DIESEL II ENGINES

for ship propulsion and power plants

FROM 0 TO 100,000 kW

Kees Kuiken

Author:
Kees Kuiken

Technical advice:
Johan Heuff, Chief Engineer ret. P&O Nedlloyd.
Jan Leekstra, Chief Engineer ret. P&O Nedlloyd.
Christiaan Kuiken, Maritime Officer, Maersk Ship Management.
Kees Verkleij, Chief Engineer ret. machinery design office Royal Schelde, Flushing, The Netherlands.

Translation:
AAG Translations, A.A. Geerts MA, Almelo, The Netherlands.

Textual corrections and proof reading:
S.E. Brent BSc, Almelo, The Netherlands.
Mimi Kuijper-Heeres, Senior Lecturer ret. Maritime Institute Willem Barentsz, Terschelling, The Netherlands.

Layout:
Little Shop of Graphics, Almelo, The Netherlands.

Technical drawings:
Niels Kuiken,
Maarten Bosch (LSoG).

Printing:
Wilco, Amersfoort, The Netherlands.

Finish:
Stronkhorst Van der Esch, Groningen, The Netherlands.

A publication of:
Target Global Energy Training
Noorderhooidijk 2
9755 PJ ONNEN, The Netherlands
Website: www.targettrainingcentre.nl
E-mail: targettraining@planet.nl

ISBN 978-90-79104-02-4

Target Global Energy Training,
Onnen, The Netherlands.

Information
The task of completing this book could not have been accomplished without many substantial contributions by a considerable number of companies both in The Netherlands and abroad. These companies each in their own manner have contributed to the realisation of this edition by allowing interviews and providing photo-material and technical information.
It is impossible to single out any specific company. However, special reference is made of MAN Diesel AG and Wärtsilä, for their generous contributions.

> Introduction

Diesel engines play an important role in today's society: we are quite dependent on them.
Over 100 years after Rudolf Diesel developed a working diesel engine, there is still no real alternative for ship propulsion and electric generators in tropical and/or remote areas.
The diesel engine is indispensable for road haulage, inland shipping, aquatics, electric power emergency systems, agriculture, and passenger transport by road or rail, oil and gas industry and various other industries. We have chosen to make use of many pictures accompanied by a written explanation.
Much highly in-depth technical theory has been omitted as these topics are covered by specialist books available on the market; these topics include thermodynamics, vibrations, materials, and electronics.
We, at Target Global Energy Training have opted for a more practical approach. This includes ample information with respect to the construction of engines, use of materials, various engine categories, maintenance, repairs, and the use of engines.
Much attention has been paid to the choice of proper graphic material. This, in our opinion, is helpful for the reader to gain insight in the various subjects. This publication is indispensable for every person who has dealings with the diesel engine industry, from the smallest engine to 'The Cathedrals of the Oceans'.

Kees Kuiken, Onnen, The Netherlands, July 2008.

> Content

CH 18

Diesel Power Plants

Caterpillar engines in the diesel power plant at Peng Hu II, China.

Electricity is generated by diesel power plants in most parts of the world. This includes mobile floating power plants.
Shown here; discharging a floating power plant with MAN-B&W engines from a semi-submersible 'Dockwise' vessel.

A large diesel power-station contains fourteen eighteen-cylinder V-engines running on H.F.O. for driving the electric generators.

On the left; the gensets and on the right the turbo-blowers with the exhaust-gas pipes on the engines.

18.1 Introduction

For over fifty years diesel power plants have been used to generate electricity all over the world. They are located mainly on islands, in remote areas and often in regions with poor or non-existent infrastructures, for instance in the tropics or in developing countries.

18.2 Classification of diesel power plants

A Four-stroke high-speed diesel engines operating on M.D.O. – generating up to ± 5 MW power output per set.

B Four-stroke medium-speed diesel engines operating on H.F.O. – generating up to ± 20 MW power output per set.

C Two-stroke low-speed crosshead engines operating on H.F.O. – generating up to ± 50 MW power output per set.

D Floating diesel power plants with four-stroke trunk piston- or two-stroke crosshead diesel engines operating on H.F.O. – Delivering from ± 5 MW up to ± 20 MW power output per set.

E Mobile gensets installed in a container with four-stroke high-speed diesel engines, operating on M.D.O. – generating to ± 5 MW power output per set.

18.3 Applications for diesel power plants

Electricity is generated in various parts of the world by diesel engines for the following reasons:

1 **The required electric power is** not large enough to justify building large-scale power stations with gas- and steam turbines. The return on capital expenditure is only acceptable when operating at full load.

2 **The electric power is required for a short time-span.** This occurs in projects for large infrastructures, special events or following major natural disasters. In these situations large emergency generators are used. These comprise diesel gensets placed in containers. This is also applied on a smaller scale with a single genset providing electricity.

Bottom left: Manufacture of a floating-diesel power plant. Make: Wärtsilä.

Bottom right: A floating diesel-power plant or power barge travelling to its final destination. Make: MAN–B&W.

At the front; the electric distribution section with the transformer, switches and pylons for connection to the high-voltage feeder on shore. Left of the engine room the air-inlet filters, with right the exhaust-gas lines and sound dampers.

▲

A transformer station fed by diesel gensets.

Many countries with diesel-power plants, such those found in developing countries, only have an electricity grid in densely populated areas or areas with a high electricity demand. Here the transformers are fed by the diesel gensets increasing the voltage from 13,800 to 110,000 volt.

3 **The local high-voltage electricity network** is not capable to transport large amounts of electric power.

4 **The electricity requirement per area is relatively small, for instance only a few dozen** megawatts. The efficiency of this power output is much higher (± 40%) than the efficiency when using a gas and/or steam turbine. A diesel power plant of 100 MW is large.

5 **The investment per kilowatt hour** is much lower than that of traditional power generators such as steam- and gas turbines.

6 **At partial load** with an installation comprising several engines, a certain number of engines are switched off, so that the remainder operate at full load. Thus the total efficiency for the generation of electric power remains high. The efficiency for the generation of electric power using steam- and gas turbines at partial loads is extremely low.

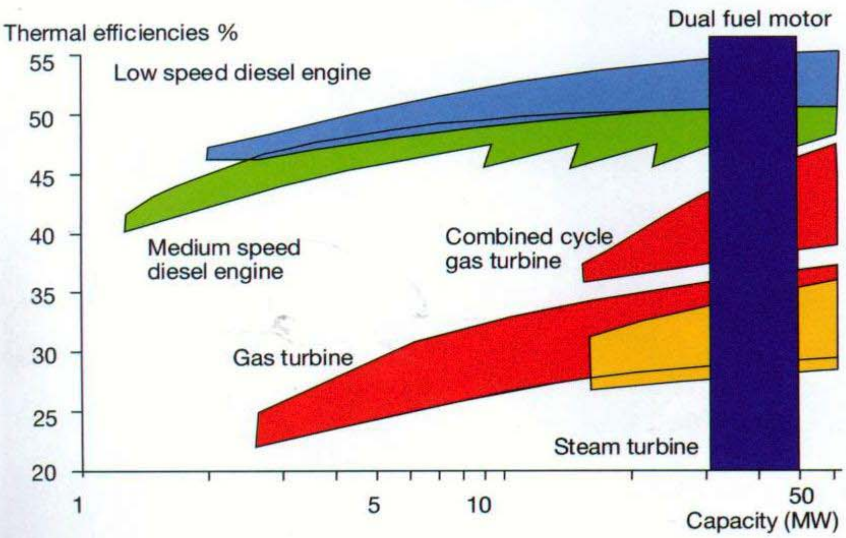
7 **Operating and maintaining** gas- and steam turbines requires highly skilled personnel. This is to a lesser extent applicable for the maintenance of diesel engines.

8 **Maintaining diesel engines** is relatively inexpensive.

9 **Stand-by diesel gensets:** The stand-by genset is automatically started with a power failure of the mains. This system is often found in public buildings such as hospitals, institutions and large office blocks, but also as 'black start' in power plants, in industry and other vital processes. These diesel gensets can be started within several seconds and immediately operate at full load.

Some disadvantages of diesel power plants

- Relatively high exhaust-gas emissions.
- Limited power output per engine:
  - four-stroke: maximum 40 MW;
  - two-stroke crosshead engine: maximum 100MW;
  - gas turbine: maximum 400 MW;
  - steam turbine: maximum 1500 MW;
- Relatively heavy.
- Relatively high maintenance costs.



◄

A total efficiency graph for the various engines and engine combinations up to 50 megawatt shaft power.

steam turbine	yellow	33%
gas-turbine	red	36%
steam - and gas-turbine combination	red	46%
medium-speed engine	green	48%
low-speed engine	purple	53%
dual-fuel engine	blue	56%

Location

- Often in scarcely populated areas (with modest power requirements).
- Especially in the tropics (poor infra-structure).
- Often in third-world countries (low investment levels).
- On islands and other isolated areas.
- In remote areas with large industrial activities, such as in mining, gas- and oil drilling industries.

▲

In diesel-power plants, four-stroke medium-speed diesel engines are often used.

Shown, an eighteen-cylinder V-engine. Considering weight, size and capital cost this is the best choice as two in-line nine-cylinder engines have a similar maximum output, but are heavier, take up more space and capital cost is higher.

18.4 Types of diesel power plants

18.4.1 Utilizing four-stroke high-speed diesel engines operating on M.D.O. Engine category II

Here the power output per diesel genset is limited to 1 to 5 MW due to comparatively expansive diesel oil.
They are used to generate electricity locally for various purposes:

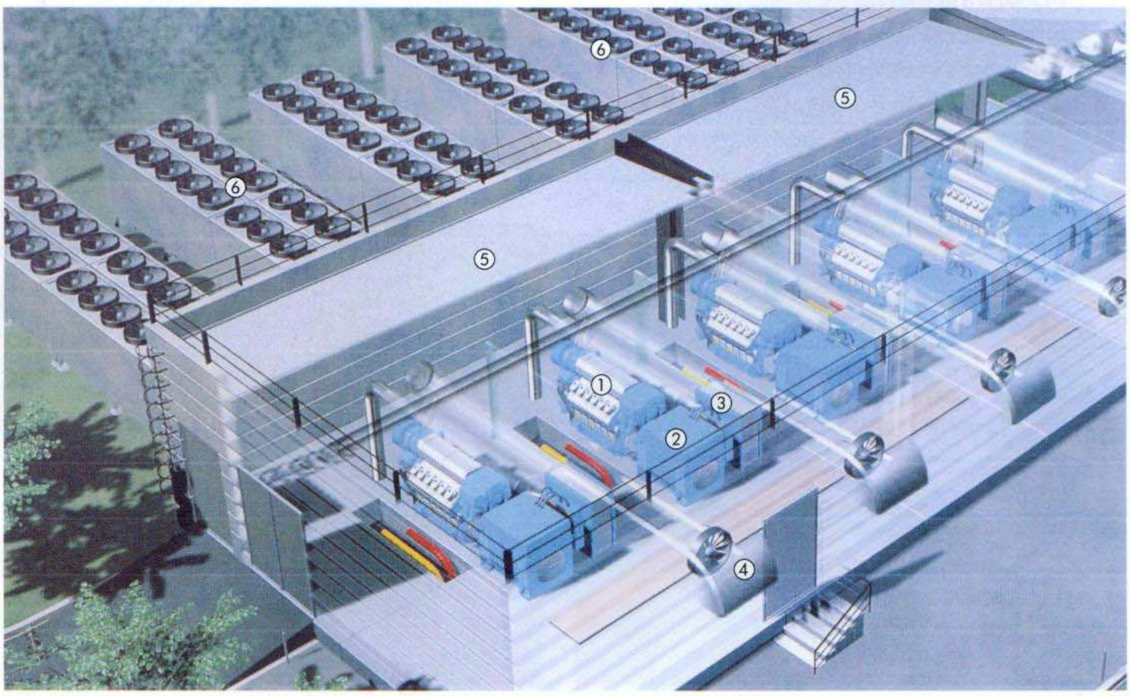
- lighting;
- infra-structure;
- production processes in manufacturing plants such as breweries and water treatment.

The number of revolutions per minute is usually 1500 and 1800, depending on the frequency (50 or 60 Hz). For the slightly larger engines the number of revolutions is lower. However, at approximately 1200 revolutions or less, there is the use of heavy fuel oil is possible. This is cheaper than diesel oil. The engine types used often fall in category II.

▼

An artist's impression of a modularly built diesel-power plant only for electric power.

1 *high-speed four-stroke diesel engine*
2 *electric generator*
3 *main switch board*
4 *engine-room ventilation*
5 *space for distribution station and transformers*
6 *air coolers for discharging the heat from the fresh water, lubricating-oil and combustion air-cooling systems.*



Familiar makes are: Caterpillar, Cummins, MTU/DDC, GM.
With the use of light diesel oils (M.D.O.), the fuel cleaning process is simple and the exhaust-gas emissions in comparison to those of heavy fuel oil (H.F.O.) are less.
The high-speed high-load diesel engines used for gensets are usually arranged in V-shape. They are compact and relatively light and are mounted either in a fixed or flexible manner on a steel frame and can immediately be positioned on the factory floor.

◄

A high-speed four-stroke genset running on diesel oil for electricity generation.

The frame can be easily positioned on the shop floor.

In the gas- and oil industry, many high-speed diesel gensets are set up on locations that lack (natural) gas for operating gas-engine gensets. The total power output varies from several megawatts to over ten megawatts.

►

A high-speed four-stroke diesel genset with its own generator cooling.

In the foreground the electric generator with its own air-cooling system. Offshore, for instance gas and oil platforms, complete sets are often substituted during major maintenance, so as little work as possible needs to be done on location. This is cost effective.

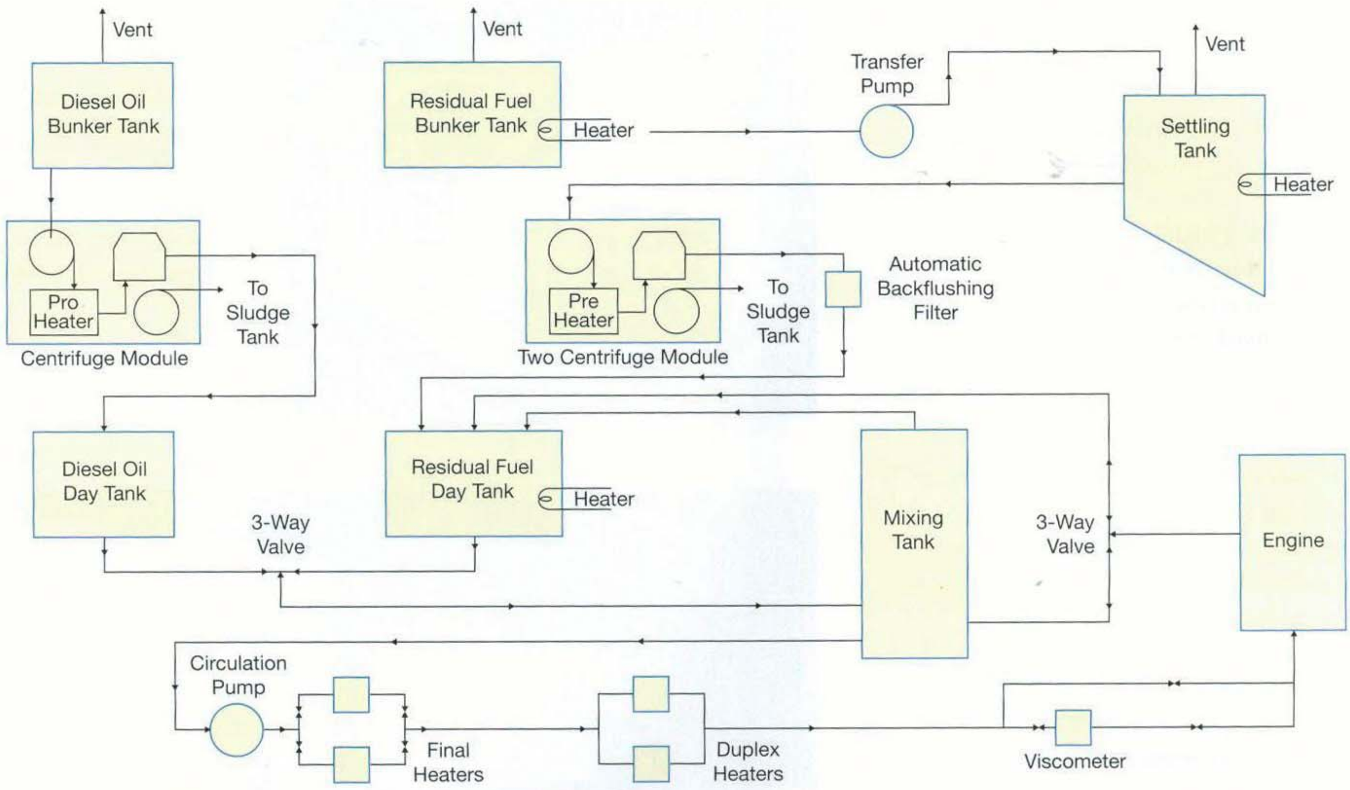
18.4.2 Utilizing four-stroke medium-speed diesel engines operating on H.F.O. Engine category III

Most of the world's larger diesel power plants are in this category. These plants have a total capacity of up to 150 - 250 MW. To reach these capacities, several dozen of sixteen or eighteen cylinder V-engines with large cylinder bores are installed. A fuel treatment and supply system is an integral part of the installation. The large amounts of fuel are delivered by pipeline, ship or fuel tankers and stored temporarily in the storage tanks, enough supply for several days of consumption.

▲

The entrance to a large diesel power-plant in the tropics. Here near Mombassa, Kenya.

Left the plant and right the electric distribution to the high-voltage grid.



▲

A fuel system with storage, cleaning and treatment of diesel oil and H.F.O..

►

A floating diesel-power plant, the 'Seaboard', in the Dominican Republic.

It is moored alongside a highway and close to a viaduct in tidal estuary. Electric power, 72 MW.

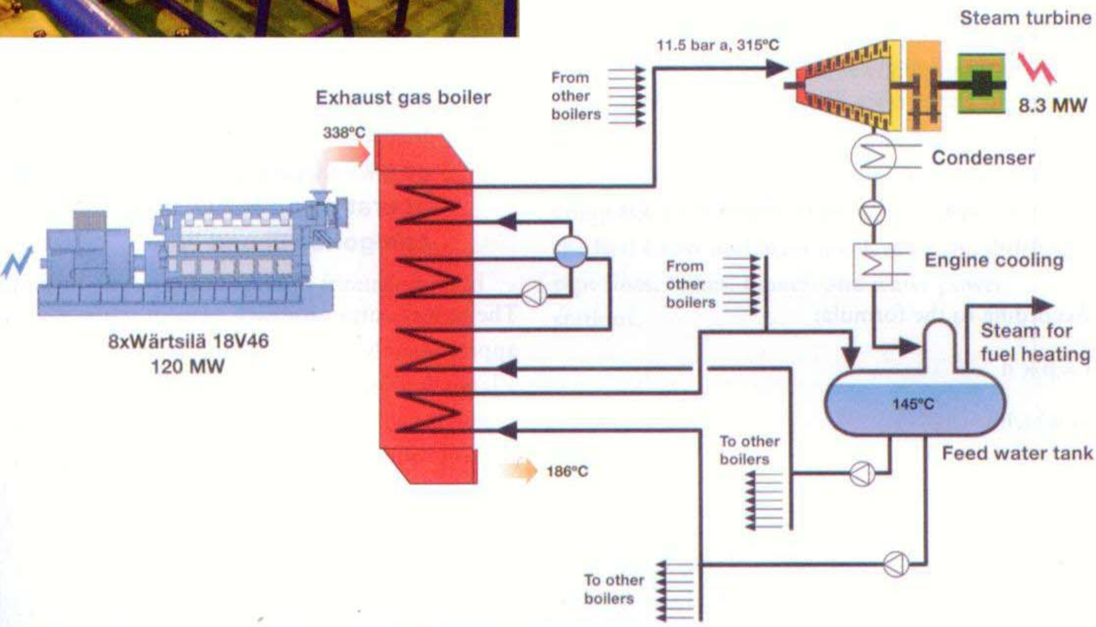
►

Arrangement of diesel engines in the same floating plant.

Left the turbo-blowers and exhaust-gas lines with right the electric generator with air cooling.

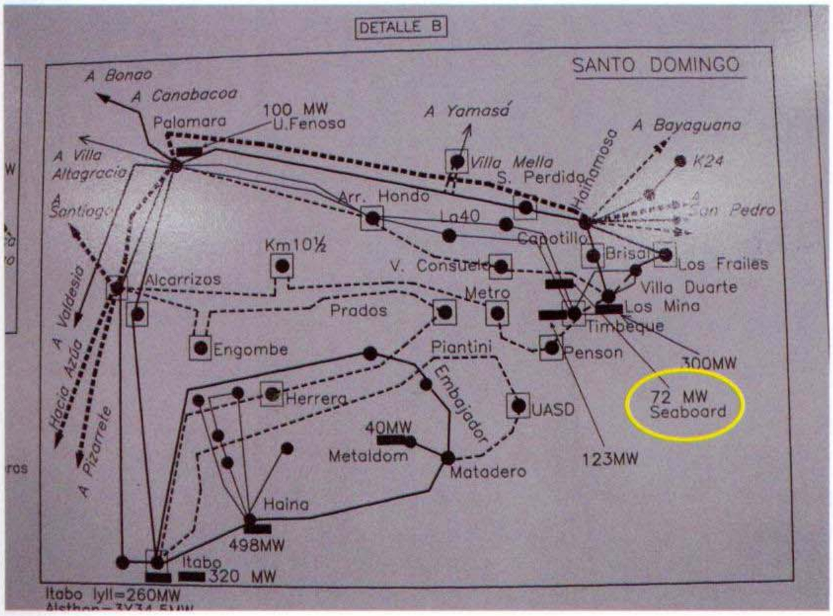
Spare pistons ready for the 12,000 operating hours between overhauls.

In these relatively large plants only a fraction of the total productivity is lost with one malfunctioning diesel genset.
If, for instance, ten engines of 8 MW each are assembled together, one stopped engine would amount to a mere 10% of the capacity, but the genset will still produce 72 MW. The engine load in these types of plants is fairly high, generally 95 to 98% of the nominal capacity using a three-phase generator.



A diesel genset producing steam using the exhaust gases and an exhaust-gas boiler.

The steam drives a turbine genset. This relatively high investment is only cost-effective in large installations with several engines connected to one exhaust-gas boiler. Shown here, eight Wärtsilä 18V46 engines. Total power output 120 MW. The steam turbine generates 8.3 MW at full load.



The electricity network near Santa Domingo, the Dominican Republic.

The black horizontal rectangles are plants. The 'Power-barge Seaboard' on the right generates 72 MW.

A large two-stroke crosshead engine of MAN-B&W for generating electricity.

In the background and right, a large disc generator with a diameter of 8 metres.
1 *disc generator*

18.4.3 Utilizing two-stroke low-speed crosshead engines operating on H.F.O., Engine category IV

Compared to the engine category III, the use of these engines is negligible. The power output for each plant varies from 20 to 240 MW. Obviously, this is dependent on the type and number of engine(s). A low number of engine revolutions requires a low number of revolutions for the generator.

According to the formula:

$f = p \times n$

in which:
f = Hz;
p = number of pole pairs;
n = number of revolutions per second, the number of pole pairs at an engine speed of two revolutions per second (120 per minute) is 25 and the number of poles therefore:
$25 \times 2 = 50$

Therefore: $50 = \frac{120}{60} \times p$.

Fifty poles at this large a capacity take up a great deal of space, and consequently the generator has a very large diameter, the so-called **disc generator or disc dynamo**. The diameters of these generators can reach as much as 5 to 12 metres.

The size and weight of these sets are huge, but compared to the more prevalent four-stroke trunk-piston engines driven by gensets, the number of moving parts is significantly lower. This generally implies that the number of malfunctions is also limited. However, the energy company's preferences play an important part.
From an economic view, diesel power plants with four-stroke medium-speed V-engines using, H.F.O., are the most cost-effective investment.

18.4.4 Floating diesel power plants with both four-stroke trunk piston or two-stroke crosshead engines operating on H.F.O., Engine categories III and IV

The power output for each plant reaches approximately 200 MW. The plant consists of a large floating pontoon equipped with all the required facilities, these include **diesel gensets, a control room, a workshop, fuel- and lubricating-oil storage tanks** and a **step-up transformer** for the medium voltage gensets.
Floating diesel power plants are completely self-sufficient; they can be supplied with fuel and parts when at sea and can be towed to most locations. The underlying concept is that of a mobile power plant, which can be deployed anywhere in the world.
Obviously, the floating plant must be connected to the medium- or high-voltage local network.

Advantages of floating plants

- The floating installation is anchored at the location where the energy is required.
- Expeditious delivery times of approximately twelve months.
- Modular construction, of both gensets and buildings.
- Very flexible deployment to locations.
- Construction in boat yards by modular construction.

- Fuel storage in double-bottomed tanks. Therefore no separate tanks are required.
- Shore power connection to the plant.
- Auxiliary systems are tested at the dockyard as opposed to on location.
- Possibility of using (natural) gas at a later phase.
- The engines are manufactured with two fuel systems (Dual Fuel).
- When natural gas is available, the floating power plant can be converted to operate on natural gas.
- Fewer risks in case of natural disasters, earthquakes and land slides.
- Sea water cooling is possible, so a simple installation with low temperatures is sufficient.
- Mobility, versatility and deployment of these plants means a low-risk investment.

Several significant advantages of using natural gas as fuel

- Very low exhaust-gas emissions.
- Clean combustion process.
- No fuel storage required.
- No danger of fuel spillage, so reduced risk of environmental damage, such as soil- or water contamination.

These two fuel systems are also frequently used for land-based diesel power plants.

Also see Chapter 8, Fuels, fuel line systems and cleaning fuels.

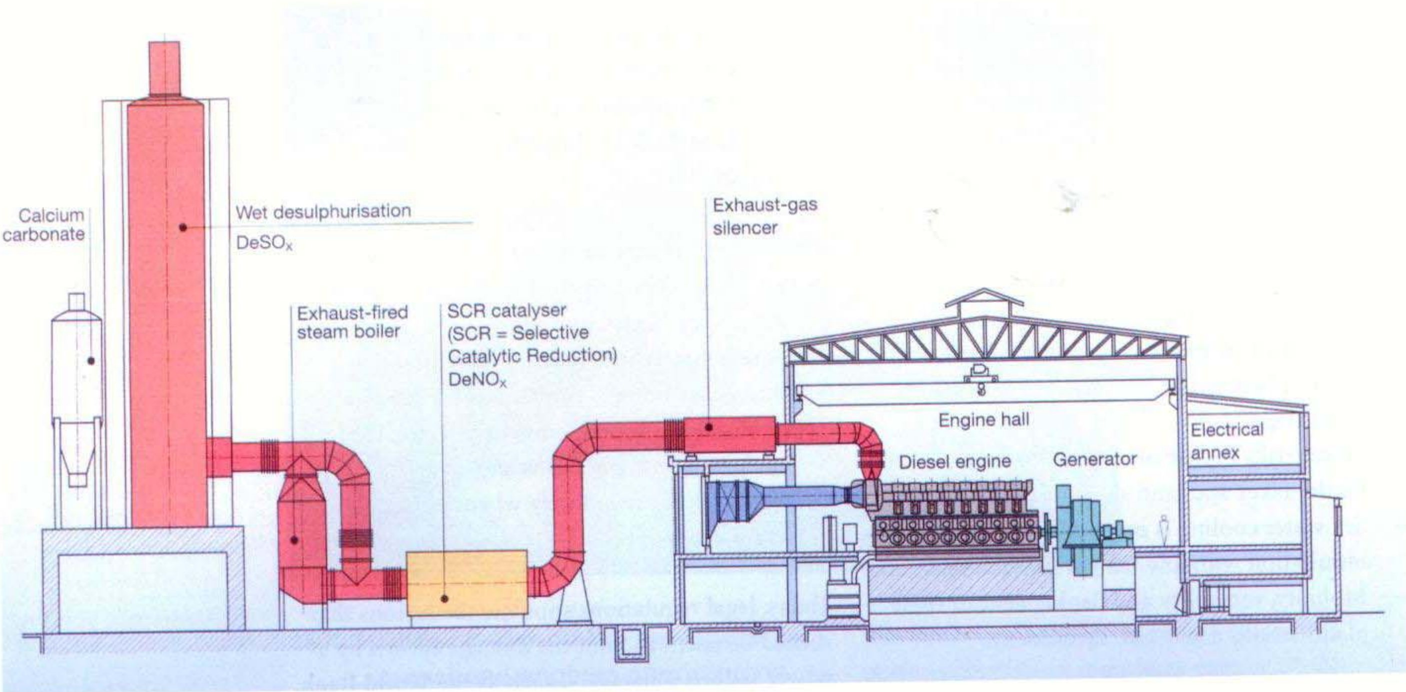
18.4.5 Mobile gensets arranged in a container using four-stroke high-speed diesel engines, fuel M.D.O. Engine categories I and II

Capacity per container is up to ± 5 MW per genset. These are often used for:

- **temporary electricity supply** for large events, building activities, military activities, fairs;
- **emergency power installations** for hospitals, public buildings, the computer-sector, the banking sector, pig farms and a variety of processes in the industry where power cuts cause losses.

Today, **legal regulations** stipulate the actions that should be taken to limit the damage caused by a power failure. Financiers, such as the **World Bank**, also make demands, for instance, with respect to the pollutant emissions. There are many **rental companies** active in this sector, which lease mobile gensets ranging from several kilowatts to several megawatts electrical power output per set.
Some companies utilise these mobile gensets on a continual basis, for instance: **gas- and oil drilling companies, construction firms, the army, circuses, the Red Cross and firms involved in constructing pipe lines, mines, tunnels and water power stations.**

◄ The 'Powerbarge Seaboard' as seen from sea. Eight Wärtsilä-38 engines with eighteen cylinders in V-arrangement generate electricity.



▲

A diesel-power plant with a two-stroke crosshead engine.

On the far right the electrical switching section. Next to it a large disc generator, crosshead engine and air inlet. Above; the sound dampers, the DeNOx installation for the removal of nitrogen oxides and to the far left the DeSOx installation for the removal of sulphur oxides.

18.5 Special applications of diesel power plants

18.5.1 Reduction of toxic flue gas emissions

Reduction of nitrogen oxides, sulphur oxides and soot particles.

Also see Chapter 22, Engine emissions.

18.5.2 Use of the residual heat

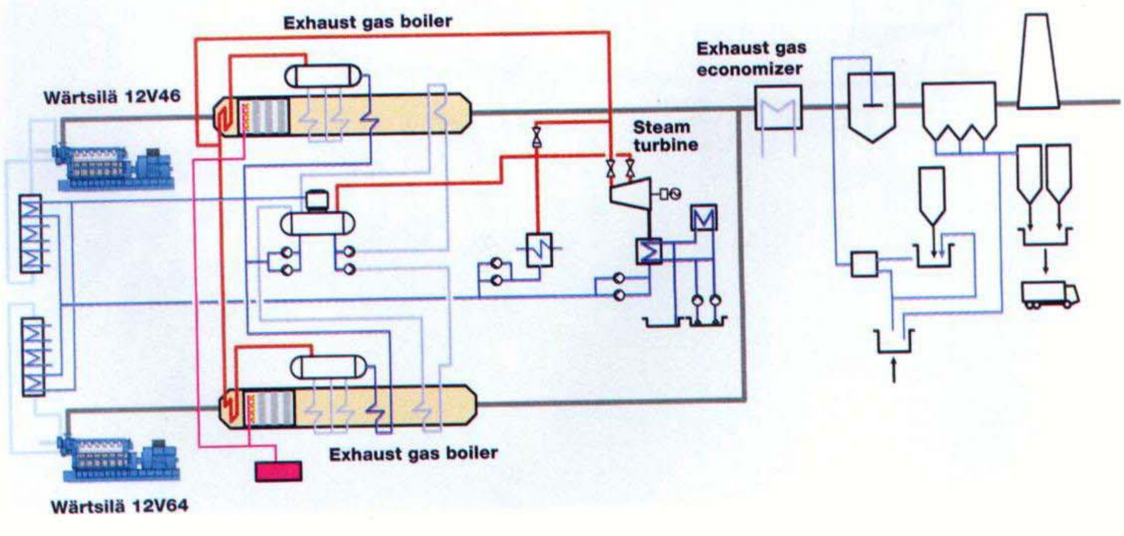
Heat generated by exhaust gases, coolant, lubricating oil and air cooling.

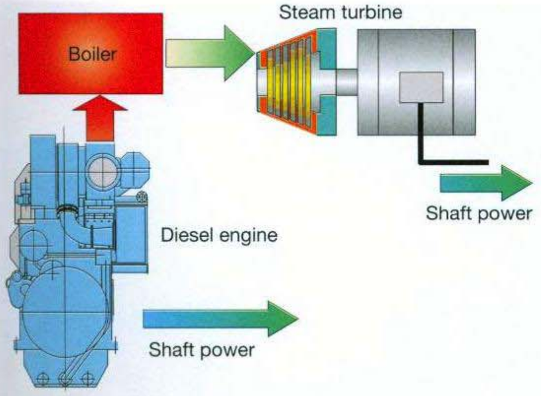
This is also referred to as combined heat and power or CHP.

▼

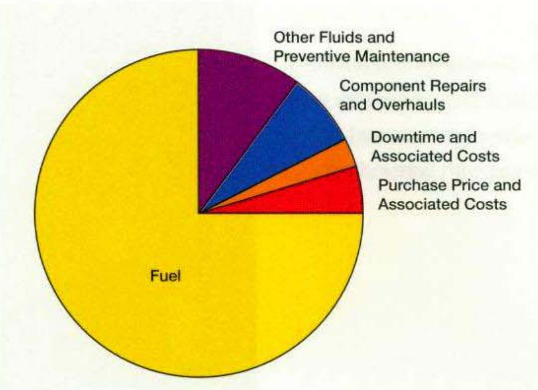
A district heating project with diesel engines in Vaasa, Finland.

Both twelve-cylinder V-engines with bores of 46 and 64 centimetres drive generators which supply electricity for the public grid. The exhaust-gas heat is predominantly used for exhaust-gas boilers. The generated steam is used for driving a steam turbine genset also connected to the network. Since the plant is located in the port near the city, the regulations with regard to the exhaust gas emissions are very strict. Here both DeNOx and DeSOx installations are required (far right).





▲ The energy in the exhaust gases is approximately 25 to 45% of the fuel energy and can be partially used for electricity generation by means of an exhaust-gas boiler combined with a steam-turbine genset.



◄ Fuel costs are the largest cost driver in the exploitation of diesel engines used in shipping and diesel-power plants.

At high power outputs and many operating hours, H.F.O. is often used as opposed to more expensive diesel oil.
Price indication March 2007 Rotterdam
heavy oil 380 cSt.:
160 USD per tonnage weight
heavy oil 180 cSt.:
170 USD per tonnage weight
diesel oil: 225 USD per tonnage weight

With the use of heat exchangers, water can be heated for use in city heating or manufacturing processes.
Using exhaust gas boilers, steam for operating a steam turbine genset is produced which subsequently generates additional electric power. The steam used in the steam turbine can then be re-used in processes that require a lot of heat energy. Much of the heat is, of course, used in the diesel power plant itself. For heating the fuel storage, heating the cleaned heavy oil and maintaining temperatures in stationary engines. Clearly, the extra energy 'gain' must be weighed against the extra investments.

General

Diesel power plants often have a high operational reliability and capability. In the relatively smaller capacities of up to approximately 250 MW, higher efficiency in comparison to steam- and/or gas turbines is usual. The fuel- and lubricating-oil costs form the highest expense posts, as is the case in the shipping industry. Diesel oil, which on average is twice as expensive, is only used for smaller capacities.

Legislation with regard to, amongst others, the pollutant emissions is also becoming increasingly strict in countries where diesel power plants are prolific. Naturally, engine suppliers capitalise on these developments.

Also see Chapter 22, Engine emissions.

◄ Example: In a country like Turkey with huge distances and relatively few industries, many diesel power plants are locally built. Here plants of one engine manufacturer; Wärtsilä.

GD stands for gas diesel. This is a diesel engine using a minimum of 5% diesel and a maximum of 95% gas.
SG stands for spark gas. This is an Otto engine running on fuel gas. All other engines are diesel engines.

▲
The first diesel-power plant at Choloma, with five MAN–B&W Pielstick medium-speed four-stroke engines running on H.F.O..

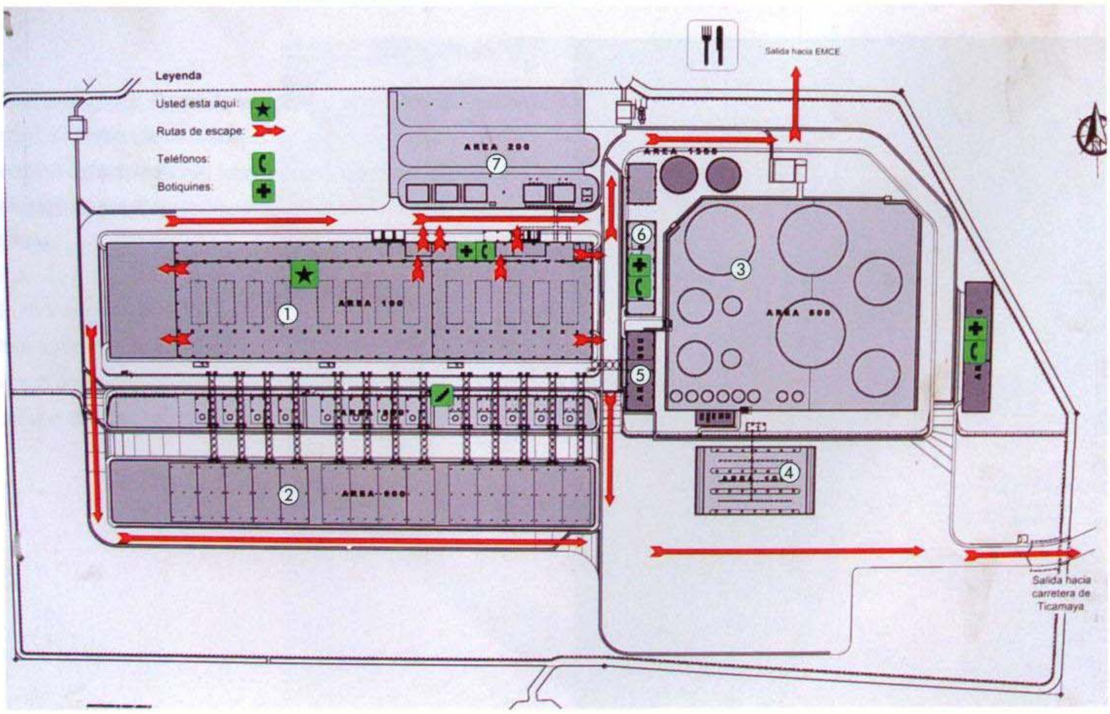
18.6 An example of a large diesel power plant for the generation of electricity

20 February 2007 – Choloma

This power station is situated in the north west of Honduras, near the village of Choloma, approximately 20 km outside the city of San Pedro de la Sula. The power company, Enersa, which has Egyptian investors, owns several diesel-power plants in Honduras and is also the owner of the airport of San Pedro de la Sula.

▼
The new diesel-power plant lay-out.

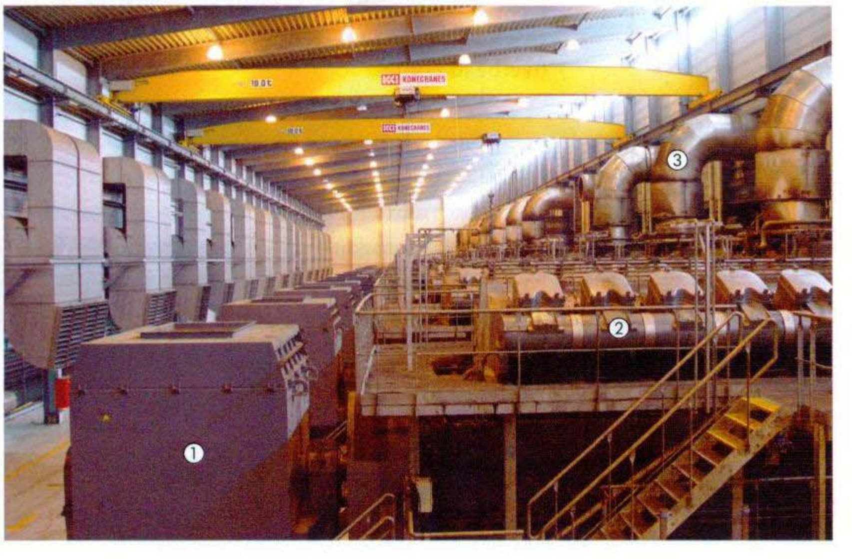
1 *fourteen MAN–B&W 48/60 four-stroke medium-speed H.F.O. engines*
2 *cooling for the H.T. and L.T. cooling-water system to the ambient air*
3 *H.F.O., M.D.O. and lubricating-oil storage*
4 *loading station for tankers*
5 *H.F.O. centrifuge building*
6 *workshop*
7 *transformer station*



The new diesel-power plant.

1 *engine building*
2 *transformer station*
3 *high-voltage lines*
4 *H.F.O. storage*
5 *contaminated-oil storage*
6 *workshops*

Using H.F.O. has very visible consequences; high emissions of visible combustion products, amongst others, soot particles.



In the engine hall.

1 *electric 18.4 MW generator*
2 *MAN–B&W four-stroke medium-speed H.F.O. diesel engines*
3 *exhaust- gas lines*

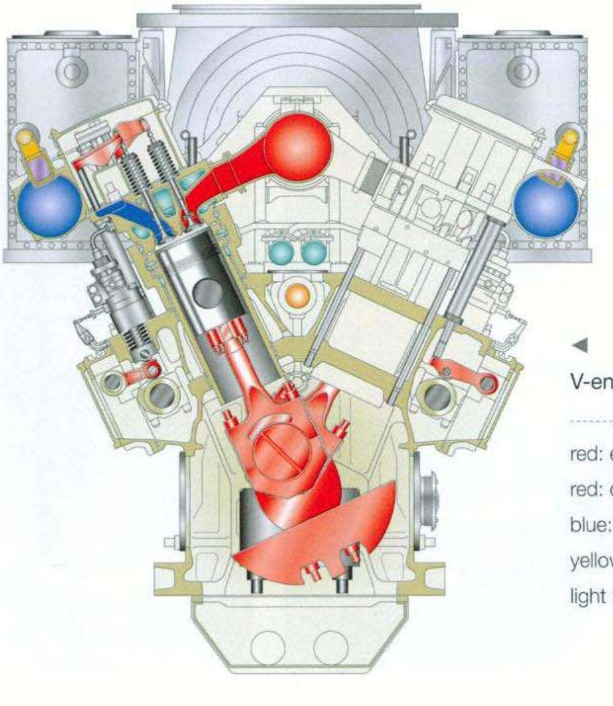
18.6.1 The Choloma III project

The Choloma III project comprises fourteen diesel gensets with the following data:
Diesel engines: Four-stroke medium-speed diesel engines operating on H.F.O., by MAN–B&W, Augsburg.
Type 18 V 48/60 of which three are type18 V 48/60 A and eleven type 18 V 48/60 B.

Electrical power output per genset 18.4 MW. Total power output 250 MW.
This diesel power plant is in the top twenty largest plants in the world.

Generator: ABB

This genset generates a voltage of 13,800, which is increased to 100,000 volt using a 'step up' transformer.



V-engine cross-sections.

red: exhaust gases
red: drive gear
blue: inlet air
yellow: lubricating oil
light blue: H.T. cooling water

▶
ABB three-phase gensets with in the foreground the 'generator row'. Here large parts are moved by 10 tons overhead cranes and trucks.

Installation

The fourteen diesel gensets are arranged along side each other in one large hall with two 10 ton overhead cranes.

18.7 Auxiliary engines

The auxiliary engines are also situated in the hall.

18.7.1 Fuel booster systems

The so-called 'booster units' heat the fuel to approximately 135°C to achieve a viscosity of 10 to 12 cSt.

18.7.2 H.T. and L.T. coolant systems

The H.T. and L.T. coolant systems comprise two pumps for each engine. The H.T. system is used to cool the fresh water- and lubricating-oil system of the L.T. system and cooling the combustion air.

▼
A low-pressure fuel-supply system. Each engine has its own system, the so-called 'fuel boosters'.

▼
The pump row almost 200 metres long.

left: H.T. and L.T. cooling-water pumps (yellow)
right: main lubricating-oil pump (grey) with heat exchangers (blue)

18.7.3 Main lubricating-oil pump

One main lubricating-oil pump with an integrated lubricating-oil cooler and a lubricating-oil filter.

18.7.4 Valve lubricating system

A valve lubricating system for lubricating the inlet valves.

18.7.5 Fuel-nozzle coolant system

A nozzle coolant system for cooling the fuel injection nozzles. Pre-heating the nozzle coolant before starting the engine is possible.

18.7.6 Starting-air system

One system for starting the engines. The starting air valves are mounted in the cylinder heads.

18.7.7 Lubricating-oil separator system

One lubricating-oil separator system for each engine, which automatically cleans the circulating lubricating oil in the engine.

18.7.8 Fire extinguishing system

A fire extinguishing system with foam as an extinguishing agent.

18.7.9 Ventilation system

A ventilation system for the engine hall to safeguard the combustion-air supply and simultaneously cool and ventilate the area.

18.7.10 Electrical generators

Each sleeve bearing in the electrical generator has a separate pre-lubricating pump, which is automatically switched on prior to start and stop.

18.8 Outside the engine hall

18.8.1 Waste-heat exhaust-gas steam boilers

Exhaust-gases boilers are connected to the exhaust-gas manifolds of all fourteen gensets. These produce steam for heating the fuel booster units, the heater for the ten fuel and fourteen lubricating oil separators, the pre heaters for the eight fuel-transfer pumps and storage-tank heating.

▼

Left in the picture the H.T. and L.T. system-cooling blocks.

Right in the picture: over 50% of the supplied fuel heat is wasted with the engine cooling system and the exhaust gases.

18.8.2 Silencers

All exhaust-gas systems have silencers (sound dampers).

18.8.3 Separator room

A dozen separators for cleaning the heavy fuel oil are installed in a separate building. Diesel oil, used for repairs to the fuel systems or when there are problems with the viscosity of the heavy fuel oil, is cleaned using fine-filters.

18.8.4 Fuel receiving station

The fuel is supplied from the storage tanks in the port of Puerto Córtes and delivered using tankers. There is sufficient storage capacity for a two-week plant operation at a maximum fuel consumption of 1100 tons per day.

18.9 Fuel quality

Usually, a heavy fuel oil with a viscosity of 380 cSt at 50 °C and a sulphur content of 2.5% is used. The C.C.A.I. number fluctuates between 845 and 860. This means that the financial department rather than the plant operator determines the fuel quality. Obviously, the quality must conform to the specifications stipulated by the engine suppliers!

18.10 A few particulars

18.10.1 Operational management

Generally, the gensets operate at full load between 06.00 and 22.00 hours. All the gensets are automatically controlled proportional to the load. When the load of all the gensets is too low, with a limit of for instance 90%, the first genset in the

▶ The separator room with ten separators for cleaning heavy fuel oil. These Alfa Laval centrifuges work fully automatically.

▶ A fuel station.

The influence of the United States is clear: all the tankers are of American make!

computer-controlled programme is automatically turned off, and vice versa.
Between 22.00 and 06.00 hours, the electricity requirement and therefore the load is low. In this period only two to three gensets are operational. The others remain on stand-by as does the entire auxiliary plant. A genset can be started within a matter of minutes and is automatically loaded to a maximum of 98%. As few gensets are operational at night, the steam production of the exhaust gas boiler is proportionally reduced. At low steam production a separate stand-by oil-fired auxiliary boiler is started.

▲
Exhaust-gas boilers for heat generation (March 2007).

18.10.2 'Turn key'-projects by MAN-Diesel AG

The entire construction of a diesel power plant including machinery, outbuildings and tank storage is outsourced to a special department at MAN-Diesel AG.
Construction occurs in three phases:
Phase I: five sets (2003/2004)
Phase II: five sets (2004)
Phase III: four sets (2005)

In 2007, extra systems were delivered to the owner 'Enersa'. At this point approximately five hundred claims, standard for such an extensive installation, must be finalised.

18.10.3 Heat exploitation of exhaust gases

In addition to the small exhaust-gas boiler, fourteen high-pressure steam boilers have been installed to drive the steam-turbine genset which has a power output of 15 MW. Separate from the fourteen high-pressure exhaust-gas boilers with fittings and piping, a separate engine hall has been built. This hall contains the steam turbine with reduction gearing, the electrical generator, a high-pressure steam collection drum, and feed-water pumps.
A large air condenser condenses the steam expelled from the steam turbine into water, which returns to the exhaust gas boiler. Unfortunately, there is no company in the vicinity that could use the low-pressure steam in production process as this would lead to a considerable increase in the total fuel efficiency. With the present operation, the power efficiency amounts to approximately 54% and after the installation this rises to about 56%.

18.10.4 Fuel supply

Diesel power plants situated close to a suitable port usually receive the fuel directly from an oil tanker via a short pipe line installed in the harbour. Diesel power plants located further inland receive fuel either via a pipe-line or fuel-tanker. Diesel power plants in third world countries often opt for transportation by fuel-

◄
Tank storage.

black: H.F.O.
white: M.D.O.
yellow: lubricating-oil

tanker, due to the fact that building transport pipelines over long distances is costly and prone to sabotage and theft.

This plant requires 1100 tons of fuel per day when operating continuously at full load!

▶

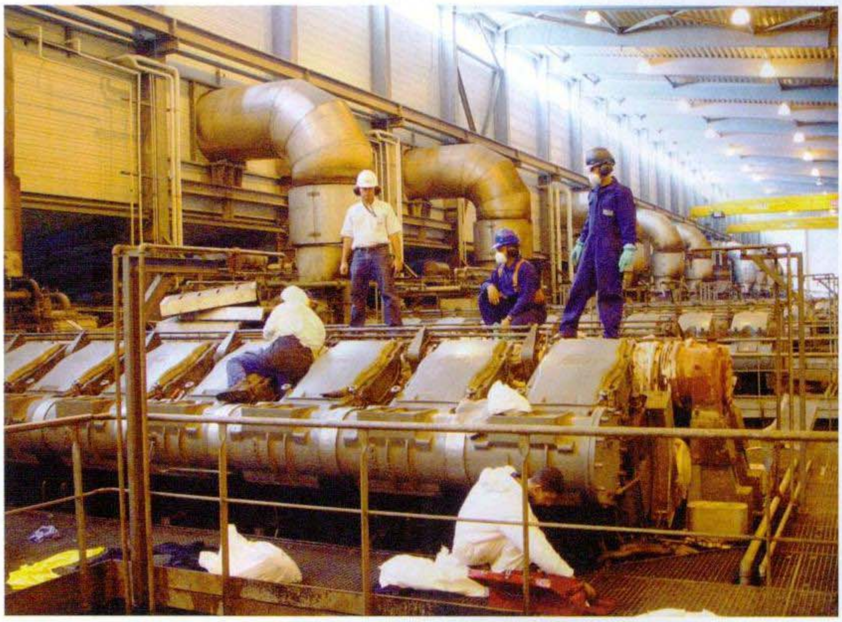
The H.T. and L.T. system for each engine consists of two pumps; one is operational and the other is a stand-by pump.

◀

A large lubricating-oil pump. Each engine has one large lubricating-oil pump with a plate heat exchanger. An automatically operating fine-filter is incorporated in the lubricating-oil system.

▼

Each engine has its own Alfa Laval automatic operating lubricating-oil separator.



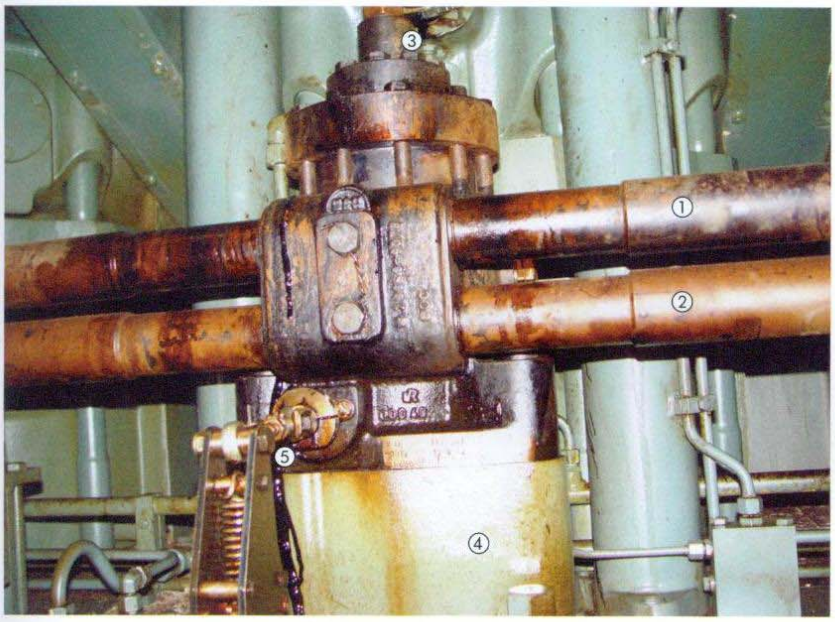
▲

Engine maintenance. Shown here, the fuel injectors are changed.

◄

Engines weighing 110 tons are flexibly arranged on steel pressure springs with centrally, a damper filled with a highly viscous liquid.

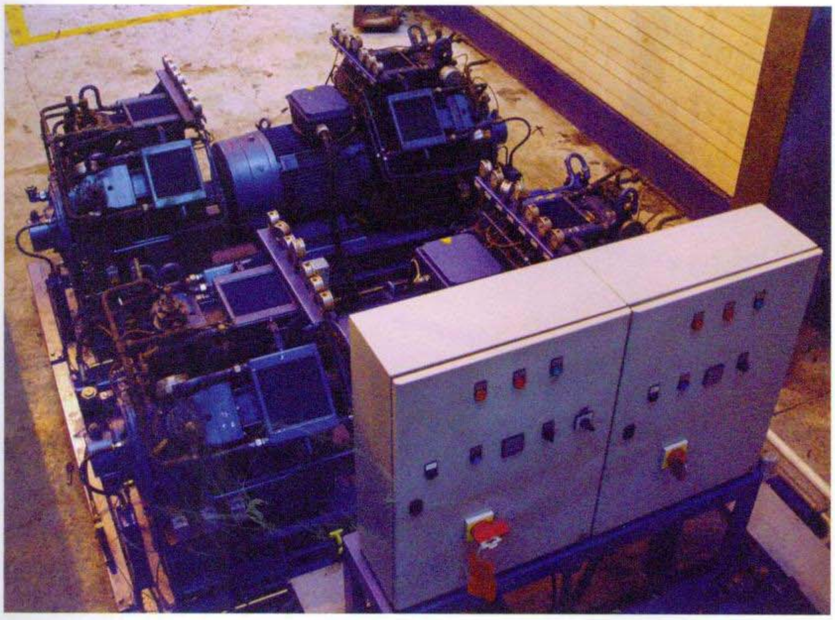
The engines oscillate approximately 2 millimetres. The flexible coupling between the diesel engine and generator must absorb this movement as the generator is positioned on a concrete foundation.



◄

The high-pressure fuel pump, one for each cylinder.

1 *fuel-supply line*
2 *fuel-return line*
3 *high-pressure line to the fuel injector*
4 *pump casing*
5 *supply-control mechanism*



◄

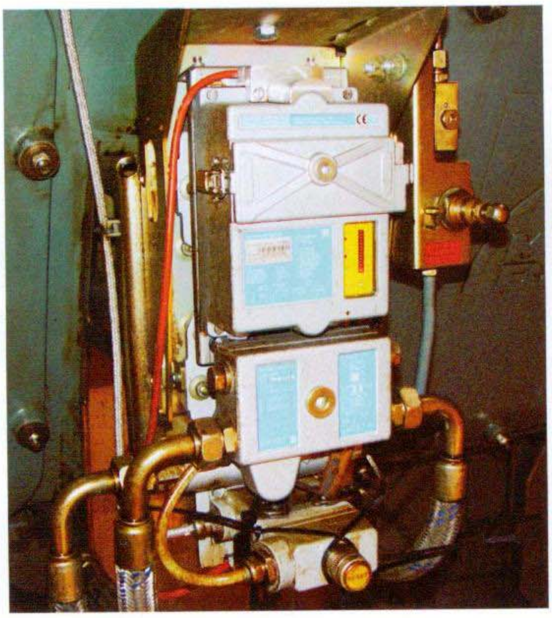
Starting-air compressors.

The engines are started with an air pressure of maximum 30 bar, supplied to the cylinder via a starting-air valve.

A foam fire extinguishing system.

Automatic 'sprinklers' are positioned near the engines. These operate when the ambient temperatures are excessive. First, a water mist followed shortly afterwards by foam.

Each engine has a crankcase mist detector, which indicates when and where a bearing temperature becomes too high.

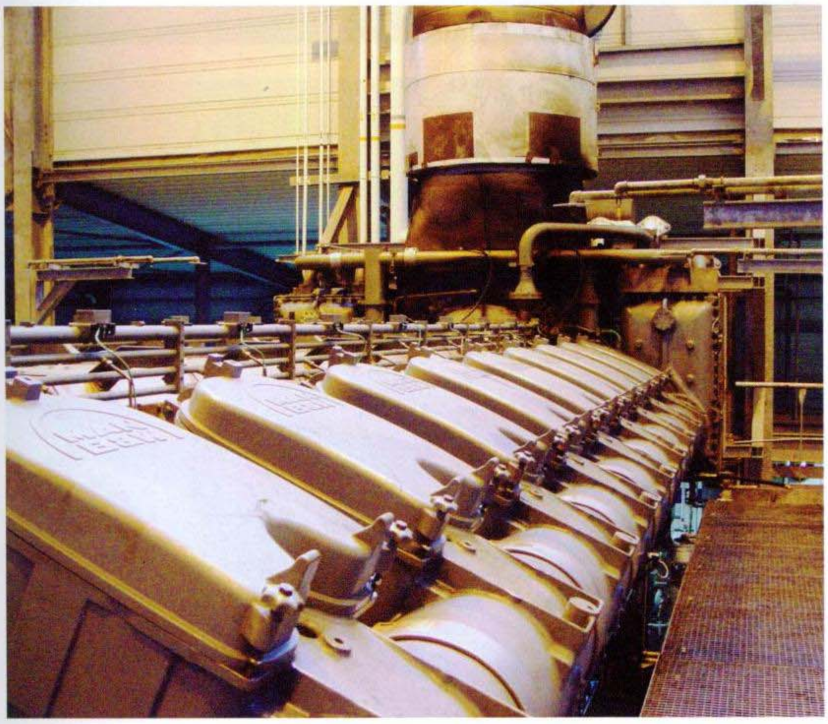


The diesel engine including generator. Note the size!

The transformer station 13,800/110,000 Volt.

The electric main rail to which all engines are connected.

The main switch board where all fourteen generators can be switched onto the network.

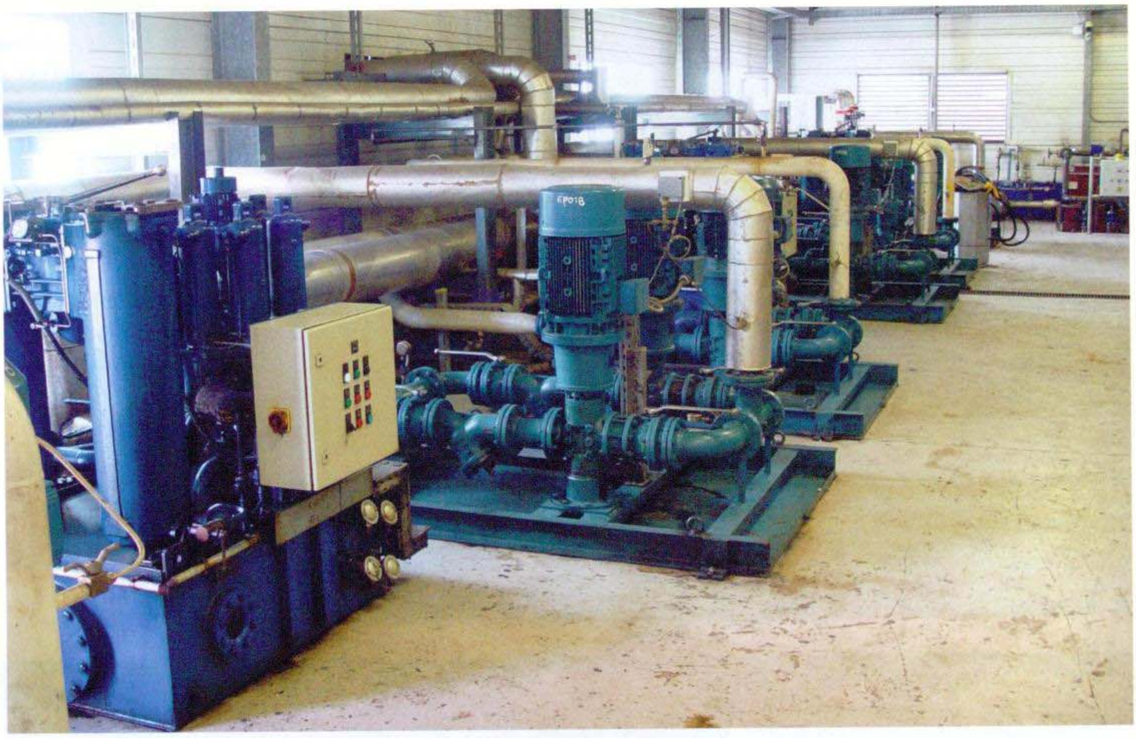


The cylinder-head platform.

Right; the cylinder inlet-air manifold.

▶

Transfer pumps for the fuel storage system.

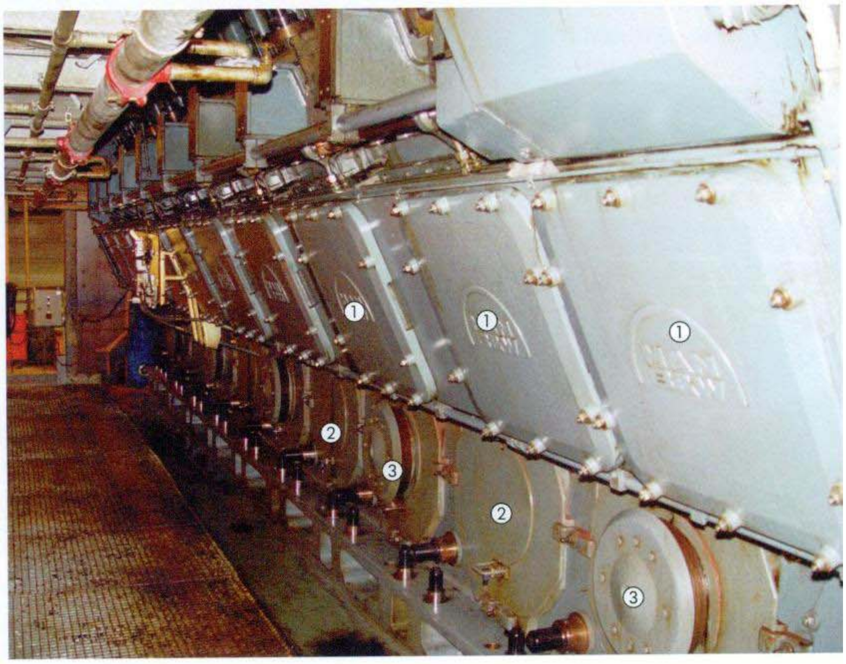


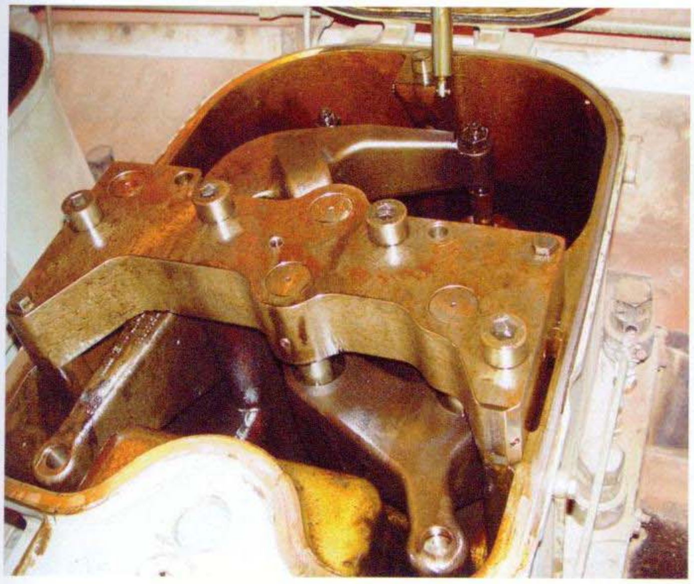
▶

The air ventilators. The combustion process and the ventilation of the building require large amounts of air.

▶

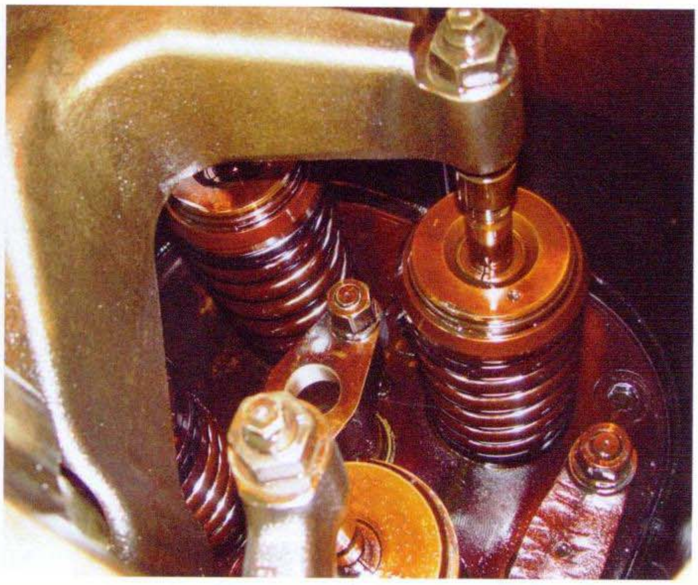
The camshaft 'doors' (1) with the crankcase covers (2) below with several relief valves (3).





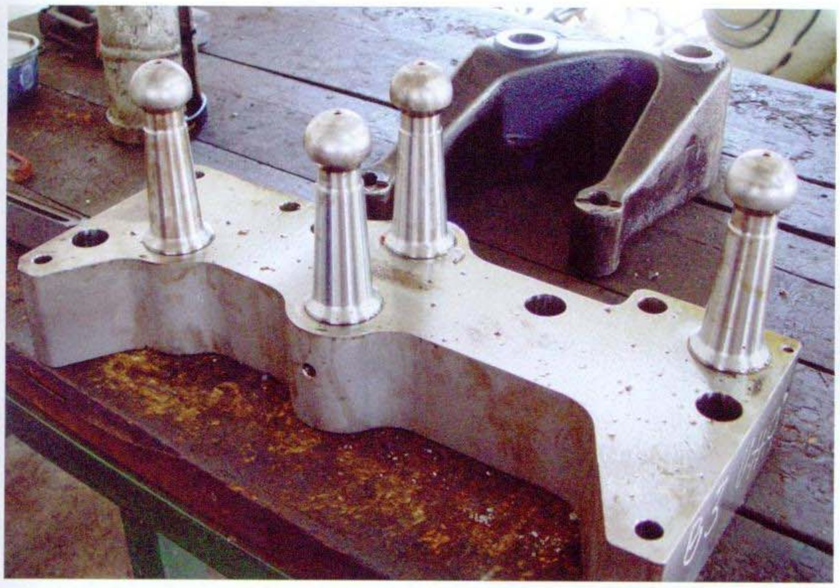
▲

The valve drive of the inlet and exhaust valves; a special construction.



▲

The fuel injector between the two inlet and exhaust valves. The high-pressure double-walled fuel supply line has been removed.

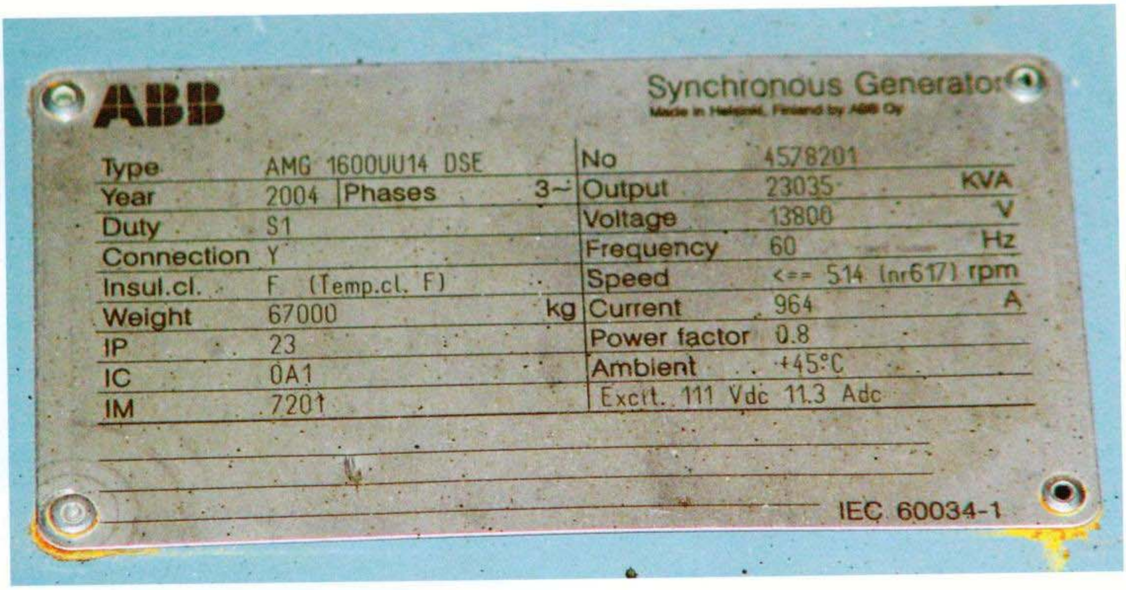


◄

The valve drive's yoke (upside down).

◄

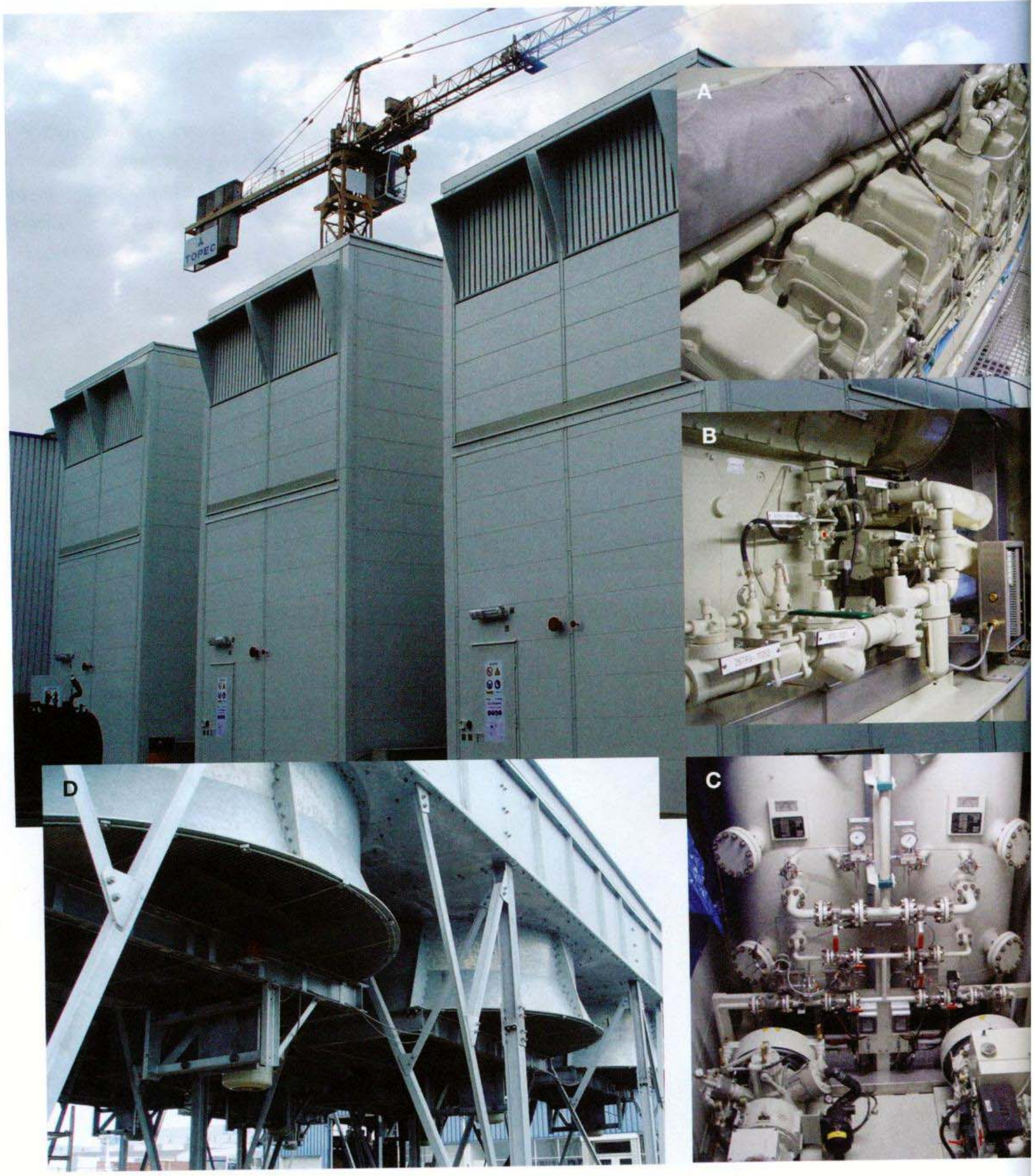
Two transfer pumps for draining tankers.



Manufacturing data of the ABB three-phase generator.

The frequency is 60 Hz instead of 50 Hz, as in Europe. A sizable generator weighing 67,000 kilograms!

18.11 Examples of power plants



Three complete diesel gensets with a power output of 5MW.

Generators are driven by Caterpillar diesel engines category II. After manufacturing, testing and release for certification, the sets are disassembled and transported to Qatar where they will be re-assembled for a refinery.
The test facility here on the premises of Topec/Pon Power in Papendrecht, The Netherlands.

A: One of the three Caterpillar V engines, the largest of this manufacturer running on M.D.O., a 3616 – C 280 – 16 with a speed of 1,000 revs/min.

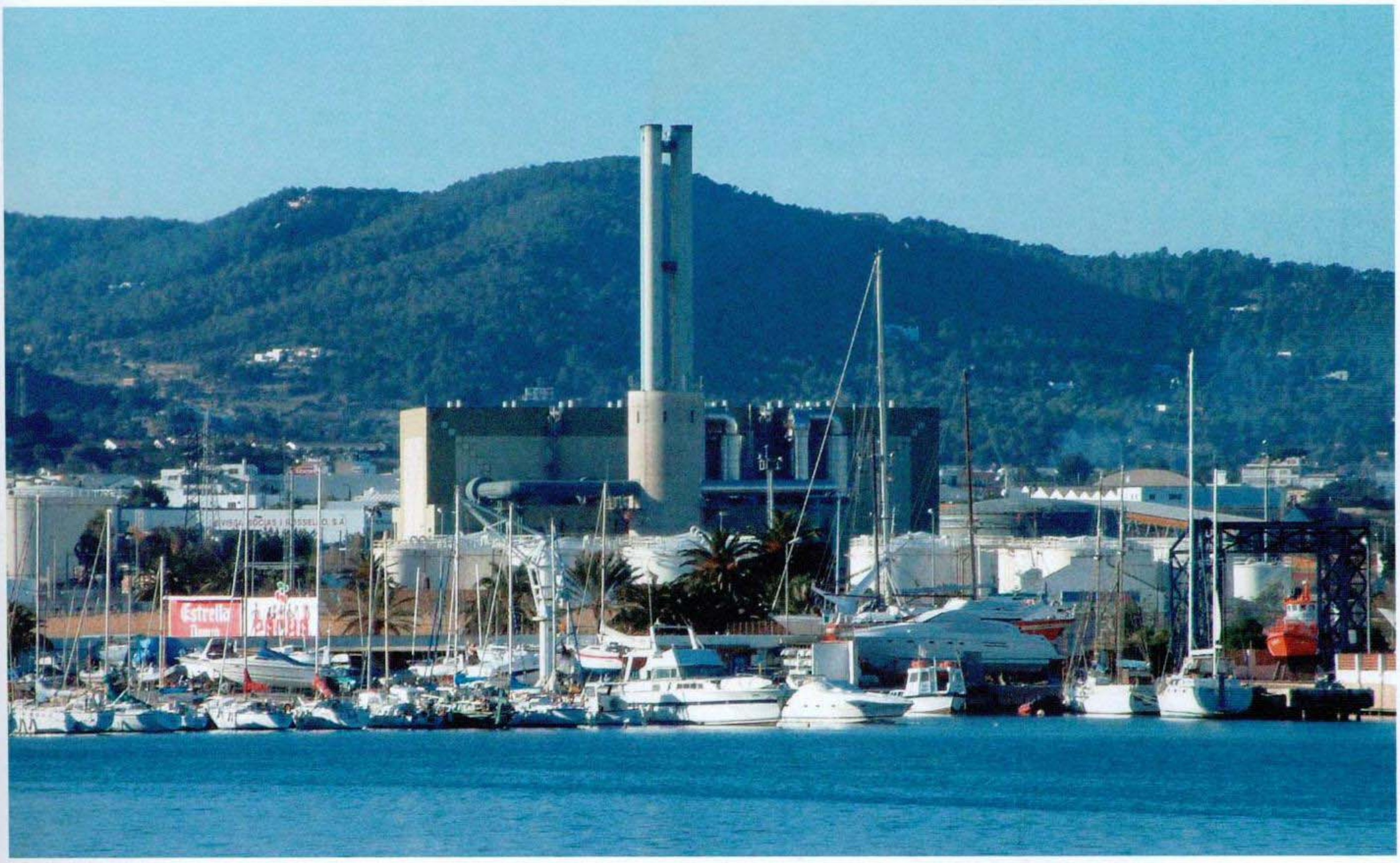
B: The two start rotors near the flywheel.

C: Both starting-air compressors, where one is electric and the other driven by a two-cylinder Hatz diesel engine. In the background are both starting-air reservoirs.

D: The air-coolers used for testing. With these air coolers, the heat of the cool-water system is discharged.

▲
A large floating power plant in the river mouth with Wärtsilä medium-speed diesel engines running on heavy-fuel oil.

▼
A power plant near the waterfront of the harbour.



CH 19

Ship propulsion

Since the introduction of the reciprocating-piston-driven steam engine in the 19th century, propellers have been used for ship propulsion.
The diesel engine has been in use as the standard propulsion engine since approximately the 1950's. The combination of a diesel engine and a propeller amounts to an propulsive efficiency of approximately 33 %. So two-thirds of the energy generated by the fuel is wasted!

Marine craft often are powered by high load, high speed four-stroke diesel engines, category II.

19.1 Introduction

Since the introduction of auxiliary power on sailing vessels by using a steam engine, ships have been fitted with a propeller shaft between propulsion engine and screw.
This is still the case today: nearly all seagoing vessels with a power output above 2000 kW, approximately 90,000 vessels, use a propeller driven by a diesel engine for propulsion. Numerous ships for inland-water navigation, including pleasure craft use propellers. Other propulsion systems such as jet- propulsion are also used.

Every vessel requires a different propeller to achieve a maximum efficiency from the chemical (fuel) energy supplied to the diesel engine. This depends, among other things, on its dimensions, hull shape, and the desired speed, kind of transport, sea area and so forth.

To obtain a basic idea of marine propulsion, the subject is subdivided into three main areas.

1 **Discussion of the most elementary issues, ship dimensions, hull shape, displacement, ship weight, design draught, length between perpendiculars, block coefficient.**
Other topics are towing resistance, comprising frictional, residual and air resistance, and the effects of these resistances in real terms.

2 **Elaborate discussion of marine propulsion and the flow conditions near the propeller. The wake coefficient and the thrust coefficient will also be dealt with.** The total engine output required for the propeller is computed from the aforementioned effective towing resistance and the efficiencies dependent on the various propellers and hull shapes. The operating conditions for a screw deduced from propeller principles for fixed-pitch propellers are discussed. These conditions vary from navigating in calm waters to conditions varying from light to heavy operating resistances, such as fouling and heavy swell.

3 **This section contains a discussion regarding the significance of selecting the correct measurement for the Maximum Continuous Rating (MCR), the maximum available continuous power and the optimising point for the propulsion engine. In this discussion, attention will be paid to the load curve of the engine in relation to the design of the screw.** The compilation of the accompanying load charts is discussed in detail using a number of examples.
It is important to be aware of how various ship resistances affect the continuous service rating of the engine.

▶ The combination of a diesel engine driving a propeller has been the most commonly used propulsion system for over 75 years. It is used by more than 90.000 large sea vessels, hundreds of thousands of inland vessels, and millions of yachts, cabin cruisers and other ships.

Category	Class	Type	
Tanker	Oil tanker	Crude (oil) Carrier Very Large Crude Carrier Ultra Large Crude Carrier Product Tanker	CC VLCC ULCC
	Gas tanker Chemical tanker	Liquefied Natural Gas carrier Liquefied Petroleum Gas carrier	LNG LPG
	OBO	Oil/Bulk/Ore carrier	OBO
Bulk carrier	Bulk carrier		
Container ship	Container ship	Container carrier Roll On-Roll Off	 Ro-Ro
General cargo ship	General cargo Coaster		
Reefer	Reefer	Refrigerated cargo vessel	
Passenger ship	Ferry Cruise vessel		

◄
The three main/largest categories of ship types.

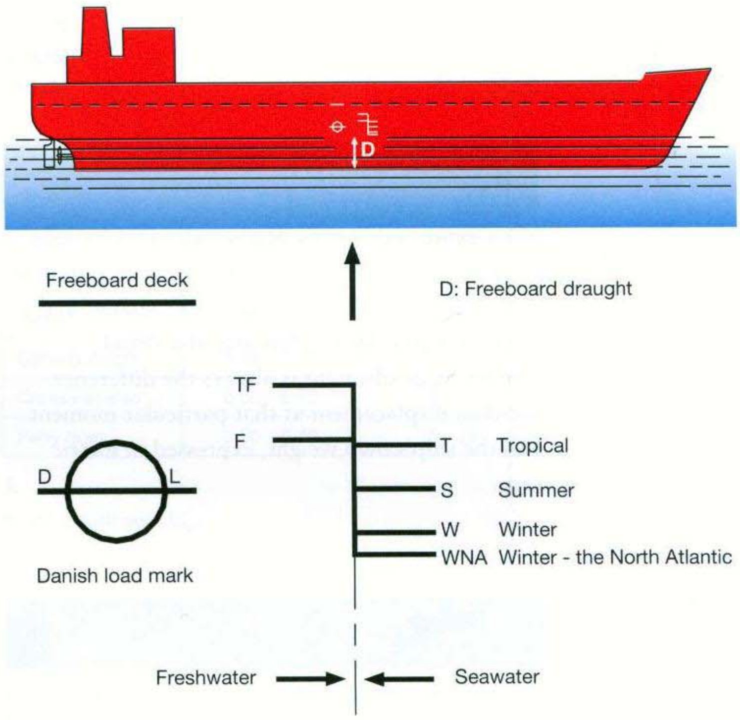
The three main ship categories are the container ships, bulk carriers (grain, ore) and tankers, which can be further subdivided into classes that are more specific and types, such as oil, gas and chemical tankers.
There are often 'multi-purpose' ships as well (which are suitable for various purposes).

19.2 Ship types and hull resistances

Depending on the cargo and loading conditions, ships can be subdivided into different categories, classes and types.

19.3 Load-lines of a ship

Every seagoing vessel has load line marks amidships on each side of the hull; this is known as the 'Plimsoll Mark'. This mark is made by the use of steel plates painted in a colour distinctive from the colour of the ship's hull, welded on port and starboard side of the hull. The lines and letters of the Plimsoll Mark correspond with the freeboard regulations as laid down by the I.M.O., the International Maritime Organisation and indicate the maximum draft to which a ship can safely be loaded. This draft is dependent on sea, season and salinity. The salinity of the water affects the level, fresh water being less dense than seawater and therefore exerts an upward force on the ship.



▲
The load lines of a ship.

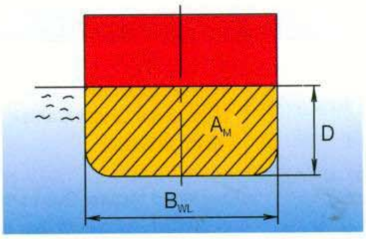
There are load lines for sailing in fresh and seawater, in tropical conditions and for sailing in summer and winter time. According to the international freeboard regulations, the summer freeboard draught is equal to "the scantling draught", which is the draught when dimensioning the hull. The winter freeboard draught is lower than summer's, as there is a greater risk of encountering adverse weather. However, the freeboard draught in tropical seas is somewhat higher than the summer freeboard draught.

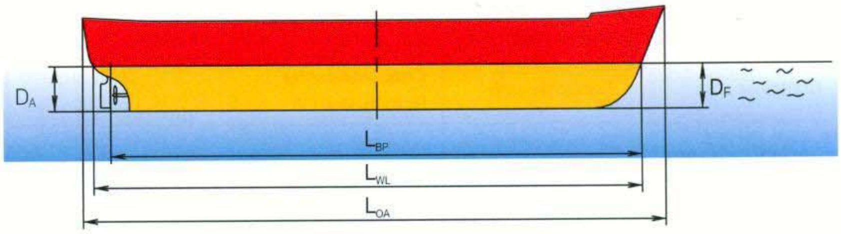
19.4 Ship dimensions and their values

19.4.1 Displacement and loading capacity

When a ship is loaded to a given waterline, displacement is the weight of the water that the ship displaces while floating.
So displacement equals the full weight of the laden ship, in seawater which has a specific mass of 1025 kg per m³.

▶
Displacement and loading capacity.





Length between perpendiculars : L_{BP}
Length on waterline : L_{WL}
Length overall : L_{OA}
Breadth on waterline : B_{WL}
Draught : $D = 1/2\ (D_F + D_A)$
Midship section area : A_M

Displacement includes the weight of the ship itself and the weight of its cargo. Loading capacity is often expressed in tons deadweight. Deadweight includes not only the cargo but also fuel and other necessities such as lubricants, water, spares, and stores required for the running of a vessel. Therefore, deadweight is always the difference between displacement at that particular moment and the ship's own weight, expressed in metric tons.

Ship type	dwt/light weight ratio	Displ./dwt ratio
Tanker and Bulk carrier	6	1.17
Container ship	2.5-3.0	1.33-1.4

▲
The relationship between a ship's dead weight, lightweight and displacement.

First column: deadweight to lightweight ratio
Second column: displacement to dead weight ratio

The lightweight of a ship is not usually used to indicate the size of a ship. Deadweight, based on the loading capacity of the ship, including fuel, lubricating oil and other supplies required for ship operation, provides a good indication of the size (read: loading capacity) of the ship.

19.4.2 Deadweight = displacement = total ship's weight

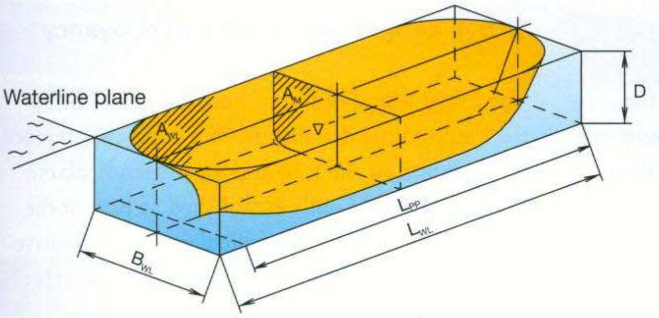
Displacement may also be expressed as the ship's displacement volume. SI unit is m^3.

19.4.3 Gross register tons/ Net Register Tons

Ship's volume can also be expressed in Gross Register Tonnage (G.R.T.) and Net Register Tonnage (N.R.T.). A gross register ton equals 100 cubic feet or 2.83 cubic metres.
These values indicate the total internal volume of the ship according to certain complex rules that apply to said values and are often used to calculate commercial fees such as port and canal dues.

19.5 Hull forms

Specific for ship propulsion is the part of the ship that is located under the waterline.
The ship measurements found below are used for the design draught. The design draught selection is dependent, among other things, on the cargo, the weather conditions, and the sea area. The degree to which the ship is loaded is also important: is the ship lightly, heavily or fully loaded? In general, the most common draught between full-load displacement and ballast displacement is used.



Volume of displacement : ∇

Waterline area : A_{WL}

Block coefficient, L_{WL} based : $C_B = \dfrac{\nabla}{L_{WL} \times B_{WL} \times D}$

Midship section coefficient : $C_M = \dfrac{A_M}{B_{WL} \times D}$

Longitudinal prismatic coefficient : $C_P = \dfrac{\nabla}{A_M \times L_{WL}}$

Waterplane area coefficient : $C_{WL} = \dfrac{A_{WL}}{L_{WL} \times B_{WL}}$

▲
Hull coefficients.

Ship's lengths

Generally, the overall length (l.o.a.), the total length of the ship, does not affect the calculation of the hull's water resistance.

The length on the waterline (l.o.w.) and the length on the load lines (l.o.l.) are factors used for the calculation.

The length between perpendiculars is the length between the first load line, usually a vertical line through the stern's intersection with the waterline, and the aft load line, which normally coincides with the axis of the rudder.

Generally, this length is slightly shorter than the waterline length and is expressed as:

$\text{Between}_{\text{the perpendiculars}} = 0.97 \times L_{OWL}$.

Draught D

The draught of the ship D, is defined as the vertical distance from the waterline to the point on the hull which is most submerged.

Draught, forward and aftmost are usually identical when the ship is loaded.

Breadth on the waterline B_{WL}: This is the largest breadth on the waterline.

Block coefficient B_E: Various shape coefficients are used to denote the shape of the hull.

Ship type	Block coefficient C_B	Approximate ship speed *V* in knots
Lighter	0.90	5 – 10
Bulk carrier	0.80 – 0.85	12 – 17
Tanker	0.80 – 0.85	12 – 16
General cargo	0.55 – 0.75	13 – 22
Container ship	0.50 – 0.70	14 – 26
Ferry boat	0.50 – 0.70	15 – 26

▲
Block coefficient C_B.

The most important of these coefficients is the block coefficient C_B, which is the ratio of displacement volume and the volume of a block with dimensions: length on the waterline, the breadth on the waterline and the draught according to the formula:

$$C_B = \frac{\text{Displacement volume}}{L_{OWL} \times B_{OWL} \times D}$$

In this instance the block coefficient refers to the length on the waterline.

However, ship manufacturers often use a block coefficient, which is based on the length between perpendiculars, thus producing a slightly larger block coefficient, as the length on the perpendiculars is slightly smaller than the length on the waterline.

$$C_B = \frac{\text{Displacement volume}}{L_{OLL} \times B_{OWL} \times D}$$

A small block coefficient means less resistance and consequently potentially higher speeds.

19.5.1 Water-plane coefficient C_W

The water-plane coefficient is the ratio of the water-plane A_W and the product of the length on the waterline and the largest width of the ship.

$$C_W = \frac{A_W}{L_{WL} \times B_{WL}}$$

In general, the water-plane coefficient is approximately 0.10 higher than the block coefficient or:

$$C_W = B_C + 0.10$$

The difference is slightly larger for fast ships with low block coefficients, where the stern is partially submerged, and therefore part of the water-plane.

19.5.2 Midship-section coefficient C_M.

The midship-section coefficient C_M, is the ratio of the submerged midship section A_M (midships between the fore and aft perpendiculars) and the product of the ship's width B w.l. and the draught D.

$$C_M = \frac{A_M}{B_{WL} \times D}$$

For bulk carriers and tankers this coefficient is approximately 0.98 to 0.99 and for container ships 0.97 to 0.98.

19.5.3 Longitudinal prismatic coefficient C_P

Longitudinal prismatic coefficient C_P is the ratio of the volume of displacement V_W and the product of the mid-hip section area A_M and the length to the waterline L_{WL}.

$$C_P = \frac{V_W}{A_M \times L_{WL}} = \frac{V_W}{C_M \times B_{WL} \times D \times L_{WL}} = \frac{C_B}{C_M}$$

The C_P is not an independent form coefficient, but entirely dependent on the block coefficient C_B and the midship-section coefficient C_M.

19.5.4 Longitudinal centre of buoyancy

The longitudinal centre of buoyancy (L.C.B.) is the longitudinal distance from a point of reference to the centre of the fore and aft perpendiculars. The distance is usually established as part of the length between the perpendiculars and is positive if the reference point is situated in the first half of the length between the perpendiculars and negative when situated in the last half.
For fast ships such as container ships, the L.C.B. is negative, while slow-speed ships such as tankers and bulk carriers usually have a positive L.C.B. The L.C.B. is usually between – 3% and + 3%.

19.5.5 Fineness-ratio C_{LW}

The length/displacement ratio or Fineness ratio (C_{LW}) is the ratio of the waterline of the ship L_{WL} and the length of a cube with a volume that is equal to the volume displaced:

$$C_{LW} = \frac{L_{WL}}{3\sqrt{V_W}}$$

19.6 Ship's resistance

A ship must move efficiently through the water and it is therefore imperative to diminish the ship's resistance to the water. The force of ship's resistance is the opposite of the propulsion force. The calculation of this resistance R plays a crucial part in the selection process of a propeller and consequently the propulsion engine.

19.6.1 General

The ship's resistance is partially influenced by the ship's speed, displacement and the hull shape. The total resistance R_T, consists of various resistances R, divided into three groups:

1 frictional resistance;
2 residual resistance;
3 air resistance.

A heavily loaded heavy-cargo ship has a large air resistance.

This may become increasingly large when operating in head winds.

The effects of the frictional and residual resistances depend on the shape of the hull and the draught; the air resistance effects are dependent on the surface area of the ship above the waterline. In container ships, or heavily laden ships transporting large objects on deck, the air resistance is high.

Water with a velocity V and a density ρ has a dynamic pressure of: $\frac{1}{2} \times \rho \times V^2$ (Bernoulli's principle). If water comes to a complete standstill as the result of an object (in this instance a hull) moving through it, the drag load of the water on the hull is equal to the dynamic pressure exerted on the hull, which results in a dynamic force, namely pressure x per surface area, on the hull.

This fact forms the basis for calculating and or measuring the source resistances R of a ship's hull, using dimensionless resistance coefficient C_ρ as related to the force K, which is dependent on the dynamic water pressure at a certain speed V on a surface area which in turn is equal to the hull's wet surface A_S. The rudder surface is also included in the wet surface area.
The general formulas for calculating ship's resistances are:

Reference force = $K = \frac{1}{2} \times \rho \times V^2 \times A_S$ and source resistance: $R = C \times K$

Based on many model resistance tests in towing tanks and using the corresponding dimensionless hull data, methods have been developed to calculate the required resistance coefficients C, and consequently the corresponding source resistances R.
In practice, the calculation of the ship's resistance for a particular ship design for a corresponding model is performed in a towing tank and provides accurate data.

19.6.2 Frictional resistance R_F

The frictional resistance R_F of the ship's hull is dependent on the wet surface area A_S of the hull and the specific frictional coefficient R_F.
The resistance increases as a result of fouling, the growth of algae, weeds and barnacles on the ship's hull. Many anti-fouling painting systems on the market limit or prevent fouling of the hull. These often contain tributyl tin compounds (T.B.T.), which are extremely toxic.
IMO has prohibited these paint systems for new building as of the first of January 2003, and a general prohibition was implemented on the first of January 2008.
T.B.T. alternatives are available, such as copper-based anti-fouling paints. However, these appear to be less effective.

When a ship moves through the water, the frictional resistance increases to approximately the square root of the ship's velocity.

The frictional resistance of the ship in the water is approximately 70 to 90% of the total resistance to which the ship is subjected. This applies to large low-speed ships, such as bulk carriers and oil tankers (deep draught).
For large high-speed cruise ships of relatively small draught, this can be less than 40%.

▲
Frictional resistance R_F. High-speed, large container ships have propulsive power outputs up to approximately 70,000 kW.

With velocities up to 25 knots, more power is required. Container ships are always loaded; containers are added and removed, but these ships rarely sail unloaded. The propeller load and therefore the engine load are generally identical.

19.6.3 Residual resistance R_R

The residual resistance R_R comprises the wave and eddy resistance. Wave resistance refers to the energy loss caused by waves produced by the ship travelling through the water, whereas eddy resistance leads to power loss, when the current separates from the surface of a hull and eddies are formed. These eddies are prevalent at the stern. Wave resistance at low speeds is proportional to the square root of the velocity. However, it increases rapidly at higher speeds.

▼
The 'Berge Stahl' with a load capacity of more than 300,000 tonnes and a draught of 19 meters is the largest bulk carrier in the world. Large tankers and bulk carriers have low operational speeds. In loaded condition, the hull is 90% below the waterline.

To achieve high speeds at these displacements and breadths amounting to 30 to 70 metres would require huge power outputs.

▼
The picture shows an empty tanker in ballast, the bow and the rudder partially protruding from the water. The propeller operates just below the water surface.

The frictional resistance ca is formulated by: $R_F = C_F \times K$.

This essentially means that a speed restriction is created, the so-called the maximum hull velocity. A further increase of the diesel engine output above this velocity will not result in increased ship's speed!
All the extra power is converted into wave energy!
The frictional resistance amounts to approximately 8 to 25% of the total resistance for low-speed ships.
For high-speed ships, this can rise from 40 to 60% of the total resistance.
Shallow water also has a significant effect on the frictional resistance, as it is more difficult for the water displaced by the ship to flow astern. (Resistance of the displaced water volume over the ground).

The specific frictional resistance is defined as follows:
$R_R = C_R \times K$.

19.6.4 Air resistance R_A

In principle, the air resistance in calm water is proportional to the square of the ship's velocity and proportional to the cross-sectional surface area of the ship above the waterline.
Normally, air resistance is approximately 2% of the total resistance.

▼

Air resistance R_A. In relation to their displacement, passenger ships, cruise ships and ferries have tall superstructures and are therefore sensitive to (head) winds.

For highly laden container ships in a head wind, air resistance can rise to 10% of the total resistance. This also applies for large passenger liners, and in particular, heavily laden freighters.

Similar to the aforementioned resistances, air resistance is expressed as: $R_A - C_A \times K$, but can occasionally be based on 90% of the dynamic air pressure with a velocity V. In this case, the formula is as follows:
$R_A = 0{,}90 \times ½ \times \rho_{AIR} \times V^2 \times A_{AIR}$, where ρ_{AIR} constitutes the specific air mass and A_{AIR} the cross-sectional surface area of the ship above the waterline.

19.6.5 Towing resistance R_T and the effective (towing) power R_E

The total towing resistance R_T is calculated as follows: $R_T = R_E + R_A$.
A corresponding effective (towing) power, P_E, is required to move the ship or allow it to be towed through the water. At a velocity V, the formula is:

$P_E = V \times R_T$.

However, the power supplied to the propeller, P_D, required to move the ship at velocity V, is slightly higher. This is due to the flow conditions around the revolving propeller and propeller efficiency itself, which will be discussed later in this chapter.

19.6.6 General total ship resistance

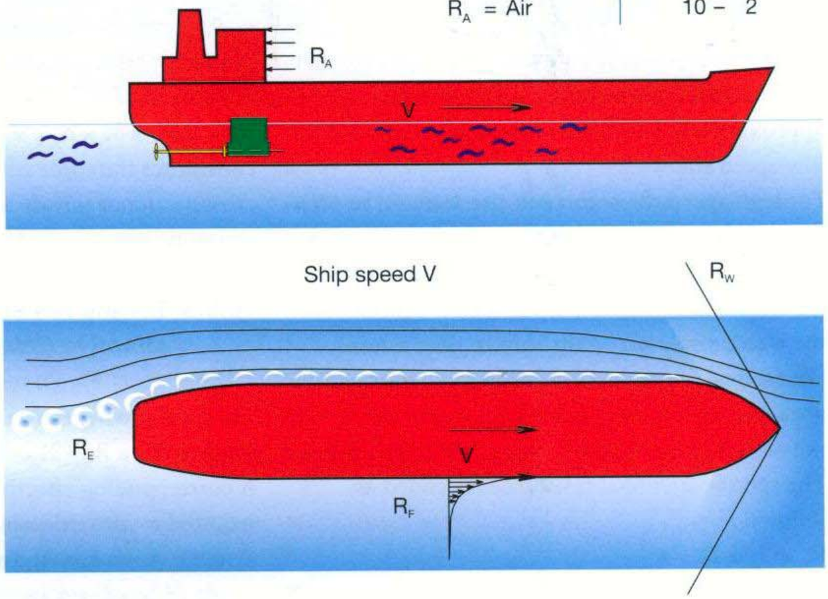
If the frictional resistance is divided into wave and eddy resistances, as described earlier, the distribution of the total towing resistance R_T can be expressed in percentages (see figure). This is applicable to both low-and high-speed ships.

▶

The total resistance of ships in general.

The right column shows low-speed ships, such as bulk carriers and tankers, the left column is valid for cruise ships and ferries. Container ships may sometimes be placed in this column or between the two columns.
The main difference between both columns is, as previously mentioned, the wave resistance.
In conclusion, all resistances are proportional to the square of the ship's velocity, but for higher speeds, the wave resistance increases more rapidly and therefore takes up a larger portion of the total resistance.

Type of resistance	% of R_T High speed ship	% of R_T Low speed ship
R_F = Friction	45	90
R_W = Wave	40	5
R_E = Eddy	5	3
R_A = Air	10	2

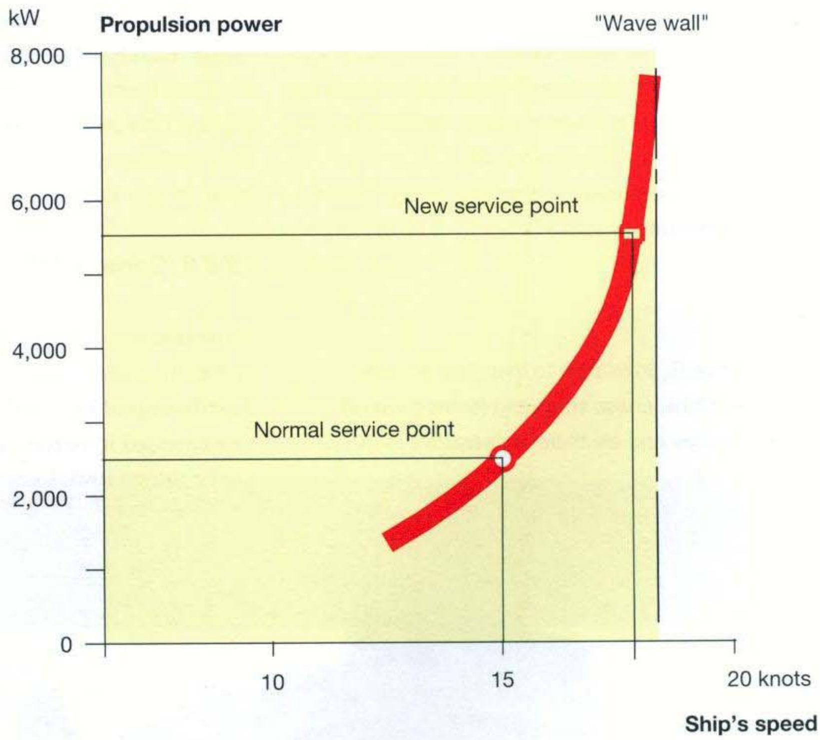


▶

This tendency is also clearly visible for a 600 TEU containership, originally designed for a service speed of 15 knots.

Without any alterations to the hull design, the speed for a sister ship of similar size was requested to be increased to approximately 17.6 knots. However, this would result in a relatively high wave resistance and would require **twice as much** propulsive power!

A further increase of the propulsive power would only produce a slight increase in the ship's velocity as the main part of the additional propulsive power is converted into wave energy.
The ship has reached the maximum hull velocity for the hull design and only produces a so-called 'wave wall'. The shape of the ship's hull will have to be modified to obtain the required speed increase.
At present (2008), there is a tendency to slightly reduce the service velocity of ships in order to reduce fuel costs.



Power and speed relationship for a 600 TEU container ship

Estimates of average increase in resistance for ships navigating the main routes	
North Atlantic route, navigation westward	25 – 35%
North Atlantic route, navigation eastward	20 – 25%
Europe – Australia	20 – 25%
Europe – East Asia	20 – 25%
The Pacific routes	20 – 30%

▲
Obviously, resistance is increased by (heavy) sea and wind conditions and currents.

In comparison to ship's resistance in calm weather conditions, heavy seas and head winds can increase the total ship's resistance by 50 to 100%.
On North Atlantic routes, the first percentage of a 50% increase corresponds to the summer period and the second percentage of a 100% increase to the winter period.
Data reveal that for a typical 140,000-ton (dead weight) bulk carrier on certain routes, especially on a northern route in the Pacific Ocean between Japan and Canada, the increased resistance when loaded shows extreme increases of 220% with a mean increase of 100%.

19.6.7 Ship resistance increase during operation

When the ship is in service, the paint coatings on the hull will gradually be attacked and degrade. Steel corrosion will slowly increase; weeds and barnacles will attach themselves to the hull.
Adverse weather conditions, perhaps together with a badly laden ship, can cause deformation of the bottom plates of the ship. This becomes evident by the beams which look like ribs protruding from the ship's hull. The hull becomes fouled and its surface is no longer smooth, which consequently results in a frictional-resistance increase. The propeller surface can become rough and fouled.
In calm water, the resistance caused by fouling can result in a total ship's resistance increase of 25 to 50%.
Generally, the larger the ship, the less the increase in resistance in the water.
However, the resistance of large, laden ships of deep draught, such as bulk carriers and tankers, may alter rapidly with hull fouling below the waterline.

In Summary

In principle and theory, the resistance increase in heavy weather is proportional to:
- the amount of adverse wind and current;
- the size of the waves.

It is difficult to distinguish between these two factors.

19.7 Screw propulsion

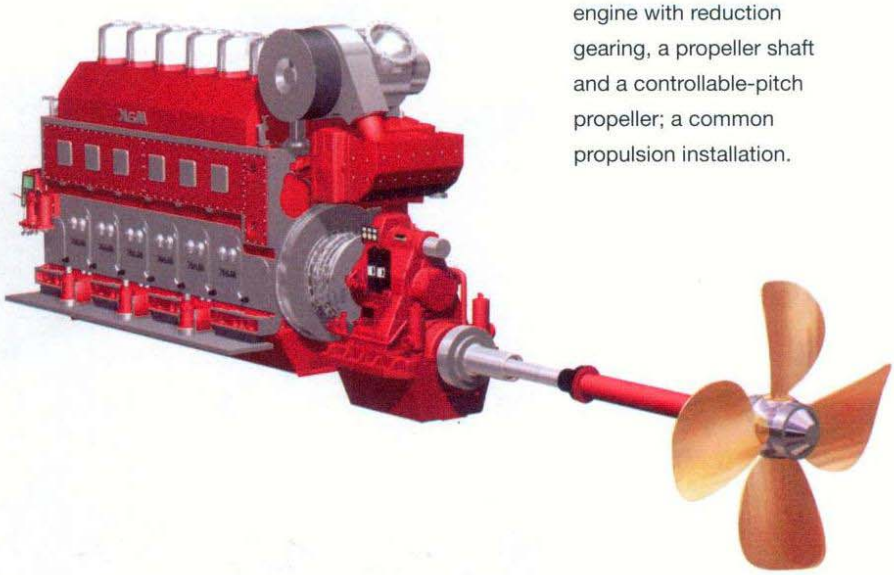
The traditional method of marine propulsion is by means of a propeller. This has been the most widely used method of propulsion for most ships for over a hundred years. Sometimes two and sporadically three propellers are fitted.

The required thrust T for the propeller enables the ship to move at a velocity V and this is usually higher than the generated towing resistance R_T. As described earlier, this divergence is caused by the water-flow differences for propulsion and towing conditions.

19.8 Propeller types

Propellers can be divided into two main groups:
1 fixed-pitch propellers (**FPP**);
2 controllable-pitch propellers (**CPP**). These are also known as **AP** (adjustable pitch) and **VP** (variable pitch) propellers.

▼
A medium-speed Caterpillar–Mak diesel engine with reduction gearing, a propeller shaft and a controllable-pitch propeller; a common propulsion installation.



Velocities

Ship's speed	: V
Arriving water velocity to propeller (Speed of advance of propeller)	: V_A
Effective wake velocity	: $V_W = V - V_A$
Wake fraction coefficient	: $\frac{V - V_A}{V}$

Forces

Towing resistance	: R_T
Thrust force	: T
Thrust deduction fraction	: $F = T - R_T$
Thrust deduction coefficient	: $\frac{F = T - R_T}{T}$

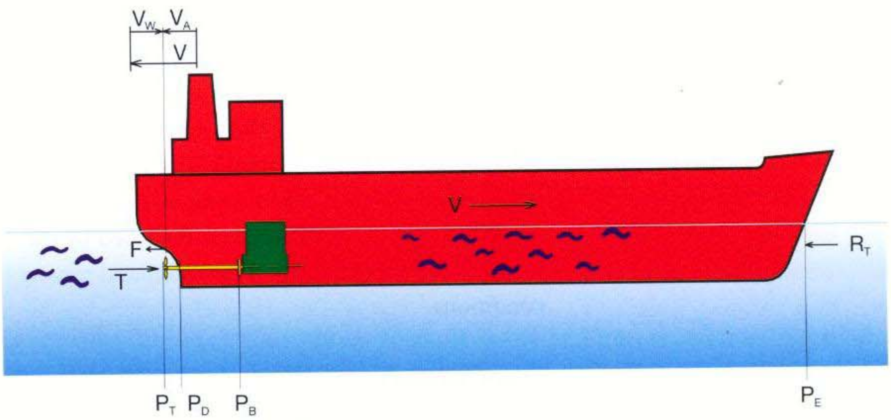
Power

Effective (towing) power	: $P_E = R_T \times V$
Thrust power delivered by the propeller to water	: $P_T = P_E / \eta_H$
Power delivered to propeller	: $P_D = P_T / \eta_B$
Brake power of main engine	: $P_B = P_D / \eta_S$

Efficiencies

Hull efficiency	: $\eta_H = \frac{1 - t}{1 - W}$
Relative rotative efficiency	: η_R
Propeller efficiency – open water	: η_0
Propeller efficiency – behind hull	: $\eta_B = \eta_0 \times \eta_R$
Propulsive efficiency	: $\eta_0 = \eta_H \times \eta_B$
Shaft efficiency	: η_S
Total efficiency	: η_T

$$\eta_T = \frac{P_E}{P_B} = \frac{P_E}{P_T} \times \frac{P_T}{P_D} \times \frac{P_D}{P_B} = \eta_H \times \eta_B \times \eta_S = \eta_H \times \eta_0 \times \eta_R \times \eta_S$$



▲
The propulsion of a ship-theory.

▶
Winter in the Baltics

The resistance of ice floes is not calculated in this chapter of ship propulsion.

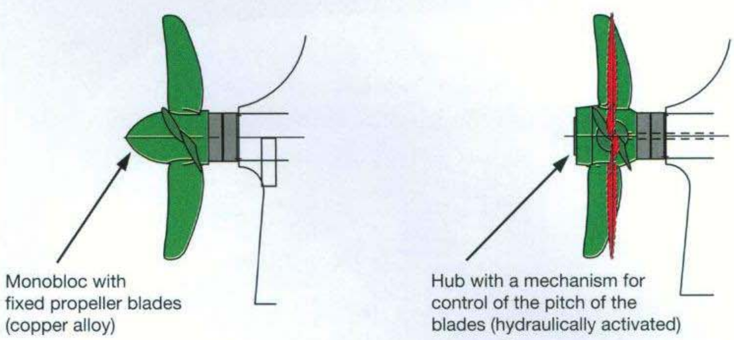
19.8.1 Fixed-pitch propellers

FP-propellers are cast in one block and usually made of a very strong copper alloy. The position of the blades and therefore the propeller pitch is fixed and cannot be modified. When operating in, for instance, poor weather conditions, propeller performance properties such as thrust will only alter in combination with the supplied engine power and the propeller speed.
Most ships not requiring sophisticated manoeuvring capabilities, such as container ships, bulk carriers and tankers are equipped with FP-propellers.
These generally travel great distances on the high seas and do not berth frequently. These ships can safely enter ports to load and discharge aided by their bow thrusters (container ships) and by tugs.

19.8.2 Controllable-pitch propellers

Controllable-pitch propellers have in comparison a larger hub as space is required to house the hydraulic mechanism that controls the pitch (angle) of the adjustable blades. The price of a controllable-pitch propeller can be three to four times that of a fixed-pitch propeller.
Furthermore, the propeller efficiency is somewhat lower due to the relatively larger hub.
The propeller is also more vulnerable in service than that of a fixed-pitch propeller.

Controllable-pitch propellers are often applied to ships that require frequent manoeuvring action. Ferries, container- feeder ships, dredgers, Ro-Ro ships and a great many other, slightly smaller ships use CP-propellers.



▲ Propeller types.

left: fixed-pitch propeller
right: controllable-pitch propeller

Another important advantage is that a directly reversible engine is no longer required. Thus the diesel-engine design remains simple and it operates at a high, fixed RPM, rapidly producing the desired power for the propeller during manoeuvring. So manoeuvring can take place in a safe and swift manner.

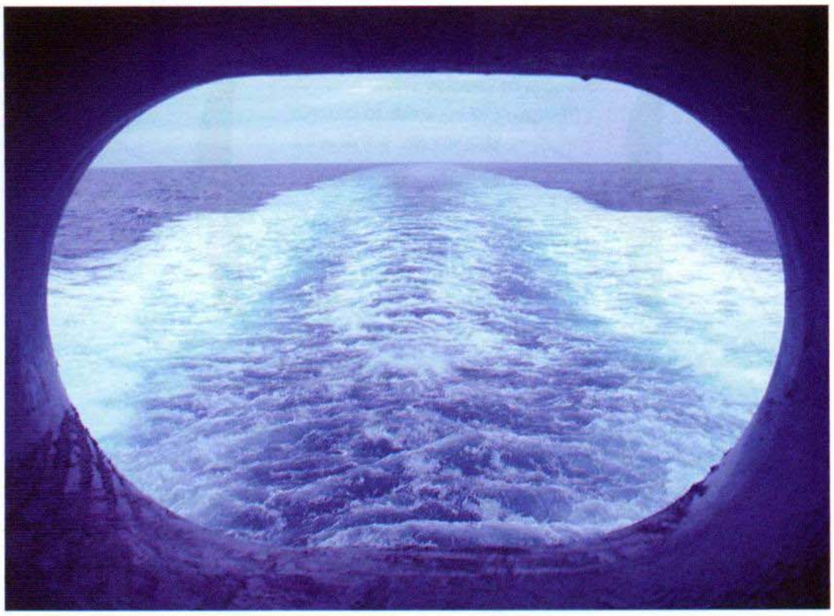
For further details regarding propellers, propeller shafts and seals, see Chapter 31, Propellers.

19.9 Flow conditions around the propeller

19.9.1 Wake coefficient w

When the ship moves through water, the friction of the hull will produce a 'friction layer' of water on the surface of the hull. In this 'friction layer' the water velocity at the surface of the hull is equal to that of the ship, but is reduced as the water distances itself from the hull surface.

◄ A large six-blade fixed-pitch propeller for a large containership (Wärtsilä Propulsion).



▲
The wake fraction after a large containership; the 'Norasia Enterprise' at an operational speed of 26 knots.

The enormous displacement of the ship's hull caused by the tremendous speed creates a 'pit' of about 1 meter behind the ship. As shown here, the water cannot immediately fill up the pit from the sides. The eddies in the water generated by the rotating propeller are also clearly visible.

Knots for ship's velocities:
1 knot = 1 nautical mile per hour = 1852 metres per hour

The water velocity equals zero at a certain distance from the hull equalling the outer surface of the friction layer.
This is the reason that water near the propeller has a wake velocity in the same direction as the ship's velocity V. This means that the speed of the arriving water, V_A, at the propeller, given as the average velocity over the propeller's wake area V_W, **is lower** than the ship's velocity V.

Therefore, the effective wake velocity at the propeller is equal to $V_W = V - V_A$ and is expressed as a dimensionless number by means of the wake coefficient w. The generally applied wake coefficient by Taylor is defined as follows:

$$w = \frac{V_W}{V} = \frac{V - V_A}{V}$$

The value of the wake coefficient is largely dependent on the shape of the hull, but also on the location of the propeller and its size, and it affects propeller efficiency.

The propeller diameter or rather the ratio of the propeller diameter d and the ship's length L_{WL} somewhat affects the wake fraction coefficient as $\frac{d}{L_{WL}}$ and gives a rough indication of the way in which the propeller performs in the wake field of the hull. The higher the ratio d to L_{WL}, the lower the wake fraction coefficient w.
Obviously, the wake fraction coefficient w increases when the hull is fouled.
For single-screw ships the wake fraction factor w is usually in the 0.20 to 0.45 region, with a corresponding flow velocity to the propeller of 0.80 to 0.55 of the ship's velocity V. The larger the block coefficient, the larger the wake fraction coefficient.
On twin-screw vessels with a conventionally shaped stern, the propellers are normally positioned outside the ´friction layer´, which results in a significantly lower wake fraction coefficient w.
A high wake fraction coefficient increases the risk of propeller cavitation, as the water speed around the propeller is often irregular and heterogeneous. A more homogeneous wake field for the propeller, involving a higher speed of advance of the propeller $V_{A,}$ is at times required and can be obtained in various ways, for instance by placing the propellers in nozzles.
Nevertheless, the preferred method is to design the stern is such a way that an optimum wake fraction is achieved.

19.9.2 Thrust-deduction coefficient

Screw rotation draws the water forward astern towards the propeller. This results in an extra resistance on the hull, called 'augment of resistance' or, in relation to the total required thrust force T on the propeller, 'thrust deduction fraction' F. This means that the thrust force T on the propeller has to overcome the ship's resistance R_T as well as this 'loss of thrust' F.
The thrust deduction fraction F is expressed as a dimensionless number.

The thrust deduction fraction t is defined as:

$$t = \frac{F}{T} = \frac{T - R_T}{T}$$

This thrust deduction coefficient t is calculated with calculation models established by research carried out on various hull shapes.

Generally, the thrust deduction coefficient increases as the wake fraction coefficient w increases.
The shape of the hull plays a key role; a bulbous stem can decrease the thrust deduction coefficient at, for instance, low speeds.
The thrust deduction coefficient t for a single-screw ship is usually in the region of 0.12 to 0.30 as a ship with a large block coefficient has a high thrust deduction coefficient.
For twin-screw vessels with a conventional stern, the thrust deduction coefficient t is much lower as the propellers draw water further away from the hull.

19.9.3 Efficiencies

The hull efficiency η_H is defined as the ratio between effective (towing) power $P_E = R_T \times V$, and the thrust power which the propeller delivers to the water $P_T = T \times V_A$, from which follows that:

$$\eta_H = \frac{P_E}{P_T} = \frac{R_T \times V}{T \times V_A} = \frac{R_T / T}{V_A / V} = \frac{1 - t}{1 - w}$$

For a single-screw vessel, the hull efficiency η_H is normally in the region of 1.1 to 1.4 with high values for ships with high block coefficients.
For twin-screw ships with a conventional stern, the hull efficiency η_H is approximately 0.95 to 1.05, again with the highest values for ships with high block coefficients.

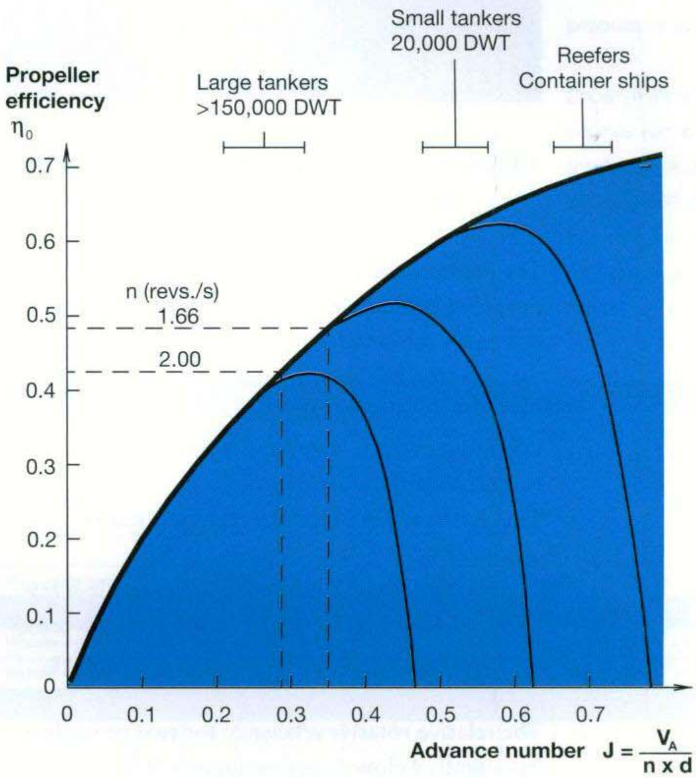
19.9.4 Open-water propeller efficiency η_O

The propeller efficiency η_O **is related to working in** open water, which is to say, the propeller operates in a homogeneous wake field with no hull in front of it.
The propeller efficiency is dependent on the speed of advance V_A, the thrust force T, the rate of revolution of the propeller per time unit (minute), the propeller diameter, the design of the propeller. Considerations are:

- the number of blades;
- the surface area of the propeller blades in relationship to the total circular surface area of the propeller circle;
- the pitch/diameter ratio, which will be discussed later in this chapter.

The propeller efficiency η_O may vary from 0.35 to 0.75, with high values for fast-running propellers.

As a rule of thumb, it can be stated that roughly a third of all the energy supplied to the engine in the form of fuel is used to provide the ship with a particular speed.



▲
The obtainable propeller efficiency η_O in relation to the speed of advance V_A.

horizontal: $J = \frac{V_A}{n \times d}$

J is **the advance number** of the propeller and is dimensionless.

vertical: propeller efficiency η_O
Furthermore, the rate of revolution per second; this is clearly a table for fixed-pitch propellers of large ships, directly driven by two-stroke crosshead engines.
So the efficiency of a propeller varies from 45 to 70%. This means that at a total engine efficiency of 50% the total propulsive efficiency is much lower.
Let's assume that the total engine efficiency is 50% and the propeller efficiency 65%.
Then the total propulsive efficiency from the fuel (100%) is 0.5 x 0.65 = 0.3250 or 32.5%.

◄
A picture of a large containership with a speed of 27 knots.

The bow wave 1 is relatively small. The propeller water 2 with the wake fraction can clearly be seen in the picture. This container ship of 5500 TEU produces fewer waves at sea than a cutter at full speed!

19.9.5 Relative rotative efficiency η_R of the propeller

The actual velocity of water flowing to the propeller behind the hull is not constant and does not reach the circular propeller surface area at the correct angle, but has a rotational flow. This explains the discrepancy between a propeller operating in open water and a propeller operating behind a ship's hull.
This difference is called the relative rotative efficiency η_R.
The relative rotative efficiency η_R on single-screw vessels lies between 1.0 and 1.07. In other words, the rotating motion of the water has a positive effect.
The relative rotative efficiency for two propellers lies slightly below 1, approximately 0.98.
Combined with w, the wake fraction factor, and t, the thrust deduction coefficient, the relative rotative efficiency η_R is often used to compare the towing-tank tests with theory.

19.9.6 Propeller efficiency η_B operating astern

The ratio between the thrust power P_T, which is delivered to the water by the propeller, and the power P_D, which is delivered to the propeller by the propulsion engine is indicated by the propeller efficiency η_B, and defined as follows:

$$\eta_B = \frac{P_T}{P_D} \times \eta_O \times \eta_R$$

19.9.7 Propulsive efficiency η_D

The propulsive efficiency η_D, not to be confused with the propeller efficiency η_O in open water, is equal to the ratio between the effective (towing-) power P_E and the required power delivered to the propeller or:

$$\eta_D = \frac{P_E}{P_D} = \frac{P_E}{P_T} \times \frac{P_T}{P_D}$$

$$\eta_D = \eta_H \times \eta_B = \eta_H \times \eta_O \times \eta_R$$

As can be seen the propulsive efficiency η_D is equal to the product of the hull efficiency η_H, the open-water propeller efficiency η_O and the relative rotative efficiency η_R of the propeller, in which the latter is less important.
In view of this, it can be inferred that a ship's hull with a high wake-fraction coefficient w, and consequently a high hull efficiency η_H provides the best conditions for the highest propulsive efficiency η_D.
However, the open-water propeller efficiency η_O is also dependent on the speed of advance V_A which increases as the wake-fraction coefficient w rises, and therefore often the opposite effect is achieved.

Generally, the best propulsive efficiency is achieved when the propeller operates in a homogeneous wake field.

19.9.8 Shaft efficiency η_S

The shaft efficiency η_S is, amongst others, dependent on the alignment and the lubrication of the shaft bearings and on the reduction gearing, if any.

The shaft efficiency η_S is equal to the ratio of the power P_D transferred to the propeller and the shaft power P_B delivered by the main engine.

$$\eta_S = \frac{P_D}{P_B}$$

The shaft efficiency normally amounts to about 0.985, but may vary from 0.96 to 0.995.

19.9.9 Total propulsive efficiency η_T

The total propulsive efficiency η_T, is equal the ratio between effective (towing) power P_E and the brake power supplied to the propulsion engine η_B and is expressed as follows:

$$\eta_T = \frac{P_E}{P_B} = \frac{P_E}{P_D} \times \frac{P_D}{P_B}$$

$$\eta_T = \eta_D \times \eta_S = \eta_H \times \eta_O \times \eta_R \times \eta_S$$

19.10 Propeller dimensions

Propeller diameter d.

The largest propeller diameter is normally selected in order to achieve the largest possible propulsive efficiency η_D. However, a number of conditions must be considered.
For instance: the shape of the hull at the stern of the ship may vary greatly and is predominantly dependent on the type of ship, the ship design and the required clearance between the tips of the propeller blades and the ship's hull. This is closely related to the propeller type.

For bulk carriers and tankers, which often sail in ballast conditions, a request for a fully immersed propeller is an option.
Container ships do not require a maximum size as they seldom sail in ballast conditions.
The above-mentioned factors attest to the fact that it is relatively difficult to calculate an exact diameter for the propeller. However, the rule of thumb with regard to propeller size is as follows:

The ratio between the propeller diameter and the draught of the ship can be taken into consideration and a large propeller diameter will result in a low speed of rotation.

For bulk carriers and tankers, the following applies:

d/D < approximately 0.65.

For container ships:

d/D < approximately 0.74.

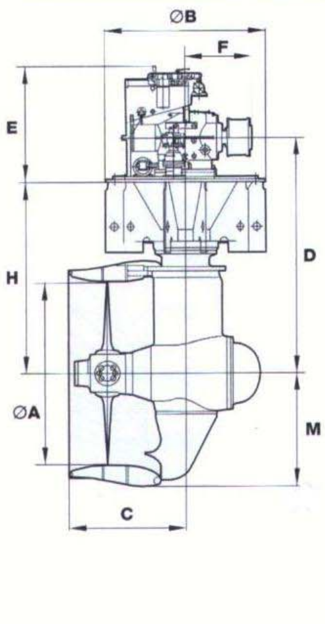
For strength and manufacturing purposes, the propeller diameter will seldom exceed 10 metres and the power output is in the region of 90,000 kW.
These values were achieved in the 'Emma Maersk' in 2006!
The largest propeller diameter (2007) is 11 metres and the propeller has four blades!

▼
Propac; a Wärtsilä propulsive system.

Shown here for in-line engines with 6, 8 and 9 cylinders and bores of 200 and 260 millimetres. Including the nozzle propeller systems with all data.

Propac ST selection table

Engine type		6L20		8L20		9L20		6L26		8L26		9L26	
MCR engine power CPP	kW	1080		1440		1620		2025		2600		2925	
	HP	1469		1958		2203		2754		3536		3978	
MCR engine power FPP	kW	1080		1440		1620		1950		2480		2790	
	HP	1469		1958		2203		2652		3373		3794	
MCR engine speed	rpm	1000		1000		1000		1000		1000		1000	
Thruster type		200		225		225		250		275		300	
Propeller diameter	mm	1900	2100	2100	2300	2100	2300	2400	2600	2600	2800	2800	3000
Propeller speed	rpm	318	318	248	248	274	274	257	257	245	245	215	215
Bollard pull with CPP													
in 19A nozzle	tonne	35	37	46	49	50	53	63	66	79	82	90	93
in HR nozzle	tonne	39	40	51	53	55	57	69	71	86	89	99	102
Bollard pull with FPP													
in 19A nozzle	tonne	35	36	46	49	50	52	62	64	76	79	87	90
in HR nozzle	tonne	38	39	51	53	55	57	67	69	83	85	95	98
MCD-type for steerable thrusters with FPP													
LD type		3000-3	3000-3	3000-4	3000-4	3000-4	3000-4	3000-5	3000-5	3000-6	3000-6	not available ->	
HD type		3000-3	3000-3	3000-6	3000-6	3000-7	3000-7	3000-7	3000-7	3000-8	3000-8	not available ->	

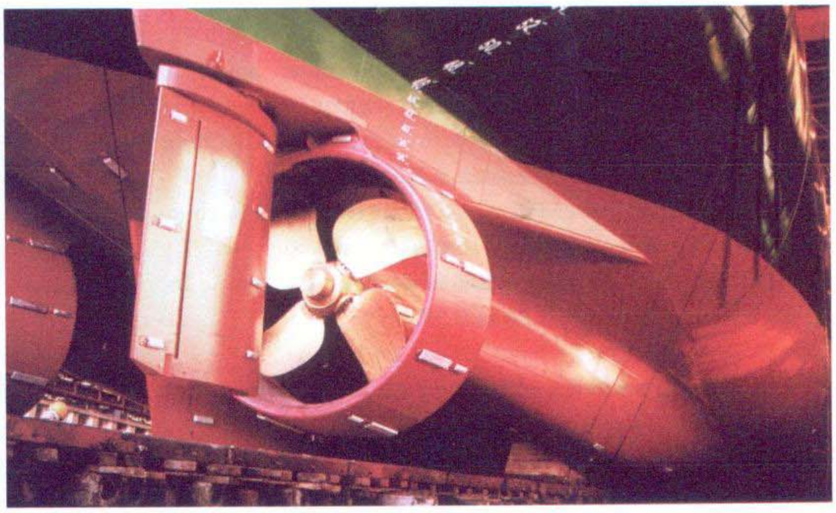


Propac ST main dimensions

Thruster type FS/CS	A mm	B mm	C mm	D mm	E mm	F mm	H mm	M mm
175	1600	1600	1200	2100	910	610	1743	985
	1800		1230	2200			1843	1109
200	1900	1900	1300	2500	1200	740	2048	1179
	2100		1350	2600			2148	1303
225	2100	2100	1400	2630	1210	740	2160	1299
	2300		1450	2830			2360	1423
250	2400	2100	1525	3100	1435	840	2512	1485
	2600		1575	3200			2612	1614
275	2600	2850	1665	3500	1465	877	2890	1609
	2800		1765	3620			3010	1731
300	2800	2850	1770	3700	1465	877	3090	1735
	3000		1870	3700			3090	1858

▶

Shown here it is clearly visible that the water should advance towards the propeller as smoothly as possible, so without eddies. The propeller rotates completely 'free' under the cut-away section of the ship.



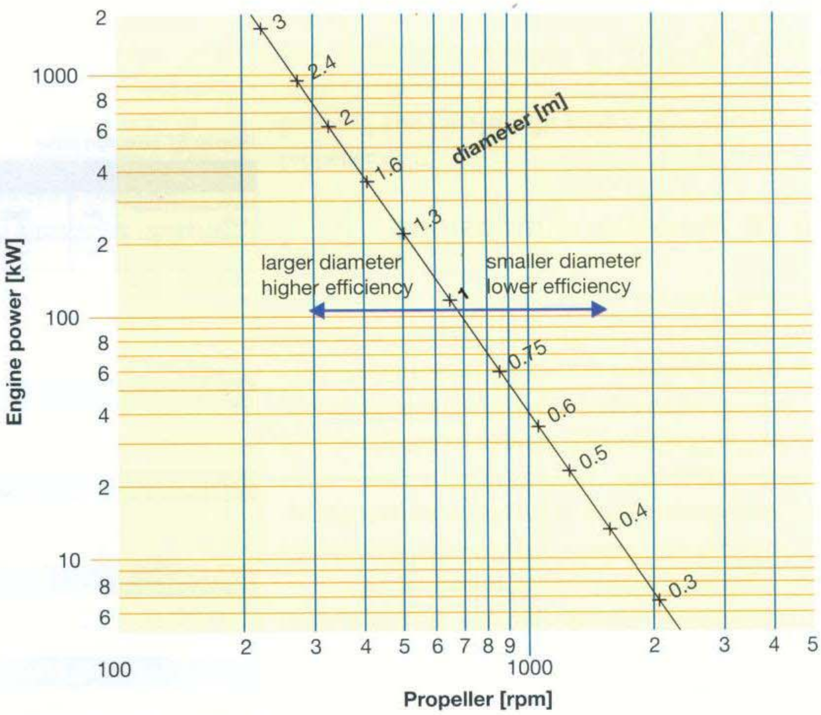
▶

This graph represents the ideal propeller diameter, here between 0.3 metres (small ships, yachts) to 3 metres (larger ships, larger fishing craft).

horizontal: propeller speed of 100 to 5000

vertical: power in kW to 2000 kW

slanted line: This represents the ideal propeller diameter of, here, 0.3 metres (small ships, yachts) up to 3 metres (larger ships, fishing craft). The larger the propeller diameter, the higher the propeller efficiency. The smaller the propeller diameter, the lower the propeller efficiency.

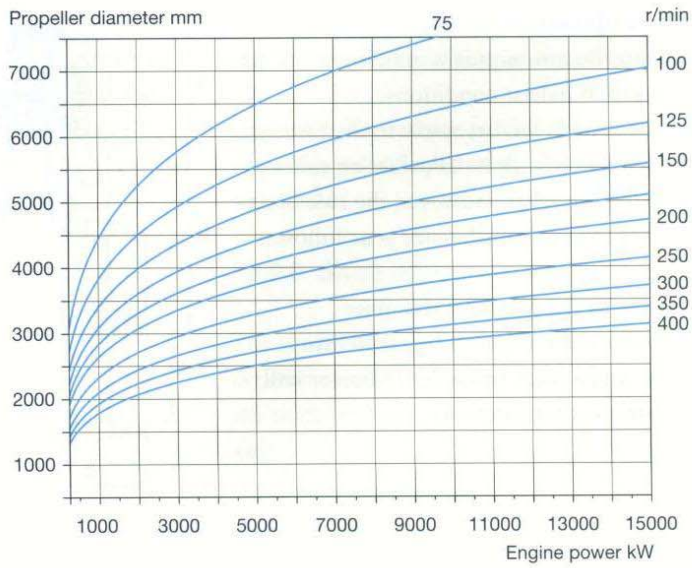


CIPS fixed pitch propeller sizes

▶

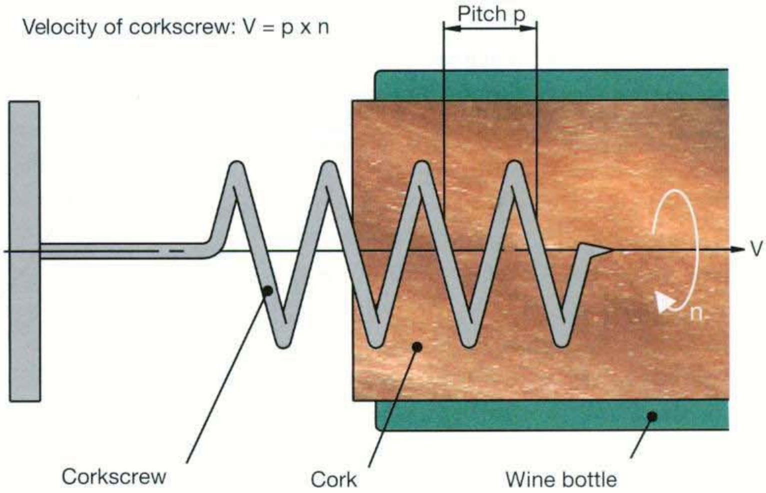
A graph by MAN–B&W showing the optimum ratio of the propeller diameter (vertical) to the engine power (horizontal) and the speed (top right).

Example: At a propeller diameter of 6000 mm (6 metres) and a speed of 100, a shaft power of 7000 kW is required. An optimum propeller efficiency of approximately 70% is obtained.



▶
A rotating 'propeller' in a solid mass (here cork).

The 'propeller' speed is equal to the distance travelled V = P1 × 1. There is no slip.



The apparent slip, which is calculated by the crew, provides valuable information as it gives the propeller load in varying operating conditions.
The apparent slip increases when:
- the ship sails against the wind and/or waves;
- the ship sails in shallow waters;
- the ship's hull is fouled;
- the ship accelerates.

Under increased resistance, this means that the propeller velocity has to increase to maintain the required ship speed.

19.11.2 Real slip S_R

The **real slip S_R**, is greater than the apparent slip S_A due to the fact that the actual speed of advance V_A of the propeller is smaller than the ship velocity V.
The real S_R, which provides a more accurate picture of the propeller's function, is:

$$S_R = 1 - \frac{V_A}{p \times n} = 1 - \frac{V \times (1 - w)}{p \times n}$$

If a ship is moored (at quay trials) and the propulsion engine operates at full load, the ship's velocity V equals 0 and the real slip 1.0. The slip is sometimes given in percentages.

19.11.3 Propeller law in general

The resistance R for low ship speeds is proportional to the square of the ship's velocity: $R = c \times V^2$, where c is constant. The required power P is therefore proportional to the ship's velocity V to the power of three.
$P = R \times V = c \times V^3$

For a ship with a fixed-pitch propeller, the ship's speed V is proportional to the size and the rate of revolution, so:

$P = c \times n^3$

which precisely expresses the propeller law which states: 'the required engine power output delivered to the propeller is proportional to the rate of revolution to the third power'.

Measurements demonstrate that the relation between the power output and the engine speed for certain weather conditions is fairly congruent, whereas the relation between the power output and the ship's velocity is often higher than the V to the power of three.
A reasonable relationship to be used for estimations in normal ship velocities could be as follows:

For large high-speed container ships: velocity ± 22 – 26 knots
$P = c \times V^{4.5}$

For medium sized, medium-speed ships such as feeder container ships, RoRo ships etc.: velocity ± 16 – 20 knots.
$P = c \times V^{4.0}$

For low-speed ships such as tankers and bulk carriers and small feeder container ships: velocity ± 12 – 14 knots.
$P = c \times V^{3.5}$

19.10.1 Number of propeller blades

Propellers can be manufactured with two, three, four, five and six blades. The fewer the number of blades the higher the propeller efficiency. For reasons of strength, two or three blades do not suffice for heavily loaded propellers.
Large ships, which are propelled by two-stroke crosshead engines generally, use four-bladed propellers. Large container ships with speeds reaching 25 knots and large propulsion engines sometimes require five- or six- bladed propellers. **Due to vibrations, propellers with a certain number of blades are not used in certain instances. Propellers with a different number of blades are opted for to avoid the generation of natural frequencies in the ship's superstructure.**

19.10.2 Disk-area coefficient

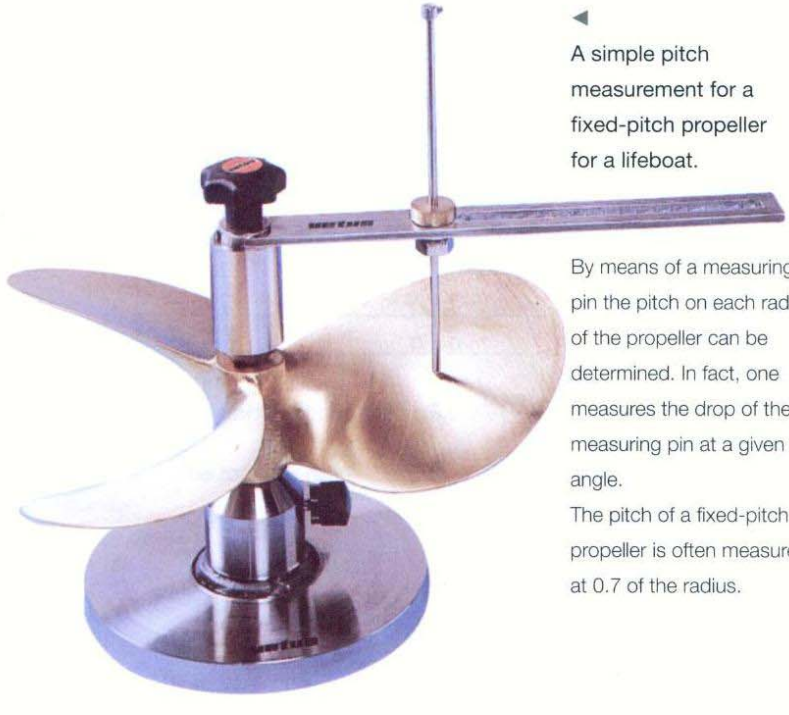
The disk-area coefficient, in older literature referred to as the expanded-blade area ratio, is the ratio between the surface of the propeller blade and its disk area. A factor of 0.55 is considered adequate.

For traditional four-bladed propellers, the disk-area coefficient is insignificant, as a value in excess of 0.55 will result in extra resistance on the propeller in the water and therefore produce a minimal increase in propeller efficiency.
Heavy propeller loads are found on large ships, they often have five- or six-bladed propellers and then the disk-area coefficients are higher. On naval vessels they can be as high as 1.2 (frigates).

19.10.3 Pitch/diameter ratio of the propeller

The pitch/diameter ratio $\frac{p}{d}$ expresses the ratio between the pitch of the propeller p and the propeller diameter d.
The pitch p is the distance that the propeller travels through the water per revolution, if there is no 'slip' (see next section).
As the pitch may vary along the length of the blade (the blade's radius), the ratio is normally related to the pitch at 0.7 of the diameter (0.7 r), where r = ½ d the radius of the propeller.
In order to obtain optimum propeller efficiency at a given propeller diameter, an optimum pitch/diameter ratio should be found, which in turn should correspond to a certain design rate of revolution.
If, for instance, a propeller is designed for a low revolution rate, the pitch/diameter ratio should be increased and vice versa for efficiency.
If a low design rate of revolution is required and providing that the draught of the ship allows this, it is possible to opt for a larger propeller diameter, which will effect an increase of the propulsive efficiency.

▼
Here the disk-area coefficient is high, about 0.7.



◄
A simple pitch measurement for a fixed-pitch propeller for a lifeboat.

By means of a measuring pin the pitch on each radius of the propeller can be determined. In fact, one measures the drop of the measuring pin at a given angle.
The pitch of a fixed-pitch propeller is often measured at 0.7 of the radius.

19.11.4 Propeller law for a propeller running under heavy conditions

Obviously, the propeller law can only be applied to identical operating conditions.
If, for instance, after a certain amount of time the hull is fouled and consequently has an increased water resistance, the wake field differs from the wake field of a 'clean' hull during for example, the trial runs.
A ship with a fouled hull is subject to extra resistance in the water, which effects an increased propeller load at an identical engine output. The rate of revolution will therefore be lower.
The propeller law is now applied to a different 'heavier' propeller curve than that of a 'clean' hull.
Similar considerations apply to ships labouring in heavy seas, in strong counter currents, in stormy weather and high waves. There will also be a heavier propeller operation in tail wind and high waves than in calm weather operation.

19.11.5 Ballast conditions

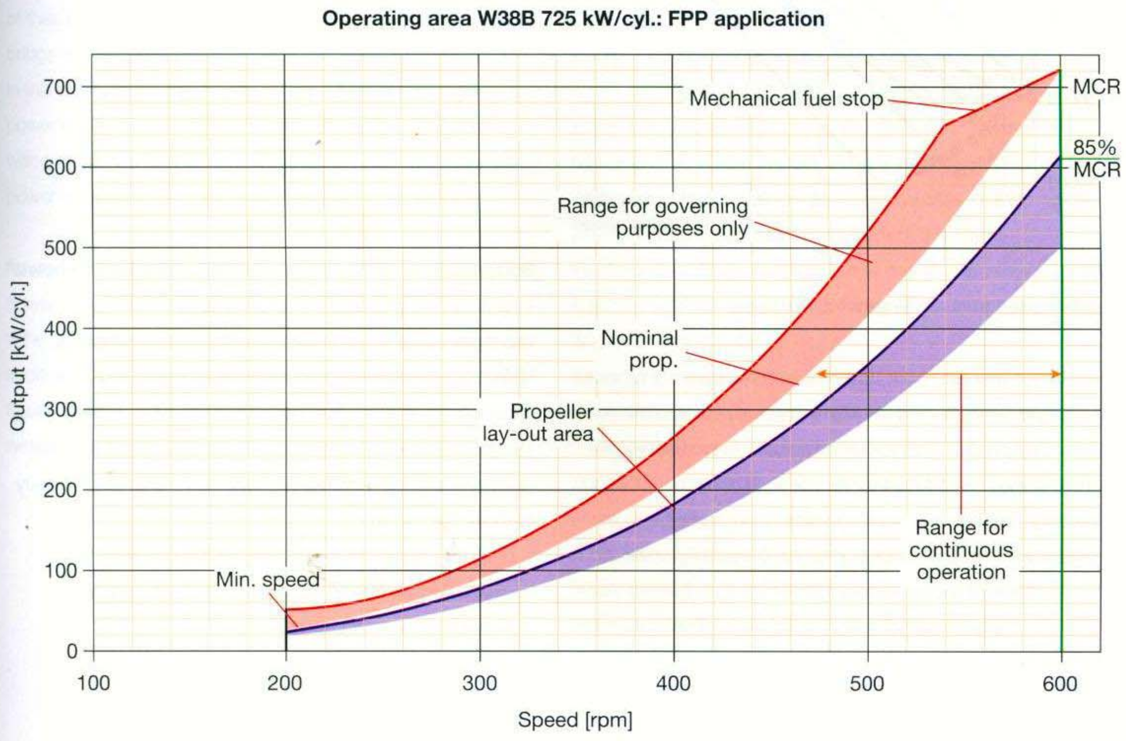
When a ships sails in ballast, which is to say, in lower displacement, the propeller law applies to a 'lighter' curve; at a similar power output the propeller speed will increase slightly.

The propeller law is predominantly applied to fixed-pitch propeller ships running at partial load. Diesel manufacturers' project guides and engine layout and load diagrams specify the engine's curves for both light- and heavy-running conditions.
These diagrams use logarithmic scales and straight lines.

▼

A load diagram for a four-stroke medium-speed Wärtsilä 38 B diesel engine operating on H.F.O. Propeller type: fixed-pitch propeller.

vertical: cylinder power in kW
horizontal: rate of revolution per minute (RPM)
minimum RPM: 200
maximum RPM: 600
MCR: at 600 revolutions and 725 kW per cylinder
normal service power: at 85% of MCR at 600 revolutions and 625 kW per cylinder
vertical hatched area: operating area for the propeller load on the engine
propeller hatched area: area in which the governor is activated
top: mechanical fuel stop



19.10.4 Propeller coefficients J, K_T and K_Q

Propeller theory is based on models, but in order to enable a general use of the theory a number of dimensionless propeller coefficients have been introduced, in relation to:

- the propeller diameter d;
- the rate of revolution n;
- the specific mass of the water ρ.

The three most important coefficients are discussed below.

The advance number of the propeller J is, as mentioned earlier, a dimensionless expression of the propeller velocity of advance V_A.

$$J = \frac{V_A}{n \times d}$$

The thrust coefficient is expressed dimensionless as follows:

$$K_T = \frac{T}{P \times n^2 \times d^4}.$$

And the propeller torque as:

$$Q = \frac{P_D}{2 \times \pi \times n} \qquad KQ = \frac{Q}{P \times n^2 \times d^5}.$$

The open-water propeller efficiency η_O can be calculated from the aforementioned coefficients, because as previously mentioned, the propeller efficiency η_O is as follows:

$$\eta_O = \frac{P_T}{P_D} = \frac{T \times V_A}{Q \times 2 \times \pi \times n} = \frac{K_T}{K_Q} \times \frac{J}{2 \times \pi}.$$

By means of special and complicated propeller diagrams, which, amongst others, include J, K_T and K_Q curves, it is possible to calculate/find the propeller sizes, the efficiency, the thrust pressure, the shaft power.

19.10.5 Manufacturing accuracy of propellers

ISO 484/1 - 1981 (CE)		
Class	Manufacturing accuracy	Mean pitch for propeller
S	Very high accuracy	+/- 0.5%
I	High accuracy	+/- 0.75%
II	Medium accuracy	+/- 1.00%
III	Wide tolerances	+/- 3.00%

▲

Prior to manufacturing a propeller, the accuracy class standard must be established.

Example ISO 484/1 - 1981 (CE) which has four different accuracy classes.
Each class describes, amongst others, the maximum tolerance of the average pitch of the manufactured propeller, including the maximum tolerance of the corresponding propeller speed (the rate of revolution).
The propeller price is dependent on the chosen accuracy class, with the lowest price for class III.
However, it is inadvisable to opt for class III as the tolerances (+ - 3%) are too high.
The tolerance of the pitch should not be in excess of +/- 1.0%.
The accuracy of manufacturing the propeller corresponds to the **accuracy** of the propeller speed (rate of revolution) tolerance and is +/- 1.0%.
Here the total accuracy of the propeller has a tolerance of +/- 2.0%.
This also corresponds with the operating conditions in heavy weather.

19.10.6 Influence of propeller diameter and the pitch/diameter ratio on the propulsive efficiency η_D

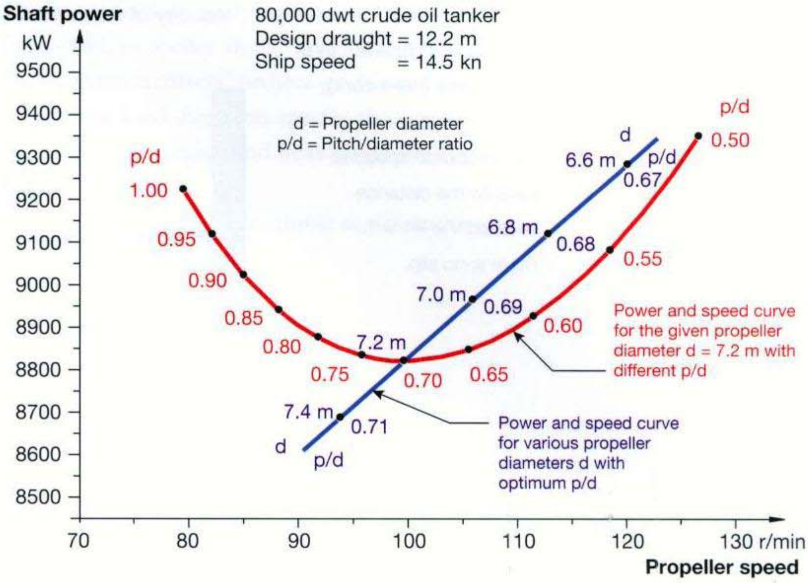
As mentioned earlier, the highest possible propulsive efficiency at a certain required velocity of the ship is achieved by using the largest possible propeller diameter d, in conjunction with the corresponding optimum pitch/diameter ratio.

▶

A 80,000 dwt crude-oil tanker, with a service speed of 14.5 knots and a maximum propeller diameter of 7.2 metres.

According to the **blue line**, a propeller with a maximum diameter of 7.2 metres has an optimum pitch/diameter ratio of 0.7 and the lowest possible shaft power of 8820 KW at a propeller speed of 100 r/min.
If the pitch for this diameter is altered, the propulsive efficiency is decreased, or the necessary shaft power will increase, see the **red curve**.

The **blue line** shows that when a larger propeller diameter is feasible, the required shaft power is decreased to 8690 KW at 94 r/min; in other words, the larger the propeller, the lower the optimum rate of revolution of the propeller.
The **red curve** shows that with regard to propulsion, it is always advantageous to use the largest possible propeller diameter even though the propeller speed could be too low at the optimum pitch/diameter ratio. This with relation to the required engine speed.



So, by choosing a slightly lower pitch/diameter ratio, comparable to the optimum ratio, the propeller and consequently the engine speed will increase and only require slight additional power increases.

19.11 Operating conditions of a propeller

19.11.1 Slip ratio S

▶

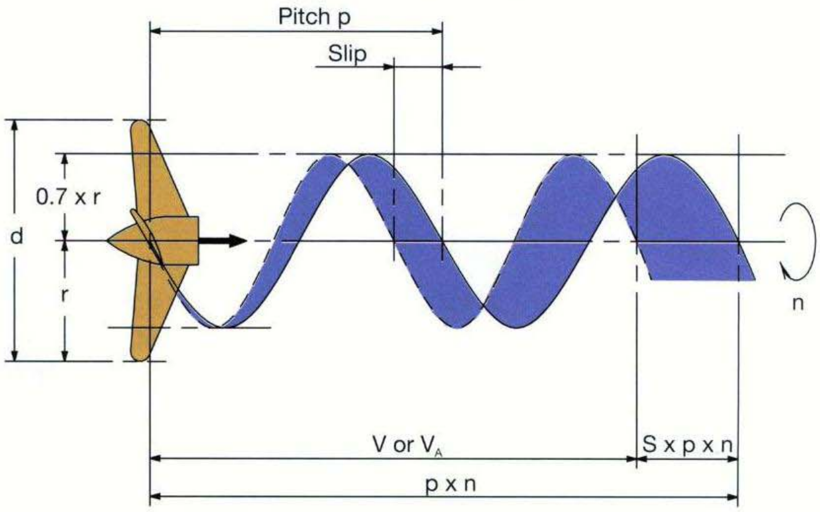
The slip S.

If the propeller had no slip in the water, or if the propeller did not thrust the water backwards, the propeller would move forward with a velocity of $V = p \times n$ in which n is the rate of revolution of the propeller and p the pitch of the propeller.

One can imagine a similar situation when the propeller would move in **corkscrew** motion (a solid mass); then the slip is zero and the propeller moves in relation to the cork with a velocity of $V = p \times n$.
However, as water is a liquid (moves backwards), the apparent propeller speed forward decreases with the slip and is equal to the ship's velocity V and the apparent slip can be expressed as: $S_A = p \times n - V$.

The apparent slip S_A, which is dimensionless, is defined as follows:

$$S_A = \frac{P \times n - V}{P \times n} = 1 - \frac{V}{P \times n}$$

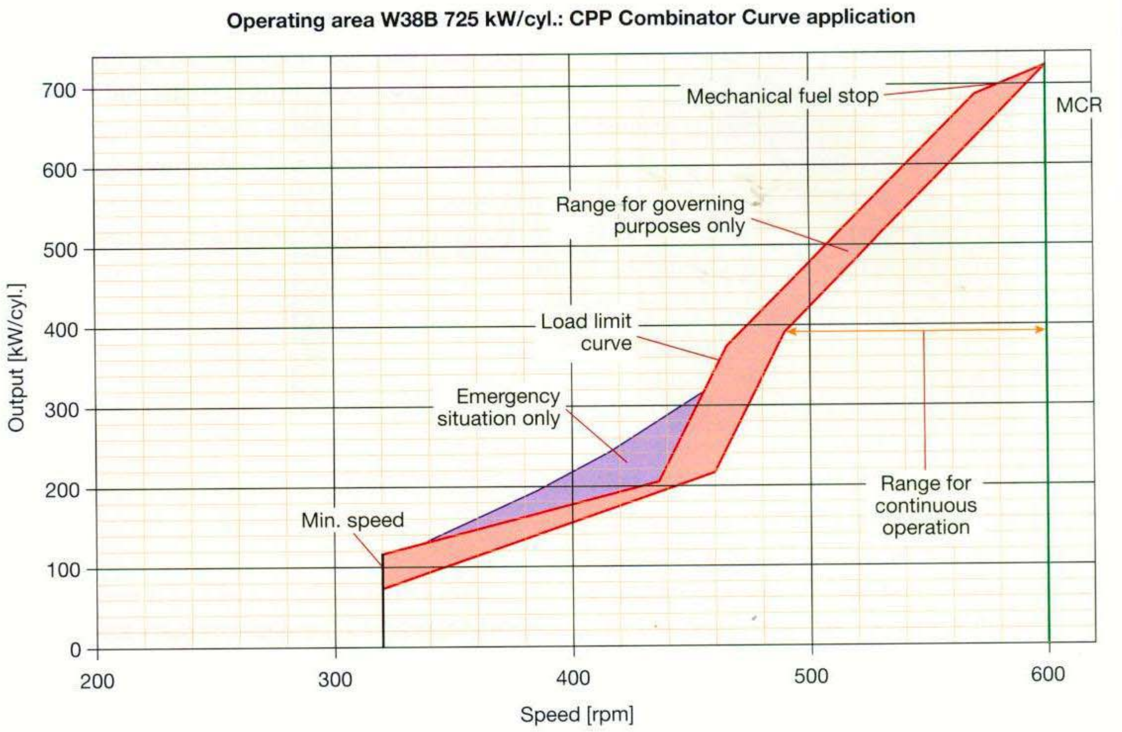


The apparent slip ratio : $S_A = \frac{p \times n - V}{p \times n} = 1 - \frac{V}{p \times n}$

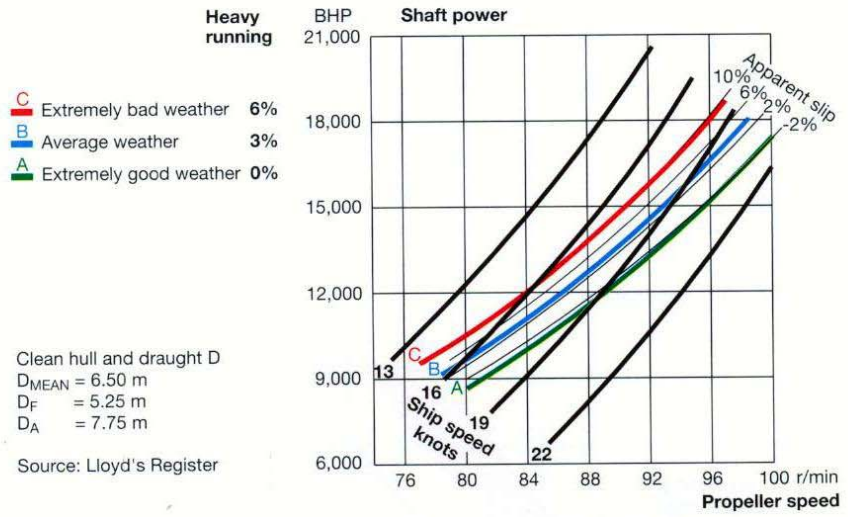
The real slip ratio : $S_R = \frac{p \times n - V_A}{p \times n} = 1 - \frac{V_A}{p \times n}$

▶

A load diagram for a four-stroke medium-speed Wärtsilä 38 B diesel engine operating on fuel H.F.O. Propeller type: controllable-pitch propeller.



19.11.6 Propeller performance in general at increased ship resistance



▲

Heavy seas and adverse wind and sea.

Resistance resulting from heavy seas can cause the propeller to operate 7 tot 8% heavier than in calm seas, or at the same power delivered to the propeller the rate of revolution of the propeller may be reduced by reduce 7 to 8%.

This chart applies to a medium-sized containership.

horizontal: the rate of revolution of the propeller per minute

vertical: the shaft power BHP (brake horse power)

Brake Horse Power: 736 Watt

draught mean: D_{mean} = 6.50 m

draught forward: D_F = 5.25 m

draught aft: D_A = 7.75 m

green line A = extremely good weather

blue line B = normal weather

red line C = extremely bad weather

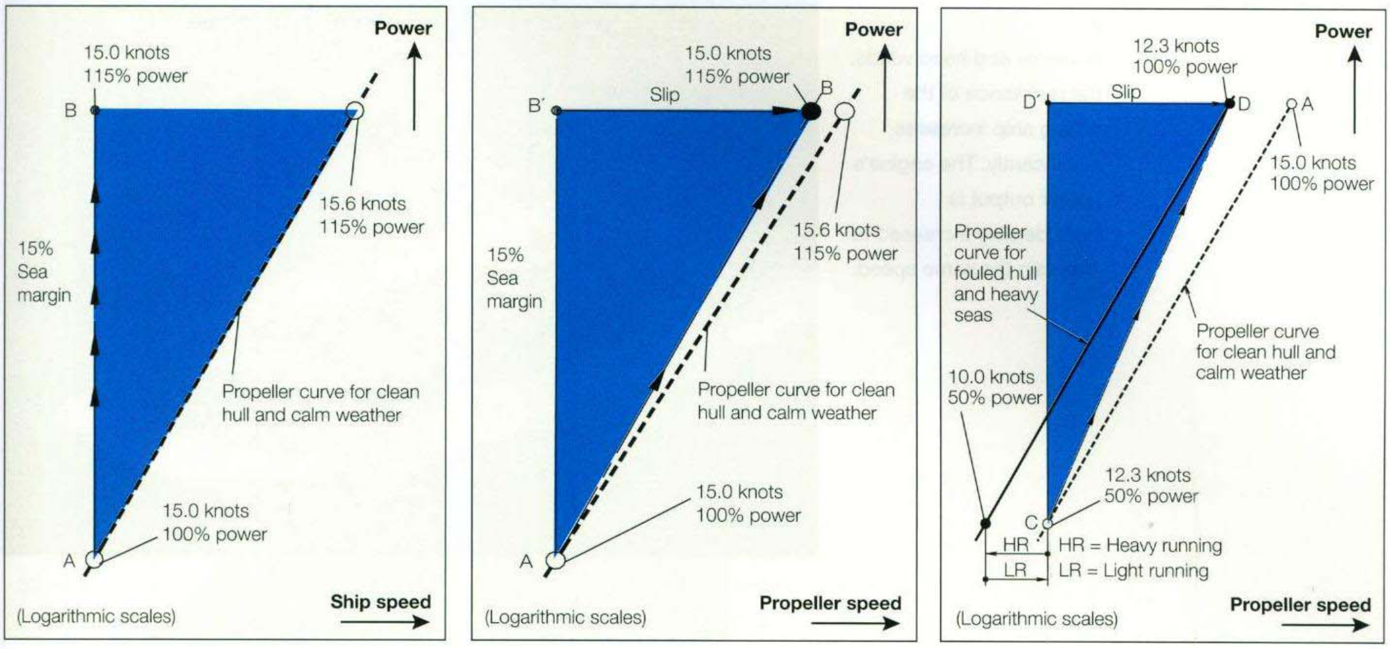
Bottom left of the figure the ship's velocity in knots (knots = 1 nautical mile per hour = 1.852 metres).

Top right of the figure the apparent slip in percentages.

The data are gathered over a period of a year and only influenced by the weather conditions!

The measurements are reduced to three average weather types and show an average 'heavy' running of 6%, in reality even slightly more.

In order to prevent heavy pitching of the ship and consequent damage to the stern, the ship's velocity is reduced to a more acceptable level.



▲

The difference between the curves for a lightly and heavily loaded propeller is clearly visible in the previous charts.

Here the ship's velocity constitutes 15 knots and the delivered engine power 100%. This applies to a clean hull just after dry docking and in calm water. With 15% extra engine power the velocity increases from 15.0 to 15.6 knots. As described earlier and corresponding to the calm weather conditions, an extra margin, the so-called 'sea margin', is established for poor weather conditions such as wind, waves and currents. This is the extra power that can be used in these bad weather conditions, if necessary.
So most propulsion marine engines usually operate at 85% of their power output. The 15% margin serves therefore to bridge the increased resistance.
In this example the ship's velocity at 100% of the engine power is 15.0 knots, at point A. At a speed of 12.3 knots with a clean hull and in calm weather, only 50% of the engine power is required, in point C at the bottom of the figure.

figure left: horizontal: the ship's velocity in knots; vertical: the power supplied to the propeller in percentages.
If the resistance increases to a level which requires an additional 15% power, the operating point shifts from A to B.
figure middle: horizontal: the propeller speed in knots; vertical: the power delivered to the propeller in percentages.

One would initially think that point A would move to B as with fixed-pitch propellers, the propeller moves through the water at the same speed. If the propeller were to rotate through a solid mass, this would be accurate. However, water is not solid and will accelerate aft and the propeller has a slip which increases at a higher thrust, produced by a higher resistance of the ship. This is why point A moves to point B which in fact is close to the propeller line of the left figure.
Operational point B is now positioned on the propeller line which runs heavier than the propeller line of the left figure in calm water.

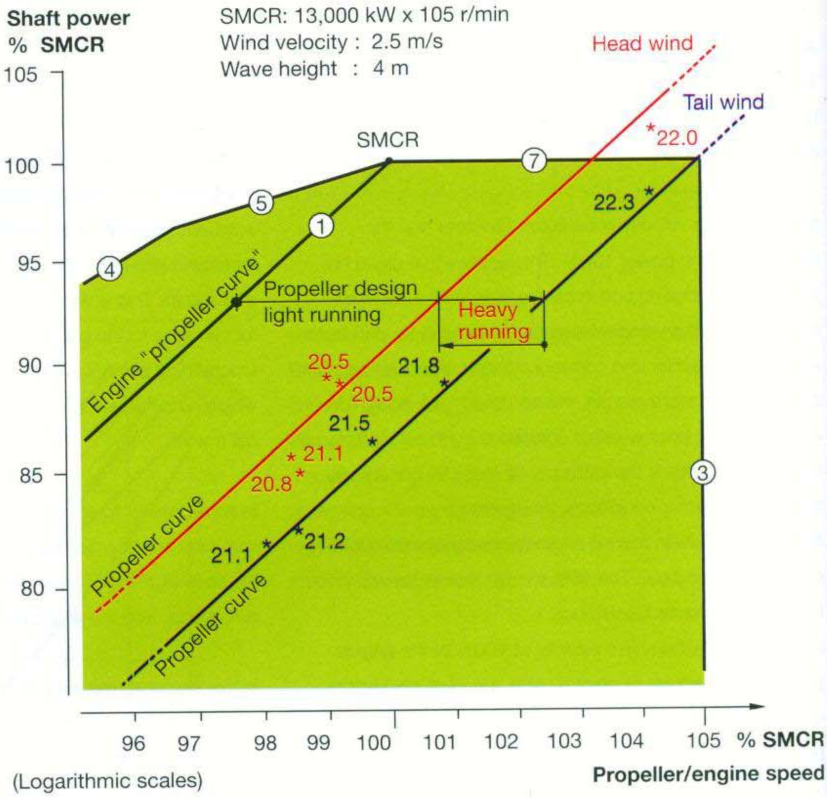
In some cases, for instance, when the hull is fouled and the ship sails in heavy seas and head winds the resistance can increase significantly and extra, up to 100% or more, power is required. See the example in the **right figure**.

In this example, the velocity of the ship at 100% engine power is 15.0 knots, at point A. At a velocity of 12.3 knots with a clean hull and in calm water, only 50% of the engine power is required at point C, bottom of the figure.
In heavy conditions similar to those mentioned previously, the ship can achieve a maximum speed of 12.3 knots at a 100% engine power output.
The operational point now shifts from point A to D.
Operational point D now is shifted to the left, relatively far from point A, which implies heavy running.
Situations such as these should be meticulously calculated when designing the propeller.
A propeller with bent blade tips is more suitable for these heavy loads than a standard propeller as the propeller is more capable to absorb the higher torque in heavy conditions.

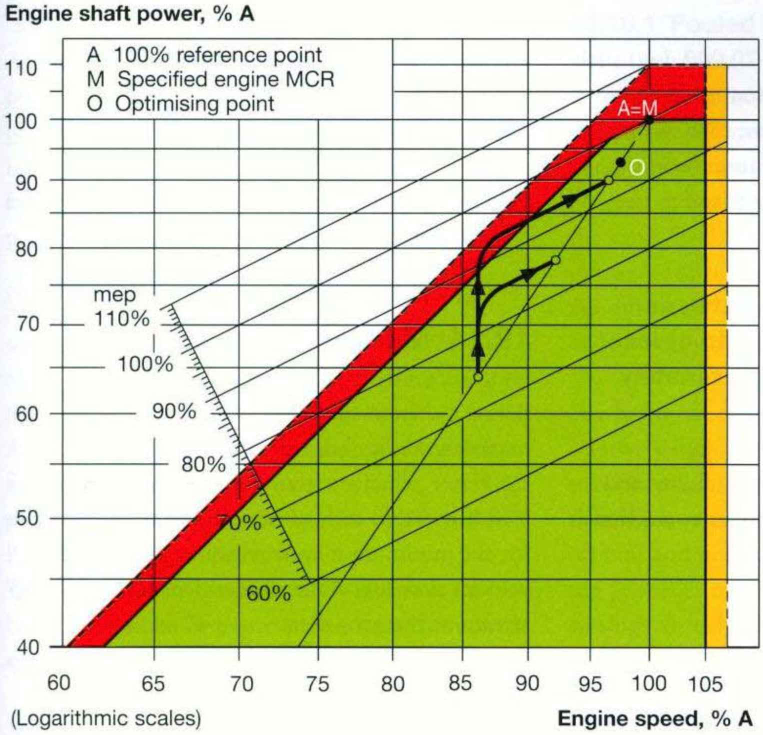
▶
In waves and head winds, the resistance of the sailing ship increases significantly. The engine's power output is considerably increased to maintain the same speed.

▶
Another example of a reefer during the trials.

horizontal: the engine/propeller RPM at a maximum shaft power of 13,000 kW and an RPM of 105.
vertical: the engine power in percentages.
weather conditions: wind velocity 2m/sec, wave height 4 metres.
Line 1 is the standard engine/propeller line for normal speed at a normal load of maximally 100% shaft power and an RPM of 100 (point SMCR).
The **red line** applies to 'heavy running', a slight head wind and waves. The speed lies between 20.5 and 21.1 knots. Here the required power is 85 to 90% and the RPM 98 to 99%.
The **black line** on the right applies for tail wind; the speed increases to 21.1 and 22.3 knots, depending on the power output and the RPM.
At 21.1 knots, the RPM is 98% and the power output 82%. Near the red line and at a speed of 21.1 knots this is approximately 98.5% and 86% and therefore more power (read fuel) will be required.



19.12 Increasing ship speeds



▲

Increasing the ship's speed.

vertical: shaft power in percentages
horizontal: engine RPM
green area: normal engine load
yellow area: maximum RPM for short periods
red area: engine over load
m.e.p. 110-60%: mean effective pressure in percentages of a maximum 100%
point A = M: 100% power, 100% engine speed, 100% mean effective pressure
A = 100% point of reference
M = specific engine power MCR (Maximum Continuous Rating)
O = optimising point

Curved lines – acceleration of the ship – the top line traverses the over load area temporarily.
When the ship accelerates, the propeller can even be loaded more heavily than during free sailing.
The engine power which at that time is higher than during free sailing is high and it may take some time for the propeller rate to achieve a new and higher revolution rate.

19.12.1 Shallow waters

Sailing in shallow waters may cause an increase in ship resistance, thus subjecting the propeller to a larger load than would be the case during free sailing in open water.

19.12.2 Influence of displacement

A ship in a loaded condition may carry more or less than under average loaded condition. This, of course, affects the ship's resistance and the required propeller power, but has only a minor effect on the propeller curve.
When the ship sails in a ballast, the displacement volume can be considerably less when compared to the loaded condition, and the propeller curve could, for instance, be 2% 'lighter'; at an identical engine power. The speed would then increase by 2%.

19.13 Parameters causing heavy-running conditions

The following examples provide an indication of the risks involved in sailing in poor weather conditions subjecting the propeller to heavy running.

19.13.1 Relatively small ships (less than 70,000 dwt)

Relatively small ships (less than 70,000 dwt) such as reefers and medium-sized container ships are more vulnerable than larger ships, such as large tankers and container ships, because the latter types are relatively large in comparison to the wave size.

19.13.2 Small ships (length between perpendiculars equal to or less than 135 m and approximately 20,000 dwt)

Small ships have a low directional stability and consequently require frequent rudder corrections, which increase the ship's resistance.

19.13.3 High-speed ships

High-speed ships are more sensitive to waves than low-speed ships, because waves have a higher reaction force to high-speed ships than to low-speed ships.

19.13.4 Ships with a flat stern

Ships with a flat (truncated) stern are more likely to be slowed down by waves than ships with a cruiser stern. Similarly, a sharp stem will cut the waves more efficiently and therefore reduce the heavy running of the propeller.

19.13.5 Fouling of the hull and the propeller

Fouling of the hull and the propeller, including corrosion of the hull, will also increase the resistance and therefore the propeller torque. The propeller and in particular the tips, as the speed and therefore the frictional force here are at their highest, should be polished each time the ship is dry docked (even when floating in a dry dock), since this has a positive effect. Using anti-fouling paints prevents the growth of living organisms such as seaweeds and barnacles.

19.13.6 Ship acceleration

Accelerating the ship's velocity will increase the propeller torque and therefore temporarily give a heavy-running propeller.

19.13.7 Sailing in shallow waters

Sailing in shallow waters increases hull resistance and reduces the directional stability of the ship.

19.13.8 Ships with a skewed propeller

Ships with a skewed propeller are able to absorb a higher torque under heavy running conditions.

19.14 Manoeuvring speed

Below a certain ship speed, the so-called manoeuvrability becomes increasing difficult, as the water velocity arriving at the rudder is too low. It is rather difficult to indicate the exact value for the minimum manoeuvrability as the water velocity arriving at the rudder depends on the slip stream of the propeller.
Often manoeuvring speeds of 3.5 to 4.5 knots are observed.
According to the propeller law, low engine power is required, but this, of course, is dependent on the weather conditions which can cause an increased resistance, so more propulsion power is required.

19.15 Direction of propeller rotation – lateral forces

The blades of a propeller of a ship afloat have better grip in their bottom position than their top position.
The resulting lateral force is larger the shallower the water, as, for instance, during port manoeuvring or in shallow waters.

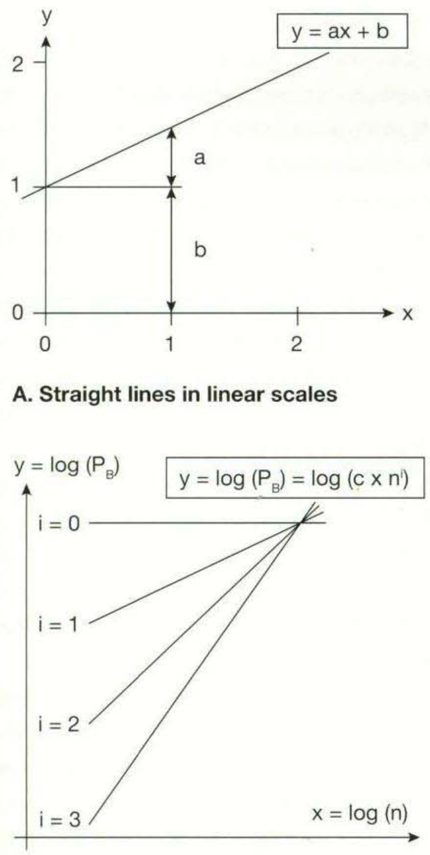
Therefore, a clockwise-turning propeller will push the ship in starboard direction and push the bow of the ship towards port side. This is counteracted by the rudder.

So, when reversing with a fixed-pitch propeller, the lateral force direction is reversed. The stern of the ship will move to port and the bow to starboard.

The actual reason for a clockwise-turning propeller to thrust the stern towards starboard is the fact that the upper part of the water flow around the propeller rotates and strikes the aft side of the ship's hull.

19.16 Engine layouts and load diagrams

As mentioned earlier, the effective brake power P_B is proportional to the mean effective pressure (m.e.p.) p_e and the engine speed (rate of revolution) n.
In the engine formula of a certain type of engine, p_e and n are the variables, whereas the rest $(\frac{\pi}{4} \times D^2 \times S \times \frac{1}{a} \times Z)$ are constant and referred to as C. $P_B = C \times p_e \times n$ and for a constant mean effective pressure $P_B = C \times n^1$ (for a constant m.e.p.).
As previously mentioned, the shaft power with a fixed-pitch propeller, may in accordance with the propeller law be expressed as:
$P_B = C \times n^3$ (the propeller law).
Therefore, for the examples shown above, the brake power can be expressed as a function of the speed n to the power output of $P_B = C \times n^i$.



A. Straight lines in linear scales

P_B = engine brake power
c = constant
n = engine speed

$$P_B = c \times n^i \Rightarrow \begin{cases} \log(P_B) = i \times \log(n) + \log(c) \\ y \quad = \quad ax \quad + \quad b \end{cases}$$

B. Power function curves in logarithmic scales

◄
The relation between the linear functions with linear scales and power functions with logarithmic scales. These are generally used by large engine manufacturers.

Propulsion and engine characteristics

19.16.1 Fouled hull

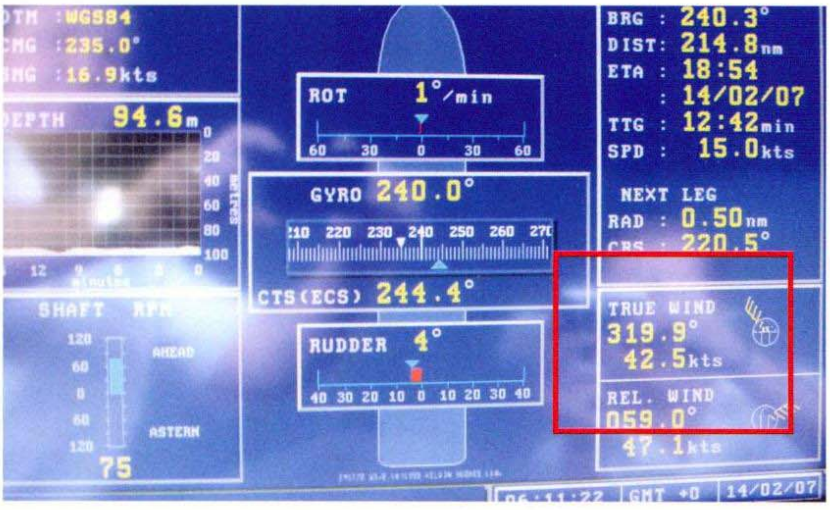
After a certain sailing time, the hull and propeller resistance will increase due to fouling and corrosion. Consequently, the ship's velocity will decrease when the propulsion power remains at the same level. If, however, the engine delivers more power to the propeller, the operating conditions of the propeller change to what is called 'heavy running'.

Furthermore, newer highly efficient ships have a relatively high velocity and very smooth hull- and propeller-surface areas (at sea trials). This means that the inevitable build-up of surface roughness on hull and propeller during regular service causes the propeller to rotate more heavily, in comparison to ships already constructed with a rougher hull surface.

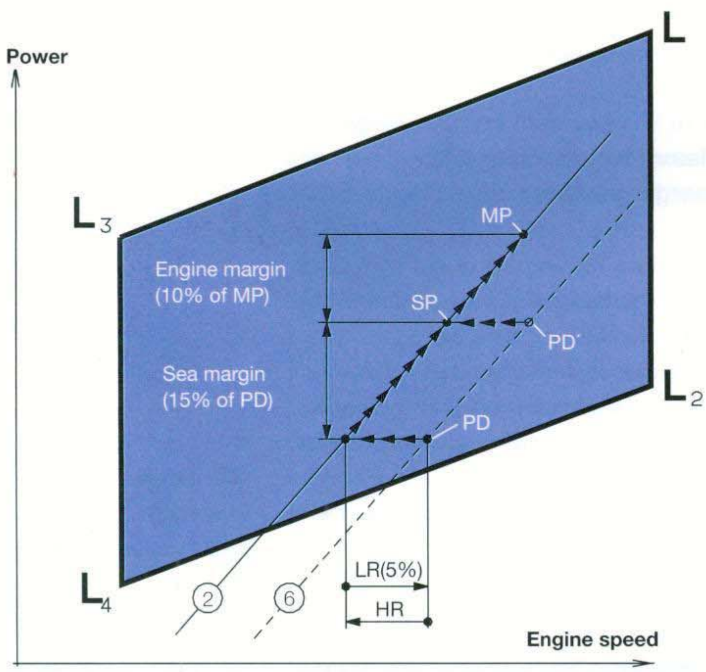
19.16.2 Heavy-weather conditions and sea margin used for engine layout

If, at the same time, the weather is bad with head winds, the ship's resistance increases significantly, which leads to even heavier running.
In determining the required engine power, it is common practice to add an extra power margin, the so-called sea margin, which is a standard 15% of the propeller design power P_D.
However, for large container ships margins of 10 to 30 % are sometimes used.

▼
With the anemometer 'True Wind', the wind speed can be judged accurately. 42.5 knots! 80 km per hour.



▸ A storm as seen from the bridge of a large container ship.



② Heavy propeller curve – fouled hull and heavy weather
⑥ Light propeller curve – clean hull and calm weather

MP: Specified propulsion point
SP: Service propulsion point
PD: Propeller design point
PD': Alternative propeller design point
LR: Light running factor
HR: Heavy running

◂ Heavy seas and 'sea margin' in the graph. The propeller design point PD.

Usually, the estimations of the required propeller power and the speed are based on theoretical calculations for a loaded ship and often (based on) towing tests executed in a hydraulics laboratory.
The combination of the speed and the required power output is referred to as the design point PD of the ship propeller. In the figure for a light running propeller, straight line no. 6.
Some shipyards and/ or propeller manufacturers use a design point PD that also partially or entirely incorporates the sea margin (power surplus).
When establishing the required engine speed of a propulsion engine, it is advisable, in comparison to line 6, clean hull, propeller and calm weather, to select the heavier propeller line 2.
Line 2 has 3 to 7% fewer revolutions.
On average a 5% difference is acceptable.
Note that the chosen 'sea margin' does not signify that a heaver propeller line was selected.

The SP point = the maximum service propulsion point for a sailing ship.

The combination of the speed and the engine power for a heavier running propeller and a sea margin is referred to as the maximum service propulsion point S.P. This applies to a fouled hull and heavy weather conditions.
The 'heavy' propeller line 2 will generally be applied as the basis for the engine characteristics during free sailing and the propeller line for a clean hull and calm weather, line 6, for a 'light' running propeller, LR.

19.17 Electronic governors with load limitation

To protect the diesel engine against thermal and mechanical overload, the accepted electronic governors include two limiter functions.

19.17.1 Torque limiter

The torque limiter ensures that the allowable limitation lines of the load diagram are adhered to at all times.

The torque limiter compares the calculated fuel pump index (fuel amount) and the actual measured engine speed with a standard reference B limiter curve, which gives the maximum allowable fuel-pump index at a given engine speed. If the calculated fuel-pump index is above this line, the actual fuel-pump index is reduced accordingly.

The standard reference limiter curve should be adjusted to correspond to the limitation lines of the load diagram.

19.17.2 Scavenging- air pressure limiter

The scavenging- air pressure limiter prevents the engine from being over-fuelled in relation to the amount of air supplied with charged-air groups, while accelerating. This is mainly the case during rapid load fluctuations such as in manoeuvring.

The scavenging-air pressure limiter algorithm compares the calculated fuel-pump index and the measured scavenging-air pressure with a standard reference limiter curve that gives the maximum fuel-pump index at a certain scavenging-air pressure.

If the calculated fuel-pump index is above this limiter line, the fuel-pump index is reduced accordingly.

The reference limiter curve should be adjusted so there is a sufficient air supply for a good combustion process.

19.17.3 Recommendations

Continuous operation within a certain time limit is only allowed in the area limited by the lines 4, 5, 7 and 3 of the load diagram.

For fixed-pitch propellers that operate in calm weather conditions with a loaded ship and a clean hull, the propeller/engine may run along or close to the propeller-design line 6.

After a certain running time, the ship's hull and the propeller grow increasingly rough by fouling and contamination, thus causing the propeller to run more heavily. The propeller line will shift to the left from line 6 to line 2. Therefore, more propulsion power will be required to maintain the ship's velocity.

If the propeller runs heavily in calm weather, this could indicate that the hull requires cleaning and possibly the propeller wants polishing.

The area between lines 4 and 1 is available for operation in shallow waters, for acceleration and if required, for non-steady operation without any time limit.

19.18 Use of charts

Examples

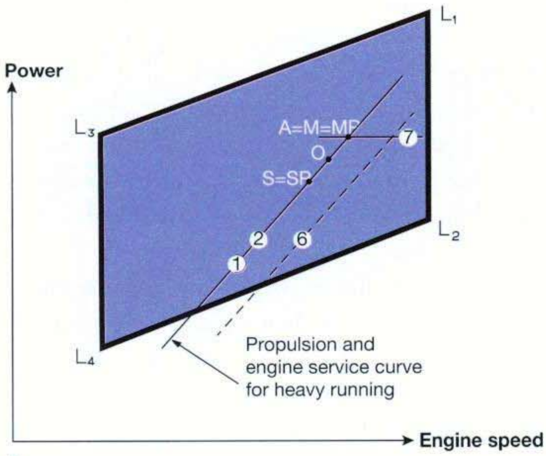
The following are four different examples of three fixed-pitch propellers and one controllable-pitch propeller. They illustrate the flexibility of these diagrams.

In this aspect, the choice of the optimum point O has an important influence.

19.18.1 Example 1 – Normal operating conditions without a shaft generator

M: Specified MCR of Engine
S: Continuous service rating of engine
O: Optimising point of engine
A: Reference point of load diagram

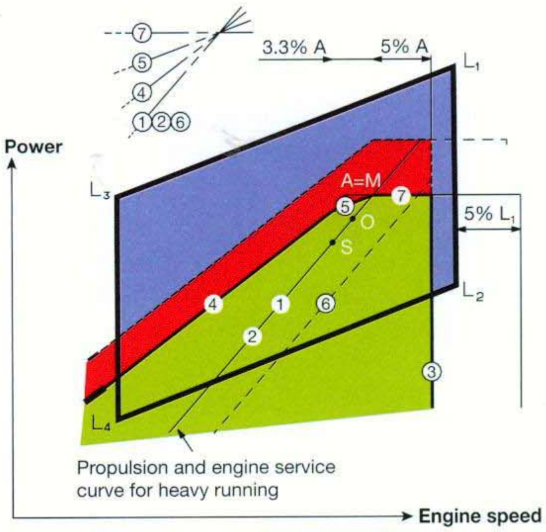
Point A of load diagram
Line 1: Propeller curve through optimising point (O)
Line 7: Constant power line through specified MCR (M)
Point A: Intersection between lines 1 and 7



▲

Example 1: normal operation without shaft generator.

Normally, the optimising point O and the engine line 1 are placed on line 2 (for heavy running).

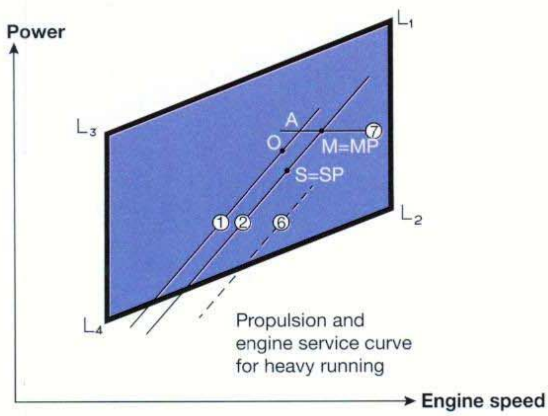


Point A can be found between propeller line 1 (2) and the line of the continuous rate line 7, through M. In this instance, point A is equal to point M.
When point A has been located, the layout diagram such as in the figure above can be drawn and the current load limitations of the diesel engine can be found.

19.18.2 Example 2 – Special load conditions without a shaft generator

M: Specified MCR of Engine
S: Continuous service rating of engine
O: Optimising point of engine
A: Reference point of load diagram

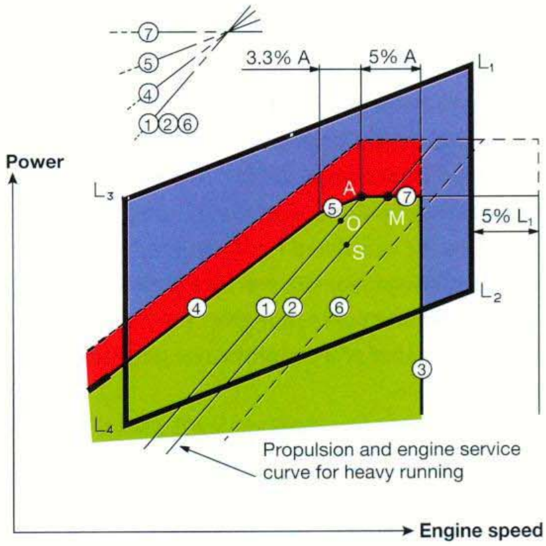
Point A of load diagram
Line 1: Propeller curve through optimising point (O)
Line 7: Constant power line through specified MCR (M)
Point A: Intersection between lines 1 and 7



▲

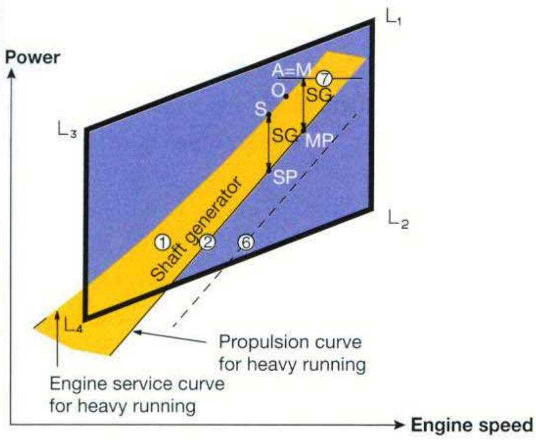
Example 2: Special load situations without shaft generator.

The propeller can be heavier loaded when the ship accelerates than during full load. The same applies when the ship is subjected to extra resistances, for instance when sailing in heavy winds with high wave resistances.
In both instances, the engine service point moves to the left of the normal service line as the propeller runs heavier.
To avoid exceeding limitation line 4 of the load diagram to the left, it is sometime necessary to reduce the speed and/or the power output.



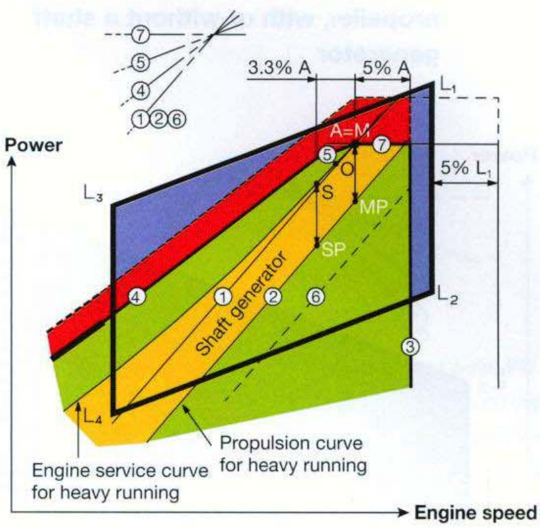
It is recommended to move the lay out diagram further right, when it is expected that the ship will sail on a route where poor weather conditions are a common occurrence.
The latter is performed by shifting the optimising point O and consequently propeller line 1 to the left.
However, this will result in a fuel cost increase in due to a slight increase in fuel consumption.
An example is shown in both graphs. The top figure shows compared to the normal situation in the first example, the limitation line 4 has been shifted to the left, thus creating more space between the lines 2 and 4; here a light load factor has been applied.

19.18.3 Example 3 – Normal operating conditions with a shaft generator



▲

Example 3: A standard case, here, however, with a shaft generator. In the figure, the engine line has been drawn for heavy running, including the extra power required for the shaft generator.



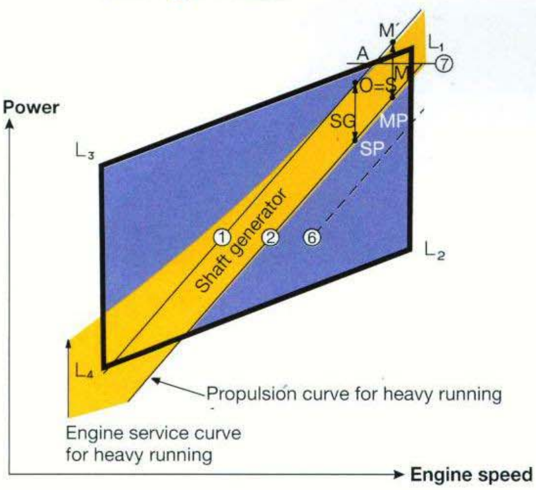
In this instance, a shaft generator has been installed and therefore the power output delivered to the propeller is reduced for the power supplied to the shaft generator.

The optimising point O and the engine line1 are then found in a similar fashion as in example 1.

M: Specified MCR of Engine
S: Continuous service rating of engine
O: Optimising point of engine
A: Reference point of load diagram

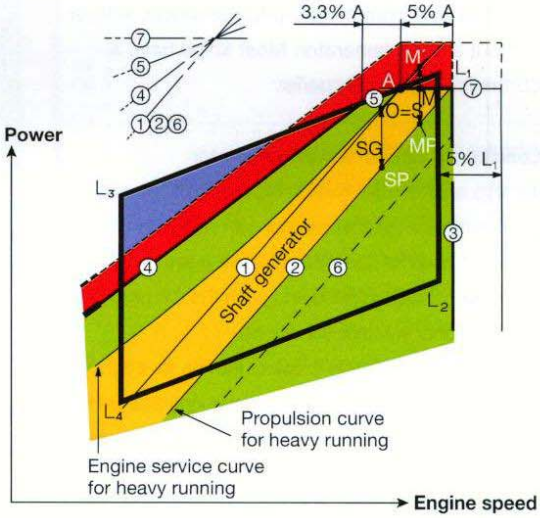
Point A of load diagram
Line 1: Propeller curve through optimising point (O)
Line 7: Constant power line through specified MCR (M)
Point A: Intersection between lines 1 and 7

19.18.4 Example 4 – A special case, with a shaft generator



▲

Example 4: A special case with a shaft generator.



M: Specified MCR of Engine
S: Continuous service rating of engine
O: Optimising point of engine
A: Reference point of load diagram

Point A and M of load diagram
Line 1: Propeller curve through optimising point (O)
Point A: Intersection between lines 1 and line L1 – L3
Point M: Located on constant power line 7 through point A and at MP's speed

Also in this special case a shaft generator is installed, but unlike example 3, the MCR.-point MP is placed at the top of the diagram

This means that the MCR of the engine, point M will be placed outside the top of the diagram.

This problem could be resolved by choosing a diesel engine with an extra cylinder (for instance a nine-cylinder- instead of an eight-cylinder in-line engine). Another and cheaper solution is to reduce the electrical power to the shaft generator when the ship is sailing at full power output. This achieved by using a diesel genset.

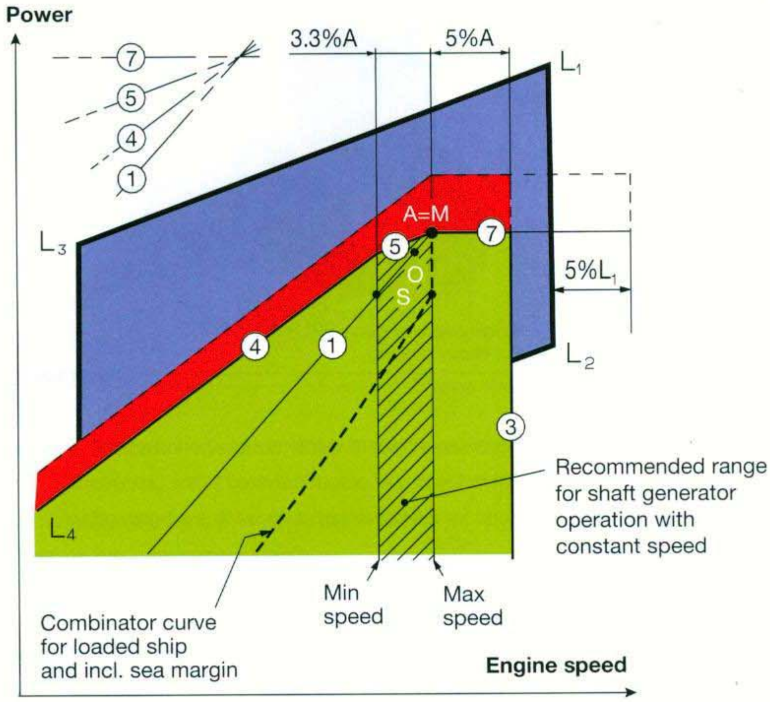
When the latter solution is choosing, the required MCR power of the engine can be reduced from point M' to M as is shown in the figure.

Situations such as these seldom occur as ships rarely have a normal operating speed with high engine capacities such as these.

In the example, optimising point O is chosen equal to point S, and so line 1 is found.

Point A, having the highest possible power can be found at the intersection of line L1-L3 and the complete load diagram has been drawn in the right figure. Point M is found on line 7 at MP's speed.

19.18.5 Example 5 – A controllable-pitch propeller, with or without a shaft generator



▲

Example 5: A controllable-pitch propeller, with or without a shaft generator. Most ships have a controllable-pitch propeller.

Load chart, without a shaft generator

If a ship is fitted with controllable-pitch propellers (CPP) the combinator line (of the propeller with optimum efficiency) is usually chosen for a loaded ship with a sea margin.
At a given propeller speed and pitch the CP propeller runs heavily in poor weather conditions.
Therefore, it is advisable to select the line for a lightly loaded propeller, such as the dotted line, ensuring a larger margin between the normal line and the lines for heavy-weather conditions such as 4 and 5.

Load chart, with a shaft generator

The hatched area indicates the recommended speed range between 100% and 96.7% of the maximum continuous rating (MCR) for an engine with a shaft generator operating at a constant speed.
The service point S can be situated at any point in the hatched area.
The combinator line of a loaded ship including a sea margin still has an ample margin indicated by lines 4 and 5.

19.19 Summarising the effects of the various types of resistance on engine operation

A brief summary of the effects of the various types of ship's resistance on a fixed-pitch propeller and the propulsion engine is shown by the chart below.

PD: Propeller design point, clean hull and calm weather

Continuous service rating for propulsion with a power equal to 90% specified MCR, based on:

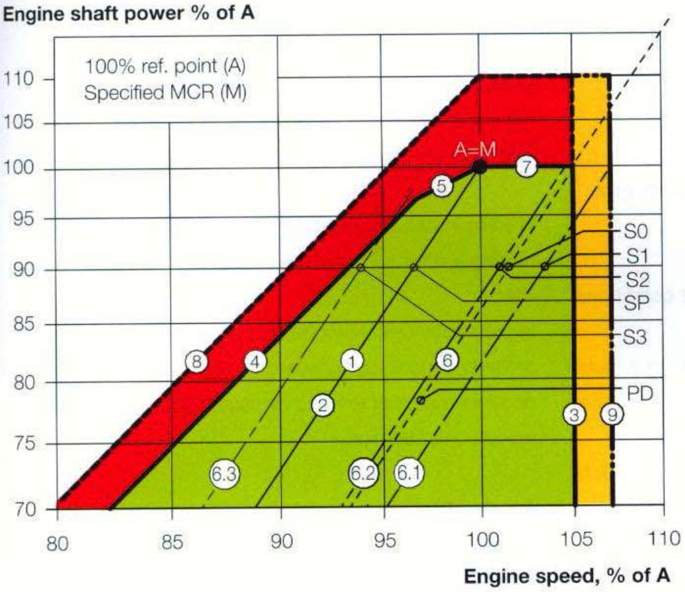
SO: Clean hull and calm weather, loaded ship

S1: Clean hull and calm weather, ballast (trial)

S2: Clean hull and 15% sea margin, loaded ship

SP: Fouled hull and heavy weather, loaded ship

S3: Very heavy sea and wave resistance



Line 1: Propeller curve through point A=M, layout curve for engine

Line 2: Heavy propeller curve, fouled hull and heavy weather, loaded ship

Line 6: Light propeller curve, clean hull and calm weather, loaded ship, layout curve for propeller

Line 6.1: Propeller curve, clean hull and calm weather, ballast (trial)

Line 6.2: Propeller curve, clean hull and 15% sea margin, loaded ship

Line 6.3: Propeller curve, very heavy sea and wave resistance

▲

The influence of the various resistances the ship is subject to, is illustrated by means of corresponding service points for propulsion with an identical propulsion power, based on the propeller design point PD, plus 15% additional power.

Propeller design point PD

As previously mentioned the propeller is generally designed for a certain ship velocity V valid for a loaded ship with a clean, smooth hull in calm weather.

The corresponding engine speed and power are indicated by point PD on the propeller line 6 of the load diagram.

Increased ship speed, point SO

If the engine power increases with, for instance, 15% and the ship has a smooth hull and operates in calm weather, point SO, the ship velocity V and the engine speed n will increase in accordance with the propeller law (more or less valid for normal speeds).

Point SO is placed on the same propeller line as point PD.

$$V_{SO} = V \times \sqrt[3.5]{1.15} = 1.041 \times V$$

$$n_{SO} = n \times \sqrt[3.0]{1.15} = 1.048 \times n$$

Normal speed ranges with clean hulls and 15% sea margin, point S_2

Conversely, still operating with a loaded ship and clean hull, but now with additional resistance due to heavy seas, extra power of, for instance, 15% is required to maintain the ship velocity V (15% sea margin).

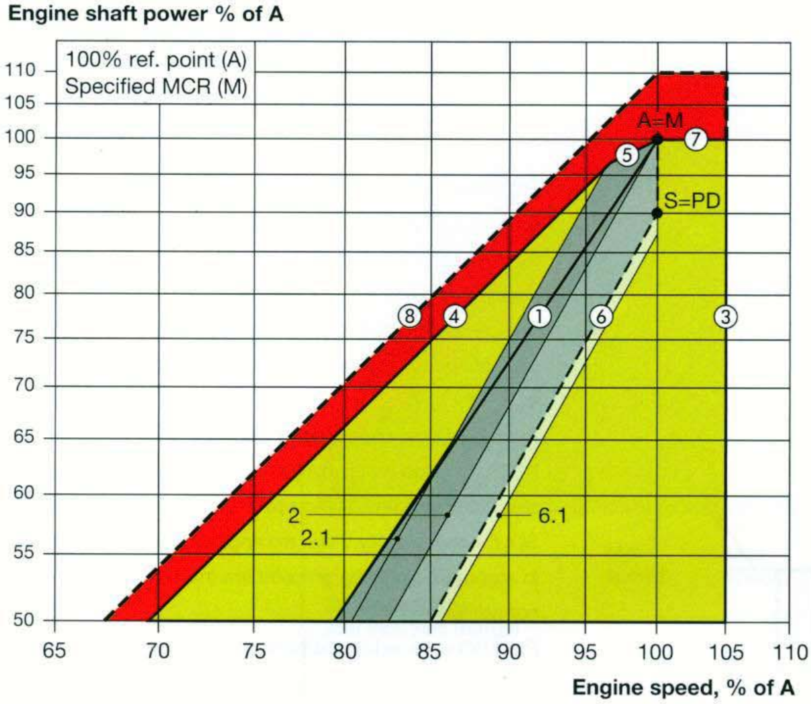
If the velocity of a ship V_{S2} = V and the propeller has no slip, the engine speed will remain constant.

However, as the water abaft the propeller gives way because the propeller does slip, the engine speed will increase and the service point S2 will be placed on the propeller curve 6,2 close to SO on the propeller curve 6.

The propeller curve 6,2 could possibly indicate a slightly heavier loaded propeller than curve 6 (approximately 0.5%). Depending on the type of ship and its dimensions, the heavy running of the propeller and therefore the factor, will be 0.5% higher or lower.

For resistances that correspond to about 30% power increase (30% sea margin) the corresponding relative heavy operating factor will be in the region of 1%.

S=PD : Propeller design point incl. sea margins, and continuous service rating of engine
Line 1 : Propeller curve for layout of engine
Line 1 . Combinator curve for propeller design, clean hull and 15% sea margin, loaded ship
Line 6.1 : Light combinator curve, fouled hull and calm weather, loaded ship
Line 2 : Heavy combinator curve, fouled hull and heavy weather, loaded ship
Line 2.1 : Very heavy combinator curve, very heavy sea and wave resistance



Sailing with a fouled hull and in poor weather conditions, point SP

When, after a certain operational time, the ship's hull is fouled and its surface consequently rougher, the wake field will alter in relation to that of a smooth ship (clean hull).

A ship with a fouled hull experiences more resistance, which results in a modified wake field and therefore a heavier rotating propeller than would be expected by deteriorating weather conditions alone.

The propeller curve will shift to the left, see propeller curve 2 in the load diagram. This propeller curve indicates that the hull is fouled and the ship operates in poor weather conditions. The propeller / engine load has increased approximately 5% in relation to propeller curve 6.

In order to safeguard the air supply for the diesel engine's combustion process, which determines the limit of the torque and the engine speed, it is common practice to ensure that the engine and the turbo-blower match each other according to the load on propeller curve 1 of the load diagram, equal to the 'heavier' propeller curve 2.

As opposed to point S2, point SP is normally used to determine the engine layout by referring this rating to 'average operating'- power at 90% of the specific MCR-point, which matches a 10% margin with the maximum engine power.

In other words, in the example the propeller design curve is approximately 5% 'lighter' than the propeller curve that is applied to the layout of the main engine.

Operating in heavy seas with extremely high waves, point S3

When operating in these conditions the propeller can run between 7 and 8% (or more) 'heavier' than in calm conditions. Consequently, at the same engine / propeller power, the rate of revolution is decreased by 7 to 8%.

For a propeller power equal to 90% of MCR, point S3 in the load diagram shows an example of this operating condition.

In some instances operating in heavy headwinds the heavy running is often such that the service point is placed left of line 4.

In such instances, when the ship begins to pitch and the propeller periodically surfaces, it is recommended to decelerate in order to avoid damage to the ship and the propulsion installation.

Accelerating and operating in shallow waters

When the ship accelerates and the propeller is works heavier than during normal operating speed, the effect on the propeller is at times equal to the operation conditions at point S3.

The accelerating effects are sometimes even greater.

The same applies to operating in shallow waters.

Trials at open sea point S1

Normally, the curve 6 for a clean hull will be equal to the trial trip propeller curve. However, as the ship is seldom in loaded condition during the trials and usually operates in ballast, the propeller curve at 6.1 has a somewhat lighter load than curve 6.

Point S1 in the diagram provides an example of the power supplied to the propeller at 90% of the MCR.

