



MAN B&W K90ME9-TII

Project Guide

Electronically Controlled Two-stroke Engines

This Project Guide is intended to provide the information necessary for the layout of a marine propulsion plant.

The information is to be considered as **preliminary**. It is intended for the project stage only and subject to modification in the interest of technical progress. The Project Guide provides the general technical data available at the date of issue.

It should be noted that all figures, values, measurements or information about performance stated in this project guide are **for guidance only** and should not be used for detailed design purposes or as a substitute for specific drawings and instructions prepared for such purposes.

Data updates

Data not finally calculated at the time of issue is marked 'Available on request'. Such data may be made available at a later date, however, for a specific project the data can be requested. Pages and table entries marked 'Not applicable' represent an option, function or selection which is not valid.

The latest, most current version of the individual Project Guide sections are available on the Internet at: www.mandieselturbo.com under Quicklinks: 'Marine Engines & Systems' → 'Low Speed'.

Extent of Delivery

The final and binding design and outlines are to be supplied by our licensee, the engine maker, see Chapter 20 of this Project Guide.

In order to facilitate negotiations between the yard, the engine maker and the customer, a set of 'Extent of Delivery' forms is available in which the basic and the optional executions are specified.

Electronic versions

This Project Guide book and the 'Extent of Delivery' forms are available on a DVD and can also be found on the Internet at: www.mandieselturbo.com under Quicklinks: 'Marine Engines & Systems' → 'Low Speed', where they can be downloaded.

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Engine Design

1

The ME Tier II Engine

The ever valid requirement of ship operators is to obtain the lowest total operational costs, and especially the lowest possible specific fuel oil consumption at any load, and under the prevailing operating conditions.

However, low-speed two-stroke main engines of the MC type, with a chain driven camshaft, have limited flexibility with regard to fuel injection and exhaust valve activation, which are the two most important factors in adjusting the engine to match the prevailing operating conditions.

A system with electronically controlled hydraulic activation provides the required flexibility, and such systems form the core of the ME 'Engine Control System', described later in detail in Chapter 16.

Concept of the ME engine

The ME engine concept consists of a hydraulic-mechanical system for activation of the fuel injection and the exhaust valves. The actuators are electronically controlled by a number of control units forming the complete Engine Control System.

MAN Diesel has specifically developed both the hardware and the software in-house, in order to obtain an integrated solution for the Engine Control System.

The fuel pressure booster consists of a simple plunger powered by a hydraulic piston activated by oil pressure. The oil pressure is controlled by an electronically controlled proportional valve.

The exhaust valve is opened hydraulically by means of a two-stage exhaust valve actuator activated by the control oil from an electronically controlled proportional valve. The exhaust valves are closed by the 'air spring'.

In the hydraulic system, the normal lube oil is used as the medium. It is filtered and pressurised by a Hydraulic Power Supply unit mounted on the engine or placed in the engine room.

The starting valves are opened pneumatically by electronically controlled 'On/Off' valves, which make it possible to dispense with the mechanically activated starting air distributor.

By electronic control of the above valves according to the measured instantaneous crankshaft position, the Engine Control System fully controls the combustion process.

System flexibility is obtained by means of different 'Engine running modes', which are selected either automatically, depending on the operating conditions, or manually by the operator to meet specific goals. The basic running mode is 'Fuel economy mode' to comply with IMO NOx emission limitation.

Engine design and IMO regulation compliance

The ME-C engine is the shorter, more compact version of the MC engine. It is well suited wherever a small engine room is requested, for instance in container vessels.

The ME-GI is a dual fuel engine burning natural gas, otherwise sharing the same compact design as the ME-C engine. It is designed for the highly specialised LNG carrier market.

For MAN B&W ME/ME-C/ME-GI-TII designated engines, the design and performance parameters have been upgraded and optimised to comply with the International Maritime Organisation (IMO) Tier II emission regulations.

The potential derating and part load SFOC figures for the Tier II engines have also been updated.

For engines built to comply with IMO Tier I emission regulations, please refer to the Marine Engine IMO Tier I Project Guide.

ME Advantages

The advantages of the ME range of engines are quite comprehensive, as seen below:

- Lower SFOC and better performance parameters thanks to variable electronically controlled timing of fuel injection and exhaust valves at any load
- Appropriate fuel injection pressure and rate shaping at any load
- Improved emission characteristics, with smokeless operation
- Easy change of operating mode during operation
- Simplicity of mechanical system with well-proven simple fuel injection technology familiar to any crew
- Control system with more precise timing, giving better engine balance with equalized thermal load in and between cylinders
- System comprising performance, adequate monitoring and diagnostics of engine for longer time between overhauls
- Lower rpm possible for manoeuvring
- Better acceleration, astern and crash stop performance
- Integrated Alpha Cylinder Lubricators
- Up-gradable to software development over the lifetime of the engine

It is a natural consequence of the above that more features and operating modes are feasible with our fully integrated control system and, as such, will be retrofittable and eventually offered to owners of ME engines.

Differences between MC/MC-C and ME/ME-C engines

The electro-hydraulic control mechanisms of the ME engine replace the following components of the conventional MC engine:

- Chain drive for camshaft
- Camshaft with fuel cams, exhaust cams and indicator cams
- Fuel pump actuating gear, including roller guides and reversing mechanism
- Conventional fuel pressure booster and VIT system
- Exhaust valve actuating gear and roller guides
- Engine driven starting air distributor
- Electronic governor with actuator
- Regulating shaft
- Engine side control console
- Mechanical cylinder lubricators.

The Engine Control System of the ME engine comprises:

- Control units
- Hydraulic power supply unit
- Hydraulic cylinder units, including:
 - Electronically controlled fuel injection, and
 - Electronically controlled exhaust valve activation
- Electronically controlled starting air valves
- Electronically controlled auxiliary blowers
- Integrated electronic governor functions
- Tacho system
- Electronically controlled Alpha lubricators

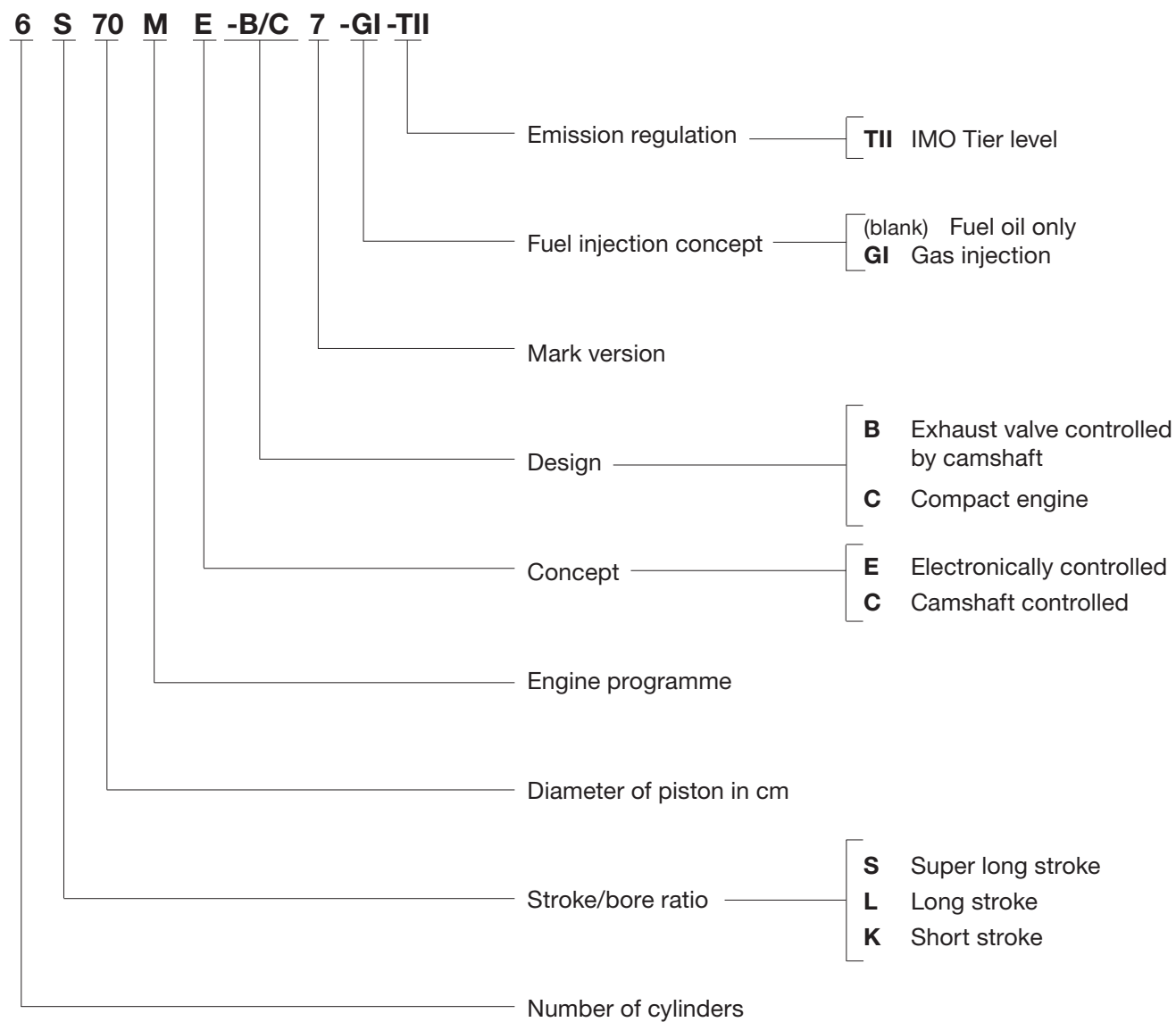
- Local Operating Panel (LOP)
- MAN Diesel PMI system, type PT/S off-line, cylinder pressure monitoring system.

The system can be further extended by optional systems, such as:

- Condition Monitoring System, CoCoS-EDS on-line

The main features of the ME engine are described on the following pages.

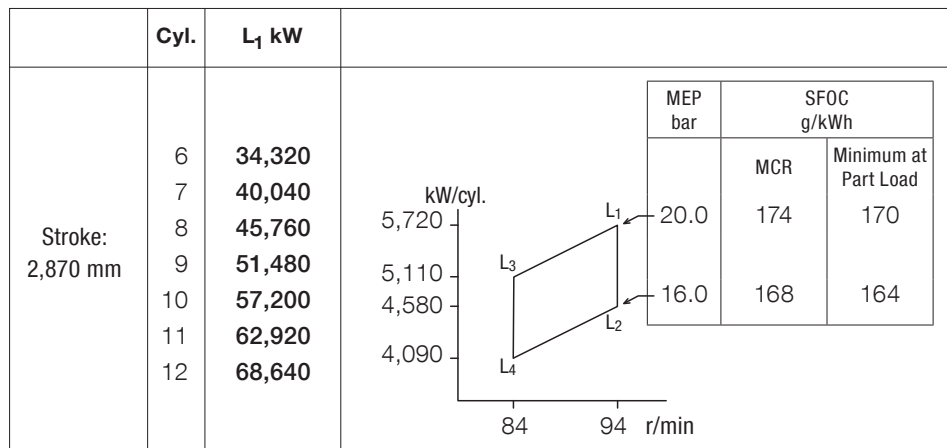
Engine Type Designation



Power, Speed and Lubricating Oil

MAN B&W K90ME9-TII

Power and Speed



Fuel and lubricating oil consumption

At load Layout point	Specific fuel oil consumption g/kWh		Lubricating oil consumption	
	With high efficiency turbocharger		System oil Approximate g/kWh	MAN B&W Alpha cylinder lubricator
	100%	70%		
L ₁ and L ₂	174	170	0.1	0.65
L ₃ and L ₄	168	164		

Fig 1.03.01: Power, speed, fuel and lubrication oil

Engine Power Range and Fuel Oil Consumption

Engine Power

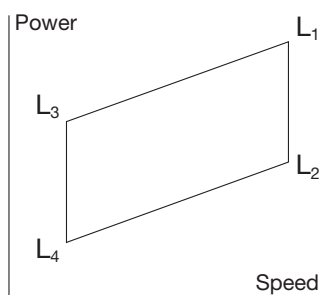
The following tables contain data regarding the power, speed and specific fuel oil consumption of the engine.

Engine power is specified in kW for each cylinder number and layout points L_1 , L_2 , L_3 and L_4 :

Discrepancies between kW and metric horsepower (1 BHP = 75 kpm/s = 0.7355 kW) are a consequence of the rounding off of the BHP values.

L_1 designates nominal maximum continuous rating (nominal MCR), at 100% engine power and 100% engine speed.

L_2 , L_3 and L_4 designate layout points at the other three corners of the layout area, chosen for easy reference.



178 51 48-9.0

Fig. 1.04.01: Layout diagram for engine power and speed

Overload corresponds to 110% of the power at MCR, and may be permitted for a limited period of one hour every 12 hours.

The engine power figures given in the tables remain valid up to tropical conditions at sea level as stated in IACS M28 (1978), i.e.:

Blower inlet temperature 45 °C
 Blower inlet pressure 1000 mbar
 Seawater temperature 32 °C
 Relative humidity 60%

Specific Fuel Oil Consumption (SFOC)

The figures given in this folder represent the values obtained when the engine and turbocharger are matched with a view to obtaining the lowest possible SFOC values while also fulfilling the IMO NOX Tier II emission limitations.

Stricter emission limits can be met on request, using proven technologies.

The SFOC figures are given in g/kWh with a tolerance of 5% and are based on the use of fuel with a lower calorific value of 42,700 kJ/kg (~10,200 kcal/kg) at ISO conditions:

Ambient air pressure 1,000 mbar
 Ambient air temperature 25 °C
 Cooling water temperature 25 °C

Although the engine will develop the power specified up to tropical ambient conditions, specific fuel oil consumption varies with ambient conditions and fuel oil lower calorific value. For calculation of these changes, see Chapter 2.

Lubricating oil data

The cylinder oil consumption figures stated in the tables are valid under normal conditions.

During running-in periods and under special conditions, feed rates of up to 1.5 times the stated values should be used.

Performance Curves

This section is available on request

Updated engine and capacities data is available from the CEAS program on www.mandiesel.com under 'Marine' → 'Low speed' → 'CEAS Engine Room Dimensions'.

ME Engine Description

Please note that engines built by our licensees are in accordance with MAN Diesel drawings and standards but, in certain cases, some local standards may be applied; however, all spare parts are interchangeable with MAN Diesel designed parts.

Some components may differ from MAN Diesel's design because of local production facilities or the application of local standard components.

In the following, reference is made to the item numbers specified in the 'Extent of Delivery' (EoD) forms, both for the 'Basic' delivery extent and for some 'Options'.

Bedplate and Main Bearing

The bedplate is made with the thrust bearing in the aft end of the engine. The bedplate consists of high, welded, longitudinal girders and welded cross girders with cast steel bearing supports.

For fitting to the engine seating in the ship, long, elastic holding-down bolts, and hydraulic tightening tools are used.

The bedplate is made without taper for engines mounted on epoxy chocks.

The oil pan, which is made of steel plate and is welded to the bedplate, collects the return oil from the forced lubricating and cooling oil system. The oil outlets from the oil pan are normally vertical and are provided with gratings.

Horizontal outlets at both ends can be arranged for some cylinder numbers, however this must be confirmed by the engine builder.

The main bearings consist of thin walled steel shells lined with bearing metal. The main bearing bottom shell can be rotated out and in by means of special tools in combination with hydraulic tools for lifting the crankshaft. The shells are kept in position by a bearing cap.

Frame Box

The frame box is of welded design. On the exhaust side, it is provided with relief valves for each cylinder while, on the manoeuvring side, it is provided with a large hinged door for each cylinder. The crosshead guides are welded on to the frame box.

The frame box is bolted to the bedplate. The bedplate, frame box and cylinder frame are tightened together by stay bolts.

Cylinder Frame and Stuffing Box

The cylinder frame is cast, with the exception of the S65ME-C which is welded, and is provided with access covers for cleaning the scavenge air space, if required, and for inspection of scavenge ports and piston rings from the manoeuvring side. Together with the cylinder liner it forms the scavenge air space.

The cylinder frame is fitted with pipes for the piston cooling oil inlet. The scavenge air receiver, turbocharger, air cooler box and gallery brackets are located on the cylinder frame. At the bottom of the cylinder frame there is a piston rod stuffing box, provided with sealing rings for scavenge air, and with oil scraper rings which prevent crankcase oil from coming up into the scavenge air space.

Drains from the scavenge air space and the piston rod stuffing box are located at the bottom of the cylinder frame.

Cylinder Liner

The cylinder liner is made of alloyed cast iron and is suspended in the cylinder frame with a low-situated flange. The top of the cylinder liner is fitted with a cooling jacket. The cylinder liner has scavenge ports and drilled holes for cylinder lubrication.

Cylinder liners prepared for installation of temperature sensors is basic execution on engines type 98 and 90 as well as on K80ME-C9 while an option on all other engines.

Cylinder Cover

The cylinder cover is of forged steel, made in one piece, and has bores for cooling water. It has a central bore for the exhaust valve, and bores for the fuel valves, a starting valve and an indicator valve.

The cylinder cover is attached to the cylinder frame with studs and nuts tightened with hydraulic jacks.

Crankshaft

The crankshaft is of the semi-built type, made from forged or cast steel throws. For engines with 9 cylinders or more, the crankshaft is supplied in two parts.

At the aft end, the crankshaft is provided with the collar for the thrust bearing, a flange for fitting the gear wheel for the step-up gear to the hydraulic power supply unit if fitted on the engine, and the flange for the turning wheel and for the coupling bolts to an intermediate shaft.

At the front end, the crankshaft is fitted with the collar for the axial vibration damper and a flange for the fitting of a turning wheel. The flange can also be used for a Power Take Off, if so desired.

Coupling bolts and nuts for joining the crankshaft together with the intermediate shaft are not normally supplied.

Thrust Bearing

The propeller thrust is transferred through the thrust collar, the segments, and the bedplate, to the end chocks and engine seating, and thus to the ship's hull.

The thrust bearing is located in the aft end of the engine. The thrust bearing is of the B&W-Michell type, and consists primarily of a thrust collar on the crankshaft, a bearing support, and segments of steel lined with white metal.

Engines type 60 and larger with 9 cylinders or more will be specified with the 360° degree type thrust bearing, while the 240° degree type is used in all other engines. MAN Diesel's flexible thrust cam design is used for the thrust collar on a range of engine types.

The thrust shaft is an integrated part of the crankshaft and it is lubricated by the engine's lubricating oil system.

Step-up Gear

In case of engine driven HPS, the hydraulic oil pumps are mounted on the aft of the engine, and are driven from the crankshaft via step-up gear. The step-up gear is lubricated from the main engine system.

Turning Gear and Turning Wheel

The turning wheel is fitted to the thrust shaft, and it is driven by a pinion on the terminal shaft of the turning gear, which is mounted on the bedplate. The turning gear is driven by an electric motor with built-in gear with brake.

A blocking device prevents the main engine from starting when the turning gear is engaged. Engagement and disengagement of the turning gear is effected manually by an axial movement of the pinion.

The control device for the turning gear, consisting of starter and manual control box, can be ordered as an option.

Axial Vibration Damper

The engine is fitted with an axial vibration damper, mounted on the fore end of the crankshaft. The damper consists of a piston and a split-type housing located forward of the foremost main bearing.

The piston is made as an integrated collar on the main crank journal, and the housing is fixed to the main bearing support. For functional check of the vibration damper a mechanical guide is fitted, while an electronic vibration monitor can be supplied as an option.

Tuning Wheel / Torsional Vibration Damper

A tuning wheel or torsional vibration damper may have to be ordered separately, depending on the final torsional vibration calculations.

Connecting Rod

The connecting rod is made of forged or cast steel and provided with bearing caps for the crosshead and crankpin bearings.

The crosshead and crankpin bearing caps are secured to the connecting rod with studs and nuts tightened by means of hydraulic jacks.

The crosshead bearing consists of a set of thin-walled steel shells, lined with bearing metal. The crosshead bearing cap is in one piece, with an angular cut-out for the piston rod.

The crankpin bearing is provided with thin-walled steel shells, lined with bearing metal. Lube oil is supplied through ducts in the crosshead and connecting rod.

Piston

The piston consists of a piston crown and piston skirt. The piston crown is made of heat-resistant steel. A piston cleaning ring located in the very top of the cylinder liner scrapes off excessive ash and carbon formations on the piston top-land.

The piston has four ring grooves which are hard-chrome plated on both the upper and lower surfaces of the grooves. The uppermost piston ring is of the CPR type (Controlled Pressure Relief), whereas the other three piston rings all have an oblique cut. The uppermost piston ring is higher than the others. All four rings are alu-coated on the outer surface for running-in.

The piston skirt is made of cast iron with a bronze band.

Piston Rod

The piston rod is of forged steel and is surface-hardened on the running surface for the stuffing box. The piston rod is connected to the crosshead with four bolts. The piston rod has a central bore which, in conjunction with a cooling oil pipe, forms the inlet and outlet for cooling oil.

Crosshead

The crosshead is of forged steel and is provided with cast steel guide shoes with white metal on the running surface.

The telescopic pipe for oil inlet and the pipe for oil outlet are mounted on the guide shoes.

Scavenge Air System

The air intake to the turbocharger takes place directly from the engine room through the turbocharger intake silencer. From the turbocharger, the air is led via the charging air pipe, air cooler and scavenge air receiver to the scavenge ports of the cylinder liners, see Chapter 14.

Scavenge Air Cooler

For each turbocharger is fitted a scavenge air cooler of the mono-block type designed for sea-water cooling, alternatively, a central cooling system with freshwater can be chosen. The working pressure is up to 4.5 bar.

The scavenge air cooler is so designed that the difference between the scavenge air temperature and the water inlet temperature at specified MCR can be kept at about 12 °C.

Auxiliary Blower

The engine is provided with electrically-driven scavenge air blowers. The suction side of the blowers is connected to the scavenge air space after the air cooler.

Between the air cooler and the scavenge air receiver, non-return valves are fitted which automatically close when the auxiliary blowers supply the air.

The auxiliary blowers will start operating consecutively before the engine is started in order to ensure sufficient scavenge air pressure to obtain a safe start.

Further information is given in Chapter 14.

Exhaust Gas System

From the exhaust valves, exhaust gas is led to the exhaust gas receiver where the fluctuating pressure from the individual cylinders is equalised, and the total volume of gas is led to the turbocharger(s). After the turbocharger(s), the gas is led to the external exhaust pipe system.

Compensators are fitted between the exhaust valves and the receiver, and between the receiver and the turbocharger(s).

The exhaust gas receiver and exhaust pipes are provided with insulation, covered by galvanised steel plating.

A protective grating is installed between the exhaust gas receiver and the turbocharger.

Exhaust Turbocharger

The engines can be fitted with either MAN, ABB or MHI turbochargers. As an option, MAN TCA and ABB A100-L turbochargers can be delivered with variable nozzle technology that reduces the fuel consumption at part load by controlling the scavenge air pressure.

The turbocharger choice selection is described in Chapter 3, and the exhaust gas system in Chapter 15.

Reversing

Reversing of the engine is performed electronically and controlled by the Engine Control System, by changing the timing of the fuel injection, the exhaust valve activation and the starting valves.

The Hydraulic Power Supply

The hydraulic power supply (HPS) filters and pressurises the lube oil for use in the hydraulic system. Depending on the engine type, the HPS consists of 2-4 pumps driven either mechanically by the engine or electrically. The hydraulic pressure is 300 bar.

An electrically driven HPS can be mounted on the engine, usually aft, or in the engine room.

The engine driven HPS is mounted aft for engines with chain drive aft (8 cylinders or less), and at the middle for engines with chain drive located in the middle (9 cylinders or more).

A combined HPS, mechanically driven with electrically driven start-up/back-up pumps for take-home power, is available as an option for ME/ME-C engines type 98-60 while basic execution for S50ME-C.

Hydraulic Cylinder Unit

The hydraulic cylinder unit (HCU), one per cylinder, consists of a base plate on which a distributor block is mounted. The distributor block is fitted with one or more accumulators to ensure that the necessary hydraulic oil peak flow is available during the fuel injection sequence.

The distributor block serves as a mechanical support for the hydraulically activated fuel pressure booster and the hydraulically activated exhaust valve actuator.

Fuel Oil Pressure Booster

The engine is provided with one hydraulically activated fuel oil pressure booster for each cylinder.

Fuel injection is activated by a multi-way valve (FIVA), which is electronically controlled by the Cylinder Control Unit (CCU) of the Engine Control System.

Further information is given in Section 7.01.

Fuel Valves and Starting Air Valve

The cylinder cover is equipped with two or three fuel valves, starting air valve, and indicator cock.

The opening of the fuel valves is controlled by the high pressure fuel oil created by the fuel oil pressure booster, and the valves are closed by a spring.

An automatic vent slide allows circulation of fuel oil through the valve and high pressure pipes when the engine is stopped. The vent slide also prevents the compression chamber from being filled up with fuel oil in the event that the valve spindle sticks. Oil from the vent slide and other drains is led away in a closed system.

The fuel oil high-pressure pipes are of the double-wall type with built-in conical support. The pipes are insulated but not heated.

The mechanically driven starting air distributor used on the MC engines has been replaced by one solenoid valve per cylinder, controlled by the CCUs of the Engine Control System.

Slow turning before starting is a program incorporated in the basic Engine Control System.

The starting air system is described in detail in Section 13.01.

The starting valve is opened by control air and is closed by a spring. The integrated Engine Control System controls the starting valve timing.

Exhaust Valve

The exhaust valve consists of the valve housing and the valve spindle. The valve housing is made of cast iron and is arranged for water cooling. The housing is provided with a water cooled bottom piece of steel with a flame hardened seat of the W-seat design. The exhaust valve spindle is made of Nimonic. The housing is provided with a spindle guide.

The exhaust valve is tightened to the cylinder cover with studs and nuts. The exhaust valve is opened hydraulically by the electronic valve activation system and is closed by means of air pressure.

The operation of the exhaust valve is controlled by the proportional valve which also activates the fuel injection.

In operation, the valve spindle slowly rotates, driven by the exhaust gas acting on small vanes fixed to the spindle.

Sealing of the exhaust valve spindle guide is provided by means of Controlled Oil Level (COL), an oil bath in the bottom of the air cylinder, above the sealing ring. This oil bath lubricates the exhaust valve spindle guide and sealing ring as well.

Indicator Cock

The engine is fitted with an indicator cock to which the PMI pressure transducer can be connected.

MAN B&W Alpha Cylinder Lubrication

The electronically controlled MAN B&W Alpha cylinder lubrication system is applied to the ME engines, and controlled by the ME Engine Control System.

The main advantages of the MAN B&W Alpha cylinder lubrication system, compared with the conventional mechanical lubricator, are:

- Improved injection timing
- Increased dosage flexibility
- Constant injection pressure
- Improved oil distribution in the cylinder liner
- Possibility for prelubrication before starting.

More details about the cylinder lubrication system can be found in Chapter 9.

Gallery Arrangement

The engine is provided with gallery brackets, stanchions, railings and platforms (exclusive of ladders). The brackets are placed at such a height as to provide the best possible overhauling and inspection conditions.

Some main pipes of the engine are suspended from the gallery brackets, and the topmost gallery platform on the manoeuvring side is provided with overhauling holes for the pistons.

The engine is prepared for top bracings on the exhaust side, or on the manoeuvring side.

Piping Arrangements

The engine is delivered with piping arrangements for:

- Fuel oil
- Heating of fuel oil pipes
- Lubricating oil, piston cooling oil and hydraulic oil pipes
- Cylinder lubricating oil
- Cooling water to scavenge air cooler
- Jacket and turbocharger cooling water
- Cleaning of turbocharger
- Fire extinguishing in scavenge air space
- Starting air
- Control air
- Oil mist detector
- Various drain pipes.

All piping arrangements are made of steel piping, except the control air and steam heating of fuel pipes, which are made of copper.

The pipes are provided with sockets for local instruments, alarm and safety equipment and, furthermore, with a number of sockets for supplementary signal equipment. Chapter 18 deals with the instrumentation.

Engine Cross Section

This section is available on request

Engine Layout and Load Diagrams, SFOC

2

Engine Layout and Load Diagrams

Introduction

The effective power 'P' of a diesel engine is proportional to the mean effective pressure p_e and engine speed 'n', i.e. when using 'c' as a constant:

$$P = c \times p_e \times n$$

so, for constant mep, the power is proportional to the speed:

$$P = c \times n^1 \text{ (for constant mep)}$$

When running with a Fixed Pitch Propeller (FPP), the power may be expressed according to the propeller law as:

$$P = c \times n^3 \text{ (propeller law)}$$

Thus, for the above examples, the power P may be expressed as a power function of the speed 'n' to the power of 'i', i.e.:

$$P = c \times n^i$$

Fig. 2.01.01 shows the relationship for the linear functions, $y = ax + b$, using linear scales.

The power functions $P = c \times n^i$ will be linear functions when using logarithmic scales:

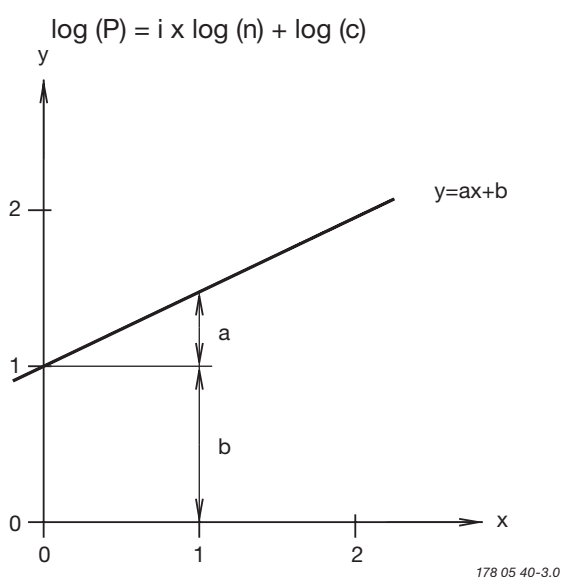


Fig. 2.01.01: Straight lines in linear scales

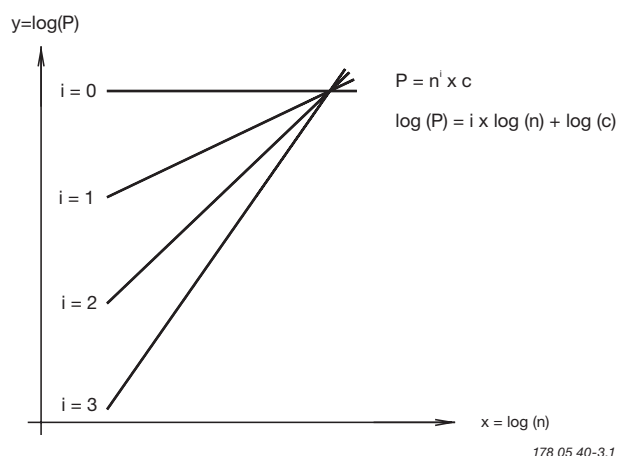


Fig. 2.01.02: Power function curves in logarithmic scales

Thus, propeller curves will be parallel to lines having the inclination $i = 3$, and lines with constant mep will be parallel to lines with the inclination $i = 1$.

Therefore, in the Layout Diagrams and Load Diagrams for diesel engines, logarithmic scales are used, giving simple diagrams with straight lines.

Propulsion and Engine Running Points

Propeller curve

The relation between power and propeller speed for a fixed pitch propeller is as mentioned above described by means of the propeller law, i.e. the third power curve:

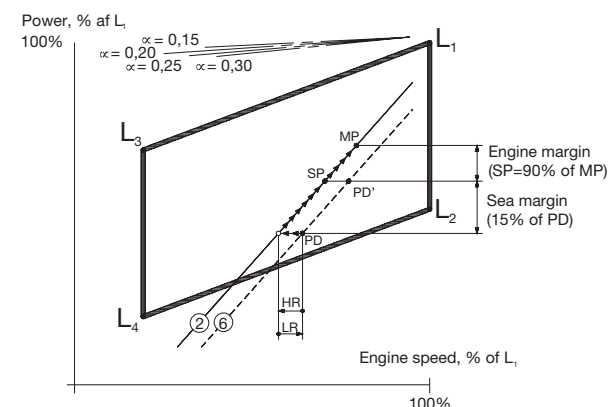
$$P = c \times n^3, \text{ in which:}$$

P = engine power for propulsion
n = propeller speed
c = constant

Propeller design point

Normally, estimates of the necessary propeller power and speed are based on theoretical calculations for loaded ship, and often experimental tank tests, both assuming optimum operating conditions, i.e. a clean hull and good weather. The combination of speed and power obtained may be called the ship's propeller design point (PD),

placed on the light running propeller curve 6. See below figure. On the other hand, some shipyards, and/or propeller manufacturers sometimes use a propeller design point (PD) that incorporates all or part of the so-called sea margin described below.



- Line 2 Propulsion curve, fouled hull and heavy weather (heavy running), recommended for engine layout
- Line 6 Propulsion curve, clean hull and calm weather (light running), for propeller layout
- MP Specified MCR for propulsion
- SP Continuous service rating for propulsion
- PD Propeller design point
- HR Heavy running
- LR Light running

178 05 41-5.3

Fig. 2.01.03: Ship propulsion running points and engine layout

Fouled hull

When the ship has sailed for some time, the hull and propeller become fouled and the hull's resistance will increase. Consequently, the ship's speed will be reduced unless the engine delivers more power to the propeller, i.e. the propeller will be further loaded and will be heavy running (HR).

As modern vessels with a relatively high service speed are prepared with very smooth propeller and hull surfaces, the gradual fouling after sea trial will increase the hull's resistance and make the propeller heavier running.

Sea margin and heavy weather

If, at the same time the weather is bad, with head winds, the ship's resistance may increase compared to operating in calm weather conditions. When determining the necessary engine power, it is normal practice to add an extra power margin,

the so-called sea margin, which is traditionally about 15% of the propeller design (PD) power.

Engine layout (heavy propeller)

When determining the necessary engine layout speed that considers the influence of a heavy running propeller for operating at high extra ship resistance, it is (compared to line 6) recommended to choose a heavier propeller line 2. The propeller curve for clean hull and calm weather line 6 may then be said to represent a 'light running' (LR) propeller.

Compared to the heavy engine layout line 2, we recommend using a light running of **3.0-7.0%** for design of the propeller.

Engine margin

Besides the sea margin, a so-called 'engine margin' of some 10% or 15% is frequently added. The corresponding point is called the 'specified MCR for propulsion' (MP), and refers to the fact that the power for point SP is 10% or 15% lower than for point MP.

Point MP is identical to the engine's specified MCR point (M) unless a main engine driven shaft generator is installed. In such a case, the extra power demand of the shaft generator must also be considered.

Constant ship speed lines

The constant ship speed lines α , are shown at the very top of the figure. They indicate the power required at various propeller speeds in order to keep the same ship speed. It is assumed that, for each ship speed, the optimum propeller diameter is used, taking into consideration the total propulsion efficiency. See definition of α in Section 2.02.

Note:

Light/heavy running, fouling and sea margin are overlapping terms. Light/heavy running of the propeller refers to hull and propeller deterioration and heavy weather, whereas sea margin i.e. extra power to the propeller, refers to the influence of the wind and the sea. However, the degree of light running must be decided upon experience from the actual trade and hull design of the vessel.

Propeller diameter and pitch, influence on the optimum propeller speed

In general, the larger the propeller diameter D , the lower is the optimum propeller speed and the kW required for a certain design draught and ship speed, see curve D in the figure below.

The maximum possible propeller diameter depends on the given design draught of the ship, and the clearance needed between the propeller and the aft body hull and the keel.

The example shown in the figure is an 80,000 dwt crude oil tanker with a design draught of 12.2 m and a design speed of 14.5 knots.

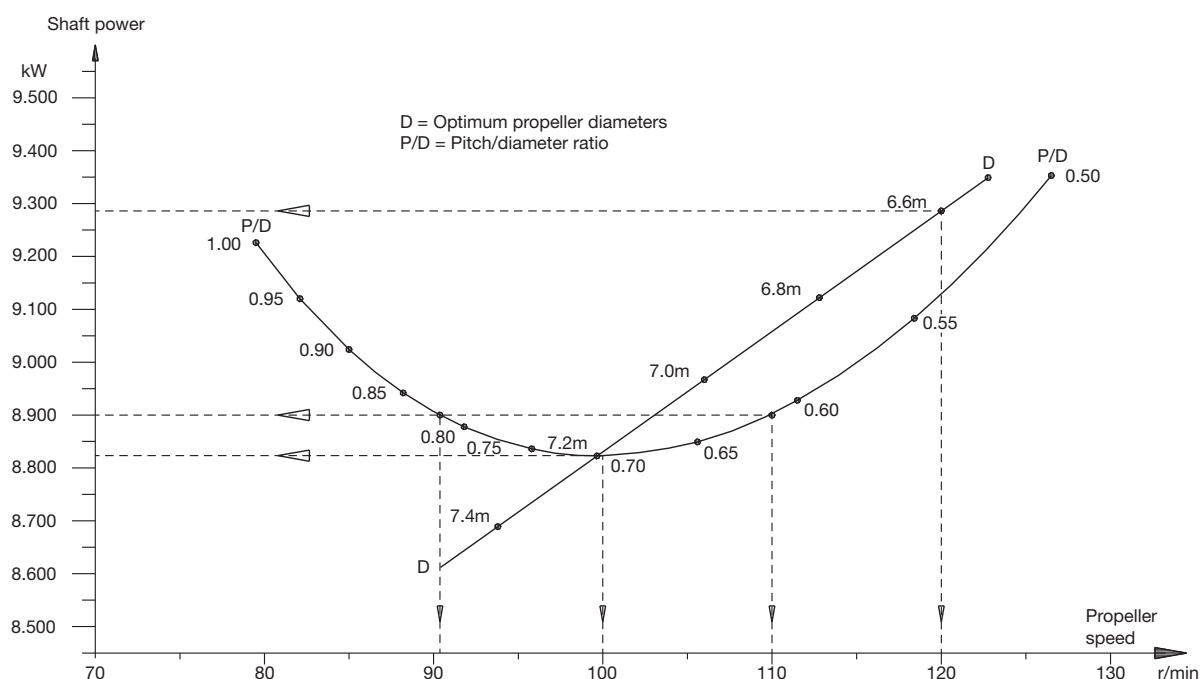
When the optimum propeller diameter D is increased from 6.6 m to 7.2 m, the power demand is reduced from about 9,290 kW to 8,820 kW, and the optimum propeller speed is reduced from 120 r/min to 100 r/min, corresponding to the constant ship speed coefficient $\alpha = 0.28$ (see definition of α in Section 2.02, page 2).

Once an optimum propeller diameter of maximum 7.2 m has been chosen, the corresponding optimum pitch in this point is given for the design speed of 14.5 knots, i.e. $P/D = 0.70$.

However, if the optimum propeller speed of 100 r/min does not suit the preferred / selected main engine speed, a change of pitch away from optimum will only cause a relatively small extra power demand, keeping the same maximum propeller diameter:

- going from 100 to 110 r/min ($P/D = 0.62$) requires 8,900 kW i.e. an extra power demand of 80 kW.
- going from 100 to 91 r/min ($P/D = 0.81$) requires 8,900 kW i.e. an extra power demand of 80 kW.

In both cases the extra power demand is only of 0.9%, and the corresponding 'equal speed curves' are $\alpha = +0.1$ and $\alpha = -0.1$, respectively, so there is a certain interval of propeller speeds in which the 'power penalty' is very limited.



178 47 03-2.0

Fig. 2.02.01: Influence of diameter and pitch on propeller design

Constant ship speed lines

The constant ship speed lines α , are shown at the very top of Fig. 2.02.02. These lines indicate the power required at various propeller speeds to keep the same ship speed provided that the optimum propeller diameter with an optimum pitch diameter ratio is used at any given speed, taking into consideration the total propulsion efficiency.

Normally, the following relation between necessary power and propeller speed can be assumed:

$$P_2 = P_1 \times (n_2/n_1)^\alpha$$

where:

P = Propulsion power

n = Propeller speed, and

α = the constant ship speed coefficient.

For any combination of power and speed, each point on lines parallel to the ship speed lines gives the same ship speed.

When such a constant ship speed line is drawn into the layout diagram through a specified propulsion MCR point 'MP₁', selected in the layout

area and parallel to one of the α -lines, another specified propulsion MCR point 'MP₂' upon this line can be chosen to give the ship the same speed for the new combination of engine power and speed.

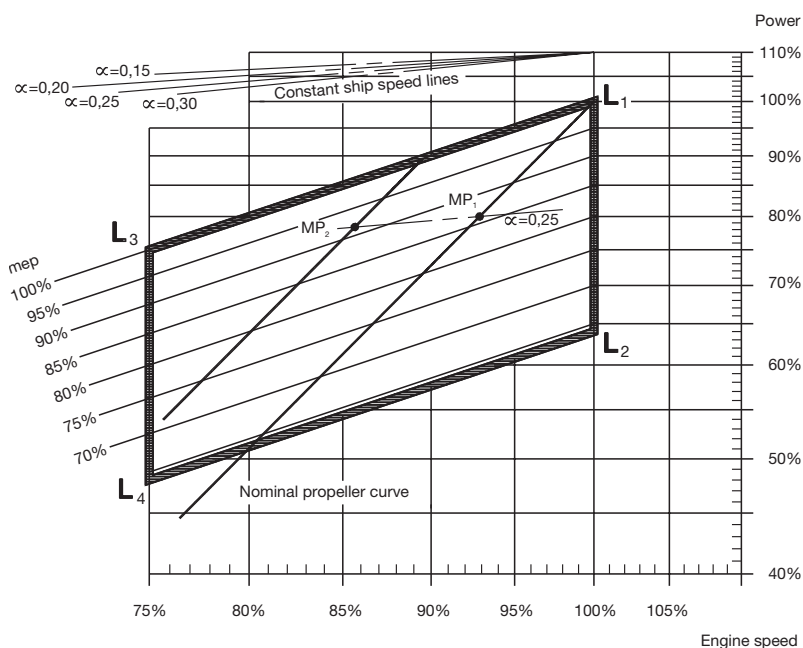
Fig. 2.02.02 shows an example of the required power speed point MP₁, through which a constant ship speed curve $\alpha = 0.25$ is drawn, obtaining point MP₂ with a lower engine power and a lower engine speed but achieving the same ship speed.

Provided the optimum pitch/diameter ratio is used for a given propeller diameter the following data applies when changing the propeller diameter:

for general cargo, bulk carriers and tankers
 $\alpha = 0.25 - 0.30$

and for reefers and container vessels
 $\alpha = 0.15 - 0.25$

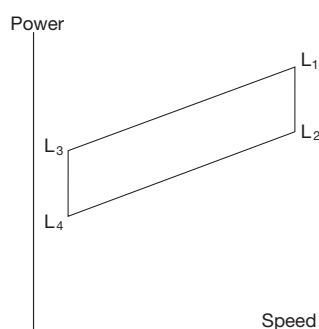
When changing the propeller speed by changing the pitch diameter ratio, the α constant will be different, see above.



