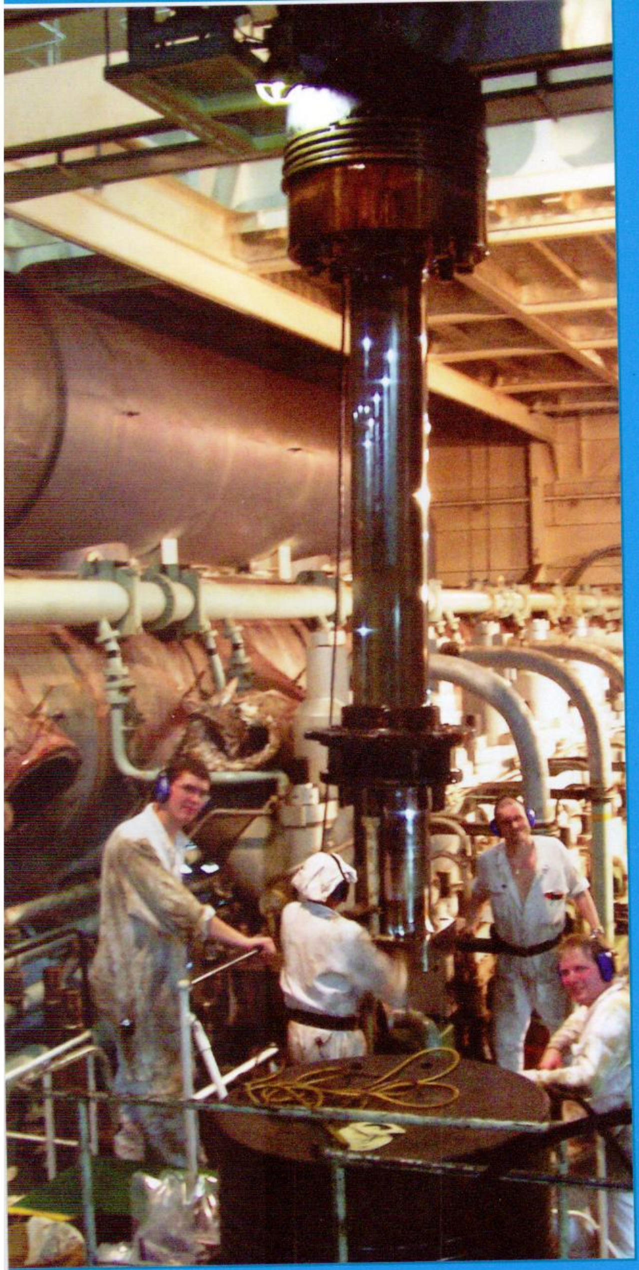
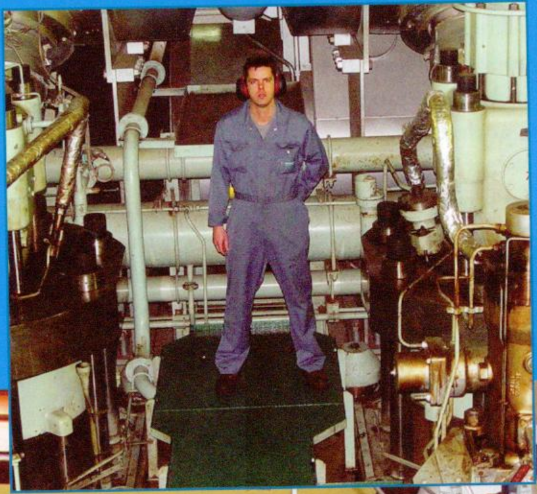
DIESEL ENGINES

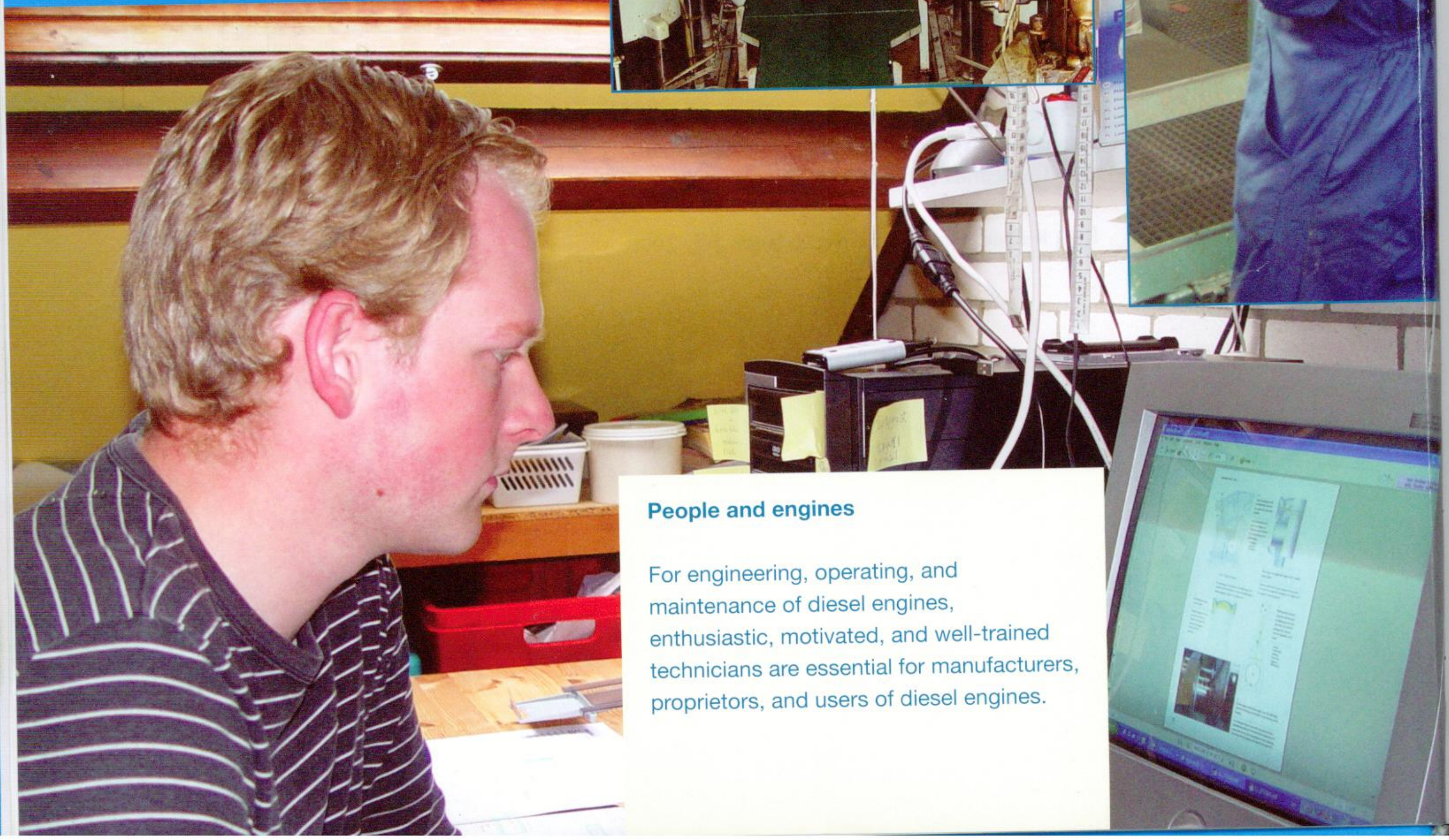
for **ship propulsion** and **power plants**

FROM 0 TO 100,000 kW

Kees Kuiken







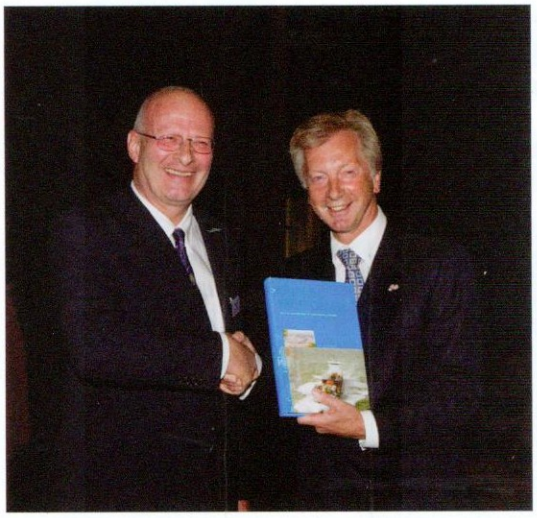
People and engines

For engineering, operating, and maintenance of diesel engines, enthusiastic, motivated, and well-trained technicians are essential for manufacturers, proprietors, and users of diesel engines.

> 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.



▲
At the special general meeting of the 60th anniversary of the VIV, de Vereniging van Importeurs van Verbrandingsmotoren (Association of Importers of Combustion Engines) at the 'Theater aan het Vrijthof' in Maastricht, author Kees Kuiken presents the first proof print to the chairman of the Association FME-CWM, Mr. Jan Kamminga.

FME-CWM is the employers' organization and trade association for the technological and industrial sector. The activities in the sector cover engineering, manufacturing, trade, industrial maintenance, and industrial automation. Some 2,750 organizations (metal, plastics, electronics and electro-technology), employing some 260,000 people, are members of FME.

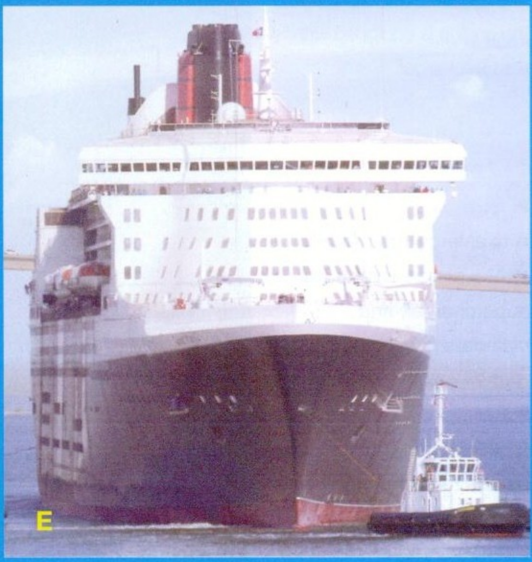
> Content

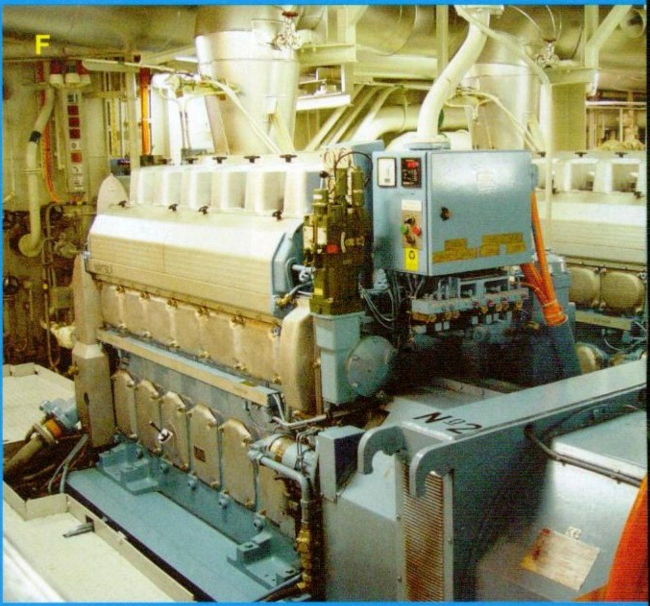
>CH1

The use of industrial diesel engines

Diesel engines are used for a variety of applications.

A Propulsion and generator sets for yachts.
B Generating electricity in the tropics.
C Propulsion of sea going ships.
D Propulsion of inland shipping.
E Generating the electric energy required on large passenger ships.
F Generating electrical energy on board ships.





1.1 Introduction

Throughout the last century the internal combustion engine has become increasingly important in a society that now relies heavily on machinery. The engine is the elementary chain in, for instance, the transportation of goods and persons by road and water, propelling various machines and generating electrical energy. The use of combustion engine equipment such as chain saws, water pumps, concrete mixers and lawn mowers has seen an explosive increase.

Modern agriculture is completely dependent on internal combustion engines for propulsion of tractors, combines, and other farming equipment.

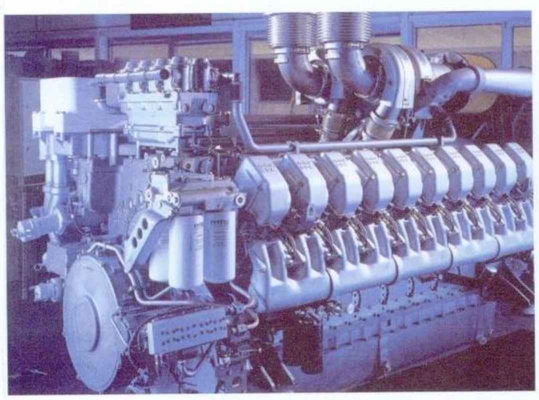
The industrial diesel engines mentioned in this book are almost all used for propelling an enormous diversity of ships. Also in generating electricity the diesel power plant plays an important role.
In the internal combustion engine one distinguishes two principles: namely: the Otto process and the Diesel process.

▶
The largest diesel engines are used in the navigation industry; here is a picture of part of a twelve-cylinder two-stroke crosshead engine on a container ship.

Shaft power 68,000 kW. The cylinders are in-line and the engine is equipped with a supercharger.

▶
A twelve-cylinder four-stroke piston engine with supercharger, and cylinders in V-position.

▶▶
A six-cylinder in-line engine for the generation of electrical energy on board a ship.



◀
A harbour tow boat needs power of several thousands kilowatts to tow large sea vessels.

◄ Container feeders with a carrier capacity of 9000 tons have an engine capacity of approximately 6000 kilowatt at 18 nautical miles per hour.

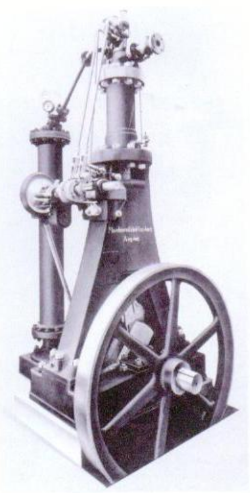
◄ A common diesel power plant in the tropics with diesel engine driven electric generators.

1.2 Otto-process

In 1876, after years of experimentation Nicolaas August Otto develops the first four-stroke engine (four strokes of the piston - two rotations of the crank shaft) which compresses a mixture of air and fuel. A spark plug provides an ignition spark at the right moment, which ignites and then combusts the mixture.
This process will not be further discussed in this book.

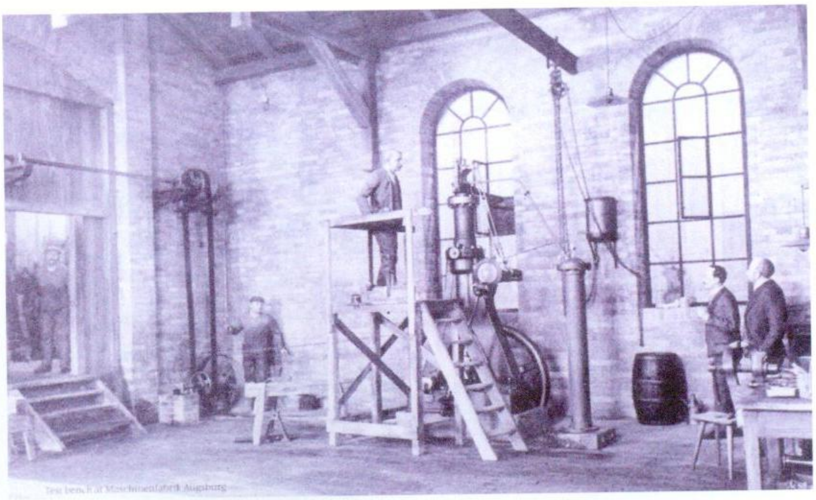
1.3 Diesel-process

In 1897 Rudolf Diesel markets the first diesel engine. In these diesel engines the air is compressed to such a degree that the high end temperature of the air effects a very swift ignition and combustion of the injected fuel. The efficiency of this diesel process is higher than that of the Otto-process, thus resulting in a huge expansion of the diesel engine industry.
Total efficiency of the internal combustion engine = the ratio of the produced shaft power and the supplied fuel power. This ranges from between 25% and 50%, depending on the engine size.

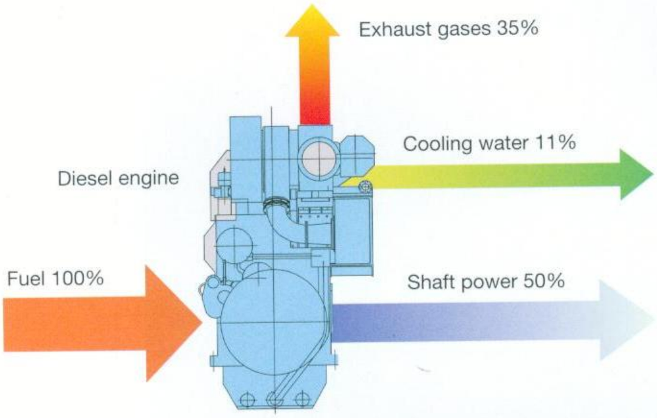


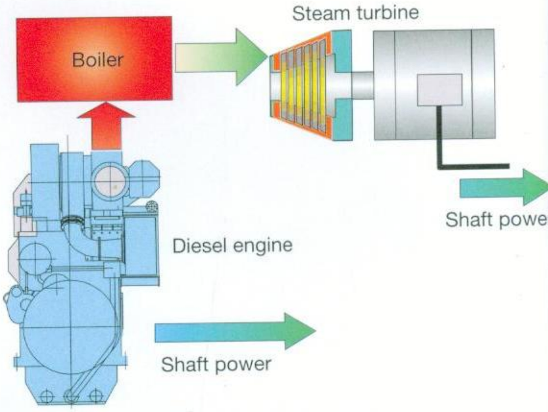
▶ Rudolf Diesel's first diesel engine built in Augsburg, Germany.

This engine never ran independently and looked more like a canon barrel than an internal combustion engine. The design pressure was 150 bar! Rudolf Diesel nearly got killed when one of his testmodels exploded.



▲ Rudolf Diesel's first diesel engine at the test bed of the Augsburg Machine factory; MAN.





▲ Diesel engine losses.

Left figure: The fuel supply of a line engine. This is always set at 100%.
Right figure: Shaft power of 50%. So half of the supplied energy(fuel) is converted into mechanical power on the crankshaft.
The remainder of the supplied energy is waste, such as cooling water- and exhaust gases losses.

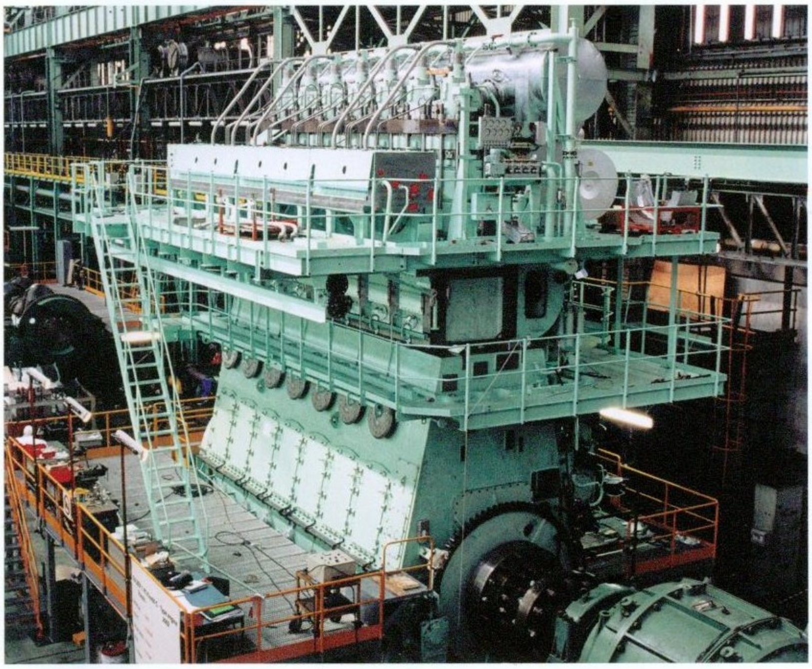
Far right figure: in order to use at least part of the 35% exhaust gas losses, steam is generated in an exhaust gas boiler. This steam drives a steam turbine which subsequently generates electricity.

1.4 The use of Otto-engines

They are often used in:
- (Hand) tools;
- garden equipment;
- automotive industry
- outboard engine
- gas engines;
- small aviation.

1.5 The use of Diesel-engines

They often used in:
- ship propulsion;
- diesel power plants;
- agriculture;
- back up generator sets
- lorries;
- earth moving machines;
- military vehicles such as tanks.



◄
A seven-cylinder diesel engine at the test bed; a Wärtsilä Sulzer RTA 7–96-C, a typical propulsion engine for large, high-speed container ships.

Generally, diesel engines are used more often when the operation hours increase and fuel costs become important. The required shaft power is often much higher than in Otto engines.

1.6 Properties of both principles

	Otto-engine	Diesel engine
Principle	mixture compression	air compression
Ignition	with spark plug ignition spark	by hot air combustion
Fuels	gases and petrol	diesel oil, heavy oil and bio-fuels
Motor weight	light	heavier
Efficiency	low: 15–45%	high: 25–55%
Capacity	maximum 10 MW (gas engines)	maximum 100 MW (ship engines)

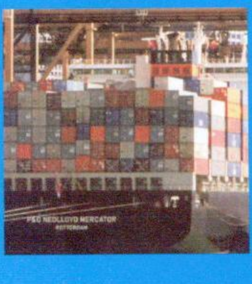
◄
The properties of both principles.

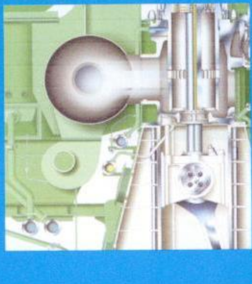
In this book we will only discuss diesel engines. They are very often used in navigation and industry. There is a book on gas engines in both Dutch and English available.

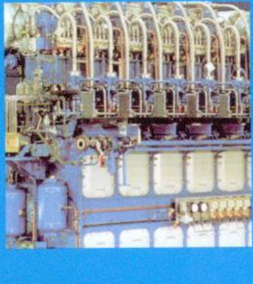
►
A small three-cylinder Volvo Penta diesel engine for yachts.

>CH 2

Classification of diesel engines







The classification of engines into four engine categories.

Category I: High-speed four-stroke engines with up to 100kW shaft power.
The number of revolutions (RPM) normally lies between 1500 and 3000 revolutions per minute (rpm).
The fuel is diesel oil.

Category II: High-speed four-stroke engines with a shaft power between 100 and 5000 kW.
The RPM normally lies between 960 and 2100 revolutions per minute.
The fuel is diesel oil.

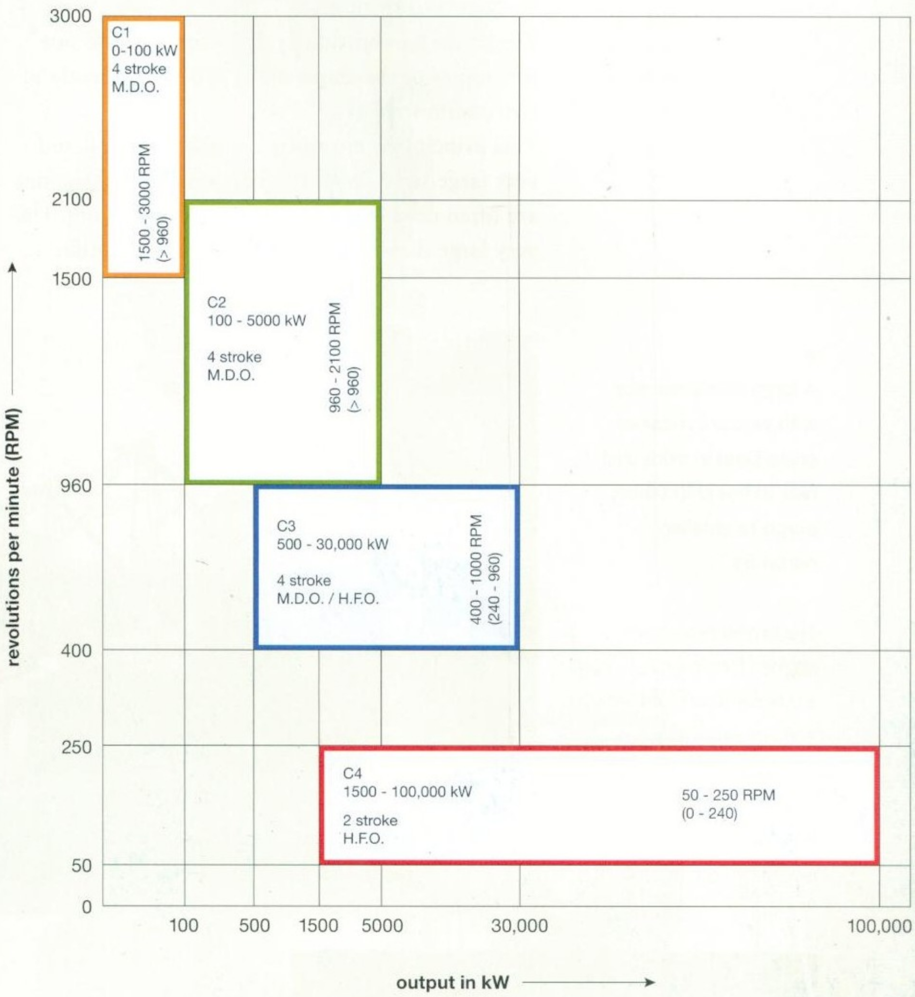
Category III: Medium-speed four-stroke engines with a shaft power between 500 and 30,000 kW.
The RPM normally lies between 400 and 1000 revolutions per minute.
The fuel is heavy oil.

Category IV: Low-speed two-stroke crosshead engines with a shaft power of 1500 up to 100,000 kW and RPM between 50 and 250 revolutions per minute.
The fuel is heavy oil.

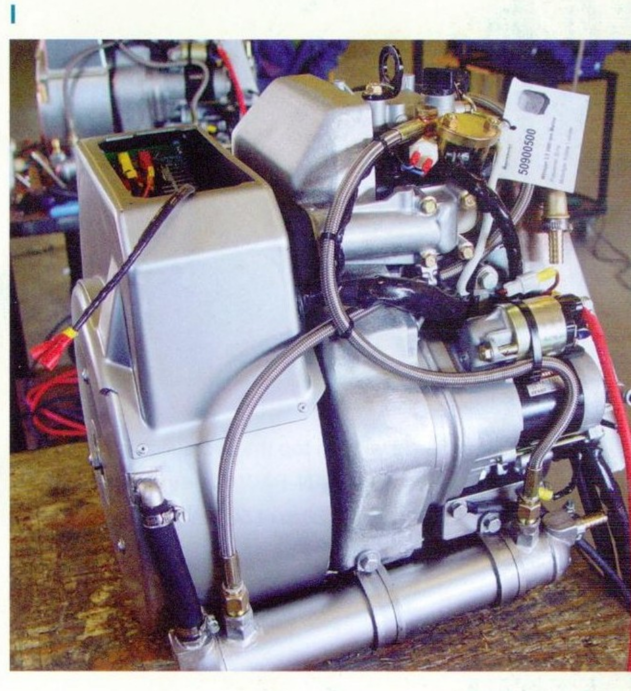
Different categories could of course overlap in some cases.

III IV

Classification of engines



II I



2.1 Introduction

Diesel engines are categorised by working principle, design, number of revolutions, power output, fuel used, usage and other characteristics.

2.2 Working principle

Two-stroke principle

The entire combustion cycle is completed in one revolution of the crank shaft, and consequently in two piston strokes.
This principle is most often applied to small and very large sized diesel engines. Small diesel engines are often used to drive a generator or a pump. The very large diesel engines are without exception crosshead engines which are predominantly used for ship propulsion and diesel power plants.

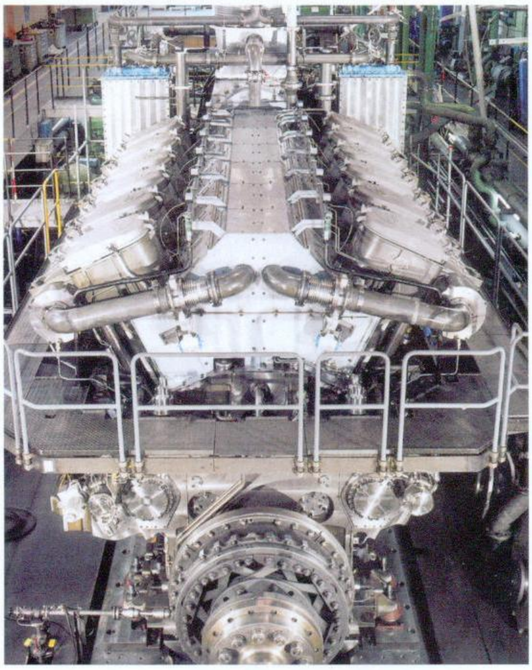
Four-stroke principle

The entire combustion cycle is completed in two revolutions of the crank shaft and consequently in four piston strokes. This principle is generally applied in high-speed and medium-speed diesel engines. Smaller propulsion engines, engines for driving generator sets, back-up generators, and also larger propulsion engines of up to ± 30MW shaft power all work according to the four-stroke principle.
Concerning industrial diesel engines, the number of four-stroke engines that are built annually far outnumber those of two-stroke engines.

▶
A large container ship with feeder container ships both in front and rear of the ship taking cargo to smaller harbours.

The largest two-stroke engines for these ships have a cylinder bore of 96, 98 and 108 cm with a fourteen cylinder in-line arrangement and a shaft power of 100,000 kW at a hundred revolutions per minute.

▲
Cylinder heads with on top the hydraulically driven exhaust valves of a MAN–B&W two-stroke crosshead engine and a cylinder bore of 500 millimetres.



▶
Installation of four-stroke trunk piston V-engine in Augsburg, Germany, in the MAN–B&W factories.

2.3 Design

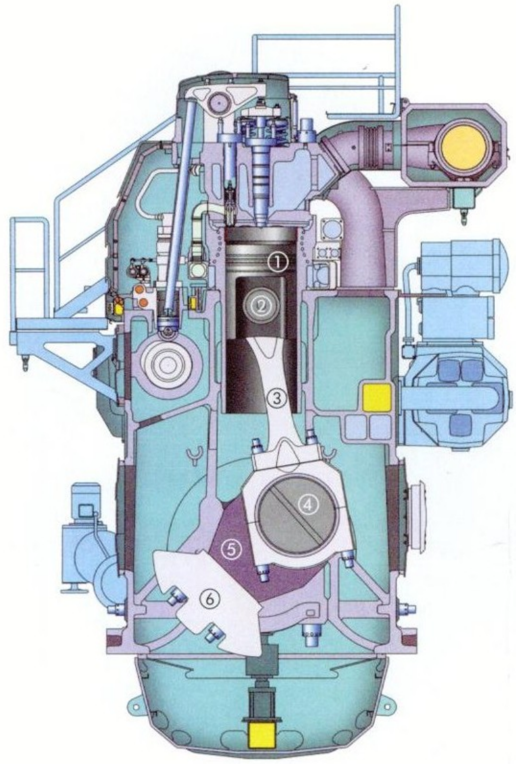
Differences between trunk piston engines and crosshead engines.

Trunk piston engines

In these four-stroke engines the connecting rod is hinged on the piston by a gudgeon pin. The engine is low in comparison to crosshead engines. The pistons have to absorb the lateral forces and transfer them to the cylinder liner and engine frame.

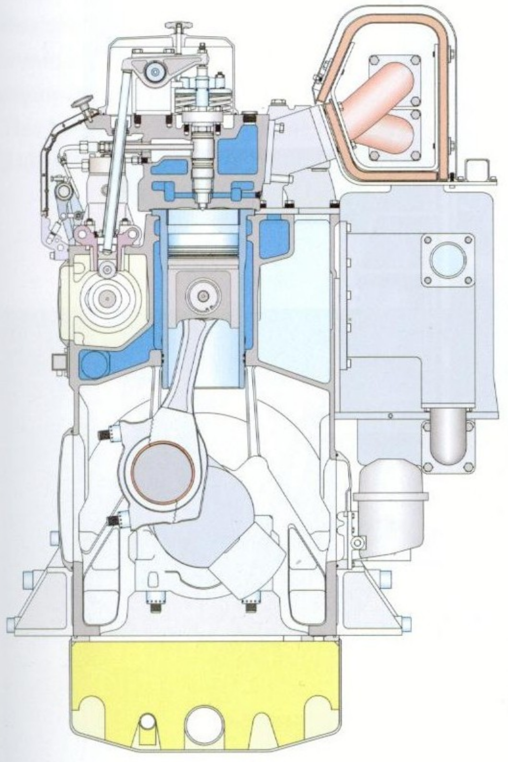
Crosshead engines

In these engines the piston rod is bolted to the piston. The crosshead is situated below the piston rod with which it is hinged to the shaft. The lateral forces of the crankshaft mechanism are transferred to guide shoes which are fixed rigidly to the motor frame via crosshead guides.

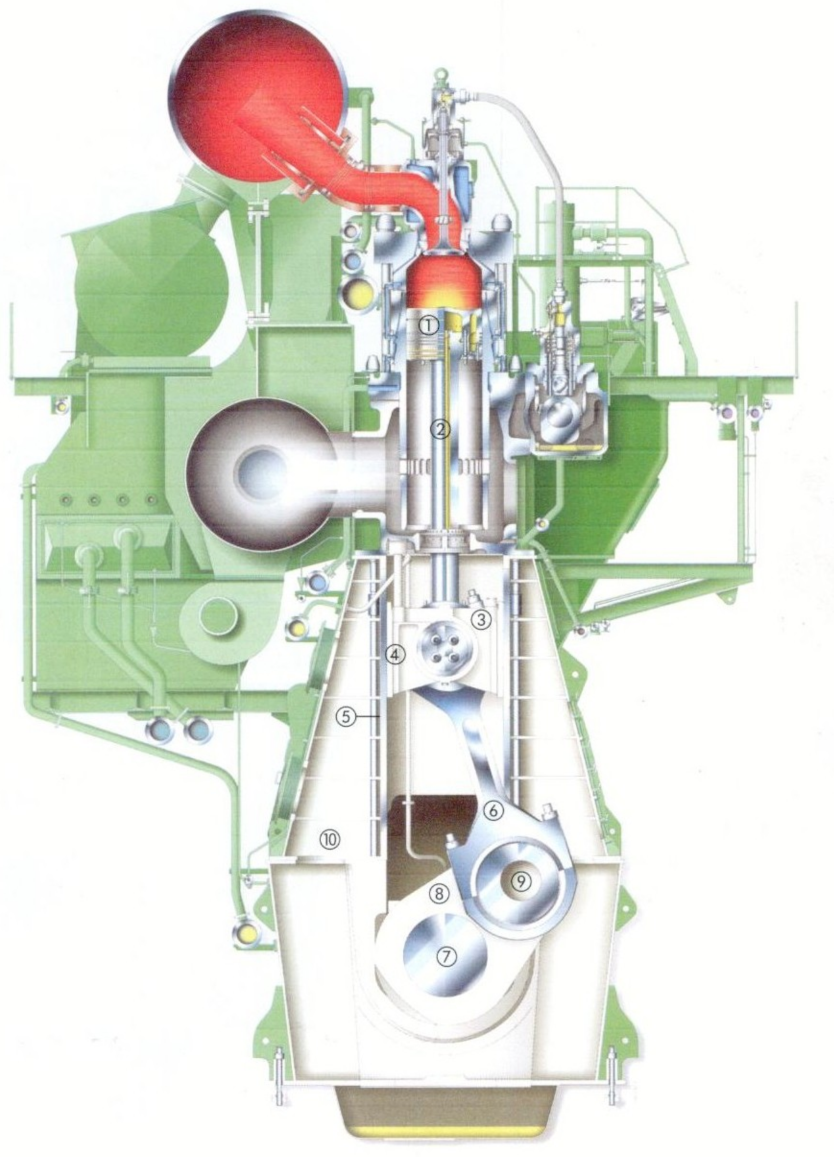


◄
Cross view of an in-line engine; the cylinders are positioned upright and in a straight row. This is a typical trunk piston engine.

1 *piston*
2 *piston pin*
3 *connecting rod*
4 *crank pin*
5 *crank web*
6 *counter weight*

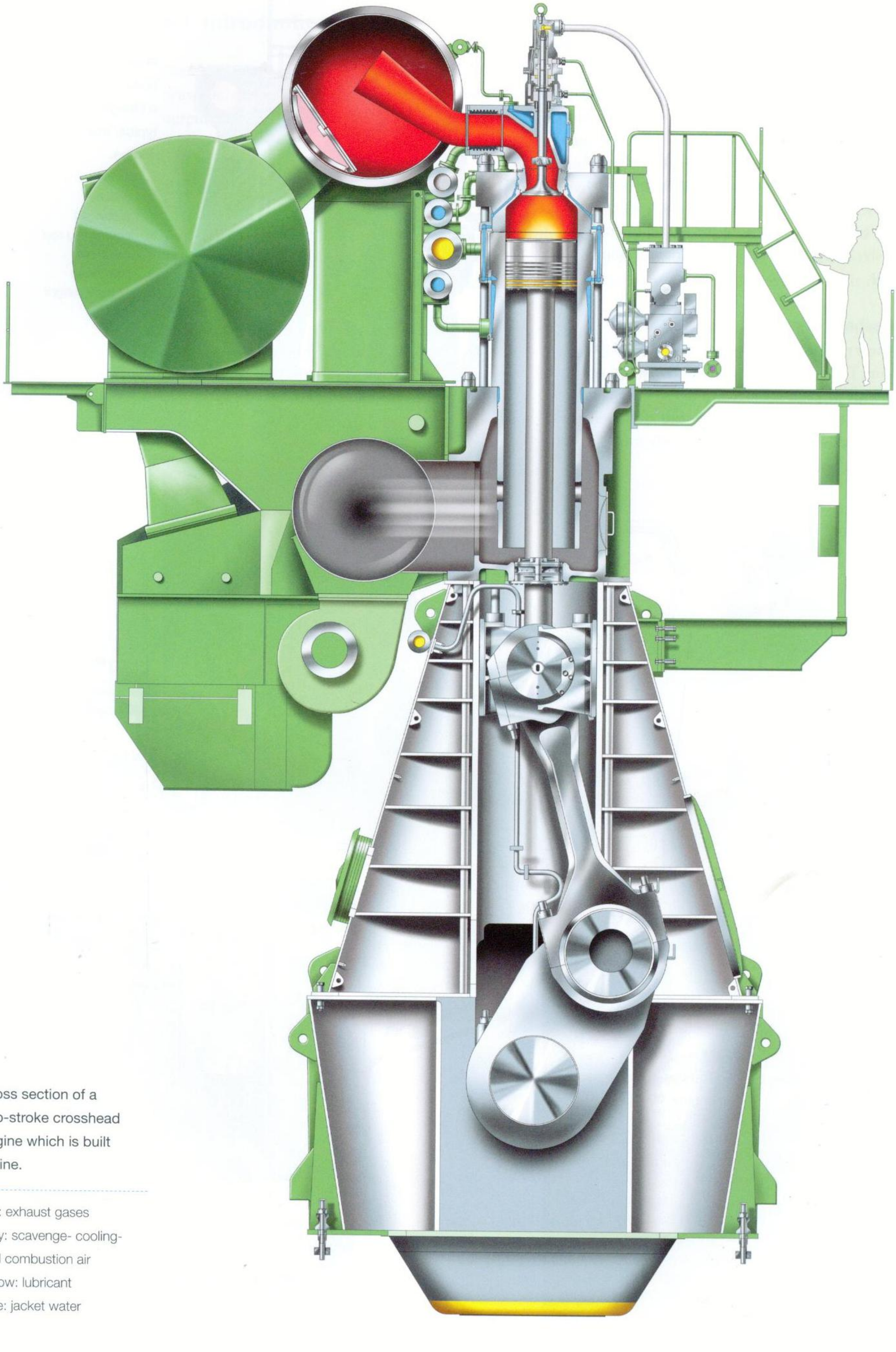


▲
Cross view of a trunk piston engine.



►
The two-stroke crosshead engine.

1 *piston*
2 *fixed piston rod*
3 *crosshead*
4 *guide shoes*
5 *crosshead guides*
6 *connecting rod*
7 *crank shaft*
8 *crank web*
9 *crank pin*
10 *A-frame*



▶
Cross section of a two-stroke crosshead engine which is built in-line.

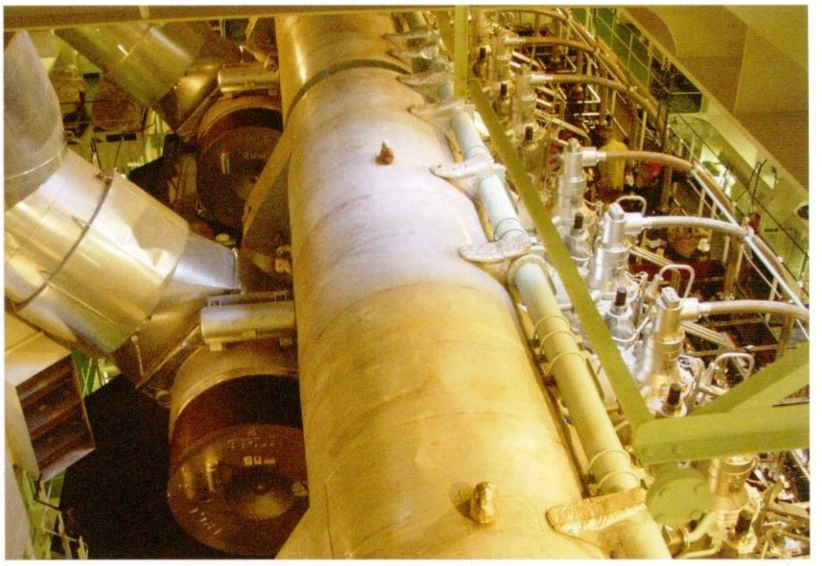
red: exhaust gases
grey: scavenge- cooling- and combustion air
yellow: lubricant
blue: jacket water

2.4 Speed of rotation

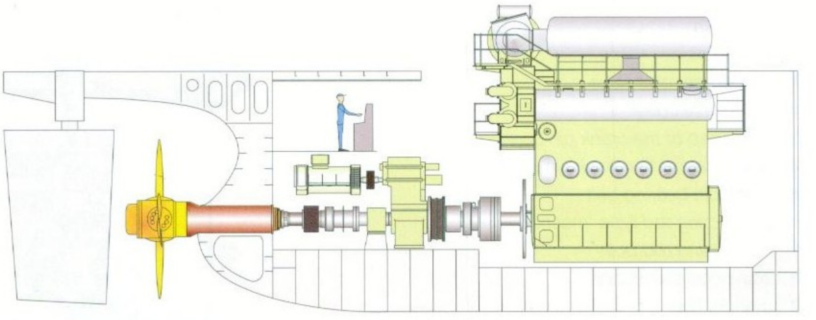
In the engine industry one still often speaks in term of revolutions per minute, while in today's technological industry one indicates the maximum permissible operational speed by means of frequency; the number of revolutions per second.

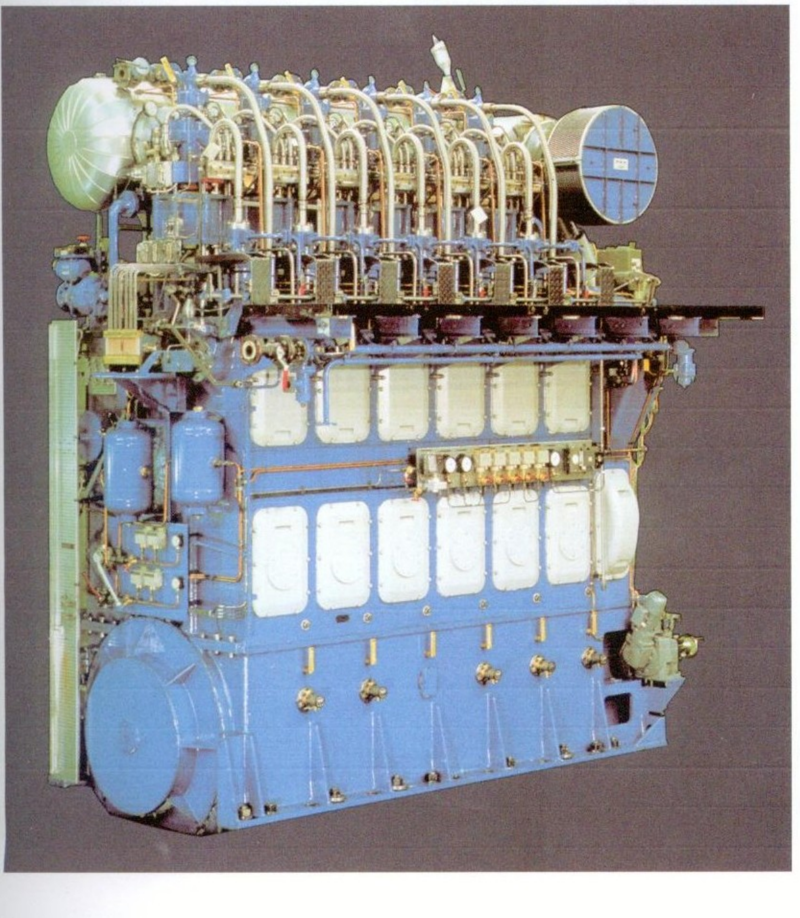
Low-speed engines

Low-speed engines have a maximum of 240 revolutions per minute or a frequency of maximum 4 revolutions per second.
These are always very large sized and heavy crosshead engines which mainly serve to propel large oil tankers, ore tankers and container ships. Some diesel power plants are also equipped with these engines.
Low-speed engines are marketed by just three engines manufacturers; those actually building the engines are also called 'cathedral builders'.
The cylinder bore varies from 260 to 1080 millimetre. For the largest diameter and a fourteen-cylinder version the shaft power constitutes 97,300 kW. The piston stroke for this engine is 2660 millimetre.
This MAN–B&W-diesel engine has a dry weight of 2300 tons and is 28 meter long and 14 meter high.



▲
View from above of part of a two-stroke crosshead engine. The diameter of the exhaust gases lead is approximately 3.5 meters!





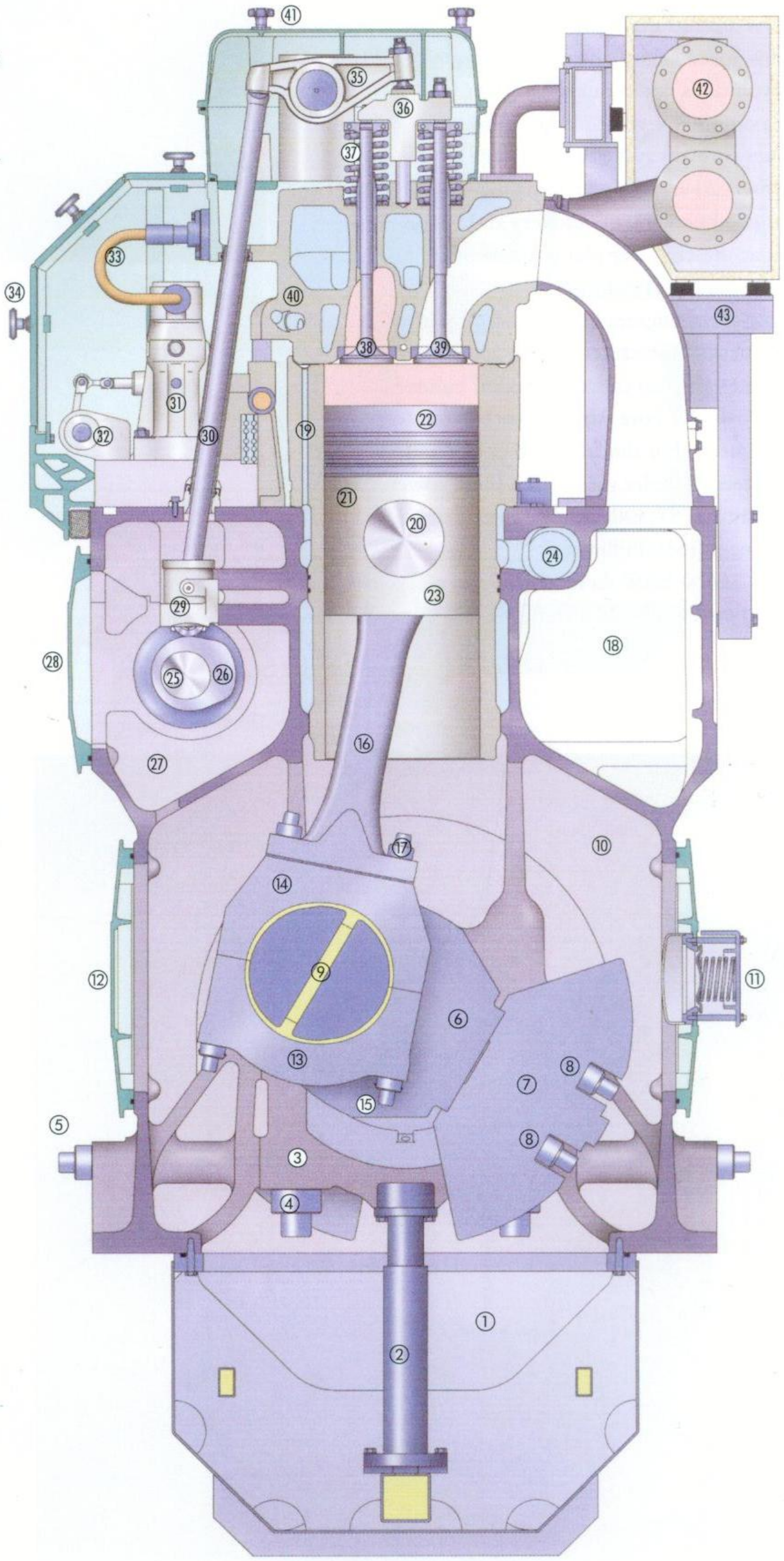
◄
View of a small two-stroke crosshead engine.

▲
A directly driven propeller is the standard arrangement for two-stroke crosshead engines in ship propulsion.

Medium-speed engines

Medium-speed engines have a RPM of between 240 and 960 or a frequency of between 4 and 16 revolutions per second (Hz). They are usually trunk piston engines using the four-stroke principle. However, one does find medium-speed two-stroke engines, and apart from the trunk piston design one occasionally finds small crosshead engines (older existing designs).

The medium-speed engines are employed in very divergent applications; for ship propulsion, generating electricity and driving various kinds of machinery.



▶

Cross view of a four-stroke medium-speed trunk piston engine with a cylinder bore of 380 millimetres, a Wärtsilä 38.

1 *oil sump*
2 *stay bolt/ lubricant supply pipe*
3 *lower bearing cap of the crankshaft bearing*
4 *vertical bolt for the lower bearing cap of the crankshaft bearing*
5 *horizontal bolt for the lower bearing cap of the crankshaft bearing*
6 *crankshaft web*
7 *counter weight*
8 *counter weight bolts*
9 *crank pin*
10 *engine block*
11 *crankcase relief value*
12 *crankcase cover*
13 *lower cap of the crank pin bearing*
14 *upper cap of the crank pin bearing*
15 *crank pin bearing bolt*
16 *connecting rod type 'marine'*
17 *connecting rod bolt*
18 *scavenging air space*
19 *cylinder liner*
20 *gudgeon pin*
21 *piston*
22 *piston gasket with piston rings*
23 *piston skirt*
24 *coolant supply*
25 *cam shaft*
26 *cam*
27 *cam box*
28 *cam shaft cover*
29 *guide sleeve*
30 *push rod*
31 *fuel pump*
32 *fuel rack*
33 *high pressure fuel pipe*
34 *hot box lid*
35 *rocker arm*
36 *spreader/bridge*
37 *valve spring*
38 *exhaust valve*
39 *inlet valve*
40 *cylinder head*
41 *cylinder cover*
42 *exhaust gases lead*
43 *turbo blower frame*

yellow: lubricating oil
light blue: jacket water
brown: fuel
white: air
pink: exhaust gases

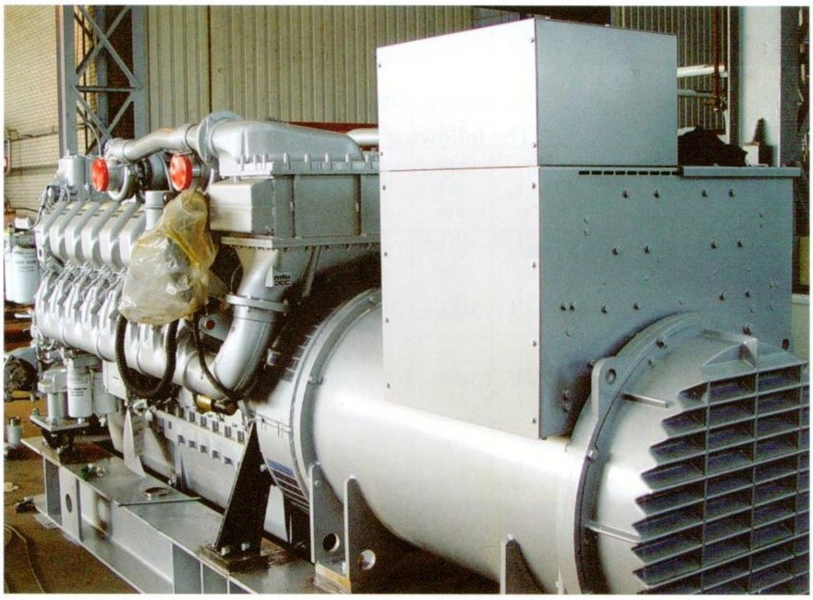
Medium-speed engines are manufactured by a limited number of engine builders. There are a large number of types when taking into account the cylinder bore, the piston stroke and the number of cylinders. The cylinder bore varies from approximately 200 to 640 millimetres. The shaft power varies up to a maximum of about 30,000 kW.

High-speed engines

High-speed engines have a RPM in excess of 960 or a frequency of more than 16 revolutions per second (Hz). Industrial engines often have a RPM of just over 2000.

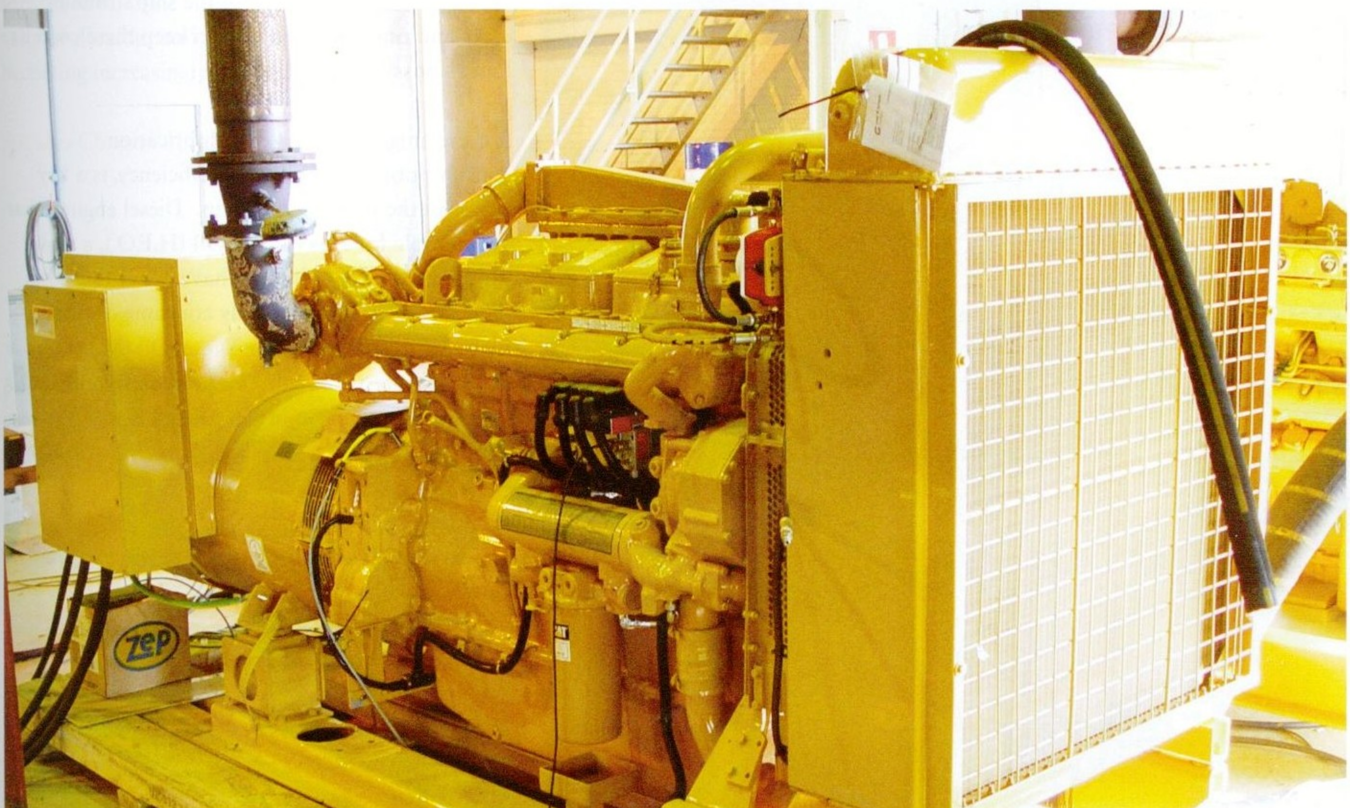
High-speed engines are built by many engine manufacturers and there are a large number of types when taking into account the cylinder bore, the piston stroke and the number of possible cylinders. The cylinder bore varies from 40 millimetres to approximately 200 to 300 millimetres.
The shaft power varies up to a maximum of about 5000 kW.

Of course, it is possible to have engines delivered with a divergent rpm, which are on the fringe of these categories. They are then usually placed into a higher RPM category.



▲
A high-speed four-stroke V-engine for a generator.

▼
A high-speed four-stroke in-line engine with radiator cooling for a generator.



2.5 Power output or shaft power

Category subdivisions

The following categories are used:

I 0 – 100 kW, fuel M.D.O., four-stroke high-speed

II 100 – 5,000 kW, fuel M.D.O., four-stroke high-speed

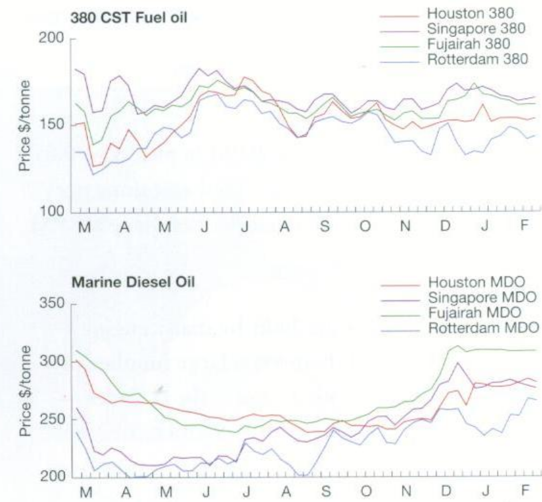
III 500 – 30,000 kW, fuel H.F.O., four-stroke medium-speed

IV 1500 – 100,000 kW, fuel H.F.O., two-stroke low-speed

These subdivisions are discussed elaborately in Chapter 5, Standard figures of various types of diesel engines.

2.6 Fuel used

Use of a particular type of fuel is related to the fuel costs, which is often expressed in dollars per metric ton.

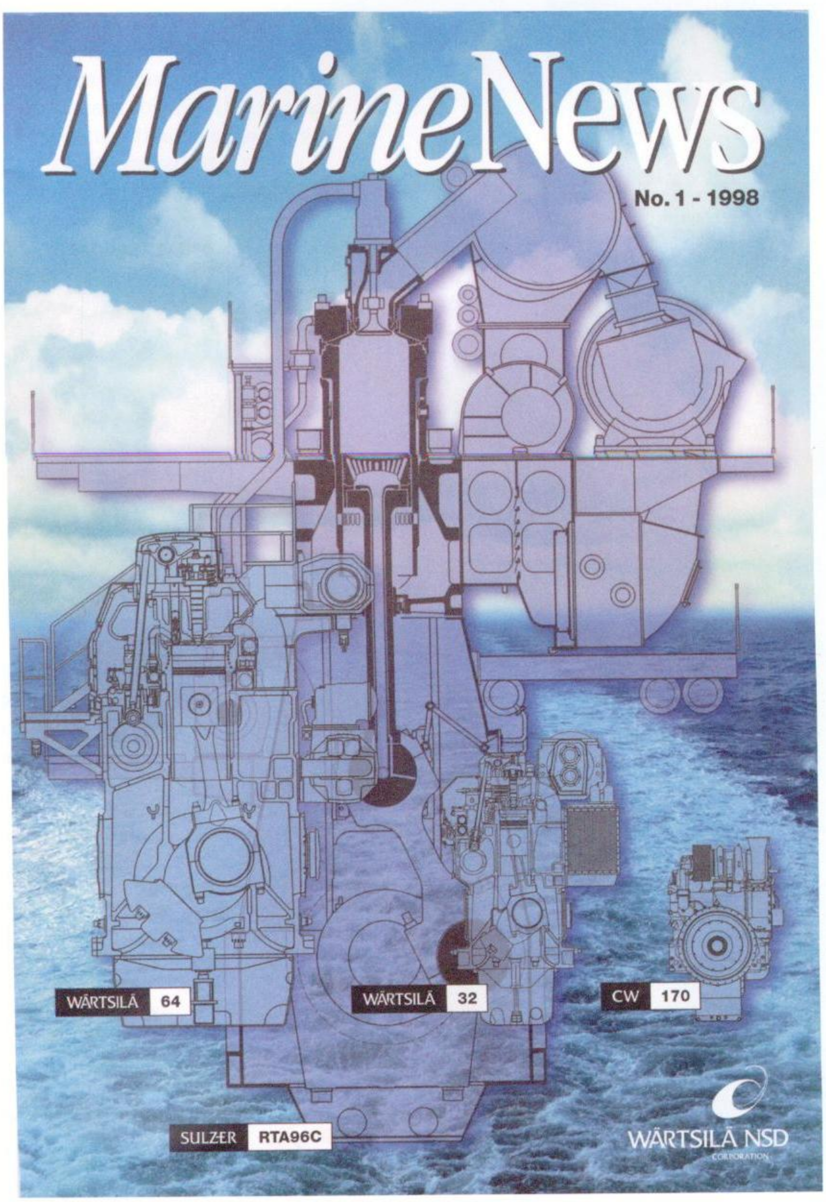


▲

The price of heavy fuel oil is about half of that of diesel oil.

The price of heavy-oil ranged from 150 to 200 dollars per metric ton in March 2007.
In ship propulsion the fuel- and lubrication costs are often more than 50% of the ship's running costs and one therefore tries to keep these costs as low as possible.

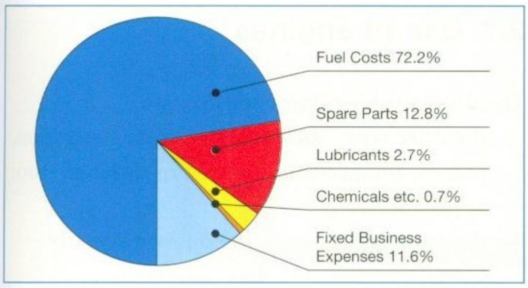
A diesel engine that has low lubrication consumption and a high fuel efficiency, is a key factor in the running of a ship. Diesel engines that are suitable for Heavy Fuel Oil (H.F.O.), a waste product from oil refineries, have lower fuel costs than diesel engines that only consume distillate fuels, for example Marine Diesel Oil (M.D.O.). M.D.O. costs are roughly twice those of H.F.O..



◄

An illustration of the large range of different engine types.

Far right: a high-speed engine with a cylinder bore of 170 millimetres, to its left a medium-speed engine with a cylinder bore of 320 millimetres, far left: a very large medium-speed engine with a cylinder bore of 640 millimetres and in the background the only two-stroke crosshead engine with a cylinder bore of 960 millimetres; they are all Wärtsilä-engines. The only diesel engine-type missing here is a small high-speed engine suitable for, for instance, yachts.



▲
The fuel- and lubricating-oil costs often constitute three-quarter of the running costs for diesel engines.

The fuel costs are 72.2%.
The spare parts 12.8%.
The lubrication costs 2.7%.
Chemicals and other materials 0.7%.
De fixed depreciation on investments etc. 11,6%.

380 IFO		Feb-04				Mar-04			
		2-6	9-13	16-20	23-27	1-5	8-12	15-19	22-26
Rotterdam	d	135	134	142	144	149	147	142	144
Gibraltar	d	160	157	156	156	158	155	149	152
Piraeus	d	148	154	145	146	150	148	144	147
Suez	d	158	156	159	159	161	161	159	164
Fujairah	d	174	168	167	165	164	162	162	162
Durban	w	n/a	n/a	n/a	n/a	n/a	n/a	n/a	n/a
Tokyo	d	201	198	195	196	197	198	197	196
Busan	d	184	181	182	183	186	185	182	179
Hong Kong	d	183	185	184	184	181	182	181	180
Singapore	d	171	172	170	167	166	164	165	166
Los Angeles	w	168	163	161	155	155	155	157	164
Houston	w	153	162	152	154	154	154	153	154
New York	w	163	163	162	158	161	161	157	158
Panama	w	160	160	160	160	161	164	168	168
Santos	d	156	142	144	146	156	156	154	153
Buenos Aires	d	n/a	n/a	n/a	n/a	n/a	n/a	n/a	n/a

180 IFO		Feb-04				Mar-04			
		2-6	9-13	16-20	23-27	1-5	8-12	15-19	22-26
Rotterdam	d	144	144	152	154	159	158	152	154
Gibraltar	d	169	166	164	164	167	164	157	161
Piraeus	d	160	155	159	160	165	164	161	164
Suez	d	164	162	165	166	167	167	166	171
Fujairah	d	181	174	173	172	171	169	169	169
Durban	w	170	167	169	166	165	163	161	159
Tokyo	d	205	202	200	200	201	202	201	200
Busan	d	192	191	192	192	195	194	192	189
Hong Kong	d	186	187	187	188	187	186	185	184
Singapore	d	175	172	175	170	171	168	170	170
Los Angeles	w	193	186	182	169	172	174	173	178
Houston	w	161	162	160	160	160	164	157	159
New York	w	183	182	177	177	179	178	172	173
Panama	w	170	170	169	171	171	174	178	179
Santos	d	159	146	147	150	160	160	158	157
Buenos Aires	d	171	170	167	165	166	164	165	165

MDO		Feb-04				Mar-04			
		1-5	8-12	15-19	22-26	5-9	12-16	19-23	26-30
Rotterdam	d	243	236	242	238	254	253	268	266
Gibraltar	d	284	281	300	306	322	319	328	327
Piraeus	d	264	261	277	282	300	298	307	307
Suez	d	321	321	321	321	325	336	335	335
Fujairah	d	310	310	310	310	309	309	309	309
Durban	w	267	263	274	285	301	299	308	311
Tokyo	d	287	288	291	288	287	292	292	290
Busan	d	332	325	322	322	321	321	319	318
Hong Kong	d	274	274	282	283	282	281	279	273
Singapore	d	277	278	282	282	284	281	279	277
Los Angeles	w	317	320	344	358	345	342	342	340
Houston	w	281	280	278	278	279	282	285	282
New York	w	345	343	335	327	325	330	331	330
Panama	w	317	317	318	321	321	323	326	327
Santos	d	350	343	342	344	347	350	345	344
Buenos Aires	d	317	311	302	306	295	303	303	302

▲
Fuel prices of diesel oil (M.D.O.), heavy oil (H.F.O.) with a viscosity of 180 cSt and heavy oil with a viscosity of 380 cSt.

M.D.O.: Marine diesel Oil;
I.F.O.: Intermediate Fuel Oil/ Heavy Fuel Oil.

Considerations for the use of M.D.O./H.F.O.

This is a complex matter. Investment and exploitation begin to play an important role in the development of a diesel engine that can use both kinds of fuel. Heavy fuel oil requires an expensive purifying- and treatment equipment, which has to be recouped within a reasonable time. Furthermore, legislation on toxic emissions such as nitrogen, sulphur dioxides and carbon dust particles is becoming increasingly important.

See also Chapter 22, Diesel engine emissions.

Generally, all large two-stroke crosshead engines, as well as the large four-stroke trunk piston engines use H.F.O.. The decision point between the use of M.D.O. and H.F.O. depends largely on the required power, engine operation hours per year, environmental legislation and numerous other factors.
In navigation ships with a propulsion power of over 3000 kW are often equipped with H.F.O. engines.

▼
Almost all large ships with a lot of propulsion power use HFO. This is less suitable from an environmental viewpoint.

Diesel power plants which generally use medium-speed four-stroke engines usually use H.F.O..
A decisive factor is the high number of operation hours per year. Then fuel costs are therefore exorbitant.
H.F.O. is seldom used for high-speed diesel engines.
In the main, because the real combustion time is too short to fully combust the fuel.

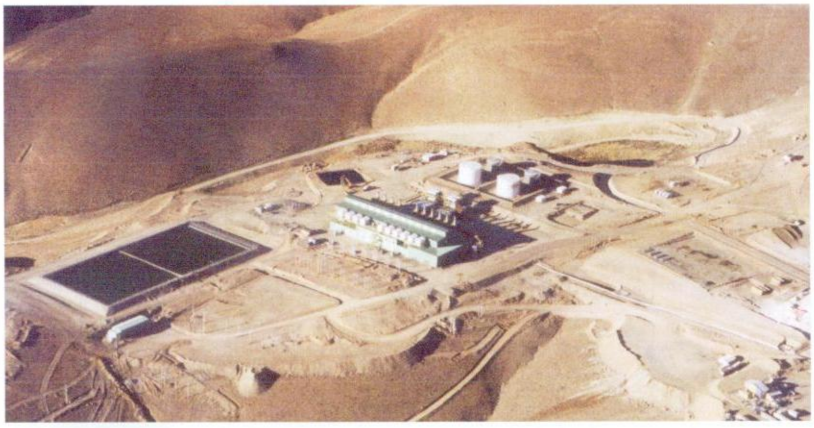
See also Chapter 8, Fuels, fuel-line systems and fuel cleaning.

2.7 Use of engines

The distinguishing characteristics are:
- the drive type , such as propulsion, generation, pumps, compressors, traction or different tool applications;
- the number of operation hours on a yearly basis;
- the degree of load and such variations;
- the atmospheric conditions such as air pressure, humidity, air pollution, minimum and maximum air temperatures.

▶ An off-shore supply ship.

▶▶ A floating diesel power plant in a port.



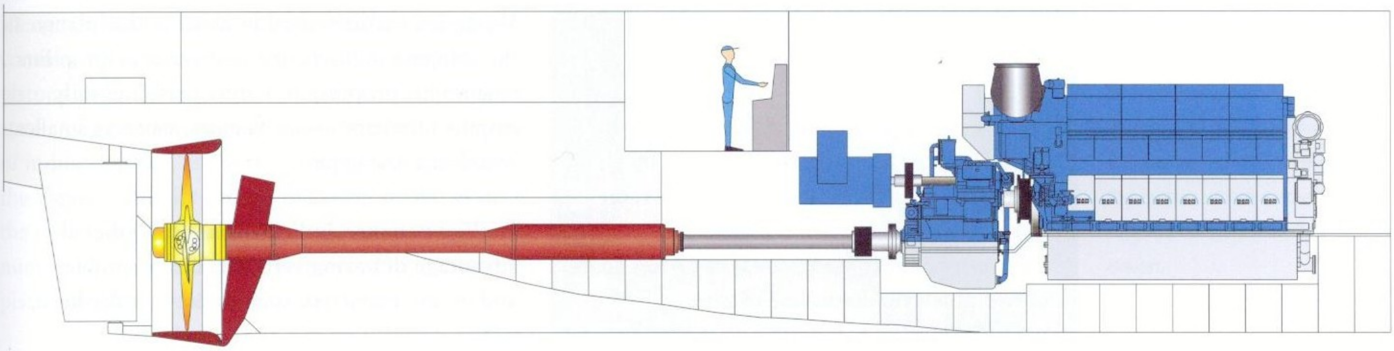
▲ An ore mine far from civilisation.

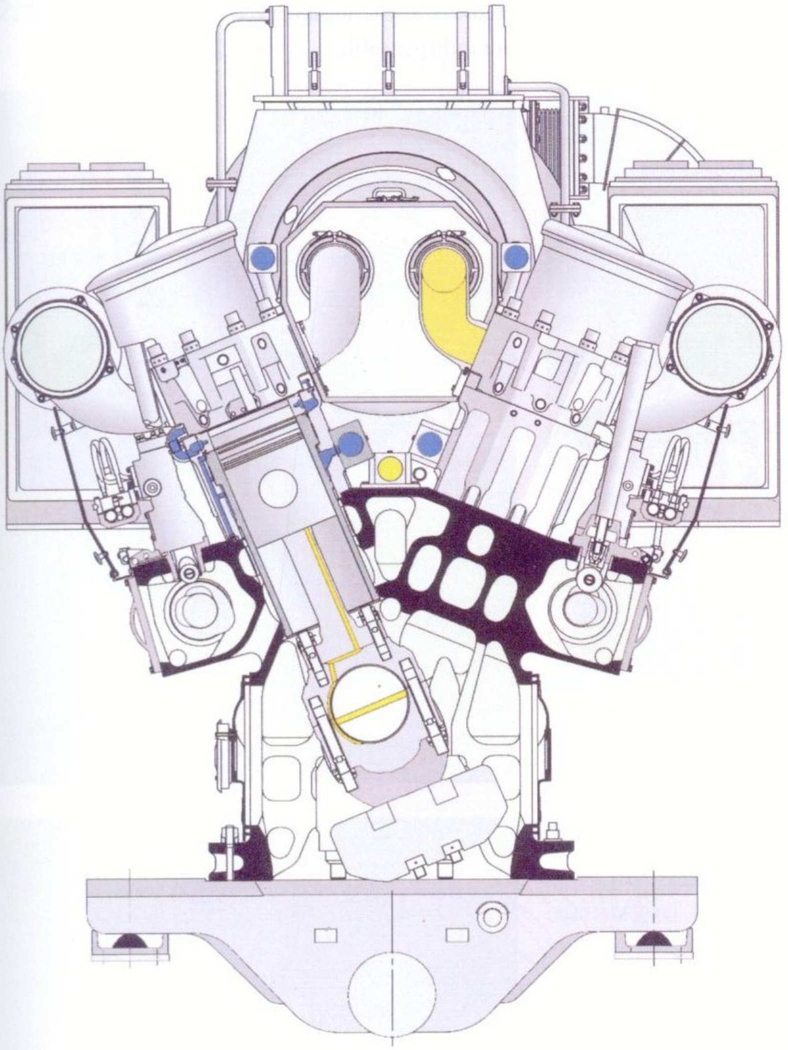
The required electrical power is generated with diesel engines. The fuel, heavy fuel oil, is supplied by a tank lorry. In the background two white storage tanks containing heavy fuel oil.

▲ Fast marine ships require a great deal of power; a high-speed four-stroke V-engine using diesel oil (M.D.O.) is most common.

◀ A relatively small container ship, a so-called 'feeder'.

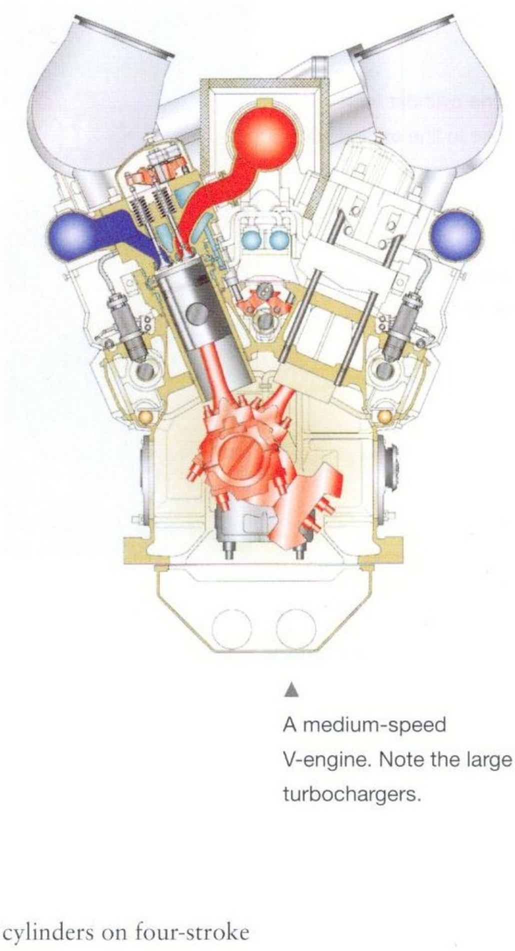
Propulsion with a medium-speed diesel engine with in-line cylinders, fuel is heavy fuel oil.





◄
A large four-stroke medium-speed diesel engine with cylinders in V-arrangement. Make MAN–B&W.

▲
A four-stroke in-line engine with reduction gear, intermediate shaft, propeller shaft, and screw propeller arrangement in a ship.



▲
A medium-speed V-engine. Note the large turbochargers.

2.8 Other characteristics of diesel engines

In-line- and V-engines

In in-line engines the cylinder axis lines are planar. The maximum number of cylinders in a two-stroke crosshead engine is usually twelve. As the size of container ships increases fourteen- or sixteen-cylinder in-line engines are also possible. The total length of the engine increases to 25 to 35 meters! The number of cylinders on four-stroke trunk piston engines does not normally exceed ten to twelve; nine cylinders being the common in-line engine.

In V-engines the cylinder axis lines are biplanar at an angle of 45° to 120° from each other. They normally have a maximum of twenty cylinders with one often seeing sixteen to eighteen cylinders in V- designs in large diesel power plants.

2.9 The use of in-line and V-engines

V-engines are most often used as they have the advantage of being shorter in length in comparison to an in-line engine with the same power output.
An eighteen-cylinder V-engine takes up less space than two nine-cylinder in-line engines.
In the engine industry everything revolves around economics. The purchase costs of V-engines are considerably lower when taking into account the same number of cylinders in-line.

V-engines are often used in diesel power plants. I the shipping industry, the preference is for in-line engines for propulsion. Larger vessels usually employ auxiliary in-line engines, whereas smaller vessels use V-engines.

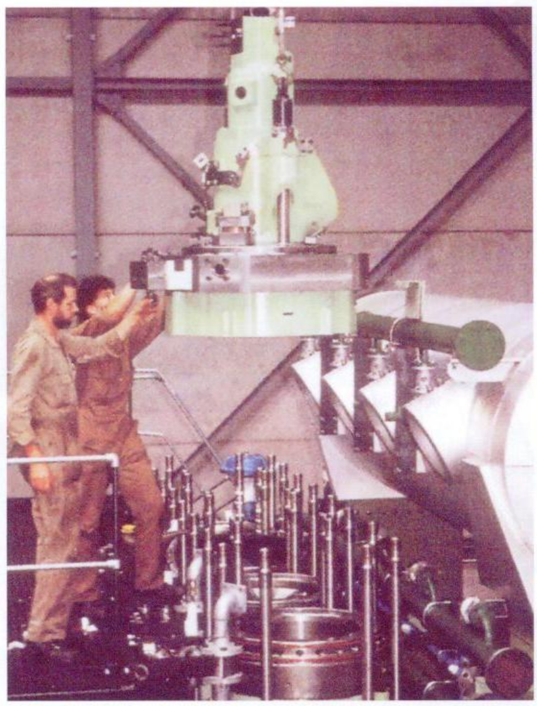
For maintenance, in-line engines have the advantage of having vertically dismountable and mountable parts, such as the cylinder head, piston, connecting rod and cylinder liner. For V-engines with a somewhat larger cylinder bore auxiliary tools and adapted hoisting machinery are indispensable.

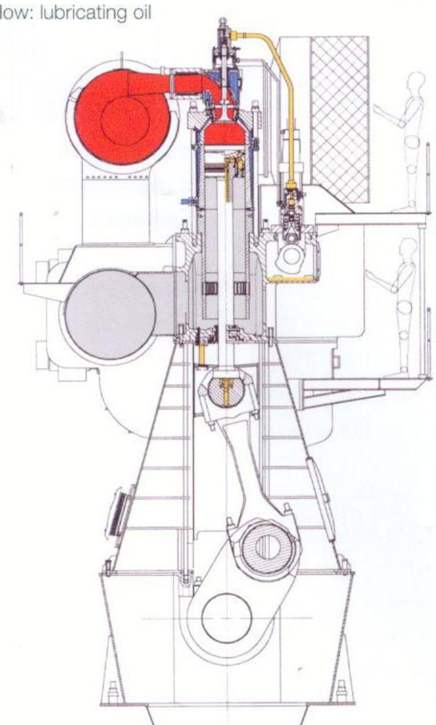
▶
Lifting a large cylinder head of a two-stroke cross head engine.

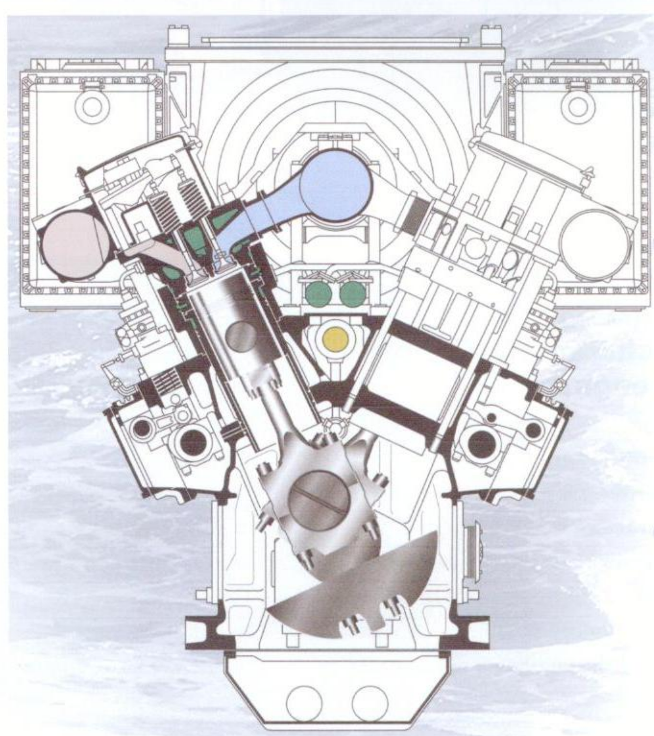
▶▶
A nine cylinder four-stroke in line engine with turbocharger.

▼
An in-line and a V-engine next to one another.

left: an in-line engine
right: a V-engine
red: exhaust gas
blue: scavenge air
blue: cooling water, left
green: cooling water, right
yellow: lubricating oil







V-engines and in-line engines seem identical when considering their components. Cylinder heads, pistons and liners are often the same. The piston stroke of some V-type engines is shorter than that of in-line engines; as there is not enough space in the V-engine block to make the same stroke as in the in-line engine. Since the stroke is shorter, the number of revolutions can be increased; the mean piston speed will then remain the same.

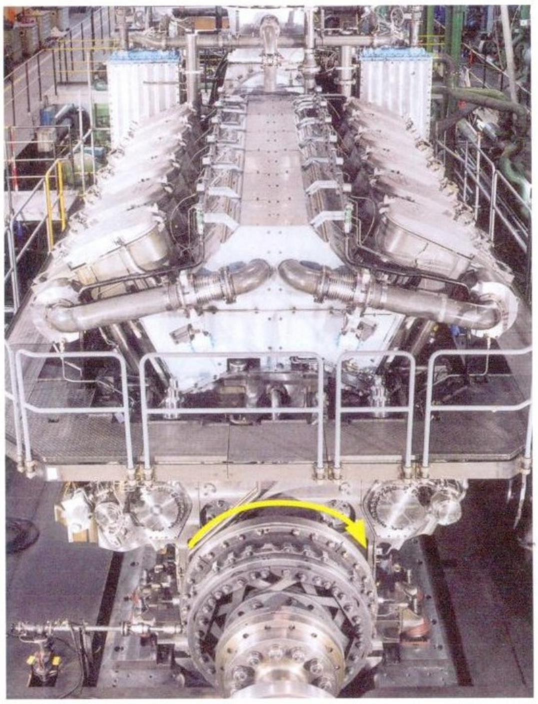
2.10 Direction of rotation of the diesel engine

If one were to stand at the far end of the crank shaft where the power output is delivered and looked at the engine, it turns clockwise. A large number of engines turn clockwise. In large in-line engines the exhaust gases leads are situated on the right hand side, seen from the outgoing engine shaft.

▼

A regular clockwise turning engine.

This picture shows a large two-stroke crosshead engine with a cylinder bore of 960 millimetres and the common rail fuel system by engine manufacturer Wärtsilä-Sulzer , where the exiting part of the crankshaft can still be seen. For a regular clockwise turning engine, cylinder 1 is the cylinder furthest removed from the exiting crankshaft. So, in this twelve-cylinder in-line engine cylinder 12 is closest to the exiting crankshaft.



◄

The revolution direction of a diesel engine.

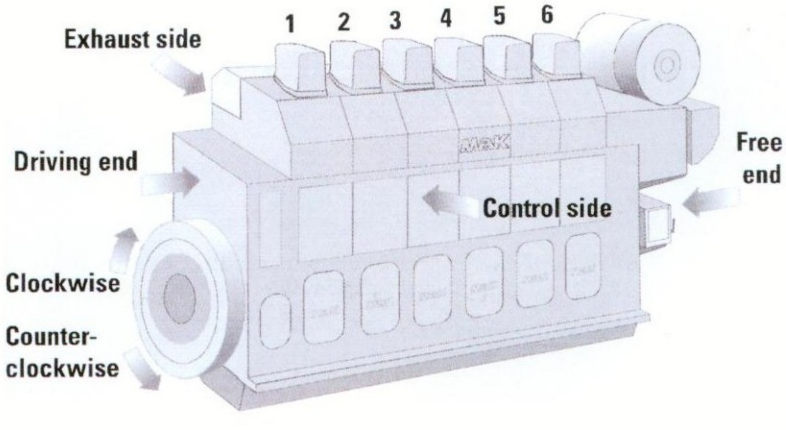
Clockwise, , on the side of the engine where a screw propeller or generator is driven.

The central exhaust gases lead of a two-stroke crosshead engine is located on the right hand side of the ship (starboard) with the turbo blowers, air coolers, exhaust gas boiler, silencers and finally the exhaust.
Because of this, there is enough space at the left hand side of the ship (port) for the propulsion gear of the exhaust valves, the fuel supply, a floor for ship repairs and a hoisting space for an overhead crane.

▼

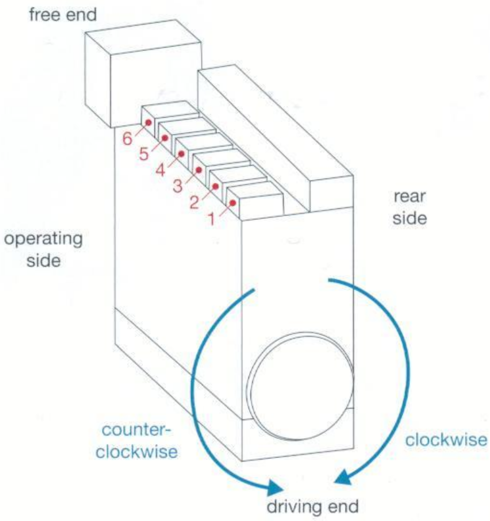
Description of the direction of rotation and cylinder numbering.

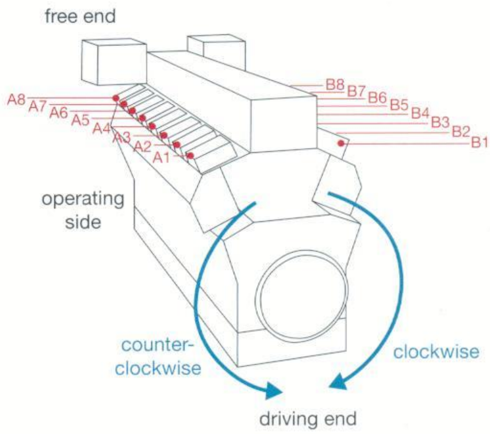
Driving end: The part of the engine where the exiting crank shaft is located.
Free end: The 'blind' part of the engine, so without exiting crank shaft.
Control side: This is the side of the engine with the cam shaft to drive the fuel pump and the valves.
Exhaust side: This is the side of the engine where the exhaust gases lead is located.



▶

Cylinder numbering and direction of rotation. For the in-line engine of this engine manufacturer numbering of the cylinders starts at the driving end. For V-engines one speaks of banks, the left bank ranging from A1 up to and including A8 and the right bank ranging from B1 up to and including B8.

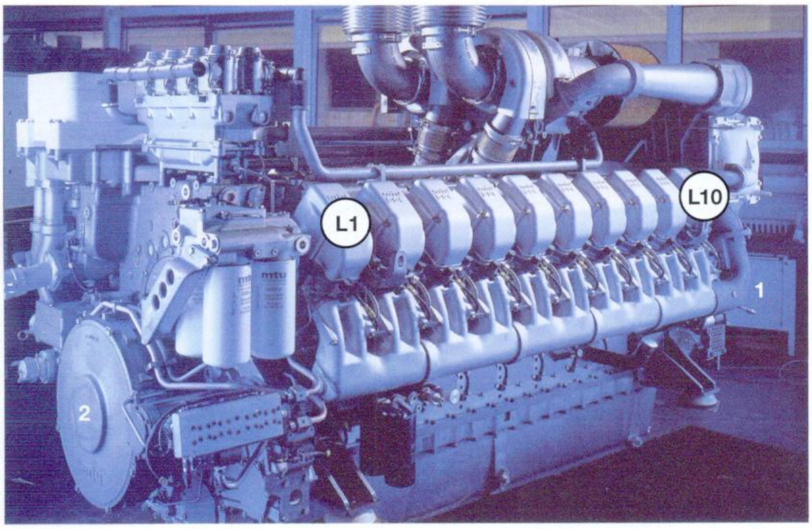


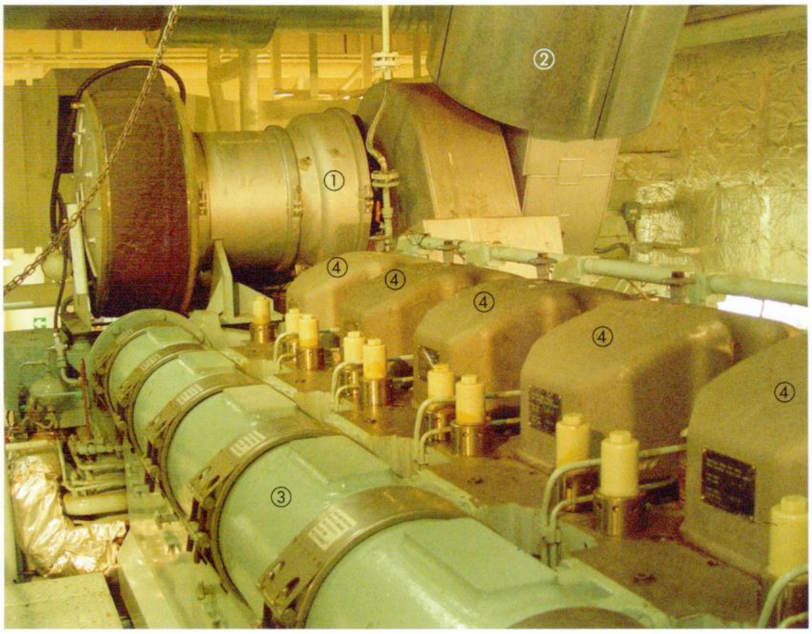


▼

V-engine numbering with a left- and right bank.

1 *driving end*
2 *vibration damper*





2.11 Cylinder number

Cylinder 1 is often the cylinder furthest removed from the crank shaft side. So in traditional propulsion engines cylinder 1 is situated towards the front end of the ship and the cylinder with the highest number towards the rear end of the ship.

Note

It is recommended that the manual be consulted in order to check the correct numbering; this also applied to V-engines!

2.12 Natural aspiration and turbo-charging

Natural aspiration

When the engine draws in air by means of the piston movement this is called natural aspiration. The air enters the cylinder with pressure slightly lower than the atmospheric pressure.

Super charger or Turbo charger

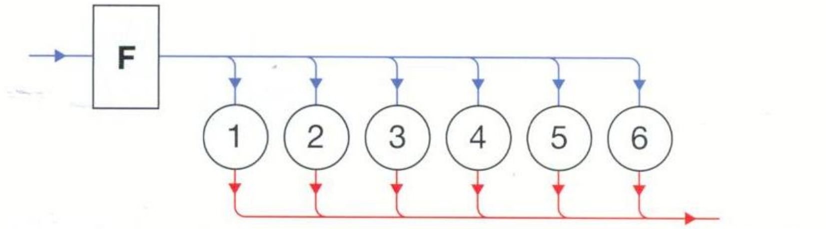
When the air supply to the cylinder takes place under pressure which is higher than the atmospheric pressure, it is called turbo or super-charging. Here air is compressed and transported to the engine by means of an exhaust gas driven turbo blower.

◀

Virtually every sizeable industrial diesel engine is equipped with a turbocharger.

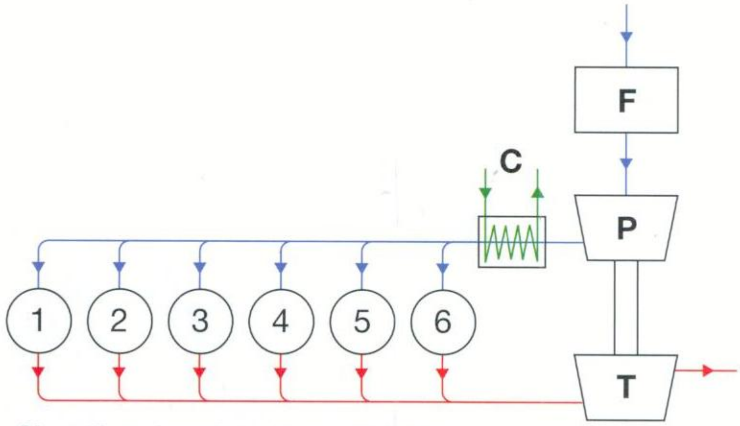
1 *turboblower*
2 *exhaust gases manifold*
3 *inlet air manifold*
4 *cylinders*

►
The principal of natural aspiration and turbo charging



Six-cylinder engine – natural aspiration

F - air filter



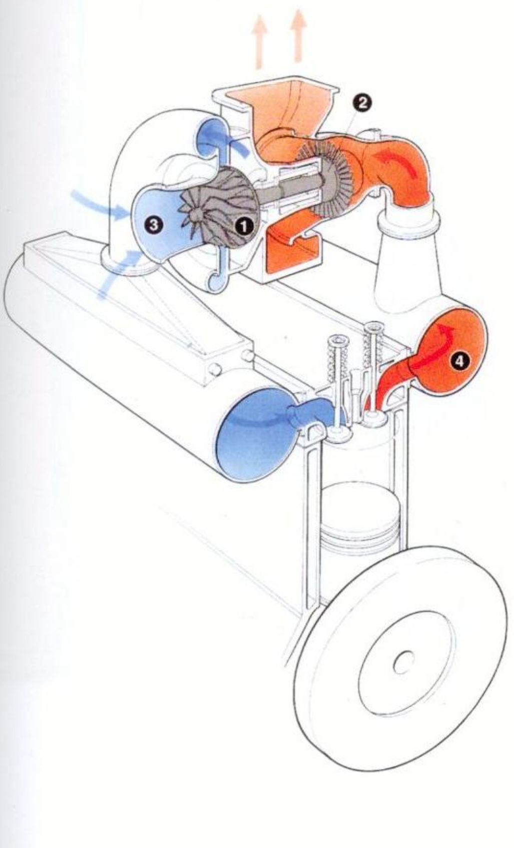
Six-cylinder engine – turbo charging

F - air filter
P - turbo-blower - air section
T - turbo-blower - exhaust gas section
C - air-cooler

▼
The turbo charger principle.

The turbo blower, driven by exhaust gases draws in ambient air by means of a centrifugal pump and forces the air under overpressure into the air inlet manifold of the cylinders. As soon as the inlet valves open the air flows into the combustion chamber. Of course the downward piston movement also has an inductive effect on the airflow.

1 *turbo blower, air section with a centrifugal pump*
2 *turbo blower, exhaust gases section with an impeller for a gas turbine*
3 *air inlet channel*
4 *exhaust gases manifold*



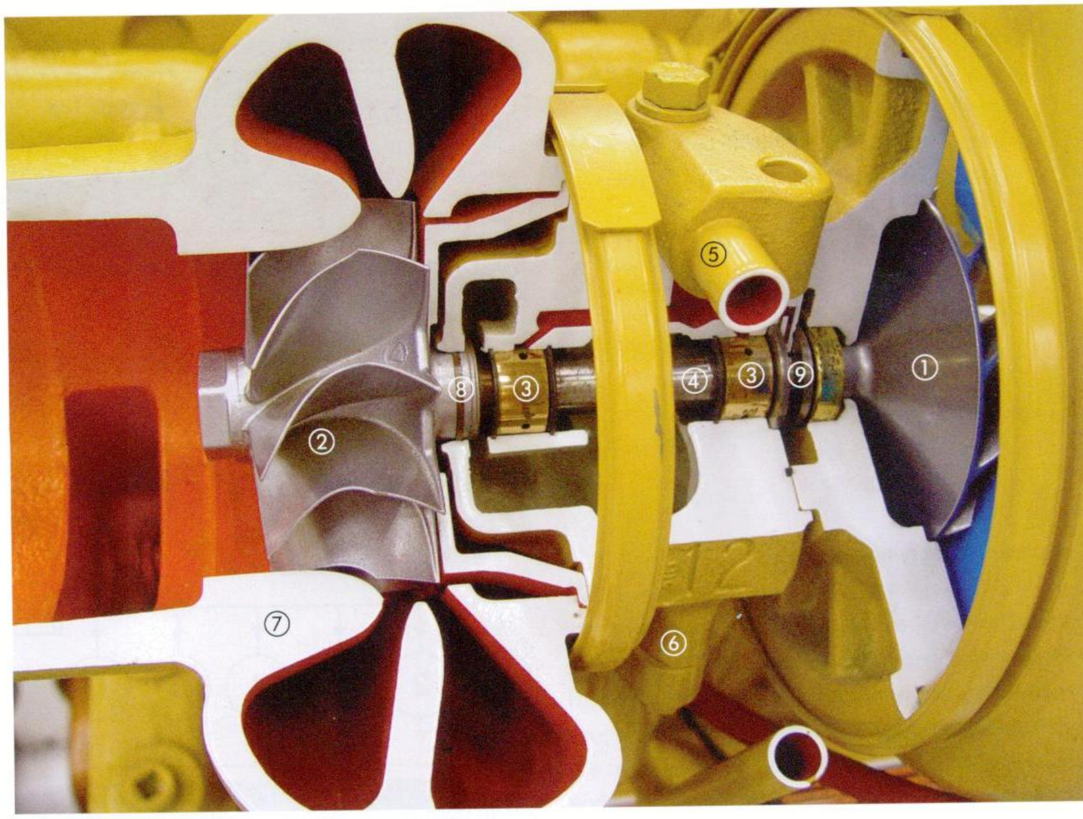
Today many engines are equipped with turbo chargers; one of the advantages is that with an identical cylinder volume, the power output can be considerably increased.
With an increased air intake, using the same stroke volume more fuel is supplied to the cylinder for combustion, which increases the power output.

Also see Chapter 12, Air supply.

▶

Cross view of a turbocharger group .

1 *impeller blower*
2 *exhaust gas turbine*
3 *bearings*
4 *turbine shaft*
5 *lubricating oil inlet*
6 *lubricating oil outlet*
7 *housing*
8 *exhaust gas gasket*
9 *air gasket*



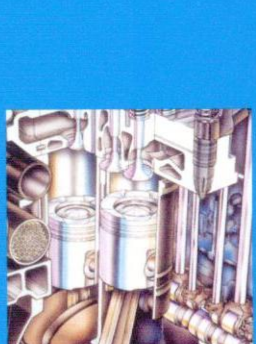
▶

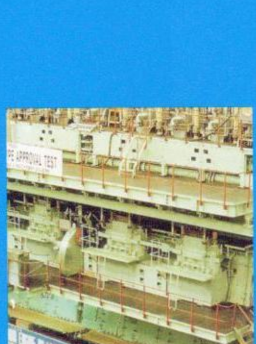
An exhaust gas turbine of a large two-stroke crosshead engine .

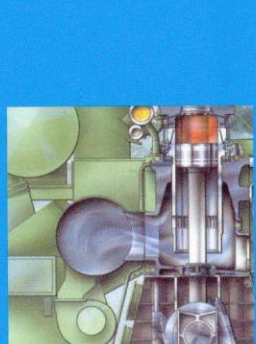
1 *cylinder*
2 *cylinder head*
3 *exhaust valve*
4 *exhaust gases manifold of the cylinder*
5 *central exhaust gases manifold*
6 *auxiliary blowers*
7 *gas turbine*
8 *exhaust lead towards exhaust gas boiler and outside air*

> CH 3

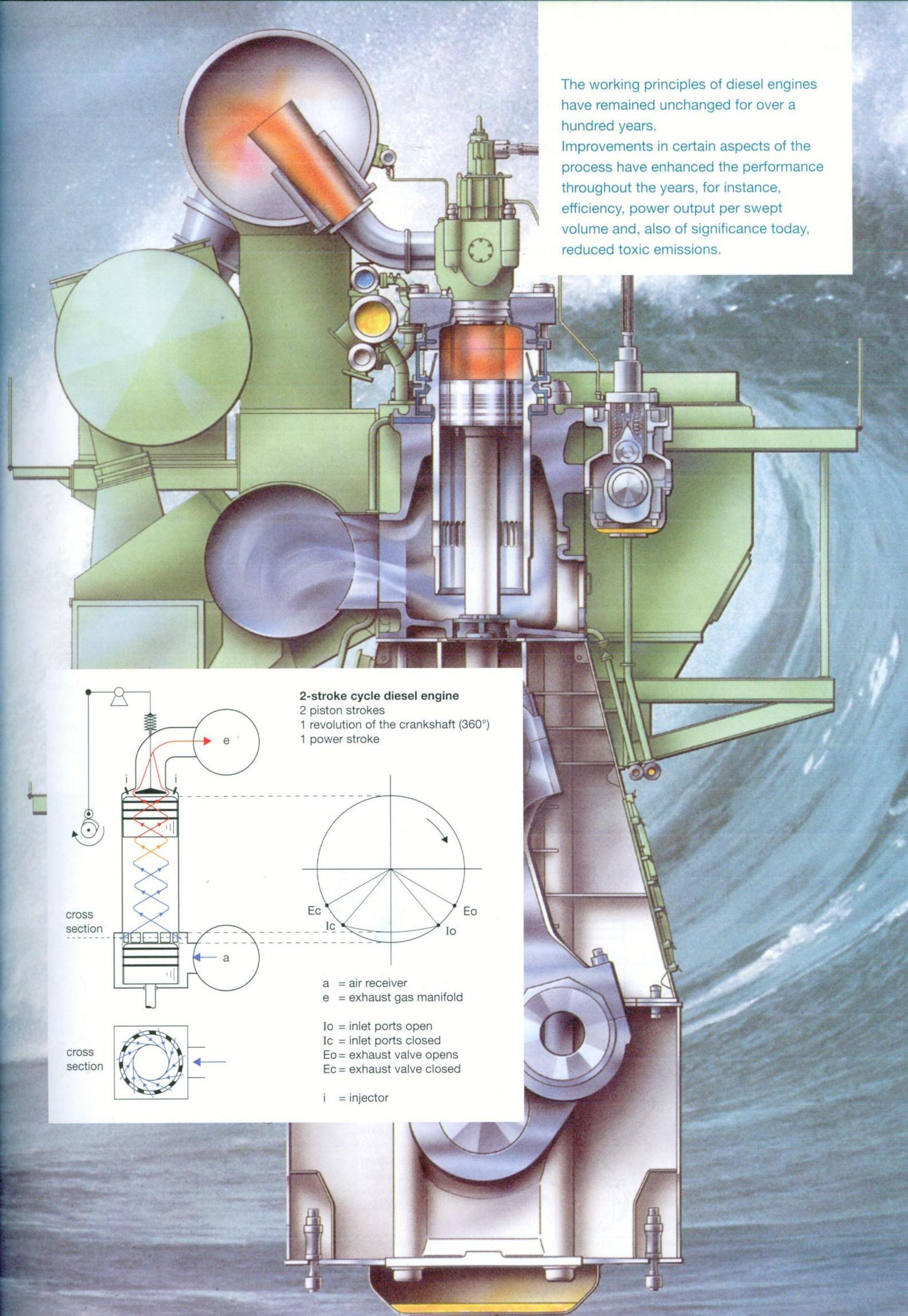
Working principles of diesel engines







The working principles of diesel engines have remained unchanged for over a hundred years.
Improvements in certain aspects of the process have enhanced the performance throughout the years, for instance, efficiency, power output per swept volume and, also of significance today, reduced toxic emissions.



3.1 Working principles

In the internal combustion engine there are two combustion cycles, namely:

1 **the four-stroke cycle,** which is four piston strokes during two crank shaft revolutions;
2 **the two-stroke cycle,** which is two piston strokes during one crank shaft revolutions.

Four-stroke engine

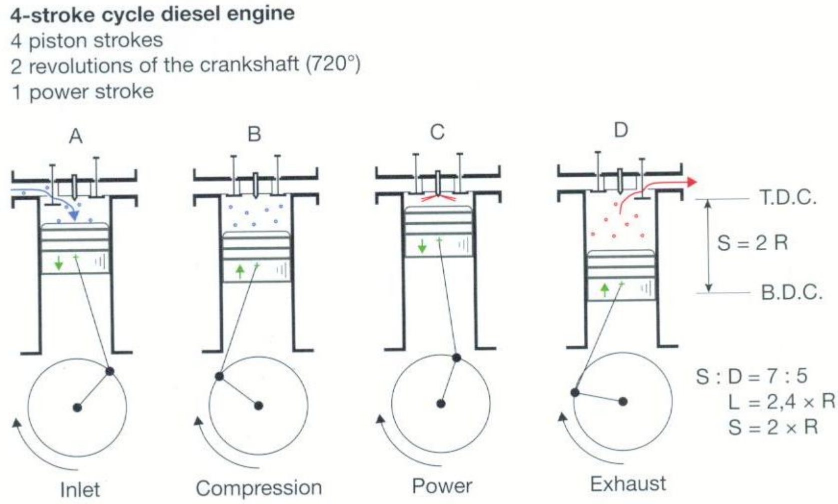
A **four-stroke engine** requires at least one inlet valve and one exhaust valve in the cylinder; today most four-stroke engines have two inlet- and two exhaust valves in the cylinder head. This has the added advantage of improved gas exchange and that the open valves are less far in the cylinder, thus allowing for a higher compression ratio.

3.2 Two-stroke engine build

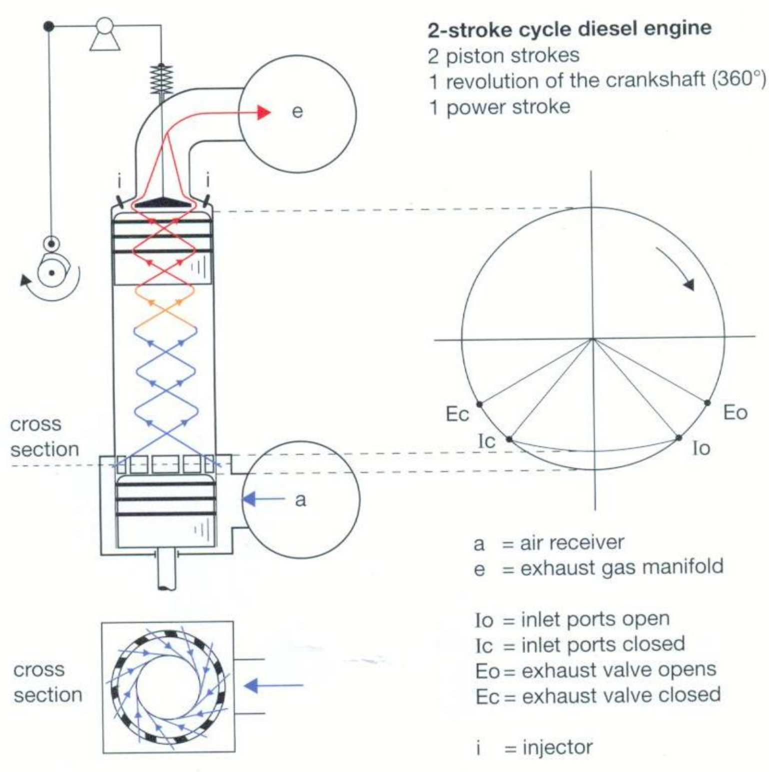
Since industrial two-stroke diesel engines of the trunk piston type are very rare today, we will only discuss the two-stroke crosshead engine principle. In a **two-stroke**-diesel engine there are two types of gas exchange:

1 **with inlet- and exhaust ports in the cylinder liner;**
2 **with inlet ports in the lower cylinder liner and a central exhaust valve in the cylinder head, the so called uniflow scavenging.**

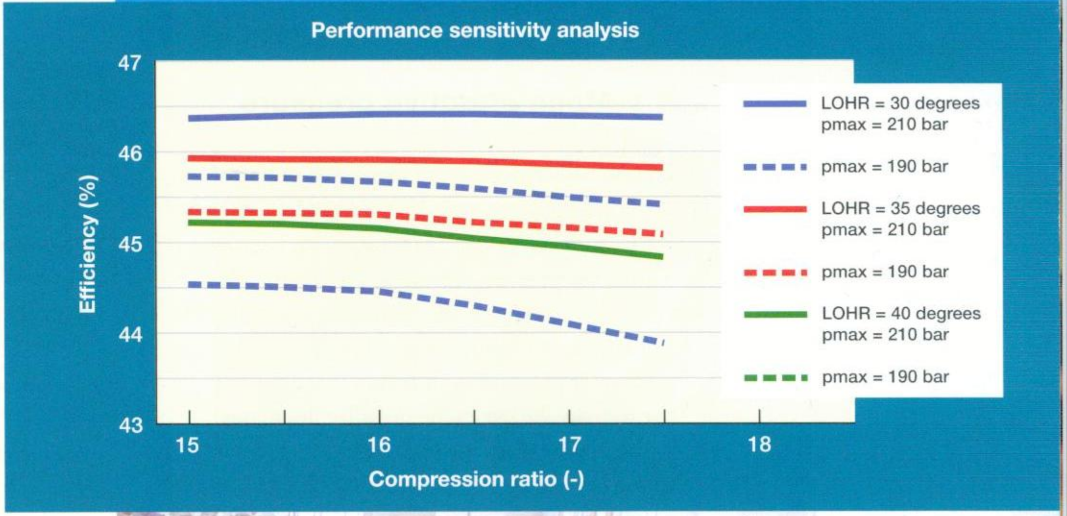
▶ The four-stroke cycle.



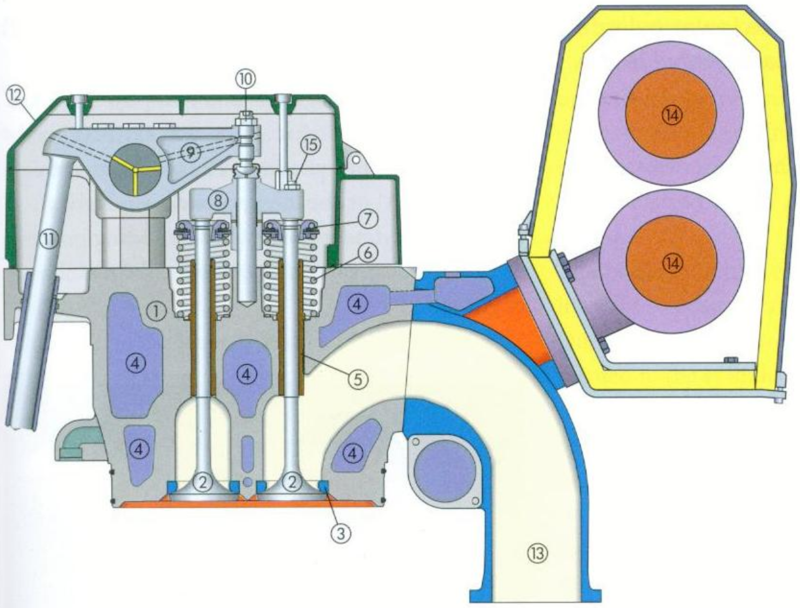
▶ The two-stroke cycle.



The compression ratio, efficiency and the maximum pressure.



In order to have a good grasp of diesel engines, it is essential to have a basic knowledge of the operating systems and their corresponding parameters. Here an excerpt from the MTU training centre for diesel engines in Friedrichshafen, Germany.



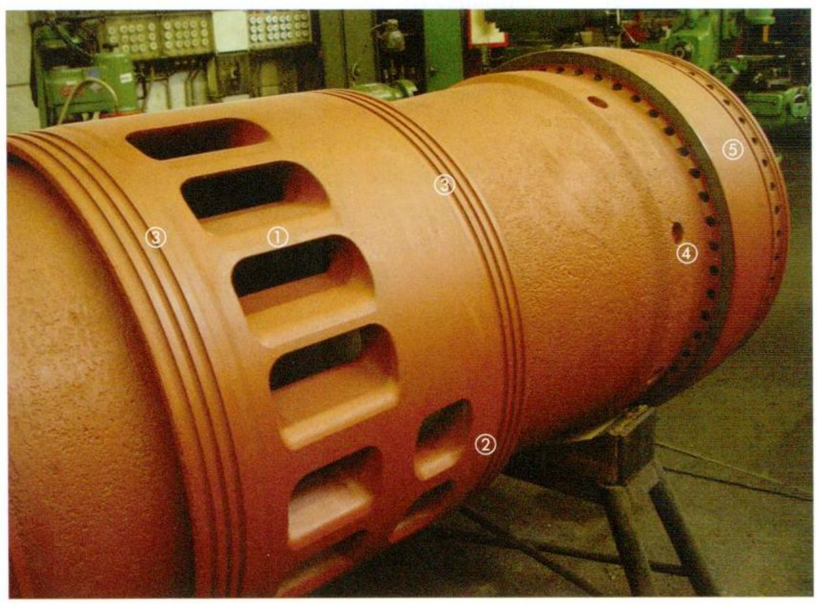
◄

Sectional view of a cylinder head of a four-stroke engine.

1. *cylinder head*
2. *intake valves (behind which are the exhaust valves, not visible)*
3. *valve seats*
4. *coolant passages*
5. *valve guides*
6. *valve springs*
7. *valve disc/rotor caps*
8. *spreader/bridge*
9. *valve lever/rocker arm*
10. *second adjusting bolt*
11. *push rod*
12. *valve cap*
13. *intake-air manifold*
14. *exhaust manifold*
15. *first adjusting bolt/clearance of the valve*

3.2.1 With inlet- and exhaust ports in the cylinder liner.

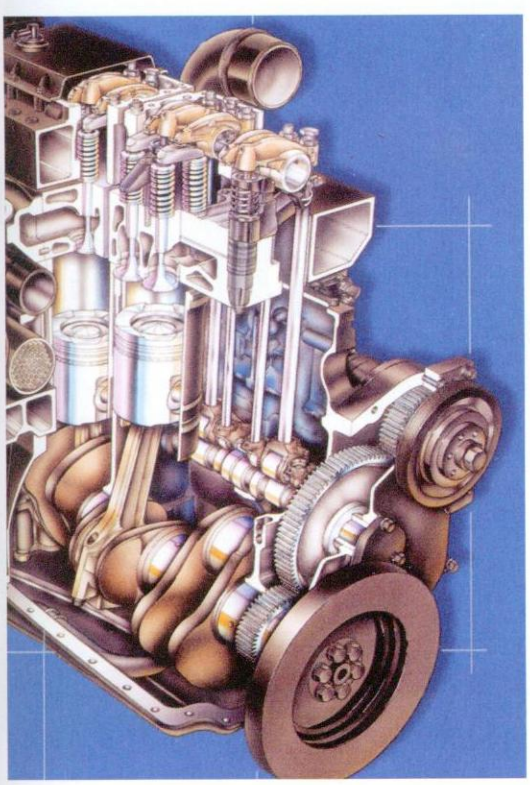
This is often the case with small engines such as hand tools, outboard engines etc. These have the advantage of having a simple and lightweight construction. Most of these engines work according to the Otto-process. Port type diesel engines are now only produced by some American manufacturers. Traditionally, the old two-stroke crosshead engines were also equipped with ports. These engines are still in use, but are no longer build.



▲

A cylinder liner of an early type two-stroke crosshead engine.

1. *intake port*
2. *exhaust port*
3. *grooves for the sealing O-rings of the coolant passages above and below the ports*
4. *holes for cylinder lubrication*
5. *liner with bore cooling*



◄

A typical four-stroke high-speed diesel engine by engine manufacturer Cummins.

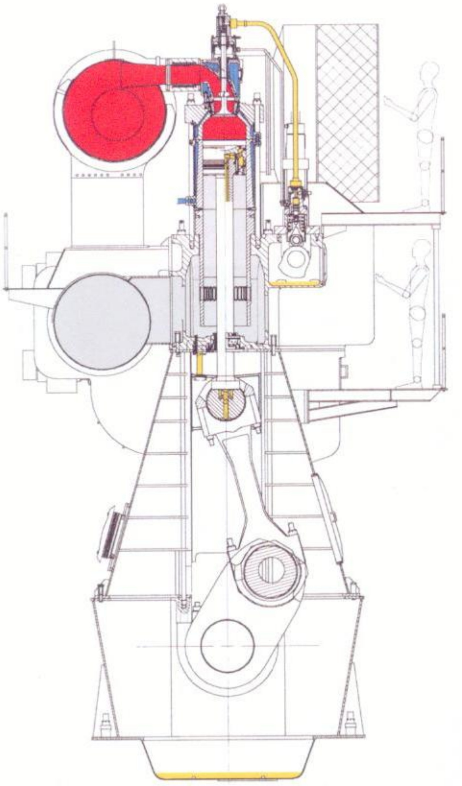
A cylinder head contains two inlet- and two exhaust valves

3.2.2 With inlet ports in the lower cylinder liner and one central exhaust valve in the cylinder head

All modern two-stroke crosshead engines are built with this configuration.

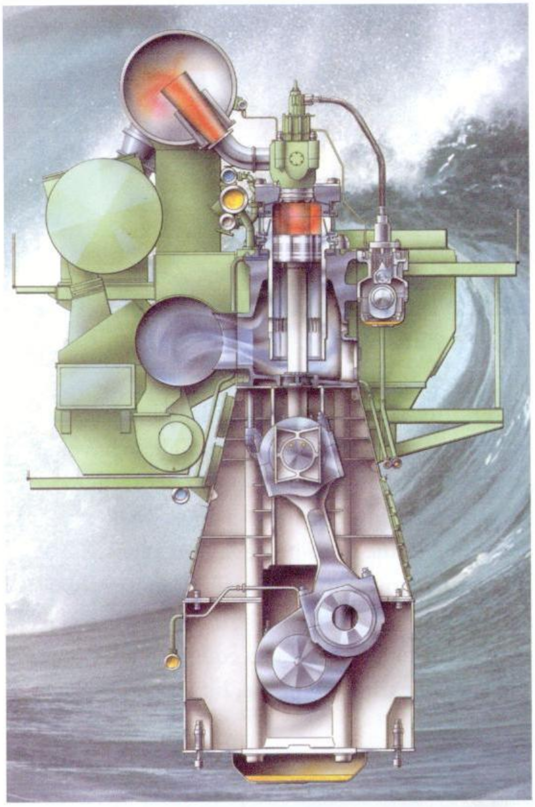
Note

The two-stroke cycle is characterised by one rotation of the crank shaft and two piston strokes in one combustion cycle. Inlet- and exhaust ports are located in the cylinder liner (small two-stroke diesel engines) or the inlet ports are situated around the cylinder liner with there being one central exhaust valve (large two-stroke diesel engines).



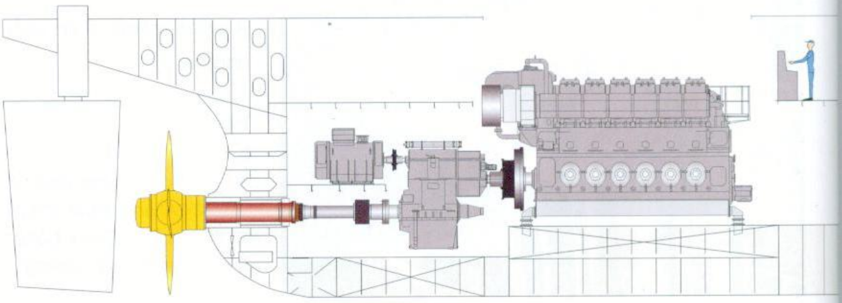
▶
A typical two-stroke crosshead engine by engine manufacturer MAN-B&W.

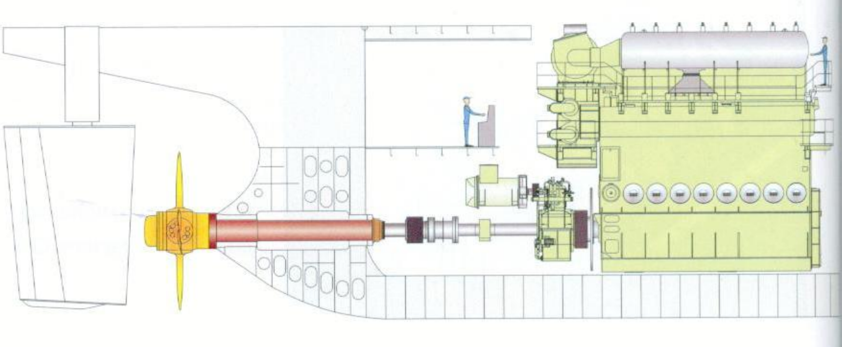
The engine is equipped with inlet ports and a central exhaust valve



▶▶
A large two-stroke crosshead engine which is almost always used for large propulsion plants.

On the left the air-intake manifold, the exhaust manifold, the turbo blower and the air coolers. On the right the exhaust-valve drive and the fuel pumps (not visible).





▶
Arrangement of two propulsion systems.

Above: propeller: four-stroke trunk-piston engine with gear box to adjustable-pitch propeller with a shaft generator on the gear box.
Propeller: RPM much lower than the engine rpm
Shaft generator: RPM much higher than the engine rpm
Comment
The built-in height of this propulsion installation is low.

Below: two-stroke crosshead engine with a direct-driven controllable-pitch propeller. The shaft generator, driven by the engine shaft via a reduction-gearing box.
Propeller: RPM identical to that of the engine
Shaft generator: RPM much higher of that of engine
Comment
The built-in height of this propulsion installation is high. In both of these systems, the shaft generators and controllable-pitch propellers can be disconnected from the engine.

3.3 Four-stroke engine set-up

There is a cylinder liner in the engine block, in which a piston makes upwards and downwards movements known as a linear motion. By means of a gudgeon pin the piston is hinged to the connecting rod which is fixed rotationally to the crank pin of the crank shaft.

When the piston has reached its highest point it stops for a moment: this is called 'top dead centre' (T.D.C.).
The bottom dead point is called 'bottom dead centre' (B.D.C.).

Intake stroke

During the downward stroke of the piston both inlet valves are open and the air flows into the cylinder under an over pressure. Both inlet valves close when the piston is almost at the bottom of its downward stroke (B.D.C.).

Compression stroke

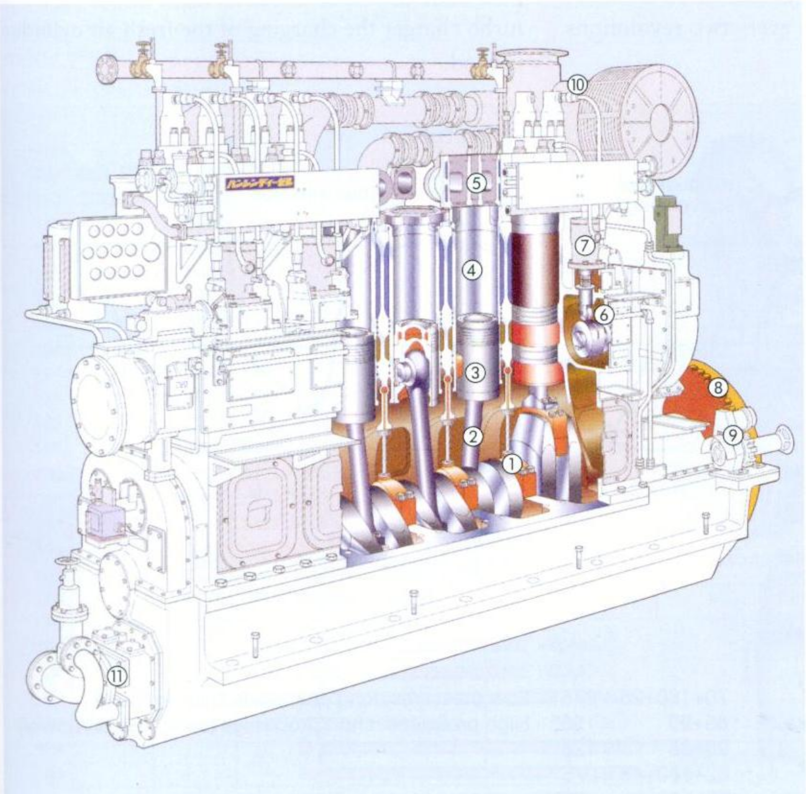
During the upward stroke of the piston the air is compressed and the pressure and the temperature of the compressed air rises considerably.

Combustion stroke

Just before the piston has reached the top of its upward stroke, an atomiser injects fuel under high pressure into the cylinder filled with hot air and the fuel/air mixture subsequently combusts swiftly. This is called the 'self combustion' of the diesel engine.
The fuel/air mixture combusts rapidly, causing an increase in pressure and temperature.
During the downward piston stroke the gases formed in the cylinder expand and exert force on the piston. This force is transferred to the crank shaft via the piston pin, connecting rod and the crank.

Exhaust stroke

Before the end of the power stroke the exhaust valves open and exhaust gases flow out of the cylinder. During the upward exhaust stroke most of the exhaust gases are expelled from the cylinder. Subsequently, the four-stroke cycle starts a new with the intake stroke.
At the top of the upward piston stroke the inlet- and exhaust valves remain slightly opened. The residual gases are removed by the scavenging air via the exhaust valves and the hot components of the combustion space, such as the exhaust valves, the cylinder head and the piston crown are cooled.



◄
A traditional four-stroke in-line engine.

1. *crankshaft*
2. *connecting rod*
3. *piston*
4. *cylinder liner*
5. *cylinder head*
6. *camshaft*
7. *fuel pump*
8. *flywheel and turning wheel*
9. *turning gear*
10. *turbo charger*
11. *lubricating-oil pump*

Note

The four-stroke cycle is characterised by the fact that one combustion cycle is completed in two revolutions of the crankshaft and therefore four piston strokes.
There are two inlet- and two exhaust valves in the cylinder head.
The four-stroke cycle is often applied to small, medium and large diesel engines. So these are the high-speed and medium-speed engines.

It is only cases requiring very high power outputs that one resorts to engines with the two-stroke principle. These are without exception crosshead engines that directly drive ship propellers or generators. These are always low-speed engines. In crosshead engines the forces moving the pistons are transferred to the crosshead via a piston rod that is fixed to the piston. The crosshead functions as a pivotal point between the fixed piston rod and the connecting rod fixed to the crank; lateral forces are absorbed in the crosshead and are transferred to the crosshead guide via the crosshead guides, which are fixed to the frame.

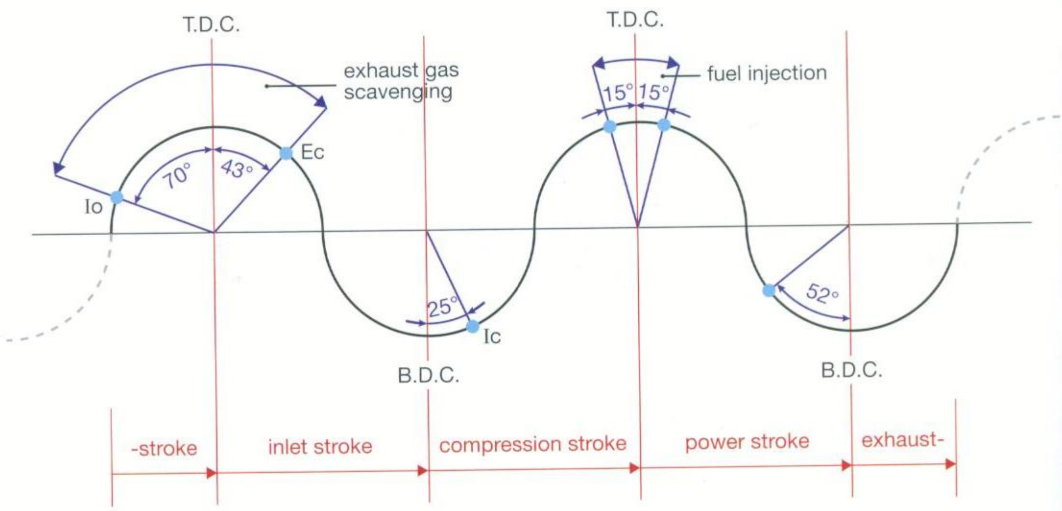
3.4 A few remarkable differences between the two-stroke and four-stroke cycles

In the two-stroke cycle there is a combustion process for every revolution, whereas in the four-stroke cycle there is one for every two revolutions. Theoretically, a two-stroke engine with a similar cylinder capacity and rpm should have twice the power output.
In the two-stroke cycle the exhaust gases flow out of cylinder under an over-pressure, at the same moment that the piston releases the inlet port, the air supplied must have an over-pressure in relation to the exhaust gas pressure, otherwise the exhaust gases will flow back into the cylinder!
The two-stroke engine requires an air supply that has a pressure exceeding that of the exhaust gas pressure and therefore far higher than atmospheric pressure (over-pressure).
In very small two-stroke engines the bottom part of the piston in conjunction with the smallest possible crank case functions as a scavenge pump. In slightly larger two-stroke engines a scavenge pump for the air supply is driven by the crank shaft. In industrial engines the scavenging air is always supplied by a turbo blower.

In the **four-stroke cycle** all stages are executed by the piston:
- intake;
- compression;
- power;
- exhaust.

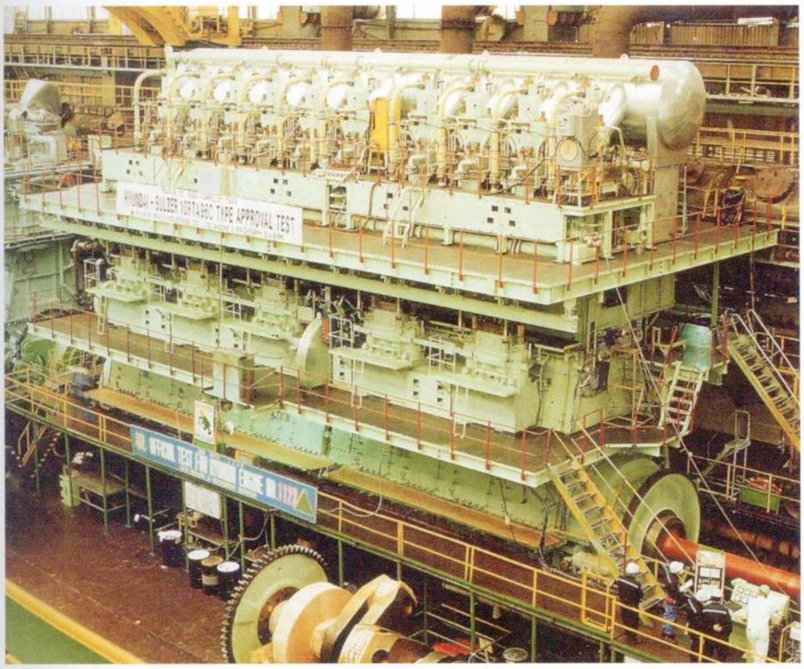
This is a simpler process. Additionally, the exhaust gases are far better expelled; the process is cleaner! No separate air supply system is required (over-pressure); also in four-stroke engines without a turbo charger the charging of the fresh air cylinder is good.

▶
The four-stroke cycle, The continual valve diagram.



Inlet stroke	70+180+25	= 275°
Compression stroke	65+90	= 155°
Power stroke	90+38	= 128°
Exhaust stroke	52+180+43	= 275°
Fuel injection	15+15	= 30°
Exhaust gas scavenging	70+43	= 113°

Low pressures: long processes (inlet/exhaust)
High pressures: short processes (compression/power)



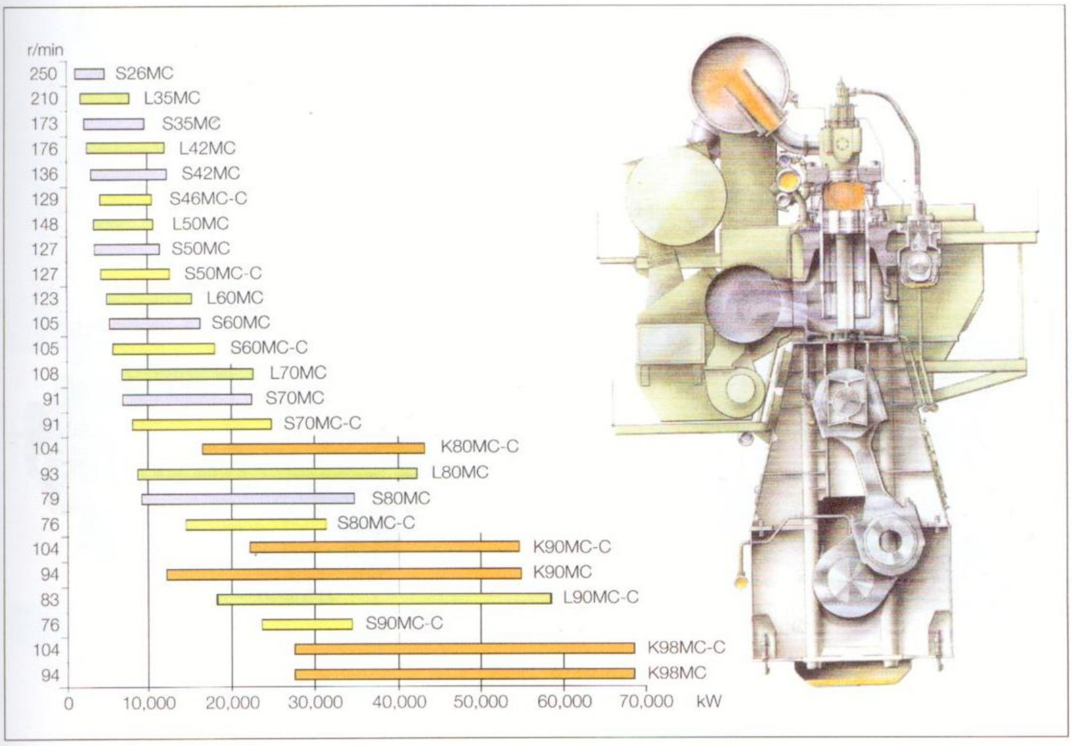
A twelve-cylinder two-stroke crosshead engine on a large container ship.

In this case a Wärtsilä Sulzer RTA 96-C, Southampton class (6800 TEU) of shipping company P & O Nedlloyd, now part of Maersk.

Even so, today most four-stroke engines have turbo charging and a turbo blower driven by the exhaust gases. This produces a considerable increase in the power output of the engine; an increase in the kilograms of air that can be fed into the cylinder means that there is in an increase in the amount of fuel combustion, thus creating a proportional increase in power output.

Note

Despite the fact that, theoretically, a two-stroke engine has the twice the power output with an identical cylinder capacity, one generally uses a four-stroke engine when a high power output in conjunction with a higher RPM are required. Only for a low power output does one use two-stroke trunk piston engines. Also in case of a very high output one generally uses two-stroke crosshead engines. In the latter case, ship propellers or generators are directly driven by the engine and therefore have the same RPM.



The engine programme of the two-stroke crosshead engines of MAN-B&W type MC.

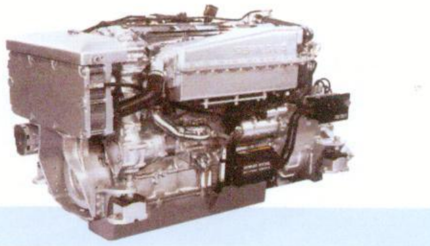
Here the smallest engine has a cylinder bore of 260 millimetres and the largest has a bore of 980 millimetres. To-day there are also 1080 millimetre engines on the market.

3.5 Examples of supply programmes of engine manufacturers

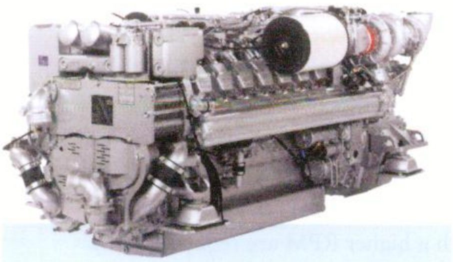
▶

The engine programme of manufacturer MTU.

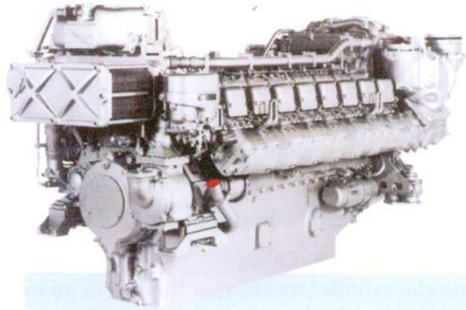
These are high-speed four-stroke diesel engines running on M.D.O.
The power output varies from 298 to 8200 kW.
RPM varies from 1150 to 2100.
The stroke volume varies from 14 to 347.4 litres.
The weight varies from 1840 to 43,000 kg.



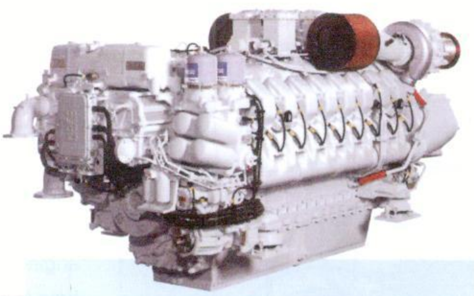
S60



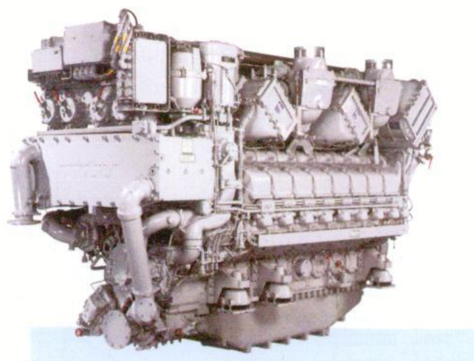
Baureihe **2000**



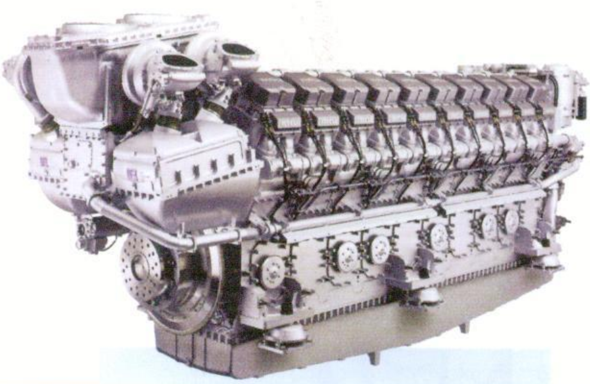
Baureihe **396**



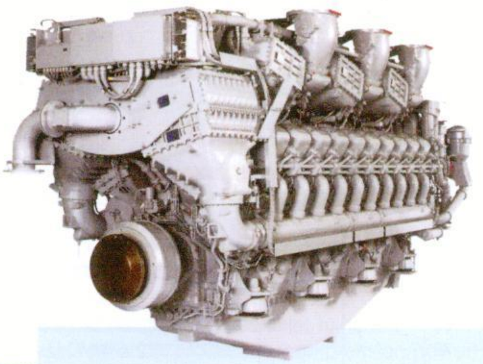
Baureihe **4000**



Baureihe **595**



Baureihe **8000**



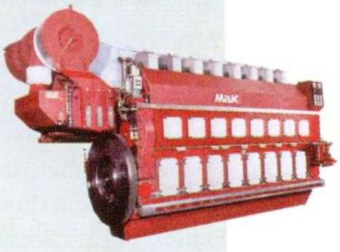
Baureihe **1163**

M 20

M 20

Bore	200 mm ∅
Stroke	300 mm
Cylinders	6, 8, 9
Output	1,020–1,710 kW
Rated speed	900–1000 rpm

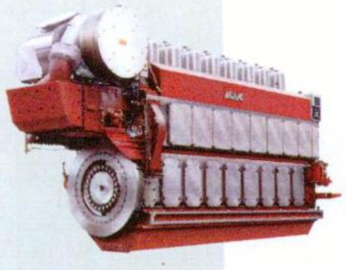
M 25



M 25

Bore	255 mm ∅
Stroke	400 mm
Cylinders	6, 8, 9
Output	1,800–2,700 kW
Rated speed	720–750 rpm

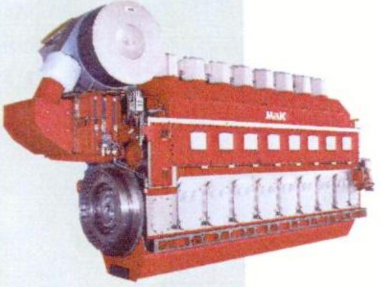
M 32 C



M 32 C

Bore	320 mm ∅
Stroke	480/420 mm
Cylinders	6, 8, 9, 12, 16
Output	2,880–2,000 kW
Rated speed	600–750 rpm

M 43



M 43

Bore	430 mm ∅
Stroke	610 mm
Cylinders	6, 7, 8, 9, 12, 16, 18
Output	5,400–16,200 kW
Rated speed	500–514 rpm

◄ The engine programme of manufacturer Caterpillar-Mak, four-stroke medium-speed diesel engines for engines that use H.F.O.

The engines' power output ranges from 1020 to 16,200 kW. RPM varies from 500 to 1000.

3.6 Important terms and definitions

3.6.1 The piston stroke (S)

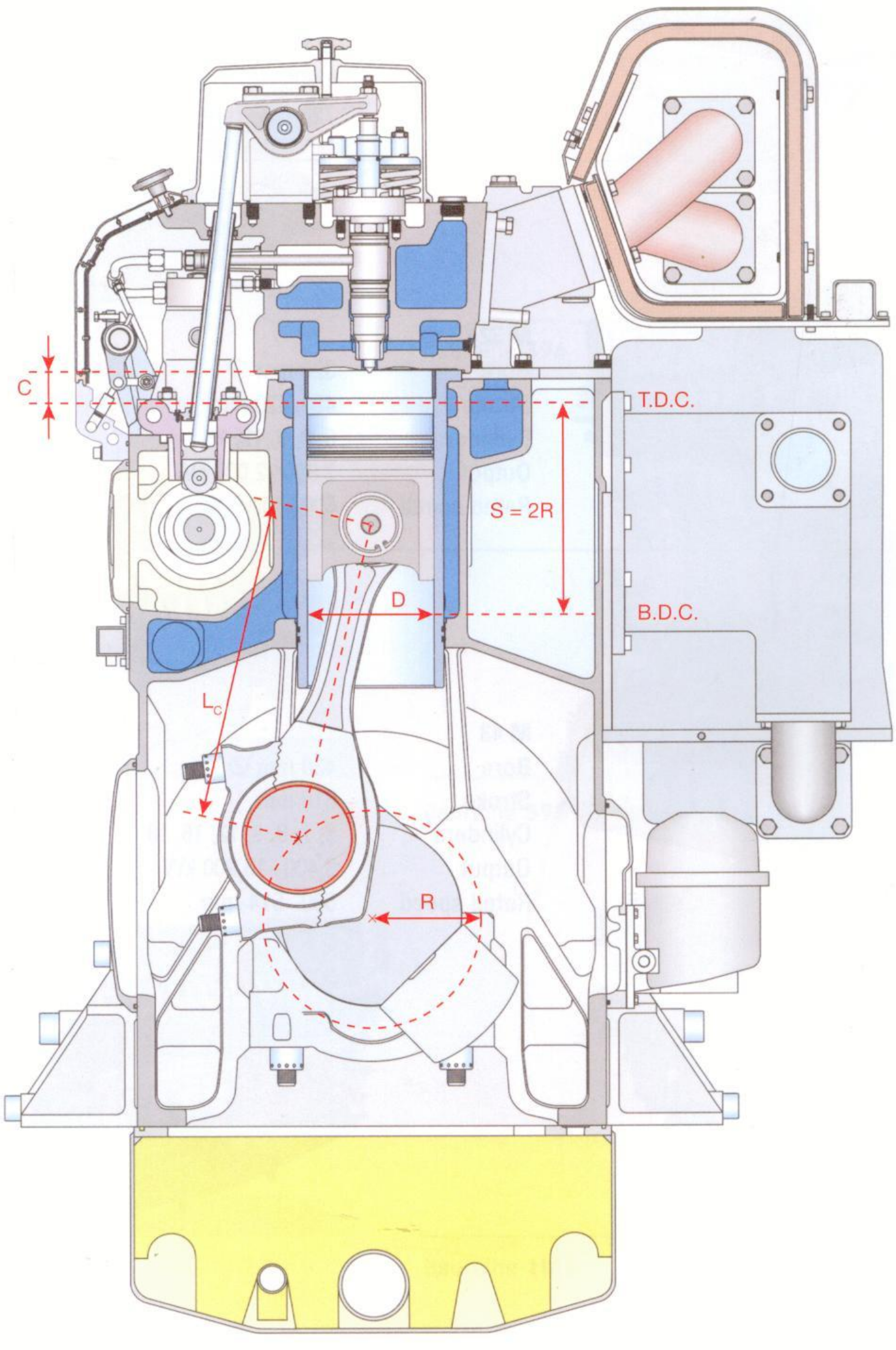
The piston moves between the bottom dead centre and the top dead centre; one can also say that the piston is either B.D.C. or T.D.C.. This means that the crank is positioned vertically upwards or vertically downwards, respectively.
The length the piston travels is the engine stroke. This varies in industrial engines from approximately 60 millimetres to 680 millimetres (four-stroke trunk piston engines) to about 3200 millimetres (two-stroke crosshead engines).

3.6.2 Cylinder bore (D)

In order to determine the cylinder bore, one takes the diameter of the cylinder liner. In industrial engines this ranges from 40 millimetres to 640 millimetres (four-stroke trunk piston engines) to 1080 millimetres (two-stroke crosshead engines).

3.6.3 The stroke/bore ratio (S/D)

The ratio varies from ca. 0.8 to 1.5 for four-stroke trunk piston engines and 2 to 4 for two-stroke crosshead engines.



The cylinder bore,
the stroke,
the connecting-rod length
and the crank circle.

3.6.4 Mean piston speed in metres per second (C_P mean)

This is directly related to the RPM and the stroke.
Mean Cp = 2 × S × n.
Two-stroke crosshead engines have a maximum piston speed of 6 to 8 m/sec. and four-stroke engines have a maximum piston speed varying from 8 to 12 m/sec.
This is determined by the acceleration- and deceleration forces of the gearing and lubrication of the pistons, rings and cylinder liner.

3.6.5 Crank length of the crank shaft (R)

The crank forms the link between the connecting rod and the crank shaft and ensures that the up- and downwards movement of the piston is converted to crank shaft rotation(R). via the piston pin, connecting rod and crank pin (four-stroke) or via the fixed piston rod, cross head, connecting rod and crank pin (two-stroke).

3.6.6 Crank circle (S)

In a frontal view of the engine the centre axis of the crank pin makes a circular movement with at its centre the centre axis of the crank shaft, which is denoted by the radius R and the diameter 2R = S.

3.6.7 Connecting rod length (L_C)

This is the distance between the centre axes of the piston pin and the crank pin. This is usually 3.5 to 4.5 times the length of the crank, so: D = (3.5 – 4.5) R.

3.6.8 Stroke volume (Vs)

This is the cylinder volume displacement between bottom- and top- dead centre.

$$V_S = \frac{\pi}{4} \times D^2 \times S$$

In smaller diesel engines the total cylinder volume is indicated in cm^3 (c.c.) or in litres.
This is less common in larger engines. Here one mentions output per cylinder, total output, the stroke, cylinder diameter and the number of cylinders.

3.7 Some engine names

MAN–B&W 12 K 80 ME – C

This is a twelve-cylinder two-stroke crosshead in-line engine with a cylinder bore of 800 millimetres (or 80 centimetres).
The stroke is 2300 millimetres (or 230 centimetres) (project guide), the RPM 104 and the maximum output 43,320 kW. Electronic fuel injection.

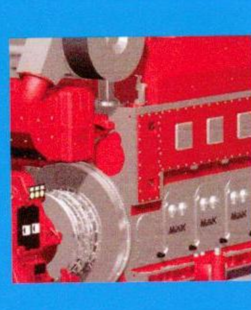
MAN–B&W 9L 16/24

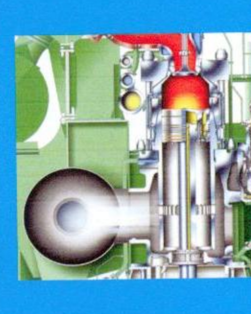
This is a nine-cylinder four-stroke trunk piston in-line engine with a cylinder bore of 160 millimetres and a 240 millimetre stroke.
Here the power output is a maximum of 900 kW at 1200 RPM (project guide).

>CH 4

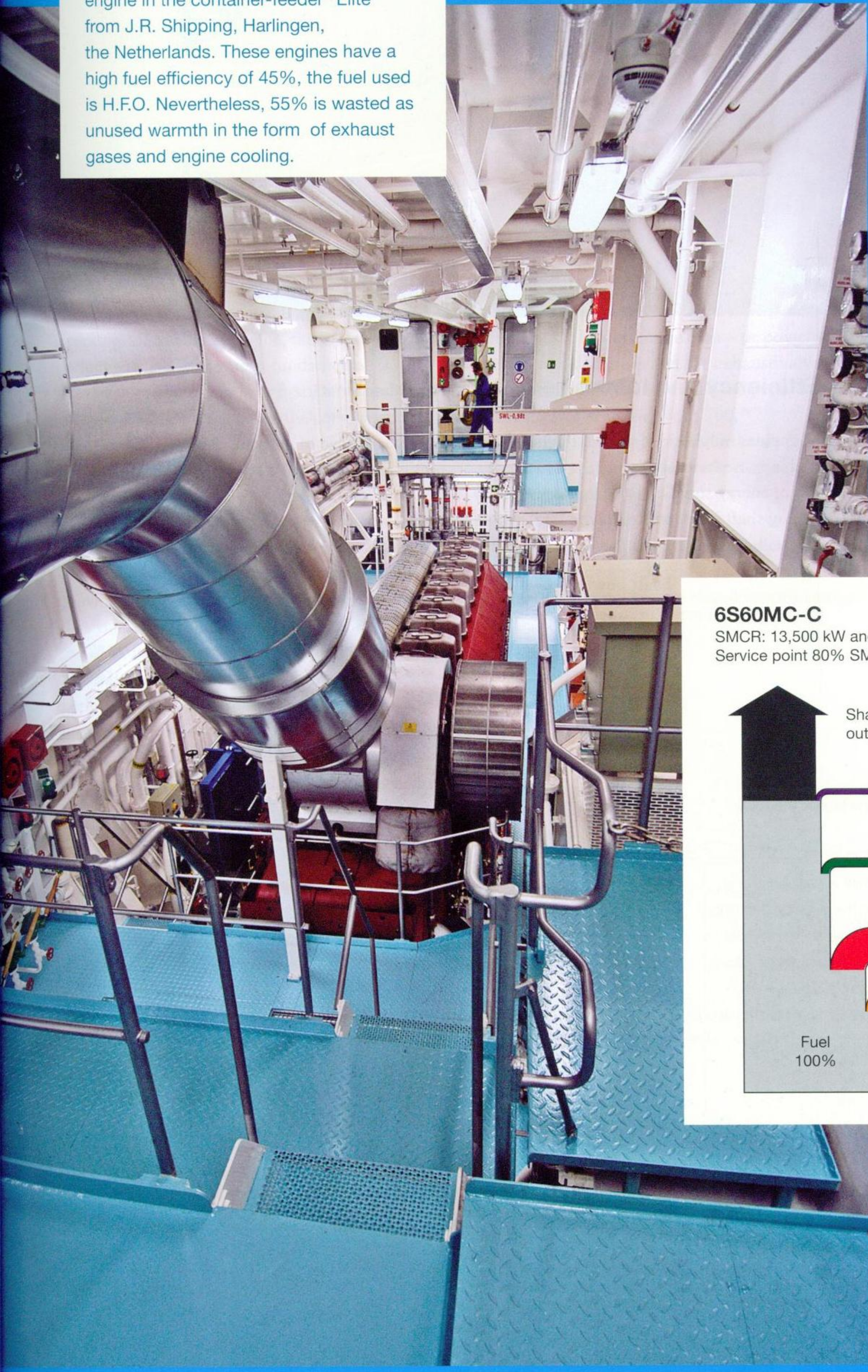
Efficiency and losses of diesel engines

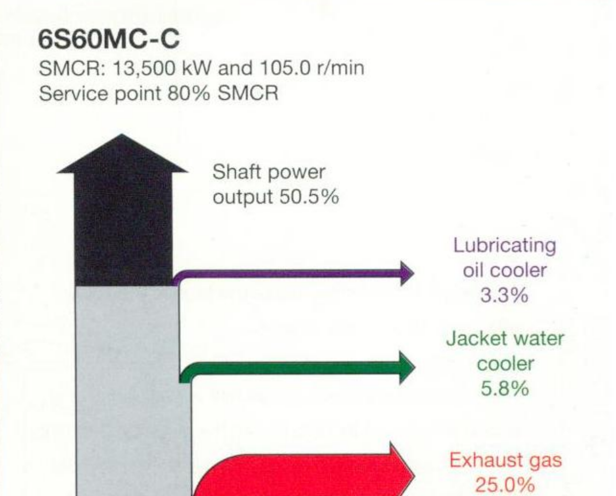






A medium-speed four-stroke Caterpillar-MAK 43 diesel propulsion engine in the container-feeder "Elite" from J.R. Shipping, Harlingen, the Netherlands. These engines have a high fuel efficiency of 45%, the fuel used is H.F.O. Nevertheless, 55% is wasted as unused warmth in the form of exhaust gases and engine cooling.





The Sankey diagram of a two-stroke crosshead engine. Efficiency is 50.5%.

▶
The total propulsion efficiency, from the engine fuel supply to the available ship thrust is approximately a third, or 33 %.

4.1 Efficiency and losses

In diesel engines only a small part of the fuel is converted into mechanical labour.
The ratio of energy supplied to energy delivered to the outgoing shaft is referred to as the total engine output.

$$\text{Engine Efficiency} = \frac{\text{Shaft power}}{\text{Energy in supplied fuel}}$$

Formulated form:

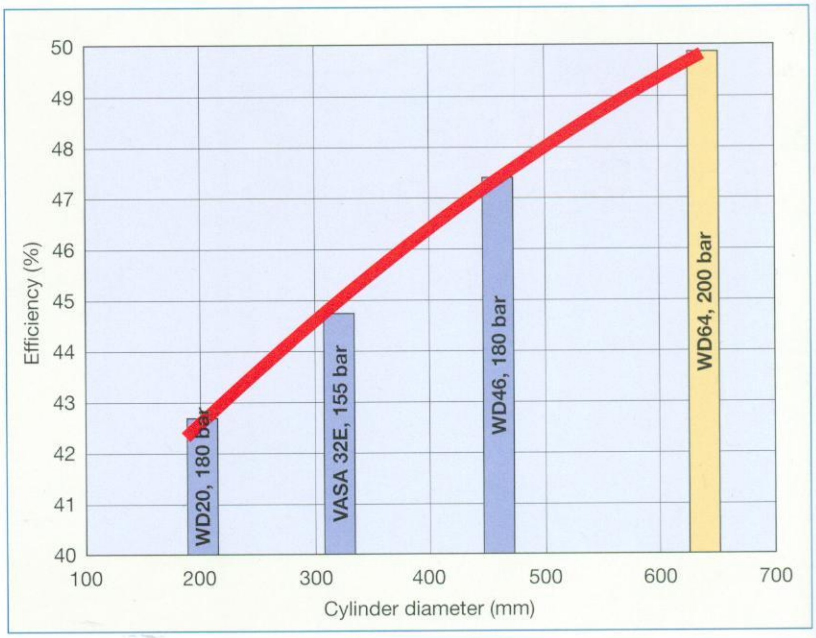
$$\eta_{tot} = \frac{P_E}{F_C \times H_L}$$

η_{tot} = efficiency as a ratio of 1
P_E = effective or shaft power in MW.
F_C = fuel consumption in kg/sec.
H_L = calorific value in MJ/kg

Small diesel engines have an efficiency of 25% whereas that of very large engines exceeds 50%.

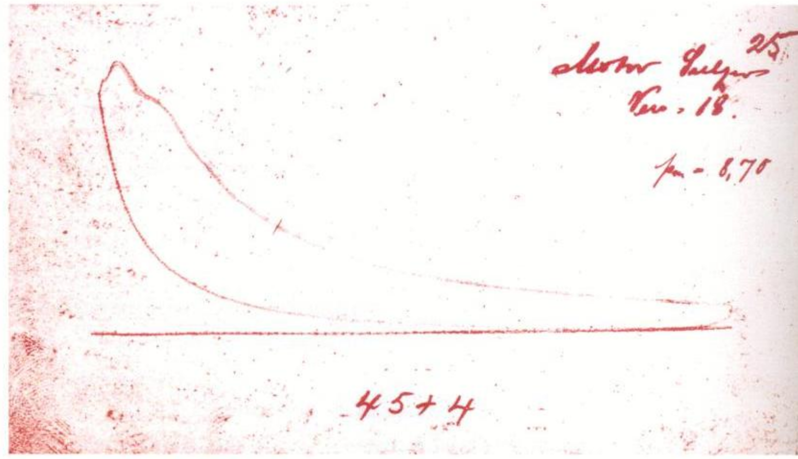
▶
The larger the cylinder bore, the higher the total efficiency of a diesel engine.

This example by Wärtsilä shows that the efficiency of an engine with a cylinder bore of 640 millimetres is 7 % higher than with an engine with a 200 millimetre cylinder bore. This is mainly due to the fact that a larger cylinder volume produces increasingly smaller radiant surfaces, liners, cylinder heads and piston bottoms, so more heat remains for power output.



All other fuel energy is eventually lost in the form of heat, namely in:
- the exhaust gases;
- the cooling water;
- the lubricating oil;
- the intercooler or air cooler;
- the heat radiation.

Additionally, energy is used to drive the valves, the fuel pumps and the engine driven cooling water pumps.
We will return to this subject later in this chapter in the **thermal energy balance or Sankey diagram.**



▲ Rudolf Diesel's very first cycle diagram.

Essentially, very little has changed over the past hundred years.

4.2 Indicator diagram

In order to obtain a good insight into the workings of the diesel engine, the Pressure–Volume- or P–V-diagram is fundamental. From the development of the very first **Diesel** and **Otto** internal combustion engines, the compression in the engine cylinder has been indicated as a function of the continuous stroke volume of the piston.
This can be measured by means of an indicator gauge. This results in an **indicator diagram,** popularly known as the 'banana- curve'. This provides a good insight into the compression in the cylinder and the work done.

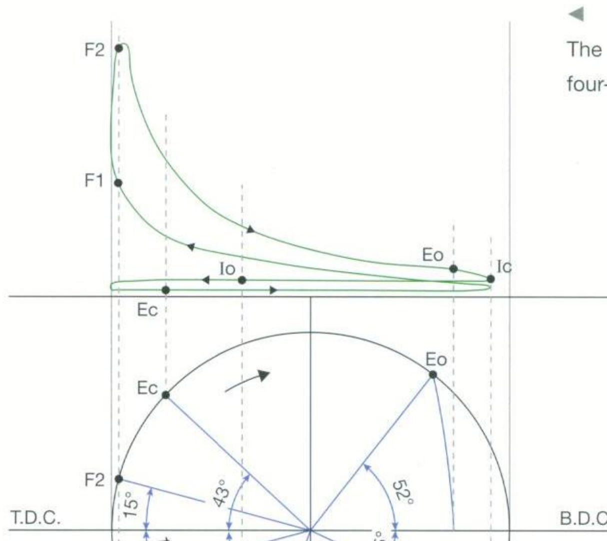
The following pictures show P–V-diagrams for both two- and four-stroke engines. Below both diagrams, the most important aspects of the crank circles are shown, such as the opening and closing of the inlet- and exhaust valves and the start and end of the fuel injection.

4.2.1 Four-stroke cycle

Let's assume that de piston is at bottom dead centre position (B.D.C.) and commences its upward stroke. From the moment that the inlet valves are closed the pressure mounts significantly from about 2.5 bars over pressure (super charging with turbo blower) to about 150 bars (final compression pressure). The internal air temperature has increased from ± 50 °C (super charged air temperature after cooler) to ± 750 °C (final compression temperature).
Just before the piston reaches top dead centre position (T.D.C.). the fuel is in a very fine spray rapidly injected into the cylinder and it then ignites.

During combustion the pressure climbs to ± 200 bar (maximum combustion pressure) and the temperature rises to ± 1600 °C (maximum combustion temperature).
In the course of the combustion – or power stroke the pressure decreases and the exhaust valves subsequently open at 6 bar, the temperature of the exhaust gases are now about 700 °C.
At this time the pressure in the exhaust pipe amounts to ± 2 bar and the temperature of the exhaust gases 450 °C.

Of course each modern four-stroke diesel engine motor has its own cycle and accompanying pressures and temperatures that diverge from the aforementioned.



◄ The P–V-diagram for a four-stroke diesel engine.

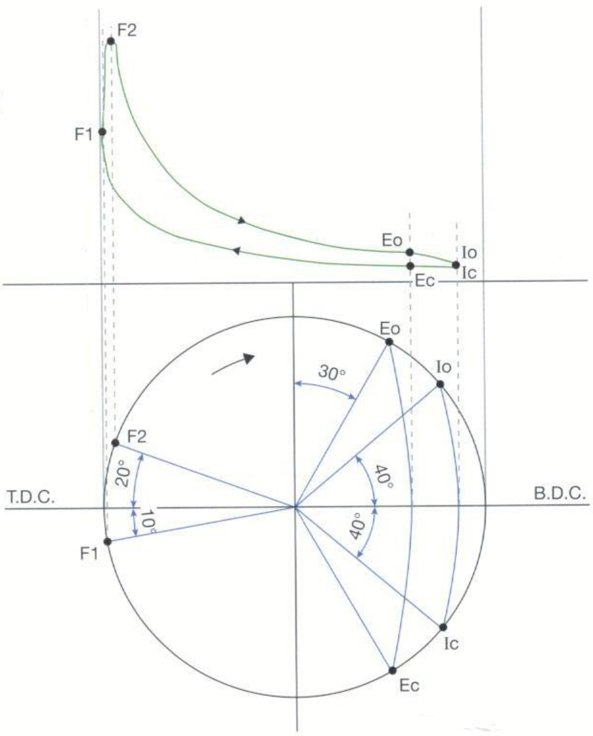
In diagram: starting on the right side
After the inlet valve is closed, the pressure increases (Ic)
Fuel is injected at (F1) until the power stroke commences at (F2)
The exhaust valve opens at (Eo) until the exhaust stroke commences at (Ec)
Exhaust gas scavenging from (Io) to (Ec), followed by the inlet stroke

The work done is positive power (a clockwise work cycle) and is the internal power of the cylinder. (heat engine)

Three strokes in this process use energy from the crank shaft; the intake stroke, the compression stroke and the exhaust stroke. This is negative power.
Only the power stroke supplies energy to the crank shaft. This is positive power.

4.2.2 Two-stroke process

Let's assume that the piston is in B.D.C. and commences its upward stroke. From the moment the piston passes the intake ports and the central exhaust valve is shut, compression begins. Pressure increases to approximately 2 bars over pressure super charging with turbo blower) to about 100 bar (final compression pressure). The air temperature has risen from van ± 50 °C (scavenge air temperature after the cooler) to ± 650 °C (final compression temperature).
Right before T.D.C. is reached the fuel is rapidly and finely distributed and injected into the cylinder and ignites.
During combustion the pressure climbs to ± 150 bar (maximum combustion pressure) and the temperature to 1500 °C (maximum combustion temperature).



▶
The P–V-diagram for a two-stroke diesel engine.

In the course of the combustion - or power stroke the pressure further decreases and the exhaust valves subsequently open at approximately 4 bar. The exhaust gas temperature is then about 600 °C. At this time the pressure in the exhaust pipe is ± 2 bar and the exhaust gas temperature ± 375 °C.

Of course, each modern two-stroke engine has its own cycle and accompanying pressures and temperatures that diverge from the aforementioned.

The work done is positive power (a clockwise work cycle) and is the internal power of the cylinder. (heat engine)

In this process one stroke uses energy (the compression stroke = negative power) and one stroke supplies energy (combustion stroke = positive power/energy) to the crankshaft.

Scavenging process

When the piston releases the intake ports, the scavenging process takes place at which point the scavenging air pressure should exceed the pressure of the exhaust gases in order to prevent reflux of the exhaust gases.

Note

A diagram of an engine with intake ports and one central exhaust valve had been made for the two-stroke process. Two-stroke industrial crosshead diesel engines with intake and exhaust ports still exist, but are no longer manufactured.

4.2.3 A comparison of both processes

At an identical RPM of the crank shaft a two-stroke engine of the same sized cylinder has twice the power strokes as that of a four-stroke engine. Theoretically, a two-stroke engine should have twice the power output of a four-stroke engine at the same RPM and identical cylinder content. However scavenging and filling for a two-stroke engine is less efficient than that of a four-stroke engine, which means less fuel is burnt. The power output ratio of four-stroke to two-stroke engines at the same RPM and cylinder volume lies just below 2: 1.6 to 1.7.
The thermal load of the material of the components of the two-stroke engine is higher than that of the four-stroke engine, so the time between two power strokes, the hottest part of the process, is reduced by half.

Cylinder scavenging in a two-stroke engine is more complex, since it is only the over pressure of the scavenging air in relation to the exhaust gases that determines the scavenging, as opposed to the four-stroke engine where a piston draws in air and forcibly expels the exhaust gases. Therefore the mean piston speed for a two-stroke engine is set slightly lower (6 to 8 m/sec. for two-stroke and 8 to 11 m/sec. for four-stroke) in order to avoid scavenging problems.

Generally, all medium-and high speed diesel engines are four-stroke engines. These can effortlessly achieve a high mean piston speed as well as a high mean effective pressure without having the problems that gas exchange or thermal load of the components can cause. Moreover, the film of lubrication oil on the piston and the cylinder liner is better maintained in a four-stroke engine.

4.3 Parameters of both working principles

The parameters of both working principles differ considerably.

4.3.1 Mean effective pressure: p effective mean – p_E

For the latest two-stroke-crosshead engines this is approximately 20 bar and for the latest four-stroke trunk piston engines about 28 bar. This disparity is predominantly caused by that fact that there is less fresh air for combustion in the cylinder; as a consequence less fuel can be injected per combustion stroke. This ultimately results in a lower mean effective pressure for two-stroke engines.

4.3.2 Mean piston speed: C_P mean

For large two-stroke engine this is around 8.5 m/sec. and for the four-stroke engine 10 m/sec. The mean piston speed is an important factor with regard to the thermal and mechanical load of the various engine components.
Due to the fact that the lubrication of, among other things, the piston rings is slightly lower in two-stroke engines as they constantly pass the ports which is not conducive to cylinder lubrication. Therefore the piston speed is lower and of course in order to obtain optimum scavenging, the piston speed is also set slightly lower.

For each engine this is connected to a maximum value.

C_P mean = 2 × S × n
C_P mean = mean piston speed in m/sec.
D = intake stroke in m
n = number revolutions per second (RPS)

Conclusion
The shorter the stroke of the engine, the higher the achievable RPM!

4.3.3 Load parameters L_P, product of the mean effective pressure and the mean piston speed

See also 4.3.1 and 4.3.2.

$L_P = p_E \times C_P$

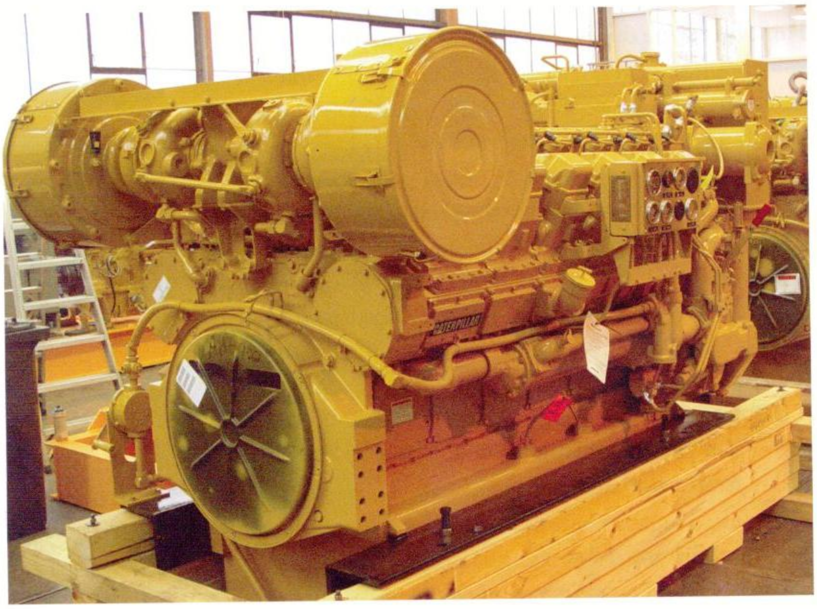
Example:

For the two-stroke cycle	$L_P = 19 \times 8.5 =$ 161.5 bar/m/sec.
For the four-stroke cycle:	$L_P = 25 \times 10 =$ 250 bar/m/sec.

4.3.4 Exhaust gas temperatures

For the two-stroke cycle:	Tgas = 325 to 375 °C
For the four-stroke cycle:	Tgas = 400 to 500 °C

▼ A large two-stroke engine and next to it, a four-stroke trunk piston engine.



Example 1

Set stroke = 0.2 m.
Mean piston speed 10 m/sec.

C_p mean $= 2 \times S \times n$
$10 = 2 \times 0.2 \times n$
$0{,}4\ n = 10$
$n = 25\ \text{RPS} \rightarrow 25 \times 60 = 1500\ \text{RPM}$

This is a high-speed engine four-stroke cycle.

◄
Fast running four-stroke Caterpillar diesel engines ready to be sent to genset manufacturers or companies for assembly in propulsion systems.

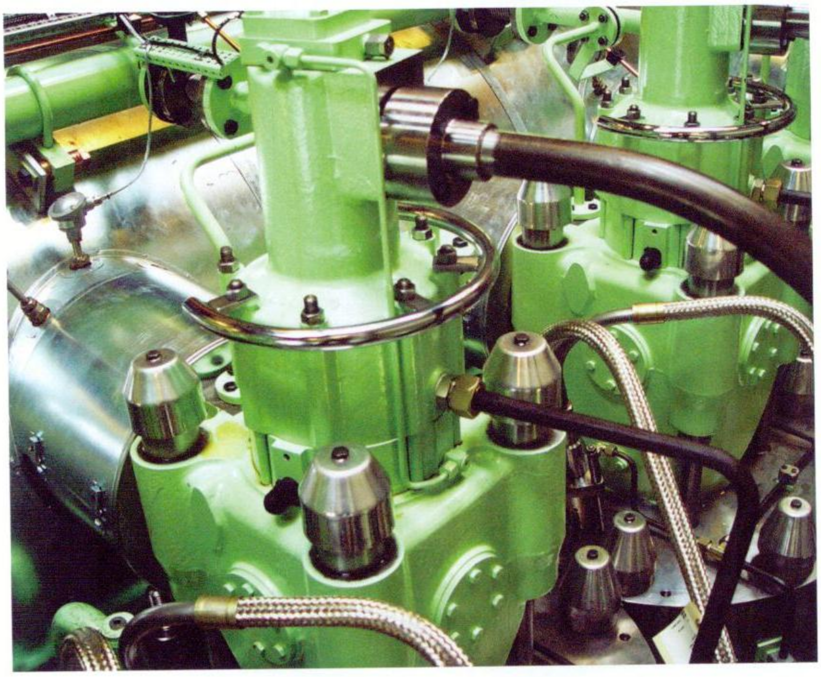
Example 2

Set stroke = 0.6 m.
Mean piston speed 9.6 RPS

C_p mean $= 2 \times S \times n$
$9.6 = 2 \times 0.6 \times n$
$1{,}2\ n = 9.6$
$n = 8\ \text{RPS} \rightarrow 8 \times 60 = 480\ \text{RPM}$

This is a medium-speed engine four-stroke cycle.

◄
A medium-speed four-stroke H.F.O. diesel engine driving a generator on a large container ship.



Example 3

Set stroke = 2 m.
Mean piston speed 8 m/sec.

C_p mean $= 2 \times S \times n$
$8 = 2 \times 2 \times n$
$4\ n = 8$
$n = 2\ \text{RPS} \rightarrow 2 \times 60 = 120\ \text{RPM}$

This is a low-speed engine two-stroke cycle.