It may be necessary to exceed the maximum engine speed of the propeller to 107%, so curve 9 in the diagram, during the trials to evaluate how the ship operates at 100% power.

19.20 Some comments

In practice, ship resistance is compared with the outcome of the towing tests in a laboratory, for instance Marin, Wageningen, The Netherlands. The results from the towing tests are also used for optimising the propeller and hull design.
When the required engine power, including the margins and the propeller speed is known, the correct engine can be selected with an engine selection programme developed by the engine manufacturer.
It is even possible to state the requirements via the internet, so the manufacturer can assess the client's demands in advance.

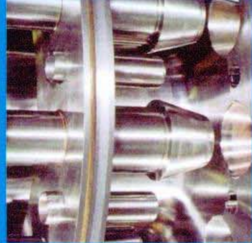
In view of this, the relation between the ship and the propulsion engine is extremely important, as well as the position of the load diagram, which is to say the choice of the engine lay-out diagram in relation to the operational curve in order to achieve an optimum propulsion engine.
In order to avoid overloading of the main engine during excessive operating conditions it is advisable to use an electronic governor with load control.

For a main engine, that also drives a shaft generator for generating electricity for the ship, the interaction between the ship and the propulsion engine is complex. Nevertheless, in practice problems very seldom occur.

CH 20

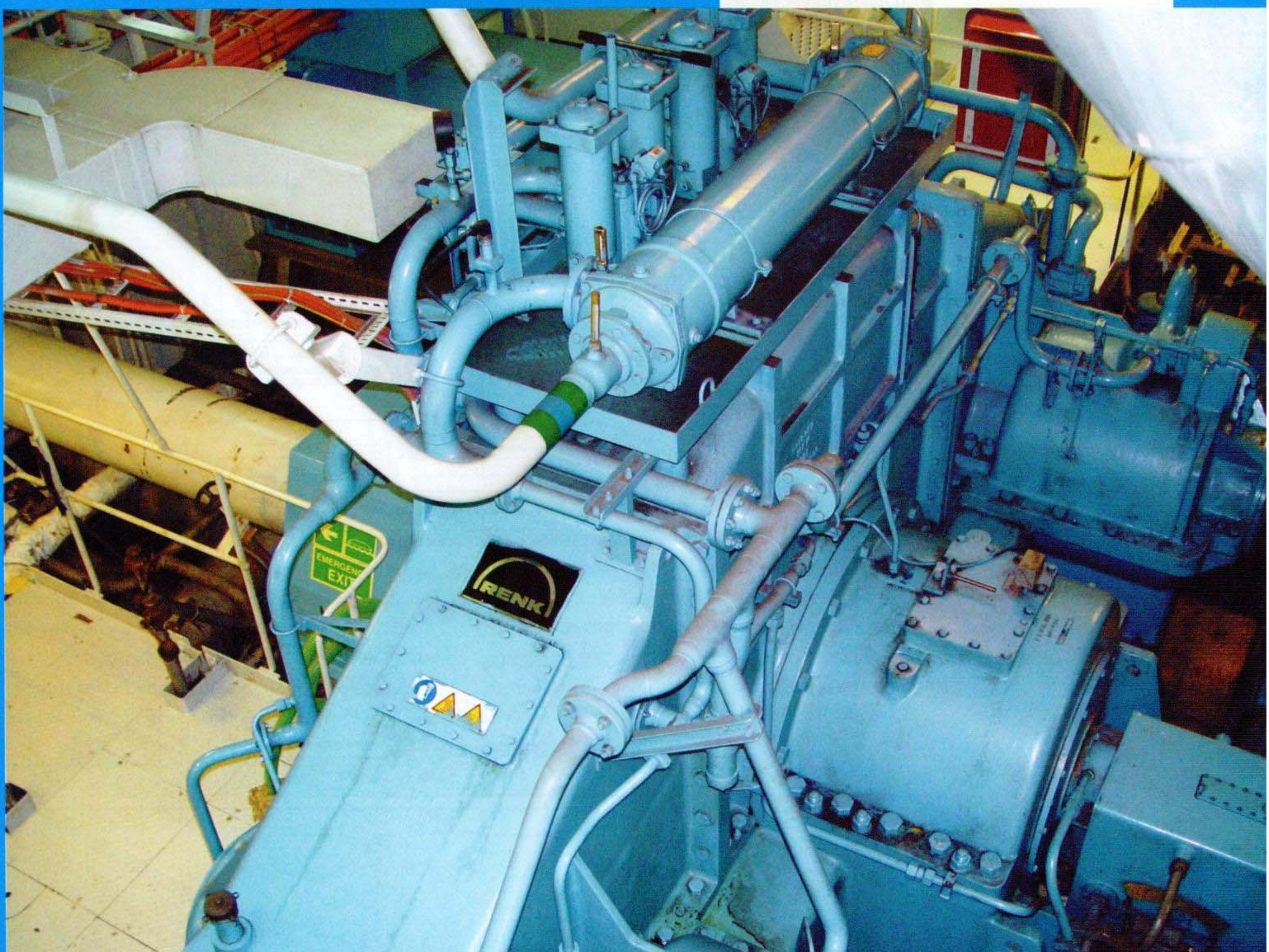
Transmission gears, flexible couplings, vibration dampers, shafting and shaft generator drives






Both Wärtsilä 46 C engines.

Propulsion of the 'Oranjeborg'
Two six-cylinder Wärtsilä 6-L 46 C medium-speed engines drive one adjustable-pitch propeller. Renk reduction gearing controls the correct propeller speed and shaft generator drive, top, at the back of the photo.



20.1 Introduction

Transmitting the engine's power output to the drives, for pumps, compressors or electric gensets requires the use of a number of provisions, such as couplings and gear transmissions.
Often, the driven items have different speeds, such as those of shaft generators and propellers. As opposed to the aforementioned drives, electric gensets in a diesel-power plant or on board ship are equipped with one fixed, specific speed, which is determined by the generator type.

20.2 Diesel-engine arrangements

The following arrangements frequently occur:

A **Ship propulsion** with a low-speed two-stroke crosshead engine.

B **Ship propulsion** with one or multiple medium-speed four-stroke trunk-piston engines.

C **Ship propulsion** with one or multiple high-speed four-stroke diesel engines.

D **Diesel-electric propulsion** with medium-speed or high-speed four-stroke diesel engines.

E **Ship propulsion with water jets** with high-speed four-stroke diesel engines.

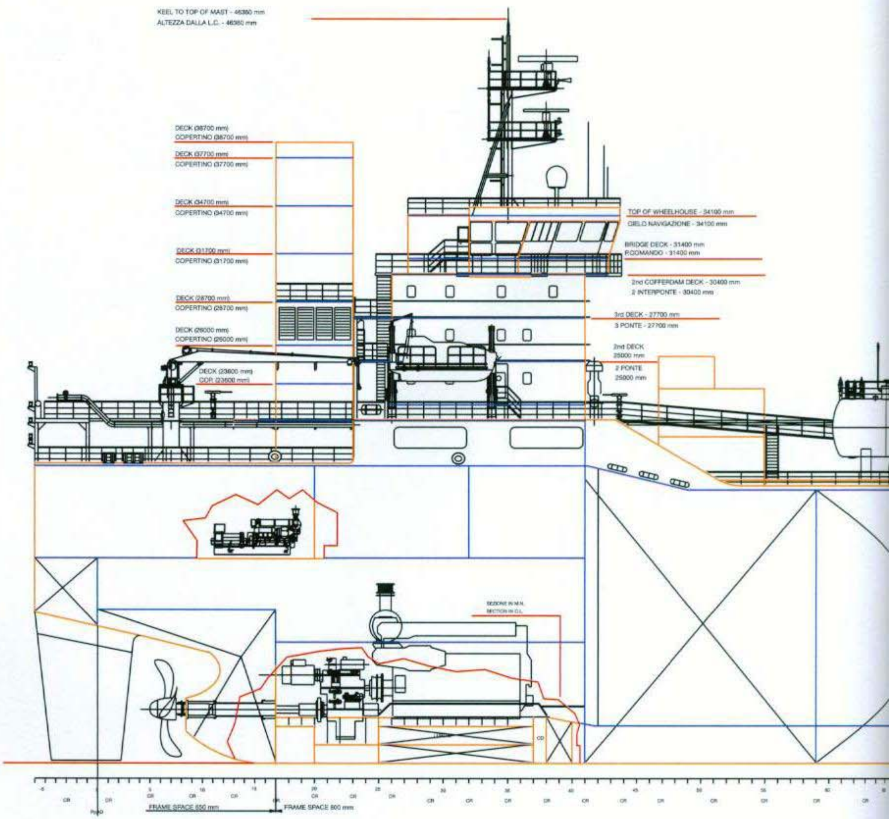
F **Diesel-power plants** with both high-speed and medium-speed four-stroke diesel engines and low-speed two-stroke crosshead engines.

G **Special drives** such as those for pumps in the dredging industry and compressors and pumps for use in the gas and oil industry with high-speed or medium-speed four-stroke diesel engines.

20.2.1 A Ship propulsion with low-speed two-stroke crosshead engines

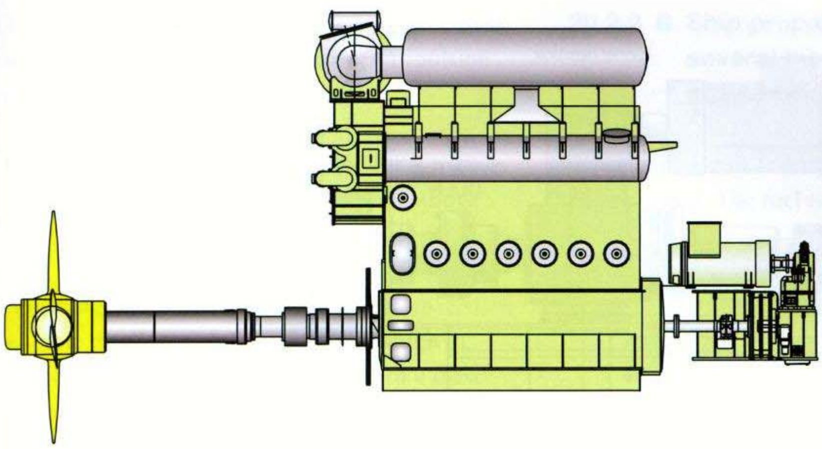
The engine directly drives a fixed-pitch propeller, is directly reversible and runs on heavy-fuel oil (H.F.O.).

This type of propulsion is generally used in large container ships, oil tankers and bulk carriers.
RPM: The low engine speeds of these category IV engines with cylinder bores from approximately 500 mm are very suitable to directly drive a fixed-pitch propeller. The efficiency of propellers is high at low engine speeds of between ± 60 to ± 120 revolutions per minute.
Engines with smaller cylinder diameters and therefore a higher speed of up to approximately 200 revolutions per minute are not often directly reversible and are equipped with a transmission gear positioned halfway between the crankshaft and the adjustable-pitch propeller.



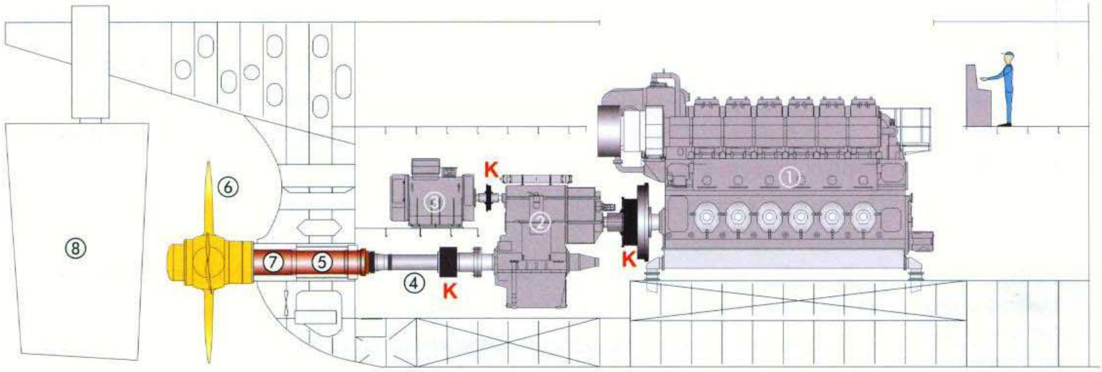
▶
Ship propulsion with a large medium-speed four-stroke engine.

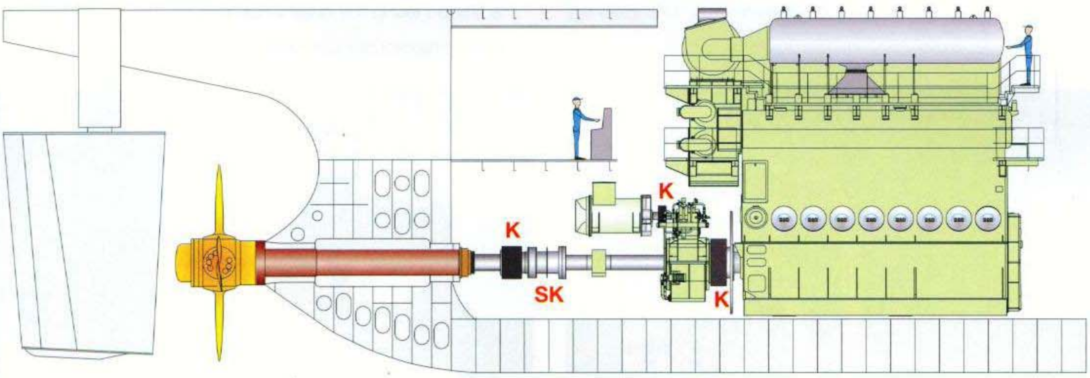
The diesel engine is connected to the propeller shaft with reduction gearing.
The pitch of the propeller is adjustable, so the engine can rotate in one direction and at a fixed speed.
A shaft generator is coupled to the gearbox. This enables electricity to be generated using heavy fuel oil, so the auxiliary engines, one deck up, do not have to operate on diesel oil, which is twice as expensive.



A six-cylinder two-stroke crosshead engine driving a controllable-pitch propeller.

At the other side of the propulsion engine a shaft generator is coupled to the engine.





Top picture:
ship propulsion with a medium-speed four-stroke diesel engine running on H.F.O.

The engine has one rotational direction and drives the controllable-pitch propeller and shaft generator.

1 *four-stroke diesel engine*
2 *gear box*
3 *shaft generator*
4 *intermediate shaft*
5 *propeller shaft*
6 *controllable-pitch propeller*
7 *stern tube*
8 *rudder*

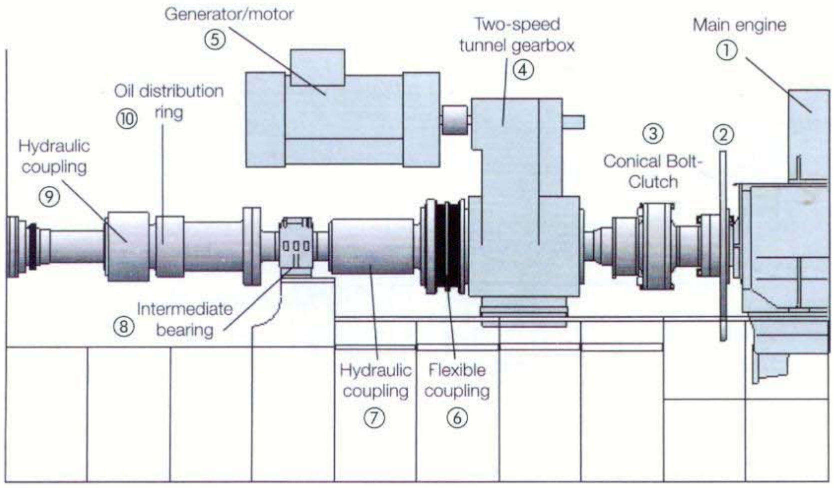
K *flexible couplings*
SK *clutch*

Bottom picture:
Ship propulsion with a low-speed two-stroke crosshead engine running on H.F.O.

Shaft generators

A number of propulsion units are provided with a shaft generator. When at sea, the shaft generator generates sufficient electric power for the entire operation of the electric systems on the ship, so the auxiliary engines can be turned off. In this way, the number of running hours of the auxiliary engines is limited as are the corresponding maintenance hours and expenditure.

The total efficiency for power generation is also slightly increased as the propulsion engine has a higher efficiency than the smaller auxiliary engines. The shaft generator is installed between the engine and the propeller shaft, and in smaller crosshead engines with a transmission gear to the propeller, on the propeller drive. This connection is often referred to as Power Take Off (P.T.O.).



▲

Two-stroke crosshead engine drive for the propeller shaft and shaft generator.

Viewed from the engine

1 *main engine with fixed rpm*
2 *flywheel/turning wheel*
3 *conical bolt clutch*
4 *two-speed gearing*
5 *shaft generator*
6 *flexible coupling*
7 *hydraulic clutch*
8 *intermediate bearing*
9 *hydraulic clutch*
10 *oil-distribution ring for controllable-pitch propeller*

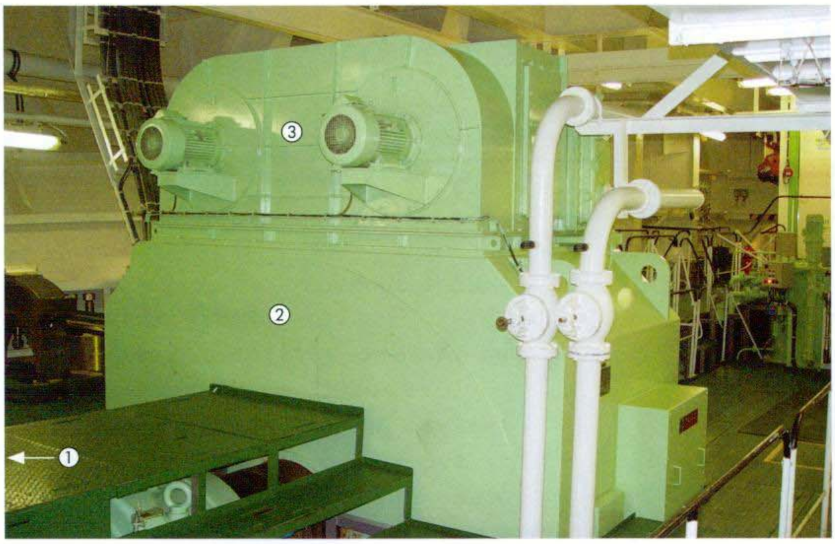
Options

I The main engine is running free of the main shaft. Shaft generator and propeller are stopped, for instance, during trials, after repairs or prior to departure.

II The main engine drives the shaft generator and the propeller is stopped.

III The main engine drives the shaft generator as well as the propeller.

VI The shaft generator is switched to the electromotor configuration and drives the propeller with power generated by the auxiliary gensets. This is an emergency operation mode during engine malfunction, the so-called 'getting home' device. This is also used when the vessel is shifted along the quay when simultaneously main-engine repairs are executed.



▲

The shaft generator on a very large container ship of 6,800 TEU.

The rotor of this 3.5 MW generator is an integral part of the shafting between the large two-stroke 68,000 kW crosshead engine and the propeller shaft. The cooling-air ventilators are positioned on top of the shaft generator.

1 *flywheel/turning wheel main engine*
2 *shaft generator*
3 *cooling-air ventilator*

▲

The shaft generator (1) above the propeller shaft on a feeder-container ship.

Space at the back of the engine room near the propeller is rather cramped.

The largest container ships being commissioned in 2008 are still equipped with one propulsion engine and one fixed-pitch propeller.

Drives with a fixed-pitch propeller

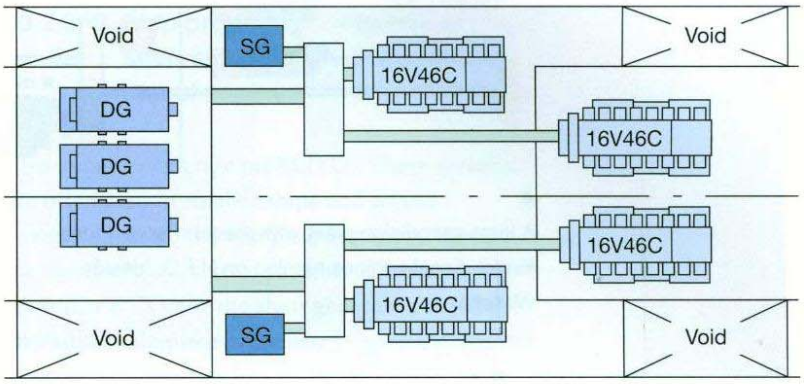
Most large ships are provided with the above mentioned propulsion system.

Advantages

- **The total propulsion efficiency is high in comparison to other systems.** With a diesel-engine efficiency of 50% and a propeller efficiency of 75%, the total propulsion efficiency is therefore:
  $0.5 \times 0.75 = 0.375$ or 37.5%. All other systems have a lower efficiency.
- **A simple installation.** A large crosshead engine has relatively few moving parts. When using the two-stroke principle, the power output generated is 1.6 times higher than that of four-stroke engines with an identical cylinder bore and engine speed.
- **The lower speed of larger two-stroke crosshead engines is particularly suitable for the direct drives for the large fixed-pitch propellers.**

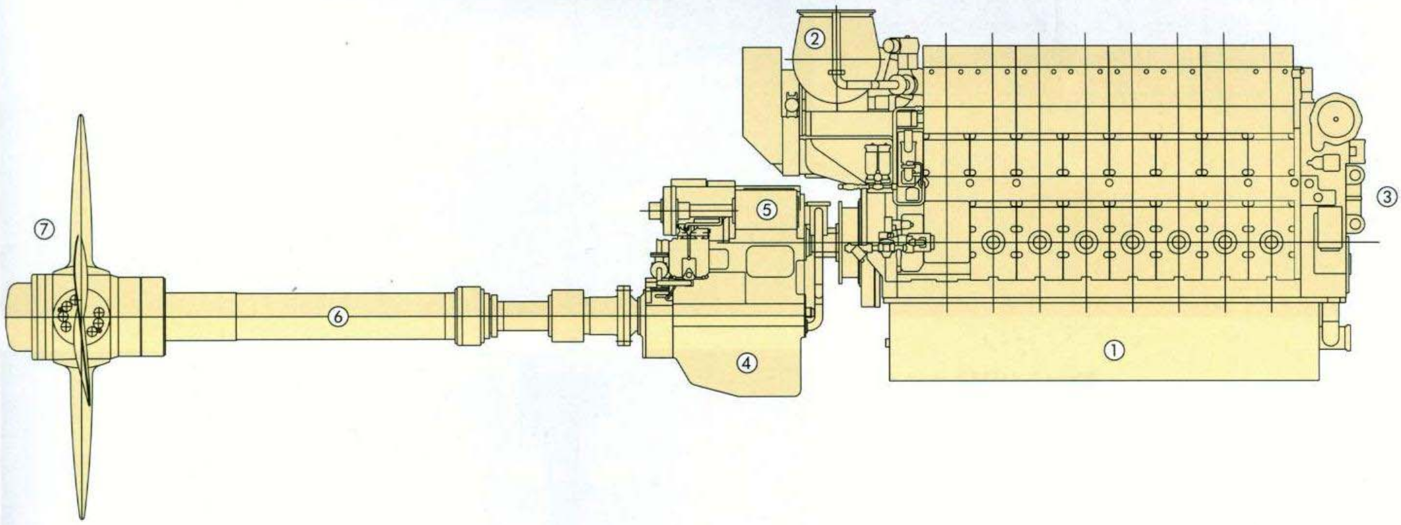
20.2.2 B Ship propulsion with one or several medium-speed four-stroke trunk-piston engines

The engine is not reversible and often has a fixed engine speed. The fuel is either H.F.O. or M.D.O..



A top view of a propulsion plant for medium-speed four-stroke engines driving twin propellers.

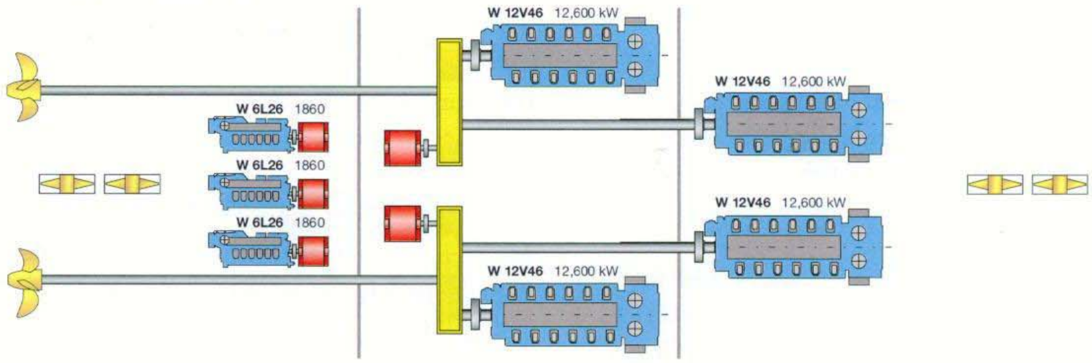
The four sixteen-cylinder Wärtsilä 46 V-engines drive the two adjustable pitch propellers via reduction gearing. Each reduction gearing has a Power Take Off-shaft with the shaft generator, SG, mounted on it.
In a separate auxiliary engine room another three diesel gensets, DG, are installed.



A standard arrangement of a four-stroke medium-speed propulsion engine.

1 *diesel engine*
2 *supercharger with intercooler*
3 *engine-driven cooling-water pumps*
4 *reduction gearing*
5 *P.T.O.-section*
6 *propeller shaft*
7 *controllable (adjustable, variable) - pitch propeller*

Diesel-mechanical (HFO)
Installed power: 55,980 kW



▲
A large propulsion plant with medium-speed four-stroke diesel engines running on H.F.O., design Wärtsilä.

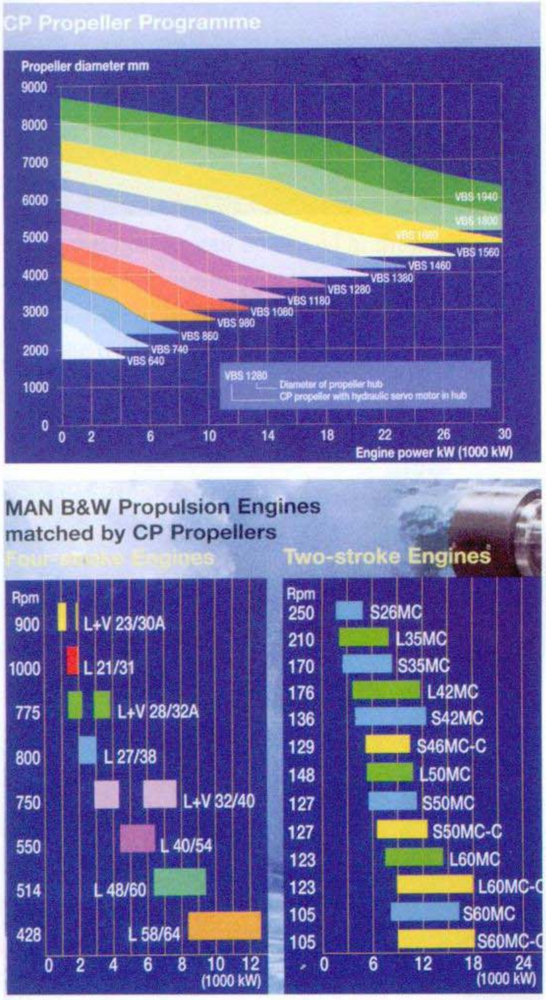
Four twelve-cylinder Wärtsilä 46 diesel engines in V-shape with a total shaft power of 50,400 KW. The three Wärtsilä 26 six-cylinder in-line engines provide a generator power of 5580 kW. The ship is equipped with two electric bow and stern thrusters.

►
Installation of one of the four large MAN–B&W four-stroke medium-speed diesel engines in the engine-room sections of a large cruise ship. This is in the hall of the Jos Meijer shipyard in Papenburg, Germany.

In view of the limited space on these ships, low four-stroke diesel engines have been chosen as opposed to the high two-stroke crosshead engines. Approximately 80% of the volume of the ship is required for the 'hotel' and the remainder is for all the technical installations.

►
The 'Norwegian Dawn' under construction in Papenburg, Germany. The cruise ship in dry-dock ready to be 'set afloat'.

This type of propulsion is often seen in smaller ships, such as container feeders, smaller oil, gas or chemical tankers, passenger ships and ferries. The shaft generator is often driven from the gear transmission. The height built for this type of propulsion is considerably lower when compared to the crosshead engine, and is therefore suitable for the low overhead space found in engine rooms of large ferries and passenger ships.



▲

The MAN–B&W program for propulsion engines with controllable-pitch propellers.

In the four-stroke medium-speed diesel engines, the RPM varies between 900 and 428 rpm, the cylinder diameter from 230 to 580 mm and the shaft power from 1000 to 12,000 kW.

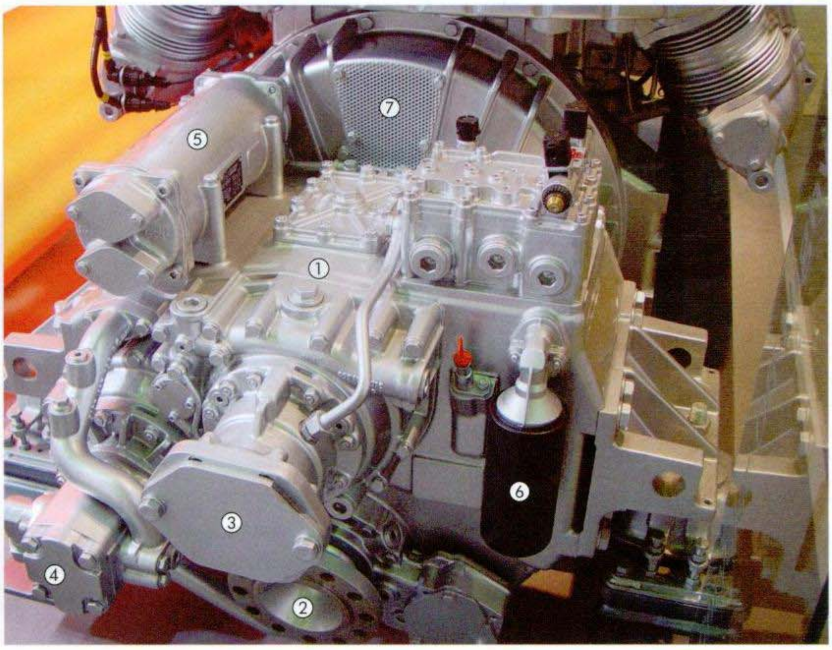
In low-speed two-stroke crosshead diesel engines the RPM varies between 250 and 105 rpm, the cylinder diameter from 260 to 600 mm and the shaft power from 1500 to 18,000 kW.

Larger crosshead engines are directly reversible and have a fixed-pitch propeller. The controllable-pitch propeller diameter varies from 1800 to 8600 mm.

A coupling section is often added to the gearbox, so the propeller can be stopped when the diesel engine and shaft generator are still running. All the engines in category III come standard with a gearbox, shaft generator, shafting and a controllable-pitch propeller.

20.2.3 C Ship propulsion with one or several high-speed four-stroke diesel engines

These engines operate on M.D.O.. These systems are often seen in smaller ships and inland navigation. Here the complete system comprises the diesel engine, a gear transmission, a coupling section, a P.T.O. for the shaft generator, shafting and adjustable-pitch propeller.



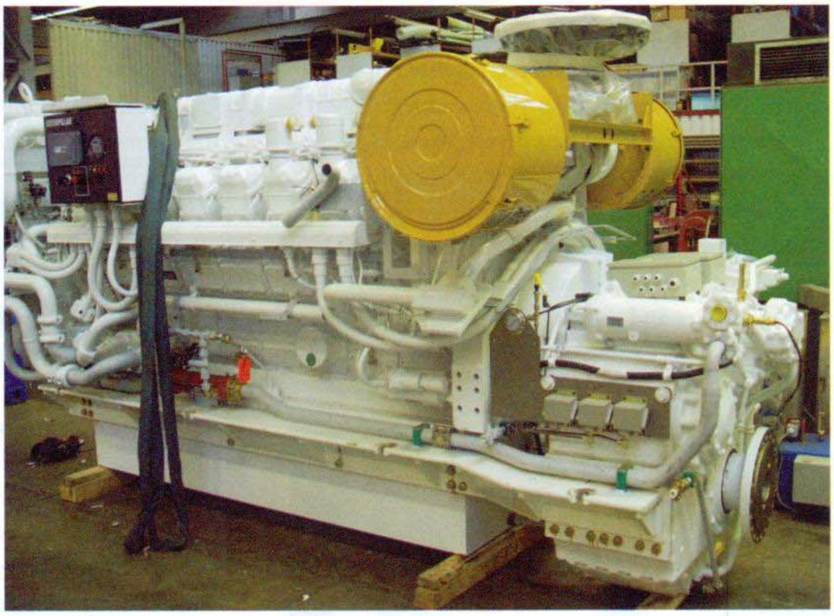
▲

A gearbox aft of a four-stroke high-speed MTU-diesel engine.

1 *reduction gearing*
2 *shaft to propeller*
3 *shaft to shaft generator*
4 *lubricating oil pump*
5 *lubricating-oil cooler*
6 *lubricating-oil filter*
7 *flexile coupling between engine and gearbox*

A four-stroke high-speed Caterpillar diesel engine for driving a large yacht.

The reduction gearing with lubricating-oil pump, lubricating-oil cooler and lubricating-oil filter attached to the engine.



Often two engines with two propellers are arranged for extra power generation. This system is also used in luxurious motor yachts.

picture Franco Pace

The world's largest private yacht, the 'Athena', has two four-stroke high-speed Caterpillar type 3516 B main engines, each of 1492 kW at an RPM of 1600.

There are three gensets of 290 kW each. The twin screws have a diameter of 1.6 metres and an RPM of 440. A maximum of 103,000 litres diesel fuel is in its bunkers.

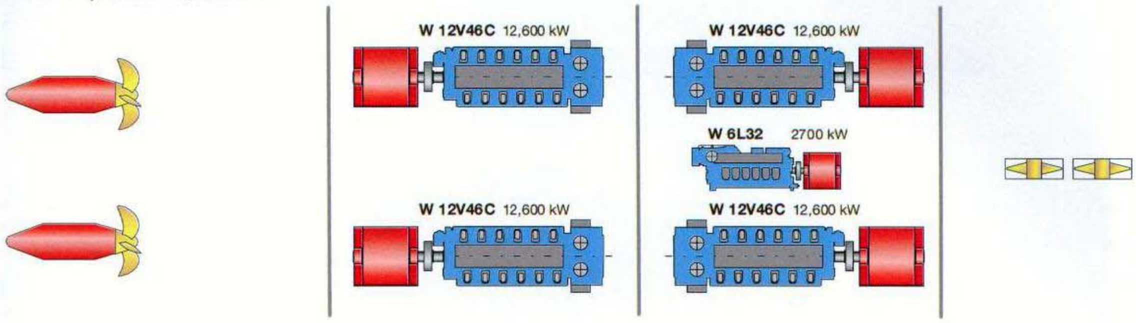
picture Hans Westerink

The engine room of the 'Athena'.

20.2.4 D Diesel-electric propulsion with medium-speed or high-speed four-stroke diesel engines

In the first instance, running on H.F.O. and in the second on M.D.O..
In this system electric gensets, which supply power to the electro-motors that drive the propeller(s) are driven by diesel engines. This system is often applied today. It is often used in passenger ships, ferries and in some self-propelled drilling platforms.

Diesel-electric (HFO)
Installed power: 53,100 kW



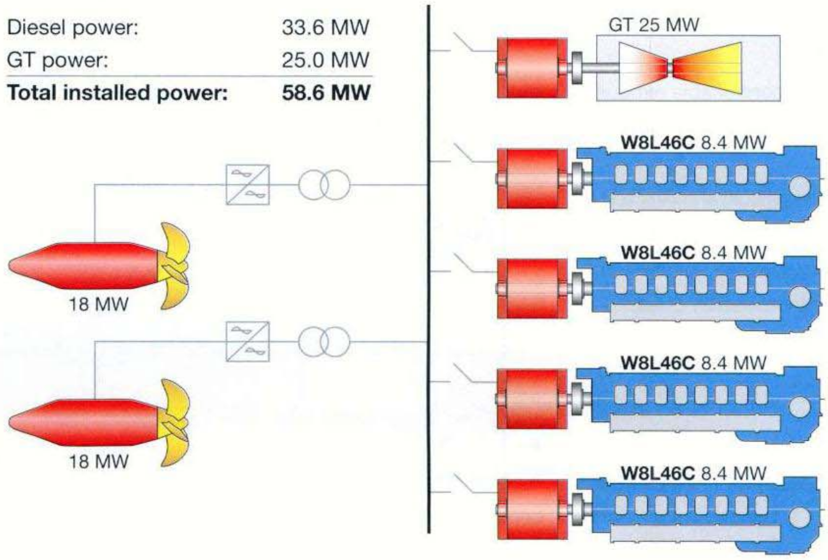
▲
A diesel-electric propulsion plant with four medium-speed four-stroke diesel engines running on H.F.O.

These plants are often installed on large cruise ships; the propellers are driven by electric motors. Depending on the required propulsion power and electric power requirement of the ship, one or more diesel generators may be deployed.

The diesel gensets can be placed in the most convenient location on the ship or vessel, as they do not have to be in-line with the propeller shaft as in the traditional arrangements.

Advantages
- **The diesel gensets operate at a constantly high RPM**, so diesel-engine contamination due to, amongst others, the low engine speed, is less at a low load.
- **At low speeds parts of the diesel gensets can be switched off.** Consequently, these have fewer operating hours and therefore require less maintenance.
- **Driving the propellers with electro-motors is independent of the diesel engines.** With adjustable propellers or frequency-controlled electric-drive motors, any type of propulsion is possible.
- **The generated electric power is also available for energy supply to the ship,** for instance on large passenger ships, ferries, self-propelled platforms and drilling ships.
  In diesel-electric propulsion, there is also the possibility to use a gas-turbine genset alongside the diesel genset. In this manner, a large amount of power can rapidly be generated with a limited size and weight. Only M.D.O. can be used. The exhaust-gas emissions of these gas turbines are much lower than those of diesel engines: ideal 'sight seeing' when sailing in ecologically sensitive areas. Well-known examples are the 'Jewel of Norway' and the 'Queen Mary II'.
- The total propulsion efficiency is sometimes even higher than for traditional propulsion when P.O.D.S. is applied.

Diesel power:	33.6 MW
GT power:	25.0 MW
Total installed power:	**58.6 MW**



▲
A diesel-electric propulsion plant.

Besides four diesel gensets running on H.F.O., a lightweight and compact gas-turbine genset running on M.D.O. is installed on the upper deck.

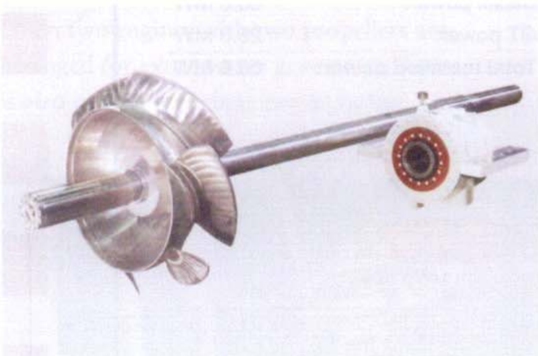
20.2.5 E Ship propulsion with water jets, with high-speed four-stroke diesel engines

A small, specific group of fast ships is not driven by propellers but by a water jet. The diesel engine drives a pump, which draws in water at a high speed and subsequently pumps this overboard. This so-called water jet can be steered in any direction thus rendering a rudder redundant.

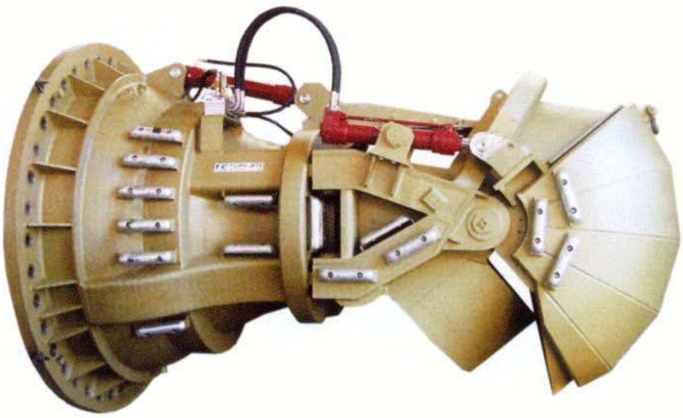
▶
Water-jets driven by high-speed four-stroke diesel engines are the ideal combination for the propulsion of fast, light catamarans for passenger and patrol craft.

▶
The controllable exhaust nozzle of a water-jet.

The amount and direction of the water can be accurately controlled for an optimum manoeuvrability of fast ships.



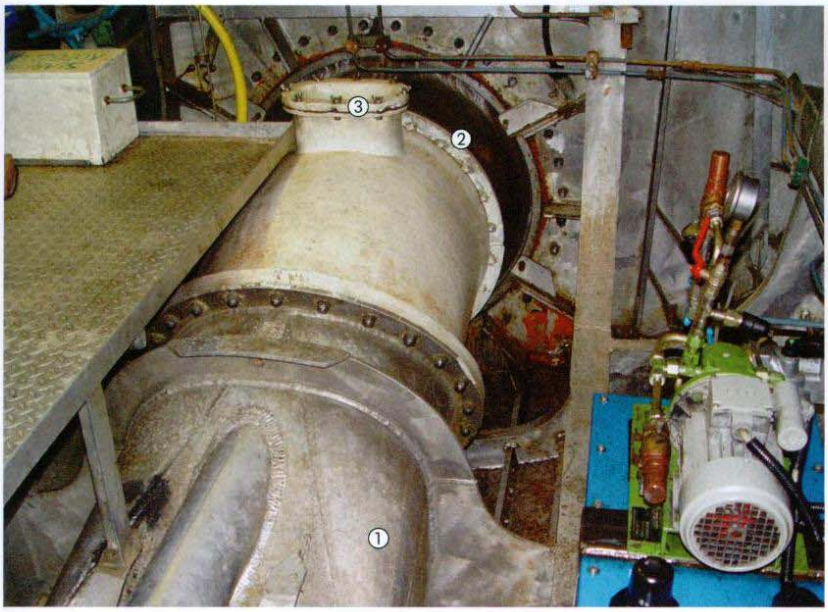
▶
A water-jet consists of a central pump section and an adjustable nozzle on the left. The pump is driven from the leftright by the diesel engine via a gearbox.



▶
A very large water-jet propulsion plant driven by a gas turbine on a luxury yacht which can achieve a speed of 40 knots.

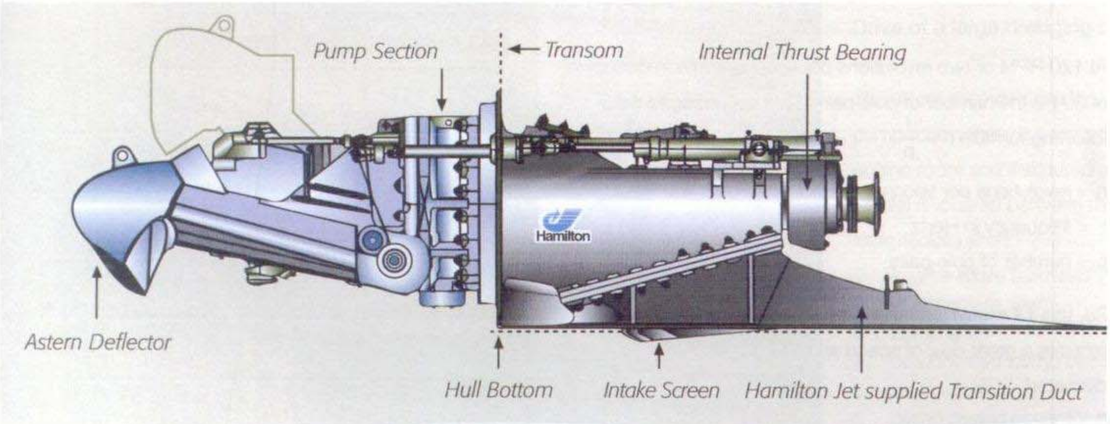
Left and right of the water jet, not visible underneath the ship are two propellers driven by high-speed four-stroke V-engines.

1 *location of the propellers*
2 *water-jet impeller*
3 *nozzle*



◄
One of the two water jets on the passenger catamaran 'Koegelwieck' of Doeksen Shipping Company, Harlingen, The Netherlands.

1 suction section: the pump is not equipped with a suction box to prevent cavitation.
2 impeller section off the stern bulkhead.
3 cover for inspection and removal of dirt.
The impeller and shaft are sturdy, so relatively heavy objects that are drawn in do not cause damage.



▲
A Hamilton water jet.

The under keel water is vertically drawn in and discharged on the left. With the 'Astern Deflector' the water-jet power is directed ahead, so that the vessel can sail astern. The pump section is situated just aft of the ship's stern.

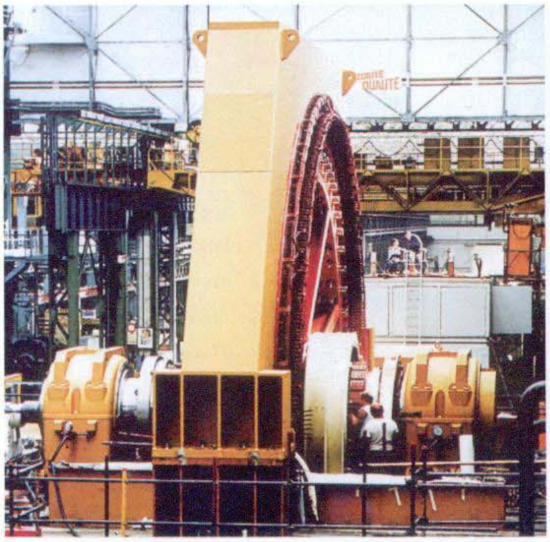
Water jets are often used in high-speed passenger transport such as catamarans, pilot boats and motor yachts.

▼
A diesel-power plant with three medium-speed four-stroke engines in V-arrangement of manufacturer Wärtsilä.

Right: the generators and left: the diesel engines.

20.2.6 F Diesel-power plants with both high-speed and medium-speed four-stroke diesel engines, as well as low-speed two-stroke crosshead engines Engine categories I to IV included

The engine operates on M.D.O. and H.F.O.. Most diesel-power plants are equipped with medium-**speed four-stroke diesel engines** running on H.F.O.. **This is often the most effective arrangement from an economic point of view.** At smaller capacities, or when no H.F.O. is available, **high-speed four-stroke diesel engines** running on M.D.O. are brought on-line. Semi-permanent plants and mobile gensets in containers also are in this category.



A diesel-power plant disc generator driven by a large low-speed two-stroke crosshead engine.

At 120 RPM or two revolutions per second and a frequency of 50 Hz the number of pole pairs 25 is according to the following formula $n = \frac{f}{p}$

n = revolutions per second

f = frequency in Hertz

p = number of pole pairs

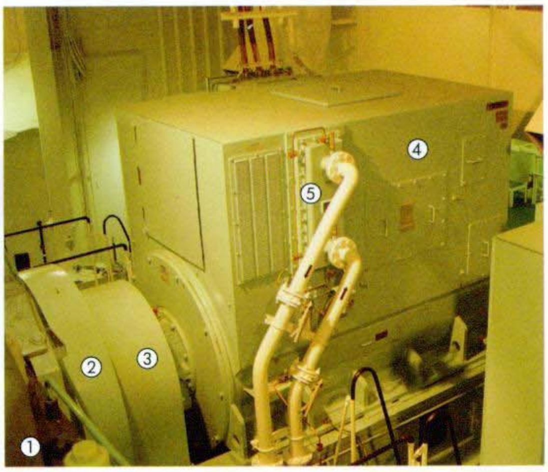
So, this **12 metre** diameter disc generator has **50** poles! This requires a great deal of space and a large generator diameter!

Large two-stroke crosshead engines are often used for larger diesel-power plants both on shore and at sea.

Genset drives

Here gear transmission is not used.
A flexible coupling between the diesel engine and the generator usually suffices.

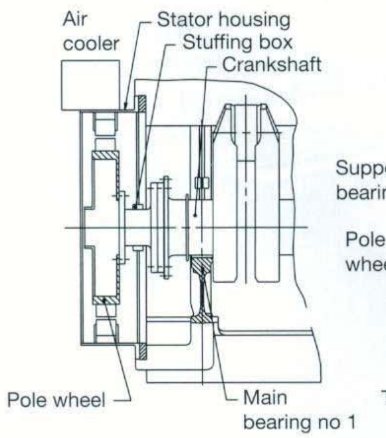
A large 3.5 MW diesel genset on a container ship.



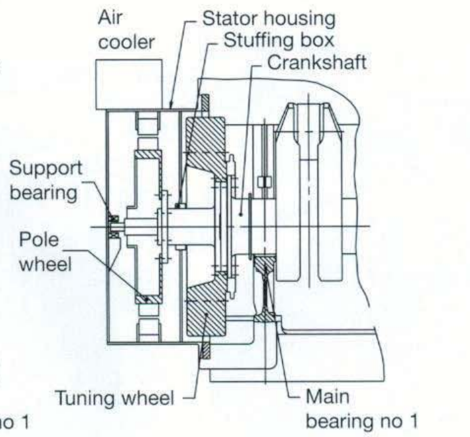
A similar 3.5 MW genset on a container ship.

1 *medium-speed four-stroke diesel engine*
2 *flywheel/turning wheel*
3 *flexible coupling, not visible*
4 *electric generator*
5 *closed cooling-air system: the air is cooled by water. Dust and dirt particles cannot enter this generator!*

Two examples of a flexible coupling between a large disc generator and a large two-stroke crosshead engine.



Standard engine, with direct mounted generator (DMG/CFE)



Standard engine, with direct mounted generator and tuning wheel

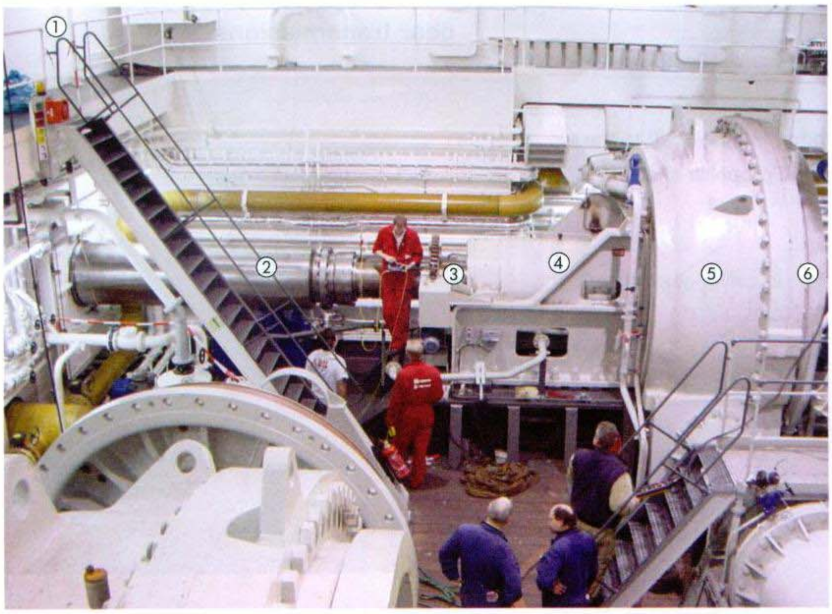
A simple flexible coupling used between the diesel engine and generator.

The rubber sections of the left shaft flange fit the holes of the right shaft flange. After approximately 30,000 running hours the rubber parts are renewed.

20.2.7 G Special drive for pumps in the dredging industry, compressors and pumps for use in the gas and oil industry with high-speed or medium-speed four-stroke diesel engines Engine categories I, II and III

The engine operates on H.F.O. and M.D.O..
In the dredging industry, large dredging pumps are either directly driven by a diesel engine or directly driven by an electro-motor. The latter is applied in larger dredgers. In smaller dredgers driven by a high-speed four-stroke diesel engine, gear transmission is generally used.

◄
Large dredgers require a lot of power for driving the dredging pumps.



◄
Drive of a large dredging pump on a dredger.

The diesel engines drive the pumps placed in a separate pump room; there is a watertight bulkhead between the engine room and the pump room. A flexible coupling (not visible) is located between the diesel engine and the intermediate shaft.

1 *engine-room bulkhead*
2 *intermediate shaft*
3 *pump shaft*
4 *oil lubricated pump bearing*
5 *pump casing*
6 *pump casing cover for inspection /repairs of the impeller*

◄
The same dredger viewed from the engine-room bulkhead.

The firm Machine Support is aligning the intermediate shaft to the pump.

In the process industry, diesel engines drive oil pumps or gas compressors. Due to increasingly strict emission regulations, the use of 'Dual Fuel' or gas engines is preferred.
The drive often uses a gear transmission. There is always a flexible coupling present.

20.3 Gear transmission

If gear transmission is used, the propeller speed can be independently selected from that of the diesel-engine crankshaft.
Clearly, the speed chosen is such that it is optimal for both the diesel engine and the propeller respectively, thus optimising the total propulsion efficiency.

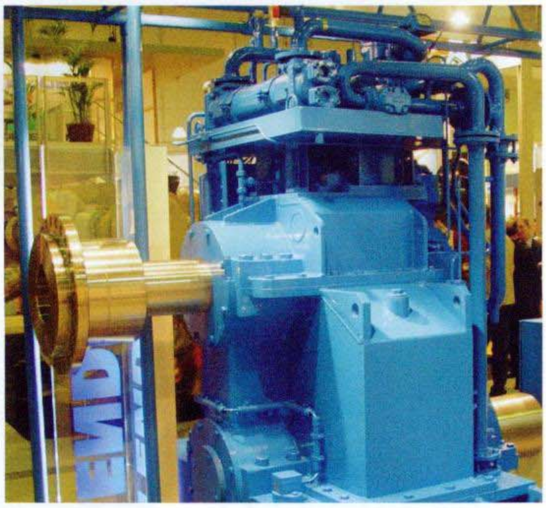
Gear transmission is also called transmission gearing. The ratio of engine speed to the propeller shaft speed is greater than 1.
It often lies between 2 and 4.

$$\text{Reduction} = \frac{\text{engine speed}}{\text{propeller-shaft speed}} > 1 \text{ often 2 to 8.}$$

In shaft generators, this is usually the reverse. The P.T.O.-shaft frequently has a speed that, for instance, exceeds the speed of the medium-speed diesel engine.

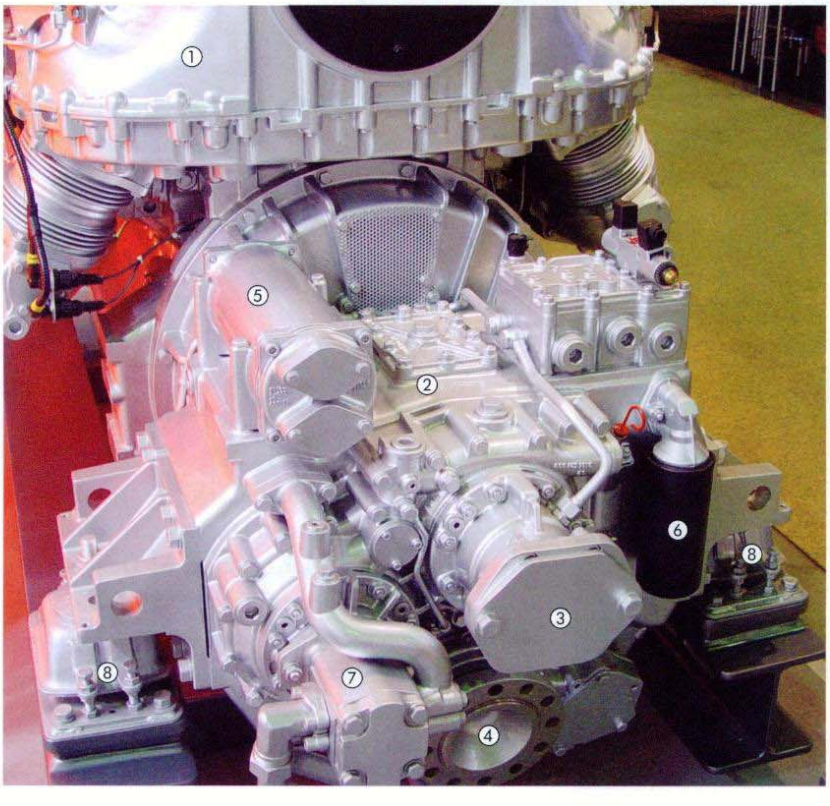
The shaft generator in this type of installation, for instance, has an RPM of 1500 and the diesel engine an RPM of 750.
The RPM of the diesel is increased by a factor $\frac{1500}{750} = 2$ via the gear box.

20.3.1 Examples of the possibilities with gear transmissions



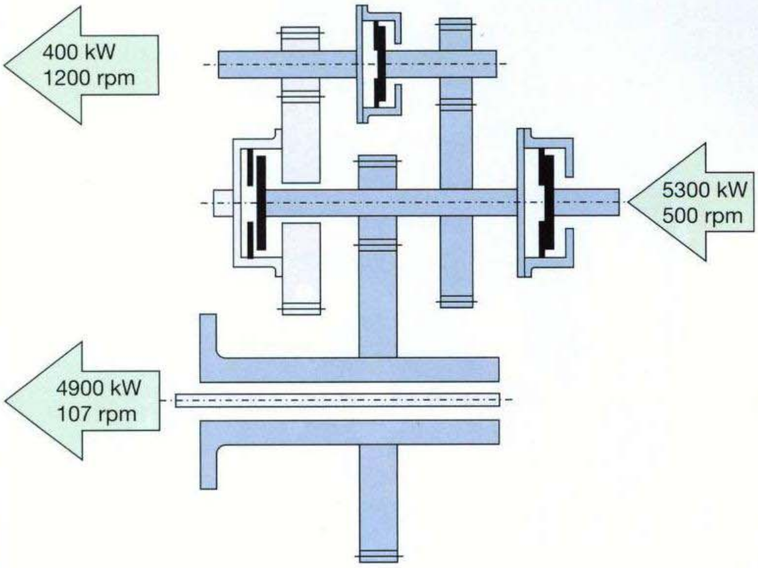
▸
A single reduction gearing with top left the shaft to the diesel engine and bottom right the propeller shaft drive.

On top of the reduction gearing, the lubricating-oil system with cooler and filters.



◂
A reduction gearing attached to a four-stroke high-speed diesel engine.

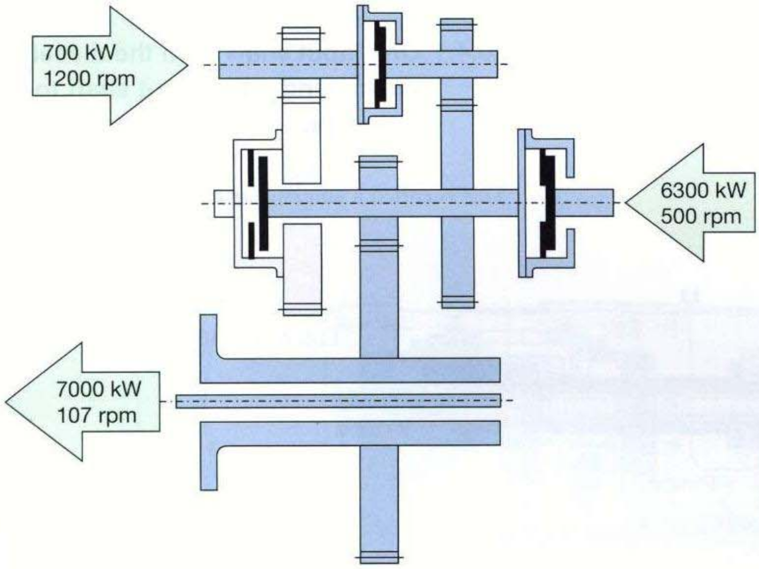
1 *diesel engine*
2 *reduction gearing*
3 *P.T.O.-shaft.*
4 *shaft to propeller*
5 *lubricating-oil cooler*
6 *lubricating-oil filter*
7 *lubricating-oil pump*
8 *vibration damper below the gear box integrated in the bedplate*



◄

Switch position I: Normal running at sea. Nominal output.

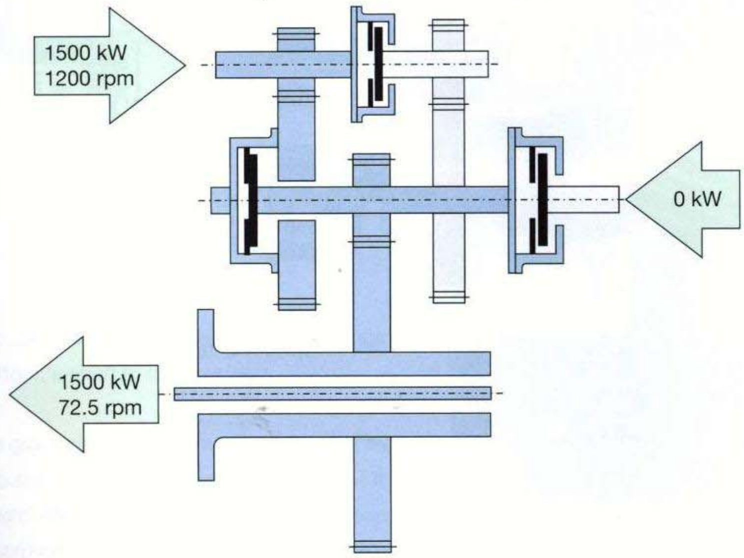
The main engine provides 5300 kW at 500 RPM and drives the controllable-pitch propeller with an RPM of 107. At an RPM of 1200, the propeller requires 4900 kW and the shaft generator 400 kW, to supply power for the ship's net.



◄

Switch position II: Extra power at sea.

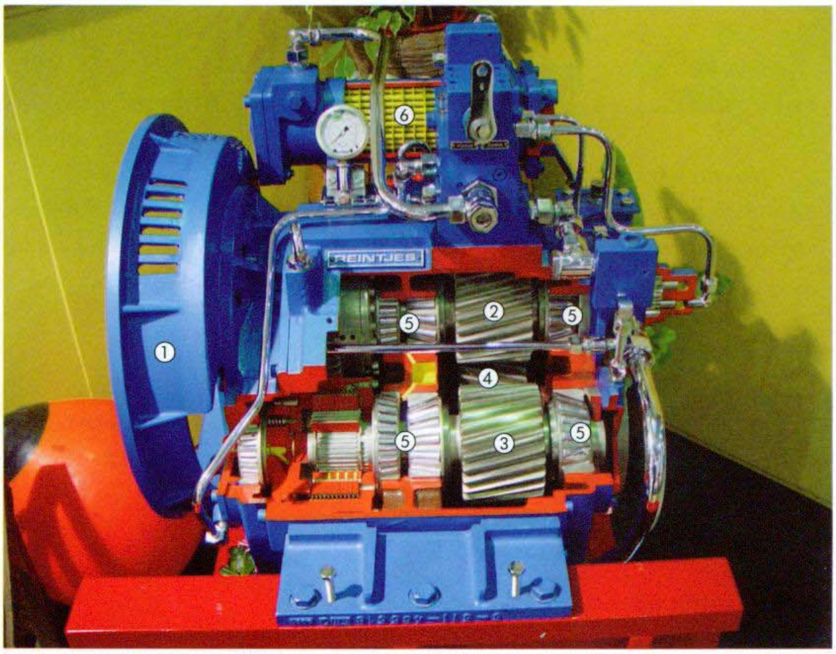
The main engine supplies its full power output of 6300 kW. The shaft generator is switched to the electromotor and supplies 700 kW. The electromotor draws power from the ship's electricity net and together with the main engine drives the controllable-pitch propeller with a maximum power output of 7000 kW.



◄

Switch position III: Emergency operation at sea or in port during shifting alongside the quay.

The main engine is stopped and repairs are being done at sea. Major maintenance can be performed in dry-dock. The switchable shaft generator/electromotor now provides the maximum capacity of 1200 kW to the c.p. propeller. This switch is referred to as the 'getting home' device. Admittedly, the speed is low; however, one can proceed to the nearest port for repairs to the main engine. In port the ship can shift berth.



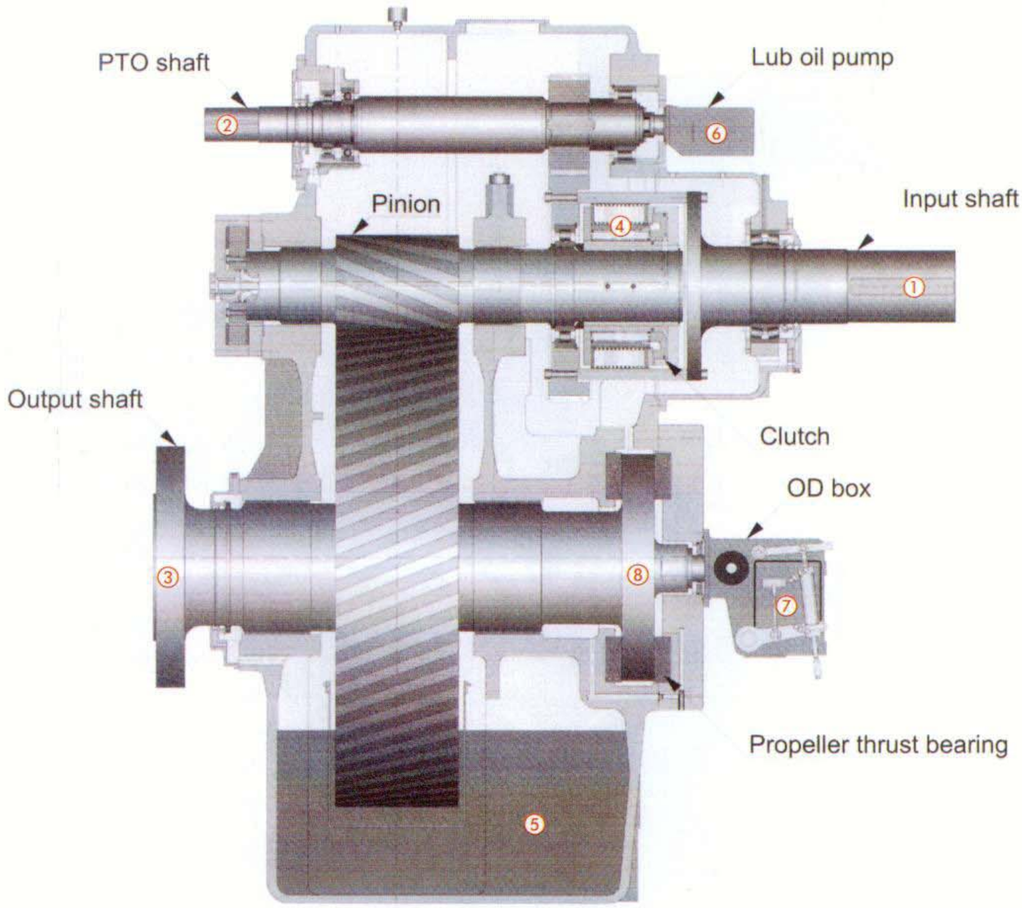
A cut-away section of a reduction gearing.

1 *engine flange*
2 *gear shaft from the crankshaft*
3 *gear shaft to the propeller shaft*
4 *helical gears*
5 *needle bearings*
6 *lubricating-oil cooler*

20.4 Various constructions and designs of gear transmissions

20.4.1 One input shaft from the diesel engine and one output shaft to the propeller.

This is called a single-gear transmission.

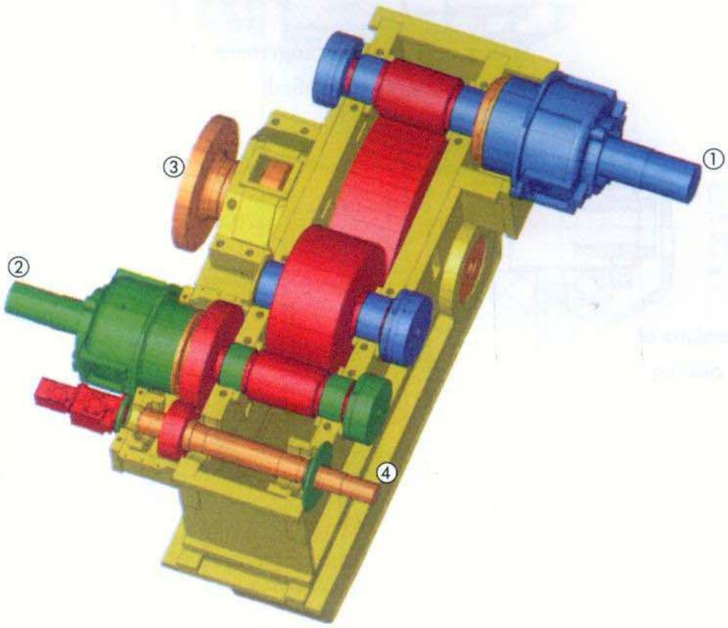


Single reduction gearing with P.T.O.

1 *diesel drive*
2 *shaft-generator drive*
3 *propeller drive*
4 *clutch*
5 *lubricating oil sump tank*
6 *lubricating-oil pump*
7 *lubricating-oil distribution unit*
8 *thrust bearing*

20.4.2 Two input shafts from two diesel engines and one output shaft to the propeller

Significant for the construction of multiple-gear transmissions is the mutual centre-to-centre distance of both diesel engines. If the centre-to-centre distance is large, intermediate shafts are used.



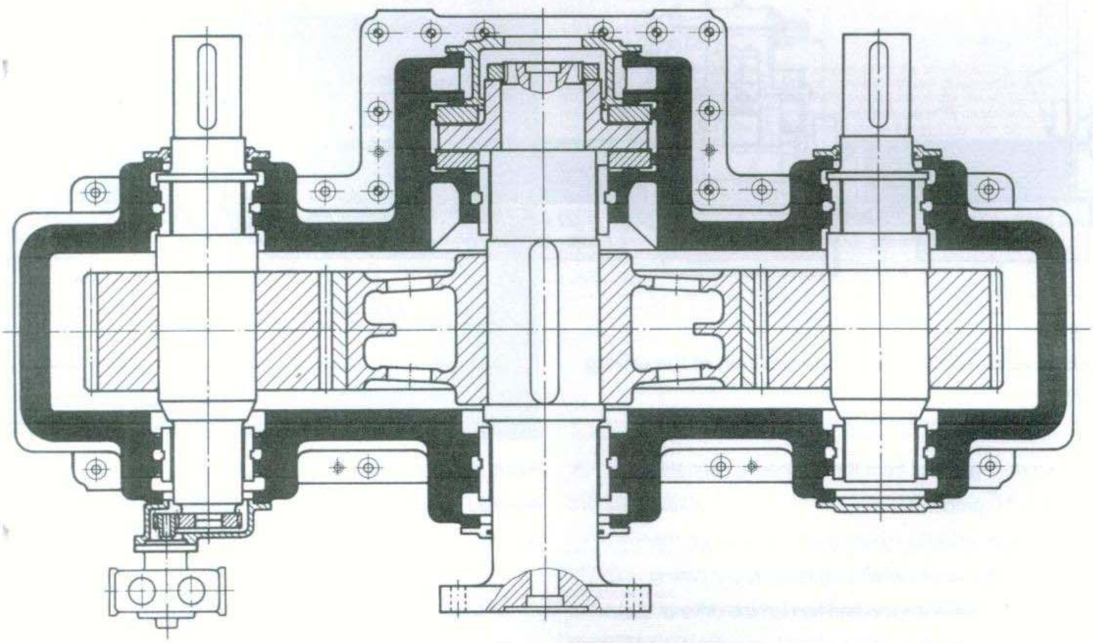
▶

Double reduction gearing.

Both diesel engines drive the propeller, one on each side.

1 diesel engine in forward engine room
2 diesel engine in aft engine room
3 shaft to the shafting/propeller
4 P.T.O.-shaft for driving the shaft generator and also the possibility to drive the propeller with the same shaft generator/electromotor going dead slow.

This arrangement with two propulsion engines in separate engine rooms is obligatory for large cruise ships. If one engine room would be flooded or on fire, 50% of the propulsion power can still be supplied from the other engine room.



▲

Top: double reduction gearing for two engines, and bottom: a shaft to the propeller.

The gear toothing is straight cut. Bottom left; the lubricating-oil pump. The outgoing pinion wheel is hollow to save weight. Above the pinion wheel; the thrust bearing for absorption of axial forces generated by the propeller thrust.

20.5 Position of the input and output shaft

20.5.1 Input and output shafts on the same side as that of the gear transmission

This is called a U-arrangement. This construction method is seldom applied.

20.5.2 One shaft on either side of the gearing

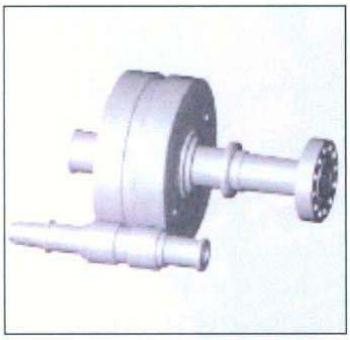
This is called the Z-arrangement. The input and output shafts run parallel to each other. Normally, the centre lines are positioned above each other, but occasionally they are placed alongside.

Designs

The following photos show various designs of gearing.

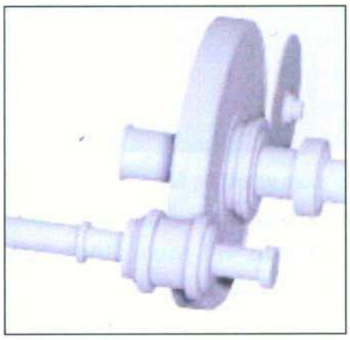
▶
Various versions of reduction gearing.

left: double helical teeth
centre: single helical teeth
right: single helical teeth with thrust collar for two rotational directions

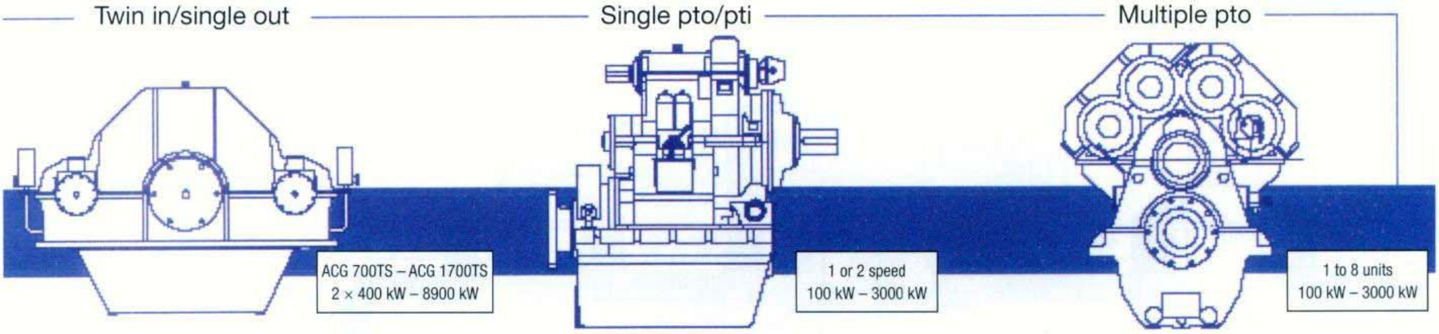


Double helical teeth

Single helical teeth



Single helical teeth with thrust collar



▲
Various versions of number and position of the driving shafts.

Left: two driving shafts from the engines; a central shaft for driving the propeller.
Centre: single-reduction gearing with bottom left the driving shaft to the propeller. Centre right, the driving shaft from the diesel engine and top left the drive of the shaft generator /P.T.O. / P.T.I.

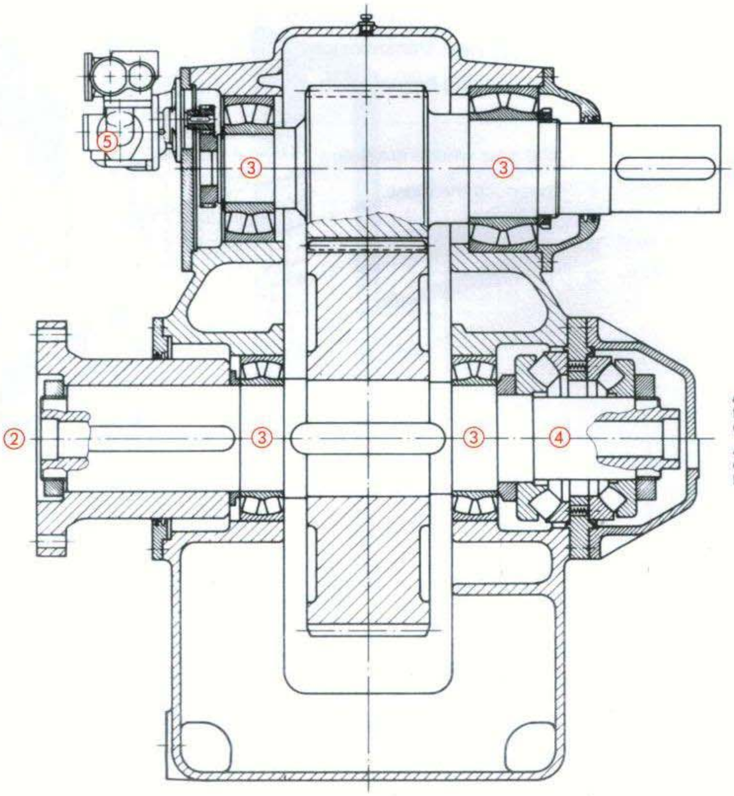
P.T.O.: Power Take Off shaft for driving the shaft generator
P.T. I.: Power Take In shaft for driving the propeller with the shaft generator-electromotor.
Right: reduction gearing and diesel engine together with several P.T.O. shafts for driving the shaft generators, pumps and other equipment.

20.6 Types of teeth

We distinguish three shapes:

1 straight teeth or spur teeth;
2 helical teeth;
3 teeth in V-shape.

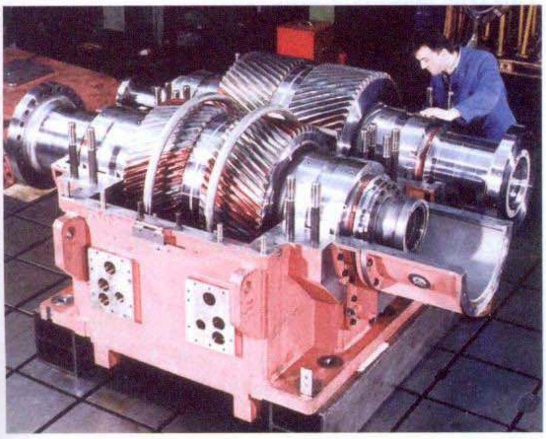
The toothed flank of these cogwheels usually has an involute shape. An involute is described as the curve traced by a point on a taut, inextensible string as it unwinds from another curve.



▶

A single transmission with spur gears.

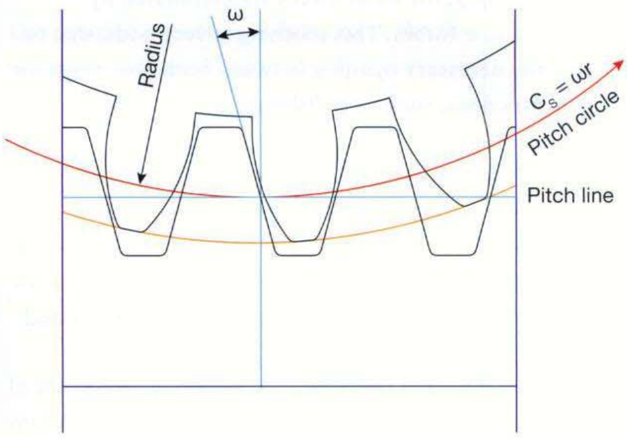
1 *driving shaft of the diesel engine, the so-called inlet*
2 *driving shaft towards the propeller, the outlet*
3 *bearing*
4 *thrust bearing*
5 *lubricating-oil pump*



▲

Double helical teeth.

The space between both rows of helical teeth is clearly visible. This is necessary as the teeth must be ground and polished. Therefore the reduction gearing is fairly long. The axial forces generated by the helical teeth neutralise each other completely.



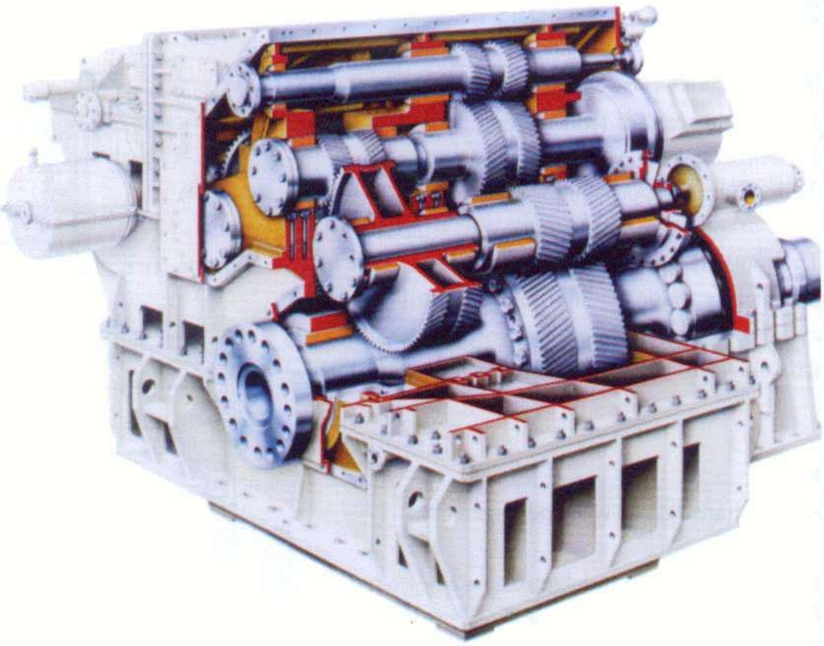
▲

Involute teeth.

In practice, in involute gearing, virtually all the teeth profiles are involutes of a circle. These are usually called involute teeth. The force line (or Line of Action) runs along a tangent common to both base circles. The points (of contact) move along the stationary force-vector 'string' as if it were being unwound by the rotating circle. The involute is entirely determined by the radius of the base circle, so all the involutes derived from the base circle are congruent.

A large gear transmission with double helical teeth.

The gear wheels are hollow to reduce the mass.



20.6.1 Comments

Helical gear teeth produce axial forces, which must be absorbed by the axial bearings. Advantages of helical teeth: more teeth are simultaneously in contact with each other, which provide a smooth operation. With double toothing, the axial forces are eliminated by counter forces. This toothing is very wide, due to the necessary opening between both gear rings for processes, such as polishing.

In this type of toothing, the pinion and the wheel are at a fixed axial position from each other. Therefore, both the pinion and the wheel must be able to move simultaneously. This is possible with the use of a coupling between the crankshaft and the drive shaft (pinion).

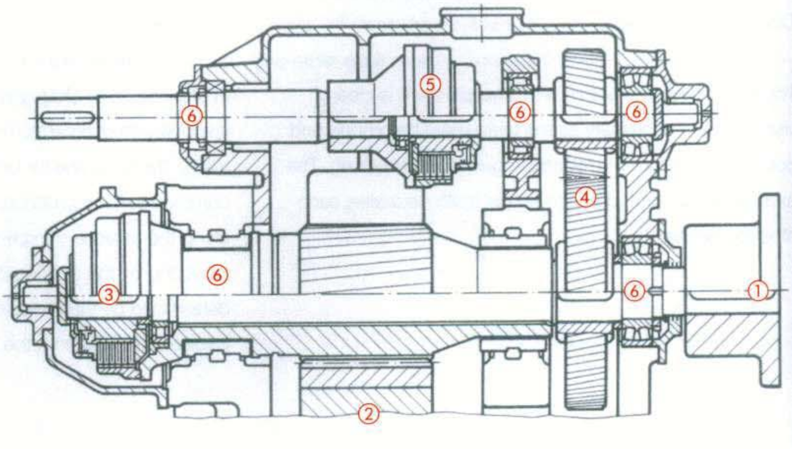
20.7 Gear transmission for shaft generators

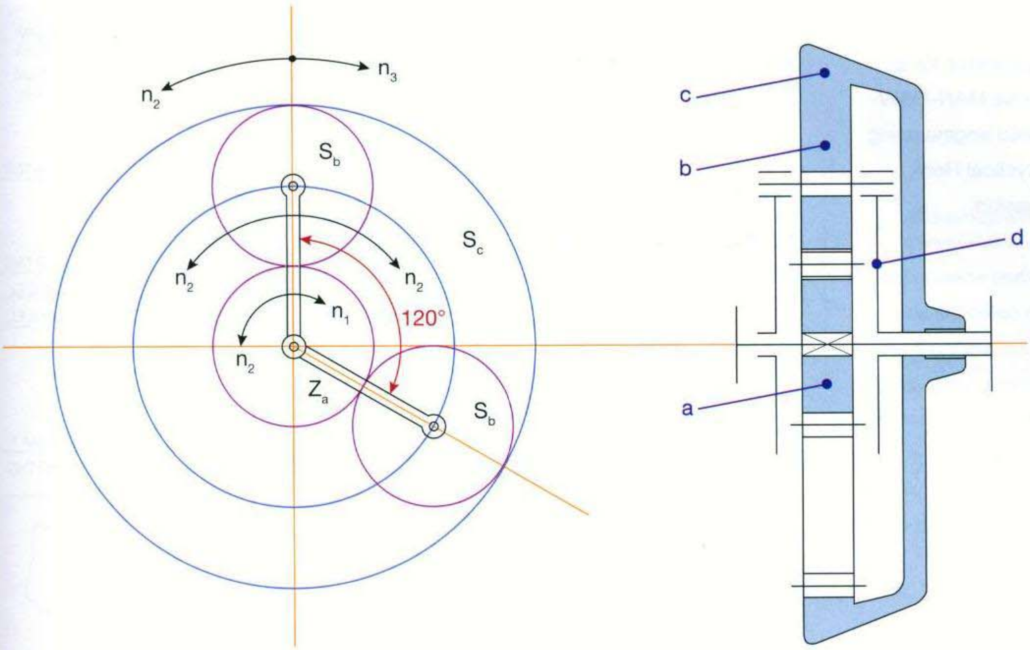
Shaft generators are driven by four-stroke or two-stroke engines and are occasionally equipped with a special kind of gearing, the so-called **epicyclical transmission.**
This type takes very little space, imperative for, for instance, a four-stroke propulsion engine, which is installed aft. Here the available space is limited due to the space that is already used for the gearbox between the engine and the propeller.

It is designed with a solar wheel, a stationary planet carrier and an output shaft with ring wheel.

Reduction gearing for a shaft generator.

1 *drive from the diesel engine*
2 *drive towards the controllable-pitch propeller*
3 *clutch to the controllable-pitch propeller*
4 *drive to shaft generator*
5 *clutch to shaft generator*
6 *shaft bearings*





◄ The principle of an epicyclical reduction gearing.

a *solar wheel*
b *planet wheel*
c *ring wheel*
d *shaft support*

Two types are commonly used.

I The driving shaft is connected to the solar wheel, the stationary planet carrier and the ring wheel.

II The driving shaft is connected to the solar wheel, the stationary ring wheel and the output shaft with a planet carrier.

In the first arrangement, the circumferential velocity of all the wheels is identical and the rotation speeds of the wheels are inversely proportional to their diameter. This is called the star type.

The transmission ratio i is equal to:

$$i = \frac{n_1}{n_3} = \frac{dr}{ds},$$

where:

n_1 = rotation speed of the solar wheel
n_3 = rotation speed of the ring wheel
n_2 = rotation speed of the planet wheel
ds = diameter of the solar wheel
dr = diameter ring wheel
dp = diameter planet wheel.

Planet and ring wheels rotate in the opposite direction to that of the solar wheel.

In the second arrangement, the ring wheel is fixed and the solar wheel is driven by the input shaft; the output shaft is driven by the planet carrier. This is called the planetary type.

The rotation speeds now differ from the first instance:

Solar wheel $ns^1 = n_1 + n_3$
Planet carrier $n_2 = n_3$
Ring wheel $nr^1 = n_3 + n_3 = 0$

In this instance the transmission ratio is:

$$i^1 = \frac{ns^1}{n_2} = \frac{n_1 + n_3}{n_3} = 1 + \frac{n_1}{n_3} \text{ also } i^1 = 1 + \frac{dr}{ds}.$$

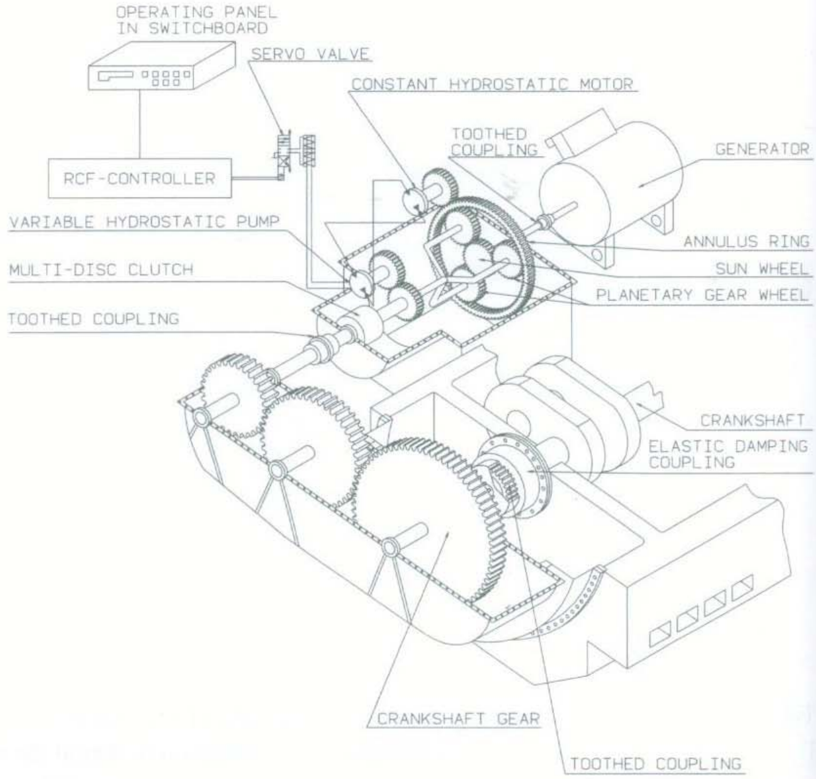
In the second example the input and output shafts rotate in the same direction.

There is also a third type, the so-called solar type. Here the solar wheel is permanently fixed.

The transmission ratio then equals $\frac{n_1}{n_2}$ or $\frac{ns}{np}$

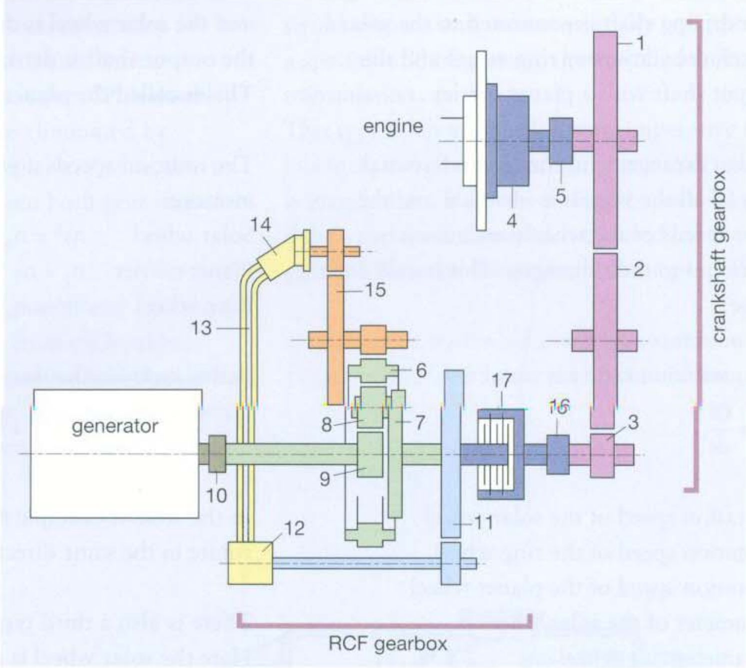
Shaft generator for a two-stroke MAN–B&W crosshead engine using an epicyclical Renk transmission.

The toothed wheel in the centre is called the solar wheel. Surrounding it are three planet wheels and the outer row of teeth is called the ring wheel.
An hydraulic oil pump is driven from the diesel engine driving shaft.



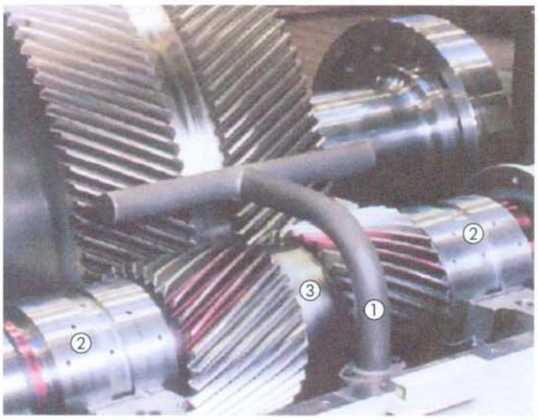
A combination of normal reduction gearing, right, with epicyclical reduction gearing, centre.

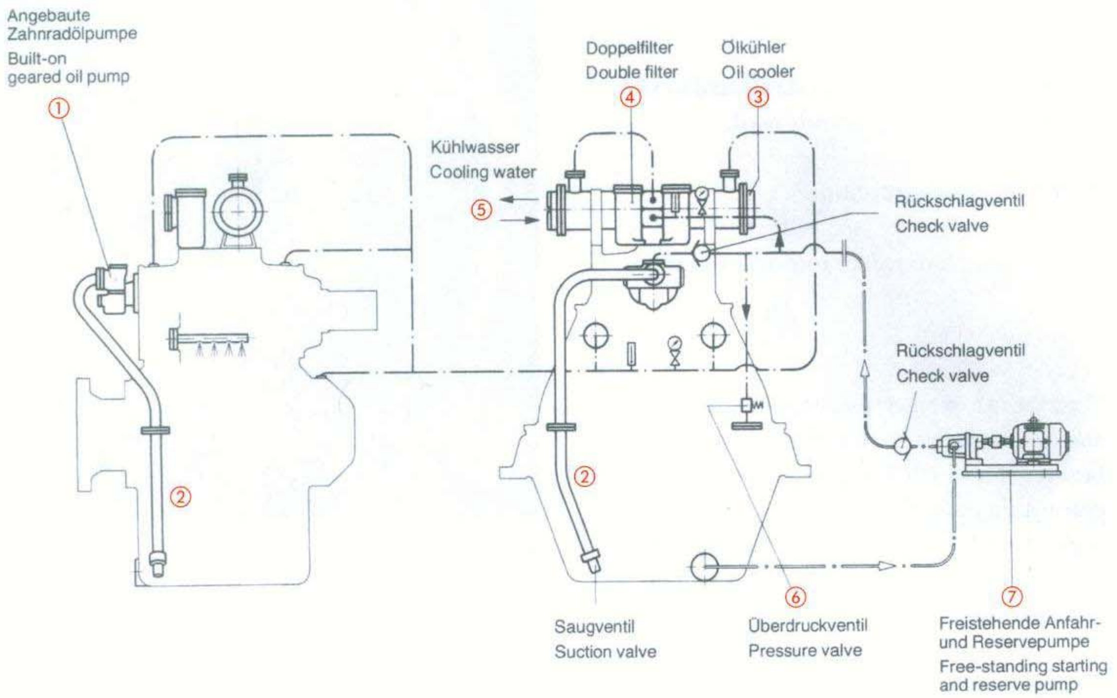
1 first gear wheel, engine speed
2 second gear wheel, engine speed slightly increased
3 third gear wheel, engine speed higher
4 flexible coupling
5 coupling flange
6 ring gear wheel
7 propeller shaft system
8 planet wheel
9 solar wheel
10 coupling flange
11 reduction gearing
12 hydraulic pump
13 hydraulic pipes
14 hydraulic motor
15 reduction gearing
16 coupling flange
17 plate coupling



The lubrication of gear wheels is usually done with a piping system where the lubricating oil is sprayed on the toothed wheels through small perforations.

1 lubricating oil supply pipe
2 upper bearing shell, the bearing caps have been removed
3 helical teeth with intermediate spaces

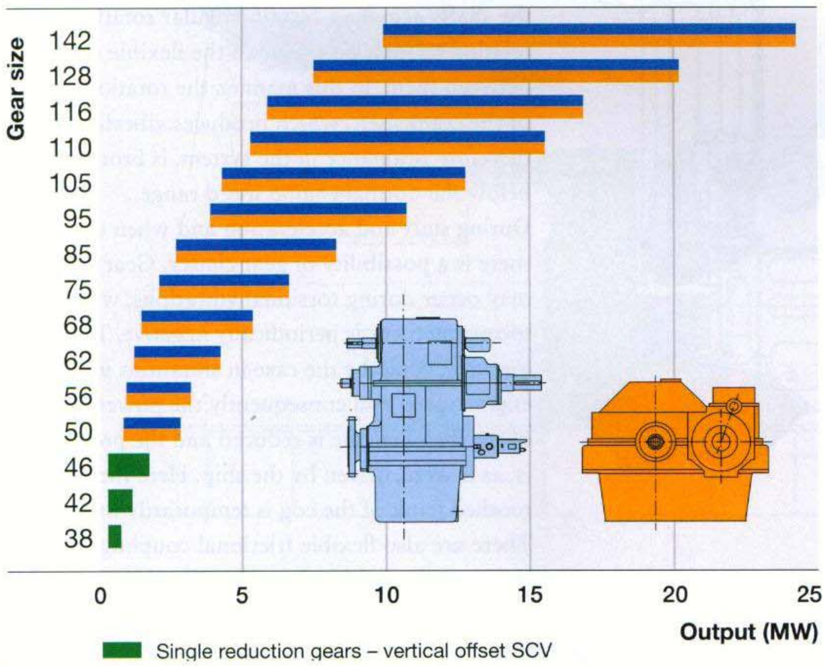




◄ The lubricating-oil system of a single reduction gearing.

1 *built-on gearwheel pump*
2 *supply line with non-return valve*
3 *lubricating-oil coolers*
4 *duplex lubricating-oil filter*
5 *cooling water*
6 *pressure valve*
7 *electrically driven spare lubricating-oil pump*

Wärtsilä reduction gears - Output range



▲ Wärtsilä reduction gearing program.

left figure

vertical: reduction gearing size

horizontal: maximum shaft power

Wärtsilä TCH gear range of twin input-single output gears

Gear size	Engine crankshaft offset (mm)*
TCH190	1 900
TCH240	2 400
TCH250	2 500
TCH270	2 700
TCH290	2 900
TCH300	3 000
TCH320	3 200
TCH330	3 300
TCH340	3 400
TCH350	3 500
TCH370	3 700
TCH380	3 800

* For the TCH type of gears the main parameter is the required horizontal offset between engine crankshafts. However, also the engine power and

right figure

A schedule with the various types of gear boxes with double diesel-engine drives.

The crankshaft diameter reaches 3800 millimetres per diesel engine.

The power output rises to approximately 25 MW; these are the largest four-stroke medium-speed diesel engines. Higher outputs are usually generated by low-speed two-stroke crosshead engines.

20.8 Couplings

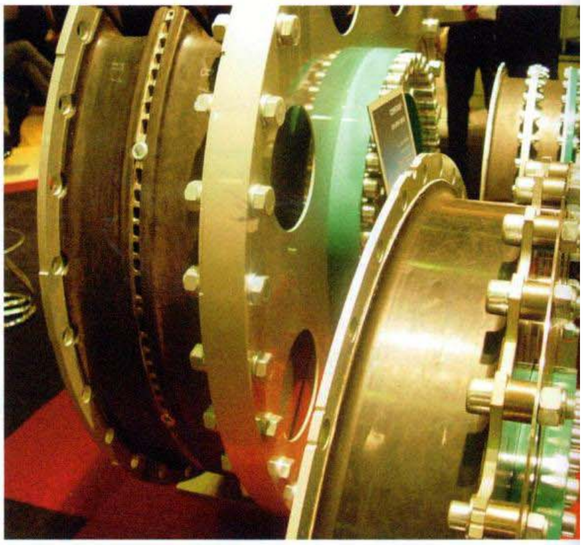
Couplings can be divided in elastic and flexible couplings. Both are frequently used.

20.8.1 Elastic couplings

The objective is to reduce torsional vibrations.

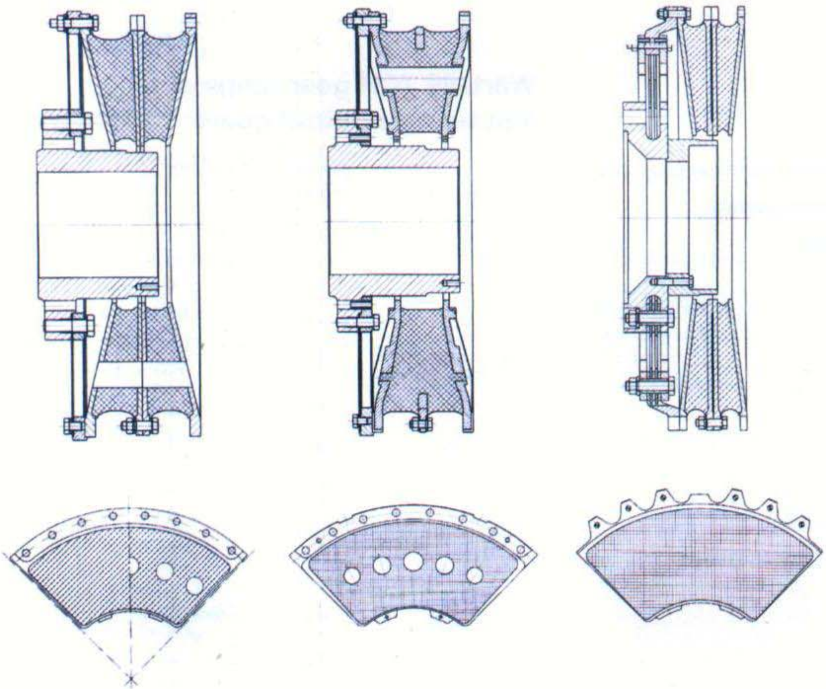
Also, see Chapter17, Vibrations and balancing.

With the use of an elastic material, plate springs and/or with hydraulic damping, the vibrations between, for instance, the engine and the reduction gearing, the reduction gearing and propeller, the engine and the generator are reduced.
Damping ultimately achieves a reduction of the torsional vibrations in the crankshaft. Due to an increase in the power density in diesel engines, vibration damping has become increasingly important.



▲
Three types of flexible couplings of manufacturer Centa.

Left, the flexible section can be clearly seen. The large holes in the coupling disc are for ventilation.

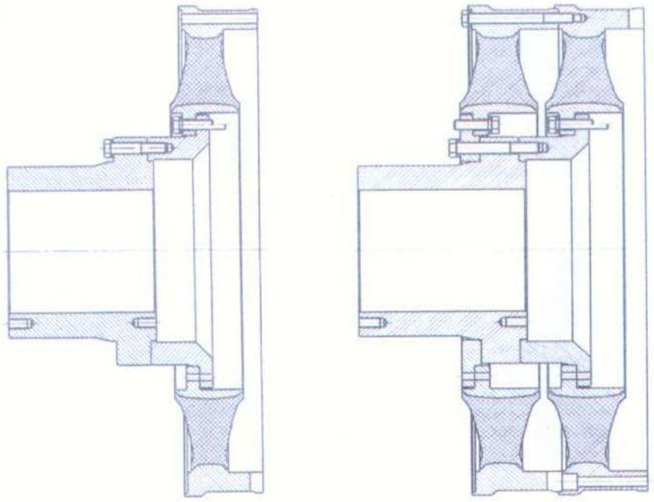


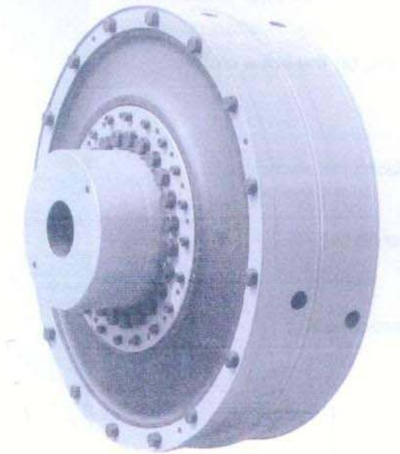
▲
The various types of Vulcan flexible couplings.

coupling 1: standard coupling with or without axial ventilation holes
coupling 2: version with wide flexible segments for maximum radial alignment deviations
coupling 3: version with narrow flexible elements for minimum power loss
The manufacturers of flexible couplings have an elaborate delivery package as flexible coupling requirements may vary considerably.

The working principle is that a coupling with a minor torsional stiffness is placed between the engine and the gearbox. Both fixed parts on the shafts acquire a certain angular rotation in relation to each other due to the flexible section between them. In this manner, the rotation speed of the crankshaft, which produces vibrations and therefore resonance in the system, is brought well below the normal engine speed range.
During start and acceleration and when stationary, there is a possibility of gear clatter. Gear clatter may occur during torsional vibrations, where the torque moment is periodically negative. This, for instance, could be the case in situations where the engine speed and consequently the power output of the diesel engine is reduced and the propeller is, as it were, driven by the ship. Here the other toothed flank of the cog is temporarily loaded.
There are also flexible frictional couplings that can prevent this problem by starting and stopping the engine with a disengaged coupling. The coupling is only put in service at sufficiently high engine speeds when gear clatter no longer occurs.

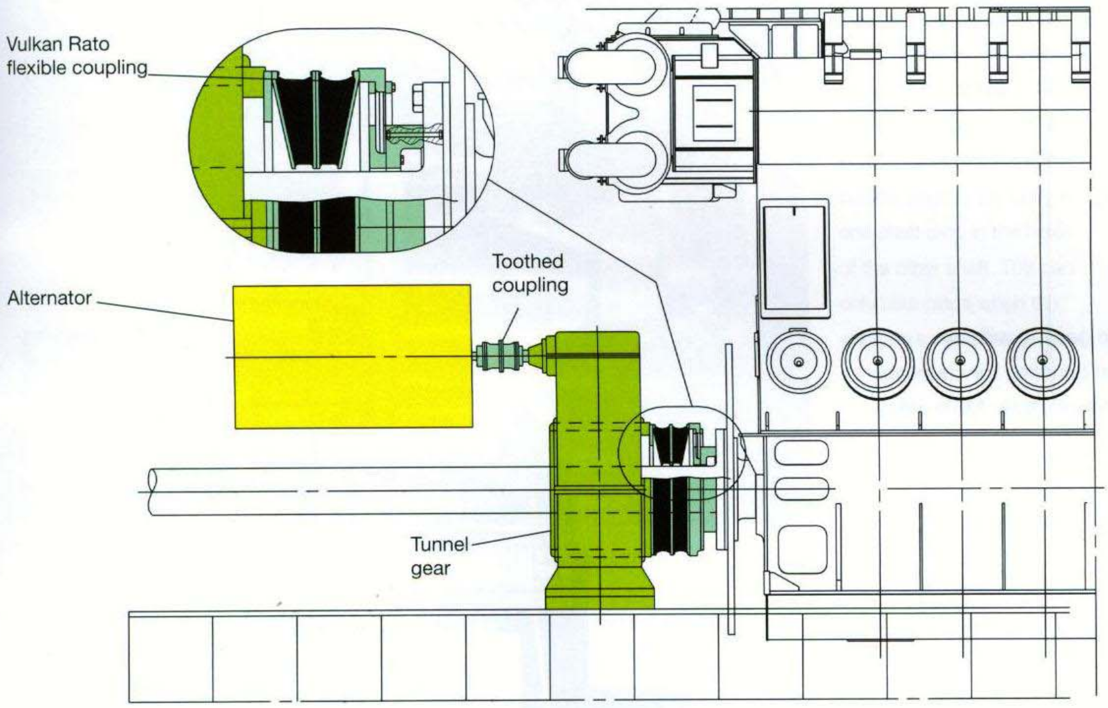
Examples of elastic couplings.





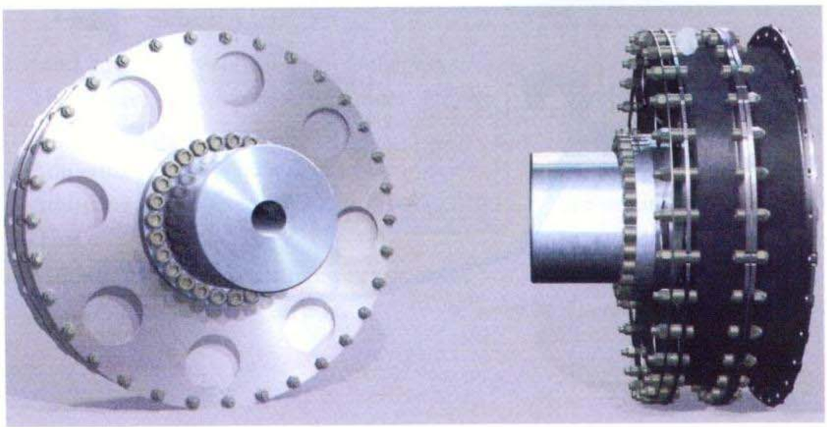
A narrow and a wide coupling.

Clearly visible in the cross-section, is that the left coupling is considerably less elastic than the centre coupling, which consists of two flexible rubber parts.



The flexible coupling between a low-speed two-stroke crosshead engine of MAN–B&W and the propeller shafting.

The shaft generator drive has a toothed sliding coupling. In this manner, the axial expansion of the reduction gear transmission that occurs between the stationary and operating temperatures can be compensated.



A flexible Centa coupling with two segments placed behind each other for maximum flexibility.

Note the large ventilation holes for discharging the heat generated in the flexible elements that can lead to deformation.

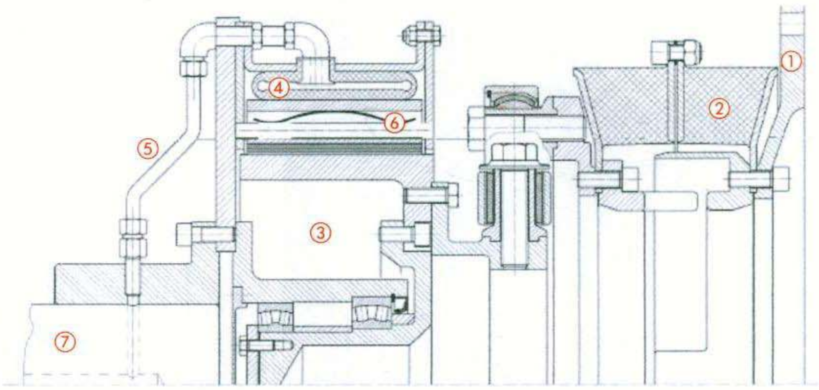
20.8.2 Examples of flexible couplings

▶

A flexible Centa coupling, right; on the left a clutch coupling which runs on air.

The coupling is achieved by inflating the elastic space (gland).

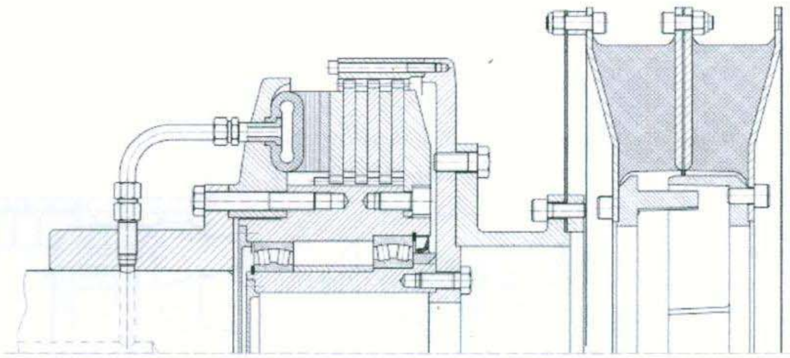
1 *flange attached to the diesel engine*
2 *flexible segments*
3 *clutch*
4 *completely circular elastic space (gland)*
5 *air pipe*
6 *spring plate*
7 *driving shaft for the controllable-pitch propeller*



▶

A Centa flexible coupling combined with a Wichita friction coupling working on compressed air.

left: The discs of which one part is fixed to one half of the shaft and the other part to the other half of the shaft. The discs are pressed together by expansion of the inflatable tube, and the engine shaft from the right drives the shaft (left) towards the propeller. When the propeller shaft is uncoupled, the friction plates are not in contact with each other.
right: the double flexible coupling

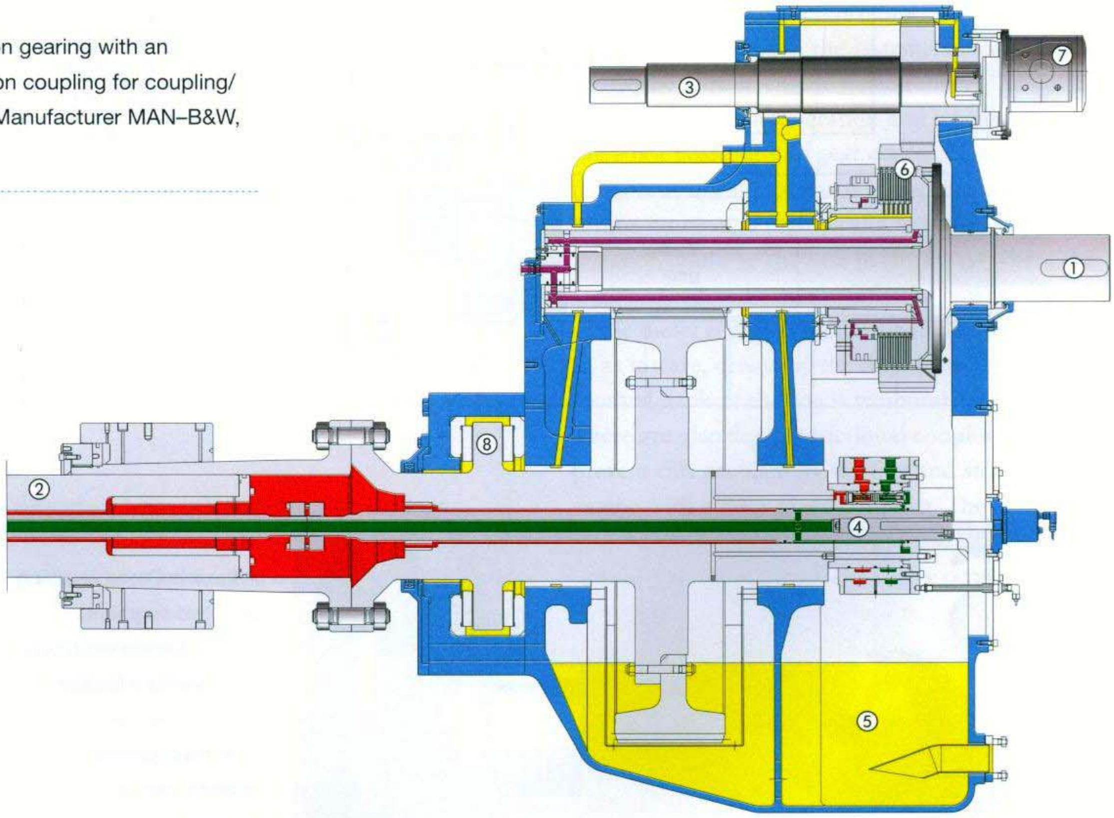


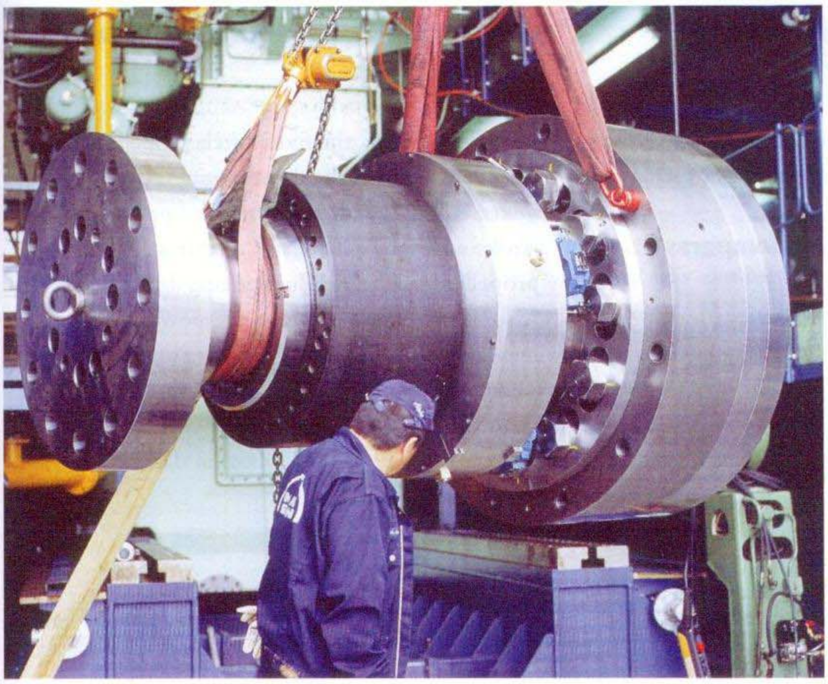
▶

A cross-section of a reduction gearing with an hydraulically controlled friction coupling for coupling/uncoupling of the propeller. Manufacturer MAN–B&W, system Alpha.

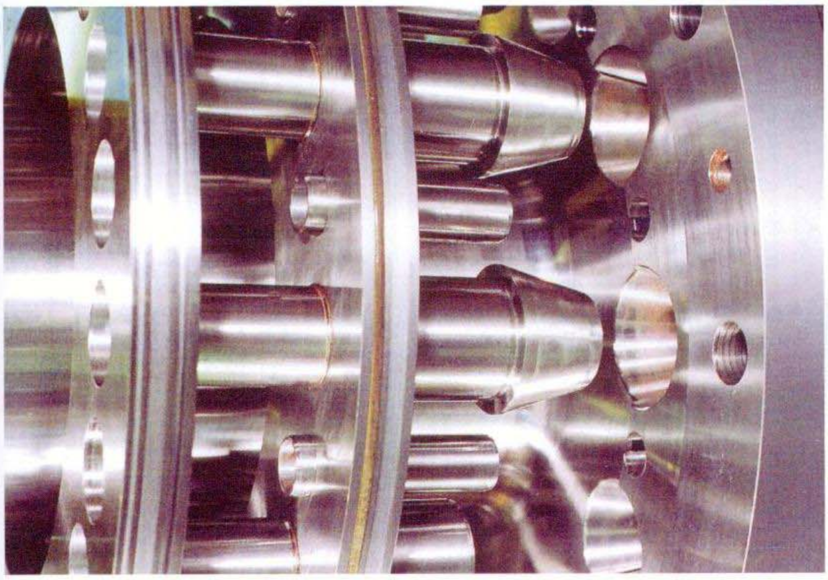
1 *diesel-engine drive shaft*
2 *propeller shaft*
3 *electric generator shaft*
4 *hydraulic section for operating the controllable pitch propeller*
5 *lubricating-oil drain tank*
6 *friction coupling*
7 *lubricating-oil pump*
8 *thrust block*

yellow: lubricating oil for the reduction gearing
red / green: hydraulic oil for operating the controllable-pitch propeller





A mechanical coupling of MAN–B&W. Type conical coupling bolts.



The conical coupling bolts of the mechanical coupling of MAN–B&W.

With coupled shafts, the conical ends of the bolts of one shaft slide in the holes of the other shaft. This can only take place when the shafts are stationary.

20.9 Torsional vibration dampers

It is imperative that the normal rotation speed does not coincide with the critical rotation speed as this could produce unacceptable vibration tensions in the crankshaft.

This can be prevented in the following ways.

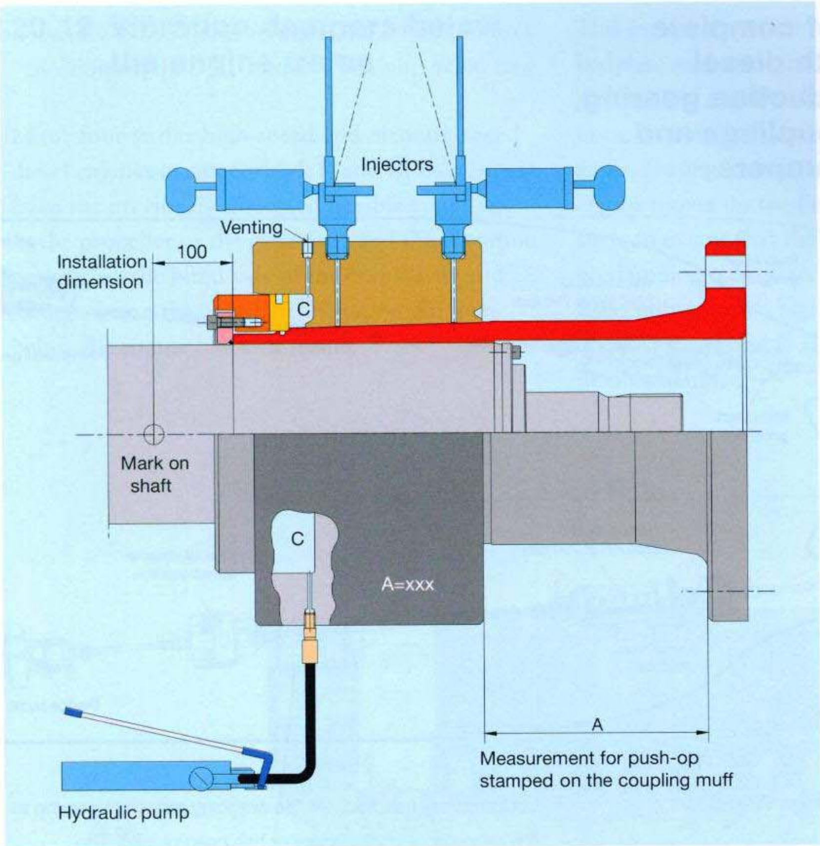
1 The correct selection of the cylinder ignition sequence.
2 Adjusting the mass inertia moments.
3 Adjusting the stiffness of certain parts of the shaft.
4 Mounting an elastic coupling between the diesel engine and the gearbox.
5 Installing torsional-vibration dampers, the torsional vibration can be reduced to an acceptable level, if points 1 to 3 are insufficient.

See Chapter 17, Vibrations and balancing for points 1 to 3.

Torsional vibration dampers

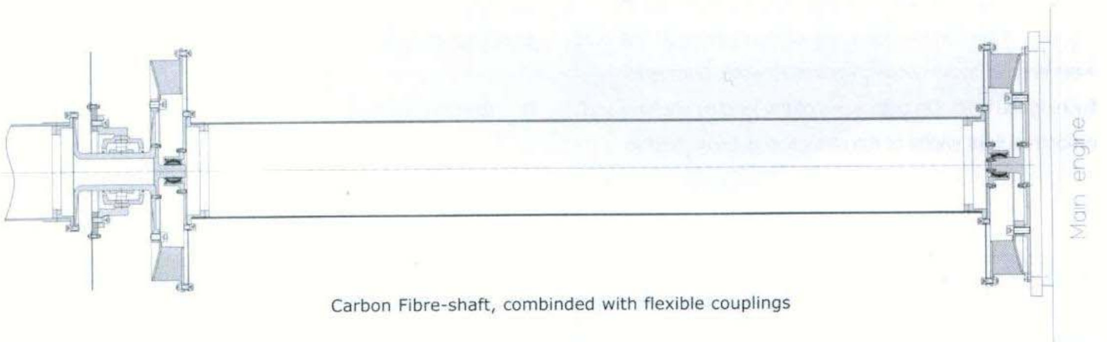
These are used in both two-stroke crosshead engines and four-stroke trunk-piston engines. They are mounted on the free end of the crankshaft and in some engines on the free end of the camshaft.

There are two types of vibration dampers.



◄
An hydraulically affixed shaft flange.

The required oil pressure is generated using a hand pump.



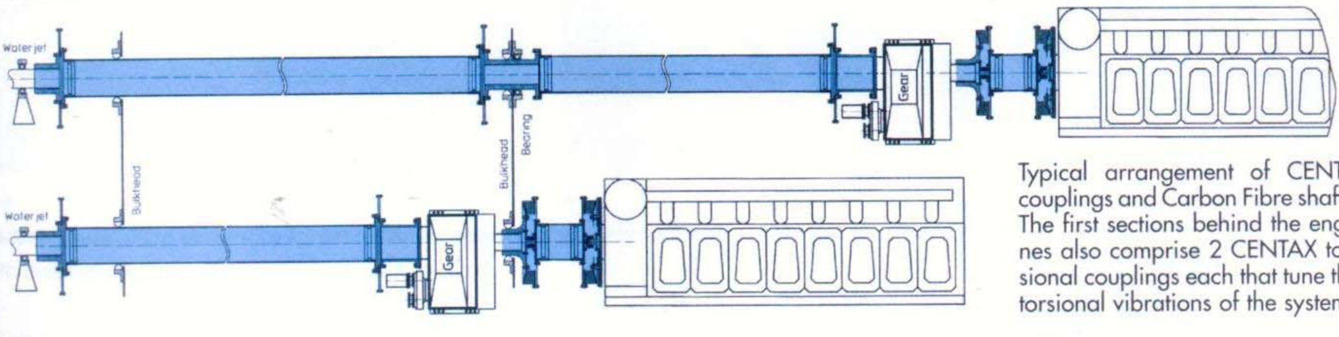
Carbon Fibre-shaft, combinded with flexible couplings

◄
A carbon-fibre flexible shaft combined on both sides with Centa flexible couplings, this is often used for catamarans and other high-speed craft.

The arrangement is very flexible; this is required for the moving hull sections between the bedplate and the stern tube.

▼
An example of the arrangement with flexible carbon-fibre propeller shafts of Centa shown here in blue.

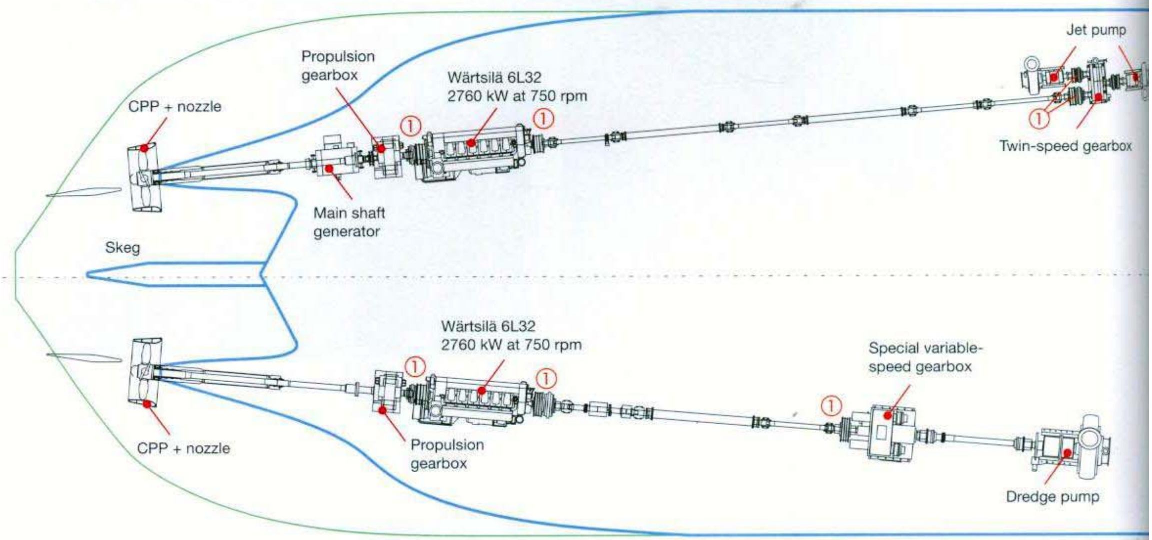
Between the diesel engines and the gear boxes are two flexible couplings which largely neutralise the torsional vibrations.



Typical arrangement of CENTA couplings and Carbon Fibre shafts. The first sections behind the engines also comprise 2 CENTAX torsional couplings each that tune the torsional vibrations of the system.

20.11 Examples of complete systems with diesel engines, reduction gearing, shafting, couplings and vibration dampers

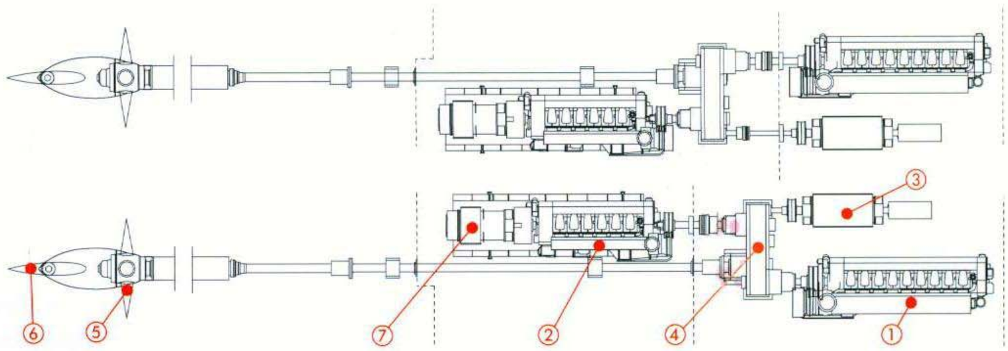
For additional information Chapter 17, Vibrations and balancing and Chapter 19, Ship propulsion.



The shafting on a dredger, design Wärtsilä.

Both main engines drive the controllable-pitch propellers in the nozzles via a flexible coupling, reduction gearing and propeller shaft. On the other side of the crankshaft, the port main engine drives two high-pressure water pumps for flushing the hold. On both sides of the lengthy shafting and on both output shafts of the reduction gearing, flexible couplings are installed. On the outgoing reduction gearing to the propeller, a shaft generator has been placed. The starboard main engine drives the dredging pump via reduction gearing with adjustable RPM with various flexible couplings built in.

1 *flexible couplings*



A propulsion system with two main engines and two gensets.

1 *main engine*
2 *genset*
3 *shaft generator*
4 *reduction gearing*
5 *controllable-pitch propeller*
6 *rudder*
7 *generator*

Switch I: Both main engines drive the controllable-pitch propellers via reduction gearing and generate electricity with the shaft generators. Normal speed.

Switch II: Both auxiliary engines generate electricity with their own generator and drive the controllable-pitch propellers at low speed. The main engines are stopped.

Switch III: The four engines drive both controllable-pitch propellers at maximum speed. Electricity may be generated with the shaft generators and/or with the auxiliary generators. More scenarios are of course optional.

Flexible couplings: All the engines are equipped with flexible couplings on the output shaft, as are the shaft generators and the auxiliary generators.

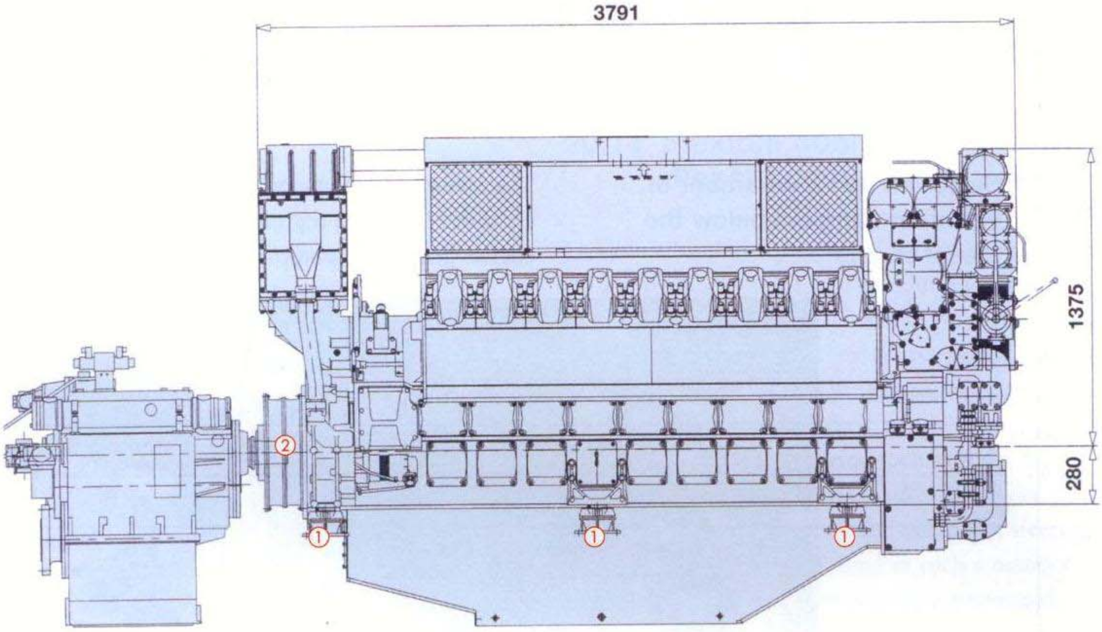
20.12 Vibration dampers below the engine frame

Many four-stroke high-speed and medium-speed diesel engines in categories I, II and III have, apart from the previously discussed flexible couplings to the propeller or the generator and the vibration damper on the blind side of the crankshaft and sometimes on the camshaft, vibration dampers below the engine block or frame.

This prevents transfer of engine vibrations to the bedplate and surroundings.

This system requires flexible connections for all lines, such as fuel, starting air, coolant and exhaust gases. During starting and stopping, the diesel engine moves on the flexible vibration dampers to such an extent that the lines would be torn off if no damping took place.

Especially in small high-speed four-stroke engines in categories I and II all the parts are installed in a flexible manner.

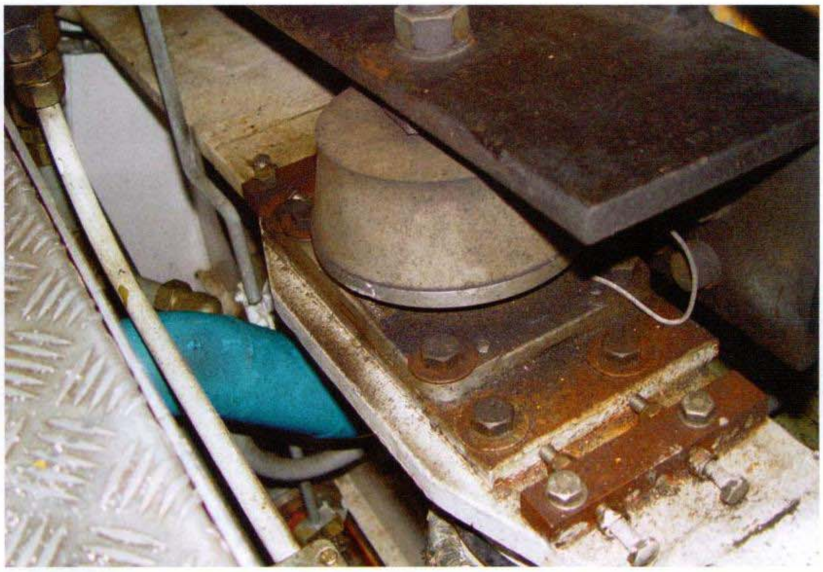


This four-stroke high-speed eighteen-cylinder V-engine with flexibly arranged reduction gearing is fitted with six vibration dampers below the engine block and a flexible coupling between the engine and the reduction gearing.

1 *vibration damper*
2 *flexible coupling*

Vibration dampers below a two-cylinder high-speed four-stroke diesel engine for yacht propulsion.

Left; the built-on reduction gearing with built-in reversing clutch. The lever for the reversing clutch is clearly visible.



A vibration damper below a propulsion engine.

Note the adjustable base below the vibration damper.

▶
Vibration dampers below the frame of a gas-engine generator set.

Both the gas engine and the generator are fixed rigidly to a mutual frame. A flexible coupling (not visible) has been placed between the engine and the generator.
1 *vibration damper*

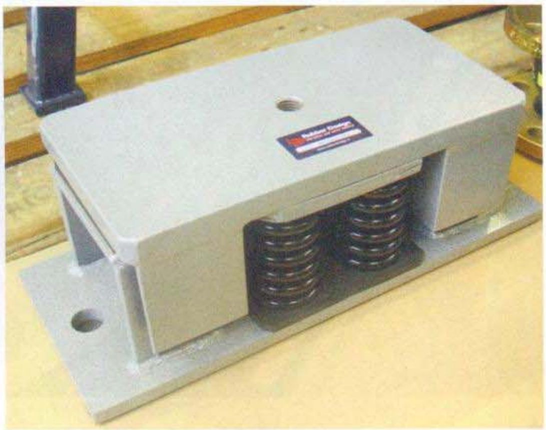
20.12.1 Selection and the number of vibration dampers below the diesel engine

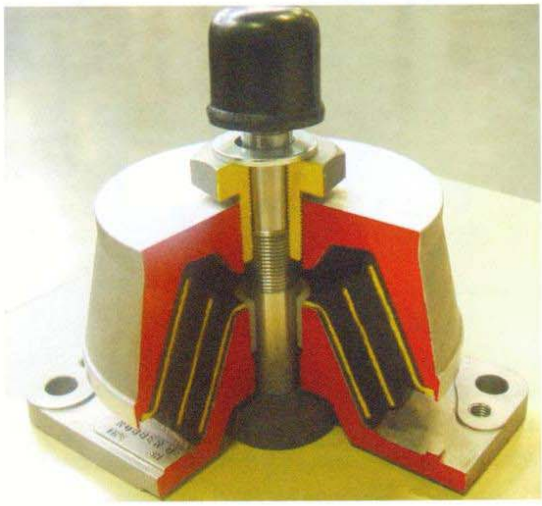
The manufacturers concerned have a very elaborate delivery program for each type of engine, genset and pump.

▶
Various types of Rubber Design vibration dampers.

▶
A large Rubber Design vibration damper for heavy engines.

The damping is done with heavy steel compression springs.





▲
A cut-away section of a Rubber Design vibration damper with in the centre the elastic rubber section.

Manufacturing diesel engines for engine category II, high-speed four-stroke engines with shaft power of 100 to 5000 kW.
Fuel: diesel oil (M.D.O.).
Here an overview of a production hall for the BR 2000 series at MTU in Friedrichshafen.

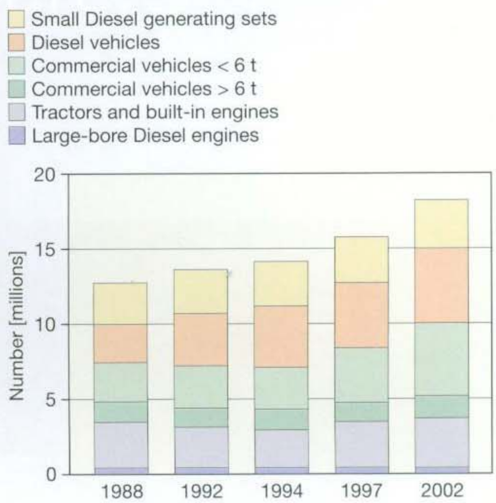
Mounting cylinder liners on a bank for a MTU – BR 8000 V-engine.

21.1 Introduction

From the time that internal combustion engines could be mass manufactured in the nineteenth century, an increasingly large number of engineering works started business. Next to their core business of manufacturing machines, they also set up engine manufacturing production lines. These companies were already manufacturing steam boilers, steam engines and, for example, cannons. Therefore the facilities for engine manufacture were present, such as a foundry and a forge; the roughly manufactured parts could be finished with simple planing machines, milling machines and/or lathes.
Most of the work was manual labour. These engine factories were often established near navigable waters, so the ships could berth in front of the premises for repairs.
The larger engines in categories II, III and IV were manufactured worldwide.
Very large engines in categories III and IV used to be and still are manufactured throughout the world.
Where ship propulsion is concerned, very large low-speed crosshead engines are normally manufactured at the same location as the shipyard. A famous example is Hyundai, the world's largest shipyard/ engine factory in South Korea.

Number of diesel engines manufactured on a yearly basis

The total number of diesel engines manufactured annually worldwide lies between 20 and 25 million (2006).
Obviously, the number of small diesel engines manufactured for, amongst others, agricultural machines, excavators, lorries and such, is the largest segment of this market, in excess of 90%.



▲
A chart showing the number of diesel engines manufactured in the course of a twenty five-year period.

The number of 'large' diesel engines, dark blue in this Bosch overview, represents approximately 300,000 pieces.

▶
Large and heavy two-stroke crosshead engines are usually manufactured at the yard where the ship is built.

Front left: a smaller MAN–B&W diesel engine with an engine frame cast in one piece. Behind it, a MAN–B&W diesel engine with a frame made of sheet steel.
In the foreground a crankshaft bed for a crosshead engine.

The figures lie below ten thousands for large diesel engines in Category II (1000 kW). For the largest engines in category II this amounts to at the most several thousands per annum.
In category III these figures are in the region of several thousand per annum and category IV, several hundreds per annum.
At present many more engines are being manufactured, in comparison to ten years ago when engine manufacturers had to shut down or downsize due to extreme overcapacity in the market.

Licence holders

Situation in 2006 – World-wide production, countries where engines are manufactured

Diesel-engine manufacturers in categories III and IV issue licences to enable others to build their engines. For instance, MAN–B&W has over 20 licence holders.

Ship propulsion with diesel engines	
South Korea	31%
Japan	26%
Europe	23%
China	12%
Other Asian countries	4%
USA and Canada	2%
Rest	2%

Source: Douglas – Westwood Ltd, Canterbury, England.

21.2 Engine categories

The divisions are based on actual running experience with diesel engines.

Characteristics

Fuel

Only medium-speed four-stroke and low-speed two-stroke diesel engines run on heavy fuel oil, H.F.O..

Operating principle

The majority of diesel engines work on the four-stroke cycle principle, and can be placed in the group of high-speed engines with over 960 revolutions per minute.
The minority are the larger four-stroke engines with a cylinder diameter above 200 mm. They fall in the category medium-speed engines and are suitable for running on heavy fuel oil.

▲ This illustration shows the long period of activity for MAN–B&W licensees.

The Japanese engine factory Mitsubishi was established in1926!

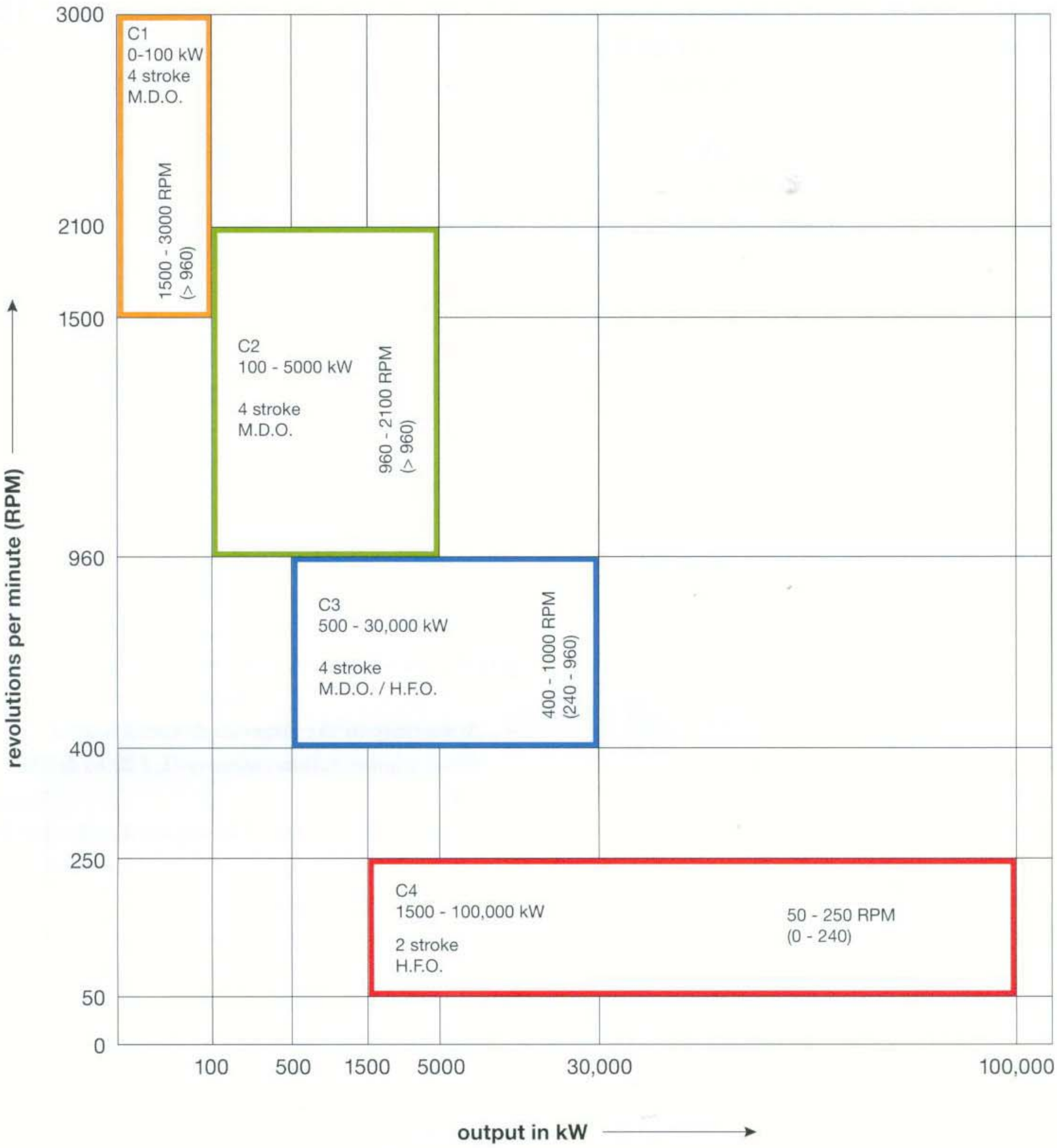
Capacity classification

A large number of small four-stroke diesel engines fall in the category I, up to 100 kW.
They are often used for propulsion of small ships, such a motorboats and yachts, drives for pumps, gensets, anchor winches and building equipment.

- A majority of the larger high-speed four-stroke diesel engines fall in category II, 100 to 5000 kW.
  They are widely used. The largest diesel engines in this category with a power output of several megawatts are commonly used in the propulsion of fast vessels, catamarans and yachts. The fuel is diesel oil: M.D.O..
- A smaller number of four-stroke engines fall in category III, medium-speed engines with engine speeds of approximately 960 revolutions per minute and usually operate on H.F.O..
- The largest diesel engines fall in category IV, low-speed two-stroke crosshead engines with a power output to almost 100,000 kW. They have a maximum engine speed of approximately 240 revs/min and operate on H.F.O..
- General comment:
  This classification allows a better insight into engine characteristics. Therefore, it is possible that in the overlapping area between two categories, there are other possibilities.

Examples

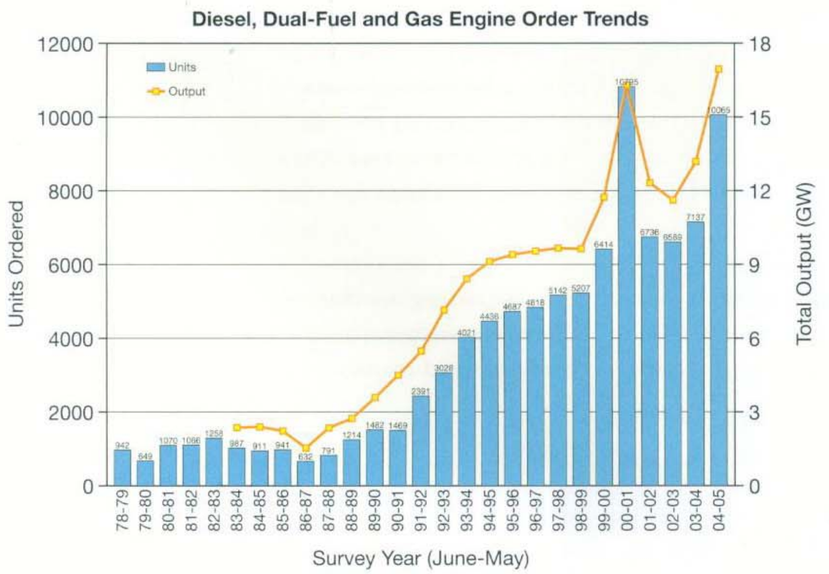
Some engine factories manufacture engines that can achieve 1000 to 1200 revolutions per minute and categorise them as medium-speed engines.



▶ The four engine categories.

Engine manufacturers

The two main suppliers MAN–B&W and Wärtsilä produce (defined in shaft power) the majority of engines in Category III and IV with Caterpillar as number 3 in the smaller capacities, category I and particularly II.



◀ A chart showing the number of diesel, dual-fuel and gas engines ordered between 1978 and 2005.

In this chart all engines from 500 kW shaft power upwards are listed, so categories II, III and IV.
The total number of units lies in the region of 10,000 per year. The average power output is a maximum of approximately 1600 kW (2000–2001).

DIESEL, DUAL-FUEL & GAS ENGINE ORDERS, June 2004 – May 2005																				
		Total Engine	Type of Generating Service (Units)			Fuel (Units)														
Output Range (MW)	Units Ordered	Ouput (MW)	Stand-by	Peak-ing	Contin-uous	Diesel Fuel	Heavy Fuel	Dual-Fuel	Nat. Gas	Western Europe	Eastern Europe & Russia	Middle East	Far East	Southeast Asia/ Australia	Central Asia	North Africa	Central W., E., & S. Africa	North America	Central America	South America
1.01 – 2.0	8662	11 465	6174	230	2258	7612	22	0	1028	1421	267	721	1800	870	874	14	55	2138	411	91
2.01 – 3.5	992	2320	616	12	363	751	25	1	214	182	54	45	93	33	33	0	34	485	20	12
3.51 – 5.0	86	362	9	3	74	40	6	0	40	21	8	6	19	7	4	5	0	9	7	0
5.01 – 7.5	136	796	9	7	120	29	33	4	70	14	5	8	36	13	18	0	26	15	1	0
7.51 – 10	137	1155	3	2	132	16	57	0	64	15	3	27	10	14	19	11	11	15	5	7
10.01 – 15	29	354	0	0	29	0	29	0	0	3	0	15	4	0	3	0	1	0	3	0
15.01 – 20	23	411	0	0	23	0	20	3	0	1	0	6	0	0	0	0	3	0	13	0
20.01 – 30	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
30.01 & above	1	43	0	0	1	0	1	0	0	0	0	0	0	0	0	0	0	0	1	0
Subtotals	10 065	16 906	6811	254	3000	8448	193	8	1416	1657	337	828	1962	937	951	30	130	2662	461	110
0.50 – 1.0	12 439	8307	9144	348	2947	11 711	6	6	716	2397	124	1177	2733	1126	1235	4	82	2923	393	245
Totals	22 504	25 213	15 955	602	5947	20 159	199	14	2132	4054	461	2005	4695	2063	2186	34	212	5585	854	355

▲

An overview from 'Diesel Publications' regarding the manufacture of engines between June 2004 and May 2005.

Here the minimum power output is 1000 kW, so these engines fall in category II.

Total number ordered 10,065, of which 8448 are diesel engines.

Most of the engines have a relatively small power output of 1 to 2 MW.

DIESEL, DUAL-FUEL & GAS ENGINE ORDERS, June 2005 – May 2006																				
		Total Engine	Type of Generating Service (Units)			Fuel (Units)														
Output Range (MW)	Units Ordered	Ouput (MW)	Stand-by	Peak-ing	Contin-uous	Diesel Fuel	Heavy Fuel	Dual-Fuel	Nat. Gas	Western Europe	Eastern Europe & Russia	Middle East	Far East	Southeast Asia/ Australia	Central Asia	North Africa	Central W., E., & S. Africa	North America	Central America	South America
0.50 – 1.0	17 614	11 840	11 315	1490	4809	16 608	13	12	981	3735	312	2378	2026	1341	2163	5	344	4466	570	274
1.01 – 2.0	11 257	15 403	6298	1280	3679	9765	285	0	1207	2430	180	1016	1610	1181	616	11	182	2827	1075	129
2.01 – 3.5	1778	4226	810	155	813	1269	209	1	298	433	60	150	77	91	29	1	35	659	220	23
3.51 – 5.0	147	574	4	1	142	43	67	3	34	35	9	3	13	24	6	1	5	3	47	1
5.01 – 7.5	129	758	4	10	115	35	15	4	75	22	6	10	51	15	10	4	3	1	5	3
7.51 – 10	151	1303	0	0	151	1	70	0	80	4	58	30	4	5	11	0	4	0	34	1
10.01 – 15	14	167	2	0	12	4	10	0	0	2	0	10	0	0	0	0	0	0	2	0
15.01 – 20	60	1067	1	0	59	13	43	4	0	13	0	9	0	2	1	0	8	0	12	15
20.01 – 30	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
30.01 & above	1	43	0	0	1	0	1	0	0	0	0	0	0	0	1	0	0	0	0	0
Totals	31 151	35 381	18 434	2936	9781	27 751	713	24	2675	6674	625	3608	3781	2659	2837	22	581	7956	1965	446

▲

An overview from 'Diesel Publications' regarding the manufacturing of engines between June 2005 and May 2006.

Here the minimum power output is 500 kW, so these engines fall in category II.

Total number ordered 31,151, of which 27,751 are diesel engines.

Most engines have a relatively small power output of 0.5 to 2 MW.

This was the first year that engines with a power output of 500 kW as opposed to 1000 kW were incorporated, which meant a significant increase in the number of engines ordered.

The number of engine manufacturers worldwide has dwindled over the last years for various reasons. Today, there are over forty diesel engine manufacturers worldwide; approximately half are active in the international market.

A number of reasons for reduction of engine manufacturer numbers:

- **Due to overcapacity in the past and heavy competition.**
- **Due to increasingly strict regulations for exhaust-gas emissions. This requires large investments to optimise the engine design which can meet present and future requirements.**
- **The customers have stricter requirements.** Important focal points are:
  - capital cost;
  - fuel and lubricating oil consumption;
  - weight and size in relation to capacity, the so-called power density;
  - worldwide fast service and availability of parts;
  - lifetime of parts.

Mass manufacture for shipping

When choosing propulsion installations, the engine type and manufacturer have usually already been established, as a different engine type will not normally fit the requirements and therefore increases costs. Manufacturing costs are kept to a minimum to compete worldwide among other shipyards.

Number of engine manufacturers worldwide

Category I – The small high-speed four-stroke diesel engines to 100 kW shaft power running on M.D.O.

The majority of manufacturers produce these engines. Several dozens for the world market and an even larger number for the local markets. Large engine manufacturers for categories I and II, such as Caterpillar and Cummins together produce hundreds of thousands to millions of engines annually.
These engines are often used in lorries, buses, earth moving equipment and cranes.

Some examples

Perkins Engine Company Limited

Perkins is an English manufacturer of diesel engines (owned by Caterpillar), predominantly for categories I and II. Diesel engines up to 100 kW shaft power and 5000 kW shaft power respectively.
A V-type engine version has a maximum shaft power of 1886 kW. These are, amongst others, used as emergency gensets in hospitals.

Ruggerini Diesel

Ruggerini Diesel is an Italian producer of very small diesel engines in category I, up to 100 kW. They are typically used for driving smaller gensets, pumps and other drive systems. Engine manufacturers German Hatz, Italian Lombardini and English Lister Petter also fall in this category.

Manufacturing Perkins diesel engines in category II.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	403C-07	67	72	0.254	3L		5.1		3600				15.3	ISO 14396
D	403C-11	77	81	0.377	3L		7		3400				21	ISO 14396
D	403C-15	84	90	0.499	3L		8.4		3000				25.1	ISO 14396
D	404C-15	77	81	0.377	4L		6.6		3000				26.5	ISO 14396
D	404C-22	84	100	0.554	4L		9.5		3000				38	ISO 14396
D	404C-22T	84	100	0.554	4L		11.4		3000				45.5	ISO 14396
D	804C-33	94	120	0.833	4L		11.8		2600		7.5		47	ISO 14396
D	804C-33T	94	120	0.833	4L		15		2600		9.5		60	ISO 14396
D	1103 Series	105	127	1.1	3L		18		2200		9.1		55	
D	1104 Series	105	127	1.1	4L		27		2500		11.6		106	
D	1106 Series	105	127	1.1	6L		34		2500		14.9		205	
D	1106-E60TA	100	127	1	6L		21.6		2500				129.5	
D	1306-E87T	117	136	1.45	6L		28		2200				168	
D	1306-E87TA	117	136	1.45	6L		41		2000				246	
D	4000 Series	160	190	3.8	6, 8L; 12, 16V		118.3		1800				1886	
D	6TG2AM	100	127	1	6L		20.1		1800		12.9		121	BS 5514
D	6TWGM	100	127	1	6L		26.8		1800		20		161	ISO 3046-1
D	4.4GM	105	127	1.1	4L		13.5		1800		8.2		54	ISO 14396
D	4.4TGM	105	127	1.1	4L		17.5		1800		10.6		70	ISO 14396
D	4.4TWGM	105	127	1.1	4L		22.75		1800		13.9		91	ISO 14396
D	4.4TW2GM	105	127	1.1	4L		29.37		1800		17.8		117.5	ISO 14396
D	903-27	95	127	0.9	3L		12.4		2250				37.3	
D	M130C	100	127	1	6L		16		2600		11.8		96	BS AU141a
D	M135	100	127	1	6L		16		2600		11.8		96	BS AU141a
D	M185C	100	127	1	6L		23.3		2100		17.4		140	BS AU141a
D	M215C	100	127	1	6L		26.3		2500		16		158	BS AU141a
D	M225Ti	100	127	1	6L		27.6		2500		16		165.5	BS AU141a
D	M265Ti	100	127	1	6L		32.5		2500		17.2		195	BS AU141a
D	M300Ti	100	127	1	6L		36.8		2500		17.2		221	BS AU141a
D	Peregrine	117	119	1.3	6L		31.1		2300				186.5	

▲

The Perkins diesel-engine programme.

▶

Small pleasure yachts such as this usually have diesel engines with power outputs between 25 and 75 kW, category I.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	RY50	69	60	0.224	1L		3.5		3600		5.2		3.5	
D	RY70	78	66	0.315	1L		5		3600		5.3		5	
D	RY75	82	66	0.349	1L		5.5		3600		5.3		5.5	
D	RY103	82	76	0.401	1L		7.3		3600		6.2		7.3	
D	RY110	86	76	0.442	1L		8.1		3600		6.3		8.1	
D	RY125	87	85	0.505	1L		8.8		3600				8.8	
D	MD151	80	65	0.327	2L;2H		6		3600				12	
D	MD191	85	75	0.426	2L;2H		7		3600				14	
D	RD211	90	75	0.477	2L		8.5		3600		6.8		17	
D	RD290	95	88	0.624	2L		10.5		3000		6.9		21	
D	MD350	85	92	0.522	3L		9.3		3600				28	
D	SP420	95	88	0.623	3L		10.3		3000		6.7		30.9	

▲

The Ruggerini Diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	1 B 20	69	62	0.232	1L		3.5		3600		5	1.4	3.5	
D	1 B 30	80	69	0.347	1L		5.4		3600		5.2	2.1	5.4	
D	1 B 40	88	76	0.462	1L		7.7		3600		5.6	3.2	7.7	
D	1 B 50	93	76	0.517	1L		8.5		3600		5.5	3.4	8.5	
D	1 B 20 V	69	62	0.232	1L		3.5		3600		5	1.4	3.5	
D	1 B 30 V	80	69	0.347	1L		5.4		3600		5.2	2.1	5.4	
D	1 B 40 V	88	76	0.462	1L		7.7		3600		5.6	3.2	7.7	
D	1D41	90	65	0.413	1L		6.4		3600		5.2	2.6	6.4	
D	1D50	97	70	0.517	1L		7.9		3600		5.1	3.3	7.9	
D	1D81	100	85	0.667	1L		10.3		3000		6.2	5	10.3	
D	1D90	104	85	0.722	1L		11.2		3000		6.2	5.8	11.2	
D	1D41 C	90	65	0.413	1L		5.2		3600		4.2	2.4	5.2	
D	1D81 C	100	85	0.667	1L		9.6		3000		5.8	4.9	9.6	
D	1D90 V	104	85	0.722	1H		11.2		3000		6.2	5.8	11.2	
D	2 G 40	92	75	0.499	2L		8.5		3600		5.7	7	17	
D	L 41	102	105	0.858	2, 3, 4L		13.5		3000		6.3	13.5	54.2	
D	M 41	102	105	0.858	2, 3, 4L		14.4		3000		6.7	14.8	57.5	
D	W 35	70	90	0.346	2, 3, 4L		5.9		3000		6.8	5.5	23.6	
D	4 W 35 T	100	85	0.667	4L		7.7		3000		4.6	13	30.6	

▲

The engine manufacturer Hatz GmbH & Co. KG. diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	15LD225	69	60	0.224	1L		3.5		3600		5.2		3.5	
D	15LD315	78	66	0.315	1L		5		3600		5.3		5	
D	15LD350	82	66	0.349	1L		5.5		3600		5.3		5.5	
D	15LD400	82	76	0.401	1L		7.3		3600		6.2		7.3	
D	15LD440	86	76	0.442	1L		8.1		3600		6.3		8.1	
D	15LD500	87	85	0.505	1L		8.8		3600				8.8	
D	25LD330-2	80	65	0.327	2L; 2H		6		3600				12	
D	25LD425-2	85	75	0.426	2L; 2H		7		3600				14	
D	12LD477.2	90	75	0.477	2L		8.5		3600		6.8		17	
D	9LD626.2	95	88	0.624	2L		10.5		3000		6.9		21	
D	11LD522-3	85	92	0.522	3L		9.3		3600				28	
D	11LD626.3	95	88	0.623	3L		10.3		3000		6.7		30.9	
D	LDW502	72	62	0.253	2L		4.9		3600		6.6		9.8	
D	LDW702	75	77.6	0.229	2L		6.3		3600		6.4		12.5	
D	LDW1003	75	77.6	0.343	3L		6.5		3600		6.6		19.5	
D	LDW1404	75	77.6	0.343	4L		6.5		3600		6.7		26	
D	LDW1603	88	90.4	0.55	3L		10		3000				30	
D	LDW2204	88	90.4	0.55	4L		9.5		3000		7		38	
D	LDW2204T	88	90.4	0.55	4L		12		3000				48	

▲ The Lombardini S.R.L. diesel-engine programme.

▼ The Lister Petter Ltd. diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	AC1	76	67	0.3	1L		5.3		3600		5.3	2.1	5.3	ISO 3046-1
D	AD1	80	73	0.37	1L		5.8		3000		5.8	2.3	5.8	ISO 3046-1
D	LT1	83	76	0.41	1L		6.15		3000		6	2	6	ISO 3046-1
D	LV1	86	83	0.48	1L		7.4		3000		6.2	2	7	ISO 3046-1
D	TR1	98	102	0.77	1L		8.6		2500		5.9	5	9	ISO 3046-1
D	TR2	98	102	0.77	2L		8.6		2500		5.9	11	18	ISO 3046-1
D	TR3	98	102	0.77	3L		8.6		2500		5.9	16	26	ISO 3046-1
D	LPW2	86	80	0.47	2L		7.7		3600		6.4	6	15.4	ISO 3046-1
D	LPW3	86	80	0.47	3L		7.7		3600		6.4	9	23.1	ISO 3046-1
D	LPW4	86	80	0.47	4L		7.7		3600		6.4	13	30.8	ISO 3046-1
D	LPWT4	86	80	0.47	4L		10.3		3000		8.9	19	40.2	ISO 3046-1
D	LPWS2 - Tier IV	86	80	0.47	2L		7.4		3000		6.4	6	14.7	ISO 3046-1
D	LPWS3 - Tier IV	86	80	0.47	3L		7.4		3000		6.4	9	22.1	ISO 3046-1
D	LPWS4 - Tier IV	86	80	0.47	4L		7.4		3000		6.4	13	29.5	ISO 3046-1
D	LPWST4 - Tier IV	86	80	0.47	4L		10.3		3000		8.9	19	40.2	ISO 3046-1
D	DWS4	94	120	0.83	4L		12.1		2500		7	29	45	ISO 3046-1
D	OMEGA 100	126	130	1.621	6L		27		2200		9.1	61	162	ISO 3046-1
D	OMEGA 200	126	130	1.621	6L		44.3		2200		14.9	101	266	ISO 3046-1

Category II – Larger high-speed four-stroke diesel engines from 100 to 5000 kW shaft power running on M.D.O.

There are at present a few dozen manufacturers active in the smaller capacities market. For engines over 1000 kW, the number of manufacturers reduces significantly and for over 3000 kW there are only a few manufacturers serving this market.

Caterpillar Inc. Engine Products Division

One of the world's largest engine manufacturers for categories I, II and III is the American company Caterpillar. Since their takeover of the German engine manufacturer MaK in Kiel, they have also incorporated medium-speed four-stroke diesel engines running on heavy oil, Category III, in their programme. The production of engines in category II is huge; hundreds of thousands of 'yellow Cats' find their way to the customers each year. There are 49 factories active in the USA and 58 in the rest of the world. An impressive number.

MTU Friedrichshafen GmbH

The German manufacturer of high-speed diesel engines running on M.D.O. is MTU in Friedrichshafen at the Boden Sea. It produces in categories I and II, highly loaded diesel engines with, in the larger engines, a mean effective pressure of 27 to 32.6 bar. This is very high. These large V-engines with a shaft power of 8200 kW (series 8000), a mean piston speed of 12.1 m/sec. and a mean effective pressure of 27 bar have a high load parameter of 326 bar/m/sec. This is the highest of all engines in category II.

Volvo Penta

This Swedish engine factory manufactures an extensive number of engine types in the categories I and II. The V-16 engines with 4.1 litre stroke volume have the largest power output of 1690 kW and a load parameter of 10.8 × 17.2 = 186.2 bar/m/sec.

Rolls-Royce

The English producer Rolls-Royce manufactures a small number of engine types in category II with a high power output. A maximum power output of 8000 kW and a load parameter of 10 × 24.9 = 249 bar/m/sec is respectable. They are built near Bergen, in Norway.

Pistons for Detroit Diesel category II engines, waiting to be assembled. Notice the two piston parts. The piston-ring section (1) and the section for the absorption of the lateral forces (2).

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	3056	100	127	1	6L	16	26	2100	2600	7.2	13.4	93	153	
D	3126	110	127	1.2	6L	44	52	2800	2800	15.5	18.6	261	313	
D	C7	127	110	1.21	6L	31	46	2400	2600	12.9	19.1	187	276	
D	C7 ACERT	127	110	1.21	6L	52	57	2800	2800	18.5	20.1	313	339	
D	C9 ACERT	149	112	1.47	6L	63	71	2500	2500	20.5	23.1	375	423	
D	C12	130	150	2	6L	42	75	1800	2300	14.1	19.5	254	448	
D	C12 ACERT	130	150	2	6L	82	88	2300	2300	21.4	22.9	492	526	
D	3406C	137	165	2.4	6L	31	72	1350	2100	8.5	16.9	187	433	
D	3406E	137	165	2.4	6L	56	100	1800	2300	15.3	21.3	336	597	
D	C15 ACERT	137	165	2.4	6L	100	106	2300	2300	21.3	22.7	597	636	
D	C18	145	183	3.02	6L	57	125	1800	2300	12.5	21.5	339	747	
D	3408C	137	152	2.25	8V	38	50	1800	2100	11.1	12.8	300	403	
D	3412C	137	152	2.25	12V	31	62	1800	2100	9.3	15.8	375	746	
D	3412D	145	162	2.68	12V	34	54	1800	2100	8.4	13.1	404	651	
D	3412E	137	152	2.25	12V	26	87	1200	2300	10.8	20.2	317	1044	
D	C30	145	152	2.5	12V	93	96	2300	2300	19.5	20.1	1119	1156	
D	C32	145	162	2.68	12V	68	103	2100	2300	14.6	20	820	1232	
D	C32 ACERT	145	162	2.68	12V	112	112	2300	2300	21.8	21.8	1343	1343	
D	3508	170	190	4.3	8V	66	107	1200	1800	12.3	16.6	526	857	
D	3508B	170	190	4.3	8V	72	140	1200	1925	12.3	20.2	578	1118	
D	3508C ACERT*	170	190	4.3	8V	72	103	1200	1600	16.2	19.5	578	820	
D	3512	170	190	4.3	12V	75	109	1200	1800	12.3	20.3	900	1305	
D	3512B	170	190	4.3	12V	68	140	1200	1925	13.8	21.2	820	1678	
D	3512B	170	215	4.88	12V	68	140	1200	1925	12.7	19.2	820	1678	
D	3512C	170	215	4.88	12V	147	158	1800	1800	20.1	21.6	1765	1895	
D	3516	170	190	4.3	16V	75	103	1200	1800	12.3	18.1	1195	1640	
D	3516B	170	190	4.3	16V	77	140	1200	1925	14.4	20.2	1230	2237	
D	3516B	170	215	4.88	16V	77	140	1200	1925	16.2	19.2	1230	2237	
D	3516C	170	215	4.88	16V	147	158	1800	1800	20.1	21.6	2350	2525	
D	3606/C280-6	280	300	18.5	6L	387	454	900	1000	20	22.8	2320	2722	
D	3608/C280-8	280	300	18.5	8L	386	454	900	1000	20	22.8	3084	3634	
D	3612/C280-12	280	300	18.5	12V	288	338	900	1000	20	22.8	3460	4060	
D	3616/C280-16	280	300	18.5	16V	288	339	900	1000	20	22.8	4600	5420	
D	3618	280	300	20.8	16V	450	450	1050	1050	24.7	24.7	7200	7200	

▲
The Caterpillar Inc. Engine Products Division diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	700	94	100	0.69	3, 4, 6L	10	37	1500	3800		16.8	30	235	
D	700	94	107	0.74	4L	12	17	2300	2600		10.4	47	67	
D	SUN	105	115	1	2, 3, 4, 6L	8	17	1500	2300		10.9	19	125	
D	900	102	130	1.6	4, 6L	19	22		2200		11.4	75	130	
D	900	106	136	1.2	6L	36	40		2200		14.5	220	240	
D	457	128	155	1.99	6L	40	56		1800		17.7	242	335	
D	460	128	166	2.13	6L	37	60		1800		18.8	220	360	
D	500	130	150	1.99	6, 8V	43	60	1800	2000		16	260	480	
D	S 40E	117	119	1.26	6L	21	31		2200		16.3	130	186	
D	S 60	130	160	2.12	6L	37	62	1500	2300		15.2	224	373	
D	S 60	133	168	2.33	6L	37	103	1500	2300		23.1	224	615	
D	1800	128	166	2.14	6H	52	65		1800		17.2	315	390	
D	2000	130	150	1.99	12, 16, 18V	38	93	1500	2350		23.6	452	1492	
D	2000	135	156	2.23	8, 10, 12, 16V	90	112	2250	2450		24.6	720	1790	
D	396	165	185	3.96	8, 12, 16V	90	134	1500	2000		20.4	725	2150	
D	4000	165	190	4.06	8, 12, 16V	87	172	1500	2100		23.9	700	2760	
D	4000	165	210	4.49	20V	110	150	1500	1860		22.3	2200	3010	
D	4000	170	210	4.77	12, 16, 20V	99	140		1800		19.6	1193	2800	
D	595	190	210	5.95	12, 16V		270	1750	1800		30.2	3240	4320	
D	956	230	230	9.56	16, 20V	220	390		1500		32.6	3520	6250	
D	1163	230	280	11.63	12, 16, 20V		370	1200	1300		29.4	4440	7400	
D	8000	265	315	17.37	20V	410	455		1150		27	8200	9100	

▲
The MTU Friedrichshafen GmbH diesel engine-programme.

▼
The 'Seeadler' of the German coast guard with MTU diesel engines, category II.

▶
The diesel engine forms an ideal propulsion engine for a variety of yachts. It is reliable, economic, and as it runs on diesel oil, the fire risk is minimal.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	D1-13	67	72	0.25	2L	4.5	4.5	3200	3200			9	9	ISO 8665
D	D1-20	67	72	0.25	3L	4.7	4.7	3200	3200			14	14	ISO 8665
D	D1-30	77	81	0.4	3L	7	7	3200	3200			21	21	ISO 8665
D	D2-40	77	81	0.4	4L	7.25	7.25	3200	3200			29	29	ISO 8665
D	D2-55	84	100	0.5	4L	10.25	10.25	3000	3000			41	41	ISO 8665
D	D2-75	84	100	0.5	4L	13.75	13.75	3000	3000			55	55	ISO 8665
D	D3-110	81	93.2	0.5	5L	16.2	16.2	3000	3000			81	81	ISO 8665
D	D3-130	81	93.2	0.5	5L	19.2	19.2	4000	4000			96	96	ISO 8665
D	D3-160	81	93.2	0.5	5L	24	24	4000	4000			120	120	ISO 8665
D	D3-190	81	93.2	0.5	5L	28	28	4000	4000			140	140	ISO 8665
D	D4-180	103	110	0.9	4L	32	32	2800	2800			128	128	ISO 8665
D	D4-225	103	110	0.9	4L	41.3	41.3	3500	3500			165	165	ISO 8665
D	D4-260	103	110	0.9	4L	47.75	47.75	3500	3500			191	191	ISO 8665
D	D5A T	108	130	1.2	4L	18	23.75	1900	2300			72	95	ISO 3046
D	D5A TA	108	130	1.2	4L	22.25	29.5	1900	2300			89	118	ISO 3046
D	D6-280	103	110	0.9	6L	38	38	3500	3500			228	228	ISO 8665
D	D6-310	103	110	0.9	6L	38	38	3500	3500			228	228	ISO 8665
D	D6-350	103	110	0.9	6L	42.83	42.83	3500	3500			257	257	ISO 8665
D	D6-370	103	110	0.9	6L	45.3	45.3	3500	3500			272	272	ISO 8665
D	D7A T	108	130	1.2	6L	18	21.5	1900	2300			108	129	ISO 3046
D	D7A TA	108	130	1.2	6L	21.67	29	1900	2300			130	174	ISO 3046
D	D7C TA	108	130	1.2	6L	24.33	32.5	1900	2300			146	195	ISO 3046
D	D9-MH	120	138	1.6	6L	36.83	52.2	1800	2200			221	313	ISO 3046
D	D9-425	120	138	1.6	6L	52.2	52.2	2200	2200			313	313	ISO 3046
D	D9-500	120	138	1.6	6L	61.3	61.3	2600	2600			368	368	ISO 8665/3046
D	D9-575	120	138	1.6	6L	70.5	70.5	2500	2500			423	423	ISO 8665/3046
D	D12-300	131	150	2	6L	36	36	1800	1800			216	216	ISO 3046
D	D12-350	131	150	2	6L	42.67	42.67	1800	1800			256	256	ISO 3046
D	D12-400	131	150	2	6L	49	49	1800	1800			294	294	ISO 3046
D	D12-450	131	150	2	6L	55.17	55.17	1800	1800			331	331	ISO 3046
D	D12-500	131	150	2	6L	61.17	61.17	1800	1800			367	367	ISO 3046
D	D12-550	131	150	2	6L	67.5	67.5	1900	1900			405	405	ISO 3046
D	D12-615	131	150	2	6L	75.33	75.33	2100	2100			452	452	ISO 3046
D	D12-650	131	150	2	6L	79.67	79.67	2300	2300			478	478	ISO 3046
D	D12-675	131	150	2	6L	82.67	82.67	2300	2300			496	496	ISO 8665/3046
D	D12-715	131	150	2	6L	87.67	87.67	2300	2300			526	526	ISO 8665/3046
D	D12-800	131	150	2	6L	95	95	2300	2300			570	570	ISO 8665/3046
D	D16-MH	144	165	2.6	6L	60.5	91.8	1800	1900			363	551	ISO 3046
D	D25A MS	170	180	4.1	6L	73.33	80.83	1600	1650			440	485	ISO 3046
D	D25A MT	170	180	4.1	6L	78.33	86.66	1600	1650			470	520	ISO 3046
D	D30A MS	170	220	5	6L	74.17	81.67	1350	1400			445	490	ISO 3046
D	D30A MT	170	220	5	6L	80	88.33	1350	1400			480	530	ISO 3046
D	D34A MS	150	160	2.8	V-12	52.83	58.42	1940	2000			634	701	ISO 3046
D	D34A MT	150	160	2.8	V-12	58.42	64.66	1940	2000			701	776	ISO 3046

▲
The Volvo Penta diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	D49A MS	170	180	4.1	V-12	73.33	80.83	1600	1650			880	970	ISO 3046
D	D49A MT	170	180	4.1	V-12	78.33	88.67	1600	1800			940	1040	ISO 3046
D	D65A MS	170	180	4.1	V-16	73.13	80.63	1600	1650			1170	1290	ISO 3046
D	D65A MT	170	180	4.1	V-16	78.13	86.25	1600	1800			1250	1380	ISO 3046
D	D5A T	108	130	1.2	4L	19.3	20.3	1500	1800			77	81	ISO 8528/8665
D	D5A TA	108	130	1.2	4L	23	25	1500	1800			92	100	ISO 8528/8665
D	D7A T	108	130	1.2	6L	19.3	20.3	1500	1800			116	122	ISO 8528/8665
D	D7A TA	108	130	1.2	6L	23.2	24.7	1500	1800			139	148	ISO 8528/8665
D	D9-MG	120	138	1.6	6L	37.8	44.2	1500	1800			227	265	ISO 8528/8665
D	D12-AUX	131	150	2	6L	48.7	61.7	1500	1800			292	370	ISO 8528/8665
D	D16-MG	144	165	2.6	6L	75	83.3	1500	1800			450	500	ISO 8528/8665
D	D25A MS	170	180	4.1	6L	85.8	99.2	1500	1800			515	595	ISO 8528/8665
D	D25A MT	170	180	4.1	6L	90.8	105.8	1500	1800			545	635	ISO 8528/8665
D	D30A MS	170	220	5	6L	99.2		1500	1800			595		ISO 8528/8665
D	D30A MT	170	220	5	6L	106.7		1500	1800			640		ISO 8528/8665
D	D34A MS	150	160	2.8	V-12	56.6	63.4	1500	1800			679	761	ISO 8528/8665
D	D34A MT	150	160	2.8	V-12	59.1	69	1500	1800			709	828	ISO 8528/8665
D	D49A MS	170	180	4.1	V-12	92.5	99.2	1500	1800			1110	1190	ISO 8528/8665
D	D49A MT	170	180	4.1	V-12	93.3	105.8	1500	1800			1120	1270	ISO 8528/8665
D	D65A MS	170	180	4.1	V-16	92.5	99.4	1500	1800			1480	1590	ISO 8528/8665
D	D65A MT	170	180	4.1	V-16	93.8	105.6	1500	1800			1500	1690	ISO 8528/8665
D	TD420VE	101	126	1	4L	18.75	18.75	2500	2500			75	75	ISO 3046
D	TAD420VE	101	126	1	4L	23.25	25.75	2500	2500			93	103	ISO 3046
D	TD520VE	108	130	1.2	4L	20.25	20.25	1800	1800			81	81	ISO 3046
D	TAD520VE	108	130	1.2	4L	26.75	29.5	2300	2300			107	118	ISO 3046
D	TAD620VE	98	126	1	6L	23.33	25.83	2500	2500			140	155	ISO 3046
D	TAD650VE	101	126	1	6L	24.5	25.5	2300	2300			147	147	ISO 3046
D	TAD660VE	98	126	1	6L	24.5	24.5	2300	2300			147	147	ISO 3046
D	TD720VE	108	130	1.2	6L	20.33	20.33	1800	1800			122	122	ISO 3046
D	TAD720VE	108	130	1.2	6L	26.17	29	2300	2300			157	174	ISO 3046
D	TAD721VE	108	130	1.2	6L	29.33	32.5	2300	2300			176	195	ISO 3046
D	TAD722VE	108	130	1.2	6L	36.67	36.67	2300	2300			220	220	ISO 3046
D	TAD750VE	108	130	1.2	6L	28.33	33.33	2300	2300			170	200	ISO 3046
D	TAD760VE	108	130	1.2	6L	30.17	30.17	2300	2300			181	181	ISO 3046
D	TAD940VE	120	138	1.6	6L	31.67	31.67	2100	2100			190	190	ISO 3046
D	TAD941VE	120	138	1.6	6L	36.67	36.67	2100	2100			220	220	ISO 3046
D	TAD942VE	120	138	1.6	6L	41.67	41.67	1900	2100			250	250	ISO 3046
D	TAD943VE	120	138	1.6	6L	46.67	46.67	1900	1900			280	280	ISO 3046
D	TAD950VE	120	138	1.6	6L	33.33	33.33	2100	2100			200	200	ISO 3046
D	TAD951VE	120	138	1.6	6L	37.33	37.33	2100	2100			224	224	ISO 3046
D	TAD952VE	120	138	1.6	6L	42	42	2100	2100			252	252	ISO 3046
D	TWD1240VE	131	150	2	6L	49	58.33	1800	1800			294	350	ISO 3046
D	TWD1240VE	131	150	2	6L	42.67	51.67	2100	2100			256	310	ISO 3046
D	TAD1241VE	131	150	2	6L	57.17	57.17	1800	1800			343	343	ISO 3046

▲

The Volvo Penta diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	TAD1242VE	131	150	2	6L	63.83	63.83	1800	1800			383	383	ISO 3046
D	TAD1250VE	131	150	2	6L	43.17	43.17	1900	1900			259	259	ISO 3046
D	TAD1251VE	131	150	2	6L	48.17	48.17	1900	1900			289	289	ISO 3046
D	TAD1252VE	131	150	2	6L	52.17	52.17	1900	1900			313	313	ISO 3046
D	TAD1641VE	144	165	2.7	6L	70	70	1800	1800			420	420	ISO 3046
D	TAD1642VE	144	165	2.7	6L	82.33	82.33	1800	1800			494	494	ISO 3046
D	TD520GE	108	130	1.2	4L	21.25	21.25	1500	1500			85	85	ISO 8528/3046
D	TD520GE	108	130	1.2	4L	22.25	22.25	1800	1800			89	89	ISO 8528/3046
D	TAD520GE	108	130	1.2	4L	13.67	13.67	1500	1500			82	82	ISO 8528/3046
D	TAD520GE	108	130	1.2	4L	27.5	27.5	1800	1800			110	110	ISO 8528/3046
D	TD720GE	108	130	1.2	6L	21.33	21.33	1500	1500			128	128	ISO 8528/3046
D	TD720GE	108	130	1.2	6L	22.33	22.33	1800	1800			134	134	ISO 8528/3046
D	TAD720GE	108	130	1.2	6L	25.5	25.5	1500	1500			153	153	ISO 8528/3046
D	TAD720GE	108	130	1.2	6L	27.17	27.17	1800	1800			163	163	ISO 8528/3046
D	TAD721GE	108	130	1.2	6L	30.5	30.5	1500	1500			183	183	ISO 8528/3046
D	TAD721GE	108	130	1.2	6L	34	34	1800	1800			204	204	ISO 8528/3046
D	TAD722GE	108	130	1.2	6L	33.5	33.5	1500	1500			201	201	ISO 8528/3046
D	TAD722GE	108	130	1.2	6L	37.5	37.5	1800	1800			225	225	ISO 8528/3046
D	TAD940GE	120	138	1.6	6L	39.83	44.17	1500	1500			239	265	ISO 8528/3046
D	TAD940GE	120	138	1.6	6L	41.33	45.5	1800	1800			248	273	ISO 8528/3046
D	TAD941GE	120	138	1.6	6L	46.5	51.33	1500	1500			279	308	ISO 8528/3046
D	TAD941GE	120	138	1.6	6L	49.3	54.3	1800	1800			296	326	ISO 8528/3046
D	TAD1240GE	131	150	2	6L	41.17	51.83	1500	1500			283	311	ISO 8528/3046
D	TAD1240GE	131	150	2	6L	50	55	1800	1800			300	330	ISO 8528/3046
D	TAD1241GE	131	150	2	6L	53.83	59	1500	1500			323	354	ISO 8528/3046
D	TAD1241GE	131	150	2	6L	57.33	64.5	1800	1800			344	387	ISO 8528/3046
D	TAD1242GE	131	150	2	6L	58.67	64.5	1500	1500			352	387	ISO 8528/3046
D	TAD1242GE	131	150	2	6L	65.17	71.67	1800	1800			391	430	ISO 8528/3046
D	TAD1640GE	144	165	2.7	6L	65.5	72	1500	1500			393	432	ISO 8528/3046
D	TAD1640GE	144	165	2.7	6L	71.83	80	1800	1800			431	480	ISO 8528/3046
D	TAD1641GE	144	165	2.7	6L	72.17	79.33	1500	1500			433	476	ISO 8528/3046
D	TAD1641GE	144	165	2.7	6L	81.67	91.83	1800	1800			490	551	ISO 8528/3046
D	TAD1642GE	144	165	2.7	6L	80.83	89.33	1500	1500			485	536	ISO 8528/3046
D	TAD1642GE	144	165	2.7	6L	88.67	97.5	1800	1800			532	585	ISO 8528/3046

▲

The Volvo Penta diesel-engine programme.

▶

Extremely large luxurious yachts generally have diesel engines of category II running on diesel oil used for propulsion. Their power output lies between 1000 and 3000 kW. Additionally, a genset is required.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	B32: 40A	320	400		6, 8, 9L		500		750		24.9	2880	4320	
D	B32: 40A	320	400		12V 16V		500		750		24.9	6000	8000	
D	B32: 40P	320	400		6, 8, 9L		500		750		24.9	3000	4500	
D	B32: 40P	320	400		12V 16V		500		750		24.9	6000	8000	
D	C25: 33A	250	330		6, 8, 9L		300		750		24.7	1400	2250	
D	C25: 33A	250	330		6, 8, 9L		300		1000		24.2	1800	2700	
D	C25: 33P	250	330		6, 8, 9L		300		750		24.7	1440	2250	
D	C25: 33P	250	330		6, 8, 9L		300		1000		24.2	1800	2700	
D	KRG	250	300		6, 8, 9L		202		750		22	1165	1820	
D	KRGB	250	300		6, 8, 9L		222		900		20	1325	1990	
D	KRM	250	300		6, 8, 9L		202		750		22	1215	1820	
D	KRMB	250	300		6, 8, 9L		222		900		20	1335	2005	

▲

The Rolls-Royce diesel-engine programme.

Category III – The large medium-speed four-stroke diesel engines of 500 to 30,000 kW shaft power running on H.F.O.

Here the number of engine manufacturers is limited to a maximum of ten. Roughly fifty percent are active worldwide.

Caterpillar Inc. (MaK)

The American engine manufacturer Caterpillar also produces diesel engines in category III since their take over of the German engine manufacturer MaK. An engine series with four cylinder diameters can be delivered by this supplier of mainly medium-speed diesel engines running on H.F.O. The largest engine, the sixteen-cylinder M 43 V-engine with 16,000 kW has a load parameter of 10.45 × 27.1 = 283.2 bar/m/sec. This is reasonably high.

Daihatsu Diesel Co., Ltd.

The Japanese engine manufacturer Daihatsu builds engines in categories II and III. The largest diesel engine, the sixteen-cylinder V-engine DK–32 has a shaft power of 6325 kW and a moderate load parameter of 9.2 x 22.5 = 207 bar/m/sec.

MAN–B&W Diesel AG

The world's largest engine manufacturer for category IV engines is unquestionably the German-Danish combination MAN–B&W, recently MAN–Diesel AG; it is also one of the principle manufacturers for category III engines. Many category III engines are manufactured in Augsburg (Germany), Holeby (Denmark) and Frederikshavn (Denmark). Larger numbers are manufactured under license in Asia. The load parameter of a popular diesel engine for use in cruise ships, the V48/60B, a twelve cylinder V-engine with a shaft power of 21,600 kW is 10.3 × 24.8 = 255 bar/m/sec. The L58/64 in-line engine, with the largest cylinder diameter, has a cylinder power output of 1310 kW.

S.E.M.T. Pielstick

The French engine factory builds four-stroke trunk-piston engines in categories II and III. The slightly lower capacity of the M.D.O.-type is more often used in naval ships and the largest engines running on H.F.O. are used in diesel power plants. In 2006 MAN–Diesel AG acquired full ownership of this engine manufacturer.

Wärtsilä Corporation

The Finnish engine manufacturer Wärtsilä is the market leader in category III, the medium-speed four-stroke engines running on H.F.O. as well as Dual Fuel engines and Spark Gas engines (Otto-engines with a large cylinder diameter). After taking over numerous European engine factories, a large number of engines with a maximum cylinder bore of 64 cm were built in Finland, Italy, France, Korea and China. The largest four-stroke diesel engine in the world has a cylinder power output of 2150 kW (!). A desired diesel engine, the latest 46 F, has a load parameter of 11.6 × 25.9 = 300.4 bar/m/sec. This is a high value for a medium-speed four-stroke engine running on H.F.O..

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	MaK M 20 C	200	300	9.5	6, 8, 9L	170	190	900	1000	24.1	24.2	1020	1710	
D	MaK M 25	255	400	20.5	6, 8, 9L	290	330	720	750	23.5	25.8	1800	2970	
D	MaK M 32 C	320	480	38.7	6, 8, 9L	480	500		600	24.9	25.9	2880	4500	
D	MaK M 32 C	320	420	33.8	12, 16V	480	500	720	750		23.7	5760	8000	
D	MaK M 43 C	430	610	88.5	6, 7, 8, 9L	900	1000	500	514	23.7	27.1	5400	9000	
D	MaK M 43 C	430	610	88.5	12, 16V	900	1000	500	514	23.7	27.1	10800	16000	

▲

The Caterpillar Inc. (MaK) diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	DC-17	170	270	6.13	5, 6L		101.7		1000		22.1	480	610	
D	DC-32	320	400	32.17	6, 8L		480		750		23.3	2905	3844	
D	DK-20	200	300	9.42	3, 5, 6, 8L		160		900		22.6	480	1280	
D	DK-26	260	380	20.17	5, 6L		267		750		22	1280	1600	
D	DK-28	280	390	24.01	6, 8L, 16V		329		750		22.1	1900	5000	
D	DK-32B	320	360	28.95	12V		368		750		21.2		4413	
D	DK-32C	320	390	31.37	6, 8L		395		750		20.3	2445	3162	
D	DK-36	360	480	48.86	6, 8L		551		600		22.6	3310	4413	
D	DK-36	360	460	46.8	12V		527		600		22.5		6325	
D	DL-16A	165	210	4.49	6L		73.5		1200		16.4		440	
D	M2	120	150	1.7	6L		27.5		1800		10.8	73	165	
D	M3	140	160	2.46	6L		44		1800		11.9	198	264	
D	M5	145	160	2.64	6L		51.3		1800		13	220	308	

▲

The Daihatsu Diesel Co. Ltd diesel-engine programme.

▶

A feeder container ship, usually driven by a medium-speed four-stroke diesel engine running on heavy oil, category III.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	L58/64	580	640	169	6-9L	1310	1400	400	428		23.2	7860	12600	
D	L+V48/60B	480	600	108.5	6-9L; 12-18V		1200	500	514	25.8	26.5	7200	21600	
D	L48/60	480	600	108.5	9L		1050	500	514	22.6	23.2		9450	
D	L40/54	400	540	67.8	6-9L	700	720	500	550	23.1	24.8	4200	6480	
D	V40/50	400	500	62.8	12-16, 18, 20V		750		600		23.9	9000	15000	
DF	L+V32/40DF	320	400	32.2	6-9L; 12-18V	385	400	720	750		19.9	2310	7200	
D	L+V 32/40	320	400	32.2	6-10L; 12-20V		500	720	750	24.9	25.9	3000	10000	
D	L+V28/32A	280	320	19.7	6-9L; 12, 16V		245		775		19.3	1470	3920	
D	L28/32H	280	320	19.7	5-9L;	210	220	720	750	17.8	17.9	1050	1980	
D	L27/38	270	380	21.7	5-9L		340		800		23.5	2040	3060	
D	L+V23/30A	225	300	11.9	6, 8L; 12V		160		900		17.9	960	1920	
D	L23/30A-E	225	300	11.9	6L		133		825		16.3		800	
D	L23/30H	225	300	11.9	5-8L	130	160	720	900		17.9	650	1280	
D	L21/31	210	310	10.7	5-9L		215	900	1000		24.1	1290	1935	
D	L16/24	160	240	4.8	5-9L	90	100	1000	1200	20.7	22.4	450	900	

▲
The MAN–B&W Diesel AG diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D/DF/SI	PA4-185 VG	185	210	5.64	6L; 18V	98	123	1200	1500		17.4	590	2215	
D	PA4-200 VG	200	210	6.6	8, 16V	132	166	1200	1500		20	1060	2650	
D	PA5	255	270	13.79	4L; 18V	210	220	900	1000		19.2	840	3960	
DF	PA5	255	270	13.79	4L; 18V	132	147	900	1000		12.8	528	2646	
D	PA6	280	290	17.86	6L; 20V	272	325	900	1050		21.8	1630	6500	
D	PA6 B	280	330	20.32	6L; 20V	325	405	900	1050		22.8	1950	8100	
D	PA6 B STC	280	330	20.32	12; 20V		405		1050		22.8	4860	8100	
D	PA6 CL	280	350	21.55	6L; 20V		295	720	750		21.9	1770	5880	
D	PA6 STC	280	290	17.86	12, 16V		324		1050		20.7	3880	5180	
D	PC2.6	400	460	57.81	6L; 18V		550	500	520		22	3300	9900	
D	PC2.6 B	400	500	62.83	12; 20V		750		600		23.9	9000	15000	
D	PC4.2	570	620	158.21	10; 18V		1215	400	429		21.5	12150	21870	
D	PC4.2 B	570	660	168.42	10; 20V	1300	1400	400	430		23.3	13000	28000	
D	PC40	570	750	191.38	5; 10L		1325		375		22.2	6625	13250	

▲
The S.E.M.T. Pielstick diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder) •	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	Wärtsilä 20	200	280	8.8	4, 6, 8, 9L		200		1000		28	720	1800	
D	Wärtsilä 26	260	320	17	6, 8, 9L		340		1000		25.5	1860	3060	
D	Wärtsilä 26	260	320		12, 16V		340		1000		25.5	3720	5440	
D	Wärtsilä 32	320	400	32.2	6, 7, 8, 9L		500		750		24.9	3000	4500	
D	Wärtsilä 32	320	400		12, 16, 18V		500		750		24.9	6000	9000	
DF	Wärtsilä Vasa 32LNGD	320	350	28.2	12, 16, 18V		375		750		21.3	4500	6750	
DF	Wärtsilä 32DF	320	350		6, 9R		350		750		19.9	2010	3150	
DF	Wärtsilä 32DF	320	350	28.2	12, 18V		350		750		19.9	4020	6300	
D	Wärtsilä 38	380	475	53.9	6, 8, 9L		725		600		26.9	4350	6525	
D	Wärtsilä 38	380	475		12, 16V		725		600		26.9	8700	11600	
D	Wärtsilä 46	460	580	96.4	6, 8, 9L		1155		514		28.8	5850	10395	
	Wärtsilä 46	460	580		12, 16, 18V		1155		514		28.8	11700	20790	
D	Wärtsilä 46F	460	580		6,7,8,9L		1250		600		25.9	7500	11250	
D	Wärtsilä 46F	460	580		12, 16V		1250		600		25.9	15000	20000	
DF	Wärtsilä 50DF	500	580		6,8,9L		950		514		19.5	5700	8550	
DF	Wärtsilä 50DF	500	580	113.9	12, 16, 18V		950		514		20	11400	17100	
D	Wärtsilä 64	640	900	290	6, 7, 8L		2150		333		27.2	12060	17200	

▲ The Wärtsilä Corporation diesel-engine programme.

Category IV – The large low-speed two-stroke crosshead engines of 1500 to 100,000 kW shaft power running on H.F.O.

These are only manufactured by three engine manufacturers: MAN–B&W (MAN–Diesel), Wärtsilä and Mitsubishi. The ratios are startling:

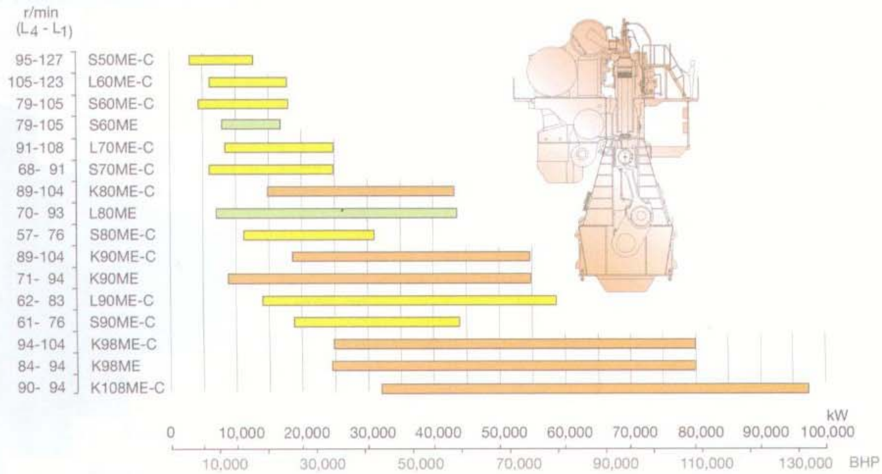
± 70% MAN–B&W;
± 22% Wärtsilä;
± 8% Mitsubishi.

Obviously, these percentages fluctuate somewhat around these averages.

MAN–B&W Diesel AG

The world's largest producer of two-stroke crosshead engines with a market share of ± 70%, they manufacture a large number of engine types with a cylinder diameter from 26 to 108 centimetres. Here the power output rises from 1080 to an incredible 97,300 kW, approximating the magical boundary of 100,000 kW per engine. The second largest engine, a fourteen-cylinder with 98 centimetre cylinder diameter was recently ordered for the latest generation of container ships with cargo capacity of ± 14,000 TEU.

▼ The MAN–B&W engine programme for the ME-series, so with electrically controlled fuel injection.



Large container ships, tankers and bulk carriers are usually propelled by the 'Cathedrals of the Oceans', large two-stroke crosshead engines, category IV.
Shown here the largest container ship in 2007, the 'Emma Maersk' when departing from the yard at Fünen, Denmark.

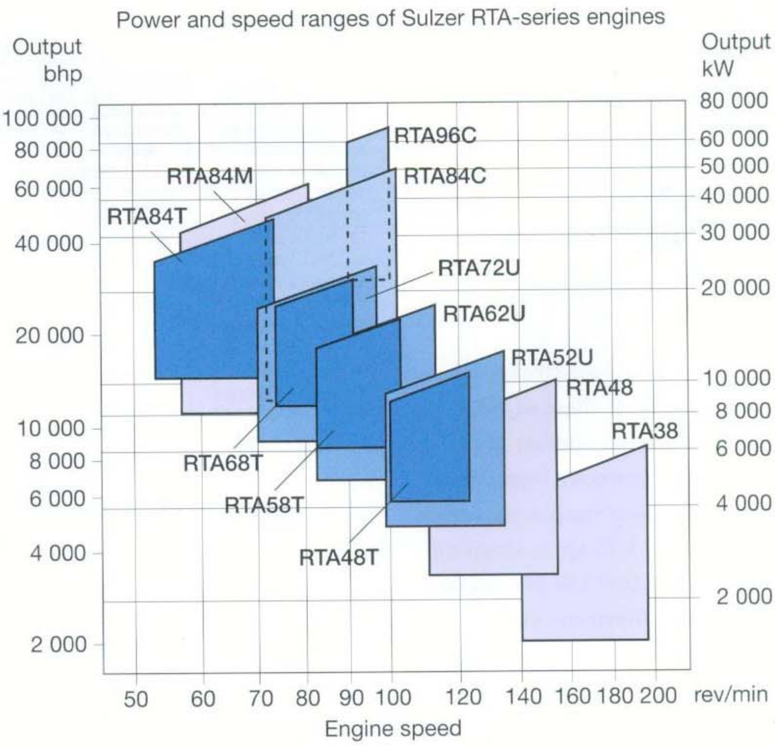
These 'Cathedrals of the Oceans' weigh approximately 2800 tons and consume approximately 300 tons of H.F.O. each day. The ME-series has electrically controlled fuel injection and the MC-series, a traditional fuel injection.

Mitsubishi Heavy Industries, Ltd.

The Japanese producer of two-stroke crosshead engines Mitsubishi builds engines with cylinder diameters of 33 to 85 centimetres. The worldwide share of 8 to 10% in the two-stroke crosshead engine market indicates that many of these engines are bought by the Japanese market for oil tankers, bulk carriers and fast container ships. At present, they are manufacturing two-stroke crosshead engines together with Wärtsilä.

Wärtsilä Corporation

The two-stroke crosshead engine's programme from Wärtsilä-Sulzer, RT Flex indicates the use of the common-rail fuel system while RTA indicates engines with the traditional injection systems. Wärtsilä manufactures the originally Sulzer two-stroke crosshead engines. Sulzer's original factory was situated in Winterthur, Switzerland. The large crosshead engines used to be transported by road and subsequently by ship to their final destination! Today, all engines are built by license holders throughout the world, often at the locations where the ships are built.



The Wärtsilä-Sulzer engine programme for the RTA-series, so with camshaft and traditional fuel injection.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	K108ME-C	1080	2660	2437	6-14L	5340	6950	90	94	14.6	18.2	32040	97300	
D	K98ME Mark 7	980	2660	2006	6-14L	4630	6230	90	97	15.4	19.2	27780	87220	
D	K98ME Mark 6	980	2660	2006	6-14L	4100	5720	84	94	14.6	18.2	24600	80080	
D	K98ME-C / MC-C Mark 7	980	2400	1810	6-14L	4510	6020	97	104	15.4	19.2	27060	84280	
D	K98ME-C / MC-C Mark 6	980	2400	1810	6-14L	4140	5710	94	104	14.6	18.2	24840	79940	
D	K98MC Mark 7	980	2660	2006	6-14L	4630	6230	90	97	15.4	19.2	27780	87220	
D	K98MC Mark 6	980	2660	2006	6-14L	4100	5720	84	94	14.6	18.2	24600	80080	
D	S90ME-C / MC-C Mark 8	900	3188	3188	6-9L	3570	5270	66	78	16	20	21420	47430	
D	S90ME-C / MC-C Mark 7	900	3188	3188	6-9L	3140	4890	61	76	15.2	19	18840	44010	
D	K90ME / MC	900	2550	1622	4-12L	2210	4570	71	94	11.5	18	8840	54840	
D	K90ME-C / MC-C	900	2300	1463	6-12L	3130	4570	89	104	14.4	18	18780	54840	
D	S80ME-C / MC-C Mark 8	800	3200	1608	6-8L	2830	4180	66	78	16	20	16980	33440	
D	S80ME-C / MC-C Mark 7	800	3200	1608	6-8L	1860	3880	57	76	12.2	19	11160	31040	
D	S80MC	800	3056	1563	4-12L	1740	3640	71	94	11.5	18	6960	54840	
D	K80ME-C / MC-C	800	2300	1156	6-12L	2470	3610	89	104	14.4	18	14820	43320	
DF	S70ME-C/ ME-GI Mark 8	700	2800	1078	4-8L	2210	3270	77	91	16	20	8840	26160	
DF	S70ME-C/ ME-GI Mark 7	700	2800	1078	4-8L	1490	3110	68	91	16	19	5960	24880	
D	S70MC-C Mark 8	700	2800	1078	4-8L	2210	3270	68	91	16	20	8840	26160	
D	S70MC-C Mark 7	700	2800	1078	4-8L	1490	3110	68	91	12.2	19	5960	24880	
D	S70MC	700	2674	1029	4-8L	1340	2830	68	91	11.5	18	5360	22480	
D	L70ME-C / MC-C Mark 8	700	2360	908	4-8L	2200	3270	91	108	16	20	8800	26160	
D	L70ME-C / MC-C Mark 7	700	2360	908	4-8L	2090	3110	91	108	15.2	19	8360	24880	
D	L70MC	700	2268	873	4-8L	1360	2830	81	108	11.5	18	5440	22640	
D	S65ME-C/ ME-GI	650	2730	906	5-8L	1960	2870	81	95	16	20	9800	22960	
DF	S60ME-C/ ME-GI Mark 8	600	2400	679	4-8L	1610	2380	89	105	16	20	6440	19040	
DF	S60ME-C/ ME-GI Mark 7	600	2400	679	4-8L	1090	2260	79	105	12.2	19	4360	18080	
D	S60MC-C Mark 8	600	2400	679	4-8L	1610	2380	79	105	16	20	6440	19040	
D	S60MC-C Mark 7	600	2400	679	4-8L	1090	2260	79	105	12.2	19	4360	18080	
D	S60MC	600	2292	648	4-8L	980	2040	79	105	11.5	18	3920	16320	
D	L60ME-C / MC-C Mark 8	600	2022	572	4-9L	1600	2340	105	123	16	20	6400	21060	
D	L60ME-C / MC-C Mark 7	600	2022	572	4-9L	1520	2230	105	123	15.2	19	6080	20070	
D	L60MC	600	1944	550	4-8L	920	1920	92	123	10.9	17	3680	15360	

▲

The MAN–Diesel AG diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	S50ME-C Mark 8	500	2000	393	4-9L	1130	1660	108	127	16	20	4520	14940	
D	S50ME-C Mark 7	500	2000	393	4-9L	760	1580	95	127	12.2	19	3040	14220	
D	S50MC-C Mark 8	500	2000	393	4-9L	1130	1660	95	127	16	20	4520	14940	
D	S50MC-C Mark 7	500	2000	393	4-9L	760	1580	95	127	12.2	19	3040	14220	
D	S50MC	500	1910	375	4-8L	680	1430	95	127	11.5	18	2720	11440	
D	L50MC	500	1620	318	4-8L	640	1330	111	148	10.9	17	2560	10640	
D	S46MC-C Mark 8	460	1932	321	4-8L	940	1380	110	129	16	20	3760	11040	
D	S46MC-C Mark 7	460	1932	321	4-8L	880	1310	108	129	15.2	19	3520	10480	
D	S42MC	420	1764	244	4-12L	730	1080	115	136	15.6	19.5	2920	12960	
D	L42MC	420	1360	188	4-12L	640	995	141	176	14.4	18	2560	11940	
D	S35MC	350	1400	135	4-12L	505	740	147	173	15.3	19.1	2020	8880	
D	L35MC	350	1050	101	4-12L	440	650	178	210	14.7	18.4	1760	7800	
D	S26 MC	260	980	52	4-12L	270	400	212	250	14.8	18.5	1080	4800	

▲
The MAN–Diesel AG diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	UEC33LSII	330	1050	90	4-8L		566	162	215		18	1230	4530	
D	UEC37LA	370	880	95	4-8L		520	158	210		15.6	1120	4160	
D	UEC37LSII	370	1290	139	5-8L		772	140	186		18	2095	6180	
D	UEC43LSII	430	1500	218	4-8L		1050	120	160		18	2280	8400	
D	UEC45LA	450	1350	215	4-8L		890	119	158		15.6	1920	7120	
D	UEC45-LSE	450	1840	293	5-8L		1245	111	130		19.6	4250	9960	
D	UEC50LSII	500	1950	383	4-8L		1445	95	127		17	3120	11560	
D	UEC50LSE	500	2400	403	5-8L		1620	99	124		19.5	5175	12960	
D	UEC52LA	520	1600	340	4-8L		1180	100	133		15.6	2560	9440	
D	UEC52LS	520	1850	393	4-8L		1330	90	120		16.9	2880	10640	
D	UEC52LSE	520	2000	425	4-8L		1705	95	127		19	3700	20460	
D	UEC60LA	600	1900	537	4-8L		1550	83	110		15.7	3360	12400	
D	UEC60LS	600	2200	622	4-8L		1770	75	100		17	3800	14160	
D	UEC60LSII	600	2300	650	4-8L		2045	79	105		17	4440	16360	
D	UEC60LSE	600	2400	2713	5-8L		2255	90	105		19	7650	18040	
D	UEC68LSE	680	2690	977	5-8L		2940	81	95		19	10050	23520	
D	UEC75LSII	750	2800	1237	4-9L		2940	63	84		17	6380	26460	
D	UEC85LSII	850	3150	1787	5-9L		3860	54	76		17	9900	34740	
D	UEC85LSC	850	2360	1339	5-12L		3900	76	102		17	10575	46800	

▲
The Mitsubishi Heavy Industries, Ltd. diesel-engine programme.