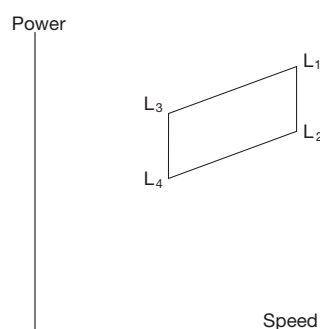
178 05 66-7.0

Fig. 2.02.02: Layout diagram and constant ship speed lines

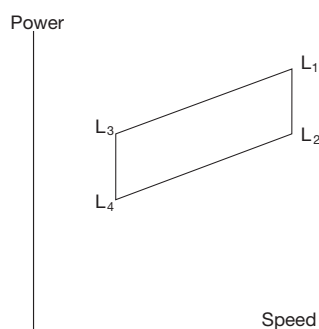
Layout Diagram Sizes



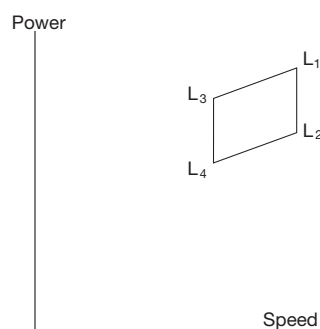
**100 - 80% power and
100 - 75% speed range**
valid for the types:
S80MC-C/ME-C7,
S80MC6,
S70MC-C/ME-C7,
S70MC6,
S60MC-C/ME-C7,
S60MC6,
S50MC-C/ME-C7,
S50MC6



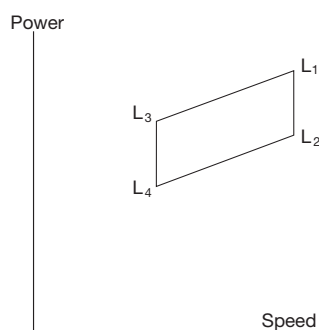
**100 - 80% power and
100 - 85% speed range**
valid for the types:
K90MC-C/6
K80MC-C/ME-C6,
L60MC-C/ME-C7/8,
S46MC-C8, S46ME-B8,
S42MC7, S40ME-B9,
S35MC7, S35ME-B9,
L35MC6, S26MC6,
S90MC-C/ME-C8,
S80MC-C8, S80ME-C8/9,
S70MC-C/ME-C/ME-GI8,
S65ME-C/ME-GI8,
S60MC-C/ME-C/ME-GI8,
S60ME-B8,
S50MC-C/ME-C8,
S50ME-B8/9



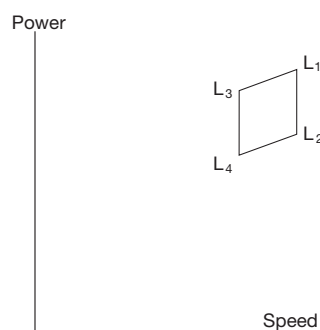
**100 - 80% power and
100 - 80% speed range**
valid for the types:
S90MC-C/ME-C7



**100 - 80% power and
100 - 90% speed range**
valid for the types:
K98MC/MC-C6,
K98ME/ME-C6,
K90ME/ME-C9,
K80ME-C9



**100 - 80% power and
100 - 84% speed range**
valid for the types:
L70MC-C/ME-C7/8,
S46MC-C7



**100 - 80% power and
100 - 93% speed range**
valid for the types:
K98MC/MC-C7,
K98ME/ME-C7

178 60 45-2.0

See also Section 2.05 for actual project.

Fig. 2.03.01 Layout diagram sizes

Engine Layout and Load Diagram

Engine Layout Diagram

An engine's layout diagram is limited by two constant mean effective pressure (mep) lines L_1-L_3 and L_2-L_4 , and by two constant engine speed lines L_1-L_2 and L_3-L_4 . The L_1 point refers to the engine's nominal maximum continuous rating, see Fig. 2.04.01.

Within the layout area there is full freedom to select the engine's specified SMCR point M which suits the demand for propeller power and speed for the ship.

On the horizontal axis the engine speed and on the vertical axis the engine power are shown on percentage scales. The scales are logarithmic which means that, in this diagram, power function curves like propeller curves (3rd power), constant mean effective pressure curves (1st power) and constant ship speed curves (0.15 to 0.30 power) are straight lines.

Specified maximum continuous rating (M)

Based on the propulsion and engine running points, as previously found, the layout diagram of a relevant main engine may be drawn-in. The SMCR point (M) must be inside the limitation lines of the layout diagram; if it is not, the propeller speed will have to be changed or another main engine type must be chosen.

Continuous service rating (S)

The continuous service rating is the power needed in service – including the specified sea margin and heavy/light running factor of the propeller – at which the engine is to operate, and point S is identical to the service propulsion point (SP) unless a main engine driven shaft generator is installed.

Matching point (O)

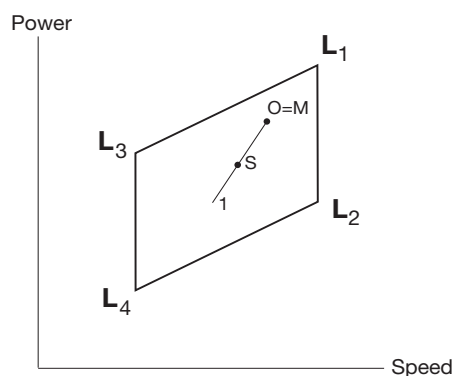
For practical reasons we have chosen to use the designation 'O' for the matching point.

The matching point O is placed on line 1 of the load diagram, see Fig. 2.04.01, and for technical reasons the power of O always has to be equal to the power of M. Point O normally coincides with point M.

For **ME, ME-C and ME-GI** engines, the timing of the fuel injection and the exhaust valve activation are electronically optimised over a wide operating range of the engine. Therefore the selection of matching point only has a meaning in connection with the turbocharger matching and the compression ratio.

For **ME-B** engines, only the fuel injection (and not the exhaust valve activation) is electronically controlled over a wide operating range of the engine, and the compression ratio is nearly constant as for an MC engine.

The lowest specific fuel oil consumption for the ME, ME-C and ME-GI engines is obtained at 70% and for ME-B engines at 80% of the matching point (O).



178 60 85-8.0

Fig. 2.04.01: Engine layout diagram

Engine Load Diagram

Definitions

The engine's load diagram, see Fig. 2.04.02, defines the power and speed limits for continuous as well as overload operation of an installed engine having a matching point O and a specified MCR point M that confirms the ship's specification.

Point A is a 100% speed and power reference point of the load diagram, and is defined as the point on the propeller curve (line 1), through the matching point O, having the specified MCR power. Normally, point M is equal to point A, but in special cases, for example if a shaft generator is installed, point M may be placed to the right of point A on line 7.

The service points of the installed engine incorporate the engine power required for ship propulsion and shaft generator, if installed.

Operating curves and limits for continuous operation

The continuous service range is limited by four lines: 4, 5, 7 and 3 (9), see Fig. 2.04.02. The propeller curves, line 1, 2 and 6 in the load diagram are also described below.

Line 1:

Propeller curve through specified MCR (M) engine layout curve.

Line 2:

Propeller curve, fouled hull and heavy weather – heavy running.

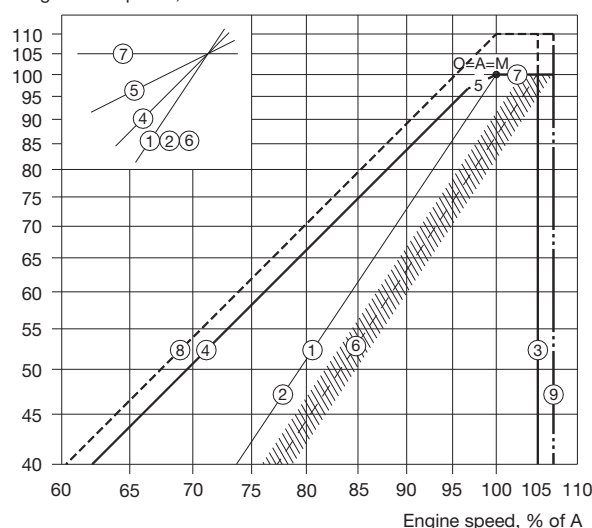
Line 3 and line 9:

Line 3 represents the maximum acceptable speed for continuous operation, i.e. 105% of A.

During trial conditions the maximum speed may be extended to 107% of A, see line 9.

The above limits may in general be extended to 105% and during trial conditions to 107% of the nominal L_n speed of the engine, provided the torsional vibration conditions permit.

Engine shaft power, % of A



Regarding 'i' in the power function $P = c \times n^i$, see page 2.01.

- A 100% reference point
- M Specified MCR point
- O Matching point

- Line 1 Propeller curve through matching point ($i = 3$) (engine layout curve)
- Line 2 Propeller curve, fouled hull and heavy weather – heavy running ($i = 3$)
- Line 3 Speed limit
- Line 4 Torque/speed limit ($i = 2$)
- Line 5 Mean effective pressure limit ($i = 1$)
- Line 6 Propeller curve, clean hull and calm weather – light running ($i = 3$), for propeller layout
- Line 7 Power limit for continuous running ($i = 0$)
- Line 8 Overload limit
- Line 9 Speed limit at sea trial

Point M to be located on line 7 (normally in point A)

178 05 42-7.5

Fig. 2.04.02: Standard engine load diagram

The overspeed set-point is 109% of the speed in A, however, it may be moved to 109% of the nominal speed in L_n , provided that torsional vibration conditions permit.

Running at low load above 100% of the nominal L_n speed of the engine is, however, to be avoided for extended periods. Only plants with controllable pitch propellers can reach this light running area.

Line 4:

Represents the limit at which an ample air supply is available for combustion and imposes a limitation on the maximum combination of torque and speed.

Line 5:

Represents the maximum mean effective pressure level (mep), which can be accepted for continuous operation.

Line 6:

Propeller curve, clean hull and calm weather – light running, used for propeller layout/design.

Line 7:

Represents the maximum power for continuous operation.

Limits for overload operation

The overload service range is limited as follows:

Line 8:

Represents the overload operation limitations.

The area between lines 4, 5, 7 and the heavy dashed line 8 is available for overload running for limited periods only (1 hour per 12 hours).

Line 9:

Speed limit at sea trial.

Limits for low load running

As the fuel injection is automatically controlled over the entire power range, the engine is able to operate down to around 15% of the nominal L_q speed.

Recommendation

Continuous operation without limitations is allowed only within the area limited by lines 4, 5, 7 and 3 of the load diagram, except on low load operation for CP propeller plants mentioned in the previous section.

The area between lines 4 and 1 is available for operation in shallow waters, heavy weather and during acceleration, i.e. for non-steady operation without any strict time limitation.

After some time in operation, the ship's hull and propeller will be fouled, resulting in heavier running of the propeller, i.e. the propeller curve will move to the left from line 6 towards line 2, and extra power is required for propulsion in order to keep the ship's speed.

In calm weather conditions, the extent of heavy running of the propeller will indicate the need for cleaning the hull and possibly polishing the propeller.

Once the specified MCR (and the matching point) have been chosen, the capacities of the auxiliary equipment will be adapted to the specified MCR, and the turbocharger specification and the compression ratio will be selected.

If the specified MCR (and the matching point) is to be increased later on, this may involve a change of the pump and cooler capacities, change of the fuel valve nozzles, adjusting of the cylinder liner cooling, as well as rematching of the turbocharger or even a change to a larger size of turbocharger. In some cases it can also require larger dimensions of the piping systems.

It is therefore of utmost importance to consider, already at the project stage, if the specification should be prepared for a later power increase. This is to be indicated in the Extent of Delivery.

Extended load diagram for ships operating in extreme heavy running conditions

When a ship with fixed pitch propeller is operating in normal sea service, it will in general be operating in the hatched area around the design propeller curve 6, as shown on the standard load diagram in Fig. 2.04.02.

Sometimes, when operating in heavy weather, the fixed pitch propeller performance will be more heavy running, i.e. for equal power absorption of the propeller, the propeller speed will be lower and the propeller curve will move to the left.

As the low speed main engines are directly coupled to the propeller, the engine has to follow the propeller performance, i.e. also in heavy running propeller situations. For this type of operation, there is normally enough margin in the load area between line 6 and the normal torque/speed limitation line 4, see Fig. 2.04.02. To the left of line 4 in torque-rich operation, the engine will lack air from the turbocharger to the combustion process, i.e. the heat load limits may be exceeded and bearing loads might also become too high.

For some special ships and operating conditions, it would be an advantage - when occasionally needed - to be able to operate the propeller/main engine as much as possible to the left of line 6, but inside the torque/speed limit, line 4.

Such cases could be for:

- ships sailing in areas with very heavy weather
- ships operating in ice
- ships with two fixed pitch propellers/two main engines, where one propeller/one engine is de-clutched for one or the other reason.

The increase of the operating speed range between line 6 and line 4 of the standard load diagram, see Fig. 2.04.02, may be carried out as shown for the following engine Example with an extended load diagram for speed derated engine with increased light running.

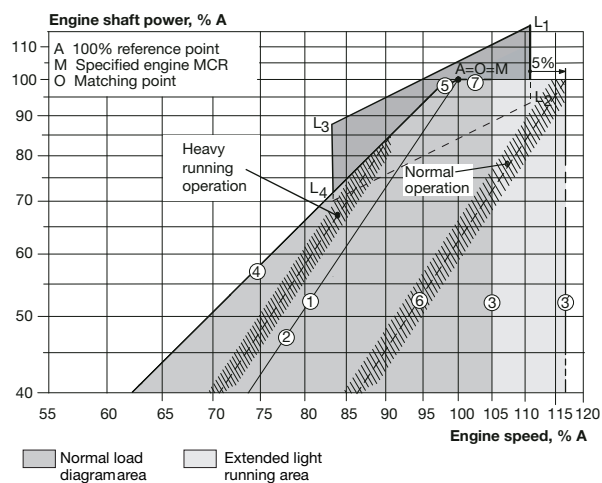
Extended load diagram for speed derated engines with increased light running

The maximum speed limit (line 3) of the engines is 105% of the SMCR (Specified Maximum Continuous Rating) speed, as shown in Fig. 2.04.02.

However, for speed and, thereby, power derated engines it is possible to extend the maximum speed limit to 105% of the engine's nominal MCR speed, line 3', but only provided that the torsional vibration conditions permit this. Thus, the shafting, with regard to torsional vibrations, has to be approved by the classification society in question, based on the extended maximum speed limit.

When choosing an increased light running to be used for the design of the propeller, the load diagram area may be extended from line 3 to line 3', as shown in Fig. 2.04.03, and the propeller/main engine operating curve 6 may have a correspondingly increased heavy running margin before exceeding the torque/speed limit, line 4.

A corresponding slight reduction of the propeller efficiency may be the result, due to the higher propeller design speed used.



- Line 1: Propeller curve through matching point (O)
 - layout curve for engine
- Line 2: Heavy propeller curve
 - fouled hull and heavy seas
- Line 3: Speed limit
- Line 3': **Extended speed limit**, provided torsional vibration conditions permit
- Line 4: Torque/speed limit
- Line 5: Mean effective pressure limit
- Line 6: Increased light running propeller curve
 - clean hull and calm weather
 - layout curve for propeller
- Line 7: Power limit for continuous running

178 60 79-9.0

Fig. 2.04.03: Extended load diagram for speed derated engine with increased light running

Examples of the use of the Load Diagram

In the following are some examples illustrating the flexibility of the layout and load diagrams.

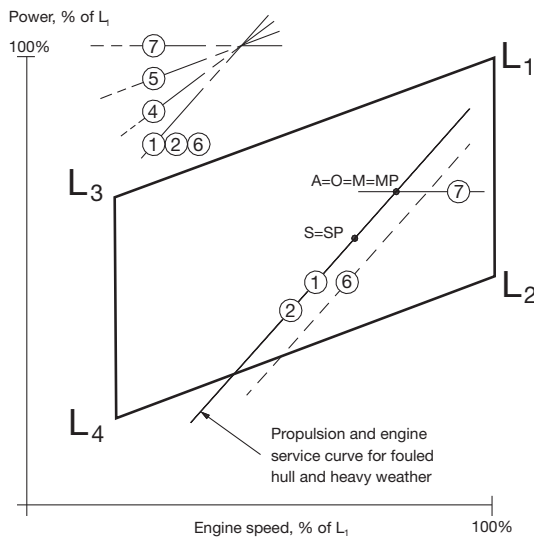
- Example 1 shows how to place the load diagram for an engine without shaft generator coupled to a fixed pitch propeller.
- Example 2 are diagrams for the same configuration, but choosing a matching point on the left of the heavy running propeller curve (2) providing an extra engine margin for heavy running similar to the case in Fig. 2.04.03.
- Example 3 shows the same layout for an engine with fixed pitch propeller (example 1), but with a shaft generator.
- Example 4 is a special case of example 3, where the specified MCR is placed near the top of the layout diagram. In this case the shaft generator is cut off, and the GenSets used when the engine runs at specified MCR. This makes it possible to choose a smaller engine with a lower power output.
- Example 5 shows diagrams for an engine coupled to a controllable pitch propeller, with or without a shaft generator, constant speed or combinator curve operation.

For a specific project, the layout diagram for actual project shown later in this chapter may be used for construction of the actual load diagram.

Example 1: Normal running conditions.

Engine coupled to fixed pitch propeller (FPP) and without shaft generator

Layout diagram

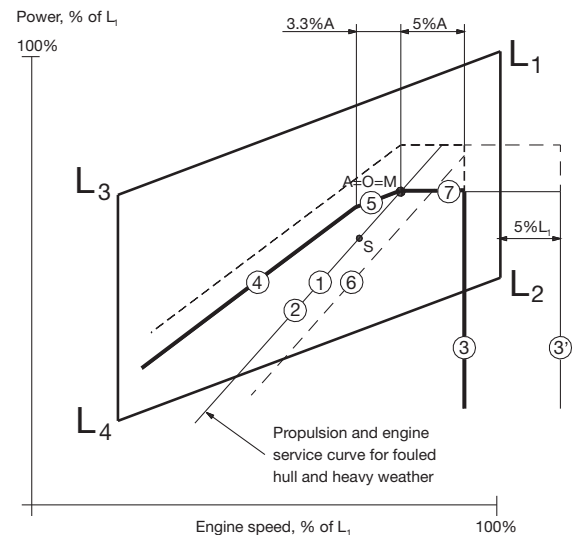


M	Specified MCR of engine
S	Continuous service rating of engine
O	Matching point of engine
A	Reference point of load diagram
MP	Specified MCR for propulsion
SP	Continuous service rating of propulsion

The specified MCR (M) and the matching point O and its propeller curve 1 will normally be selected on the engine service curve 2.

Point A is then found at the intersection between propeller curve 1 (2) and the constant power curve through M, line 7. In this case point A is equal to point M and point O.

Load diagram



Point A of load diagram is found:

Line 1 Propeller curve through matching point (O) is equal to line 2

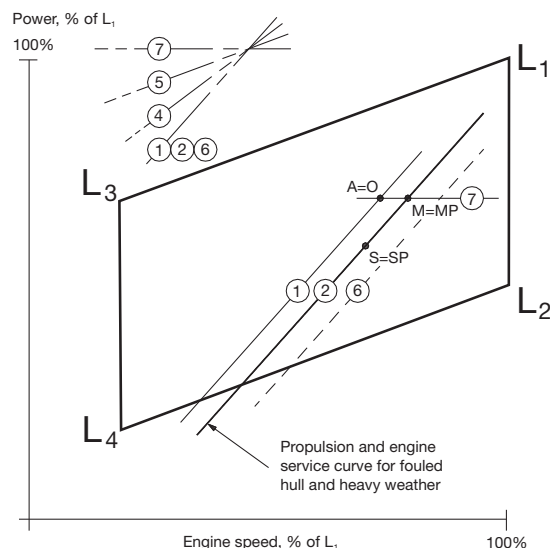
Line 7 Constant power line through specified MCR (M)

Point A Intersection between line 1 and 7

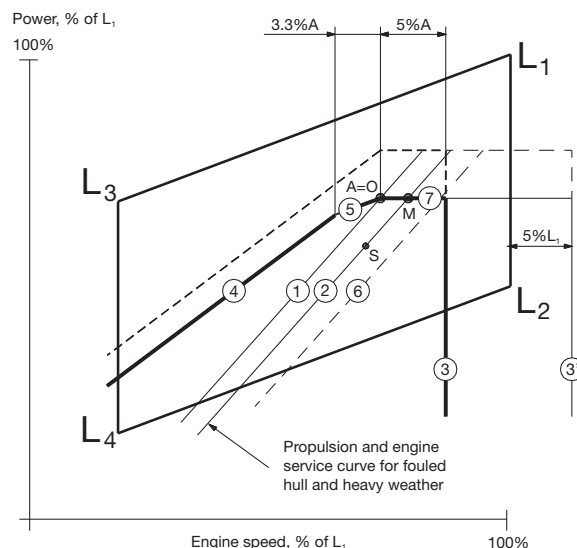
Once point A has been found in the layout diagram, the load diagram can be drawn, as shown in the figure, and hence the actual load limitation lines of the diesel engine may be found by using the inclinations from the construction lines and the %-figures stated.

178 05 44-0.8

Fig. 2.04.04: Normal running conditions. Engine coupled to a fixed pitch propeller (FPP) and without a shaft generator

Example 2: Special running conditions.**Engine coupled to fixed pitch propeller (FPP) and without shaft generator****Layout diagram**

M	Specified MCR of engine
S	Continuous service rating of engine
O	Matching point of engine
A	Reference point of load diagram
MP	Specified MCR for propulsion
SP	Continuous service rating of propulsion

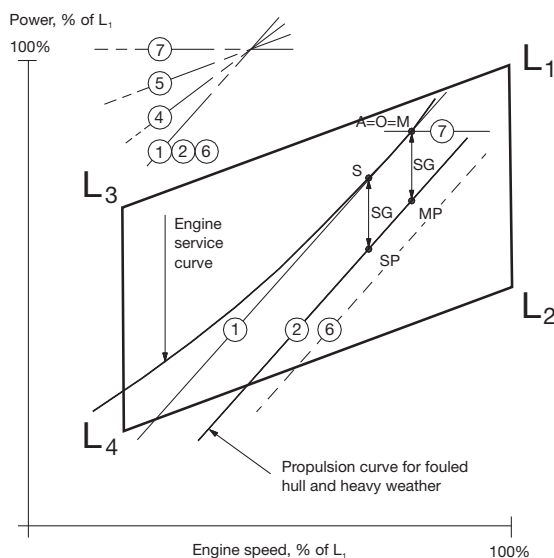
Load diagram**Point A of load diagram is found:**

- Line 1 Propeller curve through matching point (O) placed to the left of line 2
- Line 7 Constant power line through specified MCR (M)
- Point A Intersection between line 1 and 7

In this example, the matching point O has been selected more to the left than in example 1, providing an extra engine margin for heavy running operation in heavy weather conditions. In principle, the light running margin has been increased for this case.

178 05 46-4.8

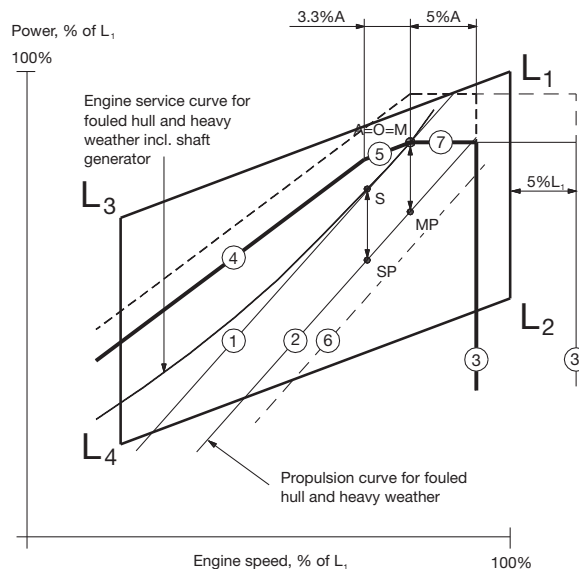
Fig. 2.04.05: Special running conditions. Engine coupled to a fixed pitch propeller (FPP) and without a shaft generator

Example 3: Normal running conditions.**Engine coupled to fixed pitch propeller (FPP) and with shaft generator****Layout diagram**

M	Specified MCR of engine
S	Continuous service rating of engine
O	Matching point of engine
A	Reference point of load diagram
MP	Specified MCR for propulsion
SP	Continuous service rating of propulsion
SG	Shaft generator power

In example 3 a shaft generator (SG) is installed, and therefore the service power of the engine also has to incorporate the extra shaft power required for the shaft generator's electrical power production.

In the figure, the engine service curve shown for heavy running incorporates this extra power.

Load diagram**Point A of load diagram is found:**

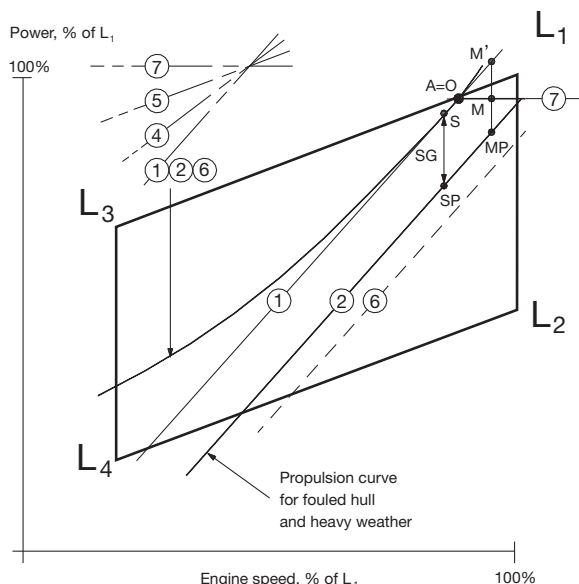
- Line 1 Propeller curve through matching point (O)
- Line 7 Constant power line through specified MCR (M)
- Point A Intersection between line 1 and 7

The matching point $O = A = M$ will be chosen on this curve, as shown.

Point A is then found in the same way as in example 1 and the load diagram can be drawn as shown in the figure.

178 05 48-8.8

Fig. 2.04.06: Normal running conditions. Engine coupled to a fixed pitch propeller (FPP) and with a shaft generator

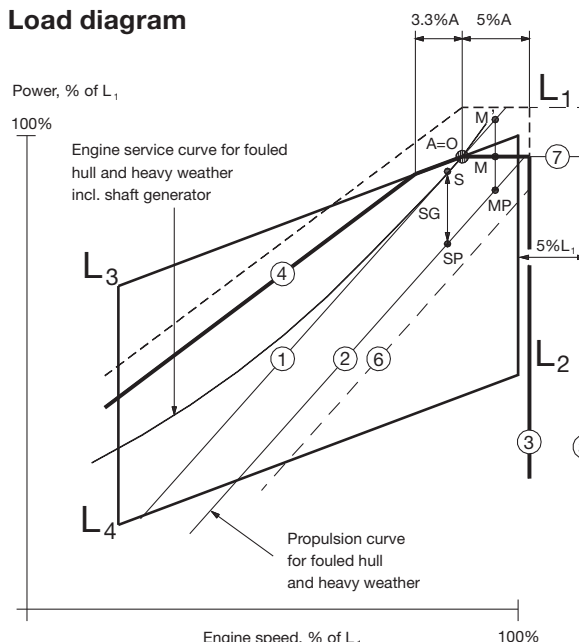
Example 4: Special running conditions.**Engine coupled to fixed pitch propeller (FPP) and with shaft generator****Layout diagram**

M	Specified MCR of engine
S	Continuous service rating of engine
O	Matching point of engine
A	Reference point of load diagram
MP	Specified MCR for propulsion
SP	Continuous service rating of propulsion
SG	Shaft generator

Also for this special case in example 4, a shaft generator is installed but, compared to example 3, this case has a specified MCR for propulsion, MP, placed at the top of the layout diagram.

This involves that the intended specified MCR of the engine M' will be placed outside the top of the layout diagram.

One solution could be to choose a larger diesel engine with an extra cylinder, but another and cheaper solution is to reduce the electrical power production of the shaft generator when running in the upper propulsion power range.

Load diagram**Point A and M of the load diagram are found:**

- Line 1 Propeller curve through point S
- Point A Intersection between line 1 and line $L_1 - L_3$
- Point M Located on constant power line 7 through point A and with MP's speed
- Point O Equal to point A

In choosing the latter solution, the required specified MCR power can be reduced from point M' to point M as shown. Therefore, when running in the upper propulsion power range, a diesel generator has to take over all or part of the electrical power production.

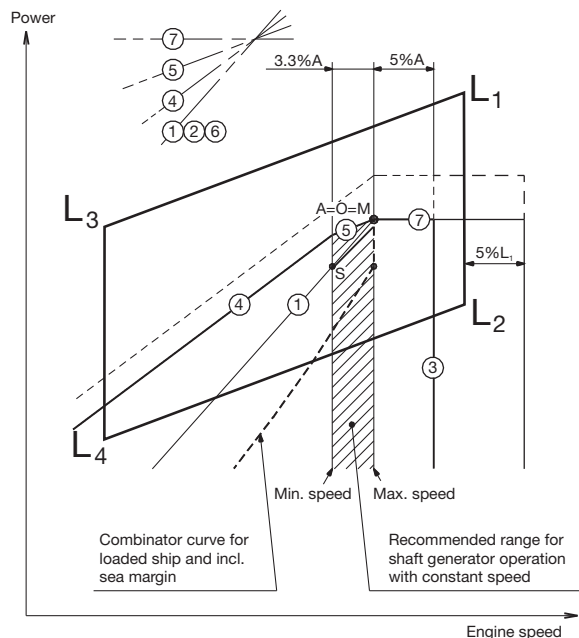
However, such a situation will seldom occur, as ships are rather infrequently running in the upper propulsion power range.

Point A, having the highest possible power, is then found at the intersection of line $L_1 - L_3$ with line 1 and the corresponding load diagram is drawn. Point M is found on line 7 at MP's speed, and point $O=A$.

178 06 35-1.8

Fig. 2.04.07: Special running conditions. Engine coupled to a fixed pitch propeller (FPP) and with a shaft generator

Example 5: Engine coupled to controllable pitch propeller (CPP) with or without shaft generator



M	Specified MCR of engine
O	Matching point of engine
A	Reference point of load diagram
S	Continuous service rating of engine

178 39 31-4.4

Fig. 2.04.08: Engine with Controllable Pitch Propeller (CPP), with or without a shaft generator

Layout diagram - without shaft generator

If a controllable pitch propeller (CPP) is applied, the combinator curve (of the propeller) will normally be selected for loaded ship including sea margin.

The combinator curve may for a given propeller speed have a given propeller pitch, and this may be heavy running in heavy weather like for a fixed pitch propeller.

Therefore it is recommended to use a light running combinator curve (the dotted curve which includes the sea power margin) as shown in the figure to obtain an increased operation margin of the diesel engine in heavy weather to the limit indicated by curves 4 and 5.

Layout diagram - with shaft generator

The hatched area shows the recommended speed range between 100% and 96.7% of the specified MCR speed for an engine with shaft generator running at constant speed.

The service point S can be located at any point within the hatched area.

The procedure shown in examples 3 and 4 for engines with FPP can also be applied here for engines with CPP running with a combinator curve.

The matching point O

O may, as earlier described, be chosen equal to point M, see below.

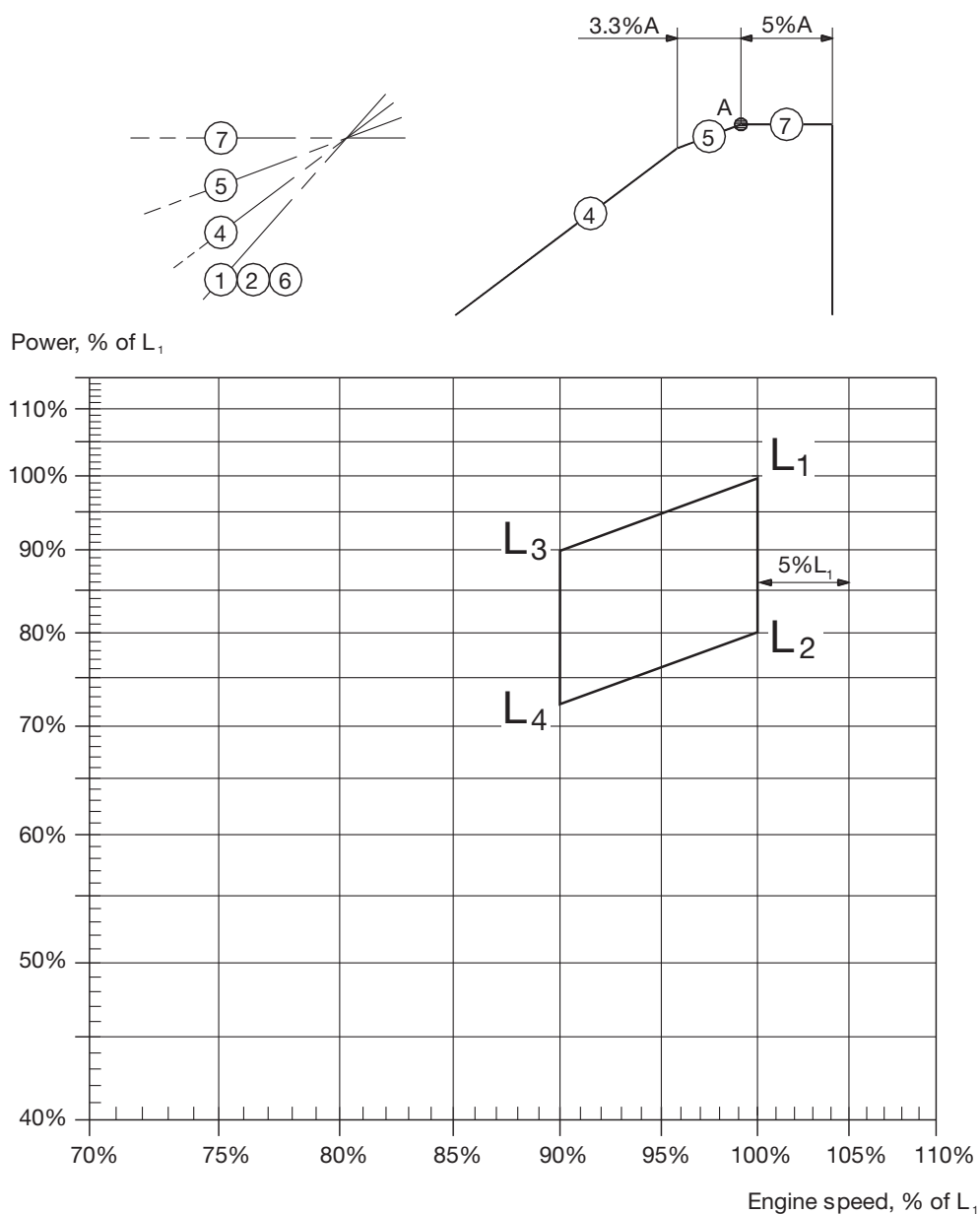
Load diagram

Therefore, when the engine's specified MCR point (M) has been chosen including engine margin, sea margin and the power for a shaft generator, if installed, point M may be used as point A of the load diagram, which can then be drawn.

The position of the combinator curve ensures the maximum load range within the permitted speed range for engine operation, and it still leaves a reasonable margin to the limit indicated by curves 4 and 5.

Diagram for actual project

This figure contains a layout diagram that can be used for constructing the load diagram for an actual project, using the %-figures stated and the inclinations of the lines.



178 56 27-1.1

Fig. 2.05.01: Construction of layout diagram

Specific Fuel Oil Consumption, ME versus MC engines

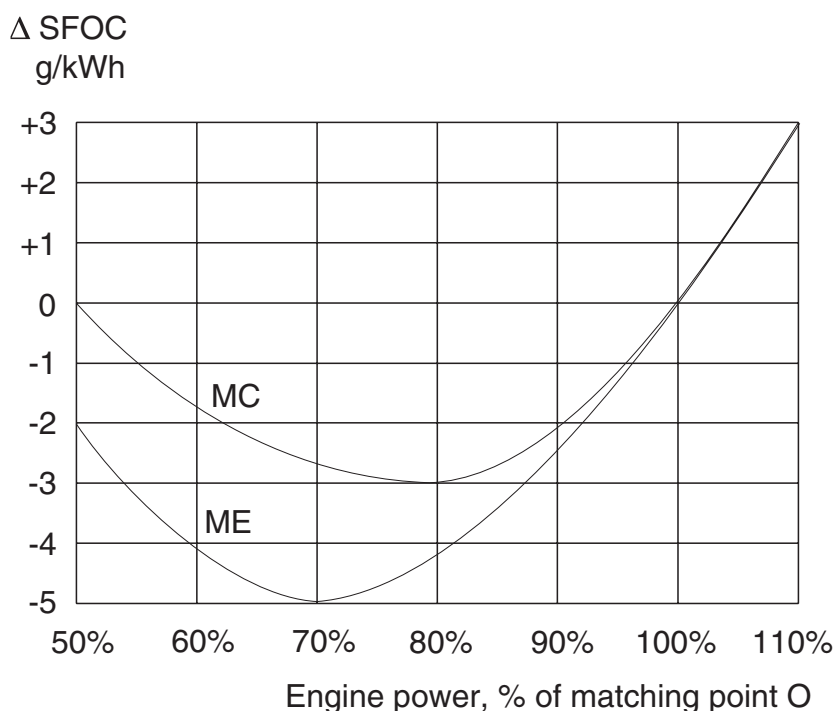
As previously mentioned the main feature of the ME engine is that the fuel injection and the exhaust valve timing are optimised automatically over the entire power range, and with a minimum speed down to around 15% of the L_r speed.

Comparing the specific fuel oil consumption (SFOC) of the ME and the MC engines, it can be seen from the figure below that the great advantage of the ME engine is a lower SFOC at part loads.

It is also noted that the lowest SFOC for the ME engine is at 70% of O, whereas it was at 80% of O for the MC engine.

For the ME engine only the turbocharger matching and the compression ratio (shims under the piston rod) remain as variables to be determined by the engine maker / MAN Diesel.

The calculation of the expected specific fuel oil consumption (SFOC) can be carried out by means of the following figures for fixed pitch propeller and for controllable pitch propeller, constant speed. Throughout the whole load area the SFOC of the engine depends on where the matching point (O) is chosen.



198 97 38-9.2

Fig. 2.06.01: Example of part load SFOC curves for ME and MC with fixed pitch propeller

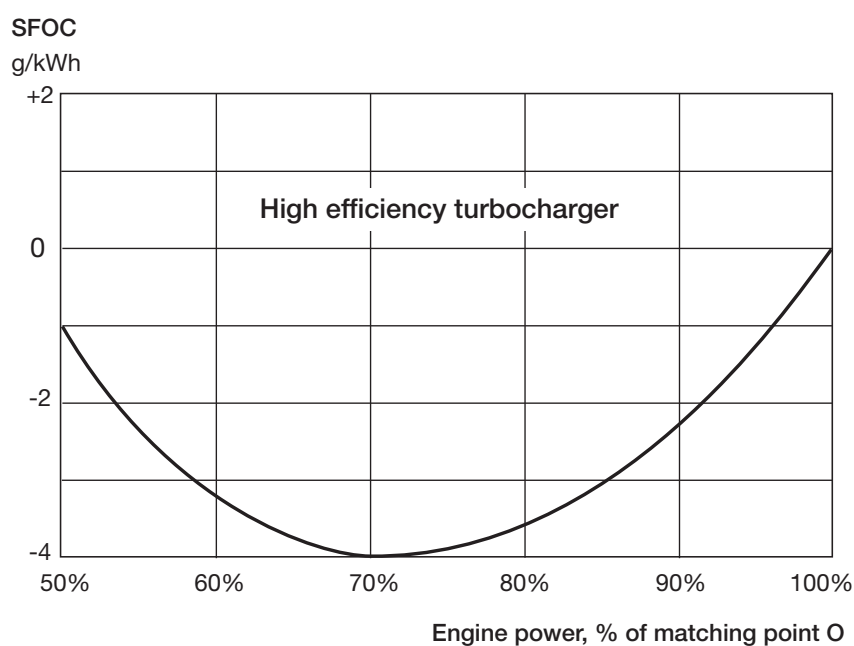
SFOC for High Efficiency Turbochargers

All engine types 50 and larger are as standard fitted with high efficiency turbochargers, option: 4 59 104.

The high efficiency turbocharger is applied to the engine in the basic design with the view to obtaining the lowest possible Specific Fuel Oil

Consumption (SFOC) values, see example in Fig. 2.07.01.

At part load running the lowest SFOC may be obtained at 70% of the matched power = 70% of the specified MCR.



178 60 95-4.0

Fig. 2.07.01: Example of part load SFOC curves for high efficiency turbochargers

SFOC reference conditions and guarantee

SFOC at reference conditions

The SFOC is given in **g/kWh** based on the reference ambient conditions stated in ISO 3046-1:2002(E) and ISO 15550:2002(E):

1,000 mbar ambient air pressure
25 °C ambient air temperature
25 °C scavenge air coolant temperature

and is related to a fuel oil with a lower calorific value of 42,700 kJ/kg (~10,200 kcal/kg).

Any discrepancies between g/kWh and g/BHP_h are due to the rounding of numbers for the latter.

For lower calorific values and for ambient conditions that are different from the ISO reference conditions, the SFOC will be adjusted according to the conversion factors in the table below.

Parameter	Condition change	With p_{\max} adjusted	Without p_{\max} adjusted
		SFOC change	SFOC change
Scav. air coolant temperature	per 10 °C rise	+ 0.60%	+ 0.41%
Blower inlet temperature	per 10 °C rise	+ 0.20%	+ 0.71%
Blower inlet pressure	per 10 mbar rise	- 0.02%	- 0.05%
Fuel oil lower calorific value	rise 1% (42,700 kJ/kg)	-1.00%	- 1.00%

With for instance 1 °C increase of the scavenge air coolant temperature, a corresponding 1 °C increase of the scavenge air temperature will occur and involves an SFOC increase of 0.06% if p_{\max} is adjusted to the same value.

SFOC guarantee

The SFOC guarantee refers to the above ISO reference conditions and lower calorific value and is valid for one running point only. The guaranteed running point is equal to the power-speed combination in the matching point (O) = 100% SMCR but, if requested, a running point between 85% and 100% SMCR can be selected.

The SFOC guarantee is given with a tolerance of 5%.

Recommended cooling water temperature during normal operation

In general, it is recommended to operate the main engine with the lowest possible cooling water temperature to the air coolers, as this will reduce the fuel consumption of the engine, i.e. the engine performance will be improved.

However, shipyards often specify a constant (maximum) central cooling water temperature of 36 °C, not only for tropical ambient temperature conditions, but also for lower ambient temperature conditions. The purpose is probably to reduce the electric power consumption of the cooling water pumps and/or to reduce water condensation in the air coolers.

Thus, when operating with 36 °C cooling water instead of for example 10 °C (to the air coolers), the specific fuel oil consumption will increase by approx. 2 g/kWh.

Examples of Graphic Calculation of SFOC

The following diagrams a, b and c, valid for fixed pitch propeller (b) and constant speed (c), respectively, show the reduction of SFOC in g/kWh, relative to the SFOC for the nominal MCR L_1 rating.

The solid lines are valid at 100%, 70% and 50% of matching point (O).

Point O is drawn into the above-mentioned Diagrams b or c. A straight line along the constant mep curves (parallel to L_1-L_3) is drawn through point O. The intersections of this line and the curves indicate the reduction in specific fuel oil consumption at 100, 70 and 50% of the matching point, related to the SFOC stated for the nominal MCR L_1 rating.

An example of the calculated SFOC curves are shown in Diagram a, and is valid for an engine with fixed pitch propeller, see Fig. 2.10.01.