◄
Cylinder heads of a large two-stroke crosshead engine for ship propulsion.

Note the hydraulically tightened bolts for the exhaust valves and the cylinder covers. Here the cylindrical nuts are covered with protective caps.

4.4 Determining cylinder output using an indicator diagram and the mean induced pressure

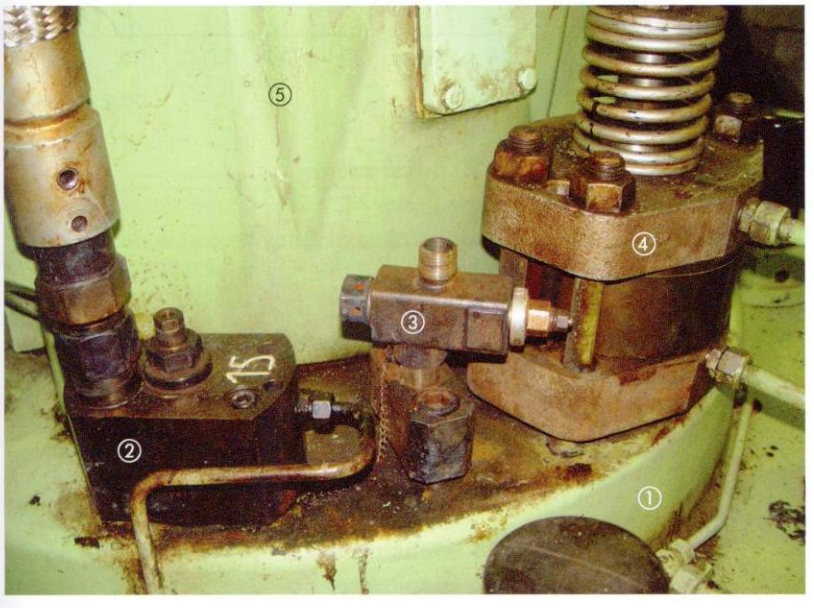
This is still done in the traditional way, but today usually by means of an electronic pressure gauge on the cylinder which computes all necessary values using a dedicated software program.

4.4.1 With the indicator gauge

◄

An indicator gauge.

1. *sleeve nut on indicator cock*
2. *piston casing*
3. *indicator spring (pressure spring)*
4. *lock pin*
5. *lever system with needle*
6. *indicator drum with indicator paper*
7. *indicator drive belt for drum*



◄

Part of the cylinder cover of a two-stroke crosshead engine.

1. *cylinder cover*
2. *one of the two fuel valves*
3. *indicator cock for making a P–V-diagram*
4. *safety valve*
5. *exhaust-valve casing*

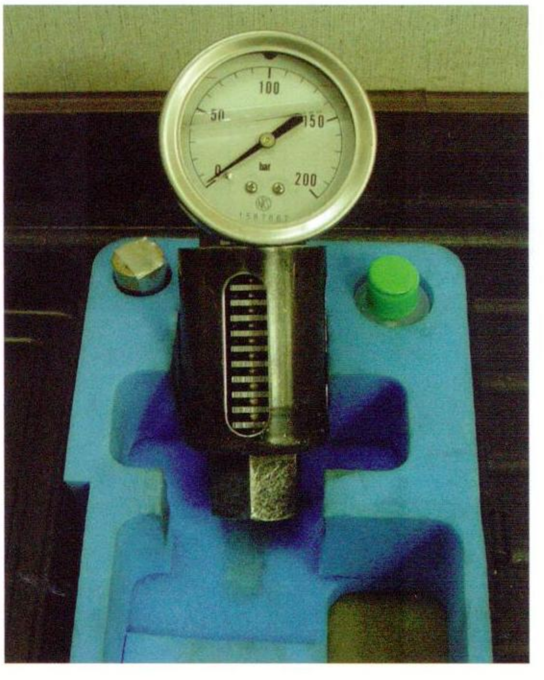
◄

An indicator.

1. *pressure-measuring staffs*
2. *extra spring*

▼

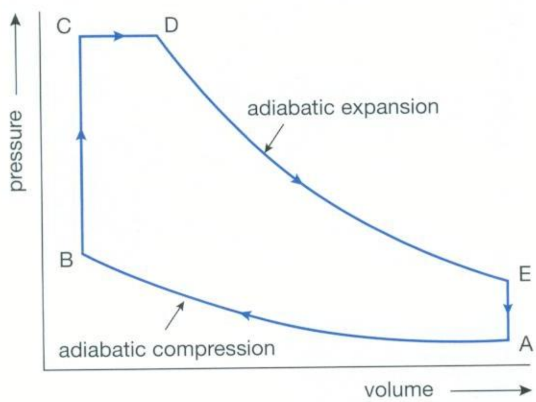
A pressure gauge for measuring the maximum combustion pressure in the cylinder.



4.4.2 Determining cylinder output using an indicator diagram

A P–V-diagram represents the working cycle of an internal combustion engine; in this case a diesel engine.

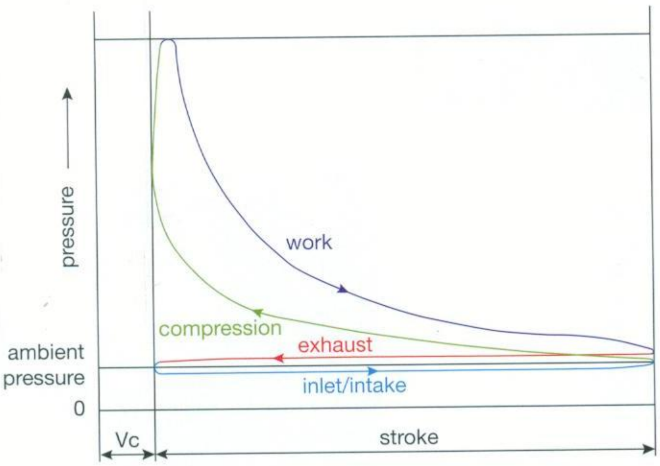
Theoretical diagram



▲ The extent of the clockwise turning process is indicative of the amount of positive power exerted on the piston.

The enclosed surface A–B–C–D–E–A is the resultant positive energy.

Practical diagram



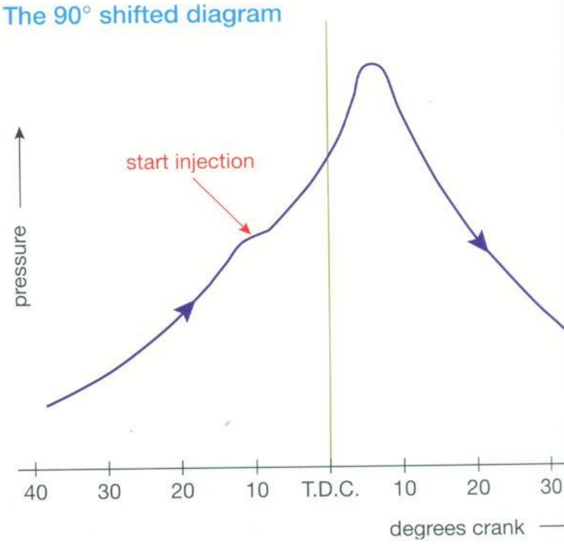
▲ The actual pressure development in the cylinder of a four-stroke engine as measured by an indicator.

Horizontal axis, the piston position and the vertical axis, the pressure development. Here the exhaust-gas pressure slightly exceeds the inlet-air pressure at the exhaust and inlet stroke respectively.
This is a diesel engine without turbo charger.

Theoretical and practical diagram

The mean induced pressure on the piston is no different from the measured (or calculated) pressure throughout the entire process.

The 90° shifted diagram



▲ The 90° shifted diagram.

This clearly shows the initial moment of fuel injection.

4.5 Determining the mean induced pressure

There are three ways in which to determine the mean induced pressure in a P–V-diagram.

4.5.1 Measure the diagram with a planimeter

With a measuring instrument, the planimeter, the diagram surface area is meticulously calculated in mm² and divided by the length of the diagram in mm.
Mean height is calculated using the formula below.

$$\text{mean height in mm} = \frac{\text{surface area diagram in mm}^2}{\text{length diagram in mm}}$$

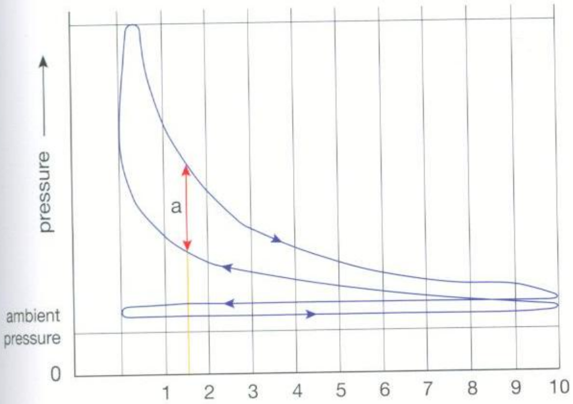
The calculated mean height in mm is divided by the linear spring scale and so the mean induced pressure can calculated.

$$\text{mean induced pressure in bar} = \frac{\text{mean height in mm}}{\text{spring scale mm/bar}}$$

To warrant accuracy one takes three planimeter measurements.

4.5.2 Oldest method, division of the acquired diagram into ten equal sized sections

The classical method in determining the mean induced pressure from a P–V-diagram: split the diagram into ten pieces of the same size. Determine the height of each part. This is roughly the height in the middle of one part.
The sum of the average heights (ten parts) and divided by 10 give the mean height in mm.
By dividing this number by the spring scale the mean induced pressure in bars is obtained.



▲
Determining the mean induced pressure from a indicator diagram.

4.5.3 Modern method, electronic

Here a pressure sensor is connected to the indicator. All cylinder data are passed on to an engine collecting unit. This collects all information such as pressure development, crank shaft position, rpm etc. All data is then processed by a central computer.

The possibilities are extensive. Here a some examples of the values that can be established:
- mean induced pressure;
- maximum cylinder pressure;
- induced power output;
- torsional vibration;
- exhaust gas pressure;
- general trends.

In this all important data of the engine are available at any given moment.

See also Chapter 25, Operational management and automation.

4.6 Engine formula

The internal and external power output of the engine can be calculated with the aid of the engine formula. The following applies to a combustion engine:

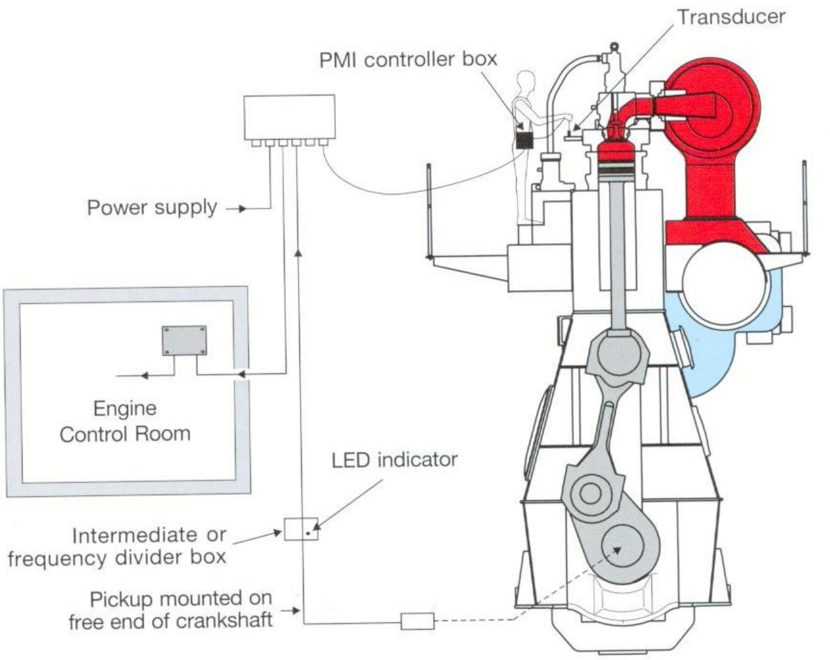
$$P_I = \frac{\pi}{4} \times D^2 \times S \times \frac{n}{a} \times Z \times p_I$$

P_I = induced power output in MW (Megawatt)
D = piston bore in m (meters)
S = piston stroke in m (meters)
n = number of revolutions of the crank shaft per second
a = 1 for two-stroke, a = 2 for four-stroke
Z = number of cylinders
p_I = induced mean pressure in MPa (Megapascal)

As a rule, pressure in engine technology is expressed in bar: **1 MPa =10 bar.**

The engine power output is usually often expressed in kW: **1 MW =1000 kW.**

▼
An overview of a modern measuring system mounted on a two-stroke crosshead engine.



The PMI System as applied on a two-stroke diesel engine

Example

How to calculate the induced (power) output of an engine by using an indicator gauge and a planimeter.

Two-stroke crosshead engine

number of cylinders	8
cylinder bore D	= 700 mm
piston stroke S	= 2360 mm
number of revolutions	
108 per minute	= 1.8 RPS
Spring scale 1 bar	= 0.3 mm
Surface area diagram	= 399 mm²
length diagram	= 70 mm

Calculate:

a The mean effective pressure in bar and MPa.
b The mean piston speed in m/sec.
c The mean induced (power) output provided that the induced mean pressure of all cylinders is identical.

Solution

a The mean height of the indicator diagram is:

$$h_M = \frac{399}{70} = 5.7 \text{ mm}$$

For a spring scale of 1 bar = 0.3 mm the mean induced pressure is now $\frac{5.7}{0.3} = 19$ bar

Or 1.9 MPa.

b The mean piston speed is:

C_p mean = 2 × S × n = 2 × 2.36 × 1.8 = 8.5 m/sec.

c The induced power output:

$$P_I = \frac{\pi}{4} \times D^2 \times S \times \frac{n}{a} \times Z \times p_I$$

$$P_I = \frac{\pi}{4} \times 0.7^2 \times 2.36 \times \frac{1.8}{1} \times 8 \times 1.9$$

P_I = 24.840 MW of 24,840 kW

This is an example of a MAN–B&W two-stroke crosshead engine type 8L70 ME – C.

Additional information:

length	11.35 meter
height	10.25 meter
width	4 tot 6.5 meter
weight (dry)	642 ton

Fuel consumption per day 101,350 kg or 101.4 ton weight.

Note

It is not possible to use the indicator gauge for high-speed diesel engines. It is generally used for low-speed two-stroke crosshead engines in order to calculate the output- compression- and combustion pressures.
For higher rotating four-stroke diesel engines it is/was merely used to measure the compression- and combustion pressure.

4.7 Induced thermal efficiency

Induced thermal efficiency is also called 'the useful effect'.
The formula in words is:

$$\text{Induced thermal eff} = \frac{\text{Induced engine output}}{\text{Fuel power input}}$$

Induced engine power	P_I (MW)
Fuel consumption	F_C (kg/sec)
Fuel energy density (also heat value)	H_O (MJ/kg)
Fuel power input	P_F (MW)

$$\eta i = \frac{P_I}{F_C \times H_O}$$

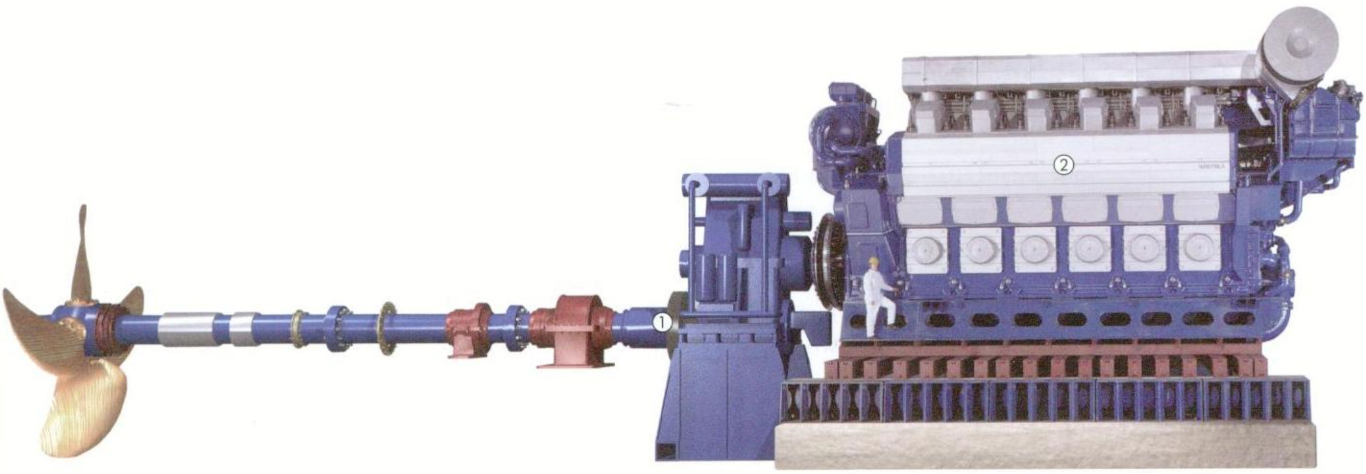
ηi = eta internal = the internal efficiency.

This ranges from about 25 to 55% for a four-stroke engine and for two-stroke engine is slightly over 65%. The remainder of the fuel energy is wasted in the form of heat.

4.8 Mechanical and total efficiency

In practice we prefer to work with a slightly more practical efficiency, namely the total efficiency or shaft efficiency of an engine.
Engine manufacturers always present the maximum engine power output as the maximum continuous capacity or shaft power.

A percentage of the induced power is not transferred to the crank shaft as useful shaft power, but instead is required to overcome frictional forces. Naturally, this is just a very small percentage as a far larger percentage of the induced power is required to drive the high



▲ The cylinder and shaft power.

1 *shaft power-P_E*
2 *cylinder of internal power-P_I*

pressure fuel pumps, the intake- and exhaust valves and the lubricating- and cooling water pumps.
For a modern diesel engine this amounts to about 8 to 10% of the induced power in total. Therefore the mechanical efficiency is approximately 90 to 92%.

The mechanical efficiency equals:

$$\frac{\text{the shaft power}}{\text{the induced/internal power}} \text{ or } \frac{P_E}{P_I}$$

The mechanical efficiency η mech. therefore is:

$$\eta \text{ mech.} = \frac{\text{effective power or shaft power on the crank shaft}}{\text{induced power}}$$

$$\eta \text{ mech.} = \frac{P_E}{P_I}$$

$$\text{Also: } \eta t = \eta_m \times \eta_i = \frac{P_I}{F_C \times H_L} \times \frac{P_E}{P_I} \text{ of } \eta t = \frac{P_E}{F_C \times H_L}$$

4.9 Specific fuel consumption

This is the amount of fuel consumed per power unit and per time unit. It is normally indicated in grams per kilowatt.
The lower the fuel consumption, the less fuel the engine uses and the higher the engine's 'return efficiency' is. In this way, one can easily make comparisons of engines.

Lubricating oil- and fuel consumption constitute by far the highest costs in ship exploitation.

Of course the measurement of fuel consumption must take place under the same conditions; these have been laid down in the ISO norm 3046 - 1 - 1995.
This stipulates amongst others:
- the intake air temperature of 25 °C;
- the temperature after the air cooler of 25 °C;
- the ambient pressure 1000 millibar;
- relative humidity of 30%;
- the heat value of the fuel.

All engine manufacturers provide these details when stating fuel consumption.
The fuel consumption in conjunction with the lubricating oil consumption (in grams per kWh) form a major consideration in decision of which engine to chose.
The fuel consumption varies from 167 gr/kWh. to 220 gr/kWh. and depends for instance on the size of the diesel engine. The larger the engine the higher the efficiency, so the lower the fuel consumption.

4.10 Mean effective pressure

We already know:

$$\text{mechanical efficiency} = \frac{\text{the shaft power}}{\text{the induced/internal power}}$$

$$\eta \text{ mech.} = \frac{P_E}{P_I}$$

$$\text{Thus follows: } \eta \text{ mech.} = \frac{\text{p eff. mean}}{\text{p ind. mean}};$$

$$\text{so also: } P_E = \frac{\pi}{4} \times D^2 \times S \times \frac{n}{a} \times Z \times \text{p eff. mean}$$

This is the more practical formula with regard to diesel engines.
Engine manufacturers also always provide the effective power output.

The difference between the induced and the effective mean pressure is called the mean frictional pressure p_F.

So: $p_F = p_I - p_E$

$$\text{Moreover}: P_F = \frac{\pi}{4} \times D^2 \times S \times \frac{n}{a} \times Z \times p_F$$

Therefore the frictional work of the engine is also:

$P_F = P_I - P_E$.

Measurements show that the (frictional) torque and therefore also the frictional pressure are barely influenced by the number of revolutions (RPM) and engine load. Consequently, the mean frictional pressure for an engine can be taken as a constant.

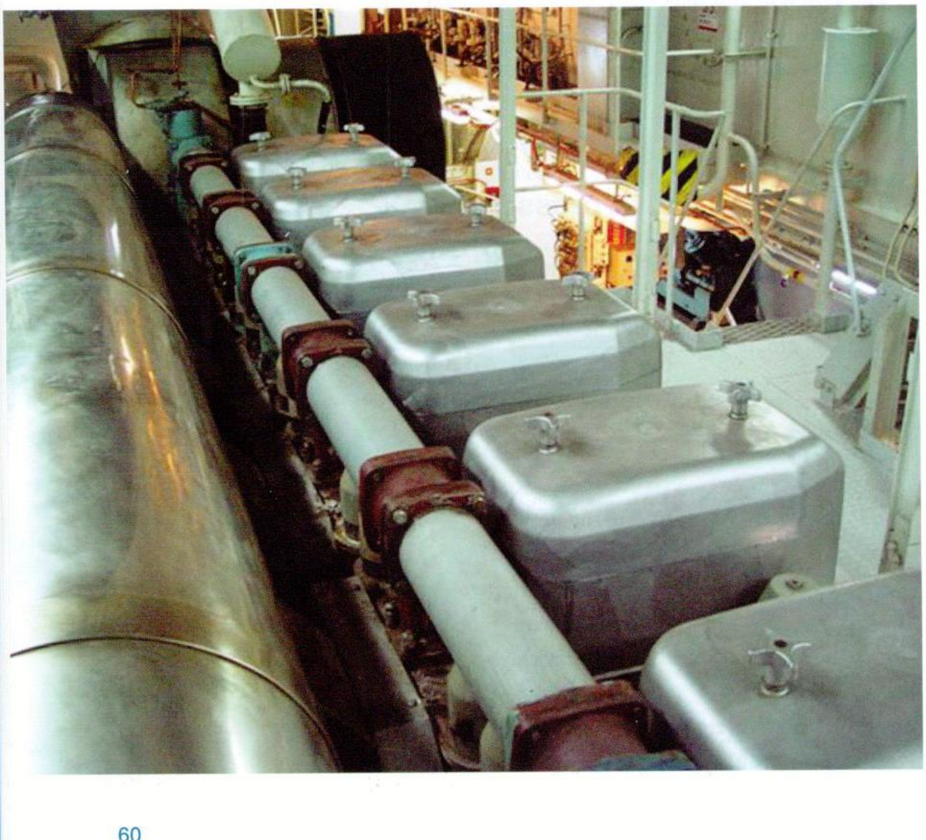
Overall, engine load is proportional to P_E and so to the generated power per cycle it follows that the engine output is only proportional to the load at a constant rpm.
It is also given that:

$$\eta\ \text{mech.} = \frac{P_E}{P_E + P_F} = \frac{p_E}{p_E + p_F} = \frac{1}{1 + \frac{p_F}{p_E}}$$

▼
One of the two Wärtsilä 9L 46 B four-stroke diesel engines on the 'Oranjeborg' of Wagenborg Shipping.

Example

At full load an engine at constant RPM runs with a mean effective pressure of 25 bar and a mean induced pressure of 28 bar.



The mean frictional pressure is:
$p_F = p_I - p_E = 28 - 25 = 3$ bar.

The mechanical efficiency at full load of this engine is:

$$\eta\ \text{mech.} = \frac{1}{1 + \frac{3}{25}} = 0.8928 \text{ of } 89.3\%$$

The mechanical efficiency drops slowly if this is calculated for a partial load.

At 75% power output p_E is= 0.75 × 25 = 18.75 bar and the mechanical efficiency is 86.2%.
At 50% power output p_E is = 0.50 × 25 = 12.0 bar and the mechanical efficiency is 80%.
At 25% power output p_E is = 0.25 × 25 = 6.25 bar and the mechanical efficiency is 67.6%.
So the mechanical efficiency falls slowly at a reduced engine power output.

4.11 Thermal energy balances or Sankey-diagrams

An energy balance or Sankey-diagram indicates how the thermal energy in heat supplied by the fuel next to the delivered shaft power is distributed over varied systems.
Exhaust gas loss is always the greatest loss, followed by the air cooler loss, or jacket coolant loss and lubricating oil loss. Furthermore, there is a radiation loss and some residual loss. The following examples are two overviews of a four-stroke trunk piston engine and a two-stroke crosshead engine.

An example of a thermal energy balance I

The Wärtsilä 9 L 46 B has the following values as detailed in the Project Guide Marine Applications:

Four-stroke cylinder bore	460 mm
Stroke	580 mm
Cylinder output at 514 revolutions	975 kW
Mean effective pressure	23.6 bar
Mean piston speed	9.9 m/sec.
Number of revolutions	514 rpm
Shaft power	8775 kW
Amount of combustion air	15.8 kg/sec.
Exhaust gas temperature after turbocharger	375 °C (C.C.P.)
Amount of exhaust gases	16.2 kg/sec.
lubricating oil cooling	1150 kW
cylinder cooling water cooling	950 kW
air cooler high temperature (H.T.)	1500 kW

air cooler low temperature (L.T.)	990 kW
Radiation heat	340 kW
Fuel consumption (C.C.P.)	173 gram per KWh

C.C.P. = controllable pitch propeller installations

Heat value fuel (calorific value) 42.777 MJ/kg (Iso-norm). Intake air temperature to the engine 25 °C (Iso/norm).

Specific heat of exhaust gases	1.04 kJ/Kg. K.
Specific heat of the air	1.0 kJ/Kg. K.

This information enables us to set-up a thermal energy balance.

Calculation

The fuel consumption is 0.173 × 8,775 = 1,518 kilogram per hour or $\frac{1518}{3600}$ = 0.4216 kg/sec.

It follows that the supplied fuel output is: 0.4216 × 42,700 = 18,006 kW. In a thermal energy balance this normally comprises the total energy supply, combustion air thermal energy supply not included.

The volume flow of the exhaust gasses is 16.2 kg/sec.

The exhaust gas thermal energy after the turbo blower is m gas (kg/sec.) × q. spec. gas (kJ.kg.K) × t gas (°C) and subsequently: 16,2 × 1,04 × 375 = 6318 kW.

The combustion air thermal energy to the engine is 15.8 × 1,0 × 25 = 395 kW.

The net loss with the exhaust gasses is 6318 – 395 = 5923 kW.

In summary

supplied with the fuel		18,006 kW
thermal energy of the combustion air		395 kW
total supplied	18,006 + 395 =	18,401 kW
exhaust gas loss	5923 kW	32,18%
absorbed by lubricating oil	1150 kW	6.24%
absorbed by cylinder cooling water	950 kW	5.16%
air cooler low temperature	990 kW	5.38%
radiation heat	340 kW	1.84%
effective power	8775 kW	47.68%
residual loss	273 kW	1.48%
total	18,401kW	100%

The high exhaust gas losses (approximately a third) is remarkable together with the high total efficiency of this relatively large four-stroke trunk piston engine.

This engine weighs 134 tons and is 10.3 meter long, 3.3 meter wide and 4 meter high.

An example of a heat balance II

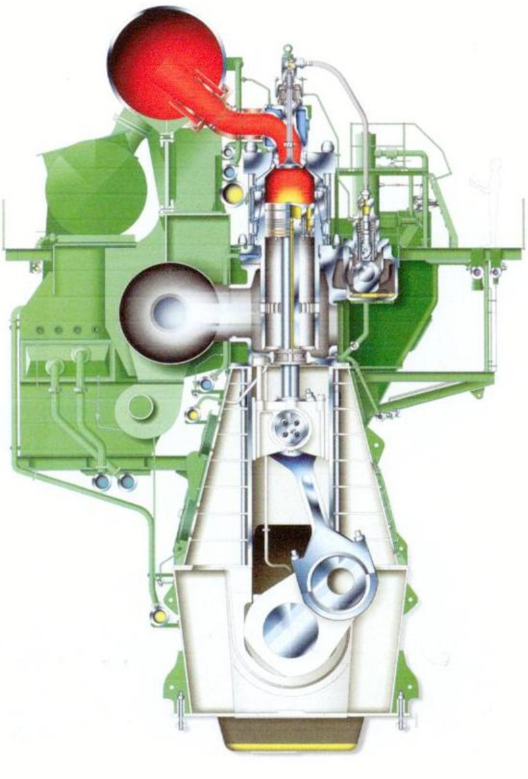
MAN–B&W-K98 – MC – twelve cylinder

Some data

cylinder bore	980 mm
stroke MC-version	2660 mm
total power output	80,080 kW
number of revolutions	94 rpm
mean effective pressure	18.2 bar
mean piston speed	8.3 m/sec.
fuel consumption	171 g/kWh
air cooler	19,416 kW
lubricating oil cooler	6770 kW
cylinder cooling water cooler	11,600 kW
amount of exhaust gases 754,600 kg/h = 209.6 kg/sec.	
amount of combustion air	205.8 kg/sec.
fuel consumption	3.8 kg/sec.
exhaust gas temperature after turbo blower	245 °C

heat value fuel (calorific value) 42.700 MJ/kg (Iso-norm)

intake air temperature to the engine 25 °C (Iso-norm)



◄ A twelve-cylinder MAN–B&W-K98 – MC – Engine.

This is the second largest two-stroke crosshead engine in the world; only the1080 mm bore version is larger.

The engine is delivered with six to fourteen cylinders.

specific heat of the exhaust gases	1.04 kJ/kg. K.
specific heat of the air	1,0 kJ/kg. K.

With this information we can set up a thermal energy balance.

Calculation

The fuel consumption per second is:

$$\frac{0.171 \times 80{,}080}{3600} = 3.8038 \text{ kg/sec.}$$

The supplied fuel output is:

3.8038 × 42,700 = 162,422 kW

In a thermal energy balance this normally comprises the total energy supply, combustion air thermal energy supply not included
The volume flow of the exhaust gases is: 205.8 kg/sec.
The heat of the exhaust gases after the turbo blower is:

m.gas (kg/sec.) × q. spec. gas (kJ.kg.K) × t gas (° C).

Therefore 205.8 × 1.04 × 245 = 52,437 kW.
The heat in the combustion air to the engine is: 205.8 × 1.0 × 25 = 5145 kW.

So the net loss with the exhaust gases is:
53,406 – 5145 = **48,261 kW.**

In Summary:

supplied with the fuel	162,422 kW
heat of the combustion air	5,145 kW
total supplied to the engine	**167,567 kW**

exhaust gas loss	48,261 kW	28.2 %
absorbed by lubricating oil	6,770 kW	4.0 %
absorbed by cylinder cooling water	11,600 kW	6.9 %
air cooler	19,416 kW	12.6 %
radiation heat (1.1% van Pe)	880 kW	0.5 %
effective power	80,080 kW	47.8 %
residual loss	560 kW	0.03%
total	**167,567 kW**	**100.0%**

4.12 Efficiencies of diesel-engine driven power plants

4.12.1 Ship propulsion

Most ships are propelled by screw propellers. The propulsion efficiency of a screw propeller is not 100%, as a screw propeller 'slips' in water. For this reason the actual shaft power made available for ship propulsion amounts to 60 to 70% of the power supplied to the propeller shaft by the diesel engine.
Therefore: the power available for ship propulsion is a part of the power supplied to the engine in the form of fuel.

▶ Ship propulsion by diesel engines has been the most effective propulsion method for over a hundred years.

P propeller = P fuel × η engine × η propeller

Let's assume that the total efficiency of the engine is 45% and propeller efficiency 60%, than the propeller output is a mere 27% (0.60 times 0.45) of the fuel power output.

Where are the losses?
1 Heat losses in the engine; 55% of P fuel.
2 Screw slippage, 40% of the shaft power delivered by the engine (P eff.).

See Chapter 19, Ship propulsion and Chapter 31, Propellers.

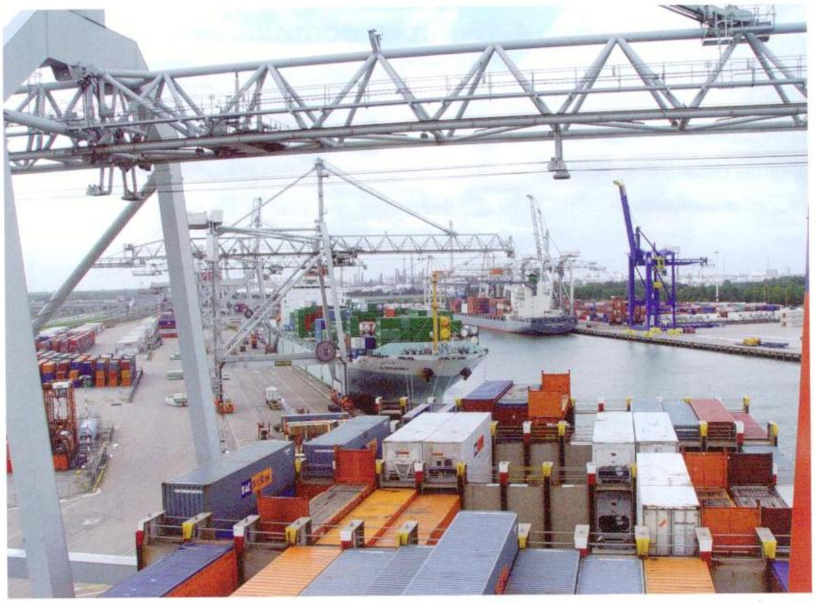
4.12.2 Gensets for ships and diesel power plants

Apart from the engine losses, diesel engine power plants or as in ordinary diesel gensets, there are generator losses. In general these are very small. The efficiencies of generators vary from 95 to 98% (100 to 10,000 kW), which means:

P elec.generator = P fuel × η engine × η gen.

Let's assume that the total efficiency of a low-speed crosshead engine in a diesel power plant is 48% and the generator efficiency 96%. The generated electric efficiency from the supplied fuel is:

0.48 × 0.96 = 0.46 or 46%.

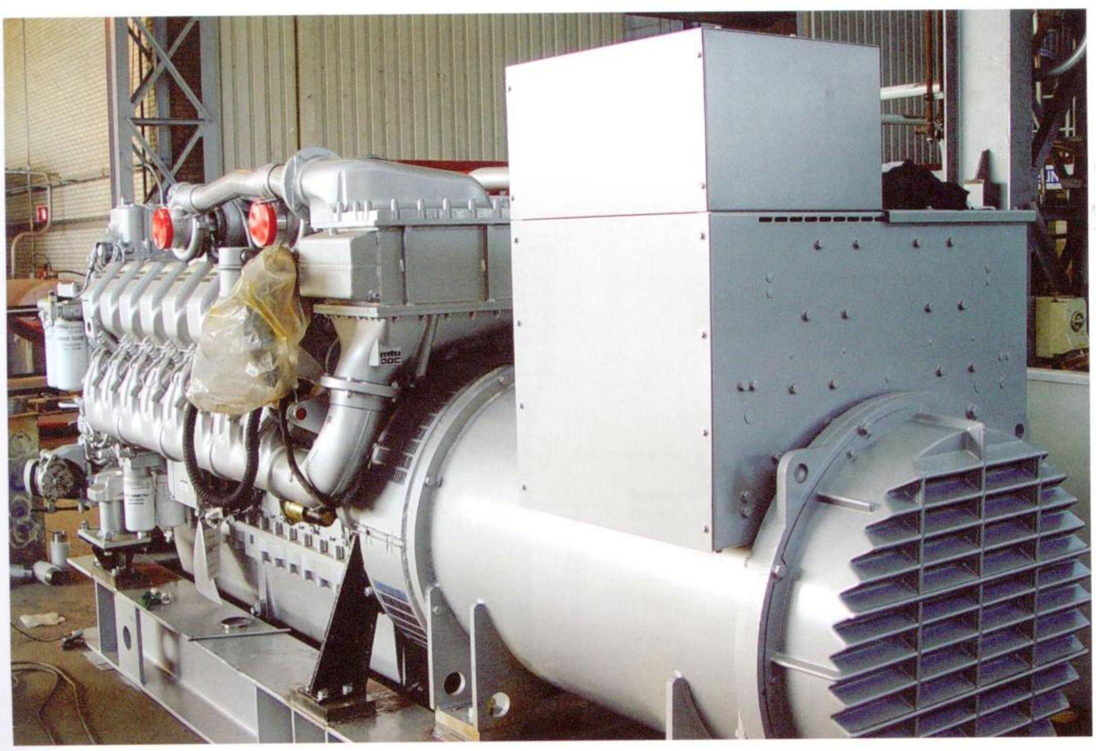


▲
All types of ships for container carriage.

The power required for power generation and propulsion is virtually always supplied by diesel engines.

Clearly this is much higher than the ship propulsion efficiency!

Where are the losses?
1 The total engine losses; 52% of the fuel power output.
2 The copper losses in the generator; these are heat losses, a mere 4% of the diesel engine's shaft power. The losses due to the friction of the bearings and the power required for driving the generator's scavenging air ventilator are minor.



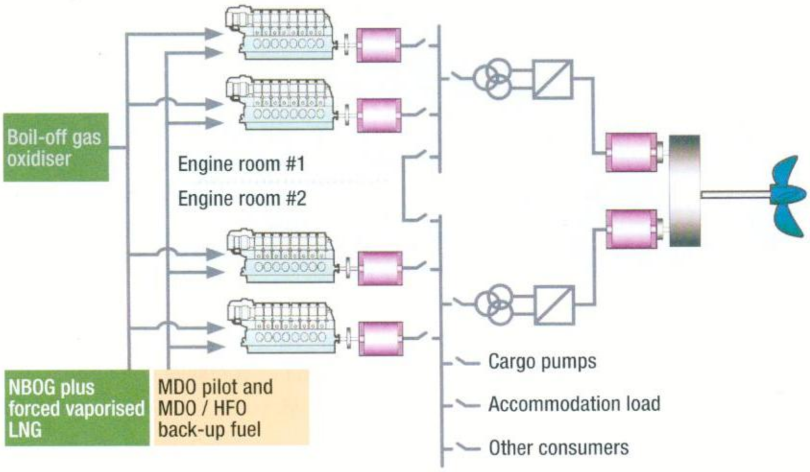
◄
A modern genset with a high-speed four-stroke V-diesel engine, category II.

▼
A diagram of diesel-electric propulsion with two rotary driven electric engines driving the propeller via the gear box.

4.13 More complex ship propulsion

4.13.1 Diesel-electric propulsion

Diesel-electric propulsion is a technology that was developed many years ago.

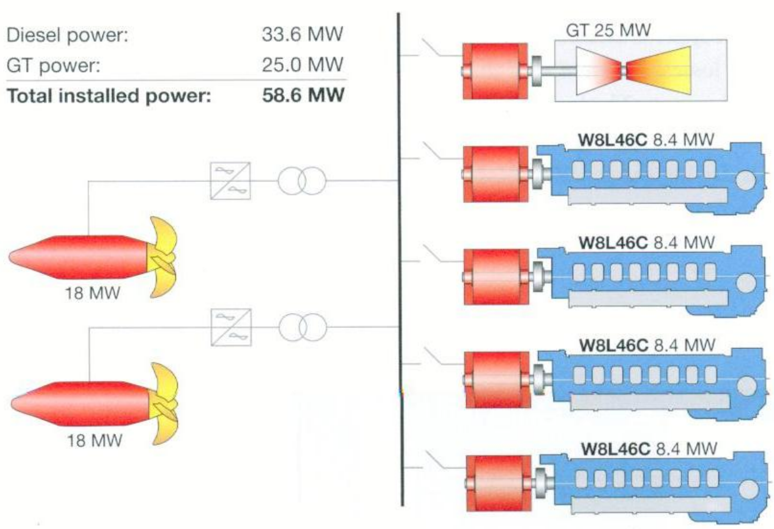


This technology uses diesel generator sets to provide electric power to a distributing panel (rail). The propeller are therefore driven by an electromotor. These generators often generate a medium voltage of, for instance, 6.6 kilovolt which is why the copper diameter or the cable diameter between the various components is acceptable.
The total propulsion efficiency from the supplied fuel is :
η propulsion = η diesel engine × η generator × η electromotor × η propeller

Example
η propulsion = 0.45 × 0.95 × 0.95 × 0.65
η propulsion = 0.2639 or 26.4%

4.13.2 Electric propulsion by means of 'PODS' or electric blade propellers

In this case the generator is driven by diesel engines or dieselengines and gasturbines, occasionally this includes a steam turbine.
The generated electric power is then made available to frequency controlled and reversible rotary current motors which drive the 'PODS', the fixed propeller or propellers, which are attached to the bottom of the ship. These are for example used on the latest passenger ships the **'Jewel of Norway' and the 'Queen Mary II'.**

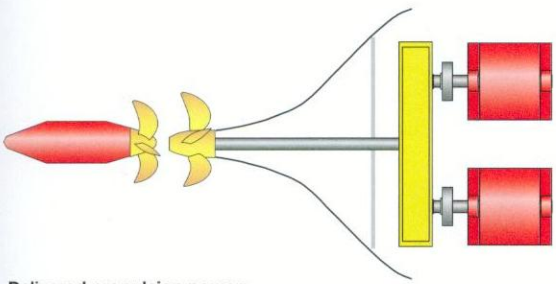


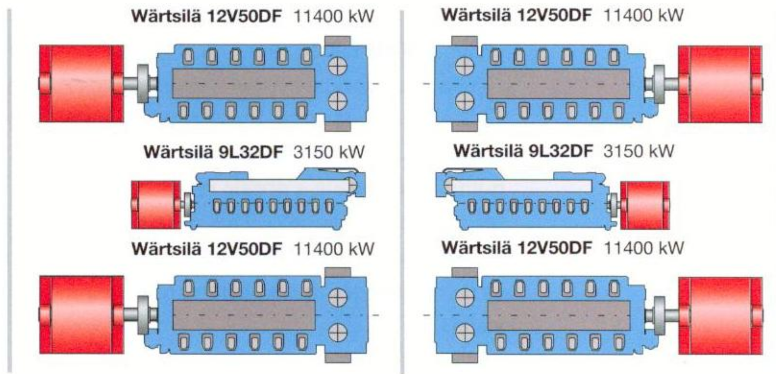
▲
The passenger liner is powered by two 360° rotatable PODS, each with 18 MW output.
PODS – podded propeller.

The required electric power is generated by four medium-speed Wärtsilä 8 L 46 C H.F.O. in-line engines of 8.4 MW each.
Gas-turbine genset with a 25 MW capacity using M.D.O. fuel can also be used for the ship's electricity network.

▼
The passenger liner 'Queen Mary II' is a good example of a diesel-/gas-turbine combination used for electric propulsion.

Total output 117 MW with four Wärtsilä 46-V-engines and one gasturbine. Propulsion by two fixed PODS and two azimuth thrusters.





Delivered propulsion power:

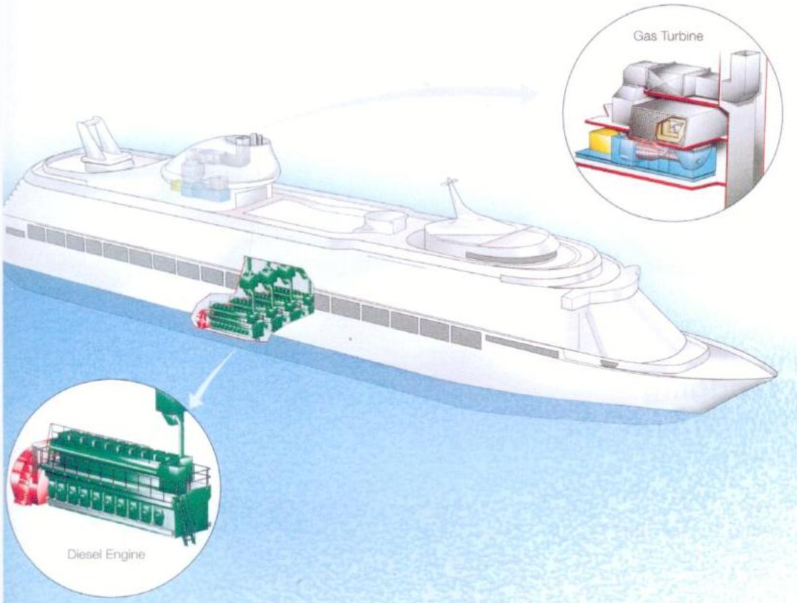
	Nominal	Service
Electric pod	17 MW 40 %	17 MW 41 %
LIPS CP propeller	25 MW 65 %	24 MW 62 %
Total shaft power	**42 MW**	**41 MW**

Installed engine power:

Mechanical transmission	0 MW
Electrical power generation	51.9 MW
Total installed power	**51.9 MW**

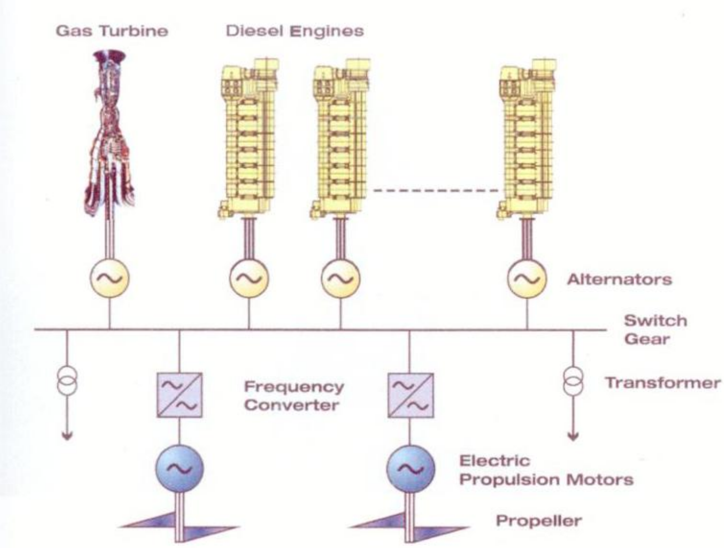
▲

Diesel-electric propulsion with four large and two smaller gensets for a controllable-pitch propeller and a contra propellor.



◄

A diesel-engine and gas-turbine combination used for electric propulsion.



◄

A propulsion system with three diesel engines and one gas turbine with two electrically driven propellers.

The gas-turbine genset has been mounted on the upper deck, so it can be replaced as a whole by an overhauled set during major repairs. The elevated position does not pose a problem for the ship's stability, since the gas turbines' weight, unlike the diesel engines', is low in relation to the generated power output.

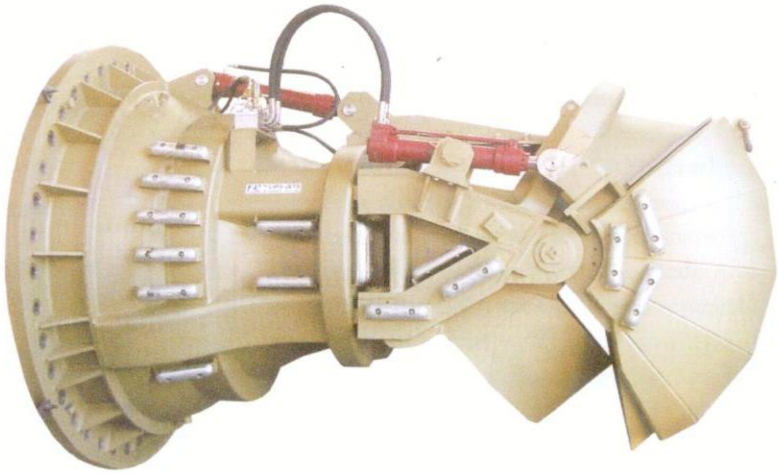
4.13.3 Propulsion by diesel engine driven water jets

This allows for several types of propulsion. A fast running four-stroke diesel engine drives a water pump which using a nozzle provide the thrust for the ship propulsion. This nozzle is adjustable and rotatable. It is often used for light ships such as fast passenger liners , catamarans and motor yachts.

For additional information about propellers see Chapter 19, Ship propulsion and Chapter 31, Propellers.

▲
A fast motor yacht with propeller propulsion.

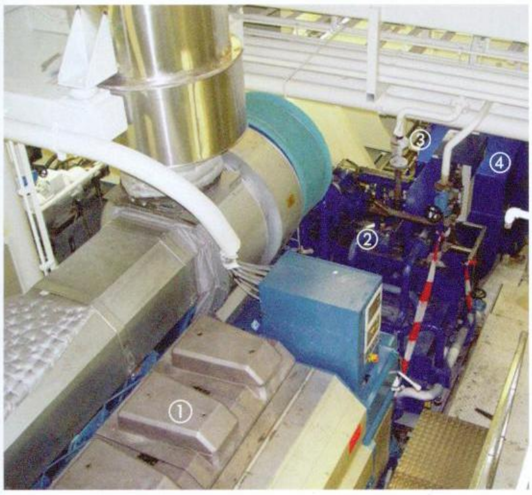
▶
A water jet with stream controls.



4.14 Water pumps, dredging pumps, crude-oil pumps, compressor drives

The most common drives are used in dredger pumps and other machines on large dredgers. They can be diesel-electrical driven- and a diesel engine driven pump. Since the dredger pump output varies significantly, the diesel engine load changes markedly and is often quite low.

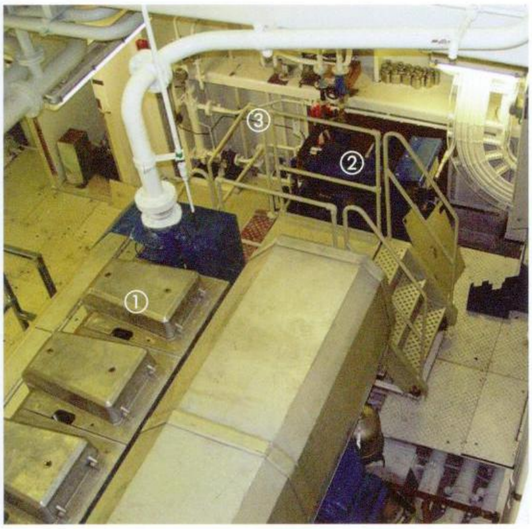
▶
On dredgers diesel engines are not only used for propulsion, but also for driving dredging pumps and water-jet systems for unloading cargo.



◄

The propulsion engine on a dredger in the foreground and back right the gear box for driving the propeller and the generator.

1 *engine*
2 *gear box*
3 *propeller shaft*
4 *generator*



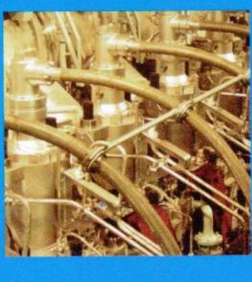
◄

The other side of the same propulsion engine with a manually controlled gear box for driving the dredger pump on the other side of the engine-room bulkhead, the pump room.

1 *engine*
2 *manually controlled gear box for on/off two speed*
3 *engine-room bulkhead*

>CH 5

Standard figures of various types of diesel engines

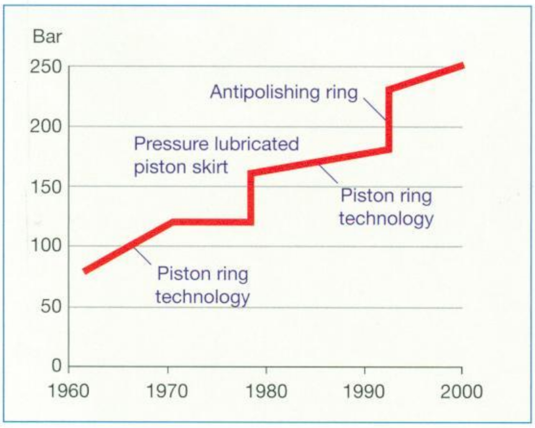




5.1 Mean effective pressure

P mean effective pressure expressed in bar or in Megapascal.

This is the pressure which is mathematically effective throughout the entire process and is available to the crank shaft. It is, of course a fictional pressure as in reality when taking the four-stroke cycle, the pressure fluctuates considerably from the inlet or intake pressure to the compression pressure, then to maximum combustion pressure and subsequently to the exhaust pressure.
The mean effective pressure is important when considering the diesel engine's performance and has seen a slow and gradual development over the past hundred years. Often new inventions such as the turbo charger, lubrication of piston rings under pressure and the anti-polishing ring have resulted in an increase of the mean effective pressure.

An improved turbo charger, that is an increased air supply with a higher filling pressure, allows for more kilograms of air to flow into the cylinders during the very brief moment that the inlet valves are opened. More air means more fuel injection, thus increasing the combustion pressures. An augmentation of the compression ratio has a similar effect. The compression ratio is the ratio of the volume when the piston is in its bottom position to the volume when it is in its top position.



▲

The development of maximum combustion pressure during the last forty years.

Shown here medium-speed four-stroke diesel engines running on H.F.O., by Wärtsilä. At this time the piston rings packets were improved. The pressure lubricated piston skirt was introduced and the greatest improvement, the anti-polishing ring.

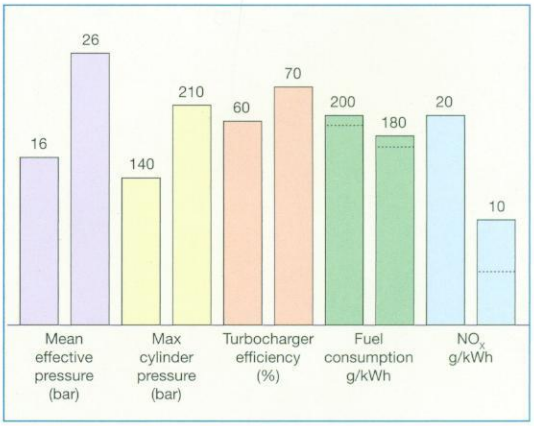
		Mk	mep bar	Cm m/s
1981	L35MC introduced			
1982	Full L-MC programme	1	15.0	7.2
1984	L-MC upgraded	2	16.2	
1985	L42MC introduced	2	16.2	7.2
1986	K-MC introduced		16.2	
	S-MC introduced		17.0	
	L-MC upgraded	3	16.2	7.6
1987	S26MC introduced		16.8	8.2
1988	K-MC-C introduced		16.2	8.0
1991	MC programme upgraded			8.0
	K and L-MC	5	18.0	
	S-MC	6	18.0	
1992	S26MC and L35MC upgraded		18,5	8.2
1993	S35MC and S90MC introduced			
	K90MC/MC-C upgraded	6	18.0	8.0
1994	S42MC introduced	6	18.5	8.0
1994	K98MC-C introduced	6	18.2	8.3
1995	K80MC-C upgraded		18.0	8.0
1996	L70MC upgraded	6	18.0	8.2
1996	S70MC-C, S60MC-C, S50MC-C		19.0	8.5
	and S46MC-C introduced		19.0	8.3
1996	S80MC upgraded		19.0	8.0
1997	L80MC upgraded		18.0	8.0
	K98MC introduced		18.2	8.3
1998	S80MC-C, S90MC-C and			
	L90MC-C introduced		19.0	8.1
	S35MC upgraded	7	19.1	8.1
1999	S42MC upgraded	7	19.5	8.0
2001	L70MC-C introduced		19.0	8.5
2001	L60MC-C introduced		19.0	8.3

mep = mean effectieve pressure Cm = mean piston speed

▲

The development of the mean effective pressure over twenty years, shown here a two-stroke crosshead engine by MAN–B&W.

An increase of the mean effective pressure by 3.5 bar and the mean piston speed by a metre per second.



▲

When the compression rate is increased both the effective mean pressure and the maximum combustion pressure increase. The specific fuel consumption decreases, and the efficiency increases.

Note: In future diesel engines with a compression rate of 18 and a maximum combustion pressure of 250 bars will have an efficiency in crease of 2,5%. Approximately 1% higher than those of current engines.

A higher effective mean pressure means that engines of a certain stroke volume can produce more power. The so-called power density increases. Due to severe competition, various engine manufacturers try to achieve increasingly more power per engine weight and - volume. For a certain engine type this can be achieved by increasing the mean effective pressure and the number of revolutions.

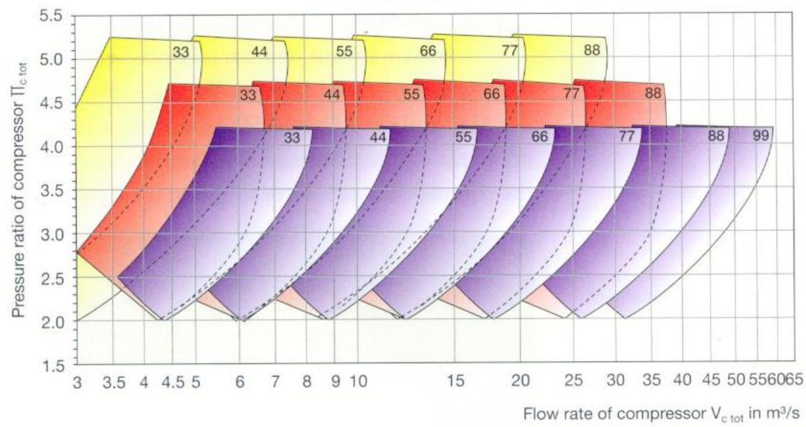
The mean effective pressure for two-stroke crosshead diesel engines is slowly but surely moving towards 20 bars. Most types achieve up to 18.5-19 bar (2008).

The mean effective pressures for four-stroke medium-speed diesel engines have reached circa 26 bars. Only a few very heavily loaded diesel engines achieve up to 28 bar (2008).

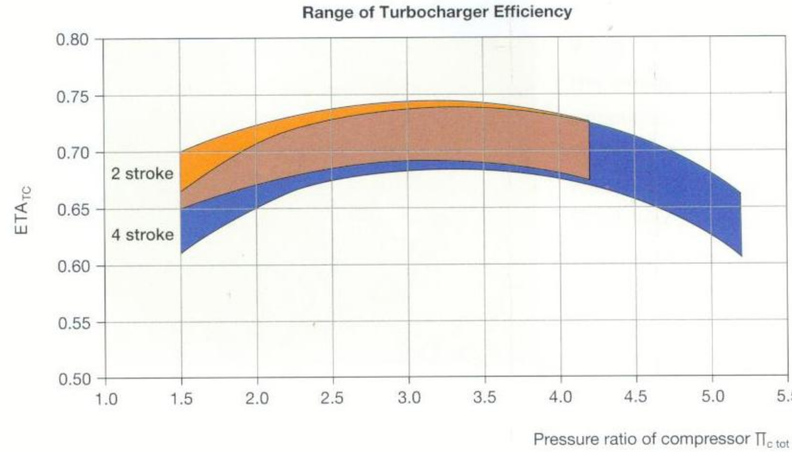
For four-stroke high-speed diesel engines these pressures lie somewhat lower but still easily exceed the 20 bar limit. Very heavily loaded engines can register pressures of over 30 bar.

The future: As far as the future development regarding flow (in m³/sec) and suction lift (in bars or metres water column) of turbo blowers is concerned, much depends on the development of turbo blower efficiency.

Turbocharger Application Range



Range of Turbocharger Efficiency



Supercharged engine output [kW]

Type/size	2-stroke engines (le = 8.5*)	4-stroke engines (le = 6.5*)
TCA 33	2,800 – 4,300	
TCA 44	4,100 – 6,200	
TCA 55	4,400 – 8,200	5,800 – 10,800
TCA 66	6,200 – 11,600	8,200 – 15,200
TCA 77	8,800 – 16,400	11,500 – 21,500
TCA 88	12,400 – 23,300	16,300 – 30,300
TCA 99	18,700 – 31,300	

* specific air consumption in kg/kWh

▲
Increasing the compression rate and the total efficiency of the turbo blowers is decisive in improving the total efficiency of the diesel engine.

An increase in the number of kilograms of air supplied per a certain time means that more fuel can be combusted, and therefore more power can be generated.

5.2 Mean piston speed

Cp mean; expressed in metres per second.
When accelerating the mean piston speed there are two important factors:

1. **The amount of the accelerating- and decelerating forces that affect the up- and downward moving piston.**
2. **The degree of lubrication of the piston rings and cylinder liner.**

The amount of the accelerating- and decelerating forces that affect the up- and downward moving piston.
When increasing the revolutions of an engine with the same piston stroke, the accelerating- and decelerating forces grow steadily, because the piston mass must accelerate and decelerate in a shorter period. In T.D.C. and B.D.C. the piston comes to a brief stop.

The forces on the piston increase according to the formula $F = m \times a$ in which:

F = the forces exerted on the piston in Newton
m = piston mass in kg
a = acceleration/deceleration in m/sec.²

In practice one tries to make the piston mass as light as possible.
Therefore low weight light metal pistons are used for high-speed diesel engines. These diesel engines use Marine Diesel Oil (M.D.O.) and exert lower pressures on the piston, which can then withstand the compression pressures found in the process.

▶ Low mass, especially of the piston is important in reducing the accelerating- and decelerating forces.

Light metal pistons of a high-speed four-stroke diesel engine.

Medium-speed diesel engines often use Heavy Fuel Oil (H.F.O.). The pressures and the temperatures are higher and light metal pistons are therefore not suitable.
For this application one uses a piston with a cast steel piston crown which can withstand the higher load forces. The piston skirt is manufactured using a light metal or cast iron.
Since the latter is much heavier, it is often milled where possible.

▶ The piston mass in medium-speed engines is reduced by keeping the skirt a light as possible.

The material surrounding the piston pin is removed where possible (1).

Degree of lubrication of piston rings and cylinder liner

Proper lubrication is of the utmost importance in maintaining a good spring pack seal, situated between the piston and the cylinder liner, for a lengthy time (for instance 15,000 operation hours). The functions of lubricating oil are among others, the reduction of friction, assisting the discharge of combustion by-products and the discharge of heat which cause problems when a certain piston speed is exceeded.
Today an average piston speed of 8 to 10 m/sec. is standard for four-stroke engines. Anything beyond this poses a potential problem. There are diesel engines available that have a mean piston speed of 12 m/sec.(2008).
Two-stroke crosshead engines usually have a slightly lower speed; this is related to the time required to scavenge with fresh air and the interruptions in the cylinder liner due to the surrounding inlet ports: all of which interfere with lubrication.
Here mean piston speed averages between the 6 and 8.5 m/sec.

5.3 Load parameters

The product of the mean effective pressure and the mean piston speed is a measurement of diesel engine load. Indicated by Lp the load parameter of the diesel engine.

L_p = p mean eff. × $C_{p\ mean}$ (bar/m/sec.)

The product of this equation gives a good indication of the degree of engine load.

Examples

Diesel engine I p mean eff. = 15 bar
$C_{p\ mean}$ = 8 m/sec.
L_p = 15 × 8 = 120 bar/m/sec.
This is low. Probably from an early diesel engine.

Diesel engine II p mean eff. = 25 bar
$C_{p\ mean}$ = 10 m/sec.
L_p = 25 x 10 = 250 bar/m/sec.
This is high. An ultra-modern diesel engine.

Diesel engine III p mean eff. = 27 bar
$C_{p\ mean}$ = 11 m/sec.
L_p = 27 x 11 = 297 bar/m/sec.
This is extremely high and is only achievable in a few finely tuned engines. The thermal- and mechanical load in this diesel engine is also very high.

Rule of thumb

L_p	=	100 – 150	**low load, earlier models**
L_p	=	150 – 250	**normal load, contemporary engines**
L_p	=	250 – 300	**heavy load, most modern engines**

5.4 Compression ratio

Also compression rate. This is the ratio of the cylinder volume above the piston at the beginning of the compression stroke (so the piston is at B.D.C.) to the cylinder volume above the piston at the end of the compression stroke (so the piston is at T.D.C.).
The compression ratio is indicated by
ε = epsilon.

So, the theoretical compression ratio is:

$$\varepsilon = \frac{V_S + V_C}{V_C}$$

In which:
V_S = Volume moved by the piston, the stroke volume.
V_C = Volume above the piston in T.D.C. or volume compression space. This is the combustion space above the piston in T.D.C.

In reality, compression only starts when, for a four-stroke engine the inlet valves and for a two-stroke engine the exhaust ports or valves, are closed.
This is called the effective compression ratio.
The distance which the piston has travelled from B.D.C. prior to the beginning of the effective compression is denoted by k.
The volume above the piston is then:
$V\ eff = (1 - k) \times V_S + V_C$.

The effective compression ratio is indicated by ε effective.

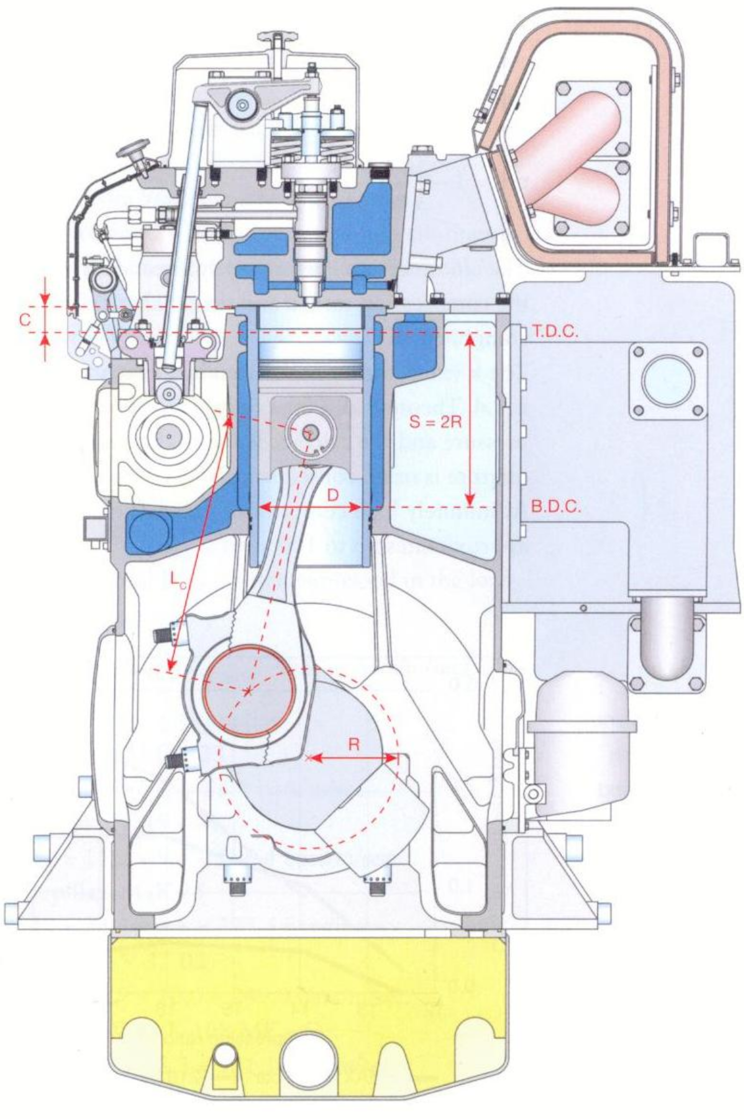
The formula:

$$\varepsilon\ \text{effective} = \frac{(1 - k)\ V_S + V_C}{V_C}$$

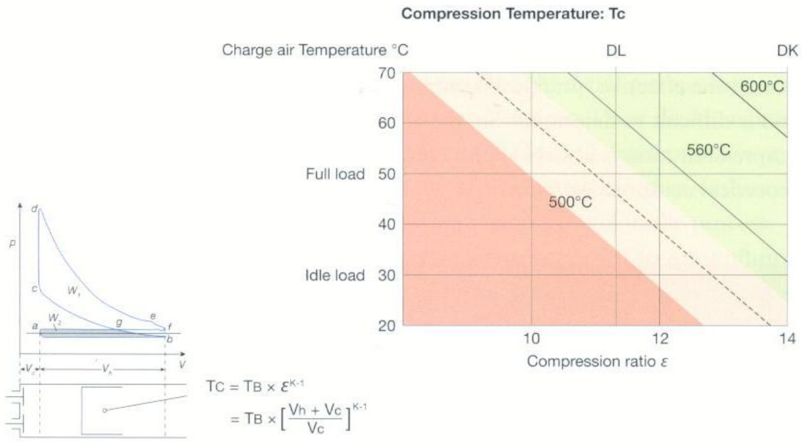
The effective compression ratio varies from approximately 6 to 20. In Otto-engines where the compressed mixture is not self igniting the compression ratio varies from 6 to 15, dependent on the air volume of the mixture; the higher the volume the higher the compression ratio.
For diesel engines used in industrial applications, the compression ratio ranges from 10 to 20. Most diesel engines have ratios of between 12 and 16. This is always the theoretical compression ratio, because the effective (practical) compression ratio is difficult to determine. So, the effective compression ratio is always slightly lower than the theoretical compression ratio.

Significance of the compression ratio

The compression ratio is of great import for the final compression pressure and the final compression temperature in the engine and is therefore key in the working of the process and consequently that of the engine.
So, dependent on the inlet air pressure and the inlet air temperature the compression ratio determines the pressure and the temperature at the end of the compression stroke.
In order to control this somewhat, the compression ratio is often slightly changed.
In earlier times a compression plate was fitted between the connecting rod foot and the top crank pin bearing cap.



▼ The theoretical compression rate is the ratio of the cylinder volume over the piston in B.D.C. to the cylinder volume over the piston in T.D.C.



▲

The final compression temperature in relation to the compression rate, the engine load and the air temperature after the inter cooler.

Shown here a Daihatsu medium-speed four-stroke engine Type DL en DK.

In small engines the compression ratio can be increased by slightly milling out the top of the engine block. Additionally, the piston height above the piston pin may be adjusted by the placement of a 'higher' (higher compression ratio) or a 'lower' (lower compression ratio) piston. Although this does not alter the stroke volume, it does change the final compression space.

Influence of the compression ratio

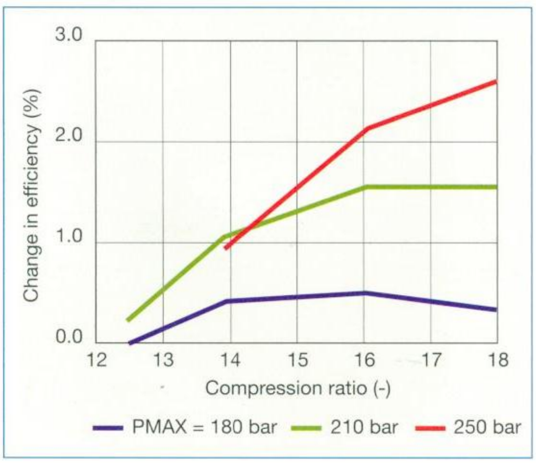
Theoretically, the thermal efficiency of the combustion cycle of an internal combustion engine is:

$$\eta th = 1 - \left(\frac{1}{\varepsilon}\right)^{k-1}$$

The efficiency is therefore solely dependent on the compression ratio ε and the adiabatic component k.
The k value, for air under standard conditions, is 1.4. Theoretically, inlet temperature, initial pressure and the calorific value of the fuel-air mixture is of no consequence.
At infinitely high compression ratios the efficiency approximates up to 100%.

▶

The increase in engine efficiency when both compression rate and combustion pressures are increased.



The example above shows that the theoretical process nowhere reflects the actual process.

In approximation the total efficiency of diesel engines is as follows:
High-speed runners (small diesel engines):
0 – 100 kW 25 – 30%
High-speed runners (larger diesel engines):
100 – 5,000 kW 30 – 40%
Medium-speed runners (large diesel engines):
500 – 30,000 kW 40 – 45%
Slow-speed runners (very large diesel engines):
1500 – 100,000 kW 48 – 53%

Only the low-speed runners are two-stroke engines, all others are four-stroke engines.

Compression ratios very seldom exceed ± 15 to 20; as the top pressures and the top temperatures would get to high, including the mechanical and thermal load on the components. The friction loss would also rise significantly.
Large medium-speed diesel engines have a compression ratio of 13 to 15. In smaller diesel engines sometimes 16 to 24. Large low-speed engines average around 11.5 to 12.5.
At high compression ratios the final compression pressures and –temperatures are significantly higher.

5.5 Power density

The term power density has come into being as a result of the continual development of industrial diesel engines and growing competition in the diesel engine industry.
The engines have become increasingly smaller in size with a corresponding reduction in weight. Furthermore, they have had to provide a power increase for a specific stroke volume.
This is why there are generally three types of power density:

- the shaft power produced in kW in relation to the weight of the engine;
- the shaft power produced in kW in relation to the total stroke volume of the engine ;
- the shaft power produced in kW in relation to the total volume of the engine.

5.5.1 Shaft power produced in kW in relation to the weight of the engine

Power density per weight =

$$\frac{\text{Total shaft power}}{\text{Total engine weight}}$$

or: produced kW shaft power per kilogram engine weight.

P density = kW/kg engine weight.

Examples

Perkins – 1103 C – 33
Propulsion, diesel fuel oil, three cylinder in-line engine, four-stroke, natural aspiration, cylinder bore 105 mm, stroke 127 mm, compression ratio 18.2 : 1, stroke volume 3.3 litre, maximum continuous power at 2200 revolutions per minute 41.5 kW.
Weight 249 kg.
Category I
per litre stroke volume 12.6 kW
per kW power 6 kg weight
mean effective pressure 6.9 bar
mean piston speed 7.7 m/sec.

Cummins KT 19 – M
Propulsion, diesel fuel oil, six-cylinder in-line engine, four-stroke, turbo charger, cylinder bore 159 mm, stroke 159 mm, compression ratio 15.5 : 1, stroke volume 19 litres maximum continuous power at 1800 revolutions per minute 317 kW.
Weight 1937 kg.
Category II
per litre stroke volume16 kW
per kW power 6.1 kg weight
mean effective pressure 11.2 bar
mean piston speed 9.54 m/sec.

Caterpillar–MaK 25
Propulsion, heavy fuel oil, six-cylinder in-line engine, four-stroke, turbo charger, cylinder bore 255 mm, stroke 400 mm, stroke volume 122.5 litres, maximum continuous power at 720 revolutions per minute 1800 kW.
Weight 22,000 kg.
Category III
per litre stroke volume 14.7 kW
per kW power 12.2 kg weight
mean effective pressure 23.7 bar
mean piston speed 9.6 m/sec.

Wärtsilä 18 V 32 B2 output
Propulsion, heavy fuel oil, eighteen-cylinder V-engine, four-stroke, turbo charger, cylinder bore 320 mm, stroke 400 mm, stroke volume 578.8 litres, maximum continuous power at 750 revolutions per minute 9000 kW.
Weight 82,500 kg.
Category III
per litre stroke volume 15.55 kW
per kW power 9.2 kg weight
mean effective pressure 24.9 bar
mean piston speed 10.0 m/sec.

MAN–B&W 14 K 108 ME – C
Propulsion, heavy fuel oil, fourteen-cylinder in-line engine, two-stroke, crosshead, turbo charger, cylinder bore 960 mm, stroke 2500 mm, stroke volume 25,320 litres, maximum continuous power at 102 revolutions per minute 80,080 kW.
Weight 2,300,000 kg.
Category IV
per litre stroke volume 3.16 kW
per kW power 28.7 kg weight
mean effective pressure 18.2 bar
mean piston speed 8.3 m/sec.

Wärtsilä RTA 96 C
Propulsion, heavy fuel oil, fourteen-cylinder in-line engine, two-stroke, crosshead, turbo charger, cylinder bore 960 mm, stroke 2500 mm, stroke volume 25,320 litres, maximum continuous power at 102 revolutions per minute 80,080 kW.
Weight 2,300,000 kg.
Category IV
per litre stroke volume 3.16 kW
per kW power 28.7 kg weight
mean effective pressure 18.6 bar
mean piston speed 8.5 m/sec.

Remarks

Today the total load is often expressed in the load parameter, L_P.

L_P = mean eff. p × $C_{p\ mean}$. Unit in bars/m/sec.

To all five examples applies:

Perkins – 1103 C – 33
$L_P = 6.9 \times 7.7 = 53.1$ bars/m/sec.

Cummins KT 19 – M
$L_P = 11.2 \times 9{,}5 = 106.4$ bars/m/sec.

Caterpillar–MaK 25
$L_P = 23.7 \times 9.6 = 227.5$ bars/m/sec.

Wärtsilä 18 V 32 B2
$L_P = 24.9 \times 10.0 = 249.0$ bars/m/sec.

MAN–B&W 14 K 108 ME – C
$L_P = 18.2 \times 8.3 = 151.1$ bars/m/sec.

Wärtsilä RTA 96 C
$L_P = 18.6 \times 8.5 = 158.1$ bars/m/sec.

Clearly, the small Perkins engine has the lowest load.
The medium speed runners (Caterpillar–MaK and Wärtsilä) have the highest load.
The two-stroke engines by MAN–B&W and Sulzer have a far lower load.
It is much more practical to compare engines that have an identical cylinder bore, similar operating processes and revolutions per minute.
In these cases one often finds small, but significant differences.
Well-known examples are medium-speed H.F.O. engines with a cylinder bore of 320 mm.

MAN–B&W L 32/40
$L_p = 24.9 \times 10 = 249$ bar/m/sec.
Wärtsilä 32
$L_p = 23.3 \times 9.6 = 223.7$ bar/m/sec.
MaK 32 C
$L_p = 24.9 \times 9.6 = 239.0$ bar/m/sec.

It is obvious that the differences in engines in the same category are far smaller.
Caterpillar–Mak for instance, has a stroke – bore ratio of $\frac{480}{320}$, while Wärtsilä and MAN–B&W have a stroke – bore ratio of $\frac{400}{320}$.

When comparing a specific category of engines the following aspects are important:
- **specific fuel consumption;**
- **specific lubricating oil consumption;**
- **the purchase price;**
- **the delivery time;**
- **the maximum continuous power output;**
- **the torque;**
- **the service;**
- **the warranties.**

Past experiences with the engines in question

For ships, and in particular ships of a certain design, the propulsion engine of choice is often pre-determined. Changing the engine often means additional costs.

5.5.2 Shaft power produced in kW in relation to the total stroke volume of the engine

Power density per stroke volume =

$$\frac{\text{Total shaft power}}{\text{Total stroke volume}}$$

Or: kW shaft power produced per litre stroke volume.

P density = kW/dm³ stroke volume

5.5.3 Shaft power produced in kW in relation to the total volume of the engine

Power density per total engine volume =

$$\frac{\text{Total shaft power}}{\text{Total engine volume}}$$

P density = kW/dm³ engine volume

This is, of course, far more difficult to determine and, so apparently used less frequently. Even so, in reality the size of a particular engine type is certainly taken into account. For example, the build-in height, the free height required to enable piston dismount, the width of a V-engine etcetera. The decision making process in choosing a certain engine automatically includes choosing the space in which to fit in the engine. For fast, compact ships the engine's measurements most certainly play an important role. The size of engine space is often very limited.
For larger engine sizes, this is also of importance. Often large car ferries have a limited height in their engine rooms. Sometimes a trap door in the ferry's deck needs to be opened in order to dismount a piston from the engine. (piston drawing).

In feeder-container ships the engine room is kept as small as possible in order to load as many containers as possible in the holds. This means that the propulsion engine is contained in the little

▲
Car ferries require that the entire main is utilized for rolling material and therefore the engine room height is limited. In this case, four-stroke engines are the only option.

space between the propeller shaft and the engine room bulk head. In this case the gear box which is used to reduce the engine speed to the most effective number of revolutions for the propeller also needs to be as short as possible.

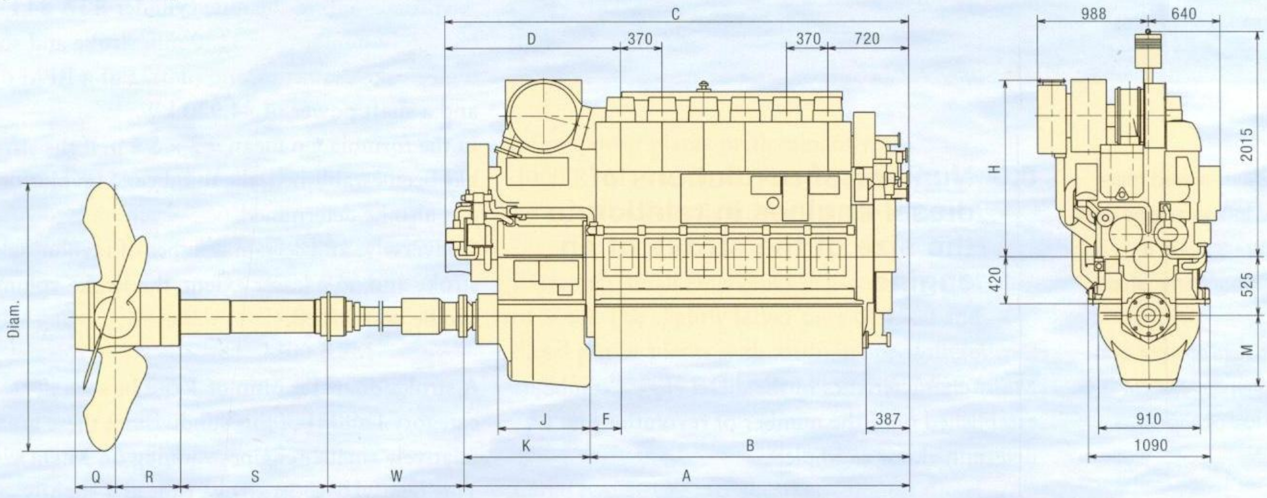
▼

Abundant data is important in running a propulsion installation.

The shaft power, as well as the number of revolutions ,the fuel consumption, the weight and sizes are required to make an informed decision.

MAN B&W Four-stroke Propulsion Systems

L23/30A-KV
800-1280kW
(1090-1740 BHP)



Standard programme

ENGINE	REDUCTION GEAR		PROPELLER			DIMENSIONS IN MM											
Type.	Series	Type	Type	Speed	Diam	A	B	C	D	F	H	J	K	M	Q	R	W
Output mcr				rpm				mm									min
6L23/30A-E	AMG 8	31KV8	VB 560	268	2200	3975	2898	4136	1566	328	1587	770	1077	636	293	553	900
800 kW	AMG 8	39KV8	VB 640	214	2450	3975	2898	4136	1566	328	1587	770	1077	636	360	595	900
1090 BHP	AMG 8	44KV9	VB 640	190	2600	3975	2898	4136	1566	328	1587	770	1077	636	360	595	900
	AMG 16	52KV13	VB 740	159	2850	4550	2898	4660	2090	318	1587	1282	1662	505	445	655	1200
6L23/30A	AMG 8	31KV8	VB 560	292	2250	3975	2898	4136	1566	328	1587	770	1077	636	293	553	900
960 kW	AMG 8	39KV8	VB 640	233	2450	3975	2898	4136	1566	328	1587	770	1077	636	360	595	900
1305 BHP	AMG 8	44KV9	VB 640	207	2600	3975	2898	4136	1566	328	1587	770	1077	636	360	595	900
	AMG 16	52KV13	VB 740	173	2850	4550	2898	4660	2090	318	1587	1282	1662	505	445	655	1200
8L23/30A	AMG 11	31KV11	VB 640	292	2350	4953	3628	5064	1754	318	1587	950	1325	639	415	595	900
1280 kW	AMG 11	39KV11	VB 640	233	2600	4953	3628	5064	1754	318	1587	950	1325	639	415	655	1200
1740 BHP	AMG 11	44KV13	VB 740	207	2800	4953	3628	5064	1754	318	1587	950	1325	639	445	655	1200
	AMG 16	52KV13	VB 860	173	3100	5290	3628	5400	2090	318	1587	1282	1662	505	445	745	1350

The propeller is calculated according to DnV, No Ice with high skew

Main data

ENGINE	BORE	STROKE	SPEED	MEP	PISTON SPEED	OUTPUT/CYLINDER	
Type	mm	mm	rpm	bar	m/s	kW	BHP
L23/30A-E	225	300	825	16,3	8,25	133	181
L23/30A	225	300	900	17,9	9,00	160	217,5

Specific consumption

ENGINE	FUEL OIL				LUBRICA TING OIL	
Type	mcr		85% mcr		mcr	
	g/kWh	g/BHPh	g/kWh	g/BHPh	g/kWh	g/BHPh
L23/30A-E	188	138	187	137	1,0	0,7
L23/30A	190	140	189	139	1,0	0,7

Weight

ENGINE	GEAR	PROP	Dry weight in tons (approx)	
Type	Type	Type	Engine/Gear	Prop*
6L23/30A-E	31KV8	VB 560	15,0	2,7
	39KV8	VB 640	15,0	3,4
	44KV9	VB 640	15,0	3,5
	52KV13	VB 740	17,6	4,4
6L23/30A	31KV8	VB 560	15,0	2,7
	39KV8	VB 640	15,0	3,4
	44KV9	VB 640	15,0	3,5
	52KV13	VB 740	17,6	4,4
8L23/30A	31KV11	VB 640	17,9	3,4
	39KV11	VB 640	17,9	3,8
	44KV13	VB 740	17,9	4,5
	52KV13	VB 860	20,1	6,1

** Weight incl 4.0 m shaft and 2.0 m stem tube*

▲
A medium-speed four-stroke in-line engine used for propulsion in a 'feeder-container ship'.

Shown here an eight-cylinder Caterpillar–MaK M43-C heavy fuel oil engine.

5.6 Number of revolutions of diesel engines in relation to the size of the stroke of an engine

Explanation

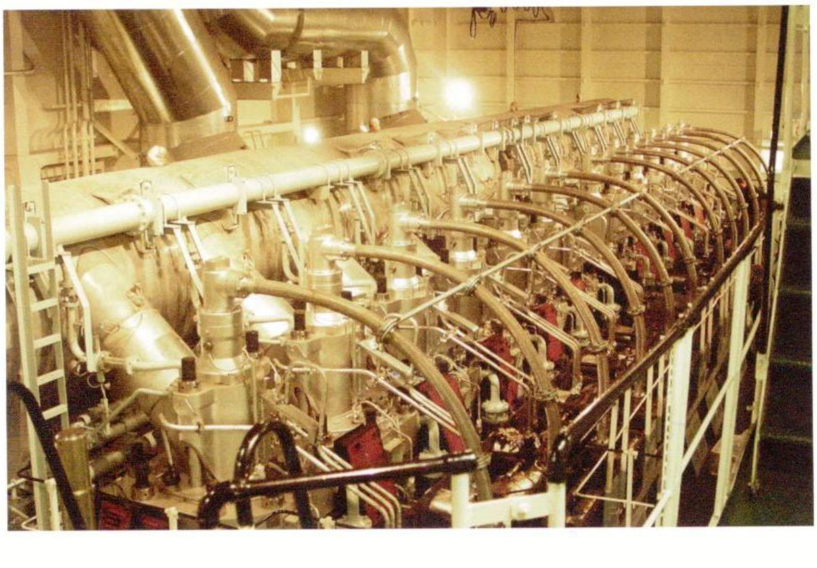
Many characteristics of the diesel engine can be established once the number of revolutions is determined, for example:

the maximum allowable mean piston speeds:
- ± 11 m/sec. for four-stroke trunk piston engines;
- ± 8.5 m/sec. for two-stroke crosshead engines.

The stroke/diameter ratio for four-stroke as well as two-stroke engines, in practice often lies within certain values.

Four-stroke	normally	1.1 – 1.3
	long stroke	1.4 – 1.6
two-stroke	normally	1.8 – 2.2
	long, ultra long stroke	3 – 4

▼
A two-stroke crosshead engine on a large container ship, a twelve-cylinder Wärtsilä Sulzer RT – FLEX, with the common rail system.



In smaller diesel engines one also sees S/D ratios of 1 or less. This is, for instance, the case in very small diesel engines with a high number of revolutions.
A **relatively long stroke** means that the engine will be tall with a long cylinder liner and a crankshaft with a large turning circle (especially for two-stroke crosshead engines).
The world's biggest engine, a fourteen-cylinder MAN–B&W K 108 ME – C has a 1080 mm bore with a 2660 mm stroke and has therefore a limited stroke/diameter ratio of 2.5 at a RPM of 94 and a shaft power of 97,300 kW.
A Wärtsilä Sulzer, the nine-cylinder RTA 84 F – B has a 840 mm bore at 3150 mm stroke and so a high stroke/diameter ratio of 3.75 at a RPM of 74 and a shaft power of 34,920 kW.
In the formula Cp mean = 2 × S × n, if the stroke has been established, the number of revolutions can also be determined.
Conversely: at a certain number of revolutions the stroke and, to a lesser extent, the stroke volume can be established.

A stroke/diameter ratio of 1 can be seen in category I and II of this subdivision; these are the relatively smaller engines running on M.D.O.. This relatively short stroke runs at a slightly higher RPM of often over 1500 revolutions, for instance, 1800 and 2100 RPM.

5.7 RPM of generators

Diesel engine gensets ought to provide a **constant voltage** in conjunction with a constant frequency under all load variations of the electrical net.
The European net frequency along with most parts of the world is 50 Hz. The United States and other countries related to the U.S. have a frequency of 60 Hz. In general the frequency on board of ships is 60 Hz.
The **frequency** generated by a rotary generator is directly dependent on the **RPM** and the **number of polar pairs** (p).
A polar pair consists of a north and a south pole.

Formula

Frequency = number of polar pairs × number of revolutions

Frequency in Hz (Hertz)
Number of polar pairs is an undefined number.
1 polar pair – bi-polar. A north and a south pole.
Number of revolutions in RPS.

◄
A four-stroke medium-speed Deutz 640 diesel engine for H.F.O. in a diesel power plant in the Gambia.

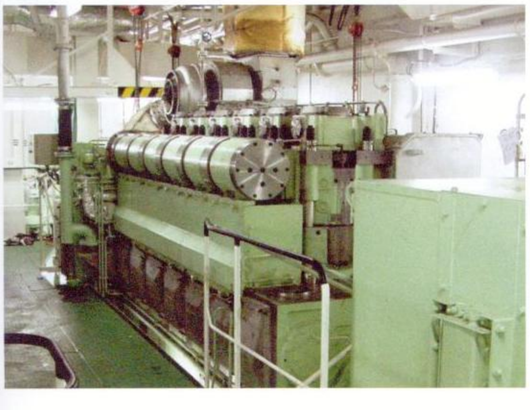
Example

Number of revolutions of diesel engines for 50 Hz gensets.

1 polar pair	bi-polar	50 RPS = 3000 RPM
2 polar pairs	4-polar	25 RPS = 1500 RPM
3 polar pairs	6-polar	16.67 RPS = 1000 RPM
4 polar pairs	8-polar	12.5 RPS = 750 RPM
5 polar Pairs	10-polar	10 RPS = 600 RPM
6 polar pairs	12-polar	8.33 RPS = 500 RPM

At 60 Hz this is then multiplied by 1.2, so 3600, 1800, 1200, 900, 720 and 600 revolutions. The diesel engines for gensets are frequently delivered for two different engine revolutions, for example 1500/1800 revolutions, especially by American companies that supply markets outside the Americas.

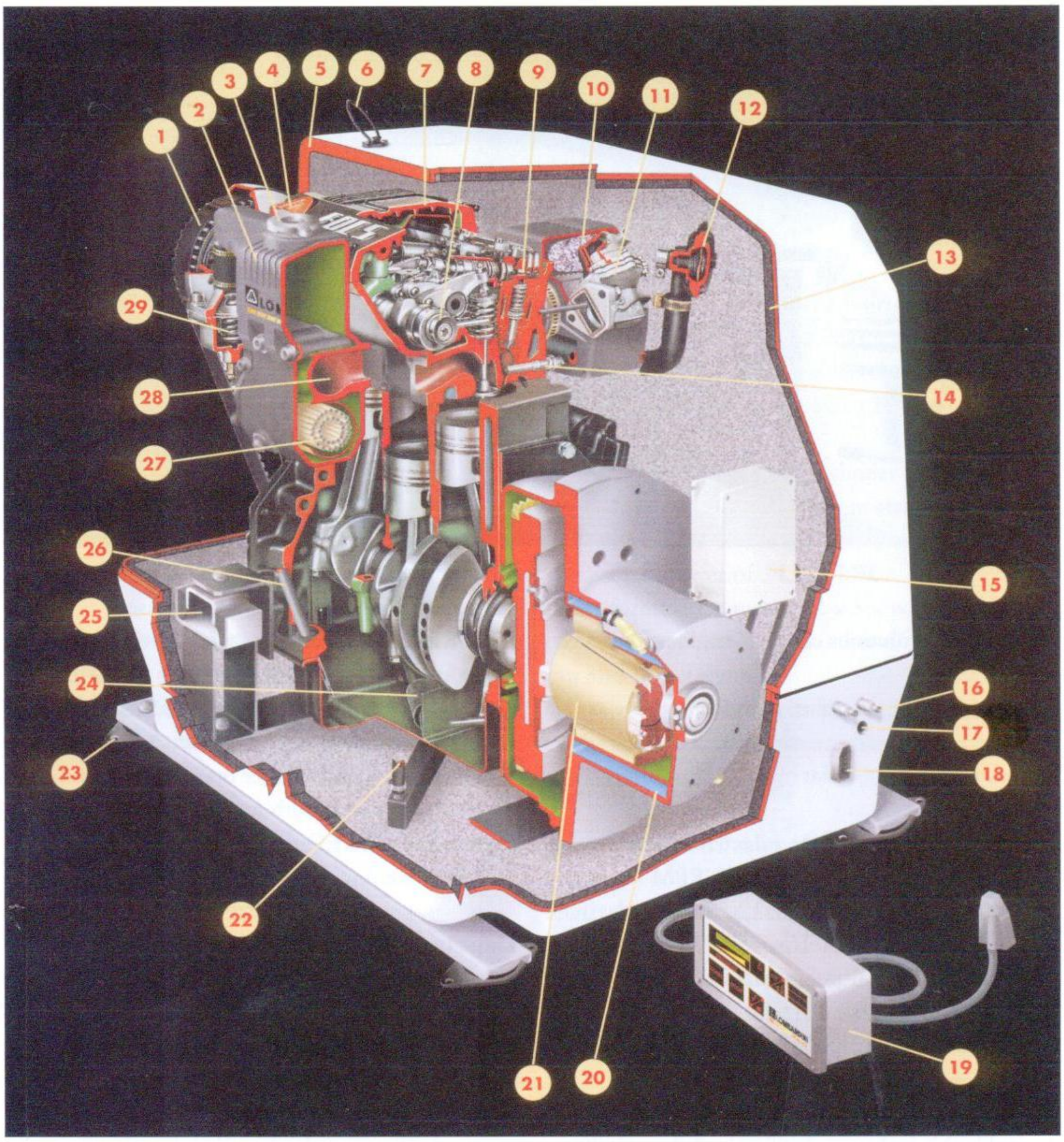
▼
A diesel genset on a large container ship.



Large power plants predominantly use 3000 RPM generators. These are directly driven by a gas- and steam turbine-combination.

Extremely small emergency gensets have a RPM of 3000. The slightly larger ones of 1500 and diesel power plants with medium-speed diesel engines using (H.F.O.) often have a RPM of 600, 750 or 1000.

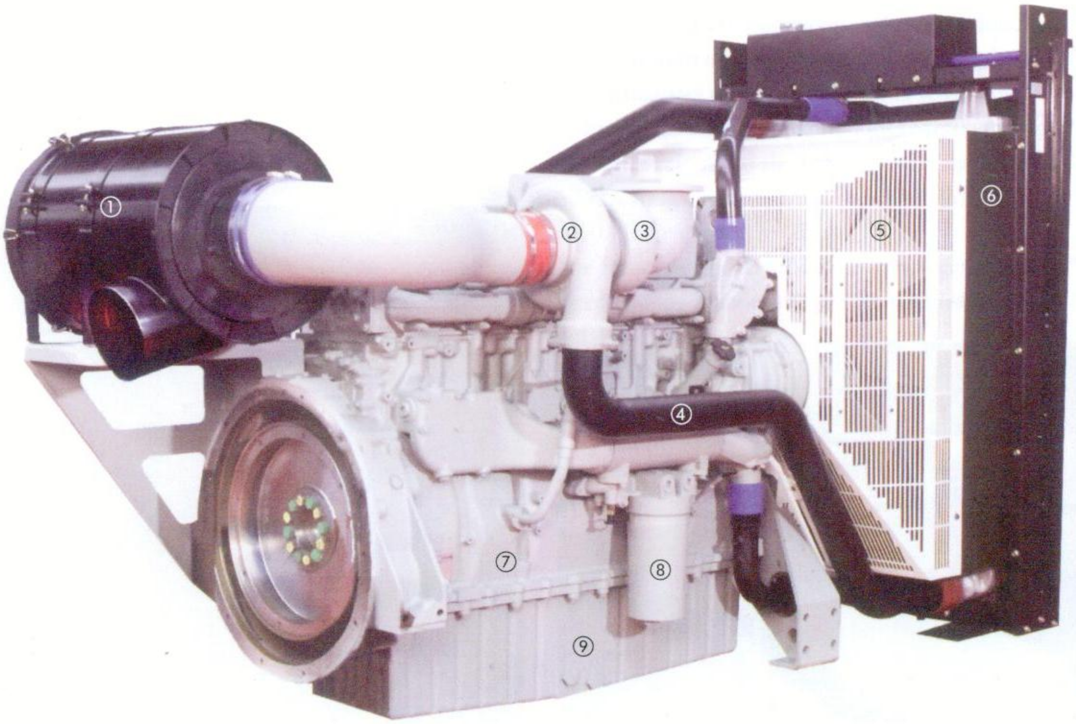
Large two-stroke crosshead engines for diesel power plants have an even smaller RPM. With eighteen polar pairs and therefore a 36-polar generator, the number of revolutions is 166.67. In order to install a 36-polar generator, a generator diameter of between 6 to 12 metres is required. These are the so-called disc generators.



▶

A small sound proof genset for yachts.

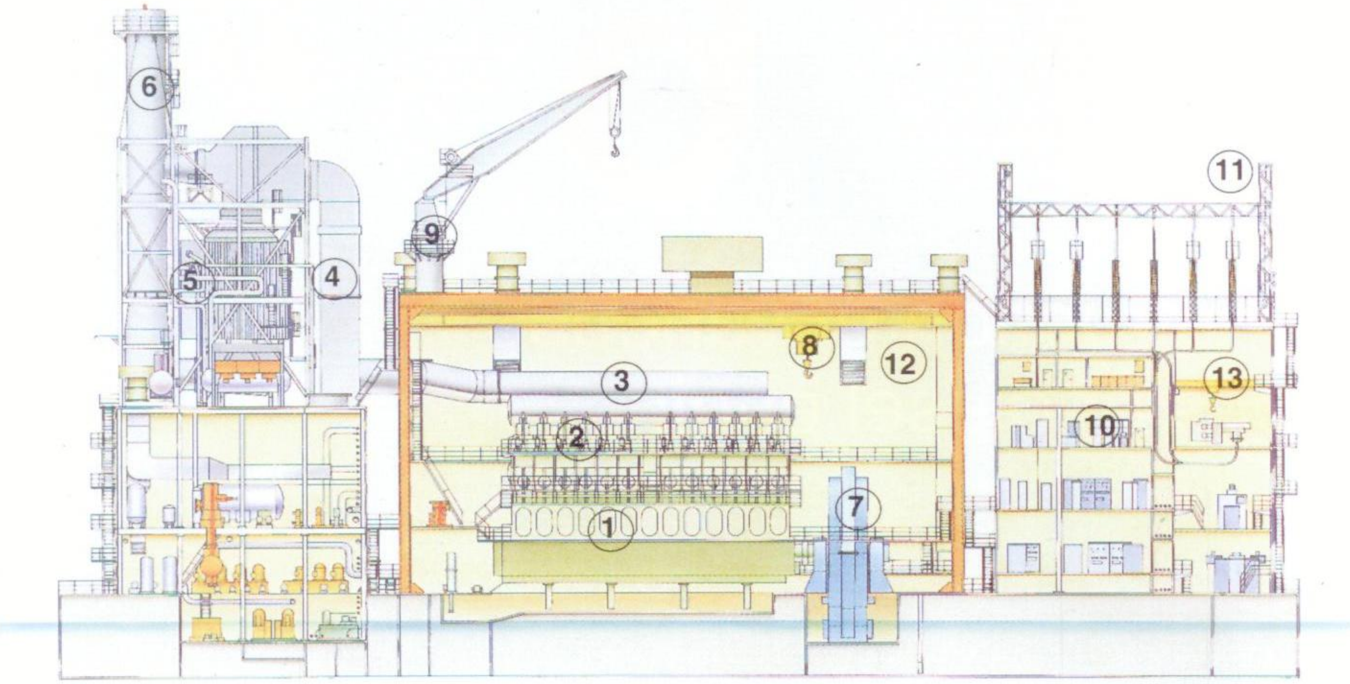
1 timing belt
2 heat exchanger
3 coolant filler cap
4 oil filler cap
5 sound damping housing
6 rubber valves
7 valve cap
8 cam shaft
9 fuel injector
10 air filter
11 fuel supply pump
12 bleeder valve
13 sound insulation
14 glow plug
15 electric junction box
16 coupling for fuel line
17 outer water inlet
18 multi connector for remote control
19 remote control panel
20 generator jacket
21 generator
22 oil cooler
23 external vibration dampers
24 sump partition wall
25 internal vibration dampers
26 dip stick
27 cooling element of the heat exchanger
28 water cooled exhaust manifold
29 thermostat



▶

A high-speed diesel power unit.

1 air filter
2 turbo blower air section
3 turbo blower exhaust gas section
4 compressed air to air cooled intercooler
5 radiator ventilator
6 radiator with cooling section for combustion air, fresh water coolant and lubricating oil
7 engine block
8 lubricating oil filter
9 crank pan



▲

A floating diesel power plant.

1 *two-stroke crosshead engine*
2 *cylinder head plateau*
3 *exhaust gas manifold*
4 *sound damper*
5 *exhaust gas boiler*
6 *chimney*
7 *disc generator*
8 *overhead crane*
9 *crane*
10 *electric control box*
11 *high voltage connection to the shore*
12 *ventilation shafts*
13 *spare parts and maintenance area*

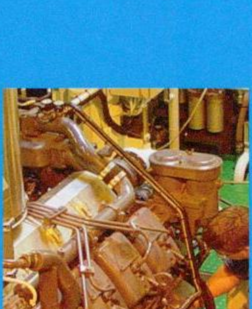
◄

The generator poles are clearly visible. The north pole is followed by a south pole, which in turn is followed by a north pole and so on.

>CH 6

Construction of various types of diesel engines





Mounting a fast running MTU diesel engine.

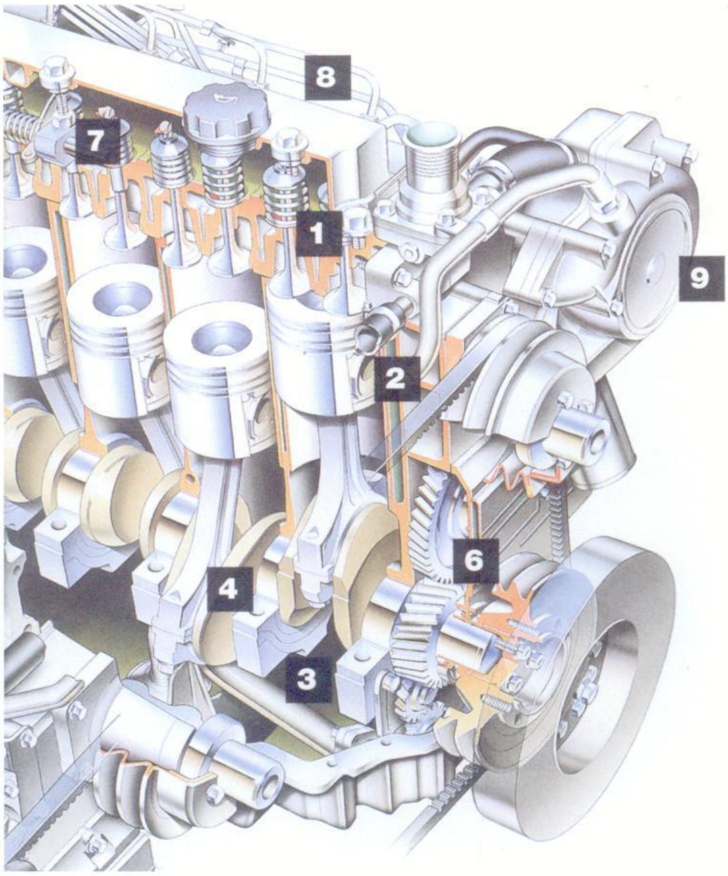
Modern diesel engine construction is a time-consuming, slow process requiring much research. Due to strict consumer requirements with regards to diesel engines, such as a high power density, low oil- and fuel consumption, low toxic emissions, long standing time of the components and simple mounting and dismounting methods. These factors make it difficult for engine manufacturers to continue to compete in this market segment.

6.1 Category I: Industrial diesel engines from 0 to 100 kW shaft power, fuel M.D.O., four-stroke, high-speed engines

▶

High-speed four-stroke diesel engine.

1 *cylinder head with two inlet valves and two exhaust valves*
2 *piston*
3 *crankcase or carter*
4 *crank web*
6 *timing gear between the crankshaft and camshaft*
7 *valve motion*
8 *high-pressure fuel lines*
9 *coolant pump*



The relatively small industrial diesel engines are used in a large number of apllications, among others:

- **propulsion;**
- **gensets;**
- **pump geardrives;**
- **traction**
- and **numerous** other **areas** which will not be discussedany further, such as transportation, earth moving, cranes etcetera.

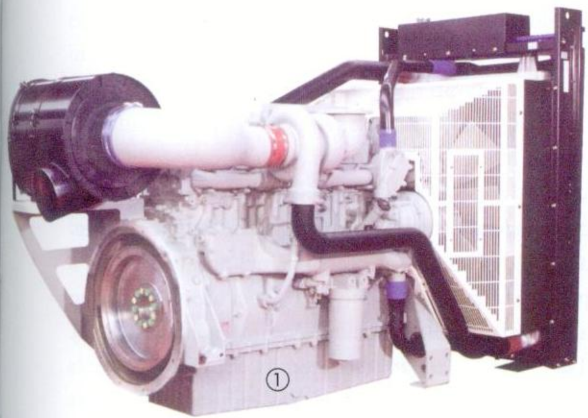
Inside the scope of this book, they can be found mainly:

- in propulsion of smaller ships and yachts;
- for generating electrical energy, such as in gensets or as back-up generators;
- for driving pumps and other equipment.

In general they are based on the four-stroke principle, trunk piston type and today increasingly more often fitted with two inlet- and exhaust valves.
Here one also finds that the modern systems such as fuel injection with, among other things, a common rail system are increasingly applied.
The engine consists of a light metal or cast iron block which in the smallest types do not have separate cylinder liners (known as 'bushes' in these small engines).
This means that with excessive wear and tear of the liner the cylinder bore needs to be rebored and then fitted with an oversized(larger piston and piston rings) piston.

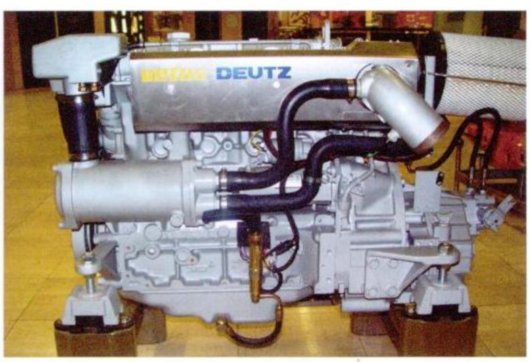
See Chapter 26, Overhauling diesel engines and their parts.

The crank shaft is fixed to the block by means of bearing caps and vertical tap bolts. In this case a four-cylinder diesel engine has five bearings (popularly said: the engine is five times bearinged).
At the bottom of the frame is a freely suspended oil sump which contains the oil pump,oil filter, and the lubricating oil among others.
The engine is not directly placed on the sump but on the engine mountings that are attached to the block.



▲
Diesel engine with suspended oil sump (1).

Traditional designs have a closed frame under the block onto which the engine is mounted.

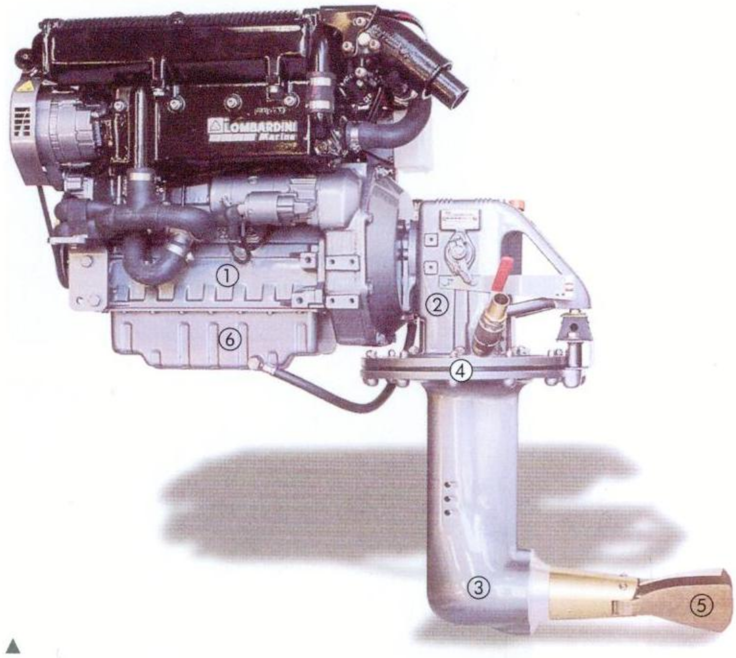


▲
A diesel engine adapted to maritime use.

Often the original uncooled hot exhaust pipes are water cooled to reduce the fire risk. The engine is mounted on vibration dampers.

Often vibration dampers are placed underneath the frame or the engine mountings in order to stop vibrations from the engine being transmitted to the engine surrounds.

▲
Vibration dampers.



▲
A small high-speed four-stroke diesel engine for yacht propulsion.

This so-called 'z-drive' is fairly common: there is no need for a long propeller shaft, so the propulsion space is kept to a minimum. The propeller is a folding propeller; at sea the two propeller blades fold back for minimum drag. At the large horizontal flange the gear drive passes through the ship's hull.

1 *diesel engine*
2 *bevel gear in the yacht*
3 *bevel gear in the water*
4 *flange in the hull*
5 *folding propeller*
6 *oil sump*

The cylinder head mounted on top of the engine block often already comprises of all the cylinders, so the engine does not have separate cylinder heads as is the case in larger engines. Of course the weight of such a multi-cylinder head in larger engines is impractical.

▲
A complete multi-cylinder head of an full -sized engine.

▶ A small diesel generator with a crank handle on the camshaft.

▶ A decompression handle to keep open the exhaust valve during cranking.

1 *compression handle for releasing the exhaust valve*
2 *cylinder head*
3 *air filter*
4 *fuel tank*
5 *lubricating- oil filler cap*
6 *engine block*

Usually one or two inlet- and one or two exhaust valves are placed in the head for the combustion air supply and the discharge of exhaust gasses. These are mechanically opened and shut at the right time by means of the cam shaft, guide pulleys, push rods and rockers. Rings around the valve seal ensure the valves shut swiftly. For smaller engines which are manually started it is possible to keep the exhaust valve open with a decompression handle until a certain number of revolutions are achieved with the crank handle. When pulling the handle again the exhaust valve(s) are closed at the right time and compression, combustion etc. takes place; the engine is running.

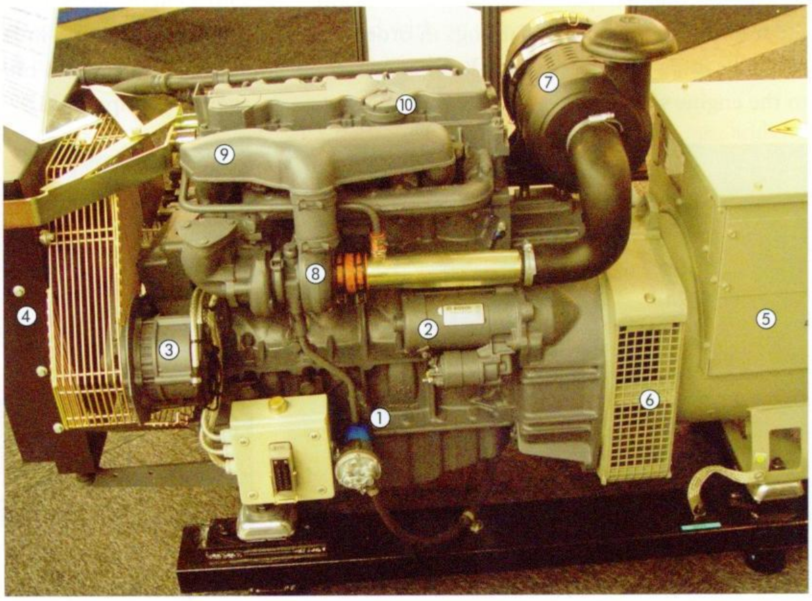
Also see Chapter 14, Starting systems of diesel engines.

Usually the fuel is injected into the combustion chamber in the centre of the cylinder head. This is achieved with a high pressure fuel pump, a high pressure fuel supply line and a fuel atomiser. Often a central so-called block fuel pump is installed. The engine speed is restricted by, for instance, a mechanical speed regulator, the regulator directly controls the amount of fuel supplied to the engine by adjusting the effective capacity of the high-pressure fuel pumps.
With a regulator it is possible to maintain the number of revolutions and therefore keeping the frequency and the voltage of the generator constant during varying generator loads.

Engine lubrication occurs by the use of a gear pump placed in the crankcase and driven from the crank shaft. Via a suction filter which protects the pump the lubricating oil is transported through lines and bores to the different engine parts that require lubrication and cooling. A lubricating oil fine filter ensures that the lubricating oil remains clean.
An oil cooler keeps the lubricating oil temperature at a normal operating temperature. For water cooled engines, the oil is cooled by the coolant circuit. A thermostatic control valve sends enough cooling water through the cooler to ensure that the lubricating oil temperature remains constant. Most engines are cooled using a closed cooling water system, an open cooling water system with for instance canal water, or with the lubricating oil

▶ A four-cylinder diesel genset with an RPM of 1500. The engine is equipped with a turbo-charger without an air cooler.

1 *diesel engine*
2 *starter motor*
3 *alternator*
4 *radiator cooling*
5 *three phase generator*
6 *generator ventilation*
7 *air-inlet filter*
8 *turbo blower*
9 *inlet manifold*
10 *oil-filler cap*



or air. An increasing number of these engines are equipped with a turbo charger and an inter cooler in order to cool the air.

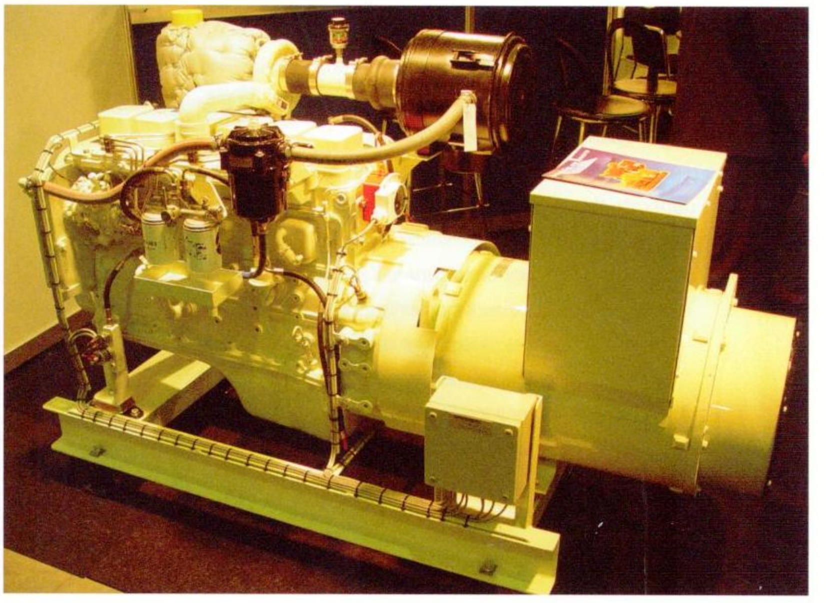
The number of cylinders varies depending on the cylinder stroke volume, the number of revolutions also vary from one to six in in-line-engines or four to eight in V-engines. Cylinder bore varies from ± 50 to 150 mm.

The engines are started either manually or electrically. There are also other types of ignition systems, **hydraulics, spring pressure** or **compressed air** which drive the ring gear.

Also see Chapter 14, Starting systems of diesel engines.

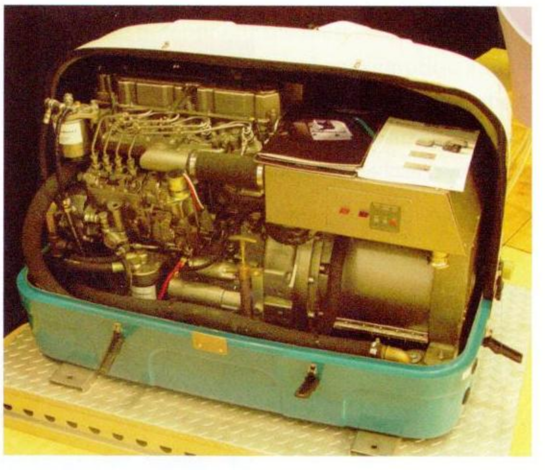
The number of manufacturers and types is large and diverse.

Also see Chapter 21, Diesel engine manufacturers.



▲

A diesel generator.



◄

A very small, compact and silent diesel genset which can placed in the most suitable position in a sailing or motor-yacht. Manufacturer Mastervolt, the Netherlands.

6.2 Category II: Industrial diesel engines from 100 to 5000 kW shaft power, fuel M.D.O., four-stroke, high-speed engines

The slightly larger industrial diesel engines ranging from 100 to 5000 kW are also used for very divergent purposes. An 2000 kW engine is a large propulsion engine for the Rhine-, inland water and coastal navigation and also has a considerable capacity for driving the generator. Applications are among others:

- propulsion of numerous ships especially in inland navigation or for private yachts;
- generation of electricity;
- engines drives for multi-purpose machines such as pumps and compressors.

▼

A Deutz propulsion engine with reduction drive to the controllable pitch propeller on a dredger of the Iran government.

▶

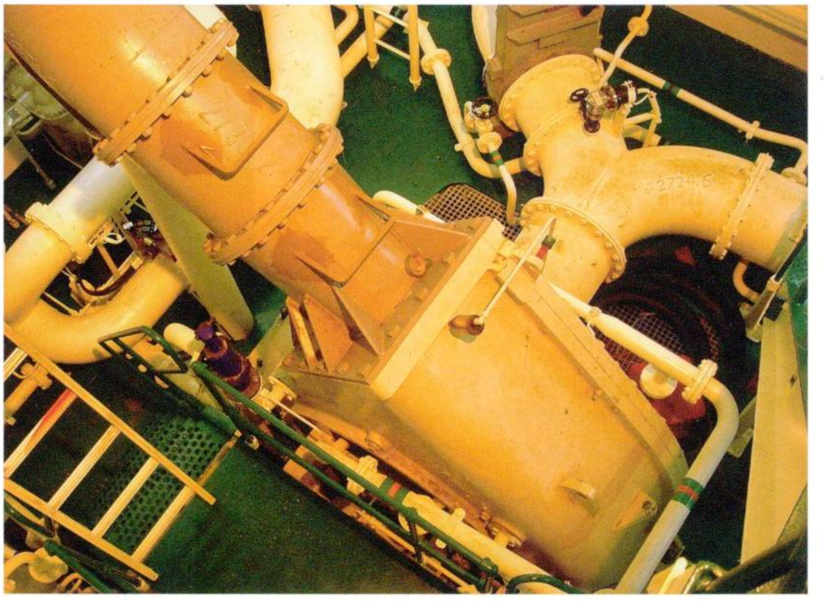
Sailing yachts also have a diesel engine as auxiliary propulsion. Larger yachts also have a diesel genset on board to generate the required electricity.

Important points of interest:

vibration-free set-up
low noise production
compact and lightweight
easy access for repairs
elaborate service network
exhaust-gas emissions below legal requirements
low fuel consumption
reasonable purchase price
easy to operate
reliable

▶

Dredger pumps on dredgers are usually driven by diesel engines.



▶

Rhine and (other) inland navigation ships are becoming increasingly larger in order to be cost-effective.

Ships of 2000 to 8000 tons carrying capacity are no exception. Propulsion usually occurs with high-speed four-stroke diesel engines with diesel oil as fuel. Heavy oil is not allowed in these navigational areas due to the increasingly strict regulations with regard to exhaust-gas emissions. The auxiliary genset for generating electricity or direct-pump propulsion is also becoming considerably larger. Here one uses a similar engine with a lower capacity.

▲

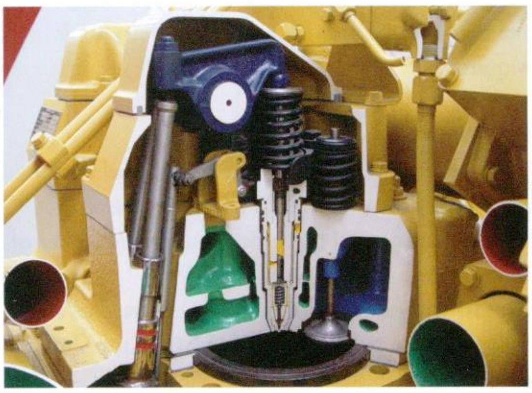
One of two propulsion engines on an inland tanker.

Dozens of engine manufacturers are active in this 'power category', many of which cover a global market.
These diesel engines still use M.D.O., because heavy fuel is not suitable for these fast running engines (over 960 RPM). The number of cylinders for a V-engine can amount to sixteen, eighteen or even twenty cylinders. Cylinder bore ± 80 tot 200 mm.

Construction

The block or frame is constructed from a single-piece of cast iron and in smaller engines sometimes of light metal (a cast alloy with a high aluminium content) and generally has separate interchangeable cylinder liners which are usually cooled on all sides by cooling water(or coolant). In cases of engines with higher power outputs each cylinder has its own fuel pump, however, block fuel pumps are also often used.

Common rail fuel systems are with increasing frequency for these engines, which considerably enhance the potential of the diesel engine with regard to process control, fuel consumption and toxic emissions.
Normally these engines are cooled by means of a closed cooling water- or coolant system. Naturally,



▲

A cross section of the cylinder of a Caterpillar diesel engine with the mechanically driven fuel injector centrally positioned.

▲

An independent fuel pump for larger engines.

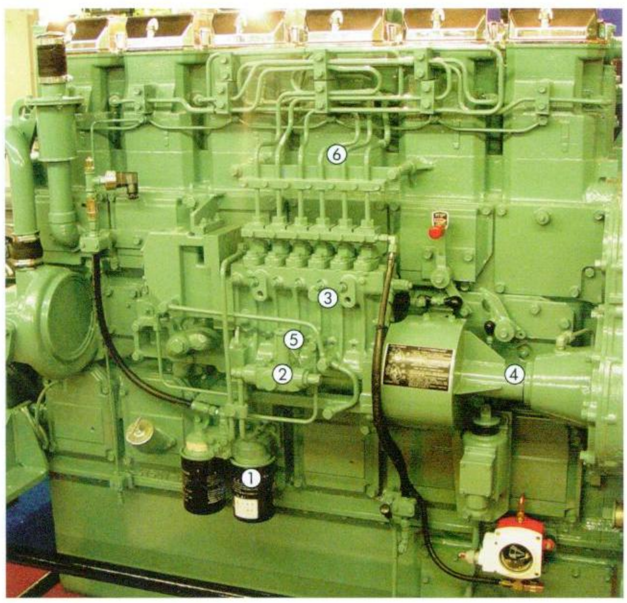
▲

A block fuel pump with six independent high-pressure in-line plunger pumps.

▼

A block fuel pump on a high-speed in-line engine.

1 *fuel filter*
2 *low-pressure transfer pump*
3 *block fuel pump*
4 *propeller shaft from crankshaft*
5 *manual fuel pump for deaeration*
6 *high-pressure pipe to the cylinder*



▲ In smaller engines the piston is cooled internally with an oil jet which is vertically directed through a nozzle.

1 *light- metal piston*
2 *piston rings*
3 *scraper and divider ring*
4 *gudgeon pin*
5 *gudgeon-pin bearing*
6 *connecting rod*
7 *crankshaft*
8 *crank web*
9 *main lubricating- oil duct*
10 *nozzle*

the warmth in the cooling water can be lost by contact of the surface water with air.
The lubrication of theses engines is becoming increasingly important. The heavy thermal and mechanical load of these engines require thorough lubrication and cooling all around the combustion space (piston, piston rings, cylinder liner and exhaust valves), but also a proper lubrication of the highly loaded mechanical parts such as the piston pin, crank pin and the crank shaft.

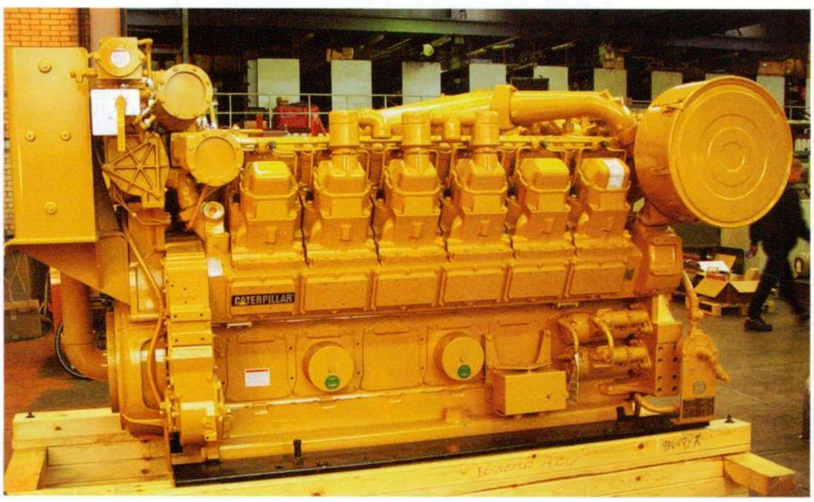
What is further noticeable is that the power density is very high; highly charged with turbo charger groups, a high compression ratio and therefore a high mean effective pressure. What is striking is that the number of V-type versions is extensive for these engines.
Arranging the cylinders in a V-shape makes the engine construction short and compact, which is especially important when there is limited space. Parts that need to be dismounted when overhauling the engine, such as the cylinder head, piston, connecting rod and cylinder liner have reasonable sizes and weights, thus restricting the number of hoists and other auxiliary tools required.

Number of revolutions

The number of revolutions of these diesel engines is generally over 1000 RPM and they therefore fall into **category II** high speed engines (number of revolutions > 960 RPM).
Almost without exception M.D.O. is used as a fuel, since the processtime of H.F.O. is too short to combust properly.
There are, of course, H.F.O. engines on the market with a RPM of 1000 to 1200, but these are exceptions. The group of engines we are presently discussing often has 1500, 1800 or even 2100 revolutions.

Also see Chapter 21, Diesel engine manufacturers.

The mean piston speeds are approximately 10 to 11 m/sec. and the mean effective pressures can go up to 25 bar.



◄ A twelve-cylinder Caterpillar-diesel engine for gensets.

Note the double electric starter motors in the fly wheel, right.

6.3 Category III: Industrial diesel engines from 500 to 30,000 kW shaft power, fuel H.F.O., four-stroke, medium-speed engines

Almost all of this relatively large group of industrial diesel engines use H.F.O.. This also means that the maximum number of revolutions is limited to 1000 to 1200 RPM. According to engine classification, these are really high-speed engines. Some engine manufacturers call these engines 'high-medium speed four stroke diesel engines'. The cylinder bore varies from 160 to 640 mm. A fair number of engine manufacturers, about a dozen, manufacture diesel engines which can run on H.F.O. under all loads and under varying operating conditions.

Definition H.F.O. diesel engine

This is a diesel engine which can start and run on H.F.O. at every load. The heavy oil then has a viscosity of at least 50 cSt at 50° C.

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

▲

A medium-speed Caterpillar Mak in-line engine.

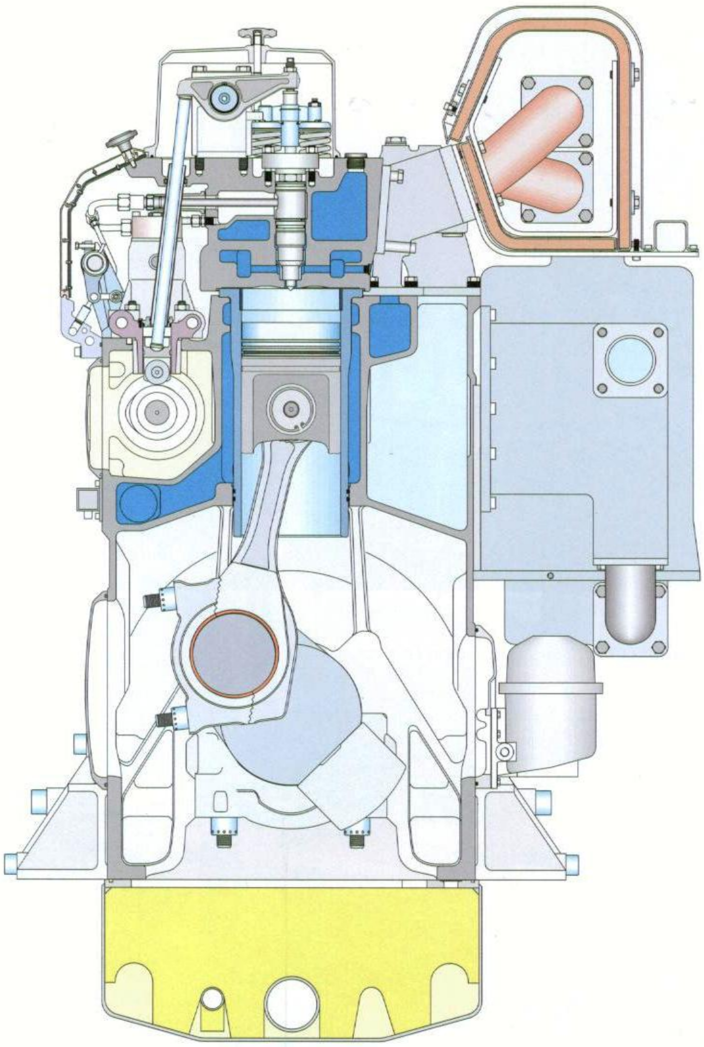
▲

A medium-speed V-engine.

Here a 18 cylinder MAN B&W 48/60 B diesel engine capable of generating an electrical power output of 18.4 MW

◄

A cut-away section of a Caterpillar–MaK-diesel engine with mechanically adjustable cam shaft. A typical example of a medium-speed diesel engine running on heavy oil.



▲

A modern four-stroke heavy fuel oil engine, a Wärtsilä 20.

All cylinder block frames are manufactured out of a single metal piece. The casting is usually from grey cast iron, nodular cast iron or cast iron with different additives used to enhance the material's properties. Due to the strong increase in power densities, the mechanical and thermal loads have increased considerably.

Where possible, the piping canals for lubricating oil, cooling water and air are incorporated into the cast engine block. This way a very robust/whole is obtained with a minimum of extra piping and other connectors (potential defects).

The one piece forced steel cast crank shaft is incorporated into the bottom of the frame and the bottom bearing caps are fixed with two vertical bolts and laterally supported by two horizontal pressure bolts. This way the bottom bearing cap becomes more or less one with the frame.

▲

Crank shaft with mounted counter weights for correct balancing.

▲

The cylinder block of a Caterpillar Mak 43 in-line engine.

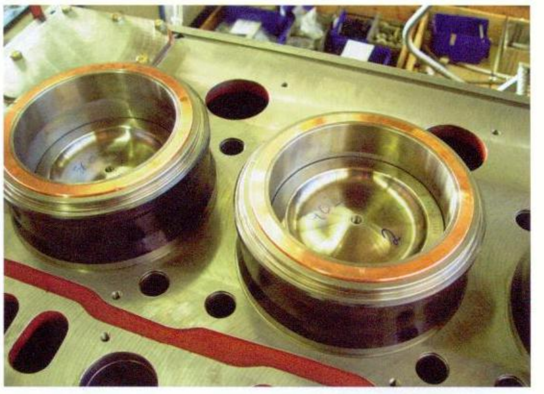
1 *engine block*
2 *position of crank shaft*
3 *scavenge air space*
4 *position cam shaft*

The air supply to the cylinders is integrated into the solid block. Here only the crank shaft bearing caps have been fitted.

▲

The engine block of a V-engine with fitted cylinder liner and cylinder head bolts. In the middle the air inlet manifold.

The cylinder liners can be fitted level in the frame, but with the larger engines in this group a substantial part of these cylinders sticks out of the block.
In the first instance they are conventionally cooled with cooling water, and in the second the liners have been furnished with bore holes through which the cooling water flows. This principle is called 'bore'cooling.



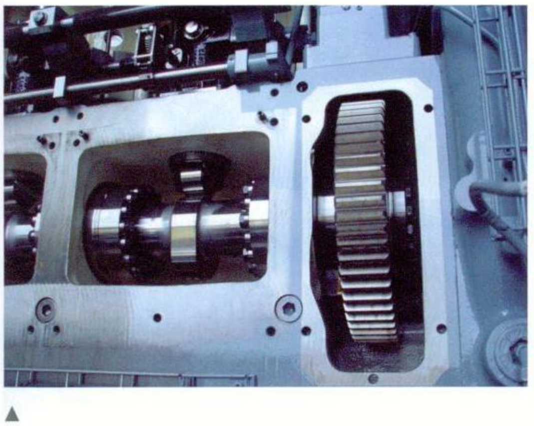
◄
A cylinder block of a Caterpillar–MaK 25 in-line engine.

It is clearly visible how the cylinder liners extend from the block.

▲
An in-line crankshaft.

The crankshaft at final inspection. Tiny burrs are also manually removed.

◄
An in-line cylinder block with mounted fuel pumps, camshaft and cylinder-head bolts.



▲
The camshaft part of one cylinder.

▲
The cylinder liner of a modern highly loaded diesel engine.

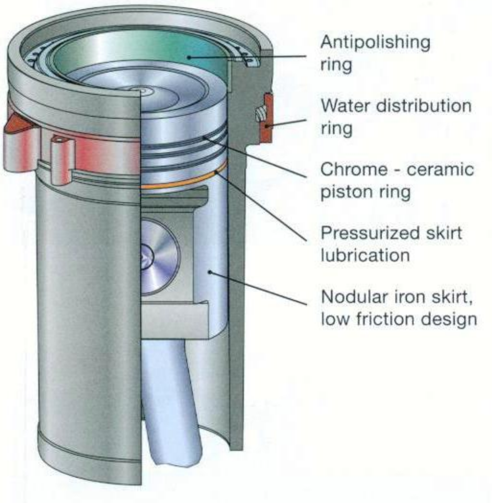
1 *upper collar, for absorbing the high compression and combustion pressures, equipped with bore-cooling*
2 *grooves for coolant tubes, above these the traditional coolant areas*
3 *edge support in engine block*
4 *ant polishing-ring location*
5 *top of the engine block*

▶
A medium-speed diesel engine piston skirt.

One can clearly see the reduction of the material around the gudgeon pin.

▶
A piston in a cylinder with bore-cooling, anti polishing ring and a piston skirt lubricated under pressure in a medium-speed diesel engine.

The piston rings have a chromium-ceramic coating.



The piston is built from two parts and consists of a wrought- or cast steel iron piston crown, which can absorb the high thermal and mechanical loads and contains the piston rings.
The skirt is made of light metal or cast iron and is used for the piston conveyance and the absorption of the lateral forces.
In order to keep the piston as light as possible due to the acceleration- and deceleration forces transmitted to the piston, all material not construction-technically required is removed, especially when piston skirt is manufactured using cast iron.
The piston pin needs to have a large diameter in relation to its length in order for it to be rigid and barely flexible. The piston ring package for larger cylinder bores is lubricated under pressure and also cooling of the piston crown receives special attention.

The connecting rod has to transfer the extremely high forces that are transmitted to the piston to the crank shaft. There are numerous possible versions to achieve this.

Also see Chapter 13, Driving gears.

In the upper part of the **cylinder liner** special rings, **the so-called anti-polish rings** are placed ensuring that the mean effective pressures are considerably increased and the sealing properties of the piston, piston rings and cylinder liner is gradually diminished.

Also see Chapter 13, Driving gears.

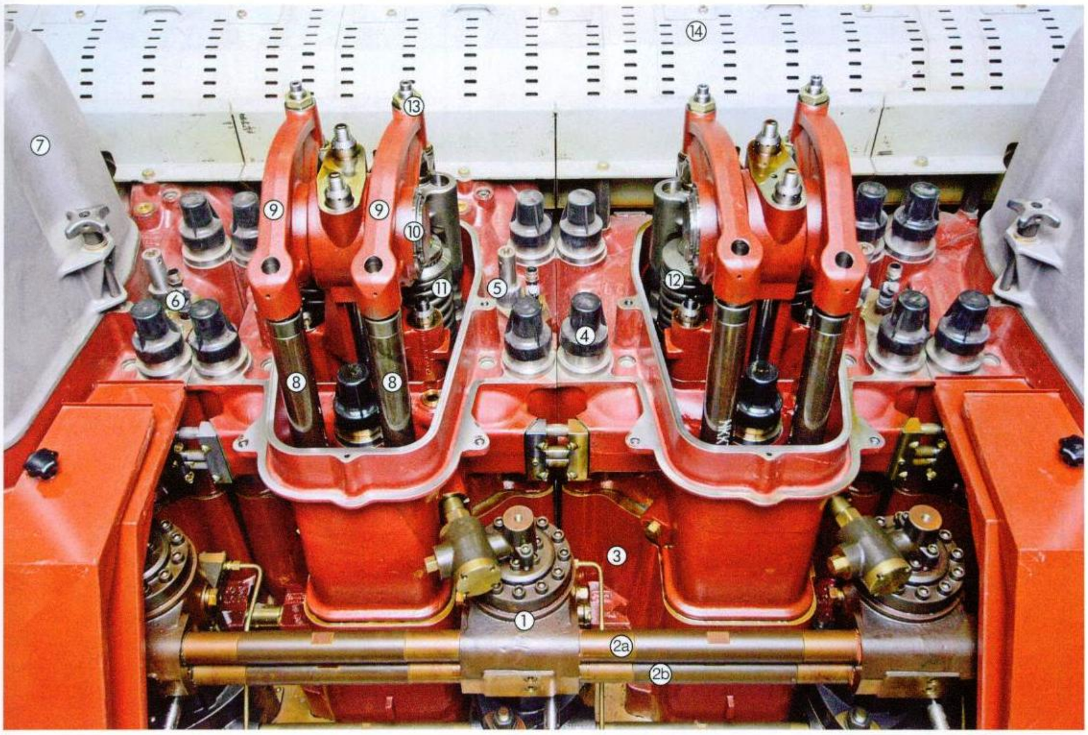
▼
The 'Marinehead'-version of a connecting rod.

By unbolting the eight studs, the piston can be removed without the crankpin bearing via the cylinder liner.

1 *connecting rod*
2 *big end bearing box*
3 *studs*
4 *connecting-rod eye*

▼
An anti polishing ring.

This can be easily removed and plays no part in the construction or sealing of the combustion chamber.



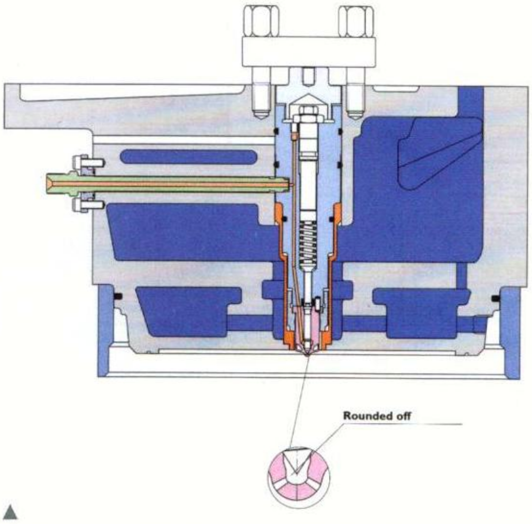
◄ Four- valve cylinder heads in a medium-speed Caterpillar-MaK 43 in-line diesel engine on the feeder-container ship 'Endurance' of JR Shipping.

In the foreground three high-pressure fuel pumps with fuel-supply lines.

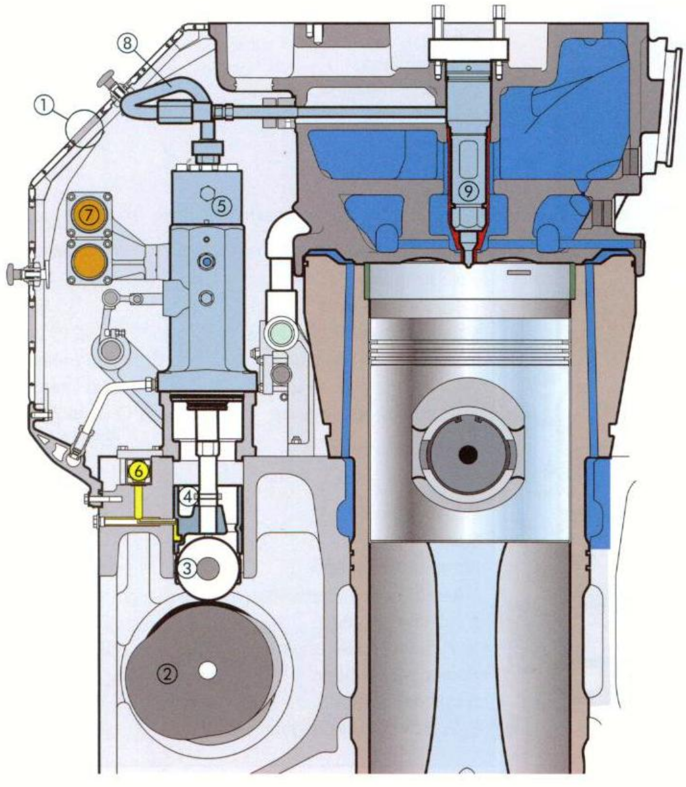
- 1 *high-pressure fuel pump*
- 2a *fuel supply line*
- 2b *fuel discharge line*
- 3 *cylinder head*
- 4 *cylinder- head bolt*
- 5 *safety valve*
- 6 *indicator cock*
- 7 *valve cover*
- 8 *push rod*
- 9 *rocker*
- 10 *rocker shaft*
- 11 *exhaust valve*
- 12 *inlet valve*
- 13 *set screw valve clearance*
- 14 *exhaust-gas manifold*

The **cylinder heads** are made of cast iron in which alloys are increasingly added in order to improve thermal and mechanical properties.
The **gas exchange** occurs **with two inlet valves and two exhaust valves** where the exhaust valves in larger engines are placed in an interchangeable housing.

The exhaust- valves and valve seats are manufactured using a special high performance alloyed steel to ensure that wear and tear are reduced to the minimum.



▲ Sectional view of a cylinder head with in the centre the injector.

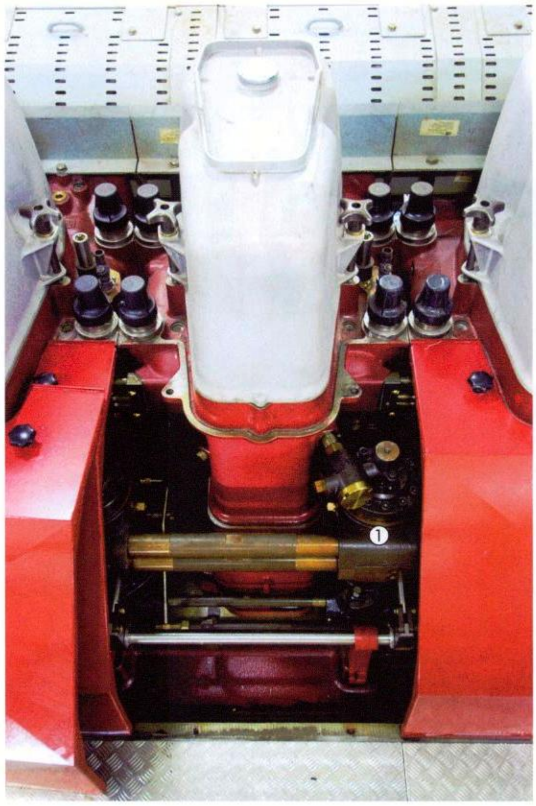


◄ A camshaft-driven high-pressure pump and the injector.

Notice the so-called 'hotbox' for maintaining the temperature of the fuel section.

- 1 *'hotbox' cover*
- 2 *camshaft*
- 3 *roll*
- 4 *roller guide*
- 5 *high- pressure fuel pump*
- 6 *lubricating-oil supply*
- 7 *fuel supply*
- 8 *high-pressure delivery pipe to injector*
- 9 *injector*

▶
View of a cylinder head with the fuel pump (1).

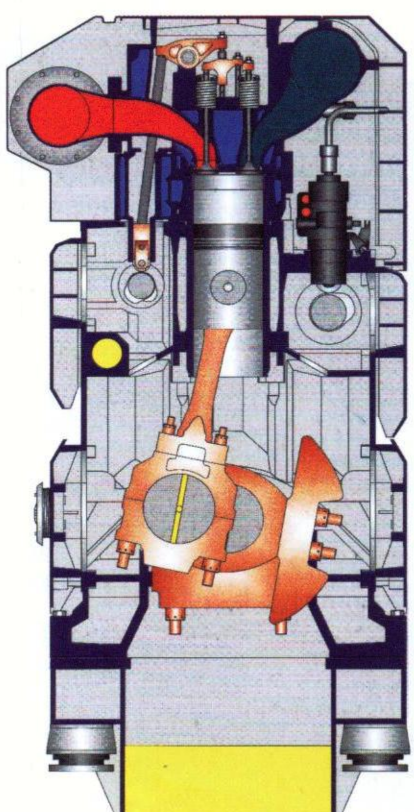


▲
The high fuel pressure pump, a very important part of the diesel engine.

The centrally placed fuel atomiser in smaller engines often has its own double-walled fuel line to prevent fuel leakage, the larger engines have a fuel supply through the cylinder head itself.

All engines, without exception, are equipped with a turbo charger by means of (a) turbo charger group(s) and fitted with an air cooler which ensure that the compressed air is adequately cooled, thus increasing the specific mass of the air and consequently the number of kilograms of air per time unit which are added to the engine. This allows for more fuel to be combusted in the cylinder and therefore extra power generation.

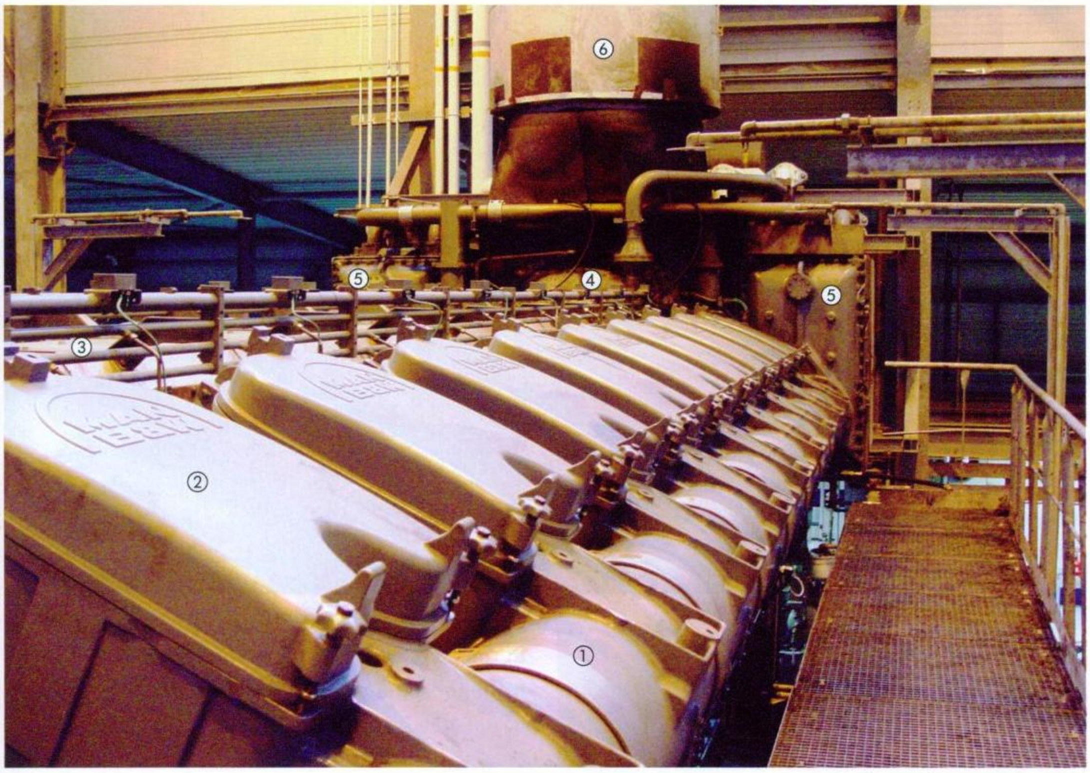
▶
Sectional view of a modern MAN–B&W four-stroke diesel engine with required details.



Main particulars

Cycle	:	4-stroke
Configuration	:	In-line
Cyl. Nos. available	:	5-6-7-8-9
Power range	:	950-1935 kW
Speed	:	900/1000 rpm
Bore	:	210 mm
Stroke	:	310 mm
Stroke/bore ratio	:	1.48:1
Piston area per cyl.	:	346 cm^2
Swept volume per cyl.	:	10.7 ltr.
Compression ratio	:	15.5:1
Max. combustion pressure	:	210 bar
Turbocharging principle	:	Constant pressure system and intercooling
Fuel quality acceptance	:	HFO up to 700 cSt/50° C (BSMA 100-M9)

Power lay-out		MCR version	
Speed	rpm	900	1000
Mean piston speed	m/sec.	9.3	10.3
Mean effective pressure:			
5 cylinder engine	bar	23.6	22.4
6, 7, 8, 9 cylinder engine	bar	24.8	24.0
Power per cylinder:			
5 cylinder engine	kW/cyl.	190	200
6, 7, 8, 9 cylinder engine	kW/cyl.	200	215



◄
In large four-stroke engines the turbo-charger groups with the air coolers placed below take up much space.

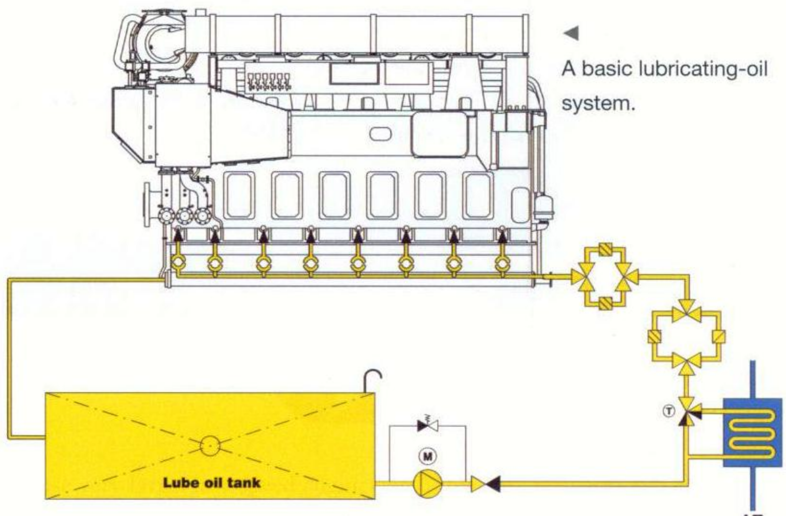
Especially in V-engines the overall width exceeds the total height of the engine. The exhaust-gas manifold can be seen in the background.

1 *air-inlet manifold*
2 *cylinder- head cover*
3 *exhaust- gas manifold*
4 *turbo- blower*
5 *air coolers*
6 *central exhaust-gas manifold*

Lubrication is taken care of by the displacement pump, which is driven by the engine. Prior to the initial start, the engine can be pre-lubricated by an external, electrically driven pre-lube pump. There are also electrically driven main lubricating oil pumps.

The lubricating oil is not only purified in fine filters on the feed side of the lubricating oil system but also in centrifuges or separators that clean the oil thouroughly by removing water and regular contaminants such as metal particles, combustion products and dust.

Also see Chapter 11, Lubrication of engines.



◄
A basic lubricating-oil system.

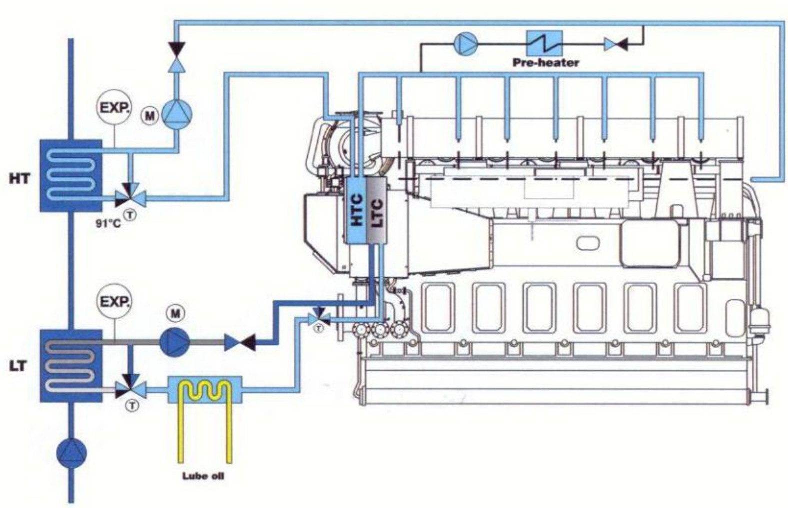
Cooling takes place by an engine driven cooling water pump in a standard closed system with pressure build up obtained using a buffertank or an open expansion tank.

The temperature of the cooling water lies between 80 and 90° C.

Also see Chapter 10, Cooling diesel engines.

This group is very frequently used on various ships for propulsion, either as an auxiliary engine for generating electrical energy or for directly driving the pumps.

▼
An overview of a coolant system with a high- and low- temperature section.



▲

Building a diesel generator set at Caterpillar–MaK in Kiel, Germany.

▶

A Caterpillar–MaK four-stroke diesel engine for the genset in the engine room of a large container ship.

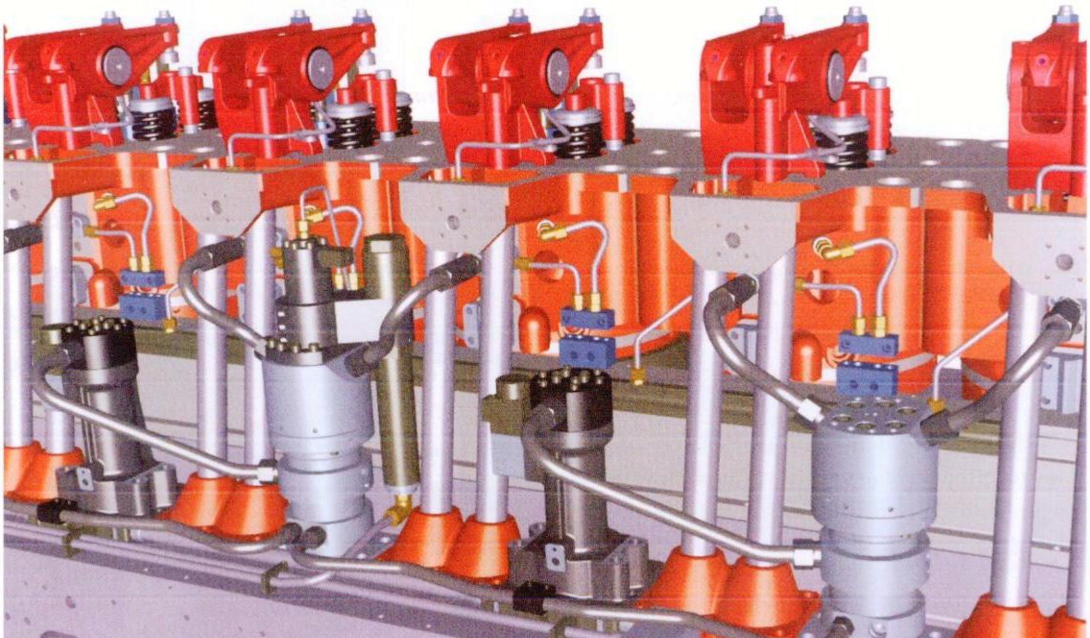
▲

Fourteen eighteen-cylinder MAN–B&W – V-48/60 type A&B diesel engines in a diesel-electric power plant in San Pedro Sula, Honduras, Central America.

In the merchant navy with a total line- up capacity from approxiametely 15 MW, the main engines as well as the auxiliary engines use the same heavy fuel. This cuts the fuel costs considerably.

These engines are very often used in diesel plants. The latest developments (2008) show that in particular the fuel injection systems continue to be further optimised with among others common rail injection systems, in which the injection period, the moment of injection and the amount of fuel are electronically regulated.

Also see Chapter 9, Fuel-injection systems.



▶

A modern fuel-injection system, the Common-Rail System for Wärtsilä four-stroke engines.

6.4 Category IV: Industrial diesel engines of 1500 to 100,000 kW shaft power, fuel H.F.O., two-stroke with crosshead, low-speed

▶

All tankers, bulk carriers and large container ships use the bulky low-speed two-stroke crosshead engine on heavy fuel oil for propulsion.

These are the immensely large and also remarkably high (due to the crosshead construction) diesel engines also referred to as the **Cathedrals** among engines. They can get as high as 14 metres at a length of 30 metres and and weighing 2300 tons.

▲

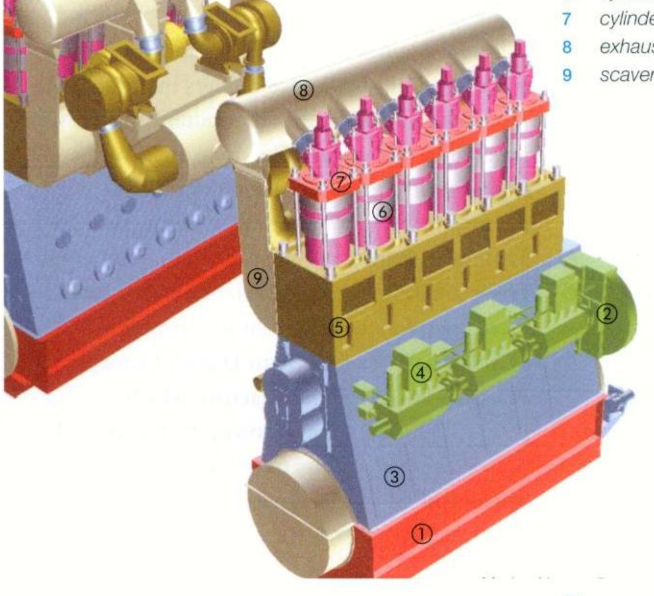
On the cylinder-head mounting floor.

The curved lines show the lubricating-oil supply ducting for for the operation of the exhaust valves. The exhaust valves (1) with valve casing are exceed two metres.

▼

Two-stroke crosshead engine construction.

1 *crankcase bedplate*
2 *crankcase*
3 *A-frame*
4 *fuel- and lubricating-oil pumps for exhaust-valve actuation*
5 *cylinder block*
6 *cylinder liners*
7 *cylinder covers*
8 *exhaust-gas manifold*
9 *scavenging-air manifold*



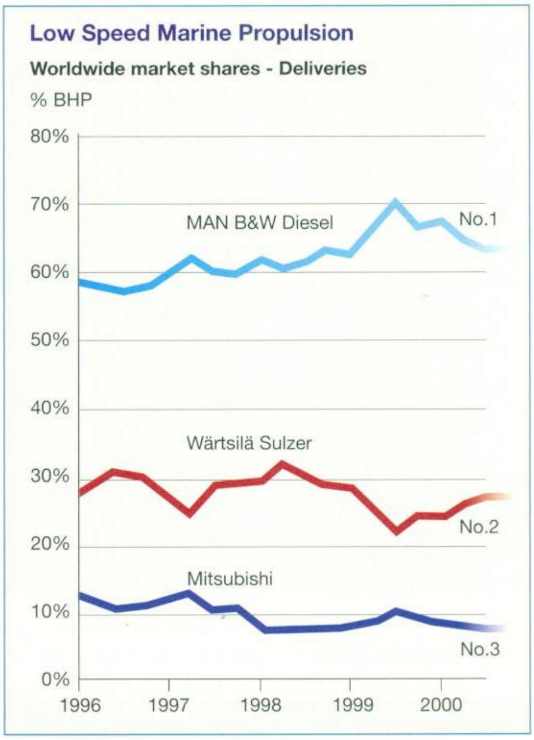
These very large, low speed engines with the number of revolutions from 60 to 250 RPM at a cylinder bore of 260 to 1080 mm and are mainly used for propulsion of very large container ships, oil- and ore tankers and sporadically in diesel powerplants for the generation of electricity.

▼

Large high-speed container ships are equipped with the largest two-stroke crosshead engines in the world with a shaft power of up to 100,000 kW.

▲
A large container ship leaves the port of Rotterdam.

They always directly drive a propeller or generator, so the number of revolutions is always equal to that of the propeller or the generator. This makes their installation transparent and straightforward.
All these engines run on H.F.O. and are for the most part built by licence holders in South Korea, Japan and China, as the shipbuilding industry is situated in this region. The dry weight of these engines range from 100 tot 2300 tonnes.



▲
There are only three two-stroke crosshead-engine manufacturers.

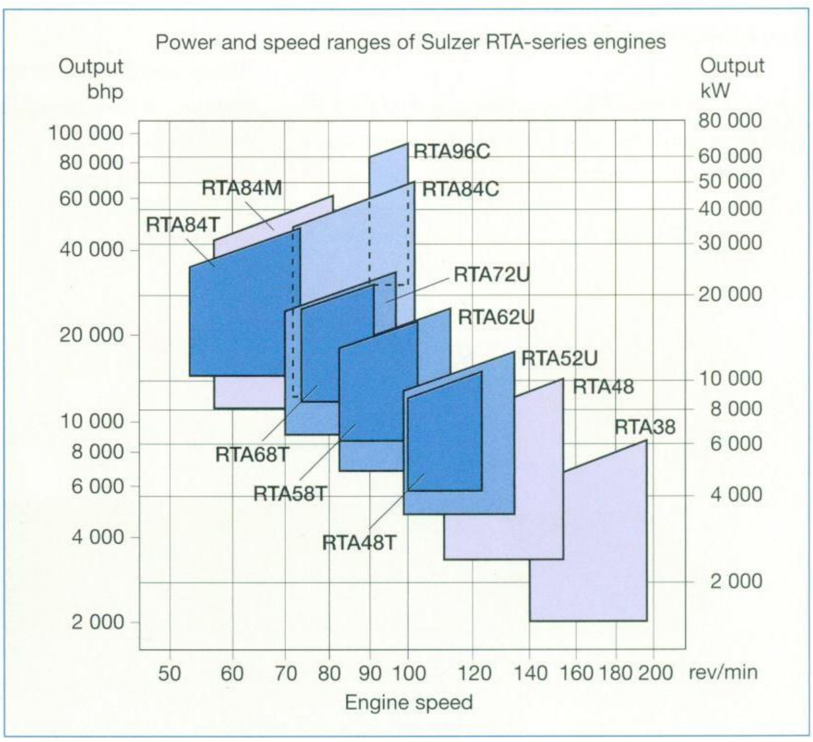
Engine builders

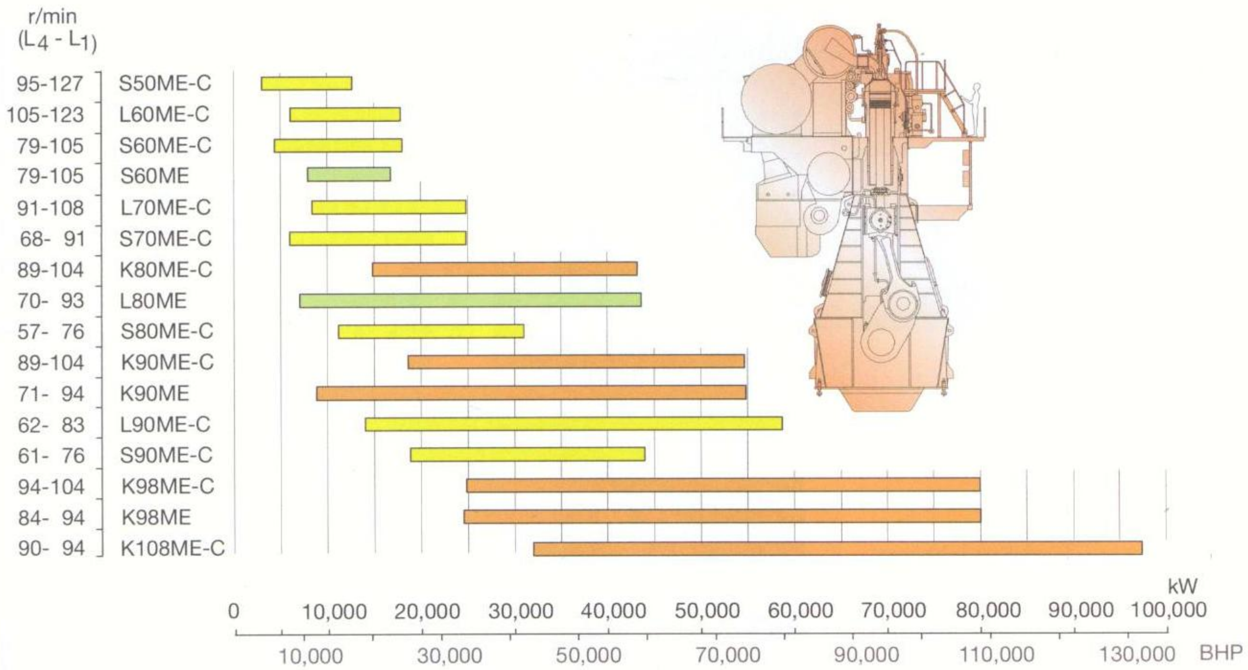
There are only three manufacturers that build these big two-stroke crosshead engines:

MAN–B&W	a German-Danish factory;
Wärtsilä–Sulzer	Finnish factory which manufactures the traditional Swiss two-stroke engines;
Mitsubishi	a Japanese factory.

▼
Globally, there are a considerable number of engine factories which build engines of a certain design 'under license'.

▼
The engine program of the Wärtsilä-Sulzer-RTA two-stroke crosshead engines with a cylinder bore of 380 to 960 millimetres. The number of revolutions varies from approximately 50 to 200 per minute.





▲

The engine program of the MAN–B&W ME-series.

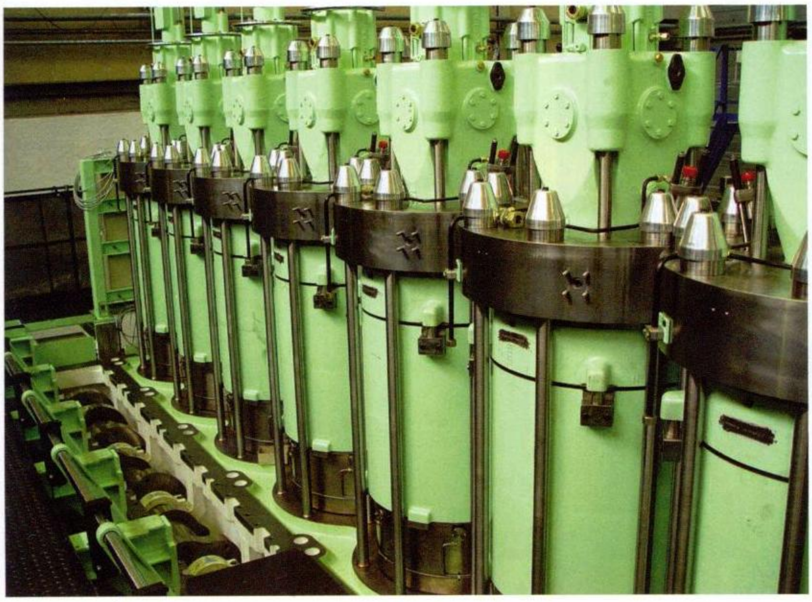
The ME is the version with the electronically controlled fuel injection. Today even smaller cylinder bores are delivered in the ME-version.

Construction

These engines are entirely manufactured from a partially welded construction, such as the engine foundation and the A-columns including cast parts such as the cylinder beam and cylinder liners. The cylinder head is made using forged steel as is the crank shaft.

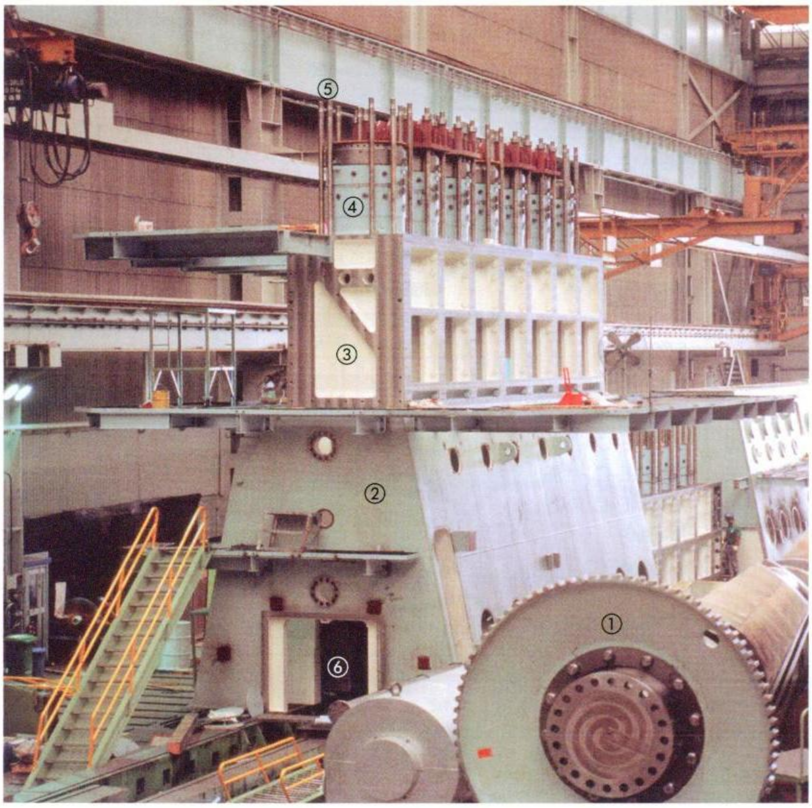
▼

A cylinder block with liners.



▼

The welded A-frame.



▶
Building a crosshead engine.

1 *crankshaft with flywheel*
2 *A-frame*
3 *cylinder block*
4 *cylinder liners*
5 *cylinder-head bolts*
6 *crankshaft location*

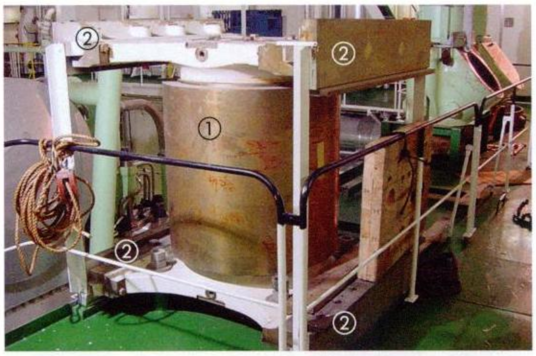
▼
The camshaft.

1 *cylinder block*
2 *cylinder liner*
3 *cylinder-lubricating point*
4 *cylinder-head bolt*
5 *camshaft*
6 *cam*
7 *camshaft bearing*
8 *sleeve-coupling bearing, hydraulic*

On the specially adjusted ship foundation plate the engine bedplate is affixed by means of many holding down bolts. The forged crank shaft is situated inside the engine bedplate. It can be as long as 25 metres with a weight of 150 tons! The welded steel plate A-columns for example two, three or four cylinders,are then placed on top of the bedplate.
These are joined together with tie bolts. The cast iron cylinder beams are placed on the columns with very long drawing rods/pull rods which join the three parts together.

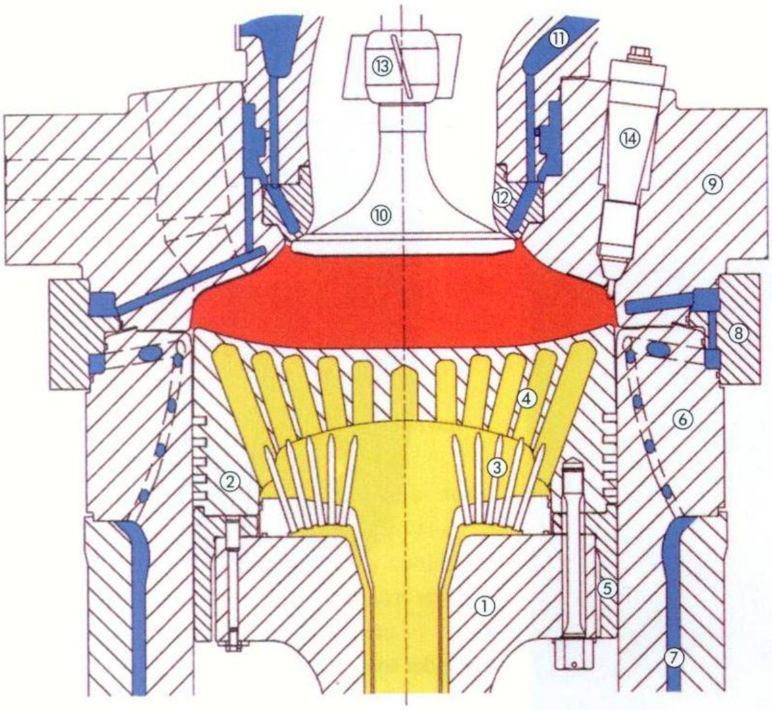
▲
A large crankshaft in a two-stroke crosshead engine with a turning circle of 1680 mm.

