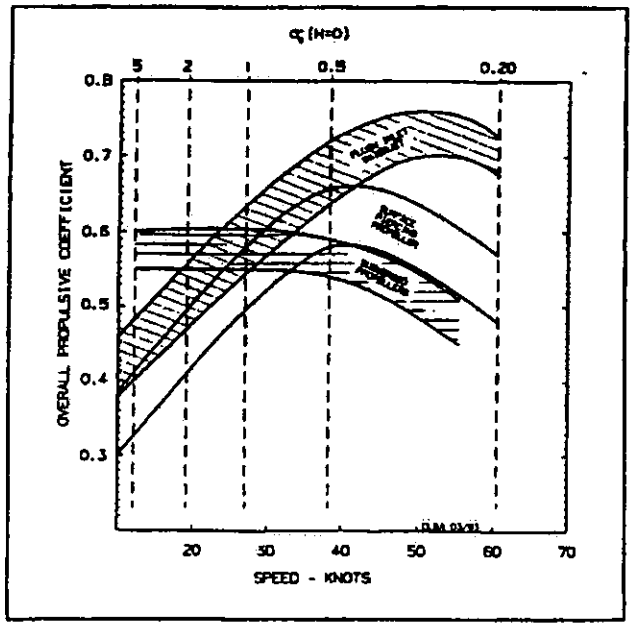


Fig. 13. Achievable overall propulsive coefficients for several propulsors

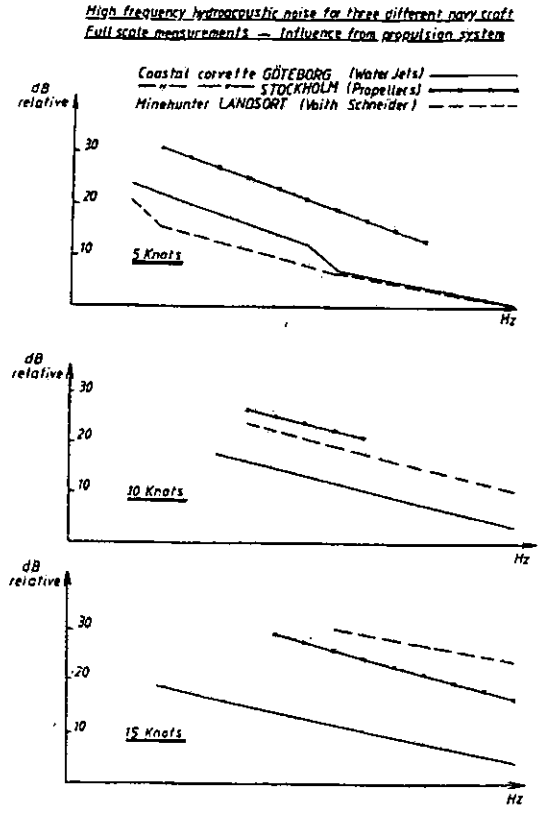


Fig. 14. High frequency hydroacoustic noise for three different navy craft

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Fast Ship Propulsion Plants
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DIESEL ENGINES FOR FAST SHIPS

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Abstract

After giving a short overview of the position of the diesel engine in the marine market the working principle of the turbocharged engine is explained briefly as well as the importance of the mean effective pressure as a main performance parameter. Then follows a discussion of the other main parameters that govern power density as well as absolute power from a given engine specification. The overall efficiency is formally introduced and split into a thermodynamic part and other losses. A basic thermodynamic analysis shows the effect of turbocharging on the ideal efficiency and highlights the importance of high peak pressures. A more practical analysis reveals that the mechanical and heat losses have a tendency to become relatively less important when increasing the charge pressure. Two-stage turbocharging is briefly touched upon. The severe limitations of the power/speed envelope of a highly turbocharged diesel engine are explained in some detail as well as other part load peculiarities. Sequential turbocharging is discussed as a means to broaden the power/speed envelope of a highly turbocharged diesel engine. Last but not least practical issues such as type of fuel and noise & vibration are briefly addressed.

Contents

ABSTRACT 1
EDUCATIONAL GOALS 4
NOMENCLATURE 5
Variables 5
Greek variables 6

DIESEL ENGINES 7

1 INTRODUCTION 7
2 MEAN EFFECTIVE PRESSURE 8
3 POWER DENSITY 10
3.1 Specific power related to stroke volume 10
3.2 Specific power related to bore area 11
3.3 Maximum power for a given specification 12
4 EFFICIENCY 14
5 THERMODYNAMIC ANALYSIS OF THE IDEAL SEILIGER PROCESS 15
5.1 Seiliger cycle definition 15
5.2 Heat and work 17
5.3 Specific input heat and thermodynamic efficiency 18
5.4 Specific work and mean indicated pressure 19
5.5 Constraints 20
5.6 Trend in mean indicated pressure and efficiency for ideal cycle 21
6 CYCLE WITH LOSSES AND OPTIMISATION OF PERFORMANCE 22
7 TWO-STAGE TURBOCHARGING 24
8 OFF-DESIGN CHARACTERISTICS OF A TURBOCHARGED DIESEL ENGINE 25
8.1 Power/speed characteristic 25
8.2 Off-design for Constant torque/Propeller law/Constant speed 26
8.3 Methods to broaden the engine characteristic 27
9 FUELS 27
10 INSTALLATION ON BOARD 28
SOME DIESEL ENGINE DATA 29
REFERENCES AND OTHER LITERATURE 30
FIGURES 31

Educational goals

The aim of this module on Diesel Engines within the WEGEMT school for Fast Ship Propulsion Plants is that the participants will be able to:

- explain the working principles of the modern turbocharged diesel engine,
- understand the most important parameters that make diesel engines light and compact, i.e. the factors determining power density,
- understand how to obtain reasonable efficiency for these light and compact engines, i.e. the factors determining fuel economy,
- explain the limits of the engine characteristics in relation with the characteristic of the propulsor,
- describe the features that can widen the engine characteristic,
- describe some special topics relating to the installation of diesel engines on board ships.

Last but not least this module will:

- provide some factual information on particular engines available in the market

Nomenclature

Variables

Symbol	Name	Unity
A_B	Bore Area	m^2
a	pressure ratio combustion (2-3)	-
afr	air/fuel ratio	-
BASP	Bore Area Specific Power	W/m^2
b	expansion ratio combustion (3-4)	-
c	expansion ratio combustion (4-5)	-
c_m	mean piston speed	m/s
c_p	specific heat at constant pressure	J/kg/K
c_v	specific heat at constant volume	J/kg/K
D_B	Bore diameter	m
f	(engine) frequency	Hz
h	specific enthalpy	J/kg
i	NR of cylinders	-
k	NR of revolutions per cycle	-
L_S	Stroke length	m
LHV	Lower Heat Value (of fuel)	J/kg
M	Torque	Nm
M_B	Shaft (brake) torque	Nm
m	mass	kg
m_a	air mass	kg
m_f	amount of injected fuel	kg
$\dot{m}$	mass flow	kg/s
$\dot{m}_a$	air mass flow	kg/s
$\dot{m}_f$	fuel mass flow	kg/s
$\dot{m}_g$	exhaust gas mass flow	kg/s
n	rotational speed	1/s
P	Power	W
P_B	Shaft (brake) power	W
p	pressure	Pa
$\overline{p}$	mean pressure	Pa
p_c	inlet receiver pressure	Pa
p_{me}	mean effective pressure	Pa
p_{mi}	mean indicated pressure	Pa
Q	Heat	J
Q_f	Fuel (potential) heat	J
Q_{comb}	(Actual) combustion heat	J
Q_i	Input heat	J
q	heat per unit mass	W/kg
R	Gas constant	J/kg/K
R_a	Gas constant of air	J/kg/K

r_c	Effective compression ratio (1-2)	-
r_e	Effective expansion ratio (5-6)	-
S	Entropy	J/K
SP	Specific Power	(k)J/kg
SVSP	Stroke Volume Specific Power	W/m^3
s	specific entropy	J/kg/K
sfc	specific fuel consumption	kg/Ws (g/kWh)
T	Temperature	K
T_c	inlet receiver temperature	K
T_{max}	maximum (process) temperature	K
V	volume	m^3
V_{BDC}	Volume at Bottom Dead Centre	m^3
V_S	Stroke volume of one cylinder	m^3
V_{TDC}	Volume at Top Dead Centre	m^3
$\dot{V}$	Volume flow	m^3/s
v	specific volume	m^3/kg
W	Work	J
W_e	effective work	J
W_i	indicated work	J
w	specific work	J/kg

Greek variables

Symbol	Name	Unity
Δh	specific enthalpy difference	kJ/kg
ε	geometric compression ratio	-
η	efficiency	-
η_{comb}	combustion efficiency	
η_m	mechanical effectiveness	-
η_q	input heat efficiency	-
η_{scav}	scavenge efficiency	-
η_{td}	thermodynamic efficiency	-
η_{tot}	overall (total) efficiency	-
η_{trap}	trapped air efficiency	-
θ	non-dimensional heat flow	-
φ	volume ratio	-
κ	isentropic index	-
λ	air excess ratio	-
λ_S	Stroke/Bore ratio	-
π	pressure ratio	-
ρ	density	kg/m^3
σ	stoichiometric ratio of fuel	-
τ	temperature ratio	-
ω	angular velocity	rad/s

DIESEL ENGINES

1 Introduction

The diesel engine is much older than the gas turbine and in the days of steam found its application in ships where the required power was modest. This of course was before the advent of turbocharging. The development of turbocharging during the last 30 years has resulted in a power increase for a given cylinder volume in the order of magnitude of two to three.

Therefore it is now possible to even power the largest ships with diesel engines (which may have an output of 30 to 70 MW, a figure that is growing still, particularly for containerships). Of course then we are talking about the large, low speed (<250 rpm), 2-stroke diesel engines that can only be installed in relatively big ships.

For smaller vessels the medium speed, 4-stroke diesel engine (with rotational speeds between 400 and 800 rpm) are popular. The power of these engines is not only lower in an absolute sense (maximum is about 18 MW) but also in a relative sense, i.e. the power *density* is higher resulting in low weight and volume for a required power.

For small and fast ships (such as fast ferries, fast patrol boats and frigates) the high speed, 4-stroke diesel engine (rotational speed >1000 rpm) offers a compact solution up to 7 or 8 MW per engine. Figure 1 gives a 3-D cur-away view of a representative engine of that class.

The principle of turbocharging as it is now invariably used on marine diesel engines can be seen in Figure 2. In a compressor the charge air is compressed to a pressure that is 2 to 4 times (or even higher) the ambient pressure. To further increase density the air then is cooled as far as possible to ambient temperature again. The bigger mass of charge air in the cylinders allows more fuel to be injected and the power output increases accordingly. The compressor is driven by a turbine that is located in the hot gas stream of the cylinder exhaust. The combination of compressor and turbine is called the turbocharger which in fact is a small, waste heat driven gas turbine.

The efficiency of a diesel engine is much better than for a steam plant and also when compared to a gas turbine. This is true for the design point but even more so for part load. The fundamental cause for this high efficiency of diesel engines is the intermittent character of the combustion which allows high peak temperatures in the cylinder without causing an extreme thermal loading of the surrounding materials. A disadvantage of course is that the harmful nitrogen oxide (NO_x) also easily forms at high temperatures.

If the load characteristic dictates that speed drops with power (as is the case for a propeller or waterjet) then any diesel engine cannot develop full power anymore but only maximum torque and turbocharged diesel engines cannot do even that. This means that the margins necessary for acceleration and a heavier load line (which will occur for instance in heavy sea state) are rather

small. Recent developments (sequential switching of turbochargers, variable geometry) aim at widening the engine power/speed characteristic. Of course the problem also can be solved by reducing pitch with a controllable pitch propeller but then there are limitations in view of propulsion efficiency and underwater noise (naval ships!).

As for any other machine involved in energy conversion power density and fuel economy are important issues for a diesel engine. A quantity specifying power density of the diesel engine is the mean effective pressure. To fully grasp this and other quantities some basic diesel engine theory must follow now.

2 Mean effective pressure

A diesel cylinder process can be depicted in a diagram plotting the indicated pressure versus volume (see figure 3).

We will start by calculating the indicated work (in kJ/stroke) in the cylinder. Assuming a reversible cycle the indicated work is equal to the enclosed area in the p-V diagram. Mathematically it is possible to define a mean value as the pressure integral over volume divided over the total displaced volume over that pressure trace. For a diesel engine normally the cylinder swept volume (i.e. the nominal volume of one cylinder as it is displaced during the expansion stroke) is taken as a reference:

$$W_i \overset{rev}{=} \oint_{cycle} p \cdot dV = \overline{p} \cdot V_s \qquad ===> \qquad \overline{p} \overset{def}{=} \frac{\oint_{cycle} p \cdot dV}{V_s}$$

Note

Note that a pure mathematical definition of mean pressure for a 2-stroke and 4-stroke engine would use two respectively four times the stroke volume.

The mean indicated pressure of a piston engine in general is taken as the mean value defined above:

$$p_{mi} \overset{def}{=} \frac{W_i}{V_s}$$

Index 'm' stands for "mean effective" while index 'i' signifies "indicated" (i.e. as measured with a pressure indicator on the cylinder).

Note

Sometimes 'imep' is used which stands for "indicated mean effective pressure". Also 'MIP' can be encountered meaning "Mean Indicated Pressure".

Between the work exerted on top of the piston and the work delivered to the output flange of the engine there are mechanical losses which may not be neglected such as friction between piston and wall, friction in bearings, power take off for lube oil pump, cooling water pumps as well as the fuel pumps, mechanical drive of the cylinder valves etc. All this is put into a mechanical efficiency:

$$\eta_m \overset{def}{=} \frac{W_e}{W_i}$$

Refer to Figure 4.

With the effective work per stroke delivered at the output flange a mean effective pressure can be defined in an analogous way as the mean indicated work:

$$p_{me} \overset{def}{=} \frac{W_e}{V_S}$$

Note

Sometimes 'bmep' is used which stands for "brake mean effective pressure". Also 'MEP' can be encountered meaning "Mean Effective Pressure". Note that "effective" is used in two different ways!

It will be clear that the mechanical efficiency also is the ratio between the mean indicated and mean effective pressure:

$$p_{me} = \eta_m \cdot p_{mi}$$

The relation between "work per stroke" and "power" is easily understood if one counts the number of work strokes the engine produces per unit time (i.e. per second). If 'n' is the rotational speed of the engine (in rotations per second), 'i' the number of cylinders and 'k' a constant equal to the number of revolutions per complete working cycle :

$k = 1$ 2-stroke

$k = 2$ 4-stroke

then the number of working strokes per unit time for the complete engine is:

$$f = \frac{i \cdot n}{k}$$

From the definition of power:

$$P_B = W_e \cdot f \qquad \Longrightarrow \qquad W_e = k \cdot \frac{P_B}{i \cdot n}$$

(where 'B' indicates "Brake" power, i.e. power at the output flange of the engine). Now divide by stroke volume of one cylinder to obtain mean effective pressure:

$$p_{me} = k \cdot \frac{P_B}{i \cdot n \cdot V_s}$$

From this expression it will be clear that mean effective pressure can be looked at as a specific power. Contrary to specific power of a gas turbine which relates power to *mass* flow, the mean effective pressure of a piston engine is a specific power related to *volume* flow through the engine:

$$\dot{V} = i \cdot n \cdot V_s$$

Another interpretation of mean effective pressure emerges when one realises that power is torque divided by angular velocity:

$$M_b = \frac{P_B}{\omega} = \frac{P_B}{2\pi \cdot n} \qquad \Longrightarrow \qquad \frac{P_b}{n} = 2\pi \cdot M_B$$

Then:

$$p_{me} = 2\pi \cdot k \cdot \frac{M_B}{i \cdot V_s}$$

So mean effective pressure is output torque normalised for the engine size in terms of its total stroke volume.

Typical values for the mean effective pressure of modern engines are 15 to 30 bar which are obtained by using charge pressures between 2.5 and 5 bar

3 Power density

3.1 Specific power related to stroke volume

Yet another significance of mean effective pressure can be discovered if the formula of the previous chapter is clustered in a different way:

$$p_{me} = \frac{k}{n} \cdot \frac{P_B}{i \cdot V_s}$$

From this expression the power density related to total engine stroke volume (SVSP = "Stroke Volume Specific Power") can be found to be:

$$SVSP = \frac{P_B}{i \cdot V_s} = \frac{p_{me} \cdot n}{k}$$

From this expression it will be clear that a high rotational speed and high mean effective pressure are directly proportional to a favourable power

density. The 4-stroke principle effectively halves the power density (if mean effective pressure and rotational speed would be equal to a two-stroke which in practice is not the case)

Power per litre stroke volume is shown for some engines in figure 5. As expected the power per ton installed volume follows the same trend, see figure 6. The same applies to power per m^3 of installed box volume, see figure 7.

3.2 Specific power related to bore area

There is yet another way to cluster the formula for the mean effective pressure. Remember that cylinder stroke volume is equal to stroke times cylinder bore area:

$$V_S = L_S \cdot A_B$$

Then mean effective pressure can be rewritten as:

$$p_{me} = \frac{k}{n \cdot L_S} \cdot \frac{P_B}{i \cdot A_B}$$

Now the power density related to the total engine bore area (BASP = "Bore Area Specific Power") can be found to be:

$$BASP = \frac{P_B}{i \cdot A_B} = \frac{p_{me} \cdot n \cdot L_S}{k}$$

Introduce the mean piston speed considering that the piston travels twice the stroke length during one revolution:

$$c_m \overset{def}{=} \frac{\text{distance}}{\text{time}} = \frac{2 \cdot L_S}{1/n} \qquad \Longrightarrow \qquad \boxed{c_m = 2 \cdot n \cdot L_S}$$

The mean piston speed is an indicator for the mechanical dynamics of the engine and in particular the construction of the crank/rod mechanism. Mean piston speed for modern engines lies between 8 and 12 m/s.

Then:

$$\boxed{BASP = \frac{P_B}{i \cdot A_B} = \frac{p_{me} \cdot c_m}{2 \cdot k}}$$

The specific power defined in this way is a measure of deck area necessary in the engine room.

Again a high mean effective pressure turns out to be favourable but now not *rotational* speed but rather *mean piston* speed is the other important factor. The product of mean effective pressure and mean piston speed:

$$p_{me} \cdot c_m$$

is often seen as an important quantity that represents the "technology" of the engine. High values of both factors within this quantity (for instance 30 bar and 12 m/s) can be found in the "high tech" high speed engines. Medium speed engines (with 20 to 25 bar and 10 m/s) are more conservative as they are designed for reliability and continuous operation. The sturdy and reliable slow speed diesel engines (where 18 bar and 8 m/s are typical values) seem even more conservative in this respect but it must not be forgotten that these engines also are 2-stroke meaning that every cycle counts a working stroke with its inherent thermal load (as expressed by the factor 'k' in the expression for 'BASP'!). Figure 8 shows the trend of the "technology parameter".

3.3 Maximum power for a given specification

From the equation given in the previous chapter one could think that, given a certain technology, any power could be extracted from a cylinder, simply by increasing the number of cylinders or the bore area:

$$P_B = i \cdot A_B \cdot \frac{p_{me} \cdot c_m}{2 \cdot k}$$

There are however limits to observe. For the number of cylinders there is a constructional length limit which in practice means that a line engine has 12 cylinders as maximum while for a Vee-engine the limit seems to be 20 cylinders. (but 14 cylinder line engines are in study for containerships while 24 cylinder V-engines have been built!).

Also the bore area cannot be increased arbitrarily since the stroke/bore ratio must remain within certain limits. A value >1 at least is necessary to obtain an acceptable geometrical shape of the combustion volume at Top Dead Centre (Diesel engines burning liquid fuel which is injected at the latest moment are more critical in this respect than Otto engines with premixed charge). Since stroke length at a given mean piston speed is fixed by the rotational speed of the engine, this also applies to the cylinder bore as the stroke/bore area is specified by the designer:

$$A_B = \frac{\pi}{4} \cdot D_B^{\,2} = \frac{\pi}{4} \cdot \frac{D_B^{\,2}}{L_S^{\,2}} \cdot \frac{L_S^{\,2} \cdot n^2}{n^2}$$

With the definition of mean piston speed:

$$c_m = 2 \cdot n \cdot L_S$$

and the stroke/bore ratio:

$$\lambda_S = L_S / D_B$$

the bore area can be expressed in parameters which deliberately must be selected during the design phase and for which there is only limited freedom:

$$A_B = \frac{\pi}{16} \cdot \frac{c_m^{\,2}}{\lambda_s^{\,2}} \cdot \frac{1}{n^2}$$

The bore *diameter* therefore will decrease with the square of nominal engine speed. Substitute this result in the expression for power:

$$\boxed{P_B = i \cdot \frac{\pi}{32} \cdot \frac{p_{me} \cdot c_m^{\,3}}{k \cdot \lambda_s^{\,2}} \cdot \frac{1}{n^2}}$$

This is a highly interesting expression:

- The maximum power that can be obtained from an engine of course is proportional to the number of cylinders - which is limited as we have seen - and also proportional to mean effective pressure. Since the latter has almost doubled in the last twenty years the maximum power from a given diesel engine has doubled in the same time frame.

- Maximum power decreases with the cube of engine rotational speed. Therefore high speed engines may have a high power *density* but are limited in the *absolute* power obtainable from a certain engine block.

- For a large power it would be best to select low values of the stroke/bore ratio. As argued before there are limits with respect to volume shape near Top Dead Centre in view of combustion. Nevertheless this explains why high speed engines often have a relatively small stroke/bore ratio (just >1) in order get some more power out of these small engines.

- The dependency on mean piston speed (third power) is remarkable. This explains the urge to go for high values of the mean piston speed for high speed engines although there is a general belief that this is not favourable for cylinder wear and engine reliability.

For the main types introduced in the first chapter (slow speed, medium speed and high speed) typical values of mean effective pressure, mean piston speed and stroke/bore ratio were assumed. Then for the three main types the maximum power was calculated as function of nominal rotational speed: see figure 9. From this figure it can be concluded that although for fast ships there is a preference (if not a necessity) for high speed engines in view of their high power density, the maximum available power is limited by the same nominal rotational speed, even if advanced engines score highly in technology (i.e. charge pressure and piston speed).

4 Efficiency

The overall efficiency of an engine is work output divided by heat input:

$$\eta_{tot} \overset{def}{=} \frac{W_e}{Q_f}$$

where heat input may be put equal to fuel injected per cycle times the Lower Heat Value of the fuel:

$$Q_f \cong m_f \cdot LHV$$

Not all heat goes into the thermodynamic work cycle, see figure 4. First of all combustion may be incomplete but more important a significant amount of the heat produced by combustion is carried away to the cylinder wall into the jacket cooling water and also to the piston (and from there mainly to the lube oil). All these effects are allowed for in a combustion efficiency and a heat input efficiency:

$$\eta_{comb} \overset{def}{=} \frac{Q_{comb}}{Q_f}$$

$$\eta_q \overset{def}{=} \frac{Q_i}{Q_{comb}}$$

So the heat that really is put into the cycle process is:

$$Q_i = \eta_{comb} \cdot \eta_q \cdot Q_f$$

In this expression the combustion efficiency is almost 100% but the heat input efficiency can be 80 to 90%. (heat loss 10 to 20%).

The "input heat" as defined above is (assuming a reversible cycle process) equal to an area in the temperature/entropy diagram:

$$Q_i \overset{rev}{=} \int_{combustion} T \cdot dS$$

In the same way that work can be represented as an area in a p-V diagram (figure 10) the heat is equal to an area in the T-S diagram (figure 11).

Note

The process must however be reversible. For irreversible processes work and heat are not equal to areas in the p-V and T-S diagram respectively. So be careful in case of compression and expansion in the gas turbine!

The heat input for the diesel engine can be recognised as the area in the T-S diagram under the three stretches that represent combustion in figure 11.

With the partial efficiencies (mechanical, heat loss, combustion) the overall efficiency can be written as:

$$\eta_{tot} = \eta_m \cdot \eta_{comb} \cdot \eta_q \cdot \frac{W_i}{Q_i}$$

The last factor in this expression can be interpreted as a "thermodynamic efficiency" that is completely determined by the shape of the cycle in the thermodynamic diagrams as presented:

$$\eta_{td} = \frac{W_i}{Q_i} = \frac{\oint_{cycle} p \cdot dV}{\int_{combustion} T \cdot dS}$$

Both indicated work (and thus mean effective pressure) as well as input heat can be calculated for the ideal Seiliger cycle. The overall efficiency of a diesel engine however is greatly influenced by the mechanical and heat losses and it is these losses that are relatively favourable for the bigger (but lower speed) engines when compared to the smaller, high speed engines. If anywhere the law of "economy of scale" applies to the diesel engine. This trend can be observed in figure 12 where engine efficiency is plotted against engine design speed (which is inversely proportional to size).

5 Thermodynamic analysis of the ideal Seiliger process

5.1 Seiliger cycle definition

The cylinder process can be described in a simple yet comprehensive way by means of the Seiliger process which in its more general form consists of six stages (see figure 13).

The geometrical compression ratio relates the volume at Top Dead Centre to the total cylinder volume at Bottom Dead Centre:

$$\varepsilon \overset{def}{=} \frac{V_{BDC}}{V_{TDC}}$$

The *effective* compression ratio that characterises the (isentropic) compression (1-2) generally is smaller than the geometric compression ratio:

$$r_c \overset{def}{=} \frac{V_1}{V_2}$$

Now in general (see figure 13):

$$V_1 < V_{BDC} \qquad \text{and} \qquad V_2 = V_{TDC}$$

Thus:

$$r_c < \varepsilon$$

The pressure rise at constant volume (2-3) is determined by a parameter 'a':

$$a \stackrel{def}{=} \frac{p_3}{p_2}$$

The expansion during isobaric combustion (3-4) is fixed by a parameter 'b':

$$b \stackrel{def}{=} \frac{V_4}{V_3}$$

The expansion during isothermal combustion (4-5) likewise is determined by a parameter 'c':

$$c \stackrel{def}{=} \frac{V_5}{V_4}$$

The effective expansion ratio of the (isentropic) expansion phase (5-6) is by definition:

$$r_e \stackrel{def}{=} \frac{V_6}{V_5} = \frac{V_6}{V_3} \cdot \frac{V_3}{V_4} \cdot \frac{V_4}{V_5}$$

If the expansion comes to an end at the same volume at which the compression begins then:

$$V_6 = V_1$$

while by definition the process between 2 and 3 is at constant volume:

$$V_3 = V_2$$

Then the effective expansion stroke turns out to be dependent on the previous parameters:

$$r_e = \frac{V_1}{V_2} \cdot \frac{V_3}{V_4} \cdot \frac{V_4}{V_5} = \frac{r_c}{b \cdot c}$$

Thus the shape of the cycle is completely defined by:

$$r_c, a, b, c$$

The complete original 6-point ("hexagon") Seiliger cycle can be defined analyticaly in terms of volume ratio, pressure ratio and temperature ratio for each stage as presented in the following table 1. Compression and expansion will be assumed adiabatic such that Poisson's law holds. Assuming perfect gas and equal fluid properties throughout then the isentropic index emerges as an additional parameter.

stage	Volume ratio φ	pressure ratio π	Temperature ratio τ
1 - 2	$\frac{V_1}{V_2} \overset{def}{=} r_c$	$\frac{p_2}{p_1} = r_c^{\kappa}$	$\frac{T_2}{T_1} = r_c^{\kappa-1}$
2 - 3	$\frac{V_3}{V_2} \overset{def}{=} 1$	$\frac{p_3}{p_2} \overset{def}{=} a$	$\frac{T_3}{T_2} = a$
3 - 4	$\frac{V_4}{V_3} \overset{def}{=} b$	$\frac{p_4}{p_3} \overset{def}{=} 1$	$\frac{T_4}{T_3} = b$
4 - 5	$\frac{V_5}{V_4} \overset{def}{=} c$	$\frac{p_4}{p_5} = c$	$\frac{T_4}{T_5} \overset{def}{=} 1$
5 - 6	$\frac{V_6}{V_5} = \frac{r_c}{b \cdot c}$	$\frac{p_5}{p_6} = \left(\frac{r_c}{b \cdot c}\right)^{\kappa}$	$\frac{T_5}{T_6} = \left(\frac{r_c}{b \cdot c}\right)^{\kappa-1}$
6 - 1	$\frac{V_6}{V_1} \overset{def}{=} 1$	$\frac{p_6}{p_1} = \frac{r_c^{\kappa} \cdot a}{c \cdot \left(\frac{r_c}{b \cdot c}\right)^{\kappa}} = a \cdot b^{\kappa} \cdot c^{\kappa-1}$	$\frac{T_6}{T_1} = \frac{r_c^{\kappa-1} \cdot a \cdot b}{\left(\frac{r_c}{b \cdot c}\right)^{\kappa-1}} = a \cdot b^{\kappa} \cdot c^{\kappa-1}$

Table 1 Analytical definition of Seiliger process

5.2 Heat and work

The theoretical heat input per kg of trapped mass is:

$$q_{in} = q_{23} + q_{34} + q_{56}$$

Note that it is also implicitly assumed that the mass is constant throughout the process. The contribution of the three combustion stages can be further specified:

$$q_{23} = c_v \cdot (T_3 - T_2) = c_v \cdot T_1 \cdot r_c^{\kappa-1} \cdot (a - 1)$$

$$q_{34} = c_p \cdot (T_4 - T_3) = c_v \cdot T_1 \cdot \kappa \cdot r_c^{\kappa-1} \cdot a \cdot (b - 1)$$

$$q_{45} = R \cdot T_4 \cdot \ln\left(\frac{V_5}{V_4}\right) = c_v \cdot T_1 \cdot (\kappa - 1) \cdot r_c^{\kappa-1} \cdot a \cdot b \cdot \ln(c)$$

As can be observed in the right hand side of these equations all heat portions can, with help of table 1, be expressed in the four parameters as well as the initial temperature, the specific heat and the isentropic index (the latter two assumed constant for all stages).

This also applies to the heat output. When the compression (1-2) and expansion (5-6) are assumed adiabatic the only heat heat rejection takes place during exhaust (6-1):

$$q_{out} = q_{61}$$

which can be idealised as a constant volume process:

$$q_{61} = c_v \cdot (T_6 - T_1) = c_v \cdot T_1 \cdot (a \cdot b^{\kappa} \cdot c^{\kappa-1} - 1)$$

Physically this heat is rejected by driving out the exhaust gases and taking in fresh and cool air.

5.3 Specific input heat and thermodynamic efficiency

The input heat is by definition equal to all heat flowing into the process:

$$q_i = q_{in}$$

With the equations derived in the previous chapter the specific heat input can be expressed non-dimensionally:

$$\boxed{\frac{q_i}{c_v \cdot T_1} = r_c^{\kappa-1} \cdot \{(a-1) + \kappa \cdot a \cdot (b-1) + (\kappa-1) \cdot a \cdot b \cdot \ln(c)\}}$$

The ideal thermodynamic efficiency can be obtained from:

$$\eta_{td} \overset{def}{=} \frac{w_i}{q_i} = \frac{w_{out} - w_{in}}{q_{in}}$$

For a cycle process the net work output is equal to the net heat input:

$$w_{out} - w_{in} = q_{in} - q_{out}$$

Thus the thermodynamic efficiency can be expressed in heat only:

$$\eta_{td} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

Making use of the equations derived in the previous chapter:

$$\boxed{\eta_{td} = 1 - \frac{1}{r_c^{\kappa-1}} \cdot \left\{ \frac{a \cdot b^{\kappa} \cdot c^{\kappa-1} - 1}{(a-1) + \kappa \cdot a \cdot (b-1) + (\kappa-1) \cdot a \cdot b \cdot \ln(c)} \right\}}$$

Note

Special cases occur for:

c = 1: Well-known 5-point "Seiliger" or "dual cycle" process,

c = 1; a = 1: Ideal 4-point "Diesel" process,

c = 1; b = 1: Ideal 4-point "Otto" process.

It is left to the reader to write out the ideal efficiency for these cases.

5.4 Specific work and mean indicated pressure

The indicated work per kg of trapped mass (assumed to remain unaltered during the cycle) is:

$$w_i = w_{out} - w_{in}$$

With the thermodynamic efficiency from the previous chapter:

$$\eta_{td} \stackrel{def}{=} \frac{w_{out} - w_{in}}{q_{in}}$$

the indicated work can be expressed in the input heat:

$$w_i = \eta_{td} \cdot q_{in}$$

With the equations derived in previous chapters the specific indicated work can be expressed non-dimensionally:

$$\boxed{\frac{w_i}{c_v \cdot T_1} = \eta_{td} \cdot \left[r_c^{\kappa-1} \cdot \left\{ (a-1) + \kappa \cdot a \cdot (b-1) + (\kappa-1) \cdot a \cdot b \cdot \ln(c) \right\} \right]}$$

The mean indicated pressure is by definition:

$$p_{mi} \stackrel{def}{=} \frac{W_i}{V_S}$$

Indicated work is trapped mass times specific work:

$$W_i = m_1 \cdot w_i = \frac{p_1 \cdot V_1}{R \cdot T_1} \cdot w_i$$

The mean indicated pressure in non-dimensional form:

$$\frac{p_{mi}}{p_1} = \frac{V_1}{V_S} \cdot \frac{w_i}{R \cdot T_1}$$

The volume ratio appearing in this relation expresses the fact that the active volume is slightly different from the stroke volume (the latter being the volume used in the definition of the mean pressure):

$$\frac{V_1}{V_S} = \frac{V_1}{V_2} \cdot \frac{V_{TDC}}{V_{BDC} - V_{TDC}} = \frac{r_c}{\varepsilon - 1}$$

The factor making specific work non-dimensional is equal to:

$$R \cdot T_1 = \left(c_p - c_v \right) \cdot T_1 = (\kappa - 1) \cdot c_v \cdot T_1$$

So the mean indicated pressure can be expressed in specific work as:

$$\frac{p_{mi}}{p_1} = \frac{1}{\kappa - 1} \cdot \frac{r_c}{\varepsilon - 1} \cdot \frac{w_i}{c_v \cdot T_1}$$

Now substitute the expression for specific work derived above:

$$\boxed{\frac{p_{mi}}{p_1} = \frac{\eta_{td}}{\kappa - 1} \cdot \frac{r_c}{\varepsilon - 1} \cdot \left[r_c^{\kappa-1} \cdot \{(a-1) + \kappa \cdot a \cdot (b-1) + (\kappa-1) \cdot a \cdot b \cdot \ln(c)\} \right]}$$

Note

Again the reader may derive for himself the expressions for the mean indicated pressure for the simpler "dual combustion" cycle, the pure "Diesel" cycle and the "Otto" cycle.

5.5 Constraints

Comparing different cycles must be done on a fair basis. Then it seems logical to keep track of two important limitations in the design of turbocharged diesel engines: the maximum pressure and the air excess during combustion.

For a turbocharged diesel engine the peak pressure must be considered as a definite constraint. The dimensioning of cylinder head, piston, connecting rod, crankshaft and bearings (and thus overall cost of the engine) is strongly dependent on the chosen design value of the peak pressure. For a given value of the charge pressure, the pressure rise during combustion and the chosen peak pressure it turns out that the effective compression ratio is a dependent variable:

$$p_{max} = p_3 = a \cdot r_c^{\kappa} \cdot p_1 \qquad \Rightarrow \qquad \boxed{r_c = \left(\frac{p_{max}}{a \cdot p_1} \right)^{\frac{1}{\kappa}}}$$

The air excess ratio determines the conditions for combustion and must be carefully selected a.o. in view of the fuel to be burned (a lower value can be chosen in case of light destillate fuels). The air excess depends on the air fuel ratio which can be expressed in the thermal load (heat input of fuel) of the engine:

$$afr \overset{def}{=} \frac{m_a}{m_f} = \frac{m_a}{m_1} \cdot \frac{m_1 \cdot LHV}{Q_f}$$

Heat of fuel per kg of trapped mass is equal to specific input heat divided by the heat input efficiency (which allows for the heat loss during combustion)

$$\frac{Q_f}{m_1} = \frac{1}{\eta_{comb} \cdot \eta_q} \cdot \frac{Q_i}{m_1} \cong \frac{q_i}{\eta_q}$$

Note that the combustion efficiency is neglected here. Further assume the trapped mass equal to the fresh air in the cylinder (100% scavenge efficiency):

$$\frac{m_a}{m_1} \overset{def}{=} \eta_{scav} \cong 1$$

Then the air excess ratio (being air/fuel ratio related to stoichiomtric ratio of the fuel) is:

$$\lambda \overset{def}{=} \frac{afr}{\sigma} = \eta_q \cdot \frac{LHV}{\sigma \cdot q_i}$$

Now substitute the expression for the input heat as derived in a previous chapter:

$$\lambda = \eta_q \cdot \frac{LHV}{\sigma \cdot c_v \cdot T_1} \cdot \frac{1}{r_c^{\kappa-1} \cdot \{(a-1)+\kappa \cdot a \cdot (b-1)+(\kappa-1) \cdot a \cdot b \cdot \ln(c)\}}$$

Now if combustion pressure rise parameter 'a' and isothermal expansion ratio 'c' are given reasonable values and for the air excess ratio is taken a chosen value it turns out that the parameter 'b' is a dependent parameter:

$$b = \frac{\left\{\left(\eta_q \cdot \frac{LHV}{\lambda \cdot \sigma \cdot c_v \cdot T_1} \cdot \frac{1}{r_c^{\kappa-1}}\right) - (a-1) + \kappa \cdot a\right\}}{(\kappa-1) \cdot a \cdot \ln(c) + \kappa \cdot a}$$

5.6 Trend in mean indicated pressure and efficiency for ideal cycle

The theoretical framework derived in the previous chapters can be used to draw some general conclusions about the thermodynamic performance of the turbocharged diesel cycle. Charge pressures up to 5 bar have been investigated while keeping peak pressures at distinct values of 120, 160 and 240 bar.

Figure 14 illustrates the increase of the mean indicated pressure with increasing charge pressure. The slight curvature of the lines is due to the fact that keeping the maximum pressure and air excess ratio constant distorts the shape of the cycle as can be seen in the p-V diagram (figure 16) and in the T-s diagram (figure 17). Further observe that the effect of the peak pressure on mean indicated pressure is far less than charge pressure.

Figure 15 presents the ideal efficiency of the diesel cycle. Apart from the fact that the values shown are unrealistically high (this will be corrected in the next chapter) it is remarkable that turbocharging seems to have a bad effect on cycle efficiency. This is not unexpected since the distortion of the diesel cycle under the constraint that the peak pressure and the air excess ratio remains constant is not favourable for thermodynamic efficiency. The reader

may verify this by taking a close at the cycle shape in figure 16 and 17. In particular the lower temperature of the combustion and the higher exhaust temperature as observed in the T-s diagram in case of the highly turbocharged cycle indicate an unfavourable effect on efficiency. In the next chapter it will be proven that the loss of thermodynamic efficiency is compensated by lower parasitic losses and that a turbocharged cycle can have a better overall efficiency than a non turbocharged cycle.

Increasing the peak pressure is a way to compensate the efficiency loss caused by the higher charge pressure. This is not unnatural since increasing the charge pressure at constant peak pressure would result in a compression ratio that is too low for auto-ignition (see limit line in figure 15). The thermodynamic reasons for the increase of cycle efficiency as a result of a higher peak pressure can be found in the p-V diagram (figure 18) and T-s diagram (figure 19). In particular the higher temperature level of the combustion and the lower exhaust temperature as observed in the T-s diagram in case of high peak pressures are favourable in that respect.

6 Cycle with losses and optimisation of performance

If the stroke volume would be filled with air of inlet receiver conditions (index 'c') the mass of air would, according to the ideal gas law, be equal to:

$$m^{\text{id gas}} = \frac{p_c \cdot V_s}{R_a \cdot T_c}$$

Since:

- pressure and temperature in the cylinder at the start of compression (inlet valve closes) normally are higher then in the inlet receiver (particularly temperature)
- the cylinder volume at that instant also is not exactly equal to the stroke volume
- some gas remaining from the previous stroke will still be present

the total amount of air present in the cylinder normally is lower than the mass 'm' above:

$$m_a = \eta_{scav} \cdot m_1 = \eta_{scav} \cdot \frac{p_1 \cdot V_1}{R_1 \cdot T_1} = \eta_{scav} \cdot \frac{p_1}{p_c} \cdot \frac{T_c}{T_1} \cdot \frac{V_1}{V_s} \cdot \frac{R_a}{R_1} \cdot \frac{p_c \cdot V_s}{R_a \cdot T_c}$$

Put all the factors together into a "trapped efficiency":

$$m_a = \eta_{trap} \cdot \frac{p_c \cdot V_s}{R_a \cdot T_c}$$

Total heat produced during combustion is approximately:

$$Q_f \cong m_f \cdot LHV$$

An important quantity is the "air/fuel ratio" which was previously defined as:

$$afr \overset{def}{=} \frac{m_a}{m_f}$$

Then the potential heat from the fuel is:

$$Q_f = m_a \cdot \frac{LHV}{afr} = \eta_{trap} \cdot \frac{p_c \cdot V_S}{R_a \cdot T_c} \cdot \frac{LCV}{afr}$$

Work done during the cycle is overall efficiency times fuel heat:

$$W_e = \eta_{tot} \cdot Q_f$$

The overall efficiency can be split into its partial efficiencies:

$$\boxed{\eta_{tot} = \eta_m \cdot \eta_{comb} \cdot \eta_q \cdot \eta_{td}}$$

The effective work of the cycle is:

$$W_e = \eta_m \cdot \eta_{comb} \cdot \eta_q \cdot \eta_{td} \cdot \eta_{trap} \cdot \frac{p_c \cdot V_S}{R_a \cdot T_c} \cdot \frac{LHV}{afr}$$

Divide by stroke volume and reshuffle quantities to obtain a neat expression for the mean effective pressure of the turbocharged diesel engine:

$$\boxed{p_{me} = \eta_m \cdot \eta_{comb} \cdot \eta_q \cdot \eta_{trap} \cdot \frac{p_c}{T_c} \cdot \frac{1}{afr} \cdot \frac{LHV}{R_a} \cdot \eta_{td}}$$

From this expression the importance of a high charge pressure and effective intercooling is directly visible. Also the a low air/fuel ratio is important but when fuel is increased too much the combustion will become incomplete and the engine even may produce visible smoke. The thermodynamic efficiency of course has a direct influence. And last but not least there is the effect of several loss factors.

From the last equation an important conclusion can be drawn. It may be observed that the power of the *a given engine* can be increased considerably. But then the losses (mechanical and heat losses) will not increase in the same proportion. Therefore it must be concluded that the losses will tend to be *absolutely* equal and thus *relatively* smaller. In other words the mechanical efficiency and heat input efficiency as defined will *decrease* with *increasing* charge pressure see figure 20.

The result is that the mean effective pressure will increase with increasing charge pressure, same as for the ideal case in the previous chapter, see figure 21.

More important the overall efficiency now is the result of two opposite effects: an increase of charge pressure will cause the ideal thermodynanic efficiency to become worse but this is more or less compensated by the relative decrease of the mechanical losses and heat losses. The net result is shown in figure 22. Now for each peak pressure there seems to be an optimum value of the charge pressure. Peak pressure as such of course remains the important parameter to improve efficiency in any case.

The results for mean effective pressure and overall efficiency can be brought together in a trade-off picture with both parameters at the axes. The charge pressure, together with peak pressure is then a running parameter along the curves, see figure 23.

Now it is clear that when the charge pressure is increased there comes a moment (say at 1.5 or 2 bar) where it is necessary to increase peak pressure as well in order to maintain overall efficiency.

This is exactly what can be seen in the development of modern turbocharged engines today. Charge pressures have reached values of 3 to 4 bar with peak pressures of 150 to 180 bar. While charge pressures continue to increase to 5 bar the peak pressure for which modern engines are designed is 200 to 220 bar.

A further discussion of these issues can be found in [Grimmelius, 1999].

7 Two-stage turbocharging

For charge pressures above 4 bar two-stage charging must be considered (although the limit for a single stage radial compressor is somewhat higher).

The principle of two-stage turbocharging is shown in figure 24. Two charge compressors are put in series to achieve the required high charge pressure. Apart from intercooling before the inlet receiver the opportunity is taken to also have intercooling between the LP and HP compressors. Each compressor is driven by its own turbine. The turbines are also put in series.

The difference between *two-stage* turbocharging and *sequential* turbocharging (which will be discussed in the next chapter) must be well understood: for sequential turbocharging the turbochargers are put in parallel. The combination of two-stage and sequential turbocharging is possible though.

8 Off-design characteristics of a turbocharged diesel engine

8.1 Power/speed characteristic

A diesel engine in general, but highly turbocharged engines even more so, have limitations with respect to the operation in the power/speed characteristic, refer to figure 25.

The limits are as follows:

- First there is a maximum speed for which the rotating and translating parts (pistons, connecting rods, crankshaft but also the valves with their actuating mechanism) are designed. The governor of the diesel engine monitors engine speed and reduces fuel as soon as overspeed tends to occur. Above a certain limit an emergency stop will be initiated (fuel to zero).

- A diesel engine also has a minimum speed: below a certain speed the engine will not run smoothly or even stop altogether. This is caused by loss of compression and as a consequence failure of ignition. Minimum speed normally is 30 to 40% of nominal speed. When driving a ship with a fixed pitch propeller this also dictates minimum ship speed. For good control of low ship speeds a controllable pitch propeller must be adopted.

- There may be a small speed range where maximum power can be delivered.

- Below this "maximum power window" the power of a diesel engine is limited by the maximum fuel injected per cycle. But remember that mean effective pressure was:

$$p_{me} = \eta_m \cdot \eta_{comb} \cdot \eta_q \cdot \eta_{trap} \cdot \frac{p_c}{T_c} \cdot \frac{1}{afr} \cdot \frac{LHV}{R_a} \cdot \eta_{td}$$

and air/fuel ratio by definition:

$$afr \overset{def}{=} \frac{m_a}{m_f}$$

Thus if all (partial) efficiencies do not vary too much the mean effective pressure is proportional to charge density and injected fuel divided by cylinder air charge:

$$p_{me} \propto \frac{p_c}{R_a \cdot T_c} \cdot \frac{m_f}{m_a}$$

The cylinder mass is proportional to charge pressure:

$$m_a \propto \frac{p_c}{R_a \cdot T_c} \cdot$$

Then mean effective pressure is proportional to mass flow:

$$p_{me} \propto m_f$$

So maximum fuel implies maximum mean effective pressure which is equivalent to maximum torque and shows in the power/speed characteristic as a straight line to the origin.

- For a turbocharged diesel engine below a certain speed even maximum torque cannot be achieved anymore. The reason is that the lower engine speed causes a lower exhaust flow. The turbine pressure then falls rapidly and although exhaust energy (= exhaust temperature) is still high the turbine cannot deliver sufficient power to the compressor and charge pressure drops. Cylinder mass, which is still proportional to charge pressure, then drops as well. When injected fuel is maintained the air/fuel ratio will drop as a result and air excess may become insufficient. This causes the temperatures, in particular the exhaust temperature to become (too) high, i.e. thermal overload of the engine. Also the compressor may surge. The net result is a "forbidden area" well below the maximum torque line.

- Below 25% to 40% power the combustion in the engine may become too "cold" and fouling of the cylinder will occur. Running a diesel engine at low power therefore is only permitted for a limited time.

A genuine characteristic of a highly charged diesel engine can be seen in figure 26.

8.2 Off-design for Constant torque/Propeller law/Constant speed

Three typical load curves, i.e. constant torque, propeller law and constant speed are drawn in the power/speed characteristic, figure 27.

The next figure 28 shows the drop in charge pressure at part load. Note that the 35% charge pressure at low power corresponds to approximately 1 bar absolute pressure, i.e. the turbocharger is not effective anymore!

These curves as well as the ones to come were drawn with a simplified theory according to [Winkler, 1980] which a.o. assumes constant turbomachinery efficiency.

The next figure 29 showing the intake air flow is crucial. For constant speed operation the intake air flow and density fall off at the same rate. For propeller law and constant torque the *engine speed decreases* together with power. Since mass flow is proportional with charge pressure *and* speed, the air mass flow decreases steeper then the charge pressure.

At the same time the decrease of fuel flow is almost the same for the three load curves and practically proportional with power: see figure 30. The specific fuel flow, figure 31, is relatively flat for diesel engines.

The fact that air mass flow decreases so much faster than fuel flow for propeller law and constant torque makes that the air excess drops fast in these cases as can be observed in figure 32.

When charge pressure is plotted against intake mass flow the trajectory in the compressor map results. High pressures at low mass flows, as particularly

occur for constant torque, indicate danger for surge of the compressor, see figure 33.

8.3 Methods to broaden the engine characteristic

To broaden the power/speed envelope of highly turbocharged diesel engines the following options are available, see figure 34:

- One can adopt the "pulse system" instead of the "constant pressure" system. In case of the constant pressure system (German: Stau Aufladung) all cylinders exhaust in one, big exhaust receiver which feds into the turbocharger(s). The pressure before the turbine(s) is constant but relatively low. One can raise the pressure before the turbines by connecting a small number (maximum of 3) cylinders through a small exhaust duct to an exhaust turbine. The result of the smaller duct volume is that high peak pressures will occur before the turbine which then will produce more power to the compressor. This pulse system (German: Stoβsystem) has the advantage of giving better charge at part load and thus a somewhat broader characteristic.

- Another possibility is to stay with the more simple constant pressure system but connect more turbochagers in parallel. At part load one ore more turbochargers can be switched off. The smaller exhaust gas flow is then distributed among the remaining turbines increasing the pressure before these turbines which then will be able to maintain speed and produce power to the remaining compressors. The inlet receivers (if there is more than one) must be mutually connected of course. This system has been named "sequential turbocharging" (German: Abgas Turbo Lader Schaltung). The principle can be understood from figure 35. An example of the wide envelope of an engine equipped with sequential turbocharging is given in figure 36.

9 Fuels

Contrary to low-speed and medium speed engines that often run on heavy fuel oil, high speed diesel engines can only run on light destillate fuels:

- Light Diesel Fuel Oil, LDF (or sometimes LDO); ISO designation DMA; density 840 to 890 kg/m^3; viscosity at 40°C 1.5 tot 5 cSt. This fuel corresponds to NATO F 76.

- Marine Diesel Oil, MDO; ISO designation DMB; density 860 to 900 kg/m^3 and viscosity 5 tot 11 cSt at 40°C.

10 Installation on board

Airborne noise of high speed engines is such that there have been (naval) applications where an acoustical enclosure has been fitted. Also for naval application a double elastic mounting may be necessary to attenuate the

structure borne noise. For both acoustical enclosure and resilient mounting see figure 37.

An example of the installation of four high speed diesel engines in a waterjet driven fast ferry is presented in figure 38.

Some diesel engine data

The number of manufacturers of light, highly turbocharged, high speed diesel engines is limited: MTU and Ruston. Also the diesel division of Fincantieri is active in the market but their GMT and Isotta Fraschini engines are not very highly turbocharged. For overview of some data see following table.

		MTU 16V 2000 M90	MTU 16V 4000 M90	MTU 16V 595 TE 90	MTU 20V1163 TB93	Ruston Rk 270
Brake Power	kW	1343	2720	3600[1]	7400[2]	6875[3]
Output speed	rpm	2300	2100	1700	1300	1000
Configurations	-	12,16V	12,16V	12,16V	12,16,20V	12,16,20V
Bore	mm	130	165	190	230	270
Stroke	mm	150	190	210	280	305
mean piston speed	m/s	11.5	13.3	11.9	12.1	10.2
spec. fuel cons.	g/kWh	220	199	212	212	198
Mean effective pressure	bar	21.7	23.9	26.7	29.4	23.9
Air mass flow	kg/s	1.9	4.1	6.2	12.6	13.2
spec. air cons.	kg/kWh	5.2	5.4	6.2	6.1	6.9
Weight	kg	3890	9280	13400	25310	33000
Specific weight	kg/kW	2.9	3.4	3.7	3.4	4.8
Length	m	2.855	3.630	3.980	5.100	5.965
Width	m	1.400	1.520	1.500	1.660	1.940
Height	m	1.275	1.820	2.855	2.905	2.820
Volume	m^3	5.1	10.0	17.0	24.6	32.6
Specific volume	dm^3/kW	3.8	3.7	4.7	3.3	4.7
Remarks		Developed with Detroit Diesel	Developed with Detroit Diesel			

Table 2 Survey of highly turbocharged, high speed diesel engines

[1] This is a 1B rating intended for fast ships with a high duty factor. The engine is also available as 1DS rating with a peak rating of 4320 kW.

[2] This is a peak power of the 1DS rating which is intended for Fast Patrol Boats and High Speed Yachts with a low duty factor. The "normal" overload for this engine is 6375 kW while continuous power is 6120 kW.

[3] This is a "restricted marine" rating. The "unrestricted marine" rating allows 6250 kW.

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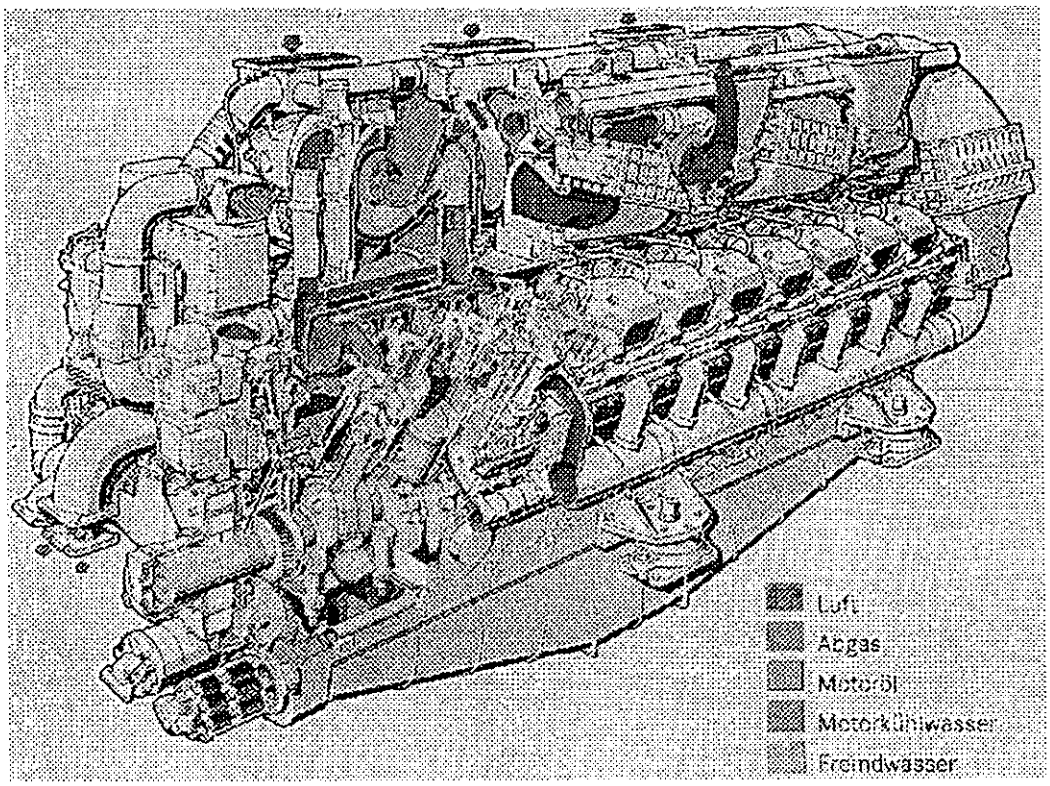


Figure 1 Advanced high speed diesel engine with two-stage sequential turbocharging (MTU 20V 1163-03)

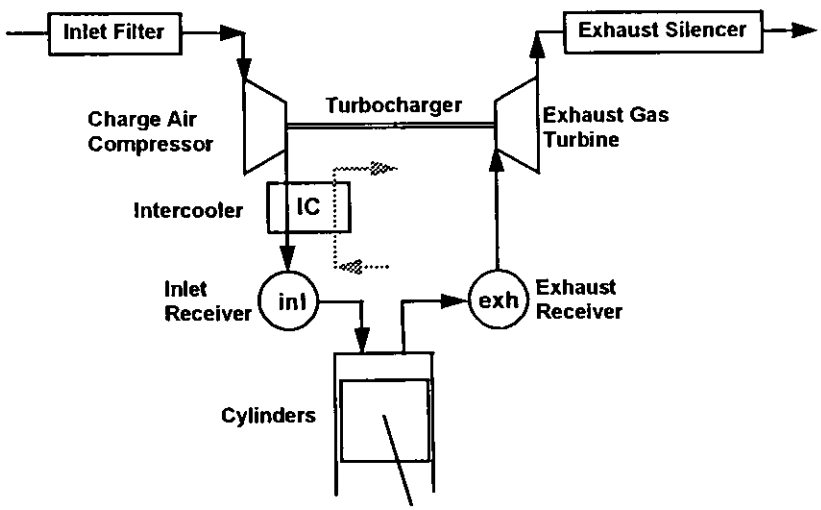


Figure 2 Principle diagram of exhaust driven charging of a diesel engine, including intercooling of the charge air

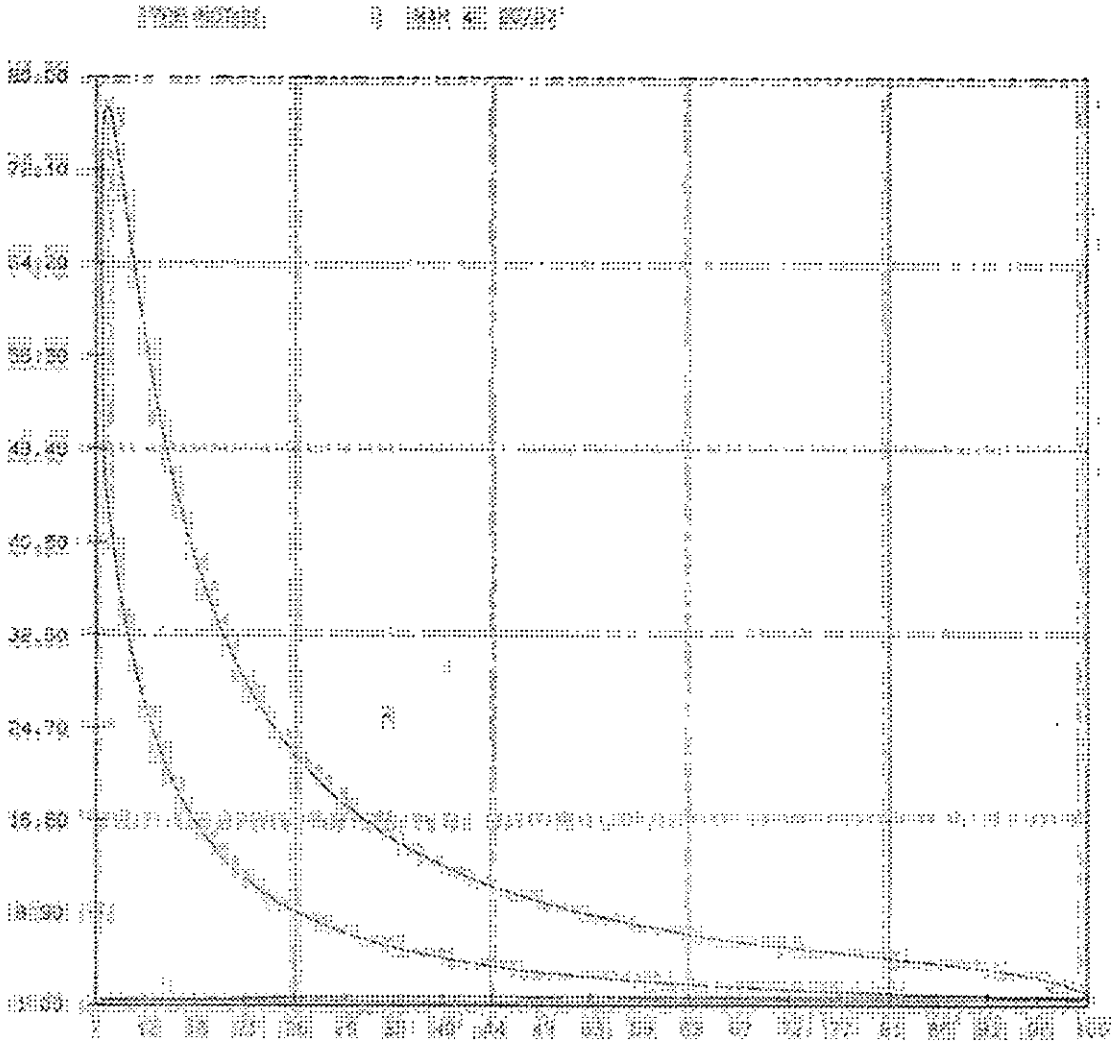


Figure 3 Cycle process of a diesel engine in a pressure-volume (p-V) diagram

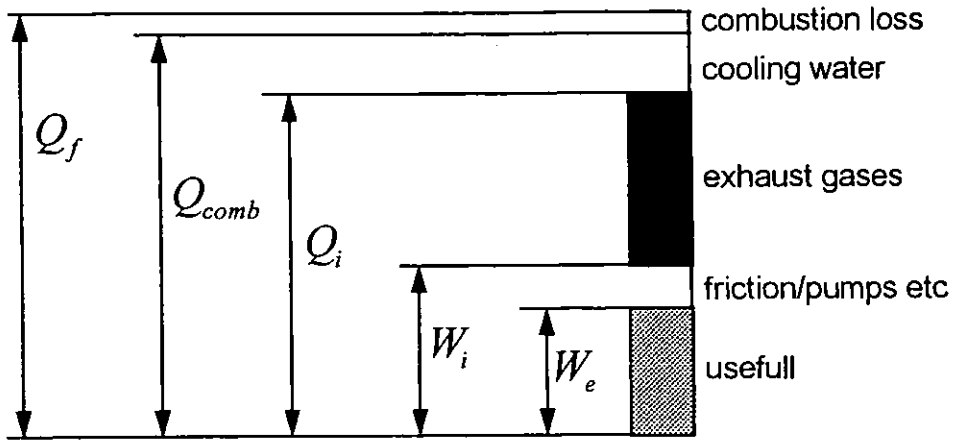


Figure 4 Energy losses in a diesel engine

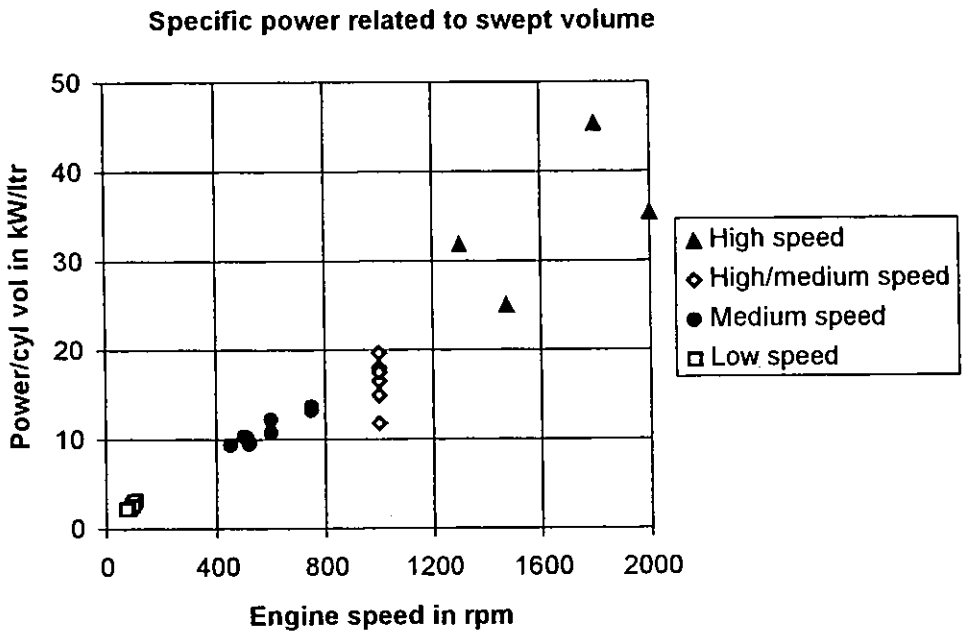


Figure 5 Trend of "Stroke Volume Specific Power" for diesel engines as function of nominal rotational speed

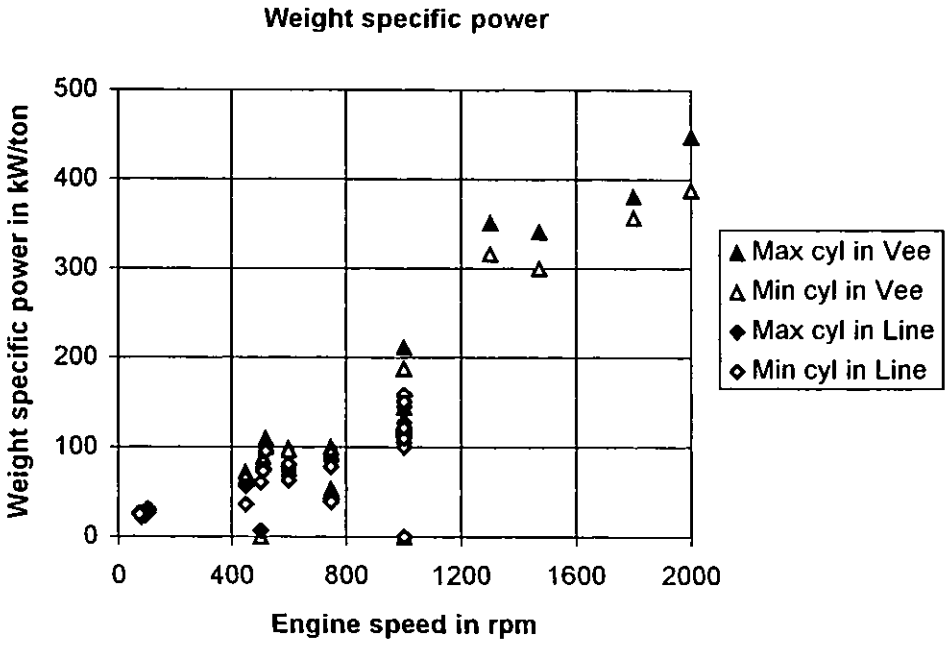


Figure 6 Trend of weight specific power for diesel engines as function of nominal rotational speed

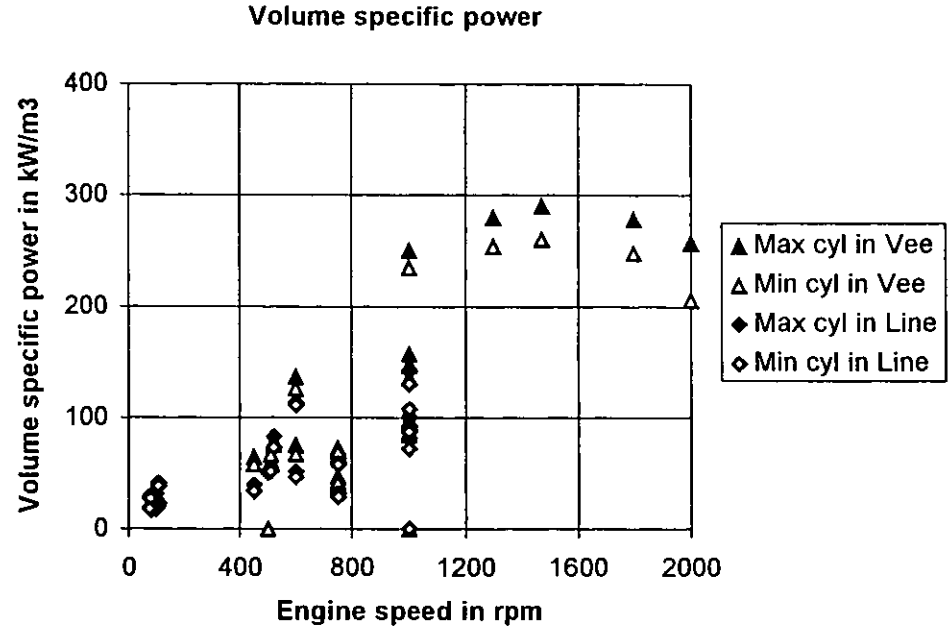


Figure 7 Trend of volume specific power for diesel engines as function of nominal rotational speed

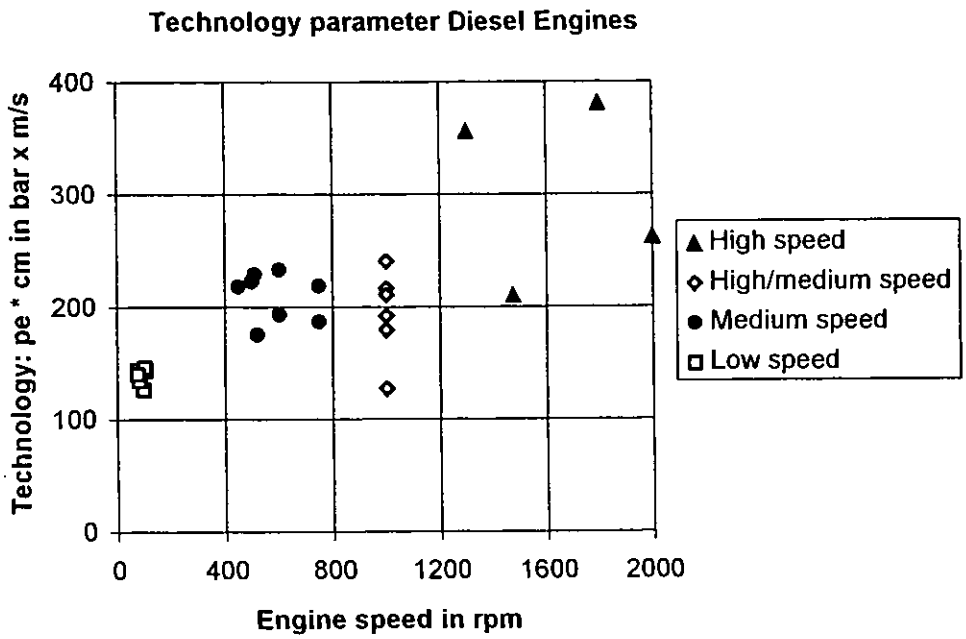


Figure 8 Trend of the "technology parameter" (mean effective pressure times mean piston speed) as function of nominal rotational speed

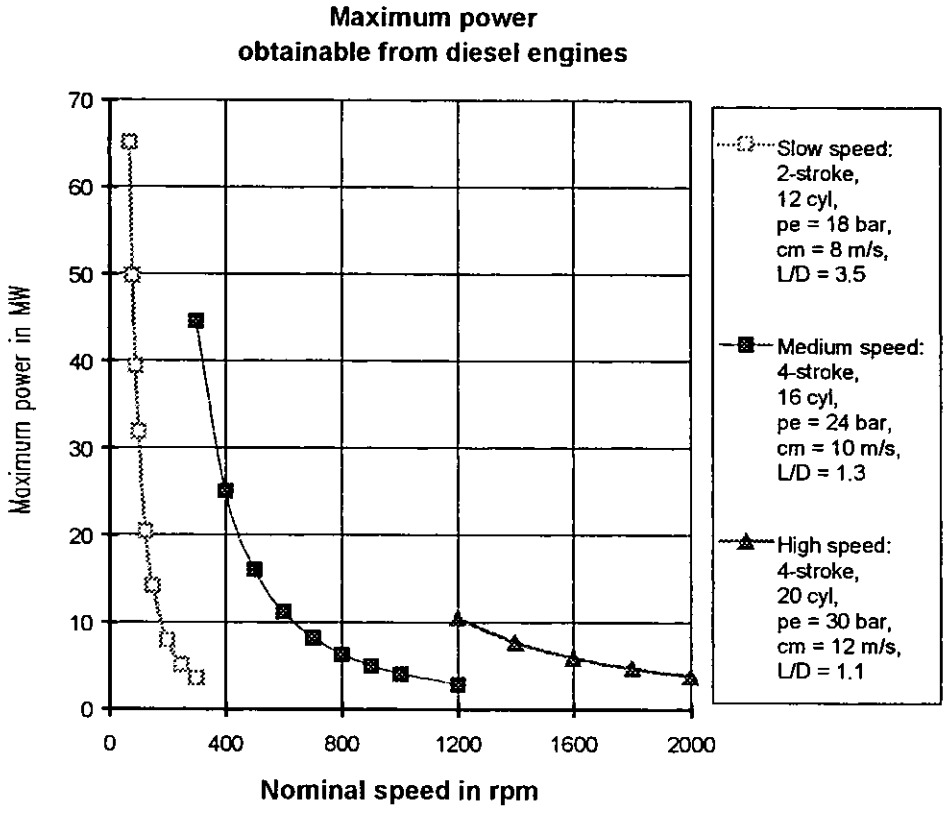


Figure 9 Maximum obtainable power for slow speed 2-stroke engines, medium speed 4-stroke engines and high speed 4-stroke engines, as function of nominal rotational engine speed

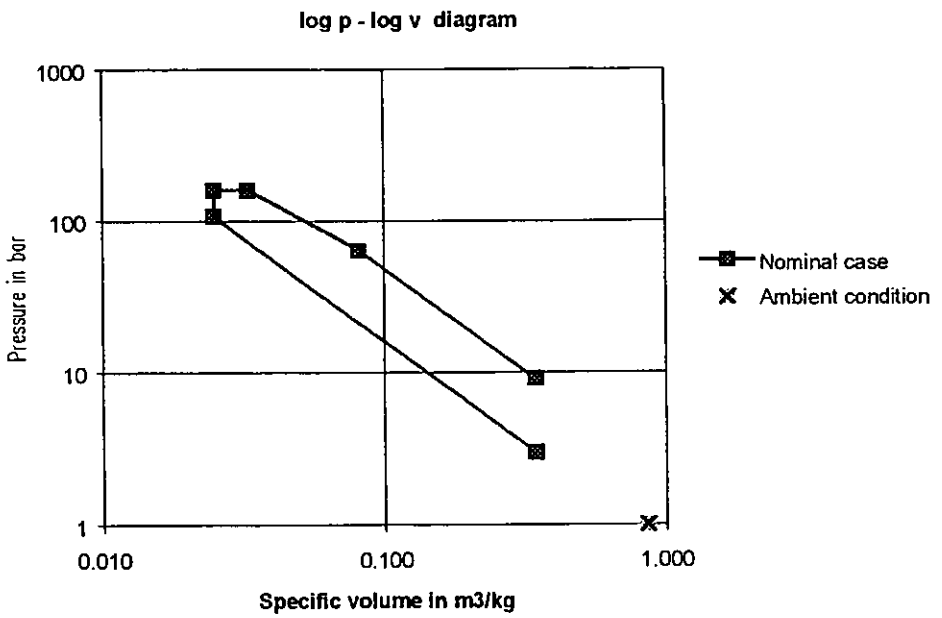


Figure 10 Logarithmic p-V diagram of Seiliger shaped diesel cycle

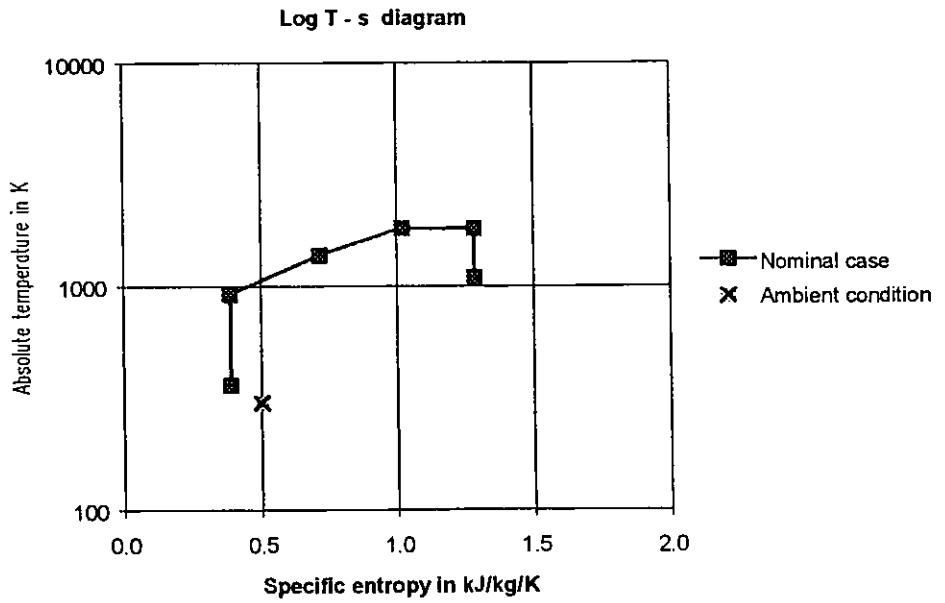


Figure 11 Logarithmic T-s diagram of Seiliger shaped diesel cycle

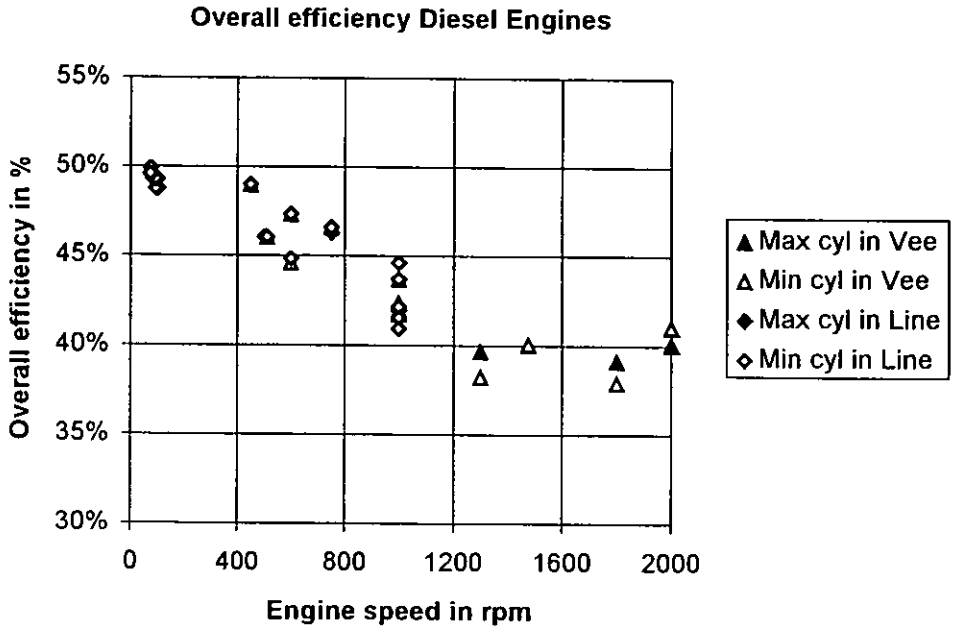


Figure 12 Trend of overall efficiency of diesel engines as function of nominal rotational speed

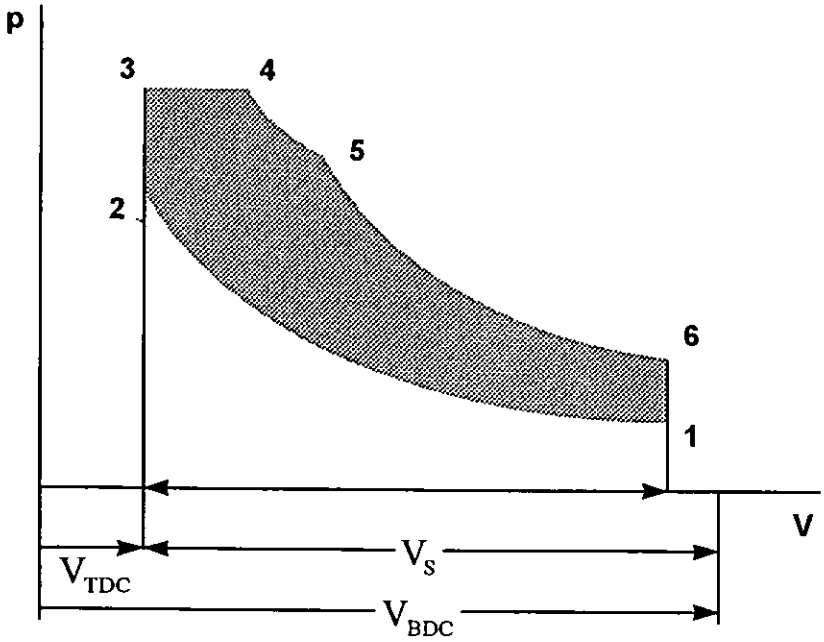


Figure 13 Seiliger idealisation of diesel process

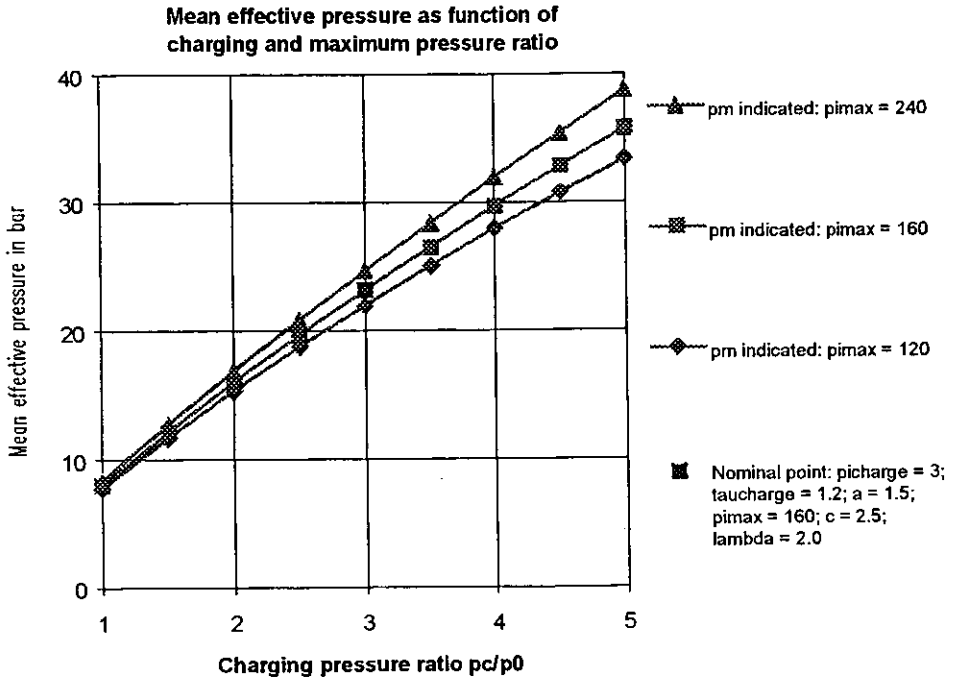


Figure 14 Ideal mean indicated pressure of Seiliger cycle

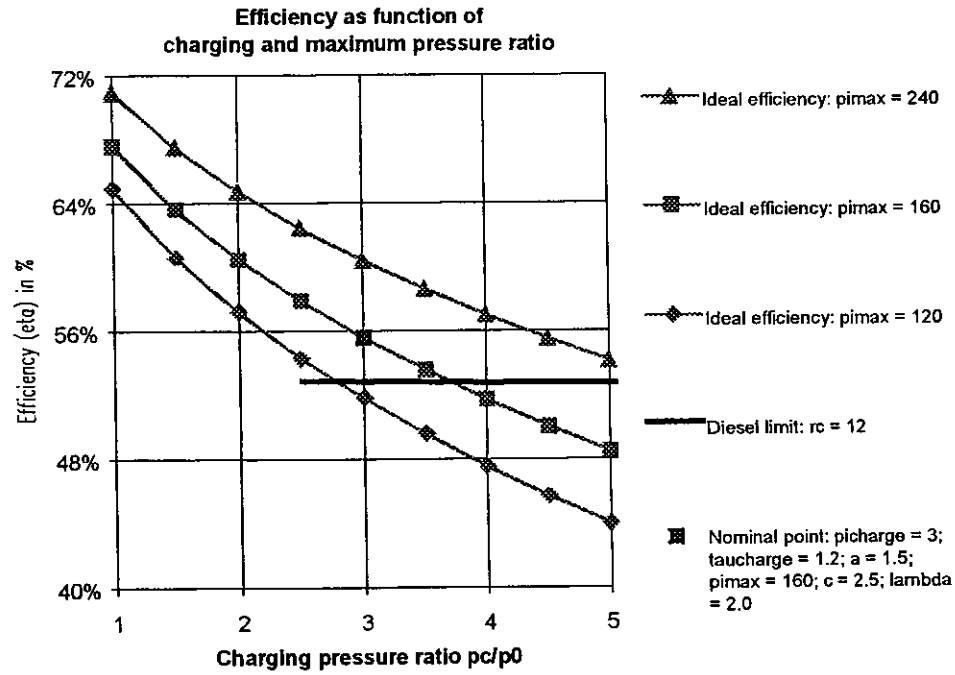


Figure 15 Ideal thermodynamic efficiency of Seiliger cycle

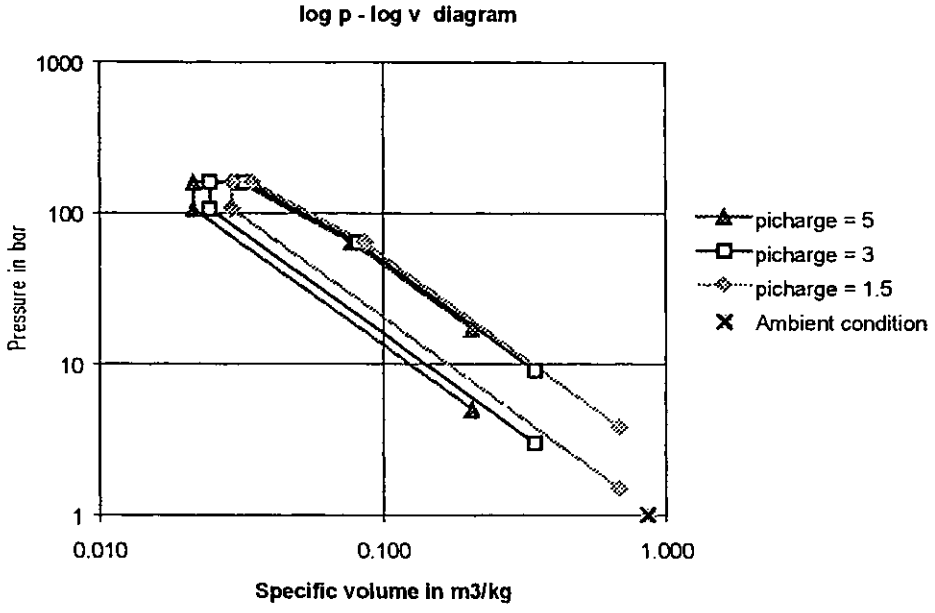


Figure 16 P-v diagram of Seiliger cycle with increasing charge pressure and constant peak pressure

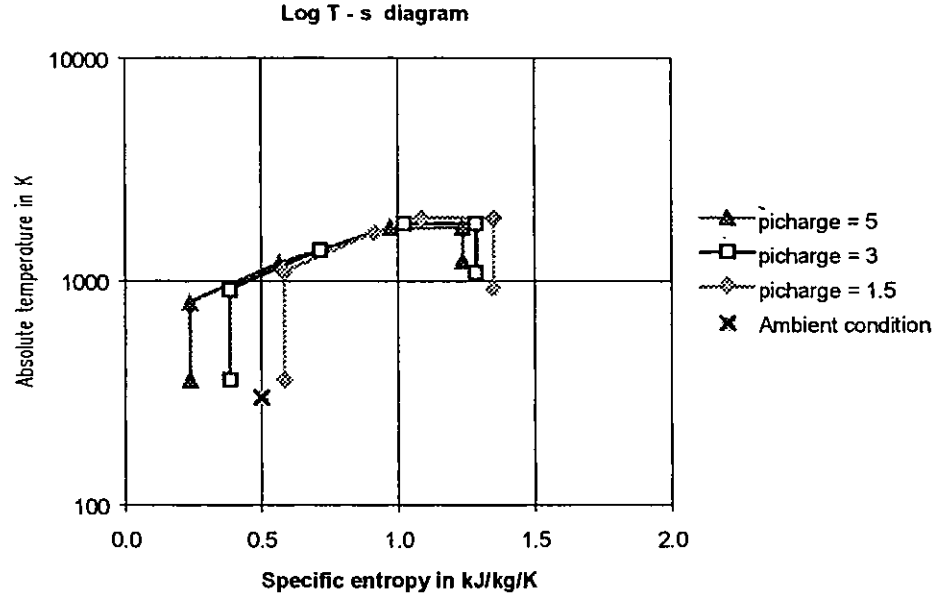


Figure 17 T-s diagram of Seiliger cycle with increasing charge pressure and constant peak pressure

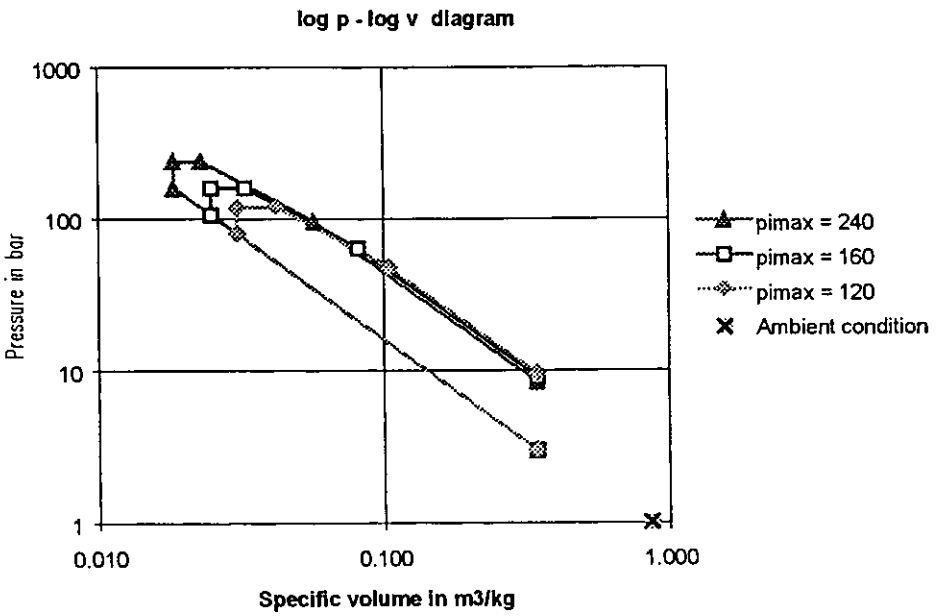


Figure 18 P-s diagram of Seiliger cycle with increasing peak pressure and constant charge pressure

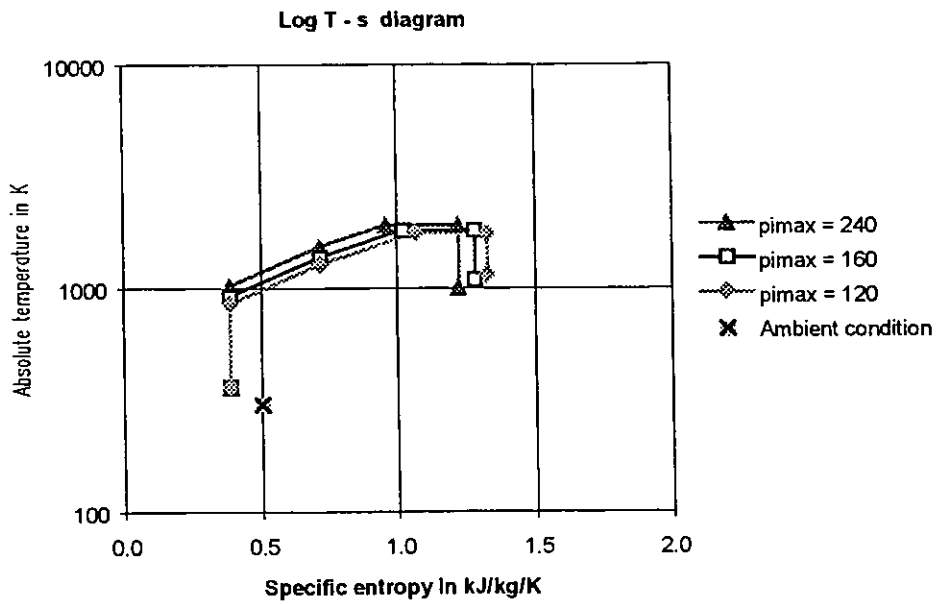


Figure 19 T-s diagram of Seiliger cycle with increasing peak pressure and constant charge pressure

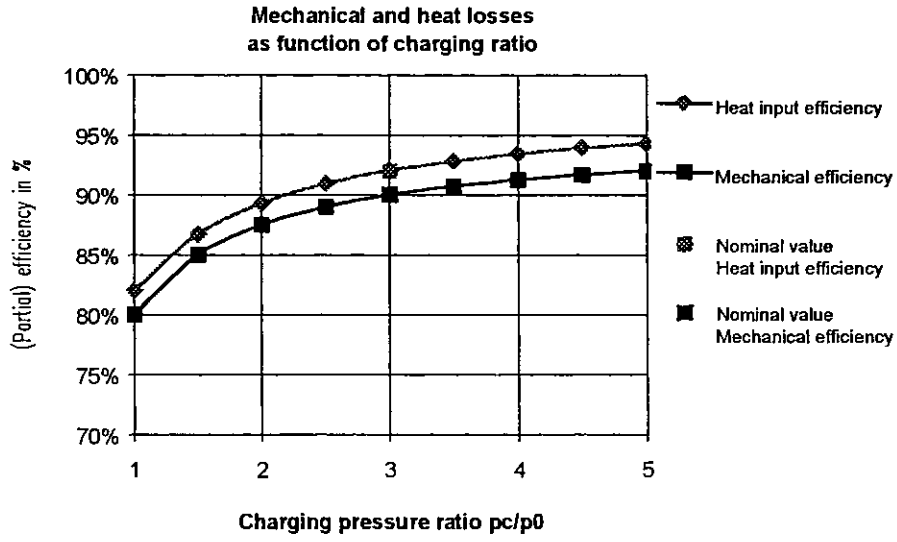


Figure 20 Mechanical and heat losses of a diesel engine expressed as partial efficiency versus charge pressure

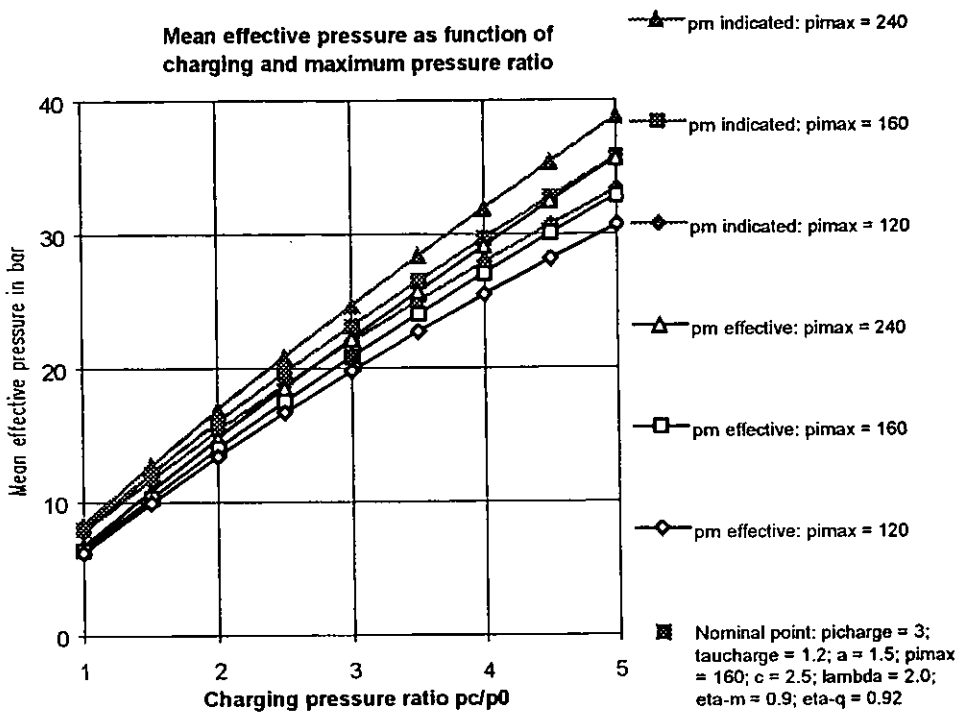


Figure 21 Ideal mean indicated pressure of Seiliger cycle compared to realistic mean effective pressure of diesel engine

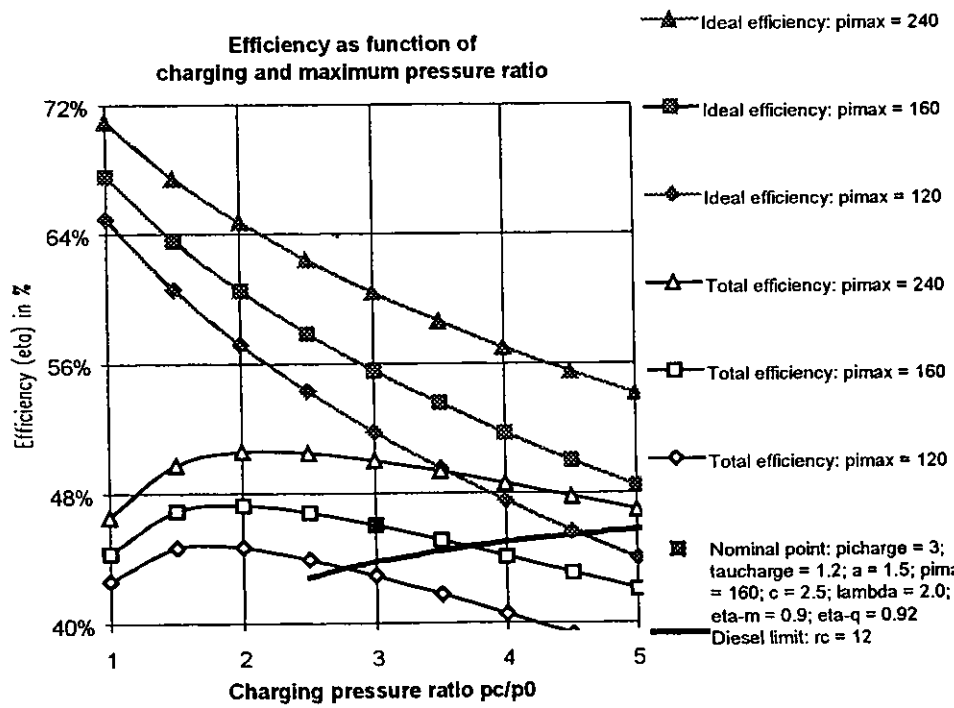


Figure 22 Ideal thermodynamic efficiency of Seiliger cycle compared to realistic overall efficiency of diesel engine

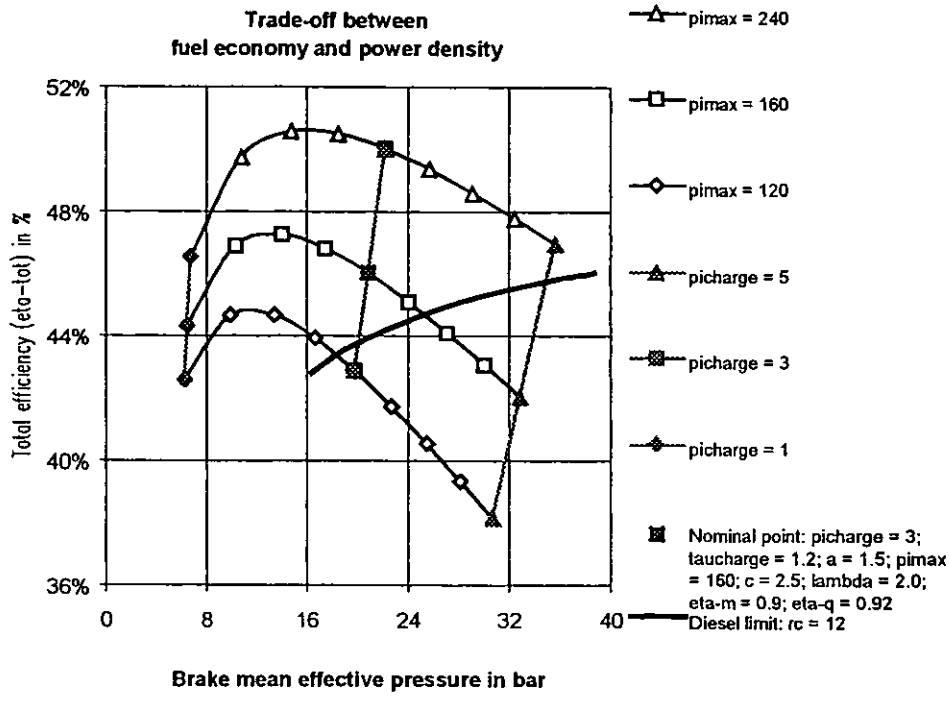


Figure 23 Trade-off of efficiency and mean effective pressure for several values of the charge pressure and peak pressure

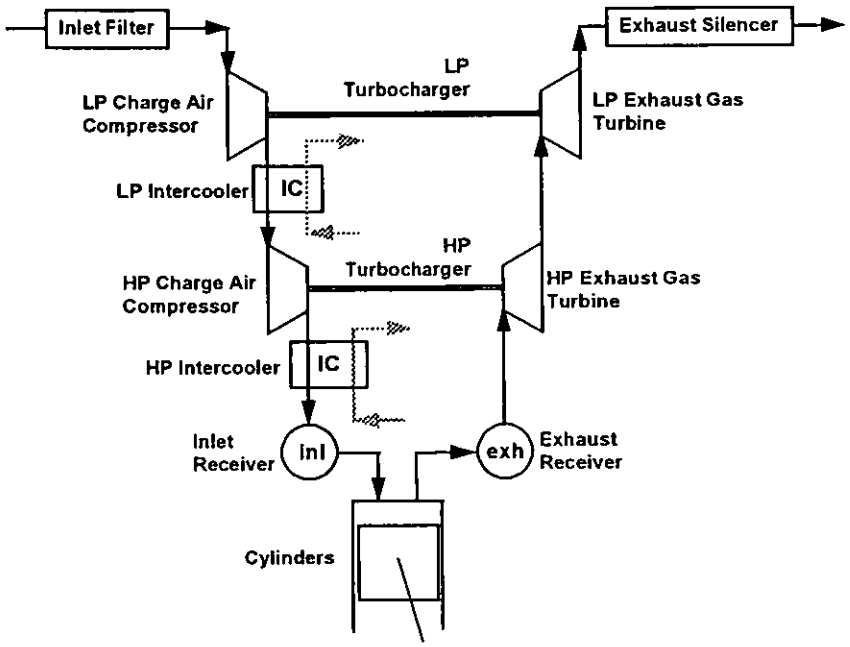


Figure 24 Principle diagram of two-stage turbocharging

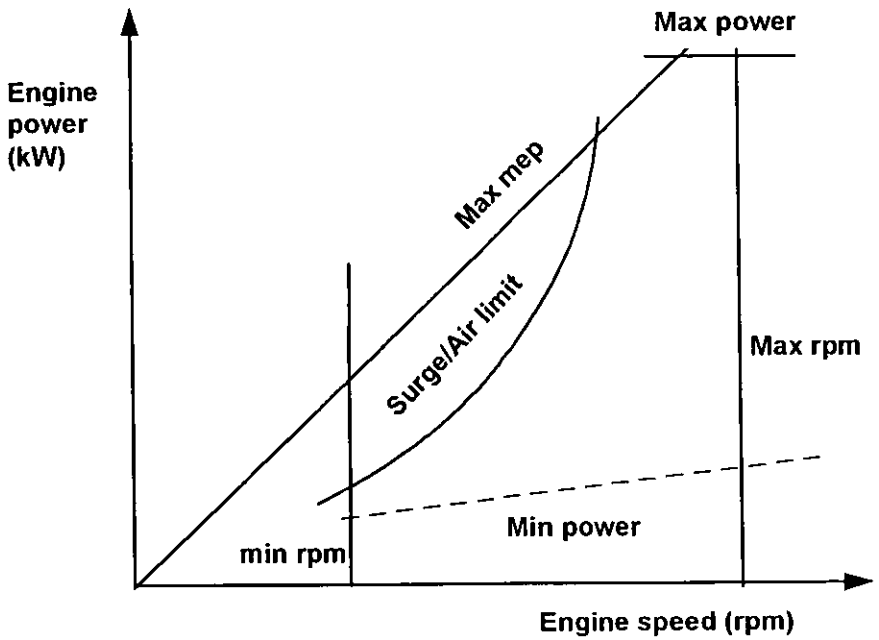
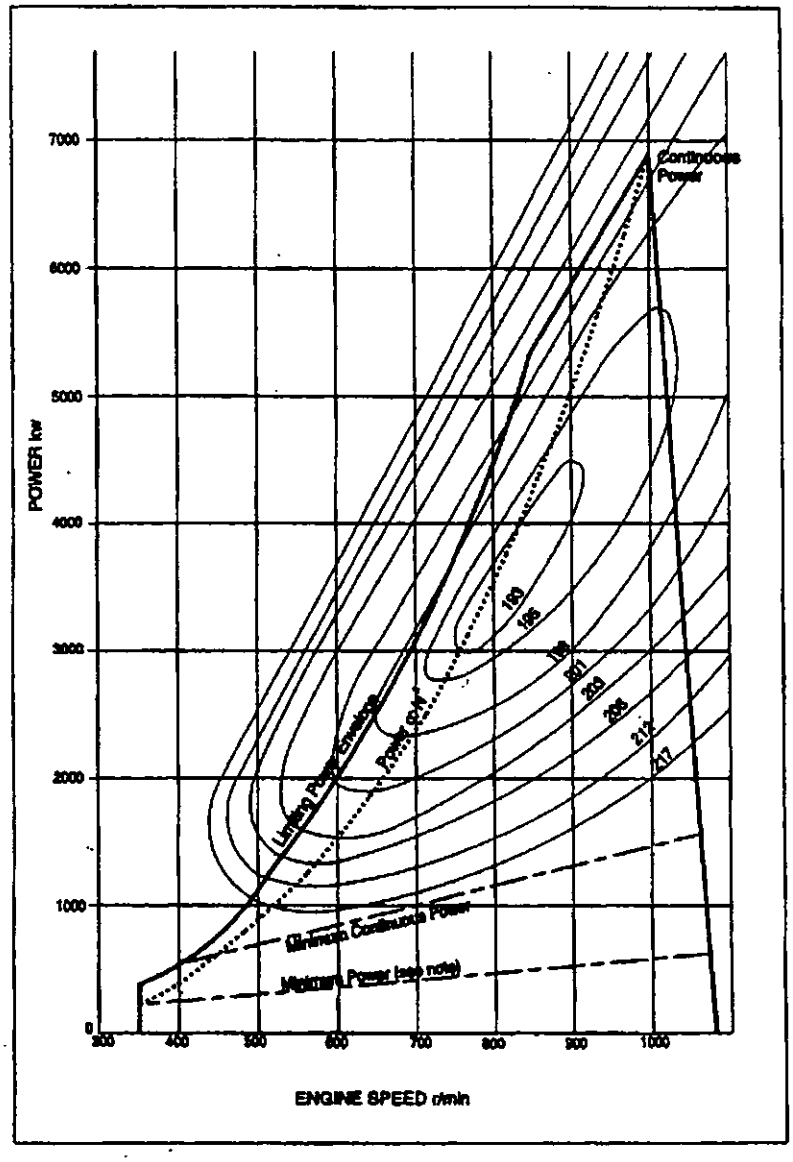


Figure 25 Limits in diesel engine power/speed characteristic

20RK270

engine power envelope



Note 1

Specific fuel consumption in g/kW h (fuel L.C.V. 42700 kJ/kg) Subject to a tolerance of +5%

Note 2

MINIMUM POWER; The minimum power is available if the power is increased above 50% full load for a period of approximately 1 hour in every 50 hours.

Note 3

This diagram is for guidance only and must not be considered binding for any specific requirement without reference to Ruston Diesels.

Note 4

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DS0149

Figure 26 Power/speed characteristic of a highly turbocharged diesel engine showing limit curve and lines of constant specific fuel consumption

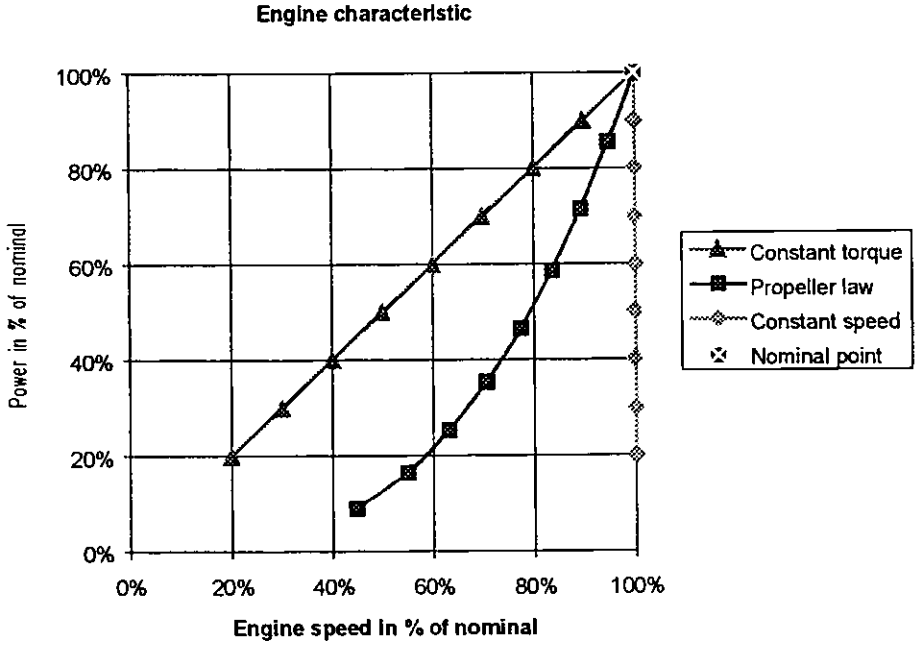


Figure 27 Three load curves in engine characteristic

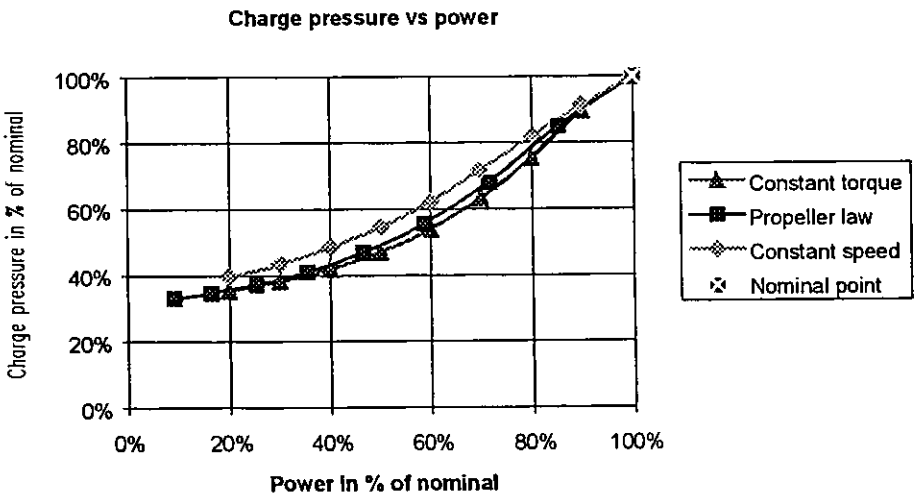


Figure 28 Charge pressure versus power for three load curves

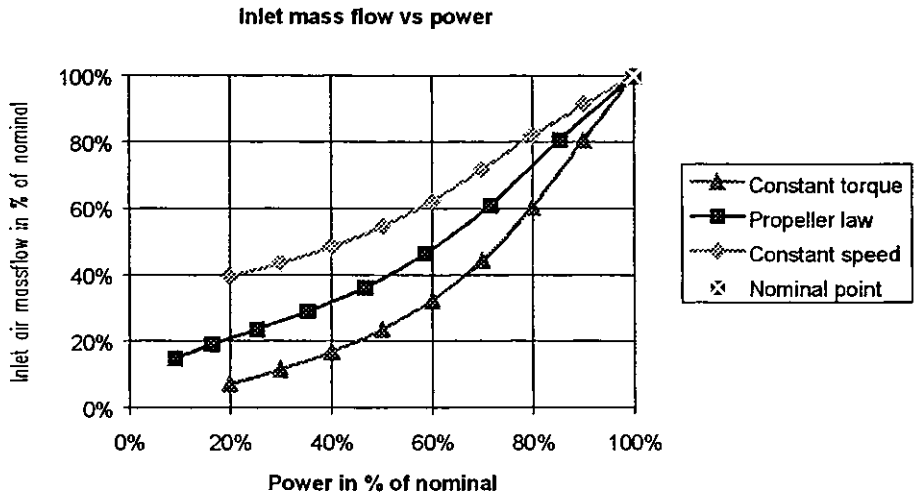


Figure 29 Inlet mass flow versus power for three load curves

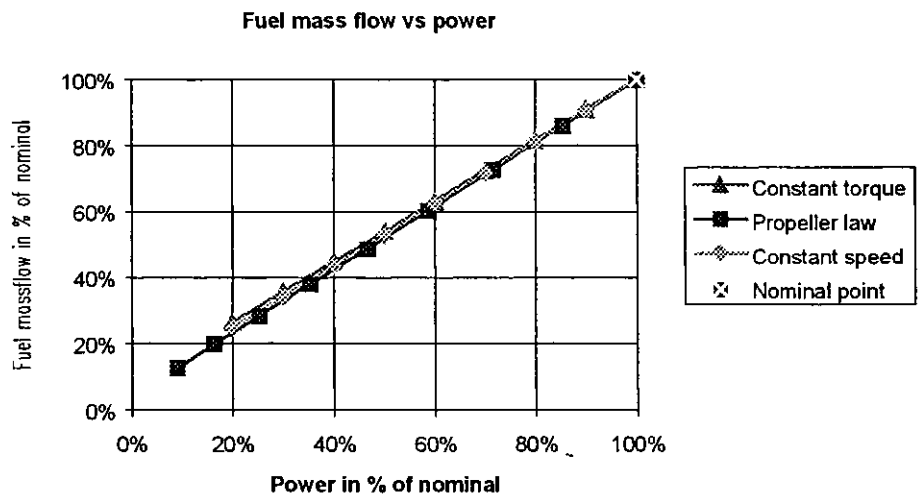


Figure 30 Fuel flow versus power for three load curves

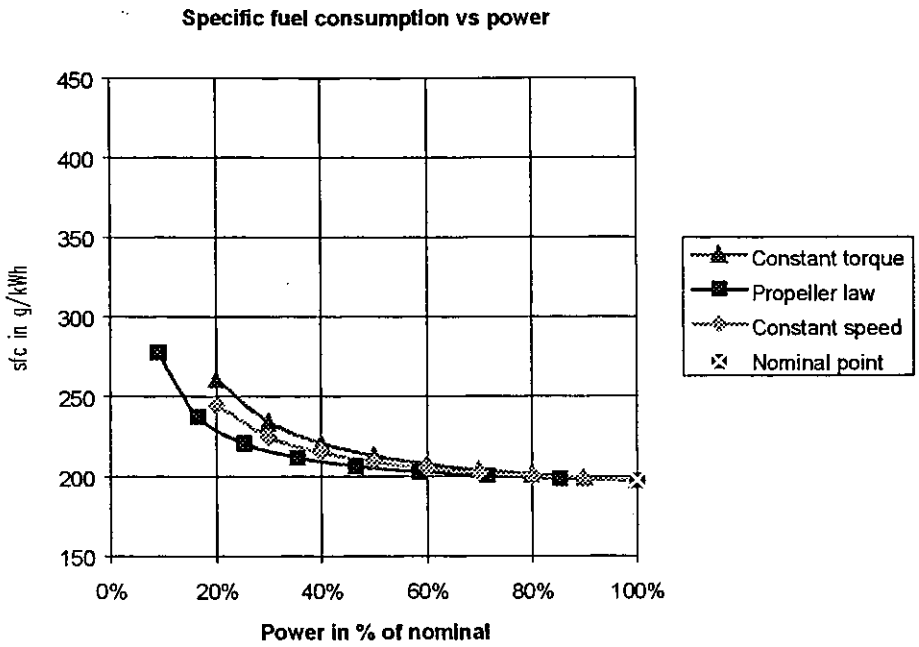


Figure 31 Specific fuel consumption for three load curves

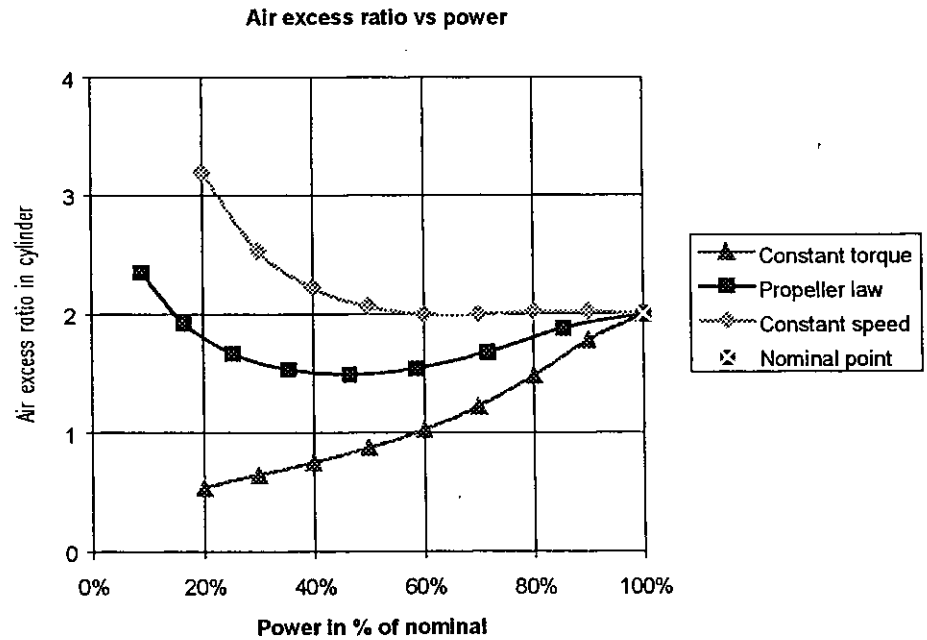


Figure 32 Air excess ratio for three load curves

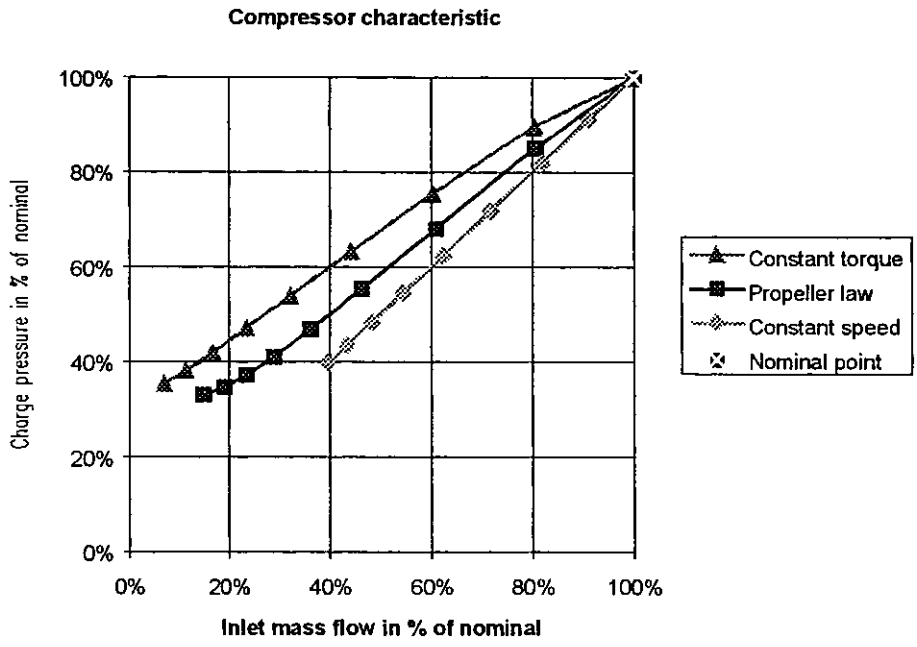


Figure 33 Three load curves in compressor characteristic

Bild 1: Drehmomentenverlauf bei verschiedenen Aufladeverfahren
Fig. 1: Tourque curve under different charging processes

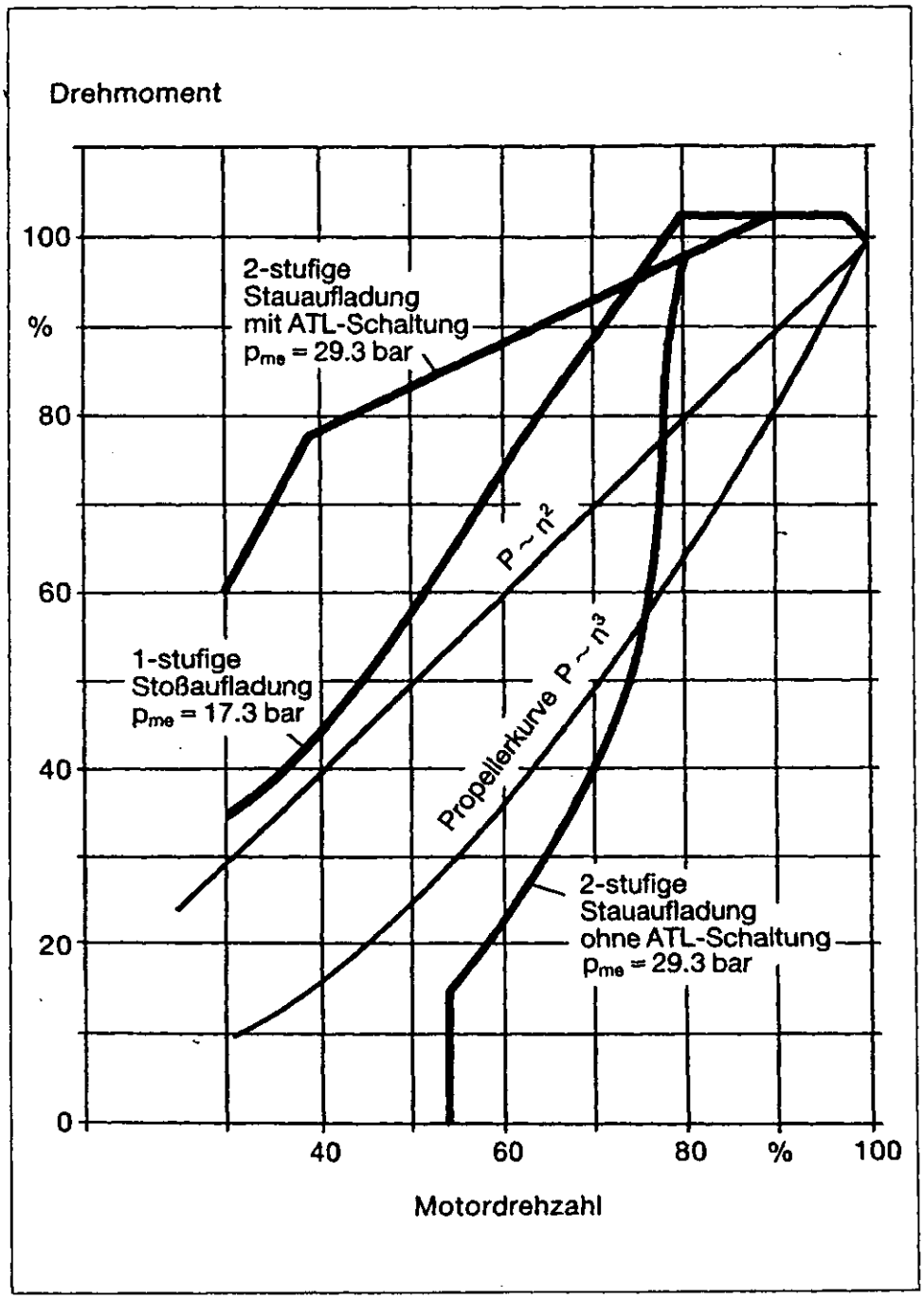


Figure 34 Several possibilities to broaden the engine characteristic of highly turbocharged diesel engines

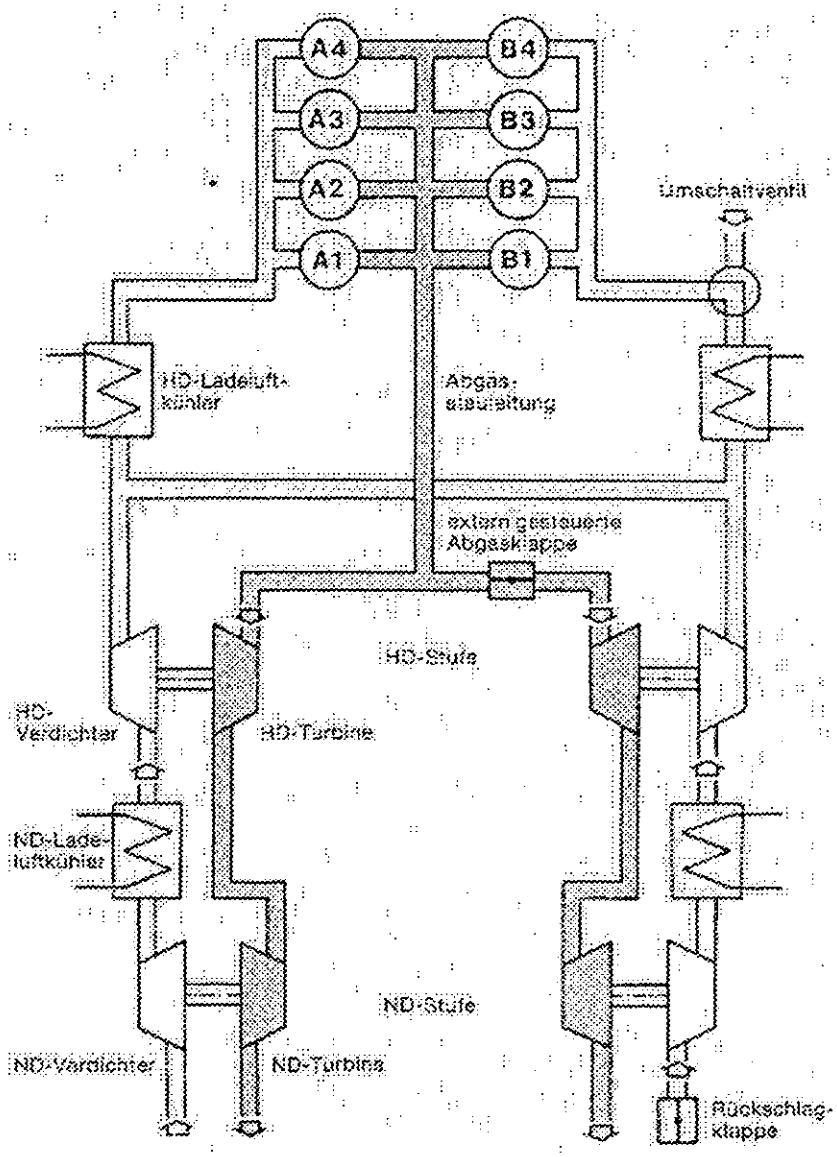
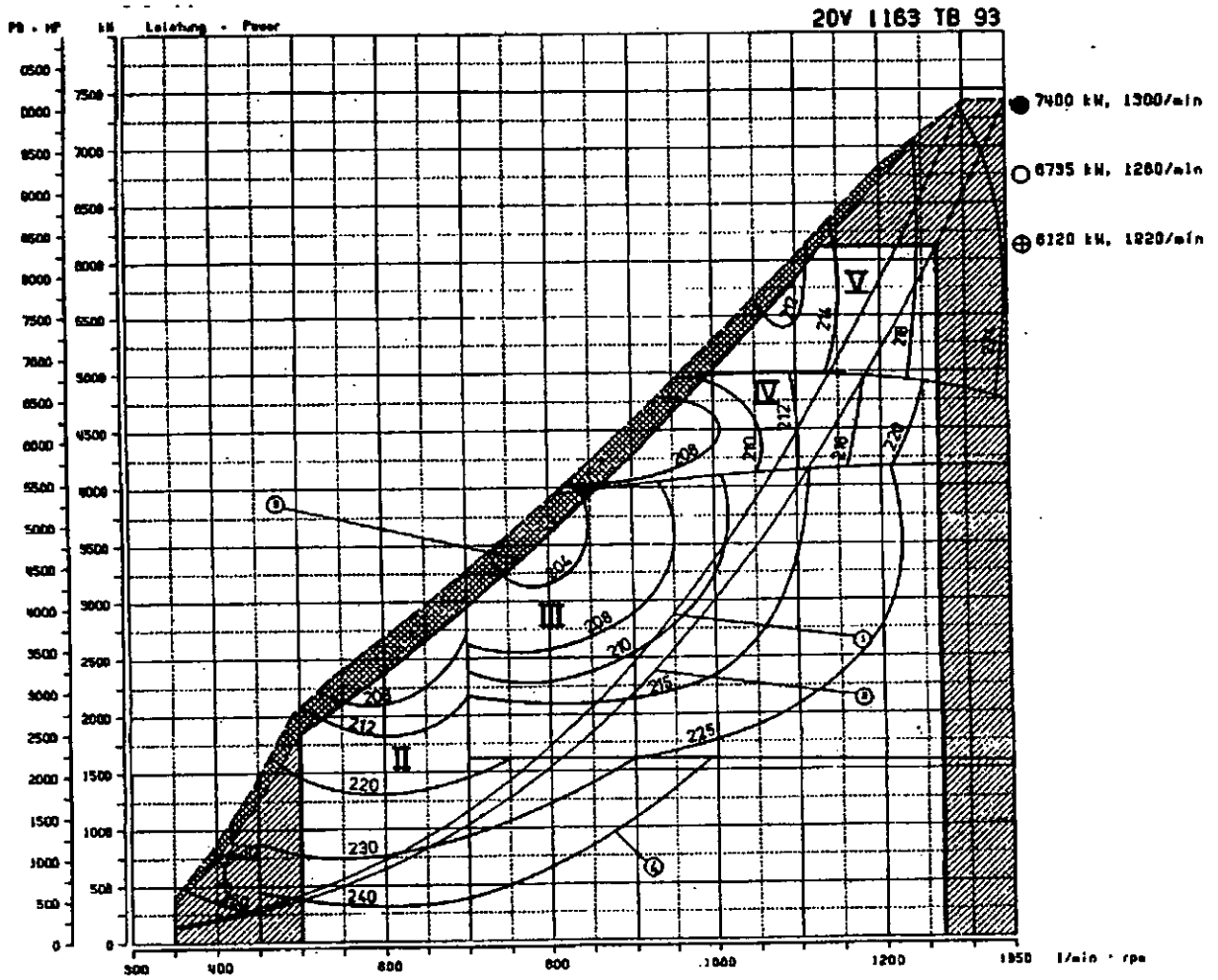


Figure 35 Principle of sequential turbocharging



Begriffsbestimmungen

- I – V Ladergruppenbetriebsbereich
- ⊕ Dauerleistung
- ○ Oberleistung
- ● Höchstleistung
- Bereich für Dauerbetrieb (zeitlich unbegrenzt)
- Bereich für Dauerbetrieb (zeitlich begrenzt)
- Leistungsreserve für Beschleunigung
- ① Propellerkurve $P = f(n^3)$
- ② Kurve für Propeller-Neuauslegung
- ③ MCR-Kurve

Bezugsdaten

Ansauglufttemperatur	27°C
Rohwasser-/Kühlmitteltemp.	27°C
Barometerstand	1000 mbar
Ansaugunterdruck	15 mbar
Abgasgegendruck	30 mbar

Spezifischer Kraftstoff-Verbrauch

④ Kraftstoffverbrauchkurven (g/kWh) Toleranz: + 5%, bei Diesel-Kraftstoff nach DIN 51 601 entsprechend MIL-F 16884 F mit einem unteren Heizwert von mind. 42 800 kJ/kg (18 390 Btu/lb)

Definitions

- I – V Operation area of charger groups
- ⊕ Continuous power
- ○ Overload power
- ● Maximum power
- Continuous operation (unlimited duration)
- Continuous operation (limited duration)
- Power reserve for acceleration
- ① Propeller curve $P = f(n^3)$
- ② Clean hull conditions
- ③ MCR-Kurve

Reference Conditions

Intake air temperature	27°C
Raw water-/coolant temp.	27°C
Barometric pressure	1000 mbar
Intake depression	15 mbar
Exhaust back pressure	30 mbar

Specific Fuel Consumption

④ Fuel consumption curves (g/kWh) tolerance: + 5%, diesel fuel to DIN 51 601 (equivalent to MIL-F 16884 F) with a minimum L.H.V. of 42 800 kJ/kg (equivalent to 18 390 Btu/lb)

Figure 36 Characteristic of a highly turbocharged diesel engine with sequential turbocharging. The engine has 5 two-stage turbocharger groups. The areas correspond to the operation of 5, 4, 3 and 2 turbocharger groups respectively.

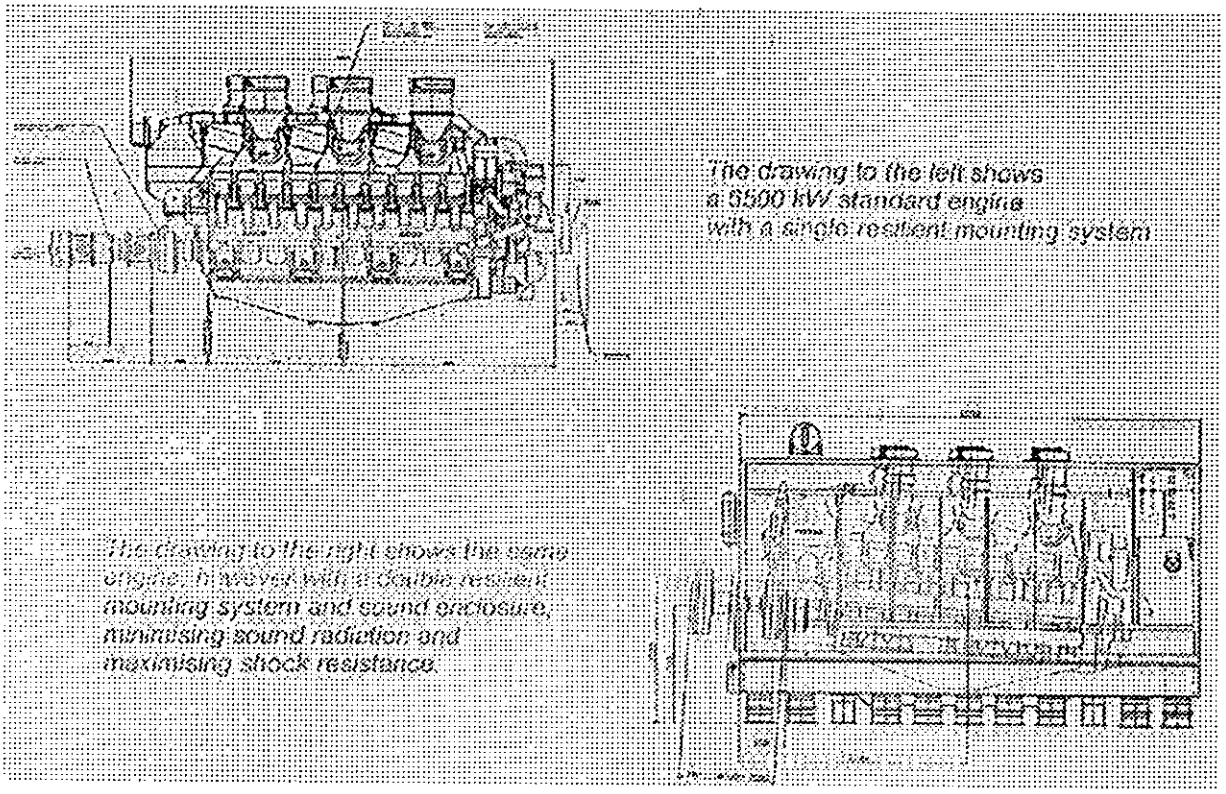


Figure 37 High speed diesel engine with single and double resilient mounting and with and without acoustical enclosure

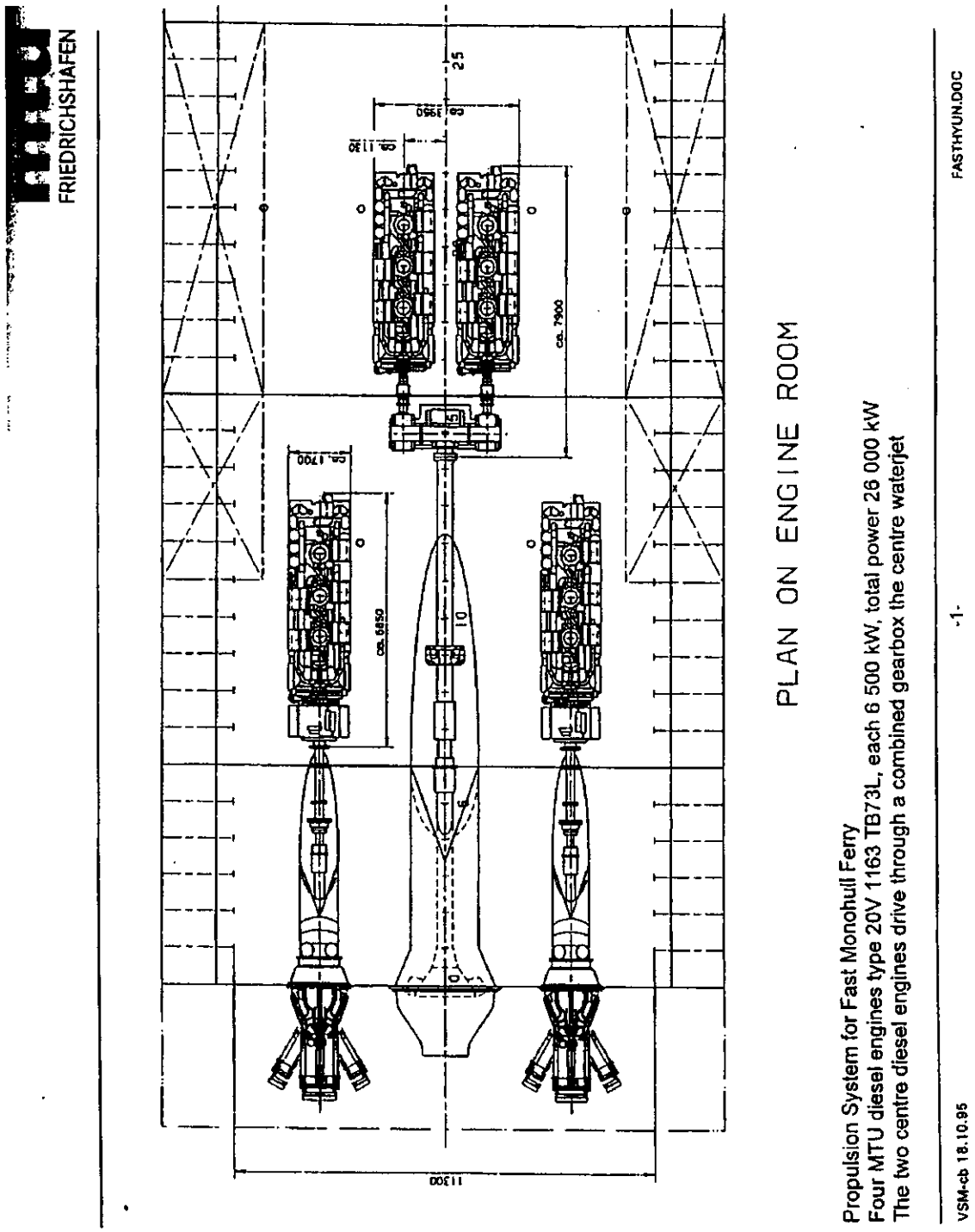


Figure 38 Installation of 4 high speed diesel engines in a waterjet driven fast ferry

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

MTU'S NEW ENGINE SERIES 2000 / 4000 IN FAST FERRY APPLICATION

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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ABSTRACT

MTU and his partner DDC are going to introduce two new engine families into the market.
For the fast ferry market they are covering the following power range:

Series 2000:	480 kW to	960 kW	at 2 100 rpm
Series 4000:	1 160 kW to	2 320 kW	at 2 000 rpm

The first part of the paper describes the technical features of the engines.