SFOC Calculations for K90ME9

Data at nominal MCR (L _r)			SFOC at nominal MCR (L _r)
			High efficiency TC
Engine	kW	r/min	g/kWh
6 K90ME9	34,320	94	174
7 K90ME9	40,040		
8 K90ME9	45,760		
9 K90ME9	51,480		
10 K90ME9	57,200		
11 K90ME9	62,920		
12 K90ME9	68,640		

Data matching point (O=M):	
	cyl. No.
Power: 100% of (O)	kW
Speed: 100% of (O)	r/min
SFOC found:	g/kWh

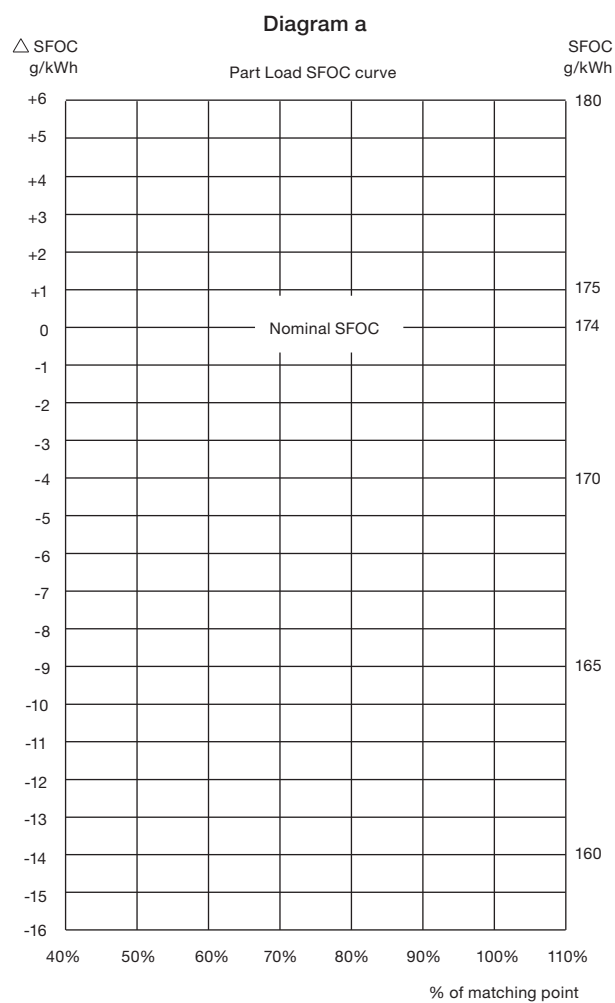
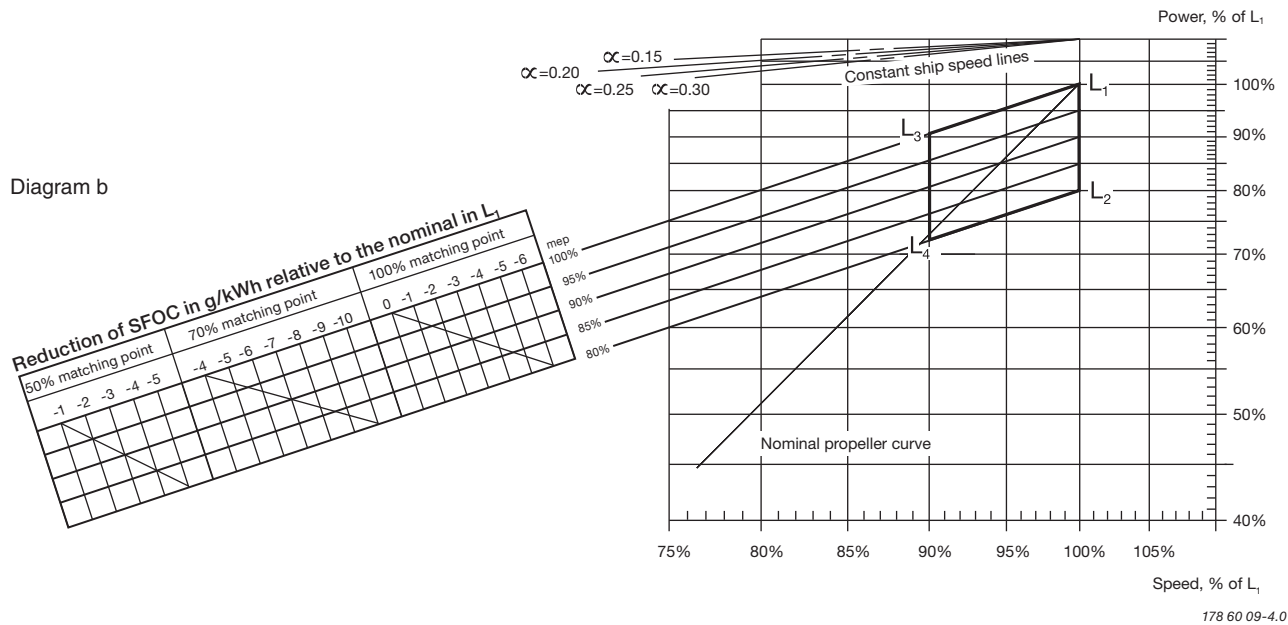
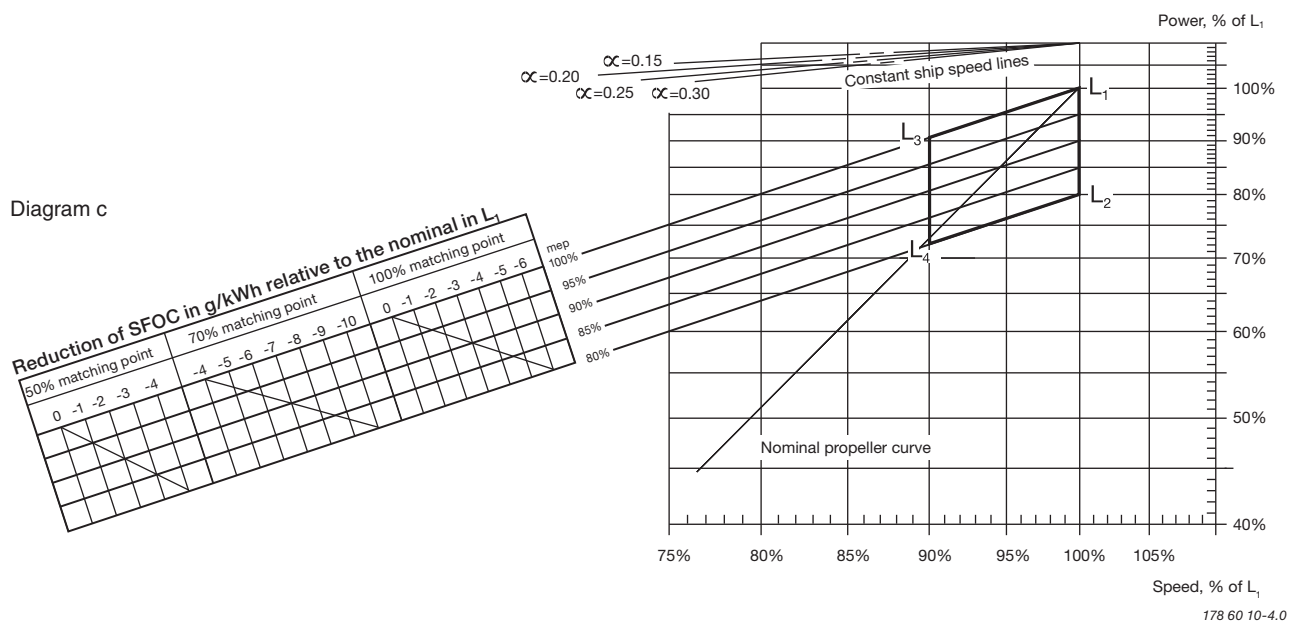


Fig. 2.09.01



178 60 09-4.0

Diagram c



178 60 10-4.0

SFOC calculations, example

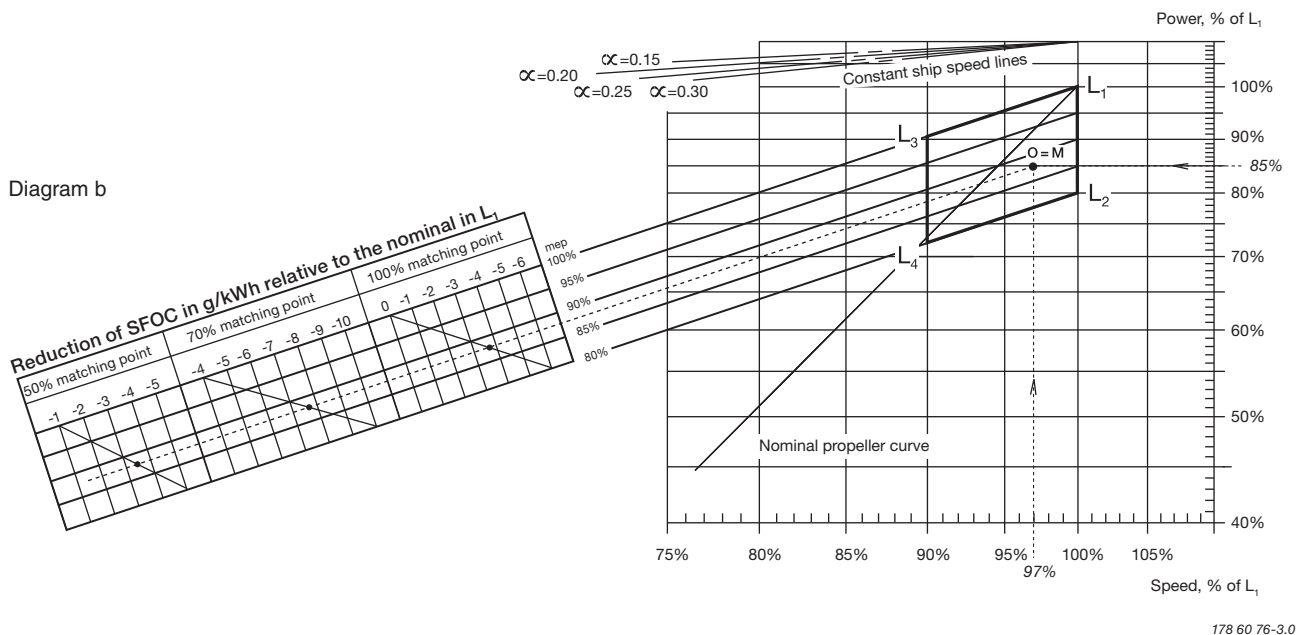
Data at nominal MCR (L ₁): 6K90ME9	
Power 100%	34,320 kW
Speed 100%	94 r/min
Nominal SFOC:	
• High efficiency turbocharger	174 g/kWh

Example of specified MCR = M	
Power	29,172 kW (85% L ₁)
Speed	91.2 r/min (97% L ₁)
Turbocharger type	High efficiency
SFOC found in O=M	170.3 g/kWh

The matching point O used in the above example for the SFOC calculations:

O = 100% M = 85% L₁ power and 97% L₁ speed

Diagram b



The reductions, see diagram b, in g/kWh compared to SFOC in L_1 :

Power in	Part load points		SFOC g/kWh	SFOC g/kWh
100% O	1	100% M	-3.7	170.3
70% O	2	70% M	-7.7	166.3
50% O	3	50% M	-3.5	170.5

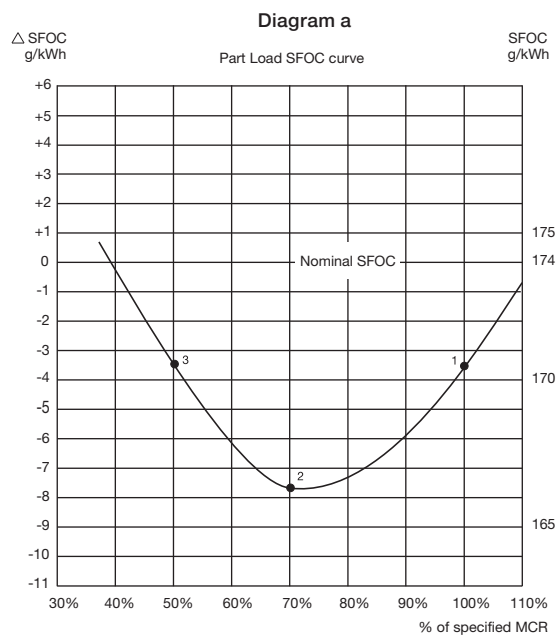
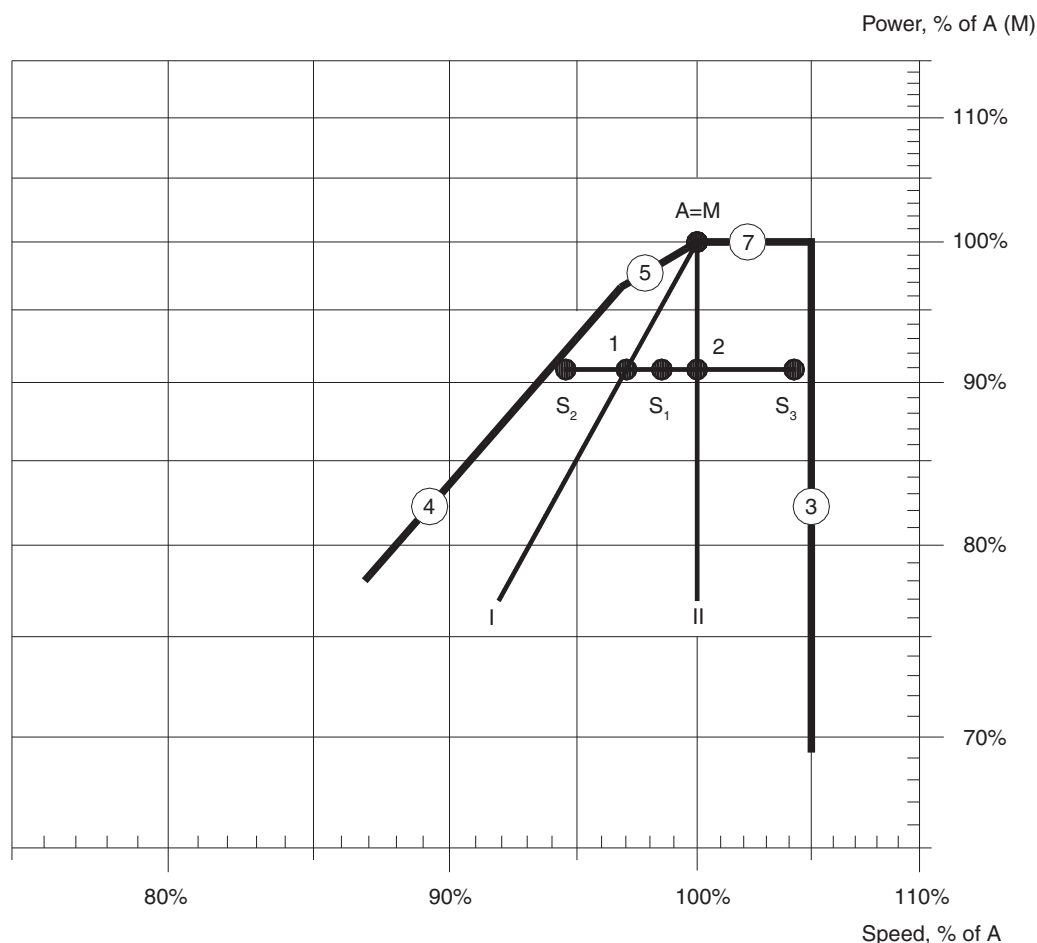


Fig. 2.10.01: Example of SFOC for derated 6K90ME9 with fixed pitch propeller and high efficiency turbocharger

The SFOC curve through points S_2 , on the left of point 1, is symmetrical about point 1, i.e. at speeds lower than that of point 1, the SFOC will also increase.

The above-mentioned method provides only an approximate value. A more precise indication of the expected SFOC at any load can be calculated by using our computer program. This is a service which is available to our customers on request.

Next the SFOC for point S_1 can be calculated as an interpolation between the SFOC in points '1' and '2', and for point S_3 as an extrapolation.



198 95 96-2 2

Fig. 2.11.01: SFOC at an arbitrary load

Emission Control

IMO NO_x emission limits

All ME, ME-B, ME-C and ME-GI engines are, as standard, delivered in compliance with the IMO speed dependent NO_x limit, measured according to ISO 8178 Test Cycles E2/E3 for Heavy Duty Diesel Engines. These are referred to in the Extent of Delivery as EoD: 4 06 060 Economy mode with the options: 4 06 060a Engine test cycle E3 or 4 06 060b Engine test cycle E2.

NO_x reduction methods

The NO_x content in the exhaust gas can be reduced with primary and/or secondary reduction methods.

The primary methods affect the combustion process directly by reducing the maximum combustion temperature, whereas the secondary methods are means of reducing the emission level without changing the engine performance, using external equipment.

0-30% NO_x reduction

The ME engines can be delivered with several operation modes, options: 4 06 063 Port load, 4 06 064 Special emission, 4 06 065 Other emission limit, and 4 06 066 Dual fuel.

These operation modes may include a 'Low NO_x mode' for operation in, for instance, areas with restriction in NO_x emission.

For further information on engine operation modes, see Extent of Delivery.

30-50% NO_x reduction

Water emulsification of the heavy fuel oil is a well proven primary method. The type of homogenizer is either ultrasonic or mechanical, using water from the freshwater generator and the water mist catcher.

The pressure of the homogenised fuel has to be increased to prevent the formation of steam and cavitation. It may be necessary to modify some of the engine components such as the fuel oil pressure booster, fuel injection valves and the engine control system.

Up to 95-98% NO_x reduction

This reduction can be achieved by means of secondary methods, such as the SCR (Selective Catalytic Reduction), which involves an after-treatment of the exhaust gas, see Section 3.02.

Plants designed according to this method have been in service since 1990 on five vessels, using Haldor Topsøe catalysts and ammonia as the reducing agent, urea can also be used.

The SCR unit can be located separately in the engine room or horizontally on top of the engine. The compact SCR reactor is mounted before the turbocharger(s) in order to have the optimum working temperature for the catalyst. However attention have to be given to the type of HFO to be used.

For further information about emission control, please refer to our publication:

Exhaust Gas Emission Control Today and Tomorrow

The publication is available at: www.mandiesel.com under 'Quicklinks' → 'Technical Papers'.

Turbocharger Selection & Exhaust Gas By-pass

3

Turbocharger Selection

Updated turbocharger data based on the latest information from the turbocharger makers are available from the Turbocharger Selection program on www.mandiesel.com under 'Turbocharger' → 'Overview' → 'Turbocharger Selection'.

The data specified in the printed edition are valid at the time of publishing.

The MC/ME engines are designed for the application of either MAN Diesel, ABB or Mitsubishi (MHI) turbochargers.

The turbocharger choice is made with a view to obtaining the lowest possible Specific Fuel Oil Consumption (SFOC) values at the nominal MCR by applying high efficiency turbochargers.

The engines are, as standard, equipped with as few turbochargers as possible, see the table in Fig. 3.01.01.

One more turbocharger can be applied, than the number stated in the tables, if this is desirable due to space requirements, or for other reasons. Additional costs are to be expected.

However, we recommend the 'Turbocharger selection' programme on the Internet, which can be used to identify a list of applicable turbochargers for a specific engine layout.

For information about turbocharger arrangement and cleaning systems, see Section 15.01.

High efficiency turbochargers for the K90ME9-TII engines - L_1 output			
Cyl.	MAN (TCA)	ABB (A100)	MHI (MET)
6	2 x TCA88-21	2 x A185-L35	2 x MET83MA
7	2 x TCA88-21	2 x A190-L35	2 x MET83MA
8	2 x TCA88-21	3 x A185-L34	2 x MET90MA
9	2 x TCA88-25	3 x A185-L35	3 x MET83MA
10	3 x TCA88-21	3 x A190-L34	3 x MET83MA
11	3 x TCA88-21	3 x A190-L35	3 x MET90MA
12	3 x TCA88-21	4 x A185-L35	3 x MET90MA

Fig. 3.01.01: High efficiency turbochargers

Exhaust Gas By-pass

Extreme Ambient Conditions

As mentioned in Chapter 1, the engine power figures are valid for tropical conditions at sea level: 45 °C air at 1000 mbar and 32 °C sea water, whereas the reference fuel consumption is given at ISO conditions: 25 °C air at 1000 mbar and 25 °C charge air coolant temperature.

Marine diesel engines are, however, exposed to greatly varying climatic temperatures winter and summer in arctic as well as tropical areas. These variations cause changes of the scavenge air pressure, the maximum combustion pressure, the exhaust gas amount and temperatures as well as the specific fuel oil consumption.

For further information about the possible countermeasures, please refer to our publication titled:

Influence of Ambient Temperature Conditions

The publication is available at: www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

Arctic running condition

For air inlet temperatures below -10 °C the precautions to be taken depend very much on the operating profile of the vessel. The following alternative is one of the possible countermeasures. The selection of countermeasures, however, must be evaluated in each individual case.

Exhaust gas receiver with variable by-pass

option: 4 60 118

Compensation for low ambient temperature can be obtained by using exhaust gas by-pass system.

This arrangement ensures that only part of the exhaust gas goes via the turbine of the turbocharger, thus supplying less energy to the compressor which, in turn, reduces the air supply to the engine.

Please note that if an exhaust gas by-pass is applied the turbocharger size and specification has to be determined by other means than stated in this Chapter.

NO_x Reduction by SCR

The NO_x in the exhaust gas can be reduced with primary or secondary reduction methods. Primary methods affect the engine combustion process directly, whereas secondary methods reduce the emission level without changing the engine performance using equipment that does not form part of the engine itself.

For further information about emission control we refer to our publication:

Exhaust Gas Emission Control Today and Tomorrow

The publication is available at www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

Engine with Selective Catalytic Reduction System

Option: 4 60 135

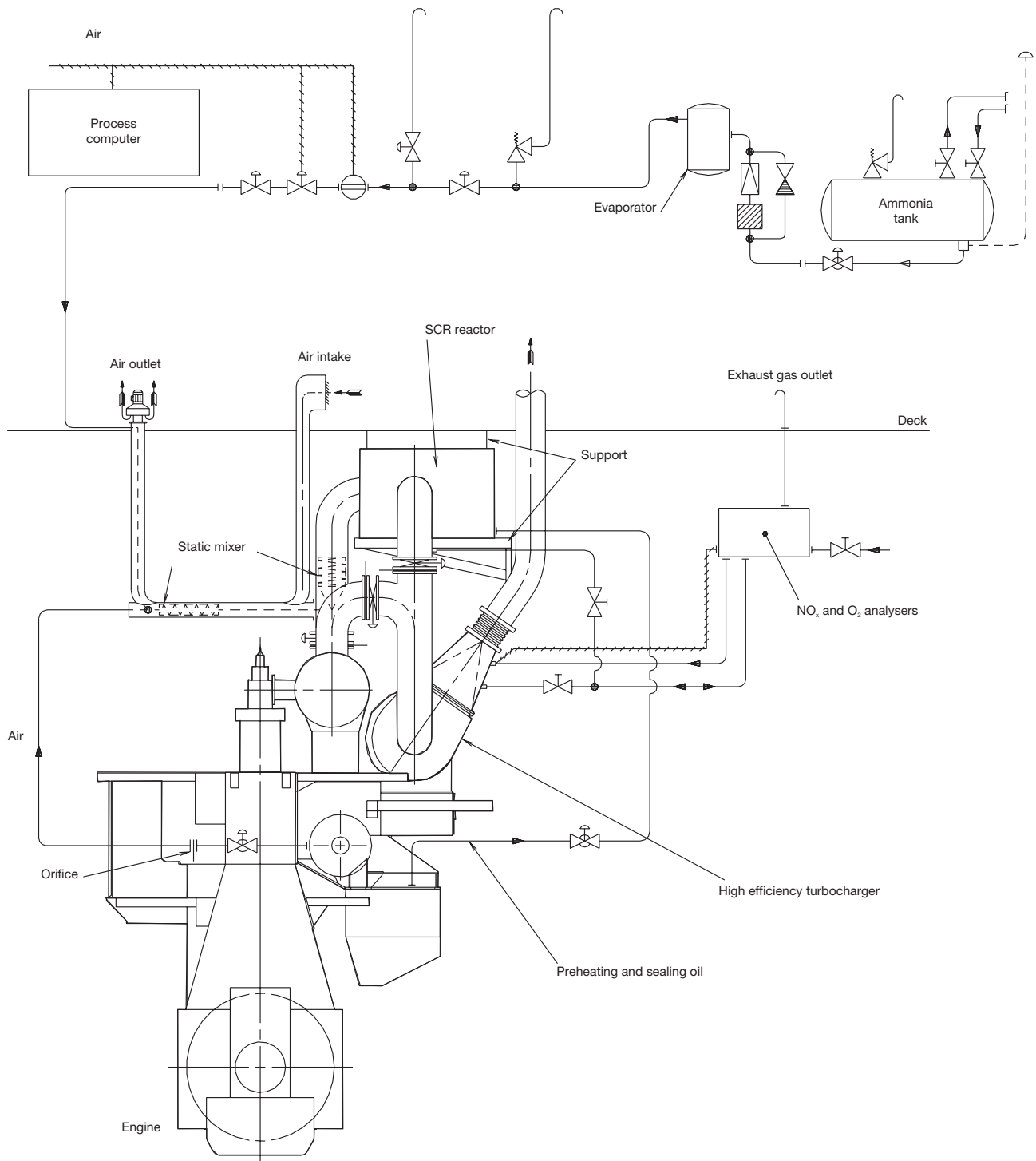
If a reduction between 50 and 98% of NO_x is required, the Selective Catalytic Reduction (SCR) system has to be applied by adding ammonia or urea to the exhaust gas before it enters a catalytic converter.

The exhaust gas must be mixed with ammonia before passing through the catalyst, and in order to encourage the chemical reaction the temperature level has to be between 300 and 400 °C. During this process the NO_x is reduced to N₂ and water.

This means that the SCR unit has to be located before the turbocharger on two-stroke engines because of their high thermal efficiency and thereby a relatively low exhaust gas temperature.

The amount of ammonia injected into the exhaust gas is controlled by a process computer and is based on the NO_x production at different loads measured during the testbed running. Fig. 3.03.01.

As the ammonia is a combustible gas, it is supplied through a double-walled pipe system, with appropriate venting and fitted with an ammonia leak detector (Fig. 3.03.01) which shows a simplified system layout of the SCR installation.



198 99 27-1.0

Fig. 3.03.01: Layout of SCR system

Electricity Production

4

Electricity Production

Introduction

Next to power for propulsion, electricity production is the largest fuel consumer on board. The electricity is produced by using one or more of the following types of machinery, either running alone or in parallel:

- Auxiliary diesel generating sets
- Main engine driven generators
- Exhaust gas- or steam driven turbo generator utilising exhaust gas waste heat (Thermo Efficiency System)
- Emergency diesel generating sets.

The machinery installed should be selected on the basis of an economic evaluation of first cost, operating costs, and the demand for man-hours for maintenance.

In the following, technical information is given regarding main engine driven generators (PTO), different configurations with exhaust gas and steam driven turbo generators, and the auxiliary diesel generating sets produced by MAN Diesel.

Power Take Off

With a generator coupled to a Power Take Off (PTO) from the main engine, electrical power can be produced based on the main engine's low SFOC and the use of heavy fuel oil. Several standardised PTO systems are available, see Fig. 4.01.01 and the designations in Fig. 4.01.02:


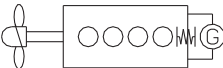

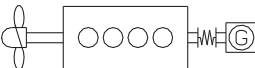

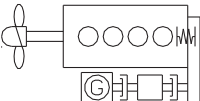

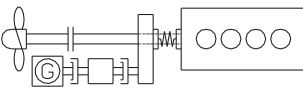

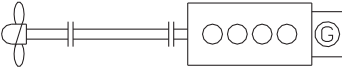

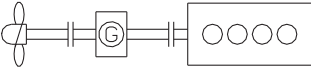
- PTO/RCF
(*Power Take Off/Renk Constant Frequency*):
Generator giving constant frequency, based on mechanical-hydraulic speed control.
- PTO/CFE
(*Power Take Off/Constant Frequency Electrical*):
Generator giving constant frequency, based on electrical frequency control.

The DMG/CFE (*Direct Mounted Generator/Constant Frequency Electrical*) and the SMG/CFE (*Shaft Mounted Generator/Constant Frequency Electrical*) are special designs within the PTO/CFE group in which the generator is coupled directly to the main engine crankshaft and the intermediate shaft, respectively, without a gear. The electrical output of the generator is controlled by electrical frequency control.

Within each PTO system, several designs are available, depending on the positioning of the gear:

- BW I:
Gear with a vertical generator mounted onto the fore end of the diesel engine, without any connections to the ship structure.
- BW II:
A free-standing gear mounted on the tank top and connected to the fore end of the diesel engine, with a vertical or horizontal generator.
- BW III:
A crankshaft gear mounted onto the fore end of the diesel engine, with a side-mounted generator without any connections to the ship structure.
- BW IV:
A free-standing step-up gear connected to the intermediate shaft, with a horizontal generator.

The most popular of the gear based alternatives are the BW III/RCF type for plants with a fixed pitch propeller (FPP). The BW III/RCF requires no separate seating in the ship and only little attention from the shipyard with respect to alignment.

Alternative types and layouts of shaft generators		Design	Seating	Total efficiency (%)
PTO/RCF	1a  1b 	BW I/RCF	On engine (vertical generator)	88-91
	2a  2b 	BW II/RCF	On tank top	88-91
	3a  3b 	BW III/RCF	On engine	88-91
	4a  4b 	BW IV/RCF	On tank top	88-91
PTO/CFE	5a  5b 	DMG/CFE	On engine	84-88
	6a  6b 	SMG/CFE	On tank top	84-88

178 57 12-1.0

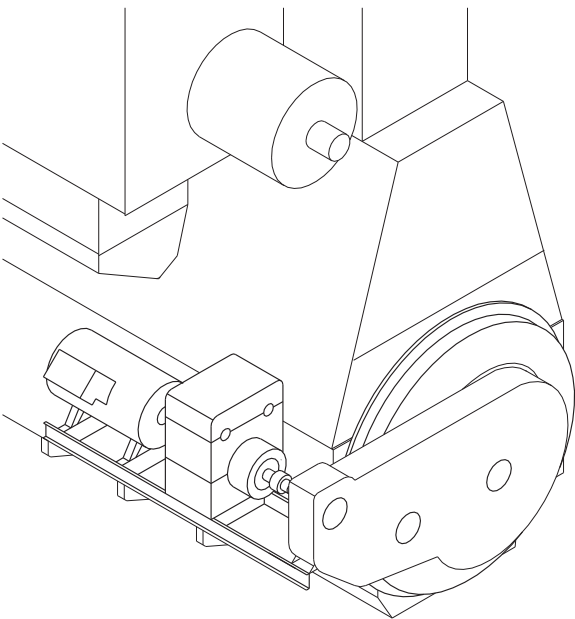
Fig. 4.01.01: Types of PTO

Designation of PTO

For further information, please refer to our publication titled:

Shaft Generators for MC and ME engines

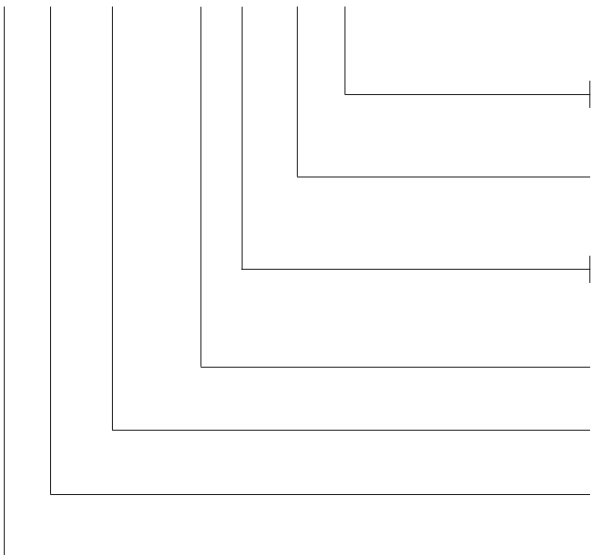
The publication is available at: www.mandiesel.com
under 'Quicklinks' → 'Technical Papers'



178 06 49-0.0

Power take off:

BW III S90ME-C7/RCF 700-60



50: 50 Hz
60: 60 Hz

kW on generator terminals

RCF: Renk constant frequency unit
CFE: Electrically frequency controlled unit

Mark version

Engine type on which it is applied

Layout of PTO: See Fig. 4.01.01

Make: MAN Diesel

178 39 55-6.0

Fig. 4.01.02: Example of designation of PTO

PTO/RCF

Side mounted generator, BWIII/RCF
(Fig. 4.01.01, Alternative 3)

The PTO/RCF generator systems have been developed in close cooperation with the German gear manufacturer RENK. A complete package solution is offered, comprising a flexible coupling, a step-up gear, an epicyclic, variable-ratio gear with built-in clutch, hydraulic pump and motor, and a standard generator, see Fig. 4.01.03.

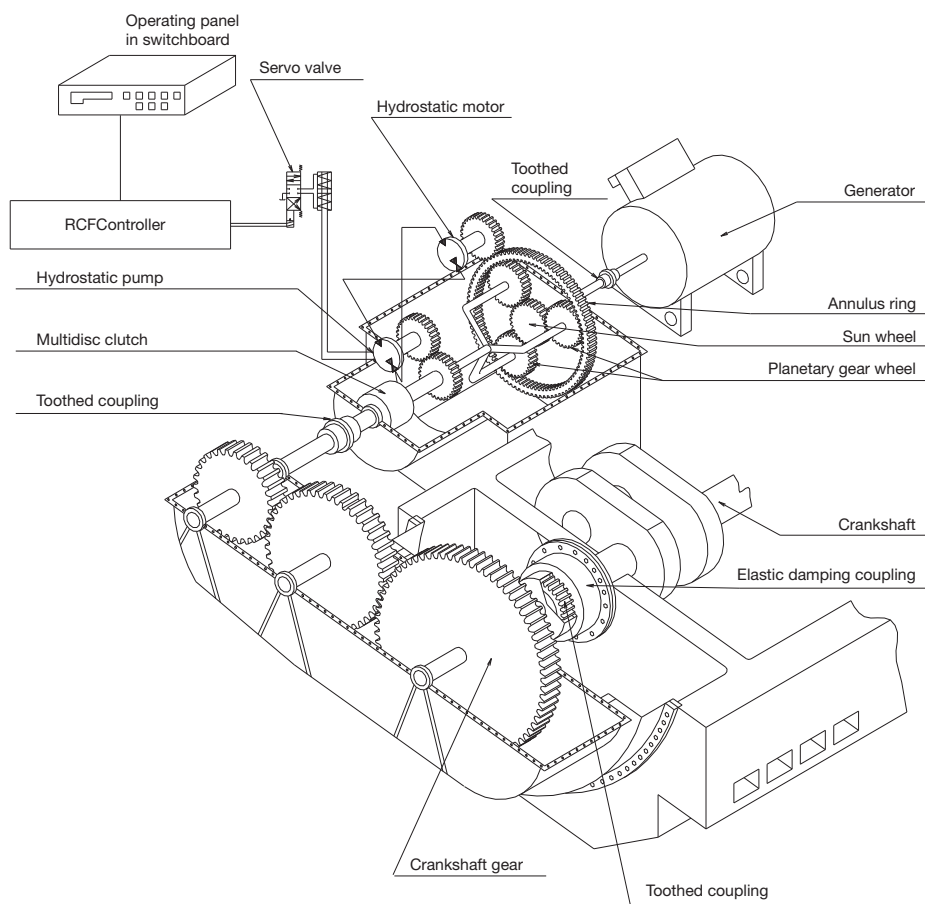
For marine engines with controllable pitch propellers running at constant engine speed, the hydraulic system can be dispensed with, i.e. a PTO/GCR design is normally used.

Fig. 4.01.03 shows the principles of the PTO/RCF arrangement. As can be seen, a step-up gear box (called crankshaft gear) with three gear wheels

is bolted directly to the frame box of the main engine. The bearings of the three gear wheels are mounted in the gear box so that the weight of the wheels is not carried by the crankshaft. In the frame box, between the crankcase and the gear drive, space is available for tuning wheel, counterweights, axial vibration damper, etc.

The first gear wheel is connected to the crankshaft via a special flexible coupling made in one piece with a tooth coupling driving the crankshaft gear, thus isolating it against torsional and axial vibrations.

By means of a simple arrangement, the shaft in the crankshaft gear carrying the first gear wheel and the female part of the toothed coupling can be moved forward, thus disconnecting the two parts of the toothed coupling.



178 23 22-2.1

Fig. 4.01.03: Power take off with RENK constant frequency gear: BW III/RCF, option: 4 85 253

The power from the crankshaft gear is transferred, via a multi-disc clutch, to an epicyclic variable-ratio gear and the generator. These are mounted on a common bedplate, bolted to brackets integrated with the engine bedplate.

The BWIII/RCF unit is an epicyclic gear with a hydrostatic superposition drive. The hydrostatic input drives the annulus of the epicyclic gear in either direction of rotation, hence continuously varying the gearing ratio to keep the generator speed constant throughout an engine speed variation of 30%. In the standard layout, this is between 100% and 70% of the engine speed at specified MCR, but it can be placed in a lower range if required.

The input power to the gear is divided into two paths – one mechanical and the other hydrostatic – and the epicyclic differential combines the power of the two paths and transmits the combined power to the output shaft, connected to the generator. The gear is equipped with a hydrostatic motor driven by a pump, and controlled by an electronic control unit. This keeps the generator speed constant during single running as well as when running in parallel with other generators.

The multi-disc clutch, integrated into the gear input shaft, permits the engaging and disengaging of the epicyclic gear, and thus the generator, from the main engine during operation.

An electronic control system with a RENK controller ensures that the control signals to the main electrical switchboard are identical to those for the normal auxiliary generator sets. This applies to ships with automatic synchronising and load sharing, as well as to ships with manual switchboard operation.

Internal control circuits and interlocking functions between the epicyclic gear and the electronic control box provide automatic control of the functions necessary for the reliable operation and protection of the BWIII/RCF unit. If any monitored value exceeds the normal operation limits, a warning or an alarm is given depending upon the origin, severity and the extent of deviation from the permissible values. The cause of a warning or an alarm is shown on a digital display.

Extent of delivery for BWIII/RCF units

The delivery comprises a complete unit ready to be built-on to the main engine. Fig. 4.02.01 shows the required space and the standard electrical output range on the generator terminals.

Standard sizes of the crankshaft gears and the RCF units are designed for: 700, 1200, 1800 and 2600 kW, while the generator sizes of make A. van Kaick are:

Type DSG		440 V 1800 kVA	60 Hz r/min kW	380 V 1500 kVA	50 Hz r/min kW
62	M2-4	707	566	627	501
62	L1-4	855	684	761	609
62	L2-4	1,056	845	940	752
74	M1-4	1,271	1,017	1,137	909
74	M2-4	1,432	1,146	1,280	1,024
74	L1-4	1,651	1,321	1,468	1,174
74	L2-4	1,924	1,539	1,709	1,368
86	K1-4	1,942	1,554	1,844	1,475
86	M1-4	2,345	1,876	2,148	1,718
86	L2-4	2,792	2,234	2,542	2,033
99	K1-4	3,222	2,578	2,989	2,391

178 34 89-3.1

In the event that a larger generator is required, please contact MAN Diesel.

If a main engine speed other than the nominal is required as a basis for the PTO operation, it must be taken into consideration when determining the ratio of the crankshaft gear. However, it has no influence on the space required for the gears and the generator.

The PTO can be operated as a motor (PTI) as well as a generator by making some minor modifications.

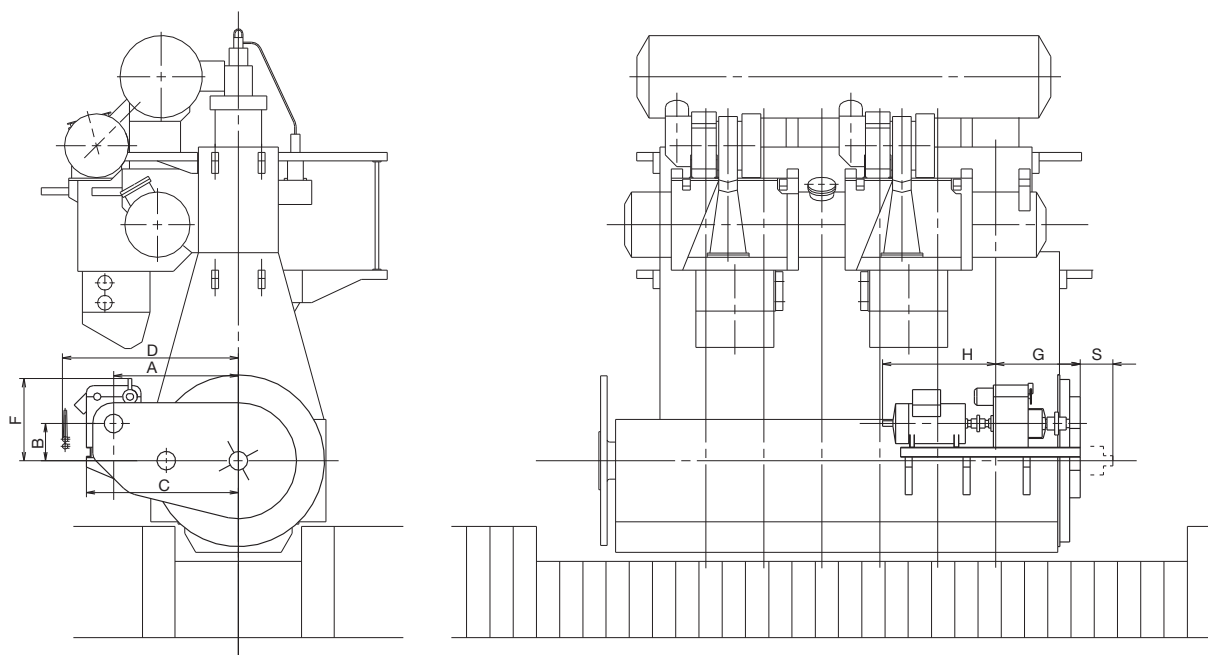
Yard deliveries are:

1. Cooling water pipes to the built-on lubricating oil cooling system, including the valves.
2. Electrical power supply to the lubricating oil stand-by pump built on to the RCF unit.
3. Wiring between the generator and the operator control panel in the switchboard.
4. An external permanent lubricating oil filling-up connection can be established in connection with the RCF unit. The system is shown in Fig. 4.03.03 'Lubricating oil system for RCF gear'. The dosage tank and the pertaining piping are to be delivered by the yard. The size of the dosage tank is stated in the table for RCF gear in 'Necessary capacities for PTO/RCF' (Fig. 4.03.02).

The necessary preparations to be made on the engine are specified in Figs. 4.03.01a and 4.03.01b.

Additional capacities required for BWIII/RCF

The capacities stated in the 'List of capacities' for the main engine in question are to be increased by the additional capacities for the crankshaft gear and the RCF gear stated in Fig. 4.03.02.



178 36 29-6.1

	kW generator			
	700 kW	1200 kW	1800 kW	2600 kW
A	3,568	3,568	3,708	3,708
B	623	623	623	623
C	4,228	4,228	4,508	4,508
D	4,620	4,620	4,900	4,900
F	1,673	1,793	1,913	2,023
G	3,025	3,025	3,385	3,385
H	1,453	1,955	2,330	3,660
S	430	530	620	710
	System mass (kg) with generator:			
	36,250	41,500	55,100	71,550
	System mass (kg) without generator:			
	34,250	38,850	50,800	66,350

The stated kW at the generator terminals is available between 70% and 100% of the engine speed at specified MCR

Space requirements have to be investigated case by case on plants with 2600 kW generator.