The forged steel cylinder heads are attached to the cylinder beam and they contain the hydraulically driven central exhaust valve, two or three fuel valves, a relief valve, an indicator cock and a starter air valve.
The driving gear comprises a piston to which a piston rod is connected, a crosshead with guide shoes attached to it which transfer the lateral forces of the crank shaft mechanism to the crosshead guide fixed to the A-frame.



◄ One spare crosshead.

1 crosshead pin
2 crosshead-guide shoe



◄ The combustion chamber.

1 piston rod
2 piston
3 cool inlet drillings for lubricating oil
4 honeycomb holes
5 guide ring
6 cylinder liner
7 bore cooling
8 coolant circulation
9 cylinder head
10 exhaust valve
11 exhaust-valve manifold
12 valve seat
13 valve rotator
14 fuel injector

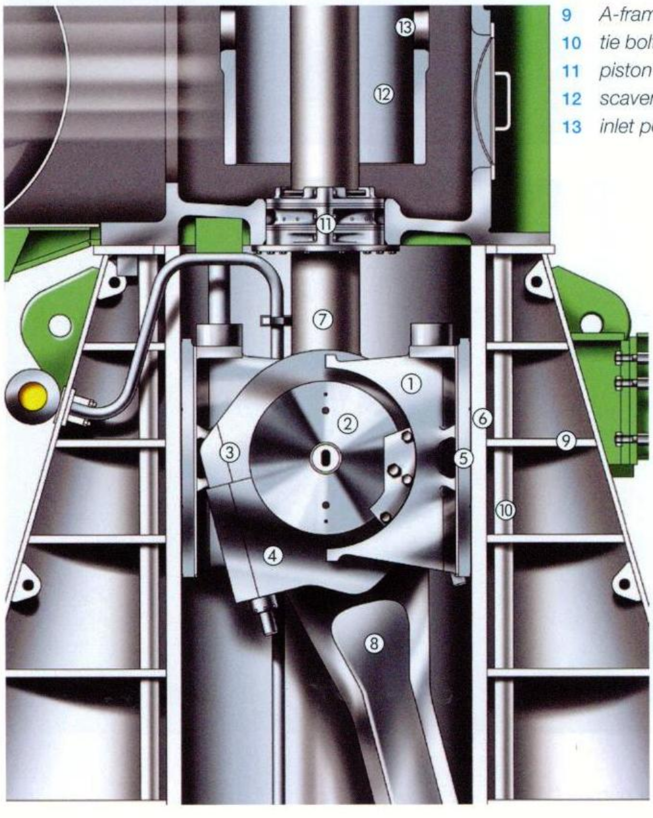
blue: coolant

yellow: lubricating-oil for piston cooling

red: exhaust gases

▲ A piston with piston rod.

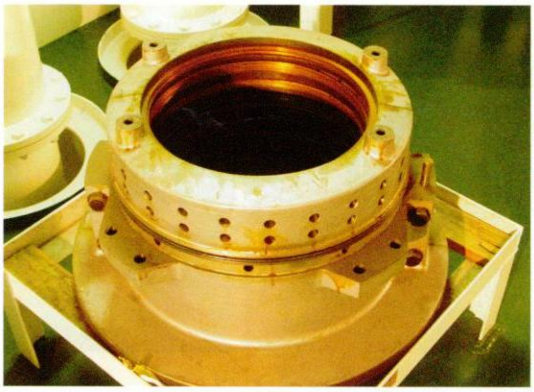
The flanged piston rod has almost the same diameter as the piston.



▼ The crosshead.

1 crosshead
2 crosshead pin
3 upper bearing cap
4 lower bearing cap
5 crosshead guide shoe
6 crosshead guide
7 piston rod
8 connecting rod
9 A-frame
10 tie bolt
11 piston-rod stuffing box
12 scavenging- air space
13 inlet port

▶
A piston-rod stuffing box for the separation between the scavenging-air space above the sealing rings and the crankcase below the sealing rings.

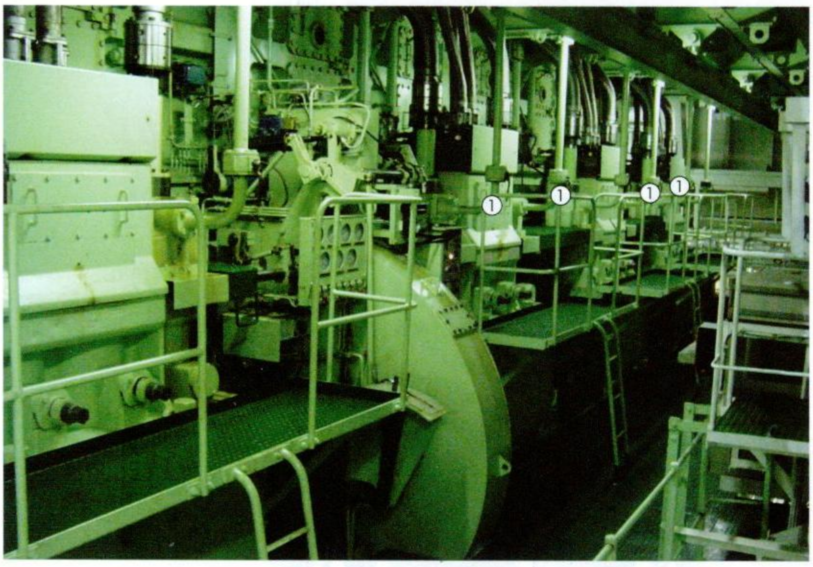


The forged connecting rod is fixed to the crosshead and forms the link to the crank shaft with a crank pin. The piston rod-stuffing box ensures a excellent seal between the scavenging space surrounding the inlet ports in the cylinder liner and the crank case.

▼
Exhaust-valves at the factory for repair.

▲
A cylinder liner.

1 *inlet ports*
2 *support collar with the cylinder block*
3 *cylinder lubricating-oil inlet holes*
4 *thickened upper collar*
5 *bore cooling*



▲
High-pressure fuel pumps (1).

For most existing engines the cam shaft for driving the fuel pump and the hydraulic pump driving the exhaust valves are driven from the crank shaft either using cogwheels or a chain.
The latest types have a very advanced fuel system such as the common rail system and no longer have their own cam shaft.

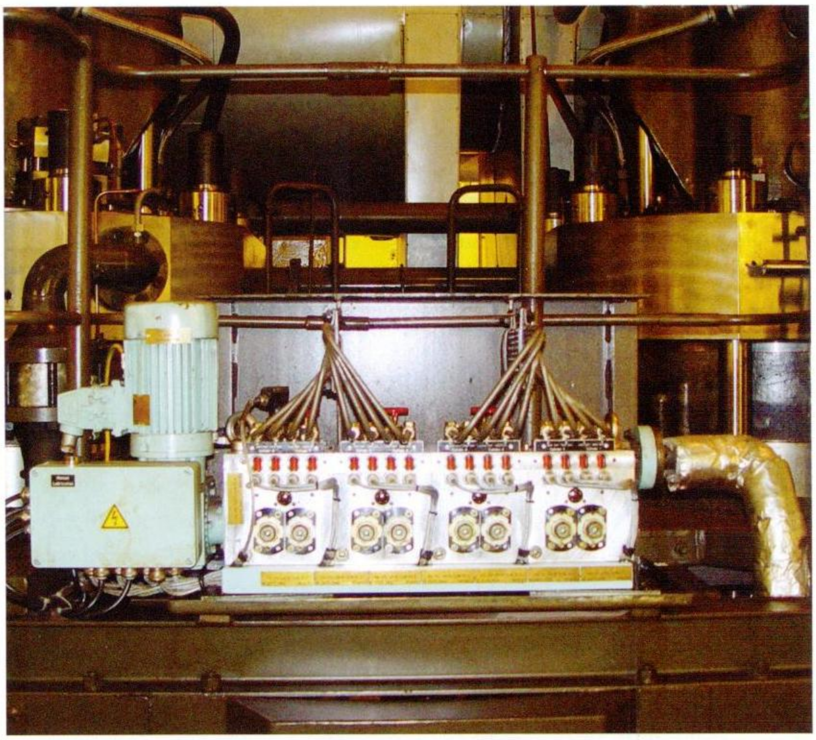
Also see Chapter 9, Fuel-injection systems and Chapter 21, Diesel engine manufacturers.

The lubrication of these crosshead engines consists of two separate systems:
- the cylinder lubrication system;
- the crank case lubrication system.

Cylinder lubrication system

A series of small plunger pumps driven either indepently by the engine itself, or by a frequency controled electro motor which forces special cylinder lubricant through the drillings in the liner and is spread around the cylinder via grooves machined in the cylinder liners. This is done to ensure that the lubricant is injected at the right time and at the most effective piston height. The cylinder lubricant lubricates the piston, the piston rings and the cylinder liner. A certain amount of the lubricating-oil should combust odourlessly and ashlessly, another part seeps along the liner and drips to the bottom of the scavenging air space from which it is manually tapped.

Also see Chapter 11, Lubrication of engines.



▶

A cylinder lubrication unit for a cylinder with eight small independent plunger pumps.

Crank case lubricating system

This system is operated by electro motor driven displacement pumps.
The lubricating-oil is pumped past the main- and crank pin bearings, the crosshead and the cam shaft. Also, the piston is often internally cooled using this circuit. Furthermore the exhaust valve drive and the reserve gear use this system.

Also see Chapter 11, Lubrication of engines.

▶

A main lubricating oil pump.

Cooling system

Large two-stroke engines have two cooling water systems with two temperature levels, the H.T.-(high temperature) and the L.T.-(low temperature) system.
The **high temperature system** is used to cool the cylinder liner, the cylinder head and the exhaust valve housing. Pistons were also sometimes cooled using this system.

The **low temperature system** cools the scavenging air and the lubricating oil.

Also see Chapter 10, Cooling diesel engines.

▶

Two plate heat exchangers.

Air supply

The **air supply** is usually provided by several turbo charger groups wich compress the combustion air; the air is subsequently cooled and led to the scavenge air pipe which runs along the length of the engine. Via the scavenging air space surrounding the cylinder liner the air flows into the cylinder when the piston is in B.D.C. and releases the inlet ports.

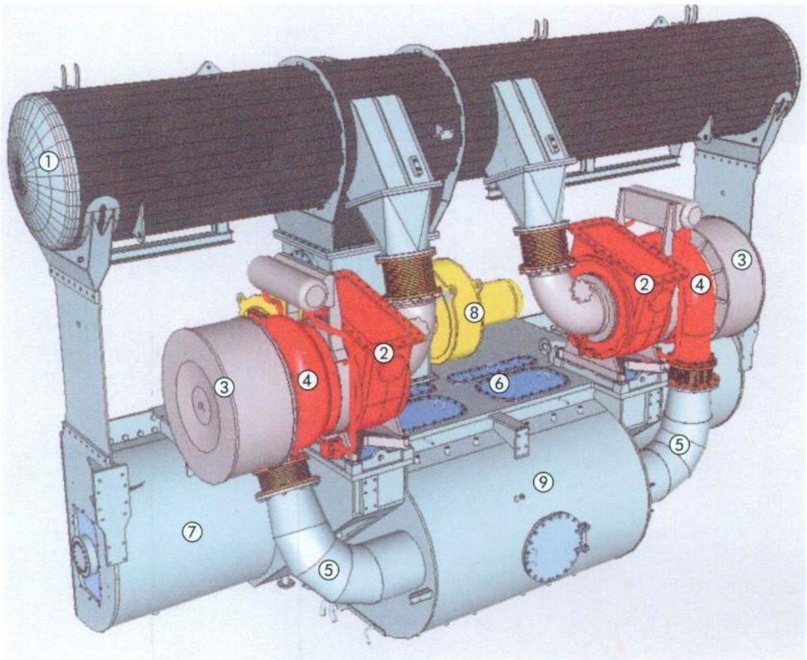
In case of a **low load** the turbo charger groups provide too little air for an optimal combustion cycle and an electrically driven auxiliary blower is automatically switched on.

Also see Chapter 12, Air supply.

▼

An overview of the turbocharger of a two-stroke crosshead engine.

1 *exhaust-gas ducts*
2 *exhaust-gas turbine*
3 *air filter*
4 *air-compressor space*
5 *compressed air*
6 *air cooler*
7 *air inlet*
8 *auxiliary blower*
9 *mist catcher*



▶

A cut-away diagram of a turbocharger group.

Fuel

For conventional fuel pumps the fuel supply is provided by a cam shaft driven plunger pump. The cylinders are fitted with two or three fuel atomisers surrounding the centrally positioned exhaust valve and injecting the fuel almost horizontally into the cylinder.

Both plunger- as well as valve-controlled fuel pumps are used. The recently (2008) marketed electronically controlled common rail systems have numerous advantages and are increasingly applied.

Also see Chapter 9, Fuel-injection systems.

▼

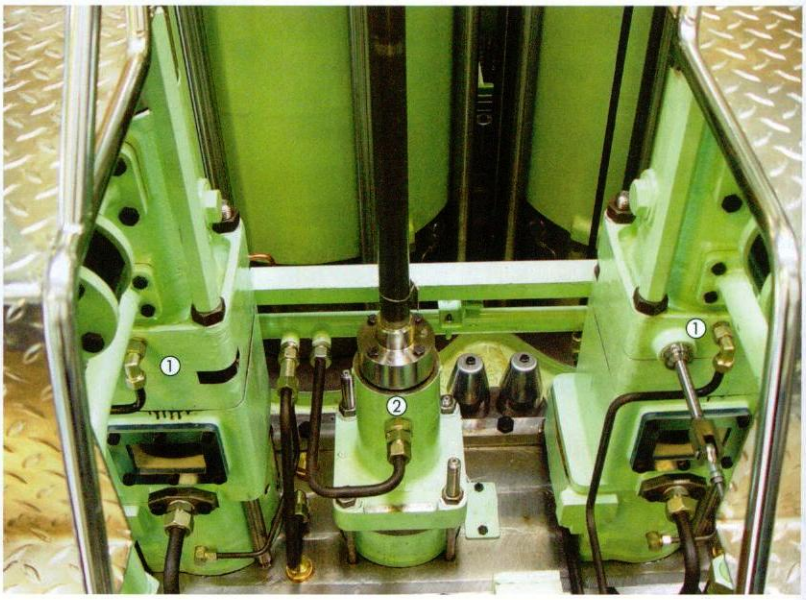
The gear drives from the camshaft of:

1 *the fuel pump;*
2 *the hydraulic-oil pump for the exhaust valve.*

▲

One of the fuel injectors.

There are usually two or three assembled around the central exhaust valve.



Reversing the revolution direction

Two-stroke engines up to a cylinder bore of about 600 mm are often equipped with an adjustable pitch propeller which allows for the engine to operate in only one direction of rotation.
For larger engines and therefore larger propellers one opts for a fixed pitch propeller, which requires the rotation direction to be reversible. This is made possible by moving or turning the cam shaft under a lubricating oil pressure causing the air start valves and fuel injection on the various cylinders in order to obtain a different operating order.

Also see Chapter 14, Starting systems of diesel engines.

Number of cylinders

This can vary from four to fourteen cylinders and the capacity from approximately 1500 to 100,000 kW. All engines have in-line cylinders.

Comment

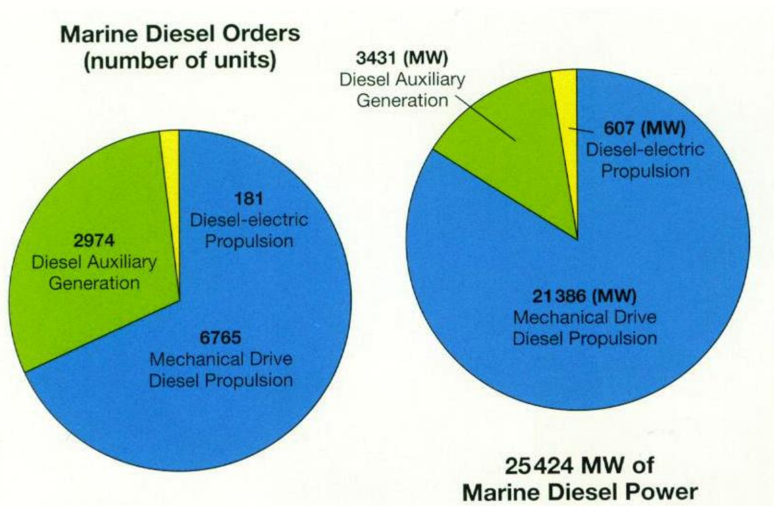
The capacity of the small two-stroke crosshead engines overlaps the power supply of the 'larger' four-stroke trunk piston engines. For the latter the power can reach approximately 30,000 kW.

The choice of a two-stroke crosshead engine or a four-stroke trunk piston engine depends on many factors such as:

- desired power output;
- desired number of revolutions;
- available space;
- weight of the engine;
- specific fuel consumption;
- specific lubricating oil consumption;
- economic aspects;
- ship yard lay-out for certain type ships;
- delivery time
- service package.

The client's preference for, for instance, other shipping companies and/or investors often plays an important role as well.

◄ The 'Berlin Express' of Hapag-Lloyd has a capacity of 8600 TEU and an engine output of 68,000 kW. It is one of the largest container ships launched in 2004.



◄ A overview of the orders per year for diesel engines with a shaft power exceeding one Megawatt.

blue: left – the number of propulsion installations
green: left – the number of generator sets
yellow: left – the number of engines used for diesel-electric propulsion
blue: right – the total power output
green: right – the total power output
yellow: right – the total power output

>CH 7

Use of materials for diesel engines





Every engine's basis since they first came onto the market in the nineteenth century is a top-quality cast engine block.
Apart from cast iron, steel and light metals are the most important materials used in engines.

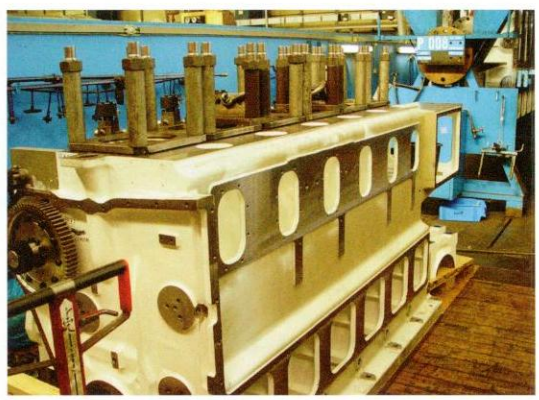
A cast engine block aligned for the initial machining process.

▲
The casting of a large block for a four-stroke diesel engine.

The casting time is very important for the mould quality. A large block with a weight of 115 tons must be cast within 100 seconds. If this time is exceeded, the quality is such that the block is rejected. The costs, approximately € 100,000.- for manufacturing the mould and the melting down of the base materials are lost.

▲
A cast engine block just out of the mould.

The deburring and cleaning of the rough block must take place.



◄
A cast block of an in-line engine in manufacture.

7.3 Steel

This is an iron-carbon alloy with a maximum carbon content of 1.7%.

Properties:

- malleable (+);
- ductile (+);
- weldable (+);
- light in welded constructions (+), for example the A-frame for crosshead engines;
- good machinability, such as planing, milling, drilling, grinding and polishing (+);
- high melting point of approximately 1450° C (+);
- **over 0.3% carbon hardness**, surface hardening possible (+);
- soft (–).

Steel in a steel plate form is often used in welded constructions such as bedplates, crosshead engine A- frames, air inlets found in crosshead engines, air cooler housings, various supports and exhaust gases manifolds.

▲
The manufacture of two-stroke crosshead engines at MAN–B&W, Frederikshavn, Denmark.

foreground: a welded steel crankshaft bedding or lower crankcase of a two-stroke crosshead engine
background: a welded steel A-frame

7.4 Cast steel

This is an iron-carbon alloy with a carbon content of between 0,5 and 2%. The base material is white pig iron.

Properties:
- can be cast in crude moulds(+);
- extremely high tensile strength (+);
- malleable (+);
- high melting point (+);
- possibility of strong yet light constructions (+);
- viscous (–);
- shrinkage is twice as high as that of cast iron (–);
- due to its viscosity, only used in simple casting moulds (–);
- requires annealing to resist high casting pressures (–).

Casting properties are improved by adding manganese and silicon. By adding 0.5 to 1.0% molybdenum, a higher tensile strength at higher temperatures is achieved. In older, still existing and larger crosshead engines, it is among other things used in cylinder heads, high pressure shut- off valve housing, piston crowns and cylinder heads in four-stroke engines.

7.5 Forged steel

This is an iron-carbon alloy with 0,15 to 0,5% carbon content.

Properties:
- malleable (+);
- high tensile strength also at high temperatures (+);
- ductile (+);
- machine able (+);
- coarse moulding which requires machining operation after casting (–).

Today, generally used for the cylinder heads of crosshead engines due to the mechanical and thermal load increases.
Crankshafts for all types of engines, connecting rods, piston rods and cam shaft components also are made of this material.

7.6 Steel alloys

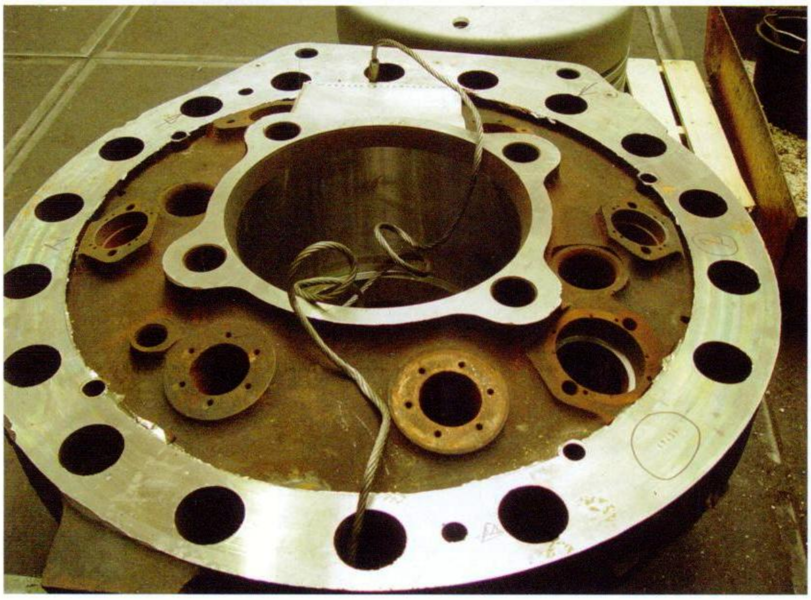
Alloys are added to steel, an iron-carbon alloy, in order to enhance certain physical properties, or for instance to create new properties.

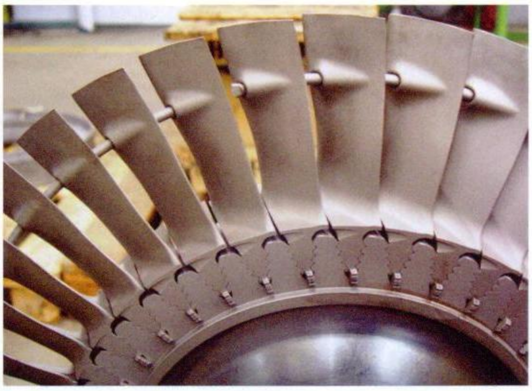
Alloy elements are metals such as chromium, nickel, molybdenum, tungsten, manganese and vanadium.

Various alloy elements provide steel with particular properties, for example:

chromium	– increases the hardness and increases toughness
nickel	– increases the tensile strength and is anticorrosive
molybdenum	– increases the tensile strength and maintains the hardness up to 600 °C
tungsten	– makes steel heat resistant
manganese	– increases the tensile strength and decreases wear and tear
vanadium	– increases ductility

▼
A cast-steel cylinder head of a two-stroke crosshead engine.





▲
The shaft and the blades in the exhaust-gas turbine of the turbo blower are manufactured from high-alloyed steel for resistance to high temperatures and corrosion caused by chemicals.

The crankshafts of high load fast running engines are often made of **chromium nickel- or chromium molybdenum steel.**
The bearing surface of the crank pins and journal bearings are then hardened, thus increasing their wearability.
Steel used in exhaust valves are often alloyed with 8 to 12% chromium and for example silicon. 'Nimonac 80A', an alloy of 80% nickel, chromium, titanium and aluminium is resistant to the high temperatures and the corrosive products contained in exhaust gases. This is, however, an expensive material and today frequently used in the exhaust valves of two-stroke engines. The cost of the material has decreased considerably in recent years.

◄
Steel Alloys.

These are used for the manufacture of parts for diesel engines, for example:
1 *exhaust valves*
2 *exhaust-valve seats*

◄
Crankshafts are the most heavily loaded parts in a diesel engine; they are manufactured from high-alloy forged steels.

Steel Alloys are, used in a large number of diesel engine parts, such as toothed wheels, fuel pumps, shafts, cams, atomisers, valve springs and valves. Each component has different material requirements, which is the reason that there are many different alloyed steel types.

7.7 Aluminium

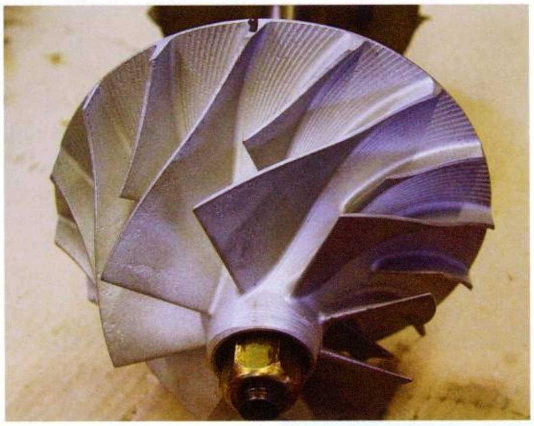
Properties:
- high fatigue resistance (+);
- good bearing properties (+);
- light weight (+);
- good heat conductor (+);
- good strength properties at high temperatures (+);
- high wear rate (–);
- high coefficient of expansion (–).

By forming alloys with aluminium certain properties are enhanced.

Copper, nickel and magnesium ensure good heat conduction and favourable strength properties at high temperatures, the so-called Y-gamma alloy. This is called an **aluminium-copper alloy.**
With **silicon, copper, nickel** and **magnesium** this 'light metal'-alloy has a low coefficient of expansion and a reasonable wear rate. This is called an **aluminium-silicon alloy.**

Use

'Light metals' with aluminium as the main component are used for engine blocks of small diesel engines, the pistons of fast-running diesel engines, the piston skirts of medium-speed diesel engines, tri-metal axle bearings in all engines and, for example, for impellers in the air inlets of the turbo charger groups. Today, many crank case- and cam shaft covers are also made of a light metal. The 'hot box' of Wärtsilä-engines, which maintains the high temperature found in the common rail fuel injection system is made entirely of light metals.



► The impeller of the air blower in a turbo blower is manufactured from an aluminium alloy, the so-called light metal.

7.8 Ceramic materials

Physically, ceramic has unique properties such as great hardness and high wear rates. It can consist of the following compounds:
Al_2O_3, ZrO_2, TiC, Si_3N_4 en $ALTiO_2$.
These are:
- oxide-based aluminium, silicon, zirconium, titanium, beryllium en magnesium;
- nitride-based silicon, boron, titanium en aluminium;
- carbide-based silicon, boron en titanium;
- composites.

It is easily applied as a thin coating on, for instance, piston rings.

▼ Light-metal pistons are generally used in high-speed diesel engines.

The main advantage is their light weight, granted the acceleration and deceleration forces of the piston remain within the maximum limits.

7.9 Specific materials for engine parts; engine classification according to the four categories

7.9.1 Engines from 0 to 100 kW, high-speed, four-stroke, fuel oil, M.D.O. – Category I: Small industrial diesel engines

The engine blocks are often constructed of cast light metal or cast iron. An added advantage of a light metal is, of course, its weight.
The **cylinder liners** or bushings of a light metal block need to be fitted separately, while a cast iron block does not require an independent cylinder liner. Moreover, a separate cylinder liner or 'bushing' is regularly applied to the somewhat larger engines of this category.

The pistons are often made of light metal and the entire cylinder head can be constructed from both light metal and cast iron.

The crankshaft of high load engines is often made of high alloyed forged steel, but for standard engines normal unalloyed carbon steel suffices. However, the crank pin and the shaft journals are especially hardened.

▼ A diesel-engine series for pleasure yachts with from front to back increasing power outputs.

All diesel engines are standard diesel engines and later modified for shipping. This is known as 'marinisation'.

7.9.2 Engines of 100 to 5000 kW, high-speed, four-stroke, fuel M.D.O. – Category II: Larger industrial diesel engines

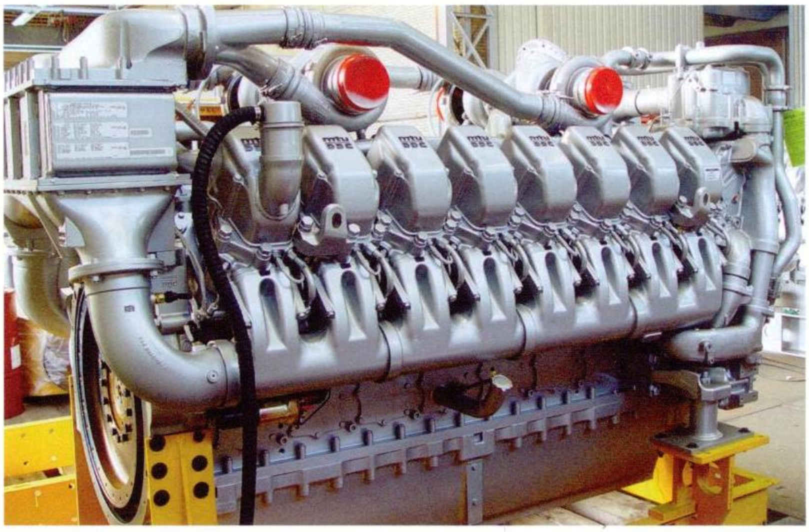
The greater the output/power the more often cast iron is used for the block. In this case the crankshaft is also made of forged iron, which, dependant on the load may or may not be alloyed. The **cylinder heads** are usually manufactured of cast iron. Pistons are often made of **light metal.** The piston mass is very important if there is a high number of revolutions, because of the acceleration- and deceleration forces working on the moving parts. The lighter the piston mass, the lower the acceleration- and deceleration forces. The piston of high load high-speed engines is often constructed from **two components**, the **cast iron piston crown because of** the high thermal and mechanical load, and the **light metal piston skirt enabling a** good piston conduction and absorption of the lateral forces exerted on the piston by the crank/connecting rod mechanism.

With a large power output cylinder liners are of cast iron and water cooled.

7.9.3 Engines of 500 to 30,000 kW, medium-speed, four-stroke, fuel H.F.O. – Category III: Large industrial diesel engines

Generally, these engines are highly powered and are both mechanically and thermally highly loaded. H.F.O. is also paramount in the decisions taken in, for instance, the choice of material for the exhaust valves.
The **engine block** is made of alloyed cast iron. This is often **nodular cast iron** compounded with magnesium creating a structure with a high tensile strength.

Cylinder liners are made of **special cast iron** with superior running properties and often have a built-up upper collar for the absorption of the high compression- and combustion pressures. The upper collar contains bore-cooling and protrudes from the engine block. This has the added advantage of reducing the size and weight of the engine block.



▲
A diesel engine with a large power output such as this sixteen cylinder MTU-engine which has a high thermal and mechanical load uses many high-alloyed parts, for example, the crankshaft, the cylinder block and the exhaust valves.

◄
A high-speed Caterpillar diesel engine for the propulsion of an inland navigation tanker.

▼
A block in an in-line engine rotated 180° for the mounting of the underslung crankshaft.

1 *engine block*
2 *crankpin*
3 *counterweight*
4 *lower bearing cap*
5 *bearing-cap bolts*
6 *hydraulic nuts*
7 *joint interface between bearing cap and block*

▶ A cylinder liner with a built-up top collar containing drilled-out cooling ducts.

The liner is conserved after repair and wrapped partially to prevent corrosion.

The crankshaft is of forged steel and as in all four-stroke engines underslung. The **piston consists** of a **cast iron crown** and a **cast iron skirt.** Tri-metal bearings are used for the crankshaft and the crank pin **bearings.** Today, this is the case in virtually all engines.

The cy**linder heads** are made of **cast iron** or **cast steel.** Especially if the **exhaust valves** are heavily loaded. Sodium and vanadium compounds in the fuel can create a sticky deposit on the valve seat which is a very aggressive corrosive. This is called high temperature corrosion, or abbreviated, H.T.C.. In contrast to low temperature corrosion, or abbreviated, L.T.C., caused by sulphur in the fuel found at much lower temperatures. This occurs, for instance, on the cylinder liners at extremely low engine loads.

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

Inlet valves are often made of high quality **carbon steel.** This certainly does not suffice for the exhaust valves and therefore high chromium steel with, for example, 8 to 12% chromium content and also silicon is used.

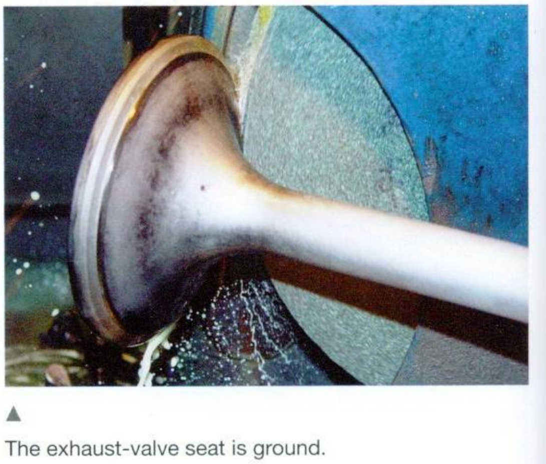
The **exhaust valve** is often rewelded with so-called armour steel, stellite. This is a very hard and wear-resistant alloy containing **chromium, tungsten, cobalt** and **carbon.**

Sometimes the entire valve or the valve disc and part of the valve stem are made of **'Nimonac 80A'**, an **alloy with 80% nickel, chromium, titanium** and **aluminium.** This material is costly, but gives a tremendous increase in the operating lifetime of the valve.

▼ Assembling of the forged-steel crankshaft must be done precisely and carefully.

The engine block is rotated 180° for the assembling of the crankshaft with bearings and bearing caps.

▲ Cylinder heads during maintenance at engine works Bolier, Dordrecht, The Netherlands.



▲ The exhaust-valve seat is ground.

To improve the resistance of exhaust valves against high-temperature corrosion, they are often completely or partially manufactured from a high-alloyed steel type, Nimonac.

7.9.4 Engines of 1500 to 100,000 kW, low-speed, two-stroke crosshead engines, fuel H.F.O. – Category IV: Extremely large industrial diesel engines

These are exceptionally large and heavy engines. Today, they are the only engines that are constructed from parts.

▲

The cylinder head of a crosshead engine with in the background the turbo blower.

◄

The welded A- frame of a crosshead engine under construction at MAN–B&W, Frederikshavn, Denmark.

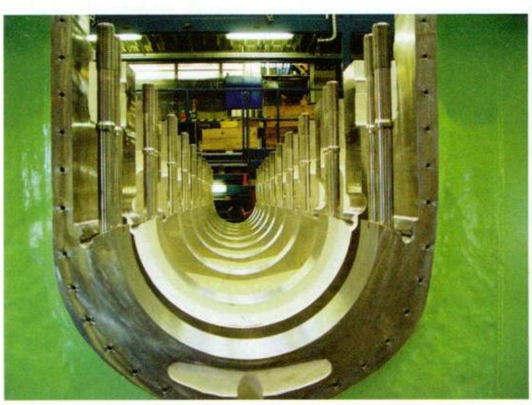
A large number of these engines are built from **welded steel plate constructions** and a smaller number from **castings.**

The crankshaft of these large engines rests on the engine bedplate, a welded construction with incorporated crossbeams in which the forged steel supports are integrated beneath the crankshaft bearings.

The A-shaped engine frame constructed from welded steel plates is mounted on the bedplate. The crosshead guides, used to convey the crosshead and absorb the shearing forces of the crosshead, are welded or bolted against the columns found in the A-frame.

The cylinder beam rests on the columns which in larger crosshead type engines consist of different parts bolted together.

Bedplates, columns and the **cylinder beam** are linked by very long vertical **alloyed steel** tension **bolts,** which run through the columns in ducts. The bolts in the ducts are fastened in some places with stay-bolts in order to prevent shearing vibrations of the tie rods. These vibrations could cause cracks in the long term. These are as in all other large linkages hydraulically pre-stressed.



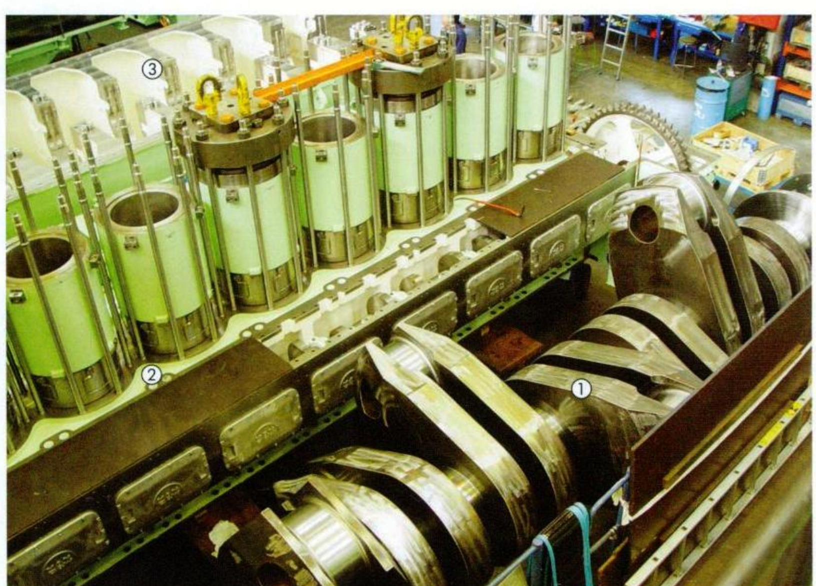
▲

The welded steel crankshaft bed of a crosshead engine; it is of utmost importance that the crankshaft is placed in line.

▼

Parts of crosshead engines under construction.

1 *forged crankshaft*
2 *cylinder block with cylinder liners and camshaft*
3 *crankshaft bedplate*



▲
The manufacture of a crosshead engine.

1 *camshaft*
2 *cylinder liners*
3 *cylinder head*
4 *exhaust-valve casing*
5 *cylinder-head bolts*
6 *exhaust-valve casing bolts*
7 *fuel injector*

▲▲
The scavenging-air space at the cylinder liner with the inlet ports.

▲
Forged steel connecting rods with mounted crankpin bearings and crosshead-pin bearings of a two-stroke crosshead engine.

The **cylinder beams** are made of cast iron.
The bottom part of the cast iron **cylinder liners** is situated in the cylinder beam. The **forged steel cylinder heads** are fixed to the cylinder beam by means of long bolts.

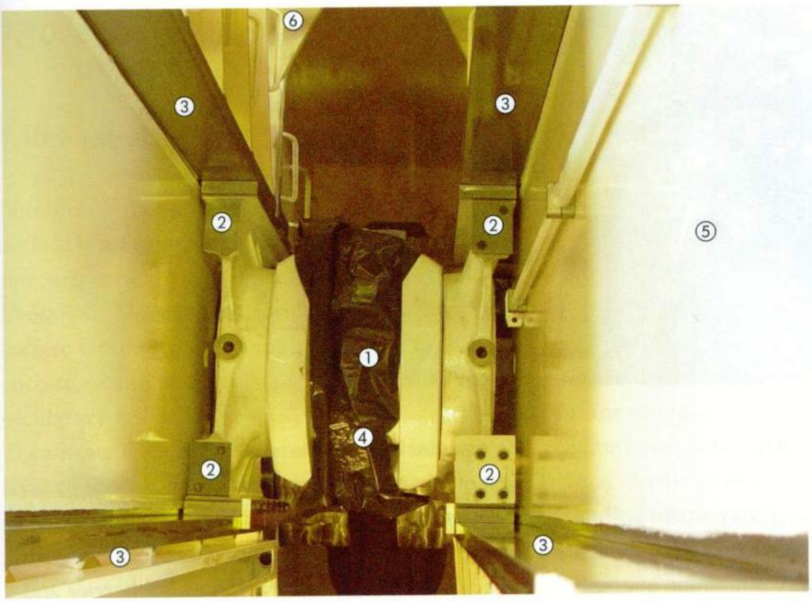
The cooling of the cylinder liner and cylinder heads occurs through drilled canals, the so-called 'bore-cooling'.

The pistons are made of forged steel or cast steel in view of their high thermal and mechanical loads.

The piston rod, connecting rod and the crankshaft are made of top quality malleable steel; special attention is paid to the crankshaft.
For crosshead engines unalloyed carbon steel usually provides sufficient strength, which is in contrast to engines with a higher load, such as medium- and high-speed engines.

▶
The forged crankshaft; manufactured in sections, and fixed together using shrinkage joints.

1 *crankshaft*
2 *crankweb*
3 *crankpin*
4 *weight economy*
5 *roundings*



The mounted crosshead, seen from above.

1 *crosshead*
2 *guide shoes – four pieces*
3 *crosshead guides – four pieces*
4 *place for the fixed piston rod; not mounted here*
5 *A-frame*
6 *step*

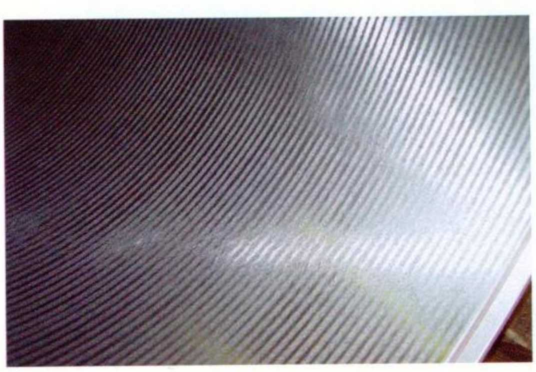
Lubrication here requires special attention. Unlike four-stroke engines, two-stroke engines have no 'rest' revolution. Therefore the crosshead bearings are always subjected to great forces from the piston. Without a lubricating oil film, wear and tear of the bearings increases.
One of the measures taken to prevent premature wear and tear at the boundary lubrication conditions is a surface treatment of the steel crosshead pin.
Hard chromium plating, a grind or super finishes are often applied.

Also see Chapter 13, Driving gears.

Piston rings

These usually consist of **fine grained perlitic cast iron** with **flake graphite** which has excellent running properties in the cast iron cylinder liner. The piston ring hardness lies between the 2000 and 2400 N/mm², hardness Brinell.

The finished surface of a cylinder liner. Shown here is a special finishing that deviates from the normal honing cross-hatching, the 'ribbed' surface of diesel engines of MAN–B&W.



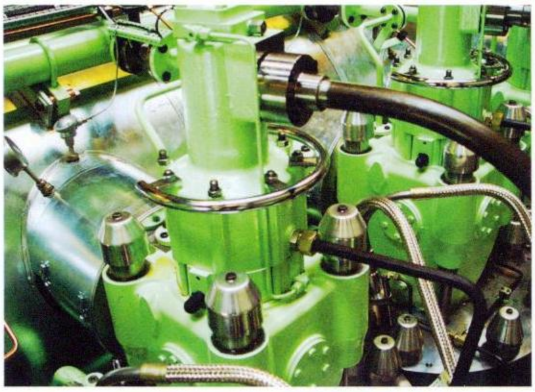
The **cylinder liners** are also made of **perlitic cast iron** with flake graphite and therefore also have good running properties. Often alloys are added in order to improve wear and tear resistance. The **hardness** is considerably lower than that of the piston rings, namely 800 N/mm² hardness Brinell.

Exhaust valve

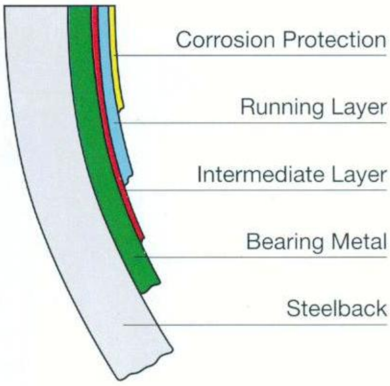
All modern crosshead engines are equipped with one centrally placed hydraulically controlled exhaust valve. In large engines the valve diameter is approximately 40 centimetres.
All material consists of a high alloy type of steel usually covered with a rewelded stellite layer. Alloys such as **'Nimonac 80A'** are also often used.

A series of exhaust valves manufactured from Nimonac, a high-grade material.

The total life of these valves given by the factory is up to 96,000 hours!



▲
The cast iron exhaust-valve casing.



▲
The manufacture of a tri-metal bearing.
Most of the layers are very thin.

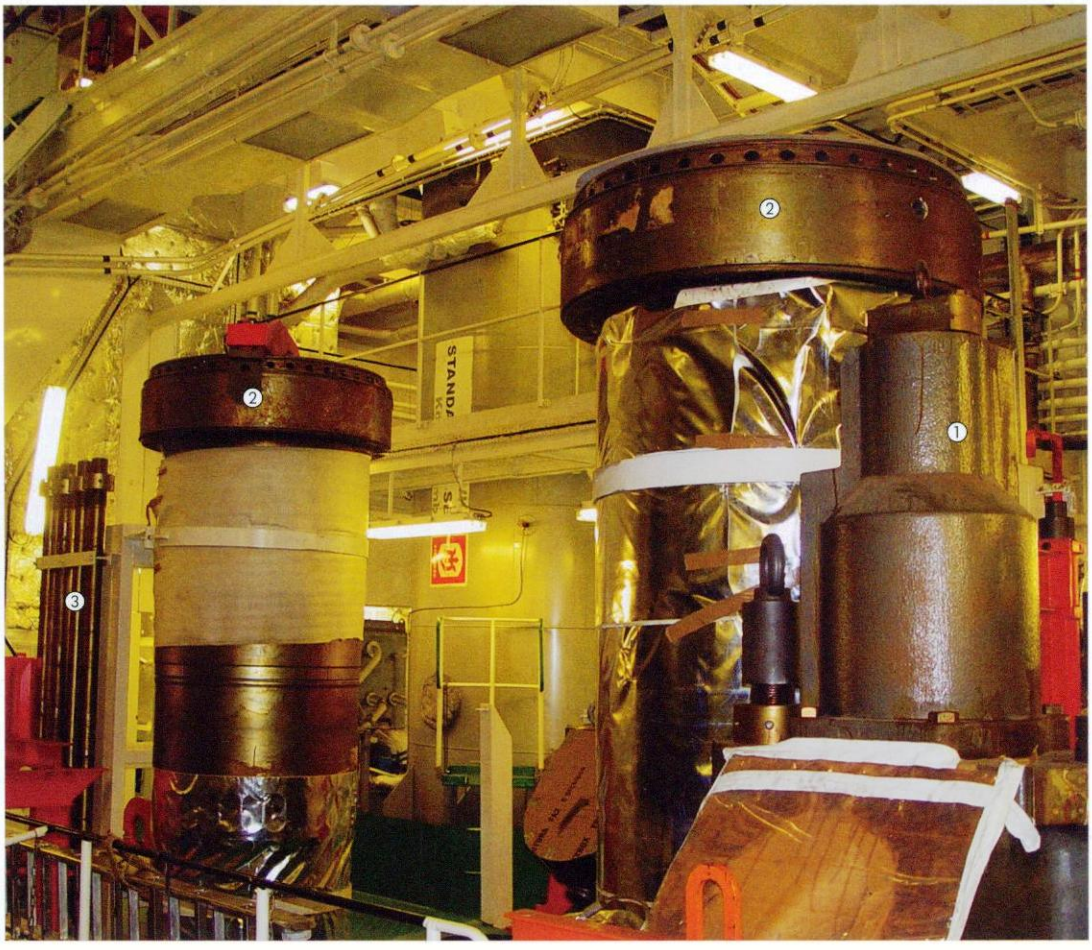
The exhaust valve housing is made of high quality cast iron and is today the only remaining engine component with cooling water ducting in the cast. The remainder of the cooled components such as the cylinder liner, piston and the cylinder head are of course cooled using 'bore-cooling' technology.

Bearing materials

Due to the heavy load exerted on the bearings today one often uses tri-metal types. These can bear high loads and do not require being custom manufactured unlike the previously used 'soft' white-metal bearings.

They generally consist of a steel bearing cap with a lead-bronze layer, a very thin nickel barrier and the actual sliding material, a 'white metal'. 'The white metal often comprises of mainly aluminium and tin in an 80% aluminium and 20% tin ratio. This material has excellent properties, such as high fatigue strength and good running properties. High loading can be achieved with an **aluminium alloy where either a zinc or silicon component is added.**

▶
A complete exhaust valve (1) right and two spare cylinder liners (2). Far left cylinder-head bolts (3).



7.10 Special finishes and heat treatments

7.10.1 Nitration

Nitration is thermal treatment in which the surface layer of virtually all iron parts can be nitrogen enriched at temperature ranges of 500 °C to 600 °C. The nitrogen enrichment increases the surface tensile strength thus achieving a higher corrosion- and wear resistance as well as a higher oscillatory resistance.
The nitrate layer consists of a very thin outer layer of some thousandths of a millimetre (micro millimetre) with a hardness of about 800 to 1200 N/mm^2, hardness Brinell.

Application

It is often applied when overhauling cylinder liners and crankshafts.

7.10.2 Annealing

This occurs in temperature ranges of 450 °C and 650 °C. It reduces the internal stresses in the constructions caused by the electric welding processes.
After a certain 'glow' time the work piece or the construction needs to be cooled slowly in order to prevent development of new stresses.

Application

Welded steel constructions in engines such as bedplates, A-frames and air inlet manifolds. Overhauled engine parts such as welded cylinder covers, pistons and exhaust valves.

7.10.3 Surface hardening

During this process the parts are only heated and hardened at the surface, thus causing only minor shape- and size changes. Heating takes place by means of induction or with a gas jet, the so-called 'flame hardening'. The latter is sometimes applied when overhauling small crankshafts and after the grinding of the valves to a smaller size, the ground surfaces must be hardened.
There are many other surface treatments. Surface hardening causes tensile stress on the surface, which subsequently enables a higher load.

Application

Crankshafts, piston pins and crosshead pins. Steel crankshafts of a 0.5 to 0.6% carbon content are heated to approximately 720 °C and then subsequently water cooled. The core of the crankshaft remains fairly soft, but on the outside a hard layer of approximately ± 0.9 to 2.0 mm is formed.

7.10.4 Applying surface layers

Some engine components' surfaces run either continuously or discontinuously across another surface, such as:

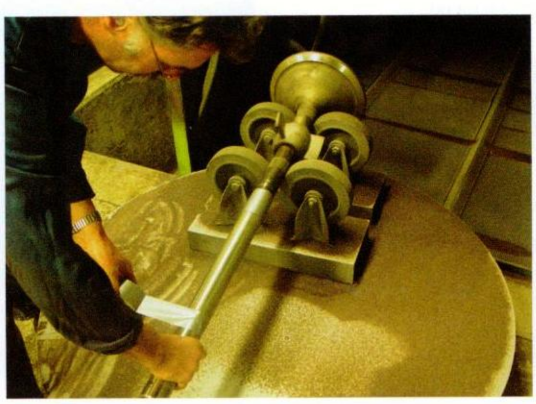
1. piston rings on the cylinder liner;
2. the stuffing box on the connecting rod of two-crosshead engines;
3. the bearings of crankshafts, cam shafts and piston pins.

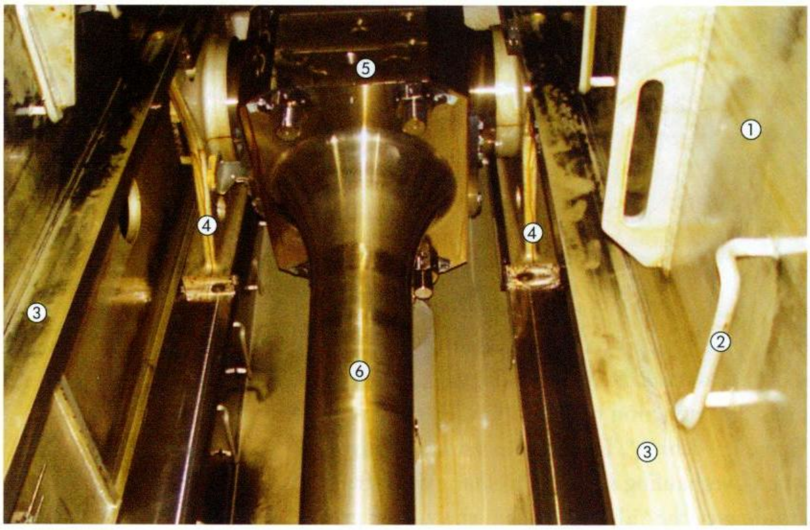
◄
The honing pattern on the running surface of a cylinder liner.

◄◄
A cylinder cover in an electric oven for annealing.

The temperature and the timing of this treatment are very important.

▼
Shown is a very hard, wear-resistant layer sprayed on the valve stem, followed by very fine sandpapering.





▲

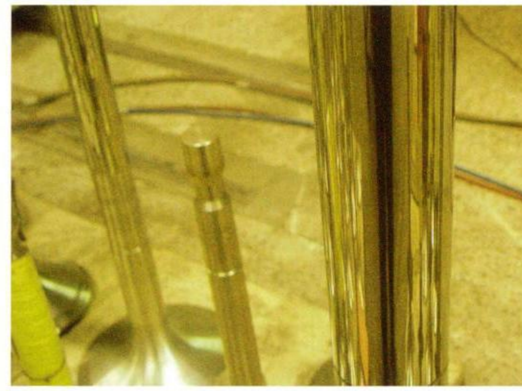
Forged steel connecting rods with above the crosshead and guide shoes, left and right are the crosshead guides.

1 *A-frame*
2 *step*
3 *crosshead guides*
4 *guide shoes*
5 *crosshead*
6 *connecting rod*

Chromium

In order to make the surfaces harder and more wear resistant they are often furnished with a thin chromium coating which increases the operating life of the components considerably. There are, for instance, chromium plated piston rings, connecting rods and crosshead pins.
Chromium has also good resistance properties with respect to the corrosive substances in the combustion products found on the piston rings. Furthermore, chromium has an extremely low friction coefficient on the generally cast iron cylinder liners.

Hard chromium plating or chromium hardening is the electrolytic plating of chromium on the components. The so-called plating is done with a chromium- salt solution.



▲

A chromed valve stem that is very wear-resistant and resistant to chemical corrosion.

Plasma layer

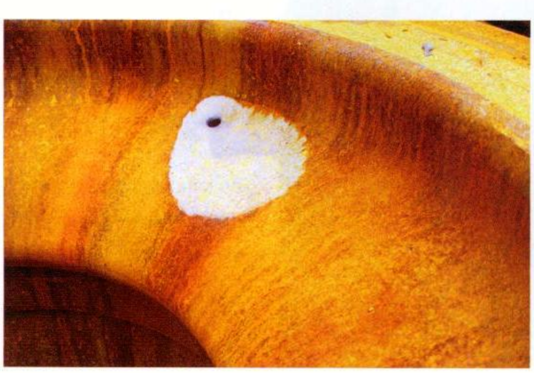
A ceramic coating is applied to chromium plated parts such as, the top piston rings. Ceramic has very good wear resistance properties under heavy operating conditions.
Both the running in of a piston in the cylinder liner, as well as the great many operating hours that follow, show that the wear and tear of the cylinder liner and the rings is much lower.

The rewelding of special material that has good corrosion- and erosion resistance; so-called component cladding. This is mainly applied in combustion areas with excessively high temperatures such as those found in RTA – C Wärtsilä Sulzer crosshead engines. Here the material is attached to the inner wall of the cylinder head, in the direction of the fuel jet. In the latest designs the material temperature of the cylinder head has be reduced to such an extent that this surface treatment method is no longer required.

▼

This cylinder head is waiting outside for repair.

The high-grade material is clearly seen near the injector nozzle; shown here a very strong welded layer of material that is resistant to the high temperature at the injector nozzle.



►

Reconditioning the injector nozzle by rewelding.

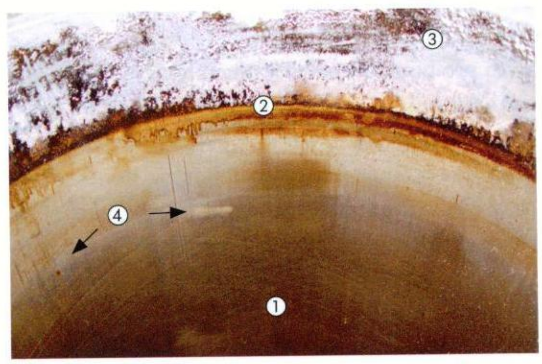
In a later phase, the liner is machined resulting in the original dimensions and the injector nozzle is drilled out and finished.

The honing of a cylinder liner.

1 *honing machine*
2 *cylinder liner*
3 *honing-stone holder*
4 *honing stone*
5 *driving shaft*
6 *oil supply*
7 *clamping strips*

7.10.5 Grinding, polishing and lapping

Grinding, polishing and **lapping** are standard expressions used in the finishing of metal surfaces. They are general terms which give an impression of the surface finishes, however, they do not provide a surface roughness value.
A **ground** and polished crankshaft journal can have a very smooth surface.
Polishing is usually a finish which removes the final roughness.
Lapping involves the use of two surfaces which are rubbed together such as seal surfaces of the different parts of a fuel atomiser.



A worn cylinder liner after thousands of operating hours.

Stop shoulder for the upper piston ring is clearly seen.

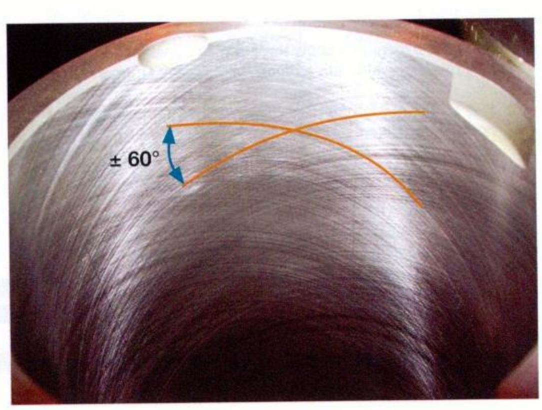
1 *worn part of the cylinder liner with still visible honing cross-hatching*
2 *stop shoulder*
3 *non-worn part of the cylinder liner with incrustation*
4 *vertical scratches in cylinder liner caused by damage, dirt or metal particles (catfines)*

7.10.6 Honing

This surface finish is applied to the running surface of cylinder liners in various engines. The objective of honing is to improve the running surface for the reciprocating piston. The honing cross hatch pattern is used to retain a certain amount of lubricating oil which reduces the wear and tear of piston rings, piston and liner.
Honing is done with a pivot shaft head on which four honing stones are held under pressure against the cylinder liner using springs. With a specified pressure, peripheral speed and longitudinal movement of the liner a rough kind of cross hatch pattern is applied to the liner in which the scratches are at an angle of about 60 °. There are a great many types of honing stones with a specified roughness. The honing oil that is used is also important. Honing stones, pressure, speed and the process length are crucial in order to achieve optimum results.

Honing.

The use of the correct honing stones, turning tension on the stones, the number of revolutions and the horizontal movement of the stones through the liner are important points of interest. The choice of honing oil also plays a role.



With a good honing cross-hatching, the scratches have an angle of roughly 50 tot 60 °.

A honing machine with a rotating and a forwards and backwards movement, both are adjustable.

Plateau honing

This type of processing is used by some engine manufacturers and reconditioning companies. It takes place in two steps, namely the pre-honing and fine-honing.

With pre-honing, coarse grained honing stones are used resulting in a surface roughness of 8 to 10 microns.

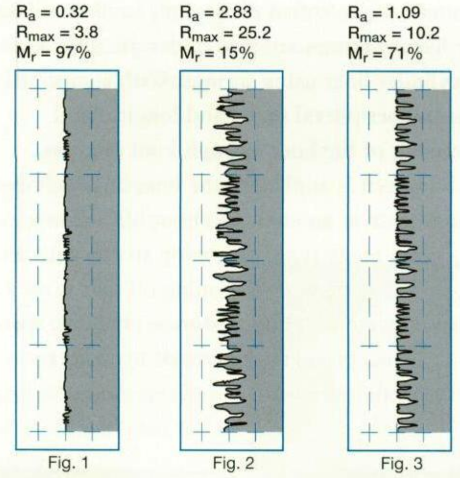
With **fine-honing,** fine grained honing stones are used where the roughness 'peaks' are ground off. Between the deeper scratches resulting from the pre-honing, peaks are obtained with finer structured profiles.

Using this method, the bearing capacity of a thus processed cylinder liner achieves the same level as conventionally honed cylinder liners after very few operating hours.

Plateau-honing.

Figure 1: Shows an almost smooth worn cylinder liner. This is known as 'glazing'. Figure 2: the surface profile after a standard honing process. The surface is still quite rough with deep grooves and high peaks. Figure 3: the surface profile after plateau-honing. Very fine honing stones remove the high peaks resulting in a flat running surface being created for the piston rings.

Surface Condition



The above diagrams show three profile curves taken from cylinder liner walls with surface roughness instrument.

Fig. 1 - Prior to honing: The diagram shows an almost completely glazed surface.

Fig. 2 - Basic honing: The diagram shows a rough surface with deep grooves and high peaks. Here, work has been done to remove ovality, scuffing marks, etc.

Fig. 3 - Plateau honing: The cylinder liner wall has been given it's final honing. The high peaks have been honed down using fine stones, and the resulting smooth surface provides a good running surface for the piston rings.

The cross angle pattern and the depth and width of the remaining grooves achieved by honing are important factors as they act as lube oil reservoirs.

Simply put: Plateau-honing reduces the running in time. This process also reduces the lubricating oil consumption without having a negative effect on the lubrication.

7.11 Examples of modern material usage

Competition in the engine building industry is fierce. Customers' requirements are always increasing, therefore engine designers (Research and Design) are continually looking for materials that have better resistance to the thermal and mechanical loads and the effects of aggressive compounds.

7.11.1 Example 1 – Design requirements for Wärtsilä Sulzer RTA - C crosshead engines, fuel H.F.O. - Engine category IV

Construction

- stiff frame.
- Gondola-type engine bedplate.
- cast iron cylinder blocks.

Bearings

- lower crankshaft bearing: thin white metal bearings.
- upper crankshaft bearing: thick in cast white metal bearings.
- crosshead bearings: thin white metal bearings.

Combustion chambers

- special forged steel with high resistance to corrosion fatigue.
- rewelding cylinder cover with heat resistant material at the fuel nozzle tips to prevent burning.

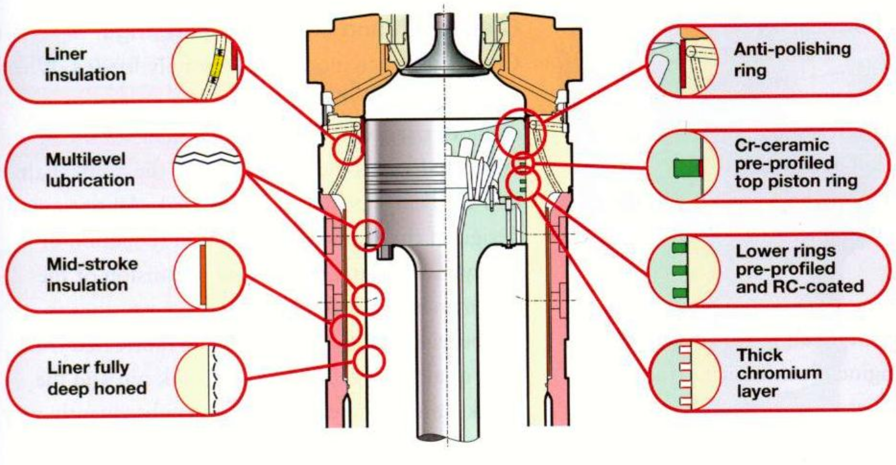
Piston

- forged steel piston crown with chromium plated piston ring grooves.

Liner

- cylinder liner of high wear and tear resistant rigid cast iron.
- honed cylinder liner.
- piston rings with a plasma coating.

Furthermore, all measures taken to improve environmental conditions during combustion summed up in the so-called **'Tribo Pack Concept'.**

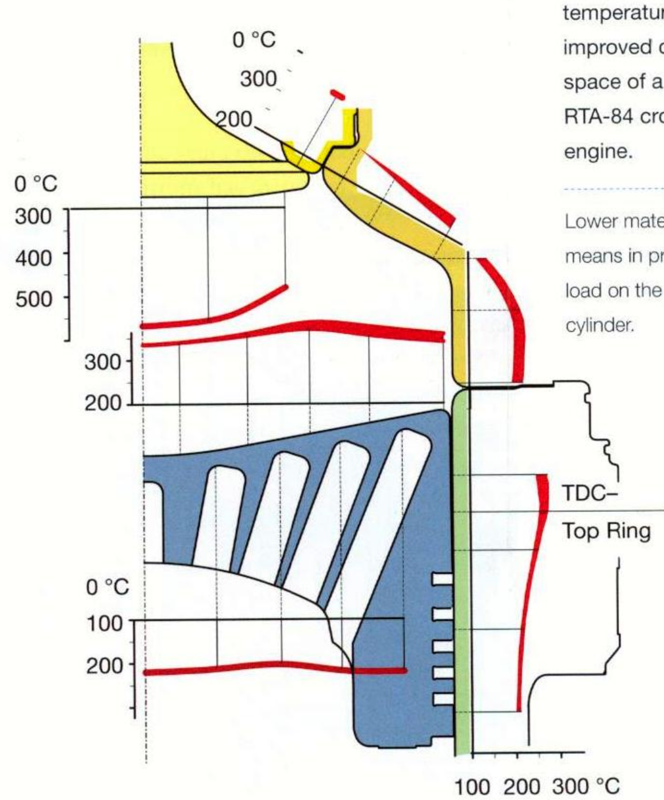


◄ Eight improvements in the combustion space of a Wärtsilä Sulzer RTA-84 crosshead engine.

This entails the following:
- isolated 'bore'-cooling holes' throughout the combustion space in piston, cylinder liner and cylinder head;
- multilevel cylinder lubrication;
- mid stroke cylinder liner insulation;
- a 'deeply' honed liner;
- an anti-polishing ring which prevents the combustion residue from settling in the ring packing;
- a chromium and ceramic treated pre-profiled top piston ring;
- applying a thin run-in layer to the other piston rings;
- thick chromium coated piston ring grooves to prevent pitting and wear and tear.

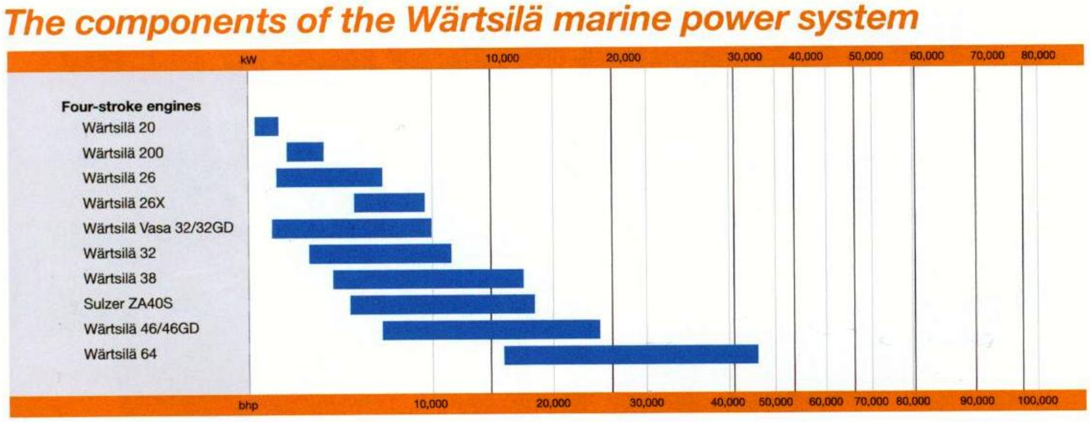
7.11.2 Example 2 – Wärtsilä-four-stroke engines, medium-speed, fuel H.F.O. – Engine category III

Wärtsilä has a programme with cylinder diameters of 20, 26, 32, 38, 46 and 64 centimetres for this type of engine. The latter is also the largest four-stroke engine presently available on the market, with a cylinder capacity of 2000 kW.



▼ The material temperatures in an improved combustion space of a Wärtsilä Sulzer RTA-84 crosshead engine.

Lower material temperatures means in principle a lower load on the parts around the cylinder.



◄ The different cylinder diameters of four-stroke diesel engines at Wärtsilä.

To designers the following elements are essential:
- reliability;
- low user costs;
- low exhaust gas emissions;
- simple installation at low cost;
- applied mounting methods need to be reliable;
- low, simple maintenance.

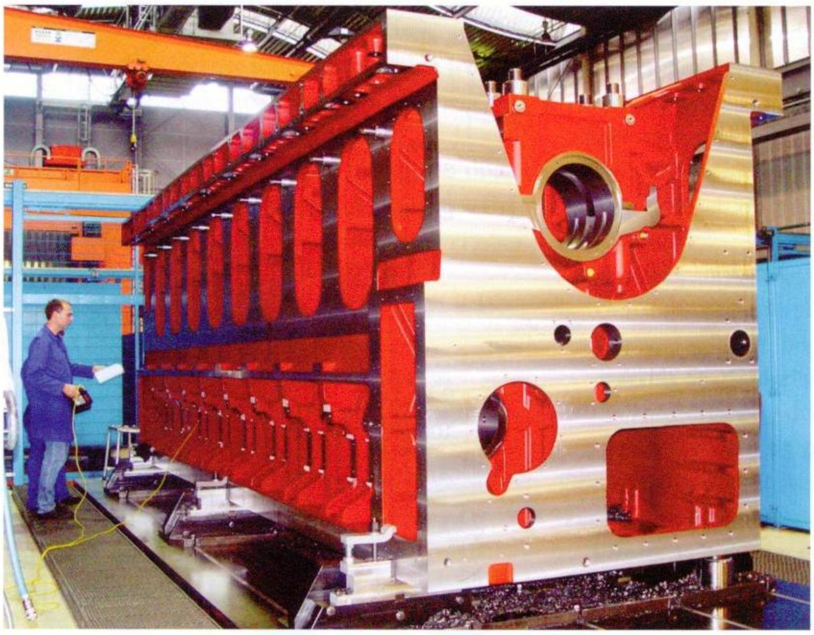
Engine block

All lubricating oil- and cooling water ducting is integrated in the engine block which results in compact and safe operating systems. The cam shafts have also been fully incorporated in the block which creates a very solid entity. The engine block is made from special nodular cast iron, which is very strong and rigid and especially suitable for casting complicated forms.

Crankshaft and crankshaft bearings

Crankshafts are increasingly heavily loaded. When increasing the cylinder capacity the pressure in the cylinder rises, which results in higher forces on the crankshaft. This means the size of the crankshaft and the bearings must be increased. An optimum design of the crankshaft such as a reduction of the cylinder centre radius and expansion of the bearing surface reduces the surface pressure in the bearings. The crankshafts are fabricated from high quality forged steel. The crank pin and the crankshaft are casehardened and subsequently very finely ground.

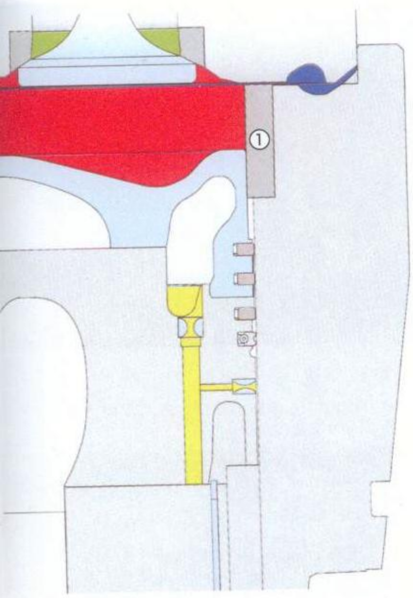
The crankshaft- and crank pin bearings are manufactured from a tri-metal type using an aluminium/tin- or antimonium/tin basis and work according to the so-called 'Thick-Pad Bearing' principle.



▶ A cast iron engine block of an in-line engine.

▶ A crankshaft of a V-engine.

Note: the two lubricating-oil holes in the crankpin for both connecting rods. Crankshaft reconditioning at Mark van Schaick, Schiedam, The Netherlands.



◄◄
Cross-section of a cylinder liner with bore-cooling in the built-up upper stroke and an anti-polishing ring (1).

◄
A cylinder liner with built-up upper stroke and anti-polishing ring visible.

The cylinder liner is made of fine perlitic cast iron with high wear resistance and excellent running properties.
The thick high-collar type cylinder liner is very stiff and therefore distortion is minimal. The collar has bore-cooling for an optimum liner temperature. The anti-polishing ring stops carbon particles from finishing up between the piston and cylinder liner which diminishes the wear and tear of these parts considerably.

Cylinder head

All modern cylinder heads have four valves: two inlet valves and two exhaust valves. This allows for the gas exchange to take place swiftly and completely. The cylinder head is usually affixed with four cylinder head bolts.
The head is manufactured from special cast steel, is very stiff and heat resistant.
The supply of combustion air, the discharge of exhaust gasses and the supply and discharge of cooling water takes place via nodular cast iron 'multi-ducts' which remain in one place when the cylinder head is dismantled.

Pistons

Throughout the years the thermal and mechanical load on the pistons has increased. Therefore all pistons consist of two parts:

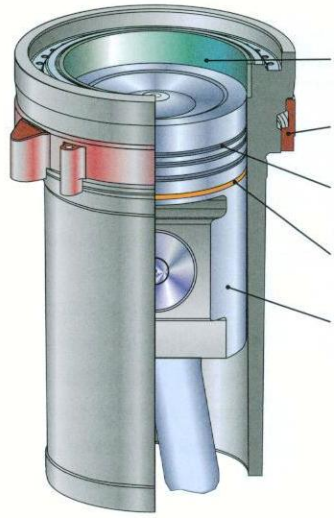
1 **a cast steel piston crown** which contains the ring packing. This has good resistance against the high thermal and mechanical loads. This material has good resistance against pitting of the piston rings in the piston ring grooves.
2 **a nodular cast iron piston skirt for surveying the piston and absorbing the lateral forces caused by the connecting rod mechanism.** The nodular cast iron has a very low expansion coefficient and retains sufficient tensile strength at high operating temperatures.

◄
A piston with a cast steel piston crown and cast iron piston skirt.

Note the gaps for the valves in the piston crown for exhaust-air scavenging.

◄◄
A modern cylinder head with four valves.

▶
The piston rings as shown here are chromed and the top ring has an extra ceramic layer.



▲
A modern diesel engine, manufacturer Caterpillar.

▲
Two diesel generator sets on board a container-feeder with a Scania diesel engines.

Piston rings

A three piston rings design has been made for the sealing package. These are made of fine perlitic cast iron where both sealing rings are chromium plated.
On top of that the upper ring has an extra plasma coating and has high wear and tear resistance.

7.11.3 Example 3 – Four-stroke high speed engines with high capacity at a relatively high RPM, fuel M.D.O. – Engine category II

Marine engines of this type are always subdivided into two types:

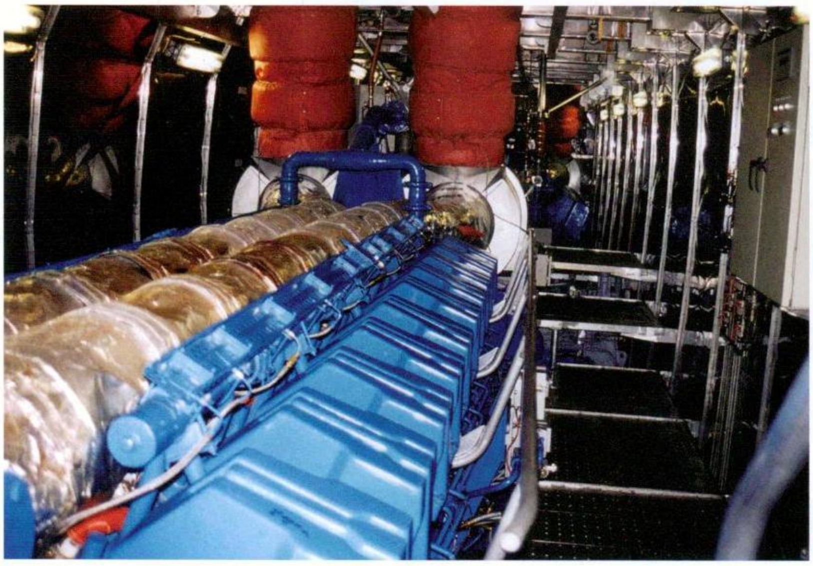
1 **High output,** often used in the military services. Additional characteristics are: complex designs, few operating hours between big overhauls, costly components, extremely high effective mean pressures and a low weight in relation to the output.

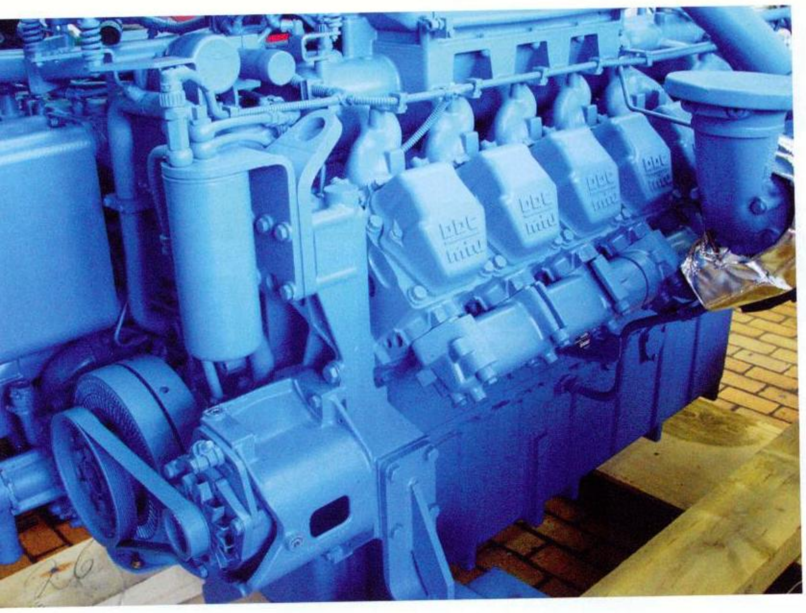
2 **Engines for heavy operating conditions** such as: earth moving equipment, gensets and other gear driven devices. Characteristics are simple and the design is robust with normal mean effective pressures and a fairly high weight in relation to the output.

The demand for the somewhat smaller high speed diesel engines for propulsion and other gear driving has resulted in designs that can be easily integrated in high-volume serial production lines. The following manufacturers are market leaders in the manufacturing of engines with a cylinder bore of about 170 mm:
Caterpillar, Cummins, Deutz MWM, GMT, Isotta–Franchini, MAN–B&W Holeby, Perkins, Mitsubishi, MTU, MTU/DDC, Niigata, Paxman, SEMT–Pielstick, Wärtsilä and others. All manufacturers produce high-speed engine types.

▼
A high speed heavy loaded twenty cylinder V-engine for propelling a fast catamaran.

The engine rooms have been placed in both pontoons. The fuel is diesel oil. Power output and weight play an important role.
The operating speed of this catamaran is 37,5 knots.





◄
A modern high-speed four-stroke diesel engine in V- form (manufacturer MTU) ready to be placed in an inland navigation tanker.

These fairly large ships with a load capacity from 2000 to 4000 tons usually have two propulsion engines. In view of the strict emission requirements all inland vessels run on diesel oil.

Characteristics

Blocks of cast iron, with integrated cooling water- and lubricating oil ducts. Nodular cast iron is often applied.

- A underslung **alloyed forged iron crankshaft with tri-metal bearings.**
- **Cast fine perlitic cylinder liners.**
- **Light metal cast pieces** for pistons, covers, inlet manifold and other engine parts. This creates a considerable weight reduction.
- **Weight** in general plays a role in these engines in that they are often used in relatively light ships, such as catamarans, high-speed single hull ships and mobile gensets.
- **Many V-engines of this type are built; therefore the size is limited in relation to the power output.**
- **Pistons** often consist of two parts, a cast steel piston crown and a light metal piston skirt.
- **Cylinder heads** are made from cast iron and contain four valves.

Further evident technical details in this type of engine are

- high mean effective pressures > 25 bar;
- high RPM up to 2100 per minute;
- high mean piston speeds up to12 m/sec.;
- high power output per litre stroke volume;
- combustion pressures up to 200 bar;
- power output/weight ratio high;
- state of the art fuel injection systems;
- high air charging with turbo charger groups up to 4 bars;
- optimized coolant temperature at varying loads;
- improved lubricating oil cooling for the piston.

▼
A Caterpillar high-speed four-stroke diesel engine with twelve cylinders in V-form.

1 *air filter*
2 *turbo blower (not visible, behind filter)*
3 *compressed air to intercooler*
4 *inter cooler*
5 *cylinder head*
6 *control console*
7 *lubricating-oil filter*
8 *lubricating-oil cooler*

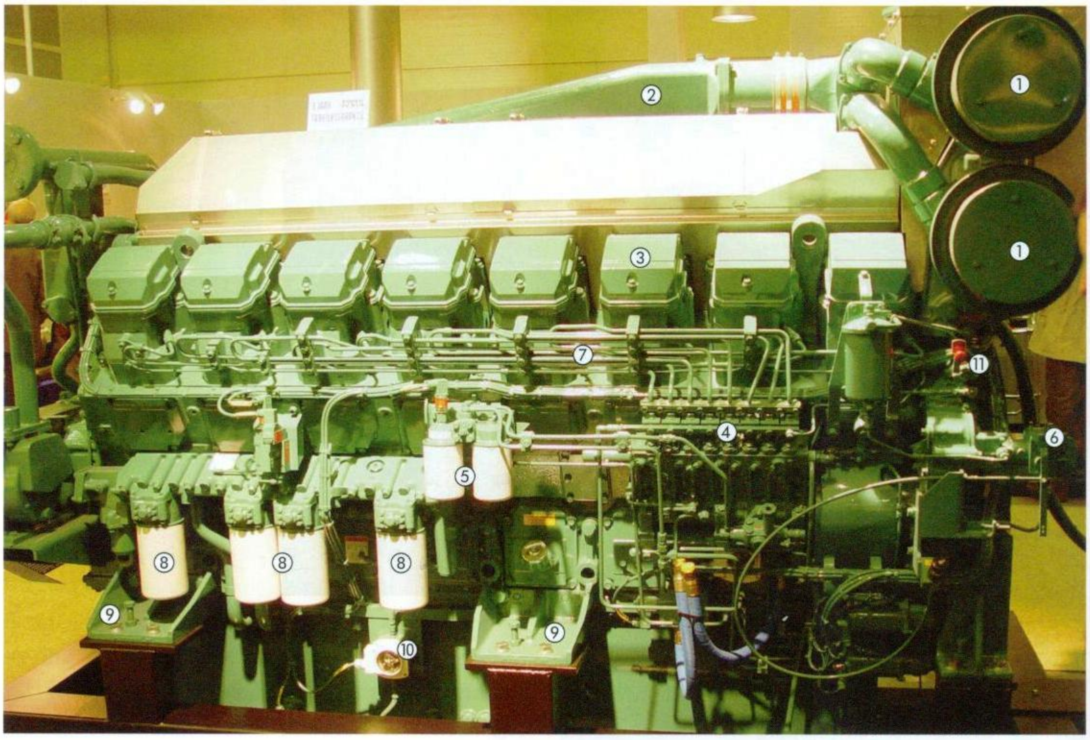
▲
A Deutz propulsion engine for a small dredging-craft.

In the foreground the gear box for the propeller.

▶

A sixteen cylinder high-speed V-engine, make Mitsubishi.

1 *turbo blower*
2 *intercooler*
3 *cylinder head*
4 *block fuel pump*
5 *fuel filters*
6 *speed controller/ governor*
7 *high-pressure fuel lines*
8 *lubricating-oil filters*
9 *engine supports*
10 *safety switch for lubricating-oil level in sump*
11 *emergency stop*



Category	Mahle 105	Mahle 109	Mahle 120	Mahle 121	Mahle 147	Mahle 226
Composition (%)						
Si	4,5 – 6,0	9,0 – 10,0	10,3 – 13,5	10,3 – 13,5	16,0 – 18,0	7,5 – 9,5
Mg	0,5 – 0,8	0,2 – 0,4	0,06 – 0,12	0,1 – 0,3	0,4 – 0,7	≤ 0,3
Cu	≤ 0,03	≤ 0,01	≤ 0,1	± 0,1	4,0 – 5,0	2,5 – 3,5
Fe	≤ 0,4	≤ 0,4	≤ 0,25	≤ 0,25	≤ 0,7	≤ 1,3
Mn	≤ 0,5	≤ 0,4				≤ 0,5
Zn						≤ 0,7
Al	rest	rest	rest	rest	rest	rest
Tensile strength Rm (N/mm^2)	≤ 260	≤ 260	≤ 140	≤ 145	≤ 200	240 – 310
Yield strength Ro 0,2 (N/mm^2)	≤ 240	≤ 200	≤ 67	≤ 80	≤ 160	160 – 240
Stretching at breaking load A (%)	≤ 1	≤ 4	≤ 5	≤ 3,5	≤ 0,5	0,5 – 3
Brinell hardness (HB 2,5/62,5)	≤ 90	≤ 80	48 – 60	50 – 70	90 – 120	80 – 110
Properties and applications	Rust proof; Alfin-cooling fin cylinder	Rust proof; cylinder block of water cooled ship engines	Ductile, rust proof, not heat cured; motor car wheel rims	Ductile, rust proof, not heat cured	high wear and tear resistance; cylinder without motor car wheel rims	Die cast alloy for cooling fin cylinder running surface protection SILUMAL- cylinder

▲

A table for the light-metal blocks for diesel engines. The main component is aluminium.

The table shows an aluminium cast alloy for cylinder blocks of water-cooled marine engines, Mahle 109.
9 to 10% silicon and 0.2 to 0.4% magnesium have been added.
The table also provides the aluminium alloy Mahle 147, which is applied to unprotected bores in cylinder blocks without separate cylinder liners.
The material is highly wear-resistant due to the alloy:
silicon: 16 to 18%
magnesium: 0.4 to 0.7%
copper: 4.0 to 5.0%

7.11.4 Example 4 – Smaller engines actually in engine category I: 0 tot 100 kW but also higher in capacity, high speed, four-stroke fuel M.D.O.

These engines are used for a large number of applications, such as the propulsion of smaller ships, gensets, pumps propulsion and traction. A noticeable difference is that, generally, many light metals are used. The main metal is aluminium with added alloys.
Properties of light metal
- low crystal density
- good heat conduction
- good corrosive properties
- many machining capabilities and especially low weight

Cylinder heads can also be made of light metal, but of course cast iron is also often used.
Generally, crankshafts are made of malleable steel, although cast crankshafts still exist, especially in older type engines.

Steel is often alloyed with **chromium** and **molybdenum.**
The journals are often hardened by means of nitration. Flame hardening is also applied.

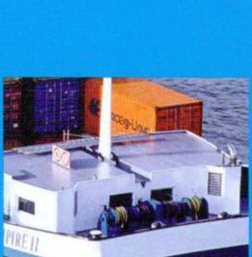
Also see Chapter 26, Overhauling diesel engines and their parts.

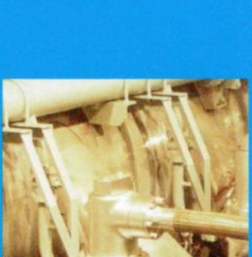
Timing belts, (also known as Toothed, Notch or Cog belts)
These are increasingly used in this engine category. They require no heavy cog wheels between the crankshaft and the cam shaft. However, they must to be replaced after a certain operating time.

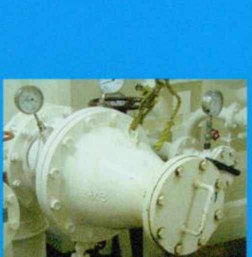
Bearings
Usually hard steel tri-metal sleeve bearings are applied. The friction metal most frequently used is an aluminium alloy.

>CH 8

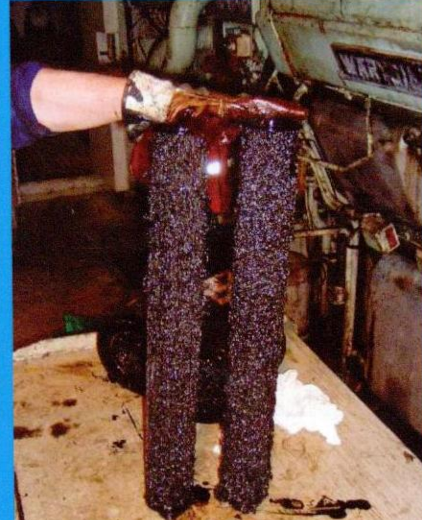
Fuels, fuel-line systems and fuel cleaning







The fuel quality injected into the combustion space is essential for an optimal running of the diesel engine.
Cleaning the fuel, here performed by Alfa Laval centrifuges for H.F.O. on a large container ship is an important aspect of fuel treatment prior to injection.

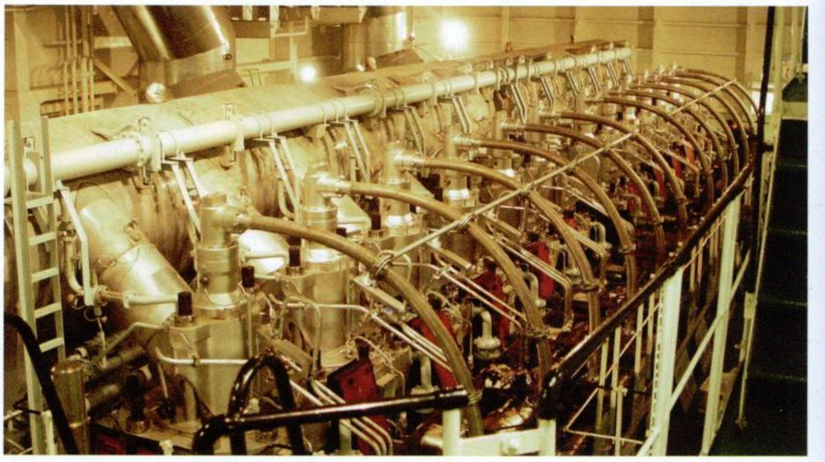


Poor quality H.F.O., these filters are completely clogged.

▸

Diesel engines have been powered by liquid carbon compounds distilled from crude oil for over a hundred years.

The refined crude oil can be used as gas oil, diesel oil and heavy oil for the four diesel engine categories. Since the year 2000, other fuels such as natural gas and bio gas are also successfully used. Bio fuels are becoming increasingly frequently used, especially in smaller diesel engines in the road transport and agriculture sectors. Today a few medium-speed engines are equipped to run on bio diesel.



8.1 Introduction

Since the first internal combustion engine was built according to the Otto-principle around 1860 and according to the Diesel-principle in 1896, to present day, 2008, mixtures of gaseous and liquid hydrocarbons are generally used.
The gaseous hydrocarbons used in the Otto-engine were predominantly a mixture of mining gases, process gases and land fill gases. Much later, after gas- and oil supplies were discovered deep in the earth's crust, mostly natural gas and oil were used. Today, natural gas resources exploration continues with gas fields being discovered and exploited. Otto-engines are also used frequently, for instance, in motor cars with liquid fuels such as petrol, a very light liquid hydrocarbon compound.
Of the gaseous hydrocarbons natural gas is generally used for stationary gas engines. However over the past five years, other gases such as land fill gas, sewer gas, bio-gas and other waste gases are becoming increasingly important due to energy saving and environmental issues.

Also see Chapter 29, Recent developments in the fuel industry.

The comparatively light liquid hydrocarbon compound diesel oil has been the most frequently used fuel since the introduction of the diesel engine. Around 1950 heavier fuels, initially used only in steam boilers, were used as a fuel in low-speed crosshead engines with a RPM of 80 to 120 revolutions per minute. As the fuel costs soared this so-called heavy oil was used more regularly.

From 1985 onwards, heavy oil has been increasingly used for four-stroke diesel engines with a RPM of about 1200 revolutions per minute. Above these RPM's, the combustion time is too short to achieve complete combustion of heavy oil. The price of gas oil is presently twice that of heavy oil.
Today, heavy oil is becoming increasingly a residual product in oil refineries with a corresponding loss in quality.

▾

Diesel oil requirements according to ISO 8217/1996.2005.

There is a subdivision consisting of four categories: DMX, DMA, DMB and DMC.

ISO 8217 standard; 2005				MGO	MDO	
International Standard for Marine Distillate Fuels	ISO Grade	2005	DMX	DMA	DMB	DMC
Characteristics	**Units**	**Limit**				
Density at 15 °C	kg/m^2	max	–	890.0	900.0	920.0
Viscosity at 40 °C	cSt	min	1.40	1.50	–	–
		max	5.50	6.00	11.00	14.00
Flash Point	°C	min	43	60	60	60
Pour Point, Winter Quality	°C	max	–	- 6	0	0
Pour Point, Summer Quality	°C	max	–	0	6	6
Cloud point	°C	max	- 16	–	–	–
Micro Carbon Residue on 10% Residue	% m/m	max	0.30	0.30	–	–
Micro Carbon Residue	% m/m	max	–	–	0.30	0.30
Ash	% m/m	max	0.01	0.01	0.01	0.05
Total Sediment Existent	% m/m	max	–	–	(–) 0.10	0.10
Water	% V/V	max	–	–	0.30	0.30
Cetane Number		min	45	40	35	–
Visual Inspection			(Clear) Clear and Bright	(Clear) Clear and Bright	–	–
Sulphur	% m/m	max	1.0	1.5	2.0	2.0
Vanadium	mg/kg	max	–	–	–	100
Aluminium + Silicon	mg/kg	max	–	–	–	25
Used Lubricating Oil (ULO)[1,2]						
– Zinc	mg/kg	max	–	–	–	15
– Phosphorus	mg/kg	max	–	–	–	15
– Calcium	mg/kg	–	–	–	–	30

1 Fuel shall be free of ULO.
2 All three elements shall exceed the same limits before a fuel shall be deemed to contain ULO.

8.2 Composition of liquid fuels

Apart from carbon and hydrogen, many other elements are found in fuels, such as oxygen, nitrogen, sulphur, and various metals which are often bound to the hydrogen carbon molecules.

8.3 Definition of heavy oil

This is a fuel which is manufactured from the remaining distillate of the crude oil distillation refining process and is characterised by its viscosity. Residual fuel is a general term for heavy fuels that are characterised by a high specific mass and a high viscosity. Heavy oil consists mainly of residues from the distillation and refining process.

Heavy fuel values

For specific mass: between 950 and 1020 kilograms per m³.
For viscosity: over 30 cSt at 50 °C.

Diesel oil consists mostly of lighter fractions of crude oil.

Gas oil: specific mass 820 to 880 kilograms per m³.
Marine diesel oil: specific mass 840 to 920 kilograms per m³ for a viscosity to 30 cSt at 50 °C.

Definition heavy oil diesel engine: A true heavy oil diesel engine can start, stop and run on heavy oil at every load.

8.4 Refining crude oil

Refining process

Step 1: Salt is removed by water washing.
Step 2: Atmospheric distillation. Crude oil is heated up to 375 °C under atmospheric pressure.

Products
gases
distillates
vapours

Step 3: Vacuum distillation at an absolute pressure of 0.05 to 0.15 bars and a temperature of 400 °C. Then a further distillation of the atmospheric residue takes place.

Products
vacuum gas oils
vacuum residues
These residues already have a high viscosity and specific mass.

Step 4: Thermal-cracking; the vacuum residue is thermally treated with high temperatures and pressures to produce products with a lower viscosity.

Products
gas
distillates
thermal-cracking residue

ISO 8217 standard; 2005				IFO (non-standard) →		180			380			
International Standard for Marine Distillate Fuels	ISO Grade	2005	RMA30	RMB30	RMD80	RME180	RMF180	RMG380	RMH380	RMK380	RMH700	RMK700
Characteristics	**Units**	**Limit**										
Density at 15 °C	kg/m³	max	960	975	980	991	991	991	991	1010	991	1010
Kinematic Viscosity at 50 °C	cSt	max	30	30	80	180	180	380	380	380	700	
Flash Point	°C	max	60	60	60	60	60	60	60	60	60	60
Pour Point, Winter	°C	max	0	24	30	30	30	30	30	30	30	30
Pour Point, Summer	°C	max	6	24	30	30	30	30	30	30	30	30
Micro Carbon Residue	% m/m	max	10	10	14	15	20	18	22	22	22	22
Ash	% m/m	max	0.10	0.10	0.10	0.10	0.15	0.15	0.15	0.15	0.15	0.15
Water	% V/V	max	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Sulphur	% m/m	max	3.5	3.5	4.0	4.5	4.5	4.5	4.5	4.5	4.5	4.5
Vanadium	mg/kg	max	150	150	350	200	500	300	600	600	600	600
Aluminium + Silicon	mg/kg	max	80	80	80	80	80	80	80	80	80	80
Total Sediment Potential	% m/m	max	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10[2]
Used Lubricating Oil (ULO)[1,2] - Zinc - Phosphorus - Calcium	 mg/kg mg/kg mg/kg	 max max max	 15 15 30	 15 15 30	 15 15 30	 15 15 30	 15 15 30	 15 15 30	 15 15 30	 15 15 30	 15 15 30	 15 15 30

1 Fuel shall be free of ULO.
2 All three elements shall exceed the same limits before a fuel shall be deemed to contain ULO.

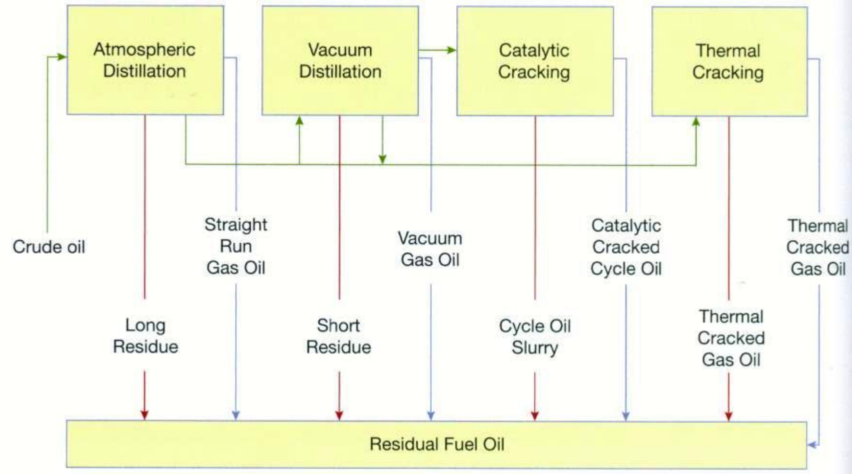
◄ Heavy oil requirements according to ISO 8217/1996.2005.

Fuels can be divided into ten different fuel classes. The viscosity at 50 °C ranges from 30 to 700 cSt. This is often expressed in centistokes; sometimes in mm² per second.

▶

A simple diagram of the processes that crude oil undergoes in order to obtain products such as gases, gas oil, diesel oil and lubricating oil.

The most popular heavy fuel is made from the waste product residue that remains after these four processes. Due to an increased demand for lighter fuels in for instance road transport, aviation and the chemical industry, refineries are increasingly looking for ways in which lighter carbon compounds can be refined from crude oil. This results in a poor quality residue, which forms the basic product of the heavy fuel and therefore poses problems for the heavy fuel powered diesel engines.



Step 5: Catalytic cracking; catalysts added to the process aiding the process of breaking down long hydrocarbon chains without requiring high temperatures.

Products
petrol
light fuels
heavy fuels

Note
Part of the catalysts, the so-called **'cat-fines'** can remain in the heavy oil. These aluminium compounds can seriously damage the fuel pumps and atomisers and must be removed by centrifuges.

Shell Hycon installation
About a dozen years ago Shell developed a special method to process crude oil, the so-called Hycon installation. During this process hydro carbon is injected into the crude oil under a pressure of 200 bar and at 500 °C and heavy asphaltenes are broken down into smaller carbon chains which can then be distilled. Sulphur and vanadium are then removed and very little residue remains. Due to the growing number of Hycon refineries that are being commissioned, the amount of residue from which heavy oil can be obtained is decreasing along with its poor quality making it less suitable for diesel engine combustion.

8.5 Chemical composition of hydro-carbon compounds

One distinguishes between **paraffins** and **aromats**.

Paraffins consist of long rigid chains.
As crystallisation occurs at relatively high temperatures, this fuel requires heating at low temperatures in order to prevent problems such as pipe blockage.
Properties: good ignition properties and high solidification point.

Aromats consist of conjugated planar ring systems. They have poor ignition properties and may contain asphaltenes that have a high specific mass. Asphaltenes are oil molecules that do not dissolve in light crude oils.
Properties: high specific mass and poor ignition properties.

Heavy oil is always a mixture of the paraffins and aromats.

8.6 Standardisation of liquid fuels

It is imperative that guidelines be established for liquid fuels. We have already seen that fuels can roughly be classified as follows.

Gas oil
This product consists of distillates and is used for small fast running diesel engines with low temperature operating conditions. Examples are back up generators and boat engines. **Category I engines.**

Marine diesel oil

Often indicated as 'M.D.O.'.
This product consists for a large part of distillates and contains a small percentage of residues.
This percentage usually lies between 15 and 20 percent. Generally, the fuel is not heated. There is risk of pipe blockage due to crystallisation of the paraffins at low temperatures. This is also called 'flocculation'.
This fuel is generally used in high-speed diesel engines. **Category I and II engines.**

Heavy oil

Often indicated as 'H.F.O.'.
This product consists mainly of various residues mixed with lighter fuels so that the correct viscosity is obtained. Depending on the type of mixture, these fuels must be heated to a maximum of 150 °C for optimum combustion. H.F.O. is used in large low-speed two-stroke crosshead engines and larger medium-speed four-stroke trunk piston engines. **Category III en IV engines.**

Fuels must meet minimum requirements

These requirements have been laid down by ISO, the International Organization for Standardization, in collaboration with the association of diesel engine manufacturers, the **Cimac**, the French **'Conseil International des Machines à Combustion'.** Cimac is an interest group in the field of the 'non-automotive' combustion engines for gas- and diesel engines as well as gas turbines.

The requirements are stipulated in the standard specification ISO-8217: 1996/2005.
Each specification has a test method indicating a minimum or a maximum value acknowledged by the ISO.

The following table contains the specifications for heavy fuels. Shown here, the standards for diesel engines make MAN–B&W, four-stroke medium-speed type V40/50. Engine category III.

▼

The kinematic viscosity is defined at100 °C, the fuel treatment mean temperature. All fuel related data are as a standard made available in English. The table provides the compliance specifications for heavy fuel oil. There are six viscosity categories, which can be further divided into five different classes of specific mass, carbon parameter number and vanadium content.
The classification is denoted with an 'R' of residue. The numbers provide the kinematic viscosity at 100 °C, so RMA 10, RMD 15, RMK 35 etcetera.

The specifications for engines built in or after 2005 are in accordance with ISO-8217/2005. Engines built before 2005 are specified under a different standard.

Marine residual fuel specifications
Fuel classes according to ISO 8217(1996)

<table>
<tr><th rowspan="2">Characteristics</th><th rowspan="2">Dimension</th><th rowspan="2">Limit</th><th colspan="15">ISO classes</th></tr>
<tr><th>RMA 10</th><th>RMB 10</th><th>RMC 10</th><th>RMD 15</th><th>RME 25</th><th>RMF 25</th><th>RMG 35</th><th>RMH 35</th><th>RMK 35</th><th>RMH 45</th><th>RMK 45</th><th>RML 45</th><th>RMH 55</th><th>RMK 55</th><th>RML 55</th></tr>
<tr><td>Density at 15 °C</td><td>kg/m³</td><td>max.</td><td>975.0</td><td colspan="2">981.0</td><td>985.0</td><td colspan="2">991.0</td><td colspan="2">991.0</td><td>1010.0</td><td>991.0</td><td>1010.0</td><td>–</td><td>991.0</td><td>1010.0</td><td>–</td></tr>
<tr><td>Kinematic viscosity at 100 °C</td><td>mm²/s [1]</td><td>max.</td><td colspan="3">10.0</td><td>15.0</td><td colspan="2">25.0</td><td colspan="3">35.0</td><td colspan="3">45.0</td><td colspan="3">55.0</td></tr>
<tr><td>Flash point</td><td>°C</td><td>min.</td><td colspan="3">60</td><td>60</td><td colspan="2">60</td><td colspan="3">60</td><td colspan="3">60</td><td colspan="3">60</td></tr>
<tr><td>Pour point (superior) [2]
winter quality
summer quality</td><td>
°C
°C</td><td>
max.
max.</td><td>
0
6</td><td colspan="2">
24
24</td><td>
30
30</td><td colspan="2">
30
30</td><td colspan="3">
30
30</td><td colspan="3">
30
30</td><td colspan="3">
30
30</td></tr>
<tr><td>Conradson Carbon residue</td><td>% (m/m)</td><td>max.</td><td colspan="2">10</td><td>14</td><td>14</td><td>15</td><td>20</td><td>18</td><td colspan="2">22</td><td colspan="2">22</td><td>–</td><td colspan="2">22</td><td>–</td></tr>
<tr><td>Ash content</td><td>% (m/m)</td><td>max.</td><td colspan="3">0.10</td><td>0.10</td><td>0.10</td><td>0.15</td><td>0.15</td><td colspan="2">0.20</td><td colspan="3">0.20</td><td colspan="3">0.20</td></tr>
<tr><td>Water content</td><td>% (V/V)</td><td>max.</td><td colspan="3">0.5</td><td>0.8</td><td colspan="2">1.0</td><td colspan="3">1.0</td><td colspan="3">1.0</td><td colspan="3">1.0</td></tr>
<tr><td>Sulphur content</td><td>% (m/m)</td><td>max.</td><td colspan="3">3.5</td><td>4.0</td><td colspan="2">5.0</td><td colspan="3">5.0</td><td colspan="3">5.0</td><td colspan="3">5.0</td></tr>
<tr><td>Vanadium</td><td>mg/kg</td><td>max.</td><td colspan="2">150</td><td>300</td><td>350</td><td>200</td><td>500</td><td>300</td><td colspan="2">600</td><td colspan="3">600</td><td colspan="3">600</td></tr>
<tr><td>Aluminium + silicon</td><td>mg/kg</td><td>max.</td><td colspan="3">80</td><td>80</td><td colspan="2">80</td><td colspan="3">80</td><td colspan="3">80</td><td colspan="3">80</td></tr>
<tr><td>Total potential sediment</td><td>% (m/m)</td><td>max.</td><td colspan="3">0.10</td><td>0.10</td><td colspan="2">0.10</td><td colspan="3">0.10</td><td colspan="3">0.10</td><td colspan="3">0.10</td></tr>
</table>

1) 1mm2/s = 1 cSt.
2) The buyers must make sure that this pour point is appropriate for the equipment, especially for ships which sail from one hemisphere to the other

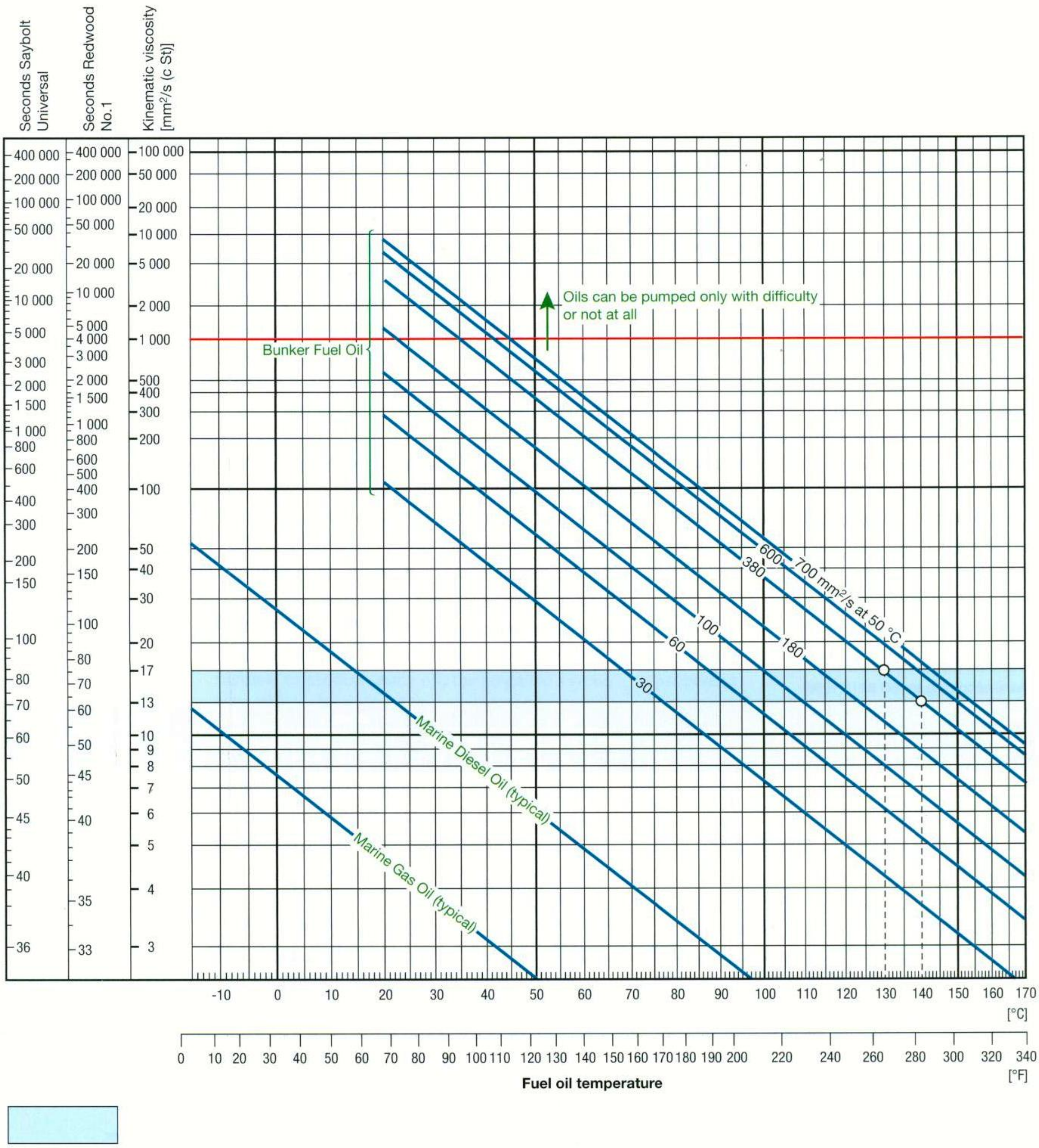
▼

A viscosity-temperature table for fuels.

horizontal: the fuel temperature between –10 and + 170 °C
vertical: the fuel viscosity in various units.

The unit to the right is most frequently used: the kinematic viscosity in mm² per second of centistokes.
The light blue horizontal band indicates the optimum fuel injection viscosity area. In this case, 13 to 17 centistokes.

The horizontal line at 1000 centistokes gives the maximum viscosity value at which the fuel is pumpable from, for instance, a storage tank to a settling tank.
The slanted lines are the lines for various fuels at a certain viscosity.
Bottom left: gas oil, subsequently diesel oil and then the heavy fuel oils of 30, 60, 100, 180, 380, 600 and 700 centistokes.



Example
To obtain the recommended viscosity before fuel injection pumps a fuel oil of 380 mm²/s (cSt) at 50 °C must be heated to 130 to 140 °C

8.7 Fuel properties

8.7.1 Viscosity – in centistokes or mm²/second

This is a measure of the viscosity or 'resistance to flow' of fuel. It is measured by time taken for a fluid to drain from a graduated cylinder through a calibrated opening. The longer the discharge time the thicker the oil and the higher the viscosity. The unit of measure is **centistokes indicated by cSt.** This can also be indicated by mm^2 per second. The viscosity is, of course, strongly influenced by the temperature; this is why the temperature at which the measurement is taken is always noted.

Examples

At -10 °C, gas oil still has good viscosity of ± 10 cSt.
Diesel oil still has good atomising qualities at 15 °C.
Heavy oil of 30 cSt. requires heating to approximately 75 °C to atomise properly.
Heavy oil of 380 cSt. requires heating to approximately 130 to 140 °C to obtain the correct viscosity.

Comments

Naturally every engine manufacturer has his own prescribed viscosity range for optimum atomisation.
Medium-speed engines with a high number of revolutions and therefore a shorter process time often have lower viscosity fuels.
Two-stroke low-speed crosshead engines with a longer process time often have fuels with a viscosity that can increase up to 15 cSt.
The maximum allowable fuel temperature is often prescribed by manufacturers and amounts to 150 °C. This is mainly done to avoid problems with the fuel pump seals. Fuels with a viscosity of 600 and 700 centistokes are rarely used as the price difference with fuels having a viscosity of 380 cSt. is negligible. The fuel temperature needs to be approximately 10 °C higher at 600 and 700 cSt.!

8.7.2 Density denoted by kilogram per m³

This signifies the mass in kilograms per m^3 at 15 °C. This is very important with regard to, for instance, the required storage space and the settings of the centrifuges. The most advanced centrifuges can clean fuel with a density of 1010 kg per m^3 at 15 °C.

Also see Chapter 24, Auxiliary systems: Fuel and lubricating-oil centrifugal separators.

The rule of thumb for the density correction at an increased temperature:

$$s.m._{actual} = s.m._{15\,°C} - 0.64 \times (t_{actual} - 15)$$

8.7.3 Flash point denoted by degrees Celsius

The flash point is the minimum temperature at which the fuel vapours formed over an oil sample that is warmed-up in a special heating device, first burns after being ignited with a test flame.
The Pensky Martens closed cup flash point tester is a well-known and often applied instrument for determining the flash point. In the shipping industry the minimum flash point of 60 °C is enforced due to safety issues (explosion- and fire hazards). Fuel for barge engines and back-up generators forms an exception. To ensure low temperature ignition of these engines a flash point of 43 °C is allowed. Modified rules apply to the M.O.B. (man-over-board-boat) which is equipped with an outboard motor and where very light petroleum is used as a fuel.

8.7.4 Flow point –denoted by degrees Celsius

This is the lowest temperature expressed in degrees Celsius at which a fuel can be treated or still flows. This is about three degrees above the solidifying point. The solidifying point is of the utmost importance in the storage and processing of fuels.
Furthermore, the fuel must be pumpable. Once heavy oil has solidified, it is extremely difficult to liquify it by using heating coils. The oil often only liquifies around the coils as the oil no longer circulates. When using heavy oil heating is an absolute necessity.

8.7.5 Carbon residue or Conradson-carbon number, also denoted by Micro Carbon Residue abbreviated as M.C.R. – indicated in percentage mass

This is a measurement for the formation of carbon and coal deposits during combustion. This is important when considering the contamination of the combustion space in which the atomiser nozzles, the exhaust valves, the pistons, the piston rings and the piston ring grooves are found and in the exhaust system containing among others the turbo blower, the exhaust gas boiler and the silencers.
The carbon residue is determined in the Conradson apparatus; fuel is heated in a crucible. At high temperatures the lighter hydrocarbons vaporise and the heavier hydrocarbons are cracked, breaking down the long chains into smaller ones. The pure carbon residue that remains is subsequently measured.

8.7.6 Ash content – denoted by percentage mass

This represents the inorganic material content that is left behind after combustion at extremely high temperatures when the carbon is removed. For ship fuels the ash content lies between 2/1000 and 9/100 percent mass.
The inorganic metal oxides found are often those of vanadium, nickel, chromium and sometimes sodium. The latter mostly resulting from salt water flooded oil fields or during transport. Sodium can also be introduced into the fuel during crude oil processing such as washing. Aluminium and silicon are also found in fuel. They are introduced during the catalytic cracking process. Fuel also contains substances such as rust and dirt, mostly from storage tanks, bunkers or double bottom tanks.
If the ash content in the fuel is considerably higher than the sum of the parts of sodium, aluminium and silicon, further research is required. Ashes may cause wear and engine corrosion.

8.7.7 Water content –denoted by percentage volume

Fuels always contain a small amount of water. This enters during transportation, processing in the refinery and storage in tanks, especially air by condensation. A maximum of 1% of water is allowed in fuel. In reality 80% of the fuels contain less than 0.3% water. At high water content values the heat value of fuel decreases.

8.7.8 Sulphur content denoted by percentage mass

This is one of the most common elements in liquid fuels. The sulphur content in heavy oil varies from 1.5 to 3.5%. Sulphur poses problems in the form of low temperature corrosion, L.T.C. (Low Temperature Corrosion). Together with water and oxygen, which are always present in air, sulphur forms sulphuric acid. Sulphuric acid can damage the cylinder liner and, for instance, the exhaust systems at low engine loads.
In the future, IMO will check ship fuels in order to reduce air pollution. Worldwide the maximum sulphur content will be set at 4.5% and in vulnerable areas, a maximum of 1.5%. This measure will become effective before 2010.

8.7.9 Vanadium content denoted by milligrams per kilogram and also in parts per million (normally indicated by p.p.m.)

This metal is found in all fuels. It is bound to hydro-carbons and can not normally be separated from the fuel. Vanadium contents can vary from 10 p.p.m. to 1000 p.p.m., dependent on the natural oil field. After combustion of the fuel in the cylinder the vanadium remains in the ash. At a certain temperatures and compositions it forms, when combined with sodium and sulphur oxides, a very aggressive precipitate especially on the exhaust valves.
Adequate cooling of the exhaust valve and valve seats and usage of, for instance, a corrosion resistant valve material such as 'Nimonac' this high temperature corrosion, (abbreviated H.T.C.), can be limited.

8.7.10 Aluminium- and silicon content – denoted by milligrams per kilogram and often p.p.m.

This occurs mainly in the form of an aluminium silicate, a catalyst used in the catalytic cracking of crude oil. These compound silicates are small (1 to 100 micro millimetres) and very hard and can cause serious wear to atomisers, fuel pumps, pistons, piston rings and cylinder liners.
A good centrifuge can remove the majority of these silicates as their specific mass far exceeds that of the fuel.
A maximum of 80 p.p.m. total of the two metals is allowed to be present in the fuel and aluminium alone has a limit of 30 p.p.m..

8.7.11 Total sediment – indicated by percentage mass

These are ash-like substances or insoluble hydrocarbons. Heavy oil contains asphaltenes which are usually soluble in the fuel. Fuels that contain insoluble asphaltenes, are called unstable fuels. These unstable fuels may contain large quantities of sediment, overloading the centrifuges and causing them to work poorly.
Fuel obtained from the catalytic cracking process contains many aromats. This is one of the main reasons that suppliers accept it as a diluent.
Fuels acquired from either thermal cracking or from the 'Hycon'-process pose more problems concerning fuel stability.
Simply put: the latter techniques increase the likelihood of unstable fuels.

8.8 Additional fuel specifications

These are not incorporated in standard specifications, but are in practice important to the user and are often prescribed by engine builders.

8.8.1 Calculated Carbon Aromaticity Index, indicated by the abbreviation C.C.A.I.: indefinite number

This is a unit with which the ignition quality of heavy oil is expressed.
This C.C.A.I. has been introduced because the quality of heavy oil continues to deteriorate. Modern refineries can increasingly refine light fuels from various residues, which leaves the last remnants proving increasingly difficult to use for diesel engines. **The ignition quality is especially important.**

The Shell system of quality indicators is used. Fuels that have a high carbon percentage ('heavy oil') which are bound to aromatic compounds produce a longer ignition delay than fuels with fewer aromats. Research has also shown that there is a relationship between the carbon aromaticity and the fuel properties of density and viscosity. The C.C.A.I. value can be established by means of a formula or a nomogram.

Formula:

$$CCAI = \rho - 81 - 141 \times \lg \cdot [\lg \cdot (v + 0{,}85)] - 483 \times \lg\left(\frac{T + 273}{323}\right)$$

Where:
ρ = fuel density in kg/m³ at 13 °C;
v = kinematic viscosity in mm²/sec. or cSt at 50 °C;
T = temperature in °C when the viscosity measurement is taken.

One often uses the above nomogram in which the viscosity and the density of the fuel are known by drawing a straight line through these two known points which bisect the C.C.A.I.-line. The value at this intersection is the C.C.A.I.-value for the fuel.

C.C.A.I.-values

Normal heavy oil has C.C.A.I.-values of 770 to 840 and is easy to ignite. Cracked residues have values that can increase to 840 or even to values exceeding 900. Most heavy oils have maximum values of approximately 850 to 870.

In many engine manufacturers' product guides the maximum C.C.A.I.-value is mentioned in addition to all other (ISO) specifications.

8.8.2 Calorific value

This is the heat that is released during the combustion of 1 kilogram of fuel, expressed in kilojoules per kilogram of fuel.

We know the higher heating value. Here the combustion gases are cooled to the point that the water vapour in the flue gases condenses thereby releasing the condensation heat present in the water vapour. In reality this condensation very rarely occurs in diesel engines. Just think of the formation of sulphuric acid!
In gas engines one often uses this condensation heat by placing a flue gas condenser in the exhaust pipe. This is manufactured from stainless steel and on top of this the gases contain very little sulphur.

▼

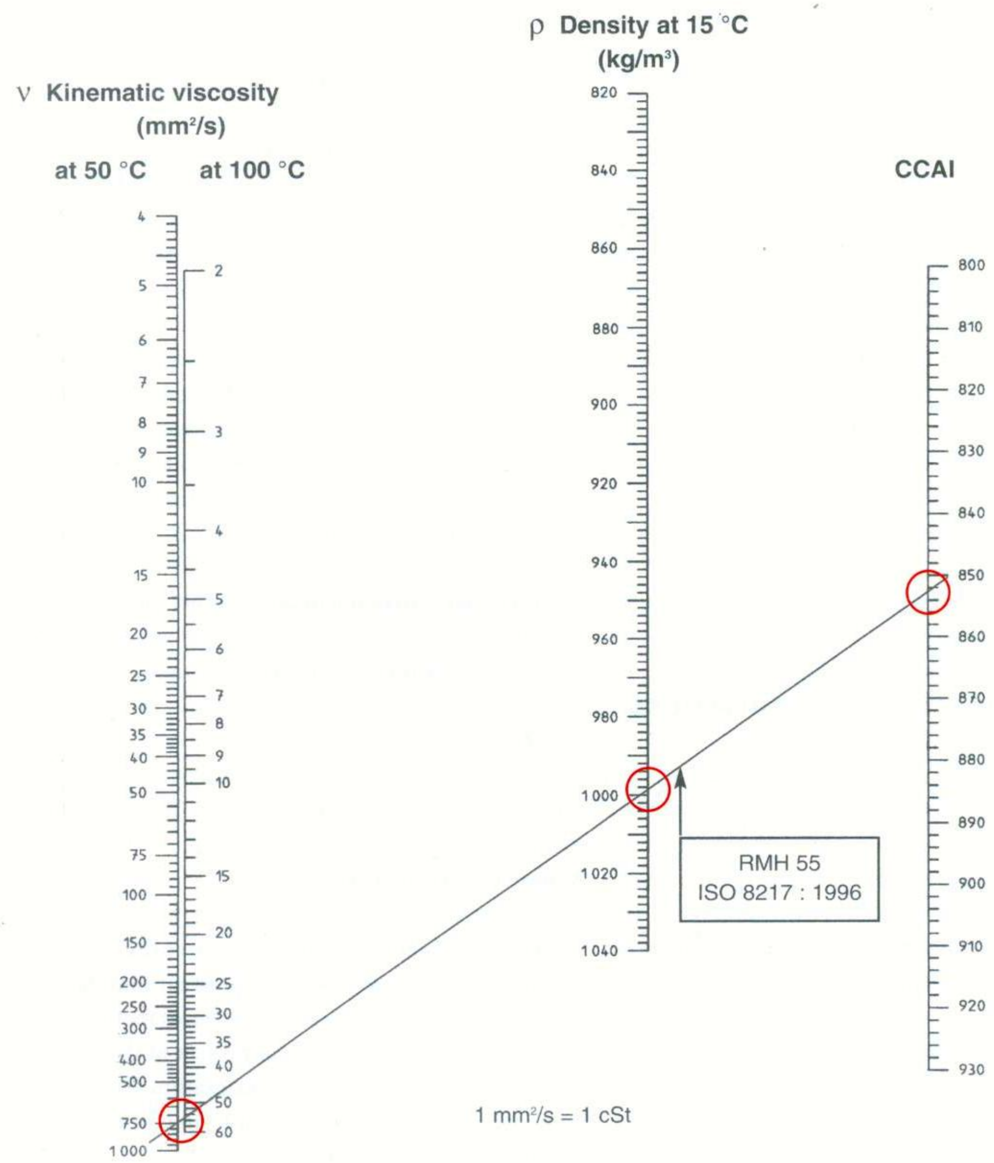
A so-called nomogram to determine the C.C.A.I.-number of a heavy fuel.

The first vertical column represents the kinematic viscosity at 50 and 100 °C. Unit in mm² per second.
The middle vertical column represents the specific mass at 15 °C. Unit in kilograms per m³.

The vertical column to the right represents the C.C.A.I.-number. This is an abstract number.
The fuel RMH 55 is an example. It has a viscosity of ± 750 cSt and a specific mass of 990 kg per m² at 15 °C.
By drawing a line to the right through the points mentioned above, the line intersects the right column at a C.C.A.I.-number of approximately 852. This is fairly high, but nevertheless acceptable.

$$\mathbf{CCAI} = \rho - 81 - 141 \log_{(10)} \left[\log_{(10)} (\nu + 0{,}85)\right] - 483 \log_{(10)} \left(\frac{T + 273}{323}\right)$$

ρ : density at 15 °C (kg/m³)
ν : kinematic viscosity (mm²/s) at T temperature
T : temperature corresponding to the kinematic viscosity measurement (°C)



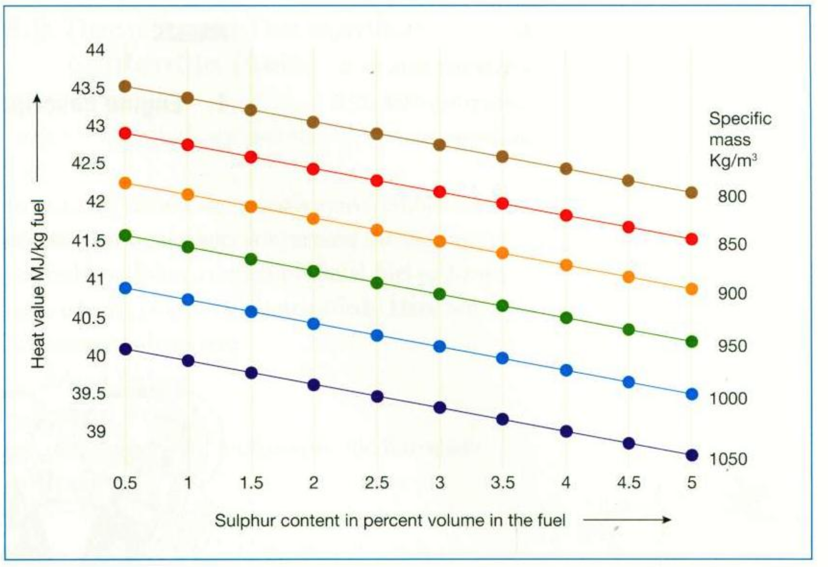
NOMOGRAM TO DEDUCE CCAI

This diagram can be used to find the CCAI number of a fuel oil when density and viscosity are known. The CCAI number can be determined by drawing a straight line through the viscosity and density values. The line for the RMH 55 (ISO 8217 : 1996) is outlined as an example.

▶
A chart which clearly shows the effects of the sulphur content and the specific fuel mass on the heat value.

horizontal: the sulphur content in percent volume in the fuel
vertical: the heat value in MJ per kilograms fuel
the slanted lines: the specific mass of the fuel in kg per m^3
The bottom line has a specific mass of 1050 kg per m^3.
The top line has a specific mass of 800 kg per m^3.

In relation to heavier fuels, light fuels contain less carbon atoms and more hydrogen atoms. Carbon has a relatively high atomic weight and hydrogen, a relatively light atomic weight.
Hydrogen generates more energy during combustion than carbon. A high sulphur content also inhibits the provision of energy.
Fuel with a low specific mass and sulphur content has therefore the highest combustion value.



We usually work with the lower heating value or net calorific value.

Lower heating value = higher heating value – heat of vaporisation.

The lower heating value is determined by, among other things, the following properties:
- the density;
- the water content;
- the ash content;
- the sulphur content.

Examples

Fuel with a specific mass of 800 kg per m^3 and a sulphur content of 0.5% has a lower heating value of 43.5 MJ/kg.
Fuel with a specific mass of 1050 kg per m^3 and a sulphur content of 4% has a lower heating value of 39.2 MJ/kg.

The lower heating value can also be established by using formulas and nomograms.
Below is an example of how to calculate the lower heating value or **'specific energy'**.
A higher density means that the fuel contains more carbon and fewer hydrogen atoms.

Lower heating values of common fuels

gas oil	43,000 kJ/kg
diesel oil	42,700 kJ/kg
heavy oil	40,500 kJ/kg

It is easy to calculate the energy supply to the diesel engine by using the lower heating value.

Examples

A six-cylinder Caterpillar–MaK 32 C diesel engine has a power output of 500 kW per cylinder.
The specific fuel consumption for the screw propulsion is 178 grams per kWh at full capacity (100%). The engine uses heavy oil with a lower heating value of 42,000 kJ per kg.
What is the total efficiency of this engine?

Energy supply to engine = P_F =
Fuel consumption = 6 × 500 × 0.178 =
534 kilogram per hour.

Supplied energy = mass fuel per second × lower heat value

$$P_F = \frac{kg}{sec} \times \frac{kJ}{kg} = \frac{kJ}{sec} = kW$$

$$P_F = \frac{534}{3600} \times 42.000 = 6230 \text{ kW}$$

$$\text{Total engine efficiency} = \frac{\text{Shaft power}}{\text{Energy input}}$$

$$\text{Total efficiency} = \frac{3000}{6230} = 0.4815 \text{ or } 48.15\%.$$

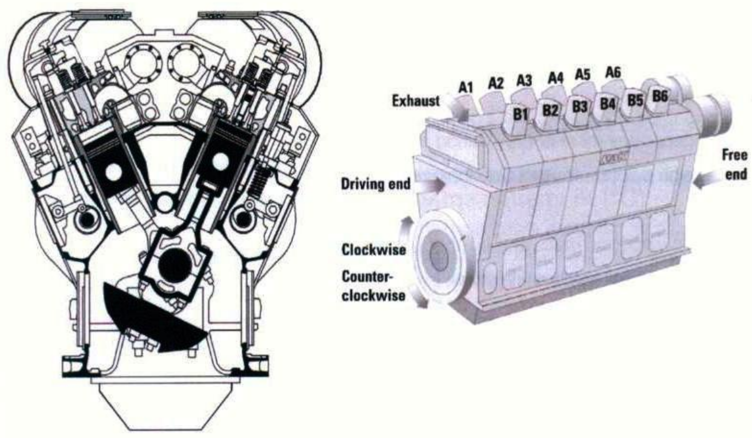
▶
Important data for a Caterpillar–Mak 32 C diesel engine.

MaK

1. Engine description

The M 32 C is a four stroke diesel engine, non-reversible, turbocharged with direct fuel injection.

V-engine M 32 C



Cylinder configuration:	12, 16 V
Bore:	320 mm
Stroke:	420 mm
Stroke/Bore-Ratio:	1,3
Swept volume:	33,8 l/Cyl.
Output/cyl.:	480/500 kW
BMEP:	23,7/23,7 bar
Revolutions:	720/750 rpm
Mean piston speed:	10,1/10,5 m/s
Turbocharging:	single log
Direction of rotation:	clockwise, option: counter-clockwise

8.8.3 Fuel stability

Fuel is stable when during long-term storage in, for example, tanks, bunkers or a double bottom on ships, heavy components do not separate and gravitate to the bottom of the storage tanks. Especially in the thermal cracking process, the residue usually contains very long and heavy chains which are difficult to dissolve in the surrounding (lighter) oil distillates.
When these residues are mixed with much lighter distillate oils in the manufacture of the heavy oil, it is a possibility that the heavy compounds slowly sink. This is called a volatile fuel. One often uses gas oil in the mixture.

Using these volatile fuels can create serious problems:
- **sludge forming in the fuel tanks;**
- **problems with the centrifuges during the fuel cleaning process. Here a large amount of sludge (sediment) is extracted, which often blocks the centrifuge entirely;**
- **fuel fine filters can get completely obstructed.**

8.8.4 Miscible fuels

It should be stressed that mixing different fuels is highly inadvisable.
There is always the possibility that this can result in a unstable fuel. Tanks should preferably be drained completely before being filled with a 'new

fuel'. In reality it is not always feasible to remove all the 'old' fuel from a tank.
To establish the fuel stability Shell has developed a test which can be performed on site. Heavy fuel instability is a common occurrence.

8.8.5 Ignition quality, the Cetane number

This only applies to light distillates such as gas oil and diesel oil.
The Cetane number is a measurement of the ignition quality of fuel. Highly aromatic fuels have a lower cetane number. Pre-chamber diesel engines require fuel that has a cetane number of at least 35. Direct injection engines require a cetane number of at least 40. The cetane number is established using a test engine.

8.8.6 Combustion quality

When a fuel ignites easily, this means it will also combust well. The hydro carbons with chain-like molecule structures (paraffins), burn faster than the hydro carbons with circular molecule structures (naphthenes). During combustion of the latter a large amount of free carbon is produced by the cracking reactions, which in turn translates into high carbon numbers. Today the carbon number is extremely high, over 20 % to be precise, this causes slow combustion.

8.8.7 Hydrogen sulphide (H_2S)

This is a toxic gas which can be present in some crude oils and heavy oil fuels. Damage can occur when present in high concentrations. Water, a combustion by-product, can form sulphuric acid (H_2SO_4) when mixed with hydrogen sulphide.

Inhaling hydrogen sulphide

This is an extremely dangerous health hazard. Death can occur rapidly at high concentrations.

▶
A table providing the concentration of H_2S in parts per million.

At a concentration of approximately 100 p.p.m., sulphur gas inhalation is life threatening. Lower concentrations can also pose a health hazard.

8.9 Decreasing the sulphur content in fuels

Regulations with regard to sulphur content in fuels.

In order to reduce the negative environmental effects, the regulations concerning the maximum allowable sulphur content of diesel fuel or heavy oil are being continually intensified. Here one differentiates between:
- **inland waters;**
- **coastal waters;**
- **regional waters, for instance the European Union.**

These stringent regulations already apply to diesel engines used in the automotive industry or, for example, cranes and trains (traction). The regulations are also being tightened for other engine types in inland navigation such as propulsion, gensets and bow thruster propulsion. Diesel engines require a certificate stating that they meet the stipulated emission values. Rules and regulations have also been implemented and are becoming more stringent for the international shipping industry and diesel power plants.

H2S concentrations and their effect on humans

H2S Concentration PPM	Physiological effect
0.2 – 0.5	First detectable by smell.
10	Limit for eight hours working, may cause some nausea, eye irritation.
15	Limit for 15 minutes working. Eye and respiratory tract irritation.
50 – 100	Eye and respiratory tract irritation after 1 hour. Longer exposures to concentrations at 100 PPM induce a gradual increase in the severity of these symptoms and death may occur after 4-48 hours exposure.
150	Loss of sense of smell.
350	Could be fatal after 30 minutes inhalation.
700	Rapidly (few minutes) induces unconsciousness and death. Causes seizures, loss of control of bowel and bladder. Breathing will stop and death will result if not rescued promptly.
700+	Immediately fatal.

It is technically fairly simple to further reduce the sulphur content of engine fuels. The rules regarding the maximum sulphur content of gas oil and diesel oil already stringent.
In the territorial waters of Europe ships are required to prove that the sulphur content of their diesel oil amounts to less than 0.2%.
So ships not running on low sulphuric diesel oil are obliged to reserve a separate tank suitable for fuels with low sulphur content.

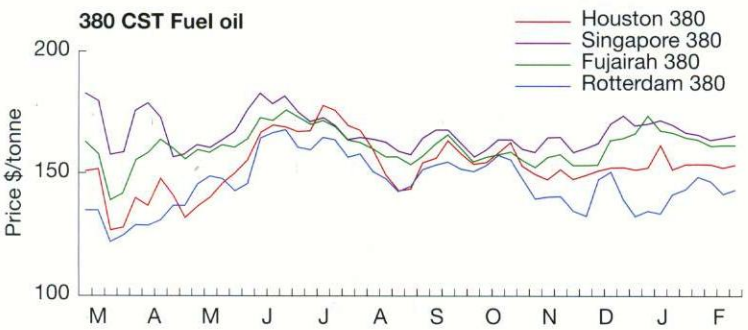
Let's assume that the regulations for heavy fuel are as follows: the sulphur limit is 1.5% according to IMO Marpol Annex VI for regions within which the sulphur rules apply, or perhaps even 1% for certain areas inside the European Union.
This means that ships which normally use heavy oil with sulphur contents exceeding these values, and this is generally the case, are required to have separate tanks for this low sulphuric fuel. Many ships are already equipped with these special tanks, so they can meet the sulphur limit when the rules become effective.

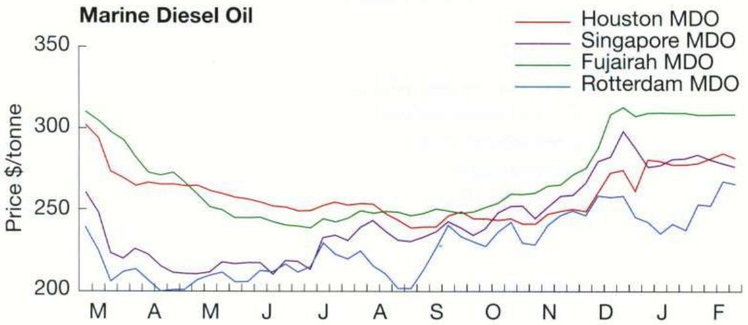
Low sulphur fuel costs

These are largely a matter of supply and demand, but the costs are considerably higher than that of regular heavy oil. Furthermore, these low sulphur fuels are generally difficult to obtain.
One can expect to pay 15 to 20 US-dollar per ton surcharge.

▼
Fuel price development of heavy oil with a viscosity of 380 centistokes and diesel oil throughout 2007 at four bunker stations: Houston, Singapore, Fujairah and Rotterdam.

The prices are provided in U.S.D. per tonnage weight. For heavy oil the average price is 150 U.S.D. per ton and for diesel oil 275 U.S.D. per ton.





At even higher surcharge costs on-shore companies, such as diesel power plants can consider using a different fuel, such as natural gas for Dual Fuel engines or bio-fuels, which are presently (2008) in great demand.

Also see Chapter 29, New fuel developments.

8.9.1 Sulphur free diesel oil

On shore, sulphur free fuel will be increasingly prescribed. In 'sulphur free' diesel oil the sulphur content is less than 1%.

A maximum sulphur content of 0.2% applies to the territorial waters of the European Union for gas oil and diesel oil, including DMA-, DMB-, DMC- en DMX-specifications.
The latest IMO guideline and the European commission states that the sulphur content of heavy oil on the North sea, the English channel and the Baltic sea may not exceed 1.5%. This new rule became effective on the 19th of May 2005!

Presently, heavy bunker oil has an average sulphur content of 2.7%. The price difference between 1.5 % and 2.7% sulphur content bunker oil currently amounts to approximately 30 U.S.D. per ton.
It is very probable that most ships will drain a fuel tank in order to use it for these low sulphur content fuels and run on the ' low sulphur tank' in said areas.

8.10 Bunkering

Everywhere diesel engines are used, fuel delivery is a periodic reoccurrence. Engines with high operating hours such as in navigation and in diesel power plants use enormous amounts of fuel.
The specific fuel consumption of engines lies roughly between 170 and 220 grams per kWh, depending on the working principle, the engine size and the load.

Example 1

A large inland boat with a power output of 1600 kW and 5000 operating hours per annum has a specific fuel consumption of 200 gr/kWh:
Fuel consumption = 1600 × 0.200 × 5,000 = 1,600,000 kg per annum.

This is approximately $\frac{1,600,000}{0.850}$ =

1,882,350 litres or 1882 m³!

380 IFO		Feb-04 2-6	Feb-04 9-13	Feb-04 16-20	Feb-04 23-27	Mar-04 1-5	Mar-04 8-12	Mar-04 15-19	Mar-04 22-26
Rotterdam	d	135	134	142	144	149	147	142	144
Gibraltar	d	160	157	156	156	158	155	149	152
Piraeus	d	148	154	145	146	150	148	144	147
Suez	d	158	156	159	159	161	161	159	164
Fujairah	d	174	168	167	165	164	162	162	162
Durban	w	n/a	n/a	n/a	n/a	n/a	n/a	n/a	n/a
Tokyo	d	201	198	195	196	197	198	197	196
Busan	d	184	181	182	183	186	185	182	179
Hong Kong	d	183	185	184	184	181	182	181	180
Singapore	d	171	172	170	167	166	164	165	166
Los Angeles	w	168	163	161	155	155	155	157	164
Houston	w	153	162	152	154	154	154	153	154
New York	w	163	163	162	158	161	161	157	158
Panama	w	160	160	160	160	161	164	168	168
Santos	d	156	142	144	146	156	156	154	153
Buenos Aires	d	n/a	n/a	n/a	n/a	n/a	n/a	n/a	n/a

180 IFO		Feb-04 2-6	Feb-04 9-13	Feb-04 16-20	Feb-04 23-27	Mar-04 1-5	Mar-04 8-12	Mar-04 15-19	Mar-04 22-26
Rotterdam	d	144	144	152	154	159	158	152	154
Gibraltar	d	169	166	164	164	167	164	157	161
Piraeus	d	160	155	159	160	165	164	161	164
Suez	d	164	162	165	166	167	167	166	171
Fujairah	d	181	174	173	172	171	169	169	169
Durban	w	170	167	169	166	165	163	161	159
Tokyo	d	205	202	200	200	201	202	201	200
Busan	d	192	191	192	192	195	194	192	189
Hong Kong	d	186	187	187	188	187	186	185	184
Singapore	d	175	172	175	170	171	168	170	170
Los Angeles	w	193	186	182	169	172	174	173	178
Houston	w	161	162	160	160	160	164	157	159
New York	w	183	182	177	177	179	178	172	173
Panama	w	170	170	169	171	171	174	178	179
Santos	d	159	146	147	150	160	160	158	157
Buenos Aires	d	171	170	167	165	166	164	165	165

MDO		Feb-04 1-5	Feb-04 8-12	Feb-04 15-19	Feb-04 22-26	Mar-04 5-9	Mar-04 12-16	Mar-04 19-23	Mar-04 26-30
Rotterdam	d	243	236	242	238	254	253	268	266
Gibraltar	d	284	281	300	306	322	319	328	327
Piraeus	d	264	261	277	282	300	298	307	307
Suez	d	321	321	321	321	325	336	335	335
Fujairah	d	310	310	310	310	309	309	309	309
Durban	w	267	263	274	285	301	299	308	311
Tokyo	d	287	288	291	288	287	292	292	290
Busan	d	332	325	322	322	321	321	319	318
Hong Kong	d	274	274	282	283	282	281	279	273
Singapore	d	277	278	282	282	284	281	279	277
Los Angeles	w	317	320	344	358	345	342	342	340
Houston	w	281	280	278	278	279	282	285	282
New York	w	345	343	335	327	325	330	331	330
Panama	w	317	317	318	321	321	323	326	327
Santos	d	350	343	342	344	347	350	345	344
Buenos Aires	d	317	311	302	306	295	303	303	302

◄ The prices of heavy oil and diesel oil in February and March 2004.

The price of fuels is affected by, amongst others, supply and demand, war, natural disasters and international politics.

d = day price

w = week price

▼ A container ship for inland shipping.

Diesel oil is the fuel for propulsion engines and power aggregates.

Example 2

A container ship of 8600 TEU (TEU = twenty-feet equivalent units= standard 20 feet container) with a power output of 72,000 kW and 6500 operating hours has at full load a specific fuel consumption of 175 grams per kWh:

Fuel consumption: $72{,}000 \times 0.175 \times 6500 =$ 81,900,000 kilograms

or $\dfrac{81{,}900{,}000}{0.950} = 86{,}210{,}526$ litres or 86,210 m^3!

On average 2000 tons of heavy oil is bunkered on these ships, but this can be increased to 8,000 tons! Moreover,, large amounts of lubricating oil are used. For the latter example, approximately 120,000 litre per annum!

◄ A large sea-going container ship.

Heavy oil is the fuel for both propulsion engines as for diesel gensets.

▲
Emergency power aggregates usually run on diesel oil.

This type of engine should start within a few seconds and run at full load. With relatively light diesel oil this is not a problem.

Example 3

A back-up genset for a hospital with a power output of 1500 kW has at full load a specific fuel consumption of 190 grams per kWh:

$$1500 \times 0.190 = 285 \text{ kilograms of } \frac{285}{0.850} =$$
335 litres per hour.

If, in case of an emergency situation or any other circumstances requiring the engine to run for two whole days, a fuel storage tank of 2 × 24 × 335 = 16,100 litres would be required.
It is for the diesel engine user of the utmost importance that the quality of the fuel meets all requirements.

8.10.1 Bunkered fuel quality – Who determines the quality?

Firstly, refineries adhere to the ISO 8.217 norm. Additionally, there are the engine manufacturers' recommendations. The latter are far stricter than the former. The owners and users can also follow their own regulations.
Gas oil makes a good example. Apart from liquid it often contains minute metal particles that are so small that most fuel filters can not filter them out. Especially aluminium silicate, the so-called 'cat-fines', that have a two to ten micron diameter and cause increased wear and tear to fuel pumps, atomisers, pistons, and cylinder liners. One also occasionally finds soot particles in filters.
According to ISO 8.217 the maximum allowed amount of metal particles in gas oil is 80 p.p.m. Engine manufacturers of modern engines recommend 15 p.p.m. as the maximum value. Most metal particles can be removed by centrifuges. Of course one must have access to such a centrifuge on an inland boat.
Another solution would be to use very fine fuel filters for these installations, which are able to filter out most of the metal particles. The mesh of these filters is only two microns!

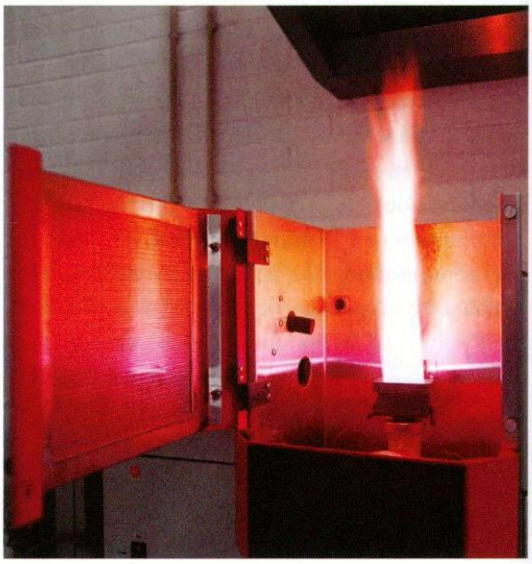
8.10.2 Fuel samples

It is always advisable to take a sample of the fuel supplied. Of course this is not practical for the small amounts purchased at petrol stations. For large to very large amounts the purchase price plays an important role, especially when one encounters problems with regard to storage, cleaning and usage of the bunkered fuel.

▶
Testing fuel samples in a laboratory.

It is very important to identify the fuel composition prior to using it in the diesel engine.

▶▶
Testing fuel in the laboratory of Inpechem Inspectors B.V. in Rotterdam, The Netherlands.



Recommendations for taking fuel samples

The sample needs to be representative. This means that it should represent the average value of the entire fuel intake. It is advisable to take various samples at regular intervals during the bunkering. There is special equipment available which automatically taps small amounts of fuel during bunkering.

8.10.3 Bunker procedures

Ensure that:
- the fuel supplier knows exactly the type and amount of fuel oil required;
- both parties are in agreement over the arrangements, the pump capacity and manner of communication during the bunker operation;
- both parties are aware of how to abandon the bunker operation in the case of an emergency situation;
- both parties follow the bunker activities;
- both parties take all safety- and anti-pollution measures into account prior to commencing the bunkering operation.

The following is important:
- Discuss and make arrangements regarding the start procedure how the opening of the valves and tanks will be executed.
- Discuss and make arrangements regarding the sampling procedures that take place during the bunker operation.
- Take collective fuel samples during bunkering at an agreed location.
- Discuss and make arrangements as to how the supply line will be closed when the bunker operation is terminated.

Make sure that both parties sign the following documents at the correct time:
- amount of fuel;
- type of fuel;
- the bunkering order;
- the pump capacity;
- safety- and anti-pollution regulations;
- starting procedures for the pumps;
- terminating procedures for the pumps;
- method used for the verification of measurements;
- bunkering data;
- sampling data;
- an agreement stipulating that all complaints be submitted in writing and that the signature on the complaint form will only be used to acknowledge receipt.

▲
The bunker station on a large container ship.

Apart from heavy oil and diesel oil, lubricating-oil can also be bunkered. The picture also shows the pipes required for the discharge of contaminated lubricating oil and separator sludge.

The sampling methods and -location and a record of the witnesses attending the sampling is very important. This forms the basis for all the discussions concerning possible ensuing disagreements over the quality of the fuel supplied.

8.10.4 Gas-oil shipping associations

The V.O.S. (Vignet Olie Scheepvaart) foundation, for instance, is active in the Netherlands and monitors the quality of gas oil and issues a hall mark for shipping gas oil.
There are recurring signals of serious malfunctions in engines in the inland shipping branch. This apparently mainly concerns the decrease in the

▼
A sample taking device.

A sample taking device which is placed in the bunker line and periodically takes some fuel from the line, in order to collect a representative sample of the bunkered fuel. This is often referred to as a 'drip sample'.

operating life time of fuel filters. This often causes engines to stall.

Possible problems

It is often suggested that paraffin could be one of the causes. Paraffin is an essential part of every (diesel) fuel. The type and the amount of paraffin in the fuel influence the **'Cloud Point'** (C.P.). The cloud point is the stage at which the paraffins start to 'flocculate' in the fuel, thus blocking the filters. In physics this is referred to as 'crystallizing out'. A process during which the fuel is slightly heated and consequently prevents the paraffin from crystallizing out could form one of the solutions. This, of course, requires some technical adjustments.
The quality of the paper used in the filters could also be one of the problems. Gas oil naturally absorbs 22 p.p.m. of water before it separates from the water. This could pose problems if the filter paper is not manufactured for these conditions. There are also instances where the filters are obstructed by a heavy substance. The composition of this particular substance remains unknown.

Metallic pollution

According to the experts iron and silicon (sand) do not naturally occur in gas oil. Gas oil is a distillate and these elements do not vaporise. They must therefore originate from an external source.

Filter sizes

In the past filters up to10 microns were used and today one uses even finer filters of less than 2 microns. In the automotive industry much coarser filters up to 25 microns are used. Tests have demonstrated that filters less than 2 microns get blocked after a period of time.

Diesel- and gas oil storage

Generally, the maximum storage time of fuel is approximately one year. Contamination with water (liquid) and other substances has to be prevented.

Fuel costs

In exploiting engines in shipping, trade, industry and diesel power plants, the fuel- and lubricating oil consumption form, especially with the engines in continuous use, the greatest cost factor.

Example

A large container ship with a cargo capacity of 7200 TEU can transport about 10,000 tons of heavy oil of which it will consume approximately 250 tons per day. For instance: a price difference of 30 U.S.D. per ton between Rotterdam and Singapore amounts to a price advantage of 250,000 U.S.D. at full bunkering of the ship in Rotterdam. For the standard eight return trips to Asia that a container ship makes on a yearly basis, this amounts to nearly 2,000,000 U.S.D.!

Fuel line systems

In using diesel engines the storage, cleaning and treatment of the fuel before it is ready for use in the engine is of the utmost importance.
Before the fuel can be used, the following applies:

1 Fuel storage
This should be such that the fuel can not be contaminated with corrosives, sand or other impurities. Any condensate should be drained.

2 The amount of fuel
This should be adequate for the planned voyage, that is: the number of operating hours.

3 Fuel quality
This should be established through sampling.

4 Cleaning fuel
The fuel should be cleaned sufficiently by means of filters and/or separators.

5 Engine supply
An adequate amount of fuel should be transported to the engine at a certain pressure and temperature.

6 Fuel specifications
The fuel needs to meet the relevant ISO 8.217 specifications as well as the local and inter-local regulations such as maximum sulphur content.

7 Fuel specifications of engine suppliers
Of course, the fuel must comply with the specifications detailed for the engine make and -type. This is also important in case there is damage, excessive wear and tear or other problems.

General storage of fuels

This can diverge tremendously according to the engine size, the fuel used, the application, such as a diesel power plant or ship propulsion and the number of operating hours.

For small category I diesel engines a simple fuel tank may suffice, whereas category III and IV engines which use large amounts of heavy oil require large storage tanks equipped with heating systems.
There are building instructions in place for both fuel storage tanks on shore and on ships. These serve to prevent fuels polluting the environment and unsafe situations such as fire and explosions. It is important for mechanical engineers to bear in mind the following points of interest:

- storage tanks must never be over-pressurised; therefore a pressure valve should be placed on top of the tank.
- storage tanks should be equipped with a drain valve which can remove water and dirt particles.
- a lid or manhole makes it possible to check for rust formation and dirt during an internal inspection.
- in order to prevent dirt particles, water and rust in the fuel system, the connection between fuel line and engine should preferably not be placed at the bottom of the tank but at a certain distance from the tank floor.
- gauge glasses on tanks should be closed at all times and only be opened when they are read.

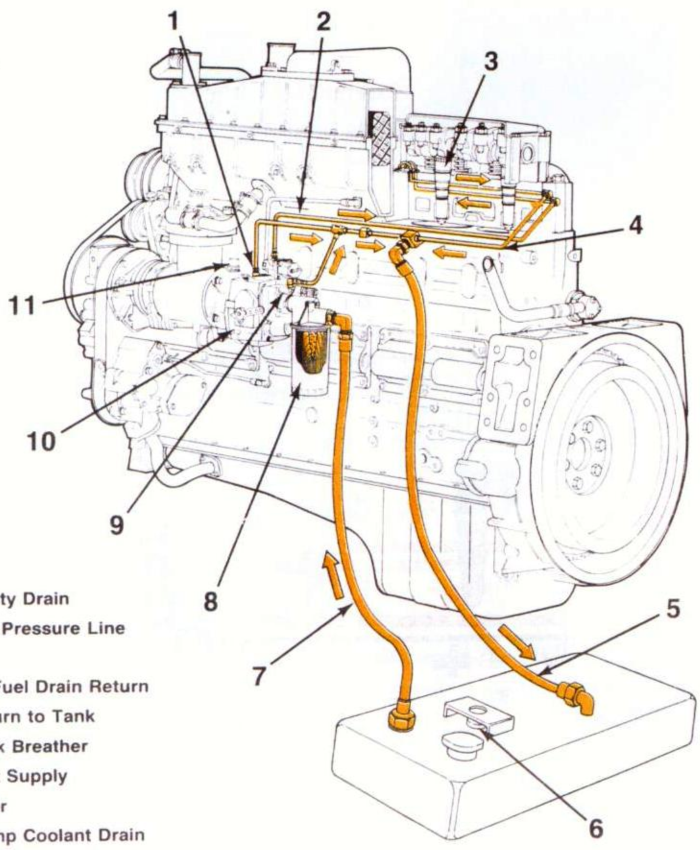
Of course, there are many specific rules and regulations with regard to the building of fuel tanks which are outside the scope of this book.

8.11 Fuel-line systems according to the engine classification

8.11.1 Category I, 0 to 100 kW, four-stroke, high-speed, fuel M.D.O.

These are often very straightforward systems. From an elevated tank the diesel oil passes through a fuel filter towards the high pressure fuel pump. The atomiser drain- or return pipe is also connected to the fuel tank. If the tank is not placed in an elevated position, the engine is equipped with a low-pressure fuel pump. It is important that the filter, transfer pumps and lines can be easily bled. It is also advisable to fit a water tap at the bottom of the fuel filter.

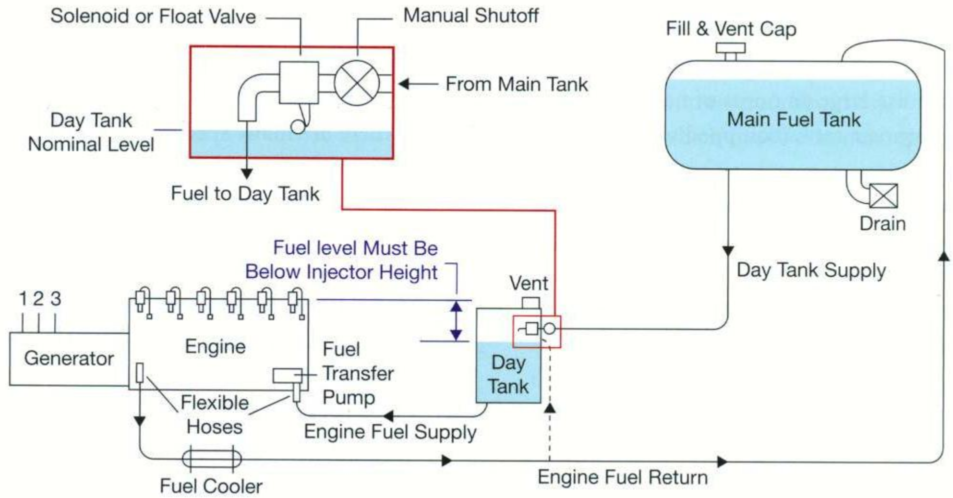
Examples



1. AFC Cavity Drain
2. Fuel Rail Pressure Line
3. Injector
4. Injector Fuel Drain Return
5. Fuel Return to Tank
6. Fuel Tank Breather
7. Fuel Inlet Supply
8. Fuel Filter
9. Gear Pump Coolant Drain
10. Fuel Pump
11. Tachometer Drive

◄ Fuel system of a Cummins NTA 855 Big Cam III diesel engine.

This is an engine with a fuel injector mechanically driven from the cam shaft in the cylinder head. This fuel injector is a combination high-pressure fuel pump and a fuel atomiser. The storage tank can be placed anywhere and is often positioned lower than the engine. The tank is equipped with a filler cap, a vent 6; a suction line 7 and a return line 5. An engine driven low pressure fuel pump 10 draws fuel from the storage tank via a filter 8 to the high pressure line on the suction side of the injectors 3. The return fuel of the injectors flows from pipe 5 to the storage tank. Mounted on top of the low-pressure fuel pump is a vent valve 1.



▲

Fuel system of a Caterpillar-diesel power unit, general.

This consists of an elevated storage tank with a fill-and bleed connection. A drain for water and dirt is fitted at the bottom of the tank. Via an automatic float system or a manual stop valve, the day tank can be maintained at the correct level. From the day tank the fuel flows to the low-pressure fuel suction pump. This provides a certain fuel pre-pressure to the mechanically driven fuel injectors. A suction filter has been placed before the fuel suction pump to protect the pump, after the fuel pump a duplex filter is installed to the clean the fuel. In the engine the supply- and return lines are fitted with a flexible section in order to avoid damage to the fuel lines as a result of vibration .The fuel return line has an integrated fuel cooler.

8.11.2 Category, II 100 to 5000 kW, four-stroke, high speed, fuel M.D.O.

Engines can be equipped with either a high-pressure fuel pump for each cylinder or one block fuel pump. This also applies to category I engines. These larger engines usually have several clean fuel tanks. Two duplex section filters are fitted in order to avoid problems with blocked filters, which could cause the engine to stall. The atomiser drain-or return pipe must be connected to tank being used.

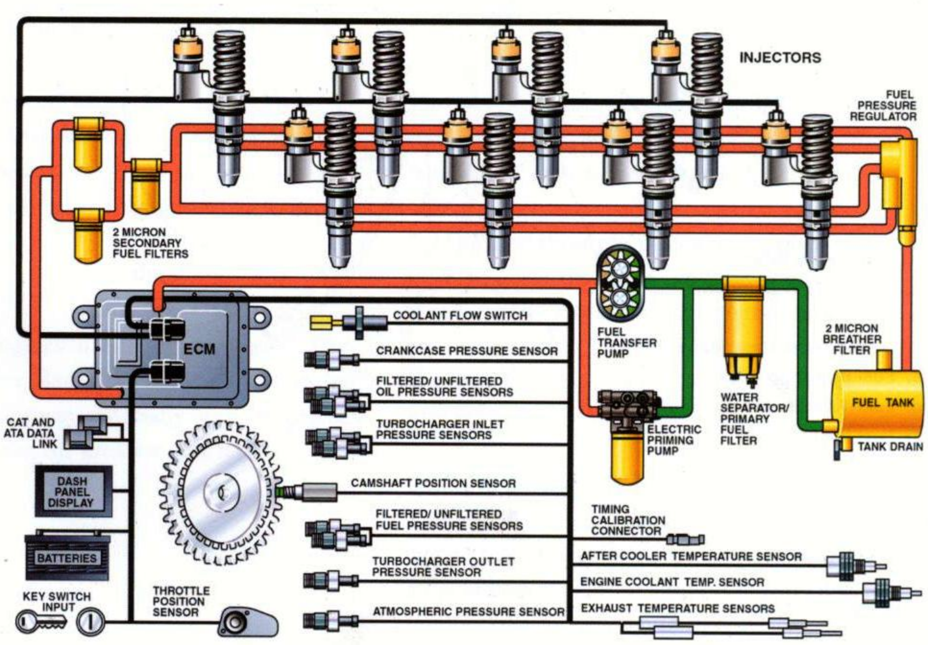
Examples

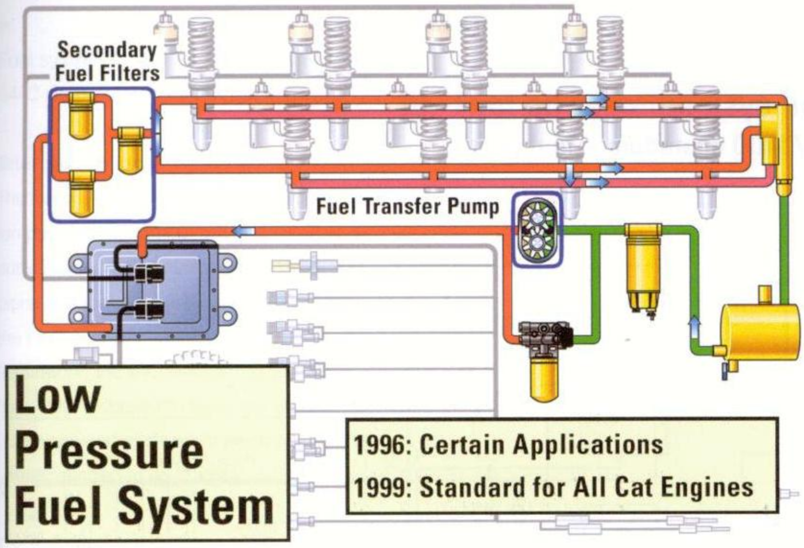
Some larger engines in this category which have a power output easily exceeding 1000 kW are generally equipped with a fuel separator which cleans the diesel oil pumped from the fuel tanks prior to storage in the clean fuel tank or the daily supply tank.

►

A fuel system for diesel oil of a Caterpillar, Series 3500 B MEUI.

The diagram is self-explanatory. The green section of the line has no pressure build-up. The pressure governor above the fuel tank by-passes the fuel to the storage tank at high pressure. The suction line for both low-pressure fuel suction pumps is located to the left of the tank. The filter in the suction line protects the pumps. Sometimes a water separator is also placed here. The grey housing left in the drawing is called the 'Engine Control Module' (E.C.M.) and controls all the important engine functions.





◄ The low-pressure fuel system of a Caterpillar diesel engine, category II.

◄ Fuel separator.

If required, engines running on diesel oil are equipped with a small separator, where dirt and water are removed. This separator or centrifuge is located between the fuel storage tank and the fuel day tank.

8.11.3 Category III, 500 to 30,000 kW, four-stroke, medium-speed, fuel H.F.O.

Engines in this category use very elaborate fuel systems. The heavy oil must be cleaned and attain the correct pressure and temperature before it reaches the high pressure fuel pipes.

Examples

Fuel systems with heavy fuel

When using heavy oil it is of the utmost importance that the fuel temperatures during the various stages of the pre-treatment are read, such as:

- the sedimentation- or settling tanks ± 60 to 80 °C
- the separator ± 95 to 98 °C
- the fine filter to the engine ±125 to 135 °C (max. 150 °C)

▶

Fuel system with diesel oil.
Shown, an example of the system of a Caterpillar–Mak 43 – C propulsion diesel engine.

Brief description.

Diesel oil can be pumped from storage tank DT 4 to the diesel oil tank DT 1 via the suction filter DF 3 and diesel oil trim pump DP 3.

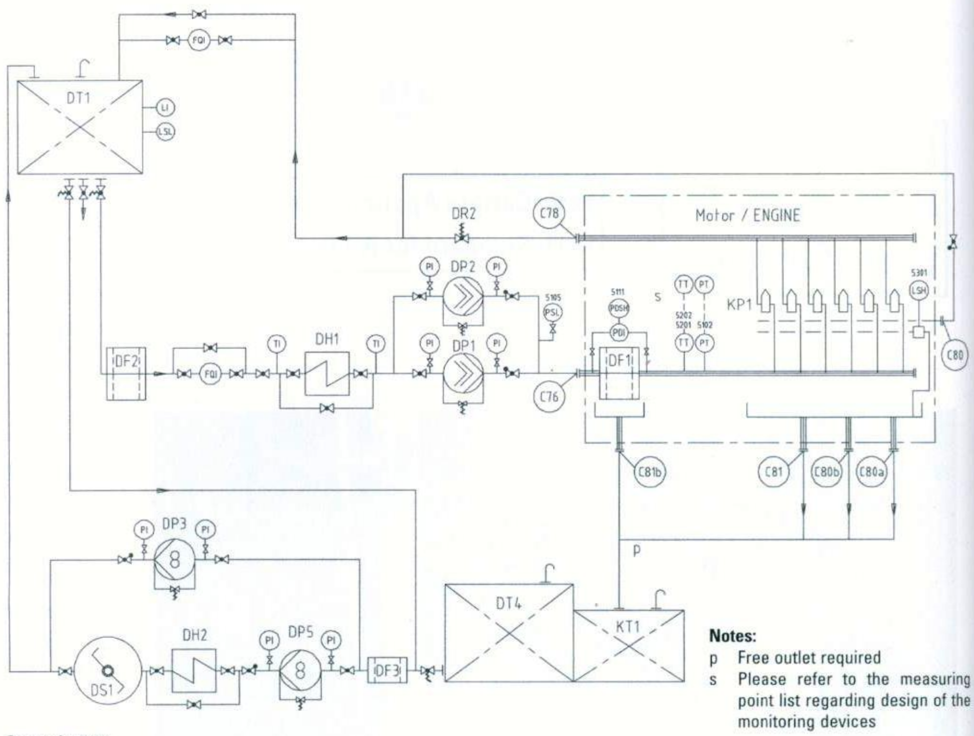
Additionally, from the storage tank DT 4, diesel oil can be pumped via an electric heater DH 2 using the diesel oil trim pump DP 5 to the diesel oil separator, where additional cleansing (water and dirt) occurs. The fuel is then pumped to the diesel day tank DT 1,

From the elevated diesel oil tank, the diesel oil flows through the duplex filter DF 2, the flow meter FQ 1 and the diesel oil pre heater DH 1. Following, one of the two diesel oil feed pumps pump the fuel through a very fine-filter to the high-pressure fuel pumps KP 1. Spilled fuel oil is collected in the spillage collection fuel oil tank KT 1. This fuel flows back into the day tank DT. The high-pressure fuel pump return line feeds the day tank DT via a flow meter FQ 1.

The difference between both flow meters is the actual fuel consumption of the engine. After the high-pressure fuel pumps, the fuel pressure is set using the adjustable valve DR 2. As there are two pre heaters and a day tank, which contains a heating element, the fuel can be maintained at the correct temperature in cold conditions. In this manner the injected fuel viscosity is acceptable, which in this engine running on diesel oil is a maximum of 14 cSt.

MaK

10. Fuel oil system
Gas oil / MDO operation



General notes:
For location, dimensions and design (e. g. flexible connection) of the disconnecting points see engine installation drawing.
DH1 not required with: - Gas oil ≤ 7 cSt/40°
- heated diesel oil day tank DT1

Accessories and fittings:

DF1	Fuel fine filter (duplex filter)	KP1	Fuel injection pump
DF2	Fuel primary filter (duplex filter)	KT1	Drip fuel tank
DF3	Fuel coarse filter	FQ1	Flow quantity indicator
DH1	Diesel oil preheater	LI	Level indicator
DH2	Electrical preheater for diesel oil (separator)	LSH	Level switch high
DP1	Diesel oil feed pump	LSL	Level switch low
DP2	Diesel oil stand-by feed pump	PDI	Diff. pressure indicator
DP3	Diesel oil transfer pump (to day tank)	PDSH	Diff. pressure switch high
DP5	Diesel oil transfer pump (separator)	PI	Pressure indicator
DR2	Fuel pressure regulating valve	PSL	Pressure switch low
DS1	Diesel oil separator	PT	Pressure transmitter
DT1	Diesel oil day tank	TI	Temperature indicator
DT4	Diesel oil storage tank	TT	Temperature transmitter (PT 100)

Connecting points:

C76	Inlet duplex filter	C80b	Drip fuel (sealing oil injection pump)
C78	Fuel outlet	C81	Drip fuel
C80	Drip fuel	C81b	Drip fuel (filter pan)
C80a	Drip fuel (injection pump)		

▶

Fuel system for H.F.O. Caterpillar–MAK 34-C propulsion diesel engine.

Brief description.

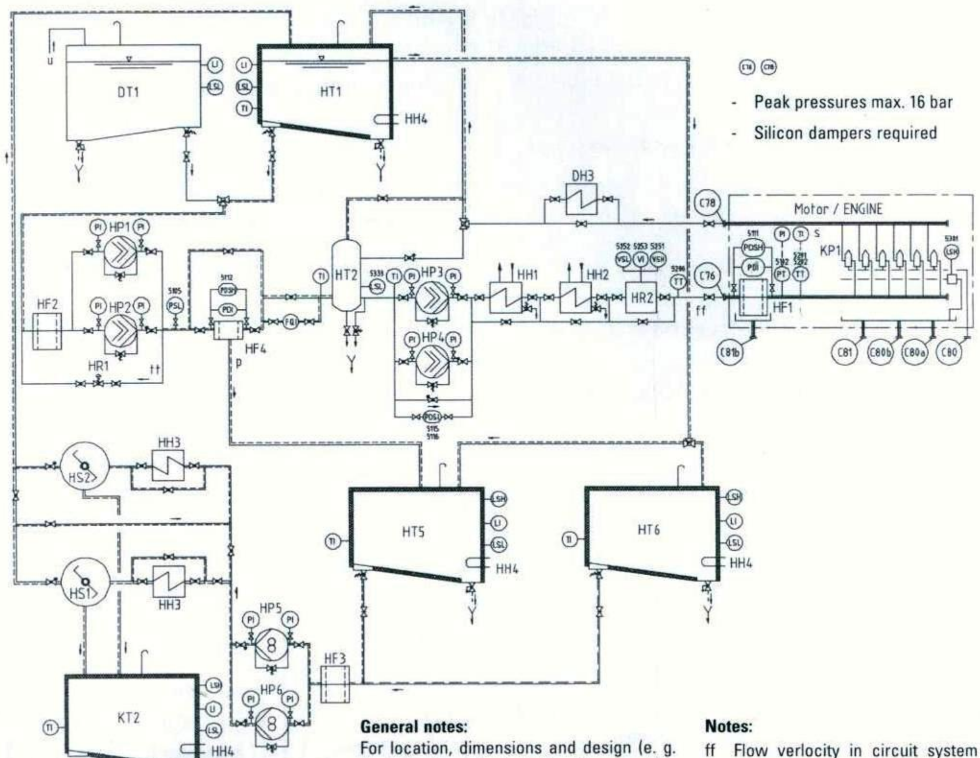
The fuel must be maintained at the correct temperature throughout the system. This entails adequate insulation of the tanks and pipes and that pre heaters are mounted in the fuel tanks trim section, the fuel separators, the fine filters and in the day tank. Moreover, some pipes are heated with warm water, steam or electricity. This is usually referred to as 'tracing'.

The fuel trim pump HP 5 or HP 6 can draw fuel up from the settling tanks HT 5 and HT 6. In these tanks, it will take at least 24 hours for the heavy oil to achieve the correct temperature and separate from the water and dirt. This sinks and can be drained (water). Via heater HH 3 the heavy oil is cleaned in separators HS 1 and HS 2. The waste product, the sludge, is stored in the sludge tank KT 2. The heavy oil is subsequently pumped from the separator to the heated day tank HT 1 by its own pump. From the day tank the fuel passes first through the filter HF2 and is then forced through a self cleaning automatic fine filter HF 4 by feed pump HPO 1 or HP 2. Via a flow meter FQ 1 the fuel flows through a mixing tank GT 2 to the circulation pumps HPO 3 and HP 4. Both pre heaters HH 1 and HH 2 ensure that the fuel reaches the correct temperature; this is controlled by the viscosity gauge HR 2. Via a final Duplex fine filter HF 1 the fuel flows to the high-pressure fuel pumps. At extremely low fuel consumption the cooler DH 3 can ensure that the maximum fuel temperature in the day tank HT 1 is not exceeded.