Fuel Type D: Diesel or Heavy Fuel DF: Dual Fuel SI: Spark Ingnited	Engine Model	Bore (mm)	Stroke (mm)	Displacement (liters/cylinder)	Number of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed	Output per Cylinder Range (kW/cyl)		Rated Speed Range (r/min)		Maximum Brake Mean Effective Pressure (bar)		Output Range Low (kW)	Output Range High (kW)	Ratings
						min	max	min	max	min	max			
D	Wärtsilä RT-flex96C	960	2500	1810	6 – 14L	4000	5720	92	102		18.6	24000	80080	
D	Wärtsilä RTA96C	960	2500	1810	6 – 14L	4000	5720	92	102		18.6	24000	80080	
D	Wärtsilä RT-flex84T-D	840	3150	1746	5 – 9L	2940	4200	61	76		19	14700	37800	
D	Wärtsilä RTA84T-D	840	3150	1746	5 – 9L	2940	4200	61	76		19	14700	37800	
D	Wärtsilä RT-flex82T	820	3375	1782	6 - 9L	3620	4520	68	80		20	21720	40680	
	Wärtsilä RTA82T	820	3375	1782	6 - 9L	3620	4520	68	80		20	21720	40680	
D	Wärtsilä RT-flex82C	820	2646	1397	6 - 12L	3620	4520	87	102		20	21720	54240	
D	Wärtsilä RTA82C	820	2646	1397	6 - 12L	3620	4520	87	102		20	21720	54240	
D	Wärtsilä RTA72U-B	720	2500	1018	5 – 8L	2155	3080	79	99		18.3	10775	24640	
D	Wärtsilä RT-flex68-D	680	2720	988	5 – 8L	2150	3070	76	95		20	10950	25040	
D	Wärtsilä RTA68-D	680	2720	988	5 – 8L	2150	3070	75	95		19.6	10750	24560	
D	Wärtsilä RTA62U-B	620	2150	649	5 – 8L	1600	2285	92	115		18.4	8000	18280	
D	Wärtsilä RT-flex60C-B	600	2250	636	5 – 9L	1650	2360	91	114		20	8450	21780	
D	Wärtsilä RT-flex58T-B	580	2416	638	5 – 8L	1530	2180	84	105		19.5	7650	17440	
D	Wärtsilä RTA58T-B	580	2416	638	5 – 8L	1530	2180	84	105		19.5	7650	17440	
D	Wärtsilä RTA52U	520	1800	382	5 – 8L	1090	1560	108	135		18.1	5450	12480	
D	Wärtsilä RT-flex50	500	2050	403	5 – 8L	1160	1660	99	124		20	5800	13280	
D	Wärtsilä RTA50	500	2050	403	5 – 8L	1160	1660	99	124		20	5800	13280	
D	Wärtsilä RTA48T-B	480	2000	362	5 – 8L	1020	1455	102	127		19	5100	11640	

▲

The Wärtsilä Corporation diesel-engine programme.

Diesel engines for ship propulsion and diesel power plants

Numbers

Obviously, the number of small engines manufactured, in particularly for categories I and II, is enormous. Every year, hundreds of thousands of engines are manufactured for various vessels and other propulsion devices by a large number of engine manufacturers. These total approximately 15 million diesel engines per annum.

In surveys of engines for shipping and diesel power plants the number of engines manufactured annually is 'counted' from a certain bottom limit, that of shaft power; often 1000 or 2000 kW. Clearly, the moment a large two-stroke low-speed crosshead engine with fourteen cylinders in the largest diameter is ordered, this attracts more attention in the industry than when a much smaller four-stroke medium-speed diesel engine is ordered.

Striking aspects of engine manufacturing

1 Major engine manufacturers produce a large number of engine types, as to cylinder diameters, speed and power output. In this manner, they endeavour to 'cover' their market.
2 Engines in category I often start with a single cylinder and move on to a six-cylinder in-line arrangement.
3 Engines in category II often start with a three- or four-cylinder in-line arrangement and move to sixteen-, eighteen- or twenty-cylinder V-arrangements.
4 Engines in category III often start with a four-cylinder in-line arrangement and move on to an eighteen-cylinder V-arrangement.
5 Engines in category IV usually start with a four-cylinder in-line arrangement and at present are produced with up to fourteen cylinder in-line arrangements.
6 Especially for category III, with the limited number of engine manufacturers, it is noticeable that they often have engine

programmes for the same cylinder diameter. Other manufacturers have 200, 250 and 320 mm cylinder diameters.

7 The mean effective pressures and the mean piston speeds are increasing due to keen competition between the manufacturers. Consequently the load parameter also increases!

8 Very few new engine manufacturers enter the market. However, the number of manufacturers that have closed their businesses or merged with others continues to increase. After a merger, certain types of engines are only maintained if they are economically viable and after several years the brand name disappears and an increasing number of renown international and national trade-marks end up in the history books.

9 Fuel used.
As a result of increasingly strict regulations regarding maximum emission levels for pollutants in exhaust gases and a growing realisation that supply of exploitable crude oil is finite, engine manufacturers and consumers are attempting to find 'alternative' fuels. The diesel process where small amounts of liquid fuel are combined with gaseous fuel, the so-called diesel/gas process, is for instance in a development. The emissions are often lower in comparison with those of regular diesel engine. The usage of bio fuels, such as olive oil, palm oil, rape oil and other vegetable oils are also being developed. These fuels work **CO_2 neutral**, which means that they emit the same amount of CO_2 to the air during combustion as they absorb from the atmosphere during their growth.

Also see chapter 29, New developments in the fuel industry.

Future of the diesel engine industry

Diesel engine performances have to meet high requirements.
The legal regulations regarding emission levels for pollutants are being applied to more areas in the world and will become more stringent in the future.
Users of diesel engines demand an optimum service network and a growing number of these consumers operate worldwide. This means that engines manufacturers must maintain an extensive service organisation. This is only economically viable for large manufacturers.
The development costs of new engine types are extremely high. These costs are made to:

a reduce fuel consumption;
b reduce lubricating oil consumption;
c create a higher mean effective pressure;
d produce a higher mean piston speed;
e create and therefore a higher load parameter;
f produce longer life for the engine parts;
g produce lower emissions;
h be suitable for alternative fuels;
i get a lower specific weight;
j devise simple construction and maintenance-friendly disassembly and assembly methods.

As well as numerous other points with which an engine manufacturer may stay ahead of the competition!
Due to above mentioned reasons, many engine manufacturers are either economically or due to organisational issues, no longer capable of developing new engines and discontinue certain engine series, merge or simply close their business. Over fifty percent of the engine manufacturers discontinued their businesses between 1950 and 2005.

Alternative methods for energy generation other than by diesel engines are:

- fuel cells;
- gas turbines;
- hydrogen;
- kite sailing.

These alternatives could become successful and are dependent on numerous factors, so very little can be said about the future developments at this time.

Complete list of 46 diesel engine manufacturers, July 2008

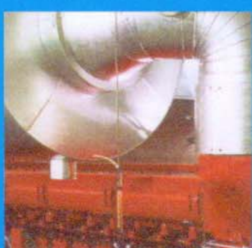
Contents Diesel Publications.

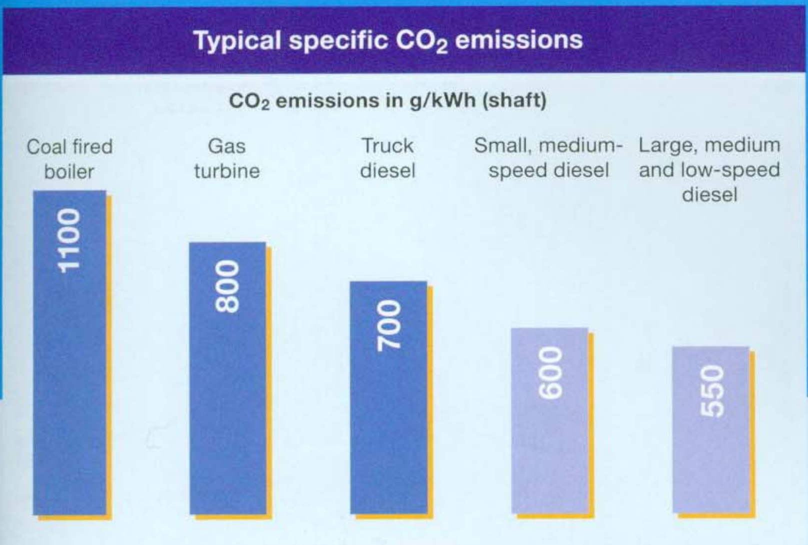
Anglo Belgian Corporation
Briggs & Stratton Commercial Power
Caterpillar Inc.
Cummins Inc.
Daihatsu Diesel Mfg. Co, Ltd.
Deutz Corporation
Deutz Power Systems GmbH & Co. KG.
Doosan Infracore Co, Ltd.
Electro-Motive Diesel, Inc.
Fairbanks Morse Engine
Fiat Powertrain Technologies SpA.
Greaves Cotton Ltd.
Greaves Farymann Diesel GmbH.
GE Energy
Guascor. S.A.
H. Cegielski , Poznan S.A.
Hino Motors , FPT.
Isotta Fraschini Motori SpA.
Isuzu Motors, Ltd.
Iveco Motors,Ltd.
JSC ZVEZDA
John Deere Power Systems
Kubota Corporation
Lister Petter Ltd.
Lombardini S.R.L.
MAN-Diesel AG
MAN Nutzfahrzeuge Aktiengesellschaft
Marine Diesel Sweden AB
Mitsubishi Heavy Industries. Ltd.
Mitsui Engineering & Shipbuilding Co, Ltd.
Motorenfabrik Hatz GmbH & Co, KG
MTU Friedrichshafen GmbH.
Niigata Power Systems(Europe) BV.
Perkins Engine Company Ltd.
Rolls-Royce
Ruggereni Motori
RUMO,JSC
S.E.M.T. Pielstick
Scania
Sisu Diesel Inc.
STEYR MOTORS GmbH.
VM Motori SpA.
Volkswagen AG
Volvo Penta
Wärtsilä Corporation
Yanmar Co, Ltd.

>CH 22

Engine emissions







The carbon dioxide emissions in grams per kWh with various energy generating methods.

In many parts of the world, electricity is generated using large diesel engines. In Honduras, where heavy-fuel oil is half the price of diesel oil, this is the fuel of choice. Pollutant emissions of heavy-fuel oil are higher than that of diesel oil. Due to increasingly strict pollutant emission requirements, engine manufacturers are using new technologies such as those of common-rail systems and exhaust-gas cleaning. Ultimately, the price per generated kWh is the deciding factor in determining which fuels and systems are selected in order to comply with emission regulations.

▶

Many human activities have an impact on the environment.

Cruise ships sailing to beautiful regions in the world pollute the environment. The emission of pollutants must be restricted as much as possible. Increasingly strict legislation ensure that this a major point when designing an engine.

22.1 Introduction to 'the fossil fuel' society

Society has been transformed since around 1750, the beginning of the industrial revolution in Western Europe. With the introduction of the steam boiler and steam engine, continuous power was available, for the first time in history, independent of wind, water or animals.
However, this required a constant fuel supply. At first coal was used as a fuel for the steam boiler. In later technical developments, steam turbines were used on a large scale.
Especially after 1850, when the Otto internal-combustion engine was developed, and just before 1900 when the Diesel internal combustion engine was introduced, liquid and gaseous fossil fuels were increasingly used.
Around 1950 the diesel engine was used more frequently for ship propulsion and soon new diesel engines suitable for operation with relatively inexpensive fuel oil (without excessive problems), initially only used for steam boilers, were introduced. Originally this was H.F.O. used in the large two-stroke crosshead engines and has since the past 25 years, been introduced for medium-speed diesel engines.

Use of fossil fuels

Use of fossil fuels has dramatically risen over the last hundred years and after ±1950s, the increase has been explosive.
In 2008 the demand for energy, in countries such as China and India in particular as well as the other emerging Asian countries, has increased significantly and the price of a barrel of crude oil (159 litres), has exceeded the hundred-dollar boundary (April 2008).

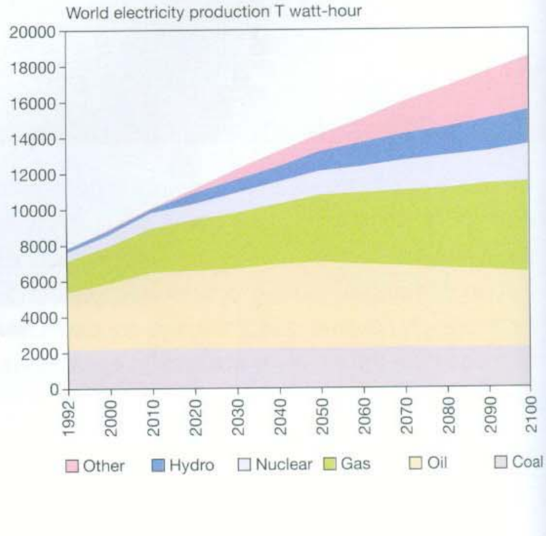
Dependence on fossil fuels

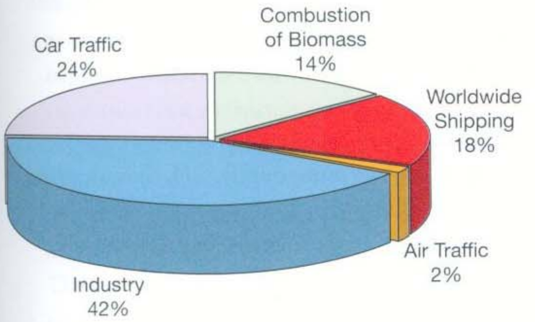
This is extreme. Apart from its use in the generation of energy for an increasing number of sectors, a large number of products are manufactured from crude oil. Among these are synthetics for building materials and clothing, fertilisers and pesticides for agricultural use as well as an infinite number of raw materials and semi-finished products in the chemical industry.

▶

A vision for various sources for the generation of energy to 2100.

The use of coal remains constant. The use of crude oil decreases slightly. The use of natural gases increases significantly. The use of nuclear energy increases as well as the use of water-power and other energy sources, such as wind and sun. It can be concluded that although the energy sources change somewhat, the total amount of energy consumption has doubled. This has enormous consequences for the environment and therefore our existence.





▲
The global energy consumption in 2000. The emission of pollutants is indicated for each consumer group.

Crude oil dependence is universal and comprehensive.
In fact, modern society is built on the availability of a continuous oil supply!

Environmental effects

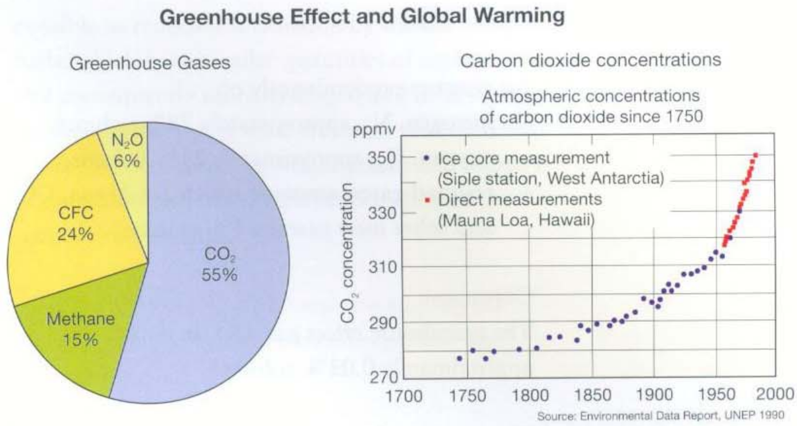
Effects of the large-scale usage of fossil fuels on the environment have long been known and researched in detail throughout the years. It has been established that fossil-fuel combustion weighs heavily on our environment.
Topics such as the greenhouse effect, acid deposition, harmful emissions and a reduced habitat quality for flora and fauna are daily recurring conversational topics.

22.2 Exhaust gas composition

A modern two-stroke crosshead engine of Wärtsilä is used as an example.
Wärtsilä builds a series of crosshead engines with cylinder diameters from 480 to 960 millimetres with power outputs between 7000 and 76,000 KW. Heavy-fuel oil with a maximum viscosity of 730 cSt at 50 °C: 150 8217, category ISO-F-RMK55 is used as the fuel. This is heavy-fuel oil with a sulphur content of ± 3.5%.

Also, see Chapter 8, Fuels, fuel-line systems and cleaning fuel.

In the combustion process overview, air, fuel and lubricating oil are added as follows to the engine.



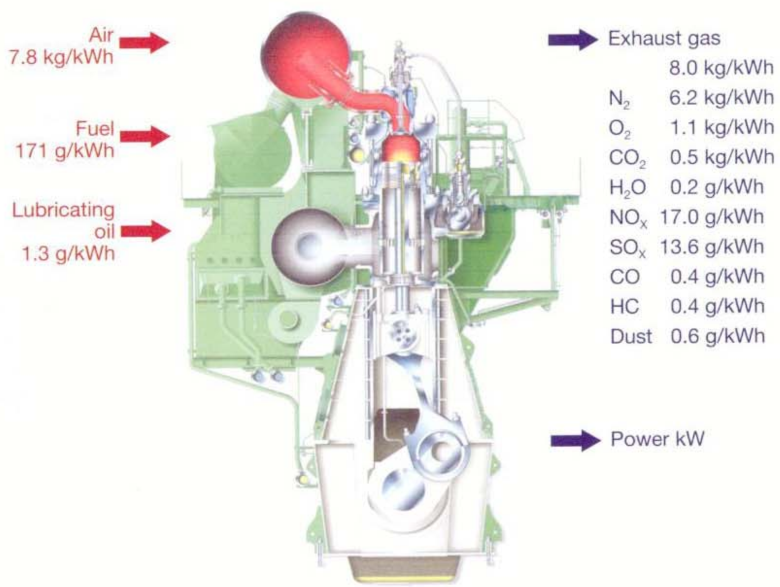
▲
This graph shows how various gases contribute to the greenhouse effect.

Note the increase starting from approximately 1750, (first industrial revolution) and the increase around 1950, (mechanisation/electrification throughout society). The graph after the year 2000 also shows an ominously sharp incline.

▼
An overview of the combustion process in a two-stroke crosshead engine running on heavy fuel.

Note the ratio between the amount of the fuel and air supplied to the engine. Here the ratio is 1: 45.6. The oxygen content (O_2) shows that there is still a considerable amount of oxygen in the exhaust gas. In the combustion of 7.8 kilograms air, the oxygen content is reduced from 1.638 kilograms to 1.1 kilogram. The consumption during the combustion process is 1.638 – 1.1 = 0.538 kilograms per kW. The excess air (Lambda) of this engine is $\frac{1.638}{0.538} = 3.04$.

Reminder: air comprises approximately 21% oxygen and 79% nitrogen. 21% of 7.8 kilograms air is 1.638 kilograms oxygen.



22.2.1 Air

Air consists predominantly of:
- **nitrogen,** N_2, approximately 78% volume;
- **oxygen,** O_2, approximately 21% volume;
- **residual gases, amongst which are** Argon, CO_2 and other inert gases, ± 1% volume.

Comment
The greenhouse effect gas, CO_2 in air is approximately 0.03% volume!

The amount of air supplied to this engine is approximately three times the amount of air required for a complete chemical combustion of the fuel, or: the excess air is 3.

Also, see Chapter 8, Fuels, fuel-line systems and cleaning of fuel.

7.8 kilograms of air are supplied per kWh.

22.2.2 Fuel

For this engine heavy-fuel oil which consists almost entirely of:
- **hydrocarbon compounds,** CH, approximately 97% volume;
- **sulphur,** S, approximately 3% volume.

Heavy-fuel oil, of course, also contains small amounts of other substances.

22.2.3 Cylinder-lubricating oil

Used lubricating oil combusts. In a two-stroke crosshead engine, cylinder lubrication is separate of the engine-lubricating system. The engine's lubricating oil almost entirely combusts and is discharged in the exhaust gases. A smaller part flows to the scavenging-air space and is drained.

Also, see Chapter 11, Lubrication of engines.

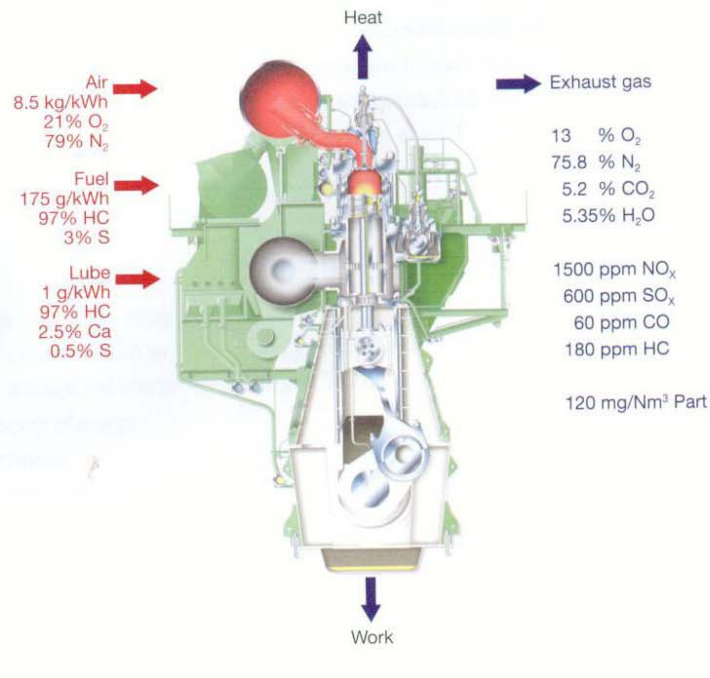
This consists of:
- **hydrocarbon compounds,** CH, approximately 97% volume;
- **calcium,** Ca, approximately 2.5% volume;
- **sulphur,** S, approximately 0.5% volume.

The three-abovementioned substances, air, fuel and lubricating oil are definitive for the combustion process.

In the Wärtsilä overview, 7.8 kilograms of air, 171 grams of fuel and 1.3 grams of cylinder-lubricating oil is supplied per kW hour (kWh), approximately 8 kilograms in total. Subsequently, 8 kilograms of exhaust gas are formed with the following composition:

– nitrogen	N_2	6.2 kg/kWh
– oxygen	O_2	1.1 kg/kWh
– carbon dioxide	CO_2	0.5 kg/kWh
– water	H_2O	0.2 g/kWh
– nitrogen oxides	NO_X	17.0 g/kWh
– sulphur oxides	SO_X	13.6 g/kWh
– carbon monoxide	CO	0.4 g/kWh
– hydrocarbon compounds	HC	0.4 g/kWh
– dust particles		0.6 g/kWh

An example for the volume percentages for two-stroke crosshead engines type MC of engine manufacturer MAN–B&W.



In this figure, the gas volume percentages are clearly shown.

Similar to fuel, the cylinder lubricating-oil comprises mainly of hydrocarbon chains. The composition of noxious substances in the exhaust gases is shown in p.p.m. (parts per million).

Striking features:

- The largest part of the nitrogen is, as in air, found in the exhaust gases. Essentially, nitrogen is an inert gas and does not normally combust with other substances. Under certain circumstances, for instance in a diesel engine, nitrogen oxide will react with oxygen. This requires process temperatures of over 1200 °C and these frequently occur, often up to 1800 °C!
- The exhaust gases still contain oxygen, also referred to as excess air.
- Water vapour is produced from the hydrogen component in the fuel.

22.2.4 Exhaust-gas composition

1 Oxygen

Diesel-engine exhaust gases always contain oxygen. Oxygen is required for the combustion process, but the air is also used for the scavenging and cooling processes in the engine. Therefore, due to excess air in diesel engines, exhaust gases always contain oxygen that also indicates the magnitude of the excess air. The oxygen content varies from approximately 13 to 16%. The excess air is often 3 or more.

2 Nitrogen

As in air, nitrogen is the main component of exhaust gases. It is inert and therefore does not react with other substances.

3 Carbon dioxide

The carbon in the fuel reacts with the oxygen. This produces carbon dioxide at perfect combustion. This gas appears to contribute to the so-called greenhouse effect. For diesel engines, it is only possible to reduce this emission by the use of light fuels, which have smaller quantities of carbon and consequently more hydrogen and to develop engines with a higher total efficiency and therefore lower fuel consumption. Maintenance and manner of operation also offer (limited) possibilities to reduce this emission.

Also, see Chapter 27, Maintenance and repairs and Chapter 25, Operational management and automation.

4 Water vapour

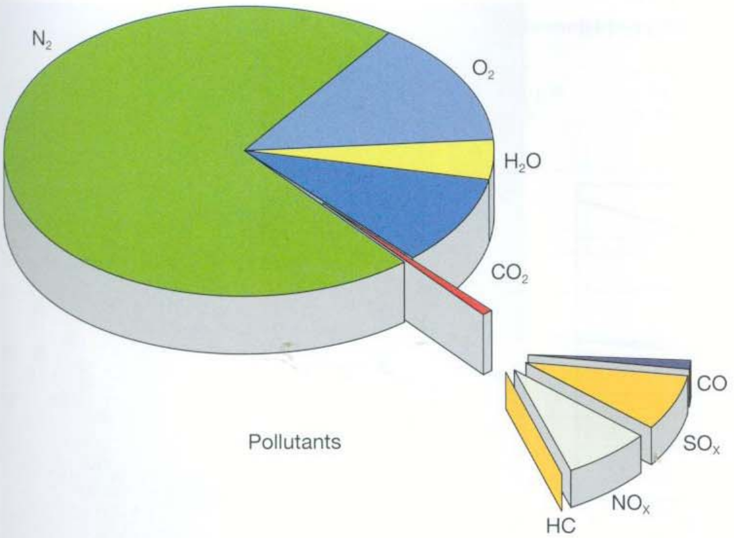
Hydrogen in fuels reacts with oxygen to form water vapour.

5 Carbon monoxide

This is produced during imperfect combustion of the fuel. In principle this occurs when too little oxygen is available for perfect combustion, for instance, close to the walls of the combustion chamber. Poor fuel and air mixture may also be a cause. Carbon monoxide can react with oxygen and this can simply occur in the engine. It is a very toxic gas for both man and fauna. **Carbon monoxide can kill instantaneously when inhaled in a closed space.**

6 Sulphur dioxide

This is produced with perfect combustion and is a reaction of sulphur with oxygen. In addition to sulphur dioxide SO_2, sulphur trioxide SO_3 is also formed. Sulphur dioxides are the main compounds in acid rain and have a destructive effect on flora and fauna, human respiratory systems; they also attack building materials, in particular soft limestone. The effects of sulphur oxide compounds are negligible at sea. Seawater



In this figure; the composition of exhaust gases.

N_2 = nitrogen
O_2 = oxygen
H_2O = water (steam)
CO_2 = carbon dioxide
SO_X = sulphur oxide
HC = hydrocarbon

is slightly alkaline and therefore they are easily neutralised. One litre of seawater can neutralise approximately 300 milligrams of sulphur. Furthermore, sulphur dioxide is not carried long distances but is relatively quickly precipitated into the sea. A greater negative effect should occur within 100 kilometres from the coast. However, measurements taken in the busy English Channel have demonstrated that the contribution of shipping to air pollution in the English Channel region is only 5%. The remainder is caused by sulphur oxide emissions from shore-based producers, such as electric-power plants and chemical factories. Sulphur dioxide emissions fall under the I.M.O., the International Maritime Organisation, regulations.

7 Nitrogen oxides

These are produced during the combustion process at high temperatures between nitrogen and oxygen. Fuel contains small amounts of nitrogen. Heavy fuels contain more nitrogen than light fuels.

Nitrogen oxides are compounded under the following circumstances:

01 **Sufficient oxygen** available. This is always the case in diesel engines. The excess air often constitutes more than two or even three.

02 **A temperature** in the combustion process in excess of **1200 °C**. In the hottest part of the process, this is normally the case. Temperatures from 1400 °C to 1800 °C are no exception.

03 **Time** in the combustion process so that nitrogen oxides are formed at this high a temperature and in an oxygen-rich environment.

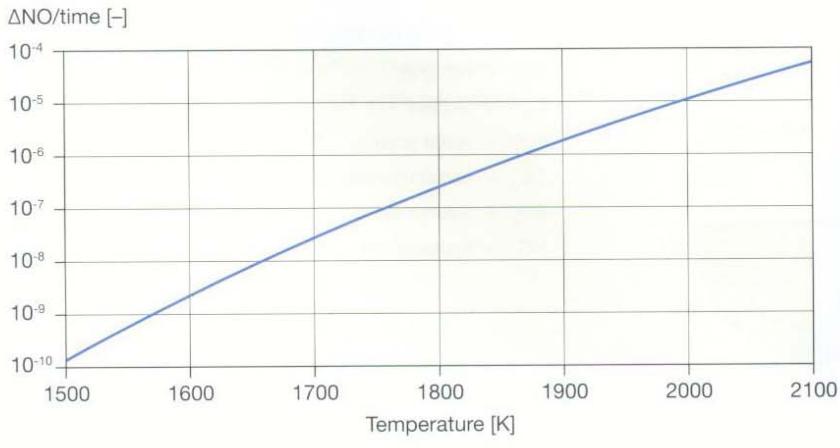
Comment

The total combustion process including expansion is for two-stroke engines, approximately 120 crank degrees and four-stroke engines about 140 crank degrees.

▼

The generation of nitrogen oxides.

Horizontal: the temperature in ° Kelvin of the combustion process and vertical: the time at which these high temperatures occur.



Example 1

Assumption: A two-stroke crosshead engine has an RPM of 94.

The total combustion process including expansion: the number of passed-through crank degrees per second is the number of revolutions per second × 360 crank degrees:

$$\frac{94}{60} \times 360 = 464 \text{ crank degrees per second.}$$

Then the combustion process takes

$$\frac{120}{564} = \mathbf{0.212\ second,} \text{ so approximately } \frac{2}{10} \text{ seconds.}$$

Example 2

Assumption: A four-stroke diesel engine has an RPM of 720.

The number of passed-through crank degrees per second:

$$\frac{720}{60} \times 360 = 4320 \text{ crank degrees per second.}$$

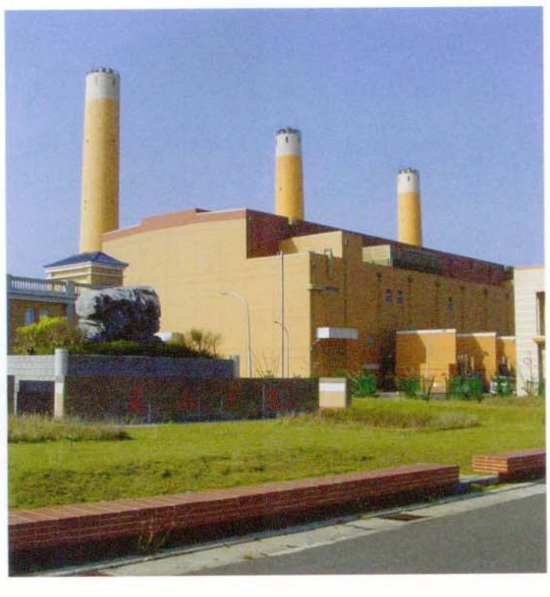
Then the total combustion process including expansion takes $\frac{140}{4320}$ seconds = 0.032,

so approximately $\frac{3}{100}$ seconds.

▼

A diesel power station in China for the generation of electricity.

Cheap heavy fuel oil is becoming increasingly restricted in its use all over the world. The use of sulphur-poor heavy fuel oil or even banning the use of heavy fuel oil is occurring more often. Relatively clean natural gases will be used as an engine fuel more often in the future. This is already taking place to a significant degree in engines with the 'Dual Fuel' (DF) principle, a combination of diesel oil and gas, and the 'spark gas' (SG) principle, a gas engine.



A diesel power plant with a engine running on heavy fuel oil.

Example 3

A small high-speed four-stroke engine with an engine speed of 2100 revolutions per minute:
The number of passed-through crank degrees per second:

$$\frac{2100}{60} \times 360 = 12{,}600 \text{ crank degrees per second.}$$

Then the total combustion process including expansion takes:

$$\frac{140}{12{,}600} = 0.011 \text{ second of } \frac{1}{100} \text{ seconds.}$$

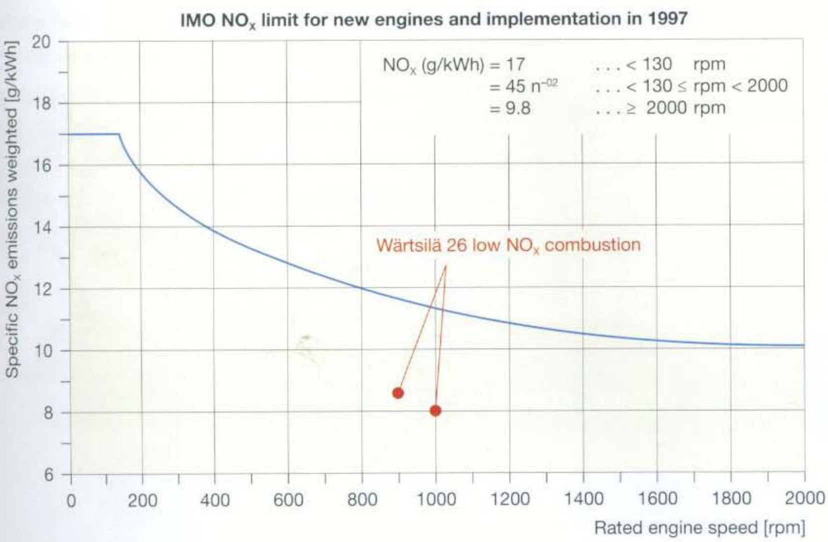
In conclusion: Combustion **process:**

I	**low-speed engine**	94 rev/min.	0.212 sec.
II	**medium-speed engine**	720 rev/min.	0.032 sec.
III	**high-speed engine**	2100 rev/min.	0.011 sec.

From the above, it is shown that a low-speed engine has more time for the formation of nitrogen oxides at high process temperatures than medium-or high-speed engines.
So high-speed engines by definition emit far less nitrogen oxide, not only because the fuel used, diesel oil, contains less nitrogen.
The higher the RPM of an engine, the fewer nitrogen oxides it produces.
The I.M.O. regulations assume a certain engine speed and a standard emission weight in grams NO_X per produced kWh.

22.2.5 Hydrocarbon compounds

These are often partially combusted or uncombusted hydrocarbons from fuel oil or lubricating oil. These are very diverse compounds with virtually any possible chemical combination of carbon, hydrogen, nitrogen, oxygen and sulphur, albeit in extremely low concentrations.



A well-known graph.

Horizontal: the nominal engine speed of the diesel engines and vertical: the nitrogen oxide emissions in grams per kilowatt-hour. For slow-speed engines, the maximum emission is 17 gram per kWh. Medium-speed engines may discharge between 17 and 12 grams per kWh and fast-speed engines between 12 and 9.8 grams per kWh, dependant on the speed. Every engine manufacturer today, builds engines with low nitrogen oxide emissions, the 'Low NOx Combustion'. Shown is an example of a Wärtsilä 26 engine.

The bonds these compounds will form during the combustion process are hard to predict, as are their effects on the environment and human and animal health.
Generally, hydrocarbon compounds are a result of imperfect combustion and in fact largely depend on the parameters of the combustion process, such as velocity, temperature and oxygen content. Well-tuned engines ensure that these HC-compounds are kept as low as possible.

Solids

These are a complex mixture of organic and inorganic substances, amongst others, carbon, ash particles, heavy metals, precipitated sulphur oxides, water, corrosion particles and a variety of partially combusted hydrocarbons from the fuel and the lubricating oil. The diameter of these substances is less than $\frac{1}{1000}$ millimetres can easily be carried to the outside air.

Inhalation of these exhaust gases can be detrimental to health. Naturally, fuel quality plays an important role.

Smoke, visible smoke

Smoke is of course visible and can be quantified by the following methods:

- Bosch smoke number;
- S.A.E. smoke number;
- Bacharach (A.S.T.M.) smoke number.

Smoke mainly consists of carbon particles, also referred to as soot.

Today, engine manufacturers are developing means in which 'visible smoke' can be reduced.

Black smoke is the result of imperfect combustion. Grey smoke may indicate the presence of water in the exhaust gases.

▲
Almost all diesel engines produce visible smoke.

This is mostly the case with a reduced power output or with quick power changes as with this container ship passing through the locks at Kiel, Germany.

▲
A ferry arriving in a harbour.

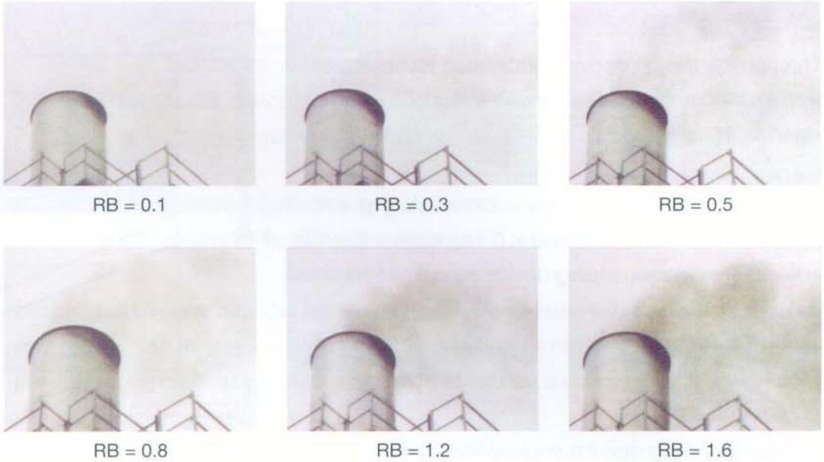
Clearly shown is the visible smoke, consisting of soot particles (carbon) and fine dust (metal particles).

▶
A large container ship approaching Hamburg terminal.

At a reduced power output, no smoke is visible. In the design of new engines types, engineers try to create the smokeless engine.

When 100 kilograms of fuel is burnt, approximately 120 kilograms of water is produced with an additional 20 kilograms of water from the scavenging-air supply.
Smoke visibility is also dependent on atmospheric conditions.

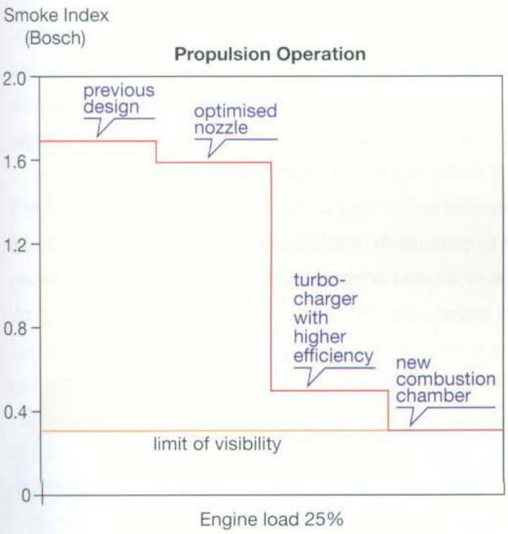
Yellow smoke indicates the presence of sulphur in the fuel.
If heavy-fuel oil contains above 2% of sulphur, this is clearly visible.

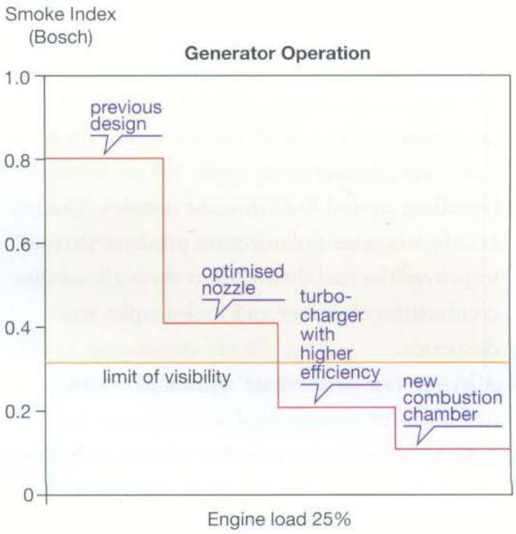


▲

Visible smoke can be defined in smoke numbers.

Shown: the Bosch method, using these pictures, smoke emissions can be determined.





▲

Two examples of the Bosch smoke index.

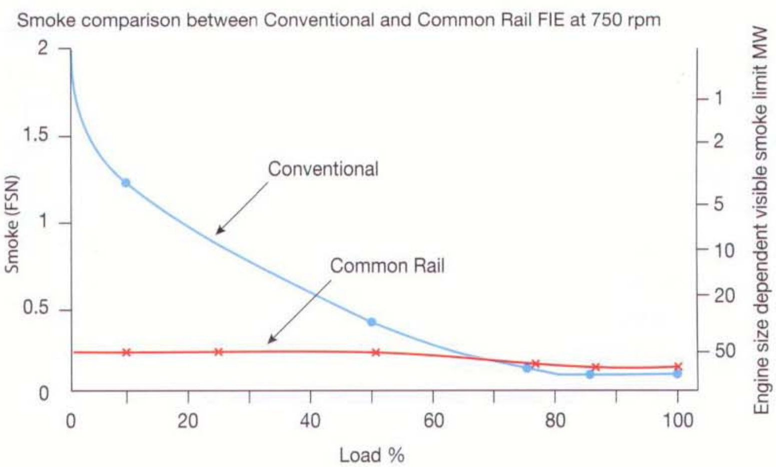
Left: an example for a propulsion engine and right: an engine for a generator.
The regular load of the generator and the constant high operational speed ensure low values. The irregular load of the propulsion engine at reduced speed has at a low load of 25% a much higher soot emission. The high constant speed of the generator set allows for a better combustion because of the higher process temperatures.

▼

The difference between a conventional fuel injection and the common rail system of Wärtsilä.

horizontal: engine load between 0 and 100%
vertical left: smoke index (FSN is Fuel Smoke Number)
vertical right: permissible visible smoke dependant on the engine power output in MW.

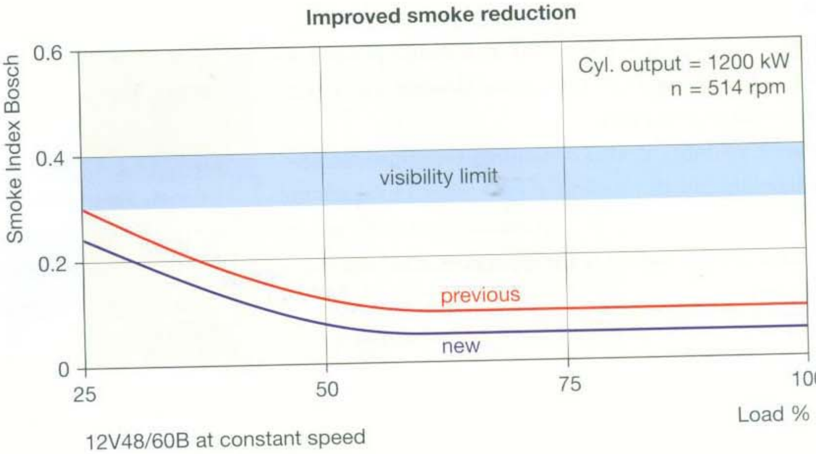
By the application of modern fuel injection methods such as the common rail, smoke emissions are strongly reduced between zero load and 80% load. At full load, there is almost no difference.



▶

Through the design of new combustion technologies with existing engine types, smoke emissions can be significantly reduced.

The visible smoke limit for this four-stroke-diesel engine of MAN–B&W has been established at 0.4 on the Bosch smoke index. The old (previous) design is represented by the red line and the new design by the green line. The engine has a constant speed at full load with a cylinder power output of 1200 kilowatt at 514 revolutions per minute. The Bosch smoke index is below 0.1.
The blue area designates the smoke visibility limits.



22.2.6 An example of smoke reduction

In response to universal concern regarding environmental issues, engine manufacturer MAN–B&W started a programme in 1996 to reduce smoke in exhaust gases, especially at low engine loads.
The following measures were taken:

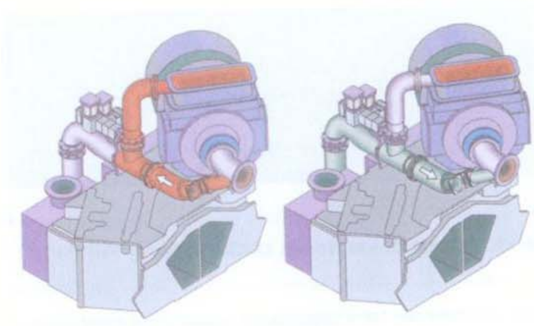
- **Pre-heating of the scavenging air.** This method involved compensating the reduced process temperatures caused by the injection of less fuel. This takes place at loads below 20%.
- **Installing partial-load injector nozzles.** This nozzle increases the injection pressure that improves the fuel distribution throughout the combustion chamber and fuel-droplet size decreases.
- A by-pass or **'waste-gate' mounted on the turboblower.**

Overall, a turbo-blower is most efficient at heavy engine loads. The efficiency can be 'advanced' at lower loads by adjusting the blower and diffuser of the air compressor. The scavenging air pressure will also often increases allowing a more effective scavenging of the cylinder. If this engine runs at

▼

Adjusting the amount of air supplied to the engine is important at variable loads.

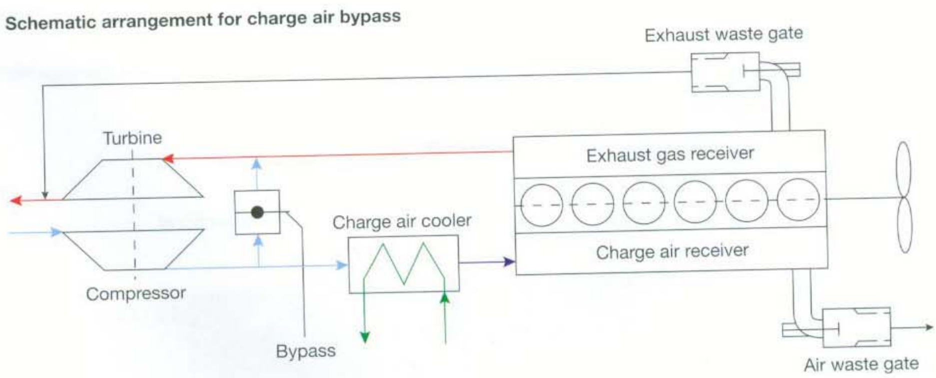
Left, the exhaust gas by-pass so that the amount of charge air is limited at full load. Right, the amount of charge air is controlled by the magnitude of charge air allowed through the by-pass. Both methods have the same goal: the optimal amount air at all engine loads so that optimum combustion can occur.



▶

An example with a mechanism on the inlet manifold and the exhaust gas manifold where a 'waste-gate' is installed for both systems.

Both valves are adjustable. Furthermore, a by-pass valve has been placed between the air compressor and the exhaust gas turbine that controls the amount of air to the engine.



full load, the scavenging air pressure will be too high. Therefore, a by-pass or waste-gate valve is placed in the exhaust-gas supply of the exhaust-gas turbine.

Fuel emulsions

The main objective of a fuel-water mixture (fuel emulsion) is the reduction of nitrogen oxides in the exhaust gases. At low loads, visible smoke is reduced as the fuel-water emulsion is injected at higher pressures.
Placing a **bypass valve** between the scavenging air receiver and the exhaust-gas manifold is also an option. This valve is opened at low engine loads. Due to reduced air resistance, the turbo-blower will revolve faster, producing a higher flow efficiency effecting a higher airflow through the engine.

Auxiliary blower

For a standard turbo-blower, the efficiency at no load or low loads lies far below the maximum value. In this instance, an electrically driven auxiliary blower can be installed to reduce significantly visible smoke.

Fuel quality

This is the main influence on the combustion-process quality and therefore smoke production. At times, fuels with different specifications are bunkered. This specification change can cause a considerable increase in smoke production. It is then advisable to order the original specified fuel for the next delivery.

22.3 Units of contamination

The substance load is often indicated in kilograms or tons per hour or year. Presently, grams of contamination per generated unit kilowatt-hour, is frequently used when referring to energy generation.
Simply: contamination is denoted in g/kWh.
The unit, parts per million or **p.p.m. is also used.**

Exhaust gas emissions must be specified at a certain oxygen concentration, for example 5, 13, or 15% O_2. This is often not related to the actual oxygen percentage found in the air in the diesel engine in question. It is, however, easy to convert, for example p.p.m. in volume to g/kWh by applying the following formula (ISO 8178 part 1).

$$\text{X corr.} = \text{X ref.} \frac{20.95\% - O_2 \text{ corr.}}{20.95\% - O_2 \text{ ref.}}$$

X corr. = actual gas concentration
X ref. = reference oxygen concentration

22.3.1 Pollutant-emissions regulations with respect to diesel engines

In the final decennia of the last century, growing environmental concern led to regulations for limiting pollutants in exhaust gases of, amongst others, diesel engines.
Universal regulations have been established for shipping. This prevents individual countries or continents from drawing up their own regulations and was initiated by the United Nations.
The I.M.O., has recorded these in Annex IV of Marpol 73/78, which for the first time in history stipulates the regulations for the maximum exhaust-gas emissions.
The annex concerns the limitation of sulphur- and nitrogen-oxide emissions and has been compulsory since 2000 for all newly built engines with a shaft power over 130 kW.
This ratification of the NO_X and SO_X limitations became effective when more than fifteen flag states, members of the I.M.O. with a tonnage of at least 50% of the total world tonnage signed this agreement.

These agreements entail:

- I.M.O. regulation for the maximum sulphur- and nitrogen-oxide emissions;
- optimising diesel-engine designs in order to comply with the I.M.O. regulations;
- maintain a 'technical' bookkeeping;
- terms such as 'single engine', 'engine group' and 'engine family';
- comments/instructions from engine manufacturers;
- comments/instructions from shipping companies.

Sulphur oxides

Fuel sulphur content limited to a maximum of 4.5% mass. In areas with local limitations, 1.5% mass sulphur.

Nitrogen oxides

The maximum limits are based on the RPM for diesel engines with a shaft power of over 130 kW. This RPM graph is only applicable for ships built after the 1st of January 2000, or to engines that have undergone extensive modifications.

The I.M.O. directives for regulating the emission of the noxious substances, NO_X and SO_X.

Ratification

- The NO_x and SO_x limitations will only be in force when a minimum of 15 Flag States in the IMO, with at least 50% of the tonnage, have ratified the regulation.

SO_x

- Limitation of sulphur content in fuel oil to 4.5% mass
- In areas with local restrictions, the limit is 1.5% mass sulphur

NO_x

- Limit based on speed-related NO_x curve (for engines larger than 130 kW)
- The speed-related curve will only apply to ships with keel laying after 1st Jan. 2000, or following a major engine modification

Test Cycles

Heavy duty diesel engines

E2 Marine Engines, constant speed:

Load point	100%	75%	50%	25%
Weighting factor	0.2	0.5	0.15	0.15

E3 Marine Engines, propeller law:

Load point	100%	75%	50%	25%
Weighting factor	0.2	0.5	0.15	0.15

D1 Stationary Engines, constant speed:

Load point	100%	75%	50%
Weighting factor	0.3	0.5	0.2

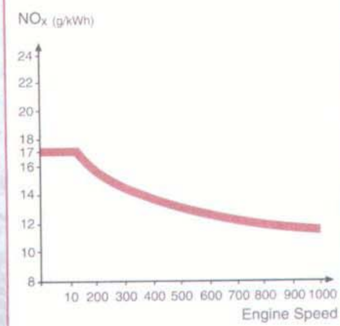
Formula:

$$NO_x\ (g/kWh) = \frac{\Sigma^{1\text{-}4}\ NO_x \times P \times F}{\Sigma^{1\text{-}4}\ P \times F}$$

F = Weighting factor
P = Engine power at each load point (kW)
NO_x = NO_x level at each load point (g NO_x/kWh)

Speed-related curve

NO_x (g/kWh) = 17 for $n \leq 130$
NO_x (g/kWh) = $45 \times n^{-0.2}$ for $130 < n < 2{,}000$
NO_x (g/kWh) = 9.8 for $n \geq 2{,}000$
where n = nominal rated engine speed in r/min



(Ref. IMO meeting BCH 23 in London)

The test methodologies to check emissions at certain engine loads have also been established. For ships built before 2000 that have not undergone extensive engine modifications or substitutions, the fuel sulphur limit is applicable. In short, each diesel engine with a shaft power in excess of 130 kW must have an Emission Certificate. The country under whose flag the ship sails or will sail is responsible for the inspection and certification of the diesel engine. So worldwide, each engine has an **'EIAPP': Engine International Air Pollution Prevention Certificate.**

Apart from the technical data, **'The Technical File'** should also state which engine parts could influence the NO_X emission values.
After the engine has been approved, these parts may not be replaced by other parts unless the substitutes have been tested and approved.

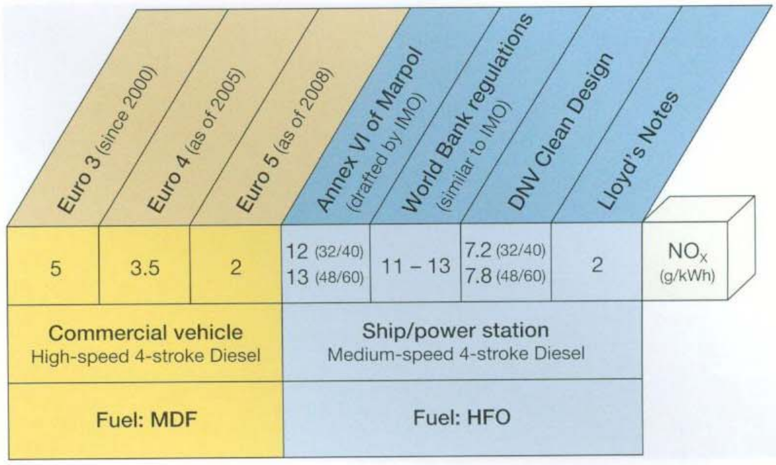
22.3.2 Additional legal regulations for diesel engines

There are regulations already in force with regard to diesel engines for road and inland shipping transport. These will become increasingly stringent in different phases.

The **Euro norms for road transport show** that the NO_X emission requirements have become increasingly strict between 2000 and 2008.

Other regulations, such as those stipulated by the **World Bank, Det Norske Veritas** or **Lloyd's,** underline that people worldwide are taking the nitrogen-oxide emissions increasingly serious. Proprietors of **diesel power plants,** predominantly built in developing countries, are obliged to comply with the World Bank emission requirements if they wish to borrow money.

In **Scandinavia,** there are local regulations that apply in coastal waters, for instance, fjords.



▲
The Euro-standards for road transport compared to the standards for ship propulsion and diesel power plants.

22.3.3 Important dates for the sulphur content in ship fuels

2005	May 19 Aug. 11	IMO EU	MARPOL Annex VI enters into force EU Sulphur Directive 1999/32 as amended by 2005/33 enters into force
2006	May 19 Aug. 11	IMO EU EU	Baltic SECA enters into force Baltic SECA enforced by EU directive 2005/33 Max.1,5% sulphur for passenger ships to/from EU ports (2005/33)
2007	Jan. 1 Aug. 11 Nov. 22	CARB EU IMO	Max 0,5% sulphur within 24 miles of California shore North Sea and English channel SECA enters into force North Sea and English channel SECA enters into force
2010	Jan. 1	EU EU CARB	Max. 0,1% sulphur bunker fuel in use at EU berths Max. 0,1% sulphur in all EU inland waterways Max. 0,1% sulphur within 24 miles of California shore
2012	Jan. 1	EU	Max 0,1% sulphur fuel in use by Greek ferries at Greek ports

Maximum sulphur content in ships fuels in the European Community

Fuel	S-%	When
Marine diesel oils	1.5	Aug. 11, 2006
Marine gas oils	0.1	Jan. 1, 2010

Maximum sulphur content in ship fuels per ship type in the European community following the EU directive 2005/33/EC

Ship type	Area	%	When
All	Baltic SECA	1.5	Aug. 11, 2006
All	North Sea + English Channel SECA	1.5	Aug. 11, 2007
All	All EU ports	0.1	Jan. 1, 2010
Passenger ships	All EU	1.5	Aug. 11, 2006
Inland waterway vessels	All EU inland waterways	0.1	Jan. 1, 2010

Combination of rules for fuels according to the Marpol Annex IV and the new EU directives

When	Ship type	Area	%	Act
May 19, 2006	All	Baltic SECA	1.5	Marpol
Aug.11, 2006	All	Baltic SECA	1.5	EU
Aug.11, 2006	Passenger ships	All EU	1.5	EU
Aug.11, 2007	All	NS+E C SECA	1.5	EU
Nov.22, 2007	All	NS+E C SECA	1.5	Marpol
Jan.1, 2010	All	All EU ports	0.1	EU
Jan.1, 2010	Inland waterway vessels	All EU inland waterways	0.1	EU
Jan.1, 2012	16 Greek ferries	Greek ports	0.1	EU

▲
A diesel power plant in Kenya using heavy fuel oil.

In countries as these, the rules for the emission of noxious substances are often inadequate or not present. When new power stations are now built, far more attention is paid to the environment and bankers such as the World Bank have their own requirements. Environmental measures cost a lot of money and therefore are often only accessible for the richer industrial countries.

22.4 Methods for the reduction of exhaust-gas emissions

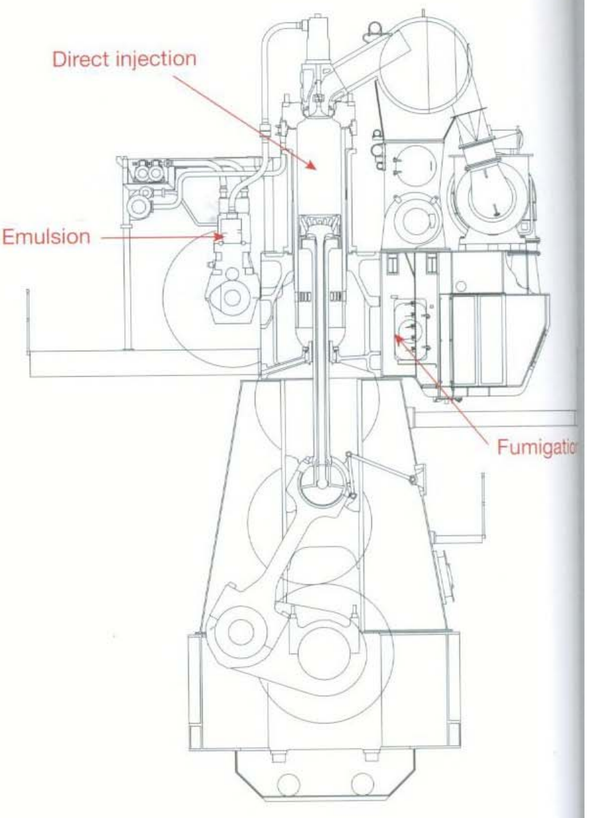
We distinguish **primary** and **secondary** methods.

Primary methods deal with the combustion space and are therefore referred to as combustion-technique improvements.
Secondary methods deal with the exhaust gas and are referred to as exhaust-gas treatment.

Primary methods

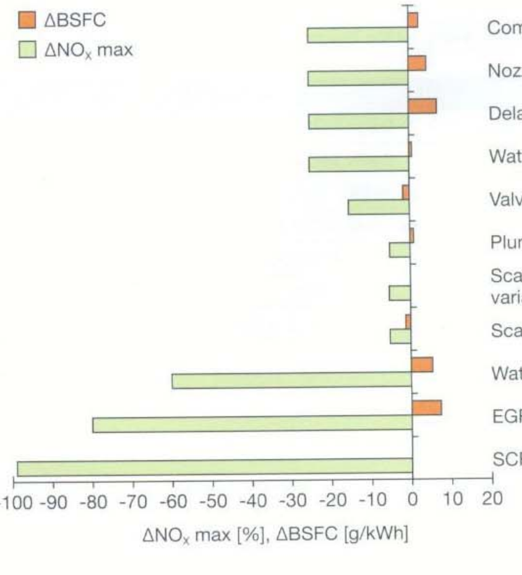
22.4.1 Nitrogen oxides – NO_X

The quantity of nitrogen oxides produced is dependent on the gas temperature in the combustion chamber. The higher the gas temperature the greater the quantity of NO_X formed. Therefore, a lower gas temperature offers a solution. However, total efficiency is somewhat reduced by these lower process temperatures.
In the following figure, all the methods aimed at nitrogen-oxide reduction.



▲
Three possible ways to influence for the combustion process for large crosshead engines using water, ensuring that the maximum combustion temperatures decrease and therefore producing less nitrogen oxides.

method 1: mixing water with fuel to obtain an emulsion (emulsion)
method 2: humidification of the charge air (fumigation)
method 3: direct injection of water in the combustion space (direct injection)



▶
Methods to reduce nitrogen oxides.

green: decrease of the amount nitrogen oxides in the exhaust gases
red: increase of the brake specific fuel consumption

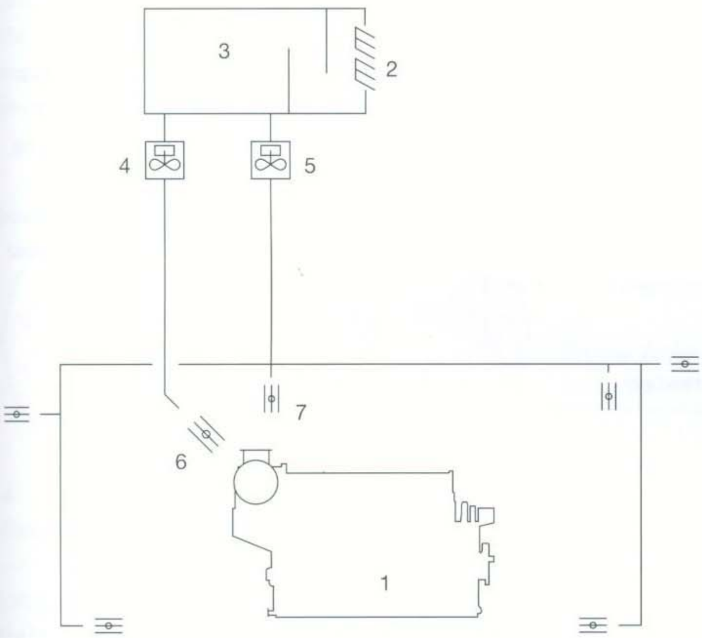
Important issues are:
- scavenging air temperature;
- Miller method for supercharging;
- turboblower cooling;
- scavenging-air pressure;
- valve timing;
- compression ratio;
- delayed fuel injection;
- injector-nozzle design;
- fuel-plunger dimensions;
- exhaust-gas recirculation;
- addition of water.

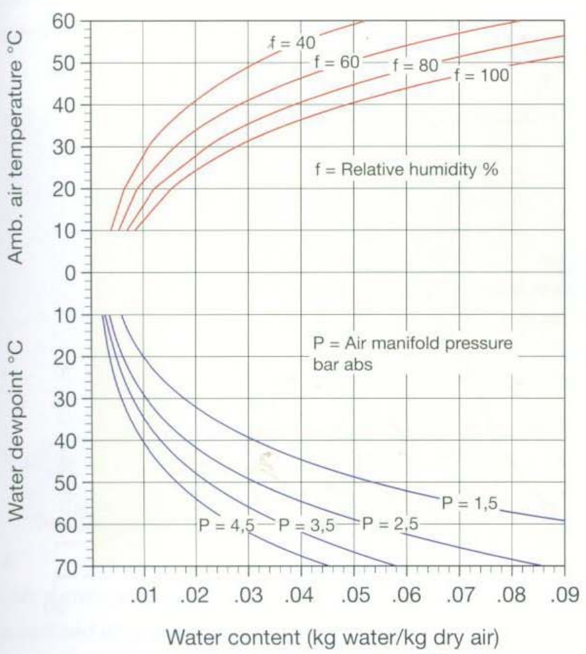
22.4.1.1 Decreasing the scavenging-air temperature and using the Miller-principle

Both entail the reduction of air temperature at the beginning of the compression stroke. Every 3 °C reduction in air temperature produces a nitrogen reduction of approximately 1%. One cannot normally cool below the dew point, as the water vapour will condense inhibiting a temperature reduction. In tropical conditions, the dew point is approximately 32 °C.

A simple NO_X reduction can be achieved by applying the Miller-process for four-stroke engines. In the **Miller-process, the inlet valve is**

Engine room ventilation (4V69E8169)





◀
Upper picture: Schematic arrangement for normal engine room ventilation on a ship.

1 *diesel engine*
2 *suction louver**
3 *water trap*
4 *combustion air fan*
5 *engine room ventilation fan*
6 *flap*
7 *outlets with flaps*

* *Recommended to be equipped with a filter for areas with dirty water (rivers, coastal areas, etc.)*

Lower picture: Graph of the condensation of steam in air coolers.

Example
- At an ambient air temperature of 35 °C and a relative humidity of 80%, the content of water in the air is 0.029 kg water/kg dry air. If the air manifold pressure (receiver pressure) under these conditions is 2.5 bar (= 3.5 bar absolute), the dewpoint will be 55 °C. If the air temperature in the air manifold is only 45 °C, the air can only contain 0,018 kg/kg. The difference, 0.011 kg/kg (0.029 – 0.018) will appear as condensed water.
- In an engine using 5 kilograms of air per second, according to the graph, 5 x 0.011 × 3600 = 198 kilograms water per hour is discharged. This is almost 5000 litres per day!

closed before the piston has reached bottom dead centre. The expansion of the scavenging air in the closed cylinder leads to a lower scavenging-air temperature!
The reduction of nitrogen-oxide emissions is remarkable: a reduction of 15 to 20% can be achieved.

22.4.1.2 Adjusting the scavenging-air pressure and valve timing

If excess air is reduced from, for instance, 2.2 to 1.8, the nitrogen-oxide reduction becomes noticeable, as there is less oxygen in the cylinder to react with nitrogen.
Additionally, the exhaust valve is closed earlier and the compression rate increased, to keep a constant ratio of combustion pressure to compression pressure.
The reduction of the nitrogen oxides can be as high as 15%. However, fuel consumption increases slightly.

22.4.1.3 Delayed fuel injection

An important factor in the production of NO_X is post-compression of the combusted gases. When fuel and air combust, the temperatures peak. If this gas is then further compressed before the piston reaches top dead centre, a considerable amount of extra NO_X will be produced.
This can only be resolved by delaying the fuel injection. This is in fact the best method to reduce nitrogen-oxide emissions.

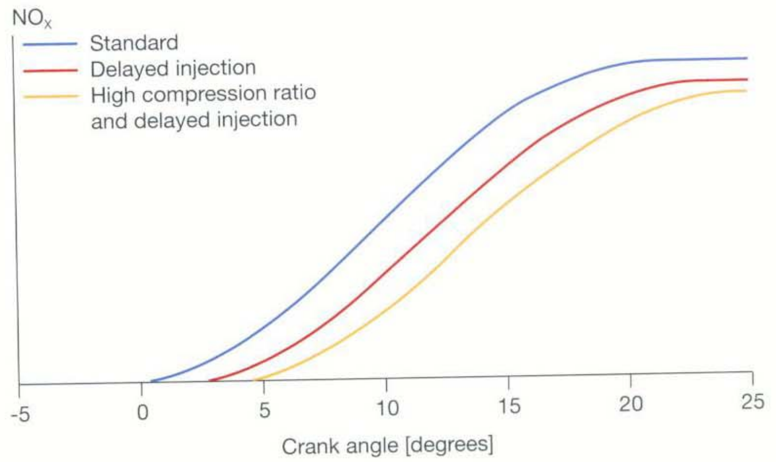
In principle, the delayed fuel injection warrants a lower maximum combustion pressure and slightly lowers engine efficiency. In two-stroke engines of Wärtsilä Sulzer RTA, a delayed fuel injection of two crank degrees produces a maximum combustion-pressure reduction of 10 bar, which in turn effects a 10% reduction in the production of NO_X and a fuel-consumption increase of 3 grams per kWh.
Delayed fuel injection can achieve a maximum NO_X reduction of 25%.

22.4.1.4 Increasing the compression ratio

The loss of engine efficiency caused by the fuel-injection delay can be compensated by increasing the compression rate.
In this way, the production of NO_X can be reduced by 2% and the air consumption will only increase by 1%.

The production of nitrogen oxides in relation to the crankshaft position with three different fuel injection methods.

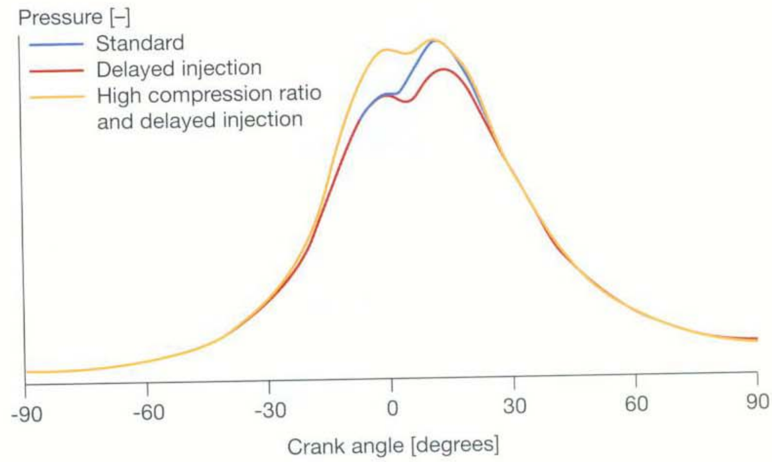
blue: standard injection method
purple: delayed injection method
yellow: delayed injection method and a high compression ratio
The last gives the lowest emission of nitrogen oxides.



If

The decrease in engine efficiency caused by the delayed fuel injection can be compensated by the increase of the compression ratio.

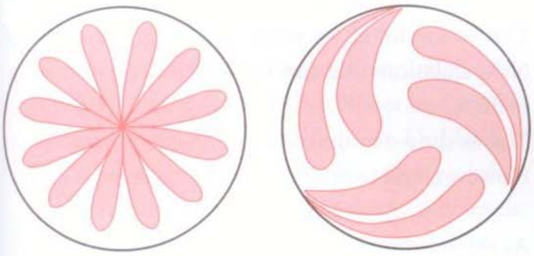
Therefore, NO_X production can be decreased by 2% and the specific air consumption only increases by 1%.



22.4.1.5 Adjusting the injector nozzles

The injector-nozzle design significantly affects combustion.
Nozzle-aperture diameter determines the injected-fuel velocity and consequently the depth of penetration of the fuel droplets in the high-pressure compressed air. This in turn is a determinant for the quality of the evaporation process, turbulence, mixing and fuel combustion.

In a four-stroke engine, the fuel is injected centrally into the relatively flat combustion chamber, related to the relatively small stroke diameter ratio of 1.1 to 1.2.
In a two-stroke engine, the fuel is injected from the external ring of the combustion chamber via two or three nozzles, positioned in a circular fashion around the rim. This can considerably influence the combustion process. Therefore, 'Low NO_X' nozzles have been designed which ensure slightly lower nitrogen-oxide emissions.



▲
Fuel Injection patterns.

Left: the injection pattern for a four-stroke engine shown from the heart of the cylinder and right: the injection pattern for a two-stroke crosshead engine from the periphery of the cylinder for three injectors.

▲
Left: a standard injector nozzle of a two-stroke crosshead engine and right the Low NO_X nozzle.

22.4.1.6 Modifying fuel-injection pressure and timing

Fuel injection can be adjusted by modifying the diameter of the injector-pump plunger or the nozzle-aperture surface area.
Both parameters influence the fuel-spray penetration, fuel-spray pattern in the cylinder and the turbulence generated in the combustion chamber.
Research has demonstrated that the effect on the NO_X production is disappointing. In the most favourable situation, a 5% reduction of the NO_X production can be achieved when prolonging the injection time by 10%.

22.4.1.7 Addition of water

A significant reduction of nitrogen oxides can be achieved by the addition of water to the combustion process. This can be carried out in one of three ways:

1 direct water injection in the combustion chamber;
2 atomised water in the scavenging air, thus increasing the scavenging air moisture content;
3 addition of water to the fuel, producing a fuel-water emulsion.

Engine designers have bruited concerns with the addition of water to the scavenging air. Mist separators extract excess moisture from the entering scavenging air in order to avoid damage of the lubricating-oil film on the cylinder liner. However, water droplets affect the lubricating-oil film rather than the moist air itself. In practice, the system for increasing the relative humidity is not used and moisture is extracted from the scavenging air.
To achieve a satisfying result with the addition of water, it must be injected in the combustion chamber at the correct time and place. This is the area where most NO_X is formed.
It is imperative that the water quality is acceptable. Generally, water is used that has been produced by the seawater evaporator heated by the H.T.-cooling-water system of the engine.

Direct injection of water in the cylinder

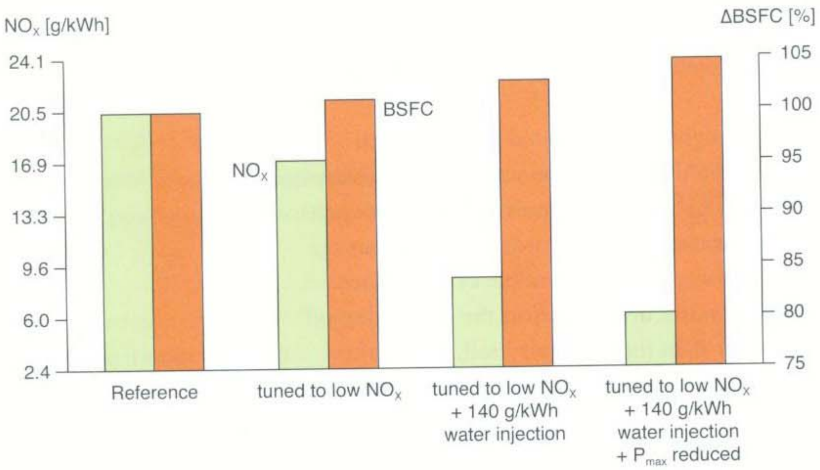
The water is injected by a completely independent injection system and monitored by an electronic control system. This allows huge amounts of water to be injected without compromising fuel injection to the engine, which would decrease power output. Moreover, the water-injection timing does not have to coincide with the moment of fuel injection.

▶
A fuel injector with two nozzles, with left: the water nozzle and right: the fuel nozzle.

Water injection takes place via the standard fuel valve, which has a combined water and fuel valve with a fuel nozzle and one for the water.
In two-stroke engines, results of a 50% NO_X reduction have been measured when injecting 140 grams of water per kWh.
For a crosshead engine of 30,000 kW
So this means that per hour $\frac{30{,}000 \times 140}{1000}$
4200 kilograms of water must be injected. A large storage tank and a good seawater evaporator are an absolute necessity!

The water is injected 60 crank degrees before T.D.C., far before the start of fuel injection. This commences approximately 20 crank degrees before T.D.C..
In the following chart is an overview of nitrogen-oxide reduction with various methods.

Therefore, there is no humidifying of the scavenging air.



▲
In this graph, another example showing the methods used for reducing nitrogen oxide emissions.

horizontal: three methods in respect to the standard engine. Far left in the graph, the reference
vertical: emission of NO_X in grams per kilowatt-hour

The green column represents the NO_X levels; the red column represents the increase in specific fuel consumption (BSFC) in percentages.

Conclusion: A large decrease in the amount nitrogen oxide results in a visible increase in the fuel consumption.

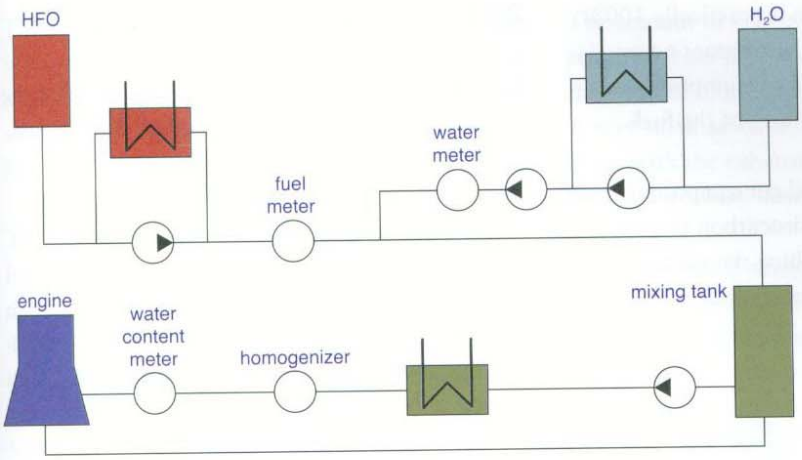
Adding water to fuel oil
This is a well-known technique in the reduction of NO_X emissions and can reduce NO_X emissions up to 50%.
This is shown in the figure on the next page.
However, fuel consumption may increase slightly.

As the fuel pump supplies extra water to the cylinder, the maximum amount of water supplied is dependent on the pump capacity. If the pump capacity is insufficient, water can displace the fuel, which will lead to a power-output decrease!
It is therefore possible that the camshaft-driven fuel-injection system requires adjustments.
It follows that the injector nozzle must also be reconfigured, as the amount of liquid pumped through it will increase. This results in more or larger apertures in the nozzle.

22.4.1.8 Exhaust-gas recirculation
This is a very efficient method for the reduction of NO_X emissions. It is generally only used in road transport engines.

This system is based on three factors.

1 The oxygen content in the exhaust gases which is reduced by the combustion process, causes, when re-circulated, an increase in the amount of gas that requires heating before combustion. This results in a decrease in combustion temperatures, which in turn leads to a reduction in the formation of NO_X. The



An installation for the supply of water before the engine.

average process temperature is not affected if the recirculating gas is cooled.
Engine efficiency does not decline when the oxygen content in the combustion process is decreased.

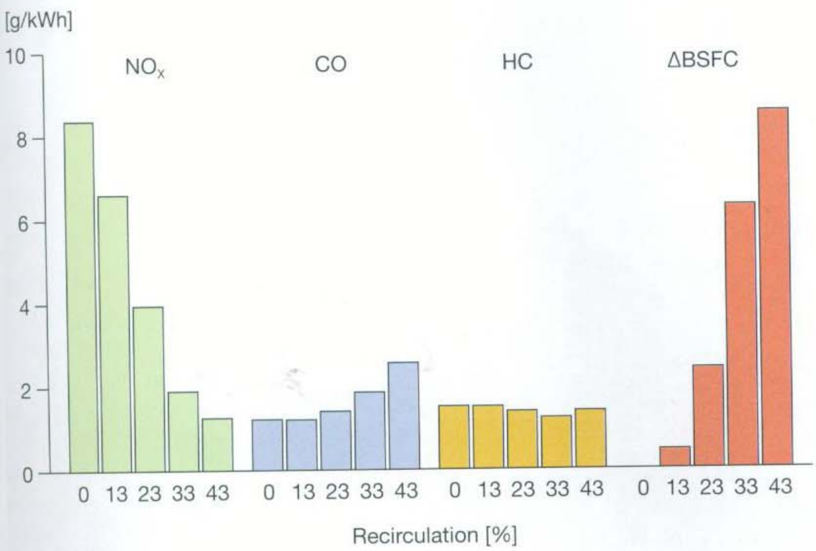
2 The reduced oxygen content lessens the reaction between the nitrogen and oxygen, which form nitrogen oxide (NO), and therefore reduces the NO_X emissions.

3 Inert components, such as water and carbon dioxide, which recirculate to the cylinder, are reheated during combustion. Water and carbon dioxide absorb more heat than air. This leads to an increase in the total heat capacity of the gas that is compressed and consequently levels local peak temperatures in the process. This, naturally, reduces the production of nitrogen oxides.

This system is not used in two-stroke crosshead engines as the exhaust-gas pressures before the exhaust-gas turbine of the turboblower are always lower than the scavenging-air pressure after the air compressor of the turboblower due to cylinder scavenging. Moreover, the exhaust gas of engines running on heavy-fuel oil is too polluted and would therefore contaminate the engine too quickly.
In fact, the recirculation method can only be applied to four-stroke engines using relatively clean fuels, such as light diesel oil, petrol and natural gas and is frequently used in Otto engines.

22.4.2 Hydrocarbon particles and other dirt particles

Due to improvements to the diesel process aimed at achieving low specific fuel consumption, uncombusted hydrocarbon particles have virtually disappeared.



A graph clearly showing the effect of the partial recirculation of exhaust gases.

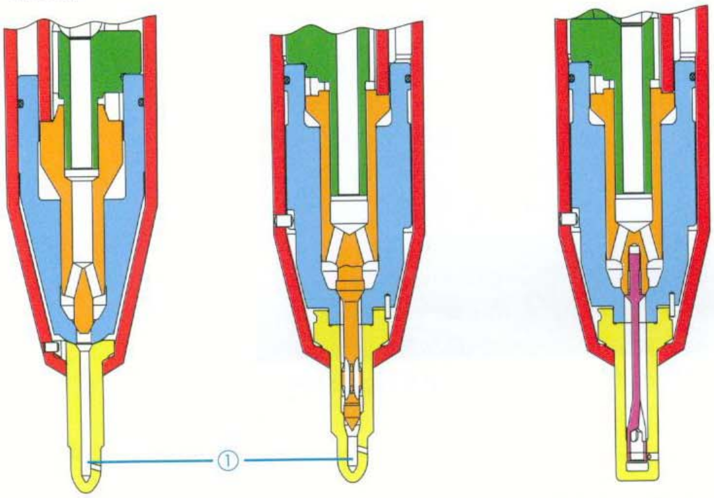
green: NO_X decrease at different recirculation percentages
blue: carbon monoxide increase
yellow: hydrocarbon decrease
red: specific fuel consumption increase

▼

Decreasing the sac volume under the injection needle can reduce the emission of uncombusted hydrocarbons.

This means that combustion is practically 100%. The hydrocarbons in the exhaust gases occur mainly due to lubricating-oil consumption and fuel evaporation in the apertures of the fuel-injector nozzle.
As the current lubricating-oil consumption already is very low, reducing the hydrocarbon content can only be achieved by diminishing the volume of the injector-nozzle space after the injector needle.

1 sac volume (space below the injector needle)



Conventional fuel valve
Sac volume 1690 mm^3

Mini-sac valve
Sac volume 520 mm^3

Slide-type fuel valve
Sac volume volume 0 mm^3

22.5 Secondary methods

After all the primary means are exhausted and the emission norms are still not attained, the exhaust gases can be post-treated.
The following methods have been in long-term use.

22.5.1 Catalytic converter

Oxidation of hydrocarbon compounds and reduction of nitrogen oxides.

22.5.2 Thermal reactors

For hydrocarbon compounds and carbon monoxide.

22.5.3 Systems and filters for the interception of fine particles

Methods used are, amongst others, electrostatic and dry filters.

22.5.4 Selective catalytic reduction

In professional literature referred to as S.C.R. For diesel engines, this is the only method that adequately works, as opposed to Otto engines where several systems work satisfactorily.

▶

In the near future, progressively more diesel power plants will use exhaust gas cleaning in order to meet the increasingly stringent requirements concerning the emission of noxious substances.

A power station using heavy fuel oil. This is clearly seen by the exhaust gasses.

This method involves the mixture of exhaust gases with ammonia, which subsequently flows through a labyrinth that has a specially coated surface area at temperatures between 300 and 400 °C.
Here NO_X is converted to N_2 and H_2O.

The nitrogen oxides are converted into the harmless compound, nitrogen and water vapour in accordance with the following chemical reaction:

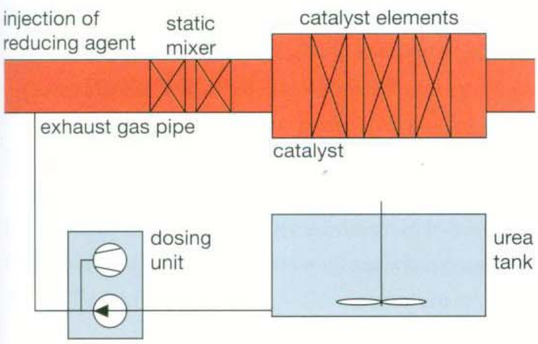
$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O$

$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12\,H_2O$

NO and NO_2 combine to form NO_x. NH_3 is ammonia.
Oxygen is necessary for this reaction and there is sufficient oxygen in exhaust gases.

The temperature at which this reaction occurs is important. Excessively high exhaust-gas temperatures combust ammonia before it can react with NO and NO_2. Low exhaust-gas temperatures produce a partial reaction and the formation of ammonium sulphates, which damage the catalyst.

The amount of ammonia used in the system depends on the amount of nitrogen oxides produced by the engine, which in turn is dependent on the engine load. A process computer controls the entire system and immediately responds to changes, for instance, more engine power means the injection of extra ammonia.

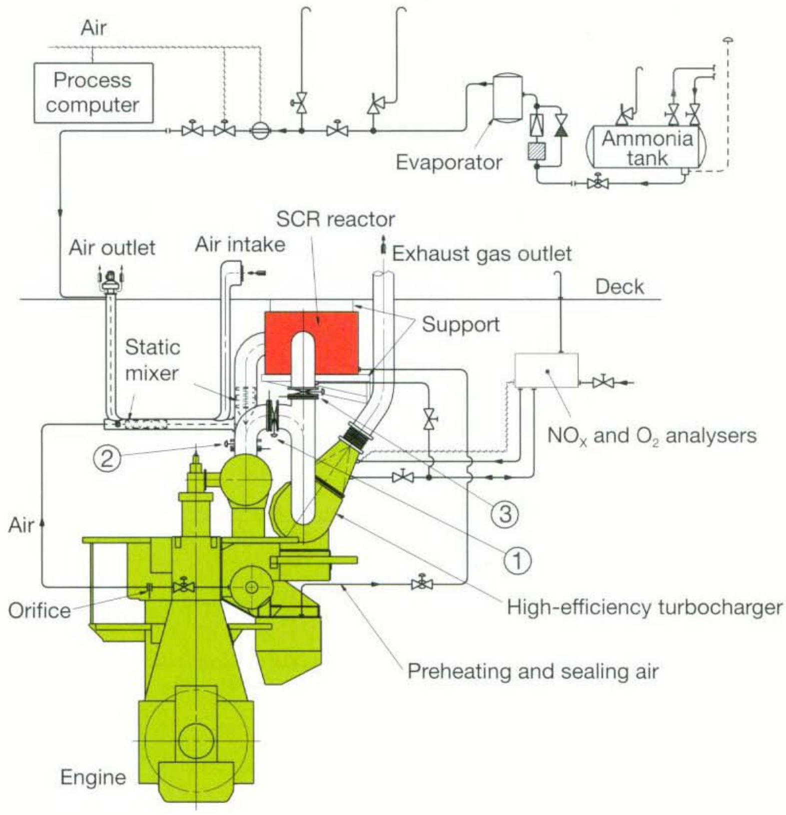


▲
Cleansing the exhaust gasses by the injection of ammonia to decrease the nitrogen content.

The amount of nitrogen oxide that can be removed from the exhaust gases is dependent on the amount of ammonia injected.
A very small amount of ammonia is not used and flows with the exhaust gases into the system. It, for instance, may flow through an exhaust-gas boiler.
When too much ammonia is injected, the unused ammonia may react with sulphur trioxide (SO_3) found in the exhaust gases and subsequently contaminate the heating surface of the waste-heat boiler with ammonium sulphates.

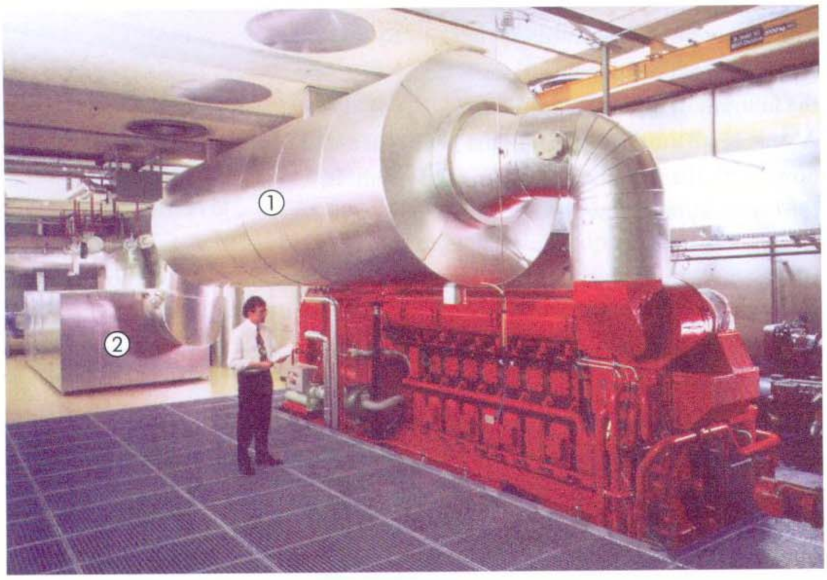
Ammonia can be:

- liquid;
- water-free under pressure;
- mixed with water under atmospheric pressure;
- urea, a dry product and dissolved in water before use.



▲
A large two-stroke crosshead engine provided with a complete system for exhaust gas cleaning.

At present, this system is seldom applied in diesel power plants. On ships, this system is never used, as the required space is too large.



▲

A S.C.R., (Selective Catalytic Reduction) installation in a diesel power plant in Zurich, Switzerland.

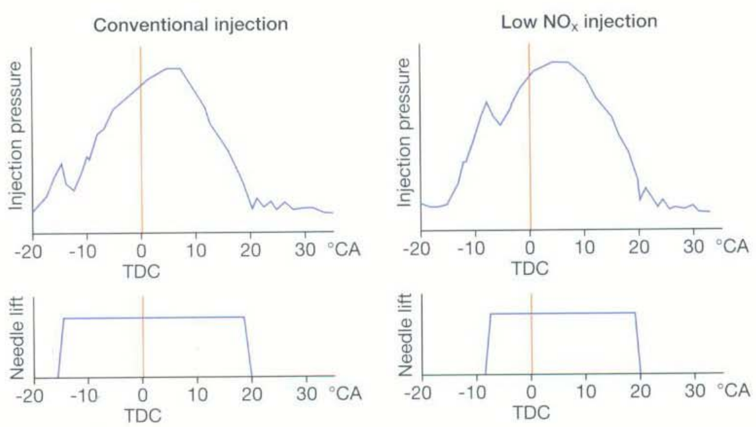
Note the dimensions of the complete upper installation (1 – sound damper) and behind (2 – S.C.R.) the diesel engine generator set.

▼

A delayed fuel injection ensures lower process temperatures and therefore lower nitrogen content in the exhaust gases.

horizontal: position of the crankshaft TDC = top dead centre

For conventional injection, needle lift is 35 crank degrees. Needle lift at delayed injection is 28 crank degrees.



In these systems, nitrogen oxides can be reduced by 80 to 90%.

Engine manufacturer Wärtsilä claims a significant nitrogen-oxide reduction in a Wärtsilä Vasa 32 using:
- delayed fuel injection: 20 to 30%;
- direct water injection: 50%;
- exhaust-gas recirculation: 50%;
- S.C.R.-systems: 80 to 90%.

22.6 Reduction of sulphur oxides in exhaust gases

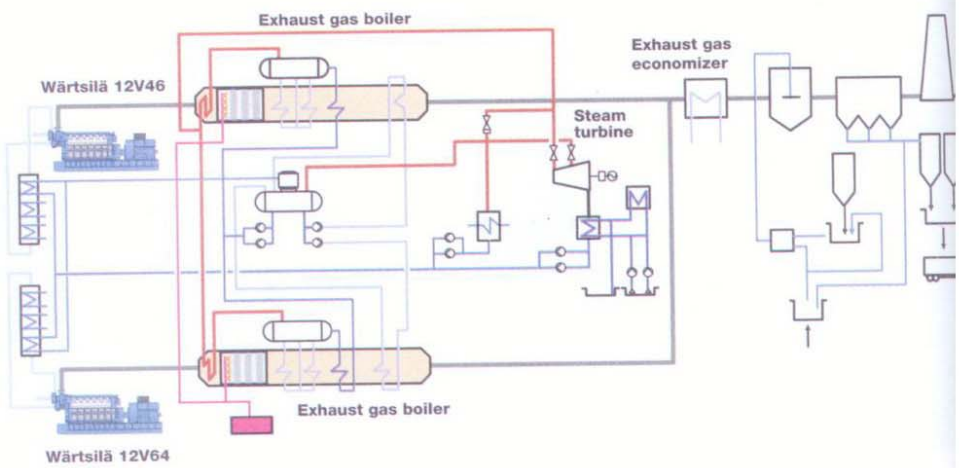
Reducing sulphur oxides is technically simple and is used in diesel-power plants. However, this has not been translated for use on board ships, as desulphurisation plants are very large and costly.

It is easier and less expensive for ships to reduce the sulphur content in fuel than decrease the amount of sulphur oxides in exhaust gases.

▼

A project for the supply of district heating in Vaasa, Finland.

Two twelve-cylinder medium-speed four-stroke V-engines with a cylinder bore of 46 and 64 centimetres generate electricity for the grid using an electrical generator. The heat from the exhaust gasses is used by two exhaust gas boilers to produce steam for a steam turbine. The spent steam of the steam turbine condenses in the heat exchanger for city heating; a very large piping network throughout the city provides buildings with warmth. The exhaust gases are cleaned in the NOx- and the SO_X systems. The normal heat discharge of the both engines is used as much as possible for heating purposes, shown here, the heating of the water supply to the boilers.



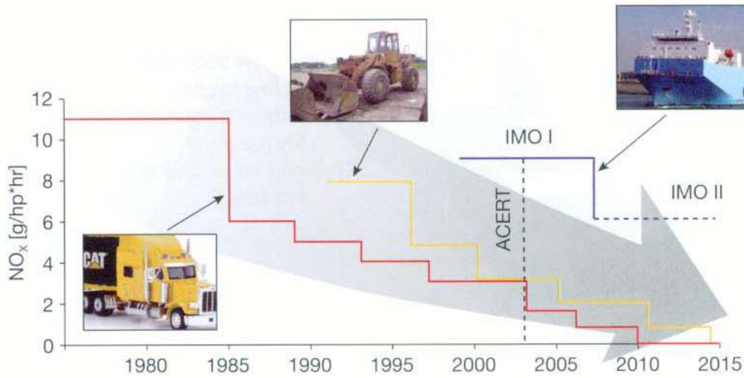
Sulphur oxides cannot be removed by catalysts. They can only be washed out. This is usually accomplished with alkaline solutions, such as calcium carbonate, sodium carbonate or caustic soda.

The reaction is as follows:

$CaCO_3 + SO_3 \rightarrow CaSO_4 + CO_2$

$CaCO_3 + SO_2 + \frac{1}{2}O_2 \rightarrow CaSO_4 + CO_2$

In this instance, calcium carbonate is used as a detergent and reacts with the SO_2/SO_3 to form gypsum (calcium sulphate).



▲

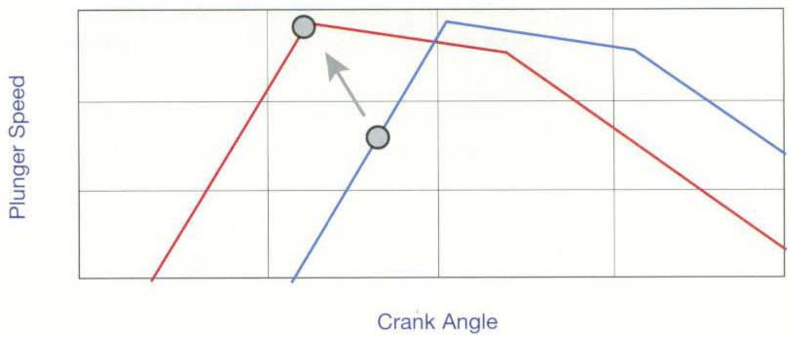
The 'Acert' technology of Caterpillar–MaK.

22.7 Removal of fine particles from exhaust gases

These minuscule dust particles often have diameters from 0.1 to 1.0 microns and cannot be removed in a cyclone, as opposed to sand grains used in desert areas for diesel-power plant air-inlet systems.

Electrostatic filters are generally used. In this system, the minuscule particles are ionised and adhere to the electrodes. Dust particles can then be washed away. The installations, however, are too bulky for ships and therefore are not used.

Tests have shown that a combination of a cyclone and an electrostatic filter removes over 60% of fine particles.



▲

The 'Flex Cam' technology of Caterpillar.

22.8 Examples of techniques engine manufacturers apply to reduce emissions

22.8.1 Engine manufacturer Caterpillar

Caterpillar developed for its medium-speed Caterpillar–MaK heavy-fuel engines, the so-called ACERT-technology.

This technology is characterised by three aspects:

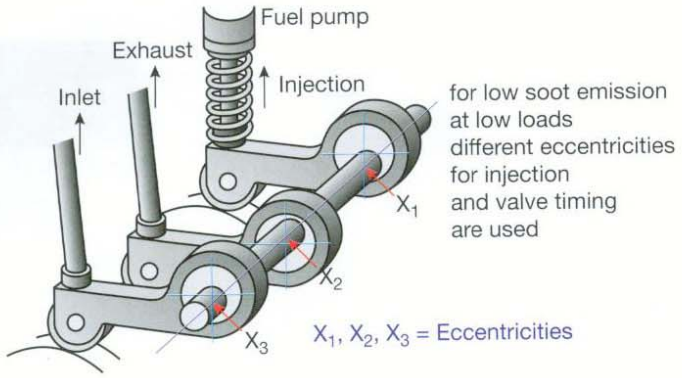
1 a reliable system when using heavy-fuel oil;
2 the highest engine efficiency in this power-output range compared with the competition;
3 the lowest engine emissions with the simplest possible system.

This is achieved with **Flex Cam-technology.** Here the inlet-valve timing and the fuel injection is adjusted to diesel-engine load.

With a mechanical adjustment device, the timing of the inlet valve and the fuel pump can be controlled. An actuator adjusts the driving gear.

This system is abbreviated: F.C.T. = Flex Cam Technology.

This first step entails the synergy between a flexible fuel system and an advanced air-supply system so attaining an optimum engine design. The fuel injection retains an extremely high pressure throughout the entire operational range while the amount of fuel and the timing of the inlet-valve opening are load-controlled. A lever system controls the injection timing and pressure, and the inlet-valve timing using an actuator.



▲
The mechanical valve and the fuel pump adjustments are made possible in Caterpillar-MaK diesel engines by the variable eccentricities X_1, X_2, and X_3.

The valve timing changes at partial loads, thus allowing effective compression to rise and consequently the maintenance of optimum combustion.
Furthermore, by modifying the fuel cam, the fuel-injection pressure increases. This method involves the fuel being sprayed in a series of finely timed microbursts in the engine-load range where it is usually difficult to prevent the production of smoke.

The **second step** has the following objectives: to reduce emission of exhaust gases below the current I.M.O. regulations and create a technology that complies with the emission requirements in vulnerable areas. Improve the mechanic adjustment system with additional built-in flexibility.

Other engine manufacturers are also improving the combustion process with the following achievements:
- reduced fuel consumption;
- reduced soot-particle emissions;
- lower nitrogen-oxide emissions.

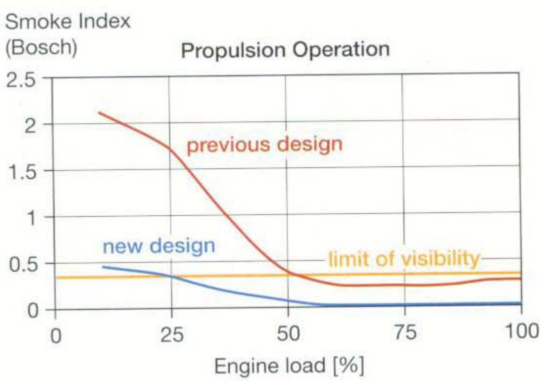
The graphs at the bottom of this page show improvements for medium-speed four-stroke diesel engines operating on heavy fuel by MAN–B&W, type 32/40.

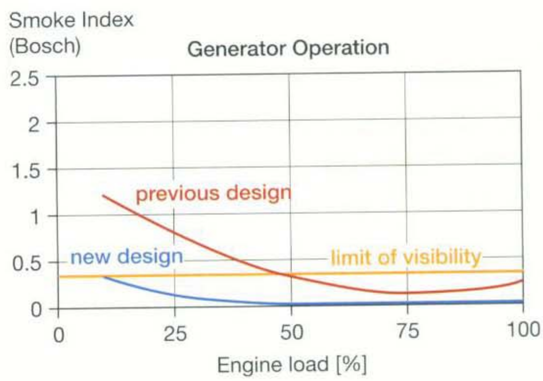
New fuel-injection techniques, such as the common rail, which is discussed in Chapter 9, Fuel-injection systems, provide a welcome contribution to the new fuel-injection techniques.

22.8.2 Engine manufacturer Wärtsilä

Engine manufacturer Wärtsilä has also taken various measures to reduce exhaust-gas pollutant emissions for its four- and two-stroke diesel engines.

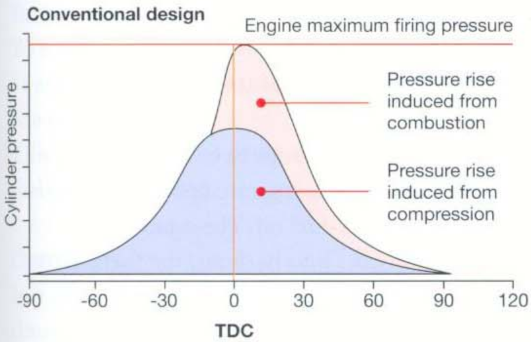
An advantage of medium-speed diesel engines is that they can operate on poor quality residues from the refining process, an inexpensive fuel source. The main disadvantage is that the pollutant emission means that extensive costly installations are required to restrict these emissions. **Ultimately, selecting fuel is a complex matter in which the financial aspect is the most important.**

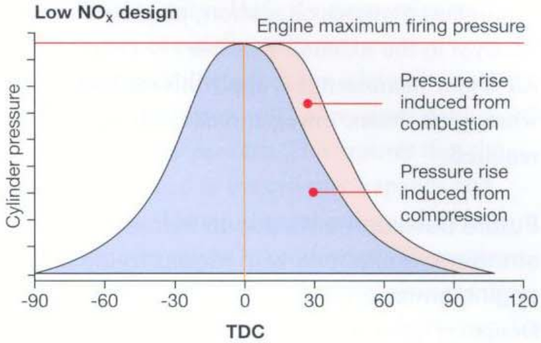




▲
In the graphs above, the decrease of smoke using new combustion technologies for MAN-B&W, type 32/40 diesel engines category III.

Some examples





◄ A very high compression ration ensures a decrease in the production of nitrogen oxides.

Example 1: Extremely high compression ratio

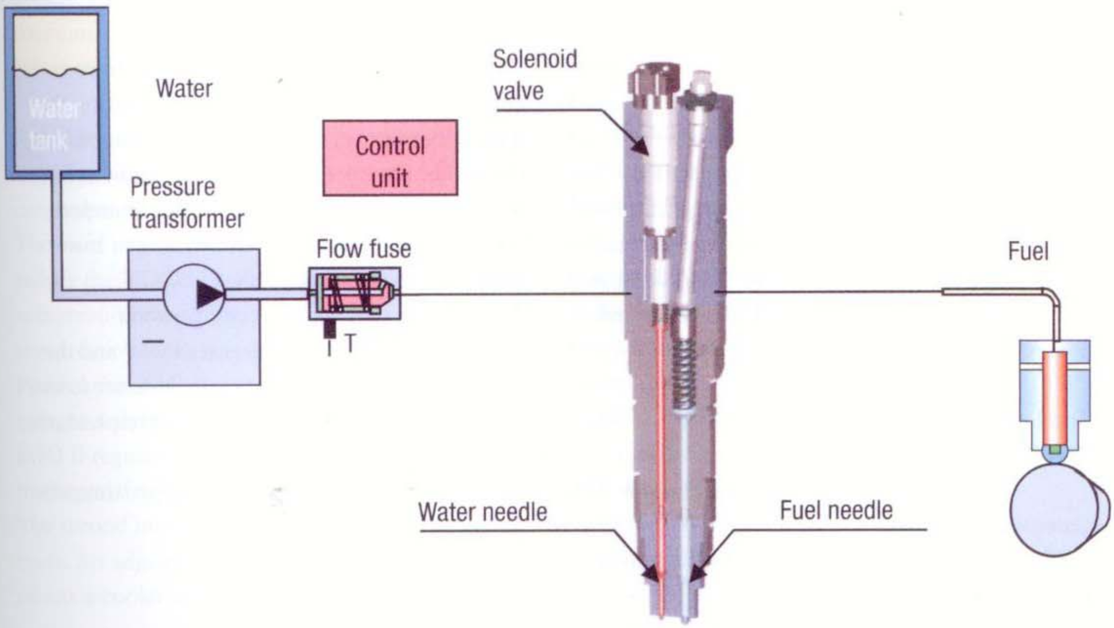
This entails that the final compression pressure is equal to the final combustion pressure. Therefore, the final compression temperature is considerably higher and the fuel combusts more rapidly. The so-called ignition delay, the time lapse between the beginning of the fuel injection into the cylinder and fuel ignition is shortened. There is, then, more time for the fuel to completely combust. As a result, the fuel injection can be delayed somewhat and combustion will occur at the most propitious time. The high peak temperatures in the combustion process are slightly lower, resulting in less nitrogen oxide in the exhaust gases.

Example 2: Water injection into the cylinder

As mentioned previously, this method comprises cooling of the combustion process by decreasing the peak temperatures. This, in turn leads to a decrease of nitrogen oxides in the exhaust gases.

▼ A technical packet for direct water injection in Wärtsilä diesel engines.

Important: The injected water must be very pure. The fuel injector has two functions: the injection of fuel and water. The water injection pressure is supplied by its own high-pressure water pump. The water flow is checked at every cylinder as a safety measure. A control unit provides the controls.



Example 3: Conversion of nitrogen oxides in exhaust gases using the Selective Catalytic Reduction method, or in short, placing a catalyst in the exhaust-gas line

All engine manufacturers apply this method whenever a drastic nitrogen-oxide reduction is required.

Future developments due to increasingly stringent regulations with regard to diesel engine emissions

Designing 'cleaner' engines means that engine manufacturers must invest time and money. A large number have to or will discontinue their business or merge. Consequently, the number of manufacturers of industrial diesel engines continues to dwindle.
The use of very heavy and contaminated fuels in certain areas will be further restricted and for a diesel oil fuel with a low sulphur content will be chosen.
The use of gas turbines running on relatively light diesel oil will increase on certain types of ships, such as cruise ships.
If heavy-fuel oil and therefore 'inexpensive' fuel is used, the secondary systems for cleaning exhaust gases will be used more frequently. In future, this will also apply to diesel-power plants.
The use of natural gas and other gases that have much lower carbon dioxide and nitrogen-oxide emissions will be used more often. The construction of natural-gas networks is on the increase in many countries. Gaseous fuels can be used for gas engines working on the Otto principle and for dual-fuel engines, which generally use gaseous fuel and small amounts of diesel oil.

Also, see Chapter 8, Fuels, fuel-line systems and cleaning fuels.

Total engine efficiency can still increase. The efficiencies in the different engine categories rise from 30% to over 50%. Efficiency increments save fuel, which in turn will effect the reduction of the hazardous emissions per kWh-generated power.

Total efficiencies 2008

Category I		
	0 – 100 kW	Maximum 35%
Category II		
	100 – 5000 kW	Maximum 44%
Category III		
	500 – 30,000 kW	Maximum 51%
Category IV		
	1500 – 100,000 kW	Maximum 54%

22.9 Measurements on board large modern sea going vessels

22.9.1 Tanks for sulphur-free fuel

In the last few years, ships have been manufactured with a separate fuel tank for sulphur-free heavy-fuel oil. The moment a seagoing ship goes into harbour; the fuel supply can be changed to this slightly more environmentally friendly fuel.

22.9.2 Electric connection

This is always installed on seagoing ships so that they can connect to the local grid while in port, for repairs and during inoperative periods. The available power, however, is small in comparison to the power required for harbour operations, (loading and, for instance, maintaining the correct temperature for containers/reefers requiring this). In 2008, large container ships with heavy shore connections will be built. This will guarantee the electricity provision for the operations in harbour. This means that all the diesel engines on board will be stopped, so no pollutants are emitted from the ship. This was first used in Los Angeles. It is known as the Alternative Maritime Power Programme (AMP).
Building electrical shore connections is costly and requires considerable on-board investments.

22.10 New techniques: the Miller process

The Miller-process principle is that in four-stroke engines, the inlet valve closes earlier during the inlet stroke, at approximately 20% time before the end of the inlet stroke. The air in the cylinder will expand and subsequently cool. Therefore, the compression stroke commences at a lower temperature, approximately 15 °C.
The result is that the final compression temperature is about 50 degrees lower and the final combustion temperature is therefore lower. It is clear that at these combustion temperatures, fewer nitrogen oxides form.
The NO_X reduction amounts to approximately 30%.

22.10.1 Adjustments.

Turboblower.

To provide the cylinder with a sufficient amount of air, a modified turboblower with a pressure of 5 bar is required.
This can be achieved using a two-stage turboblower system.

Camshaft.

Is it difficult to start the engine with the Miller timing and a large amount of soot is produced at partial load due to the turboblower-pressure being too low.
An adjustable camshaft allows adjustments to be made to valve closure and fuel injection during ignition and at partial load.
The Miller timing is then switched off so the engine operates normally and the soot emissions are reduced.

Tests with a Caterpillar Mak 7M43C have shown that the system can also be installed in existing engines. The costs in this case amount to 15 to 25% of a new engine.
The engine is then IMO phase II-norm compliant.

Example 1

In Rostock, the MaK 7M43C main engine of the chemical tanker Fure West (144 x 21 metre, 16,000 dwt) was converted into a Low-Emission Engine (LEE). It is the first ship with a heavy-fuel engine that complies with the future IMO phase II-emission norms. The Fure West is owned by the Swedish Furetank Rederi. The NO_X-emission at present amounts to 8.3 grams per kWh and the soot emission is less than 0.5 FSN (Filter Smoke Number).

According to Caterpillar Engines in Kiel, the engine achieves low emissions without a fuel-consumption increase.
The most important modification made to reduce the NO_X emission is the increase of the compression ratio. Ten years ago, the standard compression ratio was 1:11 or 12.
Prior to the IMO I-emission requirements, the ratio had already been reduced to 1:14 or 15 and IMO II regulations require compression ratios in the region 1:17.
The second important modification is the Miller cycle. An adjustment of the inlet-valve timing effects a cooler combustion process.
In order to achieve the IMO I-norm, a relatively small Miller effect of five per cent was required. However, to achieve IMO II-norms, requires a Miller effect of twenty per cent.
To attain this Miller effect, the inlet valve must close when the piston still has to move downwards a further twenty per cent. This ensures that the inlet air is locked in the cylinder, expands and further cools. Therefore combustion is less hot, which reduces the NO_X production. To supply a sufficient amount of air in the brief moment the valves open, requires an adjusted turboblower. The inlet air must be supplied at a pressure of five bar. Combining the higher compression ratios with the Miller effect, the NO_X emission is reduced by approximately thirty per cent while efficiency remains the same.

22.10.2 Camshaft technique

The main drawback of the Miller timing is difficult start-up and the engine produces a great deal of soot when operating at partial load, because the turboblower pressure is too low to provide the cylinders with sufficient air.
To prevent this, the LEE-system works with an adjustable camshaft. This 'Flexible Camshaft Technology' (FCT) allows valve-closing times and fuel injection during start and at partial load to be adjusted. Next, the Miller timing is switched off, the engine operates normally, and soot emissions are reduced. By accelerating the fuel injection and increasing the injection pressure, combustion is improved, also reducing soot emission.
According to MaK, the flexible-camshaft technique reduces soot emissions by 75 per cent, while the operational performance of the engine is also improved.
Essentially, the FCT-system comprises a modified camshaft with a pneumatically controlled actuator and Programmable Logic Controller (PLC).
Retrofitting and certifying the engine took four days. During these days, the engineers adjusted the compression ratios and the camshaft.
A higher compression ratio leads to a smaller combustion chamber as the piston is slightly raised, which means that the anti-polishing rings must be replaced by shorter rings.
Parts of the valve drive, high-pressure fuel pumps and the injector nozzles were also replaced.
ABB specialists modified the turboblower to meet the higher requirements. After the retrofit, trials were held and the engine was certified by Germanischer Lloyd.

The engine has operated on this new system for five months.
All existing Mak M20C, M25C, M32C and M43C engines can be adapted to meet the future IMO II-standards.
The retrofit costs amount to 15 to 25 per cent of new engine costs.

Example 2

The Wärtsilä Sulzer four-stroke medium-speed engines of the ZA40 type with a 400 mm bore and a 480 mm stroke, the ZA 40 A type with a 400 mm bore and a 560 mm stroke have been tested for important improvements regarding emission reduction.
Two systems have been installed for trials.

The dry system
An improved piston, injector nozzle, anti-polishing ring and a variable inlet-valve timing system.

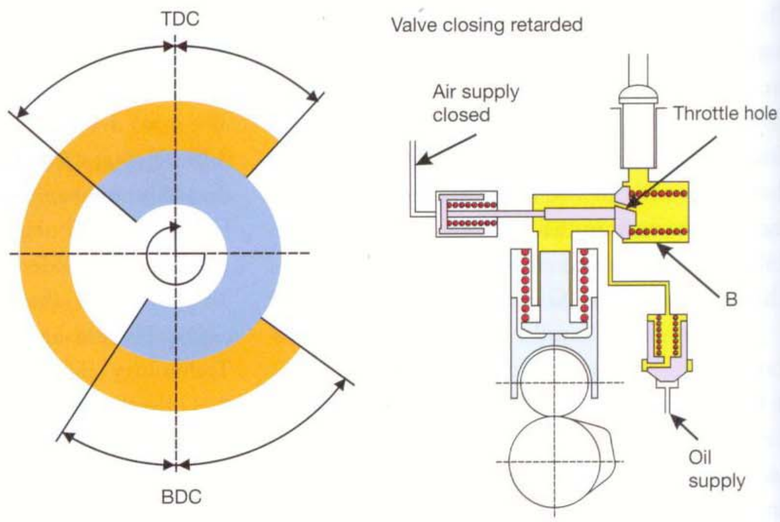
The wet system
Based on water- and heavy-fuel emulsions and is referred to as a micro-emulsion.

In the dry system, the Miller principle is incorporated with the so-called Variable Inlet Valve Closing or VIC-system.

The system works using a hydraulic system between cam roller and the push rod.

The mode of operation of the Miller principle for Wärtsilä Sulzer ZA 40 engines.

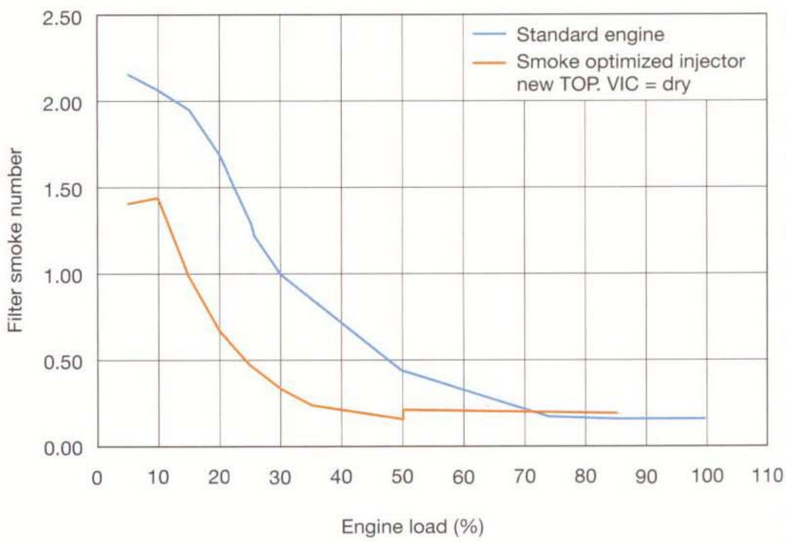
At conventional full engine load, the throttle valve B is open, the oil can flow freely, and the push rod moves according to the cam lobe and cam height.
At low loads, the throttle valve is closed by a control air system, which results in the closing of the inlet valve. The inlet valve is opened in the normal way by the increase in oil pressure and the inlet valve closes at the correct time, the bottom.
VIC is built on the engine and no external mechanical system is necessary.
A simple automation system is used for control air.
This can be simply placed in existing engines.



Delayed timing
for improved smoke behaviour at low load

Soot emissions of a standard engine compared to the 'dry' system of Wärtsilä for Sulzer ZA 40 medium-speed engines running on H.F.O..

Clearly shown are the large differences between zero loads to approximately 70%.
After this, there is no improvement!



Example 3

The Wärtsilä 32 series.

In this series, the objective was to reduce nitrogen-oxide emissions while retaining a high efficiency. This is difficult from a physics point of view as, in theory; an internal combustion engine achieves high efficiency with a high compression ratio and the highest possible maximum combustion temperature.

Carnot shows this.

$$\text{Efficiency Carnot} = \frac{T_1 - T_2}{T_2}$$

Where T_1 is the highest temperature in the process, that is, the maximum combustion temperature with T_2 being the lowest temperature, the exhaust-gas temperature.

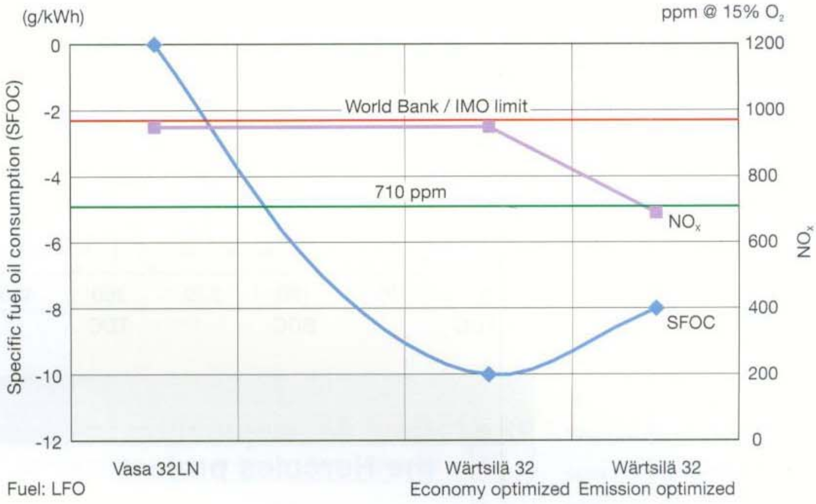
Improvements.

For the Wärtsilä 32 engines, this entailed the following.

- High compression ratio.
- New fuel-injection technique.
- New injector nozzle.
- New valve timing; the Miller principle.
- Increased turboblower pressure ratio.
- The SPEX charging-air system for both in-line and V-engines.

▶ Wärtsilä 32 performances.

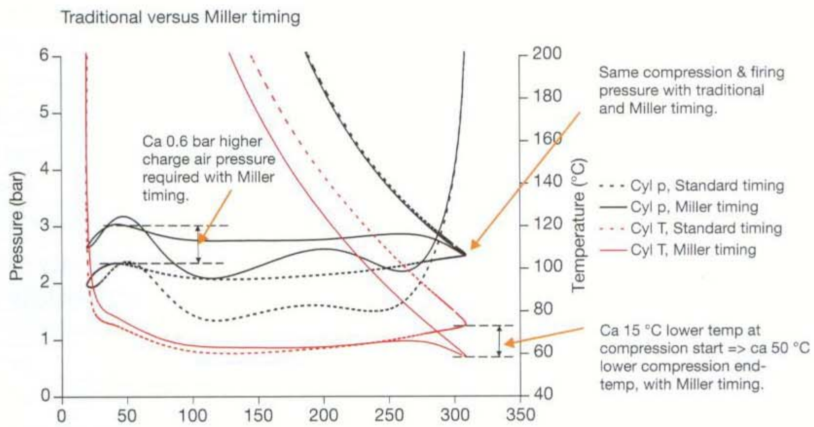
Clearly shown is that the fuel consumption is lowest in the 'economy' type and the NO_x emissions are lowest in the 'emission' type.



▶ A comparison of the traditional valve timing and Miller valve timing.

Horizontal: The number of crank degrees.
Vertical:
Left: The pre-charge pressure in bar.
Right: The air temperature.

Red dotted line: cylinder temperatures, traditional valve timing.
Black dotted line: cylinder pressures, traditional valve timing.
Red line: cylinder temperatures, Miller valve timing.
Black line: cylinder temperatures, Miller valve timing.



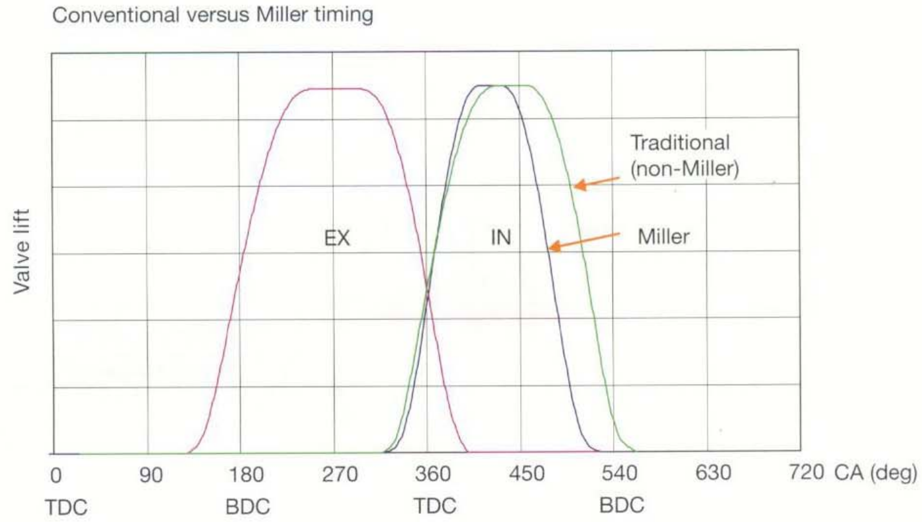
The graph shows that the pressure in the cylinder in the Miller system is higher than that of traditional valve timing. Obtaining this high pressure requires approximately a 0.6 bar higher pre-charge pressure.

It is also shown that the initial compression temperature for the Miller system is lower than that of the traditional system. This results in a 50 °C lower final compression temperature.

The difference in the valve timing of the inlet valve between the traditional system and the Miller system.

Horizontal: The number of crank degrees.
TDC – Top Dead Centre.
BDC – Bottom Dead Centre.

The graph shows that with the Miller system, the inlet valve closes earlier, just before the bottom dead centre (BDC) of the piston.



22.11 New developments: the Hercules project

This is an extensive project with as objective; further optimisation of the modern internal combustion engine.
A large number of companies and institutes are involved in this project; each has their own expertise in the engine field. Collectively the participants have approximately 80% of global turnover in the medium-speed four-stroke and low-speed two-stroke crosshead engines branch. The project is supported by the European Commission and the Swiss authorities.

Hercules is the abbreviation for High Efficiency R & D on Combustion with Ultra Low Emissions for Ships.

A few project objectives

- Drastic reduction of pollutant emissions and dirt particles in exhaust gases.
- Improvements in the total efficiency and reliability of the engine. This reduces fuel consumption and at the same tine decreases the carbon-dioxide emissions. Maintenance costs are also reduced. This project should be ready between 2010 and 2020.

Clearly, this requires technical improvements to the engine.

- Extremely high cylinder pressures.
- Improved combustion.
- Special turboblowers with several pressure steps.
- A diesel engine with an exhaust-gas boiler producing steam for the steam-turbine genset.
- Emission-reduction systems with water in the combustion chamber.
- Emission-reduction systems with exhaust-gas recirculation.
- Emission-reduction systems using post-treatment outside the engine.
- Reduction of the driving-gear friction in the engine.
- A diesel engine with elaborate electronic controls for optimal combustion-process control.

Hercules Vision	Year 2010	Year 2020
Reduction of fuel consumption and CO_2 emissions	-3%	-5%
Reduction of NO_X (Relative to IMO 2000 standard)	-30%	-60%
Reduction of other emission components (PM, HC)	-20%	-40%
Improvement in engine reliability	+20%	+40%
Reduction of time to market	-15%	-25%
Reduction in lifecycle cost	-10%	-20%

◄ The Hercules project goals to 2020.

Hercules Objective	BAT-IS (2003)	Year 2007 Targets	Year 2010 Targets
Reduction of fuel consumption and CO_2 emissions	2-stroke: 170 g/kWh 4-stroke: 175 g/kWh	-1%	-3%
Reduction of NO_X (Relative to IMO 2000 standard)	IMO 2000 limits (g/kWh) 17 N < 130 rpm 45 x (rpm)-0.2 130 < N < 2000 rpm 9.8 N > 2000 rpm	-20%	-30%
Reduction of other emission components (PM, HC)	< No limits for marine engines > Visible smoke limit FSN 1.1 Opacity 20%	-5%	-20%
Improvement in engine reliability	18,000 hours to overhaul of major components	+10%	+20%
Reduction of time to market	5 Years	-10%	-15%
Reduction in lifecycle cost - Initial cost - Fuel/lub-oil cost - Maintenance	< Costs depend on engine size >	0% -1% -4%	-1% -3% -6%

◄ The improvements to 2010.

Potential Exploitation Items		
Workpackage	Item	Time
WP1: Extreme design parameters	Engine components for extreme output operation (pistons, rings, bearings)	M
	Extreme value engine	L
WP 2: Advanced Combustion Concepts	Combustion models	S
	Chemical kinetics models	S
	Full cycle simulation tools	S-M
WP 3: Multistage/Intelligent turbo-charging	Variable geometry turbocharger	M
	Power take-in, take-out systems (Integration motor/generator/turbocharger)	M
	Multistage intercooled turbocharger	M
WP4: Turbo-compound / hot engine	Composite structures for hot-engine	L
	Engine compounding systems and components (boilers, TG, TCS)	S
WP6: Emission reduction methods (internal-water)	Direct water injection system	M
	Inlet air humidification system	S
	Control systems for above	S
WP7: Emission reduction methods (internal-Exh. Gas)	Exhaust gas recirculation system	M
	PM measuring techniques	S
WP8: Emissions after treatment	In-service emissions monitoring system	S
	Non-Thermal Plasma Technology	L
	Wet Scrubber Technology	M
	Select-cylinder emission measurement technology	M
WP9: Reduced friction engine	Low friction engine components (liner, pistons, rings, bearings, injection)	M
	In-service monitoring system for cylinder and lub feed rate adjustment	S
	Low friction engine	M
WP11 Adaptive engine	Onboard engine electronics	S-M

Timespan: Short = S, Medium = M, Long = L

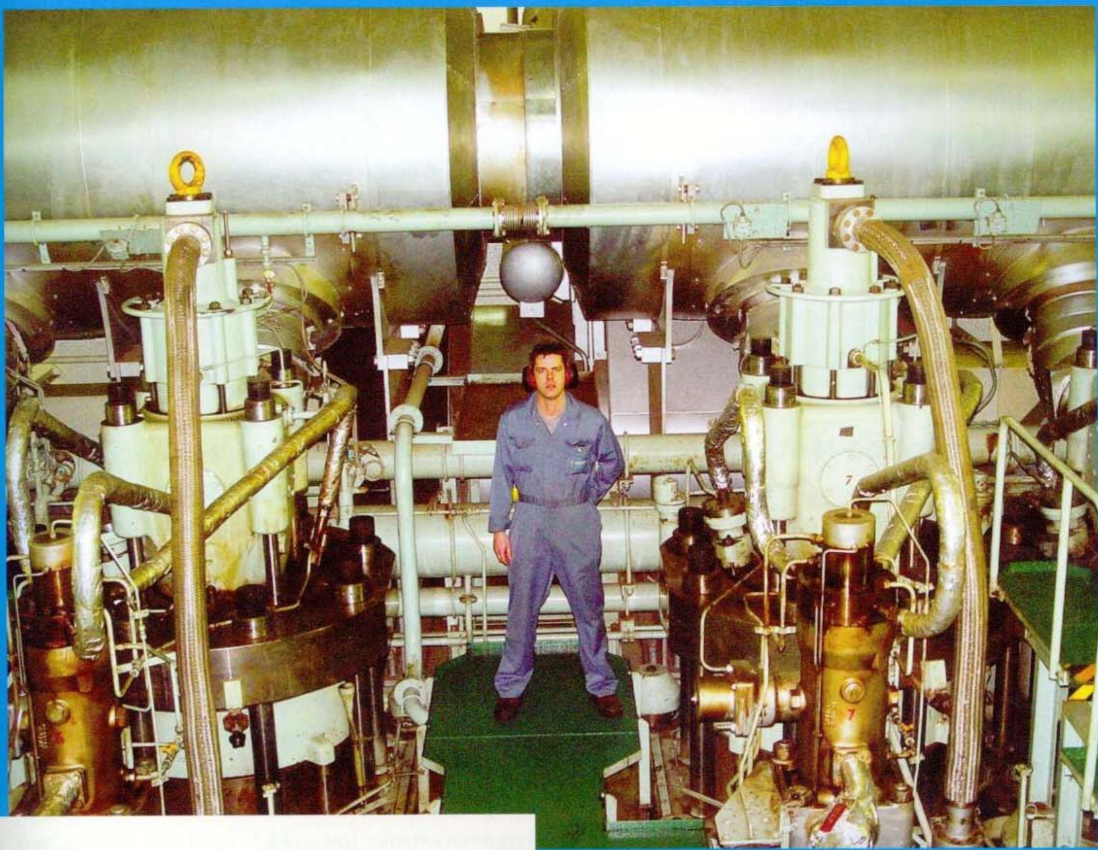
◄ The ten items chosen for specialised studies by separate groups.

>CH 23

Calculating fuel and lubricating-oil consumption







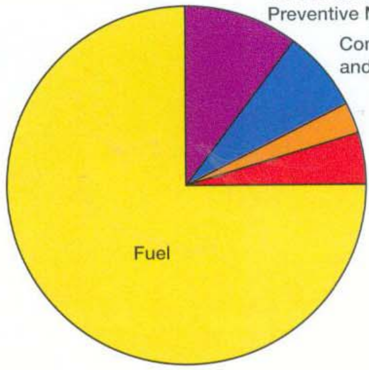
Fuel and lubricating-oil consumption of larger engines with a large number of running hours forms the most important cost factor. The fuel and lubricating-oil consumption of this two-stroke RTA 96 C crosshead engine in a large container ship is huge and costs millions of euros each year.

The annual fuel and lubrication costs for this luxury yacht with few operating hours are negligible.

23.1 Introduction

Fuel and lubricating-oil consumption involves high costs. The fuel and lubricating-oil costs of ships or diesel-driven power plants constitute between 40 and 70% of the total exploitation costs. Obviously, this is dependent on, for instance, the number of operating hours, engine category, and depreciation (writing-off) of the plant, personnel expenses, maintenance, and insurance.

Naturally, harbour dues, towing and pilotage costs are also important. Costs involved for ships sailing through the Suez or Panama canals amount to € 100,000 or more (one-way only!).



▲

An overview from Caterpillar.

The total running costs of a diesel engine in its life-span of between 10 and 25 years.
Here fuel costs alone amount to 75%!

23.2 Diesel-engine efficiency

Dependent on engine type and the cylinder volume, the efficiency varies from ± 25 to 50%.
Small four-stroke high-speed diesel engines in category I have an efficiency of over 30%, and the large two-stroke low-speed crosshead engines have an efficiency of over 50%.

Efficiency is equal to: $\dfrac{\text{Shaft power}}{\text{Fuel power}}$

23.3 Shaft power in kW or MW

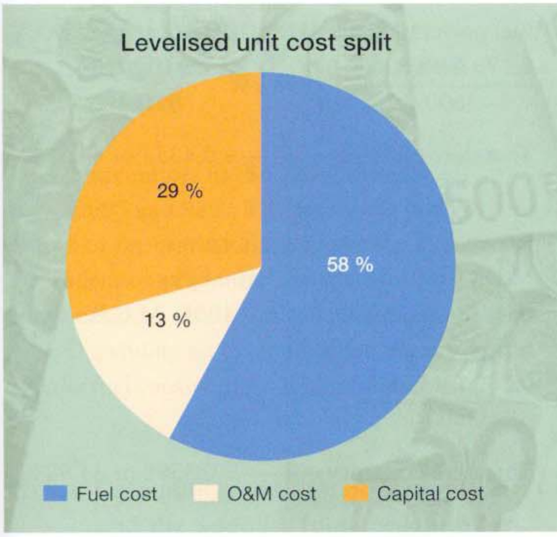
Shaft power can be measured with a torsion meter placed on the output shaft. It can also be measured with the engine formula where the mean induced pressure is measured by an indicator.

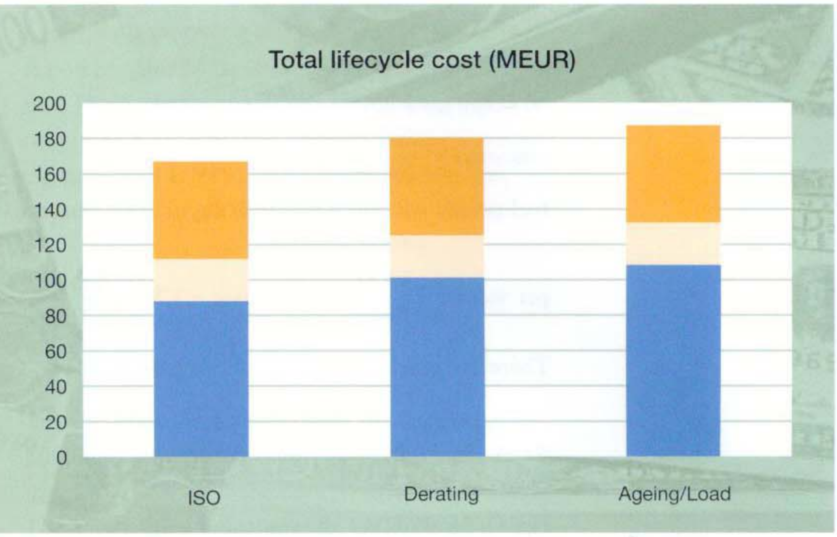
▼

Over-all shipping costs.

There are many cost factors in shipping:
- interest and amortisation on purchase costs;
- insurance;
- maintenance and repairs;
- quay and pilot dues;
- fuel and lubricating oil;
- canal and locks dues;
- crew;
- loading/discharging.

One cost factor is considerable: the cost of fuel and lubricants. This is the largest cost factor for ships that travel long distances; between 40 and 70% of all costs.





▲ An overview of Wärtsilä.

The total costs of a diesel power plant. The fuel costs are 58%.
O&M is operation and maintenance costs.

At present, this is performed with an electronic measurement system where all the data from the engine formula are directly processed to an effective cylinder output and consequently the shaft power for all the cylinders, so the entire engine.
In practice, these mean induced pressures are measured for engines in categories III and IV.

The fuel power is equal to: mass fuel in kg/sec × calorific value of the fuel in kJ/kg.

$$\text{Fuel power} = \frac{\text{kg}}{\text{sec}} \times \frac{\text{kJ}}{\text{kg}} = \frac{\text{kJ}}{\text{sec}} = \text{kW}$$

The amount of energy supplied to the engine in fuel form is set at 100%. The shaft power therefore always plays a part in this.

23.4 Specific fuel consumption

At present, this is the standard unit for fuel consumption and is indicated by grams per kW hour, or **g/kWh.**

Some values

The specific fuel consumption varies from approximately 240 to 160 grams per kWh at 100% load or Maximum Continuous Rating (M.C.R.), depending on size and type of engine. **The calorific value of H.F.O. is set at 42.700 kJ/kg and that of M.D.O. at 43.500 kJ/kg.**
Manufacturers of large engines usually provide a calorific value of 42.700 kJ/kg for H.F.O.. This is in accordance with ISO 3046-1 lower calorific value, without engine-driven pumps. The tolerance is +/- 5% and is valid at an 85% load for propulsion engines according to the propeller law. Obviously, in practice, the calorific value stated on the bunkering bill or the fuel specifications should be used.

◄ A Lemag electronic pressure sensor, the Premet XL.

1 *pressure sensor*
2 *data storage*
3 *electric-power plug*
4 *pc data cable*

Example 1 – Diesel engine category IV – MAN- B&W K98 MC ± 80% of M.C.R.

be = 160 g/e kWh. H.F.O. Fixed screw.

$$\text{efficiency} = \frac{\text{shaft power}}{\text{fuel power}}$$

fuel power = 0.160 × 42,700
= 6832 kJ/hour

$$\text{per second} = \frac{6832}{3600} = 1.97 \text{ kJ/sec} = 1.897 \text{ kW}$$

$$\text{Therefore efficiency } \frac{1}{1.897} = 0.5271 \text{ or } 52.71\%$$

Example 2 – Diesel engine category III – Wärtsilä 38 B

H.F.O. 85% of M.C.R. controllable-pitch propeller
be = 173 g/e kWh.
fuel power per second =

$$\frac{0.173 \times 42,700}{3600} = 2.051 \text{ kW}$$

$$\text{Therefore efficiency } \frac{1}{2.051} = 0.4875 \text{ or } 48.75\%$$

Example 3 – Diesel engine category II

M.D.O. calorific value 43,500 kJ/kg
be = 190 g/e kWh, genset at 100% M.C.R.

fuel power per second =

$$\frac{0.190 \times 43,500}{3600} = 2.2958 \text{ kW}$$

$$\text{Therefore efficiency } \frac{1}{2.2958} = 0.4355 \text{ or } 43.55\%$$

Example 4 – Diesel engine category I

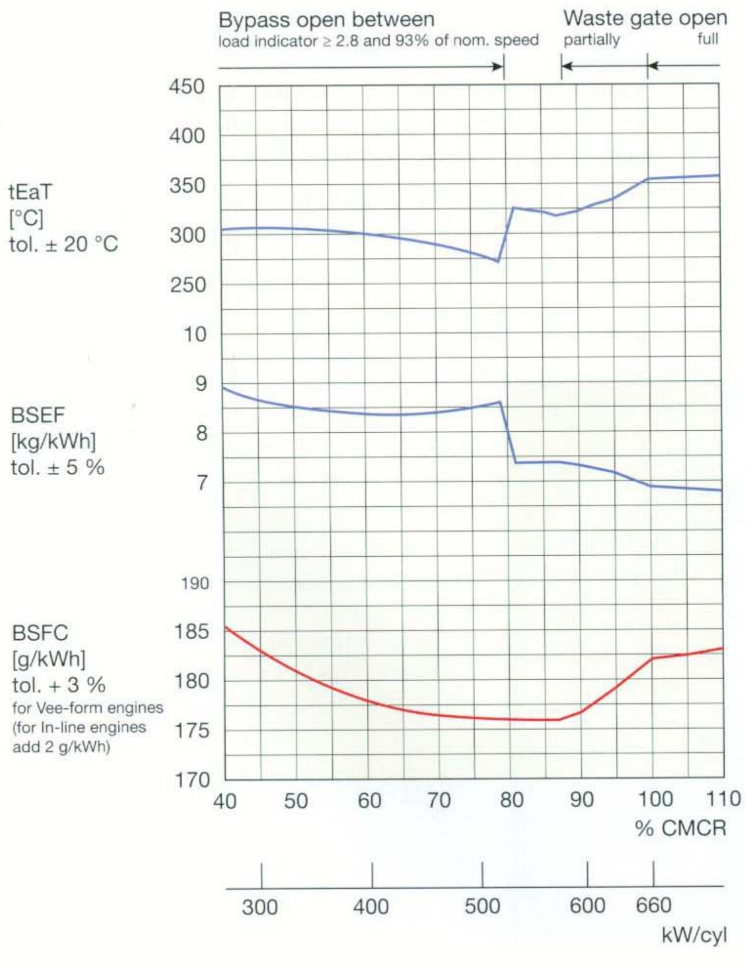
M.D.O. calorific value 43,500 kJ/kg
be = 230 g/e kWh. genset at 100% M.C.R.
fuel power per second =

$$\frac{0.230 \times 43,500}{3600} = 2.7791 \text{ kW}$$

$$\text{Therefore efficiency } \frac{1}{2.7791} = 0.3598 \text{ or } 35.98\%$$

In these examples, the efficiency varies from ± 36 to 53%. Note: this applies to a new engine operating under optimal circumstances!
For a diesel engine with a large number of running hours and consequently increased wear of the parts, the efficiency may be several percent lower! Due to:

- deficiencies in the fuel injection and consequently combustion;
- inefficient turbo-blower;
- ineffective intercooler;
- lower compression due to wear of piston rings and cylinder liner.



Specific fuel consumption and engine load
The lowest specific fuel consumption is usually not achieved at full-load, but at approximately 85% of full-load.

23.5 Fuel consumption

Fuel gauges in systems usually provide the fuel quantity in litres and m³. To convert from litres to kilograms, the specific mass and the fuel specifications must be known. The specific mass varies from 950 to 1010 kg per m³ at 15 °C for H.F.O. and 840 to 920 for M.D.O..

$$\text{Fuel in kilograms} = \frac{\text{Fuel in litres}}{\text{Specific mass of the fuel}}$$

◄

The specific fuel consumption of a Sulzer ZA 40 S diesel engine, cylinder power capacity 660 kW at 100% M.C.R, category III.

Between 40 and 100% M.C.R., the specific fuel consumption varies from 177.5 grams per kWh at 75% M.C.R. to 185 grams per kWh at 100% M.C.R.

23.6 Fuel consumption for engines in diesel-power plants

Electro-generators have a high efficiency of between 95 and 98%. The higher the power output of the generator, the higher the efficiency. Dependent on generator load, the fuel consumption is directly proportional to the engine load. Depending on the load of the engine, the specific fuel consumption seldom varies.

23.6.1 Calculating the fuel consumption and the specific fuel consumption for diesel power plants

Example

An engine drives a three-phase generator, which feeds an extensive electricity network with a step-up transformer where the voltage generated is transformed to that of the power grid. For instance, when 6600-volt is stepped up to 100,000-volt mains voltage, frequency 50 Hz. then:

$$\text{fuel consumption per hour} = \frac{\text{supply gauge reading} - \text{return gauge reading}}{\text{number of hours measured}}$$

The results are denoted either in litres or in m^3 per hour. At the same time the electricity production is measured at the main switchboard =

$$\frac{\text{final kWh} - \text{initial kWh}}{\text{number of hours measured}}$$

From this calculation, the specific fuel consumption can be calculated (be). It is important to note that the specific mass of the fuel must be given for the temperature measured at the fuel gauge.
Specific mass of the bunker fuel is known (often measured at 15 °C). This specific mass must be converted for the fuel temperature at the gauges.

$Sm_2 = Sm_1 - \Delta t \times \alpha$

Sm_2 = specific mass during measuring
Sm_1 = specific mass from bunkering note
Δt = temperature difference
α = expansion coefficient of the fuel

The amount of fuel per second can now be calculated in grams per kWh.

Example

A power plant consumes 200,000 kg H.F.O. per day. There are five diesel generators with a power output of 8000 kW. The electrical efficiency of the generator is 98%. What is the specific fuel consumption in g/kWh?

Solution

The shaft power of one diesel engine is:

$$\frac{8000}{0.98} = 8163 \text{ kW.}$$

The fuel consumption per hour per engine is:

$$\frac{200{,}000}{5 \times 24} = 1667 \text{ kg.}$$

The specific fuel consumption per kWh is:

$$\frac{1.667{,}000}{8163} = 204 \text{ grams per kWh.}$$

23.7 Fuel consumption for propulsion diesel engines

The fuel consumption per unit time or the specific fuel consumption is not the only important factors, but also the fuel consumption per nautical mile, or tonnage cargo per nautical mile or kilometre (inland).
Engines in categories III and IV often have electronic measuring devices, which are connected to the indicator cock of each cylinder. The measuring device gives the average induced pressure at a certain RPM.
The mechanical efficiency of the engine is more or less fixed: usually 90% for four-stroke engines and 95% for two-stroke engines. Therefore the mean effective pressure can also be calculated:
$p \text{ eff.} = p \text{ ind.} \times \eta \text{ mech.}$
The shaft power can now be calculated using the engine formula and measured RPM. The fuel consumption per unit time can easily be calculated if both an input and an output gauge have been incorporated in the fuel-supply system. One fuel gauge is also sufficient if placed before the fuel-supply system.
The specific fuel consumption is then simply calculated as shown in the previous example.
It is also difficult to measure the mechanical efficiency. However, using a torsion gauge together with the fuel consumption, it can be calculated accurately.

Note
It is advisable to perform regular calculations and check if the results diverge from previous calculations or the protocol data provided by engine manufacturers. Data provided by engine manufacturers are gathered from trials and often under ideal conditions. They may have a maximum deviation of 5 % according to the project guides of the engine concerned.

The fuel consumption per nautical mile or voyage. It is important to know that the fuel consumption per unit of the distance covered is proportional to the velocity cubed.

23.7.1 The most economical speed

According to the propeller law, the ship's resistance and therefore the required thrust is proportionate to the square of the velocity. The following comparisons pertain to ships with displacements well below the hull velocity. Thrust is force times the velocity, and the velocity of a ship is more or less proportionate to the propeller and engine speeds, which means that the required thrust is the velocity cubed. Therefore, the graph below has a curve to the third power.

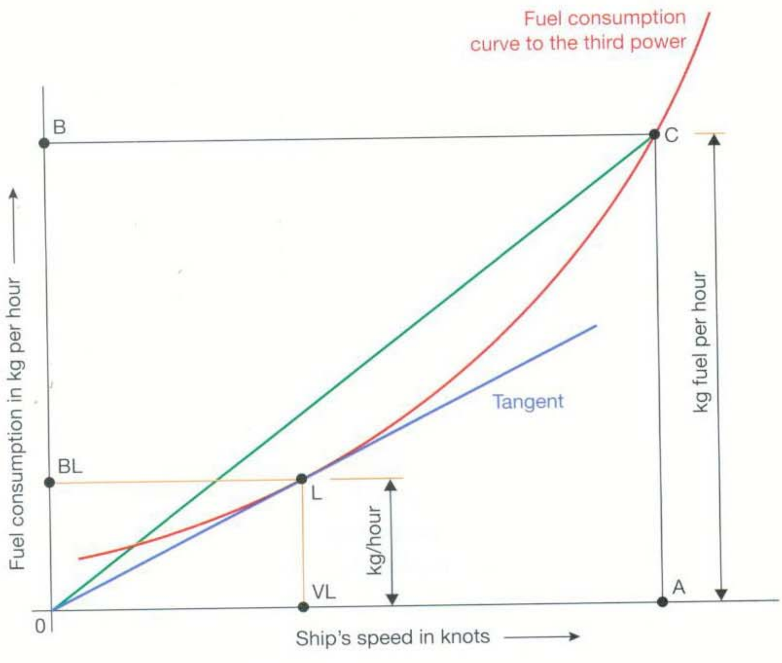
See also Chapter 19, Ship propulsion.

Note
A slight reduction of the speed provides a significant decrease in fuel costs.

Example
Speed reduction to 70% of maximum. The fuel consumption according to the curve is considerably reduced:
$0.7^3 = 0.343$ or 34.3%. Therefore, for an identical time unit the fuel consumption has significantly decreased.

A fuel-consumption curve.

horizontal: ship's speed in knots
vertical: fuel consumption per hour
The angle of inclination to point C is a measure of the fuel consumption per distance. The lowest fuel consumption per nautical mile is shown by the curve to the third power, point L.

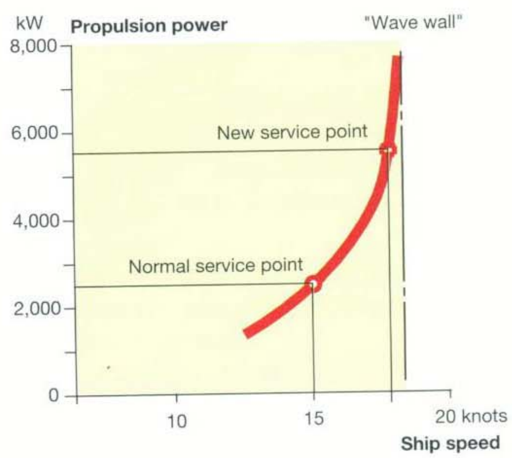


A = Service speed at 100% M.C.R.
B = Fuel consumption at 100% M.C.R.
C = Service point at 100% M.C.R.

OC = The angle of inclination
L = Lowest fuel consumption in kg/hour/nautical mile

The required engine-power output for a certain speed.

horizontal: speed in nautical miles per hour
vertical: required engine output in kW
Sometimes increasing the output and therefore the propeller speed does not necessarily increase the ship's speed; the maximum hull velocity has been reached.
At 15 nautical miles per hour, the output is merely 2500 kW; and it subsequently increases rapidly per extra nautical mile per hour.
When increasing the engine speed, the velocity and required output increase proportionately; therefore, the fuel consumption increases by the power of three.
When reducing the engine speed, the velocity and required output decrease proportionately, therefore, the fuel consumption decreases by the power of three.



Power and speed relationship for a 600 TEU container ship

23.7.2 Consumption for a certain route

In the example, speed is reduced by 30%. This means that it takes $\frac{100}{70}$ or 1.43 times longer to cover a certain route.
Ultimately, the fuel consumption for the route is equal to $1.43 \times 0.343 = 0.49$ or 49%.
The fuel consumption for the entire route is therefore halved at a velocity reduction of 30% and a 43% longer voyage time.

The speed at which the fuel consumption for a certain distance is lowest can be read from the graphs available for each specific propulsion plant.

Minimum consumption is the point where the tangent meets the curve to the third power. This tangent has a certain angle with the horizontal: the slope.

23.7.3 Modern calculation programmes

The propulsion engine data, the fixed-pitch or the controllable-pitch propeller data and various other data concerning the ship as well as the weather conditions en route allow the expected speed and fuel consumption to be calculated using special programmes. This in conjunction with current satellite navigation that can accurately provide the distance covered making the calculations increasingly accurate.

23.7.4 Sailing schedules

Ships receive a sailing schedule from their shore-based organisations, giving the distances to be covered between ports and some extra time allowing for increasing and decreasing speed. An average speed can be calculated. Using the ship's slip table produces a certain RPM.

23.7.5 Weather routeing

The SPOS system of Meteo Consult provides the weather forecast for a certain voyage. Onboard, by calculating alternatives based on, for instance, wind, sea, swell, ocean-current, ice information, the master can optimize voyage planning.
Maersk Lines works with the V.E.S.-system, Voyage Efficiency System.
This system calculates the most economic ship speed and takes the currents, tides, route information and port related data into account. This system allows sharing of ocean-current data between ships sailing this route. The users exchange information regarding their positions and speed allowing the strength and direction of currents to be determined.
These data collected by the ships are processed and the results are then made available to other shipping. When new information is received by the system about the route of a certain ship, the results of the recalculations are transmitted to that ship, so an alternative route and/or speed can be chosen.
Apart from the most favourable speed and propeller speed, the estimated expected fuel consumption is calculated and relayed.
The VES-system gives the amount of fuel consumed for each voyage in the past, the distance from one port to the next, the fuel quantity to be ordered and the amount of fuel in shore storage.
Vescos, the central Databank in Copenhagen, stores all this data. Virtually all variables are taken into account, for instance, calm sea routes or shallow water routes.
The VES system also supplies:
- harbour information;
- safety instructions;
- port details;
- information for masters and officers.

23.8 Lubricating-oil consumption and specific lubricating-oil consumption

The lubricating-oil consumption of an engine is also important. Not only regarding the lube oil costs but also as above-average lube-oil consumption is indicative of engine wear.
Increased lubricating-oil consumption may be caused by:
- **worn piston rings**
  Too much lubricating oil remains on the cylinder liner where it is burnt and contaminates the exhaust-gas system via the exhaust valve. Pollutant emissions increase.
- **worn valve stems and exhaust valves in particular**
  This results in lubricating-oil leakage to the exhaust system.
- **sealing leakage** of valve covers, crankcase covers.
- when a **lubricating-oil separator** is connected to the oil system, oil may be lost due to a poorly functioning separator. The sludge tank used to collect the soiled discharge from the lubricating oil system should therefore be regularly checked.

- **leaking lubricating-oil cooler**
  This results in lubricating oil leaking into the cooling-water system. In order to prevent cooling-water leakage to the lubricating-oil system, which would cause severe damage to the engine, the lubricating-oil pressure must always be higher than that of the cooling-water system.

23.8.1 Specific lubricating-oil consumption

This is denoted in grams per kWh; litres per hour at full load is also used frequently.

For four-stroke trunk-piston engines specific lubricating-oil consumption is between 0.6 and 1.0 gram per kWh with a comment that the actual lubricating-oil consumption is dependent on operating conditions. This often adds an extra consumption of 5%.

For two-stroke crosshead engines there are two types of consumption:

1 **cylinder-lubricating oil**
This varies from approximately 0.8 to 1.4 grams per kWh and purely denotes consumption.
A part is expelled by the exhaust gases to the environment or is burnt and deposited in the exhaust-gas system, such as the turbo-blowers, the waste-heat boiler, if any, and the silencers.
The other part seeps along the cylinder liner around the piston rod into the scavenging-air duct and is drained. This part is regularly checked by taking measurements. The cylinder lubricating-oil supply to the cylinder lubricator is also often measured in order to establish the amount of cylinder lubricating-oil lost along with the exhaust gases. Remember: the amount of cylinder-lubricating oil supplied to the cylinder = amount of drained cylinder-lubricating oil + the amount of cylinder-lubricating oil lost in the exhaust gases.

2 **driving-gear lubricating oil**
This consumption is extremely low as the system is entirely separate from the combustion section and is expressed in litres per day at full load (M.C.R.).

A MAN–B&W K98 MC engine uses between 7.5 and 11.0 litres of lubricating oil per day. The power output varies from a six-cylinder engine with 34.320 kW to a twelve-cylinder engine with 68.640 kW. In grams, this equals at a specific lubricating-oil mass of 950 kg per m^3 respectively 7.125 to 11.020 litres:

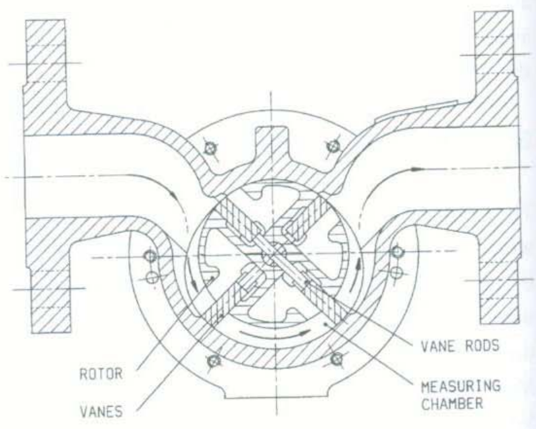
Per kWh this is approximately $\frac{7.900}{34.320}$ = 0.2 to 0.3 gram. A small amount.
Highest level of consumption occurs when there is leakage at the crankcase covers or at other seals. Certain oil consumption occurs during shooting of the lubricating-oil separator, despite the fact that the lubricating oil in the centrifuge bowl is first pumped away before shooting.

23.8.2 Choice between a four-stroke or a two-stroke engine

Cylinder lubricating-oil consumption is one of the decisive factors when choosing between a four-stroke and two-stroke engine. This is on average 15 to 35% per kWh higher for two-stroke engines than the lubricating-oil consumption for four-stroke engines.

23.9 Measuring fuel consumption

It is important to obtain an accurate measurement of engine-fuel consumption, not only from a financial viewpoint, but also to establish the specific fuel consumption for comparison with previous measurements. It indicates whether engine efficiency has reduced as a result of, for instance, a poorly functioning engine. This could be caused by insufficient fuel atomisation or a contaminated turbo-blower providing less air.



▲
A cross-section of a VAF fuel gauge. The rotating rotor and its four well-sealed vanes must be meticulously placed in the casing.

◄
Different types of fuel gauges ready for shipping.

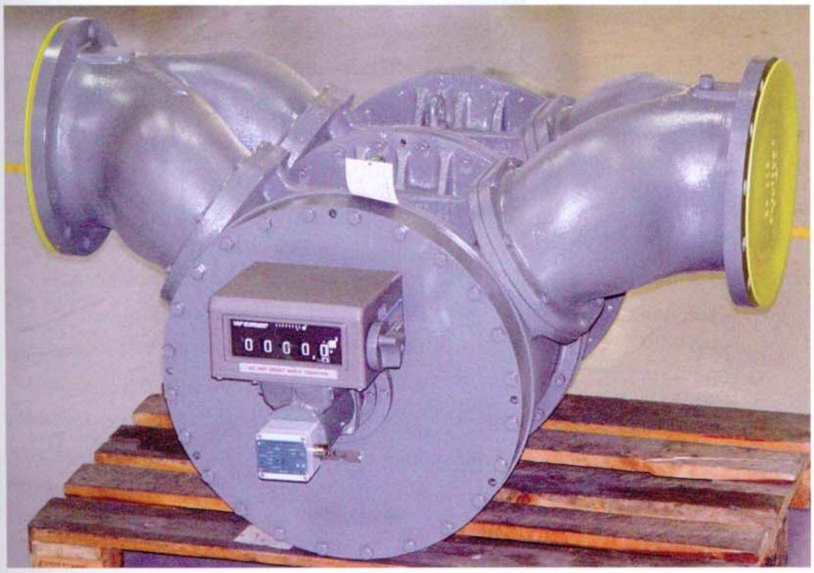
◄
A dismantled VAF fuel gauge.

1 *The casing*
2 *The rotor*
3 *The end covers*
4 *The vanes*

Fuel-gauge construction

Most fuel gauges comprise a casing with moveable rotor. The rotor speed can be measured. The rotor with vanes is mounted in a casing with very little clearance. The inflowing fuel rotates the rotor and continues its path towards the engine.

The gauge is equipped with a pulse transmitter which indicates the amount of fuel that has flowed through.
The fuel gauge is also provided with a temperature transmitter that can be remotely read.



◄
A large fuel gauge. This may be installed in a supply line on a bunker boat delivering bunkers to a receiving vessel.

▲

The casing. Note the manner in which the fuel flows through the partitions.

The shape of this patented casing ensures that the measurement is very accurate, even at minimal flow.

◄

The rotor of this large fuel gauge.

A filter with a mesh size detailed by the manufacturer should be fitted before the fuel gauge.

Measuring fuel consumption with several quantity gauges.

Fuel systems are often equipped with inlet and outlet lines. Small engines in categories I and II often have simple systems with a supply system.

In heavy-fuel oil systems, where the supplied fuel is maintained at a certain temperature, the temperature of the return fuel is often considerably lower. This means that the fuel-temperature difference must be taken into account when calculating the correct fuel consumption

An example:

A diesel engine with H.F.O. consumes 5 m^3 per hour and has a fuel circulation of 20 m^3 per hour, the temperature difference between the fuel supply and return is 25°C.

When the temperature difference is not compensated, the deviation is 2.5 % of the amount of circulating fuel. In this instance 2.5 % of 20m^3 is 0.5 m^3 or 500 litres!

The measuring deviation in fuel consumption is:

$$\frac{500}{5000} \times 100\% = 10\%$$

$$\text{Temperature difference} = \Delta T = T - T_{ret}\ (°C)$$

$$\text{Expansion} = j_{tot} = j \times \Delta T\ (\%)$$

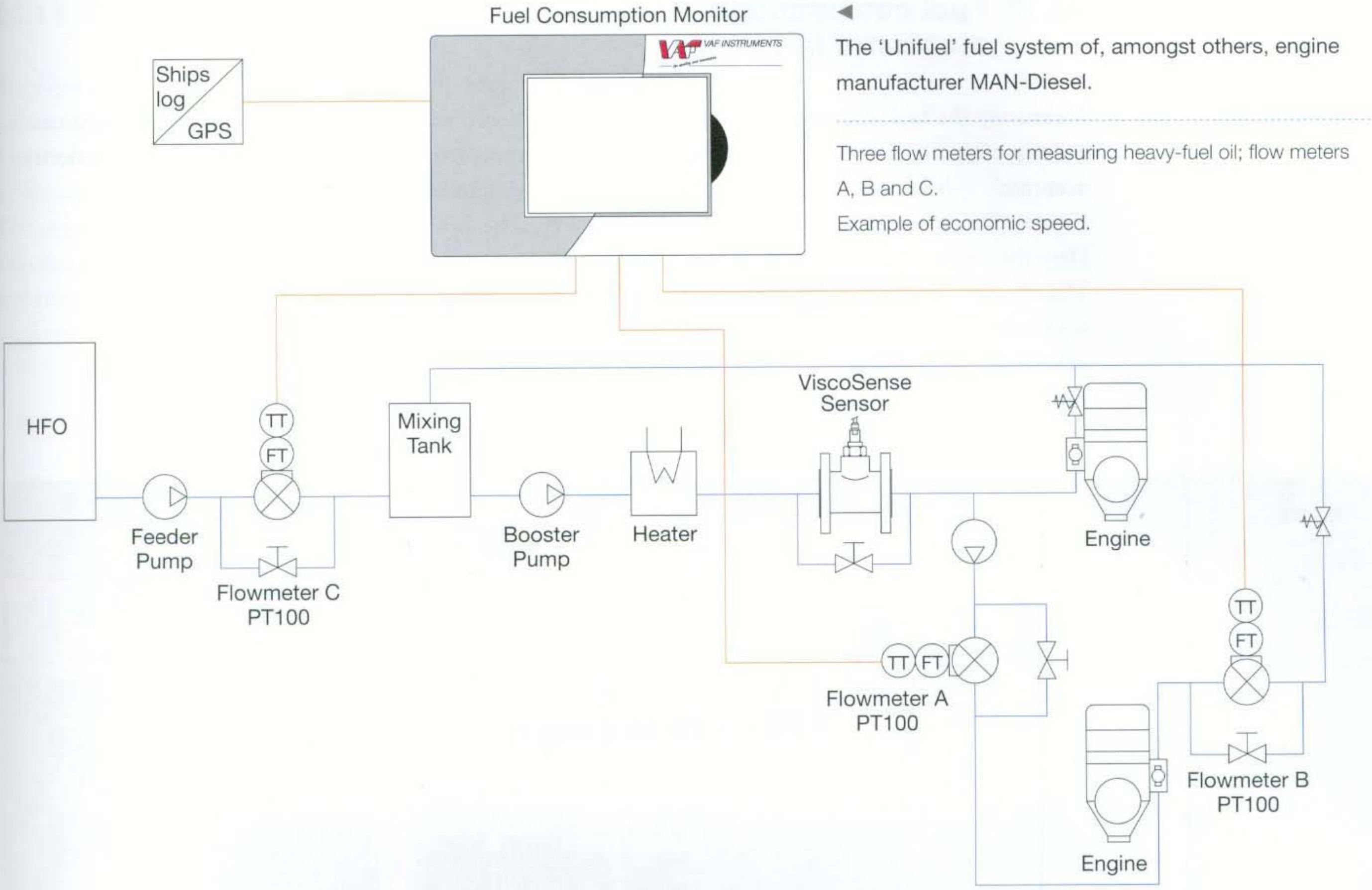
Corrected fuel amount is:

$$Q_{cor} = \frac{Q_{measurement}}{1 + \frac{j_{tot}}{100}}$$

Here T_{ret} = 15°C

$Q_{measurement}$ = measured amount

J = expansion coefficient in %



◄
The 'Unifuel' fuel system of, amongst others, engine manufacturer MAN-Diesel.

Three flow meters for measuring heavy-fuel oil; flow meters A, B and C.

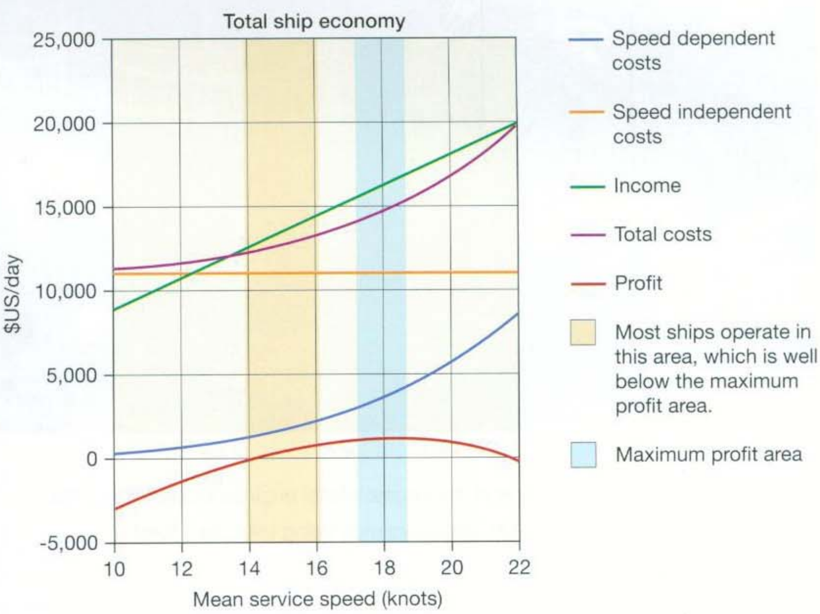
Example of economic speed.

Fuel measurements in the 'Unifuel' systems.
One of the fuel supply systems is used for the main engine and auxiliary engines. This requires three fuel gauges: one for the total amount of fuel supplied, one for the supply of fuel to the auxiliary engine(s) and one return gauge for the auxiliary engines.

Consumption main engine = total supplied amount of fuel – (supply auxiliary engines – return auxiliary engines)

There is often a temperature difference: 145 – 70 = 75° C between the fuel supplied (cold section) and the fuel supplied after the heat exchanger (hot section). Therefore, all three measurements must be meticulously adjusted with respect to temperature to avoid large deviations in actual fuel consumption.

As the fuel temperature fluctuates slightly during operation, temperature corrections should be made constantly. In this way an accurate measurement of the fuel consumption for all the engines supplied with fuel via the 'Unifuel' system is obtained.



▲
An example of the costs for a certain ship per day in $ U.S.

23.10 Fuel consumption measured in trials

Measuring the fuel consumption, and consequently the specific fuel consumption, accurately is best done during the obligatory tests for acceptance of the engine.
Here the engine is loaded with a water brake. The test data are performed in accordance with regulations, and the results are a solid basis for calculating and comparing the operative engine of a ship or a diesel-power plant. It should be noted that the engine load in a diesel-power plant can be measured very accurately by means of an electric generator.
This is not the case in ship propulsion.

Also see Chapter 19, Ship propulsion and Chapter 32, Regulations for propulsion engines, classification, repair and damage.

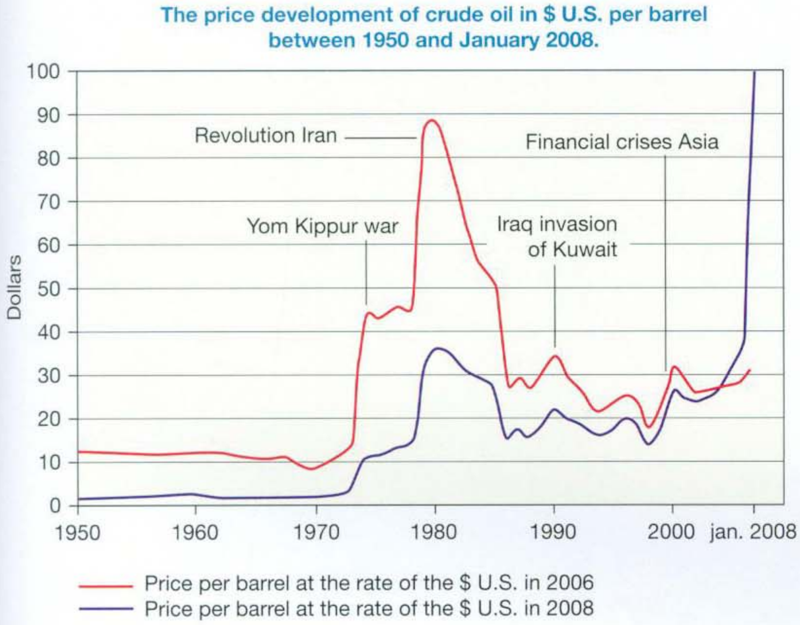
▲
One of the largest diesel engines being tested; the 'blue' water brake is being used as a load.
This is a Wärtsilä Sulzer RT Flex 96 C.

Note the size in relationship to the engineers!

23.11 The oil-price development

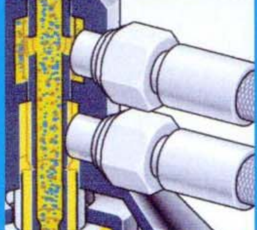
The price of crude oil is and has traditionally been calculated in U.S. dollars per barrel (159 litres). The barrel price was below twenty dollars in the 1970s and is presently over one hundred dollars! This price is dependent not only on supply and demand but also on political and financial scenarios.

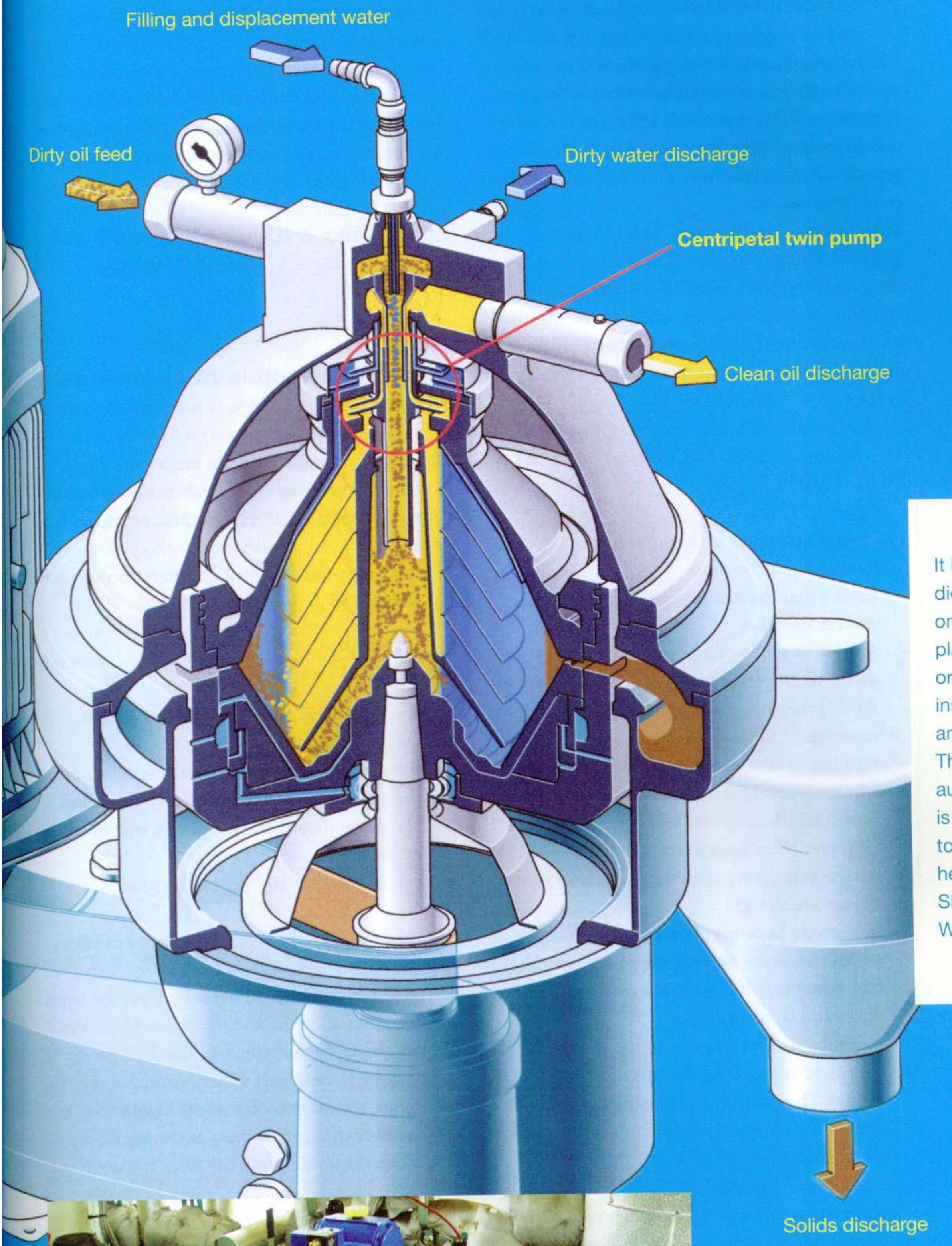
▼
The price development of crude oil in $ U.S. per barrel (159 litres) between 1950 and January 2008.



># CH 24

Auxiliary systems: Fuel and lubricating-oil separators

It is necessary that large diesel-engine plants on ships and in power plants have centrifuges or centrifugal separators installed to clean fuel and lubricating oil. The use of a modern automatic centrifuge is imperative given today's inferior heavy-fuel oil quality. Shown is a cut-away Westfalia separator.

A 'booster unit' with an automatic back-flushing fine-filter.

24.1 Introduction

Cleaning fuels and lubricating oils is crucial for optimal and long-life operation of diesel engines. Fuels, especially inexpensive H.F.O., contain various contaminants that are detrimental to the fuel system and cause problems when injected in the combustion space.
Lubricating oil is the 'blood' of the engine and serves many purposes. The quality of the circulating lubricating oil is vital. Many combustion and corrosion products are absorbed by or carried in the lubricating-oil system and must be removed in time to prevent severe damage to the engine.

24.2 Fuels

The following points are important for 'contaminant free' fuel oil:

- Ensure that the fuel tanks are clean and have no condensation.
- A drain at the lowest point of the tank is only useful when periodically used.
- Bunker fuel at well-known firms with a regular turnover.
- Fuel has to comply with minimum requirements. This is usually officially organised when bunkering large quantities. However when refuelling small amounts as with, for instance, yachts, it is assumed that 'things are all right'.
- In wintertime or in cold regions additives are added to fuels to prevent the paraffins in the fuel from coagulating. This would clog the fuel filters and stop the engine.
- From the smallest engines to the largest crosshead engines, proper fuel filters are crucial. They must be regularly changed, or manually or automatically back-flushed; this is dependent on engine size.
- Small diesel-engine plants often have a simple water separator and a magnet, to which the iron particles adhere.
- Duplex filters are also useful in small engines. One filter is operational while the other filter is cleaned for use when the operational filter is contaminated.
- In larger installations, settling tanks are also used. In settling tanks, water and impurities are separated from the fuel oil and drained off. In heavy-fuel oil systems, the contents of the settling tanks are heated to a temperature at which the specific mass difference between water and fuel is greatest, thus facilitating the separation.

Nevertheless, this does not suffice for the cleaning of heavier fuels such as H.F.O. Adequate cleaning is achieved with the use of centrifuges or separators; the centrifugal force is a decisive factor.

See also Chapter 8, Fuels, fuel-line systems and cleaning fuels.

24.3 The principle of centrifugal separators

When solid or liquid particles are carried in a solution that is under the influence of a centrifugal force, they will assume a constant speed after some time.
This phenomenon is known as separation speed, also referred to as Stoke's Law:

$$Vg = d^2 \left(\frac{s.m._d - s.m._V}{18 \times \eta} \right) \times g$$

Vg	=	velocity in metres per second
d	=	diameter of a particle in metres
$s.m._d$	=	specific mass for a particle in kg per m^3
$s.m._V$	=	specific mass of the liquid in kg per m^3
g	=	gravitational acceleration in metres per sec^2
η	=	viscosity in 10^{-3} per metres second

From the formula, the following assumptions can be made:

- the separation of solid particles from the liquid is dependent on the specific mass of both the particles and the liquid;
- the larger the difference between the specific masses, the more complete the separation;
- the lower the specific mass of the liquid, the greater the separation.

24.4 Separation in a settling tank

This is limited as specific mass difference alone aids separation. In the following assumption, a particle is 8 microns= 8×10^{-6} metres.
The specific mass difference $s.m._d - s.m._V = 50$ kg per m^3.
The viscosity $\eta = 10^{-3}$ kg per metre second in accordance with Stokes law:
$Vg = 1{,}75 \times 10^{-6}$ metres per second.

This velocity is very low! Fuel must therefore remain in the settling tank for a significant time in order to achieve a basic separation. Fuel, especially heavy-fuel oil, is heated to a certain temperature in order to create the largest possible specific mass differences and so improve the separation.

24.5 Separation with a centrifugal separator

Stokes law reads: $\frac{d^2 \times \Delta p}{18\eta} \times g.$

The acceleration g is constant, as the centrifuge rotates at a constant speed.
The force exerted on a particle in the centrifuge is related to the engine speed (the angular velocity) and the radius to the centre of the centrifuge.
The larger the centrifuge's diameter and the higher the engine speed, the greater the force exerted on the particle:

$a = r \times \omega^2$
a = accelerating force
r = radius to the centre
ω = angular velocity.

In accordance with Stokes law:

$g = r\,\omega^2$

$$V = \frac{d^2 \times \Delta p}{18\eta} \times r \times \omega^2$$

$$V = Vg = \frac{r\omega^2}{g} \times Vg \times Z$$

$$Z = \frac{r\omega^2}{g} = \text{accelerating force}$$

Example

In the settling tank, Vg was 1.75×10^{-6} metres per second. Assuming that in the centrifuge:
R = 0.2 metre. RPM n = 5000.

$$\omega = \frac{\pi \times n}{30} = \frac{\pi \times 5000}{30}$$

$$Z = \frac{r \times \omega^2}{g} = \frac{0{,}2 \times \pi^2 \times 5000^2}{30^2 \times 9.81} = 5600$$

$$V = Vg \times Z = 1.75 \times 10^{-6} \times 5.6 \times 10^{-3} = 0.98 \times 10^{-2}$$

Therefore, the ratio of the separation force in the centrifuge to that in the settling tank is:

$$\text{Ratio} = \frac{0.98 \times 10^{-2}}{1.75 \times 10^{-6}} = 5600.$$

On average, the separation force of centrifuges is 6000 times larger than that in settling tanks!
Adequately functioning centrifuges are imperative in properly cleaning fuels and lubricating oils in larger installations/diesel engines. Naturally, extremely fine pressure filters, either automatically back-flushing or manual, are also used in both instances.

24.6 Types of separators

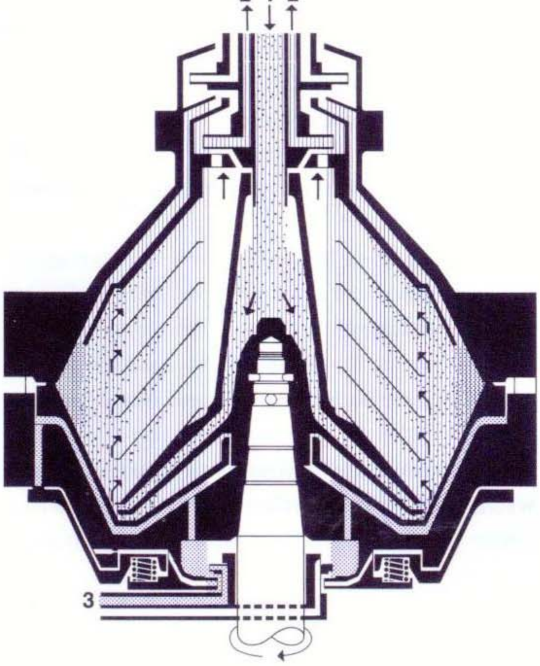
Essentially, there are two types of separators:

1 Clarifiers: for the separation of solid particles from the fuel or lubricating oil.
2 Purifiers: for the separation of solid particles and water from the fuel or lubricating oil.

▼

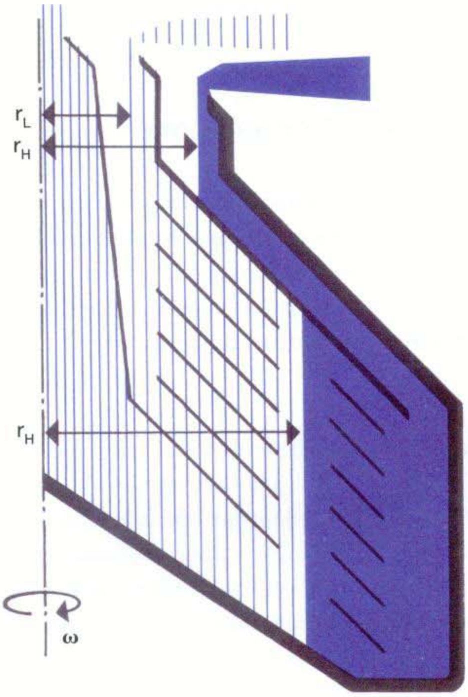
A standard clarifier of Alfa Laval.

The untreated oil is supplied by a fixed pipe to the centre of the bowl.
This untreated oil flows into the rapidly rotating bowl or drum.
The dirt particles are separated and collected in the peripheral space, the sludge space.
The cleaned oil floats to the surface in the drum and is discharged at 2.
The bottom of the bowl periodically drops. After the bowl is emptied, the bowl is high-pressure cleaned with water.
Subsequently, the bowl supplied with water from 3 is closed.



24.6.1 Interface

The interface is crucial for the operation of the centrifuge, as an equilibrium must be created between the various liquids in the rotating bowl.



A cross-section of half a separator bowl showing the radius for liquids.

r_L = radius to clean-oil discharge
r_H = radius to water discharge
r_S = radius to interface with water

Here the radius between the bowl and the various liquid interfaces and the specific masses of the liquids are the decisive factors.
The liquid pressure in the bowl is expressed as follows:

$$p = \frac{\omega^2}{2} \times \rho(r^2 - r_0^2)$$

p = liquid pressure
ω = angular velocity
ρ = specific mass
r_o = inner radius of the liquid
r = outer radius of the liquid

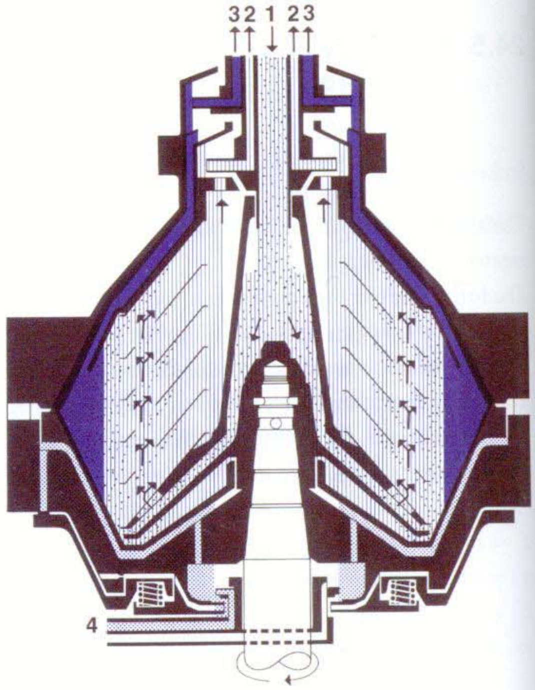
$$p_s = \frac{\omega^2}{2} \times \rho_1(r_s^2 - r_0^2) = \frac{\omega^2}{2} \times \rho_H(r_s^2 - r_H^2) \text{ of}$$
$$\rho_1(r_s^2 - r_1^2) = \rho_H(r_s^2 - r_H^2)$$

24.6.2 Purifiers

These separate both impurities and water from the fuel oil.
The separation line between both liquids (oil and water) is called the interface.

The Purifier of Alfa Laval.

1 *contaminated-oil supply*
2 *clean-oil discharge*
3 *water discharge*



All the larger modern purifiers are self-cleaning. Self-cleaning is performed as follows. The contaminated oil supply and water supply is closed, the bowl is emptied so there is no more dirty oil present. Then the bowl is cleaned, by 'shooting'. The bottom drops, which allows the solids to be discharged at great force through grooves in the outer bowl. The bowl closes, the water supply opens and then the contaminated-oil supply opens. After a large amount of water has been discharged at opening (3), the equilibrium between water and oil is restored.

Limits for conventional purifiers

- Fuels with a specific mass of maximum 991 kg per m^3 at 15 °C.
- Optimal performance is dependent on the correct specific gravity disc, which corresponds to the average density, the viscosity, the desired capacity, and the temperature of the fuel oil.
- Checking and mounting of the specific-gravity disc is time-consuming and often done by 'trial and error'. If the incorrect gravity disc is used, the water seal breaks and fuel oil is discharged via the water output to the contaminated-oil and sludge tank.

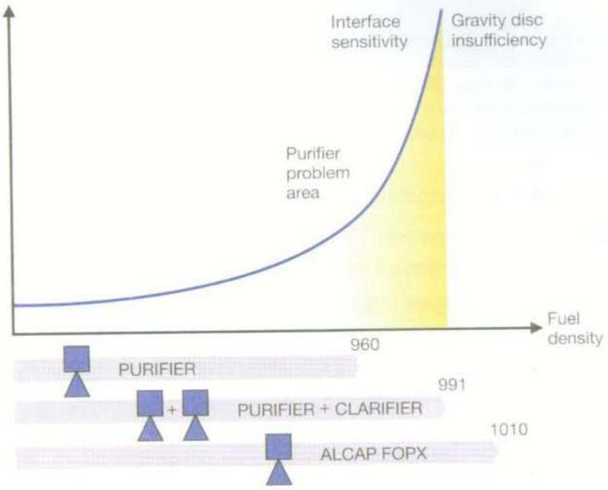
- A second purifier placed in series is required to ensure that the fuel is properly cleaned. This **clarifier** works as a kind of safety net. The impurities not yet separated in the purifier are removed from the fuel with this step. Here the maximum specific mass of 991 kg per m³ at 15 °C also applies.

24.6.3 Clarifiers

Limits for conventional clarifiers
The main problems with the cleaning of fuels, irrespective of density, are fuel-oil losses and water-discharge limitations. When a centrifuge has been arranged as a clarifier, during shooting/cleaning of the clarifier, no water is supplied to displace the oil and only water and impurities are removed, so all the fuel oil present is discharged. This centrifuge type, clarifier, does not remove water because the water discharge is closed and can only be removed with the sludge during shooting.
Placed in series with a purifier (water and dirt) and a subsequent clarifier (remaining dirt), it is a useful type of separator.

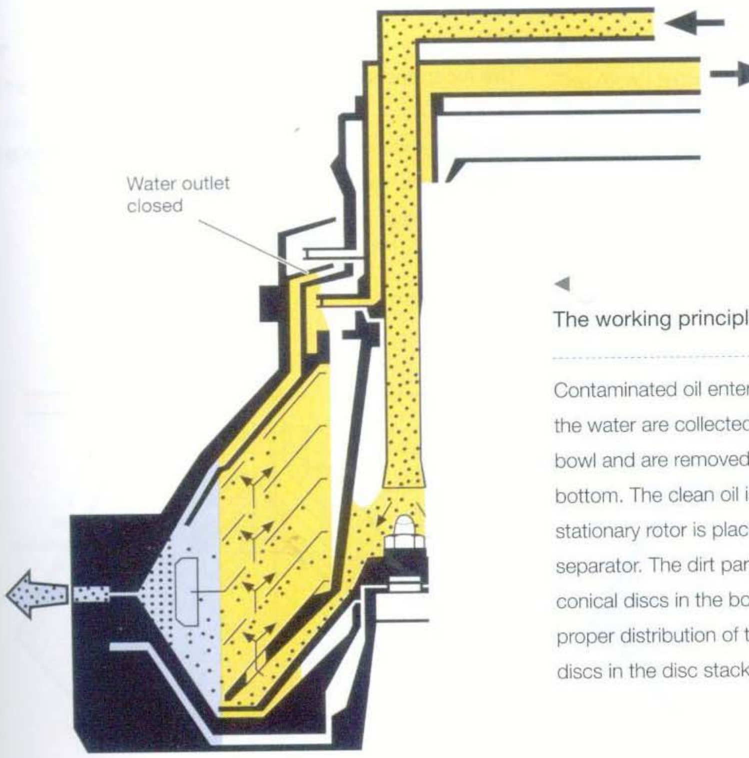
24.7 New separators by Alfa Laval

New types of separators have been developed to clean fuels with a high specific mass.



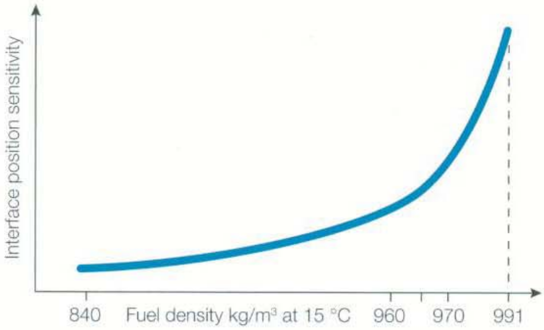
▲
The working areas of various Alfa Laval centrifugal separators.

horizontal: specific density of the fuel in kg per m³.
vertical: sensitivity of the interface.
reduced functioning of the gravity disc.
The purifier functions adequately to approximately 960 kg per m³.
The combination of purifier and clarifier is sufficient to approximately 991 kg per m³.
The new Alcap FOPX of Alfa Laval functions to approximately 1010 kg per m³.



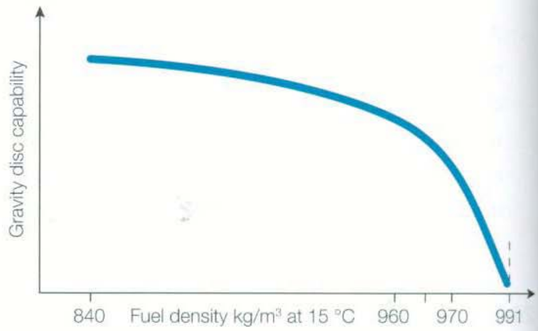
◄
The working principle of a conventional clarifier.

Contaminated oil enters the centre of the bowl. The dirt and the water are collected in the sludge space (periphery) of the bowl and are removed periodically by dropping the inner bottom. The clean oil is discharged via a disc stack. Here a stationary rotor is placed, which pumps the oil from the separator. The dirt particles are easily separated between the conical discs in the bowl. The vertical perforations ensure a proper distribution of the contaminated oil. Without these discs in the disc stack, separation would take much longer.



A graph indicating interface position's sensitivity.

As shown, interface sensitivity (deteriorating operation) increases significantly over 960 kg per m^3.



A graph for the capacity of the gravity disc.

As shown, the capacity of the gravity disc decreases significantly above an approximate specific gravity of 950 kg per m^3.

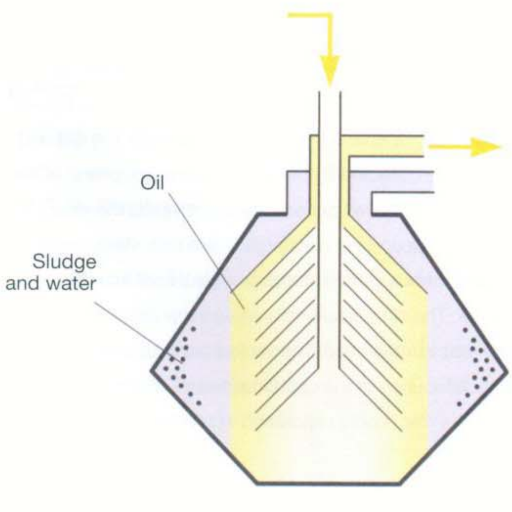
24.7.1 'Alcap'-technology by Alfa Laval

Contaminated pre-heated fuel oil is fed continuously to the 'S' type separator, which essentially functions as a clarifier. Clean fuel oil is continuously pumped away. When separated water approaches the disc stack, traces of water start to escape with the cleaned oil. This minor increase in water content of the cleaned oil is detected by the transducer MT 50, which is installed in the clean-oil outlet. Sludge and water accumulate at the bowl periphery. When separated water reaches the dirty disc stacks, traces of water escape with the clean-oil flow to the clean-oil outlet.

A measuring device MT 50 detects the water content in the fuel. At an unacceptable increase in water content, the clarifier is triggered to initiate an automatic discharge of the water via a water drain controlled by a magnetic valve.
When shooting the sludge – discharging the dirt – the contaminated-fuel supply is closed and the fuel remaining in the bowl is displaced by the supply of water and discharged to the clean-fuel supply. The separator is cleaned at great force.
So in principle, this centrifuge is a clarifier with an automatic water drain!

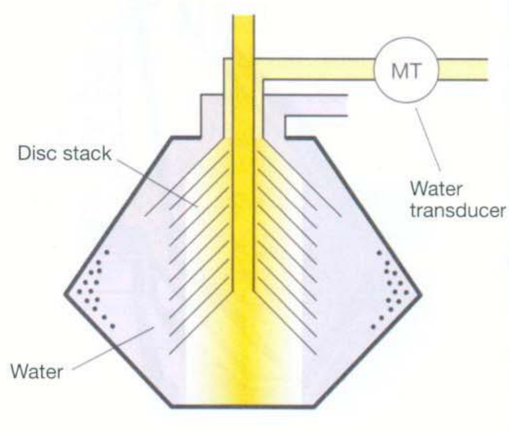
The Alcap technology I.

Low free-water in the fuel. The water discharge opens automatically as soon as the water content carried in the clean oil is above a preset limit.

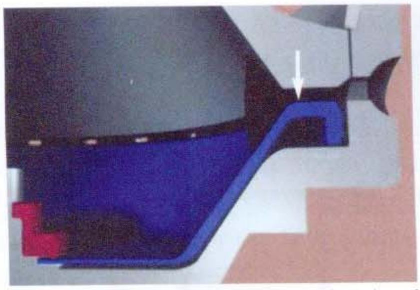


The Alcap technology II.

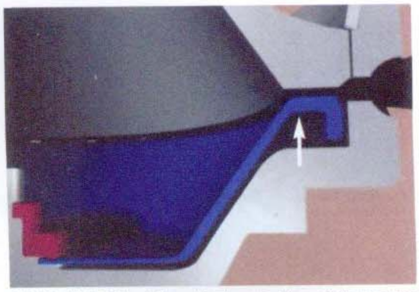
Higher free-water content in the fuel. The measuring device detects more 'free' water in the clean fuel and opens the water discharge via the process controller.



STEP ONE: The CentriShoot discharge slide is fixed at the centre. During separation it blocks the discharge ports.



STEP TWO: During sludge discharge, the edge of the slide flexes downward, exposing the ports.

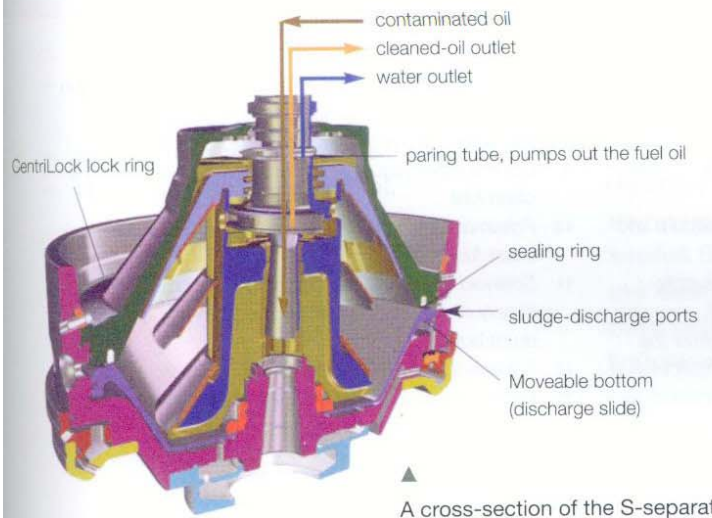


STEP THREE: After discharge, the slide moves gently back into position, closing the ports. Closing is done hydraulically, without any springs.

▲

Opening and closing the bowl by moving the flexible inner bottom.

Normally, the water pressure, maintained by the centrifugal force, keeps the bottom closed in top position. When opened, the bottom drops slightly.



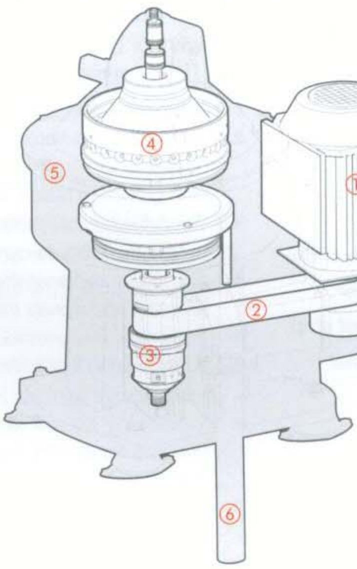
▲

A cross-section of the S-separator.

▼

Construction of an S-separator of Alfa Laval.

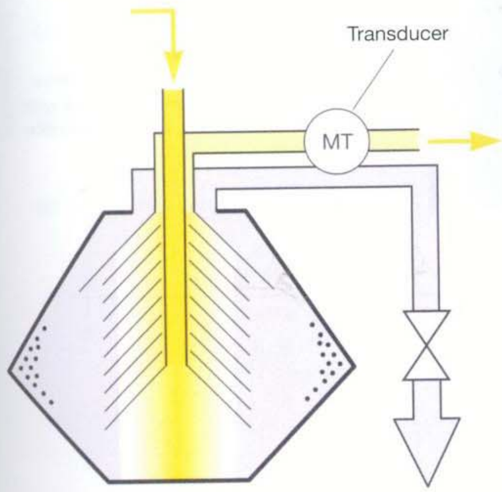
1 *electromotor*
2 *belt*
3 *bearing with special springs for a flexible drive*
4 *bowl*
5 *casing*
6 *sludge and water discharge*



▼

The Alcap technology III.

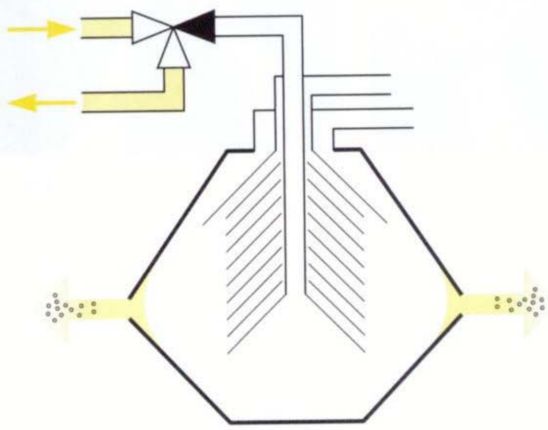
The water is discharged.

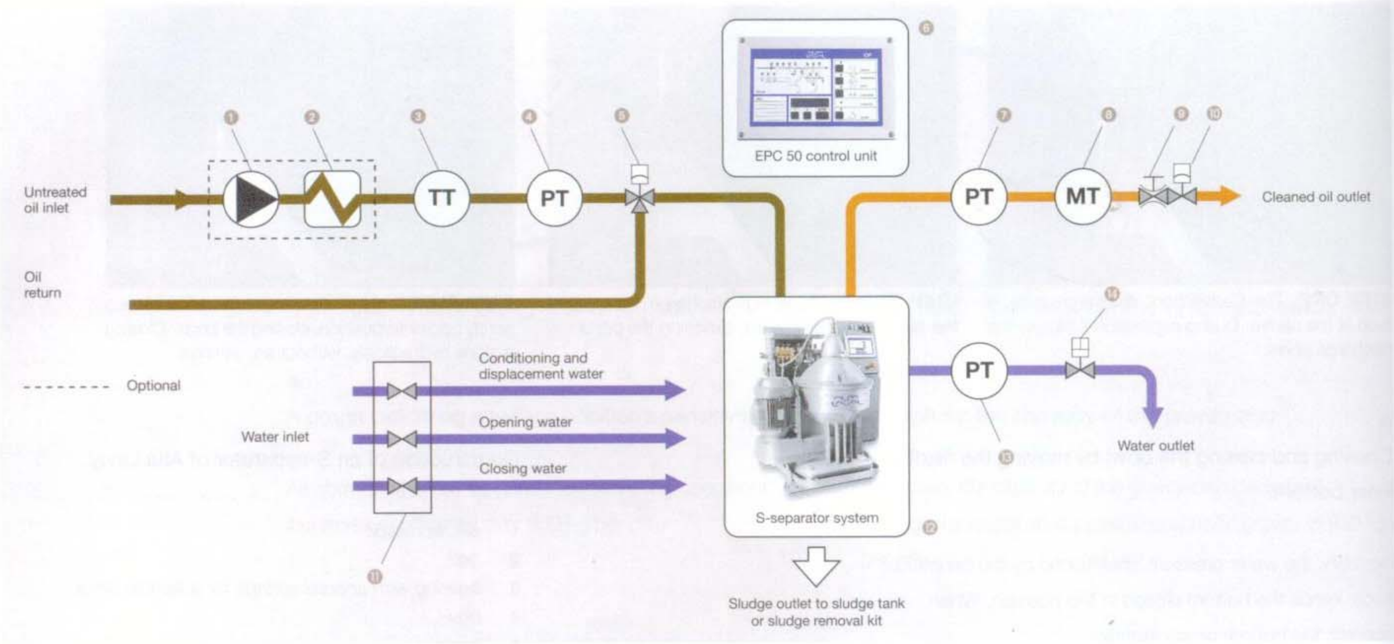


▼

The Alcap technology IV.

'Shooting' of the separator to remove the solids. The contaminated fuel is returned, water displaces the remaining fuel and the bowl opens which causes the sludge to be discharged.





▲

The complete flow diagram of an Alfa Laval S-separator.

1 *Feed pump for contaminated fuel.*
2 *Pre-heater, pre-heats the fuel to the separator temperature; for heavy oil often just below 100 °C (the boiling point of water), approximately 98 °C.*
3 *Temperature transmitter measures the temperature and signals the process-control unit.*
4 *Fuel-pressure transmitter measures the fuel-supply pressure and signals the process-control unit.*
5 *Pneumatically controlled three-way valve controls the contaminated fuel to the separator or to the recirculation pump line.*
6 *Process-control unit regulates the seperator.*
7 *Pressure transmitter, measures the discharge pressure of the cleaned fuel and signals the process-control unit.*
8 *Transducer continuously monitors the water content in the cleaned fuel. Ensures that the separator is functioning correctly. A water surplus, shoot!*
9 *Control valve, regulates the discharge pressure of the clean fuel.*
10 *Pneumatically controlled shut-off valve. Closes the clean-fuel outlet.*
11 *Solenoid valve for water discharge which regulates the supply of separator water and conditioning water in the drum/bowl.*
12 *Separator, removes the solids and water from the fuel.*
13 *Water-pressure transmitter measures the water pressure in the drain outlet and signals the control unit.*
14 *Drain valve, magnet valve, opens when the water is drained from the seperator.*

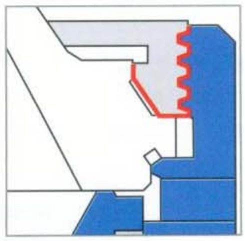
▼

The new lock rings of the Alfa Laval seperator: a simple flexible steel strip, fixed with hexagonal socket screws.

A sledge hammer to obtain the correct mark and a special ring spanner belong to the past.

CentriLock can be only removed with an Allen key. No sledge hammer is necessary.

CentriLock lifts out and snaps in easily – without any threads to wear.



Threaded lock rings must be removed with a sledge hammer. Over time, metal-to-metal wear between bowl and lock ring, leads to expensive bowl repair or replacement.

24.8 Effects of separators

At present, heavy-fuels contain residues of catalysts used during the refining process, the so-called cat fines.
These aluminium oxides can cause severe damage to a fuel-injection system, such as, the high-pressure fuel pumps and injectors. They can also damage the remainder of the engine via the combustion space.
These must be removed from the fuel by the separator as efficiently as possible.

24.8.1 Separation enhancing factors

The following factors optimise separation:
- fuel temperature in the separator;
- viscosity in the separator;
- feed rate of the separator;
- use of the disc stack;
- specific mass of fuel and water;
- size and number of the solid particles in the 'untreated' fuel;
- amount of water in 'untreated' fuel;
- chemical properties of fuel oil.

24.8.2 Rule of thumb: the lower the capacity the cleaner the fuel!

This is why most separators have a far higher capacity than the maximum fuel consumption. Separators always only operate at a percentage of their maximum capacity.

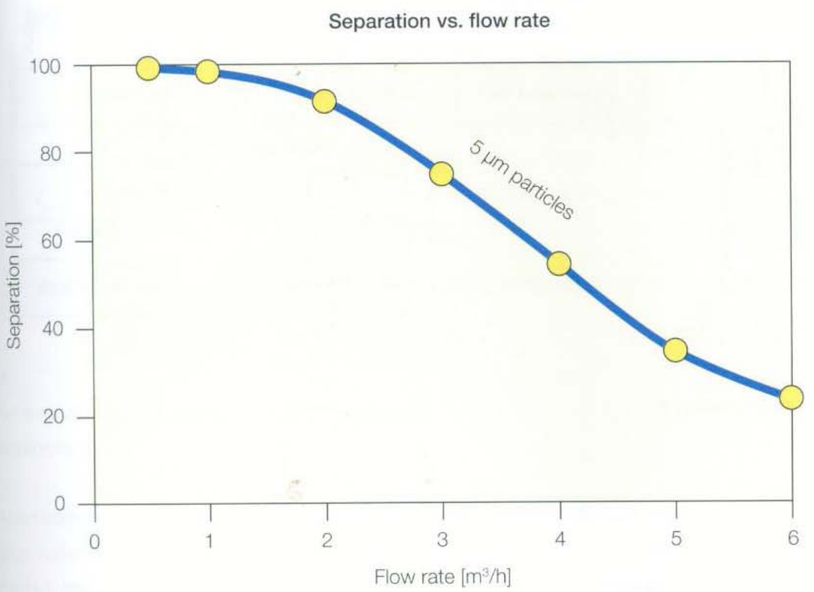
24.8.3 Cat fines content

According to the ISO 8217 regulations, the maximum cat-fines content may amount to 80 parts per million (ppm). However, engine manufacturers require, if possible, a maximum content of 15 ppm. This makes it necessary to reduce the cat-fines content in the fuel from 80 to 15 ppm. Separators are capable of separating particles as small as 2 microns.