Dimension H: This is only valid for A. van Kaick generator type DSG, enclosure IP23,
frequency = 60 Hz, speed = 1800 r/min

Fig. 4.02.01: Space requirement for side mounted generator PTO/RCF type BWIII K90/RCF

Engine preparations for PTO

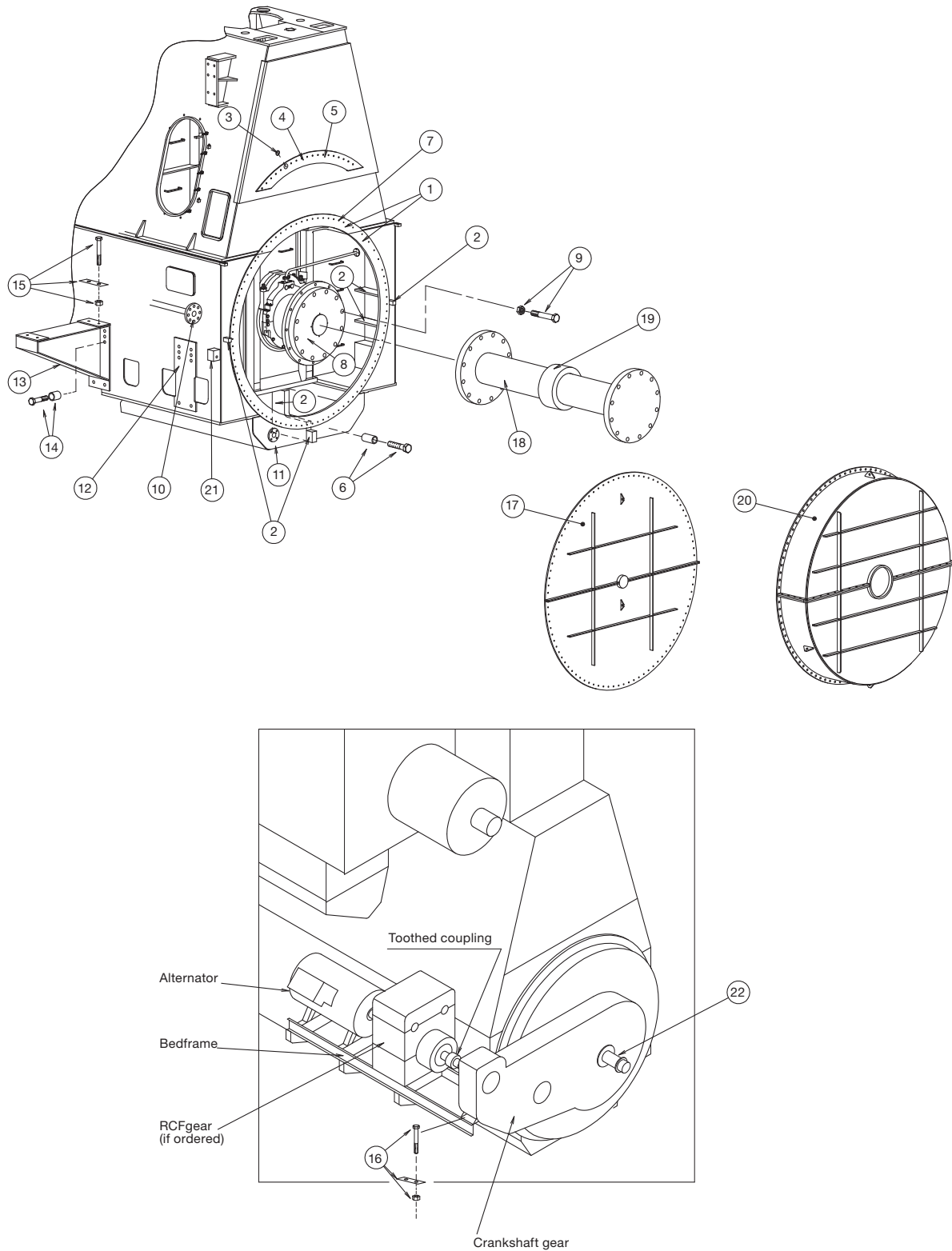


Fig. 4.03.01a: Engine preparations for PTO

178 57 15-7.0

Pos.

- 1 Special face on bedplate and frame box
- 2 Ribs and brackets for supporting the face and machined blocks for alignment of gear or stator housing
- 3 Machined washers placed on frame box part of face to ensure that it is flush with the face on the bedplate
- 4 Rubber gasket placed on frame box part of face
- 5 Shim placed on frame box part of face to ensure that it is flush with the face of the bedplate
- 6 Distance tubes and long bolts
- 7 Threaded hole size, number and size of spring pins and bolts to be made in agreement with PTO maker
- 8 Flange of crankshaft, normally the standard execution can be used
- 9 Studs and nuts for crankshaft flange
- 10 Free flange end at lubricating oil inlet pipe (incl. blank flange)
- 11 Oil outlet flange welded to bedplate (incl. blank flange)
- 12 Face for brackets
- 13 Brackets
- 14 Studs for mounting the brackets
- 15 Studs, nuts and shims for mounting of RCF-/generator unit on the brackets
- 16 Shims, studs and nuts for connection between crankshaft gear and RCF-/generator unit
- 17 Engine cover with connecting bolts to bedplate/frame box to be used for shop test without PTO
- 18 Intermediate shaft between crankshaft and PTO
- 19 Oil sealing for intermediate shaft
- 20 Engine cover with hole for intermediate shaft and connecting bolts to bedplate/frame box
- 21 Plug box for electronic measuring instrument for checking condition of axial vibration damper
- 22 Tacho encoder for ME control system or Alpha lubrication system on MC engine
- 23 Tacho trigger ring for ME control system or Alpha lubrication system on MC engine

Pos. no:	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
BWIII/RCF	A	A	A	A		B		A	B	A	A	A	A	A	B	B	A				A	A	
BWIII/CFE	A	A	A	A		B		A	B	A	A	A	A	A	B	B	A				A	A	
BWII/RCF								A	A									A	A	A	A		A
BWII/CFE								A	A									A	A	A	A		A
BWI/RCF	A	A	A	A		B		A	B								A				A		A
BWI/CFE	A	A	A	A		B		A	B	A	A						A				A		A
DMG/CFE	A	A			A	B	C	A	B								A				A		A

A: Preparations to be carried out by engine builder

B: Parts supplied by PTO-maker

C: See text of pos. no.

178 89 34-2.0

Fig. 4.03.01b: Engine preparations for PTO

Crankshaft gear lubricated from the main engine lubricating oil system

The figures are to be added to the main engine capacity list:

Nominal output of generator	kW	700	1,200	1,800	2,600
Lubricating oil flow	m ³ /h	4.1	4.1	4.9	6.2
Heat dissipation	kW	12.1	20.8	31.1	45.0

RCF gear with separate lubricating oil system:

Nominal output of generator	kW	700	1,200	1,800	2,600
Cooling water quantity	m ³ /h	14.1	22.1	30.0	39.0
Heat dissipation	kW	55	92	134	180
El. power for oil pump	kW	11.0	15.0	18.0	21.0
Dosage tank capacity	m ³	0.40	0.51	0.69	0.95
El. power for Renk-controller	24V DC \pm 10%, 8 amp				

From main engine:

Design lube oil pressure: 2.25 bar

Lube oil pressure at crankshaft gear: min. 1 bar

Lube oil working temperature: 50 °C

Lube oil type: SAE 30

Cooling water inlet temperature: 36 °C

Pressure drop across cooler: approximately 0.5 bar

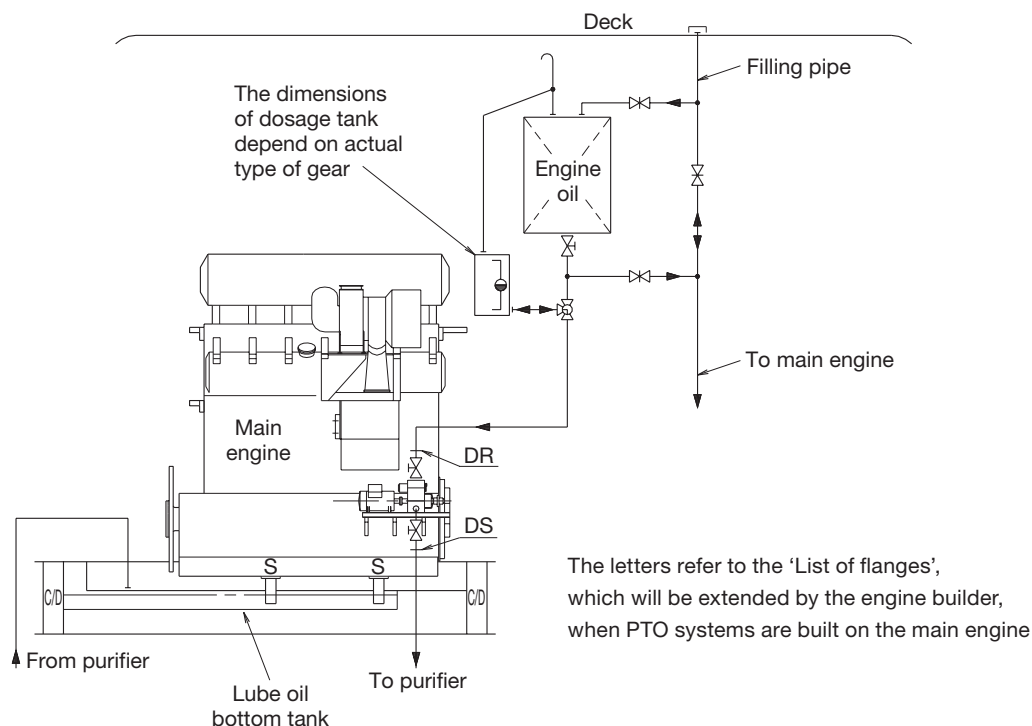
Fill pipe for lube oil system store tank (~ø32)

Drain pipe to lube oil system drain tank (~ø40)

Electric cable between Renk terminal at gearbox and operator control panel in switchboard: Cable type FMGCG 19 x 2 x 0.5

178 33 85-0.0

Fig. 4.03.02: Necessary capacities for PTO/RCF, BW III/RCF system



178 25 23-5.0

Fig. 4.03.03: Lubricating oil system for RCF gear

DMG/CFE Generators Option: 4 85 259

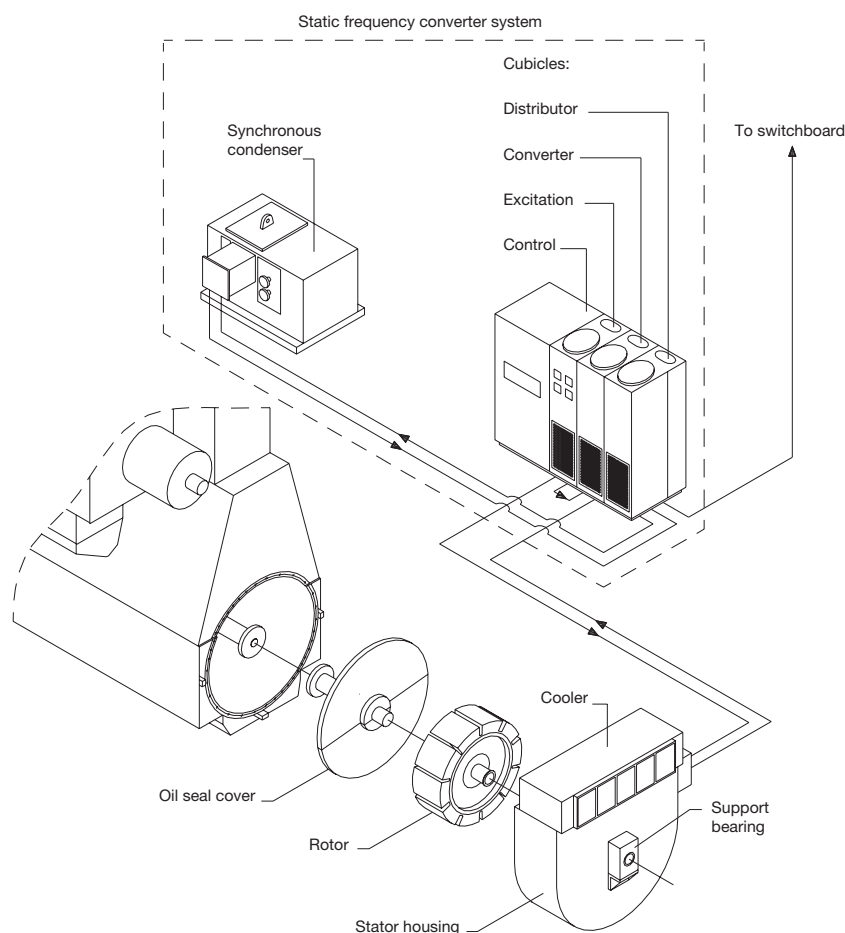
Fig. 4.01.01 alternative 5, shows the DMG/CFE (Direct Mounted Generator/Constant Frequency Electrical) which is a low speed generator with its rotor mounted directly on the crankshaft and its stator bolted on to the frame box as shown in Figs. 4.03.04 and 4.03.05.

The DMG/CFE is separated from the crankcase by a plate and a labyrinth stuffing box.

The DMG/CFE system has been developed in co-operation with the German generator manufacturers Siemens and AEG, but similar types of generator can be supplied by others, e.g. Fuji, Taiyo and Nishishiba in Japan.

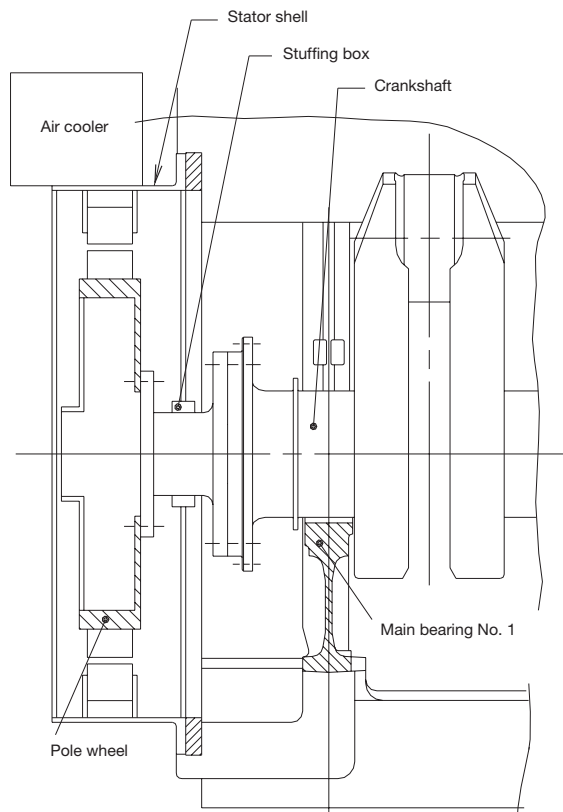
For generators in the normal output range, the mass of the rotor can normally be carried by the foremost main bearing without exceeding the permissible bearing load (see Fig. 4.03.05), but this must be checked by the engine manufacturer in each case.

If the permissible load on the foremost main bearing is exceeded, e.g. because a tuning wheel is needed, this does not preclude the use of a DMG/CFE.

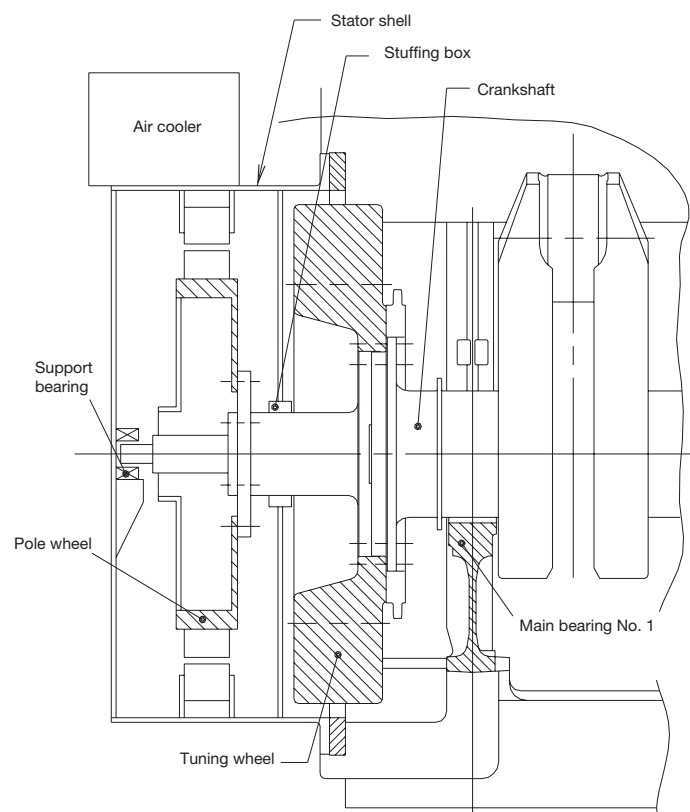


178 06 73-3.1

Fig. 4.03.04: Standard engine, with direct mounted generator (DMG/CFE)



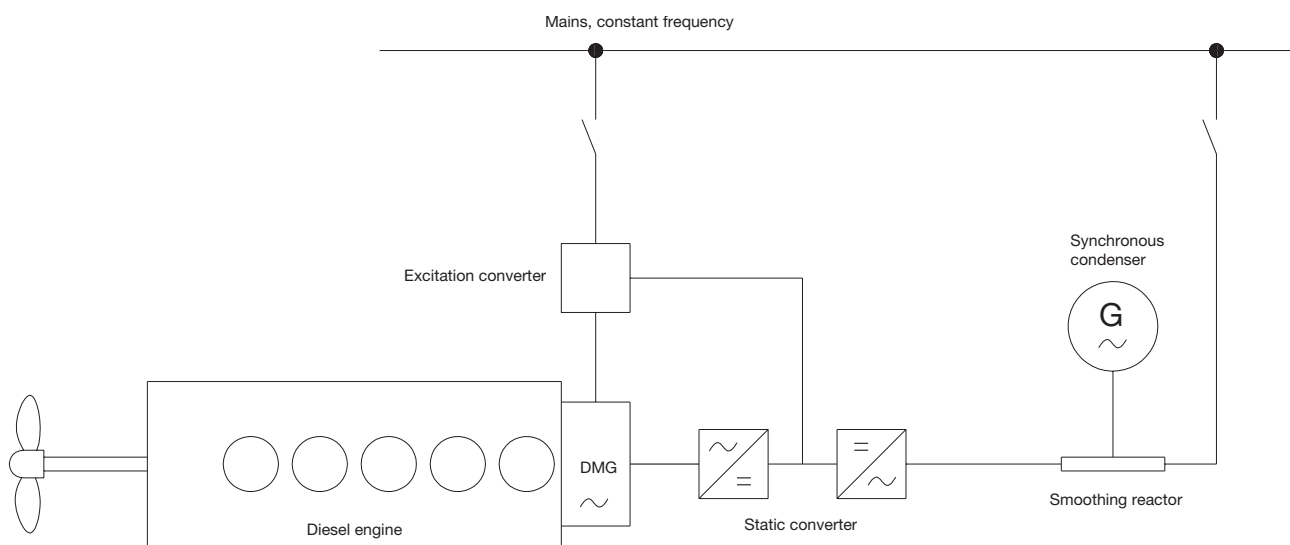
Standard engine, with direct
mounted generator (DMG/CFE)



Standard engine, with direct mounted
generator and tuning wheel

178 06 63-7.1

Fig. 4.03.05: Standard engine, with direct mounted generator and tuning wheel



178 56 55-3.1

Fig. 4.03.06: Diagram of DMG/CFE with static converter

In such a case, the problem is solved by installing a small, elastically supported bearing in front of the stator housing, as shown in Fig. 4.03.05.

As the DMG type is directly connected to the crankshaft, it has a very low rotational speed and, consequently, the electric output current has a low frequency – normally of the order of 15 Hz.

Therefore, it is necessary to use a static frequency converter between the DMG and the main switchboard. The DMG/CFE is, as standard, laid out for operation with full output between 100% and 70% and with reduced output between 70% and 50% of the engine speed at specified MCR.

Static converter

The static frequency converter system (see Fig. 4.03.06) consists of a static part, i.e. thyristors and control equipment, and a rotary electric machine.

The DMG produces a three-phase alternating current with a low frequency, which varies in accordance with the main engine speed. This alternating current is rectified and led to a thyristor inverter producing a three-phase alternating current with constant frequency.

Since the frequency converter system uses a DC intermediate link, no reactive power can be supplied to the electric mains. To supply this reactive power, a synchronous condenser is used. The synchronous condenser consists of an ordinary synchronous generator coupled to the electric mains.

Extent of delivery for DMG/CFE units

The delivery extent is a generator fully built-on to the main engine including the synchronous condenser unit and the static converter cubicles which are to be installed in the engine room.

The DMG/CFE can, with a small modification, be operated both as a generator and as a motor (PTI).

Yard deliveries are:

1. Installation, i.e. seating in the ship for the synchronous condenser unit and for the static converter cubicles
2. Cooling water pipes to the generator if water cooling is applied
3. Cabling.

The necessary preparations to be made on the engine are specified in Figs. 4.03.01a and 4.03.01b.

SMG/CFE Generators

The PTO SMG/CFE (see Fig. 4.01.01 alternative 6) has the same working principle as the PTO DMG/CFE, but instead of being located on the front end of the engine, the alternator is installed aft of the engine, with the rotor integrated on the intermediate shaft.

In addition to the yard deliveries mentioned for the PTO DMG/CFE, the shipyard must also provide the foundation for the stator housing in the case of the PTO SMG/CFE.

The engine needs no preparation for the installation of this PTO system.

PTO/BW GCR

This section is not applicable

Waste Heat Recovery Systems (WHR)

Due to the increasing fuel prices seen from 2004 and onwards many shipowners have shown interest in efficiency improvements of the power systems on board their ships. A modern two-stroke diesel engine has one of the highest thermal efficiencies of today's power systems, but even this high efficiency can be improved by combining the diesel engine with other power systems.

One of the possibilities for improving the efficiency is to install one or more systems utilising some of the energy in the exhaust gas after the two-stroke engine, which in MAN Diesel terms is designated as WHR (Waste Heat Recovery Systems).

WHR can be divided into different types of sub-systems, depending on how the system utilises the exhaust gas energy. Choosing the right system for a specific project depends on the electricity demand on board the ship and the acceptable first cost for the complete installation. MAN Diesel uses the following designations for the current systems on the market:

- **PTG (Power Turbine Generator):**
An exhaust gas driven turbine connected to a generator via a gearbox.
- **STG (Steam Turbine Generator):**
A steam driven turbine connected to a generator via a gearbox. The steam is produced in a large exhaust gas driven boiler installed on the main engine exhaust gas piping system.
- **Combined Turbines:**
A combination of the two first systems. The arrangement is often that the power turbine is connected to the steam turbine via a gearbox and the steam turbine is further connected to a large generator, which absorbs the power from both turbines.

The PTG system will produce power equivalent to approx. 4% of the main engine SMCR, when the engine is running at SMCR. For the STG system this value is between 5 and 7% depending on the system installed. When combining the two systems, a power output equivalent to 10% of the main engine's SMCR is possible, when the engine is running at SMCR.

As the electrical power produced by the system needs to be used on board the ship, specifying the correct size system for a specific project must be considered carefully. In cases where the electrical power consumption on board the ship is low, a smaller system than possible for the engine type may be considered. Another possibility is to install a shaft generator/motor to absorb excess power produced by the WHR. The main engine will then be unloaded, or it will be possible to increase the speed of the ship, without penalising the fuelbill.

Because the energy from WHR is taken from the exhaust gas of the main engine, this power produced can be considered as "free". In reality, the main engine SFOC will increase slightly, but the gain in electricity production on board the ship will far surpass this increase in SFOC. As an example, the SFOC of the combined output of both the engine and the system with power and steam turbine can be calculated to be as low as 155 g/kWh (ref. LCV 42,700 kJ/kg).

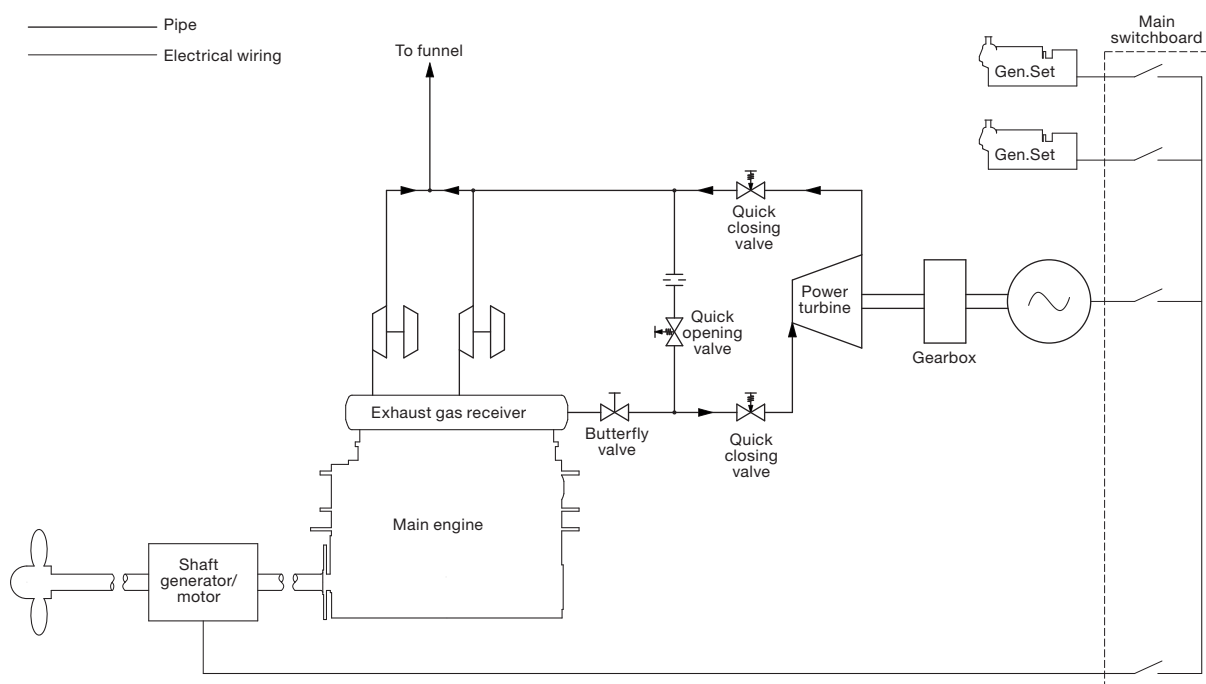
Power Turbine Generator (PTG)

The power turbines of today are based on the different turbocharger suppliers' newest designs of high-efficiency turbochargers, i.e. MAN Diesel's TCA, ABB's TPL and Mitsubishi's MA turbochargers.

The power turbine basically is the turbine side of a normal high-efficient turbocharger with some modifications to the bearings and the turbine shaft. This is in order to be able to connect it to a gearbox instead of the normal connection to the compressor side. The power turbine will be installed on a separate exhaust gas pipe from the exhaust gas receiver, which bypasses the turbochargers.

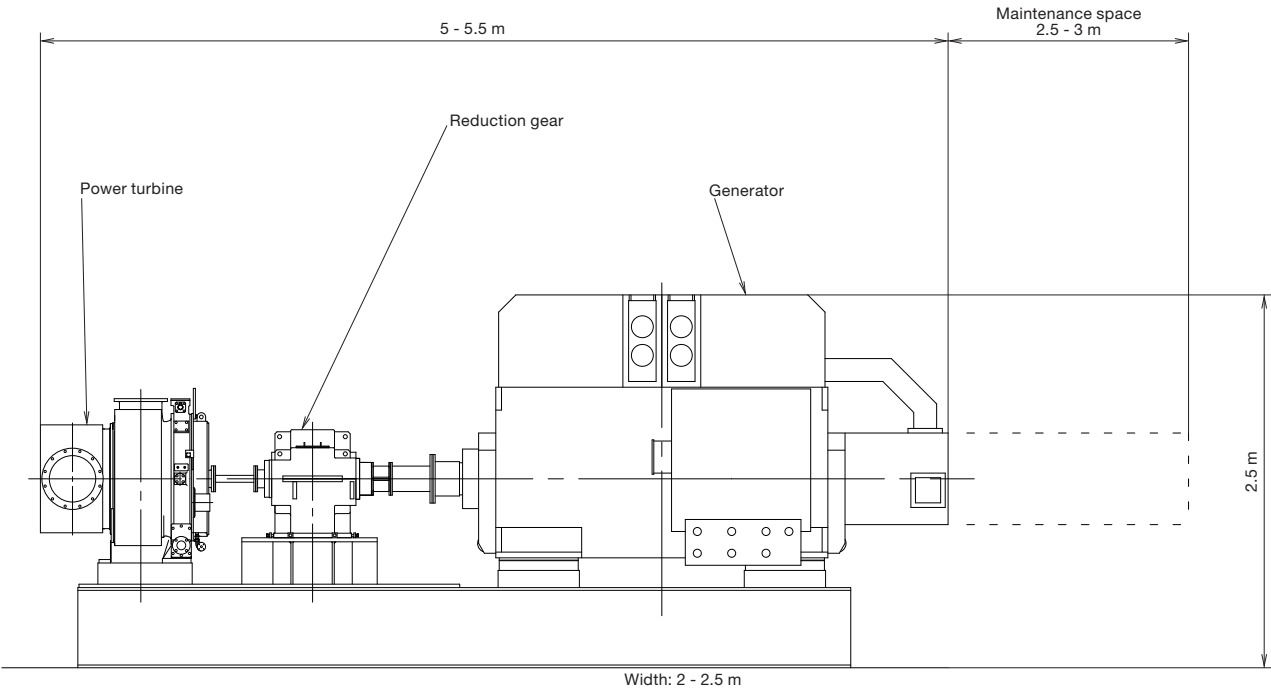
The performance of the PTG and the main engine will depend on a careful matching of the engine turbochargers and the power turbine, for which reason the turbocharger/s and the power turbine need to be from the same manufacturer. In Fig. 4.05.01, a simple diagram of the PTG arrangement is shown. The quick-opening and quick-closing valves are used in the event of a blackout of the grid, in which case the exhaust gas will bypass the power turbine.

The newest generation of high-efficiency turbochargers allows bypassing of some of the main engine exhaust gas, thereby creating a new balance of the air flow through the engine. In this way, it is possible to extract power from the power turbine equivalent to 4% of the main engine's SMCR, when the engine is running at SMCR.



178 57 09-8.0

Fig. 4.05.01: PTG diagram



178 56 93-9.0

Fig. 4.05.02: The size of a 2,000 kW PTG system depending on the supplier

Steam Turbine Generator (STG)

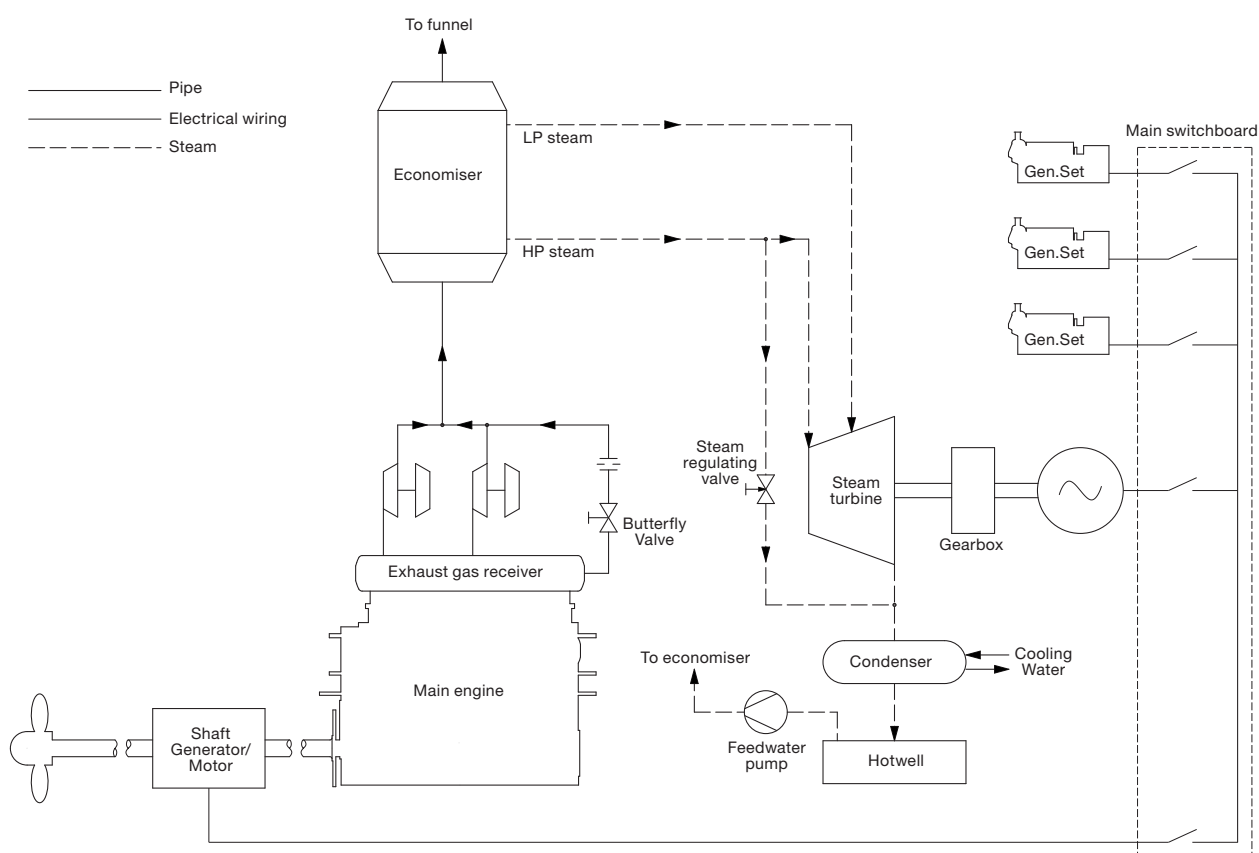
In most cases the exhaust gas pipe system of the main engine is equipped with a boiler system. With this boiler, some of the energy in the exhaust gas is utilised to produce steam for use on board the ship.

If the engine is WHR matched, the exhaust gas temperature will be between 50°C and 65°C higher than on a conventional engine, which makes it possible to install a larger boiler system and, thereby, produce more steam. In short, MAN Diesel designates this system STG. Fig. 4.05.03 shows an example of the arrangement of STG.

For WHR matching the engine, a bypass is installed to increase the temperature of the exhaust gas and improve the boiler output.

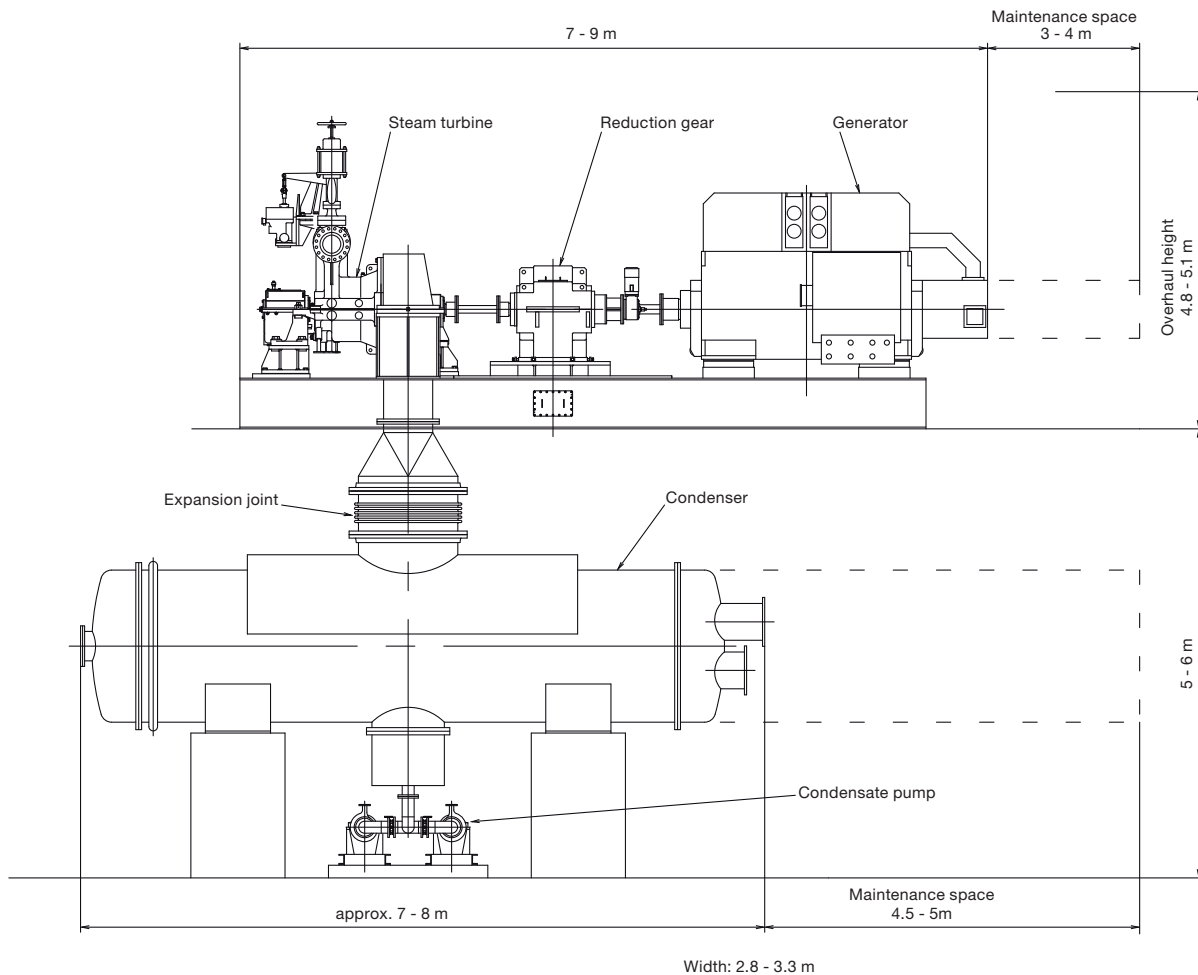
The extra steam produced in the boiler can be utilised in a steam turbine, which can be used to drive a generator for power production on board the ship. An STG system could be arranged as shown in Fig. 4.05.04, where a typical system size is shown with the outline dimensions.

The steam turbine can either be a single or dual pressure turbine, depending on the size of the system. Steam pressure for a single pressure system is 7 to 10 bara, and for the dual pressure system the high-pressure cycle will be 9 to 10 bara and the low-pressure cycle will be 4 to 5 bara.



178 56 96-4.0

Fig. 4.05.03: Steam diagram



178 57 00-1.0

Fig. 4.05.04: Typical system size for 3,000 kW STG system

Combined Turbines

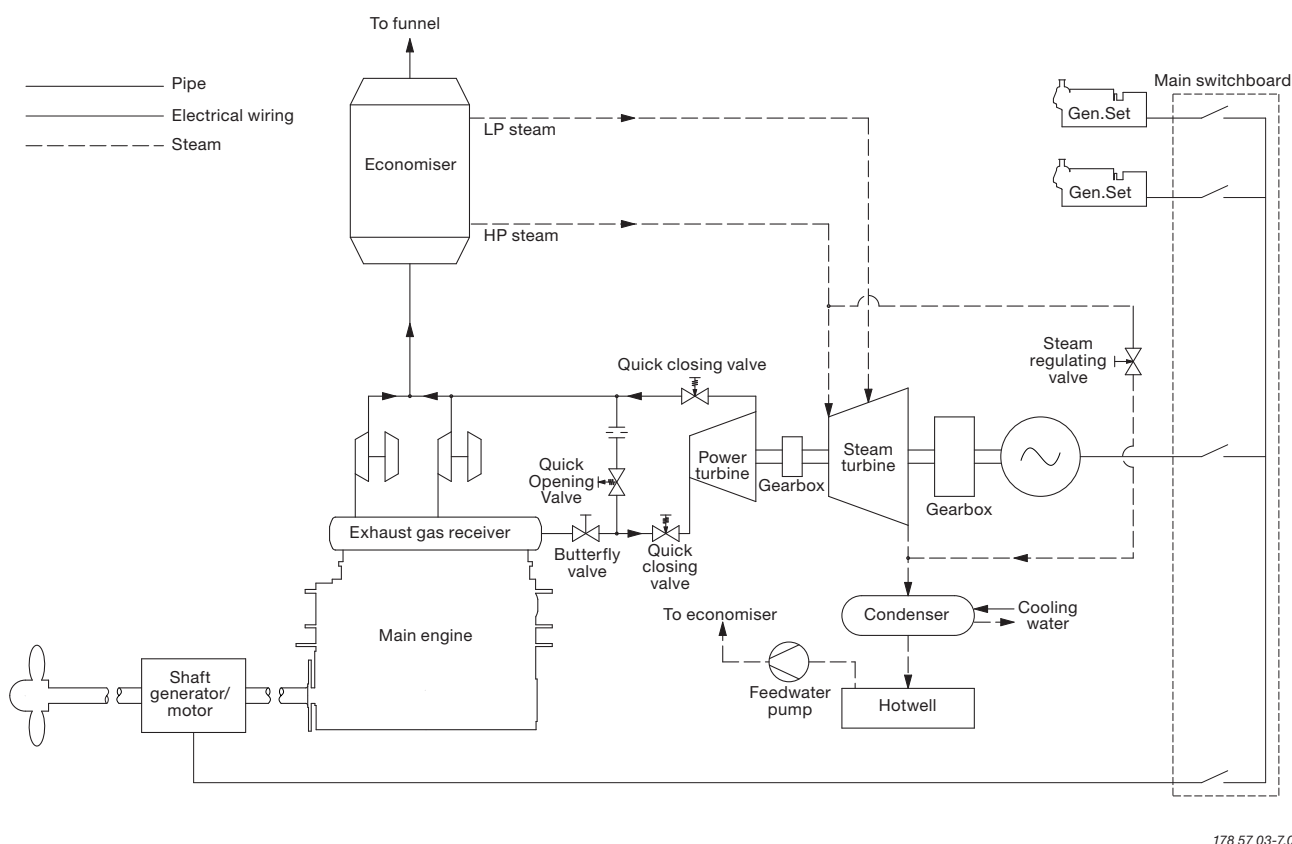
Because the installation of the power turbine also will result in an increase of the exhaust gas temperature after the turbochargers, it is possible to install both the power turbine, the larger boiler and steam turbine on the same engine. This way, the energy from the exhaust gas is utilised in the best way possible by today's components.

When looking at the system with both power and steam turbine, quite often the power turbine and the steam turbine are connected to the same generator. In some cases, it is also possible to have each turbine on a separate generator. This is, however, mostly seen on stationary engines, where the frequency control is simpler because of the large grid to which the generator is coupled.

For marine installations the power turbine is, in most cases, connected to the steam turbine via a gearbox, and the steam turbine is then connected to the generator. It is also possible to have a generator with connections in both ends, and then connect the power turbine in one end and the steam turbine in the other. In both cases control of one generator only is needed.

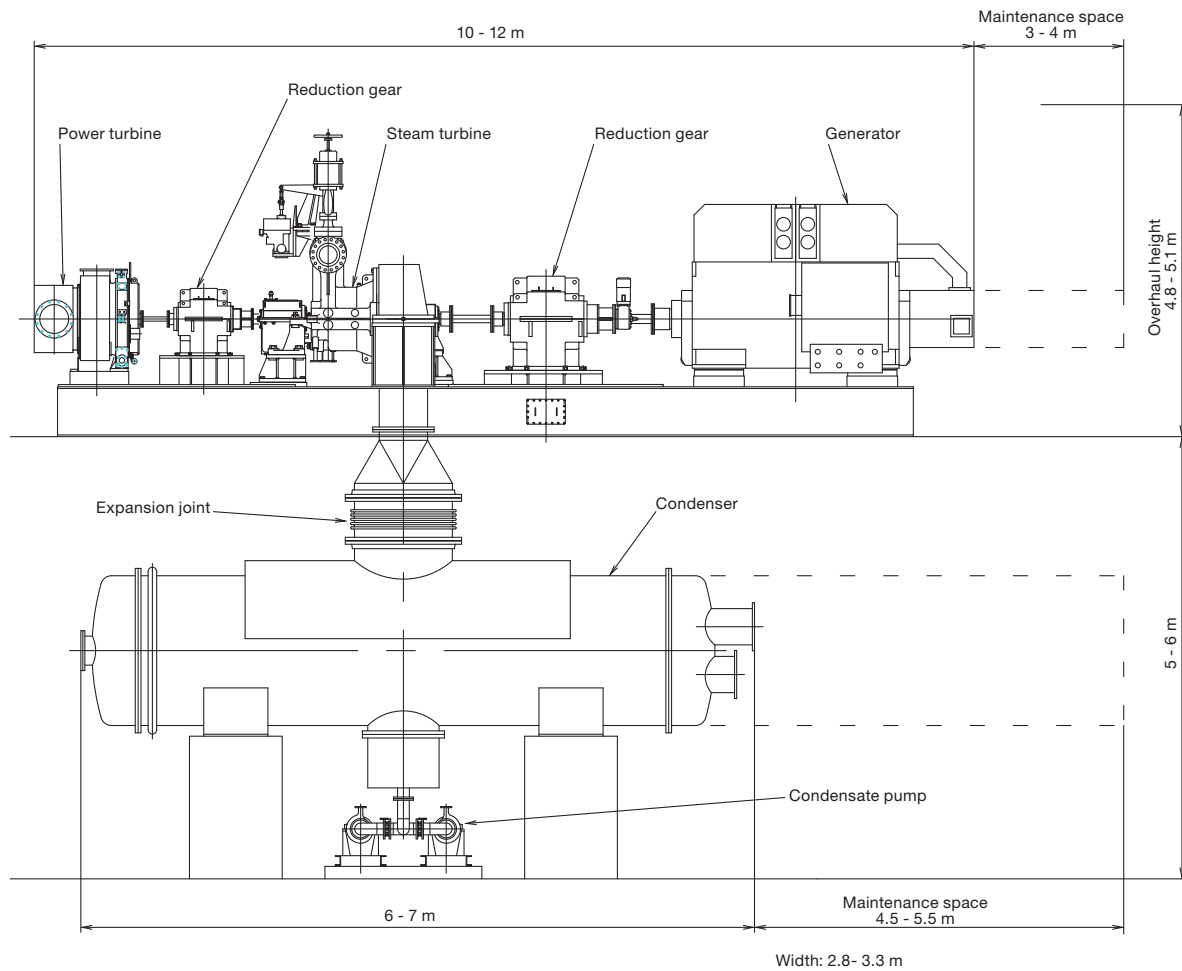
For dimensions of a typical system see Fig. 4.05.06.