Special attention was drawn during the development phase to the customer related aspects like:

- Small dimensions
- Low weight
- High power to weight ratio
- Wide performance range
- Accessories integrated to minimize installation work
- Reduced life-cycle costs based on
  - reduced fuel consumption,
  - long TBO's
  - extended maintenance intervals
  - reduced maintenance work.

The second part of the paper shows these advantages specially for the fast ferry market.

1 - INTRODUCTION

The development of diesel engines is a never-ending process.

New ideas are continually being born and existing designs improved. In all cases, the fundamental aim is to offer customers the best possible support in solving their particular problems.

Before the first design drawings for the new engines could be produced and the first calculations made, the demands articulated "in their own language" by our customers had to be translated into technical specifications for the development teams. MTU Friedrichshafen and its partner Detroit Diesel Corporation decided to take up

the challenge to design two new high-power diesel engine families.

In recognition of the joint contribution of the two co-operative partners, the new engines are being marketed under a common name, DDC / MTU series 2000 and MTU / DDC Series 4000.

The 2000 and 4000 Series represent the successful symbiosis of the capabilities of two companies, both of which are among the leaders in the production of high-speed diesel engines. They are the rigorous combination of expertise and experience gained from decades of operation with our customers.

The 2000 and 4000 Series set new standards in the 400-2720 kW output range. They are intended for use in ships, railway trains, heavy trucks and stationary power units.

This paper describes their advantages in ship application.

2 - TECHNICAL FEATURES SERIES 2000

Fig. 1 The 8, 12 and 16-cylinder Series 2000 engines owe the basic structure of their main components and subassemblies to the Mercedes-Benz volume production truck engines OM 500. As a result, they have profited from the significantly broader development base of a design built in large numbers. MTU and DDC could thus concentrate on bringing their specialist development expertise and experience to bear in all those areas where the specific requirements of the particular application demanded it, or where the greater number of cylinders as compared with the commercial vehicle version necessitated more extensive design input.

Fig.2 Initial market research indicated that the new engine would need a capacity of approximately two litres per cylinder.

Fig. 3 The power setting should be positioned within the output range of between 400 kW and 1 340 kW.

Fig. 4 The exceptionally rugged, torsionally rigid, but nevertheless lightweight grey cast iron crankcase holds the wet cylinder liners, the crankshaft and central camshaft bearings and the injection pumps.

It also contains the main oil and coolant circulation channels along with the attachments for the engine mountings, the pumps and various engine subassemblies

All engine and external auxiliary systems are driven by a system of gears which includes all pumps required like engine water pump, raw water pump and fuel pump.

The four-valve cylinder heads with centrally positioned injector in combination with the specially designed Aluminium pistons with oil cooling create the optimum conditions for low-emission and low-consumption combustion. The injection system consists of a separate high-pressure injection pump for each cylinder.

Those injection pumps are controlled by an electromagnetic valve and inject the atomised fuel into the cylinder at a pressure of 1800 bar. This system with its electronically controlled optimised-characteristic injection parameters, provides a high degree of flexibility with regard to optimisation of combustion and adaption to a wide variety of engine operating requirements.

Fig. 5 In the exhaust and turbocharger system we have integrated the full spectrum of the latest technological advantages already in use for a number of years on many MTU engines. The triple-walled exhaust system with an outer water-cooled aluminium casing ensures that the surface temperatures do not exceed the permissible levels at any point whilst also providing absolute gas tightness.

Inside that cooled outer casing there is another, high-temperature-resistant exhaust pipe designed to ensure, that the flow of exhaust gases comes into direct contact with the cooled outer case as little as possible.

This retains valuable energy within the exhaust gases for the turbochargers. In addition the heat dissipation via the cooling system is reduced.

Fig. 6 Available for the first time on engines of this size, we have introduced our well known sequential turbocharger system. This system is used in order to provide high torque at low engine speeds. It enables rapid acceleration without providing black smoke emission.

The performance diagram for the 16V 2000 M70 shows the two areas with different numbers of turbochargers in operation.

For fast passenger vessels we can provide a specific fuel consumption figure below 210 g/kWh.

Fig. 7 A comparison is made in some important operating parameters like fuel consumption, exhaust temperature and smoke emissions for the high performance rating (1DS) and the max. continuos rating (1B) which is relevant for fast passenger vessels.

Fig. 8 The practice-proven split-circuit coolant system for engine and intake aircooling complements the turbocharger system and acts as an "intelligent" cooling system keeping the engine coolant, oil and intake air at the optimum temperature for all operating conditions.

When idling or at low load, the temperature of the intake air is raised in order to ensure smooth and complete combustion without providing white smoke emission. At full load, maximum cooling of intake air maximises engine performance.

Fig. 9 The arrangement drawing shows the compact design of the 16V 2000-engine.The width is only 1400 mm and the height above crankshaft approx. 800 mm. The different options for the exhaust connection (horizontally or vertically) to the ships exhaust pipework gives greatest freedom for installation work.

Based on our philosophy all accessories are installed on the engine and tested together with the engine. Therefore the installation work provided by the shipyard can be kept to a minimum. In the most powerful version the 16V 2000 has a weight to power ratio of 2.56 g/kW.

For the fast ferry application we have a factor of 3.6 kg/kW.

These figures include all accessories and the flanged-on gearbox.

3 - TECHNICAL FEATURES SERIES 4000

Fig. 10 The demand for innovative diesel engine designs in the power range upwards the Series 2000 was the motivating force behind the development of the Series 4000 with a bore of 165 mm and a stroke of 190 mm resulting in a swept volume per cylinder of 4.06 l.

The engine family is available in 8, 12 and 16 cylinder versions covering an output range from 700 to 2720 kW.

Development of the engine followed the guiding principle of "maximum reliability" and greatest possible degree of service friendliness. To that end, a combination of tried and tested design concepts like the sequential turbocharging system and innovative and trend-setting solutions like the new injection system was used and adapted to the product according to customer demands.

Fig. 11 Reliability starts with the design of the crankcase. It was deliberately developed with maximum strength and optimum rigidity in mind. Integration of the main lubrication and coolant circulation channels as well as the engine mounting fixings also reduced the number of separate components and thereby the complexity of the system.

Generously proportioned access ports in the crankcase make it possible for all servicing operations to be carried out without having to remove the engine, even in cramped conditions.

Fig. 12 The crankshaft and the other relevant parts are designed to withstand maximum combustion pressures of up to 200 bar. The crankshaft is machined all over with bolted on counterweights. The use of the most reliable and wear-resistant sleeve bearings which are currently available are an important factor contributing to longer service intervals.

Fig. 13 The pistons with a steel crown and an aluminium skirt are fitted with exceptional wear-resistant chrome-ceramic coated piston rings which in combination with the plateau-honed cylinder liners guarantee low oil consumption and long service life.

The four-valve cylinder heads with integrated coolant channels have a specially stiffened base. The optimised combustion chamber and the bowl of the composite piston also enable minimal emission levels and low fuel consumption levels to be achieved.

All application-specific versions of the Series 4000 are capable of fuel consumption figures previously only attainable with medium-speed diesel engines.

These exceptional fuel consumption levels have only been made possible by the use of the completely new common-rail injection system.

Fig. 14 Whereas previously, on conventional systems, the injection pressure was a variable that was virtually incapable of variation over the engines operating range, the common-rail system offers the advantage of infinitely adjustable and controllable injector pressure and timing.

Fig. 15 With the common rail system also at low engine speed a high injection pressure can be provided which results in advantages specially in reducing of smoke emissions. To that end, the high-pressure pump and the electronically controlled injectors are fully integrated in the electronic engine management system.

By doing away with the separate injector pumps, the load on the camshaft and the gearing system is reduced. The removal of the mounting holes for the separate injector pumps also increases the rigidity of the crankcase.

Fig. 16 On the "front" end of the engine there is an integral service block positioned for best accessibility to suit for ship application which significantly simplifies all standard servicing operations.

For example, the standard oil centrifuge is mounted on top, which acts as a bypass oil cleaner, making it possible to extend service intervals, thereby, providing longer periods of uninterrupted engine operation.

Fig. 17 Also the Series 4000 has the benefit of the triple-wall cooled exhaust system which has proved itself over millions of hours of service. It is positioned centrally between the two rows of cylinder of the Vee-configuration engine.

As described with the 2000-engine we are also using for the 4000 the sequential turbo-charging system to achieve a wide power band with the advantages of excellent acceleration capabilities and high efficiency characteristics which results in low fuel consumption over the whole performance range.

Efforts to achieve high levels of turbocharger efficiency have been assisted by the use of the split-circuit cooling system for engine coolant and intake air.

Highest availability with reduced maintenance cost is a main requirement coming from operators of fast ferries.

Of the conventional maintenance concepts, MTU prefers the preventive approach.

In the case of preventive maintenance, the TBO (Time Between Overhauls) refers to the period of time in which no wear related failures will occur. In our case, this represents the time to the first scheduled "W6" major overhaul.

MTU has been running tests on the wear characteristics and reliability of power units since the

beginning of the Eighties.

Simulation calculations, model research and assessment of the condition of all components relevant to performance during "W6" overhauls have provided the technicians with reliable information on the wear of engine components.

This information now enables us to make precise predictions of the service life of engines and engine components.

Fig. 18 Service life and TBO are dependent on the engine workload. Load profiles can now be adapted and updated at sea according to the actual operating profiles. For this purpose, the electronic engine governor has an integrated computer which calculates the level of wear of the running gear components on the basis of the actual load profile under operational conditions, thus enabling maintenance intervals to be optimised.

Thanks to the optimised design of the new 4000-Series, the TBO can be increased from 9000 hrs (396-engines) to 15 000 hrs based on an operation profile of a fast ferry which means 75 % time with 100 % power and 25 % time with less than 15 % power. This increase results in a significant reduction of maintenance costs.

4 – ELECTRONICS

MTU engines for marine applications are provided with an Electronic Control System matched to special marine requirements. This is also valid for the new 2000 and 4000 engine series. The high functional efficiency and simple system design with plug connectors and pre-fabricated system cables for engine installation makes incorporation into ships an easy operation. This system ensures optimized engine functioning under all operating conditions.

Economical engine operation with low fuel consumption and minimum exhaust emission over the complete load range are guaranteed by the electronic governor.

Fig. 19 The products are designed in accordance with classification society guidelines, and are divided into the following product families:

MTU Digital Engine Control (MDEC):

ECU Engine Control Unit builds the heart of the Engine Control System and is responsible for the following functions:

- Regulation, Control and Monitoring
- Integrated Safety System
- Integrated Test System (Watchdog)

EMU Engine Monitoring Unit is additionally used for:

- Separate Safety System
- · Interface for additional sensors according to classification regulation

LOP Local Operating Panel with integrated local control and monitoring for installation in engine room for:

- Monitoring and operation of the engine and gearbox locally
- Display of operating values, status and fault messages.
- Start/Stop/Emergency Stop processing
- Interface between the assemblies EMU/ECU and GMU/GCU
- Interface to peripheral assemblies Monitoring and Control System MCS, RCS

GCU/GMU Gearbox Control and Monitoring Unit:

In addition to the engine monitoring and control system, the gearbox can be monitored and controlled by the GCU/GMU Gearbox Control and Monitoring Unit which provides the same features as the ECU/EMU.

Fig. 20 Remote Monitoring and Control Systems MCS-5

Flexible automation, offered as a customised, competition-orientated accessory, provides the basis for the market success to those systems designated MCS (Monitoring and Control System). The MCS product range is employed in marine applications and is characterized by extremely modern, modular, intelligent subassemblies of small size and with a minimum of wiring complexity (data bus communication).

The graph shows the monitoring and control system with color graphic screen, for the display of all information of the diesel engine, gearbox. which can be easily extended for displaying information of the propeller and other shafts.

Even the extension, to monitor and control all information of the complete ship, like gensets, Tanks, Bilge's, etc can easily be incorporated into the MCS-5 system.

5 - SUMMARY

Two new engine families were introduced into the market end of 1997.
Mean while we have delivered till end of 1999

Series 4000	576 in total
Series 2000	1380 in total.

For ship application

Series 4000	112
Series 2000	843

Special attention was drawn during the development phase to the customer related aspects like:

- Small dimensions
- Low weight
- high power to weight ratio
- Wide performance range
- Accessories integrated to minimise installation work
- Fulfilment of IMO-Nox-rules

- Reduced life-cycle-costs based on
  - reduced fuel consumption
  - long TBO's
  - extended maintenance intervals
  - reduced maintenance work

MTU's New Engine Series 2000 / 4000 in Fast Ferry Application

Dipl. Ing. Gerhard Haussmann
Director
Marine Application
MTU Motoren- und Turbinen-Union
Friedrichshafen GmbH

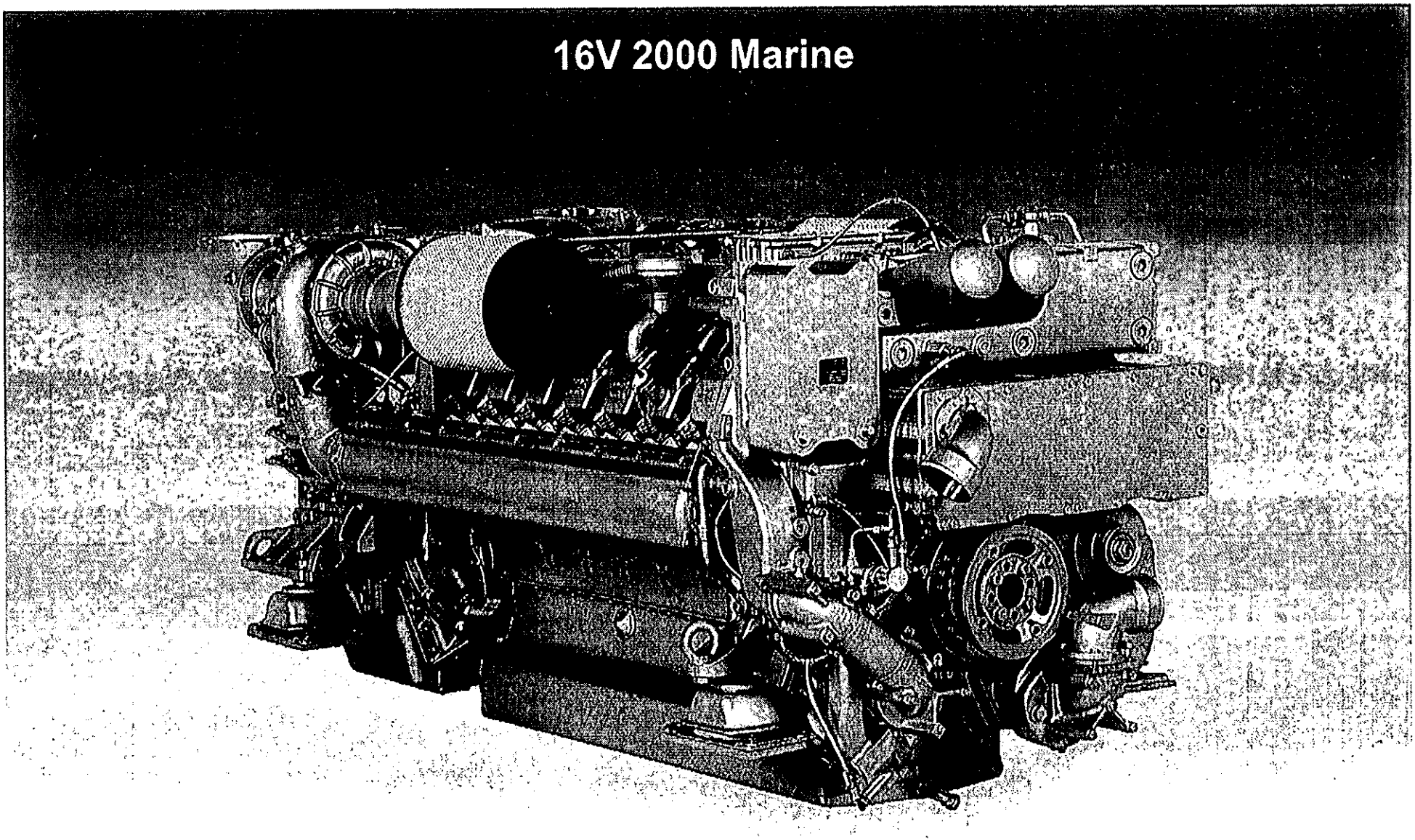
XXXIIIrd Wegemt School

Fast Ship Propulsion Plants

May 22nd - 26th, 2000

Genova, Italy

MTU's New Engine Series 2000 / 4000
in Fast Ferry Application



Technical Features
Series 2000

Displacement, cylinder	1.99 l
Displacement, total	15.92 l 23.88 l 31.86 l
Bore	130 mm
Stroke	150 mm
Crankcase	cast iron
Direction of rotation	c.c.w. (facing main PTO end)
Valves per cylinder	2 inlet, 2 exhaust
Cyl. arrangement	90° V
Compression ratio	15.2
Mean piston speed	11.5 m/sec.
M.E.P.	21.7 bar

Series 2000
Power Setting

8V 2000 M90	672 kW	2300 rpm
12V 2000 M90	1007 kW	2300 rpm
12V 2000 M91	1103 kW	2300 rpm
16V 2000 M90	1343 kW	2300 rpm

Max. Continuous Rating **

8V 2000 M70	480 kW	2100 rpm
12V 2000 M70	720 kW	2100 rpm
16V 2000 M70	960 kW	2100 rpm

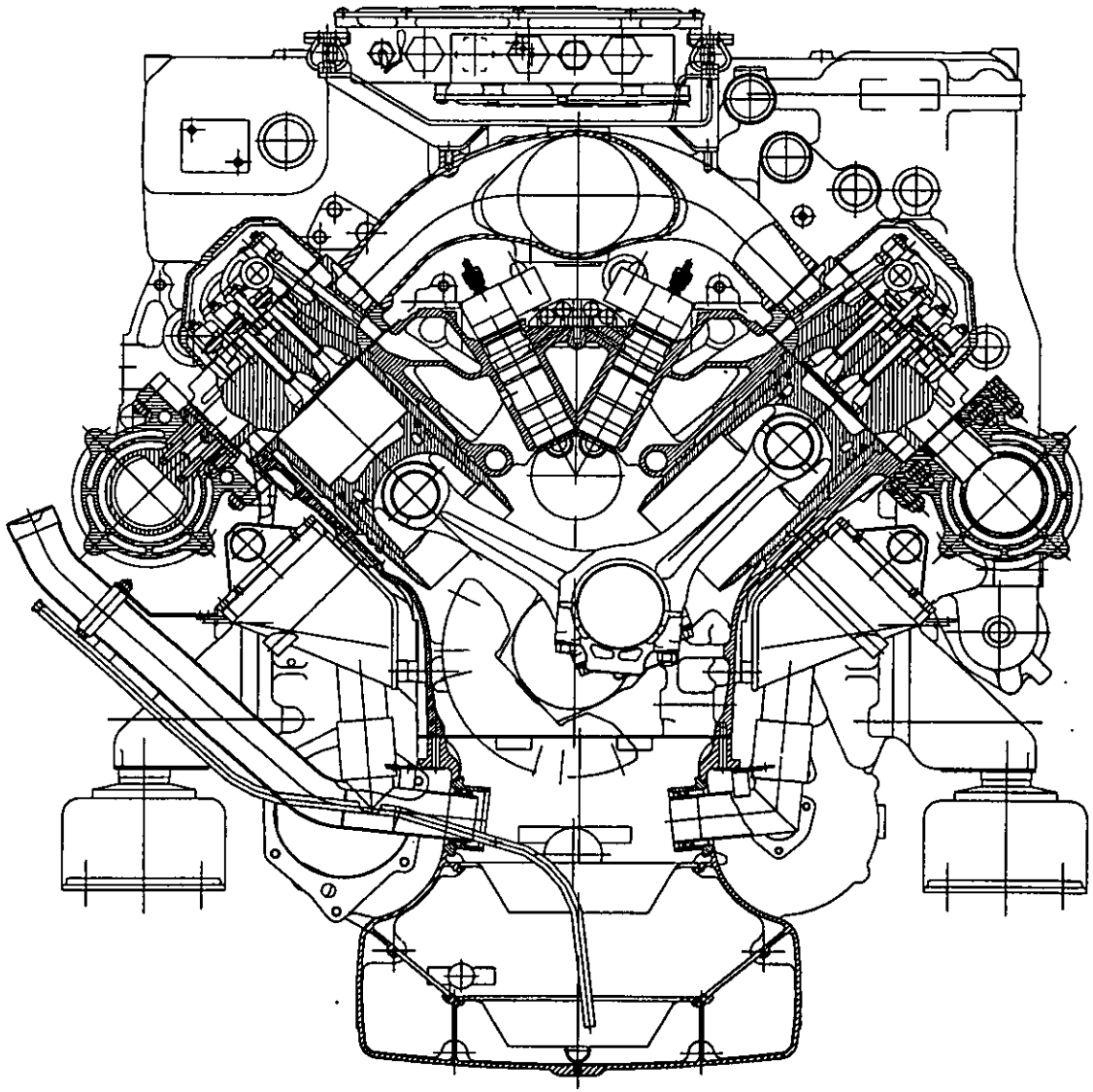
Unrestricted Continuous Rating **

8V 2000 M60	400 kW	1800 rpm
12V 2000 M60	600 kW	1800 rpm
16V 2000 M60	800 kW	1800 rpm

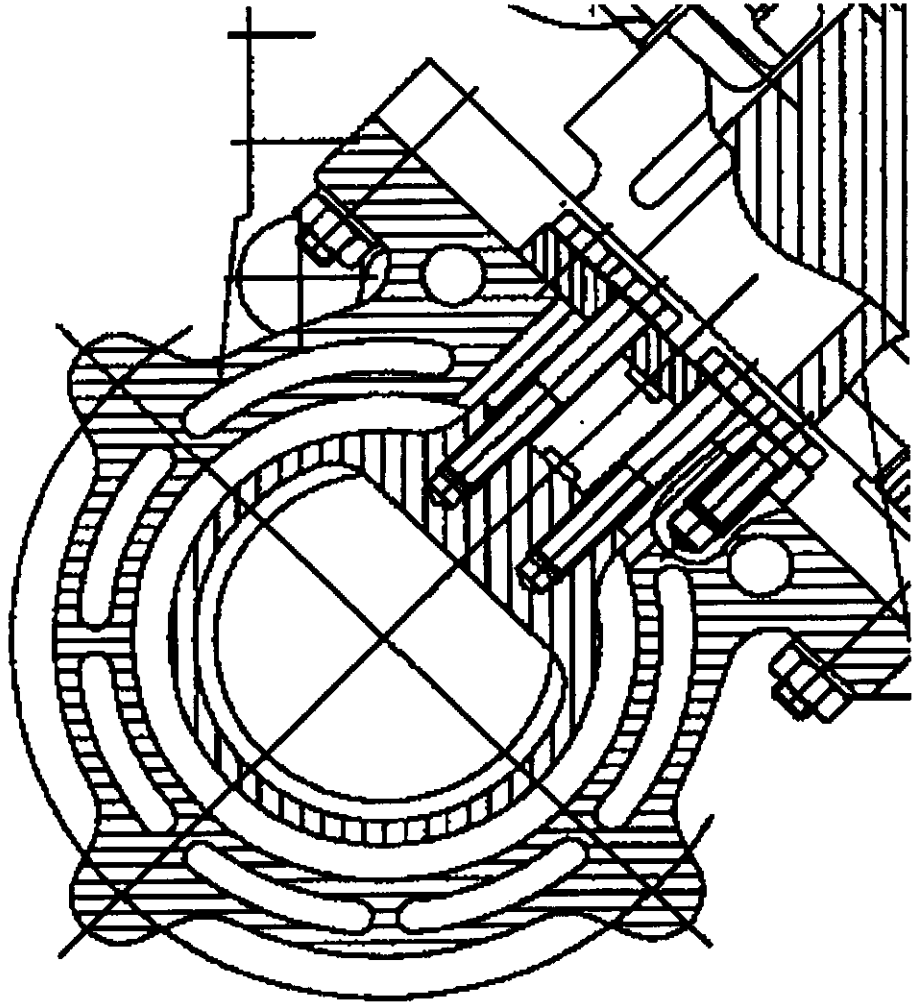
Ambient conditions: * 25 °C air intake
25 °C raw water

** 45 °C air intake
32 °C raw water

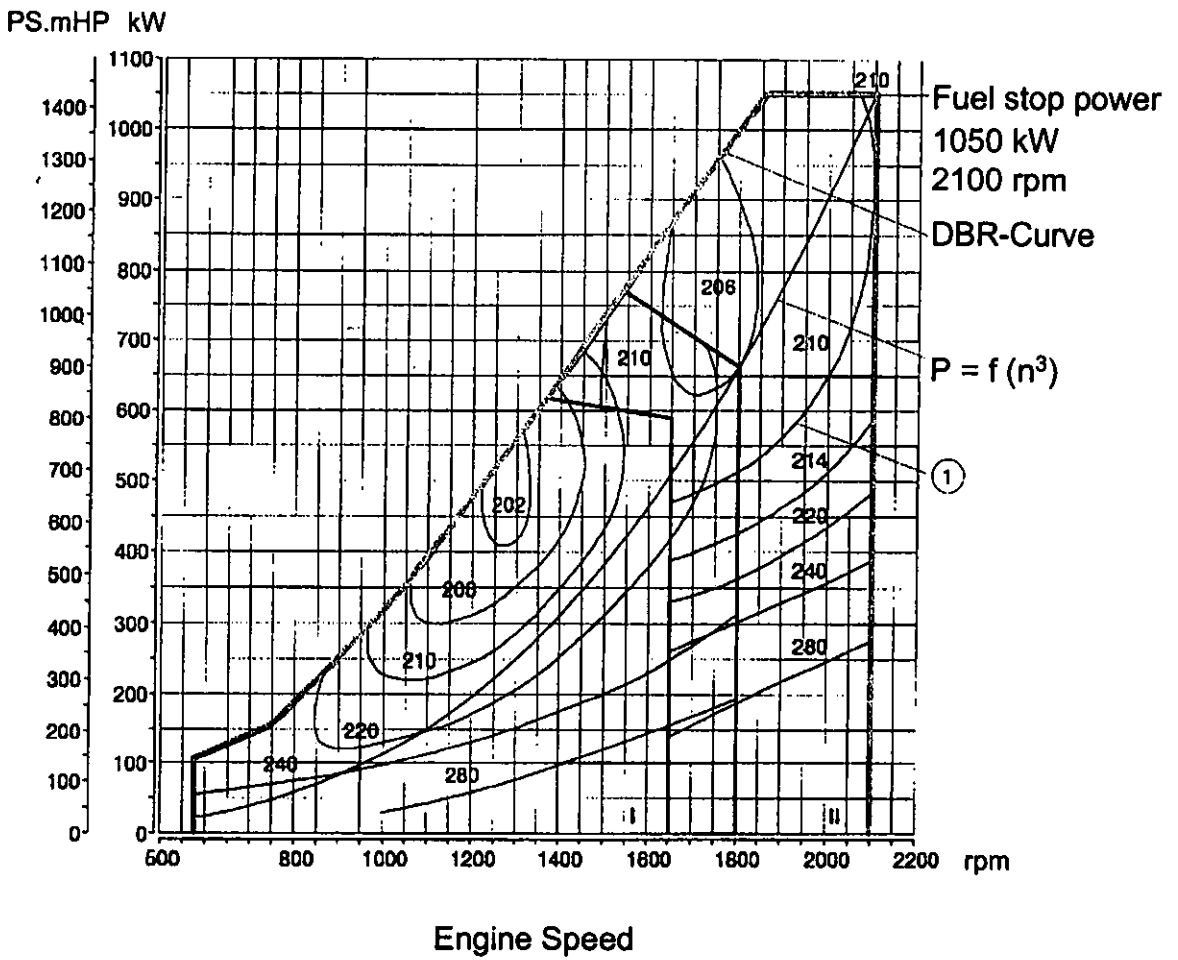
16V 2000
Engine Cross Section



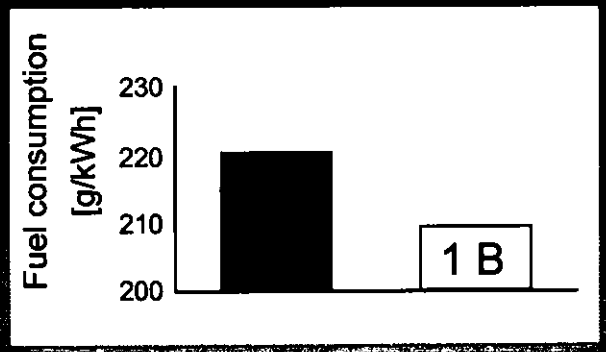
16V 2000
Triple Walled Exhaust Pipework

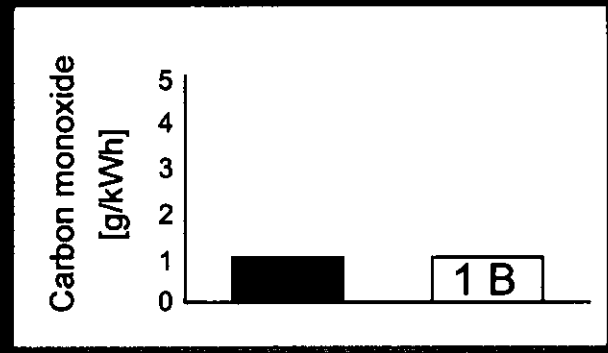


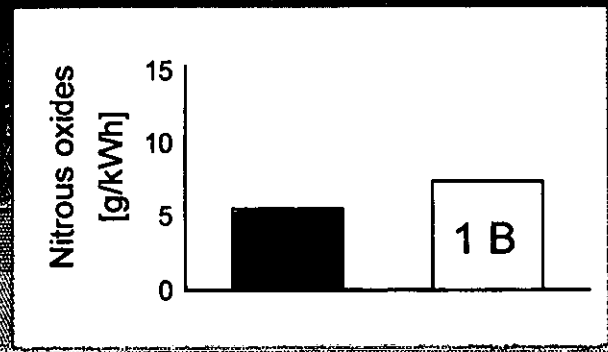
16V 2000 M70
Performance Diagram

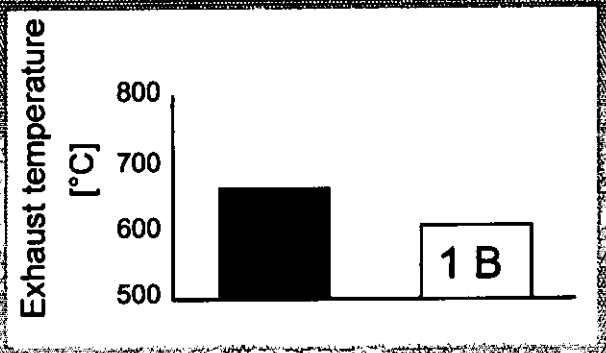


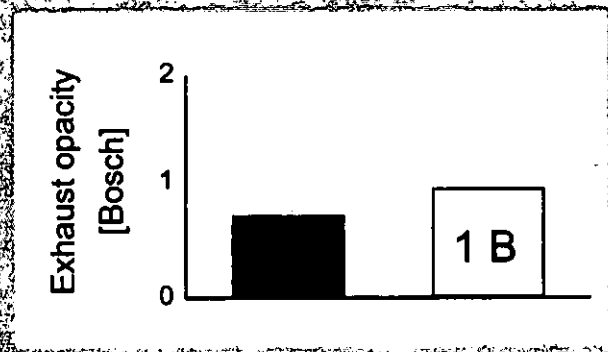
Series 2000
Operating Parameter



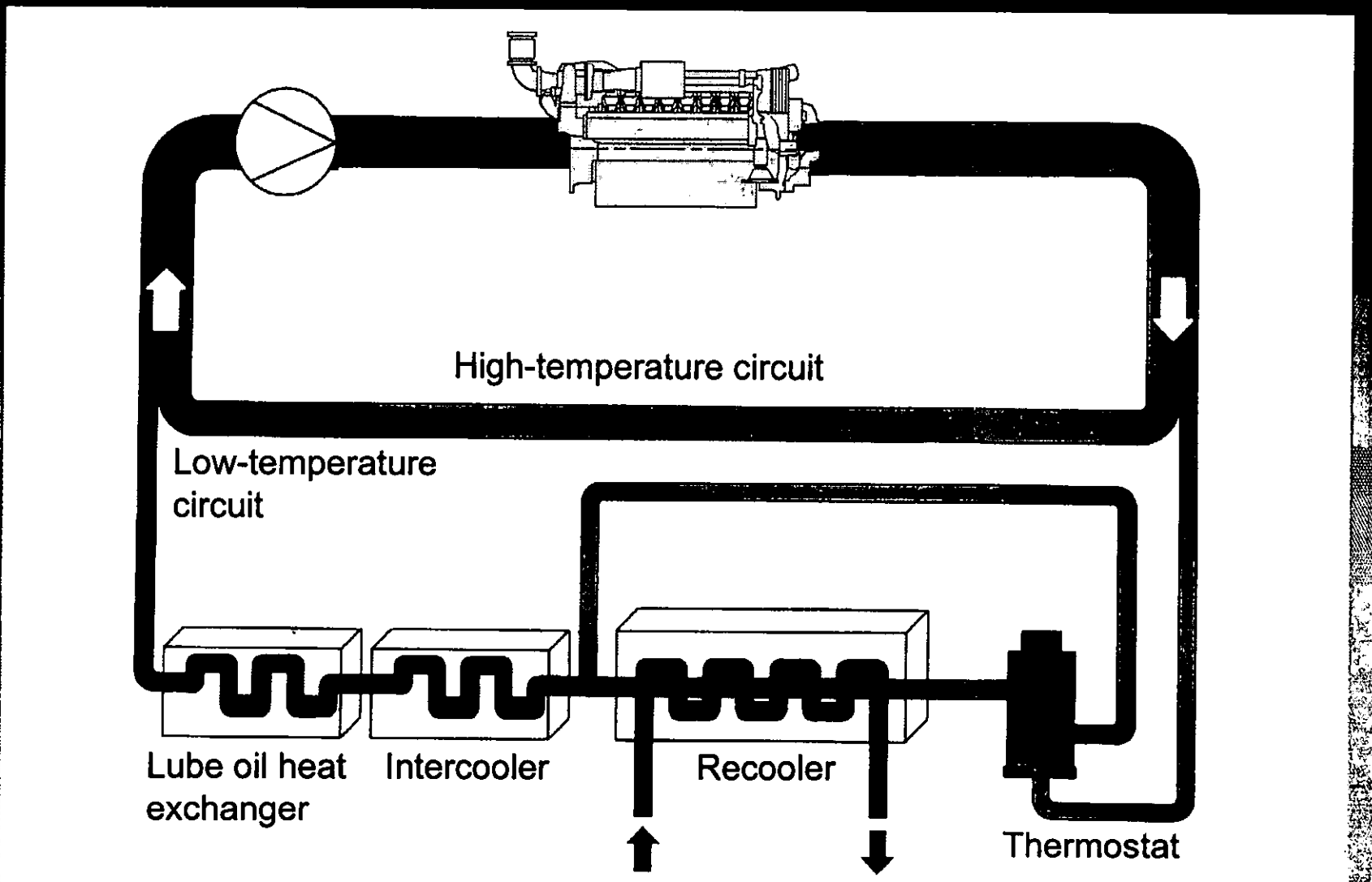






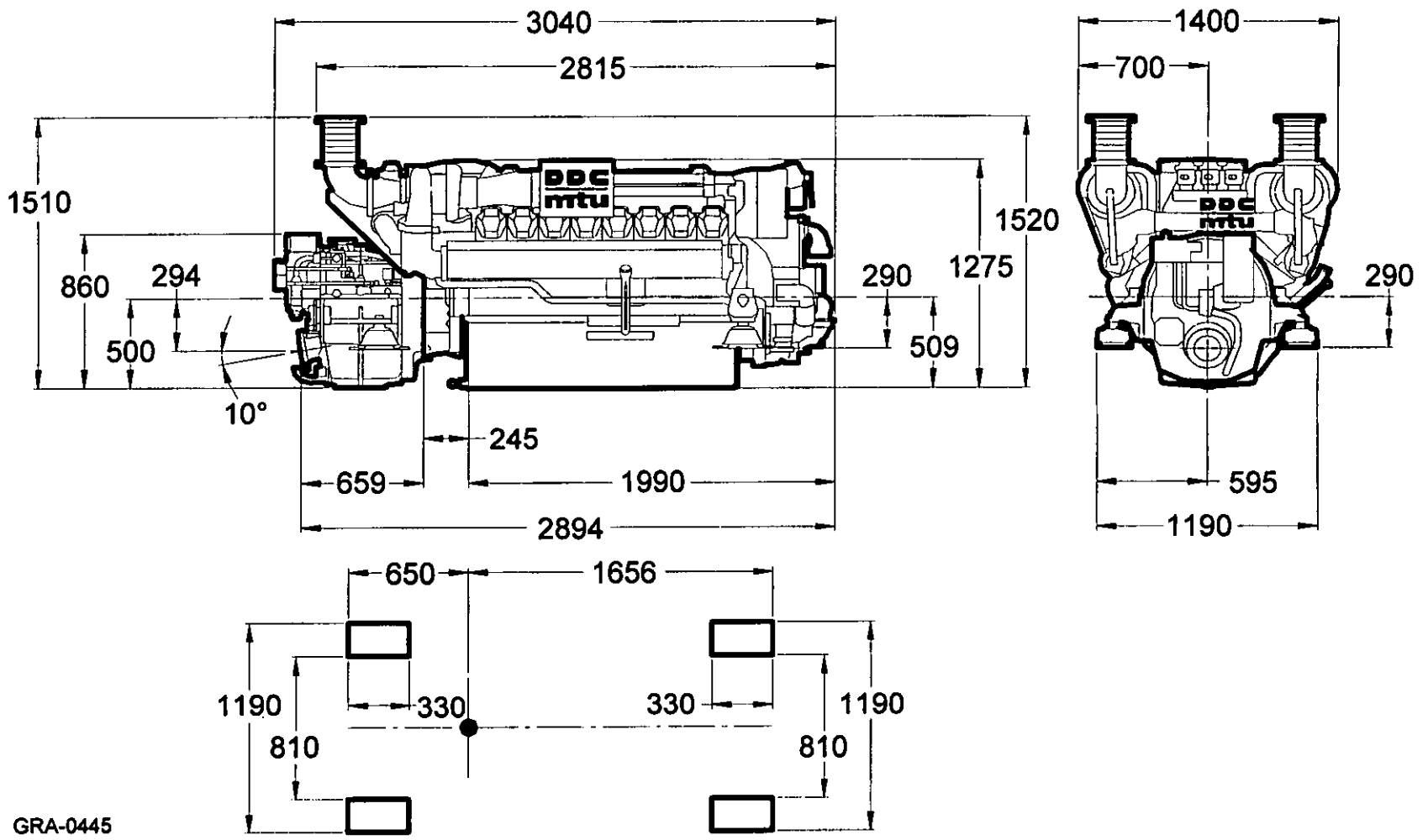


Coolant system, functional diagram

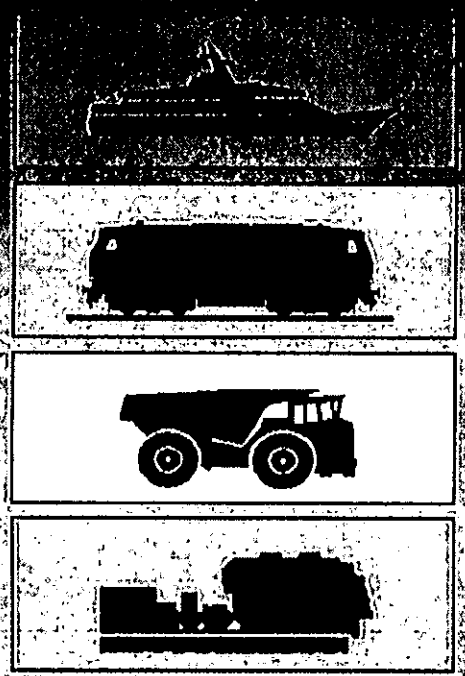


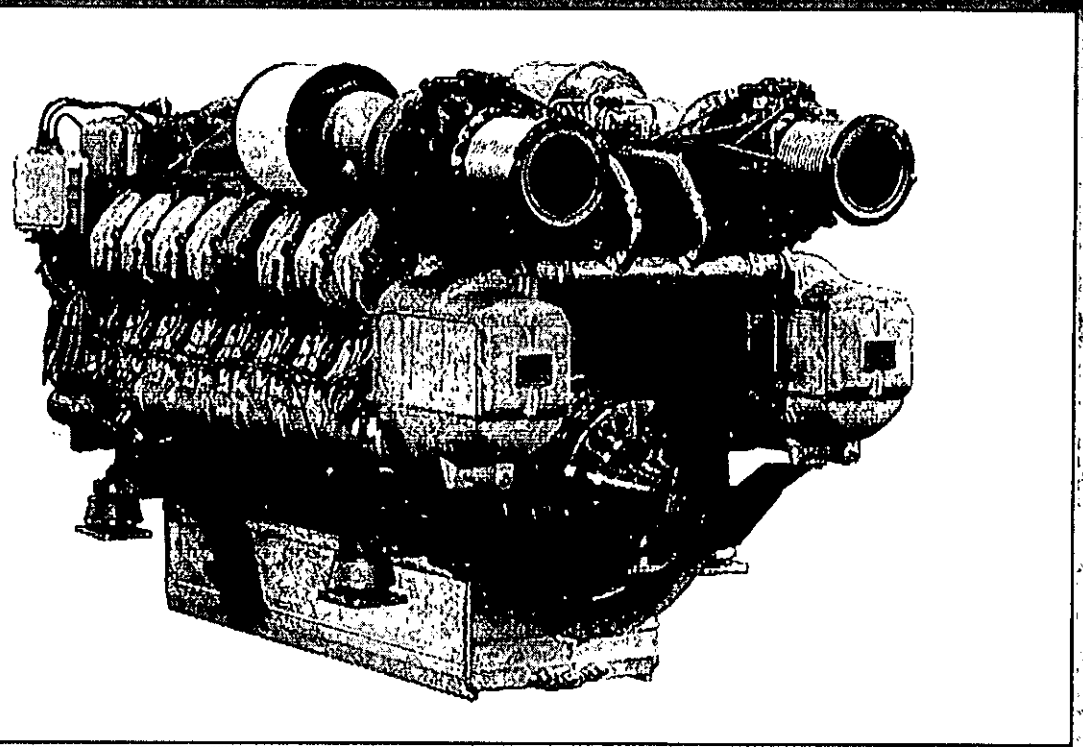
16V 2000 M90 + BW255AP

(Flange-mounted arrangement)



Series 4000

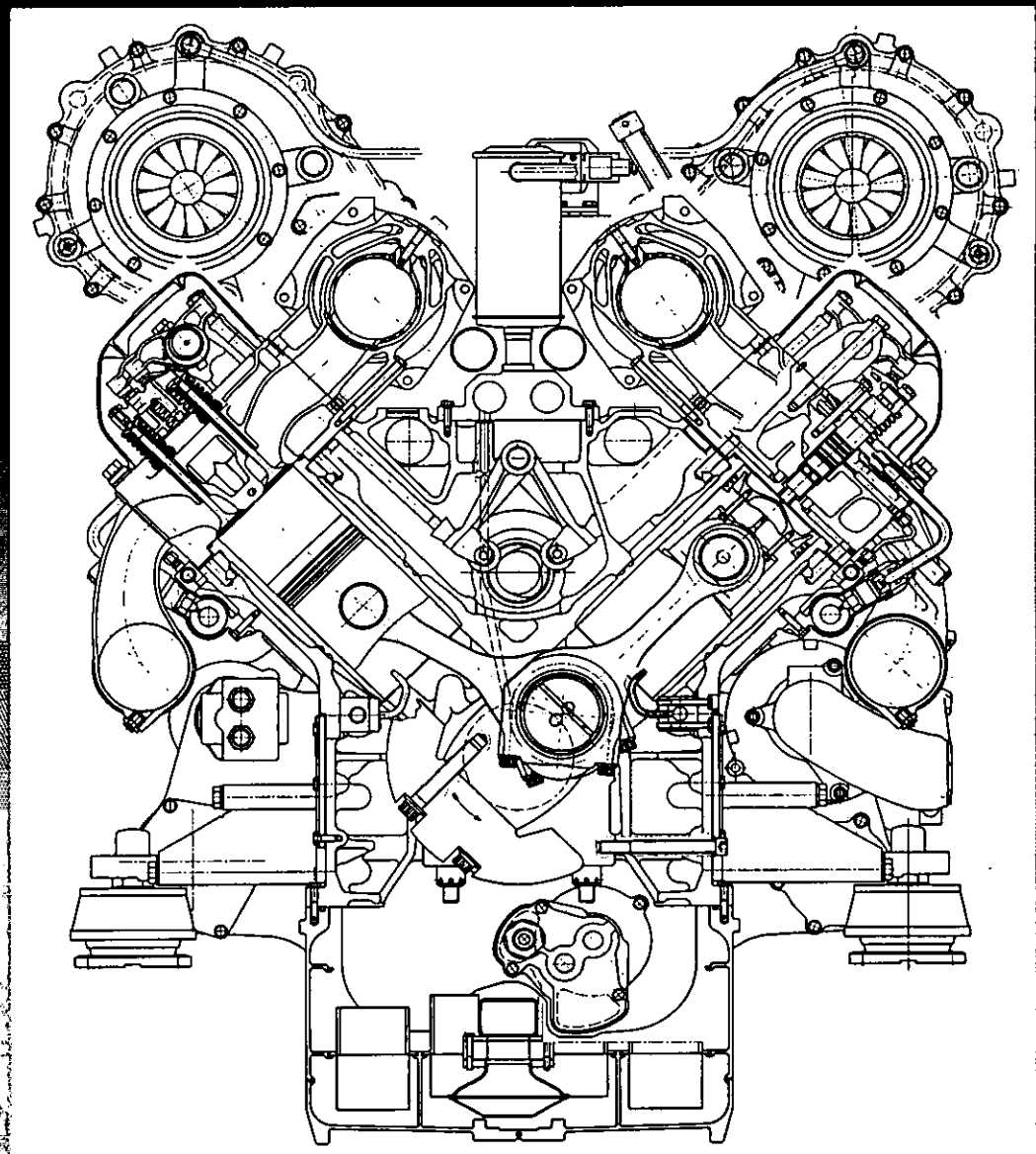




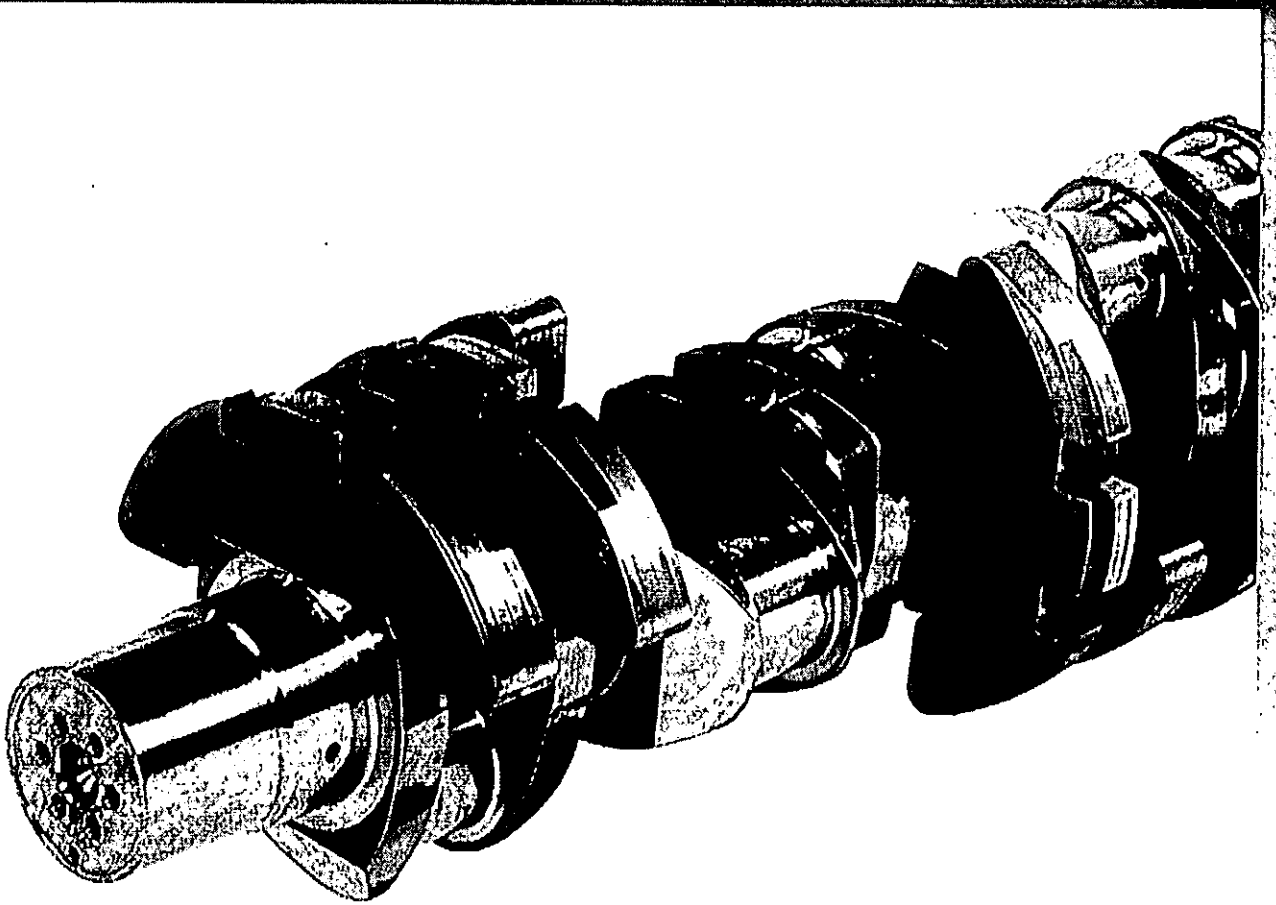
☐	8 V, 12 V, and 16 V	90° V configuration	☒ TBO 15,000 operating hours (1B)
☐	Bore / Stroke	165 / 190 mm	☐ Common rail fuel injection system
☐	Displacement per Cylinder	4.06 liter	☐ Optional: Cylinder cutout
☐	Power range	700 - 2,720 kW	☐ Exhaust turbocharging
☒	Max. speed	2,100 rpm	☐ Optional: Sequential turbocharging
☐	Specific fuel consumption	195 g/kWh (1B)	☒ Electronic engine management

Series 4000
Engine Cross Section

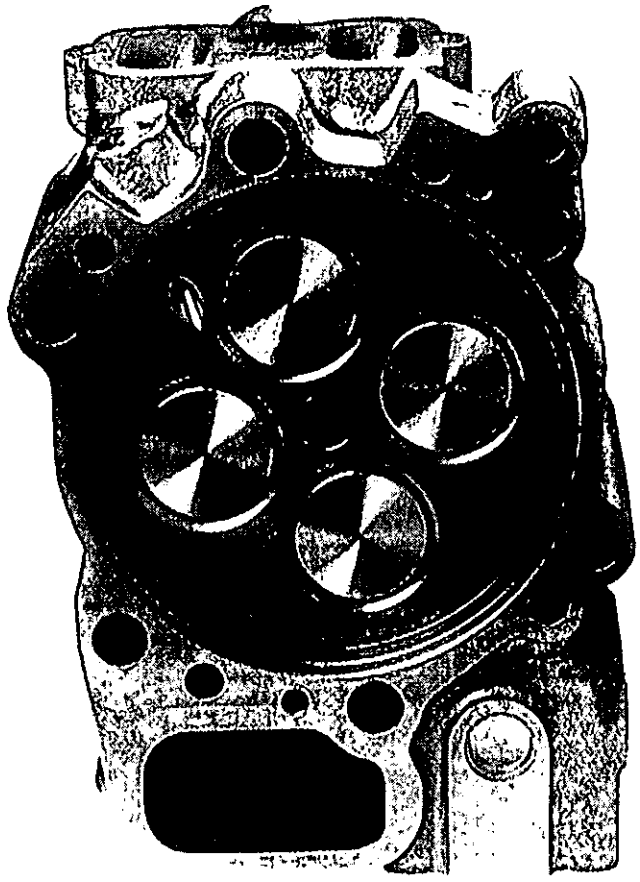
Marine Application



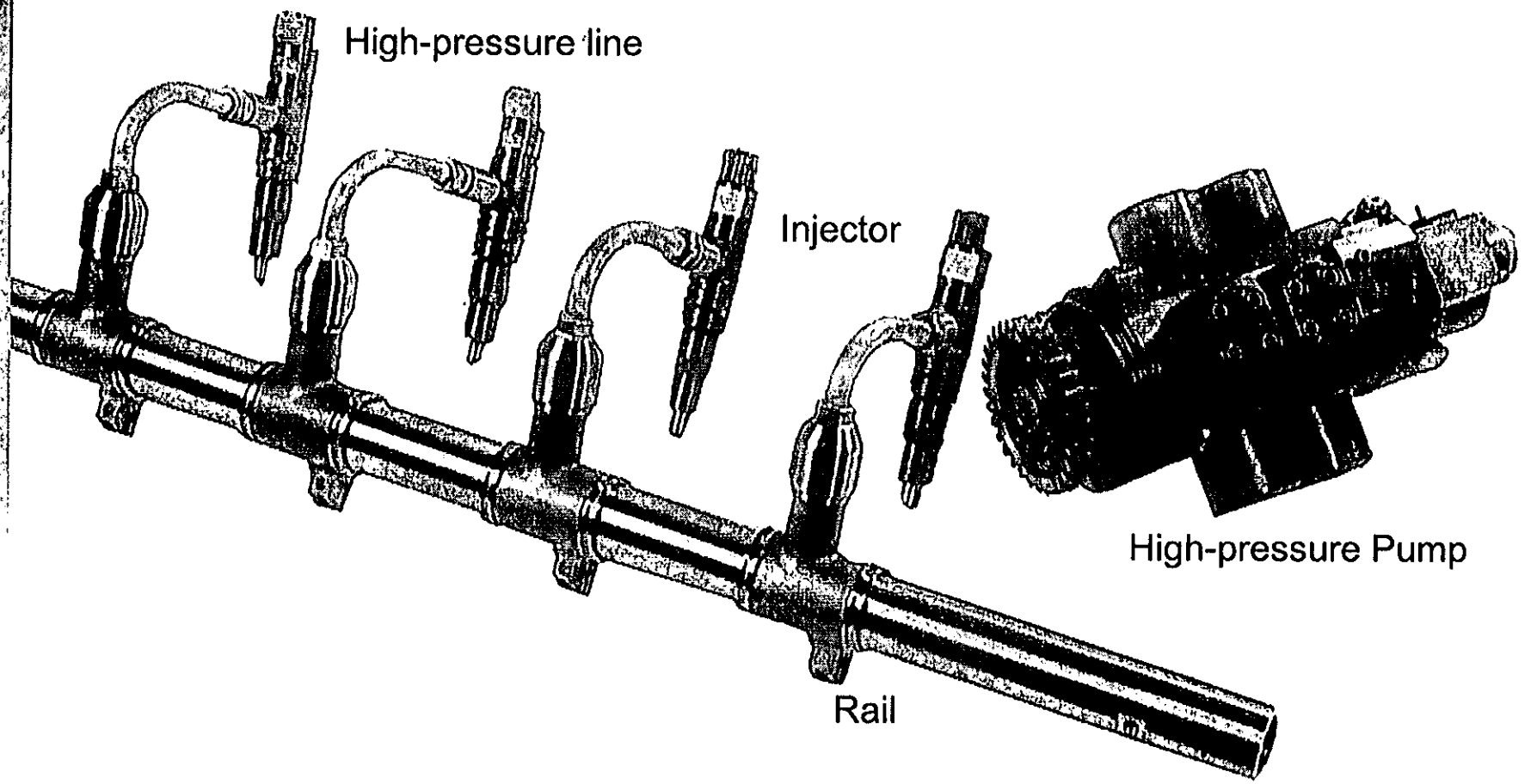
Series 4000
Crankshaft



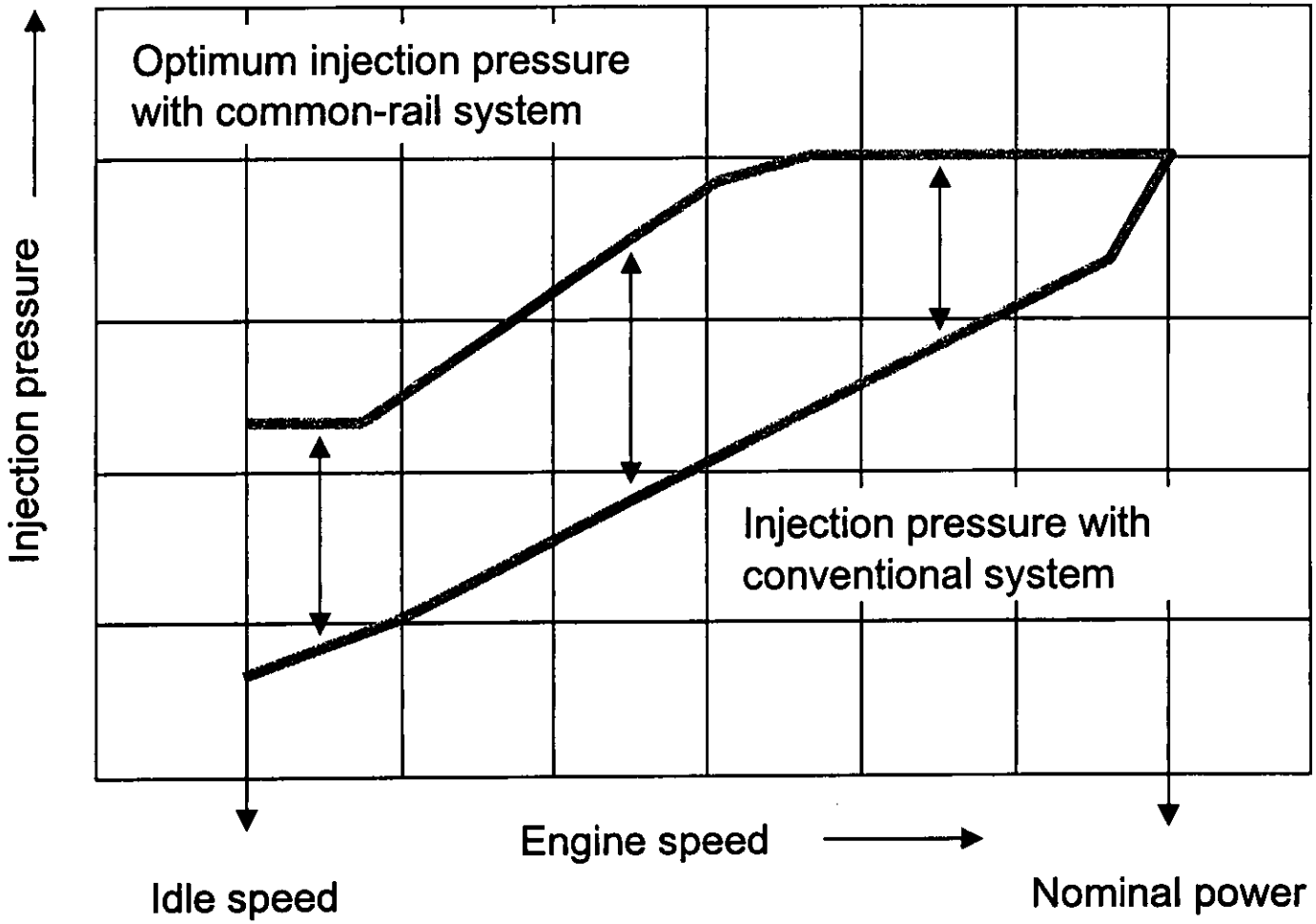
Series 4000
Cylinder Head



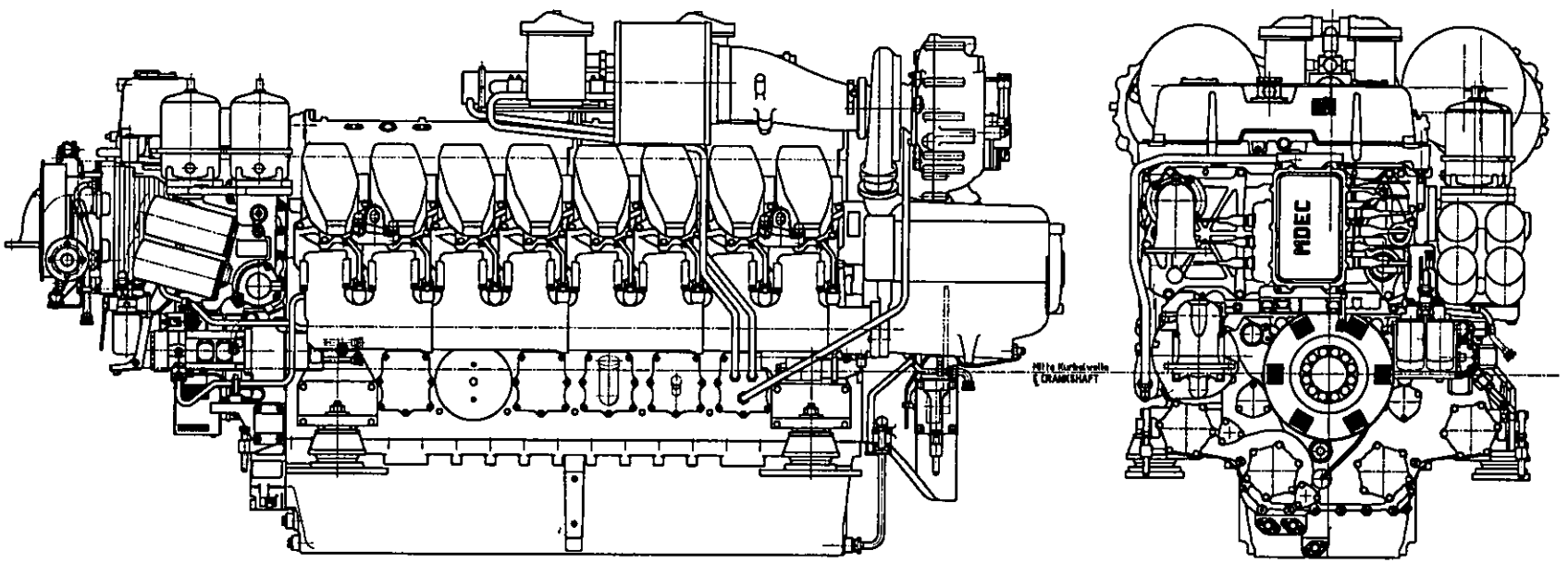
Series 4000
Commonn-Rail Components



Commonn-Rail-Injection System

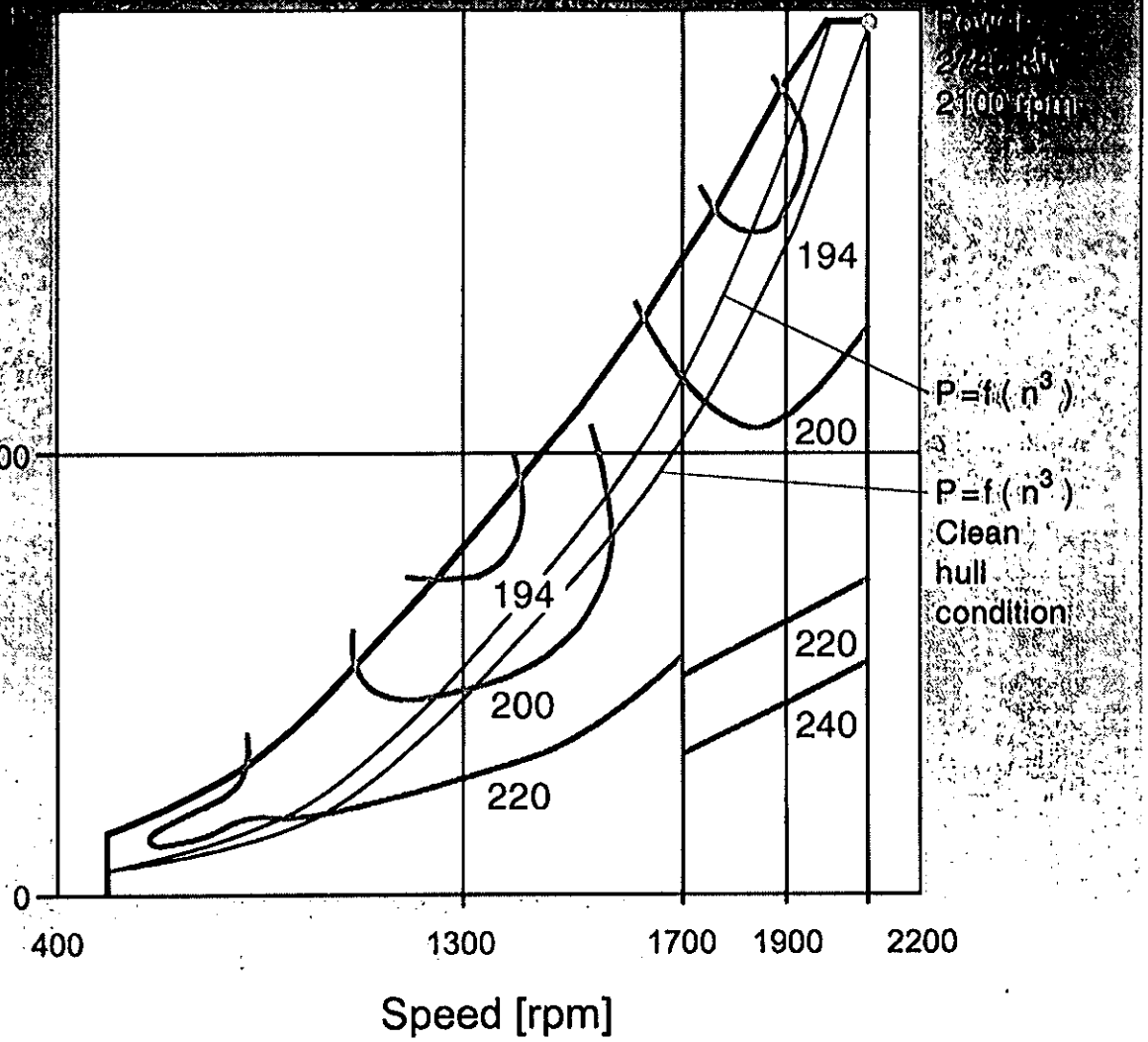


16V 4000 M90

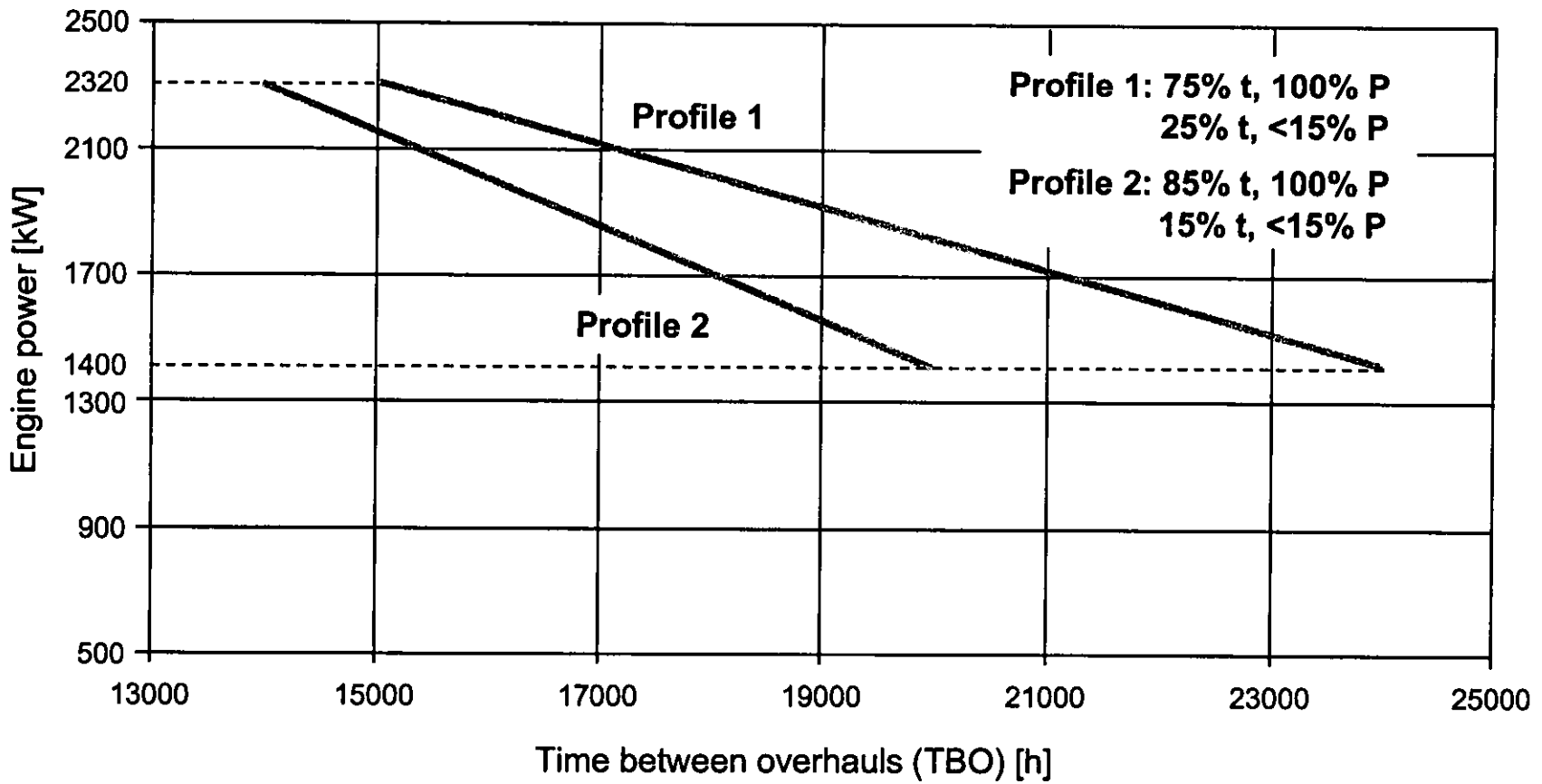


Fuel Consumption Map

16V 4000 M90



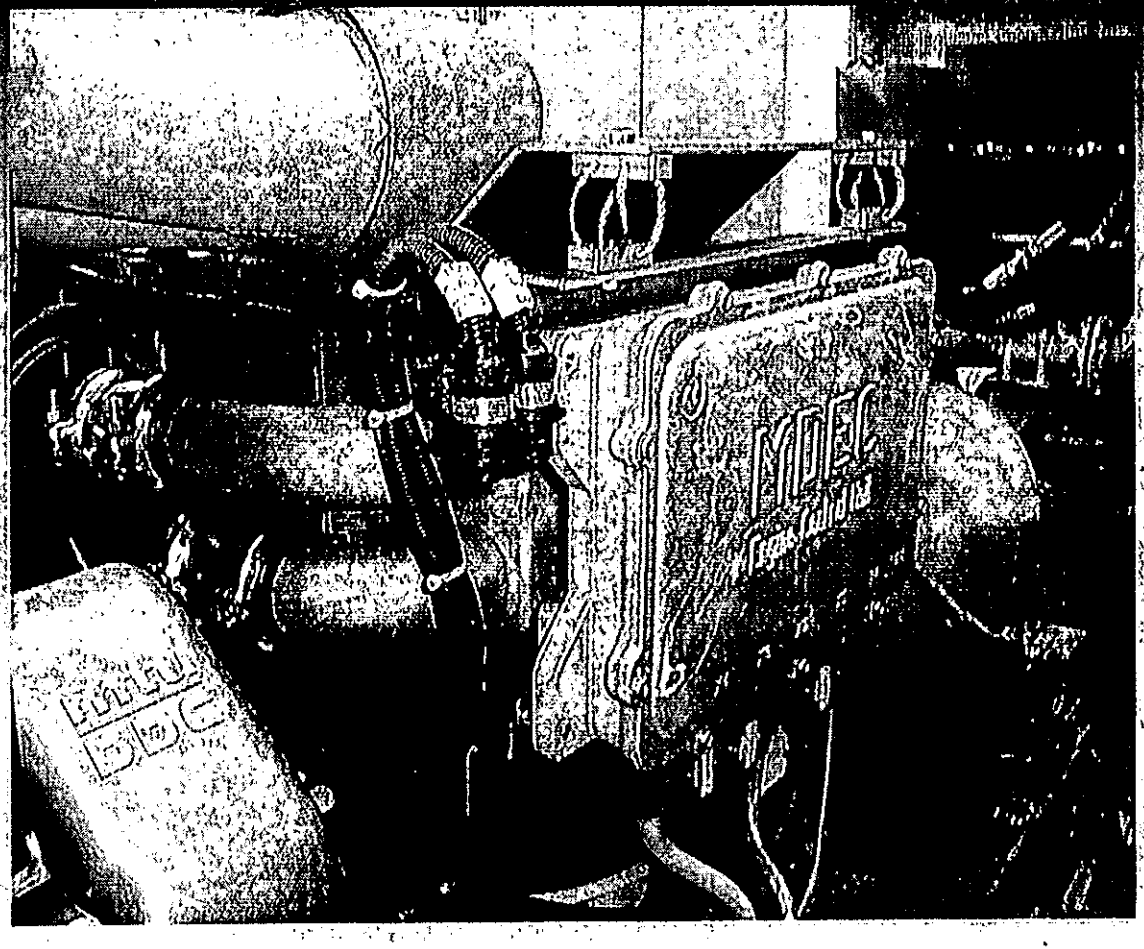
Engine Power vs TBO 16V 4000 M70



Electronic Engine Governing and Control Unit ECU-MDEC

Function:

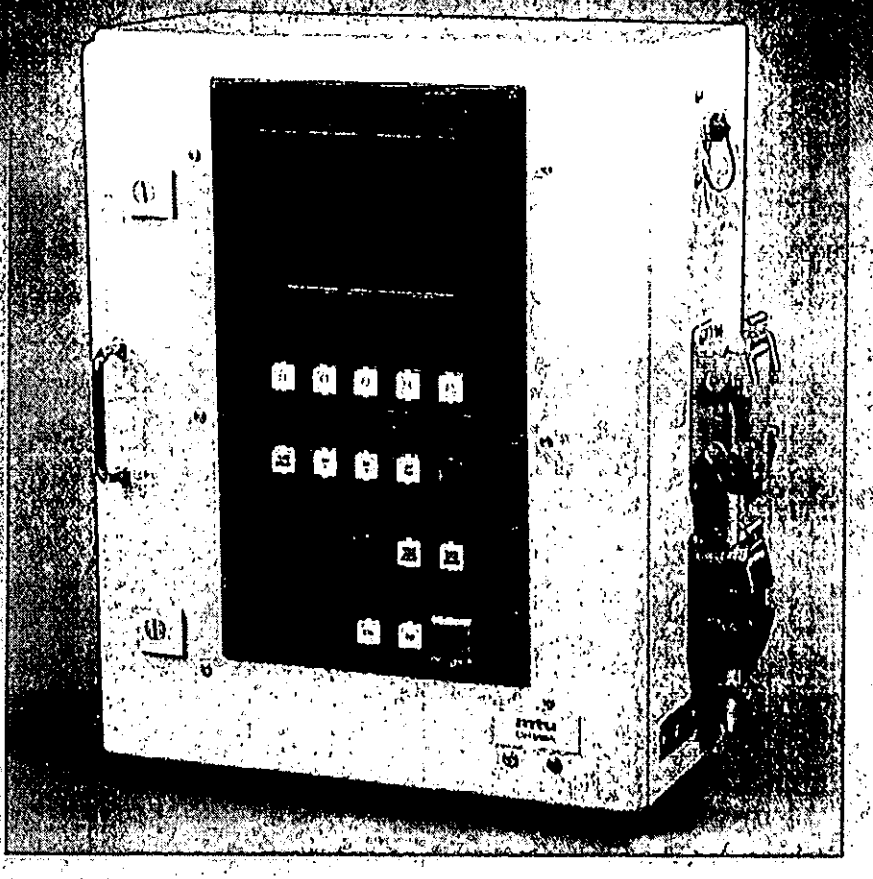
- Regulation
  - Engine speed
  - Fuel injection
- Control
  - Charger switching
  - Cylinder cutout
- Monitoring
  - Engine speed
  - Charger speed
  - Lube oil pressure
  - Coolant temerature
  - Coolant level
  - etc.
- Integrated safety system
- Integrated test system



Local Operating Panel LOP-MDEC

Function:

- [redacted]
- Interface between the assemblies ECU, EMU, GCU and GMU
- Interface between the peripheral assemblies MCS and RCS
- Monitoring and operation of the engine and gearbox locally
- Display of operating values, status and fault messages
- Generates audible and visual alarms in case of failures
- Indication if faults have been detected of the hardware and software



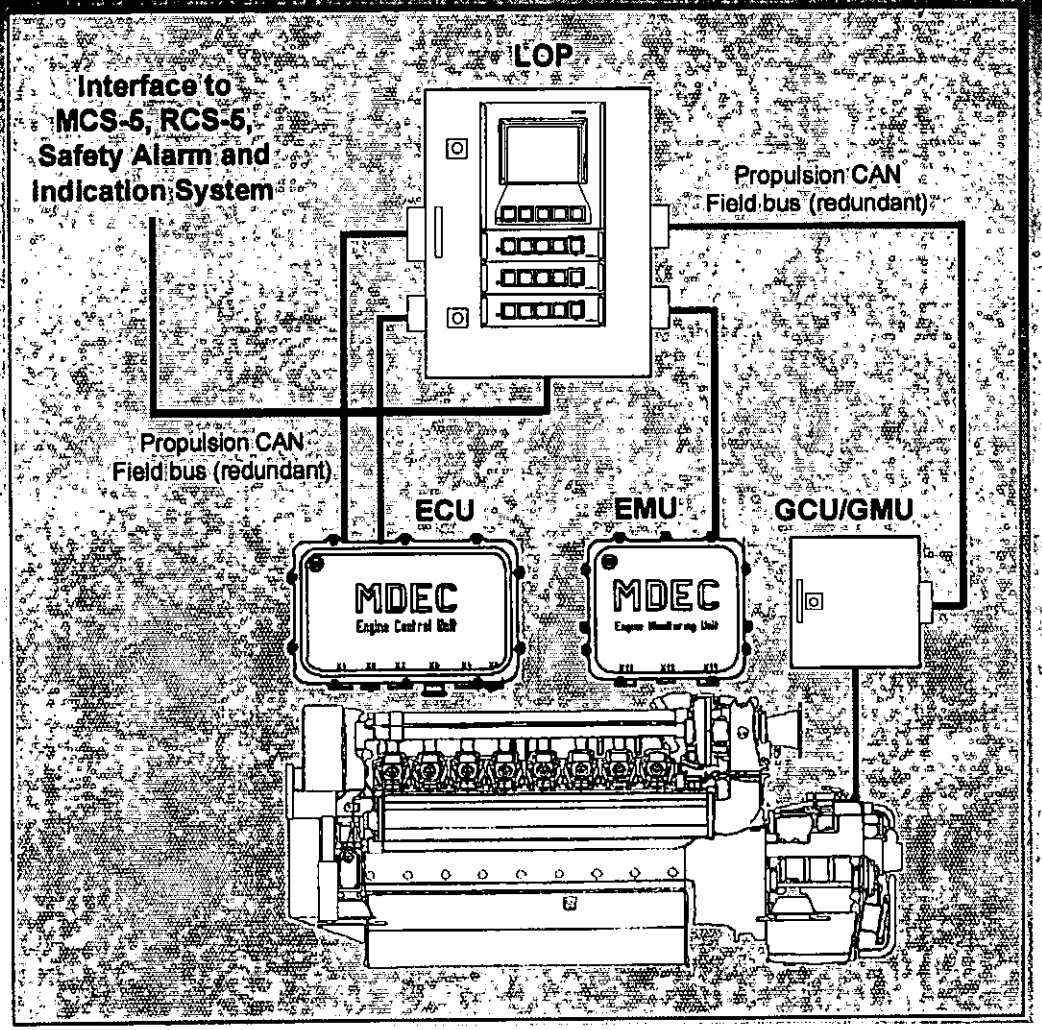
ECU: Engine Control Unit
EMU: Engine Monitoring Unit
GCU: Gearbox Control Unit
GMU: Gearbox Monitoring Unit
MCS: Monitoring and Control System
RCS: Remote Control System

Electronic Engine Management

Designed in accordance with classification society guidelines

The Engine governing and control unit ECU assembly is a speed and injection governor for engine series 2000/4000. The Engine monitoring unit EMU is an independent, auxiliary safety system intended to project the engine. This is realised by an extended range of measuring points and additional (redundant) evalution of appropriate variables.

MCS-5: MTU Monitoring and Control System
RCS-5: MTU Remote Control System
LOP: Local Operating Panel
ECU: Engine Control Unit
EMU: Engine Monitoring Unit
GCU: Gearbox Control Unit
GMU: Gearbox Monitoring Unit
MDEC: MTU Diesel Engine Control
CAN: Contoller Area Network



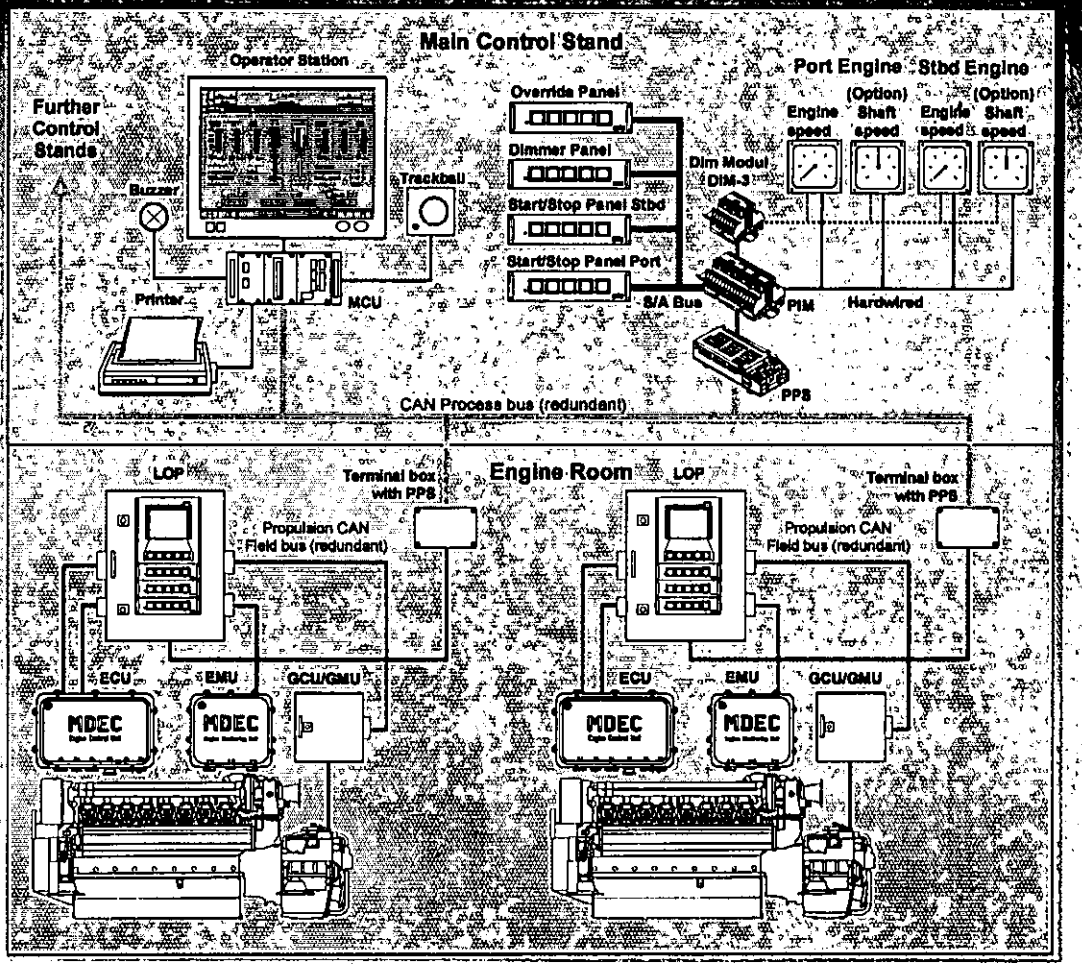
Monitoring and Control System MCS-5 Type 2

System with Colour Graphic Operator Station

The alternative solution

MCS-5 Type 2 is an integrated Monitoring and Control System for all marine applications and can be used for all in-ship systems, e.g.

- Propulsion Plant
- Electrical Power Plant
- General Ship Area



Configuration example: Two Shaft Propulsion plant with engines 2000 and fixed pitch propeller

MCS-5: MTU Monitoring and Control System
MCU: Management Computer Unit
PPS: Programmable Process Station
PIM: Peripheral Interface Module
LOP: Local Operating Panel
ECU: Engine Control Unit
EMU: Engine Monitoring Unit
GCU: Gearbox Control Unit
GMU: Gearbox Monitoring Unit
MDEC: MTU Diesel Engine Control
CAN: Contoller Area Network

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

EXPERIENCES WITH PROPULSION SYSTEMS IN THE LARGE FAST FERRY BUSINESS

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D 88040 Friedrichshafen (Germany)

Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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Fast Ship Propulsion Plants
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ABSTRACT

In 1992 MTU entered into the developing market for large ferries with the size for vessels more than 60 meters in length and top speed requirements up to 45 knots.

The 20V1163 and 16V 595 engine types were ideally suited to this new market.

More than 120 engines of the 20V 1163 engine type are in service with power settings between 6000 kW and 6500 kW.

This paper shows various propulsion systems and describes associated experiences. In addition, some recommendations for future arrangement design configurations are outlined.

In the chapter „future aspects" a description of the new developed 8200 kW-engine for fast ferries is made.

1 - INTRODUCTION

MTU is very well established in the fast ferry market since 1953.

The company and its predecessor companies have supported and influenced the fast ferry business through its products from the very beginning.

More than 2000 diesel engines have been delivered for this application for ships up to 45 meters in lengths.

In 1992 MTU entered into the developing market for large fast ferries with the size of vessels more than 60 meters in length and top speed requirements up to 45 knots.

The 20V 1163 and 16V 595 engine types were ideally suited to this new market.

More than 120 engines of the 20V 1163 engine type are in service with power settings between 6000 kW and 6500 kW.

In the power range from 3500 to 3900 kW we have delivered 46 engines from our engine type 16V 595.

High power-to-weight ratio, optimum and compact installation dimensions together with experience gathered from the operation of more than 200 engines of the same series in naval surface vessels in all areas of the western hemisphere have led to this much envied market success.

This paper shows various propulsion systems and describes associated experiences. In addition, some recommendations for future arrangement design configurations are outlined.

In the chapter „future aspects" a description of the new developed 8200 kW-engine for fast ferries is made.

2 - PROPULSION SYSTEMS

Various propulsion arrangements are currently in service. Examples of principle arrangements are made with a short description of each.

2.1 - CODAG System

Fig. 1 The Rodriguez Aquastrada I monohull named „Guizzo" entered service in Italy in 1993 and the sister vessel „Scatto" in 1994.

The 101.75 m vessels have a triple engine installation. MTU 16V 595 TE70 Diesel engines port and starboard, and a General Electric LM 2500 gas turbine in the centre.

All three propulsion engines drive their respective waterjet. The steerable waterjets are driven by the diesels via a planetary gear unit (gear ratio i = 3.028). The fixed centre waterjet is used as a booster which is driven by the gas turbine via a reduction gear unit (gear ratio i = 9.18). With the total installed power of 27 930 kW the ship can reach a top speed of 43 knots. Out of season or during the night if reduced speed is required to save costs, the vessel can be operated with the diesels only and in this case 18.5 knots are possible.

2.2 - CODAD System

Fig. 2 The Aquastrada III has a four engine installation. In this case, the gas turbine is replaced by two 20V 1163 engines driving the booster waterjet through a combined gearbox. The wing engines are also 20V 1163. The total power of the engines is 24 000 kW and the top speed of the vessel is 36 knots, while operating the vessel only with the wing engines, a top speed of 24 knots can be achieved.

2.3 - Single line arrangements

Fig. 3 This plant arrangement offers a system with total redundancy.

All diesels are completely independent from each other. The four waterjets version with the steering device on each waterjet seems to be more advantageous as the efficiency of the smaller waterjets is higher and the gearbox weight is slightly lower by using four single gearboxes.

Fig. 4 In 1996 the Austal built TT-line catamaran „Delphin" entered service on the route from Rostock (Germany) to Trelleborg (Sweden).

The catamaran carries up to 600 passengers and 175 cars with a service speed of

36 knots. The propulsion system consists of 4 x 20V 1163 with an output of 6000 kW each. The engines meanwhile have reached 17 000 operating hours.

3 - OPERATION

The fast ferry industry demand is for payload and speed.

The vessels have to earn money and downtime is not part of the operators vocabulary.

Vessel service schedules from 6 o'clock in the morning until midnight or one o'clock in the morning are not unusual.

Compared to a large conventional ferry there are a lot of stops and starts per day.

The operating profile for the engine is very high.

By entering into this market segment we have discussed with operators a profile:

3 % time	100 % power
82 % time	85 % power
15 % time	<15 % power

Meanwhile with approx. 20 large fast ferries in operation with our engines we have found out through the evaluation of our operating profile recorder, that the profile can reach higher levels. An operating profile of

3 % time	100 % power
82 % time	90 - 95 % power
15 % time	<15 % power

is not an exception.

4 - EXPERIENCES WITH THE ENGINES AND THE INSTALLATIONS

4.1 - Air

Fig. 5 Gas turbines and diesel engines must have clean air to work satisfactorily.

The gas turbine installation requires ducting air directly from the outside to the engine.

With the diesels, there is also the possibility to draw the air from the engine room.

Our experiences show that the engine room air quality varies significantly depending on many factors. We have found air borne particles of a critical concentration in the charge air coolers, the air pipe work and also in the combustion chamber/cylinder piston area of the engines.

To prevent these particles from entering the engines it was necessary to install paper filters directly on the engines as a standard solution, except when the air intake is drawn from outside.

4.2 - Oil

Experience has been gathered that the use of oil type I requires 500 hours replacement intervals. This short interval was not acceptable and therefore we have worked out a new guideline.

Oil replacement intervals can be increased to 1000 hours if:

- an approved type II oil is used;
- the fuel sulphur content is below 0,5 %.

Together with the operator MTU is verifying the oil condition after these operating periods and then decide whether it is possible to extend the replacement interval.

4.3 - Fuel Injection System

In some cases we have found irregular combustion resulting from the injector:

- drop of opening pressure combined with blow back from the cylinder
- loss of material strength and premature ageing of the injector
- uncontrolled combustion with timing and amount of injected fuel out of tolerance.

It was necessary to develop a new fuel injector with an increased opening pressure from 350 to 400 bar to prevent combustion gas blow-back and premature injector needle wear.

The replacement can be done during maintenance intervals without affecting the fuel consumption figures.

4.4 - Cylinder liner

As a consequence of the tough operating profiles and environmental conditions in the engine room, in some cases premature cylinder liner wear has been observed.

This has led to an increased risk of piston seizures after approximately 5000 running hours.

Our investigation program has shown, that the wear pattern is as follows:

- build up of carbon residues on the piston crown
- polishing of the housing structure
- insufficient lubrication due to blank spots

End of 1997 we have started the introduction of new cylinder liners with an inserted scraper ring (anti polishing ring).

The effect of the scraper ring can be summarised:

- Removal of carbon at each piston stroke
- no contact between piston crown and liner surface
- reduced oil consumption

4.5 - Gearboxes

In some cases operators were faced with severe damages to the gearboxes coupled to MTU-engines, but also those coupled with engines from our competitors.

As the gearboxes are normally not to a standard design (special requirements in respect of installation) in a damage case no spare gearbox is normally available, therefore the vessel has to be operated over a long period using only 3 of the 4 engines hence a serious loss of income for the operator as the time schedule cannot be fulfilled.

Our recommendation with respect to the gearbox design is as follows:

- Reliability has highest importance and the priority over weight considerations.
- Essentially, a casted housing is preferred over a welded fabrication.
- The gearbox must withstand the loads induced by the diesel engine and its coupling systems:

The alternating moment level is influenced to a large extent by the torsional resilient coupling characteristic as well as by the gearbox and propulsion shaft stiffness and rotary mass.

The real operational vibration moments level must be between the levels indicated by the torsional vibration calculation for normal and one cylinder misfiring mode.

The classification rule to consider up to 33 % of the nominal torque for the gearbox design incorporates this recommendation.

Depending on the shaft offset compensation coupling characteristic axial forces up to 2.6 kn (for Voith Tripode cardan shaft and Centalinc coupling) are transmitted to the gearbox input.

4.6 - Shaftline

Fig. 6 Further loads can be induced to the gearbox output shaft through the propulsion shaft train and the movement of the ships structure in demanding sea conditions.

As a safety precaution we have learned, that the best solution is to isolate the movements of the waterjet flange and the gearbox flange through a misalignment element which could be a double tooth coupling or use of the new composite shaft, solutions which reduce the forces to the gearbox output side.

5 - MAINTENANCE

The basis for good maintenance work is fixed at the initial design phase of the vessel.

It is imperative that the designer, the shipyard and the engine manufacturer work very close together to find the right solutions for proper maintenance work.

As we have learned there is normally limited time for regular maintenance work the shipyard has to ensure that piping, ducts or frames do not interfere with component removal or service personnel maintenance envelope.

To reduce the downtime of the vessel, MTU's philosophy for the main overhaul (W6) of the engine, is to replace the engine with a spare engine.

Shipbuilders and operators must be involved from the beginning of the planning stage.

They are supplied with installation diagrams on which every interface is numbered and its fitting and removal times detailed.

Fig. 7 1163-Series engines in large vessels can now be exchanged and replaced just as quickly as the smaller 396 models.

New installation and maintenance techniques such as redesigned interfaces and simplified engine alignment procedures have made this improvement possible.

Universal shafts and vertically adjustable engine mountings do away with the need for the involved and time-consuming process of adjusting the engine base when aligning the engine.

In conjunction with optimised hatchways through which the engines can be removed the design features effect replacement of a 20V 1163 engine within 24 hours.

MTU is also prepared to offer to the customer, maintenance contracts up to and including the main overhaul.

In discussion with the operator the maintenance work which can be undertaken by the crew must be determined.

6 - FUTURE ASPECTS

The application area of large, fast ferries has brought new, practical experience for all concerned. At the start of the 90's, it was 75-m vessels, growing to catamarans of 80 to 90 metres in length, with monohulls of over 95 meters toward the middle of the decade.

Fig. 8 Three catamarans of 125 m length were being built in Vancouver for B.C. Ferries equipped with 4 x 20V 1163 TB73L engines with a total power output of 26 000 kW per ship.

Four monohulls of 150 m are in service in Italy for Tirrenia as operator. In each of these, a CODAG plant with four 20V 1163 engines and two LM 2500 gas turbines provides the propulsion power totalling 71,000 kW per vessel. Even larger ships - particularly for cargo applications - are in the project phase.

MTU has made great efforts to advance the Series 1163 especially for this demanding new market segment. It was clearly recognized that these engines - which provide trouble-free service in many military applications - would have to be improved for these expanding commercial applications. Therefore, modifications have been developed which have been retrofitted to existing vessels before the 1998 season.

As stated earlier, the vessels become bigger and faster and besides the combined systems we have found out in close discussions with operators and shipyards more diesel power is required.

As there are limits with the 1163-engine it was decided to develop a new diesel engine family mainly for the fast ferries market.

The family is named „MEGA" which stands for „Medium speed Engine for General Applications".

The most powerful engine is the **20V MEGA** with a power output of **8200 kW** at ambient conditions 45°C/32°C. The rated speed is 1150 rpm.

Fig. 9 Shows a picture of the new engine. The engine is equipped with all accessories which are developed and tested together with the basic engine. In the design is brought in all our experience coming out from the large fast ferry business with the 1163-engine.

Fig. 10 gives an impression about the cross section of the new MEGA. In the following a description of the main components is made:

The crankcase has a rigid design and is made out of nodular cast iron with integrated charge air ducting. Large crankcase doors are available for maintenance work and to take care, that also the main bearings can be replaced in situ.

The crankshaft is made out of special steel with bolted-on counterweights.

For injection the new developed common rail system is used.

The power unit concept allows to replace a complete cylinder assembly with the cylinder head, piston, cylinder liner, conrod and the intermediate cooling water housing.

The cylinder head is made out of nodular cast iron with two inlet and two exhaust valves. The valves have cooled inlet and exhaust valve seat inserts and rotators.

The piston is a composite design with a forged steel crown and a steel skirt. The piston is oil cooled via spray nozzle and carries two compression rings and one oil control ring.

The cylinder liner is bolted onto the cylinder head from below and is equipped with an anti-polishing ring.

The turbochargers are located on the main PTO-end. Four turbochargers of radial/ radial types are installed in a water cooled housing.

The exhaust piping is located in the engine-V. Two pipes are isolated with special material.

The service module at the free end carries the integrated pumps for oil, cooling water, raw water and fuel.

In addition the automatic lube oil filter, the lube oil centrifugal filters, the fuel filter and the oil cooler are installed on the engine.

The modern engine is equipped with an electronic-controlled engine management system. An interface to the propulsion plant management system is available.

The main dimensions of the MEGA-engine are shown in ***Fig. 11***.

Special attention was drawn during the design phase to provide a small width of the engine.

Thanks the accessories, installed on the engine and fitted to the engine, the installation work of the shipyard can be reduced.

With the described components, we do see the following advantages in using the new MEGA-engines:

- Quick acceleration behaviour
  (avoiding black smoke)
  ⇨ Sequential turbocharging with turbochargers with low mass inertia

  High torque at low speed
  ⇨ Sequential turbocharging.

Fig. 12

- Low fuel consumption over the entire operation range in the performance map
  ⇨ Sequential turbocharging
  ⇨ Common rail fuel injection system
- Low oil consumption
  ⇨ anti-polishing ring
- Low emissions (fulfilment of IMO)
- Electronic-controlled engine

  (controlling all parameters - avoiding overloading)
- No power reduction up to 45 °C / 32°C
- Adjustment of propeller operating point
  60 rpm speed band at full power

MTU believes in fast sea transportation and in year 2001 there will be a new diesel engine on the fast ferry market with a power setting of 8 200 kW and

- high availability
- low maintenance cost
- low fuel consumption
- low emission figures
- ease maintainability
- reduced maintenance work

Experiences with Propulsion Systems in the Large Fast Ferry Application

Dipl. Ing. Gerhard Haussmann
Director
Marine Application
MTU Motoren- und Turbinen-Union
Friedrichshafen GmbH

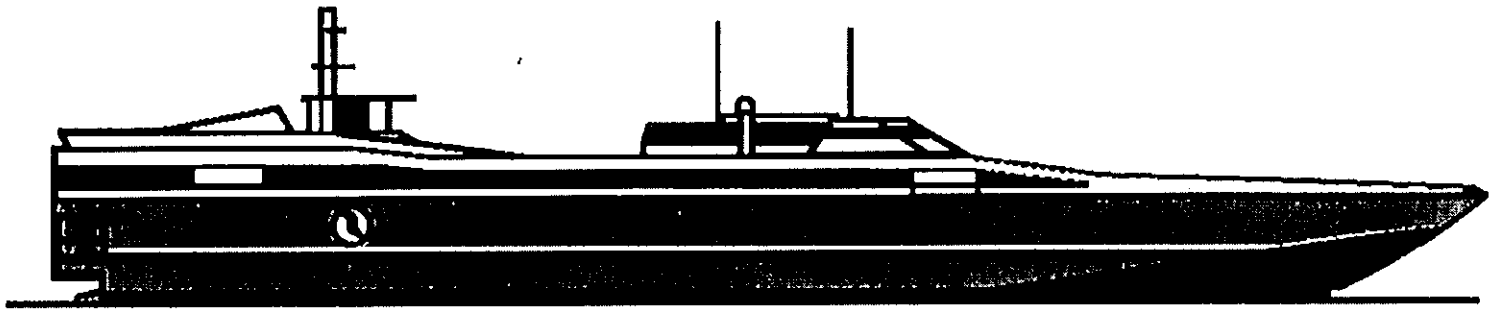
XXXIII[rd] Wegemt School

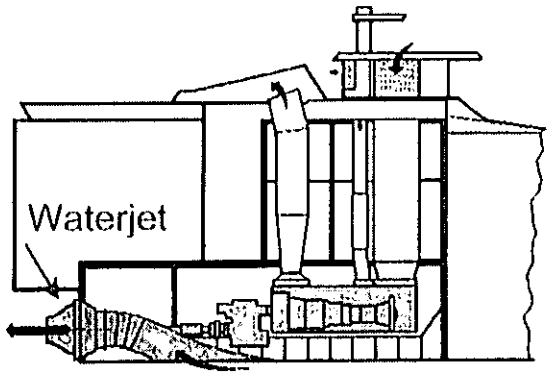
Fast Ship Propulsion Plants

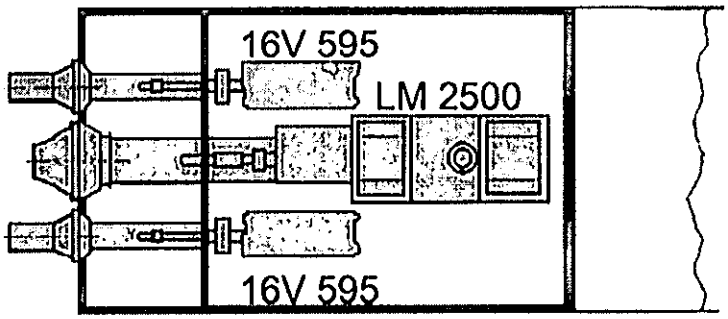
May 22[nd] - 26[th], 2000

Genova, Italy

Fast Car / Passenger Ferry *"Aquastrada"*





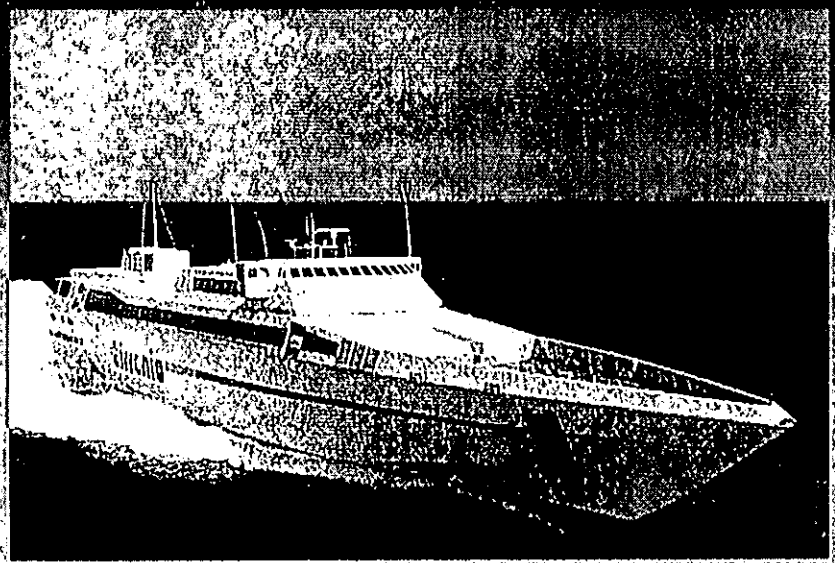


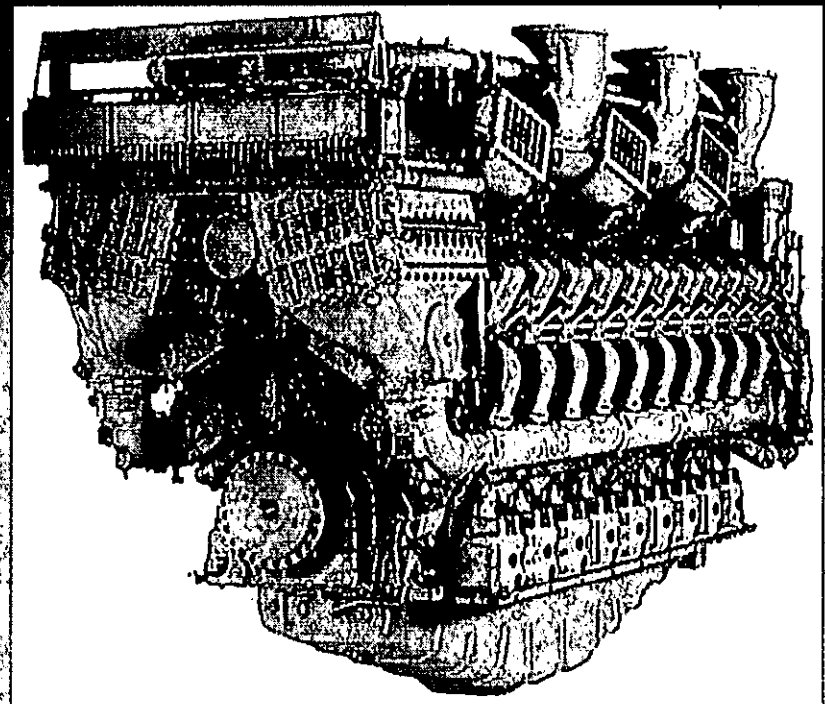
Length:	**101.75 m**
Beam:	**14.5 m**
Speed:	**43.0 kn**
Displacement (full load):	**1032 t**
Pax:	**450**
Cars:	**110**

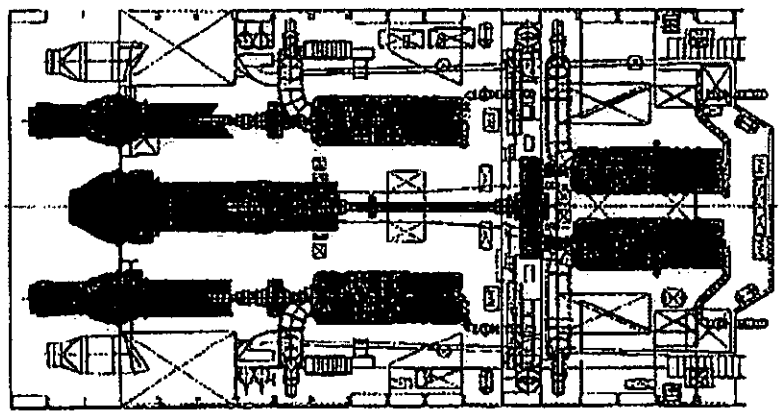
Propulsion	
1 x LM 2500	**20 800 kW**
2 x 16V 595	**2 x 3563 kW**

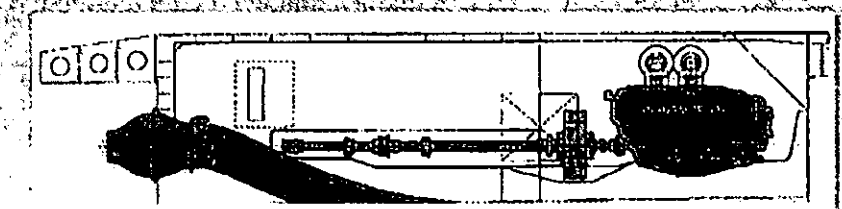
CODAD-Propulsion for Fast Ferries

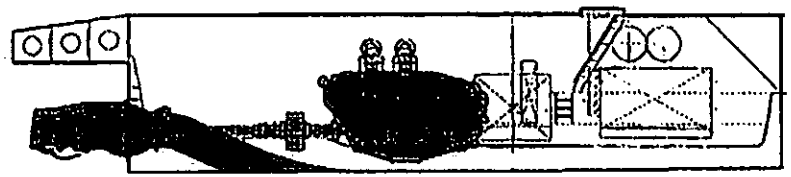
4 x 20V 1163 TB93, 24 000 kW Total Power



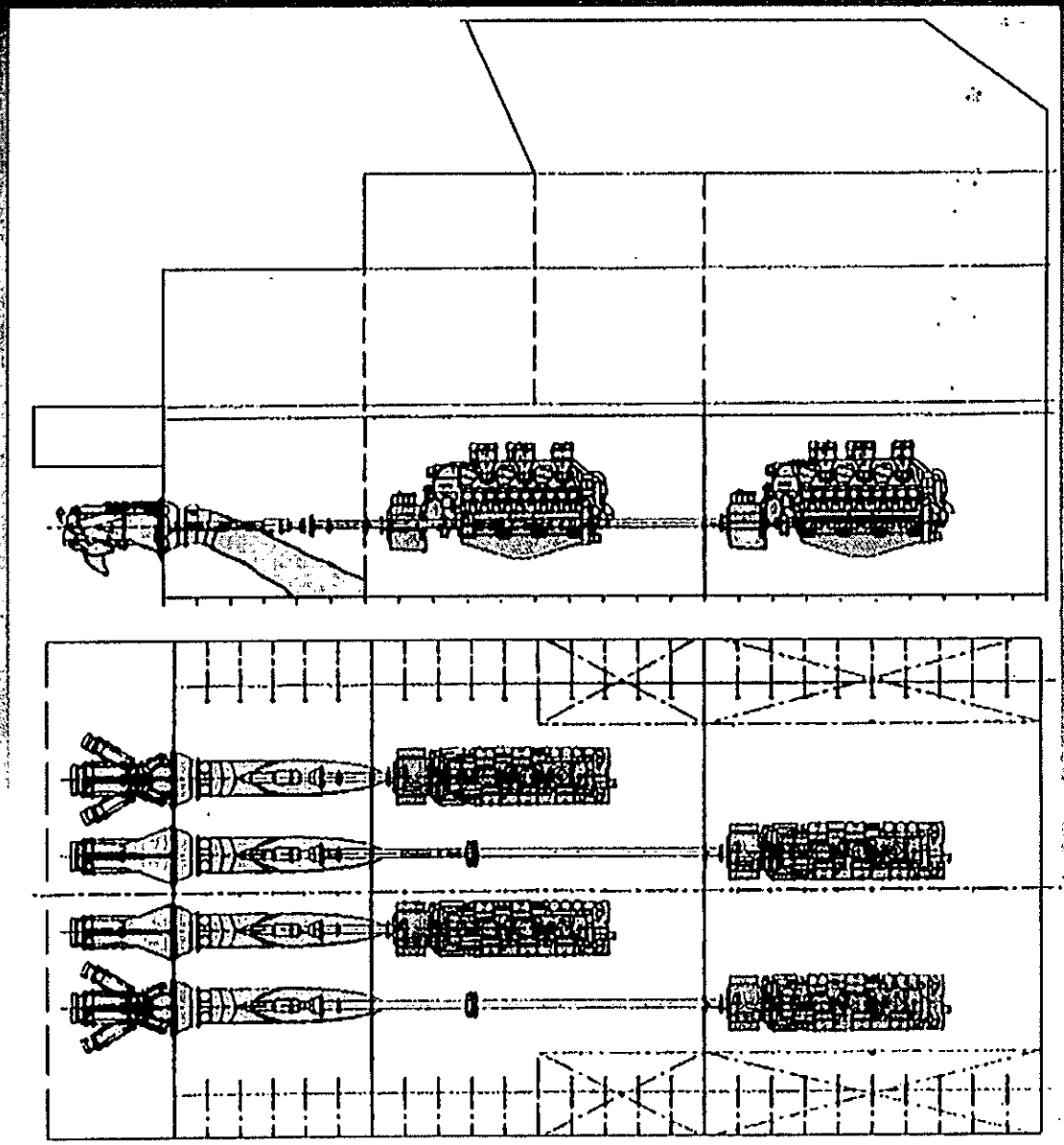






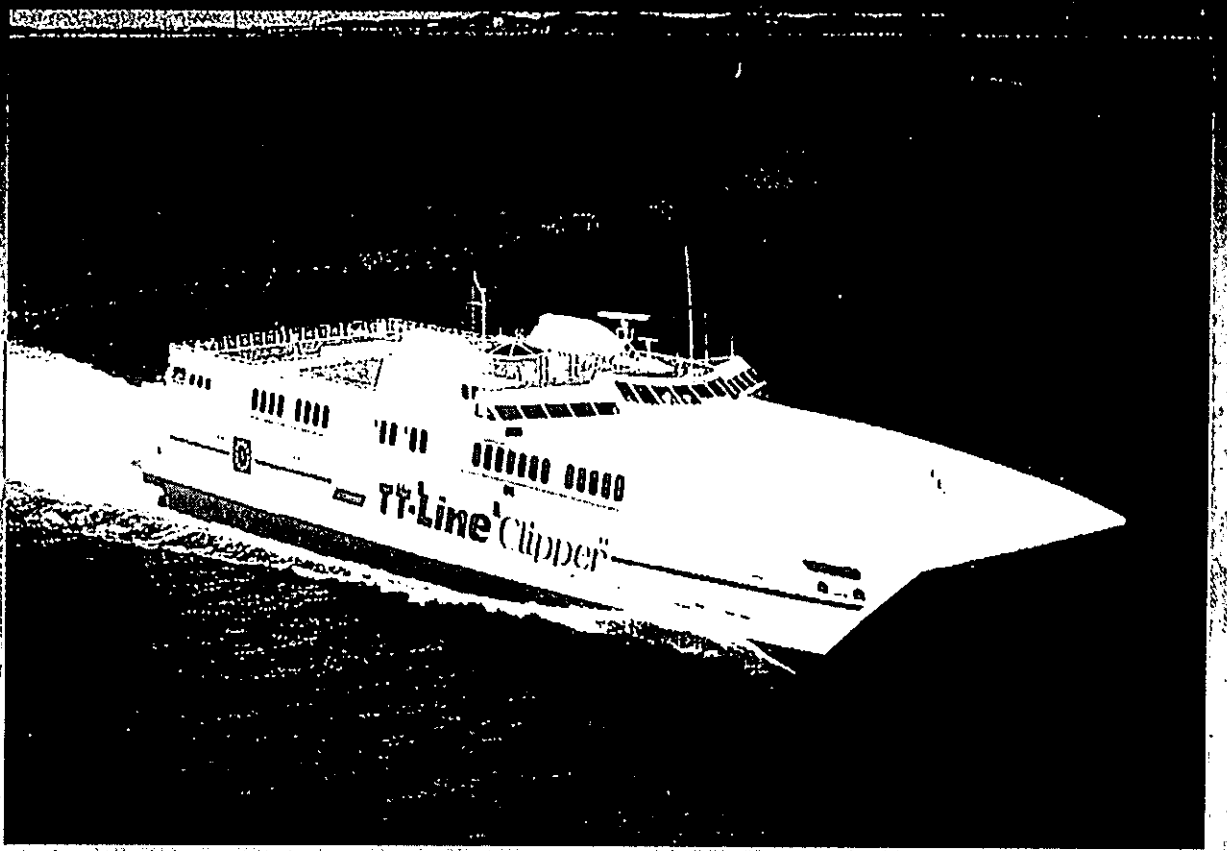


Arrangenment Waterjet Propulsion Plant



Fast Commercial Ship Market

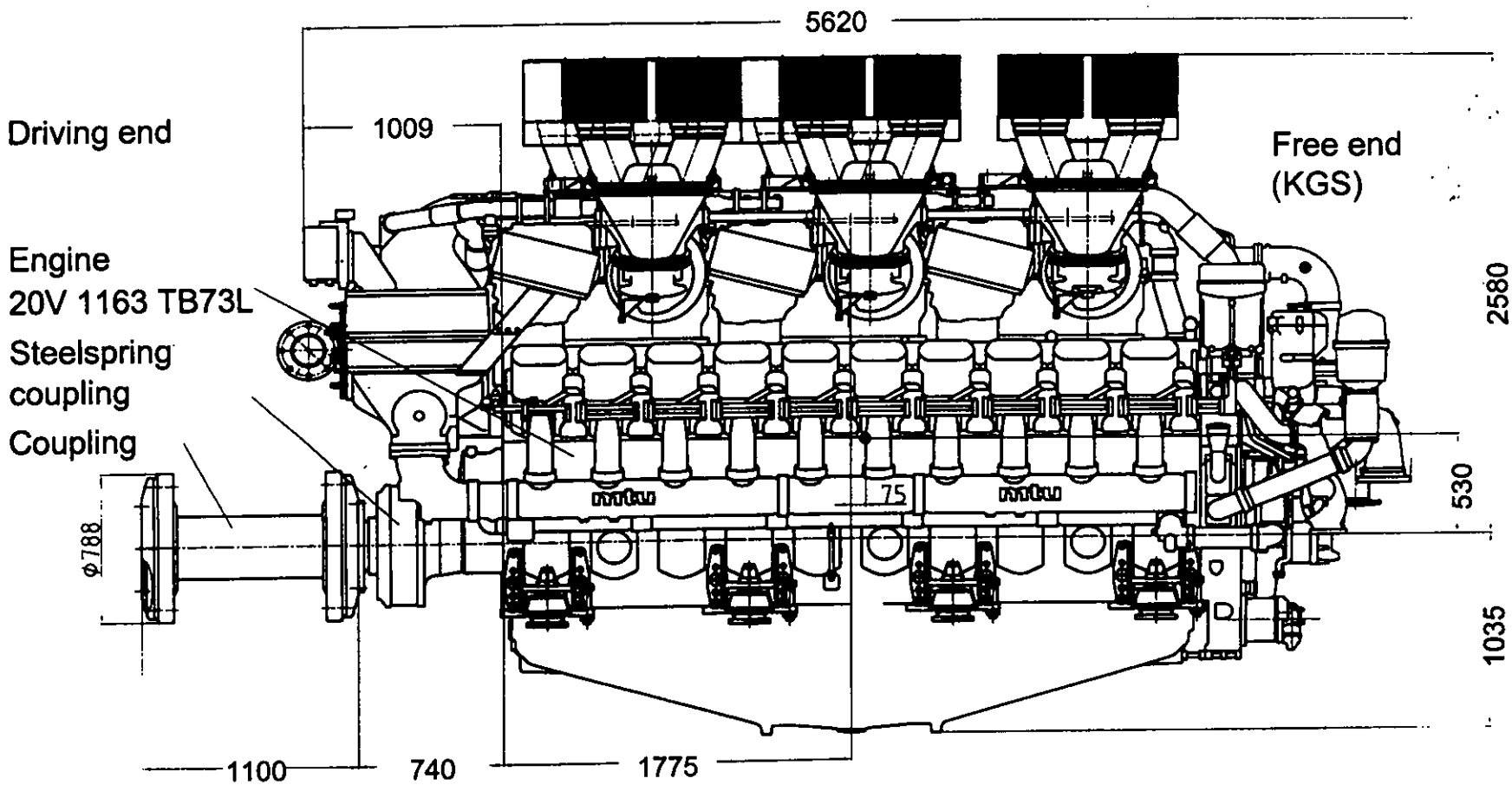
Fast Ferry *"Delphin"*



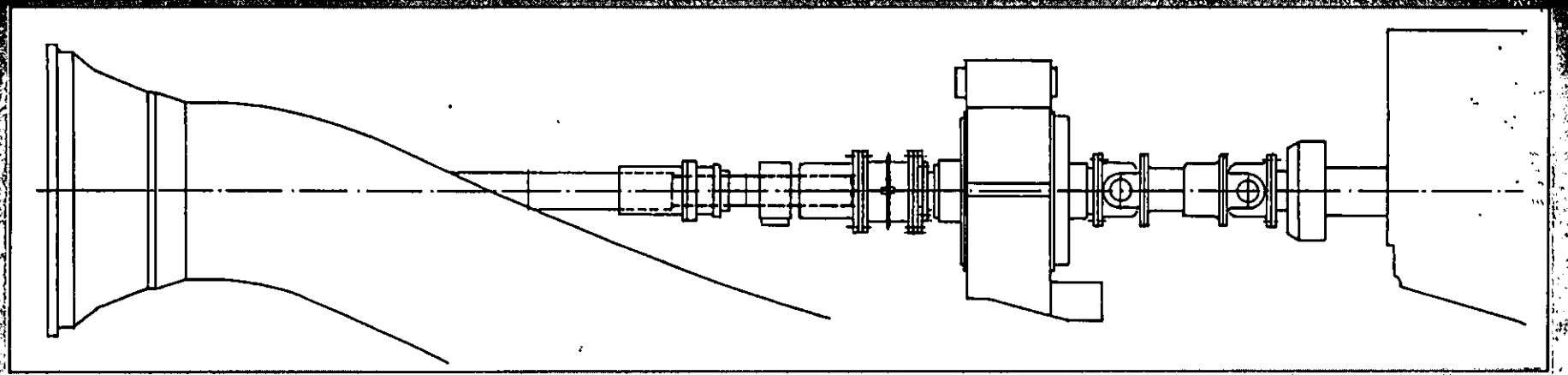
Length:	82 m
Speed:	36 kn
Propulsion:	4 x 20V 1163
Power:	24 000 kW

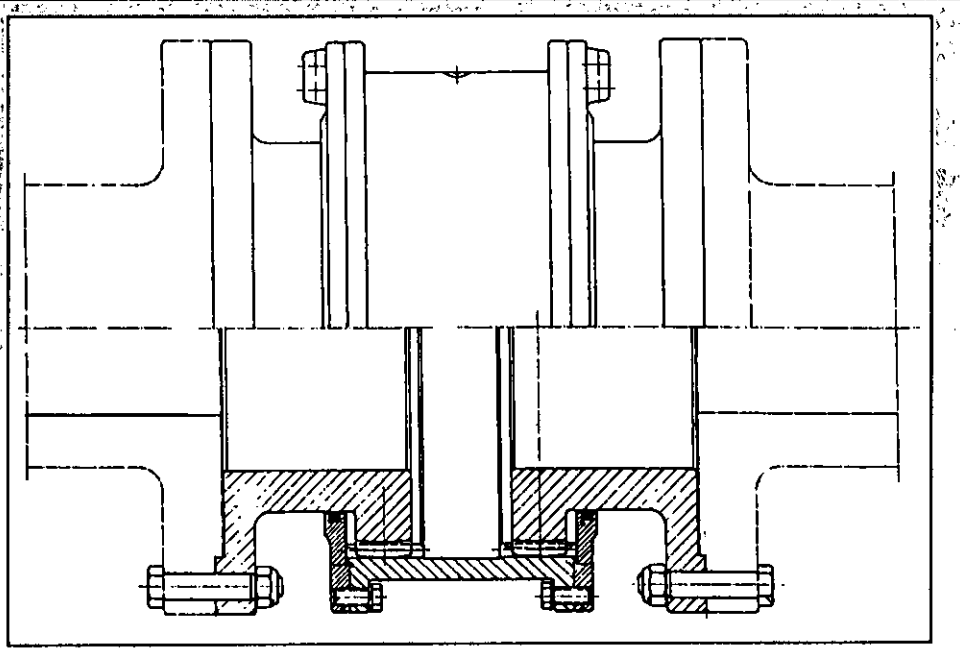
Shipyard:	Austal Ships
Operator:	TT-Line

Arrangement Engine 20V 1163 TB 73L



Arrangement Shaftline with Tooth-coupling

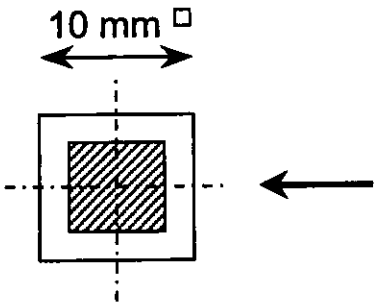


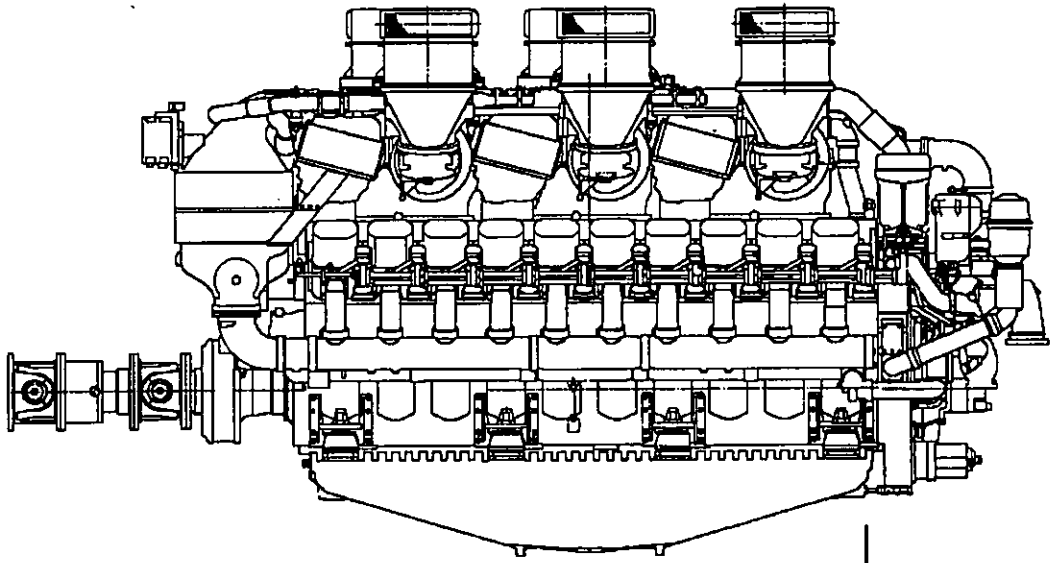


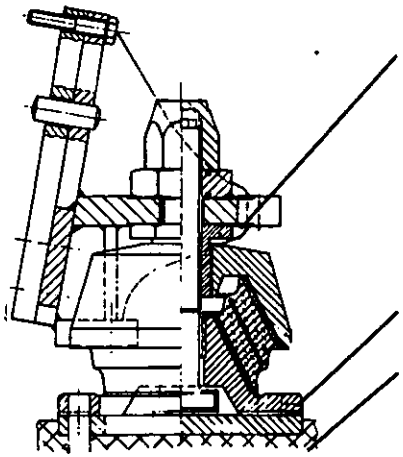
Maintenance Engineering

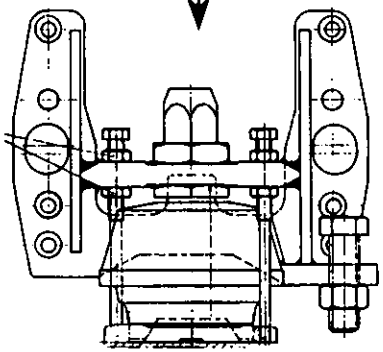
Ease of Alignment

Max. permissible range 10 mm

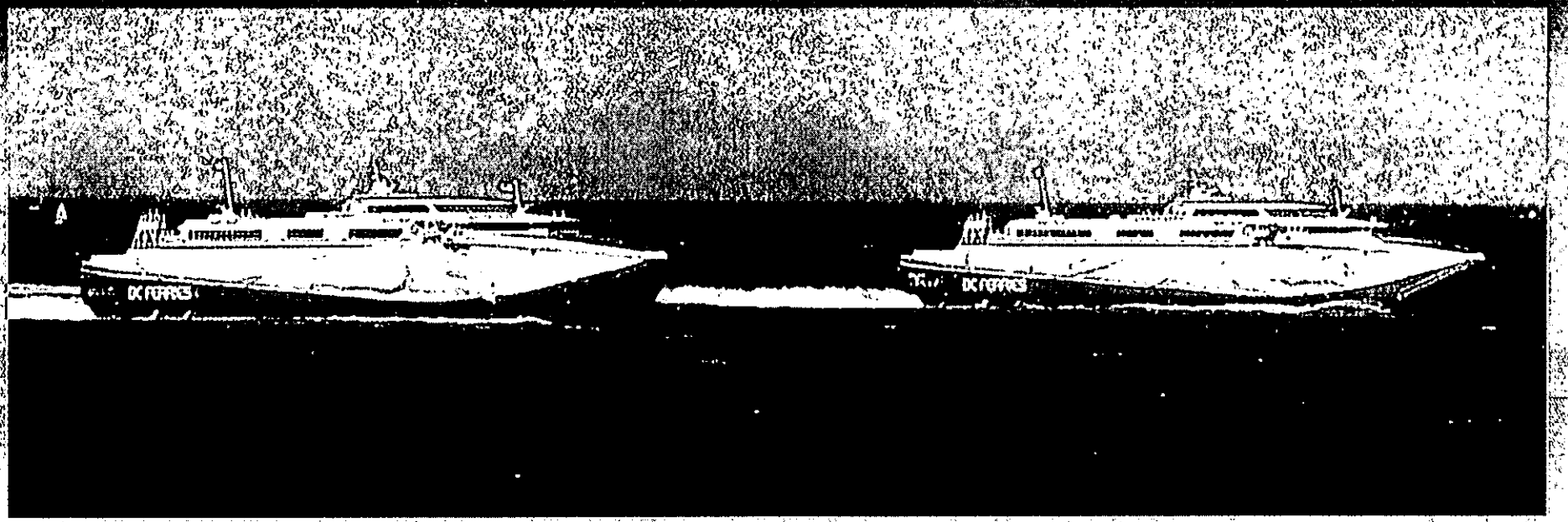




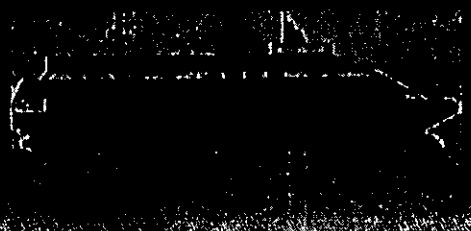


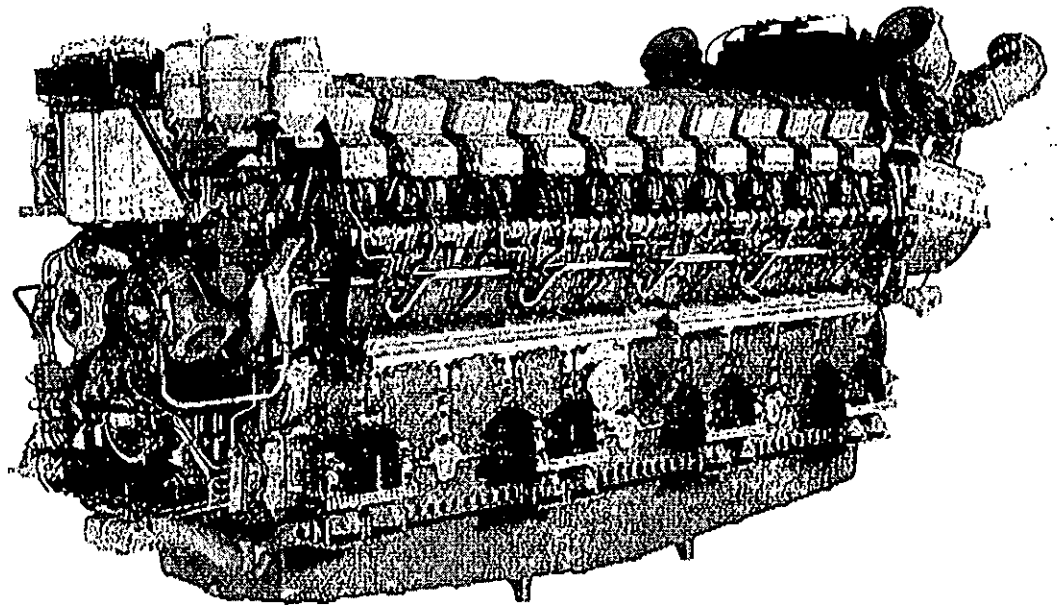


BC Ferries *"Pacific Car"*



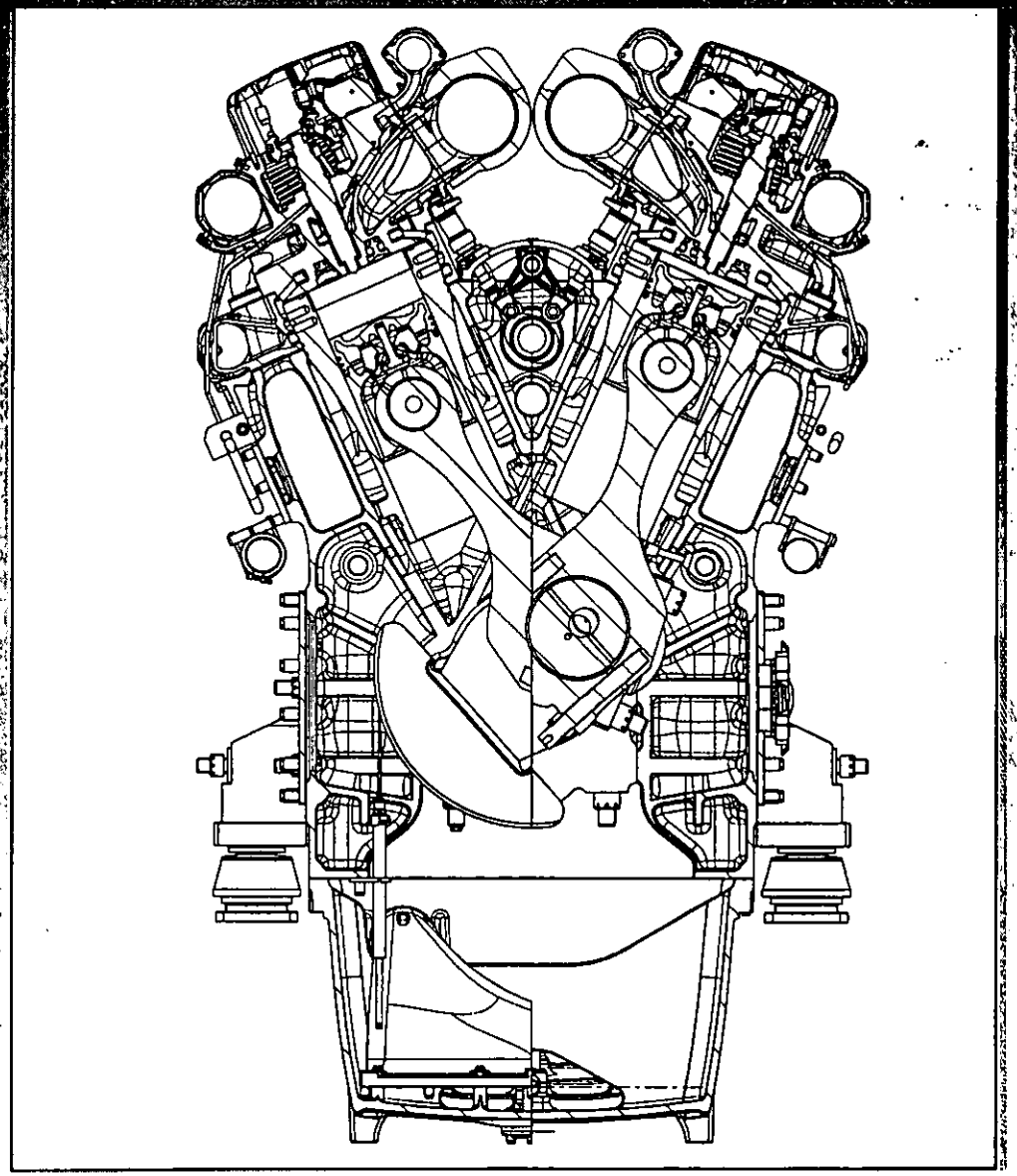
Series MEGA



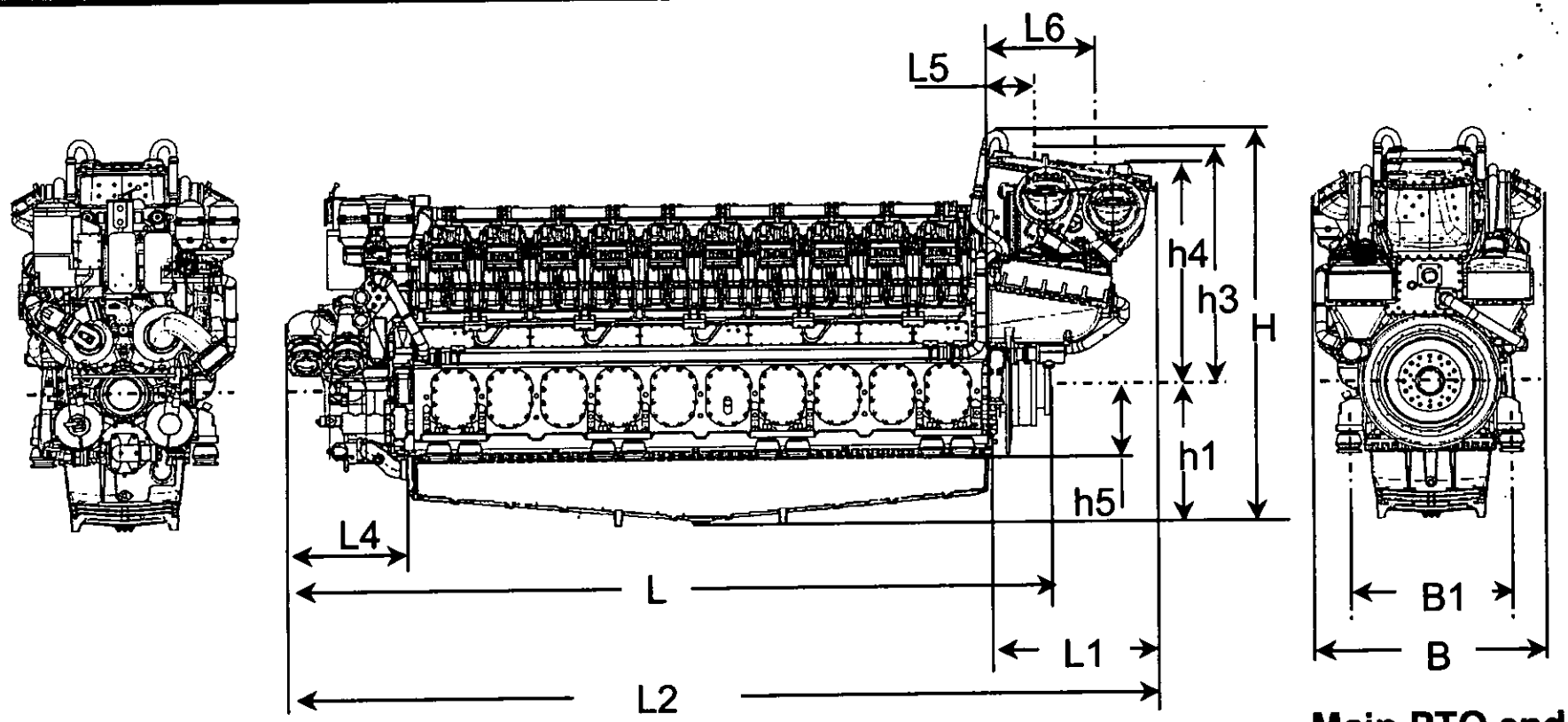


☐ 16V and 20V	48° V configuration	☐ Common rail fuel injection system
☐ Bore / Stroke	265/315 mm	☐ Exhaust emission: IMO-certified
☐ Displacement per cylinder	17.37 liter	☐ Exhaust turbocharging
☐ Power range	6,560 - 9,000 kW	☐ Optional: Sequential turbocharging
☐ Max. speed	1,150 rpm	☐ Electronic engine management
☐ TBO (1B)	24,000 op. hours	

Series MEGA



Main Dimensions



Aux. PTO end

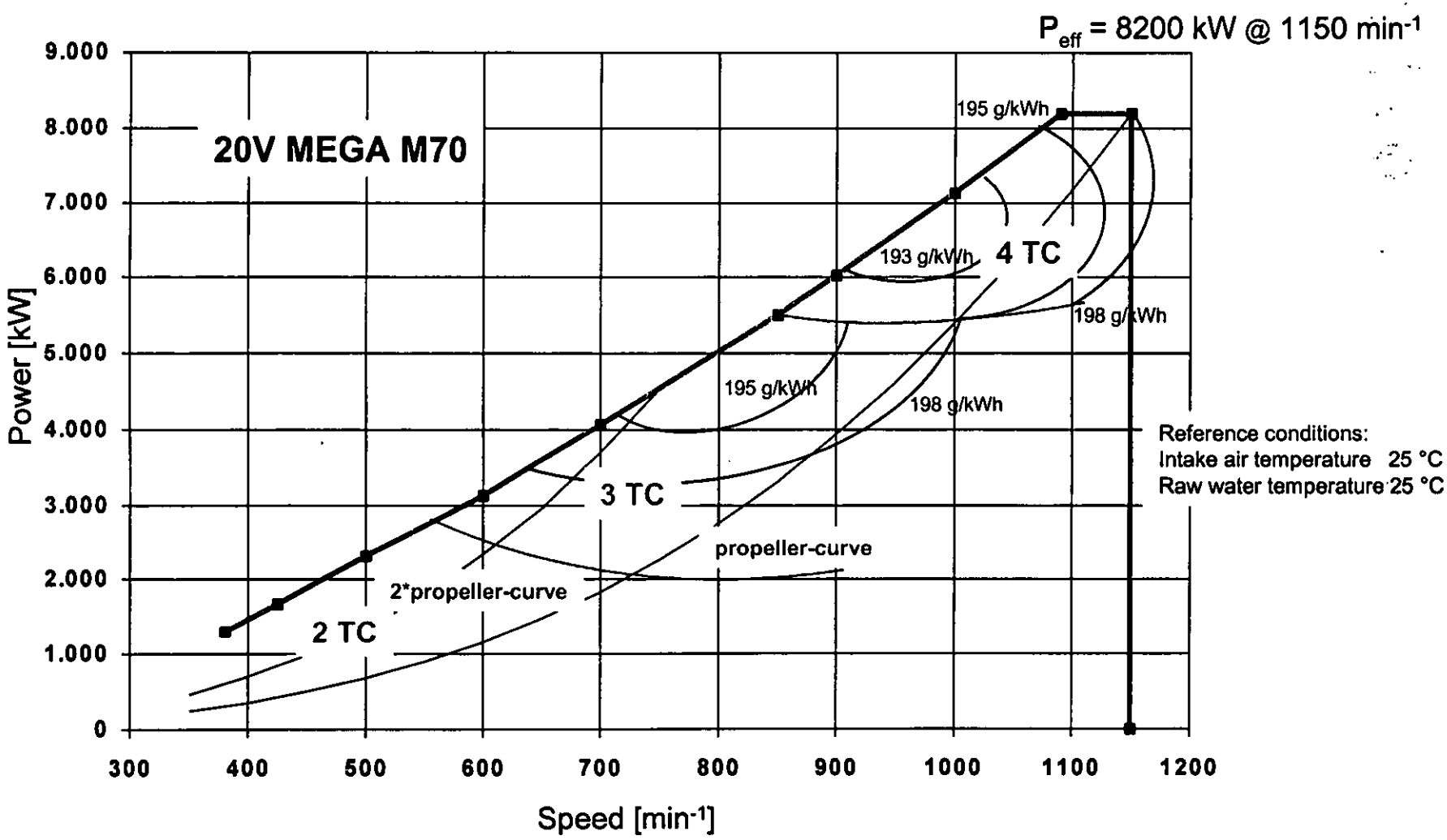
Main PTO end

	L	L1	L2	L4	L5	L6	H	h1	h3	h4	h5	B	B1
16V	5560	1440	6460	800	380	950	3600	1210	2040	1940	642	1900	1600
20V	6500	1440	7400	800	380	950	3600	1210	2040	1940	642	1900	1600

incl. Geislinger coupling

Dimensions in mm

Performance Map with Sequential Turbocharging



WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

GEAR UNITS FOR FAST SHIPS

Karl-Heinz Merk

RENK AG
Gogginger Strasse 73 D86159 Augsburg (Germany)

Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

33rd WEGEMT School
Fast Ship Propulsion Plants
May 22nd – 26th , 2000 Genova, Italy

GEAR UNITS FOR FAST SHIPS

Karl-Heinz Merk
RENK AG, Augsburg (Germany)

General

The fast vessels of the new generation are driven by lightweight diesel engines and/or by the light, spave-saving gas turbines. Besides the high reliability of the propulsion plants, the low weight is of decisive importance.