24.9 Examples of cleaning systems for lubricating oil, fuel, sludge and bilge water

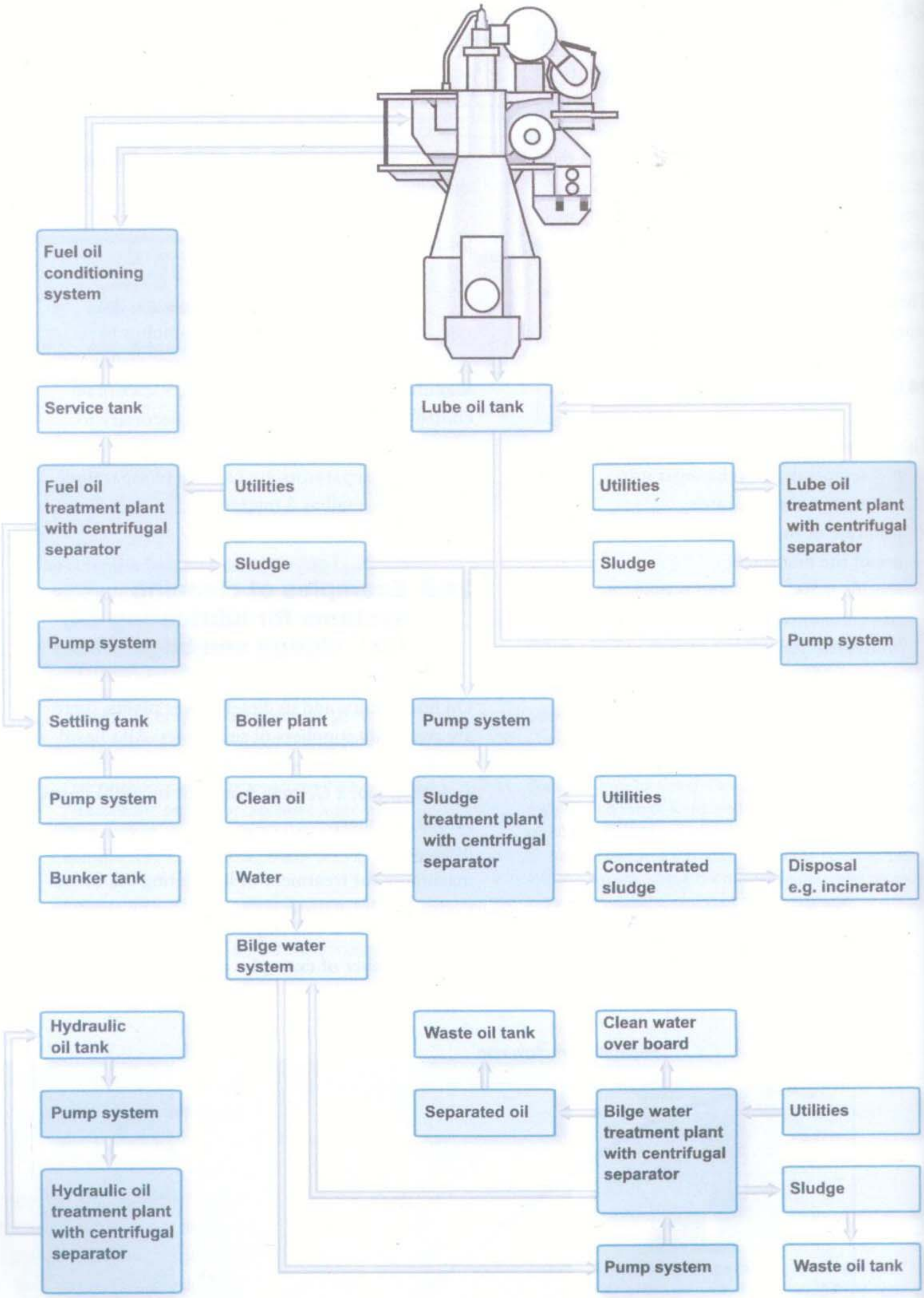
On board ships and in diesel-power plants, there are two main suppliers of separators: Alfa Laval, a Swedish firm with its well-known blue machinery, and Westfalia, a German firm with its well-known green machinery.
Both supply an elaborate program of systems and machinery for treatment of lubricating oil, fuels, sludge and bilge water.

Here a number of examples of Westfalia.



◄ Separation of aluminium and silicon particles, size 5 microns.

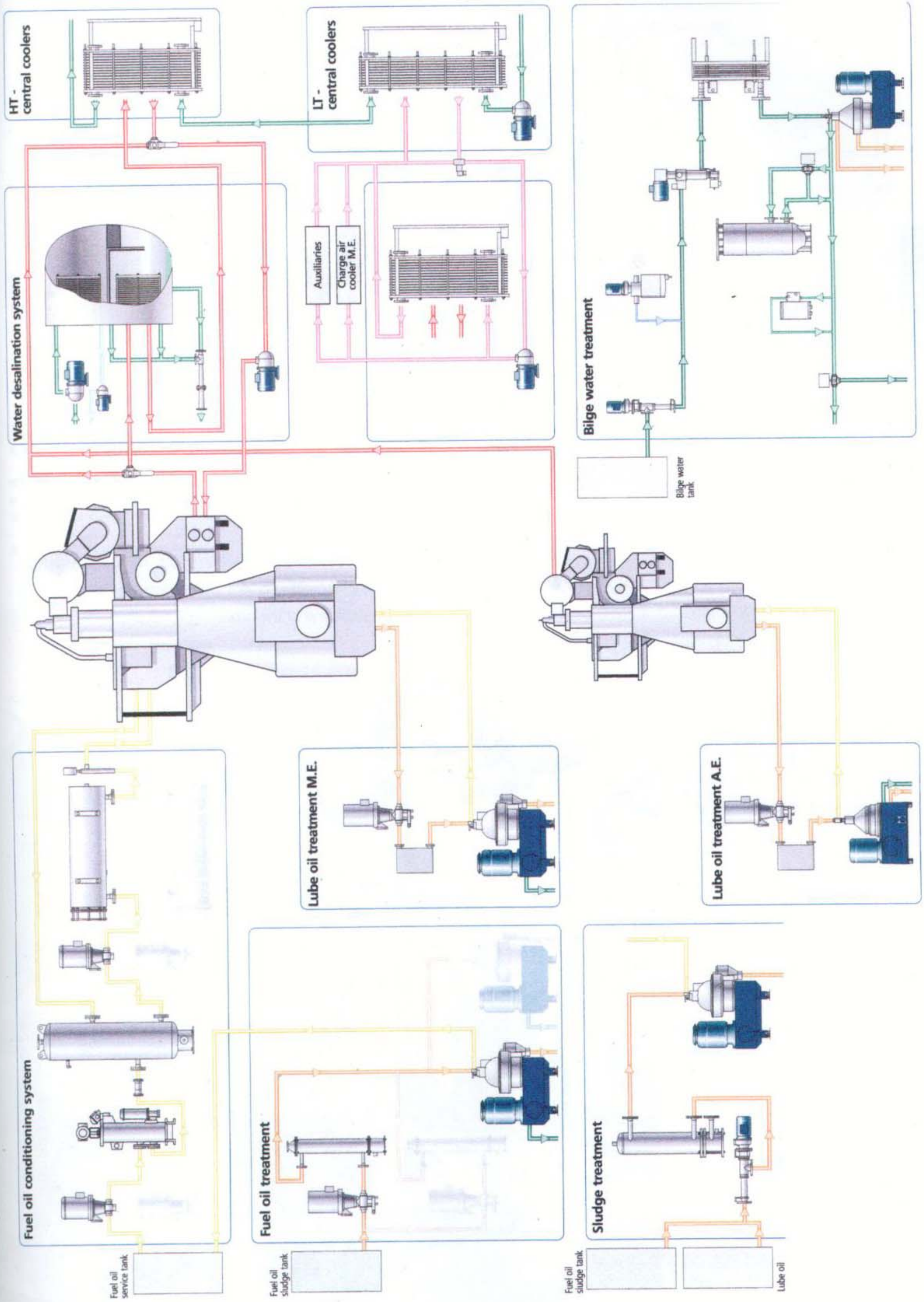
horizontal flow in m^3 per hour
vertical: efficiency of the separation in percentage.
It is clear that separation decreases as the flow increases. This is logical as the fuel remains in the bowl for a shorter time span, so the separator cannot remove all the small particles.



▲

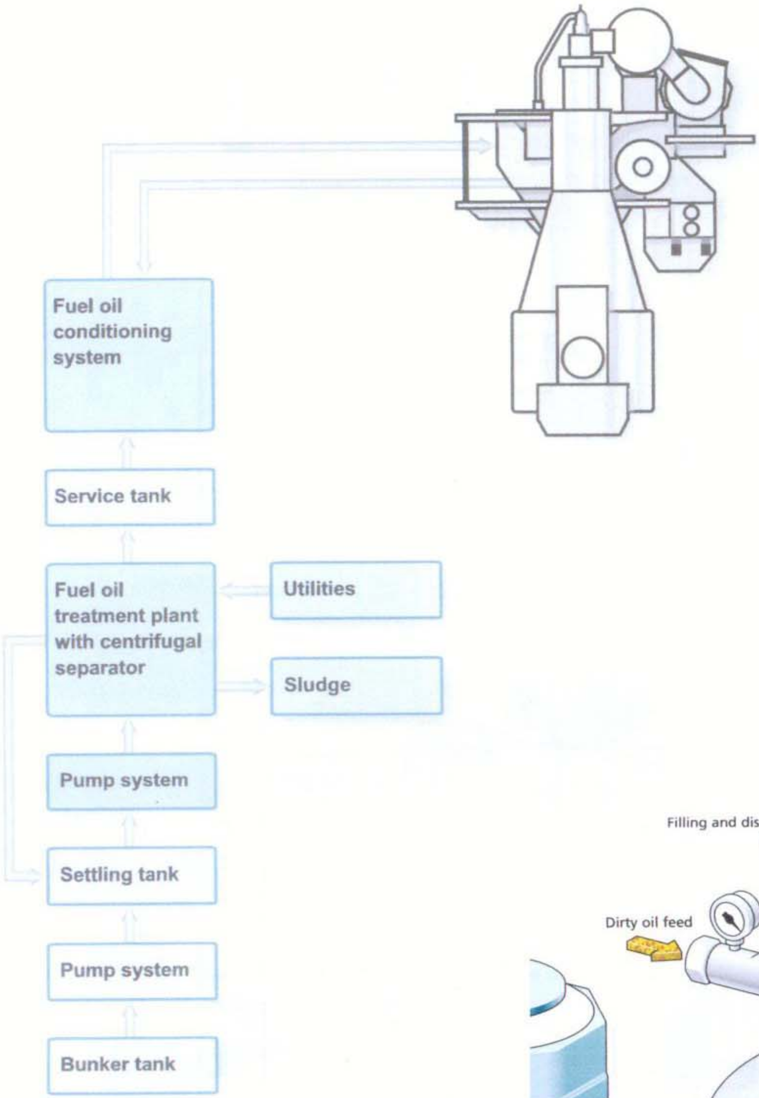
An overview of all the systems surrounding a two-stroke crosshead engine.

In large plants each system has a separator: shown here for heavy-fuel oil, lubricating oil, sludge and bilge water. Hydraulic systems may also be equipped with a separator. This happens mainly in very large hydraulic installations and seldom on board ship.



▲
An overview of the piping with all the essential equipment.

Apart from cleaning heavy oil, lubricating oil, sludge and bilge water for both the main engine and auxiliary engines, the H.T. and L.T. fresh-water cooling systems are also incorporated as well as a sea water evaporator, which is cooled with H.T. cooling water.



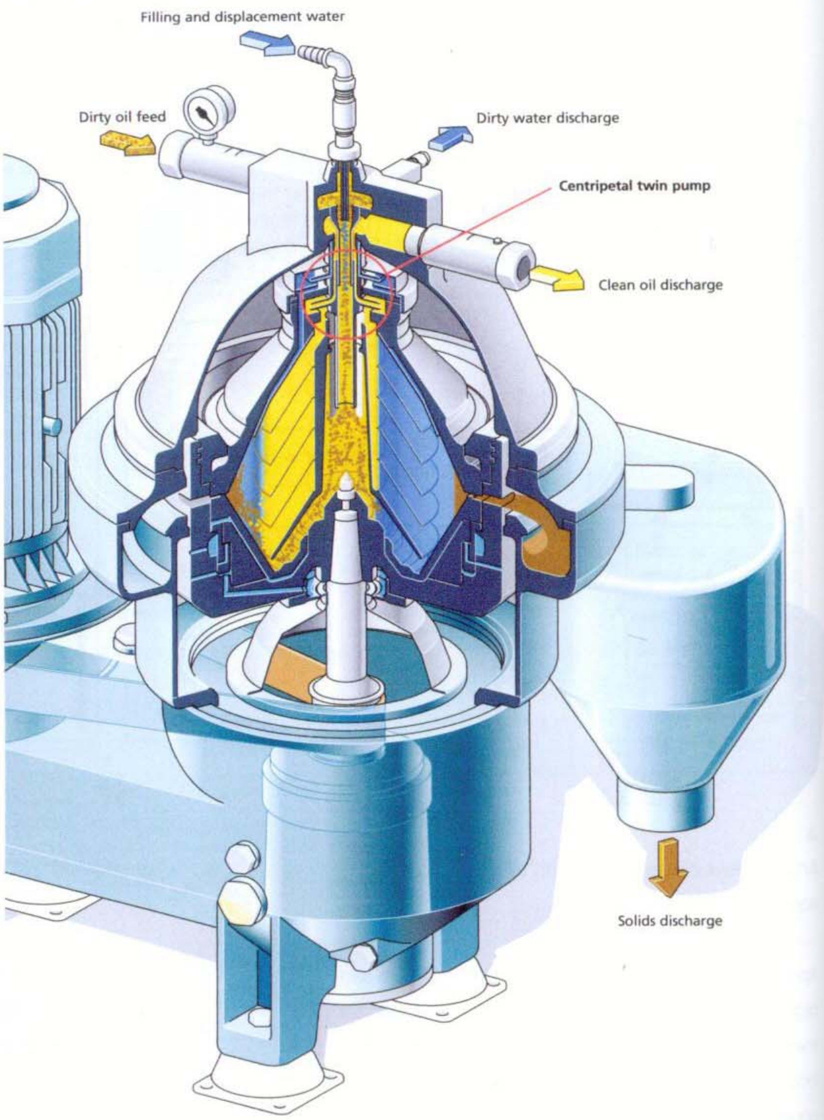
◄

A flow diagram for the cleaning of heavy oil for the main engine.

▼

A cross-section of a self-cleaning fuel and lubricating-oil separator of Westfalia.

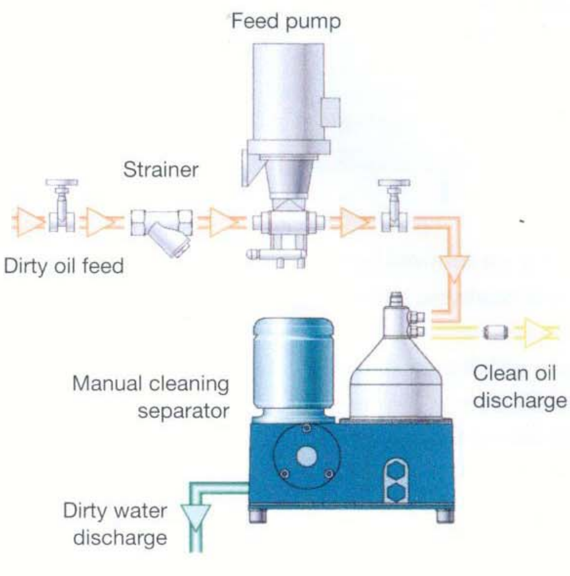
The sludge is discharged, bottom right. The contaminated fuel or dirty lubricating oil is supplied centrally along the vertical centre line of the bowl.

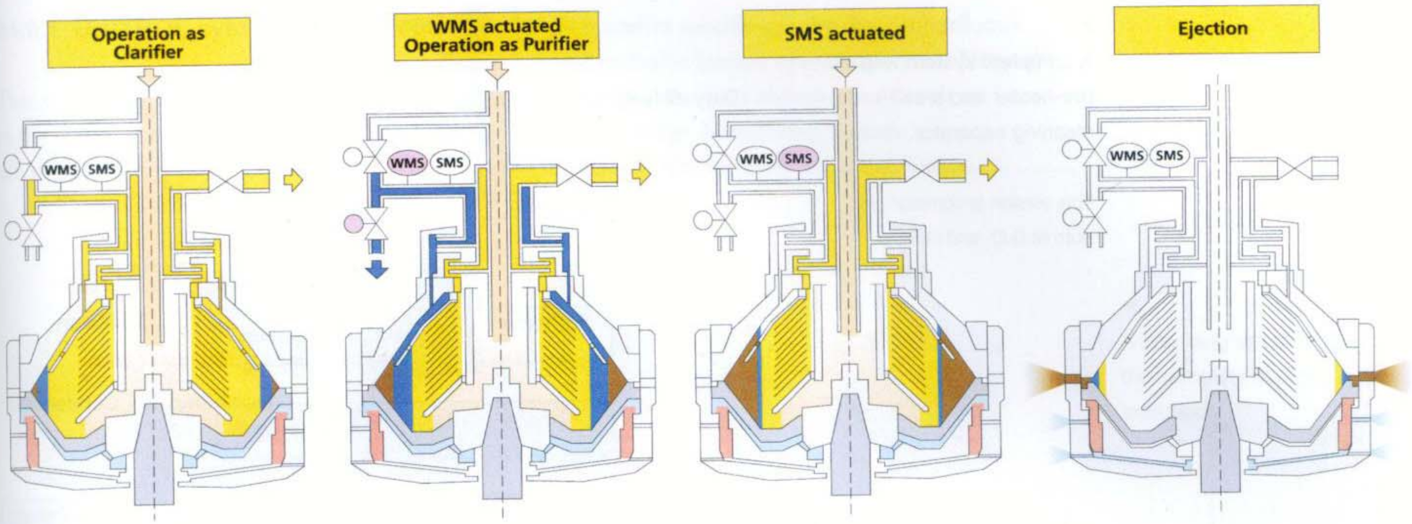


▼

A simple system with a separator for a small plant operating on M.D.O..

The separator must be manually cleaned at regular intervals. There is no fuel pre-heater.





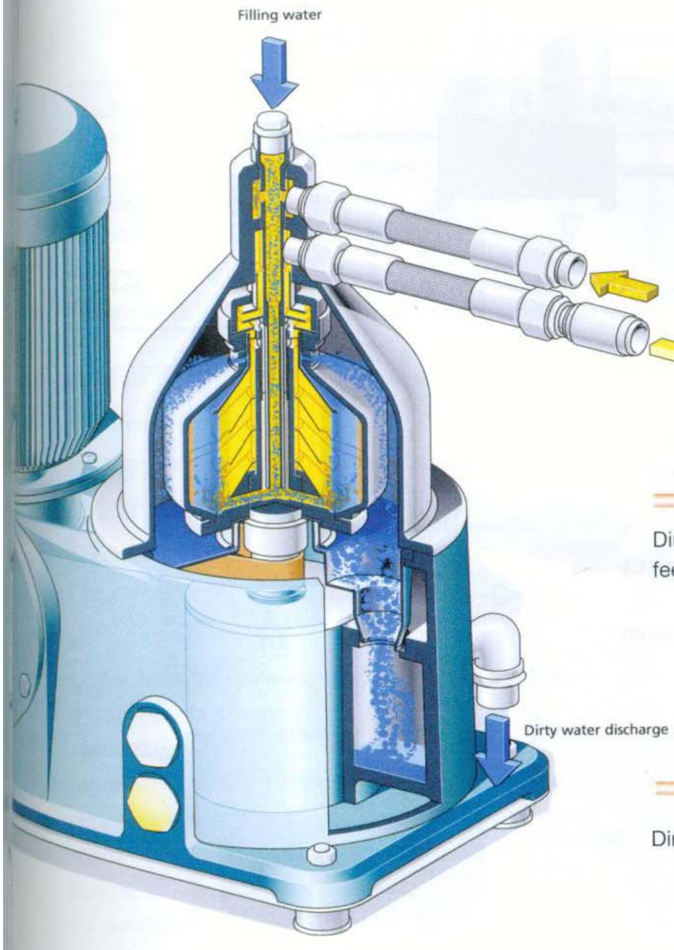
Features:
- UNITROL® system in one stage WMS and SMS
- Treatment of fuel oils up to 1.01 kg/l
- Regulating rings are not required
- High separation efficiency even in critical situations
- Automatic adjustment as clarifier or purifier
- Automatic bowl ejection
- Highly concentrated solids
- Continuous processing
- Self-cleaning effect of the bowl and the disc stack by total ejection
- Versatility of application
- Safe operation
- Belt drive
- Low noise level.

▲

The working principle of the Clarifier and Purifier of Westfalia.

left: operation of the Westfalia Clarifier which separates water and solids periodically by shooting

right: operation of the same separator as a Purifier which separates solids periodically and water continuously



▲

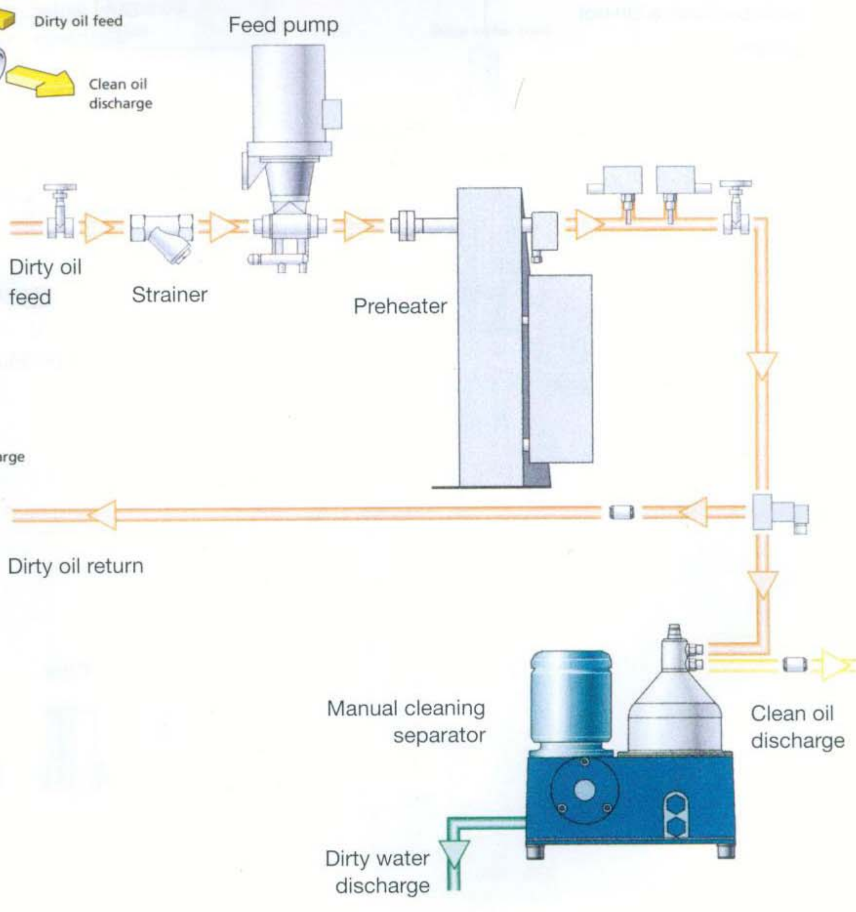
A cross-section of a simple separator.

M.D.O. usually has few contaminants; the separator can remain operational for several days before cleaning is required. From experience, it is learnt when cleaning is required.

▼

A simple system with a fuel pre-heater.

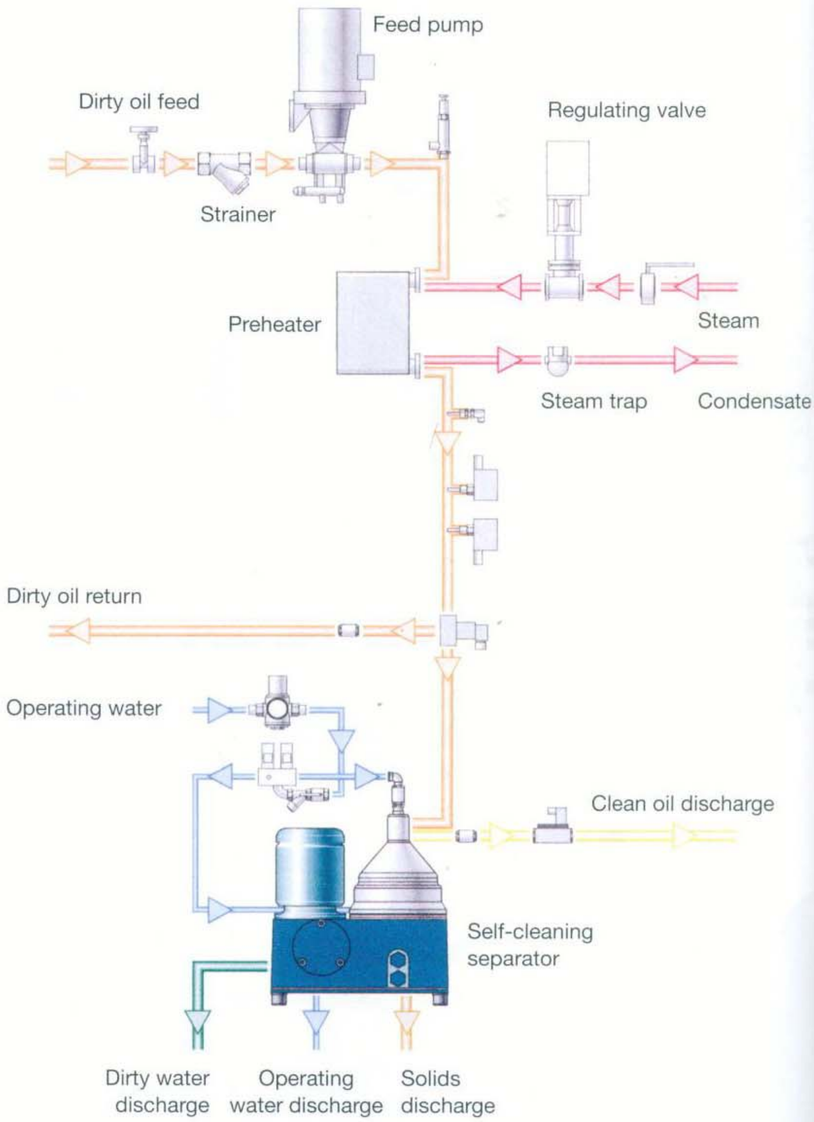
The correct oil-separation temperature can be maintained, also for instance, in arctic circumstances.



▶

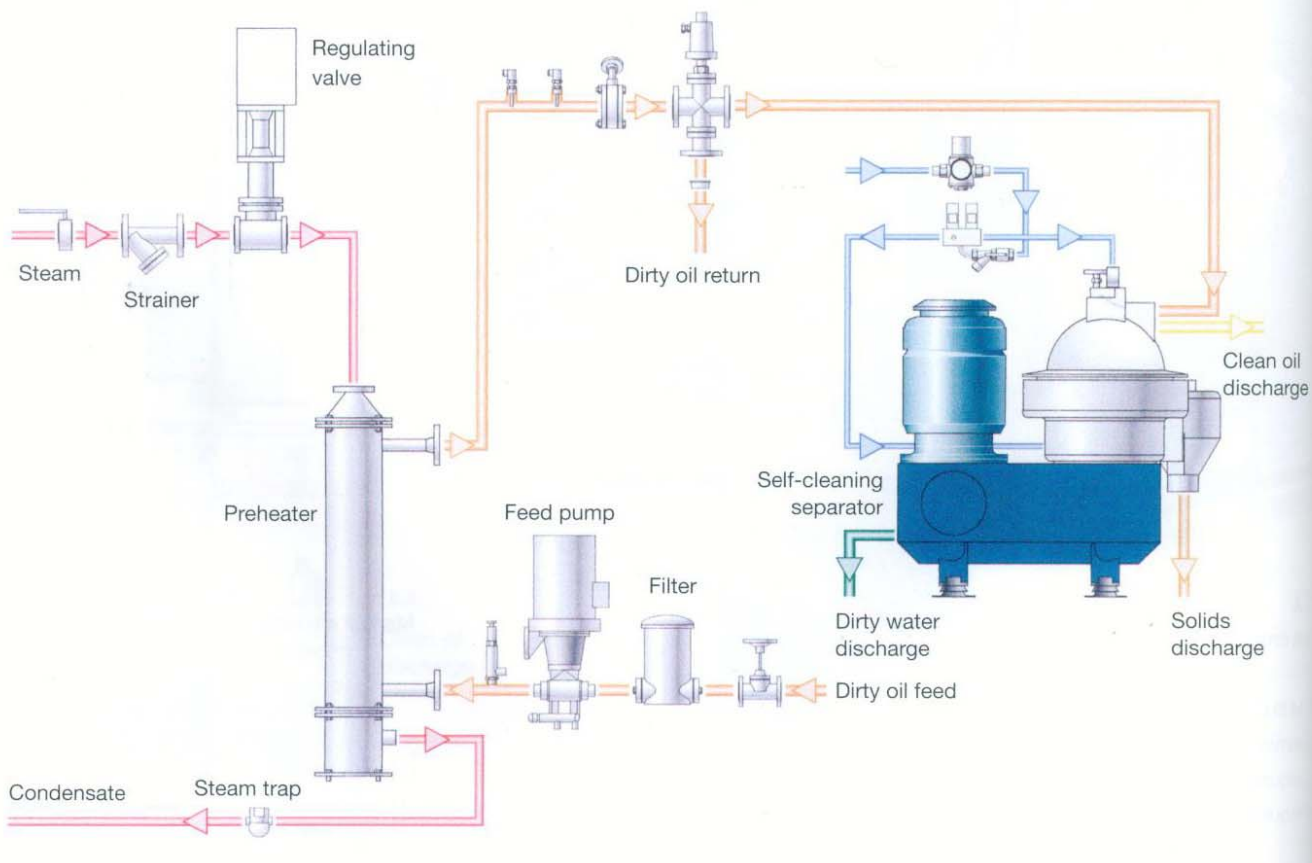
A complete system with pre-heater and a self-cleaning separator.

This version is common, for both M.D.O. and H.F.O.



▼

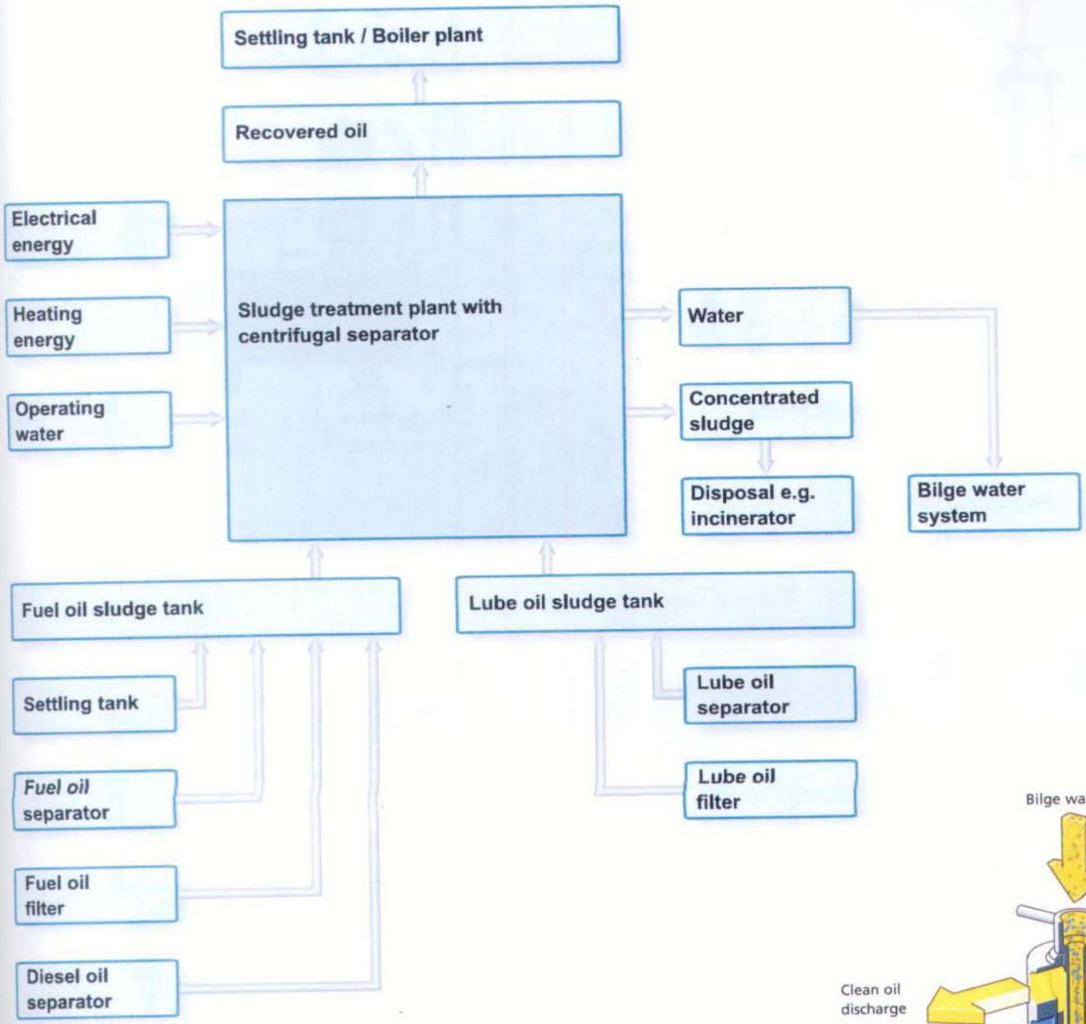
A similar system with a self-cleaning separator equipped with a Unitrol system.



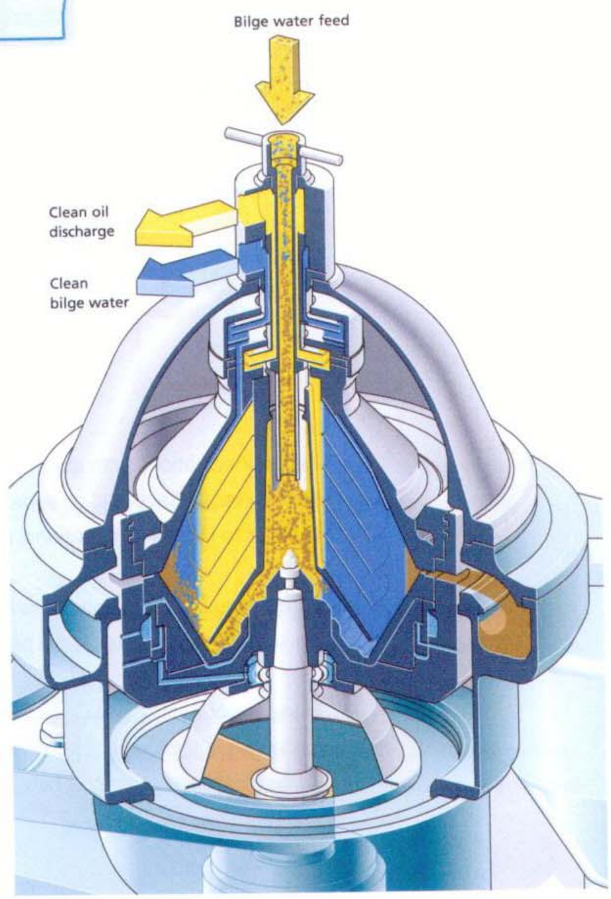
24.9.1 UNITROL system of Westfalia

This system constantly measures the water content in the clean-oil output and the amount of sludge in the sludge tank.

This makes specific-gravity discs superfluous and excludes human operating errors. Naturally, Westfalia also uses abbreviations such as:
WMS: Water Monitoring System;
SMS: Sludge Space Monitoring System.

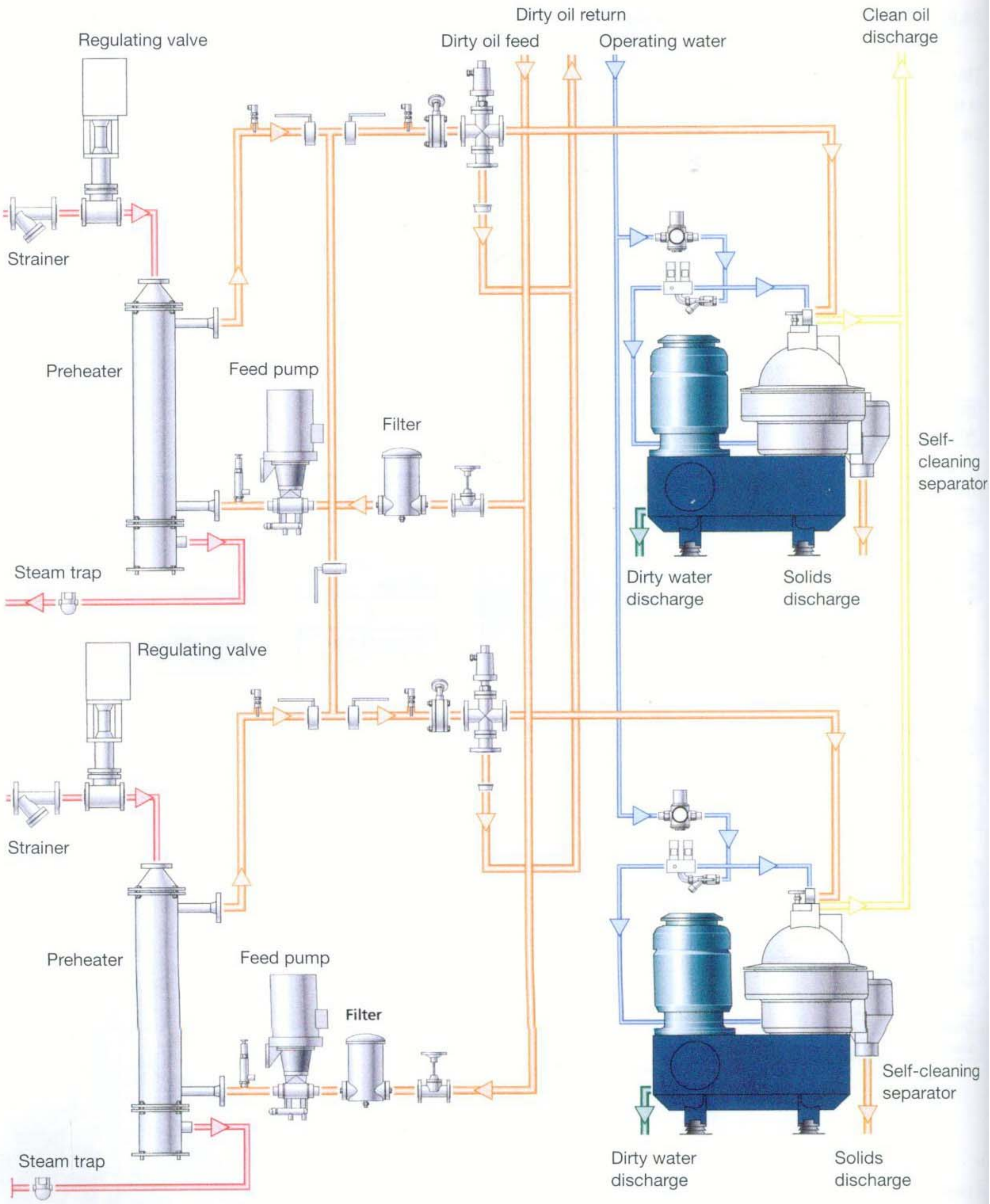


◄ A flow diagram of the complete sludge treatment.



► A bilge-water separator, virtually identical to the fuel and lubricating-oil separators.

Bilge water: this is water mixed with the dirt and oil which gathers on the bottom of the ship's engine room. The double bottom often has drain tanks in which the bilge water collects.
It is prohibited to discharge bilge water directly overboard (I.M.O.). Therefore bilge water is treated. A bilge-water separator and a strainer remove as much dirt and oil as possible.
A measuring device in the over-board pressure line prevents bilge water with too high an oil content, measured in p.p.m., being pumped over board. In this case a three-way valve returns the water to the bilge.



▲

Larger installations with marine diesel engines often have several heavy-fuel oil separators placed in series in order to increase operational reliability.

An arrangement of two separators in series which operate on the same supply line to ensure a purified oil discharge.

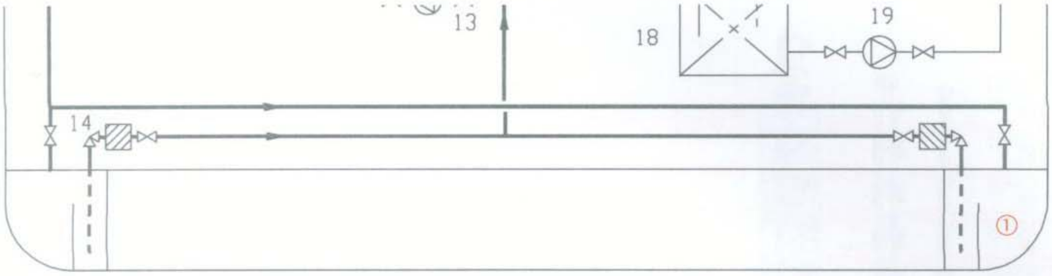
Sludge treatment

It is increasingly important to concentrate the sludge of various centrifugal separators because:

- at present, more often, the sludge must be disposed of on shore: it is prohibited to pump it overside. There are often costs involved in sludge disposal;
- part of the sludge can be re-used as fuel, which is a cost-saving.

The residues after post-treatment are:

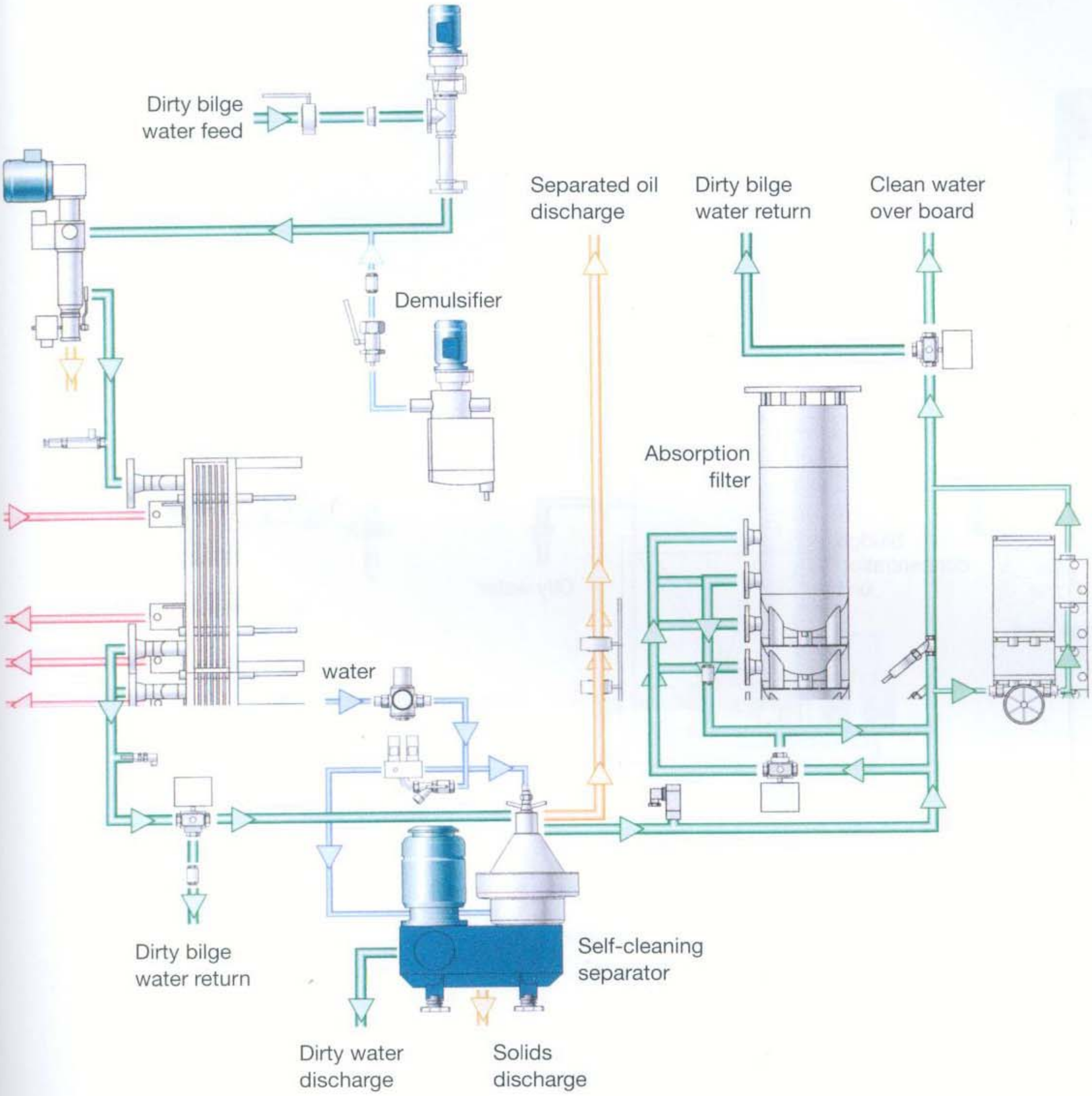
- **water**; this is pumped to the bilge-water cleaning system;
- concentrated sludge; this must be disposed of on shore by certified firms;
- dirt and such; this can be burnt in an **incinerator**.



Cross-section of a ship.

The bilges on port and starboard side in the engine room of this ship.

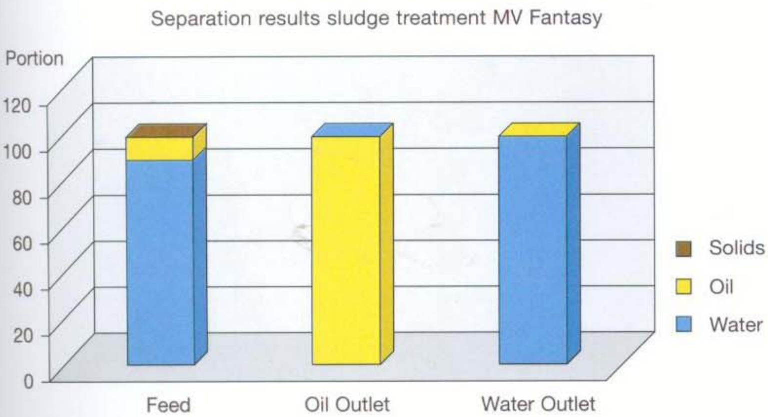
1 *bilge*



A complete bilge-water cleaning installation.

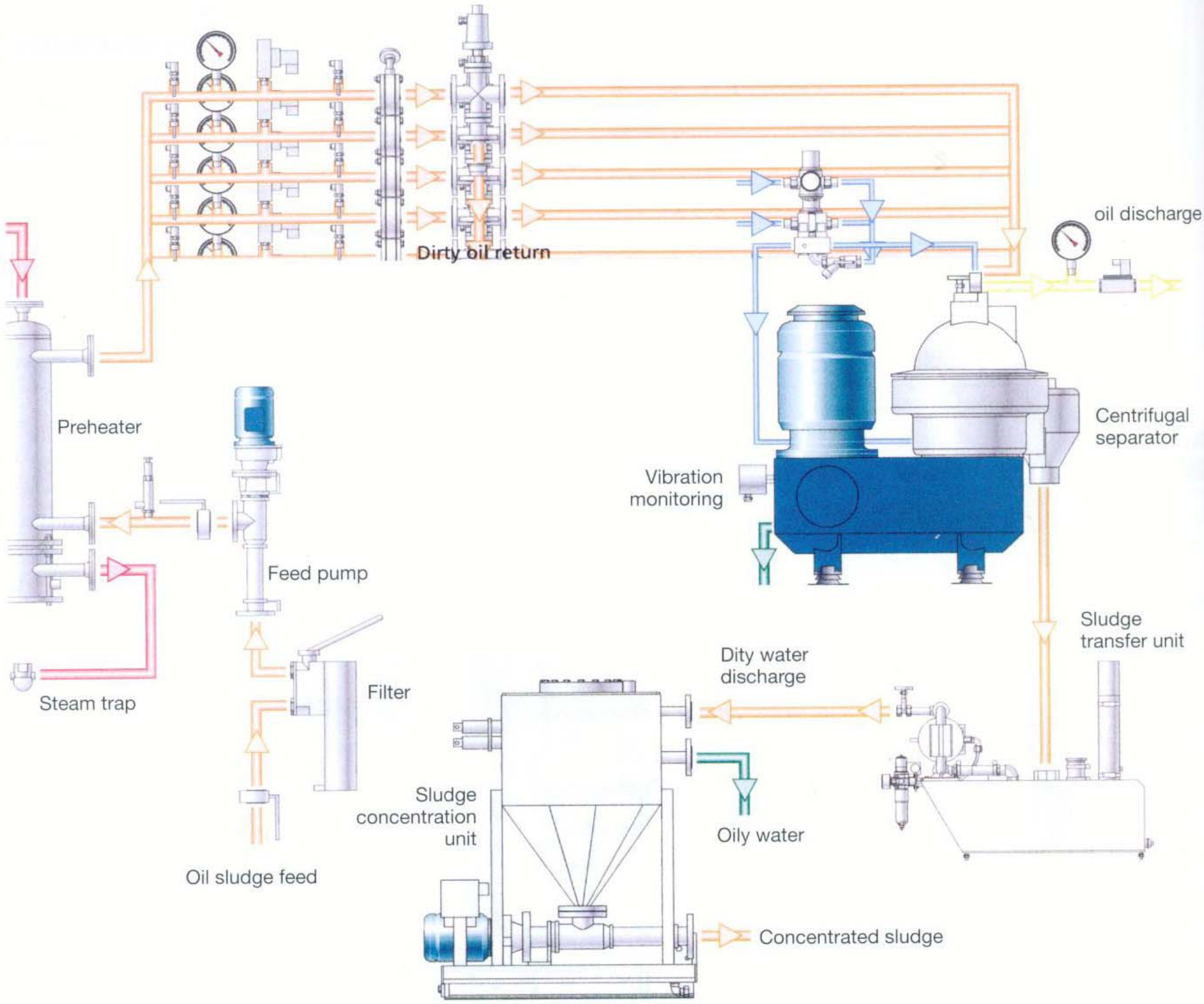
For adequate separation the bilge water must have a temperature at which the specific mass difference between water and oil is at its highest.

A special filter removes the last oil compounds from the water.



A graph of the effects of a bilge-water cleaning installation.

left: contaminated bilge water with 90% water and 10% oil
centre: oil outlet which consists of 99.6% oil and only 0.4% water
right: water outlet with 99.5% water and 0.5% oil
The 0.5 % oil percentage is cleaned in an absorption filter, so the quality of the 'over-board water' is compliant with all legal requirements.



▲

A flow diagram of a sludge-treatment system in the near future.

The sludge from the separator is treated to ensure as small a mass as possible. Eventually, sludge concentrate remains.

CH 25

Operational management and automation

25.1 Introduction

Today, an increasing number of engines in all categories are operated and monitored remotely. With modern communication technologies and general use of computers, the diesel-engine industry has changed dramatically over the last 25 years. It was common practice to have personnel on location when engines were operational. However, today the reverse applies: there is no one in situ.
At night the engine rooms of virtually all modern ships are unmanned. This applies to small coasters as well as to the largest container ships.
There are some exceptions, such as passenger vessels where legal requirements stipulate that personnel must be present in the engine room at all times.
The largest container ships with power outputs of 100,000 kW and a displacement of 70,000 tons can sail the oceans at 25 knots with one officer on the bridge, assisted by one crew member.
Relatively few accidents occur considering the number of sea-going and inland vessels plying the globe's waters at one time.

25.2 Automation of diesel engines

1 Diesel-engine data

All relevant information is displayed on the monitor and saved in a database. Therefore, it is possible to refer to the history which is extremely important when encountering problems.
The data can be sent anywhere, for instance, to the engine manufacturer or the ship's proprietor. The large engine manufacturers have a special service department continuously provided with relevant comparative data. This means that assessments can be made from a distance. This is also often the case with diesel-power plants.

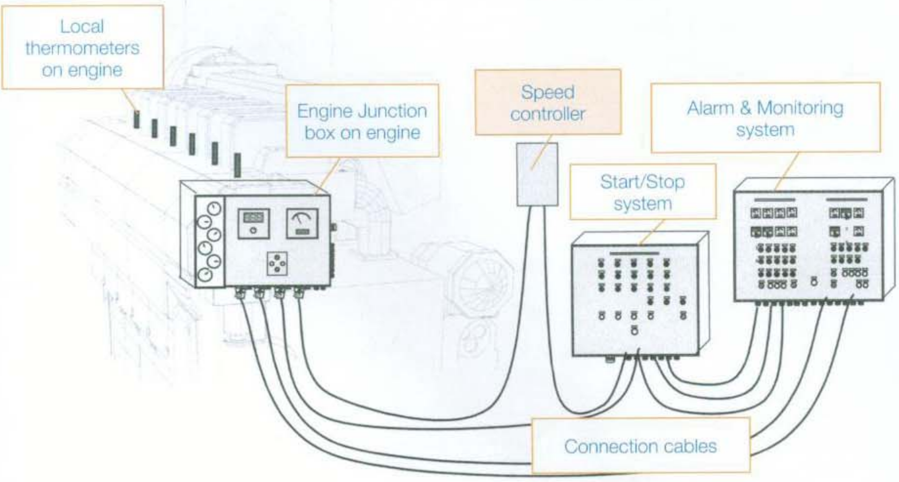
2 Operating the engine

This is remotely controlled: the engine load and speed can be adjusted or the engine can be put off line. This may occur in various ways, either mechanically, hydraulically, pneumatically, electrically, electronically or even with radio-controls.

3 Engine alarms

All important pressures, temperatures, loads, speeds, liquid levels and such are measured. The measuring gauges are fitted with an alarm, often with a pre-alarm. If there is no response a second alarm automatically intervenes in the process, if required.
In a small sloop engine only a few lamps will flash and a buzzer is activated. However, in a large marine engine, an elaborate and programmable flow-chart is put into action. The alarms in large installations are not restricted to the engine, but comprise all the auxiliary systems, such as those for cooling water, lubricating oil, fuel, refrigerating equipment and starting air.
The groups have been so categorised that the nature of the problem can be speedily detected.

25.3 Examples of automation systems

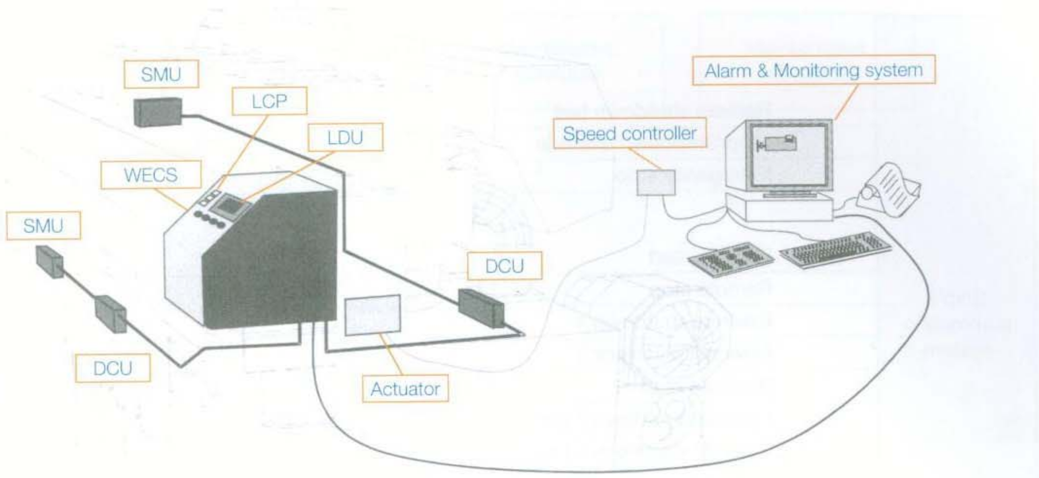


◄
Standard automation system.

The system that Wärtsilä can supply for a W 26 in-line diesel engine comprises:

- various types of meters mounted on the engine, such as temperature, pressure and speed;
- standard junction boxes on the engine with important values which can be read on the spot;
- remote-controlled starting and stopping system;
- remote-controlled alarm system;
- speed-control system.

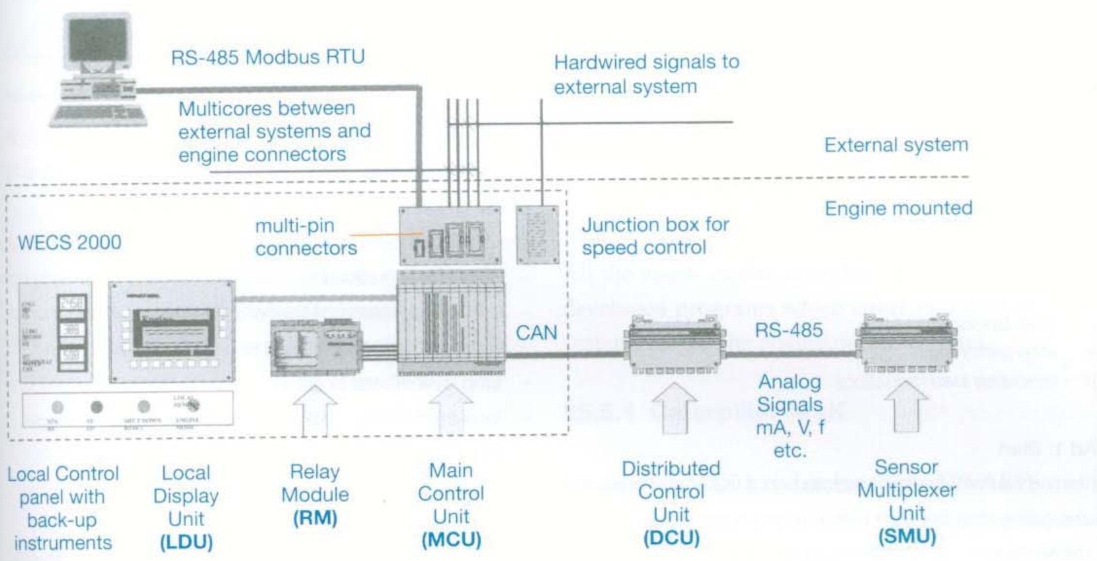
The engine is also provided with an 'Overspeed' module. Naturally, the actuator which regulates the fuel supply to the engine is also mounted on the engine. The speed-control and start/stop system are not mounted on the engine.



◄ The Wärtsilä WECS system.

This is an automation system for engines that registers all the data and monitors and ensures the safe operation of the engine.

Monitoring and alarm system



Wärtsilä uses the WECS system: the 'Wärtsilä Engine Control System'

This system has the following functions:

- measure analogue and digital signals;
- measure both the engine and the turbo-blower speed;
- engine protection;
- start/stop the engine;
- start-blocking;
- switch off engine automatically;
- adjust load;
- sensor alarms;
- reading important parameters on a monitor;
- exchange data with other systems such as general alarm systems and safety systems via MODBUS communication.

Abbreviations used by the WECS 2000 system

D.C.U.'s: Distributed Controls Units, a sensor interface based on micro-processors.

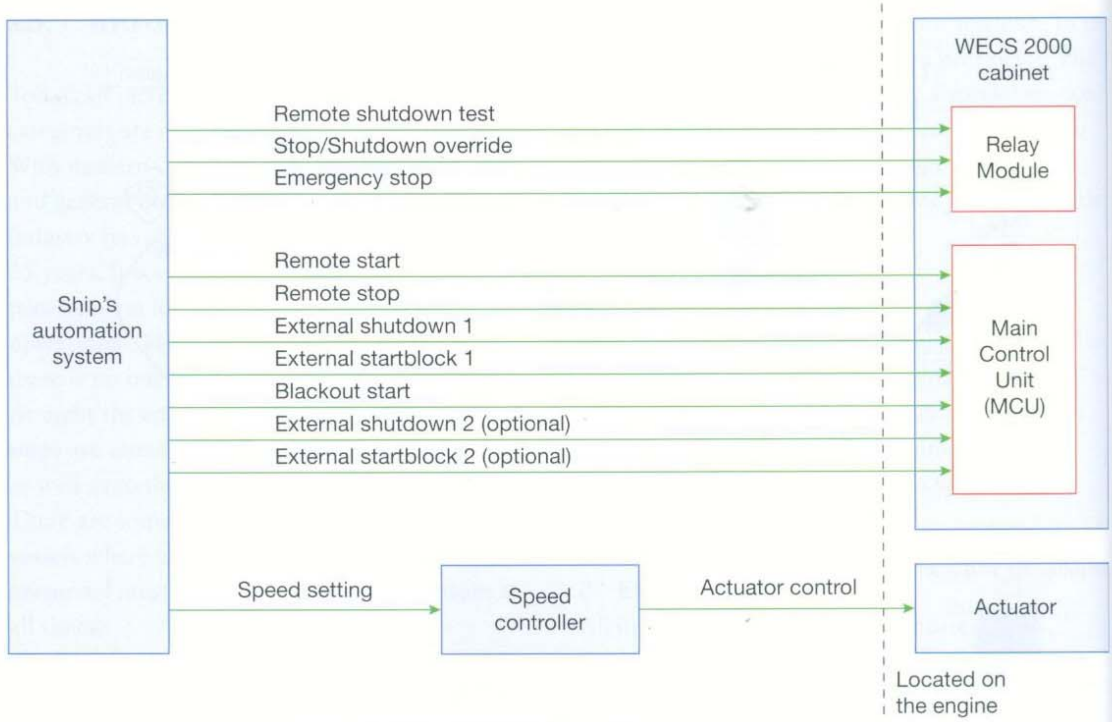
S.M.U.'s: Sensor Multiplexer Units, a sensor interface based on micro-processors.

M.C.U.: Main Control Unit, central main control unit.

R.M.: Relay Module, a safety back-up system. This is required by many classification societies.

L.D.U.: Local Display Unit, a monitor on location, often vibration-free suspended from the engine.

L.C.P.: Local Control Panel, an on-site panel with important data concerning the engine.



▲
The safety system in WECS 2000.

There are five elements:

1 *start*
2 *stop*
3 *start-blocking*
4 *emergency stop*
5 *executing load reductions*

Ad 1: Start

Starting can be remotely controlled or at the engine. After an emergency stop the reset button must be pressed.
'Black-out start'. This is an emergency start which is selected despite the fact that starting may cause damage to the engine. Only when the turning engine is in operation, the engine will not start. The pre-lubrication operation can be by-passed during five minutes.
Emergency engine start. This can be done at the engine.

Ad 2: Stop

Normal stop.
Emergency stop. This can be mounted in various places.

Ad 3: Start-blocking

Blocking the start.
The start is blocked under the following circumstances:

- engine is operational;
- low pre-lubricating oil level of the turbo-blower;
- turning engine is on stand-by;
- stop lever in stop position;
- local start control is set on remote control, which blocks the start at the engine;
- emergency stop is active;
- external start-blocking.

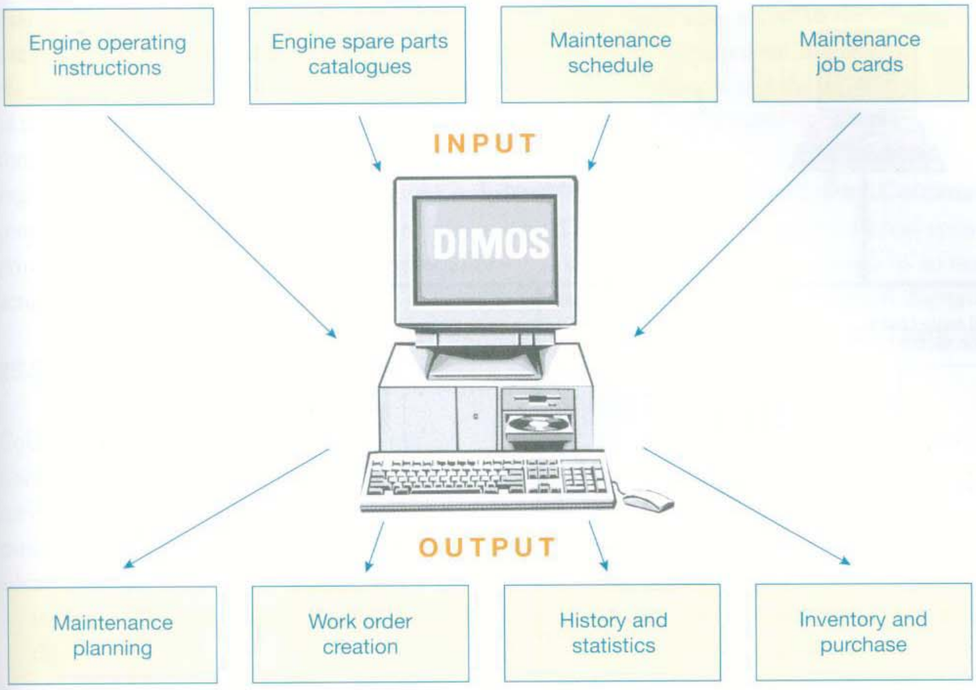
Ad 4: Emergency stop

Similarly to a normal stop, an emergency stop is performed by activating two solenoid valves. An emergency stop must be reset prior to the start.

Ad 5: Performing load reductions

There are many options, such as:

- high lubricating-oil temperature;
- high coolant temperature;
- high bearing temperature;
- high exhaust-gas temperatures.



▲
The diesel management system DIMOS of Caterpillar.

A complete system for maintenance, spare-part control management, statistics and expense monitoring.
All conceivable information is saved and processed during operation. Operating hours, preventative maintenance action and breakdowns are categorised and stored here.

25.4 Operational management

For operational management of diesel engines on ships and in power plants, it is ideal that the data from the installations can be stored in computer systems.
Where only a handwritten engine journal and operating hour's lists for all the instruments was once available, one can at present save and process all the relevant data and use them in, for instance, the maintenance system, the spare-parts storage, management of the propulsion installation or the power generation on ships or shore.

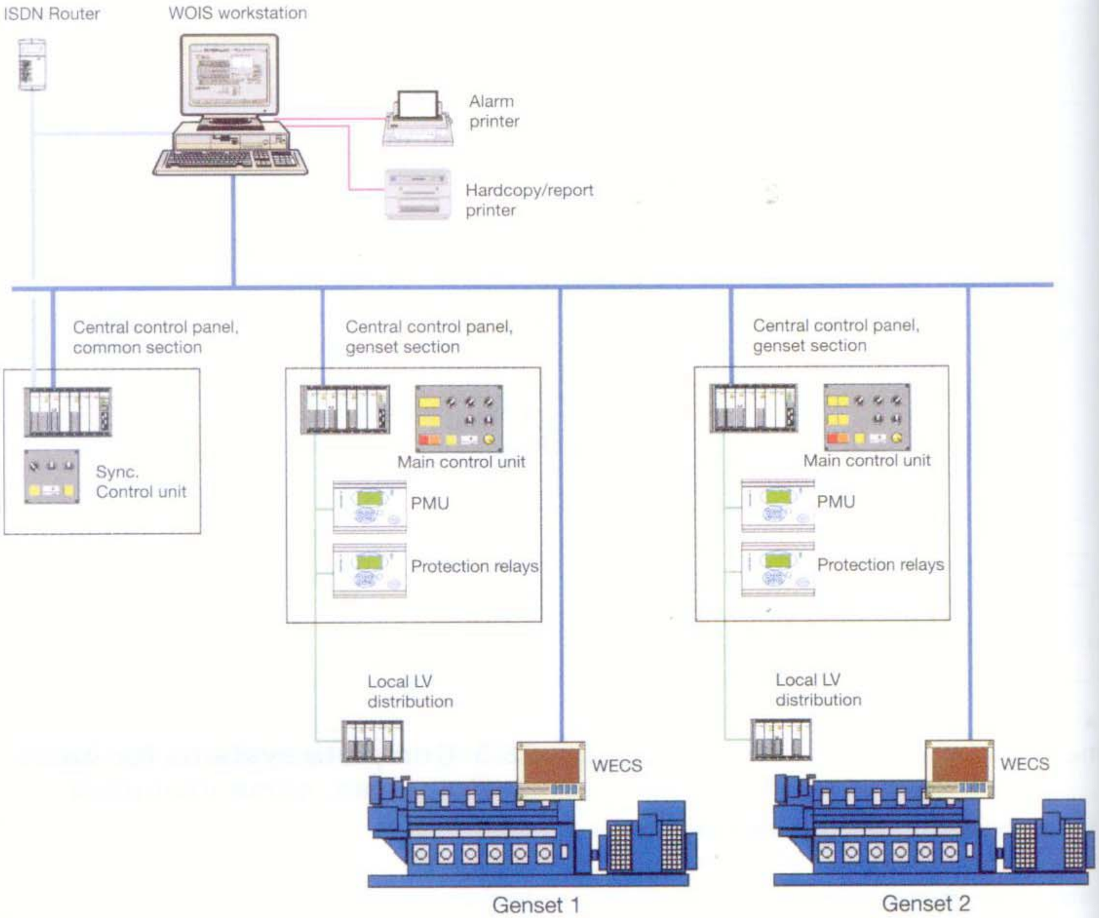
25.5 Complete systems for diesel engines, some examples

All the major engine manufacturers have developed programs which cover monitoring systems for all the engine operating data.

25.5.1 Caterpillar-MaK

Caterpillar-MaK has developed the DICARE-system.
According to Caterpillar-MaK this system has the following advantages:
- early discovery of wear and tear;
- optimal operational control as all the alarms generated by data that deviate from normal conditions are made clearly visible;
- reduction in maintenance costs by using predictive-trend analysis;
- longer life of parts by comparing current and required values;
- information with regard to the maintenance status of the entire engine;
- allows personnel to anticipate the planning of maintenance and spare-part ordering.

The engine management system at Caterpillar is known as MaK DIMOS. This is a computer-supported maintenance and spare-part management programme.



▲

A complete system for Wärtsilä generators.

To every diesel genset, a pc and monitor is attached showing all the data.
In the Wärtsilä Operators Interface System (W.O.I.S.) in the control room of for instance a diesel-power plant (or the generator of a ship) all the operating details are stored and alarms processed.
This W.O.I.S.-system can be connected to various control systems available in the industry.

25.5.2 Wärtsilä Faks system

Faks–Fault Avoidance Knowledge System is an elaborate software system for monitoring and fault avoidance for large engines. The system retrieves various values from the monitoring systems of the engine, and analyses these for possible failures. This is not 100% watertight, but nevertheless indicates what and where the most likely fault will be and when it will occur.

The system:

- safeguards the engine by using early warning systems;
- performs measurement inspections;
- monitors the maintenance status;
- performs research using the knowledge base;
- communicates with other web-enabled applications.

For example, special sensors measure the exhaust-valve seat temperature, the temperature of the cylinder liner near the combustion space and the surface temperature of the main bearing.

Faks is the end result of years of experience with engine data and fault-finding and-remedying. All the data can be sent to the service department of the engine manufacturers via modern communication systems. Subsequently, the engine manufacturers can either on request or by contractual obligations perform fault analysis and provide advice to prevent or resolve potential or actual problems.

25.5.3 CoCos system of MAN–B&W

CoCos is the abbreviation for: Computer Controlled Surveillance, a computer-controlled surveillance system. According to the manufacturer, this system has the following advantages:

- increase of the availability and the reliability of diesel engines;
- considerable reduction in the operating costs;
- effective planning of engine maintenance;
- easy and simple identification of spare parts;
- integrated stock control.

The CoCos system comprises four parts:

1 CoCos–EDS – Engine Diagnostics System

An engine diagnostics system that simplifies decision-making on board, at the diesel power plant or in the office.

- it assists in making decisions regarding measures that are to be taken;
- it improves the availability and reliability of diesel engines;
- it reduces operating costs and losses due to engine failure.

These main objectives are attained by:

- collecting, monitoring and storing operating data;
- analysis of the stored data;
- timely detection of irregularities.

2 CoCos–MPS – Maintenance Planning System

An all-in-one maintenance system which ensures the timely execution of preventative diesel-engine maintenance, and maintenance of auxiliary equipment (in the proximity of the engine). The system works in conjunction with CoCos–SPC and CoCos–SPO.

This allows easy access to the number and type of new parts required for maintenance, the amount of parts in stock and the amount of parts to be ordered immediately.

3 CoCos SPC Spare-Part Catalogue

The spare-part catalogue is a good system to track of the number of parts in stock in an orderly manner. The detailed records are digitally available including all the drawings and diagrams.

Main objectives of the system are:

- to chart the engine parts and other equipment in a simple fashion
- to provide extensive information with respect to these parts.

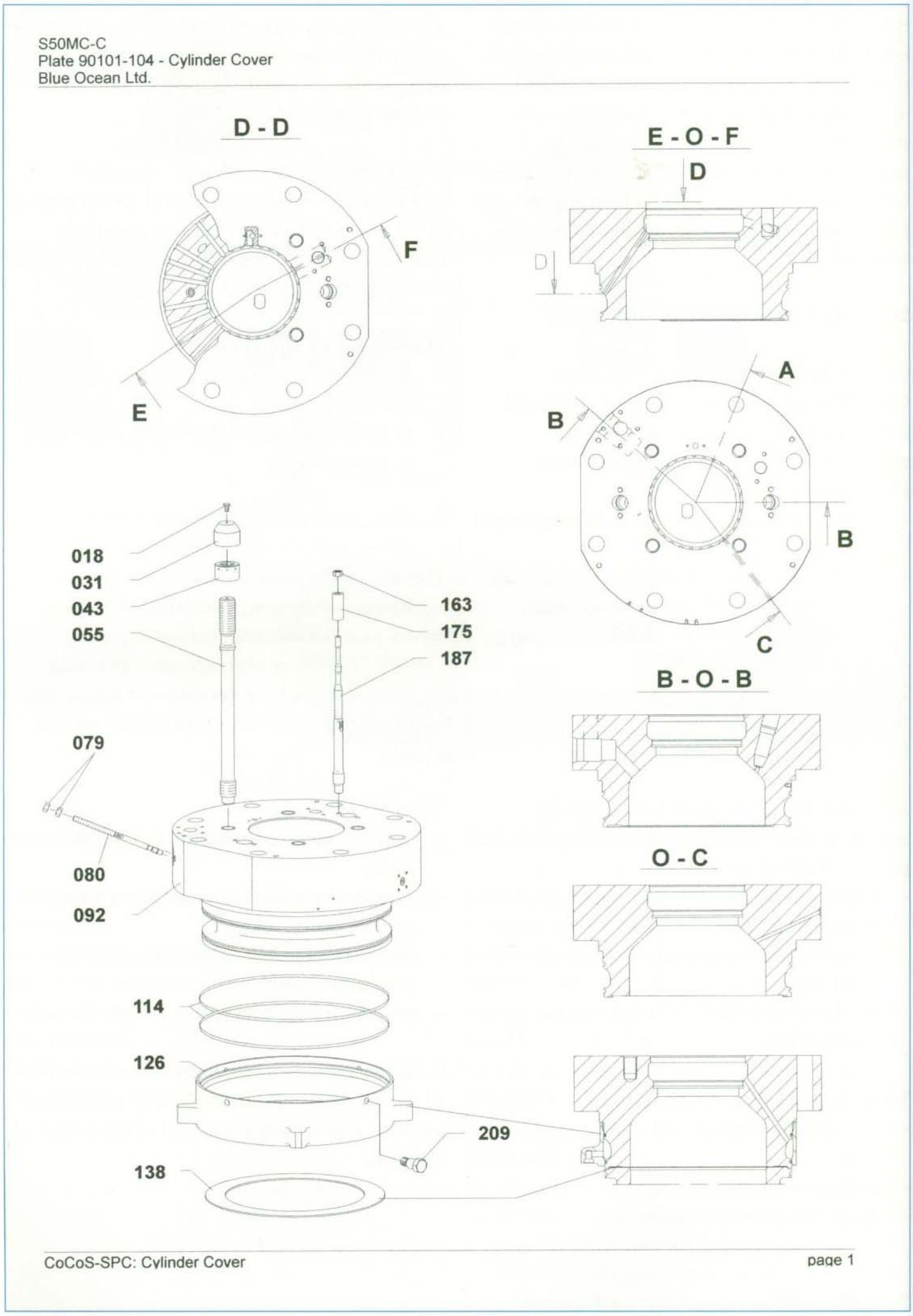
4 CoCos–SPO – Stock-control and spare-part procurement

This sets up the stock administration and registration of the spare parts effectively. This system provides accurate updated details of location and history of components including spare parts in stock for the engine or power plant. This system is elaborate and, of course, entirely digital.

The main objectives:

- provide assistance in handling the spare-part stock;
- provide up-to-date information regarding current stock;
- forecast spare-part availability for future overhauls;
- provide assistance in spare-part procurement.

If this CoCos system is used correctly, it provides all the necessary information so that an accurate overview regarding stock control of spare parts is obtained.



▲

The beginning of the CoCos system.

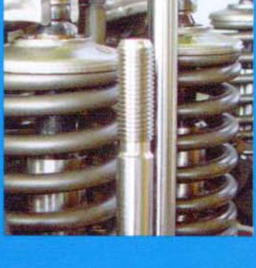
All the parts are numbered. Shown here, the cylinder head of a two-stroke crosshead engine type MAN–B&W S50 MC-C. If one clicks the numbers, the details of the part are listed according to the assembly / sub-assembly, including the tools required for assembly and dismounting, stock-control numbers can also be obtained. Ultimately, a very complete system for the operation of diesel engines.

Comment

Naturally, it is of the utmost importance to include all the modifications of each part of the CoCos system into the system. Once again man is the weakest link!

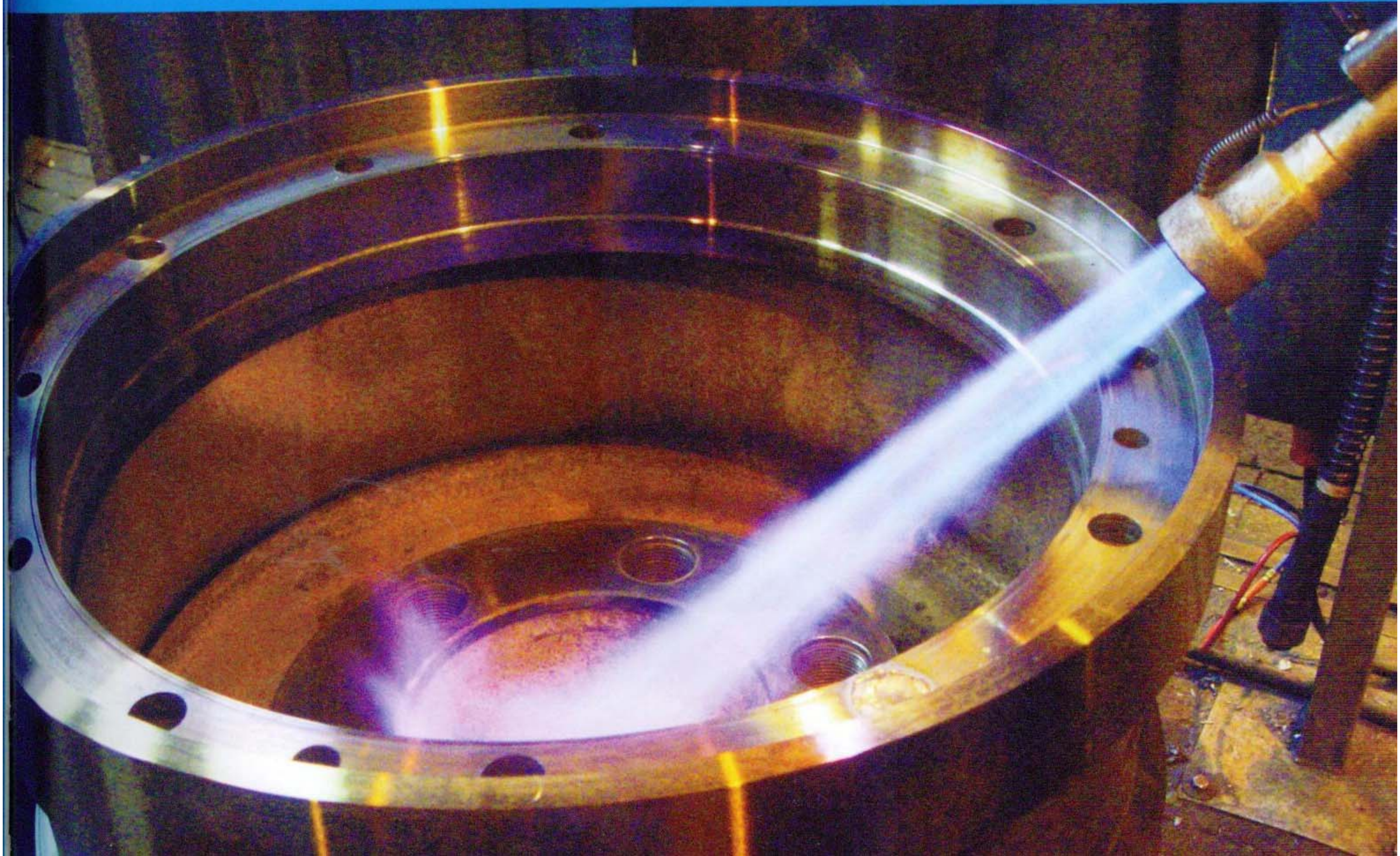
CH 26

Reconditioning engines and their parts



Reconditioning engine parts requires specialist knowledge. The overhaul and repair of worn parts must be technically possible as well as financially attractive in relation to the cost of purchasing new parts. Mark van Schaick, Schiedam, The Netherlands,
has an extended and unique machine park, such as this Schmaltz crankshaft lathe used to grind a crankshaft of a fourteen-cylinder MAN 52-55 medium-speed four-stroke diesel engine.

Keeping a two-stroke crosshead-engine piston warm before rewelding at the MAN Diesel AG Reconditioning in Hamburg.



26.1 Introduction

Mechanical, thermal and chemical factors cause engines to become worn. The worn parts must be replaced by new ones or reconditioned within a certain time span.

Several examples of parts that require replacement are:

- cylinder heads, especially the exhaust valves;
- pistons and piston rings;
- cylinder liners;
- shafts en shaft bearings;
- super-charging groups;
- pumps;
- governors;
- starting motors;
- fuel pumps and injectors.

▲ An exhaust valve affected by high-temperature corrosion.

◄ Worn parts.

After a certain number of operating hours or after having established that particular parts no longer function properly, they must be inspected.
Shown a disassembled turboblower.

▲ A filthy turboblower casing.

▲ Soiled exhaust valves are checked for wear, hair cracks and size, after cleaning.

◀
Exhaust-valve seats are heavily loaded.

AFK HAARS = scrapped after hair-crack test. Hair-cracks result in valve leaks or fractures.

◀
Various engine parts requiring maintenance. This is referred to as 'overhauling' or 'reconditioning'.

◀
Bores at the big end for the crankpin and the eye for the piston pin of connecting rods often require trueing.

▲
If repairs are postponed for too long, an exhaust valve may start to leak severely, the so-called blow-through.

▲
A microscopic view of an exhaust valve on the combustion-chamber side, seriously affected by high-temperature corrosion.

Engine manufacturers provide specifications with regard to the normal life of the various parts, and the engine proprietors or operators put together a maintenance plan based on this information.
Regular maintenance is performed at the most favourable time in shipping: during dry-docking, surveys or while the ship is in port.
Diesel power plant maintenance is executed preferably when there is a reduced electricity demand, for instance, at night or in weekends.

In the case of failures or damage, action must immediately be taken.
In the discussion on the reconditioning of engine parts, the following subjects come up.

Reconditioning:
- cylinder heads including inlet and exhaust valves;
- pistons and piston rings;
- cylinder liners;
- crankshafts and bearings;
- camshafts and bearings;
- geared wheels;
- fuel pumps and injectors;
- turboblowers or supercharging groups;
- governors;
- engine blocks;
- coolers;
- pumps;
- connecting rods and piston rods.

The four-stroke engines will be discussed first and subsequently the two-stroke crosshead engines.

26.2 Four-stroke engines

26.2.1 Cylinder heads with inlet and exhaust valves

Valves

Perfect combustion of fuel (100%) is purely theoretical. In practical terms, the fuel combustion is approximately 96 to 98%. This means that a

▶
The lifetime of parts may vary considerably. Fuel quality is a deciding factor.

Component	Time between overhauls		Expected lifetime [h]	
Fuel quality	HFO	MDO	HFO	MDO
Piston	12000 – 20000	20000 – 24000	24000 – 40000	40000 – 48000
Piston rings	12000 – 20000	20000 – 24000	12000 – 20000	20000 – 24000
Cylinder liner	12000 – 20000	20000 – 24000	60000 – 100000	60000 – 100000
Cylinder head	12000 – 20000	20000 – 24000	60000 – 100000	60000 – 100000
Inlet valve	12000 – 20000	20000 – 24000	24000 – 40000	40000 – 48000
Exhaust valve	12000 – 20000	20000 – 24000	12000 – 20000	24000 – 32000
Injection valve nozzle	2000	2000	4000 – 8000	8000
Injection pump	16000	16000	16000 – 24000	32000
Main bearing	16000 – 20000	16000 – 20000	32000 – 40000	32000 – 40000
Big end bearing	12000 – 20000	20000 – 24000	12000 – 20 000	20000 – 24000

small amount of the fuel either does not burn or does not completely burn.
Unburnt carbon particles are hard and cause damage in the combustion space.
Other heavy-oil components produce high-temperature corrosion, (H.T.C.). The **Vanadium** in fuel is chemically bound with sodium from (sea) water and forms a sticky slag, which can affect the oxide coating on the valves.

Also, see Chapter 8, Fuels, fuel-line systems and cleaning fuels.

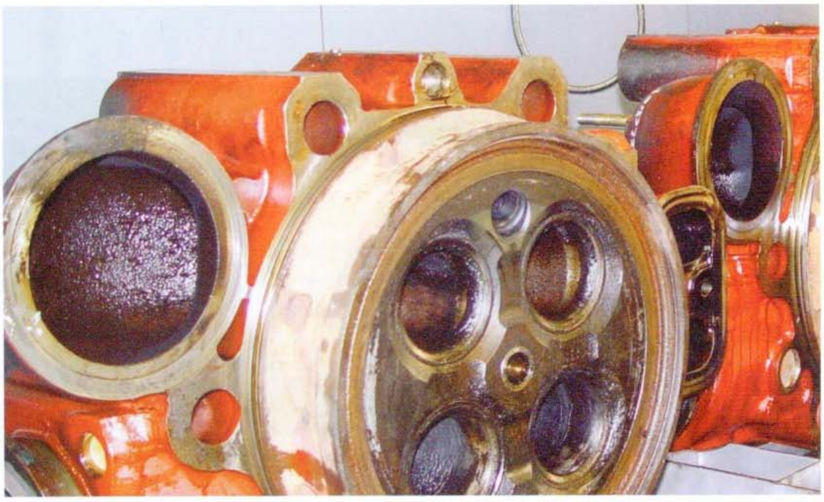
Moreover, the slag is oxygen-rich and therefore forms an extra oxide coating. This process is continuously repeated, in an aggressive, rapid fashion.
Special valve materials may be able to prevent this process.

Also, see Chapter 7, Use of materials for diesel engines.

An RPM of 1500 comprises 750 complete processes in a four-stroke diesel engine, one every second revolution. Therefore, valves are opened and closed 750 times! At a life expectancy of 15,000 operating hours for valves, this amounts to 750 × 60 × 15,000 = 750,000,000 times!

Key points for exhaust valves

Valve seat: pitting due to the impact of **solid particles.** Blow through, wear and tear.
Seat in the head: ditto, pay attention to cracks.



▲
Cylinder heads after disassembly of the valves and valve drives, ready for cleaning.

Valve disc: corrosion of material by H.T.C., usually visible by porous surface material.
Valve stem: damage to the surface; if present, crumbling of the chromium top layer; top of the valve stem: crater forming due to valve drives.

Key points for inlet valves

Obviously, these do not experience as high a pressure as the exhaust valves. The airflow is cool and uncontaminated.
Valve seat: check for pitting and wear.
Seat: ditto, pay attention to cracks.
Valve stem: Check for wear, pitting and damage to the top of the stem.

▼
High-temperature corrosion frequently occurs in engines running on H.F.O..

The valve material exhibits craters and cracks.

▼
Two cleaned multi-cylinder heads of Caterpillar engines.

▶
A conveyor washing machine for cleaning engine parts at Engine factory Bolier in Dordrecht, The Netherlands.

Carbon deposits and other solids are removed effectively.

Procedure for reconditioning valves

- Cleanse in a special washing machine, tank or other device with a carbon and degreasing agent.
- If required, scrape off the hardened layers, blast-clean them in a blasting box.
- Check for surface damage, burns and cracks.
- Check the quality of the sealing surface. Often a thin layer of valve material has peeled off; this occurs in, for instance, rewelded valves.

▼
A dry-blast cabinet for cleaning parts.

1 *entry hatch*
2 *viewing window*
3 *gloves for handling the parts for blasting inside the cabinet*

▼
After cleaning, inlet and exhaust valves are inspected for hair-cracks and the dimensions are checked.

▶
Installing the inlet and exhaust valves in reconditioned cylinder heads of Caterpillar – MaK 32 medium-speed heavy-fuel engines.

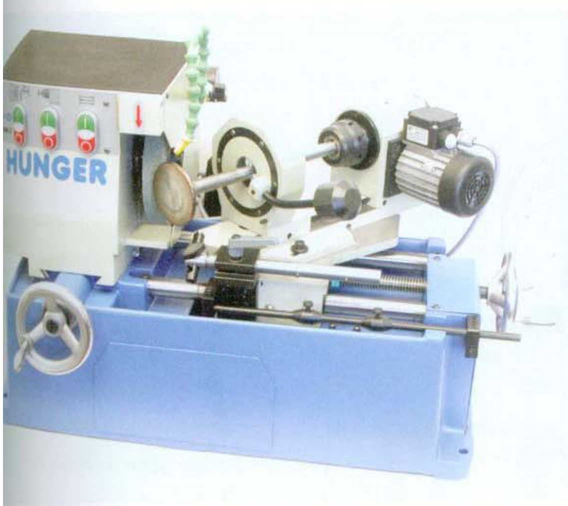
The valves are polished by hand when mounted on the new seats.

Trueing up the valve seat

Essentially, two methods are used.

1 Grinding the seat with a grinding machine.

2 Turning the seat with a chisel housed in a cutting device.



▲

A valve-grinding machine.

The valve seat is ground at the correct angle until it is smooth and clean.

▲

This machine enables the valve seat to be reconditioned using a chisel.

Here a seat with a new valve guide used as the centre line is overhauled using a rotary cutting machine with a chisel mounted on the valve guide.

◀

Grinding of a valve seat using a grinding stone mounted on a grinding machine.

The grinding angle is crucial.

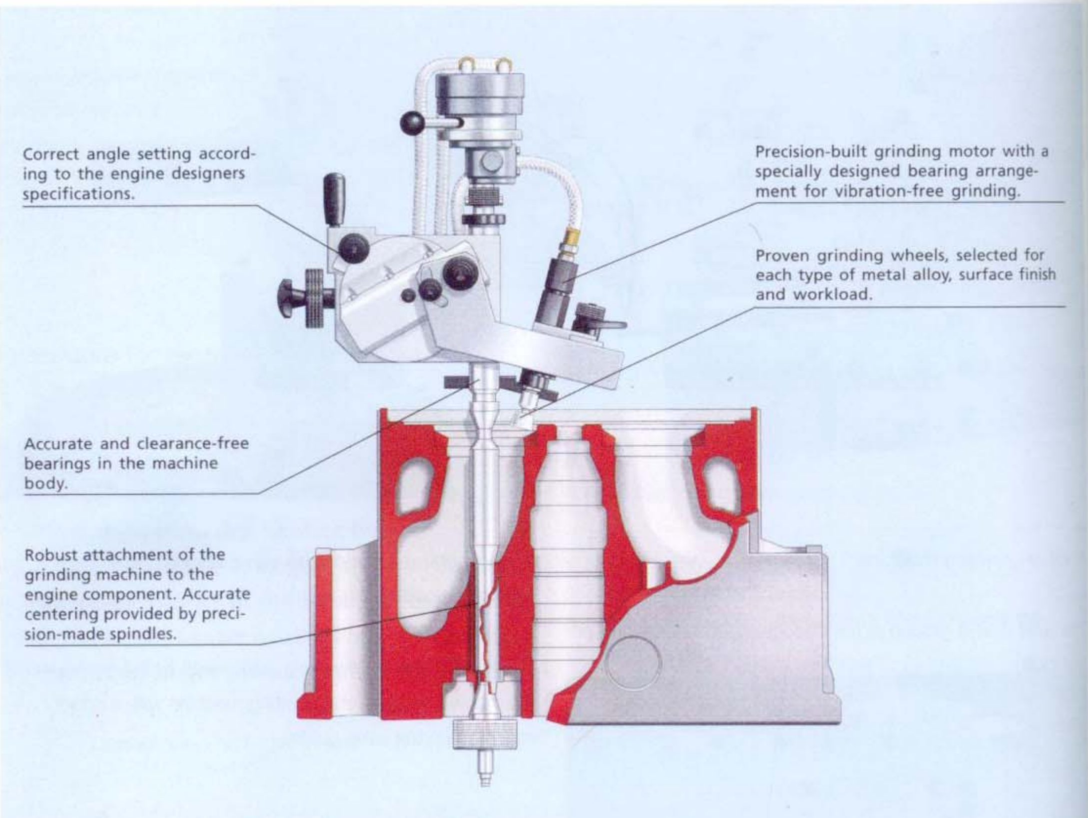
Trueing up the valve-seat casing

The machining is similar to that used for the valve seats:

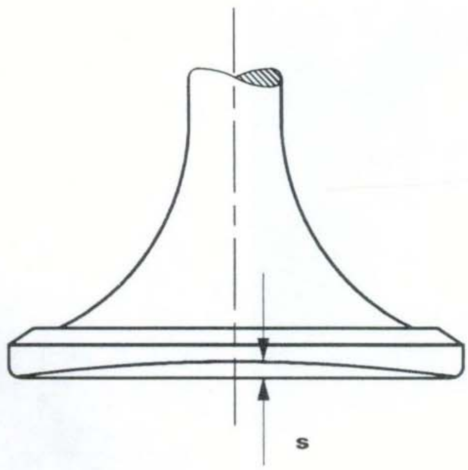
1 grinding
2 turning using a small chisel.

▶
A valve-seat grinding machine.

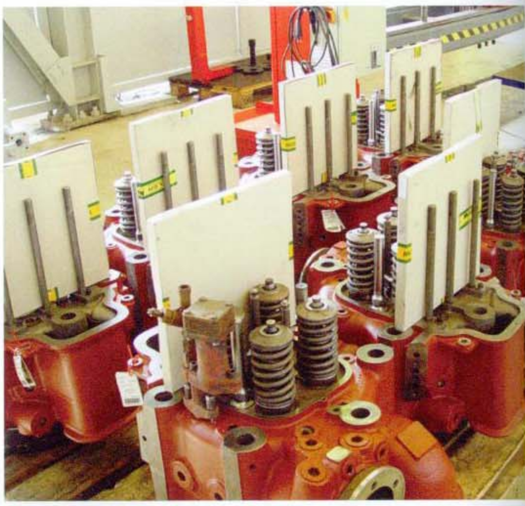
Small grinding stones are used.



Chris-Marine Valve Seat Grinding Machine type MSD on a 4-stroke cylinder head.



▲
S is the maximum burn depth for a valve before scrapping.



▲
Cylinder heads after thorough reconditioning, ready to be mounted on an engine.

The cardboard envelop contains a complete packing set. Each head is provided with a data label.

Drives

Obviously, a general overhaul of the cylinder head includes inspection of the valve drives and the valve lever mechanisms.

Procedure

- degrease the parts
- inspect for damage, cracks and pitting
- measure the pivot clearances
- restore excessive clearances

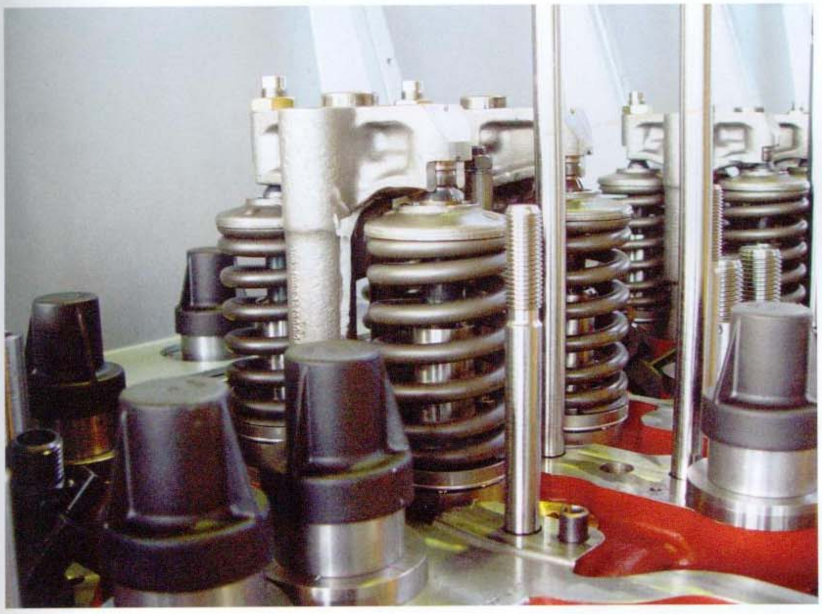
Bushes of, amongst others, valve levers are usually renewed.
Valve springs are checked for fractures and the relaxed spring length is measured. If the springs have become too short due to material fatigue, they are scrapped and replaced.

Push rods must be straight. Rolling them along a level surface shows if they are bent.

Cylinder heads

Due to increasing pressures in the cylinder, the mechanical load on the cylinder heads is constantly increasing. Most four-stroke engines with individual cylinder heads use **four head-bolts,** which in large cylinder bores are hydraulically tensioned. The cylinder-liner seal is heavily loaded requiring a precise fit of the surfaces.
Both can be machined in various ways. The cylinder head on the cooling-water side must also be cleaned and meticulously inspected.
The heads are submerged in a cleansing tank, for the removal of deposits. Prior to inspection, they are rinsed with clean water for the removal of residual solvents.
The freeze plugs are re-fitted.

The cylinder-head flame plate is checked for burning, corrosion and cracks.



◄ Valve springs and the entire valve movement require thorough inspection for damage and corrosion.

◄ Cylinder heads of a Caterpillar-MaK engine under construction.

1 *freeze plugs*

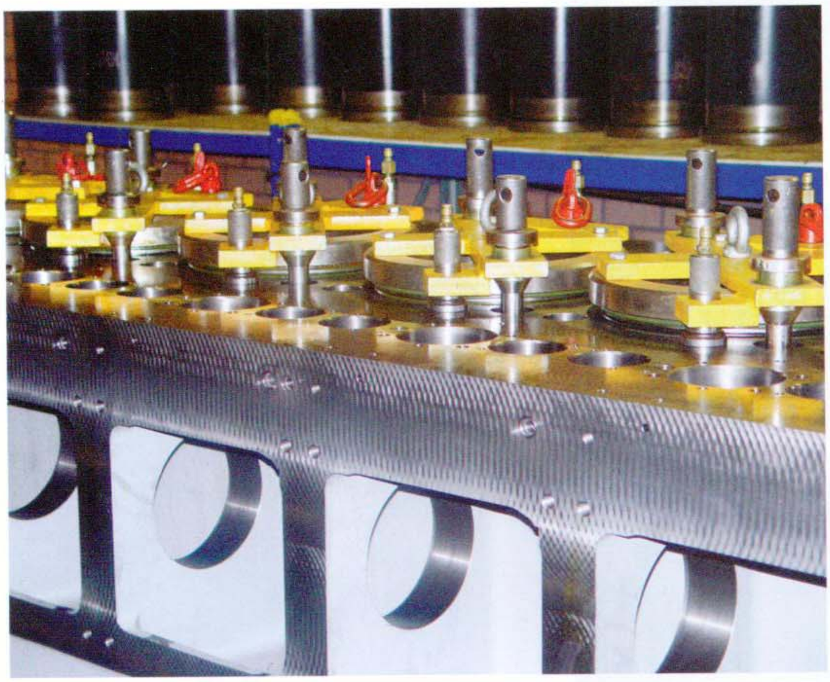
The cylinder-head flame plate.

1 *inlet valves*
2 *exhaust valves*
3 *bore for starting-air valve*
4 *bore of fuel injector*
5 *bore of relief safety valve*
6 *bore for indicator cock*

A damaged cylinder head.

Clearly visible is a piece of material, probably part of an exhaust valve, lodged between the cylinder head and the piston.

The cooling-water spaces of the heads are 'pressure tested' with clean tap water to check for leaks caused by tearing, covers or freeze plugs.



Pressure testing the cooling-water space containing the cylinder liners of a newly built medium-speed four-stroke H.F.O. engine.

The yellow glands are used to place the cylinder liner under the same pressure as the cylinder heads will exert on them when assembled. For 24 hours the cooling-water pressure is checked for pressure loss. Above a certain pressure-reduction percentage it is concluded that leaks have occurred.

◄

Valve seats are cooled in liquid nitrogen to a temperature of 180 °C below zero to facilitate mounting in the cylinder heads.

When the seats have adopted ambient temperature, they are then firmly fixed. Top right is the starting-air supply bore. In the centre of the head is the fuel-injector bore.

Valve seats are usually nitrogen-cooled and then fitted. When the seat is heated, it expands to fit snugly in the cylinder head.

▲

An insulated liquid-nitrogen drum.

◄

Levelling a cylinder head with a portable planing machine.

In multiple or multi-cylinder heads the sealing surface is often refaced so achieving a true seal on the block.
This is done using a grinding stone on a surface-planing table.

The safety valve and indicator cock, if present, must be removed and cleaned. The sealing surfaces must be inspected and re-assembled after renewing gaskets and refacing the sealing areas. Safety valves must be tested for opening at the correct pressure.

26.2.2 Pistons and piston rings

These parts are particularly heavily loaded in diesel engines.
Together they must seal the cylinder liner at increasingly higher pressures.
Furthermore, the piston must transfer the pressure in the combustion space to the crank-connecting rod mechanism.
Piston and piston-ring wear is caused by:
- abrasive combustion products and particles;
- metal-on-metal contact of piston rings, piston-ring grooves and the cylinder liner. This mainly occurs at the beginning of the combustion stroke.
- corrosion, chemical damage at low engine loads; low temperature corrosion occurs caused by the formation of sulphuric acid. Sulphur is found in heavy fuel.

Light-metal pistons in smaller engines are cleaned and inspected. When the total number of approved operating hours has been reached, they are substituted together with the piston rings.

Pistons of larger engines are dismantled, cleaned and inspected. If the dimensions are correct and there is still an adequate number of operating hours remaining, both parts are used again. The three-part piston-ring package is usually replaced.

If the top piston-ring groove is chromium-plated, it will require a new chromium coating. During inspection of the piston, it is imperative that the inside is adequately cleaned and checked. Cooling of the piston with lubricating oil is vital.

▲
Damage to engine parts can be considerable.

Shown a piston (1) of a four-stroke trunk-piston engine, which has seized in the cylinder. The camshaft (2) is bent. A connecting-rod bolt probably broke allowing it to move freely in the crankcase, thus causing damage to the engine.

26.2.3 Cylinder liners

Pistons, piston rings and cylinder liners all have a certain 'service life': the number of operating hours they should achieve without failure between major overhauls.
Abrasion by combustion products and **chemical damage** instigated by low-temperature corrosion found at low engine loads are the main causes of wear.

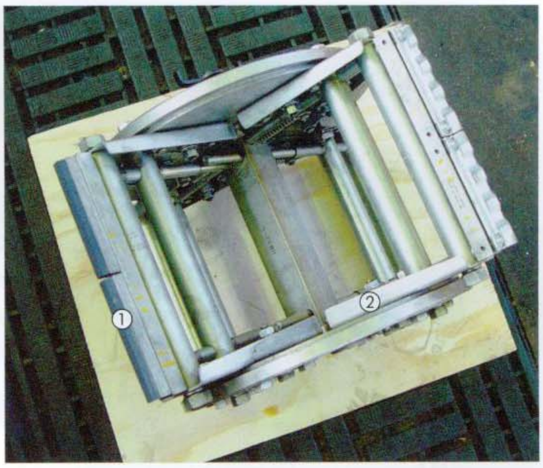
▼
Pistons made from two parts, as is the case in medium-speed engines, are taken apart, cleaned, measured and, if necessary, repaired.

Foreground: pistons with dismantled crown.
Background: the piston crowns have been disassembled.
Comment: note the notches for the valves in the piston crown.

1 *piston skirt*
2 *piston crown*
3 *bolt holes for attaching the crown/skirt*
4 *cooling-oil drilling for piston-crown cooling*

▼
New piston, consisting of two parts.

▲
Cylinder liners prior to honing.



◄
Honing-stone holder with honing stones.

1 *honing stone*
2 *holder*

Procedure

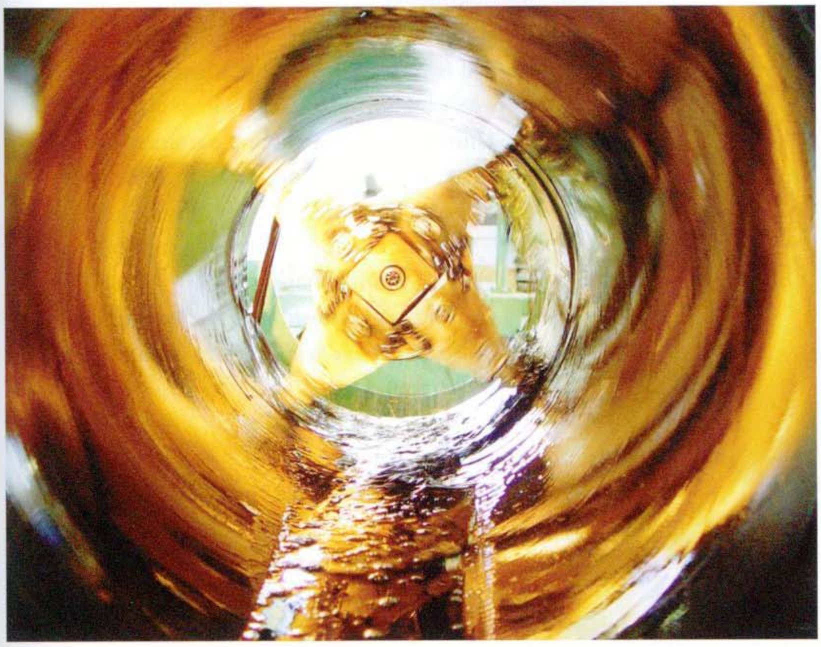
- Decarbonise, degrease and clean the entire liner.
- Inspect rim at the top of the cylinder.
- Inspect the honing pattern on the cylinder wall.
- Inspect the sealing areas.
- Check the cooling bores.
- Measure the liner diameter at various heights and at 90°.
- Wear of the rim.
- Plateau-honing of the cylinder liner.
- Measure the liner diameter, see above.

There are many theories regarding proper honing, subjects such as honing patterns, honing times, honing liquids and honing movements are just a few of the many.
Generally, plateau-honing is performed after coarse honing, creating a polished honing pattern producing a 'run in' liner surface.

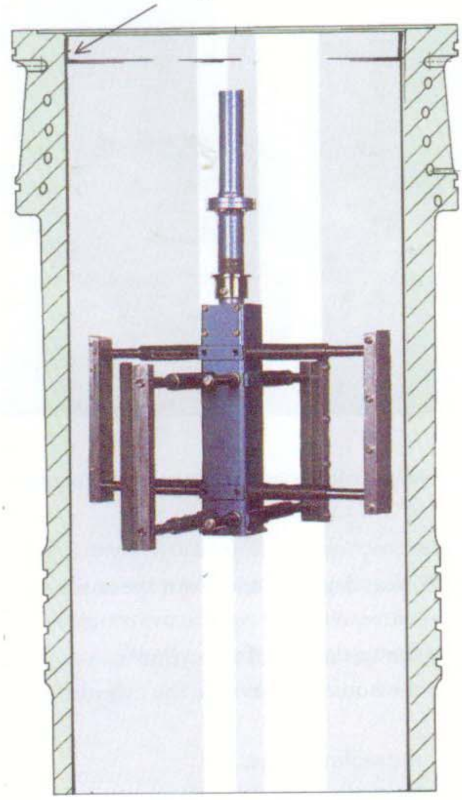
▼
The honing pattern after honing in a used cylinder liner.

Glazing

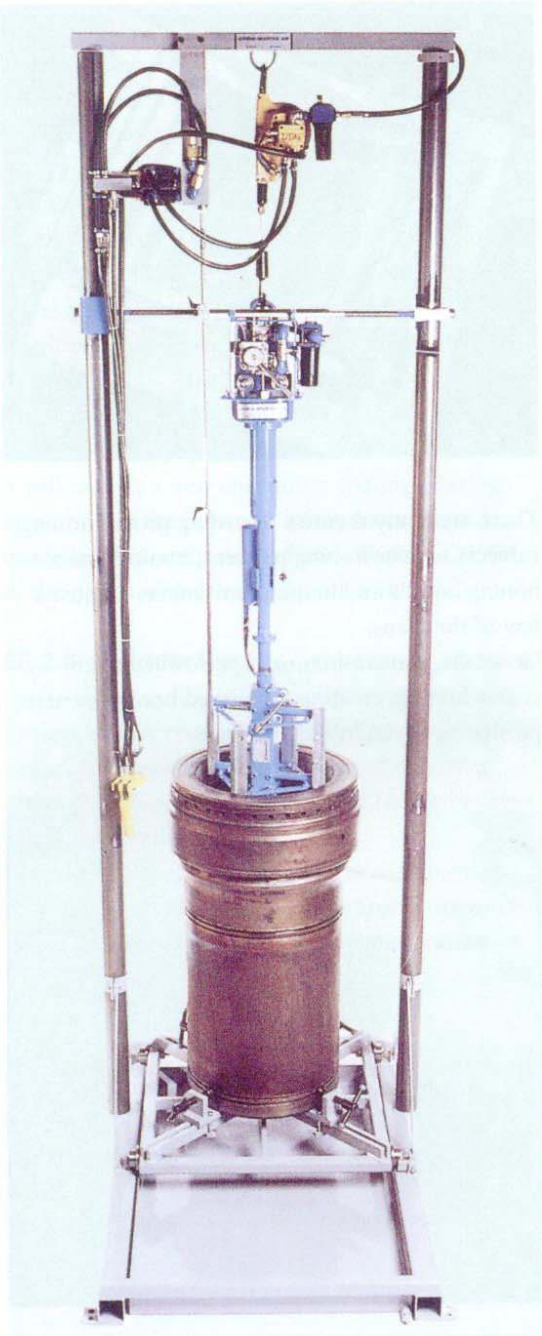
After a certain operating time, cylinder liners can show signs of glazing. The honing pattern has completely disappeared and the surface area has been polished by carbon particles produced during the combustion process. Honing can remove this glazing completely.



◄
Honing gives a precise finish of the cylinder-liner running surface.



▲
Cylinder-honing set, shown here with four honing stones. Cylinder-honing sets for larger cylinder diameters may have as many as eight honing stones producing a true honing, also for larger cylinder diameters.



◄
This mobile honing machine allows cylinder liners to be honed without disassembly of the engine. It can also remove glazing.

▼
Honing a cylinder liner in the engine.

Several features of correct honing (Mahle)

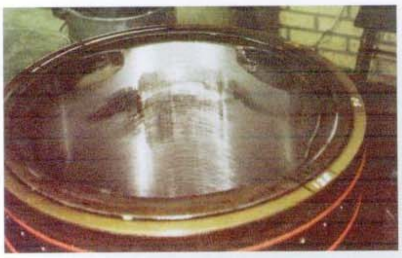
1 **Uniformity of the honing pattern**
In general, the cylinder wall must be shiny, uniform and true.

2 **Honing groove shape**
Cutting edges must be clean and sharp, uniform groove width with a maximum width of 0.1 mm.

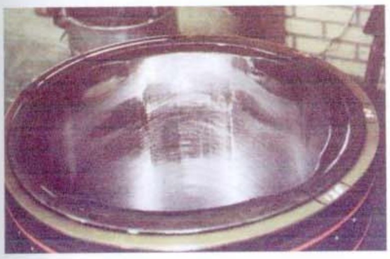
3 **Honing groove position**
Uniform cutting pattern for reciprocating stroke.

4 **Honing angle**
Must lie between 30 and 60 degrees.

Used liner
The overall condition of a liner may not be apparent until after a few honing strokes.

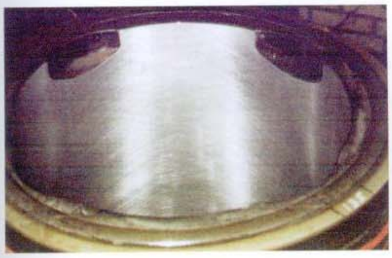


After 3 minutes of basic honing
Soon the surface exhibits marks, ovalty and imperfections.



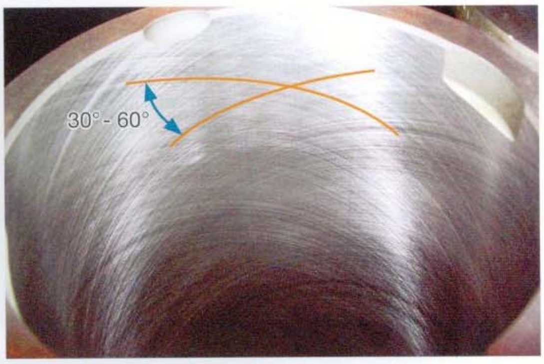
After 8 minutes of basic honing
The full extent of material to be removed becomes clear.

After 14 minutes of basic honing
The imperfections are beginning to disappear.



After 22 minutes of basic honing
Roundness has now been restored and surface marks removed.

After 3 minutes of plateau honing
The plateau honing has removed the coarse peaks, providing a good carrying surface for the piston rings.



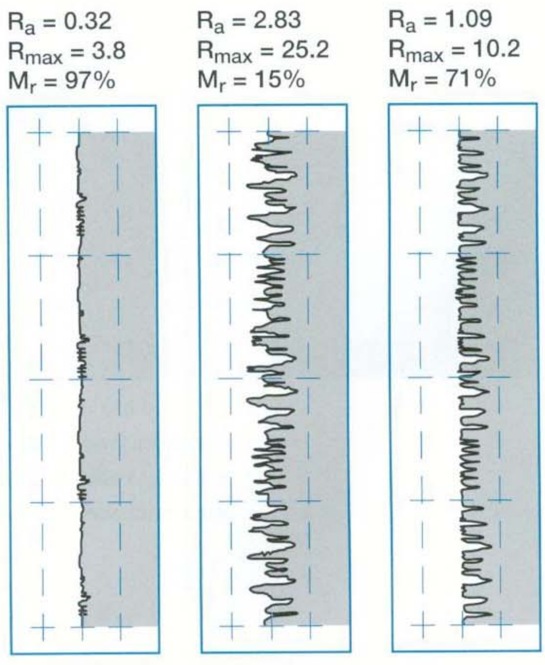
▲
Normal honing angles lie between 30 and 60 degrees.

◄
An overview of cylinder-liner honing according to the procedure performed by the company Chris Marine, so-called plateau honing.

The entire procedure of transforming a 'glazed' cylinder liner to an optimal running surface takes approximately one hour.

▼
Plateau honing; the ideal carrying surface is created in three phases.

left picture: worn and almost completely glazed surface.
centre picture: roughly honed surface with deep grooves and high peaks (R max.) that are still too large.
right picture: the third finish hones down the peaks and creates an ideal honing pattern, giving an ideal running surface for the piston and the piston rings, yet leaving sufficient groove depth and width for the lubricating-oil.



5 **Honing pattern**
Sharply cut, neither blocked nor flattened and no burring.

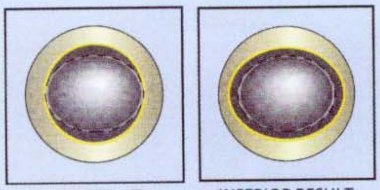
6 **Plateau-honing.**
By pre-honing as well as post-honing.

7 **Roughness of the honing pattern.**
The guide values for diesel engines are
Rz = 4 – 8 µm (microns)
Ra = 0.8 – 1.2 µm (microns)

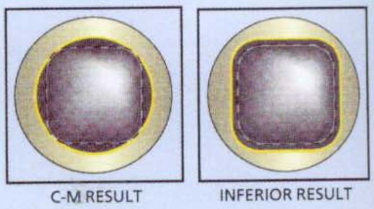
TR – value	plateau-honing pattern	A	1.5 µm
		B	83%
	normal honing pattern	A	2.5 µm
		B	91%

Examples of correct cylinder-liner honing methods from Chris Marine.

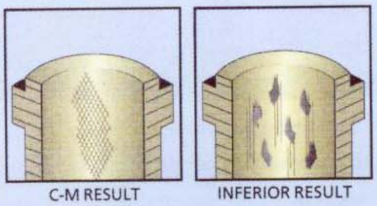
C–M Result = Chris Marine Result.



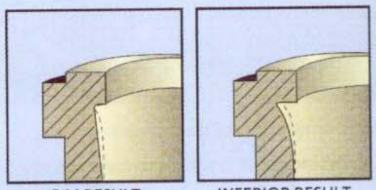
Oval cylinder liners are a common problem that should be regularly dealt with. Round cylinder liners can be achieved by using machines that have:
- adequately designed 4 or 8-armed honing head with a rack and pinion system
- a totally reliable mechanical stone pressure adjustment system



Under certain circumstances, a four-leafed clover wear can occur in a cylinder liner. To restore good roundness requires a machine that has:
- adequately designed 8-armed honing head with a rack and pinion system
- a totally reliable mechanical stone pressure adjustment system



Scuffing marks may be caused by worn piston rings, carbon deposits or fuel-borne contaminants. Such a condition can be difficult to improve and therefore requires a high capacity machine that:
- comes with a variety of tested and proven honing stones
- has a suitable stroke length to fit the cylinder liners
- provides for effective operator control of the honing work



As cylinder liners usually wear at the top sooner than the bottom, a trumpet shape can sometimes occur. This needs to be removed leaving a long transition that reduces the load exerted on both the piston and the piston rings. A cylindric or slightly tapered geometry can best be achieved with a machine that:
- is specifically designed for demanding cylinder liner maintenance work
- provides for the correct honing methods for different kinds of cylinder liners

Inspection methods

1 **Visual inspection, finger nail and coin tests**
These are frequently employed. Experienced technicians can give assessments in combination with other tests, as these methods are inadequate and indefinable.

2 **Fax-film**
An imprint is made of the cylinder surface using a plastic cellulose acetate film of approximately 0.2 millimetres. This imprint is magnified which allows for precision assessments of the honing features as detailed in paragraph 26.2.3, points 2, 3, 4, 5 and 6. **Procedure:** Clean the test surface with acetone (a degreaser), retreat with acetone and press the sheet of fax-film on the surface until the acetone has evaporated (10 to 15 seconds). Remove the film, and place the imprint of the cylinder running surface under a microscope or a microfiche reader and assess.

3 **Surface measurement**
The cylinder surface is probed by an electronic device and the roughness (Rt, Rz or Ra) is measured according to a certain scale. Furthermore, it is advisable to use a profilometer, which registers the roughness on a profile diagram. Preferably, a roughness measurement according to Abbott's bearing curve.

Important aspects for the achievement of good results in the surface processing of cast-iron cylinders

- Only use suitable machines and accessories.
- When tensioning, take note of the tensioning regulations drawn up by the engine factory, levelling, equal tension forces and prevent twisting of the part.
- Precision drilling to approximately 0.04–0.006 mm, dependent on the diameter.
- Hone for a minimum of 0.03 mm material removal, dependent on the diameter. This is required as the groove depth and the resulting hard zone can reach 0.03 mm during precision drilling.

- Do not hone dry.
- Hone cylinders throughout the entire surface so that the buffer edge is completely removed.
- Use the prescribed honing oil (correct viscosity) for the honing stones and material. Cool, if required.
- The honing oil must be sprayed against the cylinder wall with force. Material and honing-stone residues must be flushed away immediately so that they are not pressed into the cylinder wall or honing pattern grooves.
- When re-used, the honing oil must be thoroughly filtered. Filters must be cleaned or replaced on a regular basis. Honing oil must be regularly refreshed.
- Use ceramic honing stones, if possible, as these allow a sharper cut than diamond stones as they have self-sharpening edge properties.
- When selecting honing stones, the hardness of the cylinder material must be taken into account.
  Rule of thumb: the harder the material, the softer the honing stones.
- Perform a visual inspection after honing: all the cylinders should have a uniform shine and a glossy-silver finish.
- After honing, clean the cylinder block and the cylinder liners thoroughly.

Machining

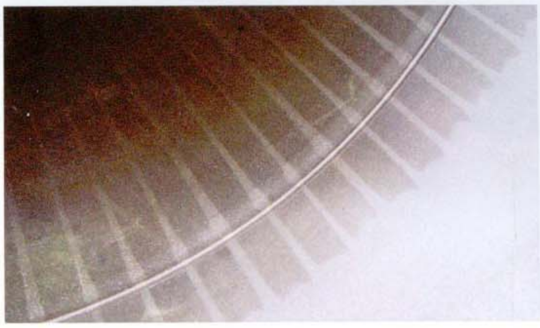
Cylindrical shape finishes.

- **Taper**
  Stroke is too long
  Stroke is set too high
  Honing stones and guides are not trued up.
  Honing stroke on the wide side is too high and too low on the narrow side

- **Bell mouth (top and bottom too wide).**
  Stroke too long
  Honing stroke on both sides too slight

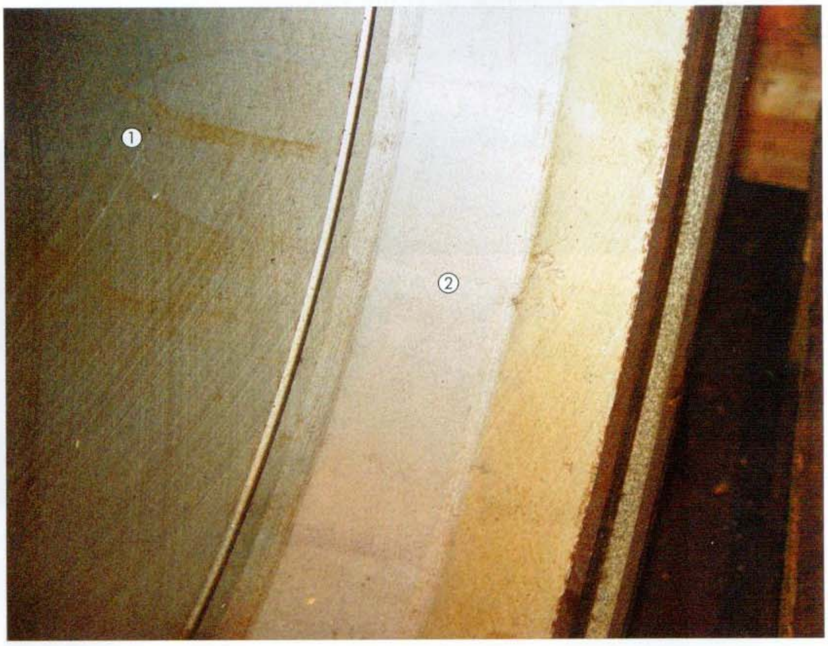
- **Barrel (top and bottom narrower than in the centre)**
  Stroke too short
  Honing stroke on both sides too slight

A worn cylinder liner.



A laser process can re-harden a used cylinder liner. This only affects the surface structure. MAN Diesel AG Reconditioning, Hamburg, Germany.

A cylinder liner of a large four-stroke engine for repair.



In used cylinder liners, a coating can be applied by spraying a material on the liner above the highest position of the piston. This has the same effect as an anti-polishing ring.

1 *running surface section of the cylinder liner*
2 *coating*

26.2.4 Crankshafts and bearings

▲

New crankshafts made from a single forging ready for the final process, such as the bolt holes for the counterweights. MAN Diesel, Hamburg, Germany.

In four-stroke engines, the engine-driving gear consists of the crankshaft, the connecting rod and the piston.
The crankshaft drives the camshaft via the geared wheels, which in turn drive the fuel pumps and inlet and exhaust valves.

In many four-stroke engines, the main lubricating-oil pump, fresh-water pump and untreated cooling-water pumps are also driven from the crankshaft and the camshaft.
The camshaft of a four-stroke engine rotates at half the speed of the crankshaft, as it makes one revolution for every two revolutions of the crankshaft.
The entire process takes 720 crank degrees.

If the engine is started using starting-air systems connected to each cylinder, additional control-cams are mounted on the camshaft.
Today, an increasing number of larger engines are started via the starter on the flywheel

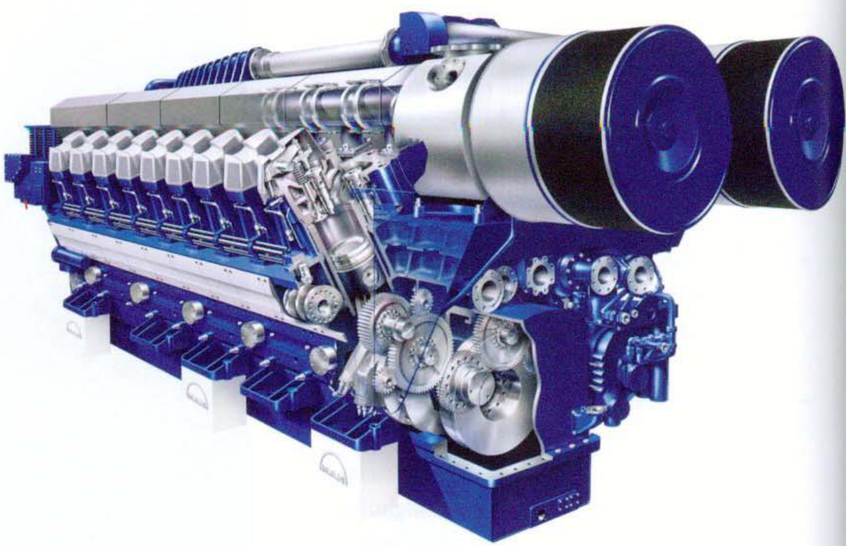
Also, see Chapter 14, Starting systems of diesel engines.

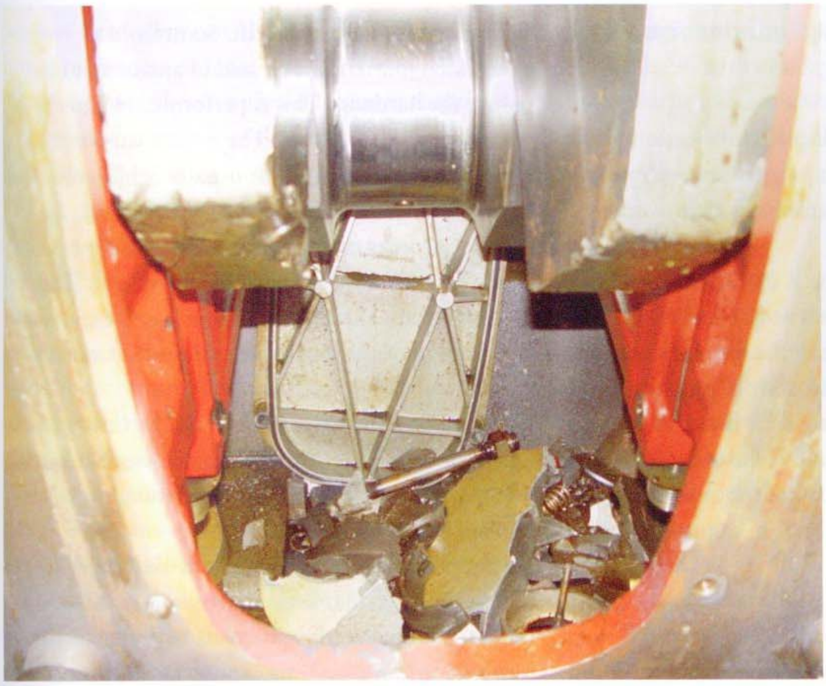
Together with the engine frame, the crankshaft is the most expensive part of the engine.
Crankshaft damage necessitates elaborate and expensive repairs. In large engines, damage to, for instance, the journals can sometimes be repaired without dismantling the crankshaft. This is also referred to as **in situ reconditioning**
Crankshaft damage or damage to the driving gearing often entails severe damage to the entire engine.

▶

The drive gearing of an engine, such as the crankshaft, gears, the camshaft, connecting rods and pistons, must be accurately installed in the engine.

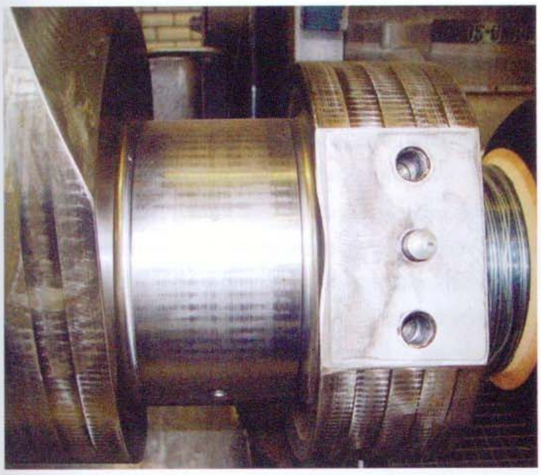
The centre lines of the crankshaft and the camshaft must run parallel. The centre line of the pistons must be perpendicular to the centre line of the crankshaft. Gear alignment is important.





◄
Damage in the crankcase usually means damage to the crankcase itself. Shown a case of total loss.

▼
A large crankshaft lathe at Mark van Schaick in Schiedam, The Netherlands. An engine-part reconditioning company.



▲
The running surface of this crankshaft journal is not damaged. It is possible to grind this journal to the next possible 'undersize'.

Classification agencies employ very detailed regulations with respect to material use, size and manufacturing methods for crankshafts.

Also, see Chapter 32, Regulations for propulsion engines, classification, repair and damage.

►
Damaged crankshafts for reconditioning.

Usually, the crankshaft journals or crankpins are damaged. After a thorough inspection, which, amongst others, consists of measuring, surface hardness check and a hair-crack test, the shaft is ground to an undersize. 'Thicker' bearing shells must be used.

26.2.4.1 Common crankshaft damage.

1 **Overheating**
Mostly a result of inadequate lubrication between the crankshaft journal or crankpin and the bearing. All crankshafts that overheat are bent.
Generally, crankshaft beds or engine blocks are rigid and the crankshaft is 'limp'. The lubricating-oil pressure in the bearings keep it 'afloat', that is, the crankshaft is supported by the oil film in the bearing so it is not in contact with the bearing material. If the bed or block as well as the crankshaft are too rigid, the rotating crankshaft will not be supported by the lubricating-oil film resulting in wear and increased friction.

2 **Fractured connecting-rod bolts**
By overloading or poor assembly of the conrod bolts, the connecting rod can become detached from the crankshaft and cause severe damage. Conrod bolts can loose their strength if re-used too often. The technical manual should always be consulted to check how often the bolts may be used!

3 **Dilution of lubricating oil**
The lubricating properties of lubricating oil can deteriorate considerably due to, for instance, ageing of the lubricating oil or fuel or coolant leaks.
Diluted lubricating oil causes a reduction in the hydrodynamic lubrication in the bearings resulting in the shaft coming into contact with the bearing material.

26.2.4.2 Crankshaft hardness

The hardness for the 'soft' crankshafts is measured according to the Brinell scale. For 'hard' crankshafts, the Rockwell C scale is used.
For spray coats, such as chromium, the Vickers scale is used, even though the base coat of thin coatings is often also measured, thereby rendering incorrect measurements!
Large crankshafts of engines in categories III and IV are generally 'soft'. The material hardness is approximately 150 to180 Brinell for crankshafts of C 45 and 220 to 260 Brinell for 42 Cr crankshafts.
Smaller crankshafts in categories I and II engines are usually 'hard'. They have been surface-hardened. The hardness is approximately 45 to 65 Rockwell C.
Soft crankshafts (category III and IV) can become 'hard' when the crankpin or the crankshaft journal is heated. The hardness can rise from 200 to 600 Brinell.

Warming a heated crankshaft. Sometimes an overheated journal is re-heated in an attempt to reduce the hardness. This is performed in situ with electric 'heat strips'. The results vary, but the original hardness is not usually achieved, as the hardness remains higher. If the engine supplier accepts the acquired hardness, approval of the classification agency usually follows. This method avoids the time-consuming and expensive dismantling of the engine to remove the crankshaft.

Hard crankshafts (categories I and II) turn 'soft' when the crankpin or the journals overheat.
Flame hardening is a surface-hardening process applied to the journal.
Method: The material is heated with a high-temperature gas flame to approximately 600 °C followed by quenching with jets of water resulting in rapid cooling. The material surface is transformed with a significant increase in hardness.

▼
Changing the crankshaft on a ship.

This is an expensive and time-consuming procedure.
In four-stroke engines, the entire block must be hydraulically supported, so the crankshaft can be 'dropped' out from below the engine. All partitions or doors must be removed so the crankshaft can be transported out of the engine room. Often a large hole is cut in the hull closest to the engine room using cutting torches.

Surface hardening can also be achieved with a nitrogen process or induction hardening. A hard chrome-plating layer with a maximum thickness of approximately one millimetre can also be applied.

Tearing of the journals often occurs due to material fatigue, torque and overloading. Microscopic tears deepen resulting in fracturing of a crankshaft and overall damage to the engine. Inferior vibration dampers may be the root cause of the problem. Vibration dampers should be regularly inspected.
The filling found in dampers usually consists of a highly viscous silicone fluid (jelly or oil). It is therefore advisable to test the vibration dampers. A special testing device is used that introduces vibrations into the coupling. The extent to which the vibrations are damped indicates how effective the vibration damper is.

Heat-induced cracks are generally superficial and can be removed by polishing on a (crankshaft) grinding machine. If the crack cannot be removed by polishing or grinding, the crankshaft is scrapped.

The material used in crankshafts can always be traced via a data base. The hardness is particularly important. The undamaged section always has the initial base hardness.

Renewing bearing shells
It is imperative that the selected bearing is manufactured from the same material as that of the original shells. Bearing shells have standard 'oversizes'.
The engineer operating a crankshaft-grinding machine knows precisely how much material can be removed in order to obtain the first oversize. Only OEM-shells should be used (OEM = Original Equipment Manufacturer).

Straightening/trueing up an overheated crankshaft
Crankshafts with crankpin or main journal that have overheated are always bent. They can be straightened hydraulically or by hammering around the radius with a hammering tool and a heavy hammer.

26.2.4.3 Grinding the crankshaft

The crankshaft is positioned at 'zero' in the centre of the grinding machine.
Depending on the size of the crankshaft, stays are placed underneath the journals to prevent the crankshafts from bending under their own weight. They also absorb the pressure exerted by the grinding process.

Grinding of the crankshaft starts from the centre plate from the left and right to the middle.
In the centre of the crankshaft, the value given on the dial gauges must be 0.000 mm!

▼
Grinding a large crankshaft at Mark van Schaick, Schiedam, The Netherlands.

1 *supports*
2 *grinding stone*

Clearance of journals and crankpins

Rule of thumb: $\frac{1}{100}$ mm or 0.001 times the diameter clearance per 10 mm shaft diameter.
A shaft with a diameter of 250 mm:
the clearance of the bearings may not exceed
$\frac{250}{10} \times \frac{1}{100} = \frac{25}{100}$ mm.

26.2.4.4 Crankpin wear

Crankpins and crankpin bearings are heavily loaded. The hydrodynamic lubrication is not optimal as the forces fluctuate in both size and direction. This often causes boundary lubrication where the lubricant film is insufficient to prevent surface contact between pins and bearings. Localised wear occurs resulting in an oval-shaped shaft. This occurs more often in a certain type of engine from certain engine manufacturers.

◄
A large crankpin of a medium-speed diesel engine.

An attempt to straighten the crankshaft by hammering has only partially succeeded.

▶
The crankpin must be true.

Measuring the roundness is important after a large number of operating hours. The pin often assumes an oval shape, which is then ground down to a smaller diameter.

▲
After repairs, the crankshaft must be approved according to the classification bureau regulations. This is referred to as the "release" of the crankshaft.

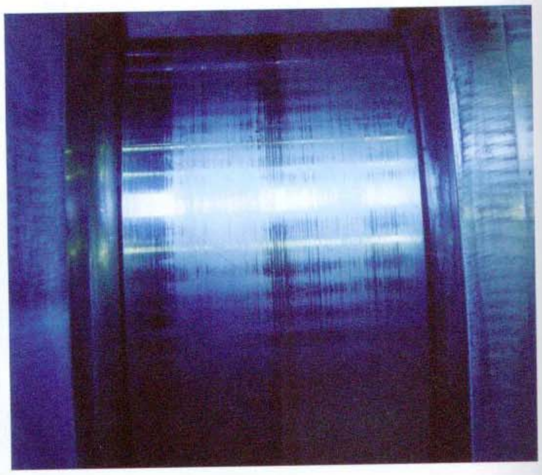
An absolutely round shape with a smooth surface area can be obtained by grinding the shaft. The main and crankpin journals of the crankshafts are manufactured with a certain oversize and may be ground-down several times without dimensioning problems. Obviously, oversized (thicker) bearing shells must be used. Standard sizes are normally used so the shaft is ground to a certain diameter (undersize).

After this machining process, a protocol is drawn up with the new measures of the crankpins and the journals. This practice is referred to as 'acceptance': the crankshaft meets classification agency standards.

Grooves in the running surface of crankpins and journals

▶
A damaged crankshaft on a crankshaft lathe, a type of grinding machine.

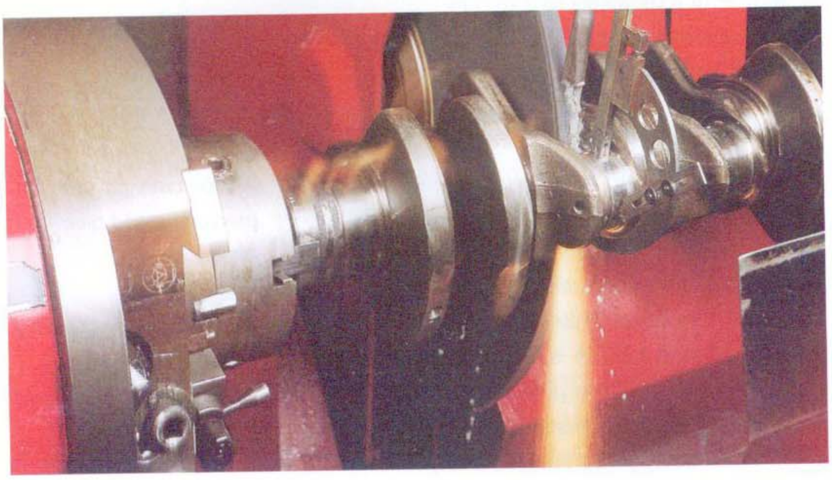
The grinding stones (1) have a large radius and varying hardness and grit sizes.



▲
A worn crankpin.

These grooves are often a result of overheating, caused in part by insufficient lubricating-oil. Moreover, dirt particles in the lubricating-oil may have come into contact with the shaft, for instance via a damaged filter or as a result of certain activities which have contaminated the cooling system. If the grooves are shallow, the shaft or pin can be ground.

▼
Grinding a small crankshaft.



26.2.4.5 Modification of the material structure

Overheating and groove formation cause a temporary temperature increase of the material. The running surfaces often turn blue. This usually means that the carrying-surface temperature has risen to approximately 700 °C.
This changes the structure of the material.

Unalloyed carbon-steel crankshafts harden.

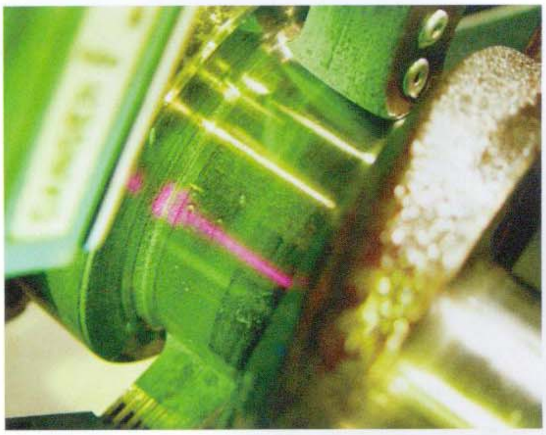
Local soft areas occur in alloyed hardened crankshafts.

These phenomena cause tension build-up in the surface layers of the material. This can result in the formation of hairline cracks, ultimately resulting in shaft fracture.
These 'areas' can be localised simply by performing a hardness test on location. Dependent on the difference between the hardness of the areas and the hardness of the shaft, the decision is made as to whether the shaft or crankpin should be ground to a certain undersize with none or very little discrepancy between the hardnesses. Clearly, the shaft must be ground to the minimum prescribed shaft diameter.

26.2.4.6 Cracking of crankshafts

▲
Check the journals for hairline cracks.

The shaft can often be repaired by grinding if the cracks are superficial.
If the cracks are too deep, the shaft is written off as a total loss.
The crack depth can be established using a small grinding stone.
Some second-rate firms weld the cracks.
This is obviously pointless and the shaft is then scrapped!



▲
A magnetic field is used to detect superficial cracking.

A coil is placed on the shaft and this generates a magnetic field. A contrast liquid is applied to the shaft and the surface inspected with a lamp. A crack can be discerned by an interruption in the colour of the shaft. This process is called 'Magnafluxing'.

Crankshaft cracks are a result of material overload.
The excessive load may be produced by torsional stress, generated by torque and torsional vibrations.
Under normal operating conditions, this need not be a problem. Often torsional vibrations are at the core of the problem. When the engine speed approximates the number of natural vibrations of the crankshaft, the torsional stress can increase to a point where the crankshaft shows cracks.
These fractures are generally easily recognisable. The cracked surface is often at an angle of approximately 45° to the centre line of the crankshaft.
Extremely large bending stress may also cause overall damage to the crankshaft when the crankshaft is not aligned.
The most important causes are:
- wear of the main shaft bearings;
- loading of the ship;
- distortion of the engine bedplate due to extreme weather conditions or the ship running aground;
- incorrect bedplate design in diesel-power plants and power barges;
- temperature influences.

26.2.4.7 Damage to the crankshaft by a seized piston

If piston rings seize in the cylinder due to insufficient cooling or poor lubrication, severe damage often follows.

The piston crown seizes at the top of the cylinder liner. The piston skirt with the piston pin is torn from the piston crown, and the moving connecting rod severely damages the frame and the crankshaft.
The connecting rod is often detached from the crankpin, when the connecting-rod bolts break and the counterweight bolts on the crankshaft detach. All these fractured parts cause denting in the crankshaft. These are irreparable resulting in the crankshaft being declared a total loss.
If the seized piston damages the frame irreparably, the engine is a total loss.

▼

The bolts of the counter weights must be tightened and secured with the utmost precision. Detached counterweights can cause irreparable damage to the crankshaft.

It can be determined if the crankshaft ever suffered an 'overspeed' with the counterweights still attached to the crankshaft.
After removal of the counterweights, the crankshaft faces and the counterweight faces exhibit smooth friction spots. This is caused by stretching of the fixing bolts to the extent that the faces of the crankshaft and the counterweight are no longer in contact, thus allowing for micro-movement independent of each other.

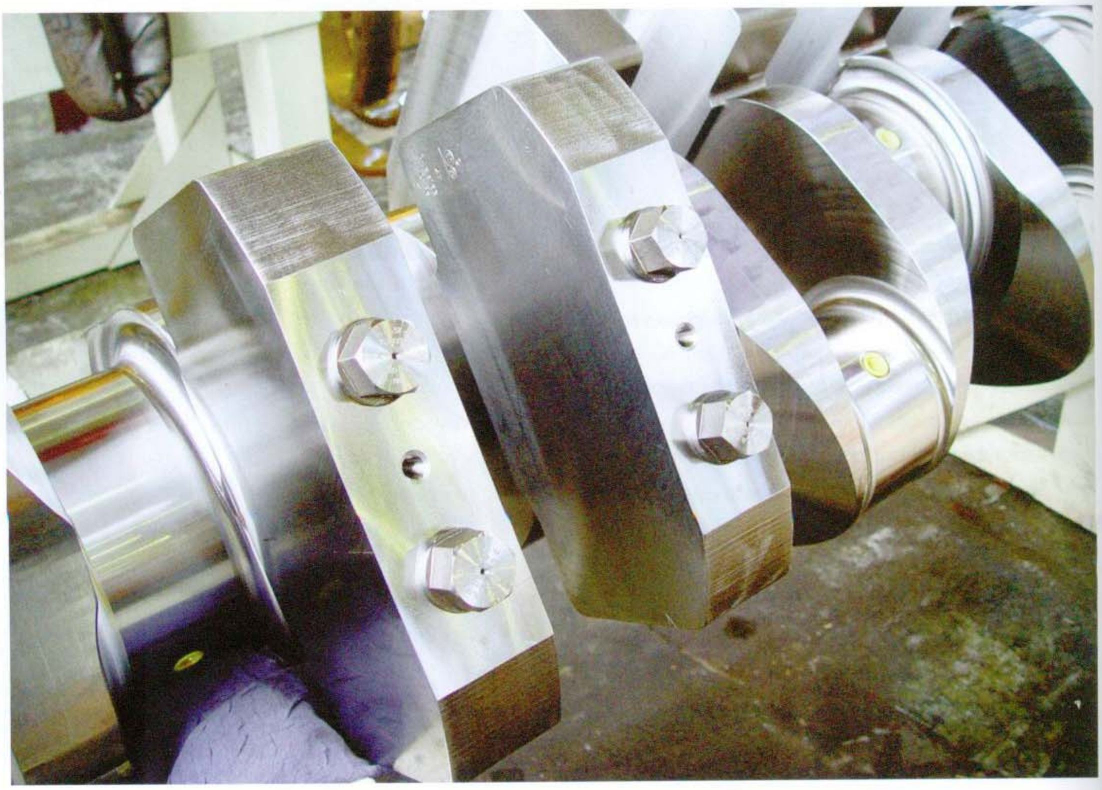
26.2.4.8 Damage caused by counterweights detaching from the crankshaft

When the engine speed far exceeds the maximum allowable speed, the counterweights may detach from the crankshaft by the breaking of the bolts attaching the counterweights to the crankshaft. In this case, the maximum allowable bolt tensile stress is exceeded. Total damage to the crankshaft follows.

26.2.4.9 Crankshaft damage as a result of incorrect mounting of the main bearing shells

This can occur when technicians do not know how to mount the bearing shells or which aspects require special attention.
Key points: the bearing shells must be clean and show no signs of damage. During fitting, no grease or oil must be applied to the back of the shell to facilitate insertion of the bearing shell. The thickness of this grease or oil layer is often greater than the clearance between the shaft and the bearing shell!
When installing the bottom bearing, the bearing-cap bolts with the bottom bearing shell must be equally tensioned. In accordance with the regulations of the manufacturer, the vertical and horizontal bearing-fixing bolts can then be attached.

Furthermore, it is important to know if a crankshaft is equipped with the originally fitted



bearings or if one or several oversized bearings have been used after crankshaft repairs.

26.2.4.10 Acid damage to the journals and crankpins

Sulphur found in fuel, in the form of sulphur dioxide formed in the combustion space, can move past the piston rings and be found in the crankcase if maintenance is not performed regularly, for instance, lubricating-oil cleaning and changing. Subsequently sulphuric acid may be formed with water vapour. Under good maintenance conditions, the acid is neutralised by an alkaline additive in the lubricating oil and is therefore not detrimental to the crankshaft.

See Chapter 11, Lubrication of engines.

If no or little attention is paid to the quality of the lubricating oil, for instance, by failing to sample and test the lubricating oil, the alkaline additive could be exhausted, thus allowing the sulphuric acid formation to damage the crankshaft. Black stains or pitting will appear on the journals or pitting which damage the running surface. **Corrosion of the crankshaft may also occur when there is leakage of cooling water or coolant, although this is less aggressive in comparison to sulphuric-acid corrosion.**
This problem is averted by taking samples at regular intervals!

Damaged bearings

Bearings in an engine have a short life-span, as they are heavily loaded; corrosive and mechanical wear in particular are significant.
In four-stroke trunk-piston engines, the quality of the fuel plays a decisive role.

26.2.4.11 Key points for an extended operating time

These are:
- suitable lubricating-oil treatment, such as filtration, separation and sampling;
- preventing external contamination of the engine during maintenance;
- capping lubricating-oil drillings in crankpins during repairs, and capping all openings leading to the lubricating-oil system;
- prelubrication of engine ensuring all the vital engine parts are properly lubricated prior to start;

▲
An in-line engine frame 'on its head'. The crankshaft counterweights must still be fitted.

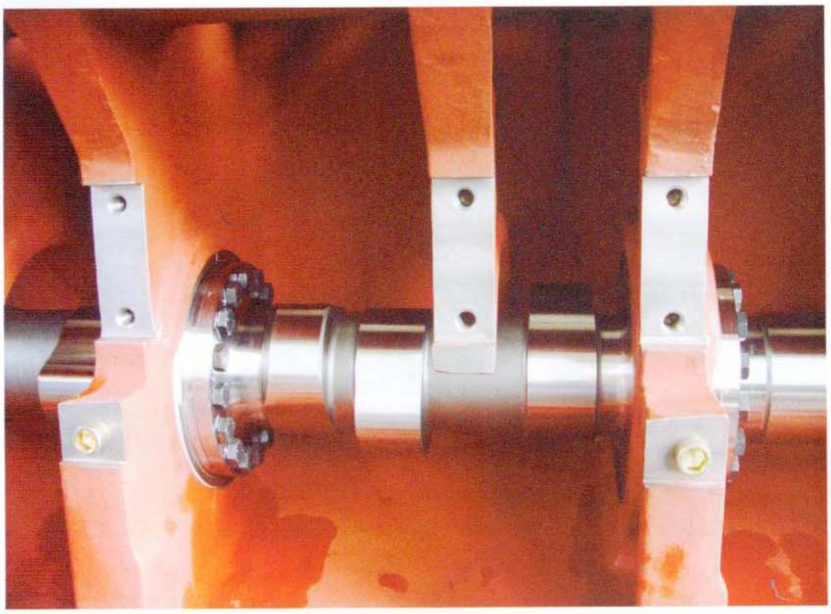
- check lubrication during operation by monitoring lubricating-oil pressures, lubricating-oil temperatures and the lubricating-oil level in the crankcase. Monitoring lubricating-oil consumption is important as this indicates the condition of engine maintenance. Inadequately functioning piston rings and exhaust-valve stems whose clearances are outside normal parameters lead to an increase in lubricating-oil consumption. Obviously, the operating life of bearings is largely dependent on operating conditions and varies from manufacturer and engine type.

Modern **trimetal bearings,** mainly used in four-stroke engines, both high-speed and medium-speed, are not reconditioned.
They are usually replaced by a new bearing after a certain number of operating hours.
The total thickness of the bearing shell can be measured in order to establish the extent of wear.

After a certain number of operating hours, the lead-bronze base coat or the emergency-running coating gradually emerges. This means that the actual friction material, an aluminium alloy, has been worn away. The criterion for bearing rejection is at a certain percentage of visible lead-bronze surfaces.

26.2.5 Camshafts and bearings

The cams on the camshaft are heavily loaded when driving the high-pressure fuel pumps and inlet and exhaust valves.



▲
A camshaft with, from left to right, the fuel cam, the inlet cam, and the exhaust cam.

▶
A finished engine block as seen from the front. The location of the crankshaft, the camshaft, the scavenging-air ducting, and the internal pipes is clearly visible.

▶
An older type camshaft of a four-stroke high-speed diesel engine.

This is made from a single forging, as opposed to modern camshafts, which are usually manufactured in sections.

The cams are often damaged by grooves and indentations.
Small camshafts manufactured from a single forging are usually replaced.
In large camshafts, which consist of separate sections for each cylinder, only the damaged section will be replaced.

In separate hydraulically mounted cams, the damaged cam can simply be replaced.
On occasion, cams are rewelded using a wear-resistant material. The original dimensions are then restored by grinding.

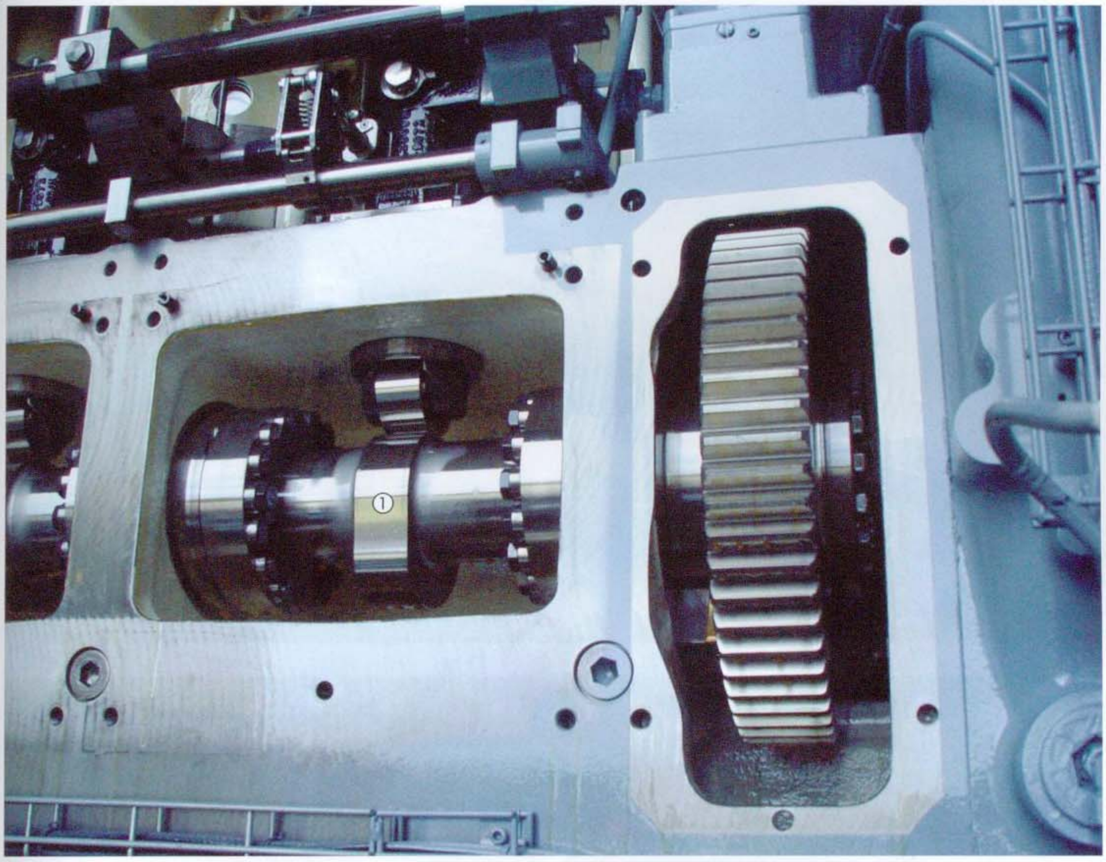
▲
Rewelding steel engine parts is a meticulous and complex procedure.

The quality of reconditioned engine parts does not always match new engine parts. Shown here, a section of a camshaft.

Camshaft bearings are replaced when scrapped.

26.2.6 Gears

Gears are subject to wear.
The shaft bearings of the gears are of primary importance.
When the valve clearance is too large, the gearwheels mesh differently causing wear.
The running surface must be checked for damage and wear.
This normally only occurs with insufficient gearwheel lubrication, contaminated lubricating oil or an irregular load on the teeth.
Generally, when reduction gears are correctly aligned, the wheels mesh smoothly and few problems may be expected.



The camshaft driven by the crankshaft by gears.

1 *fuel cam*

26.2.7 Fuel pumps and injectors

Fuel pumps

Pressures and temperatures in the fuel pumps can rise considerably.
Peak fuel-pressures of 1000 to 2000 bar and fuel temperatures of 140 °C are common when running on H.F.O..
Extreme pressures also occur when running on diesel oil, however, the fuel temperature is significantly lower, 15 to 30 °C.
The camshaft drive with fuel cam, cam and plunger are also heavily loaded.

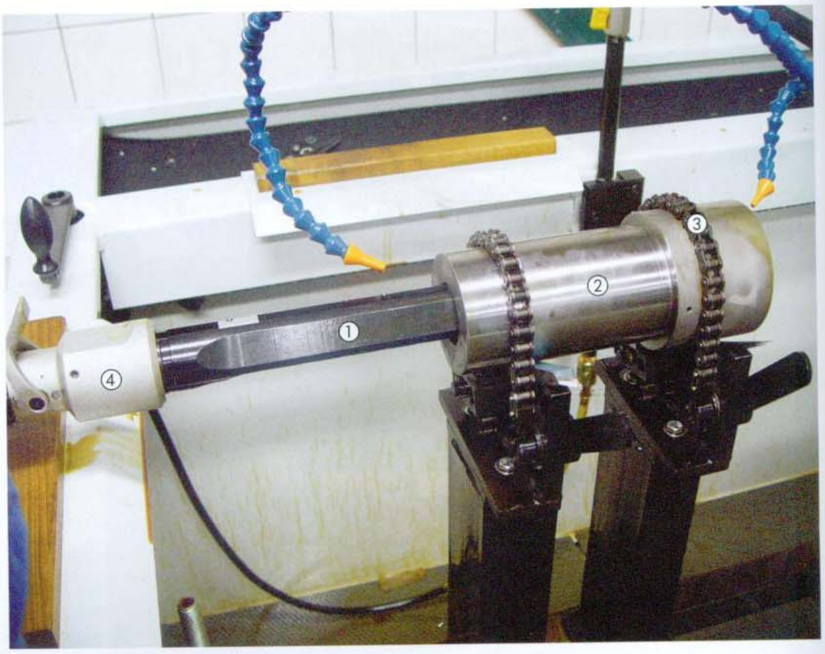
Fuel-pump casing in the reconditioning workshop.

Honing a fuel-pump cylinder of MAN-Rollo in Zoeterwoude, The Netherlands.

During the honing process, very little material is removed. The running surface is finished so that the seal between the fuel-pump plunger and the cylinder is restored.

Honing the pump cylinder.

1 *honing machine with honing stones (not visible)*
2 *pump cylinder*
3 *fasteners*
4 *mechanical rotating drive, the reciprocating movement is done by hand*



A variety of honing tools and honing stones.

Reconditioned fuel pumps.

All the openings are capped to seal out moisture and dirt. Each pump is labelled.

Procedure

The fuel pump is disassembled and cleaned in a washing machine.
Visual inspection is performed on parts subject to wear, such as, plungers, casings, pressure valves, cavitation bolts and the driving mechanism.
The dimensions of, in particular, the plunger and cylinder are measured.

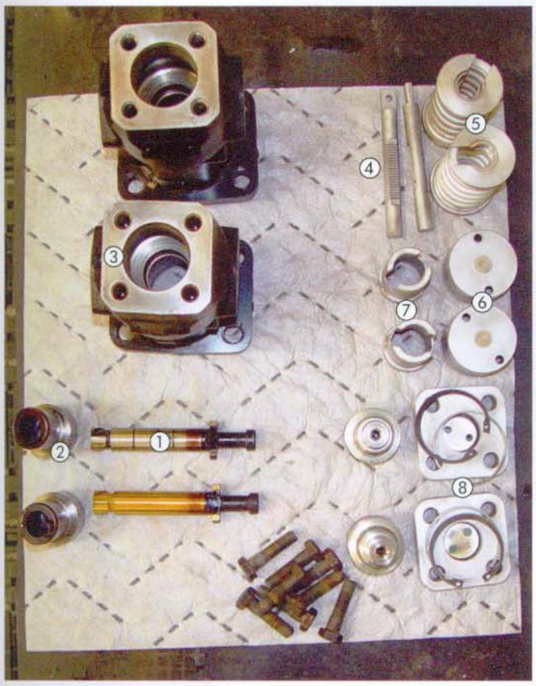
Fuel pump and parts.

1 *worn fuel plunger*
2 *reconditioned fuel plunger*
3 *cylinders or 'barrels'*
4 *reconditioned fuel pump*

Plungers and barrels (cylinders).

A polishing machine for trueing up parts of atomisers and fuel pumps.

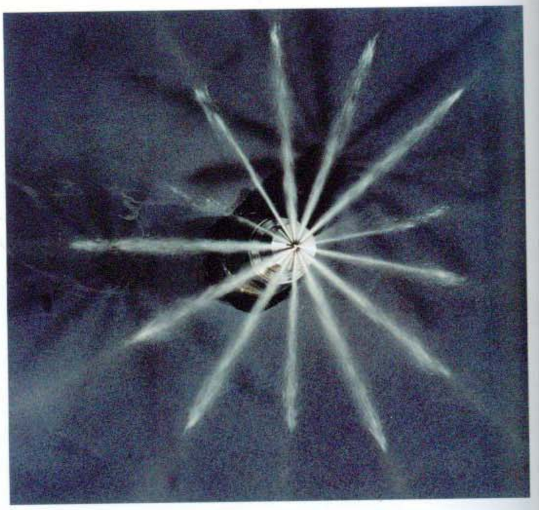
The parts are trued up by the application of a fine sanding paste applied to the vibrating plate.



Disassembled fuel pumps, cleaned and ready for inspection.

1 *plunger*
2 *barrel*
3 *pump casing*
4 *gear rack*
5 *springs*
6 *buffer plate*
7 *adjusting sleeve*
8 *cap with locking gasket*

▶ Fuel atomisers require regular and meticulous maintenance. They are essential for optimum engine operation.



Injectors

Adequate fuel-injector performance is essential for a perfect combustion process in the cylinder. The injectors must atomise the fuel at the correct pressure.

See also Chapter 9, Fuel-injection systems.

Injectors must be removed from the cylinder head after a prescribed number of operating hours and tested by an injector-testing device. These are simple manual test pumps, or mechanically driven test devices performed by specialised firms. During the test, the opening pressure and fuel atomization is checked, as well as possible injector leaks after the needle valve is closed.

The latter is indicative of poor sealing of the needle valve.

Furthermore, the injector casing and connection to the fuel line is inspected for leaks.

When the injectors are inspected by specialised companies, the entire injector is disassembled, the various parts cleaned, measured, repaired or renewed.

During inspection, the following is checked:

- the sealing surface of the injector needle;
- the length of the relaxed spring;
- the shape and dimensions of the injector perforations;
- the wear of the plunger and the casing;
- the pressure valve or the residual pressure valve, if present;
- the shape of the nozzle and possible corrosion. High temperatures may cause distortion of the nozzle.

After assembly, the injectors are tested. The spray patterns are also checked by placing special paper or cardboard underneath the nozzle.

▼ Injectors of large diesel engines. The contact faces have been buffed, a fine-sanding process.

◀ The nozzles. Each diesel engine has its own nozzle type.

◄
Fuel pumps of all the Wärtsilä four-stroke engines. The injectors are tested at varying speeds and loads.

The pumps are mounted on an actuating machine attached to a camshaft, which measures injector performances at varying speeds.

▼
An injector-test machine.

1 *injector*
2 *fuel- supply line*
3 *test machine*

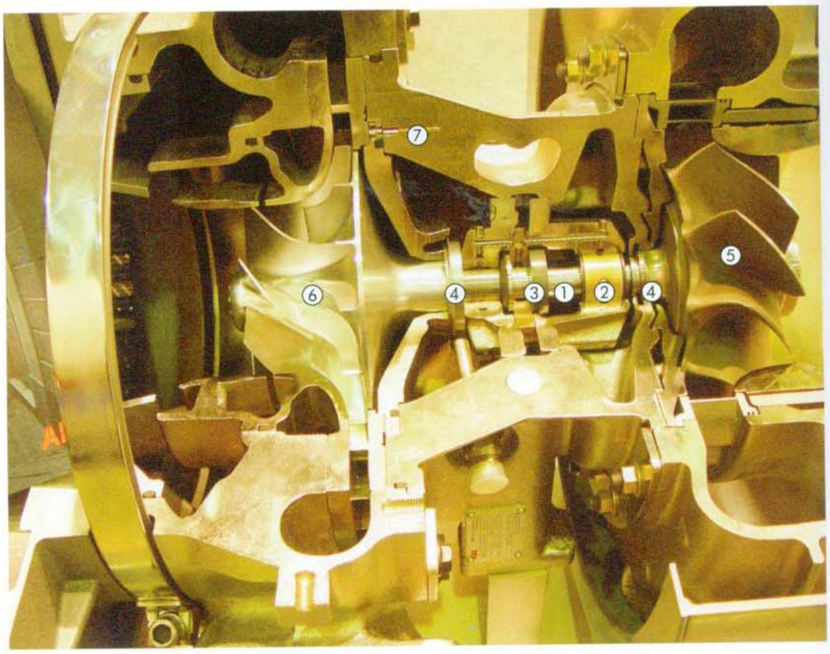
Warning: Always stay clear of an operational injector. The speed of the fuel jet is such that accidents can occur!

Comment

A regular and meticulous inspection of the injectors is important for the optimal operation of diesel engines.

A cut-away section of a turboblower.

1 *turbine shaft*
2 *exhaust-gas side bearing*
3 *thrust bearing*
4 *seal*
5 *exhaust-gas turbine*
6 *centrifugal compressor*
7 *turbine casing*



A series of blades of an exhaust-gas turbine of a turboblower that are irreparably damaged. This is most likely caused by fragmented exhaust valves.

26.2.8 Turboblowers or super charging groups

Supercharging groups are essential for the operation of the diesel engine.
Sufficient air supply for complete fuel combustion, cylinder scavenging and cooling of hot engine parts in the combustion space, are important for efficient engine operation and for engine operating life. The turboblower must supply sufficient air at a certain pressure.

Optimal turboblower performance requires the following:
The degree of contamination and damage to the exhaust-gas turbine.

Damaged exhaust-gas rotor, possibly caused by fractured piston rings.

A piston-ring catcher is sometimes placed before the exhaust-gas turbine inlet (two-stroke).

Combustion products such as carbon deposits may settle on the blades and interfere with the flow of hot exhaust gases, thus causing the blower speed and consequently the capacity and the supply pressure to decrease: the engine receives less air.

26.2.8.1 Contamination and damage of the air compressor

Obviously, this occurs less frequently. The air-inlet filter captures virtually all solid particles and the air temperature on the suction and delivery side is relatively low. The diffuser, the stationary section of the air compressor that converts the high air velocity generated by the impeller to pressure, is usually in good condition.

26.2.8.2 Rotor-shaft bearings

The speed of extremely large blowers is high, approximately 4000 to 6000 rpm. Very small blowers for high-speed diesel engines can reach speeds of 60,000 to 120,000 rpm.
The pump impeller and the turbine wheel are attached to a rotating blower shaft. Correct balancing of this shaft is essential for the operating life of both bearings.
Therefore, the turboblower is always dynamically balanced again after turboblower maintenance. Both bearings are disassembled, cleaned and all the dimensions measured. The rotating parts are replaced.

▼

Appropriate and regular maintenance is essential for diesel-engine performance.

This air-inlet filter of a turboblower is heavily contaminated and impedes the engine-air supply.

◄

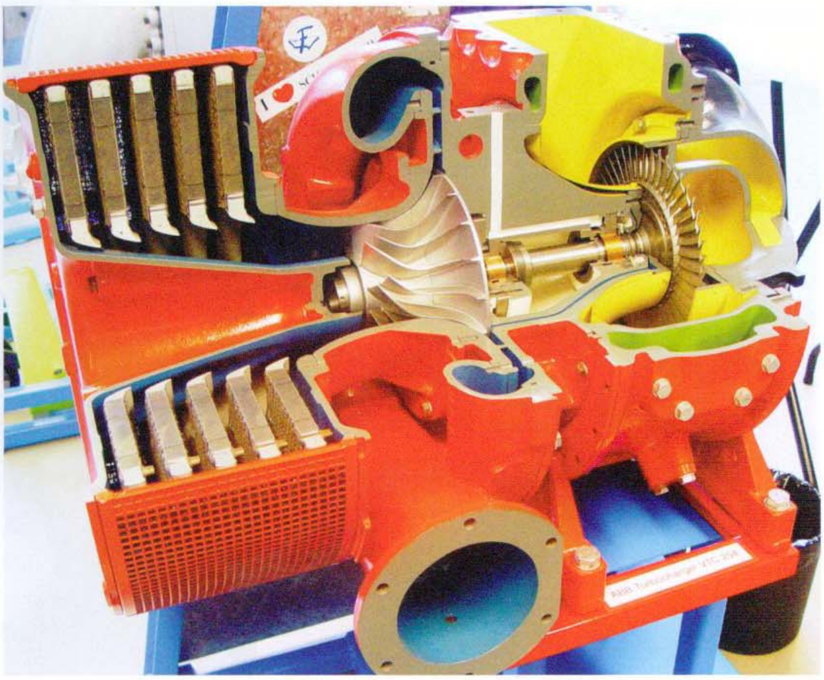
A complete rotor ready to be used.

▼

A cross-section of an ABB-VTC 254 turboblower. Note the large air filter with sound damping to the left in the picture.

▲

A contaminated exhaust-gas turbine reduces the capacity of the turboblower.



▲
Bearing casings are cleaned, inspected and measured, and then assembled and mounted with new bearings.

▲
All the moving parts of the turbine-rotor bearings, such as ball bearings, spring packings and bushes are always replaced by new parts with a major turboblower overhaul.

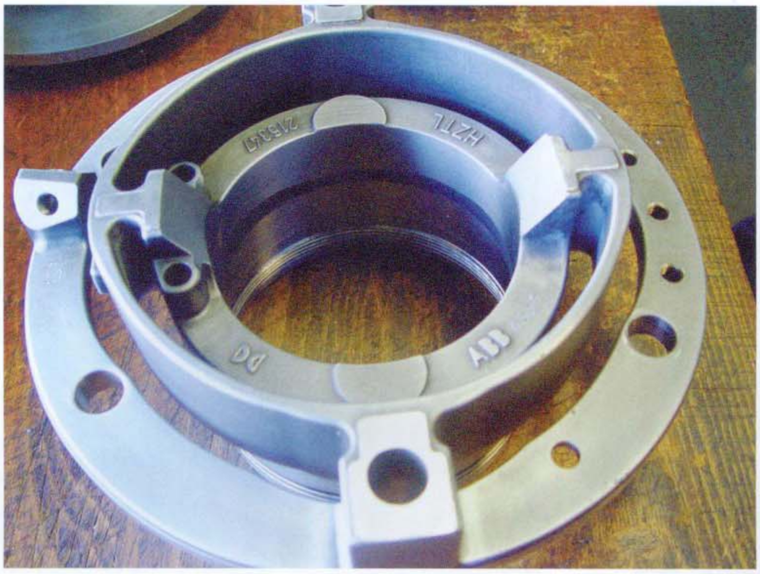
▼
Each rotor is dynamically balanced at approximately 15% average speed. Subsequently the other parts are mounted, as shown here.

▼
After arrival at the reconditioning workshop, the turboblowers are taken apart, cleaned, and measured.

▶
A new compressor-impeller is heated and then shrunk on the compressor shaft.

▶
A new bearing-casing design.

The connection between the outer ring, attached to the casing and the ball-bearing mounting ring is flexible. This is a self-aligning bearing.



◀
The exhaust-gas turbine of larger turboblowers is equipped with individual blades; these are attached to the turbine-wheel rim using blade feet.

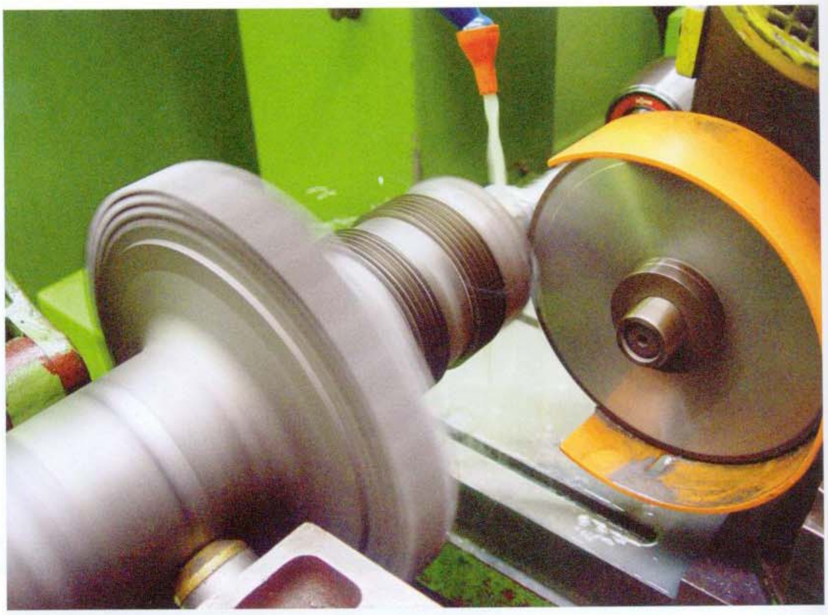
1 *turbine shaft*
2 *labyrinth seal*
3 *turbine wheel*
4 *turbine blade*
5 *lacing wire to dampen vibrations*

Casing

The materials used in the manufacture of the exhaust-gas section are exposed to exhaust-gas temperatures of 300° to 600 °C. Until recently, the exhaust-gas section was double-walled and large blowers were cooled with cooling water circulating through this section. This is still common practice in some new types of blowers and, naturally, in older types. **Corrosion and contamination** of the casing are the most frequently occurring problems. There is also the possibility of tearing at high temperatures, particularly of uncooled parts or turbine casings.

▲
Turboblower casings after painting.

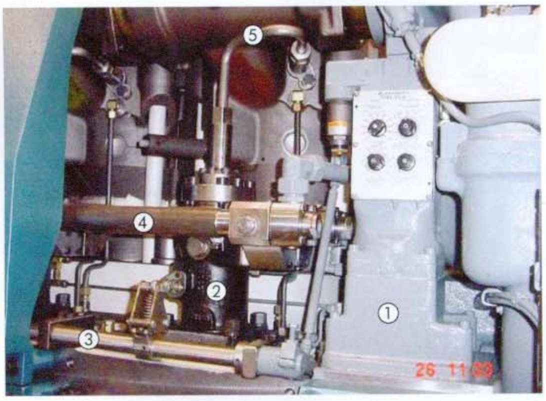
▲ A blade-attachment system of the exhaust-gas turbine on the turbine wheel. This is called the 'Christmas tree configuration'.



▲ Polishing a built-up shaft section where it was spray coated.

▶ Spare impellers and nozzle rings in a stock room.

1 *impellers*
2 *nozzle rings*



▶ A small but vital part of a diesel genset is the engine speed governor.

1 *governor*
2 *fuel pump*
3 *fuel- adjusting spindle*
4 *supply line fuel pump*
5 *delivery-line fuel pump*

26.2.9 Governors

Minor maintenance to governors is performed when they are mounted on the engine.
This maintenance comprises refilling or changing the lubricating oil, lubricating the shaft system to the fuel pumps and inspection of the governor during operation.
Reconditioning is performed by specialised companies, such as Woodward and Europa.
The governor is disassembled, cleaned and inspected.
Worn parts are replaced and the governor is tested subsequent to assembly.

See chapter 15, Speed control.

10.12.4 Temperature control is not operative or works poorly

This is probably produced by a defective temperature gauge, the control itself or a stuck three-way valve. In smaller engines the thermostatic control valve is often defective and needs to be replaced. It is easy to check if a control valve is functioning by dismantling it and placing it in a basin of water. The temperature must be at least 10 °C higher than the normal cooling water temperature. The valve should open!

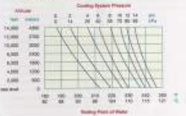
10.12.5 Contaminated coolers

This may occur on the fresh water cooling side when there is insufficient cooling water treatment. A thin corroded metal layer and other deposits on the material reduce the heat transmission. On the untreated water side or seawater side fine dirt particles that have passed the suction filter of the pump can form a thick layer of, for instance, fine silt, sand, vegetable and animal remains to the growth of shell fish such as 'Balanus' barnacles, a mollusc.

10.13 Damaged engine parts

- When cooling is insufficient the material temperature will increase to the point where the material properties will show signs of deterioration, this could cause tearing. It could also cause irreparable damage, for instance, the piston seizure due to inadequate cooling of the piston or the cylinder liner.
- Inadequate cooling of the lubricating oil system may cause the lubricating oil to decompose which leads to the deposition of carbon particles on, for instance, the lubricating oil cooler, the inside of the piston or behind the piston rings, the so-called 'ring sticking'.
- When an air cooler is contaminated the fresh air combustion temperature, and subsequently all temperatures in the combustion process increase. The temperatures of the material near the combustion chamber could also rise.
- Adequate cooling of among other things: injectors, exhaust valve seats and turbo-blower bearings and shafts. Components that are provided with bore-cooling, such as pistons, cylinder liners and cylinder heads in large engines.
  Lubricating oil that cools the pistons via a nozzle in small engines: a minimal temperature increase could cause complications.
- Material tension as a result of rapid temperature fluctuations can be fatal for an engine part. A stopped engine should not be pre-heated too quickly and increasing the shaft power of a running engine is bound to a minimum time.

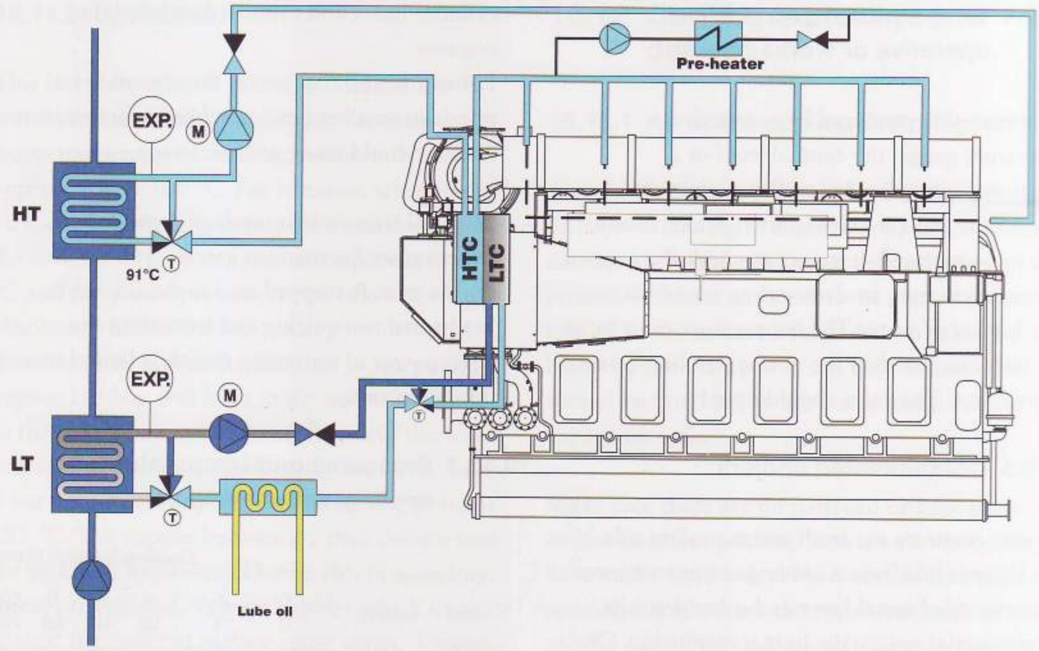
10.13.1 Pressures and temperatures in cooling-water systems



A graph with on the vertical axis the seawater level in metres and feet, on the bottom horizontal axis the boiling point of water in degrees Celsius and Fahrenheit and on the top horizontal axis, the cooling-water pressure in kPa and p.s.i. (pounds per square inch).

Example: At an altitude of 3000 metres above sea level (for instance, driving in the mountains) and a (maximum) cooling water pressure of 0.3 bar (30 kPa) excess pressure, water boils at 104 °C. At a cooling water pressure of 0.7 bar (70 kPa) excess pressure water boils at 110 °C. Without cooling water pressure water boils at 90 °C at an altitude of 3000 metres.

The pressure in cooling water systems should be sufficiently high to prevent the forming of vapour bubbles or boiling. If the cooling water pressure is lower than that of the vapour pressure at the current temperature, the water will begin to boil. Cooling water temperatures of 80 to 90 °C are fairly standard. In dual-fuel- and gas engines, cylinder head cooling water temperatures can reach 123 °C! The cooling water pressure when pumping is usually 3 to 4 bar.



A combined H.T. and L.T. cooling-water system for a four-stroke medium-speed diesel engine.

Here the lubricating-oil cooler is mounted on the L.T. system. The scavenging-air cooler consists of two heat exchangers; first the air for the engine is cooled with the H.T. system and subsequently with the L.T. system. A pre-heater is placed in the H.T. system.

10.14 Standard cooling-water system

A simple closed fresh water cooling system always consists of a fresh water cooling circulation pump, the engine parts that must be cooled, an air venting- and expansion system, a fresh water cooler and a temperature control valve, usually a three way thermostatic control valve.

10.14.1 Cooling systems according to the engine classification

Engines with a shaft power of 0 to 100 kW, four-stroke, high-speed, M.D.O. – Engine category I

The small industrial diesel engines usually have a very simple but effective cooling system.
There a various cooling methods:

A with air;
B with untreated water or seawater;
C with a closed fresh water cooling system;
D radiator cooling;
E with coolant in a closed system;
F with lubricating oil in a closed system.

A Cooling with air

Engines with air cooling are still fairly common, especially in smaller power outputs.

Cooling method

The heat of the engine is discharged at the cooling fins found on the cylinder. A ventilator driven from the crankshaft by a V-belt or an impeller integrated into the fly wheel, sends cool air

Cooling fins on the cylinder liner of a small high-speed diesel engine.

A small single cylinder air-cooled diesel power unit (arrow = air supply).

through sheet steel ducting along the fins. The heated air must be expelled far from the engine block in order to avoid it short circuiting with the cool air supply. For instance, in sailing- and motor boats or back-up gensets attention is given to the hot air discharge and cool air supply.
This system can only be applied to low load engines.

Advantages

- simple
- no risk of frost damage
- no cooling-water system required
- economical

Disadvantages

- Dirt and dust settles on the cooling fins.
- Mediocre cooling capacity.
- No temperature control.
- No sound proof cooling-water jacket.
- Supply and discharge ducting require a great deal of space in built-in engines.
- The power output has to be reduced at high air temperatures.

In practice this cooling method is seldom used.

B With raw water or seawater

This method is often applied. The canal-, river- or seawater is drawn in via an inlet- or seaweed chest by a engine driven cooling-water pump. The coolant then flows through the engine block and is discharged back into the untreated water header above the water level. The temperature is regulated by a temperature controlled valve placed in the discharge pipe after the engine. Since this system is often absent, the flow of the untreated water depends on the number of revolutions of the water pump and therefore those of the engine.

Advantage

- Simple system

Disadvantages

- The cooling-water is neither clean nor treated; the dirt particles can settle on the cooling surfaces and heat transmission is reduced.
- Potential blocking of the weed chest.
- All cooling-water needs to be drained during periods of frost.

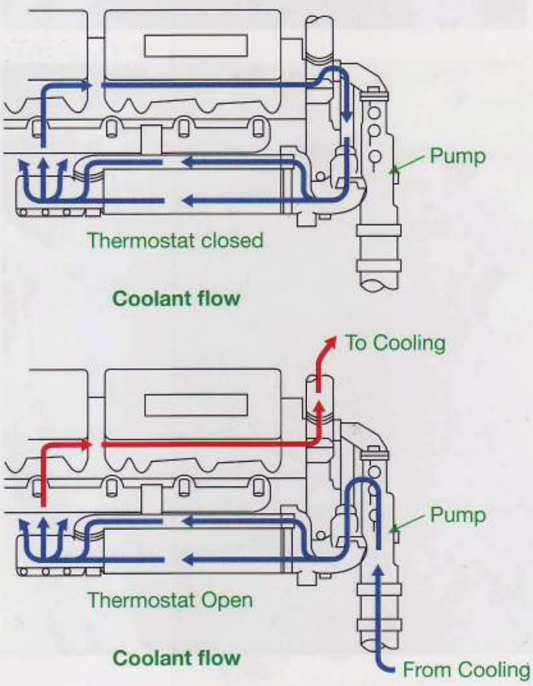
In the direct seawater cooling system it is possible for salts to be deposited on the cooling surfaces when the cooling-water temperature exceeds approximately 50 °C. This will cause scaling. Due to the high chloride content of seawater, the metal corrosion is severe.

C With a closed fresh water cooling system

This is commonly applied. The closed cooling-water system is filled with distilled water and provided with a cooling-water treatment. The system is often maintained at an over pressure to prevent from boiling and is provided with a closed expansion tank.
The heat of the engine parts that is absorbed by this closed fresh water cooling system is transmitted to a raw water or seawater system through a heat exchanger (cooler). The system, including the cooler, is designed to be as small as possible, and consists mostly of seawater resistant materials, such as stainless steel, brass and cunifer. In smaller engines the hoses may be manufactured from synthetic materials. The temperature of the first cooling-water system is controlled with a thermostatic three way valve; in an initial start in a cold engine the coolant circulates through the engine; only as the water achieves the correct temperature and in proportion to the engine load increase, does more water flow through the cooler.

Advantage

- possibility of cooling-water treatment; no corrosion



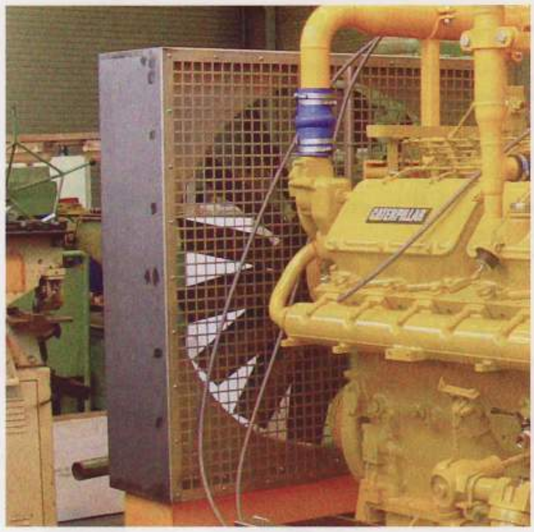
◄ A simple cooling-water system.

Top picture: The cooling system is cold. The water circulates through the engine.
Bottom picture: The cooling system has achieved the set temperature. The thermostatic control valve allows a certain amount of water through the heat exchanger, thus ensuring that the pre-set cooling-water temperature of the engine remains constant.

▶

Radiator cooling.

Whenever the heat in the cooling water cannot or with great difficulty be emitted to the surface of the cooling water, radiator cooling is used where the heat is released into the ambient air.



▲

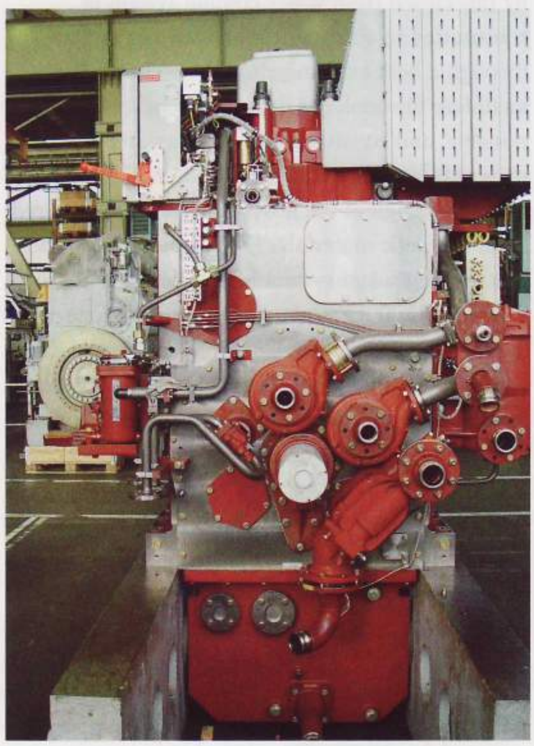
Regular deaeration prevents problems.

Properly deaerated cooling water is imperative for adequate cooling of each part. Here a (manual) deaerator vent. Regular dearation prevents problems.

▶

Cooling-water pumps mounted on the engine.

Many four-stroke medium-speed diesel engines drive several cooling-water pumps. These are driven from the crankshaft using cog wheels.



▼

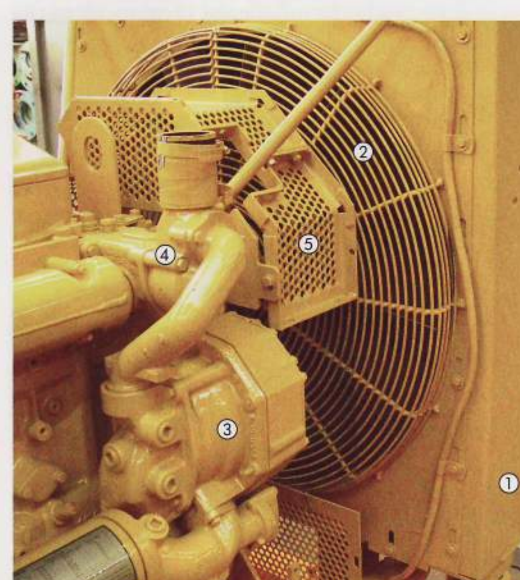
Radiator cooling.

A radiator is relatively large in comparison to the usual fresh-water cooler. Cooling with ambient air temperature sometimes in excess of 40 °C requires a large cooling surface.
The ventilator is operated by the engine by means of a belt- or gear-belt drive from the crankshaft or the camshaft.

1 *heat exchanger*
2 *ventilator (behind grill guard)*
3 *cooling-water pump*
4 *thermostatic three-way valve*
5 *grill guard.*

▼

A modern cooling-water system.



D Radiator cooling

If there is no untreated water available or if one would prefer not to use untreated or salt water in the 'second' system, one could cool to the ambient air temperature with a radiator. A radiator resembles a heat exchanger which has warm fresh water on the one side and cool air on the other, and discharges heat with an engine-driven ventilator. This is often used for 'shore' installations such as diesel gensets, mobile diesel power units in for instance large buildings such as hospitals, office buildings, factories and other facilities requiring electricity.

E With coolant in a closed system

There are engine manufacturers that prefer to have their engines filled with a ready-made coolant. This has the following properties:

- The agent contains an anti-corrosion additive; all the oxygen atoms are bound.
- The agent is resistant to temperatures below zero.
- The agent does not attack packings, rubber seals or metals.
- The agent forms a strong, thin coating on the metal parts preventing corrosion.
- This kind of cooling system often contains a liquid filter which filters out sand grains, metal particles and dirt to avoid abrasion.

Advantages

- No cooling-water treatment.
- Agent meets all the required properties.
- No problems at temperatures below zero.

Disadvantages

- Refilling after leaks is expensive.
- Prevent contact with eyes and skin. Note the user instructions.

F With a closed system with lubricating oil

Some manufacturers build a small engine series provided with this cooling system.

▲ A diesel generator with on the left the radiator cooler and on the right the electric generator with its own closed-air ventilation system. The air for the electric generator is cooled by fresh water in a heat exchanger.

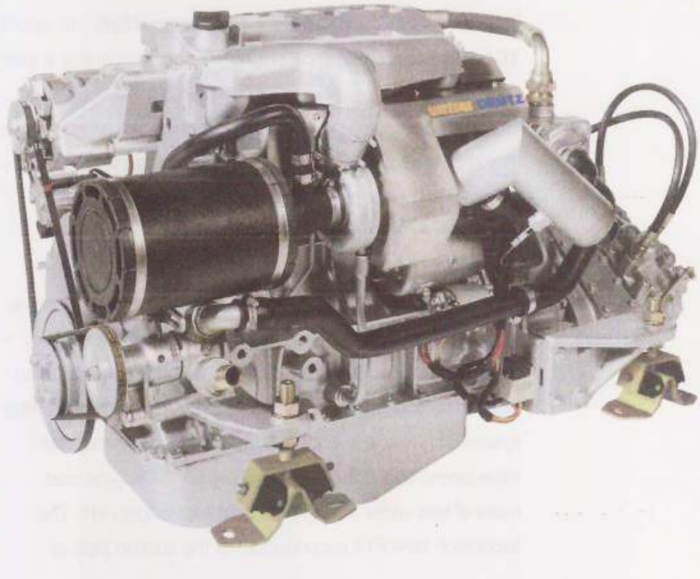
10.14.2 Engines with a shaft power of 100 to 5000 kW, four-stroke, high-speed, M.D.O. – Engine category II

These large industrial engines all have closed fresh water cooling systems:

- **with a secondary raw water or seawater system;**
- **with a secondary system to the ambient air cooled with a radiator.**

Both the fresh water circulating pump and the raw cooling-water circulating pump are driven by the engine. Since most of these engines are provided with a turbocharger, an inter cooler cooled by the circulating water in the block has been placed after the turbo-blower. This is insufficient for highly loaded engines, and therefore the cooling system is provided with a high- and low temperature system. This is often referred to as the H.T.- and L.T.- system.

▼ The Deutz D 4.29 with natural aspiration and the Deutz DT 4.29 with supercharged air; both diesel engines are cooled with lubricating oil.



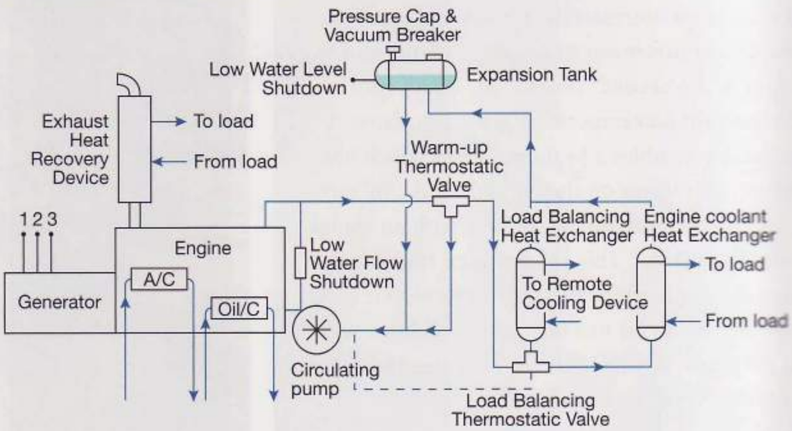
▶

A H.T. cooling-water system by Caterpillar.

This has a:

- low-water flow shut-down;
- exhaust-heat recovery device;
- two thermostatic control valves; a warm-up thermostatic valve and a load-balancing thermostatic valve.
- the expansion tank is under excess pressure.

High Temperature Water System



10.14.3 Engines with a shaft power of 500 to 30,000 kW, four-stroke, medium-speed, H.F.O. – Engine category III

These large industrial engines have closed fresh water cooling systems, often with a H.T.- and L.T.-system where cooling of both the air cooler and the lubricating oil cooler is done by the L.T.-system and the engine block by the H.T.-system.

At these high shaft outputs, a large amount of heat is released; the cooling systems therefore increase in size proportionally .

Following is an example of a Caterpillar– MaK 12 M 32 C diesel engine, a V-engine running on H.F.O.

shaft power Pas = 5760 kW

heat dissipation – engine block	800 kW
air cooler I	1880 kW
air cooler II	530 kW
lubricating oil cooler	840 kW
gear box and generator coolers	223 kW
total	4273 kW

The H.T.-cooling absorbs heat from the engine block and air cooler I = 2680 kW.
The L.T.-cooling circuit absorbs heat from air cooler II, the lubricating oil cooler and the gear box /generator coolers = 1593 kW.

▶

A cooling-water system of a Caterpillar–MaK 16 M 32 C diesel engine .

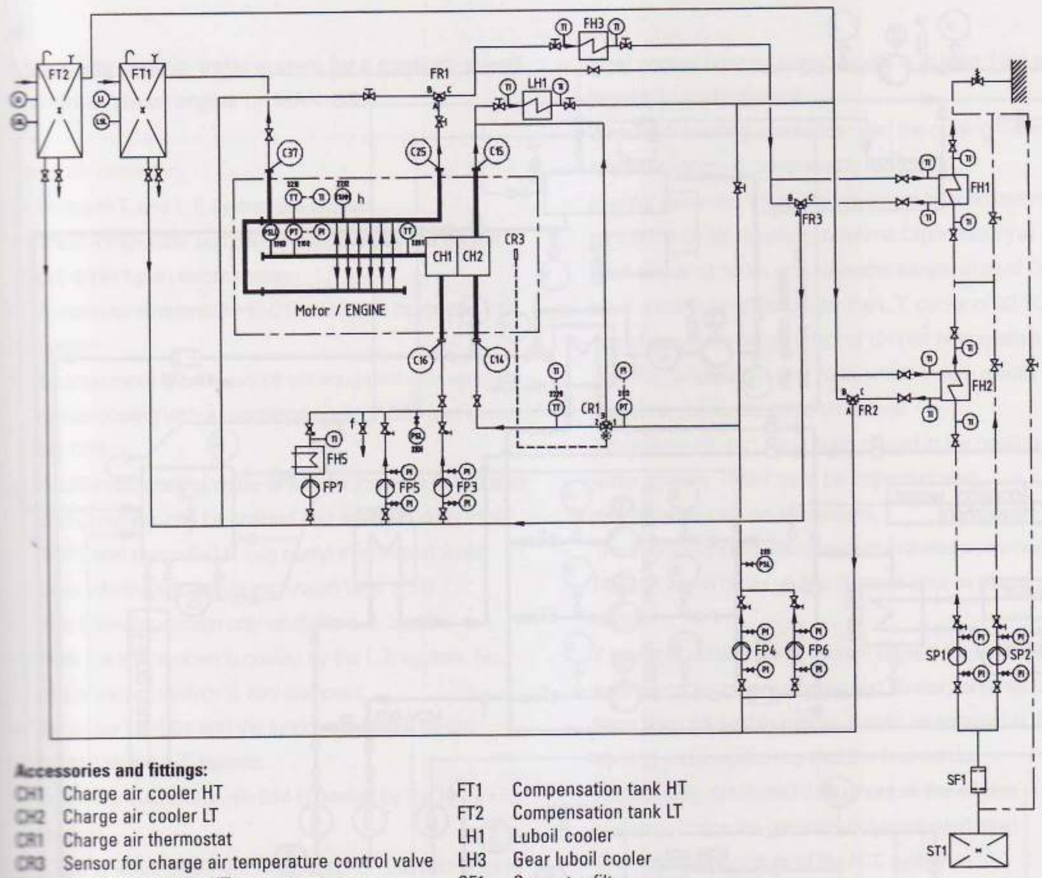
Description

The engine has a two-stage air cooler with CH1 and CH2. The system consists of a H.T. system and fresh-cooling water pumps FP3 and FP5.

The H.T. cooling-water system cools air cooler CH1 and the engine block and then flows to the thermostatic three-way valve FR1. In a cold system it circulates back to the suction side of the cooling-water pumps and at the moment that the system achieves the correct temperature, the three-way valve opens and delivers, depending on the engine load, more of less water through the heat exchanger FH1. The expansion tank FT1 is connected to the suction pipe of pumps FP3 and FP5. It exerts a sufficient amount of static pressure on the pump to have it start effortlessly as the pump is always filled with water.

Pre-heating

The H.T. system has its own pre-heater pump FP7 and pre-heater FH5. The L.T. system cools the air cooler CH2 and lubricating-oil cooler LH1. The L.T. circulating pumps FP4 and FP6 first pump the water through the air cooler in order for the air temperature to drop as much as possible. This is usually to 40 °C.
The thermostatic three-way valve FR2 ensures that the L.T. temperature is constant and delivers more or less water through the L.T. cooler FH2.
This L.T. system also has its own expansion tank FT2 with a connection to the suction side of the pumps.



Accessories and fittings:

- CH1 Charge air cooler HT
- CH2 Charge air cooler LT
- CR1 Charge air thermostat
- CR3 Sensor for charge air temperature control valve
- FH1 Freshwater cooler HT
- FH2 Freshwater cooler LT
- FH3 Heat Consumer
- FH5 Freshwater preheater
- FP3 Freshwater pump (separate) HT
- FP4 Freshwater pump (separate) LT
- FP5 Freshwater stand-by pump HT
- FP6 Freshwater stand-by pump LT
- FP7 Preheating pump
- FR1 Temperature control valve HT
- FR2 Temperature control valve LT
- FR3 Temperature control valve HT
- FT1 Compensation tank HT
- FT2 Compensation tank LT
- LH1 Luboil cooler
- LH3 Gear luboil cooler
- SF1 Seawater filter
- SP1 Seawater pump
- SP2 Seawater stand-by pump
- ST1 Sea chest
- LI Level indicator
- LSL Level switch low
- PI Pressure indicator
- PSL Pressure switch low
- PSLL Pressure switch low
- PT Pressure transmitter
- TI Temperature indicator
- TSHH Temperature switch high
- TT Temperature transmitter (PT 100)

Notes:

- f Drain
- h Please refer to the measuring point list regarding design of the monitoring devices

Connecting points:

- C14 Charge air cooler LT, inlet
- C15 Charge air cooler LT, outlet
- C16 Charge air cooler HT, inlet
- C25 Cooling water, engine outlet
- C37 Vent

General notes:

For location, dimensions and design (e. g. flexible connection) of the disconnecting points see engine installation drawing.

With skin cooler not required:
- Seawater system (SP1, SP2, SF1, ST1) Temp. - control valve FR3 required, if heat recovery installed.

Raw-water cooling system

This system is designed to be as short as possible due to potential corrosion and contamination risks. Via the untreated or seawater inlet filter SF1 the sea- water circulating pumps SP1 and SP2 draw in cooling water, pump it to the L.T. and H.T. coolers FH2 and FH1 and then the water is discharged overside. Re-circulation is also possible using a manual valve. Furthermore, both coolers are equipped with a 'bypass' valve.

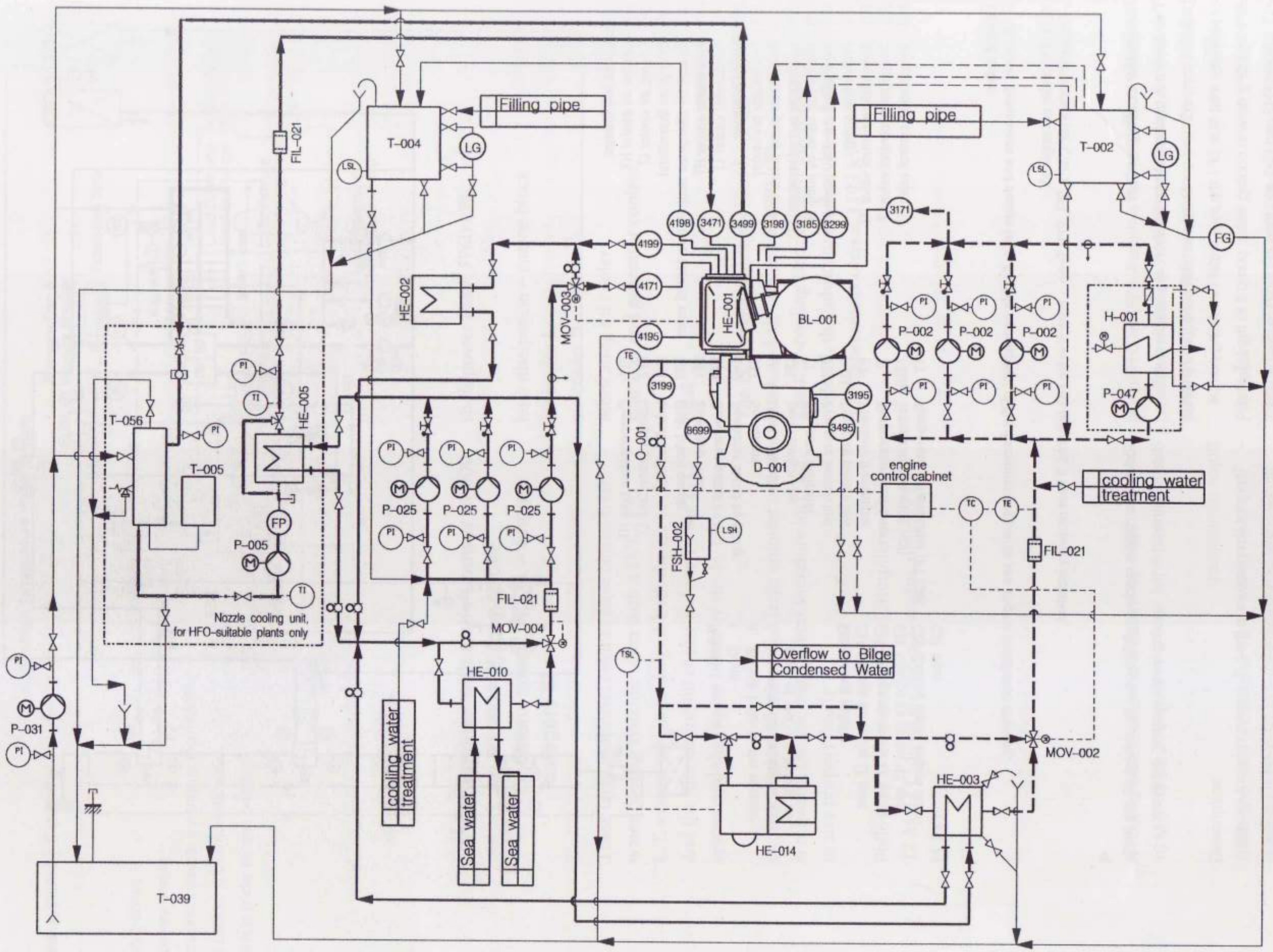
General

Pressure and temperature gauges are fitted throughout the system. Furthermore, the expansion tanks are provided with low-water alarms. Before and after each cooler; thermometers are installed in order to provide accurate readings of temperature differences in the cooler.

Today, this data is made available on a pc, where all required data is gathered in the 'cooling-water system' databank.

Some commentary

- The inlet temperature in the L.T. system may not exceed 38 °C. These temperatures are only achieved in tropical areas, shallow coastal waters and estuaries.
- Expansion tanks should be installed at least four metres above the centre line of the crankshaft of the engine.
- There is a possibility of inserting an oil detector in the cooling-water system.
- Flow velocities in the fresh-water systems on both the suction and delivery sides are 2 to 3 m/sec.