As mentioned, the systems with steam turbines require a larger boiler to be installed. The size of the boiler system will be roughly three to four times the size of an ordinary boiler system, but the actual boiler size has to be calculated from case to case.



178 57 03-7.0

Fig. 4.05.05: Combined turbines diagram



178 57 06-2.0

Fig. 4.05.06: Typical system size for 4,000 kW combined turbines

WHR output

Because all the components come from different manufacturers, the final output and the system efficiency has to be calculated from case to case. However, Fig. 4.05.07 shows a guidance of possible outputs based on theoretically calculated outputs from the system.

Detailed information on the different systems is found in our paper 'Thermo Efficiency System', where the different systems are described in greater detail. The paper is available at: www.mandiesel.com under 'Quicklinks' → 'Technical Papers', from where it can be downloaded.

Guidance output of WHR for K90ME9 engine rated in L1 at ISO conditions				
Cyl.	Engine power	PTG	STG	Combined Turbines
	% SMCR	kWe	kWe	kWe
6	100	1,350	2,097	3,177
	80	855	1,494	2,133
7	100	1,575	2,457	3,717
	80	1,008	1,755	2,493
8	100	1,800	2,826	4,266
	80	1,152	2,016	2,871
9	100	2,034	3,186	4,806
	80	1,296	2,277	3,231
10	100	2,259	3,537	5,346
	80	1,440	2,529	3,600
11	100	2,484	3,906	5,895
	80	1,584	2,790	3,969
12	100	2,718	4,266	6,435
	80	1,728	3,051	4,329

Table 4.05.07: Theoretically calculated outputs

GenSet Data

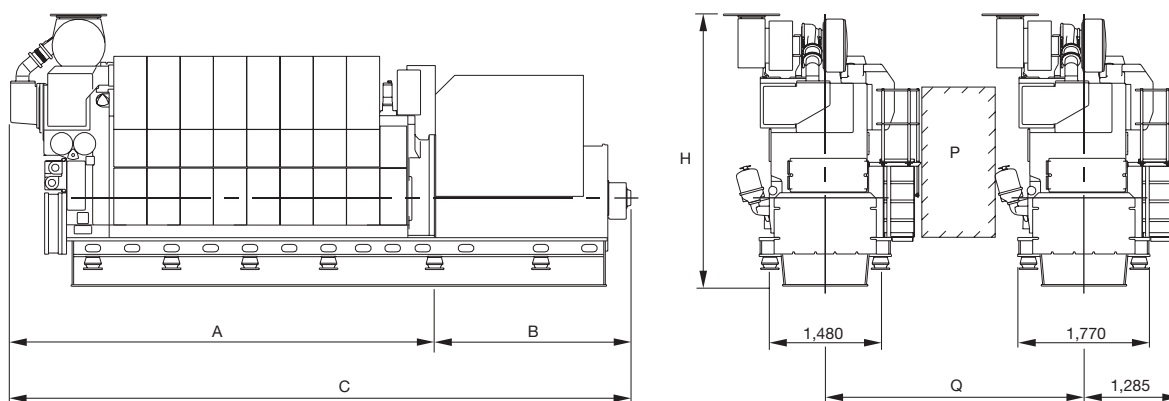
This section is not applicable

L27/38 GenSet Data

Bore: 270 mm

Stroke: 380 mm

	Power layout					
	720 r/min	60 Hz	750 r/min	50 Hz	720/750 r/min (MGO/MDO)	60/50 Hz (MGO/MDO)
	Eng. kW	Gen. kW	Eng. kW	Gen. kW	Eng. kW	Gen. kW
5L27/38	1,500	1,440	1,600	1,536	-	-
6L27/38	1,980	1,900	1,980	1,900	2,100	2,016
7L27/38	2,310	2,218	2,310	2,218	2,450	2,352
8L27/38	2,640	2,534	2,640	2,534	2,800	2,688
9L27/38	2,970	2,851	2,970	2,851	3,150	3,054



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No. of Cyls.	A (mm)	* B (mm)	* C (mm)	H (mm)	**Dry weight GenSet (t)
5 (720 r/min)	4,346	2,486	6,832	3,628	42.3
5 (750 r/min)	4,346	2,486	6,832	3,628	42.3
6 (720 r/min)	4,791	2,766	7,557	3,712	45.8
6 (750 r/min)	4,791	2,766	7,557	3,712	46.1
7 (720 r/min)	5,236	2,766	8,002	3,712	52.1
7 (750 r/min)	5,236	2,766	8,002	3,712	52.1
8 (720 r/min)	5,681	2,986	8,667	3,899	56.3
8 (750 r/min)	5,681	2,986	8,667	3,899	58.3
9 (720 r/min)	6,126	2,986	9,112	3,899	63.9
9 (750 r/min)	6,126	2,986	9,112	3,899	63.9

P Free passage between the engines, width 600 mm and height 2,000 mm

Q Min. distance between engines: 2,900 mm (without gallery) and 3,100 mm (with gallery)

* Depending on alternator

** Weight includes a standard alternator

All dimensions and masses are approximate and subject to change without prior notice.

178 33 89-8.2

Fig. 4.09.01: Power and outline of L27/38

L27/38 GenSet Data

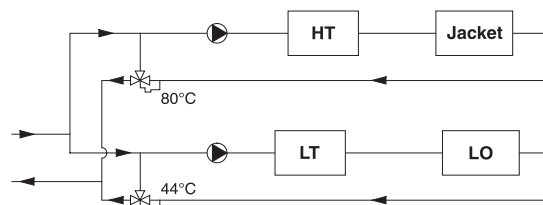
		Cyl.	5	6	7	8	9
Max continues rating	720 RPM	kW	1,500	1,980	2,310	2,640	2,970
Engine driven pumps:							
LT cooling water pump	(2.5 bar)	m³/h	58	58	58	58	58
HT cooling water pump	(2.5 bar)	m³/h	58	58	58	58	58
Lubricating oil main pump	(8 bar)	m³/h	64	64	92	92	92
Separate pumps:							
Max. Delivery pressure of cooling water pumps		bar	2.5	2.5	2.5	2.5	2.5
Diesel oil pump	(5 bar at fuel oil inlet A1)	m³/h	1.02	1.33	1.55	1.77	2.00
Fuel oil Supply pump	(4 bar at discharge pressure)	m³/h	0.50	0.66	0.76	0.87	0.98
Fuel oil circulating pump	(8 bar at fuel oil inlet A1)	m³/h	1.03	1.35	1.57	1.80	2.02
Cooling capacity:							
Lubricating oil		kW	206	283	328	376	420
Charge air LT		kW	144	392	436	473	504
Total LT system		kW	350	675	764	849	924
Flow LT at 36°C inlet and 44°C outlet		m³/h	38	58	58	58	58
Jacket cooling		kW	287	486	573	664	754
Charge air HT		kW	390	558	640	722	802
Total HT system		kW	677	1,044	1,213	1,386	1,556
Flow HT at 44°C inlet and 80°C outlet		m³/h	16	22	27	32	38
Total from engine		kW	1,027	1,719	1,977	2,235	2,480
LT flow at 36°C inlet		m³/h	38	58	58	58	58
LT temp. Outlet engine (at 36°C and 1 string cooling water system)		°C	59	58	61	64	68
Gas Data:							
Exhaust gas flow		kg/h	10,476	15,000	17,400	19,900	22,400
Exhaust gas temp.		°C	330	295	295	295	295
Max. Allowable back press.		bar	0,025	0,025	0,025	0,025	0,025
Air consumption		kg/h	10,177	14,600	17,000	19,400	21,800
Starting Air System:							
Air consumption per start		Nm³	2,5	2,9	3,3	3,8	4,3
Heat Radiation:							
Engine		kW	53	64	75	68	73
Alternator		kW	(see separate data from the alternator maker)				

The stated heat balances are based on tropical conditions.
The exhaust gas data (exhaust gas flow, exhaust gas temp. and air consumption). are based on ISO ambient condition.

* The outlet temperature of the HT water is fixed to 80°C, and 44°C for the LT water

At different inlet temperature the flow will change accordingly.

Example: If the inlet temperature is 25°C then the LT flow will change to $(46-36)/(46-25)*100 = 53\%$ of the original flow.
The HT flow will not change.



178 48 63-6.1

Fig. 4.09.02a: List of capacities for L27/38, 720 rpm, IMO Tier I. Tier II values available on request.

L27/38 GenSet Data

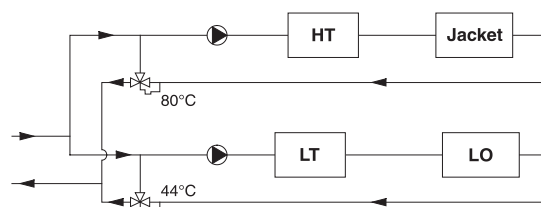
		Cyl.	5	6	7	8	9
Max continues rating	750 RPM	kW	1,600	1,980	2,310	2,640	2,970
Engine driven pumps:							
LT cooling water pump	2.5 bar	m³/h	70	70	70	70	70
HT cooling water pump	2.5 bar	m³/h	70	70	70	70	70
Lubricating oil main pump	8 bar	m³/h	66	66	96	96	96
Separate pumps:							
Max. Delivery pressure of cooling water pumps		bar	2.5	2.5	2.5	2.5	2.5
Diesel oil pump (5 bar at fuel oil inlet A1)		m³/h	1.10	1.34	1.57	1.79	2.01
Fuel oil supply pump (4 bar discharge pressure)		m³/h	0.54	0.66	0.77	0.88	0.99
Fuel oil circulating pump (8 bar at fuel oil inlet A1)		m³/h	1.11	1.36	1.59	1.81	2.04
Cooling capacity:							
Lubricating oil		kW	217	283	328	376	420
Charge air LT		kW	155	392	436	473	504
Total LT system		kW	372	675	764	849	924
Flow LT at 36°C inlet and 44°C outlet		m³/h	40	70	70	70	70
Jacket cooling		kW	402	486	573	664	754
Charge air HT		kW	457	558	640	722	802
Total HT system		kW	859	1,044	1,213	1,386	1,556
Flow HT at 44°C inlet and 80°C outlet		m³/h	21	22	27	32	38
Total from engine		kW	1,231	1,719	1,977	2,235	2,480
LT flow at 36°C inlet		m³/h	40	70	70	70	70
LT temp. Outlet engine (at 36°C and 1 string cooling water system)		°C	62	55	58	61	64
Gas Data:							
Exhaust gas flow		kg/h	11,693	15,000	17,400	19,900	22,400
Exhaust gas temp.		°C	330	305	305	305	305
Max. Allowable back press.		bar	0.025	0.025	0.025	0.025	0.025
Air consumption		kg/h	11,662	14,600	17,000	19,400	21,800
Starting Air System:							
Air consumption per start		Nm³	2.5	2.9	3.3	3.8	4.3
Heat Radiation:							
Engine		kW	54	64	75	68	73
Alternator		kW	(see separate data from the alternator maker)				

The stated heat balances are based on tropical conditions.
The exhaust gas data (exhaust gas flow, exhaust gas temp. and air consumption). are based on ISO ambient condition.

* The outlet temperature of the HT water is fixed to 80°C, and 44°C for the LT water

At different inlet temperature the flow will change accordingly.

Example: If the inlet temperature is 25°C then the LT flow will change to $(46-36)/(46-25)*100 = 53\%$ of the original flow.
The HT flow will not change.



178 48 63-6.1

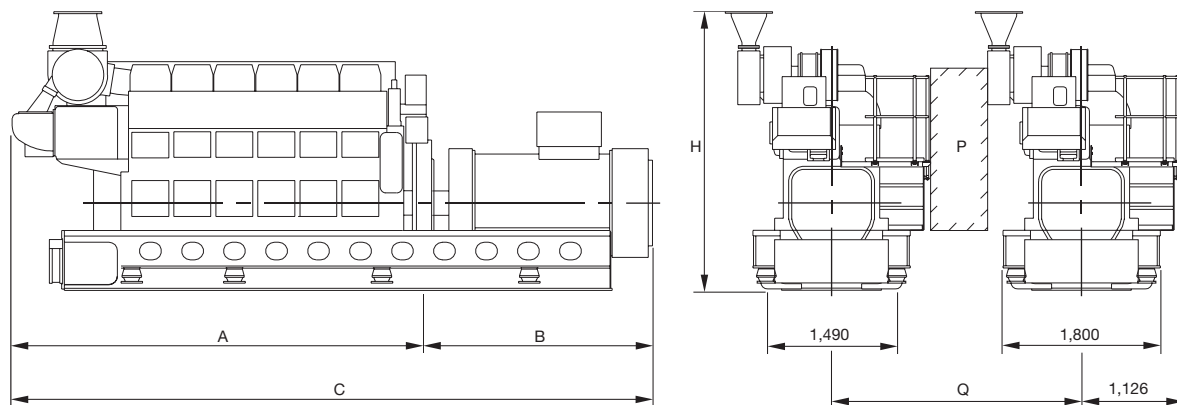
Fig. 4.09.02b: List of capacities for L27/38, 750 rpm, IMO Tier I. Tier II values available on request.

L28/32H GenSet Data

Bore: 280 mm

Stroke: 320 mm

	Power layout			
	720 r/min Eng. kW	60 Hz Gen. kW	750 r/min Eng. kW	50 Hz Gen. kW
5L28/32H	1,050	1,000	1,100	1,045
6L28/32H	1,260	1,200	1,320	1,255
7L28/32H	1,470	1,400	1,540	1,465
8L28/32H	1,680	1,600	1,760	1,670
9L28/32H	1,890	1,800	1,980	1,880



178 23 09-2.0

No. of Cyls.	A (mm)	* B (mm)	* C (mm)	H (mm)	**Dry weight GenSet (t)
5 (720 r/min)	4,279	2,400	6,679	3,184	32.6
5 (750 r/min)	4,279	2,400	6,679	3,184	32.6
6 (720 r/min)	4,759	2,510	7,269	3,184	36.3
6 (750 r/min)	4,759	2,510	7,269	3,184	36.3
7 (720 r/min)	5,499	2,680	8,179	3,374	39.4
7 (750 r/min)	5,499	2,680	8,179	3,374	39.4
8 (720 r/min)	5,979	2,770	8,749	3,374	40.7
8 (750 r/min)	5,979	2,770	8,749	3,374	40.7
9 (720 r/min)	6,199	2,690	8,889	3,534	47.1
9 (750 r/min)	6,199	2,690	8,889	3,534	47.1

P Free passage between the engines, width 600 mm and height 2,000 mm

Q Min. distance between engines: 2,655 mm (without gallery) and 2,850 mm (with gallery)

* Depending on alternator

** Weight includes a standard alternator, make A. van Kaick

All dimensions and masses are approximate and subject to change without prior notice.

178 33 92-1.3

Fig. 4.10.01: Power and outline of L28/32H

L28/32H GenSet Data

		Cyl.	5	6	7	8	9
Max. continuous rating at	720/ 750 RPM	kW	1,050/ 1,100	1,260/ 1,320	1,470/ 1,540	1,680/ 1,760	1,890/ 1,980
Engine-driven Pumps:							
Fuel oil feed pump	(5.5-7.5 bar)	m³/h	1.4	1.4	1.4	1.4	1.4
L.T. cooling water pump	(1-2.5 bar)	m³/h	45	60	75	75	75
H.T. cooling water pump	(1-2.5 bar)	m³/h	45	45	60	60	60
Lub. oil main pump	(3-5 bar)	m³/h	23	23	31	31	31
Separate Pumps:							
Diesel oil Pump	(4 bar at fuel oil inlet A1)	m³/h	0.73/0.77	0.88/0.92	1.02/1.08	1.17/1.23	1.32/1.38
Fuel oil supply pump ***	(4 bar discharge pressure)	m³/h	0.36/0.38	0.43/0.45	0.50/0.53	0.57/0.60	0.64/0.68
Fuel oil circulating pump	(8 bar at fuel oil inlet A1)	m³/h	0.74/0.78	0.89/0.93	1.04/1.09	1.18/1.25	1.33/1.40
L.T. cooling water pump*	(1-2.5 bar)	m³/h	45	54	65	77	89
L.T. cooling water pump**	(1-2.5 bar)	m³/h	65	73	95	105	115
H.T. cooling water pump	(1-2.5 bar)	m³/h	37	45	50	55	60
Lub. oil stand-by pump	(3-5 bar)	m³/h	22	23	25	27	28
Cooling Capacities:							
Lubricating Oil:							
Heat dissipation		kW	105	127	149	172	194
L.T. cooling water quantity*		m³/h	7.8	9.4	11.0	12.7	14.4
SW L.T. cooling water quantity**		m³/h	28	28	40	40	40
Lub. oil temp. inlet cooler		°C	67	67	67	67	67
L.T. cooling water temp. inlet cooler		°C	36	36	36	36	36
Charge Air:							
Heat dissipation		kW	393	467	541	614	687
L.T. cooling water quantity		m³/h	37	45	55	65	75
L.T. cooling water inlet cooler		°C	36	36	36	36	36
Jacket Cooling:							
Heat dissipation		kW	264	320	375	432	489
H.T. cooling water quantity		m³/h	37	45	50	55	60
H.T. cooling water temp. inlet cooler		°C	77	77	77	77	77
Gas Data:							
Exhaust gas flow		kg/h	9,260	11,110	12,970	14,820	16,670
Exhaust gas temp.		°C	305	305	305	305	305
Max. allowable back. press.		bar	0.025	0.025	0.025	0.025	0.025
Air consumption		kg/s	2.51	3.02	3.52	4.02	4.53
Starting Air System:							
Air consumption per start		Nm³	2.5	2.5	2.5	2.5	2.5
Heat Radiation:							
Engine		kW	26	32	38	44	50
Generator		kW	(See separat data from generator maker)				

The stated heat dissipation, capacities of gas and engine-driven pumps are given at 720 RPM. Heat dissipation gas and pump capacities at 750 RPM are 4% higher than stated. If L.T. cooling are sea water, the L.T. inlet is 32° C instead of 36°C.

Based on tropical conditions, except for exhaust flow and air consumption which are based on ISO conditions.

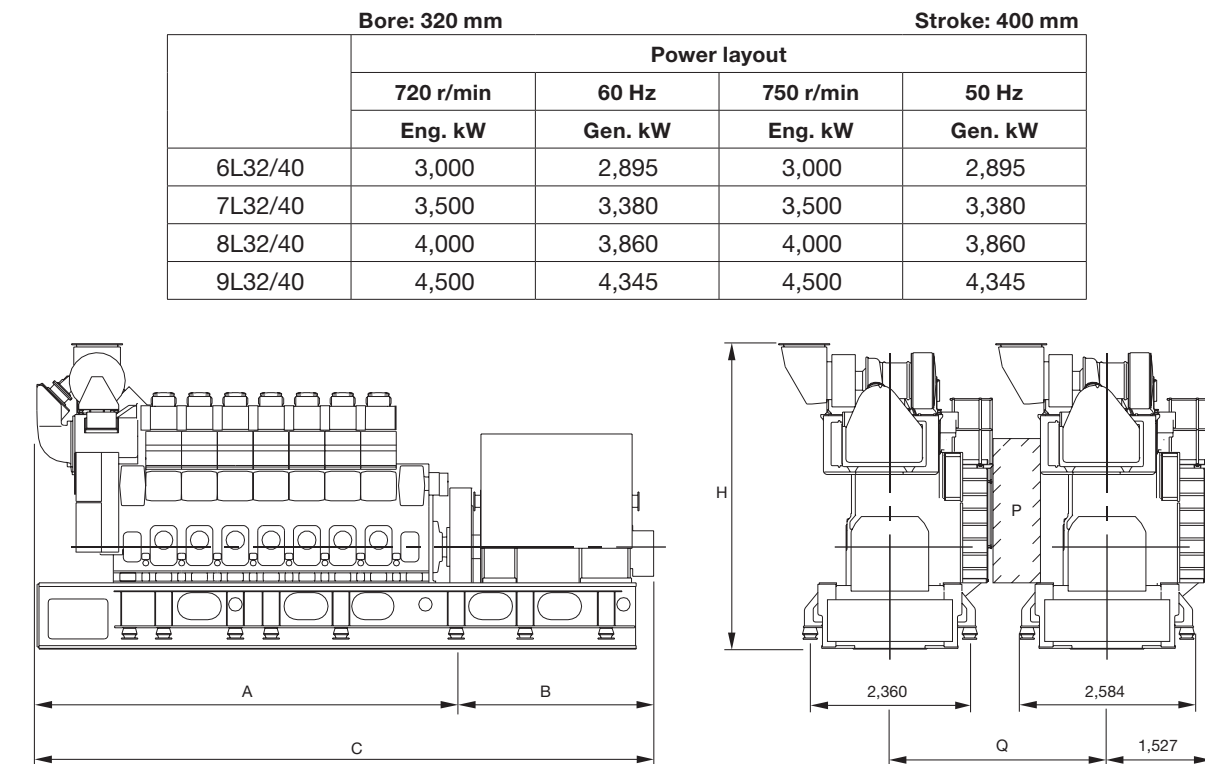
* Only valid for engines equipped with internal basic cooling water system nos. 1 and 2.

** Only valid for engines equipped with combined coolers, internal basic cooling water system no. 3.

*** To compensate for built on pumps, ambient condition, calorific value and adequate circulations flow. The ISO fuel oil consumption is multiplied by 1.45.

Fig. 4.10.02: List of capacities for L28/32H, IMO Tier I.

L32/40 GenSet Data



178 23 10-2.0

No of Cyls.	A (mm)	* B (mm)	* C (mm)	H (mm)	**Dry weight GenSet (t)
6 (720 r/min)	6,340	3,415	9,755	4,510	75.0
6 (750 r/min)	6,340	3,415	9,755	4,510	75.0
7 (720 r/min)	6,870	3,415	10,285	4,510	79.0
7 (750 r/min)	6,870	3,415	10,285	4,510	79.0
8 (720 r/min)	7,400	3,635	11,035	4,780	87.0
8 (750 r/min)	7,400	3,635	11,035	4,780	87.0
9 (720 r/min)	7,930	3,635	11,565	4,780	91.0
9 (750 r/min)	7,930	3,635	11,565	4,780	91.0

P Free passage between the engines, width 600 mm and height 2,000 mm

Q Min. distance between engines: 2,835 mm (without gallery) and 3,220 mm (with gallery)

* Depending on alternator

** Weight includes an alternator, Type B16, Make Siemens

All dimensions and masses are approximate and subject to change without prior notice.

178 34 55-7.3

Fig. 4.11.01: Power and outline of 32/40

L32/40 GenSet Data

500 kW/cyl

		Cyl.	6	7	8	9
Max continues rating at:	720 RPM	kW	3,000	3,500	4,000	4,500
	750 RPM	kW	3,000	3,500	4,000	4,500
Engine driven pumps:						
LT cooling water pump	4.5 bar	m ³ /h	70	70	140	140
HT cooling water pump	4.5 bar	m ³ /h	70	70	70	70
Lubricating oil main pump	8 bar	m ³ /h	115	115	135	135
Pre-lubrication oil pump	1.5 bar	m ³ /h	21	21	27	34
Separate pumps:						
Diesel oil pump	(4 bar at fuel oil inlet A1)	m ³ /h	1.99	2.32	2.65	2.98
Fuel oil supply pump	(4 bar discharge pressure)	m ³ /h	0.97	1.14	1.30	1.46
Fuel oil circulating pump	(8 bar at fuel oil inlet A1)	m ³ /h	2.01	2.35	2.68	3.02
Fuel nozzle pump	3 bar	m ³ /h	1,0	1,2	1,4	1,6
LT cooling water pump	3 bar	m ³ /h	57	70	74	85
HT cooling water pump	4.3 bar	m ³ /h	42	49	56	63
Cooling capacity:						
LT charge air		kW	379	442	517	581
Lubrication oil engine		kW	456	532	608	684
Lub. Seperator heat		kW	25	29	33	38
Total Lub. Oil heat		kW	481	561	641	721
Total heat dissipated LT side incl. Heat from Lub. Seperator		kW	860	1,003	1,158	1,303
LT flow at 36°C inlet engine		m ³ /h	57	70	74	85
Lub. Oil		m ³ /h	100 + z	110 + z	120 + z	130 + z
HT charge air		kW	774	871	1011	1105
Jacket cooling		kW	436	508	581	654
Total heat from HT side		kW	1210	1380	1592	1759
HT temp. Inlet engine		°C	60	60	60	61
HT flow at 85°C outlet engine		m ³ /h	42	49	56	63
Nozzel cooling		kW	12	14	16	18
Gas Data:						
Air consumption		kg/h	21,600	25,200	28,800	32,400
Exhaust gas flow		kg/h	22,200	25,900	29,600	33,300
Exhaust gas temperature at turbine outlet		°C	336	336	336	336
Starting air system:						
Air consumption per start incl. Air for jet assist		Nm ³	2,4	2,5	3,6	3,7
Heat ratiation:						
Engine		kW	109	127	145	164
Alternator		kW	(See separate data from alternator maker)			

The stated heat balances are based on 100% load and tropical condition.
The mass flows and exhaust gas temperature are based on ISO ambient condition.

Pump capacities of engine-driven pumps at 750 RPM are 4% higher than stated.

z = Flushing oil of automatic filter.

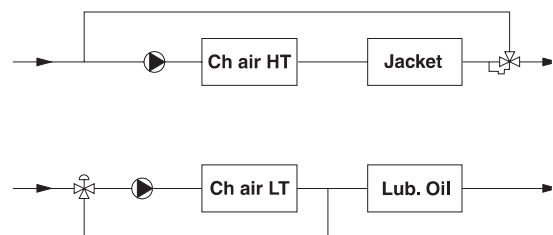


Fig. 4.11.02: List of capacities for L32/40, IMO Tier I.

178 23 11-4.0

Installation Aspects

5

Space Requirements and Overhaul Heights

The latest version of most of the drawings of this section is available for download at www.mandiesel.com under 'Marine' → 'Low Speed' → 'Installation Drawings'. First choose engine series, then engine type and select from the list of drawings available for download.

Space Requirements for the Engine

The space requirements stated in Section 5.02 are valid for engines rated at nominal MCR (L_1).

The additional space needed for engines equipped with PTO is stated in Chapter 4.

If, during the project stage, the outer dimensions of the turbocharger seem to cause problems, it is possible, for the same number of cylinders, to use turbochargers with smaller dimensions by increasing the indicated number of turbochargers by one, see Chapter 3.

Overhaul of Engine

The distances stated from the centre of the crankshaft to the crane hook are for the normal lifting procedure and the reduced height lifting procedure (involving tilting of main components). The lifting capacity of a normal engine room crane can be found in Fig. 5.04.01.

The area covered by the engine room crane shall be wide enough to reach any heavy spare part required in the engine room.

A lower overhaul height is, however, available by using the MAN B&W Double-Jib crane, built by Danish Crane Building A/S, shown in Figs. 5.04.02 and 5.04.03.

Please note that the distance 'E' in Fig. 5.02.01, given for a double-jib crane is from the centre of the crankshaft to the lower edge of the deck beam.

A special crane beam for dismantling the turbocharger must be fitted. The lifting capacity of the crane beam for dismantling the turbocharger is stated in Section 5.03.

The overhaul tools for the engine are designed to be used with a crane hook according to DIN 15400, June 1990, material class M and load capacity 1Am and dimensions of the single hook type according to DIN 15401, part 1.

The total length of the engine at the crankshaft level may vary depending on the equipment to be fitted on the fore end of the engine, such as adjustable counterweights, tuning wheel, moment compensators or PTO.

Space Requirements

This section is available on request

Crane beam for overhaul of turbocharger

For the overhaul of a turbocharger, a crane beam with trolleys is required at each end of the turbocharger.

Two trolleys are to be available at the compressor end and one trolley is needed at the gas inlet end.

Crane beam no. 1 is for dismantling of turbocharger components.

Crane beam no. 2 is for transporting turbocharger components.

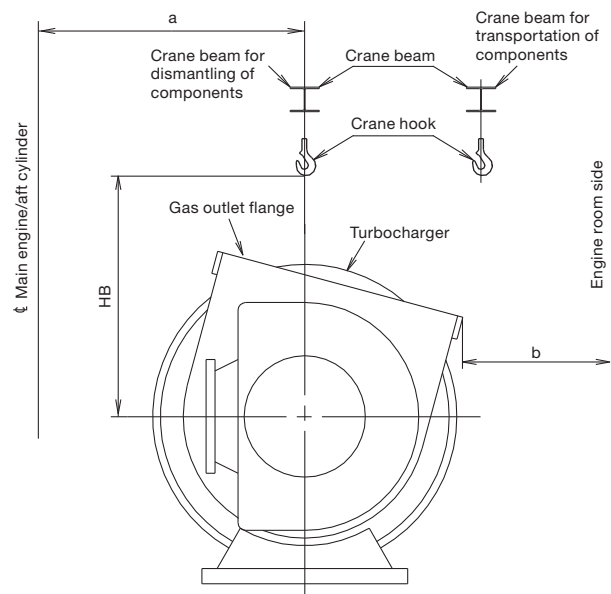
See Figs. 5.03.01a and 5.03.02.

The crane beams can be omitted if the main engine room crane also covers the turbocharger area.

The crane beams are used and dimensioned for lifting the following components:

- Exhaust gas inlet casing
- Turbocharger inlet silencer
- Compressor casing
- Turbine rotor with bearings

The crane beams are to be placed in relation to the turbocharger(s) so that the components around the gas outlet casing can be removed in connection with overhaul of the turbocharger(s).



178 52 34-0.1

Fig. 5.03.01a: Required height and distance

The crane beam can be bolted to brackets that are fastened to the ship structure or to columns that are located on the top platform of the engine.

The lifting capacity of the crane beam for the heaviest component 'W', is indicated in Fig. 5.03.01b for the various turbocharger makes. The crane beam shall be dimensioned for lifting the weight 'W' with a deflection of some 5 mm only.

HB indicates the position of the crane hook in the vertical plane related to the centre of the turbocharger. HB and b also specifies the minimum space for dismantling.

For engines with the turbocharger(s) located on the exhaust side, EoD No. 4 59 122, the letter 'a' indicates the distance between vertical centrelines of the engine and the turbocharger.

MAN B&W			
	Units	TCA77	TCA88
W	kg	2,000	3,000
HB	mm	1,800	2,000
b	m	800	1,000

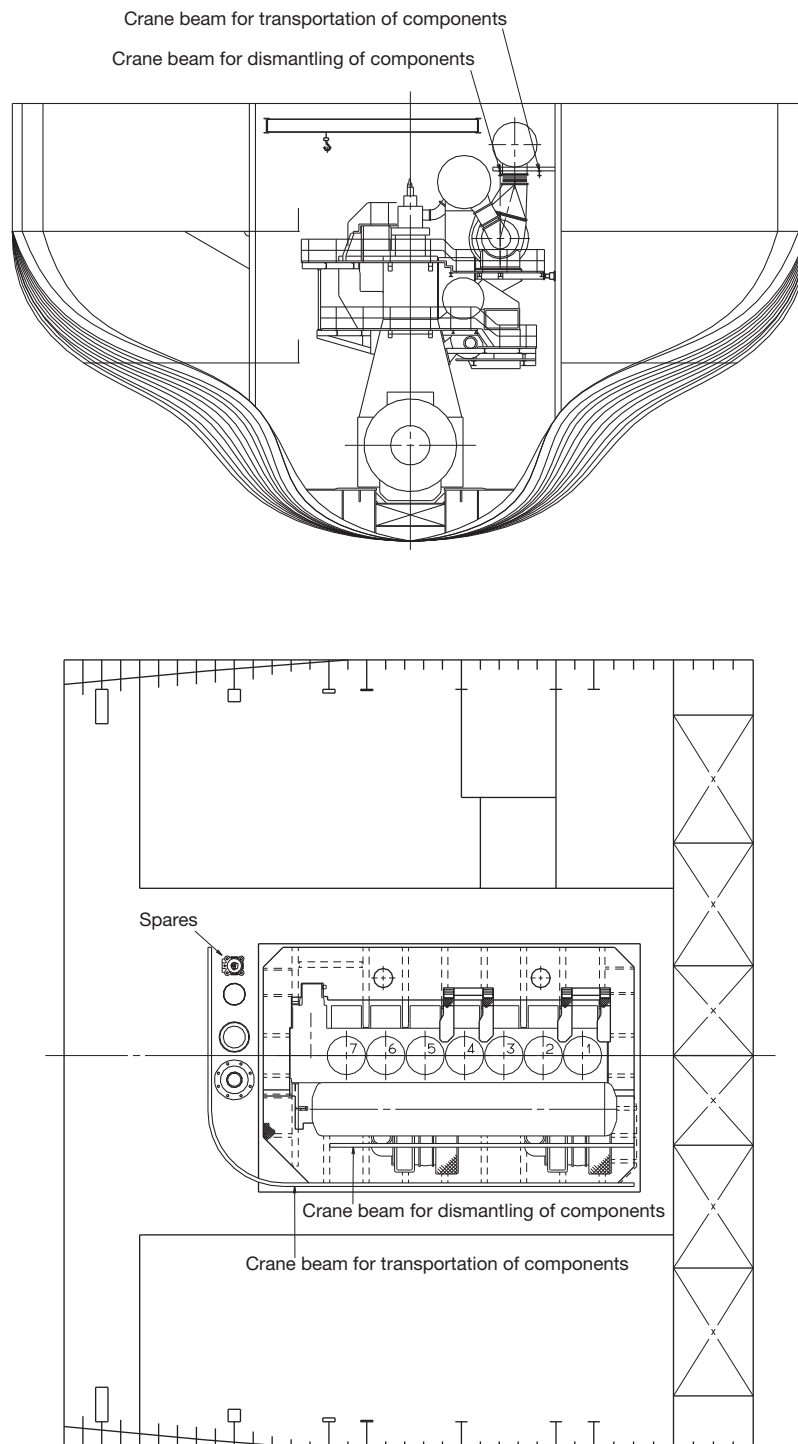
ABB			
	Units	A185	A190
W	kg	Available on request	
HB	mm		
b	m		

Mitsubishi			
	Units	MET83	MET90
W	kg	2,700	3,500
HB	mm	2,200	2,200
b	m	800	800

The figures 'a' are stated on the 'Engine and Gallery Outline' drawing, Section 5.06.

Fig. 5.03.01b: Required height and distance and weight

Crane beam for turbochargers



178 52 74-6.0

Fig. 5.03.02: Crane beam for turbocharger

Crane beam for overhaul of air cooler

Overhaul/exchange of scavenge air cooler.

Valid for air cooler design for the following engines with more than one turbochargers mounted on the exhaust side.

1. Dismantle all the pipes in the area around the air cooler.
2. Dismantle all the pipes around the inlet cover for the cooler.
3. Take out the cooler insert by using the above placed crane beam mounted on the engine.
4. Turn the cooler insert to an upright position.
5. Dismantle the platforms below the air cooler.
6. Lower down the cooler insert between the gallery brackets and down to the engine room floor. Make sure that the cooler insert is supported, e.g. on a wooden support.
7. Move the air cooler insert to an area covered by the engine room crane using the lifting beam mounted below the lower gallery of the engine.
8. By using the engine room crane the air cooler insert can be lifted out of the engine room.

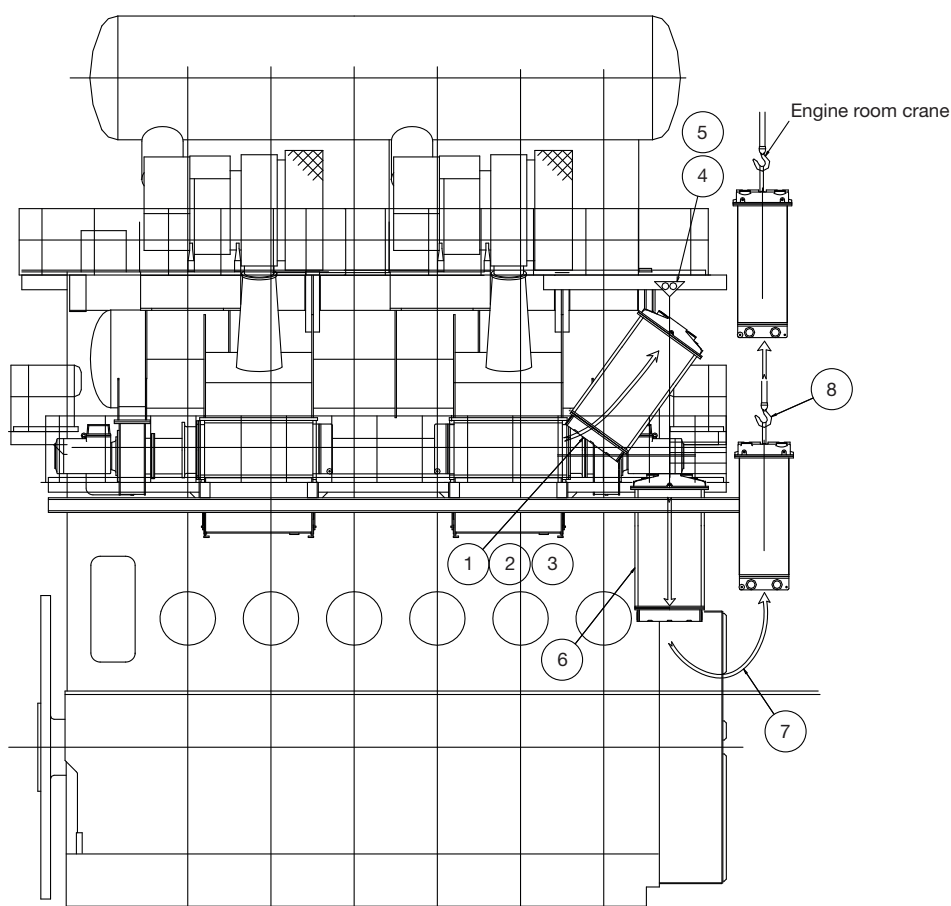
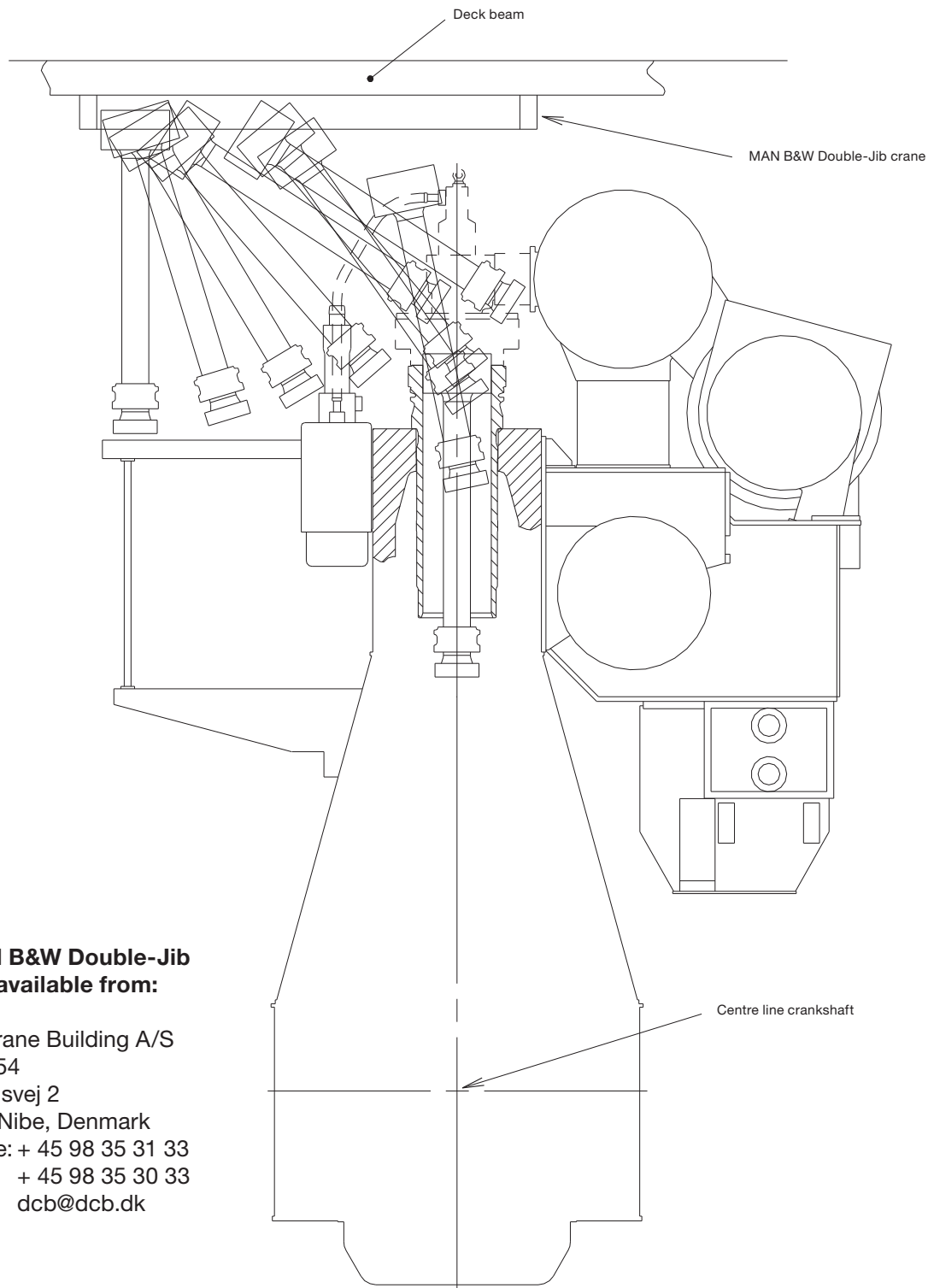


Fig.: 5.03.03: Crane beam for overhaul of air cooler, turbochargers located on exhaust side of the engine

178 52 73-4.0

Engine room crane

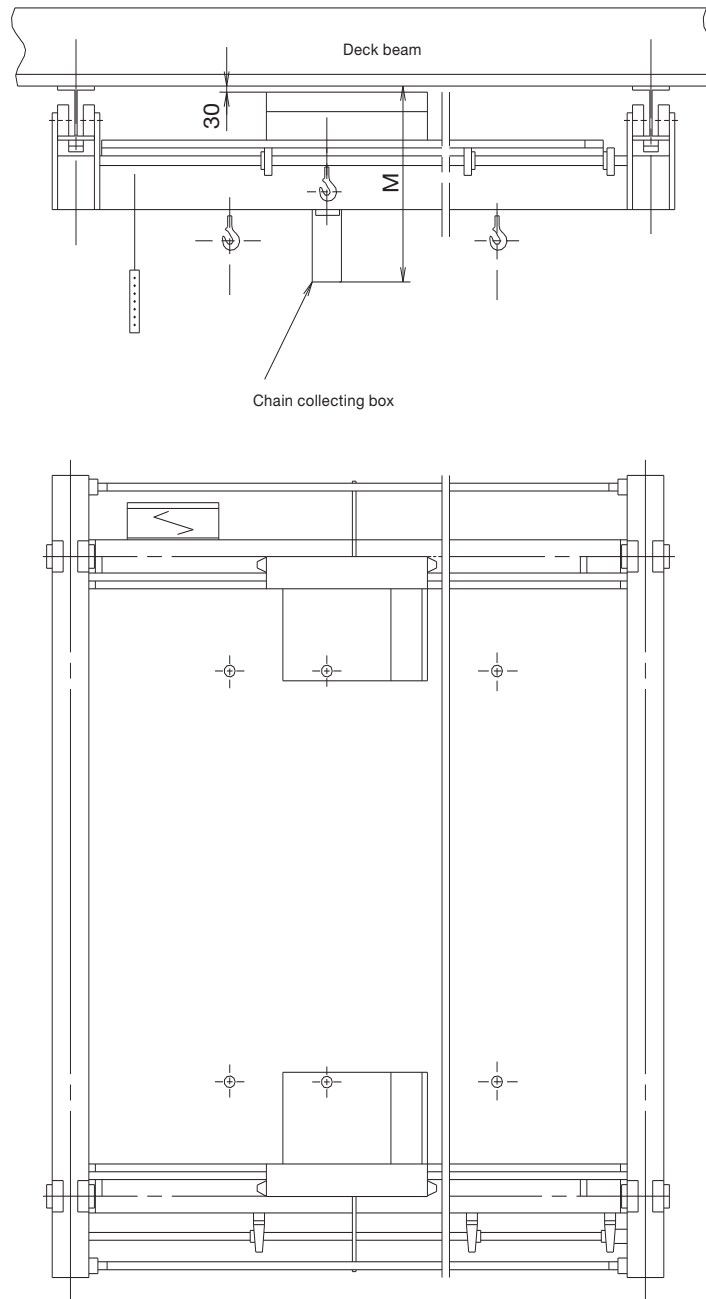
This section is available on request

Overhaul with MAN B&W Double-Jib Crane**The MAN B&W Double-Jib crane is available from:**

Danish Crane Building A/S
P.O. Box 54
Østerlandsvej 2
DK-9240 Nibe, Denmark
Telephone: + 45 98 35 31 33
Telefax: + 45 98 35 30 33
E-mail: dcb@dcb.dk

178 24 86-3.2

Fig. 5.04.02: Overhaul with Double-Jib crane

MAN B&W Double-Jib Crane

178 37 30-1.1

This crane is adapted to the special tool for low overhaul.