The gear unit manufacturer has to supply both, lightweight gear units in parallel shaft, planetary and bevel gear unit design and additional components such as gear unit type and diaphragm couplings. Besides this, nowadays the supplier of these propulsion components must be in a position to provide system engineering such as axial, lateral, bending and torsional vibration analyses as well as measuring technological information of the plants.

General

Calculation of load capacity in marine gears units

Production of high-quality gear units
- Gear cutting
- Heat treatment
- Gear unit grinding
- Measuring of gear unit teeth

Testing of gear units
- Idling test
- Load test
- Back-to-back test

Noise level
- Air-borne noise
- Structure-borne noise

Lubrication
- Pump arrangements
- Oil type
- Cleaning of oil

Ship and system related influences
- Input side
- Output side
- Foundation
- Torsional vibrations

Marine gear unit design possibilities
- Co-axial gear units
- Single input single output gear units
- Twin input single output gear units
- Multi-speed gear units
- Combined gear units CODAG, CODOG, CODAG, COGAG

Clutches and couplings
- Multi-disc clutch
- SSS-clutch
- Fluid coupling

Weight-reducing measurements
- Case-hardened gear units
- Gear wheel design
- Gear casings
- Planetary gear units

Power Take-off

Marine gear unit system engineering

General

Fast vessels have to be of lightweight design and to be equipped with lightweight components as a higher amount of water displacement calls for much higher power values.

The use of high-speed diesel engines of up to approx. 8000 kW and gas turbines with a power range of up to 30 000 kW is recommended for the propulsion of fast vessels.

The max. speed for high-speed diesel engines totals 2000 rpm and 16000 rpm for gas turbines.

Today, the optimum speed at a waterjet is approx. 400 rpm for fast and large-size ships, such as ferries. In general, the output speed of watejet-installations ranges between 250 and 1000 rpm.

Calculation of load capacity for marine gear units

Gear unit rating is normally based on ISO 9085. This calculation method is accepted worldwide. Most of the classification societies fully approve of the contents of this calculation method. Even the basis calculations of Dubbel and Niemann do refer to ISO 9085 for toothings. Moreover, they provide pure engineering information as to the shaping of teeth, shaft connections and other general mechanical engineering components.

Fig. 1

Time will certainly be to short to go too much into details of tooth rating but there are a few very important parameters I am going to explain briefly:

Determination of contact stress σ_H

$$\sigma_H = Z_{B,D} \times Z_H \times Z_E \times Z_\varepsilon \times Z_\beta \times \sqrt{\frac{F_T}{d_1 \times b_u} \cdot \frac{u+1}{u} \times K}$$

$$K = K_A \times K_y \times K_v \times K_{H\beta} \times K_{H\alpha}$$

Determination of tooth root stress σ_F

$$\sigma_F = \frac{F_T}{b \times m_n} \times Y_{FA} \times Y_{SA} \times Y_E \times Y_\beta \times K_A \times K_v \times K_{F\beta} \times K_{F\alpha}$$

F_T	=	transverse tangential force at reference cylinder
U	=	gear unit ratio
D_1	=	Pinion diameter
B_U	=	face with
$Z_{B,D}$	=	single pair tooth contact factors for the pinion, for the wheel
Z_H	=	zone factor
Z_E	=	elasticity factor

Z_ε = contact ratio factor
Z_β = helix angle factor
K_A = application factor
K_V = mesh load factor
K_{H_β} = face load factor
K_{H_α} = transverse load factor
Y_{FA} = tooth form factor for load application at the tooth tip
Y_{SA} = stress concentration factor for load application at the tooth tip
Y_E = contact ratio factor
Y_β = helix angle factor
K_V = dynamic factor
K_{F_β} = face load factor
K_{F_α} = transverse load factor

The gear unit calculation programs consider the entire passage through the toothing, i.e. from the first meshing shock over the entire mesh with the counter-tooth and finally the unmeshing shock. Load fluctuations are calculated in detail.

Fig. 2

Besides the calculated tooth load, gear unit toothings are also tested to their limit for example by the Technical University in Munich. In doing so, there is either a focus on tooth fracture or flank stiffness in combination with most differing lubricants.

In conclusion it can be said that all details of gear unit toothings have been investigated. Except for undiscovered material flaws the risk of a gear unit damage is practically excluded.

Production of high-quality gear units

Only highly alloyed materials of maximum purity are selected for the manufacture of gear units. This applies both for the chemical composition of the material and the ultrasonic test. All calculation programs for gear unit toothings and the applied tests are based on a KSR-factor ≤ 1.5 mm^2. Such value can be achieved by conventional equipment and has proven to be correct. After material testing the gear units are pre-machined, e.g. by profile cutting of single parts or by a hobbing procedure.

Highly-alloyed materials, which are not case-hardened or nitrided but only tempered can be finish-machined by hobbing to quality grade 4-7.

Heat treatment

The hardening of work pieces provides for a substantial increase in load bearing capacity. There is a difference in nitriding depths compared with case hardening. The nitrided layer has a depth of approx. 0.6 mm and beyond this section the material experiences a sudden hardness decrease to the strength of the base material. Since the section of maximum compressive strain is located at a depth of approx. 0.5 mm - especially where larger modules are concerned - this hardening procedure is only applied in exceptional cases.
The maximum load bearing capacity of toothings is attained with case-hardened components using 17CrMo6v i.e. as base material. The case-hardening procedure allows the selection of a random case-hardening depth which varies with the duration of the hardening process. Carbon enters the material surface and provides for a structural transformation. The subsequent quenching process provides for a material hardness of 60 Rockwell. One important advantage is the fact that there is a moderate transition between hardened surface and the base material in contrast to the nitriding process where there is a risk of spalling on the surface.

Fig. 3

Gear unit grinding

The hardening process is followed by the finish-machining actions. Nitrided work pieces should not be retouched or only to a minimum extent since this would remove hardened material from the already thin layer. However, a certain amount of residual distortion is unavoidable so that high-precision toothings can hardly be manufactured without regrinding. Case-hardened work pieces, for example, have to be retouched as a matter of principle but the grinding time can be reduced considerably if the skiving is used. The roll grinding process and for generating grinding by indexing has been applied for many years and MAAG and HÖFLER machines have been leading in this sector.
Nowadays this machining technique is almost outdated. Profile grinding techniques are applied to reach a precision of stage 1-2 according to DIN 3962 while the surface roughness is halved against conventional manufacturing methods.

Fig. 4

Measuring of gear unit teeth

After grinding the toothings have to undergo testing on independent measuring equipment. Therefore testing on the manufacturing machine is unacceptable since a default inherent in the machine would not only be transmitted during manufacture of the toothing but also during its testing. Three-dimensional measuring machines may be used, however, preference should be given to special measuring machines which have been designed especially for the measurement of gear units. These machines allow an effective measurement with a precision of $<1\ \mu$.

Ground toothings are finished to maximum precision. The tooth profile correction can be adapted exactly to the load imposed. By targeted tooth tip and root relief the meshing and unmeshing shock is reduced to a minimum. End relief provides for a reduction of tooth root stresses at free tooth ends.

Fig. 5

Testing of gear units

Already during the assembly phase the toothings are caused to enter into mesh and rotate to allow a check of the contact pattern. The contact pattern obtained in unloaded condition is different from that under full load which builds up when passing the different load stages.
Normally gear units are only subjected to an idling test where the gear unit speed is increased to idling speed. Such test provides for detailed information as to efficiency and the lube oil system.

Fig. 6

A performance test run - feasible up to approx. 2000 kW - will provide exact information as to the gear units noise behaviour in addition to the tooth contact and the oil system.
A back-to-back test allows to introduce any amount of power into a gear unit. Two gear units arranged „back-to-back" by means of hydraulic equipment allow an exact simulation of the torques that occur under realistic operating conditions. Measurements will only be obtained from one gear unit since the other one is loaded on the reverse flank. This test provides information on the effective tooth contact patterns of the tested gear under nominal load, maximum bearing temperatures and the noise behaviour to be expected under realistic operating conditions.

Noise level

As a matter of principle, a difference is made between air- and structure-borne noise. While air-borne noise is of secondary importance since adjacent engines/motors and gas turbines create much more sound, structure-borne noise bears great importance. Gear units which are rigidly connected with the foundation tend to transmit the pure tone yielded by the toothing via the ship's structures. Therefore, excessive structure-borne noise originating from the toothing is audible everywhere on the vessel which is, no doubt, annoying. An important aspect for the military sector is the fact that structure-borne noise propagates via sound waves in the water so that a ship's position can easily be localized.

Gear unit noises can be reduced by:

- high toothing precision
- optimum tooth corrections

Fig. 7

An important aspect is the reduction of the meshing and unmeshing shock of the toothing which is achieved by the increase of the overlap ratio. At the same time this means that the straight toothing experiences rather high stiffness fluctuations during the gear unit contact. Such stiffness fluctuation diminishes with an increasing overlap ratio. Since the helix angle of a single helical gear unit is limited to approx. 12° because of the high axial forces, a thrust-collar design allows a helix angle of up to 15°.

The ideal but rather costly solution is the realization of a double-helical toothing with a helix angle of up to 32° and an overlap ratios of max. 5. This means that 5 teeth are in mesh at a time which would reduce load fluctuations to an absolute minimum. Tests have been performed which proved that an increase in overlap ratio by 1 provides for a noise reduction by 3 dB.

Fig. 8

Lubrication

Pump arrangements

Sufficient lubrication is of vital importance for the gear unit and a decisive factor for a long service life. Pressurized oil is supplied by built-on pumps. There are primary-driven and secondary-driven (trailing pumps) on the one hand and electrically driven on the other.

As a matter of principle, gear units which are equipped with plain bearings do require prelubrication to ensure that the bearing is already lubricated before the initial start-up. This is the only way to minimize mixed friction during the start-up phase. Gear units equipped with antifriction bearings do not require prelubrication since they have been designed in a way to ensure that a certain amount of oil is retained in the bearings. Built-on pumps will supply oil immediately which facilitates the operation and allows to reduce monitoring to a minimum.

The installation of primary and secondary pumps is recommended for multi-propeller plants. Primary pumps are rotating as soon as the motor is started, while secondary pumps - i.e. pumps which are arranged beyond a clutch - will rotate with the waterjet or the propeller thus ensuring emergency lubrication under any operational condition. The attachment of primary and secondary pumps provides for an independence of the on-board electricity system which is in line with the latest safety requirements.

Oil type

Engine oils of viscosity class ISO VG 68 to ISO VG150 are used for the lubrication of marine gear units. For diesel driven gear units, the same oils are used as for the diesel engines. EP-additives allow to reach the FZG 12 load stage acc. DIN 51354without problems. Every oil supplier is able to provide several oils of sufficient quality. Synthetic oils, are normally not used for marine gear units, but could be.
Beside load stage FZG 12, air separation factor – LAV, acc. DIN 51381 and grey staining (micro pitting) acc. FVA 54/1 GFT>10 is to consider. LAV - air separation factor acc. DIN 51381 is getting worst in case of water or particle content in the oil.

Regular oil analyses are very important. Recently it has become evident that the viscosity behaviour of oils deteriorates considerably after several months of operation. Moreover, an oil analysis allows to determine whether there is abrasion in the gear unit and even where it comes from since the individual metal types are listed separately and/or can be analysed. Regular oil analyses are an important part of every safety check and highly recommendable.

Ship and system-related influences

Input side

Elastic couplings of rubber or steel spring design are required for the purpose of vibration damping between diesel engines and gear units. Moreover, additional offset-compensating, cardanic shaft elements are needed if engines are placed on soft supports. Gas turbines are usually connected with the gear unit by diaphragm couplings allowing a necessary offset compensation. Gas turbines do not require torsional vibration-damping couplings.

Output side

The connection between output shaft and the continuing propeller shaft is of same importance as on the input side, as already mentioned. Basically, it must be distinguish between a complete and flexible separation on the output using a flexible coupling and a conventional shaft train - which is rigidly connected with the gear unit and fixed in its position by one or several bearings in additional to the stern tube. After many years with problems on the diesel engines, gas turbines, flexible couplings in gear units and the waterjets aboard fast ferries it has now become general practice to separate the gear unit at the input via flexible and/or elastic couplings and by flexible couplings at the output

Fig. 9

A novelty on the market are high-capacity composite fibre materials used for flexible couplings and shafts. The extra costs involved for such components are compensated by a corresponding weight reduction which depends upon the arrangement chosen.

Foundation

Ship foundations are not absolutely stiff but are subjected to deformations and changes as a consequence of sea disturbances and temperature fluctuations. The degree of deformation is higher on lightweight ships and increasing with higher speed. For this reason, gear units have to be adapted to these changed conditions, by keeping the gear unit's contact surface on the foundation as small as possible. Where feasible, a three-point support should be chosen (RENK patent no. 3029336).

Fig. 10

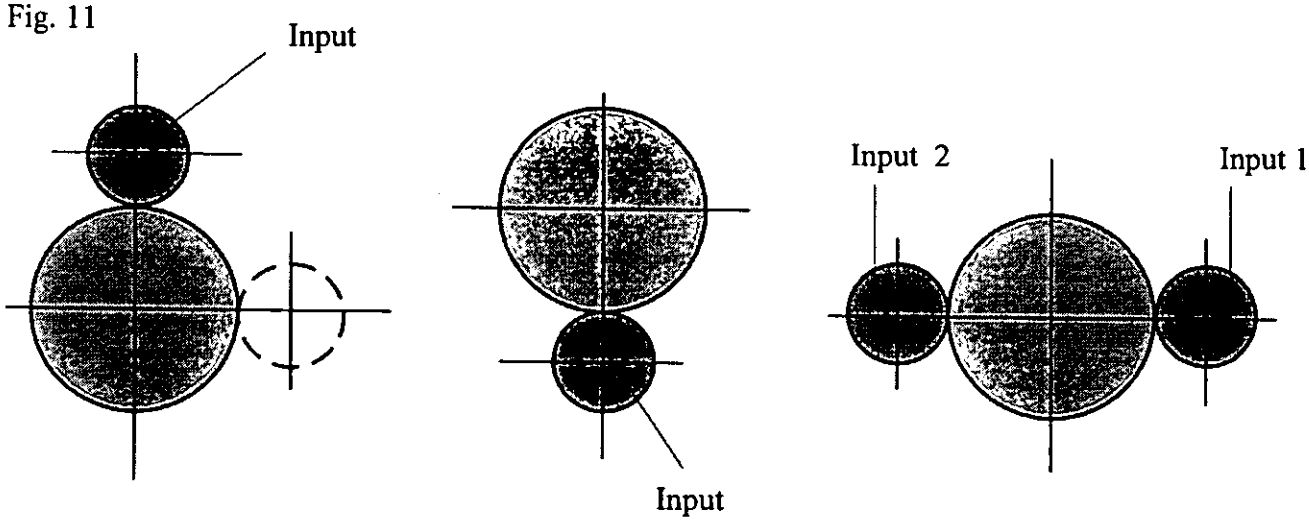
Both measures call for a reinforced structure of the gear unit casing. An optimum casing design allows the gear unit to be supported by elastic elements with no negative influence on the operational safety. Finite element calculations are helpful with regard to an approximation of suitable gear unit constructions. Final values can, however, only be determined by corresponding calculation. By use of double helical gear units unsymmetrical gear unit casing deformation can easily be avoided.

If this target is pursued consistently, gear unit damage - which has occurred frequently on the first fast ferries - can be avoided since the gear unit's behaviour is very close to the calculation model, however, this requires a correct foundation.

Marine gear design possibilities

Gear units are used to provide for a speed adaptation of the driving equipment to that of the waterjet or the propeller. Besides, gear units allow an optimization of the arrangement of the engines towards the waterjets. Co-axial gear units, gear units with vertical offset, horizontal offset and all variants in between are available. Gear units where the power of one engine is distributed to two outputs and gear units where the power of 2 or more engines is summarized to one output only, are also common standard.

Fig. 11



An ever increasing number of compact and lightweight planetary gear units is used as reduction gear units for fast vessels. The extraordinarily compact design of these gear units is achieved by the use of case-hardened toothing systems in combination with the torque distribution inside the planetary gear units. The torque is divided among up to 8 planets in the planetary gear unit. Thus, the size of pinions and bull gear units is reduced significantly with no influence on the safety level. Depending on the ratio, the planetary gear unit reaches a weight of only 0.33 kg per kW. Moreover, a planetary gear unit offers decisive advantages when it comes to the foundation design with a position influence on both weight and ship construction costs.

Fig. 12

Single input single output gear units

Marine reduction gear units with one diesel engine at the input, a multi-disc clutch for engagement and disengagement of the waterjet and one output are considered common standard. The most evident advantage is their favourable price. Another advantage is the possible variation of the centre distance and of the horizontal/vertical position of the input shaft towards the output shafts. Space constraints aboard the ship can be compensated inside the gear unit. However, it must be considered that this type of gear unit requires a relatively strong and thus expensive foundation.

Fig. 13

Twin input single output gear units

Optionally, two or more engines can be summarized to one output gear via multi-disc clutches and pinions. However, in such case the propeller curve has to be rated for the sum of engines and it has to be considered that the waterjet curve cannot be changed. If, for example, you want to drive the waterjet with only one diesel engine, you need a two-stage gear unit. With restrictions one engine could also be used with a single-stage gear unit, however, the engine can only reach the waterjet's torque within the lower speed range.
Figs 14a/14b

In the past, multi-stage gear units were often used in combination with fixed-pitch propellers. Nowadays these gear units are experiencing a renaissance in modern shipbuilding with waterjets.

Figs 15a, 15b

One aspect is the torque adaptation to the waterjet curve in multi-engine plants.

Multi-speed gears

Beside the torque adoption, the efficiency can be used remarkably by the use of a multi-stage gear unit if c.p. propellers are used. This means that the multi-stage design is not only meant to cover an emergency operation but to take influence on different travel conditions. The present example shows two three-speed gear units which are connected with each other via a Cross-Connect-Gear Unit. The three different speed ranges are required for:

- the operation of one engine on both shaft trains
- one engine each on one propeller, with the Cross-Connect Gear unit disengaged
- one engine each on one propeller in addition to the operation of one centrally located gas turbine on one waterjet.

Since c.p. propellers are concerned, an operation would also be possible without multi-stage design if the propeller pitch was reduced, however, there would be a great loss of efficiency and an increase of noise.

Combined gear units CODAD, CODOG, CODAG, COGAG

Normally combined gear units are referred to as CODAD, CODOG, CODAG and COGAG gear units. „D" stands for diesel drive and „G" for gas turbine. CODAG and COGAG plants are often also mentioned in combination with fast ferries. In this sector, however, this term means that one gas turbine transmits power to one central waterjet and two diesel engines or two smaller gas turbine transmit power to the wing waterjets.

Clutches and Couplings

Multi-disc clutch

Multi-disc clutches are used for the engagement and disengagement of diesel engines. These clutches have proven their reliability. A special valve arrangement allows to increase the pressure inside the clutch very slowly. As a consequence the diesel engine speed is not reduced during engagement which results in a very smooth and clean operation since the diesel engine speed is kept constant.

Fig. 17
Friction clutches which can additionally be used as slip clutches, are provided for a further speed reduction. Different design variants of these clutches are available. While the simple version is just controlled by a „Stop" and „Go" mechanism, improved clutches are operated as modulated clutches which means with an absolutely continuous and freely selectable slip. During low-speed travel, the main engine is running at constant speed at a possibly low level. The waterjet and/or the propeller speed can then be reduced to practically „0" and be operated at random within a range of „0" up to idling speed.

SSS-clutches

SSS-clutches are used in combination with gas turbines. They engage automatically as soon as the gas turbine rotates faster than the gear unit. This means that it can be used both for combined gear units and for a engagement of the gas turbine in single operation only.

Fig. 18

Fluid couplings

Fluid couplings bear the disadvantage that a permanent slip of approx. 2% has to be considered which needs to be cooled as powerloss via the oil. However, this clutch type engages and disengages very smoothly, especially in combined plants. Moreover, the torsional vibration system is almost entirely interrupted by the fluid coupling. This means that torsional vibrations resulting from the diesel engine are not transmitted via the gear unit. The gear units are running very smoothly which results in an increased safety and a noise reduction. Much steadier conditions are reached at the output shaft by the reduction of alternating torques at the gear unit input shaft. Inside the fluid coupling the torque is transmitted through the rotating oil flow. Heat is dissipated by a constant exchange of oil.

Fig. 19

Weight-reducing measured carried out on gear units

Case-hardened toothings

The technology of modern marine gear units is defined by two main characteristics:

- high rate of transmitted power and
- utmost reliability

This can be materialized by case-hardened and high-precision ground gear wheels. If calculation methods are based on DIN and ISO standards or the guidelines of leading classification societies, weight reductions of as much as 50 % are possibly by the application of case-hardening procedures.

Moreover, there is a difference of up to 35 % in maximum admissible torque values presently specified by the individual classification societies, however, those societies who are leading in the construction of fast ferries, have mainly approached to the calculation method according to DIN and ISO.

Gear wheel design

The weight of a gear unit is mainly influenced by the weight of the output gear unit. A welded output gear unit would be the lightest but also most expensive variant. In this way it would be irrelevant whether the gear unit is of though-hardened or case-hardened design. Both material variants have been tested and proven their reliability in practice. Another weight-reducing alternative are special cast wheel body shapes, although their extent is by far less. The most sensitive variant is a gear unit rim. It is essential to correctly consider additional stresses in the shrinkage between gear unit rim and wheel body and when rating the toothing.

Fig. 20

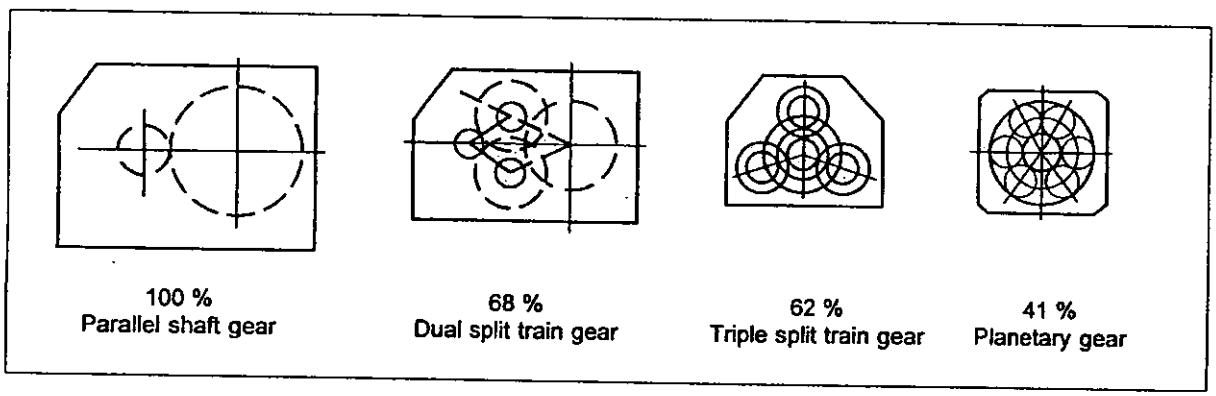
Gear unit casing

Makers of special-purpose gear units for ship propulsion systems have been using aluminium casings for approx. 60 years now. In the past, aluminium casings were mainly used for antimagnetic gear units while this material is nowadays used to reduce the gear unit's weight. Of course, it would be wrong to build a steel casing with identical dimensioning from aluminium. However, if the physical values and an optimum selection of aluminium alloys are considered, one will find out, that this material even offers advantages as compared to steel. Special care and corresponding technical measured have to be devoted to the manufacture of welded casings. The necessarily high differences in material thickness of gear unit casings call for sophisticated welding techniques, not comparable to the welding of ship hulls. It is essential, that the different heat extension between steel and aluminium is considered.

Planetary gear units

Now as before the greatest weight saving potential is achieved by the division of the torque to several gear unit meshes. The dual-split train is used in CODOG gear units and three triple-split train is frequently used for industrial applications. Planetary gear units are the best solution whenever weight reduction is demanded. This applies for ratios of ≥ 2.8.

Fig. 20a



Power take-off

Very often, engageable and directly-driven PTOs have to be mounted on gear units for fast vessels to provide for an optimization of on-board operation. Directly-driven and continuously operating PTOs are used for the drive of pumps for steering of the waterjet and the control of trim flaps.

Engageable PTOs are required for bow thrusters. The mechanical configuration of these PTOs entails a remarkable reduction of required on-board energy and a further weight and cost reduction of the overall systems.
It is of high importance that these PTOs are considered in the torsional vibration calculation.

Fig. 21

Summary

Gear units as such are fully investigated and analyzed machine elements with outstanding safety values. It is important for a gear unit manufacturer to be informed precisely on the exact peripheral details of the individual plant. Besides parameters such as power and speed such details comprise the maximum occurring alternating loads resulting from the propulsion system, additional forces at the input and output originating from the relevant connection elements, the foundation deformation to be expected, other requirements as to weight, noise, engagement possibilities and configuration. Only if all forces and deformations not directly resulting from the gear unit itself are determined in advance and passed on to the gear unit manufacturer, a safe and perfectly operating gear unit can be designed. It is therefore very important to incorporate the gear unit supplier into the project at a very early phase.

Iso Draft International Standard	ISO / DIS 9085
G. Niemann, H. Winter	Maschinenelemente B. II
W. Beitz, K.-H. Küttner	Dubbel
Renk	Technic No. 4, 5, 7, 8, 13, 15, 17, 20, 24
Prof. Dr. Hirt	Economic and Technically optimised gear units
Karl-Heinz Merk	Gear units for fast vessels

DRAFT INTERNATIONAL STANDARD ISO/DIS 9085

ISO/TC 60/SC 2 Secretariat: **DIN**

Voting begins on **1998-11-19** Voting terminates on **1999-04-19**

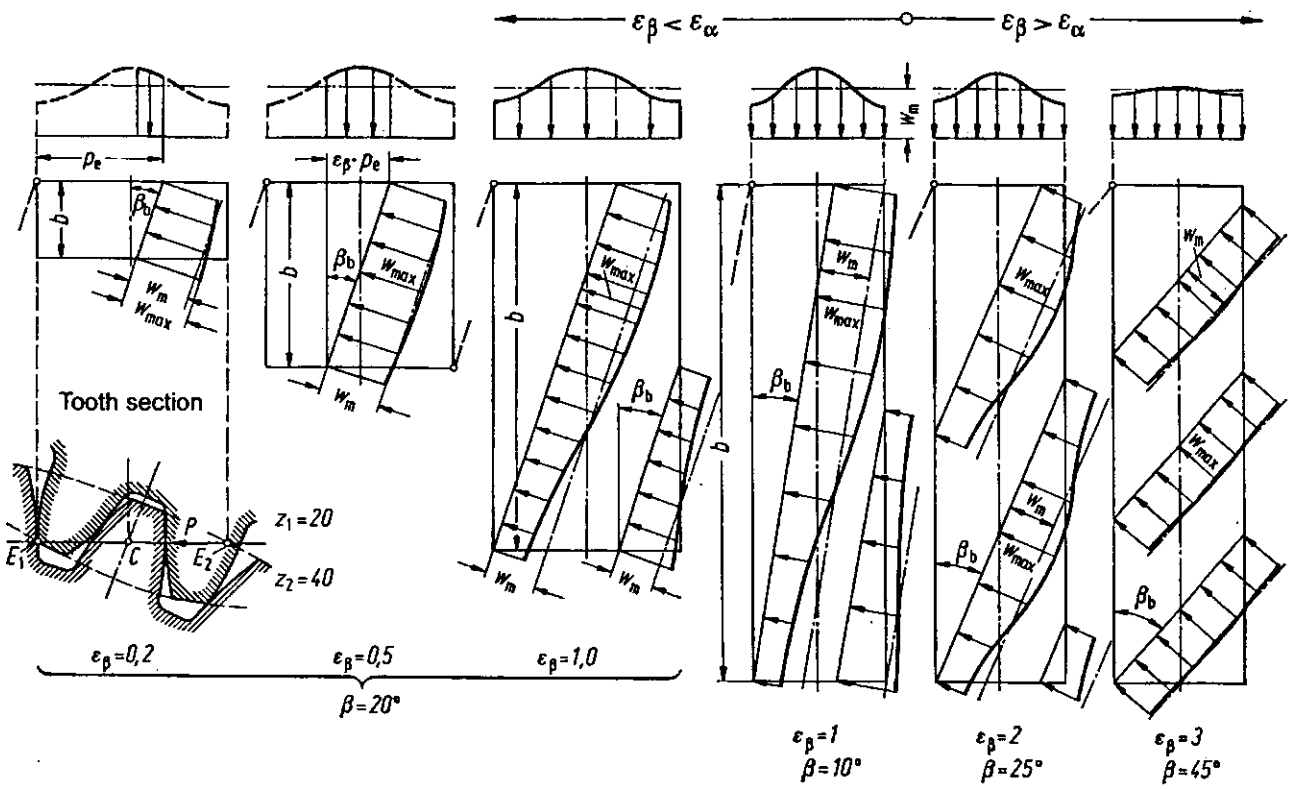
INTERNATIONAL ORGANIZATION FOR STANDARDIZATION • МЕЖДУНАРОДНАЯ ОРГАНИЗАЦИЯ ПО СТАНДАРТИЗАЦИИ • ORGANISATION INTERNATIONALE DE NORMALISATION

Calculation of load capacity of spur and helical gears — Application for industrial gears

Calcul de la capacité de charge des engrenages cylindriques à dentures droite et hélicoïdale — Application pour les engrenages industriels

ICS 21.200

Fig. 1


$\varepsilon_\beta < \varepsilon_\alpha$
$\varepsilon_\beta > \varepsilon_\alpha$
p_e
$\varepsilon_\beta \cdot p_e$
w_m
w_{max}
β_b
b
Tooth section
P
C
E_1
E_2
$z_1 = 20$
$z_2 = 40$
$\varepsilon_\beta = 0{,}2$
$\varepsilon_\beta = 0{,}5$
$\varepsilon_\beta = 1{,}0$
$\beta = 20°$
$\varepsilon_\beta = 1$
$\beta = 10°$
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$\beta = 45°$

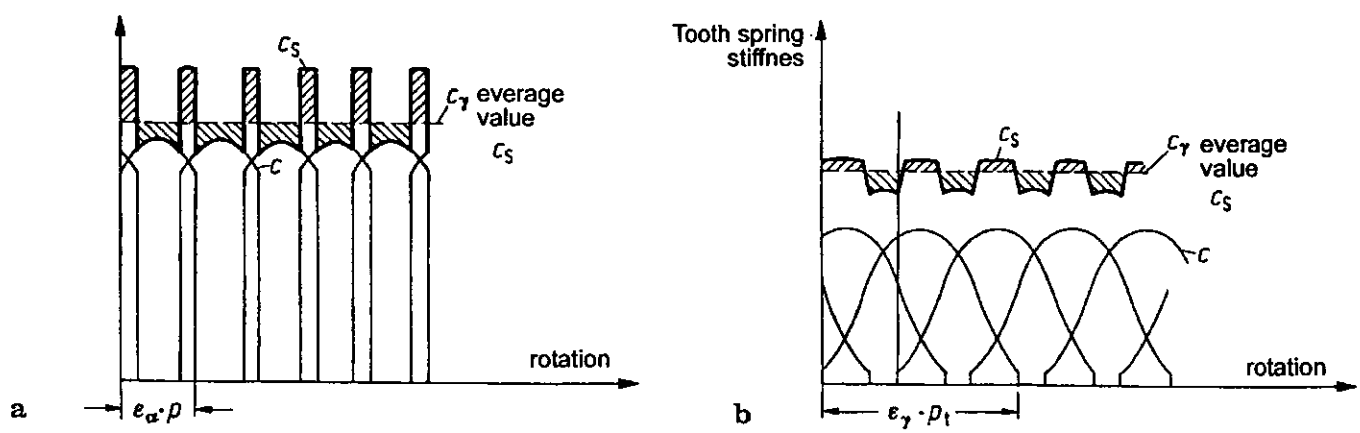

a
c_S
c_γ everage value c_S
c
rotation
$e_\alpha \cdot p$
b
Tooth spring stiffnes
c_S
c_γ everage value c_S
c
rotation
$\varepsilon_\gamma \cdot p_t$

Fig. 2

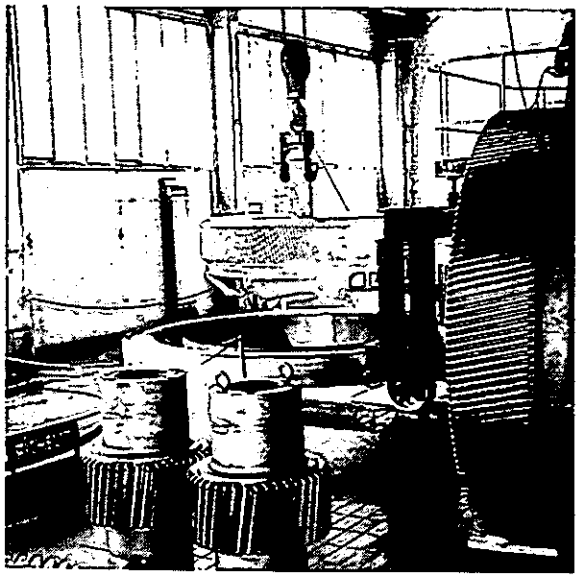


Fig. 3

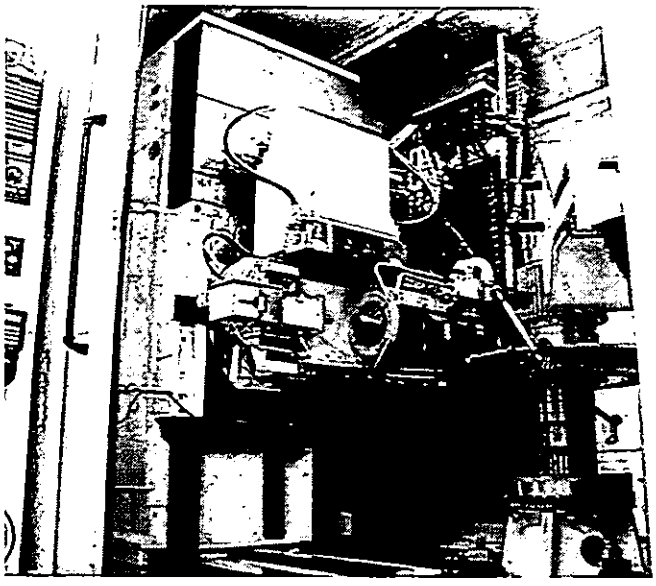


Fig. 4

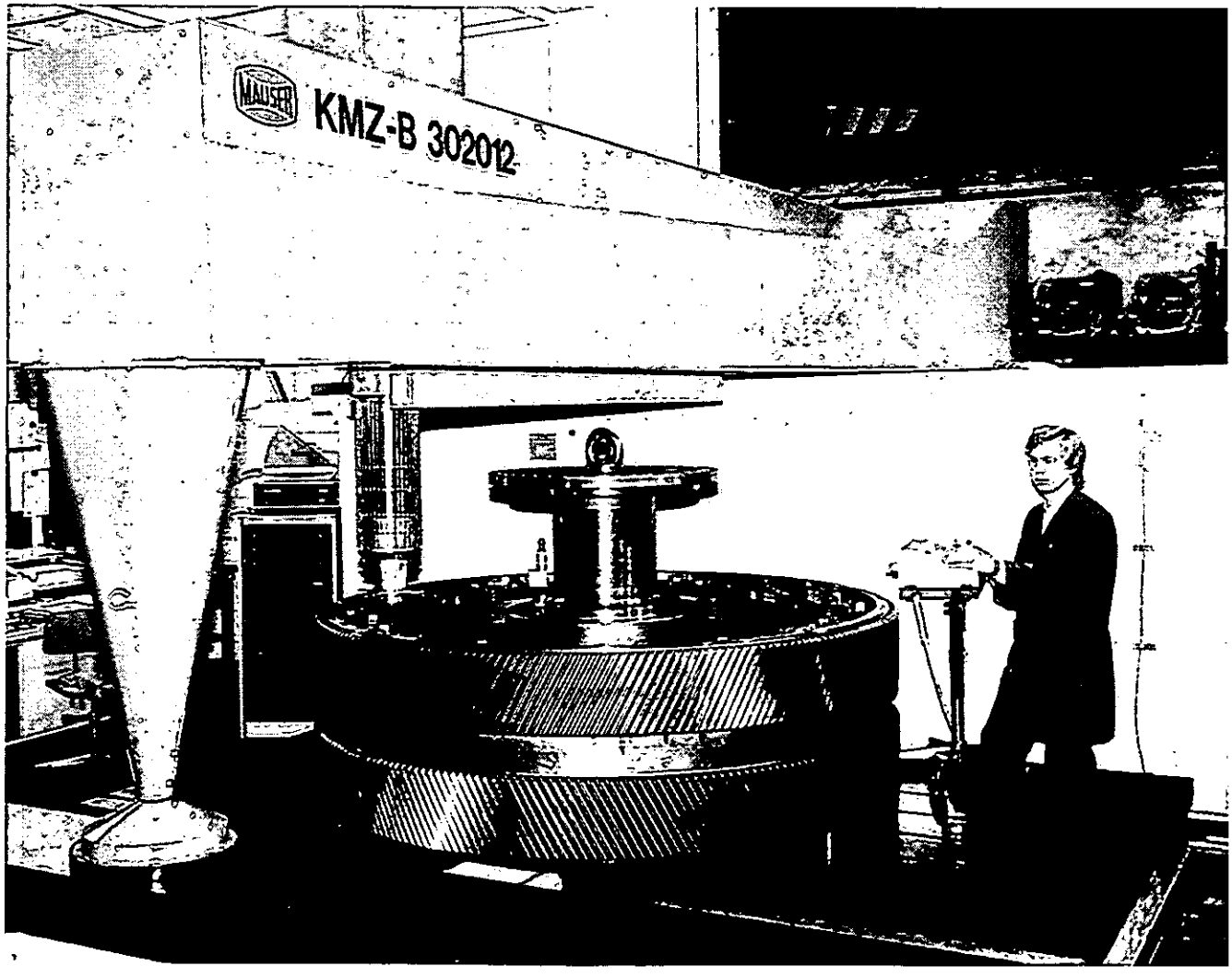
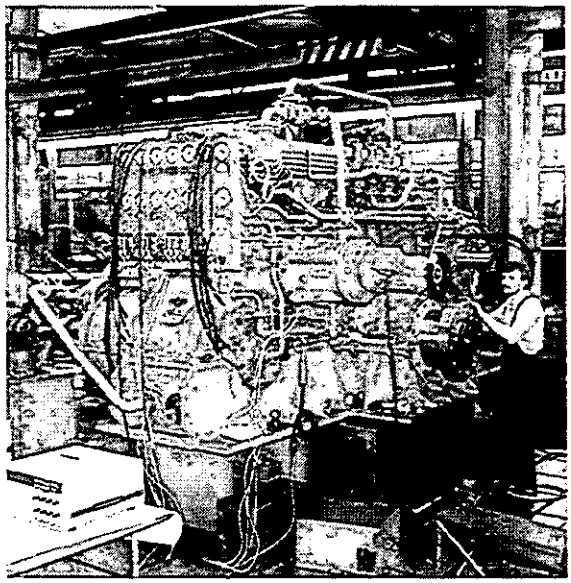
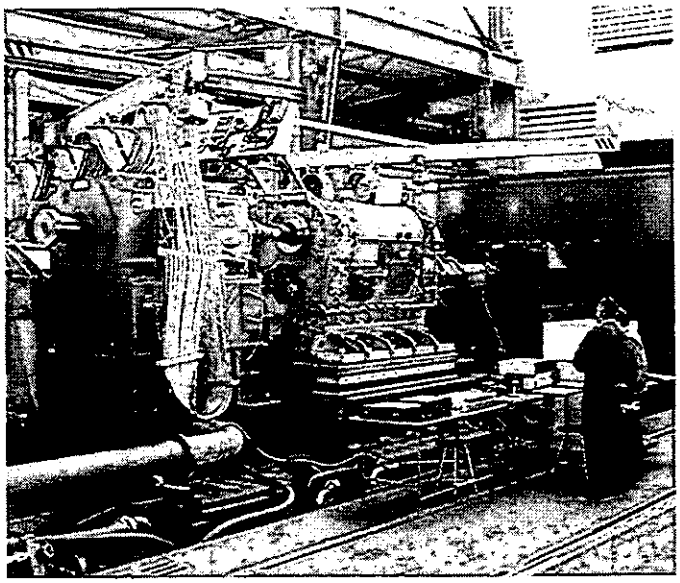


Fig. 5





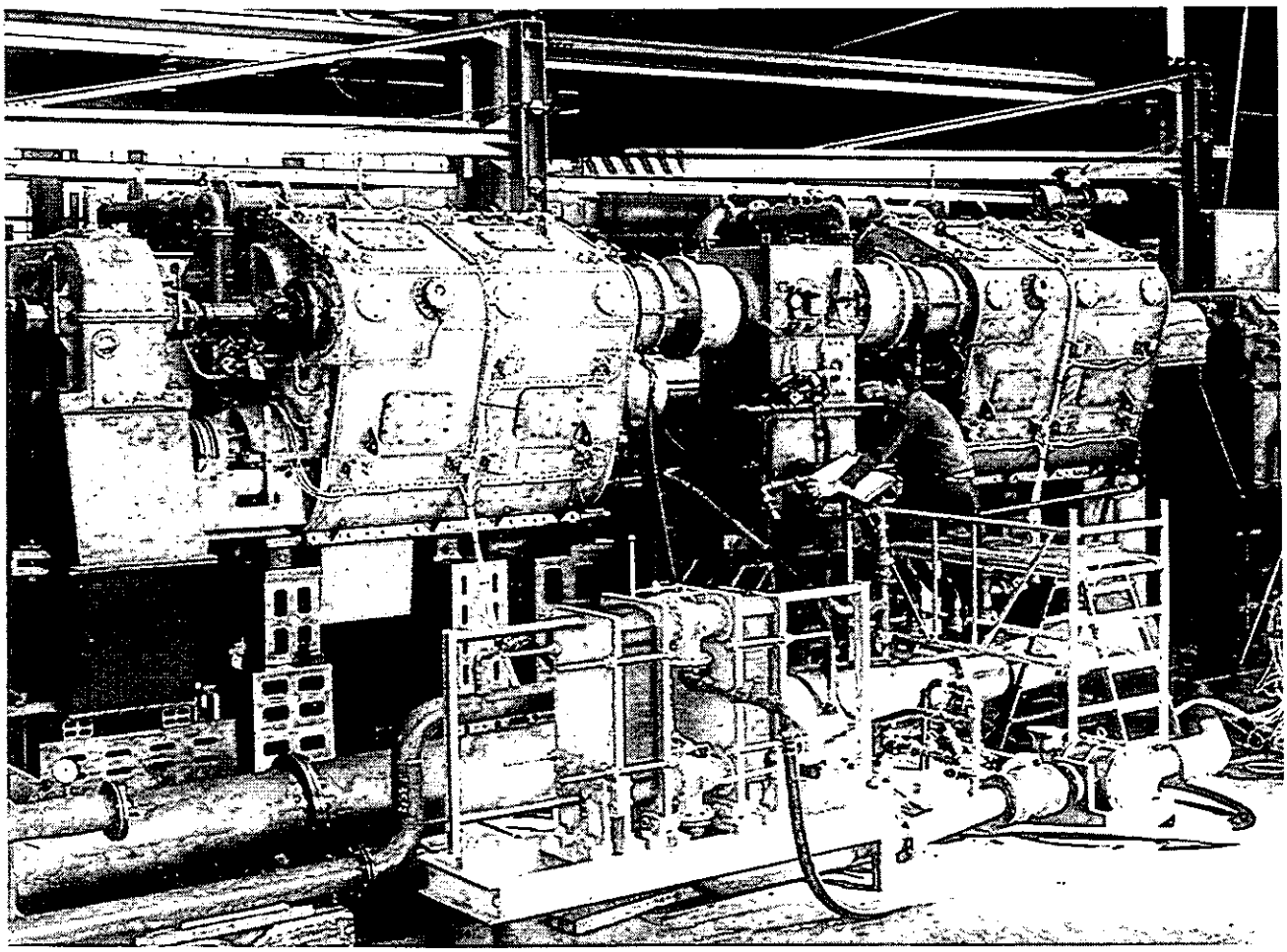


Fig. 6

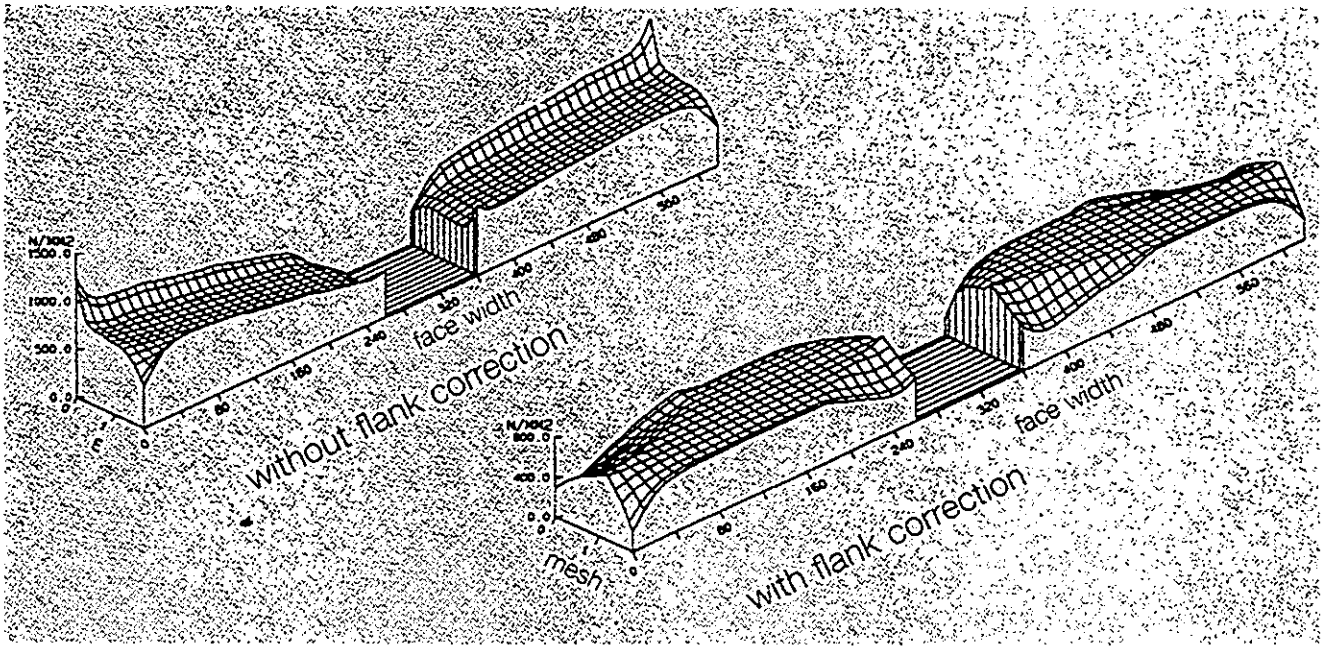
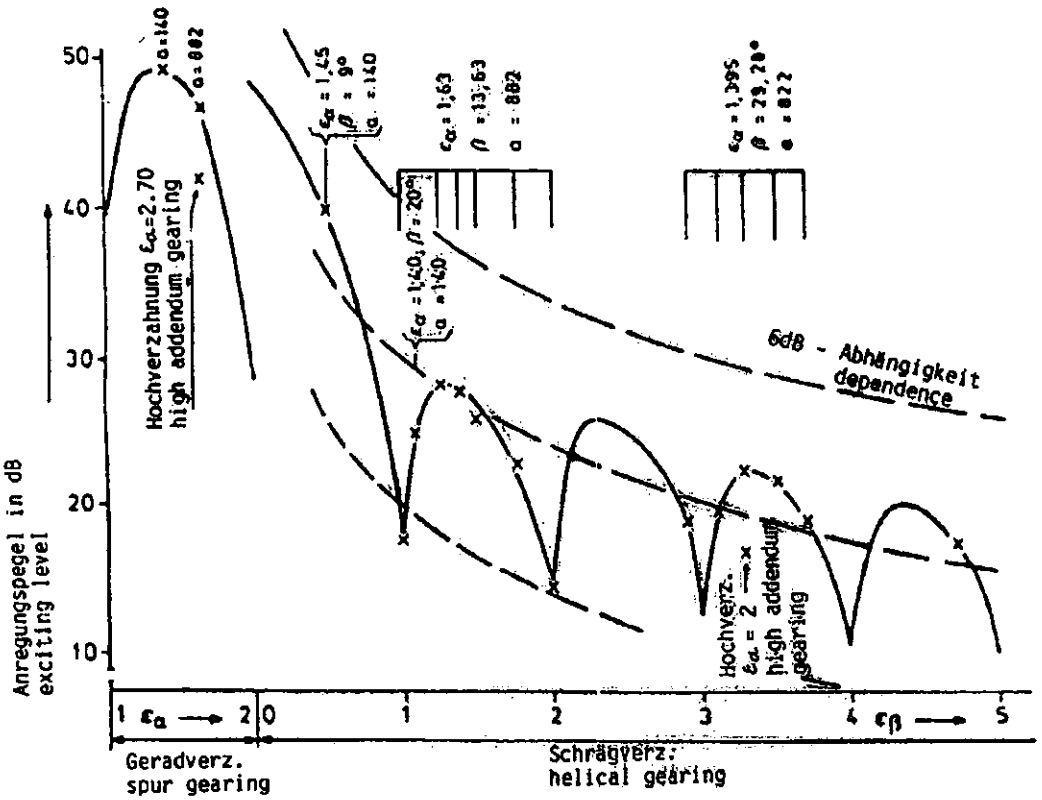

without flank correction
face width
with flank correction
face width
mesh

Fig. 7



Influence of overlap ratio on the excitement level

Fig. 8

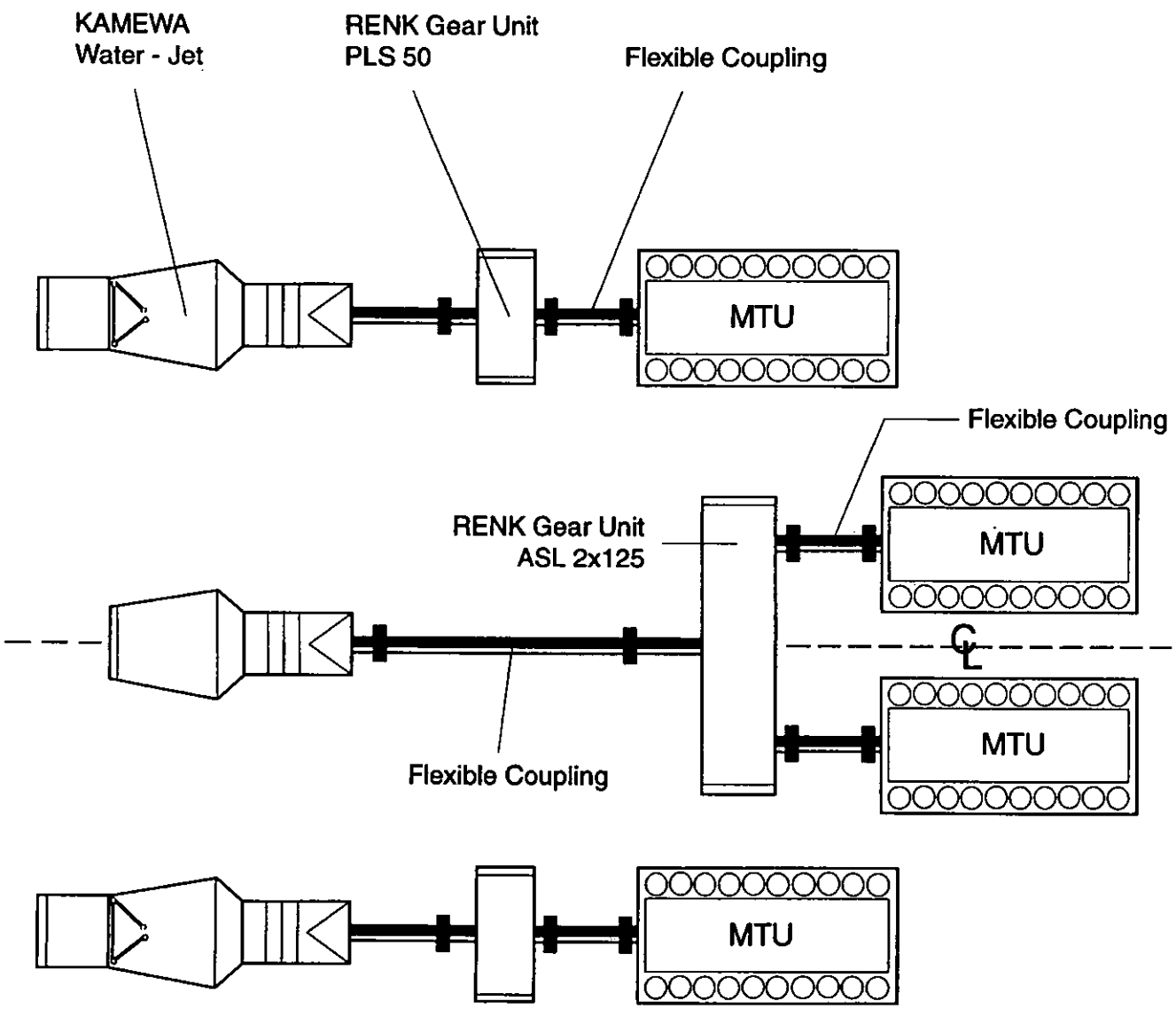
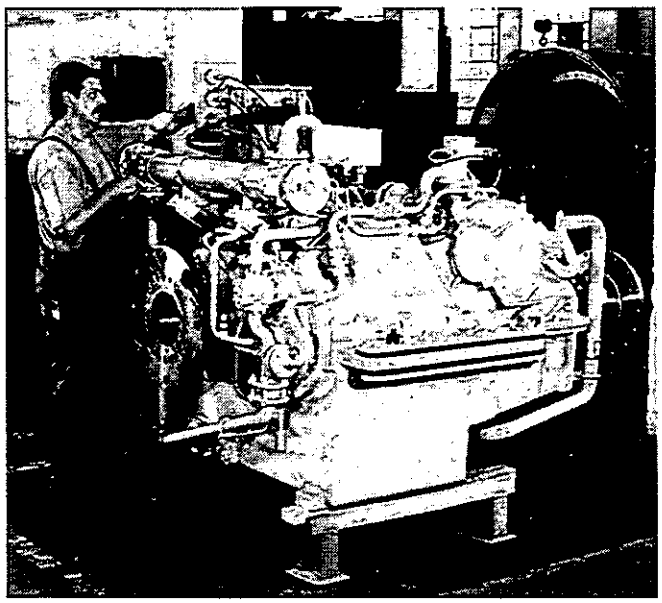
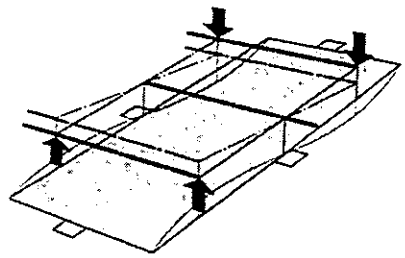
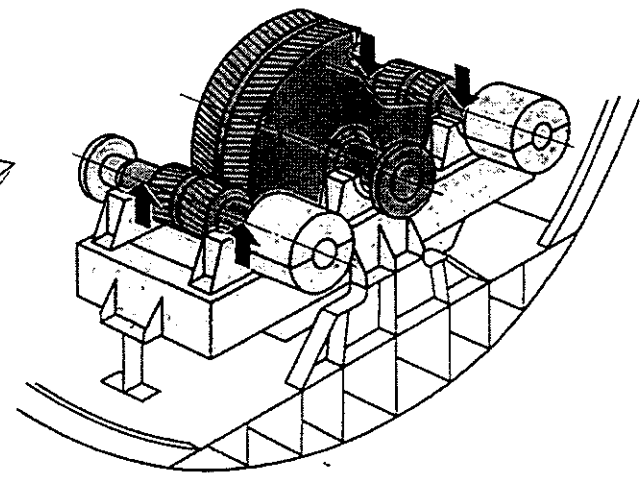
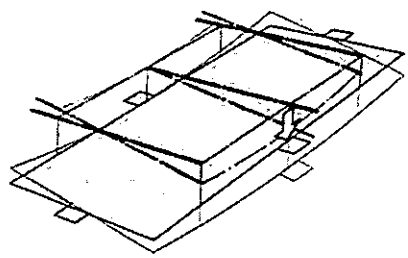

KAMEWA
Water - Jet
RENK Gear Unit
PLS 50
Flexible Coupling
MTU
Flexible Coupling
RENK Gear Unit
ASL 2x125
MTU
MTU
Flexible Coupling
MTU

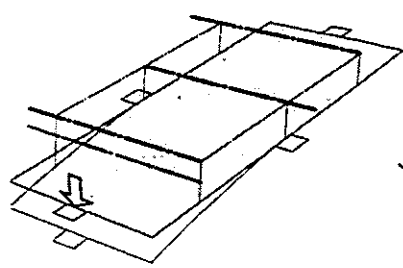
Fig. 9











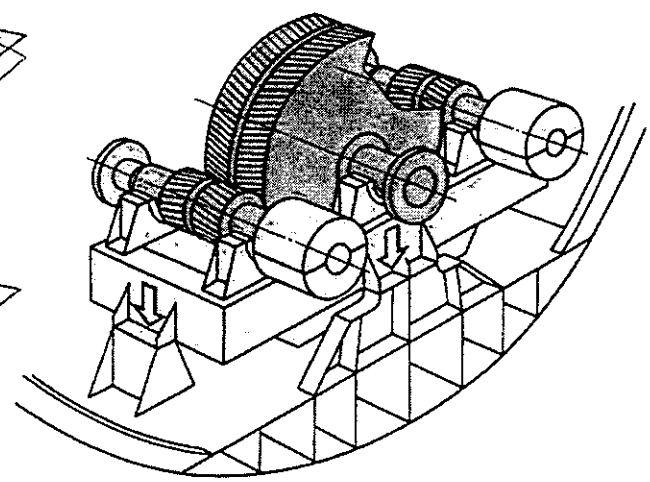


Fig. 10

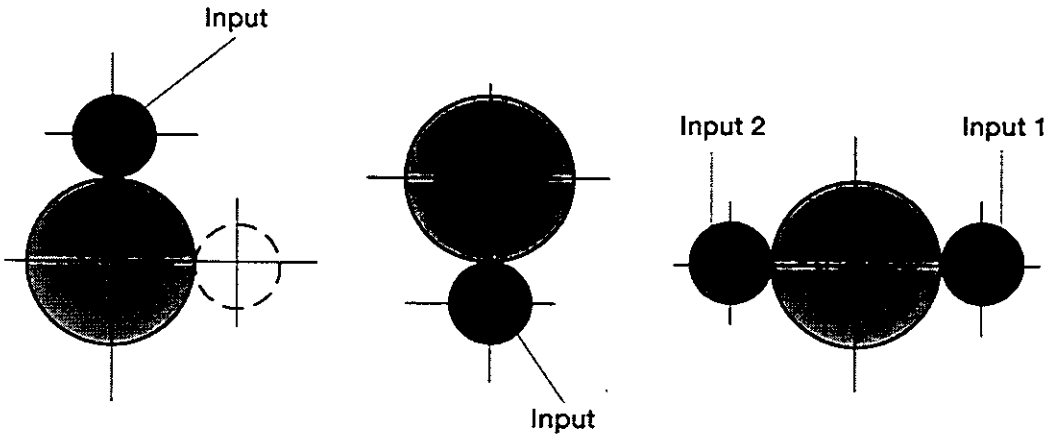


Fig. 11

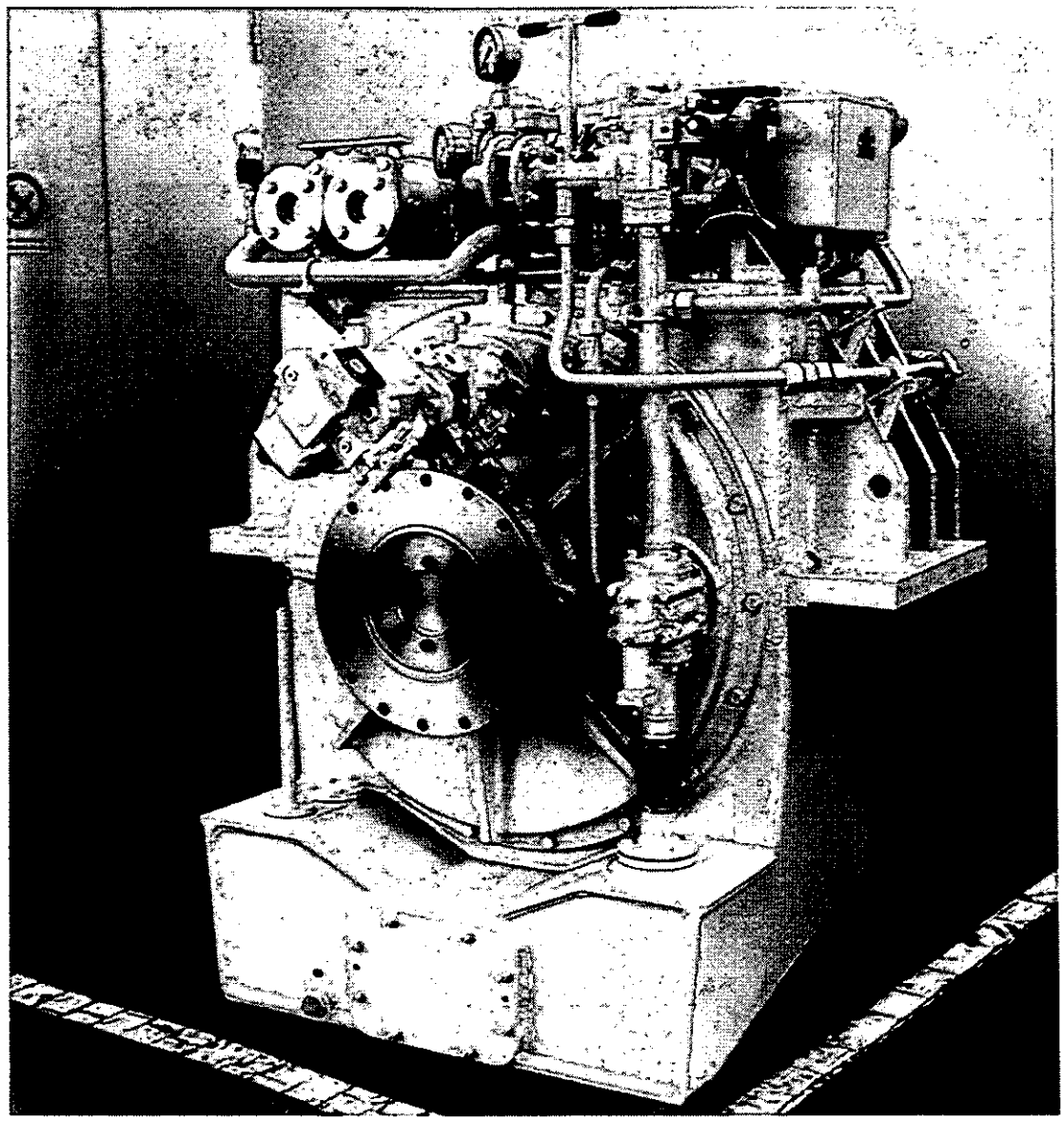
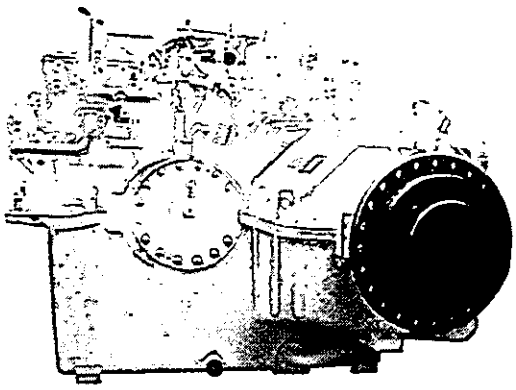
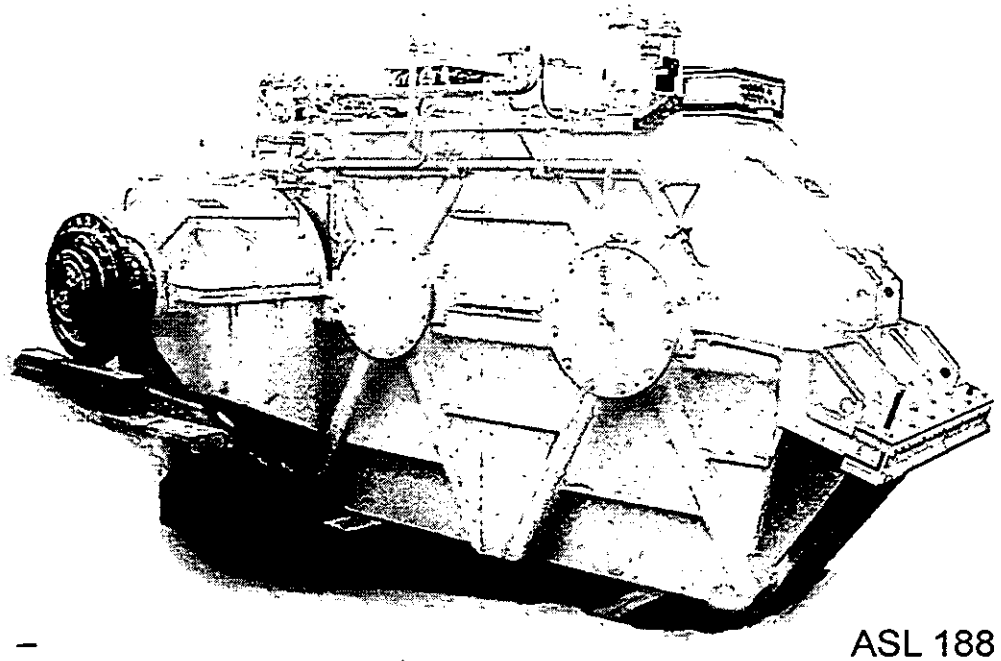


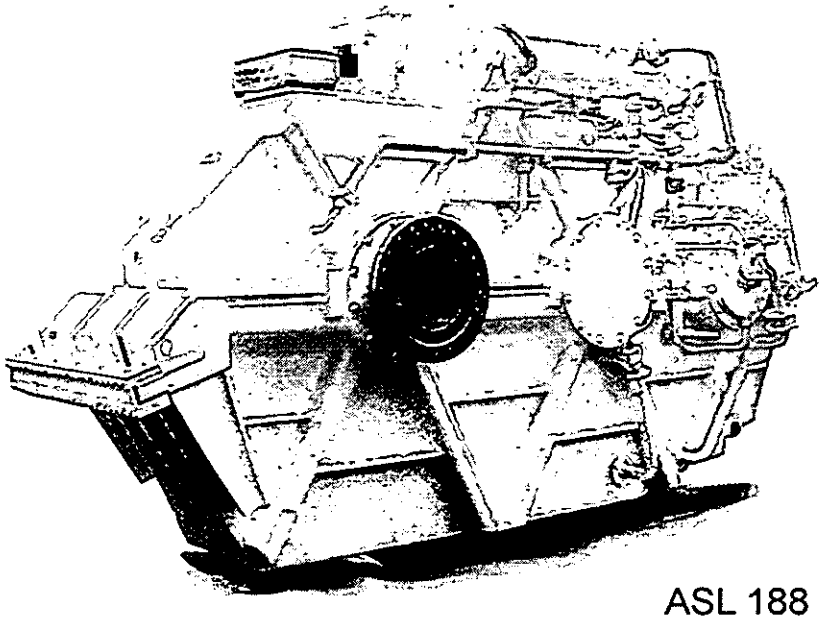
Fig. 12



ASL 62



ASL 188



ASL 188

Fig. 13

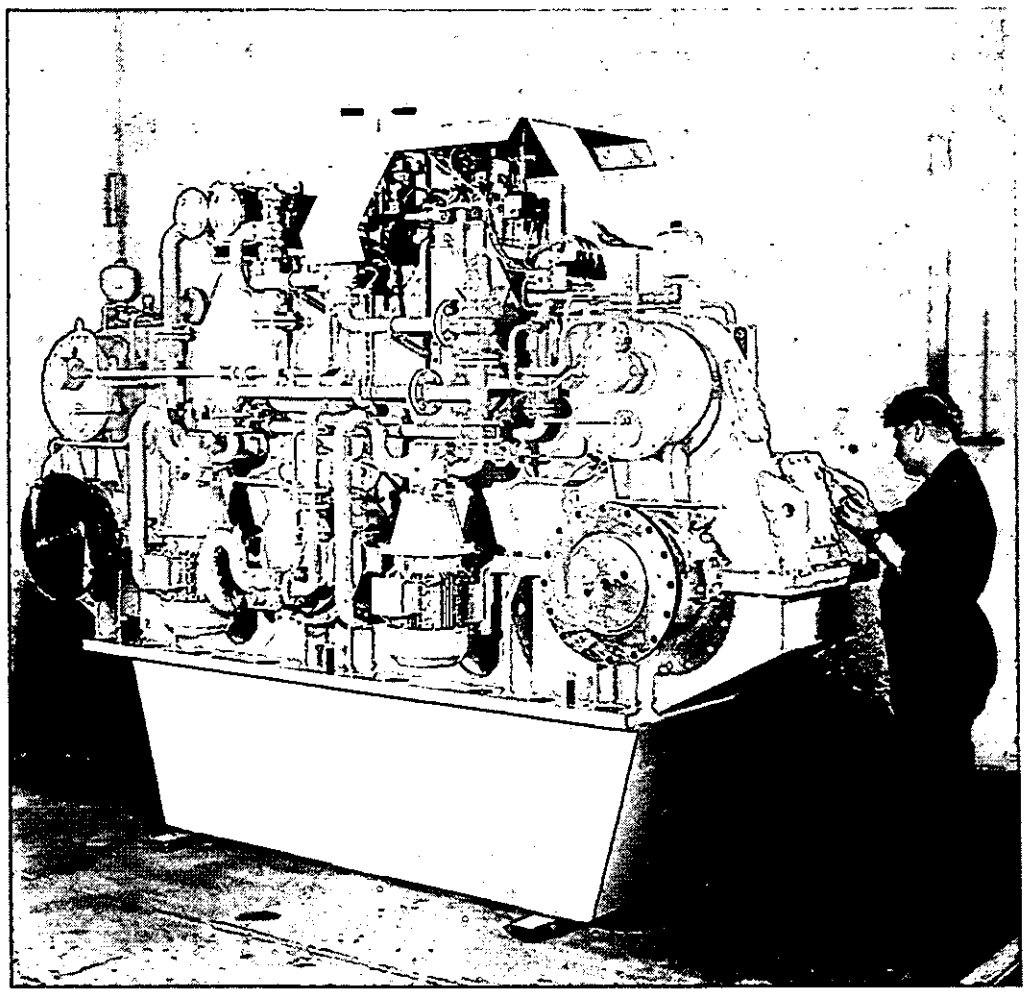


Fig. 14

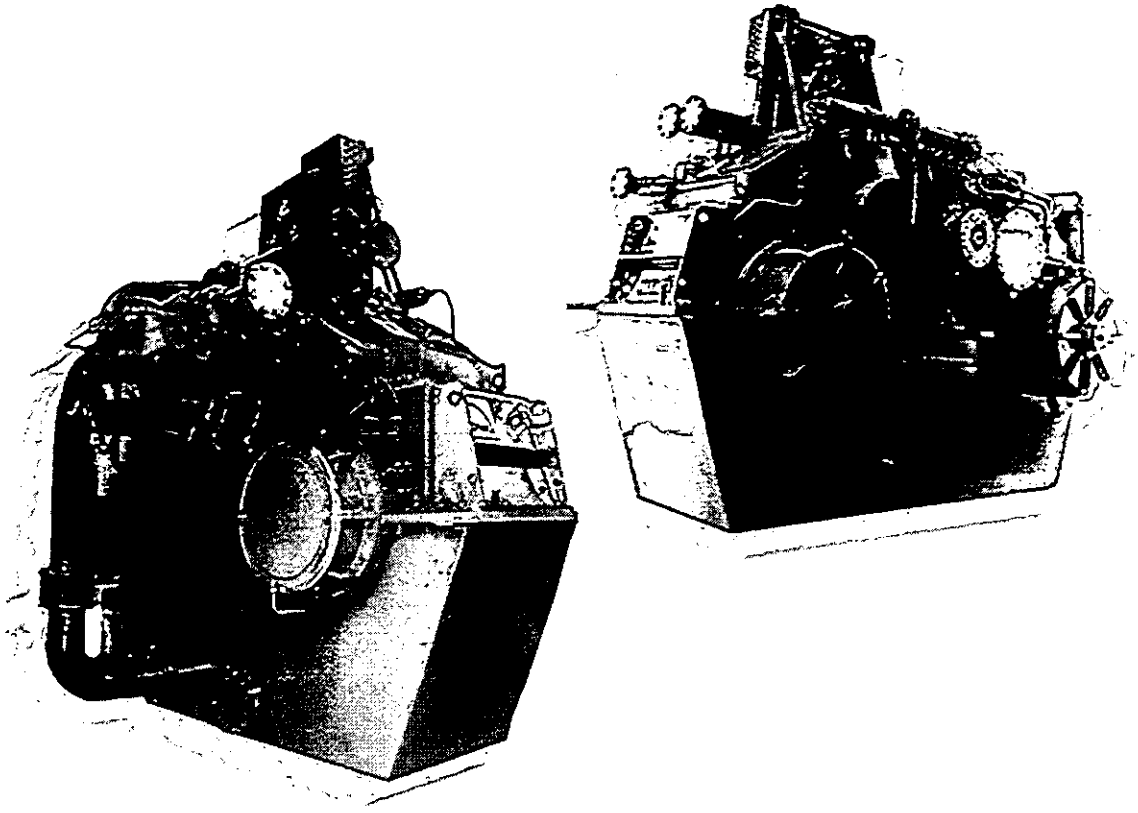


Fig. 14a

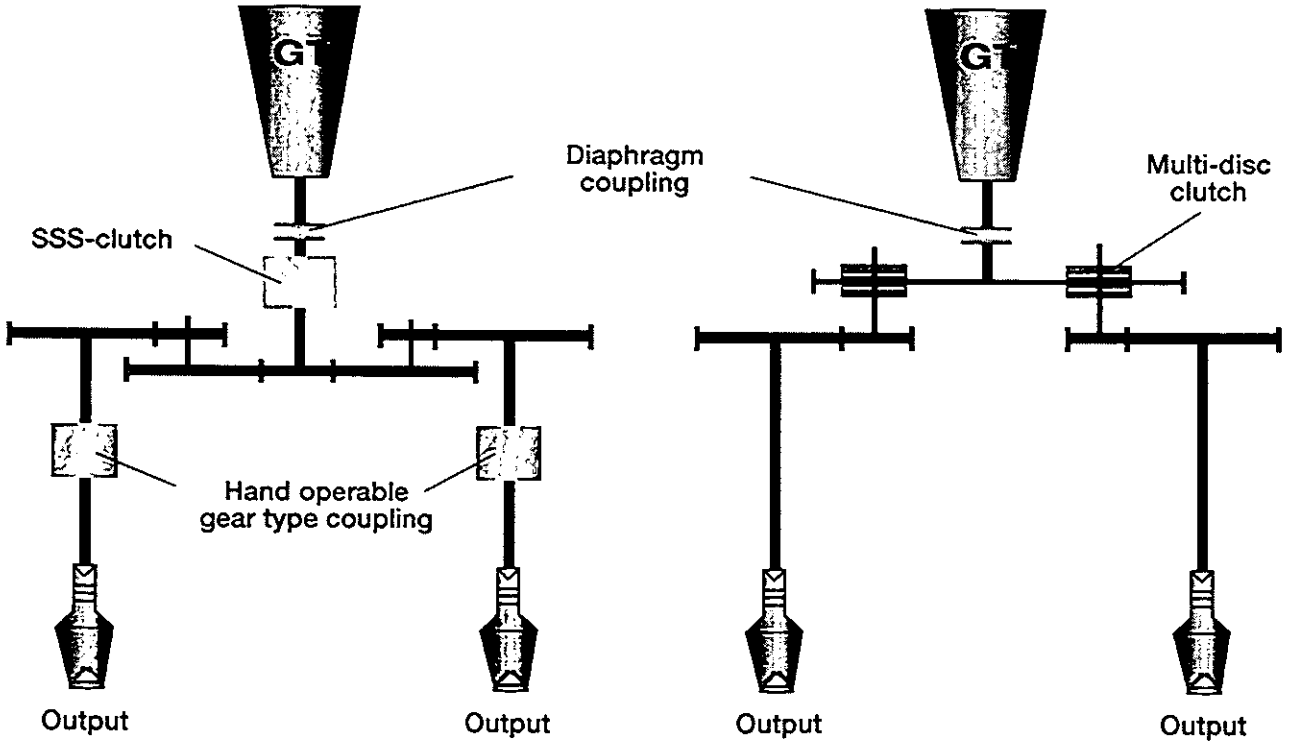


Fig. 14b

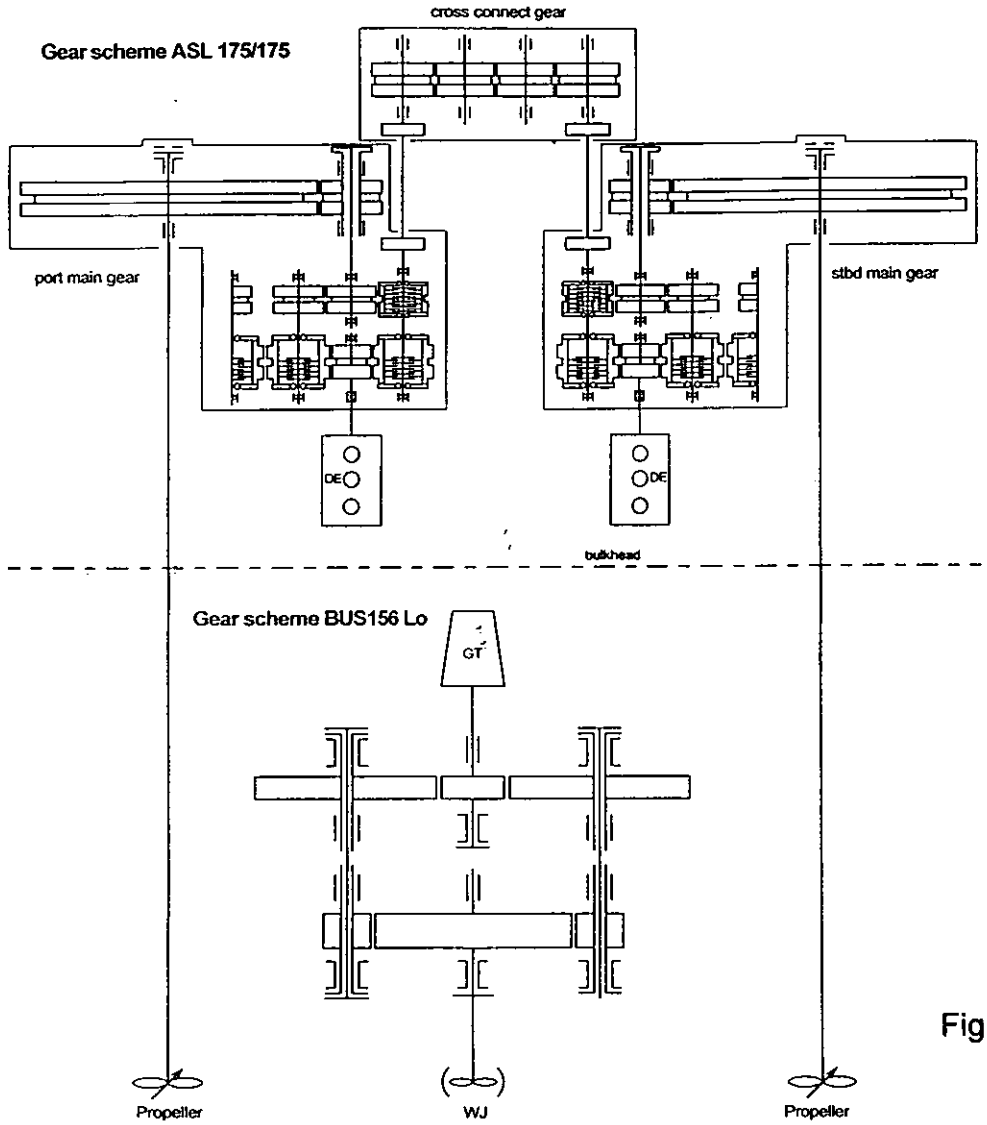


Fig. 15a

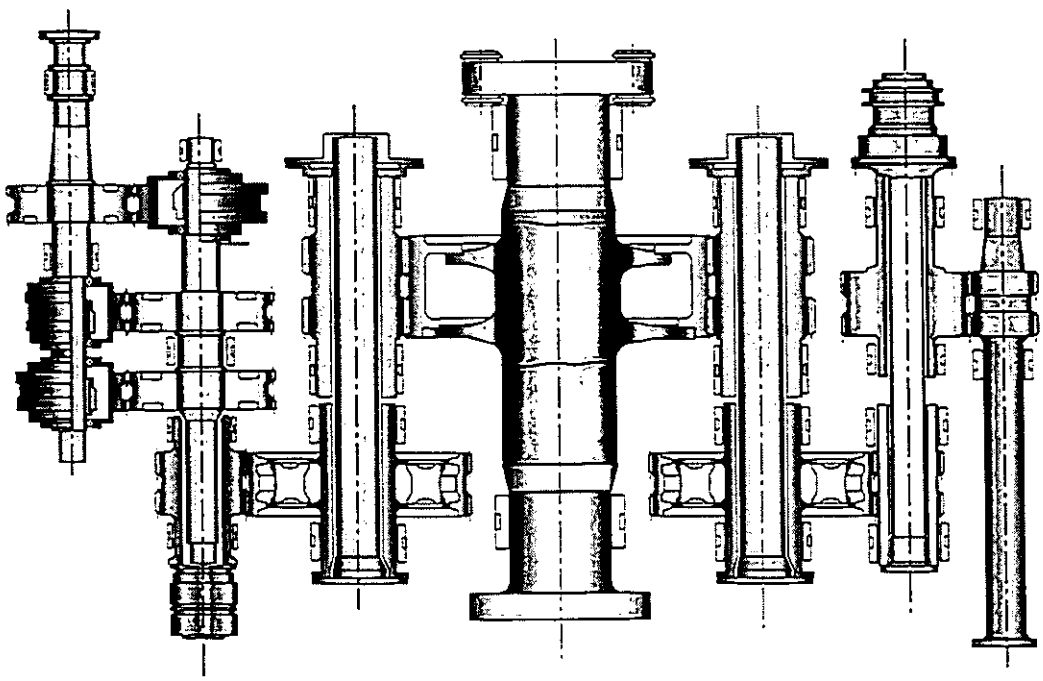


Fig. 15b

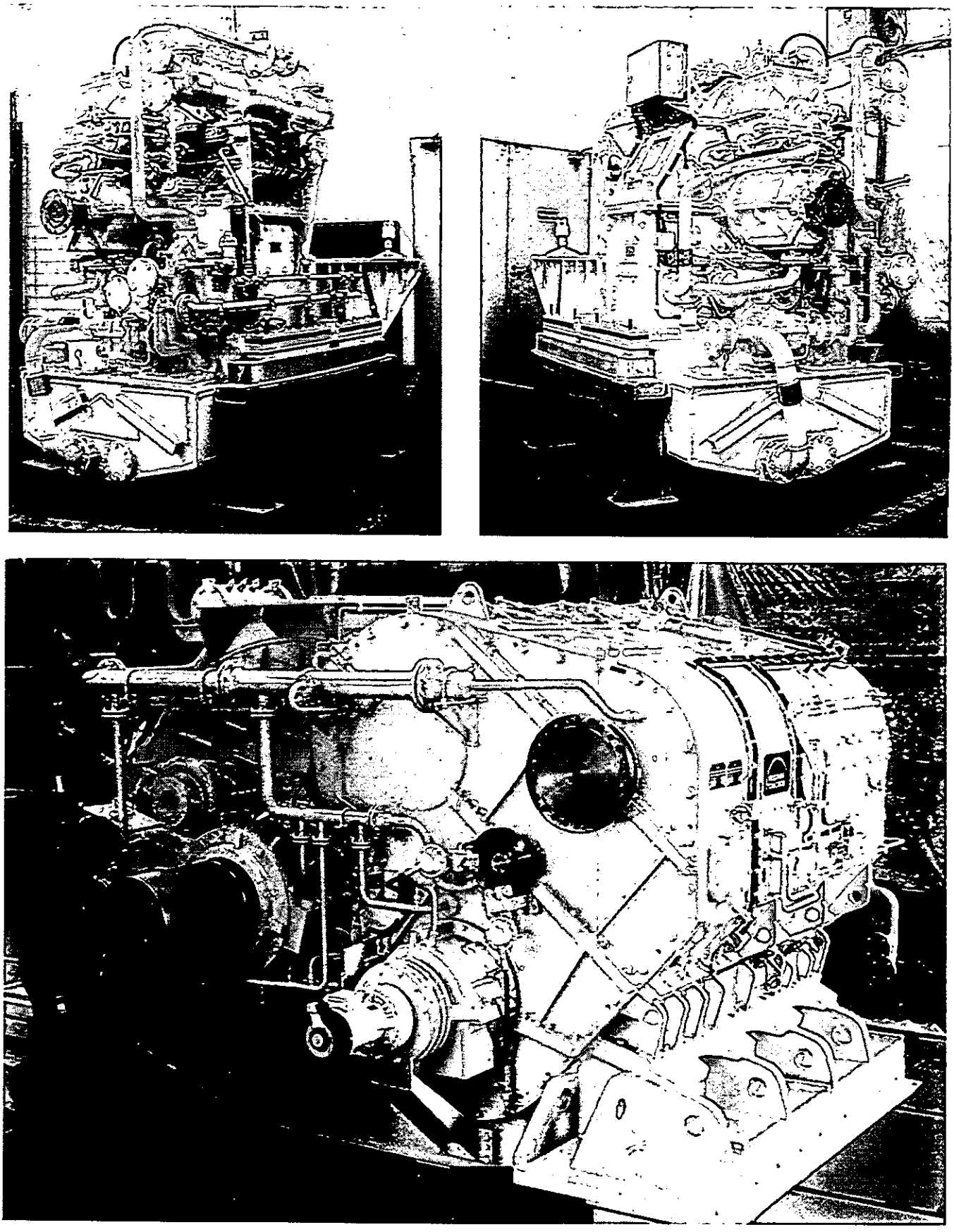
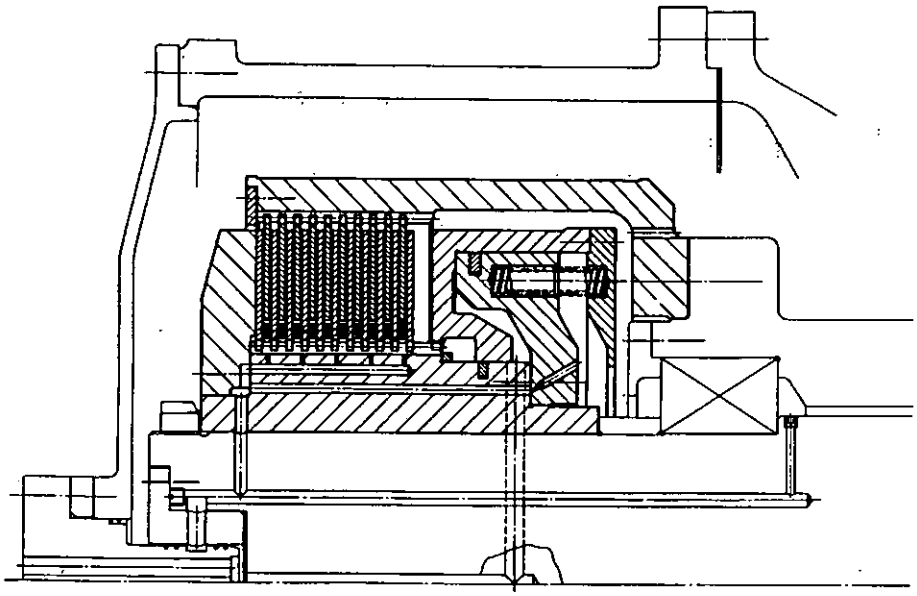
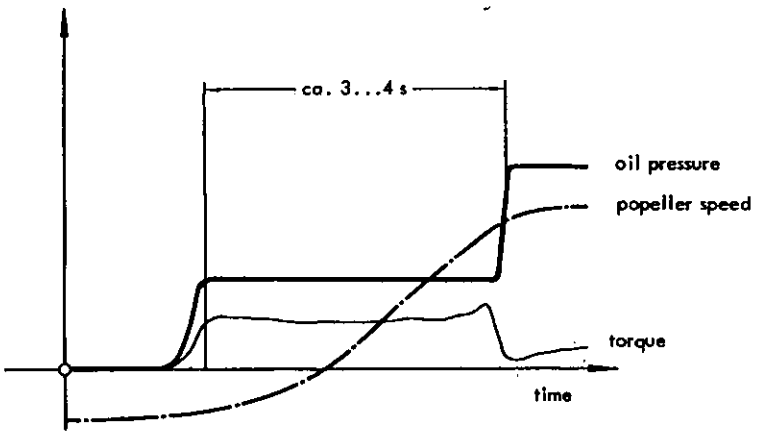


Fig. 16




ca. 3...4 s
oil pressure
popeller speed
torque
time

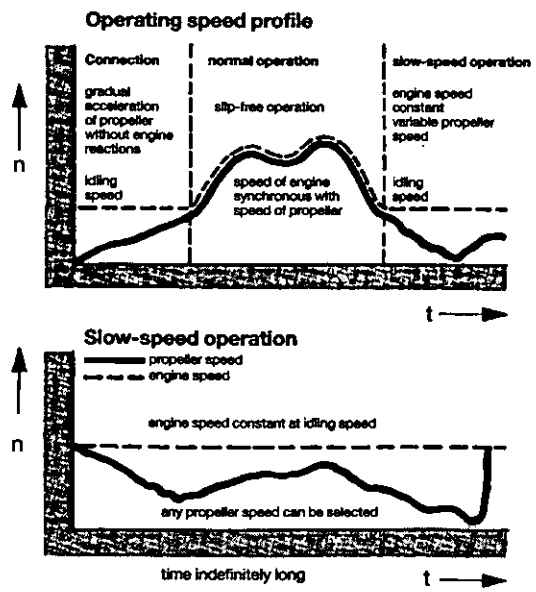
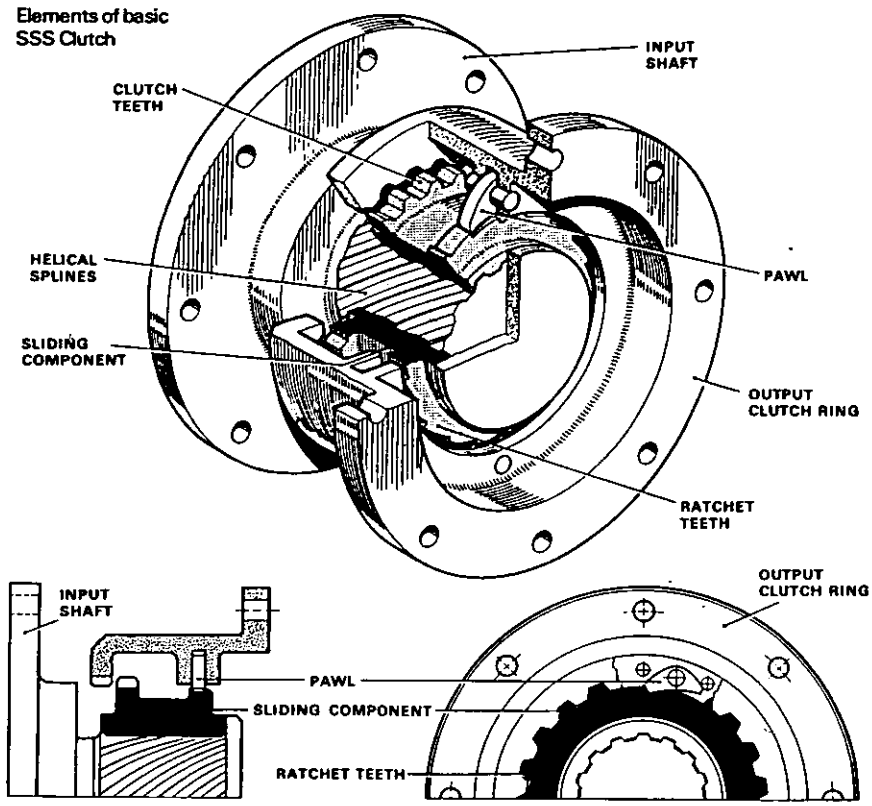
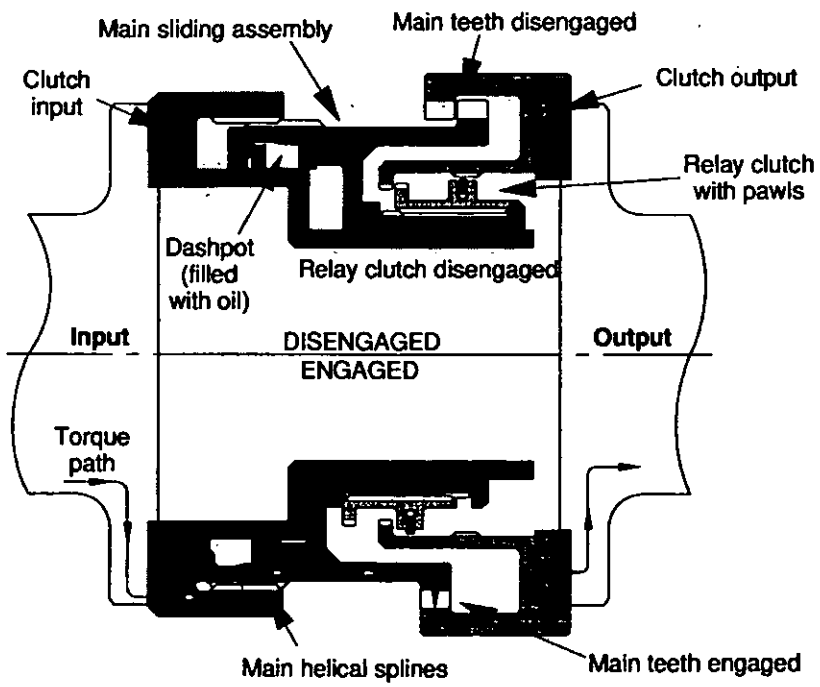

Operating speed profile
Connection
gradual acceleration of propeller without engine reactions
idling speed
normal operation
slip-free operation
speed of engine synchronous with speed of propeller
slow-speed operation
engine speed constant variable propeller speed
idling speed
n
t
Slow-speed operation
propeller speed
engine speed
engine speed constant at idling speed
any propeller speed can be selected
n
time indefinitely long
t

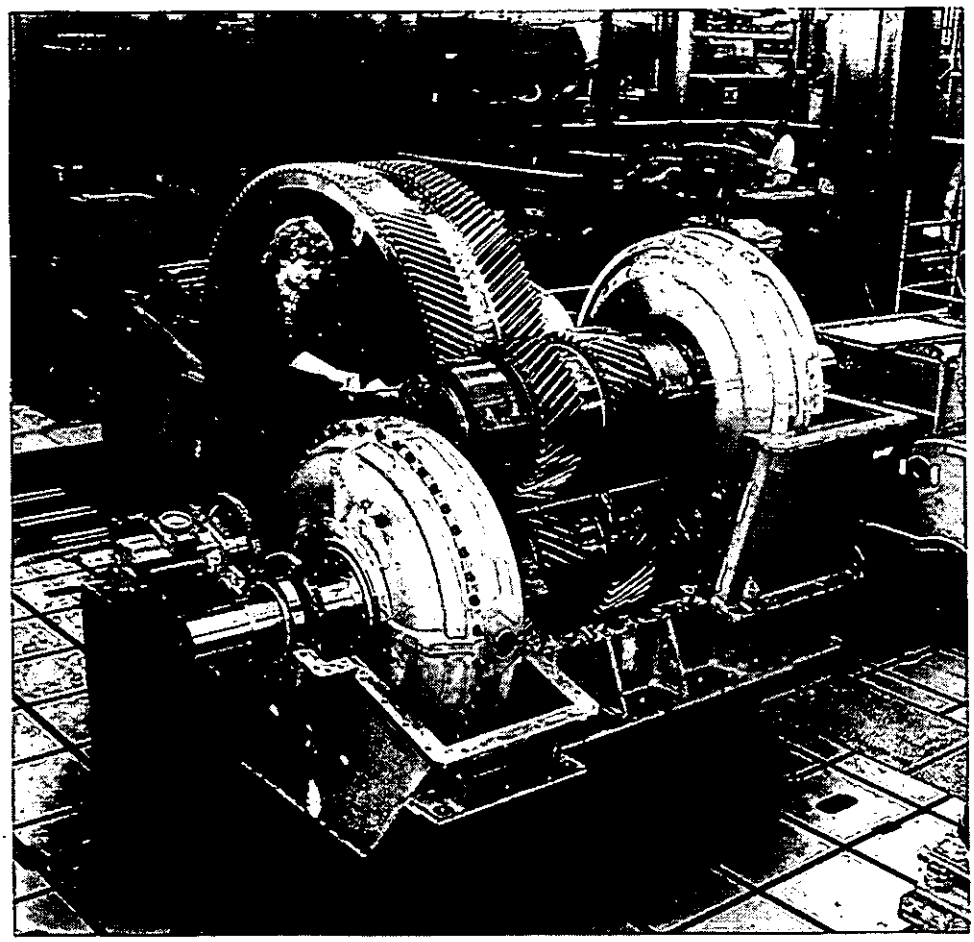
Fig. 17



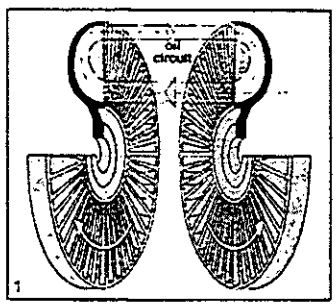


Typical Marine Drive SSS Clutch.

Fig. 18



The Principles of Hydrodynamics



1

A hydrodynamic coupling consists of a centrifugal pump (pump wheel) on the drive shaft opposite and adjacent to a turbine (turbine wheel) on the driven shaft, both in a closed housing. Energy is transmitted from the prime mover to the driven machine by the working fluid, generally oil, with a minimum of transmission losses. Fig. 1 illustrates the Foettinger principle.

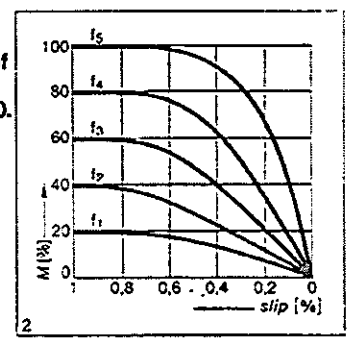
When energy is transmitted in this manner, the law of hydromechanics applies. The sum of the moments within the »fluid coupling« system must equal 0.

$$\Sigma M = 0$$

It follows that the output torque must always equal the input torque applied to the coupling.

In theory, a hydrodynamic coupling with a constant filling transmits higher torque with increasing slip. The increase is, however, not linear but in the form shown in Fig. 2.

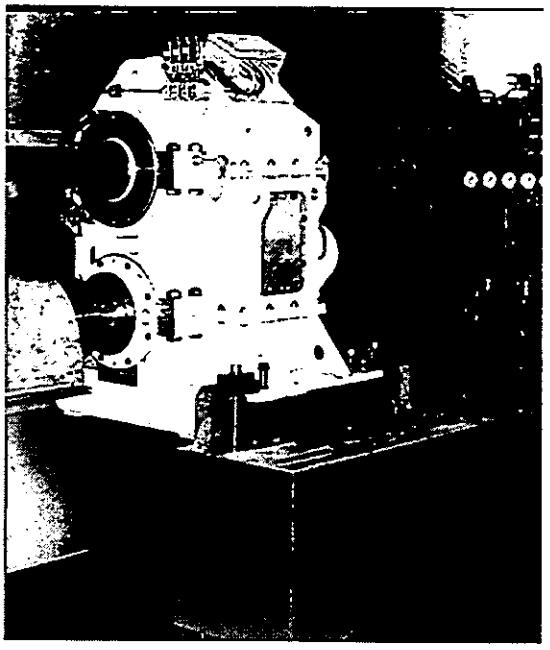
If the amount of fluid in the coupling is reduced, the flow of fluid from the pump wheel to the turbine wheel is also reduced so that the transmitted torque decreases. In Fig. 2, this characteristic is illustrated by the torque curves with the parameters f_1 to f_5 with f_5 representing the maximum possible oil filling.



2

The torque characteristics illustrated in Fig. 2 can be altered to suit the coupling application of which there are a number of varying technical possibilities.

Fig. 19



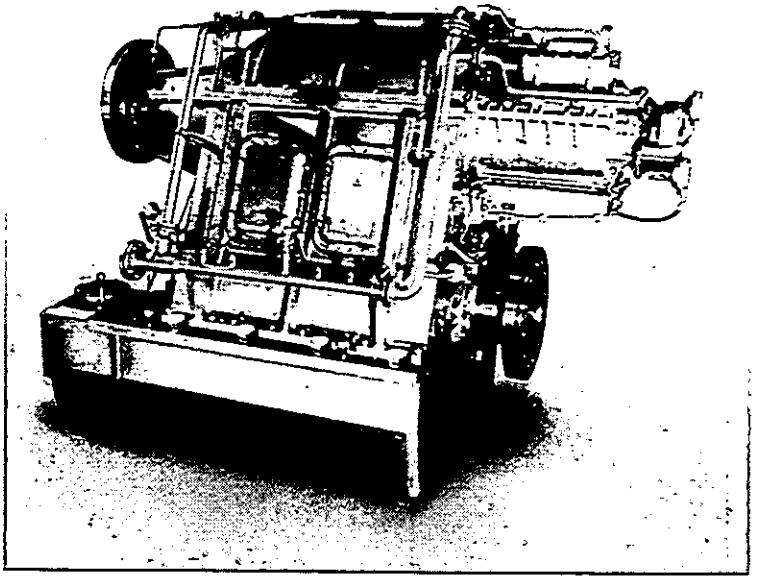
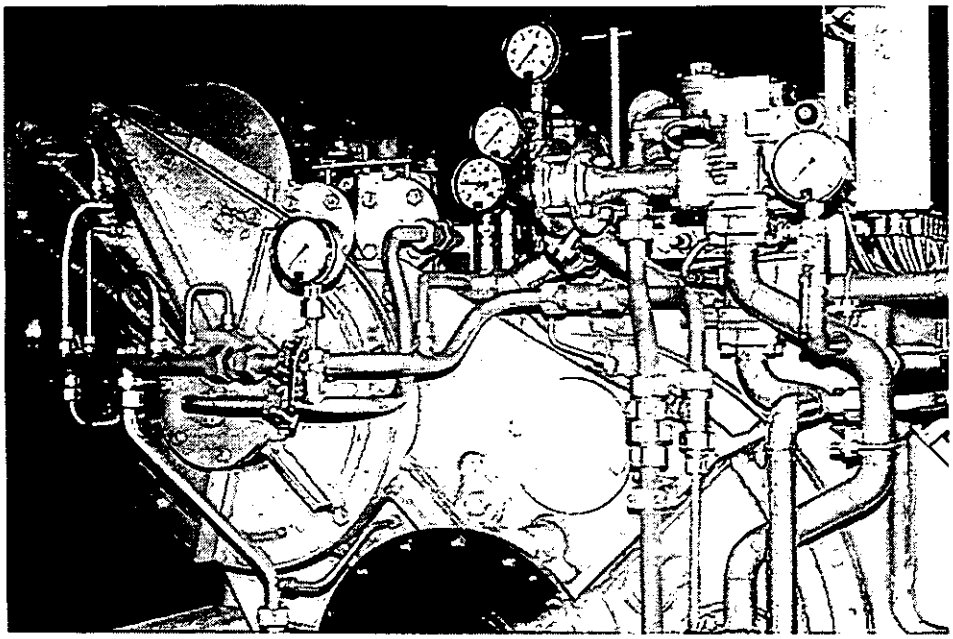


Fig. 20



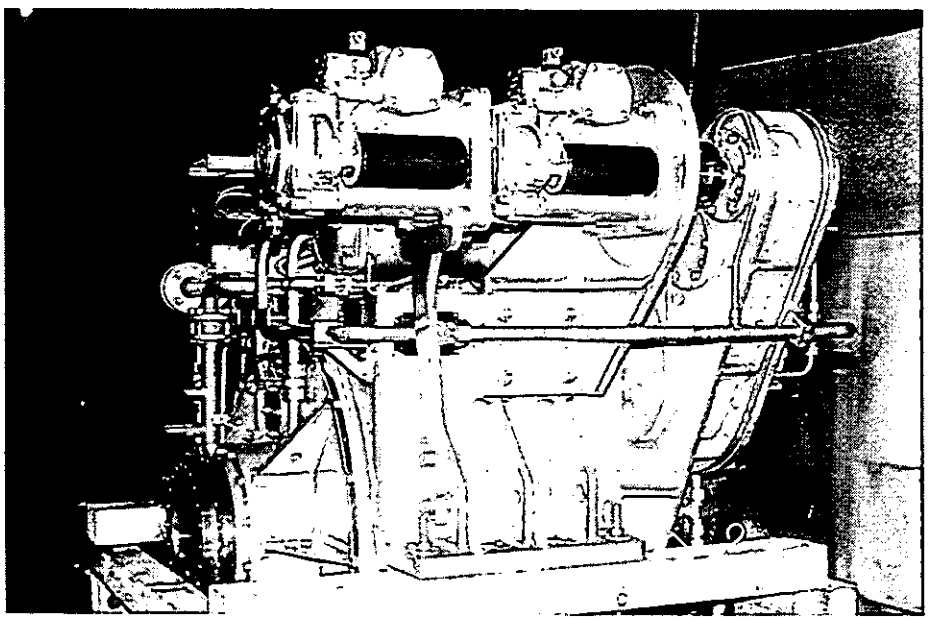


Fig. 21

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

ZF MARINE TRANSMISSION TECHNOLOGY

George E. Marsland, Franco M. Bennati

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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ZF MARINE TRANSMISSION TECHNOLOGY

George E. Marsland, Franco M. Bennati
ZF MARINE SW-V
Friedrichshafen (Germany)

1. INTRODUCTION

In common with most manufactured engineering products, a marine transmission must be supplied not only to meet the requirements of a particular specification, but also to perform reliably throughout its operational life.

Traditional marine transmissions have been developed for the straightforward need to:

- Transmit the drive from the prime mover to the vessel propulsion unit.
- Introduce a gear ratio between the input and output drive shafts to achieve correct matching of engine and propulsion unit characteristics.
- Offer a means of engaging and disengaging the output from the input drive.
- Offer a means of reversing the transmission output drive rotation.

Although there are a number of well established marine transmissions which have been designed to meet the above needs currently available from the various manufacturers and which appear to have many similarities, factors such as operational reliability, engine matching characteristics, overall transmission dimensions, operational weights, sales and service activities, responses to changing market requirements etc. are more connected with the design and operating philosophies of the individual transmission suppliers than with the actual solutions.

Any discussion regarding the applied technology and the particular features and special developments for ZF Marine transmissions for fast ferries can only therefore begin with an overview of the ZF organisation and the standards for quality and reliability which have been created to apply to all ZF manufactured products.

2. ZF FRIEDRICHSHAFEN AG

2.1 The Company Background:

The Company ZF Friedrichshafen AG is a Foundation for which the only shareholder is the Zeppelin Foundation Friedrichshafen.

As indicated in Figure 1., the company has 5 divisions for:

- Commercial and Special Vehicle Transmissions.
- Car Transmissions.
- Driveline Systems and Axles for Off-road Vehicles.
- Steering Systems.
- Chassis and Suspension Components

Additional subsidiary companies deal with Marine and Aviation products.

The organisation has 22 Sales and Service Corporations and representative offices which operate together with approximately 500 world wide trading partners and customer service centres.

2.2 Company Plants and Employees:

Figure 2 Indicates the locations of the 36 ZF Production Plants and shows the distribution of the 32,900 employees.
The importance of Company Research and Development and Training are also indicated by the numbers of personnel employed in these sectors.

2.3 ZF Companies:

The ZF organisation includes 23 wholly owned Sales and Service companies which operate together with the Affiliated Companies listed in Figure 3.
World wide coverage is completed with a number of Representative Offices.

2.4 ZF Product Range:

The complete product range is divided into 18 segments as illustrated in Figure 4.

2.5 ZF Marine Division:

The ZF Marine Division has a total of 5 manufacturing plants located in Italy Germany, France and Brazil.

2.6 ZF Marine transmissions - Quality and Reliability:

In the introduction it was noted that Marine Transmissions could be expected to perform reliably throughout service life. Given correct design, the reliability of the product becomes a function of quality and all that this means.

The company overview shows that ZF is a leading internationally active supplier of a wide range of transmissions, systems and components including:

- Commercial vehicle and special transmissions
- Car transmissions
- Off-road transmissions
- Marine transmissions
- Aircraft transmissions
- Railway transmissions
- Axles, steering systems and suspension products
- Industrial drives

2.7 The ZF attitude to Quality and Reliability:

The nature of the industry into which ZF supplies its products automatically demands the highest possible level of product Quality and Reliability.

The large number of transmissions and components manufactured by ZF makes extensive use of mass production techniques which means that a relatively minor component or transmission fault can affect a complete batch of many units.

A product recall action or a failure to meet a delivery schedule due to unsatisfactory design, faulty material or manufacturing error would be unacceptable in the industry and could result in both financial loss and severe damage to reputation.

The ZF Company Standards for Quality and Reliability are therefore laid down to ensure that all ZF products meet the stated conditions for sub-suppliers to the automotive and aircraft industries and these Standards apply to all ZF activities from initial concept and design, through manufacture, inspection and testing, to packing and dispatching the finished product.
Maintenance and adherence to the Standards for Quality and Reliability internally is considered to be essential in order to preserve the reputation of ZF and its products.

In 1991, ZF introduced a fundamental re-orientation process to apply to all divisions of ZF, referred to as the 'Total Quality Management ' concept.

TQM started from the premise that only highly motivated and qualified employees are able to take an active role in the necessary changes the company has had to face.
With long term customer satisfaction as the final objective, TQM defines processes and duties to be implemented in the individual business sectors.

2.8 ZF Company Research and Development Centre (FEZ), (Figure 5):

The core of ZF technology is the Corporate Research and Development Centre located in Friedrichshafen in Germany. The results of much of the activity of this organisation, particularly with regard to material technology, calculation procedures and manufacturing techniques, continually serve as the basis for overall Company Standards for Quality and Reliability.

The ZF Company employs a total of almost 2000 personnel, including over 500 only for the FEZ specialist research staff. Its services are available to all ZF product divisions.

Research and Development Centre activities and services include:

- Concept considerations and special prototype construction
- Product improvement
- Material research and inspection
- Electronics
- Component and transmission system research
- Transmission component calculation techniques and procedures for gear design
- Calculated procedure verification with practical testing
- Causes of, effects of and techniques to reduce noise
- Lubrication research
- Component and transmission load, noise and vibration testing
- System simulation techniques
- Patents

2.9 ZF Marine Product Design at Division Level:

Each of the marine transmission manufacturing plants has its own dedicated Product Design Department where local specialist product design and application know-how is resident.

A uniform standard of Quality and Reliability is however maintained as all design departments operate under the same TQM organisation and work with same ZF Company design standards and procedures.

The standard CAD software program is now Pro Engineering and a number of ZF internal software programs for dealing with calculations for gear tooth geometry, shaft stress and deflection, shrink fit connections, bearing life, clutch plate geometry,

classification acceptance etc. etc. may be called up locally in the design departments on networked PC's.
ZF internal software programs are supported by the Research and Development Centre which also carries out all final gear tooth geometry calculations to optimise for correct running and minimum noise. The FEZ is in fact the brain of the ZF group. The knowledge and the experience of the different divisions are used to help each designer in his work. In this way ZF Marine can use the knowledge of other divisions to develop new products (Figure 6).

Comprehensive Internal ZF Standards to be applied in design deal with:

- Materials and heat treatment
- Detail design and drawing information
- Tolerance information
- Calculation procedures
- Production techniques
- Machine elements and components
- Inspection and testing procedures
- Documentation

2.10 Manufacture:

The Production techniques and methods necessary for successful operation in the series and mass production environment associated with the automotive industry strongly influence product design, production planning, logistics and plant organisation.

Component interchangeability is essential and transmissions are designed for easy, rapid assembly so that traditional fitting work is avoided. Assembly techniques make wide use of special jigs and fixtures which are also made available to the world wide service organisations.

Component manufacture is achieved using the most modern automated machining processes and procedures, and an ongoing program for plant replacement ensures that production facilities are always in line with up to date technology.

Component parts for the ZF Marine transmissions are generally manufactured in the ZF plants and therefore make full use of the methods and procedures available for other ZF products with the same attention to Quality and Reliability.
ZF Marine has its own dedicated purchasing, logistic/stock control and assembly departments to ensure that a tight control over the supply of parts for assembly, spares and service is maintained and that deliveries are completed as required.

The ZF Standards for Quality and Reliability apply also to external sub-suppliers and incoming goods and materials are systematically inspected for conformity before acceptance into the production system.

2.11 Testing Facilities:

ZF Marine has a number of test beds, the largest of which can be used for load testing at ratings of up to 10 000 HP, (Figure 7).

Transmission testing falls into 2 distinct categories.

New product and prototype transmission testing are handled by the ZF Marine Research Department and products are subjected to thorough investigation regarding performance, noise, vibration, temperature effects, gear tooth contact area assessment etc.
The ZF Marine Research Department works in close co-operation with the main ZF Research and Development Centre and result are documented for use by the Design and Application Engineering Departments.

Series production transmission units all pass through an inspection procedure which includes a partial load and function test on a production test bed.

No marine transmission leaves the factory before satisfactory completion of the test procedure.

2.12 Classification:

ZF Marine works very closely with all classification authorities and material and performance documentation for all transmissions is kept on record.
Manufacturers Survey Arrangements (MSA) e.g. with DnV have been agreed.

2.13 Service organisation facilities:

ZF Marine has its own Service Department which has instant access to stocks of parts and which can also influence logistics and stock control.
The Service Department works in very close co-operation with ZF branch offices and other service centres world wide and the number of personnel is adequate to enable fast response to customer needs anywhere in the world.

The Service Department is also responsible for ZF Marine transmission service and spares documentation which again conforms to the standards set for the automotive industry.

Training programs are available for ZF Marine service centre staff and customer personnel.

Service, overhaul and repair work on ZF Marine transmissions is generally easily and quickly completed as the nature of component production guarantees complete interchangeability of parts and the method of assembly using jigs and fixtures avoids the need fitting and adjustment.

2.14 Sales and Application Engineering Departments:

ZF Marine sales support is available from the 23 wholly owned ZF sales and service subsidiary companies and the approximately 500 independent trading partners situated at locations all over the world.

Central Sales Departments are located at the various ZF Marine plants to provide support both to the external sales offices and also direct to the customers as and when may be necessary.

Lines of communication between customers and ZF Marine Technical, Logistic and Service Departments are therefore well established via the external and internal Sales Network.

A vast amount of application experience is documented in the ZF Marine Application Engineering Departments and this serves to provide a source of information for response to any customer questions on particular propulsion system installations.

3. ZF MARINE TECHNOLOGY

The transmission shown in figure 8 is a typical example of standard series produced unit and the design is based on the use of three shafts, 5 gears and two clutches. Other components parts include an oil supply pump for lubrication and clutch pressure, a clutch control unit and a system monitoring arrangement.

The shaft, gear and bearing layout is shown in Figure 9., and this arrangement may be used to consider the technology employed to achieve the desired result.

3.1 Transmission Housing Design:

A typical transmission housing as illustrated in a standard spare parts list is shown in Figure 10.
Weight is an important factor for all Fast Craft transmissions and for this reason transmission housings and associated components for these applications are of a sea water resistant aluminium alloy. General housing design is largely a matter of experience where suitability for recognised manufacturing and machining processes is concerned but performance of the housing structure and connected ribbing under load becomes the subject of finite element assessment techniques. Structural stresses and deflections are examined to assess possible housing distortions in order to determine possible effects on e.g., gear tooth contact, (Figure11).

3.2 Shaft design, (figure 12):

Stresses occurring in rotating shafts subjected to bending are cyclic and it is therefore essential that stresses in all such components should be below the fatigue endurance limits.

Shaft diameter changes, radial oil holes, screw threads etc. Are all elements which raise stresses and must therefore be identified and taken into account in calculations.

The procedure to correctly dimension rotating shafts is therefore to consider the shaft arrangement as a whole, noting shaft positions relative to each other, shaft geometry and dimensions, positions of supporting bearings, positions of gears, effects of shrink fitted components and calculating resulting stresses in shaft material with load applied to the system.

Choice of materials and accurate calculation procedures are important as components must be designed for minimum weight.

Shaft centre distances are functions of gear diameters, gear widths and bearing layouts but are selected again for minimum unit weight.

ZF Marine employs computer techniques with in-house software programs for shaft stress calculation and to obtain optimum layout solutions.

3.3 Shaft bearings, (Figure 13):

Bearings in any application are components which have a limited service life and in must therefore be renewed before this is reached. The actual bearing layout is largely a matter of experience but for bearing calculation purposes ZF Marine uses the SKF Mamut program to select and dimension bearings.

Bearing life is related to engine service conditions and bearings are chosen to ensure that replacement will not be necessary before engine overhaul becomes due.

3.4 Shrink fit connections, (Figure 14):

Connections between gears and flanges are made using tapered shafts and bores. The technique is to slide the hub onto the shaft until the tapered connection is made with only a slight axial force. Fluid is then injected under pressure to separate the taper surfaces and reduce friction to a very low level at the same time as the hub is forced axially along the shaft. The axial distance travelled along the taper determines the fit and hence the torque carrying capacity of the joint.

Joint dimensions and preloads and torque capacities are calculated using the computer and results ensure that material stresses are within those allowable for the materials.

The procedure lends itself to series production methods and results are more predictable than those obtained using e.g. parallel shafts and keys.

3.5 Clutches, (Figure 15):

The clutches used for drive engagement and reversal are of a multi-disc design using hydraulic pressure on the disc pack for engagement.

The torque capacity of a particular clutch is generally determined by the number of discs and the applied hydraulic pressure to the clutch.

During engagement however the clutch will be slipping with the hydraulic pressure applied and this results in heat generation at the disc surfaces. Discs are alternately steel and steel with a sintered lining and oil grooves are provided to allow a flow of cooling oil through the disc pack.

Clutch disc design depends much on experience and testing but established computer programs are used to relate torque and heat dissipation capacities to disc dimensions, numbers of discs required and quantities of cooling oil to be supplied.

Particularly important are the disc drive connections and those used in ZF Marine designs ensure that resulting stresses in the materials are always below the fatigue endurance limits.

Theoretical results obtained by calculation are often verified using appropriate test facilities, (Figure 16).

3.6 Gears:

ZF gear technology has been the subject of development over many years and a vast amount of knowledge and experience has been collected in this time.

The first requirement for a geared transmission is that it should be fit for purpose in terms of performance and reliability. With particular reference to the Automotive and Aviation Industries however, there are also e.g. restriction on weight and noise which have to be taken into account.
Much of the on-going Research and Development work in ZF has been to produce reliable gear calculation procedures which can be applied with confidence to new transmissions of all kinds and these procedures are now used for the Marine transmissions.
Further work has been in connection with noise and how this is affected by gear tooth geometry.

Gears can now be reliably designed for minimum weight and dimensions with a guaranteed performance using computer techniques and full load housing and shaft distortions can be compensated for with specially corrected gear tooth geometry.

This has been made possible with years of test bed work to verify theoretical considerations and all calculation procedures are base on tested and proven results.

Figure 17 shows a graphic illustration of a cad approach to gear tooth design.

Figure 18 is the output from a program to study the actions of gears under load.

Figure 19 indicates how tooth deflection and tooth root stress can be predicted.

Figure 20 shows the contact stress pattern for a pair of gears in a transmission obtained by calculation.

Examples of verification of predicted results by testing are:

Figure 21 showing a gear with strain gauges for both static and dynamic stress measurement.

Figure 22 is a photograph of a truck transmission set up on a test rig in a noise laboratory for noise investigation.

Figure 23 showing a 3 dimensional representation of transmission noise level in terms of frequency and order.

3.7 Clutch Control Arrangement:

The clutch control unit is mounted on top of the transmission unit and can be actuated either mechanically or electrically. Within the control unit is a hydraulic valve arrangement which provides a ramp function for the clutch hydraulic pressure to avoid 'hard' clutch engagement and also controls clutch slipping time to avoid overheating.

3.8 Autotroll Clutch Control, (Figure 24):

The traditional propulsion system solution for many fast craft is to install one or more drive lines, each consisting of a main engine coupled to a conventional reverse/reduction transmission unit and a fixed pitch propeller.

This arrangement can be found in a large number of existing vessels and in spite of the relatively recent growth in interest in waterjet and controllable pitch propeller (CPP) propulsion systems, will continue to be the accepted standard solution for many applications.

Fast craft are by Definition high performance vessels and resulting hull designs are therefore intended to offer lowest possible hull resistance at high speed.

The idling speeds of main engines used in these applications are generally about 30% of the maximum speed so that with the transmission engaged and the engine running at idling speed, the slowest ship speed is often too high for manoeuvring.

To solve this problem, transmissions have been fitted with 'Power Take In' (PTI) drives, vessels have been equipped with small auxiliary propulsion systems or transmissions have been fitted with trolling controls.
PTI drive systems and auxiliary propulsion systems are costly, limited in power and need considerable space for installation.
The trolling control system uses the main engine for the power but enables the propeller speed to be reduced below the normal minimum by allowing the transmission drive clutch to slip.

The ZF Autotroll' patented control system from ZF Marine makes use of the trolling feature but the additional electronic controller also allows the propeller speed to be

accurately set and automatically regulated down to almost zero when moving ahead and also during reversing.

3.8.1 The ZF Autotroll control unit:

The ZF Autotroll control unit dimensions are approximately 340 X 420 X 120 mm and the weight is 30 kg. The unit is mounted on top of the transmissions and is connected into the transmission terminal box.

3.8.2 The clutch selection control unit:

The forward/reverse clutch selection unit is electrically controlled and contains the hydraulic valves for the clutch engagement oil supply.

3.8.3 Trolling valve:

The trolling valve is an electro-hydraulic proportional valve which responds to signals from the ZF Autotroll unit to control the hydraulic pressure applied to engage the clutches.

3.8.4 Speed sensors:

Speed sensors are fitted to monitor transmission input and output speeds.

3.8.5 The Autotroll automatic propeller shaft speed controller:

The shaft speed controller is a closed loop control system which serves to set and hold the shaft speed according to the position of the bridge control lever.

The output signal from the bridge control lever corresponding to a required shaft speed is compared with that from the shaft speed sensor and the difference is then used to vary the clutch pressure to bring the shaft speed into line with the lever setting.
The essential feature of the ZF Autotroll slow speed control system is therefore that the shaft speed always corresponds to the bridge control lever position, whether in ahead or astern.
The design and components are derived from those which apply to controls associated with ZF transmission products from e.g. the Truck and Bus Division, and have therefore the similar levels of quality and reliability as those required for commercial and military applications.

3.8.6 Bridge speed control lever positions, (Figure 25):

Neutral:

Lever Range:	Zero to +15° or −15°
Engine Speed:	Idling
Clutches:	Disengaged

Moving out of neutral through trolling range

Lever Range:	+15° or –15° from zero to control point preset according to engine spec.
Engine speed:	Rising from idling to 45% of maximum engine speed
Clutches:	High clutch slip at start of range with steady reduction in slip to full engagement of clutch at preset control point

Moving out of trolling range through to maximum speed

Lever Range:	Preset control point to end stop
Engine speed:	Rising from 45% of maximum to full maximum
Clutches:	Fully engaged throughout range

3.8.7 Clutch pressure control:

Although the nominal clutch pressure corresponds theoretically to the setting on the bridge speed control lever, the ZF Autotroll control is automatically and continuously varying the clutch pressure to maintain the correct propeller speed.

3.8.8 Trolling duration:

The trolling control and transmission are designed for continuous operation the trolling range.

3.9 Transmission Configurations, (Figure 26):

The basic 3 shaft, 5 gear arrangement referred to above can configured in different ways to provide for installation in as various engine room layouts as required.

3.9.1 Vertical offset parallel input and output shafts:

Using this arrangement the transmission can be flanged on to the engine to form a complete engine/transmission unit but because the transmission shaft are parallel, the engine must be tilted to the propeller shaft angle. The transmission can also be free standing but in either case the engine is positioned forward to the transmission.

3.9.2 Down angle output shaft – A Drive:

Using a specially designed gear tooth form the gear set can be arranged so that the input and reversing shafts remain parallel but the output shaft is angled down at the propeller shaft angle. This design however allows the engine to be installed horizontally. As with the vertical offset arrangement, the transmission

can also be free standing but again the engine is positioned forward to the transmission.

3.9.3 Vertical offset parallel input and output shafts – U Drive:

This configuration is similar to the vertical offset transmission described in 3.9.1 above but in this case both input and output flanges are located on the same side of the unit. This arrangement allows the engine to be installed in the stern of the vessel but the transmission must be free standing and the engine is usually tilted. A cardan shaft is required to make the engine/transmission connection.

3.9.4 Down angle output shaft – V Drive:

This configuration is similar to the Down Angle version described in 3.9.2 above but as in the U-Drive both input and output flanges are located on the same side of the unit. This version allows the engine to be installed horizontally in the stern of the vessel and only a relatively short cardan shaft is required to make the engine/transmission connection.