Filling pipe
T-004
LG
LSL
FIL-021
4198
3471
3499
3198
3185
3299
3171
Filling pipe
T-002
LG
LSL
FG
4199
4171
4195
MOV-003
HE-002
HE-001
BL-001
P-002
P-002
P-002
H-001
P-047
T-056
T-005
HE-005
FP
P-005
Nozzle cooling unit,
for HFO-suitable plants only
P-025
P-025
P-025
3199
3195
3495
8699
O-001
D-001
engine
control cabinet
cooling water
treatment
TC
TE
FIL-021
FSH-002
LSH
FIL-021
MOV-004
HE-010
Overflow to Bilge
Condensed Water
TSL
MOV-002
HE-003
HE-014
cooling water
treatment
Sea water
Sea water
P-031
PI
T-039

◄

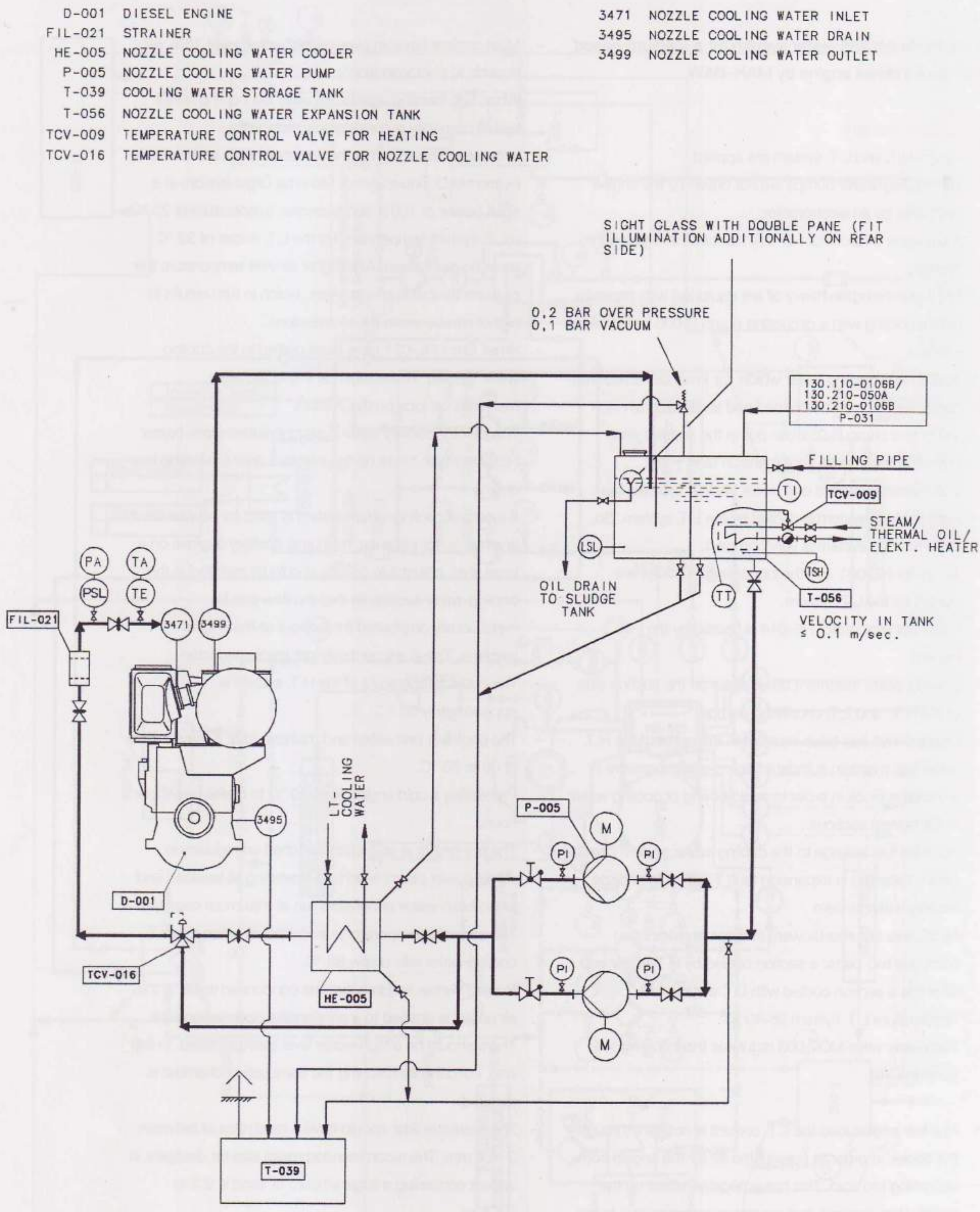
A complete cooling-water system for a medium-speed four-stroke diesel engine by MAN–B&W.

Some commentary

- Here a H.T. and L.T. system are applied.
- The cooling-water pumps are not driven by the engine but rather by an electromotor.
- A seawater evaporator HE-014 has been integrated into system..
- Engines running on heavy oil are equipped with separate nozzle cooling with a circulating pump P-005 and cooler HE-005.
- Additionally, cooling water which, for instance, is required during repairs, can be drained and stored in drain tank T-039 and pump P-031 can pump the drained water back into the system via expansion tank T-002.
- The seawater system only cools the L.T. system.
- Here the H.T. system is cooled by the L.T. system. So, this seawater system is very compact.
- Air cooler HE-001 and the lubricating-oil cooler are cooled by the L.T. system.
- Seawater evaporator HE-014 is heated by the H.T. system.
- Cooling-water treatment takes place on the suction side of the H.T. and L.T. circulating pumps.
- Orifice 0-001 has been inserted to the outlet of the H.T. system to maintain sufficient cooling-water pressure in the engine block in order to avoid boiling of cooling water in the hottest sections.
- Potential fuel leakage to the cooling-water system can be clearly detected in expansion tank T-005 of the nozzle cooling-water system .
- Air coolers of turbo-blowers of these engines often comprise two parts: a section cooled by H.T. water and after this a section cooled with L.T. water.
- Temperature L.T. system 36-40 °C.
- Three-way valve MOV-003 regulates the following temperatures.

1. At a low engine load the L.T. coolant is not sent through the cooler, in order to prevent the air for the engine from becoming too cold. This has a negative effect on the combustion process and promotes corrosion due to the presence of sulphur in heavy oil.
2. It reduces the amount of condensation which is formed in the cooler in tropical conditions.
   The air-inlet temperature is continuously compared with the outside-air temperature and if this rises, the air-inlet temperature increases accordingly in order to avoid condensation of water vapour.

- Most coolers have an over-capacity of at least 15% with regards to contamination.
- When anti-freezing agents are used the cooling-water system capacity is considerably reduced.
- In order to remain within the maximum NOx emissions set by the I.M.O. (International Maritime Organisation) at a shaft power of 100% and seawater temperature of 25 °C, a L.T. system temperature for the L.T. cooler of 32 °C must be maintained. At a higher air-inlet temperature the process temperature also rises, which in turn results in higher nitrous-oxide (NOx) emissions.
- Small filters FIL-021 have been placed in the cooling-water system. These must be inspected after maintenance jobs on the system.
- The injector-cooling water system includes a pre-heater HE-500 which heats up the injectors prior to starting the engine.
- If a central cooling-water system is used for several diesel engines or, for instance, main and auxiliary engines on a large ship, adjustable orifices should be installed in the cooling-water system so that the flow can be meticulously distributed throughout all the engine systems. These are generally not readjusted later!
- The outlet temperature of the H.T. system is approximately 90 °C.
- The engine is preheated and maintained at a temperature of up to 60 °C.
- Preheating a cold engine from 10 °C to 60 °C takes four hours!
- The pre-heater is also often switched on regularly in diesel power plants which are operating at low load and when fresh-water generators run at maximum capacity.
- The seawater evaporator is switched off when the H.T. cooling water falls below 88 °C.
- Bureau Veritas requires that the condensed water in the air cooler is drained to a condensate observation tank. There should be a high-water level alarm installed. In this way, condensate reaching the combustion chamber is avoided.
- The seawater filter should have a mesh size of between 2 to 4 mm. The recommended mesh size for dredgers in waters containing a large amount of sand is 0.3 to 0.5 mm.



▲
The nozzle cooling-water system in MAN–B&W four-stroke diesel engines.

Some remarks

The expansion tank is fitted as an observation tank to detect potential leakage of fuel into the cooling water for the nozzle. Possible oil residue can be funnelled off. A fine-filter FIL-021 prevents dirt particles from obstructing the narrow passages in the nozzles.

- The cooling water is preheated to between 60 and 80 °C.
- Depending on the engine type the cooling-water inlet temperature for the nozzle is:
  type 40/54 and 58/64 80 to 85 °C
  type 32/40 and 48/60 55 to 60 °C

There are complete modules available for nozzle cooling-water systems. These modules are frame-mounted.

10.14.4 Engines with a shaft power output of 1500 to 100,000 kW, two-stroke, low-speed, H.F.O. – Engine category IV

The large two-stroke crosshead engines have beside the standard cooled parts, two additional special systems:

1 a **separate piston cooling system**; cooling with both water and lubricating oil;
2 a **cooling system for the crosshead guides**; this is connected to the lubricating-oil system.

General

The engines in this category with shaft power outputs up to 100,000 kW have to discharge enormous amounts of heat. For a MAN–B&W 12 K 98 MC with a shaft power of 68,640 kW the following values can be read in the 'Project Guide':

air coolers	27,390 kW
lubricating-oil cooler	5,760 kW
cylinder cooling-water cooler	9,920 kW
	63,070 kW

A total amount of 63 MW of heat is discharged into the seawater!

The circulating pumps also have a high capacity:

Fresh water cooling pump	590 m^3 per hour
Sea water cooling pump	770 m^3 per hour

In these extremely large diesel engines the 'consumption figures' are also large!

air consumption per hour	635,040 kg
fuel consumption per hour	11,737 kg
lubricating-oil consumption per hour	61 kg
shaft power output of this giant	68,640 kW

Details

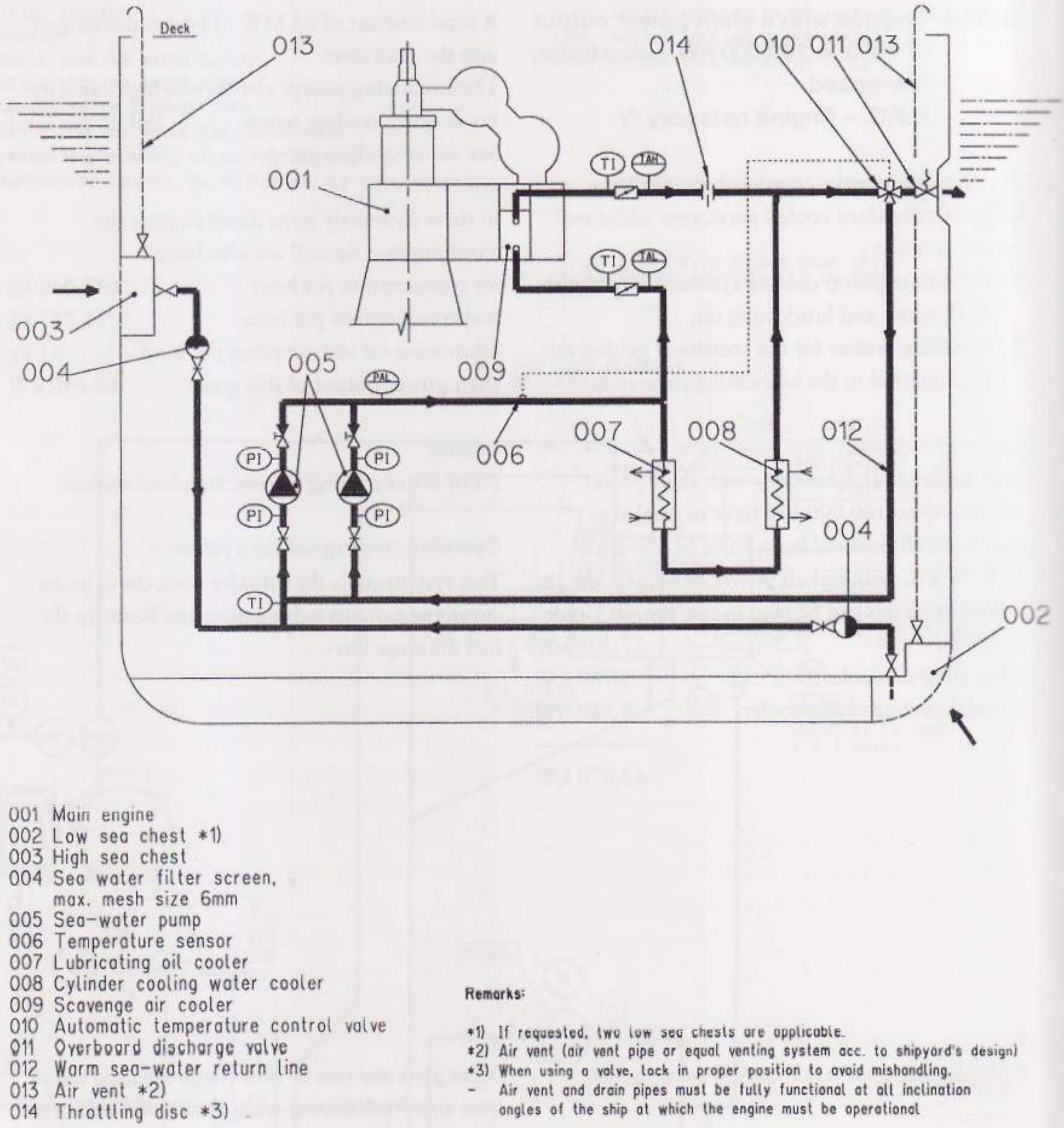
Fresh water cooling system. Standard system.

Cylinder cooling-water system

This system cools the cylinder liner, the cylinder cover, the exhaust valve casing and heats up the fuel drainage lines.

▼ Both plate coolers of a H.T. cooling-water system.

▼ Diesel generator sets on board large seagoing ships often share their cooling-water system with that of the main engine when at sea.



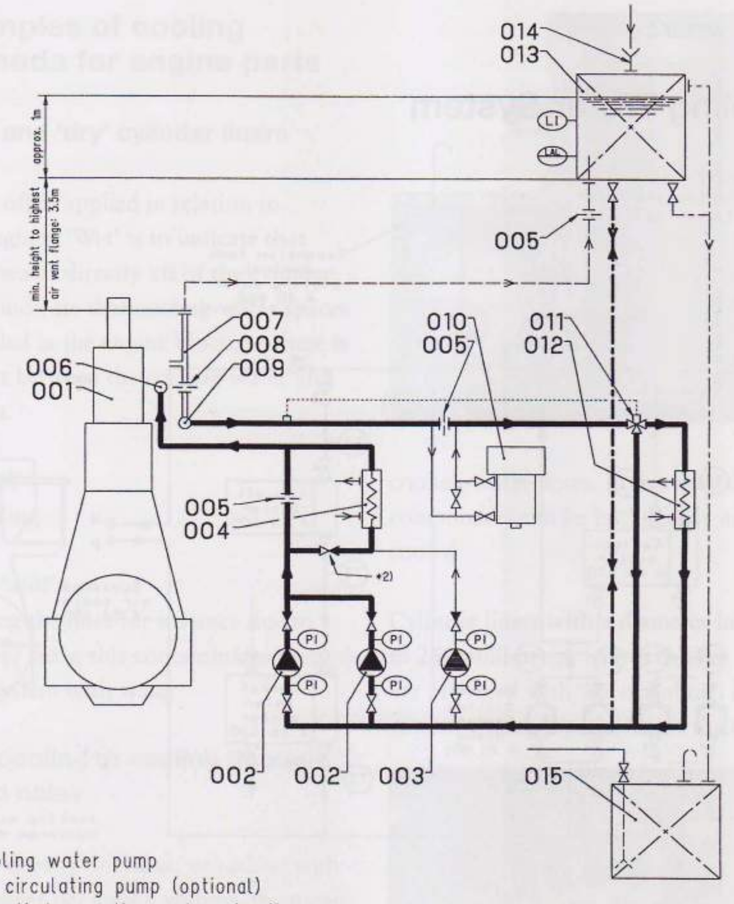
▲
The cooling-water system of a two-stroke crosshead engine of MAN–B&W.

The cylinder cooling-water pumps draw the water from the cylinder cooling-water cooler and force the water through the engine.

- An orifice has been fitted to the cooling-water outlet to maintain sufficient cooling-water pressure in the engine.
- The preheated cooling water can also be used to heat the seawater evaporator and is then returned to the cylinder cooling-water cooler.
- The thermostatic valve sends more or less water to the cooler and responds to the temperature sensor which is placed in the engine cooling-water outlet.
- A deaerator tank prevents air from entering the system. The air rises to the cooling-water expansion tank. If air is in the system, this is indicated by an alarm.
- Several pipelines are connected to the expansion tank. They are all mounted on the top of objects, such as the:
- main engine;
- deaerator tank;
- cylinder cooling-water cooler;
- auxiliary engines.
- The heater of the seawater evaporator should be opened gradually in order to avoid fluctuations in the cooling-water temperature. The minimum is a period of three minutes.
- The expansion tank must be placed at least five metres / higher than the highest point of the discharge cooling-water pipe of the engine. The discharge cooling-water temperature of the three generators is kept constant at 80 °C using three-way valves.

Port operations

A stopped main engine is heated by stand-by generators. As the preheating capacity of a low-load generator is probably too low, a pre-heater must be installed.
The pre-heater is on stand-by when the stop valves A are closed and the stop valve B is open. The engine-driven generator cooling-water pumps pump the heated water from the top to the bottom through the engine towards the thermostatic three-way valve.



001 Main engine
002 Cylinder cooling water pump
003 Pre-heating circulating pump (optional)
004 Heater for cylinder cooling water circuit
005 Throttling disc *1)
006 Cylinder cooling water inlet
007 Air vent pipe
008 Throttling disc (adjustable, on engine)
009 Cylinder cooling water outlet
010 Freshwater generator
011 Automatic temperature control valve
012 Cylinder cooling water cooler
013 Cylinder cooling water expansion tank
014 Filling pipe/inlet chemical treatment
015 Cooling water drain tank

Remarks:
*1) When using a valve, lock in proper position to avoid mishandling.
*2) Only when pos. 003 is installed
- Air vent and drain pipes must be fully functional at all inclination angles of the ship at which the engine must be operational.

Fresh water pipes
Balance pipe
Water drain pipes
Air vent pipes
Pipes on engine/pipe connections

▲
A standard cooling-water system with a relatively large seawater-cooling system in a two-stroke crosshead engine.

Brief description

The two main seawater cooling-water pumps 005 draw the seawater in via a low-inlet chest 002 or in shallow water via the high-inlet chest 003. The sea- water passes through the seawater filter 004 with a mesh size of between 2 to 4 mm. The water flows through the lubricating-oil cooler(s) 007 and subsequently through the cylinder cooling-water cooler 008. Parallel to these two coolers the water flows through the scavenging- air coolers 009 on the engine. A temperature control valve 010 measures the inlet water temperature before the cooler and circulates the water or partially discharges the water with an overboard discharge valve 011.

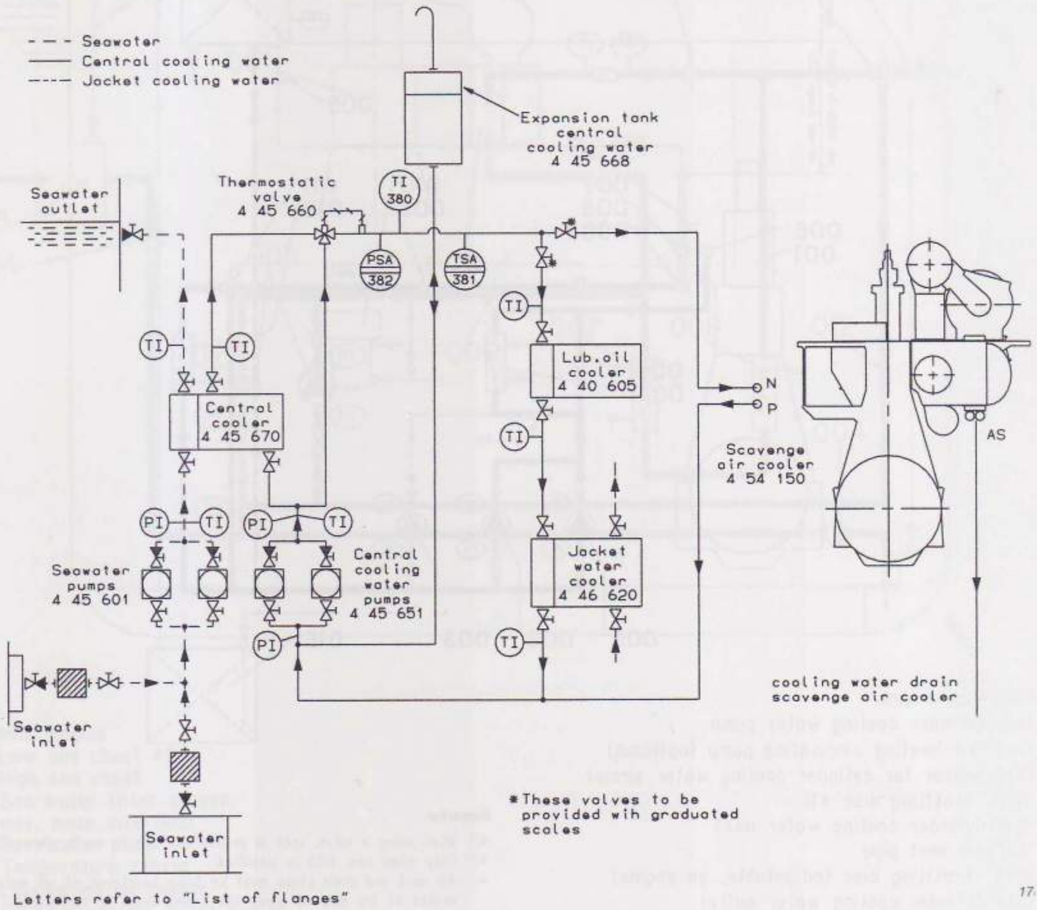
Temperature control conditions

1. The inlet seawater temperature before the lubricating-oil cooler should not drop below 10 °C to prevent the lubricating oil from becoming viscous.
2. The lowest possible seawater inlet temperature for the air cooler must be applied to keep fuel consumption as low as possible.

All the parts of the seawater system are manufactured from seawater resistant materials.
The auxiliary engines are provided with seawater cooling for the lubricating-oil and air coolers via a non-return valve.

Central cooling-water system

Central Cooling Water System



▲

A modern cooling-water system with a short seawater-cooling section.

In a short seawater cooling system, corrosion is kept to a minimum. Furthermore, the seawater section is manufactured from corrosion-resistant materials such as stainless steel and cunial, a copper nickel aluminium alloy. The most remarkable feature of this system is that the seawater only passes through one cooler, the central cooler. The seawater system is, therefore, very short with high and low seawater inlets, a suction filter, two seawater circulating pumps, a central cooler after which the water is pumped overboard.

The fresh-water circulating system consists of two pumps which deliver the water through the central cooler and the air cooler which is arranged parallel to the lubricating-oil cooler and the cylinder cooling water.
An elevated expansion tank has a vertical connecting pipe to the suction side of both fresh water circulating pumps, so they can pump water immediately the engine is started.
A temperature-control valve ensures a constant temperature in the engine supply.
The cooling-water temperature to the air cooler must remain as low as possible in order to achieve optimum engine efficiency.
Consequently the temperature of the fresh-water cooling system is kept as low as possible by the thermostatic control valve and follows the seawater temperature when it rises above 10 °C.

10.15 Examples of cooling methods for engine parts

10.15.1 'Wet' and 'dry' cylinder liners

These terms are often applied in relation to smaller diesel engines. 'Wet' is to indicate that there is cooling-water directly aft of the cylinder liner. 'Dry' is to indicate that cooling-water spaces have been installed in the engine block, so there is no direct contact between the cooling-water and the cylinder liner.

Advantage 'wet'

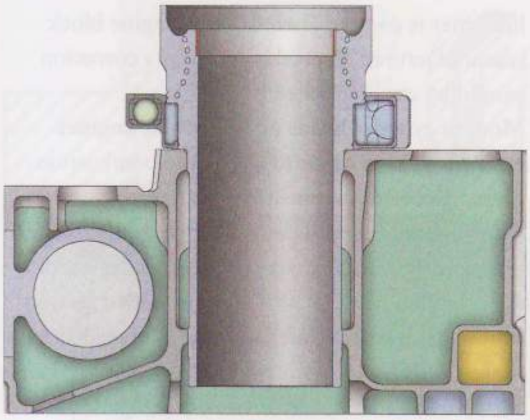
- superior cooling

Disadvantage 'wet'

- leakages along the liner for instance due to leaking rubber seals; this contaminates the lubricating system with water

10.15.2 Bore cooling or cooling through drilled holes

With the 'power density' increase, or rather: with an increase in power per stroke volume, the mean effective pressure increases. This can only be accomplished when compression and combustion pressures increase significantly. Therefore, the forces on the cylinder head, the cylinder liner and the piston increase. Consequently, the wall thickness of the engine components must be increased, but at the same time the material must be adequately cooled. This is made possible by drilling holes fairly close to the surface of the combustion chamber walls through which the cooling-water flows. In this way the various components can be both sturdy and thoroughly cooled

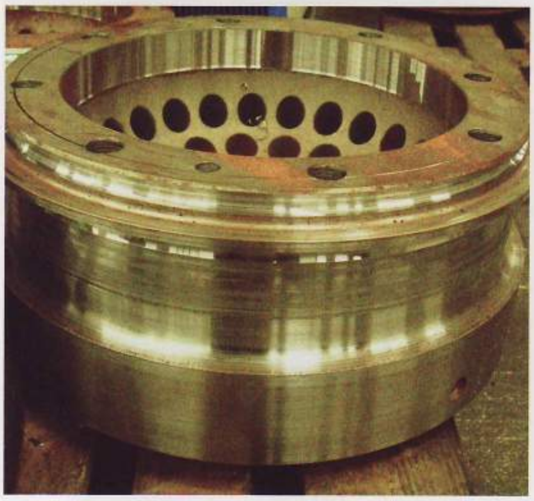


◄

A cylinder liner of a four-stroke medium-speed diesel engine of the Wärtsilä type 64 series.

Only the cylinder liner protruding from the block is cooled. The entire block is dry.

Cylinder liners with a diameter in excess of 200 to 250 millimetres have a thicker upper rim and are provided with 'bore cooling'. In some of the current engine brands, this is the only part of the



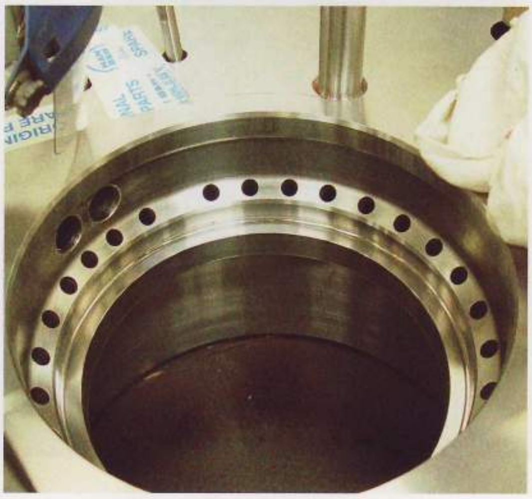
◄

A piston in a MAN–B&W four-stroke medium-speed diesel engine with bore cooling.

▲

Pistons are usually cooled with lubricating oil.

Shown here a piston crown of a two-stroke crosshead engine of MAN–B&W. Note the drilled holes in the circumference.



▲

A cylinder liner in a MAN–B&W four-stroke medium speed diesel engine with bore cooling.

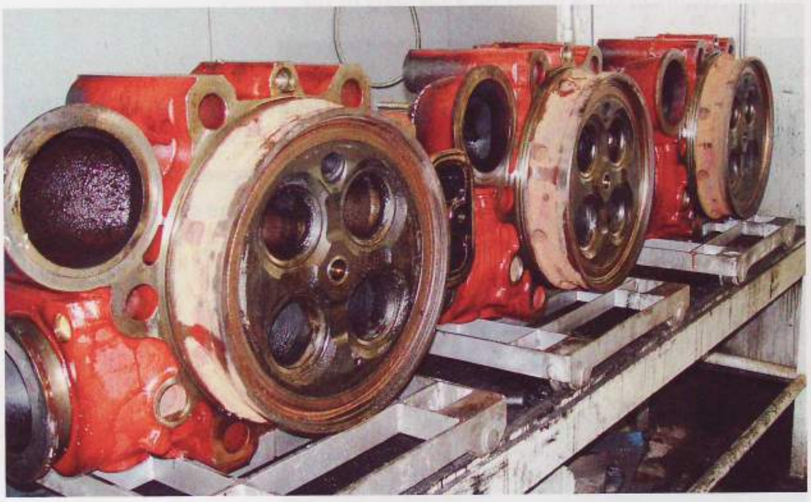
liner that is cooled. Therefore the engine block is manufactured 'dry' which prevents corrosion caused by cooling-water.

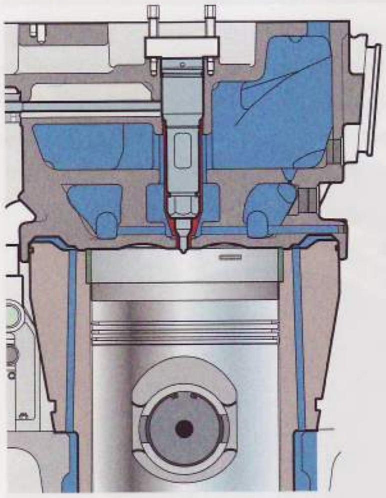
Modern cylinder heads of four-stroke engines are very rigid in order to withstand combustion pressures of 200 bar or more without deformation; this can cause gas- or cooling-water leakages. The bottom part of the cylinder head, the so-called flame plate is a thick walled section of the cast iron cylinder head and is provided with 'bore-cooling'.

Modern cylinder heads of two-stroke-crosshead engines are manufactured from forged steel and entirely solid. Centrally positioned is a hole for the exhaust valve including casing, holes for the injectors, starting valve, discharge valve and an indicator cock.

This solid head is provided with bore cooling to ensure effective cooling.

The thick flame plates of the cylinder heads in a Caterpillar–MaK 32 four-stroke engine.





A cross-section of the combustion space in a four-stroke diesel engine.

Note the thick flame plates are provided with bore cooling.

In modern diesel engines **pistons are both** thermally and mechanically heavily loaded. This requires them to be both strong and adequately cooled. Here the forged- or cast steel piston crown has also been provided with drilled holes in the crown.

In two-stroke crosshead engines the piston crown is provided with drilled holes which are injected with lubricating oil by means of nozzles.

Large cylinder covers in crosshead engines.

Large cylinder covers in crosshead engines are almost completely solid and provided with bore cooling. The water capacity of these large cylinder covers amounts to a mere tens of litres.

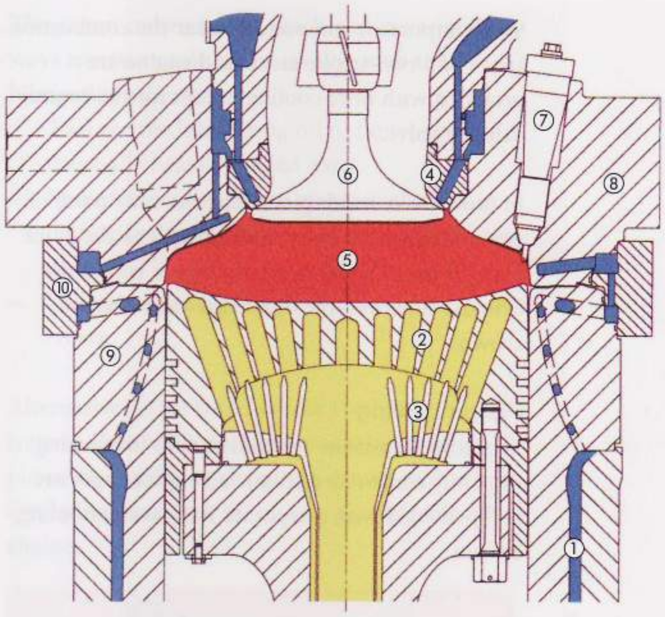
Piston crowns in modern four-stroke medium-speed diesel engines are highly loaded and therefore thoroughly cooled with lubricating oil.

▶

A cross-section of a large Wärtsilä Sulzer RTA two-stroke crosshead engine.

The parts around the combustion space are solid and provided with bore cooling.

1 *drilled channels for fresh-water cooling .*
2 *drilled channels for lubricating-oil cooling*
3 *lubricating-oil spray lances*
4 *deep-cooled valve seat*
5 *combustion space*
6 *exhaust valve*
7 *fuel injector*
8 *cylinder head*
9 *cylinder liner*
10 *cooling-water ring*



▲

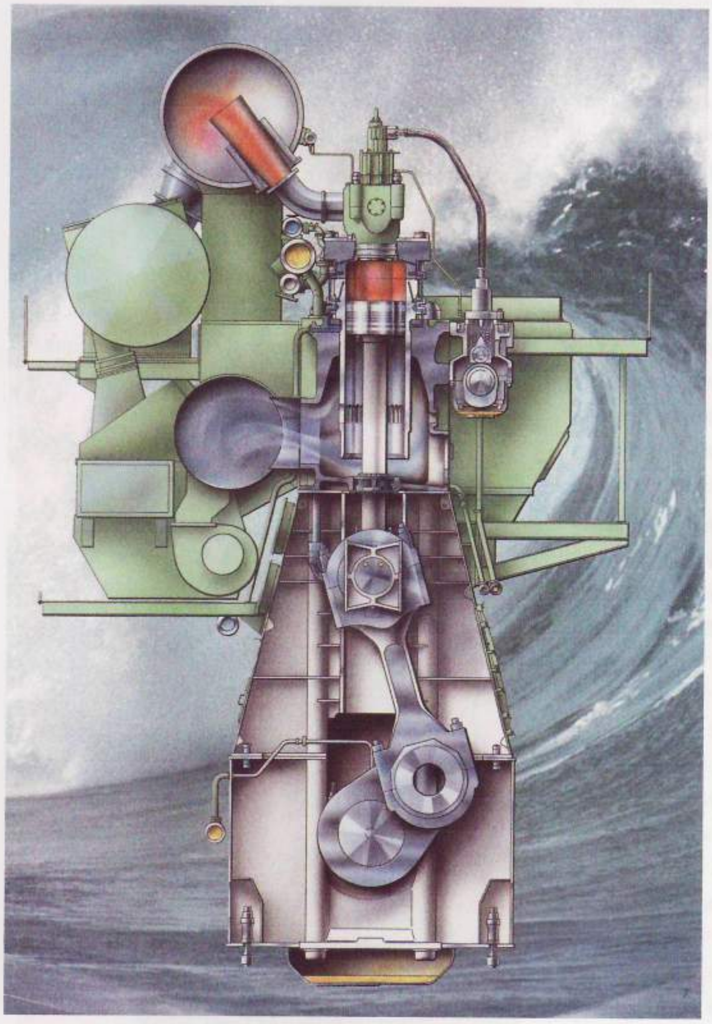
Nozzle cooling with lubricating oil in a two-stroke crosshead engine, a Wärtsilä Sulzer RTA.

The spray lances fit exactly in the drilled holes of the piston crown shown here.

▲

Nozzle cooling with lubricating oil of a two-stroke crosshead engine, a Wärtsilä Sulzer RTA.

The spray lances shown above fit exactly in the drilled holes of the piston crown.



▲

The combustion space of this two-stroke crosshead engine from MAN–B&W type MC has a completely solid combustion space.

So all important components near the combustion space of a two-stroke crosshead engine are provided with bore cooling except for the central exhaust valve.

In **four-stroke trunk piston engines** only parts of components have bore cooling, such as the flame plate of the cylinder head, the upper rim of the cylinder liner and the outer rim of the piston crown.

Piston cooling

Virtually all pistons are cooled with lubricating oil. Only some older crosshead engine types are still provided with a separate fresh water cooling system.

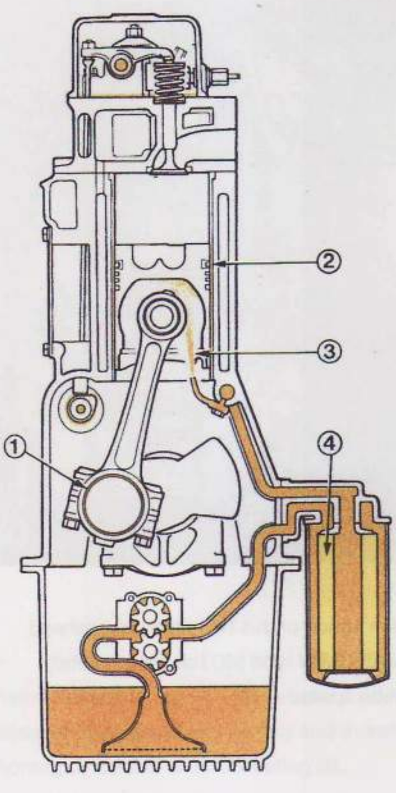
10.16 Examples according to the engine classification

10.16.1 Engine category I – 0 to 100 kW, four-stroke, high-speed, M.D.O.

All pistons are cooled with lubricating oil. Lubricating-oil supply to the piston can occur in various ways.

A. Via a nozzle mounted on the bottom rim of the cylinder liner.

The oil jet sprays **continuously** and directly into the piston and this way cools the inside of the piston.



▶ Nozzle cooling of the piston in a small high-speed four-stroke diesel engine of DAF.

The heated lubricating oil falls back into the sump. The cooling is rather limited; a relatively large amount of lubricating oil must be supplied. In this simple system this is referred to as the absorption gradient; this is the part of the lubricating oil which emerges from the nozzle and is used as the piston coolant.

It is very important that the lubricating-oil injection path is parallel to the heart of the cylinder.

If the nozzle including the supply line is only a few degrees misaligned, the absorption gradient and consequently the piston cooling is adversely affected.

It is then possible that the piston could stick.

▲ A completely seized piston.

The subsequent damage can be vast.

B. The lubricating oil is pumped under pressure from the crankshaft through a perforation in the connecting rod. Via the piston pin and a through perforation it is led into the cooling spiral in the light metal cast piston.

This system has a far better cooling effect. The lubricating oil velocity and flow in the coil has to be reasonably high. After cooling, the oil freely flows back into the sump.

C. A spray nozzle in the small or large end of the connecting rod. This is a also form of spray cooling. Here the cooling effect is moderate.

10.16.2 Engine category II – 100 to 5000 kW, four-stroke, high-speed, M.D.O.

All the pistons are cooled with lubricating oil. In engines with a high load which require intensive cooling, the simple cooling systems with spray cooling are less prevalent. As aforementioned, the lubricating oil is pumped to the piston through the connecting rod and cools the piston via a cast cooling spiral. For the somewhat larger cylinder bores, cast iron pistons are sometimes still used. Here the stroke/bore ratio is normally slightly larger than that of the light metal pistons and directly linked to the somewhat higher stroke, the RPM is lower so that the mean piston speed does not increase.
In this category II one also finds assembled pistons, but since the various categories have some overlaps; these will be discussed in category III.

It is evident that the construction of the piston and also the cooling of the larger engines is far more complex than that of smaller engines.
This is primarily due to the much larger power output that, for instance, is generated per litre stroke volume. The mechanical and thermal load are consequently also larger!

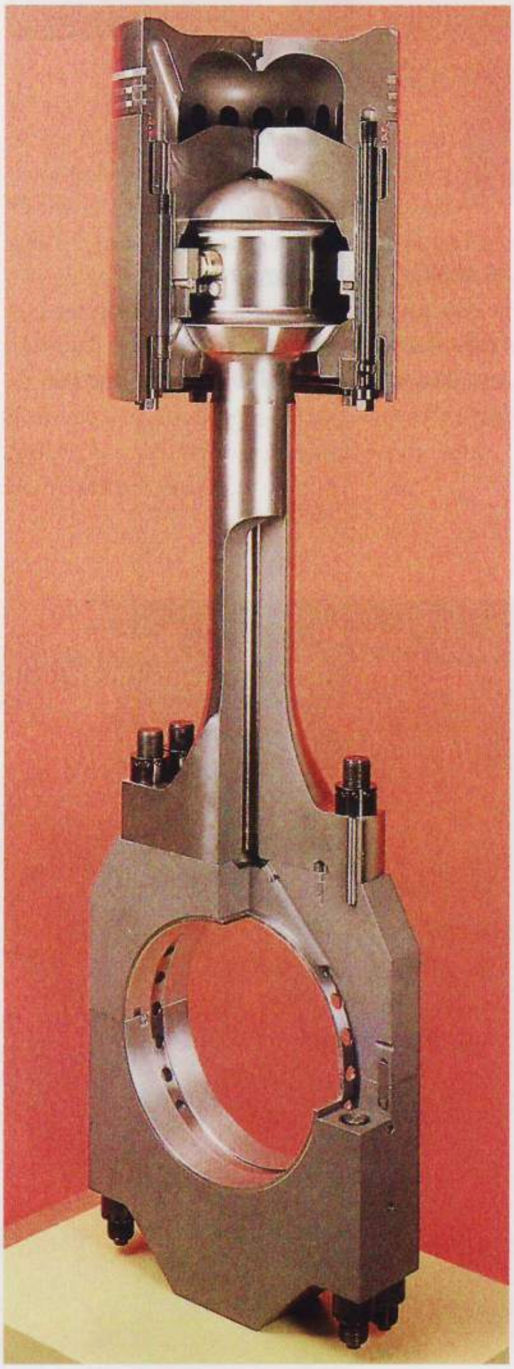
10.16.3 Engine category III – 500 to 30,000 kW, four-stroke, medium-speed, H.F.O.

Piston construction

These pistons usually consist of two parts; the steel piston crown and the cast iron or light metal skirt.
The oil supply to the inside of the piston can take place in various ways:
- via the connecting rod to a nozzle on the head of the small end where it sprays against the piston crown;
- via the connecting rod to the gudgeon pin and from here to the piston and the piston crown via bore holes.

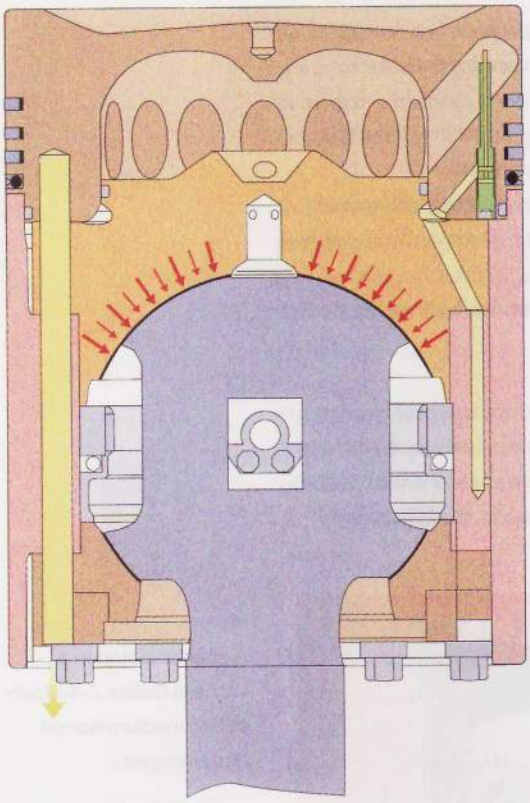
The piston crown can be machined in various ways to ensure maximum cooling with the lubricating oil.
- To force the lubricating oil to flow past the thermally heavily loaded parts.
- Manufacture the piston crown surface in an undulating wave pattern to expand the cooling surface.
- To apply bore cooling to the rims of the piston crown.

Alternatively, one could reduce the heat transmission from the combustion space to the piston by furnishing the piston crown with a heat insulating layer as is the case in the Wärtsilä 64 engine.



◄ A special piston in a Wärtsilä Sulzer Z-40 four-stroke medium-speed diesel engine.

The lubricating oil is pumped from the crankshaft via openings in the bearing caps, through the perforations in the connecting rod to the gudgeon pin and then to the piston-ring pack and the piston for cooling. The top of the connecting rod has a ratchet mechanism which during the up- and downward movement of the connecting rod allows the piston to rotate around the vertical axis. This improves the distribution of the heat flow through the piston, which provides a more evenly distributed temperature for the entire piston, preventing localised overheating. The rotating motion of the piston ensures that the cylinder liner is run in properly. The back and forth motion forces of the connecting rod are stored temporarily in a spring; the moment the gas and mass forces are balanced and the spheroid bearing is not under any load, the piston is turned.

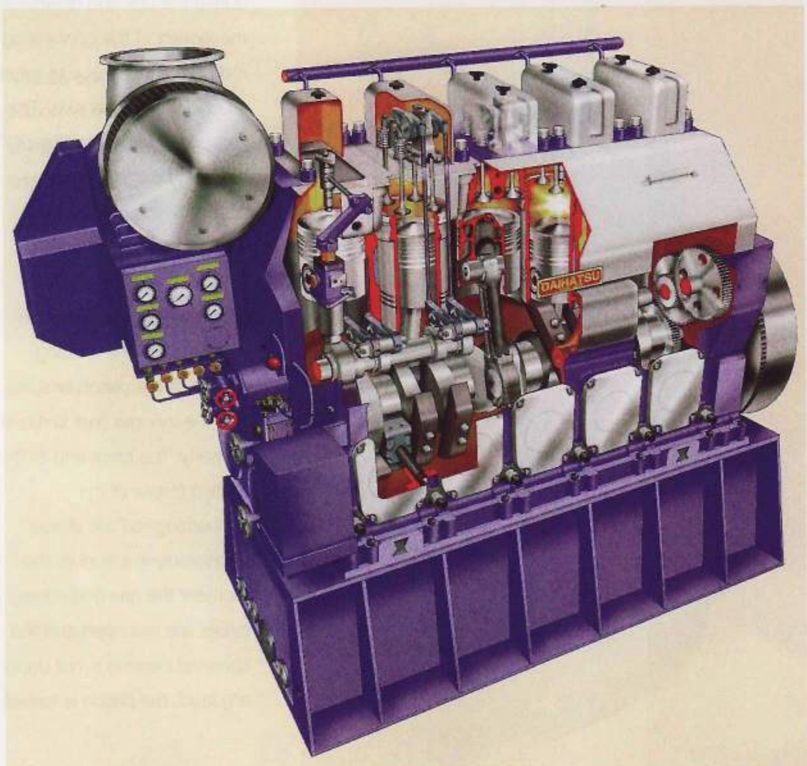


◄ A cross section of a piston in a Wärtsilä Sulzer Z-40 engine.

'Jet shaker'-cooling system

The reciprocating movement of the piston can be used to cool the piston lubricating oil. This is referred to as 'the cocktail shaker'-effect or the 'splash effect'.

▼ This Daihatsu-diesel engine series DK have one-piece cast iron pistons. They are cooled with lubricating oil.



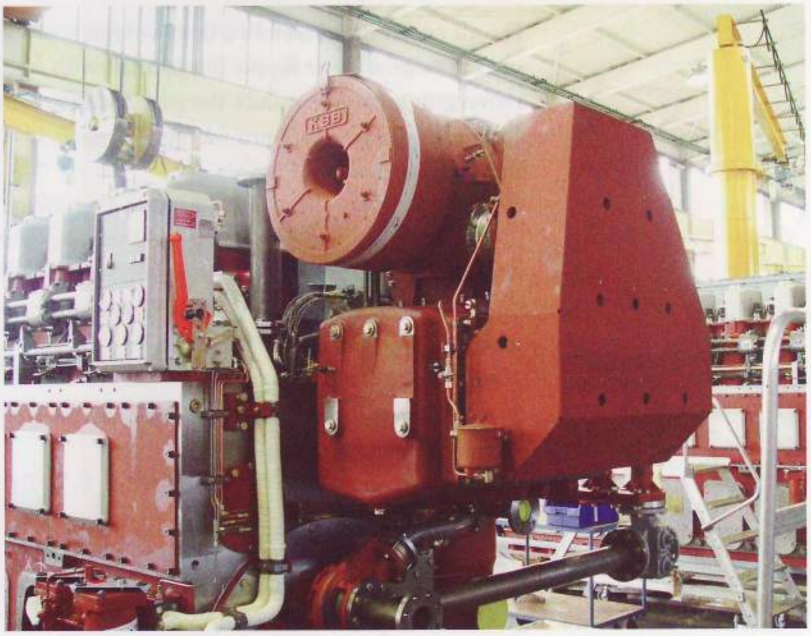
Some engine manufacturers still use one piece pistons manufactured from cast iron, such as in the Daihatsu DK-series with 20, 26, 28, 32 and 36 centimetre bores.

10.16.4 Engine category IV – 1500 to 100,000 kW, two-stroke crosshead engines, low-speed, H.F.O.

These engines have a deviating method in the supply and discharge system for piston cooling. Due to the crosshead construction it is possible to install the lubricating oil supply and -discharge in the crosshead and subsequently via the hollow piston rod to and from the piston.
Today all piston cooling is performed via the piston rod. In the supply and discharge of the crosshead, telescopic pipes and swivel pipes for the lubricating oil or fresh water cooling are used.
All modern type crosshead engines have piston cooling with lubricating oil. The main advantage is that, in case of leakages, there is no water contamination of the lubricating oil which may form an emulsion.
Disadvantages are the reduced heat absorption capacity and possibility of contamination of the metal surfaces with carbonised lubricating oil which occurs at high material temperatures.
Special attention is required to limit the amount of contamination in the piston.

▲ All the pistons of modern two-stroke crosshead engines are cooled with lubricating oil.

Extra attention should be paid to the contamination of the inner side of the piston. If the temperatures are too high the lubricating oil will carbonise and the cooling of the piston will deteriorate. This could cause the piston to overheat, crack and/or seize.

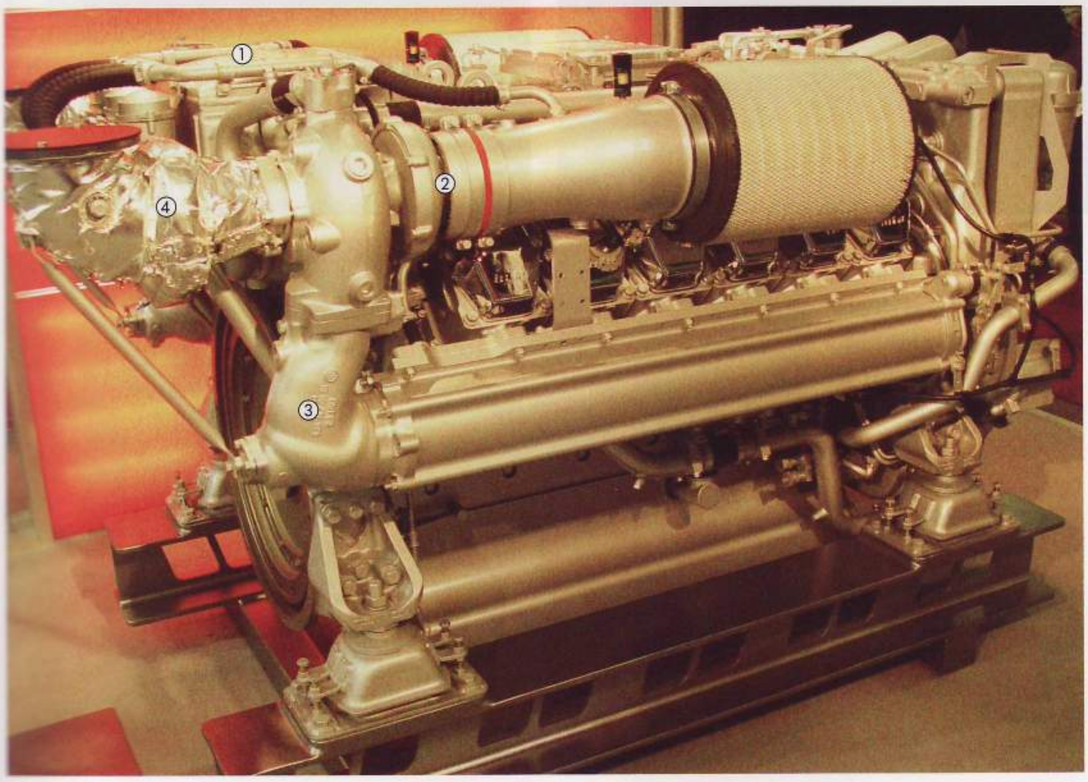


From the scavenging-air section of the turbo-blower the air channel widens towards the air cooler.

This produces a decrease in the air velocity which improves the air cooling as the time available for exposure of the cold cooling-water pipes to the air in the cooler is increased too. After the cooling section a tapered part, or converging section, increases the velocity as the air is forced towards the inlet valves or the inlet ports of the cylinders.

10.17 Combustion-air cooling

After the turbo-blower the air is compressed to a higher pressure and temperature. The compressor capacity generated by the exhaust gas turbine converts this capacity to a higher air pressure and a higher air temperature.
Most compressors operate with pressures of 2 to 5 bar (over pressure) and an air temperature of 100 to 160 °C. To generate ample power requires a good deal of fuel and also large amounts of air. When the air is cooled, the specific mass of the air increases which allows an increased amount of air to be injected into the cylinder or engine. Therefore enabling an additional fuel supply which results in an increase in the engine capacity. The air coolers after the turbo-blower usually are pipe coolers in which low temperature coolant flows, the so-called L.T.-cooling-water system. The air circulates around the pipes which are often provided with cooling fins to increase the cooling surface. Low air velocity is critical for optimum heat transmission.



A highly loaded high-speed four-stroke diesel engine in V-shape.

Both air coolers are placed between the cylinder rows.

1. *air cooler*
2. *air section turbo-blower*
3. *inlet-air manifold to the cylinders*
4. *exhaust-gas turbine*

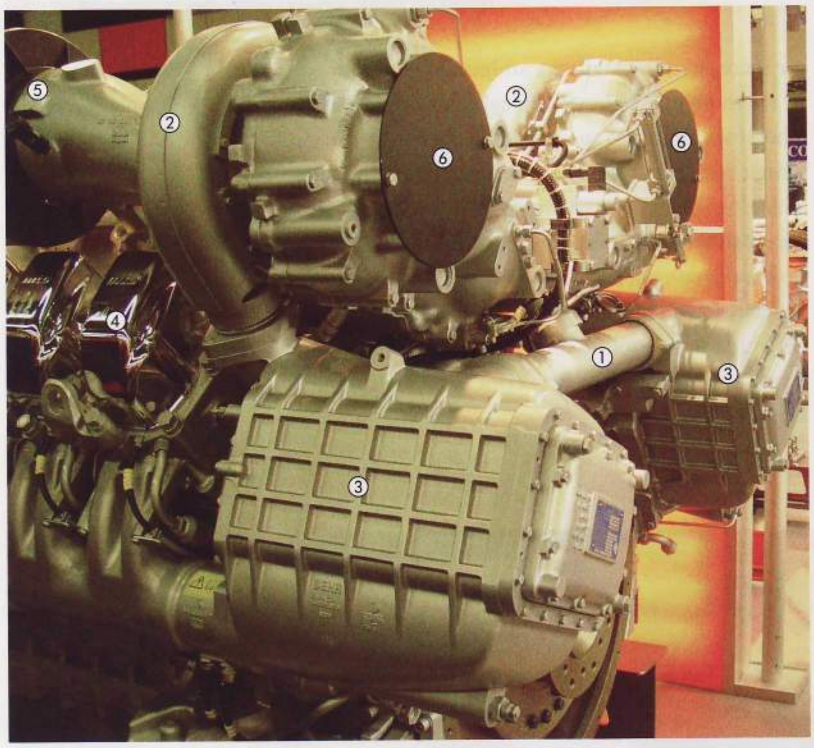
The air cooler has therefore a large volume and the turbo-blower supply line is fitted with a divergent section to reduce the air velocity in the cooler and therefore provide more time to intensify the cooling of the air.

▼

Supercharged air groups in a V-engine with both air coolers positioned below.

The air coolers are connected in order to avoid pressure differences between both banks.

1 *connection pipe*
2 *turbo-blower*
3 *air cooler*
4 *cylinder*
5 *air-inlet filter*
6 *exhaust flange*



10.17.1 Condensation

When air is cooled it can, dependent on vapour pressure, begin to condensate at a certain temperature.
When the water that is produced is carried to the cylinder with the air flow this can adversely affect the piston movement. This can also lead to an increase in the corrosive wear and tear of the piston rings and the cylinder. The water droplets interfere with the lubricating oil coating on the cylinder liner. This is why the water is drained and two-stroke engines have a partition installed in the air supply to the cylinder, the so-called **'mist catcher'**.
In humid tropical climates where the air has high humidity, large quantities of water are drained from large crosshead engines: sometimes **several thousands of litres per hour!**
Standard air temperatures after the cooler are 40 to 45 °C for two-stroke engines. The four-stroke engine air temperatures are similar or slightly higher.

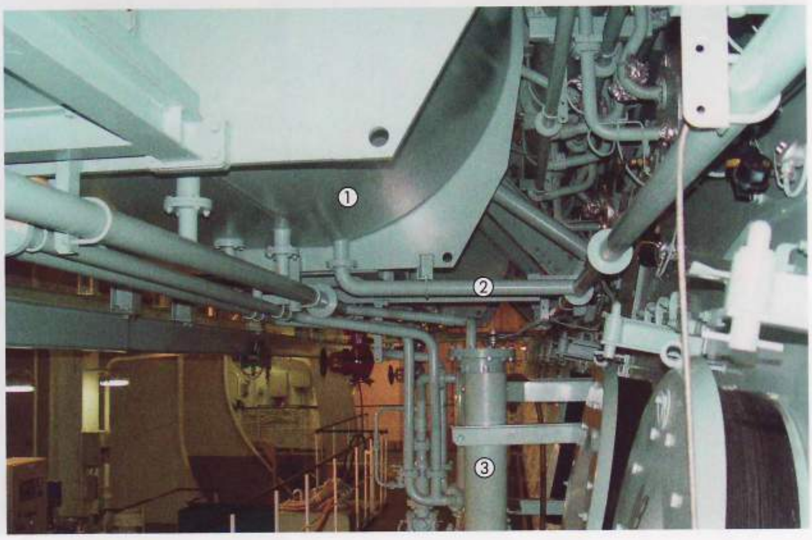
10.17.2 Bore cooling

This type of cooling has been applied for dozens of years in engine parts that are in the proximity of the combustion space, such as the cylinder liner, cylinder head and the piston. They are now often applied to large highly loaded four-stroke engines running on heavy fuel oil. Due to the increased mechanical load ensuing from combustion pressures that sometimes exceed 200 bar, the walls of the combustion space are becoming increasingly thicker. The piston crown and the cylinder head are steel; cast steel or forged steel. The cylinder liner is manufactured from cast iron in view of its advantageous running properties of, for instance, perlitic cast iron.

►

The scavenging-air condensate outlet in a large two-stroke crosshead engine.

1 *condensate sump*
2 *condensate discharge pipe*
3 *collection tank with high-water level alarm*



As opposed to the cast iron parts, it is not possible to install a cooling-water space in the mould of the steel piston crown and the cylinder head. The forged parts are today machined and through drilled cooling holes, the materials near the high process temperatures of the combustion space are cooled sufficiently without a decline in component strength.

Two-stroke crosshead engines

The cylinder liner designs of all three existing manufacturers have a thicker upper rim provided with bore cooling.

In Wärtsilä Sulzer RTA engines the other parts of the cylinder liner are cooling water free!
In MAN–B&W and Mitsubishi the middle section is cooled with water.
In all three engines the bottom section including the scavenging ports are cooled with scavenging air.

10.17.3 Turbo-blower cooling

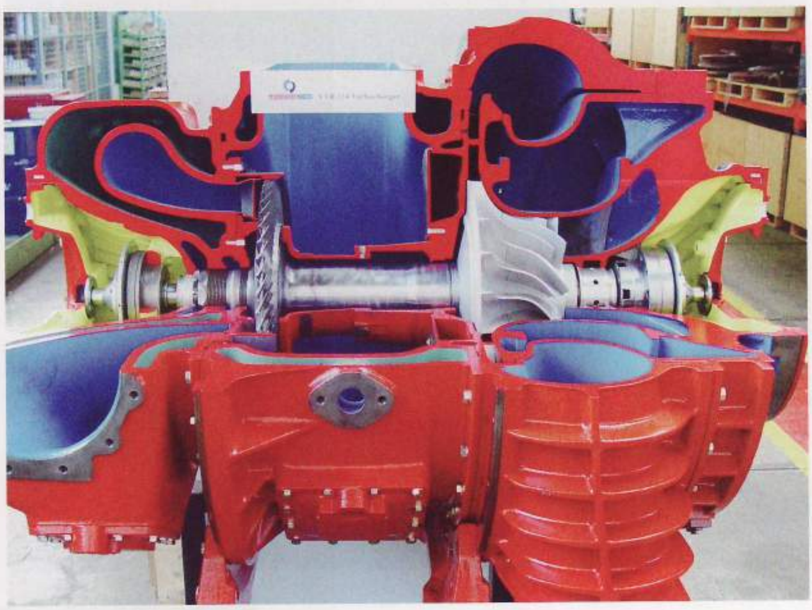
One speaks of uncooled and cooled turbo-blowers in order to indicate whether they have water cooling. In smaller turbo-blowers the thick robust bearing casing is cooled and lubricated with lubricating oil .

▼
The thickened upper collar of a cylinder liner in this two-stroke crosshead engine is provided with bore-cooling channels.

This is known as 'bore cooling'.
1 *thickened upper collar for 'bore-cooling'*

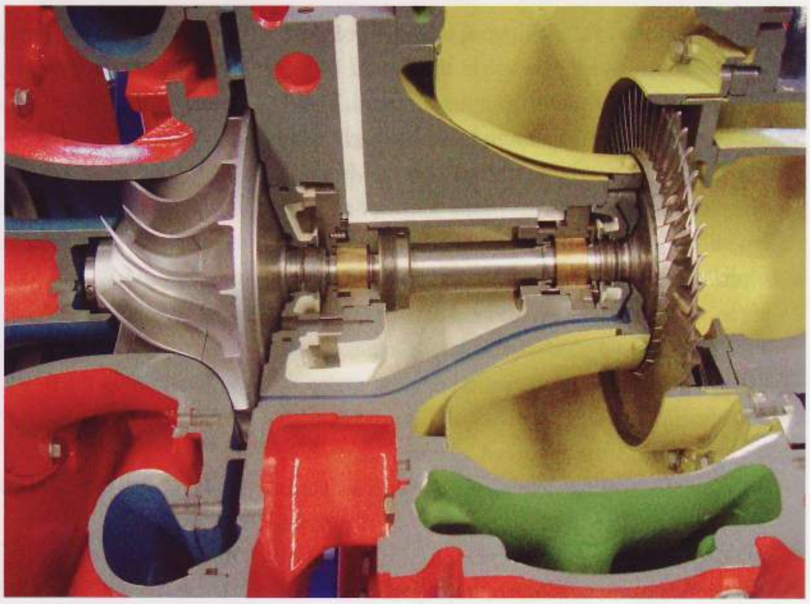
▼
An ABB-VTR turbo blower with water cooling.

Modern versions no longer have water cooling.
light yellow: lubricating-oil part
light blue: cooling-water part
dark blue: left - exhaust-gas part, right - scavenging-air part



◀
Drilled channels in a cylinder cover in a MAN–B&W two-stroke crosshead engine.

The large holes are for the cylinder-cover studs.



A turbo-blower with an uncooled casing. Manufacturer ABB.

yellow: exhaust gases
blue: scavenging air
white: lubricating and cooling oil for both journal bearings and the thrust bearing
blue canal: here; the scavenging air flows from the delivery side of the air part of the blower to the shaft seals of the exhaust-gas part. The scavenging-air pressure is higher than that of the exhaust gases so the dirty hot exhaust-gases do not flow in the vicinity of the right shaft bearing.

Small blowers often only have a combination of lubricating and cooling with lubricating oil. There are also turbo-blowers which have a water cooled casing. This ensures lower material temperatures. Furthermore, it reduces radiant heat emission into the environment. This is also fitted to exhaust gas ducts for the same reason.

Large blowers were traditionally cooled with fresh water and manufactured with double walls. However, today their construction resembles that of the smaller uncooled turbo-blowers and they are no longer cooled. This makes the turbine design simpler, cooling-water related maintenance such as leakages and cleaning of the water spaces is no longer required. The material of the casing is now subject to much higher temperatures and its quality has consequently been improved.

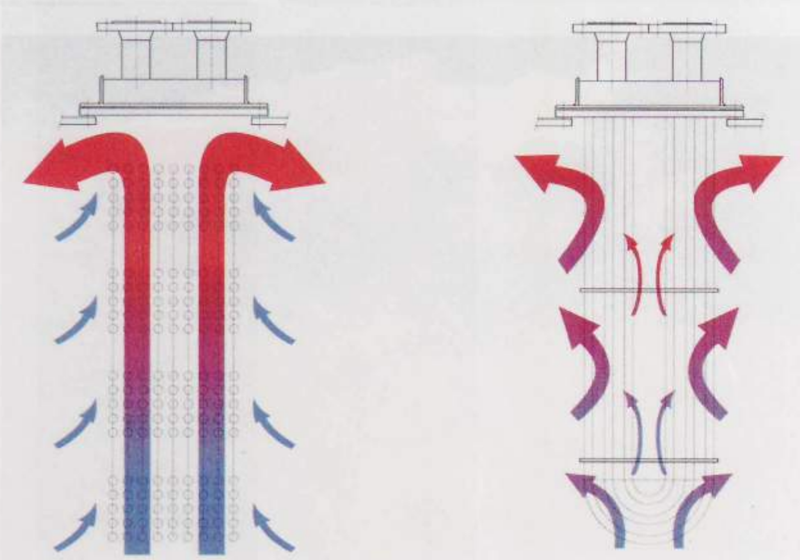
10.17.4 Cooling crosshead engine guides

In older engine types the hollow cast guides used to be cooled with relatively cold seawater. Today, guides are usually welded into the frame and cooled and lubricated with lubricating oil.

10.18 Special cooling systems

10.18.1 Box cooling

This is a system in which the heat exchanger of the closed fresh water cooling system and the raw- or salt water system has been integrated in a casing in the ship's hull along which the outboard water can freely flow during sailing. The box cooler can be mounted alongside the ship, but also in the bottom of the ship. The system is often applied in inland navigation, coastal navigation and for



Keel or box coolers have an open connection with water and therefore seawater circulating pumps are not needed.

◄

The top view of a box-cooler series.

The connections with the various cooling systems are always made on the top side and provided with cut-off valves. Note the vent cocks at the highest point of the piping on the left hand side.

1 *vent cocks*

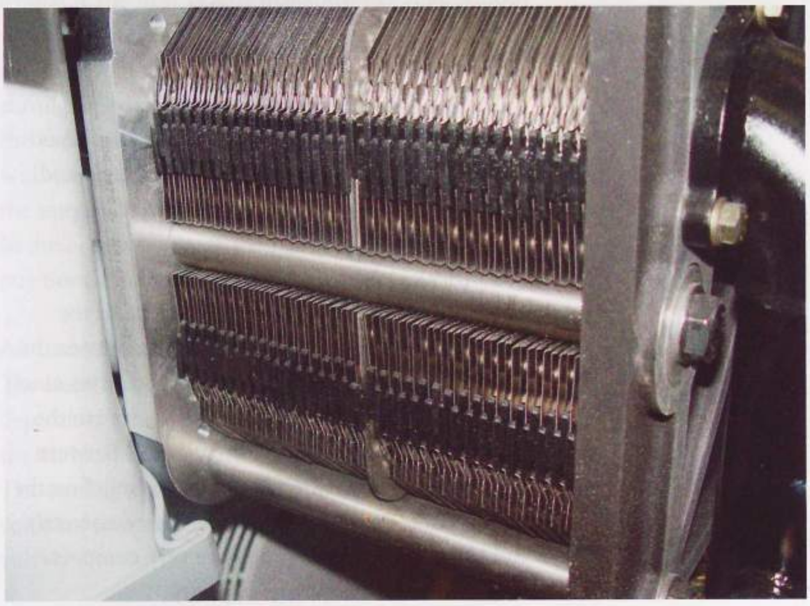
instance on dredgers. It has the advantage that any kind of dirt in the untreated cooling-water can not enter the pumps and pipes do not come into contact with dirt.

10.18.2 Bilge- or keel cooling

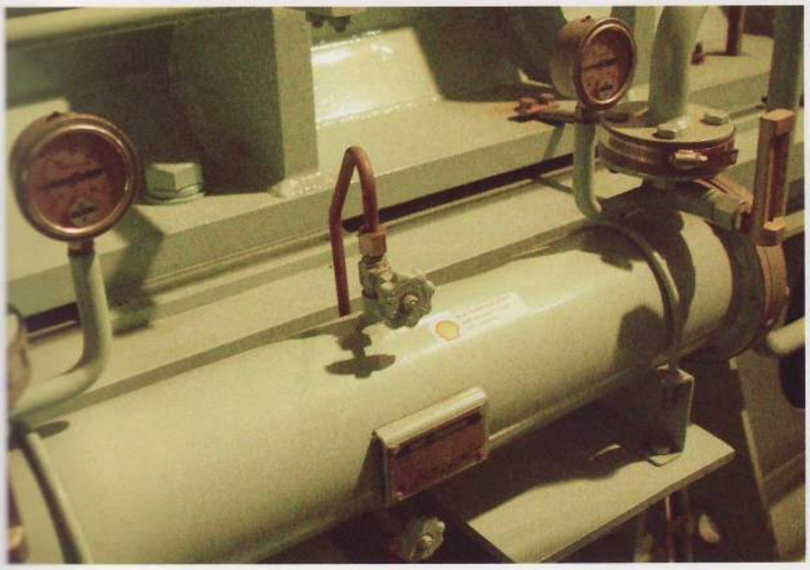
This is frequently used in cabin cruisers and other comparatively small propulsion systems.
The heat exchanger is often simple: a long cooling pipe is mounted below the surface often next to the keel. Sometimes several pipes are fitted.

▼

Plate coolers are getting used more frequently.



10.19 Pipe coolers and plate coolers



◄

A lubricating-oil pipe cooler in a four-stroke medium-speed diesel engine.

10.19.1 Pipe coolers

Principle: The heated water flows through pipes, and the cooling liquid or gas moves around those pipes.
By means of guide panelling the flow can be directed so that an optimum heat transmission can take place. Traditionally, the **pipe cooler** was often used for lubricating oil, fresh water cooling and raw water cooling. They are arranged in a flow- or counter-flow principle.

Advantages and disadvantages of pipe coolers

The pipes can be cleaned on the inside by removing the caps. This is often the raw water side. They can be easily cleaned with pipe brushes, compressed air or a high-pressure creeping hose. If the pipes need cleaning the entire assembly must be removed. Therefore, there always has to be ample room available and in modern engine rooms, there is insufficient space.

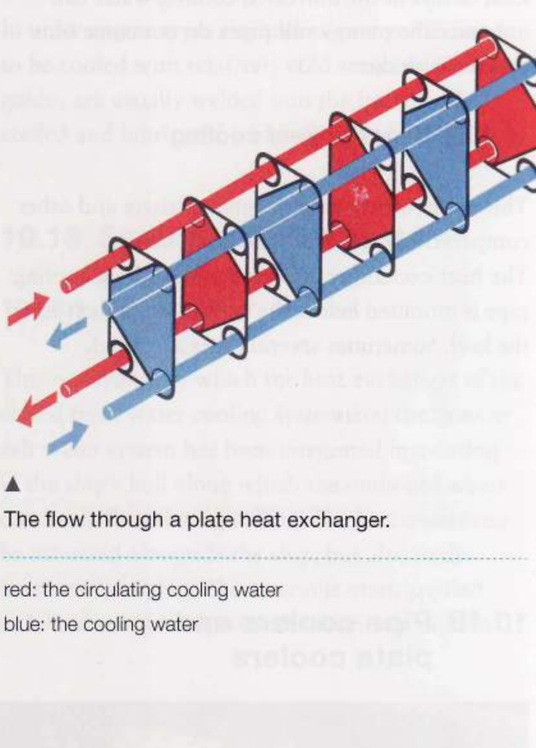
10.19.2 Plate coolers

The plate coolers are increasingly used for liquid cooling: fresh water cooling, lubricating-oil and fuels.

Principle

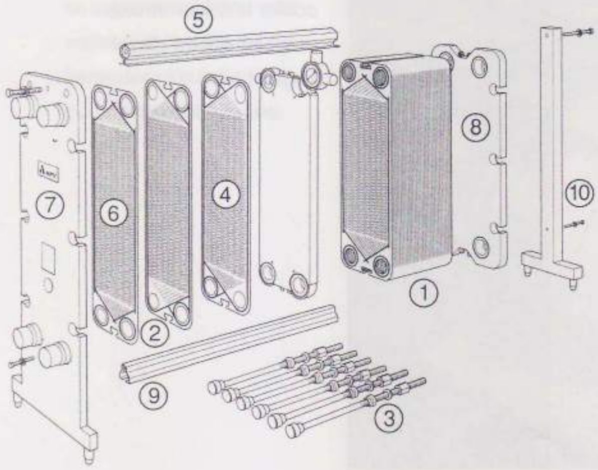
The cooling water, flows along one side of the sequentially arranged plates and flows over to the next plate via holes. The cooling liquid flows along the other side of the plates and flows over to the following plate through different holes. Between the plates, packings are fitted. Surface relief on the plates ensures that the plates are spaced correctly and can affect the flow. They are very compact and have a relatively large capacity.

To clean, the plates are simply slid off the pull rods. Each plate can then be cleaned individually. **Sealing with the packing is vulnerable.** However, cleaning the plates on location requires a lot of space. After mounting the clean plates there is a chance of leakage.
The capacity of the plate cooler can easily be modified by increasing or decreasing the number of plates. As a result of the compact construction of engine rooms there is an increase in the number of plate coolers used. In smaller engines – category I and part of II –they are seldom used, as the much smaller pipe coolers are applied here.



▲
The flow through a plate heat exchanger.

red: the circulating cooling water
blue: the cooling water



◄
The construction of a plate cooler.

1 *cooling plates packet*
2 *ribbed cooling plates with an inlet and outlet*
3 *draw bolt*
4 *gasket with gasket groove*
5 *upper beam*
6 *first plate*
7 *stationary end cover with inlet and outlet connections*
8 *back plate*
9 *lower beam*
10 *column*

Air coolers in a large diesel-power station in Honduras.

Approximately 25% of the heat supplied by the fuel is wasted into the environment.

10.20 Cooling systems in a diesel-power plant

Most diesel power-plants, particularly in tropical regions, do not use the exhaust gases. **Nevertheless, this constitutes over 50% of the heat content of the fuel!** In co-generation the total efficiency can amount to 80% and rise up to 90%. The remainder consists of flue gas losses.
Let's assume that the diesel genset efficiency is 45% then at least 25 to 35% of the heat from the fuel must be discharged. The remainder are exhaust gases. In a diesel power plant of 100 MW this is between 55 and 78 MW!
If there is river- or seawater available, the heat can simply be discharged with the normal and aforementioned cooling systems using a H.T.-, L.T.- and raw water system.

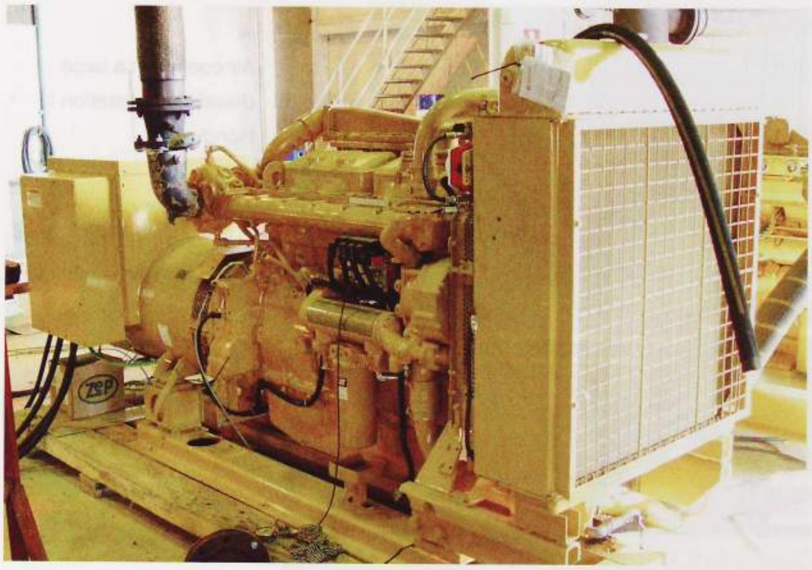
If there is no or insufficient cooling-water available the heat is discharged using a range of air coolers. These heat exchangers operate similarly to a radiator, a ventilator forces air through the heat exchanger. In view of the space the radiators are usually arranged vertically and with the ventilator placed at the opposite side to the output shaft of the diesel engine.
In diesel power plants the heat exchanger is positioned horizontally and blows air upwards.

Air cooler circuits

The number of heat exchangers in operation depends on the required power production and the ambient air temperature.
The cooling systems themselves are kept at a constant temperature by thermostatic three way valves.

A large diesel-power station in the tropics.

1 *medium voltage substation, 13,8 kV ,to high voltage, 110 kV*
2 *plant with large four-stroke medium--speed diesel engines operating on H.F.O., category III*
3 *exhaust-gas pipes with silencers and underneath, a building containing the auxiliaries for the diesel engines*
4 *diesel-engine heat exchangers that discharge their heat into the outside environment*
5 *drinking water storage tanks*



▲
Many diesel generators are provided with a radiator that in a simple manner discharges the engine heat into the outside environment.

These air coolers are also often used in large back up gensets. Smaller back up gensets are usually provided with a radiator behind the engine. Here the radiator is normally mounted on the outside wall of the building and is in direct contact with outside air. If the ambient air temperatures fall below freezing point it is important to add an anti freezing agent to the cooling-water.

10.21 Cogeneration systems

When the large amounts of heat released in engines are used for other purposes this is referred to as cogeneration.
Heat emitted by exhaust gases, lubricating oil, cooling-water and air cooling is suitable for this purpose.

The heat is applied to:
- room heating in residential areas, large buildings and companies;
- heating of swimming pools and factories;
- generation of electricity with an exhaust gas steam boiler and a steam turbine genset;
- heating auxiliary systems for engines such as fuel storage and cleaning, pre heating lubricating oil and cooling-water and the heating of fresh water evaporators.

It should be noted that although the total efficiency of the installation increases the total shaft power does not.

Comments
- **In small cabin cruisers the engine heat is sometimes used to heat up water for showers and cooling.**
- **In shipping distilled water** is made by means of a seawater evaporator which is heated with the H.T.-fresh water cooling system.
- **With an exhaust-gas boiler,** low pressure steam is formed which is used to heat the accommodation, the cargo and fuel. Exhaust gas boilers are also often used in diesel power plants.
- **In power plants** with engines, heat is used to heat buildings, for city heating, and also the process industry.

In countries where engines that run on fuels such as diesel oil or heavy oil are prohibited due to environmental regulations, gas engines are used.

Also see Chapter 29, New fuel developments.

◄
The use of cooling-water and exhaust-gas heat is applied on a large scale.

This is the case in countries with a temperate climate. City heating, production processes, swimming-pool heating and greenhouse/glasshouse cultivation projects are some of the applications. A problem with the use of diesel-engine exhaust gases is the high emission of harmful substances. Flue gas cleansing occurs, but gas engines are more often used as they produce less harmful emission. Shown here, a company in The Netherlands where paprikas are cultivated. The heat generated by the gas engines is used in this greenhouse complex. An emergency cooler is used in the case of very high external ambient air temperatures.

Diesel engine flue gases are sometimes also cleaned.

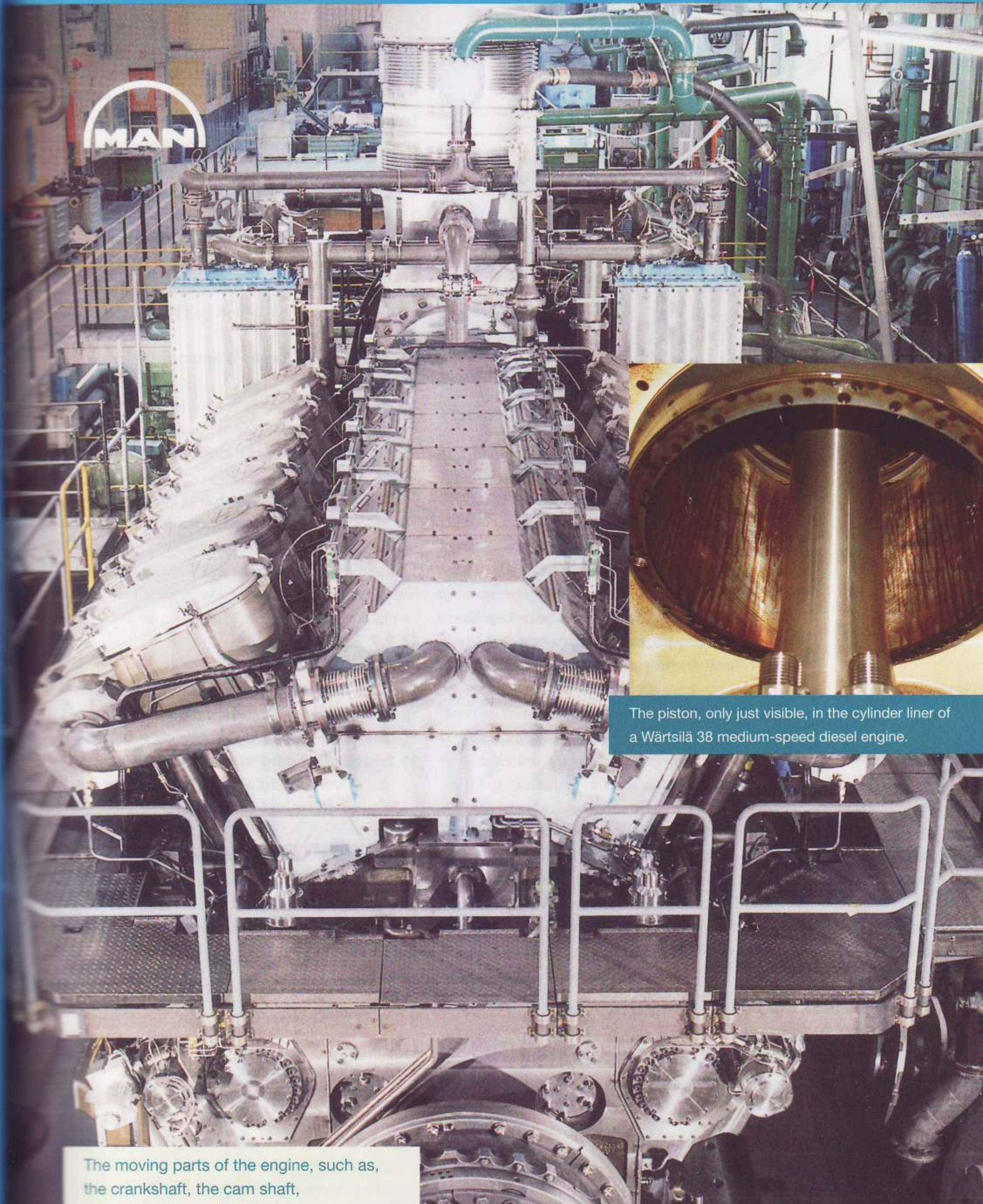
Also see Chapter 22, Engine emissions.

10.22 Summary cooling-water systems

- Only use pure water for cooling-water.
- Cooling-water treatment is imperative. Follow manufacturer's instructions!
- Cooling-water tests must take place regularly. Test procedure: see manufacturer.
- Deaeration of cooling-water is very important.
- In case of repairs, collect drained cooling-water and re-use it.
- Perform surface inspections during the repair of cooled engine parts.
- Monitor cooling-water pressures and temperatures. Fluctuations are indicative of malfunctioning pumps and control systems.
- Pre- and post heating of cooling-water must take place gradually to minimise heat tension in the engine material.

>CH 11

Lubrication of engines

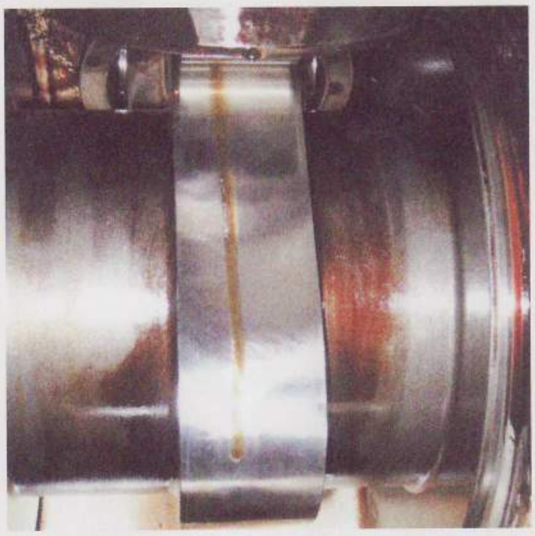


The piston, only just visible, in the cylinder liner of a Wärtsilä 38 medium-speed diesel engine.

The moving parts of the engine, such as, the crankshaft, the cam shaft, the connecting rods, the pistons, the valves, the cog wheels as well as the elements that are in contact with these parts, such as cylinder liners and bearings, require lubrication and constant cooling. Adequate lubrication and proper cooling are crucial to keep maintenance costs as low as possible. Here a MAN B&W 12 cylinder four-stroke diesel engine in V- alignment.

11.1 Introduction

Lubricating engines is imperative.



▶ Almost all the moving parts in a diesel engine require lubrication, such as the cam of a fuel pump shown here.

11.2 The purpose of lubrication

11.2.1 To reduce the friction between the moving parts and consequently limit frictional wear and tear.

Obviously the friction in the engines must be minimal because friction causes wear and tear. Parts that are subject to friction are, amongst others: pistons, piston rings, cylinder liners, bearings, shafts, valve stems, tooth wheels and fuel-pump plungers. They have an operating life ranging from10,000 to 60,000 hours.

11.2.2 Discharge of frictional heat

The amount of frictional heat is minimal.

▼ Various engine parts must be lubricated. From left to right: a section of a cam shaft, a connecting rod, a cog wheel, a cylinder liner, valves, gudgeon pin bushing and a piston.

11.2.3 Cooling parts

Lubricating-oil also serves to cool parts. Here the amount of heat discharged is considerable. For example, piston cooling.

11.2.4 Protecting surfaces from corrosion

Many engine parts do not receive any special anti-corrosive treatment. Lubricating-oil is very suitable for the protection of surfaces.

11.2.5 Lubrication as a sealant

Sealing the piston, piston rings and cylinder liner, the stem valves and, for instance the piston- rod stuffing box in two-stroke crosshead engines.

11.2.6 Flushing and keeping various damaging substances in solution

This could be: debris from wear, water, dirt, acids, corrosive particles.

11.2.7 Sound damping

Liquids have a sound damping effect; this also applies to lubricating-oil.

11.2.8 Neutralising acids to avoid severe damage to engine parts by chemical corrosion

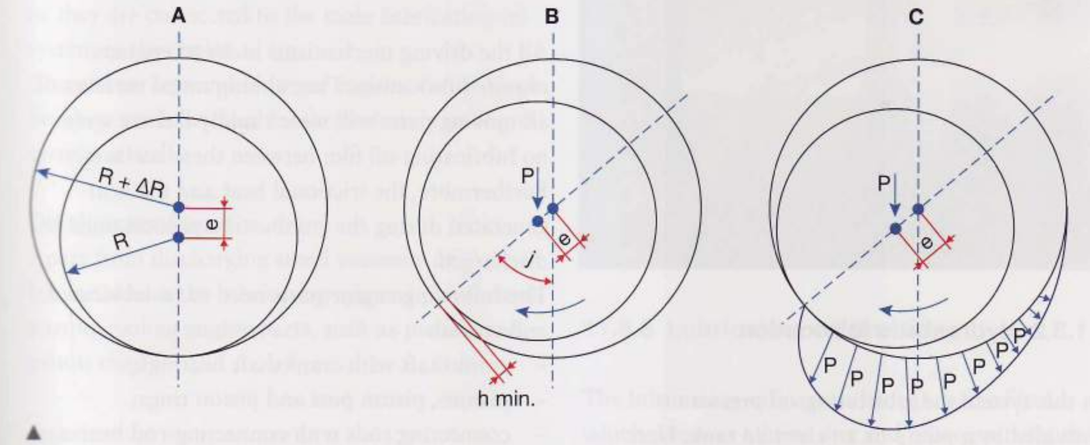
Here one should think of sulphuric acid found in fuel. The sulphur content of heavy fuel is particularly high (1 to 4%). Lubricating-oils have very specific properties and must meet strict requirements.

11.3 Three types of lubrication

11.3.1 Hydrodynamic or fluid film lubrication

This is obtained when two meeting surfaces are completely separated from each other by a thin layer of lubricating-oil, the so-called lubricating film. This film is several hundred microns to several tens of microns thick. **The lubricating-oil pressure in, for instance, the crankshaft bearing carries the shaft and is itself generated by the rotating shaft.** The pressure in the bearing can amount to 1000 bar in some places! The lubricating-oil pressure of the lubricating systems provides sufficient supply to these places. Most rotating shafts work with this principle; these include the crankshaft and the camshaft.

Pressure build-up of a rotating shaft in a plain bearing



P = lubricating oil pressure in plain bearing

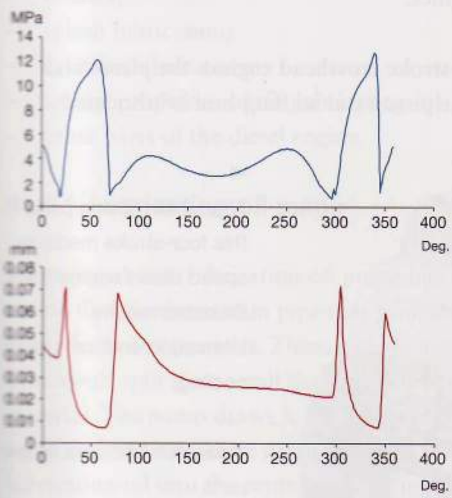
▲

The hydrodynamic lubrication of a crank shaft bearing.

Left figure: The stationary shaft is supplied with lubricating oil from the main lubricating oil pump at a pressure of 2 to 4 bars.

Centre figure: The crank shaft rotates at a regular speed. There is some clearance between the shaft and the bearing which prevents the metal parts from coming into contact. The minimum clearance is several tenths of a millimetre.

Right figure: The lubricating oil pressure build-up in the bearing. The rotating shaft carries the lubricating oil. Due to the wedge-shape, the lubricating oil pressure increases. This lubricating oil pressure when distributed over part of the bearing surface must provide the force exerted by the combustion process from above the piston downwards. This **pressure can vary from 100 to 1500 bars!** Therefore, the bearing material must be adequately resistant to these high pressures.



◀

The lubricating oil pressure in the bearing and the minimum oil film thickness.

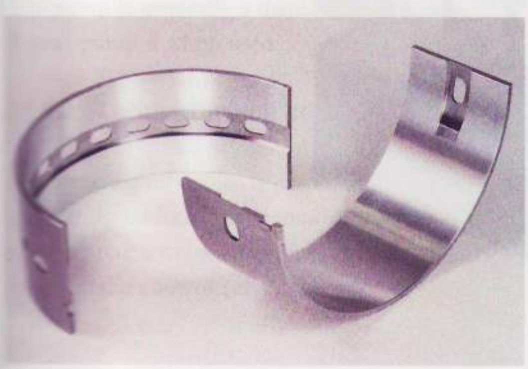
Top figure

horizontal axis: the position of the crank shaft in a four-stroke engine

vertical axis: the lubricating oil pressure in the bearing in MPa. The maximum is approximately 13 MPa or 130 bars

Bottom figure

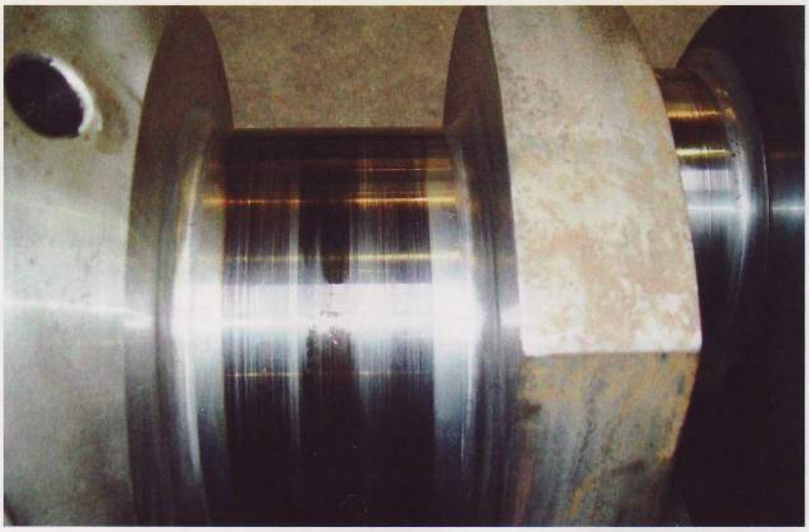
vertical axis: the minimum oil film thickness in hundredths of a millimetre. Minimum film thickness is approximately 0.01 mm, maximum thickness 0.07 mm



◀

Main bearings.

The clearance between the crank pins and the main bearing caps is so small that even the smallest dirt particles (several tenths of a millimetre) can cause severe damage to the bearing surface. This usually results in a damaged crank pin.



▲
A damaged crank pin.

Insufficient lubrication often causes such damage.

11.3.2 Hydrostatic lubrication

In this system the lubricating-oil pressure is provided by a pump or an elevated tank. Here both parts are also separated by a continuous lubricating film. It is often applied in crossheads of two-stroke engines. The pressure can vary from 20 to 30 bar before the crosshead to over a 100 bar to **'lift and support'** a steam turbine's heavy shafts from their bearings during start-up.

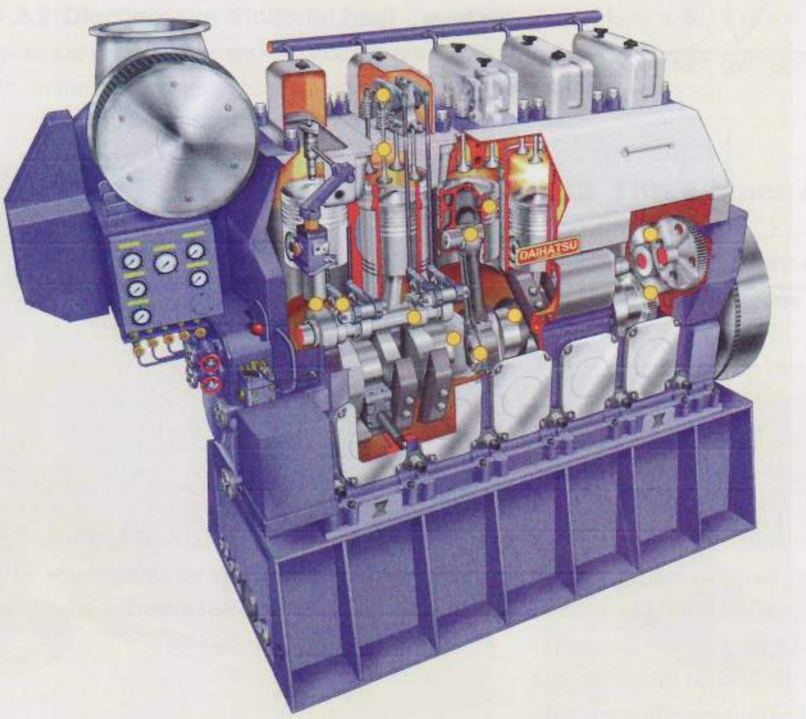
11.3.3 Boundary lubrication

In boundary lubrication, there is direct contact between both surfaces as is the case with a bearing and a rotating shaft. If boundary lubrication cannot be avoided, in spite of taking all possible measures, great care should be taken to reduce the surface roughness of both the shaft and the bearing. Chrome plating, grinding and polishing are common machining operations.

11.4 Engine parts that require lubrication and cooling

All the driving mechanisms in diesel engines require lubrication. The rubbing metal surfaces of all moving parts will wear rapidly if there were no lubricating-oil film between these surfaces. Furthermore, the frictional heat and the heat generated during the combustion process must be discharged.
The following engine parts need to be lubricated and cooled:
- crankshaft with crankshaft bearings;
- pistons, piston pins and piston rings;
- connecting rods with connecting-rod bearings;
- camshaft with camshaft bearings;
- cams, guide pulleys, push rods, rockers, valve springs and valve stems;
- parts of the fuel pumps;
- cylinder liners via the pistons;
- gear transmissions between the crankshaft and the cam shaft;
- additional shafts such as balance shafts or separate cam shafts with bearings for fuel pumps as found in MAN–B&W four-stroke engines.

In **two-stroke crosshead engines** the piston rod with the piston rod stuffing-box is lubricated



◄
Almost all moving parts in this four-stroke medium-speed diesel engine from Daihatsu require lubrication and/ or cooling.

Yellow dot: lubricated engine parts.

together with the crosshead including the bearings, guides and guide blocks. There is separate lubrication for the cylinder liner.
All engine governors have independent lubricating systems.
Turbo blowers' bearings and shafts are either equipped with their own lubricating-oil system or they are connected to the main lubricating-oil system in the engine.
Thrust bearings and in larger engines thrust block bearings are lubricated by the main lubricating-oil system as well.

Discharge of heat

Apart from discharging small amounts of frictional heat the lubricating systems plays a vital part in cooling engine parts, such as pistons and piston rings.

11.5 Common lubricating-oil system

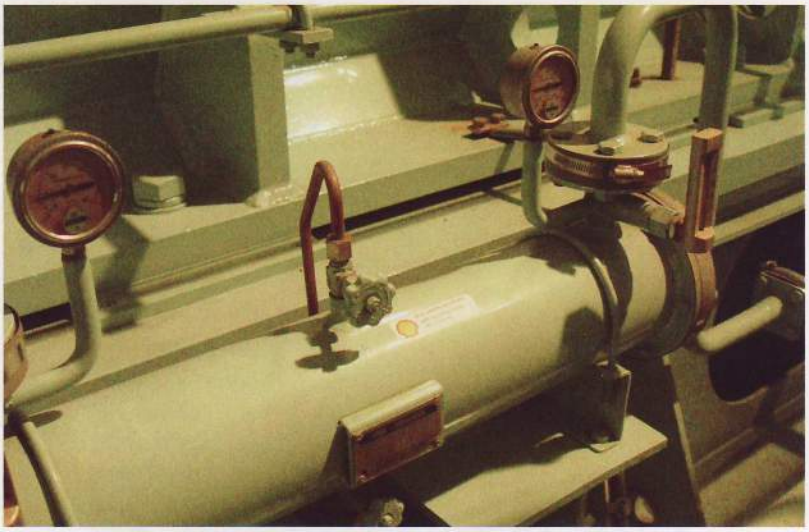
A common lubrication-oil system usually consists of the following components:
- lubricating-oil pump;
- lubricating oil coolers;
- lubricating oil fine filter;
- lubricating-oil pipes;
- drilled-through holes;
- nozzles;
- splash lubrication;
- cylinder lubrication;
- special properties of the lubricating-oil
- other parts of the diesel engine.

11.5.1 Lubricating-oil pump

An engine-driven lubrication-oil pump has a coarse filter in the suction pipe that protects the pump from dirt particles. These include metal, broken-off split pins, small nuts and packing material. The pump draws lubricating-oil from the sump or from a separate drain tank and forces the lubricating-oil into the central lubricating-oil line. The pump is generally a displacement pump. A gear pump is often used.

11.5.2 Lubricating-oil cooler

A lubricating oil cooler is placed in the main lubricating-oil line, this maintains the lubricating oil at a predetermined temperature using a thermostatic control valve.



▲
A lubricating oil cooler positioned next to a diesel engine.

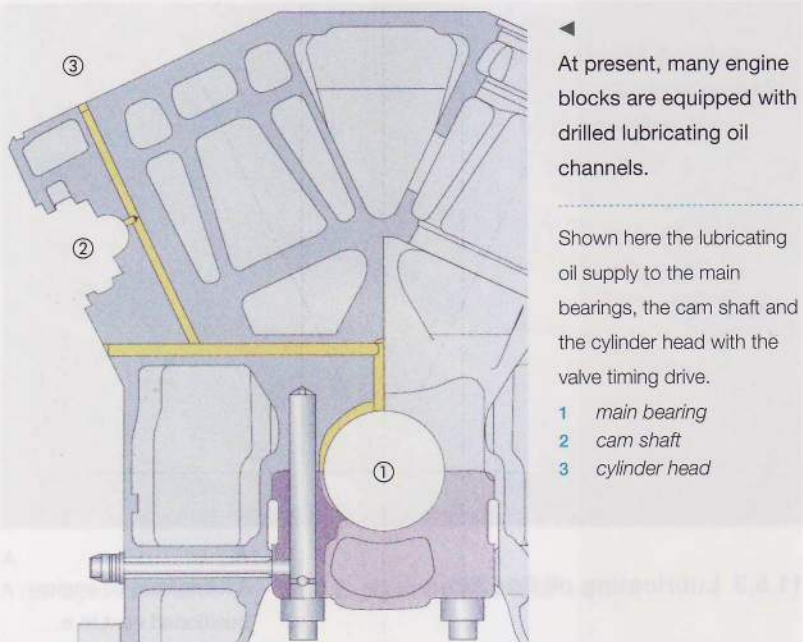
11.5.3 Lubricating oil fine filter

The lubricating-oil passes through a fine filter which removes all fine dirt and other particles that could cause wear from the lubricating-oil. There are two filters in the casing, the so-called duplex filter.
In larger diesel engines self-cleaning fine filters make releasing, cleaning and replacing of the filter cartridges redundant.
Many types of fine filters are fitted with pressure difference gauges which signal contamination of the filters and allow the necessary precautions to be taken.

11.5.4 Lubricating-oil lines

Lubrication-oil lines are installed throughout the engine. Today many lubricating-oil ducts are often drilled in the engine block. The lubricating-oil is delivered to the crankshaft bearings, cam shaft bearings and tooth wheels.

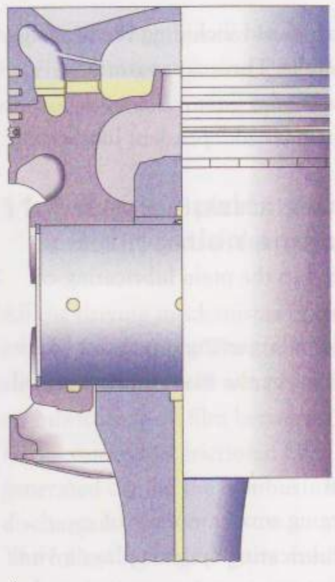
▼
Lubricating oil system automatic fine filters of a large two-stroke-crosshead engine.



At present, many engine blocks are equipped with drilled lubricating oil channels.

Shown here the lubricating oil supply to the main bearings, the cam shaft and the cylinder head with the valve timing drive.

1 *main bearing*
2 *cam shaft*
3 *cylinder head*

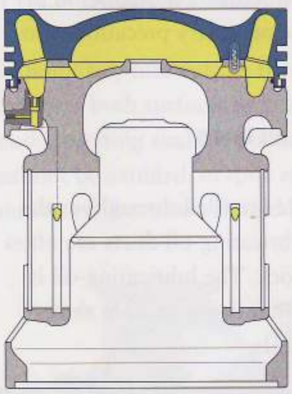


This piston is cooled using the 'cocktail shaker' effect.

The piston is partially filled with lubricating oil. The piston crown is intensively cooled by the reciprocating piston motion.

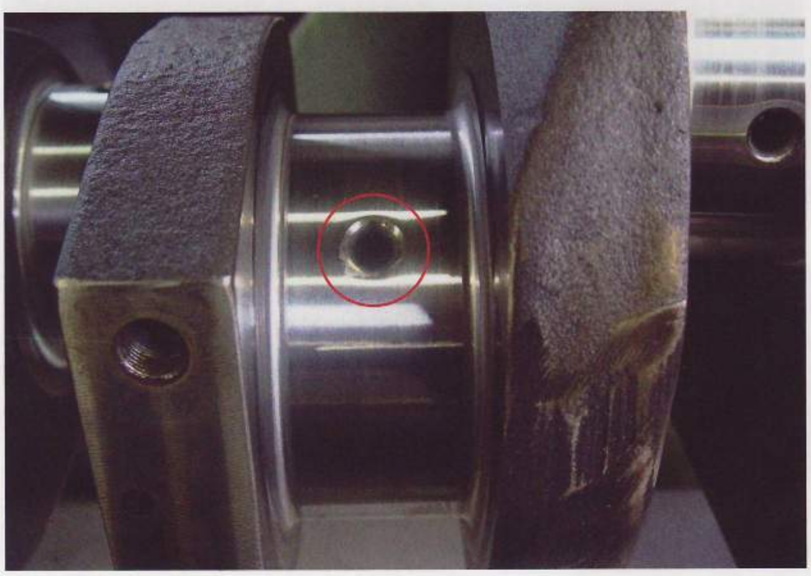
11.5.5 Drilled-through holes

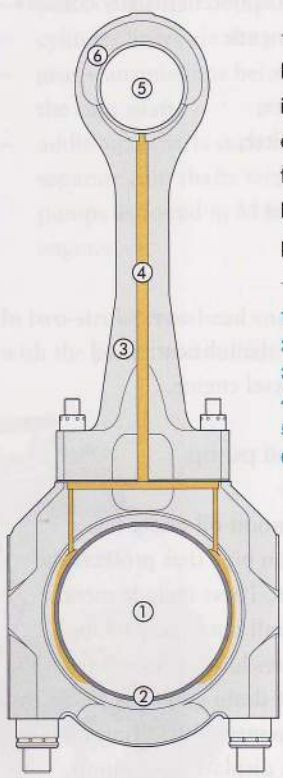
Drilled-through holes in the crankshaft, the connecting rod and the piston pin allow the lubricating-oil to be delivered to the crank pin bearings, connecting rod bearings, pistons and piston rings.



Cooling and lubricating a piston.

The entire piston crown is filled with circulating lubricating oil. The piston ring pack and the gudgeon pen are lubricated under pressure.



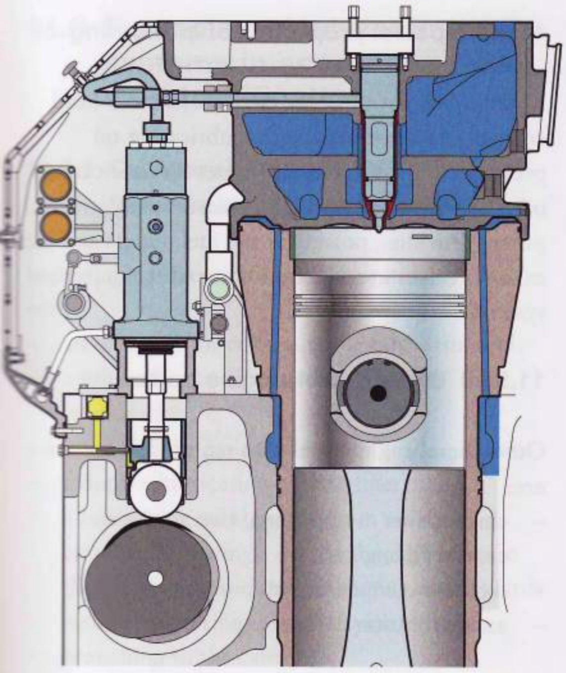


In larger engines a drilling in the connecting rod enables the lubricating oil flow from the crank pin bearing to the gudgeon pin and the piston.

1 *crank pin*
2 *crank pin bearing*
3 *connecting rod*
4 *drilling*
5 *gudgeon pin*
6 *connecting rod eye*

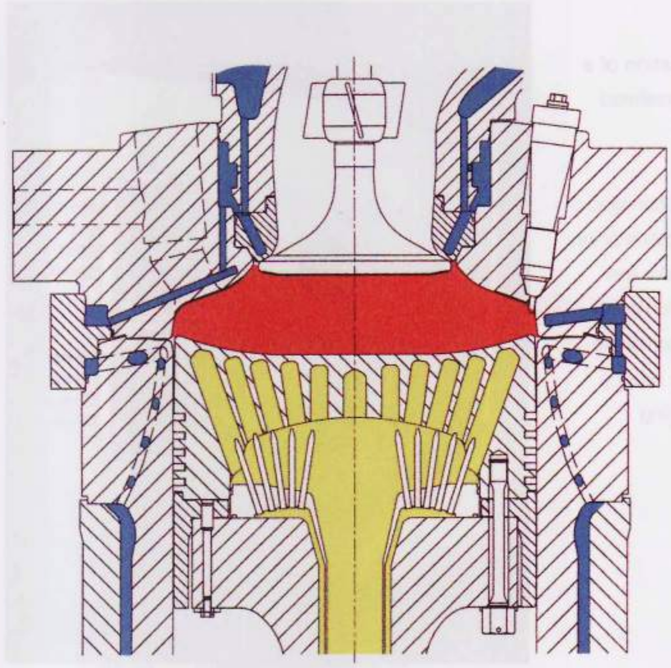
Drillings in the de crank shaft take the lubricating oil from the crank shaft to the crank pin.

The crank shaft bearing is supplied with lubricating oil from the lubricating oil supply line. The lubricating oil flows to the crank pin bearing via the crank shaft drillings and subsequently to the gudgeon pin and the piston rings via the rod drillings.



▲
Lubrication of the high-pressure fuel pump drive.

It is important that any fuel from the fuel pump drive does not spill into the lubricating oil. Lubricating oil mixed with fuel does not adequately lubricate!

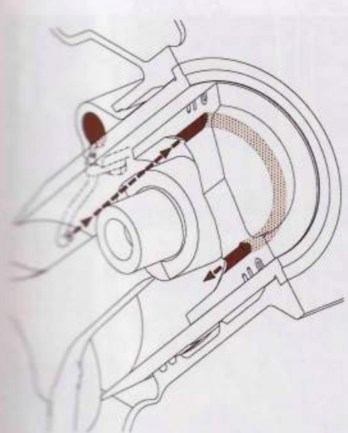


▲
Piston cooling using lubricating oil in two-stroke-crosshead engines from Wärtsilä Sulzer; type RTA.

The nozzles directed at the matching drilled cooling channels in the forged iron piston crown. This allows for each hot part to be intensively cooled.

11.5.6 Nozzles

Especially in small engines, lubricating-oil is sprayed on the cog wheels and pistons by **nozzles.** Sometimes pistons of large two-stroke crosshead engines are cooled in this manner as well.



▲
Spray jet cooling of the pistons in a four-stroke high-speed MTU-diesel engine.

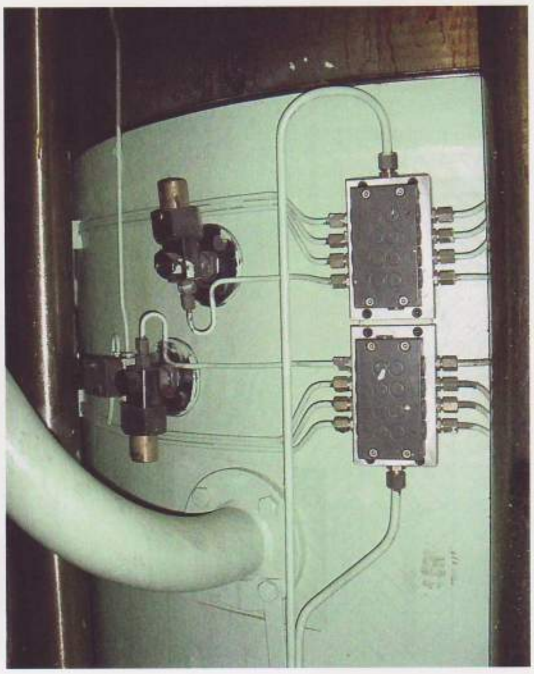
It is important that the jet is correctly directed. After maintenance, it is imperative to check that there is no deviation through incorrect mounting or damage.

11.5.7 Splash lubrication

Obviously, **splash lubrication** is mostly applied in small diesel engines. Moreover, in every engine the sump and the space directly below the piston are filled with swiftly moving oil droplets in a fine mist. It is therefore ill-advised to unscrew and open up the crankcase covers/doors while the engine is running!

11.5.8 Cylinder lubrication

In some large medium-speed four-stroke engines and in all two-stroke crosshead engines the cylinder liners and consequently the pistons and piston rings are lubricated separately.
To achieve this, cylinder liners with a number of lubricating points are found around the circumference of the cylinders. By means of small plunger pumps the lubricating-oil is forced through the drilled holes onto the cylinder liner at the correct time. The plunger pump capacity can be modified by mechanically adjusting the engine gearing. At present, it is possible to electronically alter the RPM of the electro motor that is commonly used.



Cylinder lubrication of a two-stroke-crosshead engine.

right: the distribution blocks for various lubrication points around the cylinder liner
left: two lubrication points which lead horizontally to the cylinder liner bearing surface via a drilling

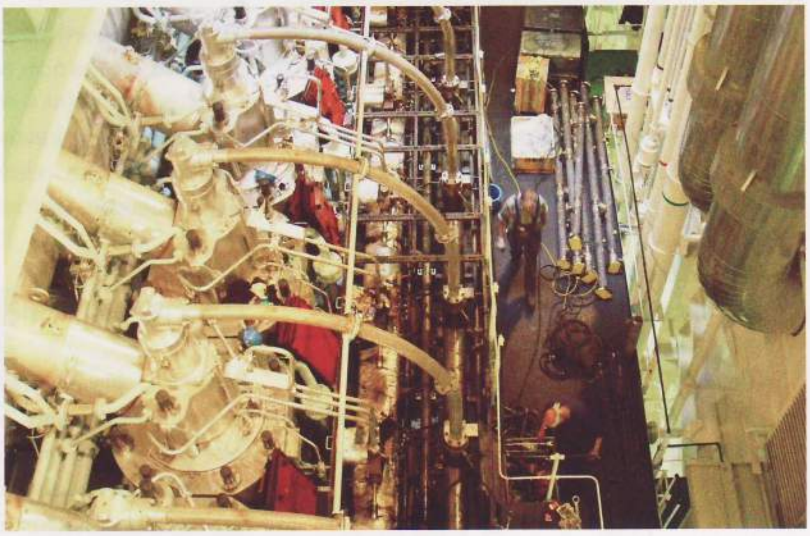
11.5.9 Special properties of lubricating-oil

Reverse gear for the large two-stroke cross head engines often operates on the lubricating-oil pressure. In the same engine types the lubricating-oil is also used for the high-pressure lubricating-oil pump, providing power for the fuel pumps and the exhaust valves in some common-rail fuel injection systems.

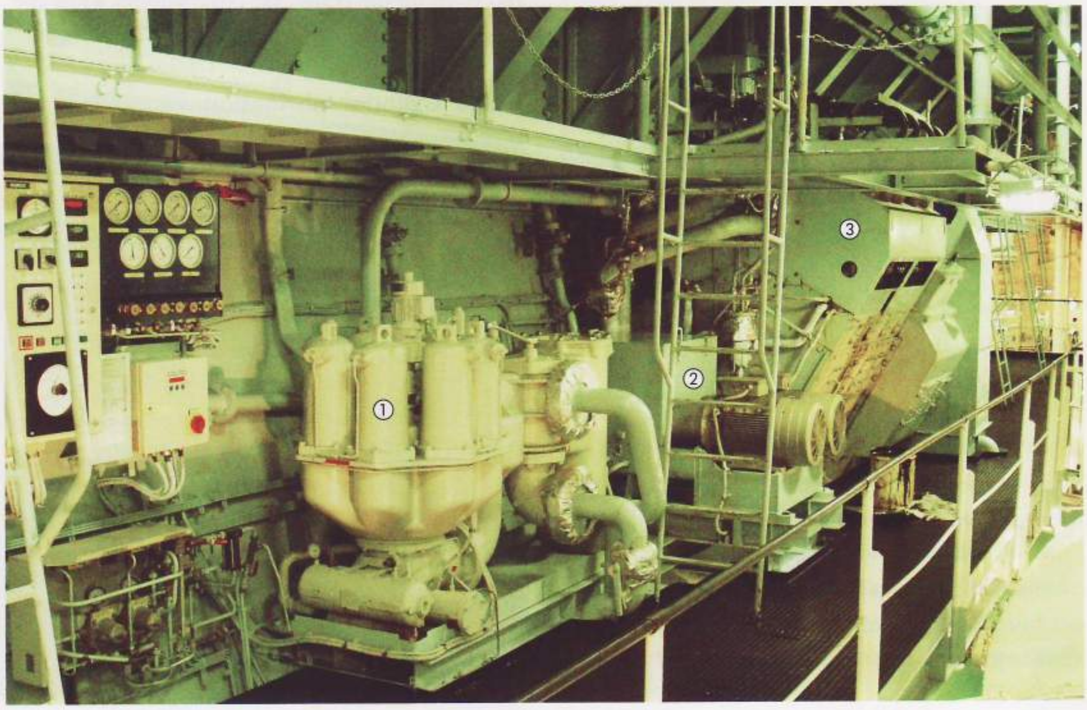
11.5.10 Other diesel-engine parts

Other diesel engine parts that require lubrication are:

- chain drives in traditional two stroke-crosshead engines;
- moment compensators;
- axial vibration dampers.



View from above of a Wärtsilä Sulzer two-stroke-crosshead engine, type RT Flex with hydraulically driven exhaust valves and hydraulically operated fuel supply systems to the fuel injectors.



An automatic fine lubricating oil filter with the lubricating oil pumps for the high-pressure lubricating oil system placed behind the filter.

This 200 bar hydraulic pressure system operates the exhaust valves and controls the fuel supply to a two-stroke-crosshead engine from Wärtsilä Sulzer; type RT Flex.

1. *automatic fine lubricating oil filter*
2. *high-pressure lubricating oil pumps*
3. *high pressure fuel pumps*

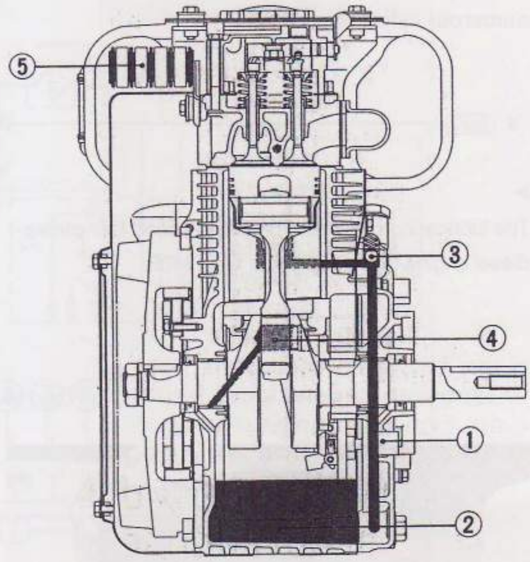
11.6 Examples of lubricating-oil systems in accordance with the classification

11.6.1 Category I, 0 to 100 kW, four-stroke, high-speed, M.D.O.

In principle these are simple lubricating systems with:

- an oil sump for collecting the lubricating-oil;
- a lubricating- oil pump driven by the crankshaft;
- a simple lubricating-oil cooler;
- a duplex lubricating-oil fine-filter;
- a lubricating-oil supply system via the crankshaft bearings, drilled-through holes in the crankshaft and the connecting rod towards piston/piston rings and the cylinder liner;
- branching to the cam shaft.

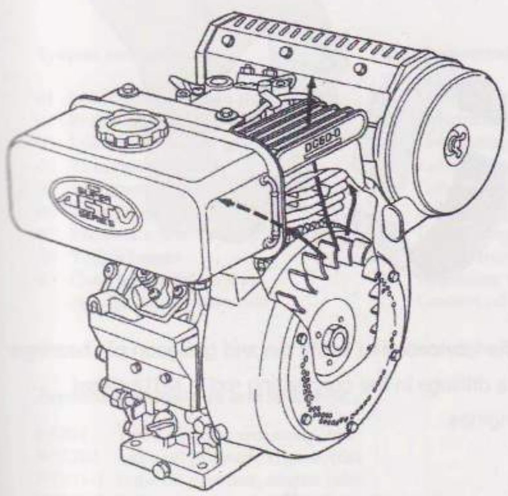
There is spray cooling as well as splash cooling of the piston.



▲

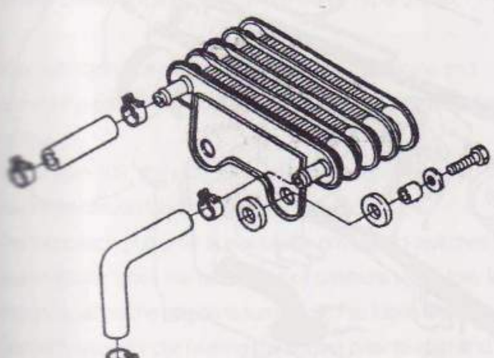
The lubricating oil system in a diesel engine from Kubota – model OC 60 – I. This is a small high-speed four-stroke single cylinder diesel engine with a power output of 4.2 kW.

The lubricating oil is drawn via a coarse filter (2) from the sump by the elevated lubricating oil pump placed behind the pressure control valve (3). The lubricating oil pressure is 1,5 to 5,0 bars, dependant on the engine speed. The lubricating oil lubricates all the moving parts such as the crank shaft and main bearings, via drillings in the crank shaft, the crank pin bearings including a piston using a spray nozzle. The cylinder liner is cooled with lubricating oil and the remainder of the engine block is air cooled. Via a small interconnect pipe (not visible) the valve mechanism fitted on the cylinder head is lubricated. The lubricating oil passes through an oil cooler which in turn is cooled by the air flow generated by a fly wheel placed on top of the diesel engine. The cooled lubricating oil is returned to the sump via a drilling.
The suction filter mesh size in the sump is 50 microns to avoid blockages. Three magnets are fitted to the filter to retrieve metal particles in the oil flow. So, this small diesel engine does not have a fine filter in the pressure pipe of the lubricating oil pump.



▲

The air cooling of the lubricating oil cooler using a fly wheel in a small Kubota-diesel engine.

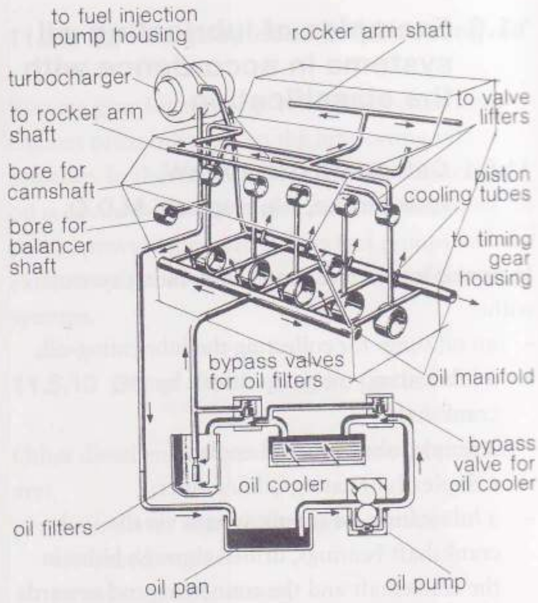


▲

A simple lubricating oil cooler in a Kubota-diesel engine.

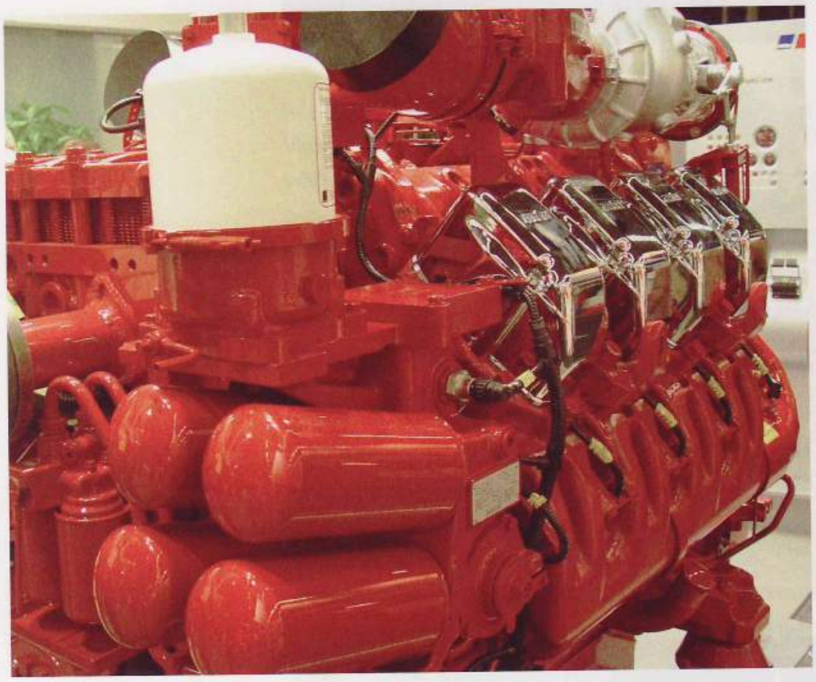
11.6.2 Category II, 100 to 5000 kW, four-stroke, high-speed, M.D.O.

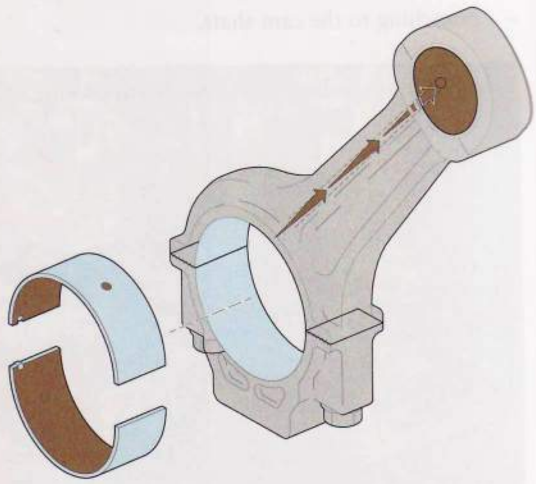
In principle this system is very similar to that of category I. It is often found in V-engines with numerous cylinders.



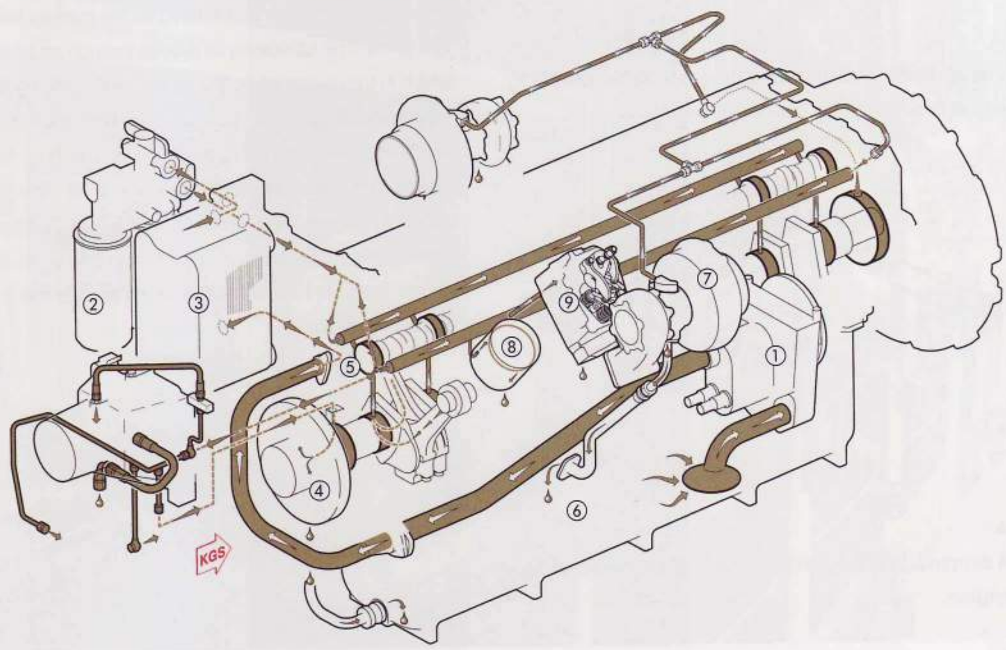
▶ The lubricating oil system of a high-speed four-stroke-diesel engine from Caterpillar, type 3408.

▼ Four fine lubricating oil filters and above a centrifugal lubricating oil filter in a four stroke high-speed MTU-diesel engine.





▲ The lubrication of crank pin and gudgeon pin bearings via drillings in the connecting rod in MTU-diesel engines.

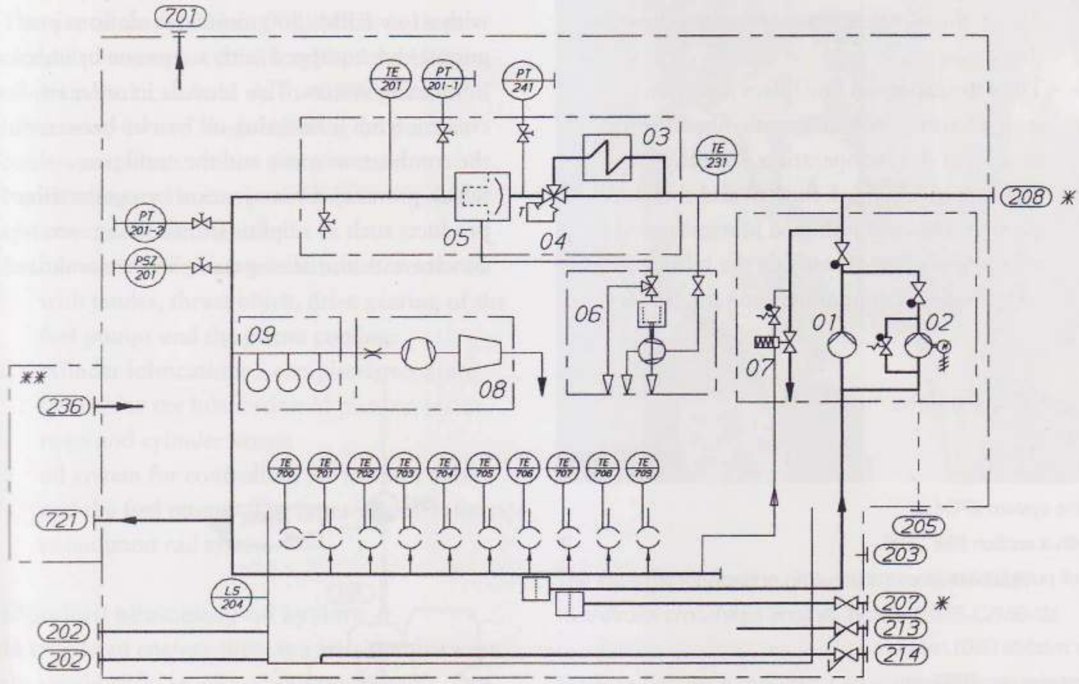


▶ The lubricating oil system of a four-stroke high-speed MTU-diesel engine.

The lubricating oil in the oil return drains into the sump.

1 *main lubricating oil pump*
2 *fine lubricating oil filters*
3 *lubricating oil cooler*
4 *crank shaft*
5 *cam shaft*
6 *sump*
7 *turbo blower*
8 *piston*
9 *cylinder head*

11.6.3 Category III, 500 tot 30,000 kW, four-stroke, medium-speed, H.F.O.



* If stand-by pump
** If common rail

System components:

01	Lubricating oil main pump
02	Prelubricating oil pump
03	Lubricating oil cooler
04	Thermostatic valve
05	Automatic filter
06	Centrifugal filter
07	Pressure control valve
08	Turbocharger
09	Camshaft bearings and cylinder head lubrication

Pipe connections:

202	Lubricating oil outlet (if dry sump)	DN150
203	Lubricating oil to engine driven pump (if dry sump)	DN200
205	Lubricating oil to priming pump (if dry sump)	DN80
207	Lubricating oil to el. driven pump	DN150
208	Lubricating oil from el. driven pump	DN100
213	Lubricating oil from separator and filling (if wet sump)	DN40
214	Lubricating oil to separator and drain (if wet sump)	DN40
236	Sludge from external filter (common rail)	DN25
701	Crankcase ventilation	DN80
721	Control oil to external filter (common rail)	DN25

All connections DIN 2576, PN10

Sensors, transmitters and switches:

LS204	Lube oil level, wet sump, low
PSZ201	Lube oil pressure, engine inlet
PT201-1	Lube oil pressure, engine inlet
PT201-2	Lube oil pressure, engine inlet (back-up)
PT241	Lube oil pressure, filter inlet
TE201	Lube oil temperature, engine inlet
TE231	Lube oil temperature, LOC inlet
TE70_	Main bearing temperature

▲
The internal lubricating oil system of a larger four-stroke diesel engine from MAN–B&W Type 32/40.

Main lubricating oil pump O1 is driven by the engine and pumps the pressurised lubricating oil to the main lubricating oil line.

Right; from 208, the lubricating oil can be supplied by an electrically driven lubricating oil pump.

Pre lubrication pump 02 is electrically driven and switches on automatically when the lubricating oil pressure is too low, for instance, when the engine is turned off. Pre lubrication is generally used for pre heating the engine prior to start and for cooling after turning off the engine.

Both pumps are fitted with a prescribed overflow valve in the bypass to avoid excessive pressures in the lubricating oil system .

In the main pipe line the lubricating oil first travels through oil cooler 03. Thermostatic three way valve 04 monitors, independent of the operating conditions, the predetermined lubricating oil exit temperature. The lubricating oil then passes through the automatic filter 05 which discharges the contaminated lubricating oil to the centrifugal filter 06.

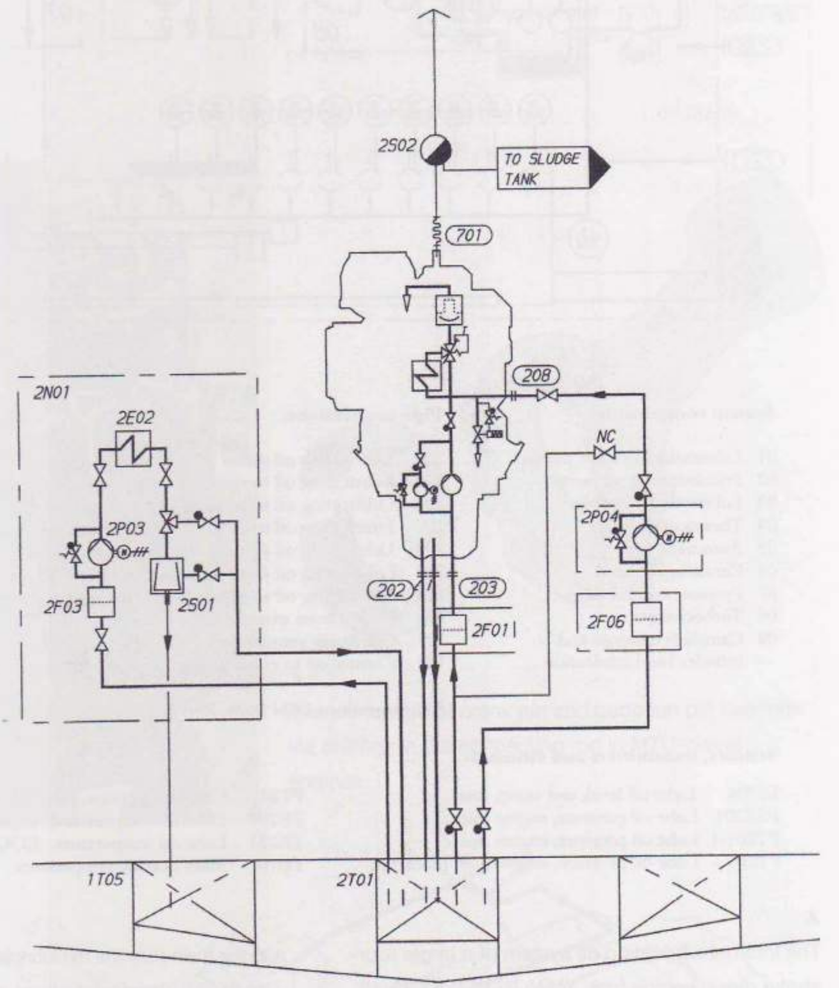
From the main lubricating oil line, the lubricating oil flows to the cam shaft bearings and cylinder head valves 09 to the turbocharger 08,

Underneath these camshaft bearings, the 10 main shaft bearing are situated. The lubricating oil continues through the drillings in the crank shaft to the crank pin bearings and via drillings in the connecting rod to the gudgeon pin, the piston rings and the inside of the piston for cooling.

– The **lubricating-oil pumps** can be driven by the engine or by electric motors. In the latter it is possible to switch on a lubricating- oil pump during the warming up and cooling down of the engine.
– **The lubricating-oil fine filters** are often manufactured with automatic filters, which mean that during operation one filter element is automatically back flushed and therefore cleaned. The contaminated lubricating-oil is subsequently led off outside the lubricating-oil system to a contamination tank. In this category most turbo blowers are lubricated by the main lubricating-oil system. Some engine types of the large four-stroke engines with a low RPM (300 to 600 revolutions per minute) are equipped with a separate cylinder lubricating system. This is made in order to create a fresh lubricating-oil barrier between the combustion space and the crankcase which prevents contamination by combustion products such as sulphur from coming into contact with the drive gearing in the crankcase.

▶
The external lubricating oil system from the same engine, a MAN–B&W type 32/40.

The **third** lubricating oil pump in the system 2PO4 is electrically driven and provided with a suction filter 2F06. The 'internal' main- and auxiliary oil pumps have a common suction filter 2F01.
Left of the engine is the separator module 2N01, which consists of an independent separator pump 2P03 with suction filter 2F03, the pre heater 2F02 and the separator 2S01.
The separator pump draws the to be cleaned lubricating oil from the system oil tank 2T01. The pre heater ensures that the lubricating oil temperature is maintained in the separator, this is the temperature at which the specific mass difference between water and lubricating oil is as high as possible to ensure optimum water separation. The cleaned lubricating oil drains to the lubricating oil drain tank. In larger four-stroke- and two-stroke engines this type of separator unit is constantly in stand-by mode (for the lubricating oil system). The separator is only switched off for repairs to the centrifuge or when the engine is out of operation for an extended period. The modern versions operate fully automatically and clean the separator bowl periodically. The dirt is discharged to the sludge tank 1T05.
The bleeder mounted on the engine contains a oil separator 2S01 which leads the oil back to the sludge tank. The gases are discharged outside the engine room.



System components:

1T05	Sludge tank
2E02	Heater (separator unit)
2F01	Suction strainer (main lube oil pump)
2F03	Suction filter (separator unit)
2F06	Suction strainer (stand-by pump)
2N01	Separator unit
2P03	Separator pump (separator unit)
2P04	Stand-by pump
2S01	Separator
2S02	Condensate trap
2T01	System oil tank

Pipe connections:		L32	V32
202	Lubricating oil outlet	DN150	DN150
203	Lubricating oil to engine driven pump	DN200	DN250
208	Lubricating oil from el. driven pump	DN100	DN125
701	Crankcase air vent	DN80	DN100

11.6.4 Category IV, 1500 tot 100,000 kW, two-stroke, low-speed, H.F.O.

The size of the lubricating-oil system in these crosshead engines is considerable. A lubricating-oil drain tank can hold up to 40,000 litres of lubricating-oil and the lubricating-oil pumps, coolers and filters are the size of a man.
In this category three types of lubricating-oil systems can be distinguished:

1 **lubrication of the crankshaft, the crosshead with guides, thrust block, drive gearing of the fuel pumps and the piston cooling;**
2 **cylinder lubrication;** a completely separate system for the lubrication of pistons, piston rings and cylinder liners;
3 **oil system for controlling the exhaust valves and the fuel pumps.** The latter is with reference to common rail system.

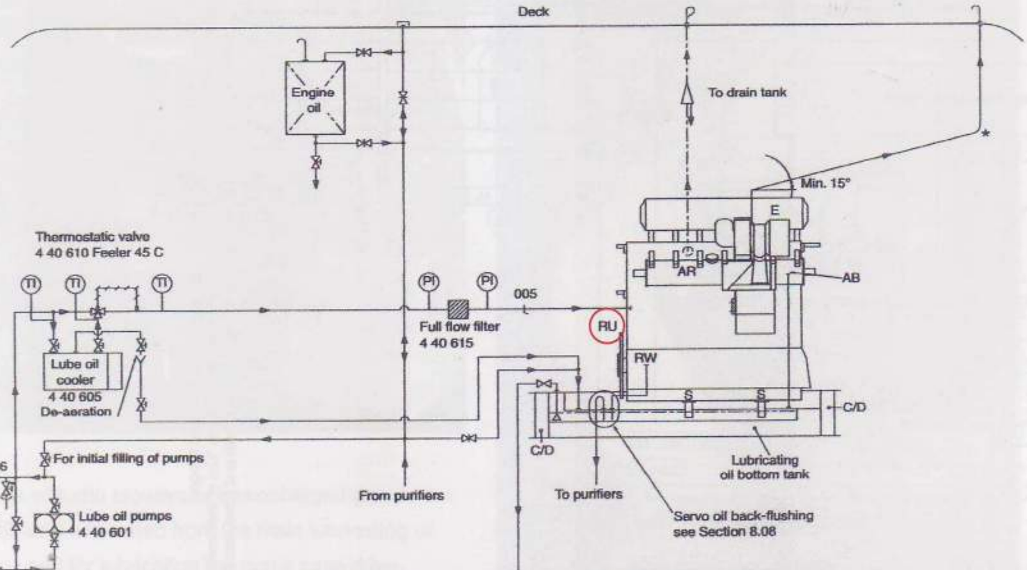
Standard lubricating-oil system

In crosshead engines there is a partition between the combustion space and the crankshaft including its gearing. The piston rod stuffing box in the scavenging-air space ensures that the contaminated lubricating-oil that leaks down the piston can not enter the crank case. **In this manner, the lubricating-oil in the crank case remains very clean.**
The system is cleaned with a lubricating-oil separator and an automatic fine filter.
The lubricating-oil- and cooling-oil system of the two-stroke crosshead engines of MAN–B&W type ME/ME-C are used as an example.

▲ Maintenance on the drive gear in the crank case of a two-stroke crosshead engine.

▼ The external lubricating oil system in a MAN–B&W two-stroke crosshead engine. Type ME/ME-C/ME-GI.

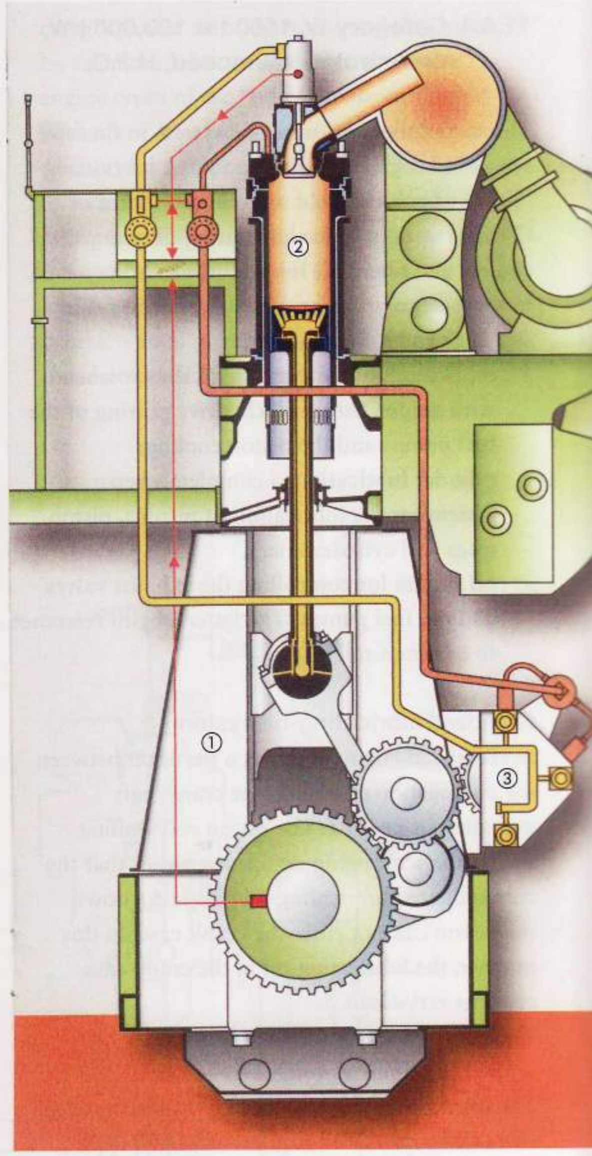
Two main lubricating oil pumps 4 40 601 draw the lubricating oil from the lubricating oil drain tank and force the lubricating oil through the lubricating oil cooler 4 40 610 and an automatic fine filter 4 40 615 to the main lubricating oil line in the engine.
Connection RU takes care of the lubrication and cooling of the main bearings, thrust block, axial vibration damper, piston cooling, crosshead bearings and connecting rod bearings.
RU also ensures the lubricating oil supply to the hydraulic system for driving the exhaust valves, the fuel pumps, the moment compensators and the torsional vibration dampers.
Lubricating oil separators which continuously clean the lubricating oil of dirt particles and water are also connected to the system .



The letters refer to 'List of flanges'
* Venting for MAN B&W or Mitsubishi turbochargers only

The lubricating oil system of two-stroke crosshead engines nowadays consists of three separate systems.

1 lubrication of the crank case drive gear and piston cooling
2 cylinder lubrication using special cylinder lubricating oil
3 the hydraulic section for driving the exhaust valves and operating the fuel injection system. The lubricating oil used here is the same as that used in the drive gear lubricating oil system.



The hydraulic system of a MAN–B&W two-stroke crosshead engine type ME.

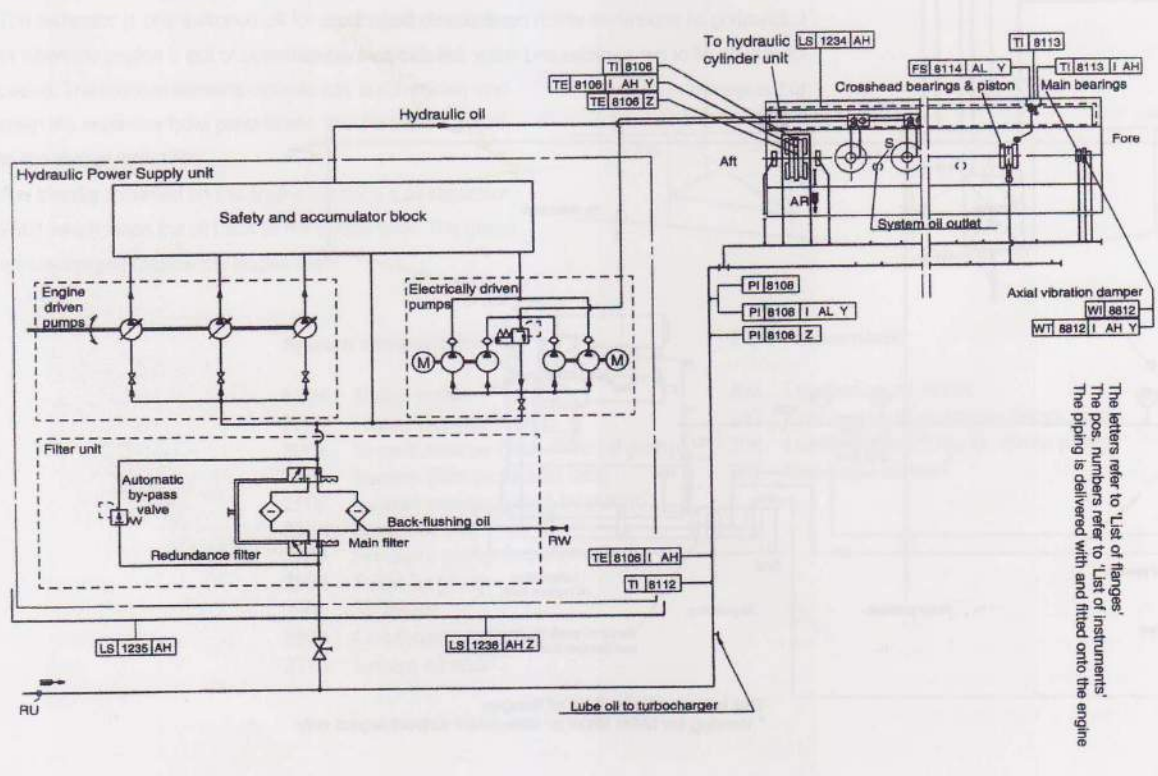
The hydraulic system consists of:

- an automatic fine filter and parallel to it a by-pass filter;
- two electrically driven lubricating oil pumps;
- three engine driven lubricating oil pumps;
- a safety- and accumulator unit

At start, one of the two electrically driven lubricating oil pumps is activated and is stopped as soon as the three engine driven lubricating oil pumps have taken over the oil supply.

The high pressure lubricating oil flows to the hydraulic units and is distributed to the electronically controlled fuel injection system and the electronically controlled exhaust valve activation system. The exhaust valve is closed by a traditional 'air spring'.

Hydraulic Power Supply – H.P.S.
Hydraulic Cylinder Units H.C.U.
Fuel Injection system and Valve Activation system – F.I.V.A.
Engine Control System, E.C.S.



Turbo-blower lubrication

As opposed to the older systems the bearings are not lubricated with a separate system by using the main lubrication system.

Mesh of the automatic fine-filter of the turbo-blower

Depending on the material used for the manufacture of the bearings:
white metal: 50 micro-millimetres;
aluminium tin: 40 micro-millimetres.

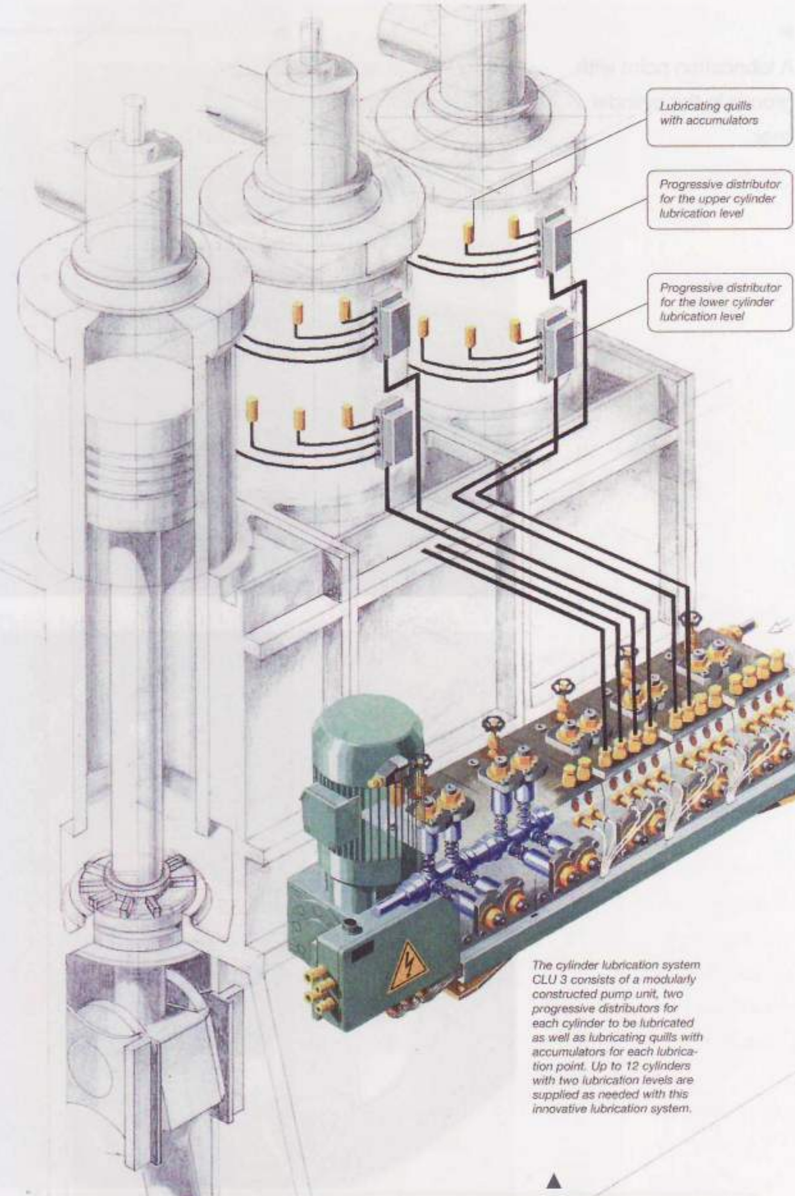
Comment

Of all metal particles larger than 25 micro millimetres, 85 to 90% remains in a filter with meshes of 40 micro millimetres, and metal particles of 35 micro millimetres in a fine filter with meshes of 50 micro millimetres.

Cylinder lubricating-oil system

The cylinders are lubricated independent from the main system with a special cylinder lubricating-oil. This cylinder lubricating-oil has to comply with strict requirements:

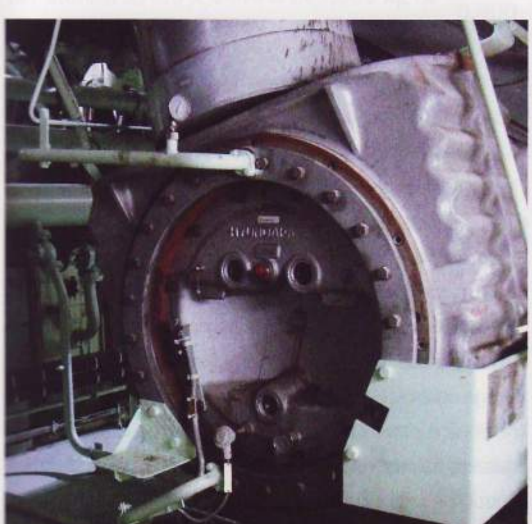
- the fuel has high sulphur content;
- a fraction of the lubricating-oil combusts in the exhaust-gas flow, this must be ashless;
- the other fraction of the cylinder lubricating-oil that leaks into the scavenging air space, is transported outside the engine in the top part of the piston rod-stuffing box;



The cylinder lubrication system CLU 3 consists of a modularly constructed pump unit, two progressive distributors for each cylinder to be lubricated as well as lubricating quills with accumulators for each lubrication point. Up to 12 cylinders with two lubrication levels are supplied as needed with this innovative lubrication system.

▲ The electrically driven cylinder lubricating oil unit

Each lubrication point is provided with special cylinder lubricating oil by its own plunger pump. Lubrication of the cylinder liner occurs at two levels.



▲ Bearings of turbo blowers or turbocharged groups are often directly lubricated from the main lubricating oil system used for lubricating the crank case drive gearing. In this large blower, lubrication takes place using its own lubricating oil system.

◄ An automatic fine lubricating oil filter.

► A lubrication point with groove in the cylinder liner.

► A large cylinder liner.

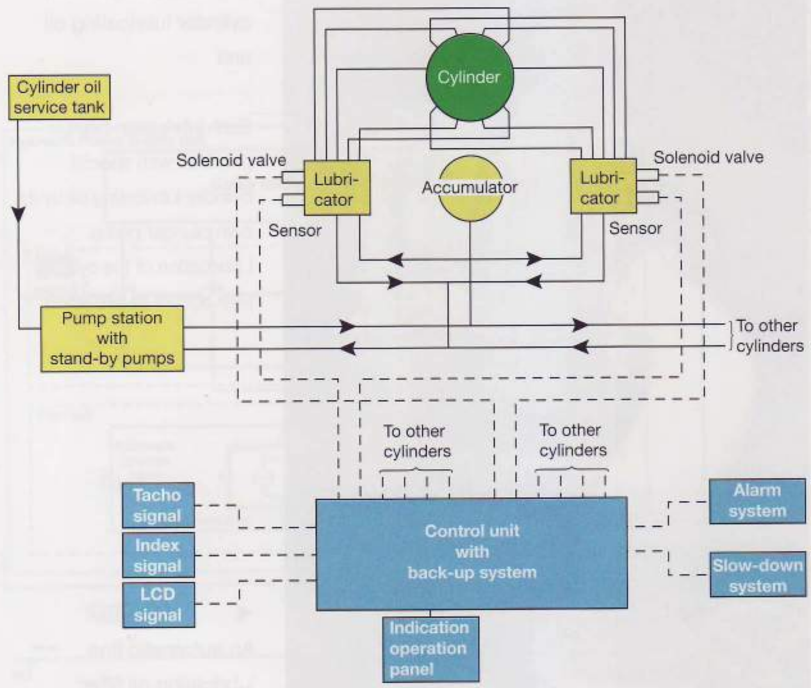
1 *inlet ports*
2 *grooves for cylinder lubrication*

- the scavenging-air spaces themselves are fitted with an oil drain, which is often set to drip in order to avoid blockage of the pipe and accumulation of lubricating-oil in the scavenging air spaces.

Next to fuel consumption the consumption of cylinder lubricating-oil is a major cost factor in the exploitation of these engines.
Too high a dosage of cylinder lubricating-oil produces extra contamination of the piston ring package, the combustion space, the exhaust gas ducting and the turbo blower. Too low a dosage causes additional wear of the piston, piston rings and cylinder liner.

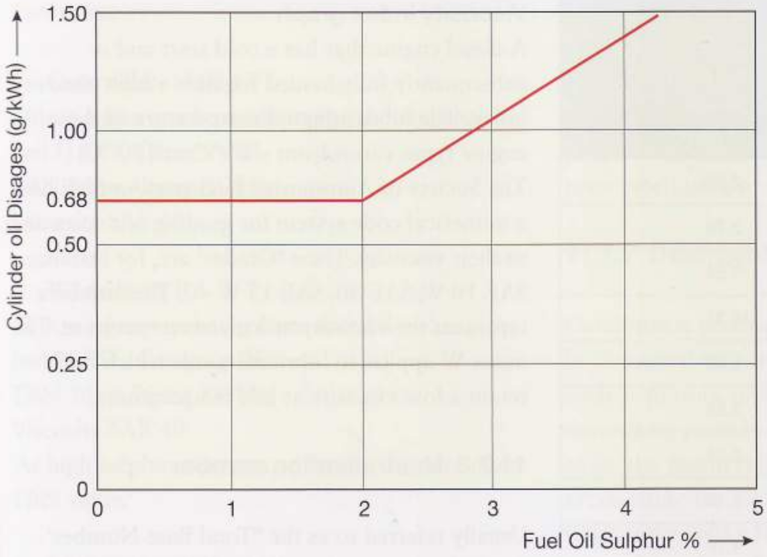
Optimal cylinder lubrication

Precise dosing of cylinder lubricant can be achieved with the '**Alpha Adaptive Cylinder oil Control System, Alpha A.C.C.-system**'.
This system takes into account engine load and fuel quality.
Sulphur causes wear and tear in the combustion space. Therefore, knowledge of the fuel's sulphur content is important in accurately measuring the cylinder lubricant quantity.



◄ The cylinder lubricating system for MAN–B&W two-stroke crosshead engines type ME.

The cylinder lubricating oil is pumped from a storage tank to a cylinder lubricating oil day tank and subsequently to the lubrication pump for each cylinder.
Each cylinder has eight lubricating points. Cylinder lubrication oil is supplied every fourth engine revolution at a constant pressure and with an electronically controlled timing and dosage control.
The dosage of the lubricating oil can be tuned with an adjusting screw. The entire system is regulated by a cylinder control unit which calculates the fuel injections frequency based on the number of rotations and also the set capacity (index) of the high-pressure pump. This is required to supply the correct dosage to the cylinder.



The dosage of the cylinder lubricating oil in the 'Alpha A.C.C.'-system from MAN–B&W.

With fuel sulphur content in excess of 2%, the cylinder lubricating oil dosage rises from 0.68 grams per kWh to 1.5 gram s per kWh with a sulphur content of 4%.

The dosage is therefore dependent on the sulphur content of the fuel and the engine load.
The following rule of thumb then applies:
Cylinder lubricating-oil dosage =
0.34 gram/kWh × S%
The minimal dosage is 0.68 gram per kWh at 2% sulphur.

Example
Cylinder lubricating-oil consumption: engine capacity 86,000 kW, sulphuric content of fuel 3.5%.
Consumption per hour:
86,000 x 0.34 x 3.5 = 102,340 grams or 102 kg
per day: 24 x 102 = 2448 kg
annually with an operating time of 6500 hours:
per annum 270.8 x 2448 = 663,000 kg or $\frac{663,000}{0.900}$
of 736.7 m³!

11.7 Lubricating-oil properties

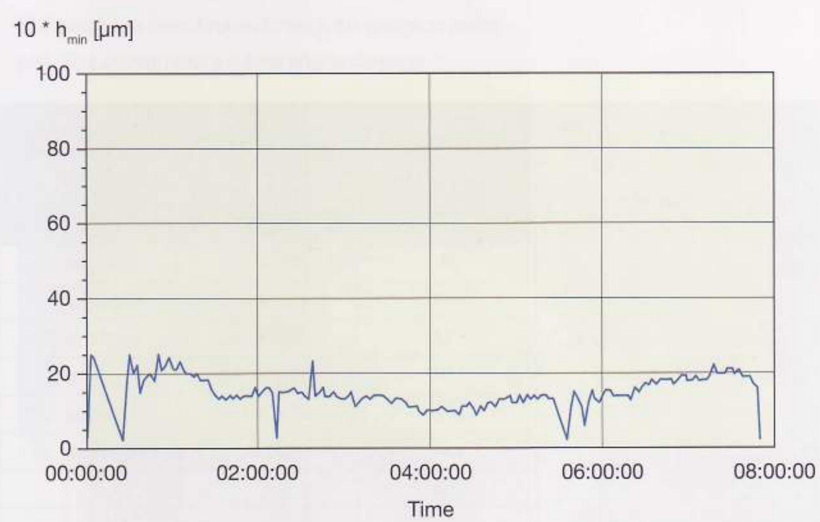
In order to acquire a good insight into engine lubrication, it is imperative to be familiar with the most important properties of lubricating-oil. The significance of these properties is shown each time there are problems with engines and in the assessment of lubricating-oil samples.

11.7.1 Viscosity expressed in centistokes (cSt) or mm²/sec. at 40 °C

To ensure that, for instance, the lubricating film between the crankshaft and crankshaft bearing is sufficiently thick, it is imperative that it has a certain viscosity.

Thick lubricating-oil, lubricating-oil with too high a viscosity produces high frictional loss and therefore a heat increase. Thin lubricating-oil, lubricating-oil with too low a viscosity seeps from the crankshaft, resulting in a pressure loss and allowing the bearing and shaft to come into contact with each other. This in turn causes wear.

Viscosity is divided into viscosity categories. This classification is incorporated in the standard specification 150–3348. Lubricating-oil viscosity is divided into eighteen areas. Each area encompasses the viscosities between two limits.



During lubrication of the engine parts, a very thin lubrication film separates the various parts.

The film thickness is usually no more than several tenths of a millimetre. Here the thickness is approximately 15 to 20 microns.

Lubricant properties

SAE Viscosity Grade	Viscosity at 40° C (mPa/sec.)	Viscosity at 100° C (mPa/sec.)	Viscosity at HTHS (mPa/sec.)
10 W	31.87	5.18	2.20
20	61.02	7.36	2.79
30	85.72	9.70	3.54
40	139.72	12.81	4.31
50	189.74	16.34	5.30
5W30(A)	47.87	8.90	3.24
5W30(B)	59.79	9.99	3.24
10W40(A)	63.66	11.06	3.10
10W40(B)	92.69	11.22	4.19
10W40(C)	74.65	12.10	3.88
20W40	89.40	11.00	3.89
10W50(A)	105.14	15.80	4.05
10W50(B)	134.19	15.92	4.80

▲
The lubricating oil viscosity is divided into a number of grades. Each grade contains the viscosities between two limits.

11.7.2 Viscosity index

The viscosity index denotes the measure in which the viscosity changes in relation to the temperature. Lubricating-oil that is barely susceptible to temperature differences where viscosity is concerned is said to have a high viscosity index.

Viscosity index graph

A diesel engine that has a cold start and is subsequently fully loaded requires a high viscosity index. The lubricating-oil temperature of these engine types varies from – 10 °C to 100 °C!
The Society of Automotive Engineers, or SAE has a numerical code system for grading oils according to their viscosity. These 'Grades' are, for instance, SAE 10 W, SAE 30, SAE 15 W 40. The numbers represent the viscosity at a given temperature. The index W applies to lubricating-oils which must retain a low viscosity at low temperatures.

11.7.3 Neutralisation number

Usually referred to as the "Total Base Number'. Fuels always contain sulphur. In light fuels the sulphur content is extremely low, for instance 0.15%. In using heavy fuels this can rise to 4%. The acids which are consequently produced by the combustion process can severely damage engine parts. These acids must be chemically neutralised by the addition of alkaline agents to the lubricating-oil.
Calcium hydroxide is in this case commonly used. Therefore, the neutralisation value is a measure of the alkalinity of the lubricating-oil. The higher the possibility of the formation of acids in combustion, the higher the required neutralisation value or the Total Base Number (TBN). Today one often speaks of the BN, Base Number.
Every engine manufacturer provides the required TBN values. They are denoted in mg KOH per gram lubricating-oil.

▶
The viscosity grades of lubricating oil; in this instance for Caterpillar-diesel engines.

The grades runs from OW to 60. 'W' indicates 'Winter'; that is low temperatures. At low temperatures the viscosity is established for operation, here cranking and pumping lubricating oil.

SAE Viscosity Grade	Low Temperature Viscosities		Low Temperature Viscosities		
	Cranking[b] (cP) max at temp °C	Pumping[c] (cP) max with no yield stress at temp °C	Low Shear rate Kinematic[d] (cSt) at 100 °C		High Shear rate (cP) at 150 °C min
			min	max	
OW	6200 at -35	60,000 at -40	3.8	–	–
5W	6600 at -30	60,000 at -35	3.8	–	–
10W	7000 at -25	60,000 at -30	4.1	–	–
15W	7000 at -20	60,000 at -25	5.6	–	–
20W	9500 at -15	60,000 at -20	5.6	–	–
25W	13,000 at -10	60,000 at -15	9.3	–	–
20	–	–	5.6	< 9.3	2.6
30	–	–	9.3	< 12.5	2.9
40	–	–	12.5	< 16.3	2.9 (0-40, 5W-40, 10W-40 grades)
40	–	–	12.5	< 16.3	3.7 (15-40, 20W-40, 25W-40 grades)
50	–	–	16.3	< 21.9	3.7
60	–	–	21.9	< 26.1	3.7

SAE: Society of Automotive Engineers
API: American Petroleum Institute
ASTM: American Society of Testing Materials

Examples

1 Caterpillar–MaK 43
This is a four-stroke medium-speed diesel engine, fuel H.F.O., Category III.
TBN 30 to 40 mg KOH/g
Viscosity SAE 40

2 Wärtsilä 38 B
This is a four-stroke medium-speed diesel engine, fuel H.F.O., Category III.
TBN 30 to 55 mg KOH/g
Viscosity SAE 40
'At high sulphur contents adhere to the maximum TBN value.'

3 MAN–B&W K 98 MC
This is a two-stroke low-speed crosshead engine, fuel H.F.O., Category II.
Lubricating-oil circulation in the crank case.
TBN 5 to 10 mg KOH/g
Viscosity SAE 30
Cylinder lubricating-oil TBN 70 mg KOH/g
Viscosity SAE 50

Generally, the following values are identical for all engine manufacturers:

- for high-speed four-stroke engines with M.D.O. as fuel (Category I + II) 4 to 6;
- for medium-speed four-stroke engines with H.F.O. as fuel (Category III) 30 to 55;
- for two-stroke-crosshead engines with H.F.O. as fuel (Category IV)
- in the crank case 5 to 10;
- cylinder lubricating-oil 70 to 90.

11.7.4 Solidification point

This is the temperature at which the lubricating-oil solidifies. This property is only pertinent to extremely cold conditions. Lubricating-oil with few paraffins have a lower solidification point.

11.7.5 Oxidation stability

There is a lot of air is in diesel engines, especially in the crank case. Lubricating-oil is therefore exposed to air. The oil in the crankcase splashes in all directions. If the lubricating-oil oxidises, acids with a high viscosity are generated which can cause the blocking of filters and the small lubricating-oil pipes. Oxidation is accelerated at higher temperatures. By adapting the refining process and an anti-oxidation additive this can mostly be averted. These additives consist of zinc or sulphur compounds.

11.7.6 Additives in lubricating-oil

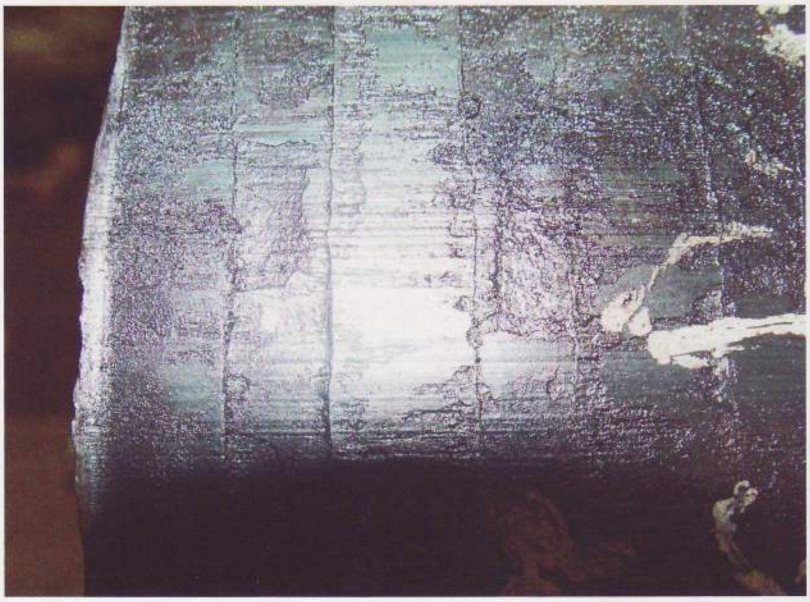
For each specific lubricating-oil application, chemicals may be added which either improve its properties, create new properties or compensate poor lubricating properties.

11.7.7 Detergents or 'solvents'

Combustion products, among which carbon, can be deposited on parts of the combustion space, such as pistons, piston rings, cylinder heads and scavenging ports in two-stroke engines. This may cause the piston rings to lodge in the grooves thus accelerating the wear and tear of the piston, piston rings and cylinder liner. Leakage of combustion gases along the ring packet will also increase. If carbon settles **behind** the piston rings the piston rings will slowly begin to protrude through the lubricating film on the cylinder liner. This may result in a corroded or seized piston. This could in fact cause complete engine failure. The seizing of piston rings is referred to in engine technology as **'ring sticking'**. The detergents often consist of sulphur/phosphor compounds.

▼
A seized piston in a cylinder liner.

The piston has been broken through the gudgeon piston pen. This usually results in total engine damage.



▲
A severely damaged piston in a two-stroke crosshead engine.

The piston rings are jammed in the piston ring grooves and are barely recognisable. The piston and rings are heavily grooved.

11.7.8 Dispersants

These are lubricating-oil additives which ensure that the combustion products do not settle and stick to engine parts but are kept in suspension. They can then be filtered by the fine lubricating-oil filter or in larger installations by a centrifugal separator.

11.7.9 Extreme Pressure (E.P.) additives or wear reducing substances

These are added in exceptional instances required for boundary lubrication. This may occur in cog wheels and very highly loaded engine bearings. They form a film on the metal surfaces. They are mostly sulphur and phosphor compounds.

11.7.10 Viscosity index improving agents

These ensure that the lubricating-oil does not get too thin at increasing temperatures. A whole range of chemicals is applied to this end, such as:
- polyisobutenes;
- polymethacrylates;
- styrene;
- polyesters.

11.7.11 Ash content

This is a non-combustible lubricating-oil residue. It is metal compounds such as barium, calcium and magnesium, which provide the correct alkalinity or TBN of the lubricating-oil. High ash contents may affect the diesel engine's performance.

11.7.12 Biocides or bactericidal agents

In spite of all the precautionary measures taken, lubricating-oil will come into contact with water. Consider moist air in the crank case of a stationary diesel engine. In this kind of environment certain bacteria develop very rapidly. The bacterial growth can become excessive and cause severe damage to engines.

11.8 Cleaning lubricating oil

Lubricating-oil absorbs many substances in the engine which adversely affect the quality and reduce lubrication and cooling of the engine. These substances are, among other things:
- water, from leaking cooling water and water vapour from the air precipitation in the engine;
- fuel, usually from leaking fuel pumps. The seals of the fuel pump and gear drives of the cam shaft are not optimal;
- metal particles, from wear of the moving parts such as bearings, shafts, pistons, piston rings and cylinder liners. Due to corrosion or cavitation metal particles from pipe walls, heat exchangers and pumps can also be launched.
- combustion products. Especially in four-stroke trunk piston engines there is always some 'deposition' along the piston rings of carbon particles; sulphuric acid, from the sulphur in the fuel, and other combustion by-products on the piston rings;
- coolants, similar to water leakage this is due to leaking cooling systems;
- carbon particles from the lubricating-oil itself. When lubricating-oil is used for cooling, and when a certain temperature, approximately 400 ° C, is exceeded, the lubricating-oil begins to decompose and carbonise;
- oxidation products of the lubricating-oil. Due to air exposure lubricating-oil may oxidise. This produces long chained compounds which thicken the lubricating-oil;
- dirt particles such as sealant materials, sand, glass droplets and fine dust.

During engine repairs, sufficient attention should be paid to working 'hygienically'. Often engines are contaminated with dirt ranging from fine dust to tools and cleaning cloths.

Note
In extreme conditions such as sand storms, for instance in the Middle East, or the spread of pollen in large agricultural areas one must take special precautionary measures to protect the combustion air from these dirt particles.

Lubricating-oil storage should also be a focal point. **It is easy for moisture in the air to condensate in a storage tank that is nearly empty.**

Lubricating-oil can be cleaned in various ways.

▼

The various metal filter constructions.

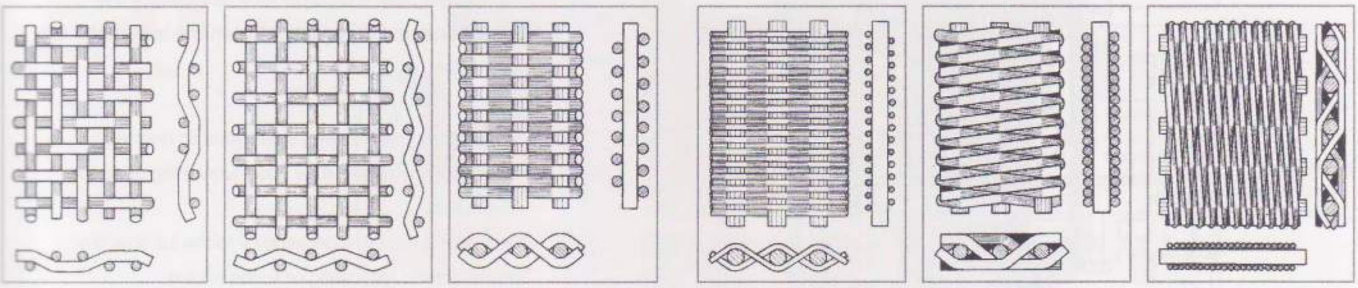
Dependant on, amongst others, the required filter mesh size, the filter surface, the flow rate and the lubricating oil temperature a certain filter element is selected. Most mesh sizes vary from 10 to 25 microns, but there are also filters with a smaller mesh size, namely 2 to 4 microns.

11.8.1 Fine lubricating-oil filters in the supply pipes of lubricating-oil pumps

This is applied to all engine types. The mesh size determines the manner in which filtering occurs. In small diesel engines filters are renewed. In larger engines they are cleaned.

11.8.2 Centrifugal filters

In increasingly more engines in categories II and III, centrifugal filters are used alongside the regular fine lubricating-oil filters. The centrifuge is driven by the lubricating-oil pressure and throws dirt and water along the outer side. After a specified number of operating hours the filters are manually cleaned.

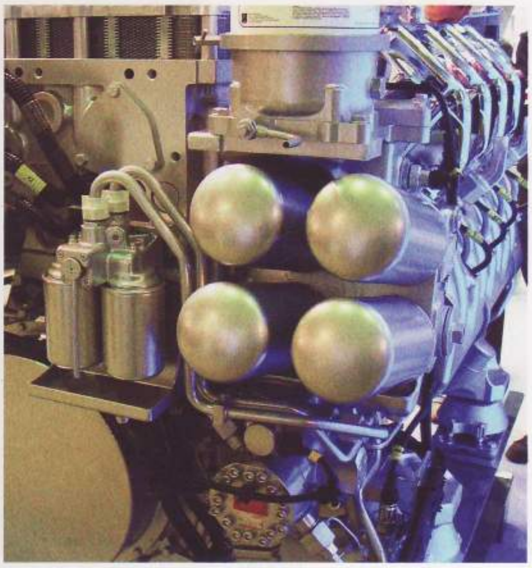


A B C D E F

▲

Two lubricating oil fine filters in a diesel engine from Volvo Penta.

Left: a fuel filter.



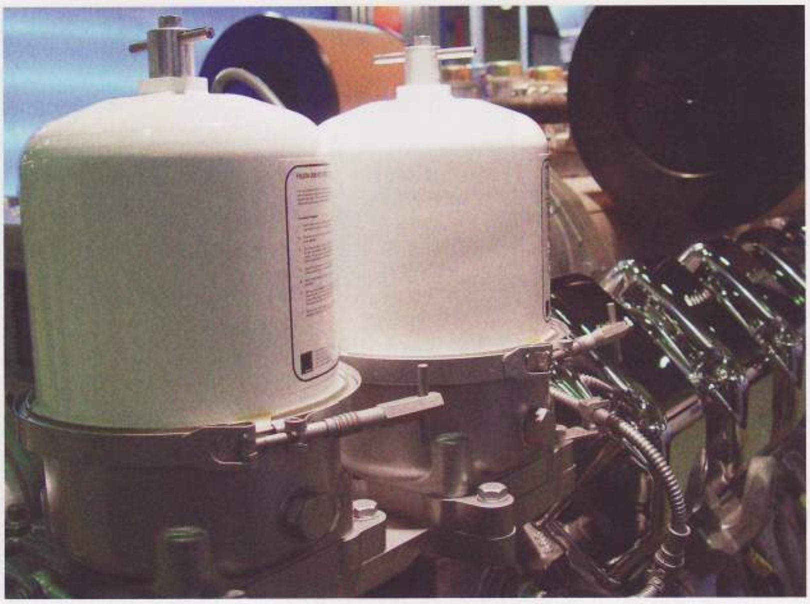
▲

Four lubricating fine oil filters in a diesel engine from MTU.

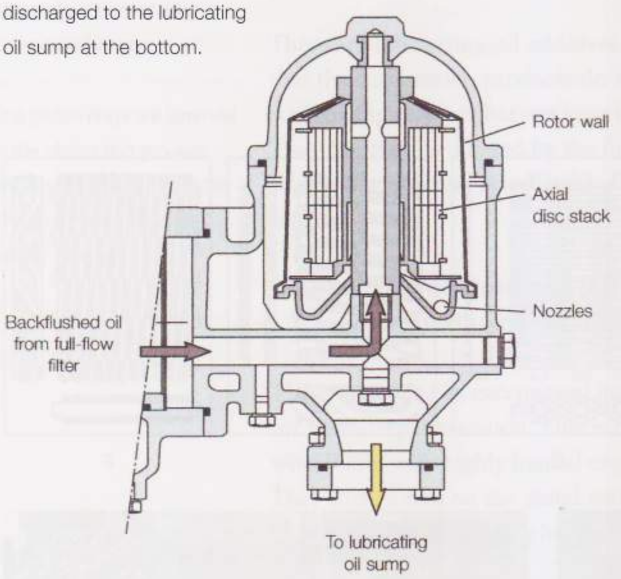
Left: two fuel filters. A centrifugal lubricating oil filter is just visible above the lubricating oil filters.

▼
A centrifugal filter driven by its own lubricating oil pressure.

Contaminated lubricating oil enters from the back-flushed filter on the left. It is thrown against the filter walls by the centrifugal forces in the rotating filter element. The nozzles spray the lubricating oil outside the element and the filter begins to turn like a garden sprayer.
The dirt is thrown against the drum wall and is periodically removed. The clean lubricating oil is discharged to the lubricating oil sump at the bottom.

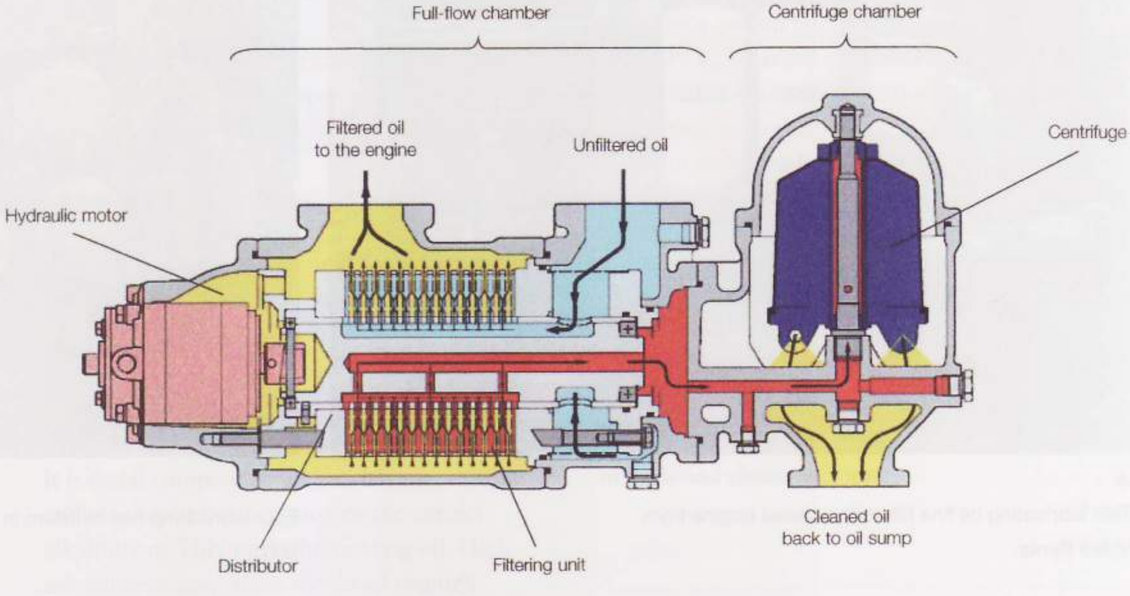


◄
Two centrifugal lubricating oil filters mounted on a diesel engine.



▼
A combination of an automatic full-flow filter and a centrifugal filter which periodically cleans the contaminated lubricating oil from the automatic full-flow filter.

The full-flow filter is driven by a hydraulic engine.
A distribution ring provides good lubricating oil supply distribution.
light blue: unfiltered lubricating oil to the full-flow filter.
yellow: filtered lubricating oil to the engine.
red: contaminated lubricating oil to the centrifugal filter.
blue: centrifugal filter.
yellow: cleaned lubricating oil returned to the lubricating oil sump.

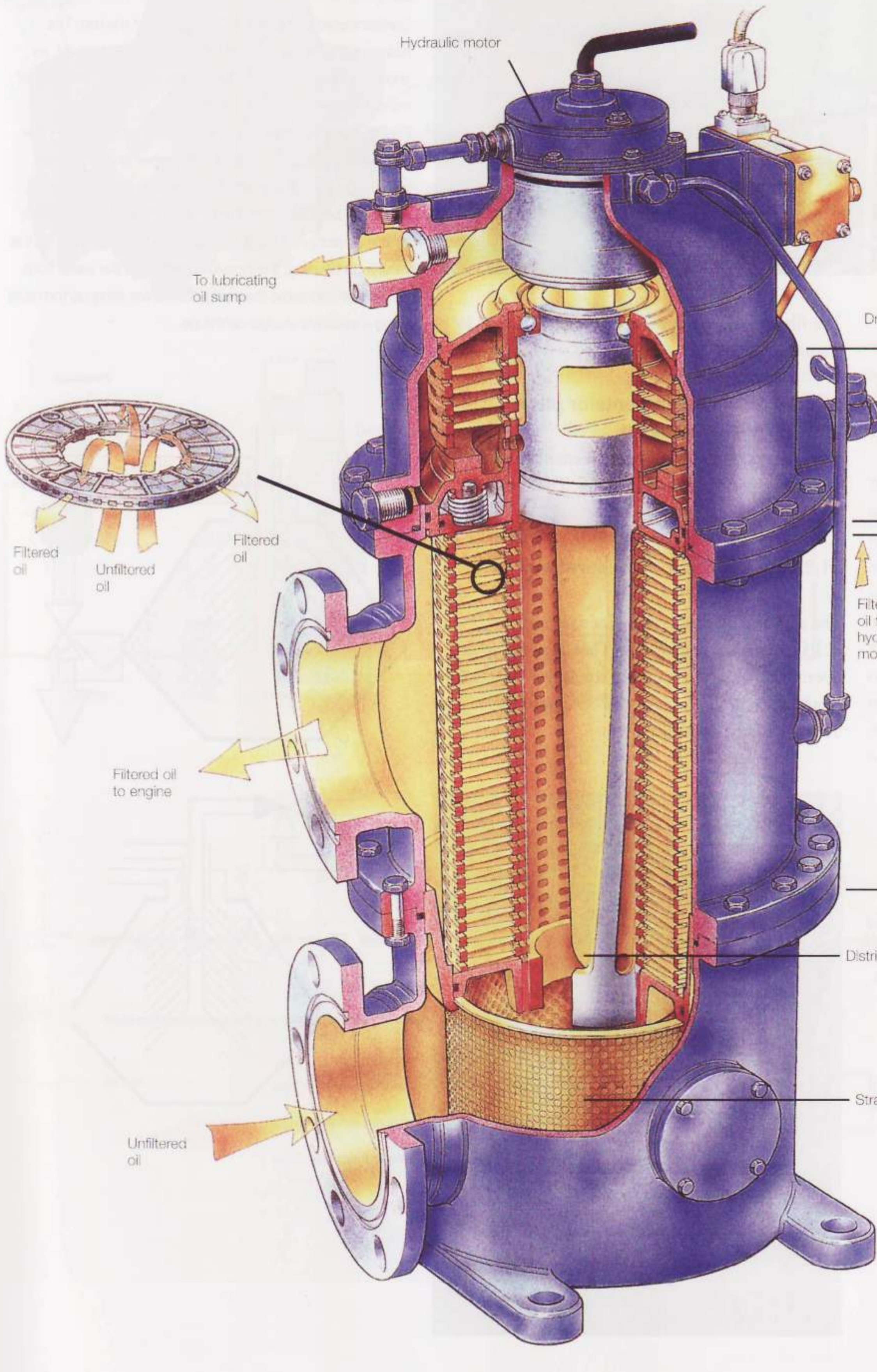


11.8.3 Automatic fine-filters

These filters are periodically cleaned by an automatic back-flushing system.

▼

The same type of automatic back-flow filter arranged vertically.



▲
An automatic lubricating fine oil filter.

The filters are always used in category III and IV engines.

11.8.4 Magnetic elements or plugs

These plugs are mostly found in small engines. They detain mostly iron particles. They require regular manual cleaning.

11.8.5 Lubricating-oil separators

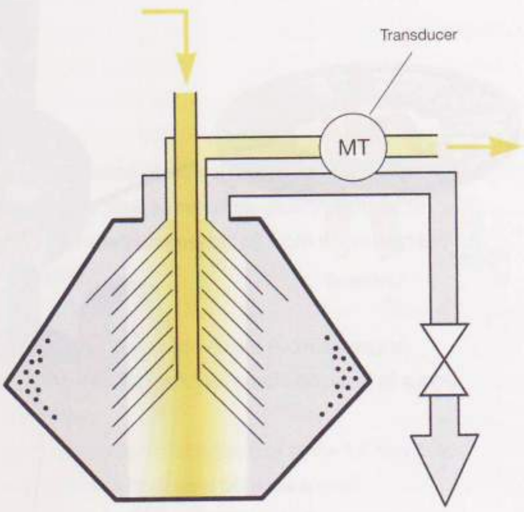
The slightly larger diesel engines running on H.F.O. often use centrifuges. They generally operate automatically and in the larger types cleaning occurs automatically on a periodic basis. This is called 'shooting'.

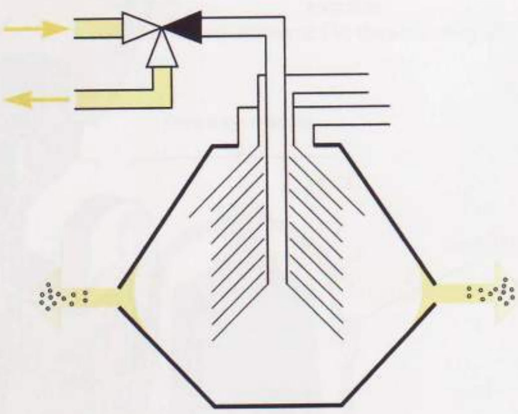
►
An automatic lubricating oil centrifuge.

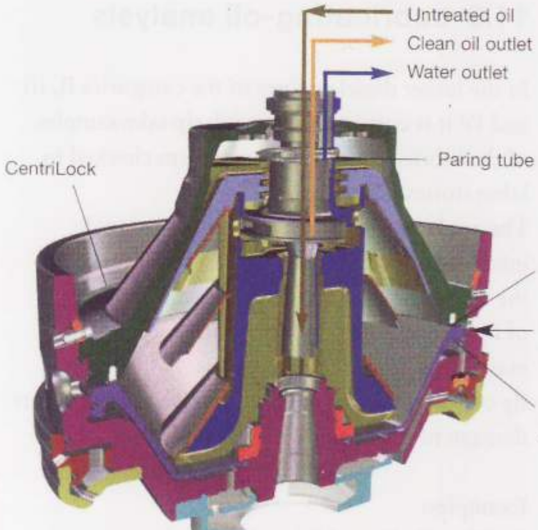
Water and dirt are separated and the dirt is automatically removed at regular intervals by the 'shooting*' of the centrifuge and is fed to a sludge tank.

▼
The operating principle of a centrifuge or separator.

Contaminated lubricating oil is supplied at the top. The heavier dirt particles are flung outwards and removed, as shown in the lower figure. This is referred to as 'shooting of the centrifuge'.
Prior to commencement of the 'shooting programme', the lubricating oil supply is automatically shut and the entire rotating bowl filled with water. At the moment that the moveable bottom drops, the dirt is removed from the bowl by water pressure and discharged to the sludge tank via the centrifuge housing. It is possible to remove the water from the sludge and avoid the sludge tank from filling up too rapid using a separate sludge centrifuge.

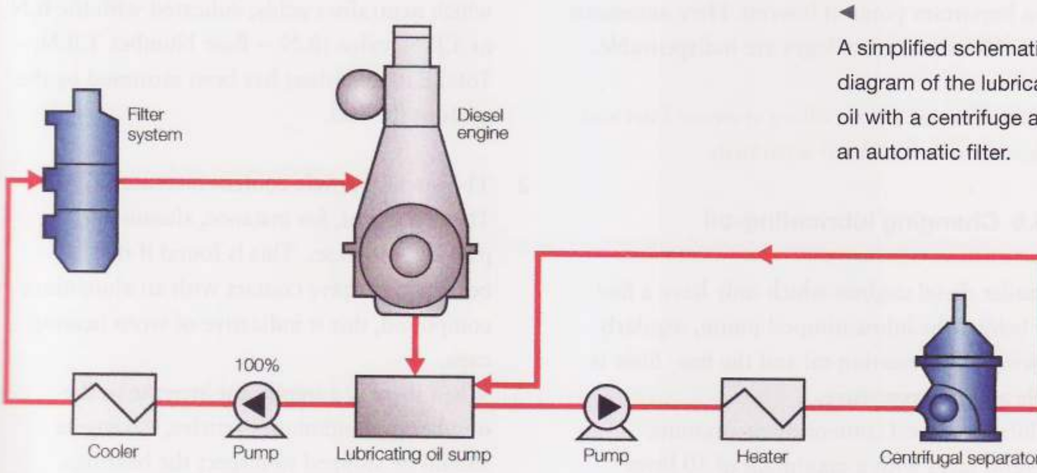




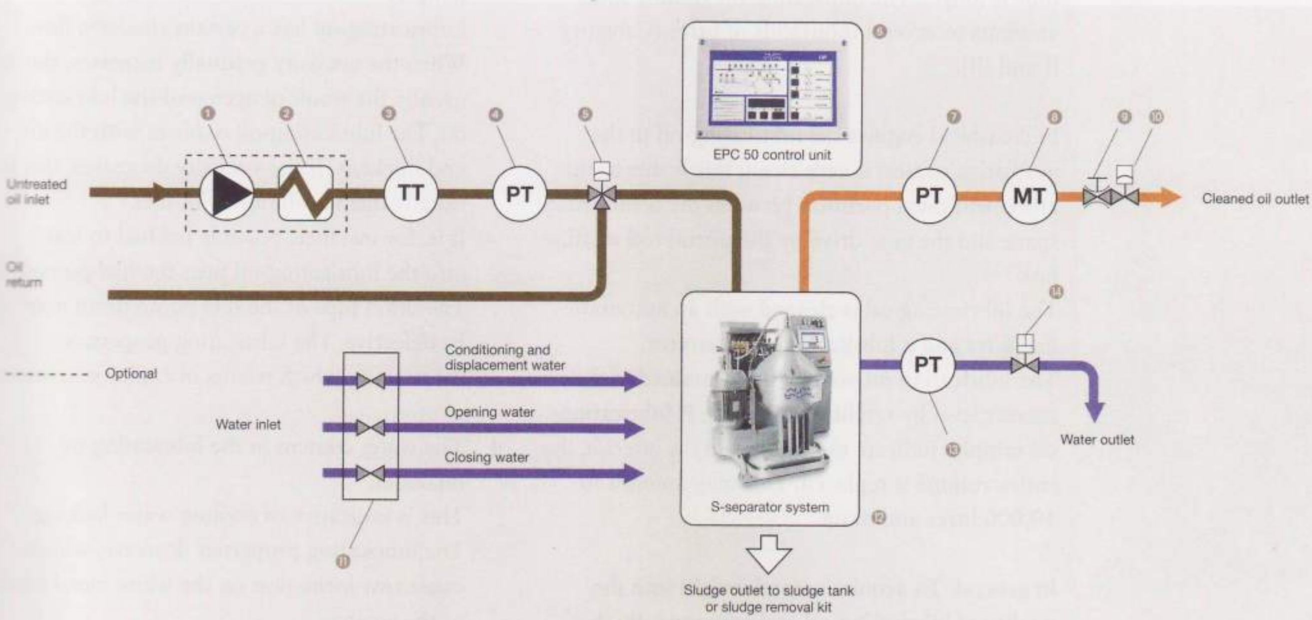


◄ The construction of a modern lubricating oil centrifuge.

Brown: the unfiltered oil supply
Orange: the filtered lubricating oil discharge
Blue: the water outlet



◄ A simplified schematic diagram of the lubricating oil with a centrifuge and an automatic filter.



▼ An automatic lubricating oil separator in an Alfa Laval type S.

This is the latest type, 2005, with the operating principle Alcap. The automated module EPC 50 is a fully automatic operational management system. Inspection: every six months or 4000 operating hours maintenance: every eighteen months or 12,000 operating hours

General commentary

- In four-stroke trunk piston engines running on H.F.O. the contamination of the lubricating-oil is considerable.
- In four-stroke trunk piston engines running on M.D.O. the contamination of the lubricating-oil is considerably less and cleaning the lubricating-oil significantly simpler.
- Due to rising fuel prices in recent years, many ships equipped with a modern propulsion engine and using M.D.O. are converted to run on H.F.O.. Naturally, the engine must be suitable for H.F.O.. They are often engines with a 200 to 260 mm piston bore with a power output of 1500 to 3000 kW. The investment is recouped in one year. Adequate cleaning of the lubricating-oil is now an important point of interest. Here automatic fine filters and centrifuges are indispensable.

Also see Chapter 24, Auxiliary systems: Fuel and lubricating-oil centrifugal separators.

11.8.6 Changing lubricating-oil

In smaller diesel engines which only have a fine filter behind the lubricating-oil pump, regularly renewing **all** lubricating-oil and the fine- filter is simple and not expensive.
The lubricating-oil content often amounts to several litres with a maximum of 50 litres (Category I). In larger systems the lubricating-oil is cleaned more intensively and the operating life time is longer. The lubricating-oil volume soon amounts to several thousands of litres (Category II and III).

In crosshead engines the lubricating-oil in the gear-drive section is very clean; this is due to the aforementioned partition between the combustion space and the gear drive by the piston rod stuffing box.
The lubricating-oil is cleaned with an automatic fine filter and a lubricating-oil separator.
The lubricating-oil volume is maintained at the correct level by refilling (leakages). If lubricating-oil samples indicate that the quality is inferior, the entire volume is replaced. This may amount to 10,000 litres and more.

In general: To acquire a good insight into the quality of lubricating-oil and consequently the state of the diesel engine, it is imperative that samples be taken systematically to monitor the lubricating-oil.

11.9 Lubricating-oil analysis

In the larger diesel engines of the categories II, III and IV it is customary to regularly take samples of the lubricating-oil and have them checked in laboratories.
The analysis results of such samples provide information with regard to the properties of the lubricating-oil after a specified number of operating hours and therefore the state of maintenance of the diesel engine.
By consistently checking the lubricating-oil, severe damage to the engine can be avoided.

Examples:

1 **The alkaline additive shows a decrease.**
The alkaline additive in the lubricating-oil, which neutralises acids, indicated with the B.N or T.B.N. value (B.N. – Base Number, T.B.N. – Total Base Number) has been saturated by the acids in the fuel.

2 **The metal -particle content increases.**
The content of, for instance, aluminium particles increases. This is found if the bearing caps have contact with an aluminium compound, this is indicative of worn bearing caps.
When there is a significant increase in the number of aluminium particles, the engine should be stopped to inspect the bearings.

3 **The viscosity decreases; the lubricating-oil is diluted.**
Lubricating-oil has a certain viscosity, flow. When the viscosity gradually increases, this is usually the result of ageing of the lubricating-oil. The lubricating-oil oxidises with the air and thickens. If the viscosity decreases, this is usually due to dilution with fuel.
It is, for instance, possible for fuel to leak into the lubricating-oil near the fuel pumps. The drain pipe of the fuel pump drain may be defective. The lubricating properties deteriorate, which results in engine corrosion.

4 **The water content in the lubricating-oil increases.**
This is indicative of cooling water leakage. The lubricating properties decrease, which cause rust formation on the white metal parts of the engine.

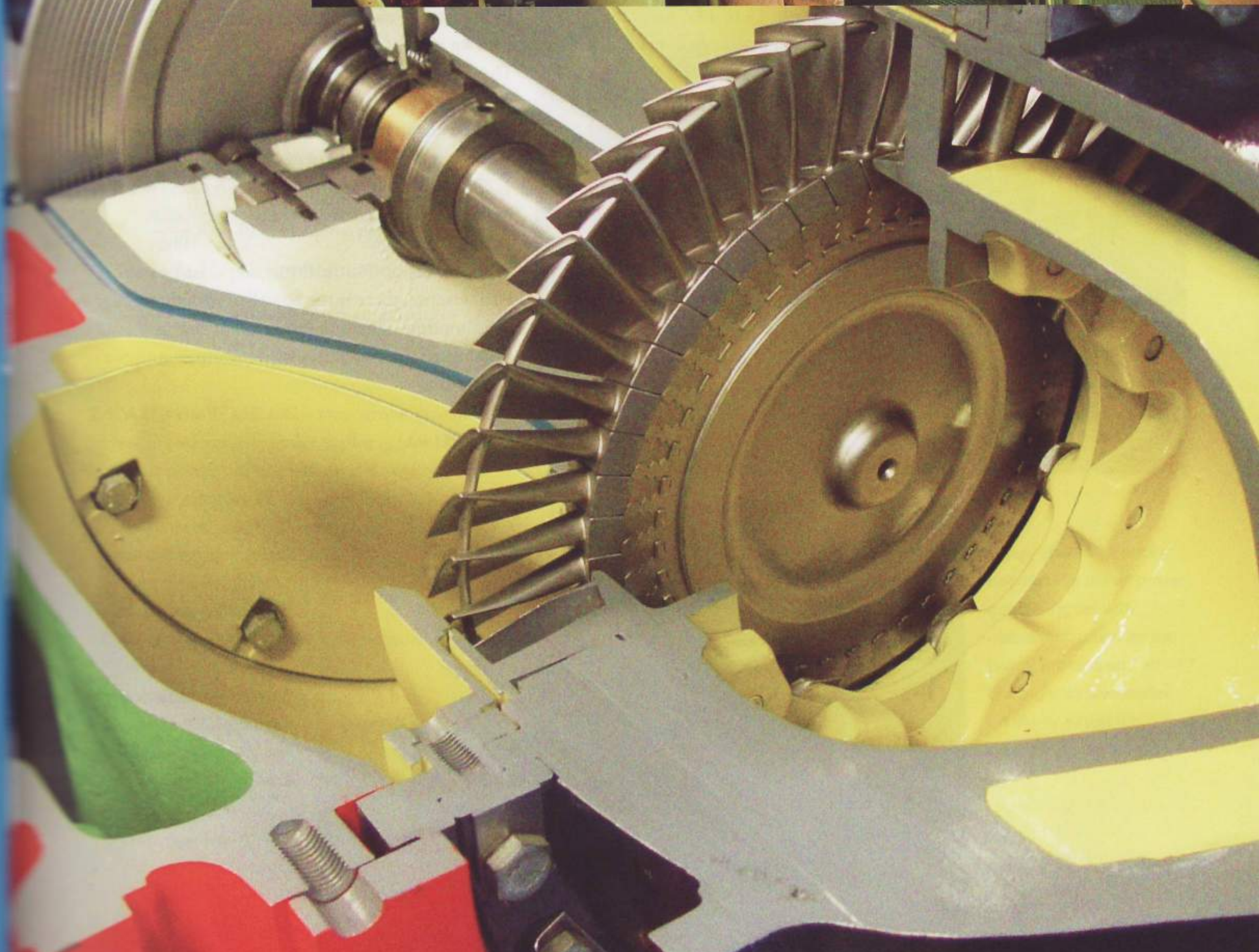
CH 12

Air supply





Diesel engines require a large amount of air in the combustion process. For example, this cross head engine, a Wärtsilä-Sulzer 12 RTA 96 C at full power output uses approximately 3 kg air per second, 11.000 kg air per hour en 201.000 kg air per day.



The turbo blower, essential for the air supply of modern diesel engines.

12.1 Introduction

Diesel engines use a considerable amount of air for their processes. Not only do they require air for the combustion process, air is also indispensable in the scavenging process and cooling hot engine parts that are in close proximity of the combustion space.

The total amount of air required must be sufficient for:

- the combustion process;
- cleaning the cylinder;
- cooling the parts surrounding the combustion space, such as the piston crown, the flame plate of the cylinder head and the exhaust valves.

12.2 The amount of air

Theoretically, the complete chemical combustion of 1 kg of fuel requires 14 to 14.5 kg of air. Lighter fuels with high hydrogen content use approximately 14.5 kg of air and the heavier fuels with lower hydrogen content approximately 14 kg.
In reality, twice the amount of air required for the combustion process is supplied. Obviously, this depends on the type of diesel engine and the combustion system.

This is referred to as the practical amount of air that is available for combustion.

Combustion air factor

$$\text{combustion-air factor} = \frac{\text{practical amount of air supplied for combustion}}{\text{theoretical amount of air supplied for combustion}}$$

λ = lambda = air factor

$$\lambda \text{ combustion} = \frac{\text{L practical}}{\text{L theoretical}}$$

The scavenging - and cooling processes also require air. The total air consumption is denoted by the total air factor – λ total. This is considerably higher than the combustion air factor, dependant on the type of engine and design, approximately 3.

Therefore for 1 kg of fuel we supply approximately 3 x 14 = 42 kg of air!
In two-stroke engines approximately 3 x 16.5 = **51.5 kg air.**
The density of air at 0 °C and 1 bar is 1.276 kg per m³.
This means that for each kilogram of fuel $\frac{42}{1.276}$ **m³ air supply, so each kilogram of fuel requires ± 33 m³ of air!**

Specific air consumption

Furthermore, there is the so-called **specific air consumption.** This is the mass of air that is supplied to the engine per power output- and per time unit.

$$\text{L specific} = \lambda_{total} \times L_{theo} \times be$$

Diesel engines use large amounts of air in the combustion process.

A well ventilated engine room is the most important requirement. Shown here an inlet grill of an emergency diesel generating set.

Specific fuel consumption

The **specific fuel consumption** is the amount of fuel that is supplied per power output- and per time unit.
This is usually provided in grams per kilowatt per hour.
L specific for four-stroke engines is approximately: 2 kg/MJ/sec and for two-stroke-crosshead engines approximately 2.5 kg/MJ/sec.

Examples

1 **Wärtsilä Vasaa 6 R 32** – four-stroke-trunk piston engine – six cylinder in-line – fuel H.F.O.
Shaft power 2220 kW at 720 rev/min. **Engine category III.**
Amount of air 4.6 kg/sec. The standard specific mass of air is taken as 1.276 kg per m³ at an atmospheric pressure of 1013 millibars at an air temperature of 15 °C.

This is therefore $\frac{4.6}{1.276}$ m³ per second or
3.6 × 3600 = 12,978 m³ per hour!

Fuel consumption 186 gram/kWh
So: 412.9 kg per hour
So: $\frac{186}{1000} \times 2220 = 412.9$ kg per hour
Per **kg fuel** is practically
$\frac{4,6 \times 3600}{412.9}$ = **40,1 kg** of air supplied.
This is in accordance with the theory!

2 **MAN–B&W MC – C** – two-stroke-crosshead engine – twelve cylinder in-line – fuel H.F.O.
Shaft power 68,520 kW at 104 rev/min.
Engine category IV
Amount of air 180,7 kg/sec.

So this is $\frac{180.7}{1.276}$ =141.6 m³ per second or
141.6 × 3600 = 50,976 m³ per hour!

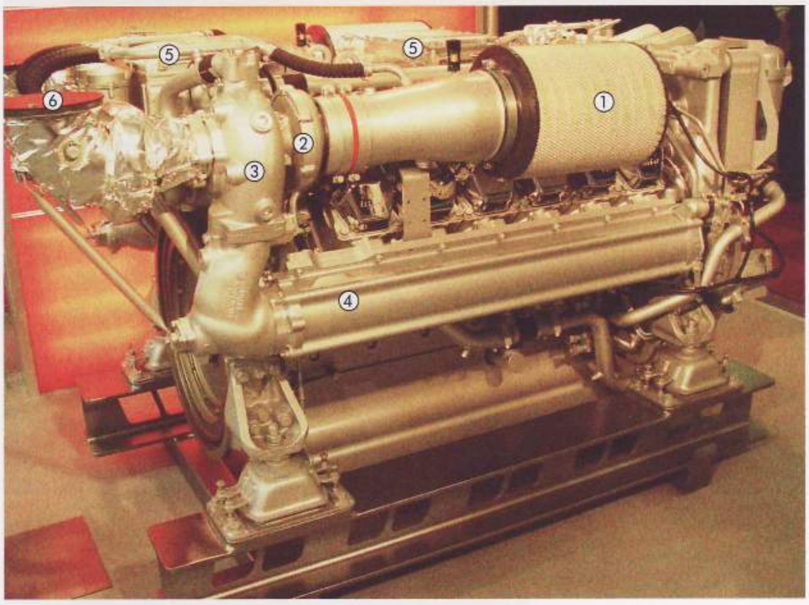
Fuel consumption 171 g per kWh
So $\frac{171 \times 68,520}{1000}$ = 11,717 kg per hour.
Per kilogram fuel $\frac{180,7 \times 3600}{11.717}$ = 55,6 kg of air is practically supplied!
This is in accordance with the theory!