In fine filters difference gauges indicate if a filter is blocked. Fine-filter HF 4 is automatically flushed clean when certain pressure differences are exceeded. The mixing tank HT 2 ensures an even transition from diesel oil to heavy oil and vice versa.

MAK

10. Fuel oil system
Heavy fuel operation



Accessories and fittings:

DH3	Gas oil cooler
DT1	Diesel oil day tank
HF1	Fine filter (duplex filter)
HF2	Primary filter
HF3	Coarse filter
HF4	Self cleaning fuel filter
HH1	Heavy fuel final preheater
HH2	Stand-by final preheater
HH3	Heavy fuel preheater (separator)
HH4	Heating coil
HP1/HP2	Pressure pump
HP3/HP4	Circulating Pump
HP5/HP6	Heavy fuel transfer pump (separator)
HR1	Pressure regulating valve
HR2	Viscometer
HS1/HS2	Heavy fuel separator
HT1	Heavy fuel day tank
HT2	Mixing tank
HT5/HT6	Settling tank

General notes:

For location, dimensions and design (e. g. flexible connection) of the disconnecting points see engine installation drawing. Valve fittings with loose cone are not accepted in the admission and return lines.

KP1	Injection pump
KT2	Sludge tank
FQ1	Flow quantity indicator
LI	Level indicator
LSH	Level switch high
LSL	Level switch low
PDI	Diff. pressure indicator
PDSH	Diff. pressure switch high
PDSL	Diff. pressure switch low
PI	Pressure indicator
PSL	Pressure switch low
PT	Pressure transmitter
TI	Temperature indicator
TT	Temperature transmitter (PT 100)
VI	Viscosity indicator
VSH	Viscosity Control switch high
VSL	Viscosity Control switch low

Notes:

ff	Flow verlocity in circuit system ≤ 0,5 m/s
p	Free outlet required
s	Please refer to the measuring point list regarding design of the monitoring devices
tt	not insulated nor heated pipe
u	From diesel oil separator or diesel oil transfer pump

All heavy fuel pipes have to be insulated.

---- heated pipe

Connecting points:

C76	Inlet duplex filter
C78	Fuel outlet
C80	Drip fuel
C80a	Drip fuel (injection pump)
C80b	Drip fuel (sealing oil injection pump)
C81	Drip fuel connection
C81b	Drip fuel (filter pan)

8.11.4 Category IV, 1500 to 100,000 kW, two-stroke, low-speed, fuel H.F.O.

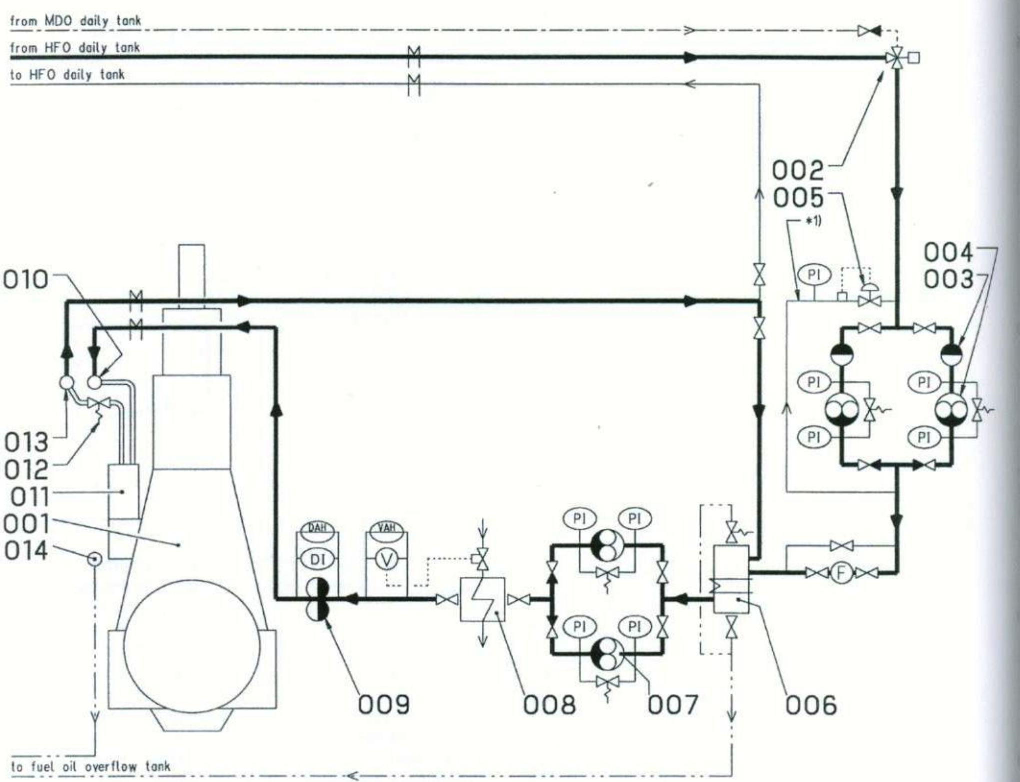
These resemble the system described above in many ways.

Example

The fuel systems of a Wärtsilä Sulzer RTA 48 T and a RTA 58 T are used as an example.

▶ Description of a complete fuel treatment system.

Via a three way valve 002, the diesel oil or heavy oil can be pumped from the day tanks with the low-pressure feed pump 004. Suction filter 003 protects these pumps. Pressure regulating valve 005 ensures that the correct pressure is found in the fuel lines leading to the mixing tank 006, where the engine's fuel return pipe exits. The high-pressure circulation pumps 007 press the fuel through the final pre heater 008, which, again, is controlled by the viscosity gauge V, via a flow meter. Subsequently, the fuel travels through a duplex fine filter to the high-pressure fuel pumps 011 of the engine. A control valve 012, also referred to as the pressure retaining valve, ensures that the fuel pressure in the suction line 010 of the high-pressure fuel pumps is correct. Drain pipe 013 leads back to the mixing tank 006.



001 Main engine
002 Three-way valve, manually or remotely operated
003 Suction filter, heated (trace heating acceptable)
004 Low pressure feed pump
005 Pressure regulating valve
006 Mixing unit, heated and insulated
007 High pressure booster pump
008 Fuel oil endheater
009 Fuel oil filter, heated (trace heating acceptable)
010 Fuel oil inlet
011 Fuel injection pump
012 Pressure retaining valve
013 Fuel oil outlet
014 Fuel oil leakage from fuel pump

Remarks:

*1) The return pipe may also be led to the HFO daily tank.
– Air vent and drain pipes must be fully functional at all inclination angles of the ship at which the engine must be operational.

HFO pipes, heated and insulated
MDO pipes
Drain & overflow pipes
Piping on engine / pipe connection

8.12 Modular fuel-treatment systems

Today complete modules mounted on frames that are connected directly to the fuel lines, alarm systems and electricity grid are delivered to ships as well as diesel power plants.
The modules are mounted, tested and inspected in the engine factory and subsequently installed on board. The advantages are short delivery times and smaller fuel treatment systems.
These fuel treatment sets are usually called 'booster units'. Worldwide there are a dozen leading companies which specialise in these systems.

8.12.1 Standard form for fuel treatment system

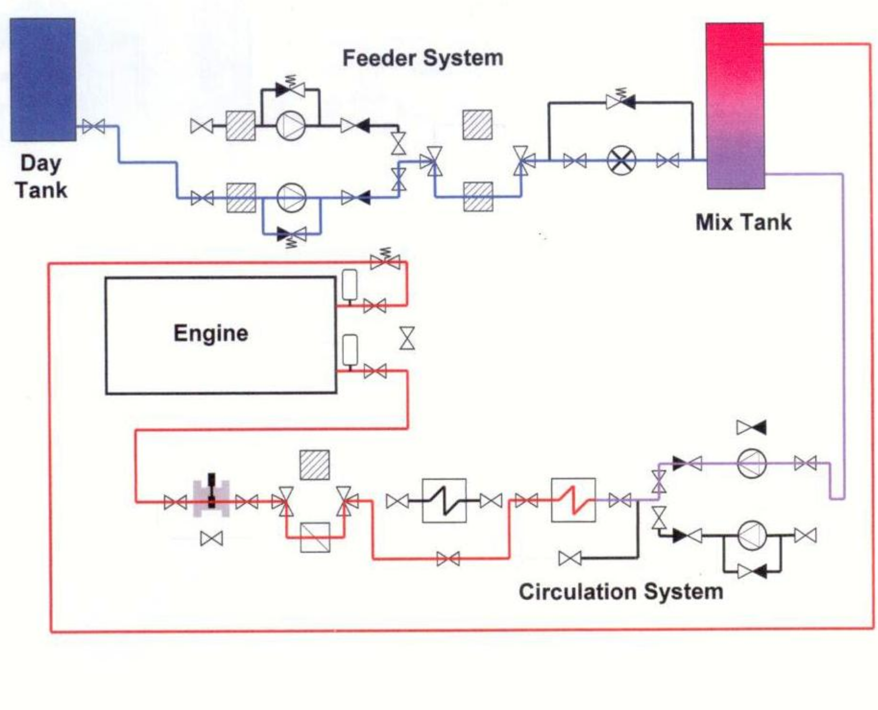
Operating mode

- One supply pump.
- One circulation pump.
- One pre-heater.
- One filter of the duplex filter where the off-line filter is clean.
- The fine filter is in automatic reflush mode.
- The viscosity sensor controls the fuel temperature via the supply valve using steam or thermal oil and regulates the pre-set viscosity.
- Shock absorbers have been fitted in order to level off the pressure impact of the high pressure fuel pumps in the suction line.
- The pressure in the suction line of the high pressure fuel pumps must be checked at regular intervals.

▲

A modular fuel treatment system by Alfa Laval.

The complete installation is mounted on a frame and can be installed between the day tanks and the diesel engines on location.



►

A standard flow diagram for a fuel treatment system.

One of the feed pumps draws fuel up from the clean fuel day tank via a suction filter and pumps it to the circulation pumps via a mixing tank. A single feed pump always has a larger capacity than the maximum fuel consumption of the diesel engine. A fraction of the pumped fuel flows back into the suction pipe via an adjustable by pass valve.

The fuel then passes through an automatic fine filter, a flow meter and is subsequently pumped through a pre heater and a viscosity gauge by one of the two circulation pumps. The fuel reaches the suction pipe of the high-pressure fuel pumps via a duplex fine filter. The pressure in the suction pipe is such that the fuel can not boil. Gas pockets can cause serious damage to the fuel pumps. This pressure can be set by means of a spring loaded overflow valve where the excess fuel flows back into the mixing tank. The mixing tank ensures an even fuel temperature before the circulation pumps. Furthermore, on top of the tank an automatic vent is fitted. The capacity of each circulation pump is considerably higher than the maximum fuel consumption of the engine.

The viscosity sensor is placed in the full fuel flow just before the engine. In order to maintain the viscosity set on the controller, a control valve regulates the steam or thermal oil flow to the fuel pre-heater.
During normal operation both viscosense by-pass valves and the control valve must remain shut. Only during repairs or damage to the sensor or the control valve is the viscosity regulated by the by-pass valve. When the control valve is fully opened while the pre-set viscosity has not yet been achieved it is possible that the steam pressure and/or the temperature will remain too low. If thermal oil is used for heating, only the temperature will be too low.

Operating principle of viscosity control, specifically, the VAF Instrument 'ViscoSense'

The sensor consists of a stainless steel pendulum which is attached to a base plate by a torsion tube. A pair of piezo-elements is integrated into the pendulum and they are actuated by an alternating signal thus causing the pendulum to move by torsional vibration. A second pair of piezo-elements, (receiver piezo-elements), measures the torsional movement and sends this signal to the processor which also ensures that the first pair of piezo-elements provides a continual torsional vibration.

Special details for the standard form

Data

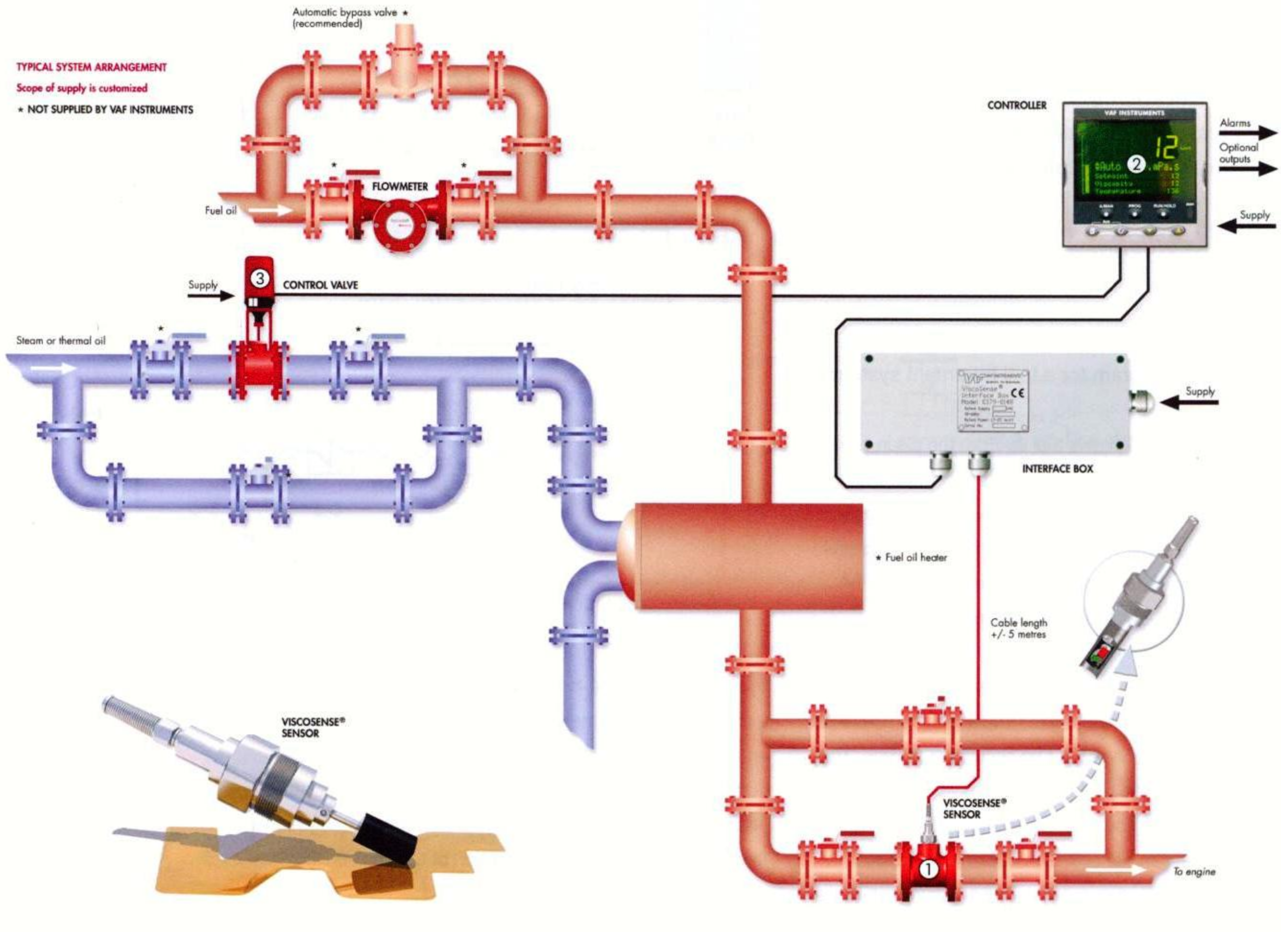
Fuel velocity in booster unit	0.6 to 0.8 m/sec.
Circulation factor supply pumps	1.5 to 2.0
Circulation factor circulation pumps	1.5 to 4.0

$$\text{circulation factor} = \frac{\text{max. capacity pump}}{\text{max. engine fuel consumption}}$$

Fuel pressure of the engine (suction line) is between 4 and 8 bars depending on the engine type and the fuel used.
Maximum fuel temperature of the engine is between 125 and 150 °C, dependent on the engine type and the fuel used.

Location of the automatic fine filter

The most favourable location is the 'cold part' of the installation directly behind the booster pumps. Here the liquid flow is much lower than past the circulation pumps; only the used fuel is filtered, this allows a considerably smaller filter (smaller filter capacity is sufficient).
The fuel is more stable at these lower temperatures; there is no separation of hydrocarbon compounds.



▼
Fuel viscosity indicator. Shown, a system by VAF-Instruments with the ViscoSense system.

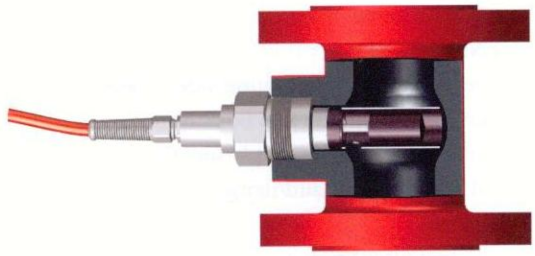
1. *The ViscoSense sensor. This measures fuel viscosity.*
2. *The ViscoSense sensor is connected to an electronic control unit which sets and controls the viscosity.*
3. *A control valve is regulated by a control unit which deals with the steam or thermal oil flow to a fuel pre heater.*

Regulations with regard to fuel systems

These are clearly described in the 'Project guide' of the engine concerned. Engine manufacturers require diesel engines with a certain fuel consumption, a certain viscosity and a certain circulation factor.
Furthermore, at full load of the diesel engine the fuel supply pressure and the fuel discharge pressure (regulated with the retaining valve or the overflow valve) require a certain value.

Viscosity sensor

The viscosity sensor is always set at the desired viscosity after which the fuel temperature is adjusted accordingly. The correct viscosity, not the correct temperature, is important. Therefore: do not adjust the temperature.
One fuel pre-heater with a shut by-pass valve should suffice in maintaining the right fuel temperature. A processor measures the phase difference between the sent and received signal. This phase difference is processed and provides the vibration damping value which is proportional to the viscosity of the liquid.



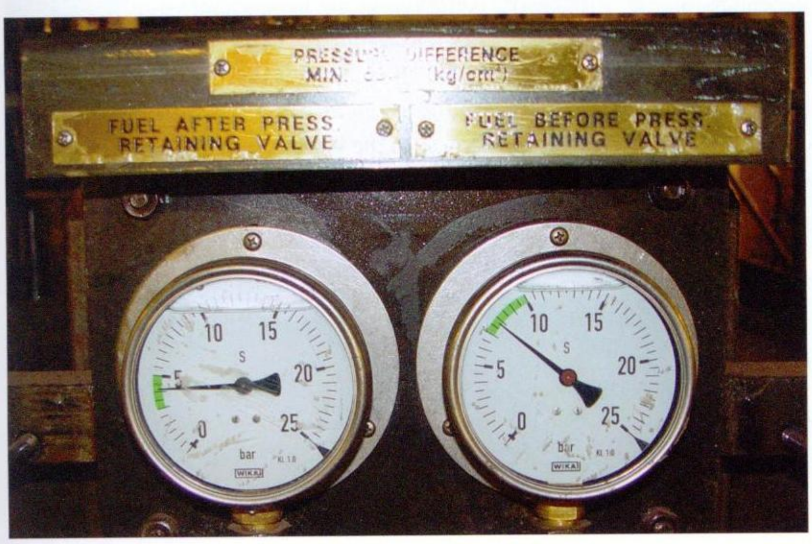
The ViscoSense sensor; this measures the fuel viscosity and temperature.

An automatic fine fuel filter in the low-pressure heavy fuel system.

The filter is cleaned by automatically back flushing the fine filter elements. This occurs automatically at a certain pressure difference in the filter.

A complete fuel treatment system with:

1 *steam control valve*
2 *fuel pre heaters*
3 *circulation pumps*
4 *mixing tank*



A retaining valve measurement in a large two-stroke crosshead engine. Also known as pressure difference.

The pressure difference between the supply low-pressure fuel line to the high-pressure fuel pumps and the fuel return pipe to the fuel treatment. This pressure difference should not exceed a certain value. If the fuel treatment unit does not supply sufficient fuel, the fuel pressure will drop below the green sector, preventing the engine from achieving the required RPM's.

The Project guide also often provides the specifications of the:
- fuel velocity at the suction side of the pumps;
- fuel velocity in the fuel booster unit;
- fuel velocity in the supply line to the engine;
- all the pressures and temperatures in the system;
- the fuel viscosity between the minimum and maximum value.

8.12.2 General comments with regard to 'booster units'

- **Ship propulsion plants often have** loads that are much lower than those of diesel power plants.
- **A propulsion engine** usually operates at **± 85%** of its capacity due to the seas, currents and draughts.
- **A diesel power plant engine** usually runs at the maximum allowed operating speed (Maximum continuous Rating or M.C.R.) of **100%** and is therefore has a heavier but more consistent loading.
- **Many two-stroke propulsion engines** have a booster system developed by the ship yard and it is normally built on a false floor in the engine room. Sizeable systems with few problems.
- **Some two-stroke propulsion engines** have a compact booster unit manufactured by a third party and delivered to the ship yard. Comparatively, one expects more problems. However, the components are often much smaller.
- **For four-stroke propulsion engines** compact booster units are generally placed with few problems.
- **Fuel consumption for four-stroke auxiliary engines on large sea-going vessels** can vary greatly and can range, depending on the electricity demand for, for instance, the number of reefers, bow thruster manoeuvring etc. from 8 to 100%. The control valve of the heater is often fully closed at such a low load and the fuel flow extremely low. However, the fuel viscosity is usually stable.
- **For container ships** the number of refrigerated containers ('reefers') is important in order to determine the load of the auxiliary engines. It is also important to pay attention to the main engine driven shaft generators. A large shaft generator has an electric power output of 2.5 to 3.5 MW and often 'replaces' one of the four auxiliary engines.
- **Main engine type:** The MAN–B&W crosshead engines are not usually equipped with mixing tanks and only have a raised tank venting system. MAN–B&W four-stroke auxiliary engines are also often equipped with this kind of system.
- **Make.** There are approximately fifteen booster manufacturers, five of whom are prominent. These are, Alfa Laval, Aura Marine, Kupke and Wolf, Westfalia and Eefting. All the sets are fairly similar. The customer plays an important role as they stipulate their specific requirements and will pay for these.
- **Most engine manufacturers** have regulations with regard to switching from **H.F.O. to M.D.O..** Most of them always operate their engines on H.F.O., also when they are stationary. Others change to M.D.O. under certain conditions, for instance, under a certain load or during repairs.
  Most fuels used are not heavier than **IF 380,** because heavier fuel does not offer any financial advantages. It is only just cheaper to purchase.
- The so-called pressure compensators are often not in place. This causes an enormous pressure pulse in the piping and consequently in the booster units.

8.12.3 Points of interest for booster units on location

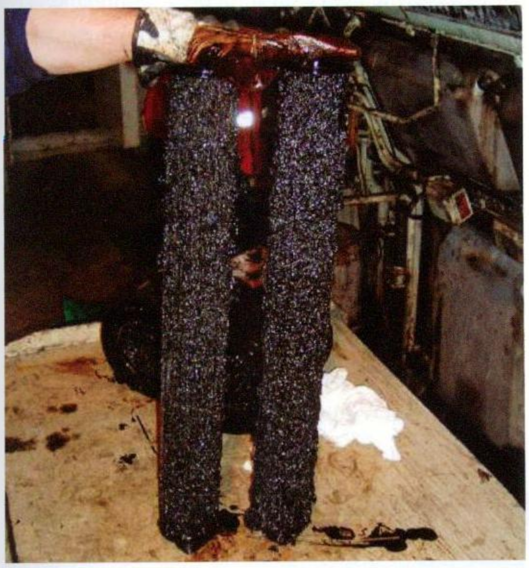
1. Does the viscosense work properly; including the software?
2. Ensure that the viscosity is regulated as opposed to the temperature. People often erroneously think that the temperature is more important than viscosity!
3. Is there one heater! Two are not normally required!
4. Is the viscosense bypass shut?
5. For four-stroke auxiliary engines on large ships. The fuel temperature of (both) auxiliary engines often diverges. The fuel temperature closest to the booster unit is often higher than the fuel temperature further away from the booster unit (usually the other side of the ship).
   The difference is often 5 °C or more. This can generally only be avoided by placing a separate viscosense/heater near the engine 'furthest away'.
6. Is there a quick conversion from M.D.O. to H.F.O. and vice versa?! The viscosense will react immediately!

7 Does one not resort too quickly to adjusting the temperature when the viscosity does not reach the correct value fast enough?
8 Does one use the correct fuel? Sampling is important!

8.12.4 Plastics in Heavy fuel oil

The presence of plastics in cleaned heavy oil is a common occurrence.
These plastics are predominantly:

polypropylene	– melting point approximately 120 °C
polystyrene atyrene monomer	– melting point approximately 100 °C



◄ Heavily contaminated fine fuel filters.

If filters are placed in a system where the fuel temperature is higher than the melting point of these contaminants, it is possible that the filter will clog up. The temperature of the duplex fine filter just before the diesel engine ranges from 125 to 150 °C and is therefore prone to blockage when these substances are present.

8.13 Bunkering fuels

This does not only apply to shipping but also to diesel power plants.
One can generally say that much attention should be paid to fuel intake, among other things:

- to the amount of fuel;
- to the available storage capacity;
- to the manner in which fuel is taken in and the supplier's capacity;
- in case of large quantities the sampling procedure;
- to prevent environmental pollution due to leakage or tank overflow..

It is self-evident that 'filling up' 200 litres of diesel oil for a small motor yacht is much easier than 'bunkering' 8000 tons of heavy oil for a large container ship.
Especially in the second example it is imperative that the entire procedure is meticulously followed.
In particular the quality of the fuel delivered, is often a moot point.

◄ A bunker boat alongside a container ship for inland shipping.

A practical way to refuel swiftly and efficiently. Operation does not have to cease for refuelling, so no time is wasted. Moreover, there are often no suitable berthing facilities.

8.13.1 Points of interest for heavy fuel

Mixing two types of fuels that were bunkered at different times should be avoided if possible. Before bunkering, the 'old' fuel should preferably be stored in tanks that are filled to capacity so the addition of 'new' fuel can be bunkered as the storage capacity is full. The 'old' fuel should be used first. If the fuel remains in the storage tanks too long it can separate into light and heavier components.

When fuels are mixed asphaltenes can separate out of the fuel and produce sludge. One is generally advised not to mix heavy fuel oil with gas oil or diesel oil on location to reduce the heavy oil viscosity. It is difficult to assess the consequences of such mixing on the viscosity and also any other potential side effects.

Volatile heavy oil or heavy oil mixed with lighter fuels often causes the following problems:
- sludge forming in storage- and sedimentation tanks;
- excessive sludge excretion by the centrifuges;
- rapid clogging of filters.

8.13.2 Cleaning of volatile fuels with centrifuges

Measures
- the centrifuge should be set at the lowest possible capacity.
- avoid multiple centrifugation of heavy oil.
- centrifuges should be cleaned or shot every fifteen minutes.
- ensure that the highest fuel temperature is achieved in the centrifuge, namely 98 °C.

8.13.3 Heating double bottomed tanks on seagoing vessels

By heating the heavy oil to make it pumpable, the temperature in the cargo holds above the fuel tanks rises, which may cause problems for certain types of cargo.
The temperature of the cargo containers in these holds may rise as well.
Depending on the type of cargo that is exposed to this temperature increase, one usually maintains a maximum hold temperature of 40 °C.
In the tropics, where many reefers are in operation, this temperature is easily reached in the hold. It is advisable to temporarily reduce the tank heating when temperatures exceed 40 °C.

▶ A completely soiled disc stack package in a heavy oil separator.

▶ A severely damaged the separator house as a result of the imbalanced overly full separator bowl.

It is absolutely necessary to regularly discharge the sludge in the separator. The backwash frequency is dependent on the amount of dirt, which in this case, is removed from the fuel.

▲ Heavy oil separators in the machine room of a large container ship.

They are crucial in cleaning the fuel and the lubricating oil for engines using heavy oil as fuel.

8.13.4 Fuel oil additives

There is an enormous amount of additives available which are used for:

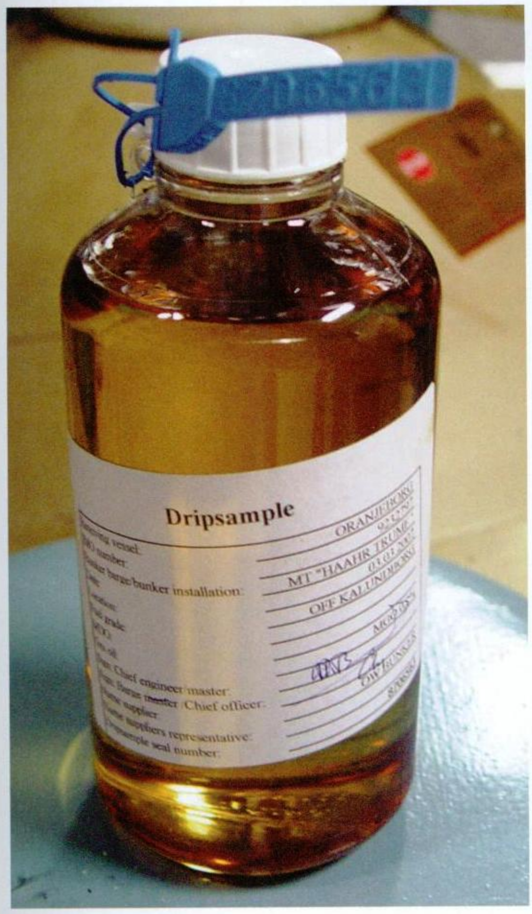
- preventing sludge and sludge forming in fuel tanks;
- separating water from fuel;
- improving combustion;
- increasing the melting point of vanadium;
- preventing sedimentation in the engine, the exhaust gas turbines and the exhaust gases boilers;
- preventing high- and low temperature corrosion.

There are no general rules for the use of dopes. Some ship yards and owners of diesel power plants use one or multiple additives and give positive feedback.

▼

A 'drip sample' of Marine Diesel Oil (M.D.O.).

Check the label detailing all required information and the tamper-proof seal.



▼

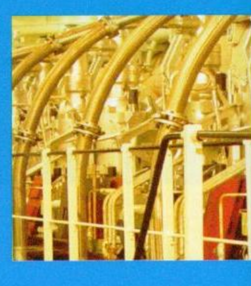
Two 'drip samples' of Heavy Fuel Oil (H.F.O.).

In this instance 380 CSt with a low sulphur content.

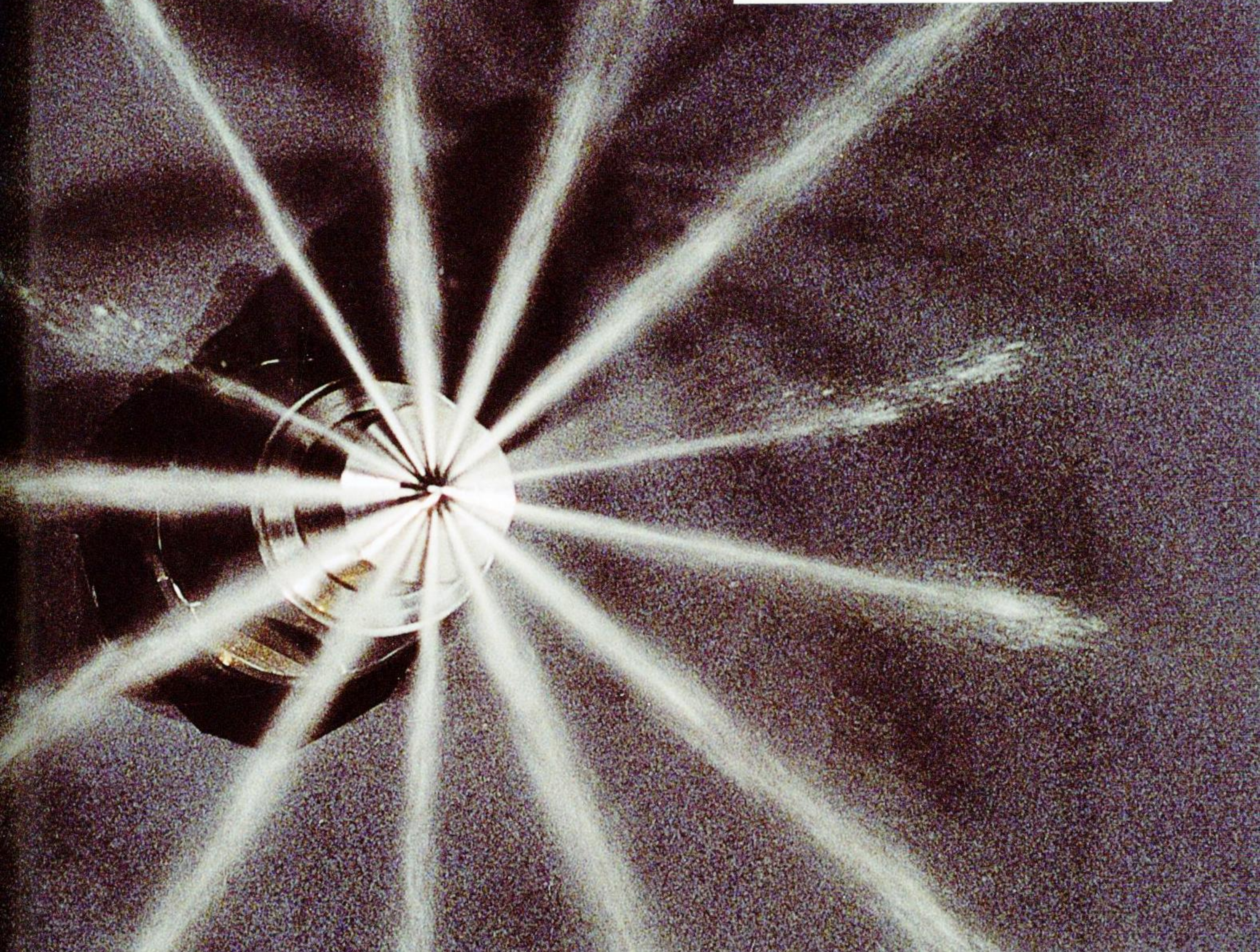
>CH 9

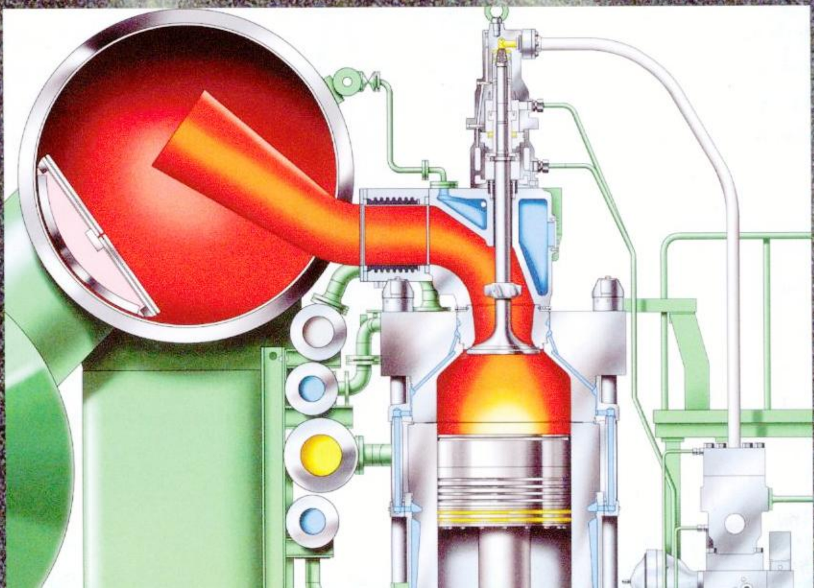
Fuel-injection systems



The most important factor in the combustion process in a diesel engine cylinder is an optimum fuel injection. The fuel injection velocity, the droplet size of the fuel spray and the compressed air temperature to which the droplets are exposed are crucial for optimum combustion.





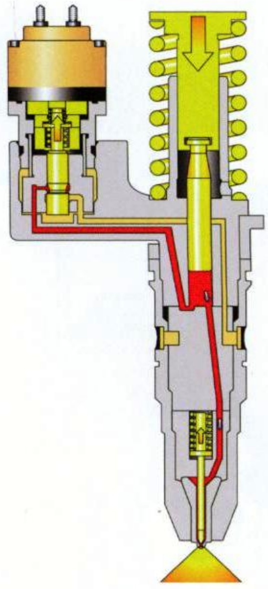
The injected fuel must fully combust in the combustion space within a fraction of a second.

9.1 Introduction

The fuel injection process has been the principal focal point from the time of the first working diesel engine. Swift and perfect combustion of the fuel is absolutely imperative for the diesel engine to work optimally.
In addition, low specific fuel consumption and a limited emission of pollutants in the exhaust gases have become increasingly important.
One must also bear in mind that the materials used in the diesel engine components manufacture experience an intensification of the mechanical, thermal and chemical load.
Especially the rising pressures in the cylinder exert very strong forces in the engine.
The fuel injection systems have continually been adjusted, modified and at times essentially re-designed over the last one hundred years that the diesel engine has been in use.
One of the most recent developments is the so-called 'common rail' injection system for four-stroke engines and the 'flexible' injection system for two-stroke crosshead engines.
These electronically controlled injection techniques have been applied to small diesel engines for a dozen years in mainly the automotive industry; motor cars and lorries.

However, the basics of these injection techniques are the same; in a short time span, they accurately regulate the amount of fuel that is injected into the cylinder in such a way that a very rapid and perfect combustion follows.

The combustion process in a diesel engine occurs in a fraction of a second.



► Fuel injectors ensure that fuel at the appropriate moment, in a very brief time span, as a finely distributed spray is injected into the combustion chamber.

9.2 Examples of injection times

9.2.1 A small high-speed four-stroke diesel engine - Engine Category I

RPM	3000
power stroke	140 crank degrees
fuel injection	30 crank degrees

Per second so 50 × 360 = 18,000 crank degrees.
Power stroke takes 140 crank degrees.

In time = $\frac{140}{18,000}$ = 0.007 seconds.

Injection time 30 crank degrees; in time: $\frac{30}{18,000}$ = 0.0016 seconds.

Number of processes per second: $\frac{18,000}{720}$ = 25.

Number of injections consequently is 25 per second.

9.2.2 A high-speed larger four-stroke diesel engine - Engine category II

RPM	1500
power stroke	130 crank degrees
fuel injection	30 crank degrees

Revolutions per second: $\frac{1500}{60}$ = 25 rev /sec.

So per second 25 × 360 = 9000 crank degrees.

Then the power stroke takes: $\frac{130}{9000}$ = 0.014 seconds.

The injection takes: $\frac{30}{9000}$ = 0.0033 seconds.

Number of power strokes: $\frac{9000}{720}$ = 12.5 per seconds.

So number of injections is 12.5 per second.

9.2.3 A medium-speed four-stroke diesel engine -Engine category III

Number of revolutions	600 rev /min.
Power stroke	150 crank degrees
Fuel injection	30 crank degrees

Number of revolutions: $\frac{600}{60}$ = 10 rev/sec.

So per second 10 × 360 = 3,600 crank degrees.

Then the power stroke takes: $\frac{150}{3600}$ = 0.041 seconds.

The injection time takes: $\frac{30}{3600}$ = 0.0083 seconds.

Number of processes per second: $\frac{3600}{720} = 5.$

Therefore number of injections per second is 5.

9.2.4 A low-speed two-stroke crosshead diesel engine – Engine category IV

Number of revolutions	94 rev/min.
Power stroke	120 crank degrees
Fuel injection	25 crank degrees

Number of revolutions: $\frac{94}{60} = 1.566$ rev/sec.

So per second: $1.566 \times 360 = 564$ crank degrees.

Then the power stroke takes: $\frac{120}{564} =$ 0.212 seconds.

Then the injection takes: $\frac{25}{564} = 0.044$ seconds.

Number of power strokes: $\frac{564}{360} = 1.566$ seconds.

Number of injections is 1.566 per second.

Comments

For both high-speed engines (3000 and 1500 rev/min.) the injection times are extremely short (0.0016 and 0.0033 sec.) also the power strokes (0.007 and 0.014 sec.).
For these types of engines only gas oil and diesel oil are considered. Heavier fuels such as heavy oil combust too slowly.
For medium- and low-speed engines, the injection times are short (0.0083 and 0.044 sec.) and the power strokes are considerably longer (0.041 and 0.212 sec.).
These engines can also use a cheaper heavy fuel as the combustion time is much longer. Heavy fuels can be used for four-stroke engines up to a 1000 or sometimes1200 revolutions per minute. For all RPM's above this value, the combustion time is too short and after burning will take place whilst the exhaust valves are open. Not only does this cause power loss, but the exhaust valves can reach a too high temperature and ignite.
Therefore, mixing fuel and air has to occur as quickly as possible and the fuel mixture should be homogeneous in order for the fuel particles to react rapidly with the oxygen.
In order to achieve a swift reaction the mixture has to have a sufficiently high temperature. This usually means a final compression temperature between 650 and 850 °C. This temperature is achieved by compression of the combustion air. The required compression ratio is between 10 and 20, dependent on the engine type. The combustion process of a diesel engine is based on the self-ignition principle. This means that the engine will not work if the fuel is compressed when the final compression temperature is too low. This is mostly the case during a 'cold' start, when the engine is not warm. Therefore many engines are equipped with devices that prevent a cold start such as hot wire filaments (small engines category I) or heating in the coolant system (larger engines).

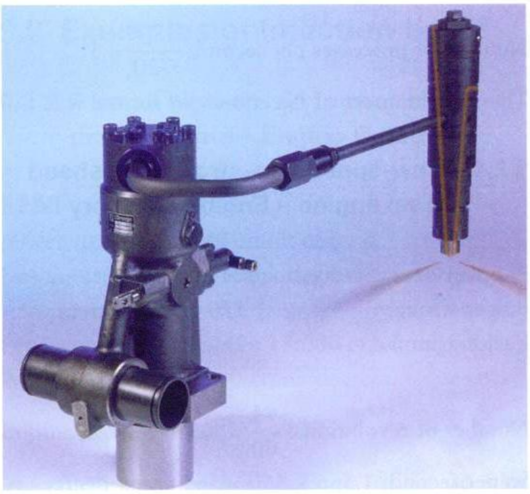
9.3 Ignition delay

In practice the mixing rate of fuel and air is lower than the chemical reaction time during ignition and combustion, which means that there is always a delay: the so-called ignition delay.
The combustion process, of course, will be longer as the number of revolutions decreases, so ensuring that the larger droplets will still fully combust within the limited combustion time. This is why there are very few problems with regard to the combustion time in low-speed two-stroke crosshead engines running on heavy oil. The shorter the combustion time, the greater the potential problems as one sees in for instance medium-speed four-stroke engines with a comparatively high RPM.

9.4 Partial-load conditions

The injection pressures typically drop at a lower RPM. This will negatively affect the combustion process. Therefore, one must take into account these partial load conditions.
Only by means of special mechanical, and today, electronically controlled injection systems can the full load injection pressures be maintained.
Virtually all new diesel engine designs have advanced, electronically controlled injections systems which, among others, achieve lower pollutant emissions in exhaust gases at partial load conditions. Generally, the fuel consumption is also reduced.
At partial load conditions, the final compression pressure- and temperature both drop as the increased compression time allows an increase in the cylinder heat escape. This of course has a negative effect on the ignition- and combustion time.
Medium-speed diesel engines, which usually run on heavy oil, are converted to diesel oil below a 25% load. Increasing the inlet air- and cooling water temperatures also positively affects the final compression temperature.

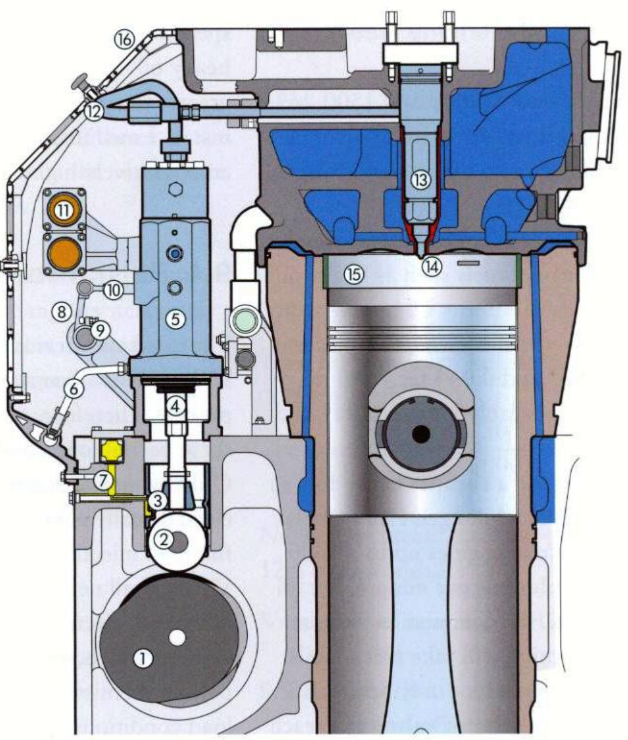
9.5 Processes in the cylinder; injection, ignition and combustion



▲
A high pressure fuel pump driven by the camshaft left, the high pressure fuel line in the middle and the fuel injector right.

The fuel is injected into the cylinder by a cam shaft driven high-pressure fuel pump. The correct amount of fuel is forced into the injector or the fuel valve at a meticulously timed moment. At the commencement of the compression stroke of the pump, the fuel is compressed in the pump cylinder, the high-pressure fuel line and the injector. An injection delay lasts until the moment that the needle valve as a result of the increasing pressure in the injector.

This is therefore a mechanical delay which is dependent on the fuel volume in the pump, the high pressure line and the injector. It is also dependent on the construction of the high-pressure fuel pump and the injector. So this injection delay can not be manipulated in an existing engine.



►
A complete high pressure fuel injection system.

1 *fuel cam on the camshaft*
2 *cam rollers*
3 *cam roll guides*
4 *pump plunger*
5 *fuel pump housing*
6 *fuel return line*
7 *lubricating oil supply to the cam roll guides*
8 *fuel pump-control rod*
9 *central adjusting spindle*
10 *fuel pump gear rack*
11 *fuel pump suction line*
12 *fuel pump high pressure line*
13 *fuel injector*
14 *injector tip*
15 *combustion space with the piston in the top position*
16 *cover over the fuel pumps, the so called 'hot box'*

9.6 The four phases by Ricardo

From the moment that the initial fuel particles are sprayed into the cylinder, the chemical process of ignition and combustion begins. This has been described by various engine builders, among whom the Englishman Ricardo. He put together a four-phase classification.

9.6.1 Phase 1: Ignition delay

The injected fuel droplets are heated by hot compressed air, then vaporise and mix with the hot compressed air, after several thousandths of a second the mixture ignites. The time that elapses between the fuel entering the cylinder and the ignition of the mixture is called the ignition delay.

The ignition delay is affected by:
- the compression pressure-and -temperature;
- the shape of the combustion space at the moment of injection;
- the type of fuel.

The heavier the fuel the longer the ignition delay. For gas oil and diesel oil this delay is very short, just a few thousandths of a second. For heavy fuel, it can be considerably longer.
When the ignition delay is represented by the number of crank degrees, the high-speed four-stroke engines obviously have the greatest value.

9.6.2 Phase II: Uncontrolled combustion

From the moment the fuel is ignited there is a rapid pressure- and temperature increase of the combustion gases in the cylinder. The rate at which the pressure rises in comparison to the crank degrees is called the pressure gradient. Sometimes, too strong a pressure gradient manifests itself as a knocking sound due to the pressure impact on the piston which travels to the driving mechanism.
The injection time in high-speed engines is very brief. This is particularly noticeable during the ignition delay. This is not the case in low-speed engines running on heavy oil. Here a minimum value has been set due to the increasingly poor ignition quality of heavy fuel.

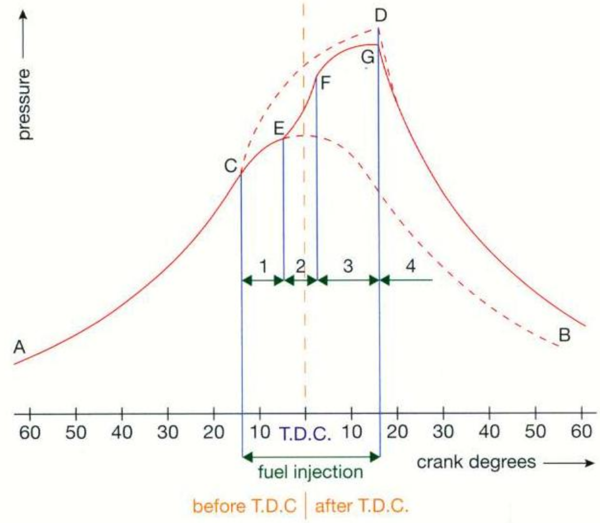
Also see Chapter 8, Fuels, fuel-line systems and fuel cleaning.

9.6.3 Phase III: Partially controlled combustion

After the ignition delay the fuel injection and combustion take place in a relatively controlled manner. The heat development increases synchronously with the fuel supply. The greater the volume of air consumed, the lesser the amount of heat generated.

9.6.4 Phase IV: After burning

The fuel injection is terminated because the effective compression stroke of the fuel pump plunger is ended and the needle valve shuts as a result of the rapidly decreasing pressure in the high pressure pipeline. At this moment the last of the fuel injected combusts. By the time the exhaust valves open all the required fuel must have combusted in order to avoid overheating of the valves and valve seats.

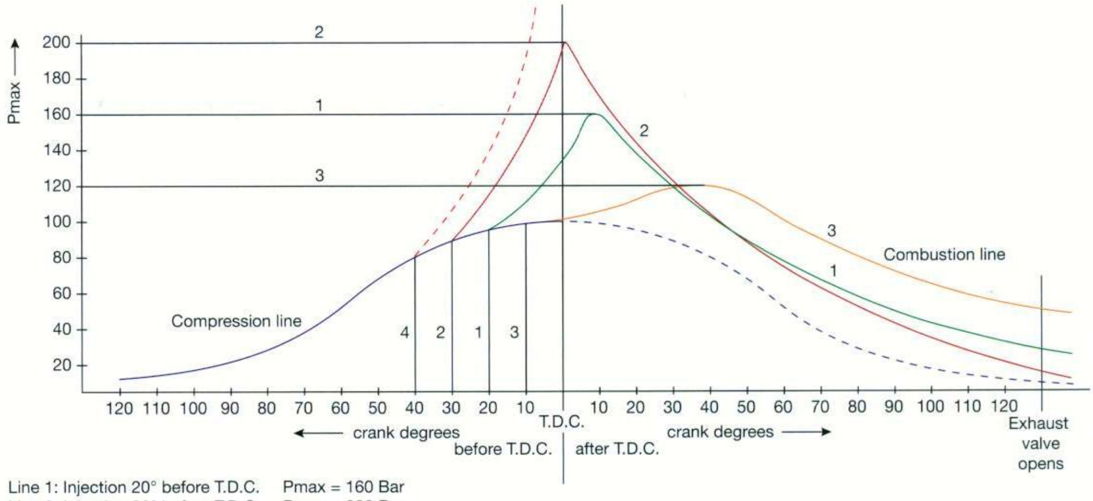


▲
The pressure curve in the cylinder for the duration of the four phases.

Line AB shows the compression pressure without fuel injection.
Line 2, starting at point E, shows the combustion pressure of the fuel injection. The maximum combustion pressure is at point G, just after the top position of the piston. Point B is the time at which the exhaust valve or -valves are open. Most of the energy formed in the combustion process is converted to pressure exerted on the piston.

The four phases of the combustion process are:
1 the ignition delay;
2 the uncontrolled combustion;
3 the partially controlled combustion;
4 the after burning.

The fuel is injected at approximately 15 degrees before top dead centre and terminates at 15 degrees past the top dead centre. Consequently, the entire injection angle constitutes 30 crank degrees. The higher the number of revolutions, the earlier the fuel is injected. The maximum combustion pressure should be achieved just after the T.D.C. of the piston in order to obtain maximum pressure on the piston as it passes T.D.C. . If the maximum combustion pressure is reached too early, for instance, with the piston at the T.D.C. position, this can cause considerable damage to the drive mechanism because of the extremely high maximum combustion pressure at the T.D.C. position. Conversely, a delayed maximum combustion pressure produces a much lower pressure as the piston has long passed T.D.C. and the piston speed is increasing rapidly. The engine generates less power and the energy created by the high pressure and temperature of the exhaust gases is lost.



Line 1: Injection 20° before T.D.C. Pmax = 160 Bar
Line 2: Injection 30° before T.D.C. Pmax = 200 Bar
Line 3: Injection 10° before T.D.C. Pmax = 120 Bar
Line 4: Injection 40° before T.D.C. Pmax = extreme high

▲

The pressure curve in the cylinder at varying fuel injection times.

Line 1 (green) is the normal pressure curve in the cylinder at an optimally tuned fuel injection. The maximum combustion pressure is approximately 15 tot 20 degrees after the TDC position.

Line 3 (orange) is the pressure curve in the cylinder with a later fuel injection where less power is transferred to the piston and the exhaust gas pressure and temperature increase.

Line 2 (red) is the pressure curve in the cylinder at a slightly early fuel injection. Here the maximum combustion pressure increases occurs too early to be effective on the piston that is almost at its top position.

Less power is produced and the exhaust gas pressure and temperature are approximately equal to line 1.

The dashed line shows the pressure curve at an early fuel injection. The piston is going through the upwards stroke and the combustions process has started. Slightly before the top position and in the top position, the combustions pressure is extremely high and causes damage to the piston and the residual of the gearing. Often this results in irreversible damage. The power output is minimal.

Altering the fuel injection seriously affects the combustion process and the power output of the diesel engine. Meticulously tuned fuel pumps and injectors are a prerequisite for optimal operation of the diesel engine.

As aforementioned, it is fairly simple to calculate the injection- and the total combustion time.

$$\text{Injection time in seconds} = \frac{\text{Injection time in travelled crank degrees}}{\text{360 times the number of revolutions per second}}$$

$$\text{Total combustion time} = \frac{\text{combustion in travelled crank degrees}}{\text{Total combustion process in crank degrees times the number of revolutions per second}}$$

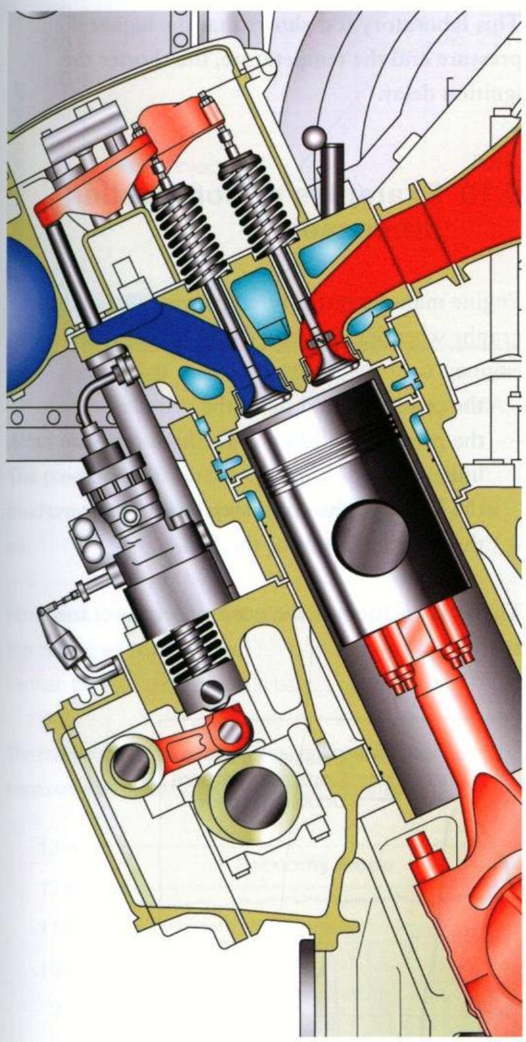
9.7 Ignition delay; causes

Ignition delay is dependent on the shape of the combustion space, the pressure and temperature at the time of the fuel injection, as well as the nature of the atomisation and the ignition quality of the fuel.

Further explanation

– **Shape of the combustion space**

From the time of the fuel injection just prior to the T.D.C. position until the end of the fuel injection just after the T.D.C. position, the combustion space is a cylindrically shaped space with little head space. It resembles a flat disc. The higher the compression rate the flatter the disc. This means that the fuel is almost horizontally injected in most four-stroke- and two-stroke diesel engines.



▲
Fuel injection is an exact occurrence; the injection angle and the fuel injection radius are continually adjusted in order to achieve optimal injection.

The crank angle a t which the fuel is injected varies, but averages between 30 and 45 crank degrees for medium- and high-speed four-stroke diesel engines, and approximately 20 crank degrees for large low-speed two-stroke engines.

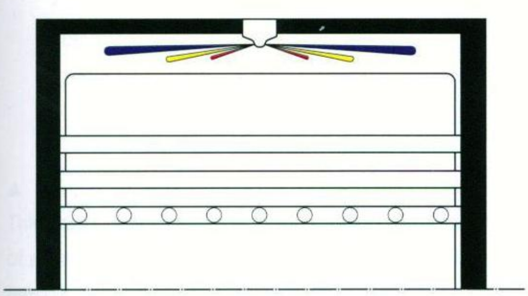
– **Pressure and temperature of the air**

The pressure and temperature of the compressed air can also greatly influence the ignition delay. The high final compression pressure is a result of a high turbo charger fill pressure in conjunction with a high compression rate. The turbocharger temperature is also of importance. Depending on the engine category one finds the following values:

turbo-charger fill pressures	**between 1.0 and 3.5 bar over pressure**
turbocharger temperature	**between 25 and 65 °C**
compression ratios	**approximately between 10 and 20**
final compression temperatures	**from 650 to 850 °C**
final compression pressures	**from 20 to 200 bar**

9.8 Nature of atomisation

The nature of the atomisation is very much dependent on the injection pressure of the injector, the diameter of the injector apertures, so therefore the velocity with which the fuel droplets travel through the 'thick' compressed air.
The droplets must move through the cylinder in just a few milliseconds.



▲
Various fuel injection possibilities.

Experimentally the number and the diameter of injector apertures are adjusted as well as the fuel injection angle and the distance that the fuel droplets travel in the compressed hot air. This is done in order to obtain optimal combustion.

9.9 Ignition quality of the fuel

The ignition quality is dependent on the type of fuel. Light distillate fuels such as gas oil and diesel oil ignite much quicker than heavy oils. The ignition quality of heavy oils can also diverge considerably. Aromats- and asphaltenes (cyclic compounds) are more difficult to crack than aliphatic hydrocarbons (chain compounds) and thus produce a longer ignition delay.
For heavy oil the C.C.A.I.-value is used: a measure for the ignition quality of heavy oil.

The ignition delay of heavy oil can sometimes be improved by mixing it with a fuel that has a better ignition quality (usually a lighter fuel). There is also a (theoretical) formula for the ignition quality.

$$t_v = \frac{k}{p^{1,19}} \times e^{\frac{4650}{T}}$$

Where:

t_v = the ignition delay in milliseconds;
p = the absolute pressure in the chamber during injection;
T = the absolute temperature in the chamber during injection;
e = radix for natural logarithms;
k = the constant; this is dependent on the ignition quality of the fuel.

This laboratory test shows that the higher the pressure and the temperature, the shorter the ignition delay.

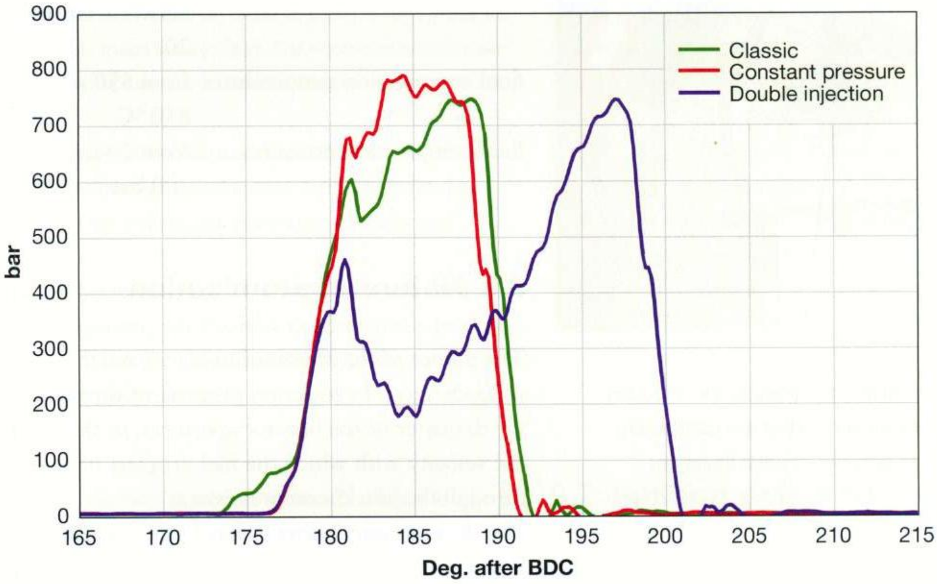
9.10 Examples of combustion processes

Engine manufacturers often provide interesting graphs when introducing a new or improved diesel engine:

- the pressure build up in the cylinder;
- the pressure build up in the high pressure fuel line;
- the lifting of the atomising needle;
- the injection angle of the fuel.

This is in relation to the position of the crank shaft.

► Fuel injection profile for the MAN–B&W ME series, two stroke crosshead engines.



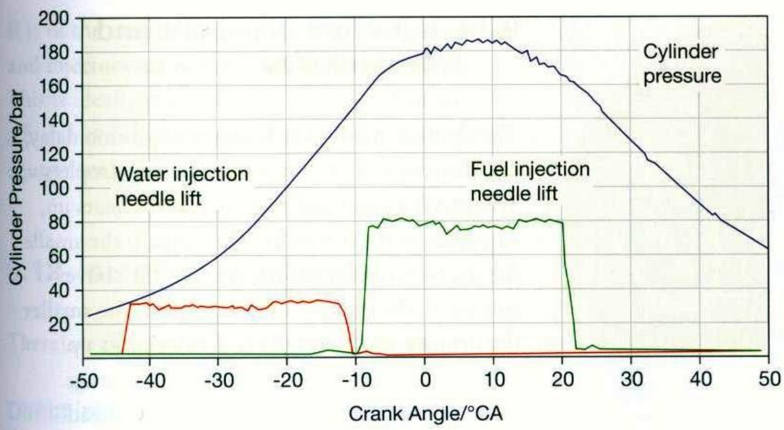
Horizontal: the position of the crank shaft in crank degrees.
At 180 crank degrees, the piston is in top 'deg. after BDC' = crank degrees after the bottom dead point (Bottom Dead Centre).

Vertical: the fuel injection pressure in bars.

Green line: the pressure curve for normal conventional injection.
Start: 173 crank degrees, this is 7 crank degrees before the top.
Finish: 192 crank degrees, this is 12 crank degrees after the top.
Total injection time: 19 crank degrees.
Maximum fuel injection pressure: 740 bar.

Red line: the pressure curve according to the constant pressure system.
Start: 177 crank degrees after the bottom, this is 3 crank degrees before the top.
Finish: 191 crank degrees after the bottom, this is 11 crank degrees after the top.
Total injection time: 14 crank degrees.
Maximum fuel injection pressure: 795 bar.
Remark: the injection time is shorter and at a higher pressure.

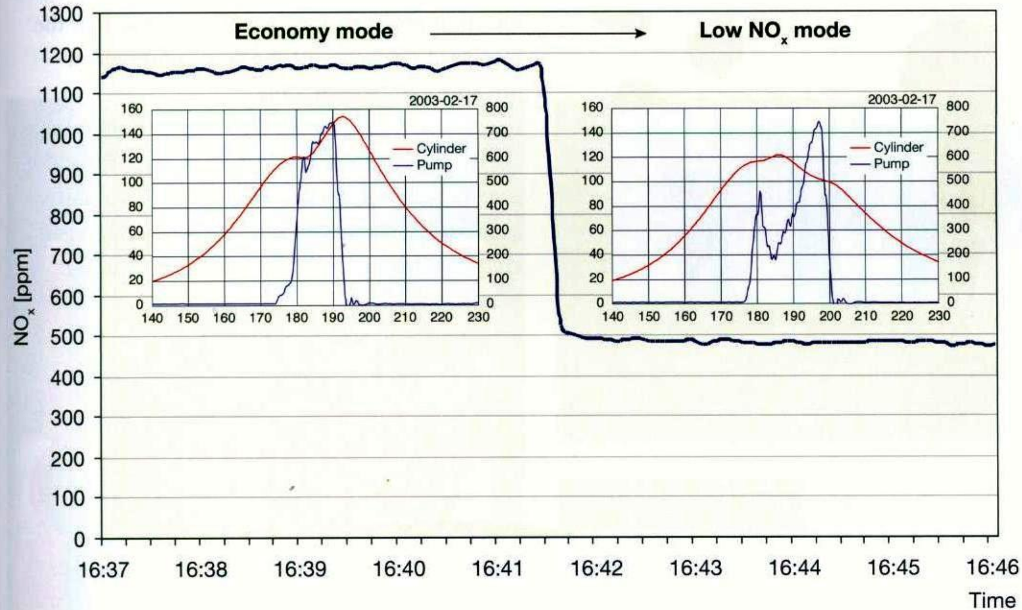
Blue line: double injection.
Start: 177 crank degrees after the bottom, this is 3 crank degrees before the top.
Finish: 201 crank degrees after the bottom, this is 21 crank degrees after the top.
Total injection time: 24 crank degrees.
Maximum fuel injection pressure: 740 bar.
Remark: the fuel injection takes place in two phases and takes longer. Due to this, the maximum combustion pressures and maximum combustions temperatures are lower. This results in lower nitrous oxide emissions.



▲
The pressure curve in the cylinder of a four stroke medium speed diesel engine operating on heavy fuel oil.

Horizontal: the position of the crank shaft with at the zero the top dead point of the piston.
Vertical: the cylinder pressure in bars.

The maximum combustions pressure in the cylinder is approximately 185 bars and occurs approximately 10 degrees after the top position of the piston. The fuel injection starts at the moment that the injector needle is opened by the increased fuel pressure from the high pressure fuel pump.
This occurs at 10 degrees before the top position to 20 degrees after the top position of the piston (green). Injection takes place in total over 30 crank degrees. This engine is also equipped with water injection to limit nitrous oxide emissions. This takes place before the top position of the piston (red).



▲
This graph shows the pressure curve in the cylinder for of a two-stroke crosshead engine operating on heavy fuel oil.

Left graph:
Horizontal: the position of the crank shaft. At 180 crank degrees the piston is in the top position.

Left vertical:	the cylinder pressure in bars
Right vertical:	the fuel injection pressure in bars
Red line:	the cylinder pressure curve
Blue line:	the fuel pressure curve at injection

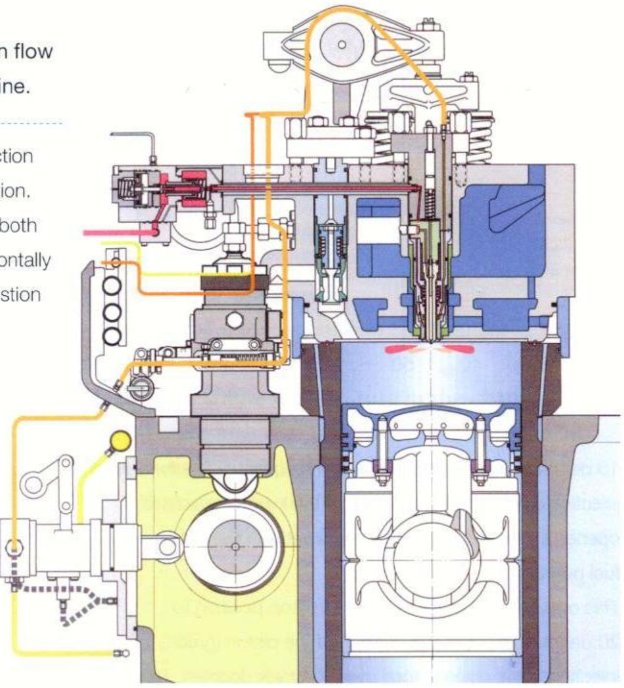
The graph shows a two-stroke crosshead engine operating at the lowest fuel usage.

Right graph:
This graph shows the same engine that has been tuned so that nitrous oxide content in the exhaust gasses is limited. The fuel injection takes place later and consists of a pre- and main injection. Due to this is the combustion is less explosive and more gradual. This results in lower maximum combustion temperatures and pressures, therefore less nitrous oxides are produced.

▶

Shown, the injection flow in a 'dual- fuel' engine.

Brown is the fuel injection and red the gas injection. Clearly shown is that both fuels are almost horizontally injected in the combustion chamber.



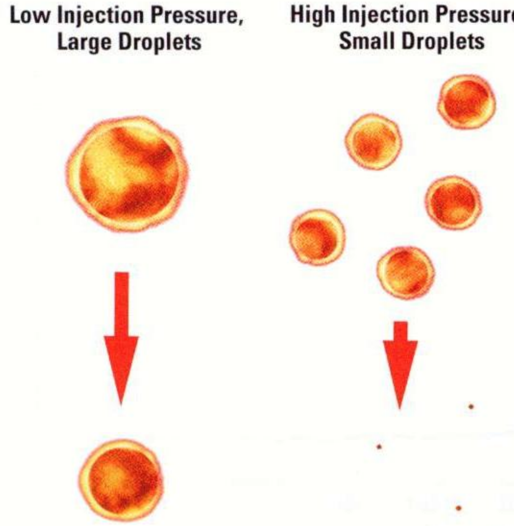
9.11 Injection pressure and droplet size

The ignition quality, including the ignition delay, is also determined by the size of the fuel droplets. The smaller the droplet the quicker it heats up, vaporises and gasifies. In other words: the smaller the droplet, the higher the ratio of the convex surface to the content of the droplet. The smaller the droplet, the faster the fuel mass heats up!
The **droplet size** is dependent on the average injection pressure. For two-stroke crosshead engines this lies between the 250 and 1000 bar and for four-stroke engines between 250 and 2000 bar. This, of course, largely depends on the engine type.
The injection pressure has increased considerably throughout the years. Due to the growing demand for an increase in cylinder capacity the effective mean pressures have risen significantly. This has been facilitated by increasing the pressures, including the final compression pressure, during the combustion process. In order to swiftly spray the fuel sufficiently far into the cylinder the fuel injection pressure must also increase so that the increased resistance of the final compression pressure can be overcome.
Electronically controlled modern fuelling systems for two-stroke crosshead engines currently (2008) use pressures of approximately 1000 bar and for four-stroke engines of approximately 2000 bar. Here the droplet size varies from 15 to about 60 micro-millimetres.

▶

The droplet size is important in the achievement of an ideal combustion process.

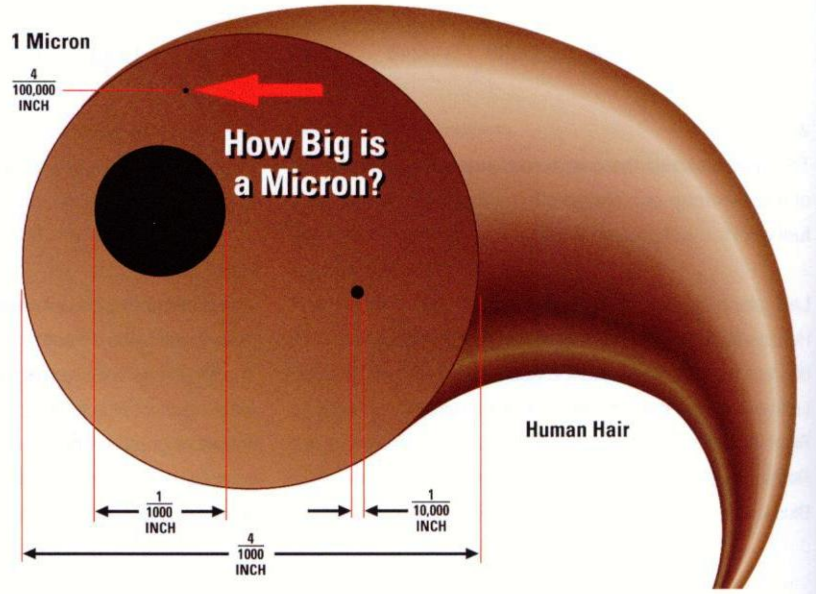
At a low fuel injection pressure, the droplets are too large and therefore incomplete combustion occurs. At higher fuel injection pressure, the droplet size is reduced and the fuel combusts faster and more completely.



▶

The droplet size relative to a human hair.

A human hair has a diameter of approximately 0,004 inch, approximately 0.1 millimetre. A fuel droplet is approximately 0,001 inch in diameter, approximately 0,025 millimetre. This is 25 microns; the ideal droplet size!



It is of the utmost importance that the fuel pumps and injectors are maintained and set correctly. This is ideally performed by specialised firms which have all the required maintenance- and test equipment in-house.

9.12 Injection principles

There are two systems: **direct** and **indirect.**

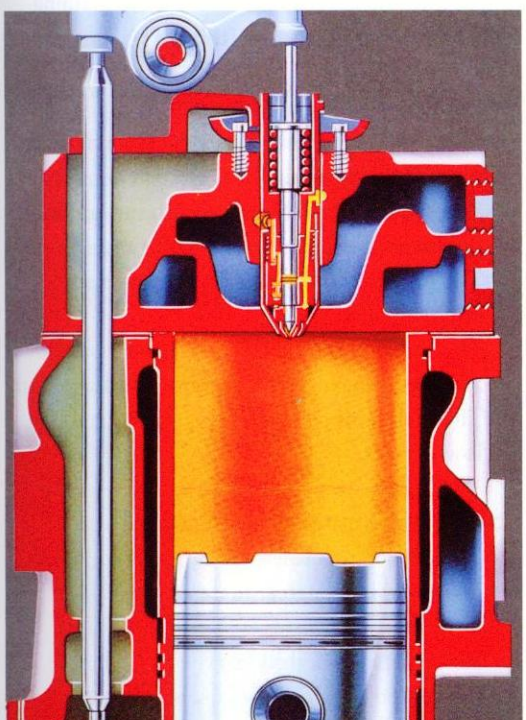
Direct injection

The fuel is injected into the single combustion space under pressures of 1000 to 2000 bar. The modern four-stroke engines have one fuel injector in the heart of the cylinder head. They are usually multiple-aperture injectors which inject the fuel almost horizontally into the disc-shaped combustion chamber at a maximum pressure, the *so-called* peak pressure, of 2000 bar. The distance that is covered during the injection amounts to approximately 0.3 D from the centre axis. In fact this means that the cylinder is injected with fuel over a diameter of 0.6 D. So in most of today's engines the fuel is not sprayed against the piston crown surface.

In two-stroke crosshead engines two or three injectors inject the fuel almost horizontally into the engines. The fuel is injected into the cylinder at a certain angle in a circular spray form. The injectors in question are always multiple-aperture injectors.

◄ The injection flow of the three injectors in a large two stroke crosshead *engine*.

As the exhaust valve is centrally positioned, the injectors are positioned around the circumference. The various colours show the gas temperatures. Red is the highest temperature then decreasing yellow, blue and purple.



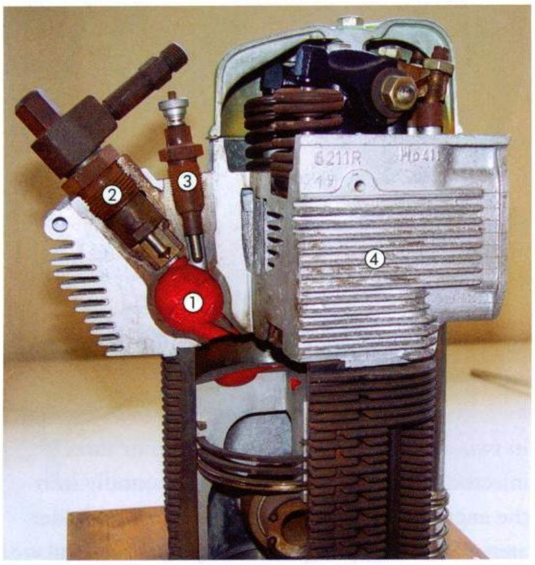
▲ The combustion process above the piston is a very quick and complicated process. The fuel injection quality is essential for the good operation of the diesel engine.

◄ The three injectors per cylinder for a two stroke crosshead engine.

◄ The injector tip of a similar engine with multiple injector apertures.

Indirect injection

The fuel is injected into a separate prechamber or pre-heating chamber at a relatively low pressure. The system is only applied in four-stroke high-speed engines.



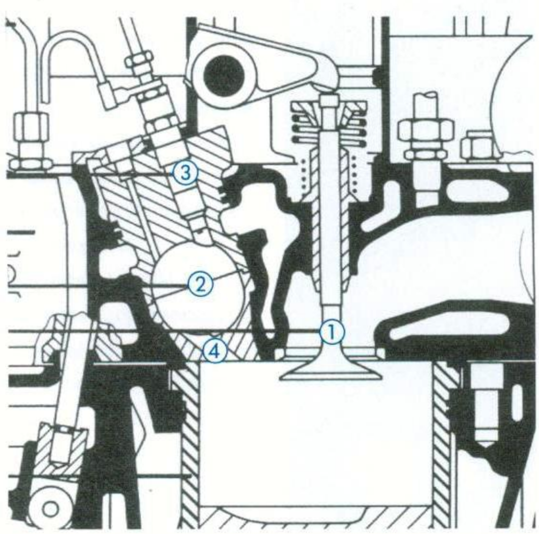
▶ A prechamber injection in an air cooled Deutz diesel engine.

Left in the prechamber the single aperture pintle type injector and right the glow plug for a cold start.

1 *prechamber*
2 *injector*
3 *glow plug*
4 *cylinder head*

There are many different types of prechambers. In prechamber injection the fuel is injected into the prechamber via a pintle nozzle at a fairly low injection pressure of a 100 to 200 bar. There are numerous different antechamber designs. Essentially, they all serve a similar purpose.

- The walls are extremely hot and rapidly heat up the injected fuel.
- Because of this the fuel swiftly ignites.
- Due to the high temperature the fuel in the antechamber cracks and breaks down the long hydrocarbons molecules into smaller chains which are easier to combust.
- As the pressure in the relatively small antechamber rapidly increases, the mixture is pushed, at a great force, through a number of specially shaped ducts into the main combustion chamber. Here there is more than sufficient air to fully combust all the fuel in the prechamber.
- As a result of the special mode in which the pipes are connected to the main combustion chamber, there is increased air movement which means that remaining fuel combusts in its entirety and swiftly.



▶ A Ricardo- prechamber in a conventional four stroke medium speed diesel engine.

Cylinder diameter 210 millimetre. Fuel M.D.O. Note the position of the single aperture injector and the channel to the main combustion chamber.

1 *exhaust valve*
2 *prechamber*
3 *pintle type injector*
4 *tangential channel*

Whirl-chamber engines have a conically shaped sphererical whirl chamber in their cylinder head. This is connected to the main combustion space by means of a tangential pipe. During the compression stroke, a part of the compressed air is fed into the whirl chamber, causing rapid air rotation. The fuel injected by the single-point injector is now intensively mixed with the rotating air. The walls of these whirl chambers are also very hot as they are only partially cooled. This causes the incoming air to achieve high temperatures very quickly and therefore the injected fuel will swiftly combust. Due to the high pressure rise in the relatively small space, the mixture which still contains a large amount of uncombusted fuel is forced into the main space where the remaining fuel combusts fully and swiftly.

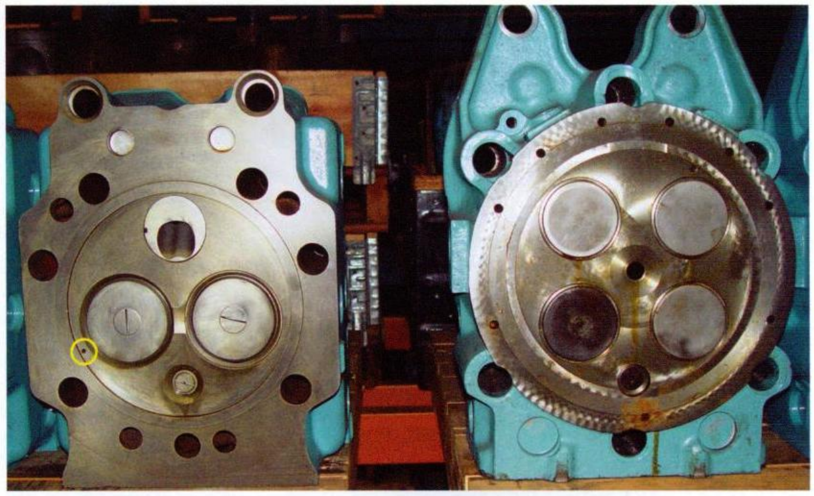
Some advantages of this system

- Swift ignition which makes the system suitable for high-speed engines.
- The system is barely sensitive to the ignition quality of the fuel.
- The maximum combustion pressures are relatively low.
- The single-point injector is relatively simple and pipe blockage will therefore seldom occur.
- The fuel pump is relatively simple due to the low pressures.

Some disadvantages of this system

- The efficiency is lower than that of a directly injected diesel engine; there are considerable flow-and whirl losses. The cooling -water losses are also slightly higher.
- Compared to a directly injected engine, starting is more difficult due to the low injection quality and the higher loss of compression warmth to the still cool ambient air. Therefore the engine often requires aids such as hot wire filaments to assist starting.
- Prechambers tend to crack with a cold start or rapidly changing loads.

During the past years, larger numbers of high-speed engines have been built on the principle of direct injection to avoid the aforementioned disadvantages of indirect injection.
The volume of an antechamber in relation to the main chamber can range from 20 to 50%.



▲
Left, the cylinder head of a older diesel engine type with above the prechamber, in the middle the inlet-and the exhaust valve; below the air starting valve. Right a modern cylinder head with direct injection.

The small hole bottom left (see circle) of the inlet valve is the relief valve. Right the cylinder head of a more recent diesel engine type with an injector placed centrally in the head the for direct fuel injection and around this aperture, both inlet- and exhaust valves. Fuel is M.D.O. and/or H.F.O.

9.13 Shape of the combustion chamber

9.13.1 Four-stroke

The space at the end of the compression stroke, so at the beginning of the combustion process is increasingly 'flat' as a result of the increasing compression rate.

Let's assume the compression rate is 14 and the cylinder bore 200 millimetres.

$$\varepsilon = \text{compression rate} = \frac{V_S + V_C}{V_C}$$

$$14\,V_C = V_S + V_C$$

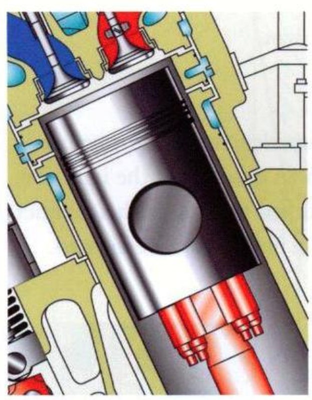
$$13\,V_C = V_S$$

The volume of the final compression space then is:

$\frac{1}{13}$ part of the stroke volume.

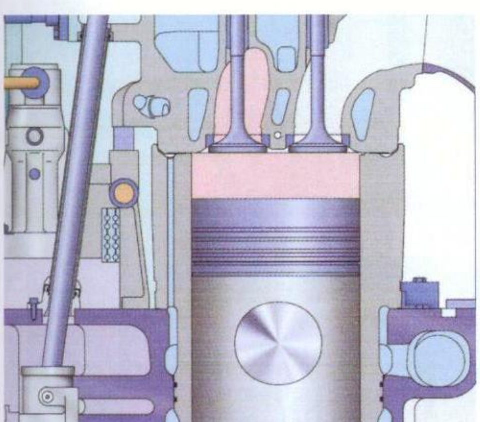
Let's assume that the stroke at a cylinder bore of 200 millimetres constitutes 220 millimetres, which gives a height of the final compression space

$$\frac{220}{13} = 16.9 \text{ millimetres!}$$

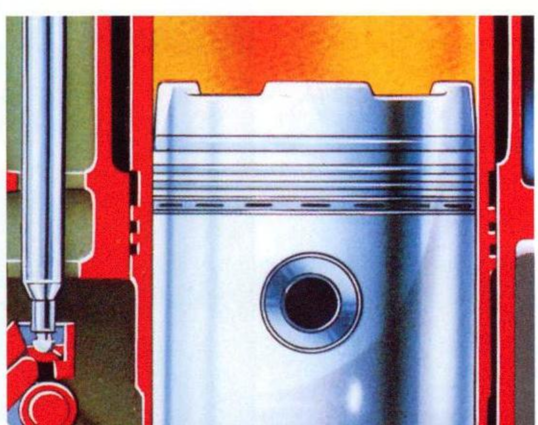


◄
The cross-section of a V- engine with the piston in top position and the both valves closed.

Clearly shown how small the space is above the piston.



▲
In this line engine with the piston almost in the top position is the space above the piston is also only centimetres high.



◄
Recesses in the piston crown of a high speed Cummins-diesel engine.

These are intended for the positioning of the inlet- and exhaust valves at the piston top position when the valves are slightly opened. Therefore during exhaust air scavenging.

▲
Recesses in the piston crown of a Caterpillar–MaK 32 diesel engine in order to create valve space.

In the background, the dismounted steel piston crowns. Note the bowl shaped combustion chamber in the piston crown.

In this 'flat disc' the fuel must be injected almost horizontally.
The **piston crown** is also often given a specific shape and the two inlet- and exhaust valves require some room.
Often there is so little space at the T.D.C. position of the piston during exhaust air scavenging that recesses have been made in the upper rim of the cylinder liner and the piston crown in order to create room for the valves.

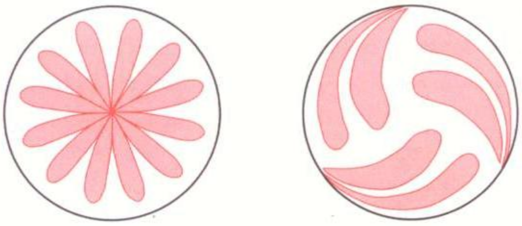
Injecting the fuel

In a test phase or in practice, the injector nozzle is often adjusted to such an extent that the injection angle and -jet are modified slightly. In this manner, the most ideal injection method is investigated.

The number of injector apertures or the location of the apertures are regularly modified. One could obviously also modify the opening pressure of the injector, the maximum pressure of the fuel injection or the diameter of the injector apertures. The injection curve and the moment of injection are important values in finding the ideal injection.

9.13.2 Two-stroke crosshead engines

All two-stroke crosshead engines work on to the uniflow- or direct flow principle. This means that the air is supplied via the scavenging ports located in the lower section of the cylinder liner and that the discharge of exhaust gases occurs by means of the hydraulically controlled exhaust valve centrally situated in the cylinder cover.

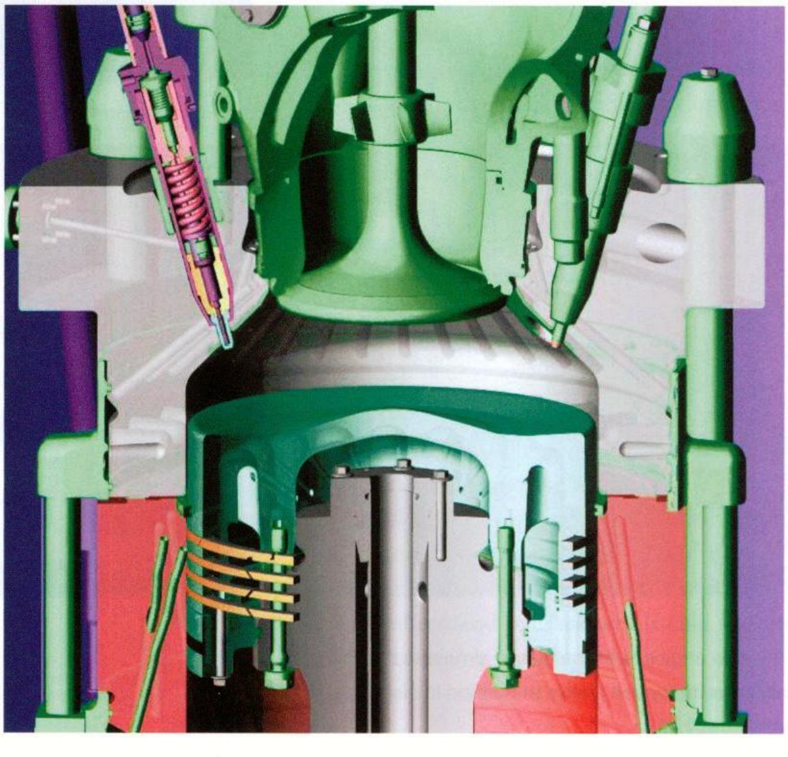


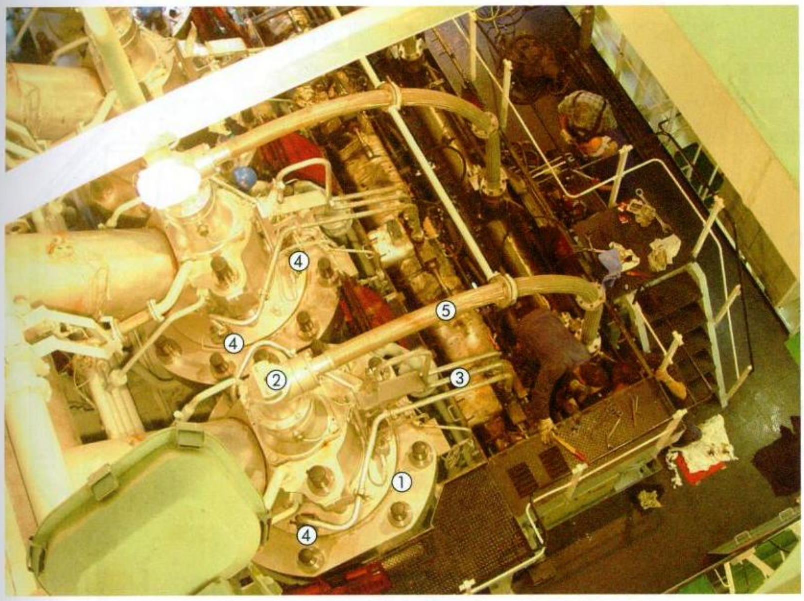
▲
The injection flow of a four stroke-trunk piston engine and a two-stroke crosshead engine.

Left, the injection flow pattern from the cylinder heart when the fuel is centrally injected.
Right, the circular injection flow pattern that is slightly directed towards the inside when the fuel is injected from the circumference.

►
The Oros combustion chamber incorpates all the latest developments and features from MAN Diesel.

Note the rounded upper angle of the combustion chamber in the cylinder cover. Central the large exhaust valve. The two injectors are visible.





A large two-stroke crosshead engine as seen from above.

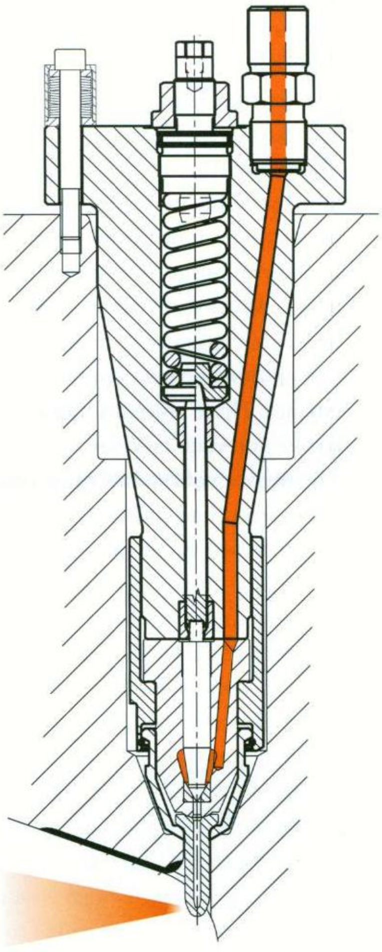
Note the dimensions of the cylinder heads in relation to the engineers.

1 *cylinder cover*
2 *exhaust valve*
3 *high pressure fuel lines to the three injectors*
4 *injectors*
5 *hydraulic pipe lines to open the exhaust valve*

Therefore, there is no space to place a fuel injector in the heart of the cylinder cover . The injectors are then mounted in a circle in between the exhaust valve and the cylinder wall.

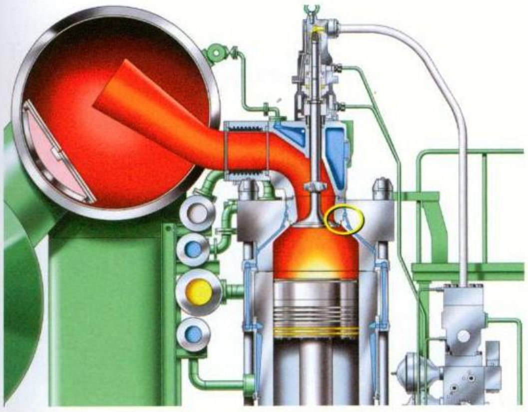
Two or three injectors, which inject the fuel more or less in the direction of this circle, are fitted.

The upper corners of the combustion chamber are rounded in order to avoid scavenging shadows. The shape of the piston bottom may vary, but is generally fairly flat and either slightly bowl-shaped or provided with an elevation in its heart ('island').



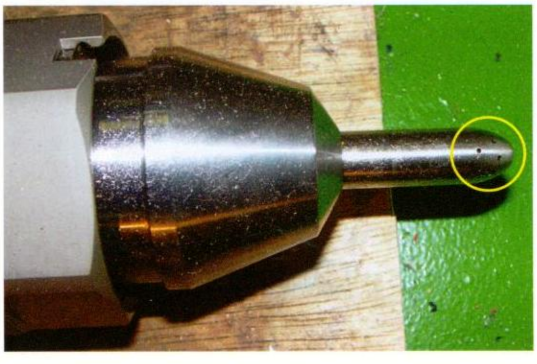
The position of the injector at the edge of the combustion chamber in a two stroke crosshead diesel engine.

The fuel is almost horizontally injected.



The upper part of the combustion space in this two-stroke crosshead engine is rounded to avoid scavenging shadows.

Scavenging shadows are areas where the scavenge air does not (fully) reach therefore exhaust gas-particles remain behind.



A injector tip of a similar engine with multiple injector apertures.

Injecting fuel

Fuel injection in two-stroke crosshead engines is frequently adjusted in order to obtain optimumal injection. The injector nozzles are often changed.

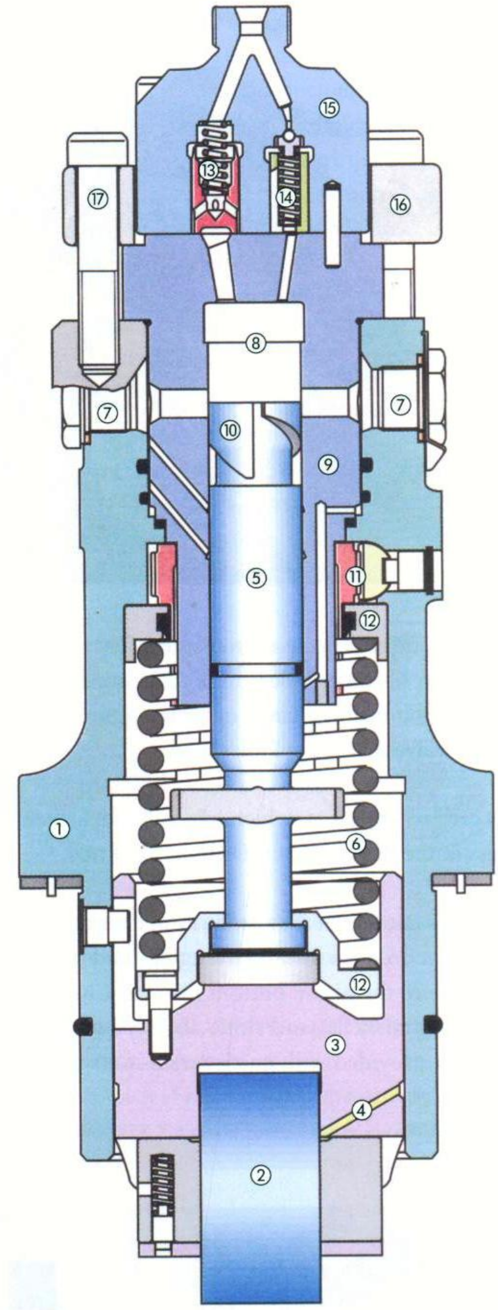
9.14 Fuel-injection mechanism: fuel pump

Fuel pumps have to meet demanding requirements since the manner in which the injection takes place is crucial for the quality of the combustion process, and therefore the optimal operation of the diesel engine.

9.14.1 Fuel pump requirements

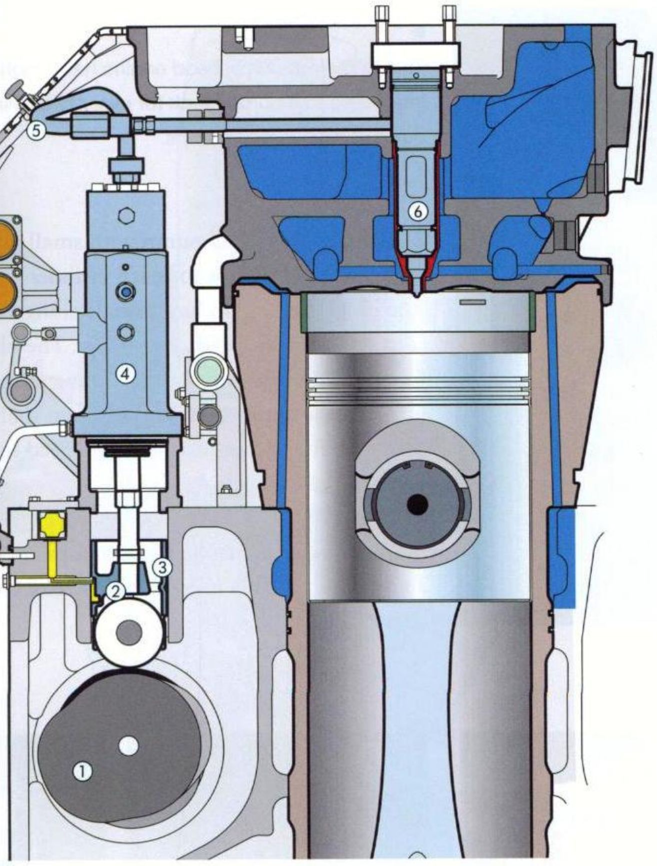
- **To swiftly put the fuel under high pressure.** The pressure time varies from several thousandths of a second to several hundredths of a second. The required pressure ranges from approximately 200 to 2000 bar.
- **To pump a very meticulously regulated amount of fuel to the injector where the quantity should be continually controllable from zero output to maximum output.** The maximum amount varies from several tenths of a cubic millimetre for small engines to several dozen cubic centimetres for the largest engines. These amounts apply to each plunger stroke and each cylinder.
- **The delivery of the fuel must commence at the correct time and last for a certain time.** The complete injection curve for four-stroke high-speed and medium-speed diesel engines amounts to approximately 30 to 40 crank degrees and about 20 crank degrees for two-stroke crosshead engines.
- **The fuel pump leakage losses should be minimal.** For the most frequently used fuel pumps, the plunger pump, fuel leakage is negligible. The long plungers, which have very little clearance in relation to the plunger housing, guarantee minimal leakage.
- **The fuel pump should be able to cope with fuel temperatures of approximately 150 °C to ensure that the heavy fuel is atomised into the cylinder at the correct viscosity.**

Heavy fuel viscosity prior to injection must lie between 8 and 12 centistokes for four-stroke engines and between 10 and 15 centistokes for two-stroke crosshead engines. The fuel temperatures lie between 125 to 150 °C.



▲
A high pressure fuel pump in a medium speed four stroke-diesel engine operating on H.F.O.

1 *pump housing*
2 *cam rollers*
3 *roll guide*
4 *lubricating oil supply*
5 *plunger*
6 *plunger spring*
7 *suction chamber*
8 *pressure chamber*
9 *pump cylinder*
10 *machined part of the plunger*
11 *gear rack for the plunger rotation*
12 *spring seats*
13 *pressure valve*
14 *overflow valve, pressure release*
15 *pump cylinder head*
16 *pressure plate*
17 *bolt*



◄ A complete high pressure fuel system of a four stroke medium speed diesel engine.

The last part of the high pressure fuel line runs through the cylinder head. The injector tip is intensively cooled around its surface perimeter.

1 *camshaft*
2 *roll*
3 *guide*
4 *fuel pump*
5 *high pressure fuel line*
6 *fuel injector*

Engine manufacturers often set an upper limit to the fuel temperature because the quality of the sealings of fuel pumps rapidly deteriorates when exposed to higher temperatures.

9.14.2 Fuel pump drives

Traditionally, fuel pumps are driven from the camshaft by means of fuel cams.

Today there are also other drive systems such as the common rail systems. This will be discussed later in this chapter.

9.14.3 Types of injection systems

Six different fuel injection systems can be distinguished.

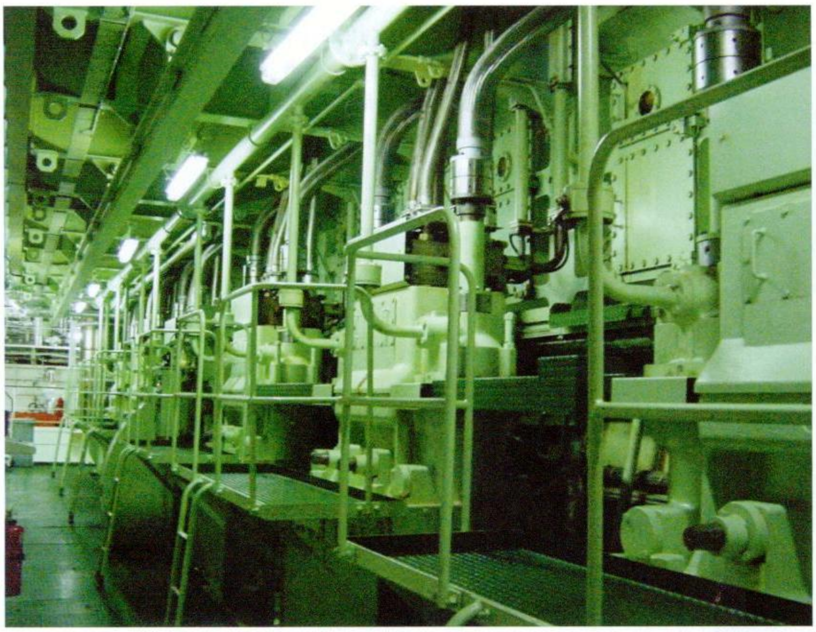
1 With plunger controlled pumps for each cylinder

Each pump is individually driven from the cam shaft by a fuel cam.

This system can be applied to each type and size of engine.

▲ In this Caterpillar–MaK diesel engine every cylinder has its own high pressure fuel pump.

▼ Two high pressure plunger fuel pumps, in the centre is the hydraulic pump for the exhaust valve. Engine type MAN–B&W 50 – MC.



▶
This two-stroke crosshead engine has a high pressure integrated fuel pump block for every two cylinders comprising two fuel pumps.

2 With a block fuel pump with small plunger controlled pumps for each cylinder

The block fuel pump has a small plunger controlled pump for each cylinder. The block fuel pump is driven from the 'timing gear train' between the crank shaft and the cam shaft. This system is often used in high-speed four-stroke engines.

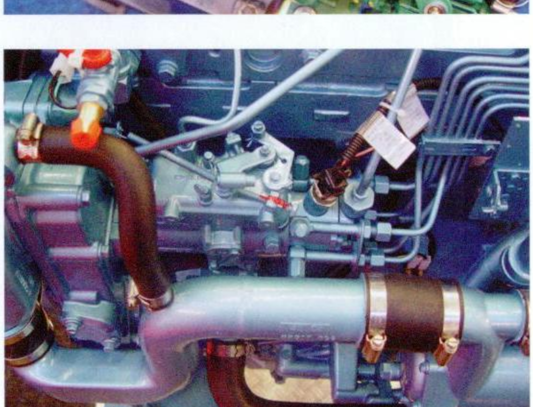
3 With valve controlled fuel pumps

This system is used in large two-stroke crosshead engines.

▶
A fuel pump block for an in line high speed six cylinder four stroke-diesel engine.

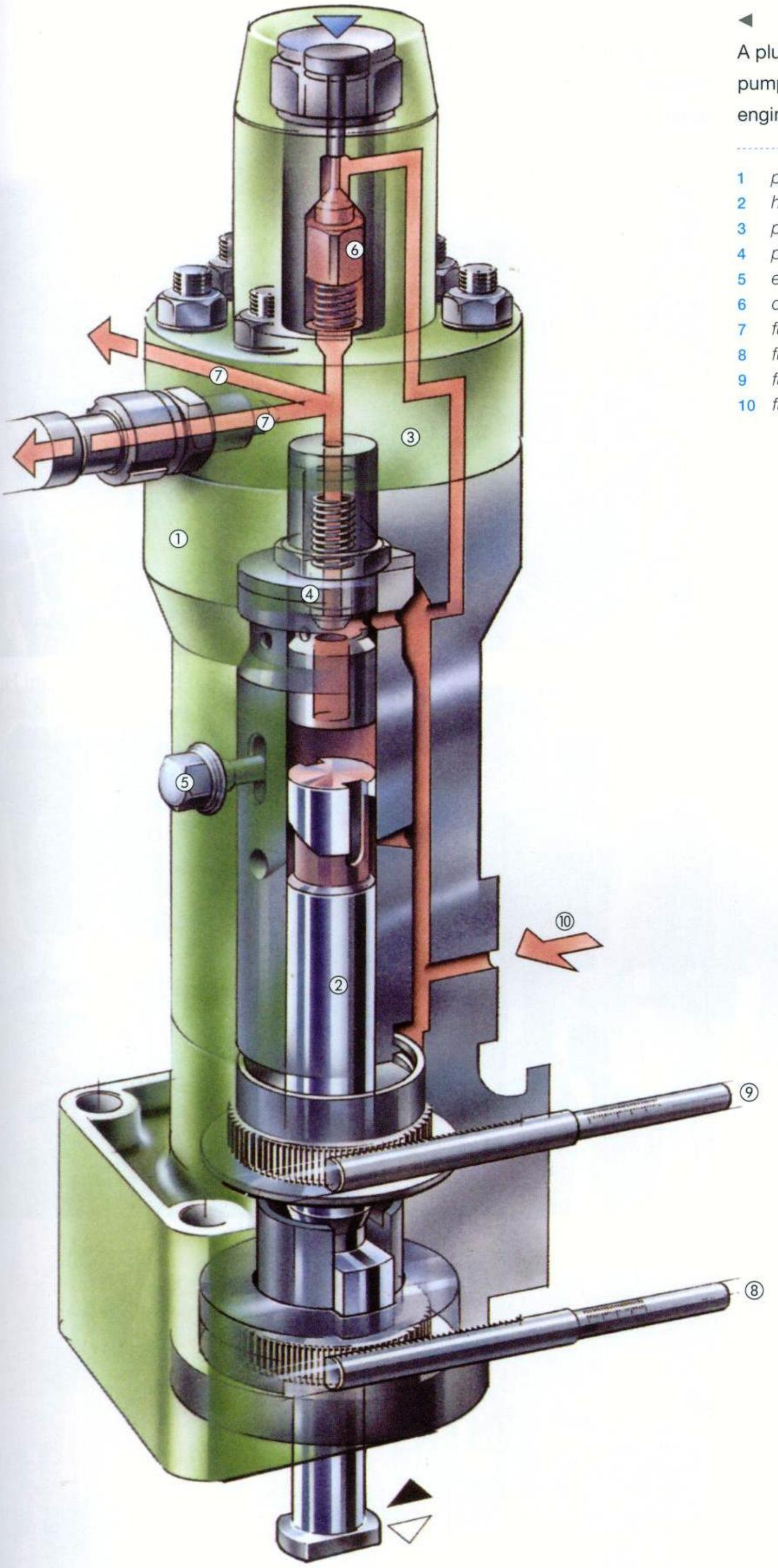
▶
A fuel pump block for a high speed three cylinder four stroke-diesel engine found in yachts.

▶
A horizontally positioned rotated block fuel pump for a small high speed six cylinder four stroke-diesel engine.



▲
A pair of valve regulated high pressure fuel pumps for two cylinders of a two-stroke crosshead engine. Manufacturer Wärtsilä Sulzer RTA.

The pump is placed close to the low positioned camshaft, therefore not at the same height of the cylinder covers. Left and right, high pressure lubricating oil pumps for the operating the exhaust valves. In this engine type, two fuel pumps are assembled in one housing.



A plunger regulated high pressure plunger type fuel pump of the Wärtsilä Sulzer RTA two stroke crosshead engines.

1 *pump housing*
2 *high pressure plunger*
3 *pump head*
4 *pressure valve*
5 *erosion/cavitation plug*
6 *overflow valve*
7 *fuel to injectors*
8 *fuel adjustment spindle, quantity*
9 *fuel adjustment, timing*
10 *fuel supply*

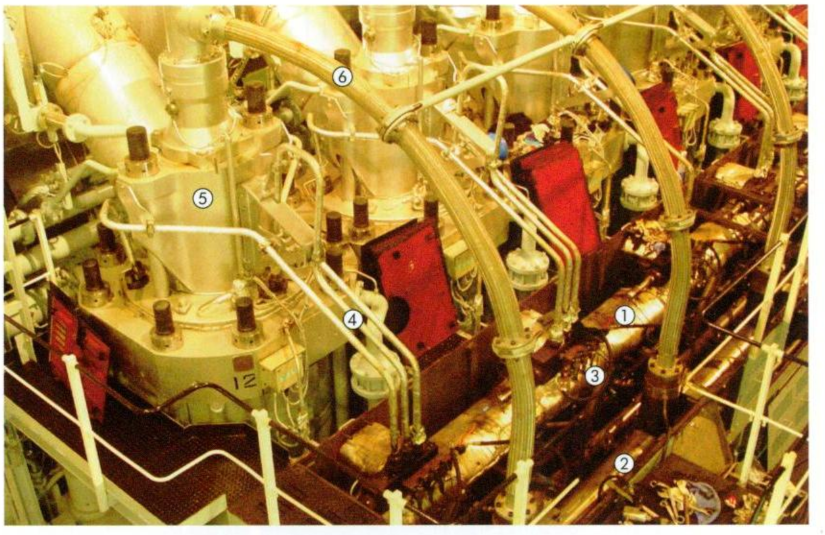
4 **Common rail system**

In this system the fuel supply is regulated electronically. There are different systems for four-stroke trunk piston engines and two-stroke crosshead engines.

▶

A two-stroke crosshead engine from manufacturer Wärtsilä Sulzer 12 RT-flex 96 C.

1 *fuel common rail 1000 bar*
2 *lubricating oil common rail 200 bar*
3 *fuel return lines*
4 *high pressure injector fuel lines*
5 *exhaust valve*
6 *high pressure lubricating oil supply to the exhaust valve*



▶

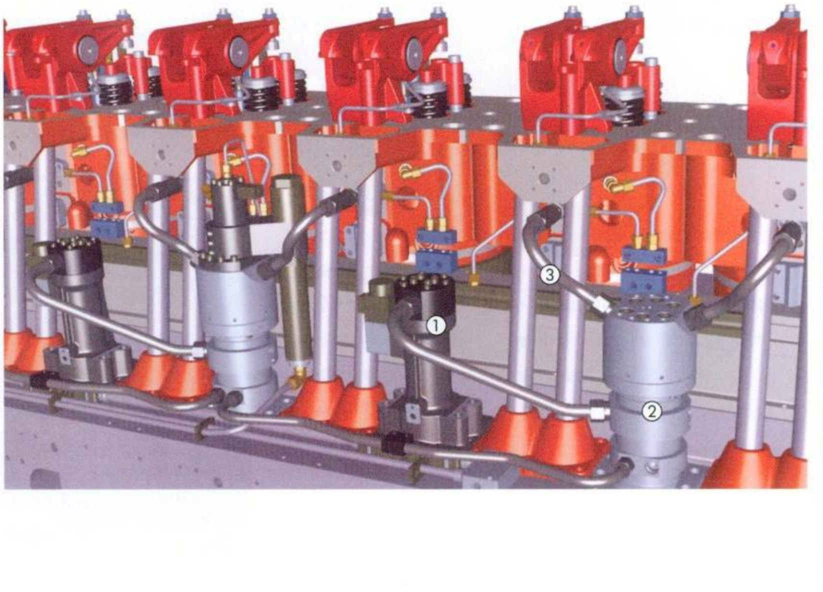
The high pressure injector of a Caterpillar high speed four stroke-diesel engine with the common rail system.

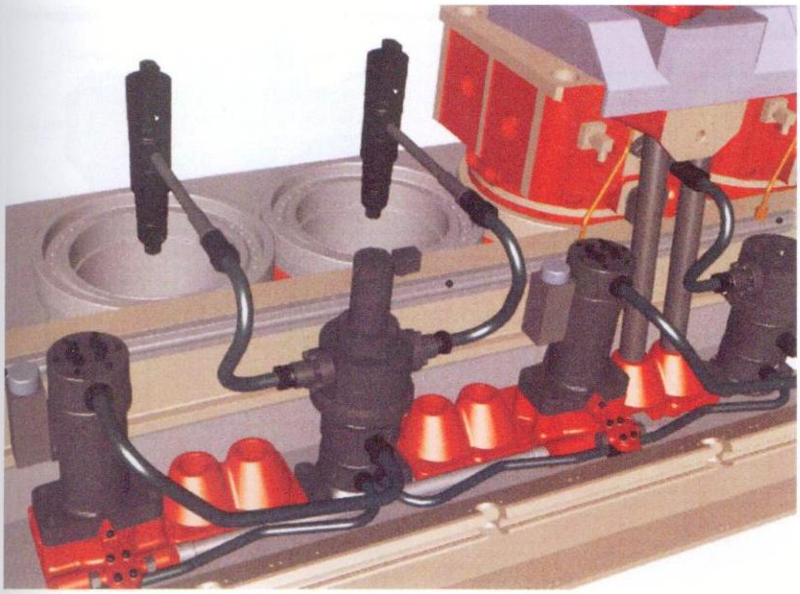
1 *fuel injector*
2 *high pressure fuel supply line*
3 *fuel return lines*
4 *exhaust valve*
5 *inlet valve*
6 *combustion chamber*

▶

The common rail system for the four stroke medium speed diesel engines from manufacturer Wärtsilä.

1 *high pressure fuel pump two cylinders*
2 *high pressure fuel accumulator for two cylinders*
3 *high pressure fuel line to the fuel injectors*





◀

The equivalent common rail system for Wärtsilä four stroke engines with the cylinder heads of the two cylinders removed.

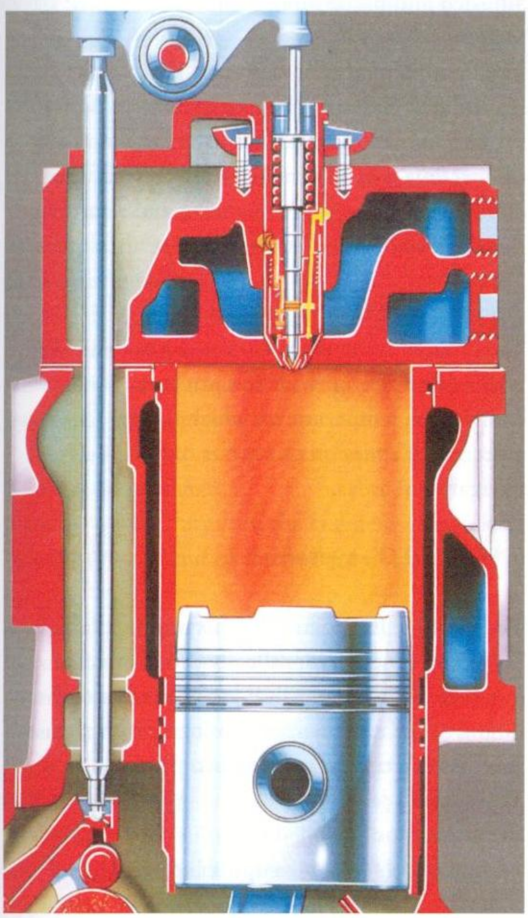
5 Injector system

In this system the high pressure fuel pump and the atomiser are located in one casing the so-called fuel injector. This fuel injector is placed in the cylinder head and is mechanically driven from the cam shaft by means of a fuel cam, guide pulley, push rod and lever, just like the inlet- and exhaust valves. The system is used in high-speed four-stroke engines of, among others, Cummins and Caterpillar.

▼

A mechanically driven fuel injector from manufacturer Caterpillar.

1 *push rod*
2 *rocker arm*
3 *pressure spring*
4 *fuel injector*
5 *combustion chamber*



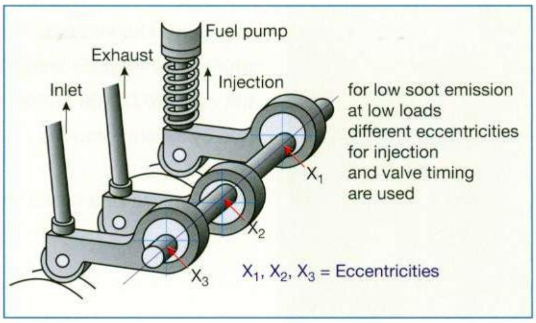
◀

A mechanically driven fuel injector from manufacturer Cummins.

◀

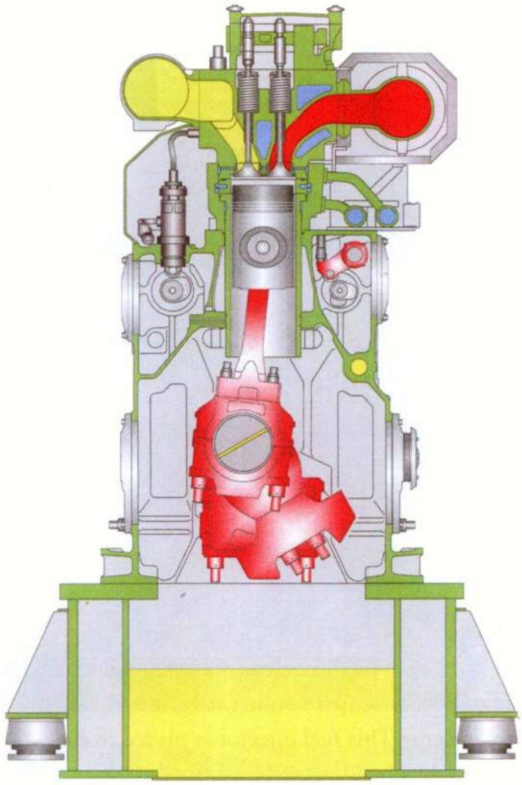
Fuel injectors with mounted the magneto valves.

6 Fuel pumps with adjustable cam drives



▶
The mechanically adjustable camshaft in the medium speed diesel engines from Caterpillar–MaK.

Through the rotation of the eccentricities, the timing of the valves and high pressure fuel pump is modified.



▶
A MAN–B&W four stroke medium speed diesel engine shown here is provided with a separate camshaft for the high pressure fuel pump and the inlet- and exhaust valves. In this engine, the timing of the valves can also be adjusted.

9.14.4 Fuel pumps with modification for fuel quality, expressed in C.C.A.I.

This system is, among others applied to Wärtsilä-Sulzer two-stroke crosshead engines. Two systems can either independently or concurrently be applied to the control system of the fuel pumps.

9.14.5 V.I.T.-system

The Variable Injection Timing-system. This is applied in order to advance the fuel injection at partial load. This way the maximum combustion pressure, and consequently the engine efficiency, remains at a high level. The system operates automatically and is dependent on the fuel control mechanism.
The V.I.T.-system operates in the engine load scope of 100% to 85% and automatically ensures that the combustion pressure retains its maximum value.
Maximum fuel economy is achieved at an engine load between 80 and 90% and amounts to two grams per kWh.
The automatic V.I.T.-system is always combined with a manually adjusted F.Q.S.-system. Engines that are unsuitable for a V.I.T.-system are equipped with a separate F.Q.S. system.

9.14.6 F.Q.S.-system

The Fuel Quality Setting-system serves to ensure that the maximum allowable combustion pressure is not exceeded. The mechanism is manually adjusted and based on the ignition quality of the fuel.
The ignition quality is expressed in C.C.A.I. (Calculated Carbon Aromaticity Index). Both systems are limited when adjusting the moment of injection. The maximum adjustment is three crank degrees for both advancing and retarding the injection.

Operation of both systems

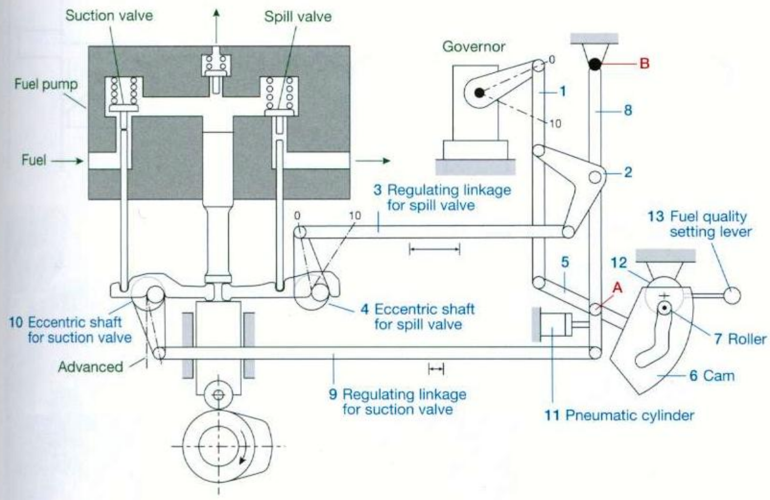
Today these systems are electronically controlled, which allows for a reduction in the maximum combustion temperatures, which in turn facilitates a decreased emission of nitrous oxides in certain navigational areas.

9.14.7 V.E.C.-system

The Variable Exhaust Control-system.
This controls the hydraulically driven plunger pump in the exhaust valve, which facilitates the timing of the opening and closing of the exhaust valve within certain restrictions.
Therefore in present times , the injection can be electronically controlled for:

- minimum fuel consumption at partial load (V.I.T.);

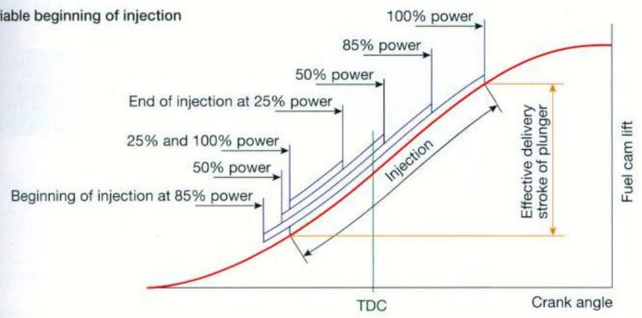
Fuel regulation with 'VIT' combined with manual fuel quality setting 'FQS'



◄ Wärtsilä Sulzer two stroke crosshead engines type RTA. The automatically regulated V.I.T. - system in combination with the manually regulated F.Q.S. - system.

The F.Q.S. - system is operated with a fuel quality setting lever 13 that is fixed using a pin 12, an eccentric positioned roller 7 moves. This moves via a recess in the cam 6. Via levers 5, 1, and 2, the position of the overflow valve is regulated and via shaft 9 the suction valve.
The V.I.T. - system is automatic and can at partial loads advance the fuel injection.

Variable beginning of injection



- an altered fuel quality (F.Q.S.);
- an optimum exhaust period of time (V.E.C.).

Naturally, the development of common-rail engines is a continuation of the aforementioned techniques.

9.15 Fuel-capacity adjustments

Essentially, three fuel capacity adjustment systems can be distinguished.

1 Initial adjustment
Here the power output of the engine is controlled by adjusting the start of the injection.

2 Final adjustment
Here the power output of the engine is controlled by adjusting the end of the injection.

3 Initial and final adjustment
Here the power output of the engine can be controlled. The initial adjustment ensures that the injection starts at the correct time at varying loads. A reduced power output and lower load will modify the ignition delay. With the final adjustment, one controls the power output.
The initial and final adjustment can also be applied to the electronically controlled common-rail systems. One is in this case, no longer bound by mechanical limitations.

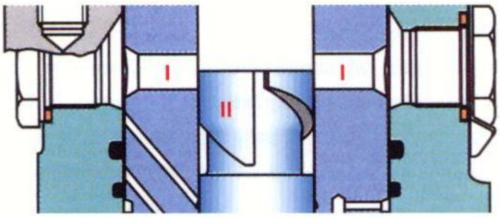
9.16 Working of a plunger pump

9.16.1 Plunger controlled fuel pumps

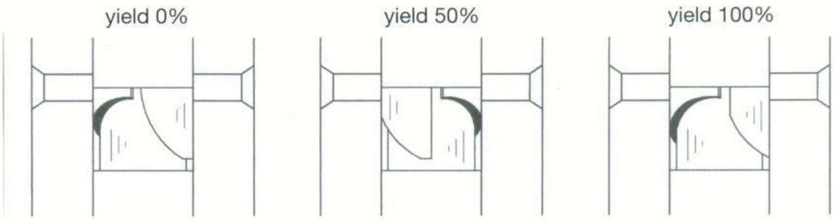
This type of pump is still often used. Not only in very small high-speed four-stroke diesel engines, but also in medium-speed four stroke engines. Two-stroke crosshead engines use both valve- and plunger controlled fuel pumps.

Principle

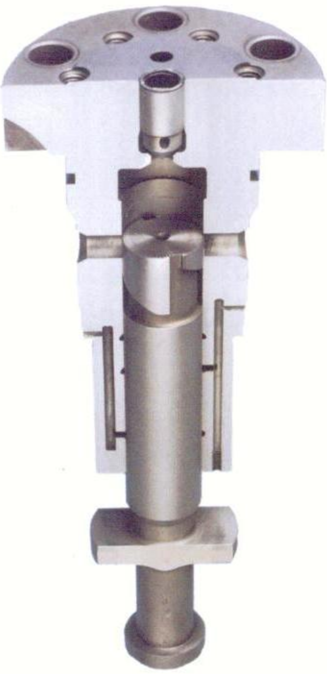
The principle of the plunger controlled pumps is based on the rotation of the plunger in relation to the plunger barrel. Due to the rotation the pressure is either earlier or later released in the fuel suction chamber The gear racks of all fuel pumps are connected to each other by pinions and form a fuel adjusting spindle, which can be moved several centimetres (depending on the size of the engine). Therefore the fuel output of the pumps can vary from 0 to 100%. In position 1, 2 and 3 of the plunger pump, the plunger capacity is 0.50 and 100%.
Material is milled out of the plunger. The plunger is fitted with two milled helical grooves which lead to the pressurised space.



▲ In this position of the ascending plunger, the suction hole I is open. After closure of the suction hole, the plunger starts the compression stroke to the milled area. The yield is approximately 50%. If the plunger is rotated to position II, the yield is 100%.



▲ Three positions with different yields/capacity. The plunger must slightly ascend before compression starts.

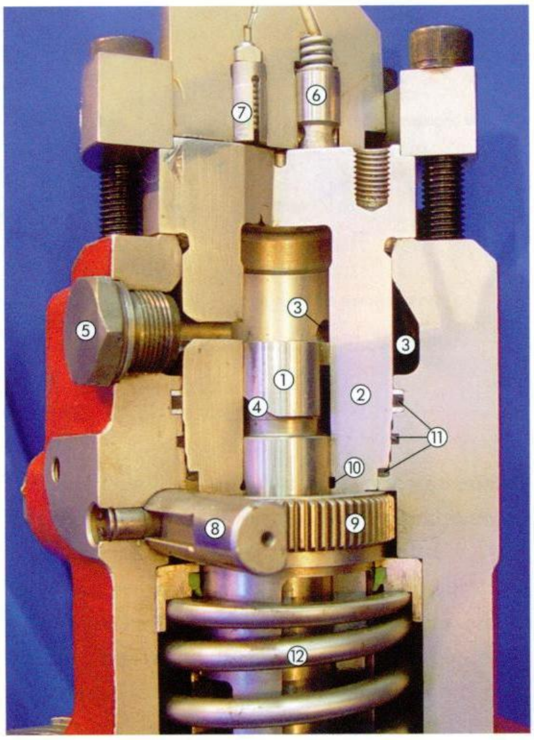


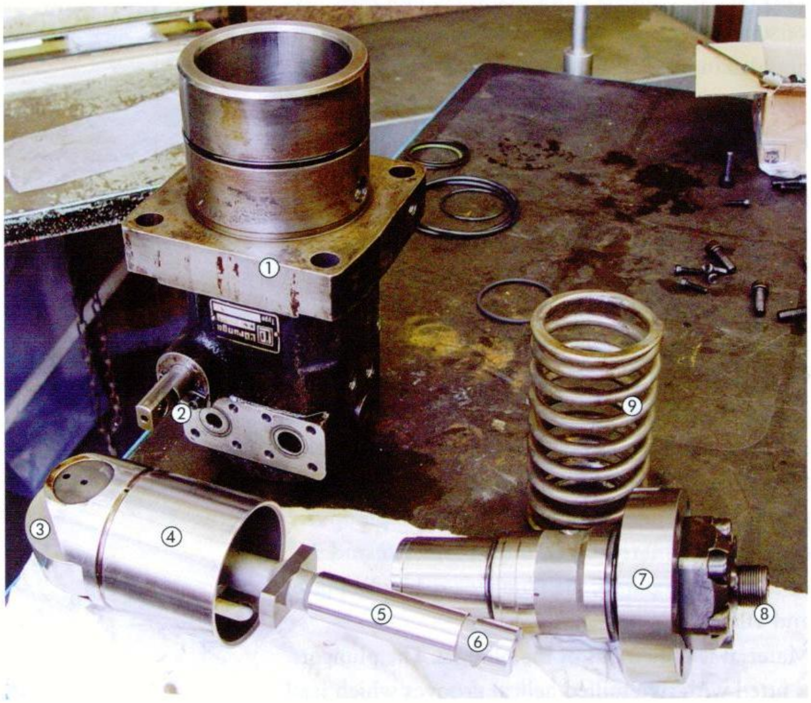
◄ A cross-section of a high pressure fuel pump.

The plunger starts the compression stroke when it ascends in the cylinder. The yield is seen below, 50%. If the channel is positioned in the middle of the suction hole, the yield is zero; there is an open system between the pressure chamber above the plunger and the milled area of the plunger. The fuel travels, but the plunger does not yield.

► A cross-section of a high pressure fuel pump.

1 *plunger*
2 *housing*
3 *suction chamber (2x)*
4 *adjustable plunger*
5 *erosion plug*
6 *pressure valve*
7 *safety valve*
8 *gear rack for plunger adjustment – yield control*
9 *gear wheel*
10 *lubricating oil channel for plunger lubrication*
11 *gaskets*
12 *spring for fast falling plunger when the fuel cam releases it for the descending stroke*





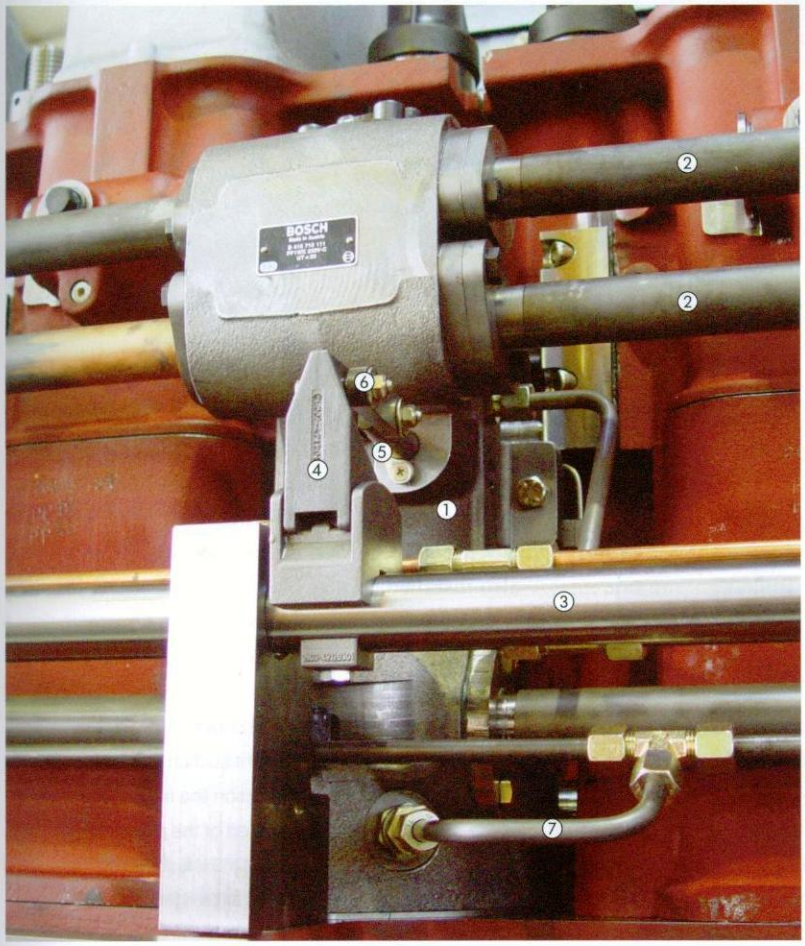
◄ A disassembled high pressure fuel pump.

1 *pump housing*
2 *yield control rod*
3 *roller*
4 *guide*
5 *pump plunger*
6 *milled area*
7 *pump cylinder*
8 *high pressure connection*
9 *spring*

►

Two plungers of high pressure fuel pumps with the associated high pressure barrels.

There are two types: one with and one without lubricating oil grooves in the plunger.



◄

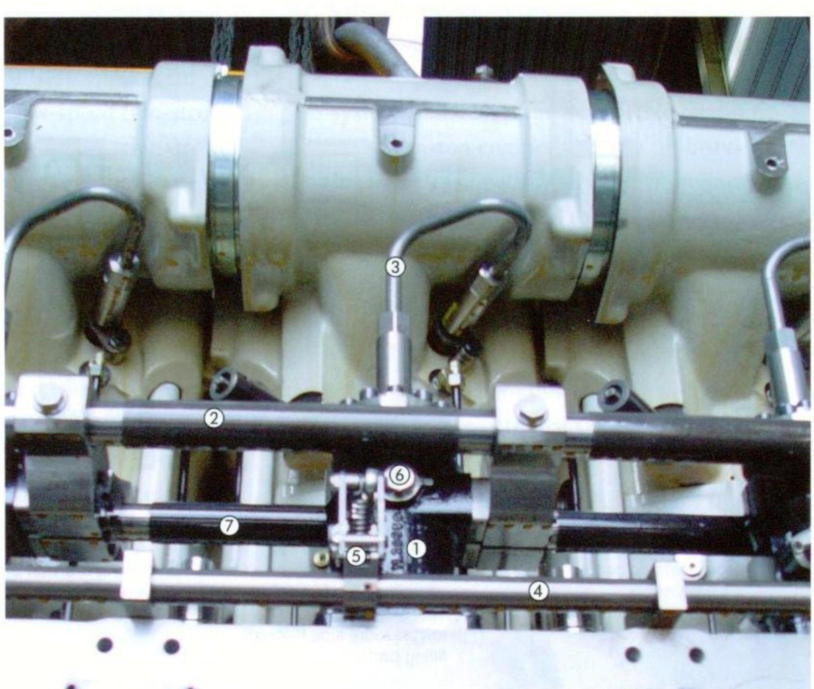
The high pressure fuel pump of a four stroke medium speed diesel engine manufacturer Caterpillar–MaK 32.

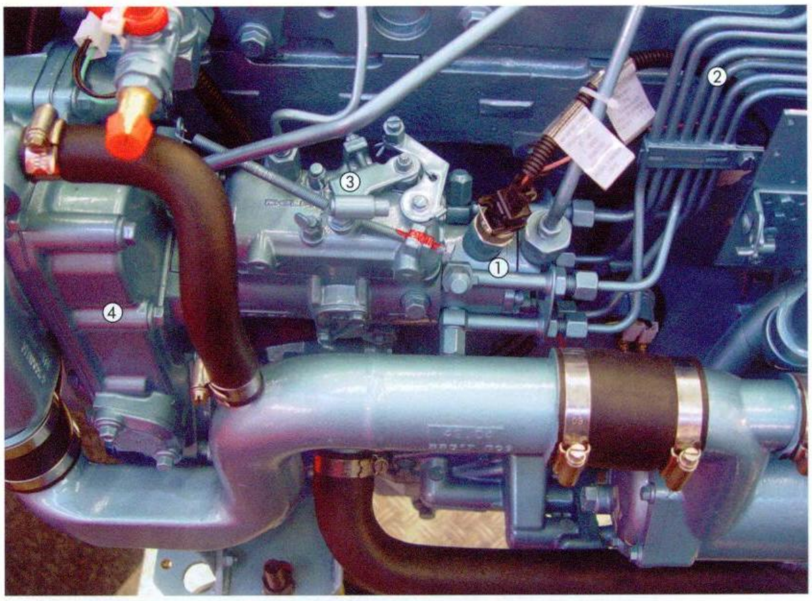
1 *high pressure fuel pump*
2 *fuel supply and discharge pipes*
3 *fuel adjustment spindle*
4 *lever*
5 *gear rack*
6 *adjustable area*
7 *lubricating pipe lines*

►

The high pressure fuel pump of a four stroke medium speed diesel engine manufacturer MAN–B&W 27-32.

1 *high pressure fuel pump*
2 *fuel supply pipe lines*
3 *high pressure fuel pipe lines to the injector*
4 *fuel adjustment spindle*
5 *lever*
6 *gear rack*
7 *fuel return pipe lines*





▶

A high pressure fuel pump in horizontally positioned in a small four stroke high speed diesel engine.

1 *fuel pump*
2 *high pressure fuel lines to the injectors*
3 *capacity adjusting fuel pump*
4 *gear drive*

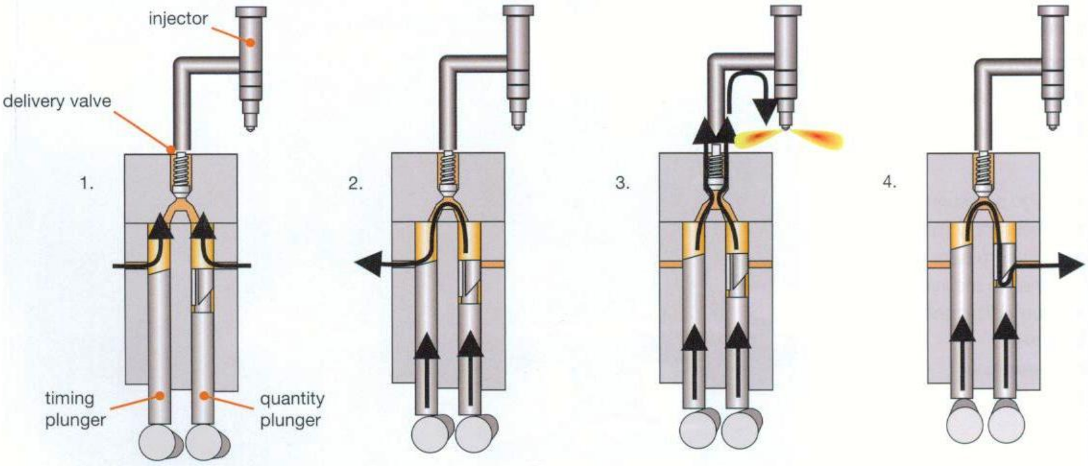
9.16.2 Plunger pumps with initial and final adjustments

The pump can also be fitted with initial adjustment controls by machining the top of the plunger. Small plungers are milled on one side of the top of the plunger. In larger plungers the lateral forces rise too significantly as a result of the uneven radial load. There are two control guides mounted opposite each other, which cancel out any radial forces. Of course two spill ports are also installed.

▼

Description of the operation of the fuel pump of a Wärtsilä 64 four stroke-diesel engine.

With the left plunger the start of the fuel injection is determined and with the right plunger the duration of the injection and so also the end of the injection.
Figure 1: Charging: both plungers are in the lowest position. Via the left and right suction lines, the spaces above the plungers are filled. The fuel cams are not touching the rollers.
Figure 2: Start of the ascending stroke. Both plungers are driven by the fuel cams. The right suction line is closed by the right plunger. Via the left suction line fuel can flow out.
Figure 3: Fuel injection. The speed of the plungers is maximal. The left plunger closes the left suction line and the fuel injection is optimal by the high plunger speed.
Figure 4: the right plunger releases the right suction line and the fuel injection quickly stops. Both plungers are in the highest position.

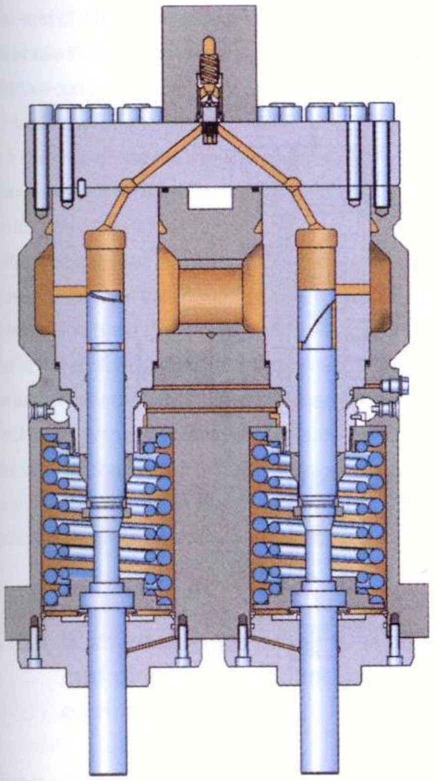


1.
- tappets on cam base circle
- filling of injection pump

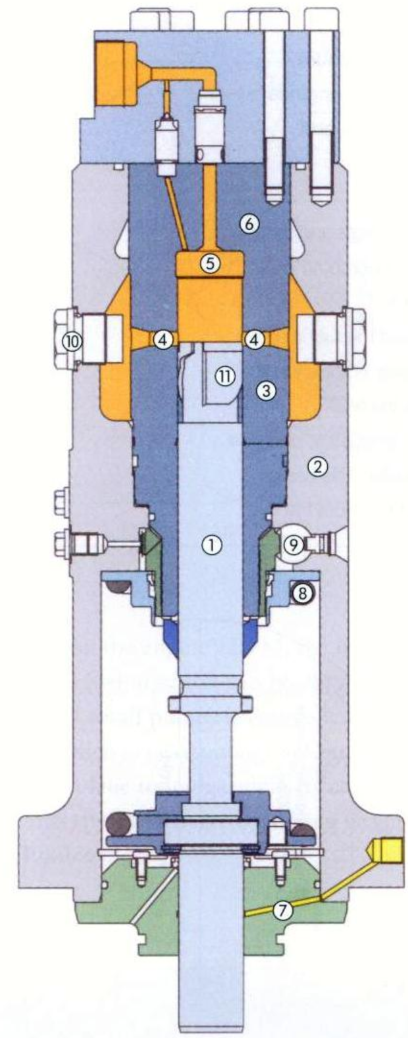
2.
- quantity plunger shuts off spill port
- excessive fuel out to lowpressure side through filling port

3.
- both ports are shut off
- delivery valve lifts
- start of injection

4.
- spill port opens
- excessive fuel out to lowpressure side through spill port
- end of injection



▲
A high pressure fuel pump with two plungers for the four stroke medium speed Wärtsilä 64 diesel engine, fuel H.F.O.



◄
A high pressure fuel pump.

1 *plunger*
2 *pump housing*
3 *cylinder or barrel*
4 *suction chamber*
5 *pressure chamber*
6 *pump cover*
7 *lubricating oil supply*
8 *spring*
9 *control rod*
10 *spreader bolt to avoid cavitation in the pump*
11 *milled area of the plunger*

9.16.3 Plunger pumps with variable injection timing

This is the so-called V.I.T.-system (Variable Injection Timing).
This is used for marine propulsion engines for ships. The system is used in low-speed two-stroke crosshead engines and medium-speed four-stroke trunk piston engines. Propulsion engines have strongly varying loads at normal operating speed. Due to the weather conditions, the load and hull fouling, these are in practice never the same. Generally, a ship propulsion engine operates far below its maximum allowable capacity. This is usually referred to as the '**Maximum Continuous Rating' or M.C.R.** Engine manufacturers have modified the engines' fuel control for this system for a load between 85 and 100%. The idea is to increase efficiency for all loads and therefore decrease the specific fuel consumption.

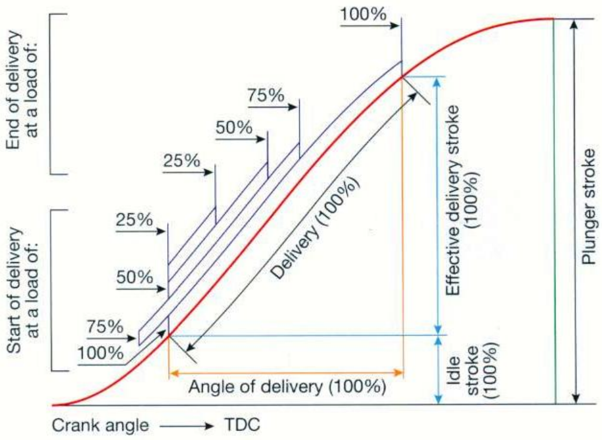
The system ensures that between an 85 to 100% engine load the maximum combustion pressure remains constant.
MAN–B&W has plunger controlled fuel pumps for two-stroke crosshead engines.

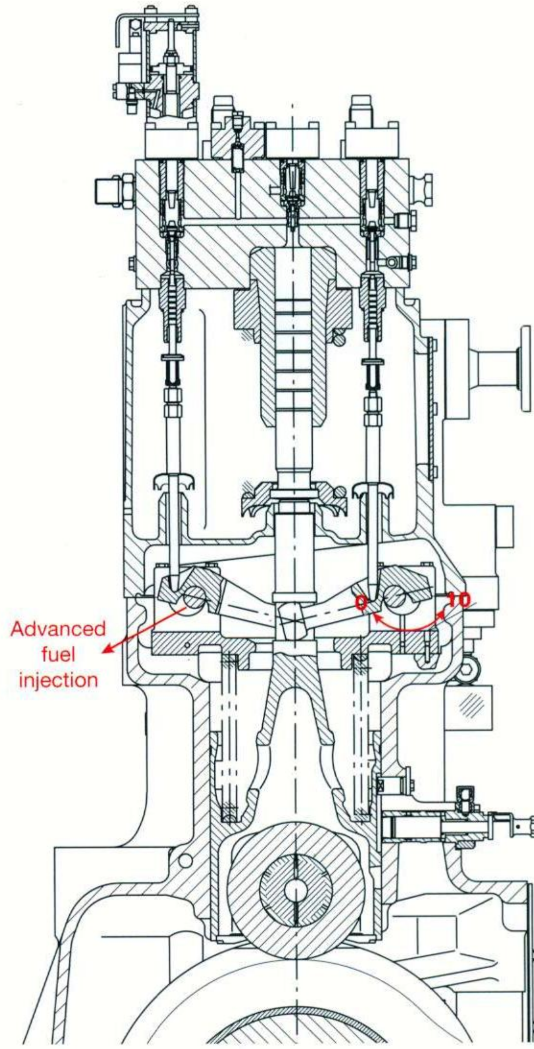
This system is also used in four-stroke medium-speed trunk piston engines that run on heavy oil. This is the case with MAN–B&W engines, in particular the largest series, the L 58/64.
Apart from the much discussed V.I.T.-system, the fuel pumps have an adjustable plunger drive which can adapt the beginning of the fuel injection to the ignition quality of the fuel over the entire engine load range. This system does not depend on the load. The objective is to keep the engine efficiency as high as possible at any speed. The pump settings are related to the fuel quality expressed in C.C.A.I.

▶

An overview of the controls of the high pressure plunger pump of the two stroke crosshead engines from Wärtsilä Sulzer RTA with suction valve and overflow valve.

Under, a diagram with horizontal the crank angle and vertical the plunger stroke of the high pressure fuel pump. In addition, the fuel injection at different loads.
Clearly shown is that at a lower engine load the fuel injection occurs earlier. Normally at decreased power output, the final compression pressure decreases and so the final combustion pressure. To retain the final combustion pressure at partial load, the fuel injection occurs earlier. This has a favourable influence on the efficiency and therefore the fuel consumption.



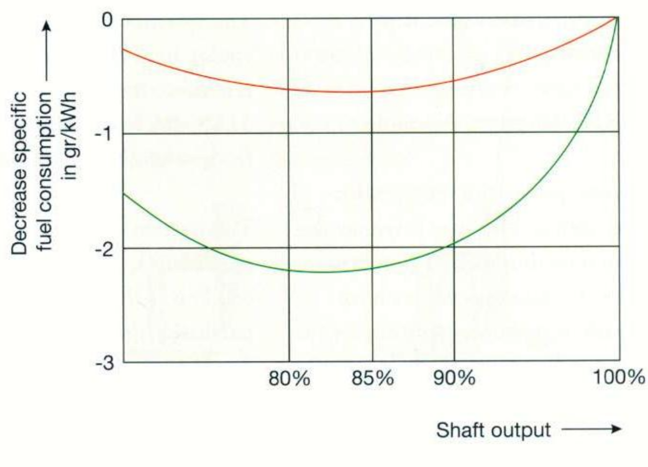


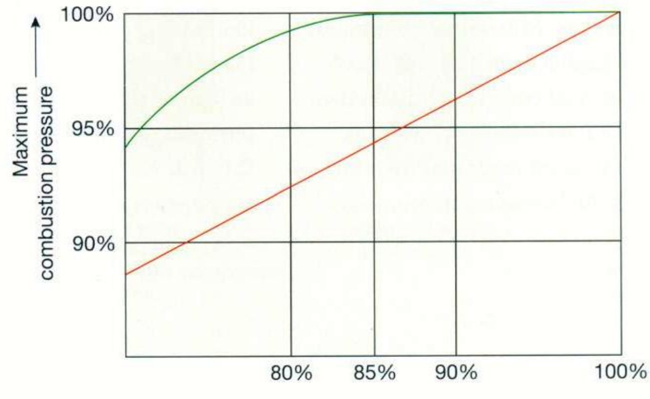
▶

Influence of timing advance at a partial load.

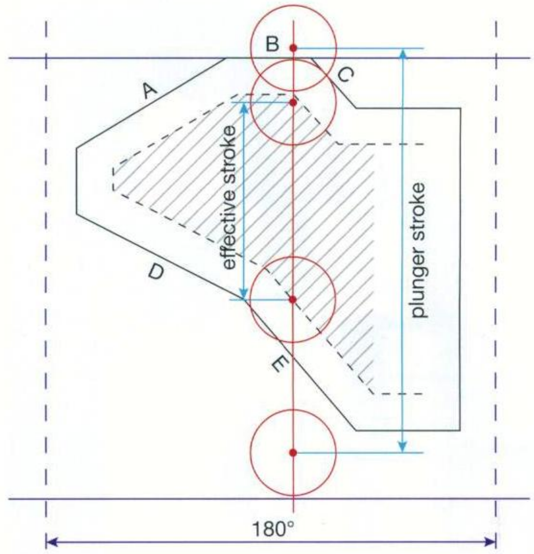
In the both graphs, the difference between a conventional injection with respect to a V.I.T. - system it is clearly shown. At partial load the final combustion pressure is maintained and fuel usage decreases.

Red: conventional injection
Green: V.I.T.- system





The start of the effective compression stroke is regulated by the above three control guides A, B and C. the end of the effective compression stroke is regulated by the lower control guides D and E.
At a low engine load the overflow channels in the pump cylinder are opened and closed by control guides A and D in the left side of the figure.
In this area, the start and finish controls are applied. At increasing load the fuel is injected earlier. From the start of control guide B the start of the injection remains constant and at the end of the injection, the engine load increases. At the end of control guide B, an engine load of 85% of M.C.R. is achieved. With a load increase to 100% of M.C.R., the effective injection start (combustion stroke) is determined by control guide C in such a way that the maximum combustions pressure remains constant.



◄

The plunger surface of a fuel pump of a MAN–B&W two-stroke crosshead engine type MC.

The plungers are machined in such a way that the profile has five control areas, A, B, C, D and E.

- A Injection starts later.
- B Injection remains constant.
- C Injection starts earlier.
- D Final yield.
- E Final yield.

9.16.4 With integrated block fuel pumps and plunger controlled pumps for each cylinder

This is often seen in small to medium-sized high-speed four-stroke diesel engine in the categories I and II (0 tot 5000 kW). One of the advantages is that, for instance, in a sixteen cylinder V-engine ,only one driving device is required: the block fuel pump as opposed to sixteen separate fuel pumps with a cam shaft drive. Another advantage is that a complete block pump can be changed and overhauled by a specialised company. To ensure that the injection delays are identical for each cylinder, the high-pressure fuel lines to the injector are of equal length. They are often intrinsically bent in order to make them fit between the block pump and the cylinder in question.

Depending on the engine's RPM, the moment of the initial fuel injection can be adjusted. The output of all small pumps is controlled by one gear rack which is horizontally mounted in the pump block. Due to increasingly strict emission regulations special systems have been designed for this technology.

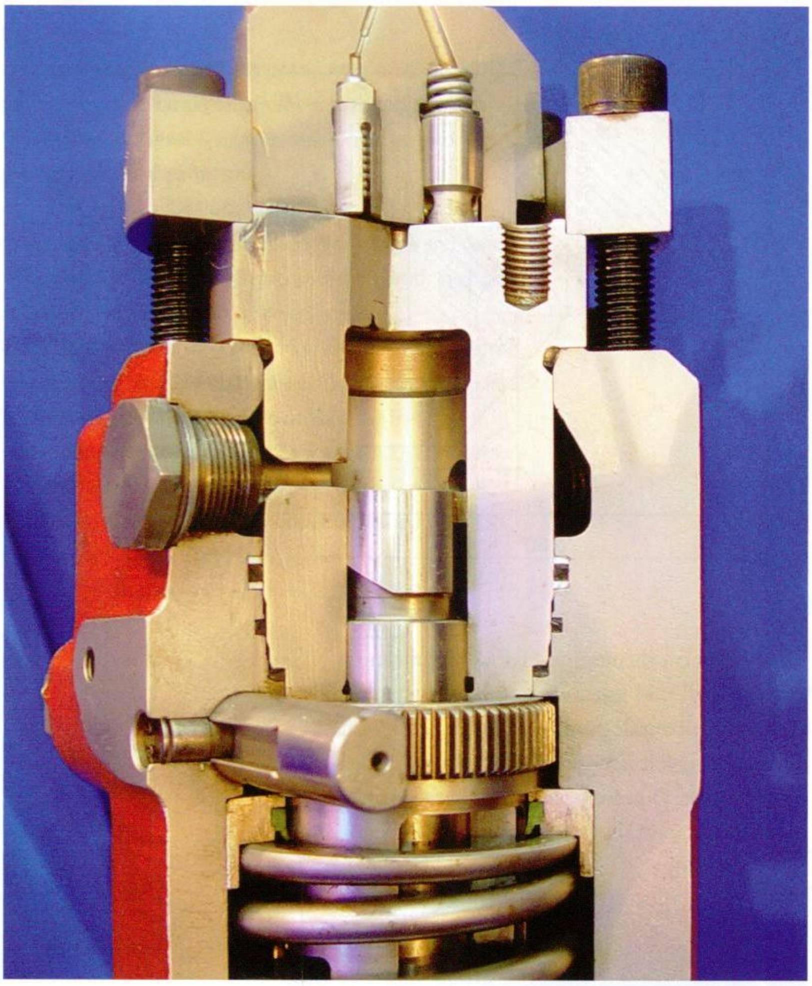
◄

A block fuel pump for a twelve cylinder four stroke high speed V-diesel engine.

►

Length of the high pressure fuel lines.

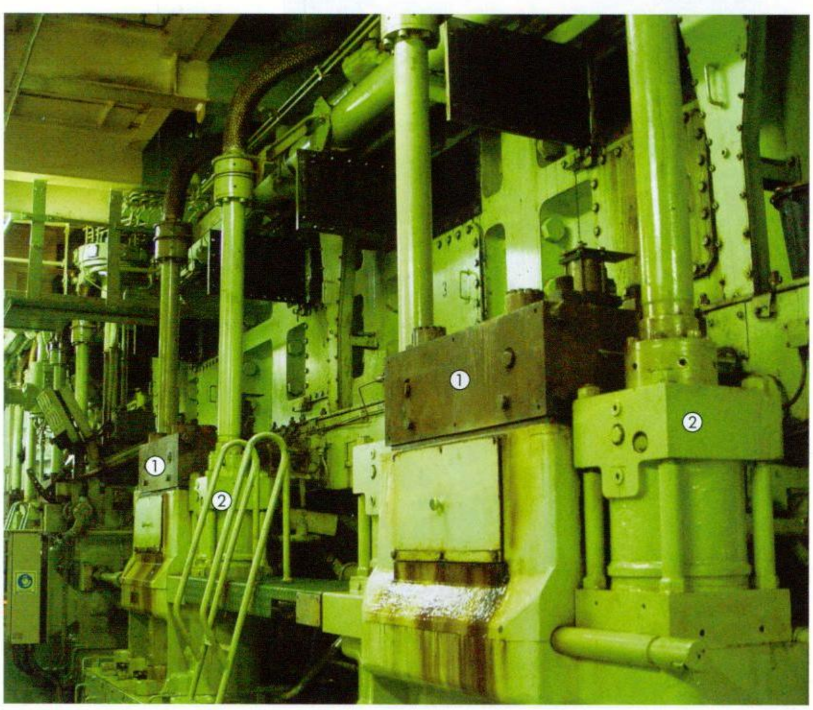
To keep the injection delay of every cylinder equal the pressure pipe lines to every fuel injector are equal in length. The cylinders furthest from the pump have high pressure fuel lines that run more or less directly to the cylinders.
The cylinders nearest to the pump have fuel lines of equal length that are so bent that they end at the relevant cylinder.



▲ An exploded view of a fuel pump.

▼ Two high pressure fuel pumps mounted in one housing in a Wärtsilä RTA 84 crosshead engine.

1 *pump housing with two high pressure fuel pumps*
2 *hydraulic pump for an exhaust valve drive*



9.17 Valve-controlled fuel pumps

These are found, for instance, in low-speed two-stroke crosshead engines by Wärtsilä Sulzer RTA.

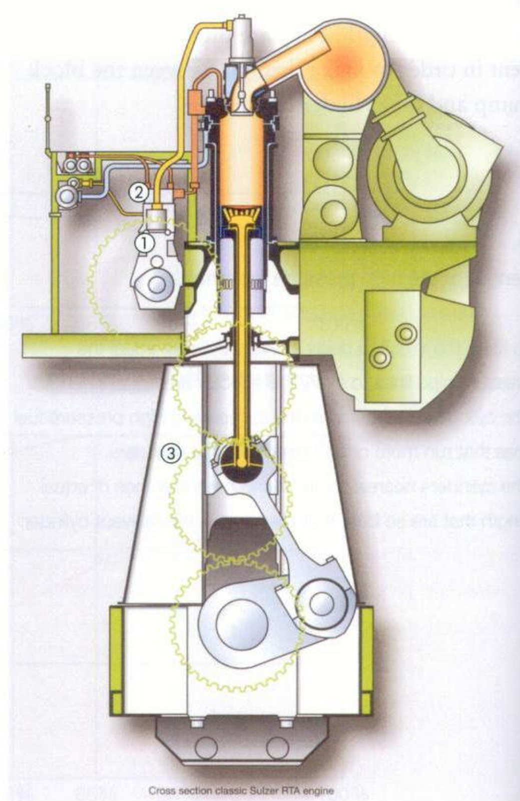
In these engines the fuel pumps of two cylinders are located in one block. Each pump is driven from the cam shaft with cam, cam rollers and cam roller guides.

Fuel quantity is regulated using a suction port-and a spill port as opposed to the rotation of the pump cylinder around a machined plunger. So the pump has initial and final adjustment controls.

▼
A low speed two-stroke crosshead engine from Wärtsilä Sulzer RTA with valve regulated fuel pumps.

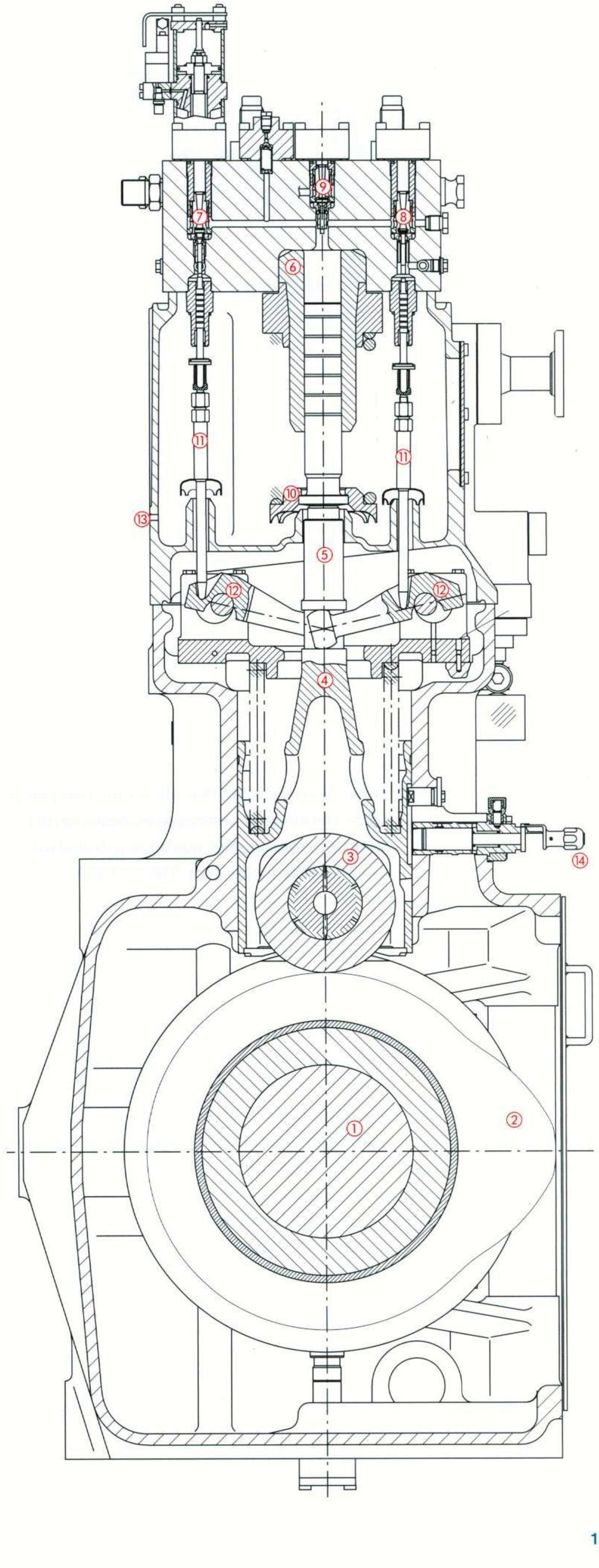
They are driven by the camshaft. The transmission from the crank shaft and camshaft is driven with gear wheels. The fuel pump positioned behind the camshaft driven high pressure lubricating oil pump for the operation of the exhaust valve.

1 *lubricating oil pump for the operation of the exhaust valve*
2 *high pressure fuel pump, behind lubricating oil pump*
3 *gear wheels*



▶ A valve regulated high pressure fuel pump for Wärtsilä RTA two stroke crosshead engines.

1 *camshaft*
2 *cam*
3 *roller*
4 *roller guide*
5 *pump plunger*
6 *pump cylinder or barrel*
7 *suction valve*
8 *overflow valve*
9 *pressure valve*
10 *plunger springs with seats*
11 *adjustable valve stems*
12 *levers*
13 *discharge excess fuel*
14 *mechanism for stopping pump*



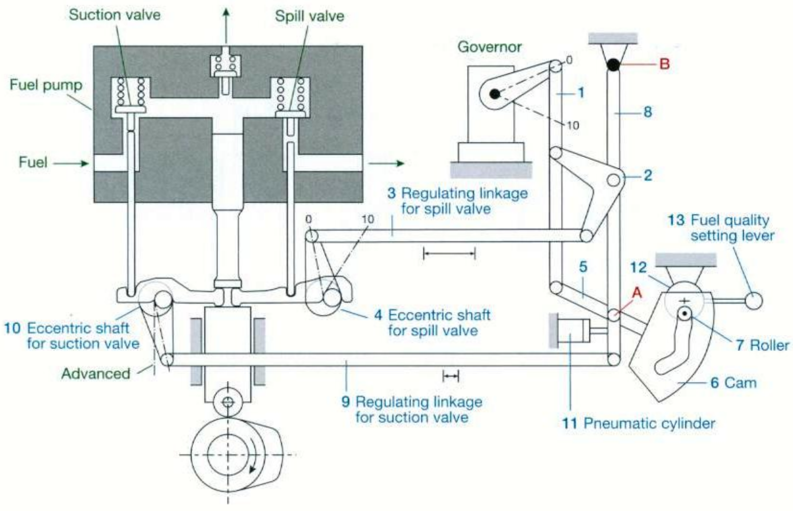
Pump operation

At the downward stroke (overpressure) of the plunger the piston valve is mechanically opened by a lever after which the fuel flows into the cylinder at a certain overpressure (5 to 10 bar). The pressure valve of the pump is still closed. At the upward piston compression stroke, the effective pressure stroke will commence as soon as the piston valve has been closed automatically by the lever. The overflow valve is closed. The pressure valve opens as a result of the rapidly increasing fuel pressure. At a certain moment the overflow valve is mechanically opened by the lever.
The fuel pressure above the plunger drops, the pressure valve closes and the effective compression stroke is now terminated. During the remainder of the upward stroke of the plunger the fuel flows back to the suction side of the pump.

Lever operation

The levers are moved up and down by the gearing mechanism of the plunger. Both sides of the lever are equipped with eccentric discs which are mounted on adjusting spindles. Rotation of the adjusting spindles alters the closure of the suction valve and the opening of the overflow valve.
If the plunger has no output, the overflow valve has opened before the suction valve has shut.
This Wärtsilä Sulzer system is also equipped with the V.I.T.-system to maintain a constant maximum combustion pressure between 85 and 100% M.C.R.

Fuel regulation with 'VIT' combined with manual fuel quality setting 'FQS'

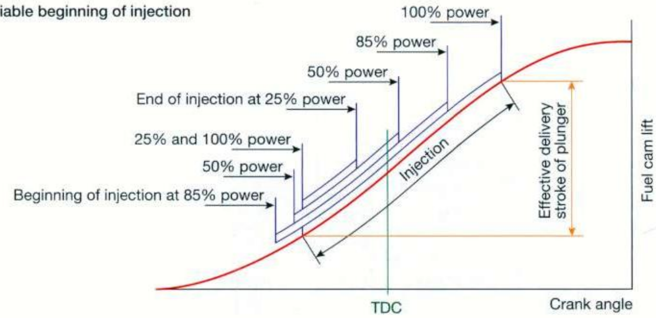


◄

A valve regulated high pressure fuel pump for Wärtsilä RTA two stroke crosshead engines.

The levers for the load and V.I.T.-controls are electronically controlled. Due to this, it is possible to, for example, to retard the injection and so lower the maximum combustion pressures (and the maximum combustion temperatures) and reduce the nitrous oxide emissions.
Furthermore in a similar fashion, the operation of the hydraulic driven exhaust valve is achieved. This is called at Wärtsilä Sulzer, the Variable Exhaust Control, V.E.C.- system.

Variable beginning of injection



Furthermore, a system that allows the fuel pumps to start the effective compression stroke earlier with a deteriorating fuel quality is as standard included in the fuel system. This system is known as the Fuel Quality Setting, F.Q.S.. Once again the fuel quality is expressed in C.C.A.I.-values. The higher this value, the earlier the fuel injection.

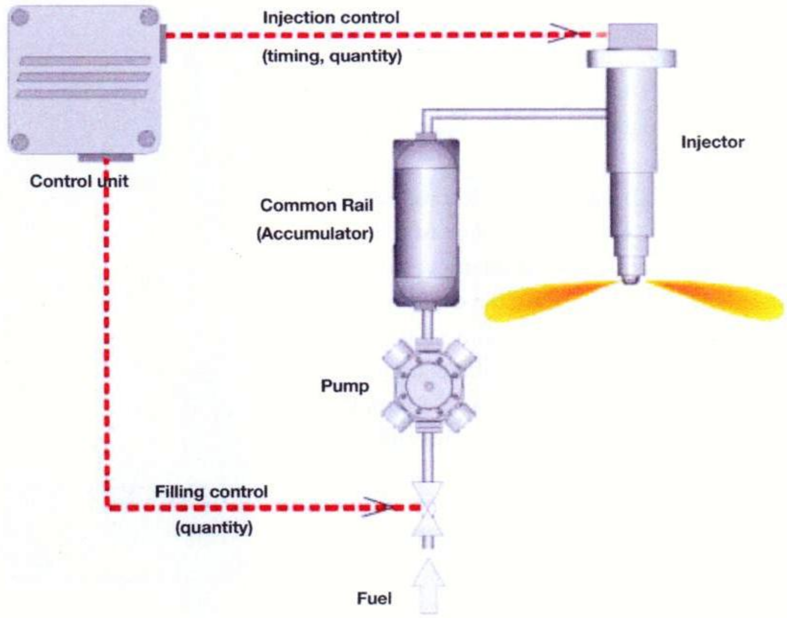
9.18 Common-rail system

In this system, the one century old system of the cam shaft driving the high-pressure pump, has been abandoned.
In the numerous systems that are currently applied, the fuel is brought to extremely high pressures of 1000 to 2000 bar in various ways. The fuel injection is electronically controlled.

►

The principal build of a common rail fuel system for diesel engines.

The fuel is drawn up by a high pressure plunger pump and pumped into a spacious collection pipe; this pipe is known as the 'common rail'. The fuel pressure in this pipe is kept constant. A control-unit regulates the fuel injection with the injector.
The greater the fuel demand, the greater the fuel supply. This is measured using a quantity measurement on the suction line of the pumps.



The common rail system was introduced to the smaller diesel engines for motor cars and lorries towards the end of the last century. **The system has advanced so quickly over the past few years that it is now applied to all diesel engine categories. In particular the increasingly strict emission regulations has required that engine manufacturers choose these tremendously flexible fuelling systems. The common rail system allows for the reduction of the specific fuel consumption, especially at partial load.**

General construction of a common rail system

The fuel supply pressure to the injectors can be attained with either one pump or with more. The pumps can be driven traditionally, from the cam shaft, but also directly from the crank shaft using a gear transmissions or a gear belt (timing belt).

In larger four-stroke engines every cylinder or every second cylinder has their own high-pressure fuel pump which is connected to a high-pressure collecting pipe. The electronically controlled and hydraulically operated supply valves to each injector ensure the fuel supply from the collecting pipe to the injectors.

Advantages of the common rail system

The injection curve of the fuel can be adjusted to guarantee optimal combustion at every engine load.

Explanatory note

Both the moment of the initial- and the final fuel injection, as well as the total injection time can be predetermined very precisely, thus allowing for optimal combustion at all rpm's and every engine load. This way the specific fuel consumption, the emission of nitrous oxides ('NOx') and fine soot particles are reduced to a minimum.

This has the following benefits:

- A lower specific fuel consumption especially at partial load.
- Lower emission of pollutants such as soot and nitrous oxides. At partial load the emission of soot particles is considerably lower.
- A consistently high fuel injection pressure, thus maintaining an optimum combustion process at every engine load. In existing diesel engines with a conventional fuel injection system the combustion quality at partial load is mediocre. This is caused by a much lower fuel injection pressure as well as decreasing final compression pressures and -temperatures.
- Some adjustment, such as the moment of injection, the time and the pressure can be made during the process. In two-stroke crosshead engines one can also modify the opening and closing of the exhaust valve, and in medium-speed four stroke engines the opening and closing of the inlet valves.
- Driving high pressure fuel pump(s) is much simpler because not every cylinder requires a separate pump.
- In conventional fuel injection systems the high-pressure fuel pumps have often been in use for many years and will therefore represent a proven technology.
- The opening and closing of the valves occurs more meticulously as a result of the electronic controlling unit, therefore the thermal load is more evenly distributed over the various cylinders.
- All data is stored and settings are improved where necessary, which increases the life-time of the engine components.
- Also, low RPM can be achieved. This is especially advantageous for large two-stroke crosshead engines, which directly drive a fixed propeller. The result is a rapid increase of the RPM and an improved stop- and turn ability in these engines. Cylinder lubrication dosage and control of the starting valves can now be regulated and adjusted electronically.
- The required software for electronically controlled systems can be upgraded throughout the entire life time of the engines.

Generally, one can say that in applying electronically controlled fuel systems, often referred to as common rail systems, it is easier to comply with present day requirements for modern diesel engines. These are lower specific fuel consumption and the reduction of pollutants in the exhaust gases. Engine manufacturers that produce for the international market certainly take these into account. Most engine manufacturers either already have common rail systems or are in the process of developing one. A significant improvement is that at partial load there is a remarkable reduction of the so-called 'visible smoke'. This smoking used to be one of the main problems of diesel engines not equipped with common rail systems. It is presently also easier to control and reduce engine vibrations.

9.18.1 Example 1: common rail systems

Four-stroke medium-speed diesel engines by MAN–B&W, fuel H.F.O. – Engines Category III, Engine types 48/60 B en 32/40

At the design stage the following was established:

- The system must be able to inject fuel up to 700 cSt. at 50 °C.
- The fuel temperature can rise to 150 °C to achieve the correct injection viscosity of 10 to 12 cSt.
- An excessive length high-pressure supply rail requires radial connections for connections with the injector. This results in extremely high material stresses. In larger diesel engines these problems increase as the internal inner diameter of the rail increases.
- In instances of a reduced accumulator volume it is virtually impossible to reach equivalent injections for each cylinder. Considerable pressure fluctuations in the pipes may ensue.
- For larger engines one would have to resort to individual rails for each cylinder.
- A large pressure accumulator also proved to be insufficient.