****

4. FIGURES

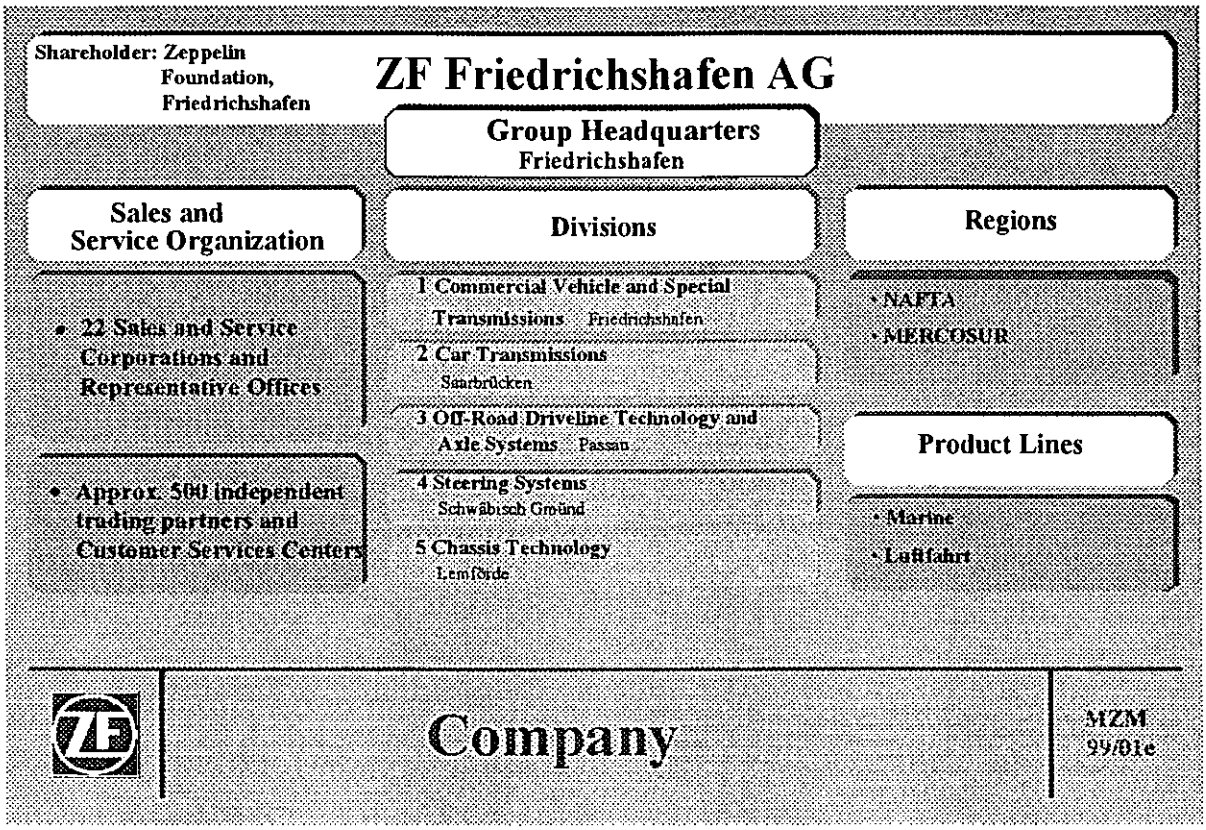


Figure 1

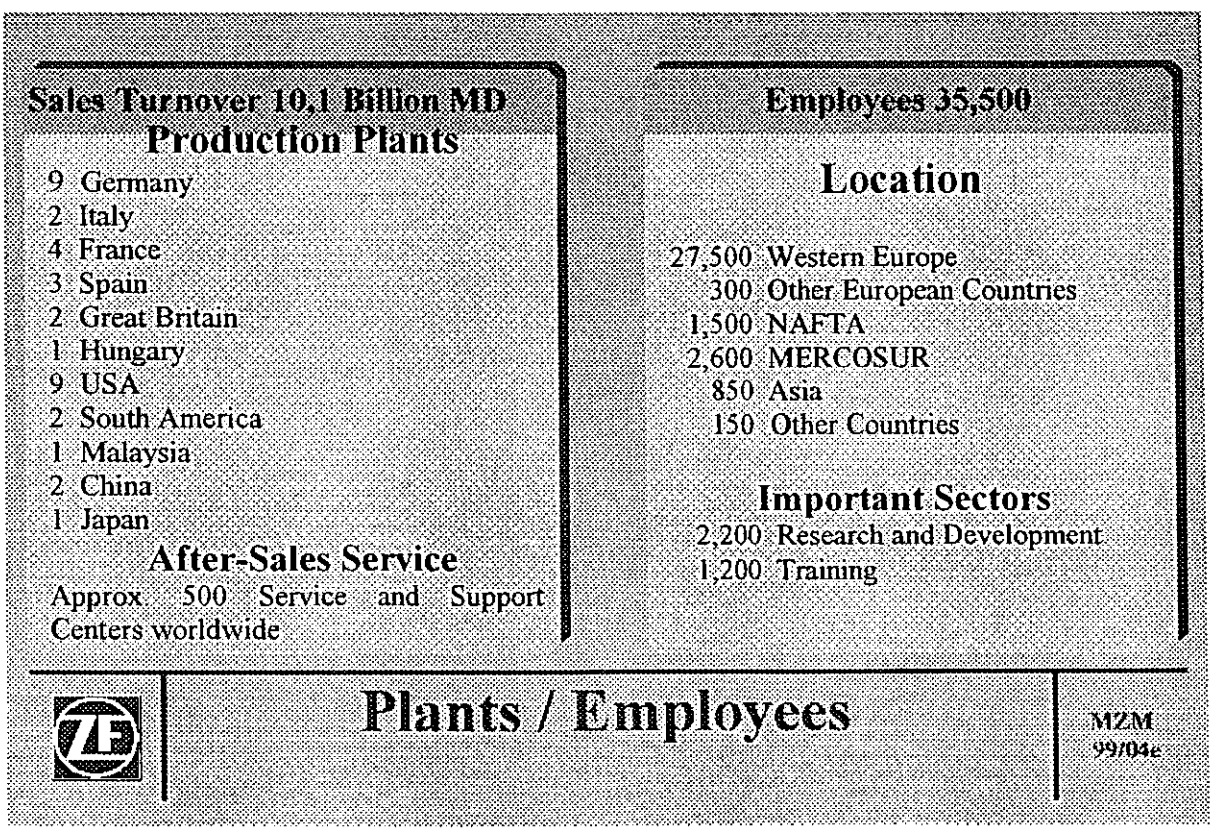


Figure 2

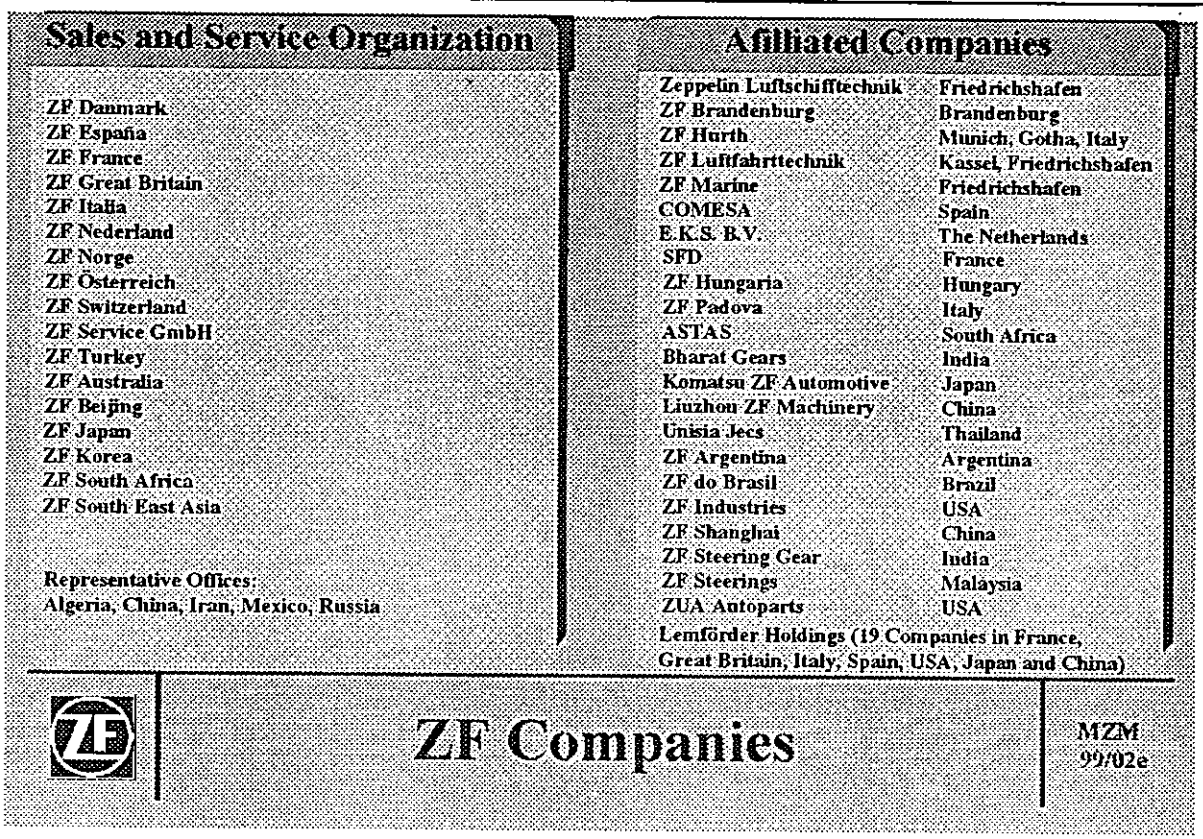


Figure 3

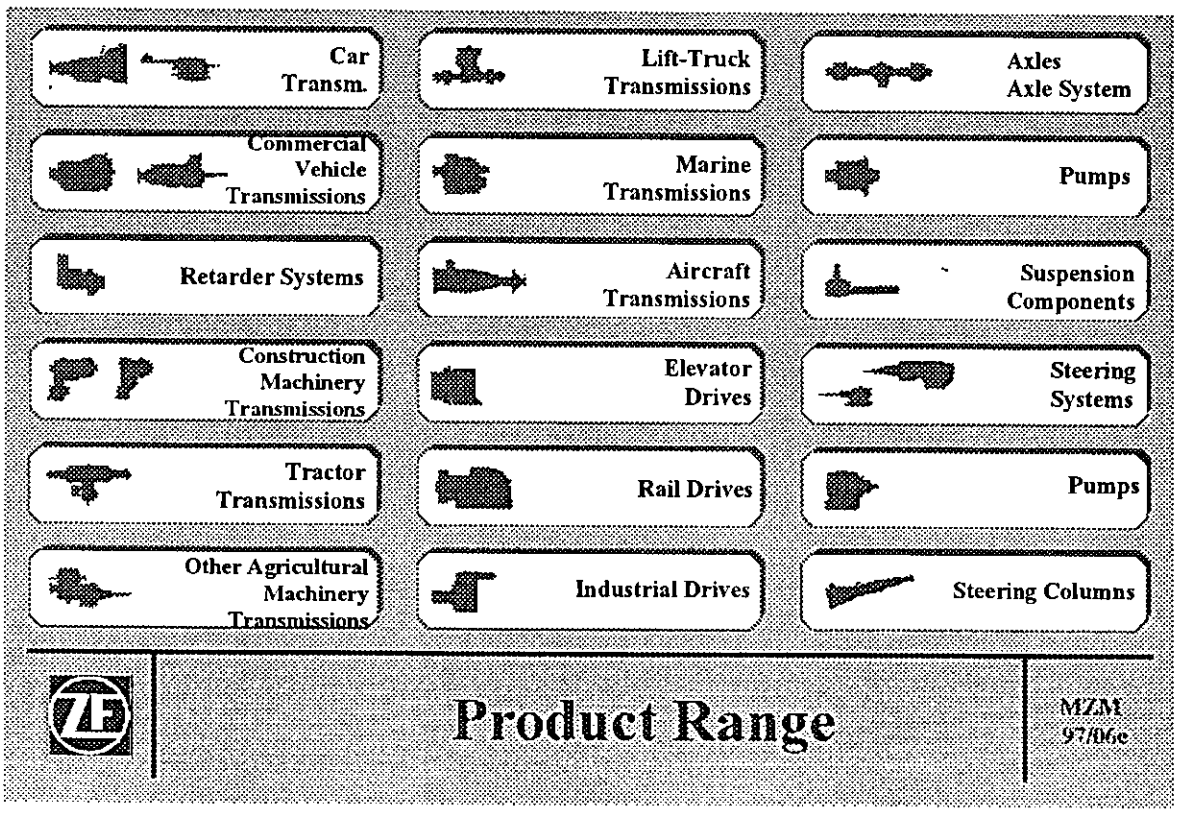


Figure 4

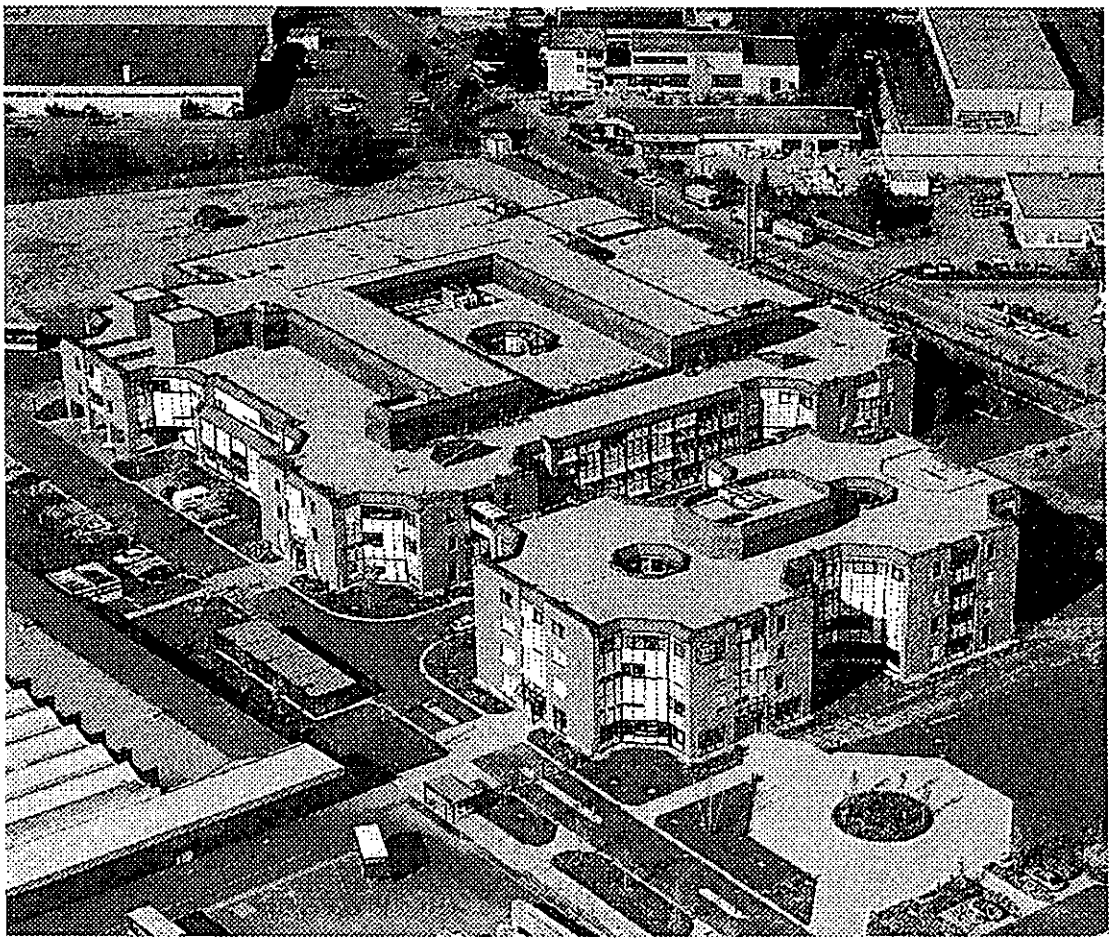


Figure 5

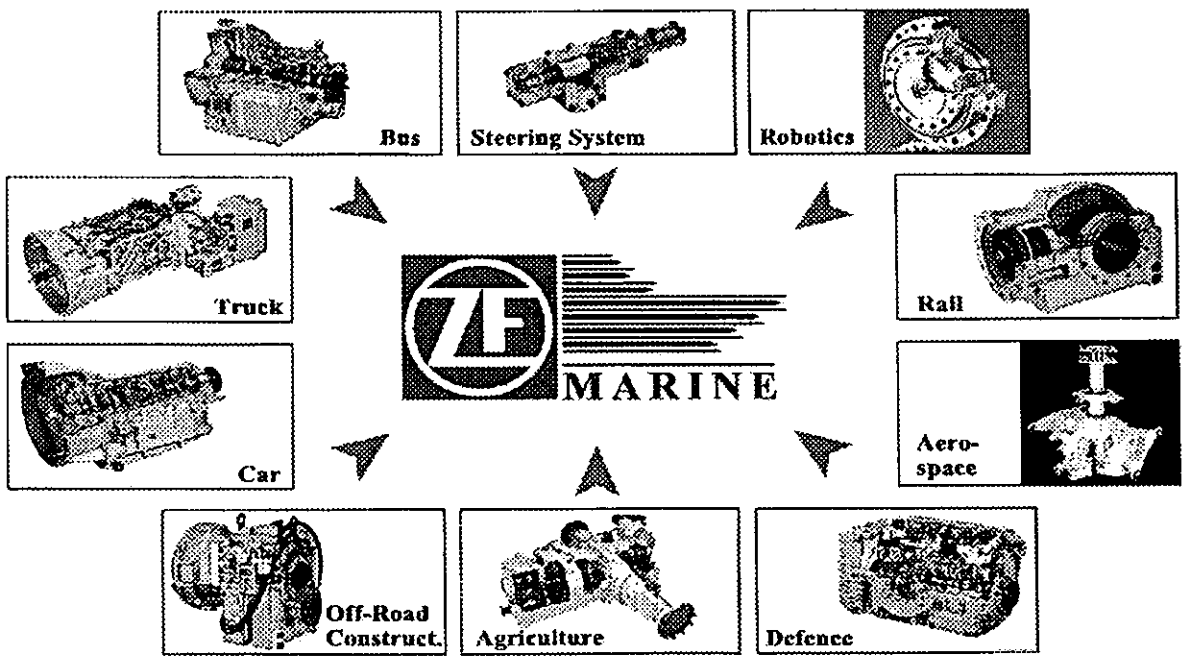


Figure 6

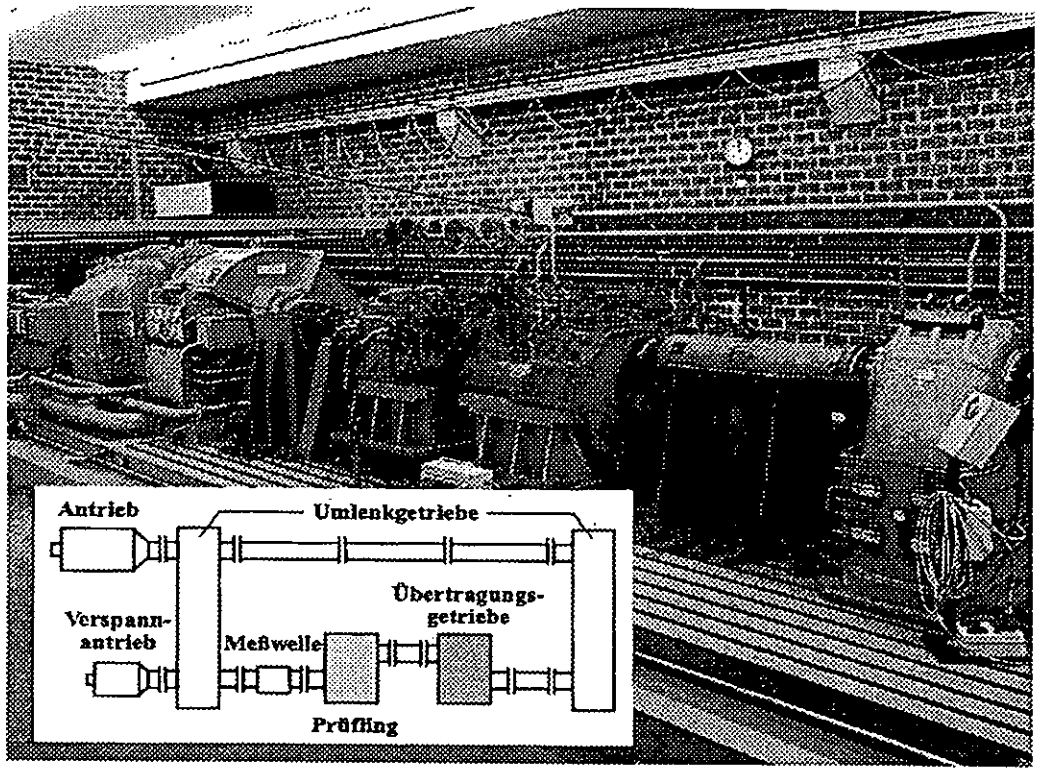


Figure 7

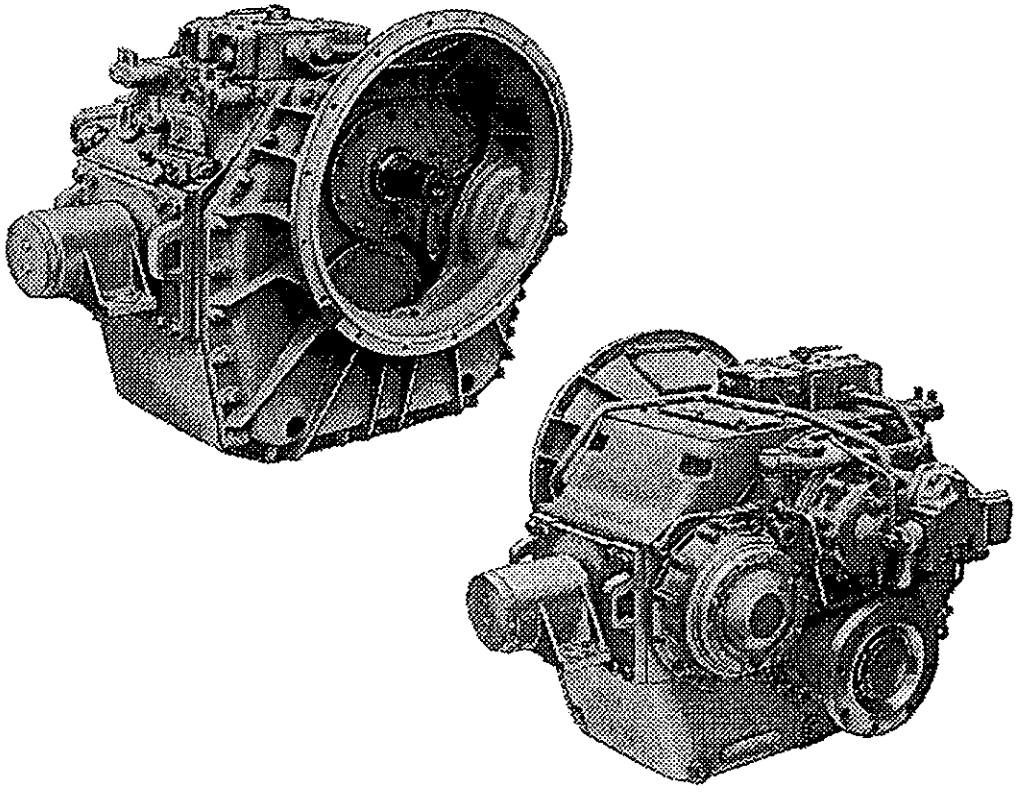


Figure 8

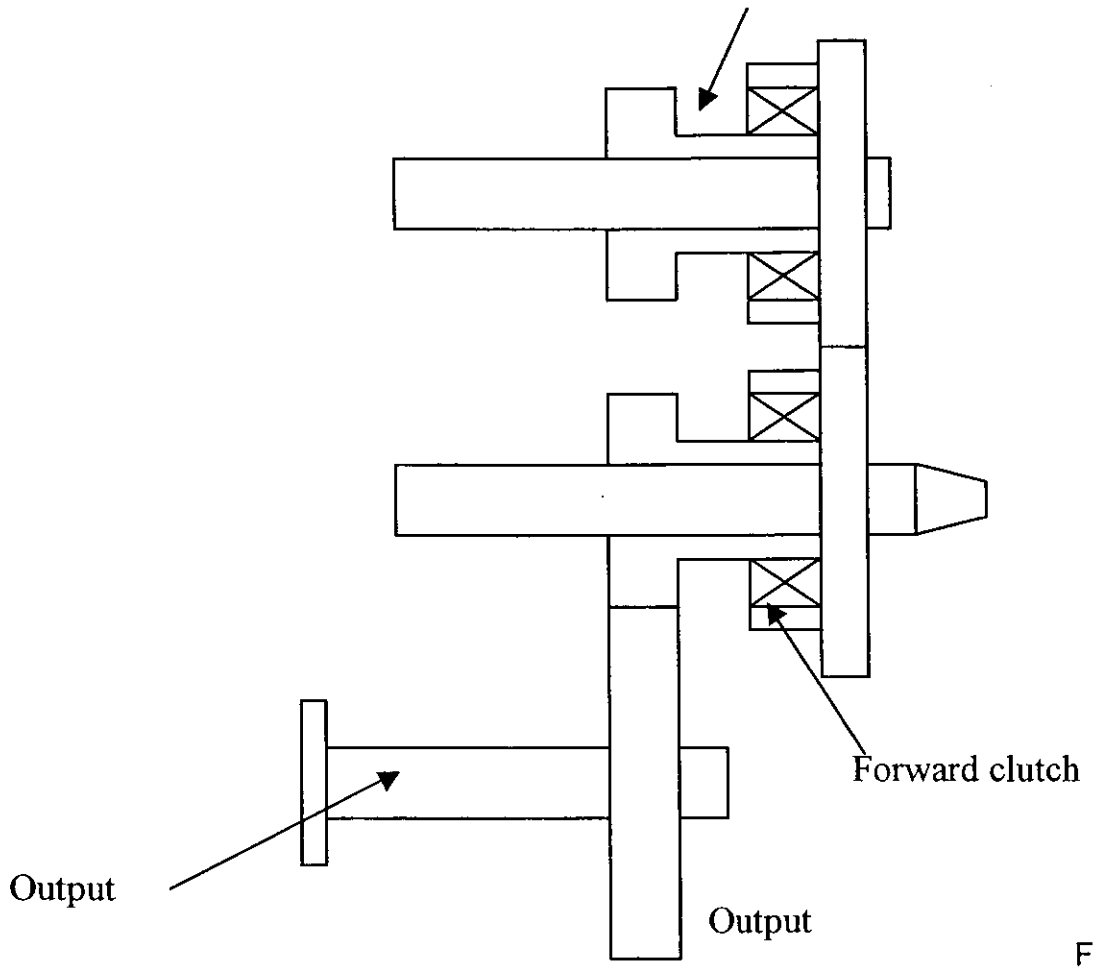


Figure 09

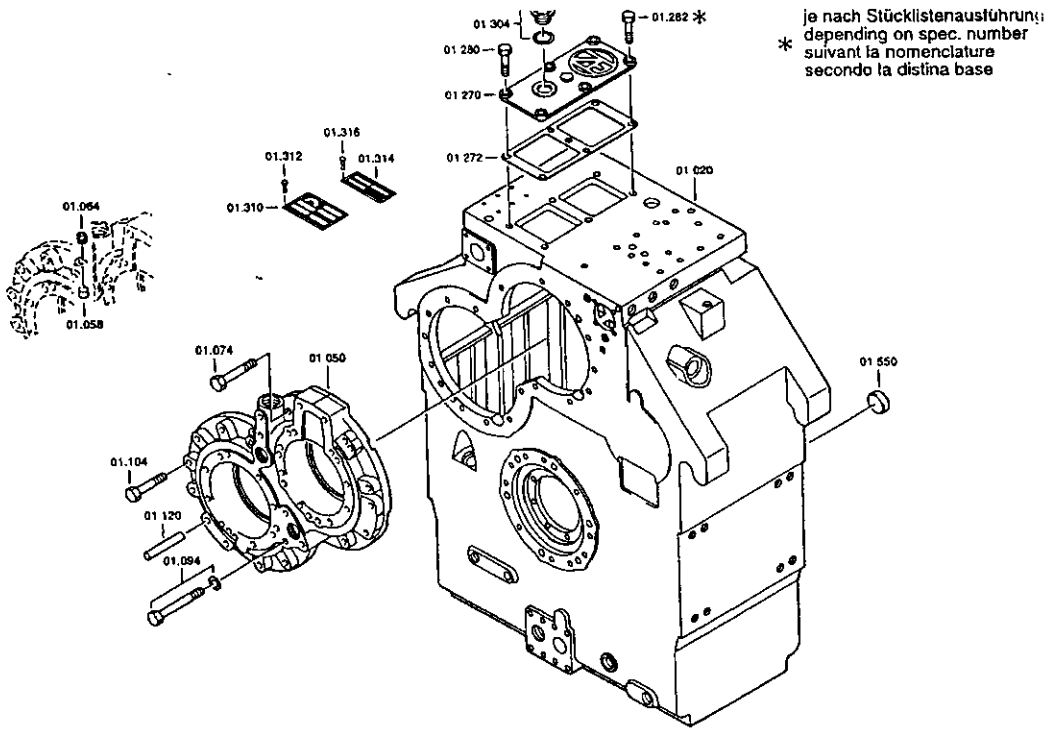


Figure 10

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Figure 11

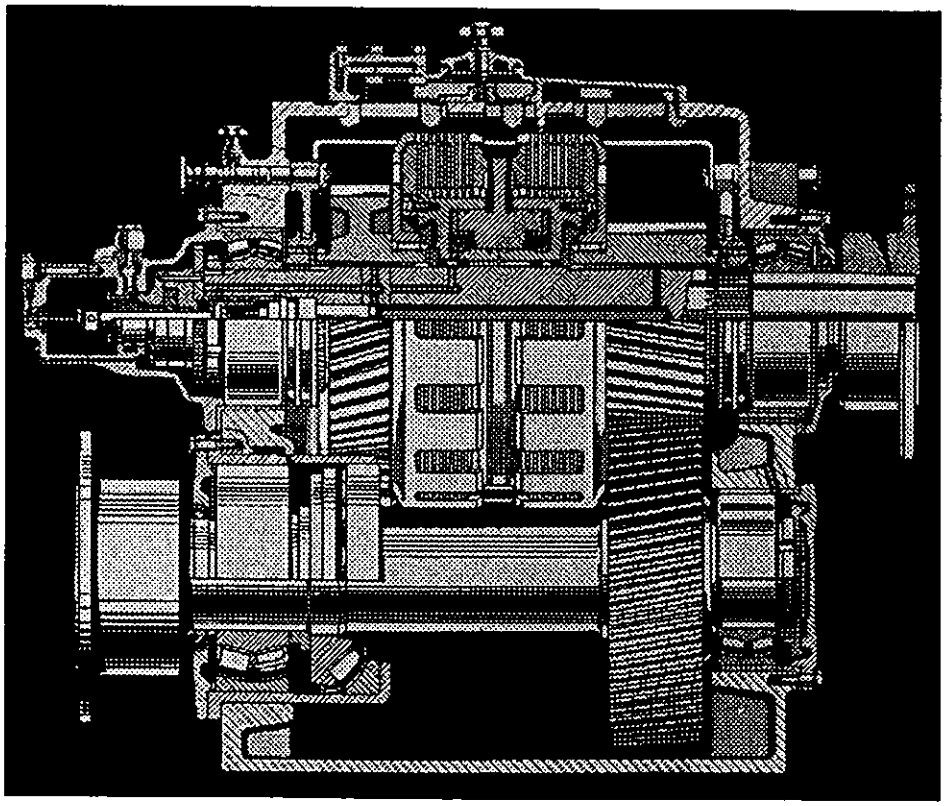


Figure 12

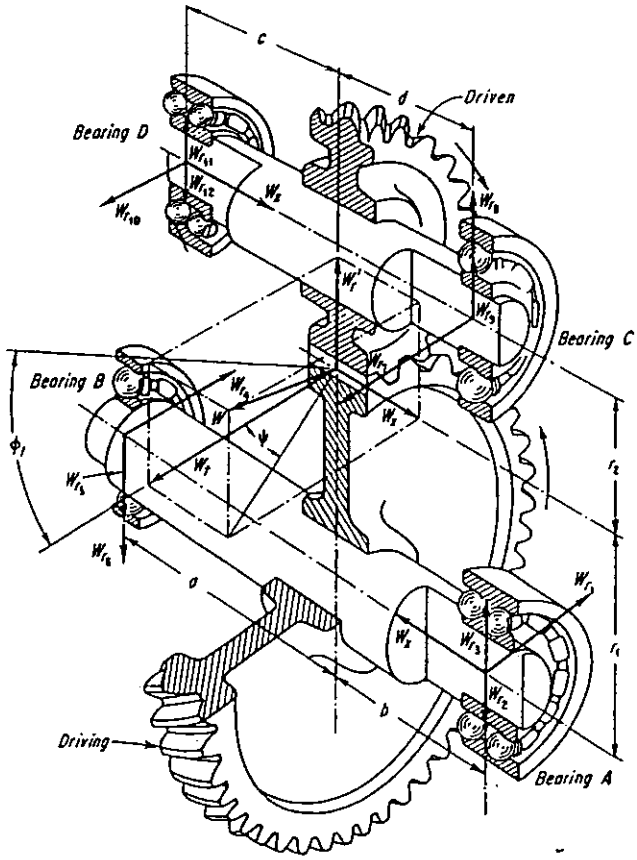


Figure 13

Fit the safety device consisting of parts 1 to 3.
Connect oil injector to flange by means of high pressure pipe and bleed the pipe. Pump pressure oil or glycerine between the mating faces until it is distibuted in the oil press-fit (pressure gauge pressure drops). Continue pumping until the oil press-fit is released suddenly and with considerable force.
Measuring push-on distance.

Note:
The complete new part (replacement part), and not just the cones and oil injection bores, must be thoroughly cleaned of any oil and grease.

The parts to be joined must be at the same temperature, preferabley at room temperature. First properly clean the tapers of the shaft and gear or flange, do not oil or grease them before measurement.
Fit the gear or flange onto the taper of the shaft but only until it is seated.
Check the push-on distance „a" in this position.
On shafts without a stop collar, the axial position of the gear or flange relative to the shaft must be measured to check on the push-on distance „a" exactly.
Remove parts again after measurement.

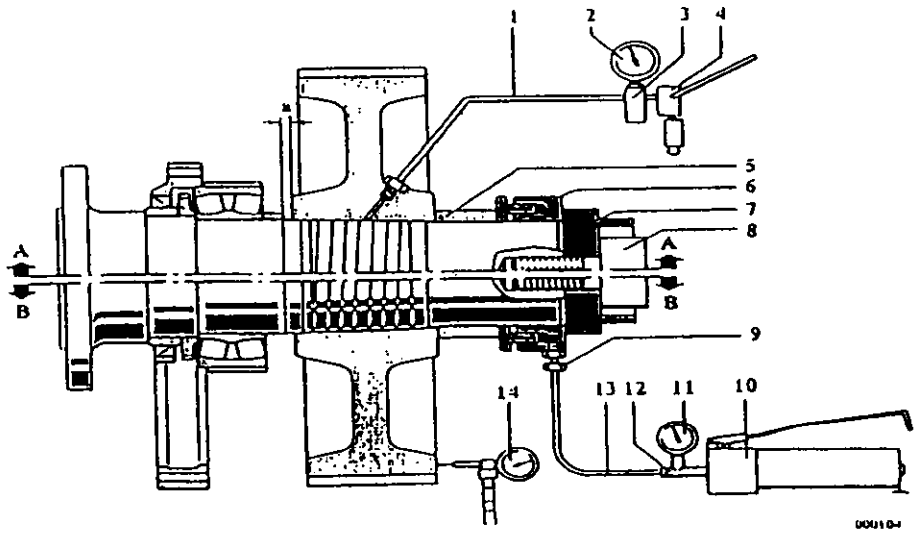


Figure 14

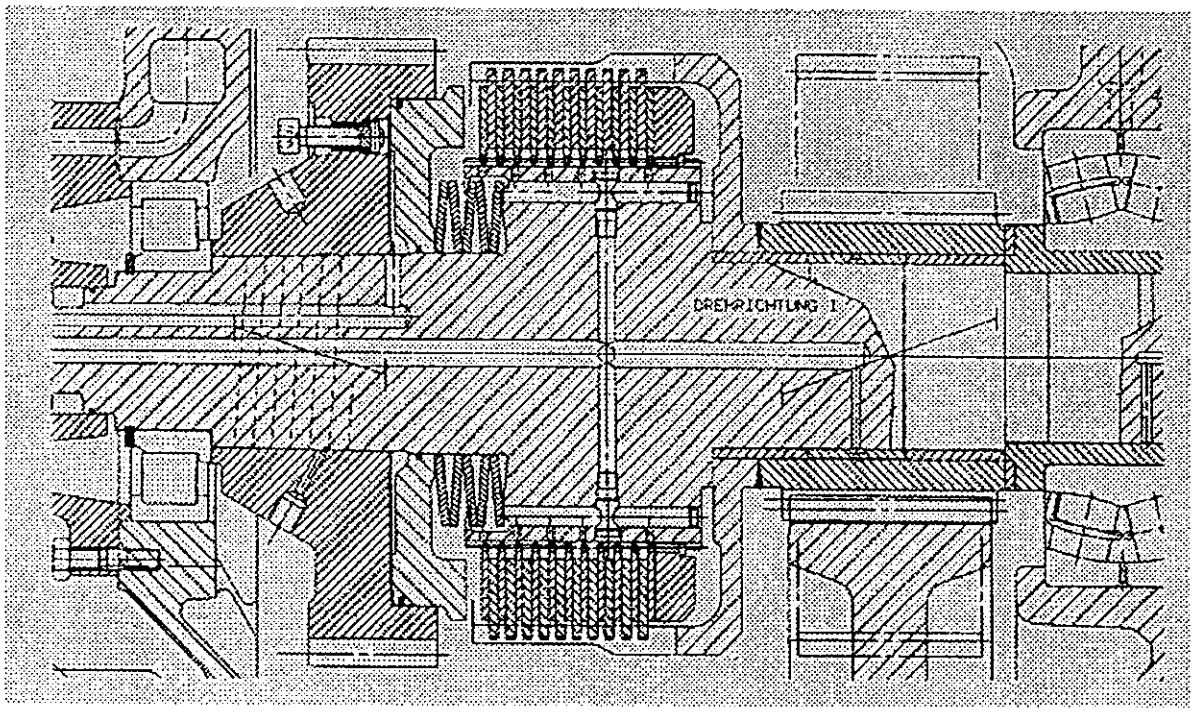


Figure 15

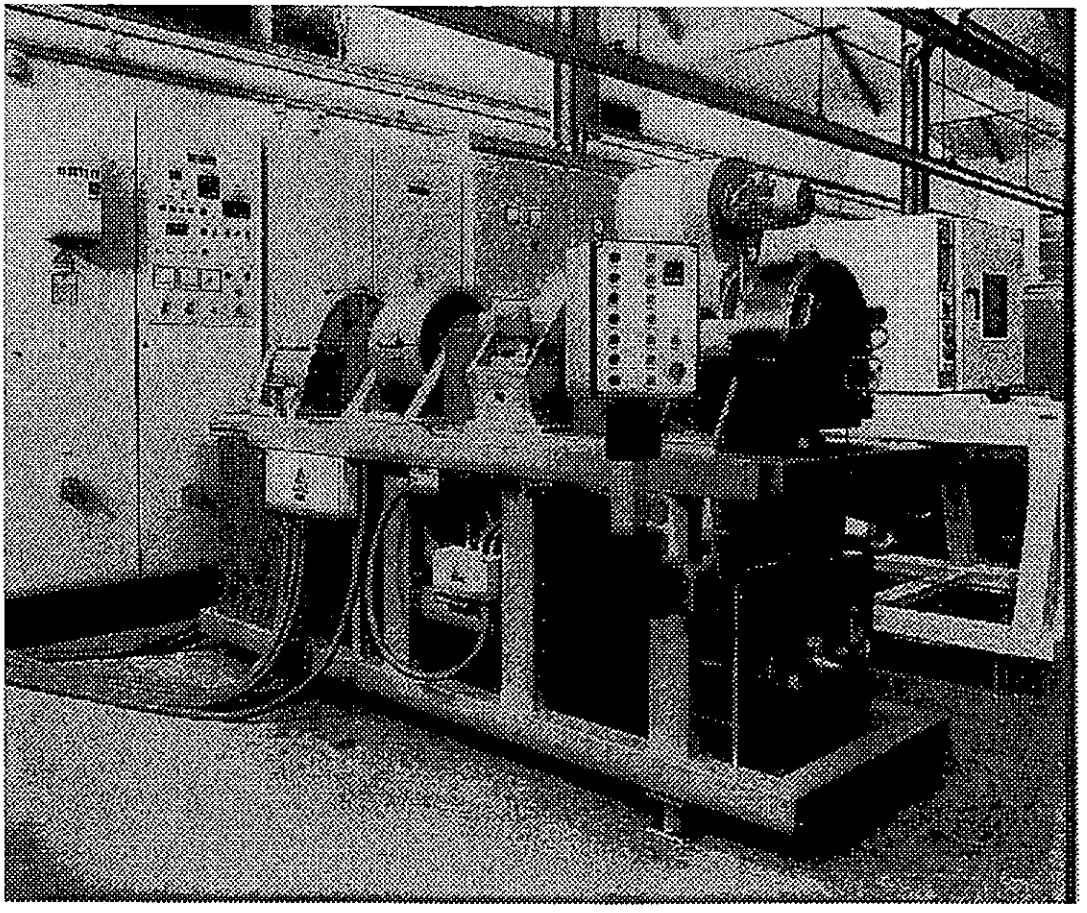


Figure 16

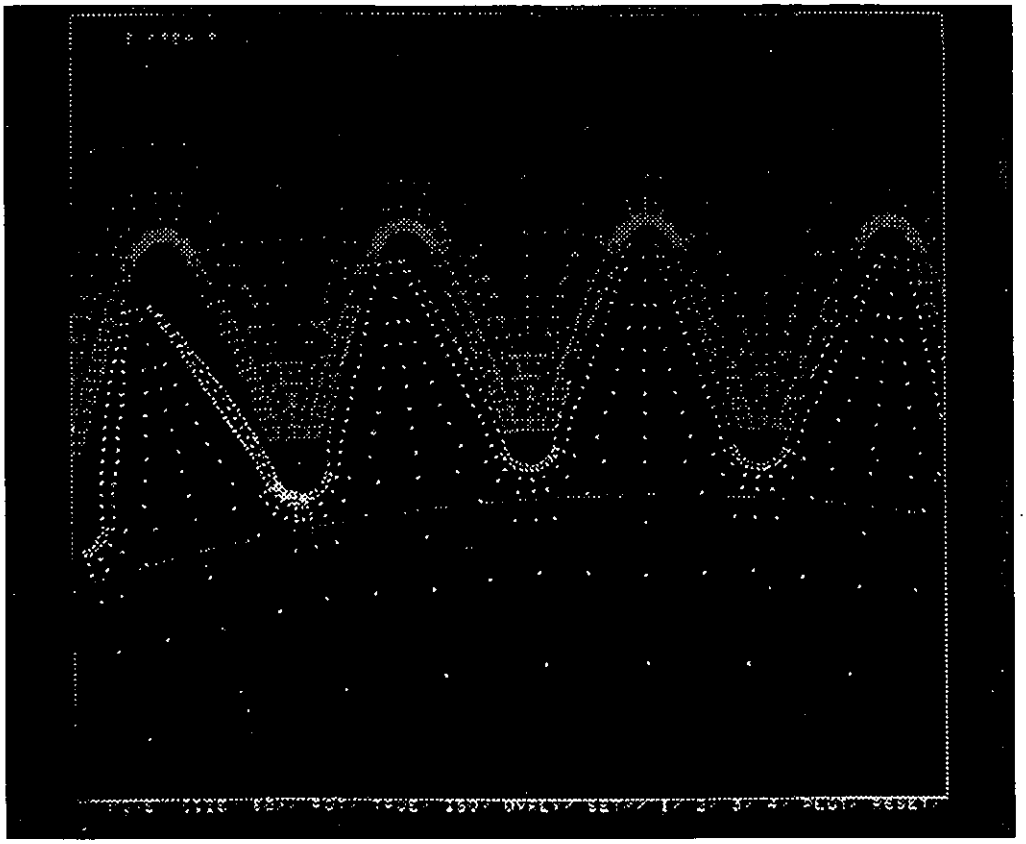


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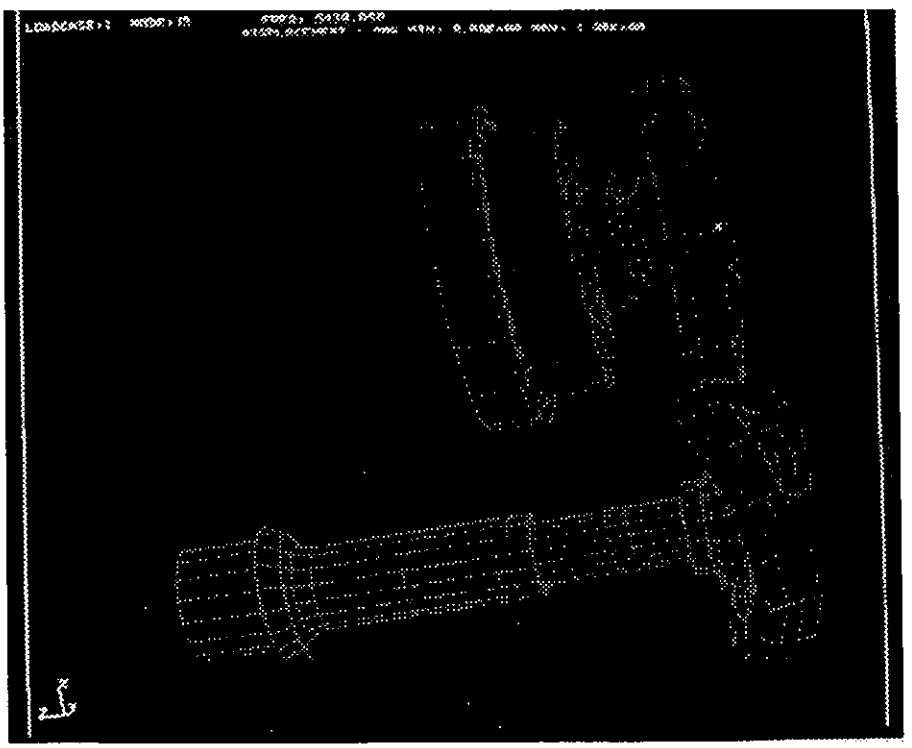


Figure 18

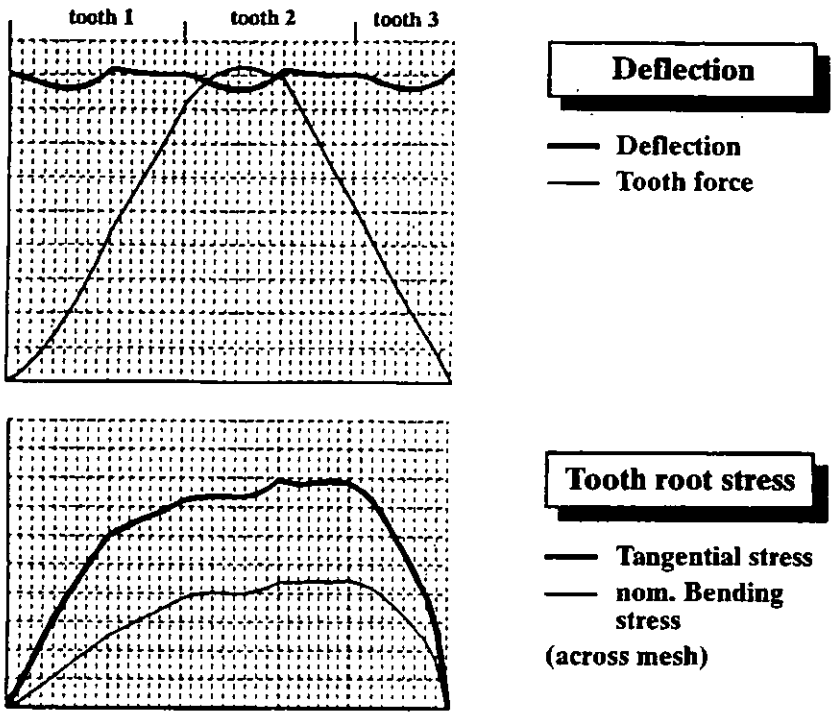


Figure 19

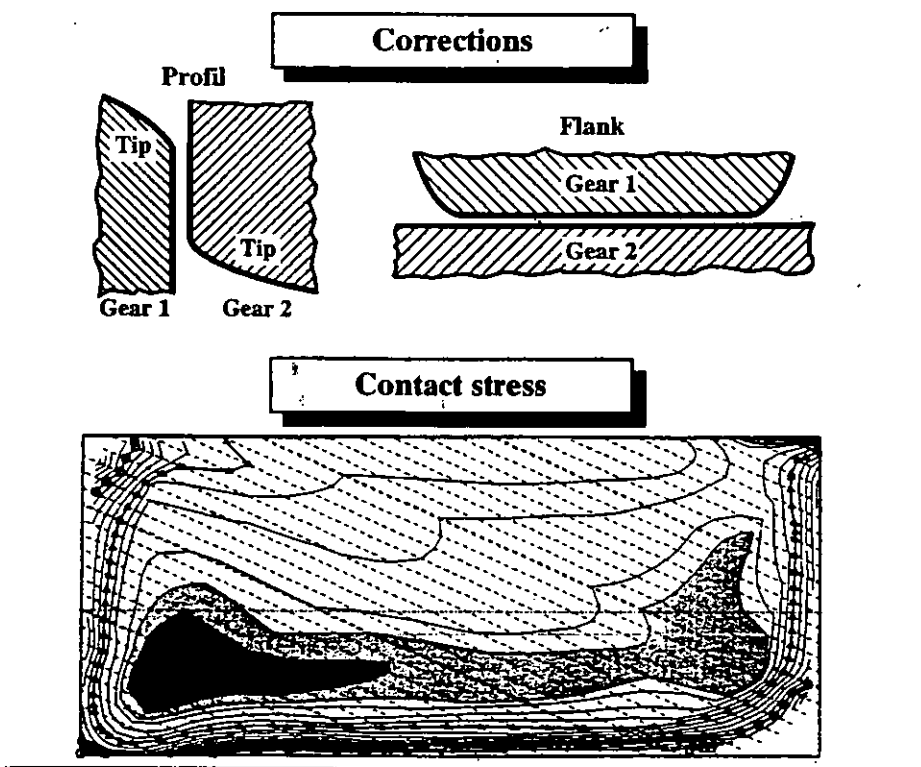


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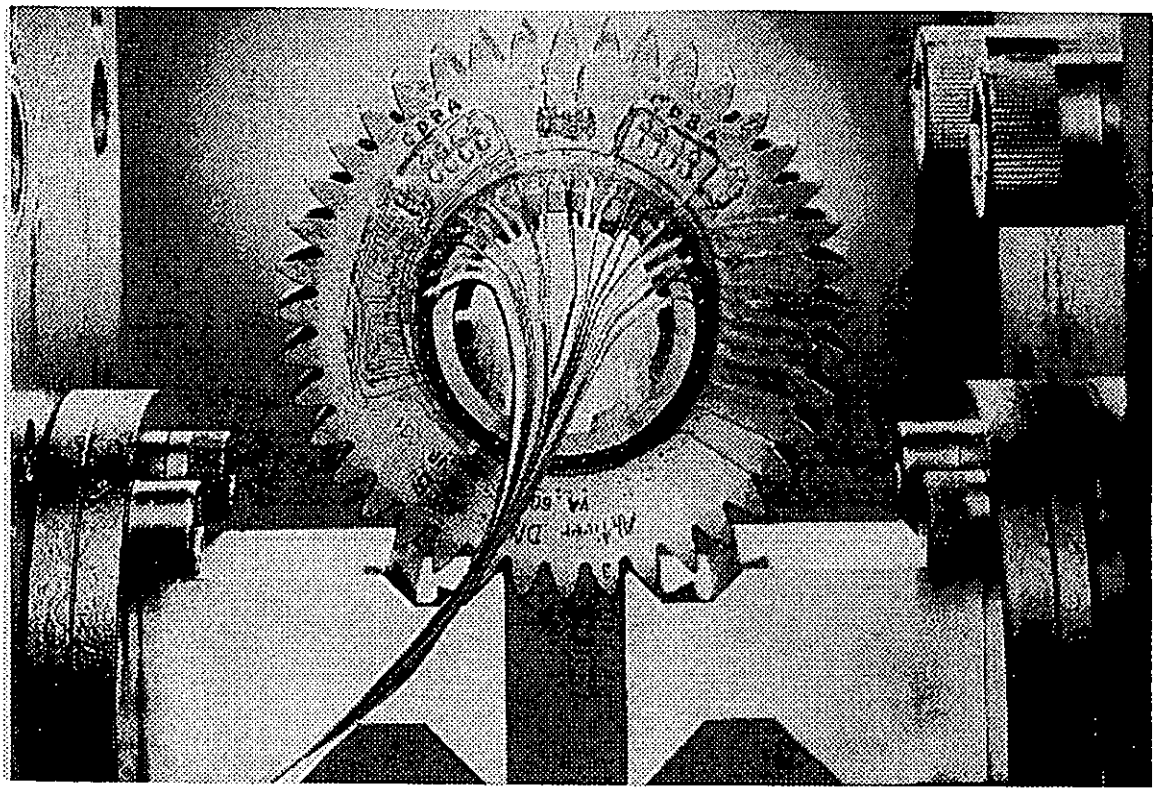


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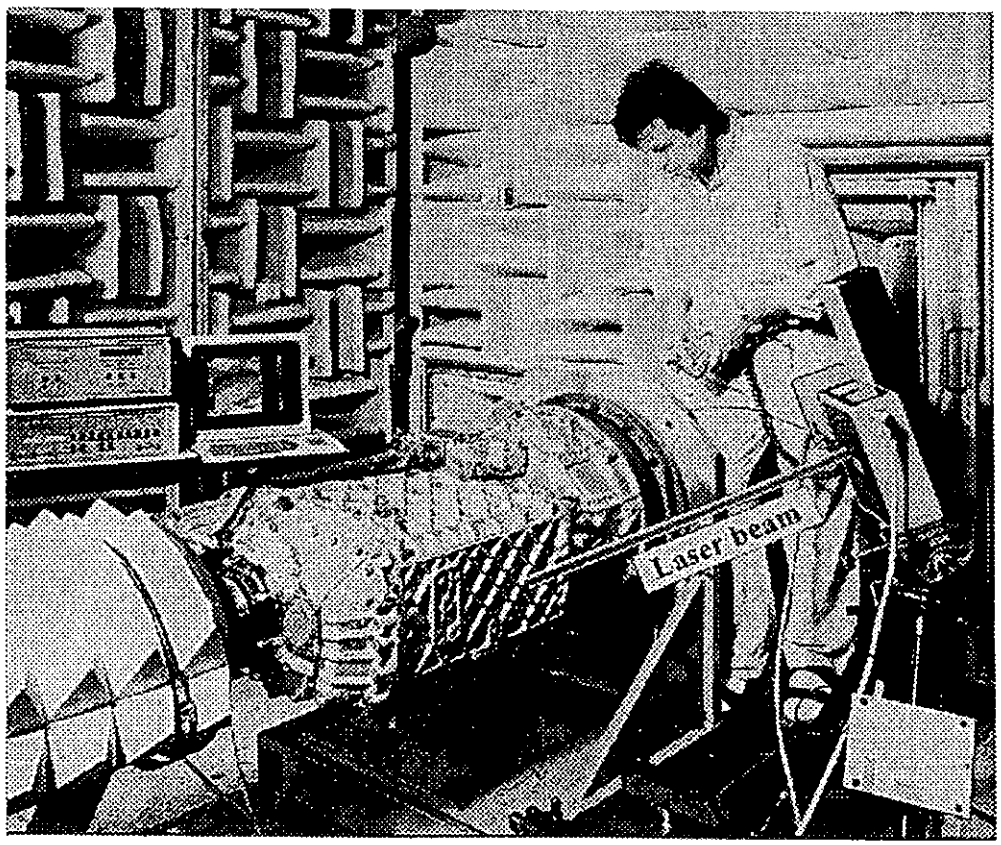


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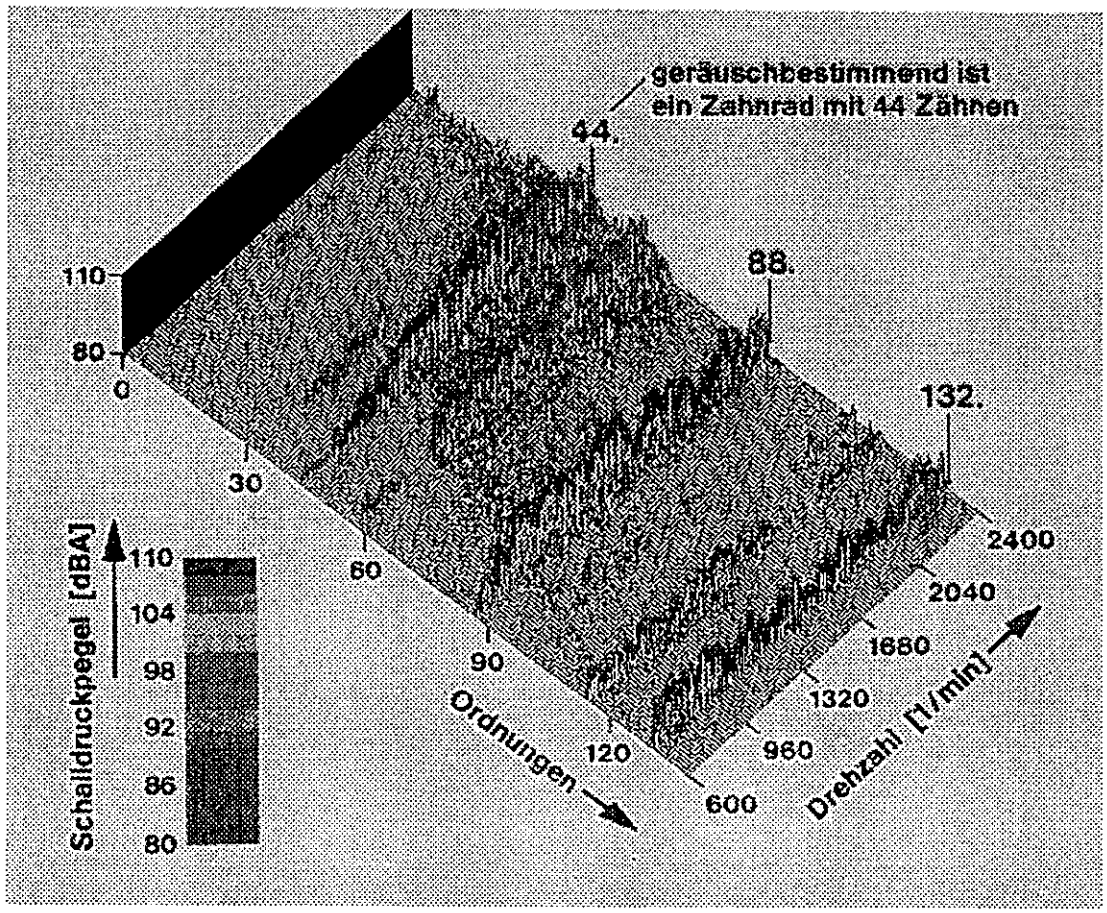


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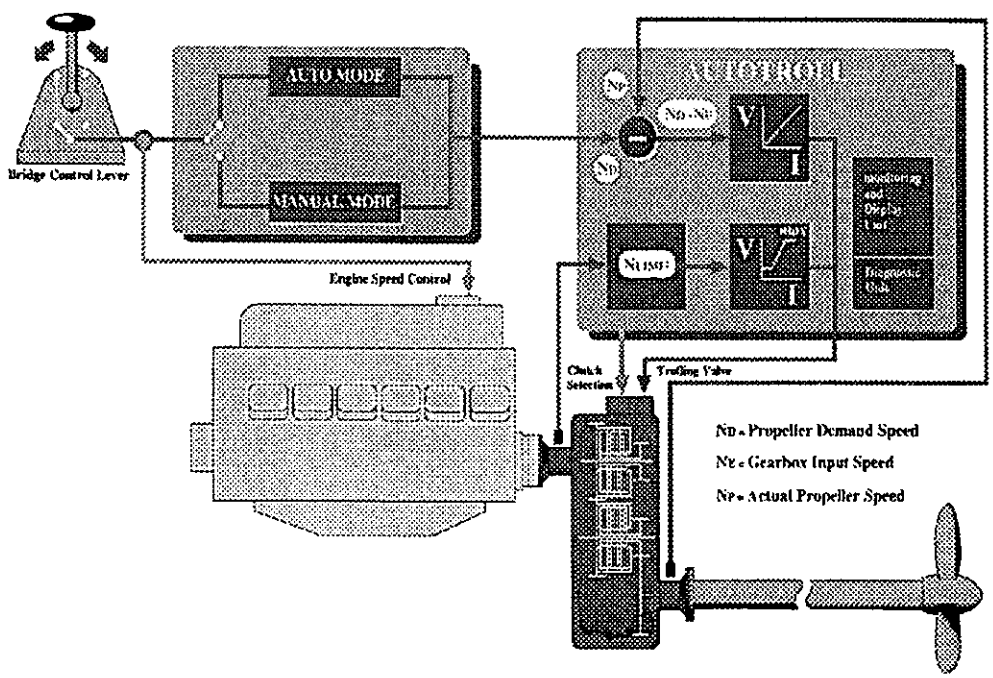


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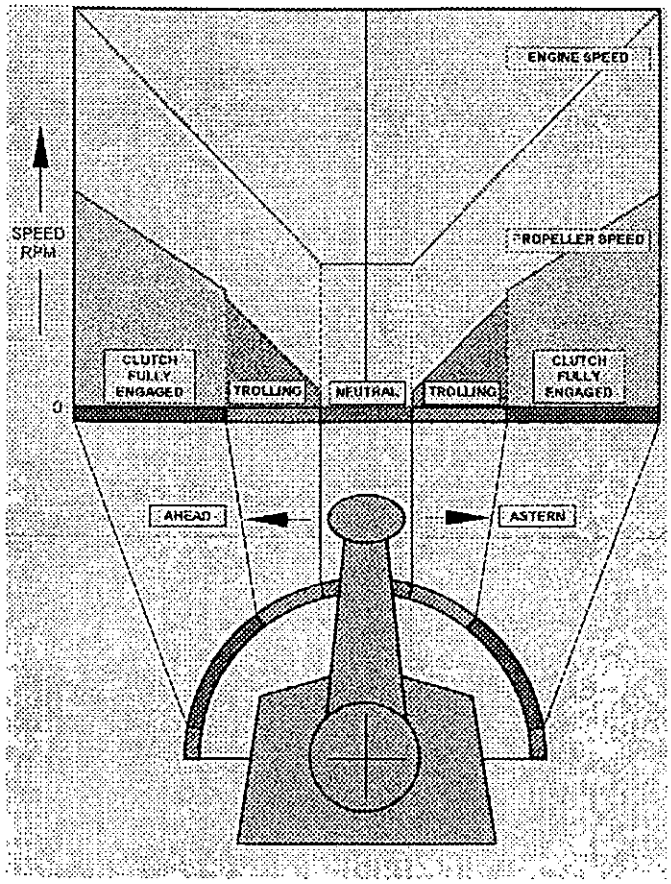


Figure 25

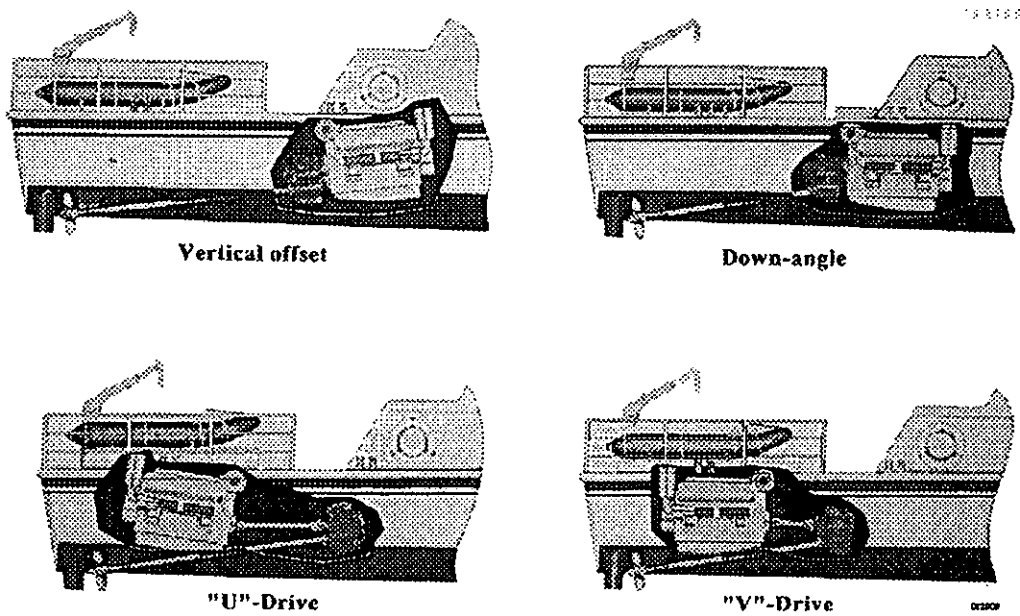


Figure 26

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

ZF MARINE TRANSMISSIONS FOR FAST FERRY APPLICATIONS

George E. Marsland, Franco M. Bennati

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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Fast Ship Propulsion Plants
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ZF MARINE TRANSMISSIONS FOR FAST FERRY APPLICATIONS

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1. INTRODUCTION

At the present time Fast Ferries appear as Mono or Multi_Hull vessels and three types of propulsion system are generally in use.

These can be defined as:

1.1 Fixed pitch propeller systems, (figure 1):

These systems are no longer suitable for the new generation fast ferries but still find applications with smaller lower speed passenger ferries.
Manoeuvring in restricted harbours can be difficult with such systems and modern Fast Ferries demand a performance which offers a minimum docking time and a fast turn around.
Fixed Pitch Propeller Systems require reversing transmissions to reverse propeller direction of rotation

1.2 Controllable Pitch Propeller Systems (CPP), (Figure 2):

These systems offer the possibility to quickly vary or reverse the propeller pitch without changing engine speed or reversing the direction of rotation of the propeller. Manoeuvring performance and system efficiency are high, and Controllable Pitch Propeller Systems find application in many present day fast ferry designs.
The system requires a reduction transmission with a single pair or a set three gears, a clutch for disengagement, a propeller thrust bearing and a rudder for steering.

1.3 Waterjet Propulsion Systems, (Figure 3):

The Waterjets are used almost exclusively in propulsion systems for larger fast ferries and provide the required performance for manoeuvrability and rapid turn round.

The 'bucket' arrangement enables thrust to be varied and reversed and steering is effected by swivelling the 'bucket' or the jet unit itself.
As with the CPP System, the Waterjet requires only a reduction transmission with a single pair of gears and clutch for disengagement.

The requirements for Fixed Pitch Propeller applications are generally covered by the ZF Marine series reverse/reduction transmissions and the range of designs for Fast Ferry applications has been specially developed for CPP and Waterjet propulsion systems.

2. PROPULSION SYSTEM REQUIREMENTS FOR FAST FERRIES:

2.1 Transmission rating and weight:

Marine engine and transmission manufacturers tend to rate equipment performance in terms of duty (power/speed profile) and type of vessel (operating time in service).
Figure 4 shows a chart with which series marine transmissions can be selected and expected to perform as required throughout service life.
Ratings and specifications for workboat applications are lower, call for deeper ratios and have heavier housings than those for vessels generally referred to as fast craft.
Fast Ferry duty can not however be categorised in this way and how this differs is shown figure 5. The indication here is that a transmission for a Fast Ferry application must not only be continuously rated for heavy duty but must also be as light as possible.

2.2 Transmission Configuration and Manufacturing Schedules, (Figure 6):

Fast Ferry service speed is of the utmost importance and this places severe restriction on hull design. This is then reflected in available engine room space and equipment layout, and is particularly critical in slender catamaran hulls.
This has the result that transmission specifications can vary widely according to individual vessel design and it is not therefore altogether possible to think in terms of a standard series design for all Fast Ferry applications. Series quality and reliability must however be observed and the task of the transmission manufacturer is therefore to maintain these standards combined with flexibility to meet customers' individual needs with limited available time.
The use of ZF R&D facilities, marine application experience and service department data enable ZF Marine to adopt a flexible but responsive approach to achieve acceptable delivery schedules with equipment as specified.

2.3 Design Philosophy:

Fitting a transmission in a particular engine room layout usually means paying special attention to input and output shaft positions and the resulting centre distance.
Available engine power ratings have usually been long established and individual tried and proven components such as clutches, gears and bearings have already been produced for these ratings.

The main consideration in a new application is the housing geometry, and the design philosophy is therefore to use existing components in a modular form to be integrated into a custom designed housing.

3. SINGLE ENGINE/TRANSMISSION/PROPULSION UNIT SYSTEMS:

3.1 An example of a Fast Ferry Transmission – The ZF Marine BU 5300:

Figure 7 shows the photograph of a ZF Marine BU 5300 transmission, custom designed for Fast Ferry applications.

This is the largest of its type currently in production and this example has a horizontal shaft offset of 600 mm and a maximum input power of 8.5 Mw at 1200 RPM.

Figure 8 shows a photograph of a similar unit using the same component modules but with a vertical offset and the input shaft below the output shaft.

Figure 9 is a photograph of a Kvaerner Fjellstrand Jumbo Catamaran in which two of the vertical offset units are installed.

3.2 Installations.

- Figure 9 shows a typical engine room layout for a Fast Ferry Catamaran with a 4 engine/transmission/waterjet propulsion system using the horizontal shaft offset BU 5300 transmissions.
- Figure 10 shows how the a similar arrangement of 4 propulsion systems can be installed in a large monohull design.
- Figure 11.1 and figure 11.2 is another monohull with 4 propulsion systems but the two centre systems are arranged one time with horizontal and one time with vertical offset transmissions to lower the engines.

3.3 Design features (Figure 12):

3.1.1 Housing:

The transmission housing is of the standard sea water resistant aluminium alloy with is used for all series manufactured transmissions.
It is anticipated that the housing will be subjected to some external forces due to hull distortion and the housing is designed accordingly. For cases where distortion may be considerable, the housing support at one side of the unit may be fitted with the ZF Marine Antiflex mounting which effectively de-couples external forces which may otherwise cause excessive stress on the housing structure.

3.1.2 Clutch:

The clutch is hydraulically actuated and is located in a separate housing fitted outside the main housing for easy maintenance. Clutch plate connections to driving components are made using involute splines.

A specially designed valve integrated into the clutch actuation cylinder allows rapid oil evacuation from the cylinder when the clutch is disengaged and external coil spring ensure rapid and complete disengagement.

The oil supply to the clutch disc pack is arranged so that no disc damage or disc distortion can occur due to overheating.

3.1.3 Bearings:

Only rolling bearings are used in the transmission and bearing rings are secured in the housing and on shafts to prevent accidental rotation. Inner bearing rings are mounted and removed using fluid injection techniques.

The bearings are arranged so that each rotating part is carried by two bearings only and one bearing only is used for axial location.

3.1.4 Connection of rotating parts:

Neither splines nor keys are used in the transmission and connection of rotating parts is made using the standard ZF Marine taper shrink fitting procedure. There is therefore no possibility for fretting corrosion to develop.

3.1.5 Output shaft seal:

The output shaft seal which is located behind the output flange is a labyrinth arrangement which has no leakage and no parts subject to wear.

3.1.6 Oil distribution system:

The top cover for the transmission housing serves an oil distribution block and oil feeds connect directly with oil holes drilled into the housing so that no oil piping is used inside the transmission.
All bearings, the gear teeth and the clutch are pressure fed with lubricating oil.

3.1.7 Time Between Overhauls:

The TBO for ZF Marine Fast Ferry Transmissions is 30,000 hours.

4. DOUBLE INPUT/SINGLE OUTPUT TRANSMISSION SYSTEMS:

4.1 Economical reasons for double input/single output transmissions:

To examine the economical reasons for the use of double input/single output transmissions it is proposed to consider a comparison between, e.g. two similar catamaran ferries, one having two single engines each driving into a transmission and a CPP propulsion unit , and a second similar vessel with the same total drive power and similar CPP units but having two double input transmissions and four engines.

4.1.1 Initial cost, (Figure 13):

It can be shown that for the same total rated power the initial cost for the two larger engines will be higher that for the 4 smaller engine. This is explained in one way by the fact that the smaller engines are produced in higher quantities.

4.1.2 Weight:

Smaller engines run at higher speeds and the 4 smaller engines are therefore lighter than the 2 larger units.

4.1.3 Operating costs (figure 14):

In the case of a 4 engine installation, the vessel can be operated with two engines shut down and by adjusting the pitch of the propeller the remaining engines can be driven at full power. This means that the vessel can be operated at about 80% of full speed with half the normal fuel consumption.
The comparison shown in Figure 14 was made by an operator in 1996 using 2 similar 35m passenger catamarans operating on a "bus stop" route in Scandinavia. The installation of different propulsion units can be regarded as incidental and the fuel savings in the case of the 4 engine installation were found to be made possible by selective use of the engines.

4.2 Practical reasons for double input/single output transmissions:

4.2.1 Safety:

The added engine redundancy achieved by the use of 4 engines means that the journey on a with an engine breakdown does not take the vessel immediately out of service and a journey can still be completed at reasonable speed with 2 engines running instead of 4.
The installation of 2 engines with the transmission between the engines avoids the use of cardan shafts and allows an additional bulkhead to be fitted between the engines which results in a higher flooding stability in case of accident.

4.2.2 Installation:

Figure 15 shows a transmission configuration and the choice of centre distances is such that the stern engine unit is raised so that the output drive shaft can be passed directly under the engine sump.
The engines are connected to the transmission using flexible couplings and cardan shafts are not riquired.
The engines are installed horizontally either in a single engine room or in two engine rooms with the additional bulkhead before or behind the transmission.
Selection clutches allow drive with either of the two engines or both together.

4.3 Double input/single output transmissions for Waterjets:

The economical and practical arguments for double input/single output transmissions also apply to waterjet propulsion systems but the advantages over the single engine arrangement can only be realised if the transmission is provided with 2 gear ratios.

Figure 16 shows a power/speed curve for single engine connected via a single input/single output transmission to a waterjet propulsion unit. The engine curve is more or less linear but waterjet curve obeys approximately a cube law.
The transmission gear ratio is selected so that the engine power at maximum engine speed corresponds to the waterjet power demand at the maximum waterjet speed. A gear ratio of 3.17 is chosen for this example.

Considering a twin engine arrangement driving the same waterjet as shown in figure 17, conditions with both engines driving are similar to the single engine system. The required transmission gear ratio is again 3.17.
If however one of the engine is shut down and the gear ratio remains at 3.17, then to avoid engine overload the engine speed must be reduced until the engine power output equates once again to the power demanded by the waterjet. This means that a twin engine system with a fixed transmission gear ratio operating with one engine shut down can never be driven with the remaining engine operating at full speed and power.

A second higher transmission reduction gear ratio is therefore necessary in order that the power output from the single engine running at full speed can be once again equated the power demand of the waterjet running at a lower speed. In the example illustrated, the second transmission gear ratio would be a certain value.

The system in Figure 15 can could be also a possible solution for a 2 speed transmission having all the features of that described for the CPP system.

ZF Marine are investigating designs for Double Input/Single output transmissions for various engine power ratings and Figure 18 shows a unit for two MTU engines designed using component modules from the well established BW 755 series transmissions.

5. FIGURES

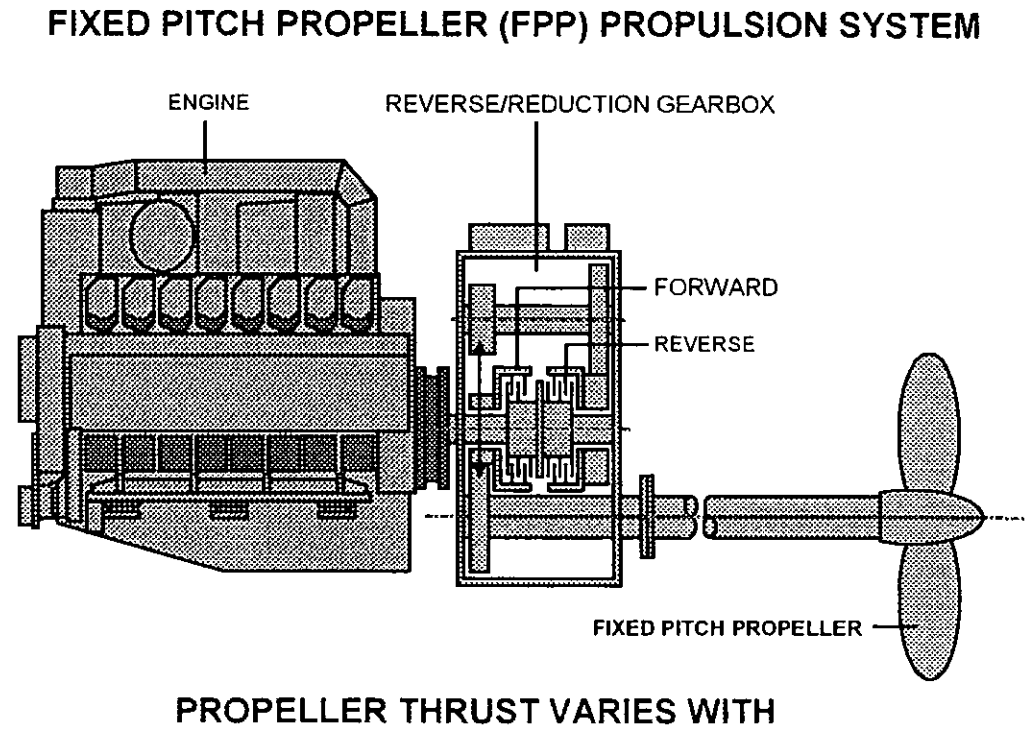


Figure 1

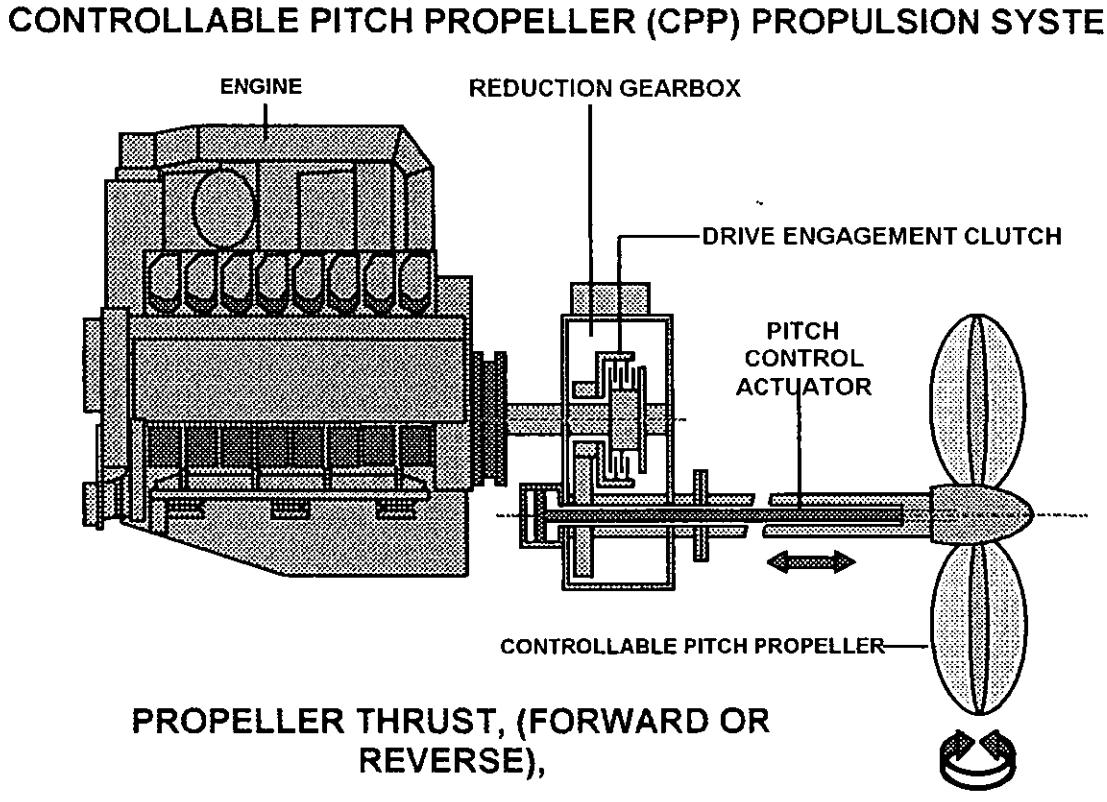


Figure 2

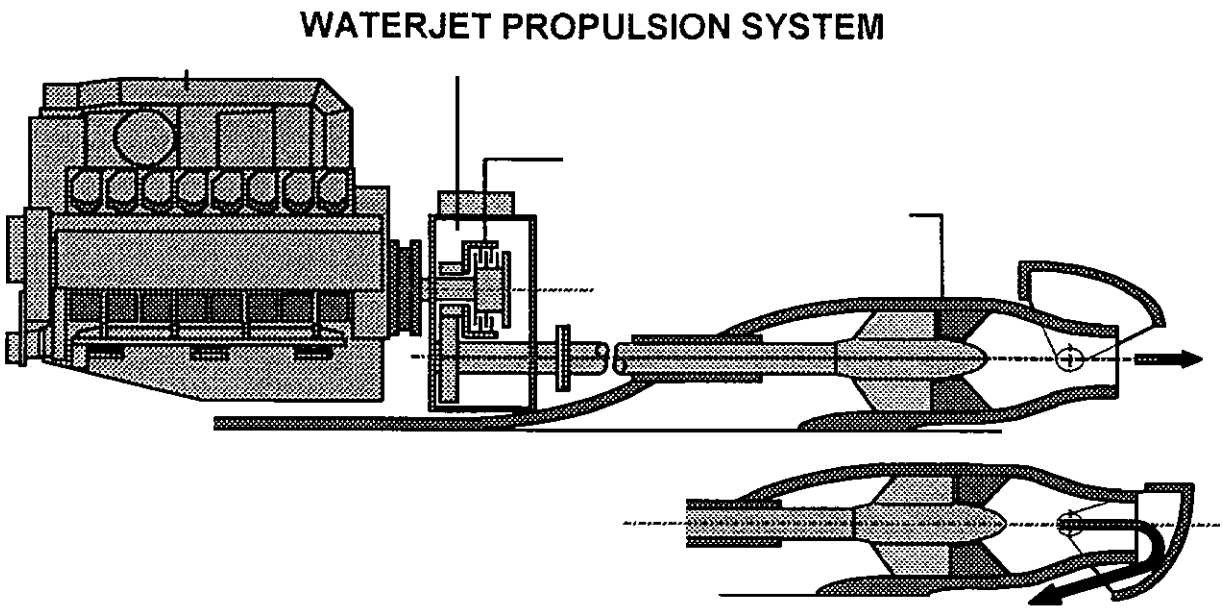
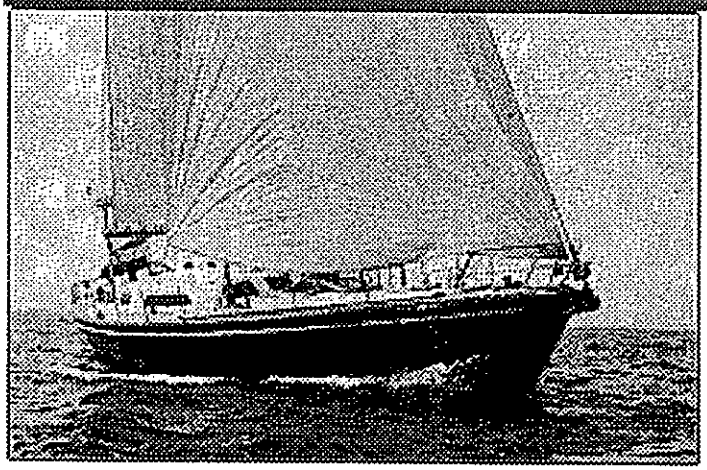


Figure 3

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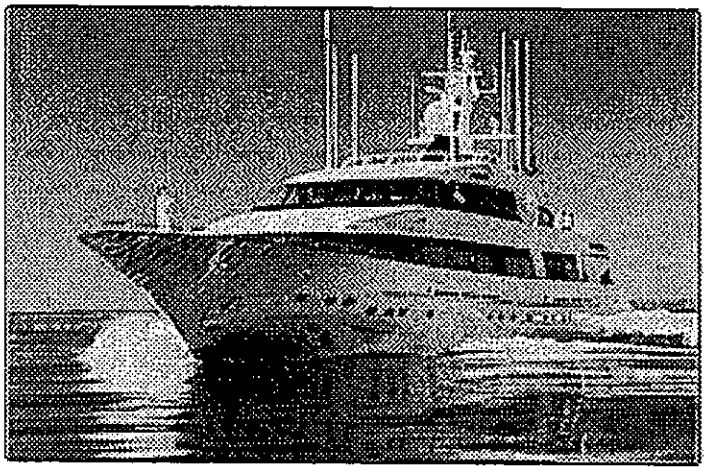
- **Highly intermittent operation with very large variations in engine speed and power**

Average engine operating hours limit: **500 hours/year (for HBW transmissions, 300 hours/year).**

Typical hull forms: **Planing.**

Typical applications: **Private, non-commercial, non-charter sport/leisure activities.**

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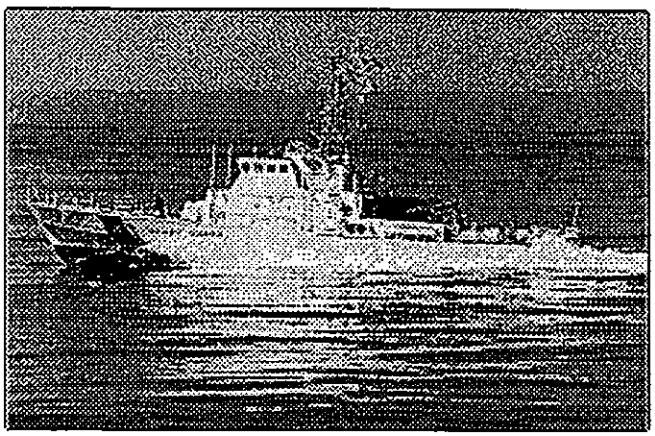
- **Intermittent operation with large variations in engine speed and power**

Average engine operating hours limit: **2500 hours/year (for HSW and IRM transmissions, 2000 hours/year).**

Typical hull forms: **Planing and semi-displacement.**

Typical applications: **Private and charter, sport/leisure activities, naval and police activities.**

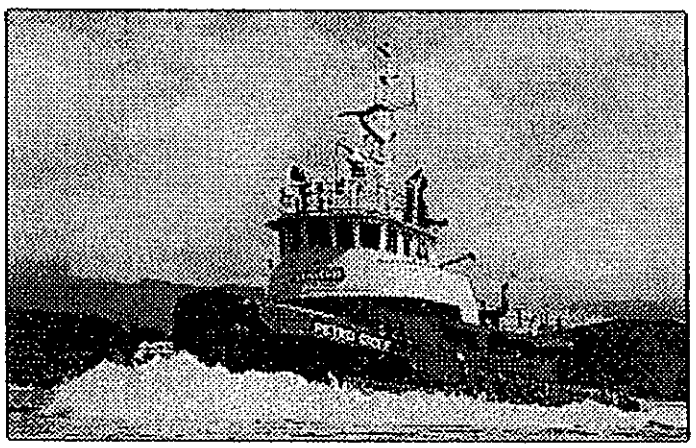
MEDIUM DUTY



- **Intermittent operation with some variations in engine speed and power**

Average engine operating hours limit: **3500 hours/year.**
Typical hull forms: **Semi-displacement and displacement**
Typical applications: **Charter and commercial craft (example: crew boats), and naval and police activities.**

CONTINUOUS DUTY



Continuous operation with little or no variations in engine speed and power

Average engine operating hours limit: **Unlimited**
Typical hull forms: **Displacement.**
Typical applications: **Heavy duty commercial vessels, tugs, fishing boats.**

Figure 4

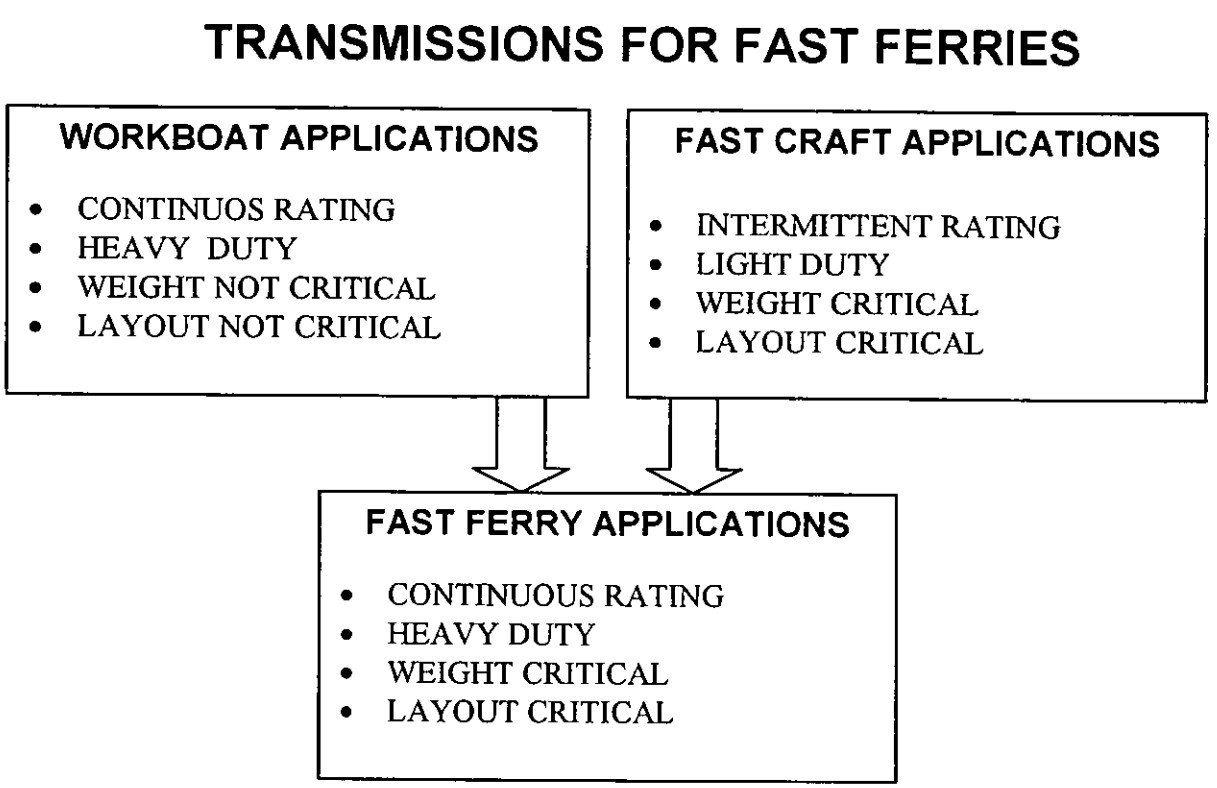


Figure 5

CUSTOM DESIGN PRODUCT DEVELOPMENT

- CUSTOMER REQUIREMENTS
- DESIGN FLEXIBILITY
- SERIES DESIGN STANDARDS
- RESEARCH AND DEVELOPMENT FACILITIES
- APPLICATION ENGINEERING EXPERIENCE
- SERVICE DEPARTMENT FEEDBACK

Figure 6

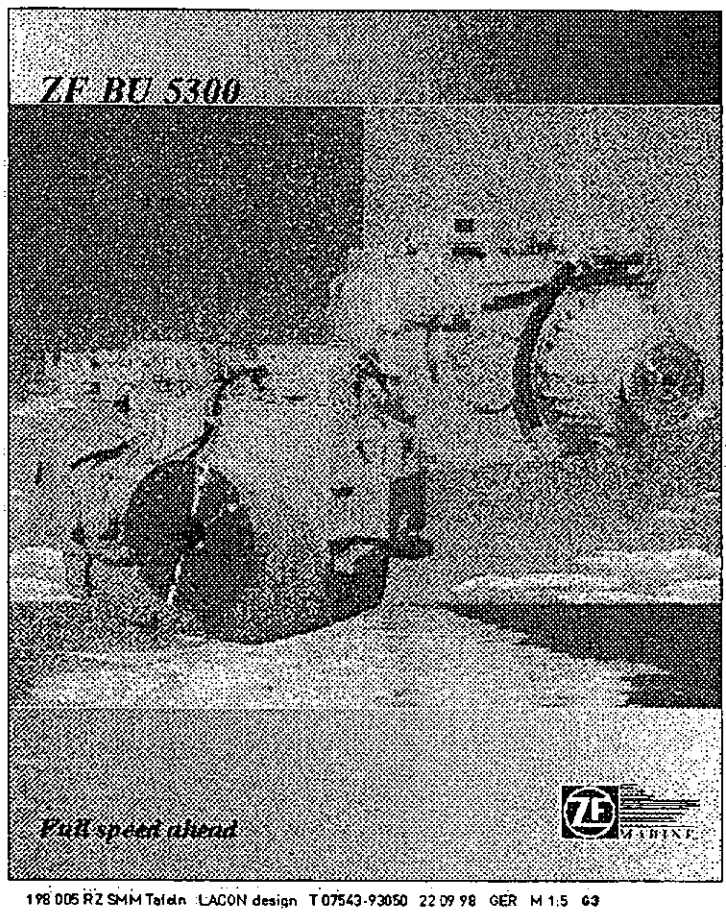


Figure 7

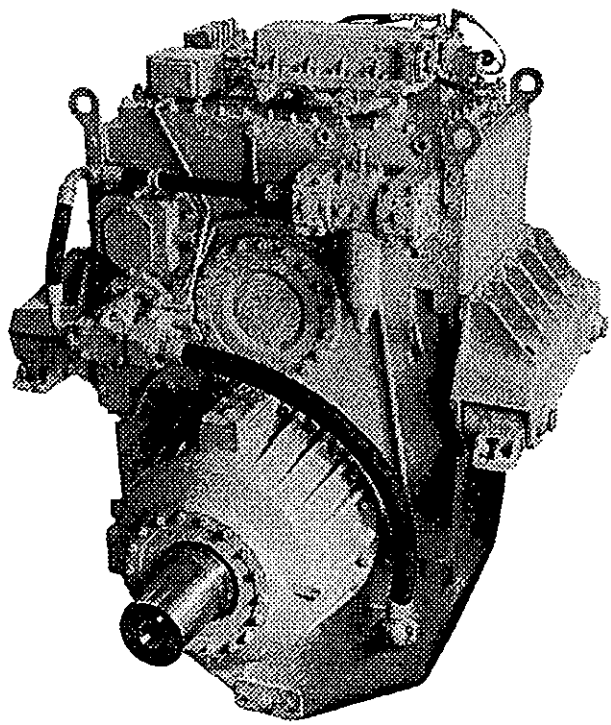


Figure 8



Figure 9

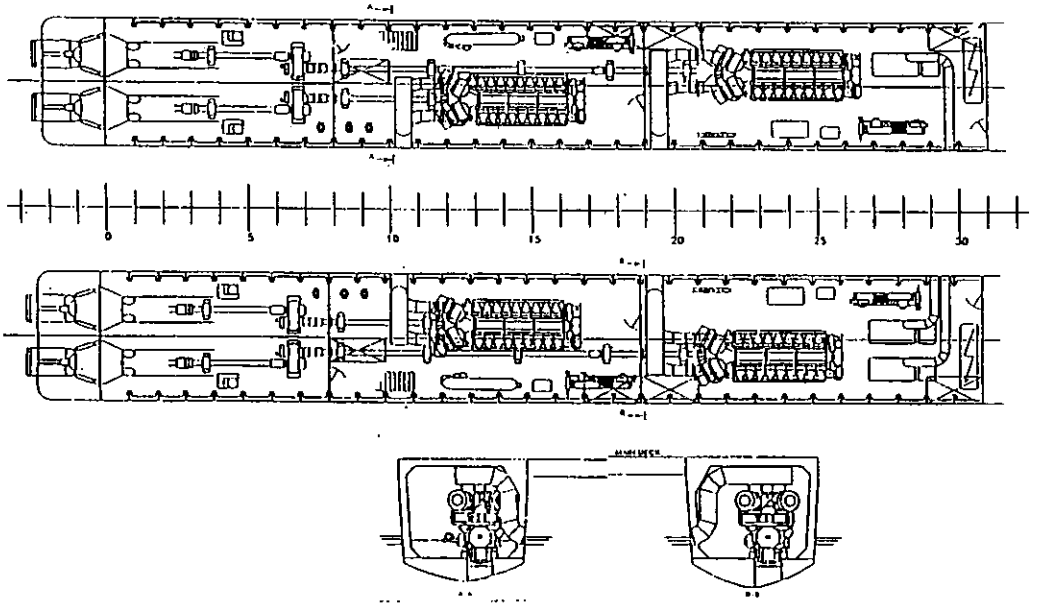


Figure 10

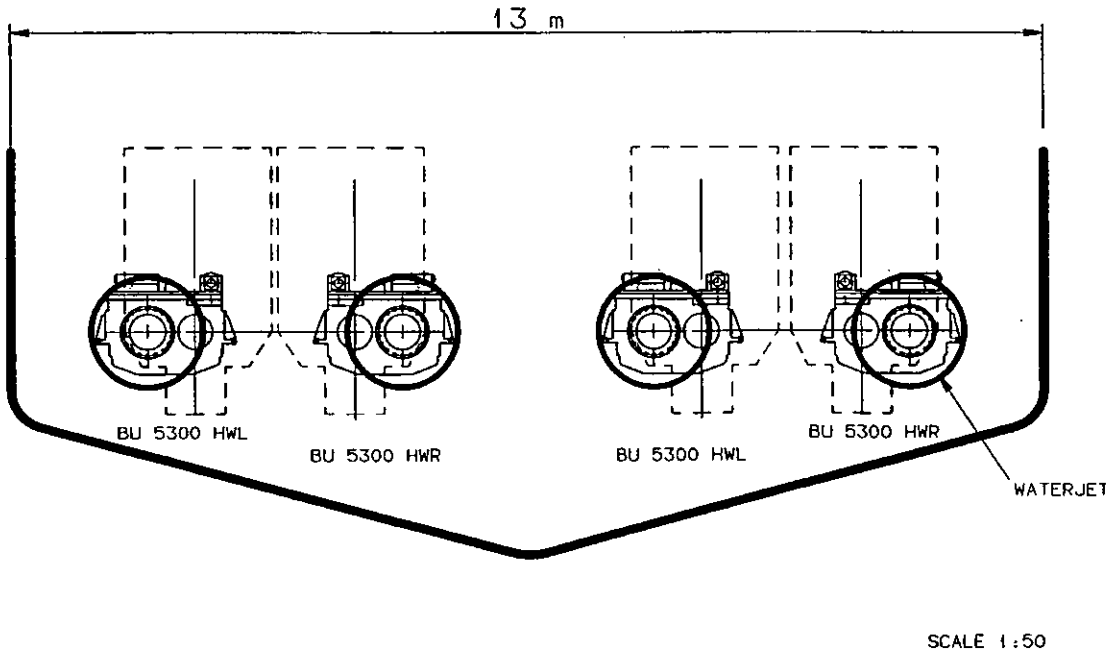


Figure 11.1

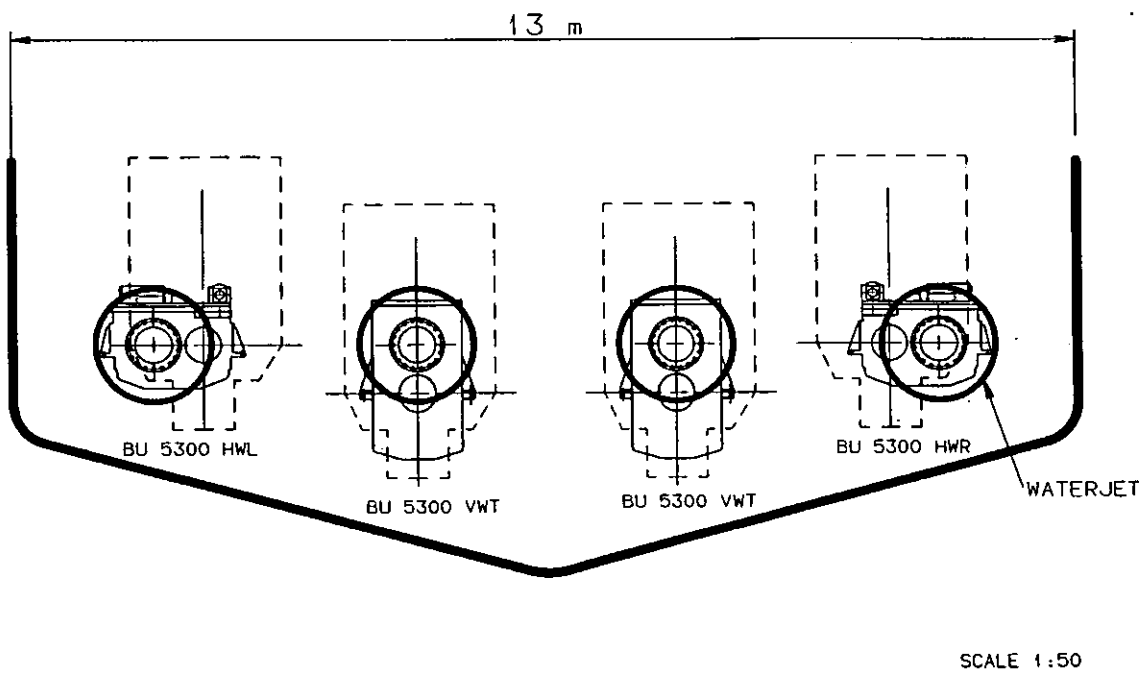


Figure 11.2

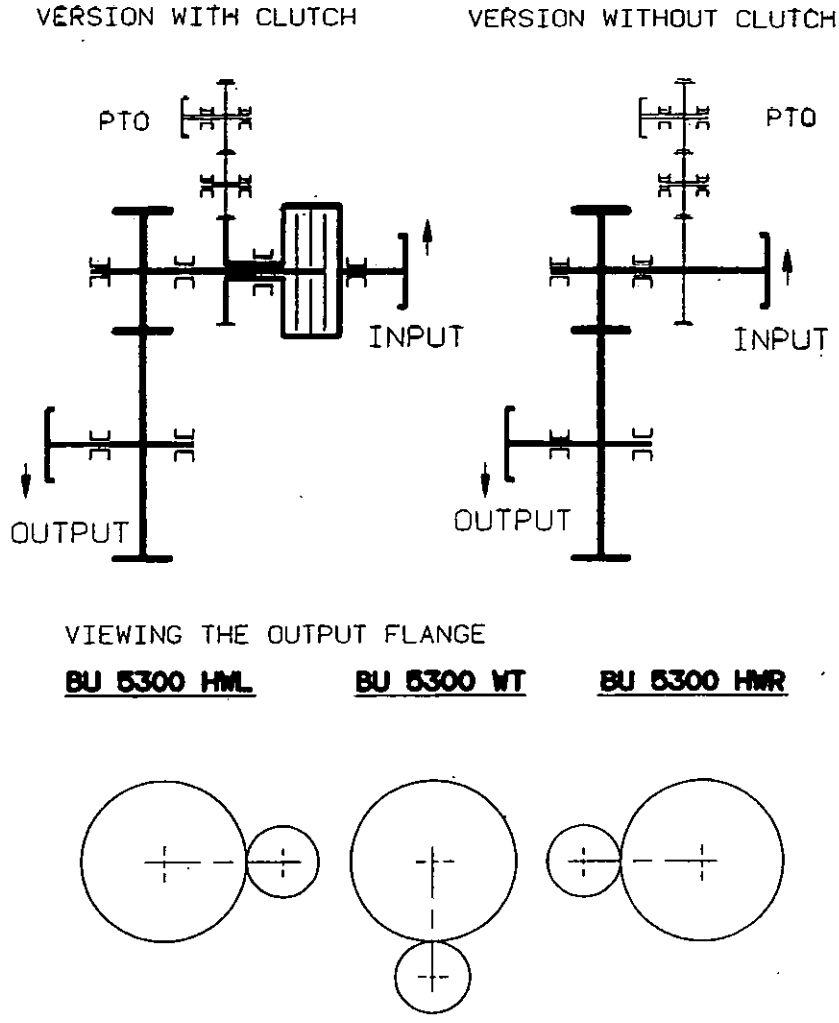


Figure 12

ENGINE PACKAGE COMPARISON

	Total Power kW	Total Weight Tonnes	Total Width mm	Total Cost GB Pounds
SINGLE ENGINE	appr. 6500	appr. 34	1830	100%
TWIN ENGINE	appr. 7500	appr. 22	1453	80%

Figure 13

COMPARISON BETWEEN TWO 35 M PASSENGER CATAMARANS OPERATING THE SAME ROUTE IN SCANDINAVIA

	VESSEL 1	VESSEL 2
• PROPULSION	2 x WATERJET	2 x CP PROPELLER
• ENGINES	1 PER WATERJET	2 PER CPP
• TRANSMISSIONS	2 x (1 IN/1 OUT)	2 x (2 IN/1 OUT)
• OPERATING MODES	1 OR 2 ENGINES	1, 2, 3 OR 4 ENGINES

THE PITCH OF THE CP PROPELLER IS SET ACCORDING TO OPERATING MODE FOR BEST ENGINE PERFORMANCE

FUEL COST PER YEAR ARE 347 TDM LESS FOR THE CPP SYSTEM VESSEL

Figure 14

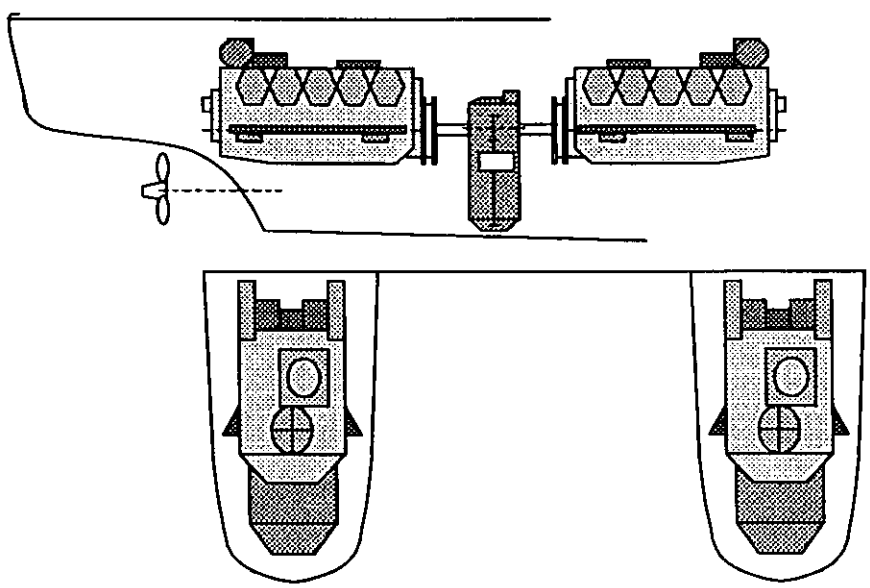
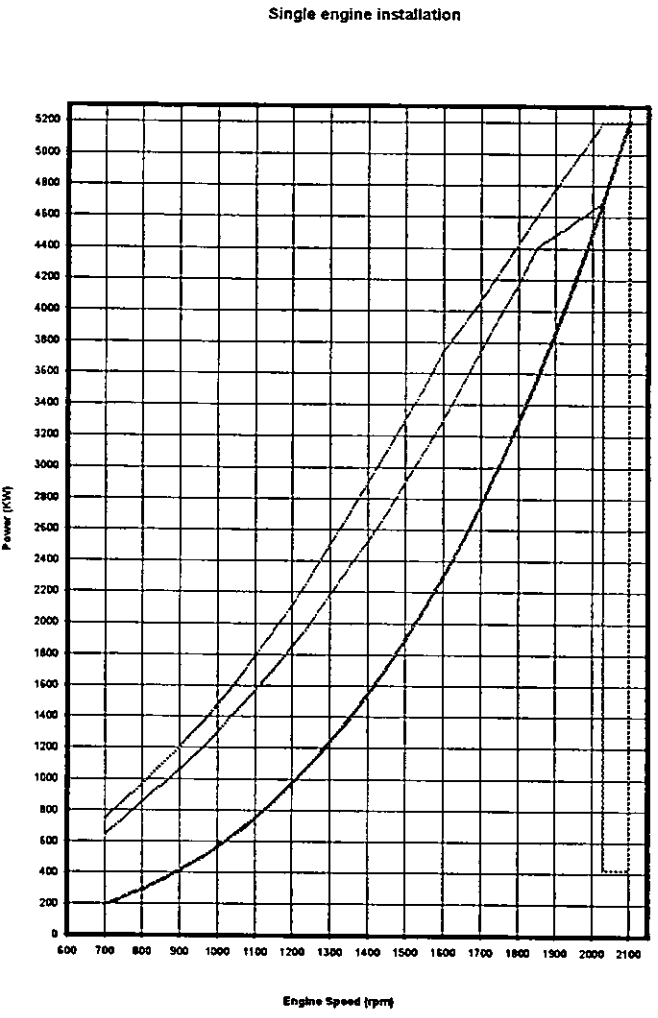
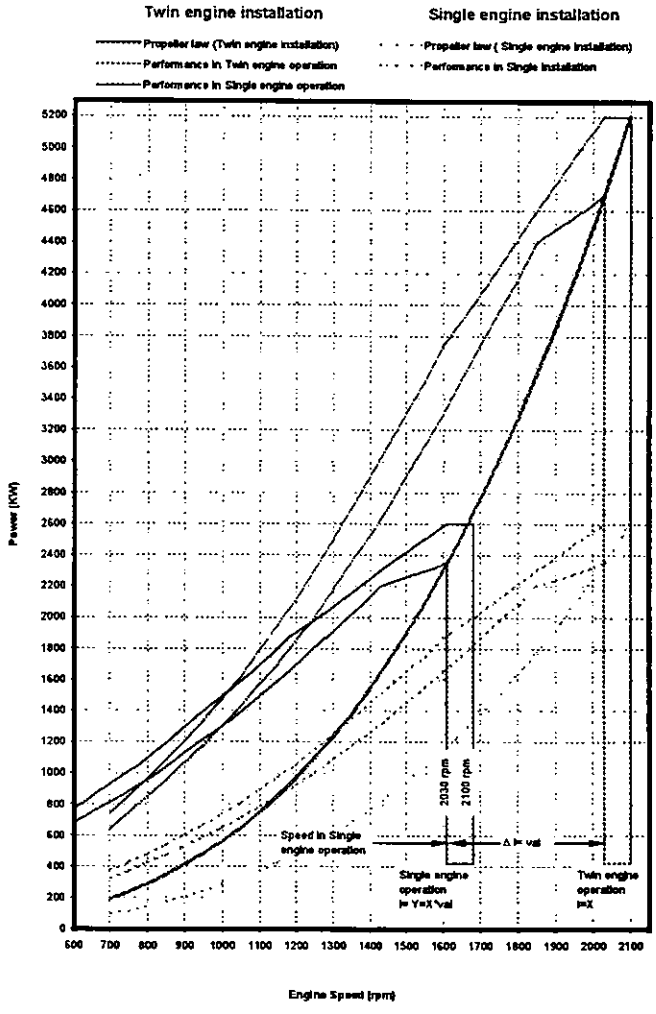


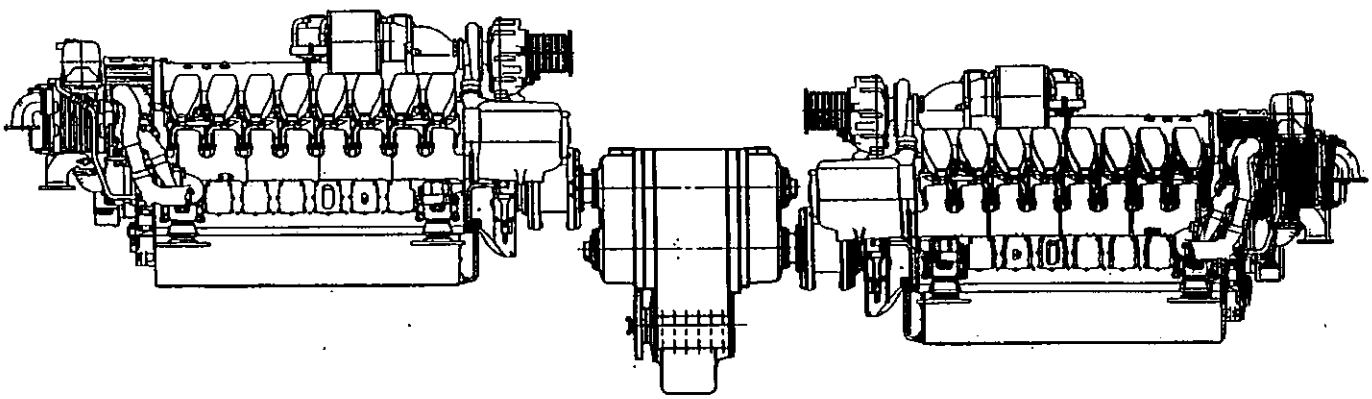
Figure 15



Figures 16



Figures 17



Figures 18

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GEAR DESIGN

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1. INTRODUCTION

Gears for marine propulsion have to fulfil various demands:
- high load capacity by low weight
- high reliability
- unlimited life
- low noise

Continuous development of gears is concentrated above all in the sectors: geometry, computing methods, strength, materials, probability of failure, lubrication and production technology (fig.1). These will influence future gear and transmission designs. Main objectives include: further power increase, improvement of noise characteristics and efficient production processes.

Figure 1: Main areas of gear development

These developments are supported by up-to-date calculation processes, test facilities, measuring and production technology[1].

2. GEAR DESIGN

An equal load distribution over the flank is the one objective of the gear design. This includes the knowledge of the deflections under load of the teeth, the shafts, the bearing and the housing. To compensate this deflections the flanks are ground with correction. Figure 2 shows the contact stress over the section of mesh of a flank with tip relief and end relief.

Figure 2: Contact stress over the section of mesh

An extensive system of programs, coupled with special gear and transmission databases support the design of gears (fig. 3).

Figure 3: Program system for gear and transmission design

- **Stitea**: designing spur and helical gears using ZF know-how and minimum input data.

- **Getwa**: transmission analysis, tooth and bearing forces, rotational speeds, durability of bearings.
- **Stifer, Stiket, plate theory program:** calculation and optimisation of spur and helical gears according to various criteria (contact stress, bending stress, temperature, dynamic forces, noise...)
- **RIKOR:** calculation of shaft and gear deflections, necessary for tooth corrections.
- **M/K sheet program:** compiling production documents.
- **StPlus:** Calculation of geometry and load capacity of cylindrical gear according to DIN3990, ISO6336 and various classification rules.

In basic research on back-to-back gear test rigs, the computational results are checked by tooth root strain measurements. For this, miniaturised strain gauges are glued into the tooth root fillets. As Fig. 4 shows, the calculation programs are now so far developed that good agreement between measurement and calculations is observed. Computational methods tested in this way enable reliable gear tooth calculations to be made in practice.

Figure 4: Comparison of measured and computed tooth stress

3. FEM

3-dimensional calculations are needed to determine the specific influences on beveloid gears and new geometrical designs. The FE-Model includes the shaft, the bearing and other elements who are involved in the force flow. Figure 5 shows a thin rim beveloid gear fitted on a friction bearing. The objective of this research is the dependence of bearing play to tooth root stress

Figure 5: FE-Model of an pinon including the shaft, the bearing and the cluth disc carrier

Figure 6: Tooth root stress of a beveloid gear

For the calculation of splines, shrink fits etc. a non-linear FE program system "Marc" can be used. The program system allows an automatic calculation of the contact zones without gap elements. Fig. 7 shows the stress distribution for a spline connection.

Figure 7: Stresses in a spline connection

4. NOISE

The fluctuation of tooth stiffness on the path of contact and the resultant torsion angle error is one cause of noise excitation. It is influenced by the macro and micro design of gearing (fig 8)

Figure 8: Options for reduction of noise generated by gears

Order analyses are **frequency analyses** relating to a reference rotary frequency (e.g. input shaft). With dynamic events (speed changes) it is therefore possible to identify and quantify speed-dependent noise-shares (e.g. of a tooth meshing frequency) as a high-resolution "order" instead of as a broad frequency band. In the case of a tooth engagement with the driven toothed wheel having 30 teeth (30 engagements per revolution) this means that its noise is

analyzed as an amplitude of the 30th order, even though a broad frequency band between 350 Hz and 1200 Hz in the speed range 700 - 2400 1/min is passed through.

Figure 9: Frequency analyse (Cascade diagram)

Figure 10: Order analyse (Cascade diagram)

- **Representation**

Cascade diagram

Representation of frequency respectively order spectrums as 3D diagrams in a dependent relation to the speed. They serve to represent noise behavior during speed variation. If needed for better clarity of representation of a cascade diagram, the 3D order spectrum data set is reduced by averaging successive spectrums. In such a case e.g. averaging [lin]/50 means that the spectrums represented are assigned speed support points at a distance of 50 1/min. The spectrums represented were then generated by means of a linear averaging of the original spectrums. Type and speed range of averaging are each indicated at the upper margin of the cascade diagram (fig. 9, 10).

Campbell diagram

Basically corresponds to the cascade diagram. The Campbell diagram is a two-dimensional representation with the third dimension (level heights) represented by different colors. Color / level assignment can be done via color chart (range of shades) next to the diagram.

Figure 11: Campbell diagramm

Averaged order spectrum

Energetic averaging across n order spectra of a speed range results in a representation as an averaged order spectrum. It can be used to arrive at the average noise share of a gearing by adding up the amplitudes of the tooth meshing frequency (e.g. 30th order) plus their upper undulations (e.g. 60th+90th+120th+... order) and stating it as an energetic single numerical value (time-average sound pressure level) in dB (A or lin).

Markings ("v") with a label above indicate the gearing order (tooth engagement frequency) and their multiples of the respective gear set. Label / gear set assignment can be indicated in a legend, which also contains the averaging level and the resolution (band width).

Figure 12: Averaged Order Spectrum

Level speed ranges

Level speed ranges indicate the overall noise level (sum of all orders) and the levels of individual orders in their dependent relation to the reference speed. The level speed range shows how individual gearing noise proportions across the speed variation behave in relation to each other and as compared to the overall noise. Moreover, resonance behaviour of a transmission can be observed.

The ranges represented can be unequivocally assigned to the respective gearing noise only if they stand out as peaks in the cascade or the Campbell diagram. If there are no peaks recognisable in the orders represented in the 3D diagram, the noise represented was caused by other vehicle noise (e.g. engine, tire, ancillary units, wind) or transmission noise (e.g. bearings).

Figure 13: Level Speed Range

5. MATERIALS

Optimised gear reliability requires specific knowledge of the permissible material strength. Systematic investigations and practical experience showed that a high material toughness of steel heat chosen is favourable for high impact and endurance strength. A process was developed for ductility tests (fig. 15). The cyclic pulsation strength of different batches of casehardened 20 Mn Cr5 with different ductility is shown[2].

Figure 15: Influence of heat selection on fatigue limits of 20 Mn CR 5

Bending strength of case hardened gears can be improved be shot peening. The degree to which performance can be increased depends on both the strength and microstructure of the material before treatment. The considerable improvement in performance is not caused by a higher amount of hardness but originates from a marked growth in compressive residual stresses.

Figure 16: Influence of shot peening (compressive residual stress) on tooth bending strength.

6. LUBRICATION

Defects on gear flanks such as scoring, scuffing, pitting, grey staining and wear are affected by the lubricant used.

Viscosity (oil film thickness), molecular structure and additives determine the friction coefficient and thus loads and stresses of the material. Lubricants may also influence the strength of the material surface due to chemical and physical interaction.

Pitting tests have shown that pitting Service life can differ by a factor of 2 if oils with the same base viscosity but different additives are used.

ZF has developed a new method (based on plate theory) to calculate the temperature spread and scuffing resistance of gear flanks, considering tip relief and crowning of the gears (fig 17)

Figure 17: Scuffing on test and calculated safety factor of scuffing

Until now, the consideration of lubricant influence in gear calculation methods, i. e. DIN 3990, has been insufficient. The choice of the correct lubricant becomes more difficult and/or expensive due to the fact that the influence of lubricant on service life can be calculated only to a certain extent. Therefore, compressive component and transmission tests are necessary in order to determine the longlife properties of lubricants as a design element.

Performance data from the lubricant producer can only be obtained with standardisation of lubricant tests. Practice orientated lubricant tests were developed largely on the basis of the FZG gear dynamometer test bench. Most of these gear-related lubricant tests can be carried out in ZF's gear lab.

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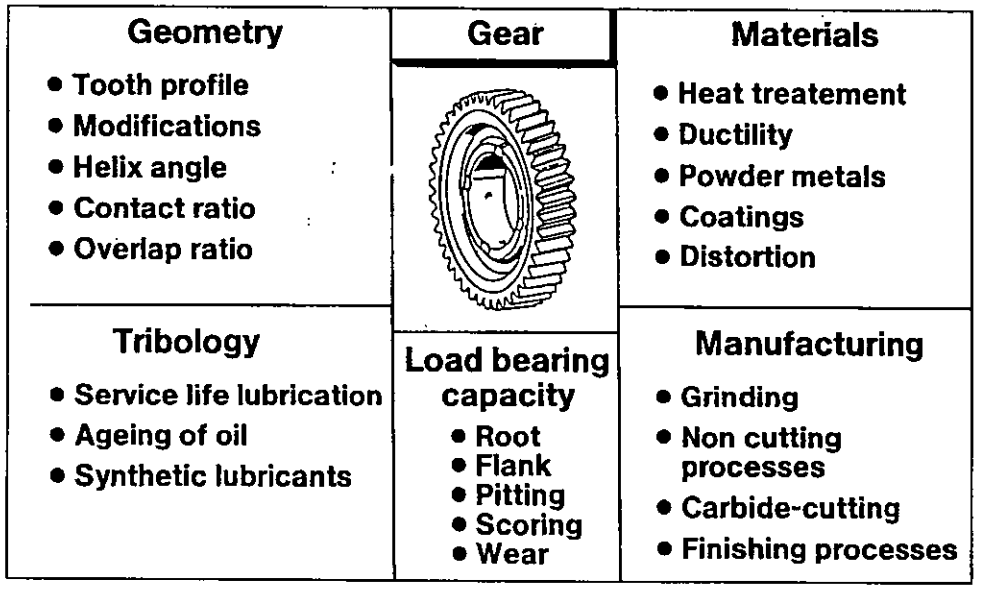


Figure 1: Main areas of gear development

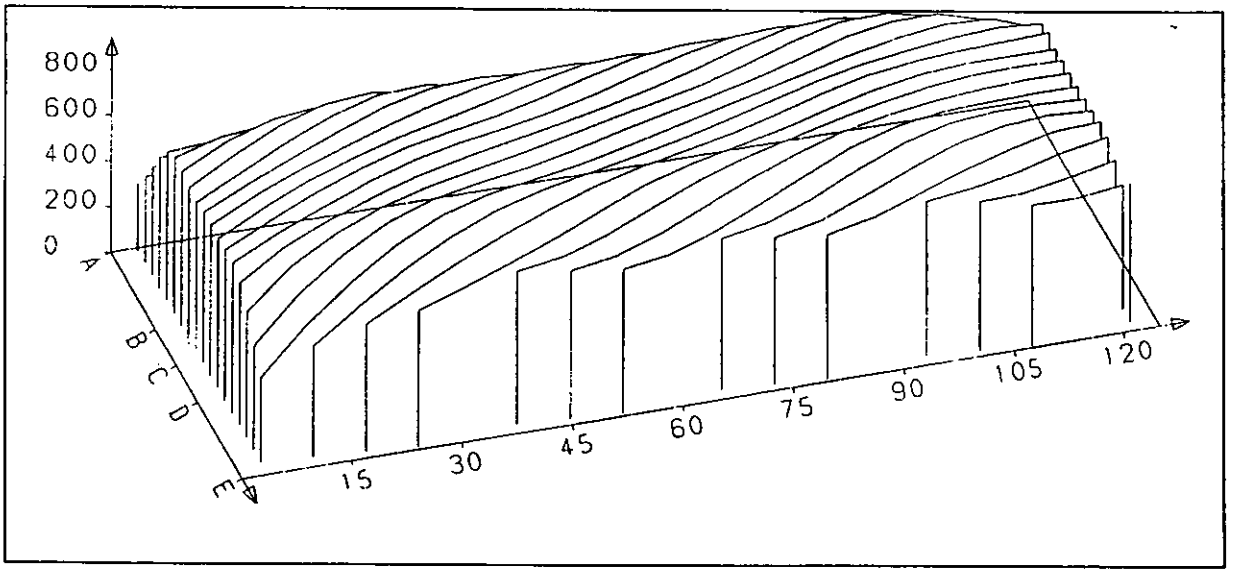


Figure 2: Contact stress over the section of mesh

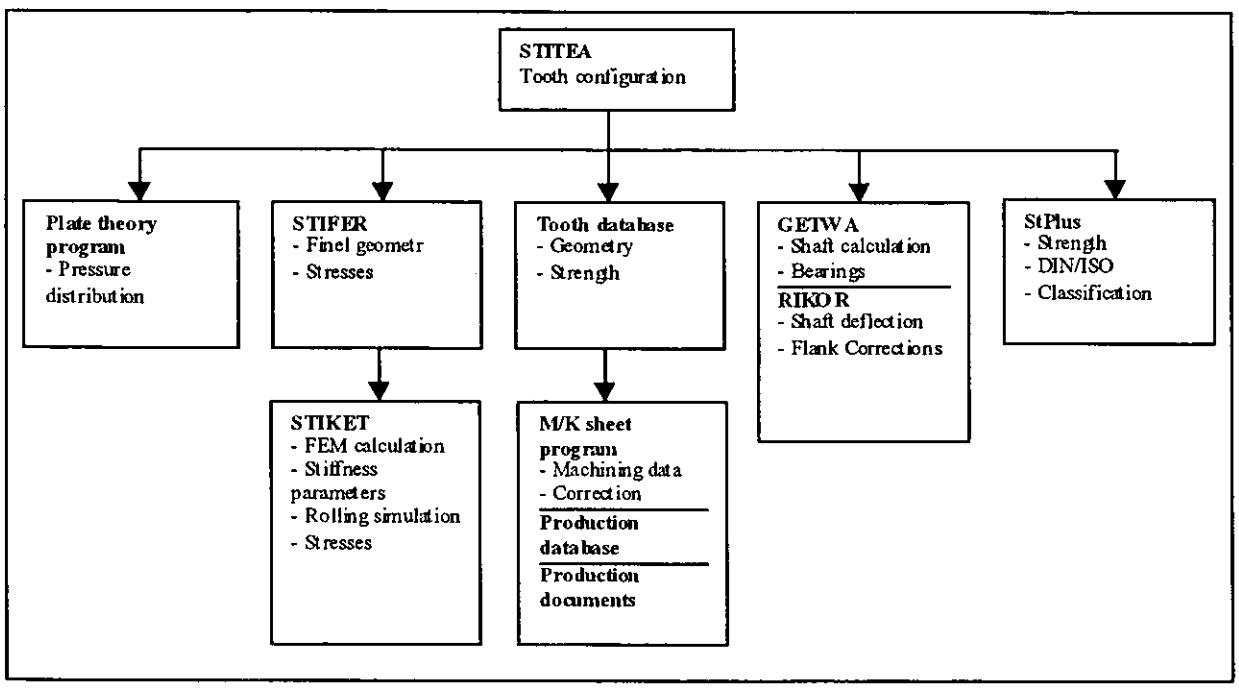


Figure 3: Program system for gear and transmission design

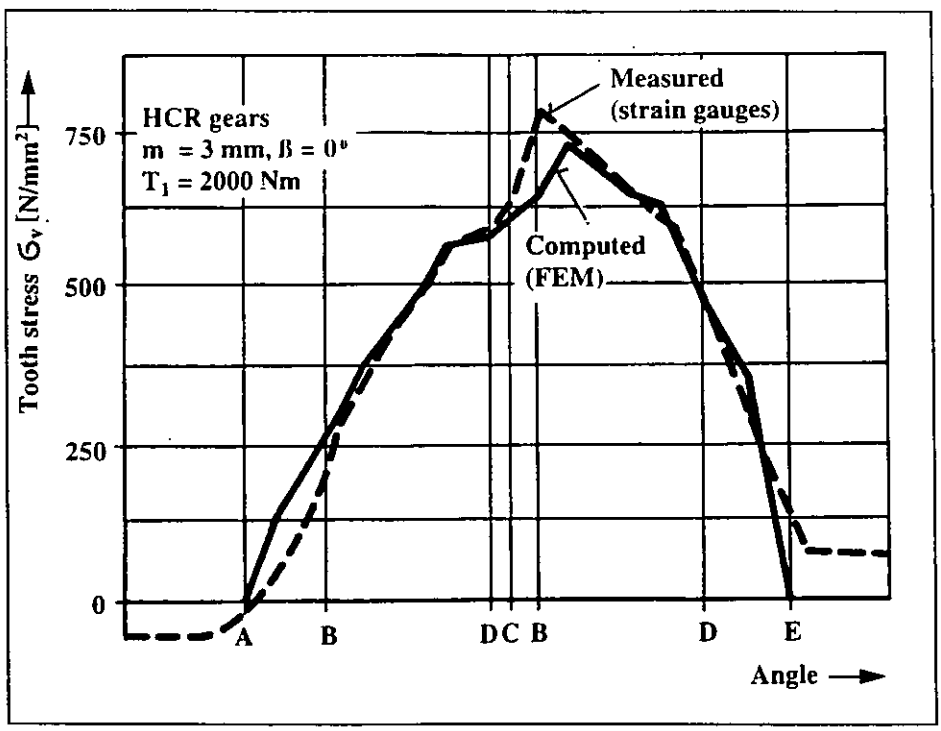


Figure 4: Comparison of measured and computed tooth stress

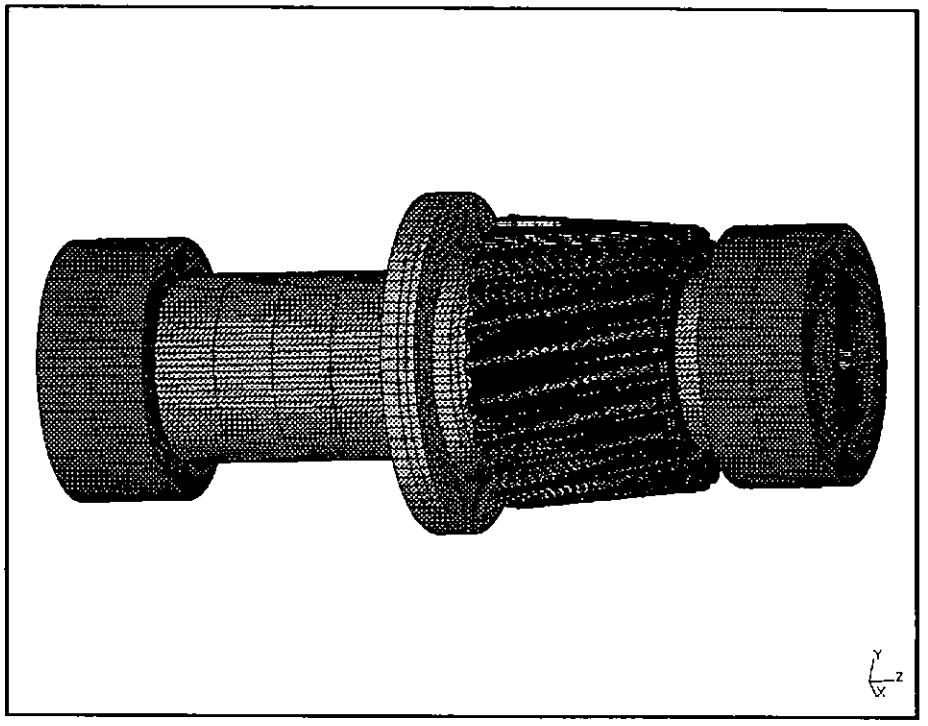


Figure 5: FE-Model of an pinon including the shaft, the bearing and the cluth disc carrier

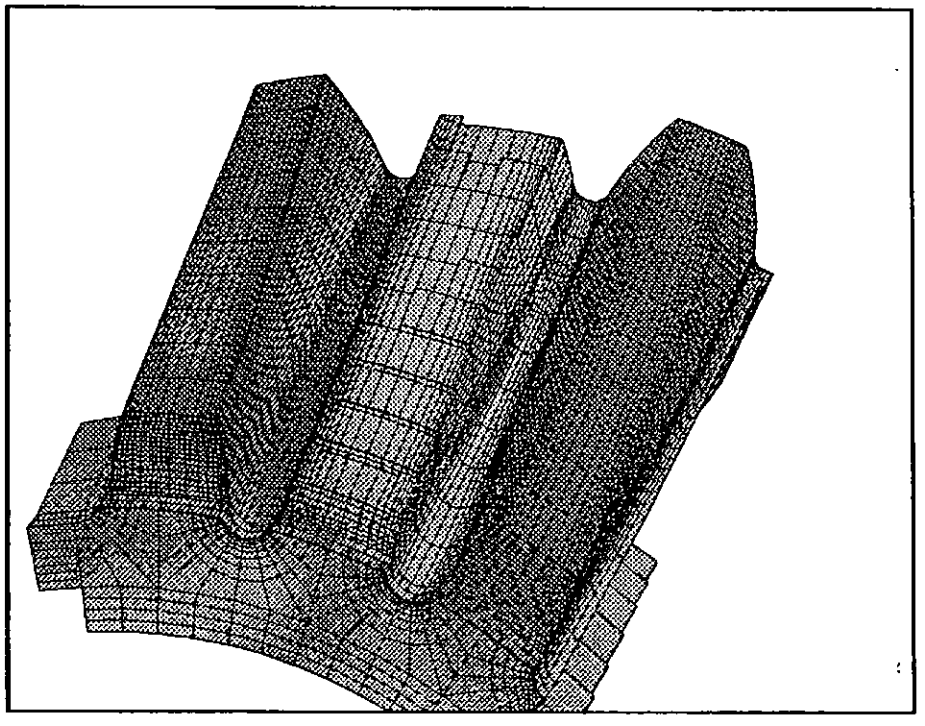


Figure 6: Tooth root stress of a beveloid gear

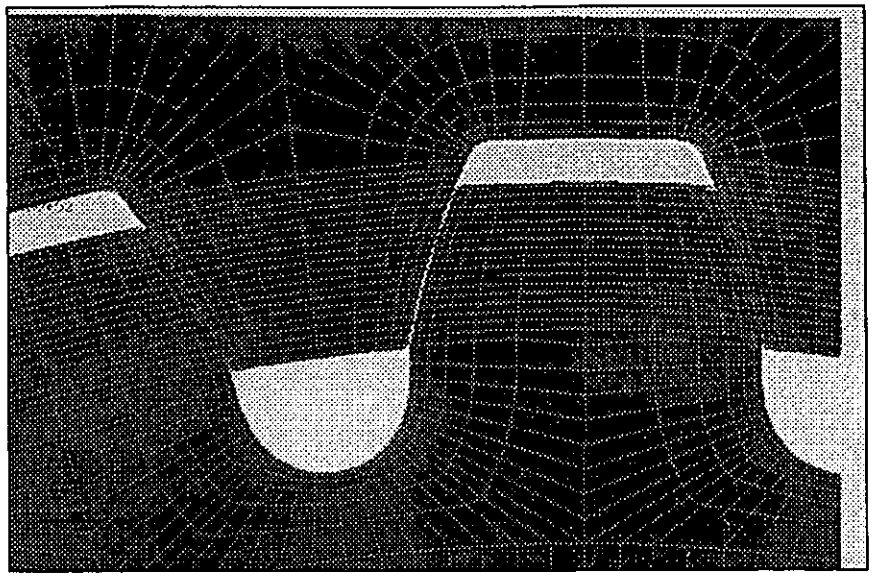


Figure 7: Stresses in a spline connection

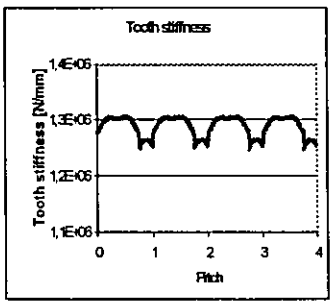
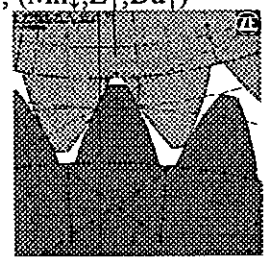
Cause	Remedy
•Gear toorh stiffness variation 	•Reduce deviation level $\varepsilon_\alpha\uparrow, \varepsilon_\beta\uparrow$, (Mn↓,Z↑,Da↑) 
•Meshing faults -Elasticity -Tooth error	•Appropriate profile modifications
•Surface	•Waviness ↓, mashine in direction of meshing contact
•Production process with deviation	•Tooth quality ↑, concenticity, tumbling
•Quality of assembly	•Bearing, contact pattern
•Shape of gear body	•Noise radiation, rigid structure
•Operating conditions	•Speed ↓, Load ↓

Figure 8: Options for reduction of noise generated by gears

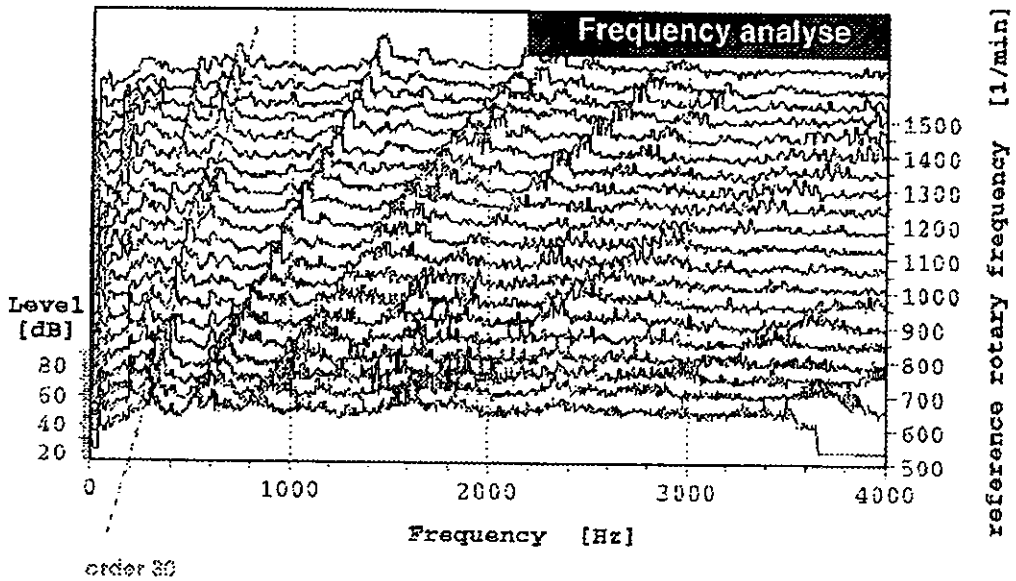


Figure 9: Frequency analyse (Cascade diagram)

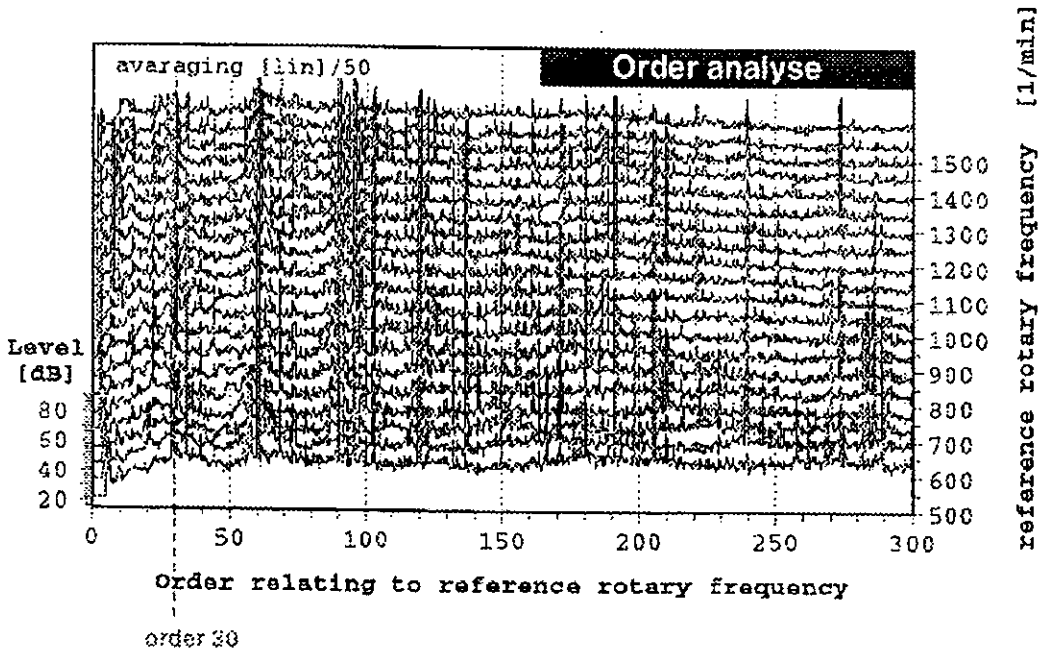


Figure 10: Order analyse (Cascade diagram)

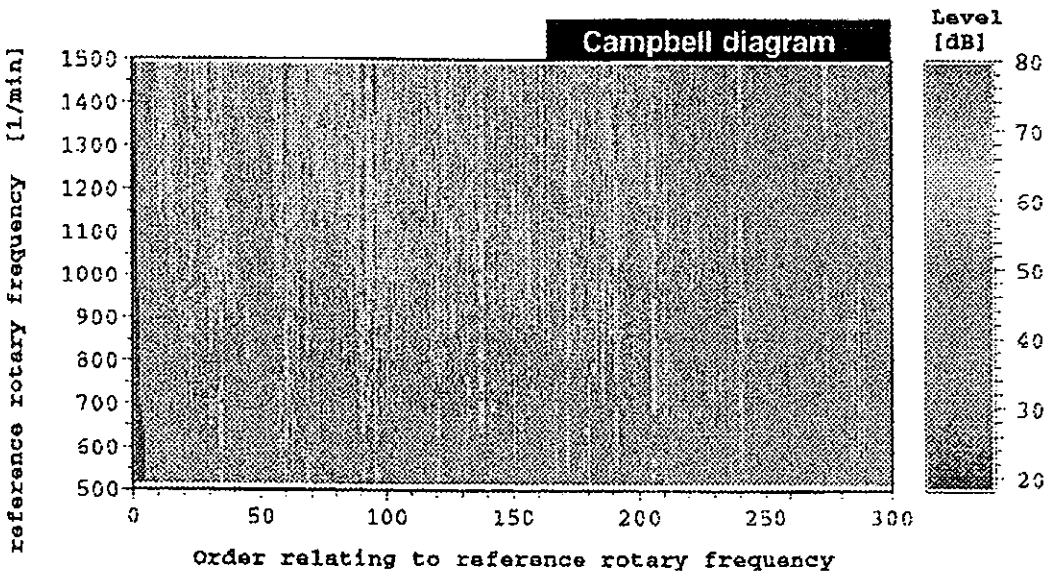


Figure 11: Campbell diagramm

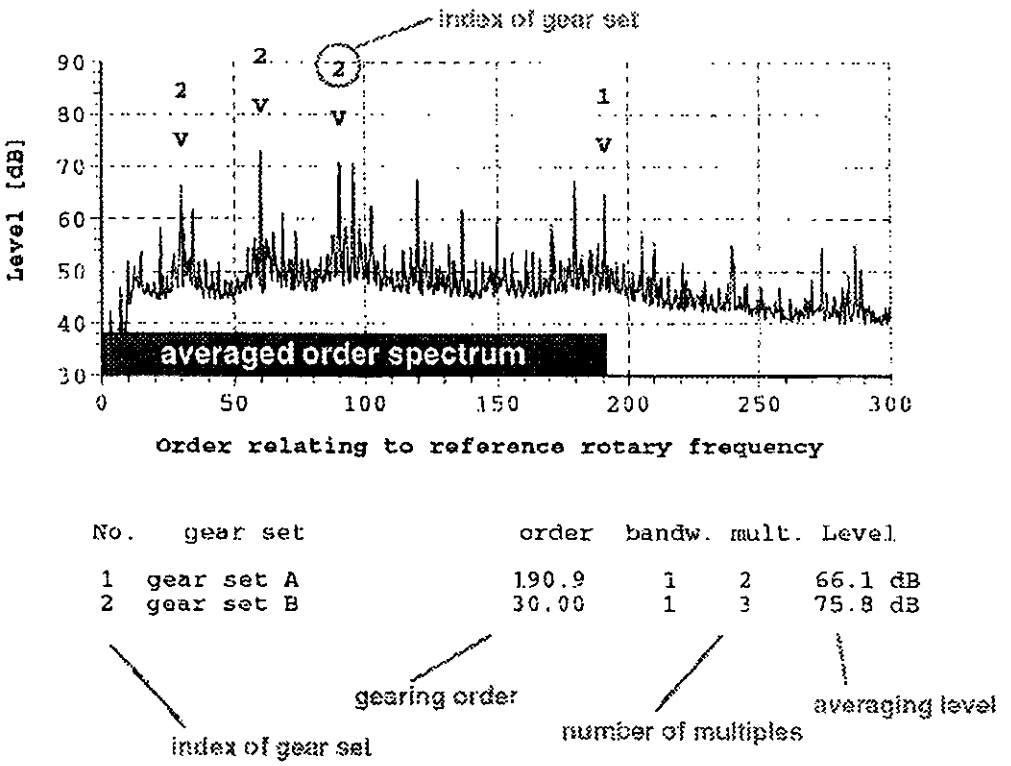


Figure 12: Averaged Order Spectrum

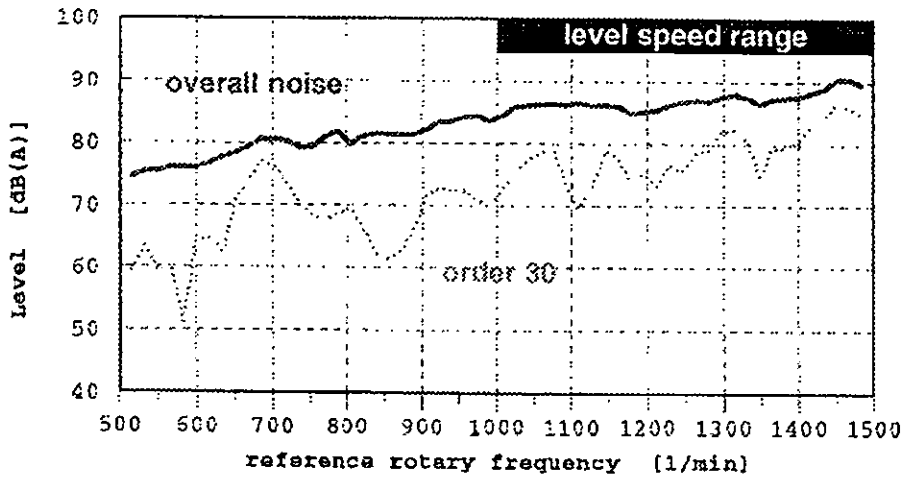


Figure 13: Level Speed Range

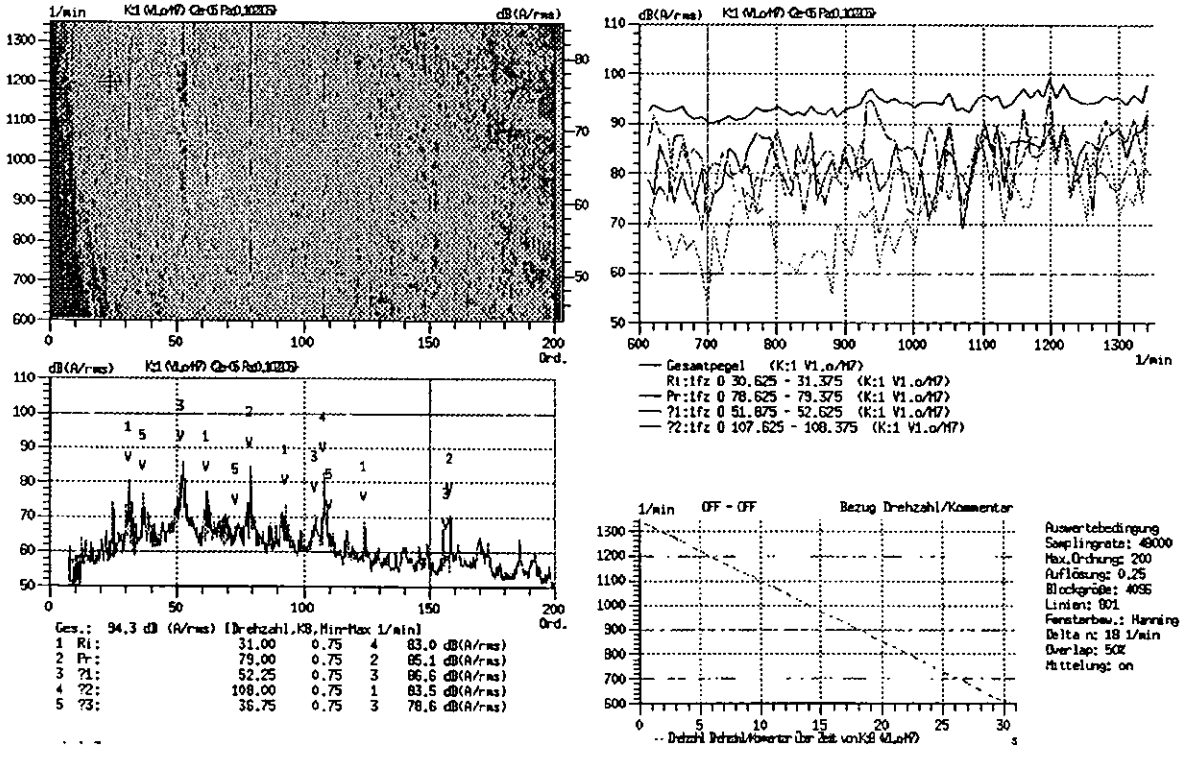
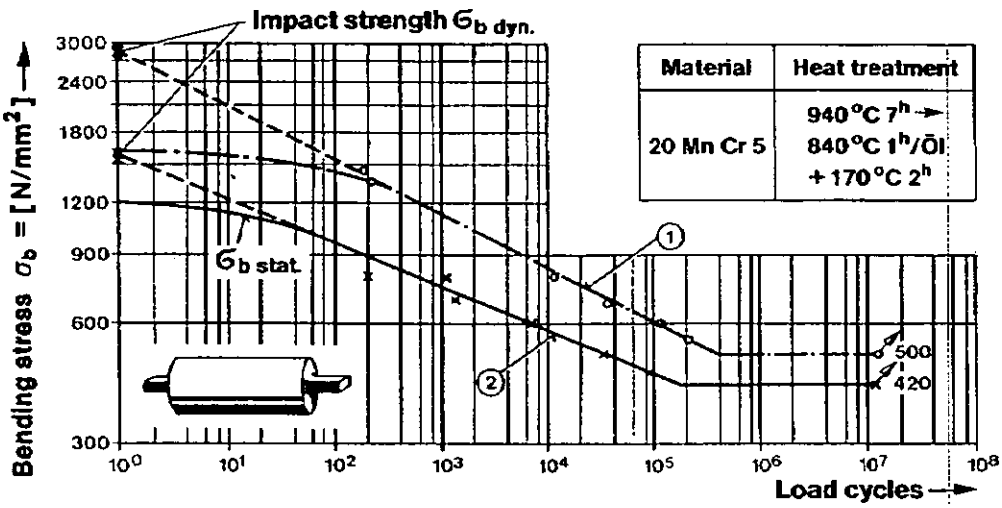


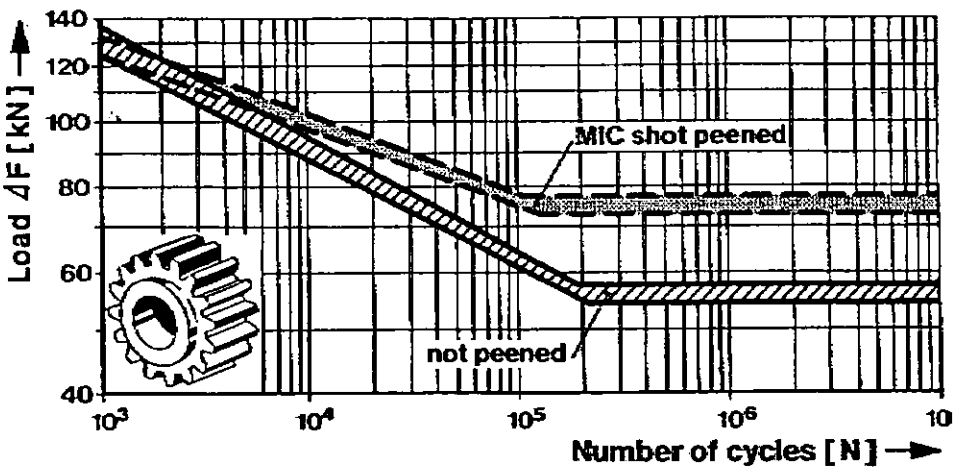
Figure 14: Diagrams overview



Material	Heat treatment
20 Mn Cr 5	940 °C 7 h → 840 °C 1 h/Öl + 170 °C 2 h

Curve	Batch	HRC	Case depth (mm)	R_{mk} (N/mm²)	$\sigma_{b\,stat.}$ (N/mm²)	$\sigma_{b\,dyn.}$ (N/mm²)
①	ductile	60 - 61	1.1	138	1600	2800
②	brittle	59.5 - 61	1.2	144	1250	1650

Figure 15: Influence of heat selection on fatigue limits of 20 Mn CR 5



m = 2.5; R =0; f = 38 Hz; 6 variants

Material	Heat treatment	Surface finish
20 Mn Cr 5	case hardened 940 °C/7 h → 840 °C/1 h/oil → 170 °C/2 h	shot peened A > 0.6

Figure 16: Influence of shot peening (compressive residual stress) on tooth bending strength.