Dimensions are available on request.

Fig. 5.04.03: MAN B&W Double-Jib crane, option: 4 88 701

Engine Outline, Galleries and Pipe Connections

Engine outline

The total length of the engine at the crankshaft level may vary depending on the equipment to be fitted on the fore end of the engine, such as adjustable counterweights, tuning wheel, moment compensators or PTO, which are shown as alternatives in Section 5.06

Engine masses and centre of gravity

The partial and total engine masses appear from Section 19.04, 'Dispatch Pattern', to which the masses of water and oil in the engine, Section 5.08, are to be added. The centre of gravity is shown in Section 5.07, in both cases including the water and oil in the engine, but without moment compensators or PTO.

Gallery outline

Section 5.06 show the gallery outline for engines rated at nominal MCR (L1).

Engine pipe connections

The positions of the external pipe connections on the engine are stated in Section 5.09, and the corresponding lists of counterflanges for pipes and turbocharger in Section 5.10.

The flange connection on the turbocharger gas outlet is rectangular, but a transition piece to a circular form can be supplied as an option: 4 60 601.

Engine and Gallery Outline

This section is available on request

Centre of Gravity

This section is available on request

Mass of Water and Oil

No. of cylinders	Mass of water and oil in engine in service					
	Mass of water			Mass of oil		
	Jacket cooling water kg	Scavenge air cooling water kg	Total kg	Engine system kg	Oil pan kg	Total kg
6	1,681	1,390	3,071	2,174	1,826	4,000
7	1,868	1,545	3,413	2,717	2,282	4,999
8	2,139	1,655	3,794	2,885	2,895	5,780
9	2,390	1,803	4,193	3,398	3,060	6,458
10	2,590	2,200	4,790	3,229	3,700	6,929
11	3,052	2,530	5,582	3,874	4,255	8,129
12	3,514	3,062	6,576	4,967	5,666	10,633

Fig. 5.08.01: Water and oil in engine

Engine Pipe Connections

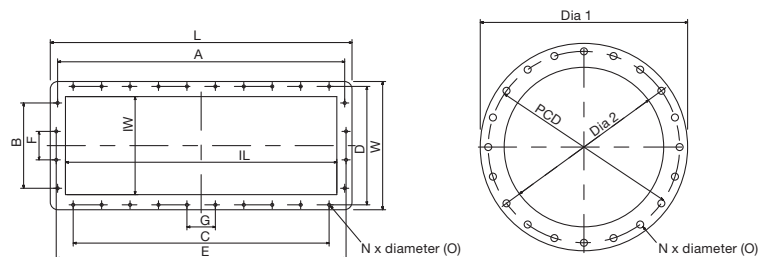
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Counterflanges

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Counterflanges, Connection D

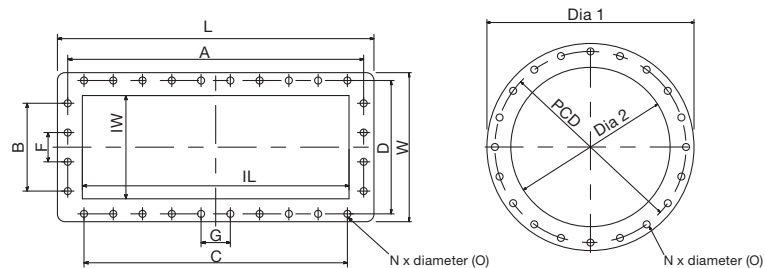
MAN Diesel Type TCA/TCR



Type TCA series - Rectangular type													
T.C.	L	W	IL	IW	A	B	C	D	E	F	G	N	O
TCA44	1,012	430	910	328	962	286	854	-	972	96	122	24	ø13
TCA55	1,206	516	1,080	390	1,143	360	1,000	472	1,155	120	125	26	ø18
TCA66	1,433	613	1,283	463	1,358	420	1,200	560	1,373	140	150	26	ø18
TCA77	1,694	720	1,524	550	1,612	480	1,280	664	1,628	160	160	34	ø22
TCA88	2,012	855	1,810	653	1,914	570	1,710	788	1,934	160	190	28	ø22
TCA99	2,207	938	1,985	717	2,100	624	1,872	866	2,120	208	208	28	ø22

Type TCR series - Round type					
T.C.	Dia 1	Dia 2	PCD	N	O
TCR18	425	310	395	12	ø22
TCR22	595	434	550	16	ø22

ABB Type TPL/A100

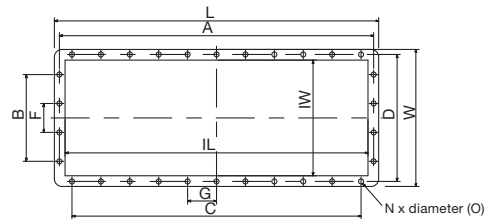


Type TPL - Rectangular type												
T.C.	L	W	IL	IW	A	B	C	D	F	G	N	O
TPL73	1,168	550	984	381	1,092	324	972	492	108	108	28	ø26
TPL77	1,372	638	1,176	462	1,294	390	1,170	580	130	130	28	ø26
TPL80	1,580	729	1,364	538	1,494	450	1,350	668	150	150	28	ø30
TPL85	1,910	857	1,740	690	1,812	700	1,540	796	140	140	36	ø30
TPL91	2,226	958	2,006	770	2,134	625	1,875	896	125	125	48	ø22

Type TPL - Round type					
T.C.	Dia 1	Dia 2	PCD	N	O
TPL69	650	500	600	20	ø22
TPL65	540	400	495	16	ø22

Type A100 series					
T.C.	Dia 1	Dia 2	PCD	N	O
A165	Available on request				
A170					
A175					
A180					
A185					
A190					

MHI Type MET



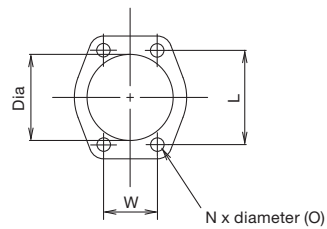
Type MET												
T.C.	L	W	IL	IW	A	B	C	D	F	G	N	O
MET33MA	Available on request											
MET42MA	883	365	793	275	850	240	630	335	80	90	24	ø15
MET53MA	1,122	465	1,006	349	1,073	300	945	420	100	105	28	ø20
MET60MA	1,230	660	1,120	388	1,190	315	1,050	500	105	105	30	ø20
MET66MA	1,380	560	1,254	434	1,330	345	1,200	510	115	120	30	ø24
MET71MA	1,520	700	1,400	480	1,475	345	1,265	640	115	115	34	ø20
MET83MA	1,740	700	1,586	550	1,680	450	1,500	640	150	150	30	ø24
MET90MA	1,910	755	1,750	595	1,850	480	1,650	695	160	165	30	ø24

503 26 38-6.0.1

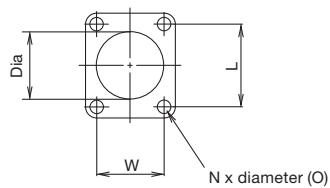
Fig. 5.10.02: Turbocharger, exhaust outlet

Counterflanges, Connection E

MAN Diesel Type TCA

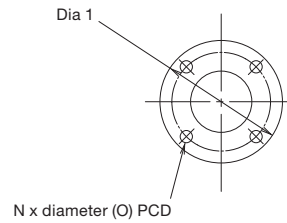


Type TCA series						
T.C.	Dia	L	W	N	O	Thickness of flanges
TCA77	116	126	72	4	20	18
TCA88	141.5	150	86	4	20	18
TCA99	141.5	164	94	4	22	24



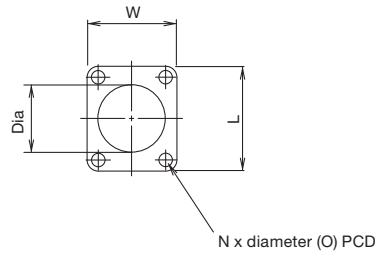
Type TCA series						
T.C.	Dia	L	W	N	O	Thickness of flanges
TCA55	77.5	86	76	4	16	15
TCA66	90.5	110	90	4	18	16

ABB Type TPL

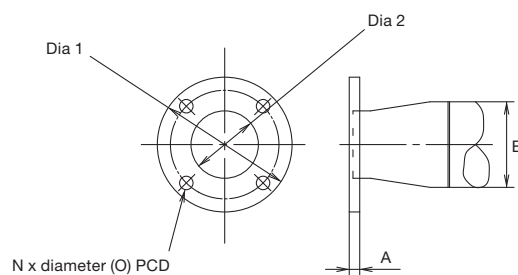


Type TPL series					
T.C.	Dia 1	PCD	N	O	Thickness of flanges
TPL65B	165	125	4	18	18
TPL69B	185	145	4	18	18
TPL73B11/12/13	185	145	4	18	18
TPL77B11/12/13	185	145	4	18	18
TPL80B11/12/13	200	160	8	18	20
TPL85B11/12/13	200	165	8	19	16
TPL85B14/15/16	200	160	8	16	14
TPL91B	210	175	8	18	19

MHI Type MET



Type MET series							
T.C.	Dia	PCD	L	W	N	O	Thickness of flanges
MET33MA	43.5	95	95	95	4	14	12
MET42MA	61.5	105	105	105	4	14	14
MET53MA	77	130	125	125	4	14	14
MET60MA	90	145	140	140	4	18	14
MET66MA	90	145	140	140	4	18	14
MET71MA	90	145	140	140	4	18	14
MET90MA	115	155	155	155	4	18	14



Type MET series - Round type							
T.C.	Dia 1	Dia 2	PCD	B	N	O	Thickness of flanges (A)
MET83MA	180	90	145	114.3	4	18	14

Fig. 5.10.03: Venting of lubricating oil discharge pipe for turbochargers

Engine Seating and Holding Down Bolts

The latest version of most of the drawings of this section is available for download at www.mandiesel.com under 'Marine' → 'Low Speed' → 'Installation Drawings'. First choose engine series, then engine type and select 'Engine seating' in the general section of the list of drawings available for download.

Engine seating and arrangement of holding down bolts

The dimensions of the seating stated in Figs. 5.12.01 and 5.12.02 are for guidance only.

The engine is designed for mounting on epoxy chocks, EoD: 4 82 102, in which case the underside of the bedplate's lower flanges has no taper.

The epoxy types approved by MAN Diesel are:

- 'Chockfast Orange PR 610 TCF' from ITW Philadelphia Resins Corporation, USA
- 'Durasin' from Daemmstoff Industrie Korea Ltd
- 'Epocast 36' from H.A. Springer - Kiel, Germany.

Epoxy Chocks Arrangement

This section is available on request

Engine Seating Profile

This section is available on request

Engine Top Bracing

The so-called guide force moments are caused by the transverse reaction forces acting on the cross-heads due to the connecting rod and crankshaft mechanism. When the piston of a cylinder is not exactly in its top or bottom position the gas force from the combustion, transferred through the connecting rod, will have a component acting on the crosshead and the crankshaft perpendicularly to the axis of the cylinder. Its resultant is acting on the guide shoe and together they form a guide force moment.

The moments may excite engine vibrations moving the engine top athwart ships and causing a rocking (excited by H-moment) or twisting (excited by X-moment) movement of the engine. For engines with less than seven cylinders, this guide force moment tends to rock the engine in the transverse direction, and for engines with seven cylinders or more, it tends to twist the engine.

The guide force moments are harmless to the engine except when resonance vibrations occur in the engine/double bottom system. They may, however, cause annoying vibrations in the superstructure and/or engine room, if proper counter-measures are not taken.

As a detailed calculation of this system is normally not available, MAN Diesel recommends that top bracing is installed between the engine's upper platform brackets and the casing side.

However, the top bracing is not needed in all cases. In some cases the vibration level is lower if the top bracing is not installed. This has normally to be checked by measurements, i.e. with and without top bracing.

If a vibration measurement in the first vessel of a series shows that the vibration level is acceptable without the top bracing, we have no objection to the top bracing being removed and the rest of the series produced without top bracing. It is our experience that especially the 7-cylinder engine will often have a lower vibration level without top bracing.

Without top bracing, the natural frequency of the vibrating system comprising engine, ship's bottom, and ship's side is often so low that resonance with the excitation source (the guide force moment) can occur close to the normal speed range, resulting in the risk of vibration.

With top bracing, such a resonance will occur above the normal speed range, as the natural frequencies of the double bottom/main engine system will increase. The impact of vibration is thus lowered.

The top bracing is normally installed on the exhaust side of the engine, but can alternatively be installed on the manoeuvring side. A combination of exhaust side and manoeuvring side installation is also possible.

The top bracing system is installed either as a mechanical top bracing or a hydraulic top bracing. Both systems are described below.

Mechanical top bracing

The mechanical top bracing comprises stiff connections between the engine and the hull.

The top bracing stiffener consists of a double bar tightened with friction shims at each end of the mounting positions. The friction shims allow the top bracing stiffener to move in case of displacements caused by thermal expansion of the engine or different loading conditions of the vessel. Furthermore, the tightening is made with a well-defined force on the friction shims, using disc springs, to prevent overloading of the system in case of an excessive vibration level.

The mechanical top bracing is to be made by the shipyard in accordance with MAN Diesel instructions.

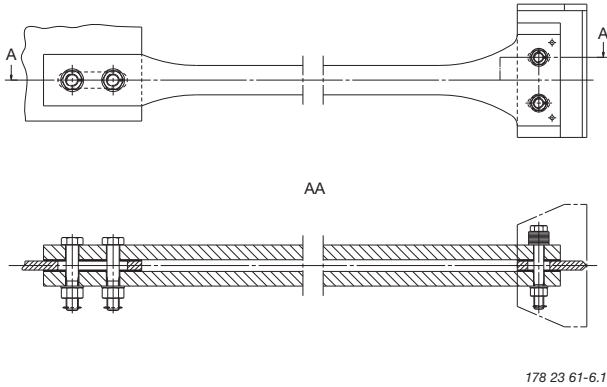


Fig. 5.13.01: Mechanical top bracing stiffener.
Option: 4 83 112

Hydraulic top bracing

The hydraulic top bracing is an alternative to the mechanical top bracing used mainly on engines with a cylinder bore of 50 or more. The installation normally features two, four or six independently working top bracing units.

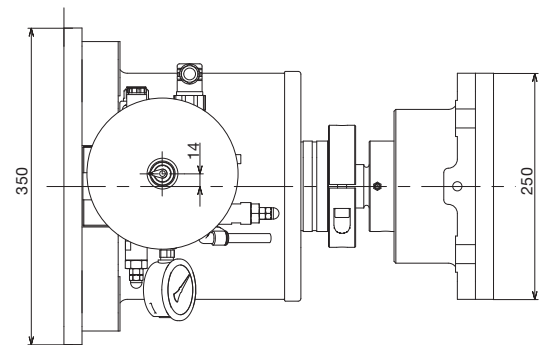
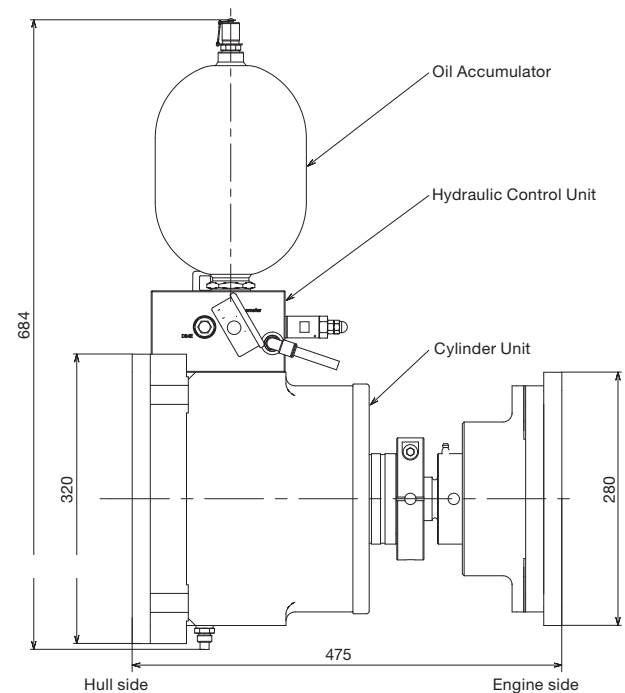
The top bracing unit consists of a single-acting hydraulic cylinder with a hydraulic control unit and an accumulator mounted directly on the cylinder unit.

The top bracing is controlled by an automatic switch in a control panel, which activates the top bracing when the engine is running. It is possible to programme the switch to choose a certain rpm range, at which the top bracing is active. For service purposes, manual control from the control panel is also possible.

When active, the hydraulic cylinder provides a pressure on the engine in proportion to the vibration level. When the distance between the hull and engine increases, oil flows into the cylinder under pressure from the accumulator. When the distance decreases, a non-return valve prevents the oil from flowing back to the accumulator, and the pressure rises. If the pressure reaches a preset maximum value, a relief valve allows the oil to flow back to the accumulator, hereby maintaining the force on the engine below the specified value.

By a different pre-setting of the relief valve, the top bracing is delivered in a low-pressure version (26 bar) or a high-pressure version (40 bar).

The top bracing unit is designed to allow displacements between the hull and engine caused by thermal expansion of the engine or different loading conditions of the vessel.



178 57 48-8.0

Fig. 5.13.02: Outline of a hydraulic top bracing unit.
The unit is installed with the oil accumulator pointing either up or down. Option: 4 83 123

Mechanical Top Bracing

This section is available on request

Hydraulic Top Bracing Arrangement

This section is available on request

Components for Engine Control System

Installation of ECS in the Engine Control Room

The following items are to be installed in the ECR (Engine Control Room):

- 2 pcs EICU (Engine Interface Control Unit)
(1 pcs only for ME-B engines)
- 1 pcs MOP (Main Operating Panel)
Touch display, 15"
PC unit
- 1 pcs Track ball for MOP
- 1 pcs PMI system
Display, 19"
PC unit
- 1 pcs Back-up MOP
Display, 15"
PC unit
Keyboard
- 1 pcs Printer
- 1 pcs Ethernet Hub

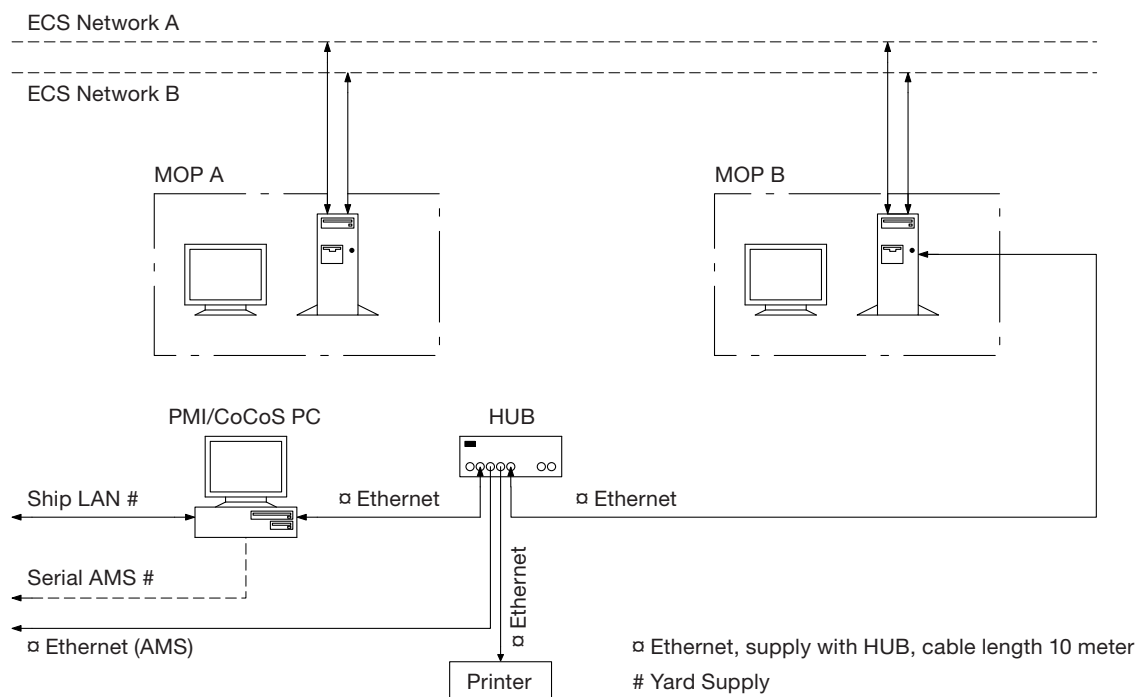
The EICU functions as an interface unit to ECR related systems such as AMS (Alarm and Monitoring System), RCS (Remote Control System) and Safety System. On ME-B engines the EICU also controls the HPS.

The MOP is the operator's interface to the ECS. From there the operator can control and see status of the engine and the ECS. The MOP is a PC with a flat touch screen.

The Back-up MOP consists of a PC unit with keyboard and display and serves as a back-up in case the MOP should break down.

The PMI offline system is equipped with a standard PC. The PMI system serves as a pressure analyse system. See Section 18.02.

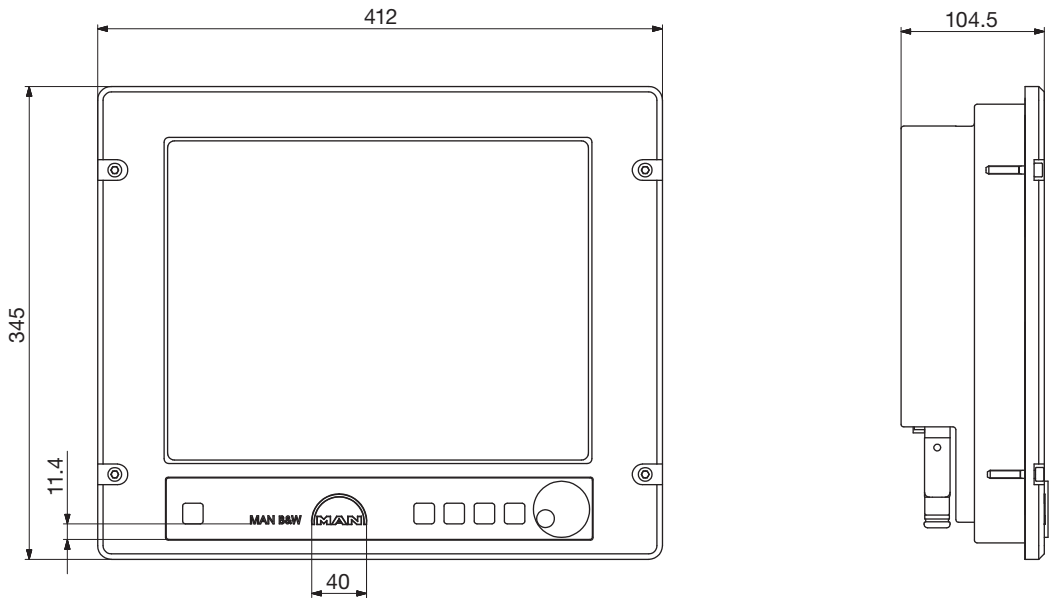
Optional items to be mounted in the ECR include the CoCoS-EDS which can be purchased separately and applied on the PC for the PMI offline system. See Section 18.03.



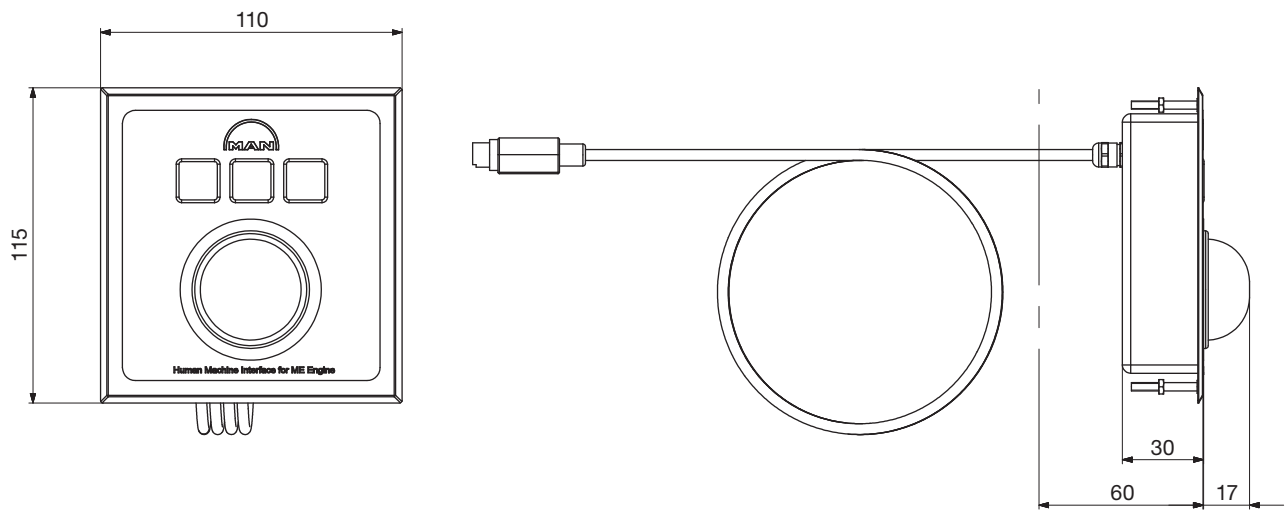
178 57 50-3.0

Fig. 5.16.01 Network and PC components for the ME/ME-B Engine Control System

MOP (Main Operating Panel)

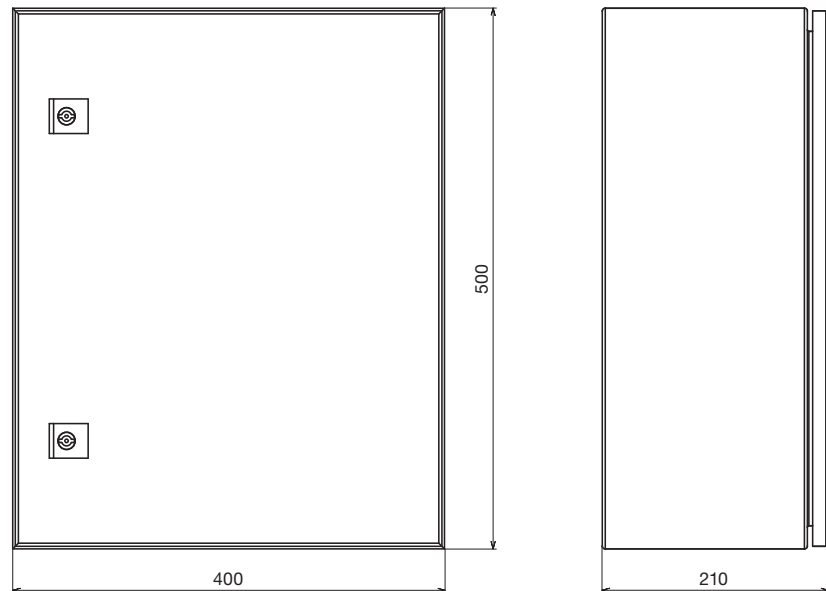
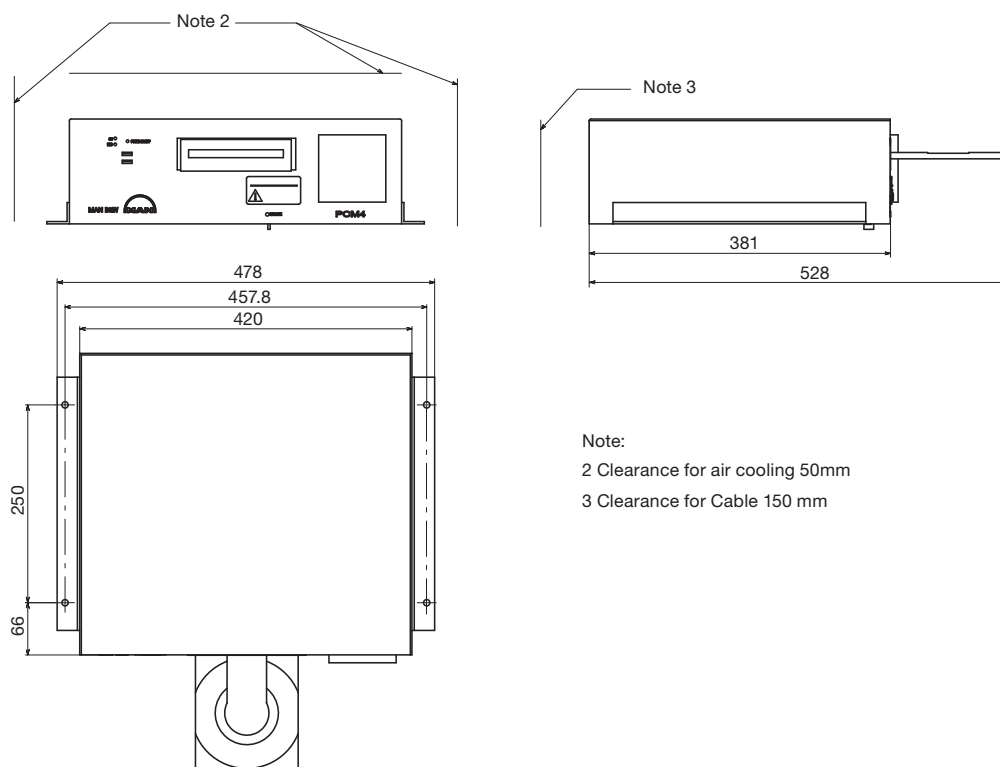


Track ball



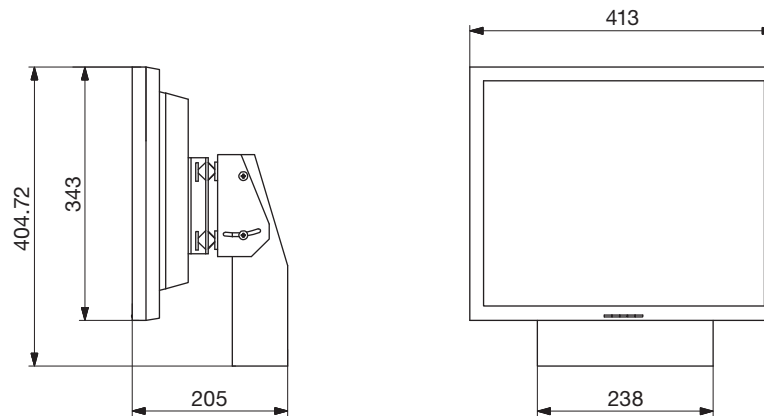
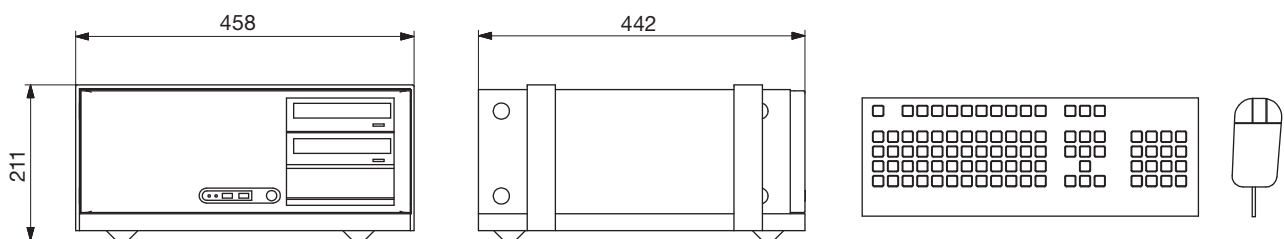
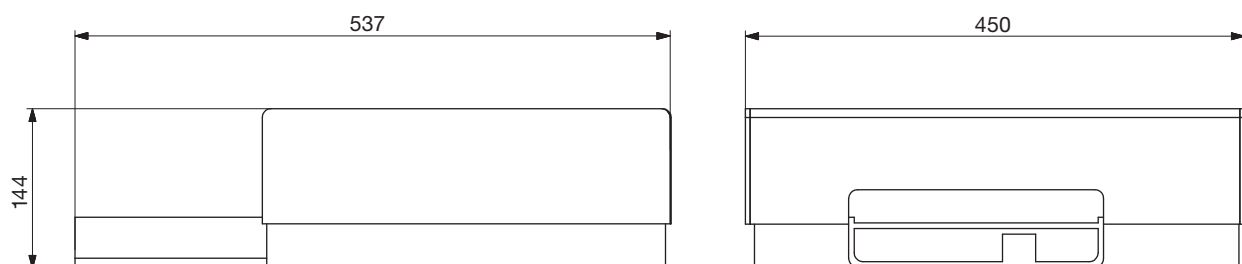
178 57 48-1.0

Fig. 5.16.02 MOP and track ball for the ME/ME-B Engine Control System

EICU (Engine Interface Control Unit) Cabinet**MOP PC unit**

178 50 14-7.1

Fig. 5.16.03 The EICU cabinet and MOP PC unit for the ME/ME-B Engine Control System

PC parts for PMI/CoCoS**19" Display****PC unit****Printer**

178 57 49-3.0

Fig. 5.16.04 PMI/CoCoS PC unit, display and printer for the ME/ME-B Engine Control System

Shaftline Earthing Device

Scope and field of application

A difference in the electrical potential between the hull and the propeller shaft will be generated due to the difference in materials and to the propeller being immersed in sea water.

In some cases, the difference in the electrical potential has caused spark erosion on the thrust, main bearings and journals of the crankshaft of the engine.

In order to reduce the electrical potential between the crankshaft and the hull and thus prevent spark erosion, a highly efficient shaftline earthing device must be installed.

The shaftline earthing device should be able to keep the electrical potential difference below 50 mV DC. A shaft-to-hull monitoring equipment with a mV-meter and with an output signal to the alarm system must be installed so that the potential and thus the correct function of the shaftline earthing device can be monitored.

Note that only one shaftline earthing device is needed in the propeller shaft system.

Design description

The shaftline earthing device consists of two silver slip rings, two arrangements for holding brushes including connecting cables and monitoring equipment with a mV-meter and an output signal for alarm.

The slip rings should be made of solid silver or back-up rings of cobber with a silver layer all over. The expected life span of the silver layer on the slip rings should be minimum 5 years.

The brushes should be made of minimum 80% silver and 20% graphite to ensure a sufficient electrical conducting capability.

Resistivity of the silver should be less than $0.1\mu\text{ Ohm} \times \text{m}$. The total resistance from the shaft to the hull must not exceed 0.001 Ohm.

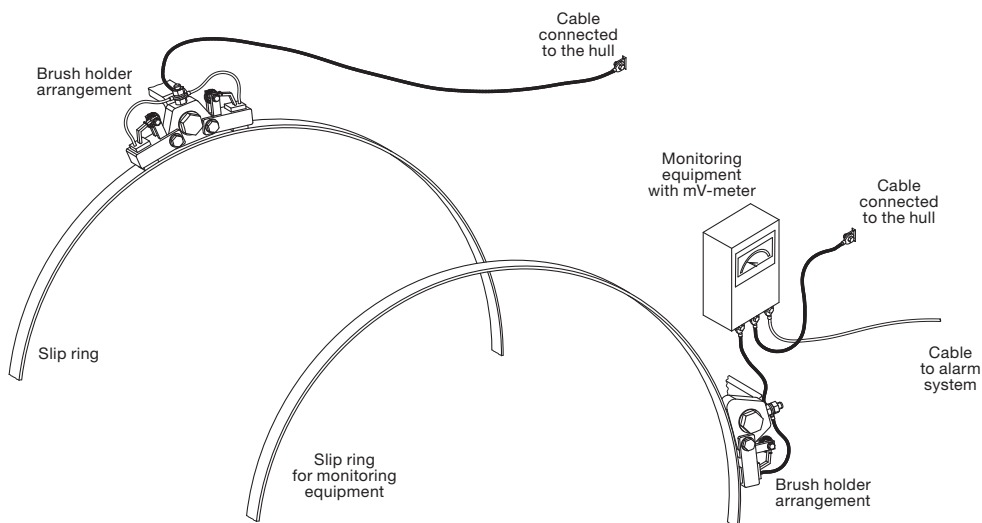
Cabling of the shaftline earthing device to the hull must be with a cable with a cross section not less than 45 mm^2 . The length of the cable to the hull should be as short as possible.

Monitoring equipment should have a 4-20 mA signal for alarm and a mV-meter with a switch for changing range. Primary range from 0 to 50 mV DC and secondary range from 0 to 300 mV DC.

When the shaftline earthing device is working correctly, the electrical potential will normally be within the range of 10-50 mV DC depending of propeller size and revolutions.

The alarm set-point should be 80 mV for a high alarm. The alarm signals with an alarm delay of 30 seconds and an alarm cut-off, when the engine is stopped, must be connected to the alarm system.

Connection of cables is shown in the sketch, see Fig. 5.17.01.

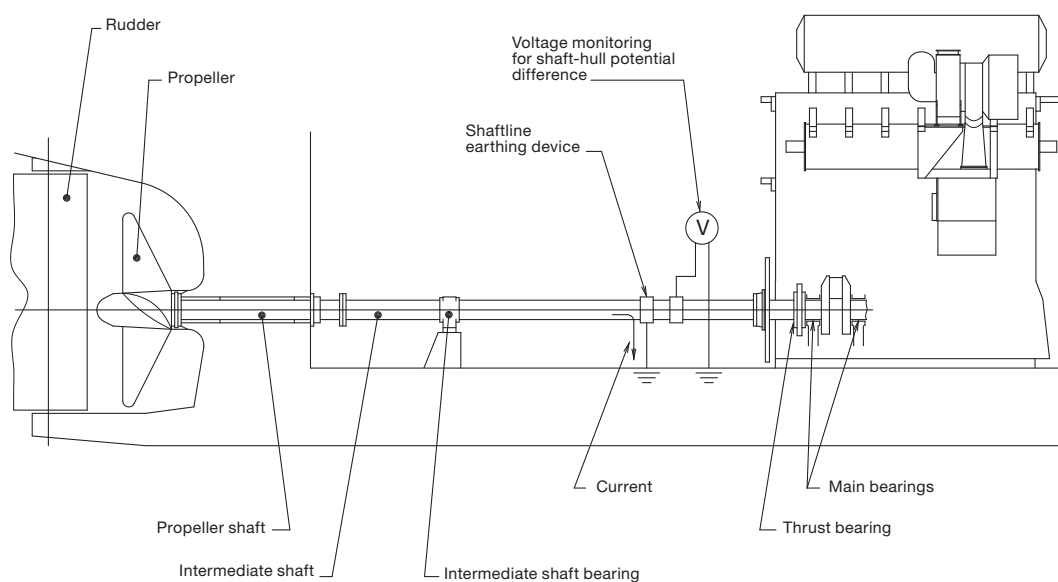


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Fig. 5.17.01: Connection of cables for the shaftline earthing device

Shaftline earthing device installations

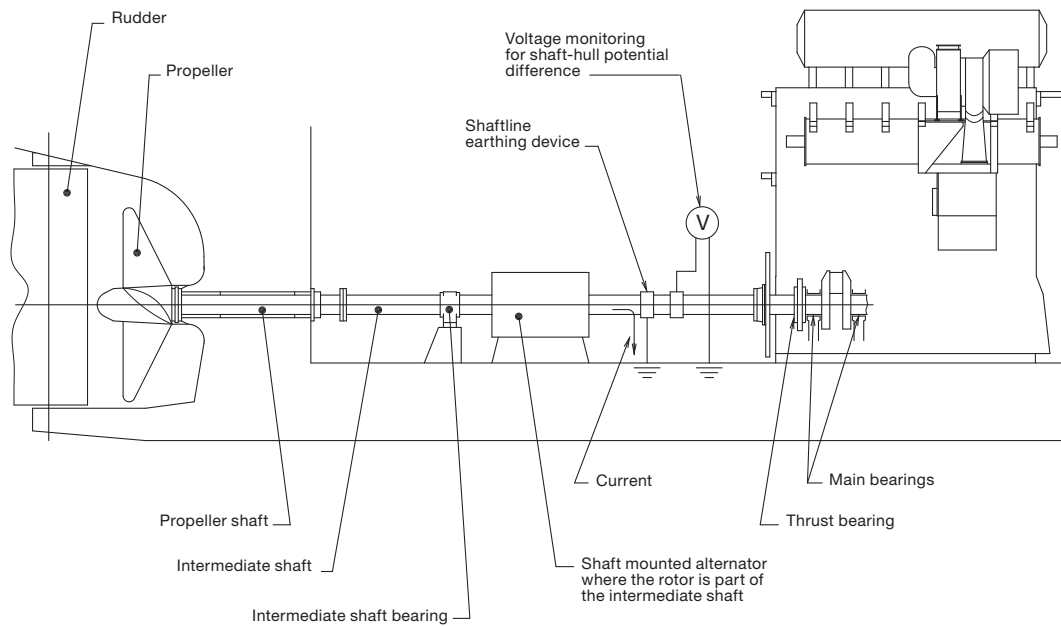
The shaftline earthing device slip rings must be mounted on the foremost intermediate shaft as close to the engine as possible, see Fig. 5.17.02



079 21 82-1.3.2.0

Fig. 5.17.02: Installation of shaftline earthing device in an engine plant without shaft-mounted generator

When a generator is fitted in the propeller shaft system, where the rotor of the generator is part of the intermediate shaft, the shaftline earthing device must be mounted between the generator and the engine, see Fig. 5.17.03



079 21 82-1.3.3.0

Fig. 5.17.03: Installation of shaftline earthing device in an engine plant with shaft-mounted generator

MAN Diesel's Alpha Controllable Pitch Propeller and Alphasonic Propulsion Control

This section is not applicable

List of Capacities: Pumps, Coolers & Exhaust Gas

Calculation of List of Capacities and Exhaust Gas Data

Updated engine and capacities data is available from the CEAS program on www.mandiesel.com under 'Marine' → 'Low speed' → 'CEAS Engine Room Dimensions'.

This chapter describes the necessary auxiliary machinery capacities to be used for a nominally rated engine. The capacities given are valid for seawater cooling system and central cooling water system, respectively. For derated engine, i.e. with a specified MCR and/or matching point different from the

nominally rated MCR point, the list of capacities will be different from the nominal capacities.

Furthermore, among others, the exhaust gas data depends on the ambient temperature conditions.

Based on examples for a derated engine, the way of how to calculate the derated capacities, freshwater production and exhaust gas amounts and temperatures will be described in details.