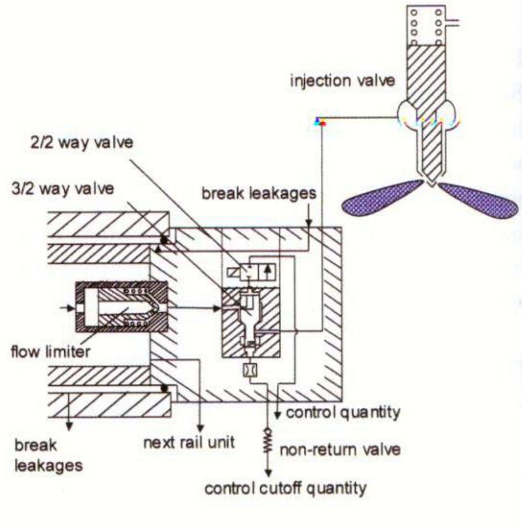
These findings have been incorporated in the design and demonstrate that a six-cylinder engine requires at least two high-pressure pumps with two independent high-pressure supply lines.

Consequences for the diesel engine

- Increasingly strict regulations with regard to the emission of pollutants create high demands on the fuel injection system and the combustion.
- The common-rail system has obvious advantages when compared to conventional injection systems.

General comment

The common rail is particularly effective at partial load operation. At full load the differences with the normal high-pressure fuel systems are negligible.



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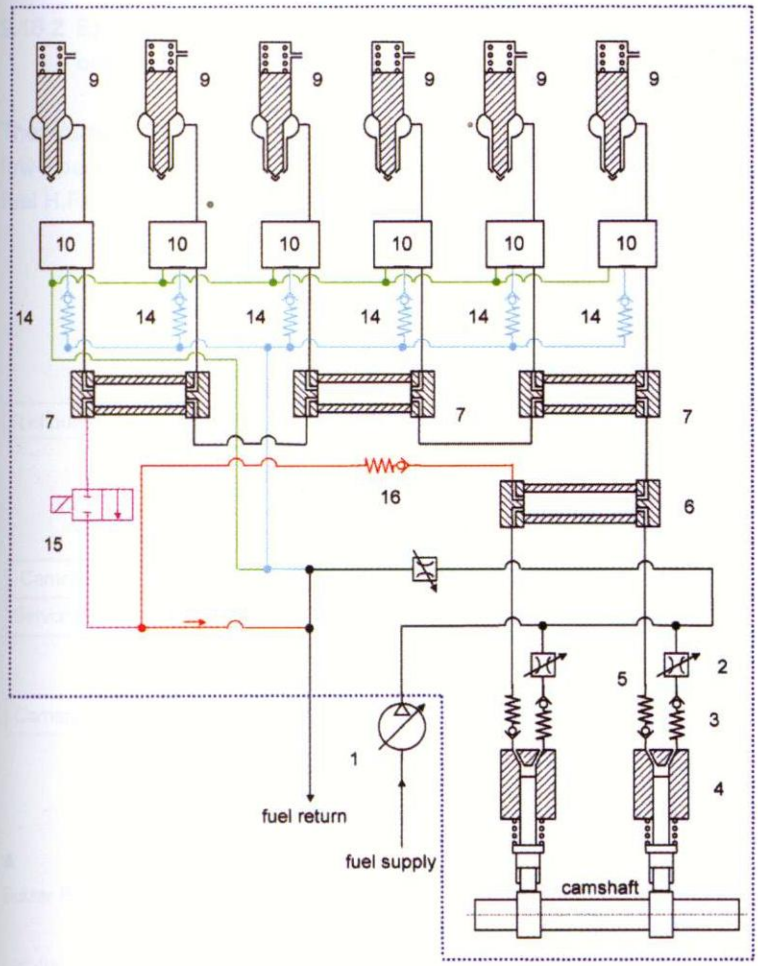
The operation of the accumulator.

Every accumulator unit 7 contains in its cover components and connections, which serve for fuel supply and transmission as well as for fuel injection control.
On its way from the accumulator unit 7 to the 3/2-way valve and then to the injector, the fuel is passed through a flow limiter. A spring-loaded piston in this component carries out a stroke for each injection, which is proportional to the injection quantity and returns in its original position in the time between the injections.
Should the injection quantity exceed however a specified limit value, the piston will be pressed to a sealing seat at the outlet side at the end of the stroke and will thus avoid permanent injection at the injector.
The 3/2-way valve inside the accumulator cover is electro magnetically activated by the control system and permits the high-pressure fuel to be supplied from the accumulator unit, via the flow limiter, to the injector.
With this valve, it is possible to change the pre- and post injection.
A safety valve (16) is arranged at the pump accumulator, which opens if a specified pressure is exceeded. The high-pressure pipes and accumulator units are designed with double walls in order to prevent fuel from penetrating to the outside in the case of leakage or break of connections. In this case, the operating personnel will be warned by means of the float lever switch. For start-up of the cold engine with heavy fuel oil, the high-pressure part of the injection system is heated by means of circulating hot heavy fuel oil. For this purpose, the flushing valve (15), located at the end of the last accumulator unit connected in series, will be opened pneumatically.
By the use of the separate 3/2-way valve there is only pressure at the injection valve during injection.



◄

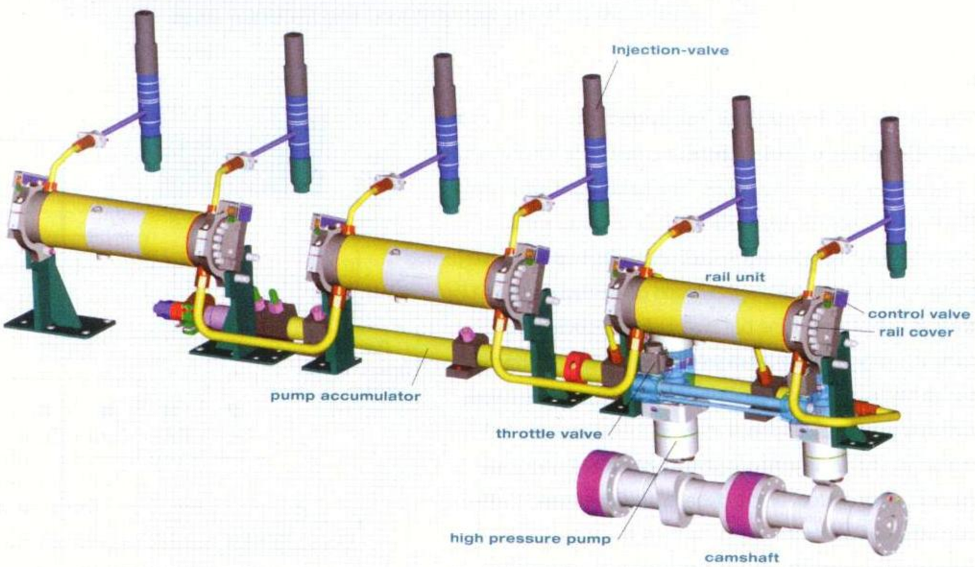
The common rail system of MAN–B&W for the four stroke engines.

A low-pressure fuel pump (1) delivers the fuel via electromagnetic activated throttle valves (2) and suction valves (3) to the high-pressure pumps (4), which force the fuel into the pump accumulator (6) by means of pressure valves (5).
Each pump is connected to the pump accumulator (6) which collects the fuel delivered by the pumps.
From the pump accumulator the fuel flow goes to the accumulator units (7), which are connected in series, to the so called common rail. The accumulator units consist of a compact tube, which is on both front sides equipped with an accumulator cover. The accumulator covers contain radial connections for the high-pressure pipes leading to the injectors (9) as well as for the connecting pipe to the next accumulator unit.
The tube itself doesn't contain any radial drillings and is therefore easy to produce and very resistant to high fuel pressures. Drive of the high-pressure pumps 4 is, as known, effected by cams arranged on the engine camshaft. The delivery quantity of the high-pressure pumps is calculated by the electronic control system on the basis of an evaluation of the fuel pressure indicated by the rail pressure sensor and the corresponding operation condition of the engine. The electro-magnetically activated throttle valve (2) in the low pressure pipe will now suitably meter the fuel quantity supplied to the high pressure pumps.

►

The common rail system of a MAN–B&W four stroke medium speed engines operating on H.F.O.

Due to this there is a complete common rail system for two cylinders.



Results

- Smoke reduction over the entire load range.
- Nitrous oxides reduction, especially at partial loads.
- Lower specific fuel consumption especially at very low loads.

Electronic controlling system

Since there is no mechanic 'back up'-system present, the electronic control has to be of superior quality and safety.

▶

Four graphs showing the changes with use of the common rail system in MAN–B&W four stroke engines.

Horizontal: the engine load

Blue line: conventional engines with a high pressure fuel pump per cylinder

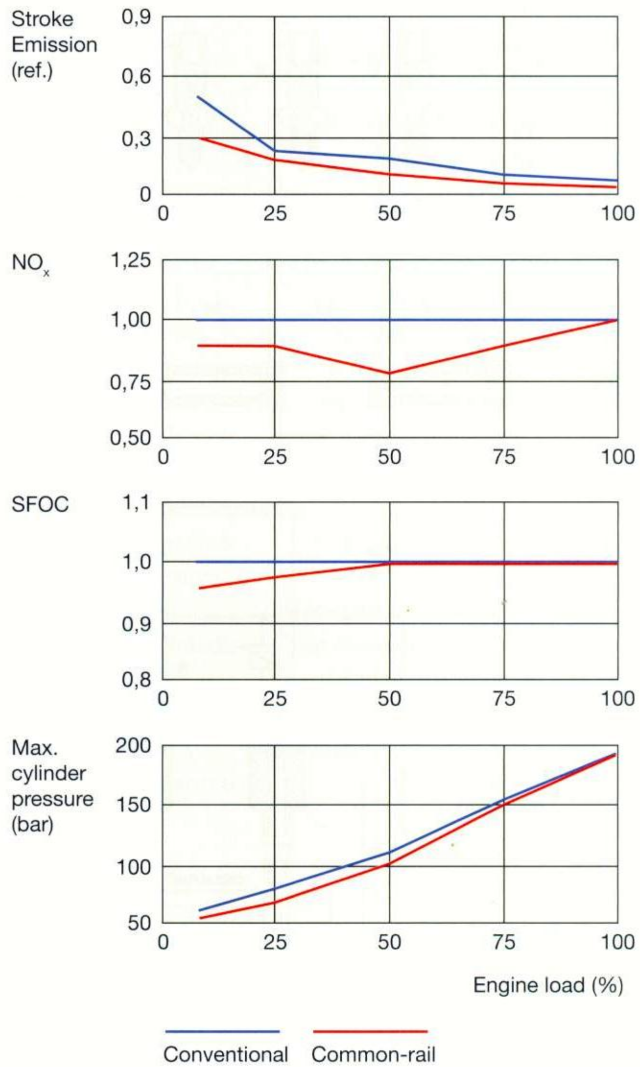
Red line: engines with a common rail system

Figure 1: vertical: the smoke opacity number, at partial load a great deal lower

Figure 2: the nitrous oxide emissions in percent, at partial load much lower

Figure 3: the specific fuel usage in percent, at partial load lower

Figure 4: the maximum cylinder pressure in bars, at partial load slightly lower



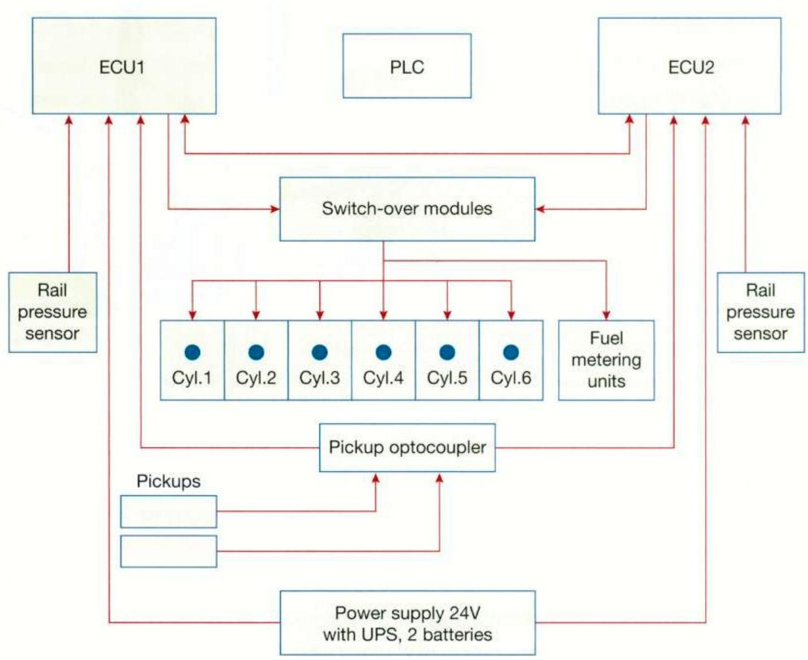
▶

The controls of the common rail system of MAN–B&W four stroke engines.

The two electronic control units ECU1 and ECU2 are responsible for the solenoid valve control, the high pressure pump control and therefore the speed governing. Each ECU controls half of the engine but is also able to control the complete engine. All necessary sensors, the power supply and all field bus connections are doubled. So a single failure will never lead to an engine stop.

The single PLC is only responsible for communication with the ship alarm system or the diesel control room and for the Man Machine Interface.

The electronic controls allow pre and post-injection and of course control the fuel pressure of the common rail system. Due to this optimal combustion occurs at every engine load.

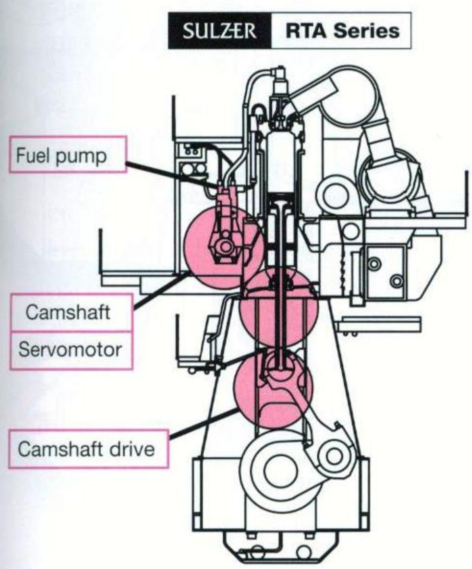


9.18.2 Example 2: common rail systems

The Wärtsilä-Sulzer RT-FLEX-systems for low-speed two-stroke crosshead engines, fuel H.F.O. – Engines Category IV

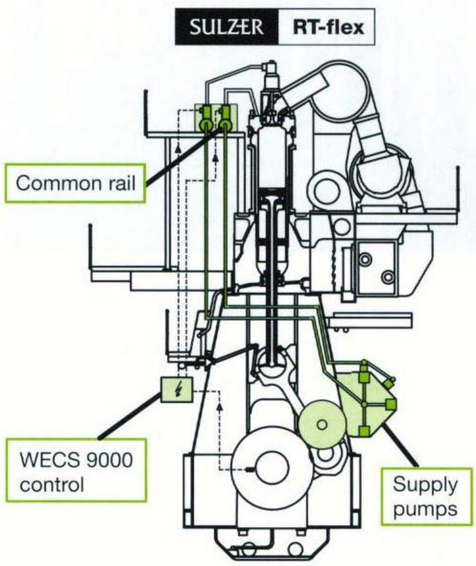
Note

There is no cam shaft! There is a common rail system for lubrication of the exhaust valves. Two electrically driven pumps incorporated in the supply unit with the fuel pumps ensure that the system retains the correct pressure.



Sulzer RTA series.

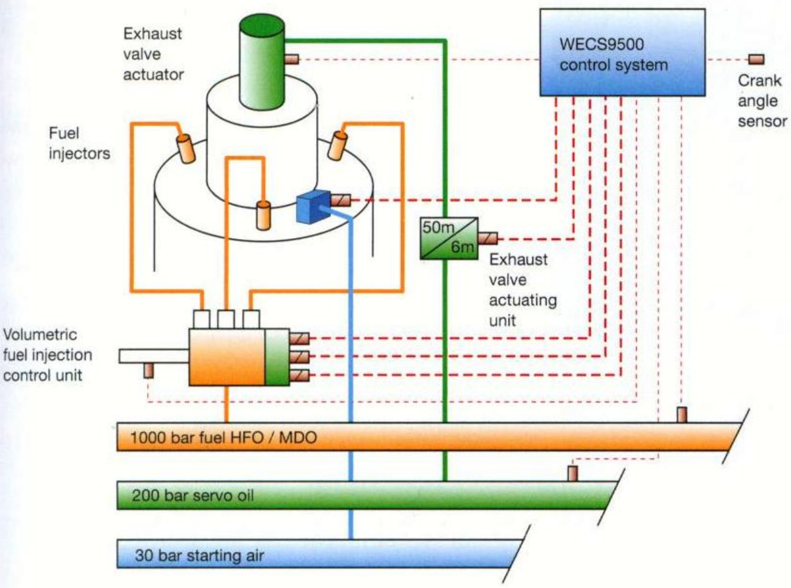
Left: the conventional system with a camshaft driven high pressure fuel pump on each cylinder. Between the crank shaft and the camshaft gear wheels.



Sulzer RT-flex series.

Right: the common rail system with a crank shaft driven series of high pressure fuel pumps in one block.
The fuel travels via the collection pipe lines or the common rail to the injectors. A engine management system controls the supply of the fuel to the injectors. This system is known as the Wärtsilä Engine Control System (W.E.C.S.) 9000, 9500 etcetera.

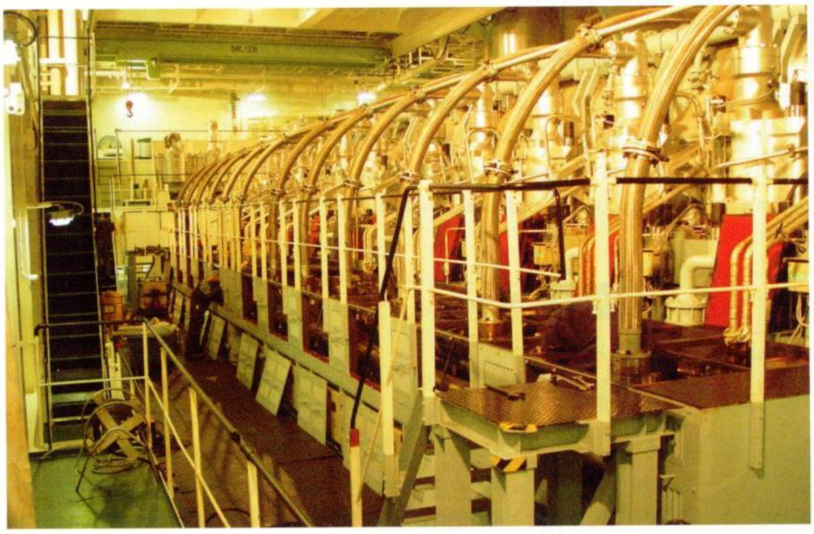
The difference between the conventional high pressure fuel system and the common rail system in Wärtsilä Sulzer two stroke crosshead engines.



A complete schematic diagram of the common rail system for the Wärtsilä RT-FLEX crosshead engines.

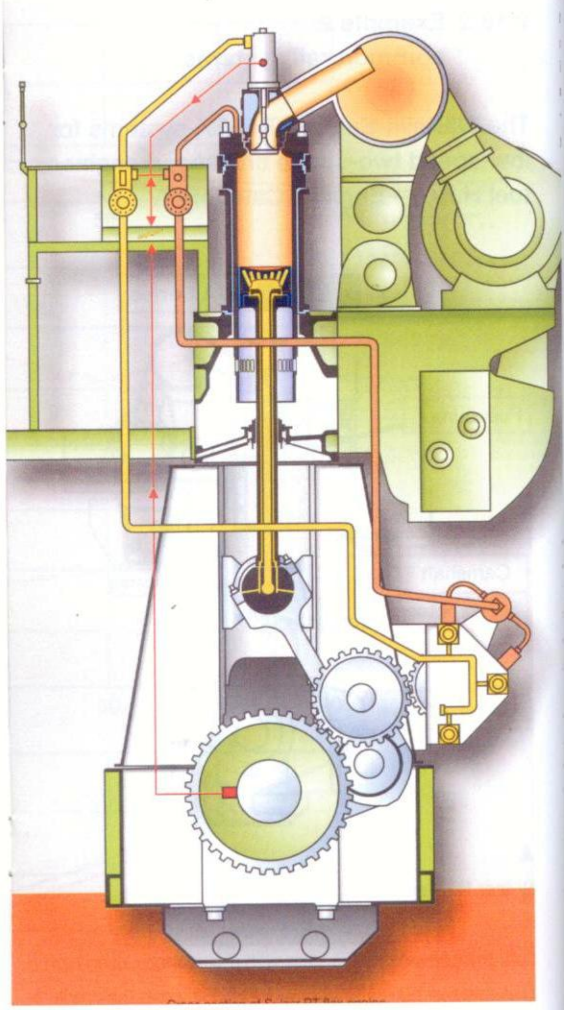
The system comprises the following components:

- A crank shaft driven high pressure fuel unit and plunger pumps with adjustable yield.
- The cam shaft for driving the individual fuel pumps and exhaust valve has been removed.
- The high pressure fuel unit pumps in a large supply fuel line (common rail) that runs past all the cylinders. Pressure approximately 1000 bar.
- With a control unit the fuel supply to the three injectors per cylinder is tuned to the requirements. The control unit is electronically driven by a W.E.C.S. 9500-system (Wärtsilä Engine Control System) and hydraulically operated by a lubricating oil system with a pressure of approximately 200 bar. This pressure is supplied by a separate pump system. The 200 bar line is also used for operating the central exhaust valve.

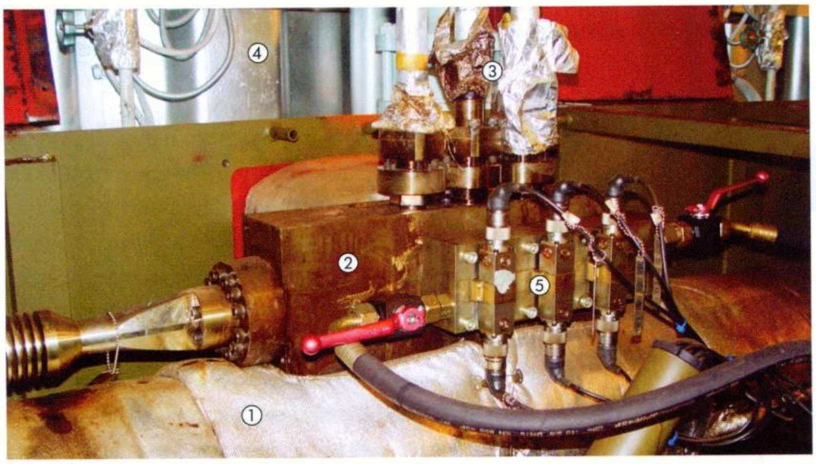


▲
The common rail system on a Wärtsilä Sulzer RT-FLEX 96 C two-stroke crosshead engine on board the containership 'P&O Nedlloyd Mondriaan'.

The space under the walkway for the cylinder heads is fitted out for the common rail system, the high pressure lubricating oil system for the operation of the exhaust valves, all appendages and control equipment.



►
The common rail system on a Wärtsilä Sulzer RT FLEX-motor.



▲
A high pressure fuel division gate for the three injectors of one cylinder.

1 *common rail with a pressure of 1000 bar*
2 *division gate*
3 *high pressure fuel line to the injectors*
4 *cylinder liner*
5 *valve control of the injector high pressure fuel lines*

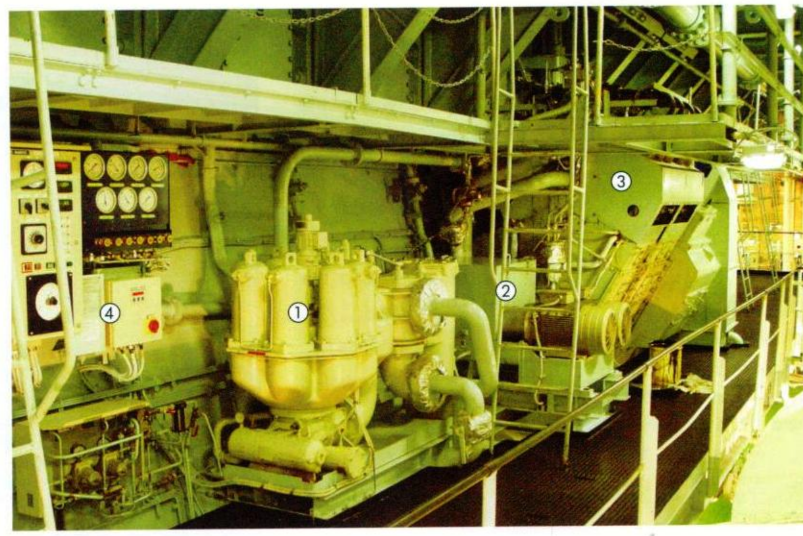
Advantages

- 'Smokeless' exhaust gases, no visible smoke.
- A potentially low RPM; the minimal RPM is approximately 10 to 12% of the nominal RPM. At this low RPM one injector per cylinder is switched off.
- Lower specific fuel consumption at partial load and longer life of the parts (less soot formation at low loads).
- Easier engine-setting and thus reduced maintenance. The running settings determined at construction are automatically maintained.
- Reduced maintenance costs through precise fuel injection controls producing a more evenly balanced cylinder power output for the entire engine as well as a more balanced thermal load of all cylinders.
- The high-pressure fuel unit consists of a number of four-stroke engine fuel pumps which already have been in use for several years.

- High operational reliability, because even if one fuel pump were to fail the capacity of the remaining pumps can still easily supply a sufficient amount of fuel.
- The common rail is situated next to the cylinders on the upper level and is easily accessible for inspection and repair.
- The fuel supply to the three injectors is regulated at once by one control unit, but is separately checked per individual injector.
- The units regulate the time of injection, the amount of fuel and the type of injection.

Comments

- Free choice of the injection pressure.
- Evenly levelled pressure in the common rail and other high-pressure lines.
- The injector may be checked and if necessary switched off independently.
- The fuel system is completely separated from the lubricating oil system of the engine.



▲
The common rail system of a Wärtsilä Sulzer RT-FLEX engine.

1 *lubricating oil filter*
2 *lubricating oil pumps for the hydraulic system*
3 *high pressure fuel pumps*
4 *local operation of the engine*

The high pressure fuel unit and the high pressure hydraulic system are positioned on the first floor of the engine. The high pressure fuel unit is driven from the crank shaft as is the high pressure lubricating oil system. Also the high pressure part of the hydraulic system which with the help of electro motors is used in stationary engines to bring and keep the system under pressure.

◄
A complete module for a Wärtsilä Sulzer RT-FLEX-motor.
Shown are the lubricating oil pumps are built into the fuel module.

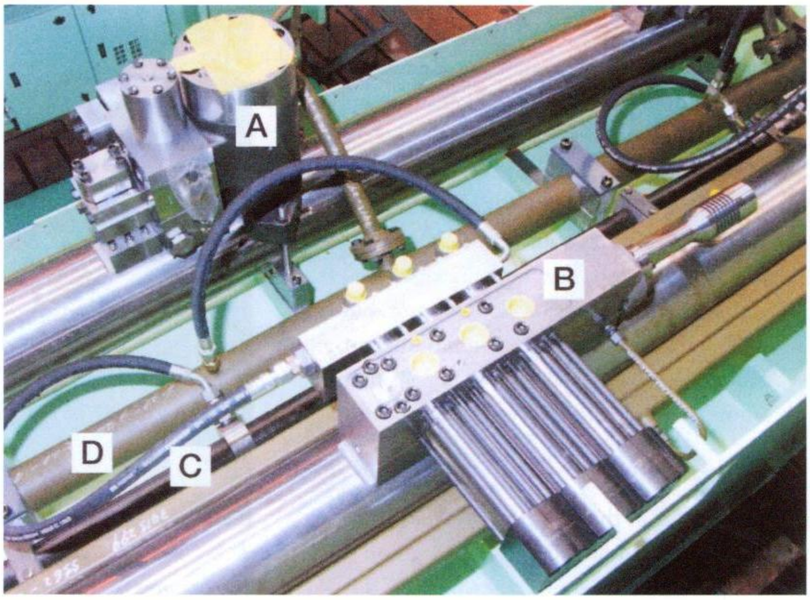
1 *high pressure fuel pumps*
2 *high pressure lubricating oil pumps*
3 *adjustment spindle of the fuel pumps*

The fuel pumps originated from Wärtsilä four stroke engines.

▶

The common rail system under the walkway of the cylinder heads.

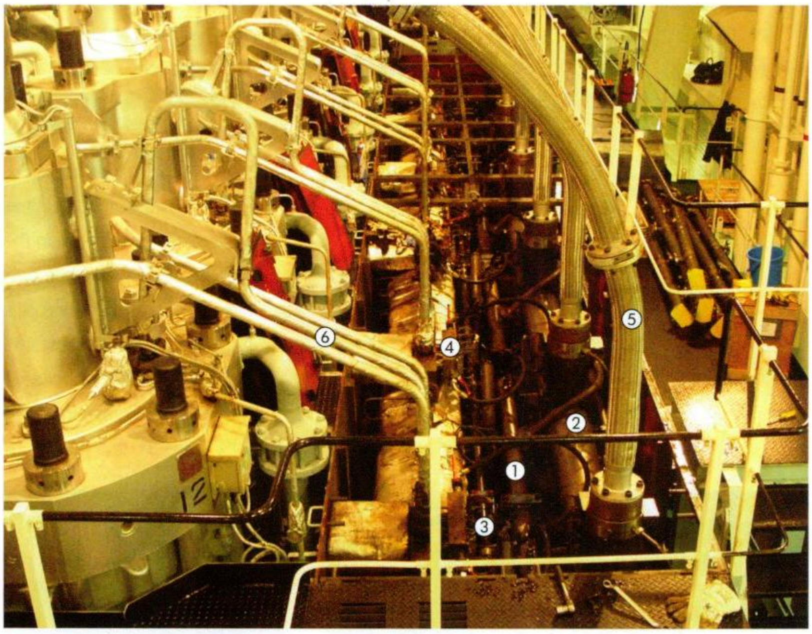
A exhaust valve actuator
B fuel injection controls
C control pipe lines for the hydraulic system
D discharge pipe lines for operating and control oil



▶

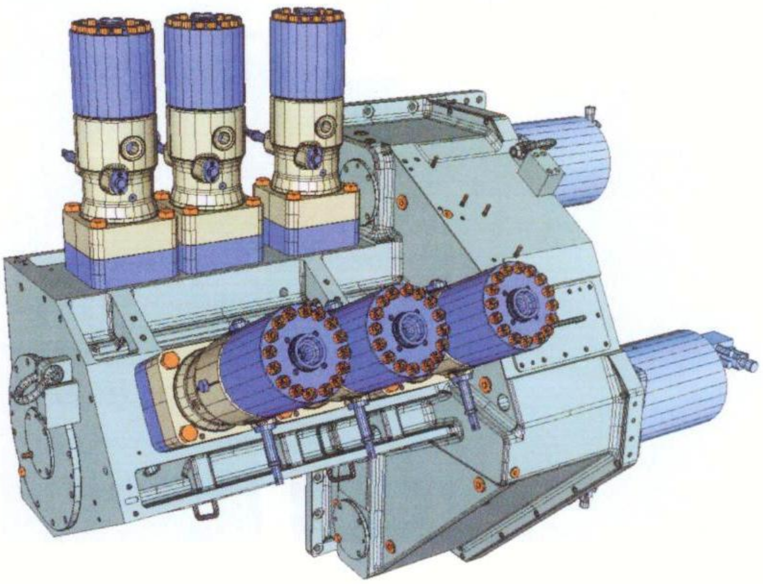
The common-rail system on a slow speed 12 cylinder Wärtsilä RTA-flex engine on the containership the "P&O Nedlloyd Mondriaan" (now Maersk Line).

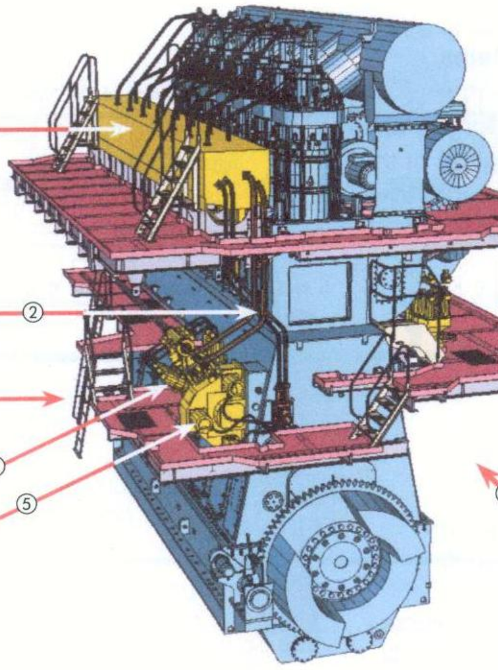
1 *high pressure fuel supply line 1000 bar.*
2 *high pressure hydraulic oil lines 200 bar.*
3 *high pressure hydraulic oil lines for operation of the fuel to the injectors.*
4 *valve block injector fuel.*
5 *high pressure hydraulic oil lines for operation of the exhaust valve.*
6 *injector lines.*



▶

A complete unit with left six high pressure fuel pumps and right two high pressure lubricating oil pumps .





◄

A overview of the system.

1 *rail unit with electronic control units*
2 *supply lines for fuel and servo oil*
3 *pump block supply unit*
4 *fuel pumps*
5 *servo oil pumps*
6 *automatic filters for servo- and control oil*
7 *crank angle sensor at the free end of the crank shaft*

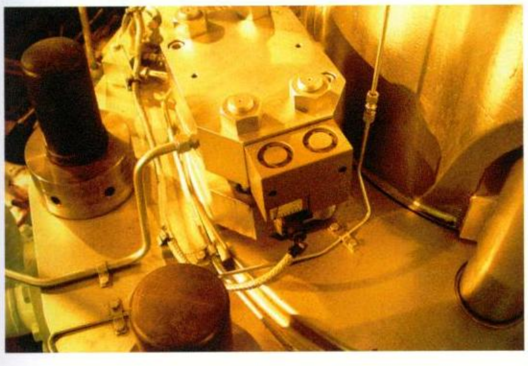
◄

The common rail system on a Wärtsilä Sulzer RT-FLEX two-stroke crosshead engine with twelve cylinders.

1 *high pressure hydraulic servo oil lines (200 bar)*
2 *high pressure fuel line (1000 bar)*
3 *fuel to three injectors*
4 *hydraulic servo oil lines for operation of the exhaust valve*

Operation of the exhaust valves and starting air valves

Apart from regulating the fuel supply, the system also controls the opening and closing of the exhaust valves and the starting air valves. All valves can be controlled and adjusted individually.



W.E.C.S. 9500-system

The whole system is controlled by a Wärtsilä W.E.C.S. 9500-system. It is a modular system with separate micro processor control-units for each cylinder and a general control and supervising system with a second series of micro processor control-units. The electronic speed regulation as well as the general monitoring system with alarm systems has also been incorporated in the W.E.C.S. 9500-system.

◄

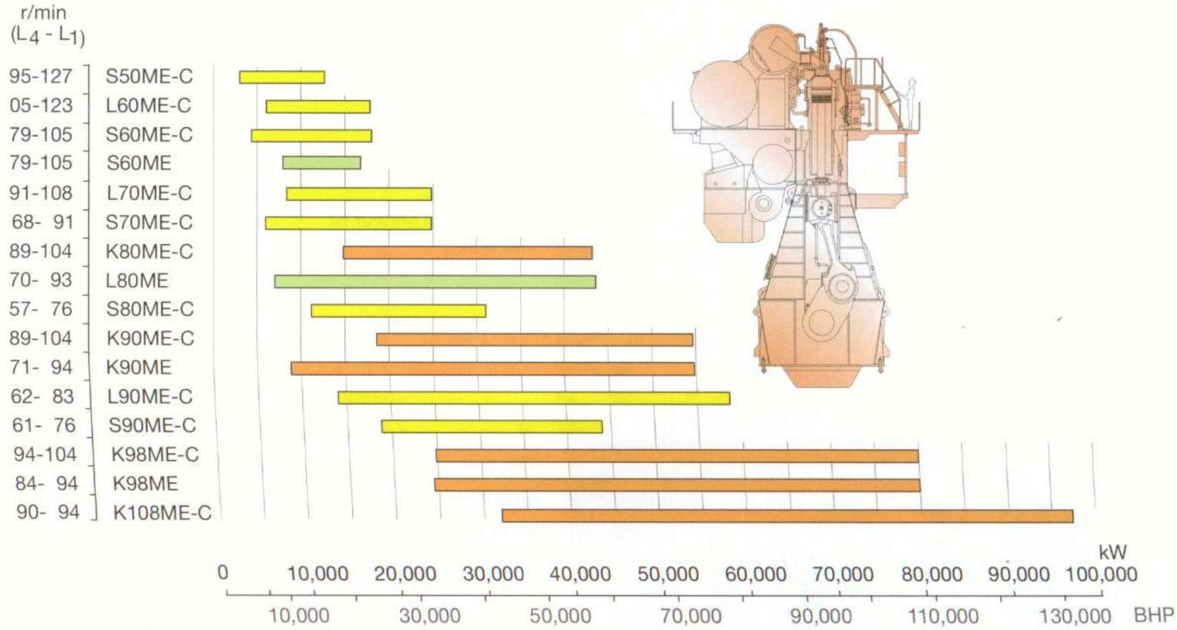
The air start valves are electronically operated from the W.E.C.S. 9500-controls system.

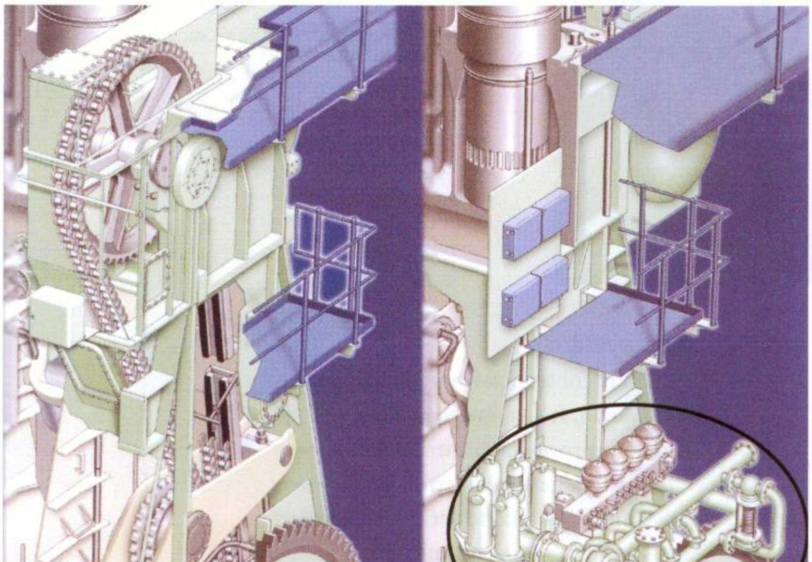
9.18.3 Example 3: common rail systems

The common rail system for low-speed two-stroke crosshead engines of the ME type by engine manufacturer MAN–B&W, fuel H.F.O. – Engine category IV

▼

A overview of the type of engines from MAN–B&W type ME-C that can be delivered. The cylinder diameter is from 500 to 1080 millimetre. Smaller cylinder diameters can at present be equipped with this system.



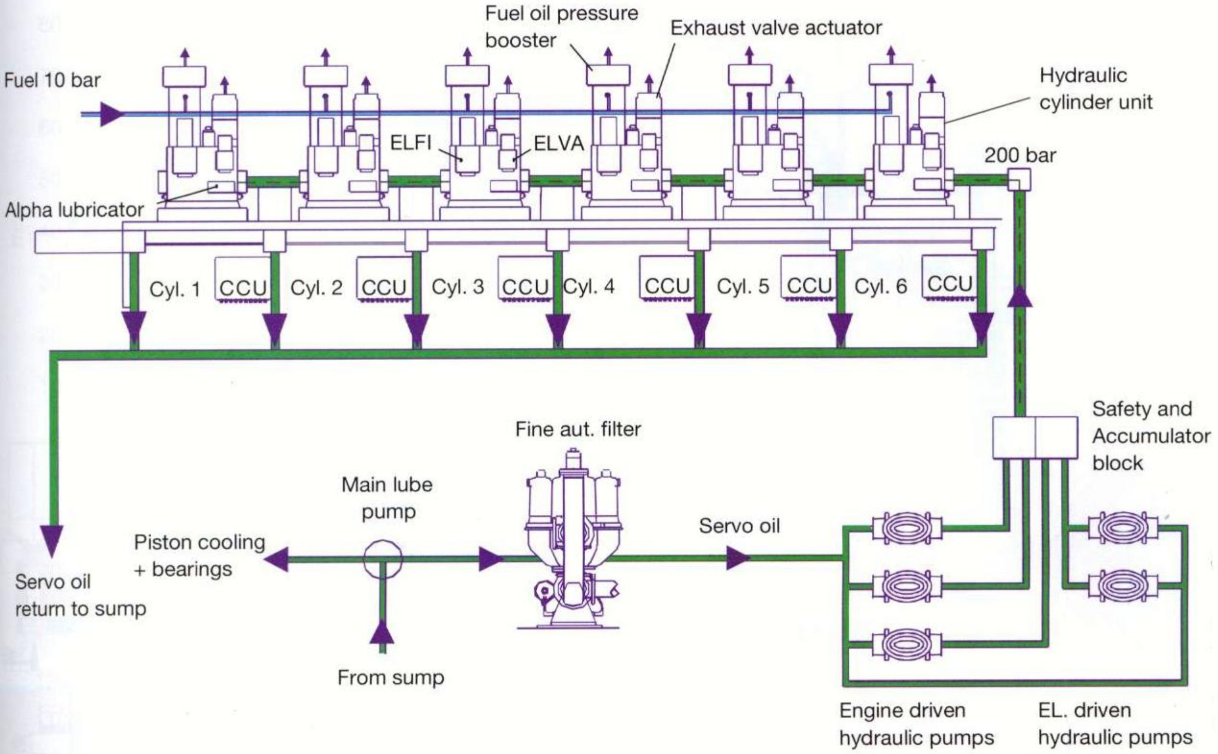


◄

The common rail system for two stroke crosshead engines type ME from engine manufacturer MAN–B&W.

Left: the normal version, with a chain driven camshaft and per cylinder a separate high pressure plunger fuel pump (MC-version)

Right: the ME-version, with electronic fuel adjustment. The camshaft with gear drives has been completely removed and in its place is a hydraulic high pressure unit positioned above the crank shaft that supplies high pressure lubricating oil for driving the fuel pumps and the operating the exhaust valves.



▲

The schematic diagram of the hydraulic system for the drive and controls of the high pressure fuel pumps and the exhaust valve at MAN–B&W type ME two stroke crosshead engines.

The system comprises the following components.

- An engine driven set of hydraulic lubricating oil pumps that supply a constant pressure of 200 bar for driving the fuel pumps and the operation of the exhaust valves. The set pumps are furnished with lubricating oil from the main lubricating oil system via an automatic fine filter. The discharge lines of the hydraulic system lead to the lubricating oil-discharge tank in the main engine.
  The lubricating-oil supply pressure is held at 10 bars.

ELVA Electronic Valve Actuator – a electronic exhaust valve operation
CCU Cylinder Control Unit – a cylinder control unit
ELVI Electronic Fuel Injection – a electronic fuel injection unit

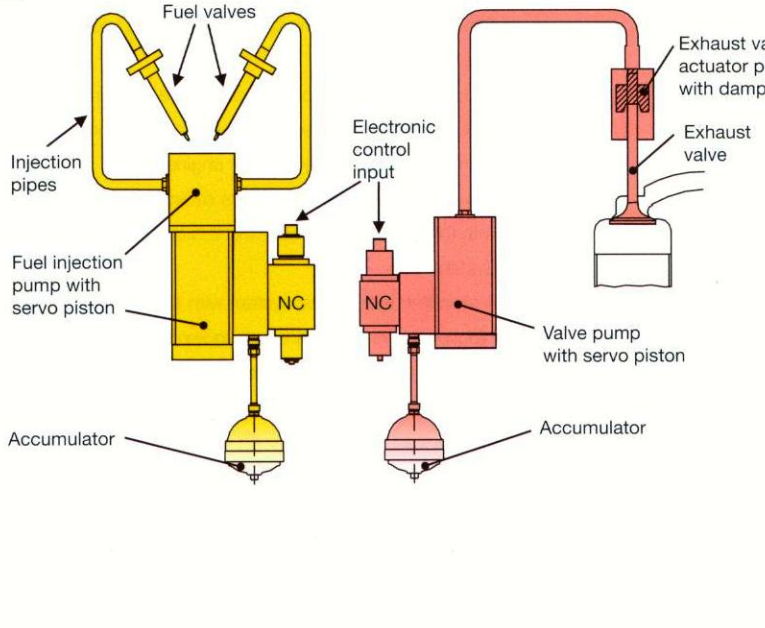
Remark

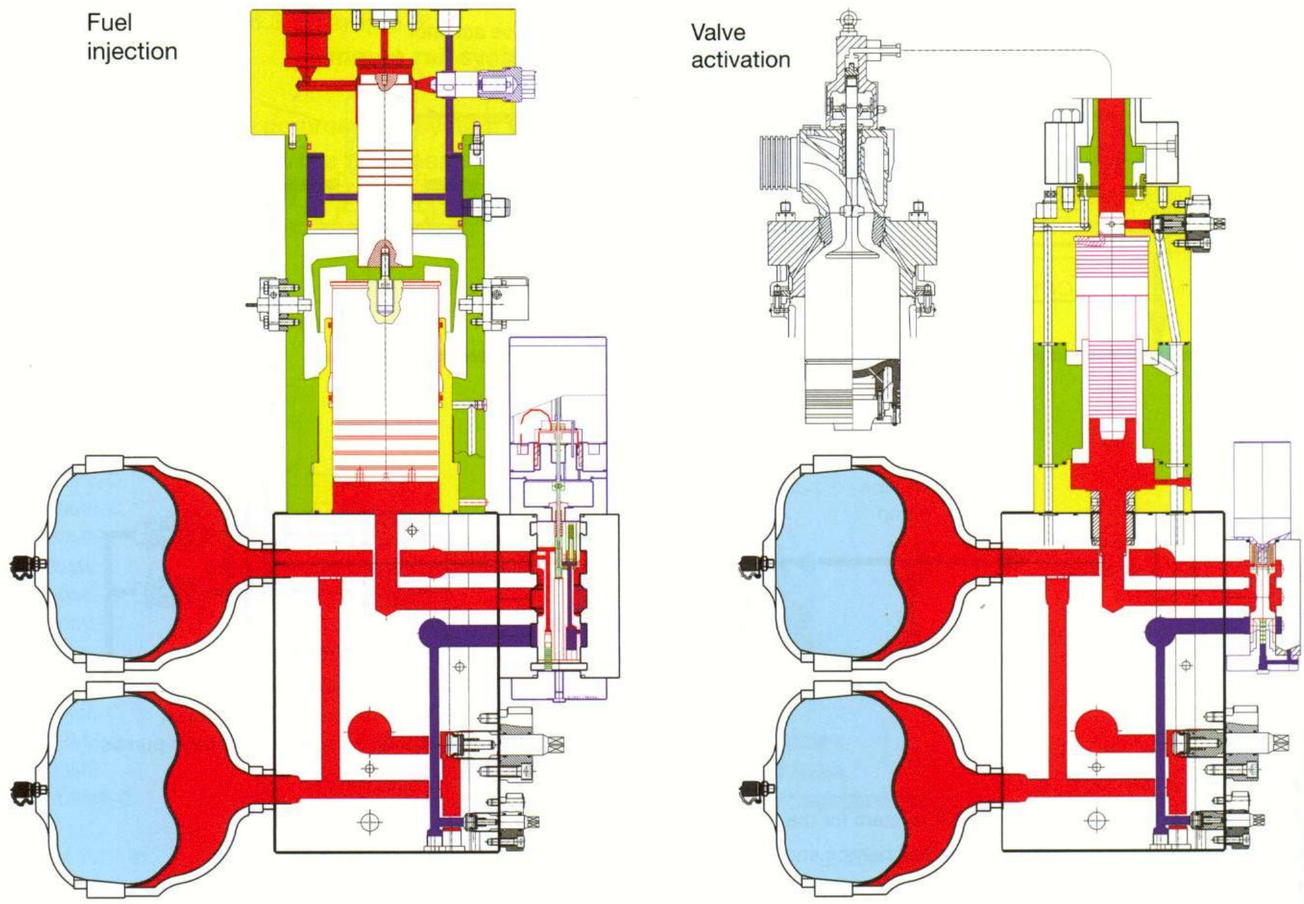
The dosing of the cylinder lubrication is automatically adjusted to the engine's operating conditions.

▼

The common rail system of MAN–B&W type ME.

Left: the fuel part where the high pressure plunger pump is driven by high pressure lubricating oil. An accumulator prevents pressure pulses. Per cylinder there are two injectors. That is electronically driven.
Right: the operation of the exhaust valve. The plunger pump of the exhaust valve is driven with high pressure lubricating oil.





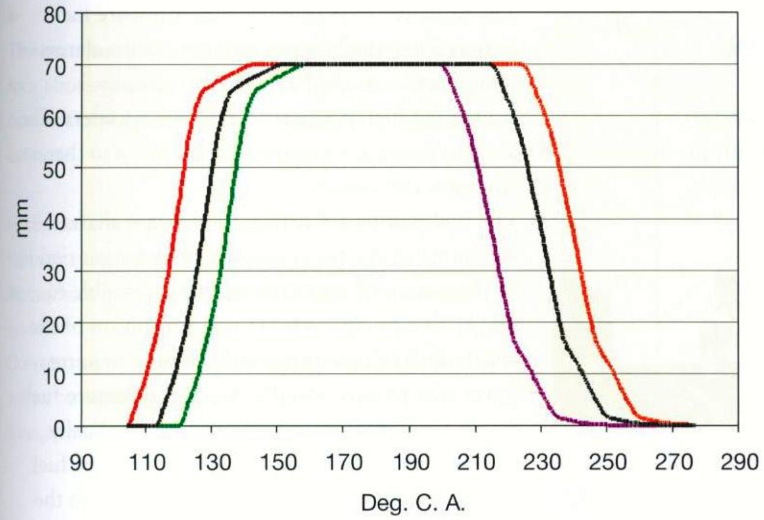
▲

The common rail system of MAN–B&W.

Left: the fuel part
Right: the operating system for the exhaust valves

Advantages of the system

- Lower specific fuel consumption and performance improvement as a result of the adjustable electronic controlled fuel injection and exhaust valve timing at every load.
- The correct high fuel injection pressure and injection time at every load.
- A improved combustion process where the nitrous oxide emissions are decreased and the soot emissions fall dramatically, especially at a low engine load.
- It is simple to modify the various settings during operation with the process computer; the E.C.S., 'Engine Control System'.
- A simplified mechanical system with solid traditional fuel injection systems that are well known to engineers.
- A control system with a more accurate timing so there is a better balance between the cylinders and a more even thermal load.
- All engine data is displayed and saved and shows as to whether adjustments should be made. Through this good diagnose the number of operating hours between maintenance can be increased.
- Manoeuvring at lower revolutions is possible due to improved combustion processes in the cylinders at low loads.
- The advancing and retarding of the revolutions, reversing, starting and running astern supply a faster change in the ship's speed in an emergency stop situation.
- The cylinder lubrication is incorporated in the electronic controls. This results in a reduction of the lubricating oil usage and therefore less contamination in the cylinder -and exhaust systems.
- It is simple to modify the software during the lifetime of the engine.
- The start air valves and auxiliary blowers are driven at the precisely at the correct moment.
- The total weight of the engine is decreased.
- Furthermore it is possible with this system to modify the timing of the opening and closing of the exhaust valve .



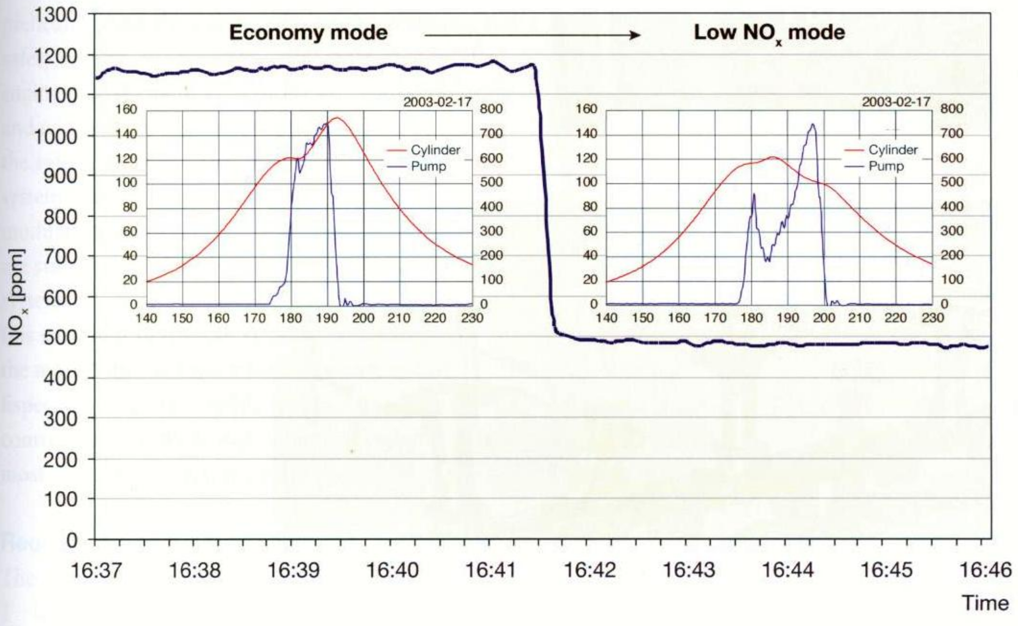
◄

Possibilities for modifying the timing of the exhaust valve.

Horizontal: crank degrees after top dead centre
Vertical: valve lift in millimetres
Purple: earlier closure
Yellow: later closure
Brown: earlier opening
Green: later opening
Black: normal line

Remark

On average the valve is approximately 140 crank degrees open.
An example of the possibilities of the ME-version.



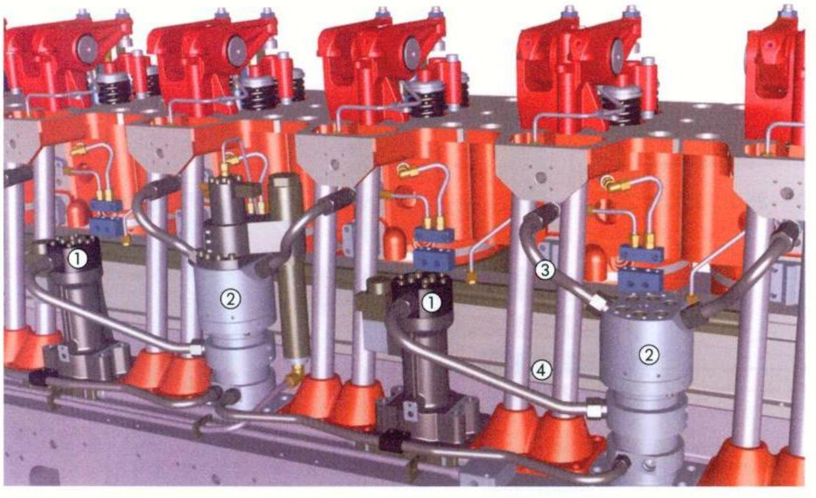
▲

By modifying the fuel injection, the engine can be tuned to different requirements such as a lower specific fuel consumption and lower nitrous oxide emissions. In the second case, the fuel injection later and in two phases.

Left figure: low specific fuel consumption
Red line: pressure curve in the cylinder
Blue line: injection pressure curve of the fuel
Right figure: the pressure in the cylinder during the entire process is smoother by allowing the fuel injection to take place in two phases. Due to this the pressure curve is more gradual and lower in temperature.

9.18.4 Example 4: common rail systems

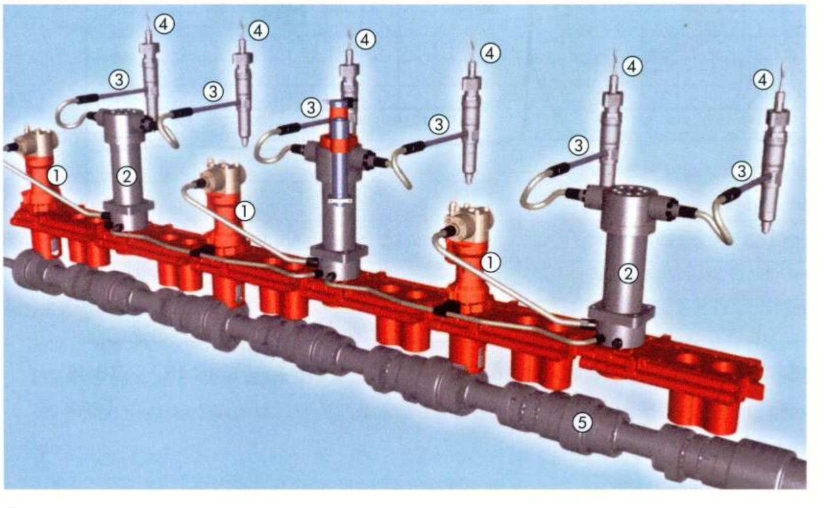
Common rail systems for four-stroke medium-speed trunk piston engines by engine manufacturer Wärtsilä, fuel H.F.O. – Engine category III



▲
The common rail system for Wärtsilä medium speed four stroke engines, fuel H.F.O.

1 *high pressure fuel pump*
2 *accumulator*
3 *high pressure fuel line to the injector*
4 *push rods for the inlet- and exhaust valves*

The principle is as follows: high-pressure fuel pumps force the fuel towards the accumulators which are connected to the electronically controlled fuel injectors of the two cylinders. The accumulators are connected with pipes to the common rail system.
The fuel pumps are driven by the cam shafts. As timing of the fuel pumping is no longer tied to the timing of injection, it is now possible to realise two pump cycles in one revolution of the crank shaft. Consequently, the engine requires fewer fuel pumps: usually one high-pressure fuel pump for every two cylinders. A hydraulically driven activation system is used to send the fuel to the injectors. The oil required is fed from the general lubricating oil system of the engine and pressurised to 200 bar by a separate, engine driven pump.

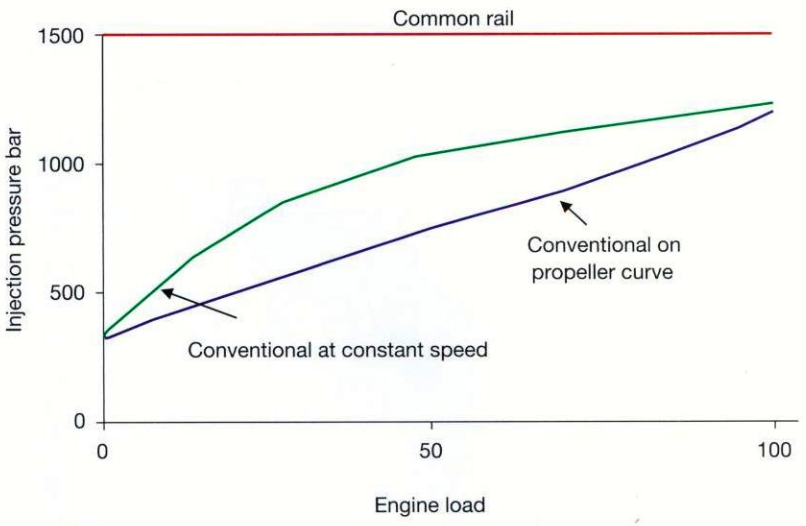


▲
The common rail system for Wärtsilä medium speed four stroke engines, fuel H.F.O.

1 *high pressure fuel pump*
2 *accumulator*
3 *high pressure fuel line to the injector*
4 *injectors*
5 *camshaft*

The high pressure fuel pumps 1 are driven by the camshaft 5 and have sufficient capacity for two cylinders.
Every pump is linked with an accumulator 2 that levels the pressure spikes and supplies fuel to two cylinders.
The accumulators are linked together by a double walled pipe. Due to this the pressure in all the accumulators is always equal and in emergencies one or two high pressure fuel pumps can be disconnected.

From the accumulators the fuel with the required pressure is supplied to the injectors 4. The fuel valves for the injectors are hydraulically operated and electronically driven. Due to this it is possible to change the timing and duration of the fuel injection in every cylinder. From a safety point of view, it is important that between two injections, the injector is pressure less, so fuel leakage to the cylinders is not possible. In a new injector design the injector needle is opened and closed at full fuel pressure. Due to this the atomisation of the fuel at every load and revolutions of the engine is optimal.
A consequence is a smokeless engine with low emissions of amongst other carbon monoxide.
Pressures: these can be adjusted between 900 and 1500 bar.



▸
The common rail system for Wärtsilä medium speed four stroke engines, fuel H.F.O.. Comparison to the conventional system with individual fuel pumps per cylinder .

Horizontal: the load of the engine
Vertical: the fuel pressure in bars
Above horizontal: the fuel pressure with a common rail system
Curved green line: the fuel pressure at constant revolutions with a conventional system
Straight blue line below: the fuel pressure at the screw drive

The same oil is used for the starting and safety valve (S.S.V.) to allow the fuel to circulate during preheating and the safety valve of the common rail safety system. All functions are controlled by the engine management system. Apart from regulating and controlling engine speed, the system monitors the safety controls in the engine. The common rail system was developed for new engines, but the modular system can also be applied to engines that are still equipped with conventional fuel injection systems.

It is expected that these systems will remain on the market for at least another twenty years. Especially since the visible smoke can effectively be controlled with these new injection systems. It is most effective at partial load.

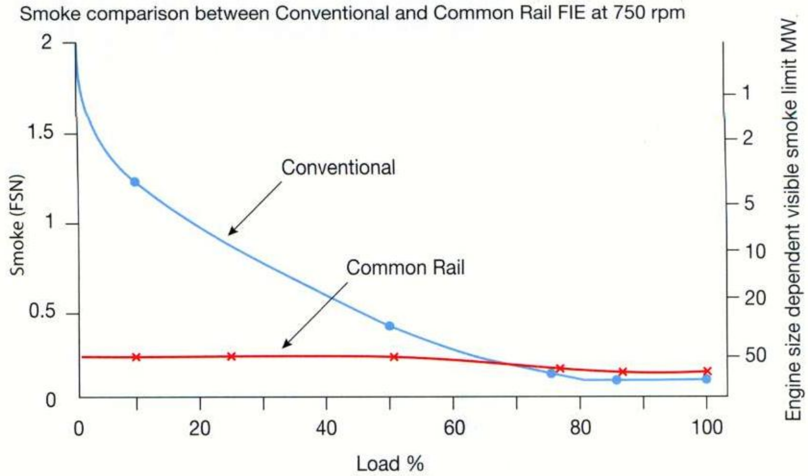
Regulating- and controlling* system

The system has two tasks:

1 to regulate the injection time and the amount of fuel injected;
2 to control the refilling and the pressure of the accumulators.

The control system also includes the monitoring of all the important data such as the pre-heating, the maximum allowable pressure in the system, the pressure in case of an emergency stop. So far, three types of four-stroke engines have been equipped with the common rail system, namely the Wärtsilä 32, 38 and 46 engines.

Smoke comparison between Conventional and Common Rail FIE at 750 rpm



▲
The effects on the soot emissions of the common rail system from Wärtsilä for medium speed four stroke engines, fuel H.F.O.

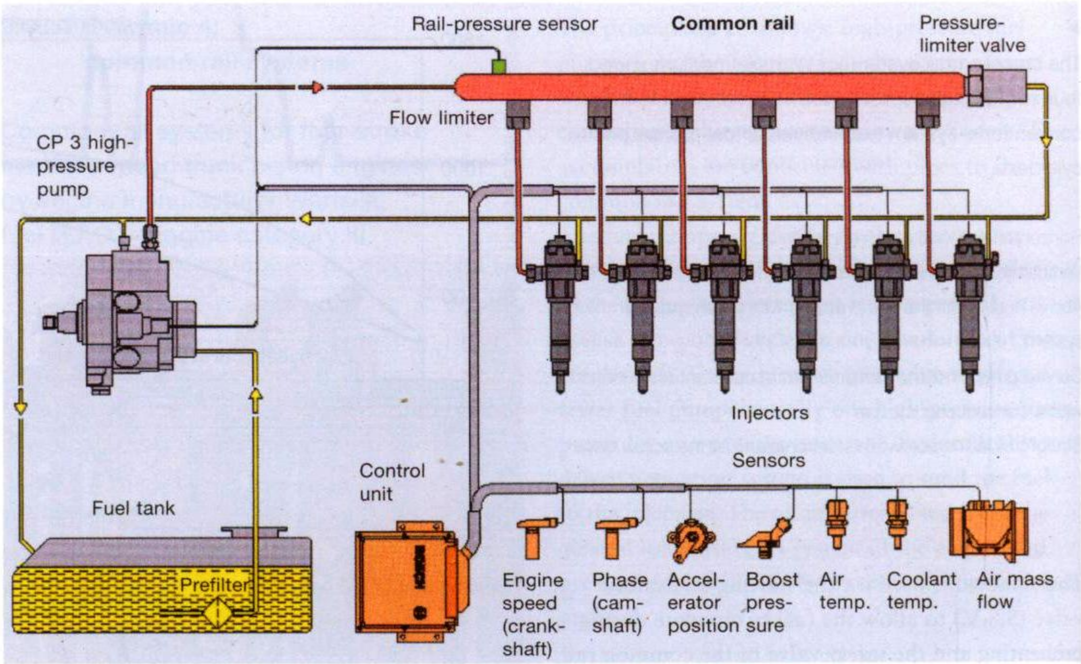
Horizontal: the load of the engine in percent
Vertical left: the soot number
Vertical right: the output power in MW of an engine, the soot number is dependant on this.
Red line: the soot emissions with the common rail system
Blue line: the soot emissions with the conventional system

From experience it has been shown that the specific fuel consumption at partial load falls slightly.
Furthermore with a common rail system every cylinder and so every individual combustion process can be adjusted for optimal performance. This limits the engine vibrations.

▶

For MAN trucks, the Bosch common rail (C.R.) system is used.

The system comprises of a fuel tank where the high pressure multi plunger pump draws up the fuel and delivers it via a spacious fuel supply line (C.R.) the fuel for the injectors. The plunger pump is driven by the pressure sensor on the rail. A safety valve ensures that the maximum allowed pressure is not exceeded. In the supply lines to the injectors there are restrictions that guarantee the maximum flow capacity. The actual fuel supply to the injectors is controlled with electronically controlled magneto valves. Here, there are also restrictions for the supply and discharge of fuel.



9.18.5 Example 5: common rail systems

Common rail systems for high-speed four-stroke diesel engines, fuel M.D.O. Engine category II

Many engines in this category have been equipped with common-rail systems. After the development of this modern fuel injection system for the automotive industry, the system is in now also applied in navigation and diesel power plants due to the increasingly strict requirements with regard to the emission of toxic substances in exhaust gases in these industries.

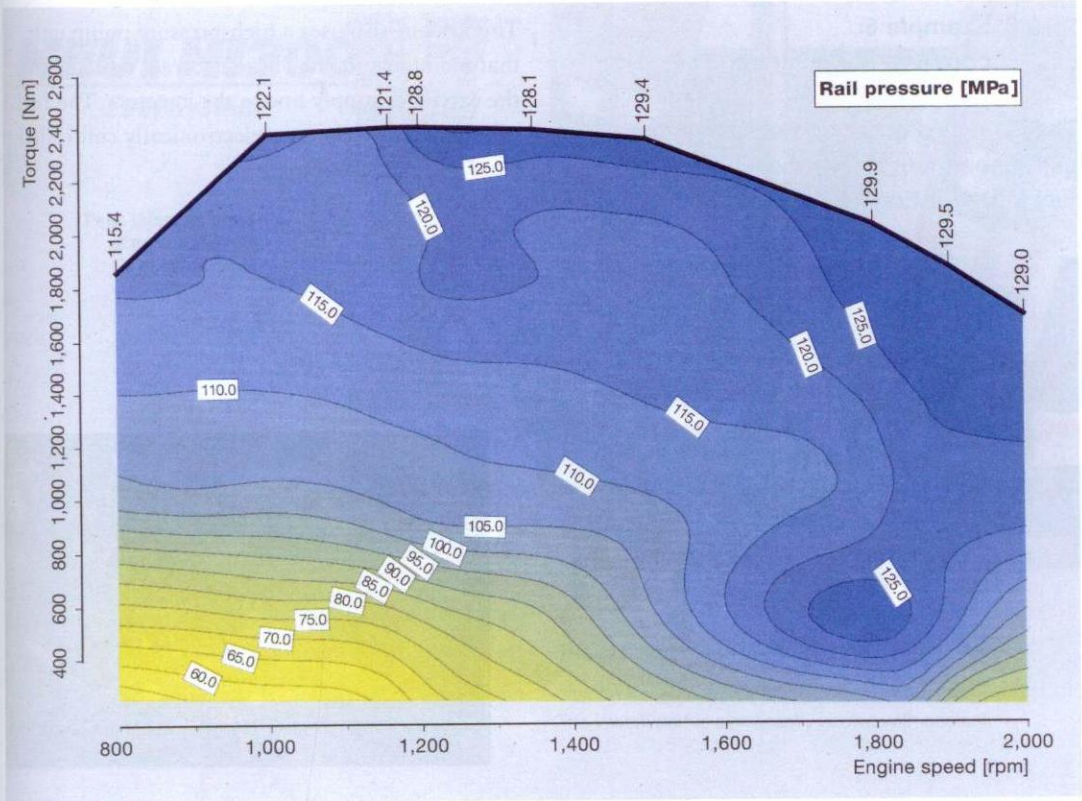
In principle these systems are very similar and all have the same objective:

1 Reduce the emission of pollutants at varying loads and numbers of revolutions.
2 Reduce the specific fuel consumption at varying loads and numbers of revolutions.

◀

The effect of the common rail pressure on the soot number SZ (Bosch), the specific fuel consumption b_e (in grams per kWh) and the nitrous oxide emissions NOx (in grams per kWh). The common rail pressure is in Mega Pascal.

Clearly shown is that at a fuel pressure increase, the soot number and the specific fuel consumption declines to a large extent due to improved combustion and that the nitrous oxide content increases because of the higher maximum combustion temperatures in this improved combustion.



◀

The fuel pressure in the common rail system with respect to:
Horizontal: the revolutions of the engine in revolutions per minute;
Vertical: the torque of the engine in Newton meter.

The fuel pressure of the common rail system varies from 1150 to 1290 bar; this is a difference of approximately 11%.
In a conventional system this varies from 1250 bar, dark blue to 110 bars, light blue to at the lowest load 70 bar, yellow. In the yellow area the engine is running at a very low partial load and the fuel pressure is so low that the atomisation and therefore combustion is poor.

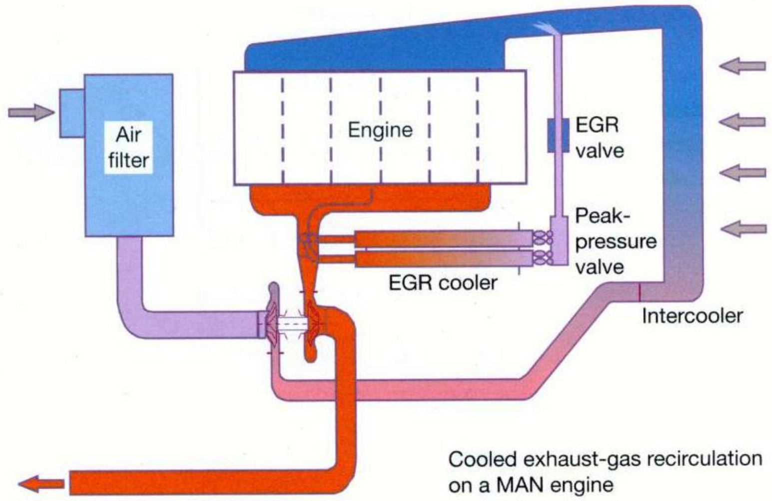
There are a few prominent details :

- Electronic engine management systems regulate the time and duration of the fuel injection.
- The fuel injection pressures are extremely high and constant and ensure adequate atomisation under all circumstances.
- The high fuel pressure can be produced by numerous different pump systems such as in-line pumps and block pumps as well as injector pumps, mechanically driven from the cam shaft by a push rod and a lever.
- The development started at the end of the last century and has continued to develop rapidly to be applied to all diesel engine categories.
- By increasing the fuel pressure the emission of soot particles diminishes noticeably, but due to the improved combustion at higher temperatures the production of nitrous oxides increases. With the application of exhaust gas recirculation with an inter cooler this effect is countered.

▼

In truck diesel engines all the systems are used for recirculation the cooled exhaust gases.

Due to this the maximum combustion temperatures are unduly lower, so less nitrous oxide is produced. As from 2007, this system is an option for all engine categories. The normal maximum rail pressure for this system is 160 MPa or 1600 bar.
For the Euro 3 standards this is sufficient. The Euro 4 standards can only be met with a pressure of approximately 180 MPa. Pressures of 200 MPa will be necessary in the future to meet the increasingly strict emission standards.

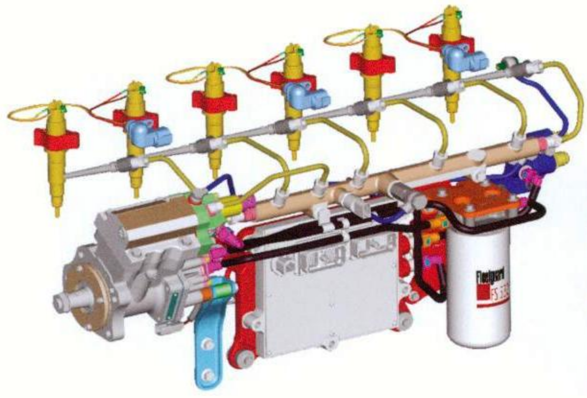


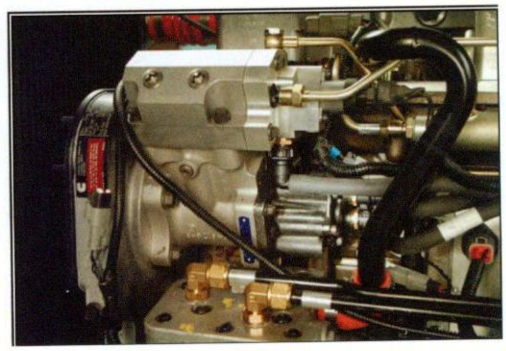
Cooled exhaust-gas recirculation on a MAN engine

9.18.6 Example 6: common rail systems

The Cummins common rail system for high-speed four stroke diesel engines, fuel M.D.O. – Engine Category II

This system also uses a high-pressure pump unit that maintains the fuel at the correct pressure in the extensive supply line to the injectors. The fuel supply to the injectors is electronically controlled from a central computer.



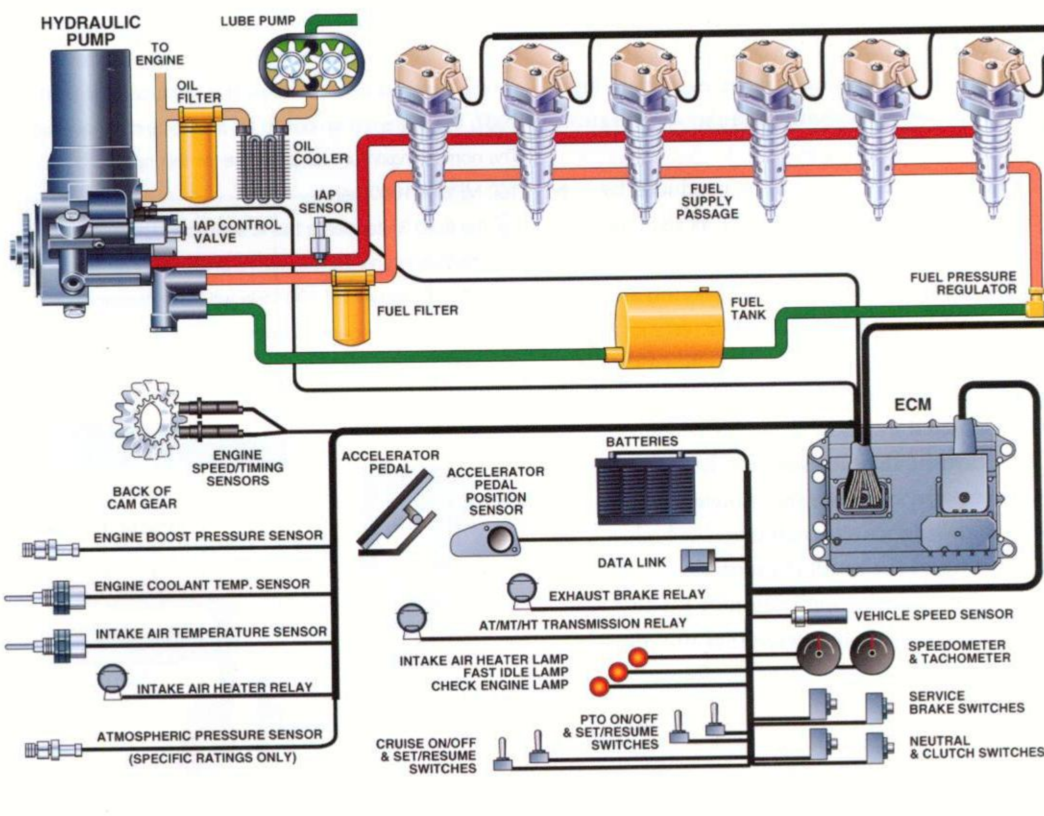


▶ The Cummins common rail system.

These are high speed four stroke engines running on M.D.O.

9.18.7 Example7: common-rail systems

The Caterpillar common-rail system for high-speed four stroke diesel engines, fuel M.D.O. – Engine Category II



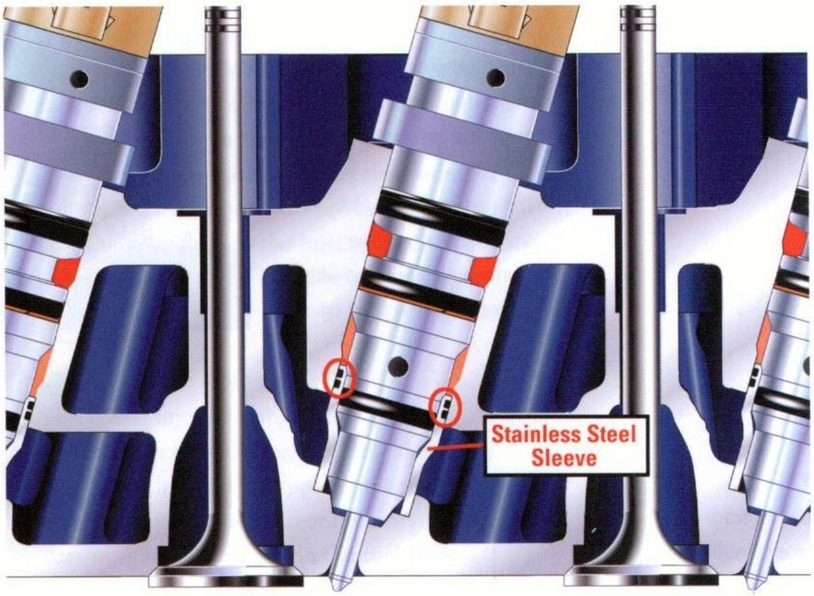
▶ The Caterpillar common rail system.

These are high speed four stroke engines running on M.D.O.. The high pressure fuel pump is driven by high pressure lubricating oil supplied by a separate pump from the main lubricating oil system of the engine.
Green: low pressure suction- and discharge lines.
Red: high pressure supply and common rail lines.
Light red: discharge lines with an adjustable minimal pressure.
Brown: normal pressure lines of the regular lubricating oil pump.
E.C.M: Engine Control Module.

▲

The injectors ensure that the fuel is injected in a very fine spray into the cylinder .

In the common rail systems of Caterpillar the fuel injection pressure can increase to 162 MPa or 1620 bar.



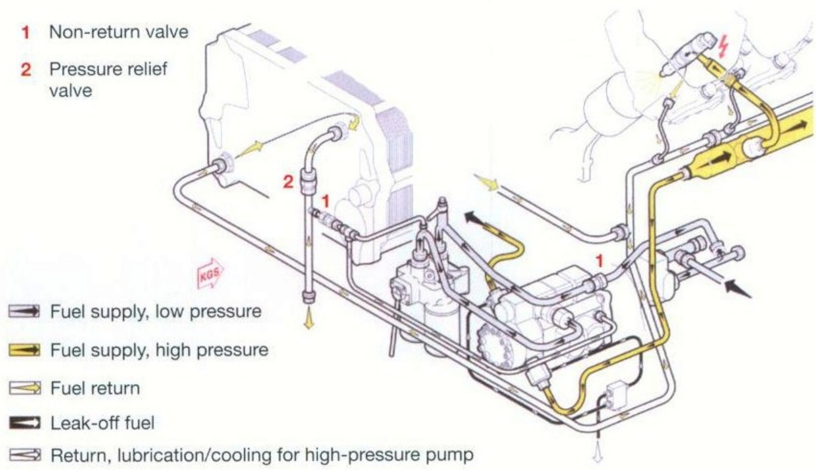
▲

The position of the injector in the cylinder head.

The lowest part is significantly cooled with coolant.

9.18.8 Example 8: common-rail systems

The common-rail system by MTU for high-speed four stroke diesel engines, fuel M.D.O. – Engine category II



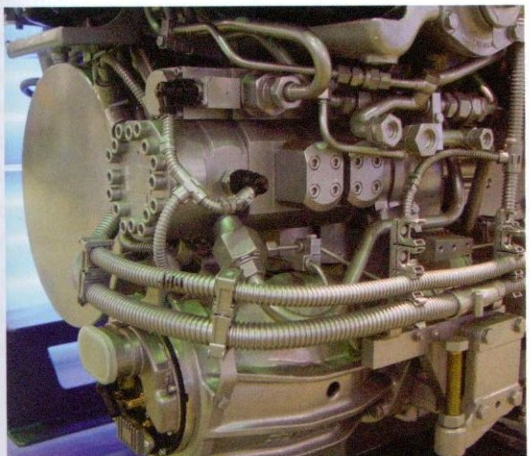
▲

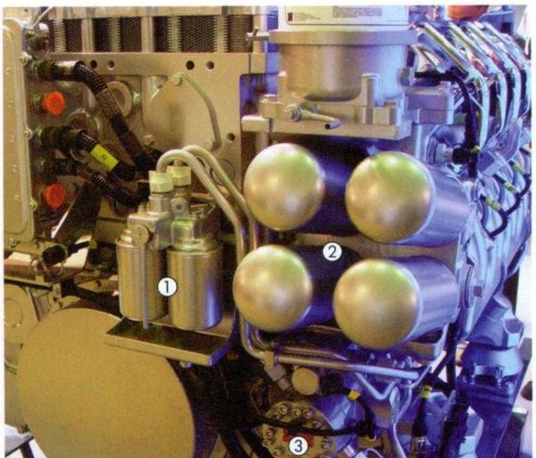
The fuel system for fast speed four stroke MTU-diesel engines, fuel M.D.O.

▼

The common rail system of MTU.

This common rail system has together with a suction filter for the engine driven high pressure fuel pump a low pressure-suction pump and a fuel cooler. The common rail pipe lines have a diameter, so that pressure waves are limited.

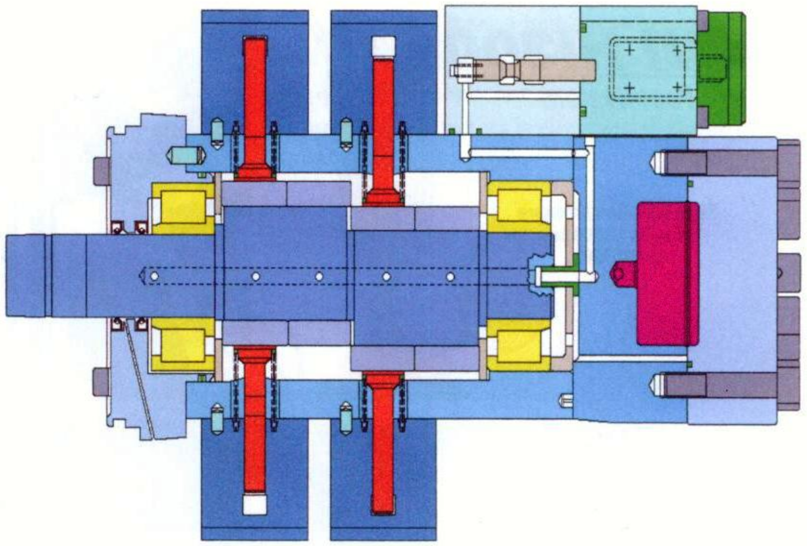




◄

The fuel filters of a MTU-diesel engine.

1 *fuel filters*
2 *lubricating oil filters*
3 *fuel pump*

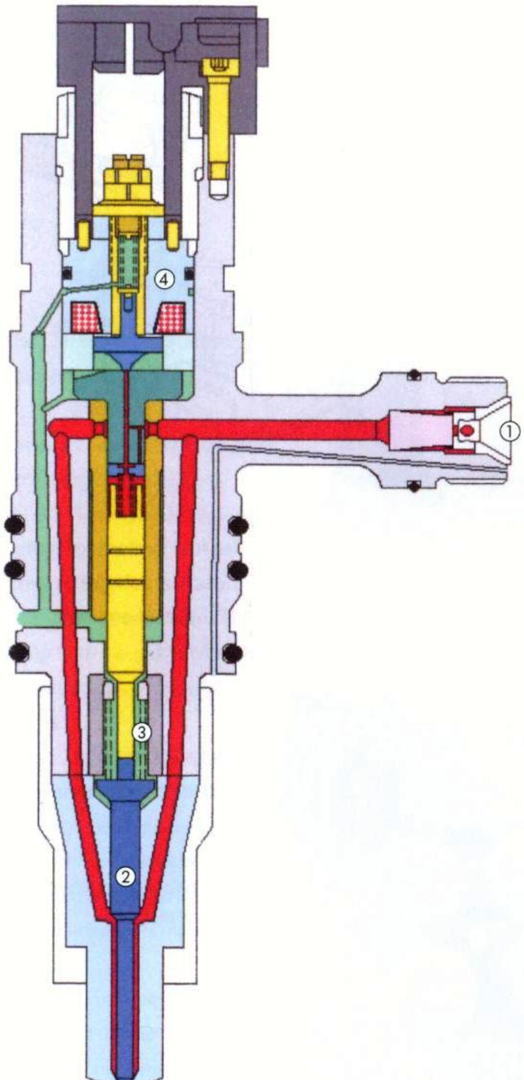


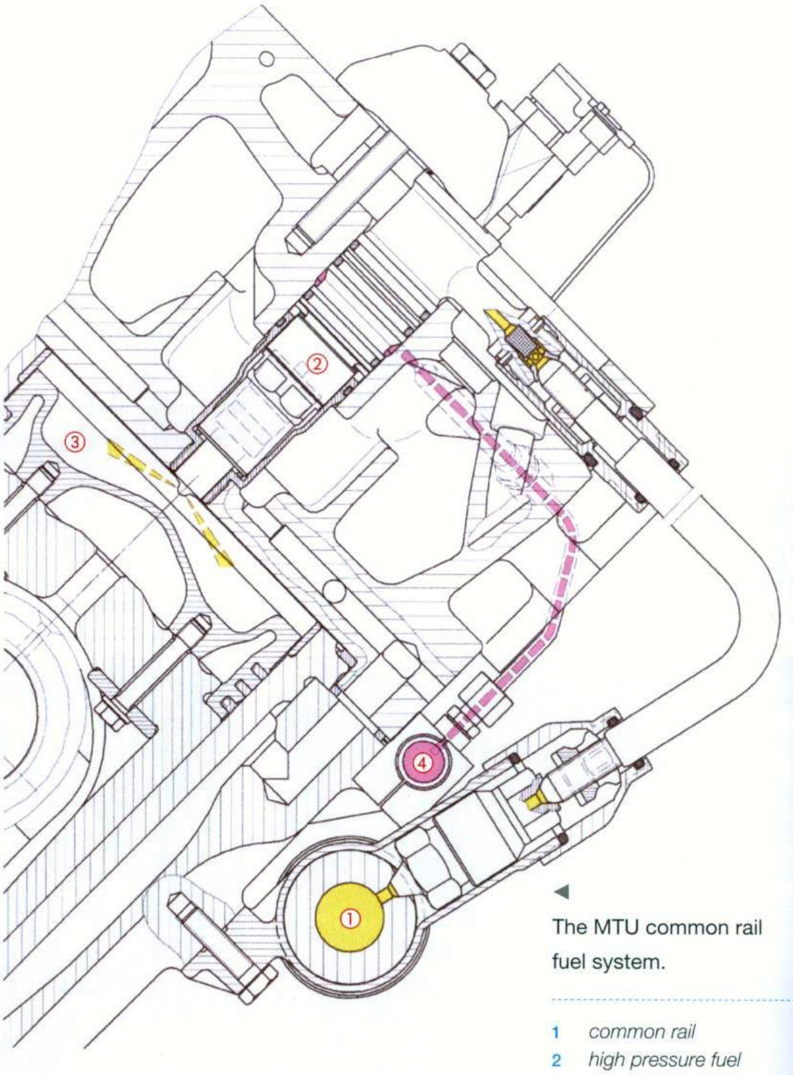
▸ The multiple plunger pump for the MTU common rail engines.

The parts of the shaft that drive the plungers are asymmetrically placed with respect to of the shaft drive.

▾ The injector in cross/section.

1 *fuel supply*
2 *injector needle*
3 *injector needle spring*
4 *solenoid valve*





◂ The MTU common rail fuel system.

1 *common rail*
2 *high pressure fuel injector*
3 *combustion chamber*
4 *discharge fuel lines*

9.19 Injector system

In this system the high-pressure fuel pump and the injector have been placed in one casing. The fuel injector is situated in the cylinder and is mechanically driven from the cam shaft, as are the inlet- and exhaust valves, by means to a fuel cam, guide pulley, push rod and lever.

9.19.1 Mechanically driven injector by Cummins-diesel engines

High-speed four-stroke engines, fuel M.D.O. – Engine category II

The PT-pump is a low-pressure system, with self compensating wear and tear; easy maintenance and repair. When compared to the conventional system there is relatively little risk of leakage in the PT-fuel system, because the pressure between the pump and the injector is low. Cummins has designed the fuel supply in such way that it runs through internal ducts in the cylinder heads to the injectors, thus further reducing the risk of leakage. Although the fuel pressure in the PT-system is low, the injection pressure is much higher than in ordinary systems. This is achieved by the mechanical movements of the injectors generated by the push rod, rocker and a separate cam on the cam shaft for each cylinder. The high injection pressure effects efficient fuel consumption, which results in low exhaust gas emissions, without requiring a complicated combustion space with high whirling as do other engines.

Fuel regulation

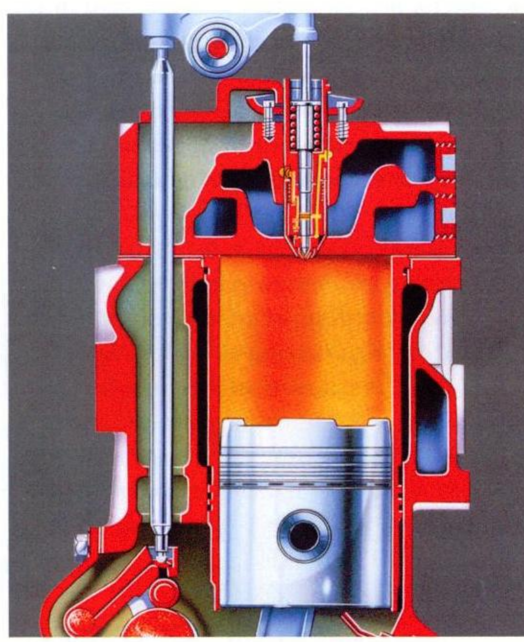
A gear pump in the fuel unit draws fuel from the tank through a filter and forces this into the pump. The fuel flows from the gear pump through a combined speed- and pressure regulator, then through a throttle valve and a shut off valve to the injectors. A central supply line transports the fuel to all the injectors of an in-line engine and to each bank in V-engines. This fuel line provides equal fuel pressure to each injector, and an evenly distributed amount of fuel and power output to cylinders. Prior to injection, cooling, lubrication of the system and bleeding, the fuel circulates through the injectors. The superfluous fuel is returned to the tank by a return pipe. The warm return fuel, which was sent back to the tank, improves the flow of the fuel, which is advantageous in a cold climate.

Fuel pumps function

Fuel consumption in individual injectors depends on the pressure in the supply line. This pressure is fully controlled by the fuel pump.
Therefore it is not necessary to set the timing of the pump, since it only has to provide the specific pressure and the flow to the injectors as a function of revolutions, load and the position of the throttle control.

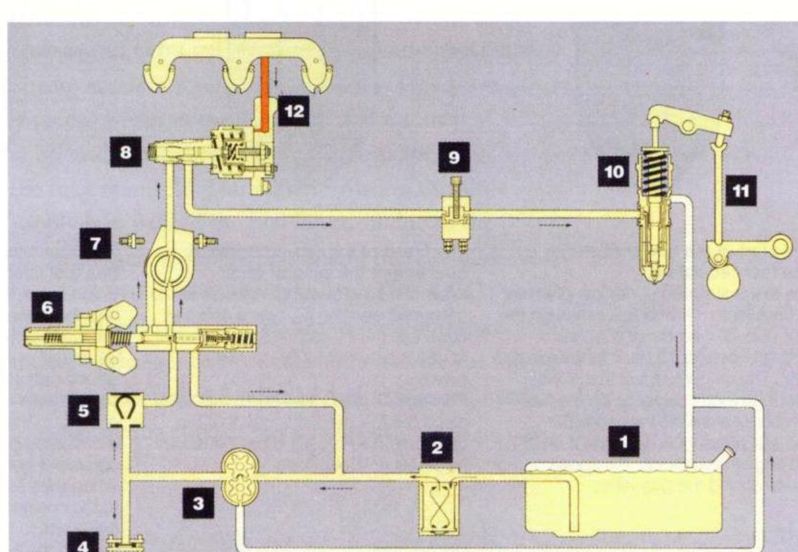
Operating under pressure

Since the high injection pressure is built up mechanically in the injector, the fuel pump only has to ensure that fuel is supplied under relatively low pressure and in the correct quantity. This is the reason that the pressure in the fuel lines is lower than 17.5 bar as opposed to a pressure of 700 bar and higher, which is required for systems with hydraulically functioning injectors.



◄ The mechanically driven injector for Cummins-diesel engines.

1 *Fuel tank*
2 *Fuel filter*
3 *Gear wheel pump*
4 *Vibration damper*
5 *Filter with magnet*
6 *Governor en pressure control parts*
7 *Throttle lever*
8 *Air -fuel control*
9 *Shut-off valve*
10 *Injector*
11 *Rocker arm, push rod and cam shaft follower*
12 *Exhaust manifold air*



▼

A complete fresh cooling-water system with a low temperature (L.T.) and a high temperature (H.T.) section for a four-stroke diesel engine.

01 High temperature (H.T.)-coolant pump

In four-stroke engines most circulating pumps are driven by the engine itself. It takes care of the closed coolant system of the engine block.

Temperatures between 80 and 95 °C.

02 Low temperature (L.T.)-coolant pump

These circulating pumps are usually driven by the engine. It takes care of the closed coolant system for the air cooler and the lubricating oil cooler.

Temperatures between 40 and 50 °C.

03 Air cooler

Cools the compressed air using a turbo-blower in combination with the L.T. system, before the air is led into the inlet line of the engine.

04 Lubricating- oil cooler

Cools the lubricating oil heated by the engine.

05 H.T. thermostatic control valve

Keeps the H.T.-coolant temperature at a predetermined value by means of a three-way valve.

06 L.T. thermostatic control valve

Maintains the L.T.-coolant temperature at a predetermined value by means of a three-way valve.

07 Adjustable restriction, usually referred to as an orifice

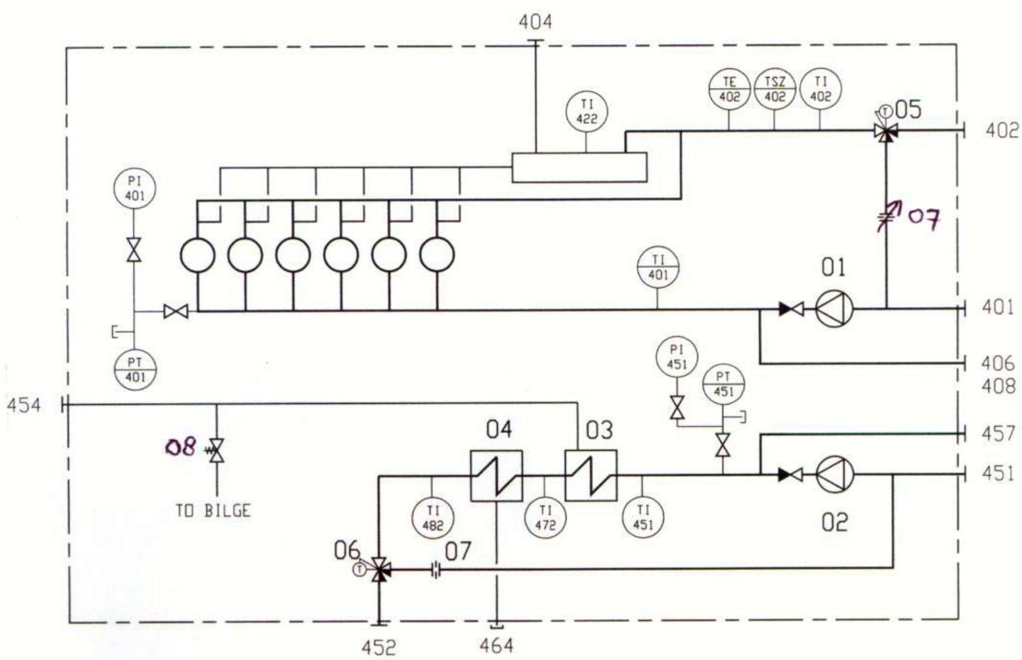
In both systems, this type of adjustable restriction is applied. These orifices are set in such a way that when the engine is operated, the three-way controlling valve does not produce significant pressure differences in the coolant system.

08 Safety valve

This is installed to avoid damage to the cooler casing when the supercharged-air pressure is too high.

Connection outside of the diagram

401 and 402:	H.T. cooler inlet and outlets
404:	H.T. system air vent.
406:	connection to the H.T. pre-heater system
408:	H.T. coolant from the stand-by coolant pump
411:	drainage H.T. cooling-water system
451 en 452:	L.T. coolant inlet and outlet from the L.T. seawater-cooler
454:	L.T.-cooler air vent
457:	L.T.-coolant from stand-by pump
464:	L.T.-coolant drain



System components

01	HT-cooling water pump
02	LT-cooling water pump
03	Charge air cooler
04	Lubricating oil cooler
05	HT-thermostatic valve
06	LT-thermostatic valve
07	Orifice

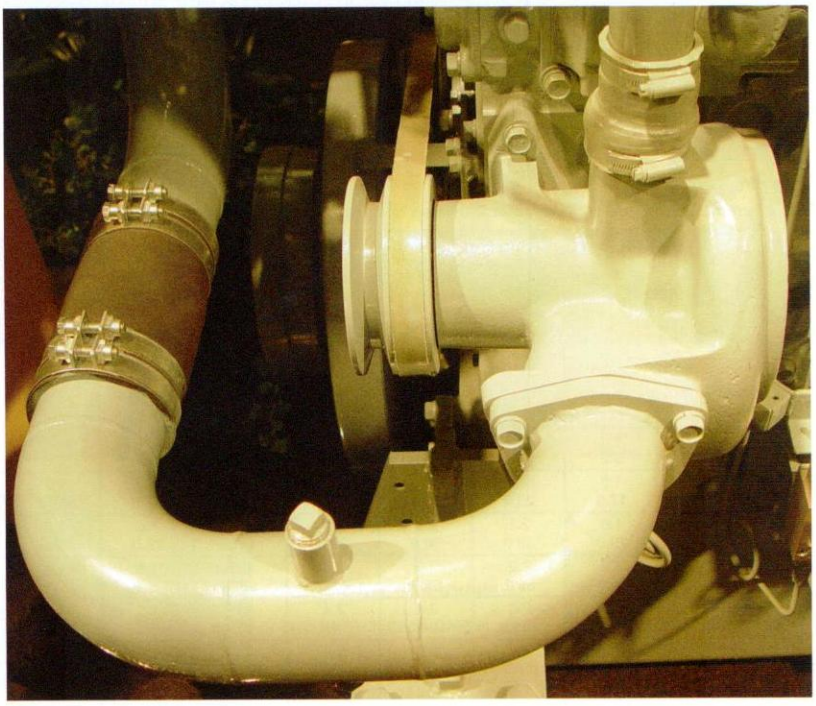
Pipe connections

401	HT-water inlet
402	HT-water outlet
404	HT-water air vent
406	Water from preheater to HT-circuit
408	HT-water from stand-by pump
451	LT-water inlet
452	LT-water outlet
454	LT-water air vent.
457	LT-water from stand-by pump
464	LT-water drain

▸
A corroded valve casing.

On board ship attempts have been made to weld the leaking corroded cast-iron valve casing with steel plate strips. This is obviously not very effective as cast iron is not suitable for welding! The welds will tear after temperature fluctuations.

▾
Centrifugal pumps are often used as coolant circulating pumps; they have a simple design, sufficient head and a consistent capacity.

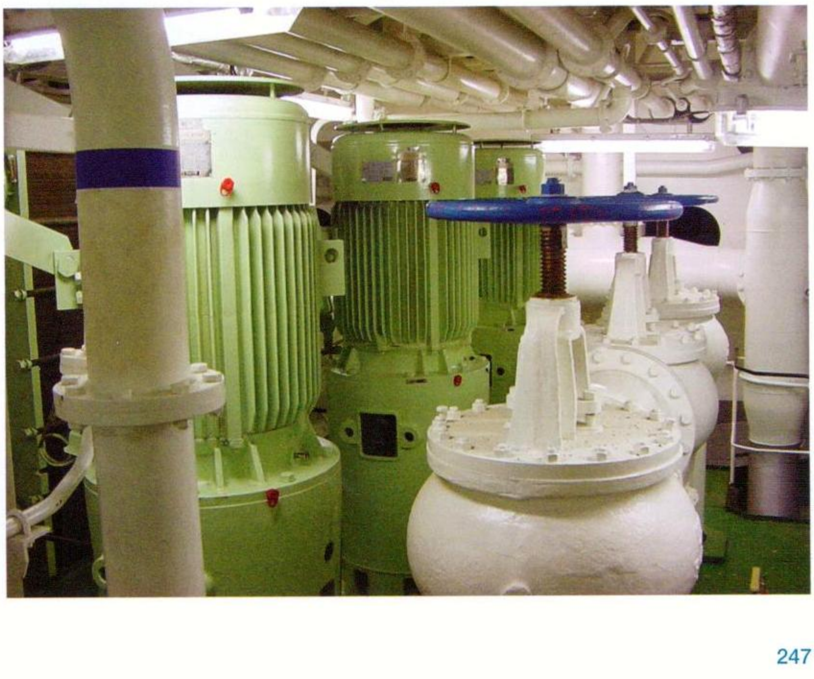


10.11.1 Pump capacities

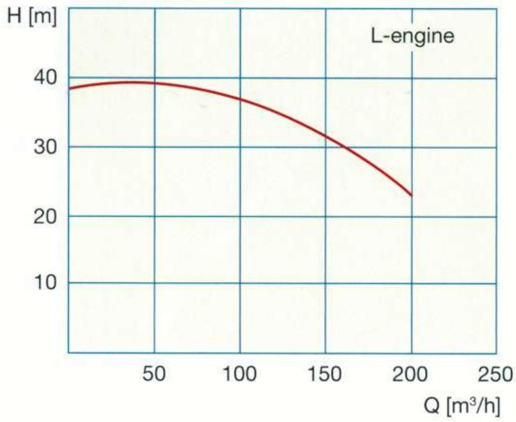
Most engine-driven cooling-water pumps are centrifugal pumps.

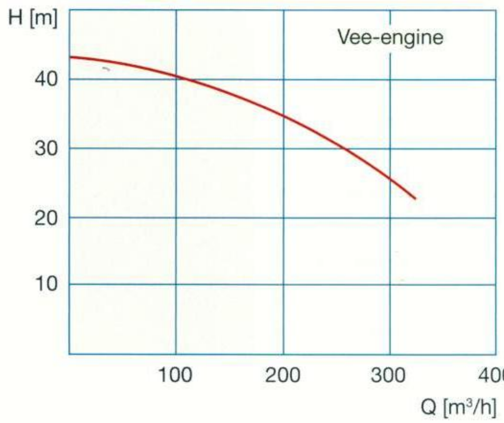
Each pump has an accompanying head/capacity curve generated at a certain RPM. Here the capacity is indicated in m^3 per hour, the discharge head in metres water column and the number of revolutions per minute.

A conspicuous detail here is that the discharge head drops considerably when the capacity is increased. Cause: the resistances in the pump casing and the discharge resistances of the whole system!
For certain engine types, the maximum capacity and head are specified.



▸
A series of coolant circulating pumps in a propulsion engine.





▶
Two pump characteristics or Q-h curves. Vertically the head is shown in metres water column and horizontally the capacity in cubic metres per hour. As shown the discharge head decreases as the capacity Q increases.

10.11.2 Pipe diameters

These are stated in a table for pipe diameters in which the velocities on the suction and delivery side of the pump are provided.

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

10.11.3 Cooler capacities

A cooler has to be designed to have some over capacity in connection with possible contamination on both sides of the heat exchanging surface.

Pipe dimensions
Recommended maximum fluid velocities and flow rates for pipework*

Nominal pipe diameter ↓	Flow rate [m/sec] Flow amount [m³/h]									
Media → Pipe material →	Sea-water Steel galvanized		Fresh water Mild steel		Lubricating oil Mild steel		Marine diesel oil Mild steel		Heavy fuel oil Mild steel	
Pump side →	suction	delivery	suction	delivery	suction	delivery	suction	delivery	suction	delivery
32	1.0 2.9	1.4 4.1	1.5 4.3	1.5 4.3	0.6 1.7	1.0 2.9	0.9 2.6	1.1 3.2	0.5 1.4	0.6 1.7
40	1.2 5.4	1.6 7.2	1.7 7.7	1.7 7.7	0.7 3.2	1.2 5.4	1.0 4.5	1.2 5.4	0.5 2.3	0.7 3.2
50	1.3 9.2	1.8 12.7	1.9 13.4	1.9 13.4	0.8 5.7	1.4 9.9	1.1 7.8	1.3 9.2	0.5 3.5	0.8 5.7
65	1.5 17.9	2.0 23.9	2.1 25.1	2.1 25.1	0.8 9.6	1.5 17.9	1.2 14.3	1.4 16.7	0.6 7.2	0.9 10.8
80	1.6 29.0	2.1 38.0	2.2 39.8	2.2 39.8	0.9 16.3	1.6 29.0	1.3 23.5	1.5 27.1	0.6 10.9	1.0 18.1
100	1.8 50.9	2.2 62.2	2.3 65.0	2.3 65.0	0.9 25.5	1.6 45.2	1.4 39.6	1.6 45.2	0.7 19.8	1.2 33.9
125	2.0 88.4	2.3 101.6	2.4 106.0	2.4 11.4	1.1 48.6	1.7 75.1	1.5 66.3	1.7 75.1	0.8 35.3	1.4 61.9
150	2.2 140.0	2.4 152.7	2.5 159.0	2.6 165.4	1.3 82.7	1.8 114.5	1.5 95.4	1.8 114.5	0.9 57.3	1.6 108.2
200	2.3 260.2	2.5 282.8	2.6 294.1	2.7 305.4	1.3 147.0	1.8 203.6	– –	– –	– –	– –
Aluminium brass	2.6 294.0									
250	2.5 441.8	2.6 459.5	2.7 477.2	2.7 477.2	1.3 229.8	1.9 335.8	– –	– –	– –	– –
Aluminium brass	2.7 447.2									
300	2.6 661.7	2.6 661.7	2,7 687.2	2,7 687.2	1.3 330.9	1.9 483.6	– –	– –	– –	– –
Aluminium brass	2.8 712.5									
350	2.6 900.5	2.6 900.5	2.7 935.2	2.7 935.2	1.4 484.9	2.0 692.7	– –	– –	– –	– –
Aluminium brass	2.8 969.8									
400	2.6 1176.2	2.7 1221.5	2.7 1221.5	2.7 1221.5	1.4 633.3	2.0 904.8	– –	– –	– –	– –
Aluminium brass	2.8 1266.7									
450	2.6 1488.6	2.7 1545.9	2.7 1545.9	2.7 1545.9	1.4 801.6	2.0 1145.1	– –	– –	– –	– –
Aluminium brass	2.9 1660.4									
500	2.6 1837.8	2.7 1908.5	2.7 1908.5	2.7 1908.5	1.5 1060.4	2.1 1484.6	– –	– –	– –	– –
Aluminium brass	2.9 2049.9									

* The velocities given in the above table are guidance figures only. National standards can also be applied.

◀
The pipe dimensions and recommended maximum flow velocities.

The velocities are denoted in metres per second.
For coolant pipe diameters of 32 to 200 millimetres, this lies between 1.5 and 2.5 metres per second on the suction side and between 1.5. and 2.7 metres per second on the delivery side. This data is shown in the column 'Fresh water – Mild Steel'.
The capacities on the suction side are 4.3 and 294 m³ per hour respectively and 4.3 and 305 m³ on the delivery side.
Too high a flow velocity generates too much resistance and consequently pressure losses in the pipes.
This table also provides a good guideline for other liquids, such as seawater, lubricating oil and fuels.
This is particularly significant for fuels when designing **'booster units'**.

An example of the identification marking of pipes.

By means of a standard coding, the identification of which gas or liquid flows through the pipes at any given point in the installation is possible.

Important data with regard to coolers

This concerns:

- the capacity in kW;
- the amount of flow in m³ per hour;
- the temperature difference before and after the cooler in degrees Celsius;
- the maximum pressure drop over the cooler on the fresh water side in kilopascal;
- the amount of flow for seawater or another liquid in m³ per hour;
- the maximum pressure drop over the cooler on the seawater side in kilopascal;
- the maximum fresh water temperature after the cooler;
- the contamination factor of the cooler. This constitutes often approximately 15%.

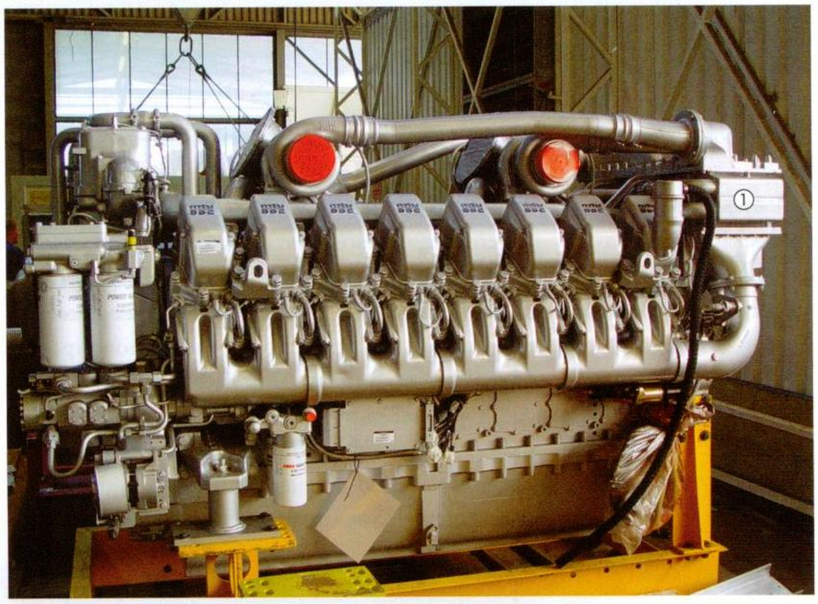
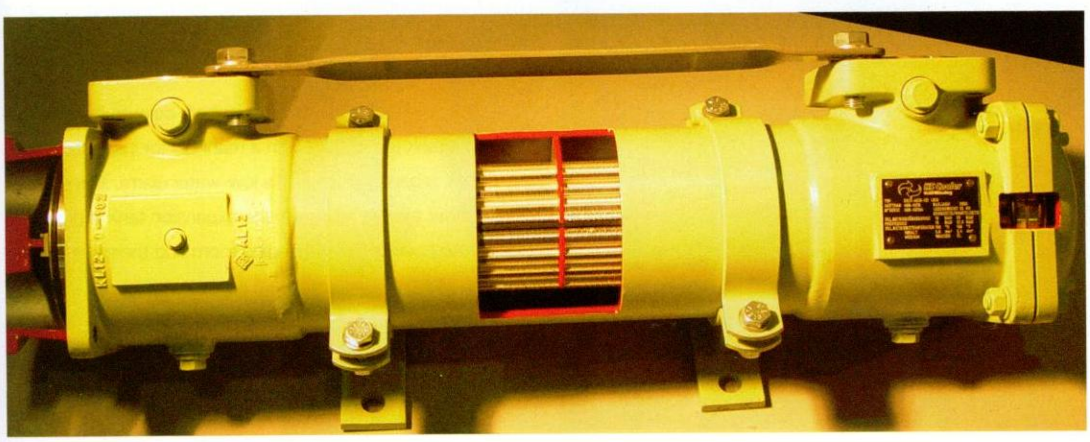


Plate coolers in a MAN–B&W four-stroke medium-speed diesel engine. Shown here the lubricating-oil and fresh-water cooler.

The air coolers (1) after the turbo-blowers in this high-speed four-stroke MTU-diesel engine are plate coolers.



In designing coolers, it should be taken into account that there may be some contamination of the cooling surfaces. Here a cut-way view of a pipe cooler.

▲
A plate-cooler detail.

On the right, the bolts are sufficient in 'length' to increase the cooling capacity by fitting more plates.

10.11.4 Fresh water-expansion-tank

This compensates for water volume fluctuations at different temperatures in the system. Furthermore, the fresh water cooling expansion-tank serves as an air bleed system, storage tank and generates the required static pressure on the suction pipe of the cooling-water pumps. So in stationary cooling-water pumps there is always pressure on the suction side of the pump which is why water is produced the instant the engine starts. **The centrifugal pumps are not self-priming and consequently experience problems with initial coolant flow when this connection is excluded.** Static pressure is formed by placing the expansion tank at a certain height in relation to the pumps. This is often between 5 and 15 metres water column. The volume is usually at least 10 to 15% of the system content. The cooling-water treatment chemicals can often be dosed in this tank. The individual bleeding pipes of each engine are fitted to the tank. **These pipes should never be placed horizontally in order to avoid air accumulation.**

10.11.5 Cooling-water drain tank

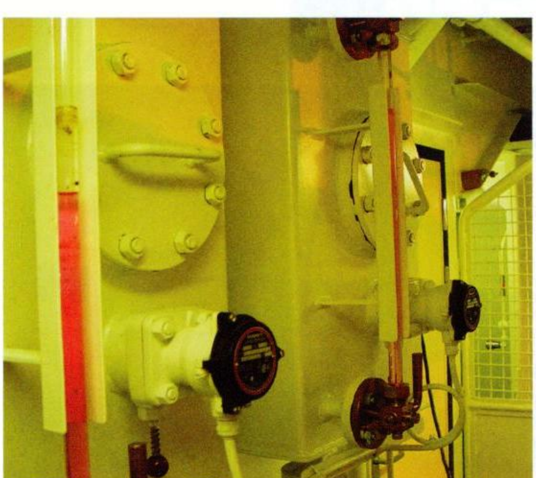
It is recommended in larger systems to drain the cooling-water into a tank so it can be re-used after cooling-water system repairs.

10.11.6 Cooling-water pre heater

For the slightly larger engines it is advisable to either preheat the cooling-water and therefore the engine block or maintain the correct temperature for an extended time before starting the engine. This is particularly important for back up gensets in order to achieve an easy start and to avoid heat tension. At sea stopped diesel engines are generally kept at the correct temperature with pre heating systems. This also applies to heavy fuel systems. As a result of short berthing times a large part of the machinery, such as pumps, remains operational and the temperatures of fresh water cooling, lubricating oil and fuel remain at a constant operating temperature.

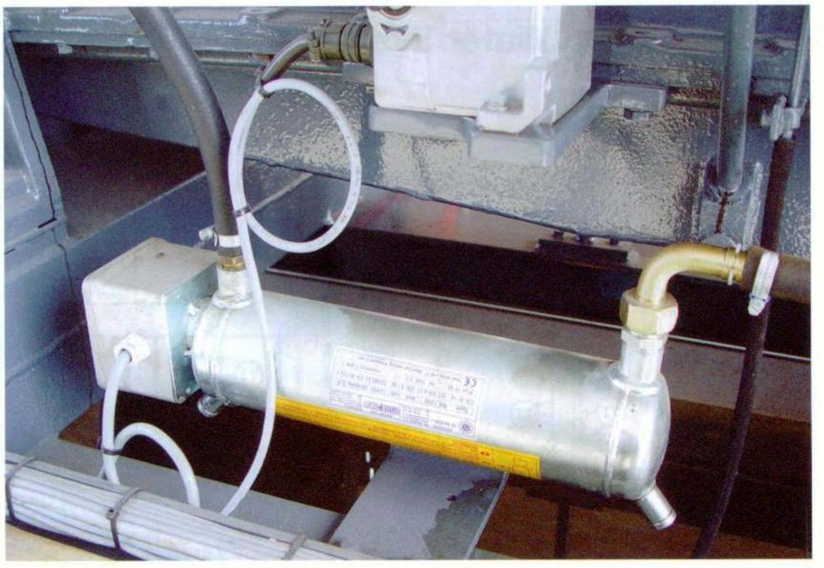
►
A coolant storage expansion tank in a small high-speed four-stroke diesel engine.

The system is under excess pressure. Opening the filler cap (1), can be very dangerous (burns).



◄
Two fresh-water expansion tanks for high-speed four-stroke Deutz-diesel engines.

The cooling-water treatment shown here colours the coolant pink. The tank is provided with a manhole for inspection purposes, a gauge glass and a low-water alarm. The system is not under excess pressure; the expansion tank is placed in an elevated position in the engine room and therefore there is a static pressure on the coolant pumps.



An electric heating element mounted on the coolant system of a stand-by diesel generator.

A stable coolant temperature guarantees a smooth quick start of the diesel engine. Obviously, this is absolutely imperative in emergencies.

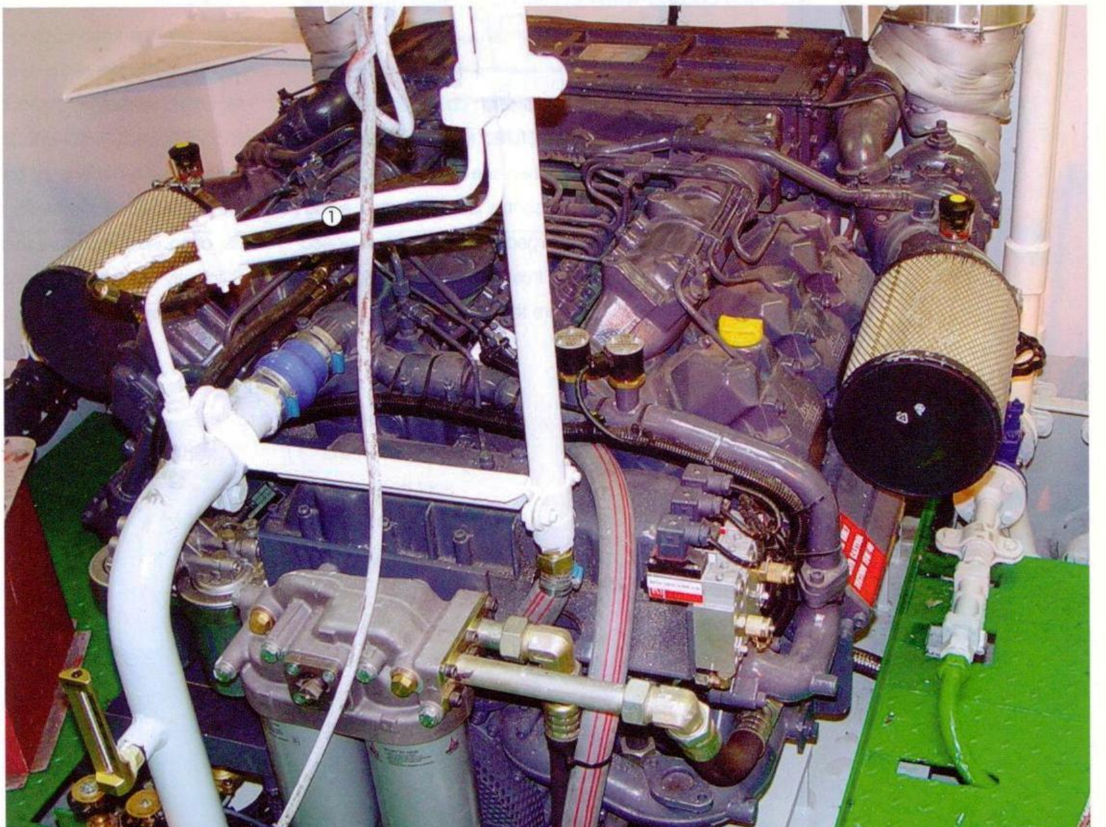
This also applies to stopped diesel engines in diesel power plants. Heated cooling-water is also often used for emergency diesel engines or back-up gensets. Finally, in cold regions, small diesel engines are warmed up or placed in heated spaces to achieve suitable temperatures prior to starting.

10.11.7 Deaerating systems

These are very important in cooling-water systems. Air and other gases can cause corrosion of materials and cool the engine parts poorly, which can cause severe damage. Pumps do not operate properly in systems containing air. Vents should be fitted on top of the systems or engines and be connected to a vent tank or expansion tank by pipes mounted vertically and preferably without bends and horizontal sections.

10.11.8 Adjustable restrictions or orifices

These have been fitted to adjust the cooling-water circulating pumps and the pressure drop when the water does not flow through the cooler.

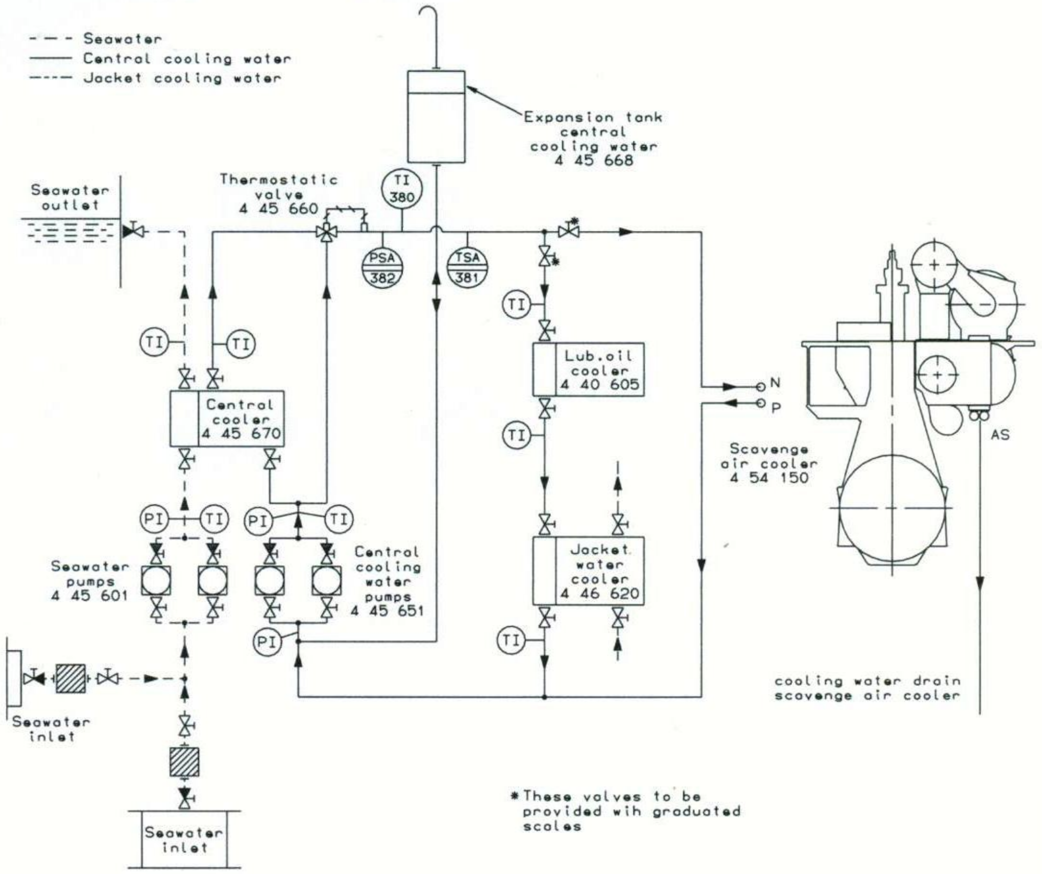


Cooling-water vents.

Cooling-water vents are fitted on the top side of a system and should run upward to the cooling-water expansion tank. In horizontally placed vents, potential air bubbles can remain in the system and cause damage to the engine (tearing).

1 *cooling-water vents*

Central Cooling Water System

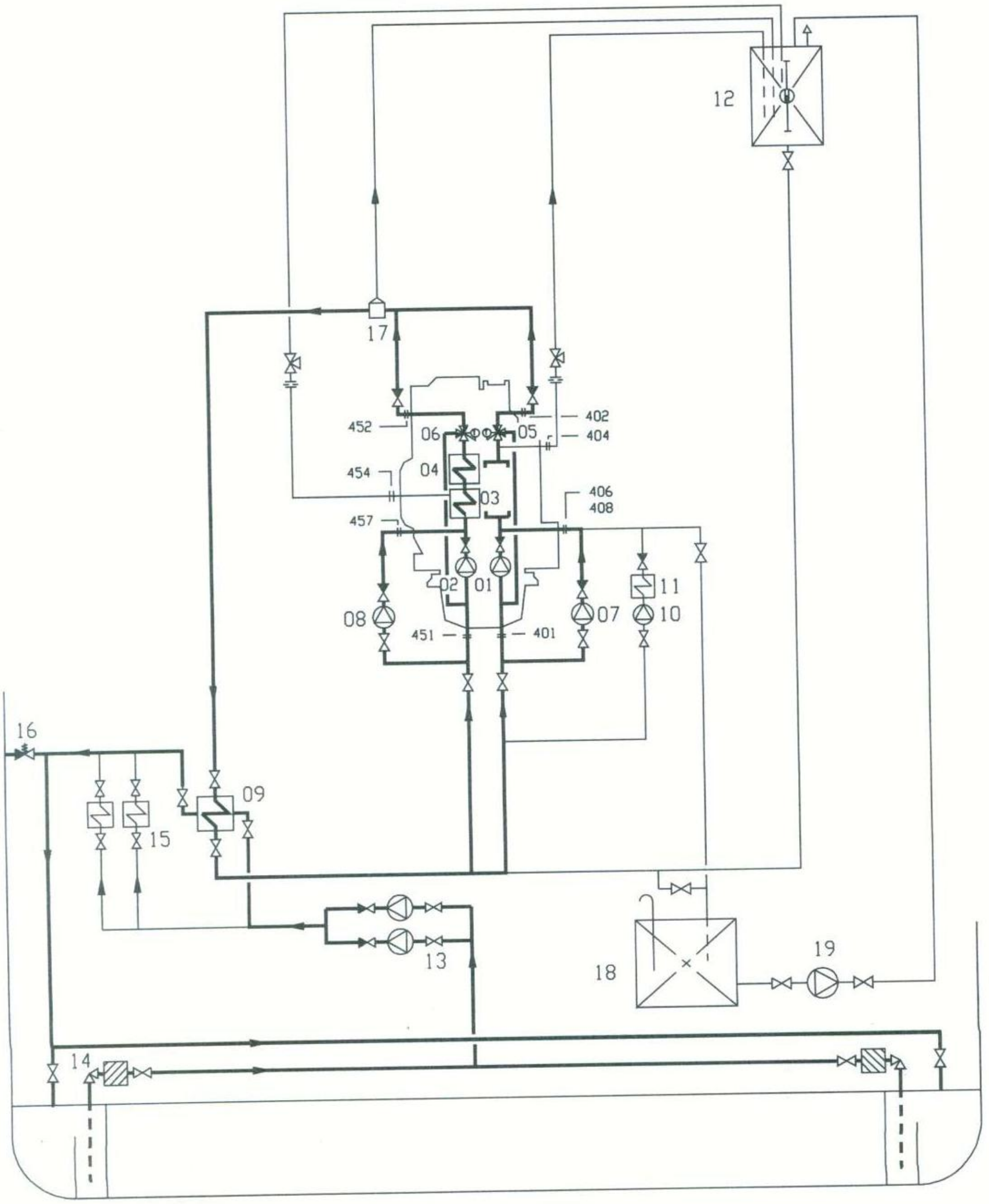


▲

A central cooling-water system for a large two-stroke crosshead engine.

The seawater cooling system has been designed to be as short as possible and is manufactured from corrosion-resistant material. It consists of two sea inlets, two cooling-water circulating pumps and a central cooler. Further along in the ship the seawater is discharged over board. The closed cooling system comprises two fresh-water circulating pumps, the central cooler where the heat is discharged to the seawater, the scavenging-air cooler, the lubricating-oil cooler and the fresh-cooling water cooler (jacket water cooler). At the top of the system the coolant expansion tank has been installed. A thermostatic control valve regulates the amount of fresh-water coolant circulating through the central cooler,. This is dependent on the required temperature of the fresh-water coolant system.

It is, of course, important that the scavenging air has a sufficiently low temperature of between 40 and 50 °C.



▲

A complete cooling-water system for a four-stroke medium-speed propulsion engine.

System parts

01 HT cooling-water pump
02 LT cooling-water pump
03 Scavenging-air cooler
04 Lubricating-oil cooler
05 Three-way valve, thermostatic HT
06 Three-way valve, thermostatic LT
07 HT level at cooling-water pump
08 LT level at cooling-water pump
09 Central cooler
10 Pre-heater pump
11 Pre-heater
12 Expansion tank
13 Seawater pump
14 Seawater inlet filter
15 Lubricating-oil cooler reduction gear
16 Seawater discharge valve
17 Vent
18 Drain tank
19 Transfer pump

10.11.9 Cavitation

This is a phenomenon that may occur at low water pressures and relatively high cooling-water temperatures. At 1 bar, water boils at approximately 100 °C. For instance, when the H.T.-cooling-water system on the suction side of the cooling-water pump, has a temperature of 75 °C and the piston pressure drops below 0.4 bar absolute, which is 0.6 bar under pressure, then the water will boil at the entrance of the pump; vapour bubbles will form in the inflowing liquid. In the compression section of the pump the water pressure is increased considerably to, for instance, 3 bar and water only achieves its boiling point at 133 °C. The vapour bubbles are then compressed the so-called imploding. **During this process huge forces are released.** Vapour bubbles which implode against the material surface cause severe damage to the material. Pump vanes begin to show an open structure and wear very rapidly. The pump capacity is then quickly reduced.

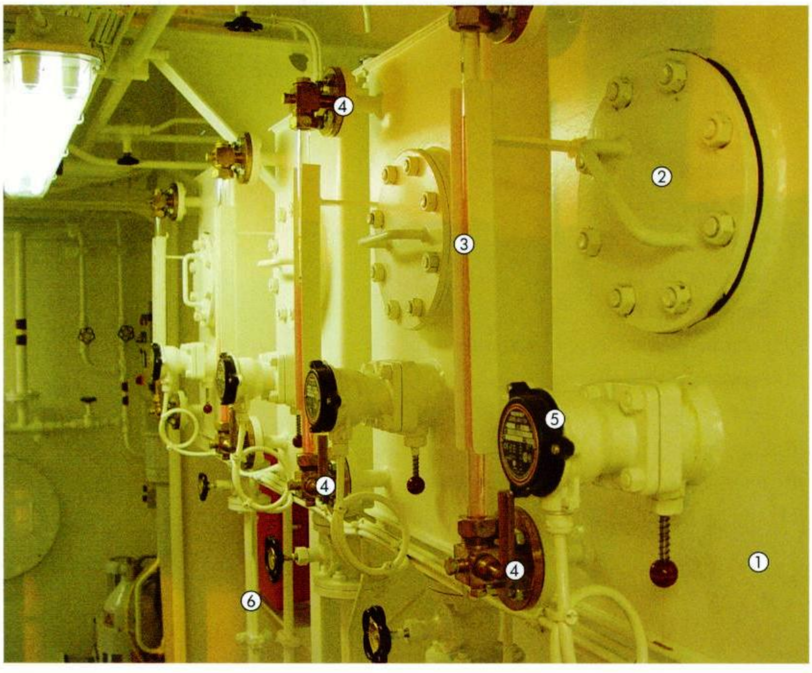
Note
Keep the pressure in the suction line at a minimum of 0.5 bar above the vapour pressure of the cooling-water present at its boiling point.

Example
Cooling-water temperature 45 °C
Boiling point at 0.1 bar absolute
Safe suction pressure 0.1 + 0.5 = 0.6 bar absolute
Boiling point at 0.6 bar absolute is 85 °C
So play is 85 – 45 = 40 °C

▼
Shown here each engine has an individual cooling-water expansion tank.

The connections of the cooling-water expansion pipes are found below the tanks.

1 *cooling-water expansion tank*
2 *manhole*
3 *liquid-level gauge*
4 *shut-off valve for the liquid-level gauge*
5 *float for low-water level indication*
6 *pipe to cooling-water system*



10.12 Cooling-water system defects

10.12.1 Air in the cooling-water system

Air in the system inhibits the flow and results in little or no cooling of engine parts, which causes damage such as tearing, cracking and piston seizure. Solution: ensure that the expansion tank is filled to the correct level and ensure that the air bleeder pipes are open to air. The air deaerators should be fitted below the water surface in the expansion tank.

Make sure there are no flattened or bent pipes and/or incorrect packing between the flanges that have under sized holes. There should be a pipe running straight from the expansion tank to the suction side of the cooling-water pump in order for the pump to always be filled with water.

10.12.2 Exhaust gases in the cooling-water system

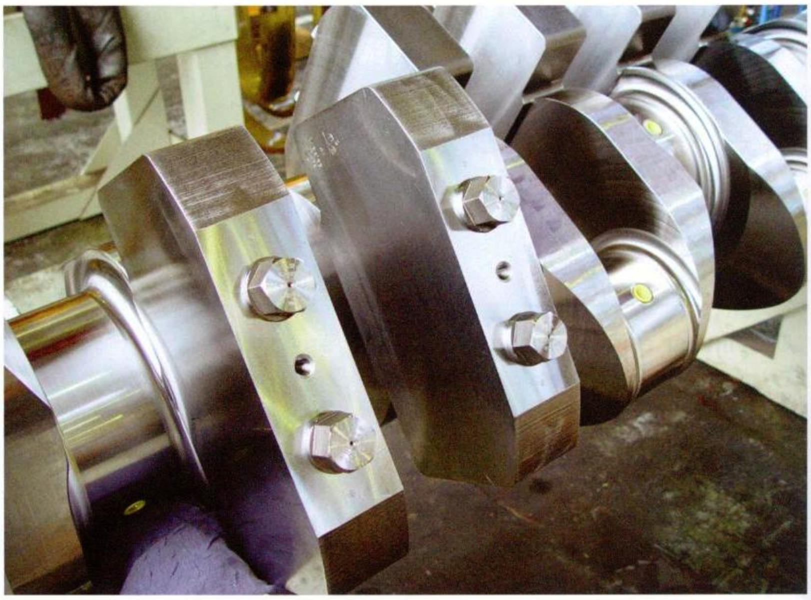
In cooled exhaust gas manifolds or double walled casings of turbo-blowers, exhaust gas can enter the cooling-water of a running engine due to tearing. This can also occur as a result of a packing leakage under the cylinder head. **In stationary engines the cooling-water can finish up in the exhaust manifold, turbo-blower or combustion space by tearing or a faulty packing and produce severe problems!** Controls and safety alarms for the water level in the expansion tank are therefore absolutely imperative. A **humming sound** in the expansion tank is usually indicative of gas leakage- often leaking cylinder head packing.

10.12.3 Cooling-water capacity is too low

Apart from (shut off) valves that are not entirely closed, the pump impeller can be affected by cavitation, thus reducing the capacity. Severe contamination in the system produces an increased head due to the increased resistance in the system. Consequently, the heat transmission to the second cooling-water system is reduced.

▶
A crankshaft with counterweights for balancing the moving parts.

The combustion sequence is very important when balancing the engine.



The manner in which the exhaust gases manifolds of each cylinder are connected to a shared pipe is entirely dependant on the selected ignition sequence. Let's assume that the ignition sequence as a result of the desired crank degrees balancing conditions is as follows 1–5–3–6–2–4. Then the cylinders should be connected as follows: 1 to 3, 2 to 5 and 6 to 4. The crank degrees between the pressure surges are 240°.

Trick

In an ignition sequence, one cylinder can be skipped each time. Consequently, for an ignition sequence of 1–2–4–6–5–3–1, 1 is connected to 4 and 5 to 2 and 6 to 3.

12.10.3 SPEX-system

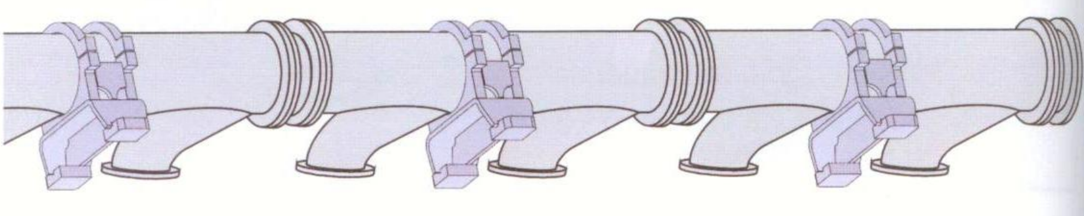
The single exhaust pipe-system comprises a single exhaust pipe in which the individual exhaust ducts from each cylinder are bent at right angles into the pipe in the exhaust gas flow direction. In this manner, the exhaust gases from the cylinders do not disrupt one another; therefore a very simple exhaust pipe is obtained.

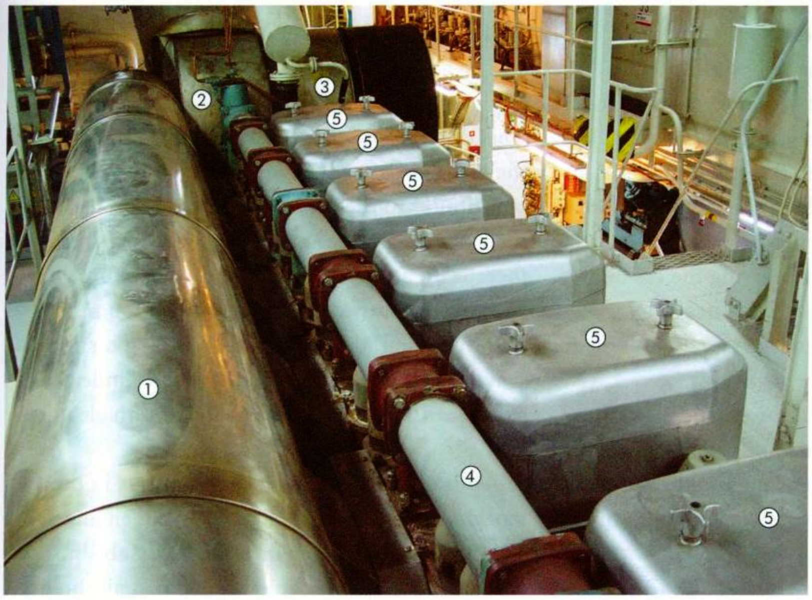
12.10.4 Pulse-Converter system

Here, the advantages of both the pressure pulse and constant pressure system are used. So, the pressure pulses, combined with the even flow in the gas turbine are used to increase the efficiency.

▶
The curved individual exhaust-gas pipes from each cylinder are attached to the common exhaust-gas receiver by a radial pipe.

▶
The SPEX-exhaust system of engine manufacturer Wärtsilä.





◄

A single turbo-blower is adequate for the air supply of this six cylinder in-line engine, a Wärtsilä 46 medium-speed heavy-fuel oil engine.

The exhaust-gas receiver is of simple construction and is not cooled.

1 *exhaust- gas pipe*
2 *turbo-blower, exhaust section*
3 *turbo-blower, air section*
4 *coolant discharge pipe*
5 *cylinder heads*

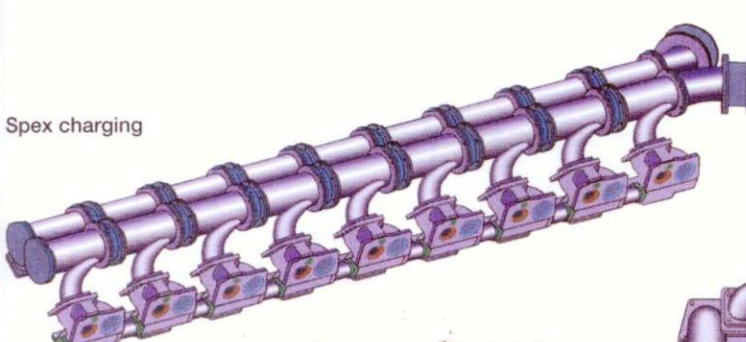
The exhaust gas manifolds of two cylinders, of which the ignition time is shorter than the time that both exhaust valves are open, are connected to a diffuser via jet nozzles, this leads to a joint collection pipe for the exhaust gas turbine. The cylinder exhaust pressure pulse is converted into speed through the nozzle. The gas from the other cylinders is not accelerated and therefore flows to the joint collection pipe at a lower speed. Here the kinetic energy is exchanged. The exhaust gas speed from the first cylinder is slowed down by the exhaust gas speed from the second cylinder; this in contrast is accelerated due to the velocity of the exhaust gas from the first cylinder.
In the diffuser the collective speed is converted into pressure and the exhaust gases then flow at a constant but increased speed to the turbine.

Advantages and disadvantages

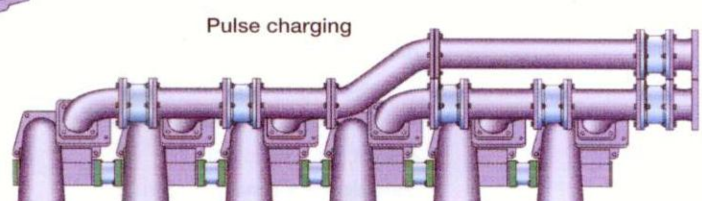
Constant pressure and pressure pulse system
In practice one would like to see the advantages and the disadvantages of the constant pressure and pulse surge system (see picture below).

Advantages constant pressure system
- The number of turbo blowers and pipes is not dependant on the number of cylinders. Generally, one turbo blower placed at the end of the central exhaust gases pipe suffices.
- The mean exhaust gas pressure is lower during the exhaust stroke in comparison to the pressure pulse system, as the gases are led to more capacious manifold, this reduces flow resistance.
- As a result of the even flow of the exhaust gases, turbine efficiency is high, and therefore the compressor can operate at maximum capacity.

Spex charging



Pulse charging



▲
The SPEX supercharging system of Wärtsilä with next to it a pulse-charging system.

Disadvantages constant pressure system

- At low engine loads, the scavenging air pressure may drop below that of the exhaust gases, it is therefore possible that the exhaust gases can flow back into the scavenging air line. This negatively affects the efficiency of the exhaust air scavenging process. Instead of the exhaust air scavenging removing the residual gases and cooling the hot parts near the combustion space, the parts are poorly cooled, thus causing various problems.
- Rapid power increase of the engine may cause temporary incomplete combustion, as the large volume of the exhaust gas manifold results in a delayed reaction in the power ramp up of the engine.

12.11 Air supply in two-stroke crosshead engines

This differs from the four-stroke engine in that the piston of a two-stroke engine in no way contributes to the aspiration of fresh air, because there is no separate suction stroke. Moreover, there is no actual exhaust stroke. Therefore, an independently driven scavenging pump is required. First, the supplied scavenging air should scavenge all exhaust gases from the cylinder and subsequently fill the entire cylinder with clean air.

Presently, there are three manufacturers in the world market that produce crosshead engines: **MAN–B&W, Wärtsilä Sulzer and Mitsubishi.** Essentially, the scavenging principle in all three systems is identical, namely:

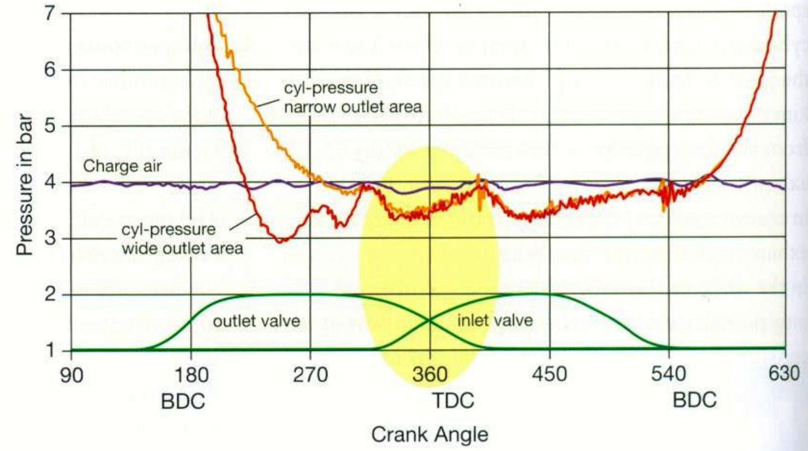
- uniflow scavenging.

Time passage chart of a four-stroke-diesel engine

▶

In this line chart of a four-stroke diesel engine, the scavenging process is encircled.

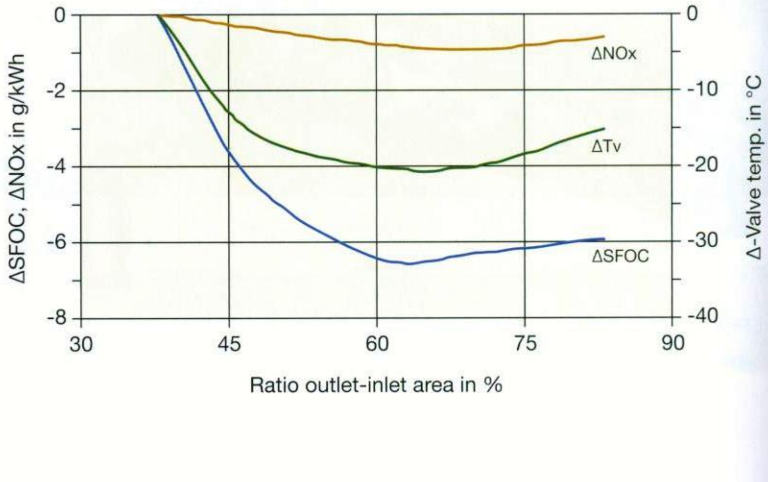
At the top position of the piston, shown here at 360 crank degrees, the exhaust - and inlet valves are momentarily open to cool the hot engine parts. The scavenging-air pressure remains at approximately four bar during scavenging. In total, scavenging takes approximately 90 crank degrees.



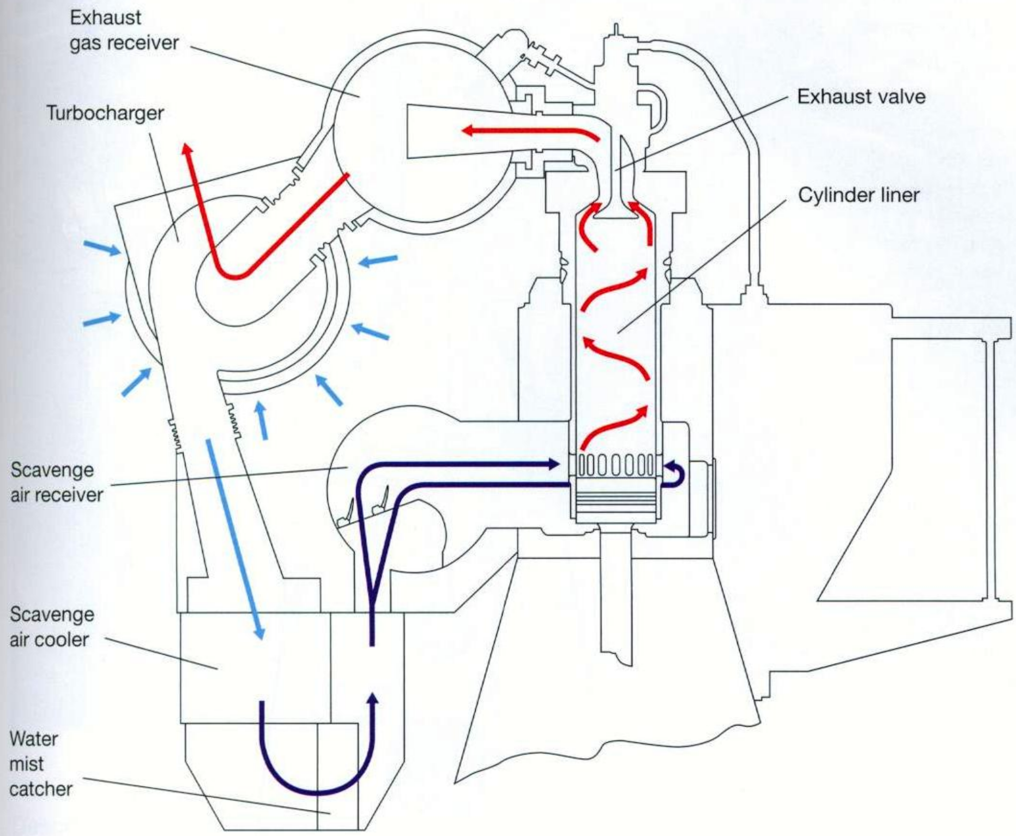
▶

The scavenging process in top dead centre with different valve overlap.

At approx. 65% the NOx content, the specific fuel consumption and the exhaust valve temperature are at their lowest level.

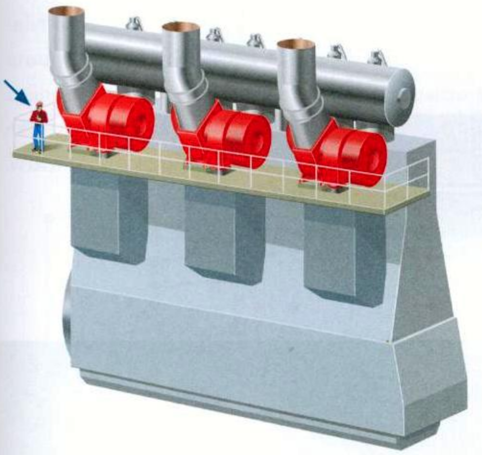


12.11.1 Uniflow scavenging



The principle of uniflow scavenging in a two-stroke crosshead engine.

When the piston is in bottom position and the inlet ports are open, the scavenging air flows from the scavenging-air space through the inlet ports in a screw movement through the cylinder in the direction of the opened exhaust valve.
In this way, the air is supplied from the scavenging ports mounted at the bottom of the cylinder liner and discharged via the exhaust valve mounted in the cylinder head.
Scavenging occurs from the bottom to the top.



Large two-stroke crosshead engines use vast amounts of air.

The capacity of the largest turbo-blowers is limited and therefore two to four turbo-blowers are mounted on the central exhaust-gas receiver in order to supply sufficient air. Note, the size of the mechanical engineer left on the platform!

12.11.2 Scavenging effect or Scavenging ratio R

In this type of engine one refers to scavenging effect or scavenging ratio.
This is the air volume which is added per cycle and per cylinder in relation to the stroke volume of the cylinder.

$$R = \frac{\text{Actual added air volume}}{\text{Stroke volume of the cylinder}}$$

During the uniflow process R is approximately 1.2 to 1.3 and over 95% of all the exhaust gases are removed. Five percent is residual gas.

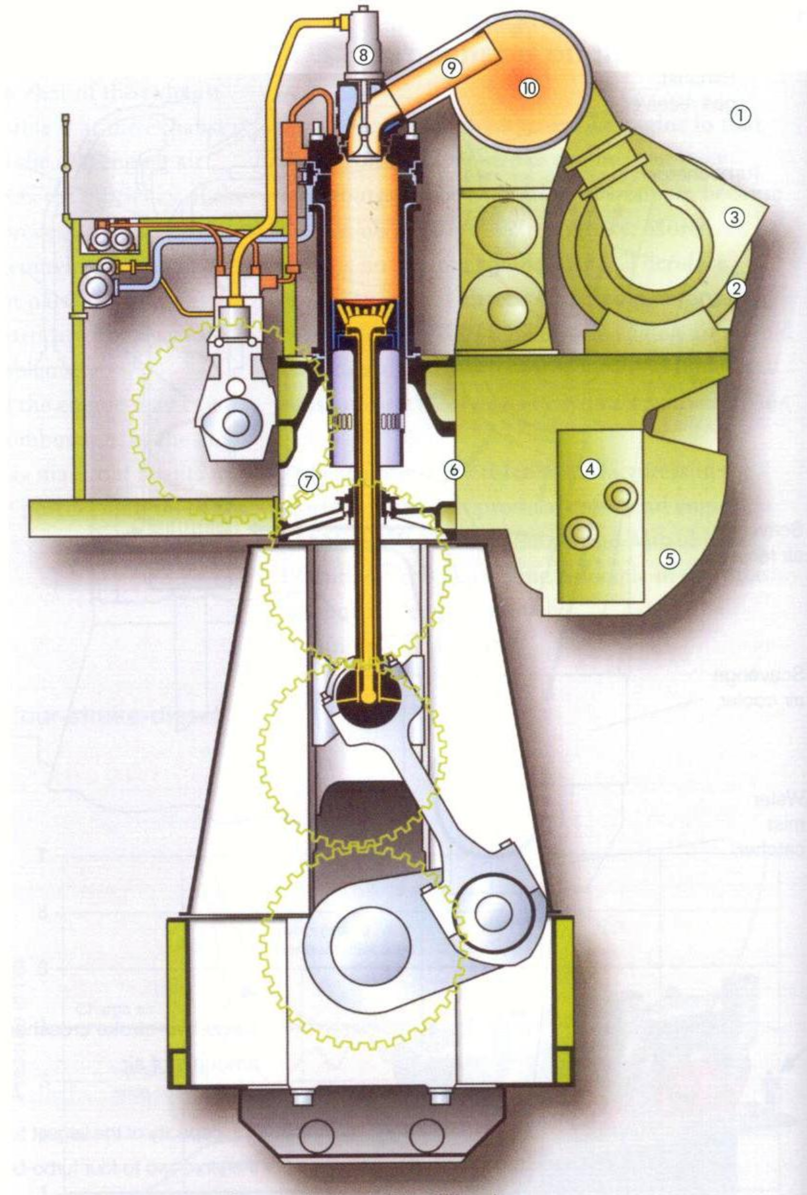
$$\text{Scavenging effect} = \frac{\text{Volume of clean air in cylinder}}{\text{Stroke volume}}$$

$$= 90 \text{ to } 95\%$$

►

The pressure-charging system of a two-stroke crosshead engine.

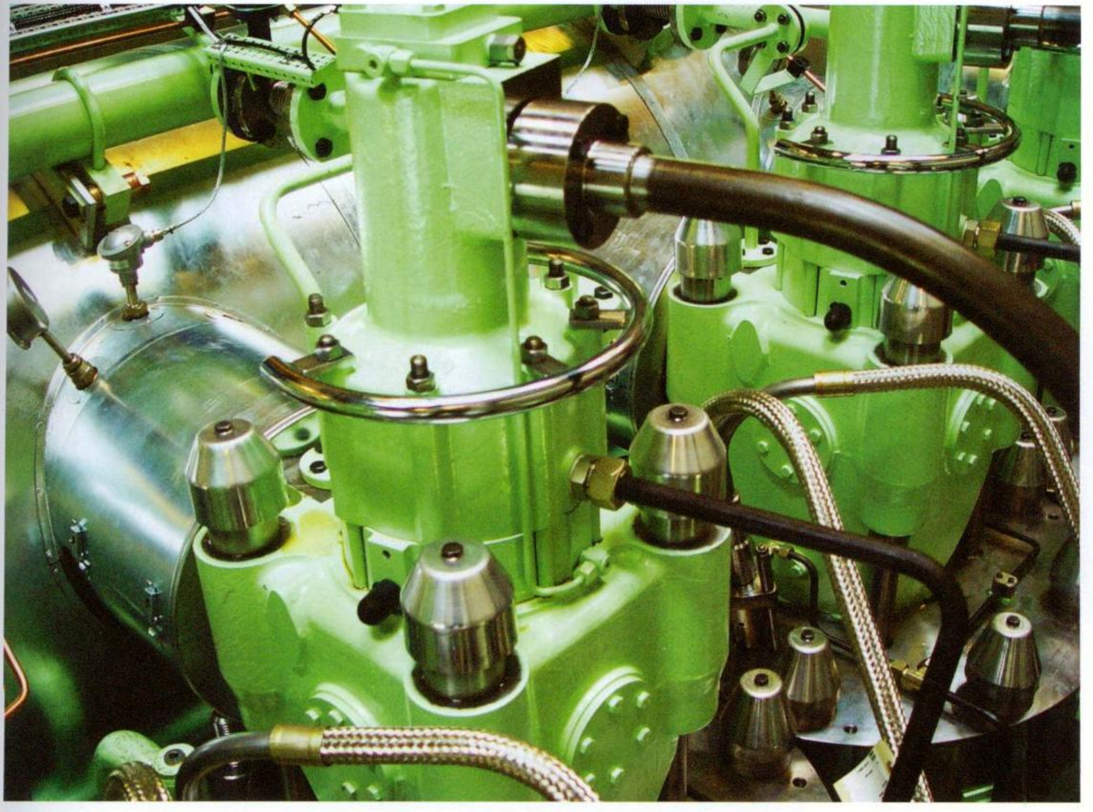
1 *ambient air*
2 *air-inlet filter*
3 *turbo-blower*
4 *air cooler*
5 *moisture separator*
6 *flap valves (not visible)*
7 *scavenging-air space*
8 *exhaust valve*
9 *exhaust of one cylinder*
10 *central exhaust gas-receiver*



Cross section classic Sulzer RTA engine

►

Via the inlet ports bottom side liner, air enters the cylinder (see arrow).

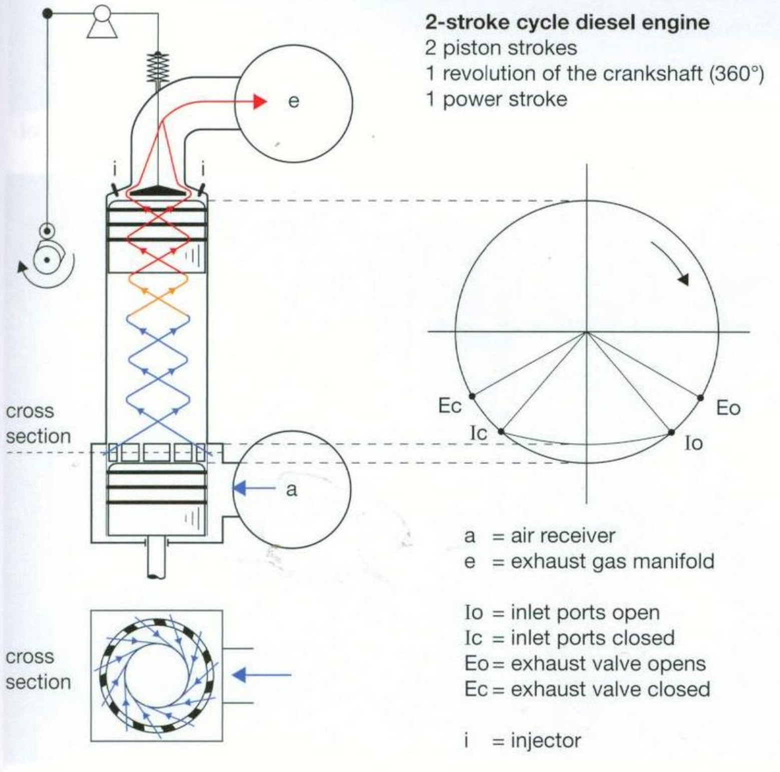


◄ The hydraulically operated central exhaust valve.

The hydraulically operated central exhaust valve of every cylinder opens during the power stroke at approximately 110 crank degrees after top to discharge the exhaust gases and scavenge the cylinder.

Description of the scavenging process

The entire circumference of the lower part of the cylinder is fitted with tangentially placed inlet ports connected to the scavenging air space around the bottom part of the cylinder. The cylinder cover has one large, centrally positioned exhaust valve, which is hydraulically opened. When the inlet ports are released by the piston during the downward stroke, the scavenging air pressure is higher than the exhaust gas pressure so scavenging occurs swiftly and effectively. This is the reason that the exhaust valve opens just prior to the release of the scavenging ports by the piston, to bring the exhaust pressure below that of the scavenging air pressure.



◄ The principle of the two-stroke engine with right the crank circle.

The height of the scavenging ports varies between approximately 10 tot 15% of the piston stroke.
From the scavenging-air receiver, the air is supplied at the moment that the piston releases the inlet ports via the tangentially positioned scavenging ports. Due to this, the scavenging air a moves in a rotating movement in the cylinder and removes all the residual gases via the exhaust pipe.

▶
Cylinder liners of a DETROIT high-speed two-stroke trunk-piston engines.

Left a indirectly cooled and right a directly cooled version . Note: the difference in port height.

Heat transfer:
- indirectly via the material of engine block
- directly via coolant in the cylinder liner

▶▶
Cylinder liners in large two-stroke crosshead engines have a height of approximately 4 metres and a mass of 11,000 kg.

The tangentially placed scavenging ports guide the incoming air along the cylinder liners, therefore scavenging shadows are minimized. The ports themselves often have 'fingers' fitted; notches which further increase the turbulence of the air and assist in the removal of the residual gases.

In this type of scavenging one also refers to the 'swirl factor'.

This is the ratio $\frac{r}{R}$; r is the radius of the circle of the inlet air pipe and R is the radius of the cylinder.
In modern engines the swirl factor is approximately 0.3. After the scavenging period the inlet ports are closed by the upward piston stroke and shortly afterwards the exhaust valve.

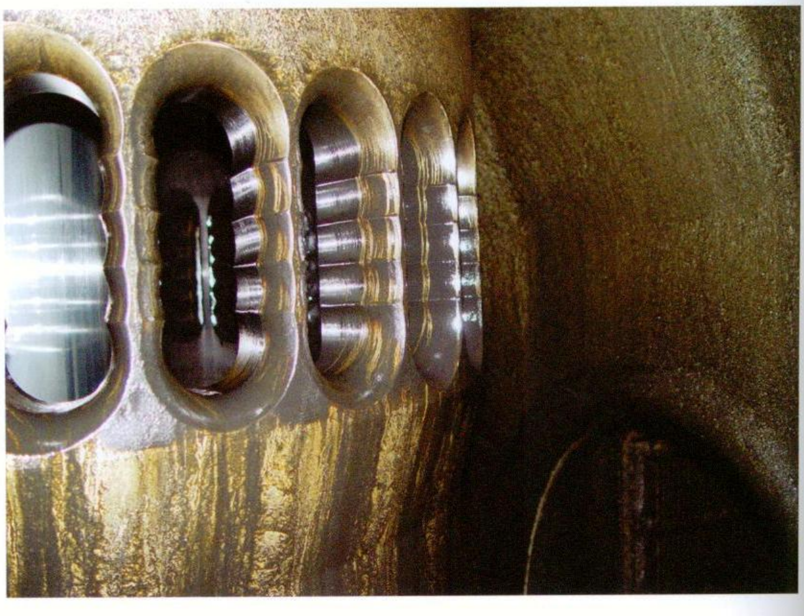
Air compression can now effectively start! At this stage at least 20% of the upward stroke has been completed!

12.12 Supercharging in two-stroke crosshead engines

Equal- or constant pressure system

Here the exhaust gases of all cylinders are discharged to a capacious, central exhaust gas manifold or 'exhaust gas receiver'.
Pressure pulses are levelled off, therefore the gas flow to the exhaust gas turbine(s) takes place at a constant pressure and velocity.
Due to the constant gas flow both turbine flow efficiency and capacity are high. The number of turbo blowers depends solely on the mass flow of

▶
The tangentially positioned inlet ports have a 'ribbed' profile to promote the scavenging -air turbulence and therefore better remove the residual exhaust gasses.



the exhaust gases and the turbo blowers capacities that are available.

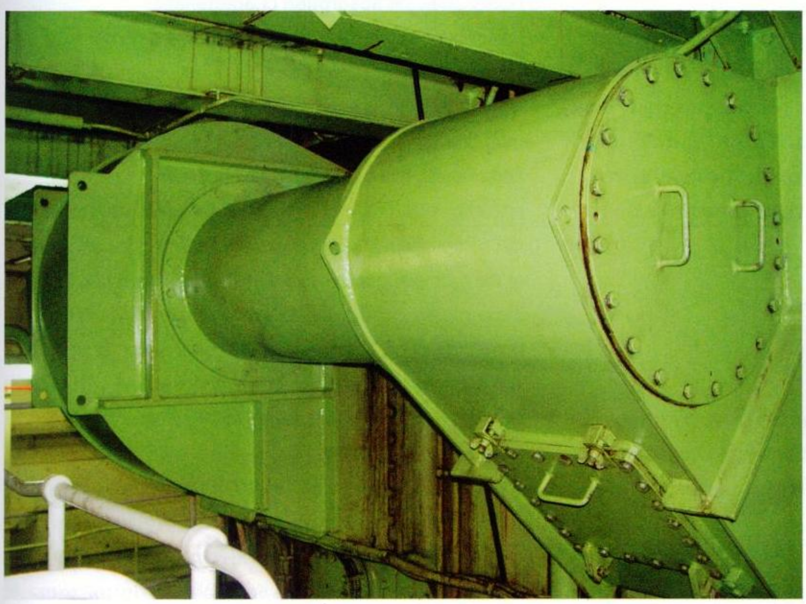
Impact- or pulse system

Each cylinder is connected to a narrow, short exhaust gas manifold, which is directly connected to the exhaust gas turbine. The kinetic energy of the discharged gases and the expansion energy of the exhaust gases can be easily converted in turbo blowers which are placed in close proximity to the cylinders concerned.

Disadvantages of this system

- The flow efficiency of the turbine is lower as the exhaust gases flow in pulses or bursts.
- The cylinder number is often limited to a maximum of three in order avoid disturbing the scavenging of the adjacent cylinders.
- A nine cylinder engine therefore has three turbo blowers.

In practice, all three types of two-stroke crosshead engines that are still manufactured are equipped with the constant pressure system. Experience has shown that the constant pressure system is the optimum system for high cylinder capacities with mean effective pressures between 1.6 and 1.9 MPa or between 16 and 19 bars.
However, diffusers are used to convert kinetic energy into pressure. Observe, for instance the RTA-series by Wärtsilä Sulzer and the MC-series by MAN–B&W.
The third two-stroke-crosshead engine manufacturer Mitsubishi also uses the constant pressure system, including a diffuser. This type has three cylinders connected to one diffuser, which leads to the central exhaust gas receiver.



◄ The electrically driven auxiliary blower in a two-stroke crosshead engine.

This starts automatically as at reduced power the scavenging-air pressure is too low.

◄ A ten-cylinder two-stroke crosshead engine with the exhaust pipes from every cylinder directly attached to the very spacious central exhaust-gas receiver.

Depending on the engine dimensions, one to four turbo-blowers can be installed.

12.12.1 Three engine manufactures

The latest design by MAN–B&W is the MC-series, for Wärtsilä 'Sulzer' the RTA-series and the UEC-LS-series for Mitsubishi.
They all utilize the equal- or constant pressure system with diffusers, uncooled turbo blowers and non-return flap valves in the supply pipes to the scavenging air spaces. In order to avoid scavenging problems in low loaded engines, an extremely low scavenging air pressure in relation to the exhaust gas pressure, all manufacturers make use of electro driven auxiliary blowers which automatically switch on when the scavenging air pressure drops below a certain point.

For an extensive engine description see Chapter 21, Diesel engines manufacturers.

12.12.2 Absorption characteristics

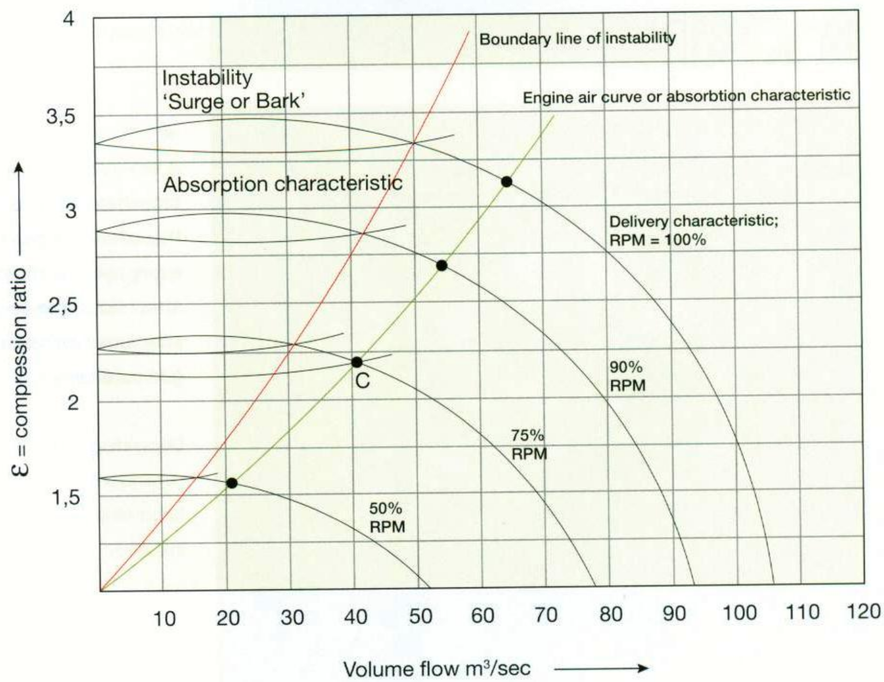
The turbo blower compressor delivers a certain amount of air to the scavenging air receiver which has a certain scavenging air pressure. It then flows through the opened inlet valves into the cylinder (four-stroke engines) or through the inlet ports released by the piston into the cylinder (two-stroke crosshead engines).

The efficiency of the centrifugal compressor depends on the suction lift.
For every RPM, there is a certain supply characteristic.

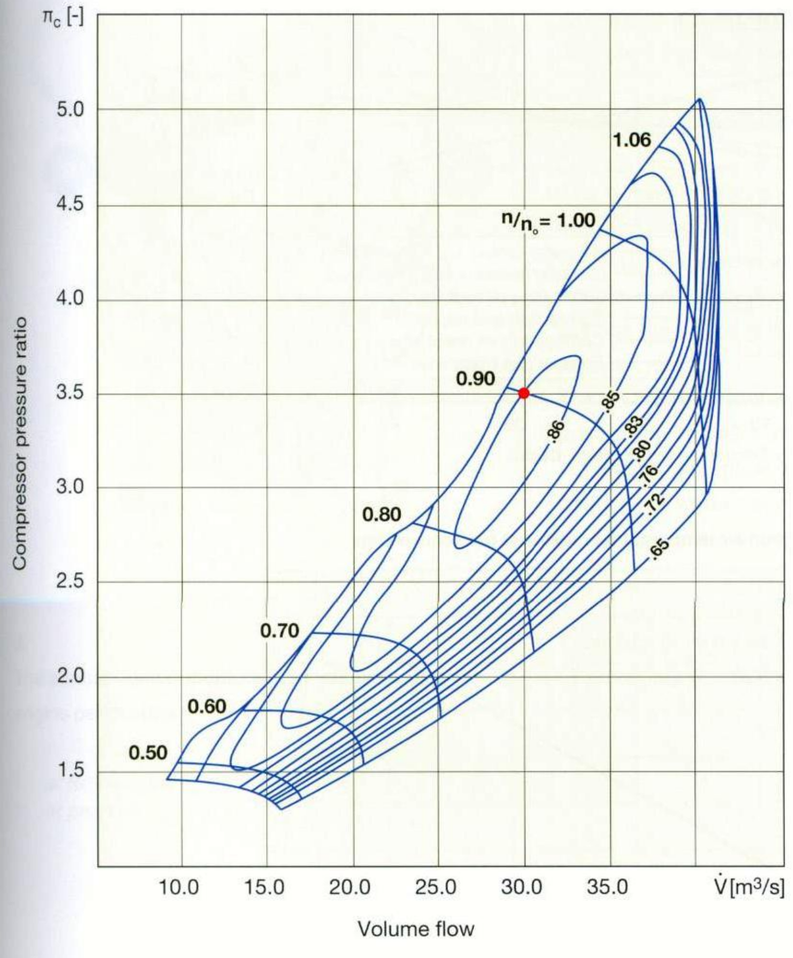
Turbo blower suppliers also use these charts. They are often called MAP's instead of charts and have the pressure ratio or the compression ratio on the vertical axis, in stead of the compressor pressure.