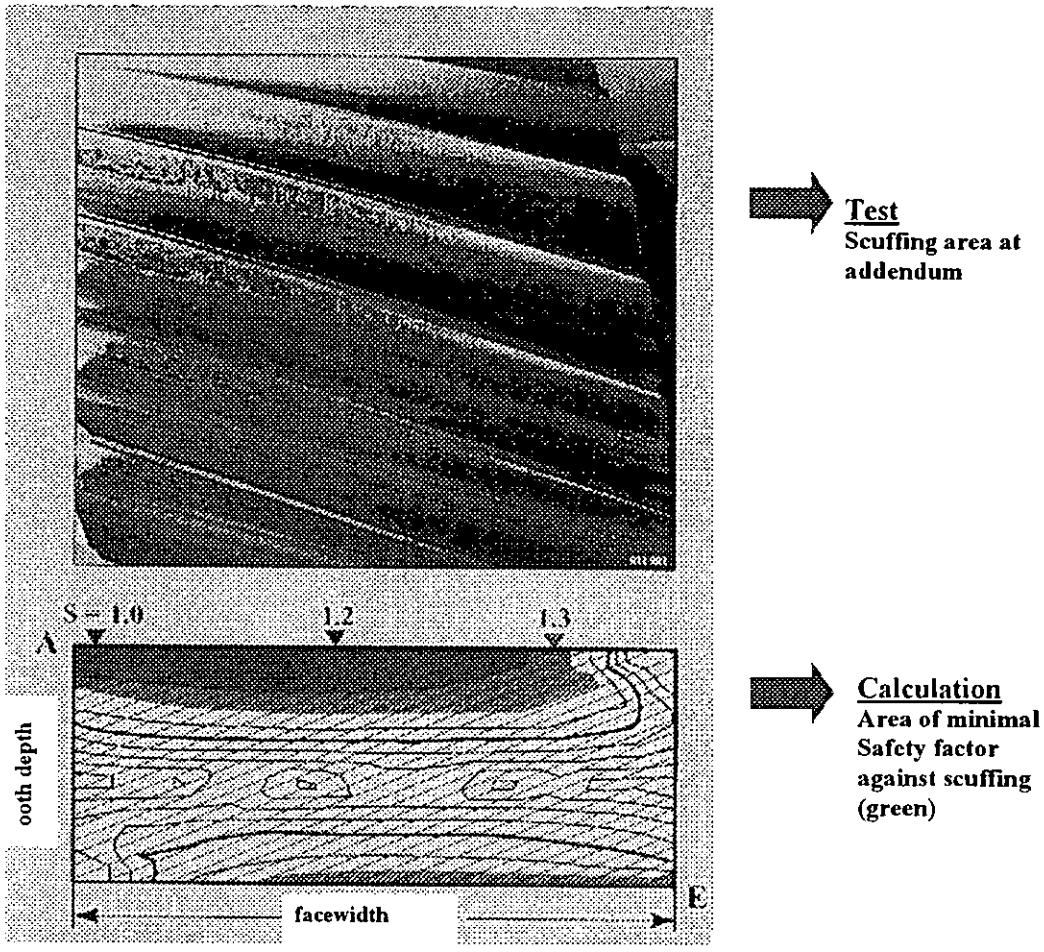


Figure 17: Scuffing on test and calculated safety factor of scuffing

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GAS TURBINES FOR FAST SHIPS

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Abstract

After giving a short overview of the gas turbine development for marine application over the years the working principles are explained in some detail. A basic thermodynamic analysis is given for the simple cycle and regenerative cycle. Based on this analysis the critical parameters for fuel economy and power density are explained as well as the need for optimising the performance and the future potential. Then some practical implications of the installation of gas turbines on board of ships are highlighted such as the sensitivity to ambient conditions and intake and exhaust losses. The engine envelope is discussed with a view on engine/propeller matching. Some peculiarities of gas turbine part load behaviour are touched upon. Last but not least fuels and fuel treatment are discussed as well as air filtration, acoustic measures and design of inlet and exhaust ducts.

Contents

ABSTRACT 1
EDUCATIONAL GOALS 3
EDUCATIONAL GOALS 3
NOMENCLATURE 4
NOMENCLATURE 4
Variables *4*
Greek variables *4*

GAS TURBINES **6**

1 INTRODUCTION 6
2 PERFORMANCE PARAMETERS OF A GAS TURBINE ENGINE 9
3 THERMODYNAMIC ANALYSIS OF THE IDEAL BRAYTON CYCLE 10
3.1 Simple (non regenerative) cycle *10*
3.2 Regenerative cycle *13*
4 BRAYTON CYCLE WITH LOSSES 15
4.1 Simple (non-regenerative) cycle with turbomachinery losses *15*
4.2 Regenerative cycle with turbomachinery losses and heat exchanger effectiveness *17*
5 OPTIMALISATION OF GAS TURBINE PERFORMANCE 19
6 POTENTIAL OF COMPLEX, ADVANCED GAS TURBINE CYCLES 21
7 ENVIRONMENTAL CONDITIONS AND INSTALLATION LOSSES 22
8 OFF-DESIGN CHARACTERISTICS OF A GAS TURBINE 23
8.1 Power/speed characteristic *23*
8.2 Fuel consumption *23*
8.3 Shaft rotation *25*
9 FUELS 25
10 INSTALLATION ON BOARD 25
SOME GAS TURBINE DATA 28
REFERENCES AND OTHER LITERATURE 29
FIGURES 30

Educational goals

The aim of this module on Gas Turbines within the WEGEMT school for Fast Ship Propulsion Plants is that the participants will be able to:

- explain the working principles of the gas turbine,
- understand the most important parameters that make gas turbines light and compact, i.e. the factors determining power density,
- understand how to obtain reasonable efficiency for these light and compact engines, i.e. the factors determining fuel economy,
- explain the limits of the engine characteristics in relation with the characteristic of the propulsor,
- describe the features that can widen the engine characteristic,
- describe some special topics relating to the installation of gas turbines on board ships.

Last but not least this module will:

- provide some factual information on particular gas turbines available in the market

Nomenclature

Variables

Symbol	Name	Unity
afr	air/fuel ratio	-
c_p	specific heat at constant pressure	J/kg/K
h	specific enthalpy	J/kg
LHV	Lower Heat Value (of fuel)	J/kg
$\dot{m}$	mass flow	kg/s
$\dot{m}_a$	air mass flow	kg/s
$\dot{m}_f$	fuel mass flow	kg
$\dot{m}_g$	exhaust gas mass flow	kg/s
n	rotational speed	1/s
n_C	polytropic exponent compression	-
n_T	polytropic exponent expansion	-
P	Power	W
P_C	compressor power	W
P_T	turbine power	W
p	pressure	Pa
$\dot{Q}$	Heat flow	W
$\dot{Q}_{CC}$	Combustion Chamber Heat flow	W
R	Gas constant	J/kg/K
SP	Specific Power	(k)J/kg
s	Specific entropy	J/kg/K
sfc	specific fuel consumption	kg/Ws
T	Temperature	K
T_{max}	maximum (process) temperature	K

Greek variables

Symbol	Name	Unity
Δh	specific enthalpy difference	kJ/kg
ε	non-dimensional power density	-
η	efficiency	-
η_C	polytropic efficiency compressor	-
η_{HE}	Heat Exchanger effectiveness	-
η_T	polytropic efficiency turbine	-
η_{tot}	overall (total) efficiency	-
θ	non-dimensional heat flow	-
κ	isentropic index	-
π	(maximum) pressure ratio	-
π_C	pressure ratio compressor (>1)	-

π_T	pressure ratio turbine (>1)	-
ρ	density	kg/m^3
τ	(maximum) temperature ratio	-
τ_C	temperature ratio compressor (>1)	-
τ_T	temperature ratio turbine (>1)	-
ω	angular velocity	rad/s

GAS TURBINES

1 Introduction

In the beginning of the seventies the gas turbine superseded the steam installation as the propulsion power for naval ships. Apart from the better efficiency (compared to the steam installation at that time) the following advantages paved the way for the gas turbine:

- *Fast starting-up time*: a gas turbine starts within a minute and can almost directly be loaded; this must be compared with the (many) hours required to heat up the boilers of a steam installation.

- *Modular construction*: the gas turbine essentially consists of a box-like unit having exchangeable sub-units; this is quite different from the complicated steam plants with all its equipment connected by piping.

- *Easily automated*: since the advent of gas turbines propulsion can be directly controlled from the bridge.

and last but not least:

- *high reliablity and maintainebility*: compared to steam the gas turbine is less prone to leakages; also the boiler in a steam plant always was a vulnerable and maintenance-critical item.

The breakthrough of gas turbines in the naval field and their application in frigates of many navies around the world has made them "proven at sea". The application in merchant ships, particularly in the larger fast ferries and recently in big cruise liners builds upon the experience gained by the navies in the last thirty years.

The gas turbine, like the steamturbine, consists only of rotating machinery and thus is from another "family" than the reciprocating dieselengine. In the gas turbine air is compressed in a rotating compressor and after combustion at almost constant pressure the hot gas expands in a turbine. Figure 1 gives a 3-D cut away drawing of a "core engine" (i.e. the bare engine without frame, enclosure etc).

The main application of gas turbines of course can be found in the aircraft industry:

- The pure "jet engine": a gas turbine that converts the thermal energy of the hot gases into a jetstream that exits the engine and propels the aircraft.

- The "turboprop engine": a gas turbine that converts all available thermal energy into a rotating movement of a propeller (in case of an aircraft) or the rotor (of an helicopter).

- The "turbofan engine": a mix of the previous types; the thermal energy of the hot gases is partly used to produce a jetstream delivering thrust and partly converted in a turbine that drives the fan (in fact a multi-bladed propeller in a duct).

The gas turbines in the aircraft industry but also the gas turbines for naval application are, thermodynamically speaking, "open cycle" machines and operate according to the "simple cycle" concept (see Figure 2):

- Compression (in one or two stages) from atmospheric pressure to the combustion pressure (in the order of 10 to 30 bar)

- Combustion at constant pressure upto a high temperature (state-of-the-art typically is 1250 C).

- Expansion to atmospheric pressure in a sequence of turbines.

The cycle consisting of adiabatic compression and expansion with a heat input at constant pressure in between sometimes is called the "Brayton cycle".

In basic applications the turbine that drives the compressor is also connected to the load: the gas turbine is of the single shaft type. In that case the load may not force the shaft speed to become too low since the compressor would not be able to deliver pressure anymore. Single shaft gas turbines are used for generator drive in which case the shaft speed is kept constant anyway.

For direct mechanical drive of a propeller shaft in a ship a gas turbine invariably has a separate turbine for the load: the power turbine. Thus it is possible that the load follows the so called "propeller law" which in an ideal case asserts that power is proportional to the cube of shaft speed. The compressor(s) and corresponding turbine(s) with the combustion chamber located in between form a separate unit: the gasgenerator (see Figure 3).

If the gasgenerator is derived from an engine used in the aircraft industry the engine is an "aero-derived gas turbine" which of course is light and often "hightech". The process of adding a power turbine and adapting the burners from kerosene to diesel fuels is called "marinising" the gas turbine (see Figure 4). A marinised gas turbine with its core-engine mounted on a frame and provided with an enclosure is a modular, box-like unit: the "gas turbine module", see Figure 5.

Note

The bigger gas turbines directly designed for stationary applications are called "industrial gas turbines" or "heavy duty gas turbines". Apart from generally being bigger (up to and over 100 MW) these are relatively heavy machines not designed for aeronautical purposes. In the past heavy duty gas turbines used to be technologically more conservative

(lower maximum temperatures) when compared to aero-derived gas turbines but this difference has disappeared in modern engines.

In order to keep the temperature within acceptable limits the gas turbine uses a good amount of air excess when compared to the steam plant (but also in comparison to the diesel engine). The intake and exhaust ducts therefore are of considerable size and require large openings in the main deck of a ship. Moreover at least the intake ducts will need silencers which often are high in the ship. The intake air also has to be clean and in particular the salt concentration must be low: it was the development of special filtersystems that made the use of gas turbines at sea possible at all.

The "simple cycle" has the disadvantage that, although the efficiency is better than for a steam plant, it is inferior when compared to the modern turbocharged diesel engine. As a first improvement it is possible to extract waste heat from the exhaust and feed it back into the cycle to heat up the air, thus diminsihing the heat supplied by the fuel in the combustion chamber. So a heat exchanger (or recuperator) has to be added to the cycle: see Figure 6. It will be clear that this form of "internal waste heat utilisation" has the potential to save fuel.

It is possible to go further: at the moment at several places in the world so called "complex cycle gas turbine" are under development, see Figure 7. The aim is to improve efficiency and also specific power by adding one or more of the following features:

- Cooling of the air between compression stages ("intercooling")
- First stage of heating of the compressed air by means of the hot exhaust gases in a heat exchanger ("recuperation")
- Additional fuel heat input between expansion stages ("reheat")

In combination with variable geometry blades in turbine and/or compressor the result may be a gas turbine with really high performance both at full and part load. The gas turbine as a unit of course will become heavier and possibly (due to the time lag of a heat exchanger) less fast reacting to load steps. An example of such an advanced cycle gas turbine is the Intercooled Regenerative (ICR) gas turbine, see Figure 8.

2 Performance parameters of a gas turbine engine

When evaluating the suitability of a gas turbine for a critical application (such as a fast ship) two technical quantities play a major role: power density and fuel economy. Apart from these the economics of the solution will be determined by the first cost, maintenance cost and the reliability of the engine.

A numerical indicator for power density is the specific power (in kJ/kg) which is equal to power (in kW = kJ/s) divided by air mass flow (in kg/s):

$$SP \overset{def}{=} \frac{P_b}{\dot{m}_{air}}$$

This quantity is a good indicator of weight and volume of the core engine and also of the intake and exhaust ducts. The volume and weight of the core engine however are by far not the installation volume and weight of the total engine since the bed frame and the acoustical enclosure are not included. In particular the volume of smaller gas turbines is dictated by the acoustical enclosure and then the proportionality with the air mass flow is lost rendering the specific power a poor indicator of specific installation volume.

A non-dimensional form of specific power is obtained by dividing power by something that can be associated with the enthalpy of the intake mass flow:

$$\varepsilon \overset{def}{=} \frac{P_b}{\dot{m}_{air} \cdot c_p \cdot T_1}$$

It will be clear that this non-dimensional quantity and the specific power in kJ/kg are related as follows:

$$\varepsilon = \frac{SP}{c_p \cdot T_1}$$

The most direct way to express fuel economy is the specific fuel consumption (normally in g/kWh), i.e fuel flow (in g/hr) divided by power (in kW):

$$sfc \overset{def}{=} \frac{\dot{m}_f}{P_b}$$

The non-dimensional form of specific fuel consumption is the thermal efficiency. Thermal efficiency may be defined as power (in kW divided by heat flow (in kJ/s), the latter being the product of fuel flow (in kg/s) and the Lower Heat Value of the fuel (in kJ/kg):

$$\eta_{tot} \overset{def}{=} \frac{P_b}{\dot{m}_f \cdot LHV}$$

Efficiency and specific fuel consumption then are inversely related:

$$\eta_{tot} = \frac{1}{sfc \cdot LHV}$$

3 Thermodynamic analysis of the ideal Brayton cycle

3.1 Simple (non regenerative) cycle

In this section it will be assumed that the simple cycle gas turbine thermodynamically operates according to the Brayton cycle. Then with some further simplifying assumptions it is possible to calculate specific power and efficiency. In Figure 9 the T-S diagram of the ideal Brayton cycle is shown together with the station numbering that will be adopted.

The power required for the compression stage (1 => 2) in the compressor (index 'C') is proportional to the intake air flow (index 'a') and the increase of the specific enthalpy of the air:

$$P_C = \dot{m}_a \cdot \Delta h_{12}$$

The increase of the enthalpy for a perfect gas can be expressed in the temperature ratio of the air across the compressor. Specific power will be based on intake air mass flow, therefore leave out index 'a' in order to make the notation simpler:

$$P_C = \dot{m} \cdot c_p \cdot (T_2 - T_1) \qquad \Longrightarrow \qquad P_C = \dot{m} \cdot c_p \cdot T_1 \cdot (\tau_c - 1)$$

Assume that compression is adiabatic and reversible. The two assumptions imply an isentropic process for which Poisson's law gives the following relation between the temperature ratio and the pressure ratio:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} \qquad \Longrightarrow \qquad \tau_C = \pi_C^{\frac{\kappa-1}{\kappa}}$$

The isentropic index 'κ' is the ratio between the specific heat at constant pressure and at constant temperature and equals the value 1.4 in case of pure air. Then the power required by the compressor in the ideal case becomes:

$$P_{C,ideal} = \dot{m} \cdot c_p \cdot T_1 \cdot \left(\pi_C^{\frac{\kappa-1}{\kappa}} - 1\right)$$

The power developed during expansion of the hot gases (3 => 4) in the turbine (index 'T') is proportional to the exhaust gas flow (index 'g') and the decrease of the specific enthalpy of the gases:

$$P_T = \dot{m}_g \cdot \Delta h_{34}$$

Neglect the increase of the mass flow as a result of the addition of the small amount of fuel:

$$\dot{m}_g = \dot{m}_a = \dot{m}$$

Also for the exhaust gases assume perfect gas behaviour:

$$P_T = \dot{m} \cdot c_p \cdot (T_3 - T_4) \qquad \Longrightarrow \qquad P_T = \dot{m} \cdot c_p \cdot T_3 \cdot \left(1 - \frac{1}{\tau_T}\right)$$

The ideal expansion process is isentropic for which Poisson asserts:

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\kappa-1}{\kappa}} \qquad \Longrightarrow \qquad \tau_T = \pi_T^{\frac{\kappa-1}{\kappa}}$$

Then the delivered power of the turbine in the ideal case is:

$$P_{T,ideal} = \dot{m} \cdot c_p \cdot T_3 \cdot \left(1 - \frac{1}{\pi_T^{\frac{\kappa-1}{\kappa}}}\right)$$

The specific power of the cycle follows from the difference between the power delivered by the turbine and the power absorbed by the compressor, both per unit mass flow:

$$SP = \frac{P_T - P_C}{\dot{m}} = c_p \cdot T_3 \cdot \left(1 - \frac{1}{\pi_T^{\frac{\kappa-1}{\kappa}}}\right) - c_p \cdot T_1 \cdot \left(\pi_C^{\frac{\kappa-1}{\kappa}} - 1\right)$$

For an ideal case the pressure losses in the intake and the exhaust as well as in the combustion chamber are neglected. Then the pressure ratio across the compressor is equal to that of the compressor; introduce the pressure ratio of the process as a whole:

$$\pi = \pi_C = \frac{p_2}{p_1} = \frac{p_3}{p_4} = \pi_T$$

Also define the maximum temperature ratio of the process:

$$\tau = \frac{T_3}{T_1}$$

Then specific power ('SP') can be expressed in these important parameters and written in non-dimensional form with the "intake enthalpy":

$$\varepsilon = \frac{SP}{c_p \cdot T_1} = \tau \cdot \left(1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}}\right) - \left(\pi^{\frac{\kappa-1}{\kappa}} - 1\right)$$

Note:

The specific heat of air is $c_p \cong 1.0$ kJ/kg/K *and the inlet temperatureof the gas turbine is of the order* $T_1 \cong 300$ K. *Thus the intake enthalpy is approximately* $c_p . T_1 \cong 300$ kJ/kg

This expression can also be written as the multiplication of two factors:

$$\boxed{\varepsilon = \left(\tau - \pi^{\frac{\kappa-1}{\kappa}}\right) \cdot \left(1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}}\right)}$$

Now it will be clear that the specific power becomes zero for two specific values of the pressure ratio:

$$\pi = 1 \qquad \text{and:} \qquad \pi = \tau^{\frac{\kappa}{\kappa-1}}$$

Question

Which are the limit cases in the T-s diagram that correspond to these two values?

Considerations of continuity require that a maximum for the specific power will occur somewhere in between the two values. Taking the derivative with respect to pressure ratio and putting this to zero yields:

$$\pi^{\frac{\kappa-1}{\kappa}} = \sqrt{\tau}$$

Refer to Figure 11 where the specific power is shown as function of pressure ratio, given a certain value of the temperature ratio. Note that increasing the temperature ratio always is favourable for the specific power. In contrast with the pressure ratio there is no extreme value for the temperature ratio

Exercise

Carry out the differentiation and verify the results above. Also prove that the maximum value of the specific power (in this ideal case) occurs for $T_4 = T_2$, *i.e. when the temperature after expansion is equal to the temperature after compression.*

Combustion takes place at constant pressure and the heat release to the process per unit time is approximately (compare the T-s diagram, Figure 10 where reversible heat is equal to an area below the process line):

$$\dot{Q}_{CC} = \dot{m}_f \cdot LHV = \dot{m} \cdot \Delta h_{23} = \dot{m} \cdot c_p \cdot \left(T_3 - T_2\right)$$

Again the mass increase due to fuel addition is neglected as well as the change of specific heat during combustion where air and fuel combine to produce a mixture of stoichiometric gas and air.

Making use of the maximum temperature ratio and pressure ratio as introduced the heat input can be written in non-dimensional form:

$$\theta \overset{def}{=} \frac{\dot{Q}_{CC}}{\dot{m} \cdot c_p \cdot T_1} = \left(\frac{T_3}{T_1} - \frac{T_2}{T_1}\right) = \left(\tau - \pi^{\frac{\kappa-1}{\kappa}}\right)$$

The total efficiency of the ideal simple cycle then is:

$$\eta_{tot} \overset{def}{=} \frac{P_b}{\dot{Q}_{CC}} = \frac{\varepsilon}{\theta} = \frac{\left(\tau - \pi^{\frac{\kappa-1}{\kappa}}\right) \cdot \left(1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}}\right)}{\tau - \pi^{\frac{\kappa-1}{\kappa}}}$$

Since there is a common factor in the power and the heat flow, this expression in this ideal case becomes very simple:

$$\boxed{\eta_{tot} = 1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}}}$$

The remarkable fact is that the efficiency is independent of temperature ratio. Also the efficiency increases with pressure ratio, see Figure 11.

Lateron it will become clear that both properties only are valid in case of ideal compressor and turbine! Turbomachinery losses will change the picture radically, not only quantatively but also qualitatively.

Question

What is the value of the efficiency for the limit case where the specific power is zero? What is the thermodynamic significance of this result?

3.2 Regenerative cycle

Adding a heat exchanger that transfers the heat of the exhaust to the first part of the heating process before the combustion chamber (see Figure 12) does not alter the expressions for the power of the compressor and turbine. Therefore the expression for the specific power of the regenerative cycle remains unaltered when compared to the simple cycle. This includes the occurrence of an optimum value at a certain pressure ratio for given temperature ratio.

The heat exchange of course has an influence on the heat addition to the cycle. For an ideal counter flow heat exchanger with infinite area the air after the compressor can be heated up to the gas exhaust temperature after the last turbine, refer to Figure 12:

$$\dot{Q}_{CC} = \dot{m}_f \cdot LCV = \dot{m} \cdot c_p \cdot \left[\left(T_3 - T_2 \right) - \left(T_4 - T_2 \right) \right]$$

The heat input then can be written in non-dimensional form as:

$$\theta \overset{def}{=} \frac{\dot{Q}_{CC}}{\dot{m} \cdot c_p \cdot T_1} = \frac{T_3}{T_1} - \frac{T_4}{T_1} = \frac{T_3}{T_1} - \frac{T_4}{T_3} \cdot \frac{T_3}{T_1} = \tau \cdot \left(1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}} \right)$$

Then the efficiency of the ideal regenerative cycle becomes:

$$\eta_{tot} \overset{def}{=} \frac{P_b}{\dot{Q}_{CC}} = \frac{\varepsilon}{\theta} = \frac{\left(\tau - \pi^{\frac{\kappa-1}{\kappa}} \right) \cdot \left(1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}} \right)}{\tau \cdot \left(1 - \frac{1}{\pi^{\frac{\kappa-1}{\kappa}}} \right)} = \frac{\left(\tau - \pi^{\frac{\kappa-1}{\kappa}} \right)}{\tau}$$

This can be rearranged in a somewhat neater form:

$$\boxed{\eta_{tot} = 1 - \frac{\pi^{\frac{\kappa-1}{\kappa}}}{\tau}}$$

The efficiency of the regenerative cycle thus is dependent on pressure ratio *and* temperature ratio. Moreover the efficiency now *decreases* with presure ratio! Refer to Figure 11 to illustrate this point. Now, contrary to the simple non-regenerative cycle, one must opt for a cycle with a *low* pressure ratio. The physical background is that at a low pressure ratio the exhaust temperature is high which is favourable for (internal) use of waste heat. On the other hand the the temperature after compression is low; this offers ample opportunity for heat exchange between the two flows.

4 Brayton cycle with losses

4.1 Simple (non-regenerative) cycle with turbomachinery losses

In a real cycle there are two major sources for losses: the compressor and turbine are not ideal and furthermore there are pressure losses in the flow path: see Figure 13 for an illustration of these effects in the T-S diagram.

The effect of turbomachinery efficiencies on the heat input to the cycle and heat rejection to the atmosphere and thus on engine performance has a major influence. This will be looked into while the pressure losses will remain to be neglected.

In case the compresion process is still adiabatic but not reversible anymore the entropy must increase. As a consequnce in Poisson's law the isentropic index 'κ' must be replaced by a polytropic index 'n_C':

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n_C-1}{n_C}} \qquad \Longrightarrow \qquad \tau_C = \pi_C^{\frac{n_C-1}{n_C}}$$

The new exponent is calculated als follows:

$$\frac{n_C - 1}{n_C} = \frac{1}{\eta_C} \cdot \frac{\kappa - 1}{\kappa}$$

where 'η_C' is the *polytropic* efficiency of the compression process. For 'η_C<1' the resulting exponent in Poisson's law will increase. Therefore the temperature rise for a given pressure ratio will be higher compared to the ideal case (also compare Figure 13). The compression work will be higher as a result. Using the *polytropic exponent* (in which the *polytropic efficiency* is hidden) the work required by the compressor becomes:

$$P_C = \dot{m} \cdot c_p \cdot T_1 \cdot \left(\pi_C^{\frac{n_C-1}{n_C}} - 1\right)$$

For the expansion process in the turbine in a like-wise manner:

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{n_T-1}{n_T}} \qquad \Longrightarrow \qquad \tau_T = \pi_T^{\frac{n_T-1}{n_T}}$$

The new exponent now is related to the isentropic case by:

$$\frac{n_T - 1}{n_T} = \eta_T \cdot \frac{\kappa - 1}{\kappa}$$

where 'η_T' is the polytropic efficiency of the expansion process. For 'η_T<1' the exponent in Poisson's law is smaller than for the isentropic case. Then the

temperature drop for a given pressure ratio is less (again see Figure 13). With the *polytropic efficiency* (in which the *polytropic efficiency* is hidden) the power delivered by the turbine is:

$$P_T = \dot{m} \cdot c_p \cdot T_3 \cdot \left(1 - \frac{1}{\pi_T^{\frac{n_T - 1}{n_T}}}\right)$$

Note

Be aware of the subtle differences between the expressions for power of compressor and turbine; the reason is that in this manner for both machines the pressure ratio >1 and the efficiency <1.

Now for the specific power of the non-ideal simple cycle gas turbine:

$$SP = \frac{P_T - P_C}{\dot{m}} = c_p \cdot T_3 \cdot \left(1 - \frac{1}{\pi_T^{\frac{n_T - 1}{n_T}}}\right) - c_p \cdot T_1 \cdot \left(\pi_C^{\frac{n_C - 1}{n_C}} - 1\right)$$

As argued above the pressure losses will still be neglected. Then the pressure ratio of the cycle is as before:

$$\pi = \pi_C = \frac{p_2}{p_1} = \frac{p_3}{p_4} = \pi_T$$

Also introduce again the maximum temperature ratio of the cycle:

$$\tau = \frac{T_3}{T_1}$$

Then the specific power expressed in these two main parameters can be written in non-dimensional form using the "intake enthalpy":

$$\varepsilon = \frac{SP}{c_p \cdot T_1} = \tau \cdot \left(1 - \frac{1}{\pi^{\frac{n_T - 1}{n_T}}}\right) - \left(\pi^{\frac{n_C - 1}{n_C}} - 1\right)$$

Unlike the ideal case this cannot be written as a product of two factors. Nevertheless it can be shown easily that there is, apart for the value 'π=1', another zero point for a large pressure ratio and that there is an optimum value again, see Figure 14.

The heat input from the fuel to the cycle again is:

$$\dot{Q}_{CC} = \dot{m}_f \cdot LCV = \dot{m} \cdot c_p \cdot (T_3 - T_2)$$

which can be expressed in the two main cycle parameters and written in non-dimensional form as follows:

$$\theta \overset{def}{=} \frac{\dot{Q}_{CC}}{\dot{m} \cdot c_p \cdot T_1} = \left(\frac{T_3}{T_1} - \frac{T_2}{T_1} \right) = \left(\tau - \pi^{\frac{n_C - 1}{n_C}} \right)$$

Then the overall process efficiency of the non-regenerative cycle with turbomachinery losses is:

$$\eta_{tot} \overset{def}{=} \frac{P_b}{\dot{Q}_{CC}} = \frac{\varepsilon}{\theta} = \frac{\tau \cdot \left(1 - \frac{1}{\pi^{\frac{n_T - 1}{n_T}}} \right) - \left(\pi^{\frac{n_C - 1}{n_C}} - 1 \right)}{\tau - \pi^{\frac{n_C - 1}{n_C}}}$$

Now the efficiency, like the specific power, generally has two zero points with respect to pressure ratio. Also there is, for any maximum temperature ratio, a maximum value for the efficiency at a certain pressure ratio, see Figure 14. As mentioned before this is fundamentally different from the ideal case where the efficiency was independent of temperature ratio and increasing monotonuously with pressure.

4.2 Regenerative cycle with turbomachinery losses and heat exchanger effectiveness

Addition of a heat exchanger again has no influence on the power of the compressor and turbine (since pressure losses are still neglected!). There is however a difference for the required heat of the fuel, see Figure 15. For a heat exchanger with finite dimensions the temperature of the heated air will be less than the exhaust temperature:

$$\dot{Q}_{CC} = \dot{m}_f \cdot LCV = \dot{m} \cdot c_p \cdot \left[\left(T_3 - T_2 \right) - \eta_{HE} \cdot \left(T_4 - T_2 \right) \right]$$

Note that 'η_{HE}' strictly speaking is not an efficiency but an effectiveness: the factor is the ratio of the real temparature increase of the air (T_2-T_2) compared to the maximum possible value (T_4-T_2).

The heat input in non-dimensional form is:

$$\theta \overset{def}{=} \frac{\dot{Q}_{CC}}{\dot{m} \cdot c_p \cdot T_1} = \left(\frac{T_3}{T_1} - \frac{T_2}{T_1} \right) - \eta_{HE} \cdot \left(\frac{T_4}{T_1} - \frac{T_2}{T_1} \right)$$

$$= \left(\frac{T_3}{T_1} - \frac{T_2}{T_1} \right) - \eta_{HE} \cdot \left(\frac{T_4}{T_3} \cdot \frac{T_3}{T_1} - \frac{T_2}{T_1} \right)$$

Now with the definitions of our main cycle parameters, maximum pressure ratio and temperature ratio:

$$\theta \overset{def}{=} \frac{\dot{Q}_{CC}}{\dot{m} \cdot c_p \cdot T_1} = \left(\tau - \pi^{\frac{n_C - 1}{n_C}} \right) - \eta_{HE} \cdot \left(\frac{\tau}{\pi^{\frac{n_T - 1}{n_T}}} - \pi^{\frac{n_C - 1}{n_C}} \right)$$

$$= \tau \cdot \left(1 - \frac{\eta_{HE}}{\pi^{\frac{n_T - 1}{n_T}}} \right) - \left(1 - \eta_{HE} \right) \cdot \pi^{\frac{n_C - 1}{n_C}}$$

Then the overall efficiency of non-ideal regenerative cycle:

$$\eta_{tot} = \frac{\tau \cdot \left(1 - \frac{1}{\pi^{\frac{n_T - 1}{n_T}}} \right) - \left(\pi^{\frac{n_C - 1}{n_C}} - 1 \right)}{\tau \cdot \left(1 - \frac{\eta_{HE}}{\pi^{\frac{n_T - 1}{n_T}}} \right) - \left(1 - \eta_{HE} \right) \cdot \pi^{\frac{n_C - 1}{n_C}}}$$

This is a complicated expression which is shown as function of pressure ratio in Figure 14.

5 Optimalisation of gas turbine performance

From the considerations in the previous section it can be concluded that the main parameters that influence the specific power and overall efficiency of a gas turbine are:

- The maximum temperature ratio ($\tau = T_3/T_1$), i.e. the temperature before inlet of the turbine(s) relative to the temperature of the environment

- The maximum pressure ratio ($\pi = p_2/p_1$), i.e. the pressure after the compressor(s) relative to the environmental pressure

Furthermore the turbomachinery efficiencies have a major impact: for good performance they must be as high as possible. This also applies to the heat exchanger effectiveness. Although not shown in the previous simplified treatment, pressure losses in the flow path deteroriate both specific power and efficiency and must be kept also as low as possible.

For the maximum temperature ratio we have seen that in general the highest value will give the best performance in terms of specific power and efficiency. The thermal load of the turbine blading forms the limit and a maximum temperature ratio of 5 already requires very advanced blade cooling techniques. This also explains the amount of research that has been put into new turbine blade materials and the development of advanced blade cooling by bleed air taken from the compressor.

The maximum pressure ratio of a gas turbine is a different story however: here the rule "the higher the better" does not apply. In the foregoing analysis it became clear that the pressure ratio must be optimised; the optimum however is quite different for maximum specific work and maximum overall efficiency!

This could clearly be seen inFigure 14. For the simple cycle the optimum value for the efficiency is reached for a pressure ratio between 20 and 40 while the optimum value for specific power (i.e. the "smallest" engine) requires a pressure ratio between 10 and 15.

In order to understand why optimising with respect to efficiency is different from optimising to specific power the cycle for the two optima is drawn in a T-s diagram, see Figure 16.

In the cycle with optimum efficiency the higher pressure ratio causes the combustion heat to be delivered at a higher temperature level which of course (Carnot!) is much more favourable. It is the turbomachinery losses that prevents the optimum to lie at even higher pressure ratio.

There is an elegant way to put both optima in one graph which a.o. can be found in [Wilson, 1984]. Efficiency and (non-dimensional) specific power are put on the vertical and horizontal axis respectively, see Figure 17.

In this trade-off picture lines for several constant maximum temperature ratios are entered whereby the pressure ratio is a running parameter along these (strongly curved) lines. A few lines of constant pressure ratio are entered as well thus completing the grid. The reader may verify that the information contained in Figure 14 for the simple cycle is directly transformed into Figure 17. The two optima now can be seen directly as the horizontal tangent (optimum for the efficiency) and the vertical tangent (optimum for specific work) of the constant temperature lines. In particular the plot shows that a trade-off must be made between efficiency and specific power, i.e. between a gas turbine with low specific fuel consumption and a small engine.

Recuperation of the exhaust heat by addition of a heat exchanger changes the relative position of the optima completely: the optimum efficiency now is reached at much lower pressure ratios between 3 and 8, again go back to Figure 14. This is a favourable characteristic of the regenerative gas turbine since it means simple turbomachinery and thus a cheap gas turbine. With a radial compressor and turbine even single stage is possible. But the heat exchanger on the other hand of course adds to complexity.

The position of the optimum for the specific power has not changed (still at pressure ratios between 10 and 15) although for a real case specific power of the regenerative gas turbine will be slightly less than that of the simple cycle due to the pressure loss of the heat exchangers (both at air and gas side).

Note

The real specific installation weight of the regenerative gas turbine of course will be greater due to the weight of the heat exchanger itself.

The explanation of the reversal of the optima becomes clear when looking at the T-s diagram: seeFigure 18. The heat exchanger only is effective in case of a considerable temperature difference between the exhaust temperature of the turbine(s) and the exit temperature of the compressor(s). This is only possible when the the pressure ratio of the engine is small and the T-s diagram is shallow.

Again the result can be presented in one trade-off graph: see Figure 19. In this figure the direction for which the pressure ratio increases along the curves has reversed when compared to Figure 17. This effect is shown for just one temperature ratio in Figure 20.

6 Potential of complex, advanced gas turbine cycles

The techniques as outlined before were used in [Stapersma, 1999] to investigate all sorts of advanced gas turbine cycles with features such as intercooling between compressor stages, recuperation of heat from the exhaust and reheat between turbine stages, see Figure 7. An overview of the potential of several complex cycle gas turbines is brought together in the trade-off plot of Figure 21.

The analysis in [Stapersma, 1999] goes somewhat further than the simplified analysis given in the previous sections. First of all pressure losses in the flow path are included. Furthermore the polytropic efficiencies of compressors and turbines are assumed to be a (decreasing) function of pressure ratio. Last but not least the heat addition in the combustion chamber is calculated in a more precise way allowing for the change in composition and associated change in specific heat. Also the mass increase due to the addition of the fuel is taken into account.

From this comparison it can be concluded that if one is not prepared to go for the complexity of a second combustion chamber (RH = reheat) the cycle with intercooling between compressor stages (IC = Intercooling) *combined* with regeneration of heat from the exhaust (HE = Heat exchanger) offers the best potential. This is the "Inter-Cooled Recuperated" cycle (ICR) for which the flow scheme and T-s diagram are shown in Figure 22 and Figure 23.

The potential of this cycle is shown in Figure 24. As a reference is included an ICR gas turbine that is in an advanced stage of development (Rolls Royce WR 21). From the figure it can be concluded that the pressure ratio is chosen between the values for optimum specific power and optimum overall efficiency, but closer to the latter (the selected pressure ratio in fact is 11.5 which is rather low thanks to the adoption of the heat exchanger). A cross section of this gas turbine was already presented in the introduction, refer toFigure 8. For the intercooler the airflow must deviate from the axial gas path through the gas turbine. The heat exchanger has a by-pass to be used in case of problems and which enables the gas turbine to be run up to normal power (but at higher fuel consumption of course).

7 Environmental conditions and installation losses

At higher ambient temperature the amount of fuel of a gas turbine must be reduced in order to keep the turbine inlet temperature below the maximum allowable value. As a result the power output of a gas turbine is greatly reduced in (sub)tropical conditions, see Figure 25. The efficiency also decreases and specific fuel consumption goes up.

The physical reason for this tendency is two-fold:

- Due to a lower density at intake the density throughout the engine is lower and the mass flow decreases. When fuel would not be reduced the temperature rise in the combustion chamber would increase proportionally to the density decrease.

- A higher inlet temperature causes all (absolute) temperatures to be higher (with the same factor if all other conditions would be the same). So fuel must be decreased also to compensate this effect

Ideally fuel would be reduced in order to maintain the temperature at the inlet of the first turbine constant. This temperature normally cannot be measured however and the practical solution is to schedule fuel based on *power* turbine entry temperature (which is an observable quantity).

At lower ambient temperatures power may be reduced in order to keep the gasgenerator speed within tolerable limits.

A gas turbine also is relatively sensitive to pressure at intake and exhaust.

In case of ambient pressure variations the pressure at intake and exhaust vary in the same manner. The influence on power and specific fuel consumption is comparable to the ambient temperature in that both a decrease of ambient pressure and an increase of ambient temperature causes the intake density to be lower than normal and a limitation of fuel is necessary to prevent too high temperatures at the critical blading.

Important also are the pressure losses in intake and exhaust ducts. Sometimes gas turbine performance is presented as "no loss". Then the values will be too optimistic and not representative for the condition "as installed" on board. It is good practice to compare gas turbines on the basis of their performance with assumed realistic intake and exhaust losses, for instance 100 mmWG (1 kPa) at the inlet and 150 mmWG (1.5 kPa) at the outlet.

Some data for installation losses:

at 1% intake loss (air filters, intake duct and silencer):

- 2% decrease of power
- 1% increase of specific fuel consumption

at 1% exhaust loss (exhaust duct, silencer and exit velocity loss):

- 1% decrease of power
- 1% increase of specific fuel consumption

8 Off-design characteristics of a gas turbine

8.1 Power/speed characteristic

A typical output power/speed characteristic of an existing (albeit outdated) gas turbine (Rolls Royce Spey SM1A) is included: see Figure 26. From this it will be clear that a gas turbine with free power turbine has a very wide characteristic, i.e. it is possible to develop high power even at low speeds. It is possible to follow a propeller curve and in each point there is sufficient power reserve for accelerating the ship. Also there is room for increased resistance in which case the propeller curve shifts to higher values.

In fact the limit curve of maximum fuel seems a parabolic curve with a flat top near nominal speed, the latter causing almost constant power behaviour around the nominal point. The parabolic shape of the limiting envelope for a gas turbine with free power turbine is almost exact and can be derived from first principles with the assumption that the power turbine behaves according to Euler's pump curve and that the mass flow of the gas generator is proportional to fuel flow. As a consequence the turbine torque almost linearly increases at decreasing output speed.

8.2 Fuel consumption

The fuel consumption (as shown along the lines in Figure 26) can be divided by power to obtain specific fuel consumption (sfc). Figure 27 gives values for sfc along the propeller curve (for other trajectories of the engine characteristic the curve usually lies somewhat higher)

From Figure 27 it is clear that sfc of a gas turbine increases sharply at part load. The fundamental cause for this state of affairs is that the air consumption of a gas turbine decreases much slower than the fuel consumption as can been concluded from the sharp rising air excess ratio inFigure 28. The air excess acts as cooling of the gases and therefore at part load the gas turbine cannot maintain the turbine inlet temperature for which it was designed. From the previous thermodynamic discussions the deteriorating effect on the overall engine efficiency will be clear.

A method to maintain turbine inlet temperature is to control the amount of air through the engine by means of variable geometry of the stator blading somewhere in the turbine. By decreasing the area the air flow will be reduced and the inlet temperature can be scheduled at any prescribed values. The result will be a flat sfc curve. The ICR gas turbine referred to in the previous section apart from an advanced thermodynamic cycle also has variable turbine geometry. This engine therefore can maintain the low specific fuel consumption it already has in the nominal point down to 40% load, see Figure 29. The flatter air excess ratio which causes this is shown in Figure 30.

8.3 Shaft rotation

Gas turbines in general are not reversible. The consequence of this is that for reversing a ship with a mechanically driven propeller ("direct drive" as opposed to "electrical drive") a solution must be found either in the gearbox (reversible gearing with couplings) or in the propeller (controllable pitch propeller or waterjet with thrust reverser).

For some gas turbines the rotation can be specified for clockwise and anti-clockwise direction (i.e. the blading of the power turbine is produced in two - symmetrical - versions). If this is not the case a solution must be found in case of the twin-shaft ship where both shafts normally have opposing direction of rotation (either inward or outward). The solution could be a different first reduction gear in the gas turbine model; another solution is an extra 1:1 pair of gears in one of the ship's main gearboxes.

9 Fuels

Aero-derived gas turbines can only run on light destillate fuels:

- Light Diesel Fuel Oil, LDF (or sometimes LDO); ISO designation: DMA; density: 840 to 890 kg/m^3; viscosity at 40°C 1.5 to 5 cSt. This fuel corresponds to NATO F 76.

Moreover the fuel must be free of water. To achieve this not only fuel centrifuges are used but normally combined filter/waterseparators are put in series. Since fuel often is used as fluid in the gas turbine control system (an heritage from its aeronautical past) the pressure at the inlet of the gas turbine sometimes has to be controlled within certain limits.

Industrial or "heavy duty" gas turbines sometimes are capable of burning somewhat heavier (and cheaper) grades of fuel, but they still need destillate fuels:

- Marine Diesel Oil, MDO; ISO designation DMB; density 860 to 900 kg/m^3; viscosity 5 to 11 cSt at 40°C.

Some manufacturers claim that their gas turbines can handle "Blended Marine Diesel Fuel", ISO designation DMC. These are destillate fuels blended with not more than 20% residual fuel.

10 Installation on board

The radiated airborne noise of the core engine is very high with maximum levels lying in the higher frequency bands for which the human ear is very sensitive. A gas turbine on board a ship therefore requires an acoustical enclosure which, together with the "bedplate" forms the total gas turbine "module", see Figure 31 and Figure 32. The structure borne noise of a gas turbine is low and a gas turbine for merchant navy application can be rigidly mounted in the ship. In case of more stringent requirements such as for

instance in naval ships the bed plate is put on (rubber) mounts; these serve the double function of noise attenuation and shock protection.

Apart from its function as acoustical enclosure the module also offers fire protection: it is "an engine room within the engine room" and it has its own independent fire-fighting system. Also the module has its own ventilation. For this purpose clean air is taken from the gas turbine intake air system (i.e. from behind the three stage filter system). Discharge of ventilation air normally is to the exhaust duct: the underpressure in the exhaust due the high velocity makes a fan in the system superfluous.

The air at the inlet of a gas turbine must be very clean. A gas turbine in particular is sensitive to salt ingress. Normally a three stage filter system is installed in the superstructure or elsewhere in the ship. In the 1st stage of a so-called inertial filter the air is forced to take a curved path. The water droplets cannot follow this and become attached to the wall where they are collected at the bottom. In a 2nd stage the remaining finer spray is forced to form greater droplets (coalescer) which are removed in another inertial 3rd stage of the filter system. Since these filters operate at relatively low speeds they require a considerable cross sectional area to handle the big air flow of a gas turbine.

Figure 33 gives a picture of a 3-stage air filter system

The high frequency noise from the inlet of the compressor must be isolated from the deck area by a silencer in the inlet duct. Depending on the noise requirements the (low frequency) noise of the exhaust gas stream is often acceptable if a certain length of duct is present and the position of the outlet is high.

The required volume for the inlet and exhaust ducts ("up and downtakes") of The air at the inlet of a gas turbine must be very clean. A gas turbine in particular is sensitive to salt ingress. Normally a three stage filter system is installed in the superstructure or elsewhere in the ship. In the 1st stage of a so-called inertial filter the air is forced to take a curved path. The water droplets cannot follow this and become attached to the wall where they are collected at the bottom. In a 2nd stage the remaining finer spray is forced to form greater droplets (coalescer) which are removed in another inertial 3rd stage of the filter system. Since these filters operate at relatively low speeds they require a considerable cross sectional area to handle the big air flow of a gas a gas turbine installation is quite large. Also the detailed flow design of bends and area changes requires careful engineering. In particular the inlet duct is sensitive in this respect since the compressor cannot handle a distorted flow. Furthermore the selection of flow velocities in the inlet and exhaust ducts always is a compromise between small dimensions and low intake and exhaust losses.

A view of the up-and downtakes for a naval ship is given in Figure 34. An example of an installation of a fast ferry with waterjet propulsion is presented in Figure 35.

Some gas turbine data

The number of manufacturers for "aero derived" marine gas turbines is limited: Rolls Royce and General Electric. For an overview of some data see the following tables.

Rolls Royce		Tyne RM1C	Allison 601 KF9	Allison 601 KF11	Spey SM1C	Olympus TM3B	WR 21
Brake power (no loss)	kW	4000	6500	7800	19500	19200	25240 max rating
Output speed	rpm	14000			5500	5600	3600
Spec. fuel cons. (no loss)	g/kWh	286	≈ 250	≈ 250	230	290	200
Air mass flow	kg/s				66		72
Specific power	kJ/kg				295		350
Spec. Air Cons.	kg/kWh				12,2		10.3
Weight	kg	14000			25700	30800	46000
Specific weight	kg/kW	3,5			1,4	1,6	1.8
Length	m	5,6	≈ 5.0	≈ 5.0	7,5	9,2	8,0
Width	m	2,1	≈ 1.9	≈ 1.9	2,3	2,6	2,7
Height	m	2,6	≈ 2.5	≈ 2.5	3,1	3,7	4.8
Volume	m^3	30,9	≈ 29	≈ 29	53,1	88,5	103.6
Specific volume	dm^3/kW	7.7	≈ 4.5	≈ 4.5	2,7	4,6	4.1
Remarks		not available anymore	Allison engine	Allison engine	SM1A not available	Not available anymore	ICR!

Table 1 Rolls Royce aero-derived gas turbines for marine application

General Electric		LM 500	LM 1600	LM 2500
Brake power (no loss)	kW	4400	14900	24000
Output speed	rpm	7000	7000	3600
Spec. fuel cons. (no loss)	g/kWh	266	233	238
Air mass flow	kg/s	16,3	47	68
Specific power	kJ/kg	270	317	353
Spec. Air Cons.	kg/kWh	13,3	11,4	10,2
Weight	kg		15440	22000
Specific weight	kg/kW		1,04	0,92
Length	m		6,5	8,3
Width	m		2,3	2,7
Height	m		3,0	3,0
Volume	m^3		44,9	67,2
Specific volume	dm^3/kW		2,5	2,8
Remarks				Workhorse of the USN

Table 2 General Electric aero-derived gas turbines for marine application

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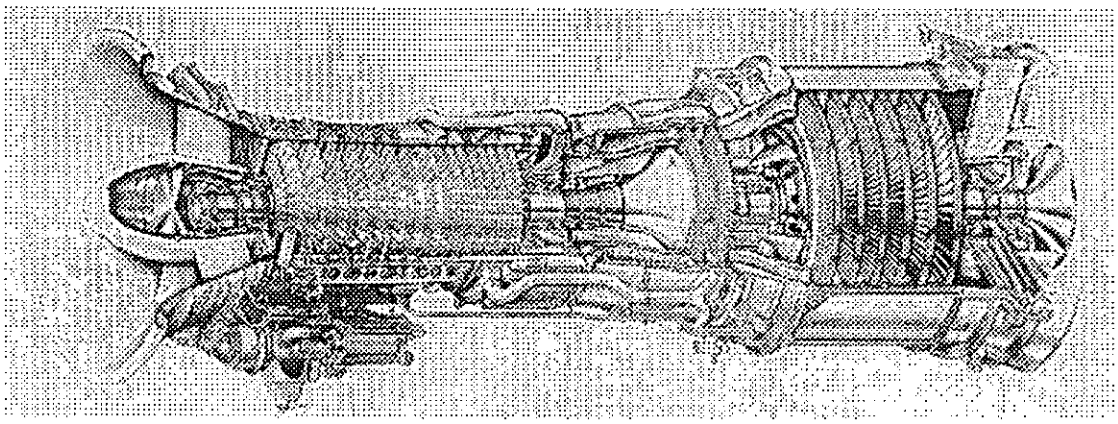
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Figures

Figure 1 3-D cut-away drawing of the core engine of a gas turbine for marine application (General Electric LM 2500)

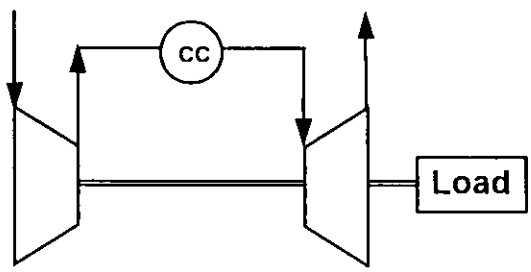


Figure 2 Single shaft simple cycle gas turbine consisting of a compressor, a combustion chamber and a turbine; compressor, turbine and load are connected to the same shaft.

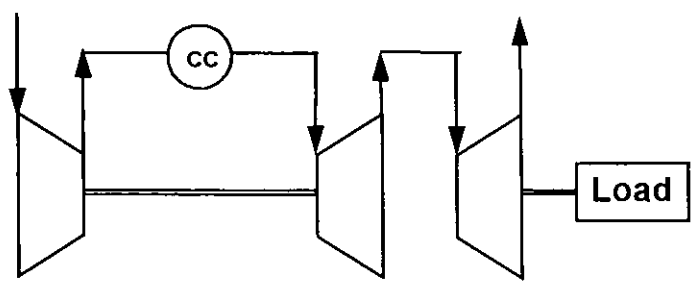


Figure 3 Two-shaft simple cycle gas turbine consisting of a compressor, combustion chamber and a turbine to drive the compressor. A separate power turbine drives the load which can be a ship's propeller or a gas compressor.

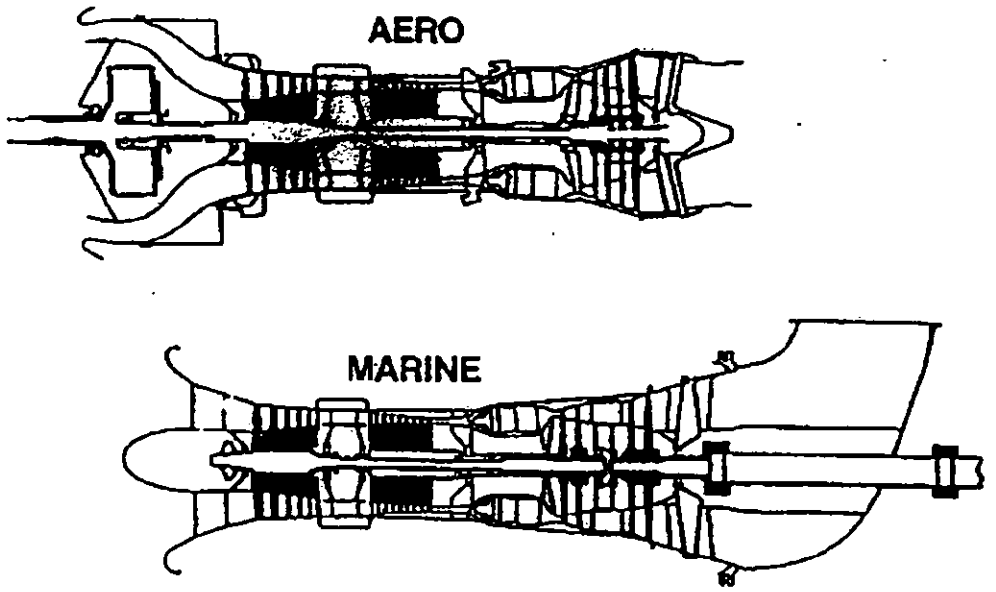


Figure 4 Difference between a "turbo-prop" engine and its derived gas turbine for naval use (Rolls Royce Tyne).

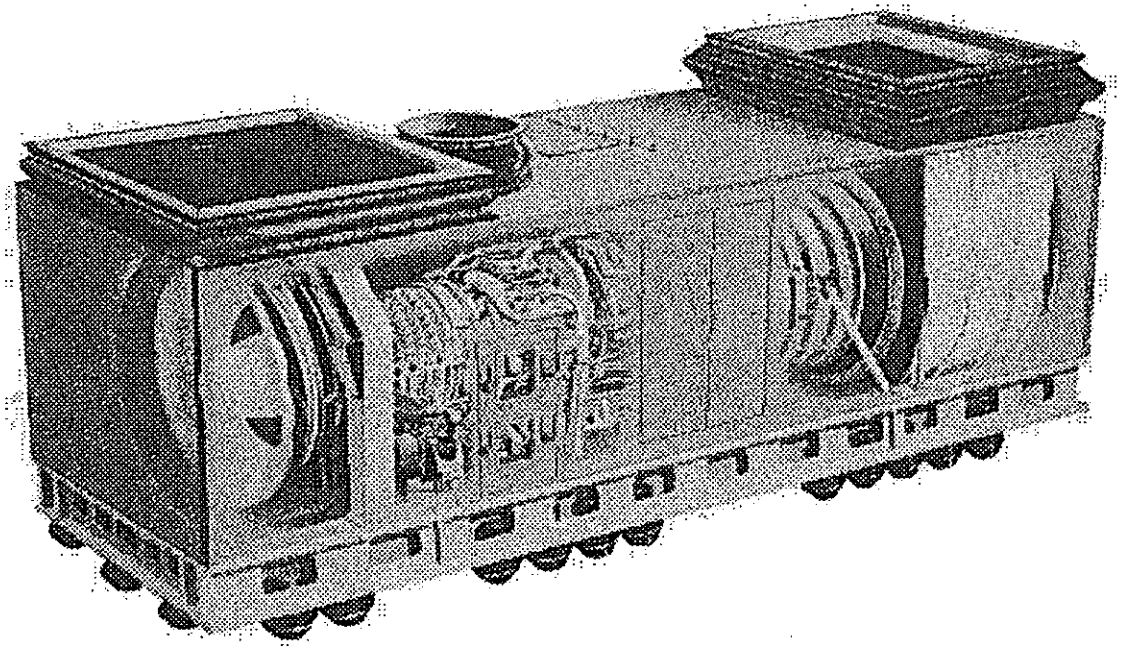


Figure 5 Gas turbine module for installation on board a ship (General Electric LM 2500)

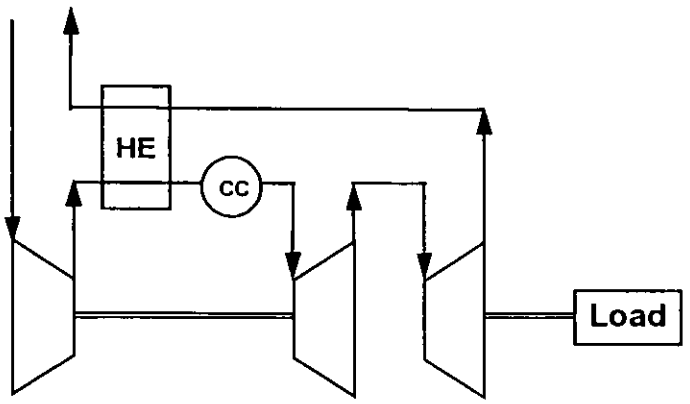


Figure 6 Gasturbine cycle with a Heat Exchanger between the exhaust gases and the air entering the combustion chamber

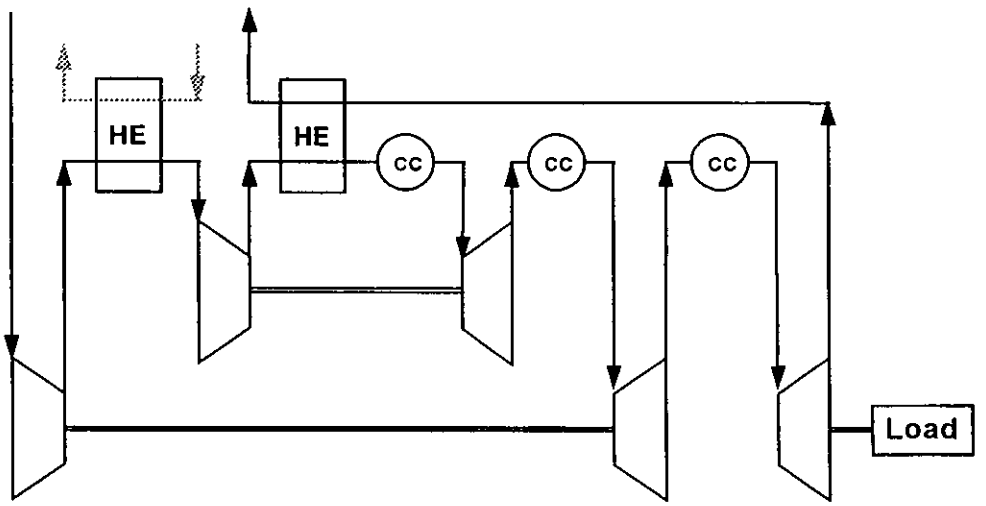


Figure 7 Survey of possible features of a complex cycle gas turbine showing intercooling of the air between compressor stages, heating of the compressed air in an exhaust heat exchanger and furthermore two reheat combustion chambers.

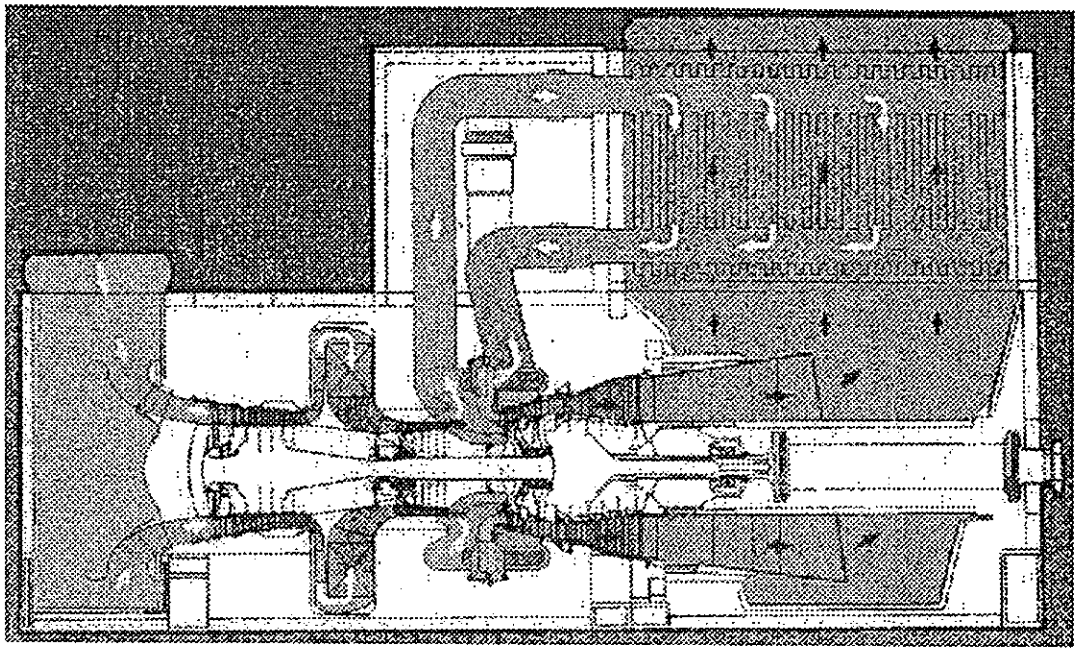


Figure 8 Cross section of a 22 MW ICR gas turbine (Rolls Royce WR 21)

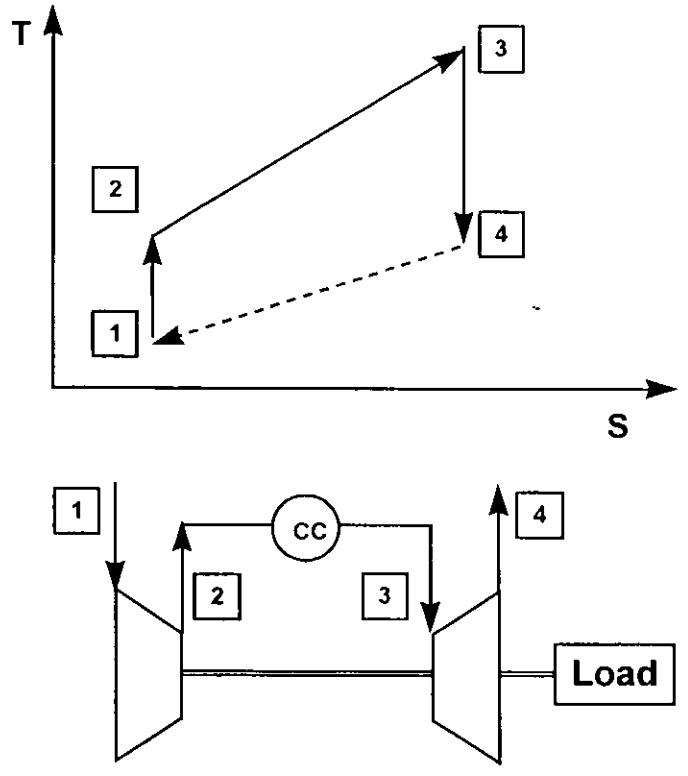


Figure 9 T-S diagram, flow scheme and station-numbering for the ideal Brayton cycle

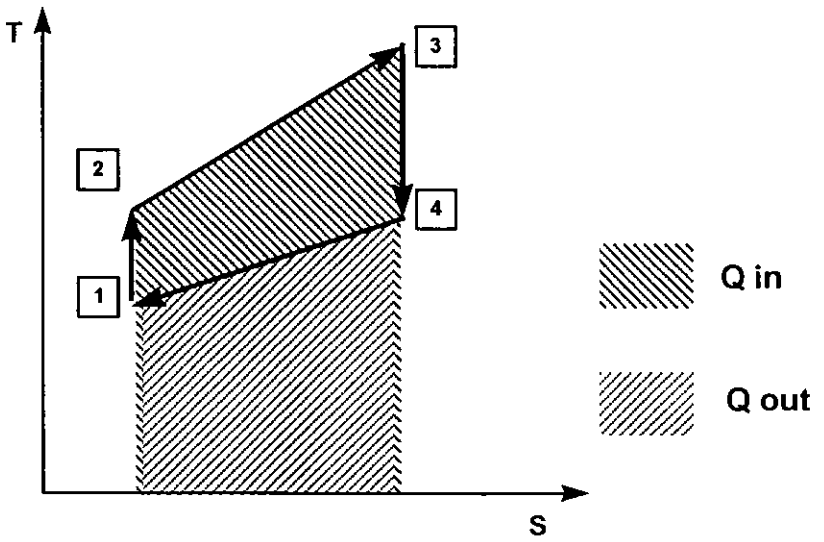


Figure 10 T-s diagram ideal Brayton process; heat input and rejection shaded

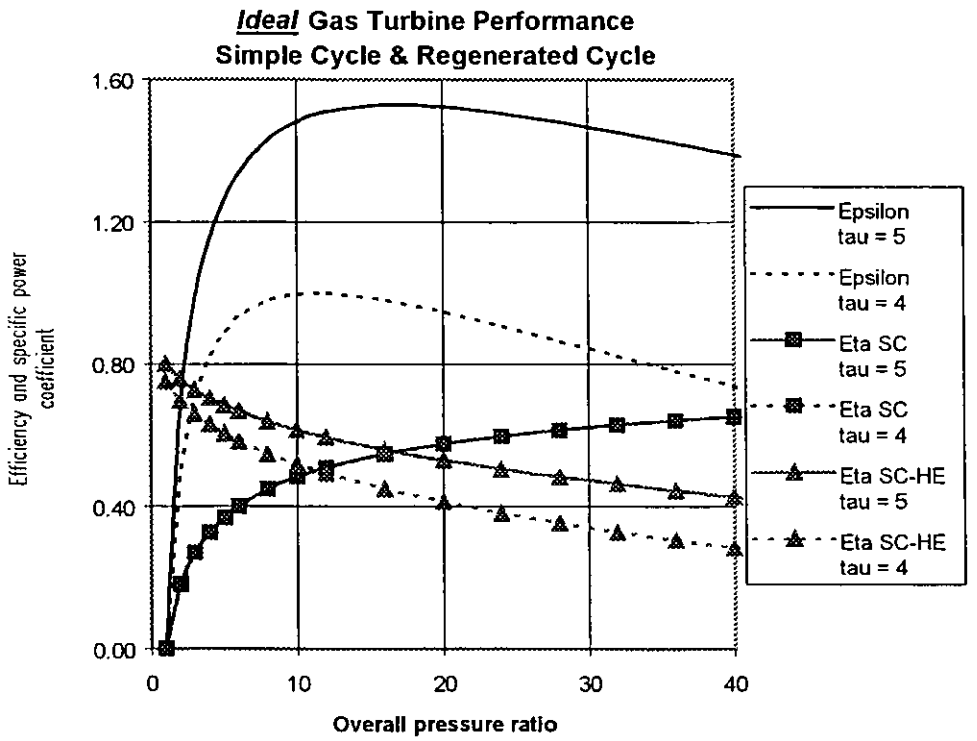


Figure 11 Efficiency and (non-dimensional) specific power as function of pressure ratio and at two values of the temperature ratio for the ideal Brayton process with and without (infinite) heat exchanger

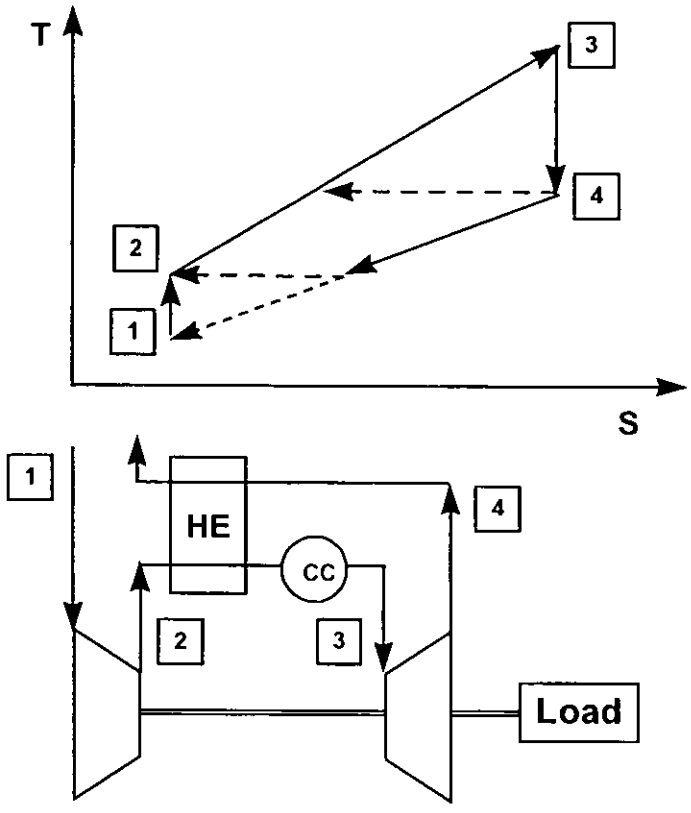


Figure 12 Temperatures in Brayton cycle with infinite heat exchange area in the recuperator

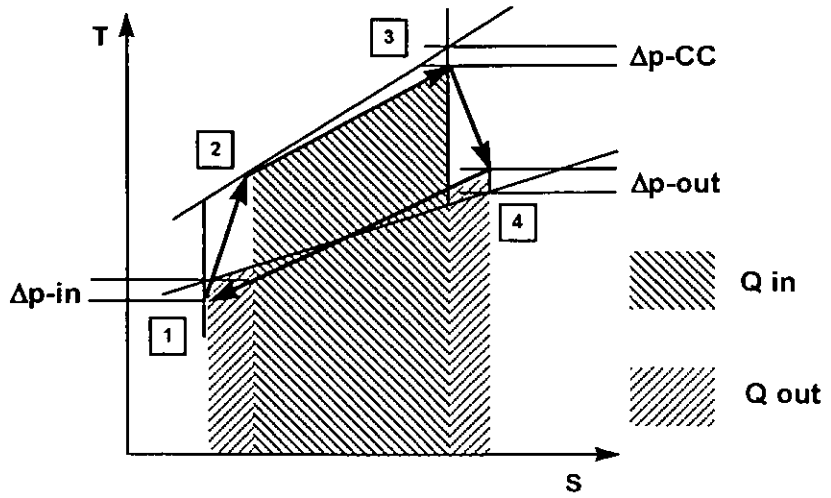


Figure 13 T-S diagram of a Brayton process with losses.

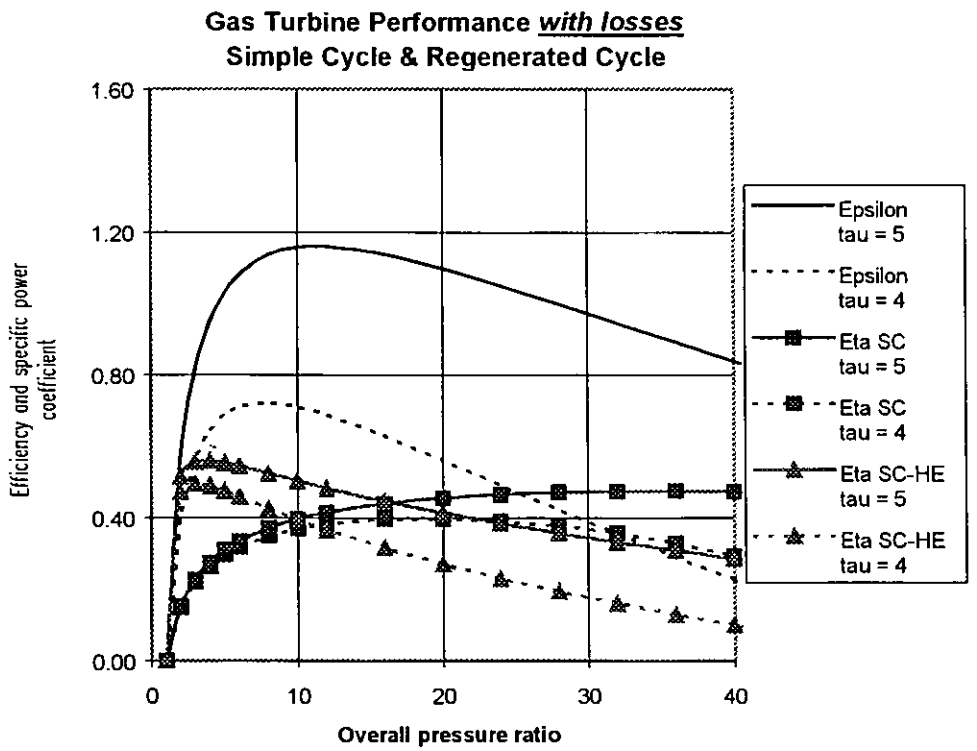


Figure 14 Efficiency and (non-dimensional) specific power as function of pressure ratio and at two values of the temperature ratio for the Brayton process with turbomachinery losses ($\eta_C = \eta_T = 0.9$) and with and without heat exchanger (finite: $\eta_{HE} = 0.9$).

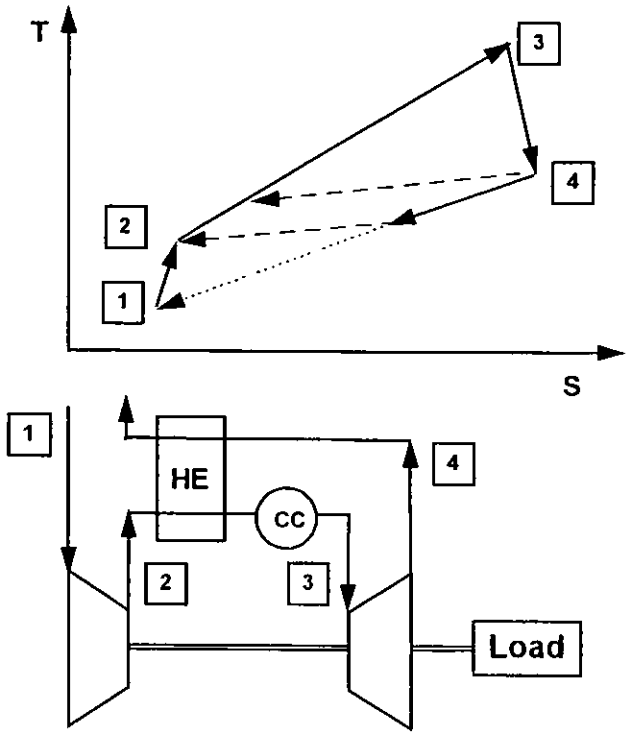


Figure 15 Effectiveness of a Heat Exchanger with finite heat exchange area in the recuperator

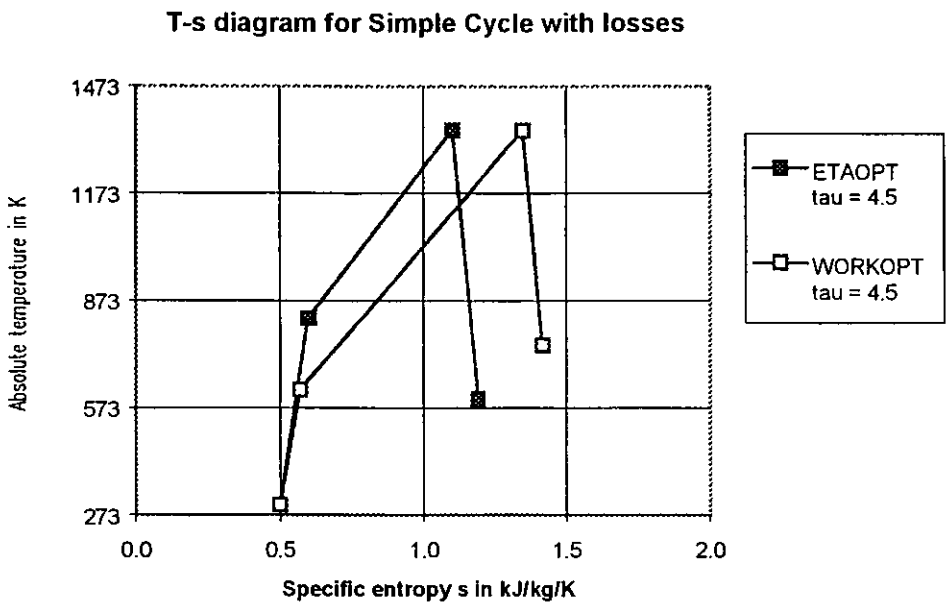


Figure 16 T-s diagram for a Simple Cycle Gas Turbine optimised for efficiency (ETAOPT) and specific work (WORKOPT) respectively

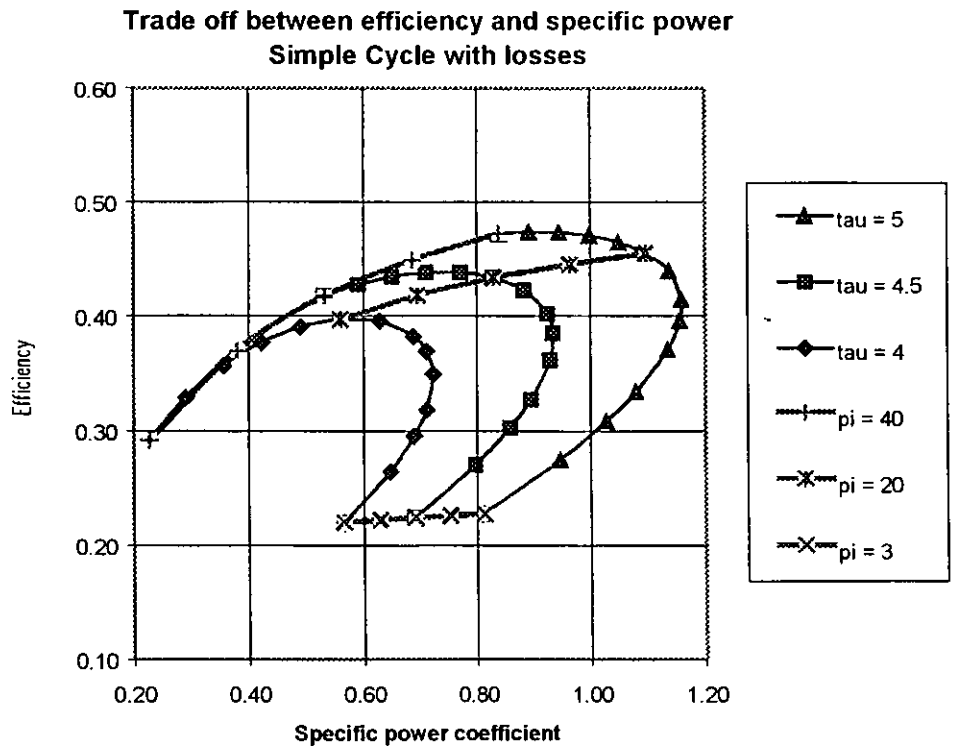


Figure 17 Trade-off between efficiency and specific power for a Simple Cycle Gas Turbine

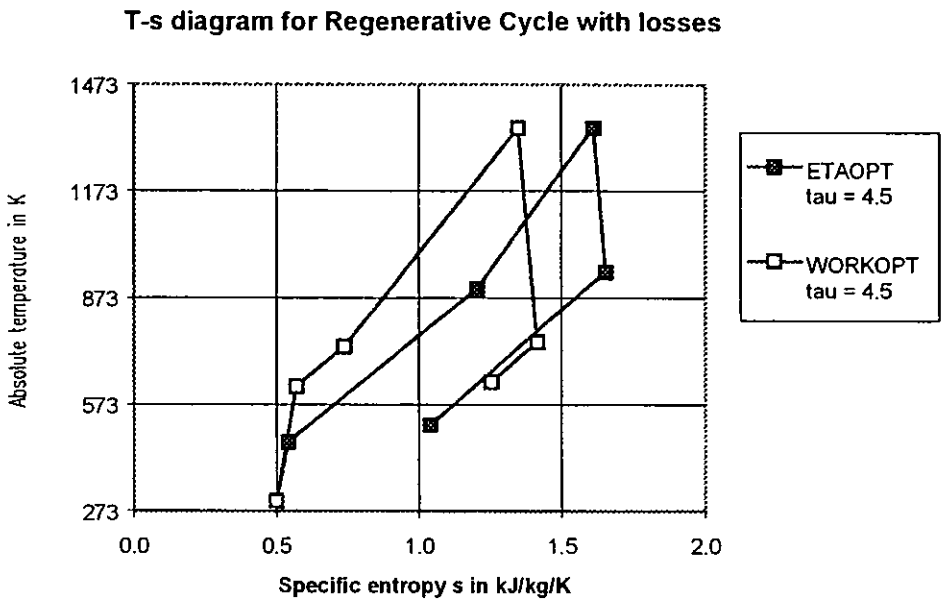


Figure 18 T-s daigram for the Regenerative Gas Turbine optimised for efficiency (ETAOPT) and specific work (WORKOPT) respectively

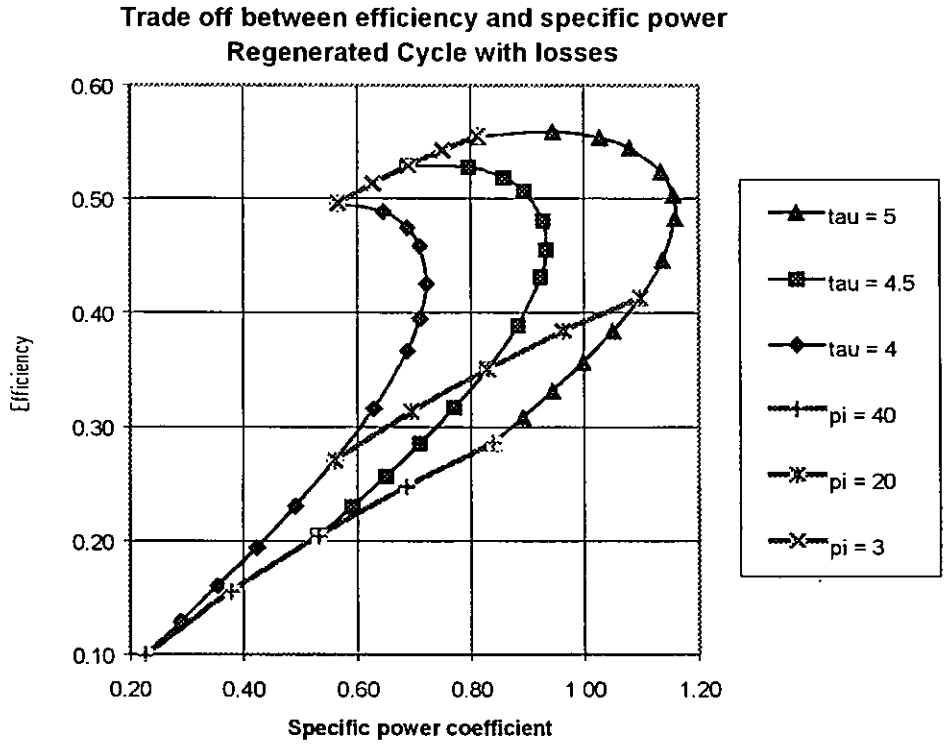


Figure 19 Trade-off between efficiency and specific power for a Regenerative Gas Turbine

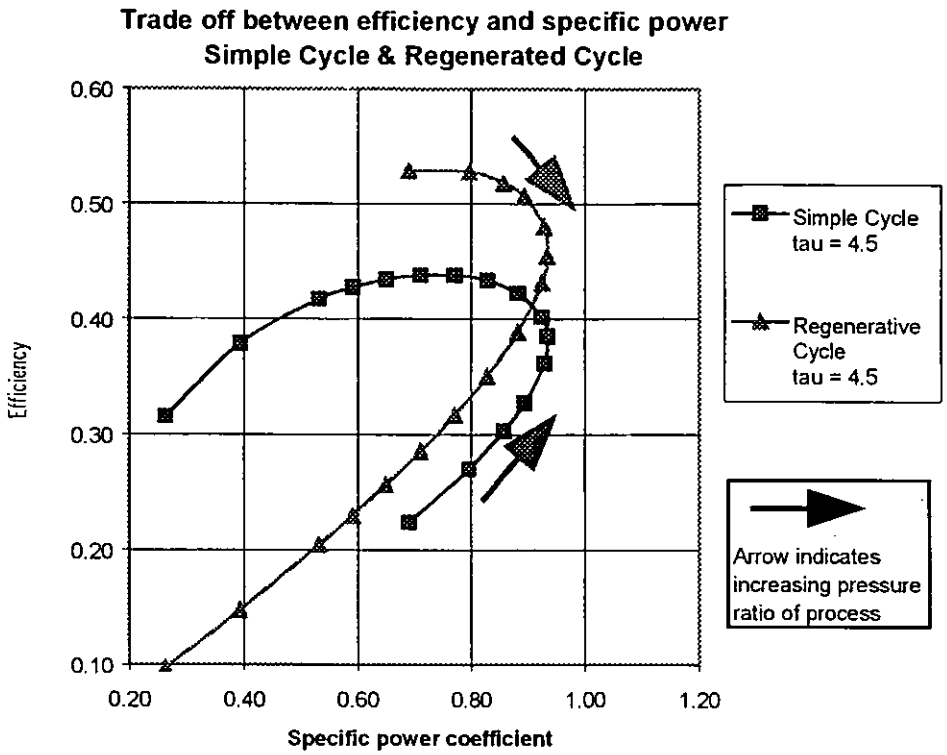


Figure 20 Trade-off between efficiency and specific power of the Simple Cycle Gas Turbine and the Regenerative Gas Turbine. The direction in which the pressure ratio increases along the curves is different for both cycles.

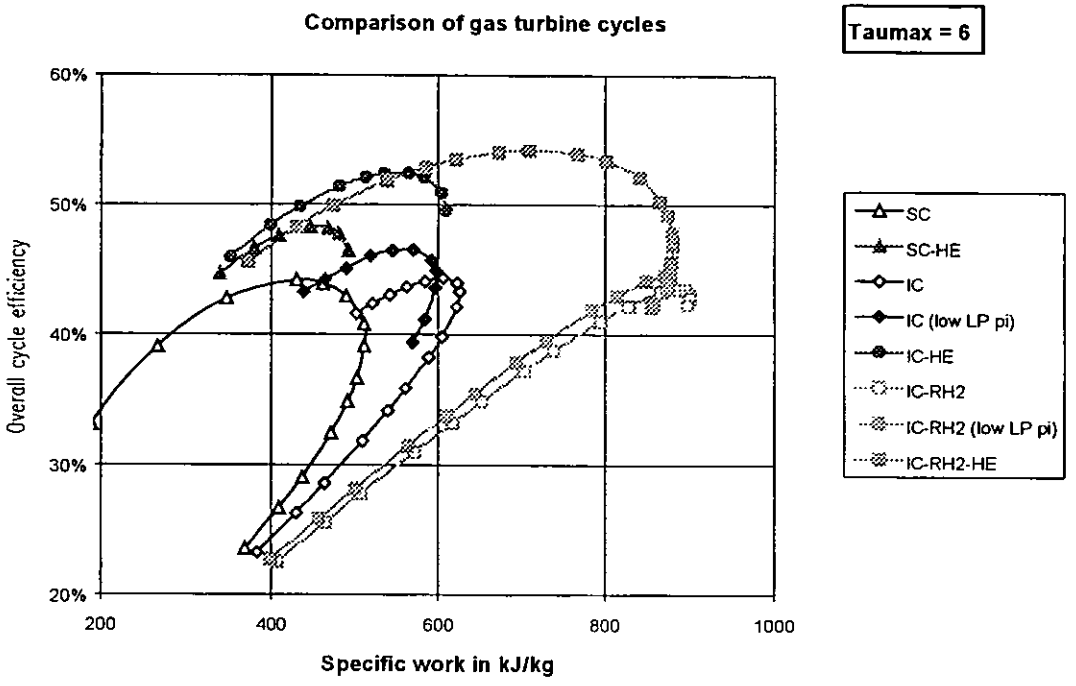


Figure 21 Comparison of several complex cycle gas turbines; maximum temperature ratio: 6; polytropic efficiency of turbomachinery: $\eta_T=\eta_T$ =0.9; effectiveness of exhaust heat exchanger and intercooler: η_{HE}=0.9

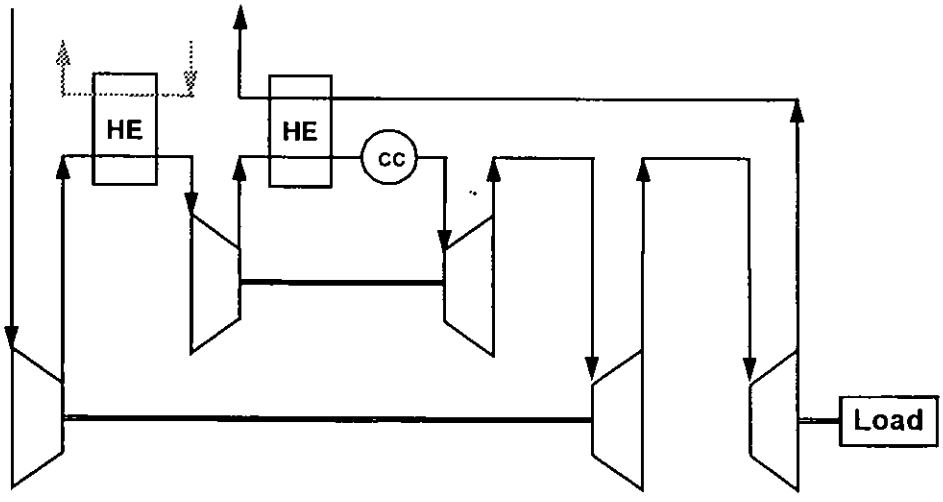


Figure 22 Flow scheme of an advanced cycle gas turbine with intercooling between compressor stages and a heat exchanger between the exhaust gases and the compressed air (ICR cycle)

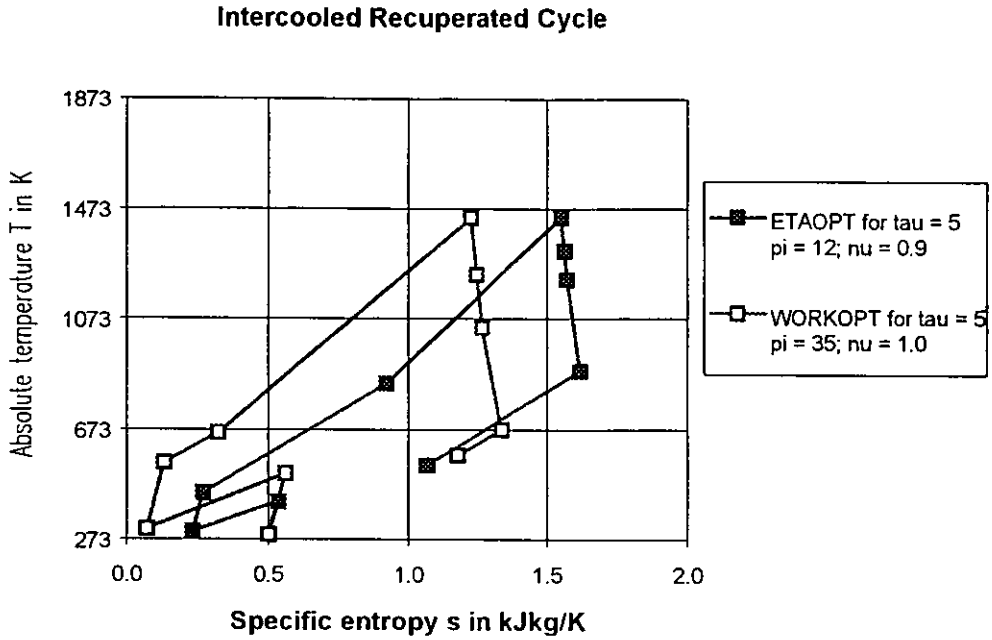


Figure 23 T-s diagram of an advanced cycle gas turbine with intercooling between compressor stages and a heat exchanger between the exhaust gases and the compressed air (ICR cycle)

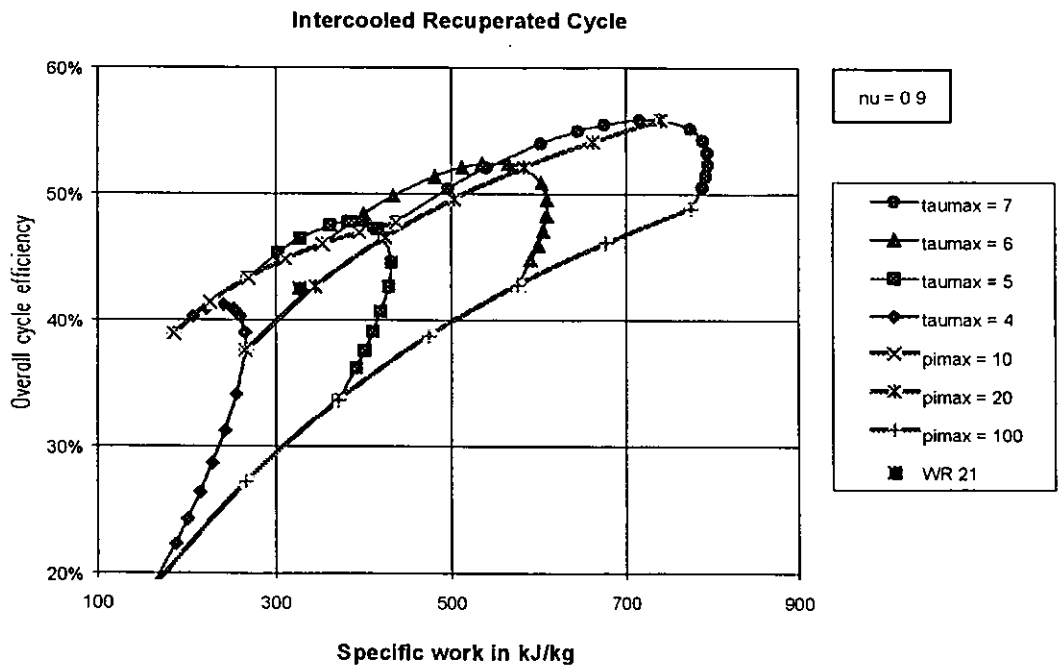


Figure 24 Trade-off between overall efficieincy and specific power of an advanced cycle gas turbine with intercooling between compressor stages and a heat exchanger between the exhaust gases and the compressed air (ICR cycle)

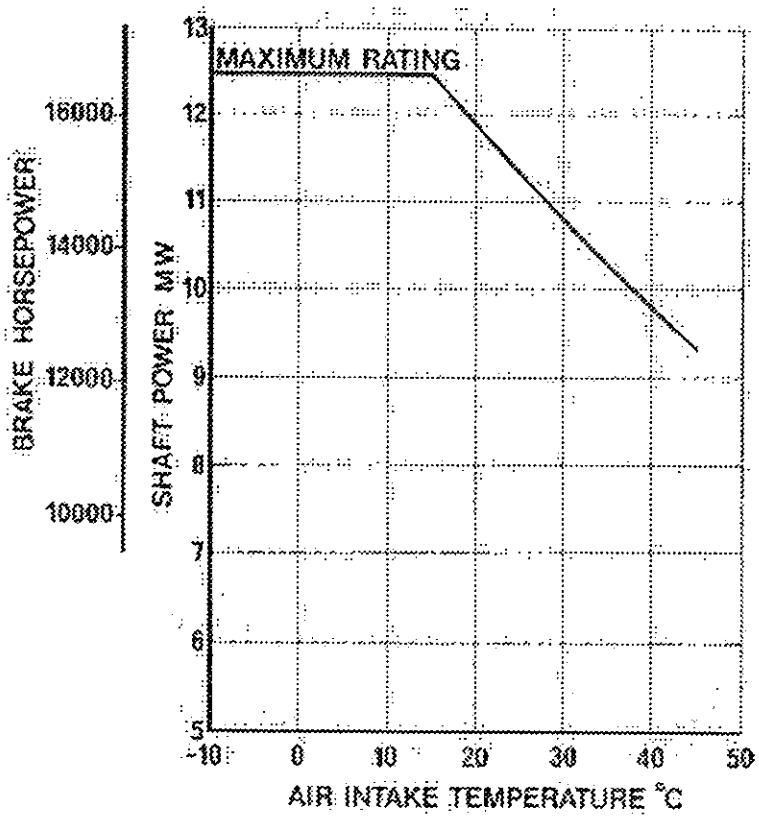


Figure 25 Power reduction of gas turbine at increasing ambient temperature

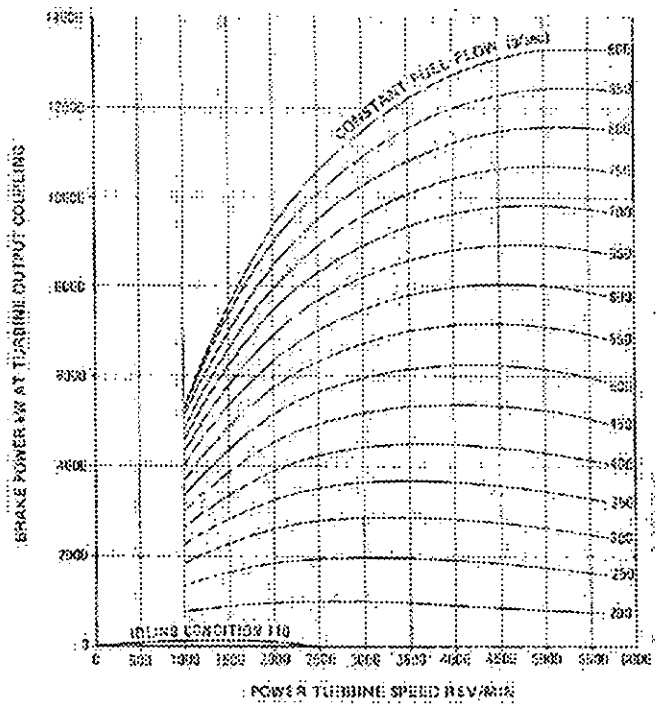


Figure 26 Characteristic for a simple cycle gas turbine with free power turbine with lines of constant fuel in a plot of power versus speed

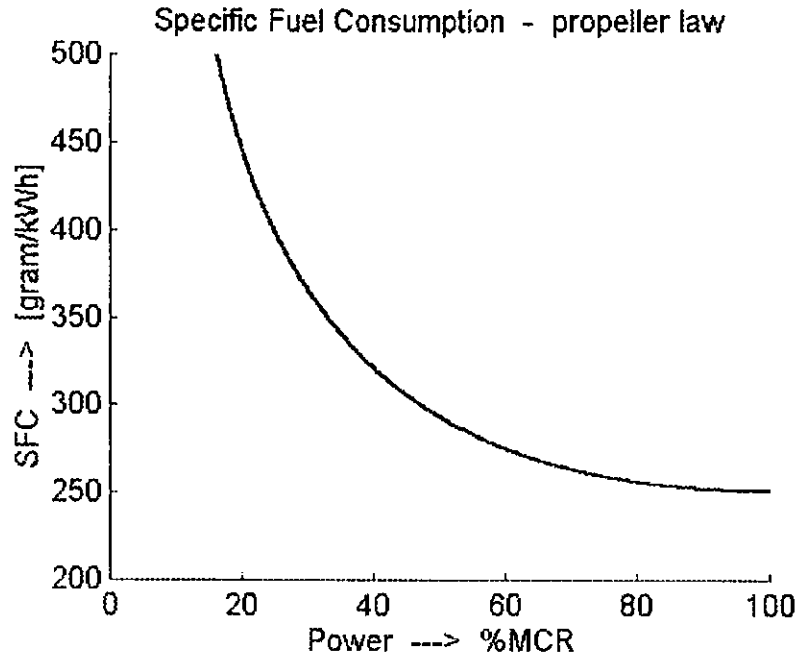


Figure 27 Specific fuel consumption of simple cycle gas turbine at part load along the propeller law

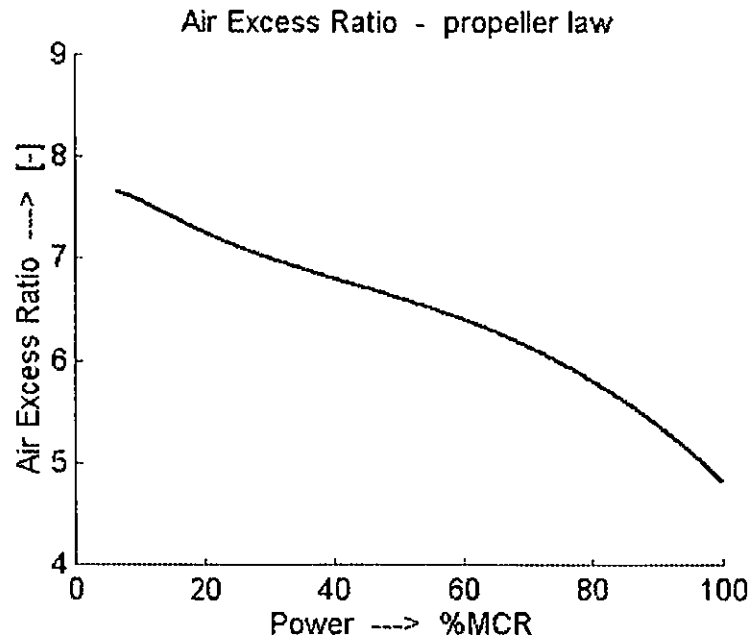


Figure 28 Air excess ratio of simple cycle gas turbine at part load along the propeller law

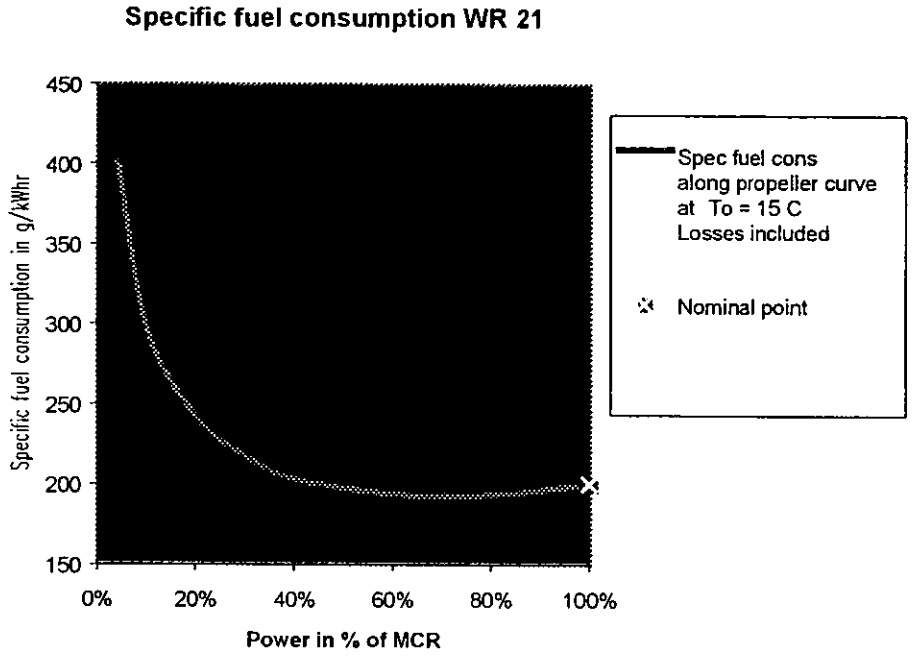


Figure 29 Specific fuel consumption of ICR gas turbine with variable blade geometry at the turbine; part load curve along the propeller law

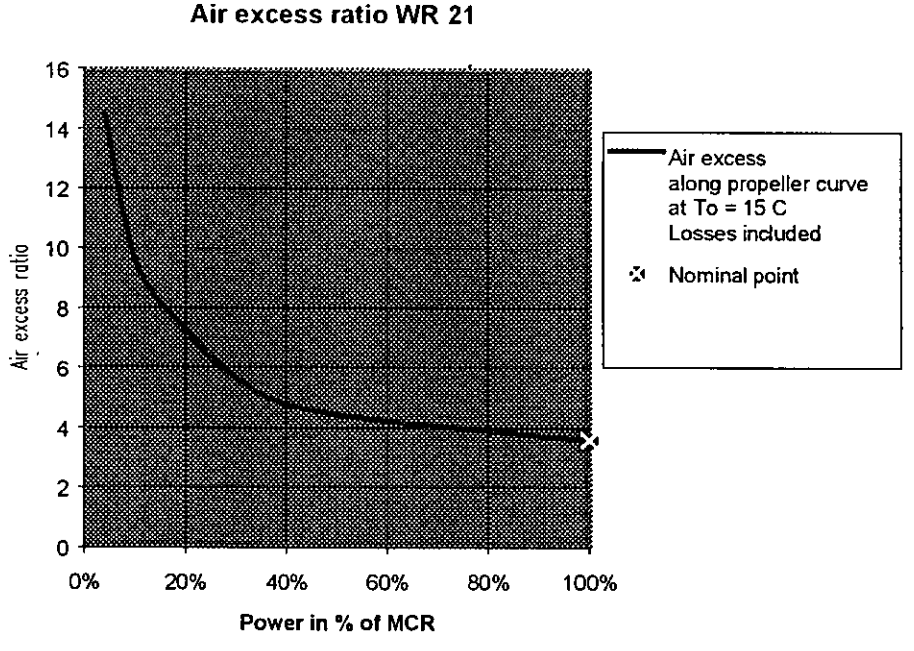


Figure 30 Air excess ratio of ICR gas turbine with variable blade geometry at the turbine; part load curve along the propeller law

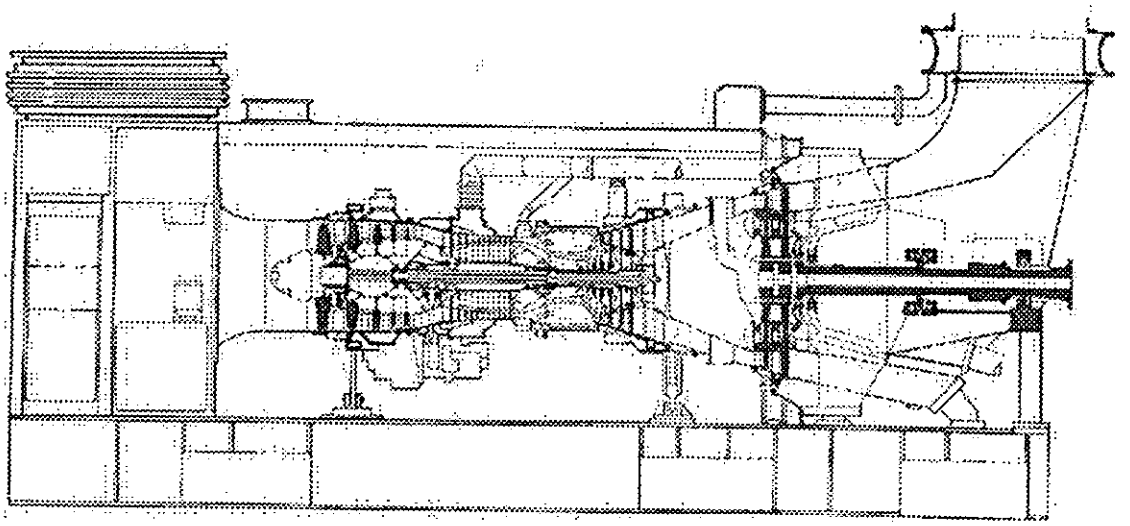


Figure 31 Gas turbine module suitable for installation on board ships (Rolls Royce Spey SM1A)

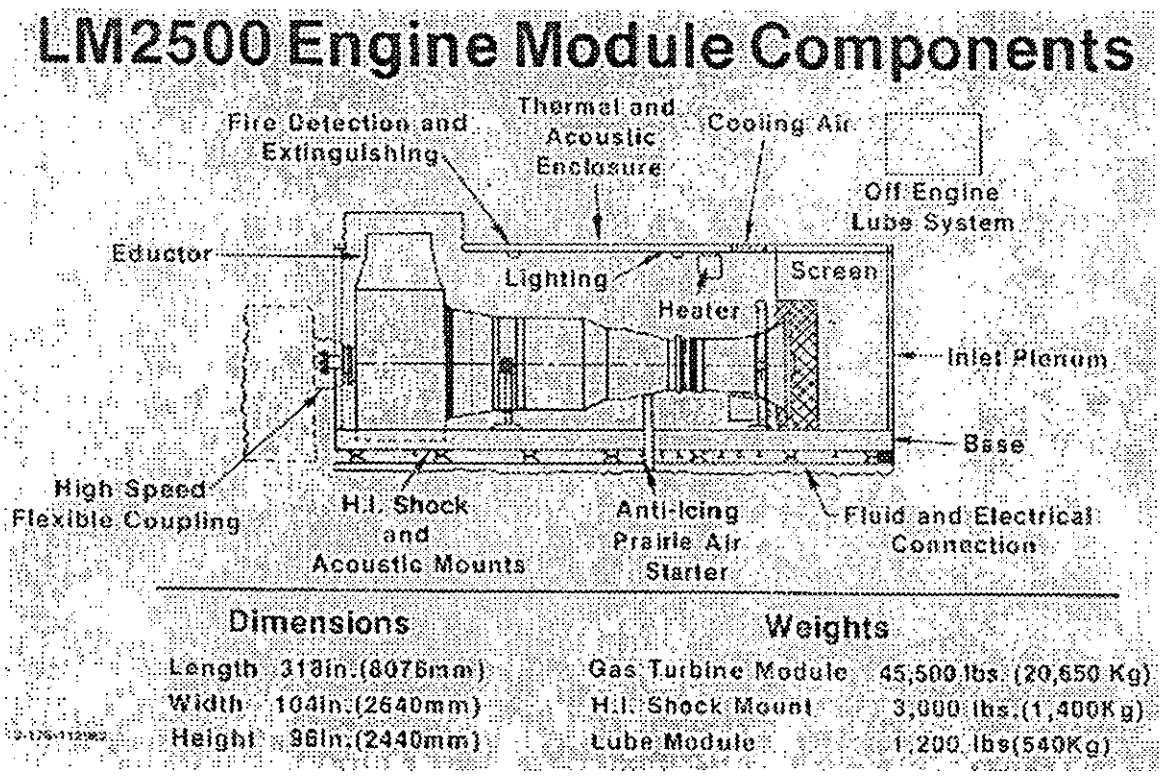


Figure 32 Overview of components in a gas turbine module suitable for installation on board ships (General Electric LM 2500)

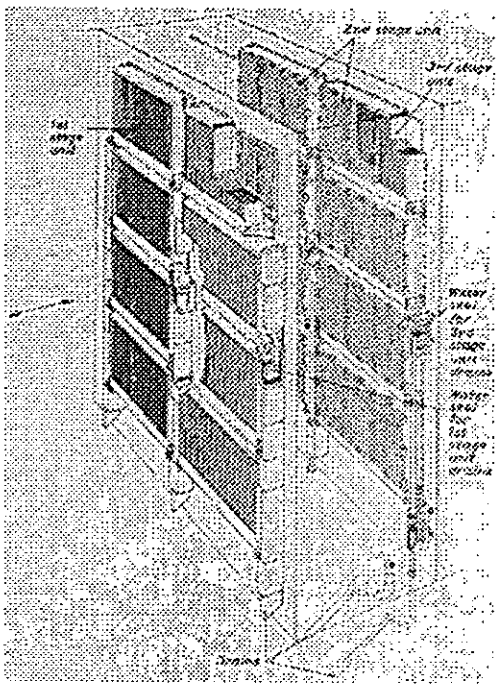


Figure 33 3-stage inlet air filter system for marine gas turbine

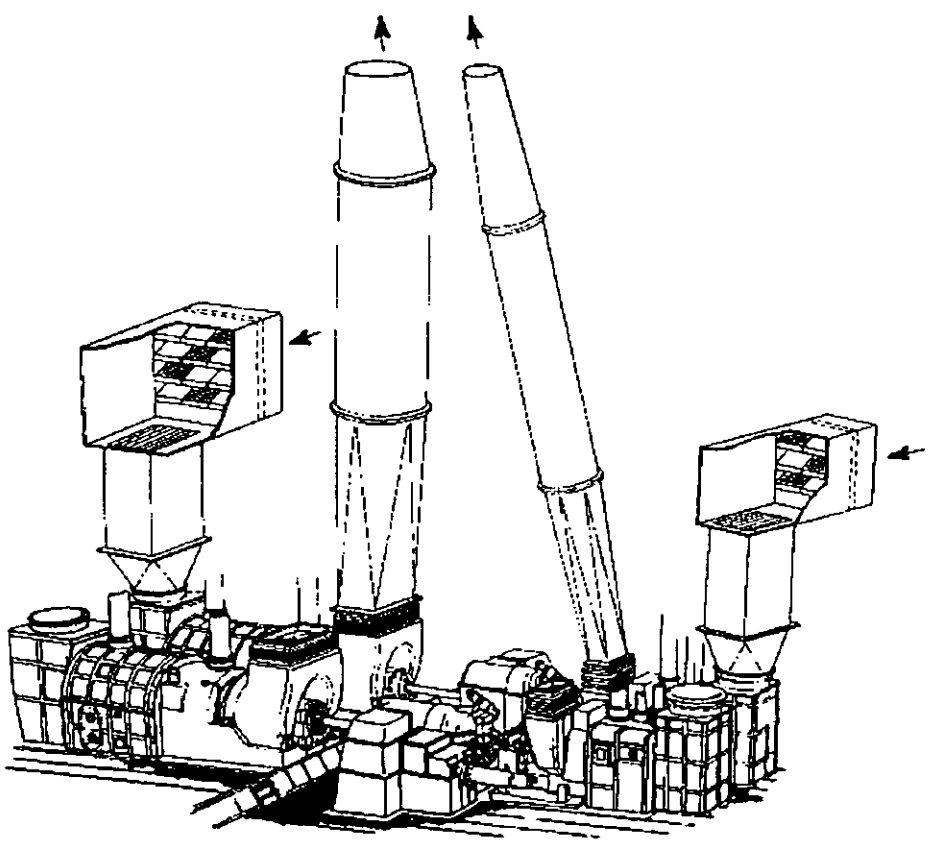


Figure 34 Up- and downtakes of a gas turbine driven frigate

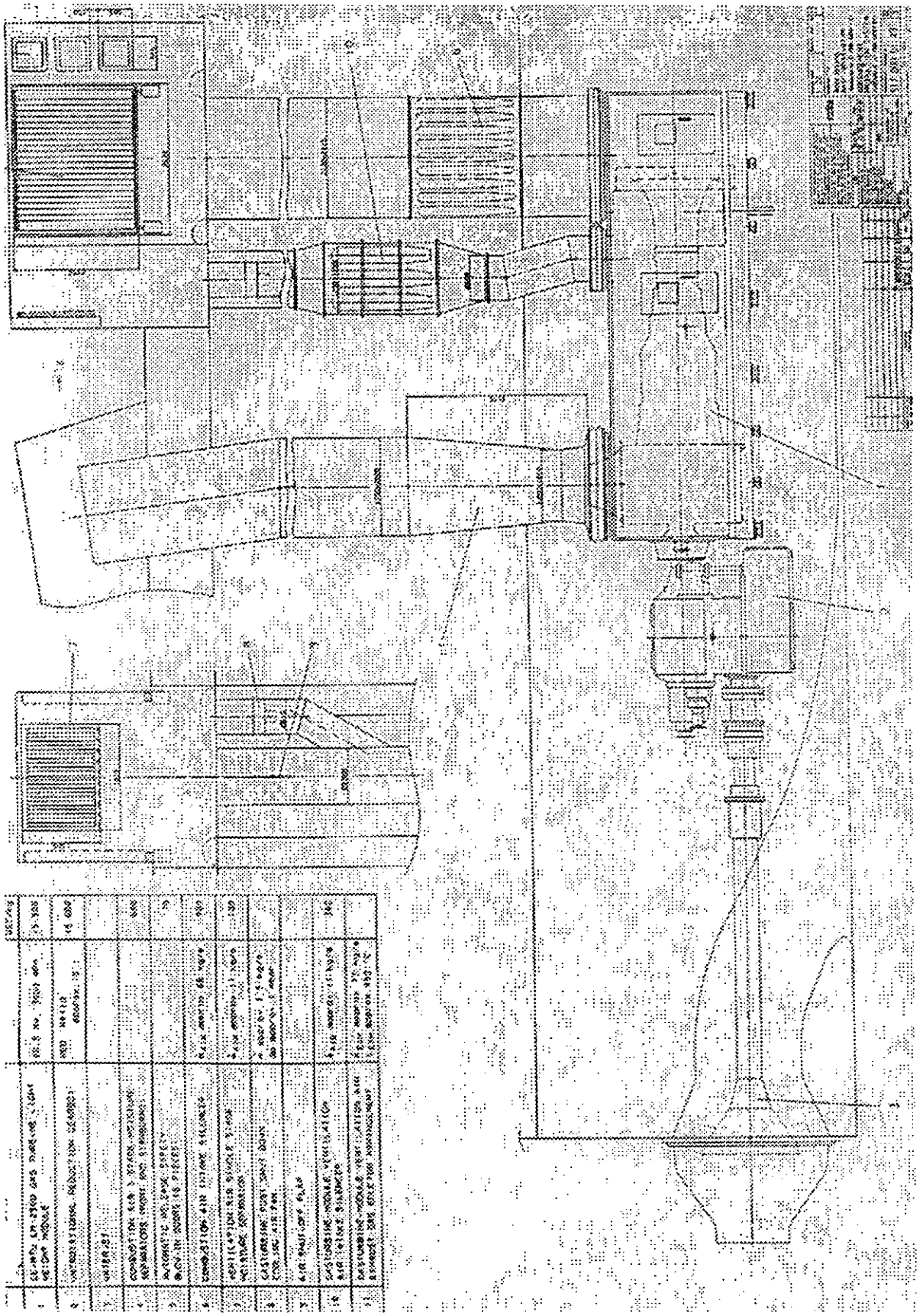


Figure 35 Gas turbine installation of a fast ferry with water jet propulsion

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

USE OF GAS TURBINES FOR MARINE APPLICATIONS

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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USE OF GAS TURBINES FOR MARINE APPLICATIONS

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ABSTRACT

This article contains a general overview of the use of gas turbines for marine applications.

The characteristics of gas turbines vs. diesel engines are analyzed taking into account, in particular, the GE LM2500, that is today one of the most commonly installed gas turbine in western Navies and in many vessels for commercial applications.

FiatAvio experience in fast ferry application is outlined, with particular reference to engine maintenance.

Finally a comparison between pneumatic and hydraulic starting system on gas turbines is depicted, as a result of FiatAvio experience both in marine and in industrial applications.

1 - INTRODUCTION

Gas turbines, as well as diesel engines, are extensively used in marine applications. A wide range of sizes of both engines can be found on board of ships, both for military and commercial applications.

Their typical use is either as prime movers in mechanical drive systems or as main drivers for on board electric generators. Only recently however a new and promising application has involved both gas turbines and diesel engines as prime movers for large electrical generators, to supply power for electric propulsion.

Paragraph 3 will focus on these applications relative to gas turbines.

2 - NOMENCLATURE

AES	All Electric Ship	FMVA	Fuel Metering Valve and Actuator
APU	Auxiliary Power Unit	FOD	Foreign Object Damage
ASW	Anti Submarine Warfare	IMO	International Maritime Organization
CODAG	COmbined Diesel And Gas	MCM	Minimum Crew Maintenance
CODLAG	COmbined Diesel eLectric And Gas	MCR	Maximum Continuous Rating
CODOG	COmbined Diesel Or Gas	RO-RO	Roll-on Roll-off
COGAG	COmbined Gas And Gas	SPL	Sound Pressure Level
COGES	COmbined Gas Electric Steam	TCS	Turbine Control System
COGOG	COmbined Gas Or Gas	VSV	Variable Stator Vane
FoC	First of Class		

3 - TYPICAL APPLICATIONS OF GAS TURBINES

3.1 - Mechanical drive

Mechanical drive is currently the most common use of gas turbines for marine propulsion systems. This has historical reasons because of the very well proven engine / reduction gear / shaftline / propeller technology, although this can also be considered a drawback, in that it requires more complex drive train calculations.

Vibration analysis shall take into account the whole shaft train, with unavoidable trials, iterative calculations and design changes before optimum design configuration can be defined.

Mechanical drive systems require a reduction gear and a more or less long shaftline for connection to the waterjet or the propeller; this is a complication for the propulsion plant installation, which should be as close as possible to the propeller to minimize the risk of dangerous vibration modes.

Long shaftlines require a high number of intermediate supports (bearings), with related necessity for lubrication, cooling, temperature and vibration monitoring; moreover the high number of mechanical components turns into a higher probability of failure, and to more difficult shaftline alignment.

There are several possible power plant configurations. In this paragraphs only those powered with gas turbines will be taken into account.

Fig. 1 shows a schematic of the lay-out arrangement for a CODOG configuration, where the propellers are driven by diesel engines for low speed, or by gas turbines for boost speed.

If, in Fig. 1, the diesel engines are replaced by a couple of gas turbines the configuration will be called COGOG. In this case the smaller couple of turbines, replacing the diesel engines, is the driver for low speed, the larger couple of turbines is for boost.

Fig. 2 shows a schematic of a COGAG configuration, where a couple of gas turbines of the same size drive the same shaftline through a common reduction gearbox. This is a power plant arrangement very common in large ships, like Destroyers and Aircraft Carriers.

CODAG configuration is the one adopted for Tirrenia/Fincantieri Fast Ferries (Jupiter MDV 3000 Project). Details of this configuration are shown in § 7.

Other power plant arrangements acronyms are used when gas turbines are coupled to electric generators for electrically powered propellers (GODLAG), or when gas turbines are combined with steam turbines in cogeneration plants (COGES). These type of propulsion plants will be better described in another section of this symposium.

3.2 - Electric propulsion

Although electric propulsion in marine application is not a brand new idea, recent applications show a promising perspective in this field due to the high flexibility in the propulsion plant layout.

As a matter of fact the gas turbine / electric generator group, in theory, can be installed anywhere in the ship, regardless of the shaftline and propeller location, while electric power can be transfered to final users by means of properly sized cables. This turns out to be a good competitive advantage for the electric propulsion with respect to the more traditional mechanical drive, in that it allows a better optimization of room available for payloads.

One of the advantages of electric propulsion is a more simplified drive train calculation and installation: in the Azipod arrangement (a device which combines, in a single unit, the propulsion motor, the propeller and the steering unit) rotating components are limited to the very short distance between electric motor and propeller. No long shaftlines required.

Another advantage in the gas turbine / electric generator arrangement is due to the fact that the gas turbine will be used at constant speed, and will generally work very close to its design point, hence with optimum efficiency.

High controllability at low speed of electric drives, low maintenance, reduced noise and vibrations, high maneuverability of Azipod type propulsion systems are further key points in favor of the AES (All Electric Ship).

On the other side there are some drawbacks, in that electric propulsion requires installation on board of large size generators, usually very large and heavy, with all the auxiliary equipment necessary for proper and safe generator operation. In fact the design engineer shall consider also installation of controls, protections, transformers and/or current rectifiers, in case of use of DC electric engines as propeller drives.

All these equipment are large and heavy; also electric engines, both AC or DC, for the power levels required for ship propulsion, are quite heavy, require adequate protection and cause increase of initial cost of the plant.

Lifecycle costs also are adversely affected by increased fuel consumption (due to electric losses and equipment efficiency) and by the more complex electric plant management.

Nevertheless current trend in large cruise ships is towards the use of electric propulsion, due to the good general flexibility granted by such a solution.

4 - GAS TURBINE / DIESEL ENGINES COMPARISON

As already highlighted gas turbines, as well as diesel engines, are widely used in marine applications.

There are several reasons however that make gas turbines today very attractive with respect to diesel engines. In this paragraph these reasons will be simply listed, as this brief pamphlet cannot claim to be thoroughly exhaustive on the topic.

The main advantages of gas turbines can be listed as follows:

1. High power to weight ratio.

   This is the most manifest and important point in favor of gas turbines, mainly in commercial marine applications, where weight and space saving is of the utmost importance. For a given output shaft power a diesel engine would be 5 to 6 times as heavy as a gas turbine. This means also much more room taken by the diesel engine: an LM2500 package, with acoustic and thermal insulation, has a power to volume ratio of about 400 kW/m^3, that is up to five times the ratio for a medium speed diesel.

   Moreover attention should also be paid to additional weight and volume of auxiliary equipment, e.g.: air intake and exhaust ducts, filters, silencers, ancillary plants. Commercially available diesel engines for marine applications have an output shaft power that does not exceed 8-10 MW, although there are examples of even larger engines; for a ship that needs about 50 MW installed power roughly five to six heavy diesel engines would be necessary with relevant auxiliary equipment, ducting, filtering and silencers; or two LM2500 plus gas turbines.

2. Comparable engine efficiency.

   In the past engine efficiency used to be one of the key points in favor of diesel engines. Last decades improvements in the design and manufacturing technology of aircraft components however, turned out to benefit also derivative engines, like GE LM2500.

   Optimized flowpath, compact design, improved materials (better resistance to high temperatures, better components life cycle), lower maintenance costs, use of coatings (improved resistance to corrosion and aggressive marine environment): all these improvements have raised gas turbines thermal efficiency up to values around 38% and over in simple cycle arrangements, comparable with diesel engines efficiency, and even more in combined cycles.

3. Low exhaust emissions.

   Nowadays there is a strong worldwide pressure to reduce emissions. This request has been accepted by the International Maritime Organization (IMO), that is pushing for a 30% reduction in NO_x emissions for new engines; restrictions may be applied to already operating engines also. Gas turbines today have emission rates that are far below Marpol 73/78 requirements, while diesel engines often require exhaust emissions abatement equipment, to comply with IMO regulations.

4. Low noise.

   Sound Pressure Level due to airborne noise in a gas turbine can be reduced by means of adequate enclosure (see § 7.2.3). Typical mean values of SPL are around 90 dB(A) at 1 m from the enclosure, which is below standard values for a large, medium speed diesel engine, that show a SPL at 1 m in the range of 110 to 115 dB(A) at max power.

   Structureborne noise also is normally lower in gas turbines than the one experienced in reciprocating engines, due to the different working principle.

   Airborne noise in intake and exhaust ducts can be reduced in both cases by use of silencers and/or thermal insulation of the duct.

5. Suitability

   Aeroderivative gas turbines were initially designed for aicraft engines; hence they are suitable for applications where start/stop cycles are very frequent, as in fast ferries, where it is possible to have over 3 to 4 start/stop sequences per day.

Another key point is automation of a gas turbine. Recent progress in digital control systems have strongly increased the capability to monitor gas turbine operation and working parameters in a user friendly way, so that it is relatively "simple" to tune and control functioning of the engine from the switchboard room.

On the other hand there are a few points where diesel engines can still be given some credit. In fact gas turbine technology is more sophisticated and requires use of special materials in some components, which is particularly true in the case of the aeroderivative turbines like GE LM2500. This turns out to cause a higher initial cost of the gas turbine with respect to diesels.

Moreover aeroderivative gas turbines were originally designed to be operated with aviation fuels, or very refined fuel oils, which makes their operating costs less attractive because of the higher costs of fuel and fuel handling (storage, cleanliness, filtration, etc.). This however turns out to be one of the aforementioned advantages of the gas turbine: in fact a more refined fuel will give less pollutants.

Recent progress in gas turbine technology has led to new or improved products, that are quickly eroding the competitive advantage of diesel engines as regards lifecycle costs. The enhanced version of the GE LM2500 gas turbine (LM2500 plus) was reported to have a thermal efficiency of 38% in simple cycle arrangements, with improved installed and lifecycle cost per unit

kW. Rolls-Royce reported that the a marinized version of the Spey can run on marine gas oil, with no smoke emission throughout the operating range.

5 - MILITARY MARINE APPLICATIONS OF GAS TURBINES

Use of gas turbines for propulsion systems in marine applications is not a recent practice.

To find the first applications of these engines we must go back to the 60's, with the U.S. "Hamilton" Class Coastguard (1967) and the RO-RO cargo ship "Callaghan" (1969), the first ship entirely powered by means of gas turbines (first naval application of LM2500).

A few years before also the Soviets had launched into service ships powered by means of gas turbines, like the Destroyer "Obraztsovyi" (1964) and the "Petya" Class Frigates (1963).

This was due to the indubitable advantages given by gas turbines with respect to steam turbines, widely adopted in the past, and to diesel engines. However only in the following decades gas turbines were more extensively used as main propulsion engines in military ships, combined with diesel engines in CODOG or CODAG configurations, or as prime movers for electric generators in CODLAG arrangements.

At the beginning of the 70's the Royal Navy started a new program of modernization of British fleet with six "Amazon" Class Frigates, entered into service in 1974-1977 (COGOG configuration), and fourteen Type 42 "Birmingham" Class Destroyers, entered into service starting 1976 through 1985.

The latter were COGOG configuration ships too, powered by means of a couple of Rolls Royce Olympus TM3B gas turbines (for boost power), and a couple of Tyne RM1C (for cruise and maneuvering).

At the end of the 70's the French Navy also launched seven "George Leygues" Class Destroyers, in CODOG configuration, powered by means of a couple of Rolls Royce Olympus TM3B gas turbines.

The Canadian Navy in 1972-1973 launched into service four "Iroquois" Class Destroyers, in COGOG configuration, powered by a couple of Pratt & Whitney FT4 A2 gas turbines, for boost speed, and a couple of smaller Allison 570-KF gas turbines for cruise and maneuvering.

Among the COGAG configuration ships mention should be made of the U.S. Navy Class "Spruance" Destroyers (31 ships, each powered by means of four GE LM2500 gas turbines), launched into service in 1975 through 1983. In this category the Russian fleet counts the Frigates "Krivak" (40 total ships in three different versions delivered in 1970 through 1990), the Destroyers "Kashin" (20 ships delivered in 1962 through 1973) and, more recently, the ASW Destroyers Udaloy (8 ships delivered in 1983 through 1995).

In the 70's the Italian Navy started a modernization program with several Frigates and Destroyers; details of these ships are reported in paragraph 6.

Gas turbines since then were widely used also for commercial marine applications and more recently, due to their high power to weight ratio, in fast ferries.

Advantages of the use of gas turbines in marine applications, compared to diesel engines, were already discussed in § 4; next paragraphs will show Fiat Avio experience in gas turbines installation, on both military and commercial vessels.

6 - FIAT AVIO EXPERIENCE IN MARINE PROPULSION

Fiat Avio experience in marine propulsion dates back to the 70's, when the first "Lupo" Class Frigates for the Italian Navy entered into service.

"Lupo" is FoC of a total of four 2500 t displacement frigates (the other three are: Orsa, Perseo, Sagittario), powered by means of two Fiat/GE LM2500 gas turbines and two diesel engines in a CODOG configuration. The four frigates entered into service in years 1977 through 1980.

These frigates were followed by the "Maestrale" Class Frigates, a total of eight 3000 t displacement ships (Maestrale, Aliseo, Euro, Grecale, Libeccio, Scirocco, Espero, Zefiro) with the same power plant configuration, but with more powerful diesel engines. These frigates entered into service in years 1980 through 1985.

In the same year (1985) also the "Garibaldi" Carrier (13,400 t displacement) entered into service; this is an "all gas" powered ship, with four Fiat/GE LM2500 gas turbines in a COGAG configuration.

In 1989 two 5400 t displacement Destroyers ("Luigi Durand de la Penne" and "Francesco Mimbelli") entered into service. The propulsion plant is always a CODOG configuration, with the Fiat/GE LM2500 gas turbines for boost power.

More recently (starting 1995) four more 2500 t displacement Class "Artigliere" Frigates were delivered to the Italian Navy. Once more the propulsion plant is a CODOG configuration.