Nomenclature

In the following description and examples of the auxiliary machinery capacities, freshwater generator production and exhaust gas data, the below nomenclatures are used:

Engine ratings	Point / Index	Power	Speed
Nominal MCR point	L_1	P_{L1}	n_{L1}
Specified MCR point	M	P_M	n_M
Matching point	O	P_O	n_O
Service point	S	P_S	n_S

Fig. 6.01.01: Nomenclature of basic engine ratings

Parameters	Cooler index	Flow index
Q = Heat dissipation	air scavenge air cooler	sw seawater flow
V = Volume flow	lub lube oil cooler	cw cooling/central water flow
M = Mass flow	jw jacket water cooler	exh exhaust gas
T = Temperature	cent central cooler	fw freshwater

Fig. 6.01.02: Nomenclature of coolers and volume flows, etc.

Engine configurations related to SFOC

K98ME/ME-C, S90ME-C, K90ME/ME-C, S80ME-C, K80ME-C, S70ME-C/ME-GI, L70ME-C, S65ME-C/ME-GI, S60ME-C/ME-GI, L60ME-C, S50ME-C, S60ME-B, S50ME-B

The engine type is available in the following version with respect to the efficiency of the turbo-charger alone:

- **A) With high efficiency turbocharger:**
which is the basic design and for which the lists of capacities Section 6.03 are calculated.

For S46ME-B, S40ME-B and S35ME-B

The engine type is available in the following version with respect to the efficiency of the turbo-charger alone:

- **B) With conventional turbocharger:**
Which is the basic design and for which the lists of capacities Section 6.03 are calculated.

For this engine type the matching point O has to be equal to the specified MCR point M.

List of Capacities and Cooling Water Systems

The List of Capacities contain data regarding the necessary capacities of the auxiliary machinery for the main engine only, and refer to a nominally rated engine. Complying with IMO Tier II NO_x limitations.

The heat dissipation figures include 10% extra margin for overload running except for the scavenge air cooler, which is an integrated part of the diesel engine.

Cooling Water Systems

The capacities given in the tables are based on tropical ambient reference conditions and refer to engines with high efficiency/conventional turbo-charger running at nominal MCR (L_n) for:

- **Seawater cooling system,**
See diagram, Fig. 6.02.01 and nominal capacities in Fig. 6.03.01
- **Central cooling water system,**
See diagram, Fig. 6.02.02 and nominal capacities in Fig. 6.03.01

The capacities for the starting air receivers and the compressors are stated in Fig. 6.03.01.

Heat radiation and air consumption

The radiation and convection heat losses to the engine room is around 1% of the engine nominal power (kW in L_n).

The air consumption is approximately 98.2% of the calculated exhaust gas amount, ie.

$$M_{\text{air}} = M_{\text{exh}} \times 0.982.$$

Flanges on engine, etc.

The location of the flanges on the engine are shown in: 'Engine pipe connections', and the flanges are identified by reference letters stated in the 'List of flanges'; both can be found in Chapter 5.

The diagrams use the 'Basic symbols for piping', whereas the symbols for instrumentation according to 'ISO 1219-1' and 'ISO 1219-2' and the instrumentation list found in Appendix A.

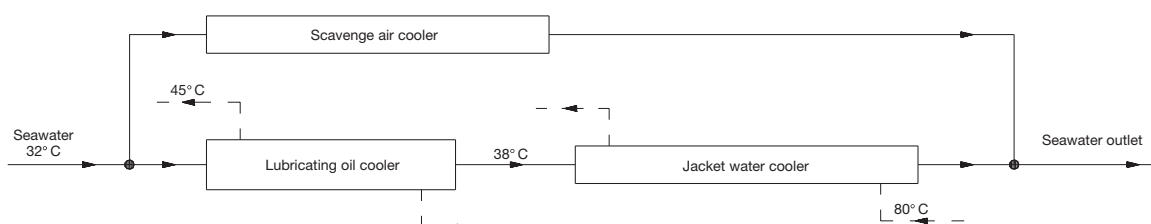


Fig. 6.02.01: Diagram for seawater cooling system

178 11 26-4.1

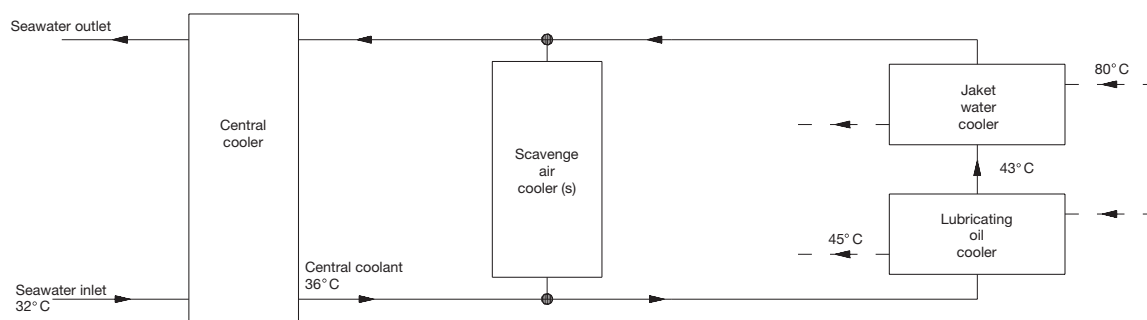


Fig. 6.02.02: Diagram for central cooling water system

178 11 27-6.1

List of Capacities for 6K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			2 x TCA88-21	2 x A185-L35	2 x MET83MA				2 x TCA88-21	2 x A185-L35	2 x MET83MA

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	13.8	13.8	13.8	N.A.	N.A.	N.A.	13.8	13.8	13.8
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	8.6	8.6	8.6	N.A.	N.A.	N.A.	8.6	8.6	8.6
Jacket cooling	m³/h	N.A.	N.A.	N.A.	215.0	215.0	215.0	N.A.	N.A.	N.A.	215.0	215.0	215.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	1070.0	1070.0	1070.0	N.A.	N.A.	N.A.	1090.0	1090.0	1090.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	600.0	590.0	600.0	N.A.	N.A.	N.A.	600.0	590.0	600.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	830	830	830

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	15,180	15,180	15,180	N.A.	N.A.	N.A.	15,100	15,100	15,100
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	491	491	491
Seawater flow	m³/h	N.A.	N.A.	N.A.	722	722	722	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	2,820	2,820	2,850	N.A.	N.A.	N.A.	2,820	2,820	2,850
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	600.0	590.0	600.0	N.A.	N.A.	N.A.	600.0	590.0	600.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	339	339	339
Seawater flow	m³/h	N.A.	N.A.	N.A.	348	348	348	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	4,200	4,200	4,200	N.A.	N.A.	N.A.	4,200	4,200	4,200
Jacket water flow	m³/h	N.A.	N.A.	N.A.	215	215	215	N.A.	N.A.	N.A.	215	215	215
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	339	339	339
Seawater flow	m³/h	N.A.	N.A.	N.A.	348	348	348	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	22,120	22,120	22,150
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	830	830	830
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,090	1,090	1,090

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 13.5	2 x 13.5	2 x 13.5	N.A.	N.A.	N.A.	2 x 13.5	2 x 13.5	2 x 13.5
Compressor cap.	m³	N.A.	N.A.	N.A.	810	810	810	N.A.	N.A.	N.A.	810	810	810

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.0	2 x 7.0	2 x 7.0	N.A.	N.A.	N.A.	2 x 7.0	2 x 7.0	2 x 7.0
Compressor cap.	m³	N.A.	N.A.	N.A.	420	420	420	N.A.	N.A.	N.A.	420	420	420

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	360	360	360	N.A.	N.A.	N.A.	360	360	360
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	318,600	318,600	318,600	N.A.	N.A.	N.A.	318,600	318,600	318,600
Air consumption	kg/h	N.A.	N.A.	N.A.	86.8	86.8	86.8	N.A.	N.A.	N.A.	86.8	86.8	86.8

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01f: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

List of Capacities for 7K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			2 x TCA88-21	2 x A190-L35	2 x MET83MA				2 x TCA88-21	2 x A190-L35	2 x MET83MA

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	16.1	16.1	16.1	N.A.	N.A.	N.A.	16.1	16.1	16.1
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	10.1	10.1	10.1	N.A.	N.A.	N.A.	10.1	10.1	10.1
Jacket cooling	m³/h	N.A.	N.A.	N.A.	250.0	250.0	250.0	N.A.	N.A.	N.A.	250.0	250.0	250.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	1240.0	1250.0	1250.0	N.A.	N.A.	N.A.	1270.0	1270.0	1270.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	680.0	670.0	680.0	N.A.	N.A.	N.A.	680.0	670.0	680.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	960	970	960

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	17,710	17,710	17,710	N.A.	N.A.	N.A.	17,610	17,610	17,610
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	572	572	572
Seawater flow	m³/h	N.A.	N.A.	N.A.	842	842	842	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	3,240	3,300	3,270	N.A.	N.A.	N.A.	3,240	3,300	3,270
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	680.0	670.0	680.0	N.A.	N.A.	N.A.	680.0	670.0	680.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	388	398	388
Seawater flow	m³/h	N.A.	N.A.	N.A.	398	408	408	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	4,900	4,900	4,900	N.A.	N.A.	N.A.	4,900	4,900	4,900
Jacket water flow	m³/h	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	388	398	388
Seawater flow	m³/h	N.A.	N.A.	N.A.	398	408	408	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	25,750	25,810	25,780
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	960	970	960
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,270	1,270	1,270

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 13.5	2 x 13.5	2 x 13.5	N.A.	N.A.	N.A.	2 x 13.5	2 x 13.5	2 x 13.5
Compressor cap.	m³	N.A.	N.A.	N.A.	810	810	810	N.A.	N.A.	N.A.	810	810	810

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5
Compressor cap.	m³	N.A.	N.A.	N.A.	450	450	450	N.A.	N.A.	N.A.	450	450	450

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	420	420	420	N.A.	N.A.	N.A.	420	420	420
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	371,700	371,700	371,700	N.A.	N.A.	N.A.	371,700	371,700	371,700
Air consumption	kg/h	N.A.	N.A.	N.A.	101.3	101.3	101.3	N.A.	N.A.	N.A.	101.3	101.3	101.3

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01g: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

List of Capacities for 8K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			2 x TCA88-21	3 x A185-L34	2 x MET90MA				2 x TCA88-21	3 x A185-L34	2 x MET90MA

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	18.4	18.4	18.4	N.A.	N.A.	N.A.	18.4	18.4	18.4
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	11.5	11.5	11.5	N.A.	N.A.	N.A.	11.5	11.5	11.5
Jacket cooling	m³/h	N.A.	N.A.	N.A.	290.0	290.0	290.0	N.A.	N.A.	N.A.	290.0	290.0	290.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	1420.0	1430.0	1430.0	N.A.	N.A.	N.A.	1450.0	1450.0	1450.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	750.0	750.0	760.0	N.A.	N.A.	N.A.	750.0	750.0	760.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	1,100	1,110	1,100

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	20,230	20,230	20,230	N.A.	N.A.	N.A.	20,130	20,130	20,130
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	654	654	654
Seawater flow	m³/h	N.A.	N.A.	N.A.	963	963	963	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	3,660	3,810	3,760	N.A.	N.A.	N.A.	3,660	3,810	3,760
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	750.0	750.0	760.0	N.A.	N.A.	N.A.	750.0	750.0	760.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	446	456	446
Seawater flow	m³/h	N.A.	N.A.	N.A.	457	467	467	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	5,600	5,600	5,600	N.A.	N.A.	N.A.	5,600	5,600	5,600
Jacket water flow	m³/h	N.A.	N.A.	N.A.	290	290	290	N.A.	N.A.	N.A.	290	290	290
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	446	456	446
Seawater flow	m³/h	N.A.	N.A.	N.A.	457	467	467	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	29,390	29,540	29,490
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,100	1,110	1,100
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,450	1,450	1,450

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 14.0	2 x 14.0	2 x 14.0	N.A.	N.A.	N.A.	2 x 14.0	2 x 14.0	2 x 14.0
Compressor cap.	m³	N.A.	N.A.	N.A.	840	840	840	N.A.	N.A.	N.A.	840	840	840

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5
Compressor cap.	m³	N.A.	N.A.	N.A.	450	450	450	N.A.	N.A.	N.A.	450	450	450

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	480	480	480	N.A.	N.A.	N.A.	480	480	480
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	424,800	424,800	424,800	N.A.	N.A.	N.A.	424,800	424,800	424,800
Air consumption	kg/h	N.A.	N.A.	N.A.	115.8	115.8	115.8	N.A.	N.A.	N.A.	115.8	115.8	115.8

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01h: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

List of Capacities for 9K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			2 x TCA88-25	3 x A185-L35	2 x MET90MA				2 x TCA88-25	3 x A185-L35	2 x MET90MA
.			

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	21.0	21.0	21.0	N.A.	N.A.	N.A.	21.0	21.0	21.0
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	12.9	12.9	12.9	N.A.	N.A.	N.A.	12.9	12.9	12.9
Jacket cooling	m³/h	N.A.	N.A.	N.A.	325.0	325.0	325.0	N.A.	N.A.	N.A.	325.0	325.0	325.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	1590.0	1600.0	1600.0	N.A.	N.A.	N.A.	1630.0	1630.0	1630.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	830.0	830.0	840.0	N.A.	N.A.	N.A.	830.0	830.0	840.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	1,230	1,240	1,240

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	22,770	22,770	22,770	N.A.	N.A.	N.A.	22,650	22,650	22,650
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	736	736	736
Seawater flow	m³/h	N.A.	N.A.	N.A.	1,083	1,083	1,083	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	4,080	4,230	4,180	N.A.	N.A.	N.A.	4,080	4,230	4,180
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	830.0	830.0	840.0	N.A.	N.A.	N.A.	830.0	830.0	840.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	494	504	504
Seawater flow	m³/h	N.A.	N.A.	N.A.	507	517	517	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	6,300	6,300	6,300	N.A.	N.A.	N.A.	6,300	6,300	6,300
Jacket water flow	m³/h	N.A.	N.A.	N.A.	325	325	325	N.A.	N.A.	N.A.	325	325	325
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	494	504	504
Seawater flow	m³/h	N.A.	N.A.	N.A.	507	517	517	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	33,030	33,180	33,130
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,230	1,240	1,240
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,630	1,630	1,630

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 14.0	2 x 14.0	2 x 14.0	N.A.	N.A.	N.A.	2 x 14.0	2 x 14.0	2 x 14.0
Compressor cap.	m³	N.A.	N.A.	N.A.	840	840	840	N.A.	N.A.	N.A.	840	840	840

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5
Compressor cap.	m³	N.A.	N.A.	N.A.	450	450	450	N.A.	N.A.	N.A.	450	450	450

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	550	550	550	N.A.	N.A.	N.A.	550	550	550
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	477,900	477,900	477,900	N.A.	N.A.	N.A.	477,900	477,900	477,900
Air consumption	kg/h	N.A.	N.A.	N.A.	130.3	130.3	130.3	N.A.	N.A.	N.A.	130.3	130.3	130.3

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01i: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

List of Capacities for 10K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			3 x TCA88-21	3 x A190-L34	3 x MET83MA				3 x TCA88-21	3 x A190-L34	3 x MET83MA

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	23.0	23.0	23.0	N.A.	N.A.	N.A.	23.0	23.0	23.0
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	14.4	14.4	14.4	N.A.	N.A.	N.A.	14.4	14.4	14.4
Jacket cooling	m³/h	N.A.	N.A.	N.A.	360.0	360.0	360.0	N.A.	N.A.	N.A.	360.0	360.0	360.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	1780.0	1780.0	1780.0	N.A.	N.A.	N.A.	1810.0	1820.0	1810.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	930.0	920.0	940.0	N.A.	N.A.	N.A.	930.0	920.0	940.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	1,380	1,380	1,380

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	25,290	25,290	25,290	N.A.	N.A.	N.A.	25,160	25,160	25,160
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	818	818	818
Seawater flow	m³/h	N.A.	N.A.	N.A.	1,203	1,203	1,203	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	4,640	4,730	4,690	N.A.	N.A.	N.A.	4,640	4,730	4,690
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	930.0	920.0	940.0	N.A.	N.A.	N.A.	930.0	920.0	940.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	562	562	562
Seawater flow	m³/h	N.A.	N.A.	N.A.	577	577	577	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	7,000	7,000	7,000	N.A.	N.A.	N.A.	7,000	7,000	7,000
Jacket water flow	m³/h	N.A.	N.A.	N.A.	360	360	360	N.A.	N.A.	N.A.	360	360	360
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	562	562	562
Seawater flow	m³/h	N.A.	N.A.	N.A.	577	577	577	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	36,800	36,890	36,850
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,380	1,380	1,380
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,810	1,820	1,810

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 14.0	2 x 14.0	2 x 14.0	N.A.	N.A.	N.A.	2 x 14.0	2 x 14.0	2 x 14.0
Compressor cap.	m³	N.A.	N.A.	N.A.	840	840	840	N.A.	N.A.	N.A.	840	840	840

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5
Compressor cap.	m³	N.A.	N.A.	N.A.	450	450	450	N.A.	N.A.	N.A.	450	450	450

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	600	600	600	N.A.	N.A.	N.A.	600	600	600
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	531,000	531,000	531,000	N.A.	N.A.	N.A.	531,000	531,000	531,000
Air consumption	kg/h	N.A.	N.A.	N.A.	144.7	144.7	144.7	N.A.	N.A.	N.A.	144.7	144.7	144.7

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01j: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

List of Capacities for 11K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			3 x TCA88-21	3 x A190-L35	3 x MET83MA				3 x TCA88-21	3 x A190-L35	3 x MET83MA

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	25.0	25.0	25.0	N.A.	N.A.	N.A.	25.0	25.0	25.0
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	15.8	15.8	15.8	N.A.	N.A.	N.A.	15.8	15.8	15.8
Jacket cooling	m³/h	N.A.	N.A.	N.A.	395.0	395.0	395.0	N.A.	N.A.	N.A.	395.0	395.0	395.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	1950.0	1960.0	1960.0	N.A.	N.A.	N.A.	1990.0	1990.0	1990.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	1010.0	1000.0	1010.0	N.A.	N.A.	N.A.	1010.0	1000.0	1010.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	1,510	1,520	1,510

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	27,830	27,830	27,830	N.A.	N.A.	N.A.	27,680	27,680	27,680
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	900	900	900
Seawater flow	m³/h	N.A.	N.A.	N.A.	1,324	1,324	1,324	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	5,060	5,150	5,110	N.A.	N.A.	N.A.	5,060	5,150	5,110
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	1010.0	1000.0	1010.0	N.A.	N.A.	N.A.	1010.0	1000.0	1010.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	610	620	610
Seawater flow	m³/h	N.A.	N.A.	N.A.	626	636	636	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	7,700	7,700	7,700	N.A.	N.A.	N.A.	7,700	7,700	7,700
Jacket water flow	m³/h	N.A.	N.A.	N.A.	395	395	395	N.A.	N.A.	N.A.	395	395	395
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	610	620	610
Seawater flow	m³/h	N.A.	N.A.	N.A.	626	636	636	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	40,440	40,530	40,490
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,510	1,520	1,510
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,990	1,990	1,990

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 14.5	2 x 14.5	2 x 14.5	N.A.	N.A.	N.A.	2 x 14.5	2 x 14.5	2 x 14.5
Compressor cap.	m³	N.A.	N.A.	N.A.	870	870	870	N.A.	N.A.	N.A.	870	870	870

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5
Compressor cap.	m³	N.A.	N.A.	N.A.	450	450	450	N.A.	N.A.	N.A.	450	450	450

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	660	660	660	N.A.	N.A.	N.A.	660	660	660
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	584,100	584,100	584,100	N.A.	N.A.	N.A.	584,100	584,100	584,100
Air consumption	kg/h	N.A.	N.A.	N.A.	159.2	159.2	159.2	N.A.	N.A.	N.A.	159.2	159.2	159.2

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01k: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

List of Capacities for 12K90ME9-TII at NMCR - IMO NO_x Tier II compliance

Seawater cooling						Central cooling					
Conventional TC			High eff. TC			Conventional TC			High eff. TC		
			3 x TCA88-21	4 x A185-L35	3 x MET90MA				3 x TCA88-21	4 x A185-L35	3 x MET90MA

Pumps

Fuel oil circulation	m³/h	N.A.	N.A.	N.A.	28.0	28.0	28.0	N.A.	N.A.	N.A.	28.0	28.0	28.0
Fuel oil supply	m³/h	N.A.	N.A.	N.A.	17.3	17.3	17.3	N.A.	N.A.	N.A.	17.3	17.3	17.3
Jacket cooling	m³/h	N.A.	N.A.	N.A.	430.0	430.0	430.0	N.A.	N.A.	N.A.	430.0	430.0	430.0
Seawater cooling *	m³/h	N.A.	N.A.	N.A.	2130.0	2140.0	2140.0	N.A.	N.A.	N.A.	2170.0	2180.0	2180.0
Main lubrication oil *	m³/h	N.A.	N.A.	N.A.	1080.0	1080.0	1090.0	N.A.	N.A.	N.A.	1080.0	1080.0	1090.0
Central cooling *	m³/h	-	-	-	-	-	-	-	-	-	1,650	1,660	1,660

Scavenge air cooler(s)

Heat diss. app.	kW	N.A.	N.A.	N.A.	30,350	30,350	30,350	N.A.	N.A.	N.A.	30,200	30,200	30,200
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	981	981	981
Seawater flow	m³/h	N.A.	N.A.	N.A.	1,444	1,444	1,444	N.A.	N.A.	N.A.	-	-	-

Lubricating oil cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	5,480	5,640	5,640	N.A.	N.A.	N.A.	5,480	5,640	5,640
Lube oil flow *	m³/h	N.A.	N.A.	N.A.	1080.0	1080.0	1090.0	N.A.	N.A.	N.A.	1080.0	1080.0	1090.0
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	669	679	679
Seawater flow	m³/h	N.A.	N.A.	N.A.	686	696	696	N.A.	N.A.	N.A.	-	-	-

Jacket water cooler

Heat diss. app.	kW	N.A.	N.A.	N.A.	8,400	8,400	8,400	N.A.	N.A.	N.A.	8,400	8,400	8,400
Jacket water flow	m³/h	N.A.	N.A.	N.A.	430	430	430	N.A.	N.A.	N.A.	430	430	430
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	669	679	679
Seawater flow	m³/h	N.A.	N.A.	N.A.	686	696	696	N.A.	N.A.	N.A.	-	-	-

Central cooler

Heat diss. app. *	kW	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	44,080	44,240	44,240
Central water flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	1,650	1,660	1,660
Seawater flow	m³/h	N.A.	N.A.	N.A.	-	-	-	N.A.	N.A.	N.A.	2,170	2,180	2,180

Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 14.5	2 x 14.5	2 x 14.5	N.A.	N.A.	N.A.	2 x 14.5	2 x 14.5	2 x 14.5
Compressor cap.	m³	N.A.	N.A.	N.A.	870	870	870	N.A.	N.A.	N.A.	870	870	870

Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine

Receiver volume	m³	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5	N.A.	N.A.	N.A.	2 x 7.5	2 x 7.5	2 x 7.5
Compressor cap.	m³	N.A.	N.A.	N.A.	450	450	450	N.A.	N.A.	N.A.	450	450	450

Other values

Fuel oil heater	kW	N.A.	N.A.	N.A.	730	730	730	N.A.	N.A.	N.A.	730	730	730
Exh. gas temp.	°C	N.A.	N.A.	N.A.	250	250	250	N.A.	N.A.	N.A.	250	250	250
Exh. gas amount	kg/h	N.A.	N.A.	N.A.	637,200	637,200	637,200	N.A.	N.A.	N.A.	637,200	637,200	637,200
Air consumption	kg/h	N.A.	N.A.	N.A.	173.7	173.7	173.7	N.A.	N.A.	N.A.	173.7	173.7	173.7

* For main engine arrangements with built-on power take-off (PTO) of a MAN Diesel recommended type and/or torsional vibration damper the engine's capacities must be increased by those stated for the actual system

For List of Capacities for derated engines and performance data at part load please visit <http://www.manbw.dk/ceas/erd/>

Table 6.03.01I: Capacities for seawater and central systems as well as conventional and high efficiency turbochargers stated at NMCR

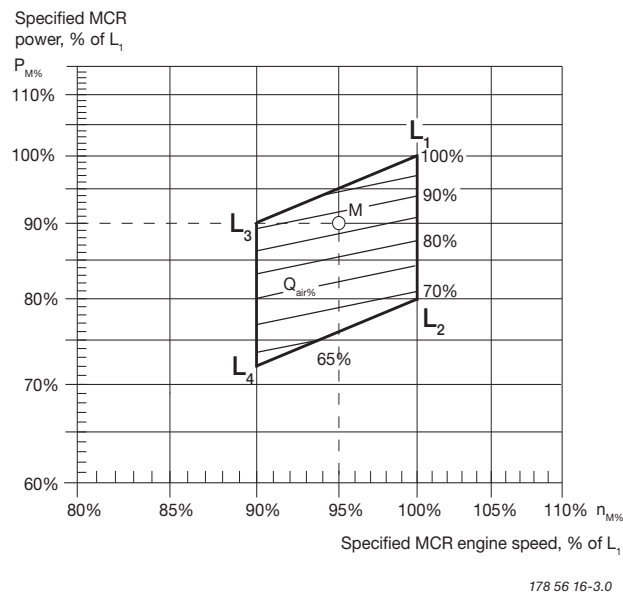
Auxiliary Machinery Capacities

The dimensioning of heat exchangers (coolers) and pumps for derated engines can be calculated on the basis of the heat dissipation values found by using the following description and diagrams. Those for the nominal MCR (L_1), may also be used if wanted.

The nomenclature of the basic engine ratings and coolers, etc. used in this section is shown in Fig. 6.01.01 and 6.01.02.

Cooler heat dissipations

For the specified MCR (M) the following three diagrams in Figs. 6.04.01, 6.04.02 and 6.04.03 show reduction factors for the corresponding heat dissipations for the coolers, relative to the values stated in the 'List of Capacities' valid for nominal MCR (L_1).



$$Q_{air\%} = 100 \times (P_M/P_{L1})^{1.68} \times (n_M/n_{L1})^{-0.83} \times k_O$$

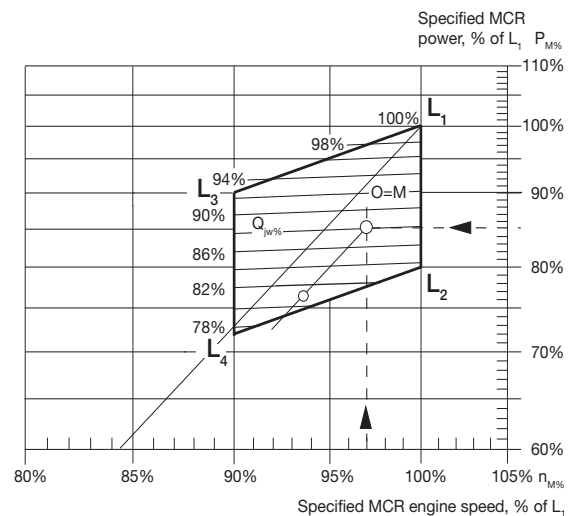
$$k_O = 1 + 0.27 \times (1 - P_O/P_M)$$

Fig. 6.04.01: Scavenge air cooler, heat dissipation $Q_{air\%}$ in point M, in % of the L_1 value $Q_{air, L1}$ and valid for $P_O = P_M$. As optimising point $O = M$, correction $k_O = 1$

The percentage power ($P_{M\%}$) and speed ($n_{M\%}$) of L_1 ie: $P_{M\%} = P_M/P_{L1} \times 100\%$

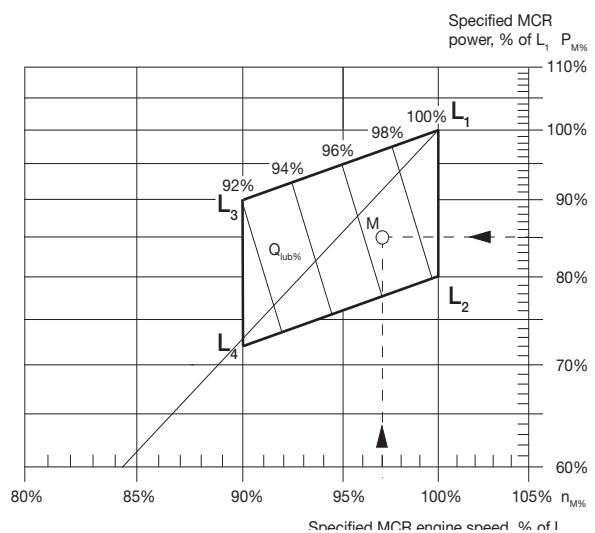
$$n_{M\%} = n_M/n_{L1} \times 100\%$$

for specified MCR (M) of the derated engine is used as input in the above-mentioned diagrams, giving the % heat dissipation figures relative to those in the 'List of Capacities',



$$Q_{jw\%} = e^{(-0.0811 \times \ln(n_{M\%}) + 0.8072 \times \ln(P_{M\%}) + 1.2614)}$$

Fig. 6.04.02: Jacket water cooler, heat dissipation $Q_{jw\%}$ in point M, in % of the L_1 value $Q_{jw, L1}$



$$Q_{lub\%} = 67.3009 \times \ln(n_{M\%}) + 7.6304 \times \ln(P_{M\%}) - 245.0714$$

Fig. 6.04.03: Lubricating oil cooler, heat dissipation $Q_{lub\%}$ in point M, in % of the L_1 value $Q_{lub, L1}$

The derated cooler capacities may then be found by means of following equations:

$$Q_{air, M} = Q_{air, L1} \times (Q_{air\%} / 100)$$

$$Q_{jw, M} = Q_{jw, L1} \times (Q_{jw\%} / 100)$$

$$Q_{lub, M} = Q_{lub, L1} \times (Q_{lub\%} / 100)$$

and for a central cooling water system the central cooler heat dissipation is:

$$Q_{cent, M} = Q_{air, M} + Q_{jw, M} + Q_{lub, M}$$

Pump capacities

The pump capacities given in the 'List of Capacities' refer to engines rated at nominal MCR (L_1). For lower rated engines, only a marginal saving in the pump capacities is obtainable.

To ensure proper lubrication, the lubricating oil pump must remain unchanged.

Also, the fuel oil circulating and supply pumps should remain unchanged.

In order to ensure reliable starting, the starting air compressors and the starting air receivers must also remain unchanged.

The jacket cooling water pump capacity is relatively low. Practically no saving is possible, and it is therefore unchanged.

Seawater cooling system

The derated seawater pump capacity is equal to the sum of the below found derated seawater flow capacities through the scavenge air and lube oil coolers, as these are connected in parallel.

The seawater flow capacity for each of the scavenge air, lube oil and jacket water coolers can be reduced proportionally to the reduced heat dissipations found in Figs. 6.04.01, 6.04.02 and 6.04.03, respectively i.e. as follows:

$$V_{sw, air, M} = V_{sw, air, L1} \times (Q_{air\%} / 100)$$

$$V_{sw, lub, M} = V_{sw, lub, L1} \times (Q_{lub\%} / 100)$$

$$V_{sw, jw, M} = V_{sw, lub, M}$$

However, regarding the scavenge air cooler(s), the engine maker has to approve this reduction in

order to avoid too low a water velocity in the scavenge air cooler pipes.

As the jacket water cooler is connected in series with the lube oil cooler, the seawater flow capacity for the latter is used also for the jacket water cooler.

Central cooling water system

If a central cooler is used, the above still applies, but the central cooling water capacities are used instead of the above seawater capacities. The seawater flow capacity for the central cooler can be reduced in proportion to the reduction of the total cooler heat dissipation, i.e. as follows:

$$V_{cw, air, M} = V_{cw, air, L1} \times (Q_{air\%} / 100)$$

$$V_{cw, lub, M} = V_{cw, lub, L1} \times (Q_{lub\%} / 100)$$

$$V_{cw, jw, M} = V_{cw, lub, M}$$

$$V_{cw, cent, M} = V_{cw, air, M} + V_{cw, lub, M}$$

$$V_{sw, cent, M} = V_{sw, cent, L1} \times Q_{cent, M} / Q_{cent, L1}$$

Pump pressures

Irrespective of the capacities selected as per the above guidelines, the below-mentioned pump heads at the mentioned maximum working temperatures for each system shall be kept:

	Pump head bar	Max. working temp. °C
Fuel oil supply pump	4	100
Fuel oil circulating pump	6	150
Lubricating oil pump	4.7	70
Seawater pump	2.5	50
Central cooling water pump	2.5	80
Jacket water pump	3.0	100

Flow velocities

For external pipe connections, we prescribe the following maximum velocities:

Marine diesel oil	1.0 m/s
Heavy fuel oil	0.6 m/s
Lubricating oil	1.8 m/s
Cooling water	3.0 m/s

Calculation of List of Capacities for Derated Engine

Example 1:

Pump and cooler capacities for a derated 6K90ME9-TII with high efficiency MAN Diesel turbocharger type TCA, fixed pitch propeller and central cooling water system.

Nominal MCR, (L_1) P_{L1} : 34,320 kW (100.0%) and 94.0 r/min (100.0%)

Specified MCR, (M) P_M : 29,172 kW (85.0%) and 91.2 r/min (97.0%)

Matching point, (O) P_O : 29,172 kW (85.0%) and 91.2 r/min (97.0%), $P_O = 100.0\%$ of P_M

The method of calculating the reduced capacities for point M ($n_{M\%} = 97.0\%$ and $P_{M\%} = 85.0\%$) is shown below.

The values valid for the nominal rated engine are found in the 'List of Capacities', Figs. 6.03.01 and 6.03.02, and are listed together with the result in the figure on the next page.

Heat dissipation of scavenge air cooler

Fig. 6.04.01 which approximately indicates a $Q_{air\%} = 78.1\%$ heat dissipation, i.e.:

$$Q_{air,M} = Q_{air,L1} \times Q_{air\%} / 100$$

$$Q_{air,M} = 15,100 \times 0.781 = 11,793 \text{ kW}$$

Heat dissipation of jacket water cooler

Fig. 6.04.02 indicates a $Q_{jw\%} = 87.9\%$ heat dissipation; i.e.:

$$Q_{jw,M} = Q_{jw,L1} \times Q_{jw\%} / 100$$

$$Q_{jw,M} = 4,200 \times 0.879 = 3,692 \text{ kW}$$

Heat dissipation of lube oil cooler

Fig. 6.04.03 indicates a $Q_{lub\%} = 96.7\%$ heat dissipation; i.e.:

$$Q_{lub,M} = Q_{lub,L1} \times Q_{lub\%} / 100$$

$$Q_{lub,M} = 2,820 \times 0.967 = 2,727 \text{ kW}$$

Heat dissipation of central water cooler

$$Q_{cent,M} = Q_{air,M} + Q_{jw,M} + Q_{lub,M}$$

$$Q_{cent,M} = 11,793 + 3,692 + 2,727 = 18,212 \text{ kW}$$

Total cooling water flow through scavenge air coolers

$$V_{cw,air,M} = V_{cw,air,L1} \times Q_{air\%} / 100$$

$$V_{cw,air,M} = 491 \times 0.781 = 383 \text{ m}^3/\text{h}$$

Cooling water flow through lubricating oil cooler

$$V_{cw,lub,M} = V_{cw,lub,L1} \times Q_{lub\%} / 100$$

$$V_{cw,lub,M} = 339 \times 0.967 = 328 \text{ m}^3/\text{h}$$

Cooling water flow through central cooler (Central cooling water pump)

$$V_{cw,cent,M} = V_{cw,air,M} + V_{cw,lub,M}$$

$$V_{cw,cent,M} = 383 + 328 = 711 \text{ m}^3/\text{h}$$

Cooling water flow through jacket water cooler (as for lube oil cooler)

$$V_{cw,jw,M} = V_{cw,lub,M}$$

$$V_{cw,jw,M} = 328 \text{ m}^3/\text{h}$$

Seawater pump for central cooler

As the seawater pump capacity and the central cooler heat dissipation for the nominal rated engine found in the 'List of Capacities' are 1,090 m³/h and 22,120 kW the derated seawater pump flow equals:

Seawater pump:

$$V_{sw,cent,M} = V_{sw,cent,L1} \times Q_{cent,M} / Q_{cent,L1}$$

$$= 1,090 \times 18,212 / 22,120 = 897 \text{ m}^3/\text{h}$$

		Nominal rated engine (L _r) High efficiency turbocharger (TCA)	Example 1 Specified MCR (M)
Shaft power at MCR		34,320 kW	29,172 kW
Engine speed at MCR		at 94.0 r/min	at 91.2 r/min
Power of matching point %MCR		100%	97%
Pumps:			
Fuel oil circulating pump	m ³ /h	13.8	13.8
Fuel oil supply pump	m ³ /h	8.6	8.6
Jacket cooling water pump	m ³ /h	215	215
Central cooling water pump	m ³ /h	830	711
Seawater pump	m ³ /h	1,090	897
Lubricating oil pump	m ³ /h	600	600
Coolers:			
Scavenge air cooler			
Heat dissipation	kW	15,100	11,793
Central water quantity	m ³ /h	491	383
Lub. oil cooler			
Heat dissipation	kW	2,820	2,727
Lubricating oil quantity	m ³ /h	600	600
Central water quantity	m ³ /h	339	328
Jacket water cooler			
Heat dissipation	kW	4,200	3,692
Jacket cooling water quantity	m ³ /h	215	215
Central water quantity	m ³ /h	339	328
Central cooler			
Heat dissipation	kW	22,120	18,212
Central water quantity	m ³ /h	830	711
Seawater quantity	m ³ /h	1,090	897
Fuel oil heater:	kW	360	360
Gases at ISO ambient conditions*			
Exhaust gas amount	kg/h	318,600	266,600
Exhaust gas temperature	°C	250	246.2
Air consumption	kg/s	86.8	72.7
Starting air system: 30 bar (gauge)			
Reversible engine			
Receiver volume (12 starts)	m ³	2 x 13.5	2 x 13.5
Compressor capacity, total	m ³ /h	810	810
Non-reversible engine			
Receiver volume (6 starts)	m ³	2 x 7.0	2 x 7.0
Compressor capacity, total	m ³ /h	420	420
Exhaust gas tolerances: temperature ±15 °C and amount ±5%			

The air consumption and exhaust gas figures are expected and refer to 100% specified MCR, ISO ambient reference conditions and the exhaust gas back pressure 300 mm WC

The exhaust gas temperatures refer to after turbocharger

* Calculated in example 3, in this chapter

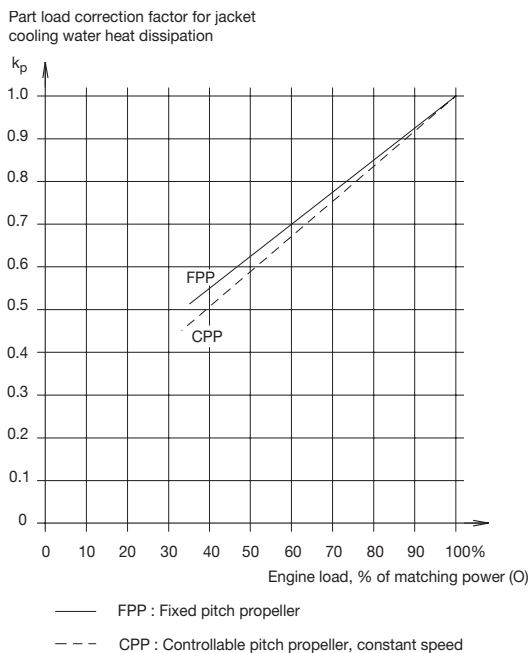
Example 1 – Capacities of derated 6K90ME9-TII with high efficiency MAN Diesel turbocharger type TCA and central cooling water system.

Freshwater Generator

If a freshwater generator is installed and is utilising the heat in the jacket water cooling system, it should be noted that the actual available heat in the jacket cooling water system is **lower** than indicated by the heat dissipation figures valid for nominal MCR (L_f) given in the List of Capacities. This is because the latter figures are used for dimensioning the jacket water cooler and hence incorporate a safety margin which can be needed when the engine is operating under conditions such as, e.g. overload. Normally, this margin is 10% at nominal MCR.

Calculation Method

For a derated diesel engine, i.e. an engine having a specified MCR (M) equal to matching point (O) different from L_f , the relative jacket water heat dissipation for point M and O may be found, as previously described, by means of Fig. 6.04.02.



$$\text{FPP : } k_p = 0.742 \times \frac{P_s}{P_o} + 0.258$$

$$\text{CPP : } k_p = 0.822 \times \frac{P_s}{P_o} + 0.178$$

Fig. 6.04.04: Correction factor 'k_p' for jacket cooling water heat dissipation at part load, relative to heat dissipation at matching power

At part load operation, lower than matching power, the actual jacket water heat dissipation will be reduced according to the curves for fixed pitch propeller (FPP) or for constant speed, controllable pitch propeller (CPP), respectively, in Fig. 6.04.04.

With reference to the above, the heat actually available for a derated diesel engine may then be found as follows:

1. Engine power equal to specified power M (equal to matching point O).

For specified MCR (M) = matching power (O), the diagram Fig. 6.04.02 is to be used, i.e. giving the percentage correction factor ' $Q_{jw\%}$ ' and hence for matching power P_o :

$$Q_{jw,O} = Q_{jw,L1} \times \frac{Q_{jw\%}}{100} \times 0.9 \quad (0.88) \quad [1]$$

2. Engine power lower than matching power.

For powers lower than the matching power, the value $Q_{jw,O}$ found for point O by means of the above equation [1] is to be multiplied by the correction factor k_p found in Fig. 6.04.04 and hence

$$Q_{jw} = Q_{jw,O} \times k_p \quad -15\%/0\% \quad [2]$$

where

Q_{jw} = jacket water heat dissipation

$Q_{jw,L1}$ = jacket water heat dissipation at nominal MCR (L_f)

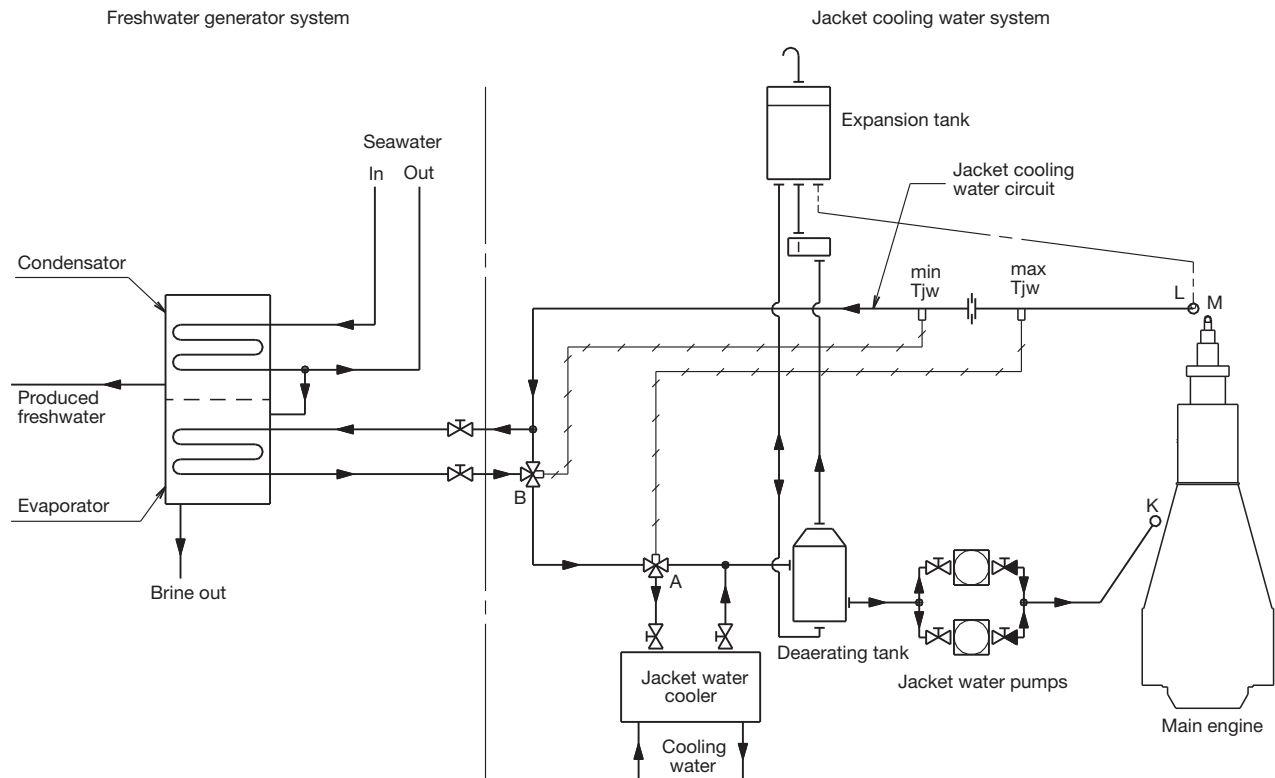
$Q_{jw\%}$ = percentage correction factor from Fig. 6.04.02

$Q_{jw,O}$ = jacket water heat dissipation at matching power (O), found by means of equation [1]

k_p = part load correction factor from Fig. 6.04.04

0.9 = factor for safety margin of cooler, tropical ambient conditions

The heat dissipation is assumed to be more or less independent of the ambient temperature conditions, yet the safety margin/ambient condition factor of about 0.88 instead of 0.90 will be more accurate for ambient conditions corresponding to ISO temperatures or lower. The heat dissipation tolerance from -15% to 0% stated above is based on experience.