▼
Absorption characteristic.

Description: Horizontal; the volume flow in m^3 per second. Vertical; compression ration, MPa of bar or as shown in the compression ratio.
Speed: In the graph, four speeds are shown with their own delivery characteristic. The point where the compressor is set is known as the operating point. Connecting these four points gives the **absorption characteristic**.
Left of the **of instability boundary line,** the compressor becomes instable. The piping characteristics are also shown and in this case also the four points where the compressor map cuts the four speed lines.
Left of these four intersection points the pressure in the scavenging air piping is higher than the pressure supplied by the compressor. The air flows backwards in a pulsing motion; at the moment that the scavenging air pressure falls below the compressor map, the flow reverses and flows in the correct direction. This is known as compressor 'Surge'. The sound produced is known as 'shock waves'. The vibrating rotor may be damaged through this.



Example of compressor map for TPL85-B



◄ A compressor map of an ABB–TPL 85-B turbo-blower.

Example: With an air output of 30 m^3 per second and a compression ratio of 3,5, the speed is 90% of full load and the compressor yield 86% (red point).

12.12.3 Bypass- valve

To prevent compressor instability at a reduced capacity in a four-stroke engine and stop engine surge or 'barking', a bypass valve placed in the pressure supply line can be opened, in order for the compressed air on the suction side of the compressor to flow back into the non-pressurized end of the intake. This requires increased compressor output and the compressor remains to the right of the instability boundary line.
This usually occurs in propulsion engines with a variable RPM.
Most four-stroke propulsion engines have a permanently high RPM and control the speed and consequently the engine load with an adjustable pitch propeller, therefore 'barking' very seldom occurs. Four-stroke engines gensets should not experience any problems either as they operate at a constant RPM.

In **two-stroke crosshead engines with a fixed propeller used for propulsion which is reversible** and has a variable RPM, instability of the compressor does not normally occur. The compressor actually becomes more stable at a reduced output and/or RPM.

12.13 Some important points of interest with regard to the air supply in diesel engines

12.13.1 Air temperature

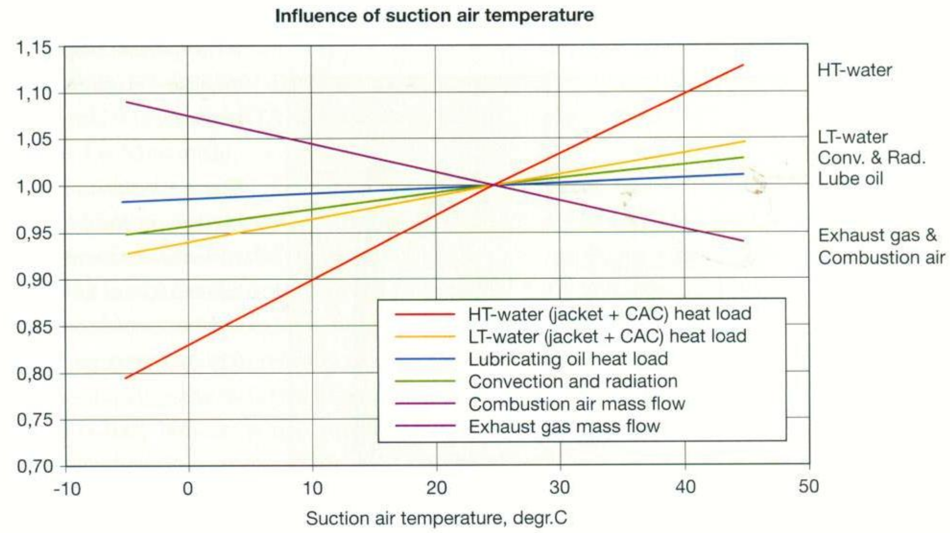
This is the temperature of the air after the inter cooler, at the instant it passes the inlet valves (four-stroke engines) or the inlet ports (two-stroke engines).
The air temperature is important for the total process.

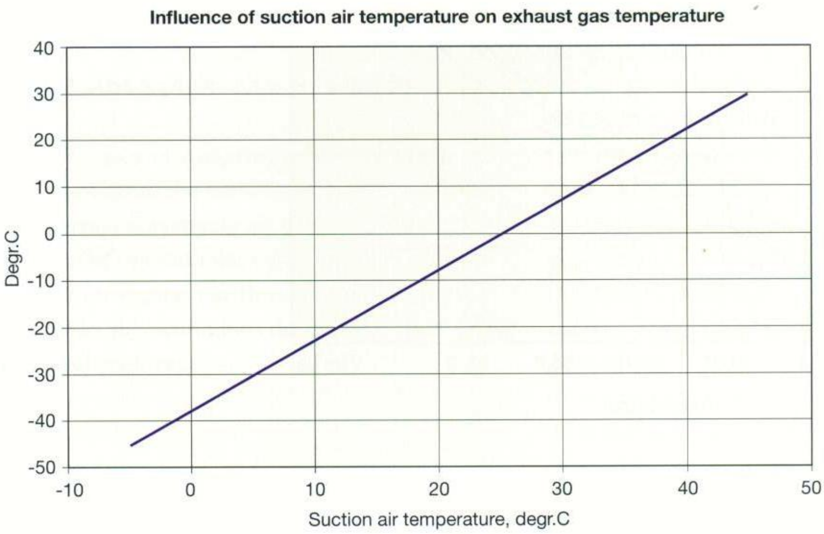
- The mass air per time unit that can be brought into the engine depends heavily on the air temperature.
- So in essence the capacity of the engine is reduced at higher air temperatures.

▶

The influence on the air temperature on the air mass that per time unit is drawn into the engine.

Upper graph:
horizontal: the air temperature
vertical: the correction factor
Clearly shown is the influence of the air temperature on the air mass per time unit drawn in the engine. The difference between – 10 °C, correction factor 1,10 and 40 °C, correction factor 0,95 is large. The HT coolant correction factor changes a lot from of 0,75 at – 10 °C air temperature to 1.10 at 40 °C.
Lower graph:
horizontal: the air temperature
vertical: the influence of the air temperature on the exhaust gas temperature
Between a suction air temperature of 0 °C and 30 °C is an increase of the exhaust gas temperature of approximately 44 °C.





▶

The cylinder of a large two-stroke crosshead engine.

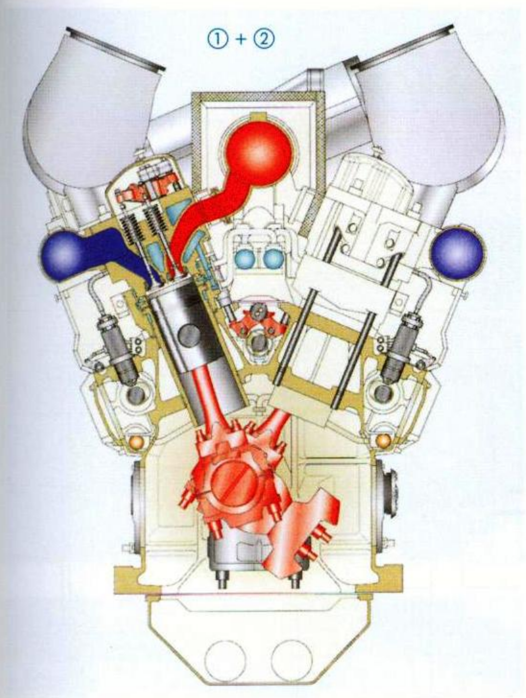
1 *central exhaust- gas pipe*
2 *exhaust gas pipe one cylinder*
3 *exhaust valve*
4 *cilinder cover*
5 *cilinder liner*

12.13.2 Effects of the air temperature on the temperature of the exhaust gases, coolant, lubricating oil and the radiant heat

The air **temperature** has a direct effect on the engine output. At higher air temperatures the power output decreases because the specific air mass is lower at higher temperatures and therefore the total air mass per time unit which is available to the engine. Therefore less fuel can be supplied to the engine, thus reducing the available shaft power.

12.13.3 Air pressure

Air pressure affects the engine's capacity, the higher the engine is above sea level, the further the air pressure decreases and the engine output.



▲
The atmospheric conditions that influence diesel-engine performance.

1 *air temperature, T_{air}*
2 *air pressure, P_{air}*

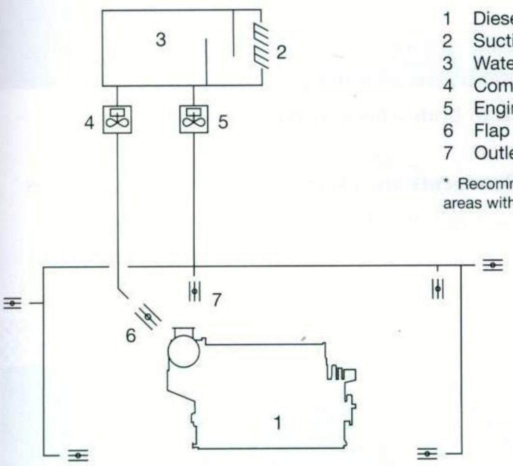
At a higher atmospheric pressure the specific air mass increases and more power can be generated. **At a lower atmospheric pressure the** specific mass decreases and less power can be generated.

12.13.4 Air moisture content

The moisture content of air is also an important factor. At too low a scavenging air temperature the water vapour in air can condensate. The water droplets will then precipitate on the lubricating oil film on the cylinder liner, causing extra wear and tear of cylinder.
For this reason, two-stroke-crosshead engines are equipped with a water mist catcher in the scavenging air space which can collect the condensed water vapour and discharge it, so avoiding additional wear and tear of the cylinder liner.

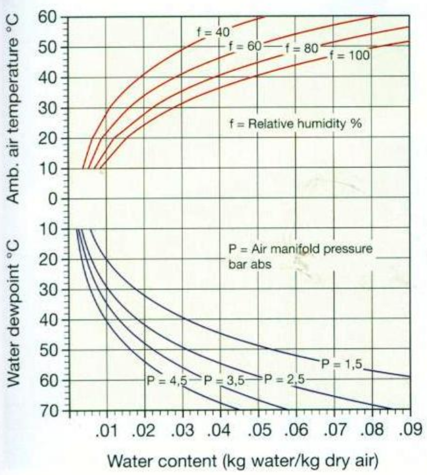
Particularly in moist tropical areas, large amounts of water are drained in this manner!

Engine room ventilation (4V69E8169)



1 Dieselengine
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.)



Condensation in charge air coolers
Example, according to the diagram:
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.

◄
The upper schematic diagram shows a standard ventilation system in an engine room.

Sufficient air supply is very important for not only the combustion process in the engine, but also discharging the radiation heat around the engine and refreshing of air in the space for personnel.

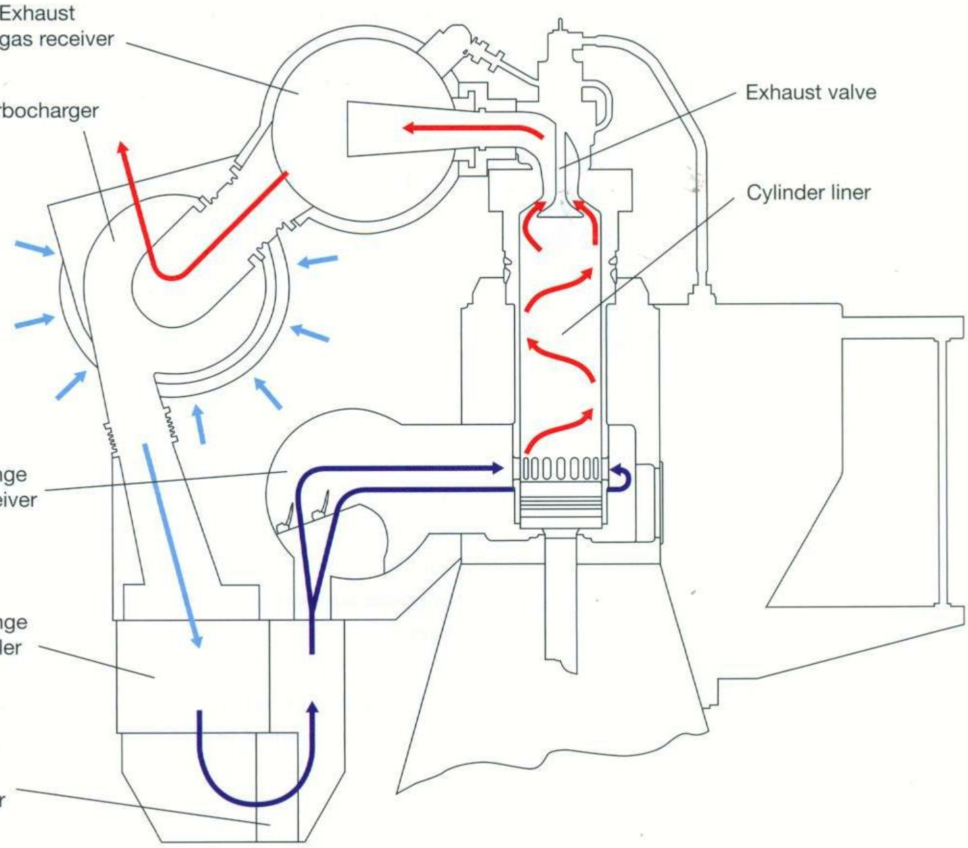
The lower graph shows the amount of condensed air in an air cooler after the turbo-blower.

Example: air conditions for turbo-blower: 35 °C and a relative humidity of 80%, the water content in the air is 0,029 kg per kilogram dry air. In the air-inlet manifold the pressure is 2,5 bar and the dew point is 55 °C. If the air temperature in the inlet manifold after the air cooler is 45 °C, the air can only contain 0.018 kg water per kilogram dry air. The difference, 0.029 – 0.018 = 0.011 kg per kilogram dry air, will appear as condensed water.

In this way in small engines, several litres of water per hour and with large engines, thousands of litres per hour are produced.

The water drops have a negative effect on lubricating-oil film on the cylinder liner; so in two-stroke crosshead engines, a water mist catcher is installed after the air cooler that discharges the water outside the engine.

▶
The water-mist catcher at the bottom of the scavenging air supply to the cylinder.



12.14 Maintenance of turbo-blowers

Proper maintenance of the turbo blowers, including air filters and air coolers is essential for optimum operation of the diesel engine.
If the air filter is contaminated the resistance on the suction side increases and the air capacity of the blower diminishes, so less air is supplied to the engine.

Contamination of the exhaust gas turbine of the air compressor also causes a reduction in produced air. Therefore, the turbo blowers are provided with a wash- or scavenging system to clean both wheels at regular intervals.

Detergents are: clean water and solid substances such as burned walnut shells or other organic materials that burns virtually ash free in the exhaust gases turbine.

▶
The water-mist catcher is placed at the lowest point in the air-piping system so that the water droplets cannot reach the cylinder.

◄

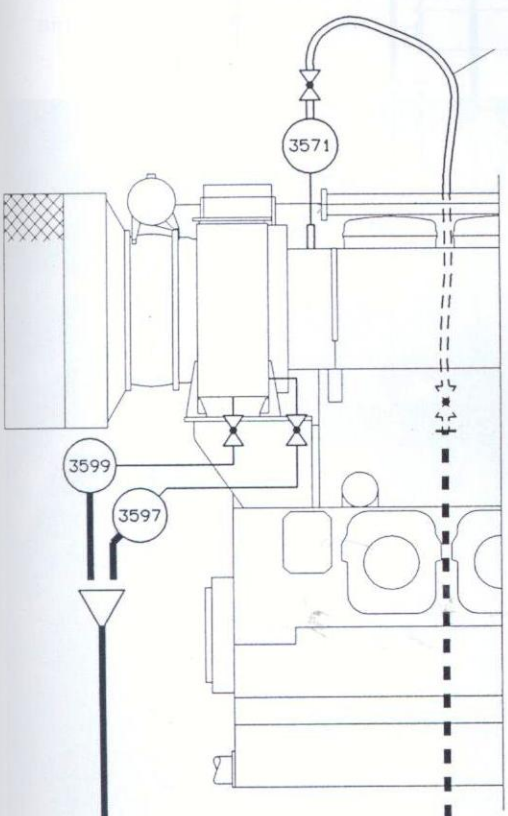
Dirty filters impede the passage of fresh air to the turbo-blowers.

Shown here, the air filter is badly soiled. A pressure difference meter will show that this filter should have been cleaned a while back!

◄

Damaged turbine blades cause a reduction in the air capacity of the turbo-blowers.

Regular cleaning and periodic maintenance can prevent this. Repair is essential.



◄

A water-wash system with the water supply above and both waste-water discharge pipes below.

▼

A water-rinsing system for the exhaust-gas turbine of the turbo-blower.

The dirty water is discharged at the bottom.

1 *rinsing pipes*

12.14.1 Air coolers

Contamination of air coolers reduce the heat transmission and increase the pressure drop over the air cooler.
Many engines are designed in such a way that the air side including the cooling fins can be cleaned without having to dismantle the cooler. By applying an isolating plate, the cooler side can be filled with a solvent and cleaned using a circulation pump. With the use of filter mats all the dirt particles are then removed from the solvent.

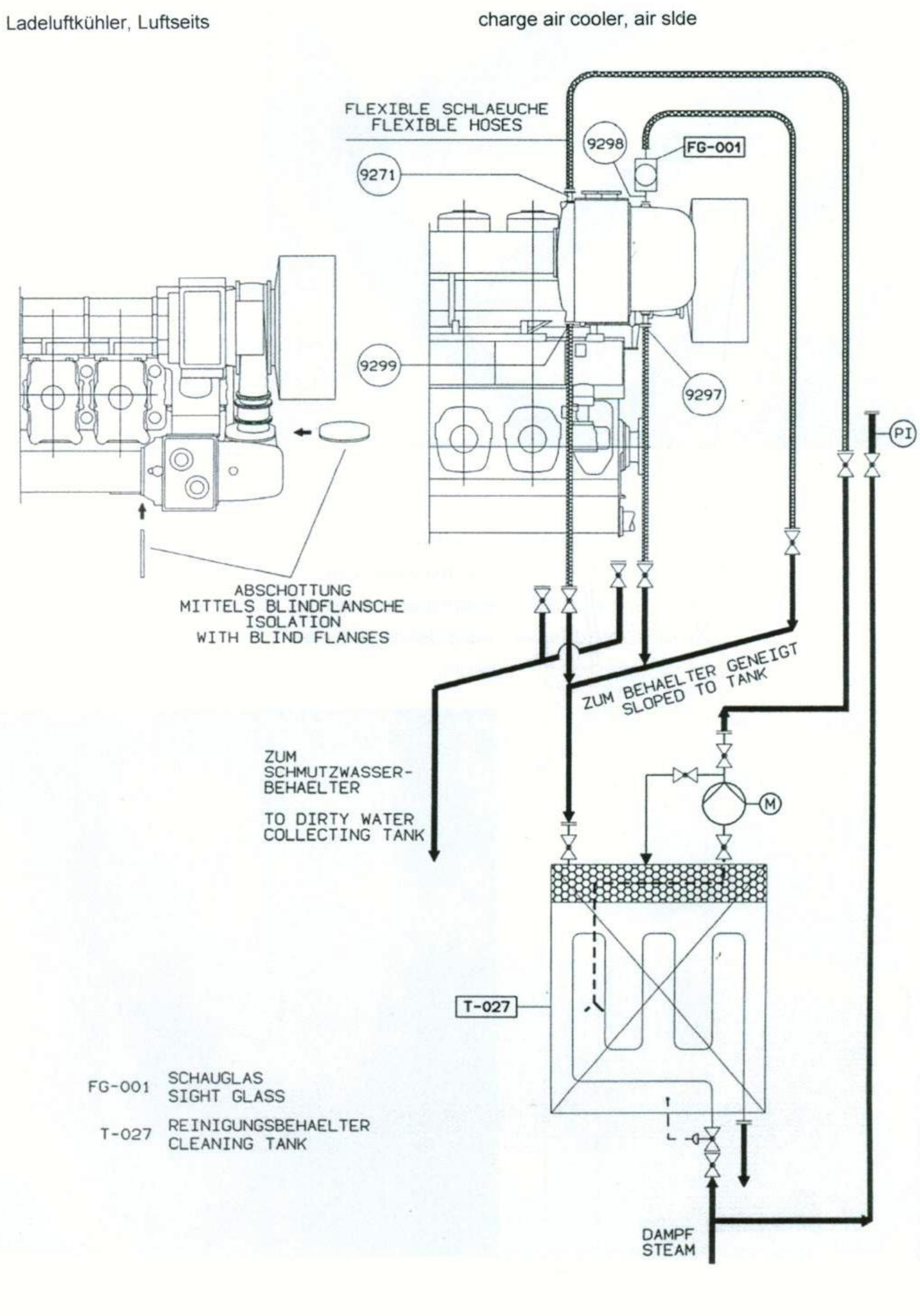
12.14.2 Arctic conditions

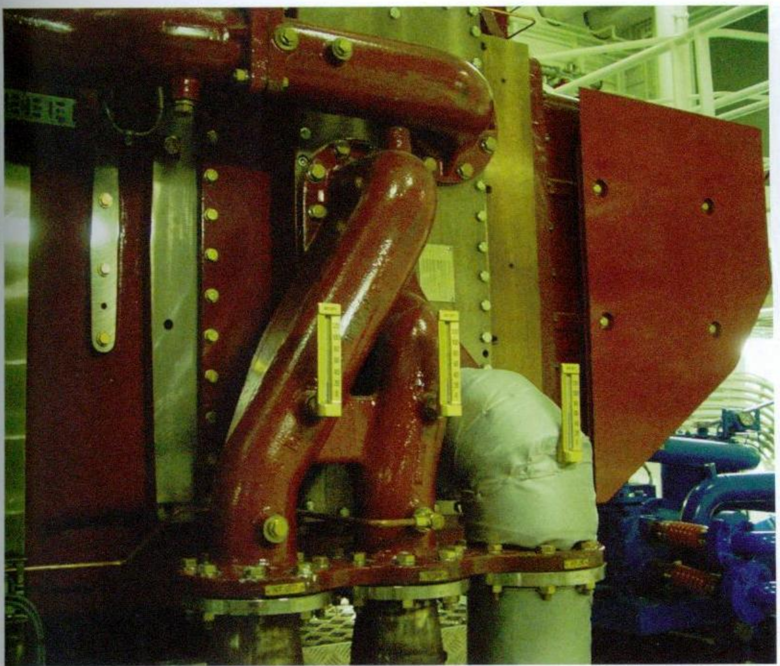
At extremely low air temperatures it is absolutely imperative to exceed the minimum scavenging air temperature.

- In order to start a diesel engine, a minimum air temperature of 5 °C is required in the engine.
- At low load, a minimum air temperature of – 5 °C.
- At full load, a minimum air temperature of – 20 °C.
- At air temperatures lower than the aforementioned, special heating to pre-heat the air is required.

▶
A rinsing system is available for air coolers.

When the engine is stopped, the cooler can be cleaned without removing the air cooler. Blind flanges are installed in the air cooler ducts and the air side is cleaned using a hot fat soluble solution and a circulating pump.

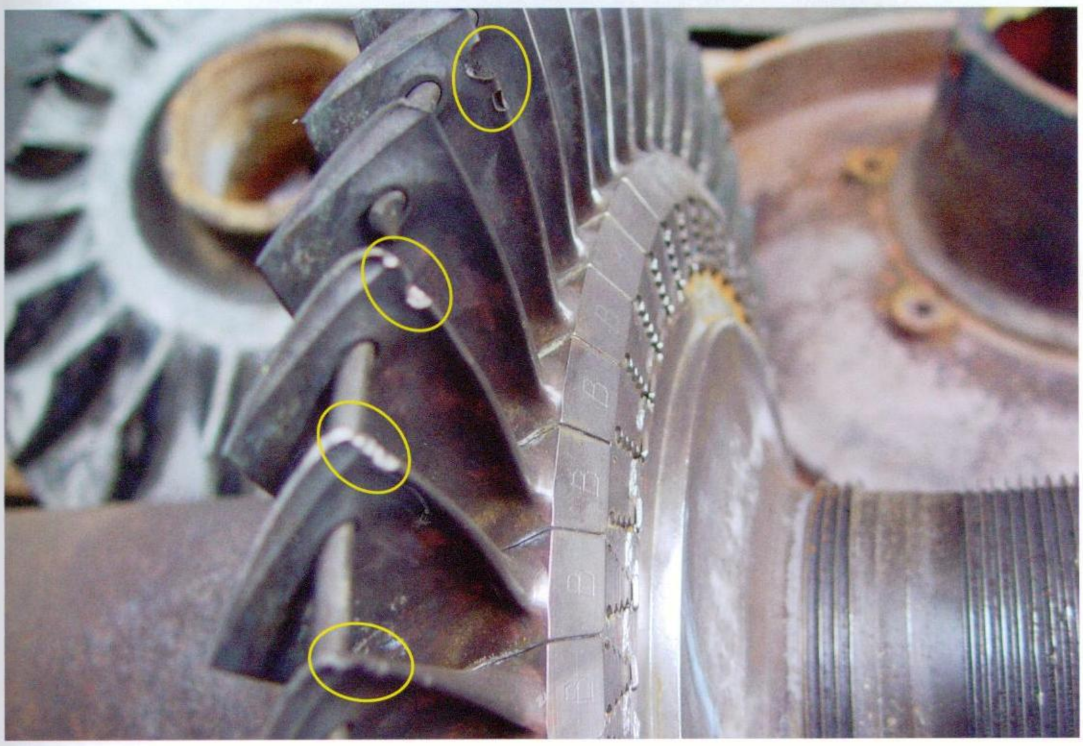




◄ It is important to monitor the air temperature after the air cooler as this air temperature affects the combustion process of the engine.

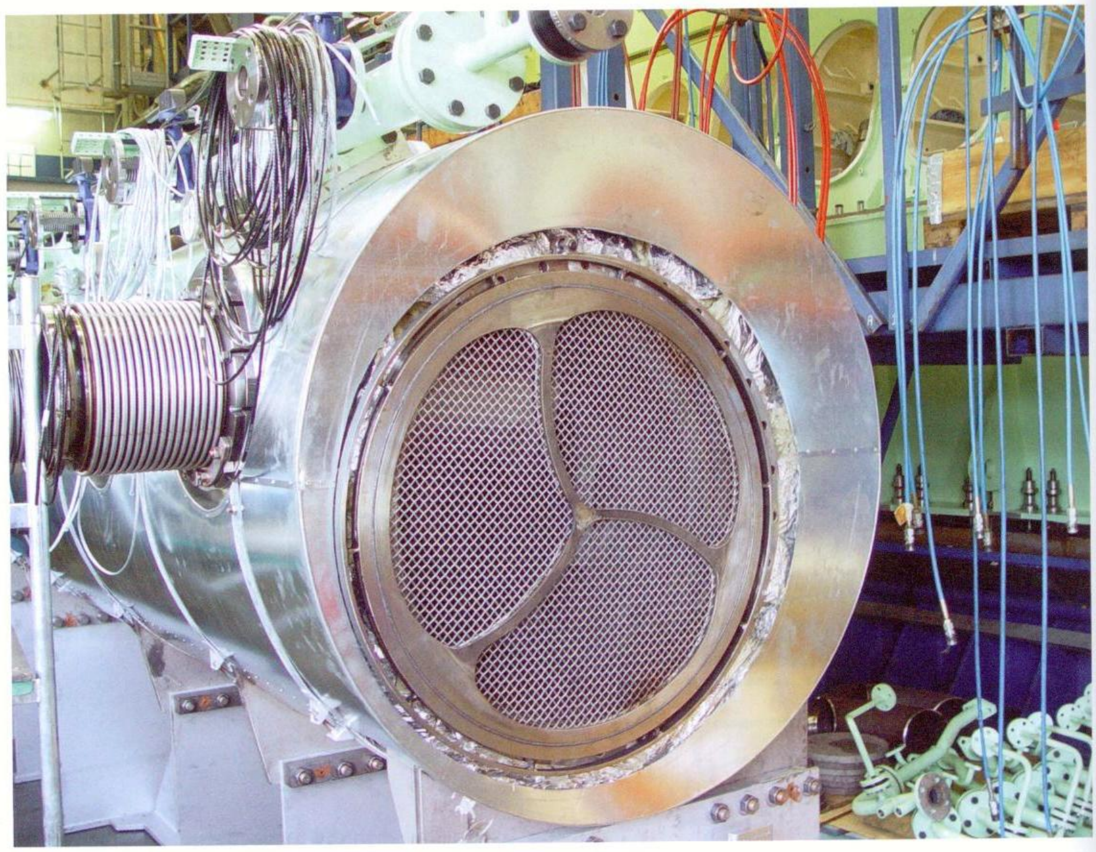
12.15 Problems with supercharging

The following problems may occur:
- contamination of the air filter;
- contamination, damage and corrosion of the gas turbines;
- contamination of the air compressor;
- leaking of the shaft sealants;
- wear and tear of the shaft bearings;
- contamination and corrosion of coolant spaces;
- turbine rotor imbalance;
- problems with the turbocharger oil lubrication system or the supply and discharge of the lubricating oil from the diesel engine;
- 'surge' of the air compressor;
- contaminated air cooler on the gas- or water side;
- too high or too low an air suction temperature, consider the ambient temperature;
- blockage of the condensate water drain of the mist-/condensate water separator.



◄ Mechanical damage of the turbine blades of a four-stroke engine.

Probably caused by a fragment of a piston ring or an exhaust valve in the turbine.



A protective screen is placed in the central exhaust-gas pipe of a two-stroke crosshead diesel engine.

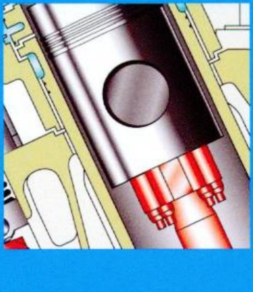
This prevents the exhaust-gas turbine from being damaged by pieces of piston rings or parts of the exhaust valves.

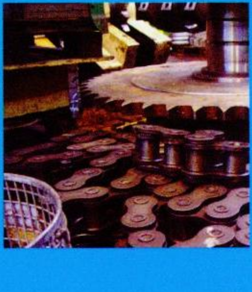
A badly contaminated exhaust-gas turbine.

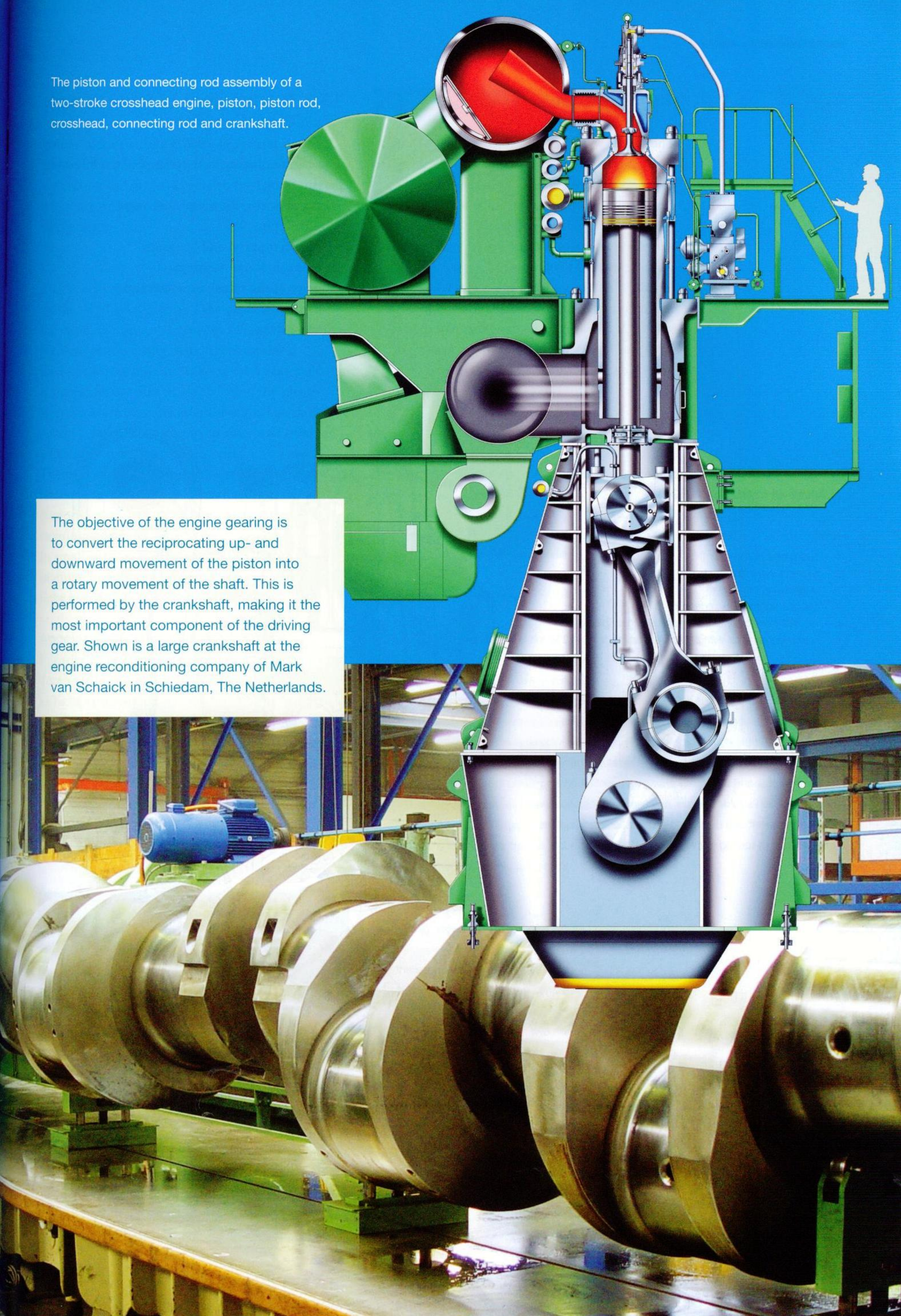
Due to this, the exhaust-gas flow through the turbine wheel is reduced making it less effective and therefore the amount and pressure of the air diminishes. The result is that there is less air available for the engine.

>CH 13

Driving gears



The piston and connecting rod assembly of a two-stroke crosshead engine, piston, piston rod, crosshead, connecting rod and crankshaft.

The objective of the engine gearing is to convert the reciprocating up- and downward movement of the piston into a rotary movement of the shaft. This is performed by the crankshaft, making it the most important component of the driving gear. Shown is a large crankshaft at the engine reconditioning company of Mark van Schaick in Schiedam, The Netherlands.

13.1 Introduction

The engine gearing is subjected to huge forces. The pressures generated in the combustion space are transferred by the piston to the piston pin, the connecting rod and the crankshaft. In today's heavily charged diesel engines, the pressure exerted on the piston during the power stroke varies from 500 kilograms for a category I engine to 1100 tons for a category IV engine. Furthermore, these forces vary continuously and swiftly. Therefore engine designers pay special attention to the construction of the gearing in order to control these tremendous forces.

13.2 Driving gear of four-stroke diesel engines

The engine running gear consists of the following parts:

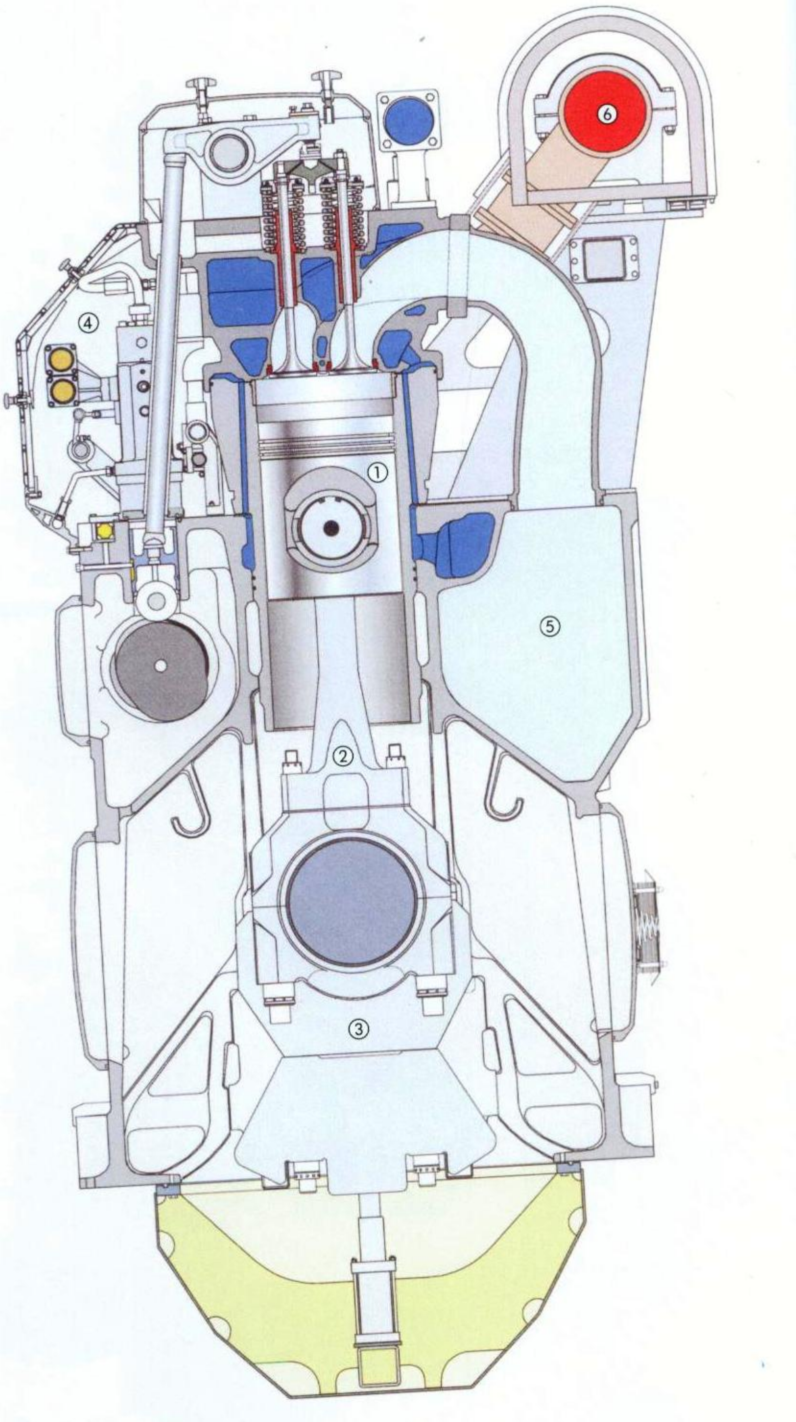
- piston including the piston rings;
- piston pin;
- connecting rod and bearings;
- crankshaft and bearings;
- drive of the camshaft and pumps;
- camshaft;
- valve drive.

▶
A modern four-stroke medium-speed diesel engine (Wärtsilä 46) suitable for H.F.O.

Note:

1 *two piece piston*
2 *big end, type 'marine head'*
3 *underslung crankshaft*
4 *hot box for the fuel section*
5 *air-inlet manifold integrated in the engine block*
6 *uncooled exhaust with protective casing*

The piston skirt with its large surface area is suitable for the absorption of the lateral forces and transmission to the cylinder liner and the engine block.
During both the compression and power strokes, the forces are considerable and constantly changing direction. During the compression stroke they are, as seen in a cross-section, directed to the left and during the combustion stroke to the right with a right-turning crankshaft.



13.2.1 Pistons

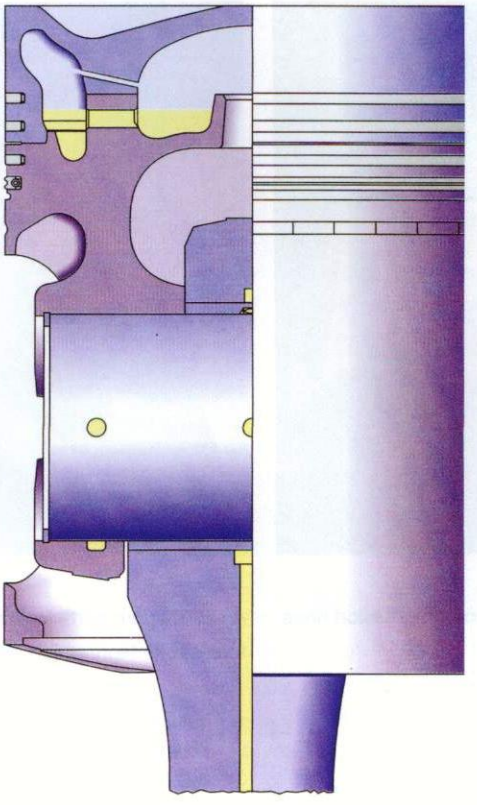
Pistons in four-stroke diesel engines not only transmit the gas forces to the crankshaft via the connecting rod, but also absorb the lateral forces which are produced by the crank-connecting rod mechanism.

Construction of the pistons

This is different for each engine category.
As the RPM decreases and the cylinder bore increases, the piston comprises two parts, namely the steel piston crown which is resistant to the huge forces that are exerted with material temperatures at 300 to 400 °C, and the piston skirt for absorbing the lateral forces. The steel crown is provided with piston rings and often an oil scraper and a distribution ring.
The light metal or cast iron skirt absorbs the lateral forces. The skirts often have special finishes comprising a special coating which enhances the running properties.
In small engines the pistons are usually manufactured from a single piece of high-grade light metal, mainly aluminium.

Accelerating- and decelerating forces

In order to restrict the accelerating- and decelerating forces, the pistons as generally kept as light as possible.



▲

A modern two-piece piston.

- The steel piston crown, transfer of the combustion forces via the gudgeon pin to the connecting rod. Heavy-loaded top rings are found in the piston crown.
  The material, cast steel or forged steel, has heat resistance properties to maintain strength with the high piston-crown temperatures which rise to 350 to 400 °C.
- The huge forces on the piston crown necessitate a stiff gudgeon pin to prevent bending of the pin and subsequent problems with the gudgeon-pin bush. The diameter of the pin is large in comparison to its length. This means that the perforation in the piston is also large! In fact this weakens the piston skirt!

◄

A cut-away view of a piston of a Scania diesel engine.

Note the special shape of the piston crown, designed for optimal combustion. The piston consists of two materials. Clearly visible is the colour difference in the piston crown. The piston only has three rings, two compression rings and one scraper ring for oil distribution. The piston is cooled with lubricating oil from the gudgeon pin oil via drillings.

▲
A piston for a small engine with two compression rings and one oil-scraper ring.

A large section of the sides where the gudgeon pin is normally positioned is missing. Only the section where the piston rings are placed is round and seals the combustion space. The skirt on the left and right in the picture absorbs the lateral forces.

▲
A set of spare parts for a modern highly charged four-stroke high-speed diesel engine (Cummins).

1 *camshaft section*
2 *connecting rod*
3 *cog wheel*
4 *valves*
5 *gudgeon-pin bushes*
6 *cylinder liner*
7 *piston, two pieces*

▼
An aluminium piston for an engine category I or II.

◀
New pistons with a steel crown and a cast iron skirt of Caterpillar–Mak 25 engines.

Note the sunken piston crown for optimal combustion space

◄
A 'cut-away' section of a two-piece piston in a four-stroke medium-speed diesel engine. In the centre; the lubricating-oil spray nozzle directed at the piston crown for cooling.

The piston rings are located in the cast steel piston crown. Note the shape of the piston crown; the atomised fuel is injected to the sides.

Piston rings

The piston rings serve to seal the combustion space and to conduct the piston in the cylinder liner. The number of rings varies from two to five, depending on the cylinder bore, the RPM and the forces exerted on the piston crown.

◄
A large piston for a four-stroke medium-speed MAN–B&W diesel engine.

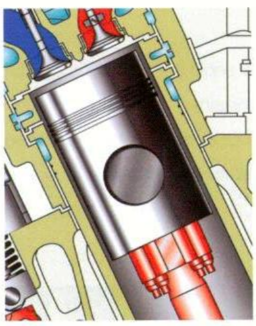
Note the amount of cast iron removed around and underneath the gudgeon pin, to reduce the weight of the piston.
At the bottom of the skirt the studs of the top section of the connecting rod and the eye, still attached to the gudgeon pin, are barely visible.

1 *piston crown*
2 *ring pack*
3 *piston skirt*
4 *gudgeon pin including seal*
5 *removed material*
6 *studs for the connecting-rod eye*
7 *cylinder-head studs*
8 *anti-polishing ring*

◄

A special aluminium piston for a Cummins–NTA 855 four-stroke high-speed diesel engine.

Due to the temperature expansion coefficient of the light-metal piston, the diameter of the piston-crown section is slightly decreased (extra clearance with the cylinder liner). The notches in the piston crown serve to create space for the valves which are not entirely closed in the piston T.D.C..



◄

A modern piston for a four-stroke medium-speed diesel engine of MAN–B&W, type 48/60.

In view of the high compression and combustion pressures the liner, with bore cooling, is very thick and placed partially above the block. In this manner the engine block is smaller and lighter! The studs for the connecting-rod eye are visible at the bottom of the cylinder liner. The exhaust valves have flaps for valve rotation.

◄

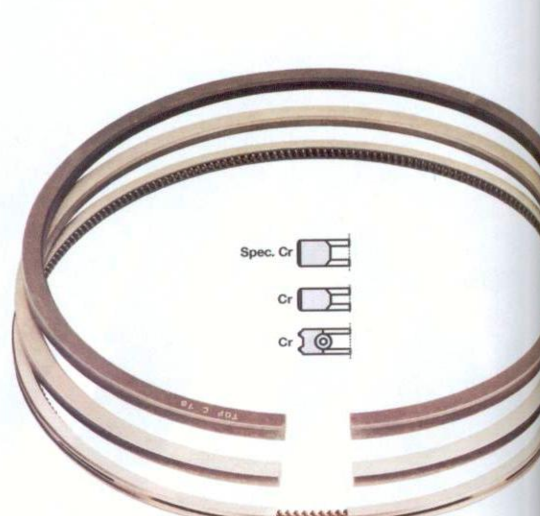
Attention is given to the piston-ring grooves.

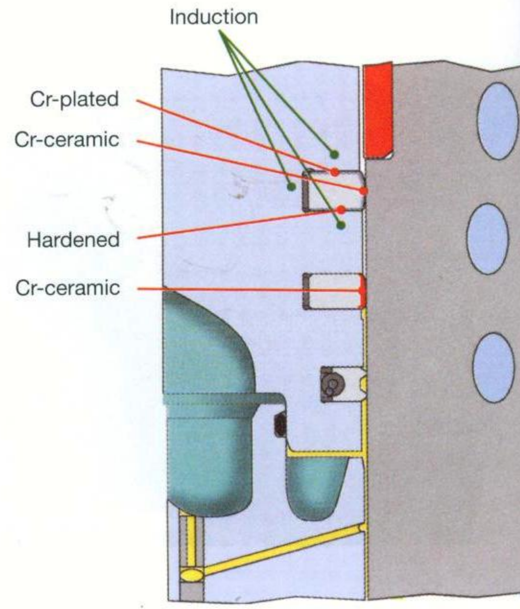
A steel piston crown has a better resistance to the 'hitting' of the hard cast iron rings than a light-metal one-piece piston. The materials used for the grooves are either hardened or chromium-plated, to increase wear resistance.

►

Piston rings are available in many variations.

Generally, each engine type and make has its own specific ring package. For high wear and corrosion and other piston-ring problems, modifications are made to improve ring-package properties.





▲

Piston rings, in combination with an adequate lubricating-oil film between the cylinder liner and the piston/piston rings, provide sealing for the increasingly high final compression and combustion pressures.

Here the following aspects are important:

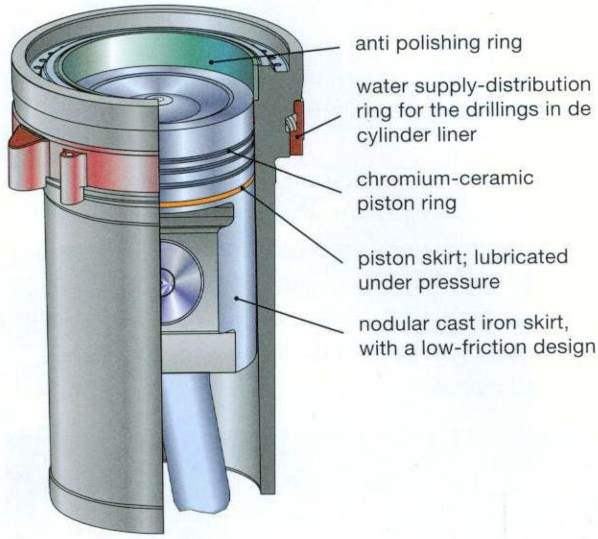
- heat absorption capacity for the lubricating oil, also expressed in µm (microns);
- adequate clearance between the piston and cylinder liner;
- correct spring and its specific properties such as tension on the liner, shape and size of the running surface area;
- the ring groove dimensions.

In the drawing above the top ring has a thin chromium coating and the ring is hardened. The running surface is chromium plated and finished with a wear-resistant ceramic coating. Induction hardening has been applied to the piston-ring groove. The cylinder liner is provided with an anti-polishing ring and has bore cooling. The piston lubrication is pushed towards the oil-scraper rings under pressure.

Piston pin

The piston pen serves to transfer the forces exerted on the piston crown to the connecting rod. The connecting rod 'swings' around the piston pin and impedes hydrodynamic lubrication. The bearing surface and therefore the piston pin and the connecting rod bearing must be large enough to ensure that the surface pressure is low. The diameter of the piston pin increases as the pressures exerted on the piston increase; this also applies to the perforations in the piston skirt. The piston pin must be rigid and so avoid problems with the piston skirt and piston pin bearing.

In order to sufficiently lubricate the connecting rod bearings, lubricating oil is supplied under pressure via perforations in the connecting rod. The piston pins are manufactured from a steel alloy and are often hardened or provided with a thin chromium coating.



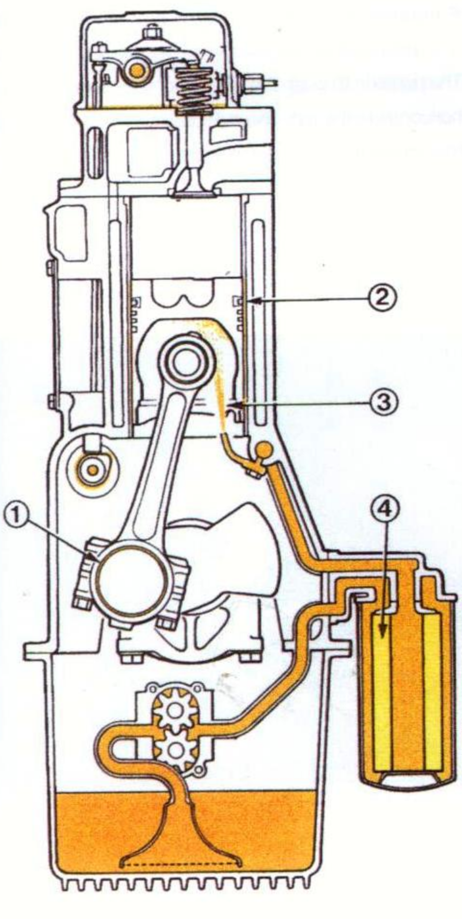
▲

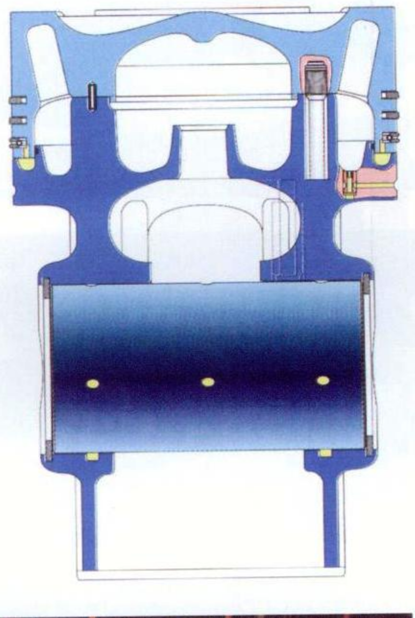
An 'optimum package' of components in the combustion space. Shown the Wärtsilä 32 four-stroke medium-speed diesel engine.

▼

Small engines often have lower loads (lower load numbers) and therefore have a less complicated construction.

Shown a DAF-diesel engine with spray cooling for the piston lubrication and cooling.





◄

The gudgeon pin placed in the cast iron skirt of the piston.

The diameter of the gudgeon pin is such that bending of the pin is prevented. Note the lubrication of the connecting-rod eye and gudgeon pin bush. The lubricating-oil supply to the piston rings is not visible; the cooling lubricating oil for the piston crown is visible.

◄

Installing the gudgeon pin in the world's largest four-stroke diesel engine, a Wärtsilä 64 medium-speed diesel engine with a cylinder output of over 2000 kW.

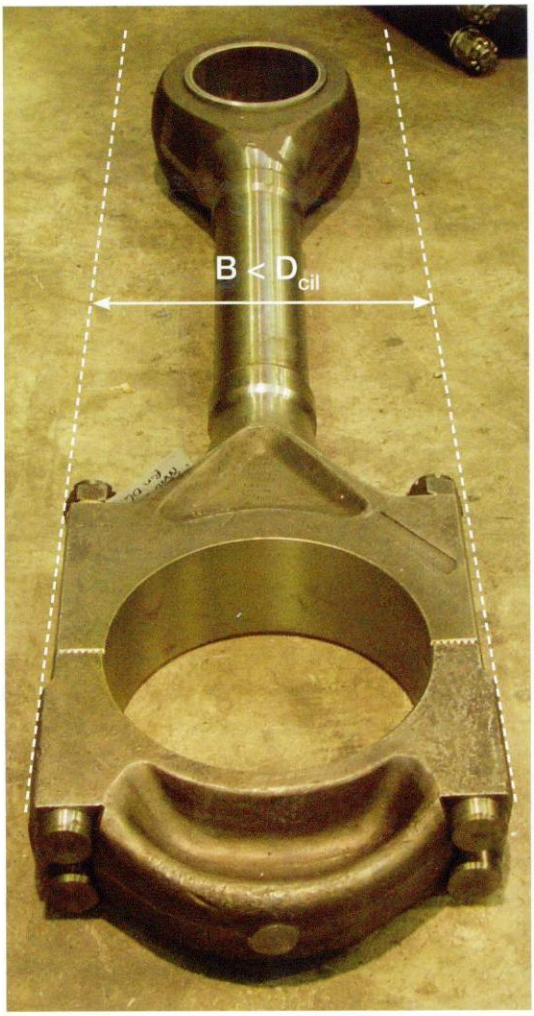
The pin is too heavy to fit manually.

13.2.2 Connecting rods

▼
Roughly forged steel connecting rods for Caterpillar–Mak engines.

During machining the connecting-rod eye is severed from the base of the connecting rod along the white dotted lines.

In modern, heavily loaded diesel engines, connecting rods are manufactured from forged steel.
Only in the smallest engines in category I connecting rods are manufactured from special cast iron. Connecting rods are often manufactured by single drop forging in a mould from a cherry-red piece of alloyed steel. It is later sawn through at the big end or at the flange connection of the connecting rod.



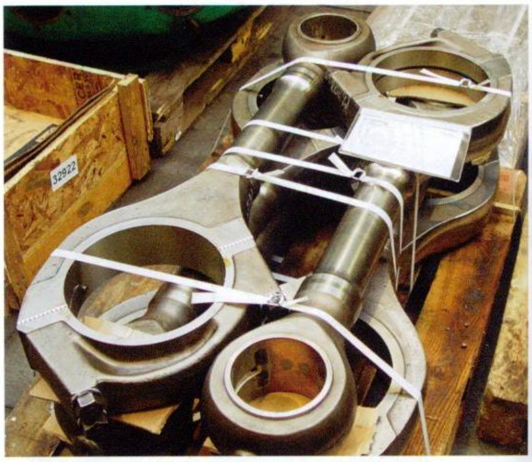
▲
A traditional connecting rod of an older engine type.

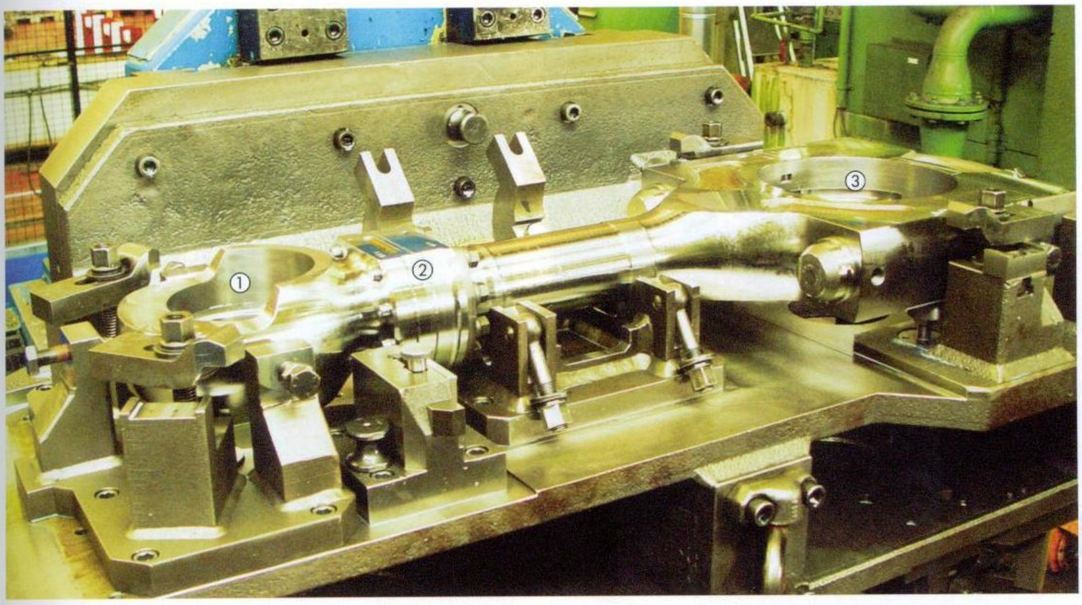
The partition through the big end of the connecting rod is horizontal to the rod. The cylinder-liner diameter is larger than the connecting rod big-end width; this means that the load parameter is low.

▲
Roughly forged steel connecting rods for Caterpillar–Mak engines.

During the machining process, the connecting-rod big end is obliquely severed.
The grain flow in the material during forging produces strong components. This occurs to a lesser extent when the pieces are forged separately.
- - - - - = saw cut

►
Overhauled connecting rods with an angled big end, a common construction.





◄
A connecting-rod big end with a partition below the connecting-rod eye for a Caterpillar–Mak diesel engine is clamped down for machining.

1 *position gudgeon pin*
2 *partition*
3 *position crankpin*

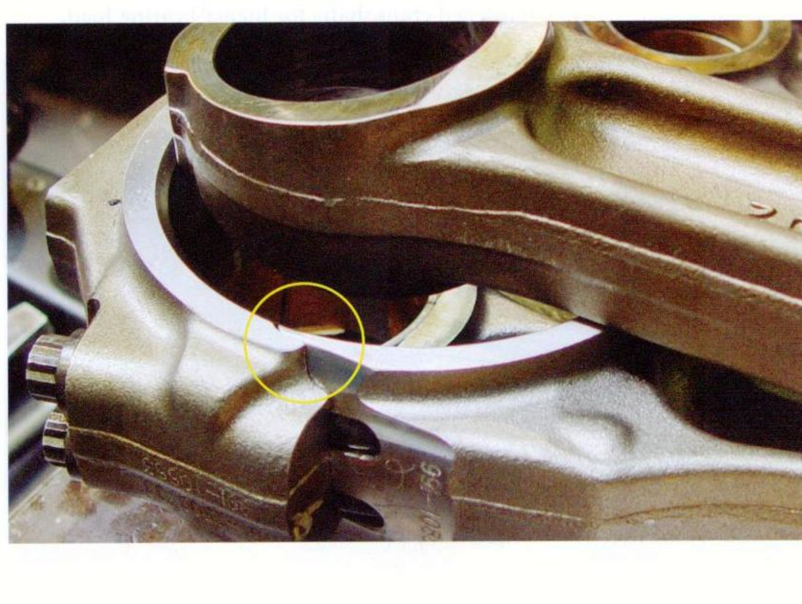
Since the diameter of the crank pin is increased to minimise the surface pressure increase on the crank pin bearings, which is the result of the increased cylinder capacity at an identical cylinder bore, the crank pin bearing dimensions and consequently those of the big end will also increase.
A horizontal big end split is no longer sufficient, as the width of the big end is too large for drawing the piston. An oblique division can also be problematic. The big end was therefore split into three sections (SWD 410) and today the marine version has a separate, small big end mounted on a separate upper- or lower cap or a division of the connecting rod close to the piston. In this construction, dismantling remains an option.

◄
Connecting rod for a Caterpillar-diesel engine. See detail.

▼
Detail of a connecting rod for a Caterpillar diesel engine.

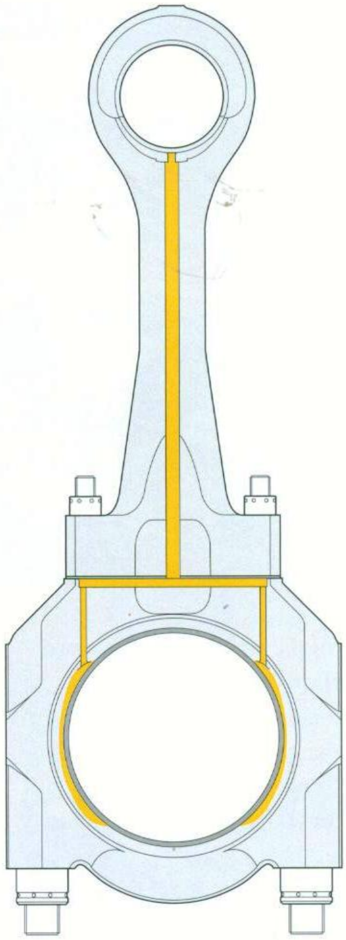
Big end and cover are fitted together at an angle (circle).

▼
A robust angled partition of the connecting-rod big end. An outdated model.



▶
'Connecting-rod eyes' for installing the gudgeon pin.

Note the large lubricating-oil grooves and the size of the bottom of the gudgeon-pin bush; these are subjected to combustion forces of approximately 250,000 kg or 250 tons!



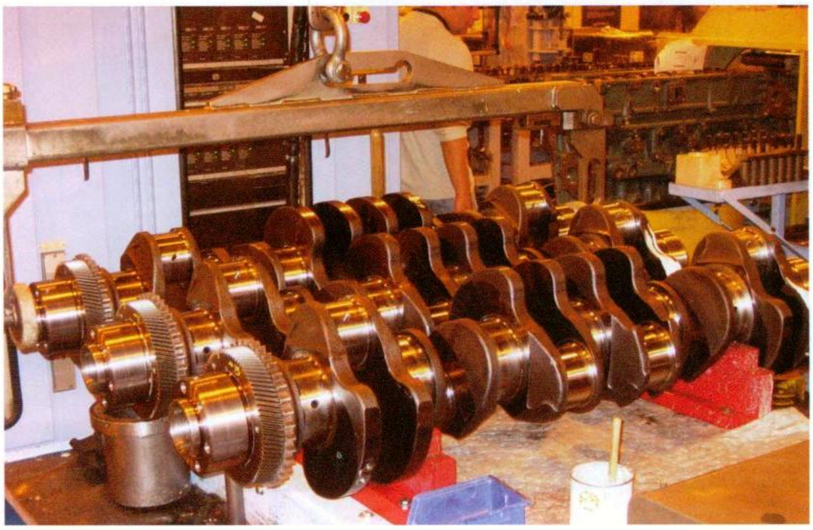
▶
The lubricating-oil channels in a modern connecting rod.

This connecting-rod version is known as a 'marine head'. The base of the connecting rod has been narrowed to such an extent that the piston including the connecting rod can be hoisted out of the engine.
The lubricating oil is supplied through drillings in the crankshaft and flows to the gudgeon pin via the crankpin, the grooves in the bearing shell, both drillings in the upper crankpin cover and the drilling in de connecting rod. Subsequently, the lubricating oil cools the piston and lubricates the piston rings.

13.2.3 Crankshaft

The crankshaft transforms the reciprocating, up- and downward, movement of the piston to a rotary motion. Small category I crankshafts are manufactured from a special cast iron. However, all larger crankshafts are manufactured from one piece forged iron, and are depending on the load, alloyed with chromium and nickel. The engines in the high-speed category usually have surface hardened crankshafts for higher bearing load.

The crank pins are often hollow and therefore restrict the centrifugal forces of high speed crankshafts. The webs are relatively thin and so ensure that the centre to centre distances between the cylinders are as small as possible. In this way the engine is shorter and the deflections between the main bearings are limited.



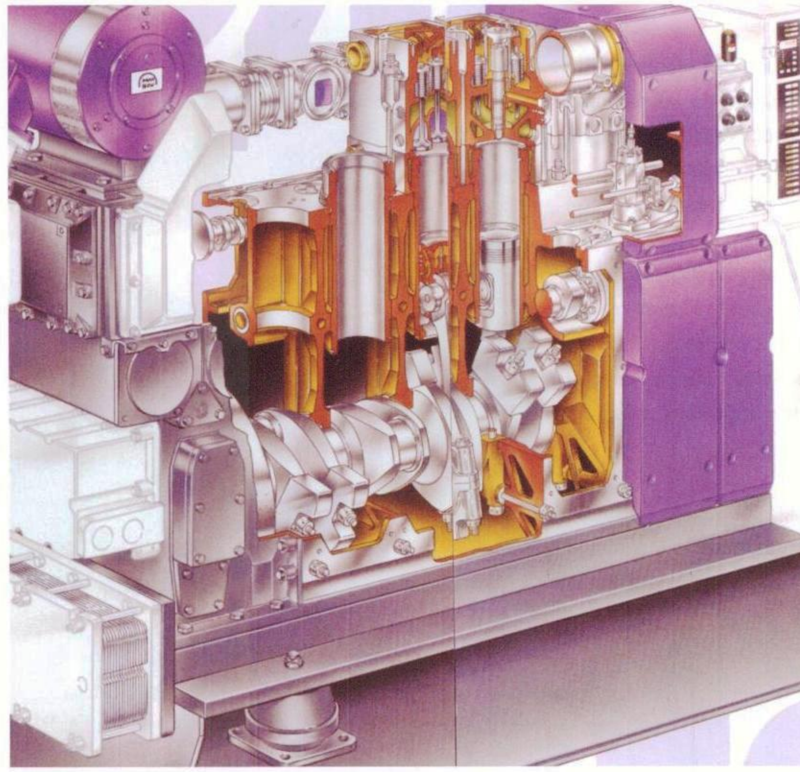
◀
Three crankshafts for Detroit-diesel engines.

Note the narrow webs and large crankshaft diameter and the crankpin.
The drive gear, amongst others, for the camshaft is installed.

▲

A crankshaft for a four-stroke high-speed engine.

The two lubricating-oil supply holes on each crankpin are indicative of a V-engine. The counterweights, which serve to balance the crankshaft, are an extension of the crank webs and not, as in larger engines, bolted to the crank webs.



▲

The crankshaft is the most elementary component of the drive gearing in a diesel engine.

Crankshaft damage often entails high costs. Replacing a crankshaft costs:

category I : ± € 1,000 – € 4,000

category II : ± € 50,000 – € 100,000

category III : ± € 200,000 – € 400,000

category IV : the overall damage can amount to millions of euros.

The main problem with the total loss of a crankshaft is the removal of the crankshaft and installation of an overhauled or a new crankshaft.

▼

A roughly forged crankshaft.

The steel is heated to a light red/yellow colour, approximately 900 degrees and is then shaped using a hydraulic press and mould.

Forging these crankshafts takes place using a crankshaft mould in which the hot steel is hydraulically pressed into shape with large forces. This is followed by a machining process.

▶

Forcing of a crankshaft in 'one throw'. Maschinenfabrik Alfing Kessler GmbH, Aalen, Germany.

Larger crankshafts are hydraulically pressed into the correct shape throw by throw; each time one crank throw and web is heated and formed.

▼

The final manual machining of the crankpin for a new crankshaft.

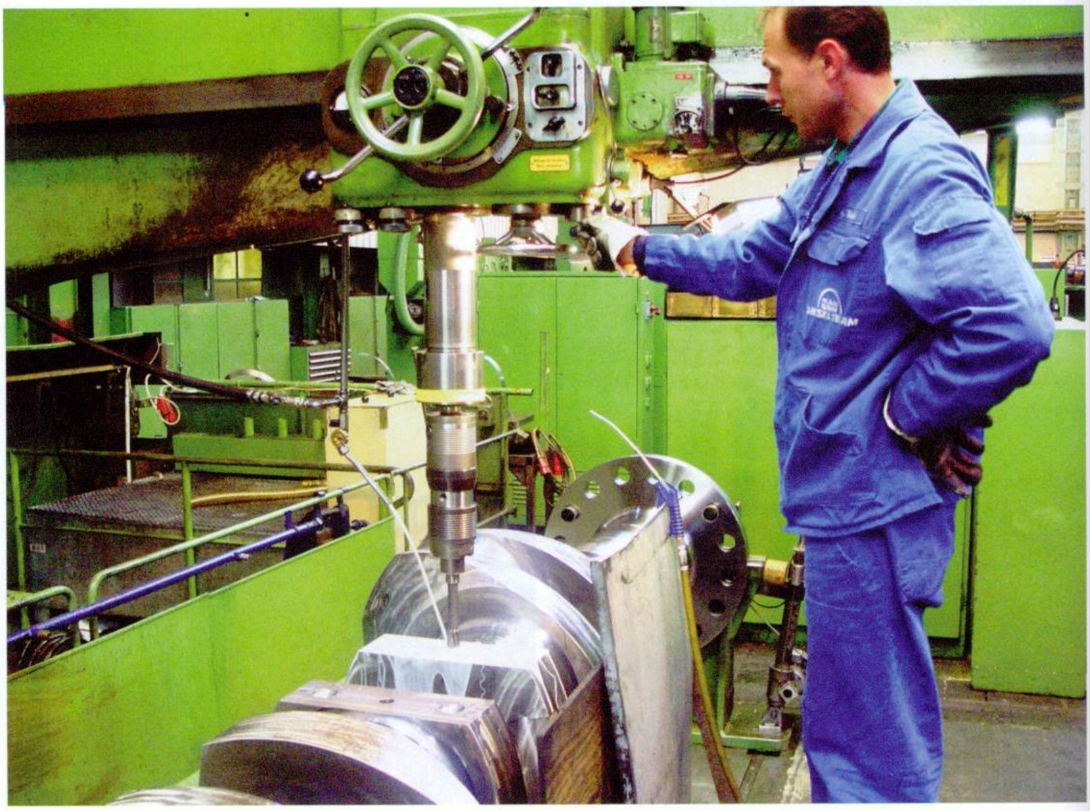
The pin surface is polished with a honing mould to remove the last surface irregularities.

The crankshafts of category III engines, the medium-speed diesel engines running on H.F.O. are also forged from steel; however, the journals are not hardened. Crankshafts with a cylinder bore of 400 to approximately 450 mm are forged in one throw over their entire length. Larger crankshafts in this category are generally forged, crank by crank from a single piece of metal using a throw by throw forging process; this entails partially heating the crankshaft in the area where the crank is to be forged.
This is followed by a machining process and then the mounting of counter weights.

▶

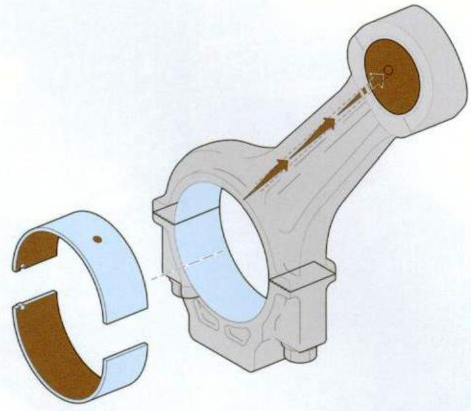
Drilling holes for fixing the studs for the counter weights.

Reconditioning workshop MAN–B&W in Hamburg, Germany. Shown, a new crankshaft.



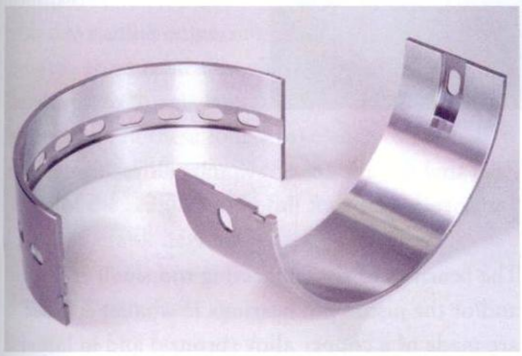
13.2.4 Bearings

The bearing caps of the main shaft and the crankpin are the so-called tri-metal bearings or the 'hard bearing caps'. The steel caps are often coated with a thin copper layer, followed by a very thin running surface layer, often an aluminium alloy.



A fitted bearing for the crankpin in a MTU-diesel engine.

Note the notches on the left of the bearing shells and the lubricating oil hole in the upper part of the bearing shell for lubricating oil flow to the connecting-rod eye and gudgeon pin.

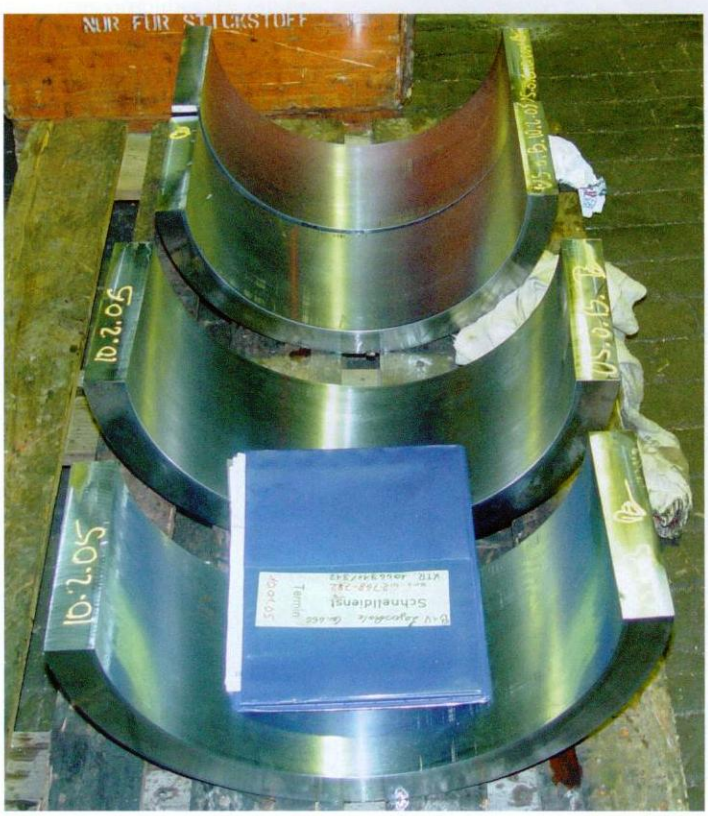


Both main bearing shells for a Wärtsilä 38 diesel engine.

left: the upper bearing with a lubricating-oil groove with holes for lubricating-oil supply to the shaft.
right: the bottom bearing with a large bearing surface for absorbing the large forces during the compression and combustion strokes.

Bearing shells for a Caterpillar diesel engine.

Bearing shells are replaced after disassembly. As maintenance intervals have become increasingly longer, new bearings are fitted, thus warranting an extended operating time without problems that could occur by re-using the bearings.



White-metal bearing shells for a two-stroke crosshead engine by MAN–B&W.

These bearings have been recast, provided with a new white-metal coating and subsequently machined to the correct specifications and dimensions. Shown are traditional 'thick' bearings for the main bearings which referred to as the "thick shell" bearings. Modern bearings are "thin shell" bearings and used in the crosshead. Due to the rising surface pressures, caused by the increasing gas forces on the piston, MAN–B&W also uses "thin shell" bearings for the main bearings. The material is a tin-aluminium alloy, Sn40Al.

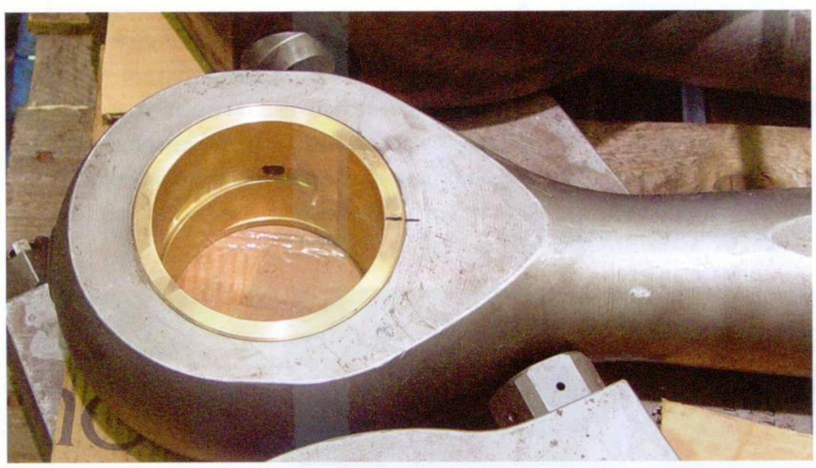
◄ The main journals and crankpins must be absolutely round and have a very smooth finish.

Naturally, the machining must occur when the crankshaft has been aligned. Alignment is done with both supports, the so-called rests. Shown: grinding a used crankshaft at Mark van Schaick, Schiedam, The Netherlands.

For the main bearings found in an engine, oversized thicker bearings are available thereby allowing the main journal to be polished and provided with an oversized bearing after being damaged. Depending on the degree of damage to the journal this can be repeated twice or three times.

Also see Chapter 26, Overhauling diesel engines and their parts.

The bearings of the connecting rod small end and/or the piston pin bearings in smaller engines are made of a copper alloy (bronze) and in larger engines, a hard tri-metal alloy.



◄ The bronze bush bearing of a connecting rod small-end.

Note the lubricating-oil groove and the drilling for lubrication of the gudgeon pin.

◄ A connecting rod for overhauling.

Clearly visible is that one side (bottom part) of the piston-rod bushing of the connecting rod small-end has a larger surface area than the other side (top part). The larger surface area is required to absorb the high combustion forces.

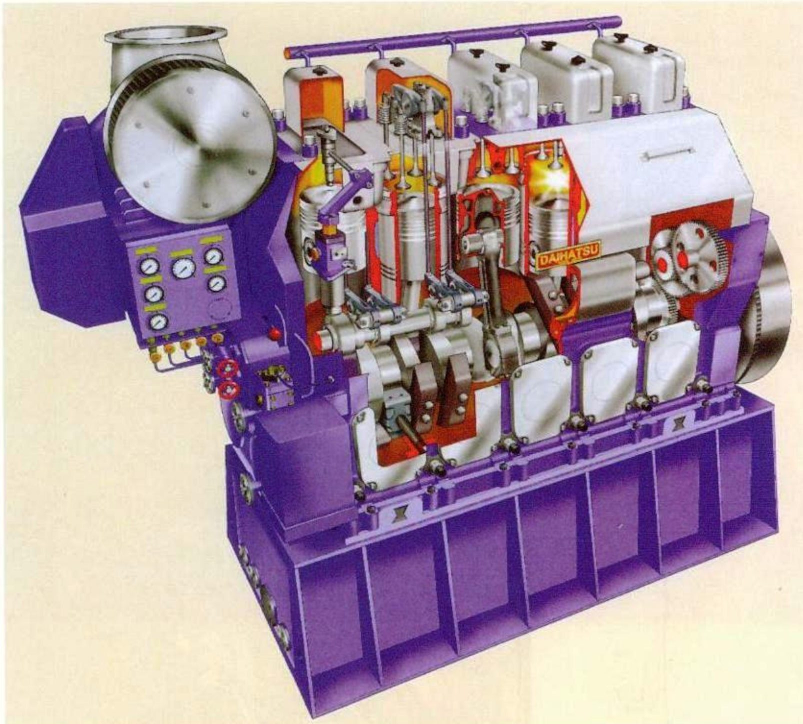
13.2.5 Camshaft drive

In small category I and occasionally category II engines, this occurs sporadically by means of a chain drive. This is light, inexpensive and takes up little space. Replacement takes place after a certain amount of operating hours, therefore avoiding propulsion problems such as cracking or chain elongation.

In most category II engines and certainly in category III engines, the camshaft is driven using tooth wheels, which as with chain drives, may also drive other engine components, such as the pumps. The steel tooth wheels have polished and hardened teeth and operate with as little clearance as possible. In these four-stroke engines the number of revolutions of the camshaft is half the RPM of the crankshaft; a full four-stroke process lasts 720 crank degrees.

13.2.6 Camshaft

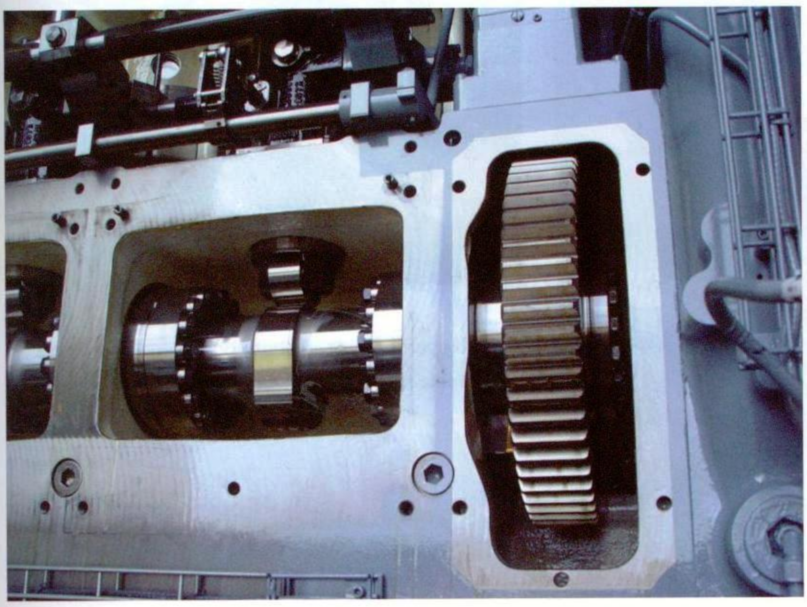
The camshaft carries the cams which control the fuel pumps and the inlet- and exhaust valves. In small engines, the steel camshaft is manufactured from a single piece of metal with fixed cams. The 'bearing surfaces' are hardened with a certain surface finish. Larger engines, in particular those in category III, have built-up camshafts in which each cylinder has its own section which is connected to the following section by means of fitted bolts. This simplifies the removal of a damaged cam.



▲

A Daihatsu four-stroke medium-speed diesel engine operating on H.F.O.

The camshaft is driven by gearwheels connected to the crankshaft; on both shafts one gearwheel and one intermediate shaft are mounted, a standard version. The four-stroke process has 720 crank degrees; therefore the camshaft rotates at half the revolution rate of the crankshaft. This is observed by the ratio between the diameter of the gear on the crankshaft and the diameter of the gear on the camshaft (twice as many teeth on the camshaft wheel).



◄

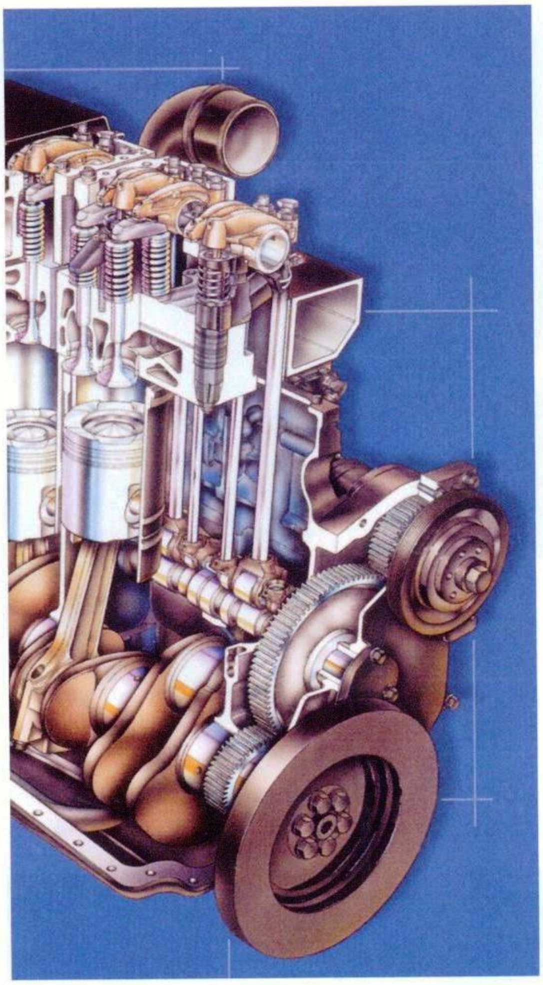
Camshaft for a MAN–B&W medium-speed diesel engine.

The high-pressure fuel pump and the inlet- and exhaust cams are operated by the camshaft driven by gears or cog belts (smaller engines category I) from the crankshaft.

▶

The camshaft drive for a high-speed four-stroke Cummins diesel engine.

The ratio between the cog wheel (gear) on the crankshaft and the camshaft (half the RPM of the crankshaft) is clearly visible.



▲

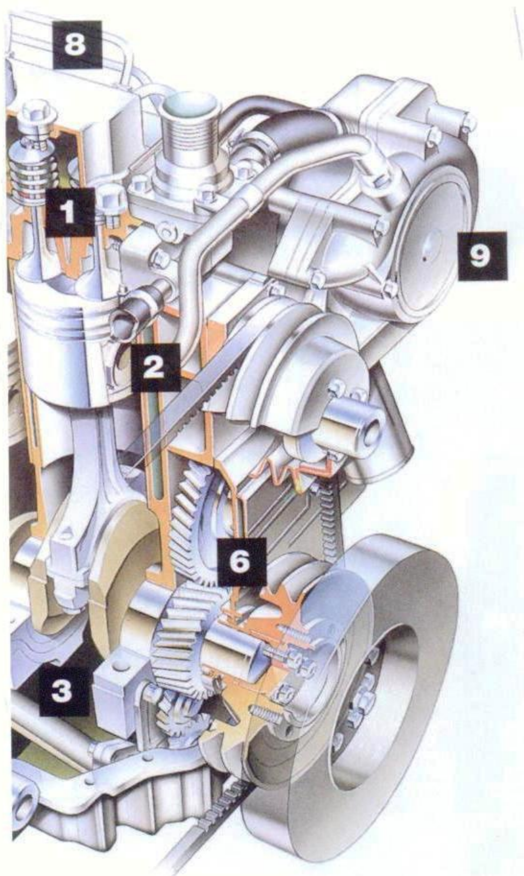
The drive of both camshafts in a V-engine, of RK 280, now MAN–B&W.

1 *crankshaft*
2 *camshaft*
3 *intermediate wheel*

▶

A helical drive.

The objective here is the pressure reduction on the gear teeth. An angled tooth has a larger surface area with an identical cog-wheel width, thereby reducing the tooth pressure. The angled teeth generate an axial force on the shafts which must be absorbed by a thrust bearing.



▲

A crankshaft and a camshaft; both forged from one piece for a diesel engine, category II.

Camshafts with separate cams

In older engines, cams were bolted to the camshaft. Today there are numerous engines with hydraulic cams. They are fitted to a finely polished shaft using oil pressure. It is therefore possible to slightly adjust the position of the cam and therefore adjust the timing of the exhaust valves or fuel pumps.

▲

Hydraulic cams.

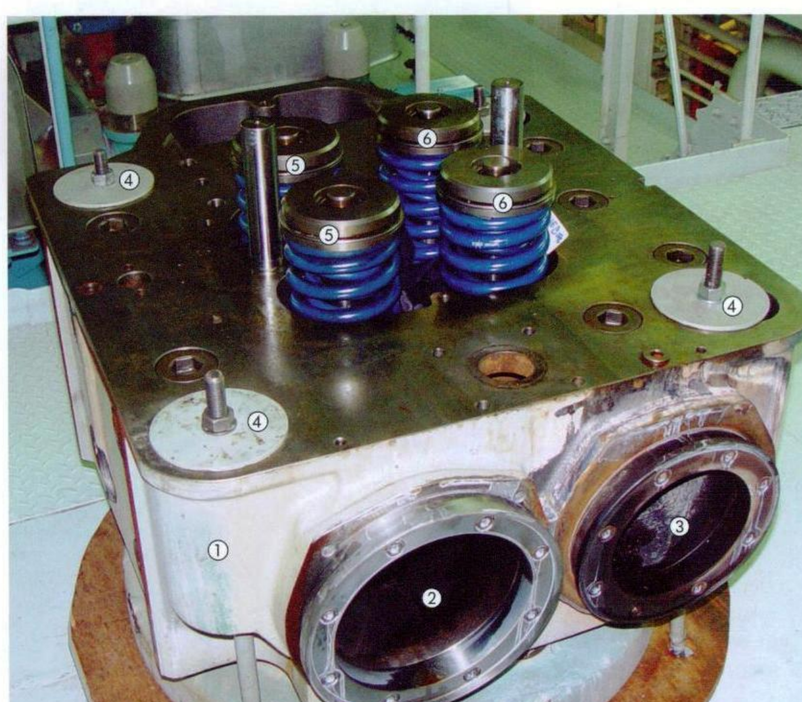
They are mounted on the camshaft using oil pressure. They can be easily exchanged when damaged.

◄

This version has three cams: for the inlet and exhaust valves and the high-pressure fuel pump.

13.2.7 Inlet- and exhaust valve drive

These are driven by the camshaft by cams which when the camshaft rotates, the lobe on the cam pushes upwards putting the valve into motion via the push rod and the leverage system of the roller guide. As most of today's engines are equipped with two inlet- and two exhaust valves, the drive of these valves on the cylinder head can be very complex. The valve springs ensure that the valves close as soon as the cam position allows this.



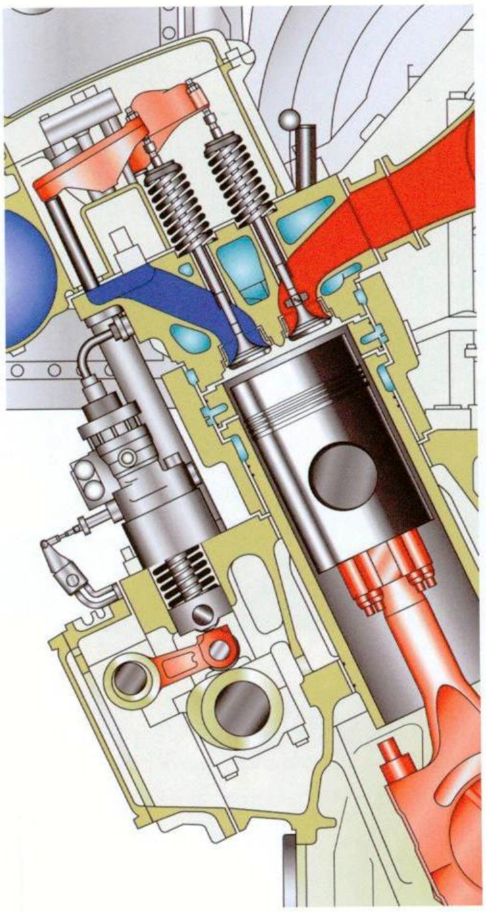
►

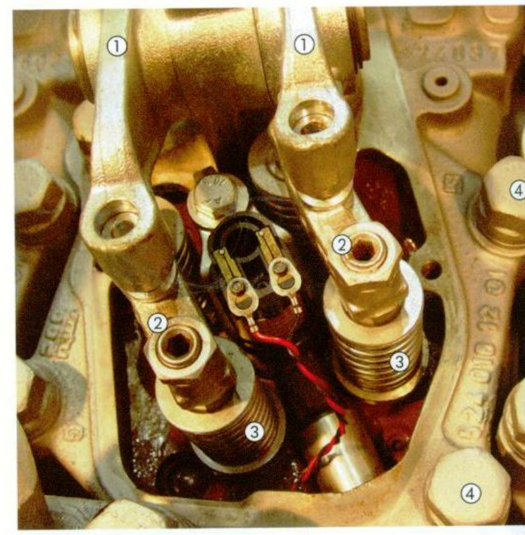
A cylinder head with mounted valves.

1 *cylinder head*
2 *inlet port*
3 *exhaust port*
4 *location for cylinder-head bolts*
5 *inlet valve*
6 *exhaust valve*

►
A cross-section of a four-stroke medium-speed diesel engine of MAN–B&W type 51/60 V dual-fuel.

The valves are driven by two rocker arms, left for both inlet valves and right for both exhaust valves.





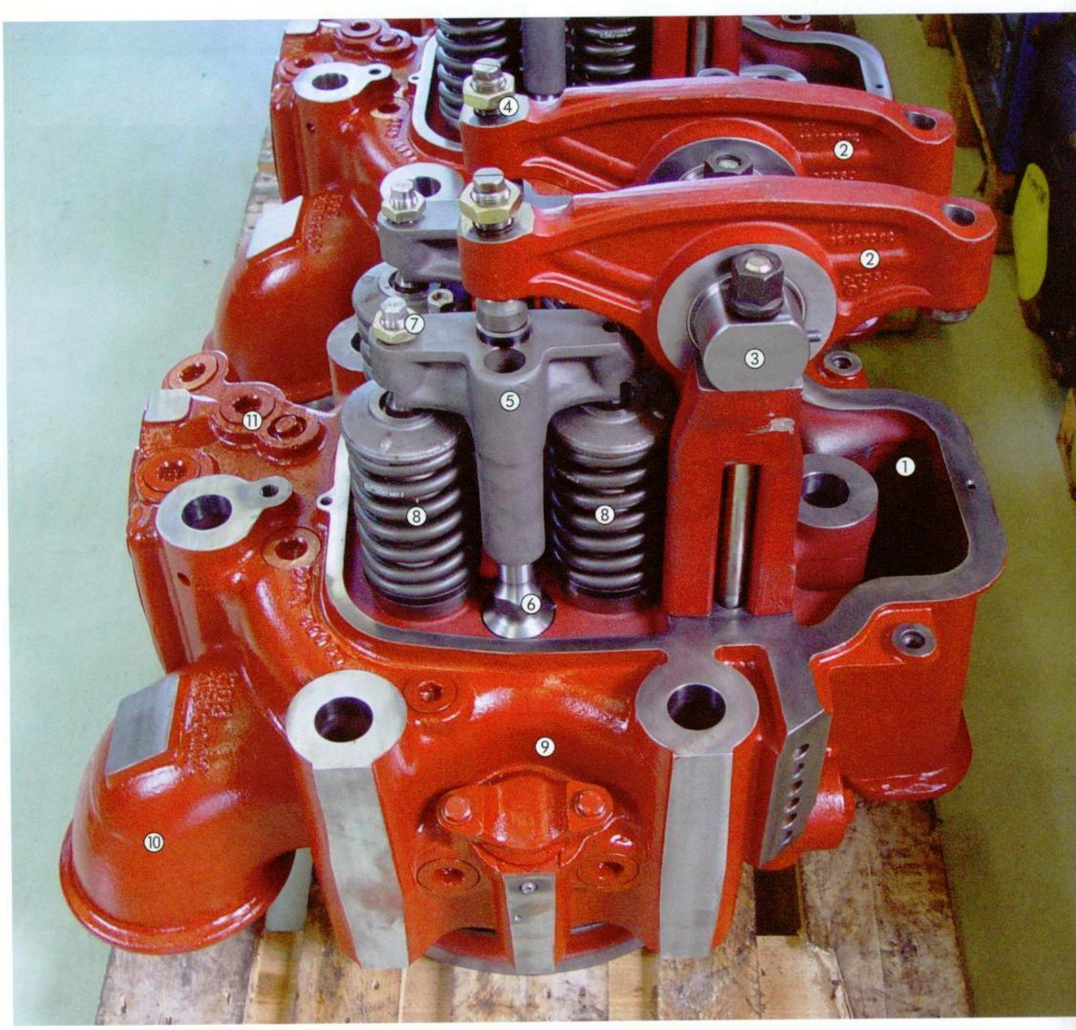
▲
Top view of a valve drive.

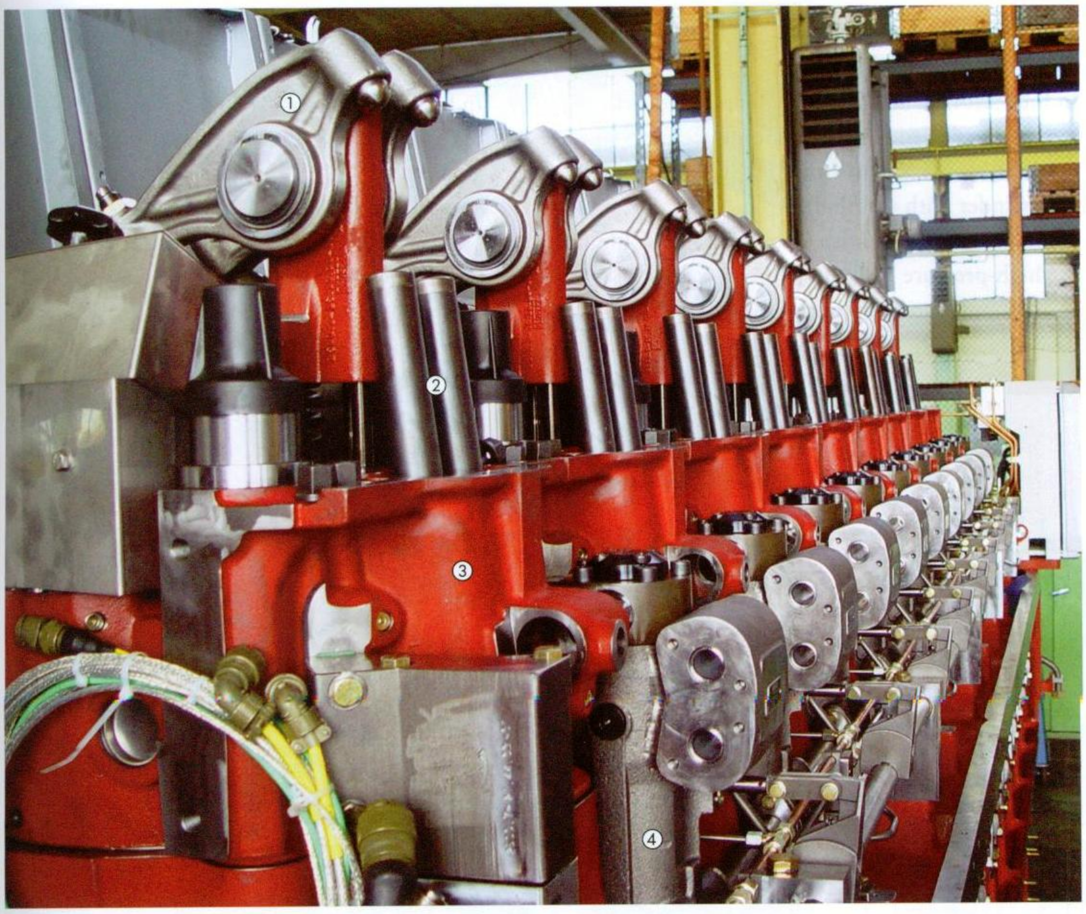
The valves are deflected in relation to the cylinder centre line. This allows a good air movement in the cylinder for the air supplied via the inlet valve.

1 *rocker*
2 *spreader*
3 *valve spring*
4 *cylinder-head bolts*

►
The drive for the inlet and exhaust valves in medium-speed Caterpillar–MaK diesel engines.

1 *location for the push rod*
2 *rocker*
3 *rocker shaft*
4 *adjusting bolt for the valve clearance*
5 *spreader*
6 *spreader guide*
7 *adjusting bolt for the valve clearance*
8 *valve springs*
9 *cylinder head*
10 *inlet duct*
11 *exhaust duct*



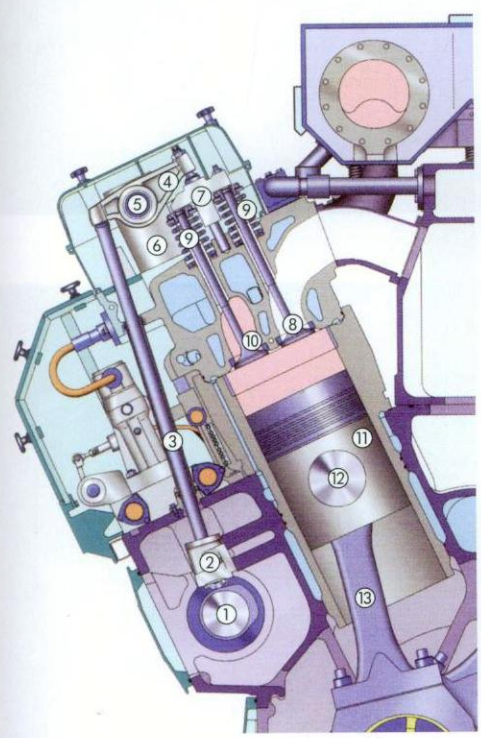


◄

Construction of a Caterpillar–MaK engine at Kiel, Germany.

The valve rockers are upright; the push rods placed over the camshaft.

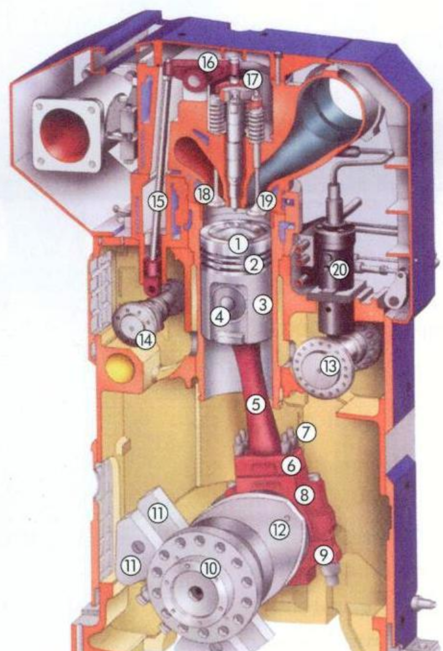
1 *valve rocker*
2 *push rods*
3 *cylinder head*
4 *high-pressure fuel pump*



▲

A complete valve drive.

1 *camshaft with inlet cam*
2 *pulley with guide*
3 *push rod*
4 *rocker*
5 *rocker shaft*
6 *rocker arm seat*
7 *spreader*
8 *inlet valve*
9 *valve spring*
10 *exhaust valve*
11 *piston*
12 *piston pin*
13 *connecting rod*



◄

The driving gear for a four-stroke MAN–B&W diesel engine.

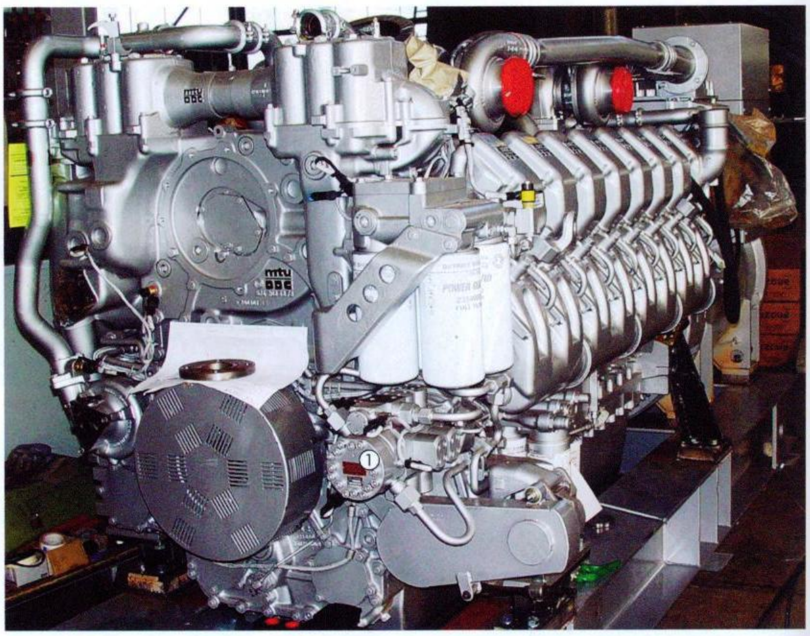
Note the position of the two camshafts; one for the valve movement and one for the high-pressure fuel pump.

1 *piston crown*
2 *piston rings*
3 *piston skirt*
4 *piston pin*
5 *connecting rod*
6 *"Marine head big-end"*
7 *connecting-rod stud*
8 *top cover big-end*
9 *bottom cover big-end*
10 *crankshaft*
11 *counterweight*
12 *crank web*
13 *camshaft for high-pressure fuel pump*
14 *camshaft for the inlet- and- exhaust valves*
15 *push rod*
16 *valve lever/rocker*
17 *spreader*
18 *exhaust valve*
19 *inlet valve*
20 *high-pressure fuel pump*

13.2.8 Fuel pumps drive

In four-stroke trunk piston engines most fuel pumps are driven in the traditional manner:
- Individual high-pressure fuel pumps for each cylinder with camshaft.
- In diesel engines with a concentrated combined high-pressure fuel pump, the so-called block fuel pump, tooth wheels are used.

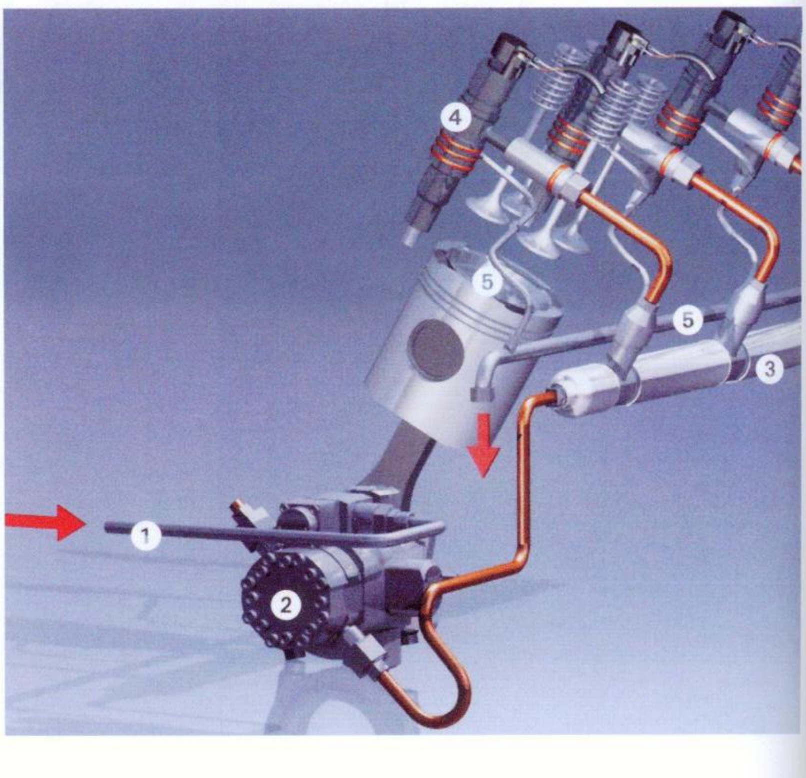
Diesel engines with the latest common-rail system use the following methods:
- In high-speed diesel engines a high-pressure fuel pump is often installed near the cog wheels.
- In medium-speed diesel engines the high-pressure fuel pump is driven by the camshaft; however fewer high-pressure pumps are present, normally half the number of cylinders. These pumps operate in tandem with a mutual pressurised buffer which leads the fuel to the cylinders.



▶
A six-cylinder MTU-diesel engine category II with a common-rail fuel system.

The high-pressure fuel pump is directly driven with the cogwheel train.

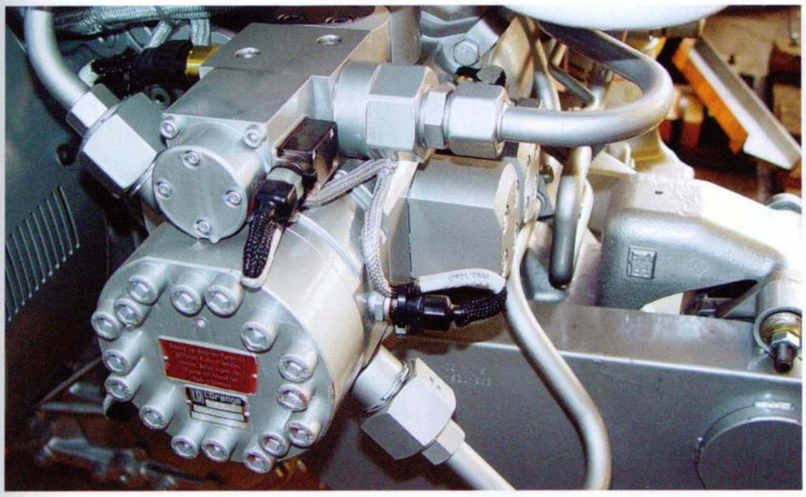
1 *high-pressure- -fuel pump*



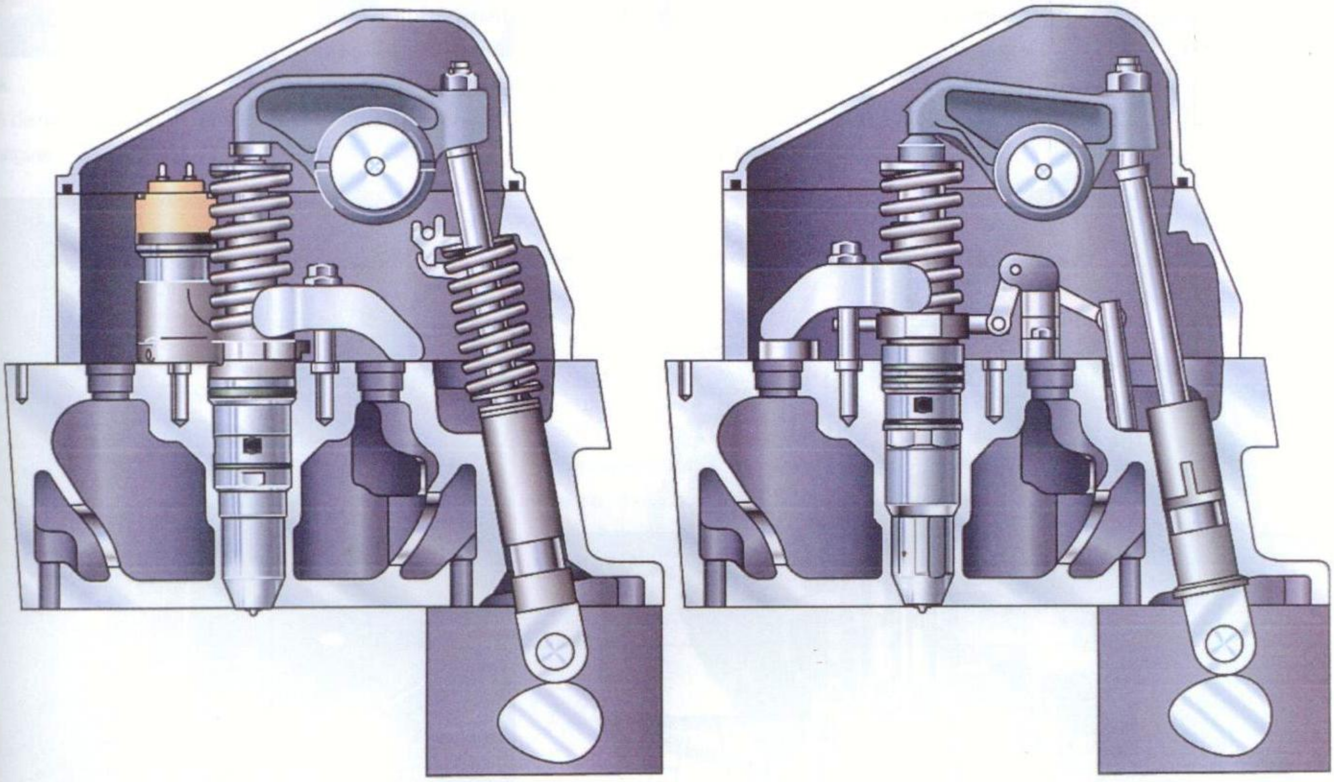
▶
The fuel system of a MTU-diesel engine, series 2000.

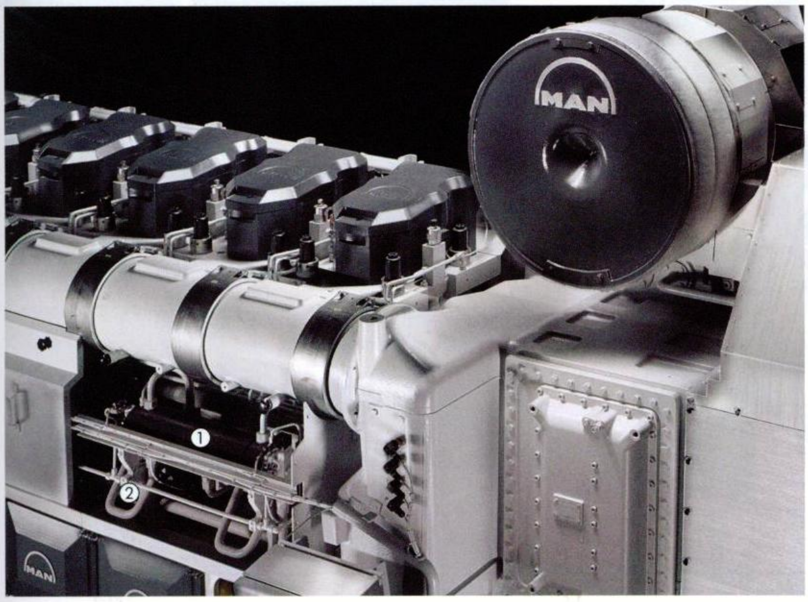
1 *fuel-supply line*
2 *fuel pump*
3 *common-rail high-pressure fuel line*
4 *fuel injectors*
5 *fuel-discharge line*

Four-stroke engines equipped with a common-rail fuel system require camshafts for driving the inlet- and exhaust valves. The main reason is that hydraulically driven valves at high engine speeds are complicated and therefore expensive.
The high-pressure fuel pump is driven with a cogwheel train connecting the crankshaft and camshaft.



◄
The high-pressure fuel pump of l'Oranje for MTU-engines with a common-rail system.





◄
The common-rail system of MAN–B&W for medium-speed engines, category III.

1 *buffer tank*
2 *high-pressure fuel pump*

▲
The fuel-injector drive in four-stroke high-speed Caterpillar engines.

Left: electronic control and right: mechanical control.

▼

Two-stroke low-speed crosshead engine of MAN–B&W MC 98.

The reciprocating movement of the piston converts combustion pressure to a rotating motion of the crankshaft via the crank connecting-rod mechanism.

1 piston
2 piston rod
3 piston-rod stuffing box
4 crosshead
5 guide shoe
6 crosshead guides
7 connecting rod
8 crankpin
9 crank web
10 crankshaft

13.3 Engine-driving gears in two-stroke crosshead engines

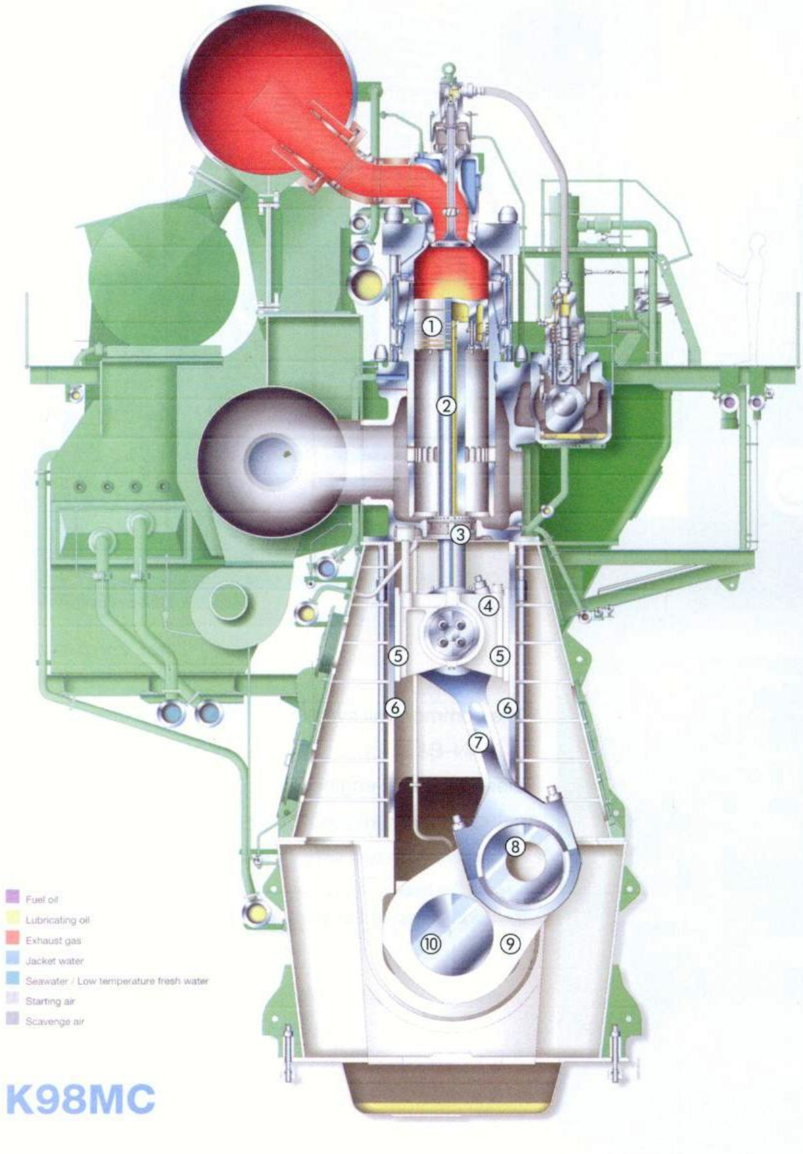
Here the drive gears comprise the following parts:

- piston with piston rings;
- piston rod;
- crosshead;
- crosshead guides and crosshead shoes;
- connecting rod with the crosshead bearings;
- crankshaft with the crank pin-and main shaft bearing;
- camshaft drive;
- exhaust valve drive;
- conventional fuel pump drive;
- common rail system drive for crosshead engines;
- reverse gears.

13.3.1 Pistons

The pistons of two-stroke crosshead engines transfer the gas forces to the piston rod, the crosshead and via the connecting rod to the crankshaft.

The lateral crank- connecting rod mechanism forces are entirely absorbed by the crosshead and via the crosshead guides conveyed to the crosshead shoes which are fixed to the welded A-frame. The piston, together with the piston rings ensures a gas proof sealing of the cylinder. When the piston is short; the cylinder diameter is often larger than the height of the piston. Therefore the piston ring package does not contain a separate scraper and distribution ring; the cylinder lubricating oil is supplied through the perforations at the beginning of the compression stroke.



▲

A new piston for a two-stroke crosshead engine of MAN–B&W.

Note the large number of hydraulically tensioned studs used to attach the piston to the piston-rod flange and the bronze guide rings in the piston skirt.

▲

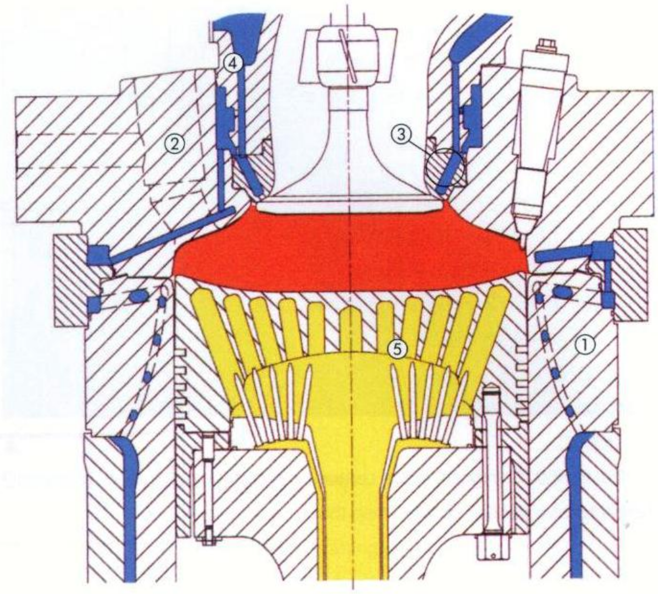
A dismounted piston for a large two-stroke crosshead engine.

Note the 'seized' top ring and the ribbed profile on the piston skirt; this profile serves as a lubricating-oil chamber.

▼

A cross-section of an earlier piston type and a more recent piston type of MAN–B&W–MC.

The differences between the combustion chambers are remarkable. The shape is different and the mean material temperature is approximately100° C lower on top of the piston crown. The piston is cooled with lubricating oil which is circulated by pumps.



▲

A cross-section of the combustion space in Wärtsilä RTA 58 T – B and RTA 68 T – B engines.

The following parts in the proximity of the combustion space are provided with drillings, also referred to as bore cooling.

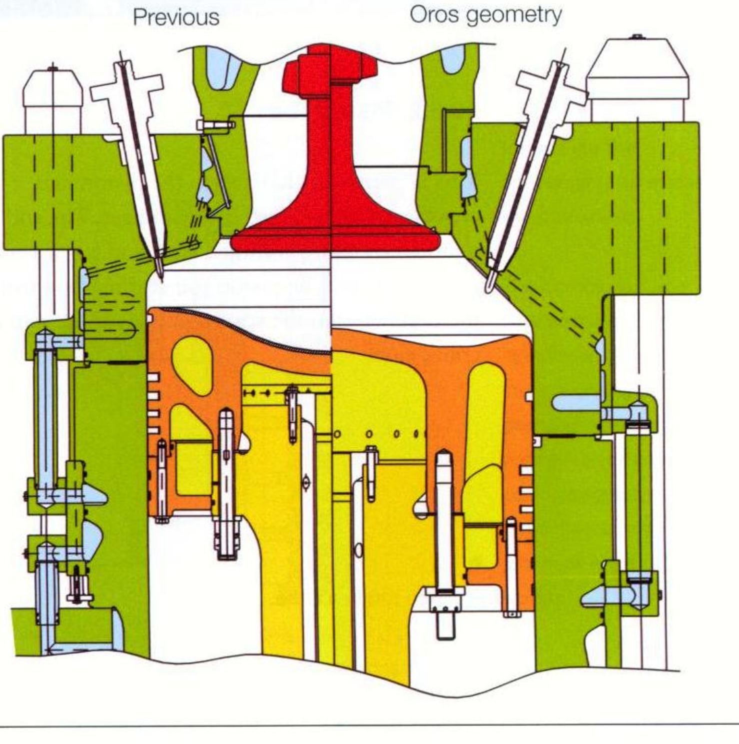
1 *cylinder liner*
2 *cylinder head*
3 *exhaust-valve seat*
4 *exhaust-valve casing*
5 *piston crown*

Comment: In comparison to other two-stroke crosshead engines these engines have very little cooling water! Therefore, draining the cooling water of a cylinder during repairs takes up little time!

Features:
- High topland
- Oros piston top geometry
- CPR top ring
- Alu-coat piston rings
- Bore cooled, forged piston of heat resistant steel
- Piston cleaning ring

Improvements:
- Approx. 100 °C lower temperature on top compared to former type piston
- Elimination of Inconel coating on piston top
- Increased chrome layer thickness in bottom of ring grooves
- Anti-erosion bushing in oil outlet in piston rod foot

Verification:
- Extensive calculations
- Comprehensive tests on K90MC and K90MC-C
- Service test on K90MC



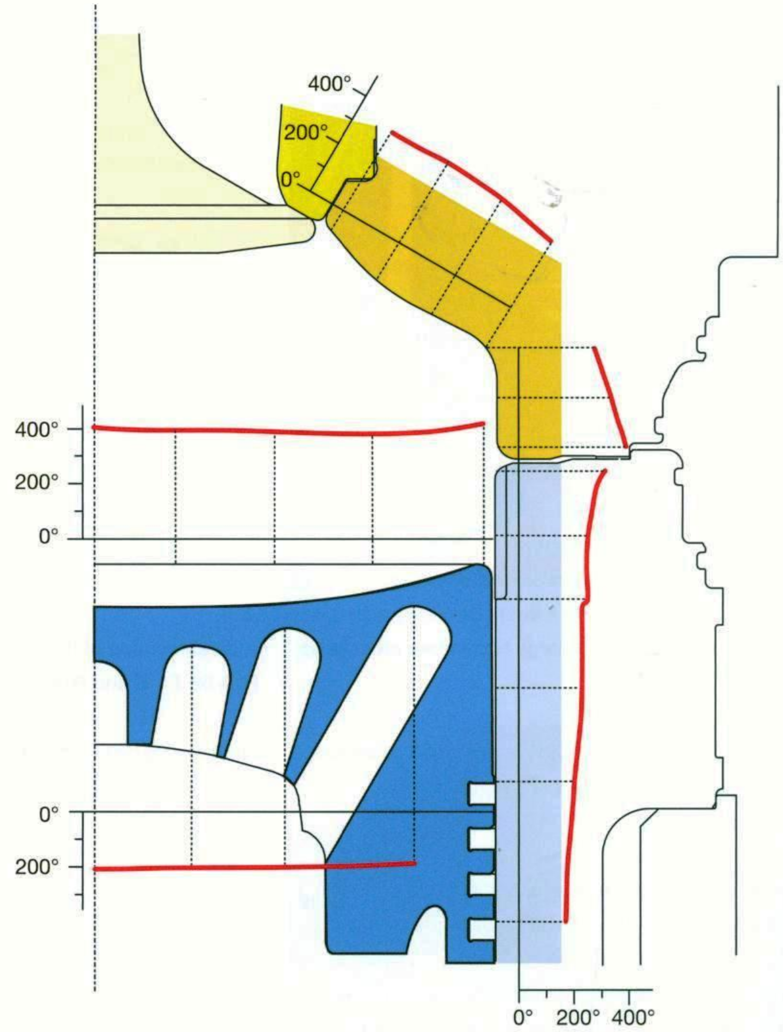
▶

The temperature developments in the parts found in the combustion space of the Wärtsilä RTA 58 T–B and RTA 68 T–B engines.

The engine design is continually being adapted to ensure that material temperatures of the parts do not dramatically increase when the cylinder output increases. In fact, temperatures of the parts are slightly lower in new designs.

Temperatures:

piston crown: ± 300 to 375 °C
piston crown, lubricating-oil cooled interior: ± 150 to 170 °C
cylinder liner: ± 200 to 260 °C
exhaust valve disc: ± 450 to 550 °C (highest in the centre, lowest at the seat; here intensive cooling occurs when the valve is closed.)
exhaust-valve seat: ± 250 to 350 °C



Today, all pistons are cooled with lubricating-oil via a separate piston cooling oil system. Due to the high load, the pistons are manufactured from forged steel, allowing repairs to take place using a rewelding process.

13.3.2 Piston rod

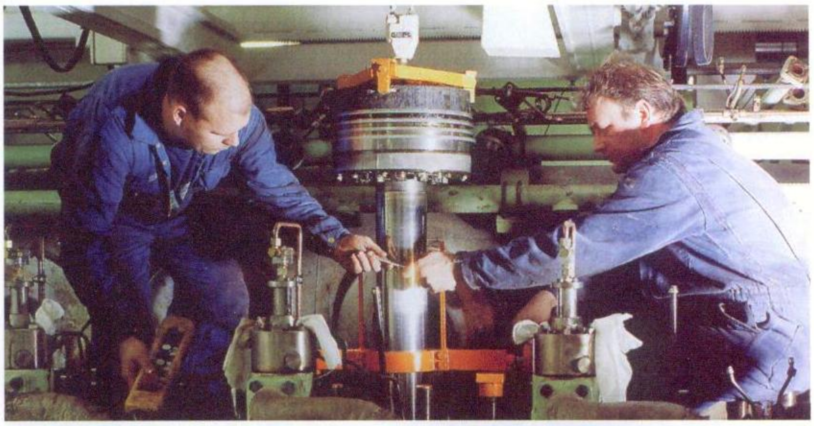
This is permanently fixed to the piston and crosshead using bolted heavy flanges. The rod is manufactured from forged steel and has a very smooth finish. The piston rod stuffing box forms the seal between the scavenging air space and the crank case.

▶

A piston rod in a lathe.

In this instance, the piston rod is to be cleaned (removal of the coating) and polished. The rectangular flange in the foreground is used to attach the rod to the crosshead.

▲
A piston with piston rod and a bare piston rod after repair.



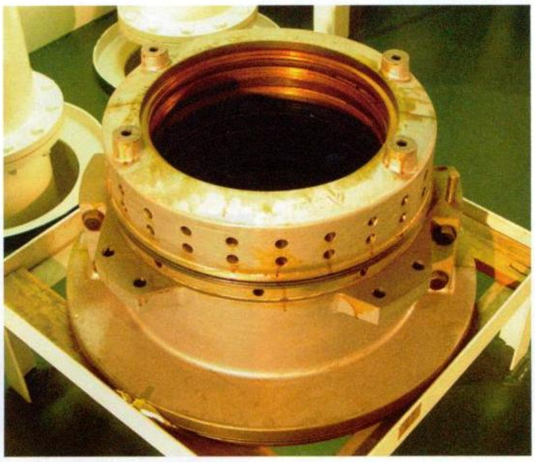
▲
Dissembling a piston with piston rod.

The piston crown is grooved so the hoisting gear can be attached. When the piston is in its top position, it is sufficiently high above the cylinder liner for attachment of the hoisting gear.

13.3.3 Piston rod stuffing box

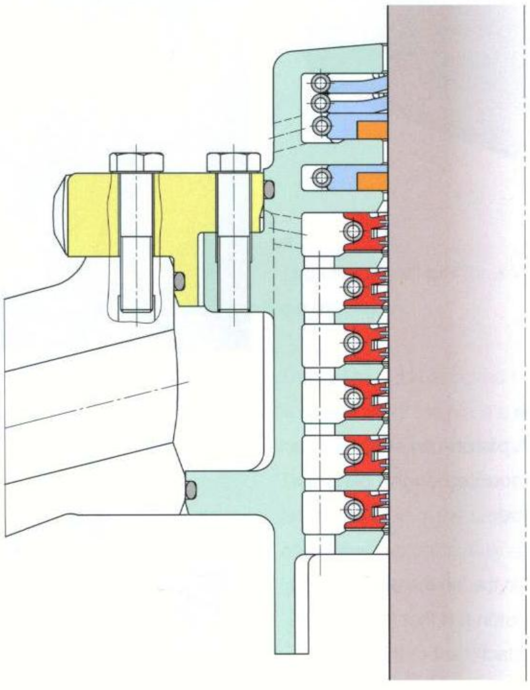
The piston rod stuffing box separates the space over the sealing rings, the scavenging air space, from the space below the sealing rings, the crank case. The scavenging air space always contains some type of dirt. Residues from the combustion process above the piston can reach this space through the piston ring package, which is subject to wear and tear. This, combined with cylinder lubricating-oil and condensed water from the inlet air, can form a soiled layer on the bottom of the scavenging air space. A scraper ring on **the upper part** of the stuffing box scrapes the dirt from the piston rod and transports it outside the engine. The base plate of the scavenging air space is provided with a drain pipe which transports dirty oil substances to a dirty oil tank outside the engine via a ball valve. The amount of contaminated oil drained from the scavenging air space is checked at regular intervals using a measuring cup. In this manner the amount of cylinder lubricating-oil that is burned in the cylinder is measured. The amount of lubricating-oil that is supplied to the cylinder is known. Once the amount of lubricating-oil drained from the dirty scavenging air is measured, the amount of combusted lubricating oil in the cylinder combustion space is also known.

There is a relatively clean space underneath de piston rod stuffing box, the crank case with drive gears such as the crosshead guides, connecting rod and crankshaft. These are provided with lubricating oil from the main lubricating-oil system which is continuously circulating the lubricating-oil in the engine. The pistons are cooled using this clean system. The **bottom part** of the piston rod stuffing box scrapes the relatively clean lubricating-oil from the piston rod and subsequently drains into the crank case.



◄
A complete piston-rod stuffing box.

Note the large holes in the upper section for contaminated cylinder lubricating-oil discharge. The packing is vertically divisible for assembly and disassembly.



◄
A piston-rod stuffing box for sealing the scavenging-air space and the crankcase.

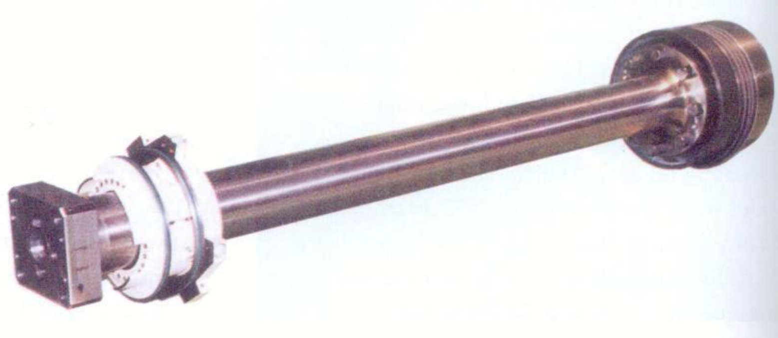
The top ring package scrapes the contaminated cylinder lubricating oil from the scavenging-air space and discharges it outside the engine. The bottom ring package scrapes the clean oil off the rod, after which it falls back in the crankcase sump.

▲
The piston with piston rod after removal from the engine. It is temporarily placed on the floor next to the cylinders. The piston-rod stuffing box (1) can now be renewed.

▲
A complete ring package for the stuffing box.

Each package-retaining ring contains three segments which are kept pressed against the piston rod by a spiral spring.

►
The two halves of the casing of the piston-rod stuffing box. Note the drainage holes for the clean lubricating oil for the crankshaft and the contaminated cylinder lubricating-oil discharge.

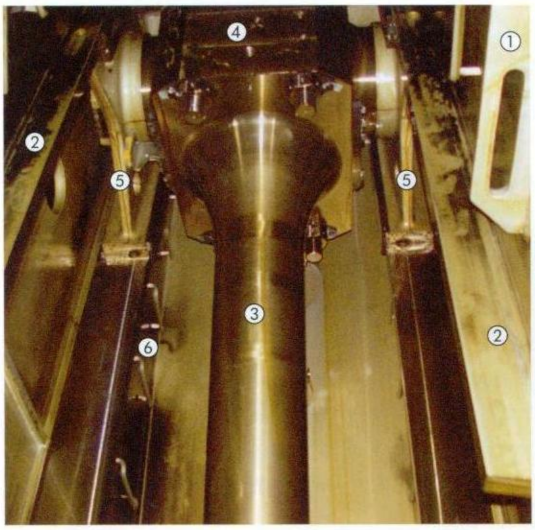


►
A piston-rod stuffing box mounted on the piston rod.

On the left the rectangular piston-rod foot for attachment to the crosshead.

13.3.4 Crosshead

The crosshead connects the piston rod and the connecting rod. The piston rod is attached to the crosshead and the connecting rod can pivot around the crosshead.
The lateral forces acting on the connecting rod mechanism are transferred to the crosshead shoes, fixed to the frame, via crosshead guides.
The forged iron connecting rod is attached to the crosshead with a forked connection, so that there are two bearings on each side of the crosshead pin. In the past the bearing caps were manufactured from a ductile white metal. Today, modern engines have hard tri-metal caps which have better resistance to the significantly increased surface pressures, produced by the higher combustion pressures exerted on the piston.



◄
A crosshead in an engine.

1 *welded A-frame*
2 *crosshead guide*
3 *connecting rod*
4 *crosshead*
5 *(crosshead) guide shoes*
6 *steps in crankcase*

◄
A complete crosshead with crosshead pin and guide shoes.

▲
Maintenance on the crosshead. Fitting hydraulic jackets to the studs for attachment of the connecting rod to the crosshead.

1 *hydraulic jack*
2 *connecting rod*
3 *crosshead*
4 *crosshead guide*
5 *hydraulic pressure bolts for attachment of the top cover of the main journal bearing*

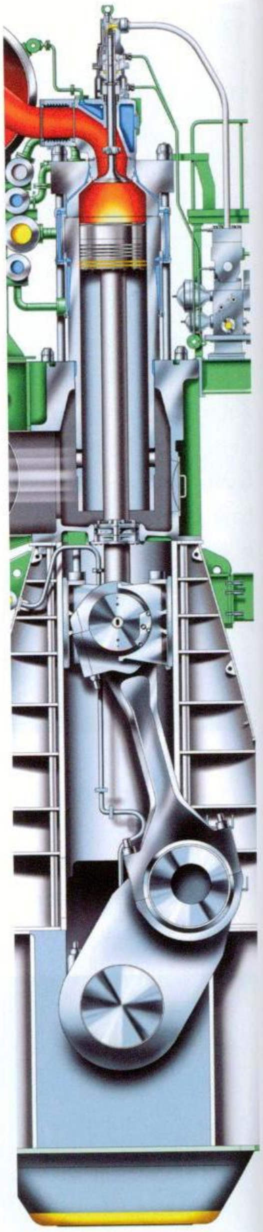
▲
The crosshead pin for a Wärtsilä Sulzer RTA two-stroke diesel engine.

The piston rod is attached to the top of the pin by four bolts. In the centre of the pin is a perforation for the cooling lubricating-oil supply and removal to and from the piston. The connecting-rod bottom bearing supports the entire bottom section of the crosshead pin. The top bearing comprises two parts; they are mounted on the left and right side of the piston rod.

13.3.5 Connecting rod

The connecting rod is bolted to the top of the crosshead by means of a 'fork' on two complete crosshead bearings. The bottom of the connecting rod is bolted to the crankpin. Connecting rods are forged from steel which is either alloyed or not. The forces in the drive gears for these category IV engines can be several hundred tons with a maximum of 1100 tons!

Therefore all drive gears, including the bearings, have a heavy-duty finish. Today, tri metal bearings are increasingly used as opposed to the soft white metal bearings, as they have a better resistance to high engine loads.



▶
The driving gears for a Wärtsilä Sulzer RTA 96 C engine.

Note the large dimensions of the parts of the driving mechanism in relation to the piston.

diameter:

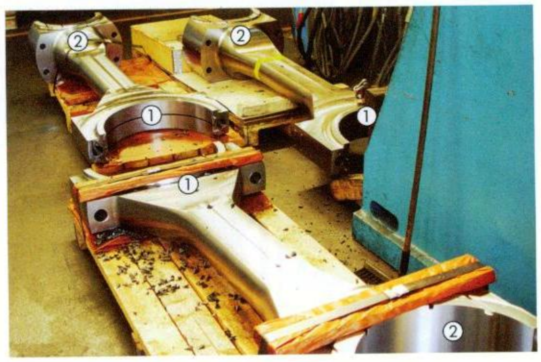
diameter piston rod: 375 mm

diameter crosshead pin: 980 mm

diameter connecting rod: 530 mm

diameter crankpin: 1350 mm

diameter crankshaft: 1200 mm



▶
The connecting rods for MAN-B&W 50 MC engines.

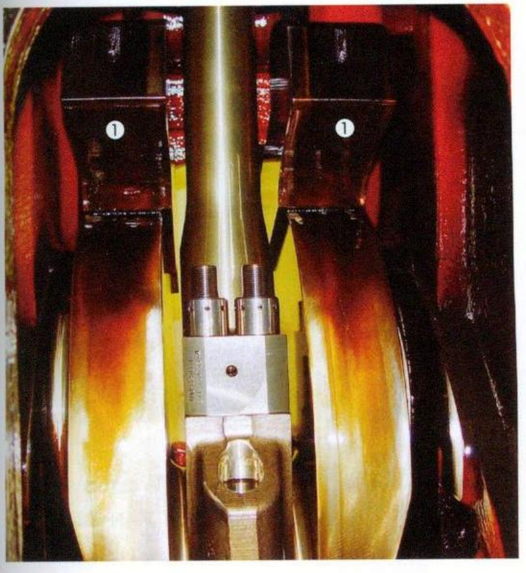
1 *crankpin side*
2 *crosshead side*

◀
Connecting rods for MAN–B&W K 50 MC engines.

Left: the connection with the crosshead and right: the connection with the crankpin.

◀
The connecting-rod head for a MAN–B&W K 50 MC engine.

Between the notches, the crosshead oscillates in relation to the reciprocating movement of the piston rod.



▲ Mounting the connecting rod on the crankpin with four hydraulically tensioned connecting-rod bolts.

The counterweights on both crank webs are visible in the crankcase (1).

▲ The crankshaft in the engine.

1 *connecting rod*
2 *big-end with connecting-rod bolts*
3 *crank webs*
4 *main journal bearing lubricating-oil supply*
5 *main journal bearing top cover*
6 *hydraulic bolt for attachment of the top cover*

13.3.6 Crankshaft

All crankshafts of two-stroke engines are constructed by joining individual forgings comprising one crankpin with two webs forged from a single piece of metal with main shafts shrunk onto the webs.

Long crankshafts are manufactured in two parts connected by large flanges.

As a result of increasing combustion forces, the crankpin and crankshaft diameters have also increased with a decrease in the distance between the crank webs to avoid distortion of the of the crank pin.

This also serves to shorten the engine in spite of increasing cylinder capacity; this translates into a smaller engine room, thus providing more cargo space. Crankshafts of large crosshead engines are 'soft'; they do not have a hardened finish. The crankpin and main shaft bearings are manufactured from a tri-metal.

13.3.7 Camshaft drive

There are two traditional methods for the drive of the camshaft by the crankshaft:

1 **With tooth wheels**
This method is still applied in the Wärtsilä Sulzer engines series RTA and the Mitsubishi UC engines.

2 **With chains**
This method is applied in the MAN–B&W M series MC.

◄ A partially built crankshaft for a low-speed two-stroke crosshead engine of MAN–B&W, type MC 50.

Crankpin and webs are made from a single forging. The main journals have been shrunk in the webs. The crankpins are hollow (weight saving) and the crank webs are positioned closely together in order to reduce the crankpin length. This results in a robust crank with little distortion.

Modern engines with the common rail system from both engine manufacturers are no longer equipped with camshafts. New series: Wärtsilä Sulzer RT FLEX and MAN–B&W K ME.

Advantages of gear drive

- Fine-tuned drive in which slight wear and tear of the teeths does not affect the 'timing' of the camshaft.

Disadvantages

- Expensive, heavy.
- Requires meticulous alignment of the tooth wheels.

Advantages of chain drive

- Relatively inexpensive.
- Light in weight.

Disadvantages of chain driving

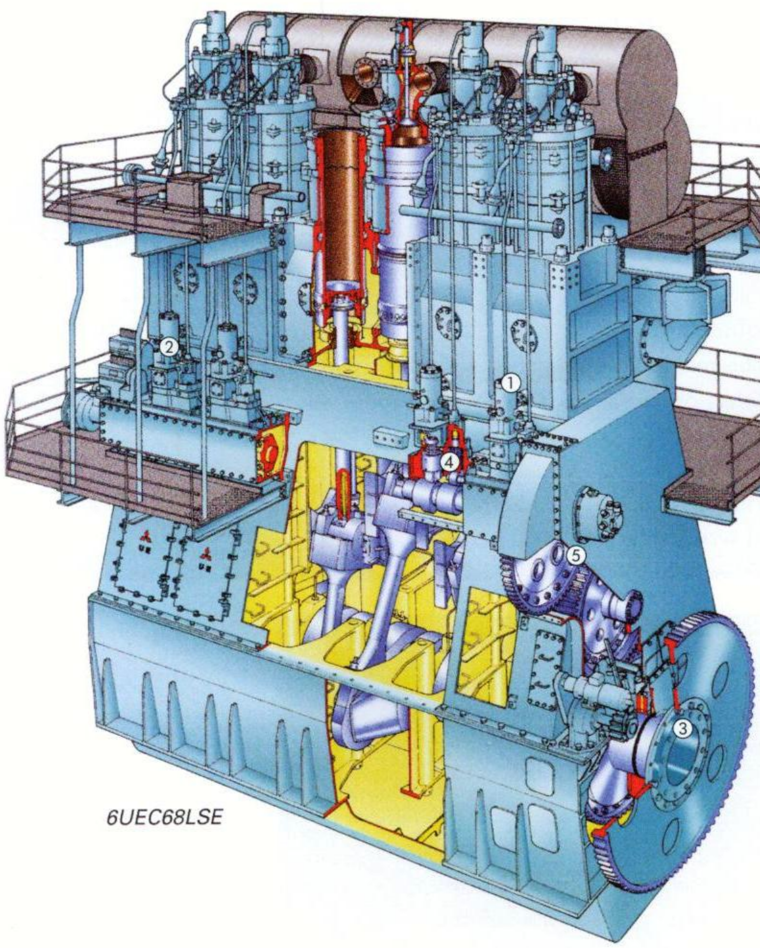
- Requires periodic tensioning due to wear and tear of the links of the chain, otherwise the camshaft timing will lag.

▼

A low-speed two-stroke crosshead engine, manufacturer Mitsubishi, type 6UEC 68 L S E.

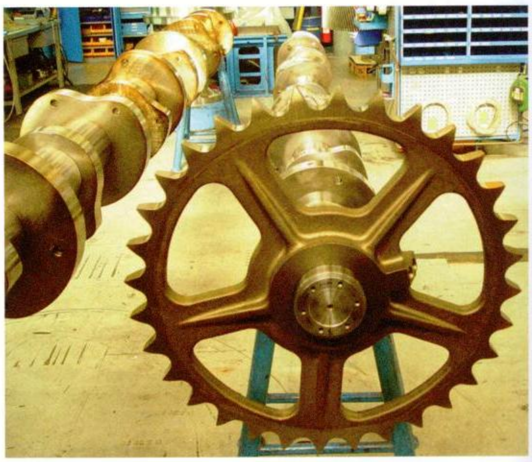
Here the camshaft is driven by spur gears.
On the right, directly next to the high-pressure fuel pumps is the hydraulic pump for the exhaust valve.

1 *high-pressure fuel pump*
2 *hydraulic pump for the exhaust valve*
3 *crankshaft*
4 *camshaft*
5 *intermediate wheel, two pieces*



▲

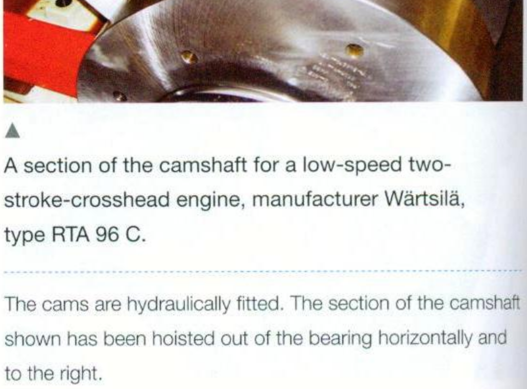
A chain and a gearwheel for driving the camshaft for a large two-stroke crosshead engine of MAN–B&W.



▲

A camshaft for a MAN–B&W K 50 MC two-stroke low-speed crosshead engine.

The camshaft is chain-driven.



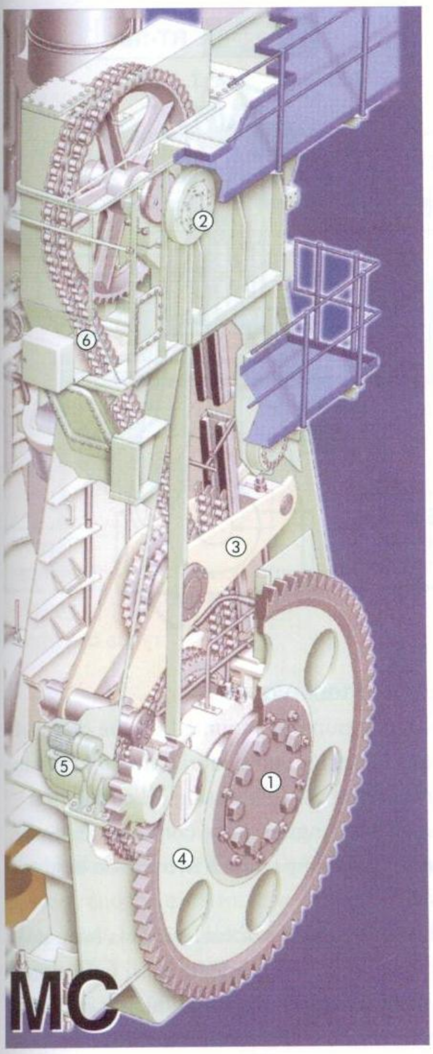
▲

A section of the camshaft for a low-speed two-stroke-crosshead engine, manufacturer Wärtsilä, type RTA 96 C.

The cams are hydraulically fitted. The section of the camshaft shown has been hoisted out of the bearing horizontally and to the right.

13.3.8 Propulsion of fuel pumps

Two-stroke crosshead engines: traditionally, fuel pumps were driven from the camshaft, with a high-pressure fuel pump placed directly above the cam. By rotating the cam or by shifting the camshaft, the fuel injection can be adjusted for the opposite direction of rotation.

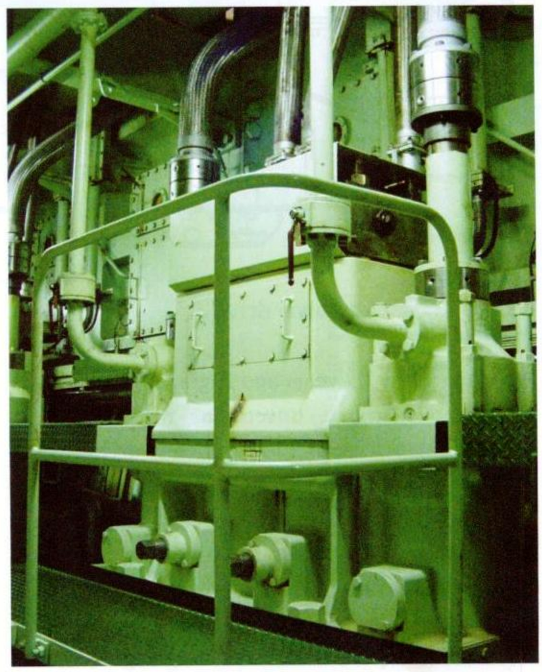


▲

The chain-drive in a low-speed two-stroke crosshead engine of MAN–B&W type MC.

The drive is installed with two chains with a tensioning device.

1 *crankshaft*
2 *camshaft*
3 *tensioning device*
4 *turning wheel*
5 *turning drive*
6 *double chain*



◀

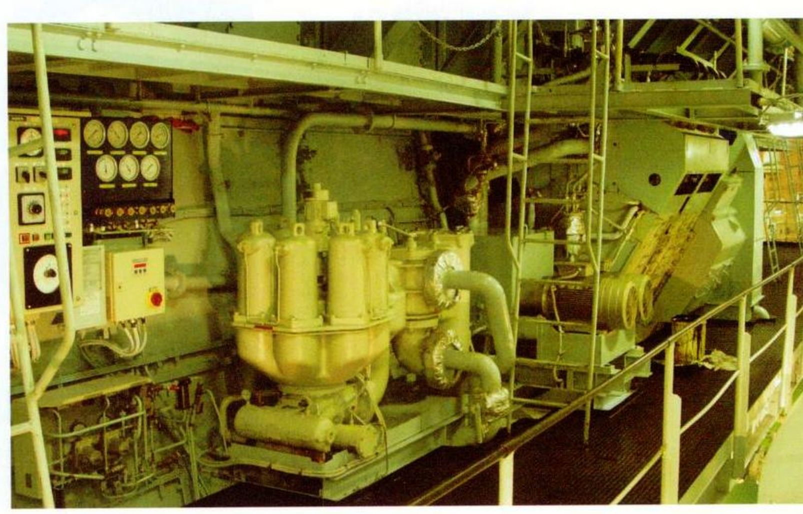
Traditional drive.

The traditional fuel-pump drive in a two-stroke crosshead engine of Wärtsilä Sulzer, type RTA 96 C. Two pumps in one casing with below, the camshaft. The pumps are valve-controlled.

◀

Two high-pressure fuel pumps driven by the camshaft of a MAN–B&W 50 MC crosshead engine.

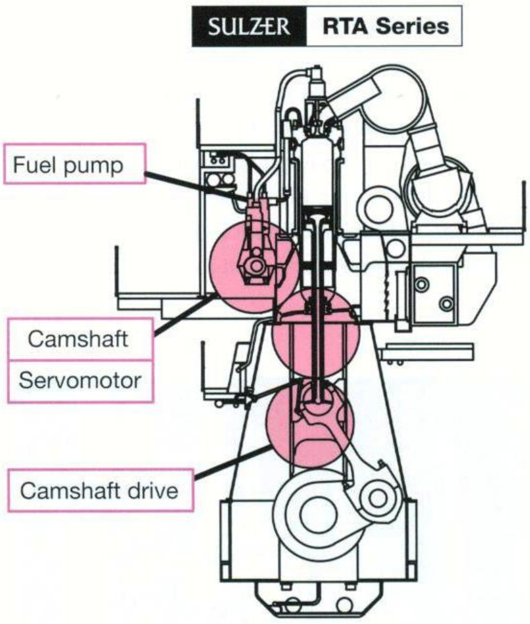
In the centre the hydraulic exhaust-valve drive.

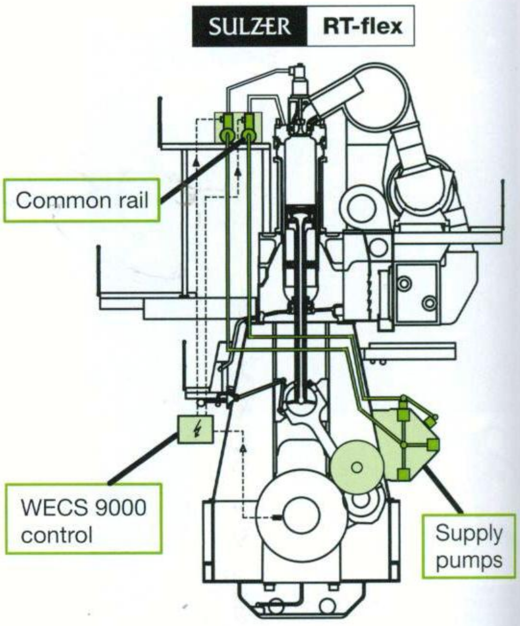


▶

In the latest two-stroke crosshead engines with a common-rail system the high-pressure fuel unit is driven directly from the crankshaft with gearwheels.

There are two camshafts in the V-shaped unit, which drive the original high-pressure fuel pumps of a four-stroke engine, the Wärtsilä 46. This engine has no camshaft.





Left: the traditional Wärtsilä Sulzer RTA series, and right: the common-rail version without camshaft, the Wärtsilä Sulzer RT-FLEX.

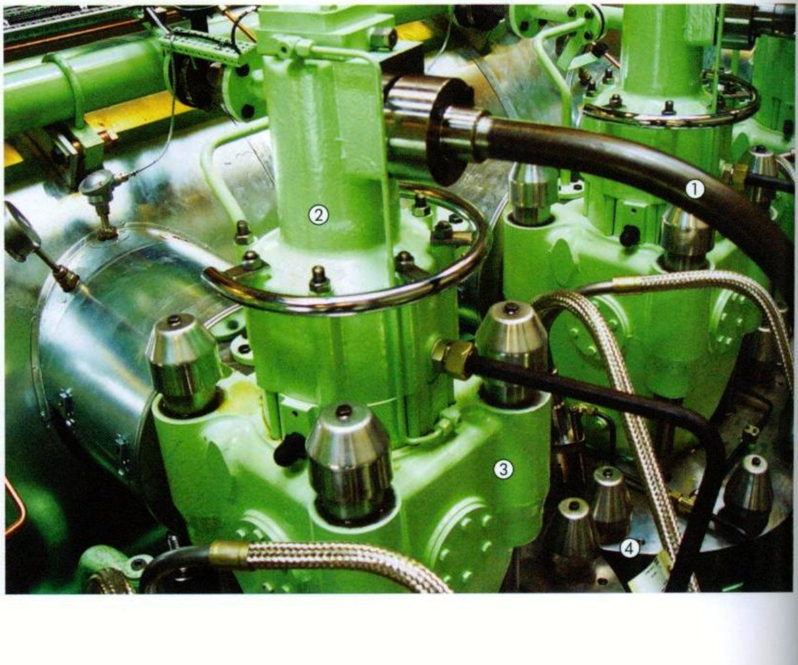
The hydraulic power-supply unit of the common-rail fuel system of the Wärtsilä series RTA engines (figure on the right) is placed at the lower section of the engine, here on the right side. The camshaft, driven from the crankshaft, in the figure on the left (pink), is not present in the right figure. The drive gearing for the engine has been simplified.

13.3.9 Exhaust valve drive

Approximately 25 years ago engine manufacturers changed over from a conventional mechanical exhaust valve drive to a hydraulically driven exhaust valve in two-stroke crosshead engines. The mechanically driven exhaust valves, driven from the camshaft with a cam lobe, guide pulley, push rod and rocker waned and never returned. Today, exhaust valves in two-stroke crosshead engines built by the three remaining engine manufacturers, MAN–B&W, Wärtsilä (Sulzer) and Mitsubishi are hydraulically operated. Closing of the valve by means of a heavy valve (pressure) spring has also disappeared. Presently, the valve is swiftly closed by an air piston.

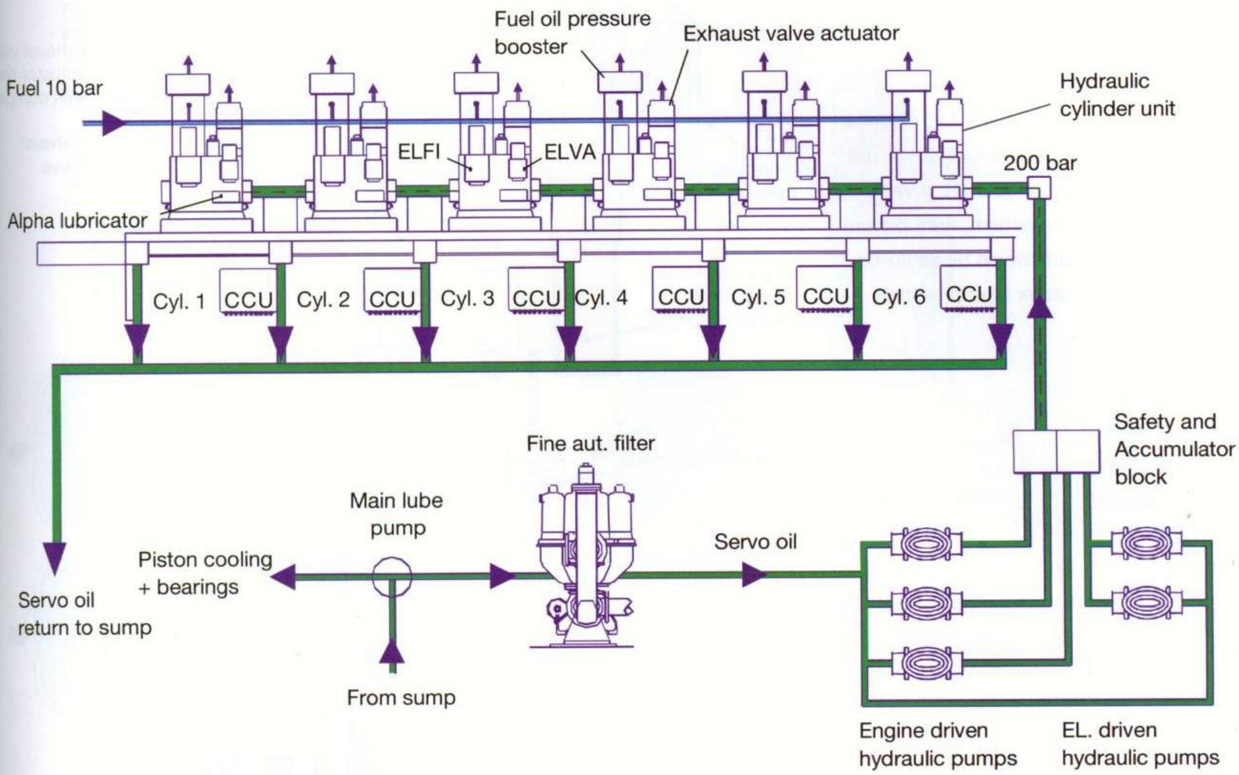
There are two techniques

1 A hydraulic pump is mounted on the camshaft. It opens the valve using lubricating oil pressure obtained from the lubricating oil circulation system. These engines have common high-pressure fuel pumps which are driven by the camshaft.
2 The latest common-rail fuel systems have no camshaft, but a fuel/lubricating oil unit placed next to the crankshaft, from which it is driven. The part of the unit containing the lubricating oil forces the lubricating oil to the exhaust valves under great pressure. These valves can be opened and closed very precisely and valve settings can be easily adjusted. In this way the valve movement can be optimised, depending on, for instance engine load, RPM and type of fuel.



The hydraulic drive for the exhaust valves in a MAN–B&W 50 MC two-stroke crosshead engine.

1 *high-pressure lubricating-oil line*
2 *servo piston for the exhaust valve*
3 *exhaust-valve casing*
4 *cylinder head*



▲
The high-pressure lubricating-oil system for fuel injection and the exhaust-valve operation of MAN–B&W, ME (E = 'electronic') two-stroke crosshead engines.

Operation: The main lubricating-oil pumps supply lubricating oil for the piston cooling, the bearings and the other parts that require lubrication; they also feed the hydraulic system through a self-cleaning fine-filter. There are two kinds of hydraulic pumps:

1 engine-driven pumps during normal operation and:

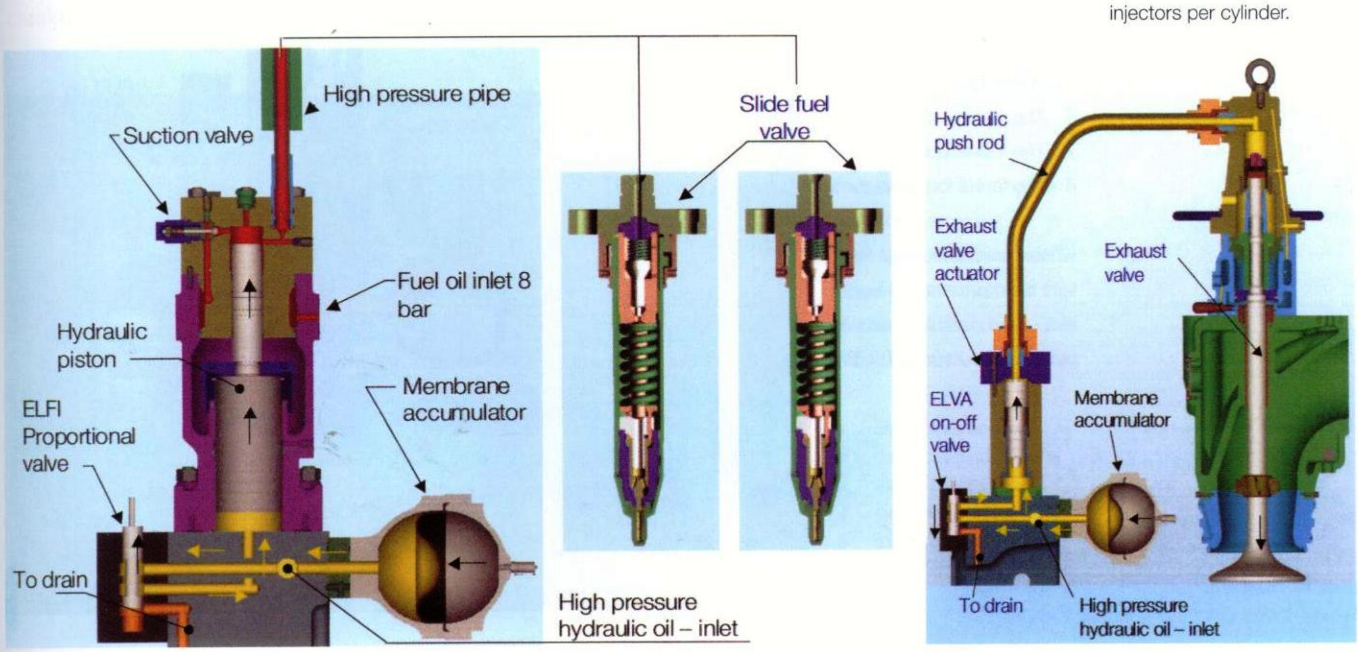
2 electrically driven pumps when the engine has stopped, has not yet started, or is non-operational.

The hydraulic pumps, via a distribution block with safety valves(to prevent an excessive build-up of the lubricating-oil pressure), supply the fuel pump hydraulic drive unit with servo-oil at a pressure of 200 bar and the hydraulic pumps for the exhaust-valve movement.

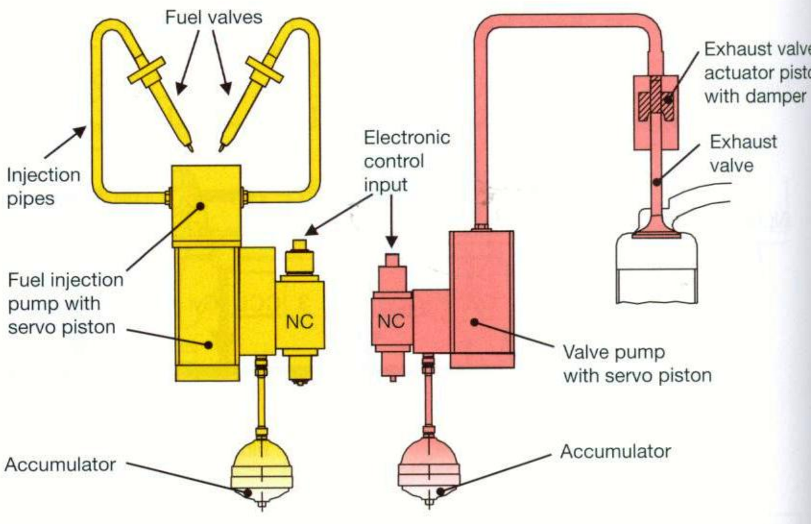
ELFI: Electronic Fuel Injection control valve
ELVA: Electronic Exhaust Valve Actuator
CCU: Cylinder Control Unit

▼
With large cylinder diameters in the MAN–B&W ME engines, both the fuel pump, left picture, and the valve movement, right picture are driven by servo-oil under high pressure.

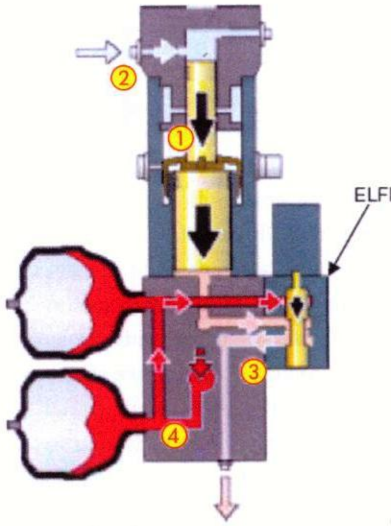
Central picture: both fuel injectors per cylinder.



▶
With large cylinder diameter in the MAN–B&W ME engines, both the fuel pump, left picture, and the valve movement, right picture are driven by servo-oil under high pressure.



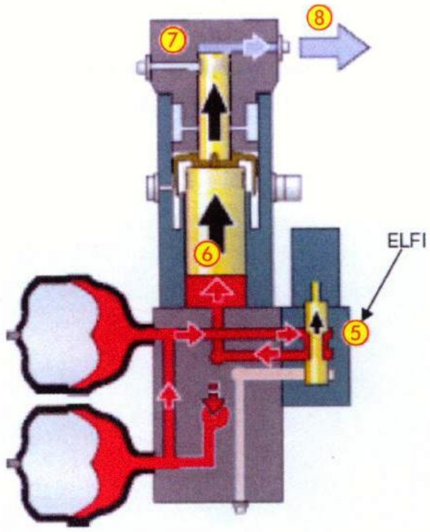
▶
The operation of the high-pressure fuel pump.
The intake/inlet stroke.



1 The plunger moves down to the bottom position and because the 'ELVI' valve moves downwards, the hydraulic oil- pressure underneath the piston which moves the plunger, is bled off.
2 The hydraulic oil leaves the hydraulic-oil chamber via valve 3
3 Fuel oil flows to the top of the fuel-pump plunger.
4 'ELVI' valve is opened and the hydraulic oil discharged.

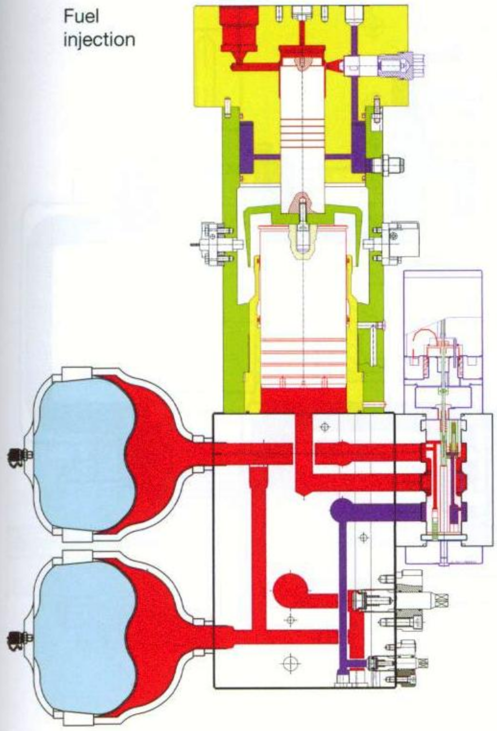
The accumulators are supplied by the hydraulic-oil pumps.

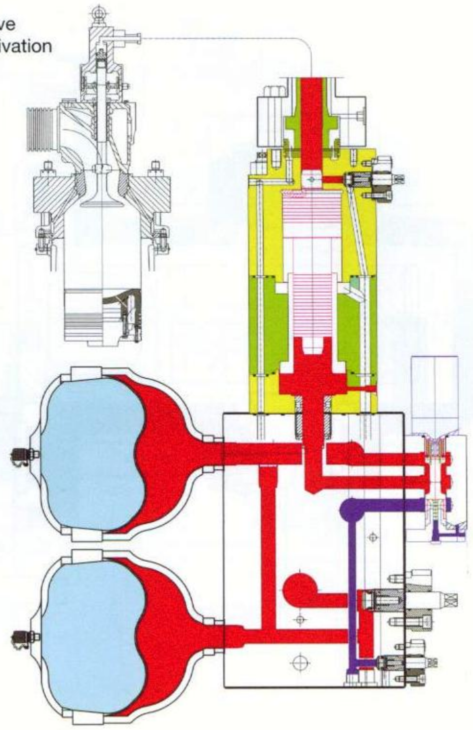
▶
The operation of the high-pressure fuel pump.
The compression stroke.



5 The ELVI-valve is closed and the hydraulic-oil pressure moves the piston and consequently the fuel plunger upwards.
6 The piston moves upwards.
7 The fuel-supply valve is closed.
8 The fuel is forced to the injectors under high pressure.

white: non-pressurised fuel
light blue: pressurised fuel
red: pressurised hydraulic oil
pink: non-pressurised hydraulic oil





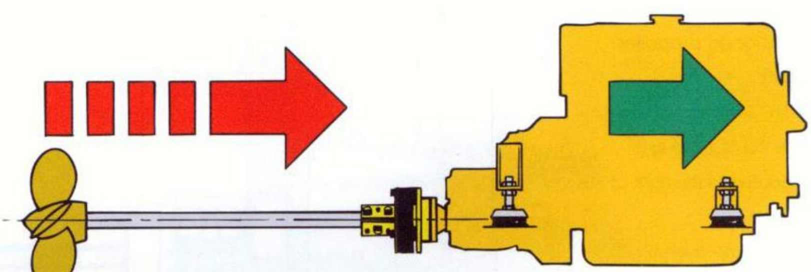
◄
The hydraulic drive for both the high-pressure fuel pump and the exhaust valve.

Red: hydraulic-oil supply, pressure 200 bar
blue: hydraulic-oil return
red: above the plunger
left: pressurised fuel to the injector
right: pressurised hydraulic oil for the exhaust-valve piston

13.4 Thrust blocks and thrust bearings

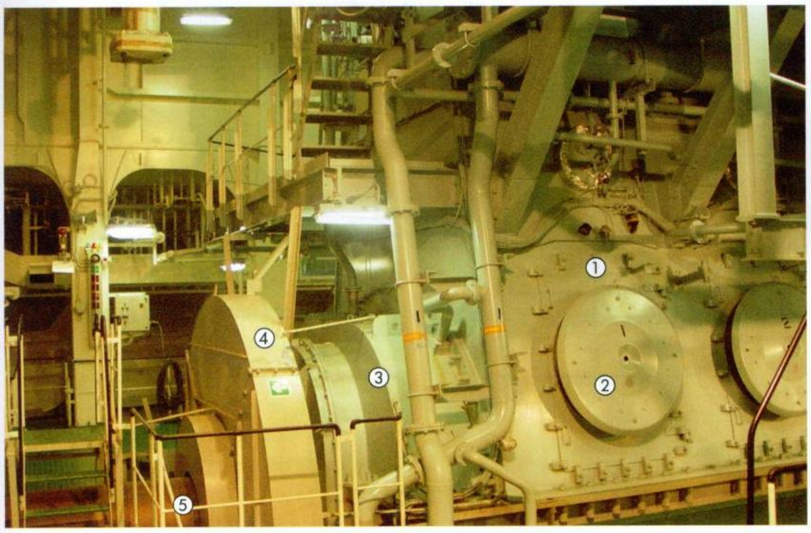
In ship propulsion, propeller rotation generates the propulsion force on the water behind the ship. This thrust is transferred to the propeller shaft which is transmitted to the ship foundation by a thrust bearing.

In large two-stroke crosshead engines the propeller shaft and the transmission shafts are directly connected to the diesel engine. The thrust bearing is placed immediately behind the engine. This ensures that no axial forces can be exerted on the crankshaft.