Besides the aforementioned ships for the Italian Navy, FiatAvio supplied the propulsion system of some ships for foreign Navies also: Venezuela (six Frigates), and Peru (four Frigates).

7 - FIAT AVIO EXPERIENCE IN TIRRENIA FAST FERRIES

In 1997 FiatAvio obtained an order for the supply of the propulsion modules, and relevant auxiliary equipment, to power four fast ferries, built by Fincantieri for Tirrenia, for service in the Mediterranean Sea. The Program was named Jupiter MDV 3000.

This was the first commercial application supplied by FiatAvio for a marine propulsion plant, and it was a good opportunity to transfer to the commercial field the past experience gained both as a marine and as an industrial or power generation packager of the Fiat/GE LM2500 gas turbine.

The total order was for two + two Fast Ferries, built by Fincantieri in two of their shipyards: Riva Trigoso, close to Genoa, for the hull and main structures, and Muggiano, close to La Spezia, for the completion of the ship.

The first couple of ferries (namely Aries and Taurus) were launched January 10 and February 28, 1998, and entered into service in June 1998.

The second two sister ferries (Capricorn and Scorpio) were launched December 11, 1998 and March 17, 1999, and entered into service in June 1999.

Each ferry is a 1,200 tons dwt class of monohull, 145.6 m overall length, 22 m beam. They are capable to carry on up to 1,800 passengers, 460 cars and 30 trucks. Top speed recorded during sea trials was 44.5 knots on "Taurus" with 800 tons deadweight and power output close to maximum (70 MW); service speed however will not exceed the contractual 40-40.5 knots at 1,200 tons deadweight (using 90% of total MCR), 33 knots using all the diesels and one gas turbine (for overnight sailing), or 23 knots when sailing on diesel engines only.

Crew number is 30 to 34 people.

7.1 – Propulsion plant configuration

The propulsion plant layout of each vessel is depicted in Fig. 3.

It is composed of two Fiat/GE LM2500 gas turbines, rated 22.2 MW nominal each, and four MTU 20V1163 TB73L diesel engines, rated 6.5 MW nominal each, in a CODAG configuration.

Two engine rooms (fore and aft), share the total propulsion plant. Each engine room contains one gas turbine, located on the vessel centerline, and two propulsion diesel engines.

Four KA.ME.WA waterjets provide for the necessary thrust for vessel motion: two central waterjets (180 BII) for boost speed, driven by the gas turbines, two side steering waterjets (180 SII) for lower speed and maneuvering, each driven by two of the four diesel engines.

A total of five smaller diesel engines (850 kW each) are coupled to electric generators for on board power generation.

Each gas turbine is coupled to a MAAG reduction gearbox with a reduction ratio of 3600 to 400 r.p.m.; the two reduction gearboxes, located one in the aft and one in the fore engine room, are separated by a bulkhead, and may be connected by means of a coupling shaft for reduced speed cruise during night sail: one single gas turbine can drive both booster waterjets through the reduction gearboxes for low thrust.

7.2 – Gas turbine module

Each gas turbine module consists of a GE LM2500 marine gas turbine housed in a base mounted enclosure.

7.2.1 – Gas turbine

The GE LM2500 gas turbine (Fig. 4) is based on the TF39 (the first turbofan with a high by-pass ratio, used on Air-Force C5A/B) and the CF6-6 aircraft engine design. It is a simple cycle, two shafts engine, composed of a gas generator and a power turbine, with the shaft power transmitted to the reduction gearbox by means of a flexible coupling extending through the exhaust duct.

The gas generator is formed of a variable geometry 16 stages axial flow compressor, an annular combustion chamber with 30 fuel nozzles, a 2 stages high pressure turbine, an accessory drive gearbox and relevant controls.

The power turbine is a 6 stages, low pressure turbine which is aerodynamically coupled to the gas generator and is driven by the gas generator exhaust flow. The power turbine shaft, mounted on a roller bearing and a ball bearing, transmits its power to the reduction gearbox through a coupling shaft, capable of absorbing angular and axial misalignments due to shafts thermal growth and to operating loads.

Hot gases exhausted by the power turbine are conveyed to the exhaust duct through a gas collector, connected to the exhaust duct by means of a flexible joint, for vibration damping and to allow for movements of the engine enclosure relative to the exhaust duct.

7.2.2 – Gas turbine base

The base is a frame that supports both the gas turbine and its enclosure. It provides for proper stiffness to the gas turbine mount system to withstand operating loads, as well as external loads due to hull movements and ship maneuvering.

It is provided with resilient mounts for vibrations and structureborne noise attenuation; the resilient mounts are assembled on the lower surface of the base, and connected to the ship' s foundations.

Service interface connections can be located at the bottom or at each side of the module base. The base is provided with drain holes to eliminate waste liquids (water, oil, fuel) that could eventually leak or flood the base during engine operation.

7.2.3 – Gas turbine enclosure

The enclosure houses the inlet air plenum, the gas turbine and the exhaust gas collector, and provides for thermal and acoustic insulation of the engine.

Combustion air is drawn through an intake duct which is equipped with moisture separators and a silencer. Intake duct is connected to the gas turbine enclosure by means of a flexible joint for

vibration damping and to allow for movements of the enclosure, relative to the inlet duct, due to ship's movements.

The inlet air plenum houses a FOD screen, that prevents the compressor blades from being damaged by any object, and the inlet bellmouth, that helps air flow uniform distribution at the compressor inlet section.

The middle section of the enclosure houses the gas turbine, duly mounted on supports and mount links, that are properly sized to withstand external loads (ship' s movement or shock loads in case of a military ship) while allowing for gas turbine thermal growth.

Air temperature inside the enclosure, during engine operation, would rise to values that are not acceptable for the electronics and the instrumentation assembled on the engine; for this reason external ventilation of the engine is necessary.

This is accomplished by means of an electric motor driven fan, located in a dedicated duct, that pumps external air, duly filtered and silenced, inside the enclosure. A vent damper, connected to the Turbine Control System, allows fresh air to enter the gas turbine compartment, and cuts air flow in case a fire is detected inside the enclosure by the on module fire detection system. Compartment cooling air is discharged through the same exhaust duct of the hot gases.

The exhaust section of the enclosure contains an exhaust gas collector, that conveys hot gases and compartment cooling air to the exhaust duct.

7.3 – Auxiliary equipment

The gas turbine is equipped with the following auxiliaries:

1. Lube Oil System
2. Fuel Forwarding System
3. Gas Turbine Starting System
4. Compressor Water Washing system
5. Off module Fire Protection System
6. Air intake and ventilation system
7. Turbine Control System

Next paragraphs will give a brief overview of each system.

7.3.1 – Lube oil system

The task of the system is:

- to provide oil to lubricate the gas turbine main bearings, located inside special receptacles in the compressor or turbine frames, called "sumps", and the components of the turbine accessory gearbox;
- to continuously filter and cool the lube oil in order to assure proper and safe use of the turbine.

Lube oil is also used as a servo fluid to power the hydraulic actuator of the compressor variable geometry vanes (VSV).

7.3.2 – Fuel forwarding system

From the ship' s service tank the fuel is pumped to the gas turbine after having been filtered in two duplex filters; the first one is a 50 μm nominal filter, the second one is a 5 μm nominal filter, with a coalescer stage for water removal.

Instrumentation and safety equipment complete the off module fuel system. Metering of the fuel is accomplished by means of the off-engine FMVA, located inside the module.

7.3.3 – Gas turbine starting system

The starting system of the gas turbine is of the hydraulic type, composed of an engine mounted starter and an off-module hydraulic power supply.

The starter is a fixed displacement hydraulic motor that receives high pressure oil (360 bar maximum with a flow rate of 210 l/min), generates motion of a splined shaft that supplies mechanical torque to the gears of the accessory gearbox for turbine motoring.

The off module power supply is described in more details in § 9.2

7.3.4 – Compressor water wash system

It is composed of a tank for water/detergent solution or rinse water, valves, instrumentation for local or remote control of washing sequence.

Fluid for compressor water washing operation is conveyed to the gas turbine module by air pressurization of the tank.

7.3.5 – Off module fire protection system

Each module is provided with a cabinet containing four bottles of CO_2, a manifold, safety valves and solenoid valves for gas release.

Release of the gas is accomplished through the control panel, with local and/or remote command.

7.3.6 – Air intake and ventilation system

Combustion air must be duly filtered before being conveyed inside the compressor section of the gas turbine. This is accomplished by means of a three stages filter located at the entrance of the air intake. The inlet duct is provided with a silencer for airborne noise attenuation.

Ventilation air is conveyed through a separated duct; this flow also is duly filtered through a three stages filter and silenced. Inside the ventilation duct an axial, electric driven, two counter rotating stages fan provides for necessary air flow rate to keep compartment temperature within safe limits.

7.3.7 – Turbine control system

Operation and control of gas turbine function is accomplished by means of the TCS.

TCS allows both local and remote (from the bridge) control and command of the gas turbines and of all the auxiliary equipment.

Hardware for the TCS is housed inside two cabinets, one for each gas turbine, located close to the gas turbine modules in the engine rooms. From each cabinet it is possible to control operation of both gas turbines, but only local engine can be operated. From the bridge it is possible to do all possible functions on both gas turbines.

Software for the TCS was specifically developed for Tirrenia Fast Ferries by FiatAvio (SEPA). It is capable to perform all necessary functions: gas turbine and auxiliary equipment operation, control, engine start/stop sequence function, protection, supervision, maintenance (self test).

Detailed description of the system can be found in a separate section of this symposium.

8 - MAINTENANCE

Being an aeroderivative engine, LM2500 gas turbine was designed taking in mind the need of frequent inspections; hence ease of intervention is one of its mostly appreciated characteristics.

In fact modular engine design allows rapid change-out of controls, instrumentation, accessories, and replacement of major components. Provision of abundant borescope ports allows inspection of the most critical engine components, which is a good practice to keep engine health status under control.

On the other hand TCS allows for continuous monitoring and recording of the engines operating parameters (performances), to take under control engine degradation or eventual unforeseen shift of parameters towards limit values.

Tests and inspections, used in conjunction with the monitoring of engine parameters during operation, are designed to alert the operator to conditions which might call for the performance of corrective maintenance tasks.

Maintenance of the GE LM2500 engine is based on a "on condition" concept. This means that great importance is given to preventive maintenance to anticipate, as far as possible, undesirable failures during operation.

Most of maintenance tasks, both preventive and corrective, are performed by specifically trained and skilled personnel; however many of these tasks can be performed by the crew, after suitable training and engine familiarization period.

8.1 – Preventive maintenance

Crew training and shipboard spare parts and tooling recommendations are based on a "Minimum Crew Maintenance" (MCM) concept.

This concept acknowledges that the crew need not be skilled in performing:

- preventive maintenance tasks that have scheduled intervals greater than the ship mission duration;
- on-board corrective maintenance tasks with low probability of occurrence during mission.

There are a total of nearly 40 preventive maintenance tasks, identified by GE, to be performed on the engine; a card number is associated to each task, that clearly explains how to perform the intervention, the procedure and the tools to use. Frequency of these tasks ranges from once per month to once per year.

Minor tasks can require more frequent intervention, but they are limited to very simple operations: compressor water wash, check/fill lube/hydraulic oil levels, check for filters clogging, etc.

MCM concept suggests to train crew and provision the ship with spare parts and tools to perform the so called "first level tasks", i.e.: those preventive maintenance tasks that occur more than twice a year or more than every 600 operating hours. Tasks and number/type of spare parts is dictated by experience and feedback from a very large population of engines (over 1000 only in marine applications), that have cumulated about 7.5 million operating hours as of today.

8.2 – Corrective maintenance

As far as corrective maintenance is concerned over 50 possible tasks have been identified. Statistical studies on occurrence of these tasks show a range from once every 7,000 operating hours to once every 600,000 operating hours.

Twenty-one, out of the over fifty potential corrective tasks, are considered "mission related", i.e.: failure of the associated component can result in inability of the gas turbine to start, reach or maintain the desired power. A "mission related" component is considered "mission essential" if it has less than 97% probability of operating satisfactorily throughout the mission.

The MCM approach suggests training the crew and provisioning the ship with spare parts to accomplish only those corrective maintenance tasks that are "mission essential".

Hence all preventive maintenance tasks with recommended intervals over 6 months or 600 operating hours, and all corrective maintenance tasks that are not "mission essential" are normally delegated to specially skilled personnel at harbor ("second level tasks").

This results in reduced maintenance costs because:

- crew is dedicated to more essential shipboard tasks, with reduced engine room personnel;
- crew training is reduced to a minimum;
- quantity of spare parts on board is minimized;
- specialist teams can be used at harbor facility on a "rotating pool" basis for the complete fleet.

8.3 – Engine removal

Most of maintenance tasks can be performed on-board. The engine and the module enclosure are designed in order to allow for inspections and on-board repair and/or parts replacement without engine tear-down; in fact it is possible to remove major components, like the accessory gearbox or the combustion chamber, or to replace compressor or HP turbine blades and vanes with no need of engine removal.

Removable side panels allow access inside the enclosure to facilitate large parts replacement or utilization of cumbersome tools. The enclosure is provided also with removable rear panels to allow output coupling shaft alignment with the reduction gearbox.

Engine removal is needed only in case of major repairs ("third level tasks"). This is usually accomplished through the air intake duct, but alternate methods are possible. Tooling for engine removal and replacement has been designed and is available as optional equipment. The U.S. Navy reported that complete engine replacement on DDG963 was performed in 26 hours (at harbor).

Fig. 5 shows engine removal through the air intake duct. A set of rails is placed inside the enclosure and the intake duct; the engine is clamped by a specifically designed tool, and made slide over the rails along the intake duct and outwards through appropriate ports.

In order to benefit of the engine modularity it is very important that also all the installation (engine room and ship general layout) is designed with the aim to facilitate extensive on-board maintenance: the U.S. Navy estimates that use of extensive on-board corrective maintenance has, through year 1990, reduced engine replacement and depot repairs by more than 81%.

9 – GAS TURBINE STARTING SYSTEM

The gas turbine starting system is a means to rotate the gas turbine main shaft from rest to a given speed, in order to perform the following operations:

- engine start up (cranking);
- engine motoring for compressor water wash, fuel line purge, exhaust duct purge;
- engine motoring for turbine cooling following emergency shut-down (if engine restart within 10 minutes is not possible);

- engine motoring for fuel metering valve (FMVA) and VSV calibration.

These operations are generally performed by means of a pneumatic or a hydraulic starter.

The starter is a rotating device mounted on the turbine accessory gearbox, that receives pneumatic or hydraulic power from an external source, and transfers it through the gear train of the accessory gearbox to the turbine main shaft.

Starting sequence of the gas turbine is shown in Fig. 6. Gas generator shaft is rotated and accelerated to 1200 r.p.m. before fuel valves are opened and igniters are energized. Acceleration of the turbine is starter assisted even after flame light off occurs, until gas generator shaft ramps up to a speed (4300-4700 r.p.m. in case of a pneumatic starter, 4000-4500 r.p.m. for a hydraulic starter) where the engine is capable of self sustaining combustion. At that point starter and igniters are de-energized and the gas turbine is capable to reach idle speed (about 4950 r.p.m.).

Engine motoring is starter assisted through a lower speed: 1200 r.p.m. for compressor water washing or purge cycle, 2000 r.p.m. for engine cooling.

9.1 - Pneumatic starter description

The pneumatic starter is basically a single stage axial turbine, that receives compressed air from an external source (usually a number of high pressure air bottles); inside the starter a gearing system reduces rotational speed to the proper range of values, and a pawl-and-ratchet device prevents the starter from being driven by the turbine during normal operation.

Fig. 7 shows a schematic of a pneumatic power supply to the engine mounted starter. The double line to the starter can be reduced to one single line if a two stages regulating valve is used to reduce air pressure to the appropriate levels (241 to 283 kPa for engine start, 131 to 172 kPa for motoring).

9.2 - Hydraulic starter description

The hydraulic starter is a hydraulic motor. It receives power from an external source, generally a circuit with oil pressurized by an hydraulic pump, usually driven by an electric motor or by a diesel engine. Power required to operate the hydraulic pump is in the range of 135 to 150 kW.

Also in this case a pawl-and-ratchet clutch is used to prevent the turbine from driving the starter, once the pressure has been removed from the circuit, following gas generator light off and ramp up to a given speed.

Fig. 8 shows a schematic of a hydraulic power supply to the engine mounted starter.

The hydraulic power supply is composed of an oil reservoir, one or more hydraulic pumps with relevant prime movers (diesel engines or electric motors), a secondary oil circuit with a heat exchanger for oil cooling, valves and oil filters as necessary. The number of hydraulic pumps

depends on the system complexity and on the level of redundancy the design engineer has selected to introduce.

The hydraulic pump is an axial piston, variable displacement pump intended for close circuit applications. The flow rate is proportional to the pump input speed and displacement. A wobble plate continuously changes pump displacement from zero to a maximum; this value is selectable by means of a electro-hydraulic system to increase main circuit flow and pressure to a maximum. A pilot pump first fills the circuit with oil at low pressure (about 2400 kPa); then by opening the proper valves in the circuit the oil flows inside the selected loop, and the wobble plate is tilted to increase oil pressure and flow rate to operate the engine mounted starter.

The hydraulic pump flows oil in the primary loop at a rate of about 210 l/min and at a maximum pressure of 36500 kPa during gas turbine start; oil mass flow rate can be reduced during motoring by changing the wobble plate inclination (pump displacement), to achieve the selected rotating speed of the gas turbine.

A secondary circuit, operated by means of an electric pump, flows a smaller amount of oil (about 8 l/min) inside the starter, to keep it flooded, lubricated, cooled, and to flow back into the reservoir the oil leaked through the starter cylinders.

Moreover the secondary circuit also flows a certain amount of oil (50 l/min) from the reservoir through a heat exchanger and back to the oil tank, to keep oil temperature below the alarm level (depending on the oil characteristics, usually about 60°C).

9.3 – Comparison of the two starting systems

Most of gas turbines installed as main propulsors in military marine application use pneumatic starters. This is mainly due to historical reasons, because:

- gas turbines on aircraft engines, from which many gas turbines for marine applications are derived, are started by means of an APU, a smaller gas turbine that gives compressed air for main engine start-up;
- the first ships in which gas turbines were installed were military combatant ships, where compressed air is available in large quantities for combat systems.

FIAT Avio experience both in marine applications and in power plant generation shows however that there are several advantages in the use of hydraulic starters.

As a matter of fact the current tendency is to install hydraulic starting systems on military vessels also, due to the many advantages that this solution offers. Only hydraulic starters have been installed for commercial applications.

Examples of hydraulic starting systems recently installed on board of military vessels are: U.S. Navy SEALIFT Program (7 transportation ships, 2 of which already in service), Korean Corvettes, Korean Destroyers, Canadian Patrol Frigates, Danish Corvettes.

Main advantages of the pneumatic starting systems are the following:

1. air availability: in military ships high pressure compressed air is normally available; this does not require installation of an extra equipment.

2. logistics: Navies that have already used pneumatic starters in previous applications could be reluctant to switch to a new system, because of the extra spare parts to stock.

3. cross connection start: for ships with two or more gas turbines installed on board there is the possibility to start or to motor all the other engines using compressed air bled out of the first one. This requires cross connection between the engines.

Main disadvantages of the pneumatic starting systems however are the following:

1. weight and space: pneumatic starting systems have a strong impact on the general ship arrangement due to the fact that air bottles, needed for compressed air storage, are quite heavy and cumbersome. Adequate room and passage must be provided for their removal for periodic maintenance. The on board compressor must be operated to refill the bottles.

   Air bottles commonly used on military engines have a capacity of 850 liters and a weight of 2000 kg each; when filled with air at 28 MPa they contain approx. 280 kg of air (at 20 °C). Air consumption during normal operation is typically as follows:

- engine start: 70 to 90 kg
- purge of the hot engine section and exhaust duct (following a maximum of 5 to 6 failed engine starts): 90 to 125 kg
- compressor water washing: 200 to 300 kg
- engine motoring for turbine cooling (if a restart within 10 minutes is not possible following emergency shut-down): 700 to 900 kg

   Number and dimensions of the air bottles is a function the air storage pressure and temperature, which depends on the characteristics of the air compressor installed on board, and of the air stock deemed necessary as a minimum for safe ship operation.

   As an average, for a vessel powered by means of four gas turbines, a pneumatic starting system is approximately three times as heavy as the hydraulic starting system for the same propulsion plant: over 9300 kg vs. nearly 2800 kg (see ref. 3).

2. sensitivity to ambient temperature: air surrounding the pneumatic starter must be kept below 82°C, which is normally accomplished by the compartment cooling system; the hydraulic starter is less sensitive to ambient conditions.

3. availability: pneumatic starters cannot be operated continuously in that they need frequent cool down cycles. The duty cycle of a pneumatic starter is as follows:

   Starting mode: 45 seconds on, 2 minutes off (for any number of cycles)
   or
   2 min on, 5 min off, 2 min on, 21 min off (for any number of cycles)

Motoring mode 5 min on, 2 min off, 5 min on, 18 min off (for any number of cycles)
or
10 min on, 20 min off (for any number of cycles)

In case of aborted start the duty cycle must be taken into account before a further start is attempted, which is hardly acceptable in military ships.

Hydraulic starters do not have any time limitation, provided that oil temperature is kept below 70 °C, which is normally accomplished by an automatic cooling system.

4. VSV calibration: limitation of the pneumatic starter to be operated for prolonged motoring strongly reduces its ability to be used for FMVA and VSV initial calibration. This operation requires prolonged gas turbine motoring, and may take several minutes. The hydraulic starter does not pose any problem in this.

5. plant complication: compressed air must be expanded to proper pressure (241 to 283 kPa for engine start, 131 to 172 kPa for motoring). This is accomplished by means of a reduction and regulating valve system; sometimes a two stages valve replaces a double line for the two needed pressure levels. In hydraulic starting systems gas generator speed (at engine start up or motoring) is selected by simply changing the tilting of the swashplate.

6. starter sensitivity: pneumatic starters are equipped with a double (redundant) safety device to avoid starter blast in case of overspeed: this system cuts air flow in case of failure of the clutch to disengage the starter, and physically cuts starter rotor blades in case of failure of the first system. An hydraulic starter is not affected by this problem.

7. maintenance: if not installed for other purposes, the air compressor needed to refill air bottles is a component that requires care and frequent maintenance.

8. cost impact: due to the air bottles and reduction valves, initial cost of a pneumatic starting system results to be about 40% higher than the one of a hydraulic system.

Moreover hydraulic starters have the advantage of availability; in fact the system is not dependent on the availability of other equipment (compressed air). In case of blackout an emergency electric generator or diesel engine would give power enough to start the turbine.

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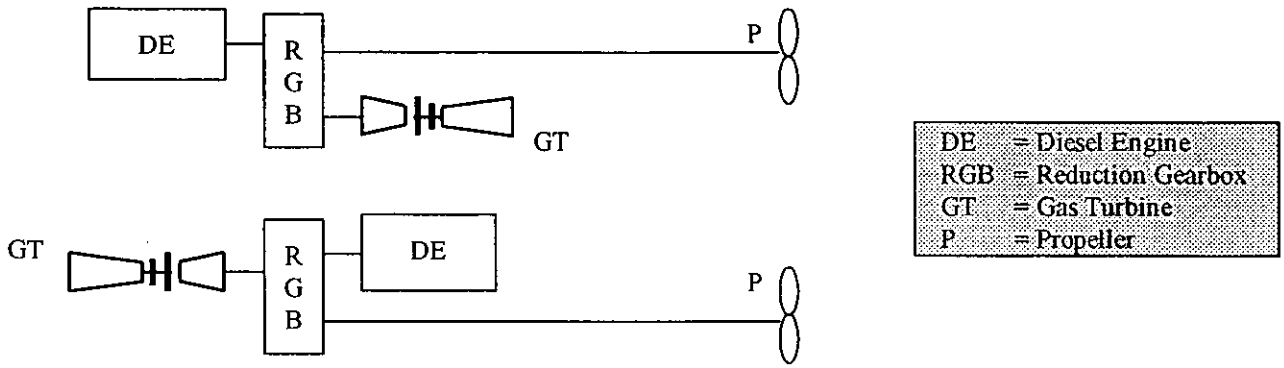


Fig. 1 - CODOG power plant configuration

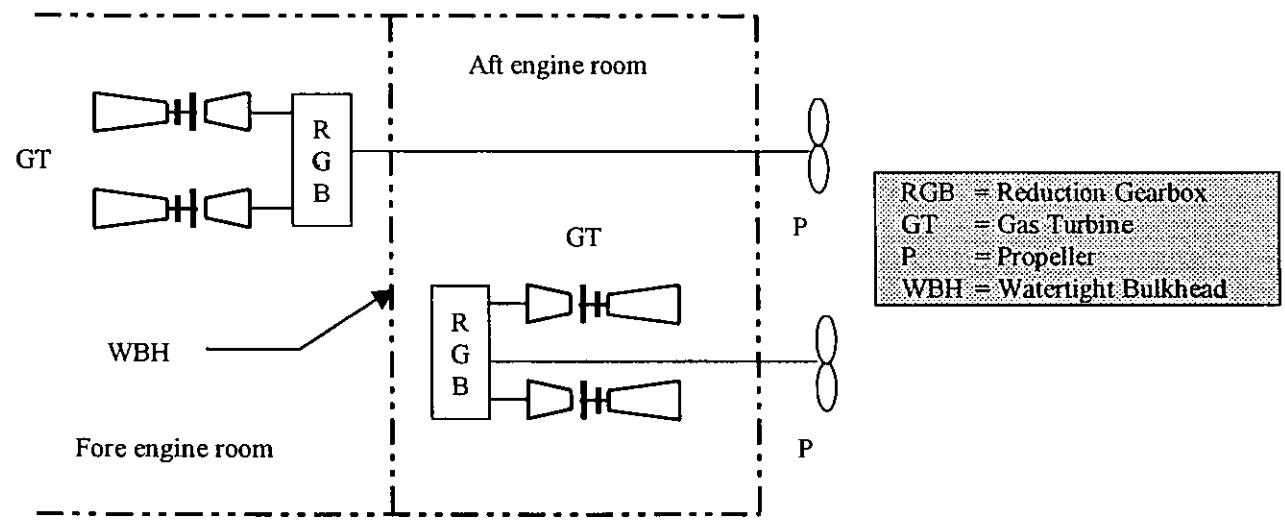


Fig. 2 - COGAG power plant configuration

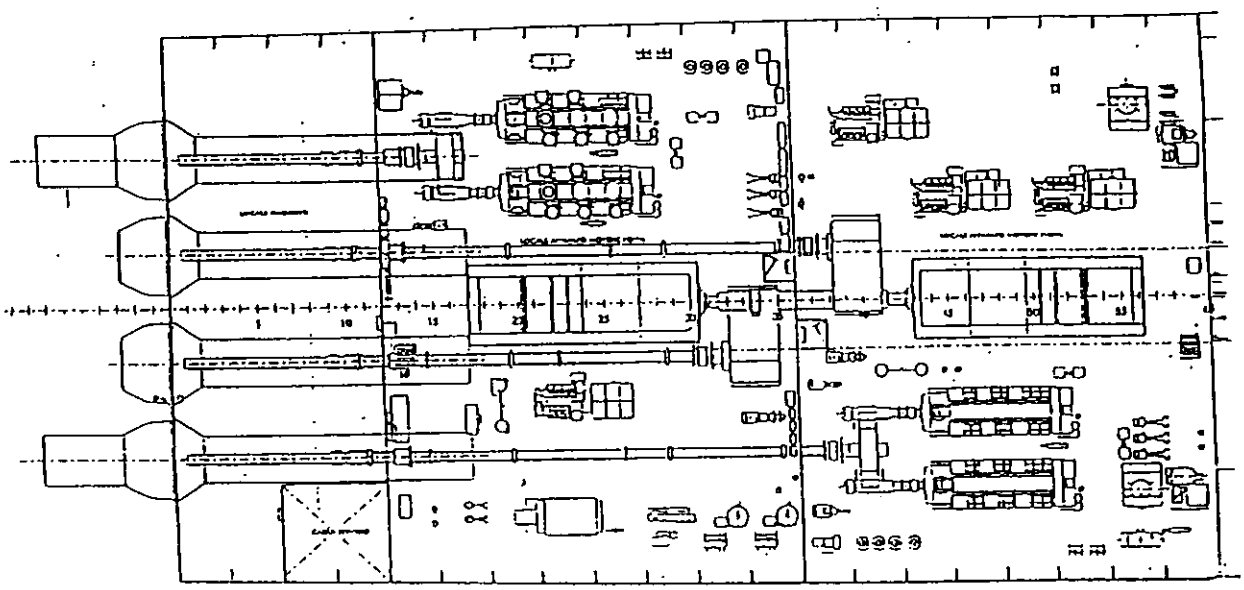


Fig. 3 - Tirrenia Fast Ferry propulsion plant

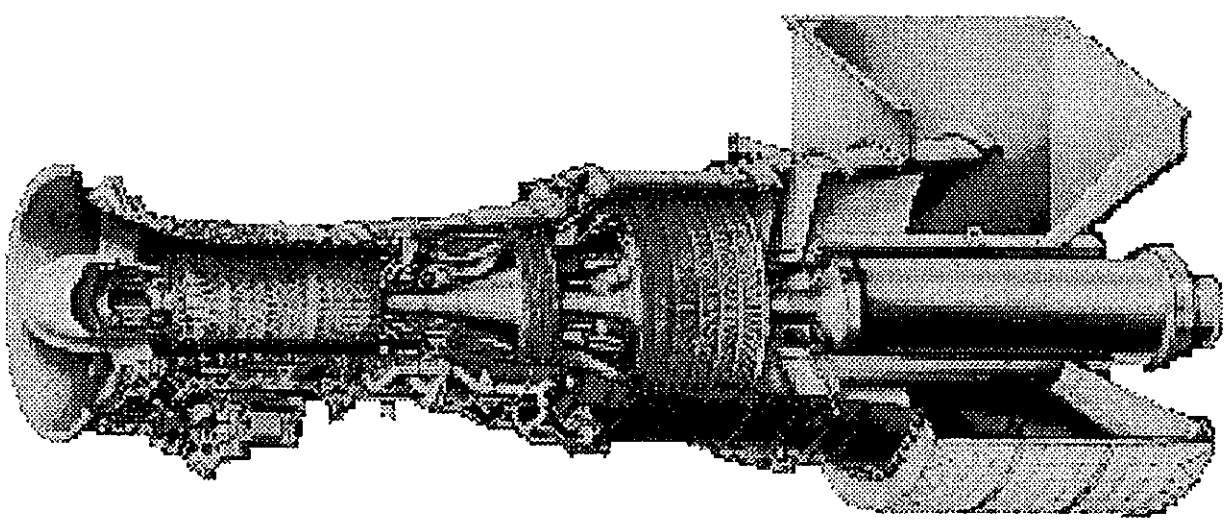


Fig. 4 – GE LM2500 Gas Turbine

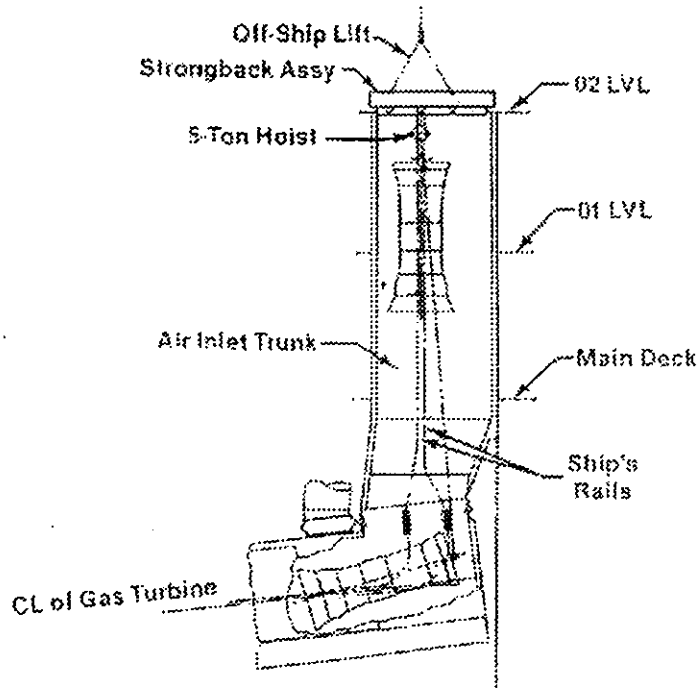


Fig. 5 – Engine removal through air intake duct

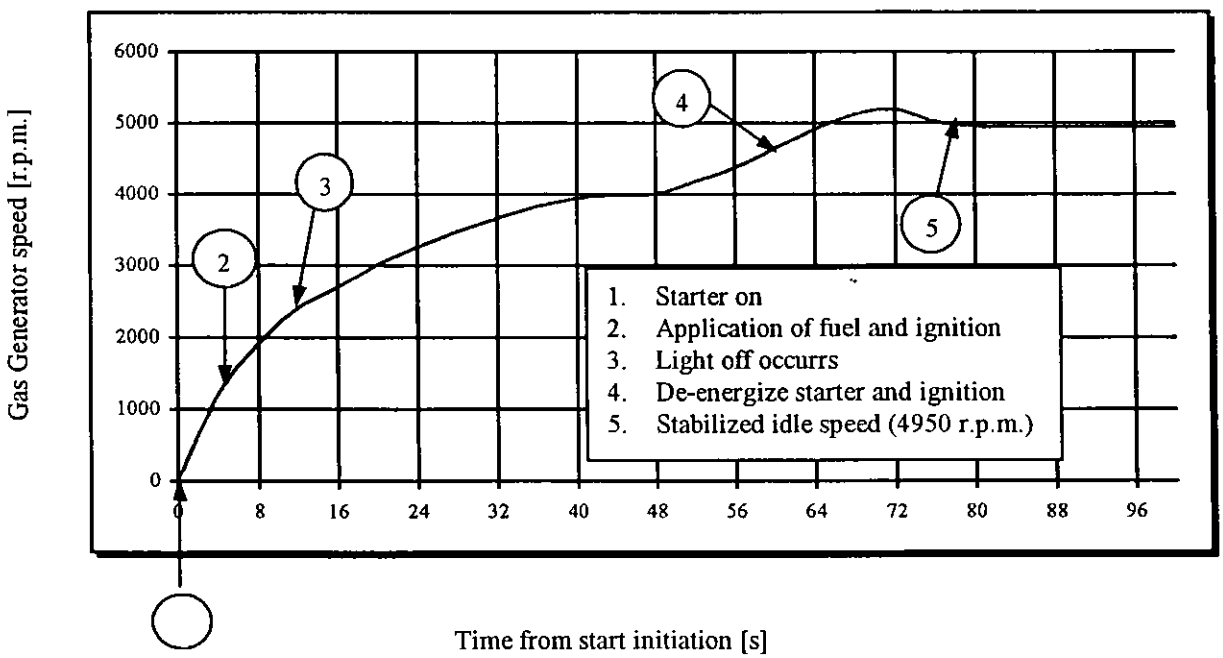


Fig. 6 – LM2500 start sequence (liquid fuel)

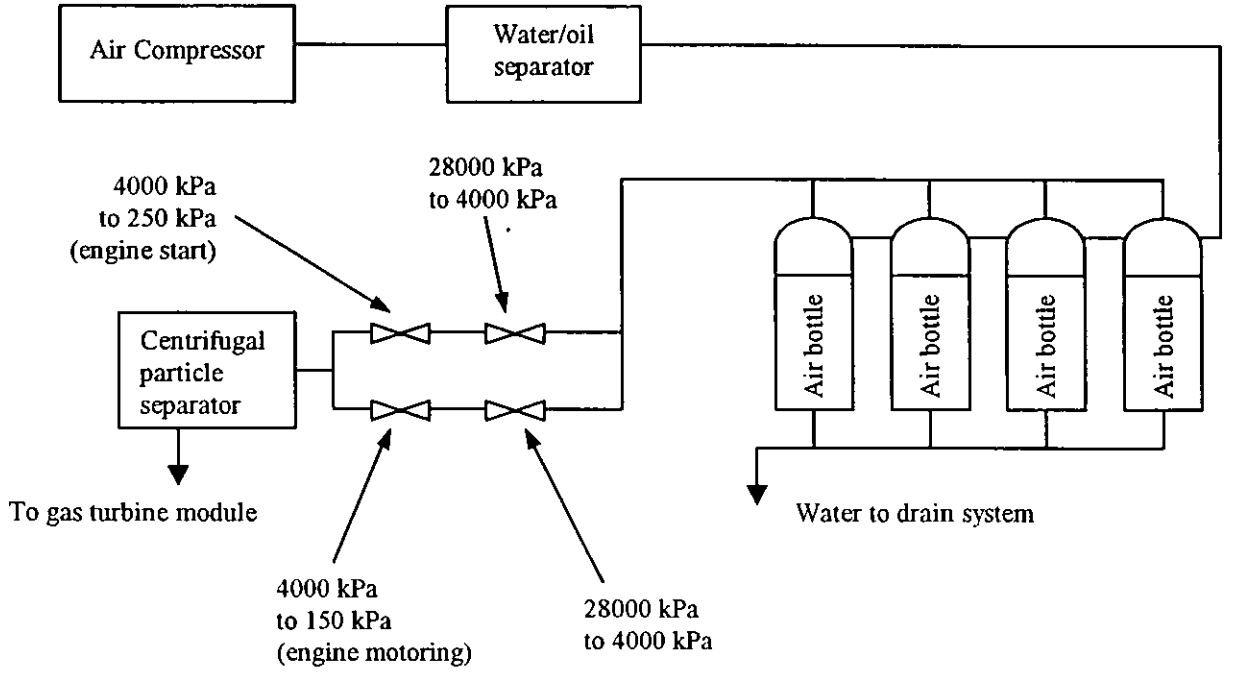


Fig. 7 – Pneumatic power supply to the turbine starter (schematic)

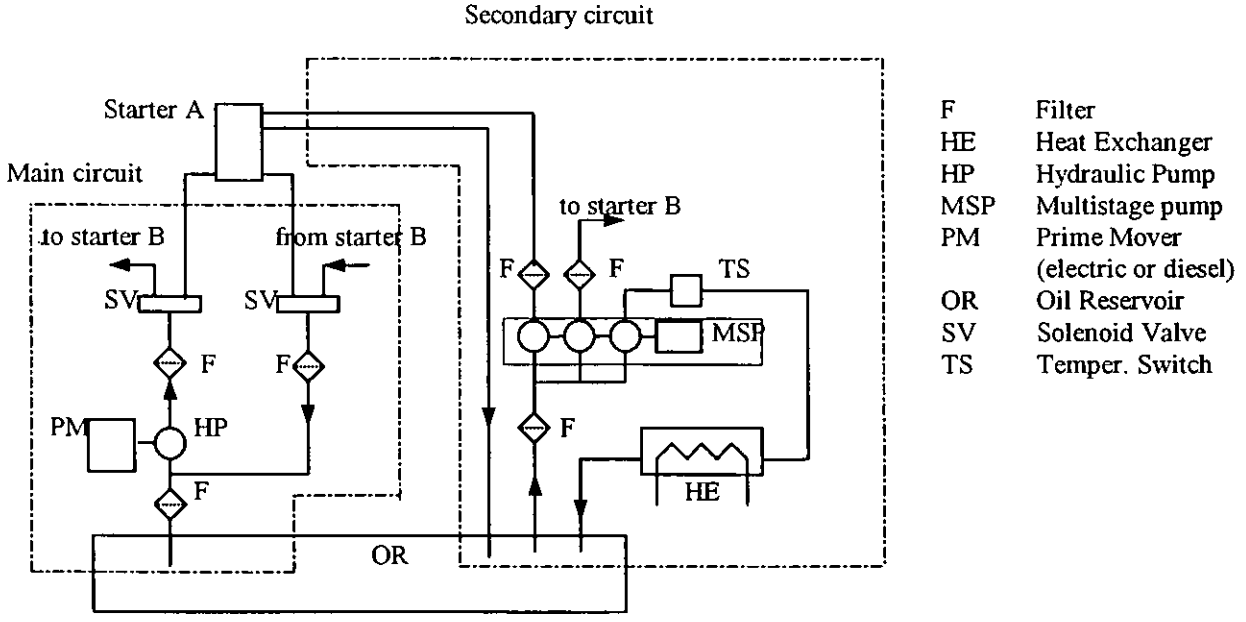


Fig. 8 – Schematic of hydraulic power supply to the turbine starter (as realized for Tirrenia Fast Ferries)

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
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GAS TURBINE FOR MARINE ELECTRIC PROPULSION

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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ABSTRACT

Some "*all electric*" cruise vessels are entering in service. Eventually electric propulsion seems ready to become operative for some specific marine civil and military applications.
This paper presents an overview of the main components of electric propulsion and their configurations, and describes its the advantages underlining the role and the benefits of the gas turbine foreseen for this type of propulsion.

The content is based on the multidisciplinary experience that FiatAvio has gained in the last decades with the General Electric LM2500 gas turbine applied to marine propulsion and industrial power plants and with the participation to NATO All Electric Ship Pre-Feasibility Study carried out in 1997-98. In particular it is believed that the *industrial* experience is very precious and can be easily "marinised". In fact this kind of application, apart from using the same kind of gas turbine, has widely proved all necessary systems that are the basis for the all-electric ship configuration.

1 - INTRODUCTION

A success product tends to migrate to different fields enhancing its characteristics and gaining advantages from this cross-fertilisation.
The gas turbine GE LM2500 is an example of this. It is called "aeroderivated" since derived directly from the CF6-6 avio jet engine. Its good features (performance, low weight and volume, easy maintenance, low emissions and high availability) induced GE to marinise it. It came out the LM2500 that has been used as a primary choice for marine propulsion for many decades. Good results as mechanical drive has led its use for industrial applications, firstly for pumping stations secondly for power generation plants. Once more this gas turbine has been a good friend in the research of high efficiency easily permitting the installation of combined cycles where the efficiency-loss have been reversed.
Now marine propulsion is moving to All Electric Propulsion (AES). And gas turbine with all the experience gained from industrial power plant is once more ready to play a primary role in this type of configuration.

2- NOMENCLATURE

AC	Alternate Current
AES	All Electric Ship
COGES	COmbined Gas turbine, Electric drive, Steam turbine
CODAG	COmbined Diesel And Gas
COGAG	COmbined Gas And Gas
COGOG	COmbined Gas Or Gas
COTS	Components Off The Shelf
DC	Direct Current
GT	Gas Turbine
IMO	International Maritime Organisation
IPMS	Integrated Platform Management System
LC	Load Centre
LV	Low Voltage
MD	Main Diesel
MV	Medium Voltage
MW	Mega Watt
MTBF	Mean Time Between Failure
MTTR	Mean Time To Repair
PWM	Pulse Width Modulation
VTOL	Vertical Take Off and Landing
VSV	Variable Stator Vanes

3 - MARINE PROPULSION CONFIGURATIONS

The history of the all electric vessel runs through the diverse configuration of the propulsion system. In this light an exhaustive survey of the different available system is necessary to fully understand the all electric advantages.

Low weight, high specific power and flexibility have been the key factors of the gas turbine success as prime mover for marine application. They have revolutionised ship design allowing faster and more versatile military and commercial vessels in many different propulsion configurations.

To obtain cost reduction and different operating profiles in military application CODOG (COmbined Diesel Or Gas), COGOG (COmbined Gas Or Gas), COGAG (COmbined Gas And Gas) configuration have been used.
CODOG utilises on the same axis a diesel or a gas turbine (**Fig. 1**). The low power diesel is used for low speed while the gas turbine for high speed. This system with variable pitch propeller is the ideal propulsion for ships in the range from corvettes to destroyers. Similar solution is the COGOG that uses on the same shaft a small gas turbine and a large one.

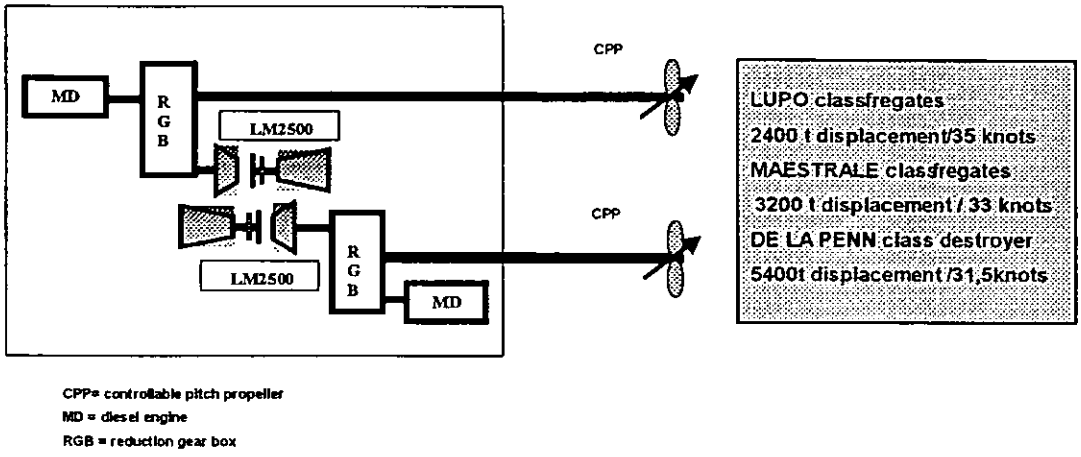


Fig. 1 - CODOG - FiatAvio installation for Frigates and Destroyers

The COGAG instead uses two turbines of the same power and it is typical for destroyers or frigates operating in oceanic missions and aircraft carriers for helicopters and VTOL's (**Fig. 2**).

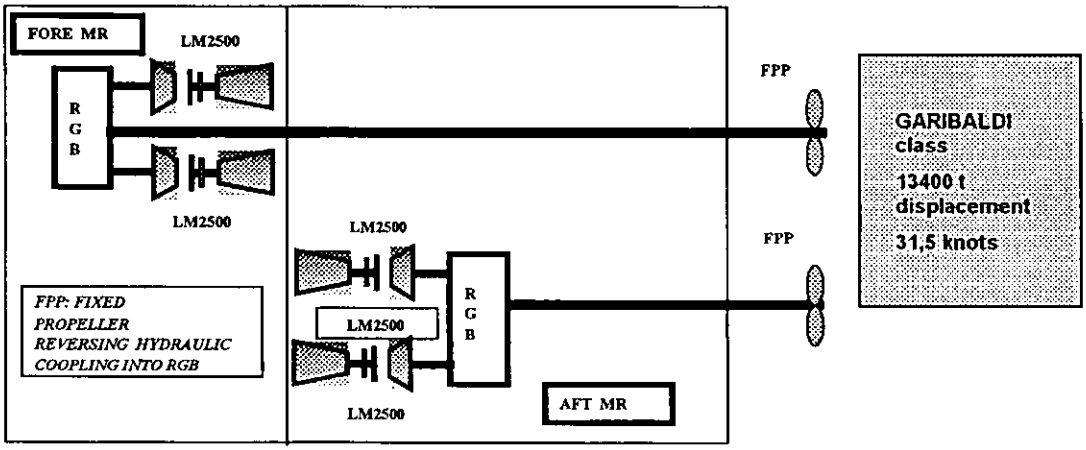


Fig. 2 - COGAG –FiatAvio installation for Aircraft Carrier

On the other way commercial markets are today demanding faster conveyance of goods and passengers and shipbuilders in some applications are replacing propellers with powerful water-jets.

Instead of the classic helm, the ship is manoeuvred by diverging the water flow with the nozzle of the water-jets and with a ducted propeller at the bow. This allows the ship to self-rotate, manoeuvring and docking easily and rapidly also in difficult conditions. This water-jet can also reverse the water by inverting the pitch (on variable pitch propeller) or inverting the shaft rotation (on fixed pitch propeller).

Using this approach fast ferries have been built which are capable of halving the navigation time necessary to connect islands or sites on the coast.

The configuration used are CODAG and COGAG: the first uses small diesels coupled to steering water-jets with thrust reversers and more powerful gas turbines to drive the booster water-jets. These ferries can cruise at up to 40 knots carrying 450 cars and 1800 passengers. The second configuration utilises small gas turbines to drive the steering water-jets and larger turbines for boosters. This configuration has permitted to build ferries capable of speeds up 50 knots (**Fig. 3**).

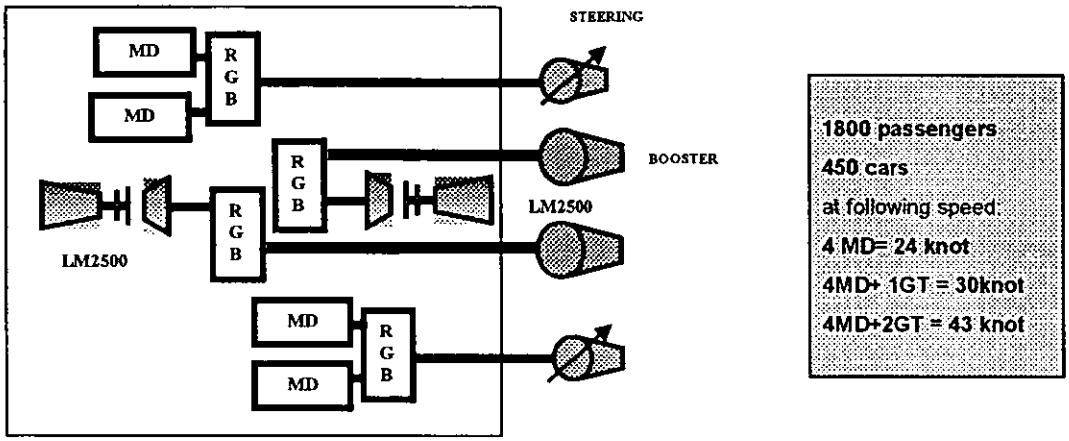


Fig. 3 - CODAG –FiatAvio application for Tirrenia Fast ferry-MDV 3000

For the future the booster water-jet, now built with a power of around 20 MW, will be enhanced up to 40 MW allowing the use of more powerful and more efficient gas turbines to move ships of increased tonnage or even ekranplanes capable of 100 – 140 knots.

Now a new architecture, very similar to the basic combined cogeneration (**Fig. 4**) and considered unfeasible until some time ago for marine, is appearing on the commercial ship. It is the base for the AES and it is provided by the installation of a COGES (COmbined Gas turbine, Electric drive, Steam turbine). And once more this type of propulsion will rely for the short and medium period on gas turbines.

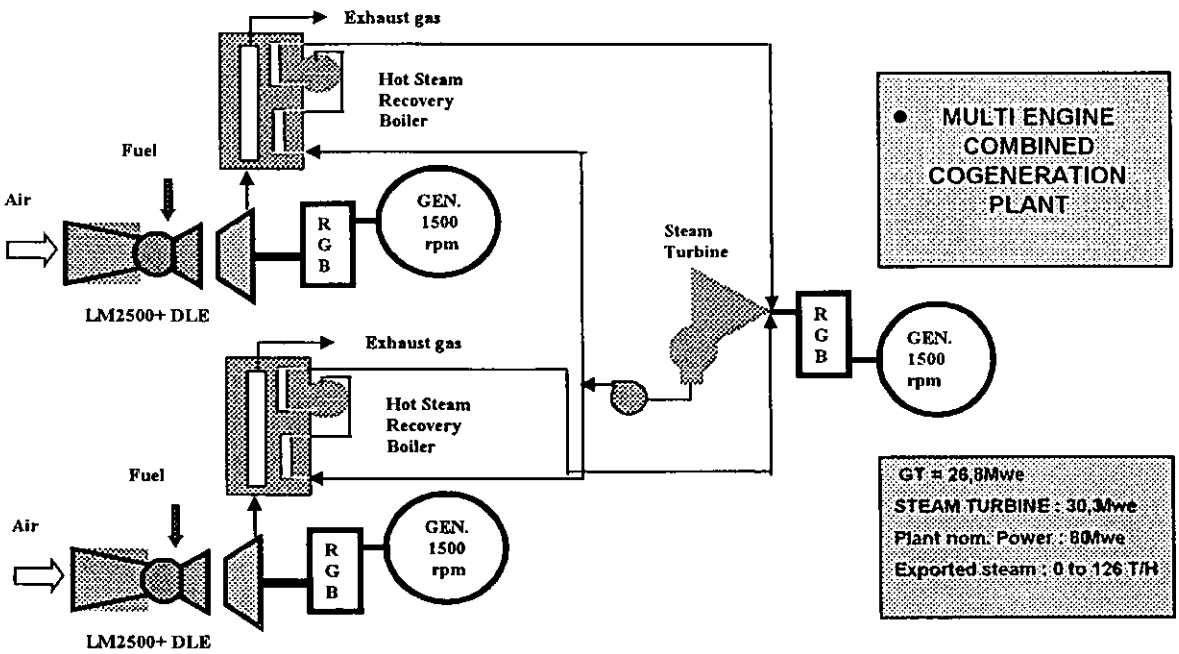


Fig. 4 - Basic combined cogeneration

4 - ELECTRIC PROPULSION

It was running the first decades of the '900 and the idea of "all electric" was applied on some US Navy such as Neptune and New Mexico. The concept embodies the use of electric means for all power needs in lieu of other means such as mechanical, pneumatic and hydraulic.
Some more US cargo carriers were built during War World II and since the 50's icebreakers and offshore vessels took advantage of the inherent torque characteristic and high controllability at low speed of electric drives. But the technology of those days was immature and on board electric generation and motors were wide and heavy. Hence the electric propulsion was not so attractive and it was generally used for special purposes only.

With the introduction of the AC drive technology in the 80's, electric propulsion became more interesting again. Alternators and electric motors became lighter, more efficient and easy to drive, even the most powerful ones. Moreover a new propulsion group, electric motor and propeller called *Azipod*, capable to rotate 360 degrees, was produced and used to easily manoeuvre large commercial ships.
Recently electric propulsion started to be used for cruise liners and from different studies carried out in Europe and US it seems to be the method of propulsion for the next generation of naval vessels.

4.1 - AES sub-systems

AES means that all the energy produced by the prime movers, gas turbines or diesel, are converted into electric power. Electricity is then delivered to the users: mainly propulsion but also ship's services and hotel loads for civil application; electric armour, electric weapons etc. for military vessels.
The overall design must minimise the number of power generations supporting the ship priorities at any time. Each power unit must be capable to supply energy to either propulsion or ship services.

This approach brings significant gains in fuel consumption and in maintenance cost due to minimised engine running hours. However many different systems must be carefully selected and integrated; some technologies are promising but require development time.

The shipboard power system can be divided into the following sub-systems (**Fig. 5**):

1) Power Generation
2) Power Conversion
3) Power Distribution
4) Propulsion Motors
5) Integrated Platform Management System (IPMS)

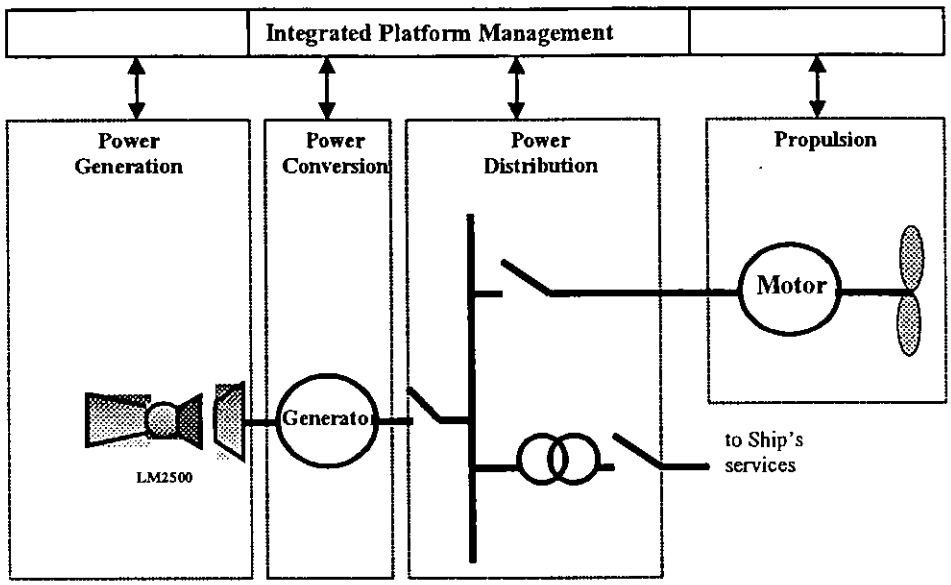


Fig. 5 - Main blocks for the All Electric Ship

4.1.1 - Power Generation

The Power Generation is the sub-system deputed to the production of power. Today technology relies on diesel and gas turbine. Diesel engine are available up to 8 MW and generally are more efficient, less expensive, require less intake/exhaust air requirements but are heavier, less powerful and more noisy than gas turbine's. These are available in the range of 7 to 29 MW (**Fig. 6**) suiting ship operating profile, redundancy and availability.
Fuel cells, that convert chemical energy directly into electrical energy, are very promising (high efficiency, no rotating part, very low emissions, modularity, low noise) but quite immature and their use is considered possible not before 10 years.

Turbine	*Power[1] [MW]*	*Simp./Comb. Gas Turbines*	*Supplier*
Marine WR21	25	Combined	Rolls Royce
ALLISON 601R	6,4	Combined	Rolls Royce
1500 TI	1,5	Combined	Turbomeca
LM1600	14,9	Simple	GE
LM2500	25	Simple	GE
LM2500+	27.6	Simple	GE
Marine Spey 1C	19,5	Simple	Rolls Royce
Allison 601-KF9 / KF1	6,5 / 7,8	Simple	Rolls Royce
Bastan	0,6 / 0,8	Simple	Turbomeca
Makila	1,1	Simple	Turbomeca
Eurodyn	2 / 3	Simple	Turbomeca
Note[1]:ISO cond.			

Fig. 6 - Turbines availability for naval applications

4.1.2 - Power Conversion

The Power Conversion sub-system converts the original mechanical power into electric power by means of an electric machine. This is conventionally performed with induction, synchronous or DC machines with a wide range of operating speeds and voltages.

Generators are generally heavy and cumbersome hence their use for marine application will require effort to reduce volume and weight. The direction is toward higher rotating speed (from 60 Hz to 400Hz), higher voltage and high speed AC generator with embedded rectifier operating into DC distribution network such as squirrel cage induction machine, wound field synchronous machine with brush-less excitation or synchronous machine with permanent magnets.

Generator type	*Length [m]*	*Width [m]*	*Height [m]*	*Weight [ton]*
Conventional synchronous 2 poles (AC)	4,6	2,9	3,2	50
Conventional synchronous 4 poles (AC)	3,9	2,7	3,5	44
Induction generator 4 poles (DC)	3,1	2,0	3,5	30
Permanent magnet synchronous 12 poles (DC)	2,9	2,0	2,0	20

Fig. 7 - Indicative dimension and weight for different technology alternators (22MW @ 3600rpm)

4.1.3 - Power Distribution

The Power Distribution is the system to supply energy around the ship to the different users. This system, providing electricity for propulsion and ship's services, must have some sort of redundancy, robust protection and reliability.
It can be realised in AC, and this is the preferred short-term solution because Medium Voltage (6.6 kV at 60 Hz) switch-gear and fault protection devices are available. Higher voltage (11 kV) may become attractive as well as 120 Hz that allows smaller generators.

4.1.4 - Propulsion Motors

There are several options for the propulsion motors:

- Conventional DC motors
  Simple to operate and control, much used for many years, they have power capability up to 5 MW and require frequent maintenance
- Synchronous AC motors
  Now used with fixed frequency AC bus and controlled through a static converter they have the advantages of being able to deliver net reactive power but they have low torque/weight ratio, high production cost and high excitation losses especially at low speed;
- Asynchronous motors
  Small, light and more rugged than synchronous motors these devices are well consolidated and much used for commercial vessels. They are capable of operating with a cyclo converter or pulse width modulation (PWM) inverter.
- Permanent Magnet motors
  These motors retain the basic advantages of the synchronous ones reducing the disadvantages because excitation is not required, and they can have high numbers of poles (hence smaller, lighter and less cooling required). Cost is higher and the assembly more critical
- Superconducting motors
  There are many expectations in this field due to very high power densities but commercial applications are far away.

4.1.5 - Integrated Platform Management System (IPMS)

Based on advanced digital automation systems the IPMS is fully integrated to provide control and monitoring of the different systems of the ship: power generation/conversion/distribution, propulsion system, auxiliaries and ship services. It can also provide damage control, maintenance on condition and embedded on-board training.
Such systems are technologically available and proven. They are generally distributed system and structured to have high degree of flexibility and redundancy.

5 - AES BENEFITS

Which are the potential benefits? AES looks attractive in the area of cost reduction and ship performance improvement but also for a more environmentally friendly approach.

In details AES ***increases***:

- *fuel efficiency*: the efficiency is better because the engines can operate at their nominal speed while the electric drives vary the speed to follow the ship operational profile;
- *flexibility and adaptability:* the elimination of long shaft lines allows more flexibility of equipment installation. Furthermore the integration of electric power and propulsion provides flexibility to shift power between propulsion and ship services as required. This increases availability as well;
- *operational characteristics:* the electric approach can enable rotational speed control from 0 to 100% with the availability of nominal torque in all the operating conditions and it can eliminate the less efficient controllable pitch propeller in favour of fixed pitch type improving overall manoeuvring (shorter response time);
- *lower signatures and better comfort*: eliminating the mechanical link between the engine and the propeller will allow noise and vibration reduction. Other noise reduction comes from the prime movers running at constant speed, which facilitates engineering for damping the structure-borne noise;
- *survivability*: the modularity and redundancy of the electric solution will allow rapid degradation of vital systems enhancing survivability. Survivability is also improved by the possible segregation of the equipment in different parts of the ship, reducing the possibility of losing propulsion and electric supply in case of damage;
- *space availability*: eliminating gears and shaft will free up space. Moreover the power sources can be located with more freedom not affecting the ship's general arrangement;
- *ship design flexibility*: the prime movers do not have to be aligned with the propeller shaft and can be aligned in any convenient direction;

and **reduces**:

- *manning*: digital automation with remote control and monitoring reduce the requirements for human operators;
- *emissions*: the prime movers, running at constant speed, can work at their optimum efficiency with better combustion hence reducing emissions;
- *maintenance*: as loads are very variable depending on the ship profile the full electric solution will allow an optimisation of the services minimising the number of genset in service. Freedom in the selection of installation can be used to select less hostile

environments thereby increasing MTBF of the equipment and allowing easier access for replacement hence lower MTTR.

On the other side there are also some disadvantages to be taken into account:

- higher purchase cost;
- increase in weight;
- more electric losses;
- potential electromagnetic compatibility (EMC) problems

5.1 - Architectures

The feasibility and benefits of employing a common power system both for propulsion and ship services are now widely accepted in particular for merchant applications (cruise vessels, cable layers, shuttle tankers etc.). In these applications the use of commercial available equipment, the advantage of space and the flexibility of equipment accommodation assure secure advantages but the solution is promising for military vessels as well.

5.1.1 - Cruise vessel example

In this type of cruise ships the hotel power demand is increasing to cope with air conditioning and galleys, show lights, swimming pools up to 1/3 of total power consumption.
Among various configurations the COGES (Combined Gas, Electric drive, Steam turbine) has been considered the most beneficial. Gas turbine is directly coupled to a power generator, the exhaust gas energy is used to produce more additional power from a steam turbine generator set without requiring additional fuel. Moreover the discharged steam before being condensed is used for fresh water production, laundry, air conditioning etc.
This avoids additional boilers and the gas turbine is used at the best in terms of efficiency, emissions and cost.

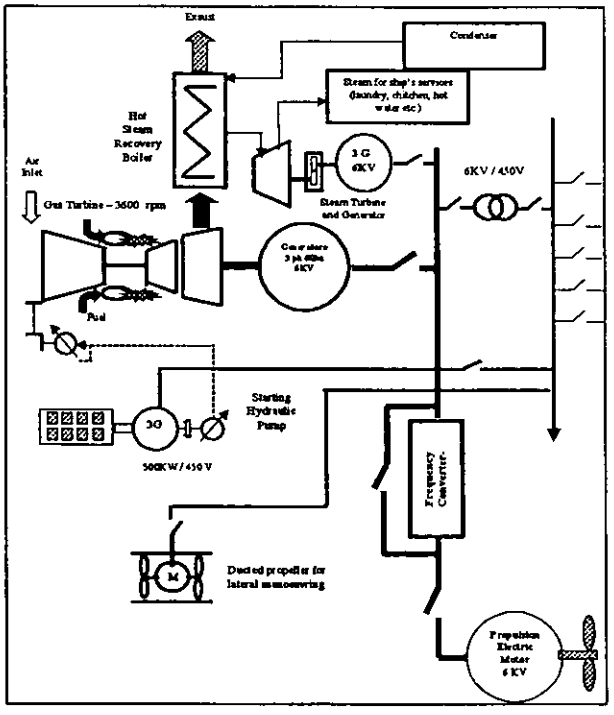


Fig. 8 - Example of combined cycle for cruise ship

5.1.2 - Frigate example

Many of the electric propulsion advantages can be beneficial for military application even if the requirements are different from the cruise liners. However the COGES configuration cannot be applied and electric propulsion is used in a simpler way.
The NATO pre-feasibility study carried out in 1997-98 was targeted for a 4000 ton frigate with full electric propulsion with the following characteristic:

- Over all length: 131 m
- Light / Full Load displacement: 3800 / 4700 ton
- Crew: 125 persons
- Maximum speed: 28 knots
- Overall ship power requirements: 33 MW
- Propulsion power requirement: 28 MW

And the short term architecture was based on existing technology with:

1) two electric propulsion motors of 14 MW each directly coupled to a fixed pitch propeller rated at 170 rpm max according to power/speed curve;
2) the partitioning of generated power among a single main generating set of about 22 MW for medium speed operations and two smaller sets of diesel 5,5 MW each for slow speed operation, stand by and to support the main set for top speed operation. The use of one or two diesels allows ship operation up to 18 knots, which is the speed range assumed for 80% of the mission time. For high speeds the gas turbine provides extra power.
3) one harbour load/emergency generator rated 0,8 – 1,5 MW supplying 440 V 60 Hz

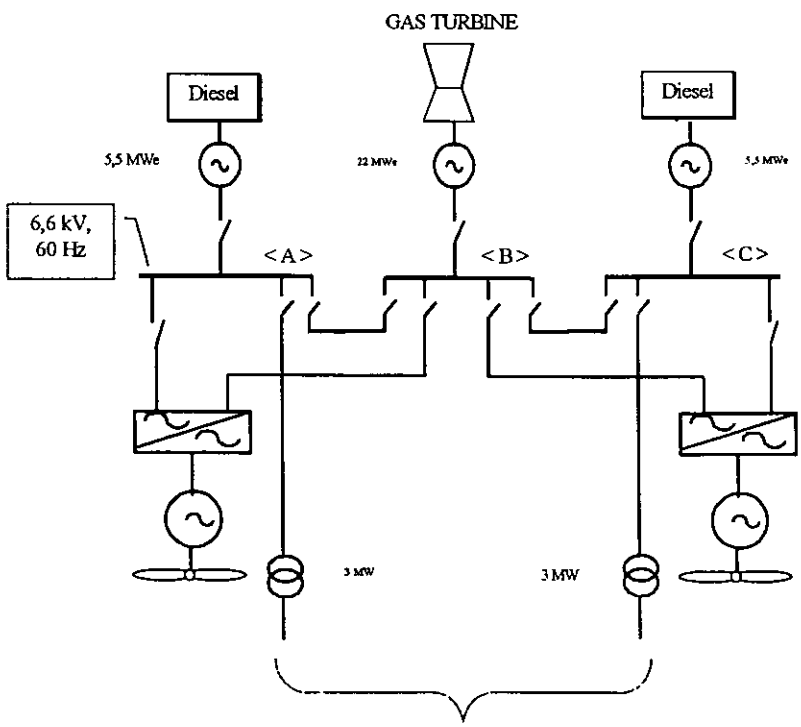


Fig. 9 - Short term configuration for the Frigate

The medium voltage (MV) bus system at 6,6 kV provides the primary supply distribution for the propulsion system and ship's services. Three switchboards (A,B,C), each physically separated with inter connectors allow a wide range of options for paralleling of busses and generators giving redundancy and flexibility.

A low voltage (LV) at 400 V 60 Hz system is fed by two MV/LV transformers and it consists of three switchboards. It is operated normally as an active ring system for reliability reasons. The Load Centres (LC) are supplied from the LV ring system with each LC fed from one side of the ring. Automatic change-over is provided with each LC to enable transfer, in case of loss of supply, to the other input feeder to the LC.

For harbour or emergency situations a small diesel provides an economic source of power.

As a general conclusion of the study it is stated that an all-electric frigate is feasible and it can be successfully developed in the short term with available technology and in the long term with improvements foreseen from different technical evolutions.

6 - THE GAS TURBINE ROLE

In the traditional propulsion systems gas turbines have proved to have:

- high efficiency: a general low initial value has been improved up to 37.5% in simple cycle and much more in combined cycle;
- low maintenance, based on "*on condition*" philosophy, enhanced by modular construction and existing skills and practices;
- low emission;
- high reliability, with mature mean time between engine removals of more than 25000 hours on gas and 15000 on liquid and credible reliability data providing failure modes and rates;
- suitability: gas turbines fit well to power classical propulsion, electric generation, steam production as required by the application. They even behave well when required to operate in high cyclic profiles (3 cycles/day of start-max-stop in fast ferry application) similar to airline;
- high availability, in excess of 99% supported by known failure modes and modular construction.

Besides the above advantages it is believed that they will perform the key role in the AES for:

1) *power*: a unique media to provide "lots of power";
2) *efficiency*: COGES systems based on LM2500 or LM2500+ show an overall fuel burn consumption rate equal or less than that of a medium speed diesel electric system. In fact the exhaust gas turbine energy produce 30% more power than the steam turbine generator. Moreover steam can be extracted from the steam turbine for ship's services (fresh water production, air conditioning etc.);
3) *exhaust emissions*: there is worldwide pressure to reduce emissions. The International Maritime Organisation (IMO) is pushing for a 30% reduction in NO_x emissions for new diesel engines compared with 1990 vintage engines. Gas turbines in COGES configuration have NO_x level that are one third of the proposed IMO levels. Moreover Dry Low Emission

technology for gas turbines, which reduces emissions of a further factor of five, is already developed for industrial land-based applications. Diesels will require exhaust treatment to meet the emission requirements and therefore cost and volume/weight will increase. Emission regulation will also induce the demise of heavy fuels (which produce a lot of NO_x, SO_x and particulate) which is today giving a cost benefit to diesel for lower costs with respect to the distillate;

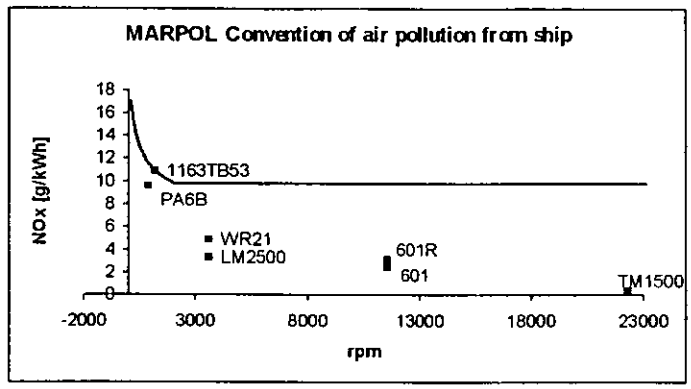


Fig. 10 - NO_x levels for some diesel and gas turbine compared with Marpol limits

4) *intake space*: modern medium speed diesel engines require about 85% of the combustion air required by a LM2500 in COGES configuration. But to generate the same power of two LM2500s, five or more diesels are required. This means the total intake cross section area of the gas turbines is equal or less than the area required for the five diesels;
5) *compactness in space and weight*: gas turbine has a very high power density compared with a medium speed diesel engine. An LM2500 package with acoustic and thermal insulation has about 400 kW/m^3 that is up to five times that of a medium speed diesel. This means space saving
6) *noise*: engine to engine room noise (**airborne engine room noise**) is generally reduced in the gas turbine by the acoustically insulated enclosure (90 dBa attenuation at 1 meter) more than a large medium speed diesel noise. The noise propagated through the engine mounts into the ship (**structure-borne noise**) is lower as well due to the continuous combustion process instead of the heavy multiple cylinders working with intermittent combustion. The **intake noise** is primarily composed of high frequency noise, which is absorbed by intake silencers normally installed in the intake duct in the engine room area without impacting upper decks. The **exhaust noise** is concentrated at low frequency, generally well insulated by the thermal insulation of the exhaust or by exhaust silencers, which do not require much space;
7) *automation*: the rapid progresses made by digital control systems have increased gas turbine friendly operation capability. Gas turbine is easy to start and shut down, rapid to erogate power and automatically monitored. The operator "*forgets*" the turbine and his workload is reduced at minimum through few user-friendly synoptic. This kind of machine seems specifically designed for remote control. Also the maintenance concept is "on condition" and the indicators related to cycles, trip, starts number etc. indicate when maintenance actions must be performed.

7 - CONCLUSIONS

AES offers many theoretical advantages especially in COGES. In the near future they will be demonstrated because electric propulsion is becoming a reality. This application requires good

architectural design and strong integration of different technologies and disciplines. The experience gained from electric industrial plants can be in great part be re-used but it is necessary to tailor and marinize these proven systems. Marinisation will also require some tecnological improvement expecially in the area of alternators and power distribution to reduce weight and dimension. This should not be seen as a limitation but as an opportunity because it means that this field is not mature and there are margins to improve the performance.
Gas turbines have proved to be excellent for classical configurations and once more will play a primary role in this new application.

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MARINE GAS TURBINE CONTROL SYSTEM

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MARINE GAS TURBINE CONTROL SYSTEM

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ABSTRACT

FiatAvio-Sepa have developed and installed many control systems for the General Electric LM2500 since the seventies both for marine propulsion and industrial applications.
The latest is a full digital Electronic Control System (ECS) for the LM2500 used as water jet drive on the Tirrenia Fast Ferries.

Based on this experience an overview of the ECS is presented with a particular emphasis on the safety features and the performance functions (sequences, protections etc.), making this type of prime mover so automated that minimum manning is required.

Moreover the benefits of such system in comparison with the old hydro-mechanical regulators is pointed out.

1 - INTRODUCTION

When the two shaft jet derivative gas turbine General Electric LM2500 appeared on the market for the first marine application the control system was a combinations of hydro-mechanical and electronic control. The first was for the main fuel control (MFC) of the gas generator, the second to command and monitor the engine.

The use of hydro-mechanical can surprise and induce the association to an old complex stuff full of cams, leverage, pipes and orifices like an expensive mechanical Swiss chronograph. It ran the seventies, electronics was a young discipline, and this jet-derived gas turbine used the simple (and safe) jet approach of a hydro-mechanical fuel regulator. The generator speed is sensed, compared with the pilot demand and the error meters the fuel flow into the combustion chambers with an automatic selection of the fuel schedule to provide optimum acceleration and deceleration (Fig. 1). At the same time compressor inlet temperature and discharge temperature were used to properly position the Variable Stator Vane (VSV).

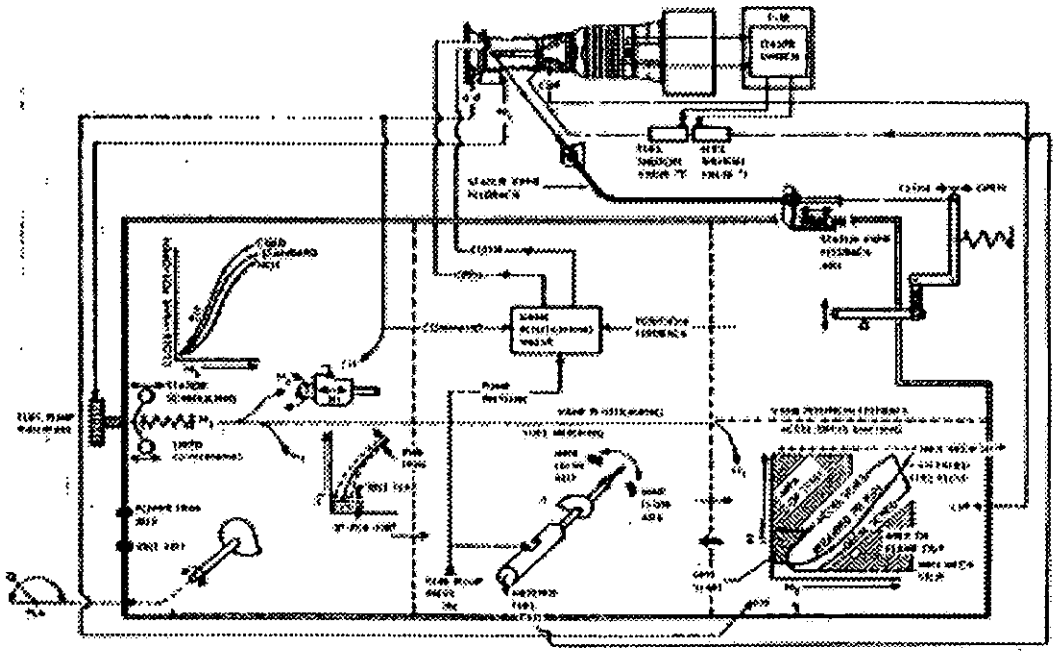


Fig. 1 - Hydro-mechanical fuel regulator

These types of controls have been widely used and worked well for many years showing excellent EMI immunity, independence from electric power, good weight and dimension characteristics, achieving acceptable fuel and VSV control performance. Only for the Italian Navy around 40 LM2500's with this combination of hydro-mechanical/electronic controls developed by FiatAvio-SEPA have been installed for aircraft cruisers, destroyers and frigates and they are still working properly.

2 – NOMENCLATURE

ECS	Electronic Control System;
GE	General Electric;
GT	Gas Turbine;
HPT	High Pressure Turbine;
HSS	High Signal Selection;
LPT	Low Pressure Turbine;
LSS	Low Signal Selection;
MFC	Main Fuel Control;
NGG	Gas Generator speed;
NPT	Power Turbine speed;
PLD	Power Level Demand;
PS3	Compressor delivery pressure;
RVDT	Rotary to Voltage Displacement Transducer;
T5.4	LPT Inlet Temperature;
T2	Compressor Inlet Air Temperature;
VSV	Variable Stator Vane.

3 - FROM HYDRO-MECHANICAL TO ELECTRONIC

When the LM2500 started to be used for the industrial application as prime mover for the pipeline pumping stations the MFC/Electronic control was still acceptable. But when it was used for the power generation it presented unacceptable limits due to:

- fast response requirements for load acceptance and rejection;
- strict frequency regulation requirements;
- safe shut-down requirements.

Moreover industrial power plants, especially in west countries, were subjected to increasing pollution limitation and this required more powerful control systems capable to manage water and steam injection with more sensors and actuators than before. In addition more and more effort was necessary to easily adapt (and at low cost) the system to the site needs and cope with the power enhancement to 32000 HP. This could only be performed using the flexibility of digital systems. Hence ECS were developed and widely used for this application.

This good experience and the advantage of the use of a metering valve and VSV driven by oil pressure instead of liquid fuel without high degree of filtration (critical for the MFC) convinced the ECS introduction on the ships.

In this respect FiatAvio-Sepa designed its own ECS for industrial application and with some modification marine-qualified it for a water jet application on the Tirrenia Fast Ferries MDV3000 (Fig. 10).

4 - ELECTRONIC CONTROL SYSTEM (ECS)

What is it? The ECS developed for the Tirrenia Fast Ferries is composed by two cabinets (each 1.8 x 1.6 x 0.6m, total weight 250 kg) installed in the engine room near to the Marine LM2500 Gas Turbine (GT). Cables connect the unit to sensors, actuators and to the auxiliaries (Fig. 2).

Inside digital electronic boards with microprocessors run its software to perform the necessary functions to control, protect and supervise the GT; conditioning cards read the different sensors on engine (thermocouples, pressure transducers, switches etc.) and drive the actuators (shut/off valves, servo-valves, electric motors etc.).

A local ruggedized console integrated in the cabinet supplies the local "*man-machine interface*" for commissioning, maintenance tasks and local management. An Ethernet link and some safety critical hardwired signals to the Platform Management System allow the communication with ship platform in order to receive commands such as start/stop and dispatch information on the status of the GT to the crew.

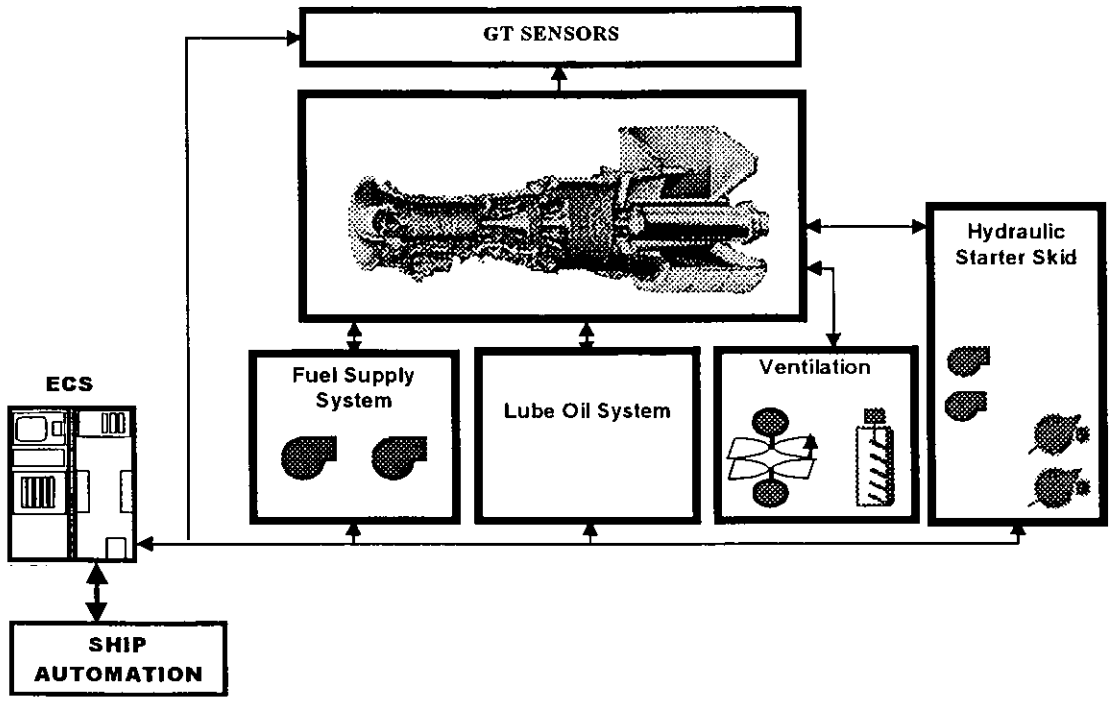


Fig. 2 - TCS Connections

The software flexibility of the digital controls, together with the necessary MIPS performance of the CPUs, have permitted to integrate more (and more complex) functions than in the previous technology increasing dramatically the level of automation. This has been done in two different directions: the first to improve GT performance and safety, the second to include the control of the auxiliaries such as ventilation, starter, fuel system etc.

The hydro-mechanical regulator has been removed and fuel and VSV position are now controlled via servo-valves electrically driven by ECS. VSV and servo-valves are actuated with oil pressurised by the oil pump that is driven by the auxiliary gearbox (Fig. 3). Engine control algorithms, based on a greater amount of GT internal parameter measurements, have been enhanced.

Moreover ECS has included system testability capability with self-diagnosis and trouble-shooting options improving engines and auxiliaries reliability.

All the ECS functions can be grouped in the following categories:

1. *CONTROL FUNCTION*
2. *SEQUENCES FUNCTION*
3. *PROTECTION FUNCTION*
4. *SUPERVISION FUNCTION*
5. *MAINTENANCE FUNCTION*

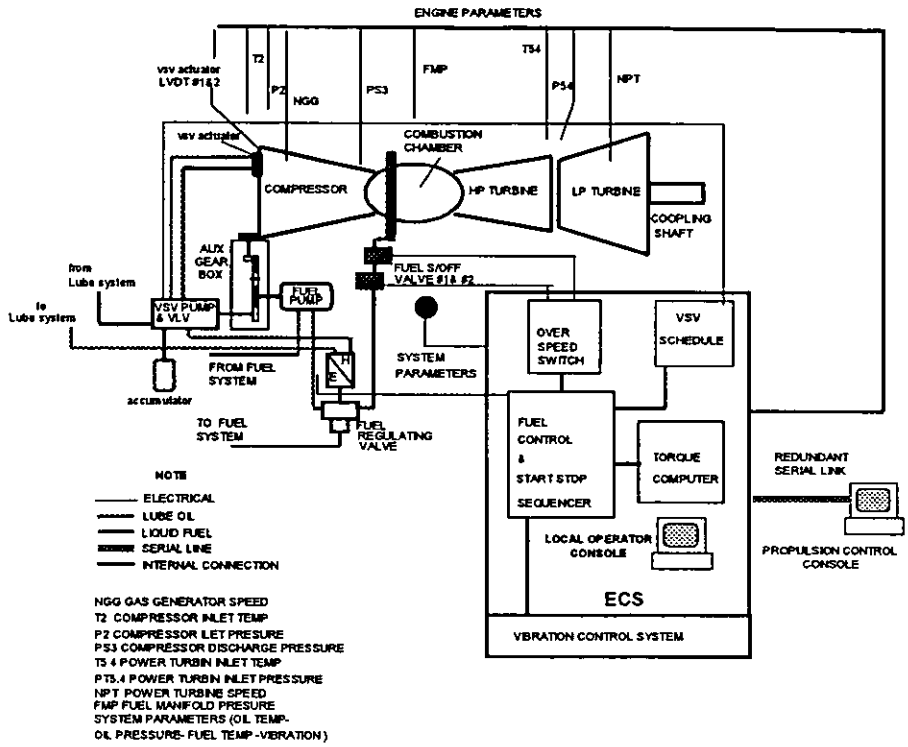


Fig. 3 - LM2500 Gas Turbine Electronic Control System

4.1 - CONTROL FUNCTION

Control functions process all parameters necessary to control properly and safely the GT during all engine operations providing the correct **fuel control** and **VGV position control**.

The **fuel control** is very articulated. The main loop control is to regulate the GT in order to supply the required power. This is what the pilot expects: to move the throttle in the pilothouse and to receive power from the prime movers.

Power is proportional to the Gas Generator rotor speed (NGG). Hence the pilot lever is converted in a speed reference equivalent to NGG generally ramped to avoid excess transient demand and limited to a max NGG value, corrected by T2. This requested value is then compared with the measured NGG and the error feeds compensating algorithms to evaluate the required fuel (Valve Position Demand). The Valve Position Demand in then transformed in current that, via a local position loop, moves the fuel metering servo-valve (Fig. 4).

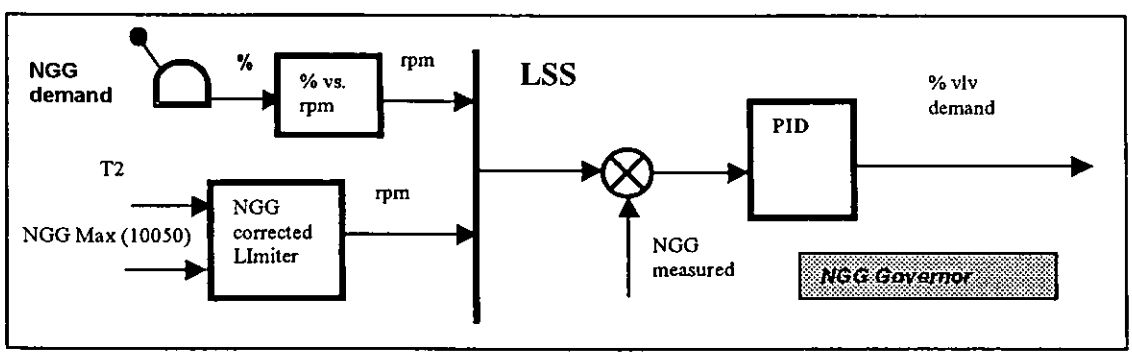
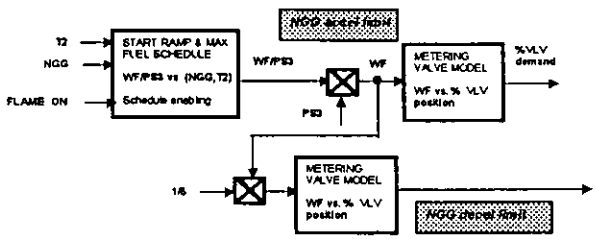
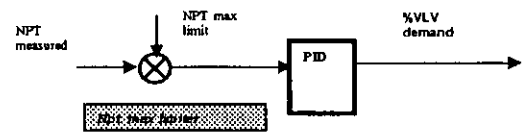
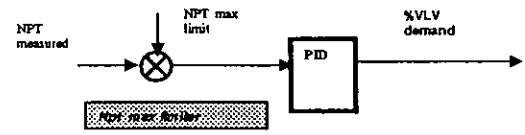
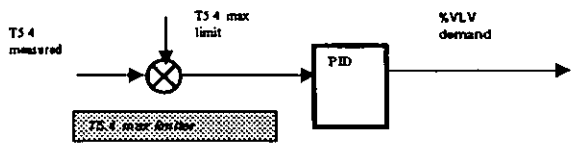
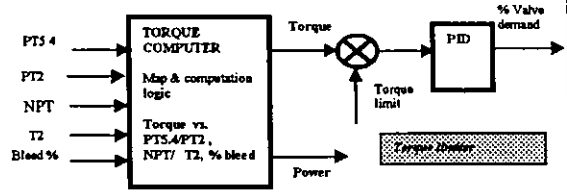


Fig. 4 - Basic control loop for the fuel

Compensating algorithms are very simple, basically a PID (Proportional, Integrative and Derivative). This corrective filter with the three gains (K_p, K_i, K_d) is sufficient to obtain good closed loop performance in terms of transient and steady state response:

$$\text{Valve Position Demand} = K_p * \varepsilon_{NGG} + K_i * \int \varepsilon_{NGG} dt + K_d * d(\varepsilon_{NGG})/dt$$

From a control point of view this loop is sufficient to regulate the power of the GT but it is not sufficient to maintain the engine through its limits during all operational envelope. Engine limits are max temperatures, forces, rotational speeds, pressures that each GT module can sustain without affecting its life. Hence control loops specific for each GT limit are added. They are similar in the structure and generally called limiters.

Limiter	*Loop*	*Purpose*
NGG accel./decel.		To avoid high acceleration/deceleration of the Gas Generator that can cause vibration, mechanical, thermal stress and sudden power to the Power Turbine. This loop also provide control for start up dosing fuel flow.
NPT acceleration		To avoid high acceleration of the Power Turbine that can cause vibration, mechanical and thermal stress. It also avoids over-torque of the shaft.
NPT topping		To limit the max rotation speed of the Power Turbine.
T5.4 limit		To limit the max temperature in combustion chamber in order to avoid turbine blades damage.
Torque limit		To limit the generated torque to protect the shaft line and the GT itself from over-torque.

In this way there are many loops. Each loop controls one particular limit of the GT and evaluates its own Valve Position Demand. Some of the loops tend to reduce the fuel (i.e.: T4.5 limit) other

to increase (i.e.: NGG deceleration). However only one loop must win at each time and the command to the fuel metering valve must be unique. This is performed using two easy and effective blocks called "***Low Signal Selection (LSS) and High Signal Selection (HSS)***" that select the lowest demand between the loops that tend to increase the fuel and the highest between those decreasing the fuel (Fig. 5).

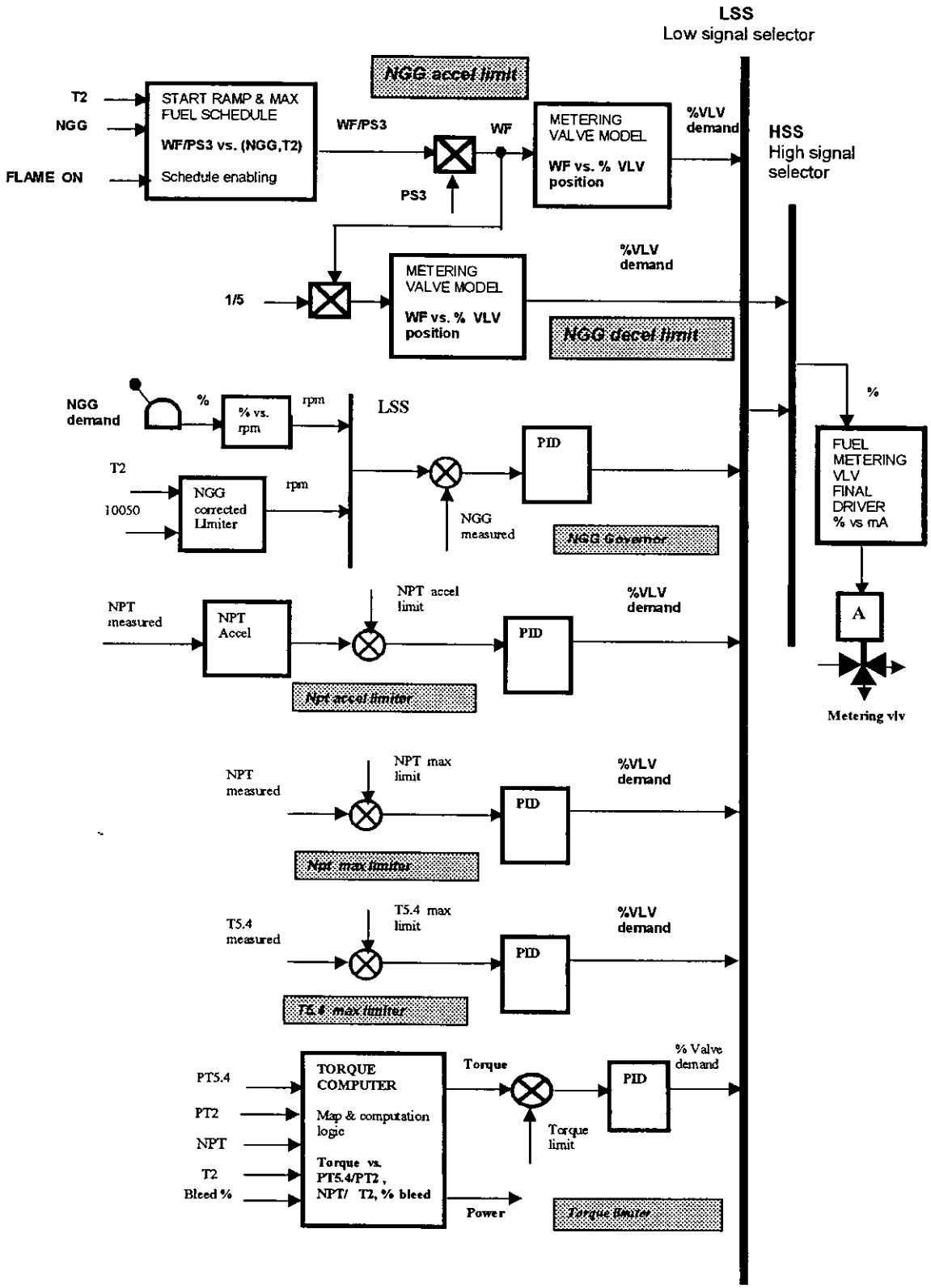


Fig. 5 - Fuel Management Block Diagram

During normal run the winning loop depends on the different ambient condition (inlet temperature, humidity etc.): at high ambient temperature (i.e. above 15 C) the GT normally runs under T5.4 limiter, below this value it is controlled by the NGG loop.

The winning Valve Position Demand will set the required position of the fuel-metering valve. This is not directly driven but a local control loop is provided to set the proper position. Valve position is measured by a RVDT (Fig. 6).

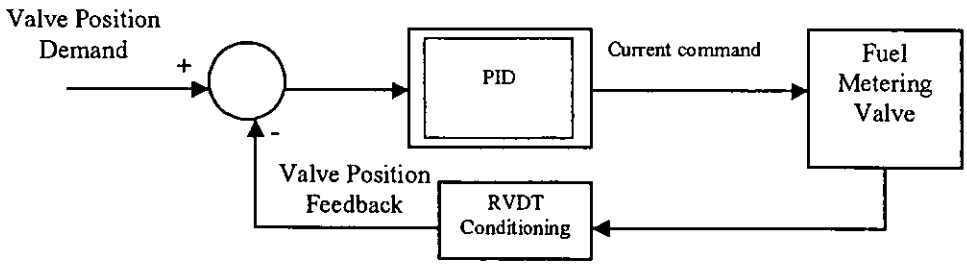


Fig. 6 - Final Driver - Fuel Metering Valve command loop

All these loops, with some more tricks to include non linearity's or variable limits depending on GT states, shall command or limit opportunely the liquid fuel metering valve position in order to maintain the engine through the limits in all operational conditions.

The **VGV position control**, a simple open loop control, is separate from the fuel management system and works in parallel with it. Its purpose is to modify the incidence of air flow on the compressor in order to optimise its performance and to support the GT during special events such as the starting and stall.

The VGV Position Demand is calculated as a function of NGG and T2 and a servo-valve loop provides to move the VGV in the requested position (Fig. 7). In this way when NGG is raised also the VSV position is raised toward open position and viceversa.

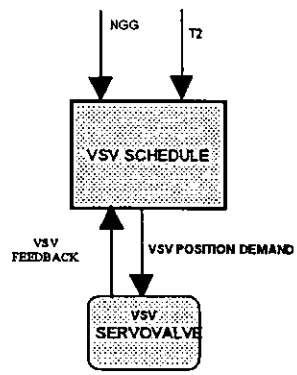


Fig. 7 - VSV Position Control

4.2 - PROTECTION FUNCTION

Protection functions ensure that if any unsafe situation occurs this is detected and properly managed by the ECS. Unsafe means that it can result in damage to the GT, to the plant or even to the human life. For instance an uncontrolled over-speed can cause rupture of blades that, if not mechanically contained, could cause severe injury to an operator in proximity of the GT.

These possible hazardous failures are identified during the faults analysis design phase (FMEA) and safety features are implemented to identify the malfunction and cope with it. This is done by software, by special hardware and by independent and/or redundant circuitry where necessary.

Basically in the ECS there are two types of protections: the **safety critical** ones and **the potentially safety critical**. The former require to shutdown immediately the GT in order to avoid damage (example: over-speed), the latter will cause damage if not managed (example: high oil temperature). In this case the strategy is to *step to idle* the GT and leave to the Operator any decision. At idle the symptom generally is reduced or even disappears and the engine can stabilise its temperature. The Operator can then decide the best strategy (normal shutdown or solve the problem) but he cannot override the ECS jeopardizing the GT if the problem persists. If the anomaly persists even at idle or in case of trespassing safety thresholds, the ECS will start a GT shut down in this case too.

Generally it is a design aim to limit as much as possible the trips due to the fact that immediate shutdown reduces the GT life for the thermal and mechanical stress.

Potential safety critical protections are:

- High inlet temperature of LPT;
- High temperature engine lube oil;
- Low pressure lube oil;
- Low pressure oil liquid fuel metering valve;
- High vibration;
- Loss of ventilation.

Safety critical protections are:

- Surge detection. A fast analogue logic based on dPS3/dt can detect the stall (Fig. 8);
- Power turbine over-speed. The system is based on an independent redundant analogue over-speed protection;
- ECS failure;
- Valves: solenoids current monitoring is provided;
- Shut-off anti recycling: it is performed using the combination of current and pilot pressure monitoring or fuel manifold pressure versus fuel demand;
- Fire hazards

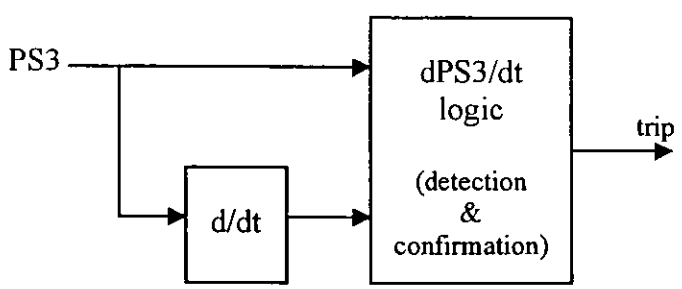


Fig. 8 - Stall protection

If the safety critical protections fail, severe damages are foreseen. As a consequence redundant circuits and mechanism are provided. A special digital over-speed circuit with double NPT

sensors as input is implemented by independent electronics. The logic is "*one out of two wins*" and the actuation is directly on the two fuel shut-off valves (Fig. 9). NGG and T2 sensors are doubled as well.

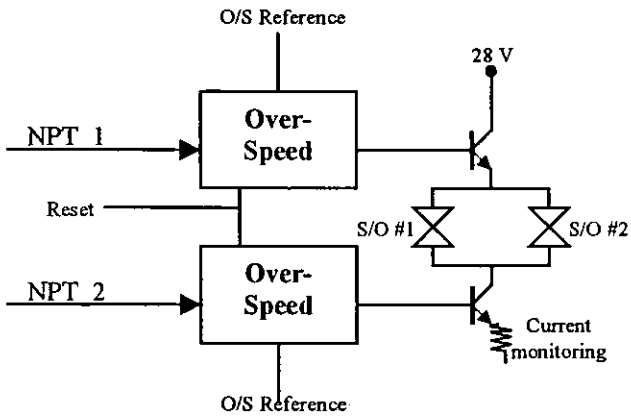


Fig. 9 - Overspeed redundant logic

To avoid possible dormant failure of the critical protections special test circuits are added to perform **cold test** (i.e. while the GT is not running) or **hot test** (i.e. while the GT is running).
A special test circuit embedded in the ECS can inject electric signal in parallel to the NGG sensors checking the over-speed chain when the GT is in halt (cold test). Moreover special software command protected by security can give the Operator the opportunity to halt the GT changing over-speed threshold and checking in this way over-speed functionality while the GT is running (hot test).

Fuel shut-off valves are doubled and to check them the GT is shutdown with only one of them alternatively.

Moreover the ECS contains Built In Test to assess the condition of the internal hardware (RAM, I/O's, CPU, timers etc.) and sensors.

All these test features assure to reach a very high safety and circuit testability. As a fallback many test options can be used during the commissioning and maintenance tasks to reduce the time schedule.

4.3 - SEQUENCES FUNCTION

In order to reduce the man efforts, automatic sequences of operations drive every activity of the GT. Sequences are basically organised as a state machine. Each state enables certain operations, special limits of the control loops, the level of protections, the operator commands in order to drive properly and safely the GT and command the transient to another state.
Besides the halt state there are three possible states: ***normal state*** , ***abnormal state*** and ***maintenance state*** . All related functions have been designed in order to have:

- high degree of automation so that the Operator pilots a very simple and easy system with few commands (start, stop, increase/decrease power);

- sophisticated sequences in order to have a very high degree of safety without the presence of any operator, even in case of malfunction;
- dedicated functions supporting the Operator during its maintenance tasks to reduce repairing time and effort.

In ***normal state*** there are all sequences that provide a single initiate automatic start-up from zero speed condition to base load and vice-versa for normal shut-down, slow to minimum, increase/decrease power in accordance with Pilot's needs. These sequences tend to minimise the Operator workload.
They include the automatic GT warm-up period after start-up disabling the Operator to increase power.

In ***abnormal state*** there is the management of the critical situation. In case of safety critical alarms an immediate shutdown is performed and manual fast cooling is provided to avoid rotor bowing. In case of potential safety critical alarm a step to idle is performed to take the GT away from the hazardous situation. The Operator will then decide the best strategy on the basis of the information provided.

In ***maintenance state*** the Operator is allowed to run cold and hot tests, to check GT and ECS parameters, to run maintenance sequences such as water wash and software reloading and perform part of normal automatic sequences for diagnostic purposes.

In order to improve the overall propulsion automation the ECS also incorporates many non-GT related functions. They command the GT auxiliaries such as fuel forwarding pump, GT ventilation, fire-fighting monitoring, ice conditioning system, skid pump for hydraulic starter etc.

4.4 - SUPERVISION FUNCTION

Supervisory functions provide the exchange of information to command the GT and to monitor the ECS, GT and auxiliary system states.

All these information are available via a man-machine-interface based on Personal Computer. One is a local console embedded in the ECS, another is the Propulsion Console on the bridge. Information are organised in video pages containing the status of the GT, all the necessary commands and alarms (Fig. 11).

These video pages have been designed in collaboration with the final user in order to easy his work and they have been based on ergonomic and user-friendliness criteria. These include start and stop sequence, emergency shutdown, fuel purge, water washing, power generation and anti-icing.

The status of the GT and its auxiliaries is also condensed in synoptic that include alarms, graphic display, real time trends, logs (first out) with the possibility of hard-copy.

Operator selectable data can be stored on suitable media (hard disk, CD rom...) for future processing.

4.5 - MAINTENANCE FUNCTION

With this type of GT the maintenance is to a certain level "on condition". The ECS collects and archive special data on a time basis used for maintenance planning and GT life management, such as fire hours, engine run hours at specific exhaust temperature, number of operations, fired starts, total trips etc.

Moreover the ECS can be configured for remote diagnostic via modem. Engine parameters and ECS internal status can be examined remotely. This can be beneficial to assist troubleshooting.

5 - CONCLUSION: MECHANICAL VERSUS ELECTRONIC

FiatAvio-SEPA has developed and installed many controls for LM2500. Quite a lot in the past decades were hydro-mechanical and analogue for different Navies, then electronic with digital fuel and VGV controls were designed for industrial power plants. This type of ECS has been recently "marinized" for water jet application in order to meet RINA and ABS rules and classification requirements on the Tirrenia Fast Ferries.

This is an example of the migration of technology from different fields and cross-fertilisation. Such migration has permitted re-usability of hardware and software allowing cost reduction. The confidence in reliability of commercial components (sensors and metering valves) gained from the power plant has also allowed to dress the gas turbine with this type of components allowing further reduction costs.

ECS's in comparison with the old hydro-mechanical or analogue ones have clearly shown improvements of:

- engine performance due to:
  - better matching of VSV position with gas turbine operating profile
  - more accurate fuel metering;
  - increase insensibility to water contamination in the fuel (up to 1000ppm in volume);
- engine protections due to:
  - more articulate control top limits (NGG, PS3, T5.4, acceleration, deceleration) that keep the engine safely running in the allowed envelope;
  - stall detection;
  - safety logic to ensure emergency shut-down and cooling;
- control performance and starting schedules due to more sophisticated control laws;
- easy tuning;
- engine monitoring, diagnostic and maintenance due to:
  - engine data capture
  - life cycle count;
  - built in test capability

ECS have proven great flexibility and performance improvements in comparison with old hydromechanical governors. But probably the real advantages have been the possibility to increase the safety of the propulsion system, the level of overall automation (to a point that the Pilot can almost forget the GT) and the maintenance benefits.

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Fig. 10 - Tirrenia Fast Ferry

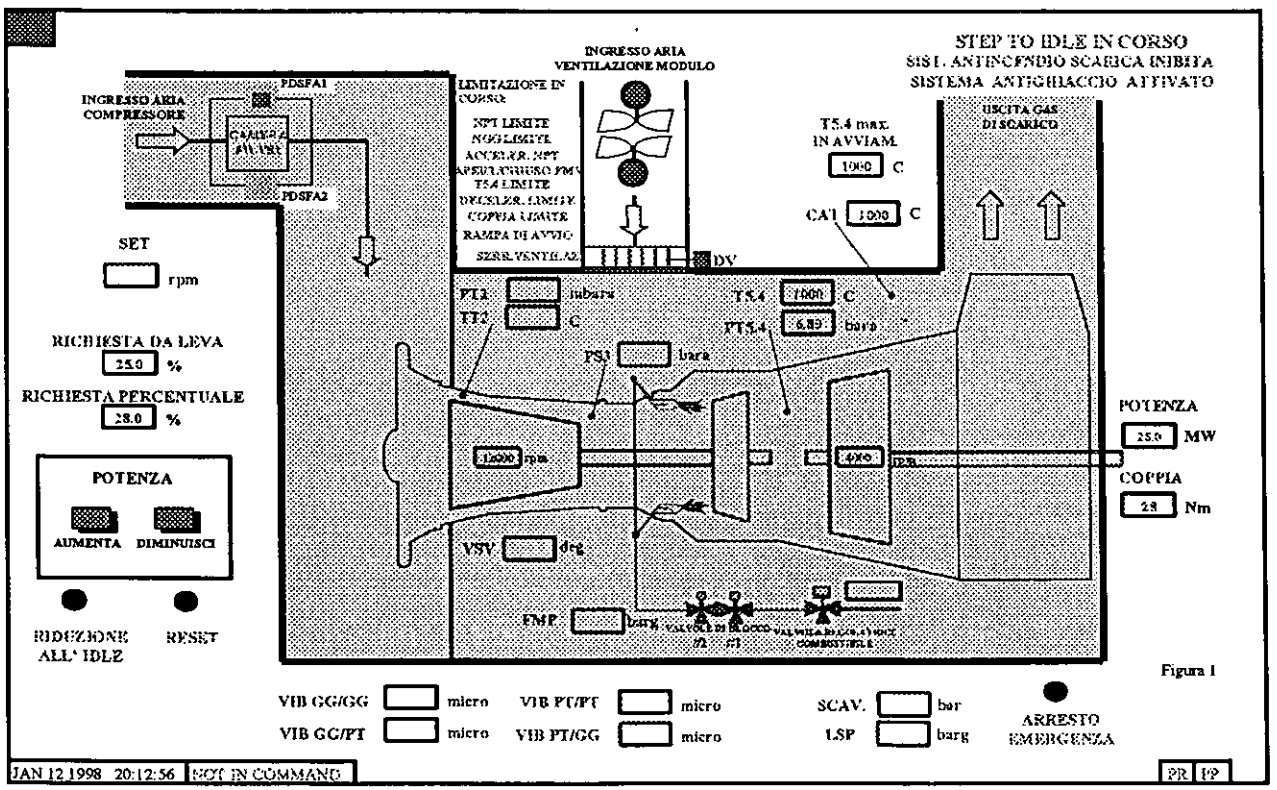


Fig. 11 - Example of Video Page

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

FMEA OF FAST SHIP PROPULSION PLANT

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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ABSTRACT

The paper describes the FMEA method (Failure Mode and Effect Analysis) and the procedure developed by Cetena for its application in accordance with the IMO HSC Code entered into force the 1st of January 1996.

In order to allow a better understanding of both FMEA method and proposed procedure, a case study concerning the failure mode and effect analysis of the propulsion plant of a high speed craft is also presented into the paper.

INTRODUCTION

Cetena, Italian Ship Research Center, always oriented to study ship's safety aspects, since the early nineties have developed advanced research projects addressed to innovative safety approaches, taking into account both technical and economical factors.

The increasing demand of higher safety standards for Fast Ferries, has focused the efforts of Cetena's R&D activities on dedicated methodological tools able to comply with the IMO HSC Code [1], entered into force the 1st of January 1996, and to ensure an acceptable level of risk.

Present paper gives a detailed description of the procedure developed by Cetena to comply with the IMO HSC Code requirements and criteria [2].

In accordance with the new approach for safety rules introduced by the Code, the procedure contains both theoretical analysis, to be developed during ship's design and construction phases, and experimental validation to be carried out during delivery sea trials.

The procedure is mainly based on the application of FMEA method (Failure Mode and Effect Analysis), a disciplined design review technique, largely employed in other industrial sectors such as: nuclear, chemical, offshore, etc., that focuses the development of products on prioritised actions to reduce the risk of product field failures, and documents those actions and the review process [3].

Since 1996, the procedure has been successfully applied to all the Fast Ferries built by Fincantieri: MDV Pegasus, Superseacat and MDV 3000 Juppiter [4-5-6-7-8-9-10-11-12].

FMEA METHOD

Failure Mode and Effect Analysis is a technique based on a single failure concept, under which each individual failure is considered as an independent occurrence with no relation to other failures in the system, except for the subsequent effect that it might produce.

FMEA is a powerful tool for new design solution evaluation, on the basis of safety and reliability characteristics, particularly useful when it is necessary to consider in detail safety and reliability of complex systems vulnerable to single equipment failure that can lead to system failure and accidents.

The analysis produces a qualitative, systematic reference list of the potential equipment failure modes, their effects on the whole system, and a list of potential corrective actions, for safety and reliability improvement, that should be evaluated by decision makers.

A worst case estimate of the consequences resulting from single failures is also provided by the analysis together with severity ranking and probability of occurrence of failure modes.

Starting from the basic failure characteristics of the elements, and functional structure of the system, FMEA systematically documents the ways equipment can fail, the possible causes, the effects these failures can produce on system performance and ranks each potential failure in terms of severity and probability of occurrence of failure mode.

Methods of detecting the failure modes, isolating the failure or malfunction and actions which can overcome or mitigate the effect of the failure are also considered and documented.

The FMEA is usually carried out on a series of worksheets, where the results are listed in a tabular format, equipment item by equipment item, following a systematic *top-down* or *bottom-up* approach.

Each individual failure mode is considered as an independent occurrence, with no relation to other failures, except for subsequent effects that it might produce, however, in special circumstances, common cause failures of more than one system's component may be also considered.

Human errors are usually not directly examined in an FMEA, however the effects of a misoperation as a result of human error may be considered as an equipment failure mode.

Using FMEA method generally requires the following data and information sources:

- system equipment list or P&ID;
- knowledge of equipment functions and failure modes;
- knowledge of system functions and responses to equipment failures;

All analysts involved in the FMEA should be familiar with the equipment functions, failure modes, and how the failures might affect other parts of the system.

The time necessary to carry out an FMEA is proportional to the size of the system and to the number of components to be analysed.

FMEA PROCEDURE

The procedure developed by Cetena to comply with the IMO HSC Code requirements and criteria, is based on the following main steps:

1. Craft description;
2. System definition;
3. Development of system block diagrams;
4. Failure mode and effect analysis;
5. Test programme development.

In accordance to the Code two different level of detail of FMEA analysis are defined:

1st level (functional failure analysis): the analysis is carried out on a system level where each function of each craft's main system is analyzed with respect to possible failure modes, causes and effects on the entire craft. This is done for all typical operational modes. The aim of the 1st level FMEA is to examine all important functional and/or operational aspects;

2nd level (detailed equipment FMEA): the analysis is carried out at lower hierarchical level, examining the components or units of a system. Each conceivable mode in which a component or unit can fail with respect to its intended function is analyzed.
The 2nd level FMEA is prescribed for the systems which failed the 1st level analysis; i.e.:

- whose individual failure can cause hazardous or catastrophic effects, and
- where a redundant system is not provided, and
- the probability of occurrence of acceptance criteria are not met.

Craft description

Basic information on ship's design and operations are collected from drawings and technical specifications, and are used to familiarise the analyst team with the design of the craft and its systems.

The operative profile of the ship is also defined in order to enable the system functional requirements to be identified within the typical operational modes of the craft.

According to the Code requirements, the following operational modes of the craft are considered:

- normal sea-going conditions at full speed;
- maximum permitted operating speed in congested waters, and;
- manoeuvring alongside.

Systems definition

In this step, as required by the Code, a study consisting in a further and more detailed review of drawings and equipment manuals, is performed for each of the following systems:

- machinery (mainly propulsion) and their associated controls;
- directional control;
- stabilization;
- electrical power generation and distribution.

A narrative description of each system and its functional requirements is drawn up including the following information:

- general description of system operation and structure;
- functional relationship among the system elements;
- acceptable functional performance limits of the system and its constituent elements in each of the typical operational modes;
- system constraints.

Development of system block diagrams (SBD)

System block diagrams are drawn up for each system being analysed, to show the functional flow sequence of the system, enabling a full understanding of its functions and operations and the various interactions between components within the system.

Each system block diagram should contain:

- breakdown of the system into major sub-system or equipment;
- all appropriate labelled input and outputs (i.e. mechanical power, electrical power, electrical signal, etc..) and identification numbers/acronyms by which each sub-system is consistently referenced;
- all redundancies, alternative signal path and other engineering features which provide *fail-safe* measures.

Generally different set of block diagrams are necessary to describe different operational modes.

Failure mode and effect analysis

The analysis is carried out with the aim of defining the important failure conditions that may exist in systems and equipment, and to comment accordingly regarding a qualitative assessment for the probability of such failures and their possible effects.

In accordance with the Code requirements, both local and end failure effects are evaluated, and the severity of end effects are classified accordingly with the categories proposed by the Code (Table 1).

Minor Effect	An effect which may arise from a failure, an event, or an error which can be readily compensated for by the operating crew; it may involve: 1. A small increase in the operational duties of the crew or in their difficulty in performing their duties; or 2. A moderate degradation in handling characteristics; or 3. Slight modification of the permissible operating conditions.
Major Effect	An effect which produces: 1. A significant increase in the operational duties of the crew or in their difficulty in performing their duties which by itself should not be outside the capability of a competent crew provided that another major effect does not occur at the same time; or 2. Significant degradation in handling characteristics; or 3. Significant modification of the permissible operating conditions, but will not remove the capability to complete a safe journey without demanding more than normal skill on the part of the operating crew.
Hazardous Effect	An effect which produces: 1. A dangerous increase in the operational duties of the crew or in their difficulty in performing their duties of such magnitude that they cannot reasonably be expected to cope with them and will probably require outside assistance; or 2. Dangerous degradation of handling characteristics; or 3. Dangerous degradation of the strength of the craft; or 4. Marginal conditions for, or injury to, occupants; or 5. An essential need for outside rescue operations.
Catastrophic Effect	An effect which results in the loss of the craft and/or in fatalities.

Table 1 –Failure effect categories.

Failure detection devices and alternative modes of operation are also indicated.

According to the Code requirements the FMEA worksheet (Fig. 1) includes several columns defined as follows:

N°	Sequential numbers to identify the item and its failure (*x-y-z*).where: *x* identifies FMEA worksheet set; *y* identifies the sub-system or equipment; *z* identifies equipment function, failure mode or failure causes.
Name	Name of the considered item.
SBD Id	Unique identifier (number/acronym) that relates the item to the system block diagram.
Function	Purpose of the item in the system.
Failure Mode	Type of failure.
Failure Cause	Causes leading to type of failure.
Local Effect	Failure effects specific to the considered system.
End Effect	More general failure effects related to the craft.
Failure detection	Means by which failure can be detected.
Corrective action	Any corrective action to prevent or reduce the failure effects. All corrective actions indicated in the worksheets are actions foreseen by designers.
Severity of failure	Ranking of end effect in accordance with the categories proposed by the Code (Table 1). If the analysis is performed for more than one ship's operational mode, the ranking of the end effect refers to the most severe consequences.
Probability of failure	Probability of failure occurring in accordance with the categories proposed by the Code (Table 2). Probability evaluation is carried out only when corrective measures or redundancy are not provided (2nd level of FMEA analysis).
Remarks/Comments	Explanation for the ranking of failure severity and probability, recommendations and general comments such as the reasons for having excluded certain failure effects.

Frequent	Likely to occur often during the operational life of a particular craft.	More than 10^{-3}
Reasonably probable	Unlikely to occur often but which may occur several times during the total operational life of a particular craft.	10^{-3} to 10^{-5}
Remote	Unlikely to occur to every craft but which may occur to a few craft of a type over the total operational life of a number of craft of the same type.	10^{-5} to 10^{-7}
Extremely remote	Unlikely to occur when considering the total operational life of a number of craft of the type, but nevertheless has to be considered as being possible.	10^{-7} to 10^{-9}
Extremely improbable	So extremely remote that it should not be considered as possible to occur.	$<10^{-9}$

Table 2 –Failure probability categories.

Test programme development

The test programme is drawn up with the aim to confirm work hypothesis and findings of the FMEA analysis by simulating failures of the system being analysed.

In accordance to the Code requirements, the programme includes all systems and system elements whose failure would lead to major or more severe effects, restricted operations and any other corrective action.

In order to prevent any duplication or overcastting of system, the FMEA tests are developed in conjunction with builder trials.

The form used to document the FMEA tests is shown in Fig. 2.

CASE STUDY

To allow a better understanding of FMEA method and its application during ship design phase, a case study regarding the analysis of the Propulsion and Manoeuvring System (PMS) of a High Speed Craft (HSC), performed in accordance with the proposed procedure, is described in the following.

Propulsion and manoeuvring system description

Due to the architecture of the system, the two functions of propulsion and manoeuvring, are accomplished by the same subsystems/components, hence they are treated together as a unique system [PMS].

The PMS consists of four propulsion lines (two side steerable and two central fixed). Each of them consists in:

- a main diesel engine including all individual auxiliaries [MENG]
- a shaft line including all elements which connect main diesel engines to waterjet [SL]
- a waterjet (steering or booster [SWJ] or [BWJ])

The common auxiliary systems are designed as follows: fuel system [FS], air combustion system [ACS], starting air system [SAS] are common for the propulsion lines of the same engine room while lubricating oil system [LOS] and fresh water system [FWS] are common for all four lines.

To allow manoeuvring in harbour, the ship is fitted with a bow thruster [BT] electrically driven.

For the control of the system each engine is equipped with a local control device [ECS], and a remote control system [RCS].

For the simultaneous remote control of the four propulsion lines, through command lever steering tiller or autopilot, the [RCS] communicate through the RCS-co-ordinating bus to the common remote control system [CRCS].

All propulsion engines together with bow thruster, electrical power systems, auxiliary systems and essential users are monitored from the bridge by a monitoring control system [MCS] mainly consisting of two screens and an alarm printer.

Looking at the general arrangement of the system, the four propulsion lines guarantee a functional redundancy with respect to both steering and supply thrust functions.

Further redundancy is given by the separation of the propulsion lines in two distinct engine rooms. In each one, a booster and a steering propulsion line are located. The choice of having two engine rooms reduces the vulnerability of the system due to external events such as fire, flooding, explosion, and also to possible internal interaction.

Redundancy philosophy is also present in auxiliary systems which are either duplicated for the two engine rooms Fuel System [FS], Air Combustion system [ACS], Starting Air System [SAS] or are functionally redundant in terms of internal components such as Lubricating Oil System [LOS] and Fresh Water System [FWS].

Table 3 shows how the functional configuration of the propulsion and manoeuvring system changes depending on the operational mode of the craft.

	Normal sea-going conditions at full speed		Maximum permitted operating speed in congested waters		Manoeuvring alongside	
Active machinery	2 Steering WJ line and aux. 2 Booster WJ line and aux.		2 Steering WJ line and aux. 2 Booster WJ line and aux.		2 Steering WJ line and aux. Bow thruster	
Controls	*Route*	*Speed*	*Route*	*Speed*	*Route*	*Speed*
	Autopilot		Autopilot	Control-lever	Tiller	Control-lever

Table 3 –PMS operational profile.

PMS block diagrams

In accordance with the relationship among PMS functional configurations and craft's operational modes, two different block diagrams are drawn up.

SBD 1: This block diagram shows the functional flow sequence of PMS and relevant controls, during normal sea going conditions at full speed and maximum permitted operations in congested waters. All four propulsion lines are considered as functioning: central lines with only thrust function and side lines with both thrust and manoeuvre functions, while the bow thruster is not included. SDB 1 is shown in Fig 3.

SBD 2: This block diagram shows the functional flow sequence of PMS and relevant controls, during manoeuvring alongside. Only the side propulsion lines and the bow thruster are considered as functioning, while the two central lines are not included. SDB 2 is shown in Fig 4.

In the two block diagrams all sub-systems and machinery, relevant for ship propulsion and manoeuvring, are included taking into account their functions in respect of the considered craft operational modes. Physical layout, electrical power inputs and electrical signals from/to local and remote control and monitoring devices are also indicated into the diagrams.

The four propulsion lines are considered as completely independent each other, both in terms of machinery, controls and auxiliaries, except for fuel, air combustion and air control systems which supply both engines of the same room.

All the different monitoring devices and alarms of the various system components, are represented in each diagram as a unique block [MCS].

Failure mode and effect analysis

The basic work hypothesis, under which the analysis is performed follows.

1. Since the functions of propulsion and manoeuvring are accomplished by the same system, a unique FMEA is performed, where for each component being analysed both propulsion and directional control functions are separately considered.
2. In accordance to the Code requirements, different sets of FMEA worksheets are compiled in order to consider separately the operational modes of the craft.
3. Since, for each considered operational mode of the craft, the complete loss of one propulsion line may at most involve a slight modification of the permissible operating conditions with a small increase in the operational duties of the crew, the effect which may arise from this failure is considered as minor effect.
4. The same concept is also applied to the directional control function of the system, which is normally performed by the two external propulsion lines, equipped with the steering water jet, but may also be performed by using only one propulsion line. This depends to the fact the craft has been designed by applying functional redundancy criteria to both propulsion and directional control systems.

The analysis is performed following a *top-down* approach, starting with the 1st level of FMEA, during which failure modes of those items (mainly sub-systems) constituting the system at top level of functional schematisation (i.e. the bocks included into SBD 1 and SBD 2) are separately analysed for each considered craft's operational mode.

As far as applicable, each item is assumed to fail in the following failure modes:

- total loss of function;
- partial loss of function;
- rapid change to maximum to minimum output;
- uncontrolled or varying output
- premature or spurious operation;
- failure to operate at a prescribed time; and
- failure to cease operation at prescribed time.

Depending on the item under consideration other failure modes are taken into account. Two examples of FMEA worksheets, compiled for the selected PMS, are shown in Figures 5 and 6.

For those items which fails the 1st level analysis, in accordance with the Code requirements, the analysis proceeds with the 2nd level FMEA, during which failure modes of basic components of sub-systems are analysed. In this case more detailed SBD are drawn up before starting the analysis.

With reference to the selected PMS, the detailed SBDs of fuel system [FS] (SBD 3) and electrical power system [EPS] (SBD 4), are respectively shown in Figures 7 and 8.

Test programme

FMEA test programme includes two Sets of tests, the first one concerns the tests necessary to prove the validity of the work hypothesis, on the basis of which the whole FMEA has been carried out and the severity of failures have been classified, while the second set includes all tests to be performed in order to prove the assumptions made on specific failure modes.

With reference to the selected PMS, some tests are listed in the following.

SET 1 (Tests to confirm work hypothesis)

- Test to confirm that the complete loss of one propulsion line, both in terms of machinery and/or controls, may at most involve a slight modification of the permissible operating conditions with a small increase in the operational duties of the crew, without any more severe consequences for the craft.
- Test to confirm that, in the event of loss of main power, the propulsion is guaranteed.
- Test to confirm that, in the event of loss of autopilot, the Craft may be safely governed.
- Test to confirm that, for all machinery in fore engine room out of operation, the craft can operate safely and can proceed its voyage under its own power. (This test cover a specific requirement of the Code).

SET 1 (Tests to confirm work hypothesis)

- Test to confirm the craft will not significantly turn when RPM of one propulsion line is set to idle.
- Test to confirm the craft remains able to be safely steered in case of total loss of one steering water jet propulsion line.
- Test to confirm that by using only one fan, set to the maximum speed, it is possible to supply the necessary fresh air to two propulsion diesel engine.
- Test to confirm that, in case of total loss of ventilation in one engine room, the propulsion is guaranteed at reduced speed.
- Test to confirm that, the complete loss of steering propulsion line, due to controls failure, may at most involve a slight modification of the permissible operating conditions with a small increase in the operational duties of the crew, without any more severe consequences for the Craft.
- Test to confirm that, that the complete loss of booster propulsion lines, due to controls failure, may at most involve a slight modification of the permissible operating conditions with a small increase in the operational duties of the crew, without any more severe consequences for the Craft.

A FMEA Trial Sheet relevant to one of the above mentioned tests is shown in Figure 9.

CONCLUSIONS

On the basis of the experience gained in this last years, during which the FMEA method has been systematically applied to all new fast ferries, the global judgement on its effectiveness is certainly positive.

In respect to the traditional prescriptive design approach the main advantages rising from the use of FMEA method include:

- providing designers and builders with a systematic approach with which to audit their proposed system designs and to indicate areas that may require further consideration in relation to acceptable levels of safety;
- providing craft operators with documented data with which to develop their training, operational and maintenance programs ensuring that full account is taken of the failure characteristics of the craft;
- providing regulatory and certification authorities with a documented investigation into the craft's failure characteristics to assist in their assessment of the levels of safety proposed for the craft's operation.

As further confirmation of the positive judgement on this new way to approach safety aspects, the attention of IMO is actually focused on the improvement of the HSC Code, through the development of the FSA methodology (Formal Safety Assessment), a rational pro-active process for assessing safety in terms of acceptable risk level.

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FMEA Worksheet

Ship:_____ **Date:**_____

System:_____ **Sheet** ____ **of** ____

Operational mode:_____ **SDB:**_____

Analyst:_____ **Drawings:**_____

N°	Name	SDB Id	Function	Failure mode	Failure cause	Local effect	End effect	Failure detection	Corrective action	Severity of failure	Probability of failure	Remark / Comments

Fig. 1 – FMEA Worksheet.

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FMEA TRIAL SHEET

Date *Time:* *Test n°*

Operational mode:

Navigational mode:

Test scope

System: *Failure Mode:*

Item: *FMEA ref.*

Function: *Failure Cause:*

Failure Simulation and Test description

	Failure detection				
	Port outer	*Port inner*	*Stb. outer*	*Stb. inner*	*Monitoring and alarm system*
Expected					
Observed					

	Local failure effect				
	Port outer	*Port inner*	*Stb. outer*	*Stb. inner*	*Others*
Expected					
Observed					

	Back-up system, Corrective action				
	Port outer	*Port inner*	*Stb. outer*	*Stb. inner*	*Others*
Expected					
Observed					

	End effect on craft	*Severity of Failure*
Expected		
Observed		

Remarks:

Fig. 2 – FMEA trial sheet.

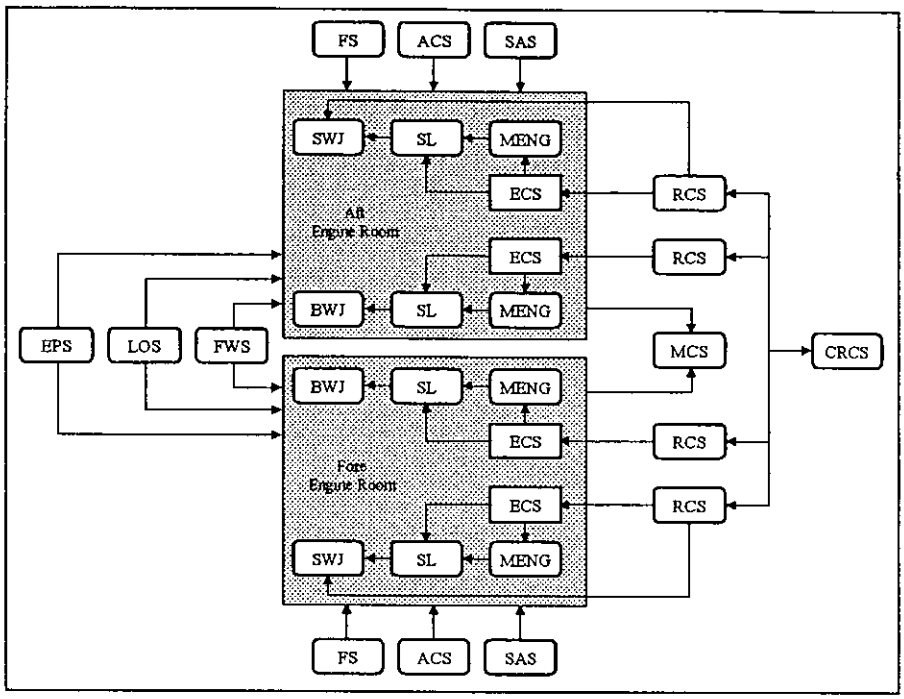


Fig. 3 – SDB 1: PMS configuration during normal sea going conditions at full speed and maximum permitted operations in congested waters.

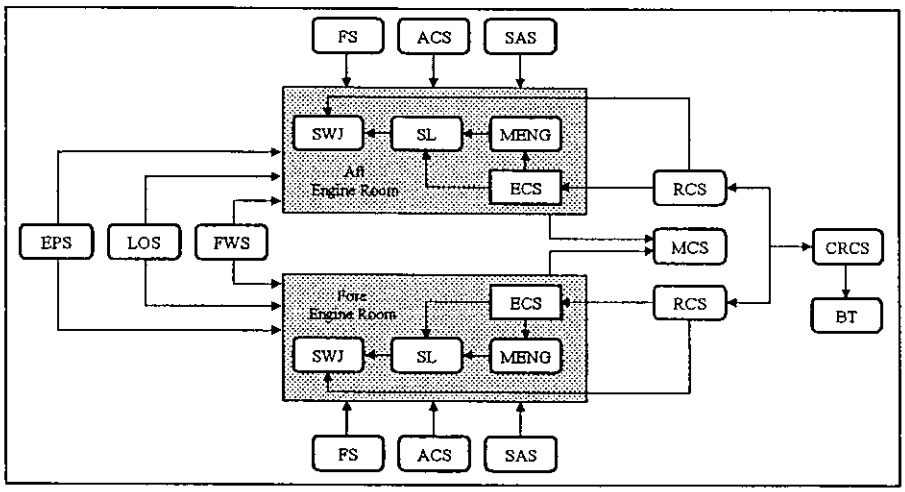


Fig. 4 – SDB 2: PMS configuration during manoeuvring alongside.

LEGEND			
MENG	Main Engine	LOS	Lubricating Oil System
SL	Shaft Line	ACS	Air Combustion System
BWJ	Booster Water Jet	BT	Bow Thruster
SWJ	Steering Water Jet	EPS	Electrical Power System
FS	Fuel System	RCS	Remote Control System
FWS	Fresh Water System	MCS	Monitoring Control System
SAS	Starting Air System	CRCS	Common Remote Control System

CETENA Private

FMEA Worksheet

System : Propulsion and Maneuvering System

Ship's Operational Mode: Normal sea-going condition at full speed *System Block Diagram Ref.: SBD 1*

Maximum permitted operating speed in congested waters *Date : 15/02/2008*

Drawing Ref.:

Team Coordinator: S. Capizzi

N°	Item Description			Failure Description		Failure Effect		Failure Detection	Corrective Action	Severity of Failure	Probability of Failure	Remarks/ Comments
	Name	SBD ref.	Function	Mode	Cause	Local	End					
1-1-1	Steering Water Jet	SW1	To supply thrust	Complete loss of output	Rotor breaking off	Complete loss of propulsion line	Small reduction (2.5%) of Craft's propulsion capabilities	Overspeed alarm	Automatic engine shut down Disconnection of propulsion line Local inspection	Minor		It is recommended to confirm by test.
1-1-2	Steering Water Jet	SW1	To supply thrust	Complete loss of output	Rotor blockage	Complete loss of propulsion line	Small reduction (2.5%) of Craft's propulsion capabilities	Loss of thrust control alarm	Engine shut down Disconnection of propulsion line Local inspection	Minor		Possible damage for components or sections of propulsion line
1-1-3	Steering Water Jet	SW1	To supply thrust	Complete loss of output	Complete obstruction of aspiration duct	Complete loss of propulsion line	Small reduction (2.5%) of Craft's propulsion capabilities	Overspeed alarm	Automatic engine shut down Disconnection of propulsion line Local inspection	Minor		The aspiration duct may be cleared by reversing the jet flow
1-1-4	Steering Water Jet	SW1	To supply thrust	Partial obstruction of aspiration duct	Partial obstruction of aspiration duct	Reduction of water jet output	Negligible reduction of Craft's propulsion capabilities	Activation of low engine power output	Local inspection	Minor		As above

Rev [2] Sheet 1 of 10

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Fig. 5 – FMEA Worksheet (example 1).

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FMEA Worksheet

System : Propulsion and Maneuvering System

Ship's Operational Mode: Maneuvering alongside

Drawing Ref.:

Items Coordinator : S. Capizzi

System Block Diagram Ref.: SBD 1

Date : 16/02/2000

N°	Item Description			Failure Description		Failure Effect		Failure Detection	Corrective Action	Severity of Failure	Probability of Failure	Remarks/ Comments
	Name	symbol	Function	Mode	Cause	Local	End					
2.15.1	Air conditioning system	ACS	To supply propulsion diesel engines with essential fresh air	Partial loss of function	One fan failed: accidental failure of few of normal power supply	Reduction of engine room ventilation	Possible depressurization and temperature increase in engine room	Fan failure alarm: low air	Stop one motor fan Local inspection and, if necessary, small reduction of engine power	Minor		It is recommended to confirm by test. The two fans are electrically separated
2.15.2	Air combustion system	ACS	To supply propulsion diesel engines with external fresh air	Total loss of function	Main switchboard black-out	Loss of two fans	Depressurization and temperature increase in engine room	Fan failure alarm: low air	Reduction of engine power Local inspection	Minor		It is recommended to confirm by test.
2.16.1	Remote control system	RCS	Steering and reversing control	Loss of steering and reversing control	Loss of power supply Hardware failure Software failure	Steering nozzle and reversing buster blocked in actual position	Partial reduction of Ceneri's directions control and manoeuvre capabilities	RCS sensors failure alarm	Engine shut down Deactivation of propulsion line	Minor		It is recommended to confirm by test.
2.17.1	Monitoring and control system	MCS	System monitoring function	Partial loss of function	Hardware failure Loss of power supply	Loss of one graphic station	Partial reduction of monitoring capabilities of propulsion and directional control systems	Visual	Use other graphic station	Minor		Monitoring system consists of two electrically separate graphic stations. Communication link by LAN.
2.17.2	Monitoring and control system	MCS	System monitoring function	Total loss of function	Loss of signal: communication	Loss of signal	Total loss of monitoring capabilities of propulsion and directional control systems	Visual	Use back up one	Minor		Each communication among the two graphic stations is connected to the MCS by a stable redundant bus.

Rev. 1.2.1

Sheet 8 of 9

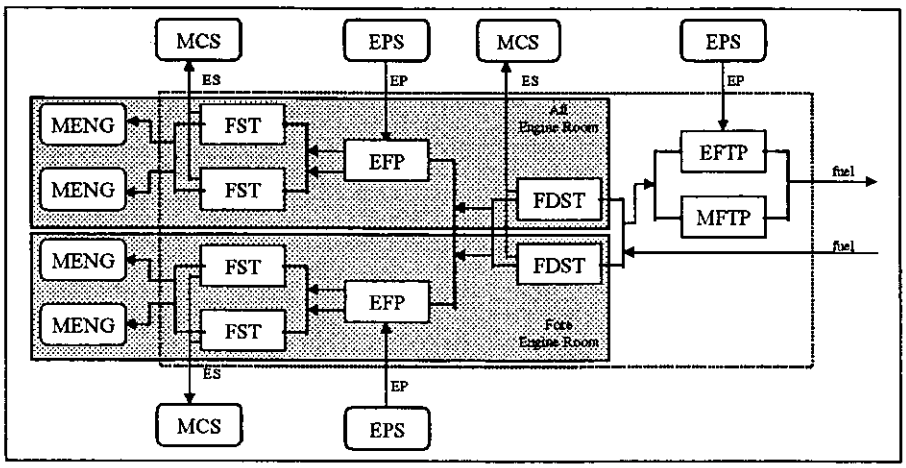


Fig. 7 – SDB 3: fuel system [FS].