12.3 Air supply to the engine

There are two ways in which air may be supplied:

1 **Natural aspiration**
This can only be applied to four-stroke engines; the piston moves down during the inlet stroke, the inlet valves are opened and the air flows into the cylinder as a result of the vacuum created.
2 **Turbo charging**
Here the air is compressed by a pump and subsequently flows into the cylinder:
- in two-stroke engines via the inlet ports;
- in four-stroke engines via the inlet valves.

Turbo charging generally occurs by means of a turbo blower, which is driven by the flow of exhaust gases. However, there are also mechanical scavenging pumps which are, for instance, driven by the crank shaft of the diesel engine.



▼
A high-speed four-stroke V-engine with supercharging and an air cooler.

1 *air-inlet filter*
2 *air compressor*
3 *exhaust- gas turbine*
4 *exhaust- gas pipe from the cylinders*
5 *air cooler*
6 *exhaust gas pipe to the exterior environment*

◄
A very small four-stroke one cylinder diesel power generating set with natural aspiration for the power process and air cooling with a ventilator on the fly-wheel for engine-cylinder cooling.

12.3.1 Naturally aspirated air in four-stroke engines

The air is drawn in by the vacuum created during the inlet stroke.
The air flows through the air filter and via one or two inlet valves into the cylinder. Today, the system is only applied in small engines with a relatively low power output.
- The power output per litre stroke volume is low.
- The mean effective pressure is low.
- It follows that the mechanical load of these engines is also low.
- The engine has a simple construction; it has no turbo blower with pipes and an inter cooler.

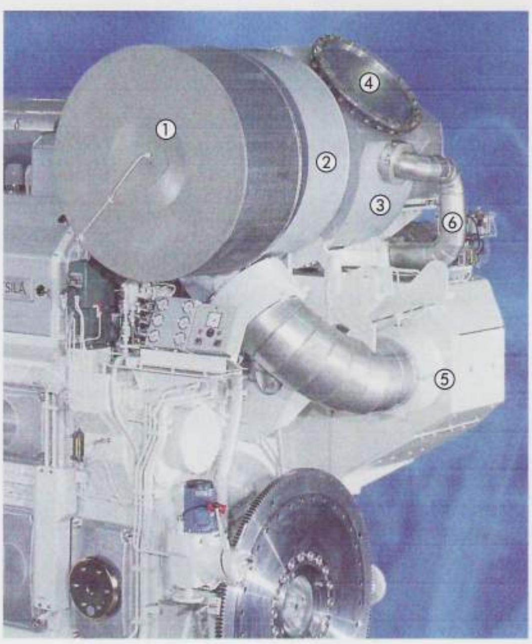
▼
Small engines with a shaft power lower than 100 kW sometimes have no supercharging.

1 air inlet filter

▶
A turbocharger for a larger four-stroke engine.

In this case, a Wärtsilä line W46, medium-speed engine on heavy fuel oil.

1 air- inlet filter
2 air compressor
3 exhaust- gas turbine
4 exhaust to funnel
5 air cooler
6 'waste-gate', a by-pass valve for exhaust gases

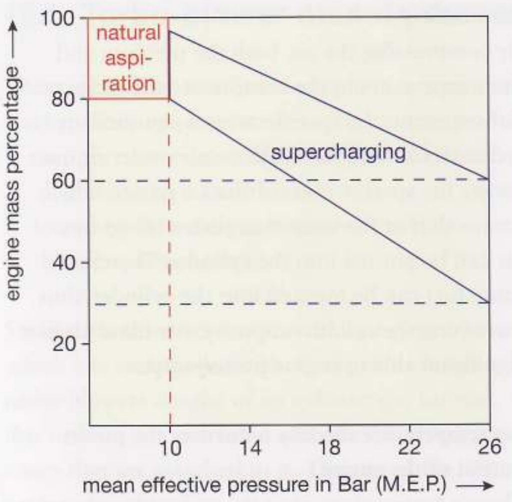


12.3.2 Turbo charging for four-stroke- and two-stroke engines

General

Today almost all larger industrial diesel engines are equipped with turbo charger.
Air is pushed into the cylinder through an over pressure. The compressed air has a higher specific mass which means that a larger air mass and consequently a larger fuel mass is inserted per fresh air charge. This allows for a capacity increase of as much as 200% of the natural aspiration capacity.

Advantages of turbo charging:
- The power to weight ratio of the engine is significantly increased.
- The torque at partial load is significantly increased.
- The specific fuel consumption at partial load is considerably reduced.
- The power output is significantly increased.
- At a great height both the power output and the specific fuel consumption stay almost constant.
- In natural aspiration at a great height both the power output and the torque reduce significantly and fuel consumption increases.
- The total efficiency is higher at all loads.
- The fuel consumption is lower at all loads.
- The sound levels with turbo charging are lower over the entire frequency range.
- The power output to volume ratio of the engine is more favourable. An engine with turbo charging is much smaller than an engine with natural aspiration with a similar power output.
- The combustion process of turbo charging is cleaner and therefore the emission levels lower.
- In combination with direct injection and the common rail systems, the turbo blower is indispensable.
- Turbo blowers can be fitted with 'waste-gates' and a variable geometry for the gas turbine with which the correct amount of air can be tuned to the engine load. In smaller engines with a fluctuating load this can be matched very precisely.



Left graph.

Horizontal: The mean effective pressure in bar.

M.E.P. = Mean Effective Pressure.

Vertical: The engine mass shaft percentage.

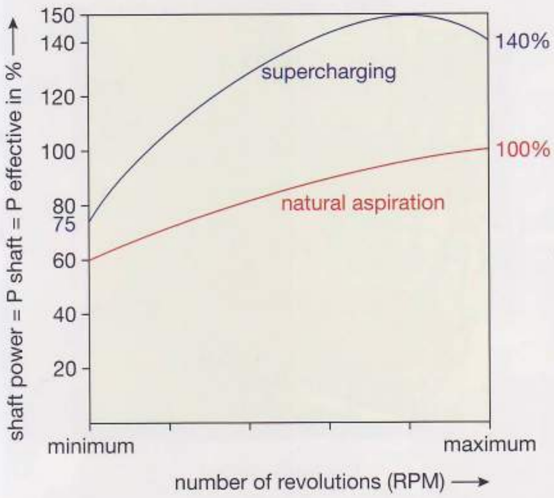
Dependent on the load the suction charging varies between 100 and 80%.

The maximum mean effective pressure is approximately 10 bar.

With supercharging the mass varies between 60 and 80%.

The maximum mean effective pressure is approximately 26 bar.

Conclusion: an engine at full power load with supercharging has less than half that mass!



Right graph.

The shaft power of supercharging shaft opposed to air charging based on the identical- stroke volume.

Horizontal: The revolutions per minute.

Vertical: The shaft power or effective power.

At the lowest RPM, an engine with supercharging produces more power.

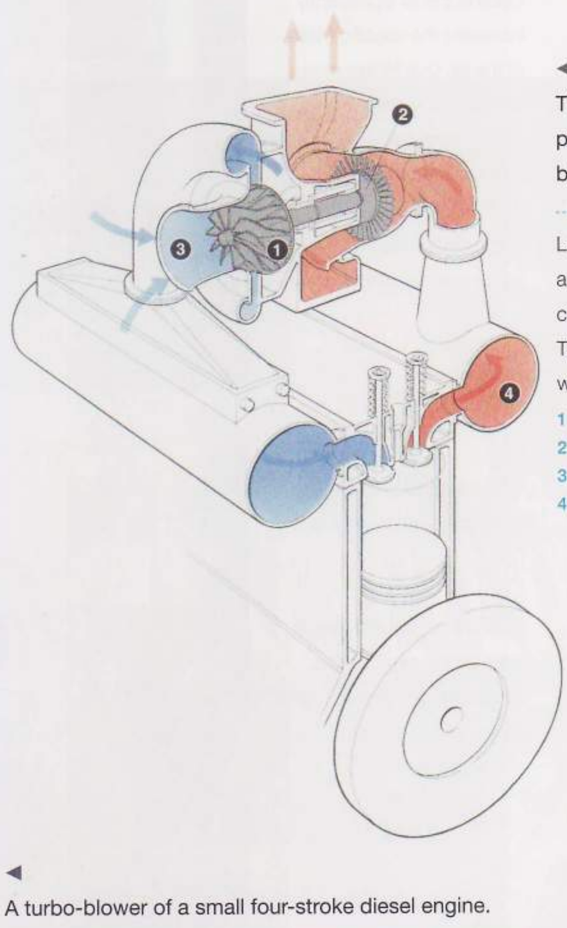
This is 150% close to the maximum revolutions per minute. 50 % shaft power compared to the same engine with air-charging.

◄ The engine performances with and without supercharging based on an identical-stroke volume.

12.4 Principle of turbo-charging

The exhaust gases of internal combustion engines retain a considerable amount of energy in the form of heat- and pressure waves. A heat balance shows that, dependant on the engine type and -size, approximately 25 to 40% of the generated fuel energy is wasted in the exhaust gases.

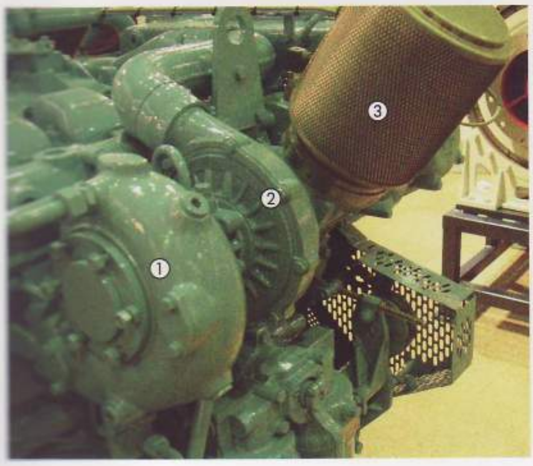
In essence, a turbo blower is a gas turbine driven centrifugal compressor.



◄ The supercharging principle with a turbo-blower.

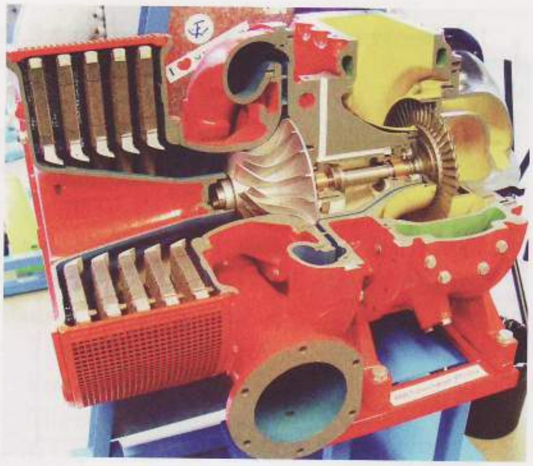
Left; the air is drawn in via an air filter through the centrifugal compressor. This is driven by a single wheel -exhaust- gas turbine.

1 *air compressor*
2 *exhaust-gas turbine*
3 *inlet air-duct*
4 *exhaust gas pipes in the diesel engine*



◄ A turbo-blower of a small four-stroke diesel engine.

From front to back: the exhaust-gas turbine (1), the air compressor (2) and the air filter (3)



▶ A cutaway view of a turbo-blower of manufacturer A.B.B.

Left the air section and right the exhaust gas section

Two-stage turbo charger

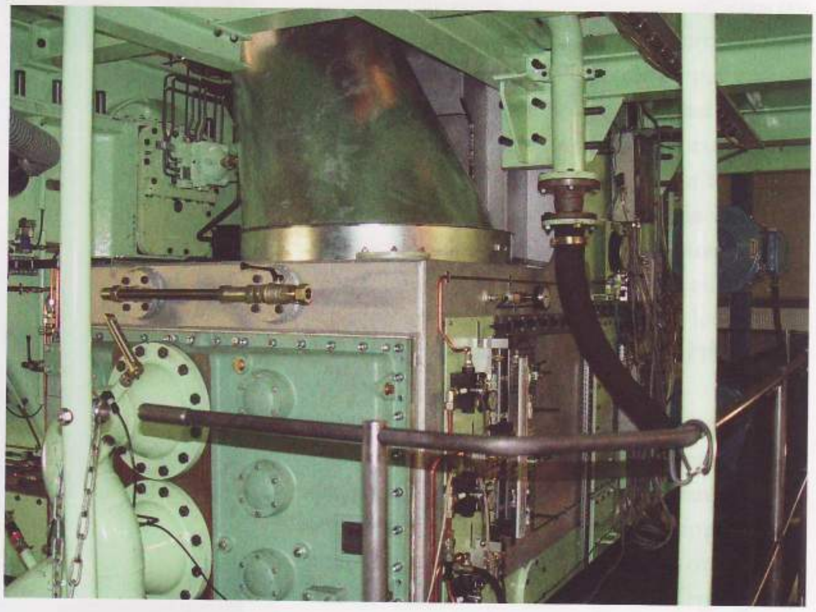
It is also possible to achieve the required inlet scavenging air pressure in two stages; this can be accomplished by placing two relatively simple turbo blowers in series.

Air cooling

By compressing the air, both the pressure and the temperature of the compressed air increase. Subsequently, the specific mass is significantly reduced. Cooling the air in an air cooler or inter cooler the specific mass of the air grows, which means that in the same time, extra kilograms of air can be pushed into the cylinder. Therefore, more fuel can be injected into the cylinder, thus increasing the cylinder capacity. **Air mass plays a significant role in engine power output.**

Air temperature directly influences the power output of the engine!

The air temperature after the turbo blower varies according to the engine type and load from 80 to 200 °C. After the inter cooler this range is 35 to 85 °C.



▶ An air cooler for a large two-stroke crosshead engine.

Cooling the air significantly increases the specific mass of the air. Due to this more fuel can be injected in the cylinder and therefore the engine power increases. Conclusion: lower air inlet temperatures can lead to higher power outputs.

▶ A turbo-blower of an eight cylinder V-engine of MTU.

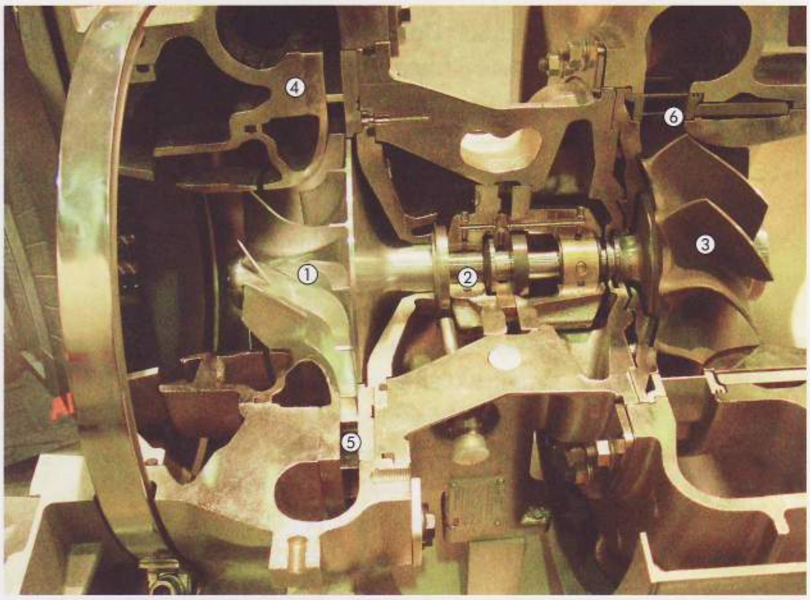
In this case with a 'waste gate' on the exhaust-gas turbine and a 'by-pass' valve on the air compressor.

1 *'waste gate'*
2 *'by-pass' valve*

12.5 Turbo-blower manufacturers

There are a large number of suppliers serving this market. They can be placed in two categories:
1 suppliers who both design and manufacture engines as well as turbo blowers;
2 suppliers who only manufacture turbo blowers, not engines.

Small turbo blowers have an exhaust gas turbine which has an impeller with vanes, while **larger turbo blowers** consist of an exhaust gas turbine, this turbine is constructed of a rotor with separate vanes that are attached to it. They resemble the rotors of gas turbines.



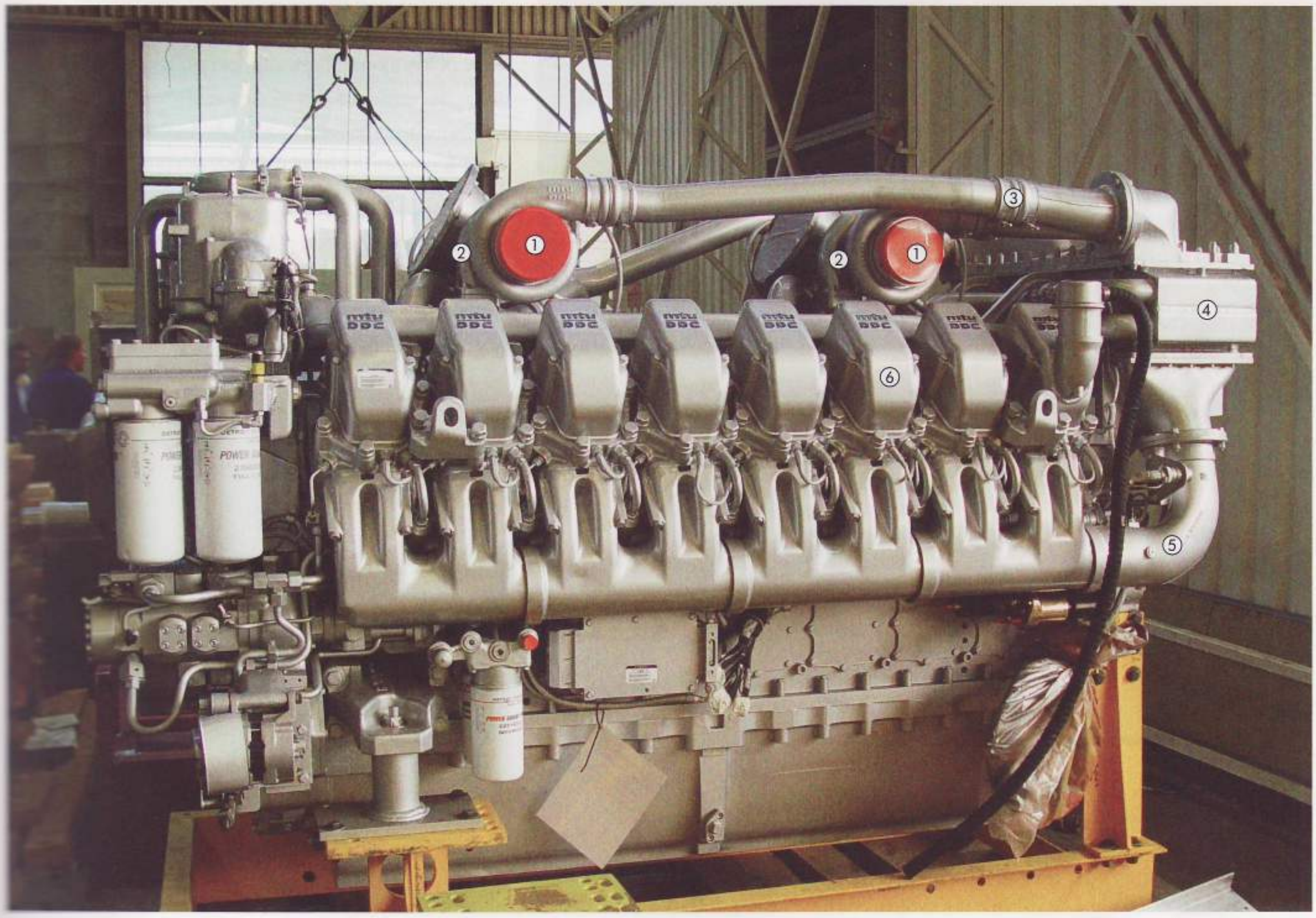
▲
A cut-away view of a modern turbo-blower.

1 *air- compressor fan*
2 *shaft with plain bearings*
3 *exhaust-gas turbine*
4 *blower casing*
5 *diffuser of the compressor section*
6 *exhaust-gas turbine nozzle ring*

▼
A high- speed four- stroke sixteen cylinder MTU-diesel engine with supercharging and intercooling – Series 2000.

Each 'bank' of this V-engine has its own turbo-blower.

1 *position for the air filters*
2 *air compressor*
3 *pressure line from the air compressor to the air cooler*
4 *air cooler*
5 *cooled inlet air to the two inlet valves of every cylinder*
6 *cylinder*



▼

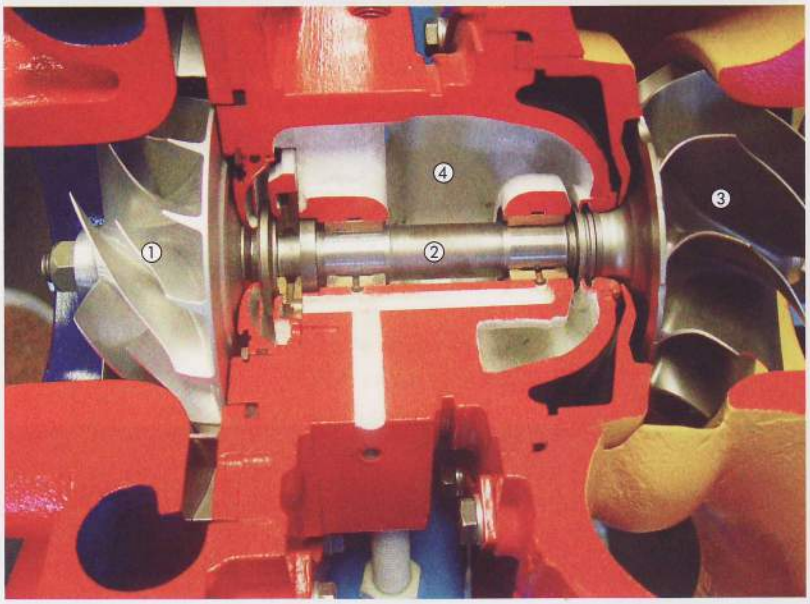
The rotor of a small turbo-blower often consists of a radial air compressor and a radial exhaust-gas turbine.

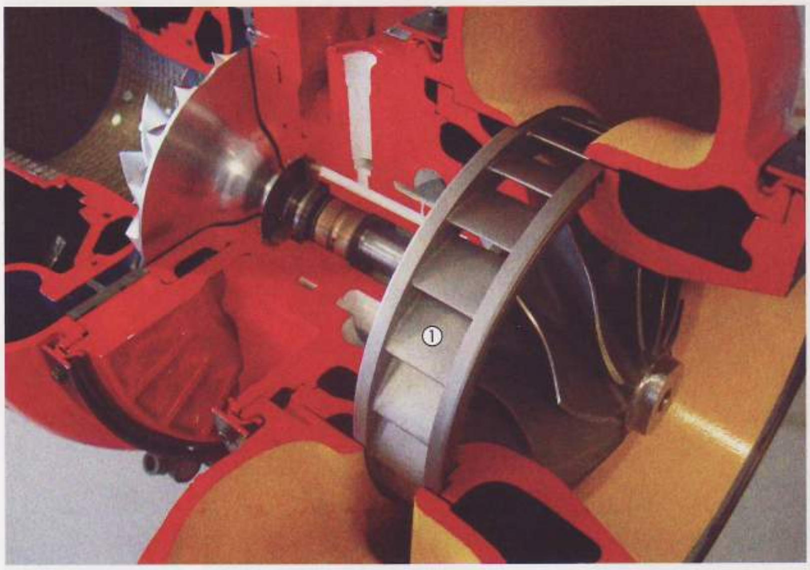
The gas flow is from the inside to the outside along the shaft.

1 *air compressor*
2 *turbine shaft*
3 *exhaust-gas turbine*
4 *lubricating-oil section*

12.5.1 Construction of turbo blowers for small diesel engines – Engine category I and II – Description I – small turbo blower

The rotor consists of an exhaust gas turbine with a fan manufactured in one piece and a centrifugal air compressor. The shaft is short and thick and is supported by two plain bearings and an axial bearing on the air compressor side. Lubrication is achieved using the engine's lubricating system, even in the very large turbo blowers found in two-stroke-cross head engines. The housings are not cooled and therefore made of high-grade heat resistant material. The turbine housings are externally insulated for fire safety.

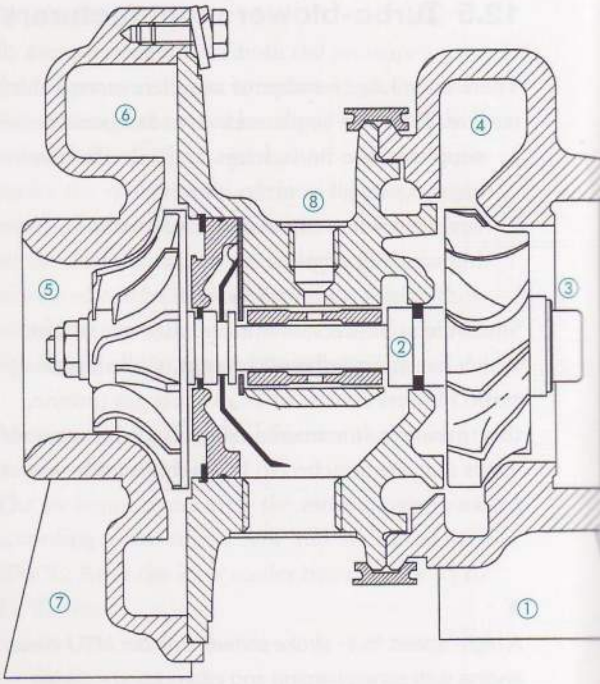




▲

A cut-away radial turbo-blower.

The inlet-segment of the exhaust gases turbine is clearly visible in the foreground (1).



▲

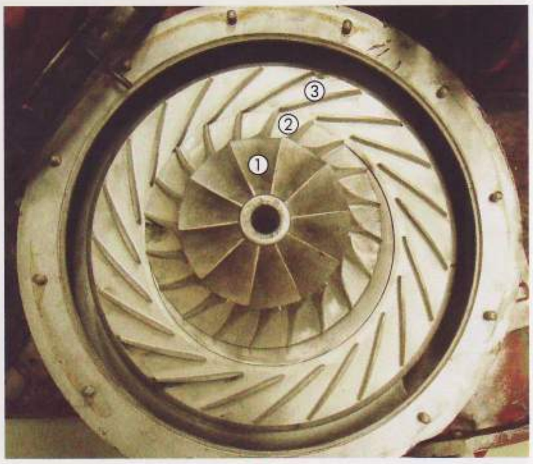
A cross-section of a small turbo-blower.

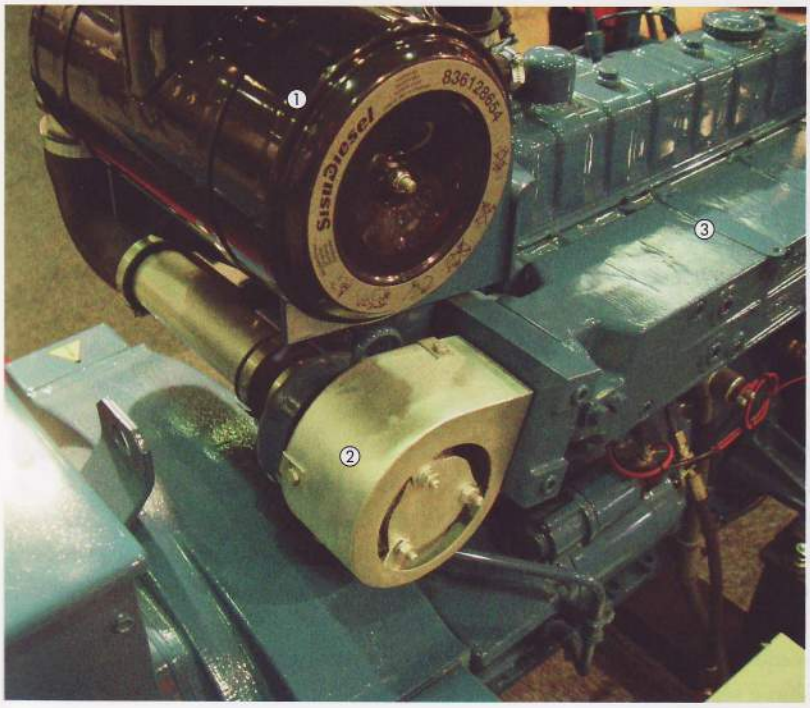
1 *inlet-turbine*
2 *shaft*
3 *outlet-turbine*
4 *turbine casing*
5 *inlet-compressor*
6 *compressor casing*
7 *outlet-compressor*
8 *oil supply*

▼

View of the air-compressor.

1 *first row of blades of air-compressor*
2 *second row of blades of air-compressor*
3 *diffusor: this stationary section converts the high air velocity, generated in the rotating impeller into pressure. The output channels of the diffusor effect a decrease in air-velocity and an increase in air-pressure.*





◄

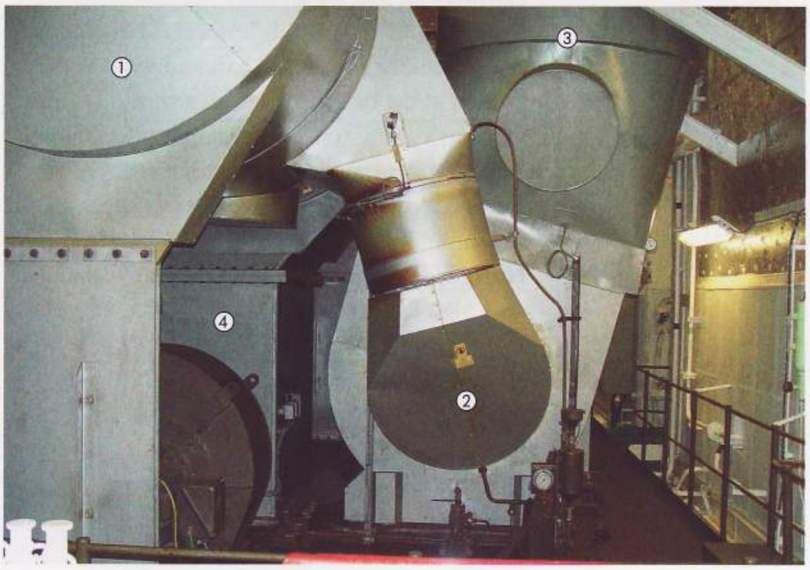
A turbo-blower of a small diesel engine.

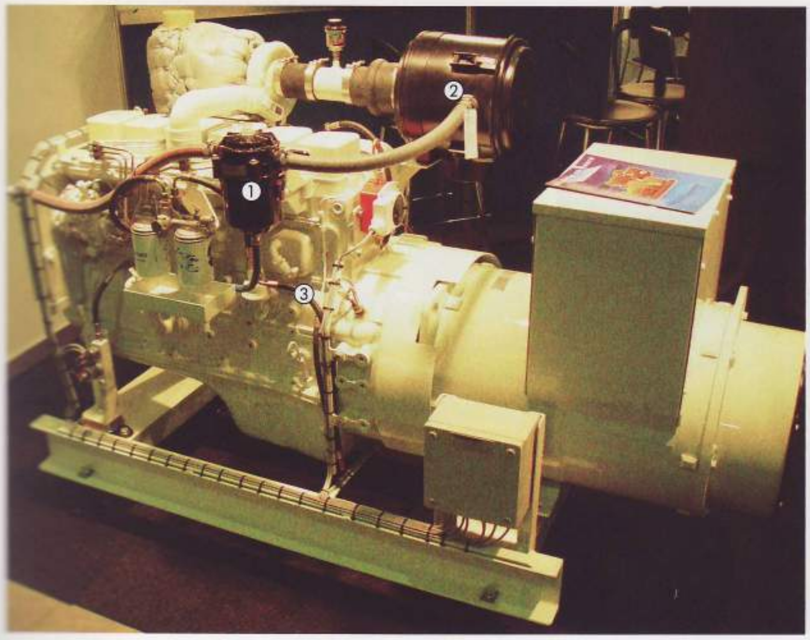
There is no air cooler.

1 *air filter*
2 *turbo-blower*
3 *engine block*

These small turbines rotate at a speed of 40,000 tot 80,000 revolutions per minute.

The exhaust gases are lead to turbine housing via the exhaust manifold. The inlet casing diameter inside the turbine housing becomes increasingly narrow which increases the speed of the exhaust gases. Due to the spiral shape the hot gases surrounding the turbine wheel are distributed and hit the fan blades, subsequently the fan starts to rotate.
The turbine housing clearances determine the rotation speed of the exhaust gas turbine. Deciding on the turbine clearance size is dependent on the cylinder content, the number of rotations and the maximum power output of the engine.





◄

A relatively small diesel genset with turbo-charging.

An oil separator is placed between the sump and the inlet- air filter for sump venting. The lubricating-oil fumes travel via the turbo-blower back to the engine and are combusted.

1 *oil separator*
2 *connection to the air filter*
3 *liquid drain*

▲

A very large turbo-blower for a two-stroke crosshead engine.

1 *exhaust-gas pipes from the cylinders*
2 *exhaust-gas inlet to the turbine*
3 *exhaust-gas discharge to the outside air via an exhaust-gas boiler and silencer*
4 *inlet air cooler*

Large two-stroke-crosshead engines have turbo blowers with a rotor diameter of 60 to 100 cm with a RPM of 5000 to 6000.
Small four-stroke-trunk piston engines have turbo blowers with a rotor diameter of 5 to 20 cm with a RPM of 80,000 to 120,000.
So large engines have large turbo blowers with relatively low RPM's, whereas small engines have small turbo blowers with relatively high RPM's.

The centrifugal compressor draws in fresh air via a large suction filter in order to reduce the pressure loss in the filter to a minimum, the air enters the compressor wheel where it is accelerated to high speeds. As a result of the increasing high velocity air passing through the blades in the diffuser, the velocity energy is converted into pressure. An over pressure of up to 5 bars can be achieved!
The short and thick shaft is equipped with two ball- or plain bearings and has a seal on the exhaust gas- and the air side.
Engine lubricating oil lubricates the shaft and is supplied to the top of the shaft and flows at the bottom of the blower back into engine oil sump.
Obviously the air output depends on the amount exhaust gases supplied to the exhaust gas turbine.
There is a capacity balance between the gas turbine and the air compressor.
High capacity - plentiful exhaust gases – abundant air – high RPM.

12.5.2 Construction of turbo blower for large diesel engines – Engine category III and IV

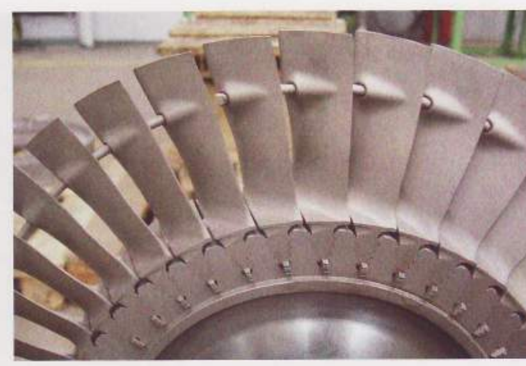
Description: Large turbo blower

At the present, there is a trend in the construction of large turbo blowers where they start to resemble the construction of much smaller turbo blowers. The construction of the exhaust gas turbine is the major difference. This consists of a wheel which has separate blades on all sides. The construction resembles the construction of the gas- and steam turbines.
Here too the engine's lubricating system is applied to lubricate and cool the turbine shaft and bearing. The shaft seals are a labyrinth construction. The lubricating oil is normally supplied from the main engine. Sleeve bearings are increasingly often applied. Occasionally, the turbine shaft housing is cooled with a separate cooling water system.

►

Renewing the labyrinth seals on the exhaust-gas turbine side.

Several labyrinth segmented have been placed closed to the wheel.



▲

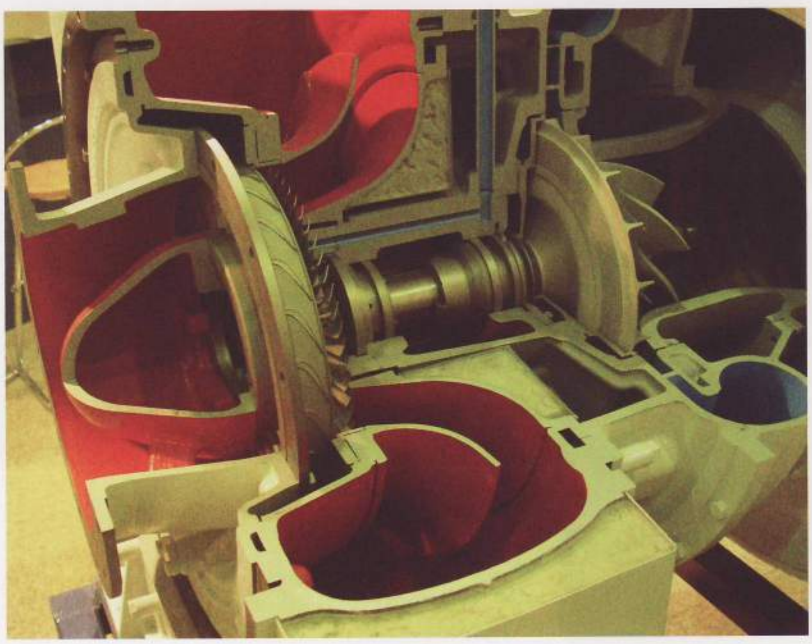
The blade wheel of the exhaust-gas turbine of a four stroke engine.

The support wire between the blades shown here is necessary to prevent vibrations; these occur especially in four-stroke engines caused by pressure surges.
Note; the blade-foot fastening.

▼

A cut-away view of a turbo-blower.

Note; the short thick turbine shaft between both plain bearings.



▲ With two-stroke crosshead engines, the pressure surges are levelled off in the large spacious exhaust-gas pipe and a support wire for the blades is not necessary.

12.6 Capacity curves

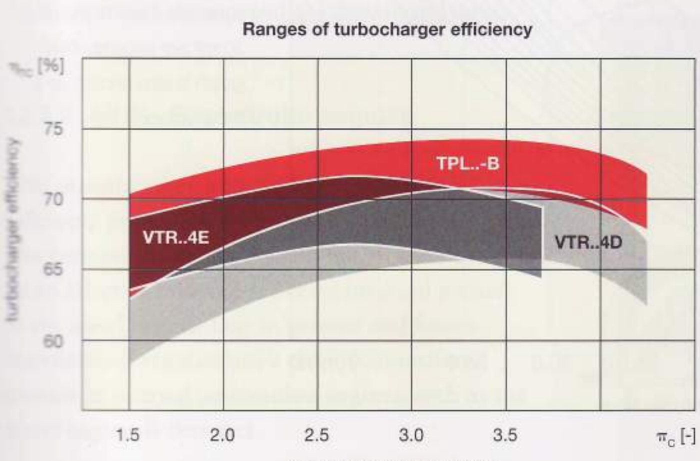
A number of graphs are used that apply to the compressor of a turbo blower.

12.6.1 Compressor efficiency in relation to the compression ratio

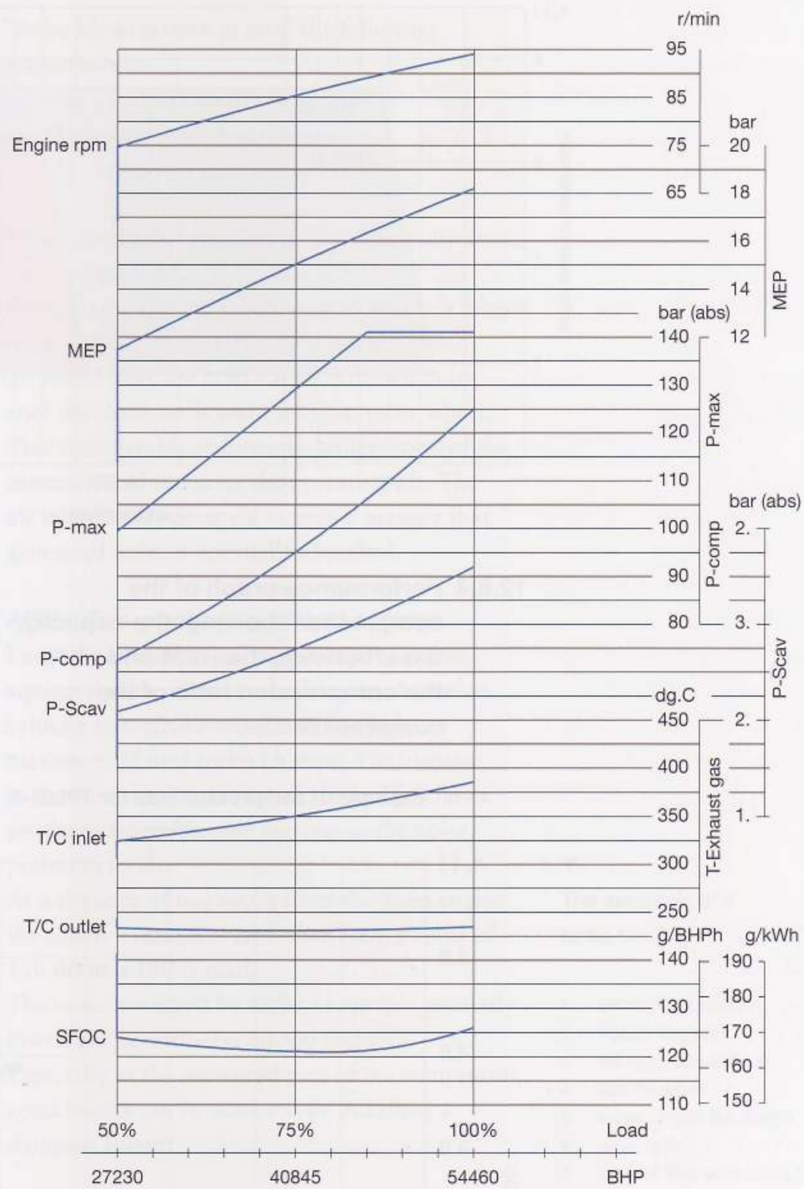
The turbo-blower efficiency with respect to the compression ratio for three types of blowers of manufacturer ABB.

horizontal: the compression ratio
vertical: the compressor efficiency in percentages
Clearly shown is the following:

- VTR – 4E (dark red) has the highest efficiency of 72% with a compression ratio of 2.7;
- VTR – 4D (grey) has the highest efficiency of 71% with a compression ratio of 3.5;
- TPL – B (light red) has the highest efficiency of 74% with a compression ratio of 3.3.



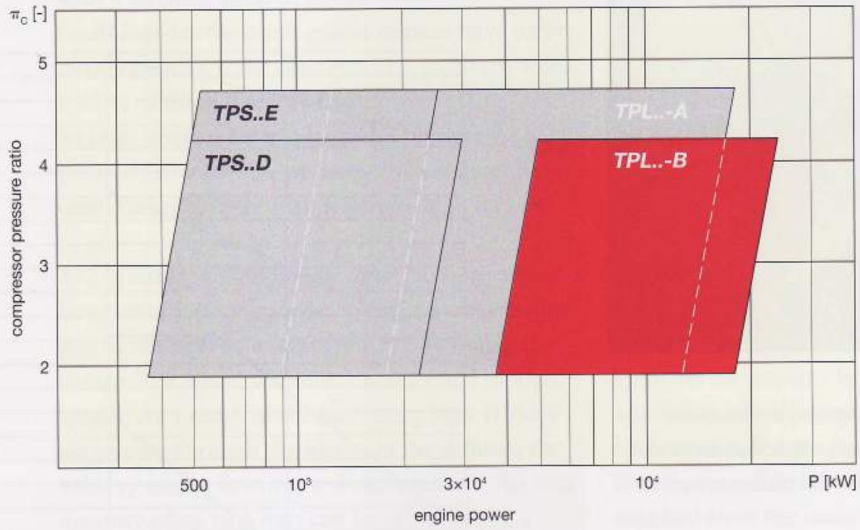
12.6.2 Turbo charger of the engine in relation to the engine load



▲ The performance chart of a MAN–B&W 98 MC two- stroke crosshead engine.

Between an engine load of 50 and 100%, all the data increase. Only the fuel consumption is at 85% the lowest. The scavenging- air pressure increases from 2 to 3,5 bar. The maximum combustion pressure increases to 85% of the engine load and then remains equal.

12.6.3 Compression ratio in relation to engine power output



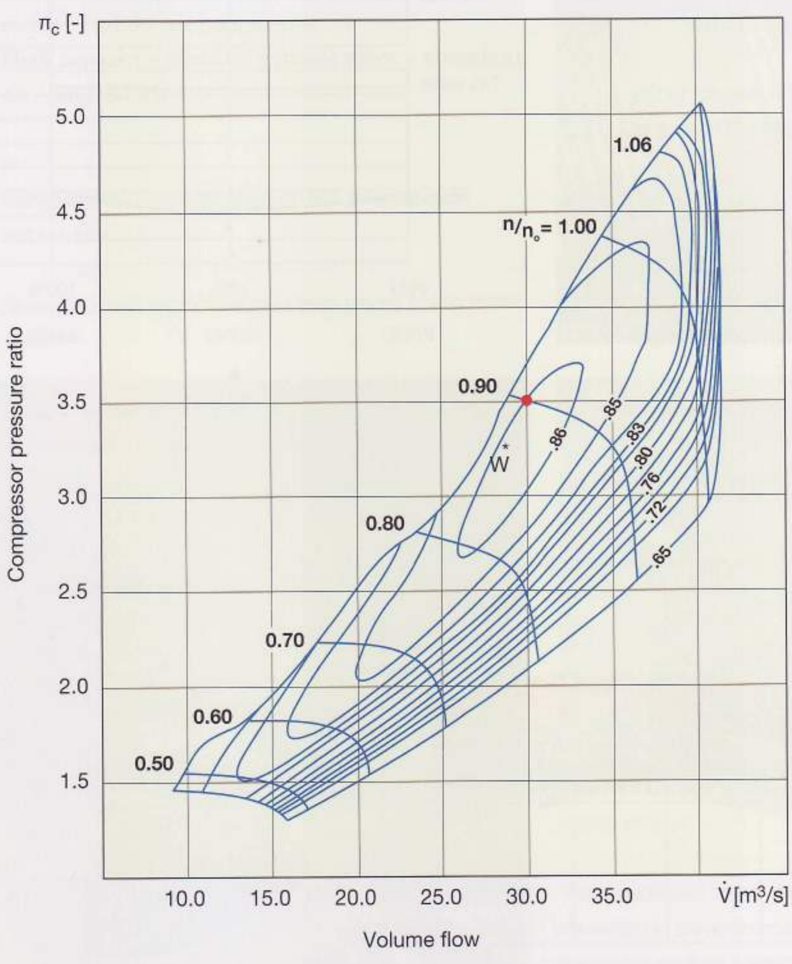
◀

The compression ratio of four types of blowers of manufacturer ABB with respect to the engine power.

horizontal: the engine power in kW
vertical: the compression ratio

12.6.4 Performance graph of the compressor showing: the capacity, the efficiency, the RPM and the compression ratio of the compressor

Example of compressor map for TPL85-B



◀

A delivery characteristic of a turbo-blower.

vertical: the compression ratio
horizontal: the air capacity
curved lines: the turbo-blower speed
closed lines: the turbo-blower efficiency
The optimum operating area is in this case **point W**. The efficiency is 86%. The speed is then approximately 85%. The compression ratio is approximately 3,3 and the suction volume of the air is 30 m³ per second. This graph is also known as a 'MAP'-graph.

12.7 Representation of three turbo-blower manufacturers – development of modern turbo-blowers

12.7.1 Mitsubishi

This Japanese manufacturer has the following principles:

- In two-stroke-crosshead engines the increased cylinder capacity produces a mean effective pressure of 2 MPa or 20 bars.
- In four-stroke-trunk piston engines a standard mean effective pressure of 2.5 to 3.0 MPa or 25 to 30 bars is achieved.
- The pressure of the inlet/scavenging air is approximately 0.4 MPa or 4 bars, this pressure will have to increase.
- The efficiency and compression ratio of the compressor must therefore increase. This entails an increase in the air velocity in the compressor and consequently the noise produced and the axial force in the turbine shaft.

Clearly, turbo blower manufacturers have a large number of focus points in the development of new turbo blowers!

1. Extremely high compressor output at low and high compression rates.
2. Special attention to the axial bearing design.
3. Reduction of noise levels.
4. To increase the operating life of the compressor wheel.
5. To protect the exhaust gas rotor, in particular against corrosion and high temperatures. Presently there are blades manufactured from a 12% chromium alloy.
6. To improve the flow efficiency in the exhaust gas turbine, the blades should be attached such that no lacing wire is required for the prevention of vibrations.
7. To optimise the air- and gas flow in the turbo blower.

12.7.2 ABB – Four-stroke engines

This manufacturer from Switzerland has the following principles:

The exhaust gas emissions and the increase of mean effective pressure are constant focal points in the diesel engine. **Due to present and future international regulations, a clean combustion process in internal combustion engines, such as the diesel engine, is decisive!**

Here the turbo blower has an important role to play.

Turbo blowers have to meet the following requirements:

- Higher compressor efficiency;
- Higher compression rates;
- Higher capacities.

Mechanical reliability and a service friendly design are also required.

Compressor cooling is provided to ensure a longer operating life of the hot engine parts. Cooled air found after the inter cooler is drawn in to cool the shafts seals and the compressor wheel. This considerably reduces the temperature of the materials and increases the operating life. The air inlet filter is designed in such a manner that generated noise is optimally absorbed.

ABB – Two-stroke-crosshead engines

Large diesel engines on board ships are often equipped with several turbo blowers. A twelve cylinder two-stroke-crosshead engine has a maximum of four turbo blowers. The increase in the compression rates and air- and gas flows produces a considerable increase in the noise pollution levels.

At a distance of one meter from the main engine the sound levels generated often have a value of 110 dB at a 100% load.

The noise produced by turbo blowers is generally lower, but nevertheless far too high.

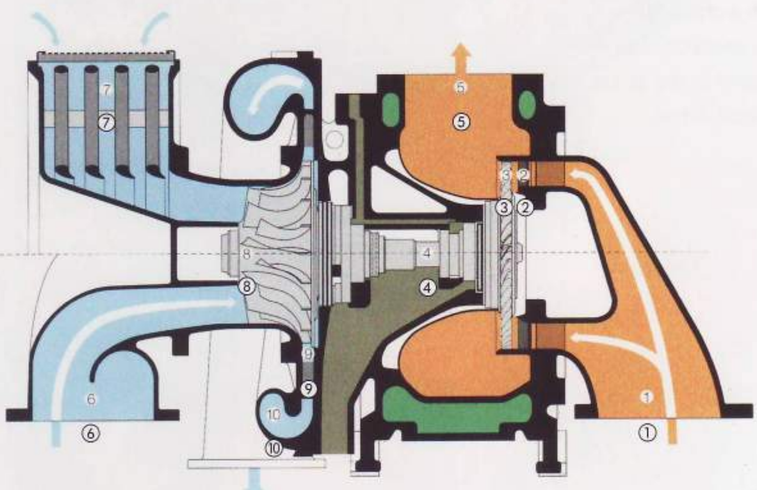
Especially in the pressured part of the compressor, good results can be achieved by installing a damping system.

Alterations to the design of the compressor fan have also led to a reduction in the sound level.

▼

The assembly of a turbo-blower.

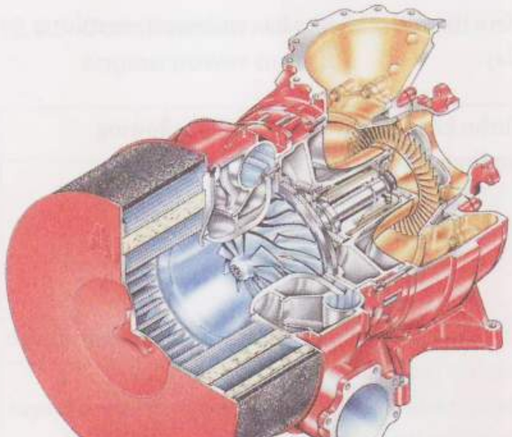
1 *exhaust-gas supply*
2 *nozzle segment*
3 *exhaust-gas turbine*
4 *turbine shaft*
5 *exhaust-gas discharge*
6 *air supply*
7 *air-inlet filter with sound damping*
8 *air-compressor rotor*
9 *diffuser*
10 *snakecasing*

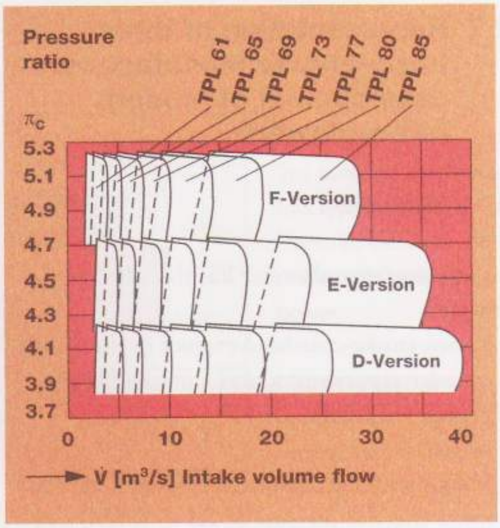


▶

A cross-section of an ABB–TPL turbo-blower with charts of the compression ratio with respect to the air capacity of the various types.

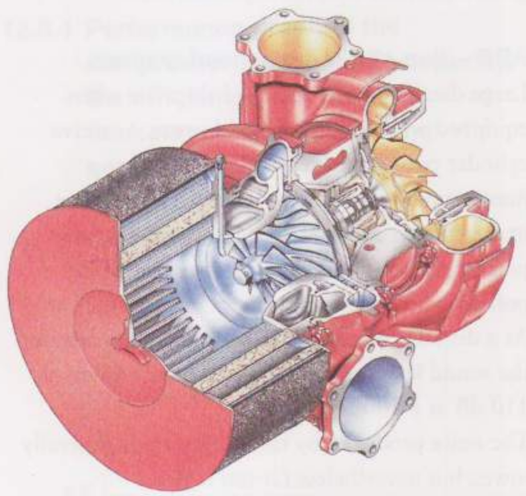
D-version: large capacity, lower compression ratio
E-version: large capacity, higher compression ratio
F-version: smaller capacity with the highest compression ratio

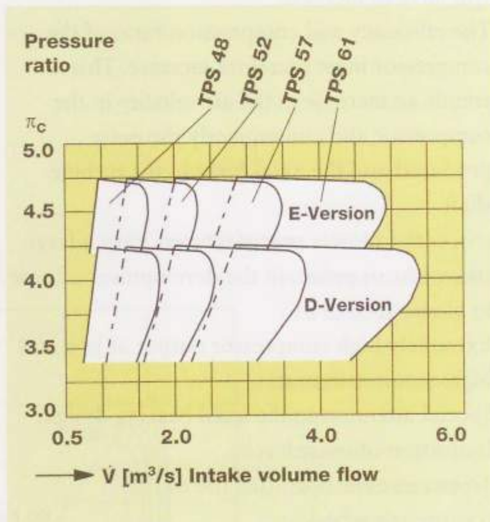




▶

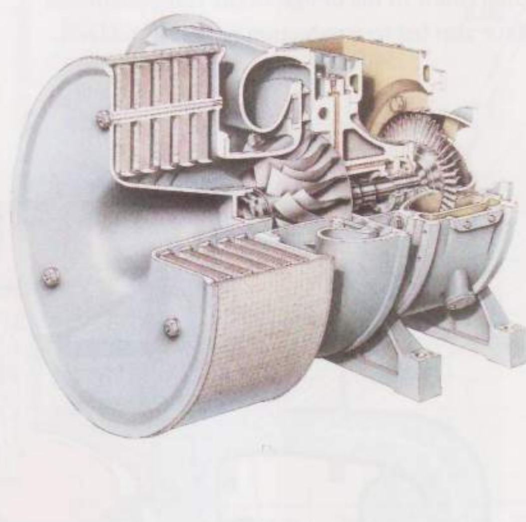
A cross-section of an ABB–TPS turbo-blower with a chart of the compression ratio with respect to the air capacity for the various types.

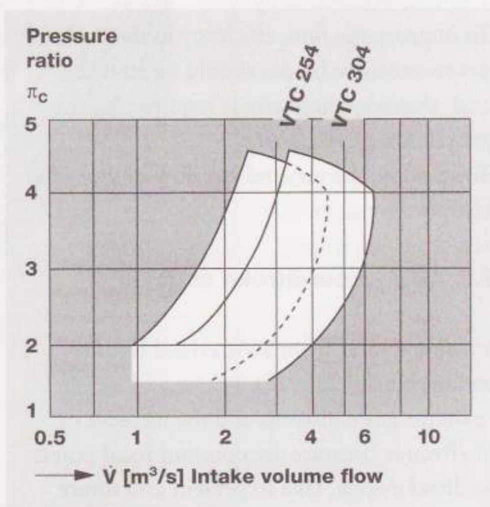


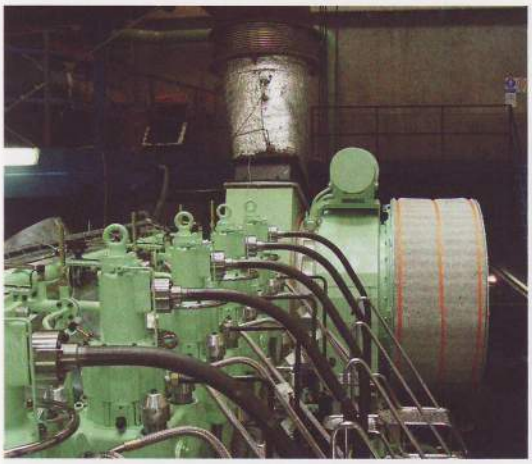


▶

A cross-section of an ABB–VTC turbo-blower with a chart of the compression ratio with respect to the air capacity for both types.

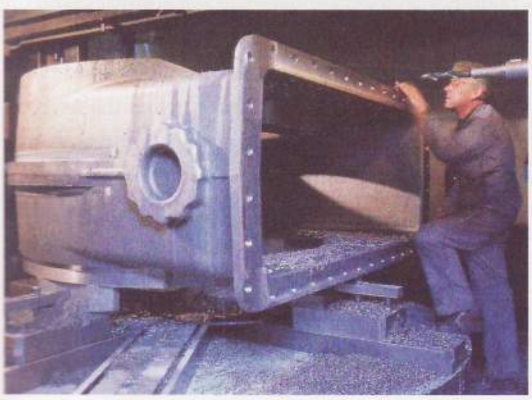






▲
A two-stroke crosshead engine always has an constant pressure system.

At the end of the engine is one turbocharger. A backup lubricating oil tank is placed above the compressor housing so that the rotating rotor of the turbo-blower can always be lubricated even with a diesel engine breakdown.



▲
The manufacture of the housing of the exhaust gas section of the largest turbo-blower of ABB.

12.7.3 MAN–B&W, a German-Danish manufacturer

Here the same requirements apply as with the aforementioned manufacturers:
- To increase the compressor efficiency;
- To reduce the noise levels;
- Service friendly;
- To increase capacity.

Small turbo blowers ABB–TPS D/E series – 500 to 3200 kW diesel engines shaft power – Engine category I and II

Due to the strict regulations with regard to the exhaust gas emissions, a growing number of diesel engines are designed according to the **Miller-principle**; in this system the closing of the valves is either accelerated or delayed. This diminishes the pressures and the temperatures in the cylinder and causes a decrease in the NOx emissions.

Also see Chapter 22, Diesel engine emissions.

In the use of this system, the scavenging air pressure needs to be increased so the combustion process can be adequately cooled; lower process temperatures generate less NOx.

Requirements

- Compression ratios of 4.7 and higher.
- Higher compressor efficiency, lower exhaust gas temperatures and higher engine shaft power.
- Larger capacity operating range.
- A simple, compact, modular and sturdy construction.
- No auxiliary systems, lubrication from the engine lubricating system.
- Corrosion resistant to sulphur found in the fuel.
- If the rotor breaks into pieces at unusually high speeds, the debris should remain inside the housing. To this end special explosion rings are fitted. **Generally, the so-called crack velocity lies at 60% over the maximum RPM.**
- A solid, vibration damped foundation is vital in order to keep the vibration levels low.

12.8 Small turbo-blowers – Engine categories I and II

Small engines are usually provided with **axial** turbines, large engines with **radial** turbines.

▼
A cross-section of a small axial turbo-blower.

▶

A 'waste-gate' of the turbo-blower of an eight-cylinder diesel engine .

1 *'waste-gate'*
2 *turbine section*
3 *air section*

Special provisions in turbo blowers

- waste-gate; an exhaust gas by pass valve
- air bypass valve
- compressor trims*
- ceramics in turbo blowers
- titanium for turbine wheels
- re-circulation valve on compressor
- water cooled turbine housing
- turbo blowers are linked in parallel
- serial connection of turbo blowers

12.8.1 Waste-gate; a by-pass valve for exhaust gases

This valve opens when a certain scavenging air pressure is exceeded; in this way the amount of exhaust gas flow to the turbo blower is controlled and therefore the scavenging air quantity and pressure and ultimately the engine capacity. Under normal conditions, this valve is not activated.

12.8.2 Bypass valve

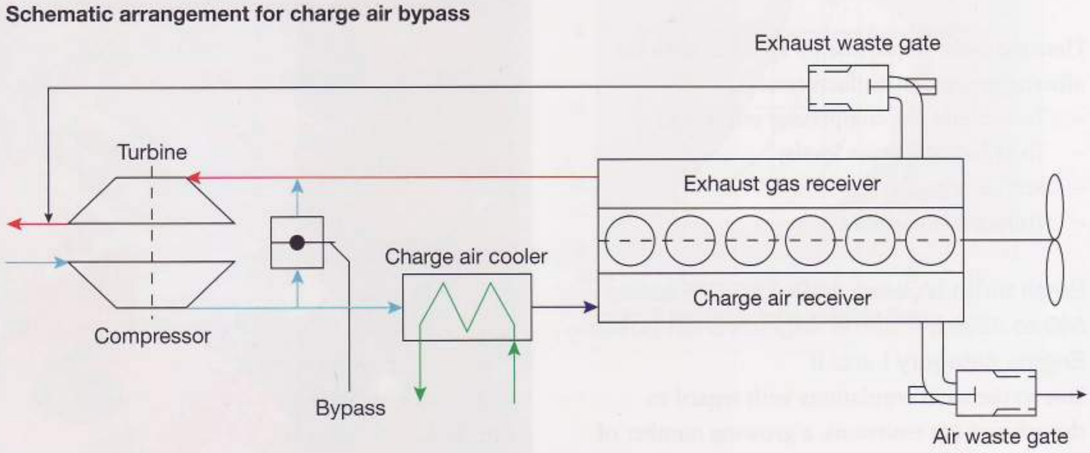
The air bypass valve is located in the air space creating a connection between the compression and suction of the compressor.
This among other things, ensures compressor stability at low loads. Effectively, when the by-pass is open, the air volume flow of the compressor exceeds the air volume flow required for the engine.

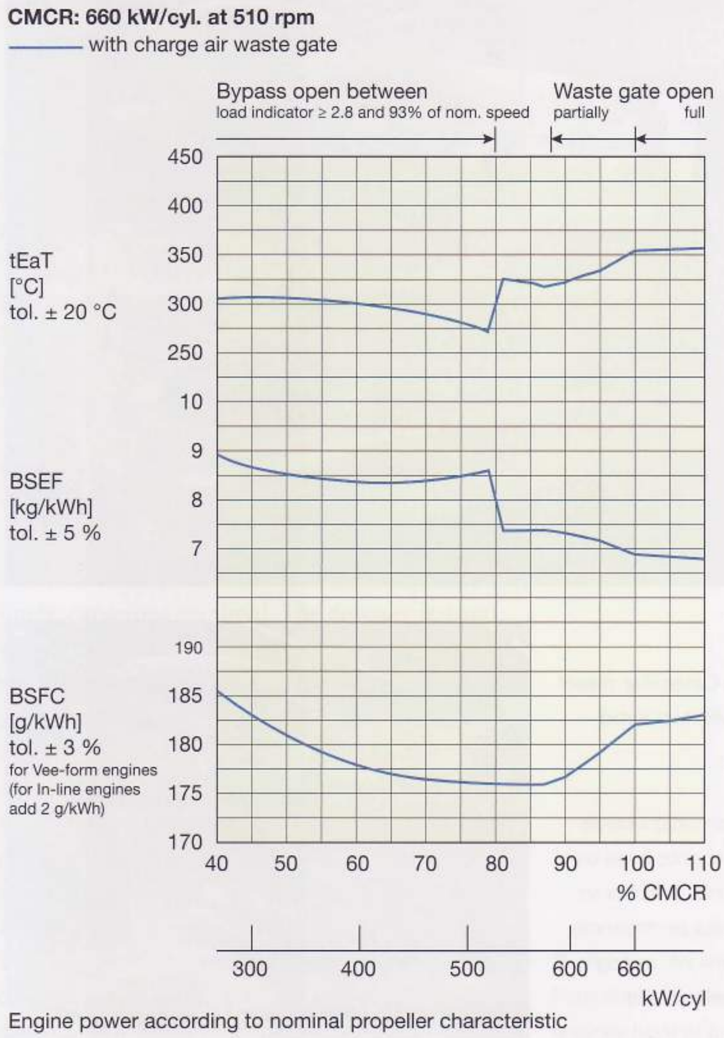
▼

The 'waste-gate' in various designs.

Using a **'waste-gate'** or by-pass valve in the exhaust- gas pipe, the amount of air to the engine is limited by reducing the flow of the exhaust gases to the turbine. Due to this the engine power is limited. A by-pass valve in the air section after the air compressor can limit the amount of air to the engine .
This is also possible with the 'air waste-gate'.

Schematic arrangement for charge air bypass





▶
The use of the 'bypass'-valve and the 'waste-gate' with different power outputs.

tEaT = temperature after the exhaust-gas turbine
BSEF = specific exhaust- gas mass
BSFC = specific fuel consumption

The specific fuel consumption BSFC is the lowest; approximately 75% power. This is in the region of the maximum power output for ship propulsion, viz. 80 to 85%. Due to the increased volume, the filling pressure to the engine increases at partial loads; this is good for scavenging!
Bypass: in air section
Waste gate: in exhaust gas section

12.8.3 Compressor trims

Small turbo blowers are typically cast with a single housing. By modifying the compressor wheel and applying various 'trims', the shape of the wheel is altered to the extent that varying capacities are generated.
The rough cast light metal wheel has identical basic measurements for an entire series. The finishing machining process gives the distinctive properties. Generally, one rough cast wheel can produce twenty different 'trims'. In these radial exhaust gases turbines, 'trims' are applied.

12.8.4 Ceramic in turbo blowers

Advantages of the use of ceramic for the turbine are:
- Resistant to high temperatures, approximately 1250 °C;
- Low coefficient of expansion; merely 1/5 of metals;
- Low weight, rapid wheel acceleration;
- Thinner housing is possible. Less mass if the light turbine wheel cracks apart.

The disadvantages are:
- Difficult to attach to the turbine shaft. Welding is not an option;
- Ceramic is brittle;
- Soot particles can damage ceramic;
- X-ray quality control is impossible.

12.8.5 Titanium for turbine wheels

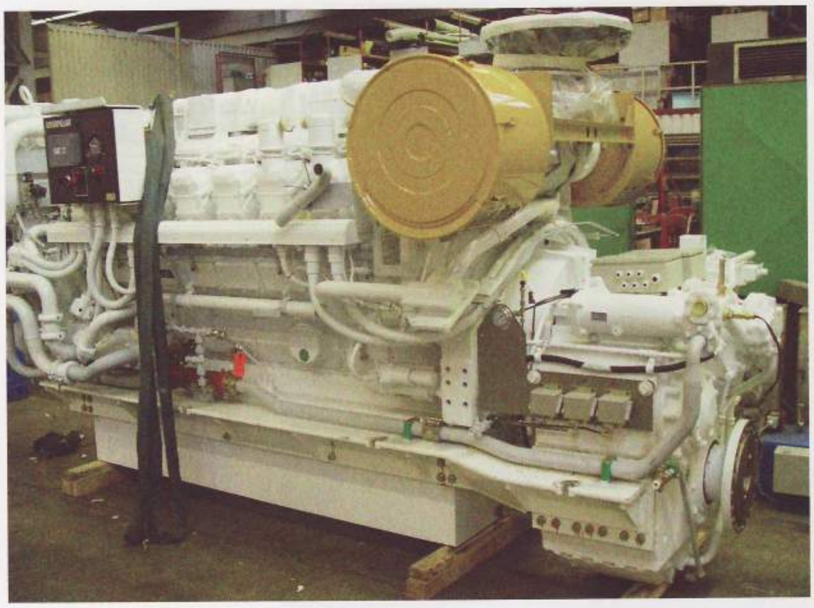
Titanium has a higher resistance to the corrosive properties of soot particles than ceramic. The traditional nickel alloys which are applied in the small to very small turbo blowers weigh twice as much as titanium. However, titanium is difficult to cast and very expensive. At present, it is not generally applied.

12.8.6 Re-circulation valve on compressor

Serves mainly to keep a high rotor speed at various engine loads. Therefore, when extra power output is required, the rotor velocity is still high. By opening the re-circulation valve, the turbo blower load is reduced and its number of revolutions increased.

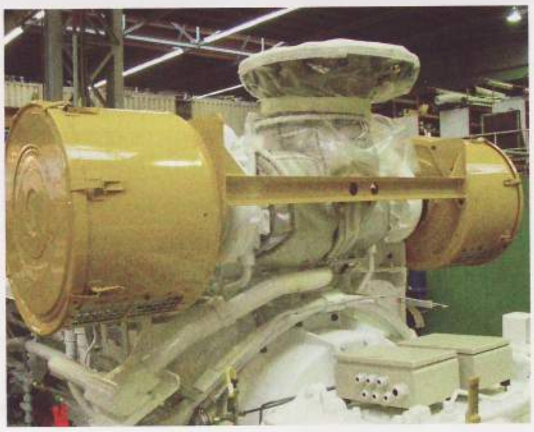
12.8.7 Water cooled turbine housing

These small turbo blowers are also provided with water cooling. This cools the housing around the rotor shaft and decreases the radiant heat near the turbo blower.



◄
A Caterpillar-diesel engine in V-shape with two turbo-blowers.

Right the attached shaft reduction gear of a luxury motor yacht.



►
A V-type Caterpillar diesel engine with two turbo-blowers.

Mostly both turbo-blowers of V-type engines press on a common inlet air-space so that air pressure differences between the left- and right engine banks are not possible.

12.8.8 Turbo blowers connected in parallel

In V-engines it is fairly common that the left- and right bank have individual turbo blowers in which the individual compressors compress the air led from the same air inlet manifold. This inlet manifold is placed at the centre of the engine with the central exhaust pipes on the either side. The advantage of this system is that the charge air pressure of each cylinder and each turbo bank is equal and therefore the power output per cylinder and bank is equally distributed compared to separate turbo blower systems.

12.8.9 Serially linked turbo blowers

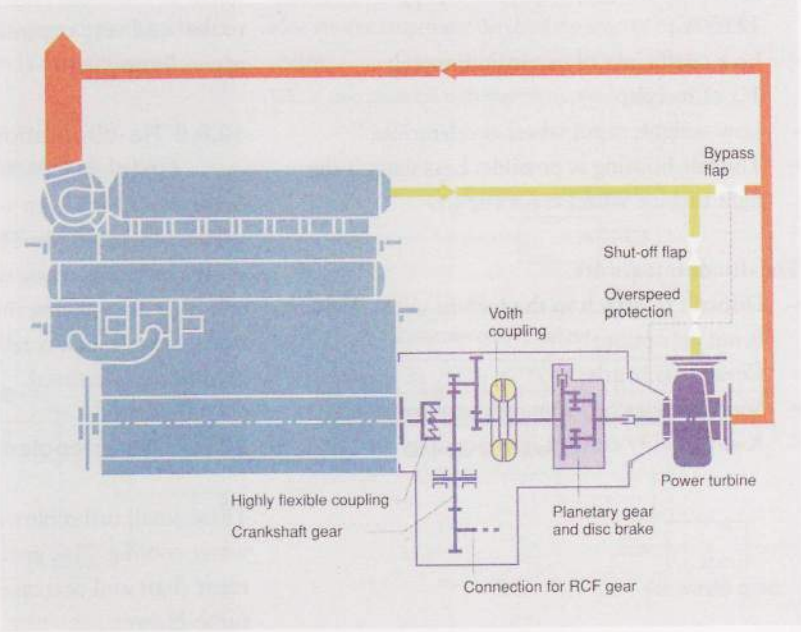
This is occasionally seen in diesel engines of category II. The efficiency of standard turbo blowers is too low when the charged air pressure is over 3 bars. Two standard turbo blowers are then placed in series. The exhaust gases first pass the high pressure- and then the low pressure turbo blower. An inter cooler is either placed after the high-pressure turbo blower, or two intercoolers, where the second inter cooler is placed after the low-pressure turbo blower.

12.9 Supercharger with a separate power turbine

This has been applied sporadically to large two-stroke-cross head engines for some years. At full load not all the kinetic energy present in the exhaust gases is required to drive the turbo

►
A schematic diagram of a power turbine driven by the surplus engine exhaust gases.

This power turbine drives the engine using a planetary gear box and a coupling. This system is seldom used as the power gain of a few percent does not compensate the high investment costs.
The fuel savings at full load are approximately 3%.



blowers. The residual energy is used for a power turbine. This energy is converted into mechanical power and supplied via special reduction gearing to the crank shaft. In this case, the power turbine is always connected in parallel.

This system receives a good deal of interest for small engines – **Category I and II** –. In these categories, both exhaust gas turbines are arranged in series.
First the normal turbo blower is driven and then the power turbine. At full load this four-stroke engine can save approximately 10%.
It should be taken into account that this type of power turbine driven requires a large investment cost!

12.10 Air supply in four-stroke engines

In the four-stroke process the exhaust gases must be expelled from the cylinder at the correct time.
- If the exhaust valves open too early, this results in power loss due to a reduced piston stroke.
- If the exhaust valves open too late, this results in an increase in exhaust gas pressure on the piston during the upward stroke at the expense of power output.
- The inlet valves open before the end of the exhaust stroke in order that there is a sufficiently open passage for the incoming fresh air at the beginning of the inlet stroke.
- The exhaust valves remain open to the end of the exhaust stroke and only close at the commencement of the inlet stroke.
- Therefore the exhaust- and the inlet stroke time periods overlap. This overlap, the so-called exhaust air scavenging, takes place from 25 to 75 crank degrees for naturally aspirated diesel engines and from 90 to 130 crank degrees for engines with a super charger system.
- The higher the RPM, the greater the overlap. The fresh air requires sufficient time to adequately cool the hot parts and discharge the remaining exhaust gases.
- A naturally aspirated four-stroke engine can always maintain its own air supply using the piston, which functions as a scavenging pump.
- A disadvantage of naturally aspirated engines is the low initial compression pressure, this is below the atmospheric pressure caused by the flow resistance of the air filter, the air ducting and the inlet valves.
- The cylinder capacity is limited as it is entirely dependent on the mass of air that is drawn into the cylinder per cycle.

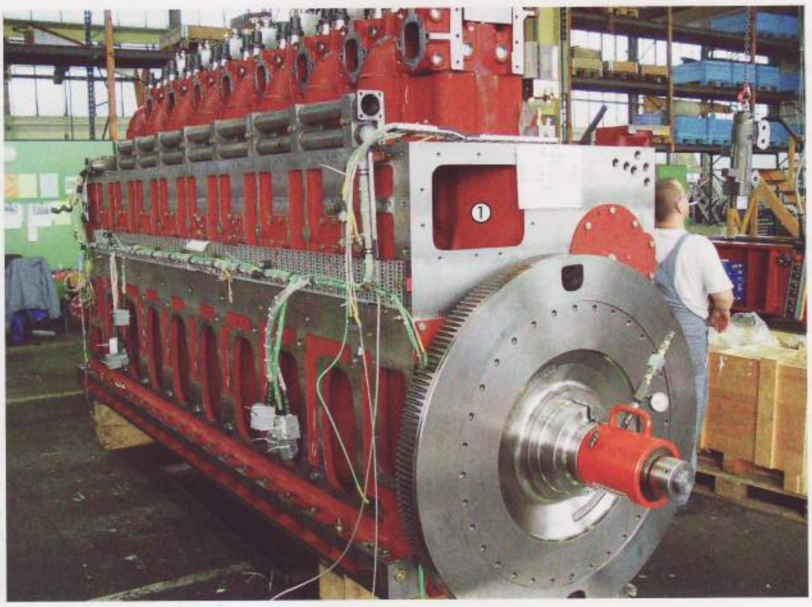
The various types of super chargers are categorised according to the manner in which the exhaust gases are discharged to the turbo blower.
There are roughly four systems.
1. **Equal pressure- and constant pressure system.**
2. **Pulse system.**
3. **SPEX-system.**
4. **Pulse-Converter system.**

12.10.1 Equal pressure- or constant pressure system

All the exhaust gases are discharged into one single, capacious manifold. The pressure impact when the exhaust valves open is levelled off in this large exhaust manifold, so the exhaust gas pressure before the turbo blower is virtually constant.
At a low engine load, it is possible that the pressure in the exhaust gas manifold exceeds the pressure in the air inlet manifold, this could cause exhaust gas flow into the air inlet manifold during scavenging.
The fresh air is drawn in by the compressor, compressed and cooled in an inter cooler and subsequently channelled into a capacious scavenging air inlet or receiver. In many four-stroke engines the scavenging air receiver is integrated in the engine block.

▼
Four-stroke in-line engines have at present one turbo-blower.

An constant-pressure system is used for cylinder charging. See also the engine manufacturer's guide for supercharging.



◄
The inlet manifold of a MAK–Caterpillar medium-speed H.F.O.-diesel engine.

1 *inlet manifold*

From the receiver the air flows to the cylinders via the inlet valve- of valves.
The exhaust gases in de cylinder flow to an uncooled exhaust gas manifold and then through a turbine.
At a certain engine load the exhaust gases flow at a constant speed through the turbine, keeping the turbine flow efficiency high.
At an engine load below 25% of full load, the exhaust gas scavenging may not function; the exhaust gases flow back into the cylinder as the scavenging air pressure is lower than the exhaust gas pressure. A fair number of engines are equipped with this type of system.

12.10.2 Pulse system

Here all cylinders are individually connected to short and relatively narrow exhaust gas manifolds in which a pressure wave, the so-called pressure pulse is produced the moment the exhaust valves are opened.

Advantages

- Reduced exhaust gas expansion loss when expelled from the cylinder.
- There is a sufficient amount of scavenging air over pressure during the exhaust air scavenging, so 'back flow' does not occur.

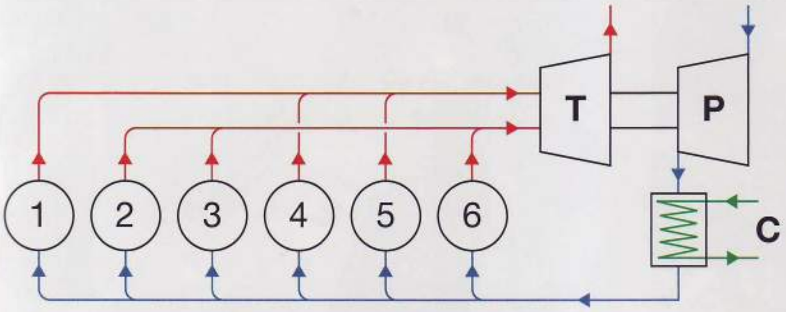
Disadvantages

- Due to the increase in the exhaust gas pressure caused by the narrow exhaust gases manifolds,

►
The inlet manifold of two medium-speed V-type engines.

Left an unmachined Wärtsilä W-38 engine block and right the unmachined MAK–Caterpillar M-43 engine block.

1 *inlet manifold*



6-cylinder engine – pulse-charging system

P - turbo-blower - compression section
T - turbo-blower - exhaust gas section
C - air cooler

The pulse-charging system.

The compressor draws air from the ambiance through a filter and forces it through a cooler to the inlet-air manifold. All the pipes are connected to this manifold.
The exhaust of every cylinder is chosen so that the cylinders do not interfere with each other. For this example; the ignition sequence is 1–2–4–6–5–3, cylinders 1, 4 and 5 and 2, 3 and 6 are connected.

the scavenging air over pressure will be lower at full load than it would be in the equal pressure system. This results in a shorter scavenging period.

- An exhaust air scavenging occurring around piston T.D.C., where the inlet- and exhaust valves are open, is selected for the pressure pulse system. This is done in order to compensate for the reduced scavenging over pressure with an extended scavenging time. This guarantees an adequate exhaust air scavenging.

Ignition timing

In order to avoid a distortion in pressure development in the cylinder during the exhaust air scavenging, the ignition times denoted in crank degrees must be sufficiently high. In an eight cylinder four-stroke engine, there are four independent exhaust gases manifolds in which each two cylinders are connected to one exhaust manifold. Six cylinder engines have two independent manifolds.

Example

An explanation based on a six cylinder engine:

The four-stroke cycle is 720 crank degrees.

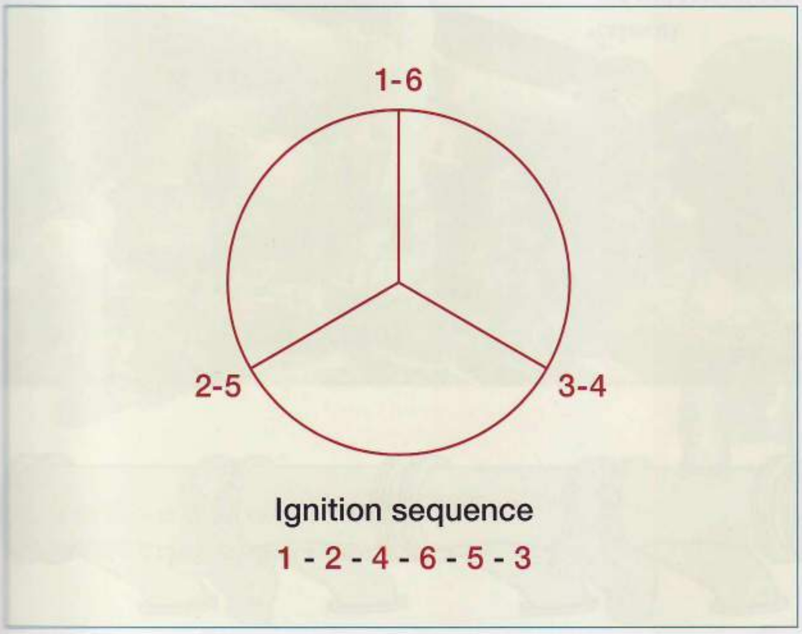
Ignition distances $\frac{720}{3} = 240$ crank degrees.

Remember, there are three cylinders connected to one manifold. Therefore, the ignition times are 240 crank degrees and the pressure surges follow in succession after 240 crank degrees. There is an optimum exhaust air scavenging between the pressure surges.
The crank position is determined by the required equal ignition times, which are:

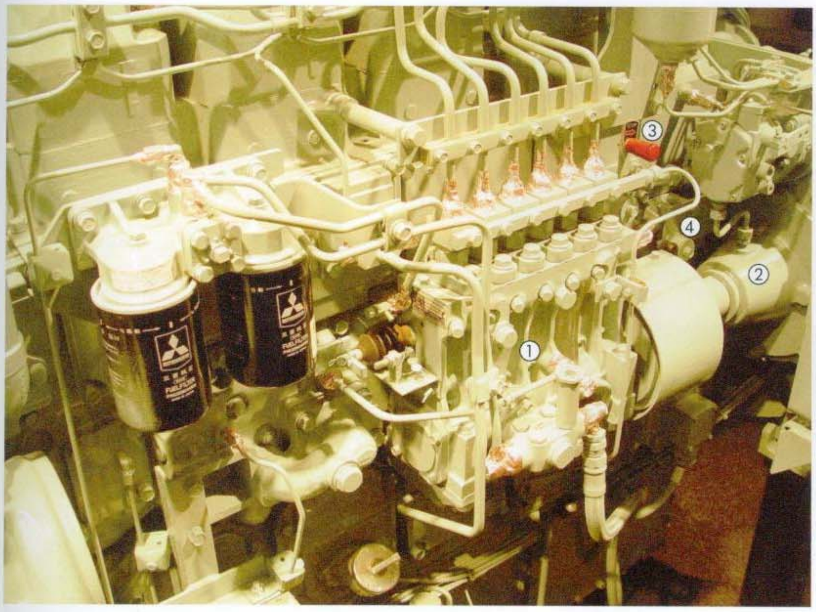
$\frac{720}{6} = 120$ crank degrees.

Balancing of the free forces and the engine moments are also pertinent.

Also see Chapter 17, Vibrations and Balancing.



The ignition sequence of a six-cylinder four-stroke in-line engine.



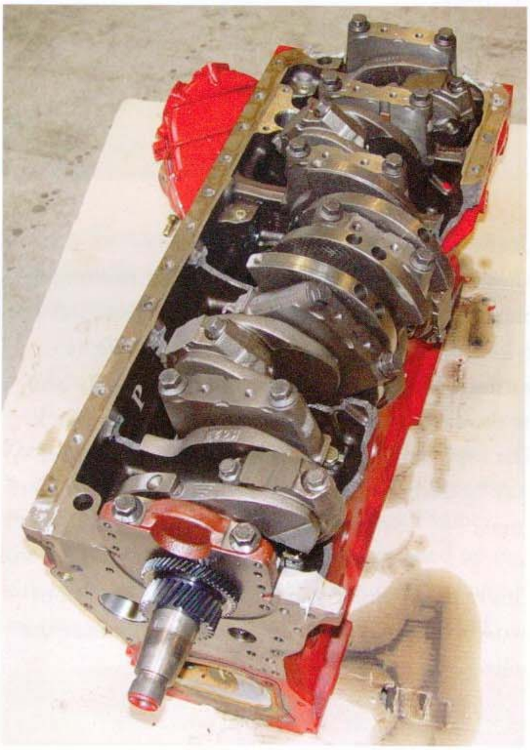
The governor is essential for speed adjustment.

A defective or poorly operating governor results in engine over-speed, which can cause engine damage. Proper maintenance and the correct settings are important. The mechanical connections between the governor and the fuel pump, a lever system, must be able to move freely.

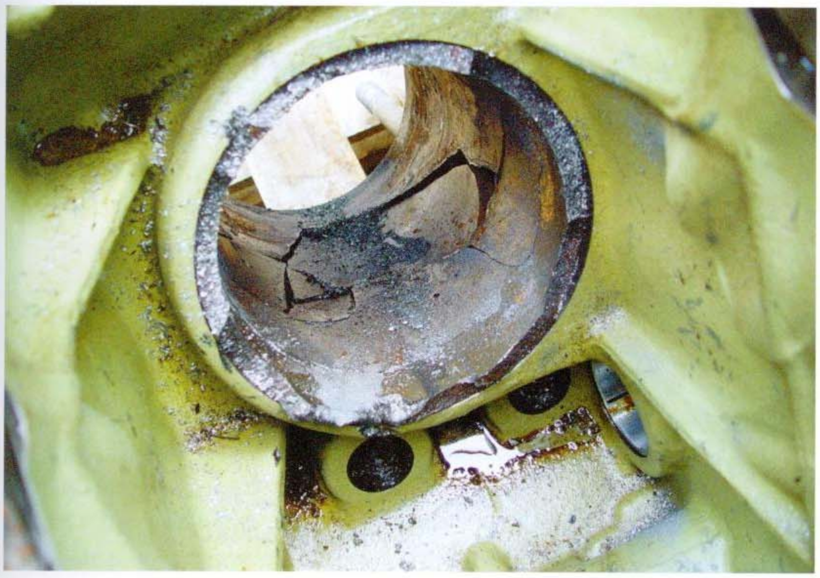
1 *fuel-block pump*
2 *drive from the crankshaft*
3 *emergency stop*
4 *adjusting spindle fuel- block pump*

26.2.10 Engine blocks

The engine block or frame of an engine may vary with respect to construction and size from a small, compact, light-metal cast block to a large, heavy cast-iron block or a built-up frame such as those in large two-stroke crosshead engines.
Generally, the following damage occurs in engine blocks.
Seized pistons resulting in fracturing of the crank-connecting rod mechanism.
This often produces severe damage to the engine block, frequently leading to engine-block scrapping, especially with four-stroke engines
The block is irreparable.



A fractured crankshaft and engine block.



A fractured cylinder liner because of a seized piston.

Cooling-water corrosion

This occurs in small engines with 'wet' cylinder liners as well as in larger engines. Most modern four-stroke medium-speed diesel engines have a **dry block**. Here the cooling water flows through the thickened upper rim of the cylinder liner protruding from the block.

▼

Cooling-water corrosion of the cylinder liner. This is pit corrosion resulting in local and deep corrosion craters.

Damaged alignment edges

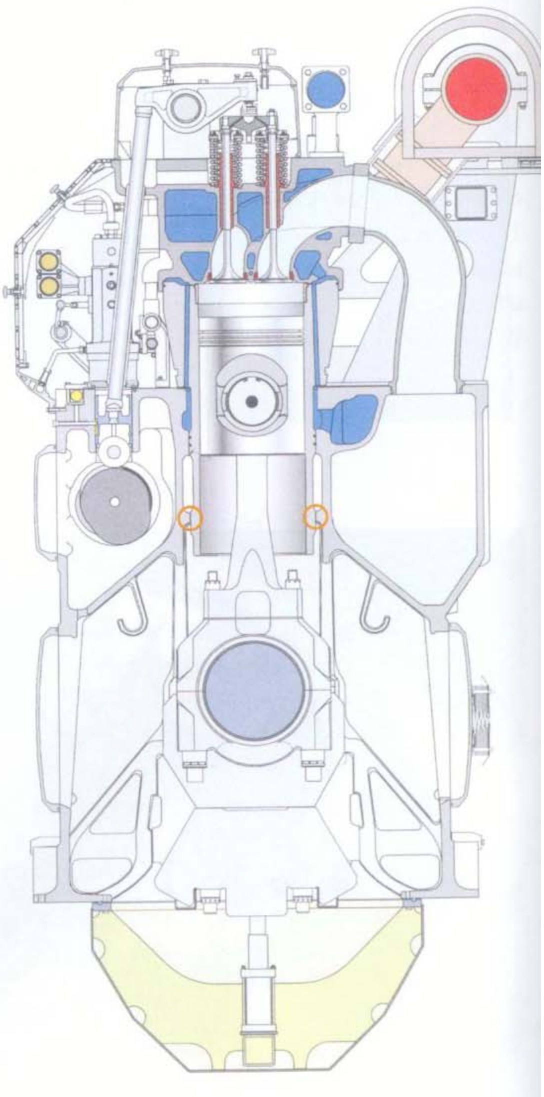
These are often dented or corroded if two engine parts micro move against one another, such as cylinder liners on the engine block. In blocks without protruding cylinder liners, this may also be the case with cylinder head and engine block. The supporting edge of the frame, for instance, may be damaged at the same level as the cylinder liners.

This occurs regularly in large built-up frames of two-stroke crosshead engines, especially near the bolted connections.

▶

A cross-section of a Wärtsilä 46 diesel engine, category III.

○ *Supports*



◄
The crankshaft must be aligned to prevent bending or fracturing of the shaft. The shaft centre line is constantly checked, from casting to assembly in the diesel engine.

Misalignment of the crankshaft and bearings

After a certain time, the crankshaft may no longer be aligned due to deformation of the engine block. This may be the result of a seized part such as a crankshaft that has seized due to an overheated bearing.
Furthermore, problems related to the attachment of the engine block to the bedplate or the bedplate itself may cause the misalignment of the frame and consequently the crankshaft.

Tearing of the cast frame in four-stroke engines

This occurs at excessively high material temperatures or at mechanical overloading of the material. Rapid temperature fluctuations and exceeding the maximum operating temperatures may produce tearing of a cast block.

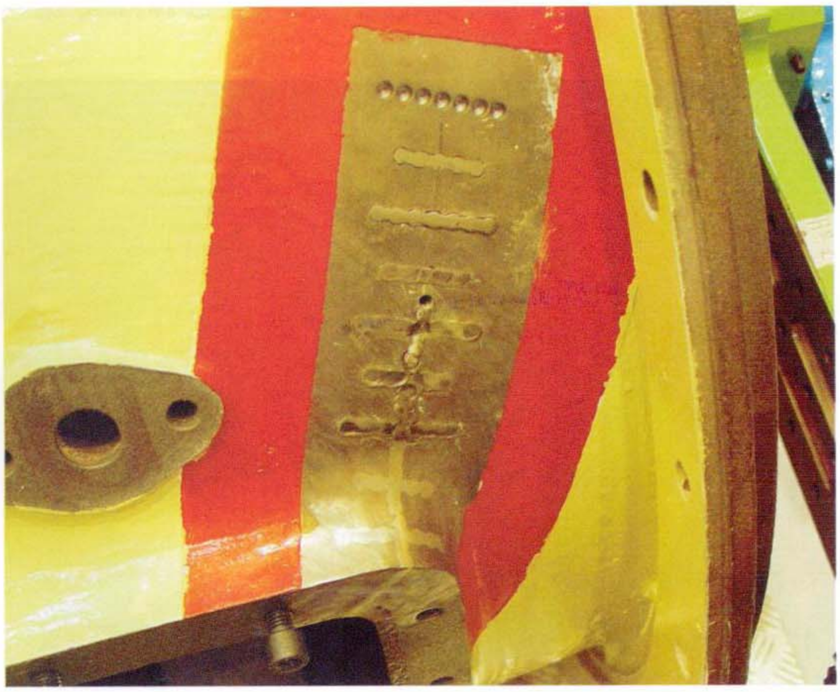
Problem solving

A damaged cast-iron block can sometimes be repaired, provided no essential parts have been damaged. This is referred to as 'meta-locking', a method where the 'fragments' are put back together with stainless steel clips.
Corrosion of the block is indicative of insufficient cooling water treatment.
If the corrosion is superficial and shallow, cleaning and an intensive cooling water treatment is sufficient. Many cooling-water treatment methods apply a film on the cooled part, therefore protecting the parts from, for instance, oxygen found in the cooling water. The oxygen is chemically bound by additives in the cooling-water treatment.

See Chapter 10, Cooling diesel engines.

Damaged alignment edges in the block can be trued up by machining processes such as boring, milling, grinding and sanding. In large engines, the alignment edges in blocks can be trued up in situ with special equipment. When performing machining operations, dimensioning is altered

▼
'Meta-lock', a method of piecing together fragments using stainless-steel clips.



▶
Special attention must be given to the position of the centre line of the crankshaft. This is applicable to new engines, inspection during operation and during major repairs of engines in operation.

▲
Trimming the top alignment edges of an engine block with a reaming machine.

due to the removal of material. At present, some alignment edges of older parts are repaired with a synthetic resin.

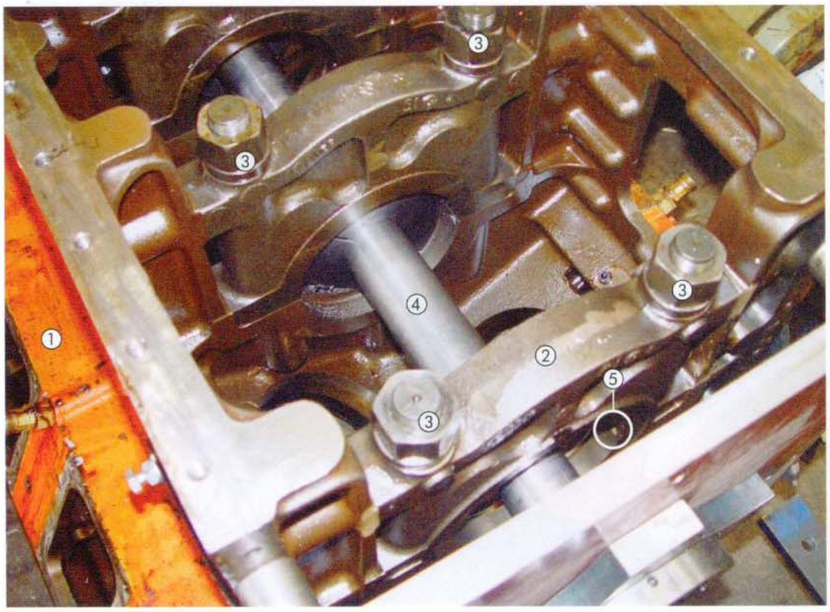
In smaller engine blocks, misalignment of the crankshaft may be fixed by means of line boring: the block is machined with line boring equipment in a specialized workshop or in situ. During this process the material is removed from the block as well as the mounted lower bearing cap to the next undersize, a standard dimension for larger bearing shells.

Tearing of the cast frame

Generally, this cannot be repaired. In cases of serious tearing, scrapping of the block must be considered. Occasionally the start of the tear is 'drilled' to prevent further tearing by drilling a hole in the direction of the crack.

▼
Line boring. The cylinder block has been turned 180°.

1 *engine frame*
2 *bearing cap*
3 *pre-tensioned bearing-cap bolts*
4 *driving shaft of line-boring machine*
5 *cutting section with small chisel near the bearing cap, barely visible*



▼
An arrangement that ensures that the mounted bearing caps can be oversized, so that the oversized bearing shells, and ultimately the crankshaft, are aligned.

1 *engine block*
2 *mounted bearing caps*
3 *line boring tool*
4 *drive*

Reconditioning a connecting rod. Line boring the connecting-rod big end/crankpin (in the background) and the line boring of the connecting-rod head/ piston pin (in the foreground).

26.2.11 Connecting rods

A common problem is oval shaped crankpin bearings.

This excessive ovality may cause damage to the crankshaft.

Inspection

- The connecting rod is cleaned.
- Visual inspection of the parts.
- Hair-crack test.
- Thereafter, the connecting rod with cap is mounted and the bolts are tightened to the normal starting tension.
- Then the crankpin-bearing bore is measured in various positions.
- The pitch and position of the serrations is measured.
- The top angle of the serrations is established.
- The position of the locating piston pin is measured.
- The straightness of the connecting rod is checked.
- The distance between the centre line of the piston pin and crankpin is measured.
- The total weight of the rod is measured for balancing purposes.

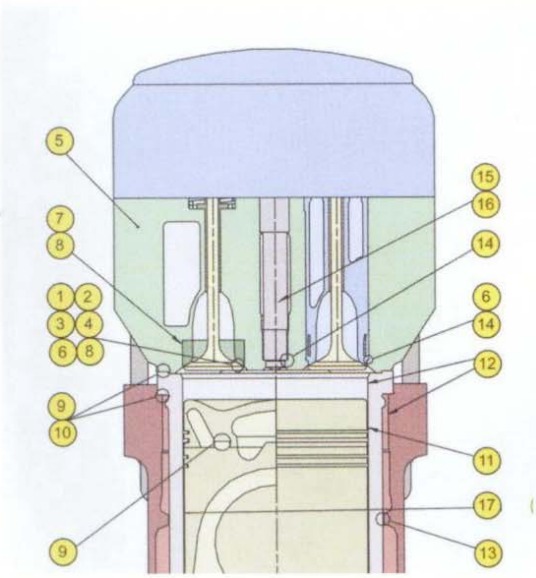
There are two repair methods

Connecting rod without cracks

- The serrations are skimmed on a milling machine.
- Both serrated sections of the connecting-rod parts are fitted again and mating faces checked with blue paste, bolts are tightened in accordance with engine-makers' instructions.
- Subsequently, the crankpin-bearing bore is machined to original dimensions using a rotating chisel.
- The complete connecting rod is labelled and lubricated to prevent corrosion.

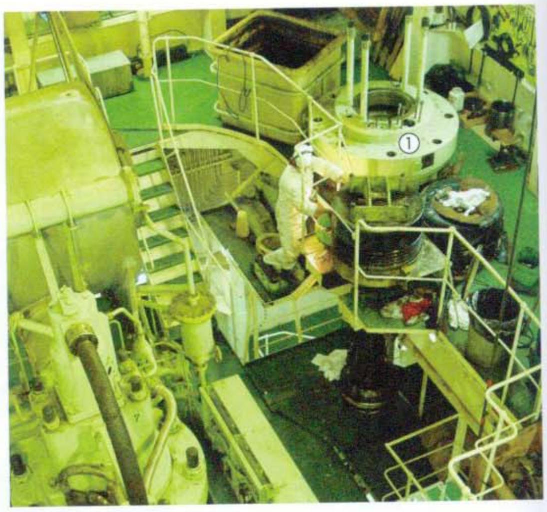
Connecting rod with cracks

- The cracked serrations are machined until the cracks have disappeared.
- After pre-heating the connecting rod in an oven, the machined section is rebuilt with an alloy certified by the manufacturer and following a strict welding procedure. The rod is then immediately stress relieved by gradually cooling. The final process is that all serrations and bores will be machined to original dimensions



All the faces that require inspection and possibly trueing up during the reconditioning of a four-stroke engine.

►
Two-stroke crosshead engines have various parts consisting of steel or steel alloys, for instance the A-frame, scavenging-air space, driving gearing, piston, cylinder cover and exhaust valve.
Many of these parts can be reconditioned using welding processes.



▲
Top right: a disassembled cylinder head (1) is cleaned for inspection.

►
Reconditioned parts are provided with a certain code, for traceability of the reconditioning company.

26.3 Two-stroke engines

General

The pistons and the cylinder covers are manufactured from forged steel or cast steel. This can be welded, so the surface can be rewelded after removal of the damaged material. This is also valid for the exhaust valve.

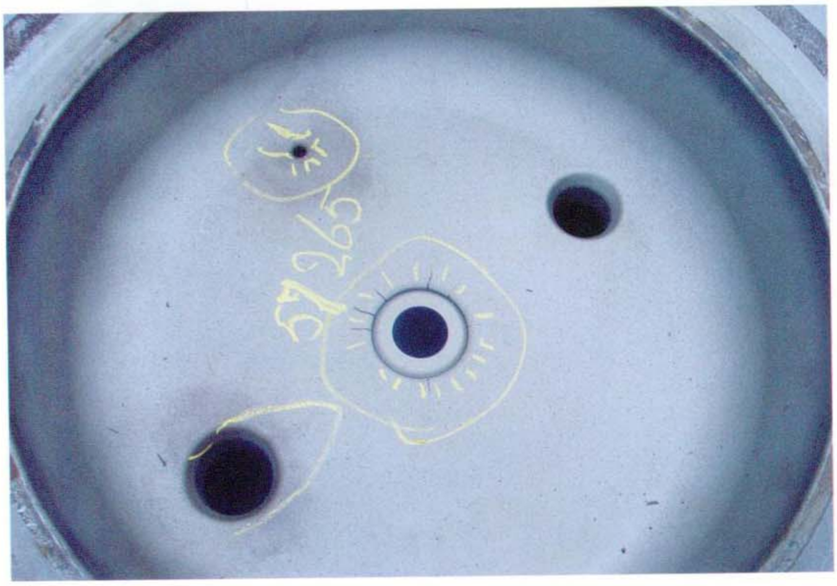
26.3.1 Cylinder heads

The section of the cylinder head, which forms the top of the combustion chamber, is heavily loaded thermally. Crack formation often is a result of this thermal loading.

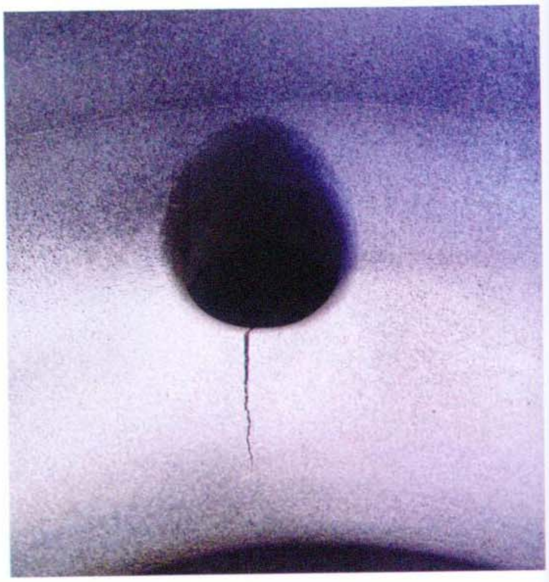
Visible cracks are often found in the section adjacent to the central exhaust valve.
The exhaust-valve casing containing the valve seat in the cylinder head is also heavily loaded. The seat is frequently damaged to the extent that the entire valve seat requires replacing.

▼
Several cracks in an older type cylinder cover of a crosshead engine with an inlet and exhaust port.

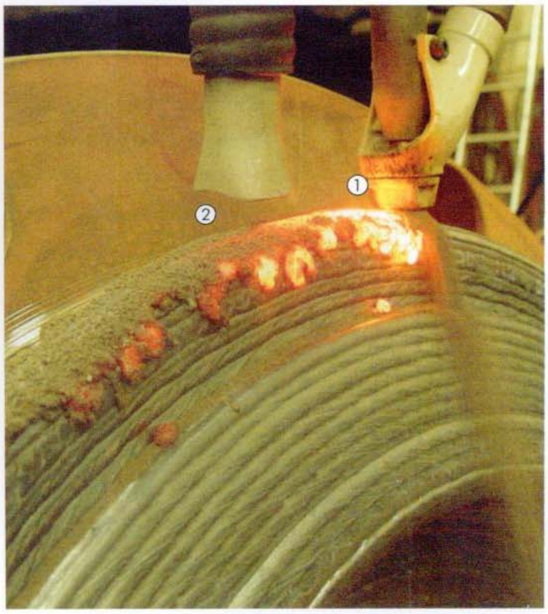
The injector bore is positioned in the centre of the cylinder. The large starting-air and the safety-valve bores. This cylinder cover can be reconditioned.



▼
Cracks in a cylinder head. These are known as thermal cracks.



▲ Machining a cylinder cover.



▲ Rewelding a cylinder cover by a fully automatic submerged-arc welding method. Note; the edge close to the exhaust valve has been damaged.

1 *welds*
2 *powder suction*

Procedure cylinder head

After cleaning in a washing machine, the head is inspected for:

- hair-cracks in the thermally loaded parts;
- irregularities of the alignment edge with the cylinder liner;
- contamination and cracking in the cooling-water ducts;
- impurities in the sealing faces of the exhaust-valve casing, injectors, starting-air valve, the safety valve and the indicator cock.

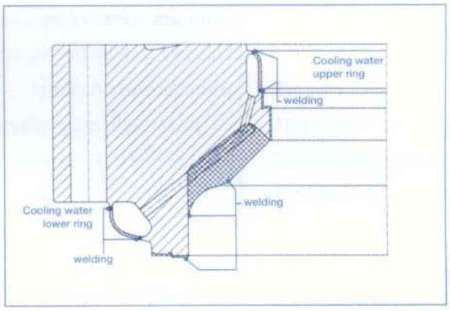
After careful analysis of the above, the decision is made whether it is technically possible and economically viable to recondition the cylinder head.

Reconditioning

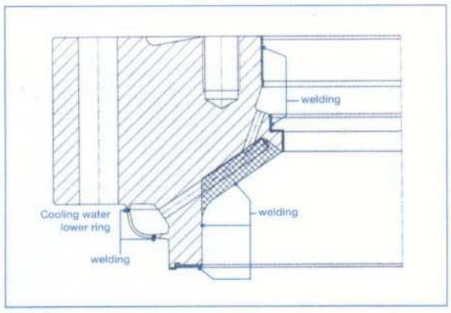
To machine all the cracks by grinding until they have disappeared.
Weld the cracks according to the 'Plasma Transfer Arc' process.
Annealing in an oven at a temperature of approximately 620 °C for 24 hours.
Polishing or metal removal of all welding irregularities to the original dimensions.
Trueing up of all alignment edges for the various parts.
Registering all data and numbering all the parts of the reconditioned cylinder head.
Treatment against corrosion and preparing for transport.

► The welding methods of various types of cylinder heads.

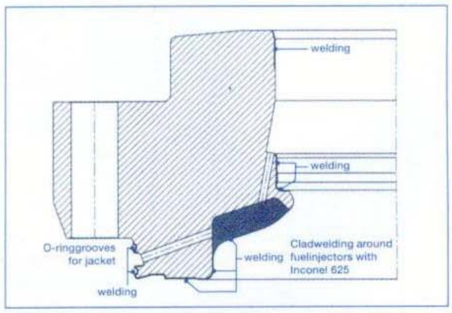
The hatched or black section of the cylinder head is first removed and then rewelded and machined, so that the original size is restored.



Cylinder cover B&W L GBE



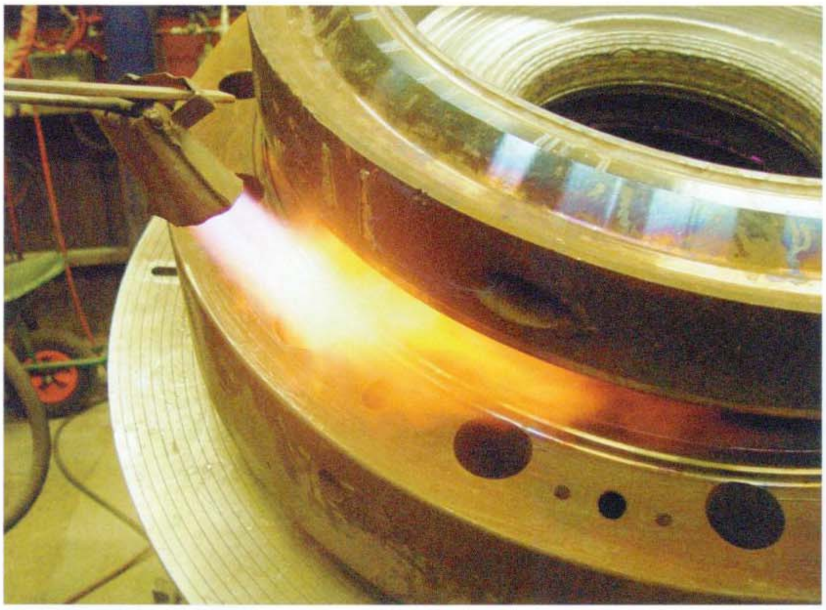
Cylinder cover B&W L MC



Cylinder cover Sulzer RTA standard repair method

▶

With a large gas flame, the slowly turning cylinder cover is kept warm for further machining. This is important for adequate adhesion.



▶

After the piston has been drawn, it is cleaned, subsequently inspected and measured.

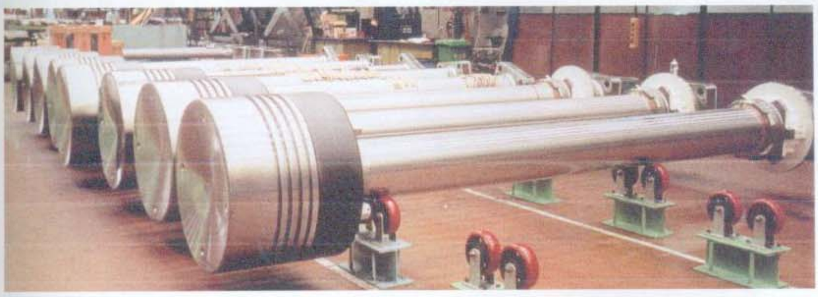
When the piston is severely damaged, it cannot be replaced and will be reconditioned on shore

1 *piston with ring package*
2 *piston rod*
3 *spare cylinder liner*

◀

Finishing a reconditioned cylinder cover, grinding the faces, trueing up the bores, and threaded holes and a visual inspection. Finally, if so requested by the customer, the various parts are mounted. Van West Holland, IJmuiden, The Netherlands.

26.3.2 Pistons and piston rings



◄
New pistons with piston rods and the piston-rod stuffing box, ready to be placed in a crosshead engine.

◄
Piston crowns of a MAN-B&W MC and a Wärtsilä Sulzer RTA crosshead engine.

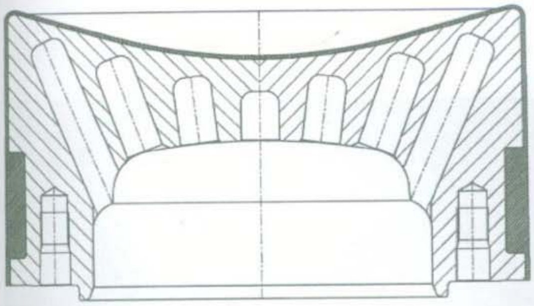
Notice the different cooling arrangements. After cleaning, they are inspected and a reconditioning report is drawn up.

◄
A piston, which has been inadequately cooled by the lubricating oil in circulation in the piston.

The centre section of the bottom of the piston has torn to such an extent that the cover has assumed a conical shape. The overpressure above the piston kept it in its place. During inspection at the reconditioning company, one tap of the hammer was sufficient to remove the cover!

▼
The piston crown of a two-stroke crosshead engine.

This damage was caused by a leaking fuel injector. The piston bottom has been torn in a star shape.



▲
If the piston-ring grooves are worn to the extent that they must be scrapped, rewelding is an option.

The piston-ring grooves and rims are removed and then the space is filled by arc welding. The piston-ring grooves are restored to the original dimensions.

Black = rewelding

Piston crowns

As a result of the high mechanical and thermal loading, the piston crowns containing the piston rings require replacement or reconditioning after a certain number of operating hours.

▸▸ Arc welding the bottom of a large piston. The side has been removed and will be rewelded. Subsequently the new piston-ring grooves are restored.

▸ A disassembled piston prior to cleaning.

The carbon deposit above the piston-ring package is clearly visible. All the piston rings are intact and moveable.

▸ A piston after a large number of operating hours.

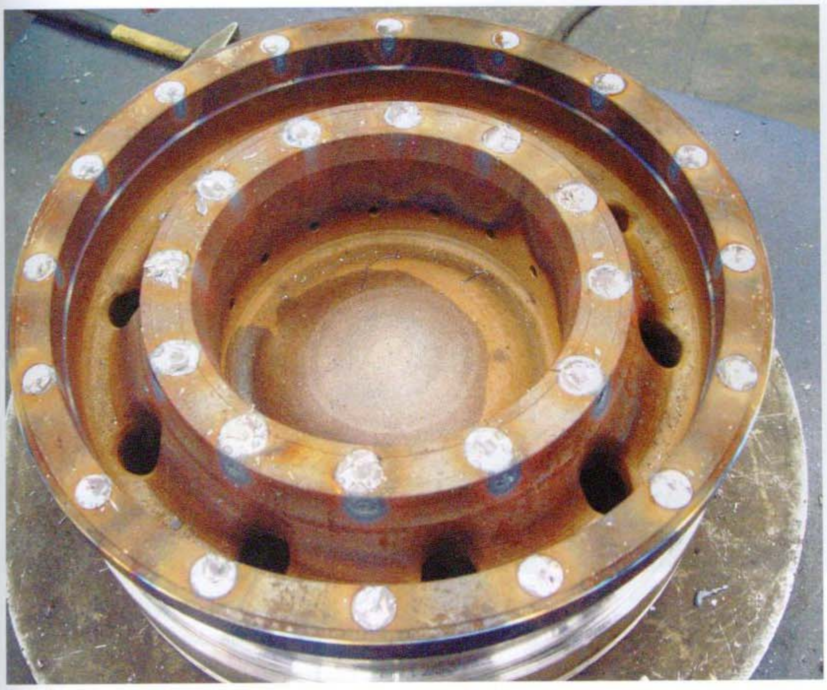
The piston crown shows signs of carbon deposit. The piston rings are in good condition and can move freely. There is no carbon build-up behind the piston rings.

▲ The rewelding is finished. Now the piston is machined to its original size in a lathe.

The following is often established:

- internal **crack formation.** These cracks must be located, removed, and rewelded.
- cracks in the piston crown. These may be rebuilt using a special controlled rewelding process.

▸ Arc welding the bottom of a piston.



◄

The piston crown is deformed to such an extent during rewelding that the connecting-bolt holes are no longer aligned.

This is the reason that they are filled when the piston is rewelded and then re-bored.

▲

Drillings in a piston crown that are torn between the positions marked in chalk.

▲

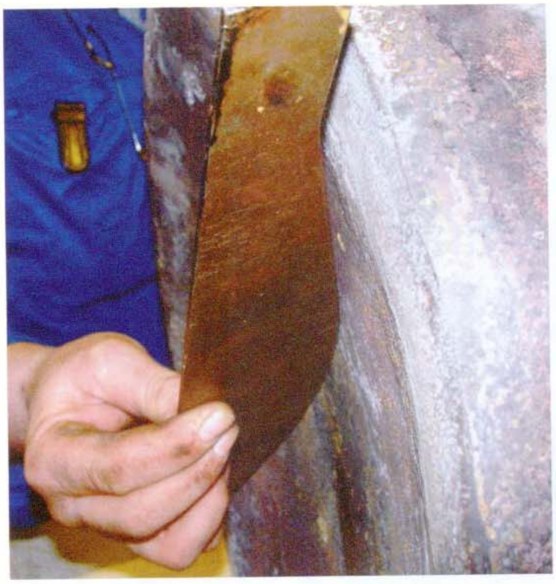
Deep craters formed by corrosion of the piston material.

▼

A piston crown showing signs of burning where the fuel injection takes place.

Crown burning. All damage and cracks must be removed; the crown must subsequently be rebuilt to the original height using a special rewelding process. A special protection coating may be applied if required.

Severe crown burning and cracking. All cracks deeper than 10 millimetres must be located and removed and the crown rebuilt to its original height and profile. In the latest types of crosshead engines, crown burning can be so severe that the crown must be replaced by a completely new section



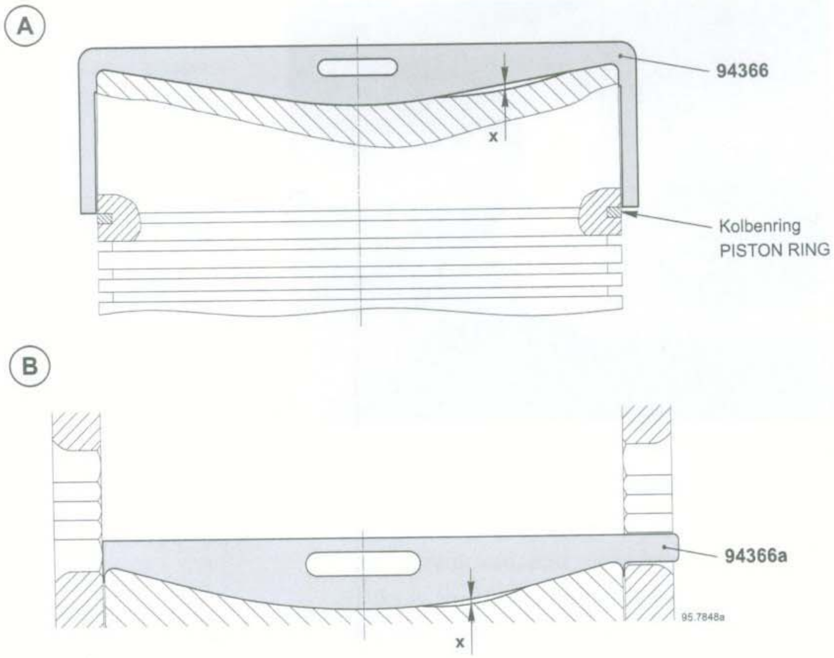
▲
Measuring piston-crown burn with a piston-crown mould.

▲
Reconditioned pistons of two-stroke crosshead engines.

The piston-ring grooves are usually worn to the extent that the complete surface must be removed, rewelded and machined.
As the piston bottom is usually partially burnt away, this is also removed, rewelded and machined. This results in a completely rewelded piston!

▼
A piston crown and measuring tool to measure the extent of crown burn.

A a drawn piston
B a piston in the engine, the measuring mould is suspended from the inlet ports



▼
After arc welding, the piston is machined to restore its original dimensions.

▲
Cylinder covers ready for repair at Wärtsilä in Kruiningen, the Netherlands. The covers are manufactured from forged steel and almost solid.

26.3.3 Cylinder liners

Obviously, cast-iron cylinder liners wear out during usage (see four-stroke engines). Inlet ports are fitted to the bottom of the cylinder liner and all the large two-stroke crosshead engines have cylinder lubrication drillings in the liner wall, and lubricating-oil grooves in the running surface. The upper thickened rim has drillings for the cooling-water circulation.
Conditions in the combustion chamber cause considerable wear and tear of the cylinder liners.

Key points

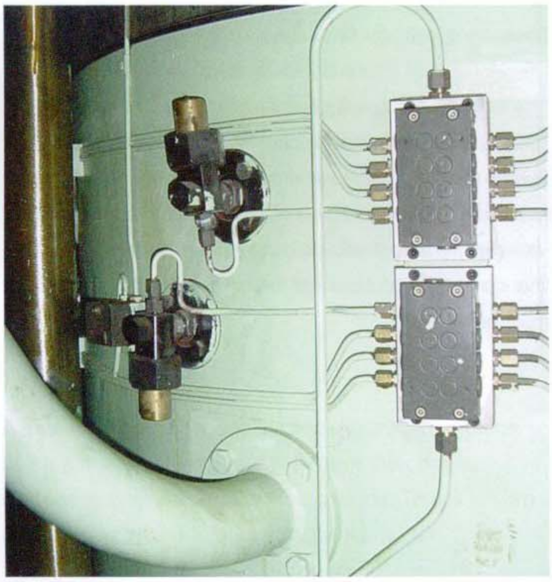
Wear of the running surface

- Normal frictional wear of the piston rings. Boundary lubrication is often inevitable.
- Abrasive wear by hard carbon and ash particles produce severe wear.
- Corrosion, especially from sulphur found in fuel. The latter is referred to as low-temperature corrosion, (L.T.C).

At present, the sulphur content of H.F.O. lies between 2 and 3.5%.

See Chapter 8. Fuels, fuel-line systems and cleaning fuels.

- Glazing of the running surface by 'polishing' by abrasive particles from wear and the combustion process.
- Contamination of cylinder-lubricating oil supply ducts. This mainly occurs due to problems with cylinder-lubricating devices or inferior quality cylinder lubricating-oil.
- Contamination of the cooling-water drillings, and/or spaces, and corrosion of the ducts.
- Damage to the alignment edges of the cylinder block with the cylinder head.



◄◄

A spare cylinder liner.

◄

The cylinder-lubrication distribution on the circumference of the cylinder liner.

◄

A modern cylinder liner with inlet ports.

Procedures

After cleaning the cylinder liner, the geometry is measured.

▲
Measuring cylinder liner wear.

The diameter of the cylinder liner is measured using a caliper gauge. This is done side to side at various heights. One person takes the measurements and a second person records all the values. These are compared to the original values of the liner. A decision is then taken as to whether the liner can remain in place until the next overhaul or if the liner must be replaced.

If a dimension is outside the prescribed limits or if the wear and tear is such that after replacement of the liner, the number of operating hours is too short, the liner is scrapped.

If the dimensions are within the prescribed limits, the following procedure is executed:

- rehoning of the running surface. First, the rim of the liner is removed by grinding;
- **cleaning** of the cooling-water ducts in the thickened upper rim;
- **machining** the liner alignment edges with the block and cylinder head;
- trueing up the cylinder-lubricating oil bores.

26.3.4 Crankshafts and bearings

Large two-stroke crosshead engines have heavy, bulky crankshafts.

A crankshaft of a Wärtsilä Sulzer FLEX, a fourteen-cylinder diesel engine with a bore of 960 millimetres and a stroke of 2500 millimetres is 27 metres long and weighs 155 tons.

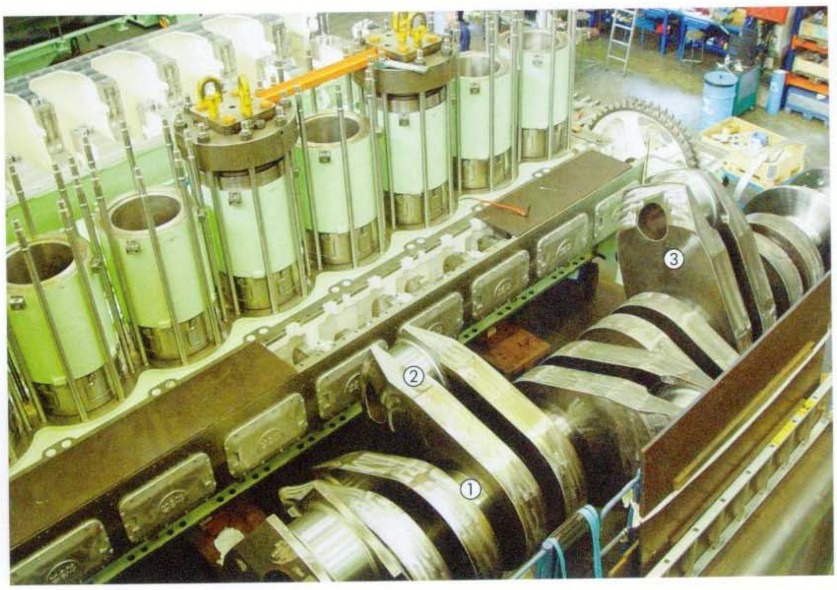
Damage to the crankshaft of this kind of engine requires extensive and expensive repairs, which can take several months. The entire engine frame must be lifted hydraulically and the scrapped crankshaft is removed through a hole in the hull.

▼
A large damaged crankshaft is removed from the engine and replaced by a new crankshaft. This procedure takes more than two months.

▼
A crosshead engine with a cylinder diameter of 500 millimetres, a MAN–B&W, in manufacture.

Note the large diameter of the journals and the very small distance between the crank webs. In order to reduce the weight, the crankpin has been hollowed out.

1 *crankpin*
2 *crankshaft journal*
3 *web*



A very large crankshaft lathe.

A very large crankshaft lathe that has a distance of 12,000 millimetres between the centres, large enough to grind nearly all the crankshafts of four-stroke engines. At the company, Mark van Schaick in Schiedam, the Netherlands. Note the large supports (chucks), which prevent the crankshaft from sagging

1 *chuck*

The new crankshaft must be carefully hoisted into the engine room and positioned in the 'crankshaft bed'.

For damaged crankpins and journals, an in-place technological approach is often used: the in-place rounding and finishing by grinding the journals. This is performed by various companies worldwide.

The damaged crankshaft is stationary while the grinding device is in operation. Removal of several millimetres of material can take up to days or weeks.

A new method for reconditioning a crankpin in-place using a manufacturing machine from the company Goltens in Spijkenisse, the Netherlands. This is known as 'in-place' reconditioning of crankpins.

1 crankpin dummy
2 manufacturing machine with:
   - guides on the crankpin radius
   - chisel holder with cutting tool
   - adjustment
3 manufacturing machine is chain-driven by an electromotor

Operation: The machine runs on the radius between the crankpin and the crank web. After trueing up in relation to the heart line of the crankpin, the crankpin is high-speed machined. 5 to 15 mm of material is removed in three to four days. Using the older method, this could take several weeks! Another advantage is that there is less fine residual material in the crankcase. The clamps for the chisel are relatively easy to remove. In this procedure, the crankshaft is stationary and the machine rotates around the crankpin.

▶

In the crankcase, also known as 'sump', of a large crosshead engine.

1 *the A-frame columns*
2 *crankshaft*
3 *crank web*
4 *connecting-rod big end*
5 *main bearing cap*
6 *bearing-cap thrust bolt*
7 *steps towards the crosshead*
8 *main lubricating-oil supply to the main bearing*
9 *crosshead*
10 *crosshead guide*
11 *hydraulic jack*

▲

The hydraulically tensioned cams on the camshaft.

▲

Manual finish of a camshaft bearing.

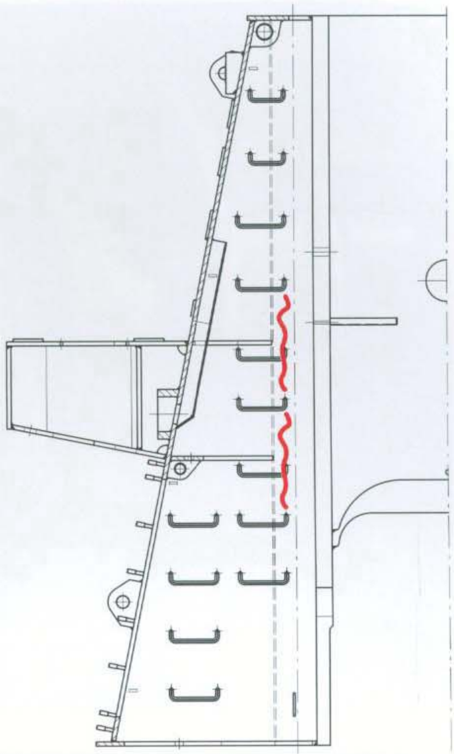
▶

Modern exhaust valves often have a long operating life. The use of material such as 'Nimonac' results in an operating life of 40,000 hours or more.

▲

This exhaust-valve seat has been partially worn away by high temperature corrosion.

Exhaust valves during reconditioning.



Tears in a welded A-frame of a two-stroke crosshead engine.

Tears can also occur in welded A-frames of two-stroke crosshead engines due to high material stresses. These problems can often be resolved to the client's satisfaction. Long-term testing is often too costly and time-consuming. Most bugs are solved in the 'field', so at the customer.

A cylinder cover. In the middle, place for the central exhaust valve.

front left: the bore for one of the two injectors

It is important that all the sealing faces close properly. They are usually machined with special equipment, which can be delivered with the engine.

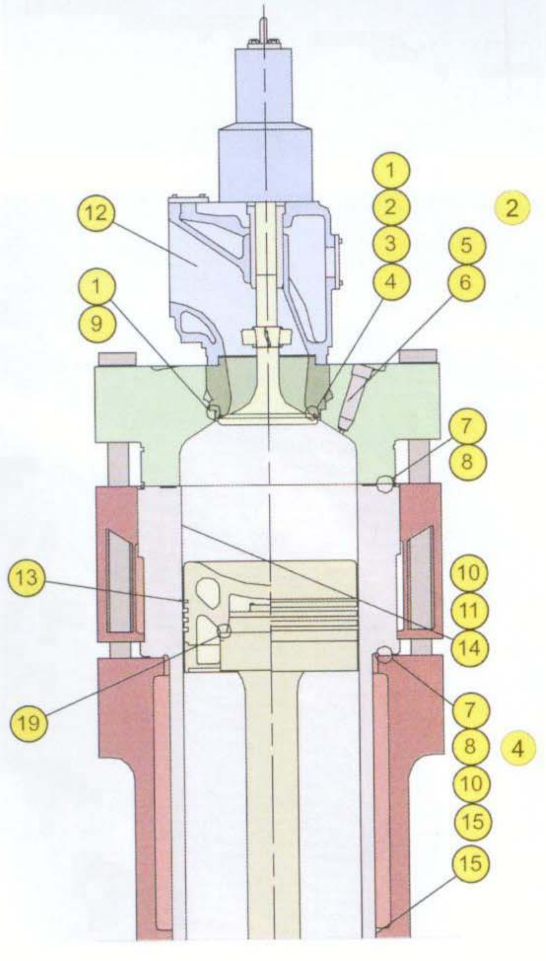
The exhaust-valve casing containing the valve seat, situated in the head is also heavily loaded. The seat is often damaged to the extent that the entire valve seat must be replaced.

Welded A-frames

A-frames manufactured of steel plate are light and strong. However, they can sometimes cause cracking.

This occurs occasionally in engines of all of the three engine manufacturers.

The power play in these large, up to 32 metre engines, is enormous. Usually, the dimensions are measured in trials to investigate the behaviour of



All the sealing faces that are checked and, if required, trued up during reconditioning of a two-stroke crosshead engine.

these welded constructions.

Therefore, regular checks for cracking are

▶

A dirty turboblower, exhaust-gas section.

▶

A seized piston in the cylinder liner.

▶▶

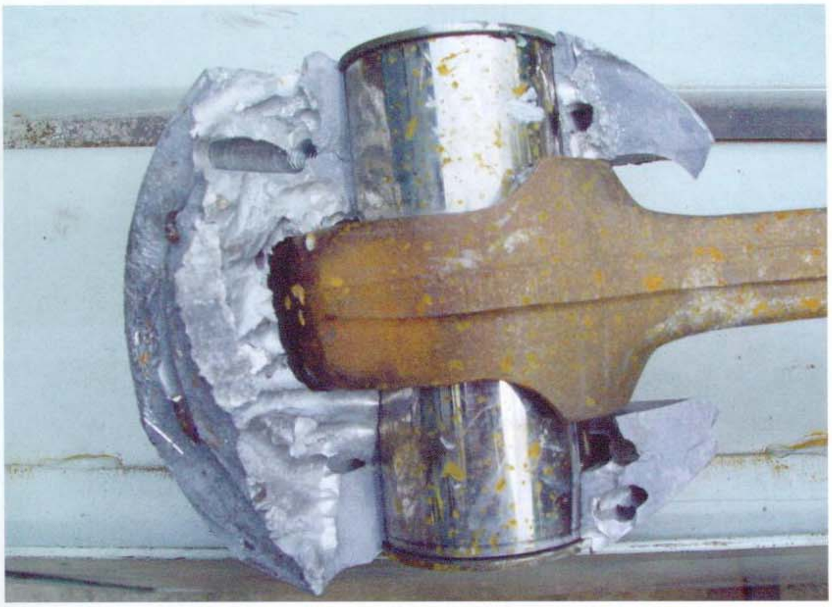
A torn piston bottom of a two-stroke crosshead engine.

▶

An attempt to weld a cooling-water jacket.

◄
An overheated and distorted cam.

▲
A split piston.

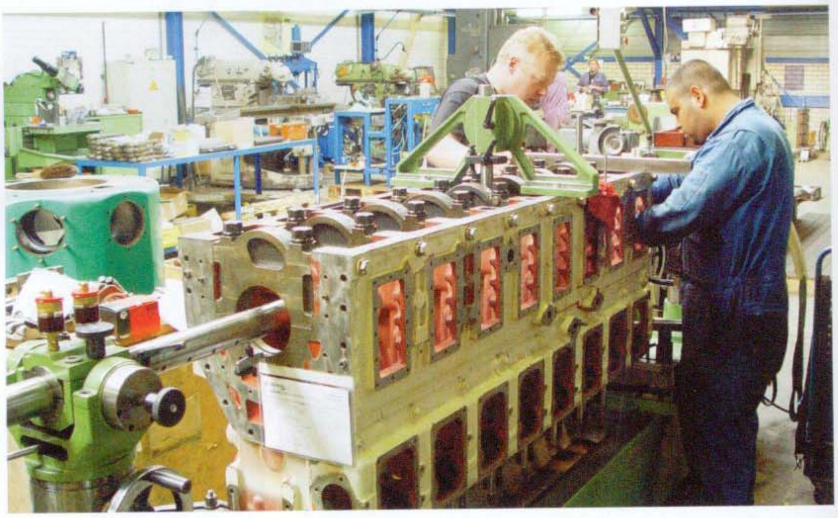


▲
A damaged piston with connecting rod.

◄
A damaged engine block.

▶
A damaged crankshaft.

▶
Line boring the crankshaft bearings in an engine block.



▶
A 'folded' connecting-rod big end due to poor lubrication.

▶
A broken piston; the fracture usually occurs at the drillings through the piston pin.

◄◄
A connecting rod has come loose and perforated the block.

◄
Damaged blading of the exhaust-gas section of a turboblower.

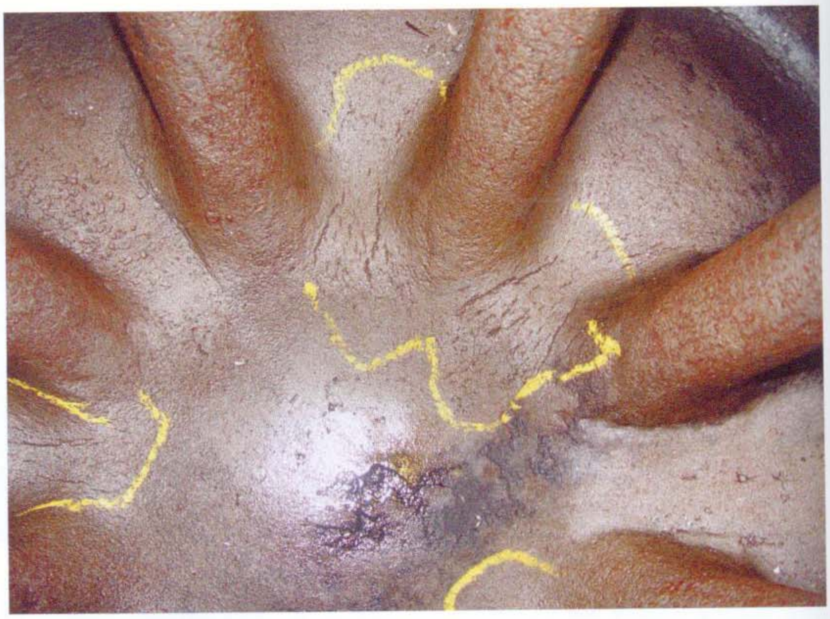
◄
A corrosion hole in the cylinder liner, viewed from the cylinder.

▲
A hole in the casing of the exhaust-gas section of a turboblower. Today, turboblowers are manufactured in such a way that the detached parts remain in the casing.

◄
A piston bottom damaged by thermal overloading. Due to the conical shape, the piston can operate 'normally'.

▶

Tearing on the inside of the piston crown of a two-stroke crosshead engine.



▲

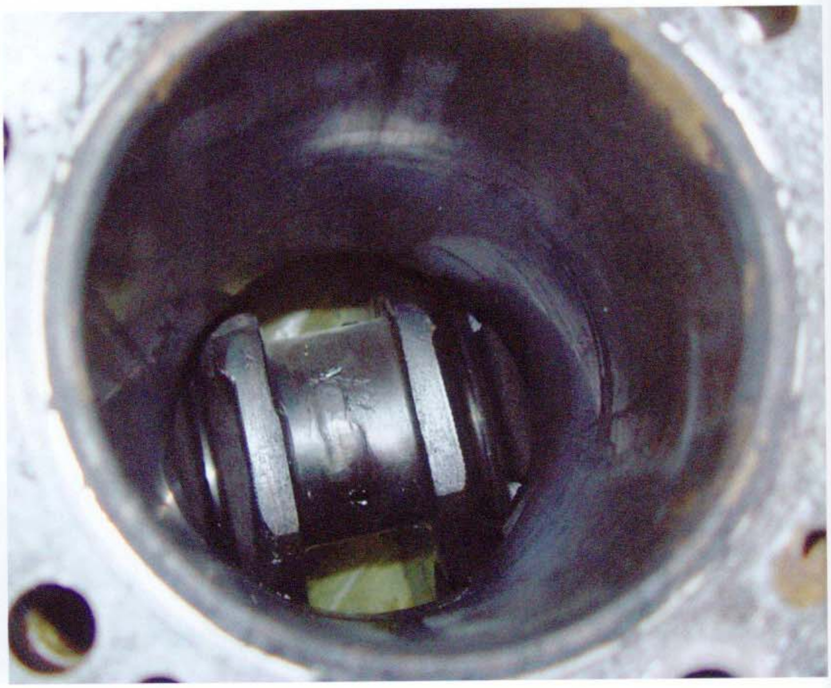
A heavily worn crosshead-pin bearing of a two-stroke crosshead engine.

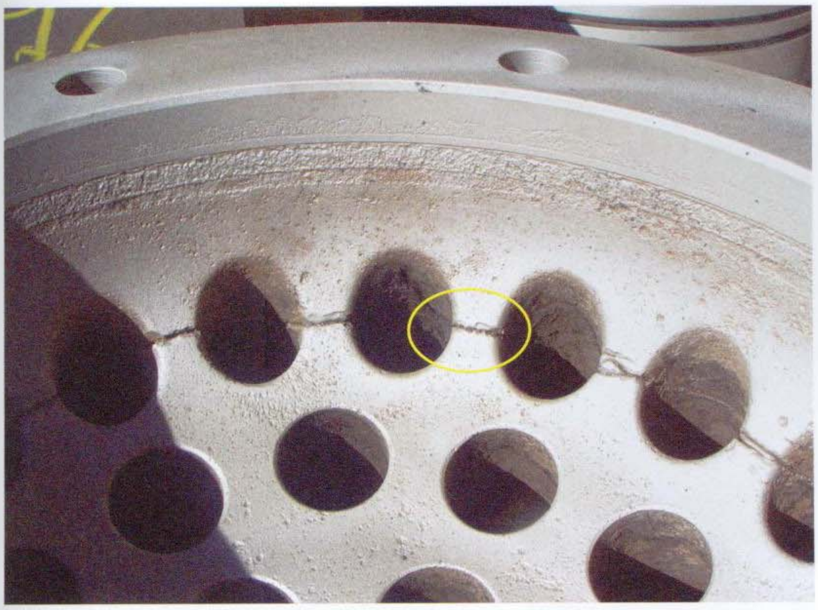
▲

Tearing at the top of the piston crown of the same type of a two-stroke crosshead engine.

▶

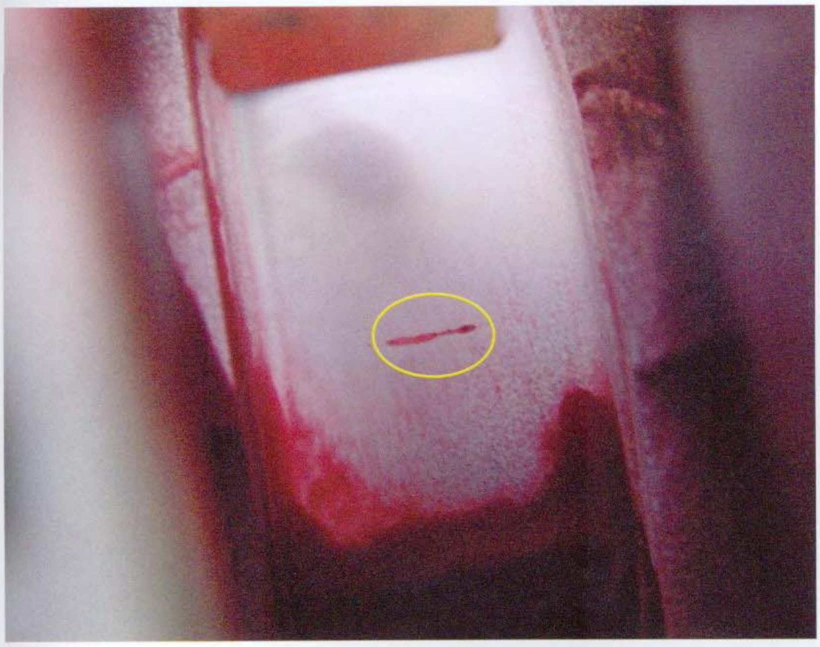
A heavily damaged crankshaft caused by a 'loose' connecting rod. This crankshaft cannot be repaired.





◀

Tearing in the cooling ducts on the inside of the piston crown of a two-stroke crosshead engine.



◀

A visible hair-crack in a crankpin seen during inspection.

◀

'Fretting' of a piston due to poor lubrication.

▶

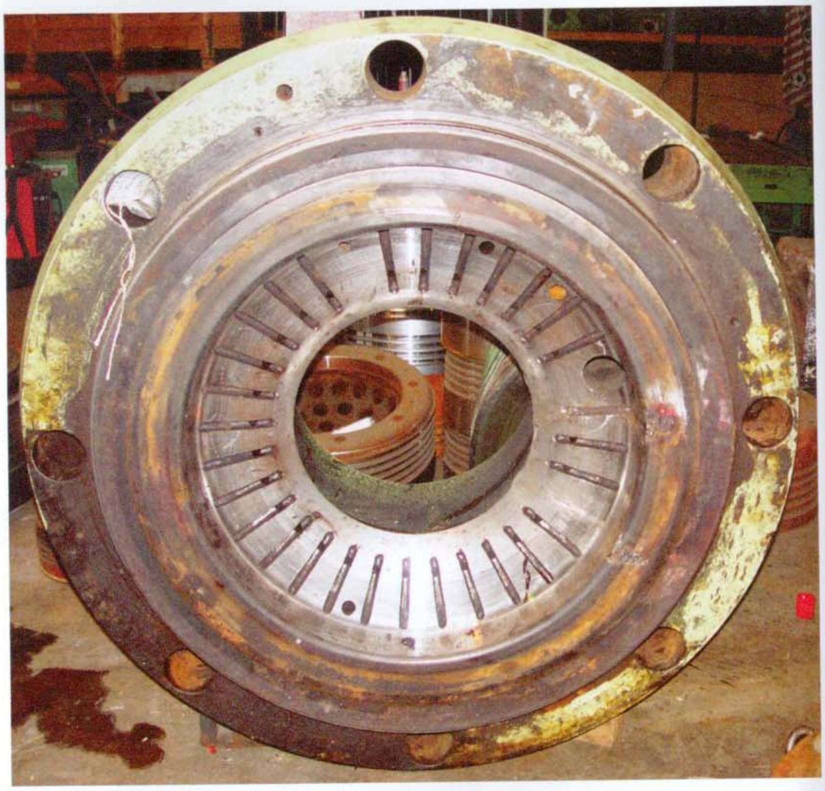
Measuring cylinder-liner wear.

The diameter of the cylinder liner is measured using a caliper gauge. This is done side to side at various heights. One person takes the measurements and a second person records all the values. These are compared to the original values of the liner. A decision is then taken as to whether the liner can remain in place until the next overhaul or if the liner must be replaced.

▶

Reconditioning a two-stroke cylinder cover at the company Polmar in Rotterdam, the Netherlands.

The affected inside of the cover is removed to the heart line of the cooling ducts.
The cooling ducts are fitted with pipes that are drilled out after arc welding.

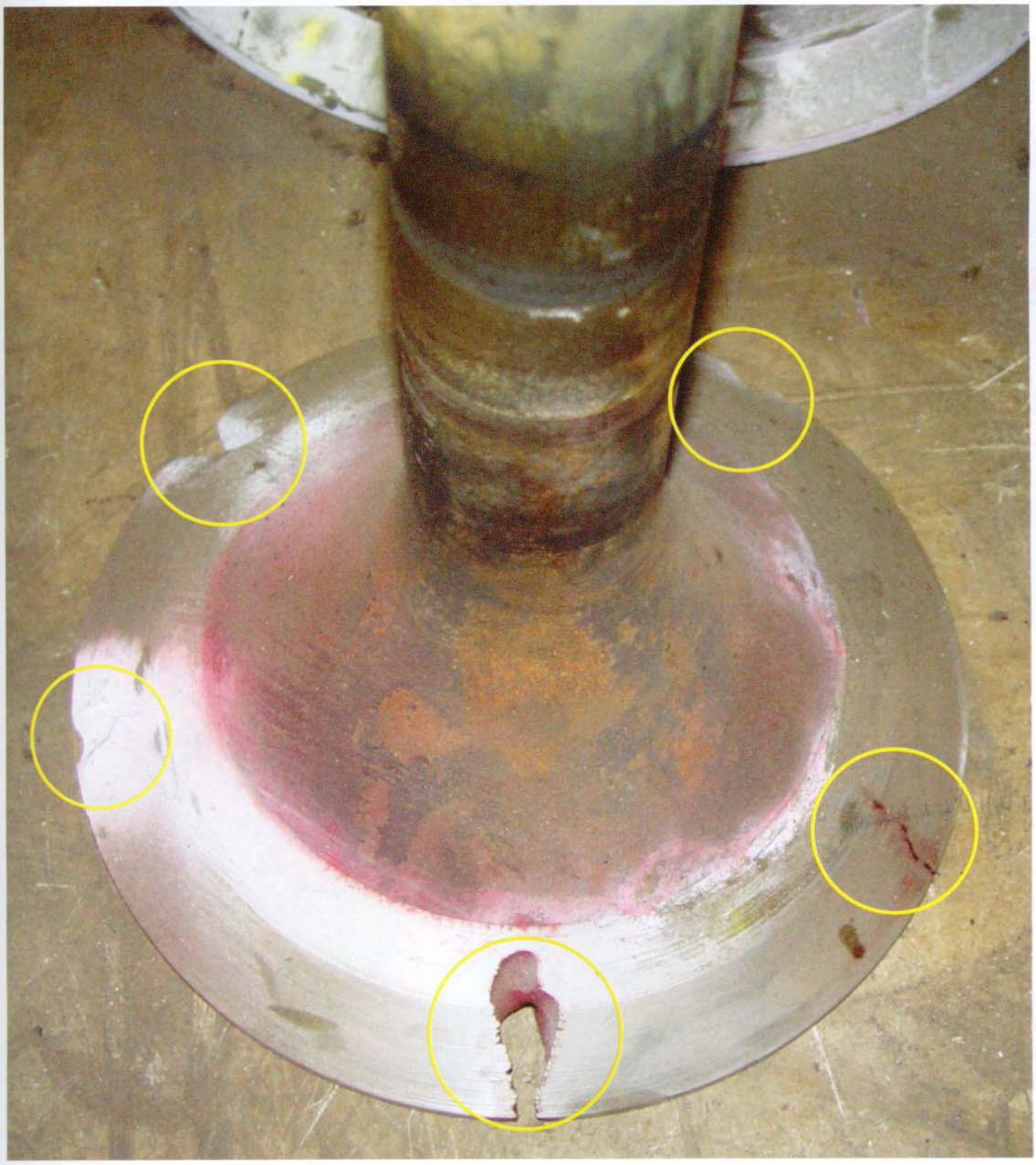


▶

These pipes are inserted in the cooling ducts and are welded shut.

A Rottler machine for trueing up the valve seats.

This machine is installed at Noord-Hollandse Motoren Revisie, Winkel, the Netherlands, and can machine worn seats quickly and effectively.



This exhaust valve of a two-stroke engine is badly damaged and scrapped.

▶

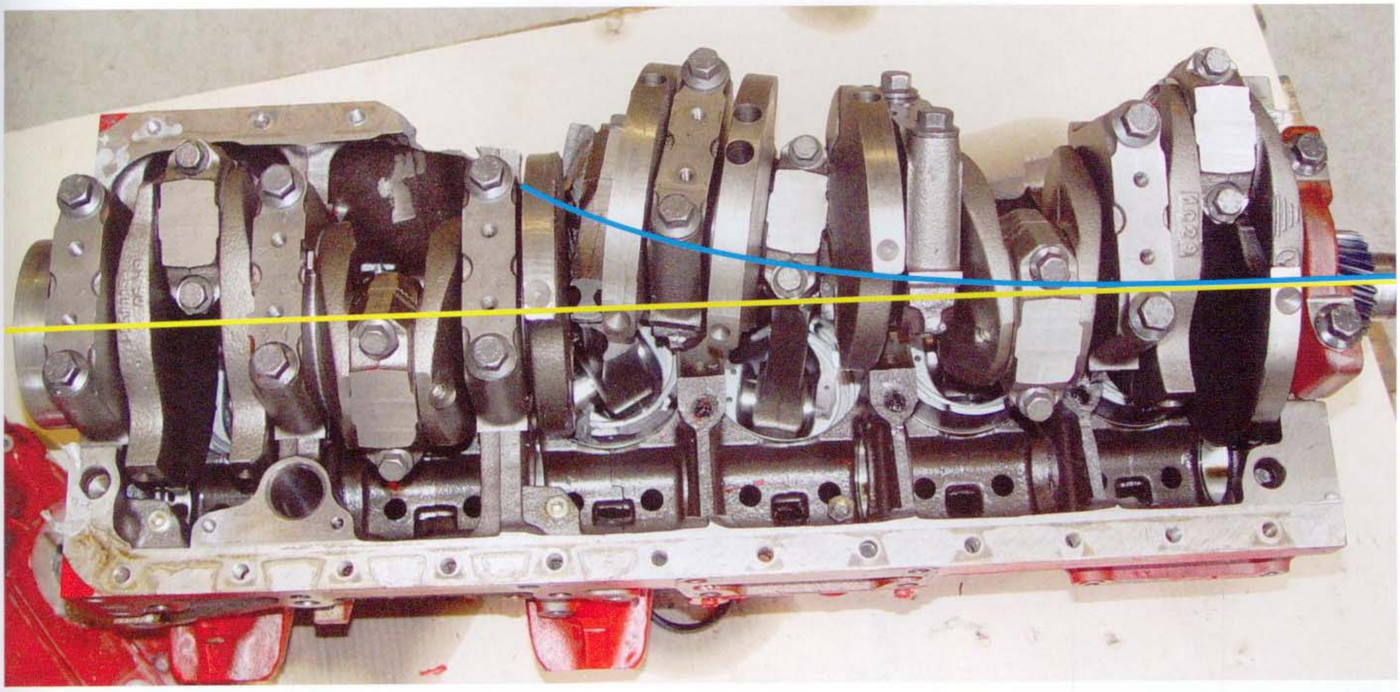
Repairing a fuel pump of a Caterpillar diesel engine at Pon Power in Papendrecht, the Netherlands. The correct measurements and settings are important.

▲

The process.

▶

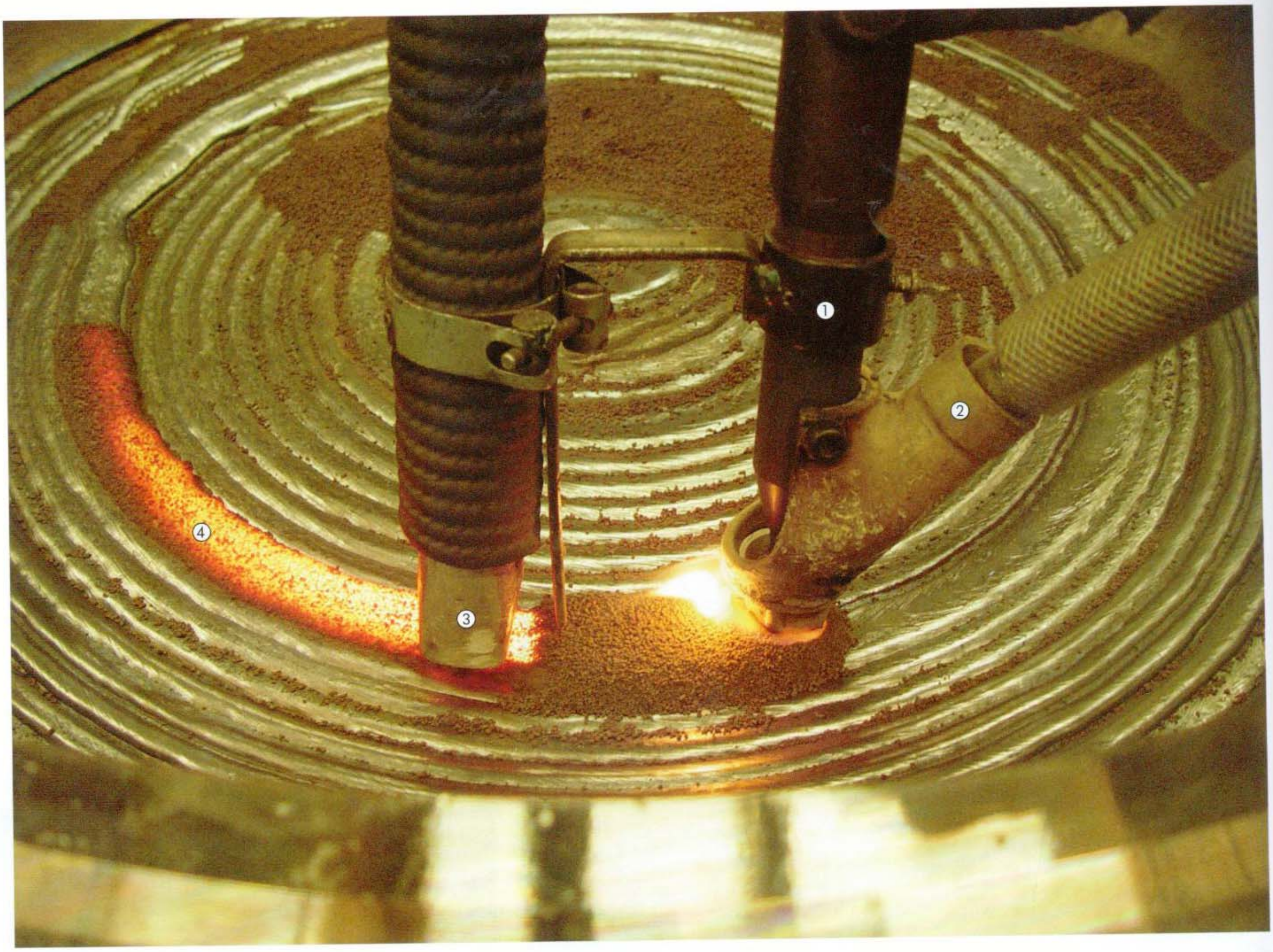
A Rottler machine for line boring engine blocks, for high-speed four-stroke diesel engines of Caterpillar and Cummins. Line boring is far less time-consuming than the traditional process. Mark van Schaick, Schiedam, the Netherlands.



centre line normal crankshaft
centre line broken crankshaft

▲
A broken crankshaft in a damaged block. This diesel engine drove a water pump on a dredger.
The engine probably suddenly drew in a large amount of water…?

◄
Machining a light-metal block of a small diesel engine.



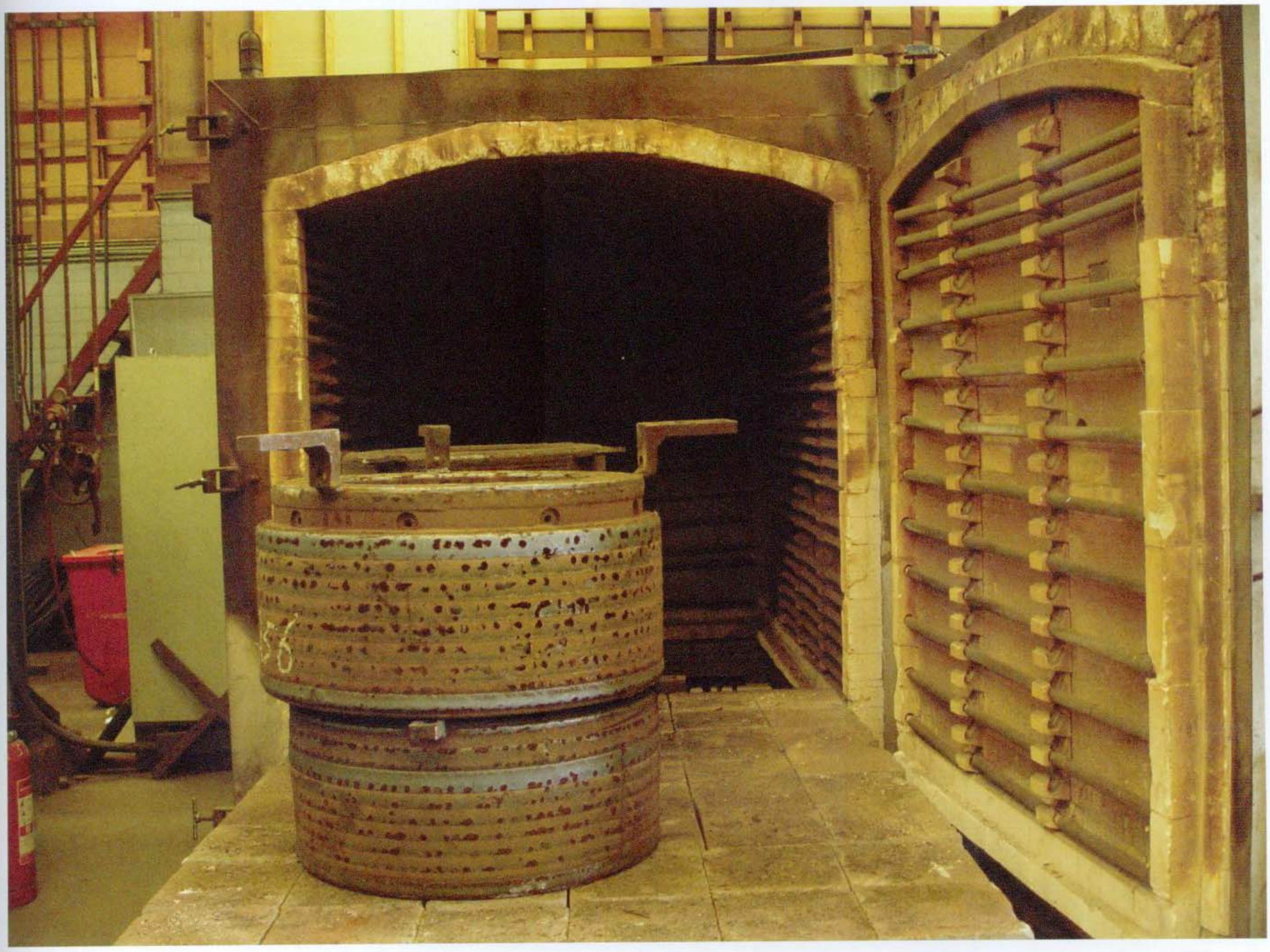
▲

Arc welding visualised.

Right, the gas supply and the electrode. Left, the powder suction.
The slag to the left is manually removed by a hammer.

1 *Electrode wire supply*
2 *Gas supply*
3 *Suction of the powder sealant.*
4 *Hard slag layer.*

A piston crown is being welded.



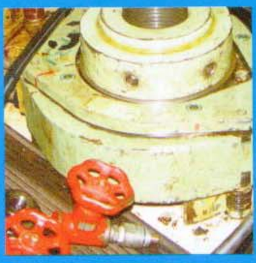
▲

An oven to anneal the welded parts. Here two piston crowns.

After heating, the welded parts are allowed to cool gradually. The stresses introduced by welding are hereby released to a large extent.

H27

Maintenance and repairs

Repair and maintenance of modern, highly charged diesel engines require expert knowledge.
A good inventory-management system, the correct tools and especially a competent and enthusiast crew that can, as shown here, keep the propulsion installation of the 'Oranjeborg' of Wagenborg Shipping in good condition, are required.

The engine control room with left, the main switchboard and in the middle, amongst others, the remote control and monitoring of the main engines.

27.1 Introduction

Engines in operation wear and require maintenance to remain in good condition. Obviously, engines that are faulty or poorly maintained present problems.
Diesel engines for shipping and electrical-power stations must be in an excellent condition. They must be capable of running at full load for most of the year.
At present, ships have short docking periods; there is barely time to perform major maintenance.
If required, engine-repair and maintenance companies are hired that either with or without the (at present small) crew, perform the activities.
It is of the utmost importance for engine manufacturers to maintain a worldwide service network in order to provide a good service standard for their clients. Quick delivery of spare parts is also essential.
Major ship maintenance is scheduled, if possible, to coincide with the planned dockings.
Under certain circumstances, there is more time available for maintenance of diesel-driven power stations, but here the demand for electricity at full-load and for an extended number of operating hours is progressively increasing.
The mean number of operating hours of a good power plant at full-load is between 90 and 95%. Sometimes higher: per year between 7500 and 8500 operating hours! In addition, here, the maintenance must be first-rate and well organised.

27.2 Types of maintenance

There are in general terms, three types of maintenance.

1 **Preventative maintenance**
This is planned maintenance aimed at the prevention of breakdowns and failures. Parts that often are not worn or in need of replacement are exchanged in accordance with a schedule.

2 **Corrective maintenance**
This is maintenance required when an item has failed or worn out, to restore it to working order.

3 **Condition-based maintenance**
This is maintenance performed based on the state-of-repair at a certain time, combined with experience. In order to prevent failures, the spare parts are removed, repaired or replaced after a predetermined of number operating hours. At times, it is apparent that the number of operating hours can either be increased or decreased. This is a matter of experience. The engine suppliers often provide spare-part lists and information regarding the maximum number of operating hours. The prescribed number of operating hours is thus that so multiple maintenance activities can take place at the same time.

27.3 Instruction manuals/ Maintenance manuals

These must be thoroughly read in order to have a good understanding of the operation, maintenance, repairs, and settings. **All too often instruction manuals are not consulted before starting the various activities.**
Often, the smallest details are decisive in performing successful maintenance.

It is advisable to follow the procedure below.
1 Read the instruction manuals and maintenance procedures carefully. This includes the drawings.
2 Find out what special tools are required and gather them.
3 Determine the required time for maintenance.
4 Discuss the activities with workers and exchange information.
5 Ensure that the working environment is clean and well lit.
6 Use approved hoisting equipment and well-maintained tools.
7 Check the hydraulic tensioning equipment for sufficient oil and ensure that the various hoses are not damaged. Make sure that the couplings are clean.
8 Avoid getting dirt **in** the engine when open.
9 Store all disassembled spare parts neatly, preferably on white cardboard, clean cloths and heavy parts on wood, and never on steel. In the latter case, cylinder-head covers may be damaged.
10 Hoist heavy parts whenever possible; do not lift them. **Mind your back!**
11 Numerous engines have special attachments to facilitate disassembly/assembly; use them and keep them in good condition.
12 Work methodically and do not rush. In the latter case, all too often mistakes are made.
13 Discuss problems with a colleague: two heads are better than one.

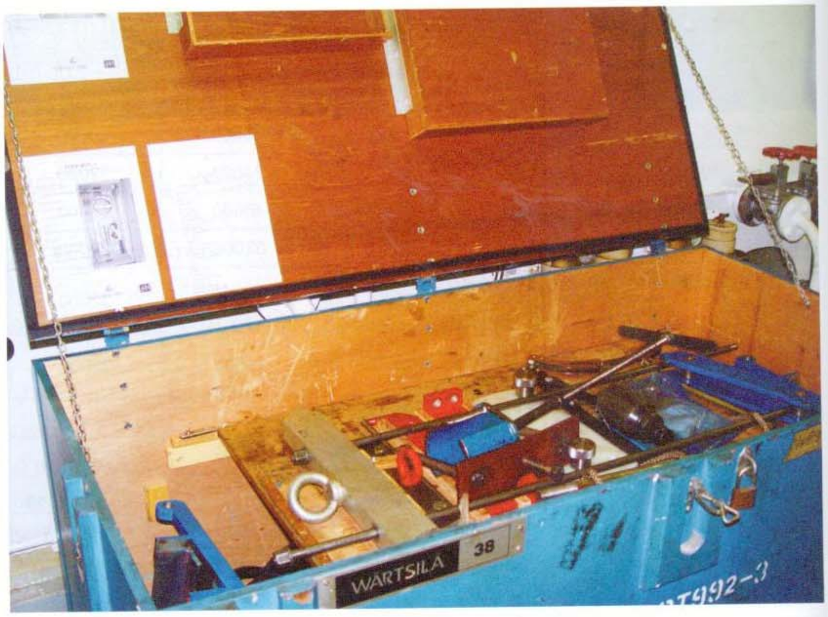
	HFO	MDF	HFO	MDF
	Time between overhauls (h)	Time between overhauls (h)	Expected comp. lifetime (h)	Expected comp. lifetime (h)
Main bearing	12000	16000	36000	48000
Big end bearing	12000	16000	24000	32000
Gudgeon pin bearing	12000	16000	48000	48000
Camshaft bearing bush	16000	16000	32000	32000
Camshaft intermed. gear bearing	16000	16000	32000	32000
Balancing shaft bearing, 4L20	12000	16000	24000	32000
Cylinder head	12000	16000		
Inlet valve	12000	16000	36000	32000
Inlet valve seat	12000	16000	36000	32000
Exhaust valve	12000	16000	24000	32000
Exhaust valve seat	12000	16000	36000	32000
Valve guide, EX	12000	16000	24000	32000
Valve guide, IN	12000	16000	36000	48000
Piston Crown	12000	16000	24000	48000
Piston rings	12000	16000	12000	16000
Cylinder liner	12000	16000	48000	64000
Antipolishing ring	12000	16000	24000	32000
Connecting rod	12000	16000		
Connecting rod screws	12000	16000	24000	32000
Valve tappet and roller			24000	32000
Injection pump tappet and roller			24000	32000
Injection element	12000	16000	24000	32000
Injection valve	6000	8000		
Injection nozzle	6000	8000	6000	8000
Water pump shaft seal	12000	12000	12000	12000
Water pump bearing			24000	24000
Turbocharger	24000	24000		
Governor	12000	12000		
Vibration damper	Acc. to manuf.	Acc. to manuf.		

◄
The number of operating hours between two overhauls and the expected lifetime. Shown, a Wärtsilä 20 diesel engine – Category III.

Clearly seen is that the service lifetime of a diesel engine running on H.F.O. is significantly shorter are that of a diesel engine running on M.D.O.. Approximately 25% less!
Many spare parts are ready for replacement by the third overhaul.
With the planning of the (H.F.O.) number of operating hours, companies will attempt to execute an optimum overhaul that is as effective as possible. An injector overhaul is planned after 6000 operating hours, and then after 12,000 hours. Most spare parts are at 12,000 operating hours ready for overhauling, with the exception of the camshaft (16,000) and the turbo-blower (24,000). On average, these types of engines operate between 6000 and 7000 hours per year.

▼
Tools stored in a well-arranged system are a primary requirement to work effectively.

14 Modern communication means allow easy access to the service organisation of the engine suppliers. Use them for advice and consulting; after all, they are there for you!
15 Note all the activities done and record all measurements in a journal.
16 Many engine manufacturers have their maintenance programs computerized. All known engine manufacturers at present have such programs. Advice from the factory can also be obtained by remote operation when using these systems.



Special tools such as these for a Wärtsilä 38 diesel engine Category III are delivered with the engine in a special 'toolbox' with detailed instructions, either on plasticized cards or digitally for use with a special laptop.

27.4 Engine maintenance

Division of the activities

In the past, many of these activities were carried out by the crew.

- There were sufficient skilled engineers.
- There was sufficient time; idle periods of ships were common and long.
- Electronics were not yet available.
- Communication with the engine factory was difficult.
- The operating life of some spare parts was so short, that they had to be frequently repaired. Notorious examples are the exhaust valves.

Nowadays maintenance is performed in a very different manner.

- There are few engineers on location.
- There is very little time, ships dock for a very short time.
- Modern engines work with numerous electronics. Working with these electronics requires specialised knowledge.
- The operating life of spare parts is much longer.
- At present, worn spare parts are removed and then overhauled or new spare parts installed. On board or in a diesel power station, few repairs are carried out, replacement is the remedy.
- Shipping companies often replace all the worn components of a propulsion engine that wear by overhauled or new components. This is done either in port or during major overhaul. After two to three years, dependent on the number of operating hours, these parts are simply replaced.
  This type of major maintenance can be performed in a couple of days or one week.
- Only minor maintenance, such as checks or adjustments, changing fuel- and lubricating-oil filters or testing/exchanging injectors, is still performed by the crew.
- The revision of cylinder heads, turboblower overhaul and for example, testing and repairing fuel pumps and injectors is executed by specialised companies.

Reminder

One should never forget that these systems do not solve breakdowns in the middle of an ocean. One should therefore always ensure that the crew/ engineers are capable of 'bringing the ship home'. **If engineers only work as operators, the risk of 'being adrift' is significant!**

27.4.1 Hydraulic tensioning tools

Many engines in category II and almost all in categories III and IV are currently equipped with hydraulic bolt-tensioning systems. This is often the case in the following:

- cylinder-head bolts;
- connecting-rod bolts;
- horizontal and vertical main-bearing bolts;
- fixing bolts for large counterweights on the crankshaft;
- bolts of the exhaust-valve casing for two-stroke engines and
- tie rods for two-stroke engines and

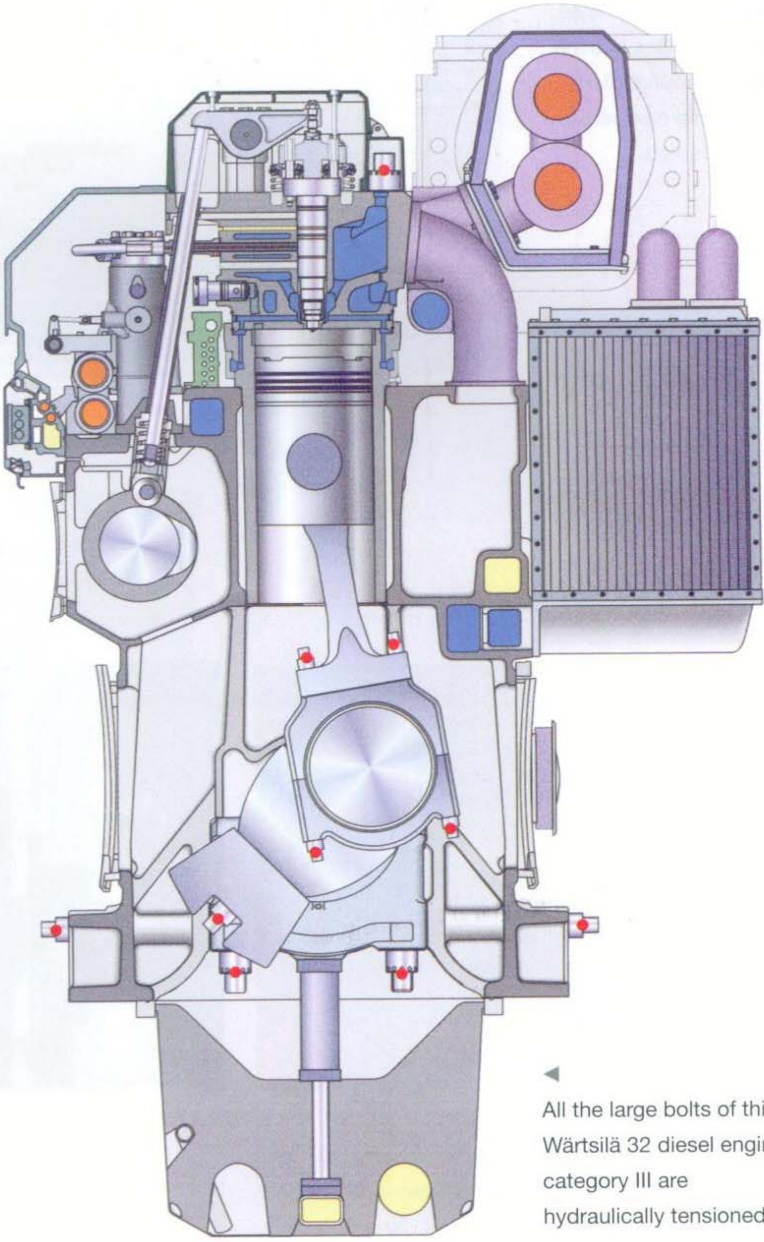
Good maintenance of the technical installation and therefore the diesel engines is of utmost importance for safe sailing, also in bad weather conditions. An enthusiastic, motivated and well-trained crew is essential for every shipping company.

- fixing the piston rod on the crosshead for two-stroke engines and
- fixing of the crosshead-bearing covers and
- foundation bolts for all the larger engines;
- cams on camshafts.

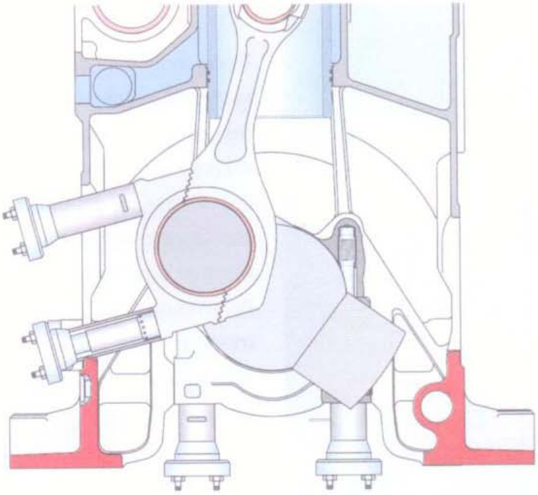
Other fastenings that are installed/disassembled with hydraulic tools are, amongst others, coupling sleeves between shaft-tunnel parts and fixing the propeller on the propeller shaft.

Advantages of hydraulic tools

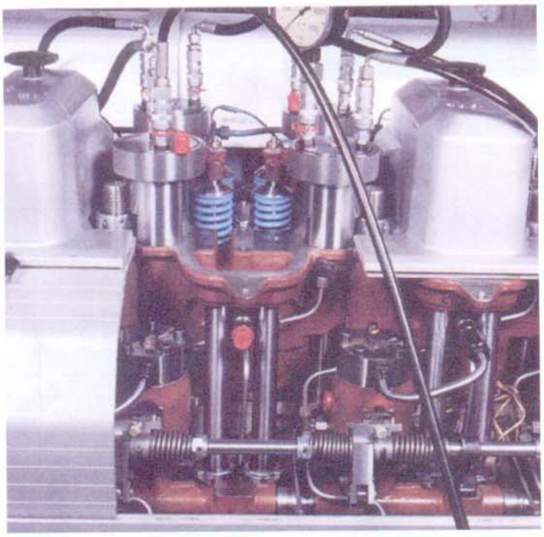
- The bolt is stretched within normal elasticity limits using a hydraulic jack. The correct oil pressure applies the exact elastic tension.
- The cylinder-shaped hydraulic bolt is simply hand-tightened with a small rod. No great force or heavy tools are necessary.
- The bolt is not under a torsion load. It does not have the inclination to loosen. The bolt does not have to be tightened with great force in the tap holes, hand-tightening is sufficient.
- The action, the tightening or loosening can be carried out in stages allowing the forces to remain stable on the bolts during assembly or disassembly.
- Hydraulic tensioning asks little time and requires less muscle. The hydraulic lubricating-oil pressure is supplied by a manual plunger pump and in larger systems by pneumatic or electric high-pressure pumps.



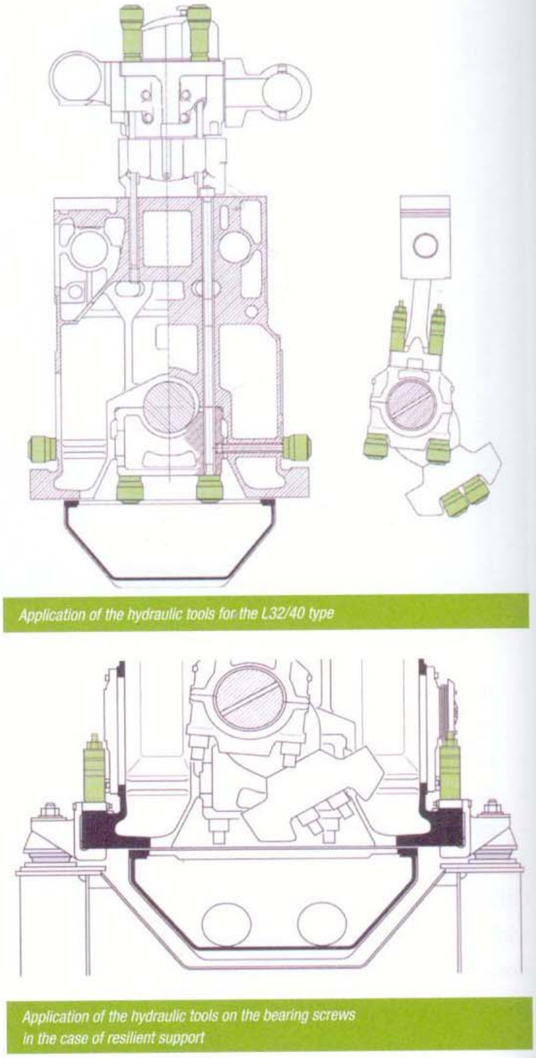
All the large bolts of this Wärtsilä 32 diesel engine category III are hydraulically tensioned.



▶ The hydraulic jacks installed on the two connecting-rod bolts and the vertical main bearing bolts of a Wärtsilä 20 diesel engine, category III, H.F.O.



▶ The four hydraulic jacks installed on the cylinder head of a Wärtsilä 20 diesel engine. The lines, the manometer, and the line (foreground) of the hand pump have already been mounted.



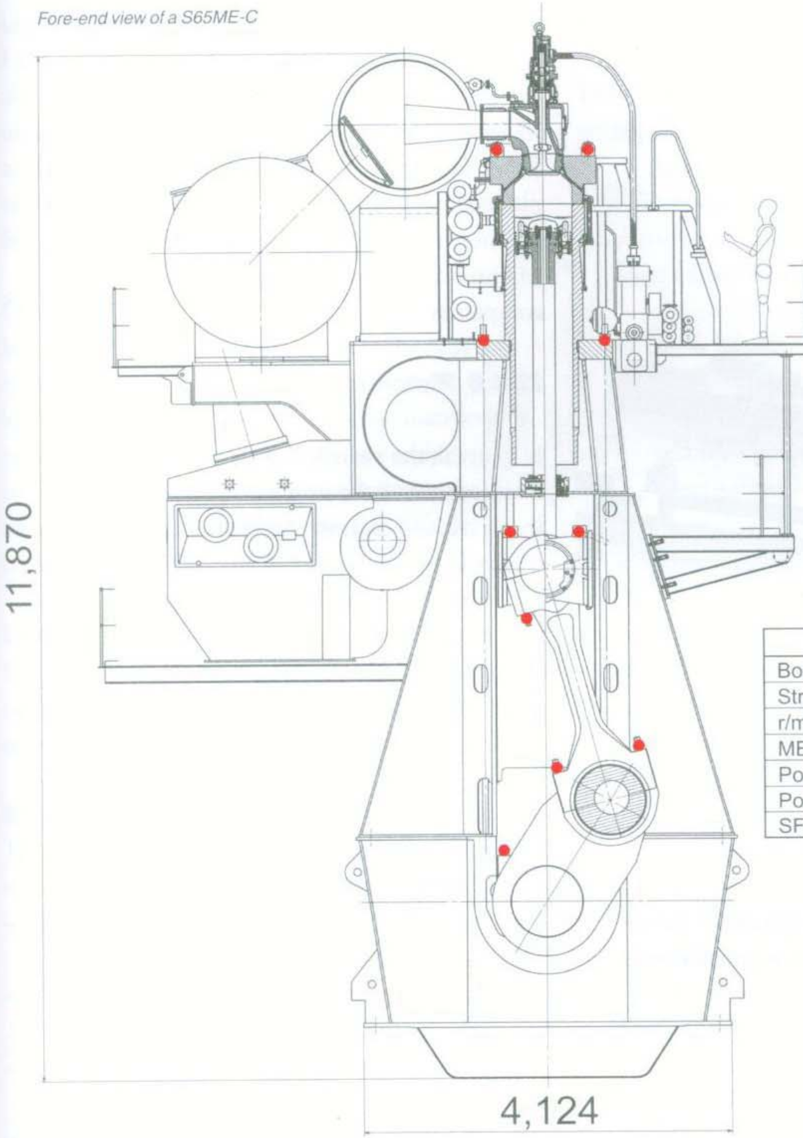
▲ The bolted connections of a MAN–B&W L32/40 diesel engine, category III.

The hydraulic jacks have been placed on all the hydraulically tensioned bolts (green).

▶ The 'loosening' of the connecting-rod big end from the crank pin.

1. *Place accessory between connecting-rod big end and bearing cap.*
2. *Loosen connecting-rod bolts.*
3. *Remove the nuts.*
4. *Hoist out the piston so that the crankpin bearing shells are released.*

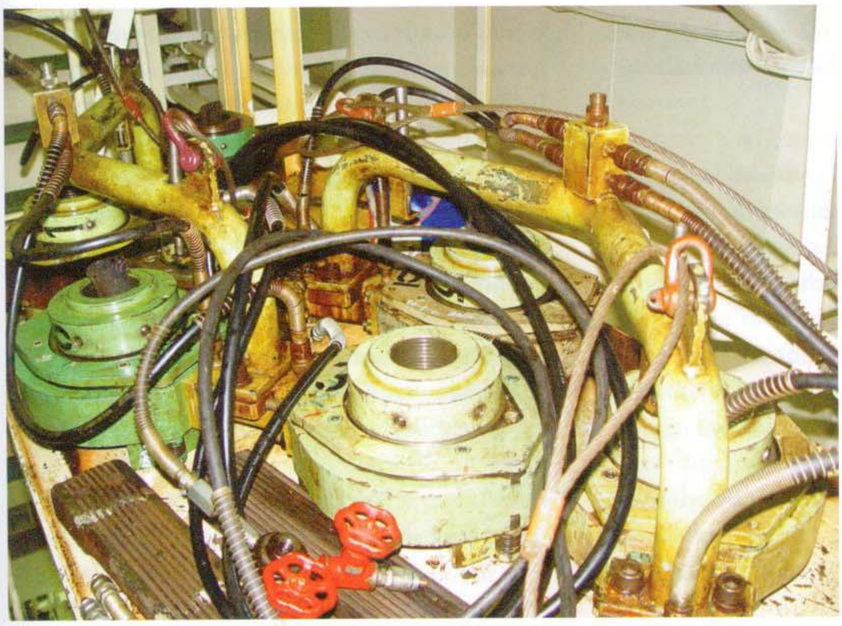
Fore-end view of a S65ME-C



7S65ME-C engine data	
Bore (mm)	650
Stroke (mm)	2,730
r/min.	92
MEP (bar)	18.5
Power/cyl.(kW)	2,570
Power (kW)	17,990
SFOC (g/kWh)	167

◄

Also with this new type of two-stroke crosshead engine, a MAN–B&W 7S 65 ME-C, all the larger bolts are hydraulically tensioned.



▲

Hydraulic tensioning device for the cylinder-head of a Wärtsilä Sulzer RTA 84 two-stroke crosshead engine, category IV.

All the tools are moved using an overhead crane.

▲

The four hydraulic jacks with hoisting equipment for a Wärtsilä 46 diesel engine category III.

27.4.2 Bolted connections

Many bolts for engine gears, as well as for cylinder heads and main bearings, are provided with rolled thread as opposed to cut thread. By 'rolling', that is, hydraulic deformation of the material, the steel-grain structure is not distorted and the bolt is stronger than a bolt with a cut thread.

▲
Hydraulic bolts with rolled thread.

The conical nut can only be turned with a steel pin.

27.4.3 Grinding and honing tools

In many marine engines and engines for power stations, tools are available for maintenance of engines.

27.4.4 Grinding and cutting tools

These are mainly used for the repair of valve seats in the cylinder head or for the other valve seats. One manufacturer of these tools grinds the seats while the other uses a cutting tool – a small chisel.

Honing tools

It is common procedure to have mobile honing tools available in the proximity of the engine. These are used to provide the cylinder liner with a good honing pattern after the piston has been removed.
Smooth areas, 'glazing', are removed ('deglazing'). It should be noted that liner wear is minimal, so it can remain in place for the full operating life of the piston.
Honing extends the running life of the cylinders.

27.4.5 Turbo blowers, fuel pumps, injectors and governors

In general, the maintenance of these parts entails either exchanging or reconditioning during planned major overhauls; progressively less maintenance is being performed on location. Maintenance is mostly executed by specialised companies.

27.4.6 Piston maintenance

In general, the vessel's own staff will carry out maintenance of the pistons. They are 'drawn', inspected and cleaned. Afterwards, the important dimensions are checked to ensure that the piston can be re-used or must be exchanged. This also is applicable for the gudgeon pin, the piston-pin bushes, and the piston rings. For two-stroke crosshead engines, also for the piston rod and the piston-rod stuffing box.

27.5 Maintenance for small engines – category I

This diesel-engine category contains numerous engines with few operating hours and they are normally lightly loaded. This is mostly the case with pleasure craft, such as yachts and motorboats, with some exceptions.

Maintenance

It is of utmost importance to carefully read the instruction manual. The long winter stops often lead to more damage caused by frost, moisture, and disuse than the wear caused by hundreds of engine operating hours in the summer season!

For adequate maintenance, the next points are of importance:
- Read the instruction manual thoroughly, all the normal activities are mentioned.
- Ensure that fuel, lubricating oil and cooling water/coolant is of good quality.

Fuel

A **fuel tank** should be left full in the winter stop. This prevents the build-up of condensate and eliminates the risk of water contamination of the fuel.
Use a good fuel filter and an oil-water separator. Fill the fuel tank with diesel oil at tank stations that have a regular supply-and-demand network.

Lubricating oil

Lubricating oil rarely reaches the number of authorised operating hours. Changing the couple of litres of lubricating oil every season is advisable as the water and dirt in the oil are also disposed of at the same time. Always renew the lubricating-oil filter.

Coolant

In a closed cooling system, use only the best refrigerant. Refrigerant must be kept at the correct level and exchanged according to the instructions. In an open coolant system or the secondary coolant system (untreated-water system), the 'sea inlet filter' must be regularly cleaned. Check, if possible, the coolant discharge.

Leaks

Stop fuel, lubricating-oil and cooling water/coolant leaks as quickly as possible; keep the outside of the engine clean and dry.

Small maintenance

This is often limited to the following points.

- Checking the valve clearance.
- Checking the fuel, lubricating-oil and coolant system, see above.
- Checking the starting system including starting motor and accumulator.
- Checking the dynamo, often belt-driven (correct tension).
- Checking the rotor of the coolant pump(s) and clean the algal filter.
- Checking the vibration dampers, the flexible couplings, and the propeller-shaft seals.
- Checking the lubricating-oil level in the high-pressure fuel pump.
- Checking the coolant pressure in the closed system. Checking the coolant for frost protection.

Testing the main engine

- Check that the engine is operating in a regular loop; if not, this is often an indication that a cylinder does not have proper combustion.
- Check pressures and temperatures.
- Check visible smoke in the exhaust gas after the engine is warm.
- Check vibrations.
- Check leaks.
- Check to see if the dynamo and accumulator are working.

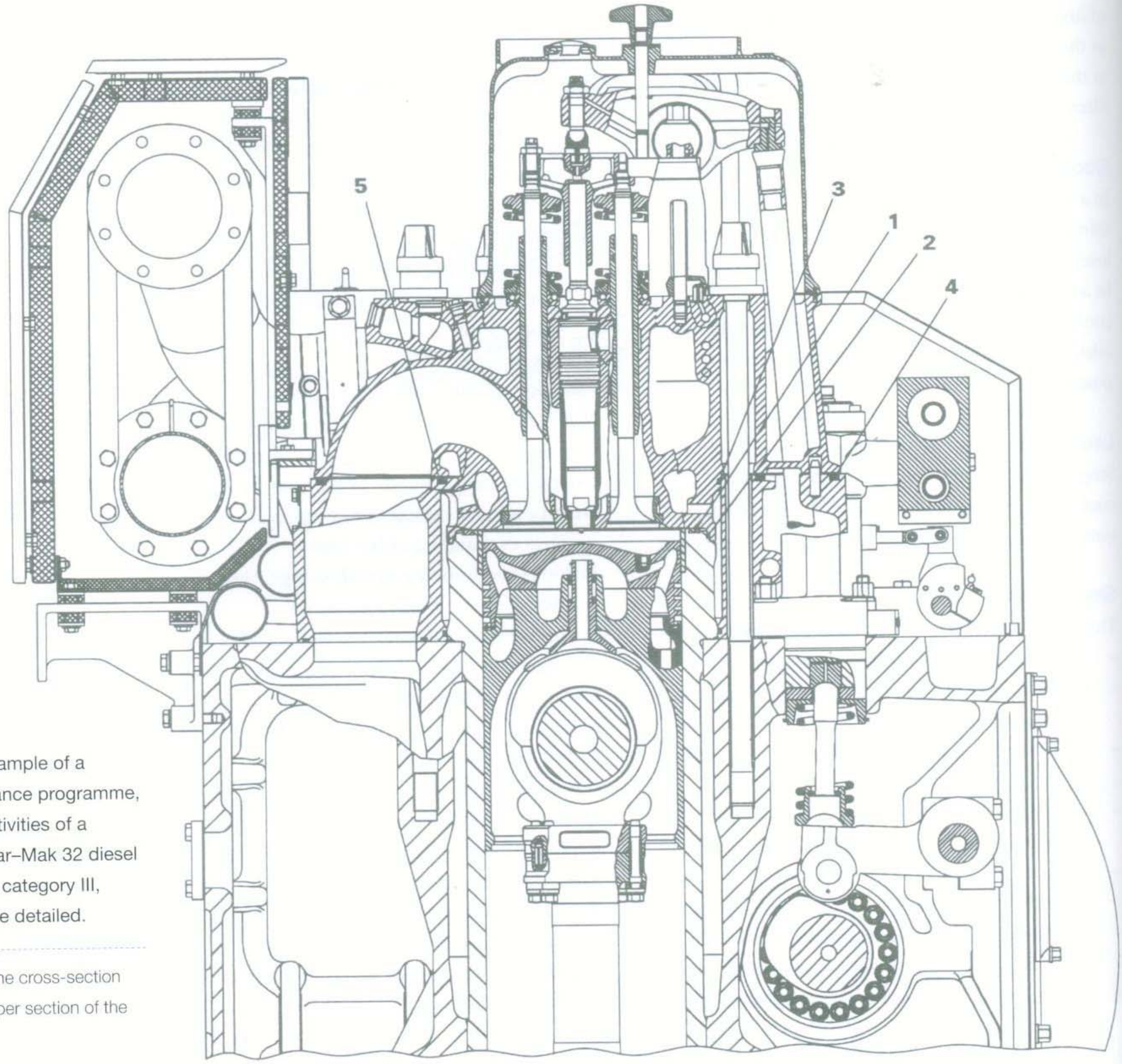
27.5.1 Major maintenance

This is mostly executed by specialised firms or the service department of the engine manufacturer.

27.5.2 Some maintenance activities

- **Reconditioning and cleaning the cylinder heads**
  - cylinder-head levelling
  - replaning the valve seats in the head
- **Reconditioning the injector(s)**
  - clean the injector nozzle (outer side)
  - test the opening pressure and the atomisation
  - check for fuel leaks
- **Honing the cylinder liner**
  When the cylinder liner shows some wear, honing with an optimal crosshatch pattern will extend the operating life of the liner. Serious wear or a loose cylinder liner in the block requires exchanging of the liner.
- **Piston reconditioning and cleaning**
  - renew piston rings, if required
  - measure the gudgeon-pin bushes dimensions and renew, if required
- **Check connecting-rod bearings and renew if required**
- **Check turboblower. If required, allow reconditioning to be performed by a specialised firm.**
- **Replace all packings, if required, for example, valve caps and crankcase doors.**
  Replace the copper rings of the injector and fuel lines.
- **Crankshafts have a long life as do the main bearings.**
  Only with serious wear, they must be disassembled. The crankshaft must be ground and over-dimensioned main bearings and crank-pin bearings must be adjusted.
- **Check governor and lubricating lines**
  - change the lubricating oil
- **High-pressure fuel pump**
  Change the lubricating oil regularly. This lubricating oil is often very thin! This indicates fuel leaks, which are not a major problem if the lubricating oil is regularly changed.
- **Crankcase**
  Clean the crankcase well with lint-free cloth. Remove all small impurities. Carefully check all nooks and crannies using a lamp.

27.6 Examples of maintenance for engines category III



As an example of a maintenance programme, some activities of a Caterpillar–Mak 32 diesel engine – category III, H.F.O. are detailed.

Shown, the cross-section of the upper section of the engine.

27.6.1 Crankshaft clock gauging

Ensure that the heart line of the crankshaft is in line.
Goal: to avoid material fatigue in the crankshaft: this could result in crankshaft breakage.
Cause: wear of the main bearing or the deformation of the engine frame (large engines).
Method crankshaft clock gauging: essentially, a dial gauge is placed in two standard centre holes between the crankshaft webs. The position of the crankshaft heart line is obtained by setting the dial gauge to zero for every cylinder in a certain piston position. Then progressively per cylinder, measurements are taken in four different positions. The dial gauge 'results' must always be within certain values.

Crankshaft clock gauging is performed for the following objective.

- As a maintenance check.
- After a collision.
- If it is suspected that the engine foundation is deformed.
- With bearing damage.
- Before and after dry-docking.

M32

Operating hours	Vessel	Date
Engine Type	Engine no.	Engineer/Service Station

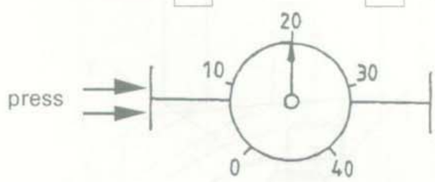
Engine cold ☐

Shut down for ☐ hours

Preheated ☐ °C

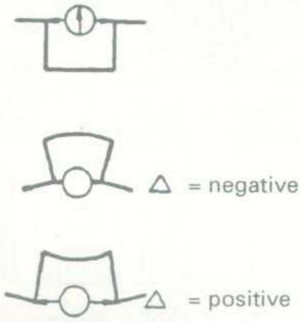
Indication of Dial Gauge

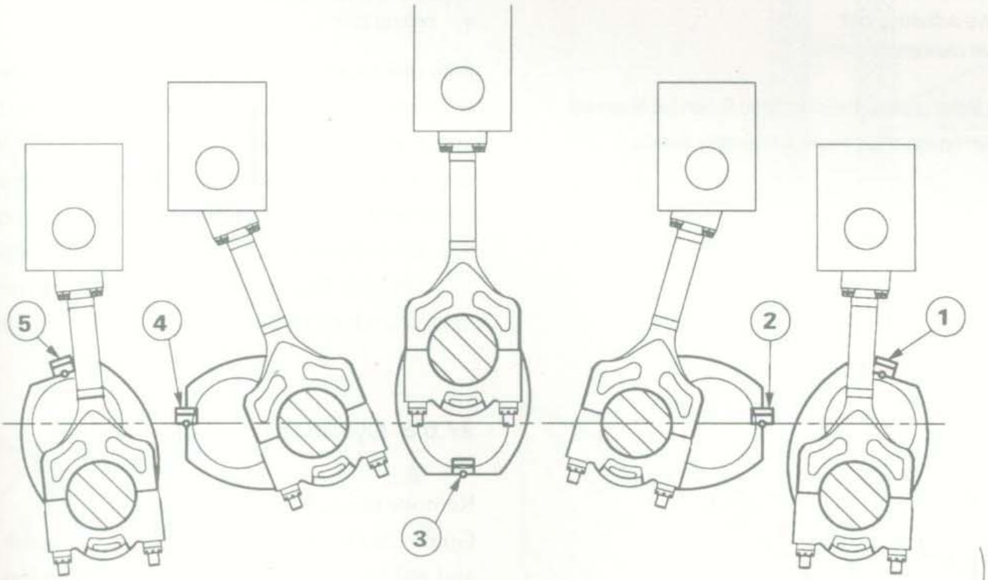
down ☐ ☐ up



Draft forward: Draft aft:

Measuring point	Cylinder 1	2	3	4	5	6	7	8	9
1	20	20	20	20	20	20	20	20	20
2									
3									
4									
5									
△ max									





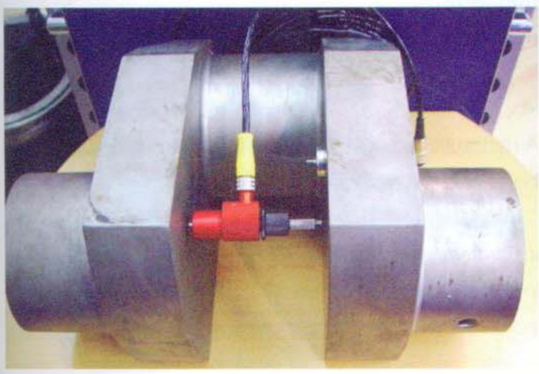
◄ A measuring table for the crankshaft position of a Caterpillar–Mak 32 engine. Shown, a nine-cylinder engine.

negative: smaller distance between the crank webs

positive: larger distance between the crank webs

In the lower drawing, the measuring positions are clearly shown.

As the connecting rod is an obstacle, measurements are taken once on the one side, position 5, and once on the other side, position 1.



◄ There are also electronic crankshaft deflection devices that can directly store the measured values in a file.

27.6.2 Valve clearance check

Checking the clearances in the driving mechanism of the inlet and exhaust valves.
Goal: Valve clearance ensures that the valves are **completely** closed if this is required for the process. The clearance is often between 0.2 and 1.2 millimetres dependent on the type and size of the engine.

The exhaust-valve clearances are usually larger; the exhaust valve is usually warmer than the inlet valve and therefore expands more.
There are different methods for cold and hot engines.
Method: A feeler gauge is used to measure the clearance between both drive parts.

▸
Measuring the valve clearance for the Caterpillar–Mak 32 diesel engine.

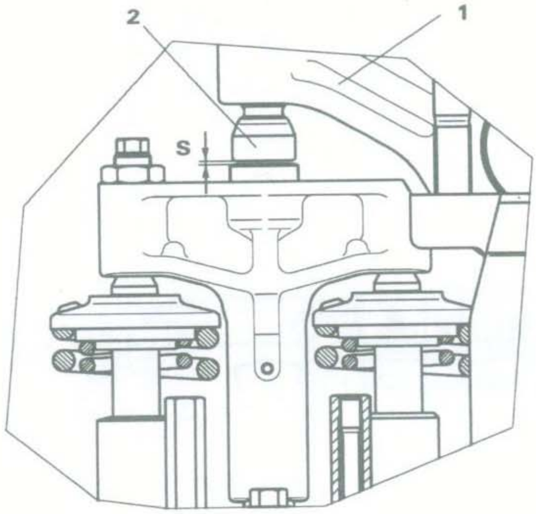


Fig. 1

In figure 1:
1 *valve-rocker arm*
2 *valve adjusting bolt*
S *valve clearance*

Using a feeler gauge, the clearance S can be checked. The feeler gauge must move reasonably heavily.

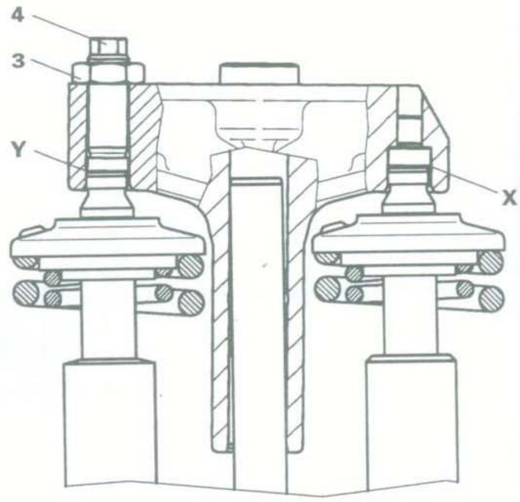


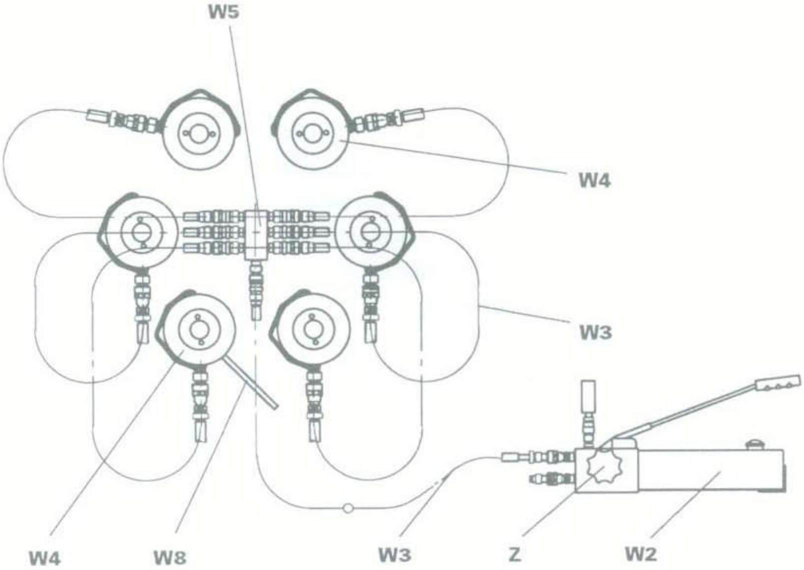
Fig. 2

In figure 2:
3 *counter nut*
4 *setting bolt*

If the clearance is not correct, the counter nut is loosened and using the setting bolt, the correct clearance can be set. Ensure when tightening the counter nut that the setting bolt does not rotate. Always recheck the clearance after any adjustments! Always press both valve yokes down so that there is no clearance.

27.6.3 Cylinder-head dismantling

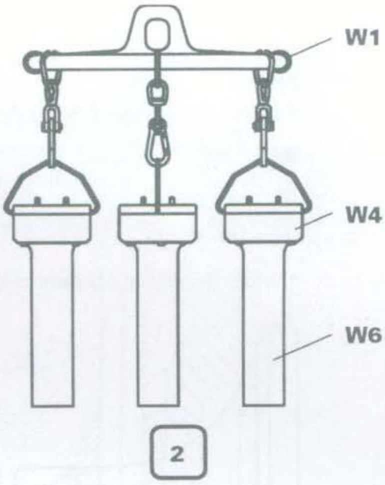
Remove the cylinder head.
Goal: Check the cylinder head, such as inlet and exhaust valves and seats, coolant spaces, contamination of the bottom side of the head and fuel injectors. Dismantle the cylinder head, so the piston and liner can be dismantled.

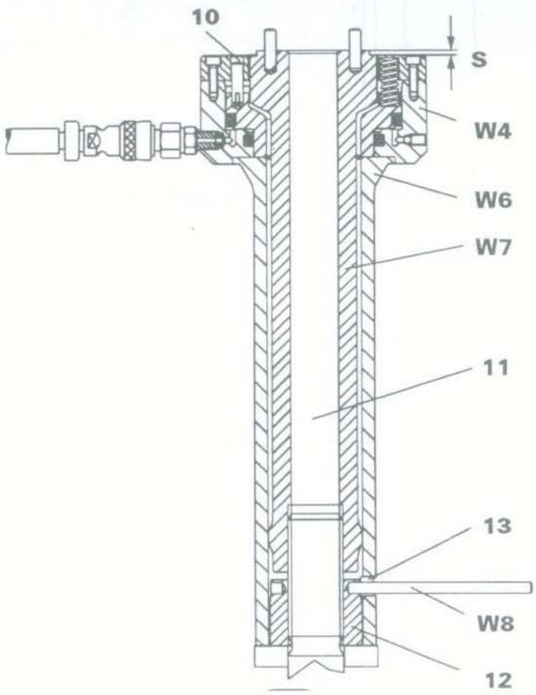


◂
A hydraulic jack set ('spider') for the six cylinder-head bolts of a Caterpillar–Mak 32 diesel engine.

In this case, the oil pressure is supplied using a hand pump. In stages, the pressure over all six hydraulic jacks can be increased until the 'hydraulic' nuts on the head bolts are released so that the cylinder head can be removed.

Method: Small engines in categories I and II have bolted connections, which can be manually loosened. Cylinders with a cylinder diameter from 200 millimetres in the categories II and III have hydraulically tensioned bolts and hydraulic jacks are used.