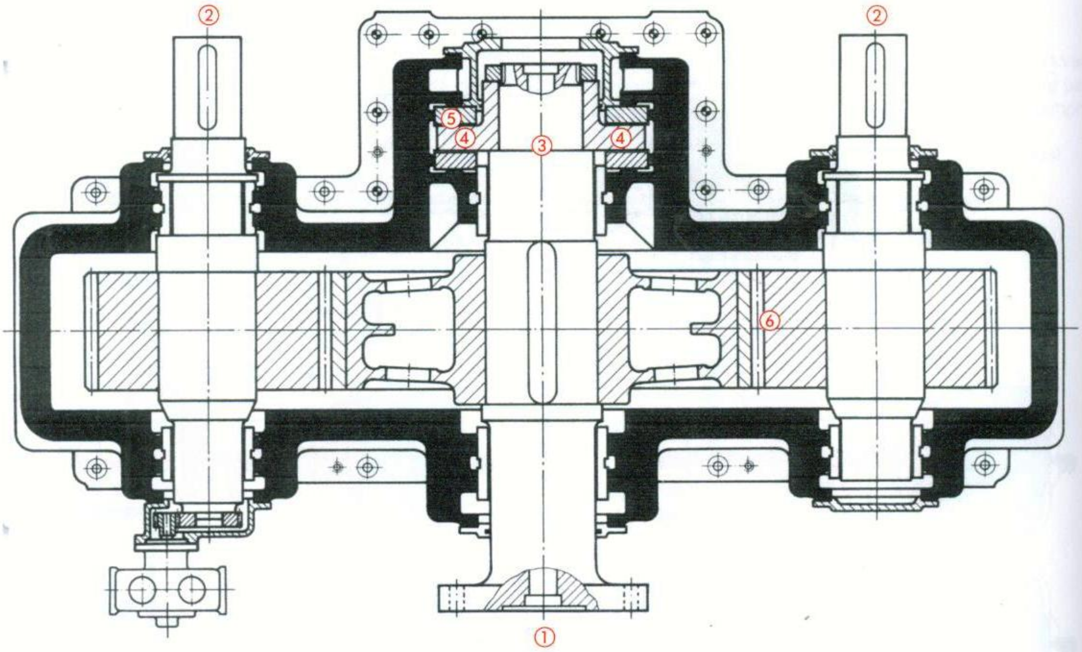
▲
The propeller thrust, the red arrow, is extended in this flexibly arranged yacht engine.

The thrust block in the reduction gearing, mounted on the left of the engine, absorbs this force.



◄
The thrust block mounted on the engine for a two-stroke low-speed crosshead engine, manufacturer Wärtsilä Sulzer, type RT fLEX 96 C.

1 *crankcase engine*
2 *crankcase relief valve*
3 *thrust block*
4 *fly/turning wheel*
5 *intermediate shaft*

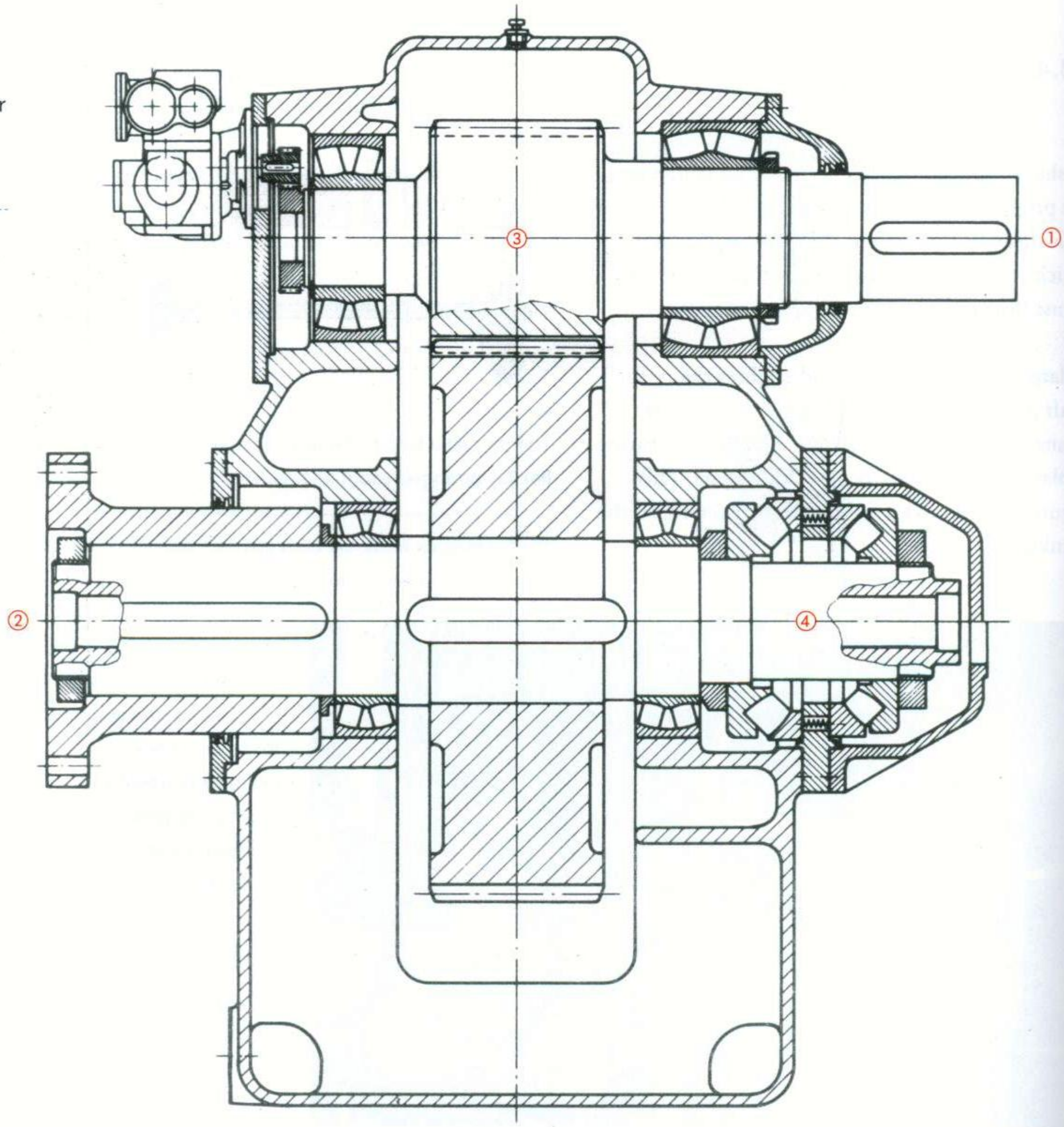


►

A Renk reduction gearing for driving a propeller with two engines.

The thrust block, located at the end of the intermediate shaft, absorbs the propeller thrust. The thrust block has been adapted for bi-directional thrust: small thrust blocks have been fitted on both sides of the thrust-block flange.

1 *propeller shaft*
2 *engine shaft*
3 *thrust block*
4 *thrust-block collar*
5 *small thrust blocks*
6 *reduction gearing*

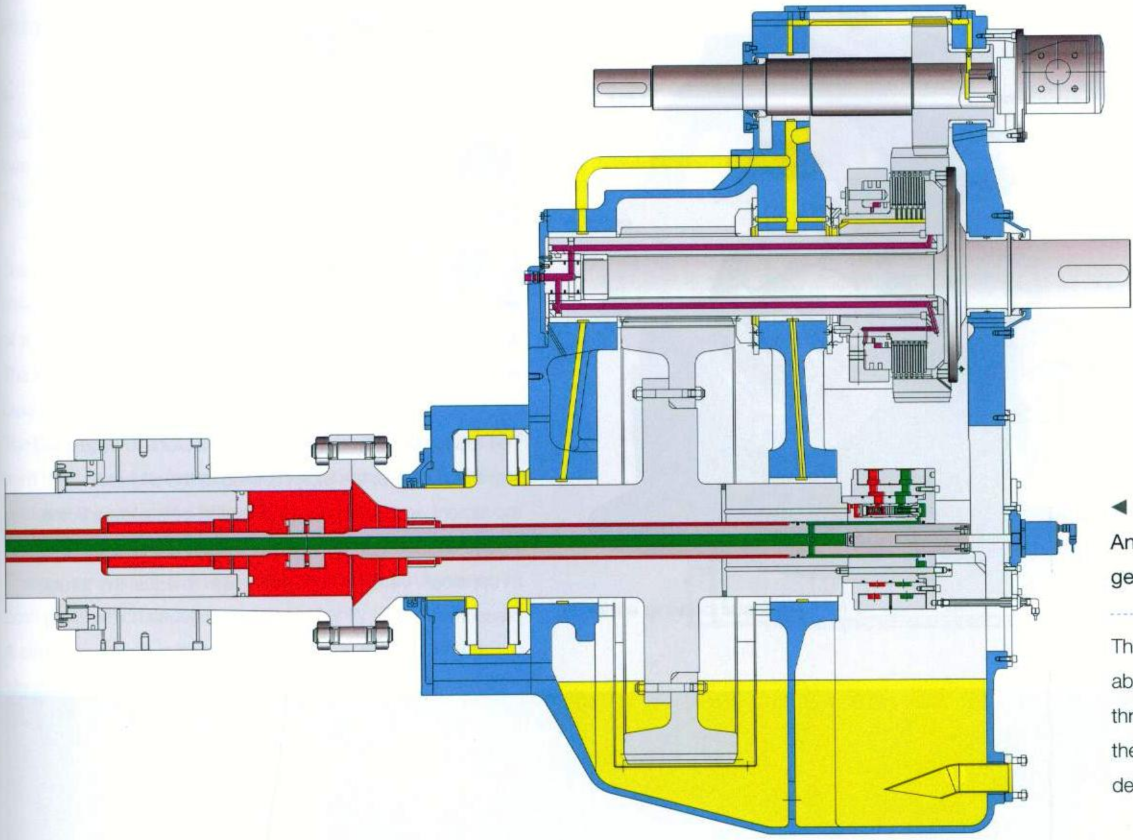


►

Reduction gearing for a diesel engine with a lower positioned propeller shaft.

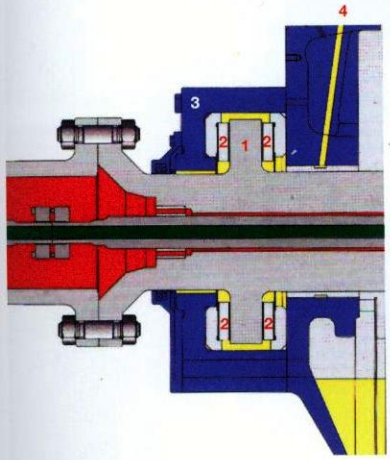
The thrust bearing is mounted to the right of the picture.

1 *engine-drive shaft*
2 *propeller shaft*
3 *reduction gearing*
4 *thrust bearing*



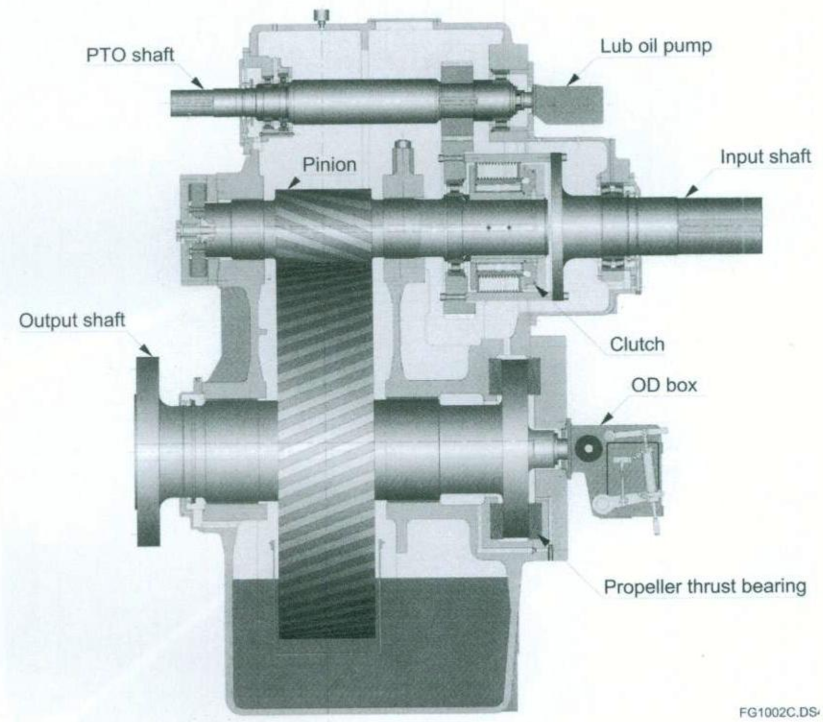
An Alpha reduction gearing of MAN–B&W.

The thrust block for absorption of the propeller thrust is situated to the left in the reduction gearing. See detail drawing.



Detail drawing of the thrust block.

1 *thrust-block flange*
2 *small thrust blocks*
3 *thrust-block casing*
4 *lubricating-oil supply from the lubricating-pump gear box*



FG1002C.DS

A gear box with a thrust block in the right side of the reduction gearing.

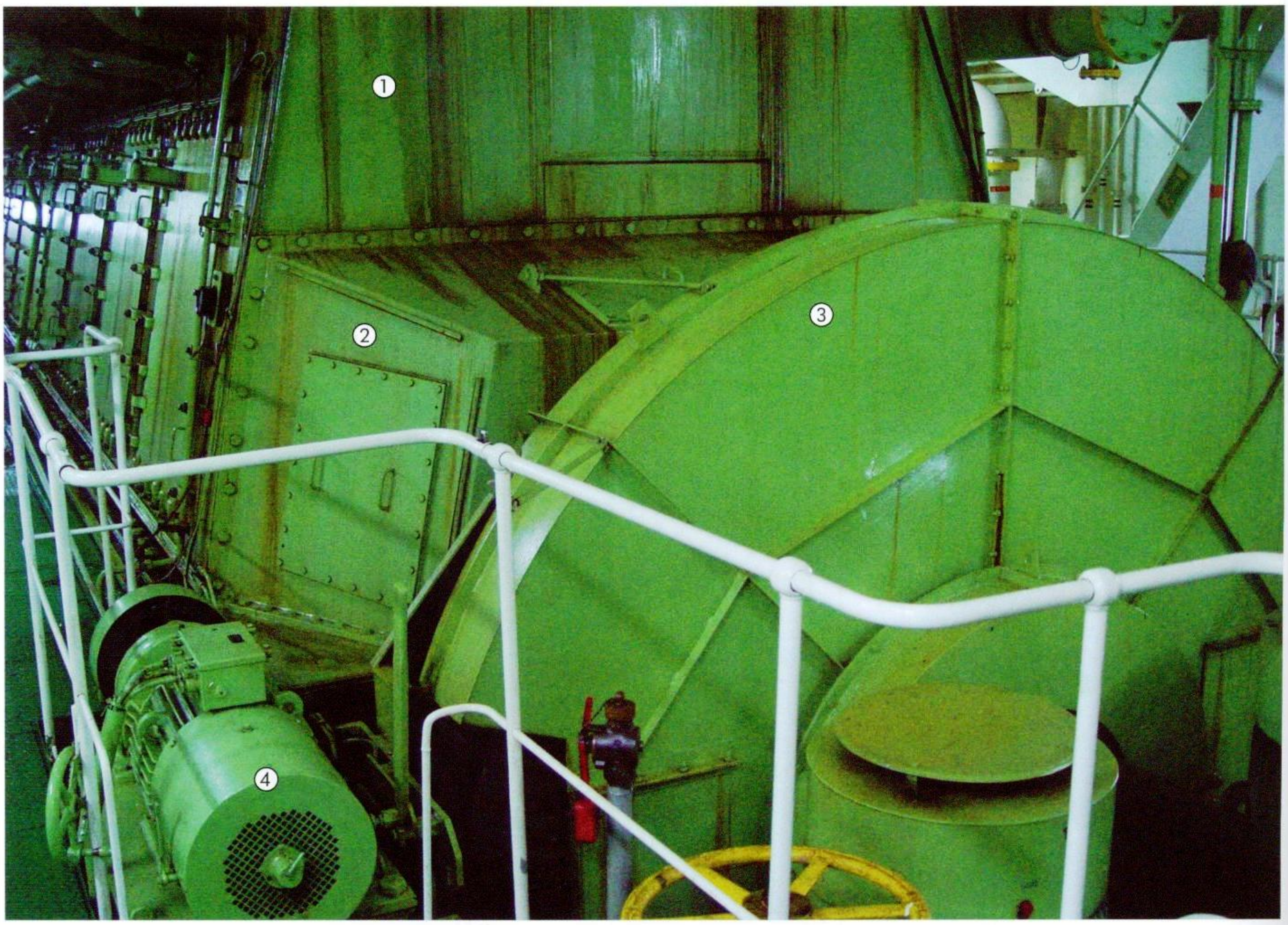
The engine shaft and the generator shaft are equipped with a thrust block/thrust bearing to maintain the shaft's position.

A Renk thrust block, normally placed in the shaft between the propeller and the engine. The circular discs are the thrust blocks.

Pedestal bearing series H.

A thrust block without top cover.

The round notch on the circular thrust block is positioned slightly off centre; the block is tilted at the slightest movement and builds up a lubricating-oil pressure in the wedge-shaped gap between the thrust-block collar and the thrust block, which is sufficiently high to equal the thrust pressure.



The thrust block in a large two-stroke crosshead engine of Wärtsilä, type RTA 84 C.

1 *crankcase*
2 *thrust-block location*
3 *fly/turning wheel*
4 *turning engine*

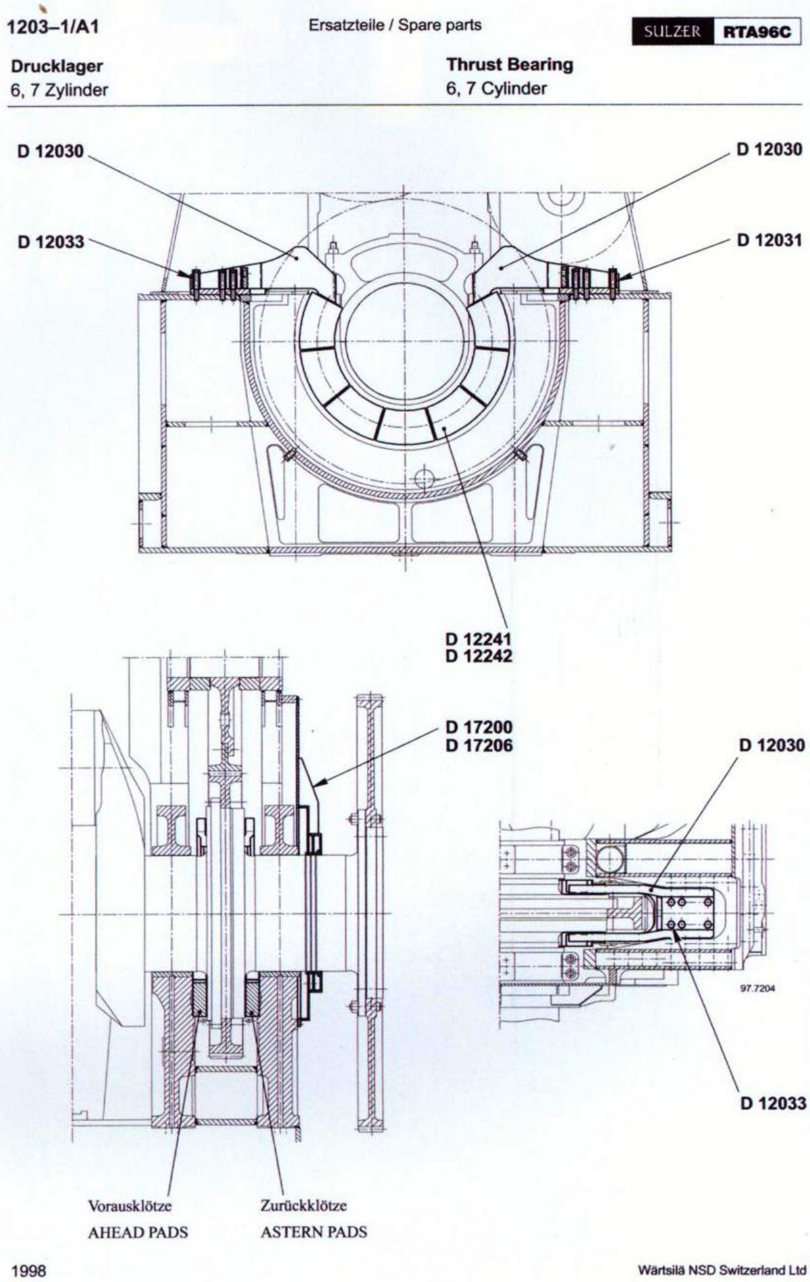
In four-stroke trunk piston engines which are normally provided with a propeller shaft driven reduction gear, the thrust bearings have been integrated in the gear box.

Principles of the thrust bearing

▶

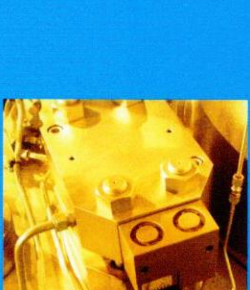
The thrust block of a Wärtsilä Sulzer RTA 96C two-stroke crosshead engine in the engine frame. The thrust block consists of a shaft with a large collar.

The casing consists of a number of tilting pads which in a directly reversible diesel engine have been fitted to both sides of the collar.
The pads have an elevation, the so-called pivot; this is placed off-centre of the pad.
The front of the pad has been rounded so the lubricating oil from the collar, which runs partially in lubricating oil, is carried between the collar and the pad.
The pad is pressed on to the lubricating film by the pivot. There is a pressure build-up in the lubricating film between both parts which causes the pad to slightly tilt.
A stable condition is then achieved as the product of the lubricating-oil forces are balanced with the force exerted on the pivot by the thrust force.
With a fluctuating thrust the equilibrium is constantly recreated which allows the thrust to be transmitted to the foundation of the ship via the thrust collar, the lubricating-oil film, the pads and the thrust-block casing.
The angle of tilt is so small that it can not be seen with the naked eye (1: 1000).
The lubricating film in the system is not interrupted, thus preventing metal-on-metal contact. Therefore, friction is negligible.
A number of tilting pads have been placed on the other side of the collar for engine reversal. Obviously these tilt in the other direction due to the changed direction of rotation.
The top of the loose thrust blocks often lack tilting pads, so the total load on the bearing surface lies below the centre line. Therefore, the bending moment on the bedplate is smaller.
The large loose thrust blocks normally have a separate lubricating-oil system. If the thrust block is integrated in the engine or part of a reduction gearing, the lubricating-oil system is often used for the engine or the reduction gearing.

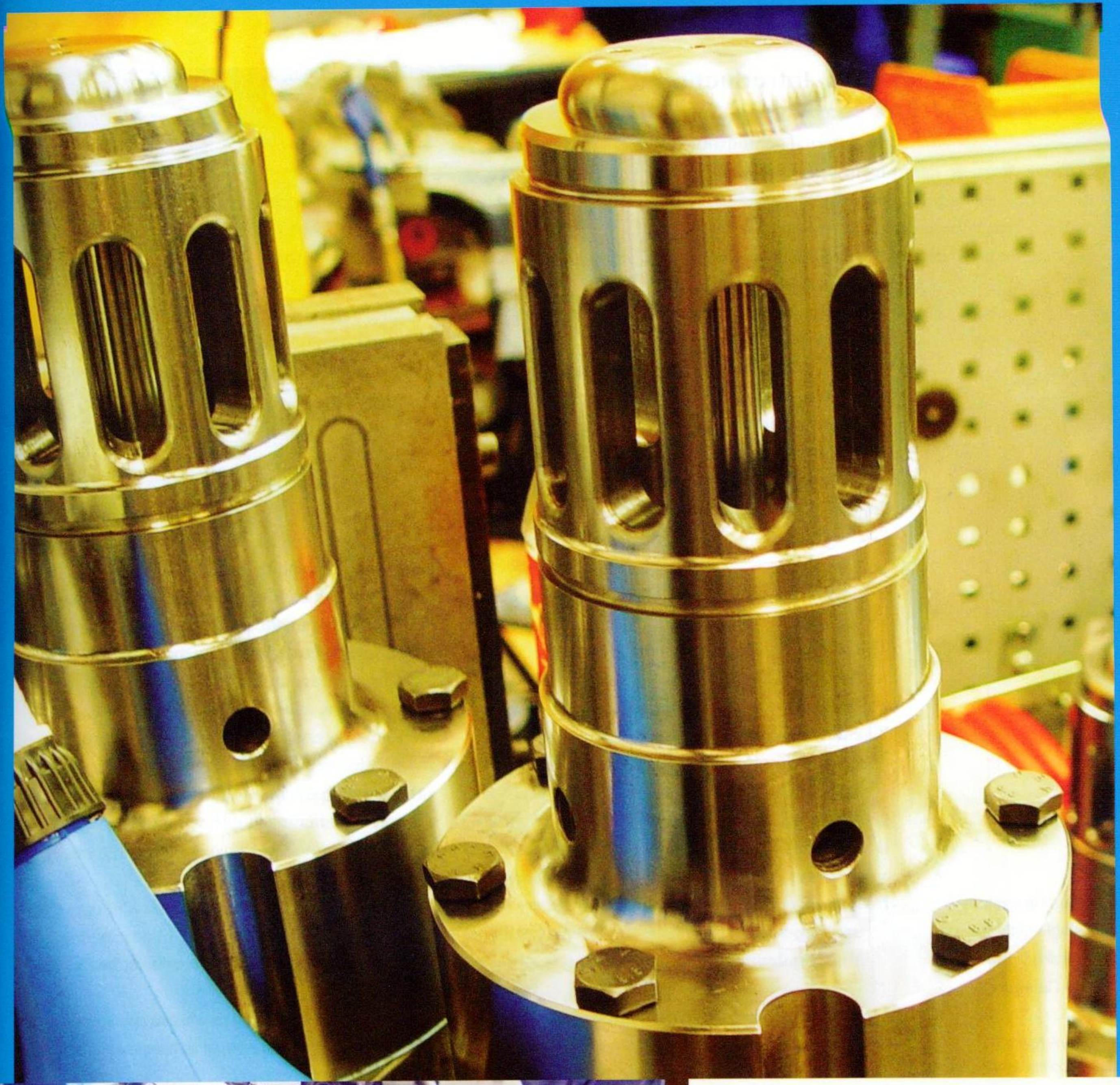


>H 14

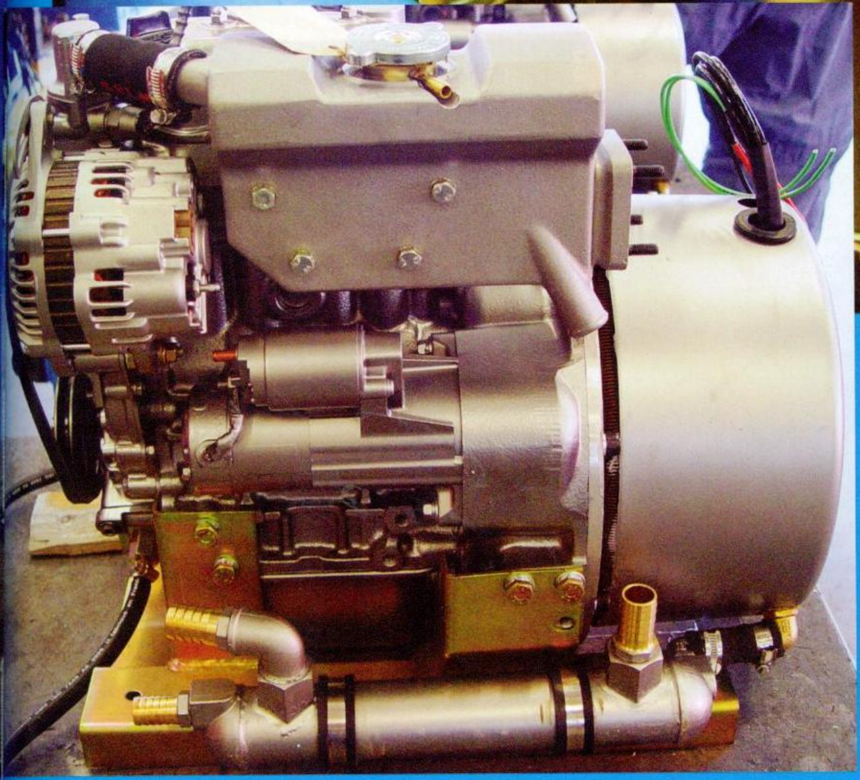
Starting systems of diesel engines







The only way in which a stationary engine can operate is by supplying power from an external source. This can be realised in various ways, such as manually, with batteries or, for example, with starting air.
Shown here is a pair of starting air valves for a large two-stroke crosshead engine. The starting air has a pressure of 25 to 30 bars.



Many category I and II engines make use of electric starters to start the engine, for example, this diesel generator from Mastervolt.
Above left, the dynamo for charging the storage batteries; in the middle, the starter/motor.

14.1 Introduction

An internal combustion engine cannot be started from a standstill mode. There is no process and consequently there are no working substances (combustion gases) present.
Therefore the engine has to be started with an external energy source. The energy supplied during starting the engine is used by the piston to compress air so fuel can be injected immediately before it reaches top dead centre position and the combustion process can commence. The compression stroke requires a considerable amount of energy. Small single- or multiple cylinder engines can be started manually. Larger engines, however, require far more energy.

14.2 Starting methods

The following starting systems are used:

A Manually combined with a decompression lever.
B Starting with a small energy source, such as a spring mechanism or a hydraulic pressure vessel.
C Electrical start with a battery.
D Starting with starting rotors, operated by compressed air, mounted on the fly-wheel.
E Air start with compressed air fed to the starting-air valve which is mounted directly on the cylinder.

Methods A, B, and C are only applied in relatively small four-stroke engines in category I and II. Method D is generally used in larger high-speed-four stroke engines. Method E is often applied in large four-stroke medium-speed engines and always in two-stroke crosshead engines.

14.2.1 Method A: Manual start in combination with a decompression lever

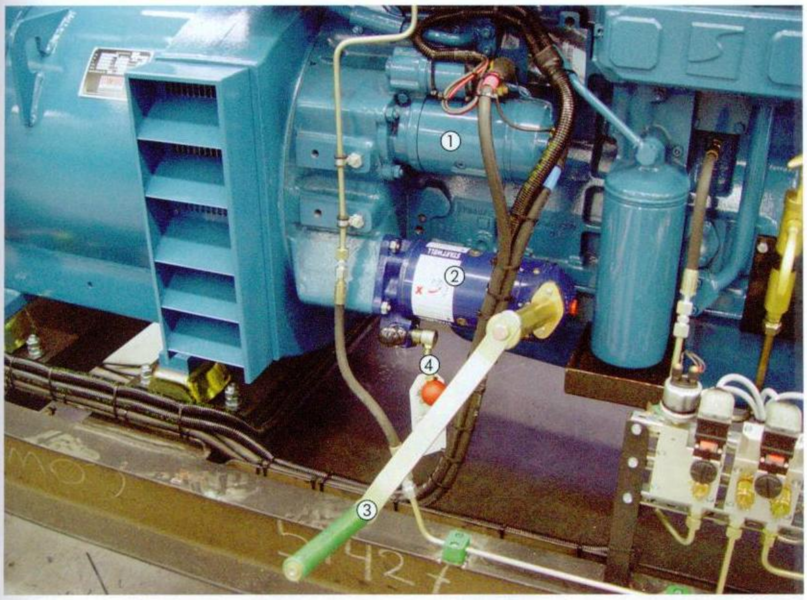
Turning a crank handle achieves a certain number of revolutions in an engine with an opened exhaust valve; due to this the piston has no compression, so the crankshaft can rapidly attain an acceptable RPM. The energy produced is stored in the fly-wheel. By operating the decompression lever which closes the exhaust valve the piston starts its compression stroke and the injected fuel ignites. The combustion stroke will switch on the engine and the engine runs.
Obviously, the hand-operated starting energy is limited, so this method is only suitable for small engines, with few cylinders.
The 'decompression lever' ensures that the exhaust valve remains open during hand cranking. There are also larger engines, which are, for instance, started electrically during which time the exhaust valves remain open automatically and are closed as soon as the crankshaft has rotated several times.

▼
The starting system of a single cylinder four-stroke Lister diesel engine Category I.

1 *crank handle*
2 *decompression lever*

▼
The decompression lever on the cylinder head.

When starting the flywheel by means of a decompression lever, the exhaust valve is opened so compression cannot take place in the cylinder.



An emergency generator built by Sandfirden Technics.

Regulations stipulate that emergency generators must be operated by two independent starting systems, an electric starting system and a second independent starting system. Shown a second starting-system door using a windable-spring starter.

1 *electric starting-air motor using a battery*
2 *spring starter*
3 *spring-winder handle*
4 *red starting button*

14.2.2 Method B: Starting with a small energy source, such as a spring mechanism or a hydraulic pressure vessel

This method is often applied in life boats as a manual start may not always be possible in poor weather conditions. The starting energy is stored in a heavy spring or in a high-pressure lubricating-oil system stored in a starting reservoir.

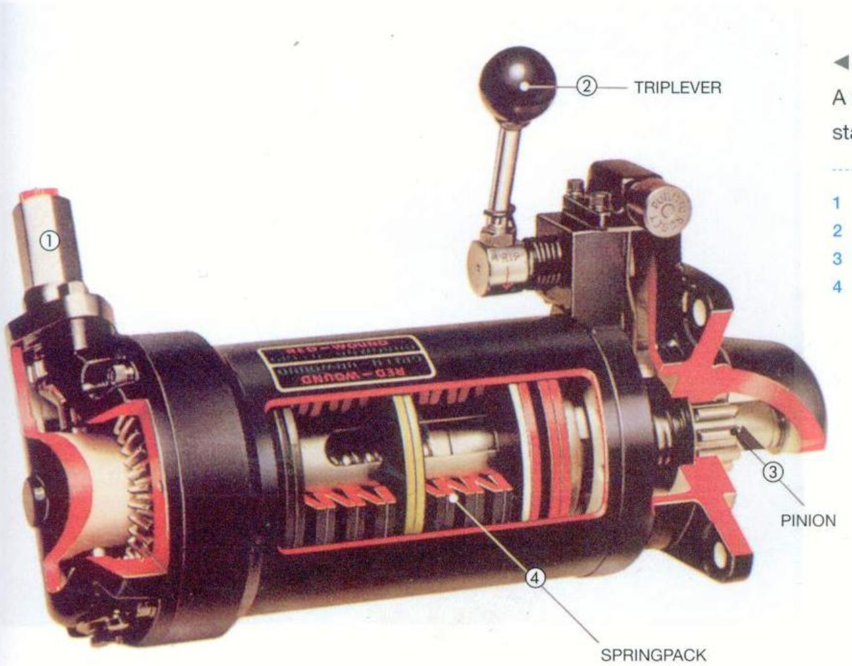
As the engine is started this energy is released and the engine starts. In this system, a decompression lever is also often used.

14.2.3 Method C: Electrical start with a battery

This method is often applied in high-speed four-stroke engines. Engines with a shaft power of over 1000 kW are often started in this manner. The size of the battery then increases accordingly.

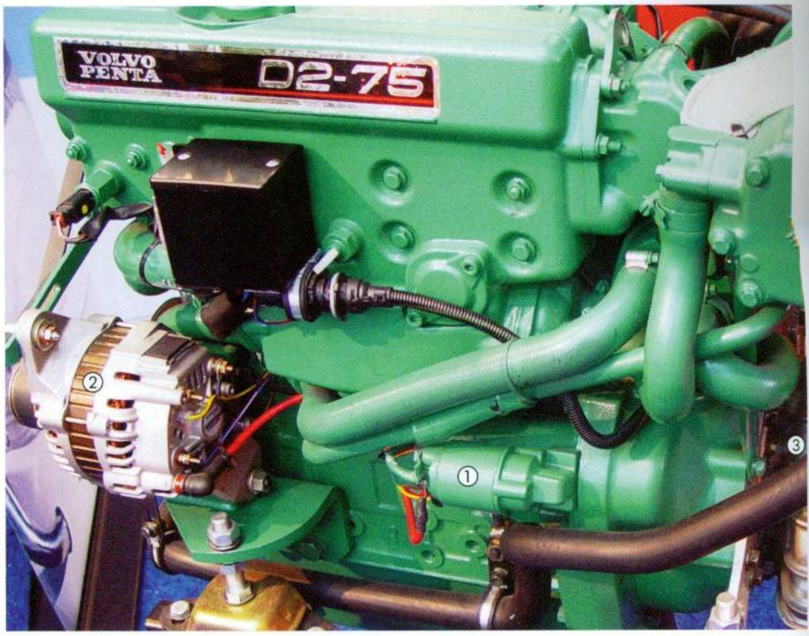
Description

The battery consists of a large number of cells which generate a voltage of 12 or 24 volt. At the starting command by a key or a button, the rotor of the starter rotor is shifted and the cog wheel on the starter shaft engages the gear ring on the fly-wheel.
After this, the starter switches on automatically and the engine begins to run. When the engine itself operates at a certain RPM, the RPM will



A different type of spring starter.

1 *spring-winder handle*
2 *starting lever*
3 *bendix, pinion*
4 *spring packet*



▶
The electric starting system on a Volvo Penta yacht engine Category I.

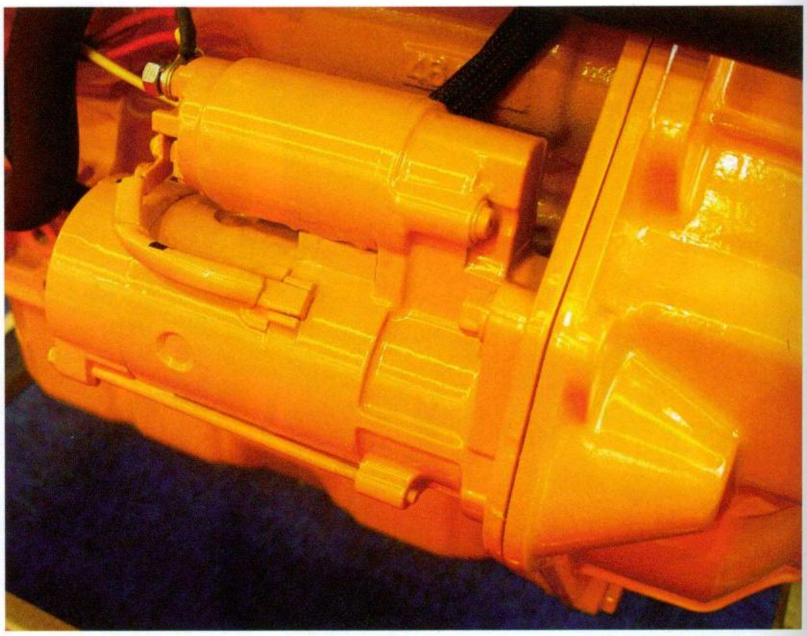
1 *starting motor*
2 *generator*
3 *reduction- gearing with shaft*

increase and when it reaches a certain rate the starter is switched off by resetting the rotor motion to null. The engine is running. The battery is directly charged by a diesel-engine driven generator. In larger installations this can also be performed by a charging system using mains electricity.

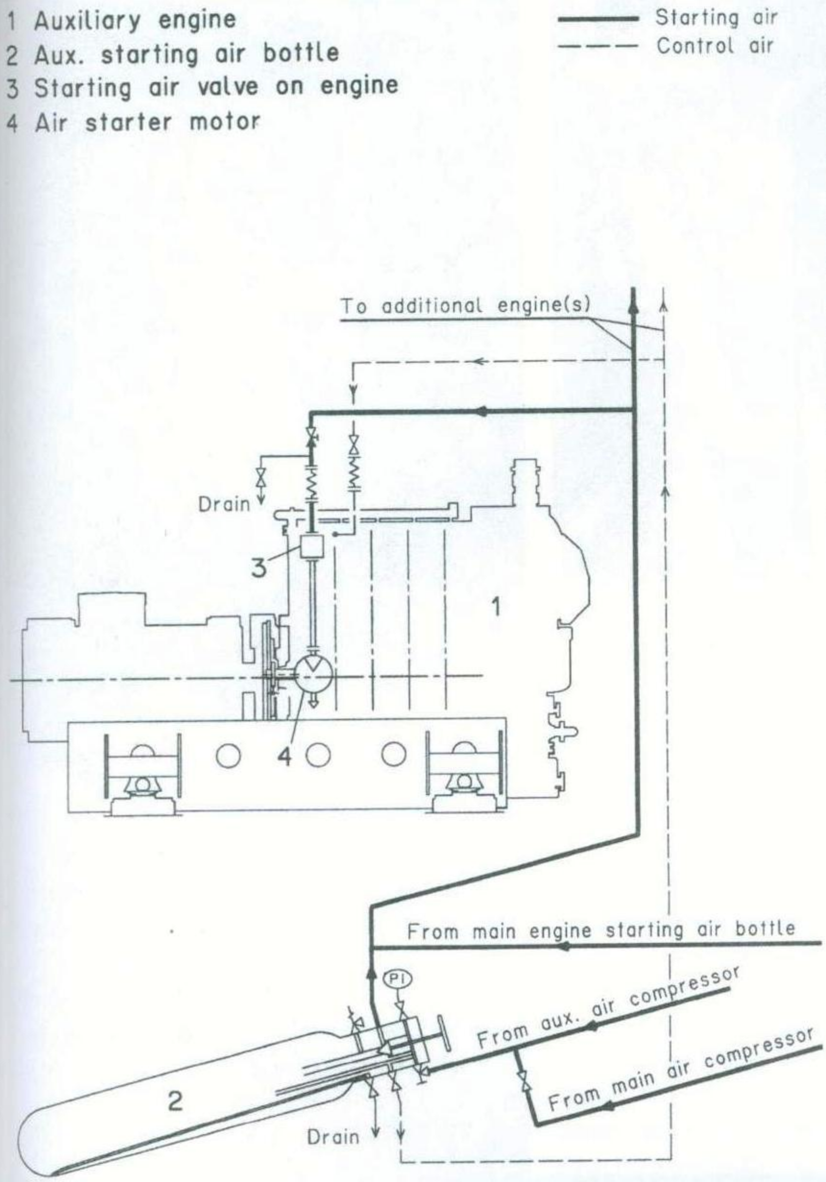
14.2.4 Method D: Starting with starting rotors mounted on the fly wheel, which are operated by compressed air

This method is increasingly used in the larger high-speed and medium-speed four-stroke diesel engines which were previously started with an extremely large battery (high-speed) or with starting air fed directly to the cylinder (medium-speed).
In the first instance, the expensive high maintenance batteries are no longer required, and in the latter, the construction of the cylinder head is simplified. Furthermore, while operating the engine when it is started for the second time, no relatively cold air must not enter the cylinder. This wave often causes cracking of the cylinder head where the starting valve is mounted.
Larger engines with several cylinders often have two or more rotors in place to provide sufficient power for the crankshaft. The starter rotors resemble simple gas turbines.

In order to have an adequate amount of air available during the start large starting air



▶
The electric starting motor of a 'Vetus' Deutz yacht engine Category I.



◄

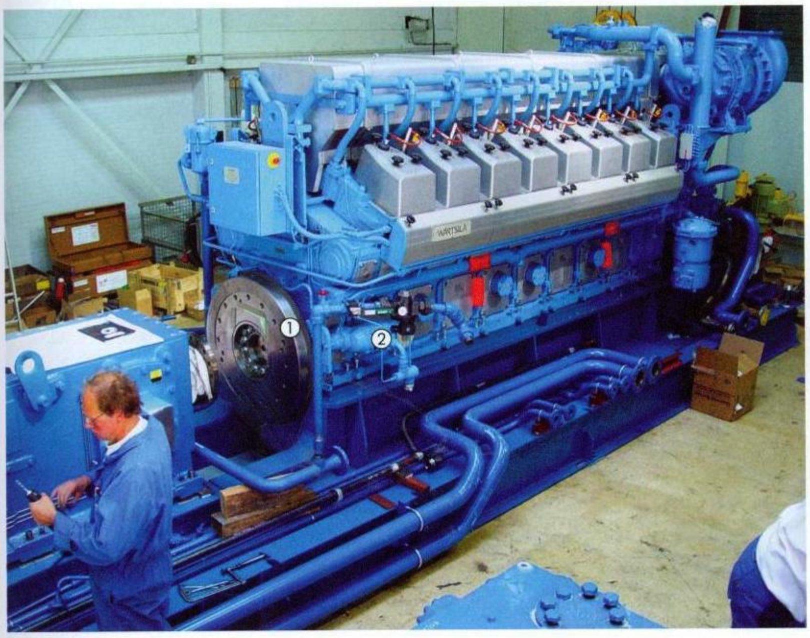
The starting-air system of a Sulzer S20 medium-speed diesel engine – Category III.

The engine is also available with an starting-air valve for every cylinder.

Description: The diesel generator set 1 is started using an air starter motor 4 that drives the flywheel. This can be done manually using starting valve 3 in the engine.

Starting-air bottle 2 is automatically kept at the correct pressure by a main compressor or an auxiliary/emergency compressor.

The manual water drain prevents the slow accumulation of condensed water vapour from the compressed air. The starting-air bottle capacity slowly decreases!

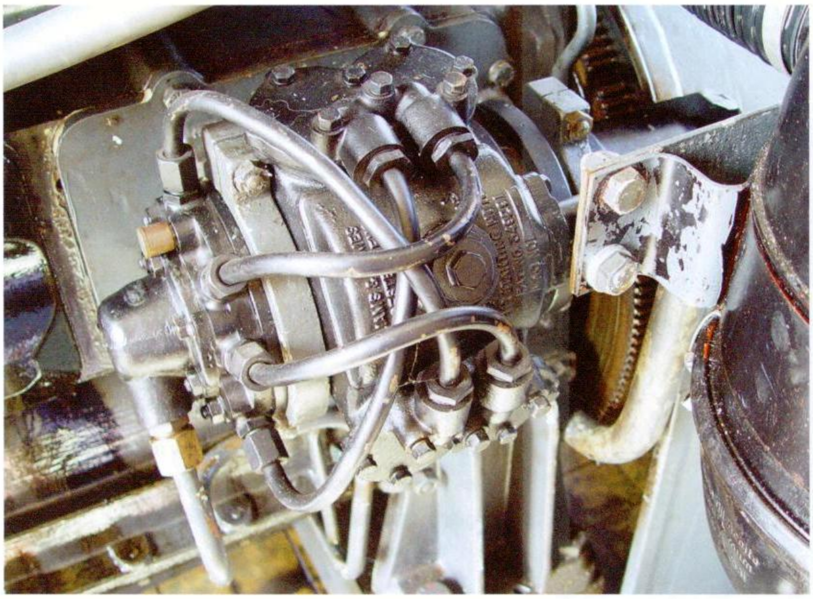


◄

The starting-air turbine on the 'flywheel' of a sixteen-cylinder Wärtsilä 25 SG gas engine.

The connection of one of the many starting-air turbines that operate the fly-wheel, is simpler than the traditional starting-air valves on every cylinder.

1 *fly/turning wheel*
2 *air starter motor*



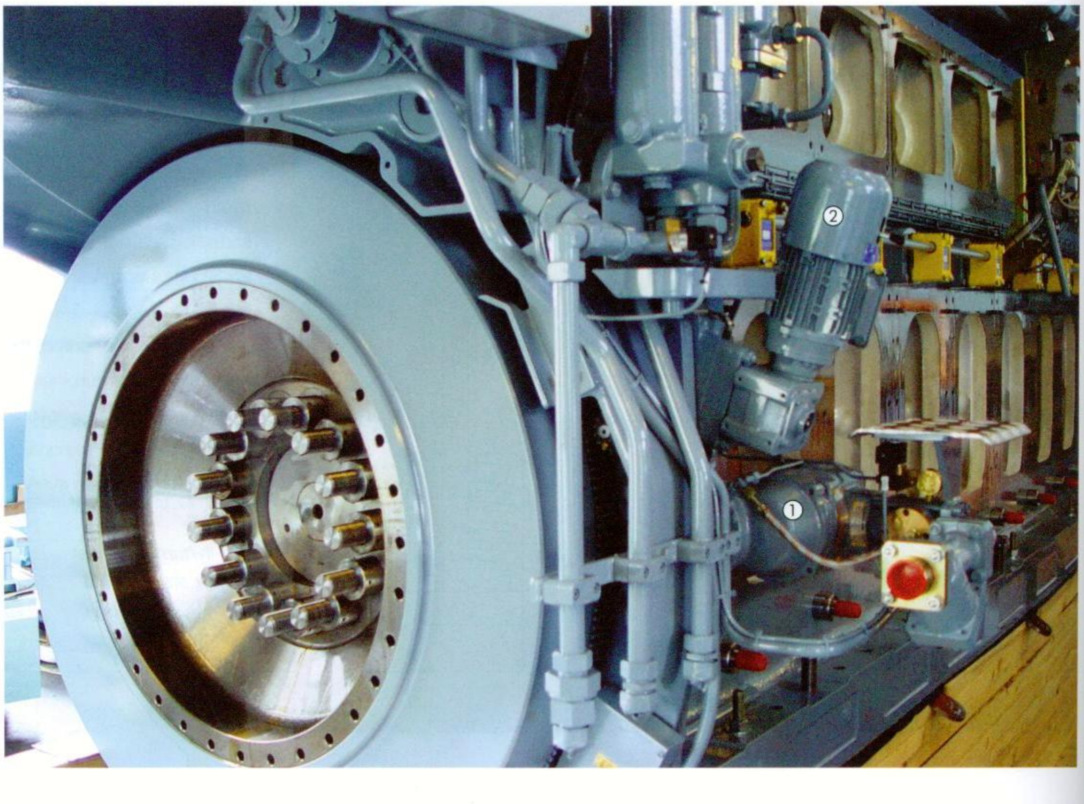
The starting-air turbine/motor on a Kromhout diesel engine. Built 1956!

A manual air stop valve for a starting-air turbine/motor on a older Kromhout diesel engine – Category I.

The lever is kept depressed until the engine starts on fuel.

reservoirs are used that are automatically kept under pressure by two starting air compressors. The pressure in the air reservoirs is approximately 30 bar. Classification Societies have laid down certain requirements for air reservoir capacities.

- **One propulsion engine, directly reversible engine:** the air reservoir capacity must be sufficient to start the engine at least twelve times ahead and/or astern in turn, without assistance from the starting air compressors.
- **Non- reversible engines;** the air reservoir capacity should be sufficient to start the engine at least six times without assistance from the starting air compressors.



The starting-air turbine/motor and the jogging motor mounted near the flywheel on a four-stroke MAN–B&W engine – Category III.

1 *starting-air turbine/motor*
2 *electric turning motor*

◄

A series of rotating starting-air turbines/motors using different air capacities.

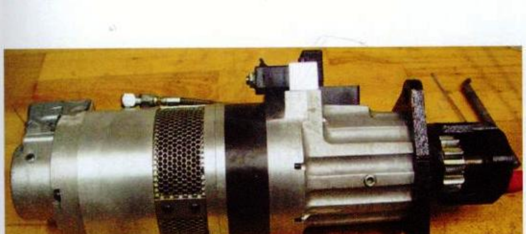
The blue section in the three cut-away starting-air turbines is the air section.

1 *rotor with small blades on the outer surface*
2 *gear box*
3 *bearing section with space for positioning the gearwheel 4*
4 *gear drive*

There are various types. Important: The air discharge of the starting motor is designed to reduce noise emissions.

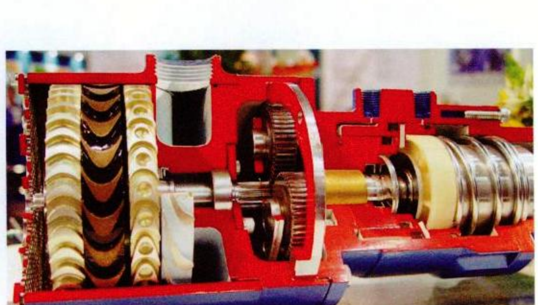
14.2.5 Method E: Air Start with compressed air fed to the starting air valve which is mounted directly on the cylinder

This system is applied in all two-stroke crosshead engines and frequently in large four-stroke medium-speed engines.
Each cylinder has an starting air valve which feeds air into the cylinder just after the T.D.C. position of the piston and puts the piston into a downwards motion.
The air start valves are opened by air start control valves. The starting air valves open when the exhaust ports- or valves are shut, as the start air would escape without exerting any force on the piston. The starting air valves remain open for a certain number of crank degrees. This is referred to as the starting arc. So, in two-stroke engines the starting air arc has a maximum of approximately 110 crank degrees and in four-stroke engines 130 to140 crank degrees. **As the combustion process of two-stroke engines is 360 crank degrees, at least four cylinders are required in order to start the engine in any piston position. Four times 110 crank degrees = 440 crank degrees.**
In four-stroke engines the combustion process is 720 crank degrees, this means that at least six cylinders are required to start the engine in any piston position.



▲

An air-starter for Yanmar diesel engines



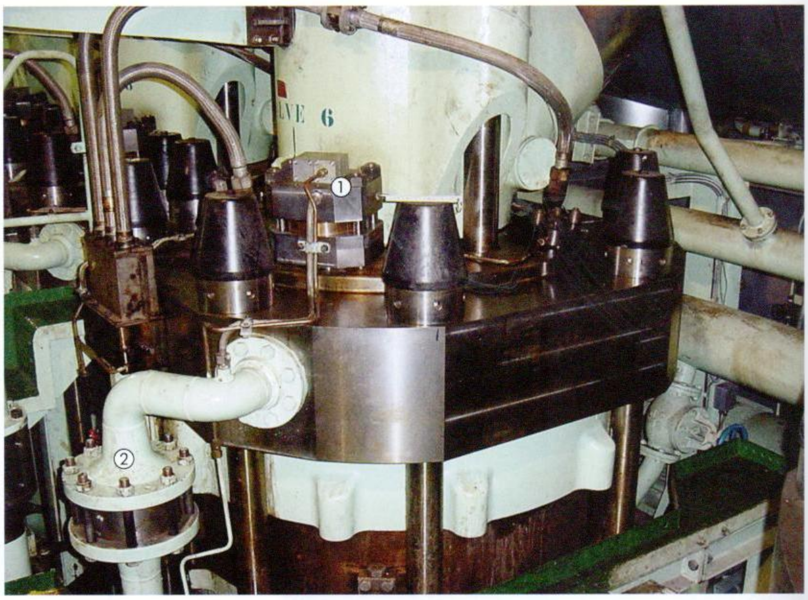
◄

A cut-away of an air-starter with rotor blades.

►

The starting-air valve mounted on the cylinder cover of a two-stroke crosshead engine of manufacturer Wärtsilä Sulzer type RTA 96-C – Category IV.

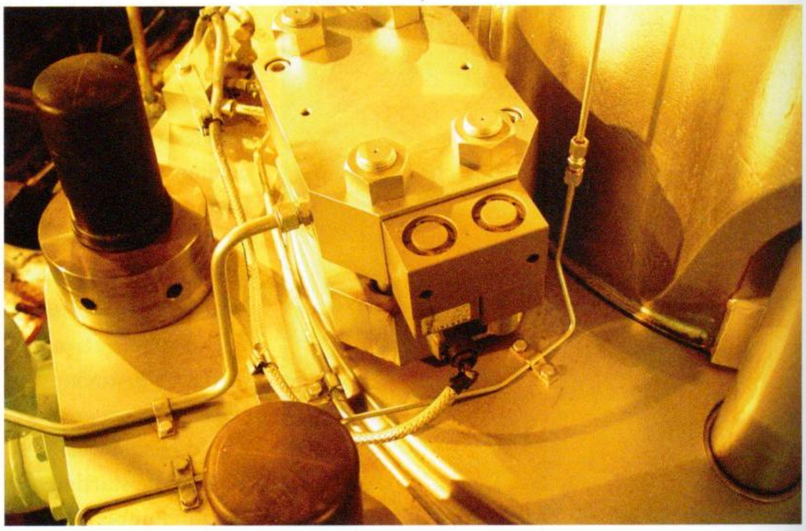
1 *starting-air valve*
2 *starting-air duct*



►

The starting-air valve of a Wärtsilä Sulzer type RT FLEX 96 C – Category IV.

This is an electronically controlled starting-air valve.



►

The water cooled two-stage high-pressure starting-air compressors for a large two-stroke crosshead engine.

They automatically keep the large air receivers under pressure.

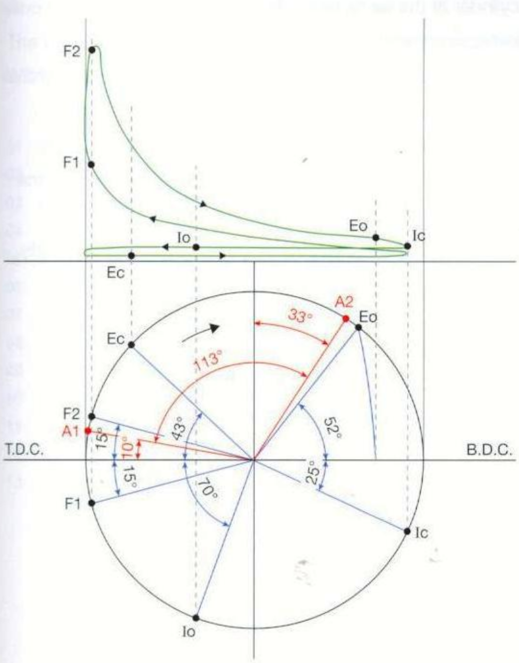
▲
The air receiver in four-stroke medium-speed diesel engines – Category III.

▲
The air receivers for a large two-stroke crosshead engine – Category IV.

The air receivers are manufactured and installed in accordance with regulations. Safety is a priority when working on these systems at pressures of 30 bar!

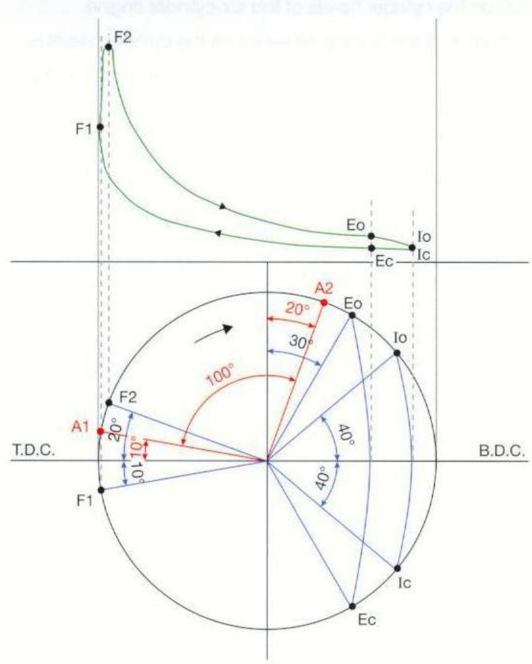
▼
The starting curve of a four-stroke medium-speed diesel engine – Category III.

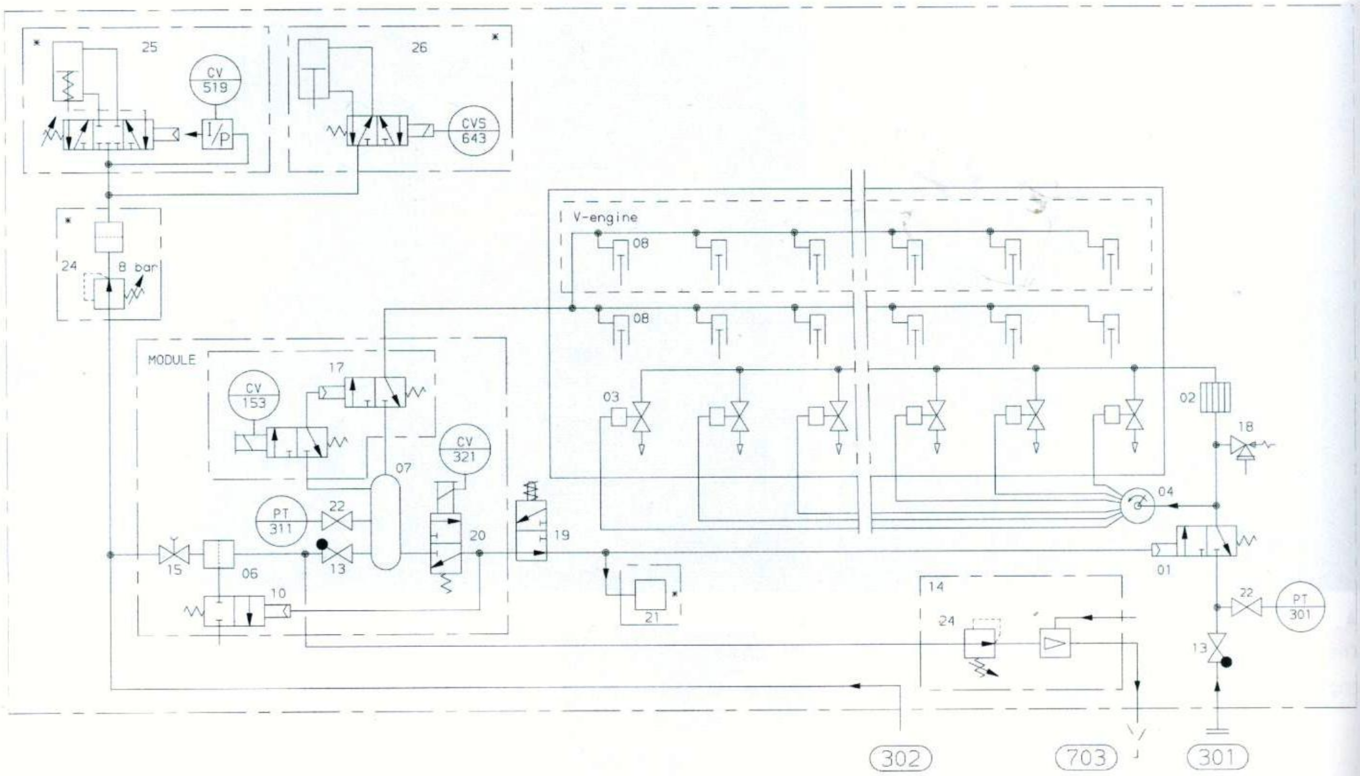
The starting curve is in this case 113 crank degrees from the start at A1 to the end of the starting air at A2.



▼
The starting curve of a two-stroke low-speed crosshead engine – Category IV.

The starting curve is 100 crank degrees, beginning at the start at A1 and ending at A2.





▲

The starting-air system of a four-stroke Wärtsilä 38 diesel engine – Category III.

Description: a starting-air valve is mounted on every cylinder. In an in-line engine on all the cylinders and in a V-engine on a bank, therefore on half the number of cylinders. At 301 the starting air under a pressure of maximum 30 bar enters the system via non-return valve 13, main starting-air valve 01, flame arrestors 02 and the six starting-air valves 003 on the cylinder heads of this six-cylinder engine. Activation of the starting-air valves on the cylinder heads is controlled by the starting-air distributor 04 mounted on the engine camshaft ensuring that the starting-air valves only open during the combustion stroke. If the electric turning motor has engaged the flywheel, safety valve 19 ensures no starting air enters the system via the main starting-air valve 01.

Via connection 302 the pneumatic control air is supplied to the control unit, to operate the waste-gate 25 and the bypass valve 28.

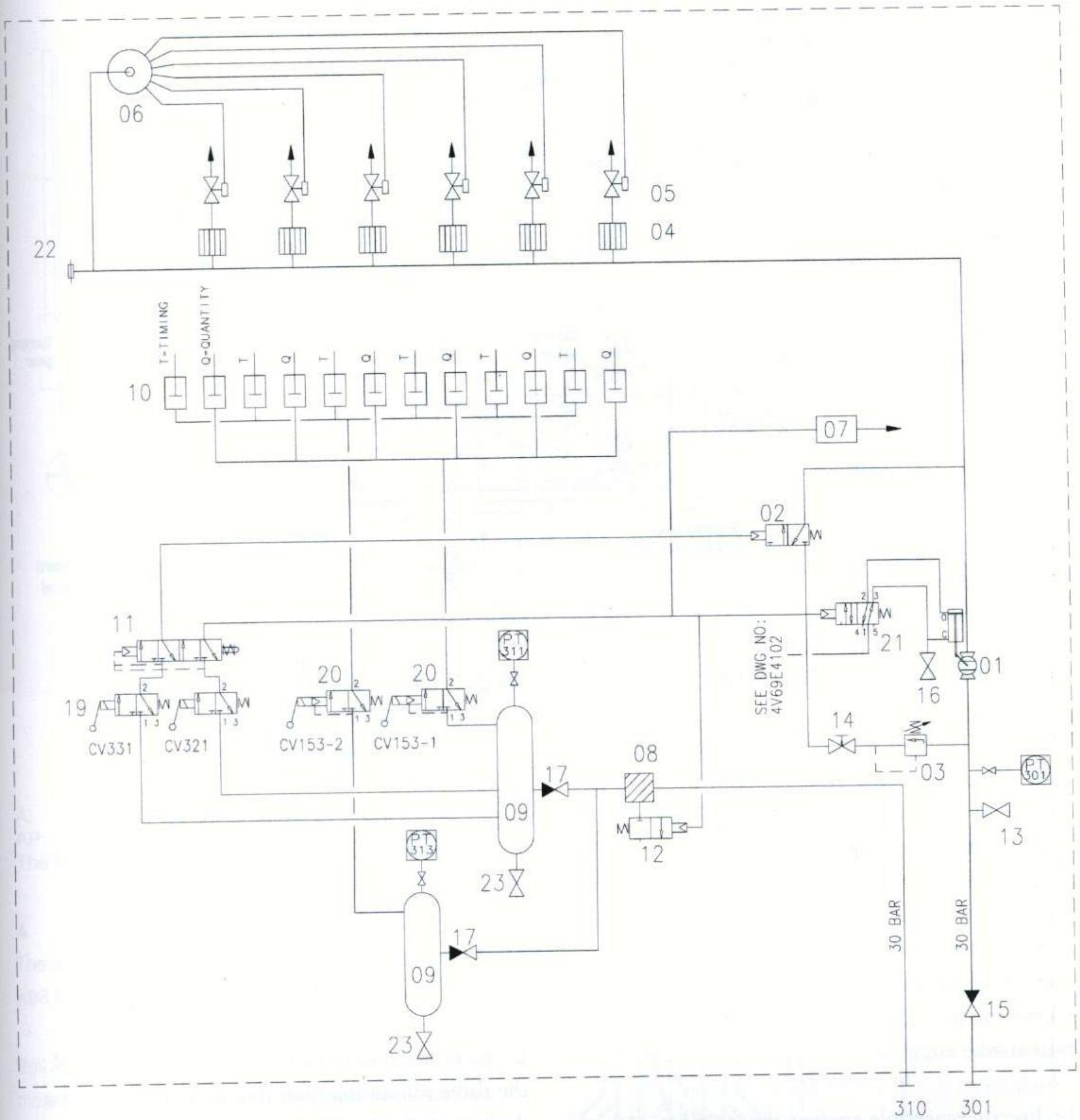
Control unit: This comprises a ball valve 15, an air filter 06, an automatic drain 10, a non-return valve 13, a buffer header 07, a stop cylinder for the high-pressure fuel pump and a starting-air valve 20.

On every high-pressure fuel pump, a pneumatic stop cylinder is mounted ensuring that air and fuel are not supplied to the cylinder at the same time. The fuel-adjusting spindle on every pump is set to 0.

The module in a normal starting operation supplies control air to the main starting-air valve 01.

The oil mist detector 14 is also supplied with control air. When mist is detected, the engine speed and therefore the engine load is decreased.

The entire system is electrically activated and electronically controlled.

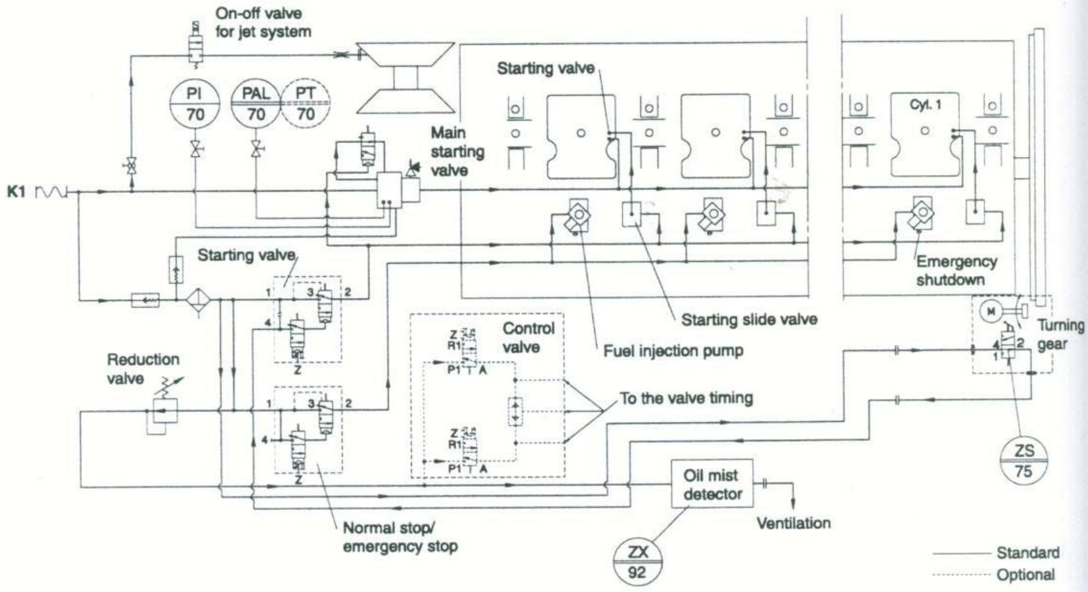


▲

The starting-air system of a four-stroke medium-speed Wärtsilä 64 diesel engine – Category III.

01 *main starting valve*
02 *slow turning valve*
03 *pressure control for slow turning*
04 *flame arrestor*
05 *starting-air valve in the cylinder head*
06 *starting-air distributor*
07 *starting booster for governor*
08 *air filter*
09 *air receiver*
10 *pneumatic cylinder*
11 *blocking valve on turning gear*
12 *valve for automatic water draining*
13 *drain valve*
14 *throttle valve*
15 *non-return valve*
16 *ball valve*
17 *non-return valve*
18 *3/2 solenoid valve*
19 *3/2 valve, pilot-controlled*
20 *4/2 valve, pressure-controlled*
21 *bursting disc, break pressure 40 bar*
22 *ball valve*

Pipe connections
301 *starting-air inlet 30 bar*
310 *control-air inlet 30 bar (including emergency stop)*



▶ The starting-air system of a medium-speed diesel engine of MAN–B&W, type L32/40.

Every cylinder has its own starting-air valve and control valve. The control valve opens the starting-air valve for the cylinder during the power stroke. The fuel pump output is pneumatically set to zero, and the emergency stop works via this system as well.

14.3 Reversing the engine

This pertains mainly to large propulsion engines with a fixed propeller. A fixed screw does not have controllable blades and therefore has to be driven by a directly reversible engine.
In practice this is often applied in large two-stroke crosshead engines and occasionally in the larger four-stroke engines. The smaller two-stroke-crosshead engines are often connected to an adjustable propeller which is also found in most four-stroke engines.

In directly reversible engines the starting- and fuelling systems must comply with the following requirements.

- The engine can start on air in two rotational directions.
- The fuel injection must commence at the correct time in both directions.
- The opening and closing of the exhaust valves in two-stroke crosshead engines with uniflow scavenging must take place at the correct time.
- Safety devices must be installed in order to prevent dangerous situations ensuing from reversing the rotational direction.

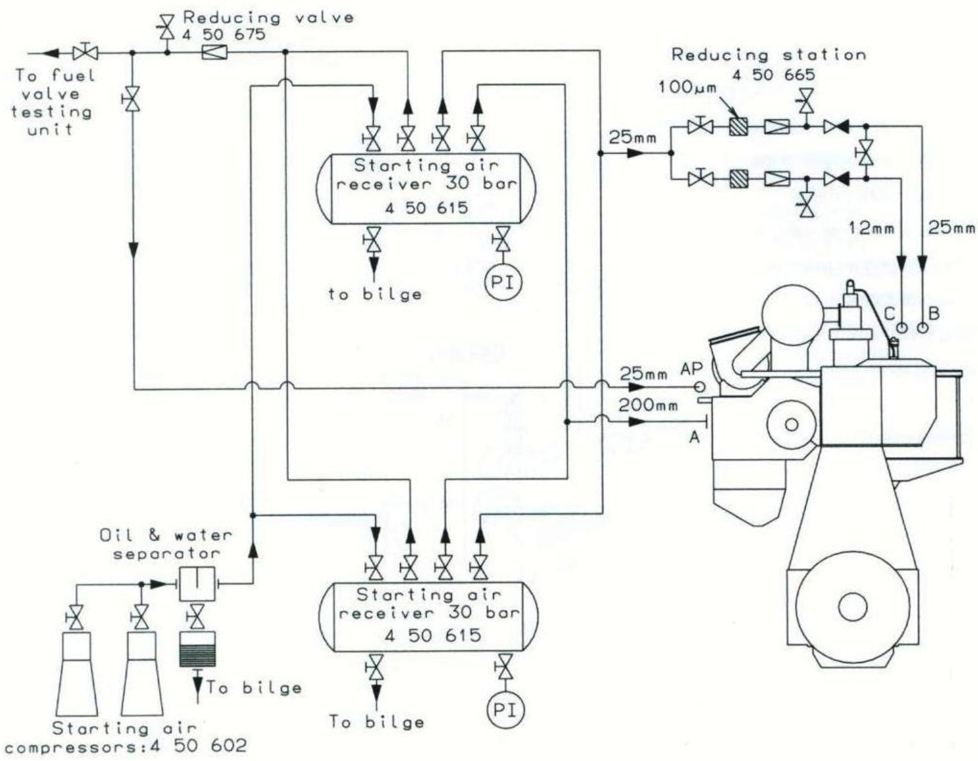
A few examples of reversing devices in diesel engines.

14.3.1 Example 1: Large two-stroke crosshead engine, direct reversing

Series MC – MAN – B&W K 98 MC.

K = short stroke piston ratio ± 2.8 (stroke = 2660 mm)
98 = cylinder bore in centimetres
M = engine program
C = camshaft controlled

In the modern two-stroke crosshead engines of the three remaining manufacturers the reversing devices are fairly simple.
The fuel cams remain in the same position as during the forward rotation.
Modification of the pressure stroke of the fuel cams occurs by reversing the roller-end of the plunger pump, which is attached to a reversing lever, against the rotation direction of the new position (reverse rotation). **See the detailed drawing of the complete starting system.**
For safety reasons, the roller-end and the lever mechanism are locked in this position.



A: Valve "A" is supplied with the engine
AP: Air inlet for dry cleaning of turbocharger
The letters refer to "List of flanges"

▲
The diagram of the starting-air system of a MAN–B&W K98 MC – C diesel engine.

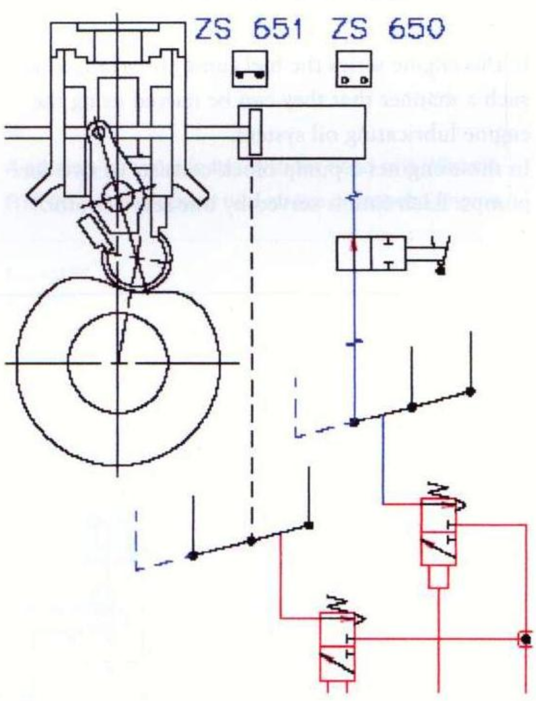
Both starting-air compressors supply air at a pressure of maximum 30 bar through an oil and water separator to two large air receivers. The receiver capacity of both is sufficient for twelve engine starts without starting the compressors.

The **connection A** with a diameter of 200 mm supplies the starting-air valve on every cylinder.

The **connection B** is fed with compressed air via a fine-filter (100μm) and reducing valve with an output pressure of 7 bar. This is control air for manoeuvring and operating the exhaust-valve air springs.

The **connection C** is fed with compressed air via a fine-filter (100μm) and reduction valve with an output pressure of 7 bar. This is safety air for the shutdown system of the main engine.

The **connection AP** is fed with compressed air via a reducttion valve with an output pressure of 10 bar and is used for turbocharging cleaning (soft blast), and a minor volume used for fuel-valve testing.

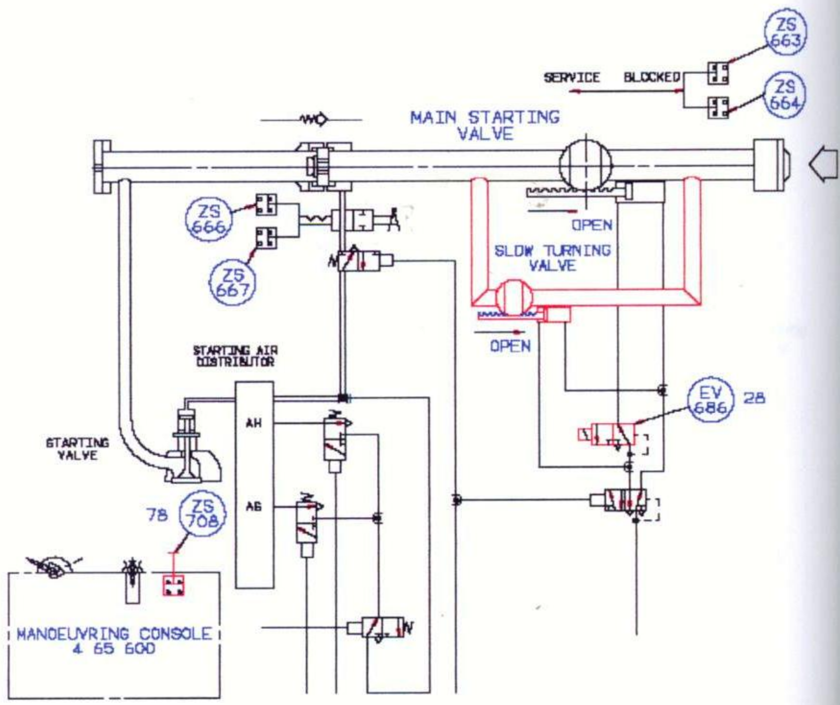


▲
For directly reversible engines, the camshaft can be moved axially allowing the fuel injection to take place at a different time. The axial movement of the camshaft occurs hydraulically.

▶

Starting-air system MAN–B&W MC 98.

The starting-air system consists of a manoeuvring console on the engine for emergency situations; for example, if the remote control from the bridge does not work or after maintenance for testing purposes. The starting-air valve is opened using control air allowing the starting air to enter the cylinder and the piston begins the power stroke.
Furthermore, there is a 'slow turning valve' which allows the engine to slowly turnover using starting air for testing purposes after maintenance.
The starting air with a maximum pressure of 30 bar is supplied via A.
Main starting valve: main starting-air valve in the supply line to the engine.
Slow turning valve: valve for slow engine turnover.
Starting valve: starting-air valve on the cylinder.
Starting-air distributor: distributes the control air to every starting-air valve for opening of this valve.
Manoeuvring console: manual starting console on the engine.



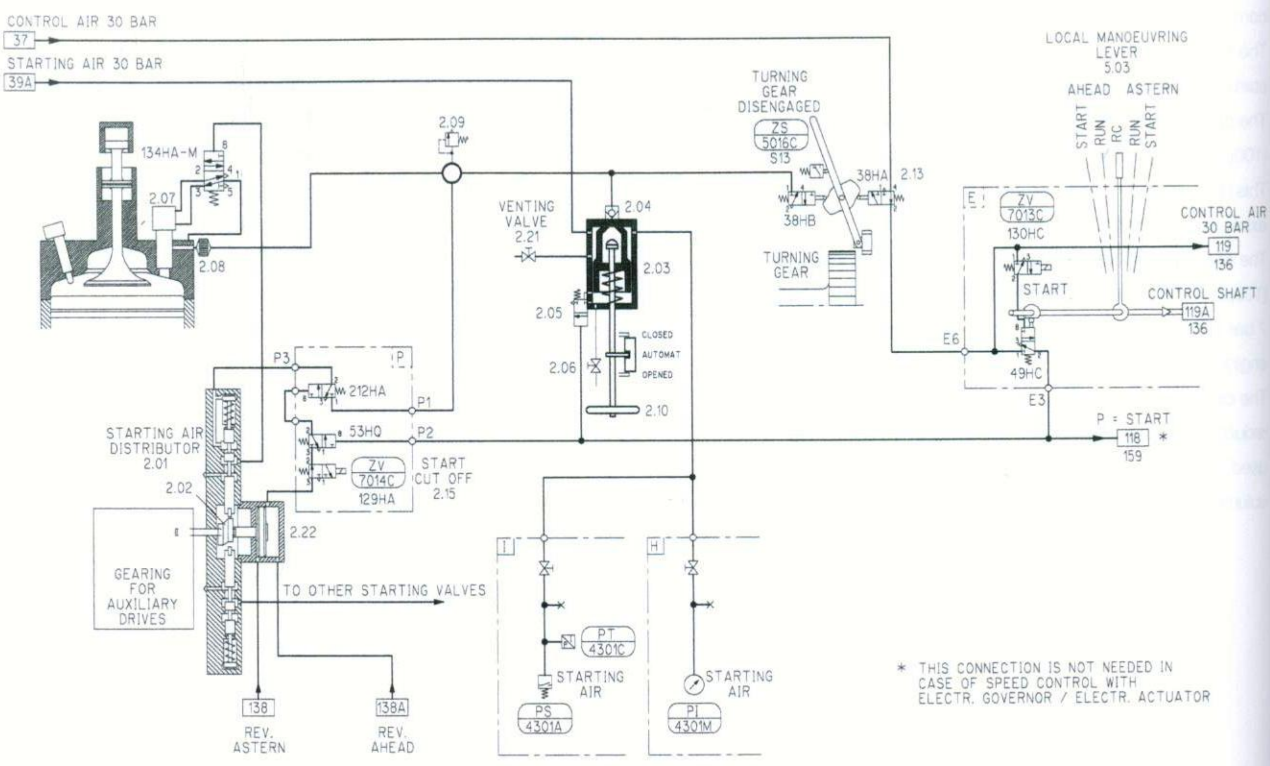
14.3.2 Example 2: Large two-stroke crosshead engine, direct reversing – Series RTA, RTA 96 C

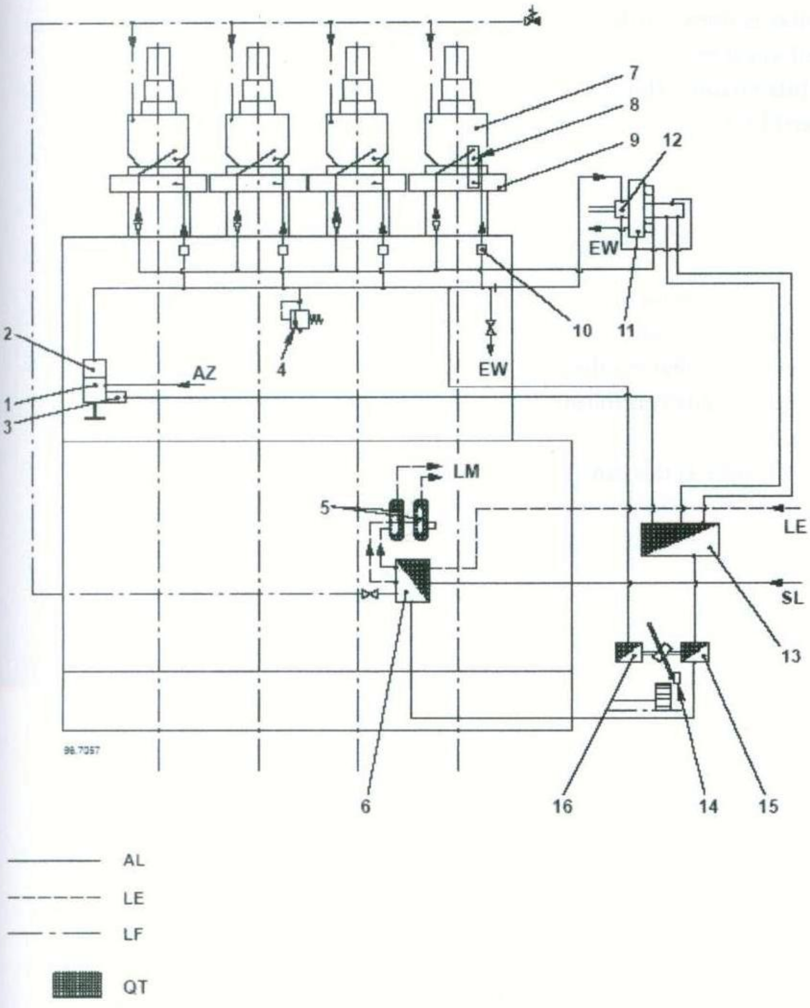
In this engine series the fuel cams are arranged in such a manner that they can be moved using the engine lubricating oil system.
In these engines a pump block consists of two fuel pumps. Each cam is served by one servo-motor.

▼

An overview of the starting-air system of the Wärtsilä RTA 96 – C diesel engine.

Above left: cylinder head with the starting-air valve
Above left middle: main starting valve
Above right middle: blocking valve on the turning gear
Above right: manual control on the engine
Below left: starting-air distributor to operate the starting-air valves in the cylinder head



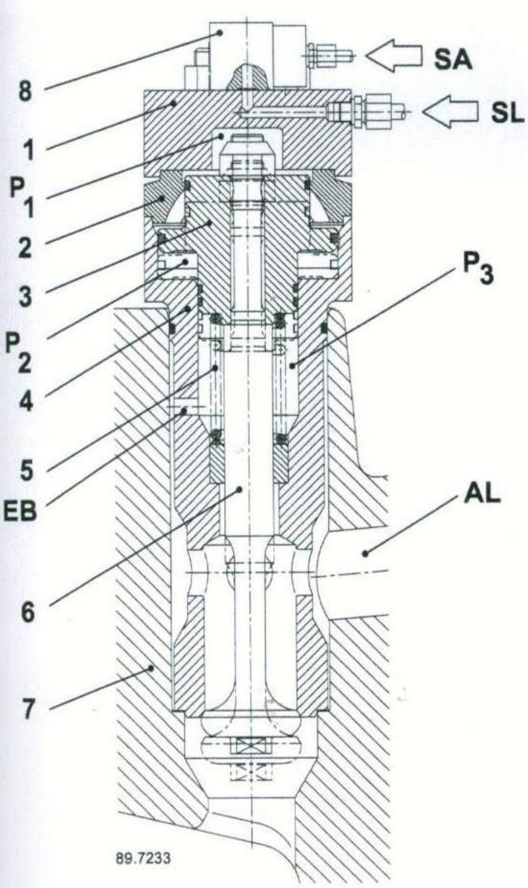


◄

Starting system of the Wärtsilä RTA 96 – C diesel engine.

1 *main starting-air valve*
2 *non-return valve*
3 *control valve*
4 *safety valve*
5 *air bottles for control air*
6 *control-air module*
7 *exhaust-valve casing*
8 *starting-air valve*
9 *cylinder head*
10 *flame plate*
11 *starting-air distributor*
12 *starting-air shut-off valve*
13 *pneumatic control unit*
14 *turning gear*
15 *shut-off valve on turning gear*
16 *blow-off valve on turning gear*

LE *control-air connection*
AL *starting-air connection*
LF *air-spring connection for the exhaust valve*
QT *components delivered by other suppliers*
SL *air connection for stand-by and control air*
LM *air for control and safety*
EW *blow-off and drain*
AZ *starting-air supply*



◄

A starting-air valve in the cylinder head of a Wärtsilä RTA 96 – C low-speed two-stroke crosshead engine.

1 *cover*
2 *ring*
3 *piston*
4 *casing*
5 *pressure spring*
6 *valve spindle*
7 *cylinder head*
8 *control valve*

SA *control-air connection from control-air distributor*
SL *control-air connection from contro- airline*
AL *starting-air connection*
EB *connection drilling for starting air*
P1, P2 and P3 *spaces*

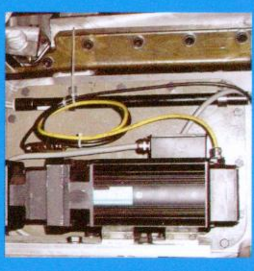
14.3.3 Example 3: Similar engines with the common-rail system – MAN–B&W ME (electronically), Wärtsilä – Sulzer FLEX (electronically)

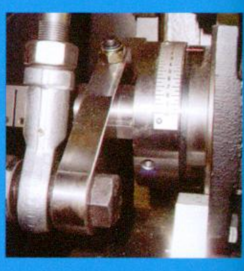
In both makes the fuel injection is controlled electronically.
As the gas exchange in modern engines occurs almost symmetrically for the current equal-pressure systems, adjustment of the exhaust valve in both mechanically and electronically controlled engines is no longer required.
Controlling the starting air is simple as this can now be electronically activated.

Also see Chapter 9, Fuel-injection systems.

>CH 15

Speed control





The governor, which is responsible for the correct and safe engine speed, is indespensible for every diesel engine. Shown, a Woodward governor, type EGB, on a four-stroke medium-speed MAN-B&W propulsion diesel engine running on H.F.O. on the passengership the “Maasdam” of the Holland-Amerika line.

The passengership “Maasdam”.

15.1 Introduction

Speed regulators or governors are an integral part of the fuel injection system in diesel engines. Generally, the fuel injection governor ensures that a diesel engine operates at a predetermined speed at varying loads.
The speed tolerances are very small, especially in gensets which require a constant frequency.

15.2 Summary

Governors have the following functions:

- fuel quantity control when the engine is started;
- fuel quantity control during engine idling;
- maintenance of a certain speed at fluctuating loads in gensets with alternating power decreases as well as in propulsion systems with varying load- and weather conditions;
- ensure that the maximum number of revolutions is not exceeded. Exceeding the maximum speed may cause damage.

15.2.1 Examples of damage

Drive gear fractures due to impermissible high acceleration and/or retardation forces.
Common damage: breakage of the connecting rod.
The piston can collide with the inlet- and exhaust valves at too high a speed, as the valve springs cannot close the valves in time.

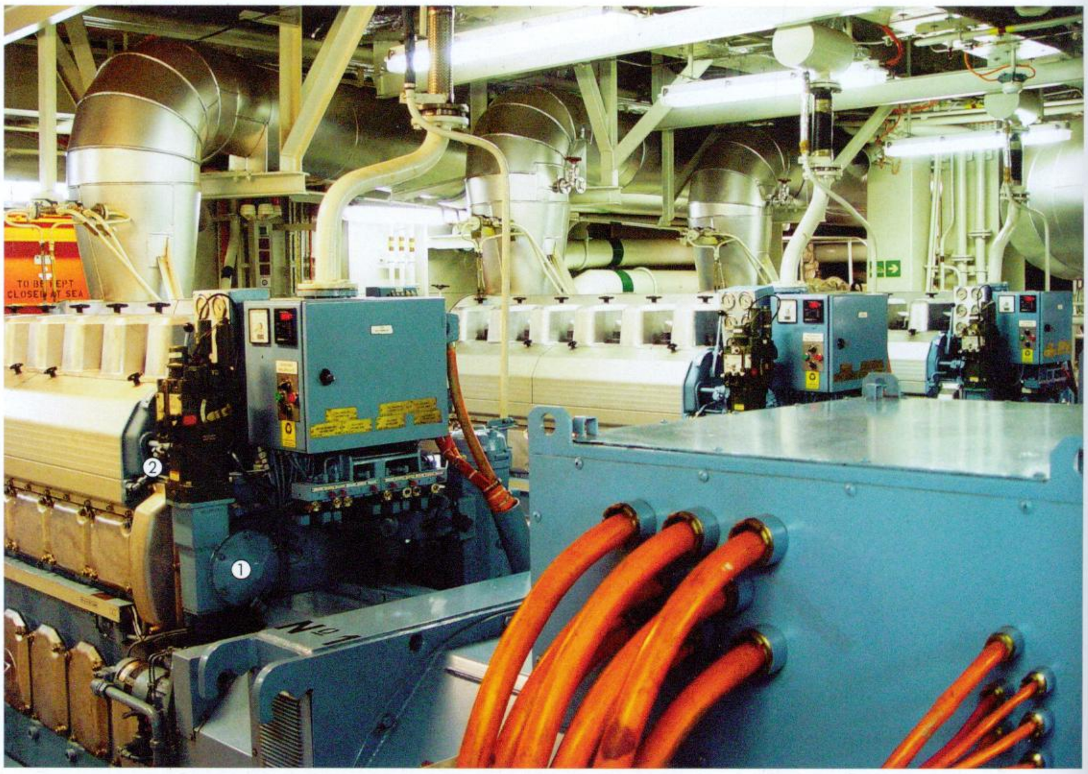
If the engine torque no longer matches the torque required by the propeller or generator, the result will be an angular acceleration or retardation: the speed alters. An adequately working governor will regulate the quantity of injected fuel via the (fuel) adjusting spindle in order to achieve the predetermined value.
So the governor operates in a state of equilibrium. The speed of the engine continuously changes slightly and time the finely tuned governor intervenes and modifies the injected fuel quantity.

15.3 Types of governors

With regard to the operation, there are roughly three different types of governor:

1 mechanical governors;
2 hydraulic governors;
3 electronic governors.

There are often combinations of all three.



▸
Woodward governors on three six-cylinder four-stroke medium-speed Wärtsilä 20 diesel engines used for gensets on the 'Oranjeborg' of Wagenborg Shipping. The gensets run on heavy oil.

1 *camshaft drive*
2 *governor type 3161*

15.3.1 Mechanic governors

These have been in use since the first steam engines, steam turbines and internal combustion engines. Today they are predominantly used in small engines in engine categories I and II. This is due to the fact that they are reliable and relatively inexpensive. Their operation is based on a principle formulated over 200 years ago by the renowned physicist James Watt.
Here the centrifugal forces work on the rotating mass of the governor weights, in counter resistance to the governor spring thrust. The centrifugal force is squared in relation to the speed.
These systems have two different types:
1. The springs with which the speed can be set are embodied in the rotating mass.
2. The springs are placed outside the rotating mass.

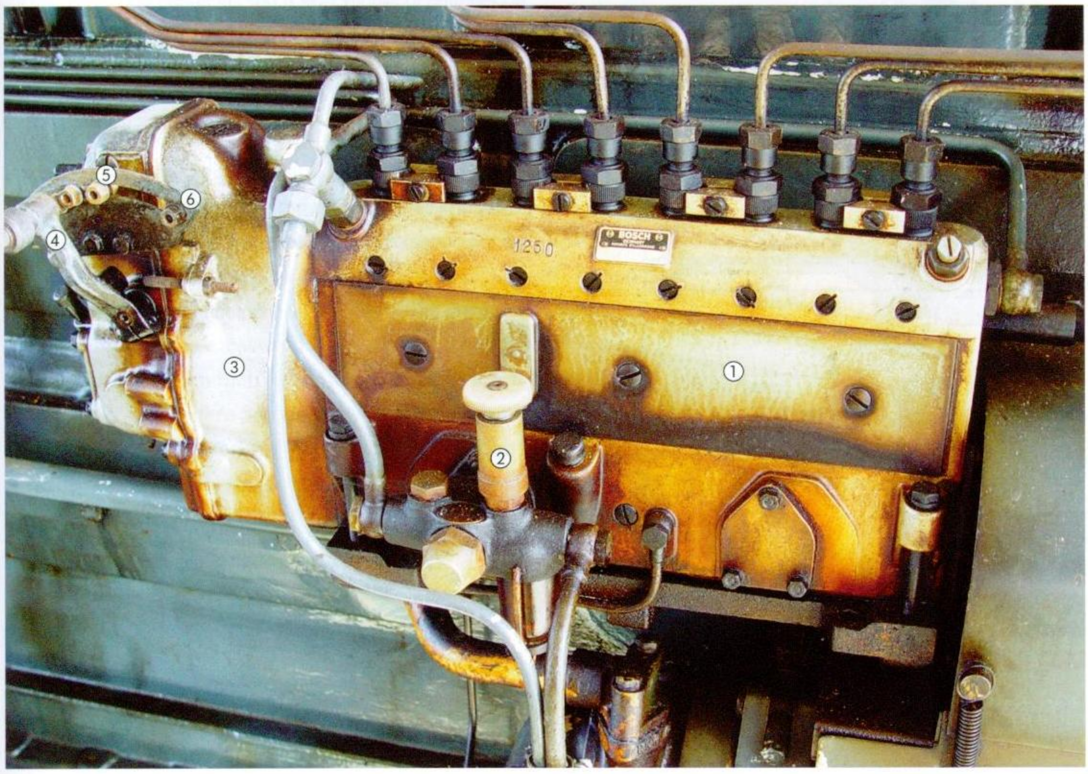
Generally, one can say that the greater the spring depression, the greater the resistance and the higher the speed before the injected fuel quantity is reduced by the governor.

Most mechanic governors operate within small deviations. The degree of irregularity amounts to between 2 and 10%, depending on the type of actuation and the type of engine.

The various parts of the governor have different functions:
- a speed adjustment mode;
- an actual speed measurement mode;
- a comparison of the actual speed and the desired speed mode;
- a manner in which the governor modifies the fuel quantity supplied to the engine;
- an engine stabilisation mode, that is, maintenance of a certain number of revolutions after the fuel quantity has been altered.

Setting the speed
The mechanic/hydraulic governors utilise a spring. The more pressure that is exerted on the spring, the higher the speed. The higher the resistance, the greater the quantity of fuel supplied to the engine.

Measuring the speed
The governor requires a force proportional to the speed of the engine. In mechanic/hydraulic governors this is supplied by the centrifugal forces of the fly weights, which are driven by a mechanical connection to the engine.
In electronic systems this force originates from measuring the frequency of a magnetic pick-up and a converter, which are directly linked to the speed of the engine.



◄ A Bosch mechanical governor mounted directly on the block fuel pump rebuilt from an old, eight-cylinder Kromhout diesel engine.

1. *block fuel pump, in-line*
2. *manually operated deaerating pump*
3. *mechanical governor*
4. *speed adjusting handle; the engine is switched off*
5. *minimum speed*
6. *normal speed*

Modifying the fuel supply to the engine

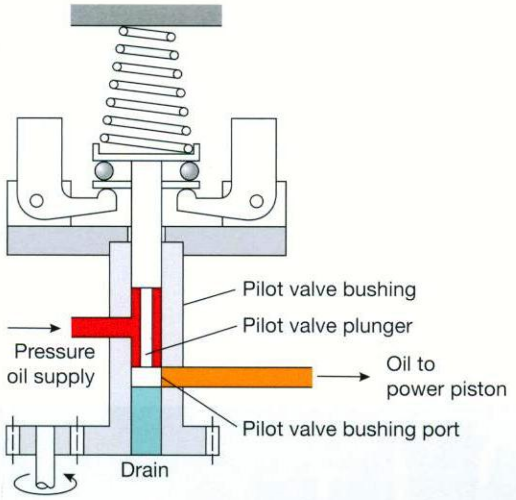
The mechanic/hydraulic governors or actuators of electronic speed measurement usually have a rotating or a linear linkage, which is attached to the fuel adjusting spindle of the engine. When the governor needs to make a modification to retain the speed or the load, the axis is moved in the required direction. At a constant fuel supply the engine generates a certain power output. Usually, the governor measures the speed and adjusts the fuel supply which in turn modifies the power output.

Fly weights

By increasing the speed, the weights spin further outwards and therefore the fuel supply to the engine is reduced and the speed returns to its set value. Obviously, the reverse also occurs. At a decreased speed the weights rotate inwards and the fuel supply to the engine is increased and the speed rises to its set value. Friction in the system produces time delays before the speed is altered. This is called the 'dead band', when momentarily nothing happens.

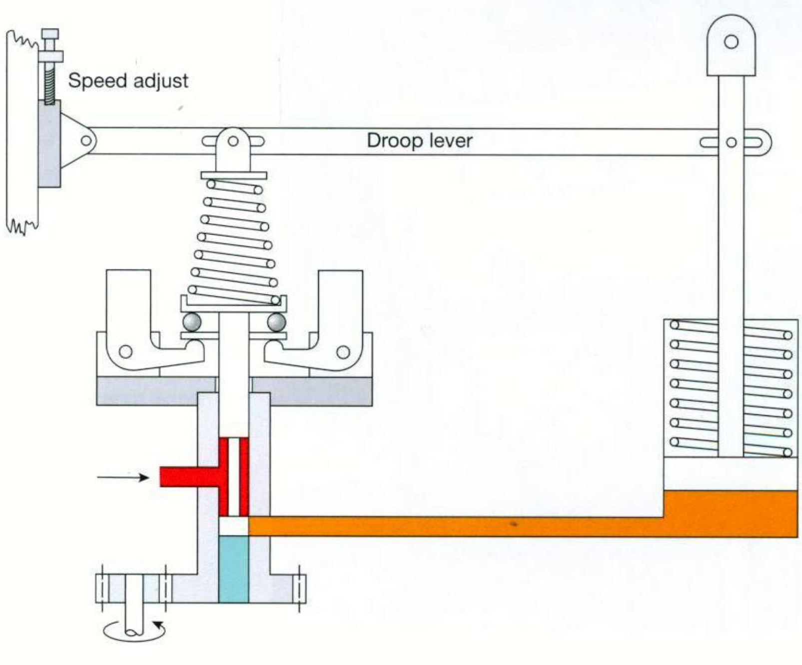
Disadvantage of fly weights

In larger engines the flying weights do not have sufficient force to serve the larger fuel systems. In order to operate these fuel systems, often the fuel adjusting spindle, an auxiliary hydraulic system is used. Hence the term: mechanic/hydraulic governor. Most mechanic/hydraulic governors and actuators use the governor drive to operate the hydraulic-oil pump. They often are gear-driven pumps.



◄
Drawing showing the working principle of the mechanical governor.

The entire device is driven by the engine form below. The governor flyweights pivot outwards exerting pressure on the pilot control valve, thus depressing the spring. The control valve, attached to the shaft regulates the oil flow to the fuel-control piston. Engine speed lower than pre-sets the spring extends and forces open the pilot valve, thus causing the piston to allow more fuel to enter the engine. The engine accelerates to slightly over e pre-set speed and continues to running more or less at that speed.



◄
The complete working principle showing the piston, spring mechanism and levers.

Top right: the governor is connected to the fuel-control system.

15.3.2 Mechanic/hydraulic governors

Here the adjustment forces used to control the fuel pump capacity are not directly provided by the measuring device, but by a hydraulic system. Due to this hydraulic system extra features are added to the governor. One of the most frequently used hydraulic governors is the Woodward UG 8.
In the world of governors one uses unusual terms such as 'speed droop'.

Speed droop

This is the governor function which reduces the governor reference speed at an increasing load.
In many governors the extent of the droop can be set on the governor.

15.3.3 Electronic speed control

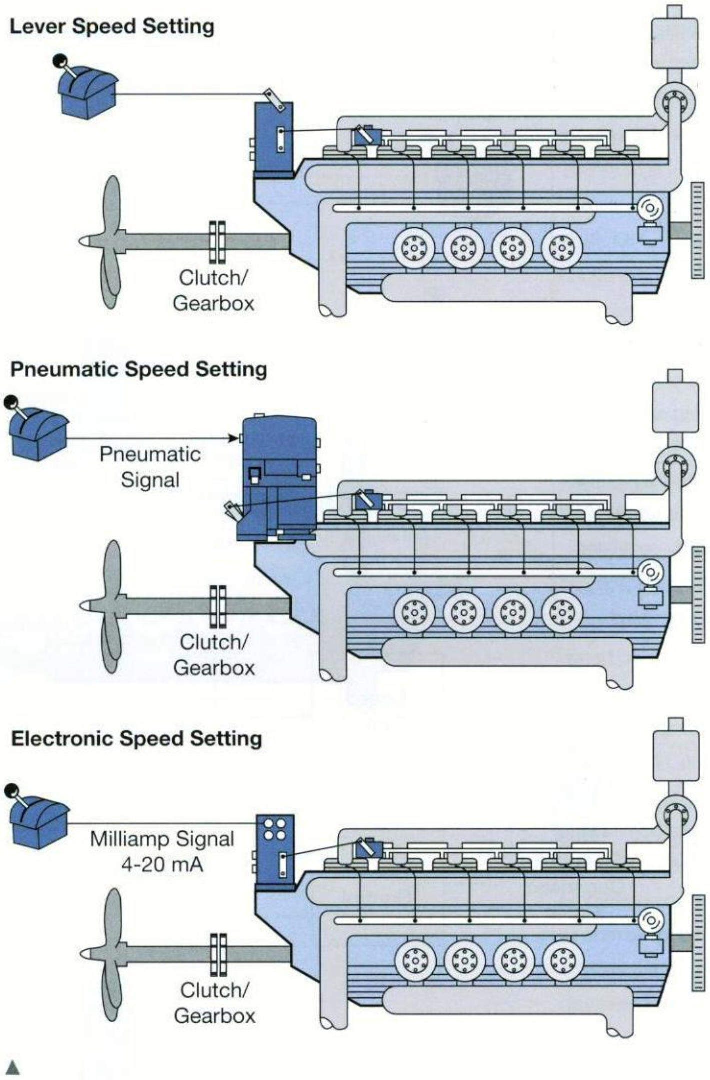
These governors are often applied in diesel engines in categories III and IV and also more often in category II.
Apart from controlling the set speed, electronic governors serve numerous other purposes:

- avoidance of excessively high speed;
- limiting the maximum fuel supply;
- limiting the maximum scavenging air pressure;
- controlling the maximum fuel quantity available when the engine is started;
- controlling the actuation of a screw with, for instance, two engines;
- controlling the delay of, for instance, switching on a coupling or in general alarms.

Operation of the electronic governor or digital speed governor

The basic principle of the electronic governor is its comparison of the set speed and the actual speed. At a certain discrepancy between these values, which can be adjusted, a signal is sent to the 'actuator' fixed to the fuel adjusting spindle and this is adjusted until the required speed is achieved.
The actual speed is measured by means of magnetic pick-ups which are often installed closely to the fly-wheel. There are often several (two to four) pick-ups installed for various purposes. The pick-up sends signals to the governor.
The number of signals is dependent on the speed and is in practice often equal to the number of teeths on the fly-wheel which pass the pick-up per time unit.
This system is often applied in high-speed and medium-speed four-stroke engines. For low-speed two-stroke crosshead engines this system is too slow and other systems are used

15.4 Examples of engine configurations with different types of governors



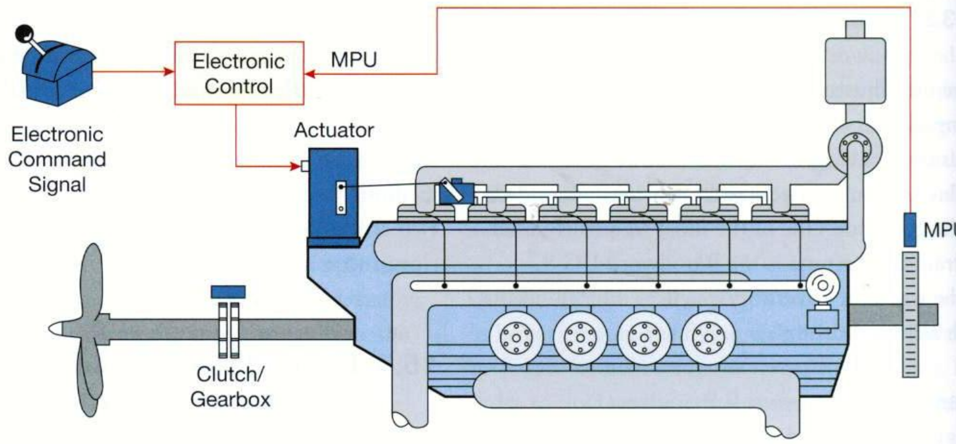
▲
Woodward governors, mechanical.

Various ways to set the speed of a propulsion engine.

First figure: mechanic transmission
Second figure: pneumatic transmission
Third figure: electronic control using a 4-20 milli-ampere control system

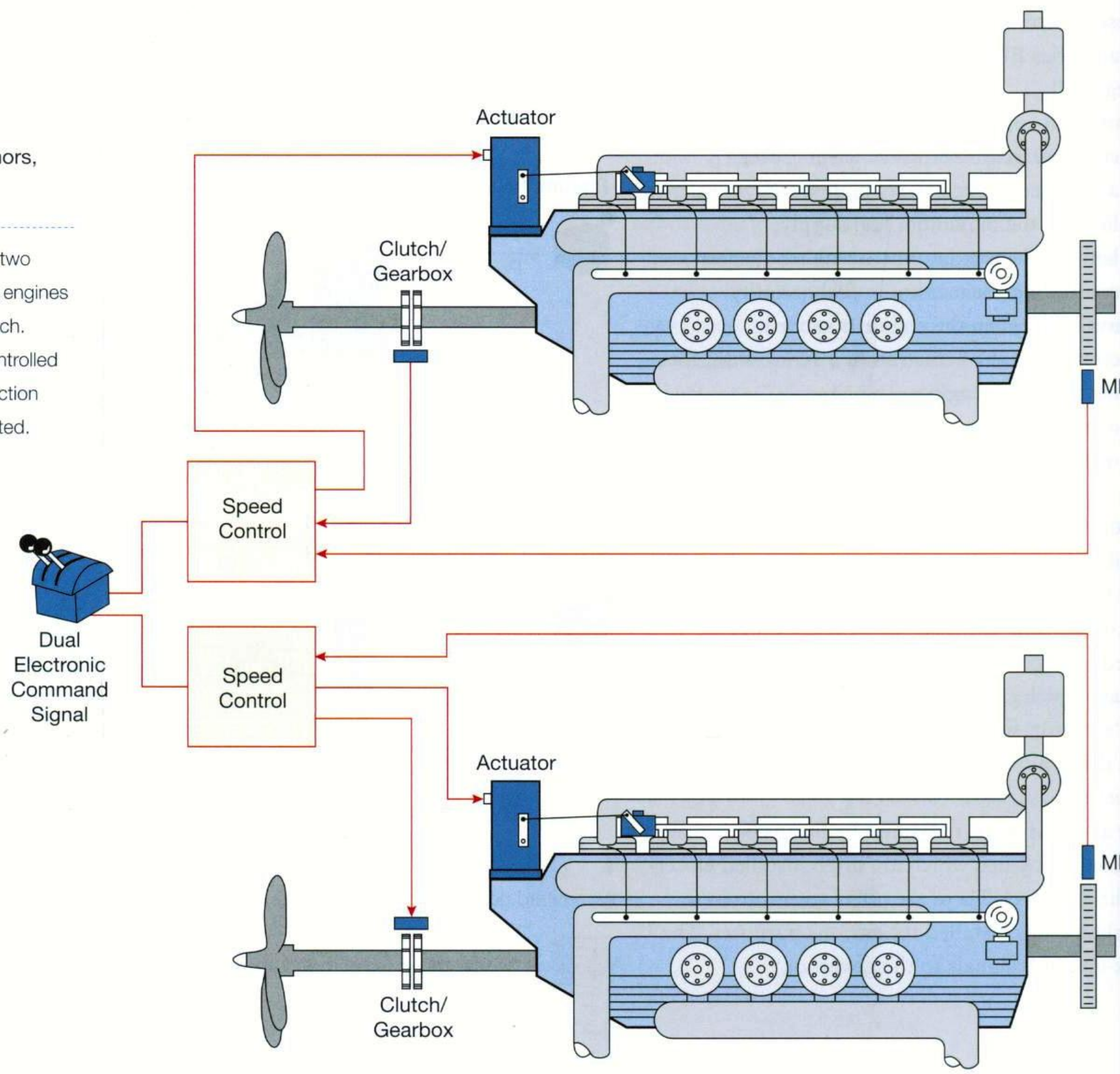
▶
Woodward governors, electronic.

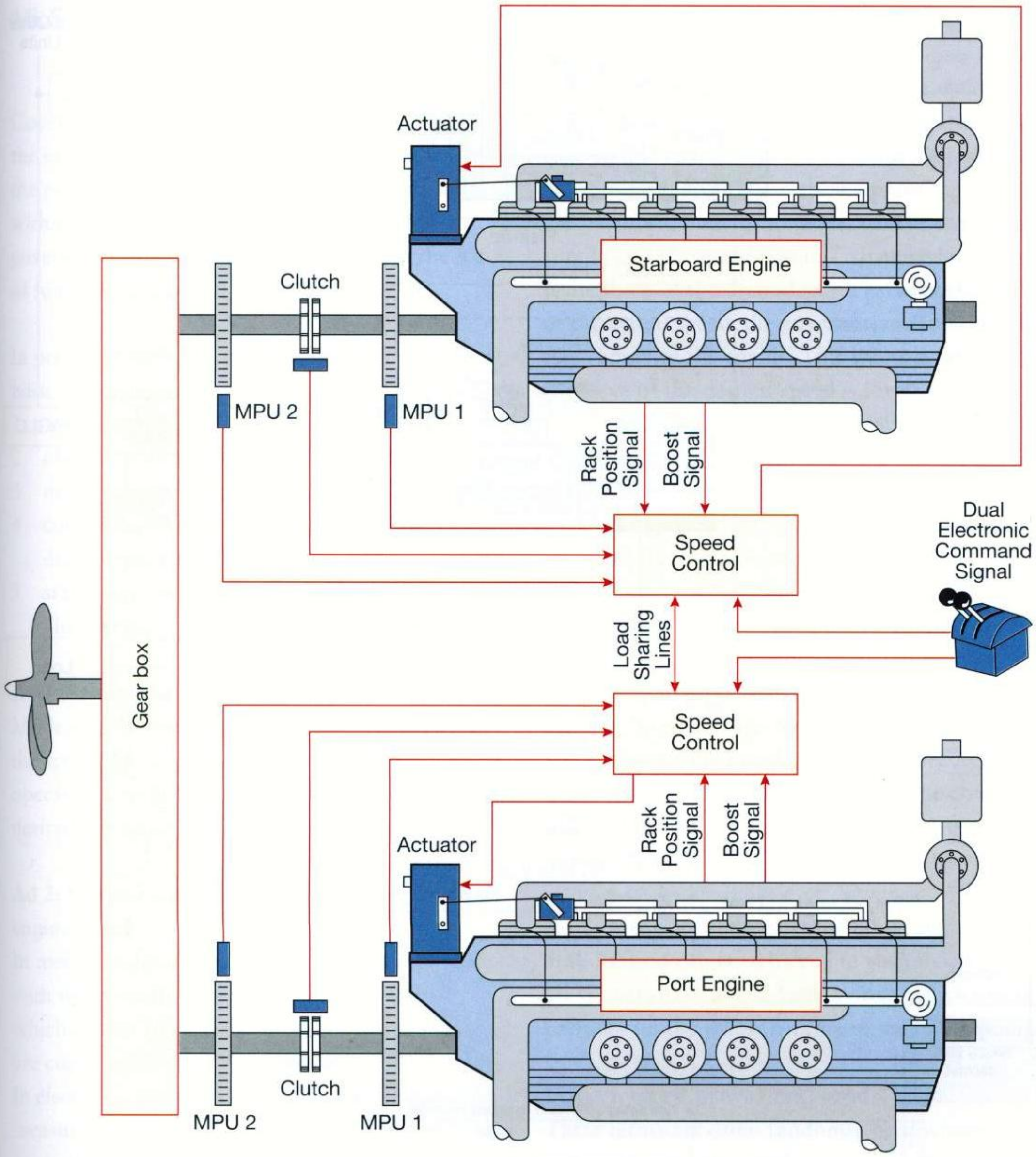
An example of a propulsion engine with an electronic governor. The engine speed is measured electronically with a tachometer attached to the gear ring on the turning/fly-wheel (MPU). The electronic control unit controls the actuator which mechanically controls the fuel control shaft. The speed can be adjusted with an 'electronic' control handle. MPU = Magnetic Pick Up



▶
Woodward governors, electronic.

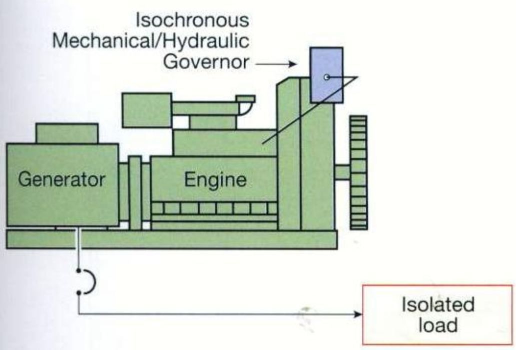
Identical system for two separate propulsion engines a single propeller each. The hydraulically controlled coupling in the reduction gearing is incorporated.





◀ Woodward governors, electronic.

A system using two propulsion engines for a single propeller. The master/slave system.
Both engines are electronically controlled.
The actuators are electronically controlled from the 'Speed Control' units.
The 'Speed Control'-units receive data from both the fuel control shaft and the turbo -blower pressures.
This determines engine load.
Two revolution counters are mounted on the output shaft: MPU1 before the coupling and MPU 2 after the coupling.
This allows a comparison between the outgoing shaft speed of the propulsion engine and the incoming shaft speed of the reduction gearing when the connection is switched off.
This is necessary in order to achieve the correct engine speed prior to the engagement of the coupling.
The connection between both Speed Control units allows a comparison between both loads in relationship to both propulsion engines.
At normal speed with two engines the load is distributed as equally as possible to both engines.
One engine has a certain speed to which the other engine adjusts its speed.



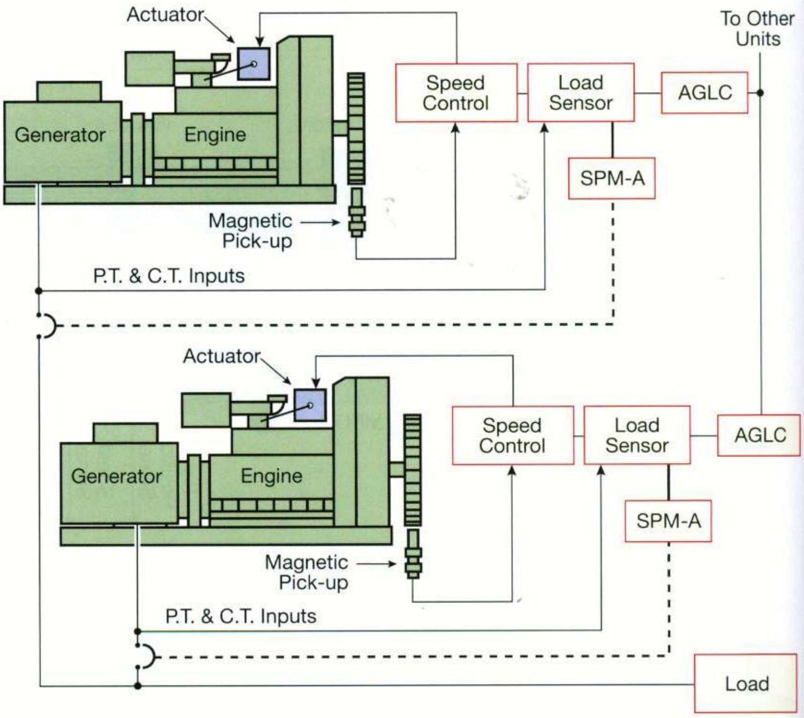
◀ Woodward governor, mechanic-hydraulic.

A single electric genset with an intrinsic load. The pre-set governor ensures a relatively constant RPM at various loads.

▶

Woodward governors, electronic.

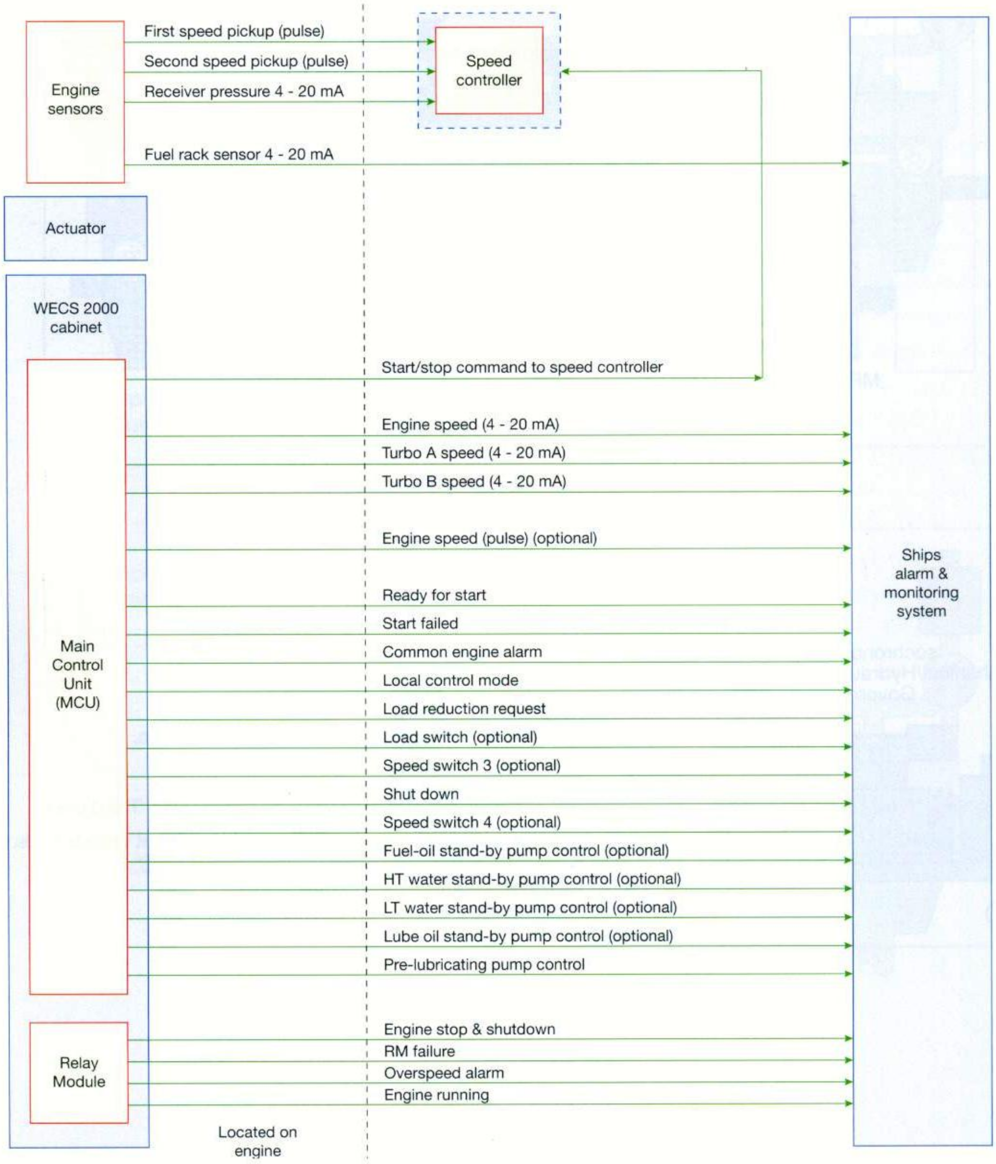
A system for single or multiple diesel gensets with automatic synchronisation of the individual sets in the utility network and separate load-control units.
Apart from the speed control with a magnetic 'pick-up' which controls the actuator and hence the fuel control, the electronic load of each set is separately measured.
AGLC: Automatic Generator Loading Control. This system serves ensure that the genset safely operates at a certain pre-set load.
SPM-A: Speed and Phase Matching Synchronizer. This provides automatic frequency and phase control matching for the generator.
This system prevents the occurrence of undesirable loads.



▶

The governor in the complete monitoring and control system for a Wärtsilä 26 diesel engine Category III.

The Main Control Unit can transmit a start/stop signal to the governor. The electronic governor has two pick-up systems.



15.5 Theoretical background of speed governors

Governors are driven by the engine and control the injected fuel quantity in such a manner that the predetermined diesel engine speed remains within strict margins. One could say that a governor controls the energy quantity, in the form of fuel that is supplied to the engine.

In principle each governor comprises a number of basic components:

1 speed set-up mode;
2 obtaining an actual speed reading;
3 modifying the actual speed to the set speed;
4 controlling the fuel supply quantity to the diesel engine;
5 stabilising the speed after fuel supply alterations.

Ad 1: Setting the required speed
Modern governors have advanced systems to set the speed. These compensate frequently occurring operating conditions in order to maintain the desired the speed.

Ad 2: Governor must receive a signal of the diesel engine speed
In mechanic-hydraulic governors this takes place with the centrifugal forces of the fly weights, which can be rotated by system of linkages that are connected to the diesel engine.
In electronic governors this signal is produced by measuring the frequency with a magnetic pick-up, converter or generator which is directly related to the speed of the diesel engine.
The frequency is subsequently converted into an electrical force which is made available for the adjustable direction of the fuel. In both instances the force increases as the engine speed increases.

Ad 3: Tuning the actual speed and the desired speed
The 'force ' of the required speed and the 'force' of the actual speed are totalled. The desired speed is a force in one direction and the actual speed is an opposing force. When these opposing forces are balanced, there is no change in the engine fuel supply and the speed remains constant.
If the force of the desired speed exceeds the force of the actual speed, the governor will increase the fuel supply to the engine. This is also applied when the force of the desired speed is lower than the force of the actual speed. In mechanic-hydraulic governors this difference in forces is absorbed by the so-called thrust block or thrust bearing. In electronic governors these forces are totalled in the so-called summing point.

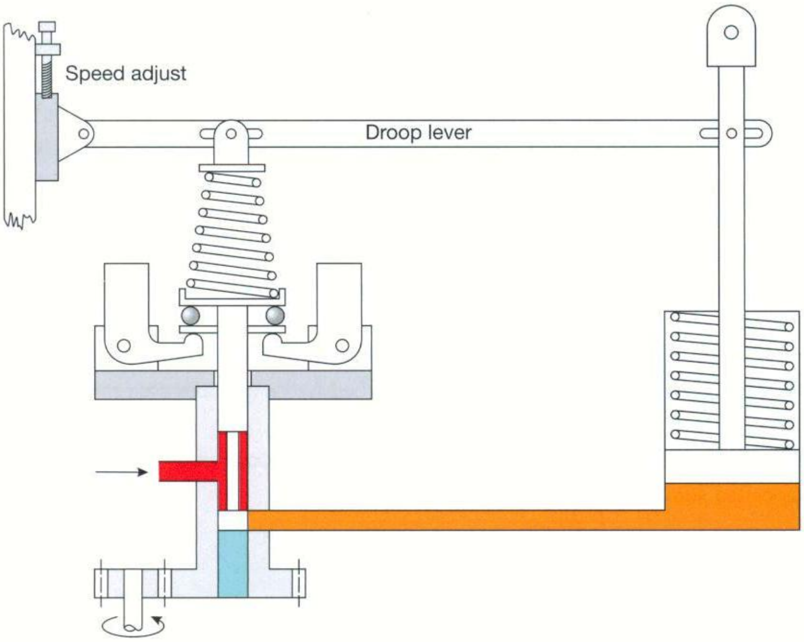
Ad 4: Means of controlling the fuel supply to the diesel engine
The mechanic-hydraulic governors usually have a rotating linear linkage which is connected to the fuel system of the diesel engine. In order to alter the speed, the linkage, and therefore the control rod which is connected to the shaft linkage in the governor, is shifted.

Ad 5: Means of stabilising the diesel engine speed
Stabilisation can be achieved in various ways, but all systems have a feed-back system which exerts pressure on the thrust bearing or summing point.

Speed, shaft power and load

These terms are often randomly used where governors are concerned.
The governor controls the fuel supply to the engine in order to maintain a constant speed or engine load.

15.5.1 Basic principle of a hydraulic/ mechanic governor



▲

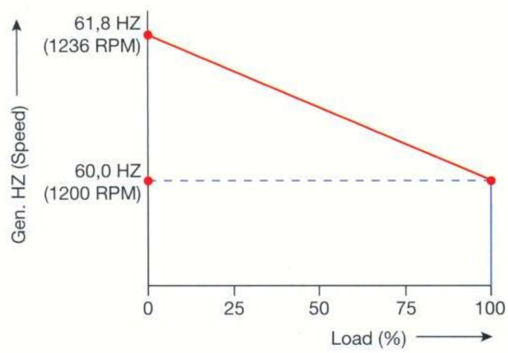
Working principle of the mechanic/ hydraulic governor.

Flyweights are mounted on the shaft, which is connected to the camshaft of the engine via a gear transmission. When the engine speed is increased, the fly-weights pivot outwards around their shaft against a predetermined force set for the spring.

Next, a control valve moves downward and allows hydraulic oil to enter the piston under pressure. The top of the piston is provided with compression spring. The hydraulic-oil pressure causes piston movement against the compression spring which subsequently moves a fuel-control rod attached to the high-pressure fuel pumps.

▼

An example of the droop.



Here the droop is $\frac{1236 - 1200}{1200} \times 100\% = 3\%$.

Droop formula:

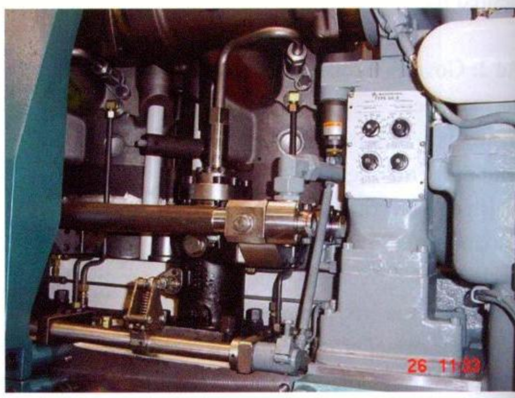
$$\%\ \text{droop} = \frac{\text{Unloaded engine speed} - \text{full-loaded engine speed}}{\text{full-loaded engine speed}}$$

15.5.2 Droop

This term is frequently applied in relation to governors. Droop is a system which prevents excessive fluctuation of the speed around the set point. This is called 'hunting'.
Droop ensures that the engine maintains a stable speed in spite of fluctuating loads.

15.5.3 Actuators or operating mechanisms

Each governor must exert a certain pressure on the fuel adjusting spindle of the high-pressure fuel pumps in order to control the fuel supply. The angle of actuation is also important. Data with regard to governors always state for which type of engine they are suitable. Information such as output and shaft travel, however, is also of significance.



▲

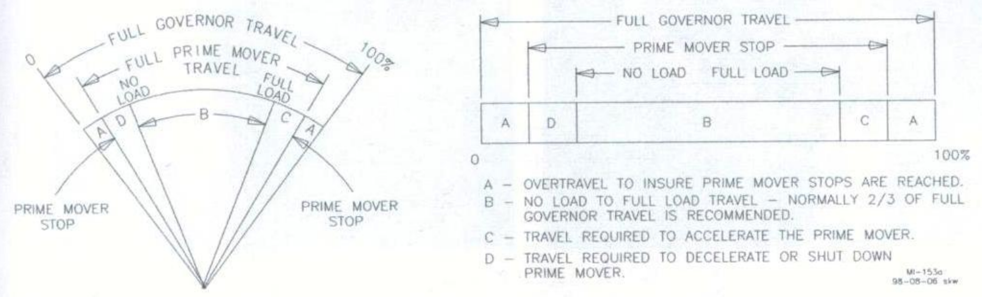
An UG–3 Woodward governor on a medium-speed four-stroke engine ofMAN–B&W type 27-38.

▲

Detail of the UG–3 Woodward governor with the 'Speed droop' and 'Speed Setting' settings.

SPECIFICATIONS

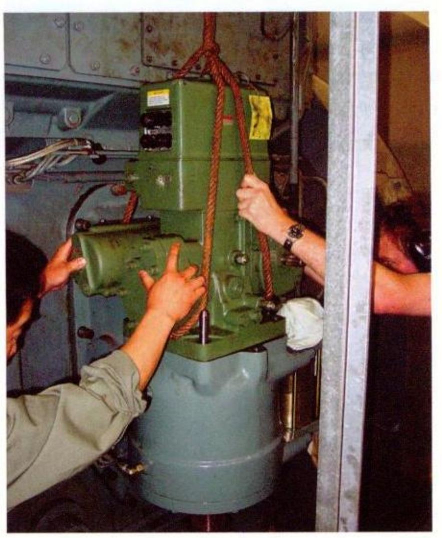
Governor Drive	Standard drive is 5/8-36 (inch) serrated. Drive shafts are available with 0.187 inch-wide keyways and threaded castle nut to carry gear.
Drive Speeds and Requirements	Maximum speed range of 375 to 1500 rpm. Recommended constant speed operating range of 1000 to 1500 rpm. Requires 970 W (1.3 hp) at normal speed and operating temperature. Rotation clockwise or counterclockwise.
Work Output	7.1 N·m (5.2 lb-ft) for the UG-5.7 13.2 N·m (9.7 lb-ft) for the UG-8 15.9 N·m (11.7 lb-ft) for the UG-10



Terminal Shaft	0.500 inch diameter, SAE-36 serrations. May extend from either or both sides of the governor. Shafts designed for specific applications are available.
Linkage	The relationship between engine torque output and governor terminal shaft travel should be linear (very important for gas or dual fuel engines).
Steady State Speed Band	±0.25% of rated speed
Variable Speed Range	375 to 1500 rpm
Droop	Adjustable on the dial governor from 0 to 12.5% at 1500 rpm and from 0 to 19% at 1000 rpm Adjustable on the lever governor from 0 to 26.5% at 1500 rpm and from 0 to 40% at 1000 rpm All droop figures are based on 42° of terminal shaft travel. If less than full shaft travel is utilized, available droop will be decreased by the same percentage as is output shaft travel.
Ballhead/Drive Configuration	A spring-driven, oil-damped ballhead and flexible drive is often used to dampen the high-frequency, low-amplitude torsional vibration which may be present in the drive to the governor. Ballheads are also available in undamped natural frequencies of: none, 50, 70, 100, or 150 cpm.
Operating Temperature	Gas-fueled engines, particularly those used on gas pipelines, often have not operated at optimum efficiency because automated controls to maintain that peak condition have not been readily available.
Pilot Valve Plunger Movement	Balanced between ballhead centrifugal force and speeder spring force.
Pilot Valve Bushing	Rotated as an integral part of governor drive shaft.
Pilot Valve Porting	A selection of chopper, 2 slotted or 8 round. Chopper gives slow response in acceleration.

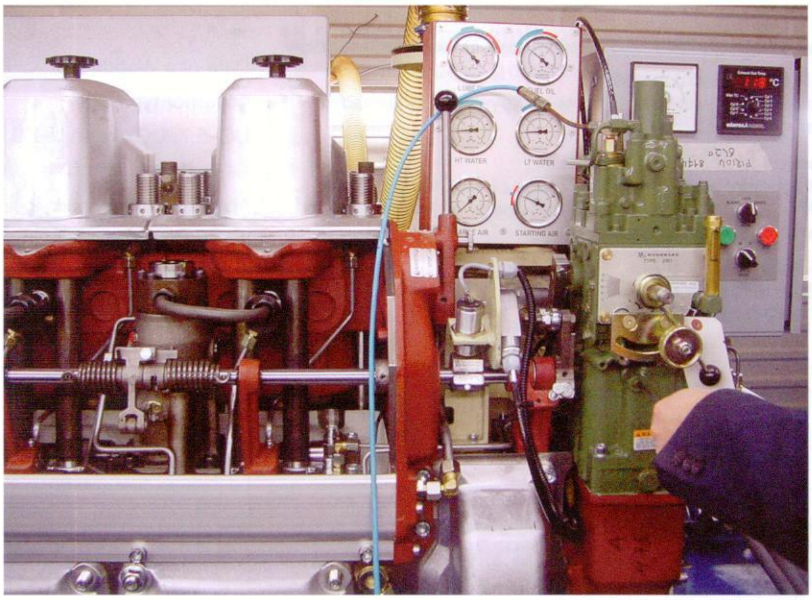
◄ Data of a Woodward governor type UG/ UG–5.7, UG–8 and UG–9.

Apart from the speed range, the mechanical stops for the output shaft travel to the fuel-control shaft and the forces they may exert on the control shaft are important. The droop is also provided. Furthermore, the operating principle of the governor.

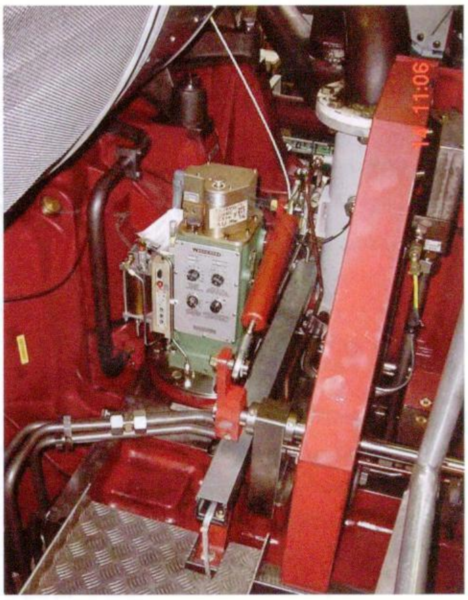


◄◄ A Woodward UG–40 governor with an actuator 4 on a 16 VM 32 C Caterpillar-MAK diesel engine.

◄ Installing a governor on the mechanical drive for the diesel engine.

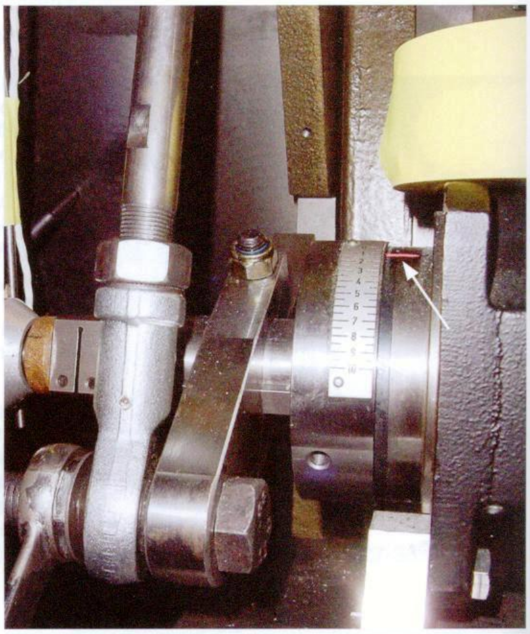


▲
A Wärtsilä 20 medium-speed four-stroke engine running on H.F.O. with a Woodward 3161 governor.



▲
An UG–40 Woodward governor on a Caterpillar–MAK diesel engine.

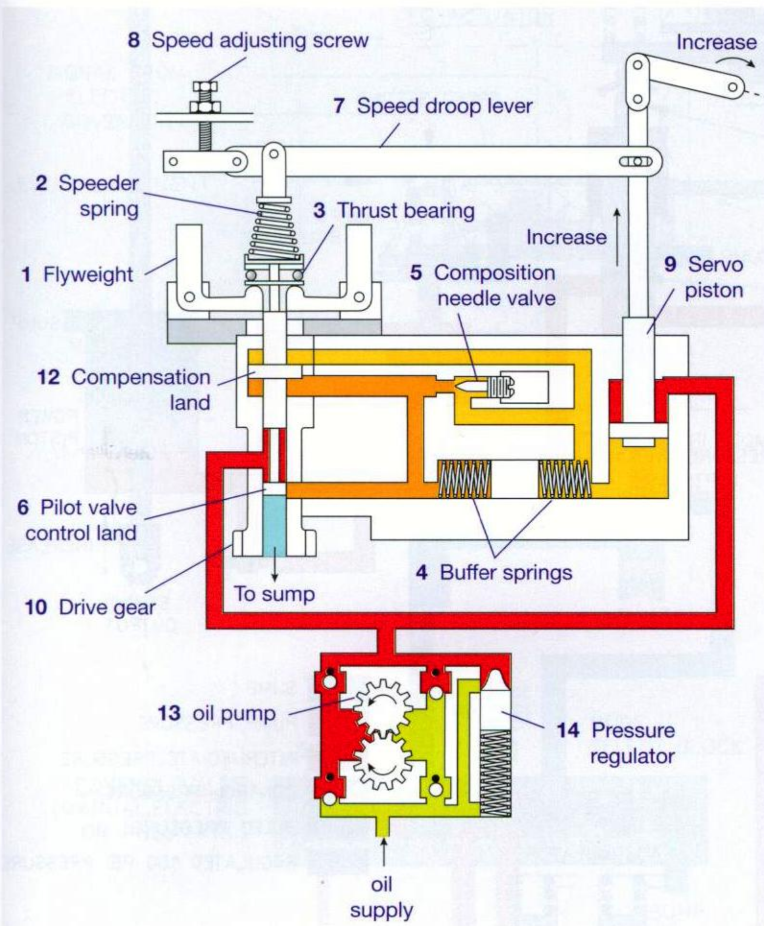
▲
A Woodward PG – governor on a Ruston 20 RK 270 high-speed four-stroke diesel engine running on M.D.O. for a high-speed catamaran of Trasmediterranea.

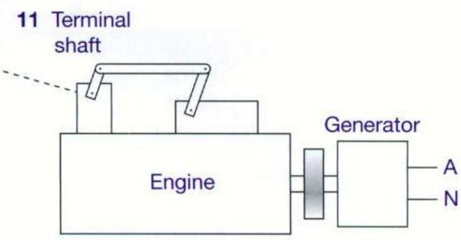


▲
The fuel control position. See arrow.

15.5.4 Examples of mechanic/ hydraulic governors with 'droop'

Basic compensated mechanical covernor with droop





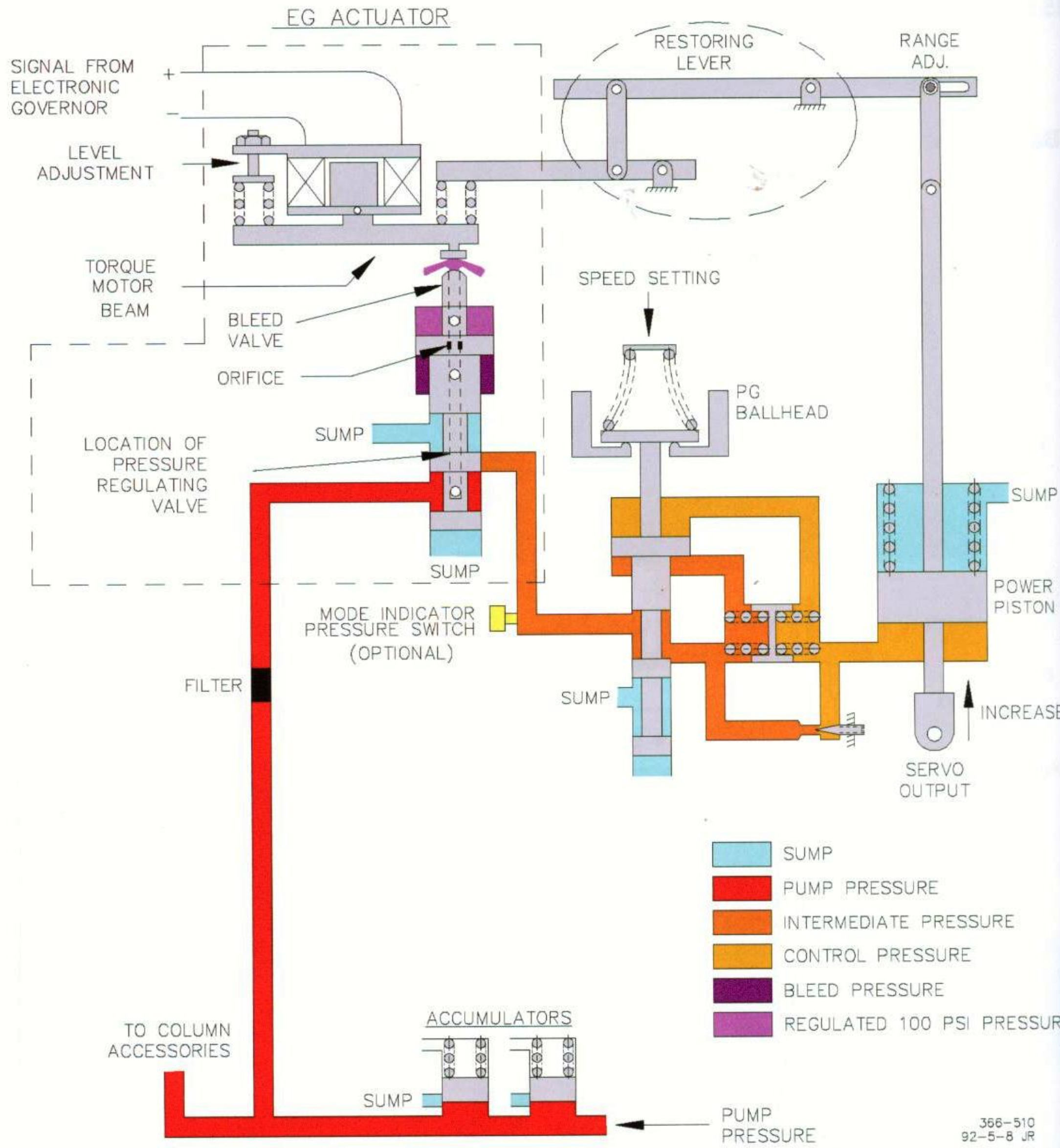
1	Flyweights	- Sense prime move speed.
2	Speeder spring	- Force for desired speed.
3	Thrust bearing	- Sum of actual speed and desired speed, droop and force from compensation land.
4	Buffer springs	- Sets gain of hydraulic circuits.
5	Needle valve	- Stabilize prime mover.
6	Pilot valve control land	- Controls flow of oil to servo piston.
7	Speed droop lever	- Feed back from servo piston to speeder spring for droop.
8	Speed adjusting screw	- Set desired speed.
9	Servo piston	- Force to move terminal shaft.
10	Drive gear	- Connected to engine to rotate pilot valve bushing and ballhead.
11	Terminal shaft	- Rotational output of actuator to move linkage.
12	Compensation land	- Transmits force diff. to thrust bearing to assist in return of pilot valve to center when fuel correction is made.
13	Oil pump	- Provides pressurized oil.
14	Pressure regulator	- Regulates pressure of oil from pump.

▲

Mechanical governor with 'droop'.

The diagram above shows a mechanical governor for an electric generator driven by a diesel engine.

1 *Flyweights, the core of the mechanical governor, react to engine speed.*
2 *Speed springs provide the force for the required RPM.*
3 *Thrust block gives the actual engine speed and the required engine speed.*
4 *Buffer springs compensate the hydraulic systems.*
5 *Needle valve ensures the smooth running of the system.*
6 *Control valve, controls the oil flow to the piston*
7 *'Droop' lever.*
8 *Speed adjusting screw. Sets the required speed.*
9 *Piston. This drives the shaft towards the actuator.*
10 *Drive gear. This is connected to the diesel engine and drives the mechanical part of the governor connected to the flyweights.*
11 *Output shaft of the actuator, drives the fuel control for the high-pressure fuel pumps.*
12 *Compensation piston transmits the force divergence to the thrust block in order to assist the control valve (6) to return to its original position when a sufficient amount of fuel has been delivered to the high-pressure fuel pumps.*
13 *Oil pump provides. oil pressure.*
14 *Spring-loaded overflow valve ensures a constant oil pressure.*

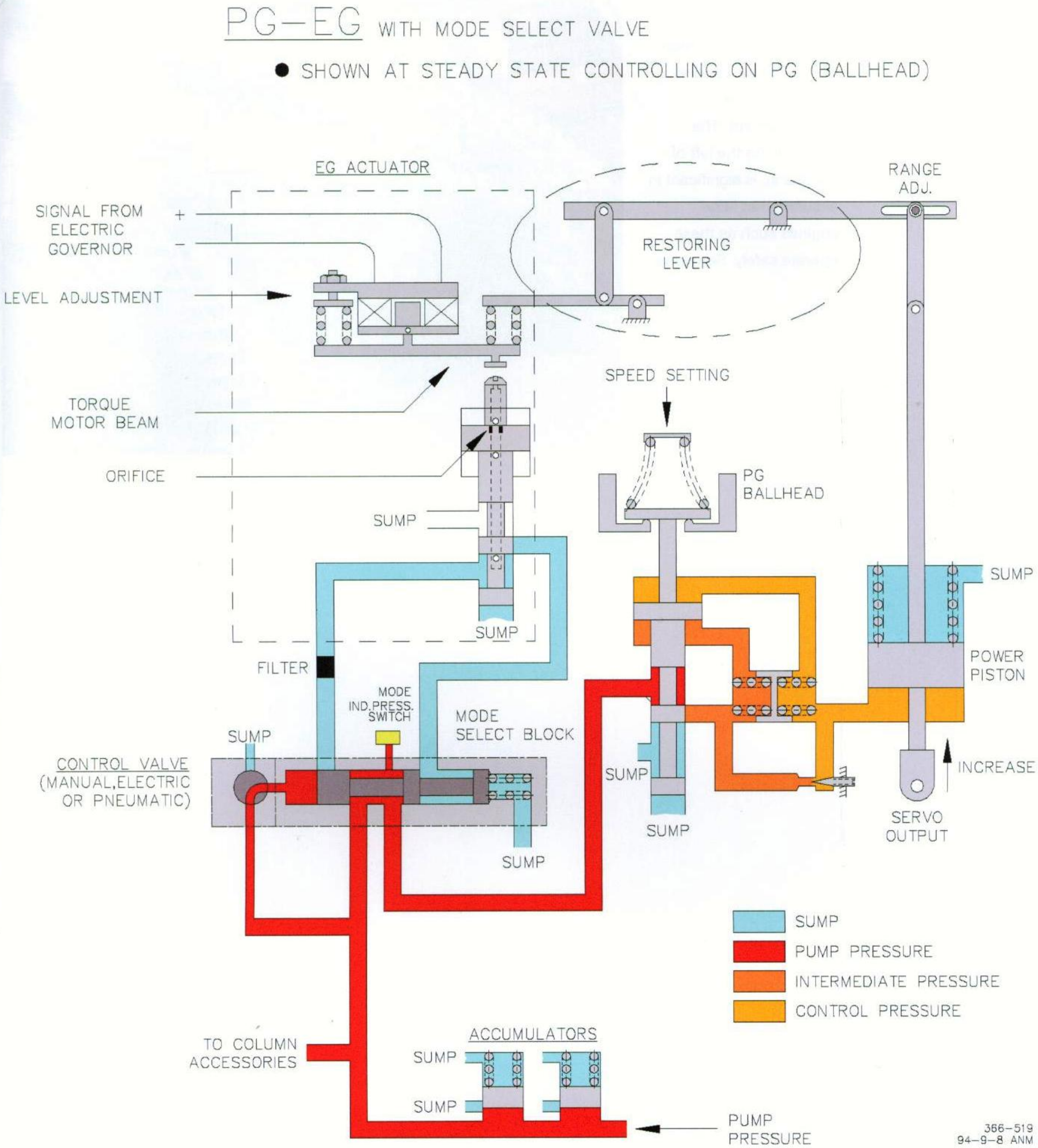


▲

A Woodward governor, PG–EG-version with electronic governor/actuator.

Flyweights control valve are opened. This governor has no 'select' setting, either manually, electrically or pneumatically.
Light blue: discharge oil to sump
Red: oil pressure
Brown: intermediate oil pressure
Light brown: control oil pressure
Purple: bleed pressure
Cyclamen: 100 PSI, constant pressure

The flyweights (mechanical) and the electronic governors (with actuators) operate continually, each for their required fuel supply.

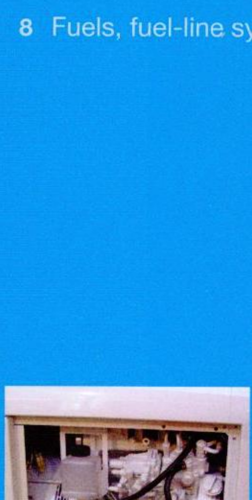
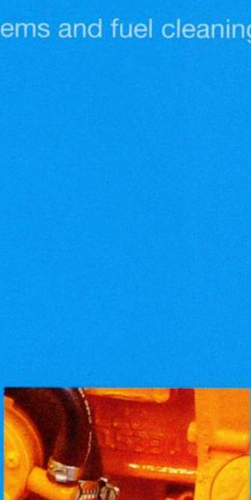


▲
A Woodward governor, PG–EG with mode selection switch, shown here controlled with flyweights. The control valve can be set to manual, electrical or pneumatic control.

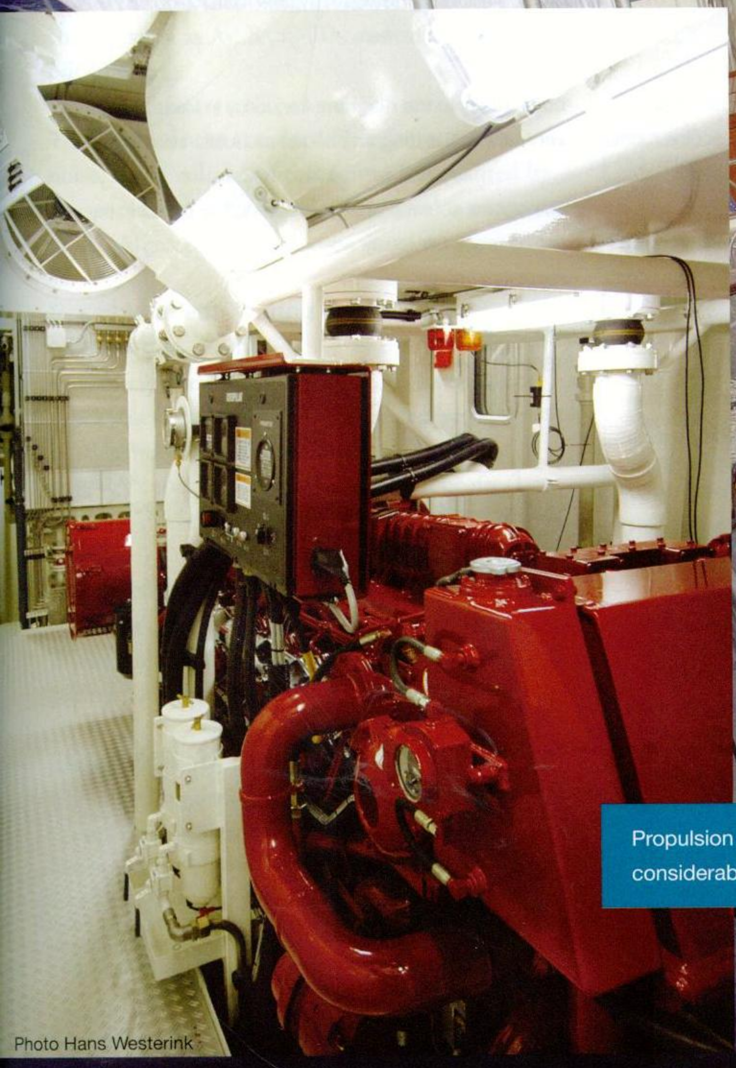
►
A large MAN–B&W four-stroke medium-speed diesel engine in V-arrangement. The governor, (to the left of the stairs), is significant in ensuring that large engines such as these operate safely. See arrow.

>CH 16

Noise, origin and damping


One of the world's largest private sailing ships "the Athena" was built by Royal Huisman, Vollenhove, The Netherlands.
These enormous yachts have elaborate technical installations.
In this instance two Caterpillar 3516 propulsion engines of 1472 kW and a large genset of 870 kW.
Today, the requirements with regard to noise (damping) on ships are very strict.
The same is applied to the reduction of vibrations.
Customers of these great yachts are very demanding regarding to noise and vibration reduction.
This forms a tremendous challenge for specialised companies!



Propulsion of a yacht with a 982 ton displacement requires a considerable power output.

Photo Hans Westerink

Photo Franco Pace

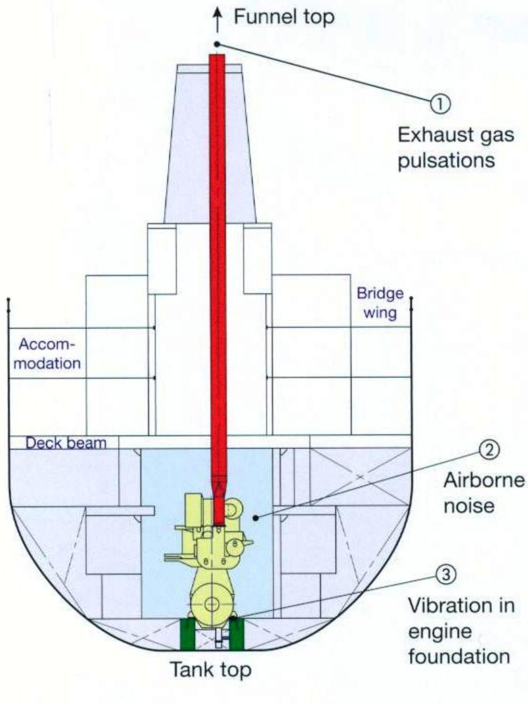
16.1 Introduction

Diesel engines produce a large amount of noise. Noise pollution is a nuisance and can be damaging to health for humans and animals. Excessive sound levels in a working environment can negatively affect an employee's well-being and result in a loss of productivity.
Measures ensuing from the guidelines pertaining to environmental and legal requirements reduce sound levels to acceptable levels. These are valid for the surroundings of a diesel engine, which is to say, in the immediate vicinity of a diesel power plant.
Anyone working with diesel engines should have proper hearing protection.
Working on or in the vicinity of extremely highly loaded modern diesel engines, with consequently high load numbers, which are operating at full load can be very unpleasant, despite the use of protective gear. Working near or on an operational diesel engine should be avoided when possible.

▼

Three sources of engine noise.

1 Exhaust gas noises: this is especially important in the vicinity of the accommodation. Noise abatement, by good silencers in the exhaust-gas ducting.
2 The sound carried by the air in the engine room, particularly important in the engine room itself. Noise abatement, by good sound proofing of the engine room.
3 The sound transmitted through the construction, particularly important with regard to the ship's accommodation. Noise abatement, by vibration proof arrangement of the diesel engine.



16.2 Origin of noise in diesel engines

Diesel engines operate on the principle of internal combustion. In the cylinder above the piston combustion processes continuously take place with a high frequency. In this process, the pressure development of, for instance, a four-stroke engine throughout the course of the inlet/intake, compression, power and exhaust strokes fluctuates rapidly. These pressure waves generate enormous amounts of noise.

▼

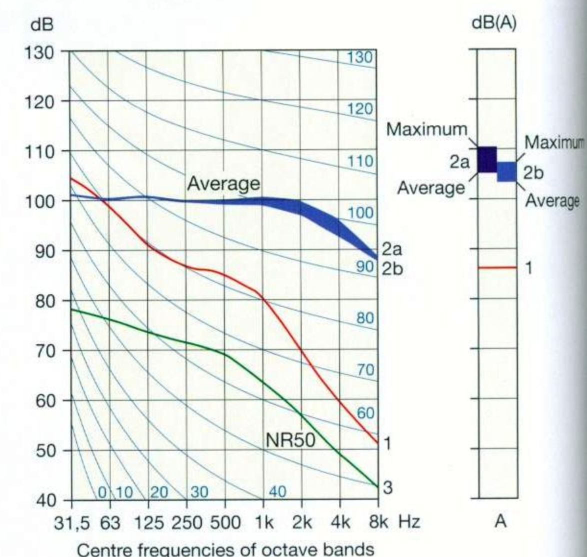
An example of a MAN–B&W 6 L 80 MC Mk 5 two-stroke crosshead engine. MCR = 20,580 kW at 93 rev/min.

1 sound of exhaust gases
2 sound carried by the engine-room air
3 sound carried by the constructions

2a high efficiency blower
2b conventional blower

horizontally in the diagram: frequency in Hertz
vertically in the diagram: sound production in dB (A)

The highest sound production is carried by the air in the engine room. Obviously, adequate soundproofing is absolutely imperative!



1. Exhaust gas noise – distance 15 metres (re 2 x 10
2. Airborne noise – average and maximum (re 2 x 10
a: High efficiency turbocharger / b: Conventional turbocharger
3. Structure-borne noise – engine foundation, vertical (re 5 x 10

16.2.1 Sound

Sound is a physical phenomenon.
One can distinguish two aspects:
1 **the pitch or frequency, denoted in Hertz;**
2 **the volume, the level.**

- The sound output level is denoted in Watts.
- The sound pressure level is denoted in N per m^2 = Pa.
- Difference in sound levels are expressed in decibel, dB.

16.2.2 Weighting filters

The human ear with respect to frequency is 'non linear' and 'level dependent'. This means that the ear is not equally sensitive to both low and high frequencies. Generally, one can say that the human ear is less sensitive to tones less than 1000 Hz and this sensitivity decreases as the frequency drops. The human ear becomes also increasingly less sensitive to tones exceeding 6000 Hz.
Both frequencies are affected by the sound level. The lower the sound level, the less sensitive the ear is to frequencies within the 1000 to 6000 Hz range and, obviously, outside this range. This is why the so-called weighting filters have been chosen, such as A-, B-, C-, D- and NR-filters.

The A filter is the standard filter for industrial noise. If the volume of this sound is measured by an operational A filter, the sound level is expressed in dB (A).

Possible effects during or after brief exposure to sound

These may vary considerably, from for instance, shock, increased heart rate, dizzy spells. Furthermore, speech impediments, decreased memory capacity, irritability and fatigue are possible side effects.

Long-term effects

Amongst others, high blood pressure, heart disease, ulcers and dizziness. Temporary and/or permanent hearing loss.

Legislation

The most important article in relation to noise issues is article 179a of the Factories or Workplaces Safety Decree (IMO).

Article 179a stipulates that machinery, the work place and the activities carried out should not produce harmful sound levels, unless such cannot reasonably be prevented. At present, the limit is the equivalent of 85 dB (A).
According to the law noise abatement should occur at the source as much as possible.

Protection

From a sound level of 80 dB (A) hearing protection equipment should be provided by the employer and at a sound level of 90 dB (A), usage is compulsory.

What is allowable during work?

For a working day of eight hours a maximum sound pressure level of Lp (A) = 85 dB (A) measured at the working place. **With every 3 dB increase the sound level is doubled and the following rule applies**

Sound pressure level	Expose to sound
80 dB(A)	8 hour
83 dB(A)	4 hour
86 dB(A)	2 hour
89 dB(A)	1 hour
92 dB(A)	30 min
95 dB(A)	15 min
98 dB(A)	8 min
101 dB(A)	4 min
104 dB(A)	2 min
107 dB(A)	1 min
110 dB(A)	30 sec

Comment

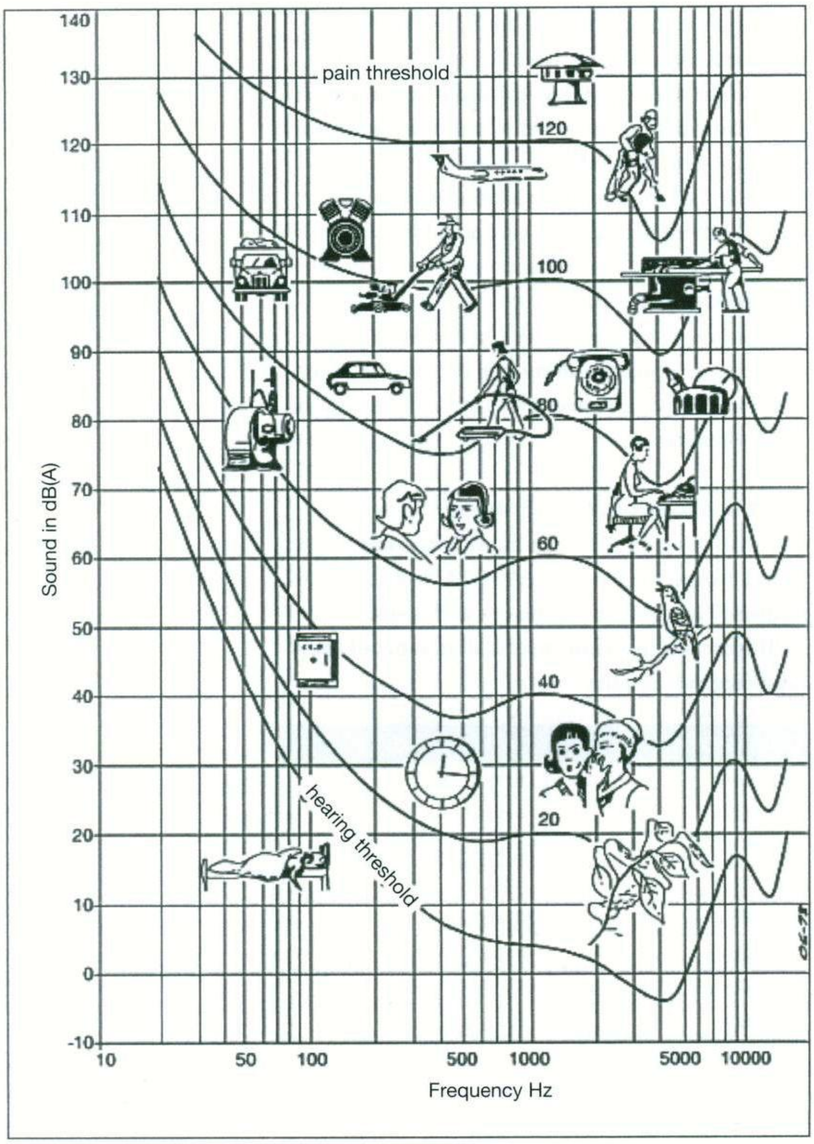
Hearing loss is always permanent!

The 120 dB (A) level is known as the 'pain threshold'.

	dB(A)
Work spaces	
Machinery spaces (continuously manned**)	90
Machinery spaces (not continuously manned**)	**110**
Machinery control rooms	75
Workshops	85
Unspecified work spaces**	90
Navigation spaces	
Navigation bridge and chartrooms	65
Listening posts, including navigation bridge wings and windows	**70**
Radio rooms (with radio equipment operating but not producing audio signals)	60
Radar rooms	65
Accommodation spaces	
Cabins and hospital	60
Mess rooms	65
Recreation rooms	65
Open recreation areas	75
Offices	65

** Ear protectors should be used when the noise level is above 85 dB(A), and no individual's daily exposure duration should exceed four hours continuously or eight hours in total.

◄ A summary table of areas on a ship indicating the maximum sound levels in dB (A).
Requirements by the IMO – the International Maritime Organisation.



◄ Environment noise production at various frequencies.

horizontally: frequency in Hertz

vertically: sound levels in dB (A)

Diesel engines are found in the upper section of the graph between 100 and 120 dB (A).

16.3 Sound transmission paths

It is important to establish through which paths sound is transmitted to the receiver. There are two paths:

- **airborne noise,** sound which is transmitted through the air;
- **impact sound,** sound is transmitted by constructions such as engine bed plates, pipes and casings.

Rigid 'couplings' between a source of sound (vibration) and, for instance, a casing may even amplify the sound; the construction behaves as an amplifier. These sound transmissions can be measured.

► A survey of the sound level found at a certain distance from the source. Shown here is the exhaust of the diesel engine above the bridge on the funnel deck.

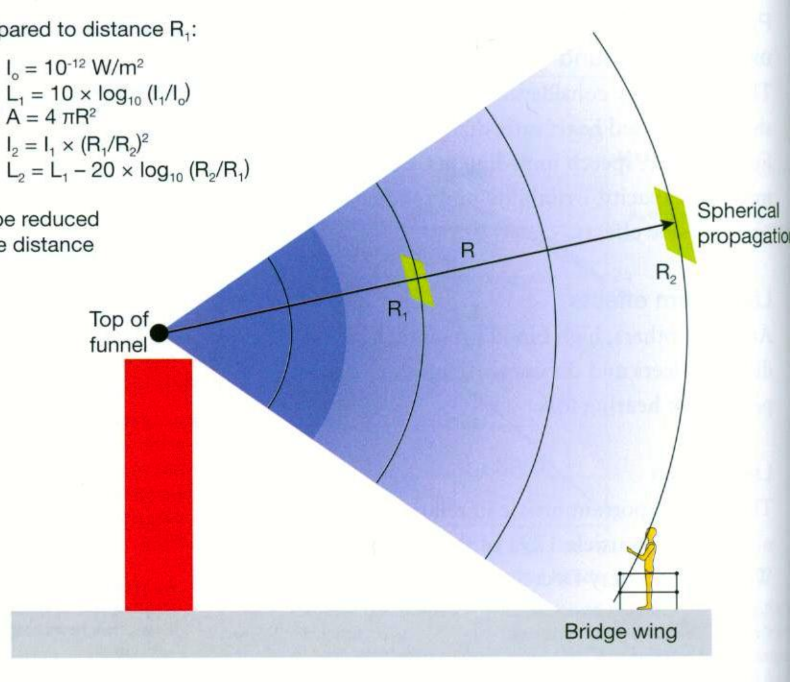
As a rule of thumb, with a doubling of the distance from the source of noise, the sound level is reduced by 6 dB (A), a significant difference!

Sound level – far field law

Sound level at distance R_2 compared to distance R_1:

Reference sound intensity	$I_o = 10^{-12}$ W/m²
Sound intensity level (dB) at R_1	$L_1 = 10 \times \log_{10} (I_1/I_o)$
Area of sphere	$A = 4\pi R^2$
Sound intensity at R_2	$I_2 = I_1 \times (R_1/R_2)^2$
Sound level at R_2	$L_2 = L_1 - 20 \times \log_{10} (R_2/R_1)$

In general, the sound level will be reduced by 6 dB for each doubling of the distance from the noise source.



16.4 Silencers for diesel engines - Choosing a silencer

There are many different versions. For two- and four-stroke engines, for instance, these are explosion-proof silencers with spark arresters and baffles for the different frequencies generated by engines.

Note

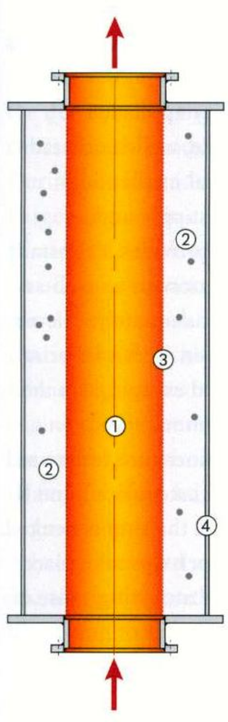
Exhaust systems of combustion engines are subjected to pulsations and other forms of vibrations. For this reason, the entire exhaust system is mounted vibration free using suitable vibration insulators.

16.4.1 How do silencers work?

The following principles apply:

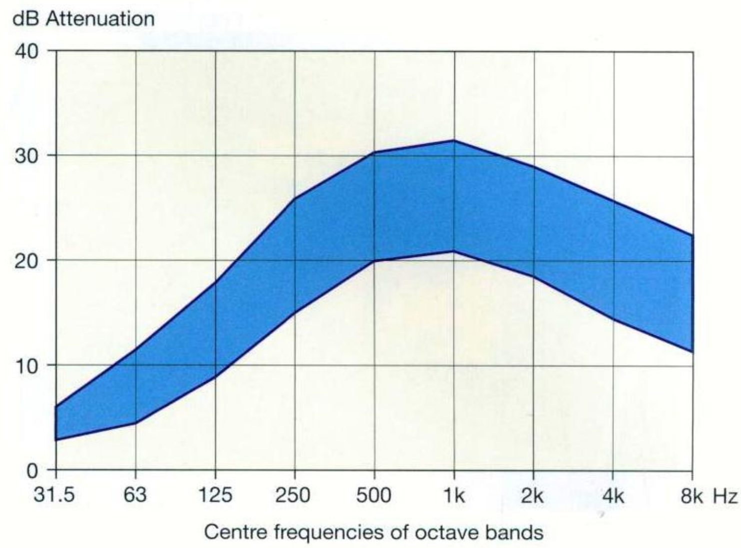
Adsorption silencers and throttle silencers. The gases flow along sound absorbent materials such as non-flammable mineral wool which significantly reduce the sound levels. These are suitable for frequencies between 150 and 5000 Hz but cannot be the first silencer in the exhaust system.

Reflection silencers. The silencer is constructed is such a manner that the sound is reflected inside the silencer and not carried out via the exhaust. The sound waves created inside the silencer cancel out each other and therefore the sound decreases. It is often the first silencer in an exhaust system. The various silencer types are often combined.



◄ A conventional absorption silencer for a two-stroke crosshead engine.

1 *exhaust gas manifold*
2 *non-flammable mineral wool*
3 *holes-pipe*
4 *steel outer wall*



▲ A good example of sound reduction in a silencer at various frequencies. Approximately 25 to 30 dB (A) at 500 to 1000 Hz is viable.

Horizontally: Sound frequency Hertz
Vertically: The decrease of sound in dB with a sound silencer

Silencers that are a combination of both principles are often used.

16.4.2 Interiors of sound silencers

Sound in a silencer is reduced to levels so that the sound produced by the diesel power plant or ship does not exceed the legally established level, as measured outside the engine.

The sound emitted by the diesel engines travels into the exhaust-gas manifold and enters the silencer; it is first led through the expansion chamber, where low-frequency impulse sounds are eliminated. Then the gases flow into the absorption section where the remaining medium- and high frequency sounds are dampened by special sound absorbent materials.

Materials

The casing and expansion section are often manufactured from carbon steel with a thickness of at least 3 mm. The absorption section contains sound-absorbing, non-inflammable mineral wool of various densities.

Model selection

Using a chart of the mass flow in kilograms per hour and the temperature of the exhaust gases, the volume flow in m³ per hour can be determined.