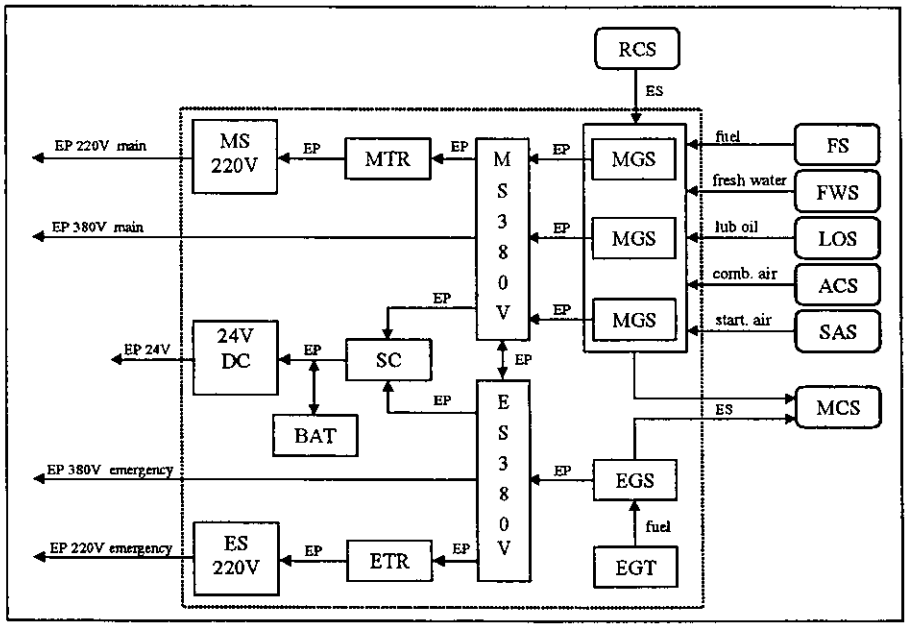


Fig. 8 – SDB 4: electrical power system [EPS].

LEGEND			
EFTP	Electrical Fuel Transfer Pump	EGS	Emergency Generator Set
MFTP	Manual Fuel Transfer Pump	EGT	Emergency Generator Tank
FDST	Fuel Daily and Storage Tank	ES380V	Emergency Switchboard 380V
EFP	Electrical Fuel Pump	ETR	Emergency Transformer
FST	Fuel Supply Tank	ES220V	Emergency Switchboard 220V
MGS	Main Generator Set	SC	Static Converter
MS380V	Main Switchboard 380V	BAT	Batteries
MTR	Main Transformer	24VDC	24V Distribution Board
MS220V	Main Switchboard 220V		

CETENA

FMEA TRIAL SHEET

Date: Time: Test n°1

Operation mode: Normal sea-going condition

Normal Operating Speed (35 kn) with invalid vessel

Test scope

To confirm that the complete loss of PORT OUT propulsion line, due to machinery failure, may at least involve a slight modification of the permissible operating conditions with a small increase in the operational duties of the crew, without any more severe consequences for the Craft.

System: Propulsion and Directional control Failure Mode: Total loss of functions

Item: PORT OUT propulsion line FMEA ref. Test sequence Set 7

Function: To supply thrust and allow Craft's directional control Failure Cause: Mechanical Failure

Failure Simulation and Test description

With the ship proceeding at normal operative speed in open water, normal navigation mode and propeller control, stop the PORT OUT diesel engine using the bridge emergency stop push-button.

Take note of the steering angle necessary to maintain a straight course with only STB steering jet operative.

Take note of the procedure necessary to restore the normal situation.

	Failure detection				
	Post cause	Post effect	Sub system	Sub system	Symptoms and alarm system
Expected	Diesel Engine, SSS and shaft will stop. Disengaging the clutch, the shaft will run. RPM measurement on	None	None	None	Port propulsion line alarm Generic Failure alarm when starting
Observed	RPM measurement on plate	None	None	None	

	Local failure effect			
	Post center	Post center	Sub center	Effects
Expected	Loss of thrust and directional control of line.	None	None	None	
Observed	As expected	None	None	None	

	Back-up system, Corrective action				
	Post cause	Post cause	Sub center	Sub center	Effects
Expected	Deactivation of line	Reduce RPM if necessary	Reduce RPM and fit steering angle to maintain the course	Reduce RPM if necessary	
Observed		RPM reduction not necessary	RPM reduction not necessary. The steering angle to maintain the course is 6°	RPM reduction not necessary	

	End effect on craft	Severity of Failure
Expected	Course deviation and speed reduction	Minor
Observed	Course deviation.	Minor
	Speed reduction from 35 kn to 30 kn.	

Remarks:
Any severe consequences for the Craft, and for other propulsion lines and systems, have not been observed.
The Operator alarms have been observed. Remote steering the engine to idle is a normal operation.
The suggested corrective action during operation of failed propulsion line.
The correct responsiveness of the ship with only one operative steering propulsion line has been verified

Fig. 9 – FMEA Trial Sheet.

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

GENERAL RINA REQUIREMENTS FOR THE AUTOMATION OF PROPULSION PLANTS ABOARD FAST SHIPS

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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ABSTRACT

The paper reports the main concepts of Registro Italiano Navale (RINA) rules with regard to the automation of propulsion plants aboard modern fast ships. The relevant criteria to accept integrated systems of automation aboard them are highlighted. Finally, some of the most used system architectures and communication protocols are described.

INTRODUCTION

One of the most innovative aspects regarding the plants aboard the ships is the more and more considerable employment of the automation systems to manage the machinery spaces. Besides the economic advantage in reducing the personnel aboard, the automation gives the possibility of the supervision of the plants, not being necessary the human intervention for the control and, at a first step, for the safety.

As to the point of view of RINA classification society, the philosophy and the criteria of acceptance of such systems are substantially reported in the "Rules for the construction and classification of ships - Section E - Rules for automation systems", in the "International Convention for the safety of life at sea" (or SOLAS) [1], and, for the high speed craft, also in the "Rules for the construction and classification of high speed craft", and in the laws in force of the flag's administration which the ship is entitled to fly.

The above mentioned rules and laws contain the requirements both for the single machinery and automation systems provided with automatic or remote control and the requirements concerning the tests required for the control and instrumentation equipment.

In the paper, the main design aspects to be fulfilled both of traditional automation systems and, taking into account the actual trend, of integrated computer based systems are illustrated, providing also an example of such a system.

NOMENCLATURE

RINA, automation, alarm, monitoring, control, safety, computer, PLC, network.

DEFINITIONS AND RINA ADDITIONAL CLASS NOTATIONS FOR AUTOMATION SYSTEMS

According to the RINA rules [2], [3], the following definitions apply to the highly automated ships:

1. machinery spaces periodically unattended with the ship both in free navigation and in normal manoeuvring conditions, i.e. from the "Stand by with engine", at the departure, to the "Finished with engine condition", at the end of voyage;
2. machinery spaces unattended, but under continuous supervision from a position where control and monitoring devices of machinery are centralised;
3. machinery spaces periodically unattended with the ship berthed at the pier.

Special notations can be assigned to ships whose automation system is in compliance with a), b) and/or c). In particular,

IAQ-1 or IAQ-1-X:
for ships whose automation systems are intended to attain the condition specified under 1). The condition of machinery spaces unattended is intended to be extended for a period of 24 consecutive hours. When such a condition cannot be attained for the above period, the notation IAQ-1-X shall be assigned, where X indicates the period, in hours, during which the machinery spaces are intended to be unattended.

IAQ-2:
for ships whose automation systems are intended to attain the condition specified under 2).

IAP:
for ships whose automation systems are intended to attain the condition specified under 3).

For the first two notations, the philosophy to be followed is to be such that the arrangements provided ensure that the safety of the ship in all sailing conditions, including manoeuvring, is equivalent to that of a ship having the machinery spaces manned.
As to the IAP notation, the automation systems shall ensure the same safety when the ship is berthed at the pier as when the machinery spaces are manned.
The main aspects involved in the automation systems are the monitoring, the control and the safety, whose main concepts will be explained in the next paragraphs. Particular reference will be made to "Category B" craft [3], defined as, according to the rules, any high speed passenger, other than a category A craft, with machinery and safety systems arranged such that, in the event of damage disabling any essential machinery and safety systems in one compartment, the craft retains the capability to navigate safely. To complete the definitions, "Category A craft" is any high speed craft:

1. operating on a route where it has been demonstrated to the satisfaction of the flag and port States that there is a high probability that in the event of an evacuation at any point of the route, all passengers and crew can be rescued safety within the least of:
   - the time to prevent persons in survival craft from exposure causing hypothermia in the worst intended conditions,
   - the time appropriate with respect to environmental conditions and geographical features of the route, or
   - 4 h; and
2. carrying not more than 450 passengers.

ALARM SYSTEM

Alarm systems should be provided which announce at the craft's control position, by visual and audible means, malfunction or unsafe conditions. Alarms should be maintained until they are accepted and the visual indications of individual alarms should remain until the failure has been corrected, when the alarm should automatically reset to the normal operating condition. If an alarm has been accepted and a second fault occurs before the first is rectified, the audible and visual alarms should operate again. Alarm systems should incorporate a test facility.
According to the "Rules for the Construction and Classification of High Speed Craft" [3], on category B craft alarm systems including their power supply have to be separate for each independent propulsion unit and its essential auxiliaries.
Alarm systems shall be designed with self-monitoring properties. As far as practicable, any fault in the alarm system shall cause it to fail to an alarm condition. Moreover, they shall be continuously powered by an independent stand-by power supply, e.g. battery in case of loss of normal power supply. In case of loss of normal power supply, the change-over to a stand-by power supply shall be automatic.

Parameter	Alarm level	Remarks
Lubricating oil pressure	low	
Lubricating oil temperature inlet	high	
Differential pressure across lubricating oil filter	high	
Pressure or flow of cooling water	low	
Temperature of cooling water outlet	high	
Level in cooling water expansion tank	low	
Deviation of each cylinder from average of exhaust gas temperature, or Exhaust gas temperature of each cylinder	high high +low	if cylinder power above130 kW
Exhaust gas temperature after turbocharger	high	if cylinders not monitored individually and if cylinder power below130 kW
Pressure of fuel oil to engine	low	if supplied by electrical pumps
Temperature of fuel oil to engine	high +low	if heated
Pressure of control air	low	
Pressure of starting air	low	
Safety system	failure	

Tab.1 - Alarms for propulsion Diesel engines

The design shall be such that, as far as practicable, a failure or malfunction in control and safety systems will not prevent the alarm from operating.

Failure of the normal power supply and of the stand-by power supply of the alarm system shall be indicated by a separate alarm providing visual and audible signals. As to the propulsion Diesel engines, the alarms reported in Tab.1 are to be provided.

The list of the alarms regarding the propulsion system is completed by the Tab.2 and Tab.3.

Parameter	Alarm level
Lubricating oil pressure to gears	low
Lubricating oil temperature of each bearing of the gears [1]	high
Servo oil pressure of gears and transmissions	low
Thrust bearing temperature	high
Stern tube temperature	high

[1] Not applicable to ball or roller type bearings

Tab.2 - Alarms for transmission, shaft gears

Parameter	Alarm level	Remark
Remote control of propulsion	failure	
Safety system of each machinery	tripped	
Override of safety system	activated	
Power of alarm system	failure	
Emergency controls	failure	
Fuel oil service tank temperature	high	if heated above flash-point
Purifier	failure	

Tab.3 - Miscellaneous alarms

The above alarms should be provided at all stations at which control functions may be exercised.

CONTROL SYSTEM

General concepts

Control systems shall be designed to "fail-safe". The characteristics of the "fail-safe" operation shall be evaluated on the basis not only of the control system and its associated machinery but also of the whole installation. All control systems essential for the propulsion, control and safety of the ship shall be independent or designed such that failure of one system does not degrade the performance of another system.

When normally operated remote or automatic controls are provided, in case of their failure the local control of the machinery or system shall in any case be possible.

Manually operated remote or automatic control systems shall be provided with shutting off devices to prevent starting of the associated machinery or system in case of conditions which can cause damage of these latter.

Failure of power supply to control systems shall operate an audible and visual alarm in the control position.

Control systems shall be designed, as far as practicable, as to ensure that a failure or malfunction of alarm and safety systems does not impair their operations.

Propulsion control requirements

It is possible to distinguish between [2]:

- *Manually operated remote control systems* of propulsion machinery are those systems for remote operations in which local actions by the operator are replaced, there being no substantial modification to the number and nature of the operations to be performed. In general a careful consideration is required to the operator before operating remote control equipment to ensure the safety of machinery.
- *Automatic remote control systems* of propulsion machinery are those systems in which the remote control is performed by such automatic devices that the number of operations to be carried out is considerably reduced and their nature is much more simplified with respect to those required by local control. In general the operation of the above automatic control devices does not require a previous careful consideration by the operator, the safe working of the associated machinery being assured by the control system itself.

The design of the remote control system shall be such that, in case of its failure, the preset speed and direction of thrust of propellers shall be maintained until local control is in operation.
In particular, the failure of electrical, pneumatic or hydraulic power supply shall not imply excessive and sudden variations of propulsive power and direction of thrust of propeller and such a failure shall operate an alarm in the control position. In the case of pitch control systems of controllable pitch propellers, the pitch position shall be maintained practically unaltered at least for a period of time sufficient for the required interventions.
When electronic speed governors of main internal combustion engines form part of a remote control system of such engines, if lack of power to the governor may cause major and sudden changes in the preset speed and direction of thrust of the propeller, back up power supply is to be provided. Moreover, it shall be possible to control the auxiliary machinery, essential for the propulsion and safety of the ship, at or near the machinery concerned.
Where the propulsion system is fitted with clutch couplings, devices to disengage the propeller shaft may also be provided in special cases.
Emergency stopping devices shall be such as to prevent their inadvertent operation.
Any remote control position shall be provided with an emergency stopping device of propulsion machinery which shall be independent of the remote control system and its supply.

SAFETY SYSTEM

Safety systems are intended to operate automatically in case of faults endangering an associated machinery or system, so that:

- normal operating conditions are restored (e.g. by the starting of stand-by unit), or
- the operation of the machinery is temporarily adjusted to the prevailing abnormal conditions (e.g. by reducing the output of the associated machinery), or
- the machinery is protected from critical conditions by shutting off the fuel or power supply thereby stopping the machinery (shutdown).

In order to avoid undesirable interruption of the operation of the associated machinery and systems, when two or more safety actions are released by one failure condition, the safety system is to intervene sequentially after the operation of the alarm system by:

- starting of standby units
- load reduction or shutdown, such that the least drastic actions is taken first.

This may be achieved activating those safety actions at different levels of intervention.
Times of intervention shall be adequate to the characteristics of the associated machinery and systems.
The operation of the safety system shall cause an alarm; in particular an alarm shall be given in the case of automatic change-over from an auxiliary machinery to the stand-by one and in the case of automatic shutdown of the associated machinery or systems.
When the safety system has been activated, means shall be provided to trace the cause of safety action (e.g. visual and audible alarm signals, indications, etc.).
The safety system shall be provided with a manual reset.
Where arrangements for overriding the safety system of the main propulsion machinery are provided, they shall be such as to preclude their inadvertent operation. Visual means shall be provided to indicate in the relevant control position when the over ride has been activated.

The safety systems shall be designed to "fail safe". The characteristics of "fail safe" of a system shall be evaluated on the basis not only of the safety system itself and its associated machinery, but also on the inclusion of the whole machinery installation as well as the ship.

The safety system shall be arranged to be powered by a stand-by power supply in the event of loss of normal power supply. If this is achieved by a change over from the normal power supply to the stand-by power supply, the change over shall be automatic.
The failure of the normal power supply and of the stand-by one shall cause visual and audible alarm indications.
The safety systems of different units of main propulsion system shall operate independently so that a failure of the safety system of one unit will not interfere with the operation of the safety system of another unit of the same propulsion system.
The aforesaid safety system intended for the functions is to be independent of all other control and alarm systems, so that failure or malfunction in these systems will not prevent the safety system from operating.
For particular parameters, the use of common sensors for the safety and alarm systems may be accepted provided that the two systems can be suitably decoupled or other arrangements to check the efficiency of sensor are fitted.
Automatic starting of stand-by units shall be initiated by the parameter which is being monitored (e.g. pressure or temperature signal, etc.) when the parameter reaches predetermined values which indicate abnormal operation of machinery or system.

For the automatic starting system of the stand-by units, the following requirements shall apply:
(a) faults in the electrical or mechanical system of the running machinery will not prevent the stand- by machinery from being automatically started;
(b) when a machinery is stand-by, ready to be automatically started, this is to be clearly indicated at its control position;
(c) the change over to the stand-by unit shall be indicated by visual and audible alarm;
(d) in way of machinery, means are to be provided to inhibit undesirable
automatic or remote starting (e.g. when the machinery is being repaired);
(e) in the case of reciprocating internal combustion engines, the number of consecutive automatic starting attempts, where applicable, shall be limited to safeguard sufficient starting power for local manually operated starting;

(f) automatic starting shall be inhibited when conditions are present which could endanger the stand-by machinery.

For shutdown system of machinery, the following requirements shall apply:
(a) the system shall in any case be independent of control and alarm systems, as far as sensors are concerned;
(b) when the system has stopped a machinery, this latter shall not be restarted automatically before a manual reset of the safety system has been carried out.
The safety systems shall in any case be tested.

COMPUTER BASED SYSTEMS

According to the RINA rules and to the applicable normative in force [4], [5] computer based systems are to provide functions in a safe, stable and repeatable manner during normal and abnormal conditions.
In case of failure of a computer based system, the process is to automatically revert to a safe condition and an alarm is to be given. Moreover, if essential services are managed by computer systems, a secondary independent means, of appropriate diversity, is to be provided.
As far as the system configuration is concerned, logical security arrangements are necessary to prevent unauthorised access to functions or alteration of configuration, program or data.
The hardware and software are to be of modular design in order to minimise the effects of a single failure and to facilitate the maintenance operation. This apply also to the communication networks, interconnecting the computer stations present in the system. Where redundant data communication links are required, switching between the links themselves shall be automatic.
Alarm control and safety functions are to be configured such that functions are mutually independent.
If integrated systems are used, the operation shall be at least as effective as it would be with individual stand-alone equipment or systems.
Failure of one part of the integrated system is not to affect the functionality of other parts, except for those functions directly dependent on the defective part.
Moreover, multi-function displays and controls shall be redundant and interchangeable. The number of units at control stations shall be sufficient to ensure that all functions may be provided with anyone unit out of operation, taking into account any functions which are required to be continuously available.

INTEGRATED AUTOMATION SYSTEMS FOR PROPULSION

At present, on the basis of the more and more growing reliability of electronic components as well as of microprocessors based systems, the trend coming from the system designers is to employ them also for the essential services aboard ships.
The integration is possible not only locally between the monitoring, control and safety for a single machine, but also in a more extended structure among several and different kinds of machines and/or devices.

In case the electronics involves essential services, like the propulsion, particular attention is to be paid for the configuration of the system, evaluating the level of integration and the consequences of the major faults, according to the well-known standard IEC 812 "Failure Mode & Effects Analysis" [5].

On the fast ships, since the common practice is to install more than one propulsion engine, the risks which could lead to total loss of propulsion are minimised, as far as the automation is concerned, being anyway necessary to evaluate and to test the entire plants, to be sure that, even with one engine out of service, the remaining one(s) is/are able to operate.

The integration of the systems is normally reached through data networks, whose communication protocols are chosen on the basis of the necessities of the application (transmission speed, MTBF, maximum cable length, etc.). The organisation between different protocols is examined in the next paragraph.

HIERARCHY OF AUTOMATION PLANT AND COMMUNICATION PROTOCOLS

As Fig. 1 evidences, it is possible to distinguish three levels of automation (following a top-down approach): the device level, the control level and the information level.
In order to meet the needs of the system in which they are utilised, particular communication techniques are dedicated to each of these.
System designers have to take particular efforts in order to get the different protocols communicated one another, without degrading the global performance.
Some details about the most used protocols at present are provided in the following paragraphs.

DeviceNet

It is a network providing connections between field devices, i.e. sensors and actuators, and hardware controllers at higher lever, such as PLC or PCs.
DeviceNet can be used following either the master/slave or the peer-to-peer philosophy. Moreover, it is an open network toward the devices manufactured by a certain number of vendors. DeviceNet is able to provide cheap communication links for interconnecting several and different devices without employing I/O units, being an open network standard: vendors are not required to use software, hardware or licenses to connect their products to it, provided that their products are included in the lists issued by the Open DeviceNet Vendor Association (ODVA). Devices being not compatible with the standard can be linked to the network by I/O modules or dedicated adapters.
The wiring is based on a radial network topology.

DeviceNet is based on the CAN specification (see further); so, its data transfer rate depends on the distance and the number maximum of nodes is 64, according to the following Tab.4:

Architecture	Master/slave or peer-to-peer
Number maximum of nodes	Generally 64
Message size	1 bit - multiple bytes
Data transfer rate	125 kbps, up to 500 meters 250 kbps, up to 250 meters 500 kbps, up to 100 meters

Tab.4 - DeviceNet main data

Launched in 1994 by Allen-Bradley, DeviceNet has been managed subsequently by ODVA, that has increased its membership up to about 140 companies.
ODVA was born in 1995 and is a non-profit independent association with the scope to manage the DeviceNet technology.
ODVA activities comprehend also compliance tests, distribution of catalogues, producer assistance and development of specific device profiles.

ControlNet

ControlNet is a product by Rockwell Automation since 1995. It is based on a peer-to-peer communication network, linking PLC and PC to I/O subsystems or subnetworks at a rate up to 5 Mbit/sec. ControlNet is able to support 99 addressable nodes along the trunk cable path and its position is identified between Ethernet and the field network. The technique of data exchange is based on the model Producer/Consumer, in which each node can be a sender and/or a receiver of data.

Ethernet

Since 1970, when it was developed at the Xerox Palo Alto Research Center, it has become the most used standard for interconnecting the computers. The first Ethernet rate was about 3MB/sec. After 10 years, a consortium formed by manufactures gave raise to the DEC-Intel-Xerox (DIX) standard, which had Ethernet become an open, high quality system operating at 10MB/sec, so that it was adopted by the Local Area Networks (LAN) Committee of the Institute of Electrical & Electronics Engineers (IEEE 802). The first formal issue of IEEE standard dates back to 1985, having the title "IEEE 802.3 Carrier Sense Multiple Access with Collision Detection (CSMA/CD) Access Method and Physical Layer Specifications". From then on, the right way to refer to this standard is "IEEE 802.3 CSMA/CD" technology, even if, practically, Ethernet is the most known and used name.
Furthermore, the IEEE standard was acquired by the International Organisation for Standardisation (ISO), which made Ethernet a standard world-wide accepted.

The above standard is continuously updated with new technologies (e.g. 10MBps, or twisted pair media, 100MBps, or Fast Ethernet), whose the latest one is the Gigabit Ethernet, or 1000Base-T.
From technical point of view, each station of the network operated with autonomy, as there is no a central controller. The interface between a station and the Ethernet is provided by the Medium Access Control (MAC) technique, contained in the network card installed in each station. The signals in the Ethernet are transmitted serially to every operating station: to send data package, the station waits for having the channel as free, and then starts transmitting. All stations on Ethernet are obliged to contend with 'democracy' the opportunity to transmit, packet by packet of data.

Profibus

Profibus was created in Germany and now is an European standard, based on the standard EN50170.
Profibus is formed by three protocol levels, designed to meet the market demands. They are Profibus FMS, Profibus DP and Profibus PA.
Profibus FMS (Field Message System), a high-level control network, is based on RS485 protocol and is used for connecting PLCs, drives and other processors at the control network level. Some leader companies employ the protocol FMS using different brand name, i.e. Siemens which produces the Simatic Net Profibus.

Profibus-DP is dedicated to high-speed remote I/O communication and is overall used at the device level.
Profibus-PA is a field bus network for the process industries, designed for use in non-intrinsically safe and intrinsically safe applications, such as in hazardous areas.

Some technical data are provided in the following table Tab.5.

	FMS	DP	PA
Employment	Control network	Device network	Field/Device network
Structure	Master/slave with token passed master-by-master	Master/slave with token passed master-by-master	Master/slave with token passed master-by-master
Transfer rate	12 MB	12 MB	about 30 Kbps

Tab. 5 - Profibus main data

Controller Area Network (CAN)

Developed by Robert Bosch for car application, CAN is a serial bus utilising the "producer/consumer" model of communication. Standardised as ISO 11989, CAN is considered a base technology also for different applications in the ambit of the automation.
As to its operation performance, after sending a message, each other node evaluates if it is relevant, and so, either the node accepts it or node ignores it: no node addressing is present. Each message contains a field, reporting both the content and its priority.
In case of failure of any node, the global performance is affected, but the failure does not have the entire system shut down.

Processes or systems having time response less than 0.5 msec should not be automated by CAN.

Some main technical data are provided in the following Tab.6:

Architecture	event-driven
Number maximum of nodes	generally 64
Message size	Maximum 8 bytes
Data transfer rate	Up to 1 MB, depending on the distance

Tab.6 - CAN main data

Some field networks, e.g. CAN Kingdom, CANopen, DeviceNet, etc., exploit the structure of the original CAN to attach new enhanced applications and new levels of more dedicated performances.

Particular mention is to be done for CANopen and DeviceNet, which are able to manage data exchange between the supervisory control level and small sensors and actuators through one only physical network, without using gateways.

EXAMPLE OF THE AUTOMATION FOR UNMANNED MACHINERY SPACES (IAQ-1)

Taking into account what exposed in the previous paragraphs of this paper, alarms may be individually signalled at a centralised control position or at subsidiary control positions. In this last case there shall be a centralised control position where the subsidiary control position signalling a fault condition is indicated.
Moreover, the alarm system shall be capable, in case of a failure to a machinery, of activating a visual and audible alarm at the engineers' public rooms and at each of the engineers' cabins through a selector switch to ensure connection to at least one of those cabins.
Group alarms may be arranged on the bridge to indicate faults conditions. Alarms associated with faults requiring speed reduction or the automatic shut down of propulsion machinery are to be separately identified.
It is noticeable that the engineer's alarm shall be capable of being operated automatically when a machinery or system alarm has not received attention locally within a limited time.

Fire alarm and alarm of protection against flooding complete the list of signals to be given for an unmanned machinery space.

Fire detection system shall initiate audible and visual alarms, distinct in both respects from the alarms of any other system not indicating fire, in sufficient places to ensure that the alarms are heart and observed on the navigating bridge and by a responsible engineer officer, e.g. initiating the alarm at the engineering personnel accommodations and at each of the engineer's officers cabins through a selection switch to ensure connection to at least one of those cabins.

As to the protection against flooding, bilge level alarms shall form part of alarm systems for machinery and shall be indicated at the control position, engineering accommodations and navigating bridge. Moreover, bilge well high level alarms shall be grouped in at least two independent systems.

An example of integrated automation system for a propulsion plant is provided in Fig.2. In the Engine Control Room (ECR) two workstations are present, being connected to the control network and continuously powered by independent supply systems. Electronic cards relevant to monitoring, control and safety are located inside two PLC cabinets, PLC1 and PLC2. PLC1 manages the two propulsion engines Eng.1 and Eng.2, installed in Engine Room (ER), by different cards, so that a failure involving one card does not affect the good operation of the other engine. PLC1, Eng.1 and Eng.2 are linked up through a network loop.
The electrical power station is integrated is integrated in the ship automation through PLC2, who is organised with the same criteria as the propulsion system.
PLC1 and PLC2 can carry out other functions, provided that following a failure there is no negative influence among the electronic cards and that a single major fault does not cause the loss of the entire propulsion and/or the entire electrical energy generation.

TESTS OF EQUIPMENT USED IN AUTOMATION SYSTEM

The compliance with the rules and the level of reliability are normally evaluated through the type approval procedure and the acceptance tests.

The type tests involves the type approval procedure, who is compulsory for sensors, actuators, control and monitoring equipment, computers and peripherals, etc., used in the automation systems.
The type tests are to be performed on specimens of the equipment to demonstrate the ability of the equipment to operate properly under the testing conditions. Vibration tests and salt mist tests may be performed on different specimens. The other tests may be performed on the same specimen. Reset between the tests may be allowed.
All tests which shall be performed with the equipment in normal operation shall be carried out in the expected supply conditions and under normal load.
Tab. 7 appended at the end of the paper reports the complete list of the above mentioned tests together with the relevant specific comments.
The acceptance tests, to be carried out in general at the manufacturer's facilities before the shipment of the equipment, are necessary for testing of those equipment whose prototype has been already type approved and for those arrangements manufactured with the above equipment. The acceptance tests include the following ones, when applicable:

- visual inspection;
- operational test;
- high voltage test;
- hydrostatic test.

No.	Test	Procedure	Test parameters	Severity degree	Other information
1	Visual inspection	-	-	-	- conformance to drawings, design data - quality of workmanship and construction - compliance with applicable RINA Rules
2	Performance test	The equipment specification	Standard atmosphere conditions: - temperature 25°C ± 10°C - relative humidity: 60% ± 30% - air pressure: 96 kPa ± 10 kPa	-	- confirmation that operation is in accordance with the requirements specified for particular automatic systems or equipment - checking of self-monitoring features (where applicable) - checking of specified protection against an access to the memory and effects of unerroneous use of control elements in the case of computer systems
3	Power supply variations a) electric	-	A.C. SUPPLY [1] recovery time 1,5 s [2] recovery time 5 s D.C. SUPPLY	-	- for each combination there shall be demonstrated that the equipment operate satisfactorily within preset limits specified by the manufacturer

Combination	Voltage variation permanent %	Voltage transient [1] %	Frequency variation permanent %	Frequency transient [2] %
1	+6	+20	+5	+10
2	+6	-20	-5	-10
3	-10		-5	
4	-10		+5	
5				
6				

Voltage ripple	Voltage tolerance continuous	Voltage cyclic variation
10%	± 10%	5%

Tab.7 - Type approval tests.

No.	Test	Procedure	Test parameters	Severity degree	Other information
	b) pneumatic and hydraulic		Electric battery supply: - +30% to -25% for equipment connected to the battery during charging or as determined by the charging/discharging characteristics, including ripple voltage from the charging device - +20% to -25% for equipment not connected to the battery during charging - pressure ± 20% - duration 15 min (minimum)		
4	Power supply failure		- 3 interruption during 5 minutes - switching-off time 30 s each case	-	- verification of the specified action of the equipment on loss and restoration of supply in accordance with the systems design - verification that there is no corruption of program or data held in programmable electronic systems (PES), where applicable
5	Dry heat	IEC Publication 68-2-2 IEC Publication 68-2-2A For non heat dissipating equipment: Test B_b For heat dissipating equipment: Test B_d	Temperature: +55°C ± 2°C Duration: 16 h Temperature: +70°C ±2°C [1] Duration: 2 h	C1 C2	- equipment operating during conditioning and testing with cooling systems on, if provided - functional test during the last hour at the test temperature to verify that the equipment operates satisfactorily within the limits stated by the manufacturer - performance test after recovery to verify that the characteristics of the equipment remain unaltered
6	Damp heat, cycling	IEC Publication 68-2-30 Amendment 1 Test D_b	- Temperature: +55°C ± 2°C - Humidity: 95% - Duration: 2 cycles (12 + 12 hours)		- measurement of insulation resistance before test - equipment operating during the complete first cycle and switched off during second cycle except for functional test - functional test during the first 2 hours of the first cycle at the test temperature and during the last 2 hours of the second cycle at the test temperature - recovery at standard atmosphere conditions - insulation resistance measurement and performance test (4 h to 6 h after the recovery period)

No.	Test	Procedure	Test parameters	Severity degree	Other information
7	Vibration	IEC Publication 68-2-6 Test F_c	– 2 +3/-0 Hz to 13,2 Hz: amplitude ± 1 mm 13,2 Hz to 100 Hz: acceleration ± 0,7 g	V1	– duration of endurance in case of no resonance condition: 90 minutes at 30 Hz – duration of the endurance at each resonance frequency at which Q ≥ 2 is recorded: 90 min – during the vibration test, operational conditions are to be demonstrated – tests to be carried out in three mutually perpendicular planes – it is recommended as guidance that Q does not exceed 5
			– 2 +3/-0 Hz to25 Hz: amplitude ± 1,6mm 25 Hz to100 Hz: acceleration ± 4,0 g [2]	V2	– duration 120 minutes, if sweep test is chosen in case several resonance frequencies are detected close to each other
8	Inclination	IEC Publication 92-504	Static: 22,5°	-	(a) inclined to the vertical at an angle of at least 22,5° (b) inclined to at least 22,5° on the other side of the vertical and in the same plane as in (a) (c) inclined to the vertical at an angle of at least 22,5° in plane at right angles to that used in (a) (d) inclined to at least 22,5° on the other side of the vertical and in the same plane as in (c) The period of testing in each position should be sufficient to fully evaluate the behaviour of the equipment
			Dynamic: 22,5° 0,1 Hz		Using the directions defined in (a) to (d) above, the equipment is to be rolled to an angle of 22,5° each side of the vertical with a period of 10 s. The test in each direction is to be carried out for not less than 15 min.

<table>
<tr><th>No.</th><th>Test</th><th>Procedure</th><th>Test parameters</th><th>Severity degree</th><th>Other information</th></tr>
<tr><td>9</td><td>Insulation resistance</td><td></td><td>
<table>
<tr><th rowspan="2">Rated supply U_n V</th><th rowspan="2">Test voltage V</th><th colspan="2">Min. insulation resistance M Ω</th></tr>
<tr><th>Before</th><th>After</th></tr>
<tr><td>$U_n \leq 65$</td><td>$2U_n$ min 24 V</td><td>10</td><td>1,0</td></tr>
<tr><td>$U_n > 65$</td><td>500</td><td>100</td><td>10</td></tr>
</table>
</td><td></td><td>- insulation resistance test is to be carried out before and after: damp heat test, cold test and salt mist test: between all circuits and earth and on the supply terminals where appropriate
- certain components e.g. for EMC protection may be required to be disconnected for this test</td></tr>
<tr><td>10</td><td>High voltage</td><td></td><td>
<table>
<tr><th>Rated voltage U_n V</th><th>Test voltage (A.C. voltage 50 Hz or 60 Hz) V</th></tr>
<tr><td>$U_n \leq 65$</td><td>$2U_n + 500$</td></tr>
<tr><td>$65 < U_n \leq 250$</td><td>1500</td></tr>
<tr><td>$250 < U_n \leq 500$</td><td>2000</td></tr>
<tr><td>$500 < U_n \leq 690$</td><td>2500</td></tr>
</table>
</td><td></td><td>- separate circuits are to be tested against each other and all circuits connected with each other tested against earth
- printed circuits with electronic components, which could be damaged, may be removed during the test
- period of application of the test voltage: 1 min</td></tr>
<tr><td>11</td><td>Cold</td><td>IEC Publication 68-2-1
For non heat dissipating equipment: Test A_b
For heat dissipating equipment: Test A_d</td><td>Temperature: +5°C ± 3°C
Duration: 2 h

Temperature: -25°C ± 3°C [3]
Duration: 2 h</td><td>F1

F2</td><td>- initial measurement of insulation resistance
- equipment not operating during conditioning and testing except for functional test
- functional test during the last hour at the test temperature
- insulation resistance measurement and the functional test after recovery</td></tr>
</table>

No.	Test	Procedure	Test parameters	Severity degree	Other information
12	Salt mist [4]	IEC Publication 68-2-52 Test K_b	Four spraying periods with a storage of 7 days after each	-	- initial measurement of insulation resistance and initial functional test - equipment not operating during conditioning - functional test on the 7th day of each storage period - insulation resistance measurement and operational test 4 h to 6 h after the recovery period
13	Electrostatic discharge	IEC Publication 1000-4-2	Contact discharge: 6 kV Air discharge: 8 kV Interval between single discharges: 1 sec. No. of pulses: 10 for polarity According to level 3 severity standard	-	- to simulate electrostatic discharge as may occur when persons touch the appliances - the test is to be confined to the points and surfaces that can normally be reached by the operator - performance criterion B [5]
14	Radiated radio frequency	IEC Publication 1000-4-3	Frequency range: 80 MHz to 1 GHz Modulation: 80% AM at 1000 Hz Field strength: 10 V/m Frequency sweep rate: $\leq 1{,}5 \times 10^{-3}$ decades/s (or 1%/3 sec) According to level 3 severity standard	-	- to simulate electromagnetic fields radiated by different transmitters - the test is to be confined to the appliances exposed to direct radiation by transmitters at their place of installation - if for tests of equipment an input signal with a modulation frequency of 1000 Hz is necessary, a modulation frequency of 400 Hz should be chosen - performance criterion A [6]
15	Conducted audio frequency	IEC Publication 945	A.C. Frequency range: rated frequency to 200th harmonic Test voltage (rms): 10% of supply to 15th harmonic reducing to 1% at 100th harmonic and maintain this level to the 200th harmonic Maximum power: 2 W D.C. Frequency range: 50 Hz - 10 kHz Test voltage (rms): 10% of supply but at least 3 V Maximum power: 2 W	-	- to simulate distorsions in the power supply system generated for instance, by electronic consumers and coupled in as harmonics - method of the test in accordance with IEC 945 - performance criterion A [6]

No.	Test	Procedure	Test parameters	Severity degree	Other information
16	Conducted radio frequency	IEC Publication 1000-4-6	Frequency range: 10 kHz - 80 MHz Amplitude: 3 V (rms) [7] Modulation: 80% AM at 1000 Hz Frequency sweep range: $\leq 1{,}5 \times 10^{-3}$ decades/s (or 1%/3 sec) According to level 2 severity standard	-	- to simulate electromagnetic fields coupled as high frequency into the test specimen via the connecting lines - if for tests of equipment an input signal with a modulation frequency of 1000 Hz is necessary, a modulation frequency of 400 Hz may be chosen - performance criterion A [6]
17	Burst/Fast Transients	IEC Publication 1000-4-4	Single pulse time: 5 ns (between 10% and 90% value) Pulse width: : 50 ns (50% value) Amplitude (peak): 2 kV line/line on power supply port; 1 kV lines/earth on I/O data control and signal lines Pulse period: 300 ms Burst duration: 15 ms Duration/polarity: 5 min According to level 3 severity standard	-	- arcs generated when actuating electrical contacts - interference effect occurring on the power supply, as well as the external wiring of the test specimen - performance criterion A [6]
18	Surge/Slow Transient	IEC Publication 1000-4-5	Pulse rise time: 1,2 µs (between 10% and 90% value) Pulse width: 50 µ s (50% value) Amplitude (peak): 2 kV line/earth; 1 kV line/line Repetition rate: ≥ 1 pulse/min No. of pulses: 5 per polarity Application: continuous According to level 3 severity standard	-	- to simulate interference generated for instance, by switching "ON" or "OFF" high power inductive consumers - test procedure in accordance with figure 10 of the standard for equipment where power and signal lines are identical - performance criterion A [6]
19	Radiated Emission	CISPR 16-1 CISPR 16-2	FOR EQUIPMENT INSTALLED IN THE BRIDGE AND DECK ZONE: Frequency range — Limits 0,15-0,3 MHz — 80-52 dBµ V/m 0,3-30 MHz — 52-34 dBµ V/m 30-1000 MHz — 54 dBµ V/m except for: 156-165 MHz — 24 dBµ V/m FOR EQUIPMENT INSTALLED IN THE GENERAL POWER DISTRIBUTION ZONE: Frequency range — Limits 0,15-0,168 MHz — 80-75 dBµ V/m 0,168-30 MHz — 75-50 dBµ V/m 30-100 MHz — 60-54 dBµ V/m 100-1000 MHz — 54 dBµ V/m except for: 156-165 MHz — 24 dBµ V/m	-	- procedure in accordance with the standard but distance 3 m between equipment and antenna

No.	Test	Procedure	Test parameters	Severity degree	Other information
20	Conducted emission	CISPR 16-1 CISPR 16-2	FOR EQUIPMENT INSTALLED IN THE BRIDGE AND DECK ZONE: Frequency range — Limits 10-150 kHz — 96-50 dBμ V 150-350 kHz — 60-50 dBμ V 350 kHz-30 MHz — 50 dBμ V FOR EQUIPMENT INSTALLED IN THE GENERAL POWER DISTRIBUTION ZONE: Frequency range — Limits 10-150 kHz — 120-70 dBμ V 0,15-1,5 MHz — 80-69 dBμ V 1,5-30 MHz — 69 dBμ V	-	-

NOTES :

[1] Dry heat at 70°C is to be carried out for equipment located in not air conditioned spaces, or adjacent to hot surfaces or subject to heating due to appliances in the proximity or exposed to sun radia tion.

[2] For severe vibrations such as, e.g. on diesel engines, air compressor, etc.
Very special conditions may exist for example on exhaust manifold of diesel engines especially for medium and high speed engines. Values which may be required in these cases are: 40 Hz t 2000 Hz: acceleration ± 10,0 g at 600°C.

[3] The test shall be applied to equipment installed in non-weather protected location or in spaces protected but not heated such as pump rooms, holds, etc.

[4] Salt mist test to be carried out for equipment to be installed in non-weather protected areas, for example on open deck.

[5] Performance criterion B: (for transient phenomena): the EUT shall continue to operate as intended after the tests. No degradation of performance or loss of function is allowed as defined in th technical specification published by the manufacturer. During the test, degradation or loss of function or performance which is self recoverable is however allowed but no change of actual operatin state or stored data is allowed.

[6] Performance criterion A (for continuous phenomena). The EUT shall continue to operate as intended during and after the test. Degradation of performance or loss of function is allowed, as define in the technical specification published by the manufacturer.

[7] For equipment installed on the bridge and deck zone the test levels shall be increased to 10 V rms for spot frequencies in accordance with IEC 945 at 2 / 3 / 4 / 6,2 / 8,2 / 12,6 / 16,5 / 18,8 / 22 25 MHz.

CONCLUSIONS AND FUTURE TREND

Automation is continuously in evolution. At present the level of the automation can be divided into three categories as already shown in Fig. 1. The latest developments of Ethernet and the other protocol technologies are bringing toward the possibility to develop and design the system architecture with only two levels: the trend consists of gradually eliminating the control level, by getting the Ethernet interfaced directly with the PLC or PC operating on the field network, as Fig.3 highlights.

At present, the opportunity of using Ethernet at the device level is being evaluated also with reference to the costs: the commercial value of an Ethernet card is about $20, while a similar device for other protocols costs up nine-ten times more.

These facts could lead to further developments based on the Ethernet technology, that has got all the characteristics to become a real standard in the world of data communication.

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[3] R.I.NA., Germanischer Lloyd, Bureau Veritas, *Rules for the construction and classification of high speed craft* - Edition 1998.

[4] IEC Publication 60092-504, *Electrical Installations in Ships - Part 504: Special features - Control and instrumentation*, Second Edition, 1994.

[5] IEC Publication 812, *Analysis and Technique for System Reliability - Procedure for Failure Mode and Effect Analysis (FMEA)*, First Edition, 1985.

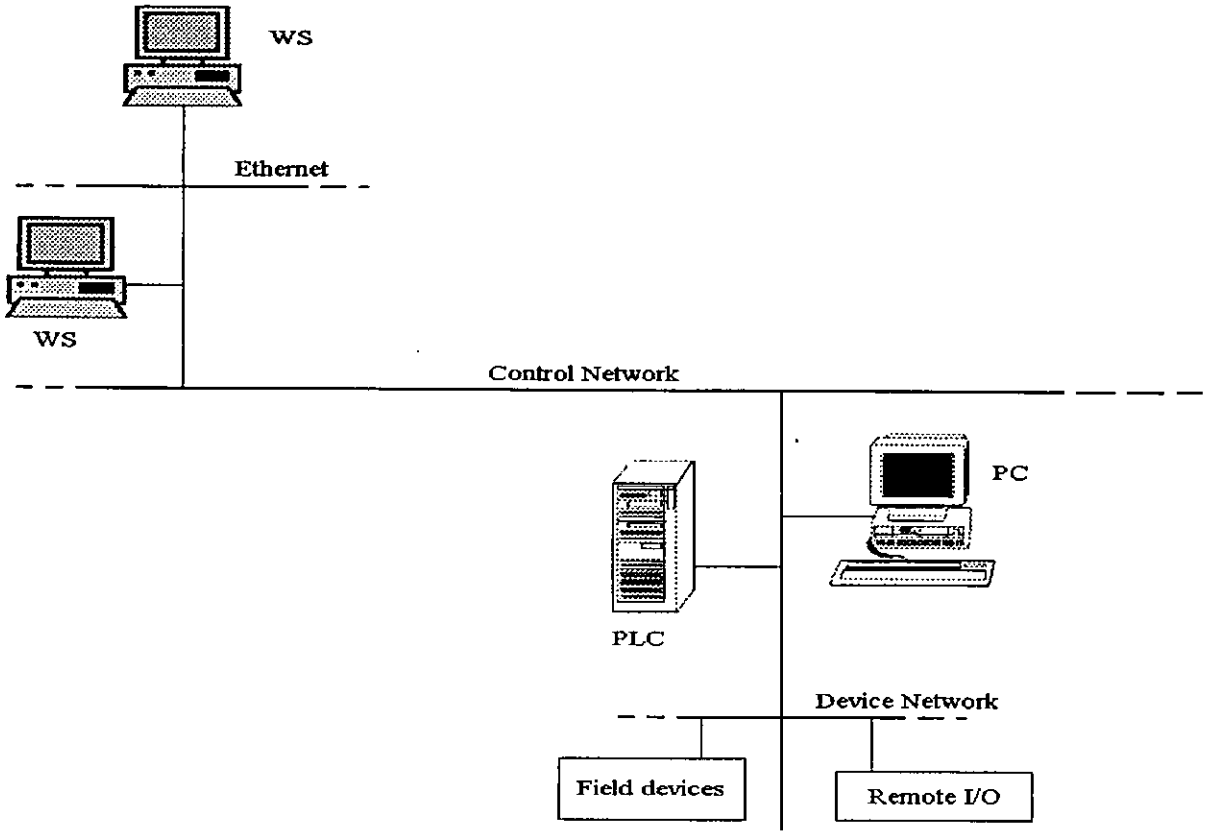


Fig.1 - Data communication levels.

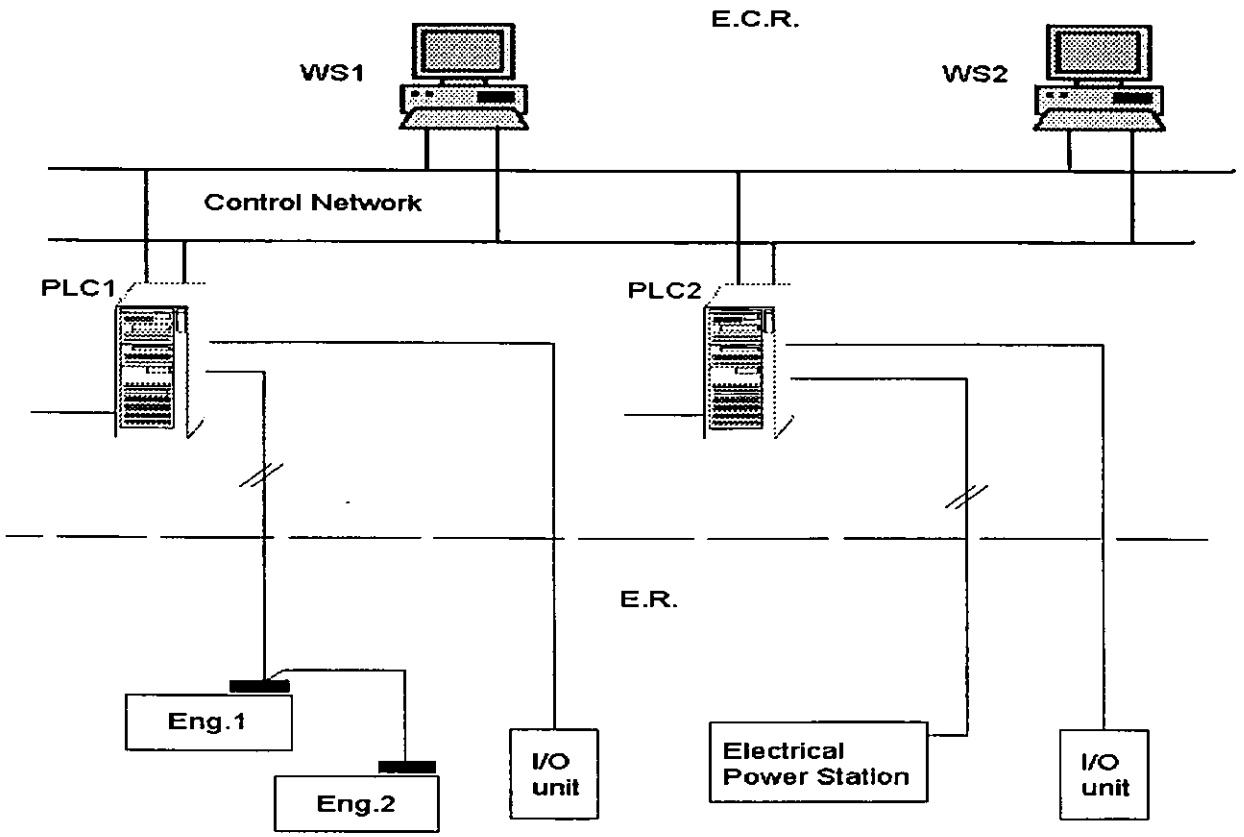


Fig. 2 - Example of IAQ-1 RINA classified automation system for propulsion plant.

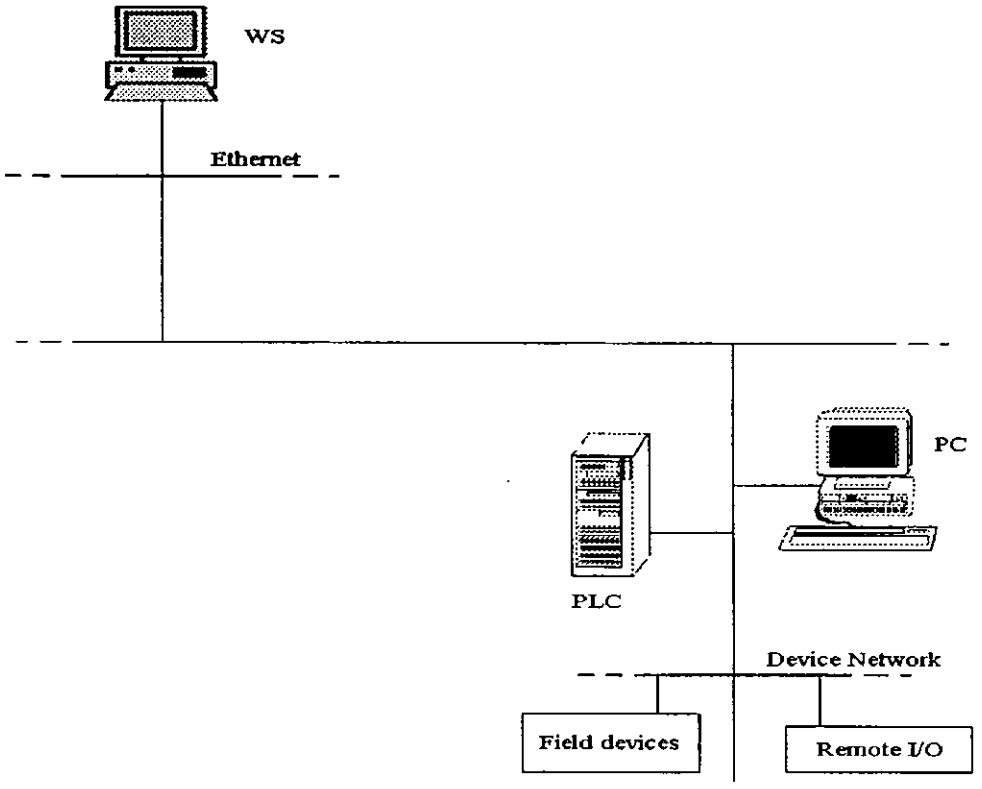


Fig. 3 - Possible future evolution of data communication levels.

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

AIR POLLUTION PREVENTION IN PROPULSION AND AUXILIARY MACHINERY

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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AIR POLLUTION PREVENTION IN PROPULSION AND AUXILIARY MACHINERY

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ABSTRACT

The paper describes the method adopted for certification of machinery installations according to the new Annex VI to the MARPOL Convention on prevention and control of air pollution from ships, adopted by The Conference of MARPOL Parties, convened in London on 26 September 1997, and in particular focuses on the engine related NOx emissions certification.
It further analyses some aspects of the existing normative for Classification which are peculiar for HSC propulsion plants.

INTRODUCTION

The Conference of MARPOL Parties, convened in London on 26 September 1997, adopted the new Annex VI to the MARPOL Convention on prevention and control of air pollution from ships [1], which is to be fully applied to ships of 400 tons gross tonnage or above, while for ships below that tonnage, appropriate survey requirements may be established by the relevant Administration.

The date of entry into force of the new Annex VI will be twelve months after the date on which not less than fifteen States, the combined merchant fleets of which constitute not less than 50 per cent of the gross tonnage of the world's merchant shipping, have become parties to the relevant Protocol.
At the time being the said new Annex is not yet entered into force and this date of entry into force cannot be anticipated as being very close.

The regulations of the new Annex VI to the MARPOL Convention are relevant to:

- the emission of ozone depleting substances from fire fighting and refrigerating plants
- the emission of NOx from main and auxiliary engines
- the emission of volatile organic compounds during cargo loading/transfer operations on oil, chemical and gas tankers
- the characteristics of incinerators the characteristics of fuels.

THE MARPOL ANNEX VI

The Annex VI foresees the issuance of an International Air Pollution Prevention Certificate (IAPP Certificate) by the ship's flag Administration or a Recognised Organisation acting on its behalf.

Ships constructed before the date of entry into force of the new Annex VI will be issued by the IAPP Certificate not later than the first scheduled drydocking after the date of entry into force, but in no case later than three years after this date.

Some regulations will apply only to ships and installations constructed or subject to major conversion on or after the date of entry into force of Annex VI or other dates indicated in the said Annex, whilst other regulations will involve also existing ships and installations.

1. *The emission of ozone depleting substances from fire fighting and refrigerating plants (regulation 12)*

   The installation of new plants containing the said substances, as defined in reg. 2(6), on or after the date of entry into force of Annex VI will be prohibited.
   Nevertheless new installations containing hydro-chlorofluorocarbons (R22) will be permitted until 1 January 2020.
   As a consequence of the above restrictions, maintenance, servicing and repairs of existing plants might suffer of the reduction of the availability of these substances.

   It is to be noted that the use of Halons in new fixed fire fighting installations has already been banned by SOLAS since 1 October 1994 (see SOLAS II-2/5.3.1).

2. *The emission of NOx from main and auxiliary engines (regulation 13)*

   The limitations of the NOx content of exhaust gases from engines will be applied to:
   - engines with a power output greater than 130 kW installed on ships constructed on or after 1 January 2000 and engaged in international voyages;
   - engines with a power output greater than 130 kW which undergo major conversion, as defined in regulation 13.2(a), on or after 1 January 2000; this includes the renewal of engines on existing ships with new engines.

   The above limitations will not be applied, independently from their power output, to
   - emergency diesel engines;
   - engines installed in lifeboats;
   - engines intended solely for use in emergencies;
   - engines installed on ships solely engaged in domestic trade provided that such engines are subject to an alternative NOx control measures established by the ship's flag Administration.

   NOx emissions from boilers and gas turbines are not regulated by Annex VI, at the time being.

Compliance with regulation 13 will be mandatory only when Annex VI will enter into force; nevertheless, being this regulation retroactive for engines installed on ships constructed on or after 1 January 2000 and engaged in international voyages or engines which undergo major conversion on or after this date, it is recommanded that the engines subject to this regulation 13 are, since now:

- *certified* in accordance with the *NOx Technical Code;*
- supplied with an *approved Technical File* ;
- maintained in continuing compliance with the relevant NOx emission limits including *documentary evidence* of them.

The term *certified* means that the engine has been pre-certified by an authorised Authority such as the ship's flag Administration or a Recognised Organisation acting on its behalf (see IMO Circular MEPC/Circ. 344).
The term *approved Technical File* means a technical file meeting the requirements of item 2.4.1 of the *Nox Technical Code* and approved by an authorised Authority such as the ship's flag Administration or a Recognised Organisation acting on its behalf.
The term *documentary evidence* means a properly completed Record Book of engine parameters or records of onboard emission testing either by the simplified measurement method or by direct measurement and monitoring all in accordance with chapter 6 of the *NOx Technical Code.*

3. *The emission of volatile organic compounds during cargo loading/transfer operations on oil, chemical and gas tankers (regulation 15)*

The requirements relevant to emission control of volatile organic compounds during loading operations of oil, chemicals and gases will be applied when annex VI will enter into force both to new and existing ships.
The ship installation and operations will have to comply with the IMO Circular MSC/Circ. 585.

The majority of involved ships will be probably found already prepared as quite similar limitations are presently requested to be applied by many Administrations.

4. *Incinerators (regulation 16)*

Incinerators installed on or after 1 January 2000 on ships engaged in international voyages will have to be constructed in accordance with the IMO Standard specification (MEPC/Circ. 76(40)).
Certain operational limitations established in the new Annex VI, such as incineration not allowed of residues of cargoes subject to Annex I,II and III to MARPOL 73/78 Convention and related contaminated packing materials as well as of polychlorinated biphenyls etc., will be applied to existing incinerators too.

Compliance with regulation 16 will be mandatory only when Annex VI will enter into force; nevertheless, being this regulation retroactive for incinerators installed on or after 1 January 2000 on ships engaged in international voyages, it is recommanded that the incinerators subject to regulation 16 are, since now:

- *certified* in accordance with the resolution *MEPC 76(40);*
- supplied with a *Manufacturer Operating Manual;*
- maintained in continuing compliance with the prescribed limits.

The term *certified* means that the incinerator is type approved by an authorised Authority such as the ship's flag Administration or a Recognised Organisation acting on its behalf.

Incinerators with capacities greater than 1500 kW, to which, by definition, resolution MEPC 76(40) is not applicable, will be considered by the flag Administration on a case by case basis.

Apart from the above Annex VI requirements, incinerators installed on or after 1 January 1999 on ships entitled to fly the flag of a European Community Country have to comply with the European Commission Directive 98/85/EC and are to be certified according to Annex 1 to the said Directive; the testing standard requested by this Directive is the same MEPC/Circ. 76(40) of the new Annex VI.

5. *The characteristics of fuels (regulations 14 and 18)*

The global sulphur content cap in fuel oils used on board ships, regardless of their use, is fixed equal to 4.5%. A stricter limit of 1,5% is required for ships operating in the *SOx emission control areas.*
Presently the Baltic Sea Area is designated a *SOx emission control area* and the North Sea States will be probably designated in the next future.

Regulation 18 identifies those additives, substances and chemical wastes which are allowed or prohibited (the same requirements are indicated in the ISO 8217 Specification for Marine Bunker Fuels).
Representative samples of fuel oil shall be taken and maintained on board for a minimum period of 12 months and a bunker delivery note shall be kept on board readily available for inspection and for a period of three years after the fuel has been delivered on board.

The above restrictions are mandatory when the new Annex VI will enter into force.

Advanced application of the new Annex

Even though the new Annex VI to MARPOL is not yet entered into force, the regulations 13 and 16 concerning diesel engines and incinerators will be retrospectively applied, as detailed above.

Therefore it is recommended that diesel engines and incinerators actually installed on board be in compliance with the pertinent parts of the above regulations, whilst not mandatory, for the following reasons:

- it will be more difficult and expensive to upgrade existing equipment to meet specifications for which it was not designed than install equipment already complying with the new specifications;
- some States could introduce local requirements which anticipate those of Annex VI.

For the above reasons, also the IMO Marine Environment Protection Committee on November 19. 1998 issued a Circular (Circ.344) recommending the use of "Interim Guidelines for the Application of the NOx Technical Code" included thereto, inviting the Administrations to issue (or authorise the issuance by organisation acting on their behalf) a "Statement of Compliance" at satisfactory compliance with the Code requirements, for each engine which will become retrospectively subject to the provisions of Regulation 13.

The Administrations are also urged to take into consideration the Statement of Compliance after entry into force of the Annex VI, whether or not the statement has been issued by their administration, or by other Governments.

NOx Emissions from Diesel Engines

According to Regulation 13 Annex VI, all the engines with:

- a power output of more than 130 kW installed on board of a ship constructed on or after 01.01.2000,
- a power output of more than 130 kW built or that undergo a major conversion on or after 01.01.2000

are subject to the limits of the nitrogen oxides emissions in g/kWh, fixed on the base of the engine speed expressed in rpm as follows:

17 g/kWh	when rated engine speed is less than 130 rpm
45 * n (-0.2) g/kWh	when rated engine speed is 130 or more but less than 2000 rpm
9.8 g/kWh	when rated engine speed is 2000 rpm or more

where "n" is the rated engine speed

Pre-certification

The tests for certification of compliance can be carried out for the Interim Certification of the single engine or of the "Parent Engine" for an engine family/engine group.

On board measurement of exhaust emission from engines, appears not feasible in particular for main engines, if they have not undergone a pre-certification on the test bed, since torque measuring instruments having the accuracy required by the Code are not available; this is why the common engine manufacturer approach has been to carry out all the required tests and checks at the test bed prior to engine delivery.

However it is to be noted that the Code has given to the engine manufacturers the chance of applying the "Engine Family" or "Engine Group" concepts, to serially manufactured engines.

In such a case, a Parent Engine is chosen to be representative of the worse case within the Family or Group, and the exhaust emissions measurement are carried out only for the Parent engine.

The engine Group concept can be adopted for small series of engines which require some minor adjustment during installation or in service on board; the effect of adjustments on NOx emissions, and the allowable range is to be preliminarily tested and approved on occasion of engine group approval.

The engine Family concept can be adopted for any series-produced engine (having all the same basic characteristics, but not specifications) which, through their design, are proven to have similar NOx emissions, are used as produced and require no adjustment or modification which could adversely affect the NOx emissions; minor adjustments through adjustable features are however permitted, provided that the engine complies with the Code requirement for any adjustment within the physically available range.

A further condition for application of engine family/group concept is that the manufacture is to demonstrate that arrangement have been made to ensure effective to control of the conformity of production.

The manufacturer is also required to choose a method among :

1. on board engine parameter check method
2. on board simplified measurement
3. on board direct measurement and monitoring

to comply with periodical survey requirement.

On Board engine parameter check method

The on-board engine parameter check method is to be described in the approved technical file, and shall include a list of all components, setting and running parameters which have influence on NOx emissions:

- for components, a method of identification is to be established.
- for settings and running parameters the allowable limits are to be detailed

in order to enable the surveys according to the engine parameter check method, the shipowner is to mantain on board:

1. a Record Book of Engine Parameters, recording all the changes made to components and settings
2. an Engine parameter list of engine designated components and settings, or load-dependant operating values, approved by the Administration
3. technical documentation of modified engine components, when modification is made to designated engine components

On- board simplified measurement method

The on-board simplified measurement method, is applicable only for confirmation tests to engines which have undergone a pre-certification test on test bed, and for periodical surveys.

Considering the conditions on board, different simplifications in the measurement procedure have been introduced, mainly 10% on the allowable NOx emission limit, but also the accuracy requirements for measuring instruments for torque, fuel consumption, and air consumption have been relaxed.

Where direct measurement of torque or fuel consumption cannot be performed, others means can be adopted such as a declared speed power curve from propeller law governed equipment such as a pump, or reference to fuel consumption from a pre-certification test.

Finally, the on–board measurement may be carried out with engine running on ISO 8217 – RM grade fuel, instead of an ISO 8217 - DM grade fuel, and in this case a further allowance of up to 10% may be granted, except that the total allowance cannot exceed 15% of the applicable limit value.

On- board measurement and monitoring method

When on-board direct measurement of NOx emissions is chosen as a method to show compliance, both spot checks and continuous monitoring with data storage are acceptable, but the device used to monitor and record is to be approved according to guidelines not yet developed, in particular the items which are to be faced are reliability of the equipment, and in particular its ability to maintain accuracy and repeatability.

REFERENCES

[1] IMO "Protocol of 1997 to amend the International Convention for the Prevention of Pollution from Ships, 1973 as modified by the Protocol of 1978 relating thereto", (Annex VI of MARPOL 73/78 and NOx Technical code), IMO Publication No. IMO-664E.

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

AUXILIARY SYSTEMS OF HIGH SPEED CRAFTS

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INTRODUCTION

Scope of this work is to remark the difference between the pumps and piping systems relating to conventional passenger ships and the ones of high speed passenger craft.

In particular it is deemed useful to point out that the safety criteria adopted in planning a conventional passenger ship are based on the necessity that she has to withstand an emergency condition due to a damage disabling any essential machinery and safety system, main and/or auxiliary machinery spaces included, even though it happens during a long voyage far off a place of refuge. Namely, in the worst condition, a conventional passenger ship is requested to wait as long as possible for rescues and, if necessary, she is equipped with life save appliances suitable for a temporary reception of all passengers and crew.

The safety criteria adopted in planning high speed passenger crafts are in principle based on the limited service and to the necessity to be based in a "Base Port" to be identified in the relevant manual route. From an operational point of view they may belong to two different categories.

Category "A" craft is any high speed passenger craft which, operating in predetermined route where it has been dèmonstrated that there is a high probability that, in the event of evacuation at any point of the route, from the base port is possible to set up in due time rescues and to save passengers and crew (max. within 4 hours). These craft cannot carry more than 450 passengers.

Category "B" craft is any high speed passenger craft, other than category "A" craft, equipped with machinery and safety system arranged so that, in the event of a fire or other casualties in any one compartment on board, the craft can return to a port of refuge, within the period of weather forecast validity, using its own power. For this reason the category "B" craft are equipped with at least two independent propulsion plants, located in separate engine rooms, so arranged that a failure of one of them cannot cause the loss of the other.

As regards the aspects of buoyancy and stability of the high speed craft, it is opportune to remark that in the HSC Code the concept of the **"DATUM"** is introduced in lieu of the "Margin Line" considered by SOLAS for the same aspect related to conventional ships.
While the "Margin Line" is a constant feature (76 mm below the upper surface of the bulkhead deck at side), the "Datum" may coincide with a watertight deck but may also be a non-watertight deck which, in such case, is to be covered by a weathertight structure.

In designing the position of openings in the shell plating and in bulkheads and the location of venting pipes, the "DATUM" is to be considered. together with the most unfavourable waterline in flooding conditions.

PUMP AND PIPING SYSTEMS

1. GENERAL

The arrangement of pump and piping systems for essential services of passenger high speed crafts, belonging to both categories "A" and "B", is usually automatic and/or remote controlled from an enclosed area outside the machinery spaces, defined by the HSC Code as "Operating Compartment". Since in such compartment means are also present for navigation, manoeuvring and communication, it usually coincides with the location of the wheel house of a conventional ship. For category "B" crafts an additional control on the spot of machinery is requested.

HSC Code does not preclude the possibility to control only on the spot the machinery and pumps located in engine rooms provided that such rooms are permanently manned. However, in practice, due to the necessity to reduce the costs of craft exercise and considering the reliability of modern automation system, there are not news of high speed crafts with machinery spaces permanently attended.

2. FUEL OIL, LUBOIL AND COOLING WATER SYSTEMS FOR ESSENTIAL MACHINERY

For conventional passenger ships intended for unrestricted service, main and stand-by pumps are required by the rules for the said system.
On the contrary, considered the reduced service of high speed passenger craft of both category "A" and "B", stand-by pumps are not requested. Where the essential machinery is a diesel engine, the main pumps necessary for the above referred systems are normally driven by the same engine.
Although what above may seem a reduction of reliability of the system, it is to be considered that:

- for category "A" crafts, for which redundancy is not requested for propulsion and auxiliary essential services, the safety of the unit is entrusted to the possibility of prompt intervention for assistance and rescue;

- for category "B" crafts, where the redundancy of whole machinery spaces and associated propulsion plant is requested, the default of the stand-by pumps does not preclude the possibility to continue the voyage, although at reduced speed.

In any case, for the piping arrangement regarding the said systems, special consideration is given in connection with the FMEA interpretation.
For the other details of the said systems, concerning the arrangement of filters, of ship side valves and of valves under head of fuel oil tanks, the rules of HSC Code are nearly similar to the ones of SOLAS Convention relating to the usual passenger ships.

3. STARTING EQUIPMENT

There are not particular rules for the starting equipment of auxiliary and propulsion engines. In any case it is opportune to remember that also for the high speed craft, like for conventional ships, the starting equipment using both pneumatic or electric source of energy, are to be enable:

- to ensure that the machinery can be brought into operation from the dead craft condition;
- to start up from a shut-down condition using the starting equipment available in the machinery space.

Also the arrangement of the starting equipment and of the relevant source of energy are to be considered in connection with the FMEA interpretation.

4. BILGE PUMPING SYSTEM

Two possibilities of planning the bilge pumping system of high speed passenger crafts may be adopted according to what follows.

4.1 - CONVENTIONAL BILGE SYSTEMS

Such system is made up by two pumps (category "A" crafts) or three pumps (category "B" crafts) connected to an usual bilge main from which the necessary suction pipes are branched off. The relevant arrangement on board is similar to the one of conventional passenger ships pursuant to SOLAS. In fact::

A) the arrangement of the pumps is to be such that at least one bilge pump is available for use in all flooding condition which the craft is required to withstand according to one of the following possibilities:
   a) one of the required bilge pumps is an emergency pump of reliable submersible type having an emergency source of power located above the "datum";
   b) the bilge pumps and their sources of power are so distributed throughout the length of the craft that at least one pump in an undamaged compartment is available;

B) all valves whose operation is necessary to control the bilge suctions are to be capable of being operated from above the "datum";

C) damage to a bilge pump or its piping connecting to the bilge main has not to put out of service the bilge system;

D) in order to prevent water flowing from one compartment to another, open ends of bilge suction lines located in a compartment different from the ones where the same lines are originating, are to be provided with automatic non-return valves.

Where a conventional bilge system is used, the diameter of the bilge main is o be calculated according to the same formula used as per SOLAS for conventional ships, which is:

$$d = 25 + 1.68 \, [L (B+D)]^{0.5}$$

where:

d = internal diameter of the bilge main, in mm;

L = length of the craft , in m, of the rigid hull, excluded appendages, at the design waterline in displacement mode;

B = for monohull craft is the breadth of the craft, in m, of the rigid hull, excluded appendages, at the at the design waterline in displacement mode;

for multihull craft is the breadth of a single hull at the design waterline, in m, in displacement mode;

D = is the moulded depth of the craft to the "datum" in m.

The requested bilge pumps are to be of self-priming type. Each bilge pump is to be capable of pumping water through the required bilge main at a speed not less than 2 m/sec. For brevity sake such capacity may be calculate with the following formula:

$$Q1 = 0{,}00565 \ d^2$$

where:

Q1 = capacity of each pump in m3/h;

d = diameter of the bilge main calculated as per above.

4.2 - BILGE SYSTEMS WITH INDIVIDUAL BILGE PUMPS

For high speed craft where such system is adopted, this has to consist of:

- at least one fixed submersible pump for each space to be drained;
- at least one portable supplied by an emergency supply for use in individual spaces.

The total capacity Q2 of bilge pumps for each hull is not to be less than 2.4 times the capacity of the pump defined in previous item 4.1.

The capacity of each submersible pump q, in m3/h, is to be not less than:

$$q = Q2 \ / \ (N - 1)$$

where N is the number of submersible pumps. In any case the said capacity is to be not less than 8 m^3/h.

Examples of bilge diagrams may be found at figures 1 and 2.

5. EXHAUST GAS SYSTEM

The exhaust gas ducts may be led in the funnel or through the hull. Where the second possibility is adopted, due to the necessity of minimising the intake of exhaust gas into accommodation and service spaces via the air conditioning and ventilation system, the said ducts are led as low as possible in the vicinity of the waterline. In this case HSC Code requires the installation of erosion/corrosion resistant shut-off flaps or similar devices on the pipe ends in order to prevent the water flooding, via the concerned engines, of the compartment where the same engines are located.

However it is to be considered that the presence of flaps or similar devices on the exhaust ducts is strongly disapproved by the manufactures of the engines for the consequent possibility of back-pressure and, in addition, the insufficient reliability of such devices for the critical working condition is also to be considered.

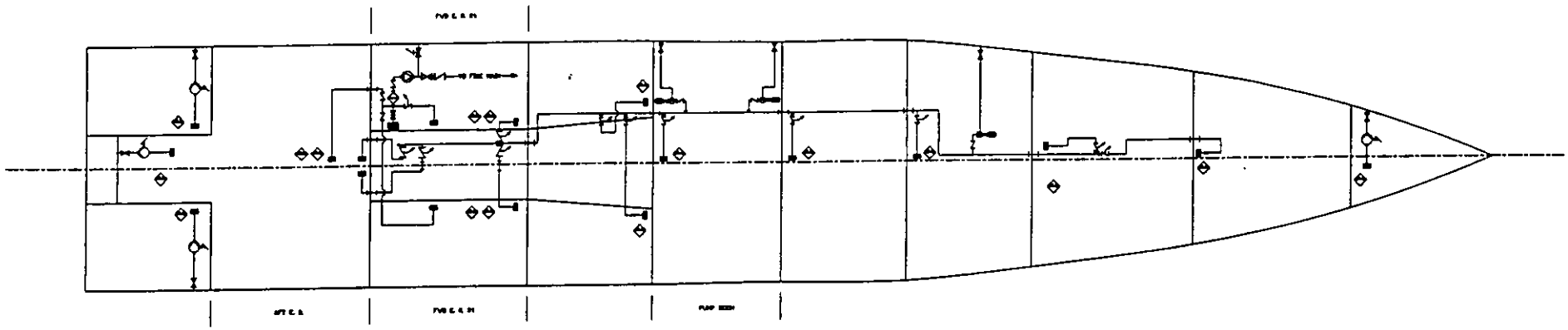
The Rules admit that the said devices may be omitted if the exhaust outlets through the craft side are arranged above the most unfavourable waterline in flooding condition. In this respect it is also to be considered that, where the exhaust outlets have a structural fabric for which, in respect of the tightness, they may be considered part of the hull, it is reasonable to accept that they are submerged in most unfavourable waterline in flooding condition provided that, in the highest point of the duct running inside the engine room, the lower edge of the ducts is above the said unfavourable waterline.

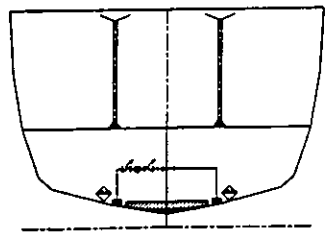
See figures 3 and 4.

CONCLUSION

For the other piping systems, like sanitary, ventilation and hydraulic (steering included) systems, and for the arrangement of sounding and venting pipes, the HSC Code has not special rules or rules different from the ones applicable to conventional ships in compliance with SOLAS, MARPOL and Load Lines Conventions.
Anyway it is to be remarked that the plastic pipes, except those fitted on open spaces or within tanks, whose use on board of high speed craft is controlled by the rules of IMO Resolution A. 753(18), are requested to be submitted for approval to fire tests including flame spread, smoke generation and toxicity tests. The same tests are not required by SOLAS for plastic pipes to be installed in conventional ships. Since the acceptance limits imposed for the said tests are very severe, there are not news of acceptance of plastic pipes for indoor arrangements on board high speed crafts.

SYMBOL	NAME
	REMOTE CONTROLLED VALVE
	BILGE/FIRE PUMP
	FOOT VALVE WITH STRAINER
	LEVEL ALARM
	HAND PUMP
	BILGE PUMP





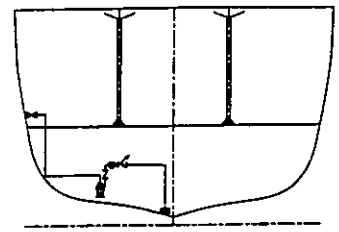


FIGURE 1

SCALE 1:200

SYMBOL	NAME
	HAND PUMP
	SUBMERIBLE BILGE PUMP
	FOOT VALVE WITH STAINER

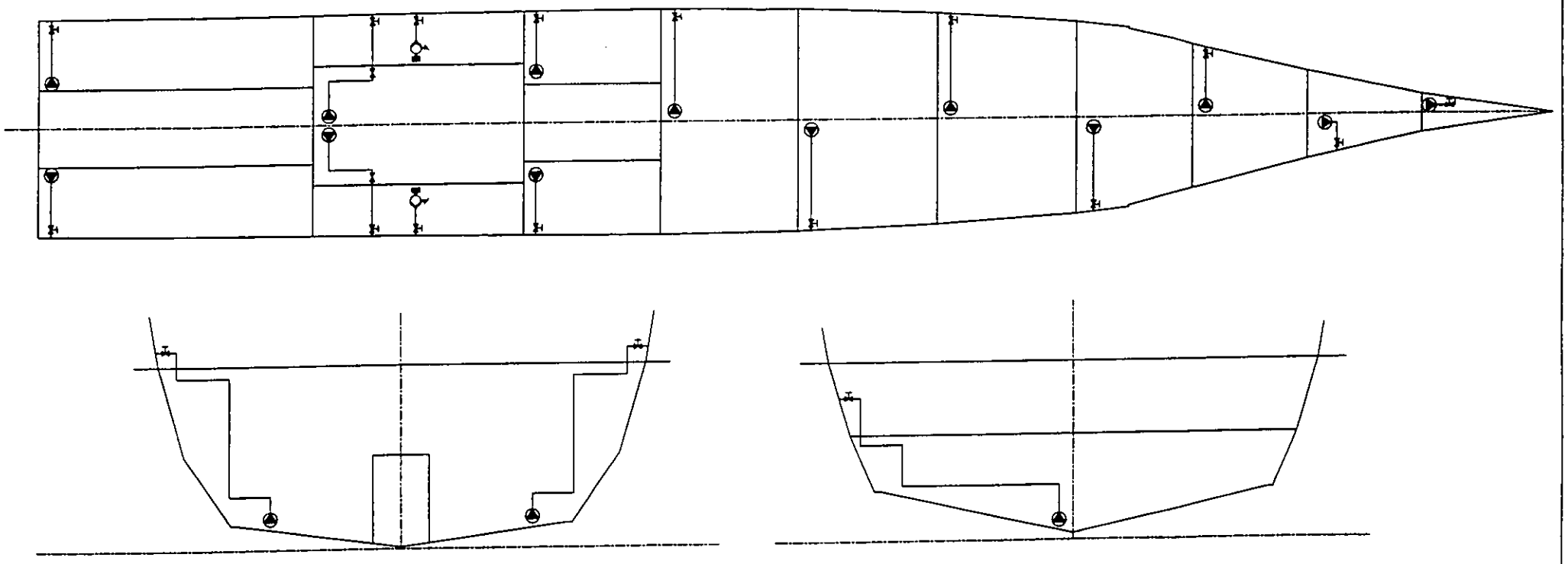


FIGURE 2

SCALE 1:250

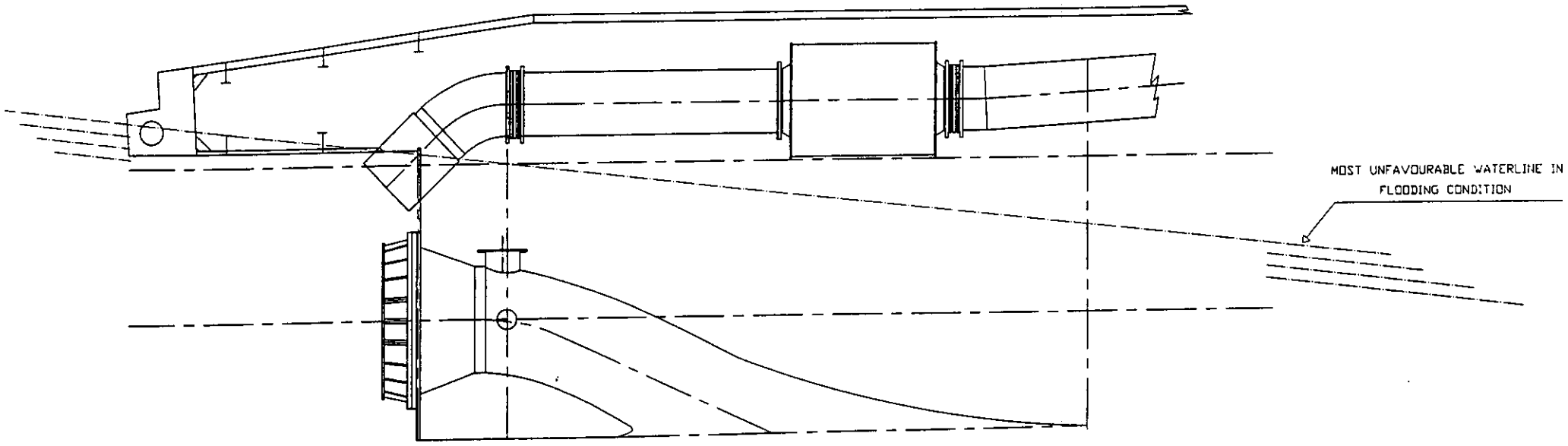


FIGURE 3

SCALE 1:50

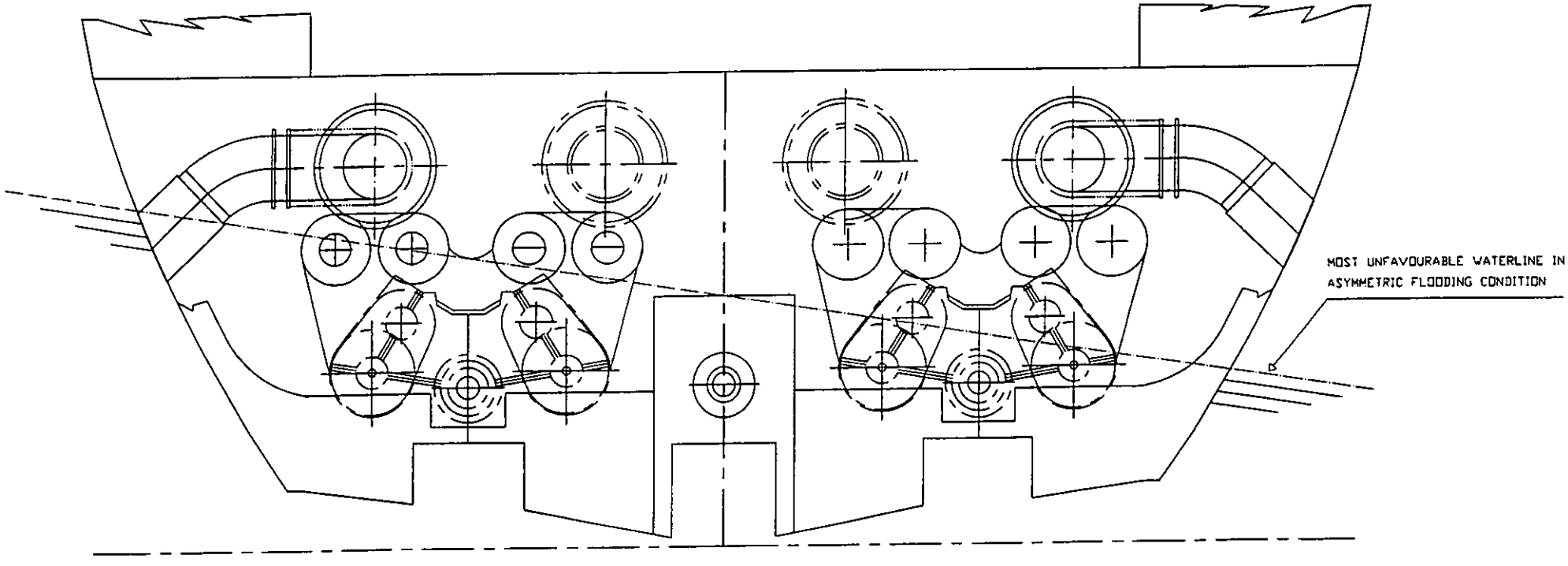


FIGURE 4

SCALE 1:50

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MATCHING SHIP, PROPULSOR and PRIME MOVER

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CONTENTS

Abstract

1. Introduction
1.1 Purpose
1.2 Definitions and assumptions

2. Ship's resistance and propeller load characteristics
2.1 Ship's resistance
2.2 Screw type propellers
2.3 Waterjets
2.4 Influence of operational conditions on the propeller load characteristic
2.5 Influence of number of engines and auxiliaries on the load characteristic

3. Prime mover drive characteristics
3.1 Diesel engines
3.2 Gasturbines

4. The matching problem
4.1 Requirements with regard to matching
4.2 A common way of matching

5. Off design conditions
5.1 The off design problem
5.2 Change of resistance/thrust load characteristics
5.3 Change of drive characteristics

ABSTRACT

The paper dicusses the matching of driving machinery with ship's propulsor.
An overview is given of the drive and load characteristics of diesel engines, gasturbines, screw type propellers and waterjets.
Further the different aspects of matching both in design condition and off-design situations are covered.

1. INTRODUCTION

1.1 PURPOSE

With the design of a ship's propulsion system, the correct matching of the prime mover (diesel engine, gasturbine) to the propulsor and ship, is of great importance. In case the matching problem is not solved adequately, the ship may have poor characteristics with regard to:
- attainable speed
- acceleration and stopping
- fuel consumption

The matching task should not be performed with the viewpoint of the engine or the propulsor only, but should be handled as an integrated problem of both.
In this lecture the subject will be discussed for screw type propellers, with fixed or controllable pitch blades and waterjets in combination with diesel engines or gasturbines as prime movers.
If a ship and its machinery plant have only one operational mode and the environmental conditions do not vary much, the matching is relatively easy. For a ship with a variety of differing operational modes (single engine-shaft versus multiple engine or multiple shaft operation; drive of propeller plus or without auxiliary machinery) more attention is needed for the matching.

1.2 DEFINITIONS AND ASSUMPTIONS

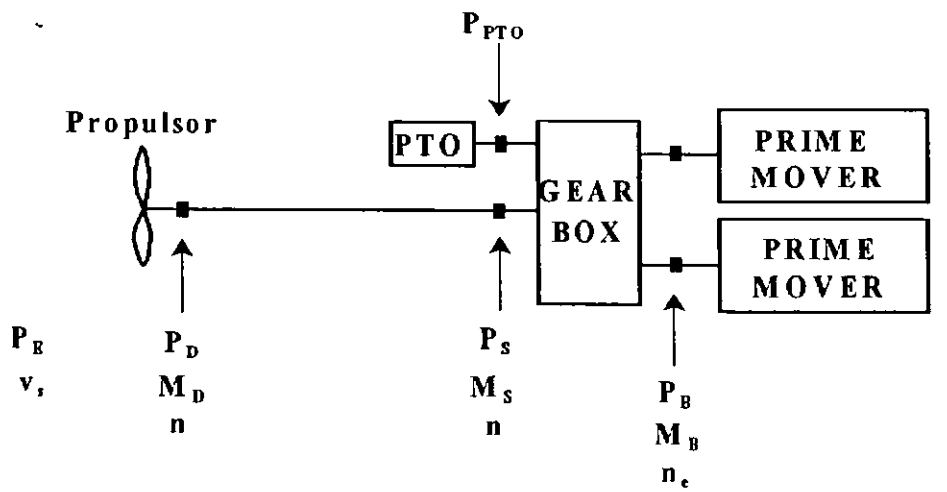


Fig. 1 - Definition powers torque and speed

Figure 1 gives an overview of the variables, with regard to power, torque and speed frequently used in the following paragraphs. It is assumed, that the theoretical basis of the given variables is known.

These variables are:

P_E = effective power [W], consisting of the total ship's resistance R [N] at a ship's speed v_s [ms^{-1}]

$P_E = R\, v_s$

P_D = delivered power [W], to the propulsor by the machinery plant. The relation between P_E and P_D is given by the propulsive coefficient (propulsion efficiency) η_D:

$P_E = \eta_D\, P_D$

The propulsive coefficient η_D consists of a number of variables as follows:

$$\eta_D = \frac{P_E}{P_D} = \frac{R\, v_s}{M_D\, \omega} = \frac{R\, v_s}{2\pi M_D\, n}$$

$$\eta_D = \frac{R\, v_s}{T\, v_A} \cdot \frac{T\, v_A}{Q\, \omega} \cdot \frac{Q}{M_D}$$

$$\eta_D = \frac{1-t}{1-w} \cdot \eta_o \cdot \eta_R$$

$$\eta_D = \eta_H \cdot \eta_o \cdot \eta_R$$

where :

ω = angular speed of propulsor shaft [s^{-1}] = $2\pi n$

n = number of revolutions per second of propulsor shaft [s^{-1}]

v_A = speed of advance [ms^{-1}], which is related to ship's speed v_s by the wake fraction w : $v_A = (1 - w)\, v_s$

w = wake fraction [-]

T = thrust [N] in open water condition, which is related to ship's resistance by the thrust deduction factor t : $R = (1 - t)\, T$

t = thrust deduction factor [-]

η_o = propulsor open water efficiency [-] = $\dfrac{T\, v_A}{Q\, \omega}$

Q = propulsor torque [Nm] in open water condition

M_D = delivered torque [Nm] to the propeller by the machinery plant, which is related to the open water propeller torque Q by : $Q = \eta_R\, M_D$

η_R = relative rotative efficiency [-] = $\dfrac{Q}{M_D}$

η_H = hull efficiency [-]= $\dfrac{1-t}{1-w}$

Further we have as variables:

P_s = shaft power [W], the power at the output flange of the gearing system, input to the shaftline. The relation between P_D and P_s is given by the shaft efficiency η_{shaft} : $P_D = \eta_{shaft} P_s$

η_{shaft} = shaft efficiency [-]. The efficiency of the thrust block is included, in case this is installed in the shaftline

P_B = brake power of the prime mover [W]

n_e = engine (prime mover) revolutionary speed [s^{-1}]

i = gear ratio [-]; $i = n_e/n$

P_{PTO} = power absorbed by auxiliary machinery [W] (such as a shaft generator) driven through a power take off on the gearing system or directly by the engine (in case of direct drive without a gearing).
The relation between P_B, P_s and P_{PTO} is given by :
$P_s = \eta_{gear} \Sigma P_B - \Sigma P_{PTO}$

η_{gear} = gear efficiency [-], which might include thrust block and coupling efficiencies

We will distinguish between load and drive characteristics:
The propulsor load characteristic gives the relation between torque or power, as required by the propulsor, and propulsor speed n.
The prime mover drive characteristic gives the relation between torque or power, which can be supplied by the engine at a certain fuel setting, and the engine speed n_e.

Frequently the prime mover drive chracteristic will be "translated" from engine brake to propeller delivered (by means of η_{gear}, η_{shaft} and i).
In order to distinguish between these two different characteristics use will be made of:
$M_{D,L}$ and $P_{D,L}$ for the propeller load characteristic.
M_D and P_D for the engine drive characteristic.

2. SHIP'S RESISTANCE AND PROPULSOR LOAD CHARACTERISTICS

2.1 SHIP'S RESISTANCE

The load characteristic of a ship in terms of resistance and ship's speed is often given as a square relation:

$R = \alpha\, v_s^2$ (see figure 2)

This only will be true for a relatively low speed with a low Froude number (F_n = 0.1 - 0.2). For higher ship speeds the resistance curve will have have higher powers than 2 and make take shapes as shown in figure 2.

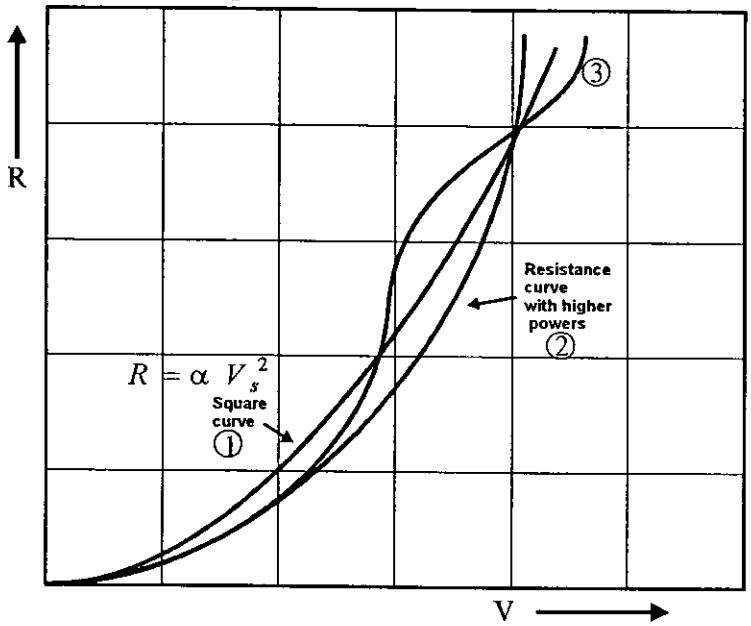


Fig. 2 - Different types of ship's resistance - speed curves. A square resistance curve (1) is a reasonable approximation for ships with low Froude numbers (F_n = 0.1 - 0.2). For higher speeds displacement ships curves of type (2) may be encountered. Planing craft, swath ships may have curves like (3).

2.2 SCREW TYPE PROPELLERS

2.2.1 OPEN WATER DIAGRAM

The propeller open water diagram gives in a dimensionless form the torque and thrust as a function of speed. Figure 3 gives such a diagram for a fixed pitch propeller. The dimensionless coefficients are:

K_T = thrust coefficient = $\dfrac{T}{\rho n^2 D^4}$

K_Q = torque coefficient = $\dfrac{Q}{\rho n^2 D^5}$

J = advance coefficient = $\dfrac{v_A}{nD}$.

ρ = density of water [kg m^{-3}]

D = propeller diameter [m]

This diagram is valid for a propeller in open water conditions, without the influence of the ship's hull.
The open water propeller efficiency can be given as follows:

$\eta_o = \frac{T v_A}{Q\omega} = \frac{K_T J}{K_Q 2\pi}$; which is also shown in the diagram.

The open water diagram, after having designed or selected the propeller, can be used to "translate" the ship's resistance characteristic to a propeller load characteristic. Therefore, in first instance, a square resistance curve is assumed, which can be converted as follows:

$R = \alpha v_s^2$

$T(1-t) = \alpha\left[\frac{v_A}{1-w}\right]^2$

$T = \frac{\alpha}{(1-t)(1-w)^2} v_A^2$

$K_T \rho n^2 D^4 = \frac{\alpha}{(1-t)(1-w)^2} v_A^2$

$K_T = \frac{\alpha}{(1-t)(1-w)^2}\left[\frac{v_A}{nD}\right]^2$

D = constant and when it is assumed that t and w may be treated as constants as well, this leads to:

$K_T = \text{constant} \cdot J^2 = \gamma \cdot J^2$

In practice there is allways some variation of w and t with varying ship's speed. This leads also to some variation of the constant γ as function of ship's speed.
We have now a square relation between K_T and J representing the ship's resistance. When this relation is drawn in the open water diagram (see figure 3), the intersection with the propeller K_T-curve gives the operational point of the propeller J. Now we can determine the torque coefficient and open water efficiency.

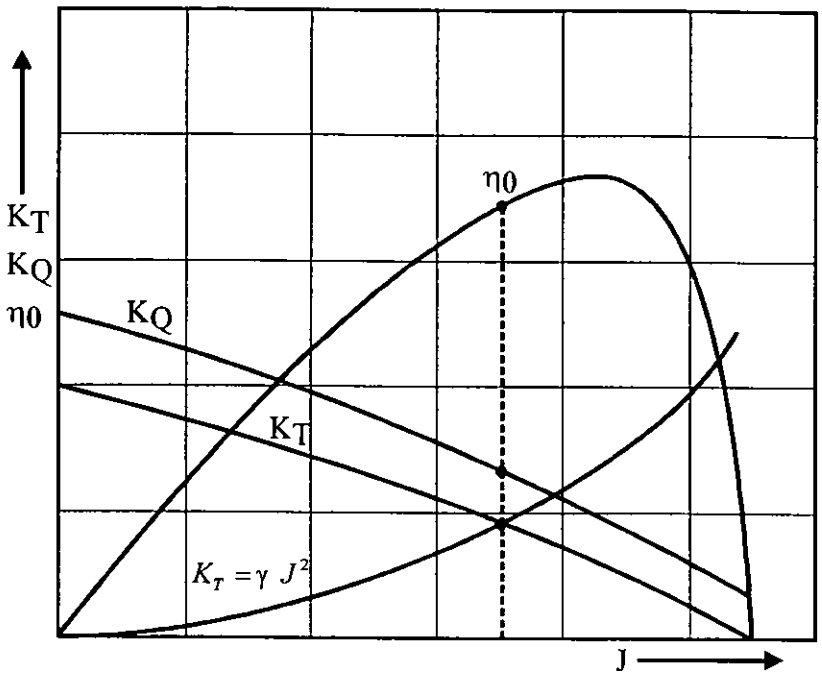


Fig. 3- Open water diagram for a fixed pitch propeller, with ship's resistance characteristic. Intersection of ship characteristic with K_T gives the propeller operational point.

It will be clear, that for a square resistance curve and constant w and t, the propeller will have a single point of operation, independent of ship's speed; this means:

- J, K_T and K_Q remain constant,

- Q and $M_{D,L}$ will be square functions of propeller speed n ($Q = K_Q \rho n^2 D^5$)

- $P_{D,L}$ will be a third power of speed n

- v_A and n have a linear relation, because J = constant.

When the resistance curve is not a square one, these relations do not exist. In the case of a non-quadratic resistance characteristic (see figure 4), the operational points of the propeller can be determined for each speed v_s by assuming, a relation $R = a\, v_s^2$.
For another speed one can assume $R = b\, v_s^2$.

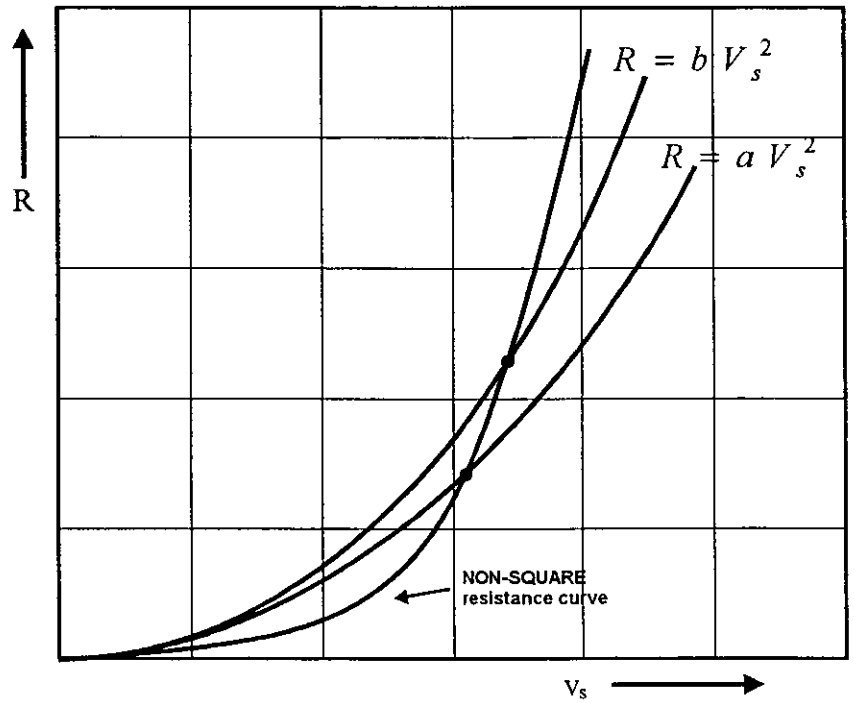


Fig. 4 - Non square resistance curve with virtual square resistance curves to determine the operational points of the propeller.

With these relations the relation between K_T and J can be determined :

for R = $a\ v_s^2 \rightarrow K_T = x\ J^2$
for R = $b\ v_s^2 \rightarrow K_T = y\ J^2$

Again the operational points of the propeller can be determined giving J_x and J_y (see figure 5).

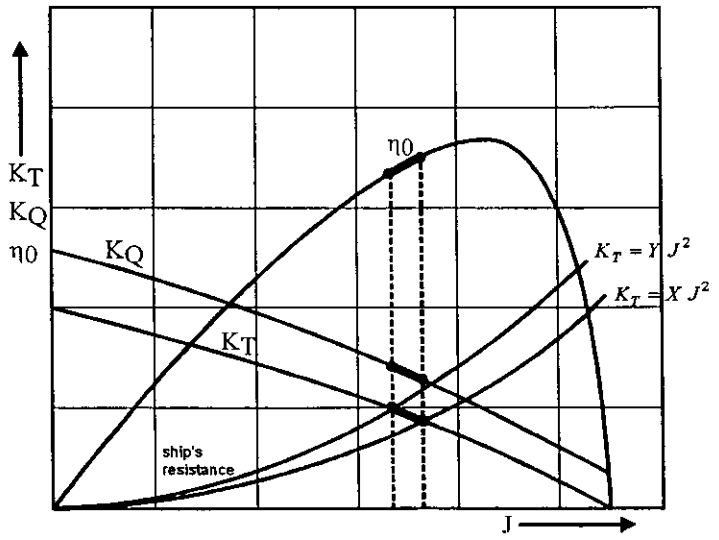


Fig. 5 - A non square resistance curve results in a range of operational points of the propeller.

From this it follows, that in the general case of a non-square resistance curve, the propeller will have a range of operational points. Consequently the propeller power load curve is no longer a cubic curve and there is not anymore a linear relation between v_s and n.

Figure 6 gives the open water characteristics for a controllable pitch propeller (cpp). The K_T and η_o curves are now given for different pitch ratio's H/D.
Again the ship's resistance can be converted to a K_T, J relation and from this a number of operational points for the propeller, one for each pitch setting can be determined.

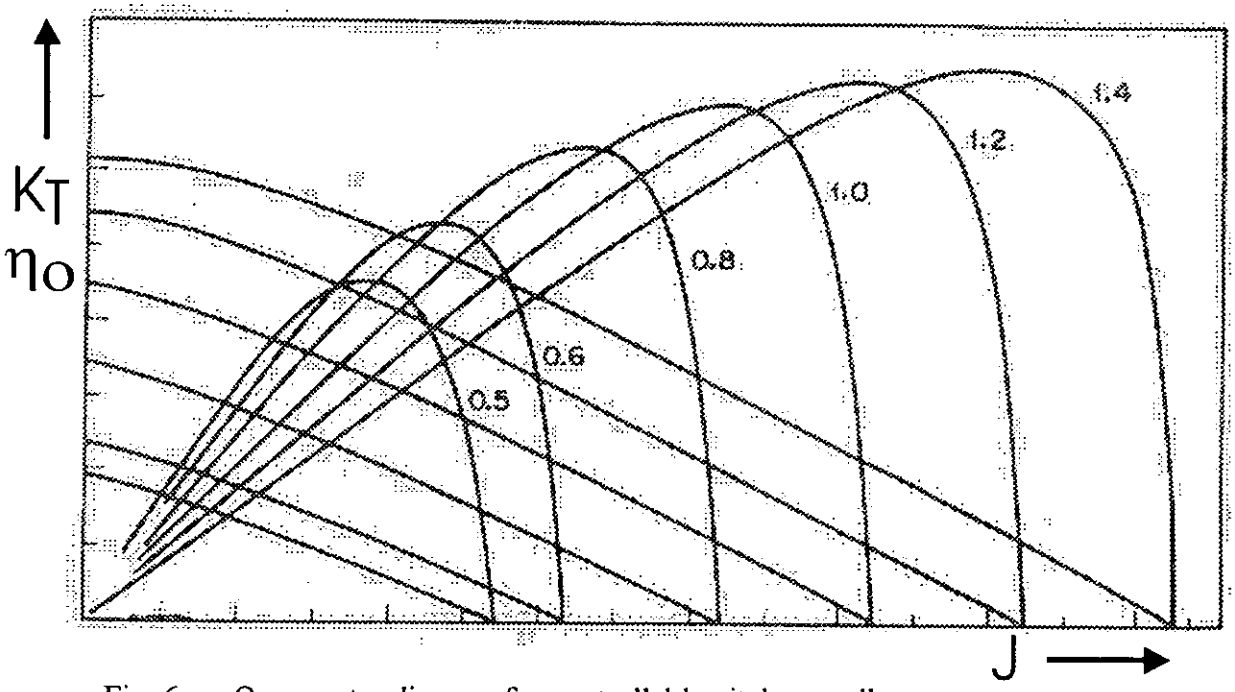


Fig. 6 - Open water diagram for controllable pitch propeller.
Lines for K_T and η_o are given for different pitch ratio's H/D = 0.5 - 1.4.
K_Q can be calculated from K_T and η_o.

2.2.2 TORQUE- OR POWER-SPEED LOAD CHARACTERISTIC OF PROPELLER

After having determined the operational points of the propeller in the open water diagram, the propeller load characteristic can be easily found, by use of K_Q.

For the controllable pitch propeller of figure 6 and a square resistance curve, the propeller load characteristics will have the form as given in figure 7.

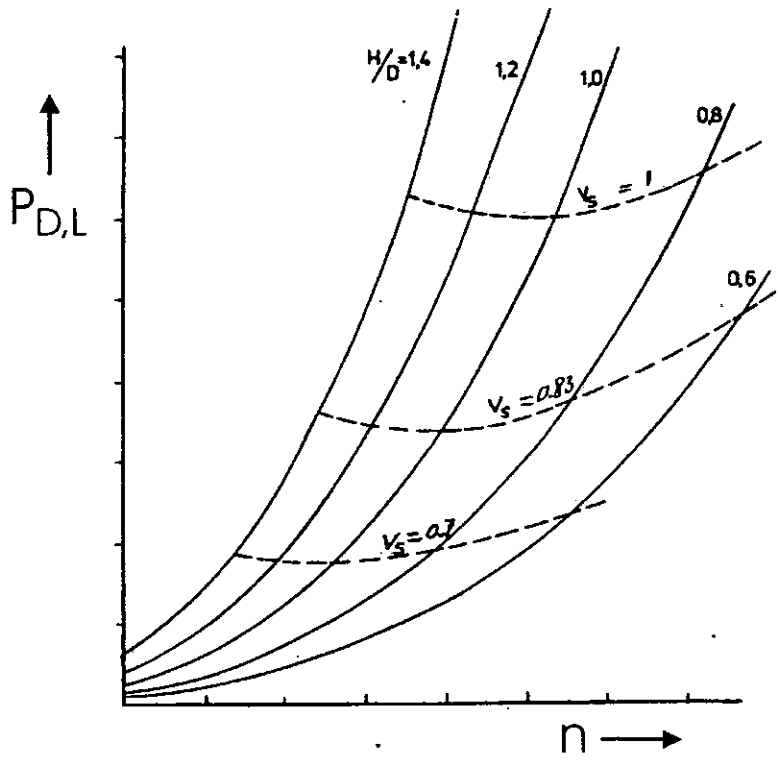


Fig. 7 - Propeller load curves as function of propeller speed for different pitch settings.

For each pitch ratio H/D a relation between $P_{D,L}$ and n follows. In the diagram also lines for constant ship's speed can be drawn. From these, it will be clear, which pitch setting is optimal for a certain speed, with regard to minimum power.
In diagram 7 the maximum efficiency is obtained for each speed at the same pitch ratio. If a non-square resistance curve would have been used, the maximum efficiency will vary with speed to different pitch settings.

Note:
When one is interested in minimum fuel consumption the optimal pitch setting might be slighty different than the pitch at which the $P_{D,L}$ is minimal. In that case one should also consider the specific fuel consumption of the driving engine. Multiplication of specific fuel consumption with $P_{D,L}$ results in fuel consumption. The minimum of the fuel consumption can be shifted somewhat compared to minimum $P_{D,L}$.

2.3 WATERJETS

In general the manufacturers of waterjet propulsors do not supply open water diagrams as for screw type propellers, discussed in paragraph 2.2.
For each application of a waterjet in combination with a certain hull and location of the jet

relative to the hull, the waterjet manufacturer may supply diagrams as shown in figure 8

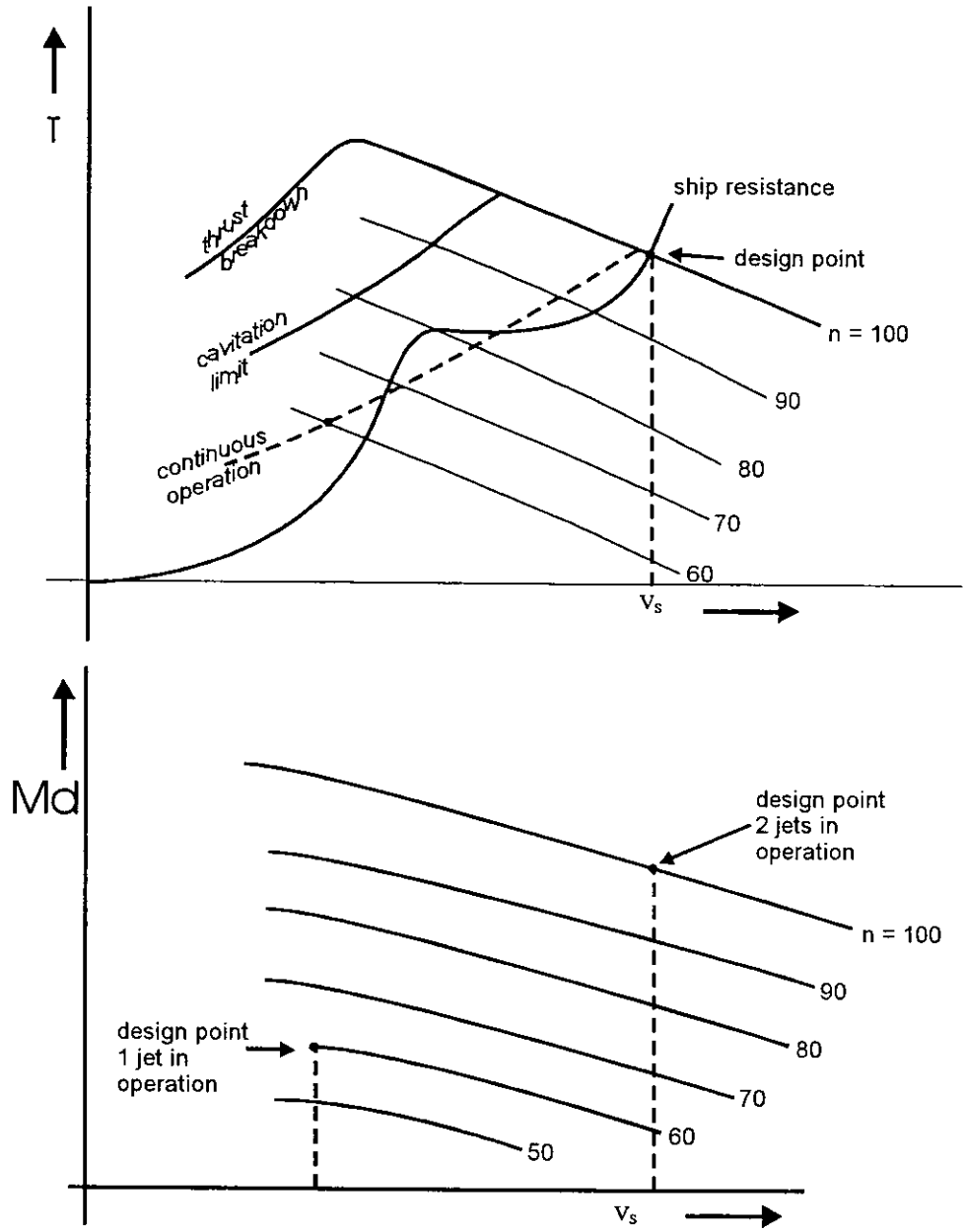


Fig. 8 -Waterjet thrust and torque characteristics as function of ship's speed and propulsor rpm.

In these diagrams are shown the lines of effective thrust and torque (M_D) to the waterjet as function of ship's speed v_s and with propulsor number of revolutions (1/s or rpm).
With effective thrust is meant the thrust which can be compared with the ship's resistance R.
In these diagrams also limits can be shown for continuous operation, a cavitation limit and a thrust break down limit.
Continuous operation is only allowed to the right of the continuous operation limit.

The zone between the line for continuous operation and the cavitation limit may be used intermittently for acceleration of the ship.
In the thrust - speed diagram also the resistance characteristic of the ship has been drawn, together with the design point of the waterjet.
In principle the curves of figure 8 can be converted to dimensionless diagrams in the form of K_T, K_Q, J-diagrams.
K_T and K_Q , in analogy with screw type propellers, are defined as:

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$K_Q = \frac{Q}{\rho n^2 D^5}$$

Where D = impeller diameter of the waterjet.
J is defined as a function of ship's speed

$$J = \frac{V_s}{nD}$$

This leads to the figure 9.

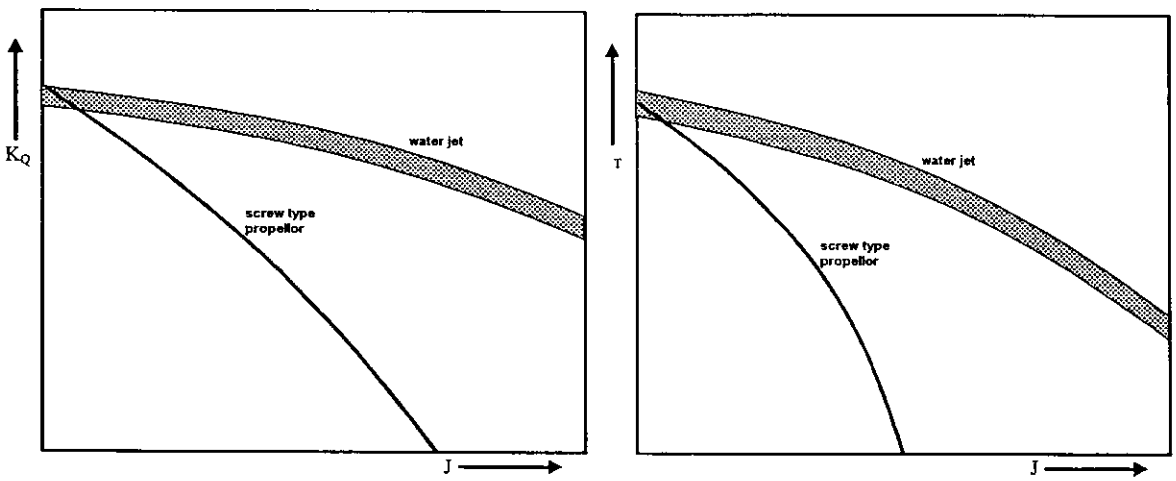


Fig. 9 -K_T, K_Q - diagrams for a waterjet propulsor.

Note that due to the influence of wake the K_T and K_Q curves are not single lines but form a band.
In comparison with screw type propellers the K_T and K_Q for a waterjet are much more flat. This means that the influence of ship's speed on thrust and torque is much less for a waterjet than for a screw type propeller. This is of course a result of the fact that entrance velocity v_A into the waterjet is, due to the location of the waterjet relative to the hull, hardly affected by ship's speed but is much more determined by the impeller speed.
The diagrams of figure 8 contain the same information as the open water diagrams, as for screw type propellers, and can be used efficiently for the matching problem.

2.4 INFLUENCE OF OPERATIONAL CONDITIONS ON THE PROPELLER LOAD CHARACTERISTIC

The propulsor load curve will change due to changes of operational conditions, as:
- ship's displacement
- hull and propulsor fouling
- sea state
- number of propulsors in operation, in case of a multiple propulsor ship.

2.4.1 SHIP'S DISPLACEMENT

The draft/displacement, and also the trim will have influence on the resistance. As a first indication one may assume, that resistance is linearly dependent on displacement to the power 0.667 :

$$\frac{R}{R_o} = \left[\frac{\Delta}{\Delta_o}\right]^{0.667}$$

Consequently increase of draft will result in increase of the propeller load curve.

2.4.2 HULL AND PROPULSOR FOULING

Fouling has an influence on the friction part of ship's resistance and on the efficiency of the propeller. This influence is nowadays less important, than in the past, due to the application of modern anti-fouling paints.

2.4.3 SEA STATE

Resistance is dependent on sea state. Compared to the situation in still water the resistance can be a factor 2-3 higher in a sea state with high waves in heavy wheather (BF 9-10). For frequently encountered sea conditions, with moderate wave heights, a resistance increase of 25 % is not unrealistic.
For proper matching of prime mover and propulsor it is essential to have reliable information on this aspect.
This can be obtained for instance from actual experience with existing ships on the same route. In that case the information will become available as an increase of required delivered power. Further it is possible to predict the resistance increase by sea keeping calculations or by modeltesting in waves.

2.4.4 NUMBER OF PROPULSORS IN OPERATION

A propulsion system, having 2 or more propulsors, normally will be designed with all propulsors in operation. The ship should however, have the possibility to operate on less propulsors.
A non-operating screw type propeller will generate an additional resistance, instead of giving thrust.
This additional resistance will be fairly high for a fixed pitch propeller (fpp), which is kept stationary by means of a shaft brake.
In case the non-working propeller is free to rotate (trailing) the additional resistance will be lower, but can still be considerable. For instance for a twin shaft ship, a resistance increase,

due to one trailing propeller, was determined to be in the order of 20-25 % of the normal ship's resistance.
The resistance increase due to a non-working propeller can be minimized by adoption of a controllable pitch type with the possibility, to put the blades in feathering position (blade sections in the sailing position).
As a consequence the propellers, remaining in operation, have to generate a much higher thrust for the same ship's speed. For instance: for one propeller in operation on a twin shaft ship, the thrust increase at a constant ship's speed is a factor 2 (because only one of two propellers is operating) plus the additional resistance of the non-operating propeller. For the operating propeller this is virtually a great increase of resistance.

2.5 INFLUENCE OF NUMBER OF ENGINES AND AUXILIARIES ON THE LOAD CHARACTERISTIC

Frequently a propulsion system consists of a number of prime movers driving a propulsor and auxiliary machinery driven by these engines.
Even in case there are no changes of ship's resistance, such an installation nevertheless will result in varying load curves, experienced by the prime mover(s).

Assume for example a propulsion system, consisting of two identical engines driving a fixed pitch propeller, through a reduction gearbox, which is designed and matched for twin engine operation. In case now one engine is switched off, the remaining engine has to supply for the same shaft speed twice of the originally supplied power.

The drive of a shaft generator means, that the load experienced by the prime movers consists of the propulsor load plus the load required by the generator.

3. PRIME MOVER DRIVE CHARACTERISTICS

The drive characteristic gives the brake power or torque of the prime mover as function of the rotational speed at a specific fuel setting.
In this paragraph the drive characteristics of marine diesel engines and marine aero-derived gasturbines will be discussed.

3.1 DIESEL ENGINES

3.1.1 TORQUE AND POWER AS FUNCTION OF SPEED

With a constant fuel rack position (this means a constant fuel mass injected in the cylinder per engine cycle) it may be assumed, as a first approximation for a diesel engine, that torque is constant with varying speed. This means that in first instance P_B is proportional to engine speed and mass of fuel injected per cycle.
In reality efficiency is not constant with varying engine speed and also torque will vary somewhat (figure 10).

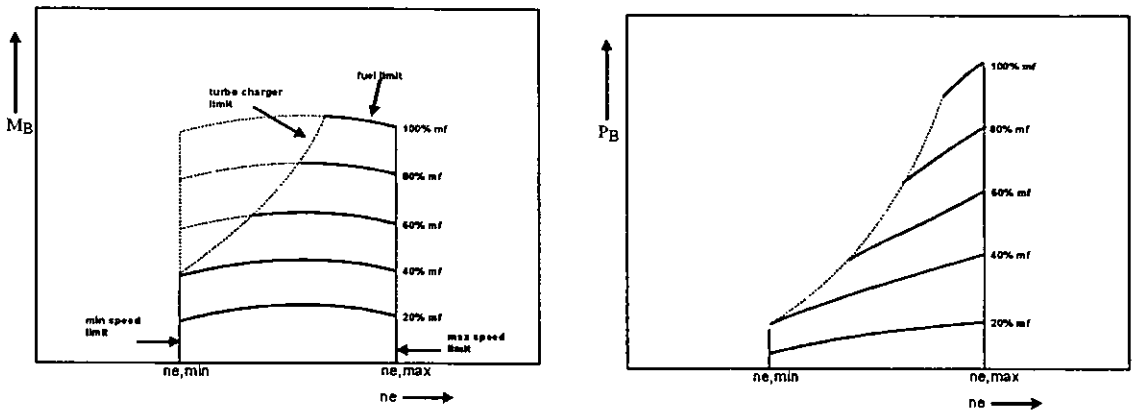


Fig. 10 - Torque and power - speed relations for constant fuel rack position for a turbo-charged diesel engine.

Marine diesel engines are nowadays allways highly turbocharged. This has the effect, that with a decrease of speed and fuel rack, the air supplied by the turbocharger to the diesel engine decreases. Furthermore with decreasing speed and constant load, the operational point of the turbocharger compressor will move to the surge line.
This results in the characteristic, that high torques cannot be maintained at reduced engine speed.

3.1.2 OPERATIONAL ENVELOPE

The operational envelope for a marine diesel engine will normally have the appearance of figure 11.
Here are shown:
MCR = maximum continuous rating [W], at the corresponding maximum speed.
An area where continuous operation is allowed, limited by:
- maximum speed
- fuel limit
- turbocharger limit

- minimum engine speed
- minimum engine torque, in order to prevent engine fouling

Further area's are shown for temporary operation:

- overload area; normally only allowed on the testbed
- area to be used for acceleration of the ship
- manoeuvring area (for low speed operation)

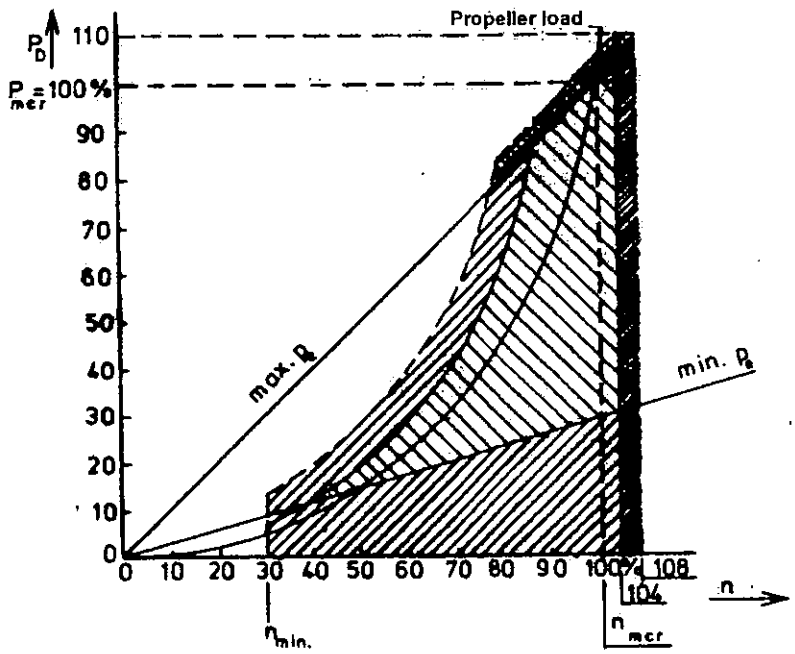


Fig. 11 -Typical operational envelope for a marine (turbo-charged) diesel engine.

- Normal continuous operation
- Temporary operation for acceleration or manoeuvring
- Overload area
- Only during trials

This diagram has for every diesel engine manufacturer its own specific form. The size of the different area's is much depending on the lay-out of the turbocharging system.

Diesel engines for high speed craft presently have very high break means effective pressures in the order of 30 bar. This can be achieved through twin stage turbocharging, as shown in the lecures on diesel engines. The disadvantages of such highly turbocharged engines is that the operational envelope becomes very narrow. To overcomes this problem the diesel engine manufacturers started approximately in 1985 to supply high speed engines with sequential turbocharging. This means that in case the engine is equipped with a number of turbocharger systems, arranged in parallel, the turbo-chargers are switched on and off on basis of the required engine load and speed.
Figure 12 shows the benefits of sequential turbocharging.

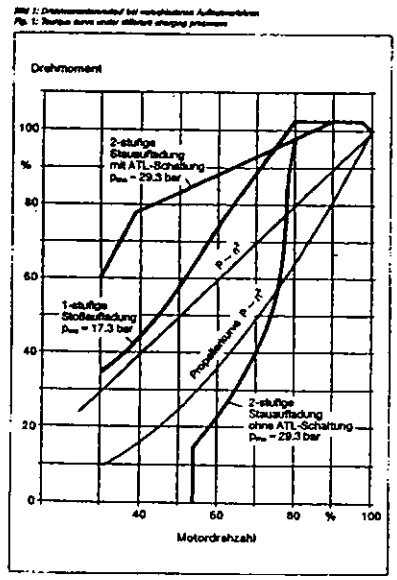


Fig. 12 - Different possibilities to improve the torque curve of turbo-charged diesel engines.

In this figure are 3 situations shown:

- S a diesel engine with two-stage turbocharging without sequential operation (bmep = 29.3 bar).
  This results in a very narrow operational envelope, which is not attractive at all for propulsion of high speed ships.
- S a diesel engine with single stage turbocharging and again no sequential turbocharger operation (bmep = 17.3 bar). This results in a somewhat wider operational envelope, still not very attractive for high speed ships, but furthermore a relatively heavy engine, not attractive at all for high speed ships.
- S a diesel engine with two-stage turbocharging in combination with sequential operation. This leads to a relatively light weight engine and a wide operational envelope, which are attractive characteristics for a high speed ship.

Figure 13 shows the operational envelope for a modern high speed, twin stage, sequential operated turbocharged diesel engine.

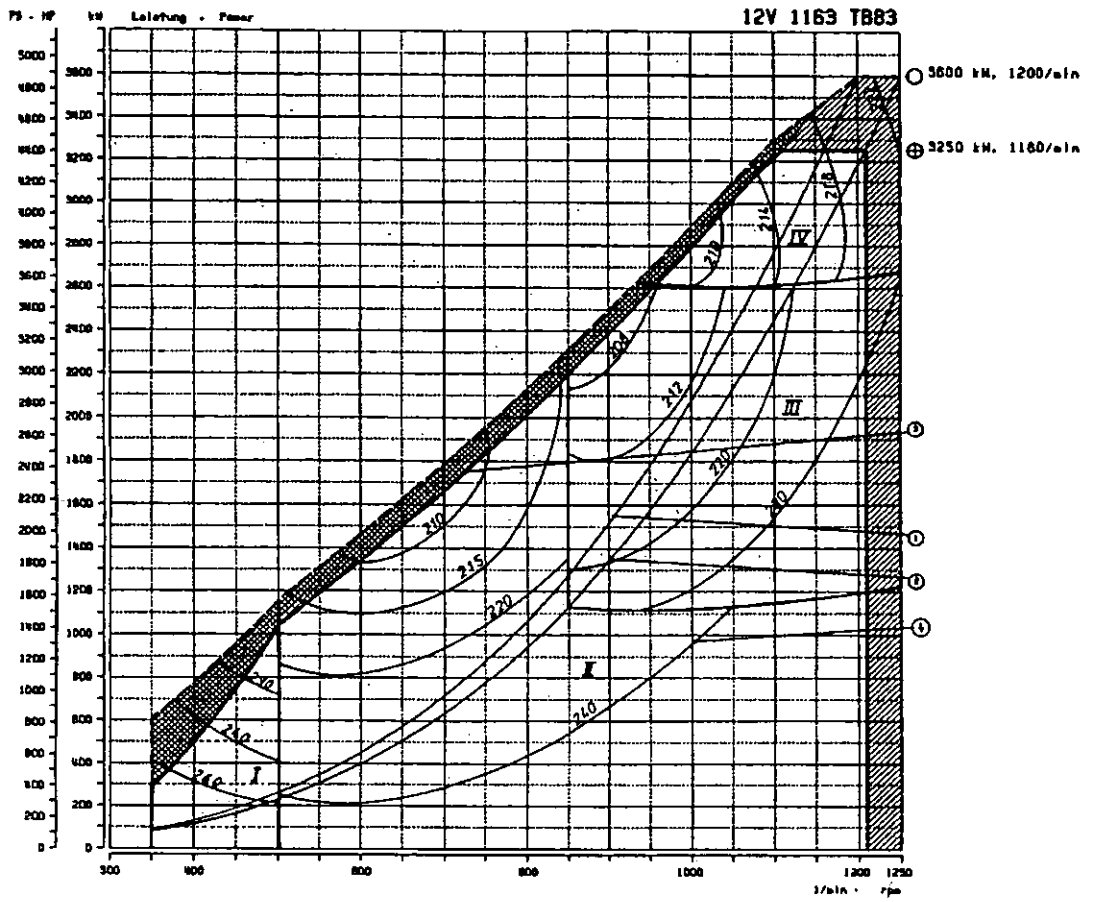


Fig. 13 -Performance diagram of a sequentially turbo-charged diesel engine.
The engine has 5 two-stage turbo-chargers. The different area's correspond with the use of 5, 4, 3 and 2 groups of turbo-chargers.

This engine is equipped with 5 pairs of sequentially operated twin stage turbochargers with LP-compressor, HP-compressor and air inter cooling.
The engine operates on 2, 3, 4, or 5 turbocharger pairs.

3.2 GASTURBINES

Gasturbines are frequently used for high speed ship application, where requirements such as: a high power/weight and /volume ratio and a low (stucture borne) noise signature are important. For these applications the marine aero derived gasturbines are an interesting choice.
The aero derived gasturbine normally is a so called simple cycle engine and consists of a single or twin spool gasgenerator plus a free power turbine. (figure 14).

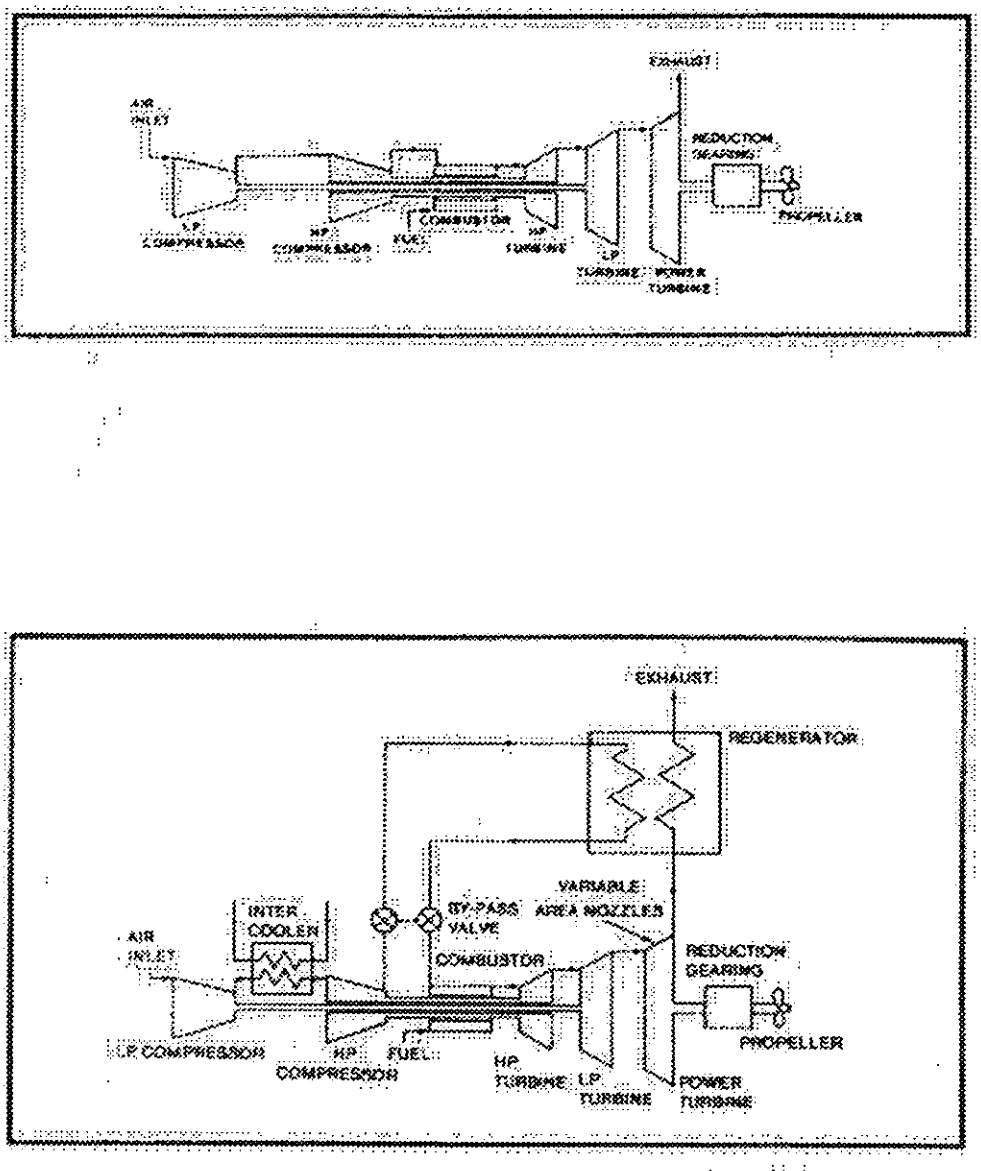


Fig. 14 - Diagrammatic arrangement of marine aero derived gas turbines.
Top: Simple cycle arrangement.
Bottom: Intercooled regenerative cycle.

This type of gasturbine has a different power speed characteristic than a diesel engine. With a constant fuel flow, the turbine has in first instance a constant power characteristic and consequently an increasing torque with decreasing speed. The characteristic is thus very flexible with high torques also at low or zero speed. Figure 15 shows the power-speed curves, the specific fuel consumption for (third power) propeller law operation is given in figure 16.

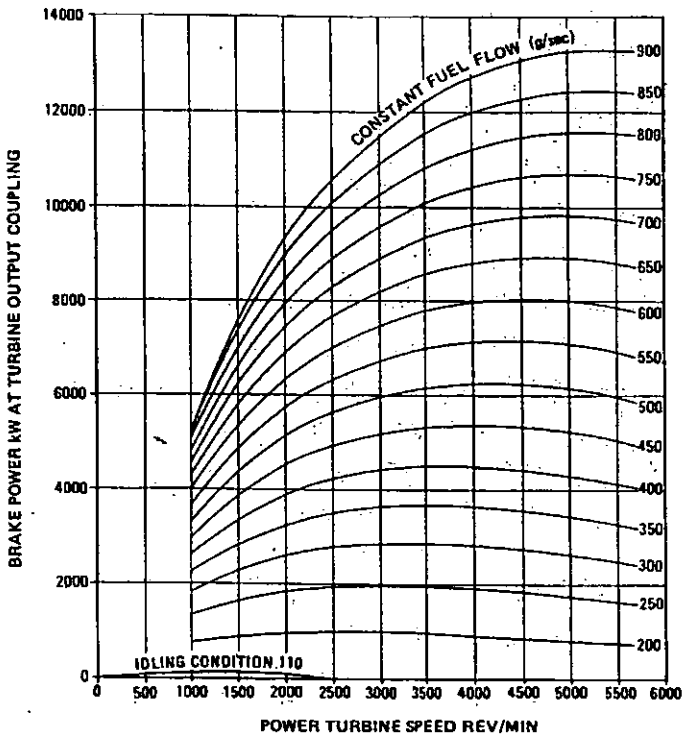


Fig. 15 - Typical power - speed curves for a marine gas turbine with lines of constant fuel flow.

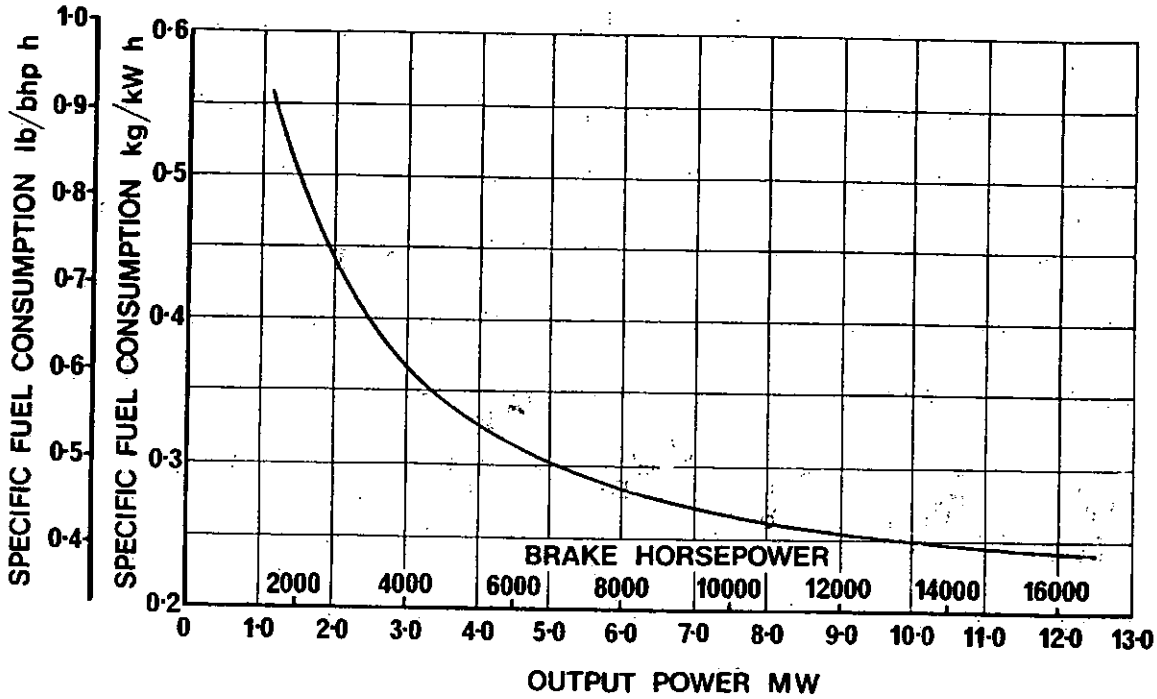


Fig. 16 - Typical specific fuel consumption curve for marine gas turbine (with a third power propeller load).

4. THE MATCHING PROBLEM

4.1 REQUIREMENTS WITH REGARD TO MATCHING

For the case of a ship with a fixed pitch screw-type propeller, driven by a diesel engine, the matching problem will be described.
The load characteristic, as experienced by the diesel engine, in terms of a power-speed relation is fully determined by the ship's resistance and the propeller characteristics. For the diesel engine a constant torque drive characteristic is assumed.
The best utilization of diesel power is obtained, when load and drive curves have their intersection at maximum engine torque and speed: figure 17, situation A.
When the ship's resistance increases, e.g. due to sea state or displacement, the load curve will shift to situation B. When the resistance decreases, a shift of the load curve to C will occur. In both these cases, it will be impossible to utilize the maximum available engine power: with curve B limited by maximum torque and with C limited by maximum speed.

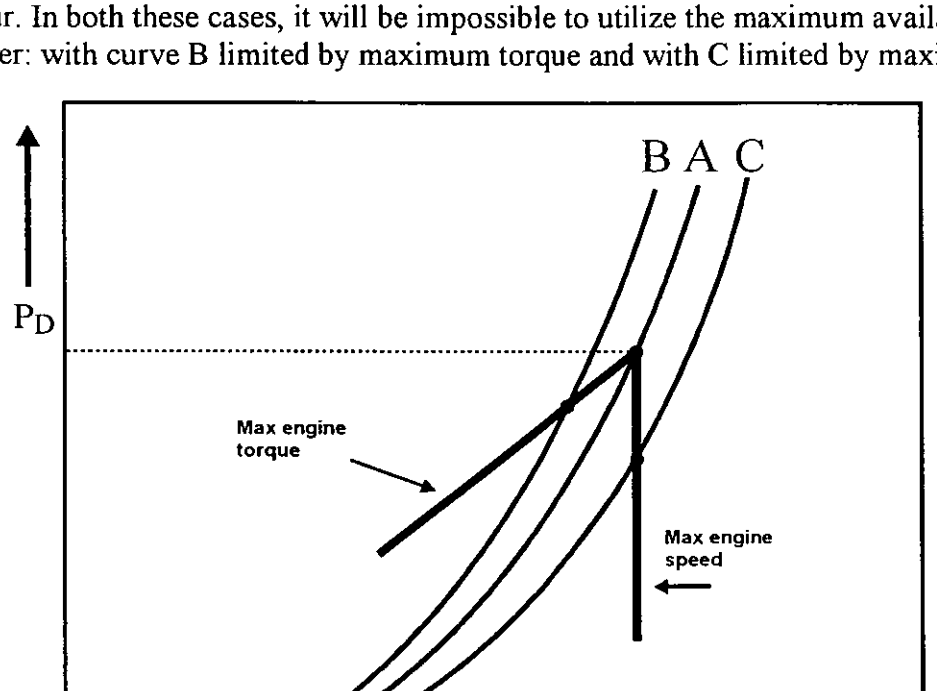


Fig. 17 - Matching of a diesel engine with a constant torque characteristic with a fixed pitch propeller.
Propeller load curve A gives the best utilization of available diesel power.
Load curve B is limited by maximum engine torque and curve C is limited by maximum engine speed.