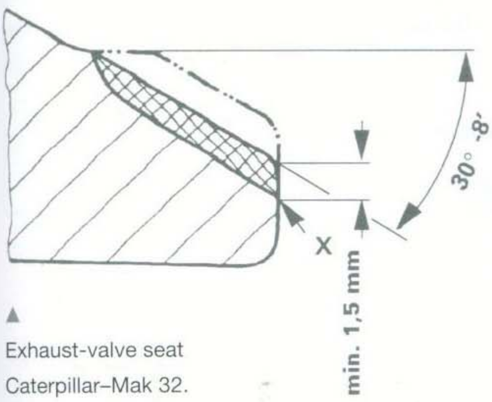


▶
Upper drawing: the hydraulic jacks hanging on a spreader and hoisted above the cylinder with an overhead crane.
Lower drawing: a cross-section of the relatively long hydraulic jack for the cylinder head bolts.

The oil is supplied from the upper left. Outside pipe, W 6, is on the cylinder head. Inside pipe, W 7, is screwed on the rolled thread of the cylinder head bolt and due to the high oil pressure, is moved slightly upwards. As a result, the bolts are slightly stretched and the cylinder-head nut 12, with a pen W 8, can be rotated upwards ('eight holes'). When the oil pressure is removed, they remain free of the cylinder head. The head can be removed.
Working pressure of the oil system is 850 bar.
13 grooves in outer pipe
10 venting
S measured distance of the screwed inside pipe, W 7

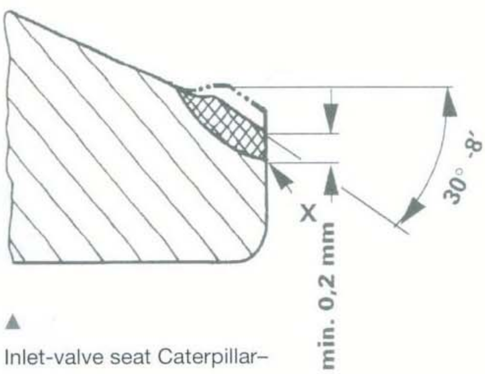
Inlet and exhaust valves

These are cleaned and measured and then the welded valve seats must be ground to the correct angle.



▲
Exhaust-valve seat Caterpillar–Mak 32.

The welded section must have a minimal height of 1.5 mm after grinding. The angle must be 30 degrees 8 minutes.



▲
Inlet-valve seat Caterpillar–Mak 32.

The welded section must have a minimal height of 0.2 mm after grinding. The angle must be 30 degrees 8 minutes.

A measuring table for the valves of the Caterpillar–Mak 32 diesel engine.

Amongst others, measurements are taken:
- if the valve stem is bent: A–B;
- to find the thickness S, the burn section caused by high temperature corrosion. S is a maximum of 1.5 mm.

M32

- when the deviation from the straightnesss does **not exceed** the following values (**Fig. 1**).

Measuring range "s" = **390 mm**

A B = contact points for test device

- when the radial runouts of the valve cone do not exceed the following values:

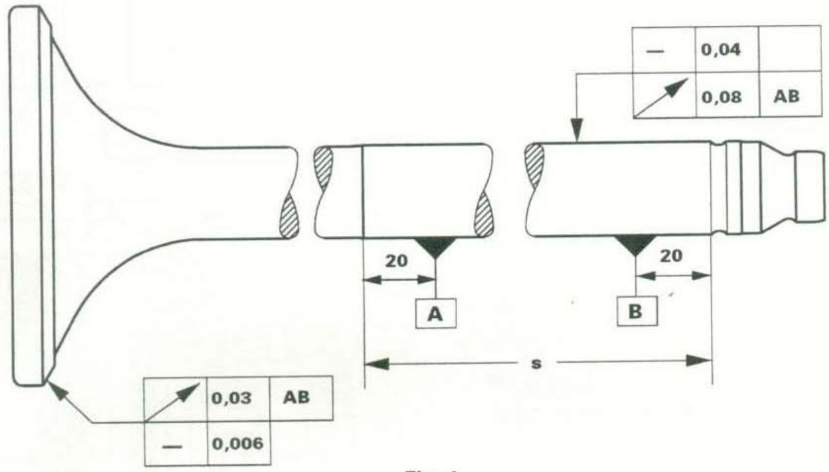
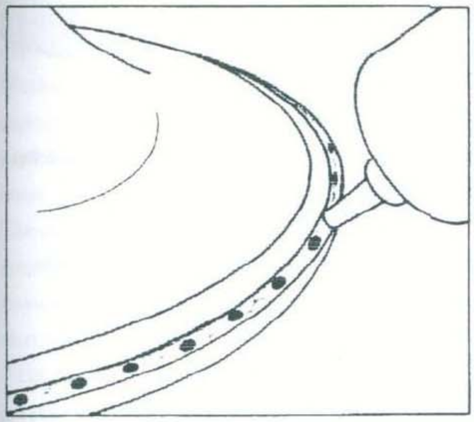


Fig. 1

1.2 The valves cannot be reused in the following cases

- the valve cone face is damaged (cracks, blowholes)
- the wear caused by corrosive material exceeds
  > 2 %
  of the valve head diameter (compare with new valve)
- concave burns "s" caused by high temperature corrosion on the underside of the valve head exceeds
  > 1.5 mm (Fig. 2)
- the underside of the valve head indicates severe pitting (formation of so-called "paving stones")
- corrosion pits and mechanical damage in zone of the valve stem, the stem transition and valve head
- excess of limit dimension for refinishing.

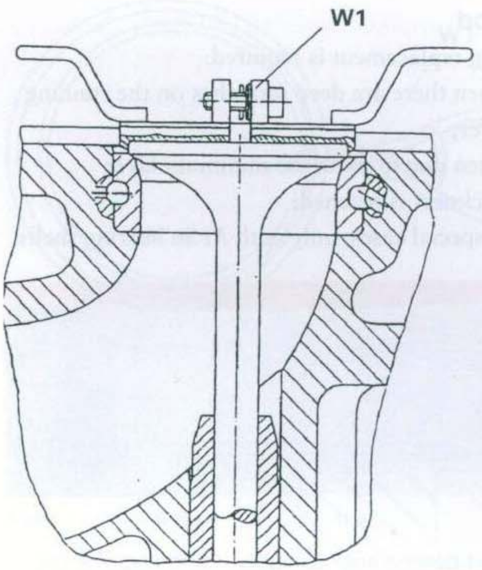
Fig. 2

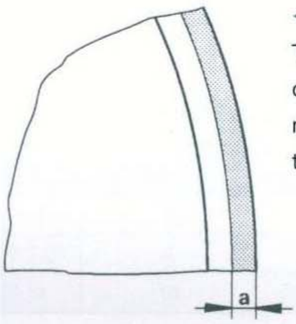


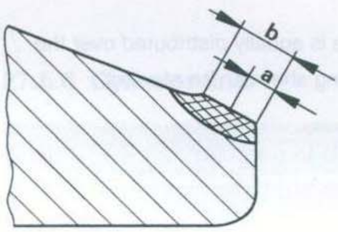
▲
Checking the valve-seat surface.

- Sand lightly by hand.
- This is done with a very fine grinding compound (Dp 30/10 – 15 microns).
- After application of the grinding compound, a solvent is sprayed over the paste to improve the polishing process (see left figure).

The handgrip is placed on the valve. The valve stem is oiled and positioned in the valve guide. After this, the valve seat is lightly polished for a very short time. The contact surface of the valve with the seat is between 30 to 50% of the seat width b.







◄
The polished outer ring of the valve seat is a maximum of 50% of the total width.

27.6.4 Check of bearing shells

Goal: Detect damage and wear.
Comment: Contaminated lubricating oil, solid contaminants, and water are the cause of 85% of all bearing damage!

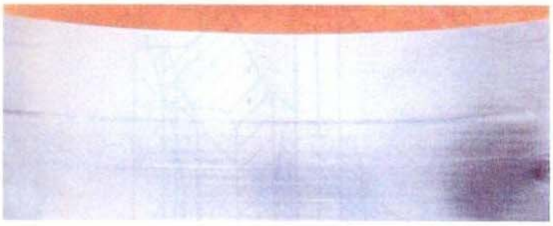
A bearing shell of a medium-speed Caterpillar–Mak 32, comprises two layers: a 10 mm-thick steel support layer and a 0.7 tot 1.1 mm-thick running layer consisting of a lead/bronze or aluminium alloy.

◄
The arrangement of the bearing shells of the Caterpillar–Mak 32 diesel engine.

The steel shell itself is approximately 10 mm. The lead bronze or aluminium tin layer is between 0.7 and 1.1 mm. The anti-corrosion layer is very thin: 0.003 mm.

Method

Bearing replacement is required:
- when there are deep scratches on the running layer;
- when due to wear, the minimum layer thickness is reached;
- in special cases, only with Al Sn bearing shells.

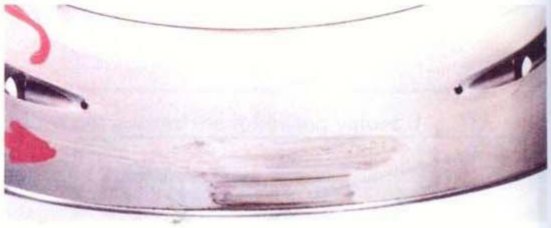


▲
The load-bearing area is equally distributed over the entire running layer. The dirt in the lubricating oil has had no negative influence on the bearing reliability. The bearing shell can be re-used.

▲
The load-bearing area is equally distributed over the entire layer. The bearing shell can be re-used.

▲
The load-bearing area is equally distributed over the entire layer. The circumferential scratches are caused by very fine solids, such as metal particles and sand in the lubricating oil.

Given that the depth of these scratches cannot be measured or felt, the bearing shell can be re-used. If the number, the depth and the size of the scratches negatively affect the lubricating-oil film build-up, then **both** bearing shells must be renewed.



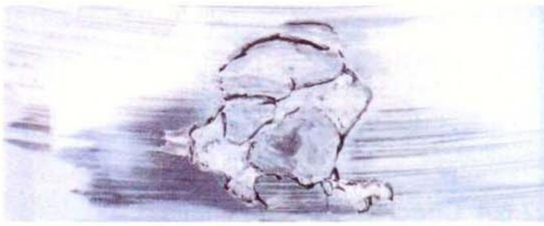
▲
The running layer shows a heavily overloaded bearing with fretting corrosion in the aluminium alloy.

This bearing shell cannot be re-used. Both bearing shells must be replaced. The cause of this damage must be investigated and resolved.

▲


The load-bearing area shows damage of the bearing in a large surface area, with in certain parts, fretting corrosion.

This bearing shell cannot be re-used, renew **both** bearing shells.
The cause of this damage must be investigated and resolved. Check the other bearing shells.



▲
The heavily loaded section of the bearing shows the stripping of bearing material, probably by overloading.

Stripped bearing material can be found in the area. Clean! This bearing shell cannot be re-used; both bearing shells must be renewed. The cause of this damage must be investigated and resolved.
Check the other bearing shells.

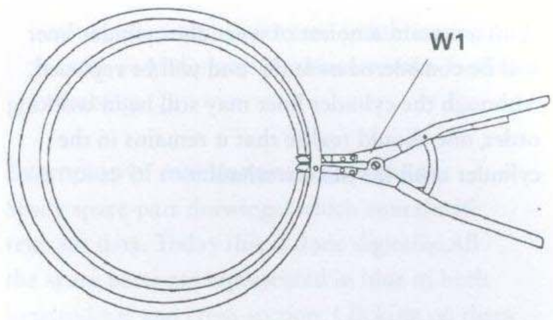
27.6.5 Piston 'withdrawal'

Goal: Check the condition of piston, piston rings, cylinder liner, gudgeon pin, connecting rod, connecting-rod bush, and connecting-rod bearings.
Method: Loosen the connecting-rod bolts at the connecting-rod division when the piston is at top position and attached to an overhead crane with a hoist strap.
The anti-polishing ring is then removed with a special tool.
Possible carbon deposition on the cylinder liner is also removed.
If the above two steps are not executed, the (sharp) piston rings could stick behind the ring!
After this, the piston rings are removed from the piston using piston ring pliers and the piston crown is dismantled. The inside of the piston crown can now be inspected for contamination.

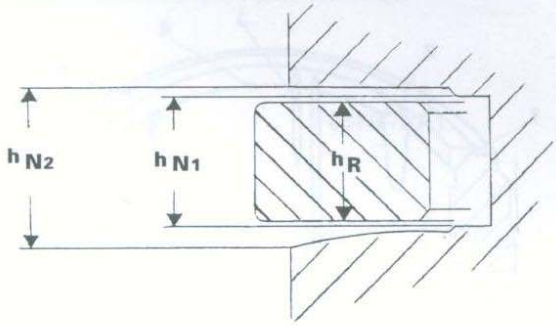
Comment

When the lubricating oil used for cooling the piston overheats, the internal piston surface area can become caked with carbon residues. This impedes the heat transfer from the piston crown to the lubricating oil, causing overheating of the piston. The pressure shocks occurring in the combustion process can load the piston crown to such an extent that the piston crown fails causing huge damage to the piston and its surroundings.
All the parts are cleaned.
The surface areas are checked for damage and all the important dimensions are measured.
Any possible rejected parts, such as piston rings, are replaced.

As a matter of course, the piston pin and the connecting-rod bushes are measured to find possible wear. The rejected connecting-rod bushes are removed in the engine factory and with nitrogen (shrinkage), a new bush is attached to the connecting-rod eye. This can also be done with CO_2 'snow'. All the bolted connections of the piston and connecting-rod eye are tightened with an approved torque spanner.



◄ Piston ring pliers used for the simple removal of the piston rings. The risk that the rings will break with the use of these pliers is small.



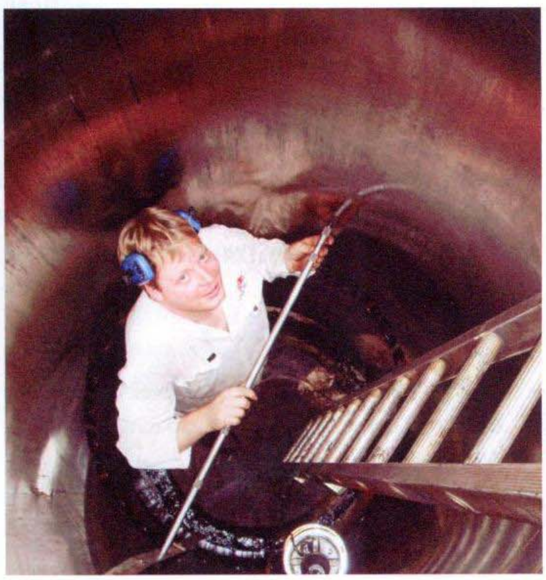
◄ Measuring the piston-ring groove in two places and the thickness of the piston ring..

Groove	Ring thickness h_R Nominal mm	Groove width h_{N2} Wear limit mm	Limit $h_{N2} - h_R$ mm
1	8	8.45	0.5
2,3	6	6.45	0.5
4	10	10.2	0.3

◄ A measuring table for the piston-ring grooves and the ring height for all the four rings.

27.6.6 Cylinder liner

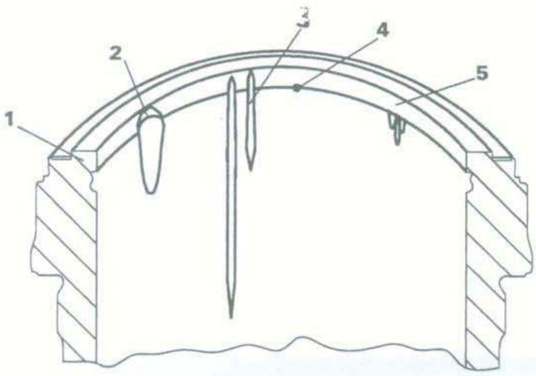
Goal: Check the condition of the cylinder liner, such as, wear, crosshatch pattern, cracks, and scratches.
Method: Both installed in the bed and disassembled, the diameter of the liner can be measured at different heights, in length and in width (fore and aft direction as well as abeam). Most wear is found on the upper side.



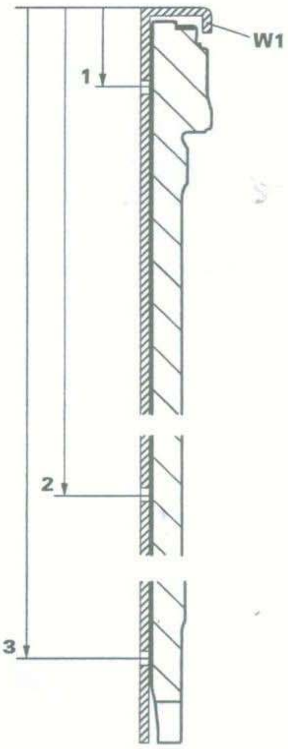
► Measuring the cylinder diameter for a two-stroke crosshead engine, a Wärtsilä Sulzer RTA 96 C.

The diameter is measured in various places of the cylinder liner to determine the wear. To position the caliper gauge correctly, use is made of steel strips with holes suspended at various heights in the liner. See next figure.

With a certain amount of wear, the cylinder liner will be considered as faulty and will be replaced. Although the cylinder liner may still be in working order, one should realise that it remains in the cylinder until the next overhaul.



Different types of scratches on the running layer of the cylinder liner. For every type of scratch, the maximum dimensions and number are defined in relation with any possible rejection of the cylinder liner.



Two measuring strips are placed in the cylinder so that the wear can be measured at three different heights by placing the caliper gauge in both of the holes. The measuring strips must be mounted exactly opposite each other in the cylinder.

27.7 Some examples of maintenance of large two-stroke crosshead engines

27.7.1 Wärtsilä Sulzer RTA 96 C – Engine category IV

In this very large engine, hardly any parts can be lifted by hand, the weight of the spare parts are impressive. Some of these very heavy parts are given as an example.

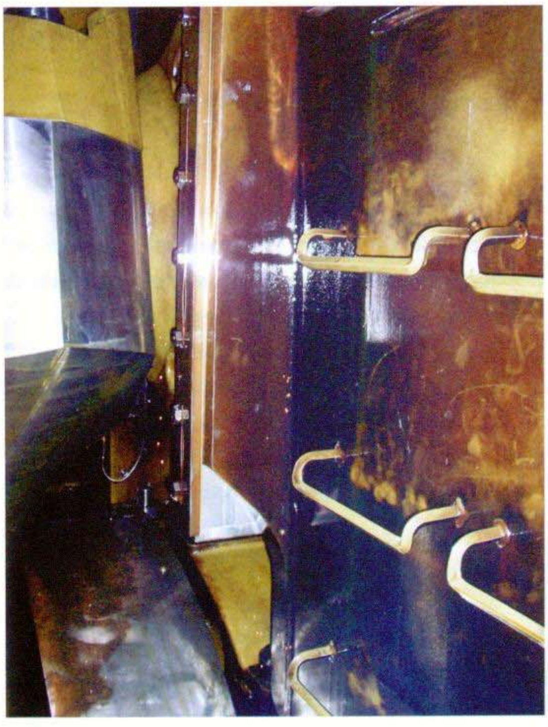
bottom-half six-cylinder crankcase	118,000 kg
main bearing shell	184-234 kg
bare A-frame	94,000 kg
tie rod	2210 kg
tie rod nut	41 kg
cylinder liner	9006 kg
cylinder cover complete	10,442 kg
exhaust valve complete with casing	2909 kg
exhaust valve	236 kg
crankshaft six-cylinder	186,000 kg
crankshaft twelve-cylinder two parts	359,700 kg
fuel cam	253 kg

This entails the following:

- Almost all the parts must be moved using an overhead crane with a hoist capacity of 10 to 15 tonnes.
- All the large bolted connections are hydraulically tensioned.

Thread diameters in millimetres

foundation bolts	M 64
foundation bolt	M 180 x 6
cylinder-head bolts	M 110 x 6
exhaust-valve casing	M 110 x 6 T
piston rod, on crosshead	M 110 x 6



The treads in the crankcase of a large two-stroke crosshead engine. Slipping during activities can be life threatening.

- The requisite hydraulic jacks are very heavy and moved by a crane.
- Manual inspection is inadequate; only by measuring one can ascertain if a part is in the correct place.
- Working with heavy parts requires a lot of attention, safety is very important. The possibility of permanent injury is always present!
- The crankcase and the scavenging-air space are large enough to walk around in; especially the crankcase, which is covered in lubricating oil, can be very slippery. Permanent stairs facilitate climbing.

Examples of maintenance schedules

Study spare-part drawings, which contain all relevant data. Today this is done digitally. All the spare parts are represented in blue in both longitudinal and cross-section. Clicking on them provides detailed spare-part drawings.

SULZER **RTA96C** Maintenance **0380–1/A1**

Maintenance Schedule
Inspection and Overhaul Intervals (Guidelines)

Component	Work to be carried out	Group	Intervals
Group 0			
Lubricating oil ...	– Laboratory analysis	0900–1	3'000 op. hours (= operating hours)
Cooling water ...	– Check concentration of inhibitor (as per supplier's instructions)		
Group 1			
Bedplate	– Check pre-tension of foundation bolts .... (first time after 1'500 operating hours)	1112–1	every 2 years
Main bearing	– Check pre-tension of waisted stud, if necessary re-tension (first time after 1 year)	1132–1	15'000 ÷ 20'000 op. hours
	– Remove main bearing upper half for inspection	1132–2	acc. to class. society
	– Remove bottom bearing shell for inspection	1132–1	as required
Thrust bearing ..	– Check axial and radial clearance	1203–1	6'000 ÷ 8'000 op. hours
	– Check bottom drain for free passage	1203–1	6'000 ÷ 8'000 op. hours
	– Remove thrust bearing pads for inspection ..	1224–1	acc. to class. society
Tie rod	– Check pre-tension, if necessary re-tension .. (first time after 1 year)	1903–1	every 4 years
Group 2			
Cylinder liner	– Establish wear in bore (in fitted condition) ...	2124–1	at every piston removal
	– Remove cylinder liner	2124–2	as required
	– Replace O-rings	2124–2	at every removal
	– Water guide jacket, replace O-rings	2124–2	at every removal
	– Grind off wear ridge in bore	2124–3	at every piston removal
	– Recondition scavenge ports	2124–3	as required
	– Reshape lubricating grooves	2124–3	as required
Lubricating quill and accumulator	– Check function and tightness	2136–1	at every piston removal
	– Replace O-rings	2136–1	at every liner removal
	– Check diaphragm of accumulator	2136–1	15'000 ÷ 20'000 op. hours

◄ A small section of the maintenance schedule for the largest two-stroke crosshead engine of Wärtsilä, the 'Sulzer' RTA 96 C.

Activities are often combined. Clicking the blue code numbers in the digital programme provides detailed information on the part. There are six pages in total.

Checking the main bearing shells

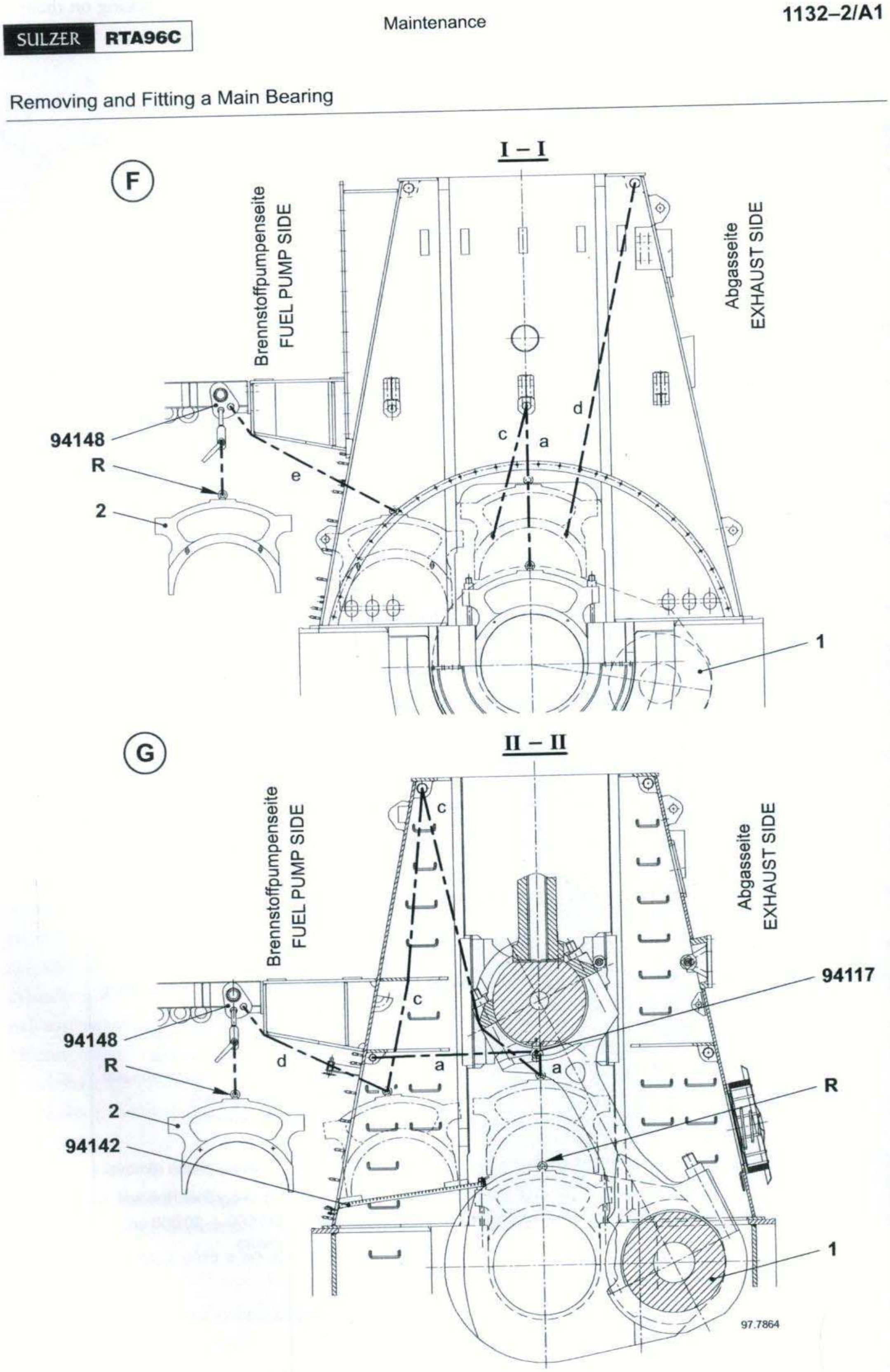
The bearing-cap bolts are loosened hydraulically. The bearing cap is hoisted from the crankcase onto a crane trolley outside the engine using a hoisting point in the crankcase. After this, the upper main bearing shell is hoisted out of the crankcase.

Subsequently the bottom shell, with the use of an attachment and the turning motor, is rotated around the crankshaft so it lies on top of the crankshaft, and can be hoisted out of the crankcase.

Now, the shells are inspected and the thickness is measured in various places. If the shells are accepted, the procedure is done in reverse order.

▸ Hoisting the bearing cap of the main bearing out of the crankcase.

Various hoisting points can be found inside the crankcase; outside the crankcase, hoist beams and a trolley.

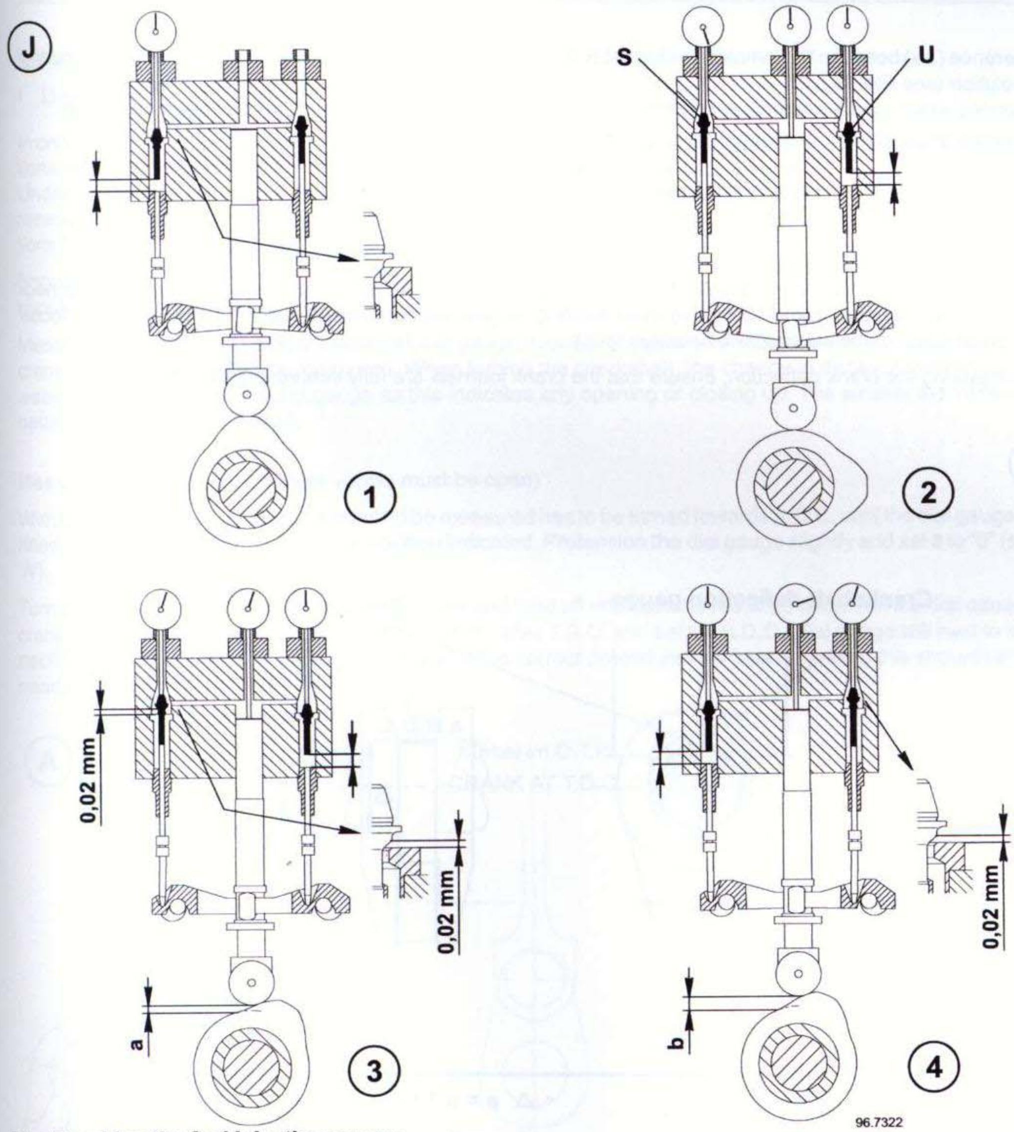


Adjusting and checking the V.I.T.-system of the high-pressure fuel pumps

This extensive procedure involves three dial gauges being placed on the suction valve, the overflow valve, and the high-pressure plunger. The start of injection, the end of injection and the effective plunger stroke are measured. In this procedure, the camshaft must be progressively re-positioned by turning the engine.

SULZER **RTA96C** Maintenance **5512–1/A1**

Setting and Checking the Control with Load-dependent V.I.T. (Variable Injection Timing)



D) Checking the fuel injection pumps

The same setting values as given for the injection pump setting must be used.
Carry out preparations in accordance with section B).

Checking the begin of injection, end of injection and of the effective plunger stroke

The description below applies to **AHEAD rotation.**

1. Turn the engine until the cam roller of the fuel pump to be checked rests on the peak of the cam. Fit the dial gauge with about 1mm pre-tension over the now closed suction valve (S) and set it to '0' (see Illustr. 'J', Fig. 1).
2. Turn engine ASTERN, until the cam roller rests on the base circle of the cam. Fit the dial gauges over the plunger as well as over the now closed spill valve (U) and adjust to '0' with a slight pretension (see Illustr. 'J' Fig. 2).

◄ Checking and adjusting the fuel pumps.

An extensive digital manual ensures that the procedure can be easily followed.

Measuring crankshaft deflection

Crankshaft deflection is measured at least once a year. When doing so, the ship must be afloat. Crankshaft deflection must be measured as soon as possible after a ship has run aground.
After changing the main bearing shells, the deflections have to be measured, if possible, within a period of 100 operating hours.

The measurements are performed for five positions of the crankshaft. From a starting position – the same for every cylinder – the dial gauge is set to zero and then read for every position. The readings can be either positive or negative (larger of smaller).
For very large crankshafts, the deviations are tenths of a millimetre. The maximum deviation for these engines is 0.66 mm!

▶
Measuring crankshaft deflections.

3103–1/A1 Maintenance **SULZER RTA96C**

Measuring Crank Deflection

The difference (Δa) between the indicated values at B.D.C. and T.D.C. shows the amount of crank deflection during one revolution (see Fig. 'B').
All influences which cause a temporary deformation of the hull and/or the engine affect the crank deflection.

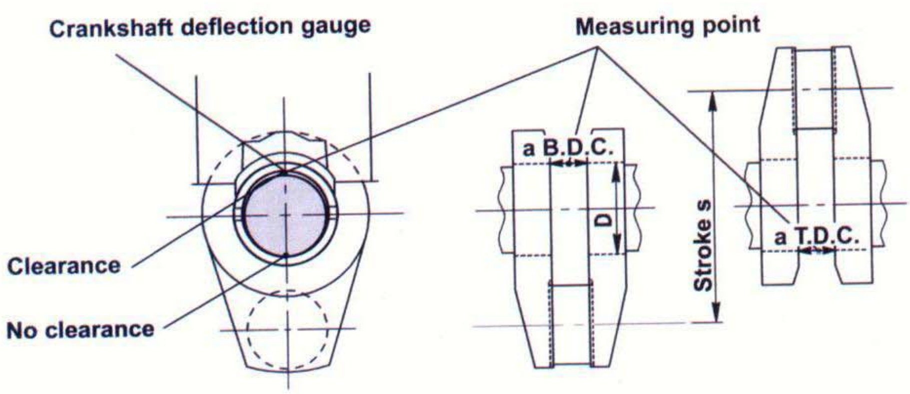
These are for example:
- loaded condition of the ship
- engine cold or at service temperature
- differing air/water temperatures
- strong sunshine

The values of Δa given below are valid for all conditions independent of outside influences.

Where values are measured which lie over the maximum permissible limits, the cause has to be found and the necessary remedial measures taken (defective main bearing, engine support altered due to hull deformation, loose holding-down bolts, defective shaftline bearings, etc.).

When measuring the crank deflection, ensure that the crank journals are fully seated in their bearings.

B



Difference	**Δ a = a T.D.C. – a B.D.C.**
Good values	**Δ a = 0 ÷ 0.33 mm**
Values still admissible	**Δ a = 0.33 ÷ 0.66 mm**

SULZER **RTA96C** Maintenance **3103–1/A1**

Crankshaft
Measuring Crank Deflection

Tools:

1 Dial gauge 94305

In order to ascertain whether the axes of the crank journals deviate from the theoretical shaft axis, the crank deflections must be measured. When doing so the ship must be floating freely.
Under normal circumstances, it is sufficient to measure the crank deflections once a year. They must, however, be measured as soon as possible in case the ship has grounded or after replacing the main bearing shells. The deflections have to be measured again approx. 100 service hours after replacing the bearing shells.

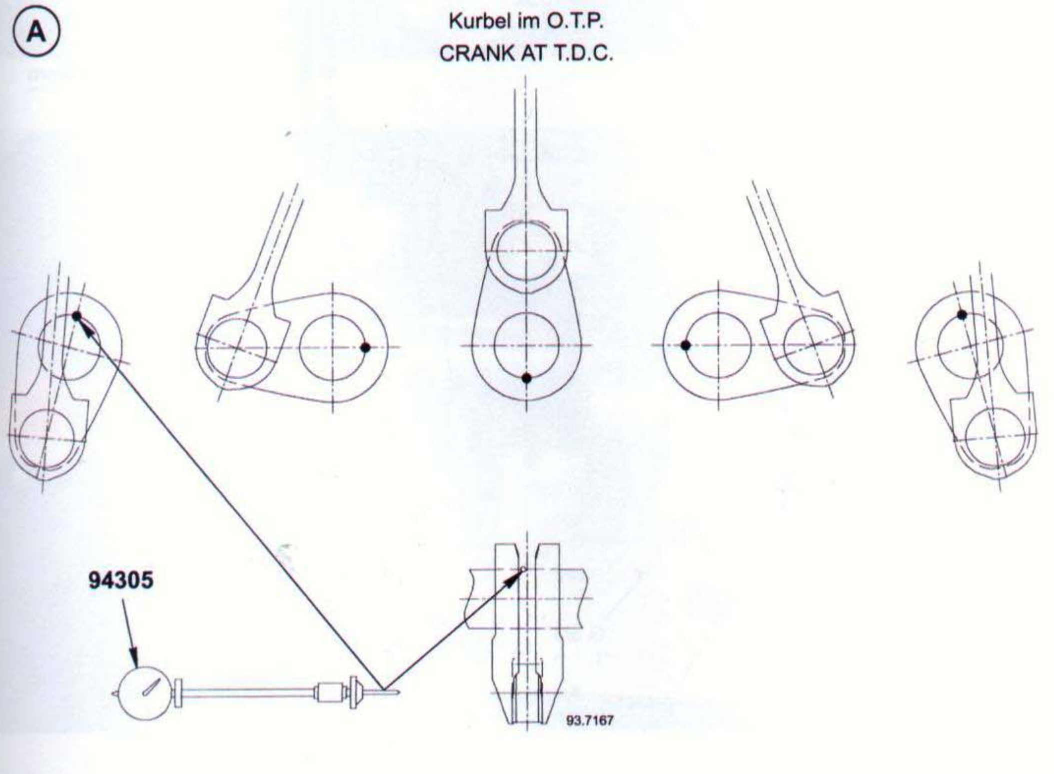
Should signs of damage of the main bearings be found during inspection of the crankcase, the crank deflections should also be measured.

Measuring is carried out using a crankshaft dial gauge (tool 94305) which, for this purpose, is inserted between the crank webs of the crank to be measured. When turning the crankshaft, the change in distance between the crank webs can be read from the dial gauge as this indicates any opening or closing up. The smaller the variations the better the position of the shaft.

Measuring procedure (indicator valves must be open)

With the running gear in place, the crank to be measured has to be turned towards B.D.C. until the dial gauge can be fitted next to the connecting rod at the position indicated. Pretension the dial gauge slightly and set it to "0" (see Fig. 'A').

Turn the crankshaft AHEAD with the turning gear and read off and note the readings shown on the dial gauge at the crank positions 90° before T.D.C., at T.D.C., 90° after T.D.C. and before B.D.C. (dial gauge still next to the connecting rod). The last value serves as a check. If the correct procedure has been followed, this should be back at nearly "0".



◄ The different positions for measuring the crankshaft deflections.

Dismantling or assembling an S.K.F.-shaft coupling of the camshaft

The camshaft of this engine consists of various parts. The shaft couplings and cams are hydraulically attached. When repairing or changing a cam, a part of the shaft is first removed from the cam casing so that the cams can be dismantled.

▶ Dismantling and assembling an S.K.F.-shaft coupling of the camshaft.

The blue codes provide information about the tools (digital info).

SULZER **RTA96C** — Maintenance — **4203–4/A1**

Camshaft

Removing and Fitting of SKF Shaft Coupling

Tools:

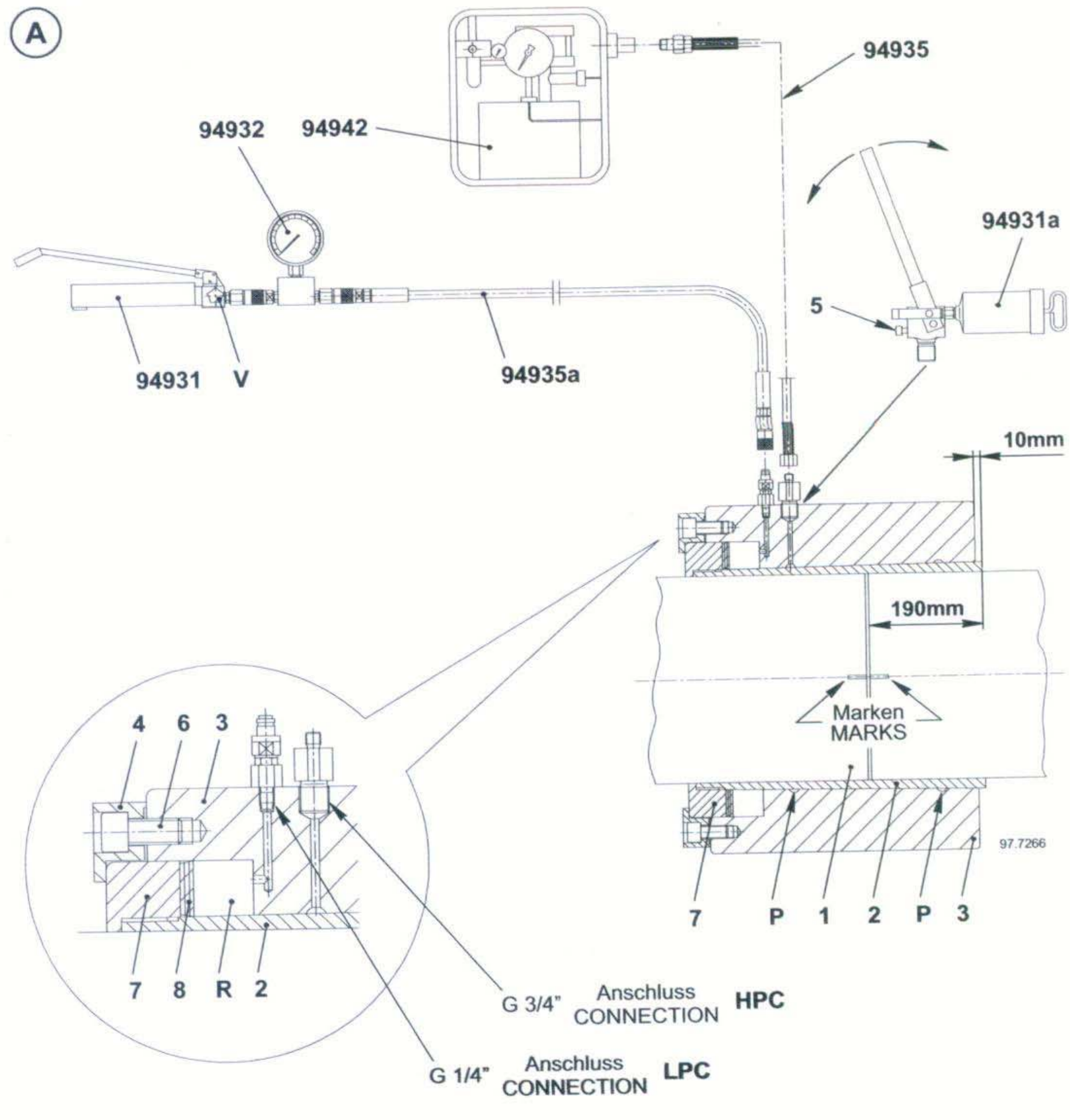
1	HP oil pump	94931
1	SKF hand oil pump	94931a
1	Pressure gauge	94932
1	HP hose	94935
1	HP hose	94935a
1	Hydraulic unit	94942

Key to Illustrations:

1 Camshaft
2 Inner bush
3 Coupling sleeve
4 Locking plate
5 Valve screw
6 Screw
7 Nut
8 Seal ring

P Fit gap
R Ring space
V Relief valve

HPC High Pressure Connection
LPC Low Pressure Connection

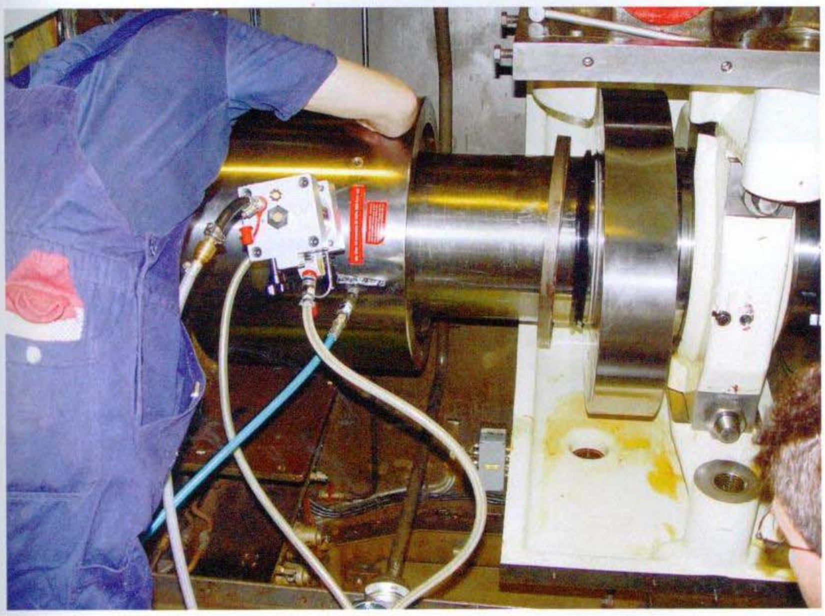


The reconditioning of a damaged fuel cam of a Wärtsilä Sulzer 10 RTA 96 C.

1 *camshaft*
2 *cam*
3 *fuel pump*

Removing a section of the camshaft.

Of course, many auxiliary tools are required such as guiding beams, hoists and hydraulic tools.



The hydraulic loosening of the S.K.F.-shaft coupling. Every cylinder has its own removable camshaft section.

Check of the foundation bolts

In order to keep the various parts of the crosshead engine, the crankshaft bed, the A-frame, and the cylinder beam together, very heavy hydraulically tensioned foundation bolts are used.
These are tensioned by using very large hydraulic jacks.

Working with these hydraulic jacks is extensively described in instruction manuals, the so-called 'Working procedures'. All the necessary detailed drawings can be found in the 'electronic' 'instruction manuals' by a simple click on the blue-highlighted text in the drawings.

▸
The hydraulic jack for the largest bolt in the engine industry: the tie rods of the two-stroke crosshead engine: 180 millimetres!

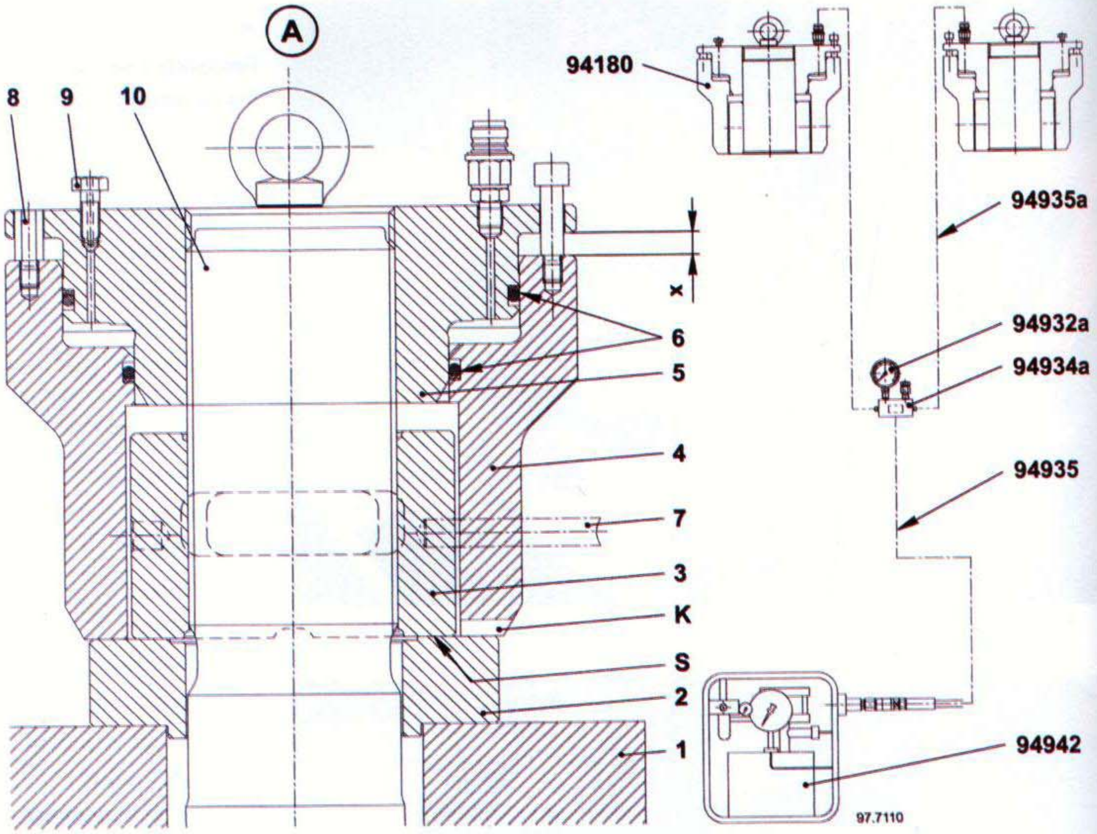
Tools:

2	Pre-tensioning jacks	94180
1	Pressure gauge	94932a
1	Connection piece	94934a
1	HP hose	94935
2	HP hoses	94935a
1	Hydraulic unit	94942

Key to Illustrations:

1 Cylinder jacket
2 Intermediate ring
3 Upper tie rod nut
4 Cylinder
5 Piston
6 Sealing ring
7 Round bar
8 Pin
9 Vent screw
10 Tie rod
11 Screw
12 Protection cap
13 Opening for solvent
14 Clamp screw
15 Column
16 Bedplate
17 O-ring
18 Lower backing ring
19 Lower tie rod nut
20 Bush

AN Drain slot
K Slot
L Thread protrusion
S Measuring place

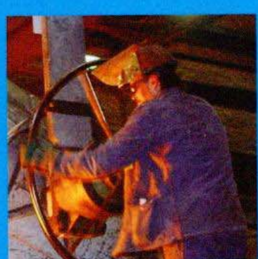


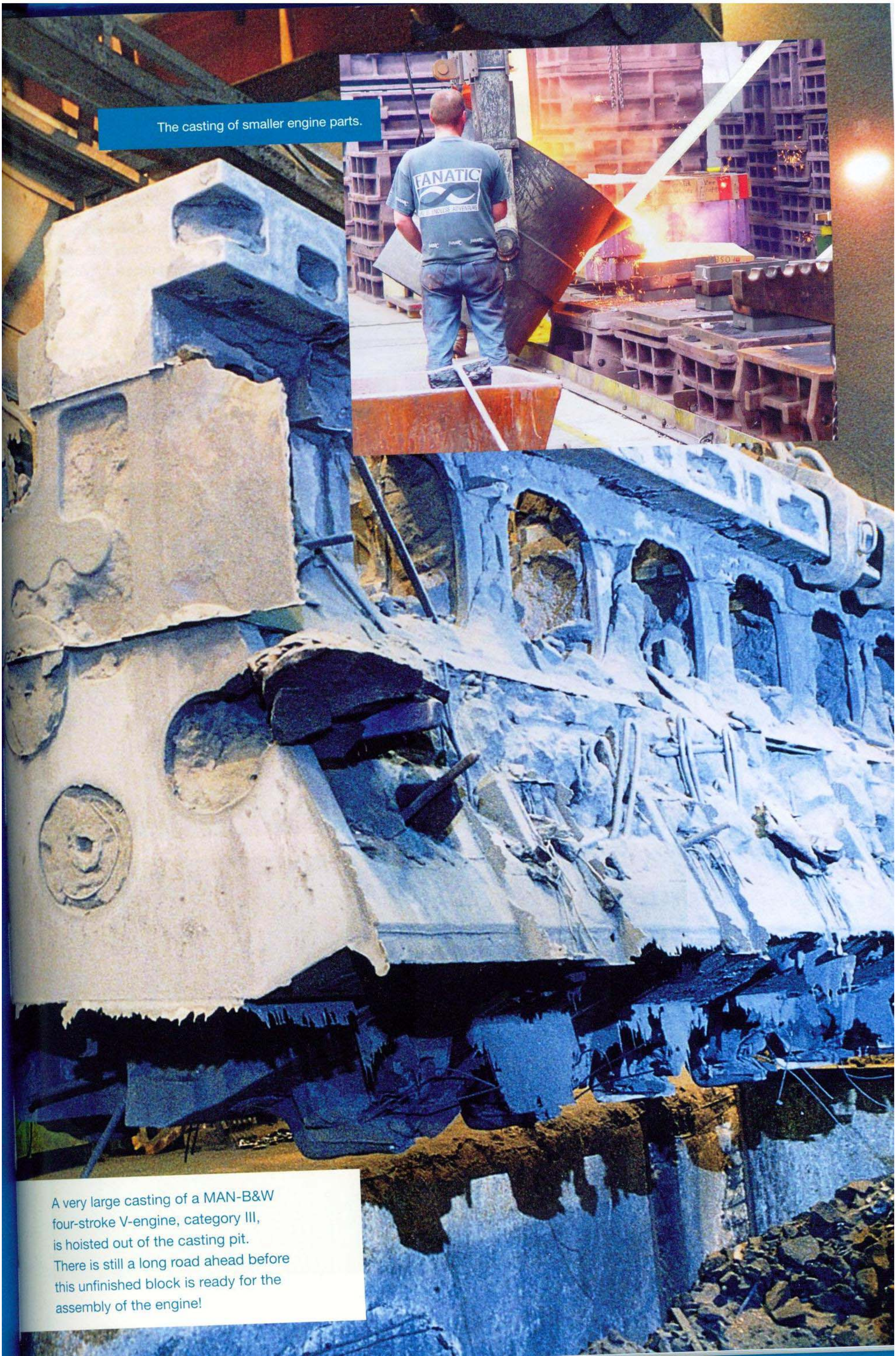
▲
Good maintenance depends for a large part on motivated and specialised personnel that, in the short time that diesel engines are stopped, can execute maintenance and repairs competently, quickly and successfully.

CH 28

Casting of engine parts



The casting of smaller engine parts.

A very large casting of a MAN-B&W four-stroke V-engine, category III, is hoisted out of the casting pit. There is still a long road ahead before this unfinished block is ready for the assembly of the engine!

28.1 Introduction

From the very first trials with Otto- and Diesel-engines, most parts, such as the cylinder block, the cylinder head and various other components, were made of cast iron.
This was a common manufacturing method halfway through the nineteenth century. One hundred years earlier, steam engines were produced in a similar fashion from cast iron. Casting was already a common craft around 1750 for weapons, such as cannons!
Many of the present foundries manufacturing engine parts have a very long history, which often goes back centuries.

28.2 Cast-iron parts of diesel engines

1 **Engine blocks**
These have been cast from the beginning of the engine industry. Only in the large two-stroke crosshead engines, the bottom half of the crankcase and the A-frame are manufactured of welded steel plates.
In the past, this was also the case in two-stroke engines of Brons and Bolnes. All four-stroke engines have as a rule a cast engine block.

2 **Cylinder heads**
Traditionally. These are cast.

▶
The foundry of Caterpillar–MaK in Kiel, Germany.

▼
The foundry of MAN–Diesel AG in Augsburg, Germany.

3 **Pistons**
Many pistons are still cast. Additionally, pistons that consist of two parts often have a cast-piston skirt.

4 **Other components**
Components such as the air-intake ducts, integrated pumps, oil sumps, rocker arms, air coolers, and governor seats are often cast.

5 **Gears**
Due to the increase in interaction forces in the engine, fewer connecting rods and crankshafts are manufactured from cast iron. These cast iron parts are now only found in small, often older engines.

▲
The top of an electric oven. The liquid cast iron is visible; it has temperatures between 1000 and 1050 °C.

28.3 Advantages of cast engine parts

Often engine parts such as the block and the cylinder head have a very complex shape. Currently, the air supply to the cylinders and the lubricating-oil supply lines are placed inside these parts where possible.
Both the block and cylinder head have shapes that are difficult to manufacture by welding steel plates.

For material properties, see Chapter 7, Use of materials for diesel engines.

▲
High-tensile scrap iron, such as steel waste from the manufacture of beams, profiles and/or metal sheeting used for ship manufacture is used as a base material for cast iron.

Waste from the thin metal sheeting used in the auto-industry is also used. Every foundry has its own preferences. There are waste-sheeting materials that contain substances that are undesirable, such as copper.

28.4 Foundries

Many engine manufacturers have their own foundry, such as, MAN–Diesel AG in Augsburg, Germany, where most of the graphic materials in this chapter were taken.
There are foundries that not only cast engine parts for engine manufacturers, but also for other industries.
The very large shipyards in the Far East often build complete ships and manufacture all the castings for their four-stroke and two-stroke engines on location.
Parts that are forged, such as crankshafts, connecting rods, pistons, and cylinder heads (two-stroke) are also manufactured in situ.

28.5 Casting process

In a smelting furnace, often an induction oven, relatively clean metal-waste materials such as those used for shipbuilding and metal sheeting used in the auto-industry, are melted together with the required alloys.

The steel turns into a liquid at a temperature of approximately 1000 °C. For every type, cast iron requires a different raw-material composition. Removal of impurities from the molten steel is of the utmost importance.

The smelting furnace at MAN Diesel AG in Augsburg, Germany, shown just before filling the casting ladles.

Transporting a casting ladle using an overhead crane through the casting hall. Prepared mould-casting assemblies ready for casting in various places.

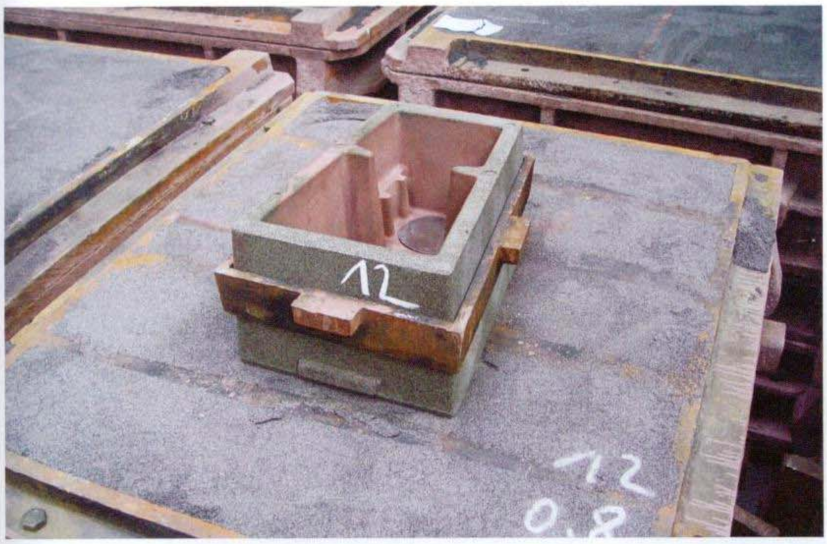
The moment that a fully loaded oven reaches the required temperature, the molten cast iron is kept at this temperature and immediately equally distributed over a number of large casting ladles; this is dependent on the size and number of the castings.

The casting ladles are insulated internally for the very hot liquid cast iron; this is done using special heat resistant fire bricks. These casting ladles are moved throughout the casting hall to every casting location with an overhead crane.

28.6 Casting location

Large castings such as engine blocks are cast in the floor of a casting pit. Square steel moulding boxes, built up in layers are used for the smaller castings.

A mould-casting assembly in a casting pit. The square openings are casting mouths for quickly filling this large engine block.



A mould-casting assembly in a moulding box. In order to take shrinkage of the cast iron into account during cooling, a part of the funnel is also filled with the liquid cast iron.

28.7 Moulds

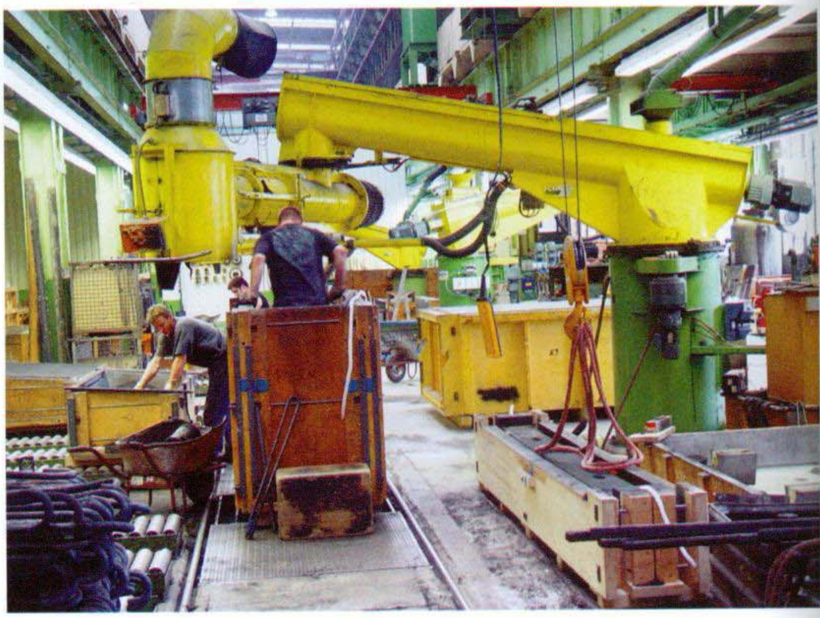
Throughout the centuries, casting has taken place by using moulds manufactured from casting sand. Today, a special substance is added that ensures that the complex and sometimes very thin shapes harden so that they can be transported either manually or by using a crane.
Casting moulds are required so that the shapes can be manufactured. These used to be made of wood but are now manufactured of wood as well as plastics.

28.8 Filling the casting moulds

Small complex moulds are made by hand. The casting moulds are then filled by hand with casting sand and intermittently compacted ('stamped down') and re-filled. When the mould has been completely filled, the casting sand is left to harden so that it can be used under normal operating conditions.
Casting moulds are partitioned in different ways so that they can be easily taken apart.
Larger moulds are filled using filling machines that vibrate and therefore compact the casting sand at the same time.

The manual process for the manufacture of smaller moulds for engine parts.

Moulding sand is often used as a base material. There are two categories: clay bound sand for small castings or chemically bound sand for large castings.



Filling machines for the larger moulds.

In order to ensure that the casting sand completely fills the moulds, the casting sand is 'stamped down'. Shown here are split wooden casting moulds.

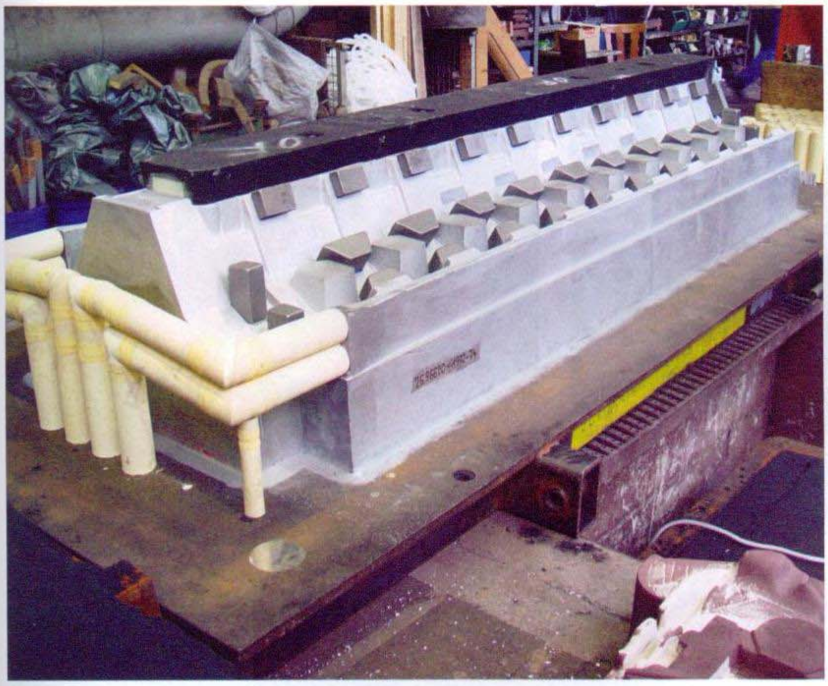
28.9 Mould assembly

28.9.1 Engine blocks

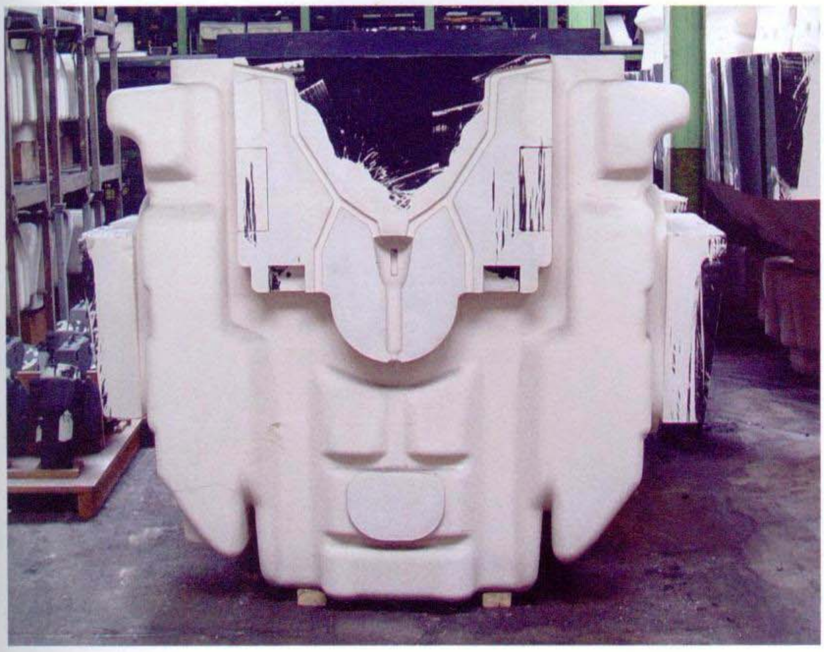
1 Almost all the space in the block must be filled with the prepared moulds, not only inside the block but also on the outside.
2 Most blocks are cast in a square casting pit built into the work floor or in large steel moulding boxes. The space between the floor, sides and the top of the block must also be filled with moulds.
3 In order to obtain a uniform material structure, casting must take place fast. The largest four-stroke engine blocks are cast in 100 seconds. The weight of the cast-iron block is 100 tonnes. A casting speed of 1 ton per second!
It is therefore necessary to have large casting openings, so ensuring that the molten cast iron is poured swiftly and filling the mould quickly 'below', while the air moves rapidly to the top!
4 It is also of the utmost importance that there are no air inclusions in the casting! This can be disadvantageous, especially near the crankshaft, where very large loads occur. This often results in the rejection of the block and consequently the loss of € 100,000.00, the production cost!

The manufacture of an engine block for a Jenbacher gas engine. In this case, a V-engine is placed on its head and filled from the bottom. The oil sump still has to be mounted on top of the cylinders. Only the sand mould is visible!

◄
One must ensure that the construction of large mould-casting assemblies does not 'float'. This is prevented by using heavy weights in large mould-casting assemblies and threaded rods for smaller moulding boxes.

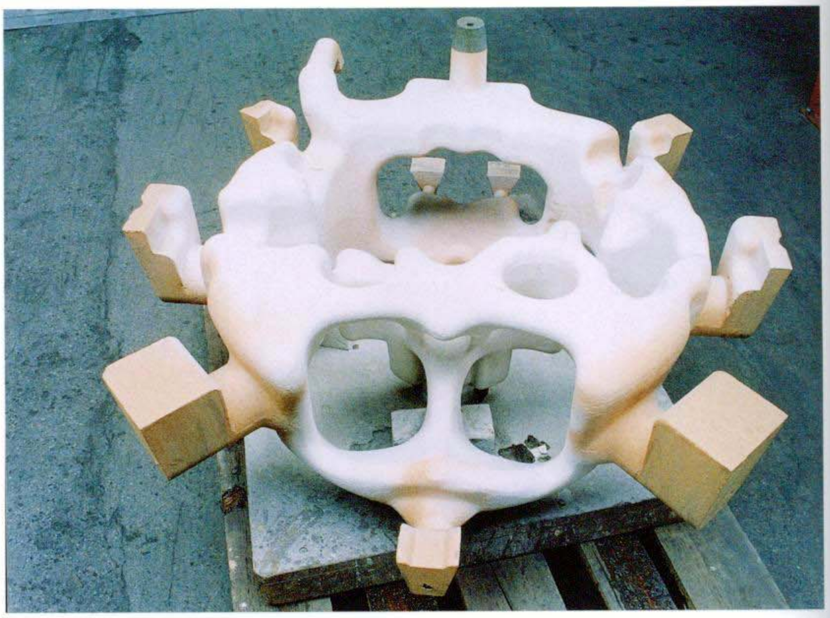


◄
The lowest part of the engine block, the crankcase. The internal piping for this Jenbacher gas motor has already been installed. All that can be seen is the sand mould!



◄
Part of the engine block in a sand mould, ready to be placed in the casting pit. Air ducting for venting has been placed in the position where the crankshaft will later be mounted, as shown in the picture.

▶
A casting mould for the coolant space in a cylinder head. The moulding sand is so hard, that the casting mould can be moved using a crane.



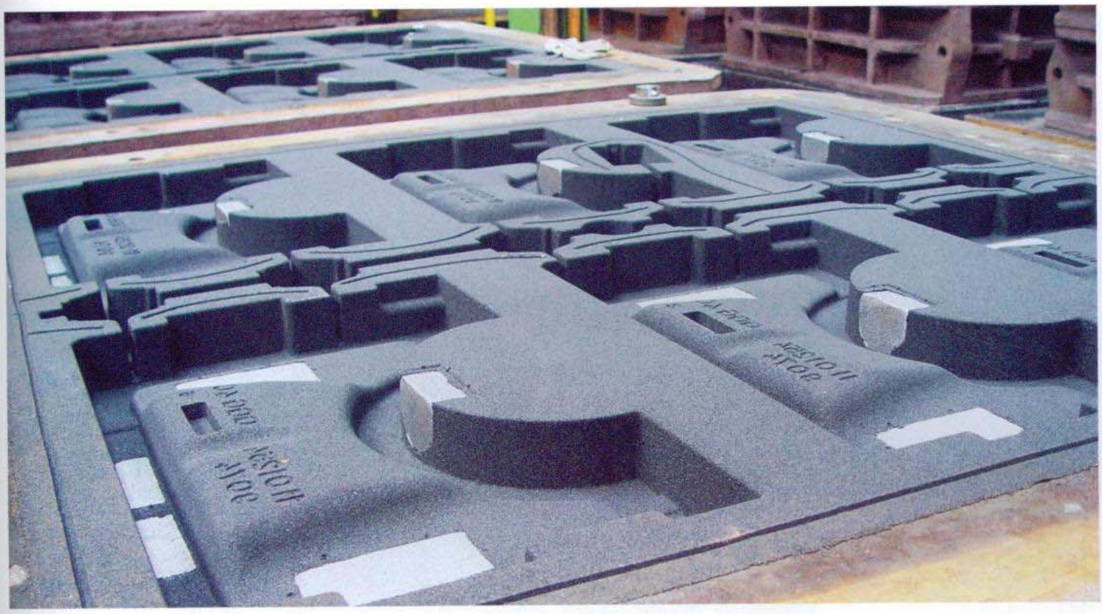
▲
A series of casting moulds, suspended by hoists. They will be used to build the complete casting.

▲
Casting. The caster is completely protected from liquid cast-iron spatters.

28.9.2 Smaller parts, such as cylinder heads

Exactly the same procedure is followed for these and similar parts, albeit less complicated in scope and process.

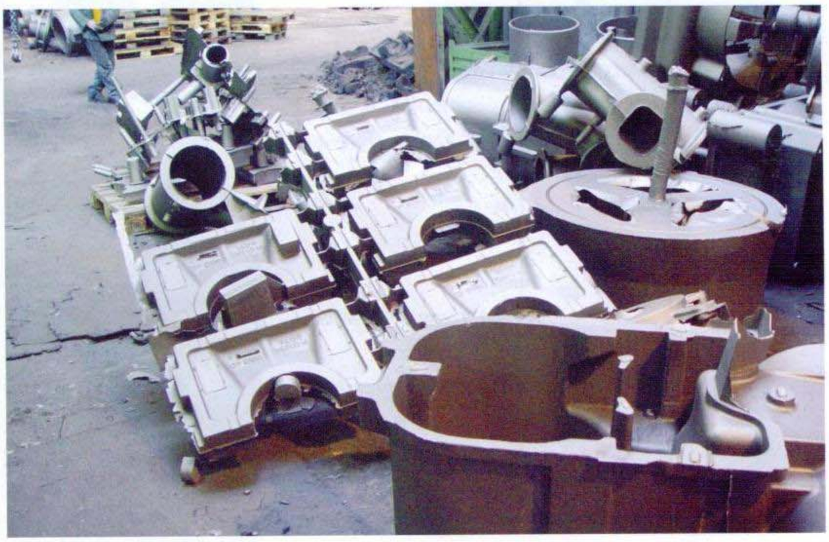
▶
The upper lid must yet be placed on this mould-casting assembly. Before a casting is cast, every casting follows a very precise procedure. Rejection of the castings is expensive.



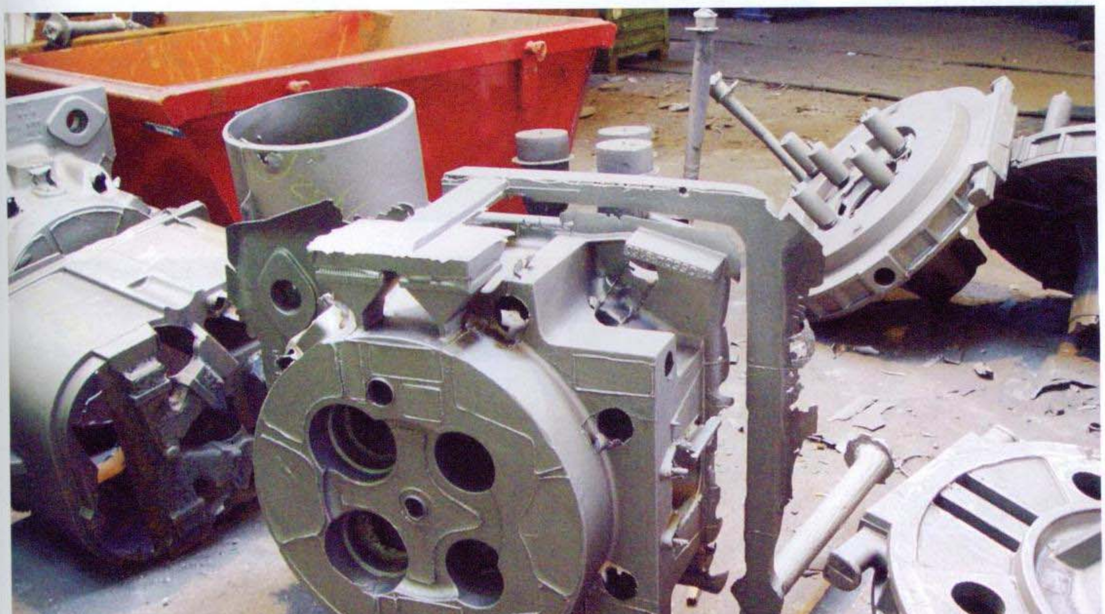
◄
The assembly in a moulding box of six lower main bearing caps of MAN–B&W four-stroke engines. The lowest mould has already been positioned. A lid is placed on top of the upper mould.

28.9.3 Serial production

The lower crankshaft-bearing caps are cast in one mould containing, for instance, six pieces.



►
The cast lower bearing caps after removal from the moulding box. The connections between the caps are clearly visible. These connections allow the immediate addition of molten cast iron and the escape of air from the mould.

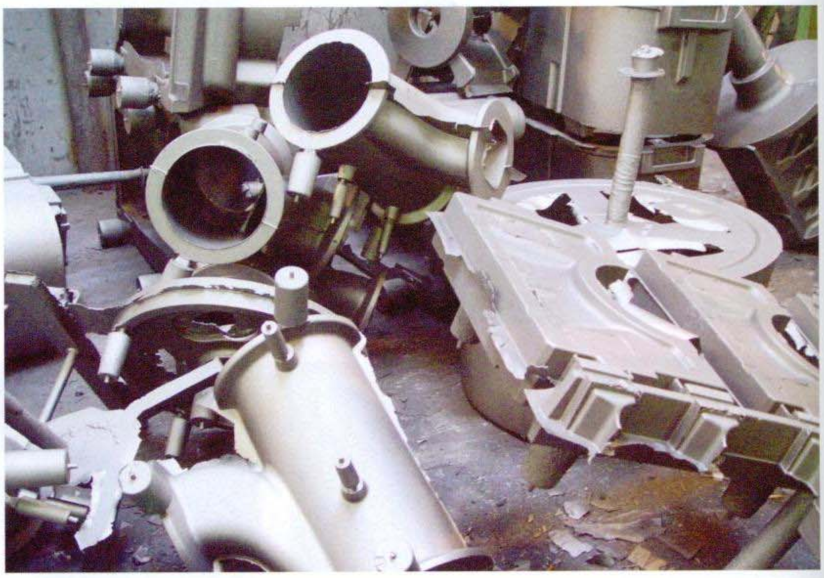


◄
A cylinder head of a four-stroke MAN–B&W diesel engine after cooling. Note the hanging cast-iron remnants!

After casting, the castings must be allowed to cool for several days. When the castings cool too quickly, the parts can crack.
Outside the casting pit or moulding boxes, flames can be seen on the coatings applied by immersion (small moulds) or lubrication (large moulds).

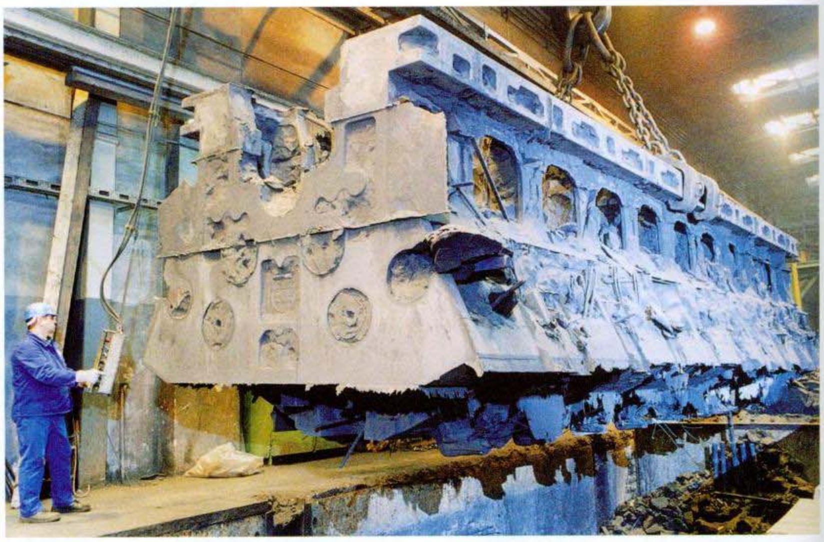
After dismantling the mould-casting assembly, the now black casting sand is collected, cleaned, and re-used.

▶
Unfinished engine parts after casting.



▶
A cast frame of a large eighteen-cylinder MAN–B&W four-stroke diesel engine type 48/60 is hoisted out of the casting pit.

Clearly shown is how much moulding sand is still hanging on the block and how rough the block is. The weight of this block is approximately 100 tons.

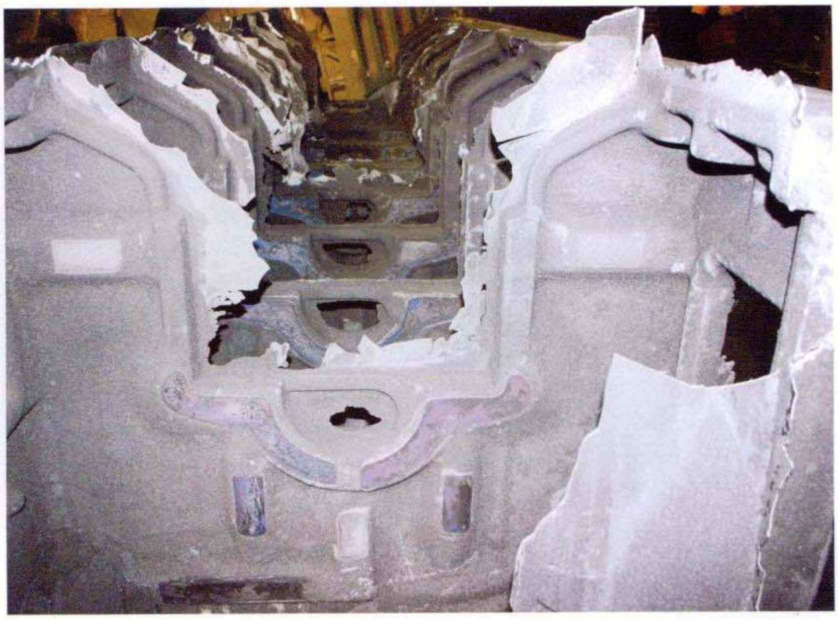


▶
A V-engine block ready for final machining.

28.10 Cleaning the castings

As shown in the images, the casting must undergo various processes before it can be machined at a different location.

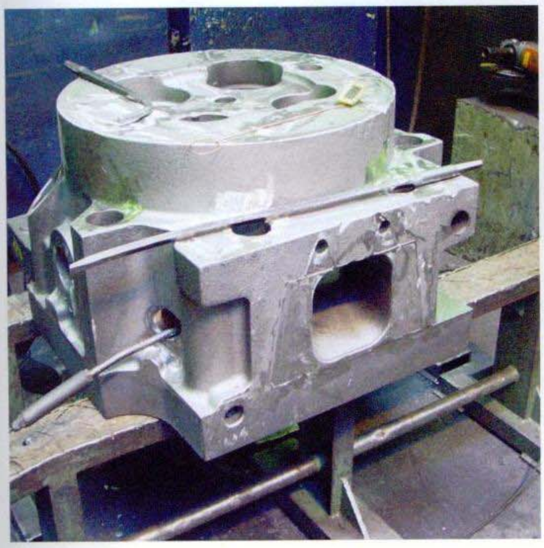
Procedure
- An initial superficial inspection is performed to examine whether the casting has been successful.
- The casting sand is then completely removed.
- Protruding 'icicles and ice sheets', the remnants of molten cast iron that has flowed out of the splits and air-vent drillings are removed by using hammers, chisels and grinding stones.



▲
'Icicles and ice sheets'. The removal of all prominences is heavy and dirty work and requires much experience.

◄
Finishing of the casting, a seven-cylinder line-engine block.

▼
A cylinder head being touched up. This is done with pick hammers, chisels and for the final finish, grinding stones and brushes.



◄
Just cast. The liquid cast iron is visible in the filling funnels. The non-stick coating is burning in some places.

The block is then de-burred using de-burring discs until all the sharp edges and prominences have been removed. The block is then 'preserved' (cast iron hardly rusts) with a varnish or paint coating.

28.11 Casting stresses

Some engine manufacturers place their cast engine blocks in storage for many months. In doing so, the stresses created during casting decrease.
At the Wärtsilä factory in Vaasa, Finland, they place the cast blocks for the W 20 engine outside in the snow during the winter months to reduce material stresses.
In view of today's growing demand for engines, this will continue in the years ahead. It will be difficult for the manufacturers to continue this practice of stress reduction!

28.12 Checking air inclusions and damage

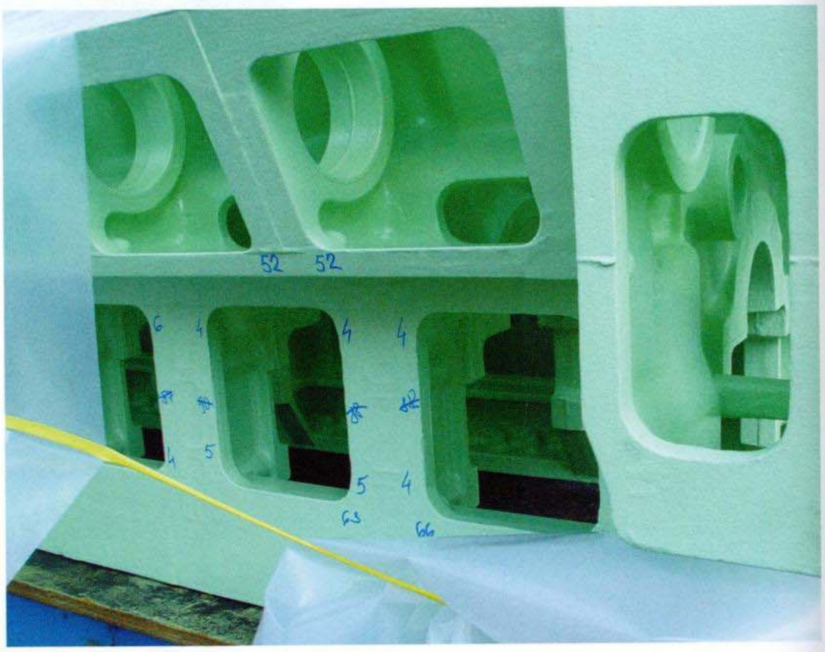
This is performed before machining, if possible. Of course, certain deviations are more serious than others.

28.13 Control of the dimensions

It is of utmost importance that the engine-block end product be machined from the unfinished cast block.
On the other hand, if too much material must be removed, this is also expensive due to the extra machining hours required.
The blocks delivered to customers are re-measured on location to ensure that they have the correct dimensions.

▶

A finished and painted casting, a block for a V- engine. The block is measured before machining.



▼

A block for a large V-engine, cleaned, painted, measured and ready for metal removal.

▲

Ultrasonic inspection at a location where the engine will be heavily loaded, specifically around the crankshaft.

28.15 Manufacturing crankshafts

28.15.1 Introduction

Crankshafts are the most heavily loaded parts of diesel engines. They are subjected to torsional and bending stresses. Due to the fluctuating loads on every crank, crankshaft manufacturing must meet high requirements.
The forces on the crankshaft are large. Even the smallest engines of Category I have maximum forces of hundreds of Newton on the crankpin during combustion.
The combustion forces exerted on the piston, and consequently on the crankshaft, vary from 0.5 to as much as 1,100 tons in the various engine categories!

▲

A crankshaft made of a one-piece forging in a single throw.

The crankpin journals and webs for this particular crankshaft are shaped by sets of heavy hydraulic presses.
The shaft is forced into a mould using huge forces of thousands of tons and the result is the rough crankshaft.

1 *rough crankshaft*
2 *bottom mould*
3 *top mould*
4 *hydraulic press*

▼

Manufacturing the largest crankshafts for the medium-speed diesel engines, category III is very expensive.

The crankshaft is forged from a solid piece of the correct material. Forging elements of such dimensions is done per crank. This is called 'throw by throw' or 'Continuous Grain Flow', as opposed to the manufacturing of smaller crankshafts, which are forged 'in one throw'.
This is followed by a very meticulous machining finish and the shaft is ultimately approved by a Classification agency.
The picture shows a crankshaft of an in-line engine for reconditioning at Mark van Schaick, Schiedam, The Netherlands.

28.15.2 Crankshafts requirements

High requirements are set for:
- material used;
- crankshaft design;
- calculation of the maximum tensions inherent in the material, such as the nominal tension, resistance to metal fatigue and the ensuing acceptable maximum values;
- drillings for the lubricating oil in the crankshaft;
- finish and the surface hardness of the journals;
- shrink fit allowances of the crankweb and crankshaft in two-stroke crosshead engines.

Also, see Chapter 32, Regulations for propulsion engines, classification, repair and damage.

28.15.3 Material used

▸

A number of steel alloys with a certain tensile strength can be considered.

A	Manganese – carbon-steel	Casting	400-550 N/mm²
B	Manganese – carbon-steel	Forging	400-600 N/mm²
C	Manganese – carbon-steel	Forging	maximum 700 N/mm²
D	Alloyed steel	Casting	maximum 700 N/mm²
E	Alloyed steel	Forging	maximum 1,000 N/mm²
F	Spherical or nodular graphite steel	Casting	370-800 N/mm²

28.15.4 Manufacturing crankshafts

Casting

This is often used for small lightly loaded crankshafts, for instance, air-cooling and industrial air compressors and small lightly loaded engines of Category I.
The liquid material is poured into moulds, where it immediately assumes the coarse shape of the required crankshaft.
The final shape is achieved by subsequent machining processes, such as turning, milling, drilling and other machining operations.
As the required manufacturing equipment is relatively simple, these crankshafts are comparatively inexpensive.
The granular structure of the material resembles sand; this is noticeable in crankshaft fractures.
Casting has fewer positive properties than forging.
The material has a lower tensile strength and is therefore more prone to fractures.

Forging

Forging large crankshafts requires large and heavy hydraulic presses. Moreover, elaborate machining is required subsequent to the forging process.
The base material, a steel shaft of the correct composition, is locally heated to white-hot and then placed between dies and pressed into the required shape.
This requires presses that can generate pressure forces of thousands of tons!
Pressing produces a continuous grain flow; continuous grain flow gives the crankshaft better fatigue resistance.
This is due to the reduced space between the molecules of the material; the molecules are more firmly joined together.
Subsequent to forging, the material is stress-relieved using heat-treating methods. They are then machined. The hydraulic machines required for forging are very expensive. Moreover, each type of crankshaft requires a specific mould.
Companies producing crankshafts or cranks (for two-stroke engines) for engine manufacturers in Categories II, III and IV who do not manufacture their own crankshafts, have a series of crankshaft dies in storage for each type with respect to cylinder bore, in-line or V-shape and the number of cylinders. This is very expensive!

Forged crankshafts are stronger and more expensive than cast crankshafts. The price of a forged crankshaft for small craft in Category I is two or three times higher than for a cast crankshaft. Large crankshafts are always forged.

◂

In the engine factory of Alfing Kessler, Aalen, Germany.

In the foreground the steel shafts, which after heating in an oven are compressed by a hydraulic press to form the rough crankshaft.
Also in the picture, a number of crankshafts ready for inspection. They are subsequently machined to the correct dimensions.

28.15.5 Material requirements and tests by Classification bureaus

There are many requirements, such as the material composition, the heat treatment, the mechanical and non-mechanical tests, such as hardness-, tensile- strength, magnetic- and ultrasonic-test methods.

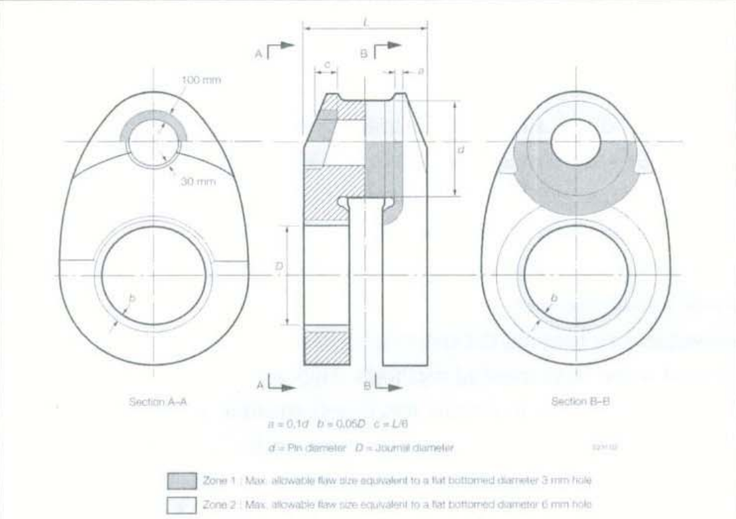


Fig. 4.4.2 Ultrasonic examination acceptance levels

▲

The prescribed inspection of the crankshaft by Lloyd's Register by the ultrasonic method. The requirements that apply to crankshafts are very stringent.

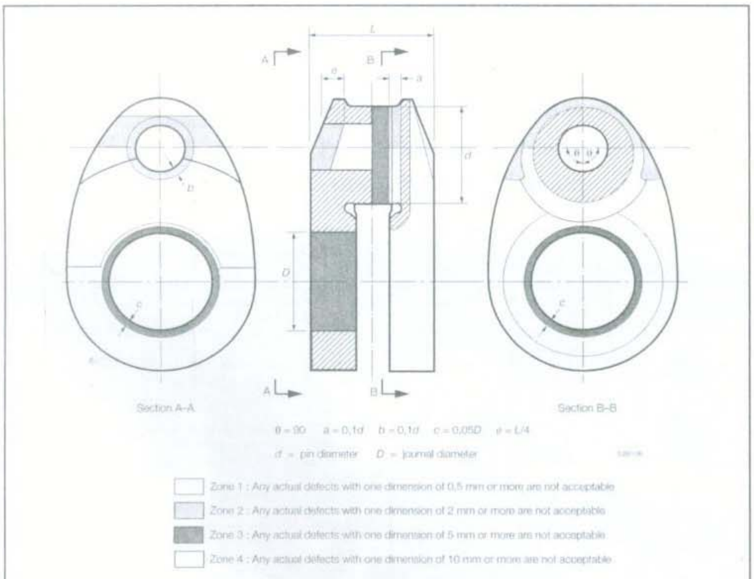


Fig. 4.4.1 Magnetic particle inspection acceptance levels

▲

The prescribed inspection of the crankshaft by Lloyd's Register with the magnetic-iron particle method.

In order to acquire a sturdy crankshaft, the dimensions of the crankshaft parts must be in a certain proportion to each other.

As the engines discussed here seldom have cast crankshafts, we will only consider the manufacturing of forged-steel crankshafts.

The procedure.

- Provide the forge with a drawing of the crankshaft and material specifications, approved by the Classification bureau. The drawings are designed and accessible in 3 D. All possible loads on the crankshaft must be established and calculated!
- Order a steel shaft of the correct dimensions and material composition.
- Prepare the hydraulic crankshaft press with the correct dies for the crankshaft.
- Heat up the shaft in a gas or electric oven until the correct temperature is achieved.
- Place the heated shaft in the press.
- The shaft is usually pre-processed in a separate machine: pre-forging.
- Press the entire crankshaft in a series of forgings or, for smaller crankshafts, in a single throw.

To ensure that the forged steel maintains the correct temperature, the rough crankshaft is removed from the press three times and heated in the oven until it achieves a temperature of 900°C. The hydraulic pressing is performed 25 times per cycle, so a rough crankshaft is completed in 75 pressings!

- Larger crankshafts: throw-by-throw pressing. One crankshaft section of the rough shaft is locally heated to 900 °C and subsequently forged in a series of throws.
  Here the cranks are also re-heated. For the following crank, the shaft is replaced in the correct position for that crank.
- Heat treatment methods are performed, such as annealing and tempering.
- Machining of the crankshaft to its final shape and dimensions.
- Post-treatments, such as hardening of the journals for crankshafts in Categories II, (generally), and III (occasionally).

◄
A rough forged crankshaft with in the centre the base material, an alloyed steel shaft.

The white marker paint is applied in order to determine if the final crankshaft can be manufactured from the material.

◄
Here the unfinished part of the crankshaft (the crankwebs with the fitted single-throw counterweights); the rough forged shape is clearly visible.

Other engine manufacturers require cleanly machined crankshafts.
Note, this crankshaft is manufactured from one massive rod!

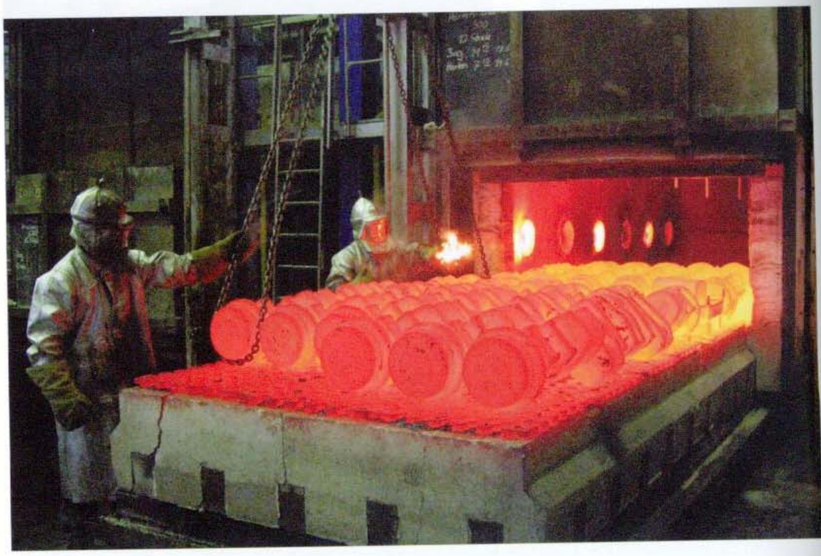
◄
A roughly forged crankshaft in the oven.

The partially forged crankshaft is re-heated to 900 °C.

▶

Forging crankshafts for diesel engines in Categories II, III and IV requires large hydraulic presses.

Shown a larger hammer press with a maximum pressure force of 3,500 ton.
Large crankshafts are forged throw-by-throw with this press. First, the base material is horizontally clamped in a lathe in order to be able to machine the correct crank angles. Then part of the shaft is heated in an oven and forged. The forged piece is transported to the oven by means of a special forklift, thus maintaining the correct forging temperature.



▶

Forged crankshafts undergo heat treatments, such as annealing, for stress relief.

Often the parts are placed in a gas-heated oven for a certain amount of time. Here, approximately 11 hours, prior to hardening. This time is required so that the centre of the crankshaft obtains the same temperature as the outer parts of the crankshaft.

▶

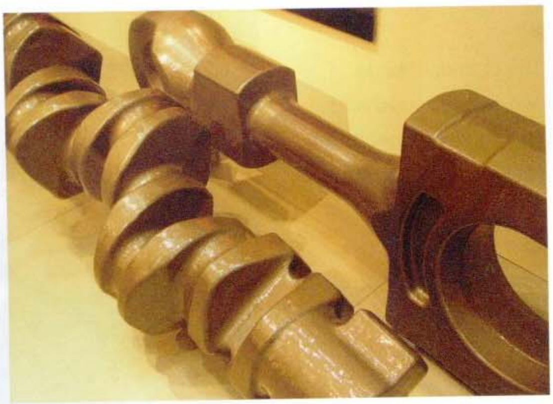
Surface hardening.

Crankshafts, which require partial surface hardening, are rapidly cooled in a liquid bath; this is known as 'quenching'. The shaft is rapidly cooled to a certain temperature, so the surface hardens. It is subsequently placed in an oven and re-heated, after which the shaft is slowly cooled.
The liquid is usually either an oil-water emulsion or oil. Induction hardening is also often applied for the running surfaces and rounding of the journals.

◄
A new crankshaft is placed in a washing machine to remove the final dirt particles before the crankshaft is placed in the engine block. Caterpillar Mak, Kiel, Germany.

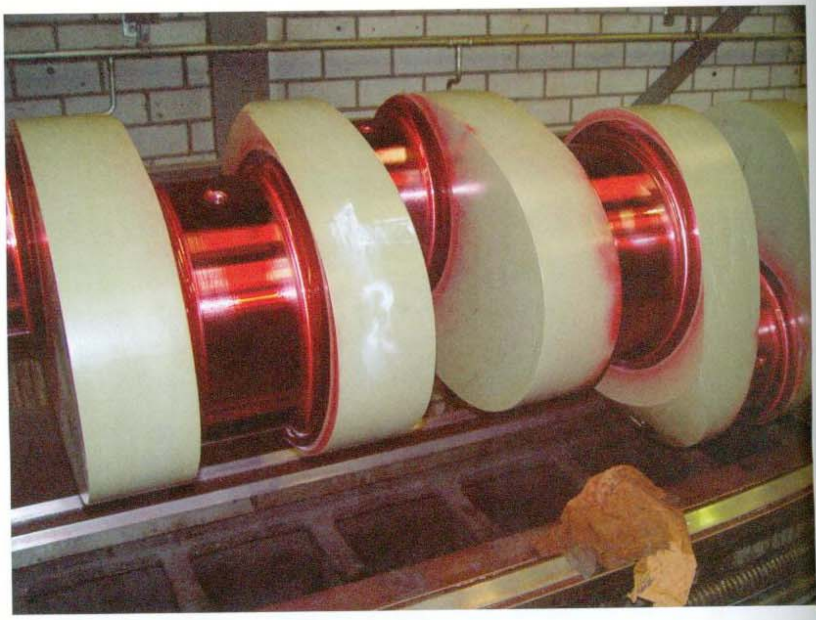
▲
A finished crankshaft for an eight-cylinder in-line engine, Category III. The crankshaft is prepared for transport to the engine manufacturer.

These final finishes are followed by a number of treatments. The crankshaft is washed in a special washing machine and transported to a dust-free room. Here a final inspection takes place and the crankshaft is tested by a Classification bureau and marked.
Most crankshafts are delivered to the client in corrosion-proof packaging. The packaging generally comprises a strong case, with a dust-proof carton lining and sufficiently supported to prevent bending of the shaft during transport.



◄
A forged crankshaft of a Category II engine and a connecting rod for a Category III engine.

This part is also forged in a single throw after which the 'small eye' and the 'big end' are milled.
It is then finished and both divisions are made.

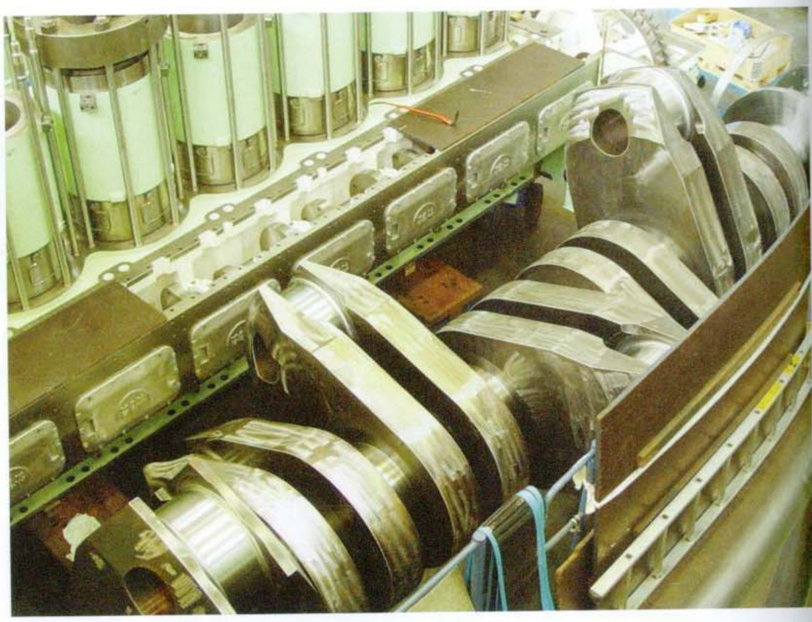


▶
When reconditioning, such as grinding journals, a hair-crack test is always performed.

The method using a white and red spray shows possible cracking of the journal after it has been cleaned; it is known as the fluorescent test.

▶
Hair-crack inspection.

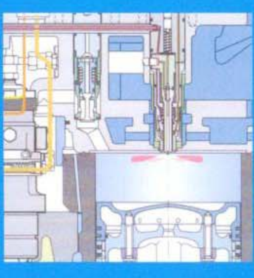
A small hole in the surface area of the shaft becomes visible after removal of the red layer with a clean cloth and a solvent.
This is usually indicative of inexpert reconditioning methods, such as rewelding a damaged journal.

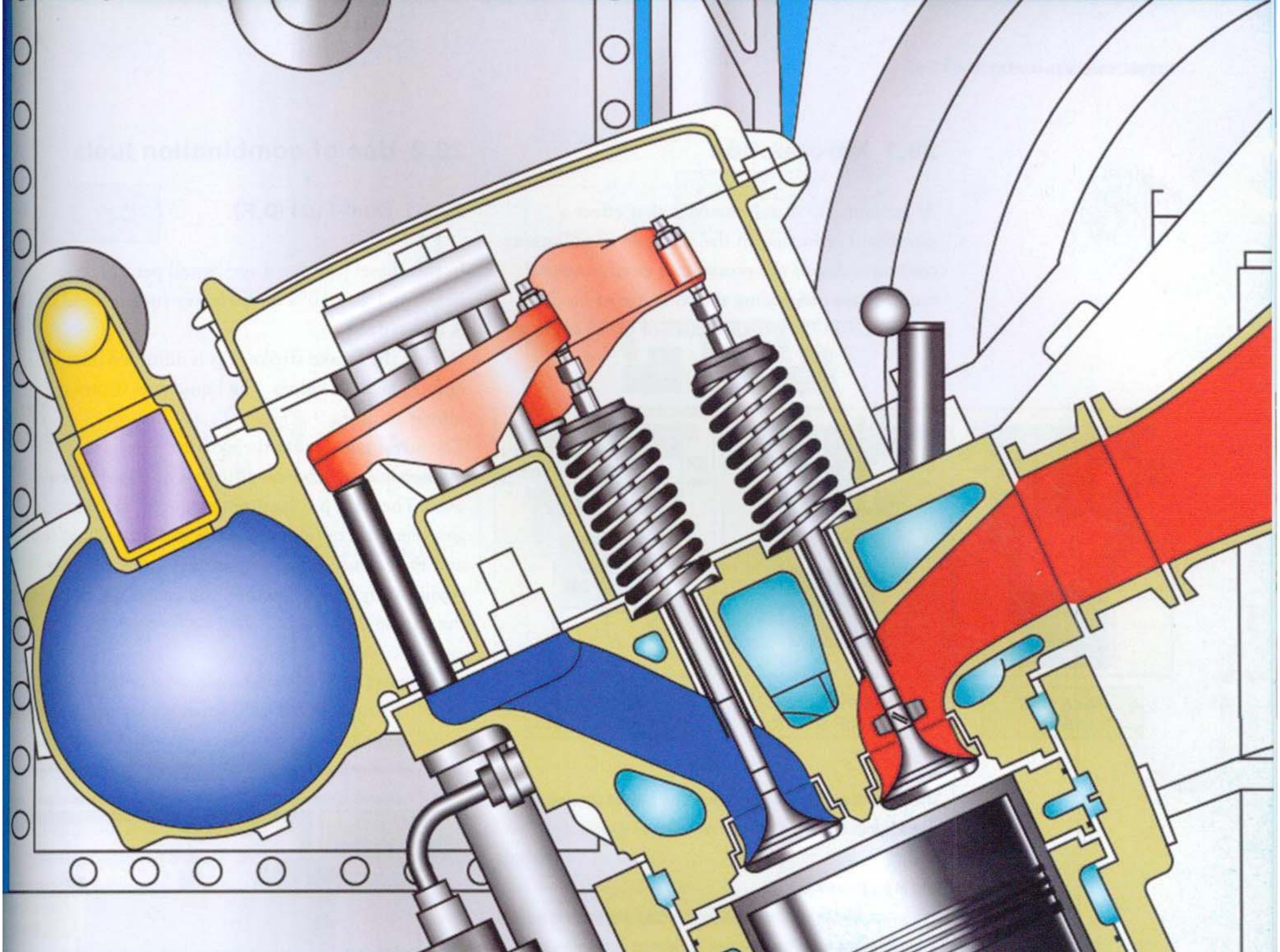


▶
The cranks of large two-stroke crosshead engines, category IV, are forged in a single throw and then bored for the journal insertion. The separately forged journals are fitted into the webs using the shrink-fit method.

CH 29

New fuel developments





A modern LNG tanker, driven by dual-fuel engines.

One of the new developments in the use of diesel engines is the enormous growth of the number of gas tankers equipped with dual-fuel engines that are being manufactured.

An original fuel pump with a special injector is used, which enables the combustion of a compressed mixture.

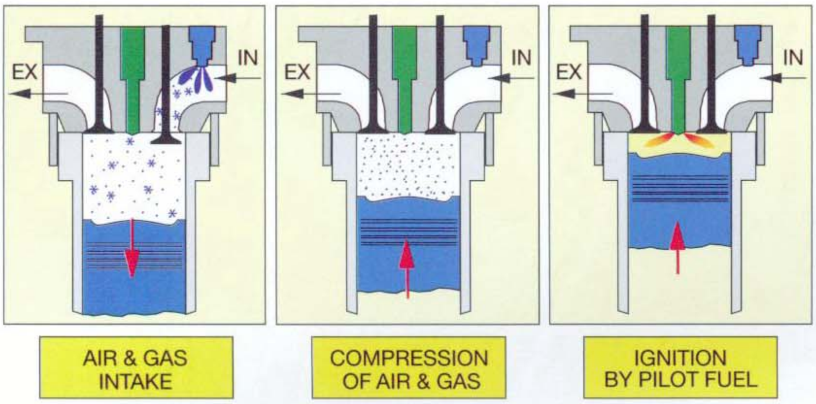
This mixture comprises air and gas, drawn from the ship's storage tanks.

Due to this, the cargo (the liquid gas) remains at a sufficiently low temperature and the emission of noxious gases is significantly decreased.

Often less than 5% liquid fuel (heavy-fuel oil) is supplied. The remainder is gas.

29.1 Introduction

At present, the search for fuels that effect a significant reduction in the emission of pollutants continues due to the pressure of environmental requirements regarding the emission of noxious substances in the exhaust gases of diesel engines.



▲

The dual-fuel engine principle. The diesel process is retained.

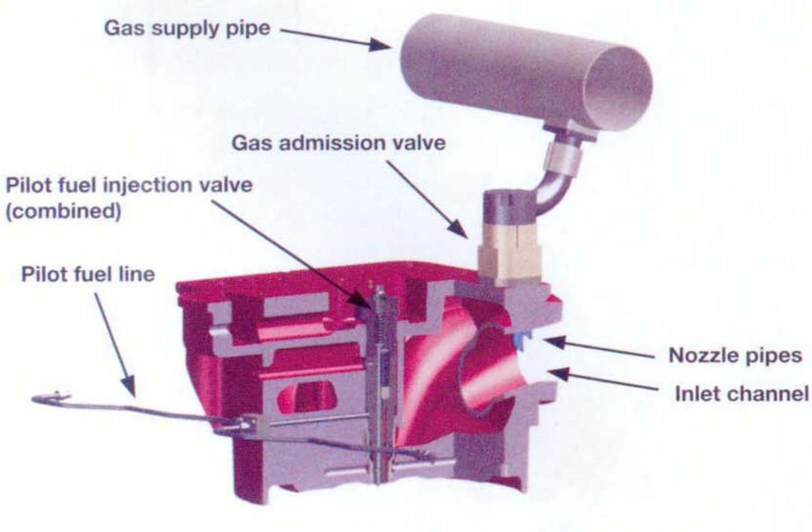
left: induction stroke

As the piston moves downwards, a mixture of air and gas is drawn into the cylinder. The gas is injected in the inlet channel via a solenoid valve.

middle: compression stroke

As the piston moves upwards, the mixture is compressed. The mixture is poor, which means that the excess air is large (approximately 2). Due to this, the mixture does not spontaneously ignite.

right: just before the top position of the piston, the fuel is injected into the cylinder under high pressure. This ignites immediately so causing the air/gas mixture to ignite. Afterwards, the normal process takes place.



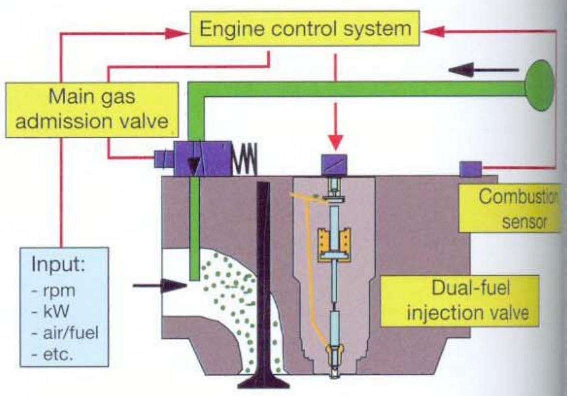
29.2 Use of combination fuels

29.2.1 Dual-Fuel (D.F.)

In this diesel process, a very small percentage of between 1 and 10% liquid heavy fuel or diesel oil is injected.

During the intake stroke, gas is admitted just before the inlet valves. The liquid fuel ignites the air/gas mixture.

The advantage is that the exhaust gases are cleaner than when heavy-fuel oil or diesel oil are used. The CO_2 production is reduced, since a gaseous fuel contains less carbon than a liquid fuel and the emission of soot particles and sulphur is significantly less. Therefore, the contamination of the engine decreases.



▲

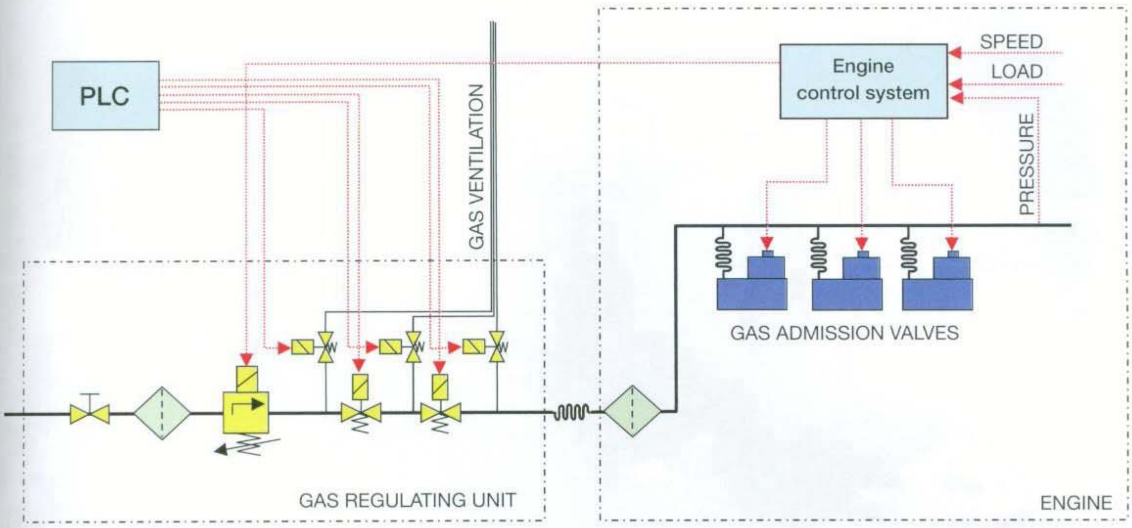
The dual-fuel principle.

The engine can automatically change over from heavy-fuel oil to gas below an 80% load. This takes approximately one minute. When there is an interruption in the gas supply, the engine automatically changes over to heavy-fuel oil.

◄

The principle of the dual-fuel system.

The one fuel, gas, is admitted just before the inlet valves during the induction stroke of the engine. Gas and air are intensively mixed. The air factor is between 1.5 and 2.0. This is known as a 'poor' mixture. 50 to 100% 'more air is present than required to fully burn all the fuel chemically. At the end of the compression stroke, the very small amount of injected fuel ignites immediately followed by the rapid ignition of the air/gas mixture. The 'Pilot'-injector replaces the traditional sparkplug. The process continues according to the diesel principle, which has a higher efficiency than the Otto-principle.

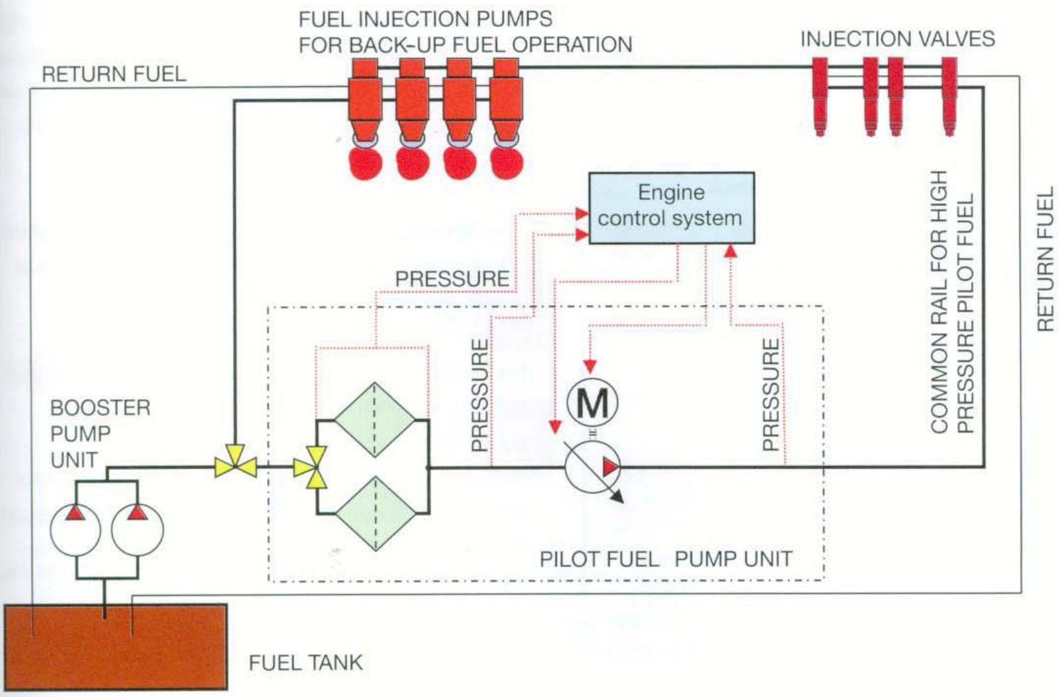


The system for gaseous fuel.

The yellow section, the 'gas valve unit' ensures that gas is safely supplied to the engine under all conditions.
A manual shut-off valve, a filter, a pressure-regulating device and two solenoid valves are standard. With engine failure and/or activation of the emergency switch, both solenoid valves quickly shut and the engine shuts down.
The 'Engine Control System' controls the solenoid valves that inject the gas in both air ducts before both inlet valves.

The liquid heavy-fuel oil system of this dual-fuel diesel engine.

The fuel before the injectors comprises 1% heavy-fuel oil and 99% gas supplied by a 'Pilot' high-pressure pump unit that via a common-rail system provides the injectors with sufficient fuel. With the transition to 100% heavy-fuel oil use, the large high-pressure fuel pumps provide sufficient fuel. The injectors have two injector nozzles. The pilot and main injector.

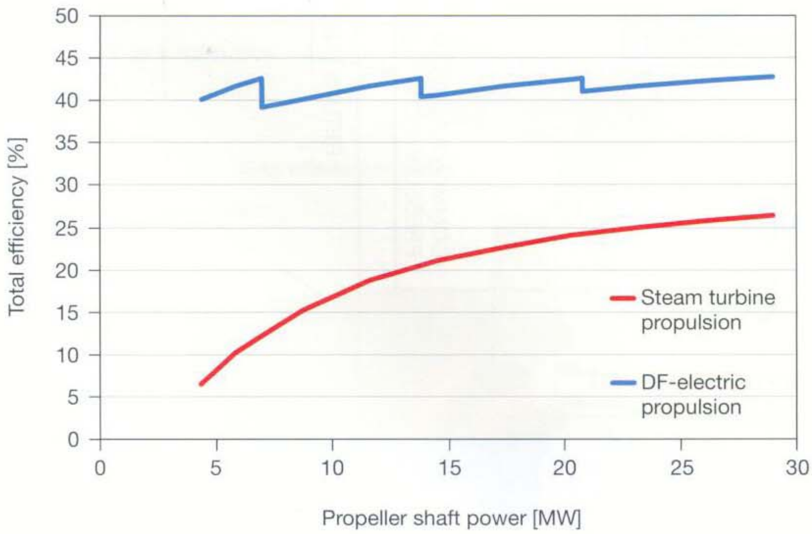


▲

Until approximately 2004, Liquefied Natural Gas (L.N.G.) tankers were propelled by steam consisting of a high-pressure water-tube boiler, a steam turbine and a condenser unit.

The fuel is supplied from tanks in which the liquid gas is kept at a temperature of -180 °C. The evaporating liquid fuel is constantly boiling during the voyage and therefore removes the warmth from the L.N.G. tanks, keeping the L.N.G at a constant low temperature. Per voyage, dependent on the time taken, between 2 and 10% of the cargo is used. The steam boiler can be fired with heavy-fuel oil when failures are experienced. At present, the dual-fuel engine continues to experience tremendous growth; all L.N.G.-tankers ordered in 2006 were with these engines.

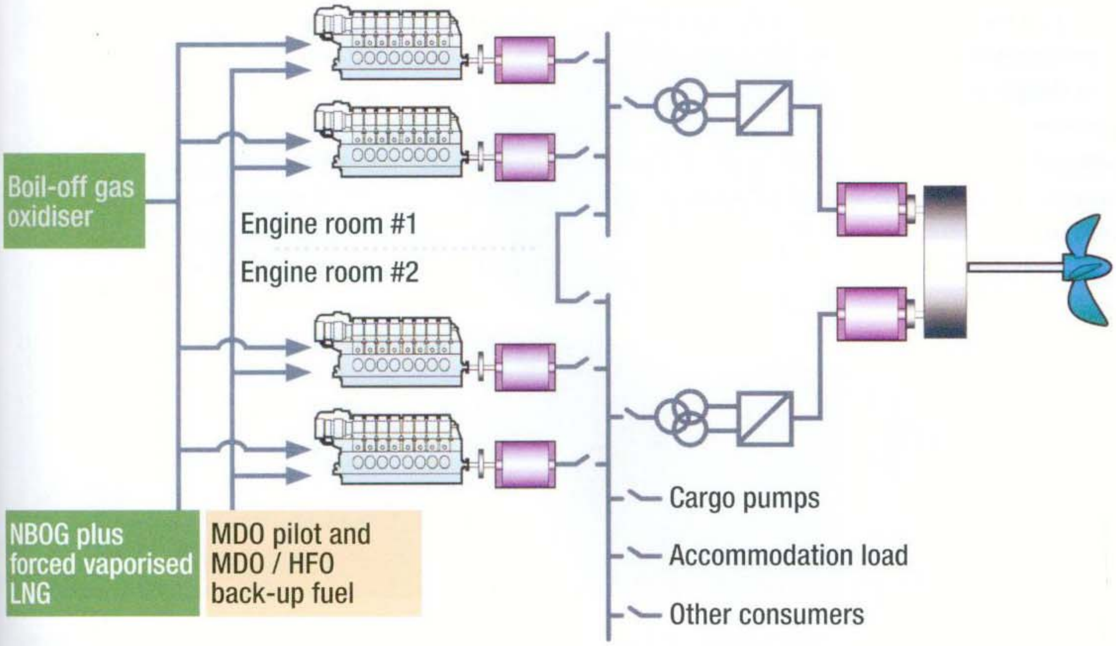
Some L.N.G.-tankers are presently equipped with a propulsion engine that mainly runs on the gas from the cargo. In order to keep the cargo (the liquid gas) at a sufficiently low temperature, a very small part of the cargo evaporates. This evaporation requires heat, drawn out of the cargo. This way the liquid gas is kept at the desired low temperature of -180 °C and gaseous fuel becomes available for propulsion. Previously, this gas cargo was also used as fuel for the steam boiler of the steam-turbine propulsion installation. The efficiency of such propulsion installations is much lower than that of the dual-fuel engine.



◄

The difference between the propulsion efficiency of the conventional steam turbine in relation to the dual-fuel diesel engine.

For this tanker, dual-fuel diesel engines were chosen to drive electric generators. At a reduced speed, one or more gensets can be shut down. Due to this, the propulsion efficiency remains high, also at partial loads. The propulsion efficiency of a steam turbine at full load is significantly lower than that of dual-fuel-engines. At partial load, the efficiency drops quickly.



▲

An example of dual-fuel propulsion with electro-motors.

The four D.F. diesel engines drive generators, which supply the electrical switchboard. The electromotors are supplied from this switchboard via a transformer and the speed is controlled by a frequency control.
The electromotors are reversible, therefore a fixed-pitch propeller is sufficient.

Dependent on the distance of the voyage, propulsion uses approximately 10% of the cargo gas. When a tanker is not carrying cargo, the main engine can run on 100% heavy-fuel oil via the main injector. The separate 'pilot' injector (nozzle) is then switched off.

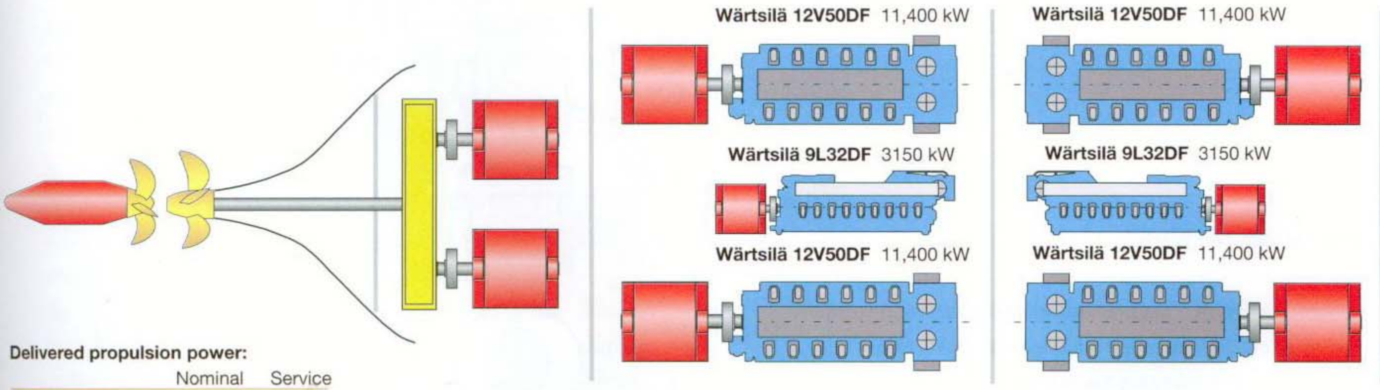
29.2.2 Gas-diesel (G.D.)

This is similar to the Dual-Fuel principle: the supply of very little liquid fuel, often less than 1% of the fuel consumption at full load and substantial amount of gaseous fuel.
During the intake stroke, only air is drawn in and compressed. Just before T.D.C. position, a small amount of liquid fuel is injected, immediately followed by the injection of the gaseous fuel.
If no gaseous fuel is present, the engine can run

▼

The same type of propulsion for a Wärtsilä design of a very fast roll-on roll-off ferry.

Note the second propeller drive. This P.O.D. can be operated in any direction for maximum manoeuvrability.



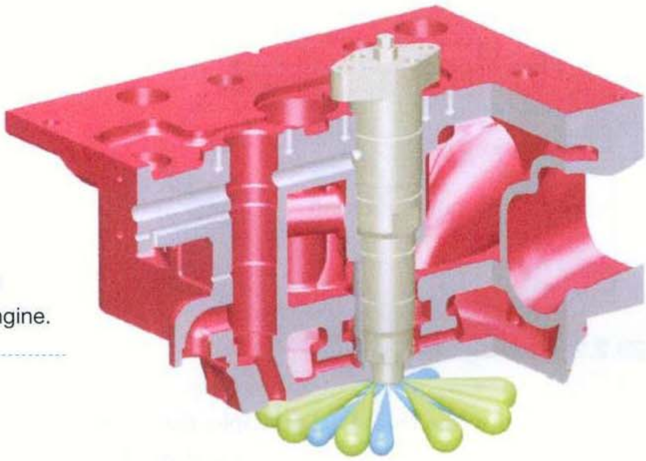
Delivered propulsion power:

	Nominal	Service
Electric pod	17 MW 40 %	17 MW 41 %
LIPS CP propeller	25 MW 65 %	24 MW 62 %
Total shaft power	**42 MW**	**41 MW**

Installed engine power:

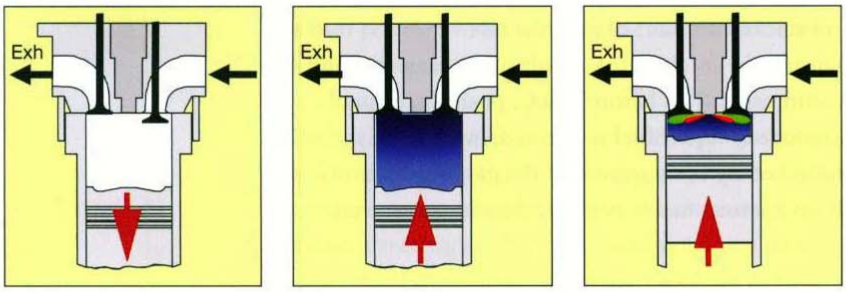
Mechanical transmission	0 MW
Electrical power generation	51.9 MW
Total installed power	**51.9 MW**

on liquid fuel. A disadvantage of this gas/diesel system in relation to the dual-fuel system is that the gas must be compressed to a very high pressure of approximately 350 bar, so it can be injected at the end of the compression stroke. This requires a considerable amount of power for gas compression.



▸ The principle of the gas-diesel (G.D.) engine.

blue: heavy-fuel oil
green: gaseous fuel



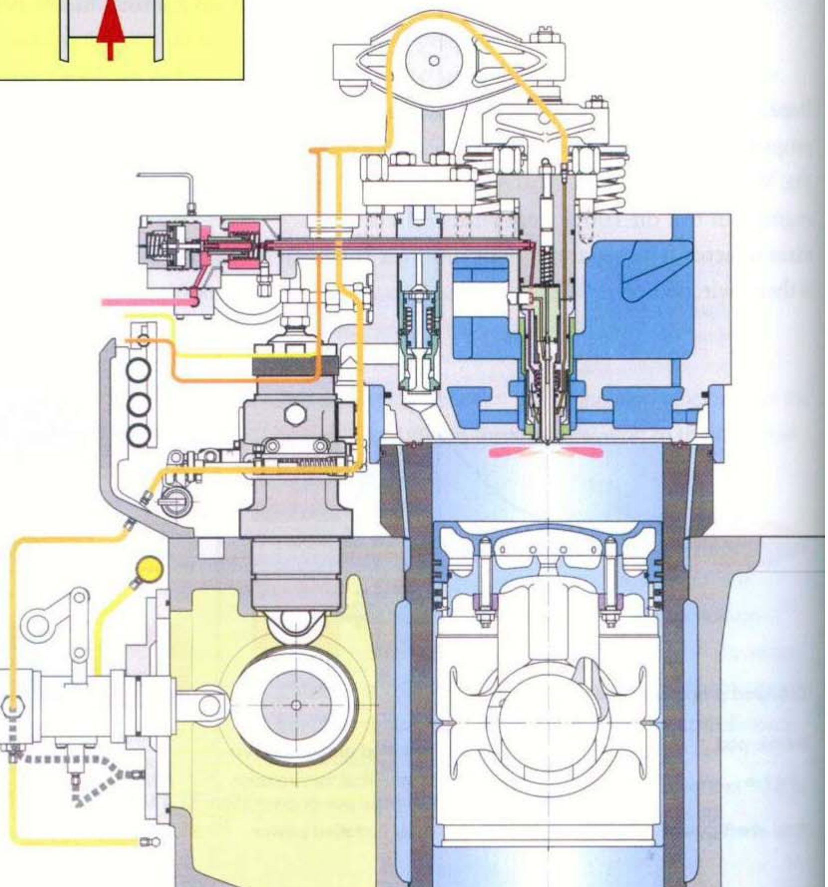
▴ The operating principle of the gas-diesel (G.D.)-engine.

white: air intake
blue: compressed air
red: heavy-fuel oil
green: gaseous fuel

29.2.3 Biofuels such as, olive oil, palm oil, rapeseed oil and other vegetable oils

In approximately 1900, Rudolf Diesel demonstrated a diesel engine that could run on peanut oil. In 2007, the interest in biofuels is huge. This is the result of the following two points.

1 **Due to the increasingly stringent requirements with regard to the emission of noxious substances in the exhaust gases, amongst which the toxic greenhouse gas CO_2, carbon dioxide.**
Biofuels absorb CO_2 from the atmosphere during their growth process; the CO_2 is converted to vegetable matter. This is known as photosynthesis. Photosynthesis is the conversion of, amongst others, CO_2, extracted from the atmosphere as a base material and light as an energy source to (energy containing) carbohydrates and O_2, oxygen, which is released into the atmosphere. This is known as 'CO_2 neutral'. Therefore, when biofuels are burnt, no extra CO_2 is emitted, as is the case with fossil fuels.



▸ A cross-section of a Wärtsilä 32 G.D. engine.

In principle, gas-diesel engines are manufactured in the same manner as diesel engines, only the fuel system differs. Note the horizontally positioned pilot-fuel pump on the side of the camshaft. Above left is the (red) gas supply. The heavy-fuel oil pump, above right of the camshaft, circulates the fuel without the main injector being activated.

CO_2 fertilization, another way of reducing carbon dioxide emissions.

The purified exhaust gases of gas engines are transported using large plastic pipes to the greenhouses, where the natural CO_2 content of the air (0.0375%) is increased, allowing many plants, such as, roses and peppers to grow faster. In this way, less CO_2 from fossil fuels is introduced into the atmosphere.

Peppers are suitable for CO_2 fertilization.

2 **Finitude of fossil fuels**

Due to the explosive growth in the use of energy, the demand for mineral fuels has increased exponentially. Countries such as China, India, Brazil and Indonesia require an ever-increasing amount of energy for their industries. Due to this, the worldwide demand is increasing, as is the fuel price. It is becoming increasingly clear how much large industrial countries are dependent on oil-producing countries. The economic and political need for the development of alternative fuel sources is increasing and in the search for alternatives, subjects such as 'less dependent on fossil fuels' are ever more important.

Some biofuel properties

The most important biofuels are olive oil, palm stearine and refined palm oil. Rapeseed oil is at present being investigated in Western Europe. Some properties:

- The pour point is high.
- A large variation in ash content.
- A large variation in acid number.
- A low calorific value.
- Very low sulphur content.

The following measures are necessary for the use of these fuels.

- Fuel systems: the storage tanks must be warmed and the fuel pre-heated before injection.
- Fuels with high ash and acid content, which cause contamination and corrosion, must not be used.

▶

The properties and composition of some vegetable fuels in comparison with diesel oil (L.F.O.) and heavy-fuel oil (H.F.O.).

Note, the high sulphur content and the high acid number of olive oil!
Refined palm oil is at present the most frequently used vegetable oil being tested.
LT = lower than.
Acid number LT3 lower than 3 mg KOH/g.

Bio-oil's physical & chemical property	Unit	LFO	HFO	Olive oil	Palm stearine	Refined palm oil
Density 15°C	kg/m³	864	993	910.7	921	915.8
Viscosity50176C	mm²/s	3.0	622	12.9	32.5	30.9
M. carbon residue	% m/m	< 0.5	0.6	0.0	0.20	0.15
Water	% v/v	0.0	0.6	0.5	0.1	0.1
Sediment total	% m/m	0.0	0.7	0.01	0.01	0.01
Ash	% m/m	0.01	0.082	0.20	0.01	0.01
Vanadium	mg/kg	LT1	370	LT1	LT1	LT1
Nickel	mg/kg	LT1	55	LT1	LT1	LT1
Sodium	mg/kg	LT1	30	594	LT1	LT1
Calcium	mg/kg	LT1	3	8	1	LT1
Phosphorus	mg/kg	LT1	N.A.	15	2	LT1
Sulphur	% m/m	0.7	2.3	0.05	0.05	0.05
Pour point	°C	-15	15	21	39	18
Acid number	mg KOH/g	LT1	LT3	136	0.08	0.12
Net calorific value	MJ/kg	42.6	40.1	36.55	36.49	36.77

▶

Noxious emissions of some vegetable fuels in comparison to diesel oil and heavy-fuel oil.

In addition, here olive oil is the least viable.

Emission	Unit	LFO	HFO	Refined palm oil	Palm stearine	Olive oil
NO_X	ppm, dry 15% O_2	860	970	990	990	1040
CO	ppm, dry 15% O_2	40	40	50	50	50
THC (as CH_4)	ppm, dry 15% O_2	170	80	LT30	LT30	LT30
SO_2	ppm, dry 15% O_2	118	463	< 2	< 2	< 2
CO_2	vol %	5.3	5.46	0.0	0.0	0.0
Particles	mg/Nm³ at 15% O_2	30	40-60	LT10	LT10	70

- The fuel-supply and high-pressure systems have larger dimensions as they must transport more fuel, due to the lower calorific value of biofuels in comparison with diesel and heavy-fuel oils.
- Biofuels can vary in composition. This is dependent on the refining process and the source of the vegetable matter supplied for the production of biofuels. It is therefore advisable to obtain the biological matter from the food industry, as they are required to investigate the quality and composition of their raw materials supplied.

Waste oils: Furthermore, used oils, such as, deep-frying fat, will play a part as alternative fuel sources for diesel engines in the near future.

▶

This large LNG tanker has two two-stroke crosshead engines of MAN-B&W, each drives a propeller.

29.2.4 Example of the propulsion of a large LNG tanker

A large LNG tanker propelled by a two-stroke crosshead engine.

Gas is a relatively 'clean' fuel and therefore there is a rising demand worldwide.
Where transporting natural gas by pipelines is not technically possible or economical, it can be transported by specially designed LNG tankers.
LNG – Liquified Natural Gas – at atmospheric pressure of $\frac{1}{600}$ the volume of natural gas.
'Natural' gas means naturally occurring gas; it is not produced in a refinery.
As expected, LNG must undergo extensive processing prior to transport and use as a fuel.
LNG is an odourless, colourless liquid with a specific mass of 450 kg per m³, 45% of the specific mass of water, which is 1000 kg per m³.
LNG comprises primarily methane. The amount of methane and other compounds is dependent on the oil fields / natural gas fields.

LNG is always transported 'cold' at atmospheric pressure; the boiling point is **-163 °C.**
The LNG is kept cold during transport by the removal of methane gas produced in the 'boil off' which is diverted to the engines and used as a fuel for ship propulsion.

The LNG tanks are normally never completely emptied when the ship is sailing because a certain amount of LNG is required to keep the tanks at low temperature.
The liquid is 'kept cold' during storage according to a well-known physical principle, as the vapour boils off. Heat for the phase change cools the remaining liquid. Only a relatively small amount of boil-off is required to maintain temperature because the insulation is very efficient. This phenomenon is also called auto-refrigeration.
There are mainly two types of gas tanks used today in LNG carriers. They are normally of the spherical (Moss) type and the membrane type.

The spherical or 'Moss' type storage tank.

The gas tankers are constructed according to the double-hull concept, including the bottom areas. Furthermore, for the LNG storage system, the cargo tanks must be installed separately in the ship's holds, and may not be part of the ship's structure.

The membrane type.

The tanks with the membrane system are rectangular and fully integrated into the hull and rely on the strength of the ship's hull. The stainless steel walls of the tank-membrane system are a very thin barrier (0.7-1.5 mm) supported by the insulation. In such tanks, only a relatively small amount of steel has to be cooled.

At present, LNG tankers are mostly being ordered with membrane tanks, because of their relatively higher utilisation of the hull volume for the cargo capacity.

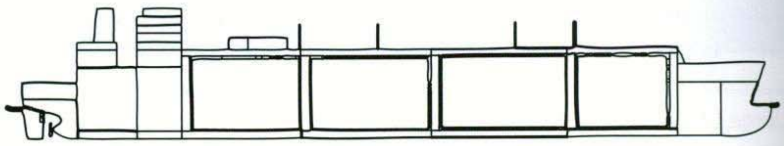
◄

One of the two-stroke crosshead engines, used for the propulsion of a large LNG tanker, a MAN-B&W 6S70ME-C engine with a shaft power of 18,660 kW.

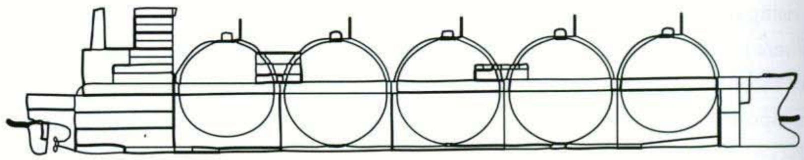
The fuel is heavy-fuel oil. These engines can be equipped with the G.I. system, the gas-injection system. Here, a high-pressure compressor together with a control system for the pressure in the LNG tanks and the gas flow to the engine comprising a complete dual-fuel system.

▶ The two containment systems used in L.N.G. tankers.

Above, the membrane type and below the spherical 'Moss' type.



138,000 m³ LNG carrier of the membrane type



138,000 m³ LNG carrier of the spherical (Moss) type

LNG tanker classes.

There are five LNG tanker classes, ranging from small up to Q-max.

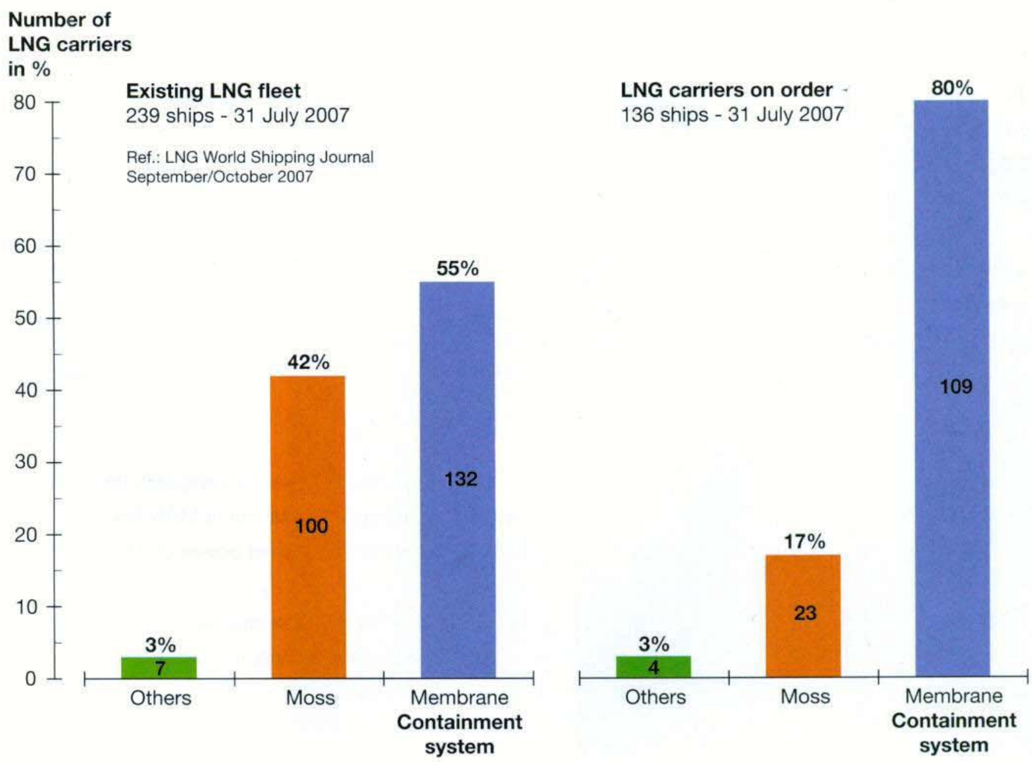
▶ The size classes of LNG tankers.

LNG tankers, as oil tankers and ore carriers, are very wide; this means that they have a slow sailing speed. Twenty knots is the maximum speed for the largest ships; Q max.

LNG carrier classes	Dimensions	Ship size - LNG capacity
Small	B: up to 40 m L_{OA}: up to 250 m	up to 90,000 m³
Small Conventional	B: 41 - 49 m L_{OA}: 270 - 298 m	120,000 - 149,999 m³
Large Conventional	T_{des}: up to 12.0 m B: 43 - 46 m L_{OA}: 285 - 295 m	150,000 - 180,000 m³
Q-flex	T_{des}: up to 12.0 m B: approx. 50 m L_{OA}: approx. 315 m	200,000 - 220,000 m³
Q-max	T_{des}: up to 12.0 m B: 53 - 55 m L_{OA}: approx. 345 m	more than 260,000 m³
Examples of special LNG carrier sub-classes: Med-max (Mediterranean maximum size) about 75,000 m³ Atlantic-max (Atlantic sea maximum size) about 165,000 m³		

▶ The number of existing LNG tankers in service per 31 July 2007, left, and the number LNG tankers on order per 31 July 2007, right.

Clearly shown is that most of the LNG tankers will be equipped with a membrane containment system.



Recently, there has been a dramatic increase in the size of the LNG carriers ordered.
Due to this extra propulsion-power output is required.
The shift from steam-turbine propulsion with a low efficiency to the internal combustion engine with a much higher efficiency allows not only new possibilities for the four-stoke engine but also larger power outputs for the two-stroke crosshead engine.

The LNG tankers on order at shipyards as of July 2007 show a clear trend.
Besides the conventional steam-turbine propulsion installations (40%), there are a large number with diesel engines using dual-fuel systems (59%).

The fuel system for LNG tankers with two-stroke crosshead engines.

This is known at MAN-Diesel as:
ME – GI system.
ME – electronically controlled liquid fuel.
GI – gas-injection system.

The fuel supply system comprises heavy-fuel oil and an LNG supply system.
Obviously, the supply of the H.F.O.–system is also in the ME version.

The LNG-supply system – the principle.

In order to be able to inject the gas in the cylinder during the compression stroke, a fuel-gas compressor is required that can deliver the LNG gas from atmospheric low tank pressure, up to a gas injection pressure of 150 to 265 bar.
The compressor must be able to deliver different types of gases with a controllable capacity at different suction pressures and temperatures.
The delivery pressure must also be variable.
In addition, the compressor must also be able to compress the gas for a gas-flaring installation, a type of burning, when propulsion is not required.

Note: the inlet-gas temperature is -160 °C!
A total of five compression stages are provided and the compressor is directly driven by an electric motor.

The ME-GI system of the engine.

The engines are identical to the existing ME version.
The only difference is that the ME-GI version uses gas as the major fuel.

LNG gas supply.

The gas is supplied to the cylinder using a gas supply double-wall piping and a gas-valve on the slightly modified cylinder cover. The gas-control block has an internal accumulator to level pressure fluctuations. In the cylinder cover, a gas injection valve is installed for control of the injected gas amount and injection timing.

In addition to the modifications to the engine/cylinder head, the following new systems have been installed.

- A Ventilation system, for venting the space between the inner and outer pipe of the double-walled piping system.
- A Sealing-oil system, delivering sealing oil to the gas valves separating the control oil and the gas oil.
- An **inert** gas system, which enables purging of the gas system with inert gas.

Additionally, there is a control and safety system consisting of a hydrocarbon analyser for checking the hydrocarbon content of the air in the double-wall gas pipes.

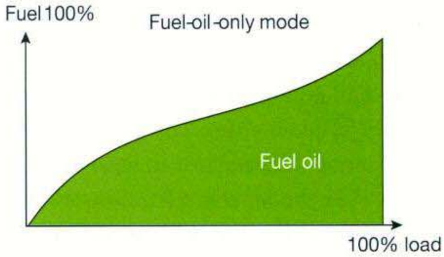
All the failures in the gas-fuel system result in an immediate changeover to the H.F.O. system or engine shutdown.
The changeover to fuel-oil mode is always done without any power loss to the engine.

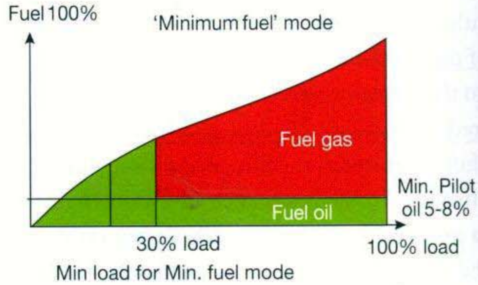
The gas-injection valve is controlled from the existing hydraulic system of the ME diesel engine.

Dual-fuel operation requires the injection of not only fuel, but also gas in the combustion space. Therefore, there are two fuel and two gas injectors in the cylinder head.

The fuel supply during operation.

One of the advantages of the ME-GI version is its fuel flexibility from which an LNG carrier can certainly benefit. At the start of a laden voyage, the natural boil-off gas holds a large amount of nitrogen and the heat value is low. If the boil-off gas is being forced, it can consist of both methane and propane, and the heat value is high. A two-stroke, high-pressure gas injection engine is able to burn these different fuels without a drop in the thermal efficiency.





▲

The possibilities of fuel use of MAN-B&W ME-G1 two-stroke crosshead engines for LNG tankers.

Left: Only fuel oil at every load.

Right: The 'dual fuel' system.

To 30% load, only liquid fuel is supplied.

From 30 to 100% load, 5 to 8% liquid fuel is supplied.

The remaining fuel, dependent on the load, is gas.

At full load, maximal 95% gas and minimal 5% liquid fuel (M.D.O. of H.F.O.).

There are two possible types of fuel supply:

- The fuel-oil mode, well-known from the ME engine. The engine is considered 'gas safe'. If a failure in the gas system occurs, it will result in a gas shutdown and it will be secured
- The minimum fuel mode is developed for gas operation, and it can only be started manually by an operator on the Gas Main Operating Panel in the control room.

The minimum fuel-oil supply is between 5 and 8% at full engine load dependent on the fuel quality. Both M.D.O. and H.F.O. can be used.
A MAN-Diesel is not able to guarantee a stable gas and pilot-oil combustion at an engine load below 30% of full load, and in that case the engine returns to Fuel-oil only mode.

Cylinder-lubricating oil.

Gaseous fuels have similar properties as fuels with low sulphur contents. Therefore, TBN 40 cylinder-lubricating oil is advisable.

CH 30

Bedplates and engine alignments, gearboxes, shafts, propeller shafts and generators

When a new diesel engine is installed in an existing ship, the existing bedplate is modified by a specialised company. Shown, the installation of a new engine at Breko, Papendrecht, The Netherlands. This is referred to as repowering.

A new engine with lower specific fuel consumption and lower pollutant emissions standing ready on the yard.

30.1 Introduction

Diesel engines for ship propulsion and as well as diesel gensets in power plants are usually mounted on welded sheet-steel bedplates. Diesel engines can be rigidly or flexibly mounted on their bedplates.

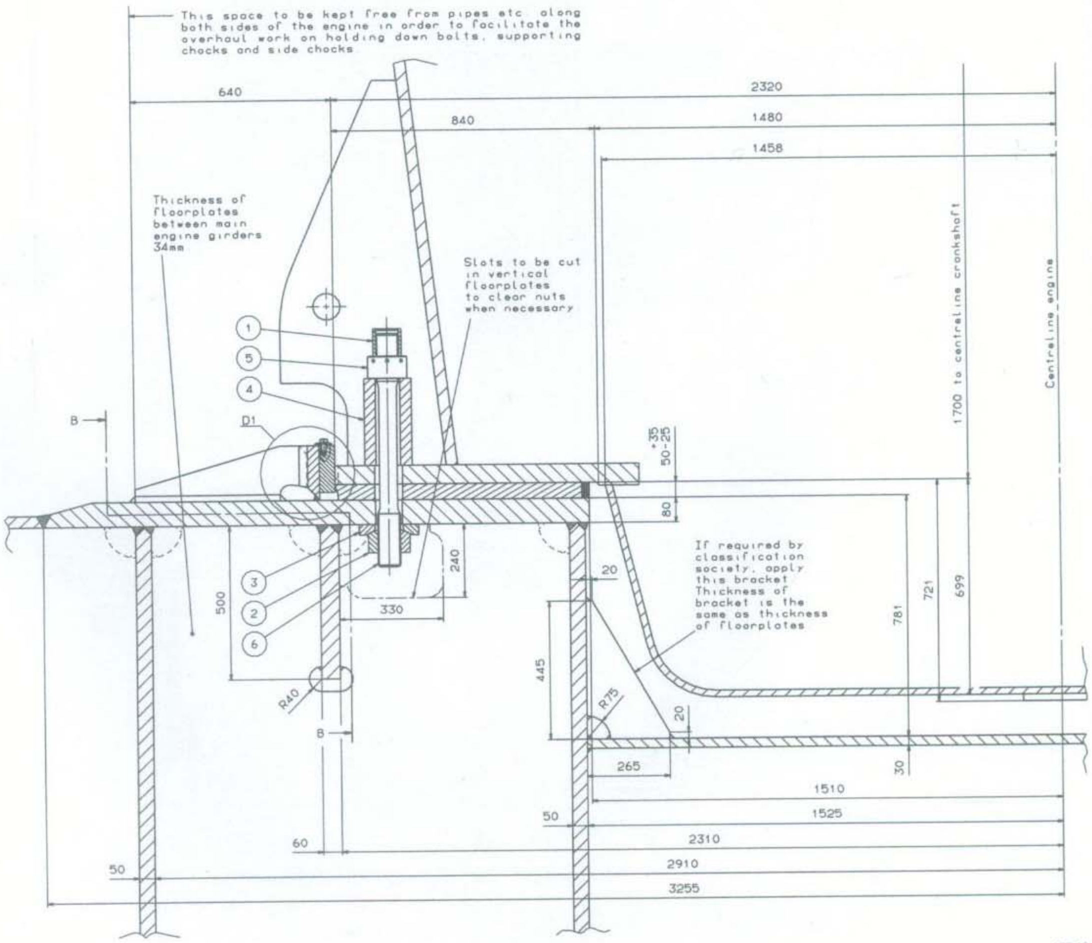
30.2 Ship propulsion

The bedplate of a propulsion engine must meet the following requirements.

- The bedplate must be able to support the weight of the engine without distortion to the bedplate or hull. The weight of the engine can vary, in category I, some tens of kilograms, in engine category IV up to nearly 3000 tons!
- The engine bedplate must be able to absorb and transmit the fluctuating forces created during engine operation.
- The bedplate must be designed and manufactured in such a way that the engine does not shift when the ship runs aground or is involved in a collision. Therefore, the engine is often provided with end stoppers.
- If the thrust block is installed on the bedplate or in the gearbox at the rear of the engine, the bedplate must be able to transmit the fluctuating forces to the ships structure.

30.3 Construction of the bedplate - Engine category IV

Larger ships usually have double bottoms; these can be used for the storage of fuels, ballast water and drinking water. In order to achieve this, the double bottom is divided into various compartments, both longitudinally and transversely.
Around the propulsion engine, there is a separate space with double bottom, often surrounded by a cofferdam.
The double bottom below the engine is reinforced with girders and often forms an integral part of the engine bedplate.
The plates are usually thicker. At the locations where the longitudinal holding-down bolts are placed, the horizontal strip is thicker and ribbed.



▶
Detailed drawing of the bedplate for a two-stroke crosshead engine of MAN–B&W ME.

The bedplate of a MAN–B&W ME engine with an epoxy resin filler below the welded engine flange. Note the long elastic holding-down bolt.

1 *protecting cap*
2 *spherical nut*
3 *spherical washer*
4 *distance pipe*
5 *round hydraulic nut*
6 *holding-down bolt*

The hydraulically tensioned holding-down bolts of a large two-stroke crosshead engine with side chocks

1 *engine frame*
2 *hydraulic holding-down bolts*
3 *side chocks*

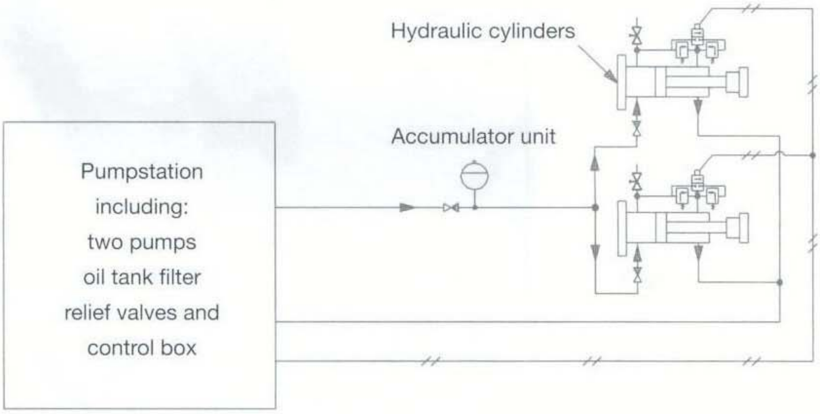
The hydraulic unit with two cylinders.

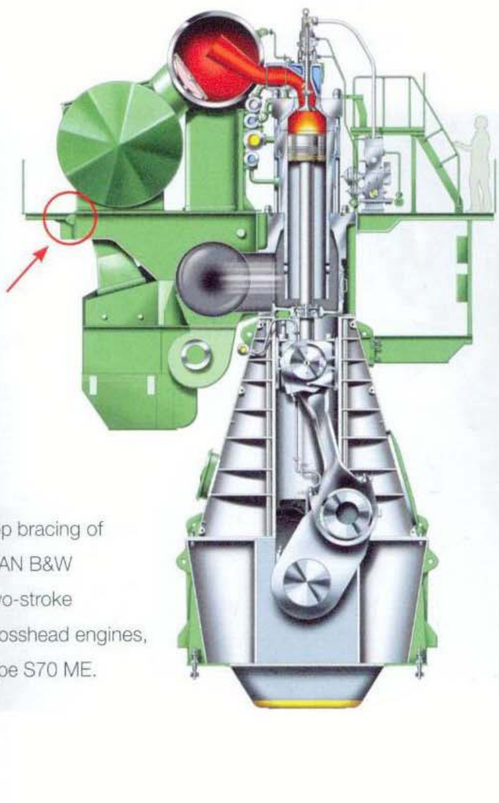
The controls are represented by the double slanted lines on the electric wiring.

'Top bracing'

The engine is equipped with a mechanical or hydraulic top bracing to absorb the lateral forces. These forces can be considerable.

Also, see Chapter 17, Vibrations and balancing.





Top bracing of MAN B&W two-stroke crosshead engines, type S70 ME.

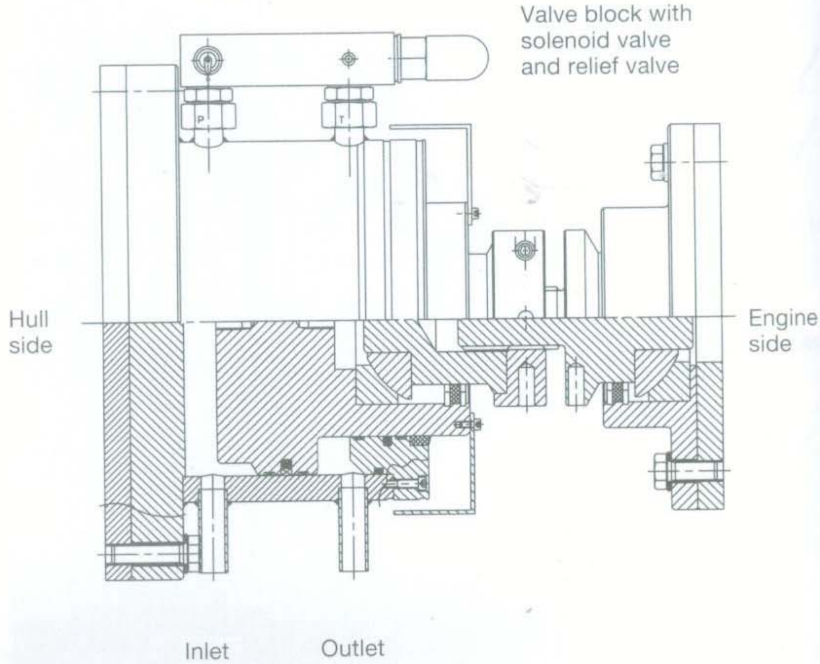
Force per mechanical top bracing and minimum horizontal rigidity at attachment to the hull		
Engine type	Force per bracing in kN	Minimum horizontal rigidity in MN/m
K108ME-C	available on request	
K98ME	248	230
K98ME-C/GI	248	230
S90ME-C	209	210
K90ME/GI	209	210
K90ME-C	209	210
S80ME	available on request	
S80ME-C	165	190
K80ME-C	165	190
S70ME-C/GI	126	170
L70ME-C	126	170
S65ME-C/GI	available on request	
S60ME-C/GI	93	140
L60ME-C	93	140
S50ME-C	64	120

Mechanical top bracing.

The forces per top bracing at attachment to the hull of MAN–B&W ME two-stroke crosshead engines. The forces per top bracing vary from 6.4 to 24.8 tons!

A cross sectional view of hydraulic bracing for the same engines.

This system consists of two to four hydraulic cylinders, two accumulators and hydraulic pump with control box. The locations on the hull are reinforced where the bracings are fitted. The hydraulic cylinder maintains a constant force between engine and hull. A relief valve ensures that the maximum allowable force is not exceeded.

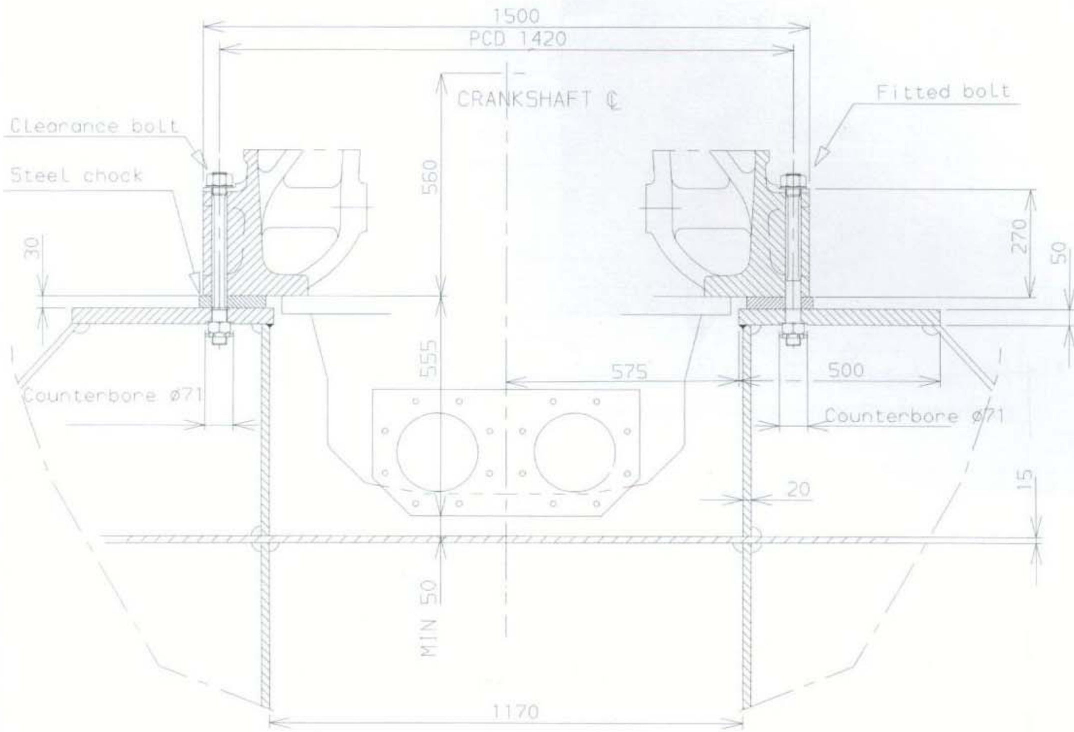


A certain type of top bracing in a Wärtsilä Sulzer RTA 84 two-stroke crosshead engine.

The use of top bracing is dependent on the number of cylinders. For instance, a five-cylinder engine, usually requires top bracing, occasionally longitudinal bracing is required! The X- and H-vibration modes of a seven-cylinder engine are low and top bracing is not required (Project Manual RT-Flex 60C).

A type of top bracing in a Wärtsilä Sulzer RTA 96 two-stroke crosshead engine.

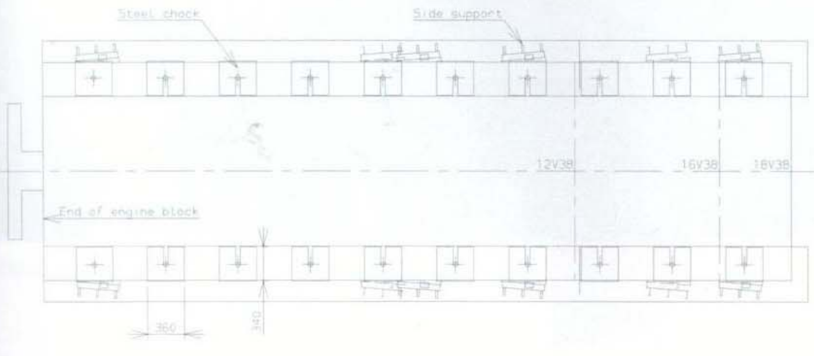
30.4 Examples of rigid mounting – Engine category III



▲
The bedplate for a Wärtsilä 38-B diesel engine.

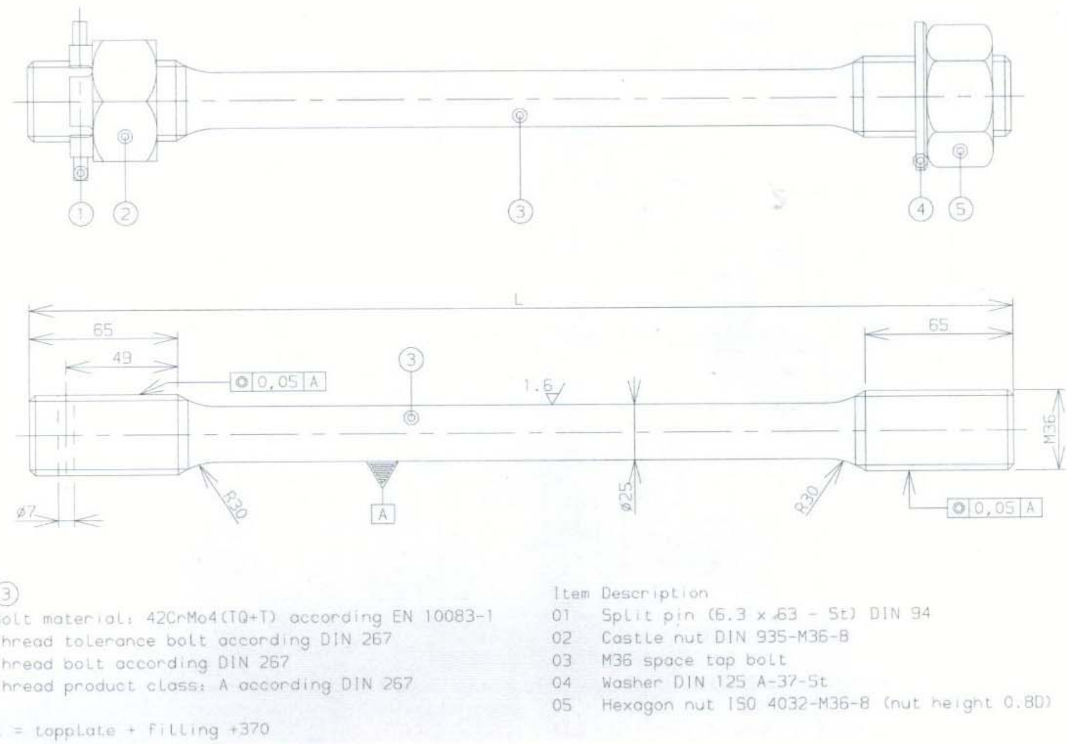
The engine is rigidly mounted on steel or resin chocks (epoxy resin), or flexibly mounted on rubber elements.
The bedplate and double bottom are as stiff as possible in all directions to absorb the dynamic forces generated by the engine, reduction gearing and the thrust block.
The steel strips below the engine are often tapered to an inclination of 1 to 100.
The engine manufacturer provides the dimensions of the wedge-type steel chocks. The chock material may be cast iron or steel.
The surfaces supporting the chocks are often levelled mechanically to produce a large contact face.

At least 80% bearing surface must be obtained.
The clearance holes in the chock and top-plate strip must be at least 2 mm larger than the bolt diameter, except when equipped with fitted bolts.
All engines are equipped with side supports. These are fitted before the alignment of the engine. The steel wedges between the engine block and the side supports must be fitted as soon as the engine and bedplate have obtained their operating temperatures.
The holding-down bolts are usually provided with a hydraulic nut at the top and a conventionally secured hexagon lock nut at the bottom. It is recommended that the bolts be manufactured from high-tensile strength steel, for instance, 42 CR MO 4 TQ + T alloyed steel or comparable. The contact faces of the nuts on the seat of the engine block (top) and in the bedplate (bottom) must be counter-bored to prevent additional bending stresses in the bolts.



◄
Wärtsilä 38 B V-engine.

The bedplate viewed from the top with ten holding-down bolts on each side, steel shims and six side supports on each side.



▶
A holding-down bolt for a Wärtsilä 38 B in-line engine.

Holding-down bolts for engines have to meet high standards

Mounting the engine on the bedplate

Hydraulically tensioned holding down bolts are used in large engines of categories III and IV. These have a large bolt elongation (strain) which improves the safety against loosening of the nuts and breakage.
The bolts are extremely large.

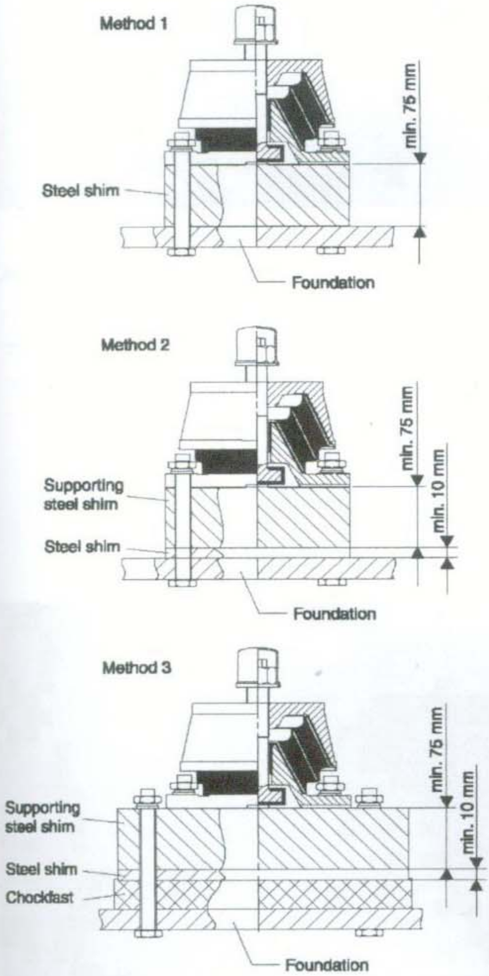
▶
Installing the engine block of a medium-speed four-stroke diesel engine, used to drive a generator on a large container ship.

1 *welded steel-plate lower frame bolted to the reinforced double bottom*
2 *hydraulically tensioned holding-down bolts*
3 *engine block*

30.5 Resilient mounting of propulsion engines – Engine categories I, II and III

This is often applied in categories I and II and occasionally in category III engines.
All two-stroke crosshead engines in category IV are rigidly mounted. Most other engines are resiliently mounted to reduce the transmission of sound and vibrations produced by the engine to the hull and consequently the ship's structure including the accommodation. A diesel engine produces low frequency vibrations and high frequency noise in the ship's structure.

Also, see Chapter 16, Noise, origin and damping and Chapter 17, Vibrations and Balancing.



▲
The options for the flexible mounting of MAN–B&W L16/24 diesel engines used for power generation.

Shown a series of three conical flexible rubbers. The compression of the rubber elements by the weight of the engine lies between 5 and 11 millimetres.

▲
Various flexible elements of manufacturer, Rubber Design, Heerjansdam, The Netherlands.

There are special flexible elements available for each type of engine and arrangement.

▲
A flexible rubber mounting below a large Caterpillar–MAK 43-L propulsion engine.

The total surface of the rubber mounting for this engine weighing 110,000 kg is 2 m².

◄
A side-support bolt for the same Caterpillar 43-L propulsion engine.

The engine can be bolted with end-chock bolts if the rubber elements are compressed.

In engine category II, flexible arrangements are often used, not only for the propulsion engine, but also for the gearbox.
In propulsion engines, the engine is frequently mounted on flexible elements. The engine and the generator of gensets are rigidly mounted on the bedplate and the bedplate is mounted on flexible elements.

In engine category I, flexible arrangements are often chosen to improve comfort on dinghies and yachts. The propeller thrust is usually absorbed by a thrust bearing.

▲
The arrangement of two diesel power units, category II.

Scania diesel engines drive the electric generators on a feeder-containership.
The engine and the generator are mounted on flexible elements.

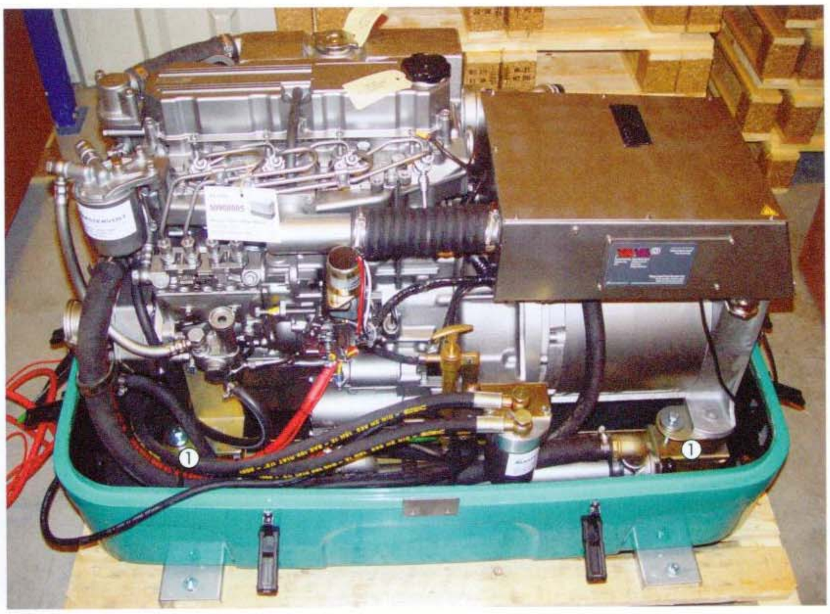
▲
A small diesel engine (Deutz) for a boat or small vessel.

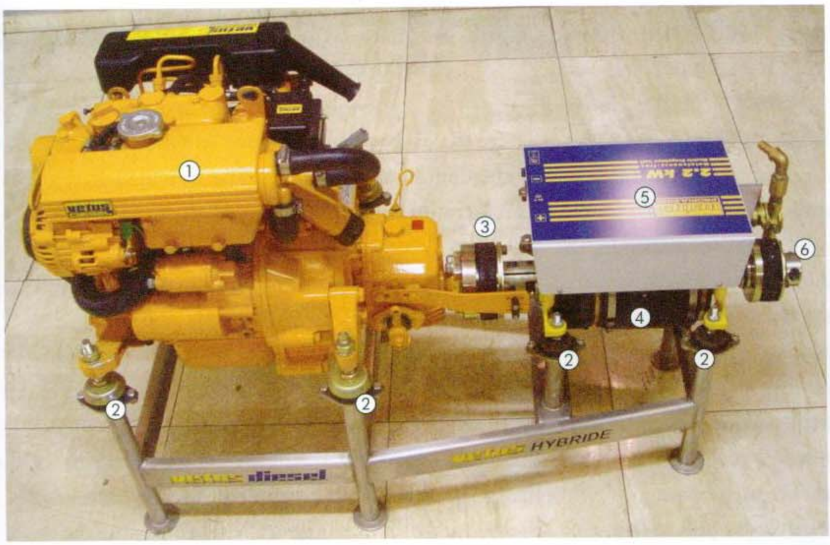
System: Vetus den Ouden, Schiedam, the Netherlands. The flexible elements ensure that the transmissions of vibrations to the hull are minimal.

1 *flexible elements*

▶
A flexible arrangement (1) of a genset of Mastervolt, the Netherlands.

Obviously, prerequisite is a vibration- and soundproof arrangement.





A hybrid arrangement of Vetus den Ouden.

A small, flexibly arranged diesel engine, category I, drives a flexibly arranged generator. A battery on top of the generator. A flexible coupling between the diesel engine and the electric generator/electromotor.

1 *diesel engine, category I*
2 *flexible mountings*
3 *flexible coupling between the engine and the generator/electromotor*
4 *generator/electromotor*
5 *battery*
6 *flexible coupling between the electromotor and propeller*

30.6 Alignment of engines

The alignment of propulsion systems with engines is crucial and vital in the prevention of damage to the system.

Objective:

- To prevent damage to shafts, bearings, couplings, gearboxes.
- To reduce the vibration levels in the engines, bedplates and their surroundings.
- To reduce the noise generated by these vibrations.

30.6.1 Alignment of stern tube, gearbox and four-stroke diesel engine of a new ship

When a ship is built, the components are, when possible, placed in their final installation position. During the construction of the engine room, the engines, gearboxes and generators are placed on the bedplate.
The available space on both sides of these components is evenly distributed.
The engine is placed as described previously and then the gearbox is positioned and provisionally aligned.

Line boring a stern tube.

This is aligned and then welded in place. The line-boring machine is then aligned, so that the stern tube can bored to the correct size.

Once the gearbox is aligned, the shaft generator can be installed in relation to the gearbox. The space is evenly distributed, dependent on the accuracy required of the section.
When the rear section, containing the stern tube and the bracket, is installed, the position is determined and checked using 3D-measuring methods and/or lasers during the welding process. The reference for positioning the rear section is the outlet flange of the gearbox.

After welding of the rear section, a reference line is established to determine how the various bearings must be positioned. The positioning of the bearings is determined with a propeller-shaft alignment calculation during the engineering phase of the ship. After positioning the bearings, the installation of the propeller shaft can continue. The propeller- shaft flange is the reference for the final alignment of the outgoing shaft of the gearbox.
In the aforementioned calculation, a so-called GAP/SAG value is obtained. The gearbox is aligned within a tolerance of +/– 0.05 mm of this GAP/SAG value. GAP is radial deflection, SAG is axial deflection.

After mounting the gearbox, the alignment of the engine can begin. The engine flywheel is aligned in relation to the input shaft of the gearbox. For this alignment, a GAP/SAG value is also calculated in advance. The engine is aligned within a tolerance of +/– 0.05 mm of this GAP/SAG-value. Subsequently the crankshaft deflection is measured. The combination of the GAP/SAG value and the measured crankshaft deflection determines if the engine is correctly aligned. When correct, mounting and fastening of the engine can be performed.
Once the engine has been aligned, the shaft generator can be aligned on the same GAP/SAG-principle.
When the installation is complete and all the components are connected, the factual bearing load at each bearing position can be established by weighing the shaft in combination with strain-gauge measurements.
The values measured must correspond with the values calculated in the engineering phase.

30.6.2 Alignment of an existing engine rigidly mounted

There may be various reasons for the re-alignment of the engine and gearbox. The flywheel is aligned in relation to the input shaft of the gearbox. A GAP/SAG value is calculated in advance. The engine is aligned within a tolerance of +/– 0.05 mm of the calculated GAP/SAG value. The crankshaft deflection is then measured. The GAP/SAG values in conjunction with the measured crankshaft deflection ultimately determine if the engine is correctly aligned. If this is the case, the main engine can be mounted and affixed to the bedplate. The alignment and the crankshaft deflection are re-checked.
After the classification agency approval, the flexible coupling is installed.

30.6.3 Alignment of an existing engine resiliently mounted

Elaborate installation instructions must be provided by the engine supplier for the alignment of an engine on flexible elements.
These instructions detail the process of engine alignment. These instructions can vary per of flexible element type.
The flexible elements must be loaded during 48 hours after installation in order to obtain the correct measurements. The elements must be levelled with respect to the front and rear within a certain tolerance (an indication is within 1 mm). Compression of the various elements is also restricted (an indication is within 2 mm).
It is recommended that the engine be filled with water and oil to achieve an optimal alignment. Once the components are aligned, the flexible elements can be fixed.

Some data with regard to damage ensuing from inaccurate alignment

Damage as a result from incorrect alignment may vary from damage to the white metal in the sleeve bearings, breakage of the couplings, generation of severe vibrations to breaking of the crankshafts. The bearings in the different components are monitored with temperature sensors. As soon as the temperature rises, an alarm signal is given. Vibration and/or temperature increases in the engine can be indicative of an alignment problem. It is self-evident that not all vibrations or temperature related problems are produced by incorrect alignment.

Important points of interest

- Ship structures distort during operation.
- Components 'grow' during operation.
- Above-mentioned factors influence the alignment.
- There are many factors that influence alignment with resilient mountings.
- Performing regular alignment check-ups extends the engine life.
- Generally, flexible couplings have a better absorption capacity than the component bearings between which they are mounted.
- Flexible couplings are used as torsional-vibration compensators, not for compensation of alignment errors.
- Economising during construction often does not compensate for the cost of repairs during operation. Using a combination of laser measurements, shaft weight and strain-gauge measurements, the actual condition of the propeller shaft can be precisely determined.
- Quick adjustment of the components after an alignment check saves money.

30.7 Flexible arrangement of diesel engines, piping, cables and other fittings connected to the engine

Large four-stroke trunk-piston engines are also often placed on flexible rubber strips to limit the vibration and sound transmission to the hull, and therefore the ship's accommodation.
Modern feeder-containerships with the propulsion engine and the accommodation aft are open to vibration and noise nuisance.
The high engine-power output, the slim stern, the light hull construction, the tall and narrow superstructure with the rotating propeller below are not conducive to a quiet and vibration-free environment!
This means that the various pipelines attached to the diesel engine must be provided with flexible connections.

Piping and fittings around the diesel engine

The exhaust-gas pipes of a rigidly mounted diesel engine have flexible compensators in the piping; this allows them to move freely during fluctuating temperatures. Vibration- related problems in diesel engines often cause tearing behind the turbo-blower, towards the sound damper. In smaller and flexibly arranged engines or engines, which have a bedplate mounted on flexible elements, all pipes and cables (ducting) connected to the engine and/ or moving bedplate must be taken into account.

◄ The stern of a feeder-containership.

The accommodation is located astern. The propulsion engine is situated directly below the rear containers. The propeller is situated directly below the accommodation. This construction is conducive to vibrations.

Two spare exhaust-gas line compensators for a two-stroke crosshead engine.

There are a large number of systems available for fuel, lubricating oil, starting air, exhaust gases and cooling water to prevent damage or tearing. It is advisable to check periodically the deflection of a certain point on the engine in relation to a fixed point in a flexibly arranged diesel engine.
If, for instance, the fuel injection of a six-cylinder in-line engine functions inadequately, this can usually established by measuring the deflection.

30.8 Mounting methods for propulsion engines and other components with a critical alignment

The following filling methods for alignment of diesel engines, gearboxes and generators are applied:
- with steel chocks;
- with steel shims;
- with a resin chock, often epoxy resin such as Epocast 36;
- with adjustable chocks.

Compensators for, amongst others, cooling-water pipelines and exhaust-gas piping required for the flexible arrangement of diesel engines.

30.8.1 Steel chocks

Advantages of steel chocks
- Relatively inexpensive.
- Easy to install.

Disadvantages steel filling plates
- It is prohibited to place them below propulsion engines.
- They do not fill the space sufficiently.
- Too many contact points are created.

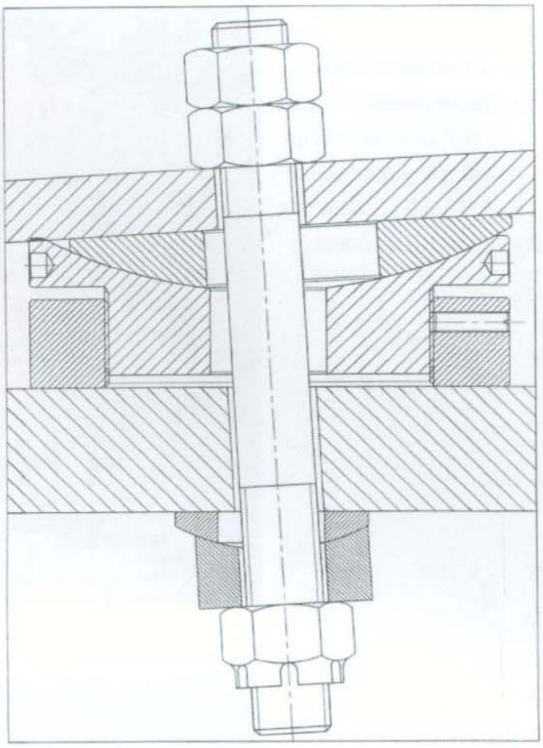
30.8.2 Epoxy resin such as Epocast 36

Advantages filler
- Provides all-over support, both for the bedplate and the engine flange.
- Cheaper than steel chocks and less time-consuming.
- Reduces high-frequency vibrations.

Disadvantages filler
- Time-consuming and more expensive in comparison to adjustable chocks.
- Hardening of the epoxy resin takes relatively long, depending on the ambient temperature 24 to 48 hours.
- The method is non-adjustable. The engine cannot be re-positioned.
- Special packaging material required for air transport.
- Installation must be performed by specialists.

30.8.3 Adjustable chocks

Advantages of van adjustable chocks
- Height adjustable
- Self-levelling, therefore they can accommodate angular differences in mounting surfaces and can compensate errors.
- Adjustable during life cycle of the machinery.
- Accurate mounting for engine alignment is simple and quick.
- Expensive milling of the bedplates is not required.
- No extra work required for the bedplates.
- There are adjustable chocks available in all sizes and shapes.
- Minimal height for confined installation spaces is 20 mm (dependent on the type).



◄
A cross section of an adjustable chock, brand 'Vibracon'.

The chock is self-levelling and therefore can accommodate angular differences in both contact faces.

◄
An adjustable chock of 'Vibracon'.

The height is adjustable and the spherical washer compensates for the angular upper face.

▼
The generator and the diesel engine installed on a fixed bedplate and provided with adjustable chocks from Vibracon.

▶

A gearbox mounted on adjustable chocks.

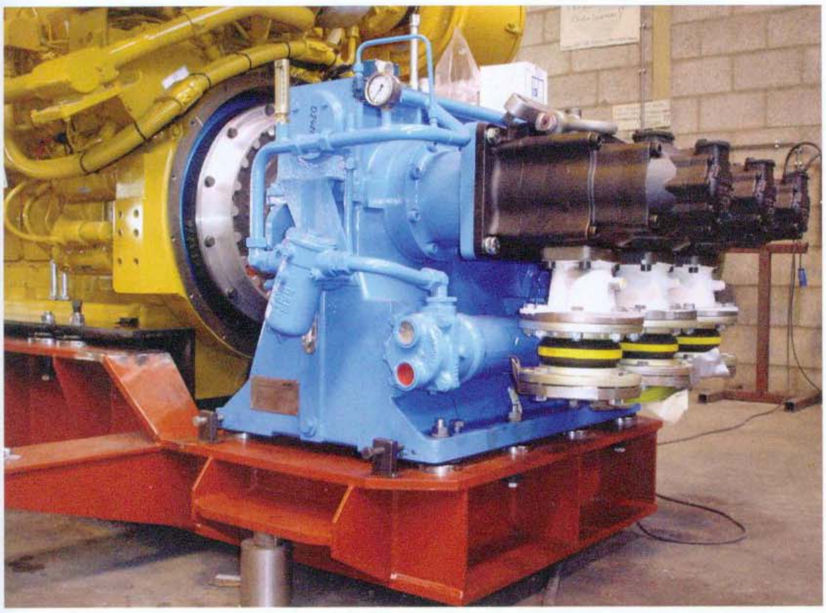
▶

Adjusting the chocks below the diesel engine.

▶

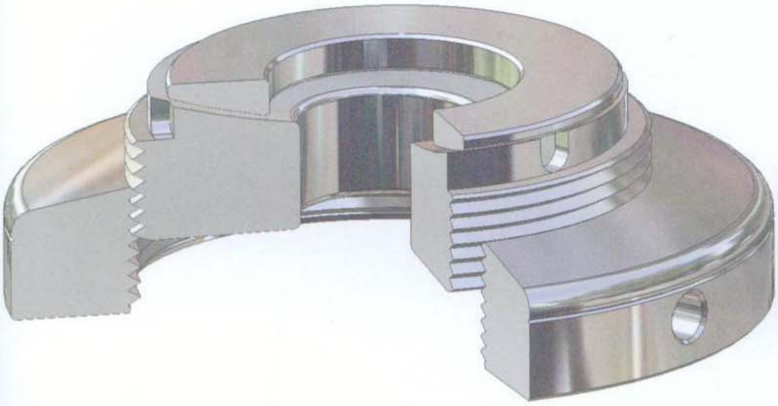
An adjustable chock with:

1 *engine frame*
2 *bedplate*
3 *adjusting bolts for alignment*
4 *adjustable chock*
5 *holding-down bolt*



◄

Vibracon Low Profile chocks below Stork gearboxes and high-pressure pumps. Driven by a Caterpillar 3512 diesel engine, Category II.



◄

The design of a Vibracon Low Profile adjustable element.

◄

An electric generator (yellow) between a Yanmar diesel engine and gearbox with reverse clutch.

The power output at a voltage of 230 volt and frequency of 50 Hz is 3 kW; ideal for a moderately sized yacht.

CH 31

Propellers





A large controllable-pitch propeller.

One of world's largest propellers; the fixed-pitch, six-bladed propeller of a large container ship weighing 85 tons.

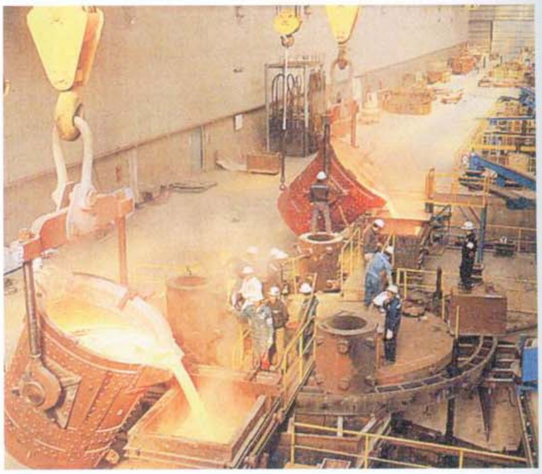
31.1 Introduction

Most propellers for seagoing vessels are manufactured of copper alloys. However, for inland shipping and small ships and yachts, propellers are also made of cast iron, aluminium alloys, stainless steel and numerous synthetic compounds.

31.1.1 Large fixed-pitch propellers

Large fixed-pitch propellers are cast in a mould and then machined.
Apart from the accurate finish of the propeller hub, a smooth finish of the blades is also essential; the propeller must be able to rotate in the water in which drag is reduced to a minimum.

Small propellers can be cast in large series and only weigh several kilograms. Larger propellers are cast individually and can weigh up to 140,000 kg!



▲
Fixed-pitch propellers and propeller blades being cast.

A large foundry of the engine works and shipyard Hyundai in South Korea. The unfinished fixed-pitch propeller sometimes weighing 100 tons, must be cast rapidly in a single casting in order to obtain a homogeneous structure.

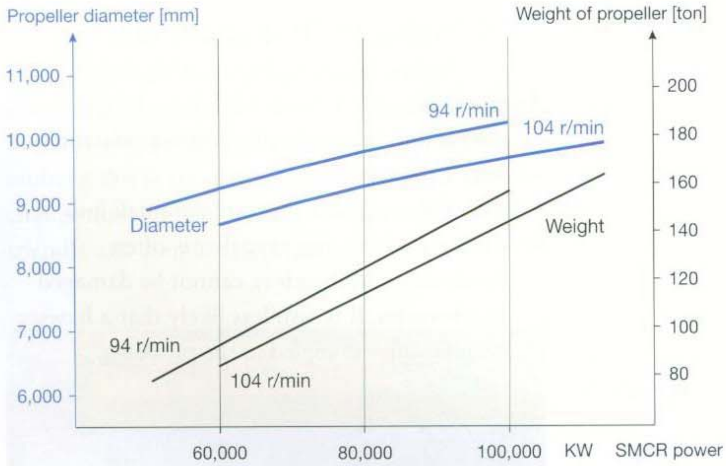
▶
Propeller blades for controllable-pitch propellers at the engine works of MAN–B&W in Frederikshavn, Denmark. They also manufacture two-stroke crosshead engines with cylinder bores up to and including 500 mm as well as four-stroke engines. The blades are of the 'high-skew design' and highly polished.

▶
The magnified surface of a propeller blade.

The smoother the surface, the less the resistance of the rotating propeller in the water.

▲
Machining a large six-bladed fixed-pitch propeller for a container ship.

Note the supports for the propeller blades of approximately 4 metres. These prevent bending during the machining processes.



▲
A table for propeller sizes.

horizontal: shaft power of the engine in kW

centre:
the black lines represent the propeller weight for 94 and 104 revolutions per minute.
The blue lines represent the propeller diameter for 94 and 104 revolutions per minute.

Example: At 80,000 kW shaft power and an RPM of 94, the propeller diameter is approximately 9700 mm and it weighs 130 tons!

31.1.2 Accuracy of the finish

It is important that the hub and the blades be finished with the utmost precision. This is required to ensure optimal operation, as well as preventing the generation of additional vibrations, which would negatively affect the propulsion system of the ship.

31.1.3 Materials used in the manufacture of large propellers

Material requirements
- corrosion-proof
- sturdy
- ease of casting
- easily machined

A well-known alloy is 'Cunial', an alloy with copper, nickel, iron and manganese. 'Cunial' represents the chemical symbols for the elements: Cu-copper, Ni-nickel and Al-aluminium.

The mechanical properties of Cunial	
Mass density	7650 kg/m^3
E-modulus (at 20 °C)	121000 N/mm^2
Poisson's ratio	0.33
Yield stress	min. 250 N/mm^2
Tensile strength	min. 650 N/mm^2
Elongation (longitudinal)	min. 18%
Coefficient of thermal expansion (from 20 to 100 °C)	$16\text{-}10^{-6}$ K^{-1}

◄
The mechanical properties of propeller material 'Cunial'.

The base material for large propellers is often 'old' coins, essentially a copper alloy.

Material	NiAl	CrNi
0.2 proof stress	min. 250 N/mm^2	min. 380 N/mm^2
Tensile strength	590 – 780 N/mm^2	600 – 790 N/mm^2
Elongation	min. 16%	min. 19%
Impact strength Charpy V notch	30 J	21 J
Brinell Hardness	min. 150	240 – 300

◄
Comparison of Cunial 625 and 13/4 chromium-nickel-steel materials.

31.2 Fixed-pitch propellers

Advantages
- simple and relatively cheap to manufacture
- very strong
- high efficiency due to relatively small hub
- during docking, the propeller is often stationary and therefore cannot be damaged by obstacles. It is also less likely that a hawser will become entangled in the propeller.

A fixed-pitch propeller fitted hydraulically to the hub. A key seat as locking device is no longer required.

Disadvantages
- The propeller must be driven in **two rotational directions:**
  1 either the propulsion engine must be reversible (large plants);
  2 or a reverse clutch must be mounted between the propulsion engine and the propeller (smaller plants).
- In extreme conditions, such as heavy-weather conditions with head-on winds and high swell the propeller can become heavily loaded.
- During manoeuvring with large plants, the maximum speed is generally the 'manoeuvring' speed, approximately 80% of the speed at full-load.

31.2.1 Use of fixed-pitch propellers

Fixed-pitch propellers are often used in the large power-output category IV engines, as well as the smaller power-output engines of categories II and I.

Engine category IV

Large container ships, bulk carriers and oil tankers are usually equipped with fixed-pitch propellers. The manoeuvring required by these vessels annually is limited and with the use of bow and stern thrusters (container ships) and tugs (container ships, in particular bulk carriers and oil tankers), manoeuvring when arriving or departing from port is facilitated.
Due to the size of the ships in this category, poor weather conditions hardly affect the hull and consequently the engine power output.
The engines are always low-speed two-stroke crosshead engines with a low speed so reduction gearing is not required.
All the engines are directly reversible.

Also, see Chapter 14, Starting systems of diesel engines.

The tunnel shaft between the two-stroke crosshead engine (background) and the propeller shaft (not visible in the foreground).

A reversible diesel engine allows a high-efficiency fixed-pitch propeller to be installed. A simple and common propulsion system for large ships.

The rotational direction of the engine can be rapidly changed using a free wheel, which disengages the propeller from the driving shaft. For very small vessels, such as yachts and cabin cruisers, this is standard. Vessels that must often change direction use a similar system, as it is user-friendly and simple to operate.

▲

A relatively small two-stroke crosshead engine of MAN–B&W, type ME.

This MC 50 can drive a fixed-pitch propeller and is directly reversible. However, a controllable-pitch propeller is often installed. This engine has a common -rail system.

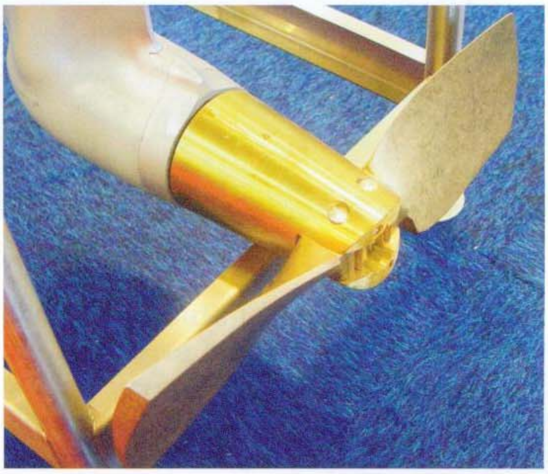
Engine categories I and II

These four-stroke high-speed diesel engines running on M.D.O. usually have a fixed-pitch propeller; a reverse gear changes the rotational direction of the propeller.

◀

A small four-stroke high-speed diesel engine ('Vetus') with an angle gear set through the yacht's hull.

The reverse coupling is located directly behind the engine. The propeller type is a foldable propeller: during sailing, it folds back due to the flow of the water giving a minimum drag when sailing.



◀

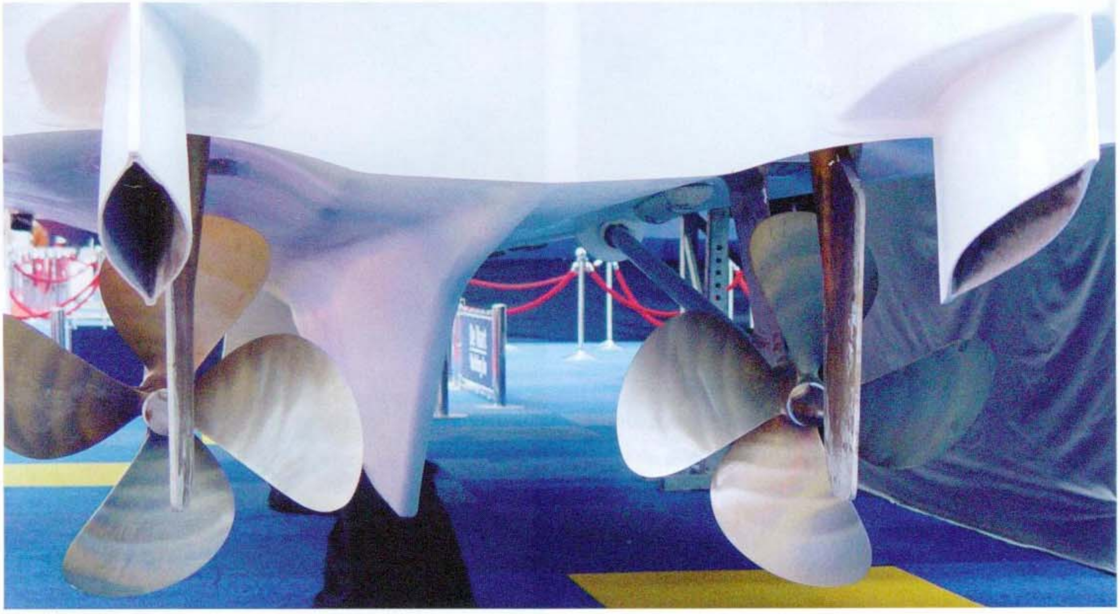
A foldable propeller in operation. The rotating blades fold out due to centrifugal forces.

◀

A propulsion installation with a four-stroke high-speed Caterpillar diesel engine running on M.D.O.. Used for inland navigation.

Reduction gearing with a reverse coupling connected to a fixed-pitch propeller is installed at the rear of the engine (back).

The propeller shaft is stationary. The yacht sails. The propeller blades fold back. There is minimum drag from the propeller.



A yacht with two fixed-pitch propellers.

Both four-stroke high-speed diesel engines are installed with reduction gearing and reverse coupling.

31.3 Use of controllable-pitch propellers of the engine categories

The position of the propeller blades can be modified in stages so that with the same rotational direction of the propeller, it can be steered from full speed ahead via the 'zero feathering position' to full speed astern. (For its operation, see below).

Advantages of controllable-pitch propellers

Controllable-pitch propellers are often used and have many advantages.

- **It allows for rapid manoeuvring,** from full speed ahead, to full speed astern; the propulsion engine continues to run in one direction at the same nominal speed, and can therefore be rapidly heavily loaded.
- **Every speed can be achieved** without having to stop the engine.
- **The average efficiency for ships with alternating loads,** such as tugs, fishing boats and dredgers can be improved by adjusting the position of the propeller blades to the load ('pitch').
- **With the correct reduction gearing** the engine speed can be reduced to a propeller speed which has the highest efficiency. This is effected by placing an electric-shaft generator drive on the reduction gearing.
- **It is possible to substitute a damaged propeller blade under water, if required.**

Disadvantages of controllable-pitch propellers

- **Propeller efficiency is lower than with fixed-pitch propellers due to the relatively large hub, which contains the mechanism for adjusting the blade pitch and the bearings.**
- **They are more prone to damage by objects in the water than are fixed-pitch propellers.**
- **On average they are three to four times more expensive** than fixed-pitch propellers.
- **The hydraulic system can fail** and will then require maintenance.

◄
A four-bladed controllable-pitch propeller. Note the shape of the blade: the 'skew' type.

This minimizes the forces on the actuating mechanism inside the propeller hub and decreases cavitation.

31.3.1 Use of controllable-pitch propellers

These are used in three engine categories.

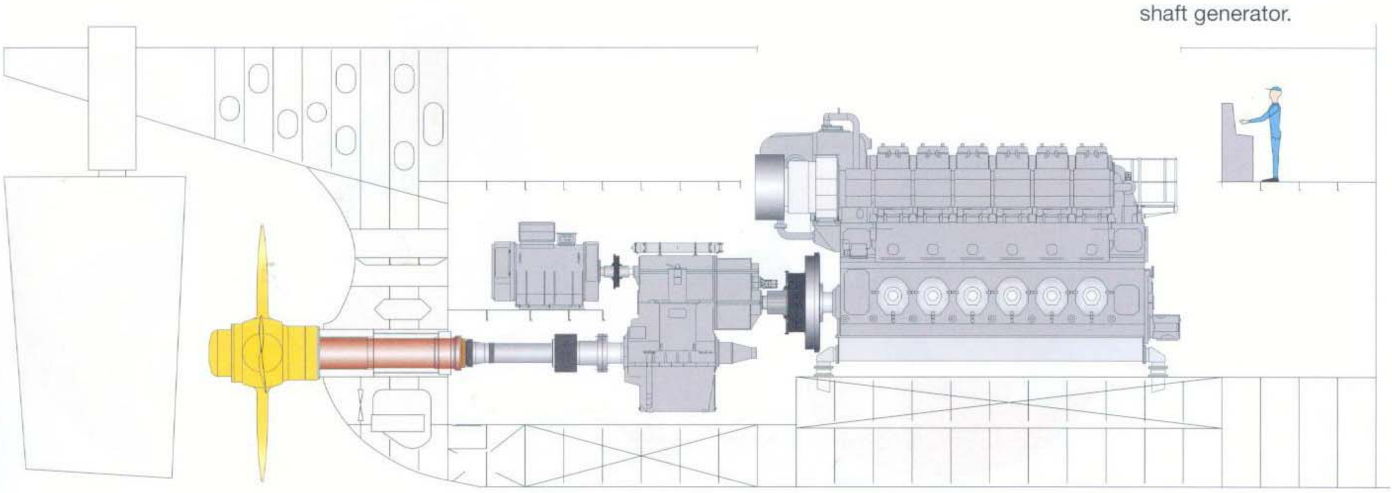
Engine category II

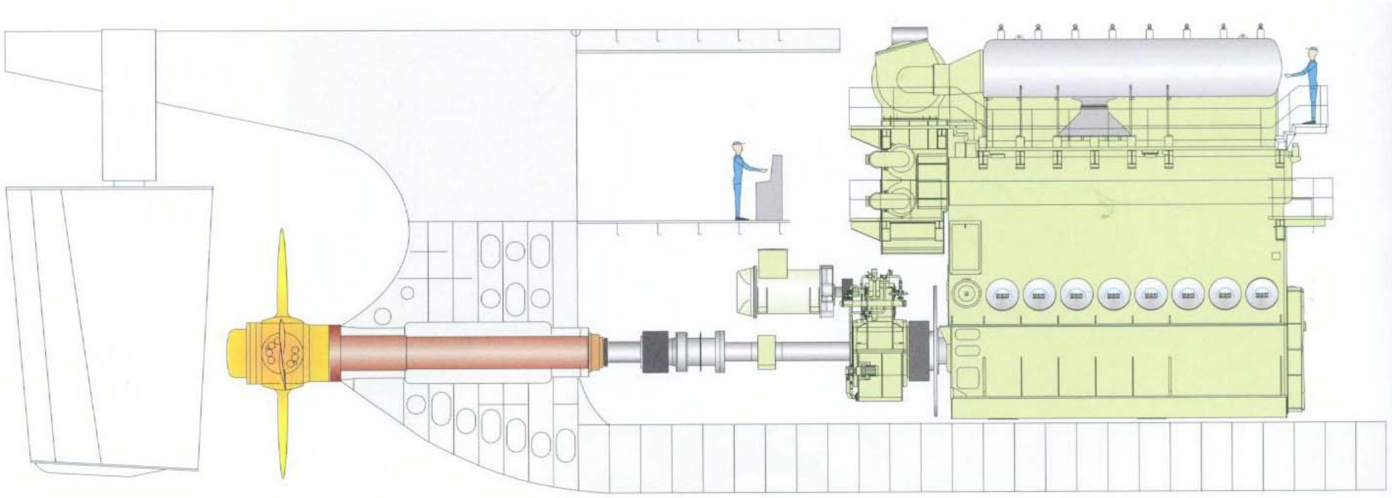
For the larger four-stroke high-speed diesel engines running on M.D.O., they are mostly used for propulsion engines with higher power outputs. Is this is not the case, then reversing/reduction gearing is installed.

Engine category III

The controllable-pitch propeller is often used in this category of four-stroke medium-speed diesel engines running on H.F.O.. Furthermore, a large number of feeder-container ships, chemical tankers, cargo carriers, ferries and passenger ships are equipped with controllable-pitch propellers.

▼
The most common propulsion plant with controllable-pitch propeller: a medium-speed diesel engine with reduction gearing and shaft generator.





▲
A smaller type two-stroke crosshead engine with shaft generator and controllable-pitch propeller. Reversing gearing, common in these types of engines, is not required.

Engine category IV

In this category of two-stroke low-speed crosshead engines with small bores, complete propulsion systems with intermediate shaft, shaft-generator drive, propeller shaft and the controllable-pitch propeller are often installed.
Engine manufacturer MAN–B&W supplies these systems up to and including a 700 mm cylinder diameter.