In a following chart, the correct model can be chosen by intersecting the pressure loss (horizontal line) and the discharge (vertical line).

In the last table the weights and measurements of the models are given.

16.6 Turbo-blower noise

The first prerequisite in order to increase the power output of diesel engines is the increase of the air supply for the combustion process. More air means more fuel can be injected and therefore more power can be generated. The supply of a sufficient amount of air to the cylinder in a short time-span through the inlet valves, in the case of a four-stroke engine, and the inlet ports in the case of two-stroke engines, requires a higher turbo-blower air pressure. The compression rate, of the centrifugal compressor of the turbo-blower, has increased throughout the years from 2 to approximately 4.5. The turbo-blower capacity, the 'flow', has also increased considerably.
Sound measurement around the turbo-blowers of large two-stroke crosshead engines show values of 110 dB (A) at full load; at a slightly lower power output they can still be as high as 105 dB (A).
On passenger ships comfort is vital and therefore the noise levels of large four-stroke propulsion engines are kept to a minimum. On cargo ships, the sound levels are kept as low as possible for the crew's comfort.

Improvements that have been introduced by turbo-blower manufacturers

The improvement of the compressor wheel and the diffuser reduce sound at its source. A silencer consisting of an absorption- and a resonance section is fitted in the compressor exhaust outlet and significantly reduces the sound output.
In this way, noise levels have been reduced to approximately 95dB (A).

16.7 Sound levels in diesel engines

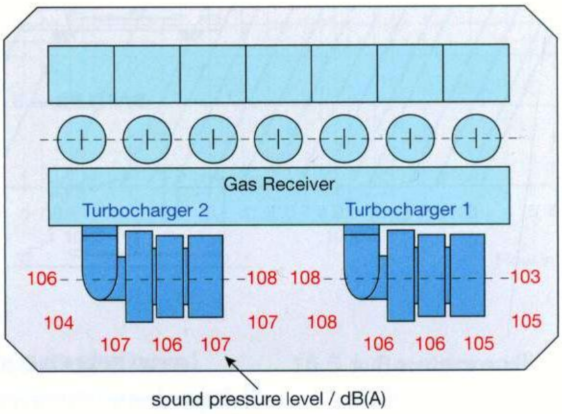
Examples of engine manufacturers.

16.7.1 Example 1 – Wärtsilä 38 A diesel engine, a category III

Measurements are taken using a specific method; in this instance one metre from the engine at the same height as the cylinder heads.

▶
The sound levels on the top floor near the cylinder heads and blowers of a two-stroke crosshead engine.

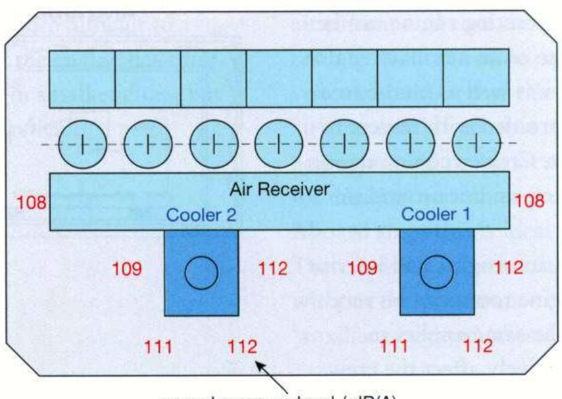
All measurements lie between 100 and 108 dB (A).



Measurements show that sounds does not remain in the turbo-blower only, but is carried to the inlet manifold and the air cooler. These sections may even reinforce the sound.

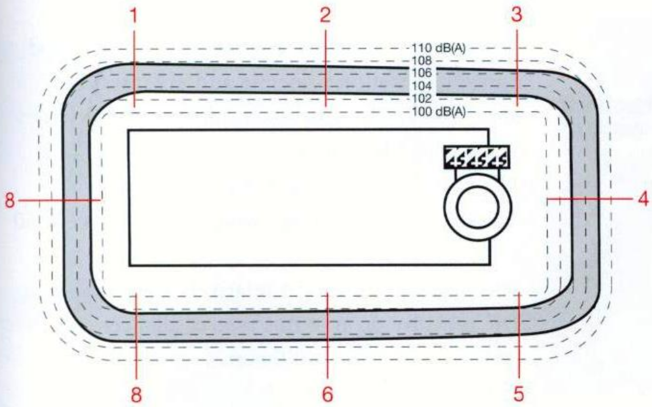
▶
The same engine. The noise levels are measured at the air-inlet manifold and at both air coolers.

This is the reason that the sound carried from the blower is boosted to a maximum of 113 dB (A).

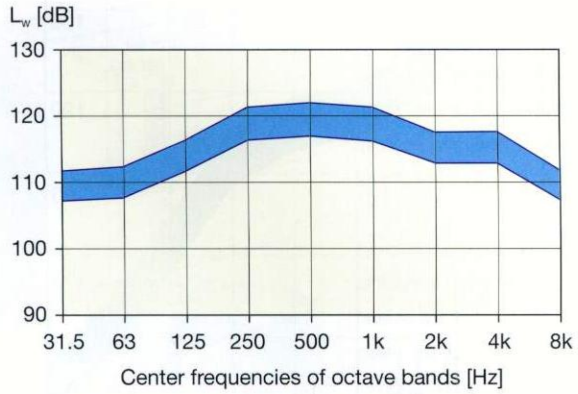


Air borne sound

8 measuring points, dB(A) re 2.10^{-5} N/m² at 1 meter from the Engine and at cylinder height.

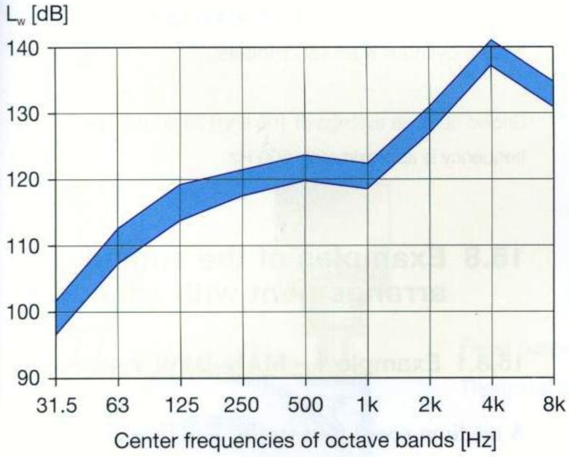


Air borne sound power level L_w (dB re 1.10^{-12} W).



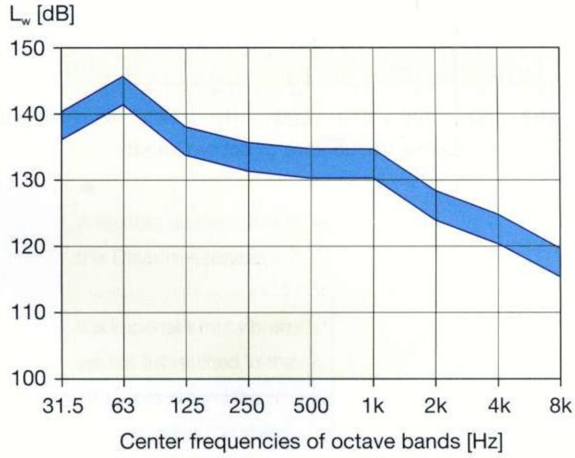
Inlet air sound

Unsilenced inlet air sound power level
$L_w = 10 \log P/Po$ dB (re $Po = 10^{-12}$w) after the turbocharger.

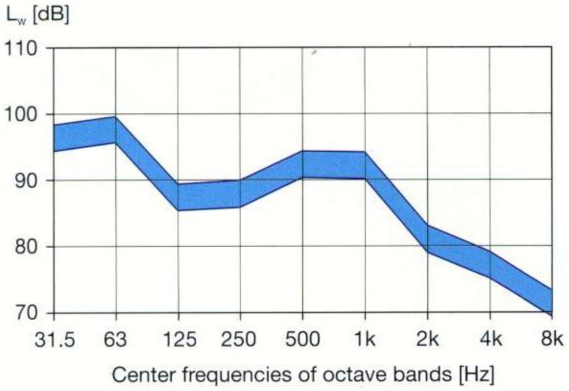


Exhaust sound

Unsilenced exhaust sound power level
$L_w = 10 \log P/Po$ dB (re $Po = 10^{-12}$w) after the turbocharger.



Structure borne sound



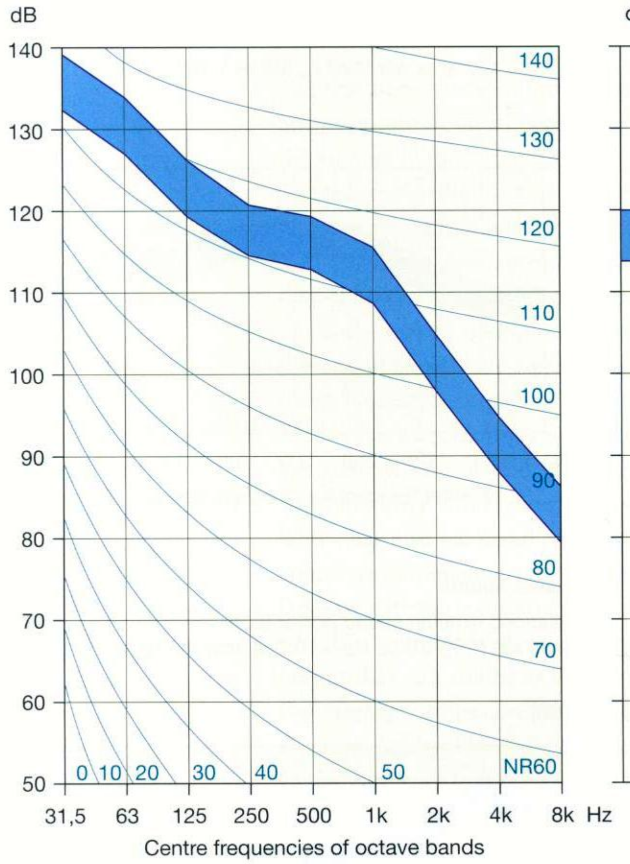
Average vibration velocity level L_v of the four corners of the eingine foundation flange in three directions. Reverence level $v_0 = 5.10^{-8}$ m/s.

$$L_v = 10 \log \frac{\sum_{i=1}^{n} (v_i / v_0)^2}{n} \text{ [dB]}$$

▲

Example of a Wärtsilä 38A diesel engine, category III.

Top left: measurement location; shown here the sound production lies between 100 and110 dB (A).
Top right: sound production in dB (A) and frequency in kHz, k = Kilohertz.
middle: sound production of the turbo-blower, a tremendous sound producer!
Left: an air inlet that is not soundproof.
Right: an exhaust that is not soundproof. Both produce high noise levels.
Bottom right: sound produced in three directions towards the bed plate measured at the four corners of the bed plate.



16.7.2 Example 2 – MAN–B&W K 98 MC, the second largest crosshead engine, engine category IV

The silencers for these engines are usually based on the absorption principle and are installed after the exhaust-gas boiler. The gas velocity is approximately 35m/sec.

The sound levels of the exhaust gas that is discharged on the bridge wing may not exceed 60 to 70 dB (A).

This is achieved using a relatively simple silencer. The pressure drop of this type of silencer is approximately 200 millimetres water column at M.C.R. power output.

◄

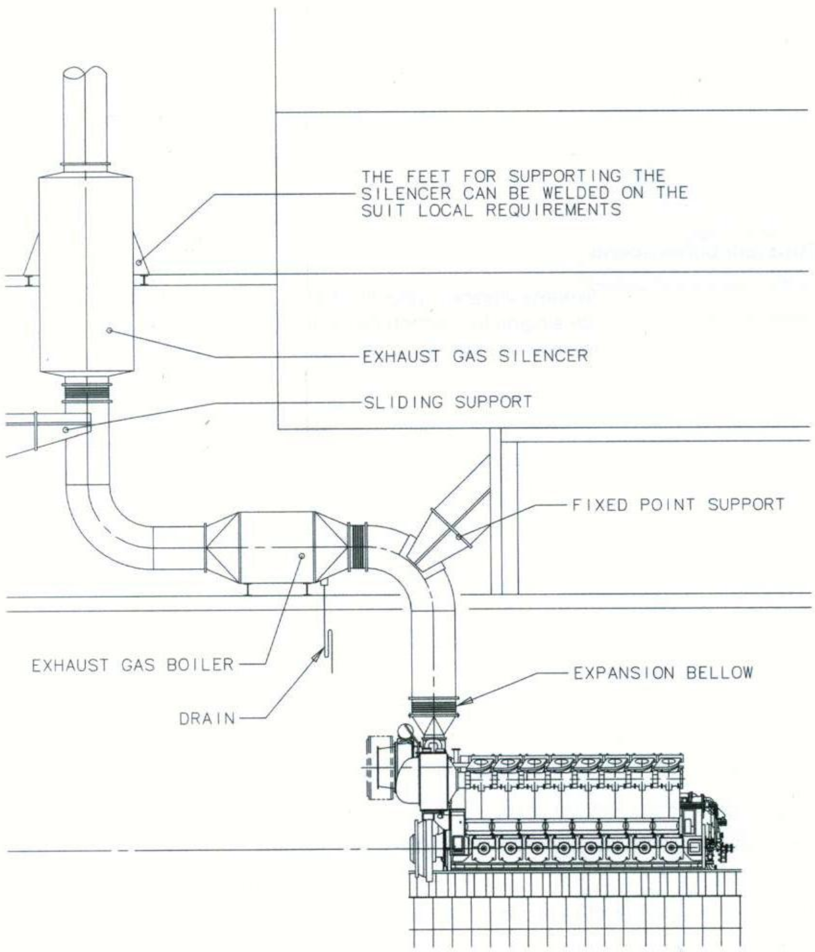
The sound in the exhaust-gas system of a two-stroke crosshead engine from MAN–B&W for the six-and twelve cylinder K 98 MC models.

Shown here, an average of 105 and120 dB (A). The frequency is approximately 500 Hz.

16.8 Examples of the engine arrangement with silencers

16.8.1 Example 1 – MAN–B&W V 40/50

A medium-speed four-stroke diesel engine – engine category III.

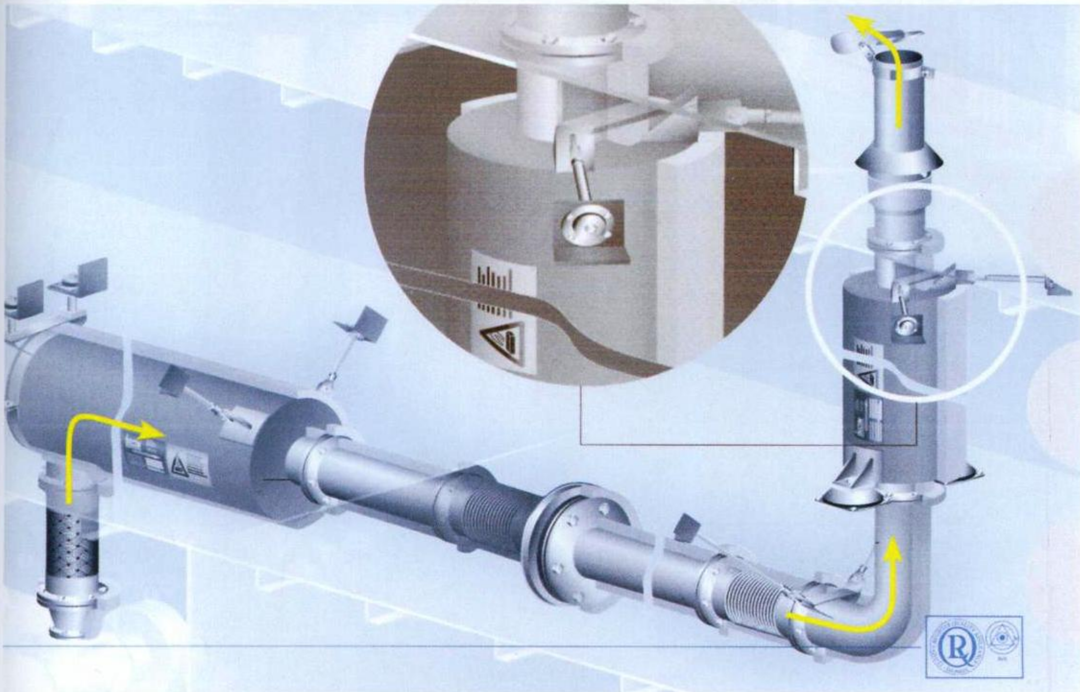


◄

An engine alignment with a silencer.

As shown here, one can appreciate the huge exhaust system dimensions in comparison to the engine when an exhaust-gas boiler and a silencer are installed. Three exhaust-gas compensators absorb the expansion and contraction of the pipes due to temperature fluctuations.

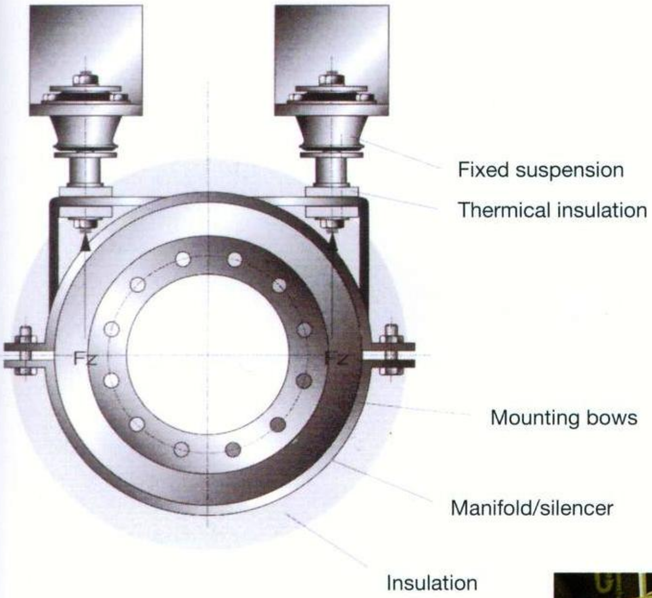
16.8.2 Example 2 – Arrangement of a complete exhaust gas system with two Discom silencers



◄

The arrangement of a complete exhaust-gas system with two Discom silencers.

Apart from the correct type of silencers, the flanges, the flexible sections and the rain cover are essential in order to obtain an adequate overall system.



◄

A flexible suspension of the Discom silencer.

It is important that vibrations are not transmitted to the structures around the exhaust gases manifolds.

►

The flexible suspension of the exhaust-gas manifold of a Wärtsilä 38 A diesel engine, category III.

▶
An inventory of silencers at Discom.

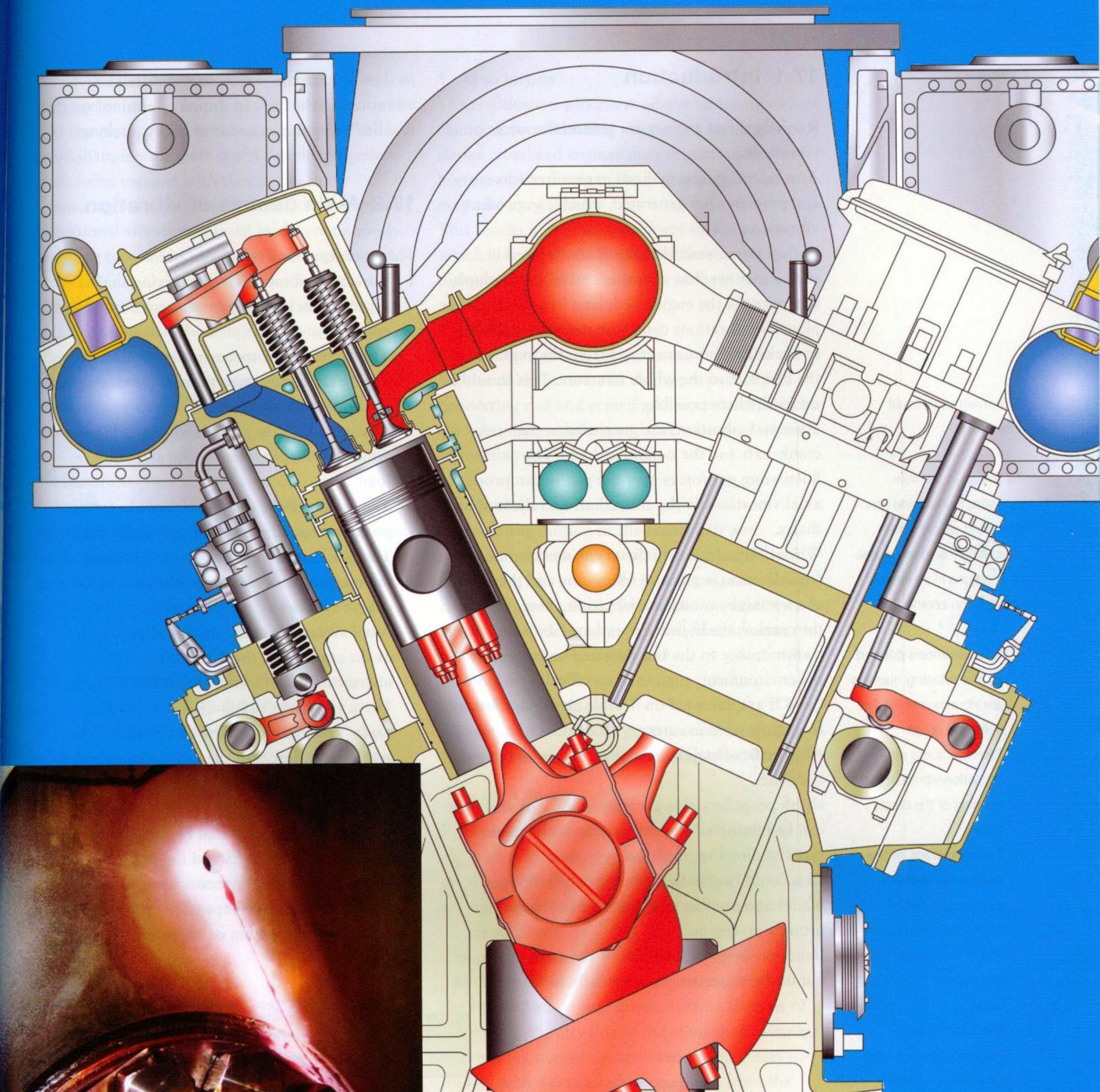
In the foreground small silencers with spark arresters (1).

>CH 17

Vibrations and Balancing







Cracks as result of the forces and vibrations in the welded A-frame of a two-stroke crosshead engine.

The combustion process above the piston produces vibration which are difficult to combat. These vibrations are generated by the pressures which fluctuate constantly during the proces.

17.1 Introduction

Running diesel engines are potential sources of vibrations.
Around an operating diesel engine free forces and moments are generated, which can produce vibrations.
In ship propulsion, the ship is constructed in such a manner that vibrations travel to the ship's structure via the engine bedplate. In a diesel power plant the vibrations can cause the entire building to shake as vibrations travel from the engine via the bedplate to the whole structure. This should be avoided where possible.
Torsional vibrations are generated in the crankshaft and the outgoing shafts. The fluctuating gas forces over the piston can produce axial vibrations in the crankshaft and the outgoing shafts.
When designing a new diesel engine type, much consideration is given to vibrations and their absorption by mounting vibration dampers on the crankshaft. In order to reduce vibration transmittance to the bedplate and consequently the environment, engines in specifically categories I and II are mounted on flexible elements. This is frequently used in category III engines.
All crosshead engines, category IV, are fixed to the bedplate.
Ship's propellers also generate vibrations and these can be reinforced in combination with certain types of diesel engines.
This could be due to the number of propeller blades and number of cylinders used in the propulsion engine.
In heavy seas, it is possible for the stern protrude out of the water and subsequently for the hull to slam against the waves, so producing heavy vibrations in the hull. In shipping terminology this is called slamming, a result of heavy pitching.

17.2 Main causes of vibration

There are three sources of vibrations.

1 **Torsional vibrations in the driving shafts,** especially in low-speed two-stroke crosshead engines.
The diesel engine torque changes constantly during the combustion process for two-stroke or the four-stroke cycles.
As opposed to the steam and gas turbines, which have a constant torque at a constant load.
The cranks of the crankshaft are also positioned at a certain angle of each other.
The position of these cranks is dependent on the combustion process and the number of engine cylinders.

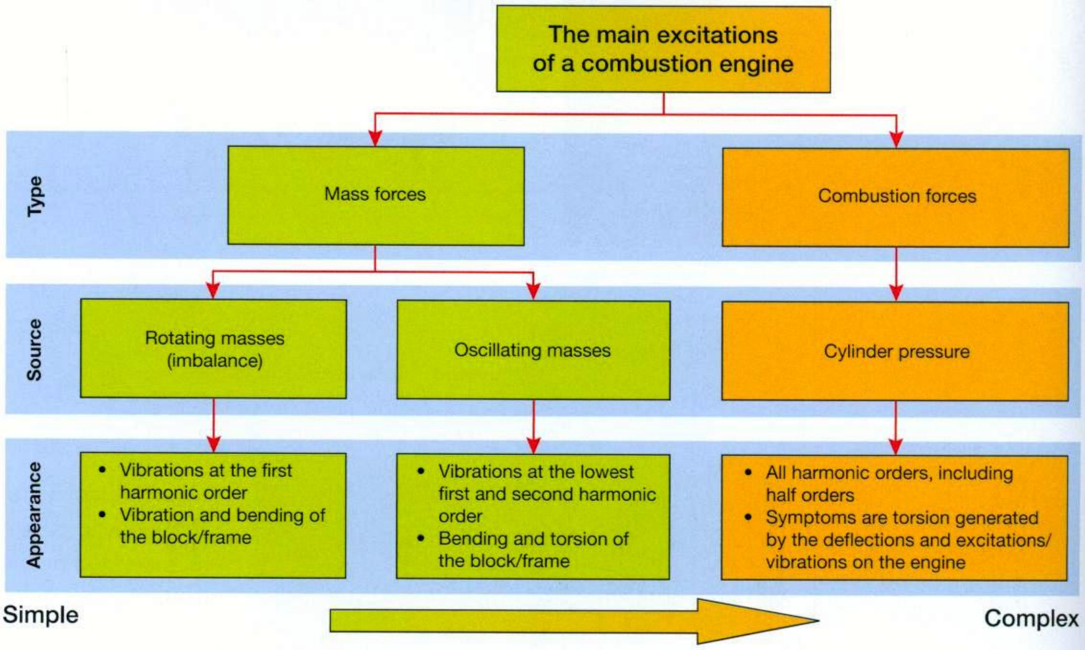
2 **Axial vibrations in the driving shafts,** such as the crankshaft, the thrust block, the intermediate shafts and the propeller shaft.
These vibrations originate from the radial loads generated from the gas forces and the acceleration forces in the individual cylinders as well as the axial forces exerted on the propeller.

3 **Bending vibrations** caused by misalignment of the intermediate shafts and the propeller shaft.
These deform by compression and tension forces and take the shape of the ship. This is referred to as hogging or sagging.

▼
The main causes of vibrations.

As shown in the flow diagram, the left side can be solved simply when evaluating the main causes of vibrations in an internal combustion engine.

- **The vibrations caused by the rotating masses** are vibrations in the first harmonic order.
  The symptoms are excitation/vibration and bending of the block/frame.
- **The vibrations of the oscillating masses**, such as the piston movement, are complex and in the lowest, first and second harmonic order.
  The symptoms are vibration, bending and torsion of the block/frame.
- **The vibrations caused by the changing pressure in the cylinder** have all the harmonic orders, also the half orders.
  The symptoms are torsion generated by the deflections and excitations/vibrations in the engine.
  This is a complex and not simply solvable problem.



17.3 Resonance

The complete driving system consisting of the crankshaft, intermediate shafts and the propeller shaft form, together with the pistons, and connecting rods, an elastic system that is subjected to torsional vibrations caused by the continuously changing torque exerted on the driving gears and cranks.
This can produce dangerous vibrations, which can result in crankshaft breakage (torsional break).
This can occur due to **resonance**, when one of the elastic system's vibration frequencies matches the frequency of one of the moments that generates a vibration.

Resonance

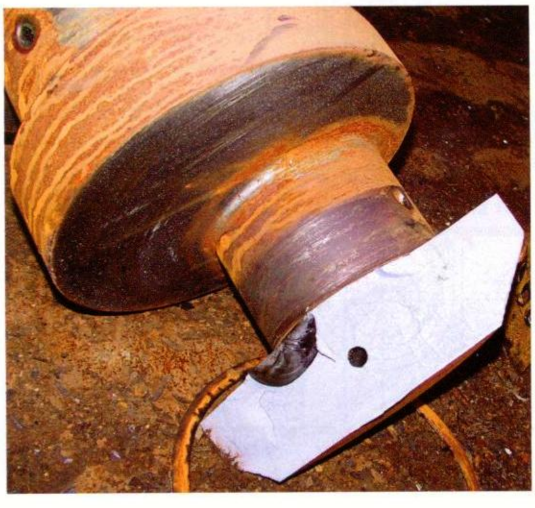
Vibrations of large amplitude produced by a relatively small vibrations near the same frequency as the natural frequency of the ship. The large amplitude vibration can lead to material fatigue which will result in breakage.

17.4 Forces exerted on the driving gear and engine block

The following are distinguished:
1 gas forces;
2 inertia forces;
3 mass forces.

▼
A broken crankshaft.

The crank web is transversely torn. The lubricating oil drilling is clearly visible. Vibrations can cause every shaft to break. The large vibrations cause material fatigue followed by breakage.



1 Gas forces

The combustion process produces constantly changing pressures over the piston, and in two-stroke crosshead engines the pressures at the bottom of the piston in the scavenging-air space may also vary slightly.
This results in a downward force on the piston, which is equal to the pressure difference on the piston multiplied by the piston surface.

2 Inertia forces

Consisting of reciprocating forces due to the motion of the piston and the upper part of the connecting rod and centrifugal forces generated by the rotating crankshaft and the lower part of the connecting rod.

3 Mass forces

The rotating motion of the crankshaft and reciprocating movement of the piston with the connecting rod in four-stroke engines; and the connecting rod, crosshead and piston rod in two-stroke crosshead engines, produce acceleration torque leading to mass forces.

Crankshaft

The rotating crankshaft generates centrifugal forces which are equal for every crank.

$$F_C = m_R \times \omega^2 \times r$$

F_C = centrifugal force
m_R = mass rotating crankshaft
ω = angular velocity of the crankshaft
r = distance from the centroid to the axis of rotation

The centroids of all these rotating engine parts are situated at varying distances from the axis of rotation, the centre line of the crankshaft. Simply, the rotating masses are replaced by imaginary masses with a crank radius equal to r from the crankshaft centre line on which inertia forces identical to the original are exerted.
The masses of the drive gears play a part in the tangential force diagram and the load of the crosshead, crankpin and the crankshaft bearings.

The value of these forces can be calculated as follows.

$$F = m \times g$$

F = mass force
m = mass of part
g = acceleration of gravity/gravitional constant.

In low-speed two-stroke crosshead engines the acceleration forces which work on the rotating mass are between 10 and 25 times higher than the weight of the rotating mass.
In a medium-speed engine these are 75 to 100 times higher and for a high-speed engine they are negligible.
This is the reason that the loads of, for instance, bearings in engine categories I, II and III are not included.

17.5 Principle of an internal combustion engine

The following can be stated for a crankshaft and connected reciprocating piston in a cylinder:

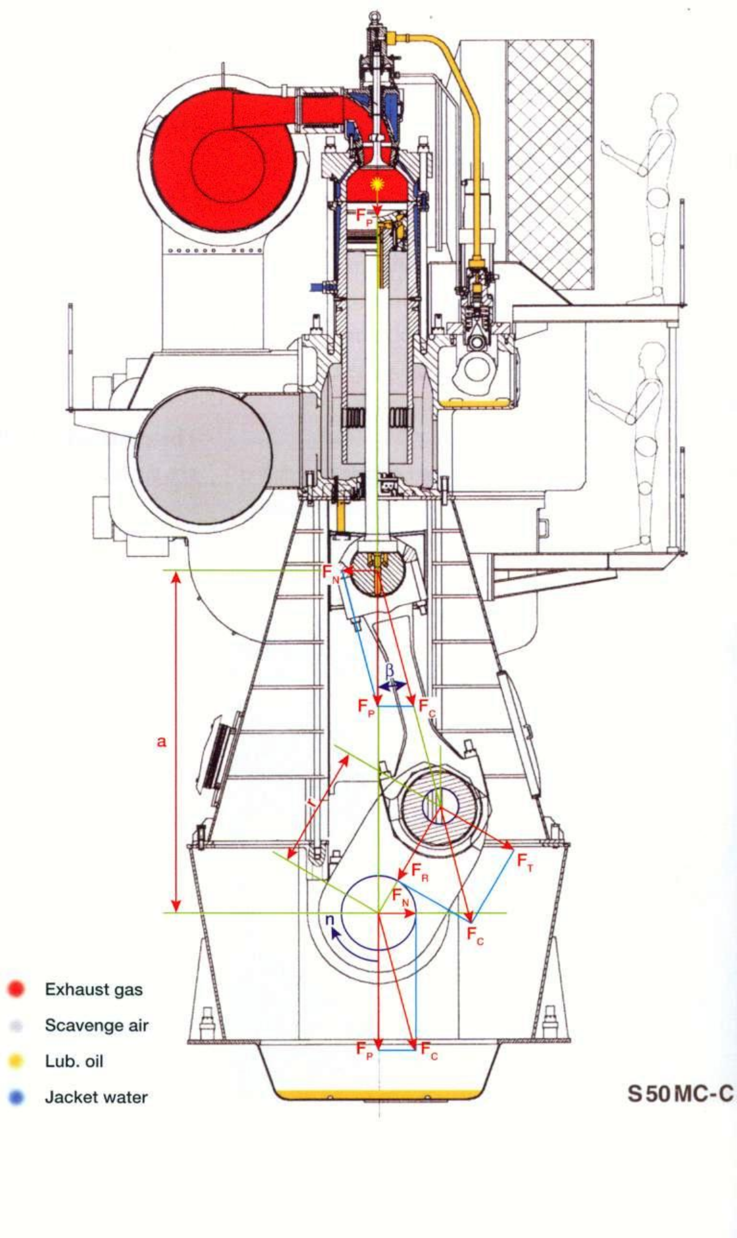
- the crankshaft rotates, as do the camshaft(s) and the gear train connecting them.
- the piston moves between the top dead centre position (T.D.C.), and the bottom dead centre position (B.D.C.) where it comes to a complete standstill!

Between these two positions the piston accelerates and decelerates continually, as does the connecting rod in a four-stroke engine or in a two-stroke crosshead engine, the connecting rod, the crosshead with guide shoes and the piston rod. The accelerating and decelerating forces can be very high, depending on the number of crankshaft revolutions and the masses of the reciprocating parts.
Therefore the piston is designed to be as light as possible.

17.6 Forces in a two-stroke crosshead engine

The following forces and moments occur:

1 tilting moment $F_N \times a$
2 turning moment $F_T \times R$
3 force exerted on the piston F_P
4 force exerted on the cylinder cover $F_H = F_P$
5 acceleration force F_A of the reciprocating parts
6 centrifugal force F_C of the rotating parts



▶
The forces exerted by gas pressures in a running engine.

Designation:
F_N = guide force
a = distance from the heart of the crosshead to the heart of the crankshaft
F_T = tangential force
r = crank radius
α = travelled crank angle

Acceleration force Fa

The accelerating force Fa is linear and gives momentum to various parts in the engine, such as the piston, piston rod, crosshead and part of the connecting rod, from the T.D.C. to the B.D.C. position.
This acceleration a, can be established with a formula (not deduced):

$$a = \omega^2 \times R \times \cos\alpha + \frac{R}{L} \times \omega^2 \times R \times \cos 2\alpha.$$

In which:
R = crank radius
L = connecting-rod length
α = travel through the crank angle

The acceleration force F_A can also be established with the formula (not deduced):

$$F_A = mass_{parts} \times a = mass_{parts} \times \omega^2 \times R \times \cos\alpha + mass_{parts} \times \frac{R}{L} \times \omega^2 \times R \times \cos 2\alpha$$

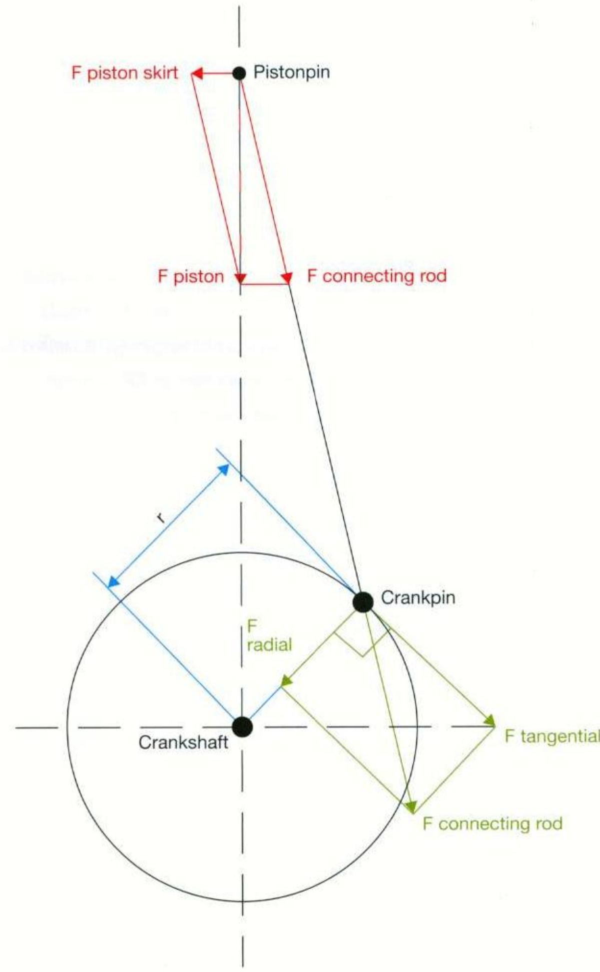
Forces exerted on the piston and the cylinder cover

Generally, these are usually of the same magnitude and opposing. Resolution of the piston force in the centre of the crosshead produces a force in the connecting rod, F_C, and a horizontal force, F_N, on the crosshead guide.
In a four-stroke trunk-piston engine this piston force is resolved in the centre line of the gudgeon pin and F_N, becomes the force in the piston skirt. Vertical accelerating and inertia forces are also active in the engine frame.
The piston force is partially used for the acceleration force F_A to provide momentum for the moving parts. A force of $F_P - F_A$ remains as the tangential force and the guide force. So this force is exerted in the direction of the engine bedplate. The force F_H is larger than the force $F_P - F_A$, so the engine has the tendency to move upwards. This force is not balanced in the engine and is therefore referred to as a free force.
So, during deceleration of the moving parts, a free force, directed downwards is generated. As these free forces fluctuate periodically, the diesel engine tends to bounce up and down, thus causing vibrations in the diesel engine and the ship via the bedplate.

17.7 Tangential force diagram

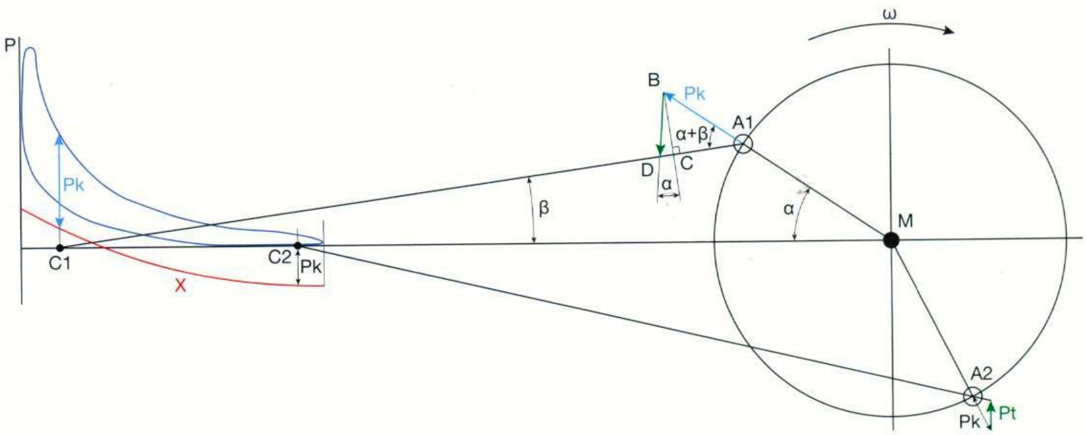
The crank-connecting rod mechanism of the diesel engine is loaded with a force F_P on the gudgeon pin in a four-stroke engine and with a force F_P on the top of the piston rod in a two-stroke engine. The pressure on the piston varies considerably and as a consequence the force F_P also changes constantly.

Calculations show that F tangential is approximately equal to F piston.
The development of these tangential forces can be observed in a diagram, with a length which is the circumference of the crank circle 2 × π × R.



▲
The gas force F_P exerted on the piston results in force F_C acting on the connecting rod and force F_N, acting on the piston skirt.

This connecting-rod force F_C when exerted on the crankpin results in a tangential force F_T and a radial force F_R.
As the gas force exerted on the piston changes constantly, all the other forces also constantly change in magnitude.
By the ever-changing piston position and so the crank, the direction of the tangential force also continually changes.



▲

A tangential-force diagram for a two-stroke crosshead engine.

Left, the indicator chart or banana and right the crank circle. In these diagrams, one works with pressures not forces. The crank MA is in a random position at an angle α and with the vertical axis (here shown shaft horizontal line MC).

AM = crank length

AC = connecting rod length

At crank position A1 the connecting rod length AC is circled and cuts the horizontal line of the diagram at C1.

At crank position A2 the connecting rod length AC is circled and cuts the horizontal line of the diagram at C2.

Line x is the reaction of the mass forces translated to pressure. Above the zero line of the operating diagram, the mass forces are assisting; below, they are counteractive. Measuring the crosshead pressure from the working diagram at crank position A1/C1 and plotting them on the extension of MA1.

A line is dropped between point B and the connecting rod A1-C1.

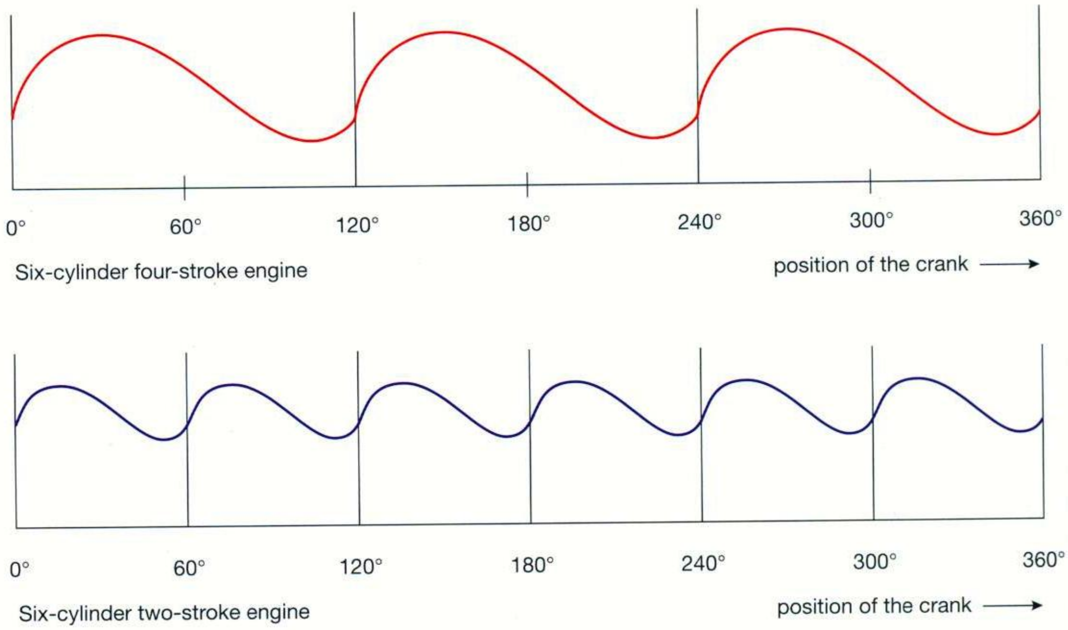
Angle β, between the connecting rod A1-C1 and the 'vertical' MC is placed left of the perpendicular line BC and PT, the tangential pressure is found. This can be found for every crankshaft position.

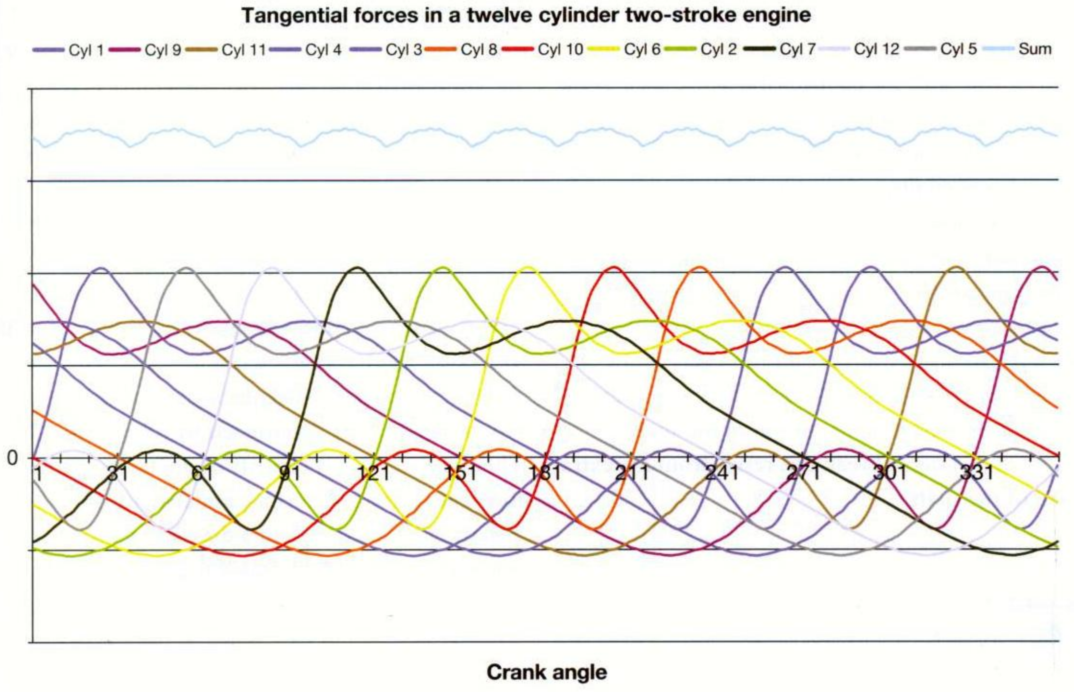
In crank position A2 the tangential pressure is directed upward and so exerts a negative pressure on the shaft.

▶

Tangential diagrams.

above: tangential force diagram for a six-cylinder four-stroke engine
below: tangential force diagram for a six-cylinder two-stroke engine
With both only the positive tangential forces are shown. **Noticeable** is that the tangential force in the two-stroke engine varies less than in the four-stroke engine.





The tangential forces in a twelve-cylinder two-stroke engine, a Wärtsilä RTA 96 C.

The ignition sequence is shown in the figure above: 1-9-11-4-3-8-10-6-2-7-12-5

Horizontal: One revolution of the crankshaft and so one process in the two-stroke engine.
Every cylinder has a different colour and one line is equal to one complete revolution and shows the tangential force at that moment.
The blue line above in the figure is the sum of all tangential forces at that specific point.
Because of the large number of cylinders, this line could be called regular.

Vertical:
Above the zero line 'positive' tangential force – directed downward.
Below the zero line 'negative' tangential force – directed upward.

Deceleration and acceleration rates

The deceleration and acceleration rates and therefore the divergence from the mean angular velocity (read RPM) will decrease when, the mass inertia moment of the rotating parts, such as the crankshaft, cam-shaft, cog wheels, fly-wheel and coupling shafts, increases. This is the reason for mounting the fly-wheel on the crankshaft.
A single-cylinder diesel engine has a rather irregular angular velocity and therefore requires a 'thick' fly-wheel. The more cylinders an engine has the thinner and lighter the fly-wheel.
In, for instance, an eighteen cylinder V-engine it is a thin disc, intended to start the engine and turn it during repairs.
The statement that an engine does not have a constant angular velocity ω is only partially correct.

17.8 Vibrations in engine frame and propeller shaft

Introduction

Construction of the crankshaft in a diesel engine is a complex matter.
The shaft has a complicated shape and it is therefore difficult to meticulously calculate the material tensions as a result of the torque exerted on the cranks. Moreover, additional material tensions may occur when the shaft vibrates.
Three types of vibrations can be distinguished:

1. Longitudinal vibrations, where the masses connected to the shaft periodically shift in relation to each other in the direction of the shaft.
2. Bending vibrations, where the masses connected to the shaft periodically shift in the direction perpendicular to the centre of the shaft.
3. Torsional vibrations, where the masses connected to the shaft periodically execute rotating movements around the centre of the shaft. This rotational vibration is superimposed on the normal rotation of the shaft. In this case, parts of the shaft positioned between the vibrating masses are torsion-loaded.

Each crankshaft, including the connecting shaft is to a larger or lesser extent subject to these vibrations; by means of the appropriate instruments the different vibration forms can be distinguished and subsequently registered.

For most types of diesel engines the **torsional vibrations** pose the most serious problems. Therefore much attention is paid to this form of vibration.
Apart from the crankshaft, vibrations occur in the entire engine frame. These vibrations produced by crosshead guide forces are divided into parallel, H-vibrations and/or X-vibrations.
Torsional vibrations originate from the fact that the torque exerted to the various cranks in the engine is not constant and varies considerably in amplitude.

The assembly of the shafting parts with the crankshaft, intermediate shafts and the propeller shaft forms with the connected masses, such as the cranks, pistons and rods and crossheads, fly-wheel and propeller, an elastic torque system that can vibrate torsionally, as a result of the constantly changing torque on the cranks. Dangerous vibration conditions may ensue with resonance, which means that one of the natural vibration frequencies of the elastic system corresponds with the frequency of the vibration generating moments. The vibration amplitudes can increase significantly resulting in tension in the shaft material. **A fatigue break can result from the extra load of the shaft.**

As will be demonstrated below, these resonance conditions occur at specific rotational velocities of the crankshaft. It is therefore sometimes necessary to avoid operating in one or more velocity areas. Furthermore, investigations are conducted to deduce which means and processes avert the production of intensive torsional vibrations, and the resulting material tensions.

The natural vibration frequency of a torsional vibration system

The natural vibration frequency of a torsional vibration system is determined by different factors. The simplest system consists of one mass, with a low mass inertia moment J, attached to the end of a shaft with length 1, and a diameter of which the polar inertia moment is equal to I_p.

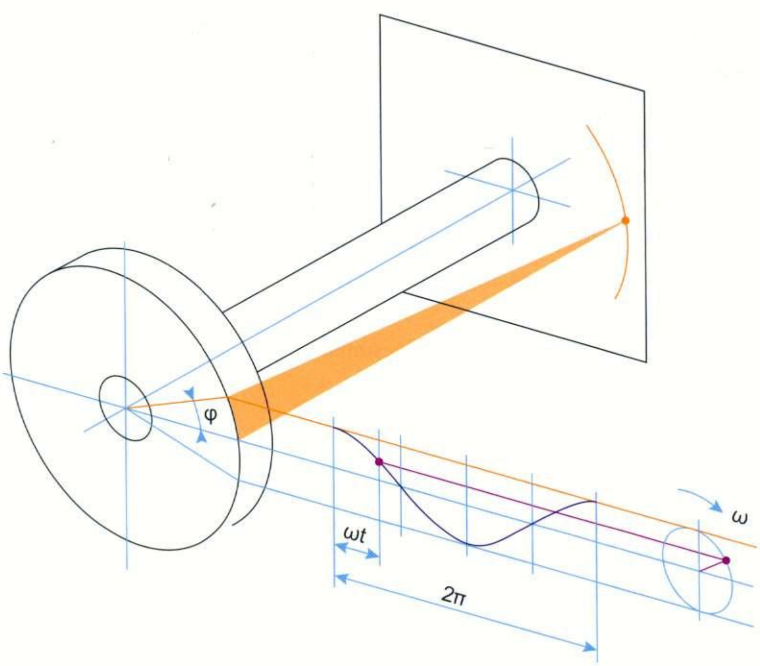
▼

The natural vibration frequency of a torsional vibration system.

An external moment T is exerted on the disk mass through which turns the disk over an angle φ (rad) from the middle position. The shaft is then loaded with a twisting tension and the following is valid:

$\varphi = T \cdot l / (G \cdot I_p)$

Where G is the shear modulus of the shaft material, the factor: $c = G \cdot I_p / l$ is shaft rigidity.



The shaft exerts a counter-moment on the disk with a magnitude of:

$-T = -\varphi \cdot (G \cdot I_p / l)$

If the external moment suddenly ceases, only the counter-moment will exert a force on the disk. The disk then has an angle acceleration $\underline{\varphi}$.
The movement comparison is then:

$T = -\varphi \cdot (G \cdot I_p / l) = J \cdot \varphi$ or: $J \cdot \varphi + (G \cdot I_p / l) \cdot \varphi = 0$

divided by J gives: $\underline{\varphi} + \varphi \cdot (G \cdot I_p / l \cdot J)$

This differential comparison is solved using the Laplace equation:

$s^2 \cdot \varphi + \varphi \cdot (G \cdot I_p / l \cdot J) = 0$
$s^2 + (G \cdot I_p / l \cdot J) = 0$
$s^2 = -(G \cdot I_p / l \cdot J)$
$s = \pm j \cdot \sqrt{(G \cdot I_p / l \cdot J)}$

So valid for reverse transformation standard: $s = \pm j \cdot \omega$
So the solution is $\omega = \sqrt{(G \cdot I_p / l \cdot J)}$ and this is called the natural circle frequency ω_e.

Often referred to as shaft natural vibration number and is:

$n_e = \omega_e / 2\pi$

The vibration frequency is:

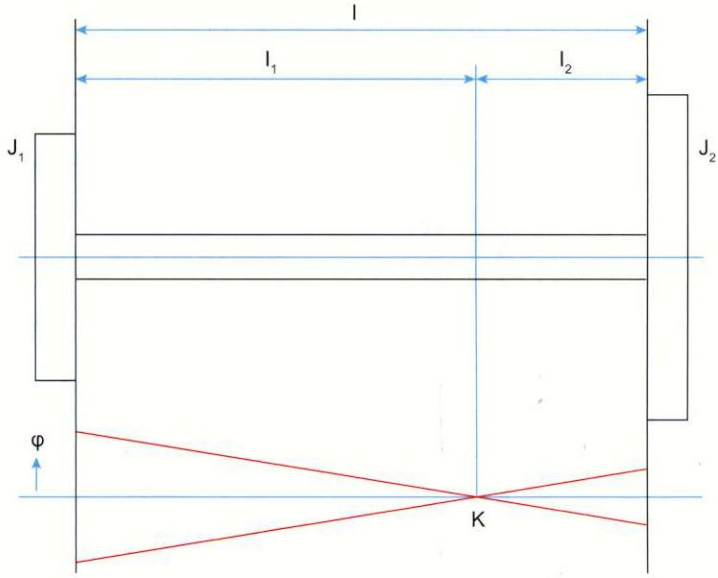
$n_e = 1/(2\pi) \cdot \sqrt{(G \cdot I_p / l \cdot J)}$

From the last formula it follows that a larger rigidity of the shaft leads to a higher natural vibration frequency; increasing the mass-inertia moment leads to a decrease in frequency. The amplitude of the vibration is not important for this calculation.

The natural vibration frequency of a shaft with two masses

▶

Torsional vibration system with two masses.



In the figure a shaft with a mass mounted on both ends is shown. When these masses are twisted in relation to each other, torsion occurs in the shaft, and thereby a counter-moment. With the release of the masses a free vibration occurs. The deflections of both masses are constantly contrary to one another. One shaft section will remain static. This section is referred to as the node of the vibration system.
At the node position, the shaft can be clamped in such a way that it does not influence the vibration. For both shaft sections, the formula for a shaft with a single mass applies as both sections have a vibration with the same frequency. Therefore the following is valid:

$1/(2\pi) \cdot \sqrt{(G \cdot I_{p1} / l \cdot J_1)} = 1/(2\pi) \cdot \sqrt{(G \cdot I_{p2} / l \cdot J_2)}$

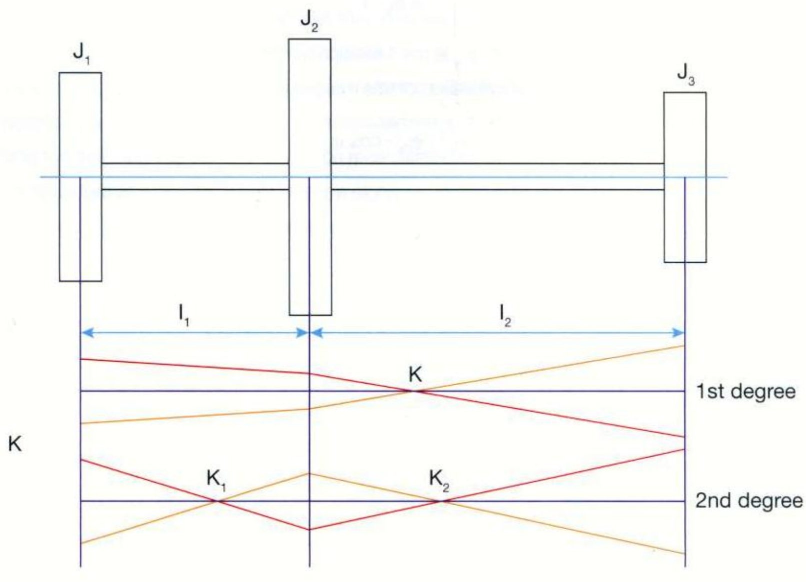
The calculation gives:

$n_e = 1/(2\pi) \cdot \sqrt{(G \cdot I_p / l \cdot J_v)}$ met $J_v = J_1 \cdot J_2 / (J_1 + J_2)$

The natural vibration frequency for a shaft with three masses

▶

Torsional vibration system with 3 masses.



A similar system can take different vibration modes:

a Mass 1 and 2 move opposite to mass 3; there is one shaft cross-section, the node. At rest, the node is positioned, in the example, between the masses 2 and 3. The natural vibration frequency for this vibration mode is referred to as being in the 1st degree.
b Masses 1 and 3 move opposite to mass 2 and a node in both shaft sections occurs. The natural vibration frequency belonging to these vibration modes is higher than that of a vibration with one node and the frequency is referred to as being in the 2nd degree.

The following relationship is again valid:
$n_{e1} = \omega_{e1} / 2\omega$ en $n_{e2} = \omega_{e2} / 2\omega$

Therefore:
$\omega_{e1}^2 = (p + q)/2 - \sqrt{(((p + q)/2)^2 - r)}$
$\omega_{e2}^2 = (p + q)/2 + \sqrt{(((p + q)/2)^2 - r)}$

In these comparisons:
$p = (G \cdot I_p/l_1) \cdot ((J_1 + J_2)/(J_1 \cdot J_2))$
$q = (G \cdot I_p/l_2) \cdot ((J_2 + J_3)/(J_2 \cdot J_3))$
$r = (G \cdot I_p)^2/(l_1 \cdot l_2) \cdot ((J_1 + J_2 + J_3)/(J_1 \cdot J_2 \cdot J_3))$

From this follows: $n_{e1} < n_{e2}$

Here the following applies: $x = l + r - l \cos \varphi - r \cos \theta$.

To eliminate φ consider that

$BB' = r \sin \theta = l \sin \varphi$, so $\sin \frac{r}{l} \sin \theta$. As generally $\sin^2 \alpha + \cos^2 \alpha = 1$ and therefore

$\cos \alpha = \sqrt{1 - \sin^2 \alpha}$ becomes $\cos \varphi = \sqrt{1 - \frac{r^2}{l^2} \sin^2 \theta}$.

The calculation is exact to this point. However, we will now apply an approach which is generally used for standard ratios for r and l.

Let's assume $\frac{r}{l} = \lambda \equiv$ crank - connecting rod ratio

$$x = r \cdot (1 + \frac{1}{\lambda} - \cos \theta - \frac{1}{\lambda} \cdot \sqrt{1 - \lambda^2 \cdot \sin^2 \theta})$$

$\sqrt{1 - \lambda^2 \cdot \sin^2 \theta}$ is according to Mc Laurin resolved as:

$$(1 - \lambda^2 \cdot \sin^2 \theta)^{½} = 1 - \frac{1}{2}\lambda^2 \sin^2 \theta - \frac{1}{8} \lambda^4 \cdot \sin^4 \theta - \ldots$$

The first terms usually suffice.

So: $(1 - \lambda^2 \cdot \sin^2 \theta)^{½} = 1 - \frac{1}{2}\lambda^2 \sin^2 \theta$.

As $\sin^2 \theta = \frac{1}{2} (1 - \cos 2 \theta)$ it follows that:

$$x = r \cdot (1 + \frac{\lambda}{4} - \cos \theta - \frac{\lambda}{4} \cdot \cos 2 \theta)$$

it then follows $\frac{d\theta}{dt} = \omega$ is the piston velocity v

$$v = \frac{dx}{dt} = \frac{dx}{d\theta} \cdot \frac{d\theta}{dt} = r \cdot \omega (\sin \theta + \frac{\lambda}{2} \sin 2 \theta)$$

and the acceleration of the piston

$$a = \frac{dv}{dt} = r \cdot \omega^2 r \cdot (\cos \theta + \lambda \cos 2 \theta).$$

The revealed expression for a is significant in studying piston engine dynamics.
On the basis of the equations for v and a, graphs can be drawn of v and a , as a function of time, or of the crank angle θ.

Crank - connecting rod mechanism

$$Sx = r (1 - \cos \alpha) + L\left(1 - \sqrt{1 - \frac{r^2}{L^2} \sin^2 \alpha}\right)$$

$\sqrt{1 - \frac{r^2}{L^2} \sin^2 \alpha}$ developed by means of $\sqrt{1 - x^2}$ Taylor or Mc Laurin?

$x = \frac{r}{L} \sin^2 \alpha \rightarrow -\frac{r}{L} \leq x \leq \frac{r}{L} \rightarrow$ so around 0!

(λ is currently 0.3 or 0.4, depending on the engine stroke.)

Developed by applying Mc Laurin!

$$f(x) = f(0) + \frac{f'(0)}{1!}x + \frac{f''(0)}{2!}x^2 \ldots$$

$$f(x) = \sqrt{1 - x^2} \rightarrow (1 - x^2)^{½}$$

$f(0) = 1$

$$f'(x) = \frac{1}{2} (1 - x^2)^{½} \cdot (-2x) = -x(1 - x^2)^{½}$$

$f'(0) = 0$

$$f''(x) = \frac{1}{2} \cdot \frac{1}{2} (1 - x^2)^{-⅔} \cdot 2x + (-2) \cdot \frac{1}{2} (1 - x^2)^{-½}$$

$f''(0) = -1$

Complete!

$f(x) = 1 + 0x \frac{1}{2} - - x^2 \ldots$ so $\sqrt{1 - x^2} \approx 1 - \frac{1}{2}x^2$

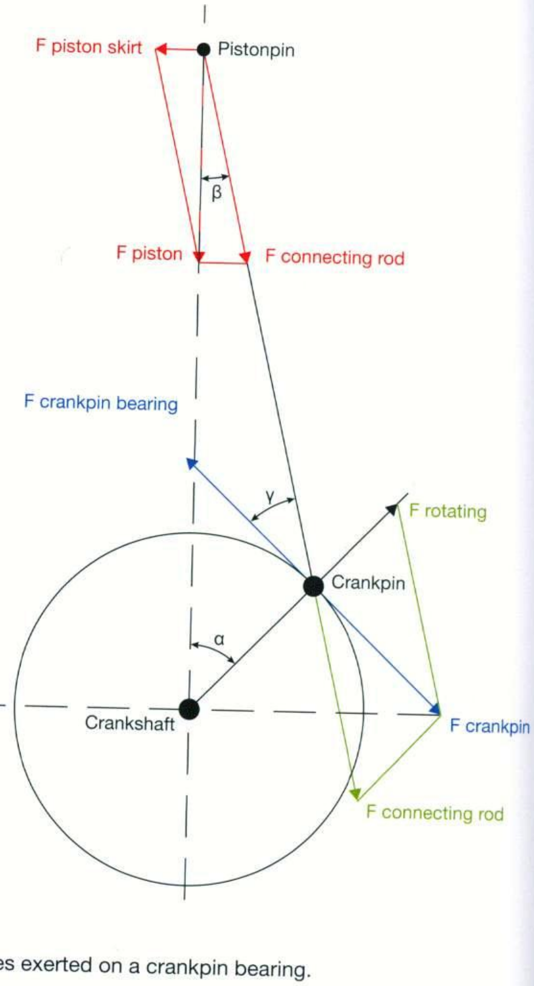
And $\sqrt{1 - \frac{r^2}{L^2} \sin^2 \alpha} \approx 1 \frac{1}{2} - \cdot \frac{r^2}{L^2} \sin^2 \alpha$.

Therefore:

$$S_x = r(1 - \cos \alpha) + \frac{r^2}{2L} \sin^2 \alpha.$$

Forces exerted on a crankpin bearing.

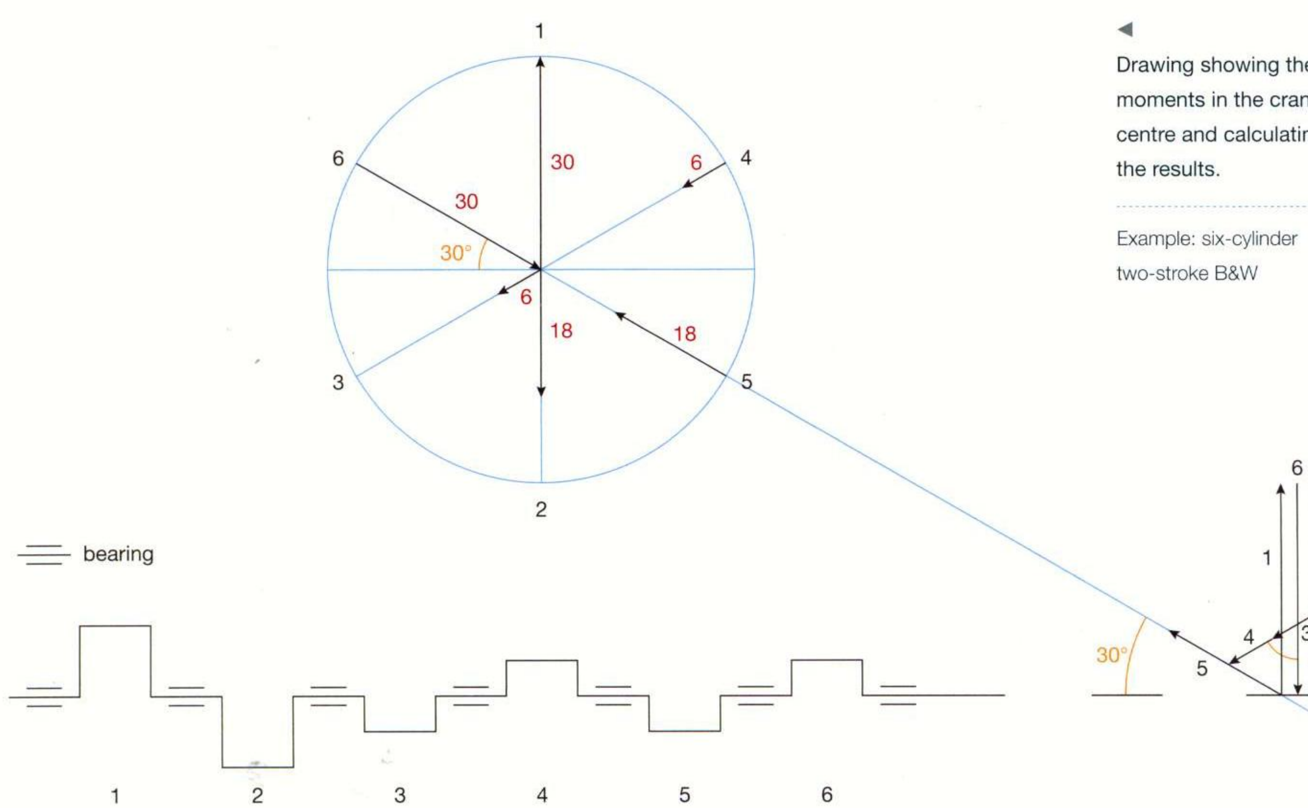
In order to obtain a clear picture of the forces exerted on the crank-pin bearing, it is assumed that the position of the piston during the power stroke is fairly random.



▲ Forces exerted on a crankpin bearing.

The force on the piston is resolved into a force exerted on the connecting rod F_{CR} and a horizontal force on the piston skirt and therefore the cylinder liner.
The rotating part of the connecting rod is placed in the heart of the crankpin.
The rotating force on this point is located on a line drawn through the heart of the crankshaft and the heart of the crankpin.
As F_{CR} is known, F_R and F_{KPL} can be either compiled or calculated.
So, the force on the crankpin bearing is opposite to the force on the crankpin and is at an angle γ with the connecting rod and this force is at an angle γ as shown in the polar load diagram for a crankpin bearing.
The force F_{CP} is subsequently used to establish the force in the main-shaft bearing. Here, it is of importance to remember that the force F_{CP} is distributed over the bearings at both sides of the crank webs! This should also be applied to the mass forces.

In order to determine the resultant moment, the forces are shifted to a so-called moment plane.
Generally, the forces will not produce a resultant moment, as most engines are homogeneous with respect to the masses and the crankshaft is usually centrally symmetric. The moments vary, but by drawing them in the centre of the crank and subsequently compiling them their moments can be determined.
Here the following sign convention is used: for cylinders in front of the moment's plane, the moment vector is in line with the crank; for cylinders behind the moment plane, the moment vector is opposed to the crank. This is applied to both the primary and the secondary crank centre.
If a considerable resultant moment remains, the reduction of the value of the resultant moment to a minimum can be attempted by adjusting the combustion sequence.
It should be remembered that optimum combustion sequence may give balancing problems with regard to torsion vibrations, bearing load and X and H moments which are exerted on the bedplate. The manufacturer should decide what is best given a certain situation.
Following are examples for establishing the resultant moment.

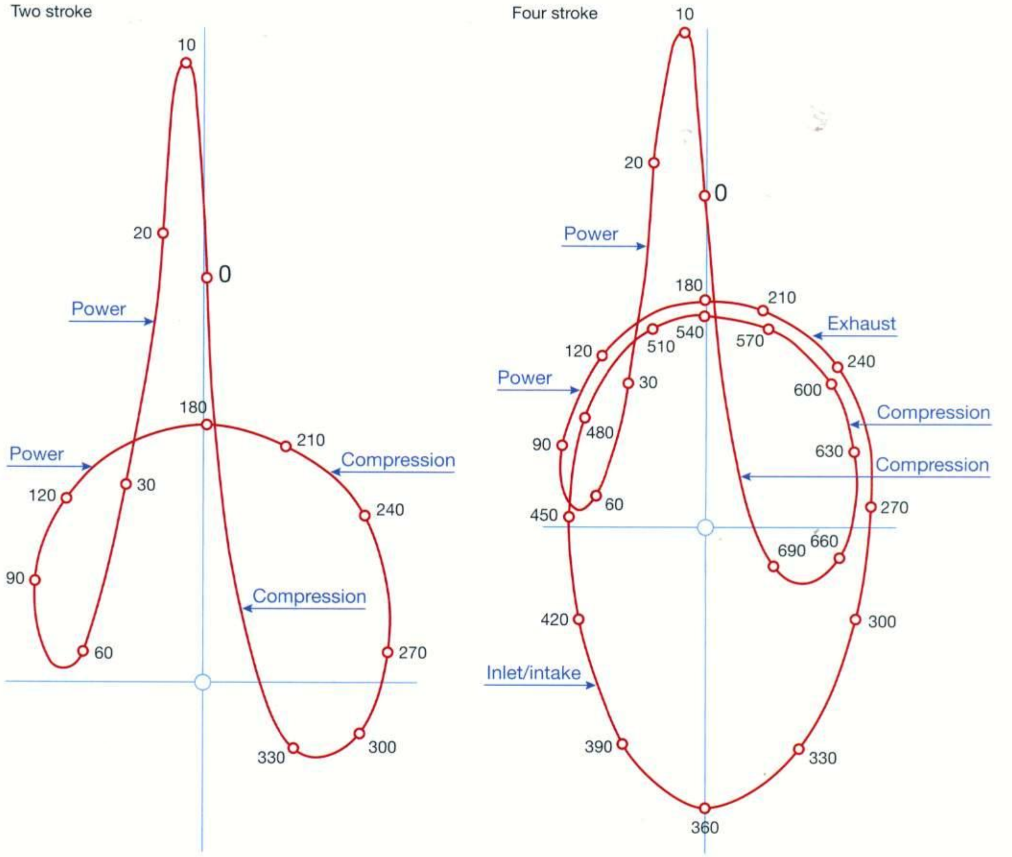


Drawing showing the moments in the crank centre and calculating the results.

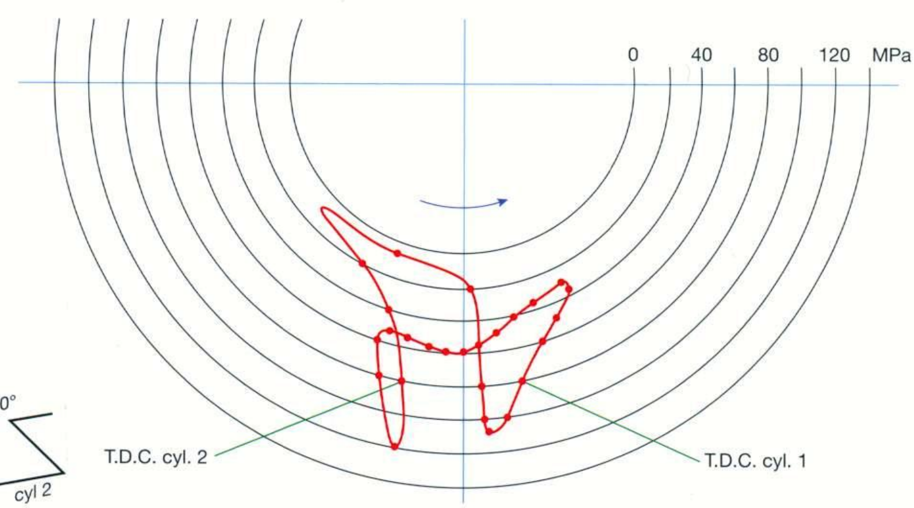
Example: six-cylinder two-stroke B&W

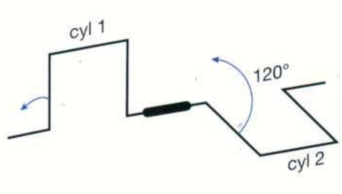
Polar load diagrams

►
Polar load charts of the crankpin bearings of a two-stroke (left) and a four-stroke engine (right).



►
A polar-load chart of a main-shaft bearing for a medium-speed four-stroke diesel engine of Wärtsilä, a Sulzer Z-40/48.





17.9 Degree of cyclic irregularity

Formula is:

$W_p = 0.5 \times J \times \omega^2$

W_p = power surplus provided by the combustion process
J = collective mass inertia moment of all the rotating parts
ω = angular velocity

From the formula it follows that: The fly-wheel may get lighter as the angular velocity ω increases, the so-called power surplus W_p, or the power shortfall is smaller.
The power surplus or shortfall can be reduced by distributing the capacity over more cylinders and ensuring that the angles between the corresponding cranks remain identical.

Example

Five-cylinder two-stroke engine:

angles $\frac{360°}{5} = 72°$.

Five-cylinder four-stroke engine:

angles $\frac{720°}{5} = 144°$.

The requirements with respect to the degree of cyclic irregularity are dependent on the apparatus driven by the engine.

Example

An alternator:

$$\delta = \frac{1}{200} \text{ tot } \frac{1}{300} = \frac{\omega_{max} - \omega_{min}}{\omega_{gem}}.$$

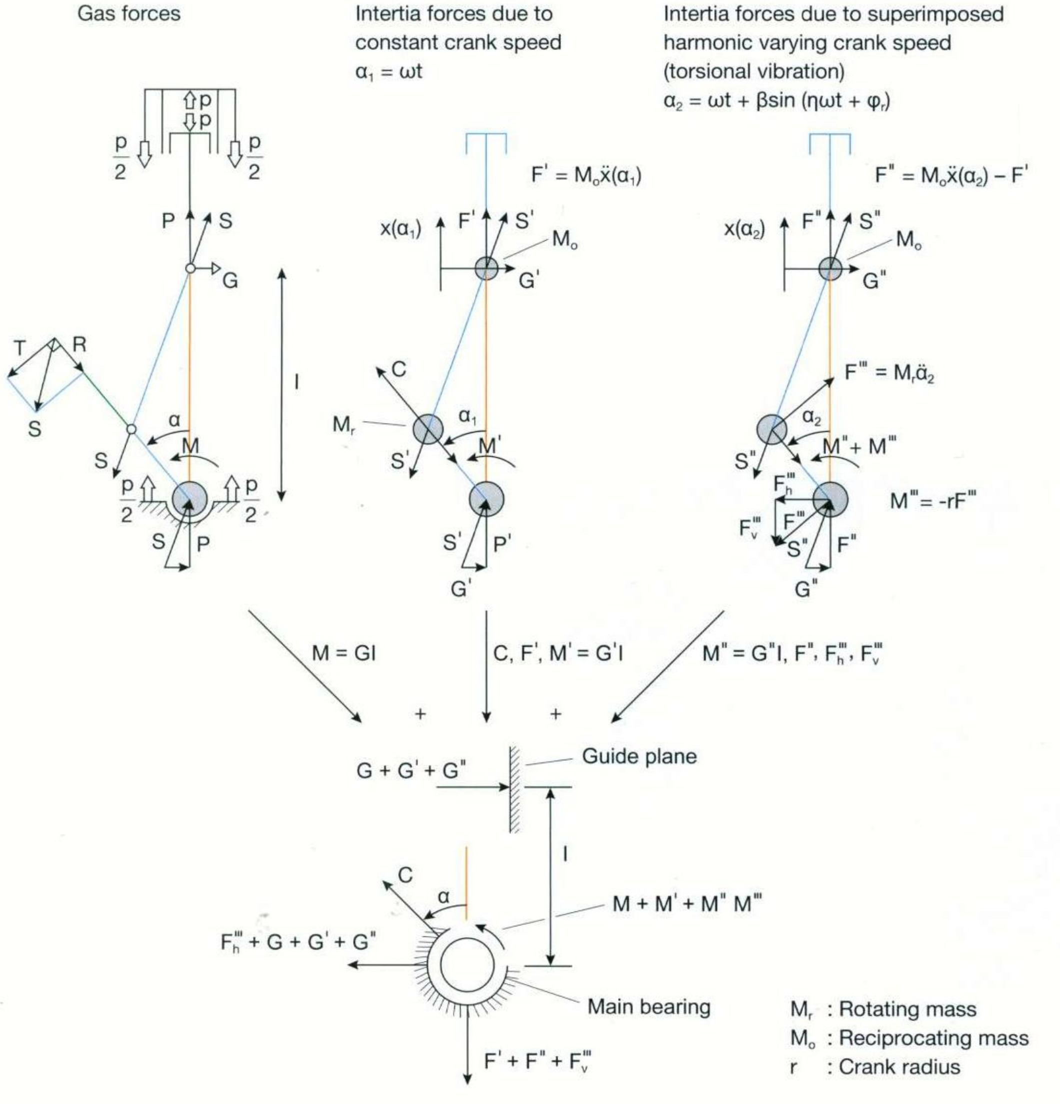
A water pump:

$$\delta = \frac{1}{50} \text{ tot } \frac{1}{100}.$$

17.10 Balancing diesel engines

Balancing engines is a matter of balancing the various forces and moments which are generated by an operating engine.
This is a complicated matter and the balancing of piston-driven engines can only be partially realised. It is simpler in a rotating machine such as a steam or gas turbine, an electromotor or the electrical genset for a centrifugal pump!

17.11 Resultant forces and moments in the engine block

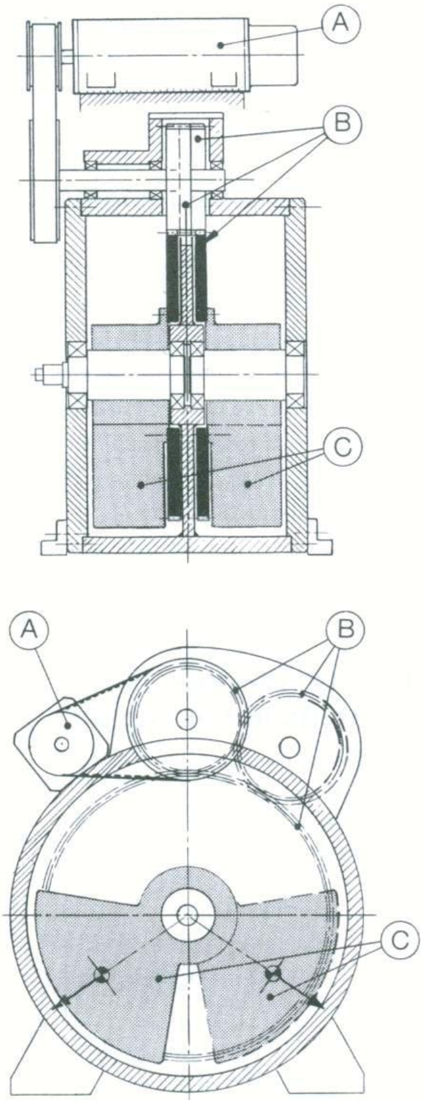


◄ The resulting forces and moments in the engine block of a two-stroke crosshead engine.

S, S^1 and S^{11}: Connecting-rod forces acting on the crosshead , equal to the connecting-rod force acting on the crankpin, equal to the force acting on the main-shaft bearings.
M , M^1, M^{11} and M^{111}: Torque on the main-shaft bearings from combustion pressure forces and inertia forces.
T and R: S, S^1 and S^{11}: At the crankpin, these are the sum of the radial component R and a tangential component T.
Resulting forces on the engine frame in vertical direction C, F^1, F^{11} and F_V^{111}.
Resulting forces on the engine frame in horizontal direction C and F_H^{111}.
Resulting moment on the engine frame $M + M^1 + M^{11} = l \times (G + G^1 + G^{11})$.

▶
A servomotor-driven compensator.

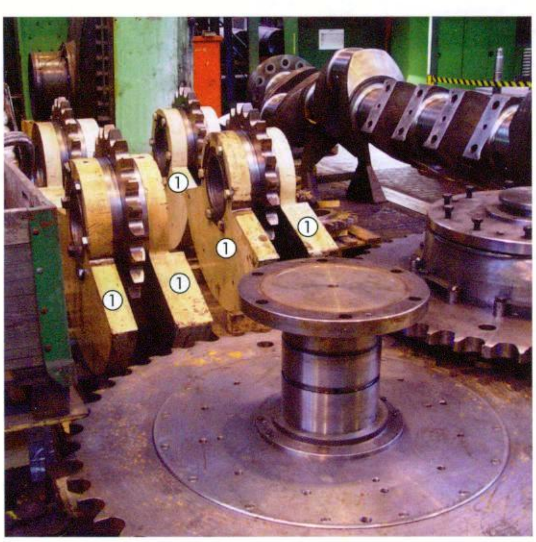
This is special vibration damper designed to reduce longitudinal ship vibrations in, amongst others, the superstructure.
The damper is designed to level out first and second order vibrations.
This damper is positioned aft, mostly in the steering-engine room as the vibrations are the largest here and so the effects are maximal.



A: AC Servomotor
B: Gear wheels
C: Flyweights

▶
The counterweights on the chain wheels in a MAN–B&W two-stroke crosshead engine.

1 *counterweights*



17.12 External forces and moments

The entire ship forms a mass-elastic system with a natural vibration frequency and generated vibrations. The horizontal and vertical bending moments of the hull construction and the corresponding natural (vibration) frequencies can be calculated. Calculating vibrations of four, five or more orders requires complicated software. Vibrations of two or three orders are easier to establish. In practice, the external forces for all MAN–B&W engines can be assumed to be zero, as their value is very low. Normally external moments of the first and second order are important.

17.12.1 Balancing the engine

Moments of the first-order work in a vertical as well as a horizontal direction.
With (the application of) MAN–B&W standard balancing these moments are of equal magnitude.
Moments of the second order only work in the vertical plane.
Procedures for four-, five- and six-cylinder engines should be considered.
In order to assess the extent of the external moments, the so-called Power Related Unbalance (PRU) system is applied.
The following applies to four-cylinder engines:

- **standard, adjustable** counter weights;
- **option**, a compensator for the moments of the first-order.

Resonance between the vertical moment and the second order of the vertical hull construction is often critical, whilst the resonance between the horizontal moment and the second order of the horizontal hull construction normally occurs at engine speeds higher than average.

The graph shows the connection between the shaft power and the vibrations of the first and second order.

horizontal: number cylinders = shaft power

vertical : Power-related unbalance in Nm/kW

Power-related unbalance =

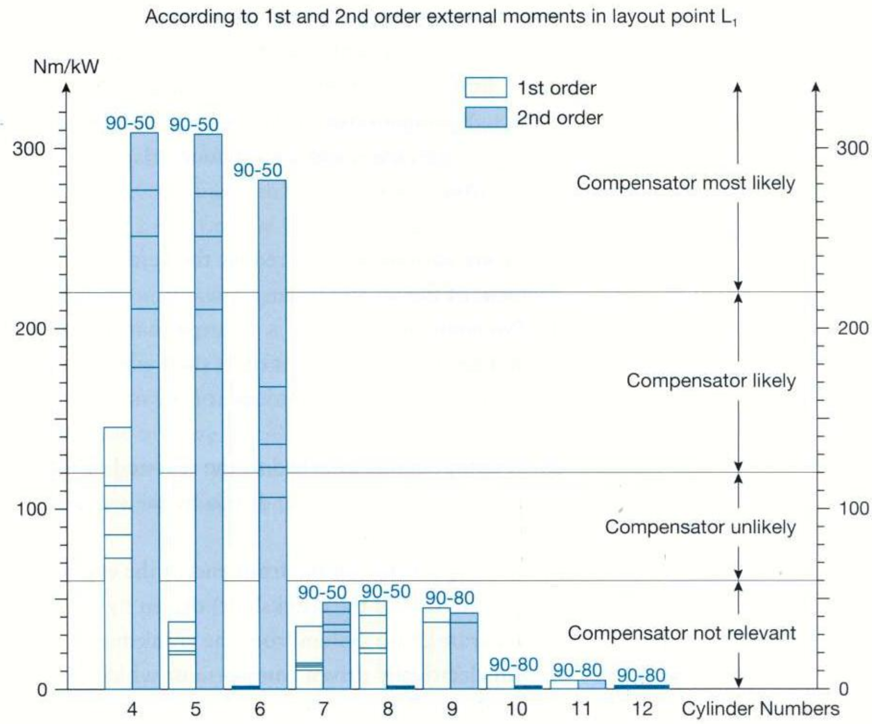
$$\frac{\text{External moments in Nm}}{\text{shaft power in kW}} = \text{Nm/kW}$$

white columns: torsional vibrations of the first order

blue columns: torsional vibrations of the second order

Clearly shown is that mostly the four-, five- and six-cylinder engines are unbalanced.

The large moments are clearly present.

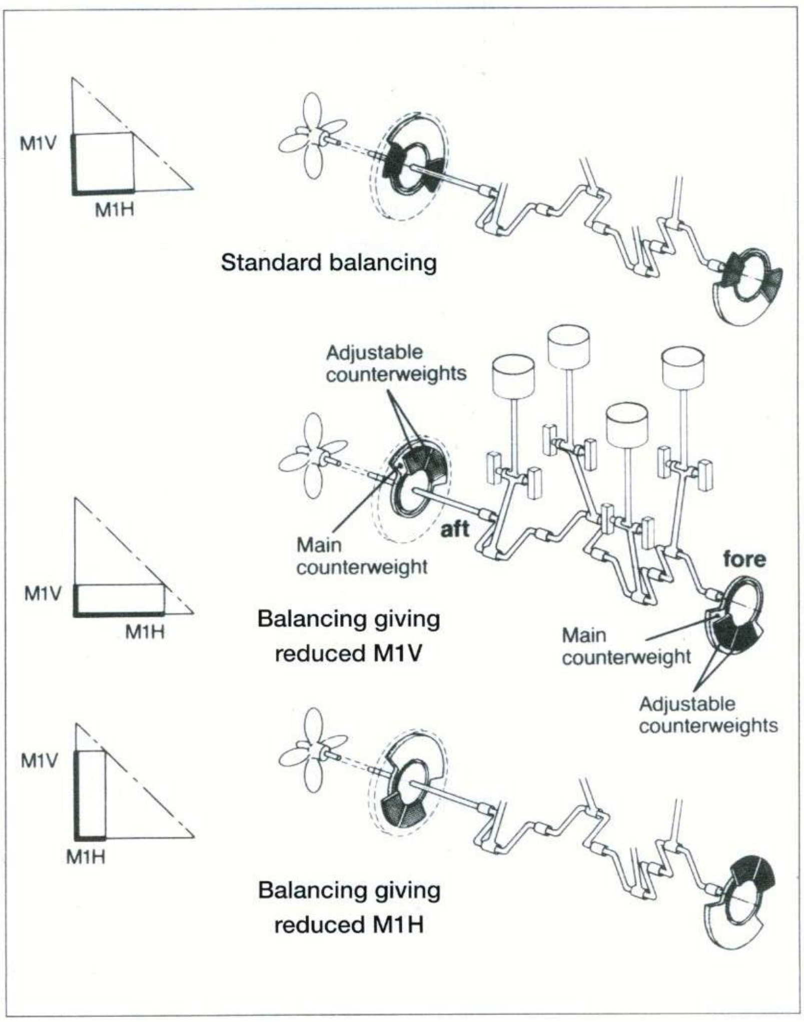


17.12.2 Adjustable counterweights

Four-cylinder engines are equipped with adjustable counter weights as standard.

These counterweights considerably reduce the vertical moments. With the use of this method, no problems should be expected with the resonance of the second order of vertical hull constructions. In exceptional cases, when the first-order moments generate a resonance at a normal engine speed in both the vertical and horizontal second order of the vertical hull construction, a first-order moment compensator can be installed. This is fitted in the chain-tensioning wheel and reduces the horizontal first-order moment to an acceptable magnitude. The compensator contains two counter weights which rotate at the same rate as the engine (two-stroke principle).

As the resonance of the vertical as well as the horizontal hull shapes rarely occur, standard engine designs do not have these installed.



Adjustable counterweights for compensating 1st order external moments.

Resonance between the vertical moment of the second-, third- , fourth- and fifth-degree vibrations of the hull construction occur at normal engine speeds. A compensator of the second order can be fitted on four-, five-, and six-cylinder engines to reduce this.

There are various ways to reduce the vertical moment of the second order:

1 **No compensators;** if it is deduced that this is not required on account of its own vibration frequency and the magnitude of the moment of the second order.
2 **A compensator attached to the rear end of the engine** (crankshaft side) driven by the chain drive of the camshaft.
3 **A compensator on the front end of the engine** ('blind' side of the crankshaft) driven by a separate chain system from the crankshaft.
4 **An electrically driven compensator** which runs in phase with the free moment.
 This type of compensator requires an additional bedplate, preferably in the steering engine room where the deviations are highest and the compensator most effective.
5 **Compensators on both sides of the crankshaft, which completely eliminate the external** moments of the second order.

Solutions 2, 3 and 4 are effective when they are operative near the belly rather than the node of the ship's vibration.
Solutions 4 and 5 must be considered when the critical value of the hull construction is close to that of the engine's.
If the electrical compensator of solution 4 is placed in the steering engine room, it has the advantage of not being as sensitive to the order position as those in solutions 2 and 3.

17.12.3 Vibrations in the bedplate/engine frame

Vibrations in the engine frame resulting from the guide forces are part of the ship vibrations.

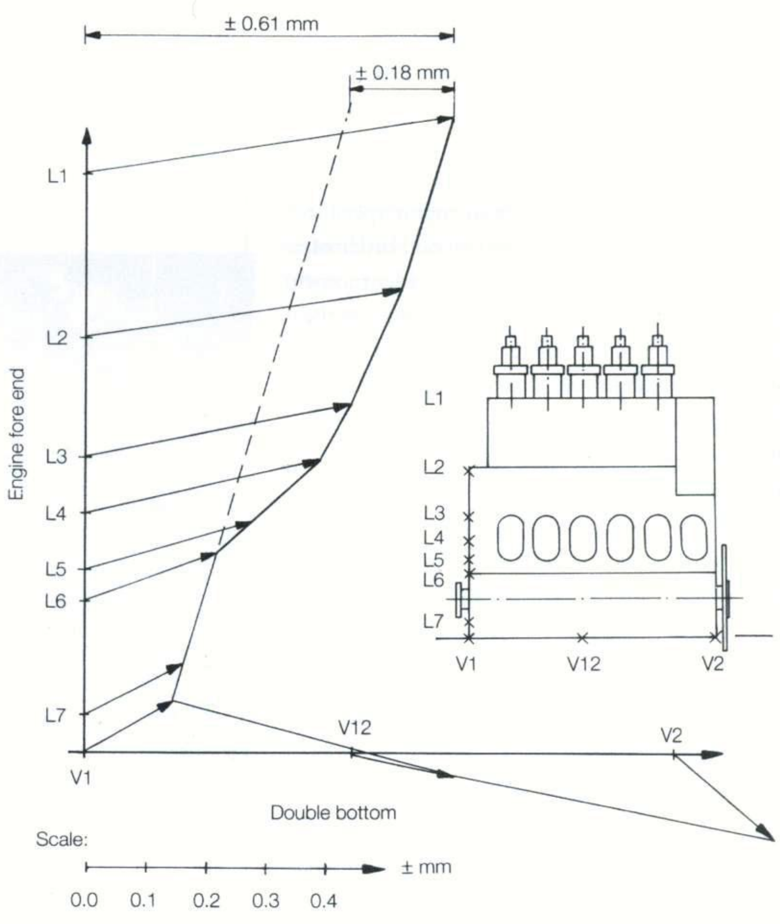
There are three forms of vibration.

1 H-mode
Works on the top side of the engine from the first to the last cylinder. It is also referred to as parallel vibration.

2 X-mode
Works on the top side of the engine at a longitudinal angle. Affects the middle of the engine.

▶ An example of the vibrations in the engine frame of the double bottom of the ship to above on the cylinder block, measured in a two-stroke MAN–B&W 5 L 80 MEC crosshead engine.

vertical: measuring points L1 t/m L7
horizontal: scale in tenths of millimetres
At the top of the cylinder block the result is ± 0.18 millimetre.
On the flat bottom surface the values V1, V12 and V2, are clearly smaller than L1.



3 L-mode
Works on the top side of the engine in the longitudinal direction of the engine.

The natural vibration frequencies of these constructions are largely dependent on the rigidity of the bedplate and the double bottom of the ship below the engine and bedplate.

H- and X-type **moments** are generated by the guide forces of the H- and X-type.
The values of these guide forces can be calculated for each engine, on the basis of the gas and mass forces.
These occur in all engines.

The secondary values of the guide moments originate from torsion vibrations and are therefore difficult to calculate.
Appendix B specifies the primary and secondary values of the guide moments for a MAN-B&W 5 L 70 MC engine with two different shaft designs. The secondary values are low.

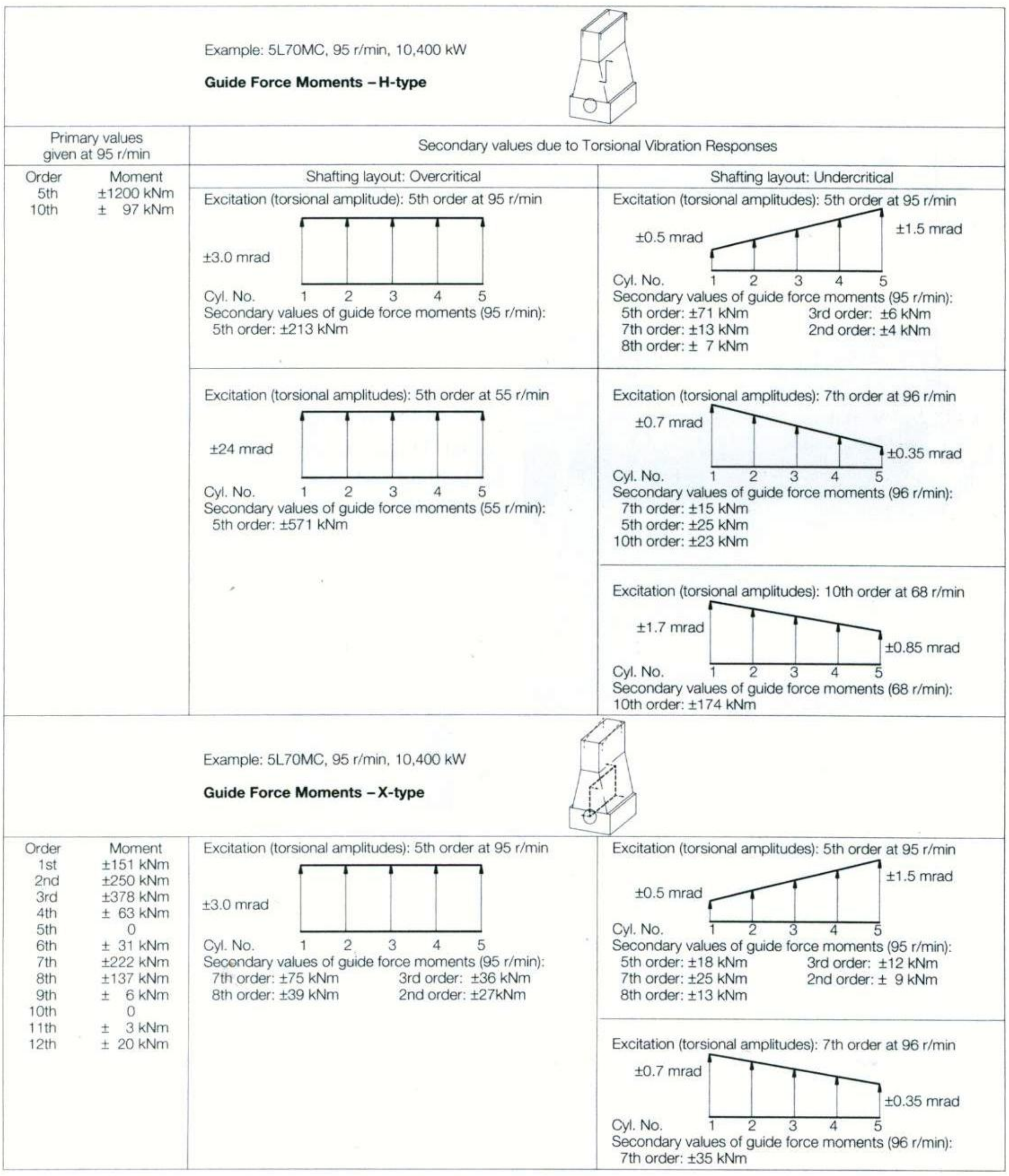
L-**type moments** are generated by installation-dependent factors.
Alternating forces in the thrust block, caused by the torsion vibrations of the propeller, are often the cause of these forces.
Furthermore, L-**type moments can** also be produced by axial vibrations of the crankshaft.

Appendix B

◄

The guide force moments of the H- and X-type.

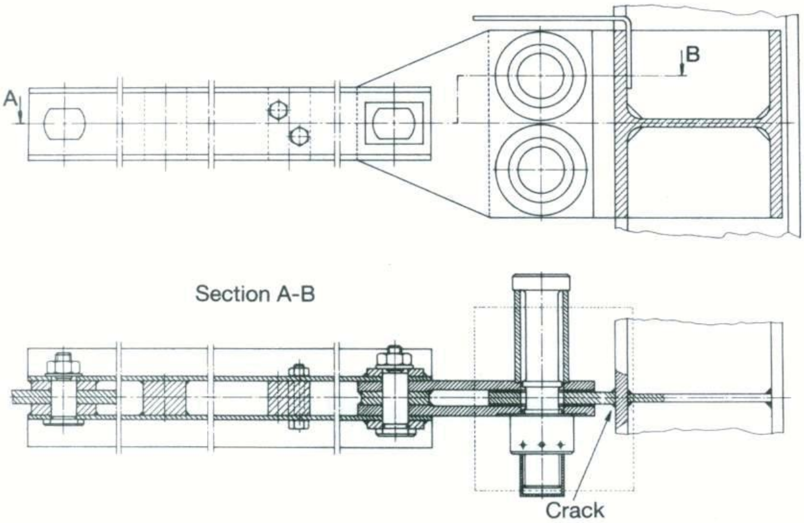
left: for shafts with overcritical speeds
right: for shafts with under-critical speeds



▸
Mechanical top bracing for a two-stroke eight-cylinder Wärtsilä Sulzer 8 RTA 84 C crosshead engine.

17.12.4 Top Bracing – Braces on the top of the engine

H- and X-type vibration-mode shapes are traditionally controlled by providing the top of the engine with braces which rest on the hull construction and so obtain resonances with critical orders **in excess** of the normal engine speed. Here the local rigidity of the construction is important. Sometimes braces are placed in the longitudinal direction of the engine in order to reduce vibrations of the L-**type moments**.
If the crankshaft forms the main source of the vibrations, an axial vibration damper can sufficiently reduce the vibration.



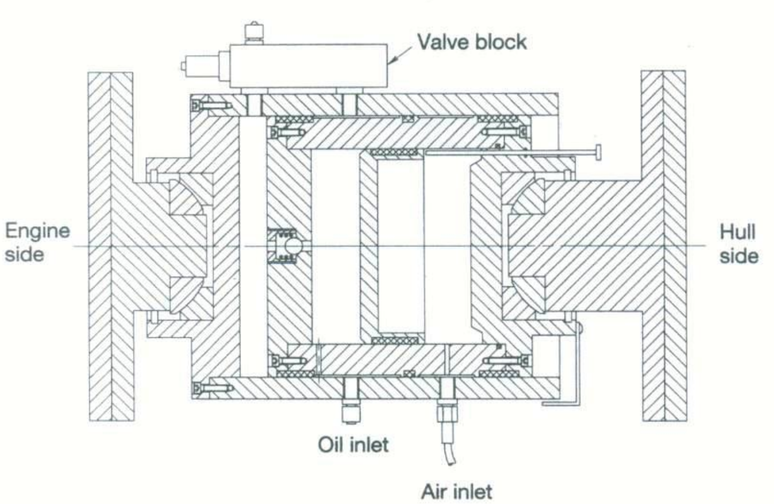
◂
Mechanical top bracing for MAN–B&W MC engines.

This is a standard support, mainly in modern ships. This is built of thin steel with a high tensile strength. Tearing occurs at the welds.
This type of ship has an elastic hull and so the forces became too large for the welds.

▸
Hydraulic top bracing for a two-stroke twelve-cylinder Wärtsilä Sulzer RTA 96 C crosshead engine.

Hydraulic Top Bracing

This is capable of hydraulically compensating the brace distortions in ships that distort more than usual. This occurs, for instance, with heavy seas or at loading and unloading.
This hydraulic top brace ensures that there is a constant force between the engine and the hull construction, independent of the distortion and also functions as a damper.

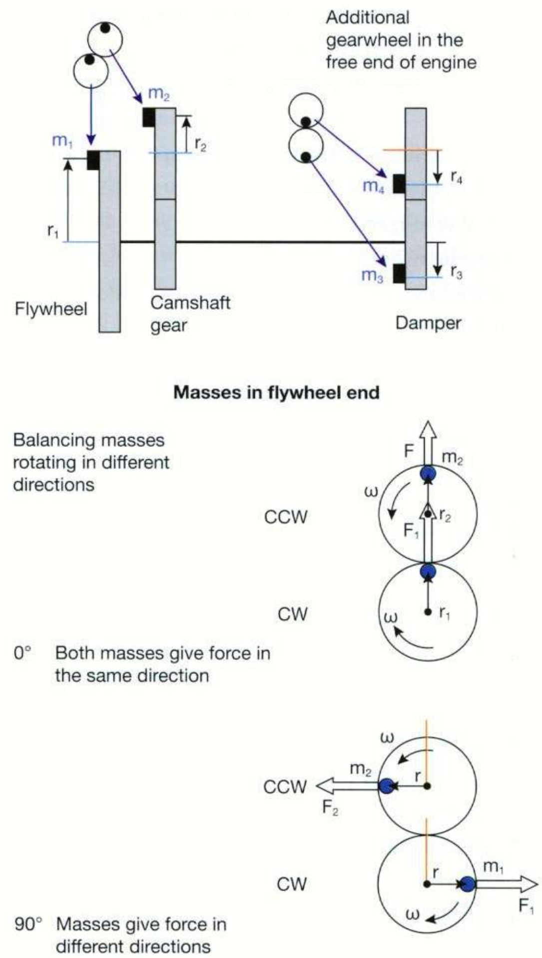


◂
Hydraulic top bracing for MAN–B&W MC engines.

This system comprises a hydraulic cylinder and two spheroid bearings.
Oil is supplied from its own oil system and an overflow valve controls the hydraulic force.
In order to simplify the system, this does not have its own lubricating-oil system but an air-driven piston. The guide moments ensure that the oil is pressurised and therefore the vibration forces are transmitted to the ship's hull.

17.13 Example of the balancing used in a Wärtsilä 9 L 46 four-stroke engine – category III

This example demonstrates how balancing and vibrations are taken care of in a modern, heavily loaded, medium-speed four-stroke diesel engine.



▶ An example of the balancing used in a Wärtsilä 9 L 46 four-stroke diesel engine, category III.

Four extra masses are mounted:

m_1 on the turning wheel

m_2 on the camshaft gear-wheel

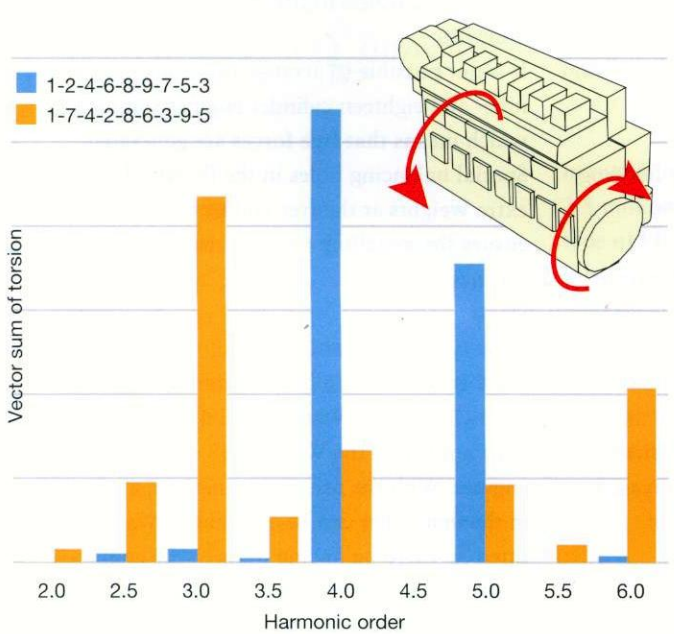
m_3 and m_4 on an extra gear-wheel transmission on the free end of the engine.

Every weight has its own radius; r_1, r_2, r_3 and r_4.

At position 0 of the crankshaft, both masses m_1 and m_2 give force in the same direction and at 90°, they give force in different directions.

ccw: counterclockwise

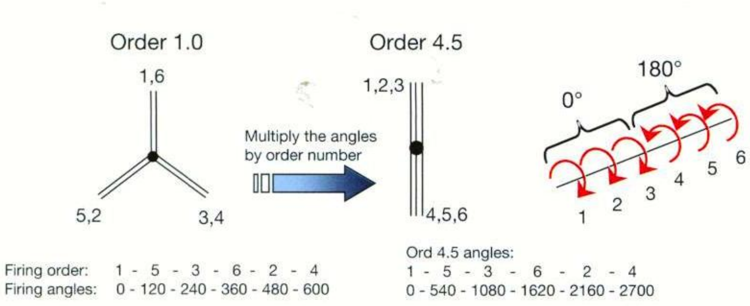
cw: Clockwise



◀ Torsional forces in the engine block at two different ignition sequences. Shown here, a nine-cylinder four-stroke engine.

horizontal: harmonic orders

vertical: sum of the torsion



◀ An example of composing the torsional excitation mode shape of order 4.5 for a six-cylinder four-stroke engine.

17.13.1 Wärtsilä four-stroke medium speed diesel engines: Several examples

Wärtsilä defines roughly the relationship between the lowest harmonic orders and vibration forms of flexibly arranged engines as follows:

Harmonic order 0.5
: **imbalance** for gas forces – movement of the rigid frame/block

Harmonic order 1.0
: mass forces – movement of the rigid frame/block

Harmonic order 2.0
: mass forces– bend

Halve orders 1.5, 2.5, ...
: gas forces, torsion

If a local resonance frequency, such as the natural frequency of the turbocharger, is situated too closely to one of these harmonic orders, the produced form of this order dominates due to the vibrations of this part.

17.13.2 Balancing

The standard manner to reduce the negative effects of mass forces is balancing.
Counter weights are predominantly used to balance the individual cranks mounted on the crankshaft in order to reduce local bending of the crankshaft as well as the bearing load.
More importantly, they can affect all the deviations in the engine block caused by bending.

Degree of balancing

The degree of balancing indicates which part of the rotating mass is balanced by counter weights. In many engines including the Wärtsilä engines the degree of balancing lies between 75 and 95%. The present (2008) trend is an increase in this percentage. In the modern Wärtsilä 46 F diesel engine, in which the mass forces have increased due to a higher engine speed, the degree of balancing is 100%.

The Wärtsilä 12 V 46 diesel engine is an exception as the degree of balancing exceeds 100% so obtaining the lowest possible bending stresses and therefore deviations in the block in both vertical and horizontal planes.

An even number of cranks in engines such as the 6 L, 8 L, 12 V or 16 V are symmetrically aligned, enabling the moving masses to balance each other. This is referred to as externally balanced engines. They do not have free forces or torques which generate vibrations in the engine block.

17.13.3 Actual differences in the weight of engine parts

In practice the moving parts never have an identical weight as manufacturers work with tolerances. This produces free forces, so vibrations in permanently mounted engines can be experienced when these forces and torques are powerful enough as they pass through a certain speed range in a flexibly arranged installation.

Resonance

Resonance often cannot be avoided, especially in engines with ever-fluctuating engine speeds. The forces generated by the weight variances in the rotating parts in the proximity of the crankshaft may decrease by 80 to 90%, if the components are manufactured with tighter tolerances. In connecting rods of high-speed engines, this can be measured in grams!

Balancing holes in the fly-wheel and other weights

It is not possible to arrange the cranks of seven-, nine- and eighteen cylinder engines symmetrically which means that free forces are generated. Special balancing holes in the fly-wheel and extra weights at the free end of the crankshaft ensure the matching of the vertical and horizontal torques.

New type of balancing

Since 2004 Wärtsilä has an added additional balancing for the Wärtsilä 9 L 46. This has also been applied in the Wärtsilä 38 and 46 F diesel engines. With the use of this method, all free forces in the first-order can be eliminated. Wartsila has fitted an extra balancing mass to the counter-clockwise turning intermediate camshaft drive and placed a counter-clockwise turning balance mass on a separate cog wheel at the free end of the crankshaft.
This, in conjunction with the clockwise turning balancing masses makes it possible to eliminate all torques!

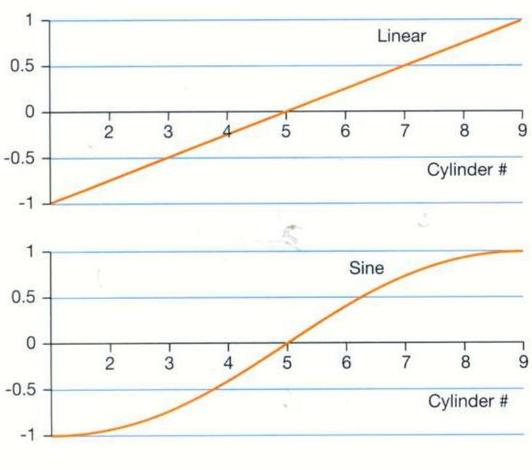
A different method for tackling free torques is the realisation of unequal forces between the various cranks.
Irregularities in the combustion process can produce small tilting torques at low frequencies. It must sometimes be accepted that some resonance will be present, provided that the resonance does not coincide with the natural vibration frequencies, such as bending-generated torsional vibration.
In the severest case of resonance, the natural vibration amplitudes together with the generated vibration amplitudes have an identical frequency. Comparing the generated vibration frequencies and the natural vibration frequencies provides information as to the expected degree of damage at a certain harmonic order to a construction.

17.13.4 Vector summation

The various kinds of torsional flexural-buckling vibrations are often analysed by summing the vectors. This is a simple method to compare generated and natural vibrations. The 'vector resultant' is the sum of the force and torque vectors for specific harmonic orders, and is calculated by:

$$V = \sqrt{(\sum_{i=1}^{c} f(i) A \sin \varphi_i)^2 + (\sum_{i=1}^{c} f(i) A \cos \varphi_i)^2}$$

▲
Here:
c = number cylinders
φ_i = ignition angle of cylinder 1 multiplied by the harmonic order
φ_i = order × ∝1
A = force of the torque amplitude
f (i) = function defining a weighting factor depending on the estimated mode shape



◄
When analyzing torsional excitations, a linear weighting function or a sinus wave can be used.

It is also possible to adjust the location of the nodal point of the mode if the torsional mode shape is well known.

17.13.5 Combustion sequence

The way that the various harmonic orders are generated is mainly dependent on the chosen combustion sequence (the ignition sequence). Altering the combustion sequence has an important influence on the vibration characteristic for an engine or genset. Today, the combustion sequence is defined in the early stages of the design process. This allows for an optimum design for engine performance. Special software has been designed to achieve these ends.

17.13.6 Vibration

There is a large amount of software on the market used to analyse the vibration activity and accurately calculate the forces generated in the engine. The analyses can be rendered in time and frequency.

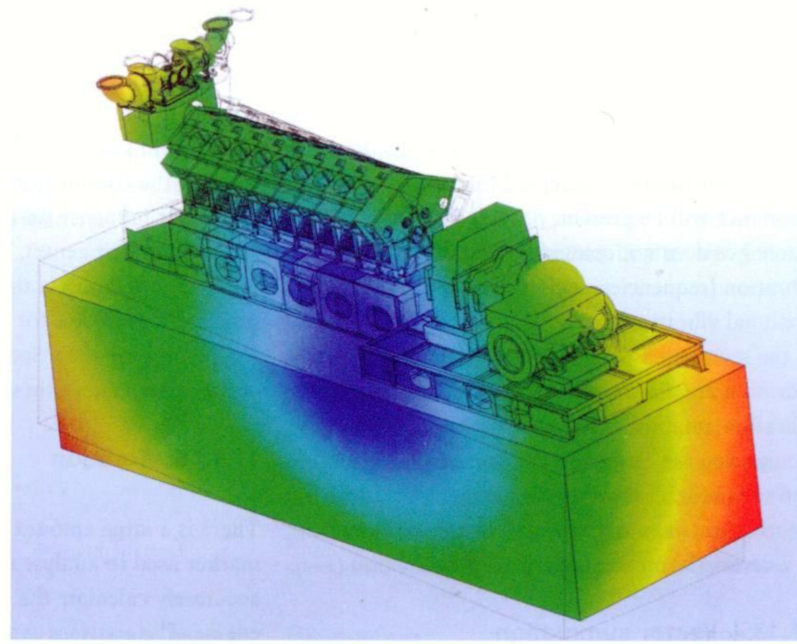
17.13.7 Finite-element method

Known as the FE model.
This is a three-dimensional computer model of, amongst others, the engine block. In this model all possible bending and loads are visible and it is widely used in the engine industry. The load of a part of the block can, for instance, be indicated with a different colour.
Bending can (or degrees of) be grossly exaggerated so a visual picture is formed.

In the Wärtsilä 6 L 46 F model, approximately 3.5 million 'degrees' must be simulated! In this type of FE model for an engine block, the working forces for each harmonic order can be calculated individually.
These forces are saved and may be re-used for calculations for components which are to be installed in the engine block, such as main bearings and cylinder liners.

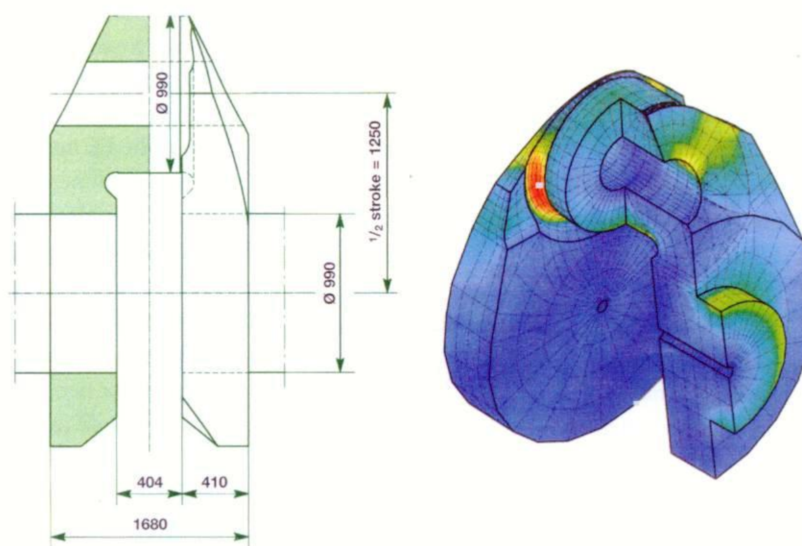
▶

A finite-element model showing the deformations in a pump assembly with a Wärtsilä V- engine mounted on a soft foundation. The colours show the degree of deformation and therefore load.



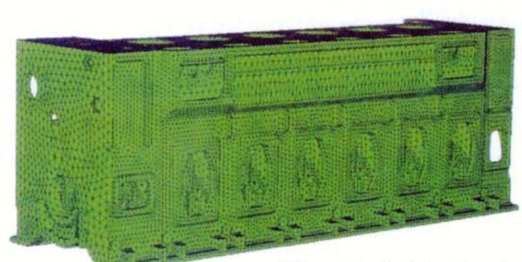
▶

The finite-element model applied to the crank for a two-stroke Wärtsilä Sulzer RTA 96 C crosshead engine. Using this model, engine designers can perform simulations detailing exactly where problems could be expected.

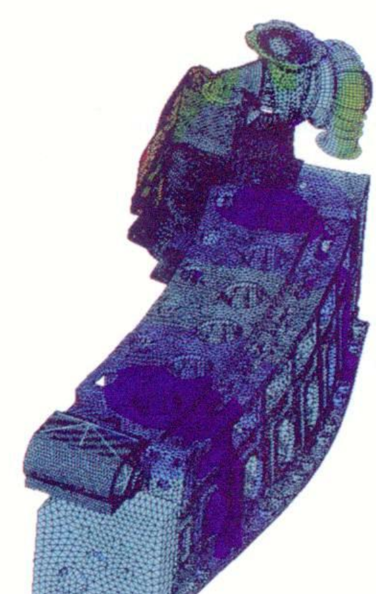


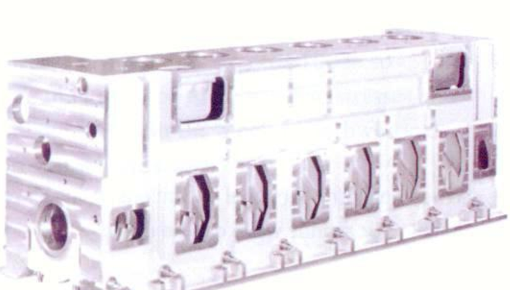
▲

The finite-element model applied to the crankshaft bedplate, the A-frame and cylinder beam for a two-stroke crosshead engine, a Wärtsilä Sulzer RTA 96 C.



FE model with 1.1 million degrees of freedom





Wärtsilä 6 L 46 F engine block

▶

The finite-element model applied to the block of a Wärtsilä 6 L 46 F engine.

17.13.8 Simulating reality

This finite-element model simulates the entire propulsion system or an electricity generating plant. The engine can, therefore, operate with a variable engine speed in accordance with the propeller law or with a constant torque in a electricity generating plant.

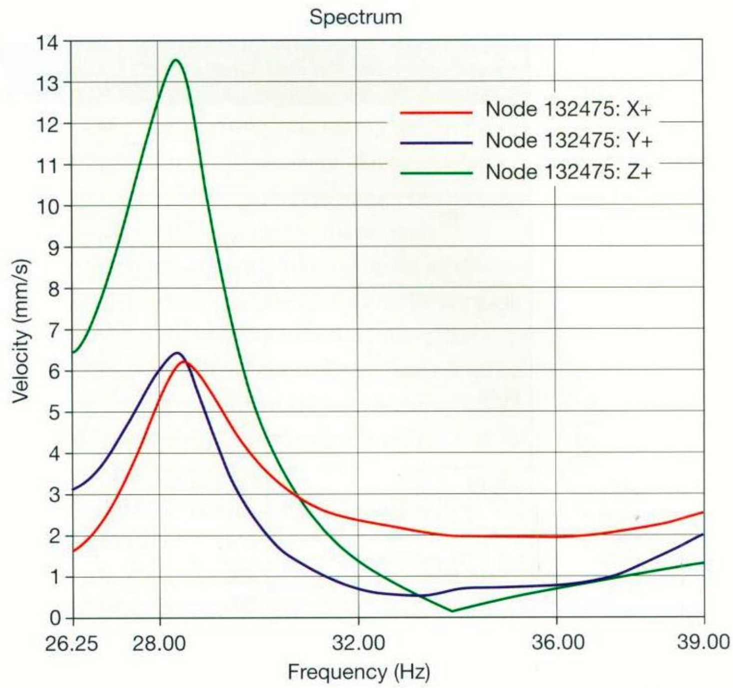
17.13.9 Assessing vibration levels

The vibration levels of engines are usually assessed according to ISO standards:

- for piston engines: ISO 10816-1/ ISO 10816-3;
- for gensets: ISO 8528-9 and ISO 10816-3.

In addition to these regulations various Classification Societies have their own regulations for finding vibration levels in the different systems.

Today, diesel engine manufacturers are often asked for their opinion the vibration levels in their engines and components.
Proper collaboration between all parties is the best way to achieve minimum vibration levels in the engines.



▲
An example of the torsional vibrations for the three axes.

17.13.10 Rule of thumb

Wärtsilä's rule of thumb is that the average vibration level of an engine block should not exceed a velocity of 28 mm per second. Limitations for standard engine blocks, added components and pipes are provided separately.

It is not easy to indicate exact values as the average vibration level for each individual point constitutes the sum of the various characteristics, such as movement in a rigid frame, elastic deviations and local effects.
Engines with a divergent number of cylinders also behave differently and consequently some engines have above average vibration levels.

17.13.11 Exceptions

It is possible for a vibration level to be unacceptable even-though the measured values are below the stipulated values. This may occur when the average vibration level is completely dominated by the vibration level in one component which is often a sign of resonance. Simulation models and calculations are necessary to eliminate this problem. These provide the maximum vibration level for the particular part and this can be compared against the average vibration level.
A good rule of thumb is that the contribution of one single vibration component should not constitute more than 80% of the average reasonable level.

An example of the torque variation for the different Wärtsilä 46 medium-speed diesel engine types at speeds of 450, 500 and 514 revolutions per minute.

Torque variation, D-rating

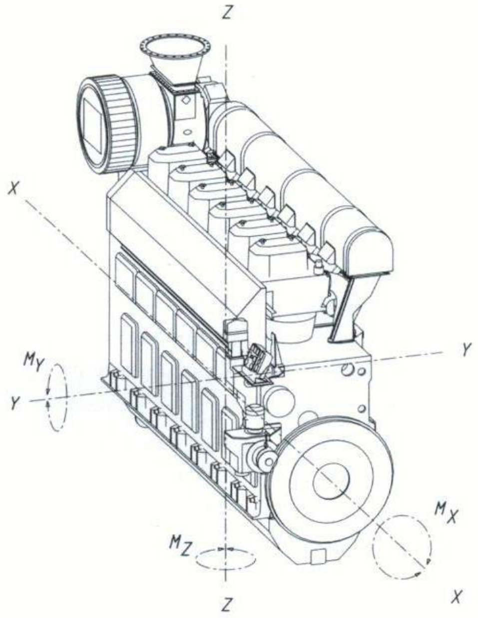
Engine	Speed [RPM]	Frequency [Hz]	M_X [kNm]	Frequency [Hz]	M_X [kNm]	Frequency [Hz]	M_X [kNm]
6L46	450	22.5	132.5	45.0	48.1	67.5	6.8
	500	25.0	101.2	50.0	49.6	75.0	9.2
	514	25.7	94.0	51.4	50.0	77.1	9.6
6L46, idle	450	22.5	48.6	45.0	11.9	67.5	3.0
	500	25.0	69.8	50.0	12.0	75.0	3.0
	514	25.7	75.8	51.4	12.1	77.1	3.0
8L46	450	30.0	176.5	60.0	18.0	90.0	2.5
	500	33.3	167.9	66.7	21.6	100.0	3.6
	514	34.3	166.9	68.5	22.2	102.8	3.9
9L46	450	33.8	158.9	67.5	10.2	101.2	2.0
	500	37.5	156.0	75.0	13.9	112.5	2.8
	514	38.6	156.0	77.1	14.4	115.6	3.0
12V46	450	22.5	101.4	45.0	68.0	67.5	12.5
	500	25.0	77.4	50.0	70.2	75.0	17.1
	514	25.7	71.9	51.4	70.7	77.1	17.7
12V46, idle	450	22.5	37.2	45.0	16.9	67.5	5.5
	500	25.0	53.4	50.0	17.0	75.0	5.6
	514	25.7	58.0	51.4	17.1	77.1	5.6
16V46	450	–	–	60.0	36.0	120.0	1.3
	500	–	–	66.7	43.3	133.4	2.2
	514	–	–	68.5	44.4	137.0	2.5
18V46	450	33.8	311.8	67.5	18.8	101.2	3.3
	500	37.5	305.9	75.0	25.6	112.5	4.6
	514	38.6	306.1	77.1	26.6	115.6	5.0

The values are instructive and are to be calculated case by case.

– couples and forces = zero or insignificant

The external forces and couples in a six-cylinder Wärtsilä 46 diesel engine.

Extend forces for all cylinders zero.
External couples: numbers are guidelines, but must nevertheless be calculated individually.



External forces F = 0 for all cylinder numbers

External couples (the values are instructive and to be calculated case by case)

Engine	Speed [RPM]	Frequency [Hz]	M_Y [kNm]	M_Z [kNm]	Frequency [Hz]	M_Y [kNm]	M_Z [kNm]	Frequency [Hz]	M_Y [kNm]	M_Z [kNm]
9L46 *)	450	7.5	25.5	25.5	15,0	30.8	–	30.0	10.5	–
	500	8.3	31.5	31.5	16.7	38.0	–	33.3	12.9	–
	514	8.6	33.3	33.3	17.1	40.2	–	34.4	13.6	–
18V46	500	8.3	283.8	283.8	16.7	135.1	55.9	33.3	–	4.0
	514	8.6	299.9	299.9	17.1	142.7	59.1	34.3	–	4.3

*) Subject to selected firing orders

– couples and forces = zero or insignificant

17.13.12 Summary for Wärtsilä four-stroke medium-speed diesel engines

The vibration level in a diesel engine is a combination of the structural properties, such as the natural vibration frequencies, mode shapes and damping. However, the vibration level also includes the vibrations generated by gas forces, mass forces of the crank-connecting rod mechanism and moving parts in the running engine.
The application of advanced software programmes allows sound calculations to be made for production series as well as experimental tests for the probable vibration performances of engines. Optimising the vibration characteristic in an engine usually consists of the analyses of the parts, such as engine block and its components, and the analyses of vibration generation.
An investigation can also be done with the resonance of vibrations between the natural vibration frequency and the generated vibration frequency. Naturally, this should be avoided as much as possible due to harmful resonances. Sometimes resonances can not be averted and must be kept as low as possible or compensated. Several manufacturers have developed good methods in this field. Due to available software and fast computers this data can be quickly made available for problems at sea.

Damage and its consequences in the engine with respect to the allowable vibration levels

It is common practice for a four-stroke engine with a below-average number of cylinders to continue to operate in the same fashion as two-stroke crosshead engines.
For the following assumption, a nine-cylinder four-stroke engine is used as the propulsion engine for a ship that is in the middle of the ocean. The combustion process is impeded due to problems with one cylinder. A high-pressure fuel pump could then be switched off, or the drive gear, piston and connecting rod removed.
It must be considered at which capacity an engine can operate without causing damage to the engine. The damage can be caused by abnormally high torsion vibrations in the crankshaft.

General procedure for vibration and balancing related problems in four-stroke engines

Communicate with the engine manufacturer! After mutual agreement an assessment can be made establishing at which engine speed can be operated safely based on the damage to the engine.
Each case must be regarded individually as there is no general true rule of thumb that can be applied. There are also engines, such as the Caterpillar–MaK series, which have crankshafts that are so powerful that, for instance, a six-cylinder MaK 25 runs 'effortlessly' on five cylinders with 83,3 % of the power output capacity.
Consulting the manufacturer is always recommended!

Special operating conditions for four-stroke diesel engines

If damage is caused to any four-stroke engine and the engine and combustion cannot occur in all the cylinders, measures with regard to vibrations and balancing are required.

- **If one or more fuel pumps must be switched off, it is no longer possible to operate at full capacity.**
- **During overheating of, for instance, the crankpin bearing,** the big end of the connecting rod, the concerning piston and connecting rod must be removed.

Here the valve drive should also be dismantled in order to prevent exhaust gases from entering the cylinder and consequently the crankcase.
Apart from the 'Actions in case of damage and emergencies', it is of the utmost importance to determine at which capacity and revolution rate the engine can operate.
In principle calculations are made for each type of engine. These indicate meticulously the allowable vibration margins.
Individual Classification Societies also prescribe these margins.
The first action is to contact the engine supplier, who can rapidly provide appropriate advice regarding action that should be taken.
It is possible to make a separate torsional vibration calculation for each situation, so damage to the crankshaft can be averted.

17.14 Balancing of V-engines

17.14.1 Introduction

V-engines are frequently used in engine categories I, II and III. They are supplied with from four up to and including twenty cylinders. One of the advantages of V-engines in relation to in-line engines is the short engine length in relation to the shaft power produced.
Furthermore, in-line engines in these three categories rarely have more than nine or ten cylinders in line. Only in engine category IV, the large two-stroke crosshead engines, the engine manufacturers deliver as standard the largest cylinder diameters with twelve cylinders.
When fourteen-cylinder engines are manufactured with an option for sixteen- and eighteen cylinders, these engines are 30 metres in length!
For a twenty cylinder V-engine in engine category I, II and III, two ten-cylinder engines are arranged. This takes up more space and is less interesting from a financial view point. Diesel gensets in general are driven by V-engines, mainly with sixteen and eighteen cylinders. In engine category II, twenty cylinders are common.

17.14.2 Arrangement and balancing

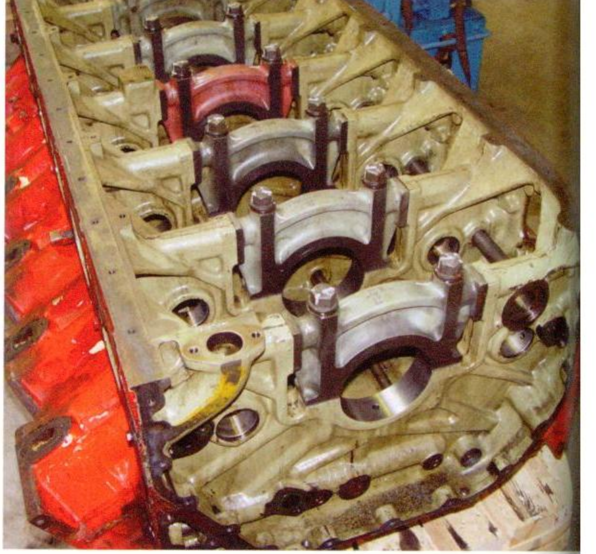
The engines are aligned in two rows at an angle varying from 60 to 120 degrees. An engine that must have equal ignition distances has an angle between the cylinder rows that is identical to the angle between the cranks of the crankshaft. Generally, an angle of 45 to 60 degrees between the cylinders is used making it impossible to obtain equally distributed ignitions for each cylinder number. For smaller engines in category II, angles of up to 120 degrees can be used.

17.14.3 Several examples of V-engines

▲

Mounting the complete cylinder unit with cylinder head, piston, connecting rod and cylinder liner in an eighteen-cylinder MTU V engine block, type 8000, category II in Friedrichshafen, Germany.

The angle between the cylinders is small, approximately 45°.



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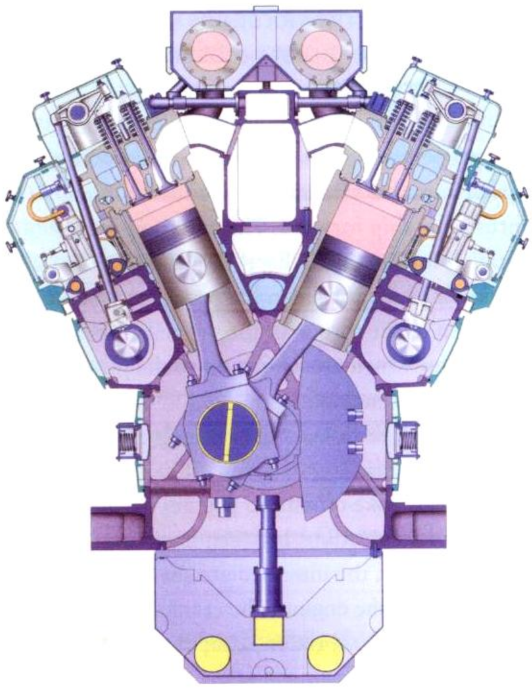
This twelve-cylinder V-engine block, category II is undergoing a revision. Seen here on its head.

The angle between the two cylinder rows is more than 90°.



▲
A Wärtsilä 38 V-engine, category III, has the traditional angle between the cylinder rows for the large bores between 45° and 60°.

The angle is 50°.

17.14.4 Arrangement of cylinder of the left and right bank

Today, in the majority of engines, both pistons with cylinders positioned opposite to each other are individually fixed on the broad crankpin. As a consequence, the cylinders are directly across from each other but placed in an 'interleave' pattern, for a certain distance as the two connecting rods adjacent to each other are attached to one crankpin.

Older engines often have one main connecting rod with a second connecting rod attached to this main connecting rod, the so-called mother-daughter construction. In these engines, the cylinders are arranged directly opposite to each other. The latter, complicated construction is no longer used in new engines.

17.14.5 Rotating forces and moments

Essentially, these are mainly dependent on the crankshaft shape but also that both bottom parts of the connecting rods are also part of the rotating forces.

The resultant rotating, primary and secondary oscillatory forces working on the engine can be found by summing the forces working on one cylinder pair.

The crankshaft is often constructed in central-symmetrical way. If the engine is homogeneous (symmetrically), the forces and the resultant forces for the entire engine equal zero.

In Wärtsilä four-stroke V-engines these are completely balanced.

Primary oscillating forces: the pistons, the piston coolant and the top part of the connecting rods, as well as the secondary oscillating forces.

17.15 Balancing examples for two-stroke crosshead engines – category IV

Example MAN–B&W

In general, there are relatively numerous two-stroke crosshead engines in use with four or five cylinders. This forms an ideal propulsion mechanism for the relatively low-speed large tankers and bulk carriers.

They have relatively few moving parts and therefore low installation and operational costs.

Due to the large number of engines with relatively few cylinders, much of the experience in the field for the reduction of vibrations has been gathered.

Reduction of vibration requirements often originate in complaints in the accommodation aboard ship where vibrations are annoying and a nuisance factor.

However, when insufficient attention is paid to vibrations, the mechanical and electronic parts of two-stroke crosshead engines and even parts of the ship could be damaged by, for instance, tearing.

Internal forces and moments can distort/misalign the engine.

External forces and moments are generated by the engine and work on the bedplate at the bottom of the engine and the bedplate at the upper part of the engine for the fitted top braces.

17.15.1 External and internal forces and moments

An engine frame should be constructed with such rigidity that the internal moments and forces can not cause excessive bending of the frame with the accompanying stresses.
If the engine bedplate is presumed to be sufficiently rigid, the internal moments and forces will not cause distortion in the ship's construction. The term 'rigid' is, of course, relative and there is always a certain amount of movement in the engine frame, the bedplate and ship construction. Consider, for example, poor weather conditions.

Information gathered by experience at MAN–B&W has shown that the internal forces and moments of the first-order caused by the mass rotating forces and the reciprocating moving masses are not capable of generating vibrations in the ship. X-type moments should be closely examined as they generate higher vibration levels and work in the less rigid directions of the engine frame, especially in multiple-cylinder engines, such as twelve- and fourteen cylinder versions.

17.15.2 Torsional vibrations

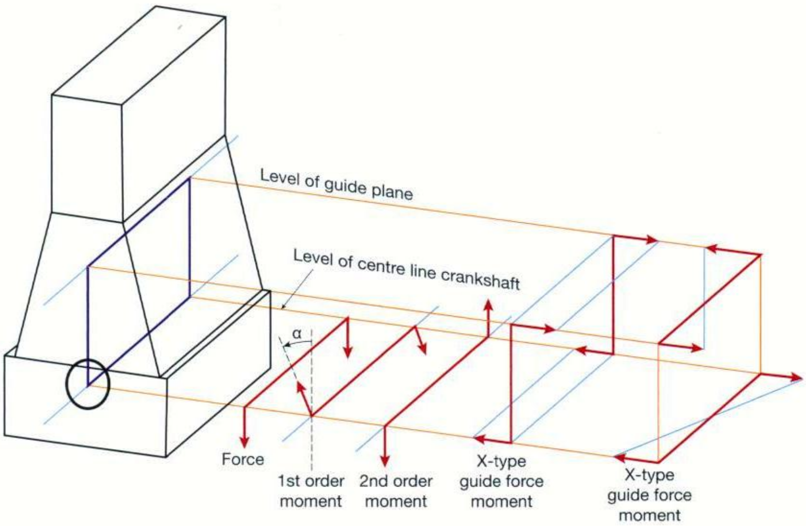
The entire shaft system of the crankshaft, intermediate shaft and propeller shaft, is subject to vibrations which are predominantly caused by the tangential force F_T.
Torsional vibrations can cause vibrations throughout the entire ship due to the connection of the engine drive gearing to the propeller.

17.15.3 Torsional vibrations generated by the thrust pressure of the propeller

The propeller is nothing but a part that with the power supplied by standard propulsion force thrusts the ship forward. The variation in the rotating torque of the diesel engine continues as a variation of the torque in the propeller. Here the variation of the torque of the propeller produces vibrations, which are passed on to the propeller shaft and the engine.
Systems with flexible torques largely eliminate these vibrations.
Large two-stroke crosshead engines usually have rigid connections between the fixed-pitch propeller, the intermediate shaft and the crankshaft of the engine. The crankshaft is equipped with a vibration damper, if required.

17.15.4 Axial vibrations in the crankshaft

These are generated by the radial force F_R and also by the tangential force F_T.
The vibrations generated by the propeller are a contributing factor to the axial vibrations in the crankshaft. Remember, large two-stroke crosshead engines have no flexible torque between the engine and the directly-driven fixed-pitch propeller.
Axial vibrations produce a reaction force in the thrust block which then itself becomes a source for vibrations in the rest of the ship.



◄
External forces and moments.

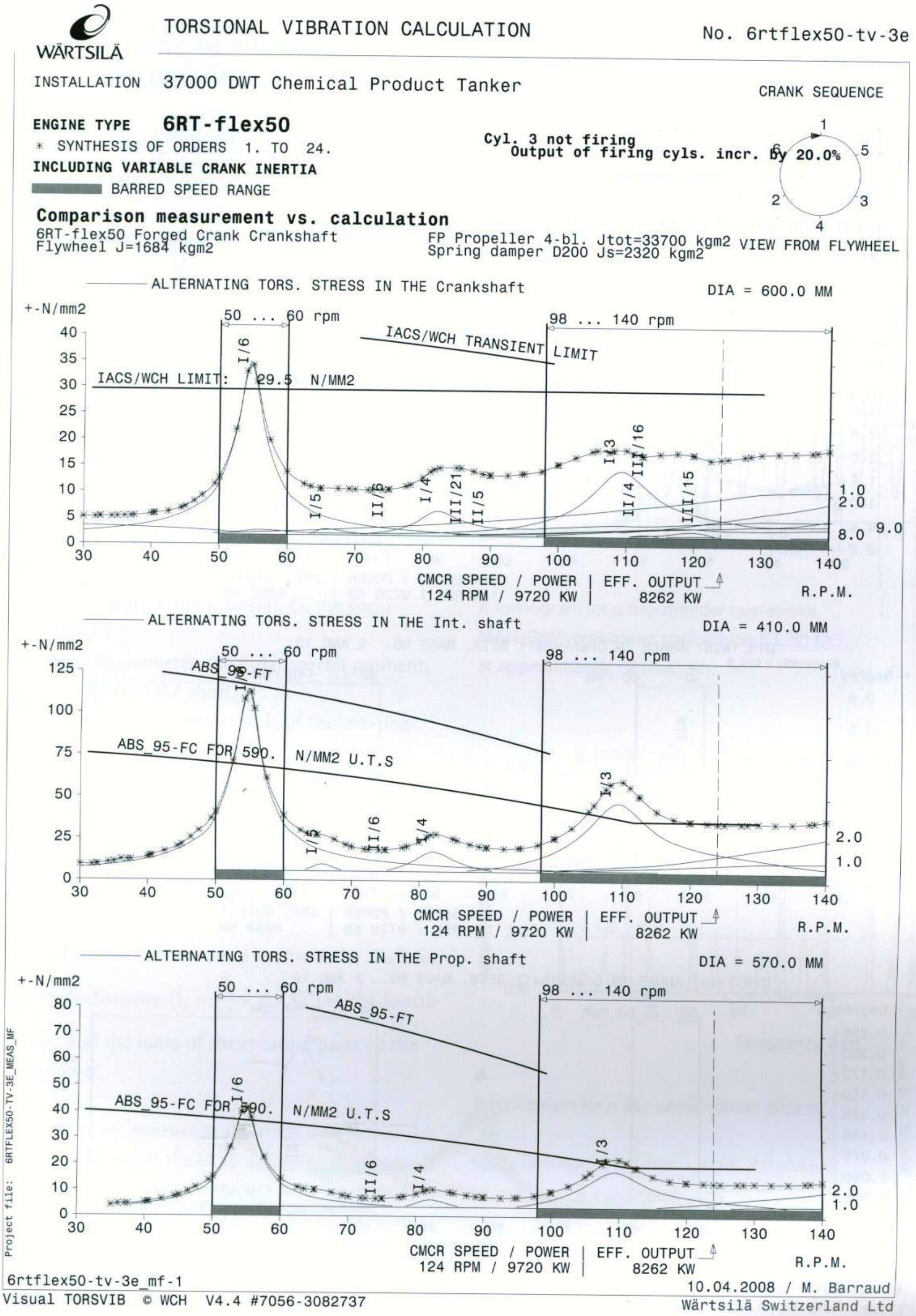
- **first-order moments in vertical and horizontal directions.**
  They are of equal magnitude in MAN–B&W engines with standard balancing.
- **second-order moments in the vertical direction.**
  Moments of the fourth order and higher occur in engines with a certain number of cylinders, but are small and can be ignored.
- **H-type moments** are found between a stationary engine frame and the rotating and reciprocating moving parts of the engine.
  parts of the engine. From a practical engineering point of view, it should be applied to the engine frame as an external moment.

17.27 Example 2, on five cylinders

▼

Simulation: Here, in the simulation, one cylinder is switched off; the engine is now operating on five cylinders and the sixth cylinder bumps along. It has compression but no combustion. The other five cylinders are now producing the same power as in the first case. The total power remains the same.

With a fixed propeller, the relationship between power and speed according to the propeller with P (kW) is equal to n^3 (rpm). The engine must supply the power demands of the propeller irrespective of the number of operating cylinders.



First picture:
The generated torsional tensions in the crankshaft.
A second barred speed area is present where above 98 revs/min., the engine may not be operated.

Second picture:
The generated torsional tensions in the intermediate shaft.
At approximately 110 revs/min., these are the largest and exceed the ABS limit.

Third picture:
The generated torsional tensions in the propeller shaft.
Here, at 110 revs/min.
A maximum value is exceeded.

Conclusion: The engine can only operate at full power when the speed is decreased from 124 to 98 revs/min.

Final conclusion

Therefore, this engine can still supply full power to the propeller with five cylinders in service and a reduced engine speed.
The engine now has two barred engine speed ranges: of 50 to 60 revs./min and over 98 revs/min.

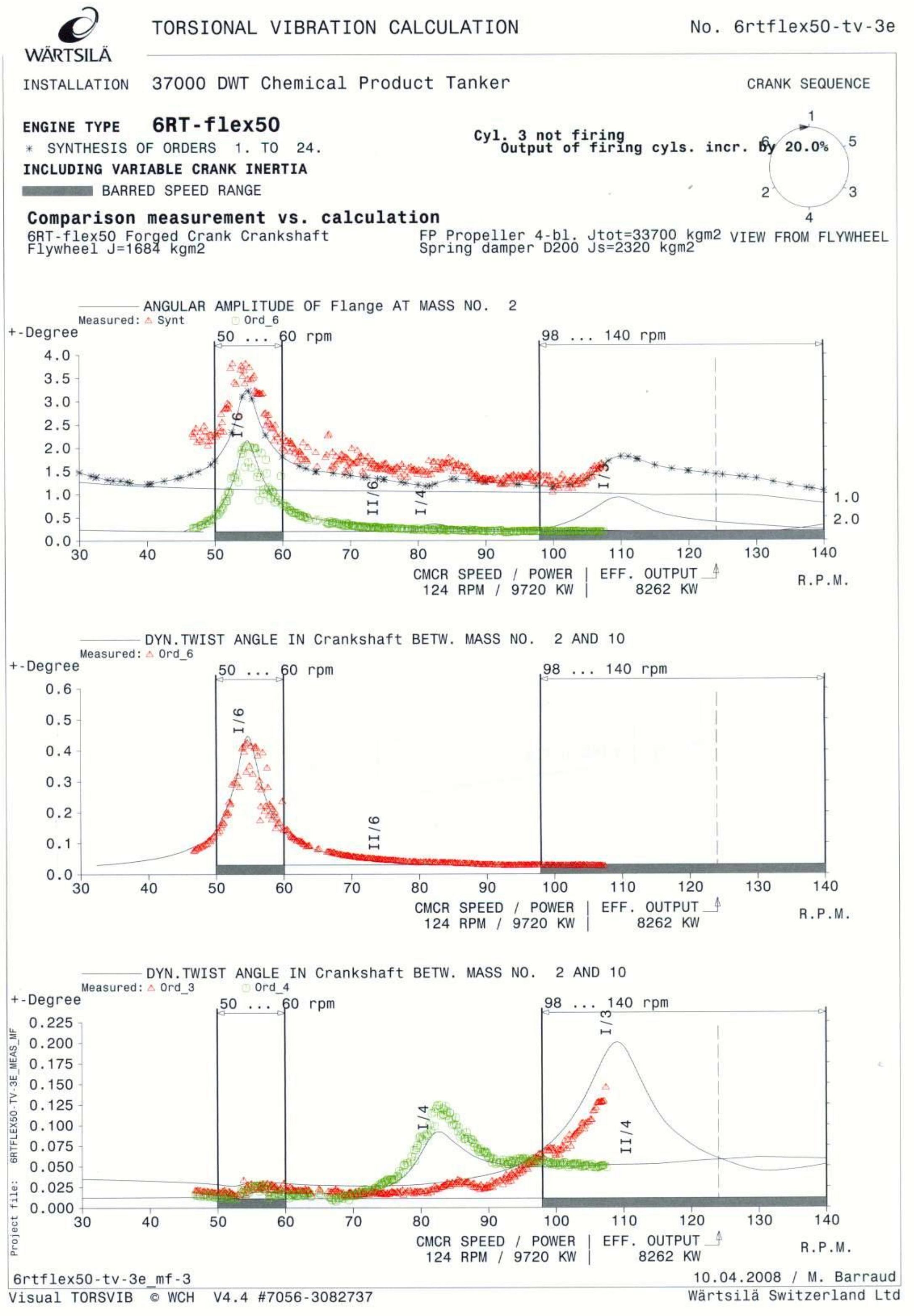
▶

Simulation (continued).

Fourth picture:
The angle of twist of the flange at mass 2.
Sixth order – green.

Fifth picture: coloured
The angle of twist in the crankshaft between mass 2 and mass 10.
Sixth order – red.
No problems at high speeds.

Sixth picture; coloured
The angle of twist in the crankshaft between mass 2 and mass 10.
Third order – red
Fourth order – green



17.28 Design of a propulsion installation

The natural (vibration) frequency of the system is calculated to determine the various critical areas for certain engine speeds and tension values.
If the exerted tensions are high, or if this area is too close to the regular engine speed, then the critical area is shifted by adjusting the natural vibration frequency of the engine or the system.

17.29 Effects of vibration frequencies

The following formula can be applied for the calculation of the vibration frequency for a single-mass system.

$$n_{tr} = \frac{1}{2 \times \pi} \sqrt{\frac{G \times I_P}{l \times J}}$$

Where:
n_{tr} = the natural vibration frequency
G = shear modulus of the material
I_P = the polar inertia moment
l = the length of the shaft
J = the mass inertia moment

The following factors influence the vibration frequency:
- the strength of the material, G, the shear modules;
- the shaft diameter, the polar inertia moment;
- the length of the shaft, l;
- the mass inertia moment, I_P of the rotating mass.

In practice, the following points may be adjusted:
- The mass inertia moment, J, of for instance, the propeller, fly-wheel and extra masses.
- The polar inertia moment, I_P.

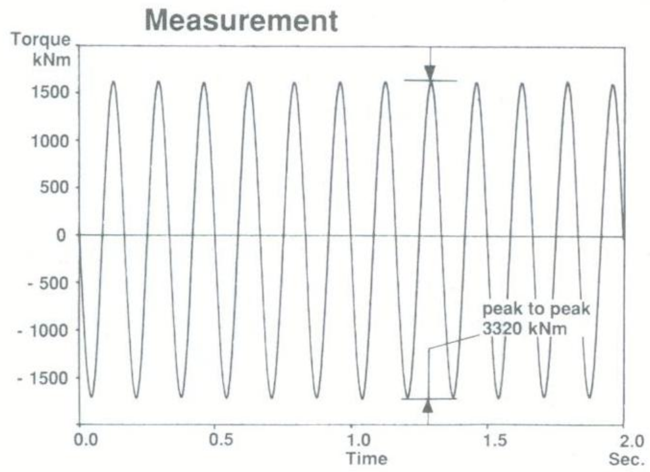
In the formula of the inertia moment the shaft is the shaft diameter ($I_P = \frac{1}{32} \times \pi \times D^4$) to the fourth power and the mass of the rotating parts of the first power.

Therefore, an increase in the shaft diameter is the easiest solution. If the shaft diameter is increased, the natural vibration frequency and the critical area of the system are moved to a higher engine speed.

In an existing propulsion installation one of the few ways to reduce the vibration frequency is to install a propeller with smaller a moment of inertia.
In this way the propeller has a slightly smaller diameter. In order to achieve an identical displacement, the propeller pitch must be increased.

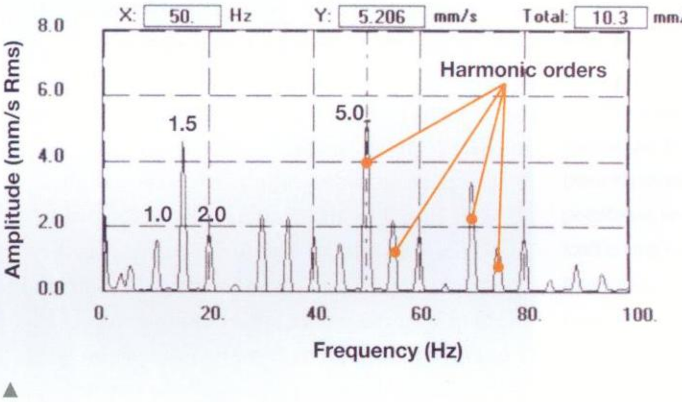
17.30 Measuring equipment

Today, measuring equipment, which meticulously registers the engine vibrations of a complete system, the so-called torsion diagram, is available.



▲

A torsiogram for a five-cylinder two-stroke MAN–B&W crosshead engine type 5 L 50 MC, at approximately 72 revs/min.



▲

A torsiogram for a four-stroke diesel engine.

Harmonic orders are shown: 1.0, 1.5, 2.0 and 5.0.

Most torsion diagrams show a wave-form measurement, which is a result of the fact that the shaft does not rotate at a constant angular velocity, but continuously decelerates and accelerates.
This is also referred to as the non-uniformity degree δ, of the engine.

δ = non-uniformity degree of the engine

$\delta = \frac{n\ max - n\ min}{n\ mean}$. This expression is as old as the internal combustion engine.

Torsional vibrations are unavoidable; this is why many engines are equipped with vibration dampers.

17.31 Mass-inertia moment of a flywheel

17.31.1 Introduction

Virtually all engines are provided with a fly-wheel or the combination fly-wheel- turning wheel. Engines with a small number of cylinders have in

By placing a fly-wheel on the crankshaft the fluctuating engine speed is levelled off. Accelerating and decelerating the rotating mass takes time and the piston strokes follow each other rapidly at a normal engine speed, thus producing a more stable engine speed.
The fly-wheel, then, serves as energy storage. The fly-wheel is no longer required to start the engine in multiple cylinder engines; these are then started electrically, mechanically or with air, but nevertheless it warrants a more regular engine speed.

3 Additional functions of the fly-wheel: use as a turning wheel

This can be manually performed with smaller engines, in which case the circumference fly-wheel has holes which allow for the turning of the crank shaft using a turning rod during inspections and repairs.
In larger engines the fly-wheels are equipped with a gear ring to which a turning engine can be connected. This usually works on electricity or air. Many starting engines working on compressed air have two circuits – one for extremely slow running or cranking, and one for rapid turning to

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17.31 Mass-inertia moment of a flywheel

17.31.1 Introduction

Virtually all engines are provided with a fly-wheel or the combination fly-wheel- turning wheel. Engines with a small number of cylinders have in proportion, a heavy 'thick' fly-wheel and multiple cylinder engines, such as V-engines usually have a light 'thin' fly-wheel.

17.31.2 Functions of the fly-wheel

1 Overcoming compression power

A single-cylinder four-stroke engine has two revolutions in one combustion process. In order to start this engine manually, a compression lever is used. This lever opens the exhaust valve, thus preventing compression. This enables the achievement a certain speed of the moving engine parts, including the relatively heavy fly-wheel by means of 'elbow steam'.
If the cranking is discontinued and the exhaust valve is closed by the decompression, the piston may compress the air. The fuel is injected at the end of the stroke and directly after T.D.C. position of the piston the power stroke is initiated.
The energy required to compress the piston is generated by the moving parts of the engine; especially by the fly-wheel. So, this fly-wheel serves as kinetic energy storage for starting the engine.

2 Levelling irregular engine speeds

In principle a single-cylinder four-stroke engine has an irregular speed. There are three piston strokes which cost energy, and only one stroke that provides energy.
By placing a fly-wheel on the crankshaft the fluctuating engine speed is levelled off. Accelerating and decelerating the rotating mass takes time and the piston strokes follow each other rapidly at a normal engine speed, thus producing a more stable engine speed.
The fly-wheel, then, serves as energy storage. The fly-wheel is no longer required to start the engine in multiple cylinder engines; these are then started electrically, mechanically or with air, but nevertheless it warrants a more regular engine speed.

3 Additional functions of the fly-wheel: use as a turning wheel

This can be manually performed with smaller engines, in which case the circumference fly-wheel has holes which allow for the turning of the crank shaft using a turning rod during inspections and repairs.
In larger engines the fly-wheels are equipped with a gear ring to which a turning engine can be connected. This usually works on electricity or air. Many starting engines working on compressed air have two circuits – one for extremely slow running or cranking, and one for rapid turning to start the engine.

4 Measuring crank positions

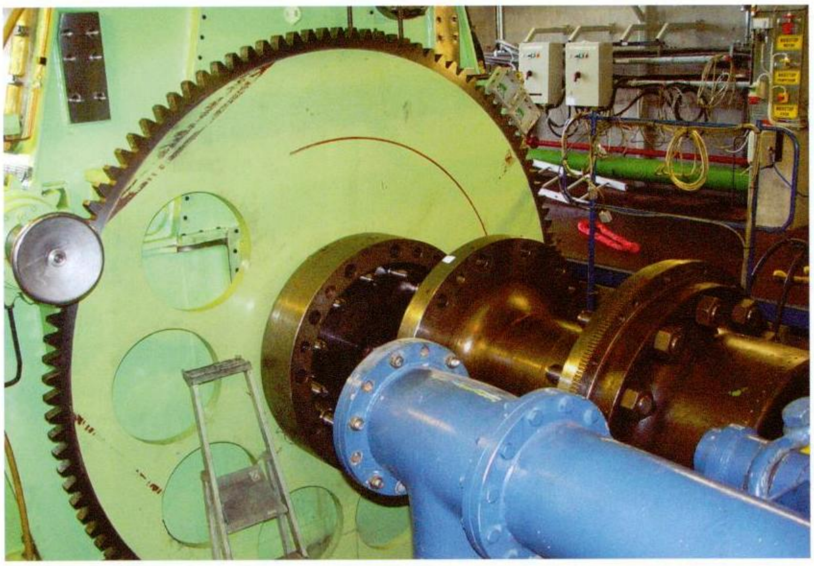
The final important function, of course, is to determine with the use of markers on the fly-wheel, for example the T.D.C. and B.D.C. position of the cylinders.

17.31.3 Examples of types of fly-wheel versions

▼

A twelve-cylinder four-stroke Perkins V-engine in the factory at Peterborough, England.

The fly wheel looks like a cylinder with a thick outer wall.

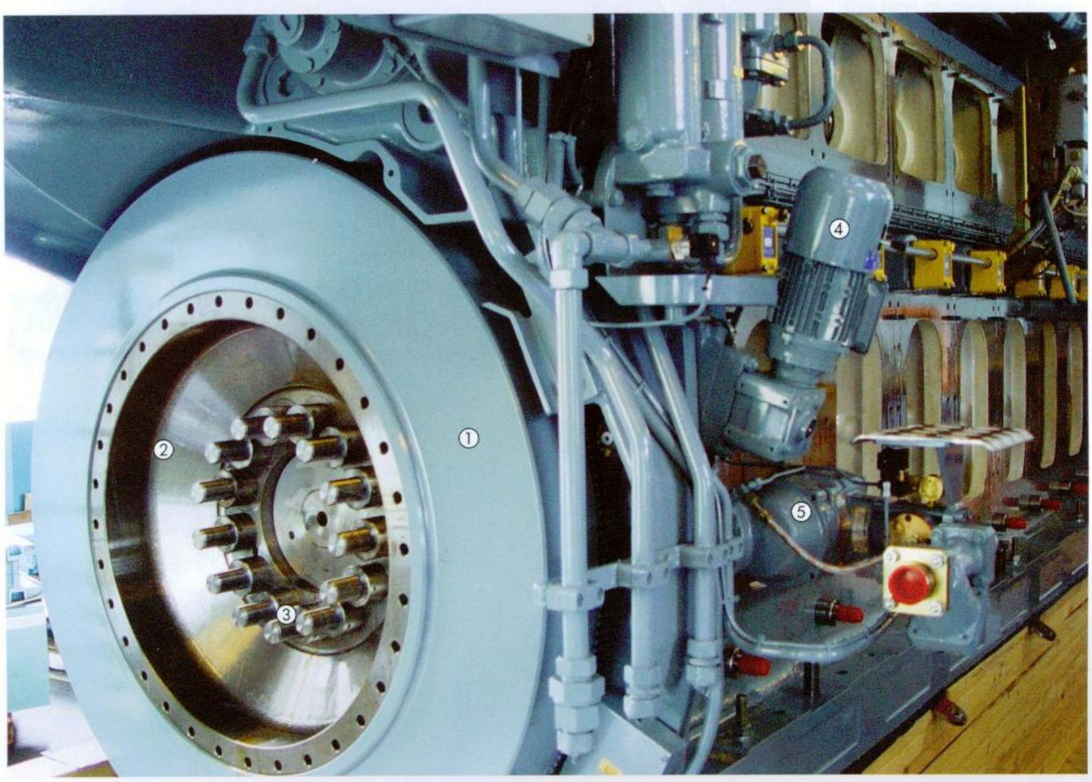


A turning-wheel/fly wheel in a two-stroke MAN–B&W 50 MC crosshead engine.

The disk is actually more like a turning wheel with left, the electric turning motor. In the turning-wheel are holes for balancing of crankshaft.

A large fly-wheel for a six-cylinder four-stroke Caterpillar–MaK diesel engine.

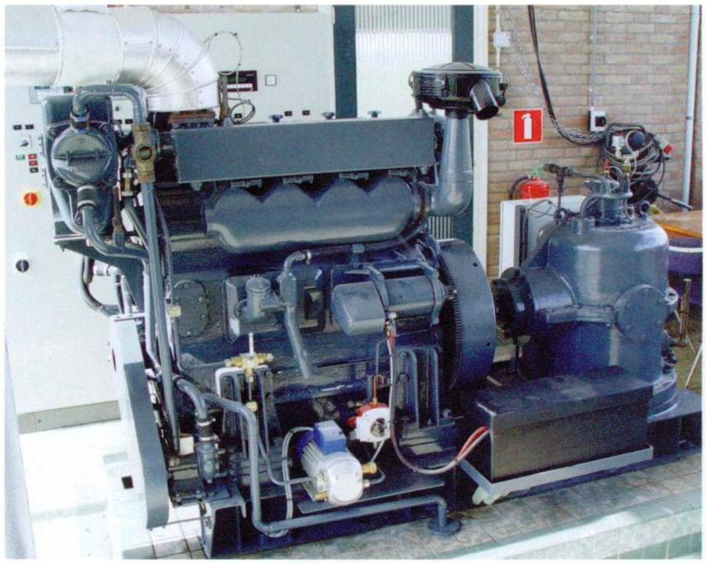
In the fly-wheel, the so-called turn holes are used to manually crank this category III engine.



The fly-wheel of a four-stroke line engine, manufacturer MAN–B&W.

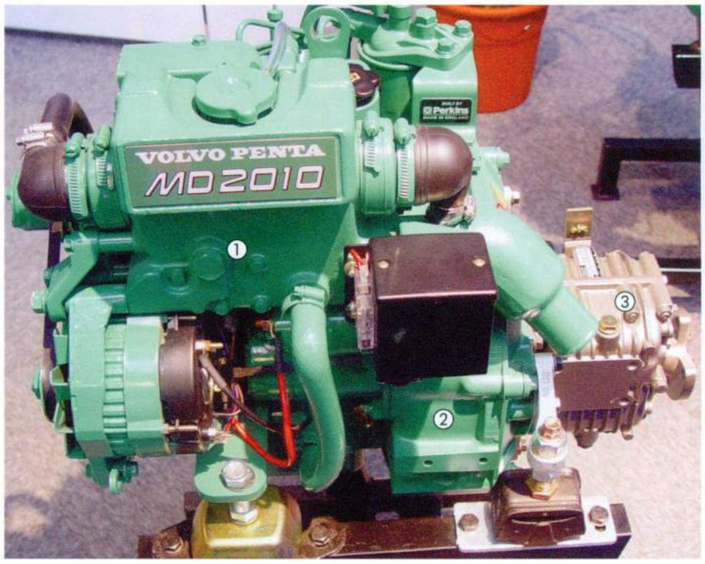
The fly-wheel is quite large. In the middle, the material has been removed as much as possible to ensure that the load on the crankshaft bearings is as low as possible.
Right; next to the fly-wheel, the electric turning motor and below, acting on the same starting-ring gear, a starting engine operating on compressed air.

1 *fly-wheel mass*
2 *removed section*
3 *fixing bolts on crankshaft*
4 *turning motor*
5 *start motor*



A four-cylinder four-stroke Kromhout diesel engine in a pumping station.

As this engine has only four cylinders and a relatively low speed (600 revs/min), a large fly-wheel is absolutely necessary. The engine is started electrically with a starting motor attached to an accumulator battery.



A small two-cylinder four-stroke diesel engine, a Volvo MD 2010, often used in yachts.

The large fly-wheel is mounted between the engine and the reduction gearing.

1 *engine block*
2 *fly-wheel casing*
3 *reduction gear*

17.31.4 The mass inertia moment of fly-wheels

The formula form of the mass inertia moment of fly-wheels:

$$J = \frac{1}{2} \times m_{disk} \times R^2$$

Where:

J = mass inertia moment in kgm^2
M_{disk} = mass of a disk in kg
R = radius of the centre of the mass

The mass inertia moment is also expressed as $G \times D^2$.

Where:

G = weight of the fly-wheel in kg
D = diameter where the entire fly-wheel mass is concentrated, expressed in metres.

17.31.5 Construction of fly-wheels

The fly-wheel is designed to be as light as possible to achieve a shaft load that is as low as possible. The further the mass centre is removed from the shaft, the higher the mass inertia moment and the greater the effects on the engine speed.
Most fly-wheels have a rim-like construction: a thin centre area for attachment to the shaft and a very thick outer ring for the required mass.

17.32 Examples of crankshafts, either with or without counterweights

▶

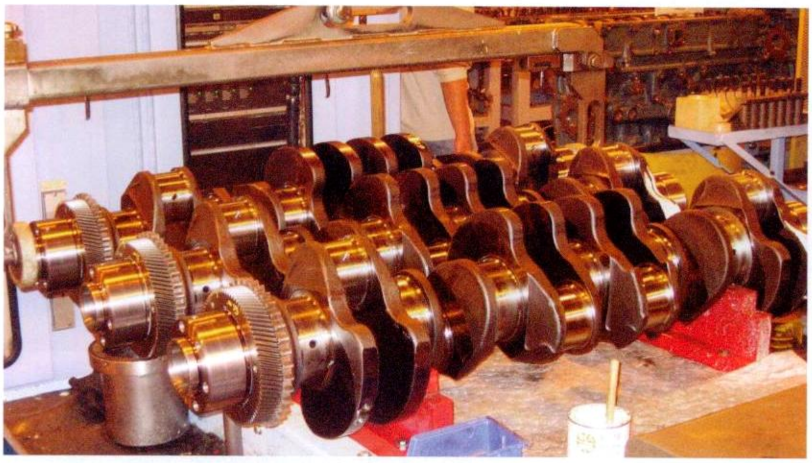
Mounting a crankshaft in a reverse V- block of a MTU-diesel engine.

Clearly shown is the circumference of the counter-/balancing weights. A radius is measured from the centre of the crankshaft.

◀

Mounting a crankshaft in an MTU-V diesel engine.

Every crankweb is has a counterweight.



◀

Three crankshafts in the Detroit Diesel factory in the United States.

This type of crankshaft has counterweights forged together with the cranks.

▶

In this V-engine crankshaft, it can be seen that the counterweights are directly opposite the crankpin, where the two pistons will be fixed.

1 *crankshaft*
2 *crankpin*
3 *counterweights*

▶

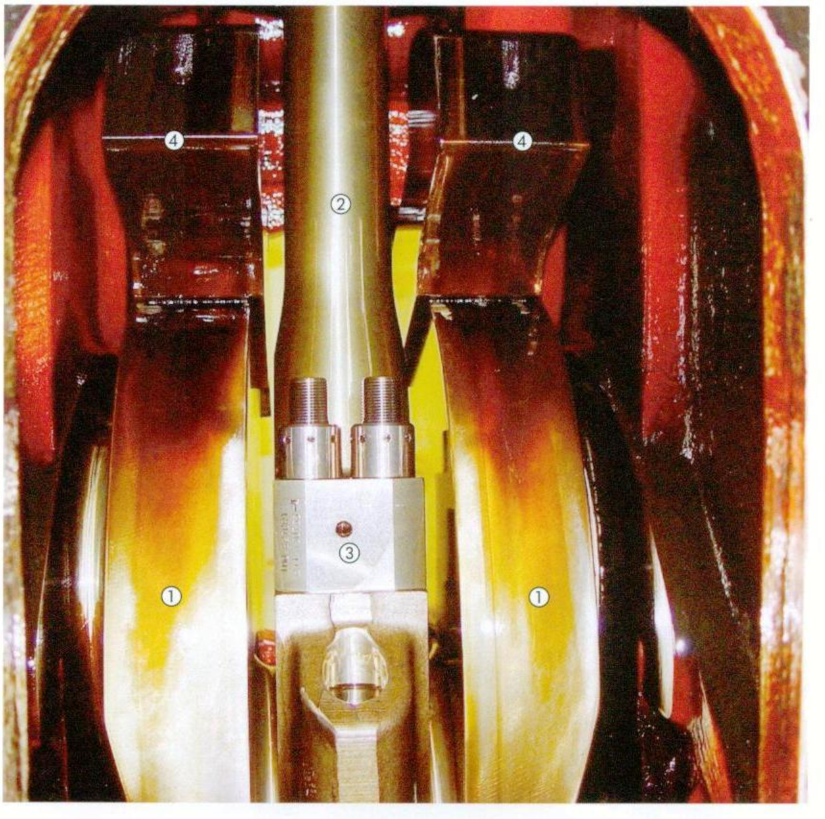
This Caterpillar V–3408 crankshaft as well as that of the V–3508 type does not have one crankpin for the two pistons, but a crankpin in two 'staggered' sections.

The reason: In this manner, the ignition angle of all eight cylinders is the same and therefore the torsional vibrations are lowest.

▶

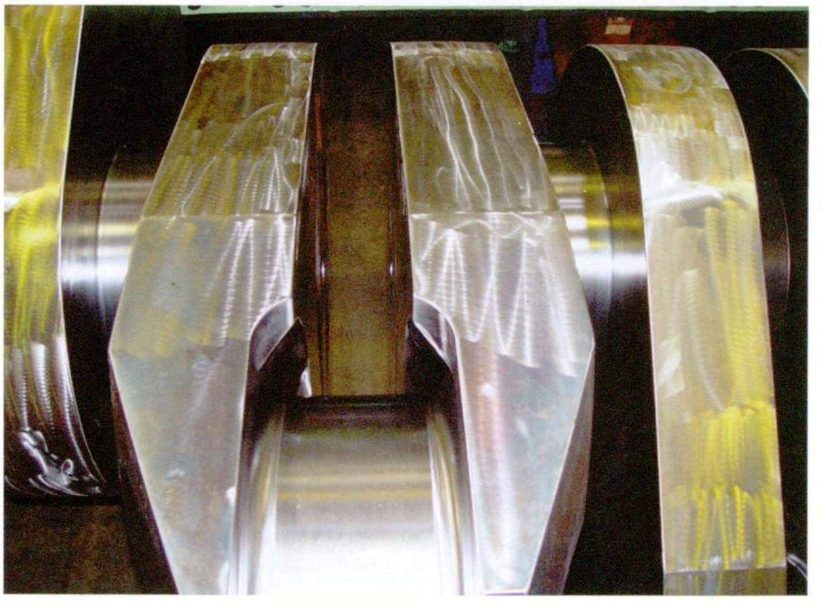
This Wärtsilä Sulzer 8 RTA 96 C is equipped with bolted counterweights.

1 *crank webs*
2 *connecting rod*
3 *crankpin bearing*
4 *counterweight*



▶

This MAN–B&W 8-50-MC two-stroke crosshead engine crankshaft does not have counterweights.



17.33 Combustion forces exerted on the driving gear

In order to get an idea of the extent of the forces, a rough calculation of the moment the piston is in T.D.C. position during the combustion stroke is made for all four engine categories.
The pressure and cylinder diameter are measured at the piston top position and put into the correct engine category.

Category I:
Pressure in top 30 bar
D = 50 mm.
Force = Surface × pressure

$$\text{Force} = \frac{\pi}{4} \times D^2 \times p$$

$$F = \frac{\pi}{4} \times 0{,}05^2 \times 3\text{MN} = 5887{,}5 \text{ N of } \mathbf{0{,}58 \text{ ton.}}$$

Category II:
Pressure in top 120 bar
D = 120 mm.
F = 135648 N of **13.5 tons**

Category III:
Pressure in top 200 bar
D = 320 mm.
F = 1,607,680 N or **160.7 tons**

Category IV:
Pressure in top 160 bar
D = 960 mm.
F = 11,575,296 N or **1157.5 tons**

These are extremely high forces which are rapidly generated and follow each other very rapidly.
All parts of the drive gear, such as the piston, gudgeon pin, connecting rod, crankpin, crankshaft and the bearings must be strong enough to withstand these pulsating loads for tens of thousands of hours. Therefore, it is of great importance that the moving parts in the drive gear are sufficiently separated by a film of lubricating oil.
The pressure of the lubricating oil film is often high and can rise from 100 to over 1.000 bar!
It is therefore important that the journals are absolutely round and that the bearing caps fit properly in order for the rotating shaft to separate the metal parts as 'a high-pressure lubricating-oil pump'.

Lateral forces of the piston movement on the cylinder liner.

As a consequence, these are also high. In four-stroke engines the piston has to transfer these forces to the engine block via the cylinder liner by means of the piston skirt. In two-stroke crosshead engines the crosshead ensures that the forces are transferred to the welded A-frame via the guides and guide shoes. These lateral forces in the various engine categories roughly vary from 100 N to 2000,000 N or 0.1 to 200 tons!