With a controllable pitch propeller there is another degree of freedom: pitch setting, which enables to shift curves B and C back to the original one A (by means of pitch decrease/increase respectively). This however does not mean, that matching a cpp is an

easier task, because incorrect matching may lead, for instance, to a bad fuel economy. With a gasturbine the matching problem also seems easier, because available power is less dependent on shaft speed. Also here, this is not true, because economy still requires optimal matching.

From the foregoing, the requirements for a correct match between ship propulsor and prime mover can be summarized, as follows:

- the engine should be able to develop its maximum, or almost its maximum, power in the design condition i.e. the condition, which the ship will encounter most frequently.
- in off design conditions the installation should be able to comply with the operational requirements, such as speed.
- the choice and matching of engine and propulsor should be such, that the total of investments and operational costs (fuel, maintenance, etc.) are minimised on life cycle basis.

As an example, in order to achieve a low fuel consumption, one should not select the most optimal propeller (with the lowest $P_{D,L}$) or the most optimal lay out point of the diesel engine (with the lowest possible sfc), but a combination, which has in total the best efficiency in terms of fuel consumption per nautical mile.

4.2 A COMMON WAY OF MATCHING

For a propulsion system, consisting of diesel engine and fixed pitch propeller, a common way of matching is shown in figure 18. Starting point is the trial condition for, which the resistance-speed relation is known, from model tests or calculations (figure 18.a). Trial condition being defined as fully loaded ship, clean hull and still water condition. To this trial condition curve, one should add a margin for resistance increase, due to fouling and sea state. Therefore a sea margin or service margin is defined:

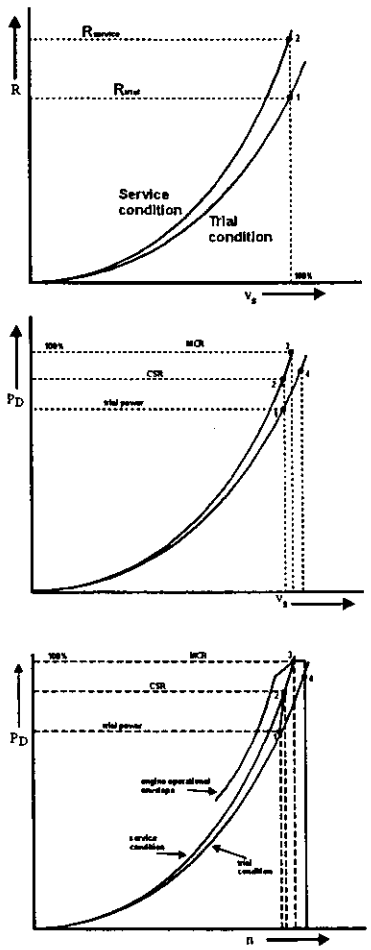


Fig. 18 - Matching of ship, propeller and diesel engine.

Sea marging = $R_{service}/R_{trial}$ ≈ CSR/trial power

Engine margin = CSR/MCR

1 = design speed in trial condition

2 = design speed in service condition

3 = engine at MCR, service condition, speed > design speed

4 = engine at max rpm, trial condition, speed > design speed.

$$SM = \text{sea margin} = \frac{\text{resistance in service condition}}{\text{resistance in trial condition}}$$

The sea margin should be chosen on basis of experience, or calculations with regard to added resistance by waves and hull fouling (see 2.4.2 and 2.4.3). Realistic values for the sea margin are 1.1 - 1.25. The thus obtained service resistance and service speed are the basis for the propeller design/selection and leads to propeller load curves for service and trial condition (figure 18.b propeller load as function of ship's speed and figure 18.c load as function of engine speed).

Often one defines the sea margin not as a ratio of resistances, but as a ratio of propeller

powers:

$$SM = \frac{CSR}{\text{trial power}}$$

where CSR = continuous service rating = the power needed by the propeller at service speed in service condition.

The ratio between these two definitions is small (the difference is the effect of change in propeller efficiency between trial and service condition).
The diesel engine could now be matched, such, that the engine can supply its maximum power in service condition. This means, that the diesel MCR point (100 % power and 100 % speed) lies on the load curve.
In order to get acceptable engine life and maintenance costs, the MCR of the engine should be larger than the required CSR. This also has the effect, that sfc at service speed is minimised.
The ratio CSR/MCR is called the engine margin :

$$EM = \frac{CSR}{MCR}$$

This margin should be in the order of 0.8 - 0.9.

This method of matching, as described, has proven to be a good one, for relative simple cases.
In service condition, the service speed can be maintained at an attractive engine load. With a further increase of resistance, there is some margin to maintain speed. With low resistance, an increase of speed is possible, without using full engine power, limited by maximum engine speed.
The matching as shown in figure 18 shows the situation with a diesel engine with a narrow operational envelope and a fixed pitch propeller. Later it will be shown that matching with gasturbines and diesel engines with a wide operational envelope on one hand and waterjets as propulsors offer a more tolerant situation.

5. OFF DESIGN CONDITIONS

5.1 THE OFF DESIGN PROBLEM

When the prime mover and propulsor have been selected and matched for the design condition, the question arises, whether changes in operational conditions lead to acceptable performance of the propulsion system. In order to answer this question, the drive and /or load characteristics (in terms of power or torque as function of rotational speed) have to be determined. After that, the new equilibrium can be established, by intersecting the load and drive curves. This leads to answers with regard to ship's speed, engine speed, power etc.
As discussed, in paragraph 2, there are two types of changes of operational conditions, which should be carefully distinguished:

- changes, which are related with ship's resistance, propulsor load or propeller pitch. In such a case the modified propulsor load curve (power against rpm) can be determined by using the open water diagram of screw type propellers or the characteristic curves of figure 8 for waterjets.
- changes, which are not related to the propulsor load, such as number of engines driving a propulsor or use of shaft alternators.

In the following sections it is assumed, that the design condition is fully known. This includes:

- resistance/ship's speed and propulsor load curves
- propeller open water diagram or waterjet characteristics curves
- prime mover drive characteristic
- the resulting operational point in terms of J, K_T, K_Q, v_s, n, P, etc.

All variables in design condition will be indicated with an index o.

5.2 CHANGE OF RESISTANCE/THRUST LOAD CHARACTERISTIC

The change of ship's resistance, propulsor load (e.g. one driving propulsor instead of two), or change of pitch lead to a modified propulsor power/torque speed curve. This new curve can, be determined with the open water diagram or with the waterjet characteristic curves. In the following the procedure will be described for screw-type propellers. For waterjets the method of calculation is similar but somewhat less complicated.
An elegant way of solving these type of problems is to work with ratio's relative to the design condition. The ratio's will be written as * variables, which in general are defined as follows:

$$x^* = \frac{x}{x_o}$$

The following relations between these ratio's exist:

$$\frac{R}{R_o} = \frac{T(1-t)}{T_o(1-t_o)} \text{ or shorter } R^* = T^* \,.\, t^*, \text{ where } t^* \text{ is defined as } t^* = \frac{(1-t)}{(1-t_o)}. \qquad (1)$$

$$\frac{v_s(1-w)}{v_{s,o}(1-w_o)} = \frac{v_A}{v_{A,o}} \text{ or shorter } v_s^* = v_A^*.\ w^*, \text{ where } w^* \text{ is defined as } w^* = \frac{1-w}{1-w_o} \quad (2)$$

$$J^* = \frac{v_A^*}{n^*} \quad (3)$$

$$K_T^* = \frac{T^*}{n^{*2}} \quad (4)$$

$$K_Q^* = \frac{Q^*}{n^{*2}} \quad (5)$$

$$Q^* = M_D^*.\ \eta_R^*, \text{ where } \eta_R^* = \frac{\eta_R}{\eta_{R,o}} \quad (6)$$

Further the propeller open water chracteristics are known:

$$K_T = f(J) \quad (7)$$
$$K_Q = g(J) \quad (8)$$

For a cpp K_T and K_Q are not only functions of J, but also of the pitch H.

For ease of calculation, it is often convenient, to linearize the K_T and K_Q curves, see figure 19. The error by linearization is normally small, because K_T and K_Q have a rather linear characteristic in the first quadrant.
When we use for the linearization the tangent of the K_T curve at J_o we obtain:

$$K_T = K_{T,o} + \left[\frac{dK_T}{dJ}\right]_O .(J - J_o)$$

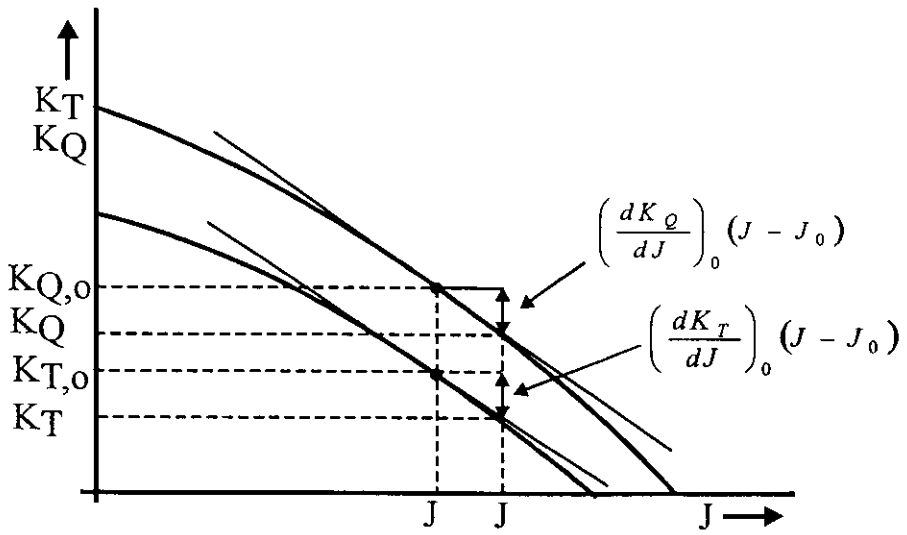


Fig. 19 - Linearization of K_T and K_Q curves at the design point J_o.
$K_T = K_{T,o} + (dK_T / dJ)_o . (J - J_o)$

$$K_Q = K_{Q,o} + (dK_Q / dJ)_o \,.\, (J - J_o)$$

The derivative of K_T is of course negative.
This relation can be written in the form of ratio's, as follows:

$$K_T^* = 1 + \frac{J_o}{K_{T,o}} \cdot \left[\frac{dK_T}{dJ}\right]_o \cdot (J - J_o)$$

$$K_T^* = 1 + a.(J^* - 1), \text{ where } a = \frac{J_o}{K_{T,o}} \cdot \left[\frac{dK_T}{dJ}\right] \tag{7}$$

In the same way for K_Q:

$$K_Q^* = 1 + b.(J^* - 1), \text{ where } b = \frac{J_o}{K_{Q,o}} \cdot \left[\frac{dK_Q}{dJ}\right] \tag{8}$$

a and b can be determined from the open water diagram at J_o.
Eight relations have been established now, which can be used to determine the modified $P_{D,L}$ - n curve, when resistance or thrust load has changed.
Some examples will explain the method.

5.2.1 INCREASE OF SHIP'S RESISTANCE

Due to fouling (or sea state, or towing) a resistance increase with a factor c at a constant ship's speed is assumed. This means:

$$R^* = c \qquad \text{at} \qquad v_s^* = 1$$

An increase of resistance must mean also an increase of thrust, which will be effected by increase of both K_T and n. The new J and K_T will follow from the relations between K_T and R and from the open water diagram. Thereafter the new K_Q can be determined and with that, the new propeller load curve. The procedure is:

$$T^* = R^* = c \qquad \text{(assuming } t^* = 1)$$

$$J^* = \frac{v_A^*}{n^*} = \frac{v_s^*}{n^*} = \frac{1}{n^*} \qquad \text{(assuming } w^* = 1)$$

$$K_T^* = \frac{T^*}{n^{*2}} = \frac{c^*}{n^{*2}} = cJ^{*2}$$

but also:

$$K_T^* = 1 + a \,.\, (J^* - 1)$$

The new operational point of the propeller is found by equalizing these two relations for K_T:

$$cJ^{*2} = 1 + a\,(J^* - 1)$$

This is a square relation in J*, from which J* can be solved. Next K_Q , Q and $P_{D,L}$ can be

determined, using the relations given in 5.2:

$K_Q^* = 1 - b\,(J^* - 1)$

$M_D^* = Q^* = \dfrac{K_Q^*}{n^{*2}}$ (assuming $\eta_R^* = 1$)

$P_D^* = M_D^*\, n^* = K_Q^*\, n^{*3}$

Herewith the new propeller power-speed relation is found. Intersection of the modified propeller load curve with the prime mover drive curve leads to the solution of the problem, see figure 20.

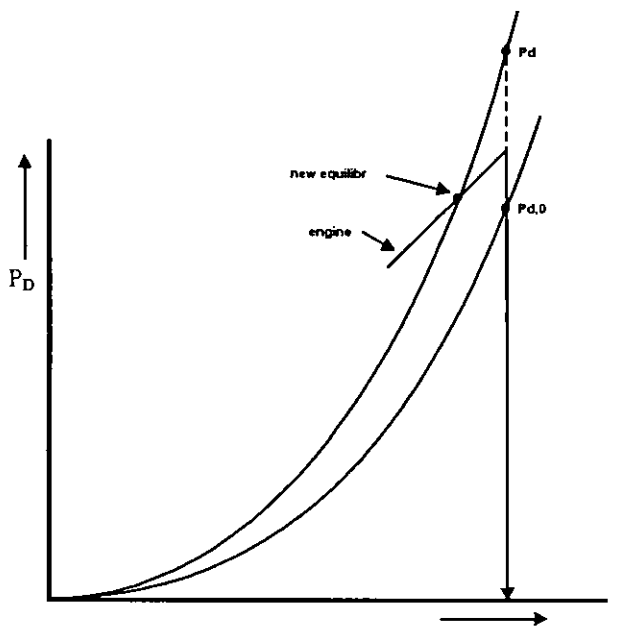


Fig. 20 - Increase of resistance has led to a new propeller load curve:

$$\frac{P_D}{P_{D,o}} = \frac{K_Q}{K_{Q,o}} \cdot \left(\frac{n}{n_o}\right)^3 \quad \text{for} \quad \frac{n}{n_o} = 1 \rightarrow P_D = \frac{K_Q}{K_{Q,o}} \cdot P_{D,o}$$

Intersection of new load curve with engine drive curve leads to new equilibrium.

5.2.2 SAILING ON ONE PROPULSOR WITH A TWIN SHAFT SHIP

When a ship is provided with two (or more) propulsors, the matching will be based on all shafts operating. In single shaft operation, with unchanged ship's speed, the working propeller has to generate a thrust, which is at least a factor 2 higher (see 2.4.4).
Assume a twin shaft ship with two identical propellers, which have as operational design point J_o, $K_{T,o}$ and $K_{Q,o}$ in figure 21.

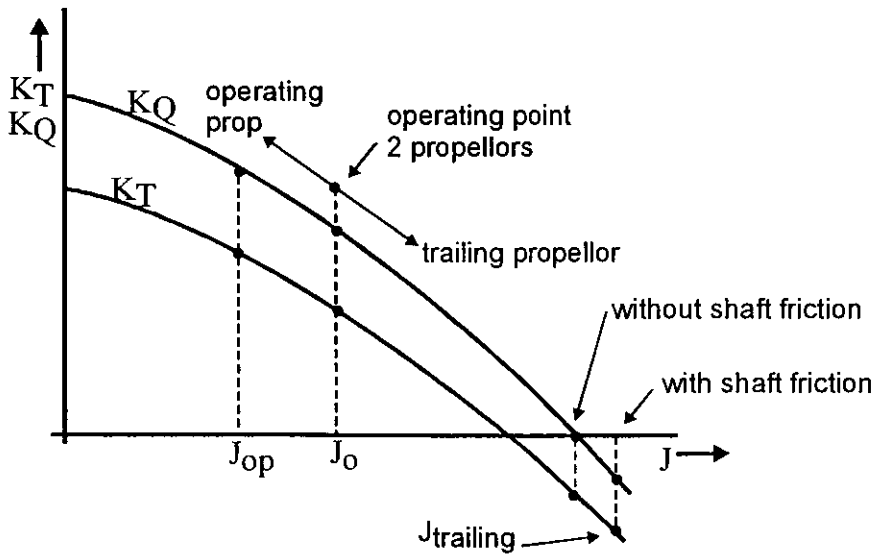


Fig. 21 - Operating points for a ship with two identical propellers.

J_o = both propellers in operation

J_{op} = operating point for one operating propeller

$J_{trailing}$ = operating point for trailing propeller with and without shaft friction.

Consider in first instance the trailing propeller. If this propeller is free to rotate, without friction loss in the shafting system, the torque to be developed by the propeller will be zero. This means, that K_Q will be zero. The operational point in the open water diagram can be established and thus K_T, which is negative. A negative thrust will be the result, which has to be generated by the operating propeller.
In reality there will be some friction in the trailing shaftline, for instance:
$M_{friction} = c\, n^2$
Now the following is valid for the **trailing** propeller:

$v_s^* = 1$ (ship's speed is assumed to be constant)

$$J^* = \frac{1}{n^*}$$

$$M_D{}^* = \frac{-M_{friction}}{M_{D,o}} = \frac{-c\,n^2}{M_{D,o}} = \frac{-c\,n^2}{d n_o{}^2} = -\frac{c}{d}\cdot\left[\frac{n}{n_o}\right]^2 = -\frac{c}{d}n^{*2}$$

$$Q^* = -\frac{c}{d}.n^{*2}$$

$$\left.\begin{aligned} K_Q^* &= \frac{Q^*}{n^{*2}} = -\frac{c}{d} \\ K_Q^* &= 1 + b.(J^* - 1) \end{aligned}\right] \quad \rightarrow \quad -\frac{c}{d} = 1 + b.(J^* - 1)$$

from which $J^*_{trailing}$ can be solved.
Then also $K_T{}^*_{trailing}$ can be determined and the negative thrust $T^*_{trailing}$:

$$T^*_{trailing} = K_T{}^*.n^{*2} = \left(\frac{K_T{}^*}{J^{*2}}\right)_{trailing}$$

Next the **operating** propeller will be considered. For the thrust to be generated can be written:

$$T^*_{operating} = \frac{2T_o - T_{trailing}}{T_o} \quad (T_{trailing} \text{ is negative})$$

$$= 2 - T^*_{trailing} = 2 - \left(\frac{K_T{}^*}{J^{*2}}\right)_{trailing}$$

This will result in an thrust load increase in the order of $T^* = 2.15 - 2.25$.
From here on the problem is the same as in 5.2.1.
Figure 22 shows the resulting situation for a twin shaft propulsion system driven by diesel engines. The drive and load curves for one shaft are shown, both in single and twin shaft operation.

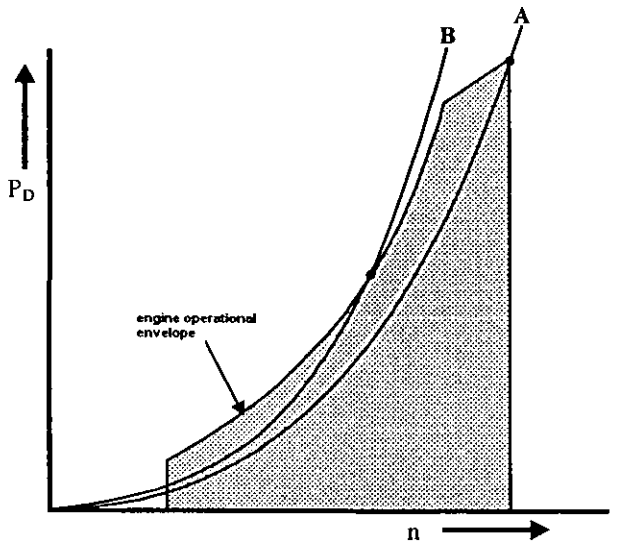


Fig. 22 - Utilization of diesel engine on a twin shaft ship with f.p.p.
A - load curve both shafts in operating
B - load curve with other propeller trailing
In single shaft operation maximum engine power is reduced considerably.

The increase of the load curve at constant shaft speed is the ratio of K_Q* and will be in the order of 1.4. This is far less than the increase of thrust at constant ship's speed. From the figure it follows that the engine cannot be used up to maximum power in single shaft mode. To overcome this problem cp propellers should be used. On high speed craft nevertheless sometimes fp propellers are used.

In case waterjets are applied the situation is easier, with regard to the computational effort to solve the problem. When the ship sails on one waterjet instead of two (identical) waterjets the vertual increase of thrust to be generated by the operating waterjet is:
$T^* = 2$ at $v_s^* = 1$
In this case there is no (or hardly any) increase of ship resistance as there is **no** trailing propeller, but only some eccentric thrust which gives a little increase of ship resistance. The result is that the full ship resistance curve can be drawn in the waterjet characteristic curve, whereas in normal operation only half of the ship's resistance has to be used. See figure 23.

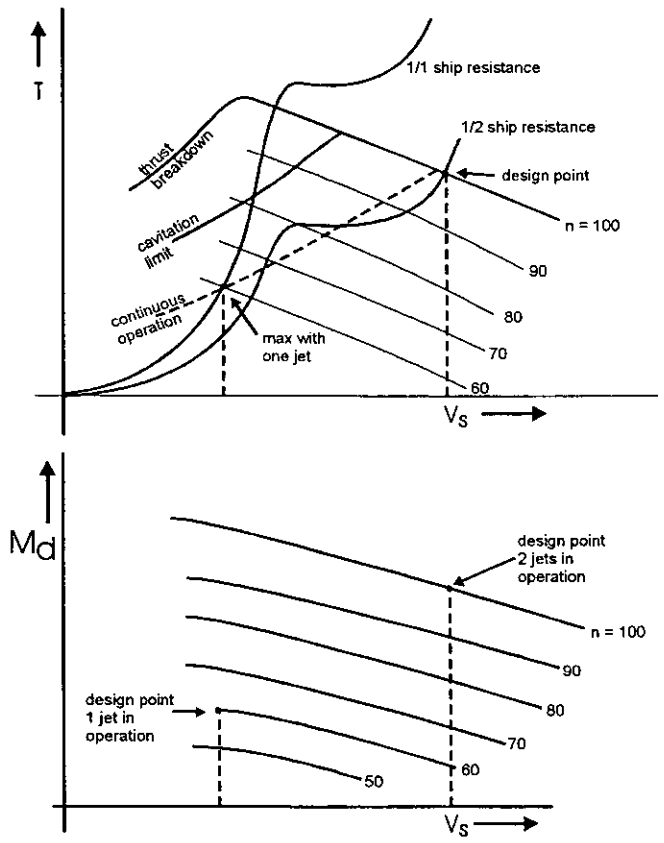


Fig. 23 - Waterjet operation with one out of two jets.

From figure 23 it will be clear that, in this example, the maximum attainable continuous speed on one waterjet, on basis of the loading of the impeller and cavitation will be $v_s^* \approx$

0.4 at an impeller speed $n^* \approx 0.6$.
From the torque curve it follows that the prime mover has to deliver a torque $M_D^* \approx 0.45$ at $n^* \approx 0.6$. A sequentially turbocharged diesel engine or a gasturbine will have ample capacity to do so.
From prime mover point of view the attainable speed may be much higher, but at the expense of operation in the cavitation zone of the waterjet.

5.3 CHANGE OF DRIVE CHARACTERISTICS

This type of off design conditions requires less calculations than the foregoing in 5.2. The propulsor load does not change, so there is no need to consider the open water diagram. In these cases there is hardly any difference between propellers and waterjets.

Figure 24 shows what happens with a twin engine installation on one fixed pitch propeller. The engines are identical and have a constant torque characteristic. With both engines in operation at maximum power, the total drive curve is matched with the load curve.

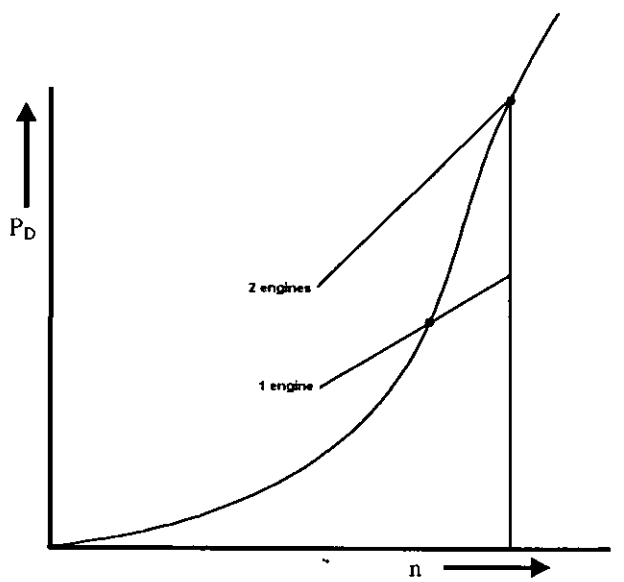


Fig. 24 - Two diesel engines with a constant torque characteristic driving a f.p.propeller
2 = two engines driving, design point
1 = maximum operating point with single engine operation.

When only one engine is operating, the drive curve is halved. The new equilibrium situation is at the intersection of the original load curve and the drive curve of one engine. Figure 25 shows the situation with two non sequentially turbocharged diesel engines, which cannot develop maximum torque at reduced speeds. It is clear, that single engine operation in this way is not a good solution. A cpp could solve the problem, enabling full power utilisation also in single engine mode. Another solution is to use an engine with sequential turbocharging, which comes close to the situation of figure 24.

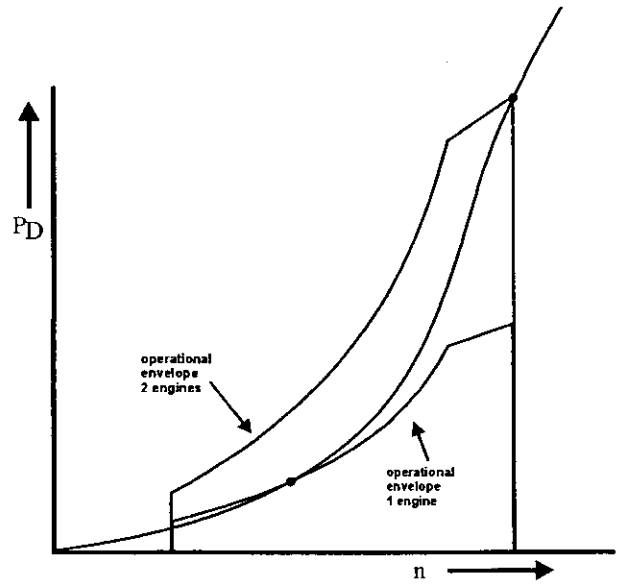


Fig. 25 - Two turbo charged diesel engines driving a f.p.propeller
2 = two engines driving, design point
1 = maximum operating point with single engine operation.

Figure 26 shows the same twin engine situation with gasturbines. In this case there is not a need for a cpp.

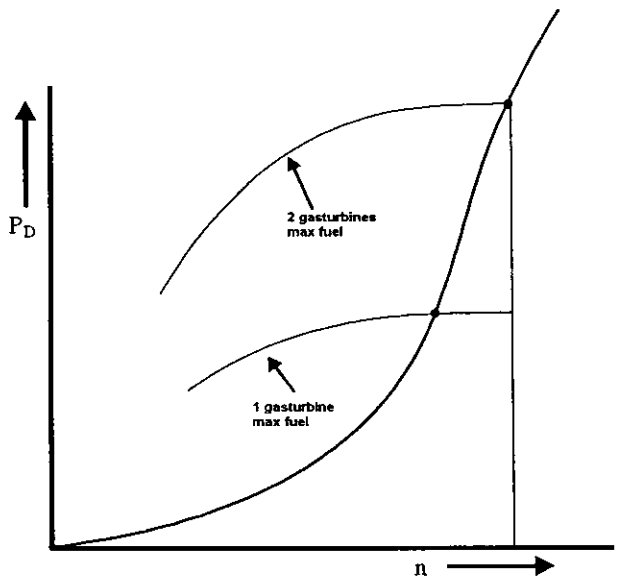


Fig. 26 - Two marine gas turbines driving a fixed pitch propeller
2 = two turbines driving, design point
1 = one turbine operation at maximum fuel setting.

Figure 27 shows the influence of a shaft generator, requiring a constant power, on a propulsion system with a diesel engine and fpp. The MCR of the engine has been matched to propulsion plus shaft generator load. The drive characteristic of a turbocharged diesel engine is such, that the margin between drive and load curves at lower speeds becomes small or sometimes even negative. This means, that with the application of auxiliary machinery not only the maximum speed/maximum load condition should be considered, but also lower speeds. It is common practice to design a shaft generator system not for the full speed range, but for instance for 0.7 - 1 of the maximum speed. This is also done because a larger speed range is difficult, with regard to maintaining a constant frequency in the electric power supply system.
When a cpp has been applied constant speed operation is possible, which eliminates the problem of overloading the engine and assures constant frequency. In that case however low speed sailing on reduced pitch and full rpm will lead to a bad propulsion efficiency.

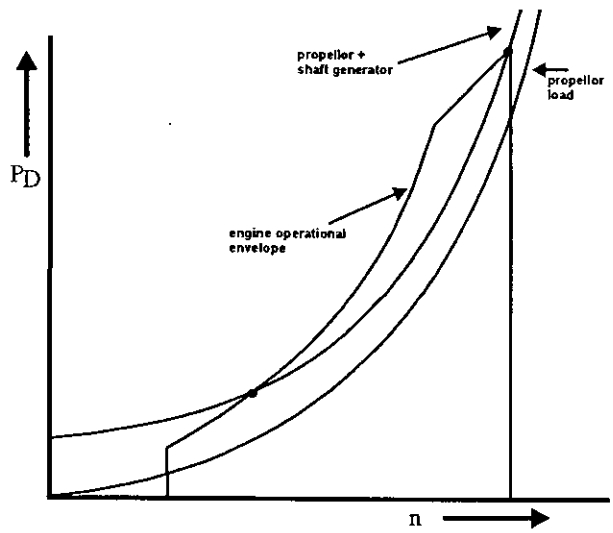


Fig. 27 - Turbo charged diesel engine driving a propeller and shaft generator, with constant load. Engine is matched for both loads at maximum speed.
At lower speeds the margin between engine capability and total load may become negative.

WEGEMT Association 33rd School
"FAST SHIP PROPULSION PLANTS"
May 22nd – 26th , 2000 Genova, Italy

MATHEMATICAL MODELLING AND SIMULATION OF PROPULSION SYSTEM

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Hosted by:

Dipartimento di Ingegneria Navale e Tecnologie Marine – Università di Genova

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ABSTRACT

Numerous symposia indicate the huge volume or public presentations and professional debates covering much of the feverish effort to become increasingly competitive by using cost-effective, new and improved systems. All of these reflect the urgent need for safe, reliable and readily maintainable systems, environmental issues and the commercial competition to remain economic and in business. As part of this pressure, marine engineers will in the 21st century be expected - perhaps required - to provide explicit evaluations of the performance of proposed designs. This means that there must be available suitable methods and analysis tools to underpin the designs. We need efficient tools, and simulation is one of the most important and useful one available to those responsible for the design and operation of complex systems in general, and marine engineering systems in particular.

The major thesis of this paper is that simulation is essential in the future of marine engineering. The state of the art of simulation as a workable tool for design and operational aspects of marine engineering systems are discussed and exemplified.

INTRODUCTION

It seems that at the present time there is a renewed interest in the subject of high-speed marine crafts. Certain aspects of the design of marine vehicles are difficult to clearly assign as the responsibility of either the naval architect or the marine engineer. At least one can say that selection of the main propulsion plant; electrical power generation and related requirements are tasks of the marine engineer.

In selecting a propulsion engine and associated equipment for a given boat, many attributes must be considered, such as

1. Weight
2. Space occupied
3. First cost
4. Reliability
5. Length of life
6. Flexibility and quietness of operation
7. Cost of upkeep
8. Cost of fuel consumed
9. Suitable for the type of propulsor to be used

Some of these factors have more or less significance for application in high-speed marine vehicles.

The question to be explored here is what must subsequently be done in design to ensure a well-engineered propulsion plant. At first glance, it might seem like very little, since the engine is to be purchased as a package, tempting the marine designer to adopt a "bolt it down, hook it up, and push the GO button" attitude. Unfortunately, the case is not that simple. Actually, there is a large engineering effort needed in spite of the package nature of the propulsion engine. The load, the environment and the service make demands upon the engine, and the designer's task might, in consequence, be summarily described as ensuring that all of these demands are successfully met,

In order to ensure that these demands are met, it would be desirable to be able to predict the system behaviour before they are built. It is highly probable that marine engineers in the 21st century will be expected - perhaps required - to provide explicit evaluations of the performance of proposed designs.

To underpin the designs of future marine engineering systems, suitable methods and analysing tools must be available. One efficient and maybe the only feasible way are to use computer studies and computer simulation. Simulation permits tracing the changes in system state through simulated time and under varying rules of operation and component characteristics. This gives means of "trying out" alternative operational policies and components without actually tampering with the physical system itself.

In this paper we will therefore, discuss what simulation is, its requirements of modelling and show by examples its use in marine engineering system evaluation.

MODELLING ISSUES

Marine engineering is not simply categorised as, for example mechanical, electrical or chemical engineering, but is an integrated engineering effort comprising parts of many engineering disciplines directed to the development and design of systems. Thus, marine engineering is mainly concerned with design and operation of systems.

The word "system" is used so often and so loosely to describe a variety of concepts that it is hard to give a meaningful definition of the word or even to see the basic concept that unites its diverse meanings. In this paper we define a system as

- As system is assumed to be an entity separable from the rest of the environment of the system by means of a physical or conceptual boundary.
- A system is composed of interacting parts.

The essence of what may be called the "system viewpoint" is to concern oneself with the design and operation of entire assemblages of parts in contrast to being concerned with the individual parts themselves.

The role of experimentation in component and system engineering provide a contrast. The component engineer can experiment readily with his actual component, but experimentation with systems is very difficult and often costly. Thus, for a system engineer to be certain of the overall performance of his system before it is built computer studies and computer simulation of proposed systems are his only method of experimentation.

Simulation is therefore, an experimental and applied methodology, which can be used to test a concept for its capacity to satisfy demands placed on it. Once the performance requirements for a system are determined and a tentative design based on these requirements is complete, there is always nagging questions such as:

- Will the actual system perform as required?
- What will happen if the operating conditions change?
- By how much does a critical factor have to change before critical response results?
- How sensitive is one factor to changes in another?

By contrasting some of the major uses of computer simulation to alternative methods of answering the questions above, one can begin to get a feel for its usefulness. In addition to providing a much more flexible and frequently less expensive vehicle for answering the questions highlighted above, simulation has other benefits. It can be used for design studies, and multiple runs using various configurations can be used to home in on design parameters. "What if" studies are easily and fruitfully done via simulations. The effects of constraints whether due to physical properties or safety policies can easily be determined. Simulation has a general applicability and is the natural link between design and operation. Simulation must therefore, be considered as a central theme in future practice and education in marine engineering.

Simulation is a very wide-open and somewhat ill defined subject. According to Webster's Collegial Dictionary, to simulate is "to feign, to attain the essence of without the reality". With some trepidation, the author defines simulation as follows: "Simulation is the process of designing a model of a real system and conducting experiments with this model for the purpose to understand the behaviour of the system or of evaluating various strategies for the operation of the system".

Thus, we understand the process of simulation to include:

- The construction of a model
- The analytical use of the model by conducting experiments with this model for the purpose either of understanding the behaviour of the system or of evaluating various strategies for the operation of the system.

Thus for modelling and simulation to be an effective problem solving methodology, three important problem solving phases are involved:

- Modelling which is by definition, the description of system behaviour by means of suitable chosen mathematical relations or equations.
- Numerical analysis and algorithm selection to solve the equations comprising the mathematical model.
- Conducting experiments with a model for the purpose to either understand the system behaviour or evaluating strategies. This phase is what we often call simulation.

Thus, for simulation to be an effective problem solving methodology, it is required that the analyst have an insight into all three of the above mentioned phases. Only in this way can there be a degree of assurance that the simulation outputs constitute meaningful and useful solutions.

This means that the idea of a model of the system is the first and central point in a simulation study. To proceed from a schematic diagram to a reasonable model, a great deal of judgement is required. For simple systems, equation formulation according to well-established rules may pose no great difficulties. However, in many practical cases the analyst is required to use considerable ingenuity to organise the component models into a system model in the absence of standard procedures. This often proves to be a major impediment to those using simulation for system design. In spite of its difficulty and outmost importance, the means for constructing a suitable model is often given relatively little attention.

Thus, central to the concept of systems engineering, simulation and indeed central to much of human activity, is the creation and use of models. It can be argued in fact, that all intellectual activities involve modelling of sorts, since we always make conceptual simplifications in order to comprehend a real world object. Models are simplifications of reality. Physical models have long traditions in engineering, but since the advent of computers we deal with what are usually called "mathematical models" which resemble the real systems they represent more abstractly than physical models. When discussing models, it is important to note that the characteristic feature of a model is that

some, but not all

of the features of the real system are reflected in the model and that no system can be modelled exactly. The idea is that the time history of important variables should be produced by the mathematical model even before the real system is constructed.

Perhaps the most universal model of representing dynamic systems is through differential equations, and it is entirely superfluous to point out the advantages of this method. On the other hand, anyone who has ploughed through page after page of equations describing an only moderately complex system will testify that despite the explicit nature of differential equations, or perhaps because of it, it may still be very hard to see just what the subsystem models are and whether some effect has been badly or incorrectly modelled. In fact, hardly any engineer of even modest experience can claim to have avoided the embarrassing experience of writing an equation set that subsequently was found to admit of no solution at all.

If we consider some of the devices that a marine engineer might encounter in modelling engineering systems; these could include rigid and flexible mechanisms in general motion, electric and electronic components, hydraulic and pneumatic devices, electromechanical devices such as solenoids and motors, and thermal and heat transfer devices. In order to adequately model such devices, the marine engineer must be prepared to deal with a number of physical domains that interact.

It is clearly unrealistic to bring the marine engineer, who must be a jack-of-all-trades to some extent, up to a level of understanding of a specialist in all energy domains. The education to which most marine engineers are exposed does not yet equip them adequately to carry out these tasks. In mathematics one learns how to solve systems of equations, but means for constructing a suitable model is often given little or no attention. Therefore, model construction still seems to be a hurdle, and it is proper to ask if this is the reason why simulation has not gained the attention it deserve in engineering. Reference [1, 2]* represent two rather complex marine engineering

simulation studies. In these examples are no mention of using a modelling methodology and the study of the used models is very difficult. It is therefore; highly desirable to have a logical procedure for abstracting a physical reality into a mathematical model which is amenable to direct computer solution.

We may imagine that the mathematical models always involve equations directly, this is not quite true. In my work I have focused on development and teaching of methods for modelling marine engineering systems and techniques for simulating their behaviour. I have found that the bond graph method is the most powerful tool in practical engineering work and teaching modelling. The bond graph method helps me to better understand the behaviour of a broad range of engineering systems. It gives a logical procedure for abstracting the physical reality into a mathematical model, which is amenable to direct computer solution. It is particularly suited to the development of models for systems in which energy occurs in several forms. I believe that bond graph methods are very useful for studying dynamic engineering system behaviour and belong in the problem-solving tool kit of every marine engineer. If you are a reader who is not familiar with bond graphs, then I hope that you will be motivated to investigate the method further.

Before one can discuss the challenge of this new approach to mathematical modelling of marine engineering systems, the basic bond graph formulation has to be mastered. Many discussions falter because people judge the new approach to superficially. Therefore, only a short introduction and example is given in the following to indicate what bond graphs are and what a model look like.

BOND GRAPHS

Bond graphs are unified graphical notation for the representation of dynamic models of interacting subsystems involving all forms of energy. This notation leads to a graphical description of a dynamic system very much resembling a chemical bond diagram in appearance. The basic concept behind the bond graph is that power is the fundamental constraint in a physical system and is the one variable that is common to the whole system. Thus, the essential feature of the bond graph approach is the representation of energetic interactions between systems and system components by an energy bond.

The basic idea of a graphical representation of the power interaction among general dynamic subsystems was invented by Professor H.M. Paynter of the Massachusetts Institute of Technology in 1960. He named his graphs bond graphs since he thought of them as analogous to bond diagrams in chemistry. The elements and topology of the bond graph representation provide a conceptual reticulation of the real system into elements drawn from a set of ideal, but physically realisable, lumped parameter components. The reticulated structure represents a conceptual separation of the energy storage and dissipation processes within the system and a description of the system.

SIGNAL VARIABLES AND BASIC ELEMENTS

Although there are textbooks available explaining the basic concepts of bond graphs [Karnopp, Margolis and Rosenberg, 1990], this section will outline the underlying ideas for those who may not be familiar with them.

The fundamental component of a bond graph is the power bond used to couple the energy ports of system components as shown in Figure 1. The half arrow shows the direction in which the positive power is flowing.

The successful use of bond graphs depends to a great extent on choosing a set of variables which describes the instantaneous power balance within a system. In a wide class of energetic systems, the propagation of energy or power, is given by the product of two quantities, an effort variable, e, and a flow variable, f.

Effort and flow variables are called "power variables" because their product represents power flowing between subsystems,

$$\text{Power flow} = e(t) \cdot f(t) \tag{1}$$

Two other variables of importance is the generalised momentum

$$p = \int^{t} e(t)dt \tag{2}$$

and generalised displacement

$$q = \int^{t} f(t)dt \tag{3}$$

The p and q variables are called "energy variables" since for specific element p and q is measures of stored energy.

$$\text{Energy} = \int^{t} e(t) \cdot f(t)dt \tag{4}$$

Table 1 shows an organisation of the above four variables in several energy domains in a scheme which build in to bond graphs a specific set of analogies. A glance at the table shows that in most cases the variables have traditional names and units in the various energy domains. The nine multiport elements function as components of subsystem and system models.

Using the classification of power and energy variables presented, it turns out that only nine basic types of multi-port elements are required in order to represent models in a variety of energy domains. They are, in many cases, idealised mathematical versions of real components, and in other cases they are used to model physical effects. The nine basic multi-ports are classified in Figure 2, according to the way they process energy. Roughly speaking, these elements account for energy supply, dissipation, storage and transduction from one mode to another (i.e. the conversion of energy from one form to another).

Using these basic elements, a graphical model can be constructed of a system. It is a remarkable fact that models based on apparently diverse branches of engineering science all can be expressed using the notation of bond graphs. This allows one to study the structure of a system model. The nature of the parts of the model and the manner in which the parts interact can be made evident in a graphical format. The resistor, capacitor and inertia elements are used to model passive elements in the various energy domains. The resistor, which relates the effort to the flow on its single bond, is often constrained to dissipate power only, as an electrical resistor

or mechanical dashpot would. The inertia and capacitor involve the energy variables, p and q, and store energy.

In mechanical systems, the I-element represents a mass and the C-element a spring. The 2-port transformer and gyrator elements are usually shown with a "through" sign convention takes the form

$$e_1 \cdot f_1 = e_2 \cdot f_2 \tag{5}$$

where the numbering of the bond provides a convenient way to identify e, f, p and q variables. In mechanics, the TF element represents lever and gear ratio and the GY represents gyroscopic coupling.

Perhaps Paynter's major innovation in inventing the bond graph formalism was to represent interconnection laws at power conserving multiport elements. In mechanics, the laws go by the names of "geometric compatibility" and "dynamic force balance" while in electrical circuit theory the corresponding laws are called the Kirchoff current and voltage laws. The 0- and 1-junction elements are shown in 3-port form although n-port versions are often used by an obvious extension of the constitutive laws given in Figure 2.

In mechanics, if three 1-ports are bonded to a 0-junction, the forces are all equal. This means that the 1-port elements are in mechanical series. If the elements are bonded to a 1-junction, velocities are common and the elements are in mechanical parallel.

Using bond graphs, the differential equations can be constructed in an algorithm manner from the graph. That is the graph is perfectly equivalent to the differential equations of the form

$$\frac{dx}{dt} = f(x, u, t) \tag{6}$$

with state vector, x, and input vectors u. The state equations can thus be written unambiguously in an algorithmic way. However, before the equations can be written from the bond graph, causal information must be added to the graph. Causality means obtaining an explicit indication of which variables for an element is to be considered independent and which are to be considered dependent. Said another way, which are the inputs and which are the output variables. The problem of assigning a particular causality to a system model is related directly to the manner in which the system equations can be written without difficulty. The way both the effort and flow signals travel are indicated directly on the graph by drawing a perpendicular line at one or the other end of the bonds, as shown in Figure 3.

The simultaneous representation of a physical and a computational structure is a unique and very powerful property of the bond graph which other representations like the linear graph or the block diagram do not have.

Thus, in the following, presentation of bond graph modelling an illustration of some useful features of causality, and its implications for writing the differential equations will be illustrated through an example system well known to marine engineers.

MODELLING OF A SYSTEM

On Figure 4 is shown a simplified schematic of a typical propulsion system that consists of a diesel engine, flexible couplings, a gearbox including a clutch and PTO, alternator and a propeller

Before starting the modelling exercise, it is important to define the objectives of the study where the model is to be used. The model presented here is to be used for a study of the transient loading in the system due to closing of the clutch. A bond graph model of the example system which can be used for this purpose is shown in Figure 5. Note that we used the expression "a bond graph model" and not "the bond graph model" because a model is actually a reflection of the modeller's understanding of reality, its components and their interrelations. There is no unique model and no unique solution to a real world problem.

As shown in Figure 5, the complete system model can be assembled by submodels in the same manner as it is done with the actual hardware. Even the graphical structure of the overall bond graph model in Figure 5 is similar to the structure of the schematic shown in Figure 4. This property of the bond graph is convenient in order to keep the physical understanding of the mathematical model through the process of modelling and analysis of the system. In addition reusability of submodels is crucial in an effective modelling environment.

The diesel engine is shown here as consisting of the total engine and flywheel mass and the engine torque is shown as an effort source modulated at signal-power level through the engine governor. A more complete model of the diesel engine is discussed later.

The flexible coupling model is shown to consist of flexibility and damping with a common velocity difference. The 0-junction between bond 2 and 9 is used to "take the velocity difference" which is distributed to both C_5 and R_6. The explanation of this model is from the following arguments.

When torque is applied to a flexible coupling, the resulting torque-angular deflection characteristic, is non-linear as shown in Figure 6. In addition to the non-linearity, a hysteresis characteristic is exhibited. This means that there is damping associated with the coupling. The significance of this damping might be questionable for this particular study. Bond graphs are particularly helpful in allowing the analyst to add or remove physical effects in order to determine their importance.

The gear is shown as a transformer (TF) with a modulus equal to the gear ratio and the moment of inertia of the gear wheels are connected to a 1-junction on either side of the transformer.

The clutch is shown as a resistor, R_{20}, between two 1-junctions representing the velocity of the two sides of the clutch. Explanation for this model is as follows.

When closing the clutch, an applied force is pressing the plates into close contact and transmitting torque from one shaft to the another by friction. This frictional torque drives the relative velocity between the surfaces towards zero. With a frictional torque less than the sticking torque is no problem. The difficulty occurs when the frictional torque is equal to the sticking torque, because then the relative velocity becomes zero and the independent masses becomes rigidly constrained. Thus, closing the clutch changes the structure of the system and the order of the system model changes, which means that we must reformulate the model after each clutching operation.

However, one solution to this problem is that we do not let the relative velocity go to zero in the stick region, but have a finite value $\Delta\omega_r$. In this way, the two masses will remain independent at all clutch states.

The transmitted torque from the resistor will have different value depending on the relative velocity between the two halves.

For $\omega_r > 0.5 \cdot \Delta\omega_r$

$$T_f = \mu \cdot F \cdot r_m \cdot n_p \qquad (7)$$

with

μ = friction coefficient
F = applied normal force
r_m = mean radius
n_p = number of friction faces

and for $-0.5 \cdot \Delta\omega_r < \omega_r < 0.5 \cdot \Delta\omega_r$

$$T_f = \frac{(e_7 - e_9) \cdot I_{22} + e_{23} \cdot I_8}{I_{22} + I_8} \tag{8}$$

The friction torque given by Equation (7) will drive the relative velocity, ω_r, towards zero, and the torque given by Equation (8) will keep, ω_r, within the band, $\Delta\omega_r$. The method proposed minimises numerical stiffness problems, allows constant causality and system structure, and computes jumps automatically. The bond Ag_r must not be too short or the integration routine will jump in and out of the bond Ag_r.

The bond graph shown in Figure 5 is augmented with causality marks, power convention and numbered bonds. The numbering of the bonds assures unambiguous referral to system variables.

When assigning causality to the I and C elements, one tries to maximise the number of elements with integral causality which means

$f = \frac{1}{I}\int e\,dt$ $\quad \xrightarrow[f]{e}\ I$

and

$e = \frac{1}{C}\int f\,dt$ $\quad \vdash\!\xrightarrow[f]{e}\ C$

If the causal strokes must be assigned at the opposite end of that shown above they have differential causality and thus become dependent elements. Detecting numerical difficulties such as implicit differential equations or algebraic loops before writing the equations is a unique feature of the bond graph causal assignment procedure compared with other approaches.

When integral causality can be assigned to C and I elements, then each energy variable is statically independent of all the others. Thus, the state variables are chosen as the, p, variable on every independent I element and the, q, variable on every independent C element.

From the causality shown in Figure 5 it indicates I_{11} and I_{25} do not have integral causality. This means that p_{11} and p_{25} are not independent, but statically related to the other p-variable(s). Thus, we can count the number of I and C elements with integral causality to determine the number of state equations to write.

The system then contains 9 independent energy-storing elements.This, as will be demonstrated when writing equations is the number of state variables and hence the systems order. One often hears of high-order and low-order models of systems. It is difficult looking at differential equations which effects are neglected. This is not so when using bond graphs since

the graph clearly shows which effects are included or neglected and the resulting system order. There is nothing said at this stage about linearity. It is valid for linear and non-linear elements.

DERIVATION OF SYSTEM EQUATIONS

A bond graph augmented as shown in Figure 5 contains exactly the same mathematical information as a differential equation. It therefore, follows that converting a bond graph into a differential equation should be a straightforward process. It is in fact, a completely algorithmic process and can easily be computerised.

The algorithmic procedure for deriving the system equations will now be illustrated. Here in this paper three state equations will be written as examples. These equations are for the engine inertia, PTO-gear inertia and the ship inertia.

In short the idea is to express the derivatives of the state variables as efforts and to proceed through the graph following the causal-stroke information until all expressions are evaluated in terms of state variables and input variables.

To derive the governing equations, starting with p_2 it proceeds as follows:

$$\frac{dp_2}{dt} = e_2 = e_1 - e_3$$

Using a proportional controller then

$$e_1 = S_{e1} = k_p \left(g_{reference} - \frac{p_2}{I_2}\right)$$

Following the causal marks indicates that e_3 is equal to and "caused by" e_4, and from summing efforts at the 1-junction we obtain that

$$e_4 = e_5 + e_6$$

From the coupling relations let us assume the following relations

$$e_5 = F_{spring} = K \cdot \theta^2 = K \cdot q_5^2$$

and

$$e_6 = F_{damp} = b_6 \cdot [\omega]^2 = b_6 \left[\frac{p_2}{I_2} - \frac{p_8}{I_8}\right]\left|\frac{p_2}{I_2} - \frac{p_8}{I_8}\right|$$

The resulting state equation then becomes

$$\frac{dp_2}{dt} = e_1 - e_5 - e_6 \qquad (9)$$

Then for the PTO gear inertia

$$\frac{dp_8}{dt} = e_8 = e_7 - e_9 - e_{19}$$

Following the causal marks again it indicates that e_7 is equal to and caused by e_4 which is determined previously.

The torque e_{19} is "caused by" e_{20} which are given by either equation (7) or equation (8) from clutch

In order to determine e_9, we must use the relations for the transformer TF given as

$$e_{10} = m \cdot e_9$$
$$m \cdot f_{10} = f_9$$

where m is gear ratio

This means that using the causal marks as "road signs" e_9 can be expressed as

$$e_9 = \frac{1}{m} e_{10} = \frac{1}{m}\left(\frac{dp_{11}}{dt} + \frac{q_{14}}{c_{14}} + b_{15}\left[\frac{1}{m} \cdot \frac{p_{17}}{I_8} - \frac{p_{17}}{I_{17}} \right] \right)$$

assuming in this case linear spring and damping in the PTO-coupling.

The term $\frac{dp_{11}}{dt}$ occur since p_{11} is not an independent variable, but statically related to other p variable(s)

The static relationship is easily determined by first noting that

$$p_{11} = I_{11} \cdot f_{11}$$

where f_{11} is the flow on bond 11. By following the causal strokes, it is seen that f_{11} is "caused by" f_{10} which according to the constitutive law for the transformer is f_9/m. The flow f_9 is "caused by" f_8 which according to the constitutive law for the inertia then is

$$f_8 = \frac{p_8}{I_8}$$

Thus

$$p_{11} = \frac{1}{m} \cdot \frac{I_{11}}{I_8} = \frac{p_8}{I_8}$$

and

$$\frac{dp_{11}}{dt} = \frac{1}{m} \frac{I_{11}}{I_8} \cdot \frac{dp_8}{dt}$$

$$\frac{dp_{11}}{dt} = \frac{I_{11}}{m \cdot I_s} \cdot \frac{dp_8}{dt}$$

Substituting this relation into e_9 and this again into the starting $\frac{dp_8}{dt}$ yields an implicit equation. Thus through manipulation the final state equation is obtained as

$$\frac{dp_8}{dt} = \frac{m^2 I_8 + I_{11}}{m^2 I_8} \left[e_5 + e_6 - e_{19} - \frac{1}{m} \frac{q_{14}}{C_{14}} - \frac{b_{15}}{m} \left[\frac{1}{m} \frac{p_8}{I_8} - \frac{p_{17}}{I_{17}} \right] \right]$$

Always when dependent elements occur then we obtain implicit equations.

So to the state equation for the ship inertia where the propeller characteristics are involved. These characteristics are given as curves where K_T and K_Q can be found when J is known. Starting as previously with

$$\frac{dp_{32}}{dt} = e_{32} = e_{31} - e_{33}$$

The speed of advance v_A which is related to ship speed v_s by the wake fraction w by the relation

$$v_A = (1-w)v_s = (1-w)\frac{p_{32}}{I_{32}}$$

The advance coefficient J is given a function of v_A, propeller diameter D and propeller revolutions per second n_p. That is

$$J = \frac{v_A}{n_p \cdot D} = \frac{(1-w)\frac{p_{32}}{I_{32}}}{D \cdot \frac{p_{29}}{I_{29}}}$$

The trust T can then be computed from the well-known equation

If we assume a square ship resistance law then

$$e_{33} = R_{33} \cdot \left(\frac{p_{32}}{I_{32}} \right)^2$$

This systematic manipulation shows that the physical understanding lies in constructing the bond graph and not writing the differential equations.

COMPUTER MODELLING AND SIMULATION ENVIRONMENTS

The topic of computer solution of differential equations is especially important to a discussion of engineering problem solving.

The most important numerical task in the simulation of continuous time dynamical systems is the calculation of the system state trajectories from the typical first-order equation in the form

$$\dot{x} = f(x, u, t)$$

with initial condition $x(t_o) = x_o$.

By integrating both sides of the differential equation with respect to time, t, we get

$$x(t) = x_o + \int^t f(x, u, t)dt$$

This indicates that integration is an important mathematical operation in solving differential equations. Generally a numerical solution to this problem is required, and the topic of numeric integration has therefore, received considerable attention over the years. As a result, an extremely large number of methods exist. No attempt will be made to discuss this subject here, but refer to the well known computer language MATLAB. In a MATLAB User Manual chapter on "Ordinary Differential Equations" is described how to use MATLAB to solve initial value problems of ordinary differential equations.

As mentioned previously, the heart of any simulation program is the integration algorithm whose goal is to approximate the behaviour of a continuous-time system on a digital computer. Thus, with the plentiful availability of good integration algorithms anyone can construct their own simulation programs.

The derivation of differential equations from an augmented bond graph follows an organised pattern and well established rules, and it should be possible to use software logic to generate the model description directly from the computer screen. Software that assists model building could be extremely useful and a break-through in utilisation of the simulation technology in the future.

Looking at the use of simulation as a tool from an industrial point of view, the added value using simulation is the use of the model to analyse the problem at hand. The time spent modelling new components and assembling sub models to produce a complete model is often found to be far too large to be justified. Our experience from an application engineering point of view, is that the problems where simulation could have given valuable contributions, often have been solved or bypassed by the time simulation models are available.

This indicates that software for modelling and simulation should in general include a good modelling module, including strong modelling support and a comprehensive model library for easy reuse of models, in addition to a powerful solver module. The broad area of applications and the diversity in experience and training of people using simulation in industry, also indicates

the flexibility of the modelling and simulation software that is required. The model building process should serve as the source for creating different applications for different purposes, and the interaction with other software should be kept open and easy. The assembly of models in different representations together with existence of software that allows people with different training in modelling and simulation to work together, is a key feature for increasing the use of simulation in the industry.

Previously we have discussed issues related to the unified modelling method of bond graphs. In order to make this method and important part of an effective problem solving environment for industrial applications, simulation programs supporting bond graphs are developed.

The three most known commercial programs are:

- 20-SIM ("Twente-SIM") from Controllab Products Inc. This is an integrated modelling and simulation environment. It is a complete integrated program with the strength and weakness this gives when it comes to integration with other systems or analysis programs. This program is perhaps the most integrated modelling and simulation environment available.
- CAMP-G from CADSIM Engineering is a pre-processor for Digital Simulation Languages. It provides the user with a tool to construct a graphic display of a bond graph on the screen. The program is only supporting bond graphs. Together with a powerful solver-module as ACSL from MGA Software, this program can provide some of the advantages that bond graph modelling gives.
- MS1-Modelling System 1 from Lorenz Simulation (Lorenz Simulation), is of the authors opinion a professional tools and its strength is its open and flexible design. It's "multi-formalism input" provides a consistent handling of several modelling representations besides bond graph, and it's many possibilities for import and export of models for other solvers makes it into a real modelling engine. It is the tool for integration of models with different origins. MS1 is also supporting a modular structure with possibilities for use of different modules for modelling, equation solving and data presentation. ACSL from MGA Software is a very well known solver, which has been used with MS1 with great success by the author.

SIMULATION OF EXAMPLE SYSTEM

Model verification is always a nagging question, especially by the sceptics. In Figure 7 to 9 are shown some results from closing the clutch. In this example is the engine running at nominal speed when the propeller is clutched in. Figure 7 shows the normal force, F, exerted on the clutch.

As indicated in Figure 8, the difference in relative velocity is so small that it can not even be seen on the plot.

Figure 9 shows the transient torque in the main engine flexible coupling closing of clutch. be constructed and utilised in detailed analysis of large marine engineering. At Norwegian University of Science and Technology in corporation

EXTENDED SYSTEM MODEL
THERMOFLUID SYSTEM

The example system shown in Figure 2 is a mechanics problem. In many instances it might be of interest to perform investigations on the complete diesel engine due to clutching or testing with an extended electrical system. We will therefore indicate how elaborate bond graph models can be constructed and utilised in detailed analysis of large marine engineering. At Norwegian University of Science and Technology in Cupertino with MARINTEK are developing an advanced modelling and simulation environment based on bond graphs and modern software.

The diesel engine is a complicated machine to model since it contains several energy domains in the same system. Even a gas turbine is included in the system. A major part of the diesel engine is thermal coupled to the mechanical piston slider crank mechanism. Thus we need to extend our modelling to thermal systems.

Traditionally, thermal systems have been presented as analogous to electric circuits, usually with temperature analogous to voltage and heat flow analogous to current. With this analogy there are then thermal resistors, capacitors and parallel and series connections and sources, but no thermal inertias. This analogy has proved useful, but there is one major hitch when using it with bond graphs. The product of temperature and heat flow is not power. Heat flow itself has dimensions of power. Therefore, any bond graph in which e and f are not power variables are called pseudo bond graph as long as the basic elements in the pseudo bond graph correctly relate the e, f, p and q variables.

Bond graphs have been used for certain types of thermodynamic systems. Basically, there is little problem when a constant amount of matter is contained within each conceptual lump. However, in many cases a thermodynamic analysis must be made of devices that involve a flow of mass into or out of the device. The procedure that is most practically used in such an analysis is the control volume approach. Mass, as well as heat, work and momentum can flow across the control surface. In modelling control volume systems the concept of power bonds is extended to that of double bonds.Pressure, P, and mass flow rate, $\dot{m}$, are paired on one bond and temperature, T, and energy flow rate, $\dot{E}$, are paired on the other bond. Bonds are then represented in Figure 10.

When using this scheme, elements analogous to the multiport elements of true bond graphs do arise. This double bond concept has been used to model a number of marine engineering systems [7, 8, 9]. In Figure 11 is shown a bond graph model of a one-cylinder diesel engine

The large multiport C element in the middle represents the cylinder where the topside represents the thermal side and the bottom side is the . crank mechanism. This crank mechanism is represented by a MTF which means a modulated transformer since the modulus depends on the crank angle.

The two multiport C elements on either side of the cylinder represent the air and exhaust receiver. The multiport R elements on either side of the cylinder represents the orifice equation for the intake and exhaust valves. The two multiport R elements at the top represent the turbine and compressor which are connected to the turbocharger inertia element. Figure 11 then shows a clear structure of a complicated machine. The lower 1-junction in Figure 11 is equal to left most 1-junction in Figure 5.

As can be seen, a model of a complicated machine containing several energy domains can be displayed in a graphical structure that maps directly into state equations.

This then shows how models of complicated machines from different energy domains can be coupled to form a total system model.

CONCLUSIONS

The formulation of a system model is the crucial step in a simulation study. Accurate simulation of a faulty model is of little use. Yet many system modellers do not have easy ways to assess the effects on their model of adding or removing elements. The paper presents bond graphs as a unified systematic approach to modelling marine engineering system.

Bond graph models render structural representation of complicated systems into a concise and uniform style which is easily communicated between people and to computers. One, if not the most important virtue of using bond graphs is the organisation provided in deriving system equations.

The basic elements used in constructing bond graphs of physical systems are discussed and the construction of a bond graph and derivation of system equations is demonstrated by example.

The most important advantages of the bond graph approach can be summarised as follows:

Causality assignment giving information about what is input and output signals. Causality assignment also reveals algebraic problems in the equation system. These problems can often be solved at an early stage in the modelling process, by simple model adjustments.

Power flow direction is defined rigorously, giving the correct signs when writing the mathematical equations, and assisting the interpretation of the simulation results.

Subsystems are modelled independently and then assembled into an overall system model in much the same way as assembling the actual hardware. This is a valuable support in the physical understanding of the model, which is often hard to obtain when working with the mathematical equations.

Modifications are easy to make, by adding new elements to the bond graph. Following the rules of causality, one can readily see the effects on the number and kind of state variables required even before equations are written.

However, realistic system parameters and final judgement of the simulation results, are still needed from experienced engineers.

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Table 1.
Identification of Variables

Energy	Effort	Flow	Displacement	Momentum
Domain	e	f	$q = \int^{t} f dt$	$q = \int^{t} e dt$
Electrical	Voltage	Current	Charge	Flux Linkage
Mechanical Translation	Force	Velocity	Distance	Impulse
Rotation	Torque	Angular Velocity	Angle	Angular Inputs
Hydrostatic	Pressure	Volume Flow Rate	Volume	Pressure Impulse
Magnetic	Magnetomotive Force	Flux Rate	Flux	(not normally used)
Thermal	Temperature	Entropy Flow	Entropy	(not normally used)
Chemical	Chemical Potential	Rate of Reaction	Advancement of Reaction	(not normally used)

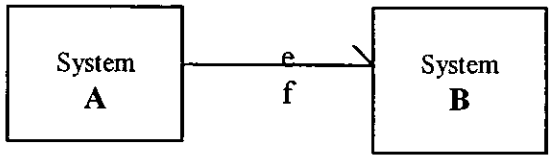


Figure 1. Bond.

Bond graph element	Elemental relations
e, f → R "resistor"	general: $e=\Phi_R(f)$ linear: $e=R\cdot f$
e, f → C "capacitor"	general: $q=\Phi_C(e)$ linear: $q=C\cdot e$
e, f → I "inertia"	general: $p=\Phi_I(f)$ linear: $p=I\cdot f$
Se → e, f "e-source"	$e=e(t)$
Sf → e, f "f-source"	$f=f(t)$
e_1, f_1 → TF → e_2, f_2 "transformer"	$e_1=m\cdot e_2$ $m\cdot f_1=f_2$
e_1, f_1 → GY → e_2, f_2 "gyrator"	$e_1=r\cdot f_2$ $r\cdot f_1=e_2$
f_1, e_1; e_2, f_2; e_3, f_3 "0-junction"	$e_1=e_2=e_3$ $f_1+f_2+f_3=0$
f_1, e_1; e_2, f_2; e_3, f_3 "1-junction"	$f_1=f_2=f_3$ $e_1+e_2-e_3=0$

Figure 2. Bond Graph elements.

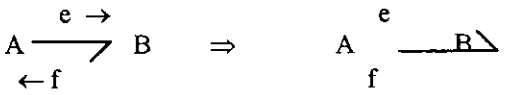


Figure 3. Causality assignment.

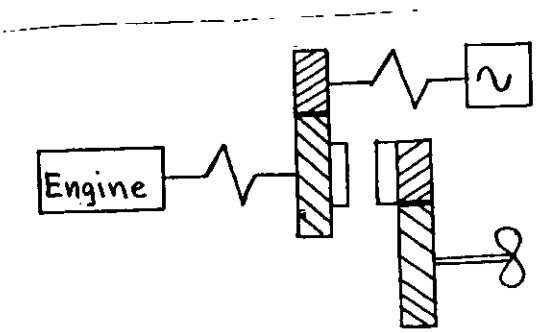


Figure 4. Example system schematic.

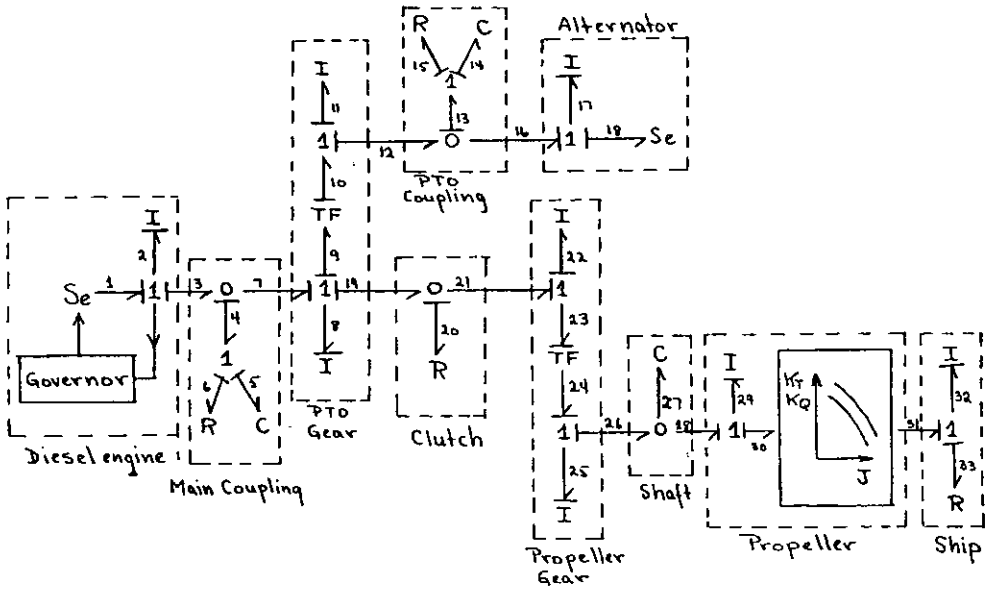


Figure 5. Diesel engine model.

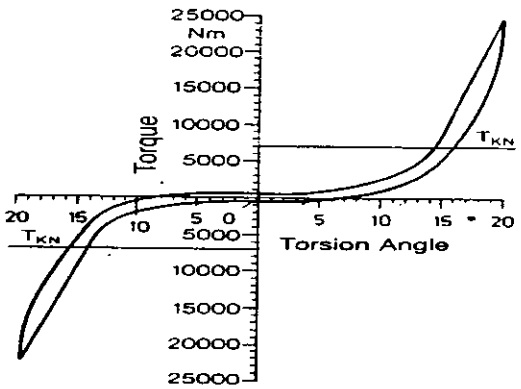


Figure 6. Coupling torque-angular deflection.

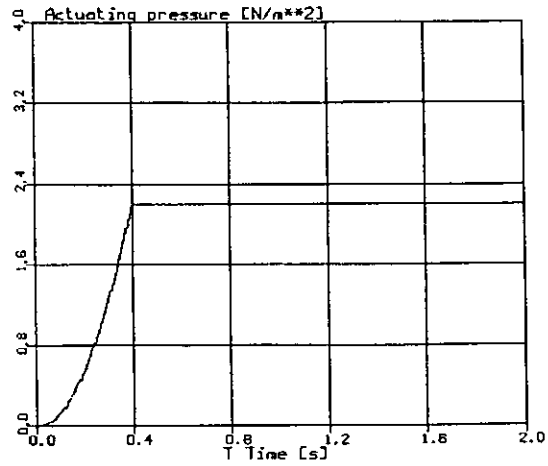


Figure 7. Actuating signal.

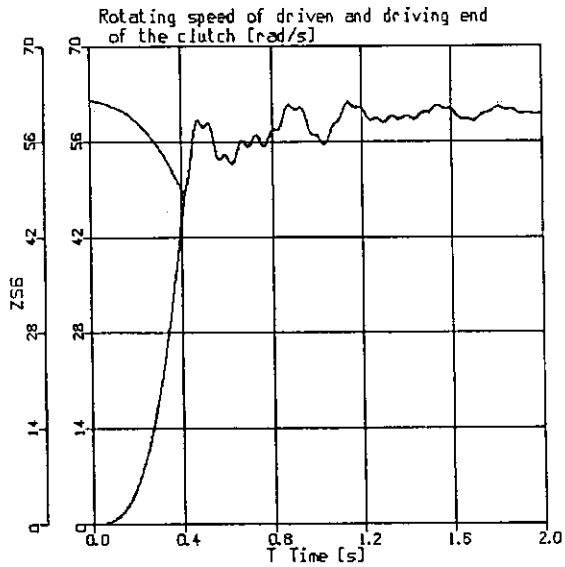


Figure 8. Rotational speeds.

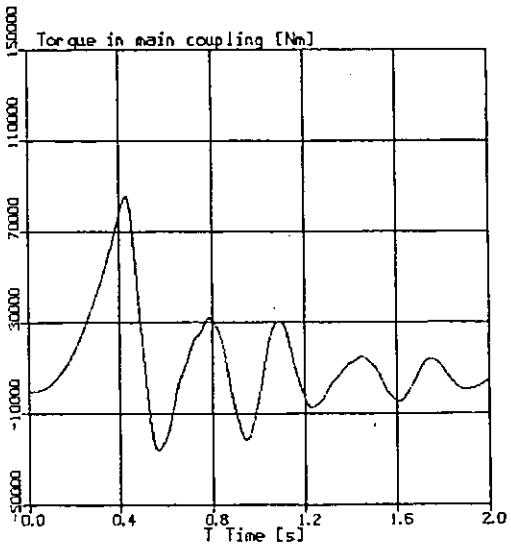


Figure 9. Main coupling torque.

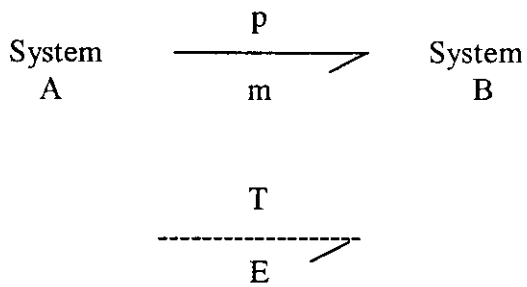


Figure 10. Thermo-fluid bonds.

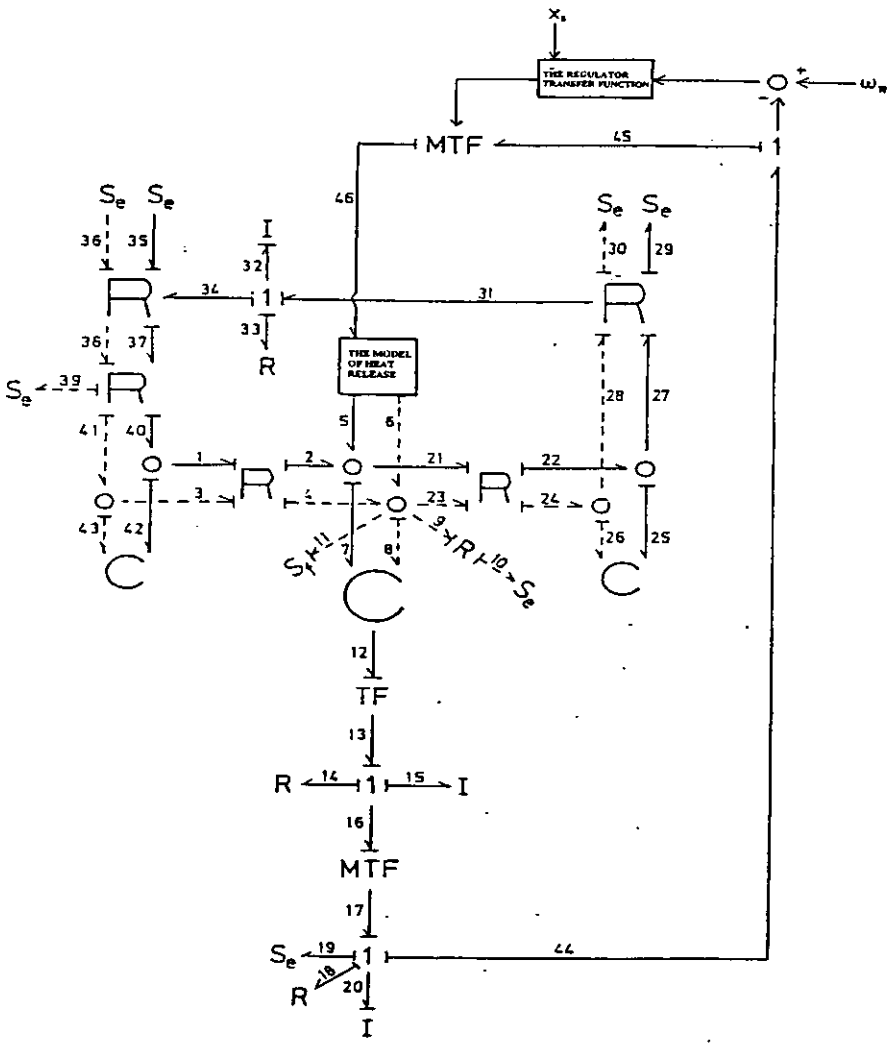


Figure 11. Diesel engine model.

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MATHEMATICAL MODELLING AND SIMULATION OF A WATERJET PROPULSION PLANT

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Hosted by:

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ABSTRACT

In designing the propulsion plant of a high-speed vessel it is important to investigate, in addition to the steady behaviour, the dynamic performance of the system over a wide range of operating conditions.

This lecture presents a mathematical model for the dynamic simulation of an high speed vessel propulsion system propelled by waterjets. The model, developed in a Matlab-Simulink software environment, is structured in modular form, in order to describe the various elements of the system as individuals blocks (hull, prime mover, gear, waterjet, etc.), taking into account their interactions. In this way it is possible to characterise the dynamic behaviour of both the single component and the whole propulsion plant.

INTRODUCTION

Simulation has been successfully used in engineering for many years as a support of the design process. From conceptual design through commissioning and operation, simulation gives the designer insight into the expected performance of the system, providing feedback which would otherwise be unavailable until the system is installed.

Marine propulsion system simulations can be used for a variety of purposes, such as machinery performance analysis, ship performance analysis, maneuvering analysis and machinery control systems development.

The requirements for the simulation are different at each stage of the design process.

The development of a simulation technique is influenced by the criteria which are used to define the simulation requirements and, in turn, by the stage of the design process to which the simulation code is intended for.
The simulation models developed by the author are intended to be used both:
in the initial phase of the design process to optimize the choice of the propulsion system components;
in subsequent phases to predict the propulsion plant behavior during critical situations or when changing the normal working conditions.
In the initial phase of the design process the simulation may be used as a useful tool to facilitate the correct matching between the prime mover (diesel or gas turbine) and the propulsor (waterjet or propeller) in the whole range of operating conditions. Of particular interest are the choice of gear ratio and the definition of the pitch-rpm law for controllable pitch propellers, at steady-state design conditions, but especially at off-design and transient conditions.
In subsequent phases of the design process the simulation may be used to evaluate maneuvering situations which can be encountered in service and/or situations the propulsion plant must comply with (slam start, crash stop, etc.). Furthermore may be used to predict the influence of adverse environmental conditions (heavy sea, wind), leading to an increase of the hull resistance, on the ship propulsion plant behaviour.

The modeling approach to be effective in the initial stage of the design is usually different from the approach that can be used in a detailed phase.
In the initial phase of the design only a few data about the system are available, so the simulation program will be based on few simplified equations or performance tables. This kind of simulation is usually sufficient to 'capture' the dynamic of the whole system but not of each single component.
In a detailed phase of the design, when more accurate data are available, a much more realistic program, based on time dependent equations, can be developed. With this kind of simulation the dynamic behaviour of each single component of the system can be represented.
In designing a simulation program we basically have to consider the types of phenomena we want to reproduce (i.e. moderate transients, hard transients) and the results we want to obtain (behaviour of the whole system, behaviour of the system and of each component).
The equations used to represent the behavior of the system incorporate a number of simplifying assumptions that should be related to the objective of the study. Nevertheless it is important that these assumptions do not change the physics of the phenomenon itself. For instance, to reproduce an actually transient phenomenon we must use time dependent equations so the performance table of the engine is not suitable in this case.

The evaluation of a simulation program is usually based on two, commonly adopted, simulation requirements: **fidelity** (realism of the prediction) and **performance** (time to produce the results).
They are in conflict each other. High fidelity is usually related to highly complex models and low performance simulators (in the sense of long time to prepare the model and to obtain the results). An example regarding a propulsion chain component is the engine, that can be modeled in two different ways, as a performance table (high performance, low fidelity) or by several equations (low performance, high fidelity).

For very complex systems, made up by several elements, it is very difficult to develop models with the same level of fidelity for all the components of the system (for instance it is actually unfeasible to evaluate the resistance of the ship from basic physical equations, it is usually represented by a resistance curve).

The lecture describes a dynamic simulation model, developed in a dedicated software environment, for the performance prediction of a ship propulsion system.
The different elements of the propulsion plant, such as hull, prime mover, waterjet, etc., have been modeled using blocks, whose behavior is described by tables, mathematical operators and algebraic or differential equations.
A library of blocks, representing the propulsion plant elements (i.e. engine, governor, etc.) modeled with different degree of accuracy, is already available. So, according to the objective of the simulation, it is possible to use the most suitable blocks.
The model may be used, as said before, to optimize the choice of propulsion system components at steady-state design conditions, but especially to analyze the system response at off-design and transient conditions.
The adopted modular arrangement allows to apply the simulation code to different plant configurations.
The model requires, as input data, a set of information regarding the hull type and geometry, the prime mover performance and operating range, the propeller characteristics. In turn the model gives, as output values, the time histories of all the considered variables (such as, for instance, ship speed, shaft r.p.m. and torque, thrust, engine power, fuel consumption and exhaust gases flow rate, etc.) which define the dynamic behavior of the system.
The developed computer simulation code has been already used to analyze the dynamic behavior of the propulsion system of different ship types: a trawler with a FP propeller, a navy ship with CP propellers and a pleasure craft equipped with waterjets. Presently a model of a large car-passenger ferry equipped with four medium speed diesel engines and two CP propellers is under development.
A recent improvement is the simulation of the propulsion plant behavior of the ship in a seaway. This feature is implemented taking into account two effects of the ship motion in head sea: the added resistance of the hull and the fluctuation of propeller thrust and torque due to propeller emergence from the water.

PROPULSION SYSTEM SIMULATION

The application of simulation techniques to marine propulsion systems represents a research field developed at the University of Genoa since some years, which has already allowed to obtain interesting results, as shown in [1]-[7].
The software used for the propulsion plant simulation is SIMULINK, a powerful toolbox of MATLAB. Within this software environment it is possible to simulate the dynamic behaviour of a system using several types of pre-defined or custom blocks connected together in a graphical scheme.
The differential equations describing the process can be solved by numerical integration in the time domain by means of integration blocks with proper initial conditions.
Among the various algorithms available in SIMULINK [8] for the solution of differential equations, the Euler and Gear methods have been adopted, giving good performances with the present model,

characterised by many non-linear blocks (look-up tables, rate and level limiters, delay blocks) and by elements with time constants very different from each other, like those related to ship mass and engine inertia ("stiff system"). Because the MATLAB is a interpreter language, in order to save computational time, some parts of the code are written in C^{++} and compiled and linked to MATLAB as Mex files. All of the physical variables of interest are stored in memory (Work-Space) for monitoring. Some post-processing programs were developed for plotting the simulation results or printing all the monitored variables as functions of time, or in correspondence of a particular simulation step.
In its original version the model was applied to simulate the propulsion system of a naval vessel, consisting of a twin shaft arrangement with controllable pitch propellers, driven, trough reduction gears, by turbo charged medium-speed diesel engines.
The main elements of the propulsion plant, modelled as separated blocks, were: the diesel engine with its turbocharging system, the engine governor, the hull, the controllable pitch propellers and the telegraph.
In the modified version of the simulation model, here described, the controllable pitch propeller block has been removed and replaced by a new block, representing the waterjet behaviour, as shown in fig. 1. However the general structure of the simulation scheme has been maintained so that, for instance, the main differential equations are the same as before and the waterjet block, even if very different from the propeller block, may be considered, at a first sight, as a black box with the same input and output variables.

MODEL DESCRIPTION

The considered propulsion plant consists of a twin shaft arrangement with medium size waterjets, driven, through reduction gears, by turbo-charged high speed diesel engines.
The main elements of the propulsion plant, modeled as separated blocks, are the diesel engine, the governor, the hull, the waterjet, the telegraph, and the shaft line. The complete model is shown in figure 1.

THE WATERJET BLOCK

General

In a waterjet propulsion system thrust is produced by accelerating a mass of water. In all waterjet-propelled vehicles the water enters through the inlet system, energy is added to the flow by the pump and the water exhaust at the exit nozzle at high velocity. The change in momentum of the water entering and leaving the waterjet system produces the thrust.
The performance prediction technique adopted here comes from ITTC'87 Waterjet test procedures [9].

The ingested flow can be defined by 3 different velocities: the average velocity, the momentum velocity, and the energy velocity, calculated as follows.
Average velocity V_a:

$$\frac{V_a}{V} = \frac{\delta}{Y} \int_0^{Y/\delta} \left(\frac{u}{V}\right) d\left(\frac{y}{\delta}\right)$$

Momentum velocity V_m:

$$\frac{V_m}{V} = \frac{V}{V_a}\frac{\delta}{Y}\int_0^{Y/\delta}\left(\frac{u}{V}\right)^2 d\left(\frac{y}{\delta}\right)$$

Energy velocity V*:

$$\left(\frac{V^*}{V}\right)^2 = \frac{V}{V_a}\frac{\delta}{Y}\int_0^{Y/\delta}\left(\frac{u}{V}\right)^3 d\left(\frac{y}{\delta}\right)$$

Where:

V ship speed
u local velocity within the boundary layer
Y distance below hull bottom
δ boundary layer thickness

In the boundary layer it is commonly assumed a two dimensional velocity profile:

$$\frac{u}{V} = \left(\frac{y}{\delta}\right)^{1/n} \qquad \frac{y}{\delta} \le 1 \; ; \; y = Y$$

$$\frac{u}{V} = 1 \qquad \frac{y}{\delta} > 1 \; ; \; y > Y$$

where n is a convenient exponent.

Thrust

Neglecting the difference between V_a (approach or average velocity) and V_m (momentum velocity), the thrust T may be calculated from the change in momentum:

$$T = \rho \cdot Q \cdot (V_j - V_a) \qquad (1)$$

where $Q = V_j A_j$ is the pump flow rate, ρ is the water mass density, V_j, A_j are the jet velocity and the jet area respectively and V_a is the effective velocity of approach (fig. 3), evaluated at a station in front of the intake ramp.

The approach velocity V_a is defined as the average velocity in the boundary layer, with a more concise notation it is possible to write:

$$Va = \frac{\int_0^{y_0} V_z(z)dz}{y_0} \qquad (2)$$

where:

$$V_z = V \quad \text{if} \quad z > \delta$$

$$V_z = V\left(\frac{z}{\delta}\right)^{1/n} \quad \text{if} \; z \le \delta \qquad (3)$$

δ is the boundary layer thickness
y_0 is the depth below the hull to which the flow is ingested and defines the idealised streamline dividing the flow entering the waterjet inlet and the flow around it.

The boundary layer thickness can be evaluated by the Prandtl formula :

$$\delta = 0.37 * L_x * Re_x^{-\frac{1}{5}} \qquad (4)$$

L_x being the distance from the bow and Re the Reynolds number.

The integration to evaluate V_a requires the knowledge of the upper limit y_0 that may be calculated from the continuity equation $Q = V_a * y_0 * d$, where d represents the effective width of the approaching streamtube. The literature reports different opinions about the value of this width, ranging from 1 to 1.3 times the inlet opening h_i. A reasonable value for the model has been found to be 1.2.

Based on the above relations and according to [9],[10] a thrust deduction fraction t and a wake fraction w may be defined in analogy with the screw propulsion coefficients, R being the ship resistance at ship speed V:

$$t = 1 - \frac{R}{T} \qquad w = 1 - \frac{V_a}{V} \qquad (5)$$

The simplification $V_a=V_m=V^*$ is justified because the three values are close together, in fact, in the case of $Y/\delta \leq 1$ (or $y_0/\delta \leq 1$, depending on the notation):

$$\frac{V_a}{V} = \frac{\delta}{Y}\int_0^{Y/\delta}\left(\frac{u}{V}\right)d\left(\frac{y}{\delta}\right) = \frac{n+1}{n}\left(\frac{y}{\delta}\right)^{1/n}$$

$$\frac{V_m}{V} = \frac{V}{V_a}\frac{\delta}{Y}\int_0^{Y/\delta}\left(\frac{u}{V}\right)^2 d\left(\frac{y}{\delta}\right) = \frac{n+1}{n+2}\left(\frac{y}{\delta}\right)^{1/n}$$

$$\left(\frac{V^*}{V}\right)^2 = \frac{V}{V_a}\frac{\delta}{Y}\int_0^{Y/\delta}\left(\frac{u}{V}\right)^3 d\left(\frac{y}{\delta}\right) = \frac{n+1}{n+3}\left(\frac{y}{\delta}\right)^{2/n} \Rightarrow \left(\frac{V^*}{V}\right) = \left(\frac{n+1}{n+3}\right)^{1/2}\left(\frac{y}{\delta}\right)^{1/n}$$

$$\frac{V_a}{V_m} = \frac{n(n+2)}{(n+1)(n+1)}$$

$$\frac{V_a}{V^*} = \frac{n}{(n+1)}\frac{(n+3)^{1/2}}{(n+1)^{1/2}}$$

for n=9 $\frac{V_a}{V_m} = 0.99$ and $\frac{V_a}{V^*} = 0.986$ that means an error of about 1%.

In the more realistic case $y_0/\delta \geq 1$ (i.e. the approaching streamline greater that the boundary layer thickness) the error is smaller.

Total Head

The total head rise H of the pump is equal to the head needed to produce the jet and to overcome the ducting losses minus the recovered head of the incoming flow, according to [9]:

$$H = V_j^2/(2g\eta_N) - (1-k)\,(V^*)^2/2g + \Delta h$$

neglecting the difference between V_a and V*, it is possible to obtain

$$H = V_j^2/(2g\eta_N) - (1-k)\, V_a^2/2g + \Delta h \qquad (6)$$

with η_N nozzle efficiency, k loss coefficient of inlet ducting and Δh elevation of jet.

The pump head can be expressed as the sum of the following contributions: the kinetic energy of the outlet flow, minus the kinetic energy of the inlet flow, plus the inlet drag, plus the elevation of the waterjet from the waterplane.

$$H = \frac{1}{2g}V_j^2 - \frac{1}{2g}V_a^2 + \text{Drag} + \Delta h$$

The energy of the flow should be reduced due to the nozzle efficiency; the inlet drag is commonly expressed as a function of the kinetic energy. The head results:

$$H = \frac{1}{\eta_N}\frac{1}{2g}V_j^2 - \frac{1}{2g}V_a^2 + k\frac{1}{2g}V_a^2 + \Delta h = \frac{1}{\eta_N}\frac{1}{2g}V_j^2 - (1-k)\frac{1}{2g}V_a^2 + \Delta h$$

that is the formula (6).

Pump Flow Rate

In order to apply the thrust equation (1) it is necessary to calculate the pump flow rate Q. For a given pump the operational point follows from the intersection of the head capacity curve of the pump with the head curve required by the system.

If the nozzle area is known, it is a simple matter to evaluate the jet area A_j; then, taking into account that $V_j = Q/A_j$, it is possible to express the total head rise required by the system, given by eq. 6, as function of Q:

$$H = \frac{1}{\eta_N}\frac{1}{2g}\frac{Q^2}{A_j^2} - (1-k)\frac{1}{2g}V_a^2 + \Delta h \qquad (7)$$

The head curve expressed by the above equation, once V_a is given and k is evaluated, has to be compared with the characteristic head-capacity curve of the pump corresponding to the given rotational speed N_p. This curve can be obtained from the pump manufacturer. Alternatively can be evaluated, as in this case, by using analytical expressions able to fit the pump experimental data and taking into account the similarity laws.

The head-capacity curve for a given revolution N_x can be expressed as:

$$H_x = aQ_x^2 + bQ_x + c$$

with N_x reference speed of rotation and a, b, c constants to be determined.

The similarity law states that:

n_x, D_x, H_x, Q_x, P_x, revolution, diameter, head, flow and power at trial condition

n, D, H, Q, P, revolution, diameter, head, flow and power for a different condition

$$x = \frac{n}{n_x} \qquad \xi = \frac{D}{D_x}$$

$$\eta = \eta_x \qquad H = H_x \cdot x^2 \cdot \xi^2 \qquad Q = Q_x \cdot x \cdot \xi^3 \qquad P = P_x \cdot x^3 \cdot \xi^5$$

$$\xi = 1 \Rightarrow \quad H = H_x \cdot x^2 = \left(aQ_x^2 + bQ_x + c\right)\cdot\left(\frac{n}{n_x}\right)^2 = \left[a\left(Q\cdot\frac{n_x}{n}\right)^2 + b\left(Q\cdot\frac{n_x}{n}\right) + c\right]\cdot\left(\frac{n}{n_x}\right)^2 \qquad (8)$$

By comparing eq. 7 and eq. 8, a quadratic equation is obtained, whose closed form solution gives the required value of the pump flow rate Q.

The pump flow rate Q is evaluated by the model in the 'waterjet block'.
The SIMULINK graphical representation of this block is shown in fig. 2, which is an expansion of the same block appearing in compressed form in fig. 1. The input variables entering the waterjet block are the effective velocity of approach V_a and the pump rotational speed N_p. The output variables are the thrust T of the waterjet, the torque Q_p required by the pump and the mass flow rate Q.

Power

Once determined the pump flow rate, the waterjet thrust may be easily calculated by using eq. 1. The pump torque may be obtained from the power demand at the prime mover:

$$P_B = \frac{\rho g Q H}{\eta_P \eta_r \eta_m} \qquad (9)$$

with η_P pump efficiency, η_r relative rotative pump efficiency and η_m mechanical efficiency [10]. If the pump rotational speed N_p is expressed in r.p.m. than the pump torque is given by:

$$Q_P = \frac{60 P_B}{2\pi N_P} \qquad (10)$$

This torque is compared, in the simulation scheme, with the engine torque, taking into account the presence of the reduction gear.

HULL MODEL

The hull model consists of the dynamic equation of the hull:

$$(M_s + M_{s\,ad})\, dV/dt = Th_P - Th_s \qquad (11)$$

where:
M_s, $M_{s\,ad}$ = mass and added mass of the hull
Th_P = thrust provided by the propeller
Th_s = thrust required by the hull

The hull block is presented in fig.4.
The added mass is considered as a constant. The resistance of the hull is modeled with a 1-D table as a function of the speed: the towing tank resistance curve R for V>0, a function of the forward resistance for V<0.
The thrust required by the hull is :

$$Th_s = \frac{R}{1-t} \qquad (12)$$

where t is the thrust deduction factor.
The input datum of the hull block (Fig. 1) is the ship speed V, coming from the inherent model, and the outputs are the thrust required by the hull Th_s and the advance velocity V_a:

$$V_a = (1-w)V \qquad (13)$$

"TELEGRAPH" MODEL

A demand lever ("telegraph") placed on the command bridge on-board selects the required speed for the engine. Each telegraph position is related to engine r.p.m. requests; the signal is then transmitted to the engine governor.
The telegraph block, shown in figure 5, represents a more general device that can also be used to control CPP by the rpm-pitch combinator law.

ENGINE MODEL

Basic Approach

In literature, various approaches are proposed for the dynamic simulation of marine diesel engines; an extensive bibliography on this topic is reported in [7].
In the most simple schemes, the engine performance is calculated by means of diagrams or empirically derived equations (i.e. engine torque or power as functions of rotational speed and fuel consumption). This approach is quite simple and is suitable for a rough simulation of the engine dynamic behavior in a short time. It does not allow a careful description of thermal and mechanical stresses induced in critical elements of the power plant.
More complicated schemes include both a description of the main thermodynamic in-cylinder phenomena and a realistic turbocharger simulation. Some authors adopted an ideal dual combustion cycle scheme, while in other papers a more sophisticated 'single-zone' approach is used. These models are characterized by a longer computational time as compared with the previous ones, but still acceptable for real time simulation.
A significant improvement in the prediction of in-cylinder phenomena is represented by the 'two-zone' approach, which provides distinct calculations for the burning and non-burning zone. This combustion model, unlike the former ones, allows to evaluate engine emissions by means of semi-empirical equations, with only a small calculation time increase
In this work, a 'two-zone' actual cycle cylinder scheme [11] is used. The model is briefly described in the following section.

Model Description

The overall scheme of the diesel engine plant, including the turbocharger group, is shown in fig. 6.
The model consists of the following modules:
1) cylinder
2) intercooler and air receiver
3) exhaust gas receiver
4) compressor

5) turbine
6) turbocharger dynamics.
Each block includes the algebraic and differential equations describing the subsystem behavior. A filling and emptying method is employed for the simulation of each block of the engine. In fig. 6 only the main mechanical and thermodynamic variables exchanged between the blocks are reported.
Basic equations
For the fluid model, the real gas equation:

$$pV_0 = MRT \qquad (6)$$

is used.
As proposed in Benson and Whitehouse (see paper [7]), specific internal energy and enthalpy are assumed to be functions of both temperature and fluid composition.
The fluid mass and energy stored in each engine volume are calculated, respectively, with the continuity equation:

$$\frac{d\rho}{dt} = \frac{1}{V_0}(m_i - m_o) \qquad (7)$$

and with the energy equation:

$$\frac{d(\rho U)}{dt} = \frac{1}{V_0}(m_i H_i - m_o H_o + m_f H_f + \Phi - P) \qquad (8)$$

The wall temperature of engine components' is determined by the energy equation:

$$\frac{dT_w}{dt} = \frac{1}{C_w M_w}(\Phi_i - \Phi_o) \qquad (9)$$

The dynamic equation:

$$\frac{d\omega}{dt} = \frac{1}{J}\Sigma Q \qquad (10)$$

holds for both crank- and turbocharger - shaft.

Cylinder

A simplified scheme of the cylinder module is shown in fig. 7, where only the main input data required are reported.
Crank angle θ is the independent variable.
From the crankshaft angular velocity $\omega$and crank angle increment $d\theta$, the time interval dt is determined.
Cylinder calculation process is pertinent only to the period when both the inlet and exhaust valves are closed (closed cylinder power process). During the gas exchange process (inlet valve or/and exhaust valve open), a constant pressure in the cylinder is assumed.
The following equation:

$$x = r\left[1+\frac{1}{r}-\left(\frac{l^2}{r^2}-sen^2\vartheta\right)^{1/2}-\cos\vartheta\right] \qquad (11)$$

gives the piston displacement from the TDC as calculated from the crank angle value.

The inlet air mass M_a in the cylinder is evaluated by means of the equation:

$$M_a = \rho_i V_0 \eta_v \qquad (12)$$

where ρ_i is the fluid density in the air receiver.
The air compression process is described with the energy equation (8), where, in this case, the inlet and outlet masses (m_i and m_o) are equal to zero.
In equation (8), the convective gas-cylinder wall heat transfer is calculated with:

$$dQ_w = h A_w (T_g - T_w) dt \qquad (13)$$

where the heat transfer coefficient h is given by:

$$h = K_{h1} V_0^{-0.06} p_{cy}^{0.8} T_{cy}^{-0.4} \left(v_m + K_{h2}\right)^{0.8} \qquad (14)$$

In equation (13), the heat transfer area A_w is given, as function of piston position x, by:

$$A_w = \pi B x + A_{co} \qquad (15)$$

The ignition delay angle (θ_{id}), is given by :

$$\vartheta_{id} = K_{id}^{n}\left(p_{cy}\right)^{-0.388}\left(\frac{40}{CN}\right)^{0.69}\exp\left(\frac{4.644}{T_{cy}}\right) \qquad (16)$$

When the combustion process starts, according to two-zone combustion calculation scheme (fig. 8), the cylinder volume is split into two parts, a burning zone and a non burning one.

At each calculation step $d\theta$, the heat released dQ_f by the combustion process, is given by:

$$dQ_f = dM_b H_f \qquad (17)$$

where the burned fuel mass dM_b is evaluated with:

$$dM_b = M_f dx_b \qquad (18)$$

M_f being the total mass of fuel injected in the cycle.
In equation (18) the fuel fraction burned dx_b in the calculation step $d\theta$, is given by the Wiebe equation (see paper [7]):

$$dx_b = 1 - \exp\left[-K_a\left(\frac{\vartheta - \vartheta_{ign}}{\Delta\vartheta}\right)^{K_m+1}\right] \qquad (19)$$

In this equation the combustion angle length $\Delta\theta$ is determined by the mass flow rate across the injector holes:

$$m_f = K_{in}\,\rho_f\left[2\frac{\left(p_{in} - p_{cy}\right)}{\rho_f}\right]^{0.5}\frac{\pi d^2}{4} \qquad (20)$$

where p_{in} is the mean injection pressure, assumed to be a function of the rotational speed and of the fuel mass injected per cycle. Always in equation (20), p_{cy} is the cylinder pressure at the start of the injection process.

In the energy equation (8), applied to the burning zone, the inlet air mass term dM_i, from the non burning zone, is evaluated as function of the fuel mass burned dM_b during the current calculation step $d\theta$:

$$dM_i = dM_b\,\alpha_s\,K_{exc} \qquad (21)$$

where the air excess term K_{exc} is imposed on the basis of data reported in literature (see paper [7]).

In the burning zone, again the heat transfer in the cylinder is computed by means of the Annand equation:

$$dQ_w = A_b\,[K_{con}\,k/B\,Re^{Kre}\,(T_g - T_w) + K_{ra}\,(T_g^4 - T_w^4)]\,dt \qquad (22)$$

which includes the effect of both convection and radiation.
In the non burning zone, the outlet mass term dM_o of eq. (8) is of course equal to the inlet mass term in eq. (21) pertinent to the same step $d\theta$.
Always in the same zone, the convective heat transfer is evaluated by eq. (13).
The temperature of the cylinder wall, supposed to have an infinite thermal conductivity, is determined by eq. (9).
The heat transfer between wall and coolant fluid is given by:

$$dQ_{coo} = h_{coo}\,A_{coo}\,(T_w - T_{coo})\,dt \qquad (23)$$

where the heat transfer coefficient h_{coo} is assumed constant.
The piston power P in the energy equation (8), for each calculation step $d\theta$, is computed by the work equation:

$$dW = p_{cy}\,dV_0 \qquad (24)$$

where: p_{cy} is the mean cylinder pressure (pressure balance is assumed between burning and non burning zone) during the step $d\theta$, and dV_0 is the cylinder volume change in the same step.

From the application of the energy equation (8) to the burning and non burning zones, the fluid temperature in each zone is determined. Hence, by means of gas eq. (6), applied to burning and non burning zones, and with the constraint:

$$V_{0T} = V_{0b} + V_{0ub} \qquad (25)$$

the cylinder pressure (p_{cy} in equation 24) is determined at each crank angle θ.
The new pressure value is compared with the pressure value employed in equation (24); a trial and error procedure is used to seek the final p_{cy} value.
NO and soot emission
During the combustion process, the changes in fluid composition are computed. Dissociation effects are neglected.
The two zone calculation scheme permits the use of correlation for NO and soot prediction during the combustion process. In the concerned model, correlation for NO and soot formation and for soot destruction are applied to the burning zone.
For NO prediction:

$$\frac{d(NO)}{dt} = K_{NO}\{O_2\}^{0.5}\{N_2\} \exp(-26.278/RT + 24.088) \qquad (26)$$

for soot formation:

$$\frac{dM_{soot\ fo}}{dt} = K_{fo}\, M_f\, p^{0.5} \exp\left(-K_{ef}/RT\right) \qquad (27)$$

and for soot oxidation:

$$\frac{dM_{soot\ ox}}{dt} = K_{ox} M_{soot} \left(\frac{p_{O2}}{p_{cy}}\right) p_{cy}^{1.8} \exp(-K_{e\,ox}/RT) \qquad (28)$$

The effective soot mass produced in a cycle is given by the eq:

$$M_{soot} = M_{soot\ fo} - M_{soot\ ox} \qquad (29)$$

In order to make the model suitable for simulating the engine over a wide range of operating conditions, numerical results of equations (26) and (29) have been compared with the output of a phenomenological multi-zone combustion model.
A 2D matrix of correction factors, having engine rotational speed and b.m.e.p. as input variables, has been introduced to match the output of the two-zone model to the numerical results of the multi-zone model.

Intercooler And Air Receiver

These components are grouped into a single block of fig. 6.
The air cooling effect is simply computed by:

$$T_o = T_i - E\,(T_i - T_{coo}) \qquad (30)$$

where the heat exchanger efficiency E is kept constant.
With equations (7) and (8), the mass and energy accumulation in the control volume is considered.
The pressure drop of the whole set is evaluated as a fraction of the inlet pressure.
Exhaust gas receiver
This block, named «Gas Receiver» in fig. 6, is treated in a very similar way to the air receiver module, i.e. by applying the continuity (7) and energy (8) equations.
The effect of the blow-down through the exhaust valve is considered, by assuming the average inlet temperature to be:

$$T_{i\,ex} = T_i\,[1 - ((k_g - 1)/k_g)\,(1 - p_o/p_i)] \qquad (31)$$

Compressor

A steady state compressor map given by the Manufacturer is employed in 2D matrix form.
Compressor pressure ratio $\beta = p_o/p_i$ and isentropic efficiency η_c are evaluated as a function of the corrected volume flow rate:

$$V'_{Cc} = V'_C \sqrt{\frac{T_{ref}}{T_a}} \qquad (32)$$

and corrected rotational speed:

$$n_{Cc} = n_{TC} \sqrt{\frac{T_{ref}}{T_a}} \qquad (33)$$

Turbine

The turbine block is very similar to that used for the compressor.
A turbine steady state map is given in 2D, flow rate and efficiency being a function of the expansion ratio $\varepsilon_T = p_i/p_o$ and of the kinematic ratio u_t/c_o.
Engine torque
The torque delivered by the engine shaft is calculated in terms of brake mean effective pressure (b.m.e.p.):

$$(b.m.e.p.) = (i.m.e.p.) + (p.m.e.p.) - (f.m.e.p.) \qquad (34)$$

where: i.m.e.p. is the gross indicated mean effective pressure;
p.m.e.p. is the pumping mean effective pressure;
f.m.e.p. is the mechanical friction mean effective pressure.
The f.m.e.p. is evaluated on the basis of test data by the Manufacturer. The brake torque is calculated in the 'torque' block of fig. 6 by:

$$Q_E = (b.m.e.p.)\,V_0 \frac{1}{4\pi} \qquad (35)$$

Shaft Dynamics

Engine and turbocharger shaft dynamics is determined by time integration of eq. (10).

SHAFT MODEL

Engine and propeller are physically joined together by the shaftline. To properly account for the mutual interaction between the elements of the propulsion chain, a model for the dynamic behavior of the shaftline has been developed.

The shaftline model links together the engine sub model and the propeller submodel into a cohesive and physically consistent system model. In fact, during transients, displacements, speed and acceleration may not be instantaneously the same for the engine and the propeller, due to the compliance of the shaft. In the common hypothesis that the shaft has no inertia, the shaftline undergoes a constant torque.

The physical model of the transmission system is shown in figure 9. The inertia of the engine and the propeller is lumped on one mass each.

Denoting:

$Q_1(t)$	engine torque
$Q_2(t)$	propeller torque
$T(t)$	shaft torque
$\theta(t)$	angular position
$\omega(t)$	angular velocity

the equation of motion of the two masses are:

$$\begin{aligned} J_1 \ddot{\vartheta}_1 &= Q_1 - T \\ J_2 \ddot{\vartheta}_2 &= T - Q_2 \end{aligned} \qquad (36)$$

where:

$$\omega_1(t) = \left(\frac{d\theta_1}{dt}\right) = \dot{\vartheta}_1 \qquad (37)$$

$$\omega_2(t) = \left(\frac{d\vartheta_2}{dt}\right) = \dot{\vartheta}_2 \qquad (38)$$

$$\dot{\vartheta} = \left(\dot{\vartheta}_1 - \dot{\vartheta}_2\right) \qquad (39)$$

$$T = K\vartheta + C\dot{\vartheta} \qquad (40)$$

$$\vartheta = \vartheta_1 - \vartheta_2 \qquad (41)$$

The torque in the shaft is assumed to be the sum of two components, one due to twist, proportional to the relative displacement of the two ends (θ_1-θ_2), and the other due to the hysteresis of the material, proportional to the shaft speed (ω_1-ω_2).
The complete system of differential equations governing the shaft speed and torque is the following:

$$\begin{aligned} J_1 \ddot{\vartheta}_1 + C\left(\dot{\vartheta}_1 - \dot{\vartheta}_2\right) + K\left(\vartheta_1 - \vartheta_2\right) &= Q_1 \\ J_2 \ddot{\vartheta}_2 - C\left(\dot{\vartheta}_1 - \dot{\vartheta}_2\right) - K\left(\vartheta_1 - \vartheta_2\right) &= -Q_2 \end{aligned} \qquad (42)$$

The SIMULINK® numerical model of the shaft in shown in figure 10. The model is arranged to solve the equations in the following form:

$$\begin{aligned} \ddot{\vartheta}_1 &= \frac{Q_1}{J_1} - \frac{C}{J_1}\left(\dot{\vartheta}_1 - \dot{\vartheta}_2\right) - \frac{K}{J_1}\left(\vartheta_1 - \vartheta_2\right) \\ \ddot{\vartheta}_2 &= \frac{C}{J_2}\left(\dot{\vartheta}_1 - \dot{\vartheta}_2\right) + \frac{K}{J_2}\left(\vartheta_1 - \vartheta_2\right) - \frac{Q_2}{J_2} \end{aligned} \qquad (43)$$

The shaft block has two inputs: Q_1 (engine torque) and Q_2 (propeller torque) which come from the other blocks of the overall model and one output ω_s (shaft speed). Inside the block other quantities are calculated such as the shaft torque, shaft twist and shaft acceleration.
Unfortunately the differential equations, as posed before, fall in a category defined as 'stiff'; this means that the solution contains different parts with very different time scales. The solution of stiff differential equations poses a lot of numerical difficulties [12] that can be partially overcome by sophisticated integration algorithms available in Matlab.

GOVERNOR

The governor senses the engine speed and controls the fuel flow in order to maintain steady state speed. The inputs to the governor are the requested speed $\mathbf{n}_{req}$ and the actual engine speed $\mathbf{n}_E$, the output is the fuel mass flow $\mathbf{m}_f$ injected in the cylinder. The requested speed is normally set by the operator by means of the so-called "telegraph lever" that also selects the propeller pitch value.

In the present study we started with a special type of PID block, named "anti-windup" selected in the SIMULINK® library. The anti-windup technique allows to limit the problem caused by the integrative action during the period in which the regulator output is in a saturation state. During the saturation the anti-windup system decreases the integrative action.
A over-limit control was added in order to avoid overloading the engine and to reduce the exhaust gas pollution. This device is present also in some mechanical-hydraulic governors. When the fuel flow decided by the PID exceeds the maximum admissible one (derived by the Limit Power Curve of the engine) at a particular speed rate, the output of the governor is limited to this value. Obviously the speed cannot be maintained during the operation of such device. As shown in the diagram of fig.11 the engine work points never exceed the Limit Power Curve.

CONCLUSIONS

The model has been applied to study the behaviour of the propulsion system of a 200 ton monohull equipped with two waterjets driven by two high speed diesel engines. Typical ship manoeuvres and some off design conditions has been simulated. In particular the presented results refer to a stop-full ahead manoeuvre obtained by imposing a certain law to the engine control system. Figure 11 shows the time histories of the main variables of the waterjet plant (wake fraction w, thrust deduction t, streamtube depth y_0, boundary layer thickness δ, ship speed V, waterjet thrust Th, shaft pump speed Np, shaft power Pb). Figure 12 shows some of the calculated variables relative to the diesel engine (maximum cylinder temperature, maximum cylinder pressure, air/fuel ratio, turbocharger speed); for these graphs the quantities have been referred to the nominal values.

The presented graphs show the regular behaviour of the curves representing the time variation of the considered parameters; this confirms the general good working of the model and particularly the stability of the numerical integration process. Furthermore the asymptotic solution of the transient simulation is in good agreement with the available experimental data measured in steady state conditions.

Finally figure 13 shows the variation of the overall propulsion coefficient evaluated by two different formulations:

$$OPC = \frac{R \bullet V}{P_b}$$

$$OPC1 = \frac{T \bullet (1-t) \bullet V}{P_b}$$

During transient situations the two definitions yield to different results because the instantaneous values of R and T(1-t) are different. Obviously when the ship reach the steady state condition the two definition give the same results.

Is the author's believe, encouraged by the recent on site validation of the new simulation model of a ferry, that propulsion plant simulation can be an useful and reliable aid to the propulsion plant design process.

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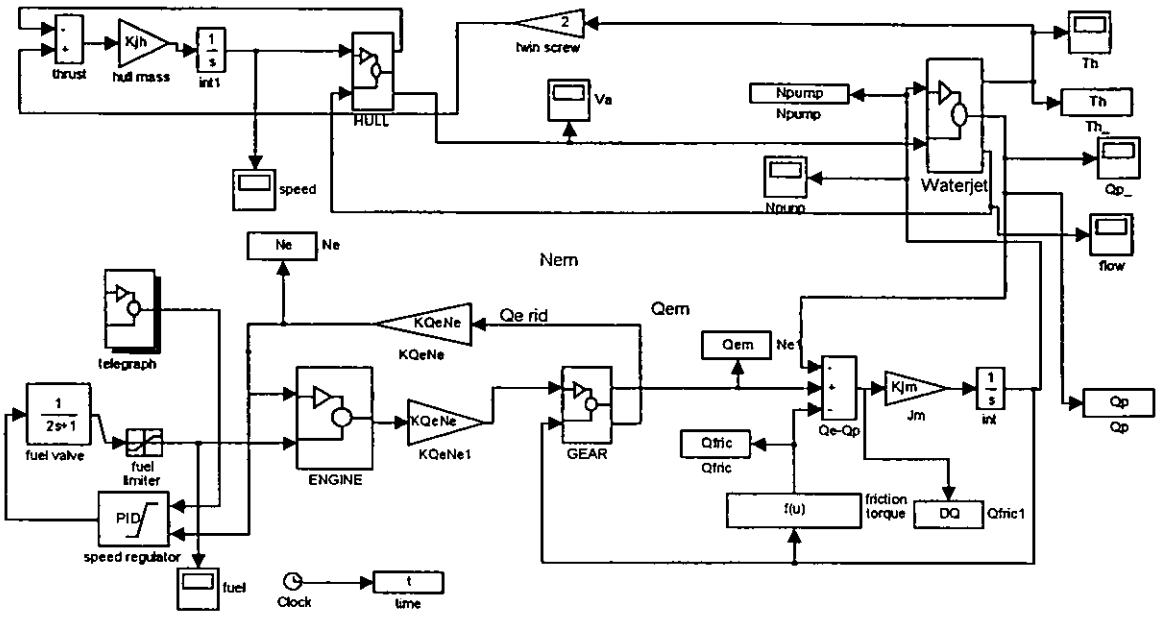


Figure 1 – Waterjet propulsion system simulator

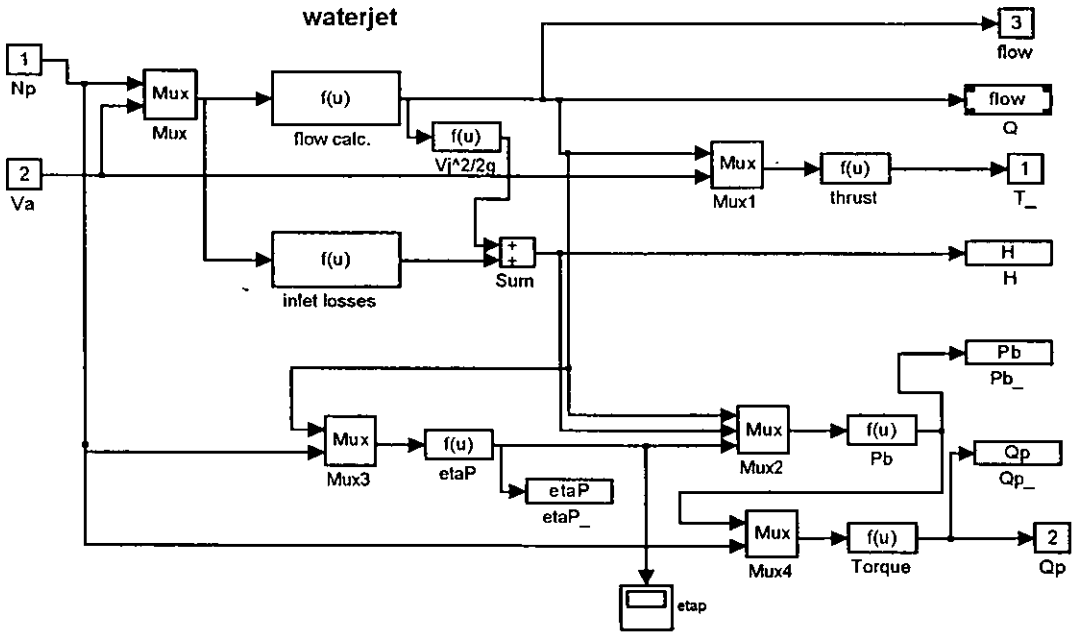


Figure 2 – Waterjet 'block'

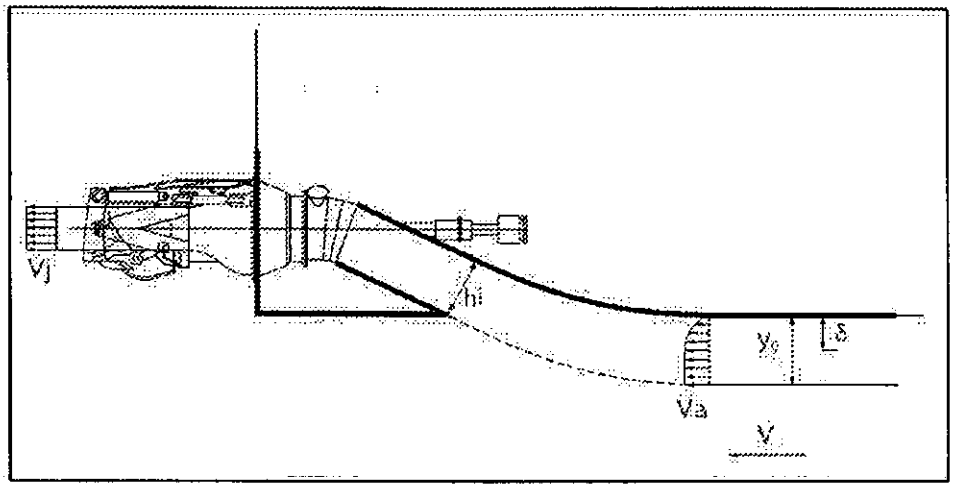


Figure 3 – Waterjet velocities definition

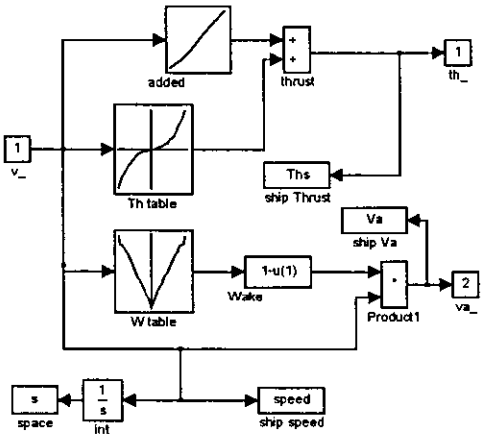


Figure 4 - Hull block

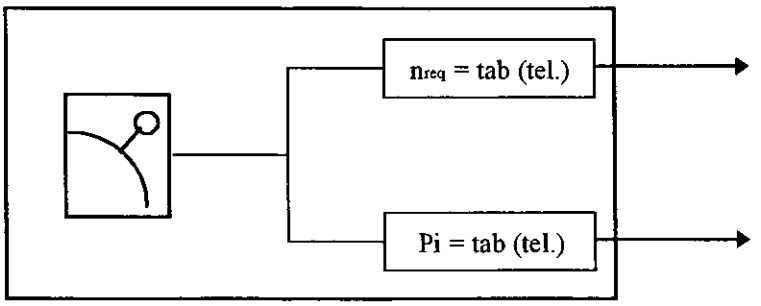


Figure 5 - Telegraph block

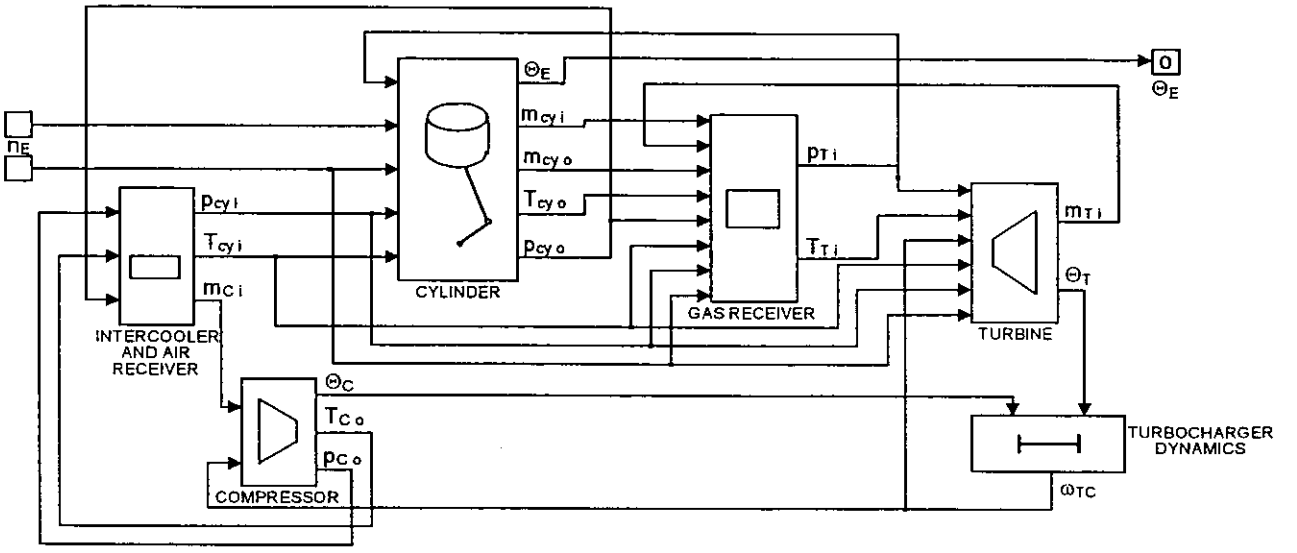


Figure 6 - Overall engine model

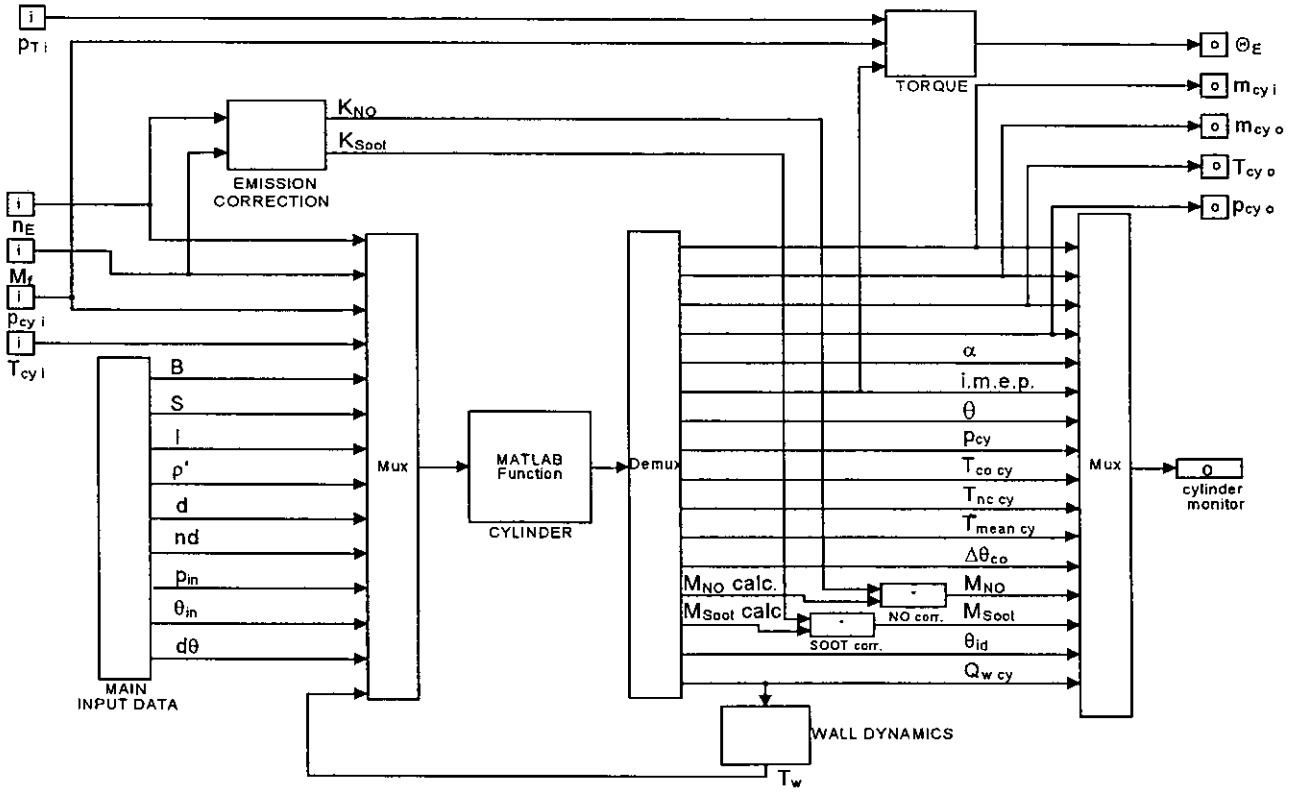


Figure 7- Overall cylinder module

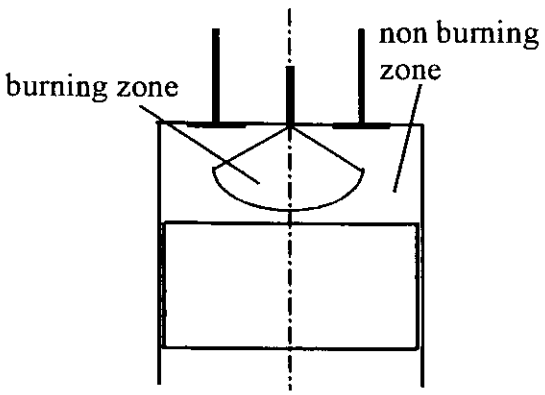


Figure 8 – Two-zone combustion scheme

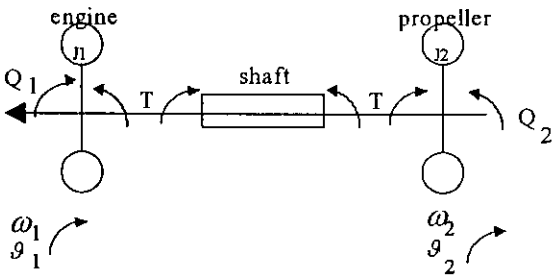


Figure 9 – Shaftline scheme

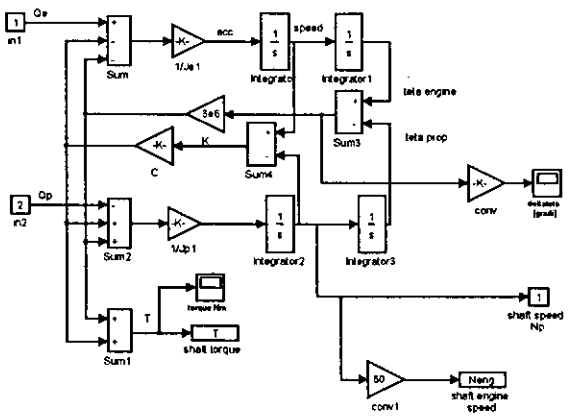


Figure 10 - Shaft dynamics block

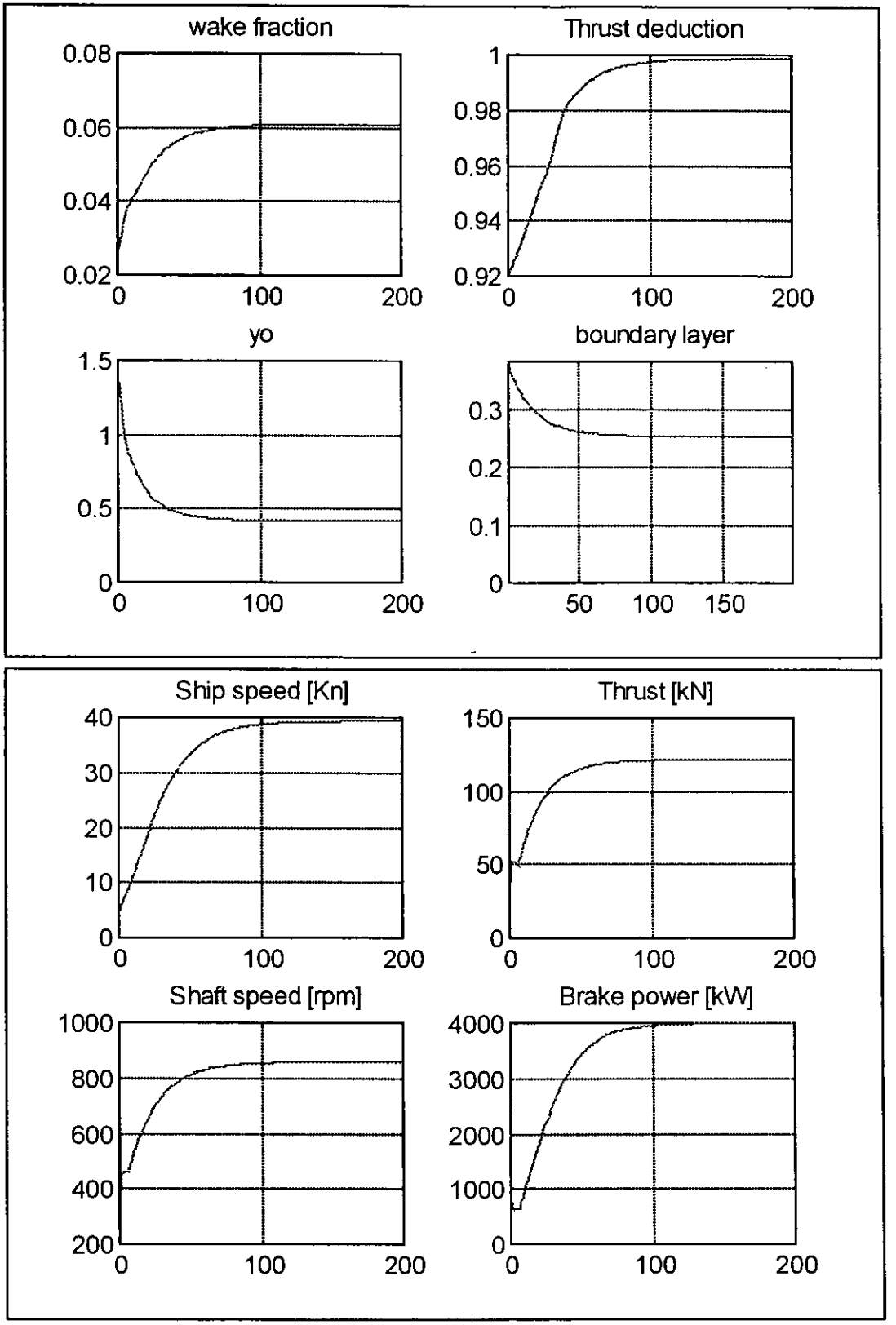


Figure 11 – Waterjet propulsion system transient results

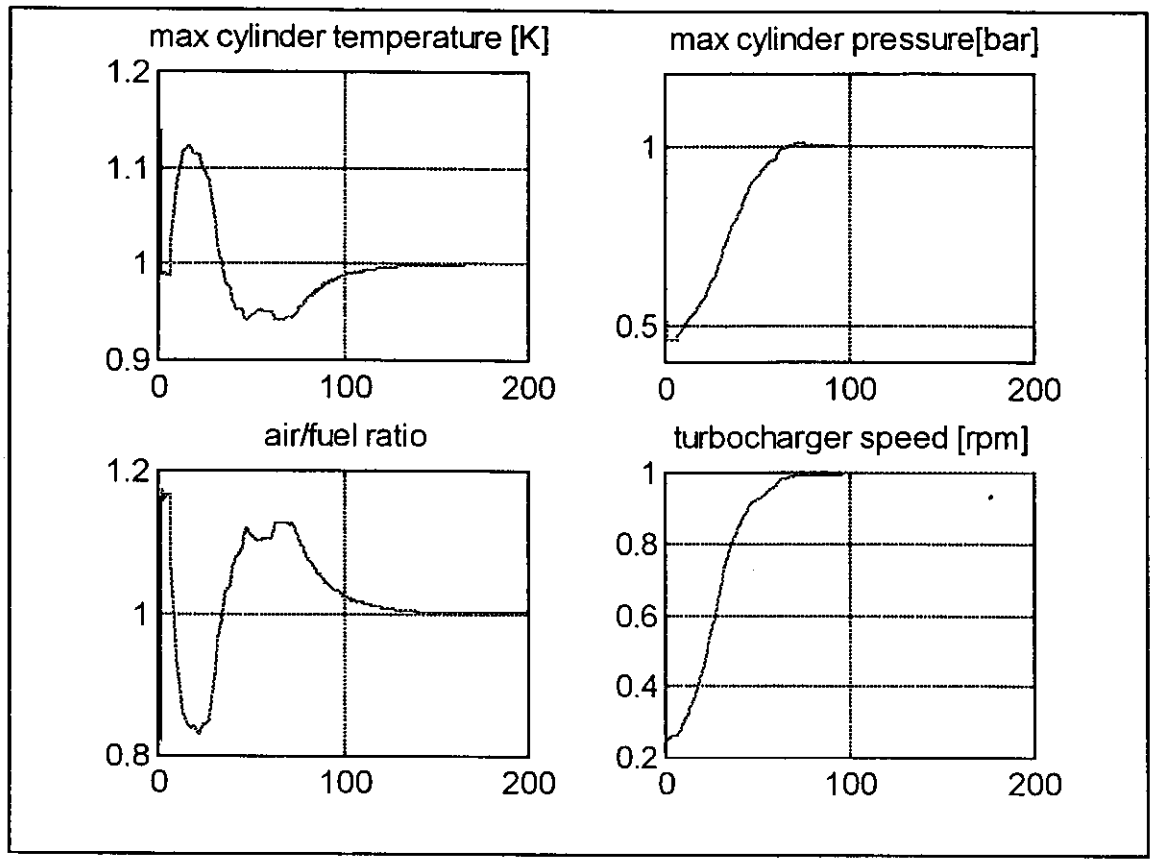


Figure 12 – Prime mover transient results

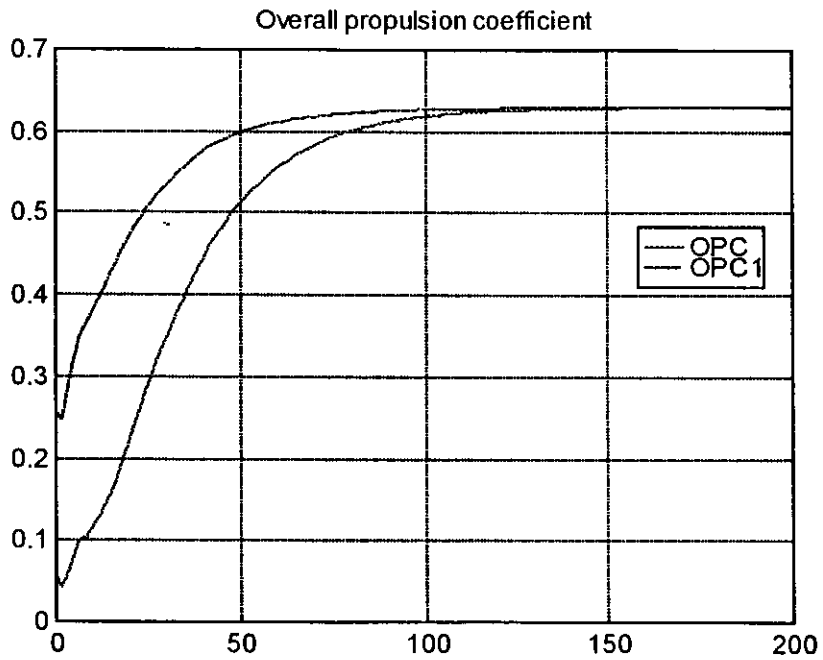
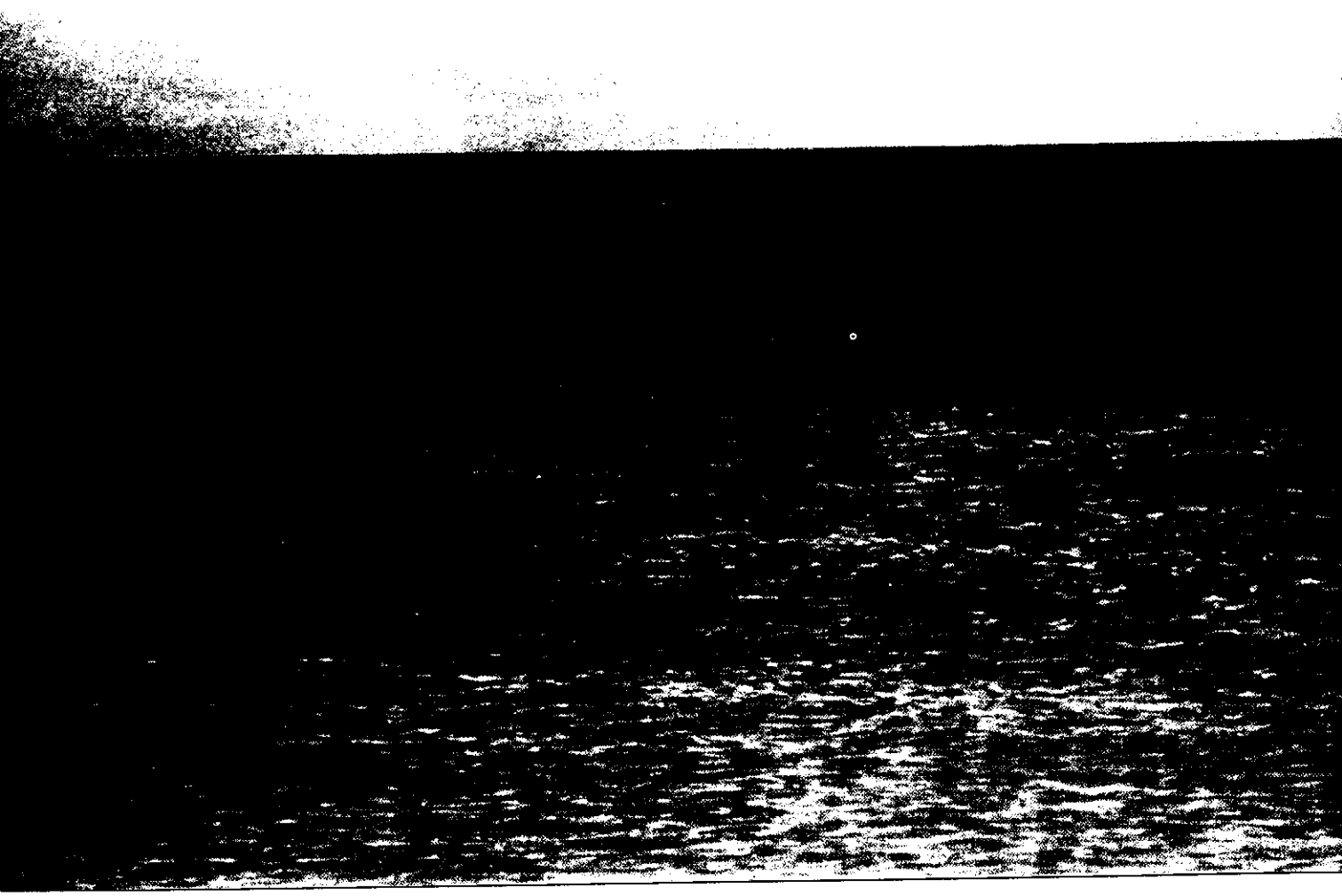


Figure 13 – Overall propulsion coefficients during transient simulation

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