31.3.2 Propeller shapes

For each ship type, propellers are designed which provide the best efficiency, produce little noise and generate minimal vibrations. The propeller shape is a decisive factor. **The propeller is custom-made for every ship!**

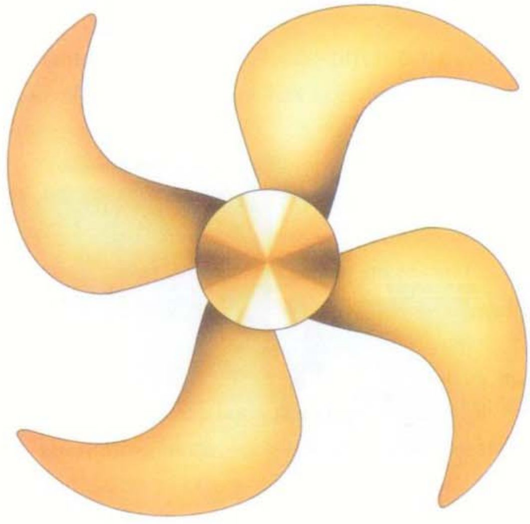
Several examples

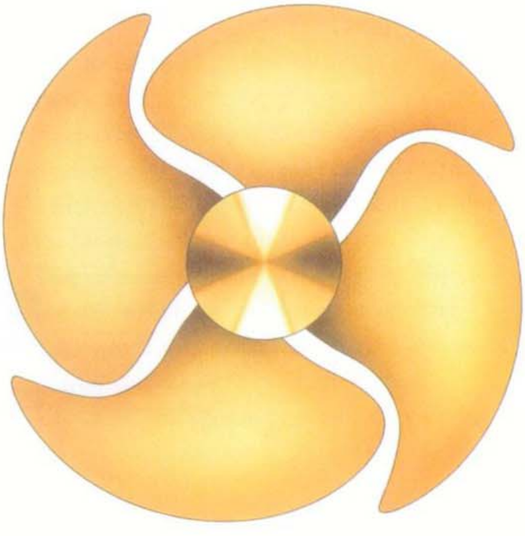
Example 1: Large container ships

As these types of ships travel huge distances each year with large engine-power outputs and of whose operational costs fuel costs are a major component, the total propulsion efficiency is of utmost importance. A large two-stroke crosshead engine with a total efficiency of 52% and a large diameter propeller with a total efficiency of 75% can achieve a total efficiency of $0.52 \times 0.75 = 039$ or 39%. This is **exceptionally high!** Therefore, 39% of the supplied fuel (100%) is ultimately used to propel the ship at a certain speed. In all other propulsion systems, this is always lower and at times significantly lower.

▶
The four-bladed fixed-pitch propeller of a large container ship. 'Skew' type blades.

▶▶
The controllable-pitch propeller for ferries





Example 2: Ferries

Modern ferries must have low sound and vibration levels. Propellers are specially designed to achieve this. The manoeuvring properties are also essential.

Example 3: Pontoons

Pontoons, which travel relatively short distances, are often provided with propellers port and starboard for efficient mooring and departure, thus allowing for the rapid transportation of passengers and vehicles.
When both propellers are set in a position in which they do not provide propulsion, they must be placed in such a position that a minimum of resistance is caused.
This also applies to ships with two propellers, where one propeller is regularly taken out of commission.

Example 4: Reversible propeller for passenger ships

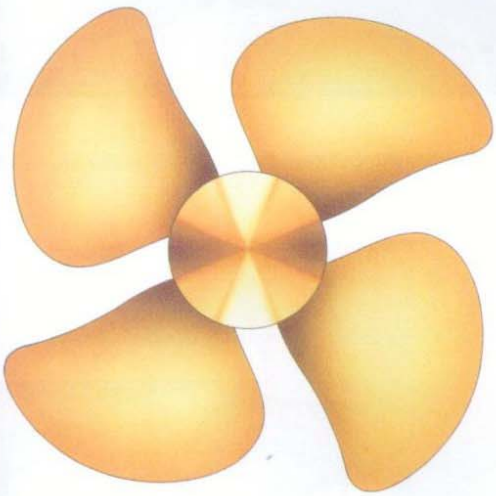
These are mainly designed for maximum 'comfort', that is: minimum noise and vibration levels. The shape of the blade is referred to as the 'skew-shape'. The blades are bent in the rotational direction.

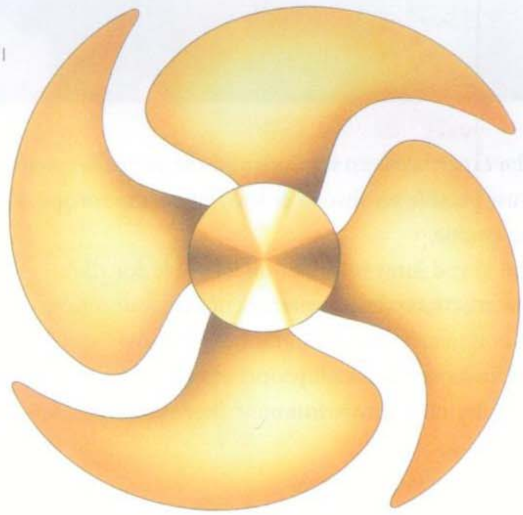
Example 5: Icebreakers

These are designed to comply with the highest ice classes stipulated by various classification agencies. Furthermore, they must be able to provide very high pulling forces ('bollard pull').

Example 6: Propellers for anchor-handling ships, tugs and tenders

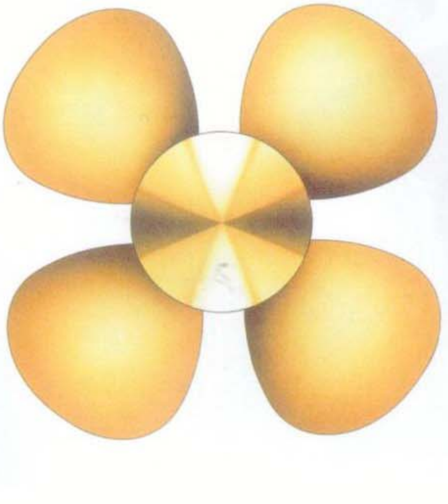
They are able to generate high pulling forces as well as having good manoeuvrability, also under extreme weather conditions.
A high pulling power is essential and is achieved with either a fixed tunnel thruster or a steerable thruster.

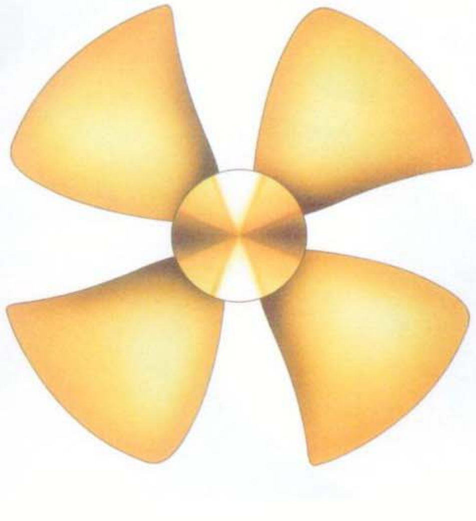




◄◄ The controllable-pitch propeller for pontoons.

◄ The controllable-pitch propeller for passenger vessels.





◄◄ A controllable-pitch propeller for icebreakers.

◄ The controllable-pitch propeller for various vessels, such as anchor-handling ships, tugs and tenders.

31.4 Fixed-pitch propellers, construction

31.4.1 Traditional fixing

The traditional attachment of the propeller to the taper end of the propeller shaft is by means of a keyed joint comprising a key and locking nut. The taper is approximately 1:12, so the hub can be firmly affixed to the shaft. The key lock does not serve to absorb the thrust, but is used for security's sake. When complications arise, it must absorb the entire torque.

▶
Disassembly of a fixed-pitch propeller from a nozzle. The hammer wrench is in position.

The taper between the hub and the propeller shaft must be able to absorb at least twice the torque of the engine.
The keyed joint is in fact undesirable for the construction of the taper or conical connection. In practice, hair cracks and fractures may occur in this location. In small propellers of, for instance, pleasure craft, attention must be paid to this issue.

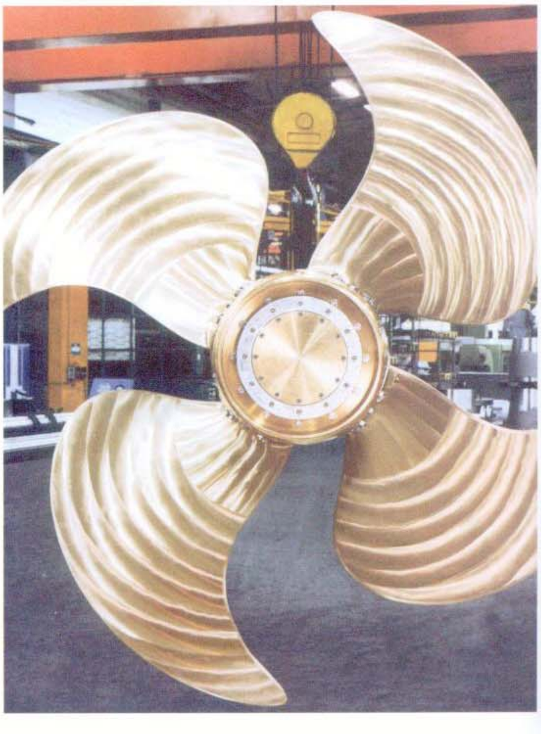
▼
A fixed-pitch, six-bladed propeller weighing 80 tons for the 6800 TEU Southampton class, a large container ship. The base material used for manufacture is 'old coins'.

31.4.2 Propeller fixing without a key, a key-less joint

The key-less assembly is often used for large propellers. The torque of this conical connection should be at least three times that of the diesel engine as this construction lacks the security of a keyed joint. Large propellers are always hydraulically fitted to the shaft. This requires a meticulous working method. As security a locking nut, protected from the corrosive seawater by a cap, is hydraulically attached to a bolt at the shaft end.

▲
Five-bladed, without key.

▼
A four-bladed controllable-pitch propeller.

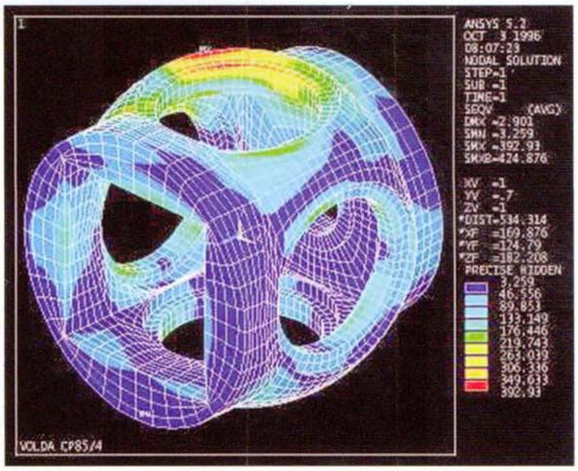


31.5 Controllable-pitch propellers, the construction

The hydraulic actuating device of the blades in larger controllable-pitch propellers is installed in the hub.

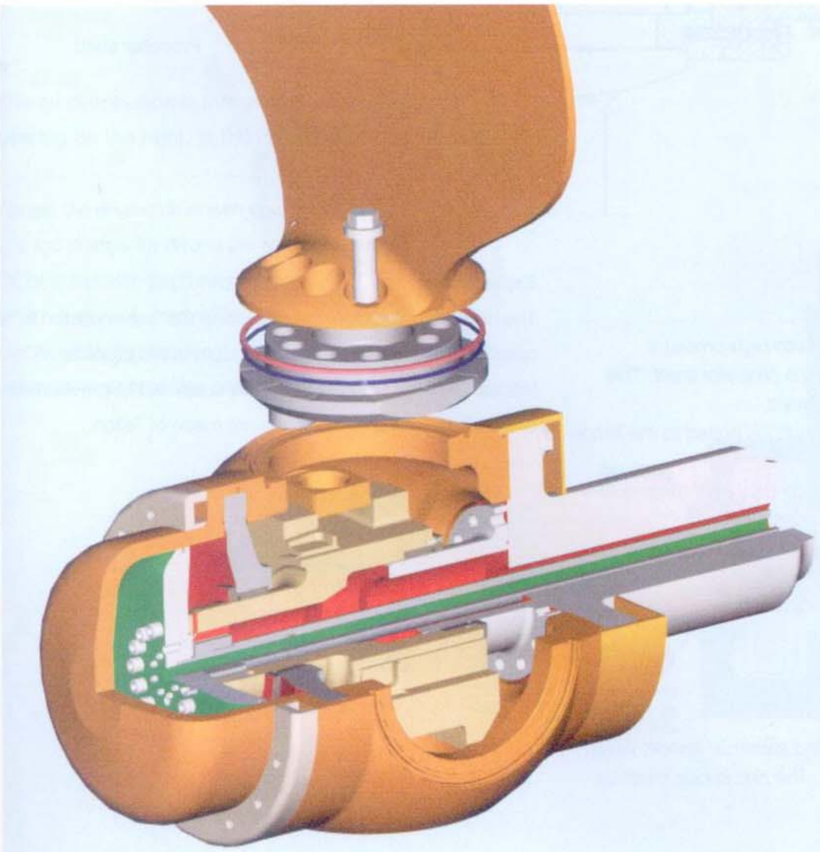
◄ The cast and machined hub prior to installation of the parts, such as the pitch-actuating mechanism and propeller blades.

▲ A hub with the propeller shaft installed.



▲ A hub must be very strong.

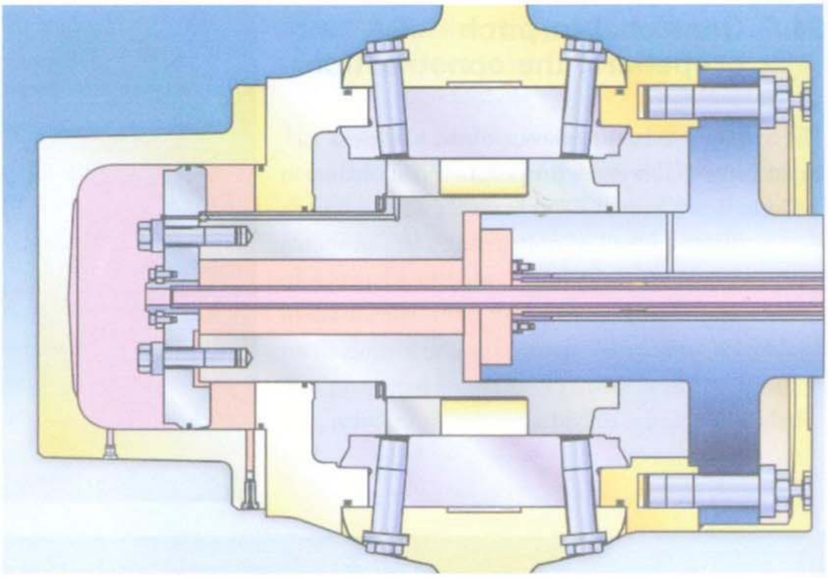
A 'finite-element method' can be used to calculate the load on each element of the hub element.
red: highest load
dark blue: lowest load

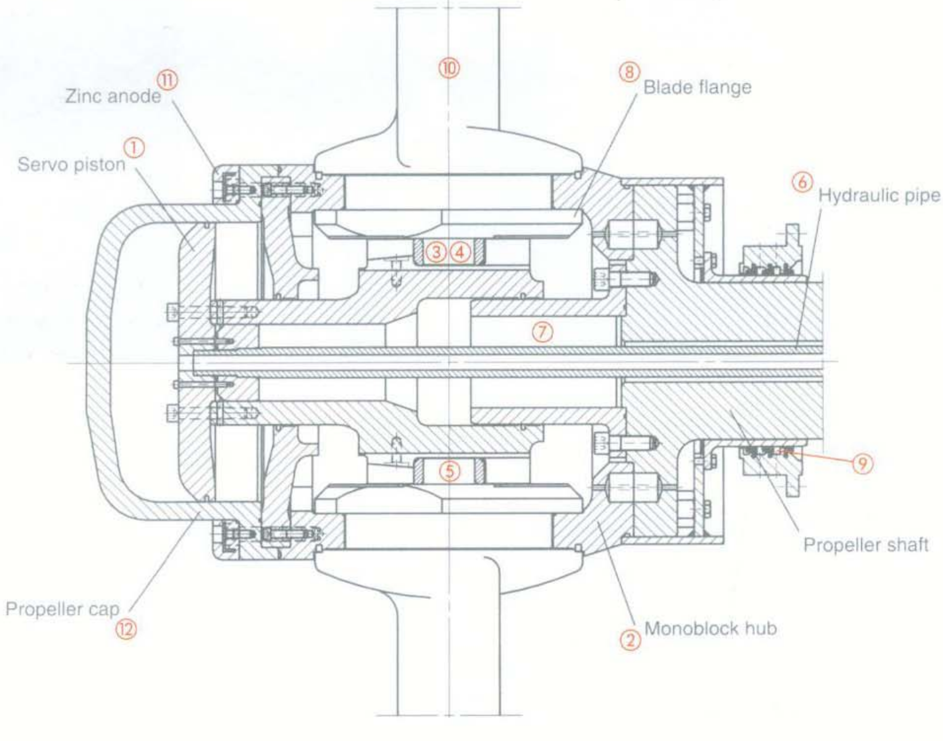


◄ A cut-away section of a hub showing the hydraulic servomotor for pitch setting.

Green: the oil-supply line and piston on the pressure side.
Red: oil discharge.
Yellow: pitch-actuating mechanism.

▶
A cross-section of a Wärtsilä Lips controllable-pitch propeller.



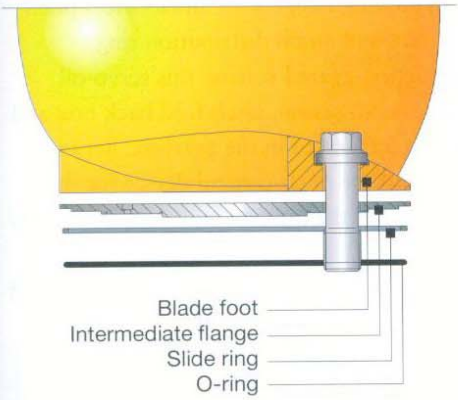


▲
A cross-section of the hub.

1 *the servo-piston is displaced with high-pressure hydraulic oil via a tube inside the propeller shaft. This allows precision-pitch adjustment.*
2 *the hub casing ('mono-block hub') is bolted to the flange of the propeller shaft.*
3 *the servo-piston moves an eccentric pin of the propeller blade with a 'slot-crosshead construction'.*
4 *slot-crosshead construction*
5 *eccentric pin*
6 *high-pressure hydraulic-oil supply line*
7 *hydraulic-oil return line*
8 *blade flange*
9 *rear-shaft seal*
10 *propeller blade*
11 *zinc anode: this is the so-called sacrificial anode, which prevents seawater corrosion. The zinc anode must be regularly substituted.*
12 *propeller cap*

Explanation

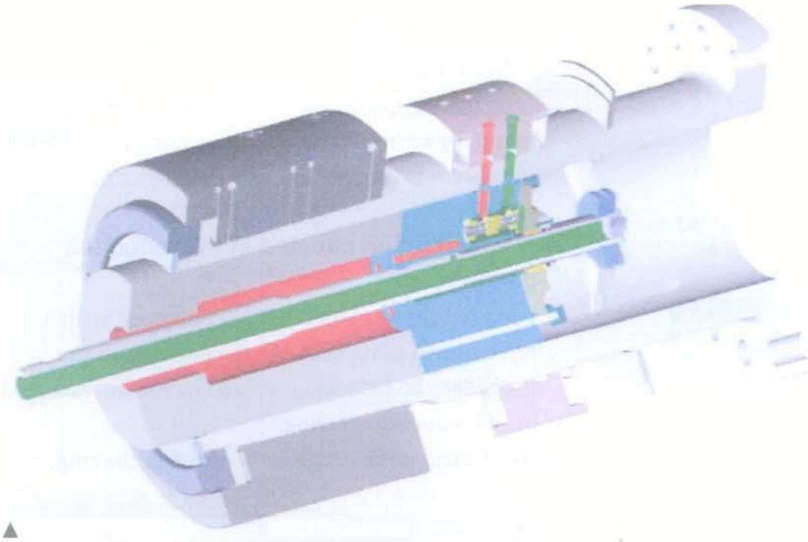
The hydraulic oil system for controlling the 'servo'-piston is completely isolated from the oil system of the propeller blades. Only the blade sealings are subjected to gravitational oil pressure. The blade sealings are made of Teflon.



▲

The insertion of an extra intermediate flange below the blade foot allows underwater replacement of the blades.

The propeller -blade flange is lubricated via a separate drilling.



▲

The same coupling flange with the oil distribution and pitch indicator (in colour).

black: hydraulic coupling muff
light blue: oil distribution
green: hydraulic supply oil
red: hydraulic return oil
dark blue: sealings

The supply and discharge of oil

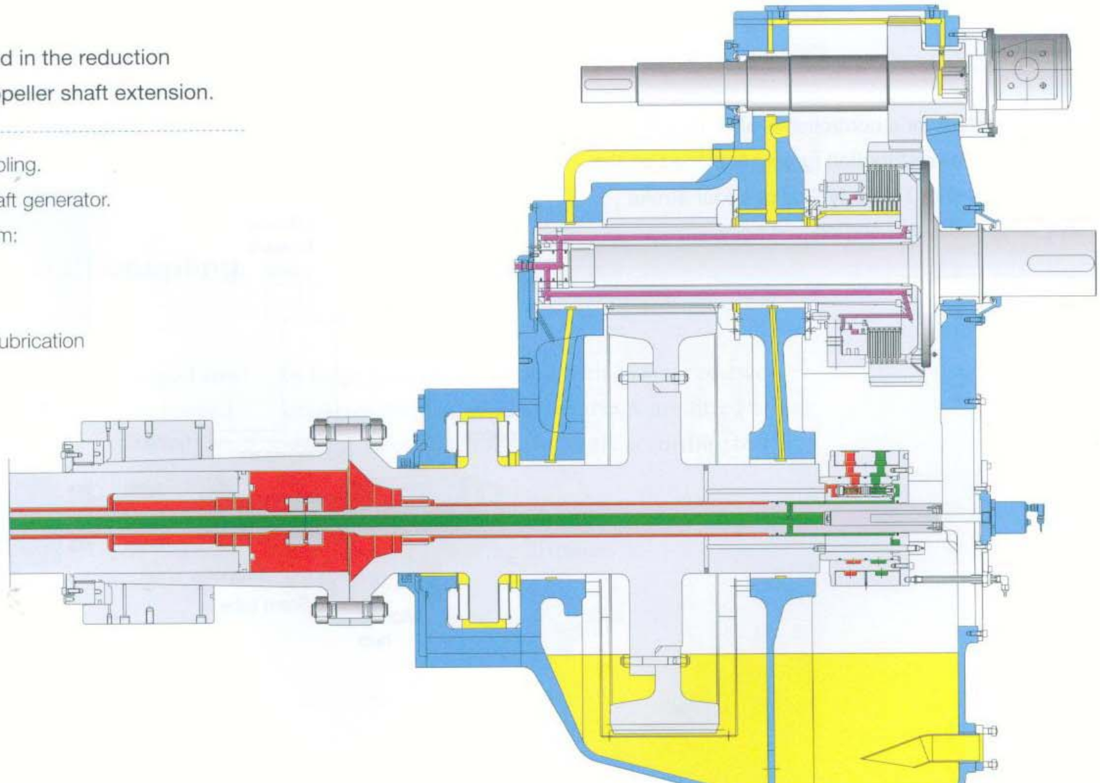
Two different systems are used.
In propulsion systems with low-speed two-stroke-crosshead engines, the oil-distribution box is positioned in the shaft line.

In propulsion systems with medium-speed four-stroke engines equipped with reduction gearing, the oil-distribution box is positioned on the forward end of the reduction gearing or in the shaft line.

►

The oil distribution is integrated in the reduction gearing on the right, in the propeller shaft extension.

Above, the engine drive with coupling.
The top shaft is for driving the shaft generator.
Oil for adjustable-pitch mechanism:
green: supply oil – red: return oil
purple: oil for the coupling
yellow: lubricating oil for bearing lubrication



Drop of the lubricating-oil pressure

When the plugs are removed, the pitch of the propeller can, with the use of a hand pump, be manually set in the required position. The isolated oil system maintains the selected position.

Oil distribution situated in the reduction gearing

For propulsion plants that are equipped with a reduction gearing between the engine and the propeller, the O.D.-ring is fitted to the forward end of the reduction gearing.
An electric-pitch gauge is mounted in the ring.

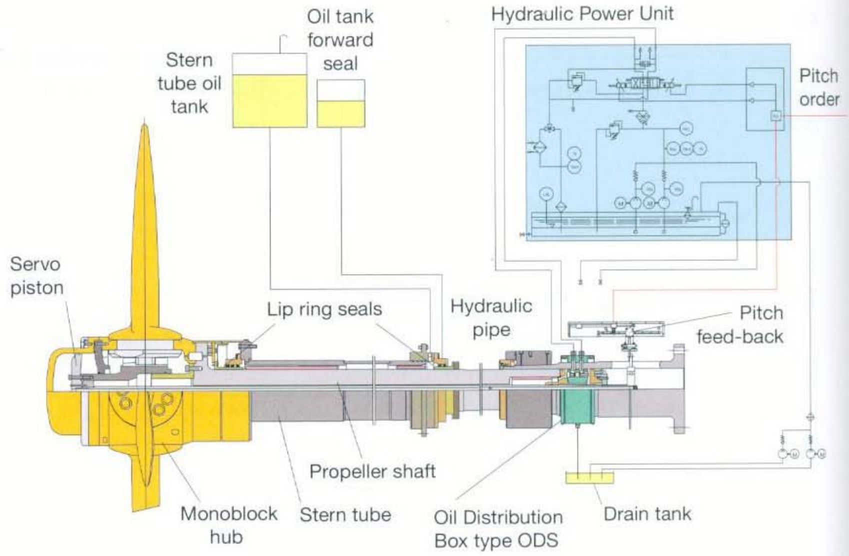
Hydraulic-oil system

The system comprises a hydraulic unit ('Hydra Pack'), a coupling flange with an electrical-pitch feedback box and an oil distribution ring.
In the reduction-geared version this servo-oil system (the servo piston, pitch-feed back box and OD-ring), is integrated in the gearbox. Between the piston and the pitch-control device for the blades located in the hub, an intermediate shaft is installed.
In case of an emergency, the pitch can be adjusted at the gearbox.

▶

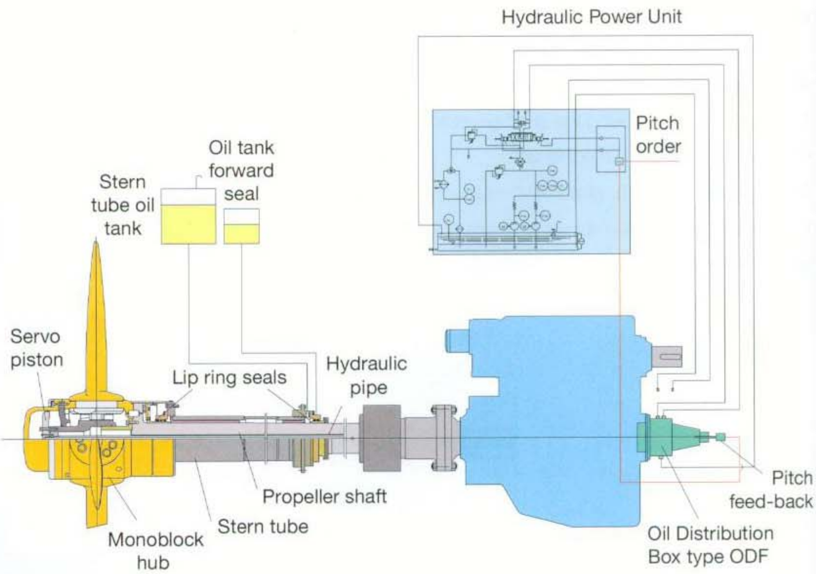
A complete system for a controllable-pitch propeller of MAN–B&W.

The hydraulic power unit controls the hydraulic-oil pressure for pitch adjustment. The tanks ensure that the propeller-shaft seals 'forward' and 'astern' are kept under static pressure.
This arrangement is used in the direct drive of the controllable-pitch propeller by a small low-speed two-stroke crosshead engine.

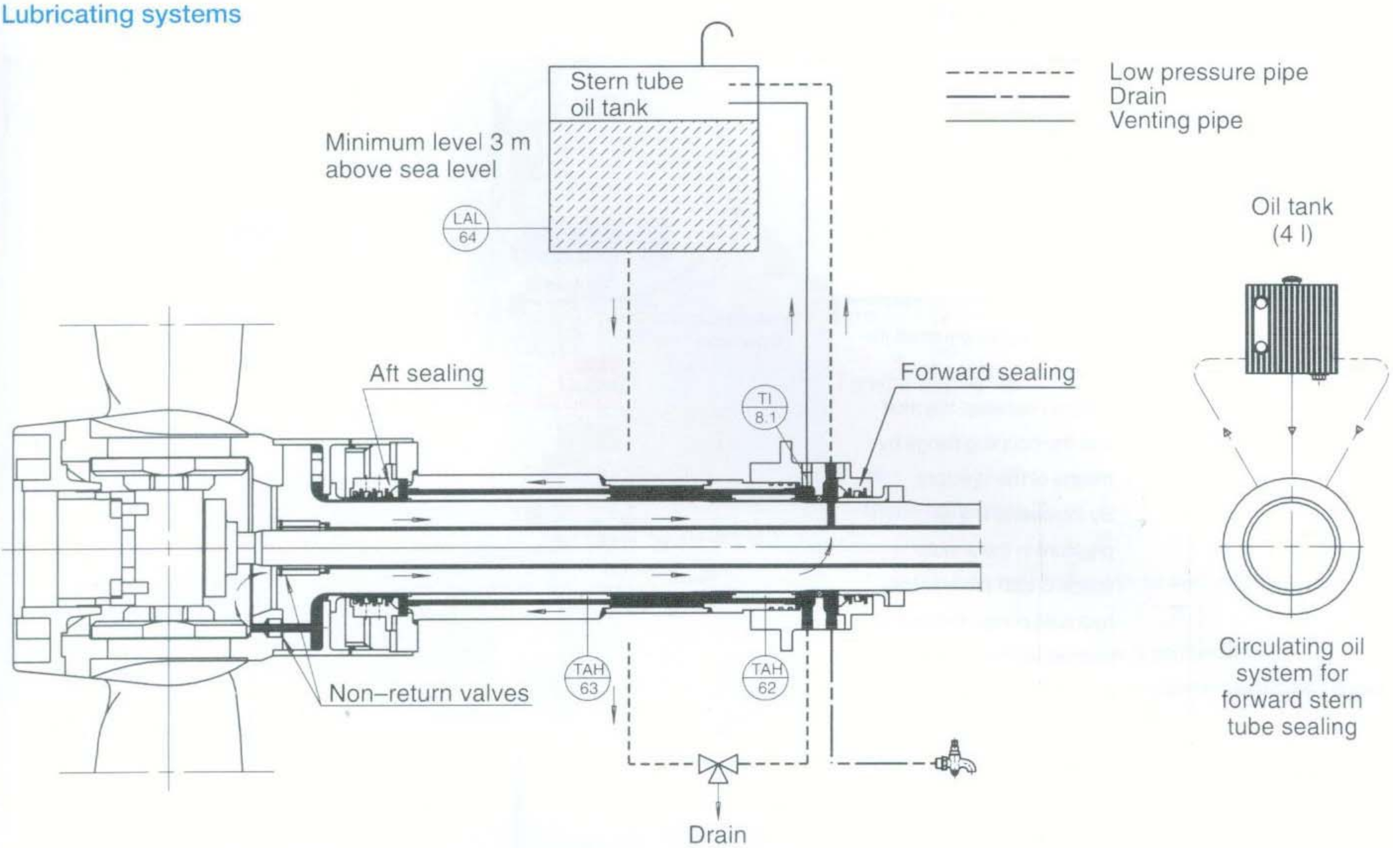


▶

A similar system for a controllable-pitch propeller. However, the oil distribution (green) is placed on the reduction gearing. This is typical of a four-stroke medium-speed diesel engine.



Lubricating systems



▲
The lubricating system.

The stern tube and controllable-pitch propeller have a common lubricating-oil system.
In order to prevent sea water penetration, a lubricating-oil system is kept under static pressure by placing an elevated gravity tank at least 3 metres above sea level, in accordance with the instructions of the manufacturer of the stern-tube seals.
Pitch changes create a pumping effect in the hub, resulting in a lubricating-oil circulation throughout the system, including the elevated oil tank.
Non-return valves in the hub and pitch-control rod ensure that the oil flow to the hub also flows to the stern-tube journal bearings and continues along the chromium steel stern tube to the adjustable-pitch mechanism.
The return oil flows back to the storage tank via the pitch-control rod.
The propeller hub is fitted with two plugs for draining and venting during docking.
The pitch-control rod is lubricated with grease where the intermediate shafts are fitted.

31.6 Propeller shaft and coupling flange

The propeller shaft is manufactured of forged steel and subsequently stress relieved. Material Forged Steel S 45P.
The shaft is mechanically finished, hollow-bored for positioning of the pitch-control rod or piping for the supply and discharge to the servo-piston.
In large propulsion plants with low-speed two-stroke crosshead engines, bearings are fitted at regular intervals along the shaft according to the formula:

$L = 450\sqrt{\text{shaft diameter (mm)}}$

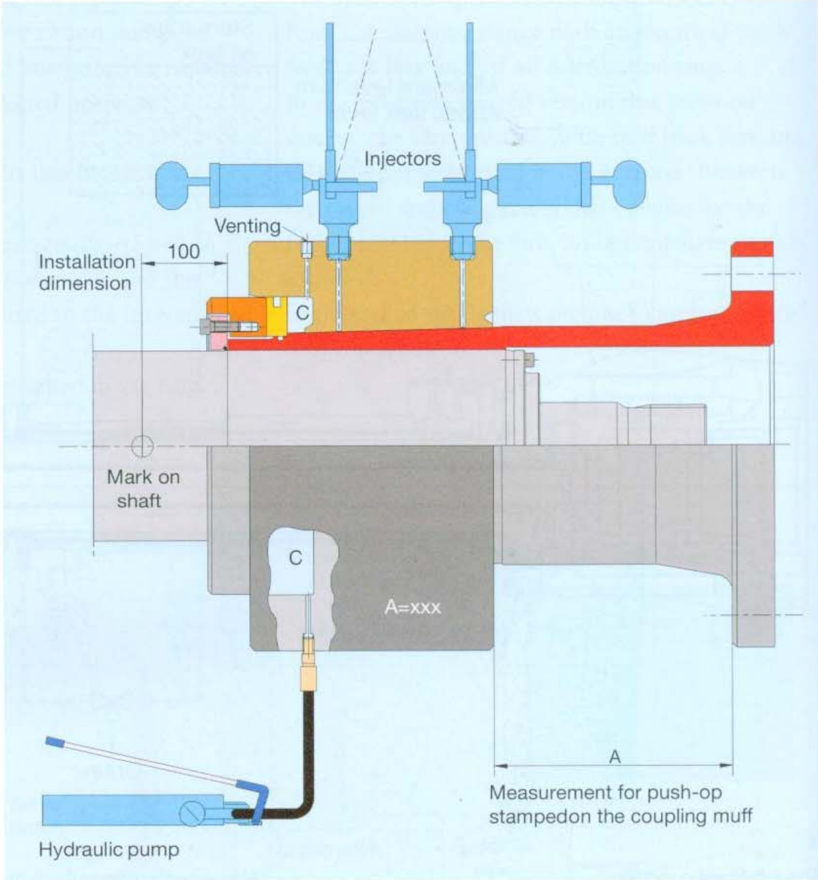
L = maximum bearing distance

RPM < 350

Coupling flange

The hydraulic-coupling flange.

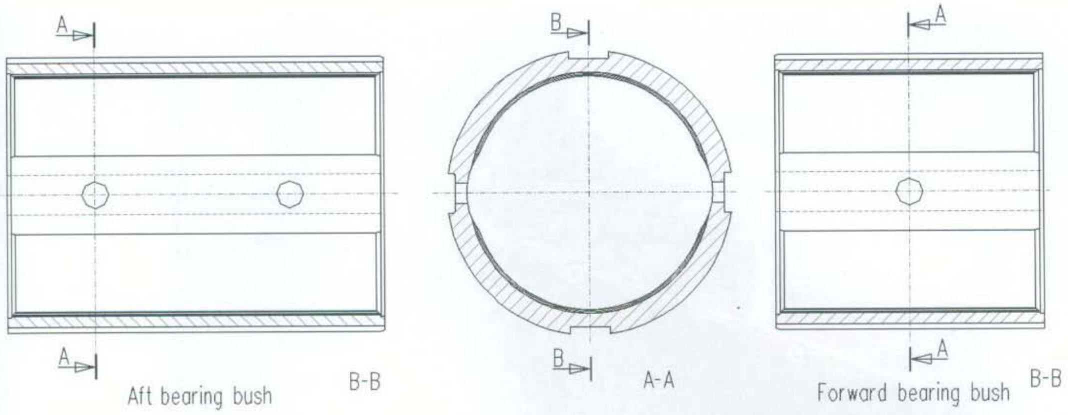
A coupling flange consisting of two parts is clamped to the propeller by oil pressure. A lubricating-oil pressure in excess of 2000 bar is pressed between the muff and the coupling flange by means of the injectors. By increasing the oil pressure in the annular space C with the manual-hydraulic pump, the muff is pushed up the conical section of the shaft in stages. Longitudinal placing of the coupling flange as well as final push-up of the muff is marked on the shaft and the muff.



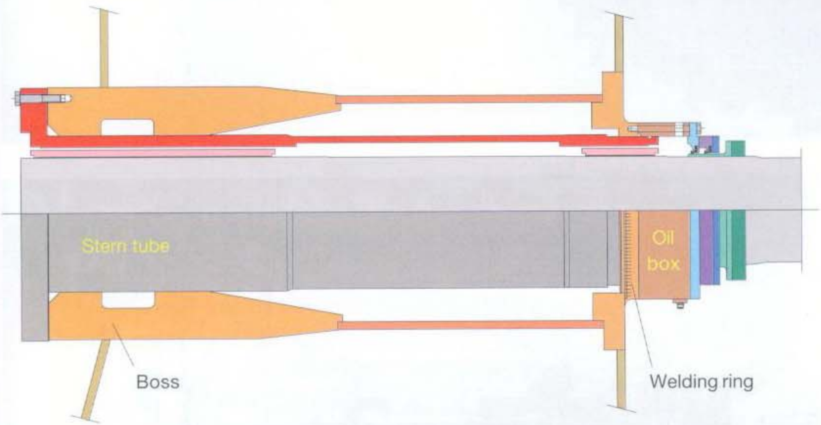
31.7 Stern tube

The standard stern tube.

The standard stern tube is fitted aft and bolted to the propeller-frame boss. The front section of the stern tube has an oil box, bolted to the stern-frame boss. This allows thermal expansion/contraction of the stern tube and facilitates the installation of the stern tube. Close tolerances of the stern tube are no longer required, and the stern tube can have a possible machining tolerance of 5 mm.



An example of a stern-tube bearing of IHC Lagersmit B.V, The Netherlands.



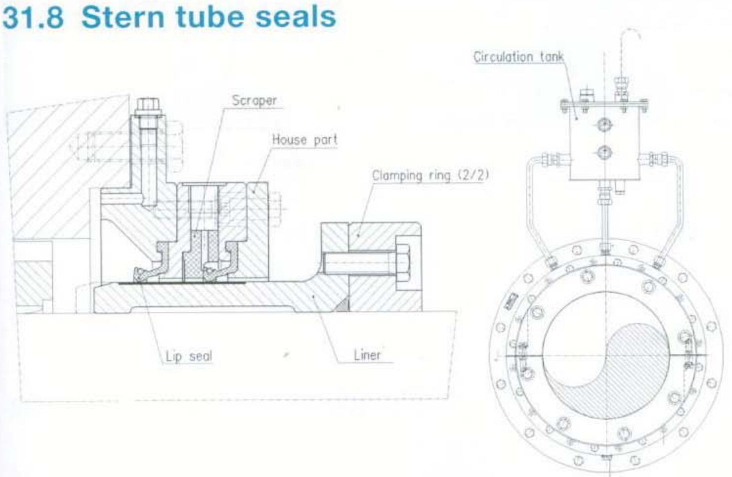
Stern-tube liners.

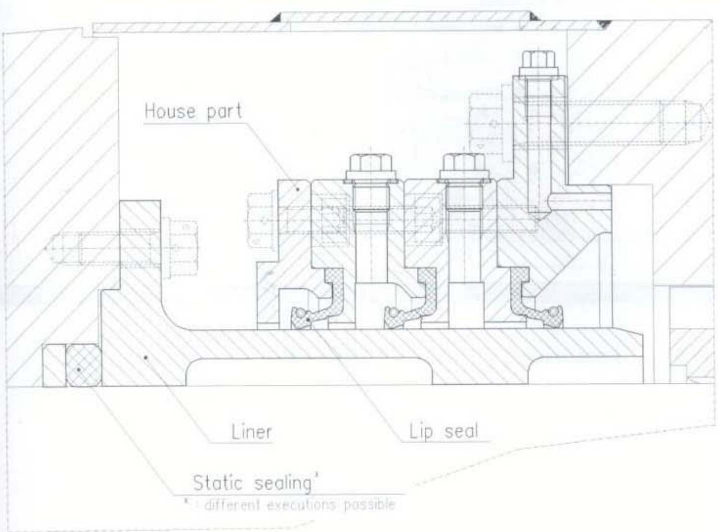
The stern tube is provided with forward and aft white-metal liners (pink).

If required, temperature sensors can be fitted.

The stern tube and the oil space can be treated with epoxy resin.

31.8 Stern tube seals

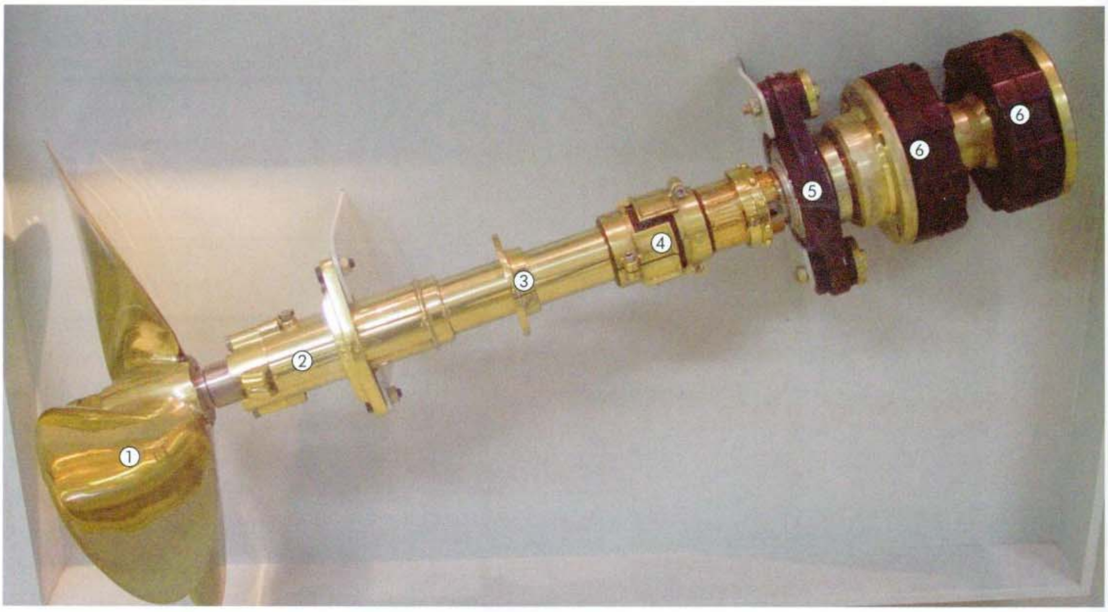




Propeller shafts are provided with forward and aft seals.
Here an example of IHC Lagersmit.

There are numerous types of seals. Their most important property is the prevention of (sea) water and dirt particles from entering the aft seal. Equally important is that leakage of lubricating oil into the (sea) water is kept to a minimum.

Today there are also seals that work without lubricating oil (large plants) or grease (small plants).



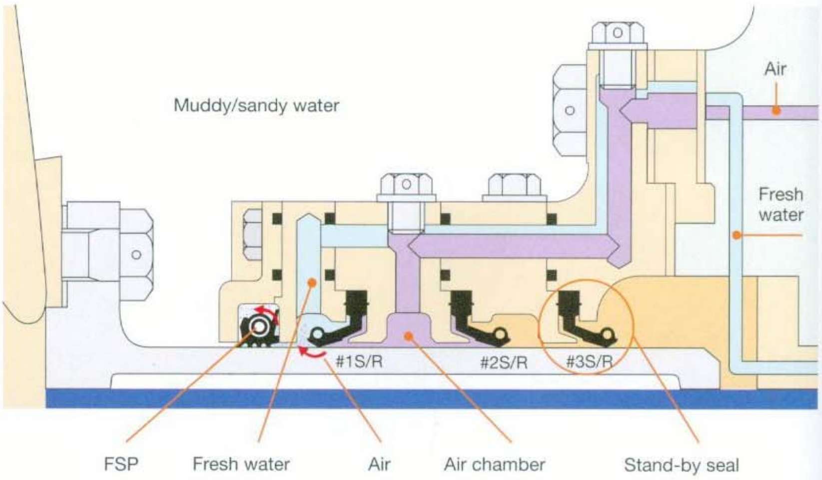
A fixed-pitch propeller with stern tube and seals for small yachts.

1 *fixed-pitch propeller*
2 stern tube
3 *shaft seal*
4 *crankshaft*
5 *thrust block for absorbing the thrust*
6 *flexible coupling towards the engine*

Various propeller systems with corresponding shafts.

31.8.1 Dry seals, pollution-free stern tubes

This system ensures that no lubricating oil from the stern tube can contaminate the environment.

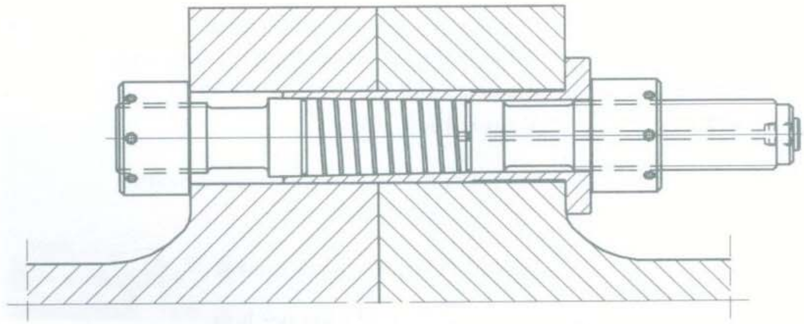


A complete shaft seal where lubricating oil is not used.

Fresh water, light blue, is sent to the far left outer seal and ensures that no sand or mud can enter the seal.
There is an air chamber in the middle of the seal, which is kept under pressure. To the right of the air chamber, another two seals are installed. The seal on the far right is a stand-by seal in case of failure of the 2S/R seal.

31.9 Hydraulic bolts

A complete plant can be fitted with hydraulic bolts, which allows swift, simple and meticulous assembly and disassembly between propeller shaft, the intermediate shafts and the flange of the (two-stroke) engine flywheel.



▲

A hydraulic connecting bolt.

The correct pressure can be applied simply and accurately to the flanges. As the bolts are pre-tensioned by an elastic stretch process, they never 'loosen' of their own accord.

31.10 Material used for controllable-pitch propellers

The I.S.O., the International Standard Organisation, has introduced a series of standards for the manufacturing of propellers (ISO 484). The accuracy class can be selected by the customer.

Propeller blades are manufactured of a nickel-aluminium-bronze (Ni, Al)-alloy or a chromium-nickel-stainless steel (Cr, Ni) -alloy.

Ice class		C		1A*	
Material		NiAl	CrNi	NiAl	CrNi
Thickness r/R = 0.35	mm	132	146	169	187
Thickness r/R = 0.60	mm	71	78	90	100
Thickness r/R = 1.00	mm	0	0	15	13
Blade weight	kg	729	877	952	1053

▲

The graph shows the properties of both materials.

Both materials have high resistance to **cavitation-erosion**. NiAl has better fatigue characteristics in a corrosive environment than CrNi.
Propeller blades are extensively exposed to cyclically varying stresses. Consequently, resistance to fatigue is a decisive factor.
According to classification societies, blades made of CrNi must have a 10% higher thickness compared to a propeller blade made of NiAl.
The thicker the blade, the lower the propeller efficiency! It is usually recommended that propeller blades be manufactured of NiAl.

Class	Manufacturing accuracy
S	Very high accuracy
I	High accuracy
II	Medium accuracy
III	Wide tolerances

▲

The accuracy classes for manufacturing propellers.

If the customer does not specify a class, the propeller is manufactured according to class 1.

31.11 Propeller design

Hydro-dynamic design

Propeller blades are computer designed. The basic data are derived from advanced hydrodynamic theories, practical experience and numerous model tests at various hydrodynamic institutes.
Each blade is specially designed for a certain ship's hull and its operating conditions. The total propeller efficiency, suppressed noise levels and vibrations are prime design objectives.

Propeller efficiency is predominantly determined by the propeller diameter and the corresponding optimum speed. To a lesser extent by the blade surface area, the pitch and the thickness distribution of the propeller blade. These also affect the propeller efficiency. The **blade surface** is selected according to the requirements for minimum cavitation, noise and vibration levels. To keep cavitation to a minimum, the pitch distribution is reduced at the tip of the blade.

Example 1

What is the tip speed of a propeller with a diameter of 8000 mm and an RPM of 82?

$$V\ tip = 2 \times \pi \times R \times n$$

$$= 2 \times 3.14 \times 4 \times \frac{82}{60} = 34.2 \text{ m/sec.}$$

Speed in kilometres per hour =

$$34.2 \times \frac{3600}{1000} = 123 \text{ km per hour.}$$

Example 2: Small yachts

How high is the tip speed of a propeller with a diameter of 300 mm and an RPM of 900?

$$V\ tip = 2 \times \pi \times R \times n$$

$$= 2 \times 3.14 \times 0.15 \times \frac{900}{60} = 14.3 \text{ m/sec.}$$

Speed in kilometres per hour =

$$14.3 \times \frac{3600}{1000} = 50.8 \text{ km per hour.}$$

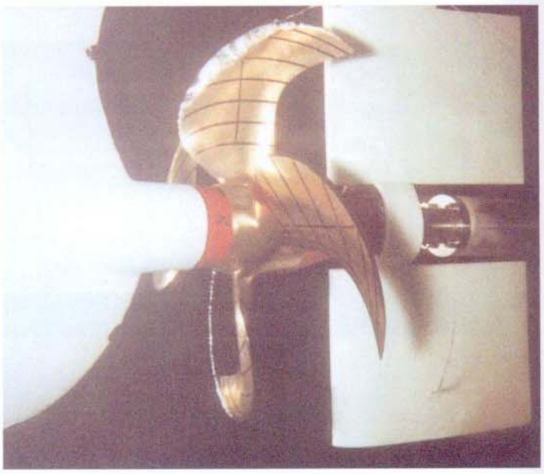
The **thickness distribution** of the propeller blade is chosen according to the requirements of the Classification agencies for 'unskewed' propellers.

31.12 Cavitation

Cavitation is caused by a localized decrease in the saturation pressure near the propeller.
At this reduced pressure below the saturation pressure, the water will boil and lead to a generation of air bubbles. The moment these air bubbles reach other regions near the propeller, the shell of the ship or parts of the rudder with a higher pressure, the bubbles implode. This is accompanied by such an amount of energy release that severe damage may be caused to the surface material close to the imploding bubbles.

Cavitation types

There are three main types of cavitation.



▲

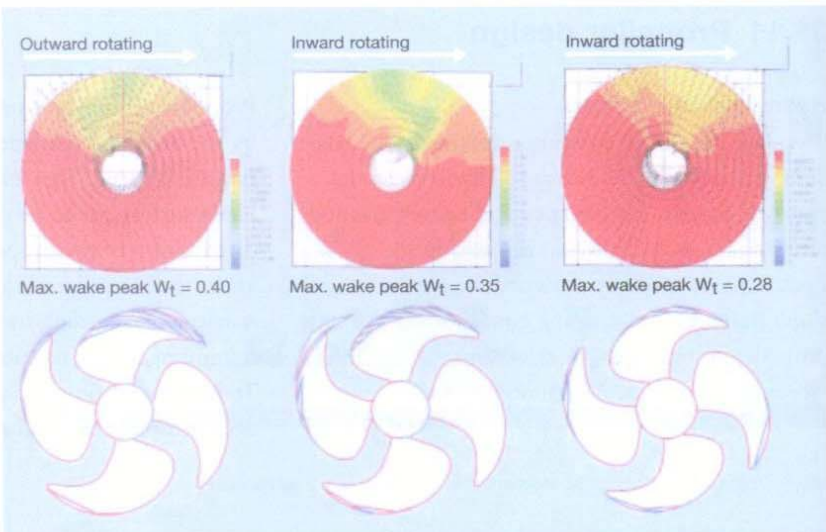
Research into cavitation is especially important in the design of a specific propeller for a specific hull.

Not only can severe damage to the propeller, hull and rudder be prevented but noise generation can also be determined in a fluid laboratory.

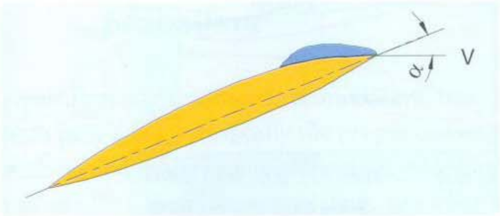
►

When testing propeller designs, much research is done to minimize the extent of cavitation.

The right propeller has a much smaller surface area where cavitation can occur than the examples on the left.
The wake peaks of these two examples are higher. The homogeneous and uninterrupted supply of an adequate amount of water is a prerequisite for minimizing cavitation.



31.12.1 'Sheet' cavitation on the 'suction side' of the blade

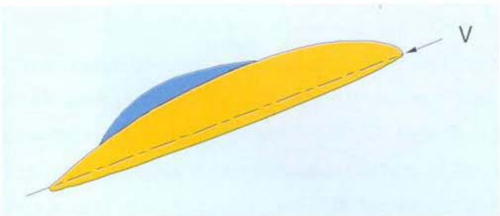


▲
Sheet cavitation on the suction side of the propeller blade.
Sheet cavitation is generated at the leading edge due to a low-pressure peak in this region

The sound and vibration levels will be low if the cavitation is limited and the clearance to the hull is sufficient.
If cavitation extends over half of the radius, it can increase significantly. This may cause erosion of the propeller-blade material and must therefore be prevented.

'Sheet' cavitation in the tip region of the propeller blade may develop into a tip vortex of the propeller blade. When this expands over the heart line of the propeller, it can lead to cavitation erosion of the blade.
If the tip vortex extends to the rudder, it may cause erosion of the rudder material.

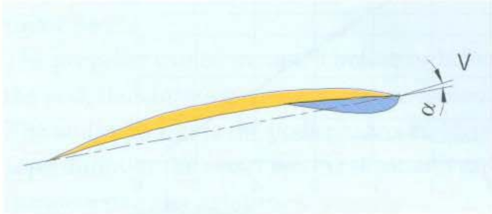
31.12.2 Bubble cavitation



▲
Bubble cavitation.

If the propeller is overloaded and the blade surface area is too small compared to the thrust required, the middle section of the centre of the propeller blade on the suction side will be covered by cavitation. This type of cavitation can be extensive and lead to erosion. When designing a particular propeller, this must be avoided.

31.12.3 'Sheet' cavitation on the pressure side of the blade



▲
Sheet cavitation on the pressure side of the blade.

This type of cavitation is similar to sheet cavitation on the suction side. However, the generated vapour bubbles tend to implode on the blade surface due to the increased pressure. This is likely to produce erosion. Therefore, the blades should be designed without any pressure-side cavitation.

31.12.4 Graphs of fuel consumption and towing force

◄
A graph depicting the required power output in kW (vertical) and the speed in knots (horizontal).

At 12 knots, a power output of 2000 kW is required and at 14 knots 4000 kW is required. Increasing the speed by 2 knots requires twice the engine output and therefore doubles the fuel consumption!

▲
A graph showing fuel consumption in kilograms per hour (vertical) and the ship's speed in knots (horizontal).

Clearly, fuel-consumption increase is significantly higher than the increase in speed. Fuel consumption is 200 kg per hour at a speed of 10 knots. This is doubled to 400 kg per hour at 12 knots and at 16 knots, it is 1200 kg per hour. Therefore, an increase in speed of six knots requires 1200 kg per hour of fuel, not the expected 200 kg of fuel. This amounts to six times the amount of fuel consumed to achieve a speed increase of six knots.

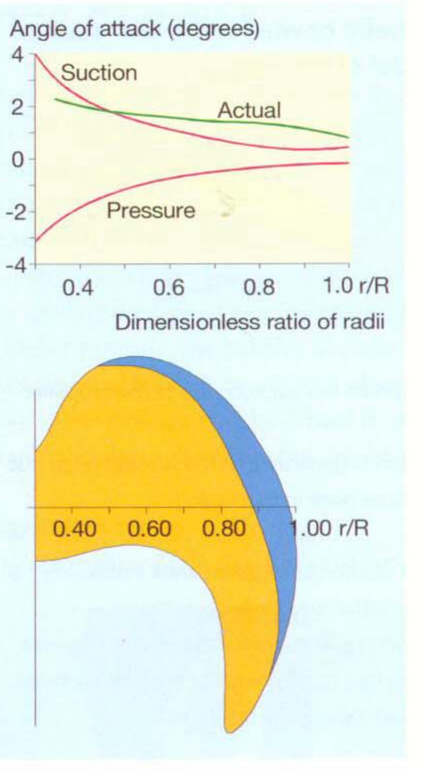
▲
The graph shows: the towing force in kN (vertical) and the speed in knots (horizontal). Increasing the speed effects a significant decrease in towing force.

31.13 Calculating the propeller-blade model

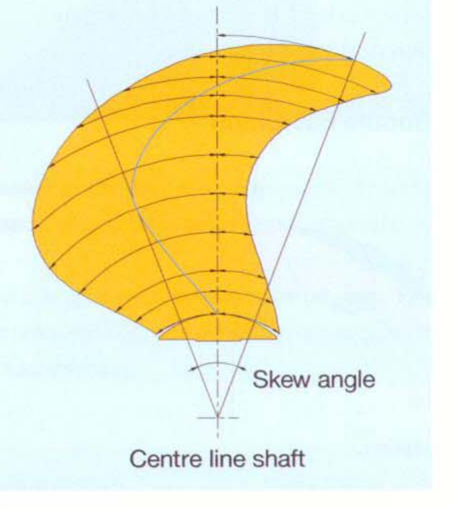
For all conditions and propeller-pitch angles abaft the hull, the flow around the blade is calculated. The extent of cavitation is evaluated with respect to noise and vibration.

31.13.1 'High skew' design of a propeller

A special design is applied to improve cavitation suppression by reducing induced-pressure fluctuations.
The blade is hereby skewed in the rotational direction. The vibrations generated by a 'skewed' propeller blade are reduced to less than 30% of those generated by an 'unskewed' design.



▲
A modern propeller, the 'skew' type. The blue area shows the extent of cavitation.



▲
The 'high-skew' design.

As this type of propeller does not affect propeller efficiency, it is used in ferries and cruise ships requiring low vibration levels.
Today the 'skew'-type propeller is bent in two directions, both rotational and contra-rotational. The blades of this kind of propeller blades can be balanced so that the least amount of force is generated in the adjustable-pitch mechanism.
A more detailed finite-element analysis is carried out for determining the optimum 'skew'.

▼
'High-skew' propeller blades ready for transportation.

31.14 Examples of other propulsion systems with controllable-pitch propellers

Apart from the traditional propulsion plants with propellers, specifically the propulsion engine with behind it, the reduction gearing, coupling, intermediate shafts, propeller shaft, propeller and their various versions, there are many other types that deviate from the standard arrangement of the propulsion engine, shafts and propeller.

31.14.1 Example 1: Diesel-electric propulsion

By driving propellers with electro-motors and generating the required electrical power with generators driven by diesel engines, the location of the propellers and diesel engines may deviate from the traditional arrangement.

Electric thruster

Here the propulsion unit, an electro-motor, is situated on the outside of the ship's hull. This system is known as its brand name 'Azipod'.

▼

Diesel-electric propulsion of Wärtsilä. Shown, an L.N.G.-tanker with Dual Fuel (DF) engines.

The four main gensets twelve-cylinder V-engines type 12 V 50 DF, each provide 11,400 kW and both auxiliary gensets, nine-cylinder in-line engines type 9 L 32 DF, each provide 3150 kW. Therefore, the total installed power is 51,900 kW. This amount of electric power can drive two electro-motors of 12.5 MW each, driving a controllable-pitch Lips propeller. An electrically driven P.O.D. serves as a contra-rotating propeller at normal speeds and as an excellent manoeuvring propeller when entering the LNG terminals.

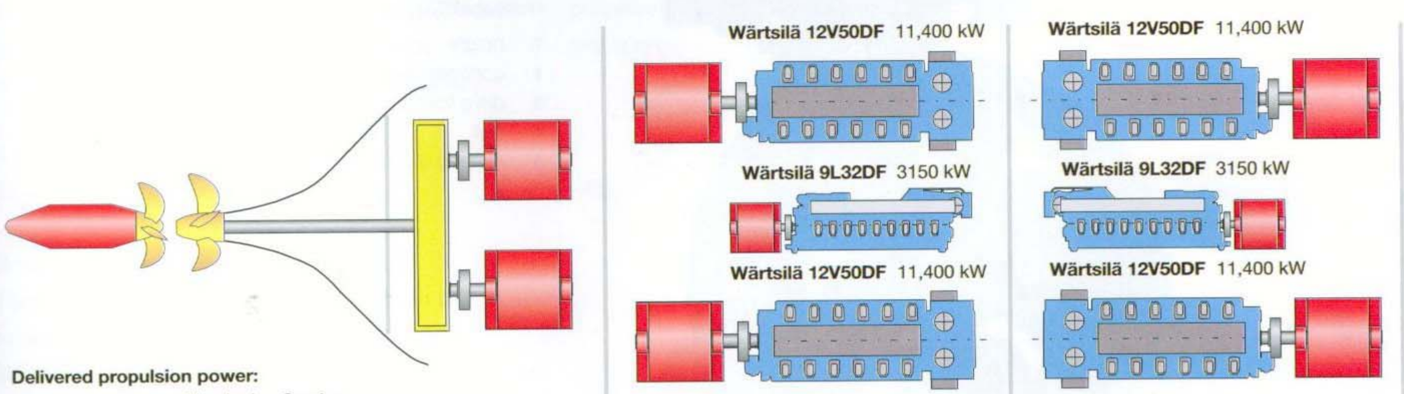
The electro-motor, which has a speed control, is installed in a separate pod below the ship. There are versions that are fixed and those that can rotate 360°.

The propeller can be mounted before or behind the pod, thus forming a tow or thrust version. **The ability to rotate the pods renders rudders superfluous as the vessel has excellent and rapid manoeuvring characteristics.**

Today, the above-mentioned system is frequently utilized in passenger ships, vessels with a dynamic-positioning system, and ferries.

Several advantages

- **Both the propeller system and the diesel gensets can be installed in the most favourable location.** It should be remembered that the 'hotel' section on passenger vessels takes up over 80% of the total space available on the ship.
- **Dependent on the required propulsion output, diesel-driven gensets can be placed off-line.** This way the diesel engines are always heavily loaded and emissions are kept to a minimum while a higher efficiency is achieved.
- In general, larger ships have medium-speed four-stroke engines, suitable for running on H.F.O. This is relatively inexpensive and consequently exploitation costs are reduced.
- **Sometimes, a larger genset driven by a gas turbine (fuel M.D.O.) is installed in the larger propulsion plants. It can be put on-line if the diesel gensets fail or when the hotel load, for instance air conditioning, increases significantly.** A well-known example is the 'Queen Mary II'.
- **Excellent manoeuvrability.** Generally towboats are not necessary and therefore not required by the port authorities.

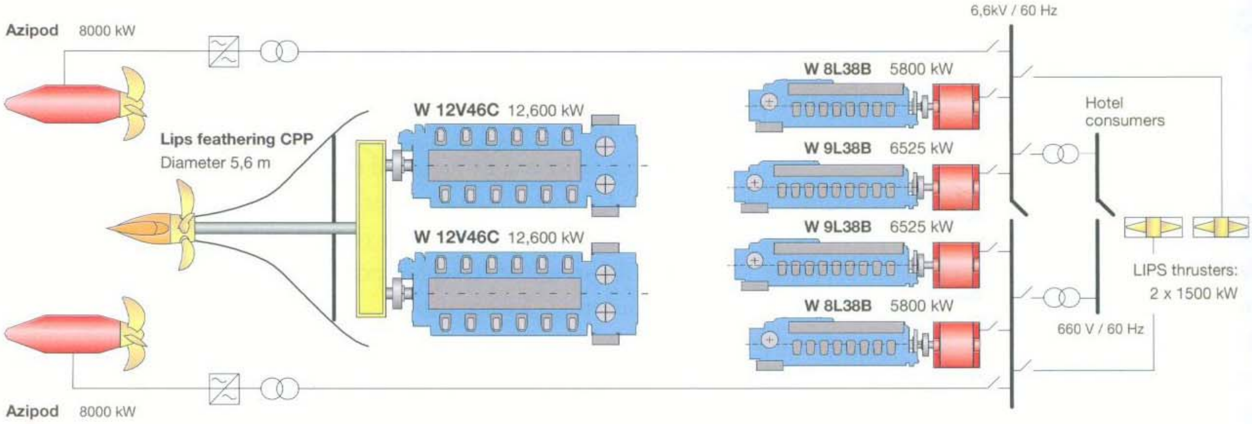


Delivered propulsion power:

	Nominal	Service
Electric pod	17 MW 40 %	17 MW 41 %
LIPS CP propeller	25 MW 65 %	24 MW 62 %
Total shaft power	**42 MW**	**41 MW**

Installed engine power:

Mechanical transmission	0 MW
Electrical power generation	51.9 MW
Total installed power	**51.9 MW**



▲

A different arrangement of Wärtsilä design.

Both main engines type 12 V 46 C, drive one controllable-pitch Lips propeller of 5.6 metres in diameter. Total shaft power is 2 x 12,600 kW.

The four gensets with 8 L 38 B and 9L 38 B engines, total output 24,650 kW, are available for driving both 'Azipods' of 8000 kW each.

The voltage of this mains is 6600 volt at 60 Hz. Both Lips bow thrusters of 1500 kW each are also connected to this mains. The normal ship's grid is supplied with a voltage of 660 at 60Hz via two transformers.

This arrangement allows various means of propulsion:

a the main engines and auxiliary engines and three propellers
b the Azipods
c one main engine driving the controllable-pitch propeller
d part of the auxiliary engines supplying power to the Azipods.

31.14.2 Example 2: Compact thruster

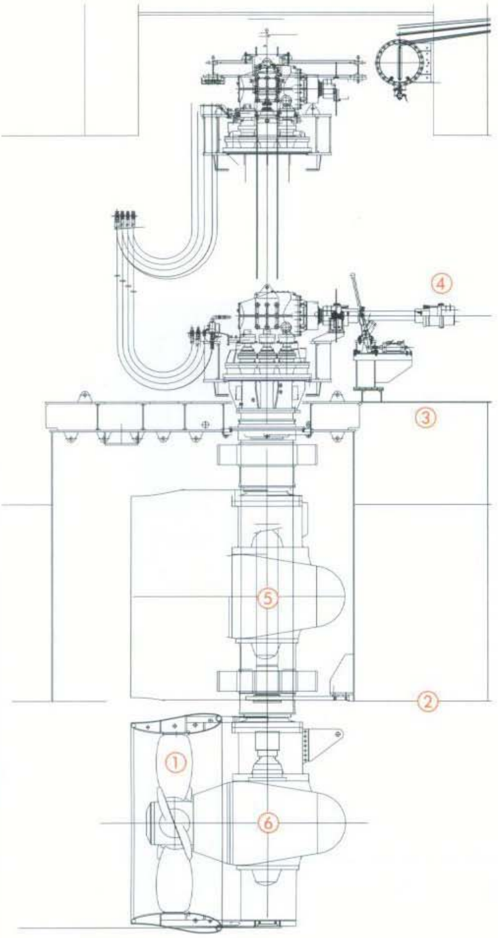
The propeller is driven by diesel engines via crankshafts. The engine output is transmitted to the propeller by two bevel gears. There are many possibilities.

◄

A Lips compact thruster.

This thruster can rotate through 360 degrees and therefore has excellent manoeuvrability. Therefore, a rudder is redundant. The thruster is driven by the diesel engine via long crankshafts.

1 *nozzle*
2 *controllable-pitch propeller*
3 *drive from the diesel engine*
4 *hull*
5 *rotating disc*



▲
A retractable thruster system where the thruster can be completely retracted into the hull.

In sailing position, the thruster is lowered out of the hull.

1 *propeller*
2 *bottom of the ship*
3 *hull of the ship*
4 *engine-driving shaft*
5 *high position inside the hull*
6 *low position outside the hull*

Fixed systems
Here the propeller can be moved vertically, depending on the ship's draught. In an unloaded/empty vessel, the propeller will always be immersed with reasonable propeller efficiency.

A retractable system
These are mainly used at sea.

Bow and stern thrusters
These are arranged in a tunnel located in the forward superstructure or astern to generate lateral thrust. This can be used to manoeuvre or to hold a position.

31.14.3 Example 3: Propellers in nozzles

These are frequently used. These nozzles serve to increase the thrust due to the water flowing around the nozzle section. When the speed of the water flow inside the nozzle exceeds that of the outside flow, a pressure difference is created, which produces horizontal thrust and therefore a propulsive force. Moreover, it often reduces the noise and vibration levels generated by the propeller. The local pressure differences decrease due to the homogeneous flow of the water through the nozzle. Therefore, cavitation is reduced.
A nozzle can be designed for any type of ship or propeller. Nozzles are predominantly used on smaller ships, for instance, inland shipping, tugs, tenders and fishing vessels.

Propulsion types of smaller ships
These diverge considerably from types for larger inland shipping and ocean navigation.
Clients often wish to install the propulsion engine in a different place, or require electric or hydraulic-driven propellers.

31.14.4 Example 4: Electric propulsion

This is used in small vessels such as dinghies.

Advantages
- whisper propulsion
- no emissions when running
- easy installation with a simple reversible electro-motor

▼
There are numerous manners of propulsion for small vessels. Shown, an electric drive of a fixed-pitch propeller. The engine is reversible and can operate at any speed.

Disadvantages

- Restricted turning radius
- Electricity storage in batteries has a low efficiency. The electricity required from the public grid has a low efficiency of approximately 40% with respect to fuel supplied and generated by the electric-power plants. An additional 50% of this is lost with loading and storage. Therefore a mere 40% – 20% = 20% of the original fuel energy remains for driving the propeller.
- Consequently, the total propulsion efficiency from the fuel supplied to the power plant at a propeller efficiency of 40% is at most 8%. A small dinghy with a small diesel engine with an efficiency of 25% does not exceed 10%.

31.14.5 Example 5: Hydraulic propulsion

This is also used in small motor vessels for recreational purposes and for water-based companies. The hydraulic unit consists of a diesel engine, which drives an oil pump.
Here the propeller is driven by a hydraulic engine. Furthermore, various other systems can be driven by this system, such as bow and stern thrusters, an anchor windlass or a light crane.

Advantages.

- The diesel-driven hydraulic set can be placed in the most favourable place in the vessel. The diesel engine operates at a constant RPM and can therefore immediately be loaded via the hydraulic pump.
- The auxiliary power does not require a separate battery.

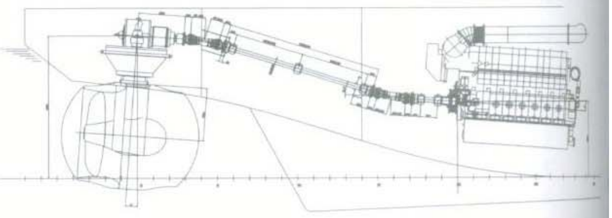
Disadvantages

The purchase price of a hydraulic installation is high and its propulsion efficiency is slightly lower

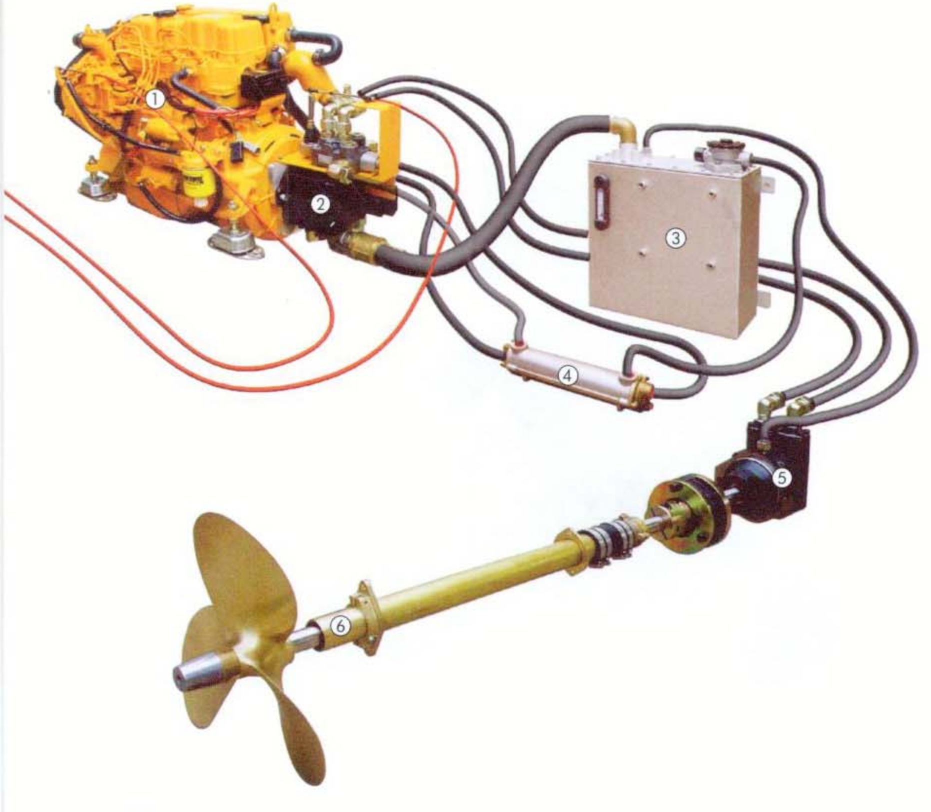
31.14.6 Example 6: Propulsion with a diesel engine and intermediate shaft system to the propeller shaft

Here the engine is not placed in line with the propeller shaft, but, for instance, astern. The propeller is driven via a so-called 'Z drive'.

An advantage is that the diesel engine can be installed in the best location on the boat. This system is frequently used for high-speed motor boats.



▲
Propulsion with a diesel engine and an intermediate-shafting system to the propeller shaft.



◄
A diesel-driven hydraulic propeller drive.

1 *diesel engine*
2 *hydro pump*
3 *storage tank hydraulic oil and buffer tank*
4 *oil cooler*
5 *hydro motor*
6 *propeller shaft with propeller*

The components are connected with flexible hydraulic lines. The diesel engine can be positioned in an ideal place when opting for this system. The hydraulic system can also be used for other purposes, such as an anchor winch.

31.14.7 Example of a large propeller

These propellers are used for huge and fast container ships.
They are fixed-pitch propellers often with six blades. The propeller diameter can increase to up to 9 metres and the weight to up to 100 tons!

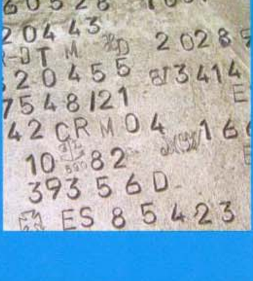
A large fixed-pitch six-bladed propeller.

Note the large amount of free space around the propeller, allowing the free flow of water around the propeller. Nevertheless, cavitation will be caused by the rudder!

Removing this type of propeller requires at least 10 engineers!

CH 32

Regulations for propulsion engines, classification, repair and damage



Each propulsion engine is tested on a test bed according to strict regulations. Shown, a large two-stroke crosshead engine, a Wärtsilä Sulzer RT Flex 96 with a shaft power of 72,360 kW at a speed of 102 RPM.

If a ship is in dry-dock for either maintenance or, in this case, for repairs, regular inspection and overhaul can also be performed.

32.1 Introduction

Many regulations are in force applying to the technical installations on ships. Adherence to these rules and regulations ensures constructive compliance with the law and with the stipulations of insurance companies. The principle is the safe operation of commercial shipping with respect to crew, ship, cargo and environment.
Many countries used to have their own regulations, all more or less aiming at achieving the above goals, but all slightly differing. As shipping is perhaps the most international of the world's industries, there is need for international standards to regulate shipping – which are to adopted and accepted by all.

32.2 The IMO: International Maritime Organization

The IMO (International Maritime Organization) has 167 Member States and 3 Associate Members, representing a worldwide range of shipping interests in 2008.
The International Maritime Organization, based in London, U.K., embodies the coordination of governments and shipping in maritime- technical issues, in order to improve maritime and environmental safety. The IMO is a specialized agency of the United Nations.
The concept of the IMO was developed after the ms Titanic disaster. The ship that 'could not sink' and sank on her maiden voyage. Until then, every country had its own regulations with respect to design, construction and equipping of ships. In 1948 at an international conference in Geneva, the IMCO (**Intergovernmental Maritime Consultative Organisation**) was formally established. In 1982, the name IMCO was changed to IMO, International Maritime Organization.

The first major treaty was the establishment of a new international convention on safety, **SOLAS** (Safety Of Life At Sea). Through the years, SOLAS has been altered and brought up-to-date by the inclusion of modern views, scientific development and new technological possibilities.

Safety was the main objectiver of the IMO, however, after the accident with the Torrey Canyon on 18th March 1967, when 119,000 tons of Kuwait crude were spilled into the sea. This grave disaster led to pollution control being put high on the agenda. In the years following, many procedures were started to prevent reccurrence of major oil spills and to reduce the effects. This resulted in 1973 in the International Convention for the Prevention of Pollution from Ships, which in 1978 was changed in the **MARPOL** protocol. Additionally, conventions were also drawn up to establish liability and compensation when accidents occur.

Safety at sea is largely determined by the ability to communicate. The establishment of the **International Mobile Satellite Organization (IMSO)** in the nineteen-seventies made communication via radio and VHF maritime radio-telephony possible. In 1992, the next step was taken with **GMDSS**, the **Global Maritime Distress and Safety System.** All ships are required to be equipped with satellite emergency-position indicating radio beacons. This ensures that in emergencies, an automatic emergency signal is sent that can be localised by satellite.

IMO, following the 9/11 terrorist attacks in the US in 2001, was highly concerned with terrorism. The fight to prevent and suppress these acts has resulted in the **International Ship and Port Facility Security Code**, where a risk analysis must be performed to determine what security measures are appropriate in harbours in order to prevent not only terrorism, but also human trafficking.

32.2.1 Rules and regulations

Below, a number of mandatory regulations adopted by the IMO.

Maritime security

- 1966, LL, International Convention on Load Lines. The Load Lines Convention contains detailed regulations on the assignment of the freeboard (the vertical distance between the top of the hull and the waterline) and the specific load-line limitations for different types of ships. Also it prescribes the required construction integrity of a ship to enable it to safely carry cargoes.
- 1971, STP, Special Trade Passenger Ships Agreement.
- 1973, Protocol on Space Requirements for Special Trade Passenger Ships.
- 1972, COLREG, International Regulations for Preventing Collisions at Sea. A set of traffic rules for shipping that can be compared with road traffic regulations.
- 1972, CSC, International Convention for Safe Containers. Regulations that set out procedures with regard to the safety of freight

containers transported by sea: size, strength, handling, storing and stacking.
- 1974, SOLAS, International Convention for the Safety of Life at Sea. Regulations which specify minimum standards for the construction, equipment and operation of ships, compatible with their safety for those on board.
- 1979, SAR, International Convention on Maritime Search and Rescue. The system covers search and rescue operations – who does what, where and under what conditions.
- 1976, INMARSAT, Convention on the International Maritime Satellite Organization. The operational requirements for a satellite communication system for maritime purposes.
- 1977, SFV, The Torremolinos International Convention for the Safety of Fishing Vessels.
- 1978, STCW, International Convention on Standards of Training, Certification and Watch keeping for Seafarers. Regulations establishing basic requirements for training, certification and watch keeping for seafarers on an international level.
- 1995, STCW-F, International Convention on Standards of Training, Certification and Watch keeping for Fishing Vessel Personnel. Regulations to improve the training, certification and watch keeping standards for crews on fishing vessels.

Marine pollution

- 1969, International Convention Relating to Intervention on the High Seas in Cases of Oil Pollution Casualties.
- 1972, LDC, Convention on the Prevention of Marine Pollution by Dumping of Wastes and Other Matter.
- 1978, (MARPOL 73/78) International Convention for the Prevention of Pollution from Ships. The prevention of accidental or operational pollution of the marine environment by ships.
- 1990, OPRC, International Convention on Oil Pollution Preparedness, Response and Co-operation. Measures for dealing with pollution incidents, either nationally or in co-operation with other countries.
- 2000, HNS Protocol, Protocol on Preparedness, Response and Co-operation as to Pollution Incidents by Hazardous and Noxious Substances, 2000.
- 2001, AFS, International Convention on the Control of Harmful Anti-fouling Systems on Ships.
- 2004, International Convention for the Control and Management of Ships, Ballast Water and Sediments.

Liability and compensation

- 1969, CLC, International Convention on Civil Liability for Oil Pollution Damage. This regulation was adopted to ensure that adequate compensation is available for persons who suffer oil-pollution damage resulting from maritime casualties involving oil-carrying ships.
- 1971, FUND, International Convention on the Establishment of an International Fund for Compensation for Oil Pollution Damage.
- 1971, Convention relating to Civil Liability in the Field of Maritime Carriage of Nuclear Material.
- 1974, PAL, Athens Convention relating to the Carriage of Passengers and their Luggage by Sea.

◄ A cargo ship docking for overhaul, repairs and periodic inspections by a classification society.

Many precisely defined regulations are in force that oversee the technical installations on ships. The underlying principle is, that merchant vessels sail safely with respect to crew, ship, cargo and environment.
Since most seafaring countries already had a highly varying set of rules and regulations pertaining to shipping and since shipping is indubitably a purely international branch of industry, it was deemed better to have the same international regulations worldwide.

- 1976, LLMC, Convention on Limitation of Liability for Maritime Claims.
- 1996, HNS, International Convention on Liability and Compensation for Damage in Connection with the Carriage of Hazardous and Noxious Substances by Sea.
- 2001, BOPD, International Convention on Civil Liability for Bunker Oil Pollution Damage.

Other

- 1965, FAL, Convention on Facilitation of International Maritime Traffic.
- 1969, International Convention on Tonnage Measurement of Ships.
- 1988, SUA, Convention for the Suppression of Unlawful Acts Against the Safety of Maritime Navigation.
- 1989, International Convention on Salvage.

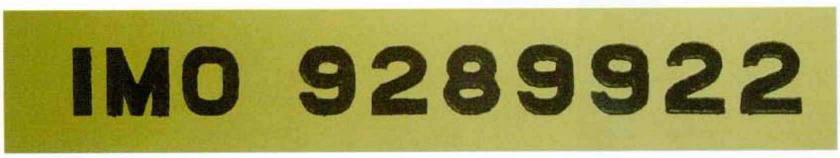
32.2.2 The MSC, The Marine Safety Committee

The legislature of the IMO is the Assembly comprising committees and sub-committees that deal with different issues.
Issues concerning safety are dealt with by the MSC (Maritime Safety Committee).
The MEPC, The Marine Environment Protection Committee, was established to handle environmental issues such as oil pollution of the sea, air pollution by the emission of noxious substances or the dumping of industrial and other waste at sea.
Over the years, two important conventions have been adopted:

- **SOLAS** – Safety Of Life At Sea.
- **Marpol** – Marine Pollution.

For both conventions, many rules have been drawn up. These must be ratified by a sufficient number of member states before the convention is adopted. It then enters into force and is implemented by all member states.

▼
Every sea-going vessel is provided with a clearly visible IMO number in the engine room and on deck.



32.2.3 Considerations for diesel engines

For diesel engines, the fuel used and the emission of noxious substances is of the utmost importance. Naturally, the discharge of oil and oily wastes at sea is bound by very strict regulations.
Seagoing vessels are required to have certain certificates on board that establish that they meet the requirements of the IMO.

The compulsory equipment on board must be approved by the flag states and/or by one of the classification societies.

The following four certificates are in use:

For SOLAS:

1 Cargo Ship Safety Construction Certificate.
2 Cargo Ship Safety Equipment Certificate.
3 Cargo Ship Safety Radio Certificate.
4 Cargo Ship Safety Certificate, this combines 1, 2 and 3.

For Marpol:
This is expressed in The International Oil Pollution Certificate 9I.O.P.C.
This applies to, amongst others, oil tankers and cargo ships above 400 G.T.
In detail, the following parts are covered:

Annex I: normal oil, such as, fuel and lubricating oil. Sludge and bilge water also belong to this annex.
Annex II: noxious liquids and chemicals.
Annex III: packed substances.
Annex IV: sewage.
Annex V: garbage.
Annex VI: air pollution, the exhaust gases of diesel engines.
Annex VII: ballast water.

32.3 Classification societies

They are active worldwide in survey and certification of systems, assets and facilities to improve quality, increase safety and protect environment.
Many classification societies have more than one discipline, of which shipping is an important one.

32.3.1 Shipping

The activities in the field of shipping mainly comprise 'classifying' ships.

Classification societies draw up standards for the quality and the reliability with respect to design, construction and operation of ships.
Besides carrying out classification surveys and statutory inspections, compulsory under international shipping conventions and regulations, a classification society gives ship-owners and operators more insight in the risks that they face and helps them to improve business performance.

32.3.2 Classification

A classification society establishes the regulations regarding the construction of a certain type of ship.
The society approves the drawings, gives advice for possible changes and modifications and verifies that the construction of the ship and all the technical installations are conform its standards.
After completion of the ship, the ship-owner will receive a **Certificate of Classification, for the hull and the machinery.**
A **Certificate** is also issued that is valid for 5 years. Every year, the certificate must be signed when the annual survey has been satisfactorily completed.
In order to provide this service, classification societies maintain a worldwide network of surveyors.
A Class certificate allows the vessel to be insured.

The society surveys the technical condition of the ship, the safety and the living conditions on board.
Flag states allow these activities to be carried out by the classification societies.
Ships carry not only class certificates but also comply with statutory regulations.
Both are issued by the societies.

32.3.3 Acceptance by a flag state

A company can have a sea-going vessel registered in a particular country, the flag state. The flag state accepts the ship as a member of its fleet.
The authorities of a country allow the shipping company to sail under their jurisdiction at a certain (financial) rate.
These rates vary considerably per country/ flag state.
The flag state's name must be visibly displayed on the stern of the ship.
The certificate of registration, the international tonnage certificate, is issued by the flag state or by the classification company on behalf of the flag state.
The official details of the ship are listed in the tonnage certificate (also important for, amongst others, harbour dues).

32.4 Periodic inspections of the diesel engine and its parts

The condition of the engine parts must be regularly checked. Classification societies have strict regulations regarding the inspection periods and activities.
In view of the brief docking times of modern ships these 'surveys' are predominantly performed during dry-docking and other periods when the ship is out of service.
Therefore, timely contact between the shipping company/proprietor and the classification society is of utmost importance.

◄ On the stern of an approved vessel, the name of the vessel and the port of registry must be displayed. It is then clear to see under whose flag the vessel is sailing.

During dry-docking, such as shown here, many activities can be carried out to the propulsion plant, without any extra delay and therefore extra expenditure.

32.4.1 Regulations for reconditioned parts

Stamped code in a cylinder liner.

Class Germanischer Lloyd. Pressure 7 bar. A repair performed in 1995. Before 1995, the logo of G.L.

Worn parts are often reconditioned.

Also, see Chapter 26, Reconditioning engines and their parts.

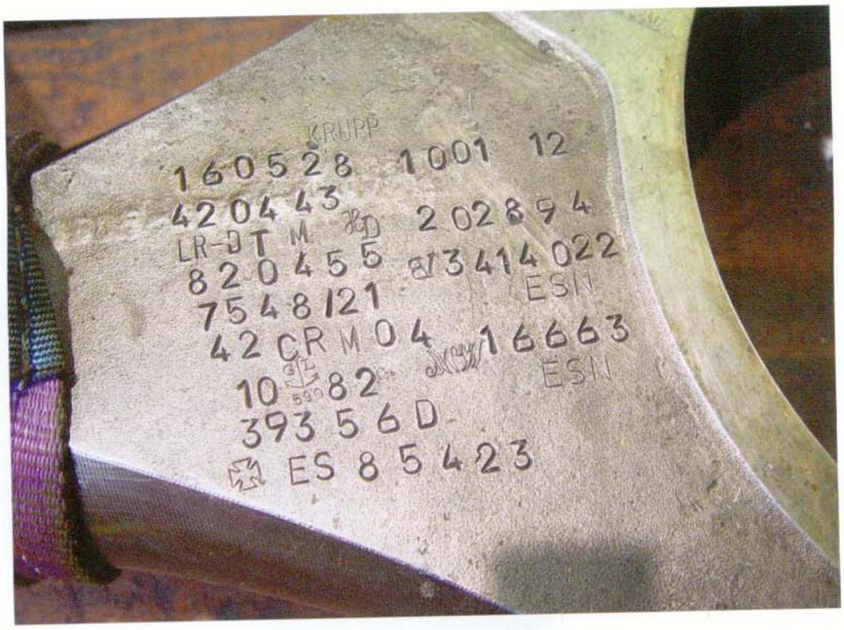
After repair, the part must be re-tested. This should be 'made available' to the classification society.
The reconditioning company makes an appointment for inspection with the surveyor of the society.
Often one (new constructions) or several (reconditioned) registration numbers are marked on the part by one or several societies.
Those who have access to all the numbers and marks can easily verify which parts have been reconditioned and/or tested!

Testing the cooling-water spaces of newly installed cylinders. Left, the foundry code. This code displays all the manufacturing data of the engine block.

▲

Stamped codes on a crankshaft web.

After a number of repairs, the web gets 'full'.



▲

Stamped codes on the connecting-rod big end.

The material composition is also displayed. 42 CR M0 4.

Vessels are constructed in accordance with the stipulations of classification societies.
The classification society approves drawings, details and material types. Every detail is clearly registered.
During construction of the vessel, the classification society monitors the activities and ensures that the vessel is constructed in accordance with the approved drawings.
When construction is complete, all the systems are checked and tested.
The final check of the vessel is a test at sea, a sea trial.
The entire construction of the vessel, all the tests and ultimately the trials are performed in accordance with a specific class.
The classification society supplies the vessel with a certificate – the Certificate of Class for the vessel and machinery.
A work certificate, valid for five years, is also issued. This certificate must be re-approved every year after an annual inspection.
The classification societies are all members of the **IACS, the International Association of Classification Societies.**
This association has its headquarters in London and is the umbrella organisation of the ten largest societies.

For a complete classification, the classification society checks the vessel from the drawing table, the beginning of construction, the skeleton, the installations and the engines.
The classification society subsequently follows the entire life cycle of a vessel.
All vessels must be surveyed in order to comply with the requirements of seaworthiness.
The flag states are accountable. They usually outsource the actual testing to classification societies.

32.4.2 Classification societies

The following societies are active worldwide:
ABS – American Bureau of Shipping
BV – Bureau Veritas
CCS – China Classification Society
DNV – Det Norske Veritas
GL – Germanischer Lloyd
KR – Korean Register of Shipping
LR – Lloyd's Register
NKK – Nippon Kayi Kyokai
RINA – Registro Italiano Navale
RS – Russian Maritime Register of Shipping

The regulations of, for instance **Lloyd's Register, comprise seven parts:**

Part 1: Rules/Regulations.
General regulations, the actual classification, periodic inspection.

Part 2: Regulations for the manufacture, for the testing and the classification of materials.
These tests are performed at foundries, ironworks and part manufacturers.

Part 3: Ship's structure.
Basis structure of the skeleton, longitudinal strength, bow and stern constructions.

Part 4: Ship's structure (ship types).
Skeleton-construction requirements for the carcass per ship type, such as towing boats, ferries, oil tankers and container ships.

Part 5: Main and auxiliary engines.
Including propeller-shaft alignment and vibrations, piping for oil tankers and steering gear.
This part is especially important for us; it provides a detailed description of the manner in which the propulsion engine must be constructed.

Part 6: Control, electrical systems, cooling systems and fire prevention.

Part 7: Remaining vessel types and plants.

32.5 Examples: Germanischer Lloyd

The following examples are a small part of the regulations.
It is not our intention to give a complete overview. The societies have comprehensive regulations. 'Rules for Classification and Construction Ship Technology/Seagoing Ships/Machinery Installations'.
Important is the acquisition of overview of the content of these regulations. This overview is not complete either!

▶
The general regulations for the lay-out, selection and arrangement of machinery with respect to the maximum angle of inclination.
Including the water temperature, the air conditions and other ambient conditions in spaces, enclosed spaces and on deck.

Table 1.1 Inclinations

Installations, components	Angle of inclination [°][2]			
	Athwartship		Fore-and aft	
	static	dynamic	static	dynamic
Main and auxiliary machinery	15	22,5	5[4]	7,5
Ship's safety equipment e.g. emergency power installations, emergency fire pumps and their drives	22,5[3]	22,5[3]	10	10
Switchgear, electrical and electronic appliances[1] and remote-control systems				

1 Up to and angle of inclination of 45° no undesired switching operations or functional changes may occur.
2 athwartships and fore- and aft-inclinations may occur simultaneously.
3 On ships for the carriage of liquefied gases and chemicals the emergency power supply must also remain operational with the ship flooded to a final athwartships inclination up to a maximum of 30°.
4 Where the length of the ship exceeds 100 m, the fore-and-aft static angle of inclination may be taken as 500/L degrees.

Table 1.2 Water temperature

Coolant	Temperature [°C]
Seawater	+ 32[1]
Charge air coolant inlet to charge air cooler	+ 32[1]

1 GL may approve lower water temperatures for ships operating only in special geographical areas.

Table 1.3 Air temperature
at atmospheric pressure = 1000 mbar
and relative humidity = 60 %

Installations, components	Location, arrangement	Temperature range [°C]
Machinery and electrical installations[1]	in enclosed spaces	0 to 45[2]
	on machinery components, boilers	According to specific local conditions
	in spaces subject to higher or lower temperatures	
	on the open deck	- 25 to + 45

1 Electronic appliances shall be designed and tested to ensure trouble free operation even at a constant air temperature of + 55 °C.
2 The Society may approve lower air temperatures for ships designed only for service in particular geographical areas.

Table 1.4 Other ambient conditions

Location	Condition
In all spaces	Ability to withstand oil vapour and salt-laden air
	Trouble-free operation within the temperature ranges stated in Table 1.3, and with a relative humidity up to 100 % at a reference temperature of 45 °C
	Tolerance to condensation is assumed
In specially protected control rooms	80 % relative humidity at a reference temperature of 45 °C
On the open deck	Ability to withstand temporary flooding with seawater and salt-laden spray

32.5.1 General regulations and instructions

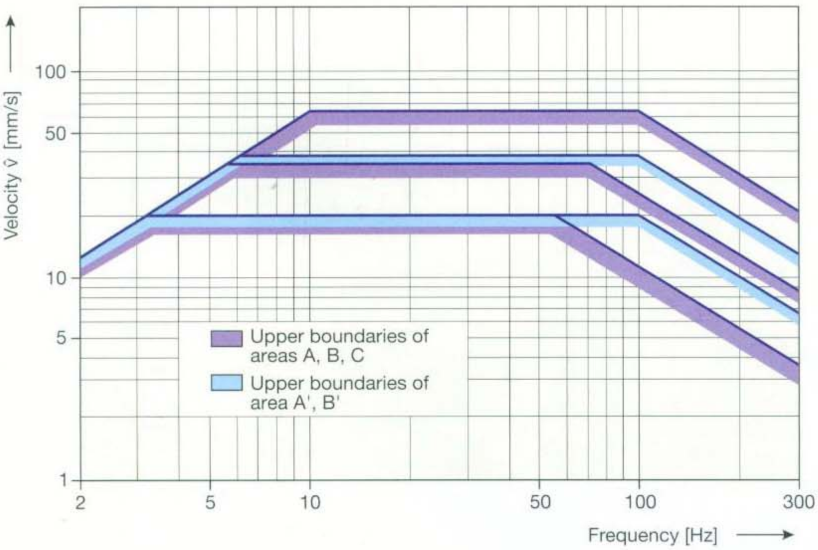
General Rules and Instructions.
These comprise general subjects, related to, amongst others:

- The propulsion plants and the auxiliary devices required to warrant the ship's safety during operation.
- The required drawings with all the necessary information.
- The atmospheric conditions such as air temperature, humidity and air pressure.
- Allowable vibrations.
- The design of the installation, in particular with regard to the stresses and loads, material types, accessibility of the engine room, operation of the propulsion plant and manoeuvring.
- Fuels used. The flash point, for instance, may not drop below 60 °C and for auxiliary engines must be equal to or higher than 43 °C.
- Additional regulations for measuring instruments, such as pressure gauges, lighting, bilge spaces/pipes, ventilation, enclosure of rotating parts, alarms and communication.
- All the equipment that is part of the propulsion plant, such as diesel generators, pumps, starting-air systems, turbo-blowers, ventilation.

32.5.2 Internal combustion engines and starting-air compressors

Internal Combustion Engines and Air Compressors.
Today, there are individual regulations for dual-fuel engines for, i.a., L.N.G. tankers. These are separate from the diesel-engine regulations.

For 'dual fuel' engines, such as on this modern LNG tanker, special regulations apply.



The maximum vibrations permissible in certain spaces.

Table 1.5 Numerical definition of the area boundaries shown in figure above

Areas	A	B	C	A'	B'
ŝ [mm]	< 1	< 1	< 1	< 1	< 1
v̂ [mm/s]	< 20	< 35	< 63	< 20	< 40
V_{eff} [mm/s]	< 14	< 25	< 45	< 14	< 28
â [9,81 m/s²]	< 0,7	< 1,6	< 4	< 1,3	< 2,6

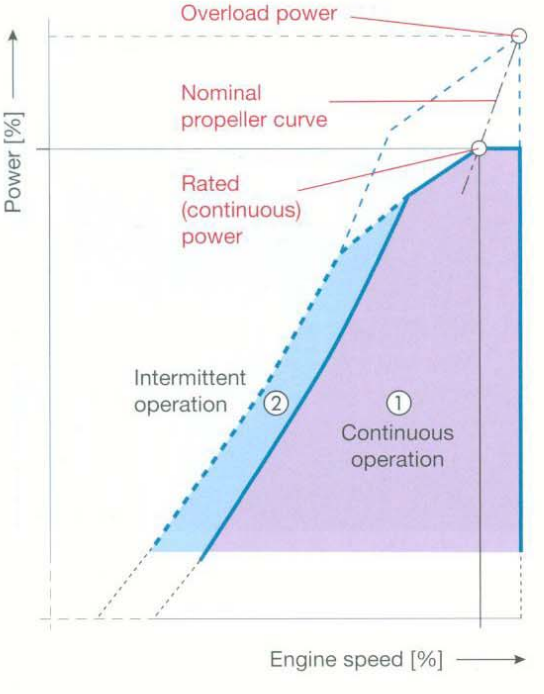
32.5.3 Important regulations

To calculate the power output of the engine the following atmospheric conditions are applied:

Air pressure	1000 millibar
Inlet-air temperature	45 °C
Relative humidity of the air	60%
Seawater temperature	32 °C

32.5.4 Standard nominal power

The use of a graph is required.



A power diagram of a diesel engine driving a propeller.

1 Continuous operation.
2 Intermittent operation.

It should be possible to exceed the standard nominal power with 10% for as long as one hour.

- After testing on the test bed, an engine overload should not be possible. Therefore, the fuel supply must be restricted to 100% standard nominal power.
- This rule does not apply for diesel generators.
- However, the rule does apply for dual-fuel engines.
- Accessibility. This is an important topic for engineers who work on engines.
- All the activities, such as the changing of components, must be able to be performed without any problems.
- Electric components and measuring instruments, which operate and control the engine, must meet certain requirements. This applies, for instance, to computer systems.
- If an electrical/electronic system fails, the propulsion engine should not stop or undergo a drastic change in the power output. This also applies to speed regulators such as governors.

According to the regulations, all engine parts must be accessible for inspection, repair and maintenance.

In practice, this is sometimes very difficult to realise. Often, the costs of the alterations are too high. Shown here, two engineers of Wärtsilä removing a cover from the 'hot box'. The upper side of this Wärtsilä 46 is easily accessible. Note, the IMO number on the engine-room bulkhead.

32.5.5 Documents of approval

The engine manufacturer must provide the classification society with drawings and data for each engine, such as those listed in the table below.

Manufacturers operating under licence are also required to present these documents; these must include a statement by the licenser when modifications have taken place.

Table 2.1 Documents for approval

◄ A table of all parts and drawings that must be submitted for approval.

Ser. No.	A/R	Description	Quantity	Remarks (see below)
1	R	Detail required on GL forms F 144 and F 144/1 when applying for approval of an internal combustion engines	3	
2	R	Engine tansverse cross-section	3	
3	R	Engine longitudinal section	3	
4		Bedplate and crankcase		
	R	– cast	1	
	A	– welded, with welding details and instructions	3	9
5	R	Thrust bearing assembly	3	3
6		Thrust bearing bedplate		
	R	– cast	1	
	A	– welded, with welding details and instructions	3	9
7		Frame/framebox		
	R	– cast	1	1
	A	– welded, with welding details and instructions	3	1, 9
8	R	Tie rod	1	
9	R	Cylinder cover/head, assembly	1	
10	R	Cylinder liner	1	
11	A	Crankshaft for each number of cylinder, with data sheets for calculation of crankshafts	3	
12	A	Crankshaft assembly, for each number of cylinders	3	
13	A	Thrust shaft of intermediate shaft (if integral with engines)	3	
14	A	Shaft coupling bolts	3	
15	R	Counterweights including fastening bolts	3	
16	R	Connecting rod, details	3	
17	R	Connection rod, assembly	3	
18	R	Crosshead assembly	3	2
19	R	Piston rod assembly	3	2
20	R	Piston assembly	1	
21	R	Camshaft drive, assembly	1	
22	A	Material specifications of main parts with information on non-destructive material tests and pressure tests	3	8
23	R	Arrangement of foundation (for main engines only)	3	
24	A	Schematic layout or other equivalent documents of starting air system	3	6
25	A	Schematic layout or other equivalent documents of fuel oil system	3	6
26	A	Schematic layout or other equivalent documents of lubricating oil system	3	6
27	A	Schematic layout or other equivalent documents of cooling water system	3	6
28	A	Schematic diagram of engine control and safety system	3	6
29	A	Schematic diagram of electronic components and systems	1	
30	R	Shieldin and insulation of exhaust pipes, assembly	1	
31	A	Shieldin of high-pressure fuel pipes, assembly	3	4
32	A	Arrangement of crankcase explosion relief valves	3	5
33	R	Operation and service manuals	1	7
34	A	Schematic layout or other equivalent documents of hydraulic system (for valve lift) on the engine	3	
35	A	Type test progam and type test report	1	
36	A	High pressure parts for fuel oil injection system	3	10

1 only for one cylinder.
2 only necessary if sufficient details are not shown on the transverse cross section and longitudinal section.
3 if integral with engine and not integrated in the bedplate.
4 for all engines.
5 only for engines with a bore > 200 mm, or a crankcase volume ≥ 0,6 m^3
6 and the systems, where this is supplied by the engine manufacturer. If engines incorporate electronic control systems a failure mode and effect analysis (FMEA) is to be submitted to demonstrate that failure of an electronic control system will not result in the loss of essential services for the operation of the engine and that operation of the engines will not be lost or degraded beyond an acceptable performance criteria of the engine.
7 operation and service manuals are to contain maintenance requirements (servicing and repair) including details of any special tools and gauges that are to be used with their fitting/settings together with any test requirements on completion of maintenance.
8 for comparison with GL requirements for material. NDT and pressure testing as applicable.
9 The weld procedure specification is to include details of pre and post weld heat treatment, welding consumables and fit-up conditions.
10 The documentation has to contain specifications of pressures, pipe dimensions and materials.
A for approval
R for reference

32.5.6 Definition of the diesel-engine type

The type specification of an internal-combustion engine is defined by the following data:
- Type number of the manufacturer.
- Cylinder diameter.
- Stroke.
- Fuel-injection method.
- Fuel used.
- Operating principle: four stroke or two-stroke.
- Scavenging system: natural aspiration or supercharging.
- Nominal cylinder power at nominal speed and the maximum mean effective pressure.
- Method of supercharging: pulsating pressure system or constant-pressure system.
- Charging-air cooling system: type of intercooler.
- Cylinder arrangement: in-line or V.

For added engine components such as turbo-blowers, heat exchangers, engine-driven pumps and other machinery, the manufacturer must provide the classification society with the required particulars.

32.5.7 Calculations of the crankshaft

The most important and yet vulnerable part of the engine is the crankshaft.
The load on crankshafts is high and damaged crankshafts can result in an engine total loss. Repairing crankshafts is costly, as they usually have to be removed from the engine room. This is in itself expensive. Consider disassembly, assembly and the time the vessel is non-operational!

There are many considerations.

- Shrink joints of built-up crankshafts of two-stroke crosshead engines are to be designed in accordance with certain regulations.
- The bolts used to connect power-end flange couplings must be the prescribed fitted bolts.

32.6 Materials for diesel engines

The mechanical properties of materials used for engine parts must meet strict requirements.

The following individually designed parts must be submitted for material tests:
- The crankshaft.
- The crankshaft-coupling flange for main power transmission.
- The crankshaft-coupling bolts.

▶
Very strict regulations are applicable for crankshafts due to their high loads.

- Pistons or piston crowns made of steel, cast steel or nodular cast iron.
- Piston rods.
- Connecting rods including bearing caps.
- Crossheads.
- Cylinder liners made of steel or cast iron.
- Cylinder covers made of steel or cast iron.
- Welded crankshaft bedplate and the bearing transverse girders made of forged or cast steel.
- Welded A-frames and crankcases.
- Welded entablatures.
- Tie rods.
- Bolts and studs for cylinder covers, crossheads, main bearings and connecting-rod bearings.
- Camshaft-drive gear wheels and chain wheels of, amongst others, some two-stroke crosshead engines, made of steel or cast iron.
- Dependent on the cylinder diameter and therefore the engine size, certain parts are submitted for material testing.
- Above a cylinder diameter of 400 mm, **all** the parts mentioned above must be submitted for material testing.

Table 2.2 Material tests

Cylinder bore	Parts to be tested*
≤ 300 mm	1 - 6 - 10 - 11 - 12 - 13
> 300 ≤ 400 mm	1 - 6 - 8 - 9 - 10 - 11 - 12 - 13 - 14
> 400 mm	all parts

* numbered according to the list under D.2.1 above

▲
The number of parts that must be subjected to material testing is dependent on the cylinder diameter of the engine. Above 400 millimetres, all parts must be submitted for testing.

- In addition, material tests are to be carried out on pipes and parts of the starting-air system and other pressure systems.
- Materials for air coolers must be delivered with test reports from the manufacturer.
- In the case of individually manufactured engines, non-destructive material tests are to be performed.
- Especially the drive gearing and other components subjected to heavy loads, such as cylinder covers, are subjected to special testing.

32.7 Tests and trials

32.7.1 Inspection during engine construction

If the engine manufacturer is approved as 'Supplier of mass-produced engines' by a classification society, special regulations apply for this manufacturer.

▼
A table of the approved material types and associated certificates.

Table 2.3 Approved materials and type of test certificate

Approved materials	Society's Rules*	Components	Test certificate†		
			A	B	C
Forged steel $R_m \geq 360$ N/mm²	Section 3, C.	Crankshafts	X	–	–
		Connecting rods	X	–	–
		Pistons rods	X^3	X^4	–
		Crossheads	X^3	X^4	–
		Pistons and piston crowns	X^3	X^4	–
		Cylinder covers/heads	X	–	–
		Camshaft drive wheels	X^3	X^4	–
Rolled or forged steel rounds $R_m \geq 360$ N/mm²	Section 3, C.	Tie rods	X	–	–
		Bolts and studs	X^1	X^2	–
Special grade cast steel $R_m \geq 440$ N/mm²	Section 4, C.	Throws and webs of build-up crankshafts	X	–	–
Special grade forged steel $R_m \geq 440$ N/mm²	Section 3, C.				
Cast steel	Section 4, C.	Bearing transverse girders (weldable)	X	–	–
		Pistons and piston crowns	X^3	X^4	–
		Cylinder covers/heads	X^1	X^2	–
		Camshaft drive wheels	X^3	X^4	–
Nodular cast iron, preferably ferritic grades $R_m \geq 370$ N/mm²	Section 5, B.	Engine blocks	–	X^1	–
		Bedplates	–	X^1	–
		Cylinder blocks	–	X^1	–
		Pistons and piston crowns	X^3	X^4	–
		Cylinder covers/heads	–	X^1	–
		Flywheels	–	X^1	–
		Valve bodies	–	X^1	–
Lamellar cast iron $R_m \geq 200$ N/mm²	Section 5, C.	Engine blocks	–	–	X
		Bedplates	–	–	X
		Cylinder blocks	–	–	X
		Cylinder liners	–	–	X
		Cylinder covers/heads	–	–	X
		Flywheels	–	–	X
Shipbuilding steel, all GL grades for plate thickness ≤ 35 mm	Section 1. B.	welded bedplates	X	–	–
Shipbuilding steel, GL grade B for plate thickness > 35 mm		welded frames	X	–	–
		Welded housings	X	–	–
Structural steel, unalloyed, for welded assemblies	Section 1, F.				

* All details refer to the GL Rules II – Materials and Welding, Part I – Metallic Materials, Chapter 2 - Steel and Iron Materials

† Test certificates are to be issued in accordance with GL Rules II – Materials and Welding, Part I – Metallic Materials, Chapter 1 – Principles and Test Procedures – Section 1, H. with the following abbreviations: A: GL Material Certificate. B: Manufacturer Inspection Certificate. C: Manufacturer Test Report

1 only for cylinder bores > 300 mm

2 for cylinder bores ≤ 300 mm

3 only for cylinder bores > 400 mm

4 for cylinder bores ≤ 400 mm

Table 2.4 Magnetic particle tests

Cylinder bore	Parts to be tested*
≤ 400 mm	1 - 2 - 3 - 4 - 5
> 400 mm	all parts
* numbered according to the list under D.2.2 above	

Table 2.5 Ultrasonic tests

Cylinder bore	Parts to be tested*
≤ 400 mm	1 - 2 - 3 - 4 - 7 - 10
> 400 mm	1 - 2 - 3 - 4 - 5 - 6 - 7 - 10 - 11
* numbered according to the list under D.2.2 above	

▲
The compulsory magneflux and ultrasonic tests are applicable for a number of engine parts at certain cylinder diameters.

▶
A number of parts must also be pressure-tested.

Table 2.6 Pressure tests[1]

Component		Test pressure, p_p [bar][2]
Cylinder cover, cooling water space[3]		7
Cylinder liner, over whole length of cooling water space[5]		7
Cylinder jacket, cooling water space		4, at least $1{,}5 \cdot p_{e,zul}$
Exhaust valve, cooling water space		4, at least $1{,}5 \cdot p_{e,zul}$
Piston, cooling water space (after assembly with piston rod, if applicable)		7
Fuel injection system	pump body, pressure side	$1{,}5 \cdot p_{e,zul}$ or $p_{e,zul} + 300$ (whichever is less)
	Valves	$1{,}5 \cdot p_{e,zul}$ or $p_{e,zul} + 300$ (whichever is less)
	Pipes	$1{,}5 \cdot p_{e,zul}$ or $p_{e,zul} + 300$ (whichever is less)
Hydraulic system	High pressure piping for hydraulic drive of exhaust gas valves	$1{,}5 \cdot p_{e,zul}$
Exhaust gas turbocharger, cooling water space		4, at least $1{,}5 \cdot p_{e,zul}$
Exhaust gas line, cooling water space		4, at least $1{,}5 \cdot p_{e,zul}$
Coolers, both sides[4]		4, at least $1{,}5 \cdot p_{e,zul}$
Engine-driven pumps (oil, water, fuel and bilge pumps)		4, at least $1{,}5 \cdot p_{e,zul}$
Starting and control air system		$1{,}5 \cdot p_{e,zul}$ before installation

1 In general, items are to be tested by hydraulic pressure as indicated in the Table. Where design or testing features may require modification of these test requirements, special consideration will be given.
2 $P_{e,zul}$ [bar] = maximum working pressure in the part concerned.
3 For forged steel cylinder covers test methods other than pressure testing may be accepted e.g. suitable non-destructive examination and dimensional control exactly recorded.
4 Charge air coolers need only be tested on the water side.
5 For centrifugally cast cylinder liners, the pressure test can be replaced by a crack test.

32.7.2 Pressure tests

Certain parts of the engine are subjected to pressure testing.

32.7.3 Type-approval testing

Diesel engines used on board ships, must be type tested in the presence of a classification society's surveyor.

32.7.4 Conditions for testing of certain engine types

The engine must be tested in accordance with the specific engine properties.
The inspections and measurements performed by the engine manufacturer for reliable continuous operation are submitted to the classification society.

32.7.5 Scope of type-approval testing

The type-approval test comprises three parts:

A Internal test.
Tests under operating conditions where the operating data of the engine, such as pressures, temperatures, power output and speed are collected.

B Type test.
The tests take place in the presence of a surveyor of the classification society.

C Part inspection.
After the test run, certain parts of the engine must be submitted to the classification society for inspection. This entails the disassembly of these parts.

32.7.6 Load during the test

In normal cases, the engine load is 25%, 50%, 75%, 100% and 110% of the maximum rated power.

a. This is performed along the nominal (theoretical) propeller curve and/or at constant speed in propulsion engines.
b. At rated speed with a constant governor setting for continuous operation for a genset.

For engines with supercharged air by the use of a turbo-blower, the achievable power output in case of turbo-blower damage is indicated as follows:

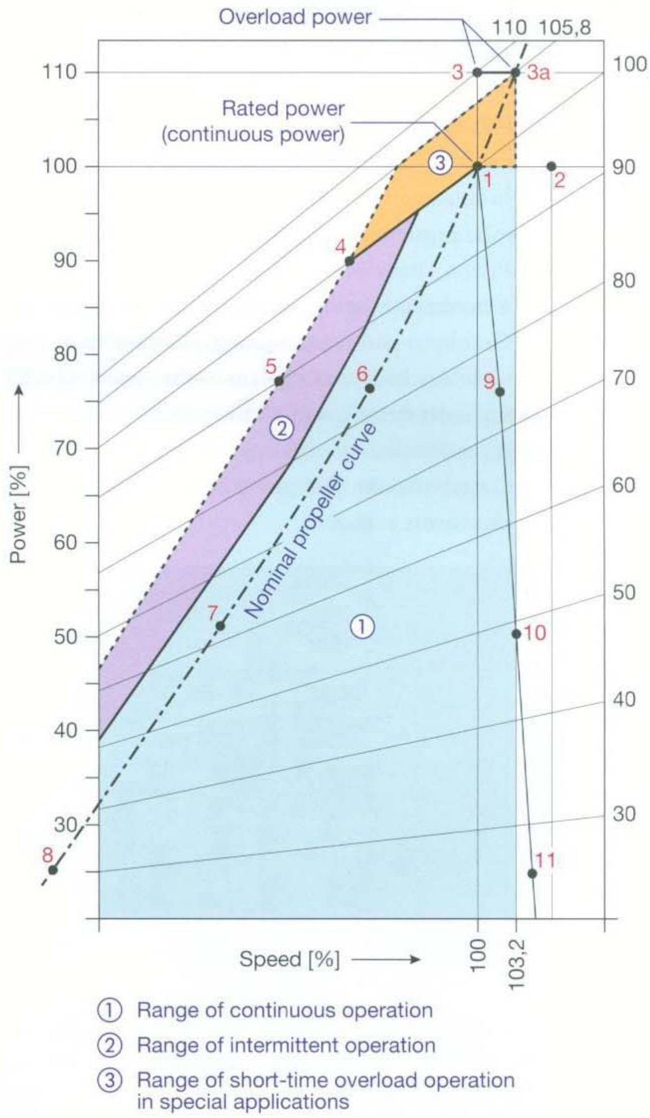
- Engines with one turbo-blower, if the rotor is blocked or removed.
- Engines with two of more turbo-blowers, if the damaged turbo-blower is turned off.

B test: Type test

This test is carried out in the presence of a surveyor of the classification society. Deviations from this program require the agreement of the classification society.

▶

This two-stroke crosshead engine is ready to be installed in the ship (docked in the background). It has been subjected to intensive testing in accordance with the stipulations on a test bed in the engine factory.



▶

An engine-load diagram.

Horizontal: the speed
Vertical left: the power output
Vertical right: the torque

- The data to be measured and recorded when testing the engine at various load points must include all the parameters necessary for an assessment.
- The operating time per load point depends on the engine size and on the time for collection of the operating values. The measurements shall in every case only be performed after achievement of a steady-state condition.
- Normally, an operating time of a half hour per load point is kept.
- At 100 % output (rated power), an operating time of 2 hours is required. At least two sets of readings are to be taken with an interval of 1 hour in each case.

The nominal maximum continuous power, 100% power.
Rated power/continuous power.

The operating point of 100 % output at maximum allowable speed corresponding to load point 2 has to be performed.

Maximum permissible torque.
The maximum permissible torque normally results at 110 % output at 100 % speed corresponding to load point.

The minimum permissible speed for intermittent operation has to be adjusted:
- at 100 % torque corresponding to load point 4.
- at 90 % torque corresponding to load point 5.

Partial load.
For partial-load operations of 75%, 50% and 25% of the rated power at speeds conform to the nominal propeller curve at load points 6, 7 and 8 is measured. Proceeding from the nominal speed at a constant governor setting, the operating points have to be adjusted corresponding to load points 9, 10 and 11.

Emergency operation for turbo-blowers

The maximum achievable power can be attained:

- At a speed conforming to normal propeller curve.
- With a constant governor setting for rated speed.

Functional tests

These must be performed as follows:

- Ascertaining the lowest speed according to the nominal propeller curve.
- Starting tests for non-reversible and/or reversible engines.
- Governor tests.
- Test of the safety system, especially for overspeed and failure of the lubricating-oil pressure.

C test: Component inspection

Immediately after testing, the parts of one cylinder for in-line engines and two cylinders for V-engines must be presented for inspection as follows:

- Piston, dismantled.
- Crosshead bearing, dismantled.
- Crankpin and main bearing, dismantled.
- Cylinder liner may remain in the engine block.
- Cylinder head/cover, all the valves disassembled.
- Camshaft, camshaft drive and crankcase, with the covers removed.
- If the classification society deems it necessary, further dismantling of the engine may be required for the inspection of other parts.

After this, the classification society draws up a test report and if this is favourable, a **'Type Approval Certificate'** is issued.

32.8 The tests of mass-produced engines

For engines with a cylinder bore smaller or equal to 300 mm, numerous conditions apply.

Some important points:
If the power of a certain type of engine increases more than 10%, a new test program is required. This occurs frequently in practice. When introducing new types of engines, the power is usually increased.

32.9 Shipboard trials

After the conclusion of the testing program prescribed by the engine manufacturer, the vessels undergo further trials in open water, both stationary and sailing.
The final trials are known as 'the sea trials'.

▶ After construction of the vessel has been completed at the fitting-out yard, the final procedure before the vessel is delivered to her new owner takes place on open water: the sea trials.

32.10 Some important points

32.10.1 Main-engine propulsion with a fixed propeller

The tests must be executed as follows:

a. At maximum permissible speed during at least four hours and an engine speed that is common for normal cruising power for at least two hours.
b. At a speed of 103% of the maximum allowable speed for 30 minutes when the engine adjustments permit.
c. Determining the minimum on-load speed to determine when the engine stalls.
d. Starting/stopping and astern manoeuvring.
e. Reverse direction of the propeller rotation at a speed of at least 70% for 10 minutes.
f. Testing of the monitoring and safety systems.

32.10.2 Main-engine propulsion systems with an controllable-pitch propeller or reversing gearing

Controllable-pitch propellers must be tested at various propeller pitches.
In combiner systems of speed and a certain pitch, a combiner curve must be plotted and verified by measurements.

32.10.3 Main engines that drive generators, such as diesel-electric propulsion

The tests are to be performed at nominal maximum speed with a constant governor setting under the following conditions:

a. 100% nominal maximum power during at least four hours and at normal continuous cruising power for at least two hours.
b. At 110% of maximum power of 30 minutes' duration.
c. In reverse direction of propeller rotation during sea trials at a minimum speed of at least 70% of the maximum speed for 10 minutes.
d. Starting/stopping procedures.
e. Testing the monitoring and safety systems.

▲ Diesel engines for generators also undergo extensive testing.

32.10.4 Diesel-engine driven auxiliaries and electric generators

These engines must be subjected to an operational test for at least four hours.
During the entire test, the set must operate at maximum permissible power.
The test must establish that the set is capable of operating at 110% of its normal maximum power, and for diesel gensets, the amount of time required for the actuation of the overload protection system.

32.10.5 Fuel type

If the main and auxiliary engines can run on heavy oil or another type of fuel oil, this must be demonstrated by a test.
The program for the sea trial can be extended for special operating conditions, such as towing and fishing.

32.10.6 Earthing

It is necessary to check the limits. These are specified by the engine manufacturers, for the potential difference between the crankshaft and propeller shaft with the hull must not be exceeded during operation.
Appropriate earthing devices must ensure that this cannot occur. Furthermore, the potential difference must be measured and give an alarm if exceeding the limits.

A slip-ring construction on the propeller shaft prevents too large a potential difference between crankshaft, propeller shaft and hull. This prevents damage to the engine, especially the crankshaft.

32.10.7 Safety devices

Here many regulations are also applicable, for instance:

32.10.8 Speed control and protection against overspeed

Some comments:

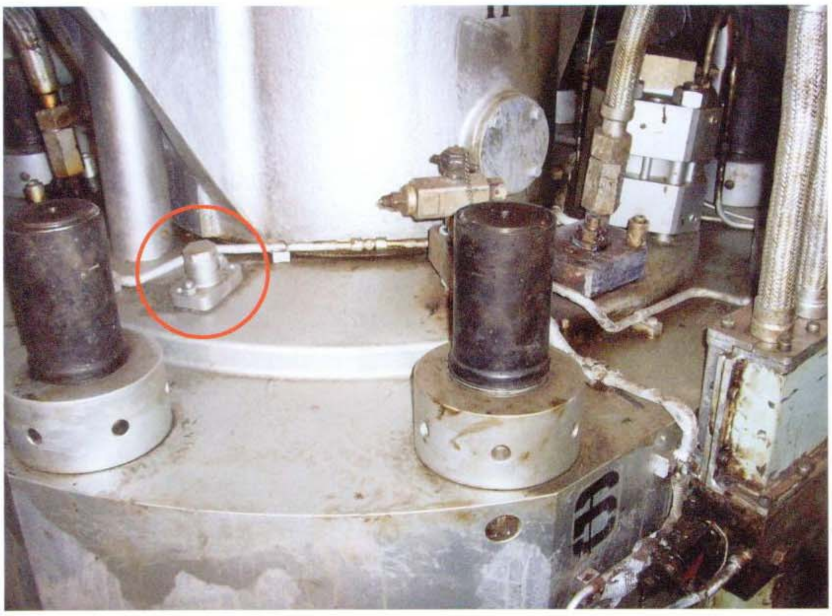
- Each diesel engine that is not used to drive an electric generator must be protected with a governor, which is set so that the engine speed cannot exceed the normal maximum speed by more than 15%.
- Each main engine with a shaft power of 220 kW or more, which can be declutched in operation or drives a controllable-pitch propeller, must be equipped with an independent overspeed protection. This must ensure that the normal maximum speed cannot be exceeded by more than 20%.

32.10.9 Engines driving gensets

Here strict regulations apply, also in view of the damage that can be caused to electrical-power consumers.

Each engine or emergency-diesel power unit must be equipped with a governor, which ensures that the frequency fluctuations in the electrical network do not exceed plus or minus 10% of the normal frequency, with a recovery time to the steady-state frequency conditions of a maximum of five seconds, when the maximum electrical load is switched on or off.

Strict requirements also apply for governors.



A safety valve on the cylinder cover of a two-stroke crosshead engine. This opens at 40% above the normal maximum combustion pressure.

Especially with a temporary shutdown, serious engine damage can occur.
A leaking injector can produce a very high peak pressure.
Cooling-water leaks can cause a great deal of damage to the piston and connecting-rod assembly of the cylinder head.
This is known as water hammer.

If a certain load equal to the maximum load is switched off, a speed deviation of 10% of the nominal maximum speed is allowed. It should be noted that the speed safety by + 15% or more, is activated.
Apart from the normal governor, each diesel engine with a power of 220 kW of more must be equipped with an **overspeed protection device**, which operates independently of the normal governor.
The protection device must prevent normal maximum speed exceeding 15%.
There are many regulations for maintaining a constant frequency and voltage of the electrical network.
When increasing the load in two stages, this is allowed from 0 to 50% and subsequently from 50 to 100%.
Here, the frequency is not allowed to deviate too much from the normal frequency, with a maximum of 10% and not for more than five seconds.
Gensets arranged in 'stand-by' systems, must operate on full load within 30 seconds after the start, also when the engine is cold, and therefore without heated cooling water.
For emergency-diesel gensets, this is 45 seconds.

32.11 Regulations for propulsion engines

To ensure that the speed of the engine is controllable in all circumstances, the following requirements apply:

a. The governor system has an independent 'back up' system.
b. There is an extra governor system that can be manually controlled with a separate, protected power supply.
c. The engine has a hand-operated fuel control, suitable for manoeuvring.

32.11.1 Safety system against excessive pressure in the combustion space

All the cylinders of engines with a bore larger than 230 mm must be fitted with a safety system that is actuated when the pressure exceeds 40% of the normal maximum combustion pressure.

32.11.2 Crankcase airing

This is not allowed.

32.11.3 Crankcase venting

The openings should be as small as possible.
In systems for extracting lubricating-oil vapours or, for instance, monitoring the oil-vapour concentration in crankcase-mist detectors, the under pressure in the crankcase may not exceed 2.5 millibar.
The vent pipes of two or more engines may not be combined.
In two-stroke engines, the oil vapour from the crankcase should not be transported to the scavenging-air receiver or the air-inlet receiver of the engine.

32.11.4 Safety valves on crankcases

Each type must be provided with a certificate. Safety valves for the crankcase must be fitted on each engine with a cylinder bore larger than 200 mm or a crankcase volume equal to or larger than 0.6 m^3.

All the other separate spaces in the crankcase, such as the gearing or chain casings for driving the camshaft or similar drives, must be fitted with their own safety valves if the volume of these spaces exceeds 0.6 m^3.

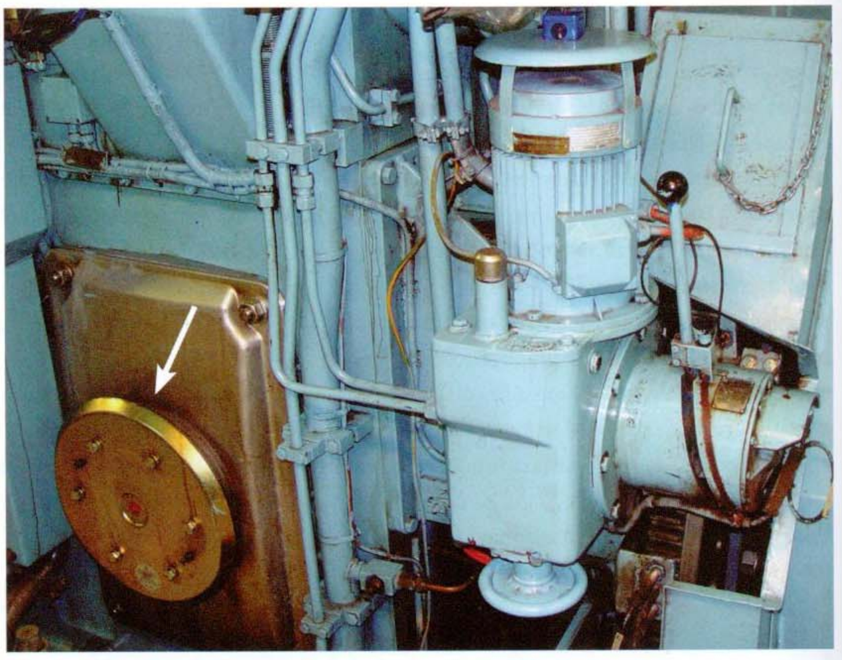
Engines with a cylinder bore larger than 200 millimetres and equal to or smaller than 250 millimetres must be fitted with at least one safety valve at each end of the crankcase.

An engine with more than eight cylinders must be fitted with an extra safety valve at the centre of the crankcase.

Engines with a cylinder bore larger than 250 mm and smaller than or equal to 300 mm must be fitted with at least one safety valve placed near every second crank web with a minimum of two.

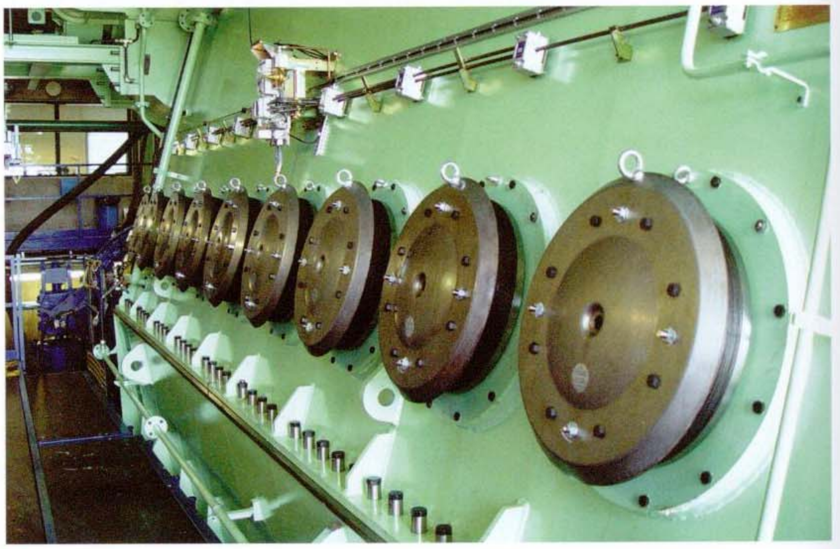
Engines with a cylinder bore larger than 300 millimetres must be equipped with at least one safety valve per cylinder.
Each safety valve must have at least a free cross-sectional area of 45 cm^2.

The total free cross-sectional area of the safety valve on a crankcase of an engine should not be less than 115 cm^2 per m^3 crankcase volume.



A relief valve.

Prescribed for a gear-casing section with a capacity of 0.6 m^3 or more.
Prescribed for engines with a cylinder diameter larger than 200 mm.



Relief valves for every crank of a two-stroke crosshead engine are required for cylinder diameters larger than 300 mm.

Clearly shown, the exiting pressure wave is directed towards the floor when the relief valves are opened. This makes it as safe as possible for people in the vicinity.

Notes about operation

The safety valve must be of the rapidly 'open and close' type.
In normal operation, they must be oil-tight when closed and prevent air from entering the crankcase.
The gas flow must be deflected by a baffling plate in order to warrant the safety of persons standing nearby when an actual explosion occurs.
The gas flow is often directed downwards.

32.11.5 Warning sign

A warning sign indicating that the crankcase covers or doors should not be opened immediately after stoppage, but only after a certain cooling-down period, must be mounted on both sides of the engine.

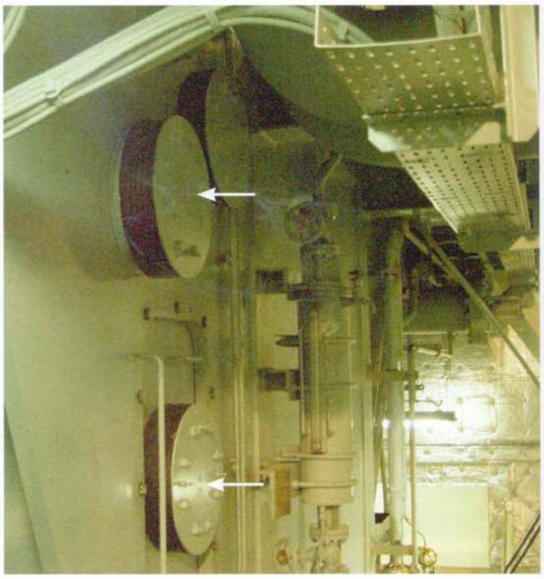
32.11.6 Oil-mist detection system

Engines with a cylinder bore larger than 300 millimetres or with a maximum power output of 2250 kW or higher must be equipped with an oil-mist detection system.
One should be able to read this at a safe distance from the engine.

An oil-mist detection system is mandatory on diesel engines with a cylinder diameter of more than 300 millimetres or with a power output of 2250 kW or higher.

The compulsory 'warning sign' mounted on the engine next to or on the crankcase cover.

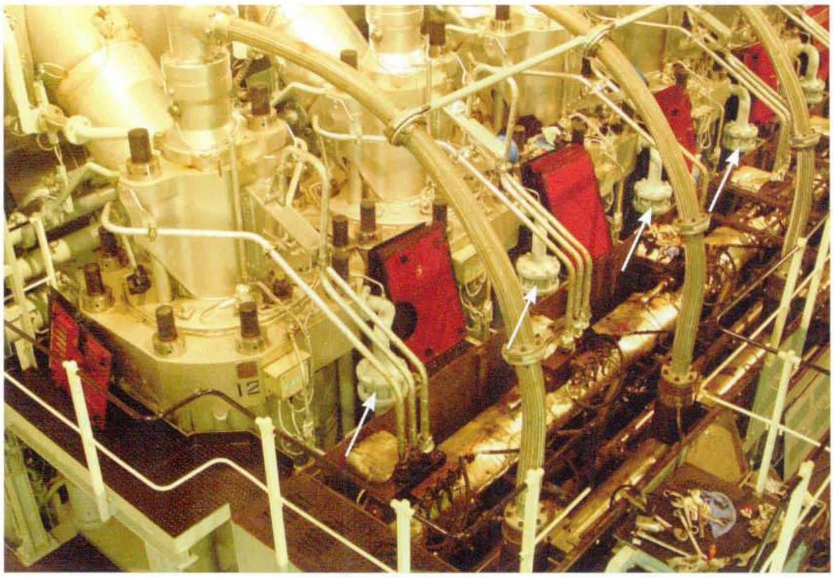
▶
The compulsory relief valves on the scavenging-air space of a two-stroke crosshead engine.



32.11.7 Safety devices in the starting-air system

Many starting-air systems work with pressures of 30 bar. This may be dangerous in certain circumstances.
A separate check valve must be placed in the starting-air line of every engine.
Engines with a cylinder bore larger than 230 millimetres must be provided with a flame arrestor under the following conditions:

a. On directly reversible engines, at the starting-air valve on each cylinder.
b. On non-reversible engines, at the supply line of the main starting-air duct of the engine.



▲
The flame arrestors in the supply line of the starting-air valve of a two-stroke crosshead engine of manufacturer Wärtsilä RT Flex 96C.

32.11.8 Safety devices in the lubricating-oil system

Each engine with a maximum continuous power of 220 kW or more must be fitted with safety devices, which automatically shut down the engine in the event of a failing lubricating-oil supply.
This is not required for engines driving emergency gensets or emergency-fire pumps. These engines require a lubricating-oil pressure alarm.

32.11.9 Safety devices in inlet-air manifolds

Inlet-air manifolds connected with the cylinders must be fitted with safety valves similar to those fitted to the crankcase.

32.11.10 Fuel systems

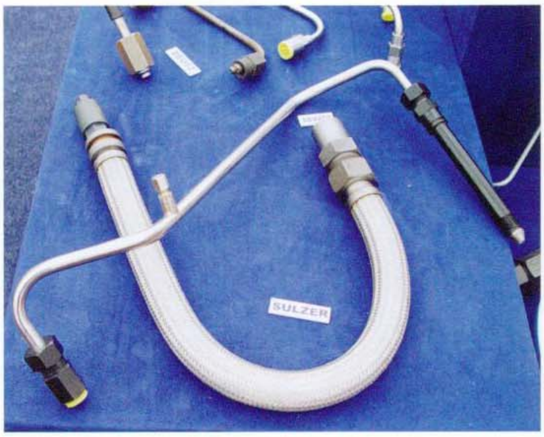
Here many regulations are also applicable.

- Fuel-line connections with flanges may only to be fitted with metal sealing surfaces or a similar fuel-line connection of approved design.
- Feed and return fuel-line systems of the engine must be designed in such a way that no excessively high pressures can occur in the system.
- All the parts of the system must be sufficiently strong to resist the maximum pressures that occur in the system.
- Fuel-oil lines should not be placed too close to high-temperature systems. Steam pipelines, exhaust manifolds, silencers and other equipment must be adequately insulated.

Shielding

Regardless of the intended use and location of internal combustion engines, all external fuel-injection lines between the high-pressure fuel pumps and injectors must be double-walled.

▶
The compulsory double-walled high-pressure fuel line, between the high-pressure fuel pump and high-pressure injector.



The fuel must be:
- safely collected
- drained away without an overpressure
- efficiently monitored
- if pressure variations of > 20 bar occur in feed- and return- fuel lines, these lines must also be double-walled.
- the double-walled high-pressure fuel lines must be of permanent assembly.

▲
The double-walled high-pressure fuel and lubricating-oil pipe for opening the exhaust valves and the three injectors of a two-stroke crosshead engine.

▲
A fuel pump for a six-cylinder Scania diesel engine.

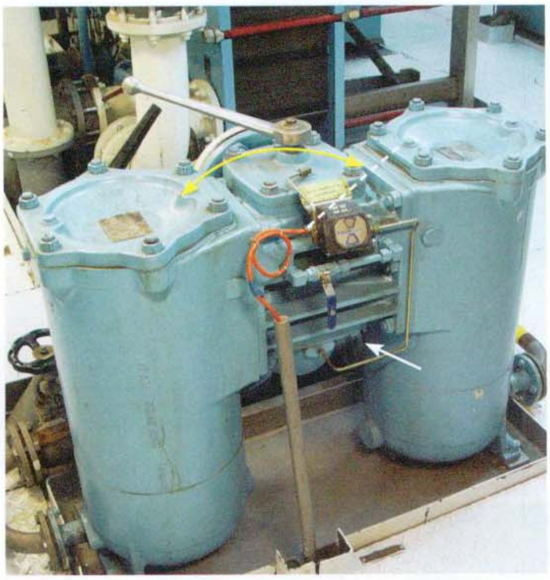
The high-pressure lines towards the injectors are double-walled.

32.11.11 Fuel-oil emulsions

Engines running on fuel emulsions or other liquid fuels must be able to convert to normal fuel-oil systems in case of failure.

32.11.12 Filters for fuel- or lubricating-oil systems

Filters that are directly mounted on the engine should not be located in the immediate proximity of rotating or hot parts. If this is not feasible, the rotating or hot parts must be shielded.



◄
A readily accessible heavy-oil fuel filter. The heavy-oil can be easily led from the one to the other filter element; it is also easy to see when the filter is dirty.

32.11.13 Duplex filters

- Filters must be provided with a drip tray.
- Filters must be able to be switched off safely and drained for cleaning purposes.

◄
Strict requirements also apply for these fuel filters.

- It should be clear which filter is on-line in the system.
- The off-line filter must be arranged so that it can be cleaned while the engine is running.

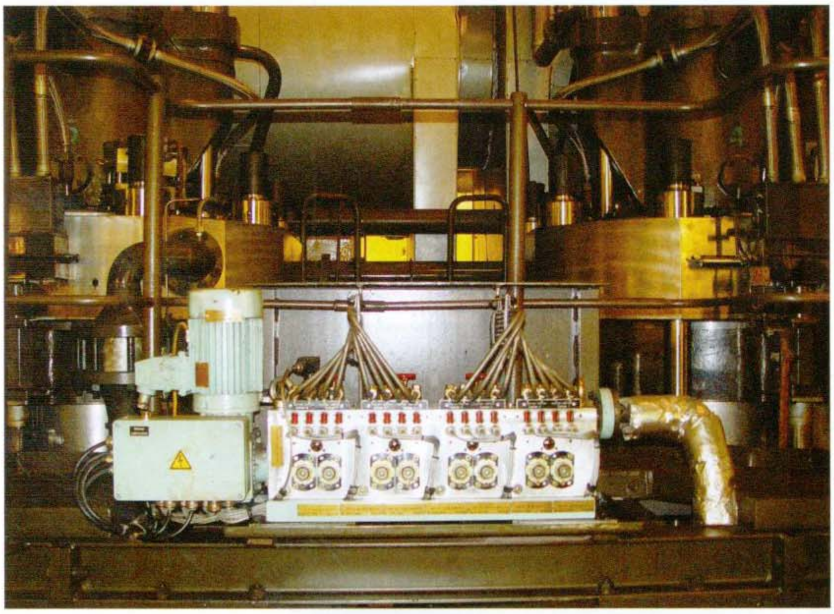
It should also be feasible to vent the filter and depressurise the off-line filter.

32.11.14 The lubricating-oil system

- The engine must be so equipped that the sumps can be topped up during operation.
- One should be able to drain the sump.
- The combination of the oil-drainage lines from the crankcases of two or more engines is not allowed.
- Main lubricating-oil pumps driven by the engine are to be designed to maintain the supply of lubricating oil under all operating conditions.
- Main engines, which drive main lubricating-oil pumps, are to be equipped with independently-driven stand-by lubricating-oil pumps.
- Cylinder lubricating-oil systems for the engine provided with electronically controlled dosing systems must be approved by a classification society.

▼

Cylinder lubricating oil systems for the engine must be approved by a classification society, as they are equipped with electronically controlled dosing systems.



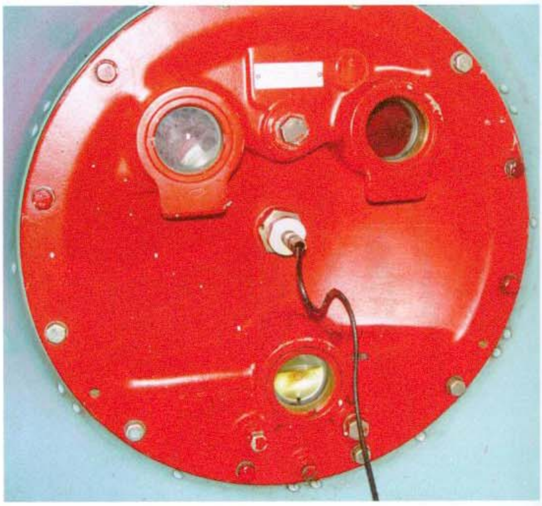
32.11.15 Cooling systems

Similar to the lubricating-oil pumps, cooling-water pumps must have enough capacity to provide a sufficient amount of cooling water for all operating conditions.
Main engines that drive main cooling-water pumps must be equipped with independently-driven stand-by pumps.

If cooling air is obtained from the engine room, the design of the cooling system must be based on an engine-room temperature of at least 45 °C.

32.11.16 Charge-air system

Exhaust-gas turbochargers may not achieve critical speed ranges under any operating conditions.
The supply of lubricating oil must also be sufficient during starting-up and running-down of the turbochargers.



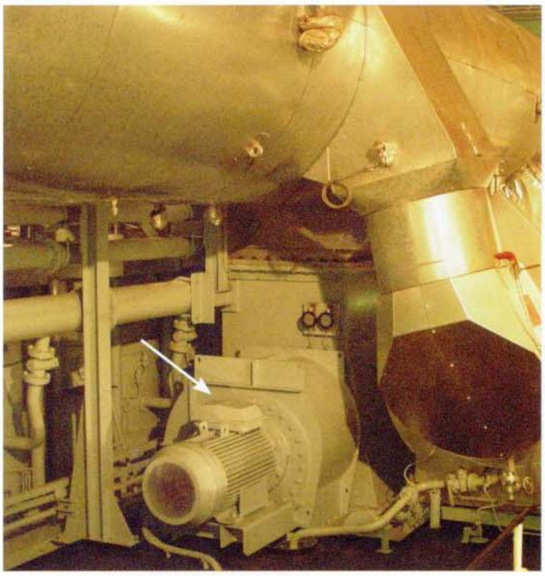
▲

The lubricating-oil system of a turboblower of A.B.B..

The turbine shaft drives the lubricating-oil pump that lubricates the bearings. The lubricating oil in the system therefore is separate from the rest of the engine and remains clean. Look at the sight glass!
Under all conditions, when the turboblower is operational, the bearings are always well lubricated.

▲

An extra lubricating-oil tank, which provides the lubricant for the bearings of this turboblower when the engine stops, until the blower stops.



▲
The electrically-driven auxiliary blower of a two-stroke crosshead engine.

◄
The steel sheet protection of the exhaust-gas line of this propulsion engine may attain a maximum surface temperature of 220 °C.

Even at low engine speeds, main engines must be supplied with a sufficient amount of charge air to ensure a reliable operation.

If required, two-stroke crosshead engines must be fitted with directly or indirectly driven scavenging-air pumps.
In the event of a turbocharger failure, the main engines must be able to maintain emergency running.

32.11.17 Charge-air cooling

Means are to be provided to ensure that the temperature of the scavenging air is within the temperature range specified by the engine manufacturer.
The air-inlet pipes of engines with charge-air coolers must be provided with a means of drainage.

32.11.18 Fire-extinguishing systems on charge-air lines

The scavenging-air line of two-stroke crosshead engines must be provided with an approved fire-extinguishing system that operates independently of the engine room's fire-extinguishing system.

32.11.19 Exhaust-gas lines

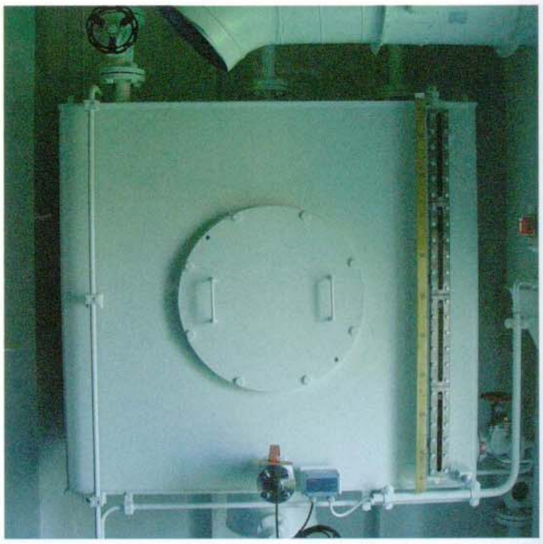
Exhaust-gas lines must be insulated and/or cooled to prevent the surface temperature from exceeding 220 °C.
The insulation material must be fireproof.

32.11.20 Starting-air systems

Here many regulations also apply.
- There must be at least two starting-air compressors for the main engines.
- At least one of the two starting-air compressors should operate independently of the main engine, and should provide at least 50% of the total compressor capacity.
- The volume of the starting-air reservoirs should be such that they can be filled to the legally required capacity from atmospheric pressure to the maximum permitted pressure.
- If the main engine is started with starting air, the available amount of starting air must be distributed over two starting-air reservoirs of similar capacity.
- The total capacity of the starting-air reservoirs must be sufficient to achieve twelve sequential starts of the engine in ahead and astern direction in case of reversible engines, without re-charging. A minimum of six starts is prescribed for non-reversible engines driving a controllable-pitch propeller or they must be equipped with another device, which allows the engine to start without load.
- For auxiliary engines with a starting-air system, this should be at least three starts.

▶
Two large starting-air reservoirs for a twelve-cylinder main engine, a Wärtsilä 12RT Flex 96 C.

The engine must start with these reservoirs **without** the assistance of the air compressors, 6 times forwards and 6 times in reverse.



▲
All the necessary equipment for an emergency-diesel generator set must be in the same space as the generator. Shown here, the diesel-oil day-tank.

32.11.21 Electric starting systems

Here a number of regulations also apply.

- Main engines that are electrically started must be equipped with two separate starting batteries, which cannot be operated in parallel.
- Each starting battery must be able to start the engine in cold condition.
- The total capacity of the starting batteries must be sufficient so that within half an hour, the previously described starts can be executed without having to re-charge the starting batteries.

32.11.22 Auxiliary engines

- At least three starts.
- Starting batteries should only be used to start the engine and the monitoring installations.

32.11.23 Starting emergency-diesel generators

- Emergency-diesel generators should be able to start at temperatures as low as 0°C.
- If it is not possible to start at this low temperature, the generator must be provided with heating.
- The set must be operational in all weather and sea conditions.
- The starting, loading and energy-storage equipment, such as the battery and the diesel-oil tank should be installed in the emergency-diesel generator location.

32.11.24 Local operating-control system

There are a number of requirements that apply to local operating control.

▶
The emergency manoeuvring position of a two-stroke crosshead engine, a Wärtsilä 10 RTA 96 C.

in circle: start and reverse lever

Table 2.7 Alarms and indicators

Description	Propulsion engines	Auxiliary engines	Emergency engines
speed/direction of rotation	I		
engine overspeed[5]	A, S	A, S	A, S
lubricating oil pressure at engine inlet	I, L, S	I, L, S	I, L
lubricating oil temperature at engine inlet	I, H	I[5],H[5]	I[5],H[5]
fuel oil pressure at engine inlet	I	I	
fuel oil temperature at engine inlet[1]	I	I	
fuel oil leakage from high pressure pipes	A	A	A
cylinder cooling water pressure or flow at engine inlet	I, L	I, L[4]	I[5], L[5]
cylinder cooling water temperature at engine outlet	I, H	I, H	I, H
piston coolant pressure at engine inlet	I, L		
piston coolant temperature at engine outlet	I, H		
charge air pressure at cylinder inlet	I		
charge air temperature at charge air cooler inlet	I		
charge air temperature at charge air cooler outlet	I, H		
starting air pressure	I, L		
control air pressure	I, L		
exhaust gas temperature[2]	I, H[3]		
oil mist concentration in crankcase or alternative monitoring system[7, 8]	I, H[6]	I, H[6]	I, H[6]
1 for engines running on heavy fuel oil only 2 where ever the dimensions permit, at each cylinder outlet and at the turbo charger inlet and outlet 3 at turbo charger outlet only 4 cooling water pressure or flow 5 only for an engine output ≥ 220 kW 6 for engines having an output > 2250 kW or a cylinder bore > 300 mm 7 alternative methods of monitoring may be approved by GL 8 an engine shutdown may be provided where necessary	I: Indicator A: Alarm H: Alarm for upper limit L: Alarm for lower limit S: Shutdown		

◄

Table of compulsory alarms and indicators that must be read in the emergency start manoeuvring position.

32.11.25 Main engine

- A local system that can operate and monitor the engine must be fitted.
- If reduction gearing and controllable-pitch propeller systems have been fitted, the above-mentioned local control system must be able to provide emergency running.
- Critical speed must be indicated with red sectors on the speed gauges.

▲

The emergency manoeuvring position of a four-stroke trunk-piston engine, a nine-cylinder Caterpillar Mak 43 engine.

Arrow: toothed and reverse lever

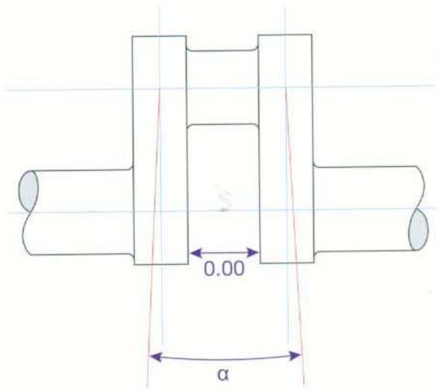
◄

Critical speeds must be indicated by red sectors on the speedometers. This engine does not have a critical speed.

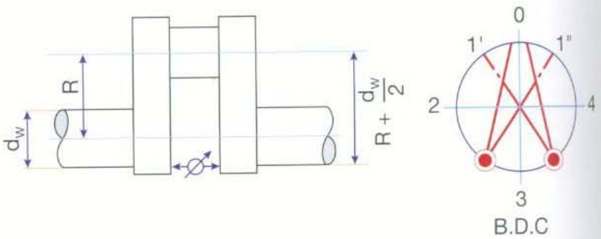
32.12 Engine alignment

Here too special regulations have been drawn up.
- The crankshaft alignment must be checked each time an engine is re-aligned on its bedplate.
- This is done by **clock gauging** of the crankshaft. The crank-web deflection of every cylinder is to be measured.

In order to compare later crankshaft deflections, note should be taken of the loading and draught of the vessel. The conditions of the engine when the deflection was measured, cold, preheated or at operating temperature.



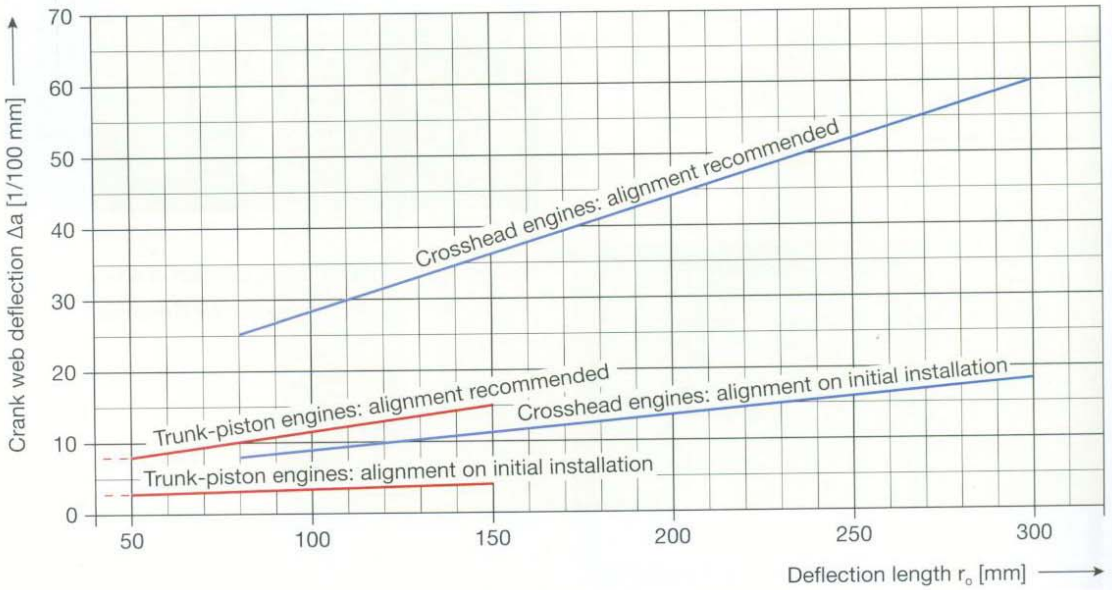
▲
If the crankwebs do not have the original values between the (engine manufacturer) stamped centres when the crankshaft rotates, they show a deviation known as the angular rotation of the webs. This is known as 'crankshaft clock gauging' or 'a deflection measurement'. It is clear that the larger a crank at the same deflection (angle α) has a larger measured value than a smaller crank.



▲
The basic standard measurements for defining the crankshaft deflection value, r_o.

▼
A chart, showing clearly how large the maximum crankshaft deflections may be.

Horizontal: the distance r_o indicating where the measurement must be taken.
Vertical: the deflection result Δa
In the figure are the lines for trunk-piston engines and crosshead engines.
Both lower lines are the recommended values after initial engine installation.
Both upper lines are the advised maximum crankshaft deflections.
The larger r_o is, the larger the crankshaft diameter and the larger the deflection may be.
Guidelines are in the process of being established for the angular rotation that a crank may deviate from the ideal line.
The same maximum angular rotation has a larger absolute value with a larger crank.

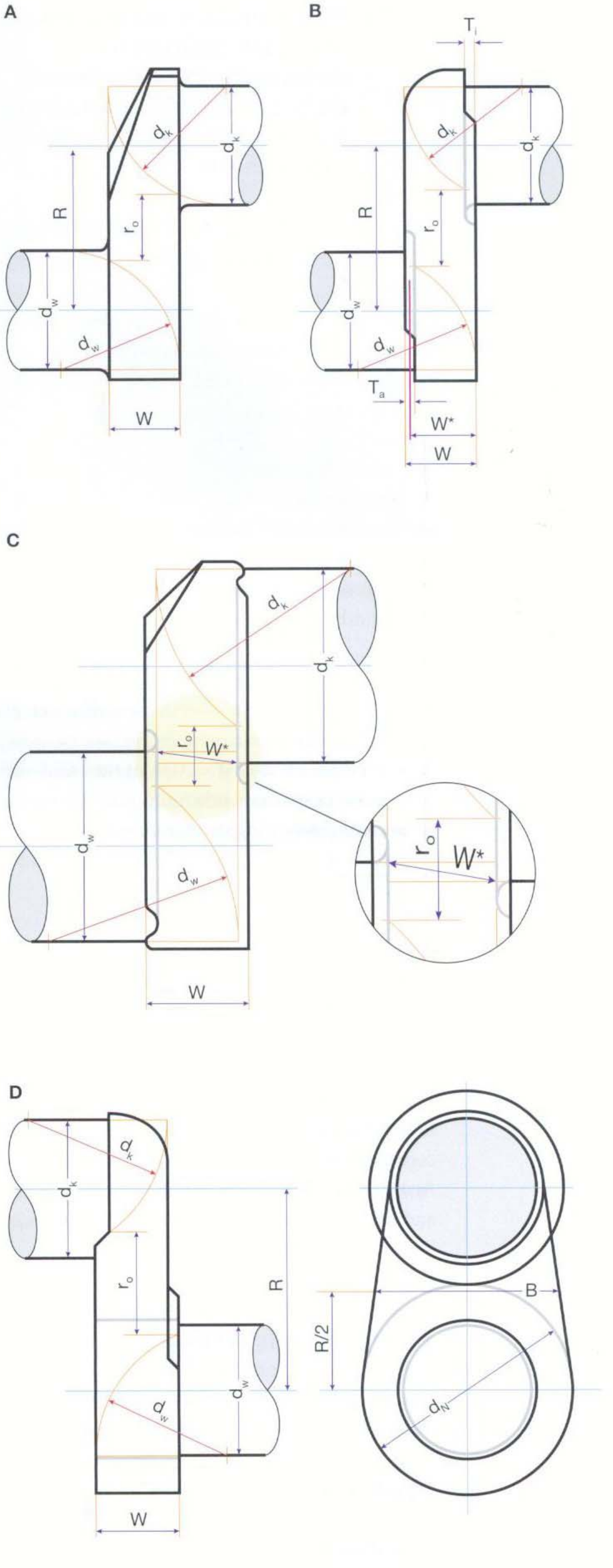


Remarks for clock gauging of the crankshaft and measuring the distance between the crank webs of each cylinder

The measurements should be performed at a certain distance from the heart line of the crankshaft

that is: $R + \frac{dw}{2}$.

In virtually all crankshafts, both webs of the crank are provided with a centre hole suitable for the crankshaft clock gauge.
In normal crankshaft clock gauging, the connecting rod remains mounted on the crankpin. Therefore, the values 1' and 1" are a mean.
The location of the centre holes is dependent on the type of crankshaft.
One can distinguish four types: A, B, C and D.



▶ The standard measurements for defining of the crankshaft deflection values, r_o, for four different types of crankshafts.

They are all forged steel crankshafts.
Types A, B and C are manufactured in a single piece forging for four-stroke trunk-piston engines.
The type D crankshaft is manufactured by the joining together of single 'throws' consisting of two webs and the crank pin and used for two-stroke crosshead engines.

Type A: standard crankshaft, negative crank pin/crankshaft formula result.
Type B: with milled sections T_i and T_a
Type C: positive crank pin/crankshaft formula result.

32.13 The standard measurements and calculations to determine the crankshaft deflection measurement r_o for four different types of crankshafts

32.13.1 Calculating the deflection length r_o according to Germanischer Lloyd

There are separate calculations for shafts forged in a single piece forging, A, B and C and for a crankshaft manufacture by joining single throws together D.

The symbols applied are:
R = crank radius/radius (mm)
H = piston stroke (2R) (mm)
d_k = crankpin diameter (mm)
d_w = crankshaft diameter (mm)
W = width of the crank web (mm)
B = width of the crank web at distance $\frac{R}{2}$ (mm)
T_i = depth of milled section in the crank web at the crankpin side (mm)
T_a = depth of the milled section in the crank web at the crankshaft side (mm)
S = crankpin/bearing overlap (mm)

$$S = \frac{(d_k + d_w)}{2} - R$$

At a negative crankpin/bearing overlap (s < o) the deflection length r_o is in accordance with figure A and is calculated as follows:

$$r_o = 0.5(H + d_k + d_w) - W\left(\sqrt{\frac{2d_k}{W}} - 1 + \sqrt{\frac{2d_w}{W}} - 1\right)$$

In the other cases of B, C and D the formula must constantly be adjusted.
Apart from the material, the shape, construction and detailing of the crankshafts, much attention is paid to the exact position of the deflection measurement.

All are forged-iron crankshafts.
Types A, B and C are forged by single piece forging and usually manufactured for four-stroke trunk-piston engines.

Type A: with a negative result is:

s < o in the formula $s = \frac{(d_k + d_w)}{2} - R$

$$r_o = 0.5(H + d_k + d_w) - W\left(\sqrt{\frac{2d_k}{W}} - 1 + \sqrt{\frac{2d_w}{W}} - 1\right)$$

When sections of a crank web have been milled away, the so-called 'web undercut', W in the formula must be modified to:

$$W^1 = W - \frac{(T_i + T_a)}{2}.$$

This is crankshaft type B.

For crankshaft type D, d_w in the formula must be substituted by $d_w{}^1$:

$$d_w{}^1 = \frac{1}{3}(d_n - d_w) + d_w$$

For crankshaft type C, with a positive crankpin/crankshaft overlap ≥ 0 in the formula $s = \frac{(d_k + d_w)}{2} - R$ the W value in the formula must be replaced by W^1.

$$W^1 + \sqrt{(W - T_i - T_a)^2} + [0{,}5(d_k + d_w - H)]^2$$

From the abovementioned formulas, it can be inferred that the position of the crankshaft measurement for each engine was calculated earlier. Clearly, an engine manufacturer applies the deflection centres when finishing the crankshaft rather than leaving it to the mechanic to guess where the centres should be located.

32.14 Procedure for reconditioning parts

Many engine parts repaired after wear has been detected must be re-tested and approved, the following amongst others:
- Crankshafts
- Connecting rods
- Engine blocks
- Cylinder covers
- Pistons

The procedure is often as follows:
- The parts are disassembled and sent to a specialised reconditioning company.
- The parts are cleaned, inspected, measured and for, amongst others, a crankshaft is the material hardness of the journals measured.
- Advice is given to the client regarding the repair. It is possible that the advice is to repair the part; however, with serious wear or damage, it can be recommended that the part should be replaced.

▲
The installation of a new sleeve bearing in the connecting-rod eye is also a part of the overhauling of engine parts.

Shown here, the new sleeve bearing is cooled using liquid nitrogen so that the sleeve can be easily installed in the connecting-rod eye. If both parts have ambient temperature, the sleeve will not fit into the eye.
Wearing special 'cold resistant' gloves is necessary, the material temperatures are under -100 °C!

▲
An overheated crank pin of a medium-speed four-stroke engine.

The question is whether this type of damage can be effectively repaired.
Together with the classification society, the specialised repair company, an overhaul plan is prepared. The final result must guarantee that the surface hardness of the overheated crank pin does not deviate too much from the surface hardness of the other crank pins.

The repair advice will also include the cost of the repair.
After the repair has been approved by the client, the part will be repaired according to the specifications of the engine manufacturer.
This means that for damaged crankshaft journals that the crank pin and/or crankshaft is ground to a certain **undersize**, and then fitted with **oversized** bearing shells.
In order to achieve this, the engine manufacturers manufacture new crankshaft in such a manner that the shafts are over-dimensioned so that grinding is possible.
Therefore, the entire repair is performed according to the manufacturers' specifications.
After reconditioning/repair, the classification society for the engine, are invited to perform an audit.
When approved, the part is provided with a code of numbers and signs by the classification society. These are stamped. Surveyors of a classification society also have their own personal number, which is stamped into the part.

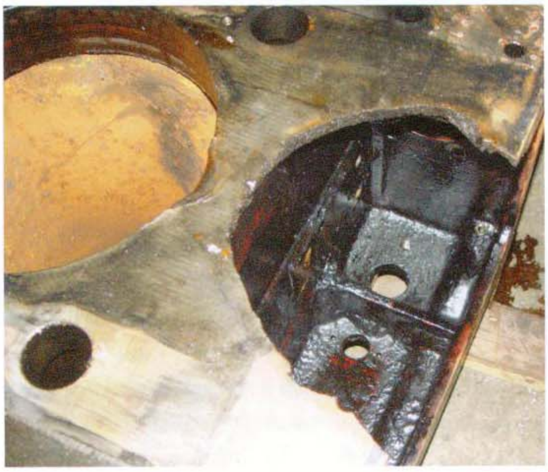
32.15 New parts

These are provided with certificates by classification societies.
This is because often after purchase by the client, the engine manufacturers do not yet know to which class the engine and its spare parts belong. Some engine manufacturers have **six**-classification society's visit!

32.16 Special cases of wear and damage to engine parts

In can occur in very special cases that damage to, for instance, an engine block or a crankshaft that the classification society will be asked to be part of the discussions regarding the repair. An example is, whether the crankshaft of an engine in a vessel is disassembled and removed, or the disassembly of an engine block in a diesel power station.

In both cases, the disassembly and then assembly cost a great deal of time and money. During the repair, the vessel or a part of the diesel power station is non-operational. A stationary ship costs a huge amount of money!



▶ A severely damaged cast-iron engine block.

The question remains: Is the repair technically possible and is the engine then dependable? What is the client's expectation for the lifetime of this engine?

▲ A damaged two-stroke crosshead engine guide.

The guide is removed from the A-frame and inspected on the engine room floor.
The white metal running surface has separated from the guide. This guide must be repaired.

▶ Inspection of this camshaft drive for a fuel pump shows that both running surfaces are still in a good state of repair. The running surfaces are smooth and have no scratches or damages.

◀ The specifications of the engine manufacturer must also be strictly adhered to when line boring commissioned engines.

32.16.1 Engine blocks and connecting rods

Occasionally, after notification by a reconditioning company, a representative of the classification society will visit for an inspection. When the notification states that the parts will be reconditioned according to the specifications of the engine manufacturer and the parties are known to each other from an earlier large number of repairs, often no control visit takes place.

32.17 Damages to the engine or engine parts

A large number of parties are involved:
- An expert of the company, which carries the insurance of the engine.
- A surveyor of the classification society.
- The engine supplier and/or manufacturer.
- The owner/user of the engine.

Usually, these parties meet on the ship or the diesel power station for deliberations.

▲
Damages to crankshafts are costly; when in repair at a specialised repair company, the crankshaft must be disassembled and removed.

The complete engine block must be placed on hydraulic jacks.
The rejected crankshaft must be replaced; the costs can reach a hundreds thousands euros; especially for category III engines.
Severe damage to the crankshaft of a two-stroke crosshead engines result in even higher costs. The ship is out of operation for a long time!

32.18 Damages

Insurance companies, that also insure diesel engines, draw remarkable conclusions regarding damages to engines.

32.18.1 Engine damages, all engine types

Claims type	Number	Total cost (USD)	Avg. Cost (USD)
Turbocharger	84 (36,2%)	16.983.474 (24,4%)	202.184
Crankshaft, Connecting rod	23 (9,9%)	13.949.870 (20,0%)	606.516
Cylinderliner	17 (7,3%)	4.267.795 (6,1%)	251.047
Entablature, Staybolts	17 (7,3%)	3.505.803 (5,0%)	206.224
Journal, Bearing	15 (6,5%)	6.653.302 (9,5%)	443.553
Fuelpump, Gears	12 (5,2%)	3.161.929 (4,5%)	263.494
Camshaft, Coupling	10 (4,3%)	3.804.377 (5,5%)	380.438
Piston, Pistonrod	9 (3,9%)	2.702.420 (3,9%)	300.269

▲
Noticeable is the large number of damages to turbo-blowers. A crucial part for modern highly-loaded diesel engines.
The crankshaft, connecting rod and cylinder liner follow.

32.18.2 Engine damages, low-speed two-stroke crosshead engines

Claims type	Number	Total cost (USD)	Avg. Cost (USD)
Turbocharger	63 (42,6%)	14.057.056 (40,7%)	223.128
Entablature, Staybolts	17 (11,5%)	3.505.803 (10,2%)	206.224
Cylinderliner	15 (10,1%)	3.810.363 (11,0%)	254.024
Journal, Bearing	9 (6,1%)	2.916.201 (8,4%)	324.022
Piston, Pistonrod	7 (4,7%)	1.609.588 (4,7%)	229.941

▲
The turboblower also has the largest number of damages for the low-speed two-stroke crosshead engines, followed by the stay bolts, cylinder liners, journal bearings, pistons and piston rods.

32.18.3 Engine damages, medium-speed four-stroke engines

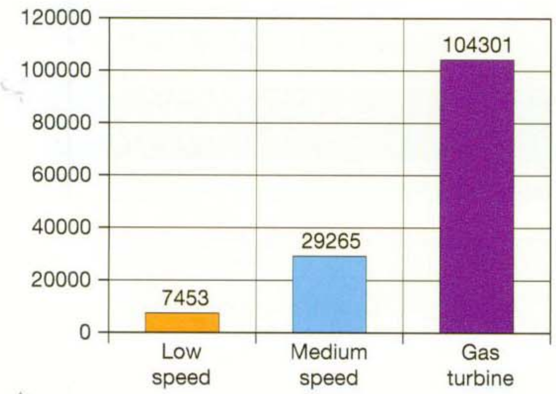
Claims type	Number	Total cost (USD)	Avg. Cost (USD)
Turbocharger	21 (25,6%)	2.926.417 (8,9%)	139.353
Crankshaft, Connecting rod	21 (25,6%)	13.593.961 (41,5%)	647.331
Camshaft, Coupling	8 (9,8%)	3.451.850 (10,5%)	431.481
Journal, Bearing	6 (7,3%)	3.737.102 (11,4%)	622.850
Fuelpump, Gears	5 (6,1%)	765.436 (2,3%)	153.087

▲

For the four-stroke medium-speed diesel engines, the damages to turboblowers and crankshafts with connecting rod are of equal number; a remarkable shift to the drive gearing.

This is the result of the high thermal and mechanical loads of these engines types. See the load numbers in Chapter 5.

The graph below gives clarity regarding the propulsion types.



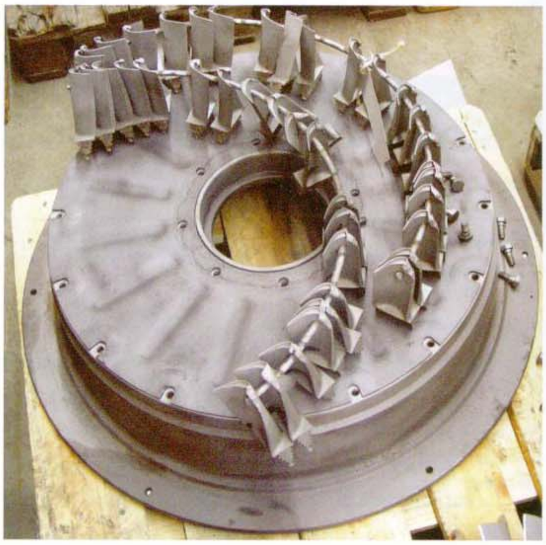
▲

Conclusion:

The damage to low-speed two-stroke crosshead engines is by far the lowest; almost four times lower than the damage to medium-speed four-stroke engines and fourteen times lower than the damages to gas turbines!

32.18.4 Some general remarks

1. Damages to propulsion plants are costly.
2. Especially medium-speed four-stroke diesel engines run large risks with respect to damages and therefore costs. They only represent 20% of the total number of insured engines, but 50% of the total costs of damages.
3. Damage to turboblowers is the largest item of loss for diesel engines.
4. The most costly damages are those of the crankshaft and connecting rod; especially for four-stroke medium-speed engines.
5. Damage to the crankshaft of a low-speed two-stroke crosshead engine is the major part in the total amount of claims paid out for damages. Think about, the disassembly and removal of the damaged crankshaft and the installation of a new crankshaft. A new crankshaft is a very expensive engine part!
6. The total loss of a medium-speed four-stroke engine, where the crankshaft and engine block are beyond repair are also part of costs that insurance companies must pay out.



▲

Damages to turboblowers are the most common occurring damage, to both four-stroke medium-speed and two-stroke crosshead engines.

32.19 Examples of certificates

A certificate for 'oil machines' and therefore for diesel engines.

The certificate is for a Mak 8M32C propulsion engine for a tanker. This has been manufactured by Volharding Shipyards B.V., Hoogezand, The Netherlands.

CERTIFIED COPY

Certificate no: HAM 0740246/1
Page 1 of 1

Lloyd's Register

Certificate For Oil Engines (Quality Assurance)

Office Kiel	Work's order number 261 956
Manufacturer Caterpillar Motoren GmbH & Co. KG	Purchaser Pritchard Gordon Ltd., Slaugham, UK
Name and address of works Caterpillar Motoren GmbH & Co. KG Falckensteiner Strasse 2 24159 Kiel Germany	Purchaser's order number Tanker Intended service Main engine Intended for ship/yard/engine number Volharding Shipyards B.V. Hoogezand, Yard No. 603

Particulars

Engine serial number 38809	Number of cylinders 8	Diameter of cylinder (mm) 320
Manufacturers designation 'MaK' 8M32C	Stroke (mm) 480	
Shaft power kW and RPM (continuous rating) 3840 kW at 600	Method of cooling Water	

Design Appraisal Document no. and date
LR EMEA Hamburg, Plan Approcal Centre ref. HMD 11808-03/3 dated 10 November 2006

Testing

The engine has been run on a dynamometer and has satisfied the agreed test schedule

Identification Marks (Serial No. Control No. and Date)

LR QAM 042 LRS HAM 0740246 (entablature stamped)

Manufacturer's Statement

This is to certify that the oil engine described above has been constructed and tested with satisfactory results in accordance with the current Rules and Regulations. A copy of the applicable Design Appraisal Document is attached.

Signed on behalf of manufacturer

Signature	Date
Name in block capitals G. HERRMANN / D. DITTRICH / Dr. GALLUN	Position in company Quality Supervisor Department CQ3

This certificate is issued by the manufacturer in accordance with the arrangements authorised by Lloyd's Register EMEA in Quality Assurance approval

Certificate number QAM 042/A1

I certify that these arrangements are being kept under review by regular and systematic auditing of the approved Manufacturing and quality control procedures

Date
18 October 2007

Lloyd's Register EMEA
Signature
Kiel Office
H. PURCZ
Surveyor to Lloyd's Register EMEA

Lloyd's Register

A member of the Lloyd's Register Group

This engine will be accepted for fitting in ships classed or intended to be classed with Lloyd's Register subject to satisfactory installation under the usual conditions of survey and testing, and where required by the rules, approval of vibration characteristics.

Form QA018 (2004.12)

Report on the main engine, an internal combustion engine with all important engine data, such as, the cylinder diameter, the stroke, the shaft power, the mean indicated pressure and the maximum combustion pressure. The data includes information with respect to the torsional-vibration damper, the crankcase volume, the number of relief valves, the type of governor, the cylinder-cover material, piston crowns and oil-mist detector.

Report no: HAM 0740246/2
Page 1 of 3

Lloyd's Register — Report on Main Internal Combustion Reciprocating Engine

Ship's Name -	Port	IMO number
Place of Survey, if different from above		Office **Kiel**
Number of visits in shops **1**	First Date **17 October 2007**	Last date **17 October 2007**
Ship built by **Volharding Shipyards B.V., Hoogezand, The Netherlands**		Yard number **603**
Engine made by **Caterpillar Motoren GmbH & Co. KG, Falckensteiner Strasse 2, 24159 Kiel, Germany.**		Year and month **2007/10**
Engine make and type **'MaK' 8M32C**		Engine Serial No **38809**
State if cylinders in V or other special formation **In line**	Number of engines **1**	Number of crankshafts **1**
2 or 4 stroke cycle **4**	Maximum kW **3840**	Number of cylinders, each engine **8**
Bore **320 mm**	Service kW **3840**	At rpm **600**
Stroke **480 mm**	Corresponding M.I.P. **27.2 bar**	Maximum cylinder pressure **191 bar**
Is engine of crosshead or trunk piston type? **Trunk piston type**	Are relief valves fitted to each cylinder? **Yes**	
Is welded construction used for: Bedplate? **No**	Frames? **No**	Entablature? **No**
Cooling medium for: Cylinders? **F.W.**	Pistons? **L.O.**	Fuel Valves? **L.O.**
Must engine be removed for overhaul of main bearings, etc.? **No**	T.V.C. approval letter date and reference number -	
If torsional vibration damper or detuner fitted, state type, e.g. spring viscous etc. **Spring type**	State where fitted **Crankshaft free end**	
Total internal volume of crankcase **7.7 m³**	Number and total area of crankcase explosion relief devices **8 = 1144 cm²**	
Are relief devices fitted to scavenge/supercharger manifold? **No**	Is engine reversible? **No**	How is engine started? **Compressed air**
Type of governor fitted **UG-40 DI**	Material of: Cylinder covers **Cast iron**	Piston crowns **Forged steel**
Are flame guards or traps fitted to: Crankcase relief devices? **Yes**	Starting air pipes at cylinder starting air valves? **Yes**	
Is an approved type crankcase oil mist detector fitted? **Schaller VN 115/87**		

Flywheel Shaft

Separate, integral with crank or thrust shaft **Integral with crankshaft**		
Material **Integral with crankshaft**	Approved tensile strength **As for crankshaft**	Actual tensile strength **As for crankshaft**
Diameter **As for crankshaft**	Flywheel weight **1405 kg**	Flywheel diameter **1439 mm**

Thrust Shaft

Separate, integral with crank or thrust shaft
Separate

Material	Approved tensile strength	Actual tensile strength	Diameter adjacent to collar
N/A	N/A	N/A	N/A

NOTICE: The particulars in this report are to be given as fully and as clearly as possible. Where the answer is "NO" or "NONE", state. Ticks and other signs of a doubtful meaning are not to be used. Wording not applicable to be cancelled.

Form 2011 (2004.11)

▼

Detailed crankshaft data, an important part. Documentation regarding the used material composition, the manufacturing method, the tensile strength and pump data.

Report no: HAM 0740246/2
Page 2 of 3

Two Stroke Engines Only

Number and type of scavenge pumps of blowers each engine and how driven **N/A**	
For E.G. blowers state makers name and type **N/A**	Where exhaust gas driven blowers only are fitted, can engine operate with one out of action? **N/A**
State any emergency arrangements fitted **N/A**	Is the exhaust discharged through ports in the cylinders or through valves in the cylinder covers **N/A**
Are undersides of pistons used as scavenge pumps? **N/A**	Scavenge air pressure at full power. **N/A**

Four Stroke Engines Only

Is engine supercharged? **Yes**	Number of exhaust has driven supercharge blowers per engine **1**
Are undersides of pistons used as supercharge pumps? **No**	Supercharge pressure at full power **2,937 bar**

All Engines

Number of blower/supercharger air coolers each engine **1**	Number of blower/supercharger oil coolers each engine **None**

Crankshaft

Number of main journals **9**	Are main bearings of ball or roller type? **No**	Distance between inner edges of bearings in way of cranks **407 mm**
Distance between centre lines of side rods opposed piston engines **N/A**	Built, semi-built or solid crankshaft **Solid crankshaft**	
Diameter: Journals **300 mm**	Centre crank pins **280 mm**	Side crank pins **N/A**
Breadth of webs at mid throw **420 mm**	Axial thickness of webs **No. 1 web = 130 mm, remainder = 101 mm**	
If webs shrunk, state radial thickness round eyeholes **N/A**	Nominal shrinkage allowance if dowel pins are not fitted **N/A**	

Material (state whether cast or forged)

Pins	Webs	Journals
50CrMo4+Ni+V	**50CrMo4+Ni+V**	**50CrMo4+Ni+V**

If forged, state method
Continuous grain flow forged

Approved Tensile Strength

Pins	Webs	Journals
860 N/mm²	**860 N/mm²**	**860 N/mm²**

Actual Tensile Strength

Pins	Webs	Journals
900 N/mm²	**900 N/mm²**	**900 N/mm²**

Material of coupling bolts
42 CrMo 4 V

Main Engine Driven Pumps each engine.

State number and purpose of each type of pump (including fuel injection and fuel surcharge), type of drive, and for bilge pumps capacity at normal rpm
1 L.O. pressure pump (gear drive), 2 F.W. circulation pumps (gear drive), 8 F.O. injection pumps (cam drive)

Form 2011 (2004.11)

▼

Identification marks of forgings and castings.

Report no: HAM 0740246/2
Page 3 of 3

Air Compressors
State number and how connected and whether can be declutched
No air compressor attached to the engine

Identification Marks of important forgings and castings (Shafting Certificates to be forwarded)
Crankshaft
QA023 STU 0700164, Heat No. 71220, Serial No. G 993 325,
Thrust/flywheel shaft
N/A
Is a detailed list of certificates attached to the report stating item, manufacturer, port of issue, certificate number, and identification markings?
Yes

Declaration
Declaration to be signed by engine builders

To the best of our knowledge this machinery has been soundly constructed in conformity with the Rules and Regulations, and the foregoing particulars of main engines are correct.

Signature – Manufacturer	Name in BLOCK CAPITALS	Date
i.A. [signature]	C. MOHR	25.10.07

Date and Port of Approval of Plans
State if general approval
General Approval
Crankshaft
LR EMEA Hamburg, Plan Approcal Centre ref. HMD 11808-03/3 dated 10 November 2006
Thrust/flywheel shaft
N/A
Air Receivers
N/A
A previous similar case was for (name)
N/A
Eng. Number
N/A
Port and report number
N/A

The machinery reported above has been built under Special Survey in accordance with the Rules, approved plans and Secretary's letters, examined, running on the test bed and found satisfactory. The materials and workmanship are good, the spare gear required by the Rules has been supplied and the machinery is eligible in my opinion to be fitted in a classed ship.

Issued at **Kiel** on **18 October 2007**

Lloyd's Register EMEA
Kiel Office
H. PURTZ
Surveyor to Lloyd's Register EMEA
A member of the Lloyd's Register Group

Form 2011 (2004.11)

▼

Certificate for forged steel crankshafts.
All the important data is shown. The identification number, method of manufacture and material tests.

Certificate no: **STU 0700164**
Page 1 of 1

Lloyd's Register

Certificate for Forged Steel Crankshafts (Quality Assurance)

Office
Stuttgart
Manufacturer
Maschinenfabrik Alfing Kessler GmbH, Aalen
Name and address of works
Maschinenfabrik Alfing Kessler GmbH
Auguste-Kessler-Str. 20
73433 Aalen

Work's order number
14560/16
Purchase order number
58369
Purchaser
Caterpillar Motoren GmbH & Co. KG, Kiel
Client
Manufacturer

Particulars
Crankshaft designation
One (1) cgf-forged, Type: 8 M 32-500
Plan number
1.94.7-25.10.01-04"d/d", Weight: 4400 kg (each)
Type and grade of steel
50CrMo4+Ni+V, quenched and tempered
Design appraisal Document no. and date
HMD 11808-3 Issue 1 dated 09.02.2004

Number of crankshafts
Forging No(s): G 993 325
Number of throws
8-throw
Stage of manufacture **finished machined**

Forging Process
continous grain flow forged

Non-Destructive Examination
Magnetic Particle
yes

Ultrasonic
yes

Identification
Cast number
71 220

Test number
N/A

QA023, STU 0700164
Forging No(s): see above

Date
N/A

This is to certify that the forgings detailed above have been inspected and tested with satisfactory results and conform in all respects to the specification and order.

Signed on behalf of (Manufacturer)
Maschinenfabrik Alfing Kessler GmbH, Aalen

Remarks
N/A

Signature – Manufacturer
J. Schmidt

Status
Quality Manager
Date
24 August 2007

This certificate is issued by the manufacturer in accordance with the arrangements authorised by **Lloyd's Register EMEA** in Quality Assurance Approval Certificate Number **QA023**

I certify these arrangements are being kept under review by regular and systematic auditing of the approved manufacturing and quality control procedures.

Remarks
For details of Mechanical Properties - see attached Works Test Certificate 38068 dated 26.06.07

Signature
O. Fuchs
Surveyor to Lloyd's **Register EMEA**
A member of the Lloyd's Register Group
Date **26 September 2007**

The crankshafts will be accepted for engines, compressors or other items of machinery which are required to be constructed in accordance with the Rules and Regulations for the classifications of ships.

Form 1281 (2006.04)

▼
Inspection report for the crankshaft from the engineering works, Alfing in Aalen, Germany.

MASCHINENFABRIK ALFING KESSLER GMBH
73433 AALEN - WASSERALFINGEN

ALFING
MASCHINENFABRIK
ALFING KESSLER GMBH
AUTOMOTIVE CRANKSHAFTS
LARGE CRANKSHAFTS
HARDENING EQUIPMENT

Inspection - Certificate
acccording to EN 10204 3.1

No. **60580**

Subject	Type	Drawing-No.	Material
1 Crankshaft, 8 throw, finished maschined	8 M 32-500	1.94.7-25.10.01-04 "d/d" 263 250 125	50CrMo4+Ni+V +QT quenched and tempere
Purchase	**Order No.**	**Alfing-Reference-No.**	**Inspection Society**
Caterpillar Motoren GmbH & Co.KG Kiel	58369 v. 19.06.2006	14560/16 HfH-4101 4010922	LR

The above-mentioned products have been manufactured and tested in accordance with the order and, where appropriate, with the stipulated specifications.
The results recorded are documented on the form/forms

Mech. Eigenschaften Nr. 38068.
crankshaft no. G 993 325.

Which as appendix / appendices to this covering sheet forms / form part of the inspection certificate in accordance with EN 10204.

Remarks:

Ultraschall-Prüfung am Schmiedeprodukt entsprechend Vorschrift AA 10/02/03/A durchgeführt.
Befund: Innerhalb der Vorschrift

Magnetische Rißprüfung gemäß Vorschrift AA 10/03/27/B durchgeführt.
Befund: Ohne Anzeigen.

We hereby certify that the results recorded for the above-mentioned products correspond to the customer's requirements or have been approved by the customer.

Maschinenfabrik Alfing Kessler GmbH
Quality Assurance Dept.

i. A. Dock

(Work's Inspector)
(J. Schmidt)

Date
24.08.2007
QWA - Schö

Enclosure(s)

ALFING

Maschinenfabrik Alfing Keßler GmbH
D - 73433 Aalen-Wasseralfingen

Mech. Eigenschaften
Mech. Properties

Nr./No. 38068

Verteiler:
C.C.:
Abnahmewesen (weiß/blau)
Quality Assurance (white/blue)
LSA (orange)

Seite/Page 1 von/of 1 Seiten/Pages

Gegenstand/Subject	Werkstoff/Material	Besteller/Customer	Type/Type	Zeichnungs-Nr./Drawing No.	Abnahme/Class. Soc
6 crankshafts	50CrMo4+Ni+V +QT	Caterpillar Motoren	6 M 32-500	1.94.7-25.10.01-04 "d/d"	NV ESN N.V.
Ident-Nr./Ident No.	**Auftrags-Nr./Reference No.**	**Rohteil-Hersteller/Forging manufacturer**	**Schmiedeart/Forging method**	**MT-Prüfung/MPI test**	LR STU QA - 023
14560/16	4010922	Alfing	HfH-4101	entsprechend Vorgabe	
Zugprobe/Tensile test piece	**Kerbschlagprobe/Impact test piece**	**Biegeprobe/Bending test piece**	**Gewicht in kg/Weight in kg**	**US-Prüfung/US test**	R QA023
14Ø×5d (RT)	ISO-V (RT)		roh forged 7040 fertig machined 4400	ja/~~nein~~ yes/~~no~~ 26.06.07	

Abnahmevorschrift/Inspection spec.: NV

Kundenvorschrift/Customer spec.: gem. Zeichnung

Stück-Nr. Serial No.	Chargen-Nr. Heat No.	Proben-Test Ort place	Proben-Test Lage position	~~R_{p}/Dehngrenze Offset yield stress~~ R_{eH}/Streckgrenze Yield point kN	N/mm²	R_m/Zugfestigkeit Tensile strength kN	N/mm²	A/Dehnung Elongation l mm	3,54 d %	5 d %	Z/Einschnürung Reduction of area Ø mm	%	A_v/Kerbschlagarbeit in Joule Impact energy 1	2	3	x̄	HB B.H.N.		Biegeprobe Bending test	Entscheid Decision
					min. 650		min. 860			min. 13		min. 45				min. 32				
G 993324	71 220	K	L	112,5	731	140,0	910	81,5		16,4	8,8	60,5	90	70	58	73	292			
		F	L	112,0	728	138,5	900	81,8		16,9	8,8	60,5	80	60	50	63	295			
325	•	K	L	112,5	731	138,5	900	82,6		18,0	8,6	62,3	98	84	78	87	295	R 24.08.07		
		F	L	116,5	757	143,0	929	81,6		16,6	8,9	59,6	54	52	48	51	292			
326	•	K	L	112,0	728	139,0	903	82,9		18,4	8,6	62,3	96	84	68	83	292			
		F	L	115,0	747	141,0	916	81,6		16,6	8,8	60,5	68	48	44	53	292			
327	•	K	L	113,0	734	140,0	910	81,9		17,0	8,8	60,5	76	70	60	69	298			Fr
		F	L	113,5	737	139,5	906	82,3		17,6	8,8	60,5	78	60	52	63	292			
328	•	K	L	110,5	718	139,5	906	82,0		17,1	8,8	60,5	80	60	60	67	292			
		F	L	112,0	728	138,5	900	81,7		16,7	8,9	59,6	76	62	50	63	292			
329	•	K	L	115,5	724	139,0	903	82,7		18,1	8,7	61,4	94	80	74	83	292			
		F	L	114,0	741	141,5	919	81,7		16,7	8,9	59,6	64	48	42	51	295			

Charge Heat	Kurzz. Code	Stahlwerk Forge	E[3]	C %	Si %	Mn %	P %	S %	Cr %	Mo %	Ni %	V %	Al%
71 220		BUD	E	0,51	0,28	0,81	0,007	0,012	1,20	0,28	0,47	0,10	0,023

Datum/Date: 26.06.07 Für Alfing/For Alfing: [signature] 26.06.07
Prüfer/Tester: [signature] Für Klassifikationsgesellschaft/For Classification Society: R QA023 N.V.

Prüfnormen/Standards: Zugversuch/Tensile test: DIN EN 10002-1/DIN 50125 Kerbschlagbiegeversuch/Impact test: DIN EN 10045/DIN 50115 Biegeversuch/Bending test: DIN EN ISO 7438 HB-Prüfung/Brinell test: DIN EN ISO 6506

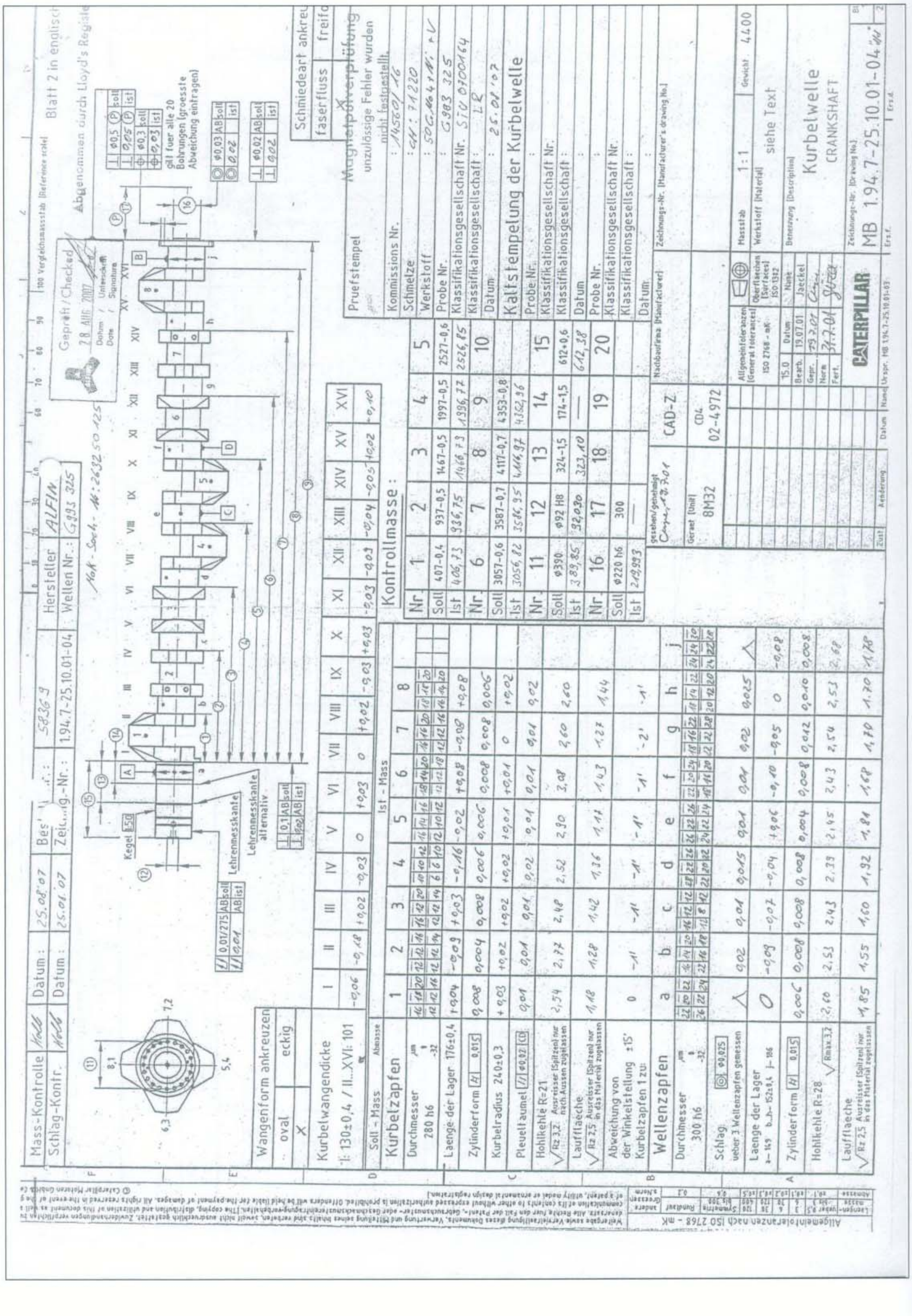
98/69084 - 06.03

Zeichenerklärung:/Key:
[1] Nichtzutreffendes streichen / Delete where not applicable
[3] E = Erschmelzungsart/Steel making process
L = Längs/Longitudinal
Q = Quer/Transverse
K = Kopf/Top end
F = Fuß/Bottom end
Fr = Frei/Free

Mechanical properties of the crankshaft drawn up by the engineering works, Alfing.

▼

The crankshaft dimensions drawn up by the engineering works, Alfing.



▼
A report from Lloyd's Register regarding the reduction gearing of a main engine.
Manufacturer, Renk AG in Rheine, Germany.

Report no: DTM 0562069/1
Page 1 of 3

Lloyd's Register

Report on Main Engine Reduction Gearing

Ship's Name

IMO number

Port
Dortmund

Place of Survey, if different from above
Rheine

Office
Dortmund

Number of visits in shops
2

First date
29 June 2006

Last date
26 September 2006

Ship built by

Yard number
NB 101

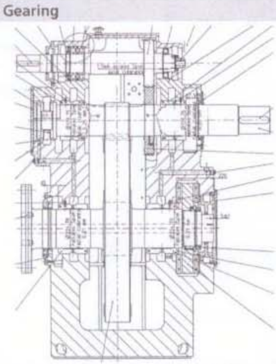
Gearing made by
Renk AG, Rheine / Germany

Gear number
857936

Year and month
2006/09

Number of sets and description of gearing, including reversing arrangements, and clutches, if any. State type of bearings.
Marine Reduction Gearbox Type RSV-630 with PTO. Center distance between the input/output shafts 630 mm. Input shafts and output shafts mounted on sleeve bearings and the PTO shaft are running in roller bearing.

Diagrammatic Sketch Showing Arrangement of Gearing



Approved maximum total kW each set
1520 KW

Corresponding rpm of main wheel
169,5 rpm

State ice class notation, if applicable
no

If single helical, state type and position of gear thrust bearing
see sketch or approved drawing of gear

Have all pinions, flexible coupling sleeves and service attachments been dynamically balanced?
no

State if wheels dynamically or statically balanced
PTO shaft dynamically balanced

Are bodies of cast or welded construction? State material.
solid forged

How are bodies connected to shafts?
shrink fitted

If rims separate, state how secured
no

If rims shrunk, has shrinkage allowance been checked and found as approved?
n.a.

Number and diameter of radial or dowels fitted
n.a.

Which gears were cut under controlled temperature conditions
Yes

How were teeth cut
ground

Machine used for finishing process

Maker's name
Niles / Pfauter Kapp

Serial number

If teeth surface hardened, state method
case hardened

Is gearcase of cast or welded construction
cast

Note: The particulars in this report are to be given as fully and clearly as possible. Where the answer is "NO" or "NONE", say so. Ticks and other signs of doubtful meaning are not to be used.

Form 2016 (2005.04)

Extra information of the same gear box.

Report no: **DTM 0562069/1**
Page 2 of 3

If welded construction, has it been stress relieved

Have means been supplied for verifying that gearcase is free from distortion when secured in ship?

n.a.

Has gearing been run light loaded in shop, and tooth contact found satisfactory? State max. rpm reached

light load run main stage at 300 KW, PTO at 300 KW

What is the backlash? (State whether measured circumferentially or normal to teeth and if no clearance bearings)

main item 5040/5020: 0,30 mm; PTO item 5060/5100: 0,57 mm (pitch circle)

	Primary H.P.	Primary L.P.	Secondary H.P.	Secondary L.P.
Pinions				
Max. kW to be delivered to primary pinions	**item 5040: 1520 KW**		**item 5100: 372 KW**	
Corresponding rpm.	**1000 rpm**		**1507,7 rpm**	
Number of teeth	**20**		**65**	
Total width of gearface parallel to axis	**150 mm**		**55 mm**	
Width of gap between helices	--		--	
Approved pinion material tensile strength	**1080 N/mm²**		**1080 N/mm²**	
Actual pinion material tensile strength	**1334 N/mm²**		**1287 N/mm²**	
Quillshafts				
Approved tensile strength	--		--	
Actual tensile strength	--		--	
Flexible Couplings				
Type of coupling	--		--	
Approved driving member tensile strength	--		--	
Actual driving member tensile strength	--		--	
Approved driven member tensile strength	--		--	
Actual driven member tensile strength	--		--	
Material of sleeve	--		--	
Approved sleeve tensile strength	--		--	
Actual sleeve tensile strength	--		--	

Wheels	Primary H.P.	Primary L.P.	Main Wheel
Number of teeth	**item 5020: 118**	**item 5060: 98**	
Approved rim material tensile strength	**1080 N/mm²**	**1080 N/mm²**	
Actual rim material tensile strength	**1208 N/mm²**	**1283 N/mm²**	
Approved shaft material tensile strength	**item 5010: 600 N/mm²**	--	
Actual shaft material tensile strength	**715 N/mm²**	--	

Form 2016 (2005.04)

▼
Representatives of the manufacturer and the surveyor from Lloyd's Register must sign the report.

Report no: DTM 0562069/1
Page 3 of 3

Declaration to be Signed by Gearmakers

To the best of our knowledge this reduction gearing has been soundly constructed in conformity with the Rules and Regulations, and the foregoing particulars of reduction gearing are correct.

Signature	Name in BLOCK CAPITALS	Date
	f. Dirk Kremkow	19 October 2006

Date and port of approval plans
LLoyd's Register EMEA, Hamburg, Ref.: HMD 16496-06 Issue no. 0 dated 02 January 2006

A previous similar case was for (name)

Port and Report Number | Gear number

Declaration to be Completed and Signed by Surveyor

The reduction gearing reported above has been built under Special Survey in accordance with the Rules, approved plans and Secretary's letters. The materials and workmanship are good; the spare gear required by the Rules has been supplied and the gearing is eligible, in my opinion, to be fitted in a classed ship. A detailed list of certificates is attached to the report stating item, manufacturer, port of issue and certificate number and identification marking.

Issue Date: 03 November 2006

Issue Location: Dortmund

Lloyd's Register EMEA
K. Allermann
Business Centre Dortmund
Lloyd's Register

K. Allermann
Surveyor to Lloyd's Register EMEA
A member of the Lloyd's Register Group

Declaration to be Completed and Signed by Surveyor at Port of Installation

The above reduction gearing has been fitted on board the MV/SS at in a fit and proper manner and found satisfactory when tested on under full power conditions for hours and when examined subsequently.

Date:

Location:

Surveyor to

A member of the Lloyd's Register Group

Form 2016 (2005.04)

▼
The used material properties of all the parts of a similar gear box are tested.
In this case, a pinion shaft in gear seating.
The composition of the material and the mechanical properties are of utmost importance.
The heat treatment is also important, in this case case-hardening.

Prüfbescheinigung nach EN 10204 - Werkstoffprüfung
Inspection document acc. to EN 10204 - Material testing

RENK

Vorgangsdaten / Process data

RENK Auftrags-Nr. / RENK order no.	Produkt-Type / Product type	Herstellerzeichen / Manufacturer's sign	Ext. Abn.-Organisation / Ext. Insp. Organisation	Abnahme-Zertifikat-Nr. / Certificate no.
857936	RSV-630	RENK	LRS	
Positions-Nr. / Position no.	**Material-Nr. / Material no.**	**Zeichnungs-Nr. / Drawing no.**	**Werkstoff-Norm / Material standard**	**Werkstoff / Material**
5040	W90662796253	0796253/0	TN70484	18CrNiMo7-6+A
Anzahl / Quantity	**Benennung/Denomination**		**Wärmebehandlung/Heat treatment**	
1	RITZELWELLE M.RADSITZ / PINION SHAFT WITH GEAR SEATING		Einsatzhärten / case-hardening	
Anforderungen / Specifications	**Rohabmessung / Rough dimension**	**Gewicht [kg] / Weight**	**Stahllieferant / Steel supplier**	**Stahlerzeuger / Steel maker**
TN70481	Ø 230	241,2	BGH- Siegen	BGH- Siegen

Werkszeugnis 2.2 nach EN 10204 über chemische Zusammensetzung
Test report 2.2 acc. to EN 10204 - Chemical composition

Schmelzen-Nr. / Heat no. 760305		C %	Si %	Mn %	P %	S %	Cr %	Mo %	Ni %	Al %	V %
Sollwerte / Theoretical values	min.	0,15	-	0,40	-	-	1,50	0,25	1,40	-	-
	max.	0,19	0,30	0,60	0,010	0,005	1,80	0,35	1,70	0,040	-
Istwerte[1] / Actual values[1]		**0,18**	**0,24**	**0,50**	**0,006**	**0,001**	**1,66**	**0,33**	**1,56**	**0,033**	-

[1] Chemische Zusammensetzung gemäß Lieferantenangaben / Chemical composition as indicated by the supplier

Abnahmeprüfzeugnis 3.1 nach EN 10204 über mechanisch-technologische Eigenschaften
Inspection certificate 3.1 acc. to EN 10204 - Mechanical-technological properties

		Zugversuch/Tensile test gemäß/acc. to EN 10002-1				Kerbschlagbiegeversuch/Impact test gemäß/acc. to EN 10045-1			
Proben-Nr.[2] / Specimen no.[2]	1866	Streckgrenze / Yield point	Zugfestigkeit / Tensile strenght	Bruchdehnung / Elongation	Einschnürung / Reduction of area	Kerbschlagarbeit / Impact work			Prüf-temp. / Temp.
Lage[3] / Location.[3]	L	$R_{p\,0,2}$ N/mm²	R_m N/mm²	(l = 5 d) %	%	ISO-V Joule			°C
Sollwerte / Theoretical values		min. 785	1080-1500	min. 11	min. 40	min. 40			RT
Istwerte / Actual values		1032	1334	13,4	60,8	95	91	73	RT

[2] Art der Probe: B nach DIN 50125 / Specimen type: B acc. to DIN 50125
[3] L = längs/longitudinal; T = tangential/tangential; Q = quer/transverse

Bemerkungen:
Remarks:

Prüfergebnisse/Test results:	Die gestellten Anforderungen sind erfüllt / The requirements have been complied with.			
20.01.2006	Ostholthoff	20.01.2006	Weiffenbach	
Prüfdatum / Date of test	RENK Prüfer / RENK operator	Datum / Date	RENK Abnahmebeauftragter / RENK Inspection representative	Externer Abnehmer / External Inspector

RENK AG Werk Rheine Qualitätsmanagement

Diese elektronisch erstellte Prüfbescheinigung ist ohne Unterschrift gültig / This electronically issued inspection document is valid without signature.

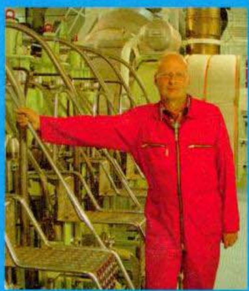
RENK Aktiengesellschaft · Werk Rheine
Qualitätsmanagement / Quality Management
Postfach 1953 · 48409 Rheine
Tel.: +49 5971 790-0 · Fax: +49 5971 790-208

DIN ISO 9001 DNV Certified Company Seit 1989

Werkstoff_de_en / 2005-06 Fo.1020







The author.

Kees Kuiken began his career in 1963 by enrolling as a marine engineering student at the 'Hogere Zeevaartschool' at Terschelling, The Netherlands.
He started his sailing career with the United Dutch Shipping Company (Verenigde Nederlandse Scheepvaartmaatschappij, the V.N.S.).
In 1978, he went on to become a lecturer in marine engineering at the Hogere Zeevaartschool in Delfzijl, Groningen and also worked in the machine construction and operational technique departments.
His great passion was the establishment of a large, modern practical lab for intermediate and higher education as well as trade and industry.
In 1995, he started the foundation of the European Training Centre for Machine technology, the E.T.M., an educational foundation.
In 2000, he left regular teaching and established Target Global Energy Training.
This enterprise provides worldwide training sessions in the field of diesel- and gas engines, gas and steam turbines, compressors and cogeneration.
Furthermore, Target addresses the solution of technical problems and publishes books and manuals.
All training programs are customised and are provided on location.

ISBN 978-90-79104-02-4

9 789079 104024

This book can be ordered directly from
Target Global Energy Training.
Email: targettraining@planet.nl

For additional information:
www.targettrainingcentre.nl
www.dieselengines.nl
Tel: +31 50 404 1554
Fax: +31 50 404 2658