Valve A: ensures that $T_{jw} < 85^\circ \text{C}$

Valve B: ensures that $T_{jw} > 85 - 5^\circ \text{C} = 80^\circ \text{C}$

Valve B and the corresponding by-pass may be omitted if, for example, the freshwater generator is equipped with an automatic start/stop function for too low jacket cooling water temperature

If necessary, all the actually available jacket cooling water heat may be utilised provided that a special temperature control system ensures that the jacket cooling water temperature at the outlet from the engine does not fall below a certain level

178 23 70-0.0

Fig. 6.04.05: Freshwater generators. Jacket cooling water heat recovery flow diagram

Jacket Cooling Water Temperature Control

When using a normal freshwater generator of the single-effect vacuum evaporator type, the freshwater production may, for guidance, be estimated as 0.03 t/24h per 1 kW heat, i.e.:

$$M_{fw} = 0.03 \times Q_{jw} \text{ t/24h } -15\%/0\% \quad [3]$$

where

M_{fw} is the freshwater production in tons per 24 hours

and

Q_{jw} is to be stated in kW

If necessary, all the actually available jacket cooling water heat may be used provided that a special temperature control system ensures that the jacket cooling water temperature at the outlet from the engine does not fall below a certain level. Such a temperature control system may consist, e.g., of a special by-pass pipe installed in the jacket cooling water system, see Fig. 6.04.05, or a special built-in temperature control in the freshwater generator, e.g., an automatic start/stop function, or similar.

If such a special temperature control is not applied, we recommend limiting the heat utilised to maximum 50% of the heat actually available at specified MCR, and only using the freshwater generator at engine loads above 50%. Considering the cooler margin of 10% and the minus tolerance of -15%, this heat corresponds to $50 \times (1.00 - 0.15) \times 0.9 = 38\%$ of the jacket water cooler capacity $Q_{jw,M}$ used for dimensioning of the jacket water cooler.

Calculation of Freshwater Production for Derated Engine

Example 2:

Freshwater production from a derated 6K90ME9-TII with high efficiency MAN Diesel turbocharger type TCA and fixed pitch propeller.

Based on the engine ratings below, this example will show how to calculate the expected available jacket cooling water heat removed from the diesel engine, together with the corresponding freshwater production from a freshwater generator.

The calculation is made for the service rating (S) of the diesel engine being 80% of the specified MCR.

Nominal MCR, (L₁) P_{L1}: 34,320 kW (100.0%) and 94.0 r/min (100.0%)

Specified MCR, (M) P_M: 29,172 kW (85.0%) and 91.2 r/min (97.0%)

Matching point, (O) P_O: 29,172 kW (85.0%) and 91.2 r/min (97.0%), P_O = 100.0% of P_M

Service rating, (S) P_S: 23,338 kW and 84.6 r/min, P_S = 80.0% of P_M and P_S = 80.0% of P_O

Ambient reference conditions: 20 °C air and 18 °C cooling water.

The expected available jacket cooling water heat at service rating is found as follows:

$$\begin{aligned} Q_{jw,L1} &= 4,200 \text{ kW from List of Capacities} \\ Q_{jw\%} &= 87.9\% \text{ using 85.0\% power and 97.0\%} \\ &\quad \text{speed for O in Fig. 6.04.02} \end{aligned}$$

By means of equation [1], and using factor 0.88 for actual ambient condition the heat dissipation in the matching point (O) is found:

$$\begin{aligned} Q_{jw,O} &= Q_{jw,L1} \times \frac{Q_{jw\%}}{100} \times 0.88 \\ &= 4,200 \times \frac{87.9}{100} \times 0.88 = 3,249 \text{ kW} \end{aligned}$$

By means of equation [2], the heat dissipation in the service point (S) i.e. for 88.9% of matching power, is found:

$$\begin{aligned} k_p &= 0.852 \text{ using 80.0\% in Fig. 6.04.04} \\ Q_{jw} &= Q_{jw,O} \times k_p = 3,249 \times 0.852 = 2,768 \text{ kW} \\ &\quad -15\%/0\% \end{aligned}$$

For the service point the corresponding expected obtainable freshwater production from a freshwater generator of the single effect vacuum evaporator type is then found from equation [3]:

$$M_{fw} = 0.03 \times Q_{jw} = 0.03 \times 2,768 = 83.0 \text{ t/24h} \\ -15\%/0\%$$

Exhaust Gas Amount and Temperature

Influencing factors

The exhaust gas data to be expected in practice depends, primarily, on the following three factors:

a) The specified MCR point of the engine (point M):

P_M : power in kW at SMCR point
 n_M : speed in r/min at SMCR point

and to a certain degree on the matching point O with the percentage power $P_{O\%} = \%$ of SMCR power:

$$P_{O\%} = (P_O/P_M) \times 100\%$$

b) The ambient conditions, and exhaust gas back-pressure:

T_{air} : actual ambient air temperature, in °C
 p_{bar} : actual barometric pressure, in mbar
 T_{CW} : actual scavenge air coolant temperature, in °C
 Δp_M : exhaust gas back-pressure in mm WC at specified MCR

c) The continuous service rating of the engine (point S), valid for fixed pitch propeller or controllable pitch propeller (constant engine speed):

P_S : continuous service rating of engine, in kW

Calculation Method

To enable the project engineer to estimate the actual exhaust gas data at an arbitrary service rating, the following method of calculation may be used.

The partial calculations based on the above influencing factors have been summarised in equations [4] and [5].

M_{exh} : exhaust gas amount in kg/h, to be found
 T_{exh} : exhaust gas temperature in °C, to be found

$$M_{exh} = M_{L1} \times \frac{P_M}{P_{L1}} \times \left\{ 1 + \frac{\Delta m_{M\%}}{100} \right\} \times \left\{ 1 + \frac{\Delta M_{amb\%}}{100} \right\} \times \left\{ 1 + \frac{\Delta m_{S\%}}{100} \right\} \times \frac{P_{S\%}}{100} \quad \text{kg/h} \quad \pm 5\% \quad [4]$$

$$T_{exh} = T_{L1} + \Delta T_M + \Delta T_O + \Delta T_{amb} + \Delta T_S \quad ^\circ\text{C} \quad \pm 15 \quad [5]$$

where, according to 'List of capacities', i.e. referring to ISO ambient conditions and 300 mm WC back-pressure and specified/matched in L_1 :

M_{L1} : exhaust gas amount in kg/h at nominal MCR (L_1)

T_{L1} : exhaust gas temperature after turbocharger in °C at nominal MCR (L_1)

Fig. 6.04.06: Summarising equations for exhaust gas amounts and temperatures

The partial calculations based on the influencing factors are described in the following:

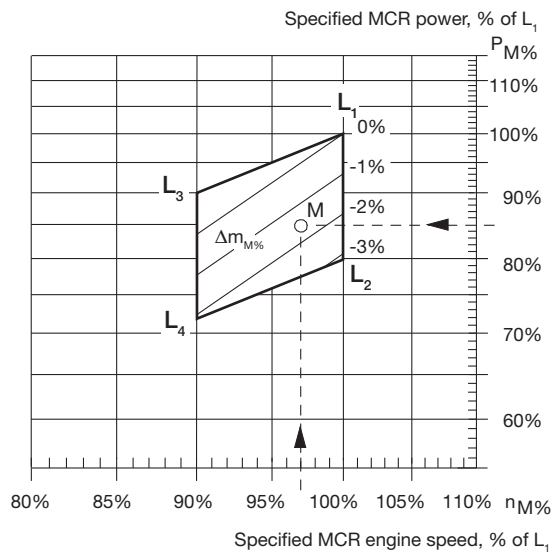
a) *Correction for choice of specified MCR point*

When choosing a specified MCR point 'M' other than the nominal MCR point ' L_1 ', the resulting

changes in specific exhaust gas amount and temperature are found by using as input in diagrams the corresponding percentage values (of L_1) for specified MCR power $P_{M\%}$ and speed $n_{M\%}$:

$$P_{M\%} = P_M/P_{L1} \times 100\%$$

$$n_{M\%} = n_M/n_{L1} \times 100\%$$



$$\Delta m_{M\%} = 14 \times \ln (P_M/P_{L1}) - 24 \times \ln (n_M/n_{L1})$$

178 59 35-0.0

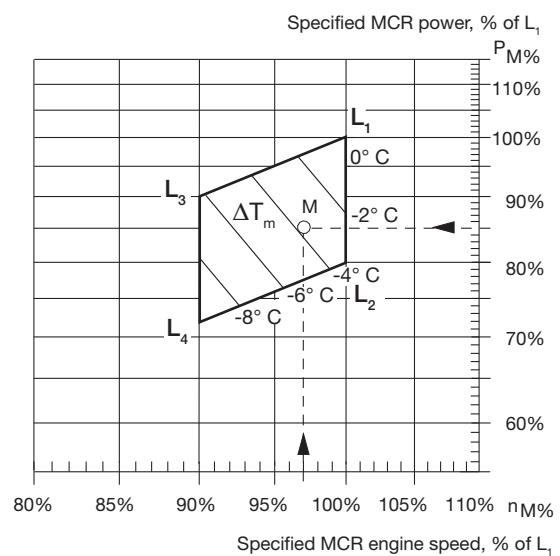
Fig. 6.04.07: Change of specific exhaust gas amount, $\Delta m_{M\%}$ in % of L_1 value and independent of P_O

$\Delta m_{M\%}$: change of specific exhaust gas amount, in % of specific gas amount at nominal MCR (L_1), see Fig. 6.04.07.

ΔT_M : change in exhaust gas temperature after turbocharger relative to the L_1 value, in °C, see Fig. 6.04.08. ($P_O = P_M$)

ΔT_O : extra change in exhaust gas temperature when matching point O lower than 100% M:
 $P_{O\%} = (P_O/P_M) \times 100\%$.

$$\Delta T_O = -0.3 \times (100 - P_{O\%}) \quad [6]$$



$$\Delta T_M = 15 \times \ln (P_M/P_{L1}) + 45 \times \ln (n_M/n_{L1})$$

178 59 36-2.0

Fig. 6.04.08: Change of exhaust gas temperature, ΔT_M in point M, in °C after turbocharger relative to L_1 value and valid for $P_O = P_M$

b) Correction for actual ambient conditions and back-pressure

For ambient conditions other than ISO 3046-1:2002 (E) and ISO 15550:2002 (E), and back-pressure other than 300 mm WC at specified MCR point (M), the correction factors stated in the table in Fig. 6.04.09 may be used as a guide, and the corresponding relative change in the exhaust gas data may be found from equations [7] and [8], shown in Fig. 6.04.10.

Parameter	Change	Change of exhaust gas temperature	Change of exhaust gas amount
Blower inlet temperature	+ 10° C	+ 16.0° C	- 4.1 %
Blower inlet pressure (barometric pressure)	+ 10 mbar	- 0.1° C	+ 0.3 %
Charge air coolant temperature (seawater temperature)	+ 10° C	+ 1.0° C	+ 1.9 %
Exhaust gas back pressure at the specified MCR point	+ 100 mm WC	+ 5.0° C	-1.1 %

Fig. 6.04.09: Correction of exhaust gas data for ambient conditions and exhaust gas back pressure

$$\Delta M_{\text{amb}\%} = -0.41 \times (T_{\text{air}} - 25) + 0.03 \times (p_{\text{bar}} - 1000) + 0.19 \times (T_{\text{CW}} - 25) - 0.011 \times (\Delta p_{\text{M}} - 300) \% \quad [7]$$

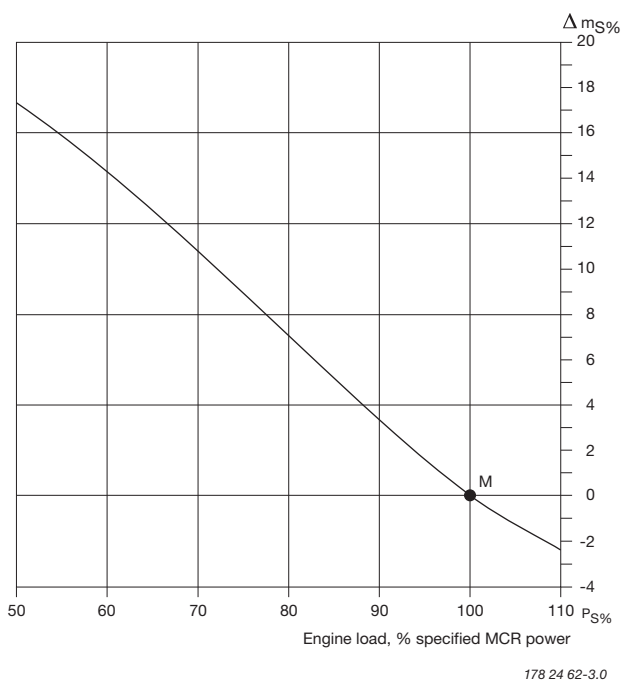
$$\Delta T_{\text{amb}} = 1.6 \times (T_{\text{air}} - 25) - 0.01 \times (p_{\text{bar}} - 1000) + 0.1 \times (T_{\text{CW}} - 25) + 0.05 \times (\Delta p_{\text{M}} - 300) ^\circ\text{C} \quad [8]$$

where the following nomenclature is used:

$\Delta M_{\text{amb}\%}$: change in exhaust gas amount, in % of amount at ISO conditions

ΔT_{amb} : change in exhaust gas temperature, in $^\circ\text{C}$ compared with temperatures at ISO conditions

Fig. 6.04.10: Exhaust gas correction formula for ambient conditions and exhaust gas back pressure



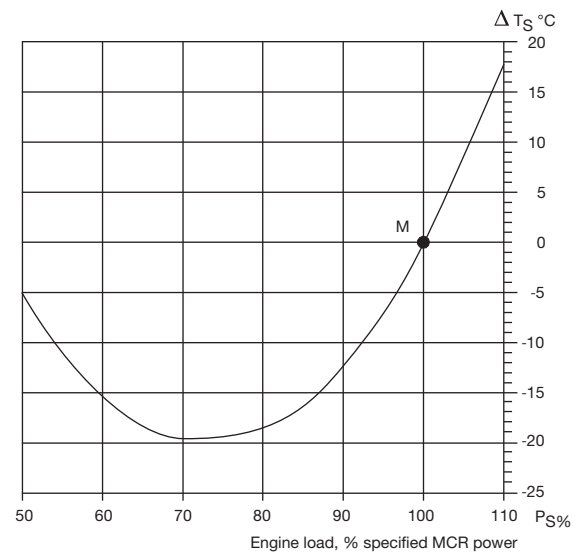
$$P_{\text{S}\%} = (P_{\text{S}}/P_{\text{M}}) \times 100\%$$

$$\Delta m_{\text{S}\%} = 37 \times (P_{\text{S}}/P_{\text{M}})^3 - 87 \times (P_{\text{S}}/P_{\text{M}})^2 + 31 \times (P_{\text{S}}/P_{\text{M}}) + 19$$

Fig. 6.04.11: Change of specific exhaust gas amount, $\Delta m_{\text{S}\%}$ in % at part load, and valid for FPP and CPP

c) Correction for engine load

Figs. 6.04.11 and 6.04.12 may be used, as guidance, to determine the relative changes in the specific exhaust gas data when running at part load, compared to the values in the specified MCR point, i.e. using as input $P_{\text{S}\%} = (P_{\text{S}}/P_{\text{M}}) \times 100\%$:



$$P_{\text{S}\%} = (P_{\text{S}}/P_{\text{M}}) \times 100\%$$

$$\Delta T_{\text{S}} = 280 \times (P_{\text{S}}/P_{\text{M}})^2 - 410 \times (P_{\text{S}}/P_{\text{M}}) + 130$$

Fig. 6.04.12: Change of exhaust gas temperature, ΔT_{S} in $^\circ\text{C}$ at part load, and valid for FPP and CPP

$\Delta m_{\text{S}\%}$: change in specific exhaust gas amount, in % of specific amount at specified MCR point, see Fig. 6.04.11.

ΔT_{S} : change in exhaust gas temperature, in $^\circ\text{C}$, see Fig. 6.04.12.

Calculation of Exhaust Data for Derated Engine

Example 3:

Expected exhaust gas data for a derated 6K90ME9-TII with high efficiency MAN Diesel turbocharger type TCA and fixed pitch propeller.

Based on the engine ratings below, and by means of an example, this chapter will show how to calculate the expected exhaust gas amount and temperature at service rating, and for a given ambient reference condition different from ISO.

The calculation is made for the service rating (S) being 80% of the specified MCR power of the diesel engine.

Nominal MCR, (L_1) P_{L1} : 34,320 kW (100.0%) and 94.0 r/min (100.0%)

Specified MCR, (M) P_M : 29,172 kW (85.0%) and 91.2 r/min (97.0%)

Matching point, (O) P_O : 29,172 kW (85.0%) and 91.2 r/min (97.0%), $P_O = 100.0\%$ of P_M

Service rating, (S) P_S : 23,338 kW and 84.6 r/min, $P_S = 80.0\%$ of P_M

Reference conditions

Air temperature T_{air} 20 °C
 Scavenge air coolant temperature T_{CW} 18 °C
 Barometric pressure p_{bar} 1,013 mbar
 Exhaust gas back-pressure
 at specified MCR Δp_M 300 mm WC

a) Correction for choice of specified MCR point M and matching point O:

$$P_{M\%} = \frac{29,172}{34,320} \times 100 = 85.0\%$$

$$\eta_{M\%} = \frac{91.2}{94.0} \times 100 = 97.0\%$$

By means of Figs. 6.04.07 and 6.04.08:

$$\Delta m_{M\%} = -1.54\%$$

$$\Delta T_M = -3.8 \text{ °C}$$

As the engine is matched in O lower than 100% M, and $P_{O\%} = 100.0\%$ of P_M

we get by means of equation [6]

$$\Delta T_O = -0.3 \times (100 - 100.0) = -0.0 \text{ °C}$$

b) Correction for ambient conditions and back-pressure:

By means of equations [7] and [8]:

$$\Delta M_{amb\%} = -0.41 \times (20 - 25) + 0.03 \times (1,013 - 1,000) + 0.19 \times (18 - 25) - 0.011 \times (300 - 300)\%$$

$$\Delta M_{amb\%} = +1.11\%$$

$$\Delta T_{amb} = 1.6 \times (20 - 25) - 0.01 \times (1,013 - 1,000) + 0.1 \times (18 - 25) + 0.05 \times (300 - 300) \text{ °C}$$

$$\Delta T_{amb} = -8.8 \text{ °C}$$

c) Correction for the engine load:

Service rating = 80% of specified MCR power
 By means of Figs. 6.04.11 and 6.04.12:

$$\Delta m_{S\%} = +7.1\%$$

$$\Delta T_S = -18.8 \text{ °C}$$

Final calculation

By means of equations [4] and [5], the final result is found taking the exhaust gas flow M_{L1} and temperature T_{L1} from the 'List of Capacities':

$$M_{L1} = 318,600 \text{ kg/h}$$

$$M_{\text{exh}} = 318,600 \times \frac{29,172}{34,320} \times \left(1 + \frac{-1.54}{100}\right) \times \left(1 + \frac{1.11}{100}\right) \times \left(1 + \frac{7.1}{100}\right) \times \frac{80}{100} = 230,993 \text{ kg/h}$$

$$M_{\text{exh}} = 231,000 \text{ kg/h} \pm 5\%$$

The exhaust gas temperature

$$T_{L1} = 250 \text{ }^{\circ}\text{C}$$

$$T_{\text{exh}} = 250 - 3.8 - 0.0 - 8.8 - 18.8 = 218.6 \text{ }^{\circ}\text{C}$$

$$T_{\text{exh}} = 218.6 \text{ }^{\circ}\text{C} \mp 15 \text{ }^{\circ}\text{C}$$

Exhaust gas data at specified MCR (ISO)

At specified MCR (M), the running point may be in equations [4] and [5] considered as a service point where $P_{s\%} = 100$, $\Delta m_{s\%} = 0.0$ and $\Delta T_s = 0.0$.

For ISO ambient reference conditions where $\Delta M_{\text{amb}\%} = 0.0$ and $\Delta T_{\text{amb}} = 0.0$, the corresponding calculations will be as follows:

$$M_{\text{exh,M}} = 318,600 \times \frac{29,172}{34,320} \times \left(1 + \frac{-1.54}{100}\right) \times \left(1 + \frac{0.0}{100}\right) \times \left(1 + \frac{0.0}{100}\right) \times \frac{100.0}{100} = 266,640 \text{ kg/h}$$

$$M_{\text{exh,M}} = 266,600 \text{ kg/h} \pm 5\%$$

$$T_{\text{exh,M}} = 250 - 3.8 - 0.0 + 0 + 0 = 246.2 \text{ }^{\circ}\text{C}$$

$$T_{\text{exh,M}} = 246.2 \text{ }^{\circ}\text{C} \mp 15 \text{ }^{\circ}\text{C}$$

The air consumption will be:

$$266,640 \times 0.982 \text{ kg/h} = 261,840 \text{ kg/h} \Leftrightarrow 261,840/3,600 \text{ kg/s} = 72.7 \text{ kg/s}$$

Fuel

7

Pressurised Fuel Oil System

The system is so arranged that both diesel oil and heavy fuel oil can be used, see Fig. 7.01.01.

From the service tank the fuel is led to an electrically driven supply pump by means of which a pressure of approximately 4 bar can be maintained in the low pressure part of the fuel circulating system, thus avoiding gasification of the fuel in the venting box in the temperature ranges applied.

The venting box is connected to the service tank via an automatic deaerating valve, which will release any gases present, but will retain liquids.

From the low pressure part of the fuel system the fuel oil is led to an electrically-driven circulating pump, which pumps the fuel oil through a heater and a full flow filter situated immediately before the inlet to the engine.

The fuel injection is performed by the electronically controlled pressure booster located on the Hydraulic Cylinder Unit (HCU), one per cylinder, which also contains the actuator for the electronic exhaust valve activation.

The Cylinder Control Units (CCU) of the Engine Control System (described in Section 16.01) calculate the timing of the fuel injection and the exhaust valve activation.

To ensure ample filling of the HCU, the capacity of the electrically-driven circulating pump is higher than the amount of fuel consumed by the diesel engine. Surplus fuel oil is recirculated from the engine through the venting box.

To ensure a constant fuel pressure to the fuel injection pumps during all engine loads, a spring loaded overflow valve is inserted in the fuel oil system on the engine.

The fuel oil pressure measured on the engine (at fuel pump level) should be 7-8 bar, equivalent to a circulating pump pressure of 10 bar.

Fuel considerations

When the engine is stopped, the circulating pump will continue to circulate heated heavy fuel through the fuel oil system on the engine, thereby keeping the fuel pumps heated and the fuel valves deaerated. This automatic circulation of preheated fuel during engine standstill is the background for our recommendation:

Constant operation on heavy fuel

In addition, if this recommendation was not followed, there would be a latent risk of diesel oil and heavy fuels of marginal quality forming incompatible blends during fuel change over or when operating in areas with restrictions on sulphur content in fuel oil due to exhaust gas emission control.

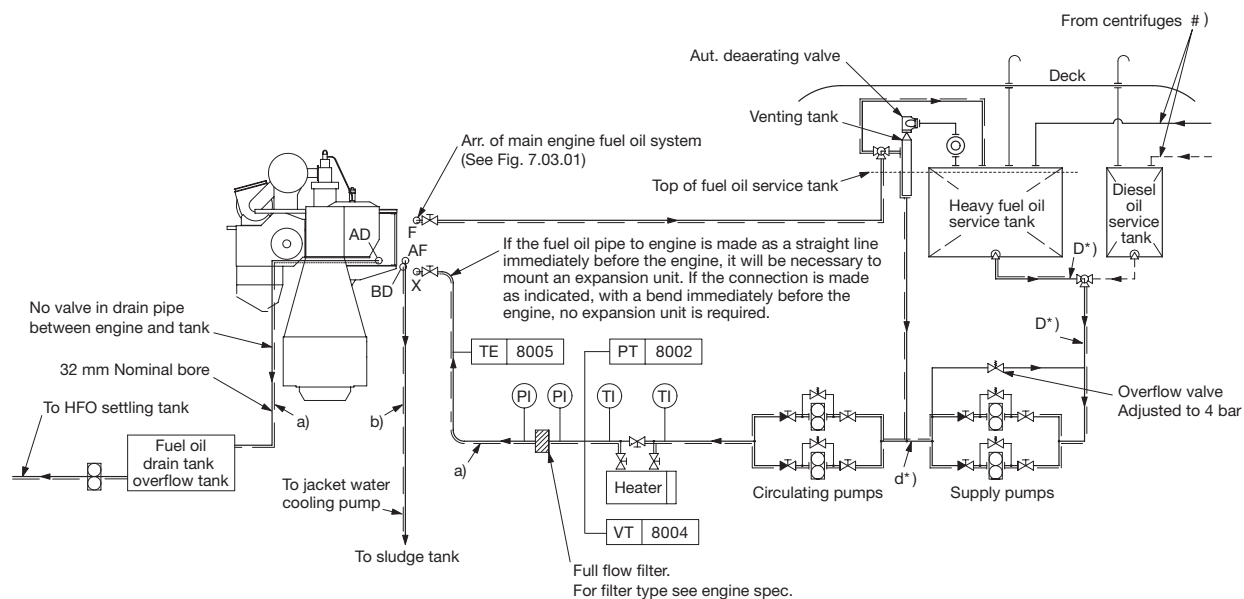
In special circumstances a change-over to diesel oil may become necessary – and this can be performed at any time, even when the engine is not running. Such a change-over may become necessary if, for instance, the vessel is expected to be inactive for a prolonged period with cold engine e.g. due to:

- docking
- stop for more than five days
- major repairs of the fuel system, etc.

The built-on overflow valves, if any, at the supply pumps are to be adjusted to 5 bar, whereas the external bypass valve is adjusted to 4 bar. The pipes between the tanks and the supply pumps shall have minimum 50% larger passage area than the pipe between the supply pump and the circulating pump.

If the fuel oil pipe 'X' at inlet to engine is made as a straight line immediately at the end of the engine, it will be necessary to mount an expansion joint. If the connection is made as indicated, with a bend immediately at the end of the engine, no expansion joint is required.

Fuel Oil System



#) Approximately the following quantity of fuel oil should be treated in the centrifuges: 0.23 l/kwh as explained in Section 7.05. The capacity of the centrifuges to be according to manufacturer's recommendation.

*) D to have min. 50% larger passage area than d.

178 52 19-7.4

- Diesel oil
- Heavy fuel oil
- ===== Heated pipe with insulation
- a) Tracing fuel oil lines: Max.150°C
- b) Tracing drain lines: By jacket cooling water

The letters refer to the list of 'Counterflanges'

Fig. 7.01.01: Fuel oil system

Drain of clean fuel oil from HCU, pumps, pipes

The HCU Fuel Oil Pressure Booster has a leakage drain of clean fuel oil from the umbrella sealing through 'AD' to the fuel oil drain tank.

The flow rate in litres is approximately as listed in Table 7.01.01.

Engine	Flow rate, litres/cyl. h.
K98ME/ME-C, S90ME-C	1.25
K90ME/ME-C, S/K80ME-C, S70ME-C/ME-GI, L70ME-C, S65ME-C/ME-GI	0.75
S/L60ME-C, S60ME-GI	0.60

Table 7.01.01: Approximate flow in HCU leakage drain.

This drained clean oil will, of course, influence the measured SFOC, but the oil is not wasted, and the quantity is well within the measuring accuracy of the flowmeters normally used.

The main purpose of the drain 'AF' is to collect pure fuel oil from the fuel pumps as well as the unintentional leakage from the high pressure pipes. The drain oil is led to a sludge tank and can be pumped to the Heavy Fuel Oil service tank or to the settling tank.

The 'AF' drain is provided with a box for giving alarm in case of leakage in a high pressure pipe.

The size of the sludge tank is determined on the basis of the draining intervals, the classification society rules, and on whether it may be vented directly to the engine room.

Drains 'AD' and 'AF' are shown in Fig. 7.03.01.

Drain of contaminated fuel etc.

Leakage oil, in shape of fuel and lubricating oil contaminated with water, dirt etc. and collected by the HCU Base Plate top plate, is drained off through the bedplate drains 'AE'.

Drain 'AE' is shown in Fig. 8.07.02.

Heating of fuel drain pipes

Owing to the relatively high viscosity of the heavy fuel oil, it is recommended that the drain pipes and the fuel oil drain tank are heated to min. 50 °C, but max. 100 °C.

The drain pipes between engine and tanks can be heated by the jacket water, as shown in Fig. 7.01.01 'Fuel pipe heating' as flange 'BD'.

Fuel oil flow velocity and viscosity

For external pipe connections, we prescribe the following maximum flow velocities:

Marine diesel oil 1.0 m/s
Heavy fuel oil..... 0.6 m/s

The fuel viscosity is influenced by factors such as emulsification of water into the fuel for reducing the NO_x emission. This is further described in Section 7.06.

An emulsification arrangement for the main engine is described in our publication:

Exhaust Gas Emission Control Today and Tomorrow

Further information about fuel oil specifications is available in our publication:

Guidelines for Fuels and Lubes Purchasing

The publications are available at:
www.mandiesel.com under
'Quicklinks' → 'Technical Papers'.

Fuel Oils

Marine diesel oil:

Marine diesel oil ISO 8217, Class DMB
British Standard 6843, Class DMB
Similar oils may also be used

Heavy fuel oil (HFO)

Most commercially available HFO with a viscosity below 700 cSt at 50 °C (7,000 sec. Redwood I at 100 °F) can be used.

For guidance on purchase, reference is made to ISO 8217:1996 and ISO 8217:2005, British Standard 6843 and to CIMAC recommendations regarding requirements for heavy fuel for diesel engines, fourth edition 2003, in which the maximum acceptable grades are RMH 700 and RMK 700. The above-mentioned ISO and BS standards supersede BSMA 100 in which the limit was M9.

The data in the above HFO standards and specifications refer to fuel as delivered to the ship, i.e. before on-board cleaning.

In order to ensure effective and sufficient cleaning of the HFO, i.e. removal of water and solid contaminants, the fuel oil specific gravity at 15 °C (60 °F) should be below 0.991, unless modern types of centrifuges with adequate cleaning abilities are used.

Higher densities can be allowed if special treatment systems are installed.

Current analysis information is not sufficient for estimating the combustion properties of the oil. This means that service results depend on oil properties which cannot be known beforehand. This especially applies to the tendency of the oil to form deposits in combustion chambers, gas passages and turbines. It may, therefore, be necessary to rule out some oils that cause difficulties.

Guiding heavy fuel oil specification

Based on our general service experience we have, as a supplement to the above mentioned standards, drawn up the guiding HFO specification shown below.

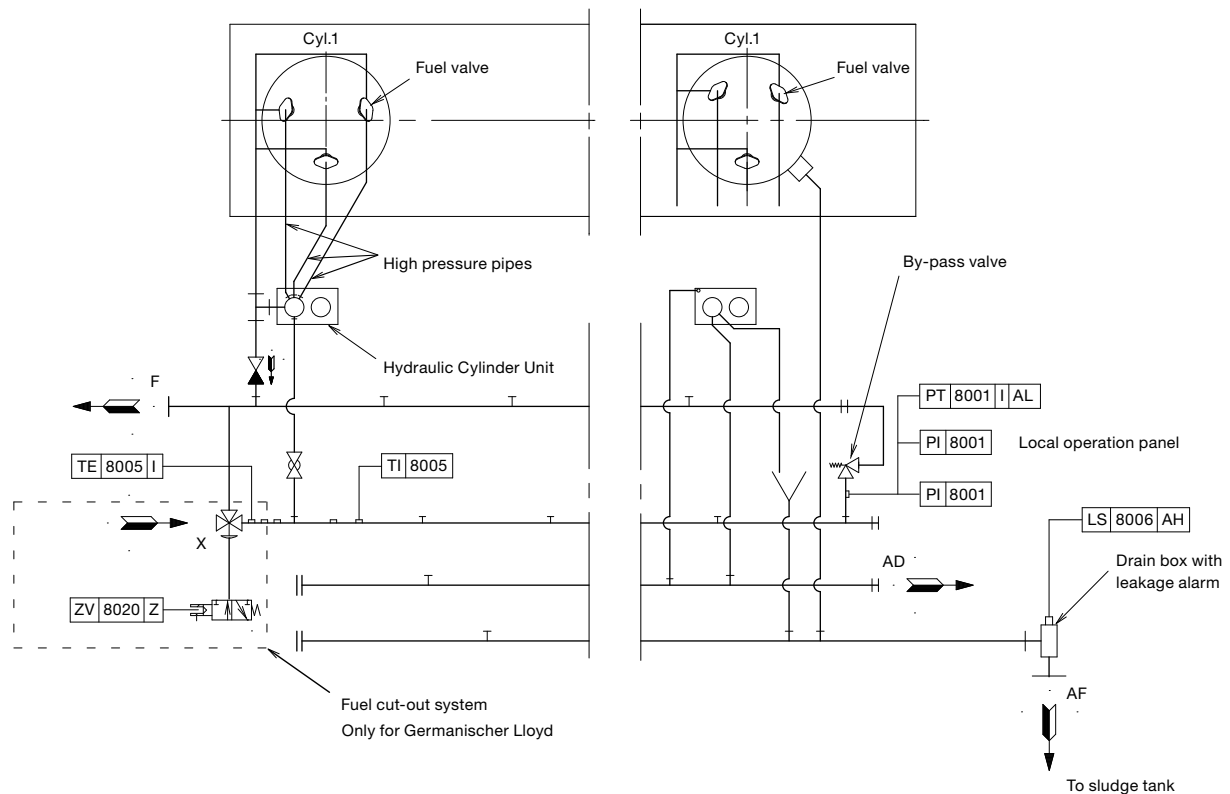
Heavy fuel oils limited by this specification have, to the extent of the commercial availability, been used with satisfactory results on MAN B&W two-stroke low speed diesel engines.

The data refers to the fuel as supplied i.e. before any on-board cleaning.

Guiding specification (maximum values)		
Density at 15 °C	kg/m ³	≤ 1.010*
Kinematic viscosity at 100 °C	cSt	≤ 55
at 50 °C	cSt	≤ 700
Flash point	°C	≥ 60
Pour point	°C	≤ 30
Carbon residue	% (m/m)	≤ 22
Ash	% (m/m)	≤ 0.15
Total sediment potential	% (m/m)	≤ 0.10
Water	% (v/v)	≤ 0.5
Sulphur	% (m/m)	≤ 4.5
Vanadium	mg/kg	≤ 600
Aluminum + Silicon	mg/kg	≤ 80
Equal to ISO 8217:2005 - RMK 700 / CIMAC recommendation No. 21 - K700		
* Provided automatic clarifiers are installed		
m/m = mass		v/v = volume

If heavy fuel oils with analysis data exceeding the above figures are to be used, especially with regard to viscosity and specific gravity, the engine builder should be contacted for advice regarding possible fuel oil system changes.

Fuel Oil Pipes and Drain Pipes



The letters refer to list of 'Counterflanges'

The item No. refer to 'Guidance values automation'

126 40 91-7.8.0a

Fig. 7.03.01: Fuel oil and drain pipes

Fuel Oil Pipe Insulation

Insulation of fuel oil pipes and fuel oil drain pipes should not be carried out until the piping systems have been subjected to the pressure tests specified and approved by the respective classification society and/or authorities, Fig. 7.04.01.

The directions mentioned below include insulation of hot pipes, flanges and valves with a surface temperature of the complete insulation of maximum 55 °C at a room temperature of maximum 38 °C. As for the choice of material and, if required, approval for the specific purpose, reference is made to the respective classification society.

Fuel oil pipes

The pipes are to be insulated with 20 mm mineral wool of minimum 150 kg/m³ and covered with glass cloth of minimum 400 g/m².

Fuel oil pipes and heating pipes together

Two or more pipes can be insulated with 30 mm wired mats of mineral wool of minimum 150 kg/m³ covered with glass cloth of minimum 400 g/m².

Flanges and valves

The flanges and valves are to be insulated by means of removable pads. Flange and valve pads are made of glass cloth, minimum 400 g/m², containing mineral wool stuffed to minimum 150 kg/m³.

Thickness of the pads to be:

Fuel oil pipes 20 mm
Fuel oil pipes and heating pipes together.... 30 mm

The pads are to be fitted so that they lap over the pipe insulating material by the pad thickness. At flanged joints, insulating material on pipes should not be fitted closer than corresponding to the minimum bolt length.

Mounting

Mounting of the insulation is to be carried out in accordance with the supplier's instructions.

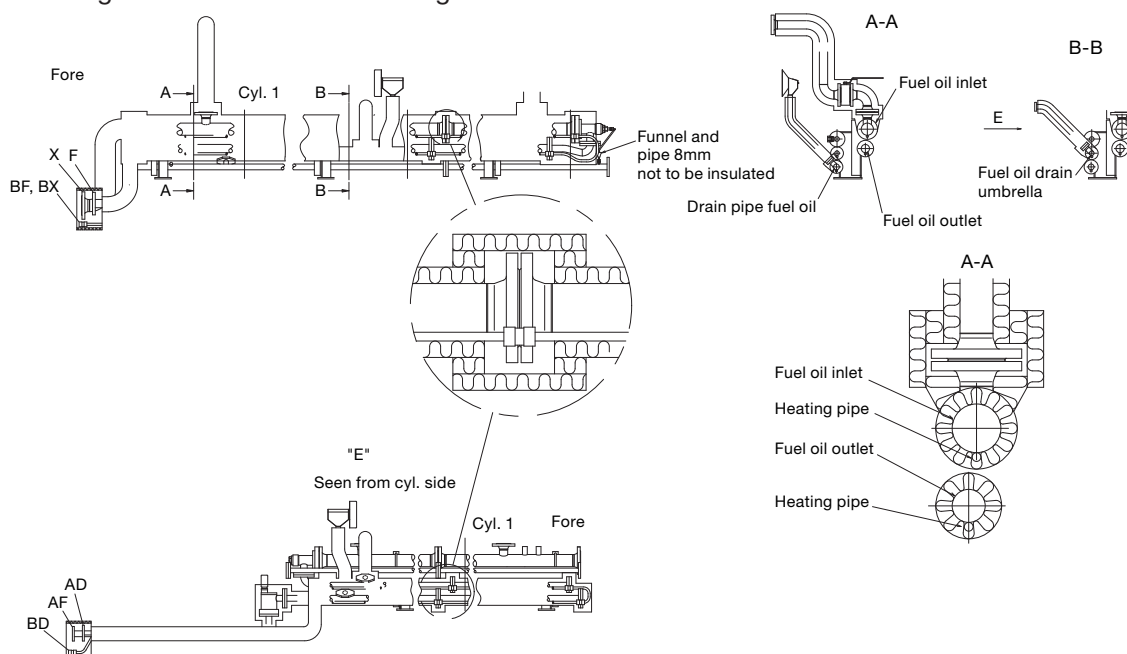
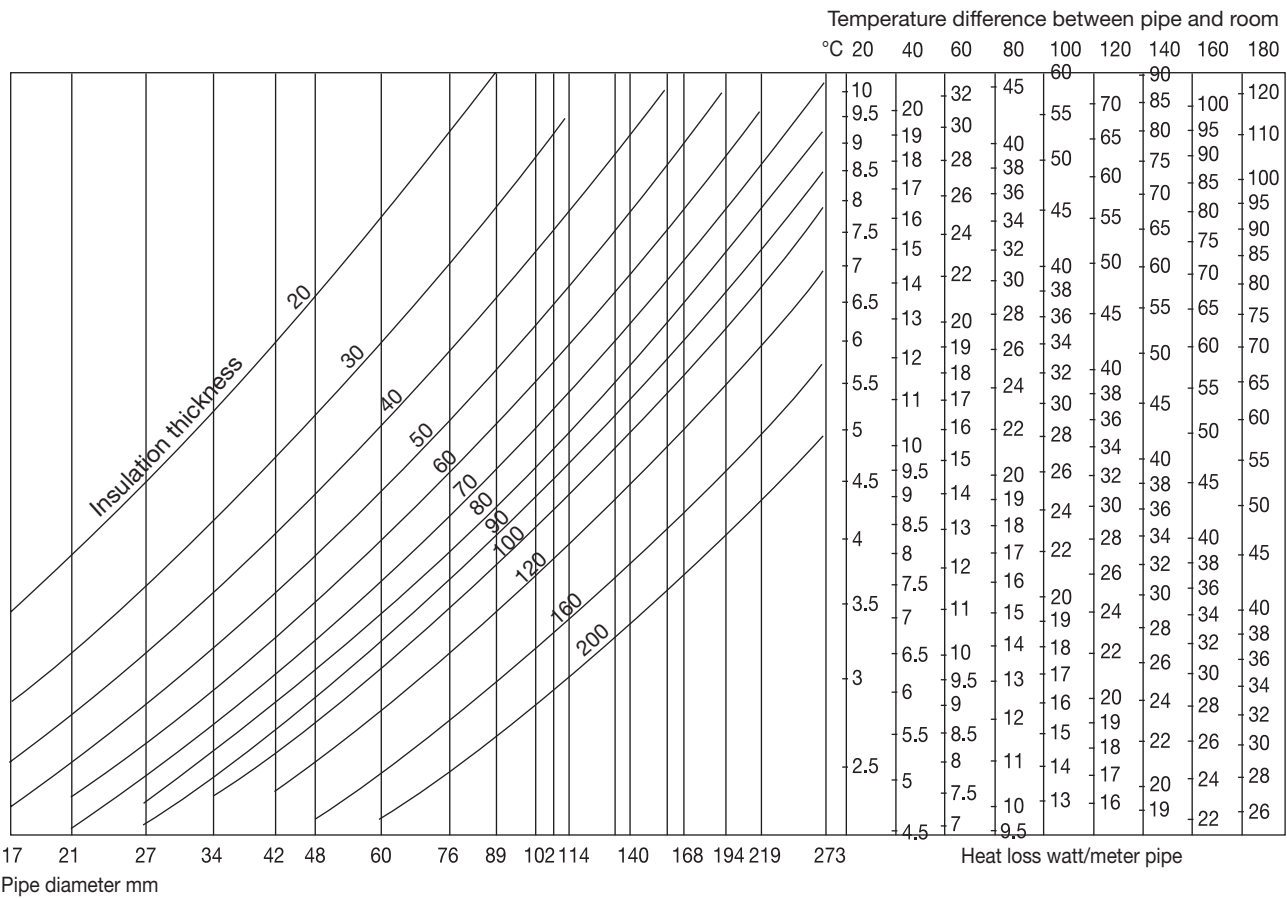


Fig. 7.04.01: Details of fuel oil pipes insulation, option: 4 35 121. Example from 98-50 MC engine

178 50 65 -0.2

Heat Loss in Piping



178 50 60-2.0

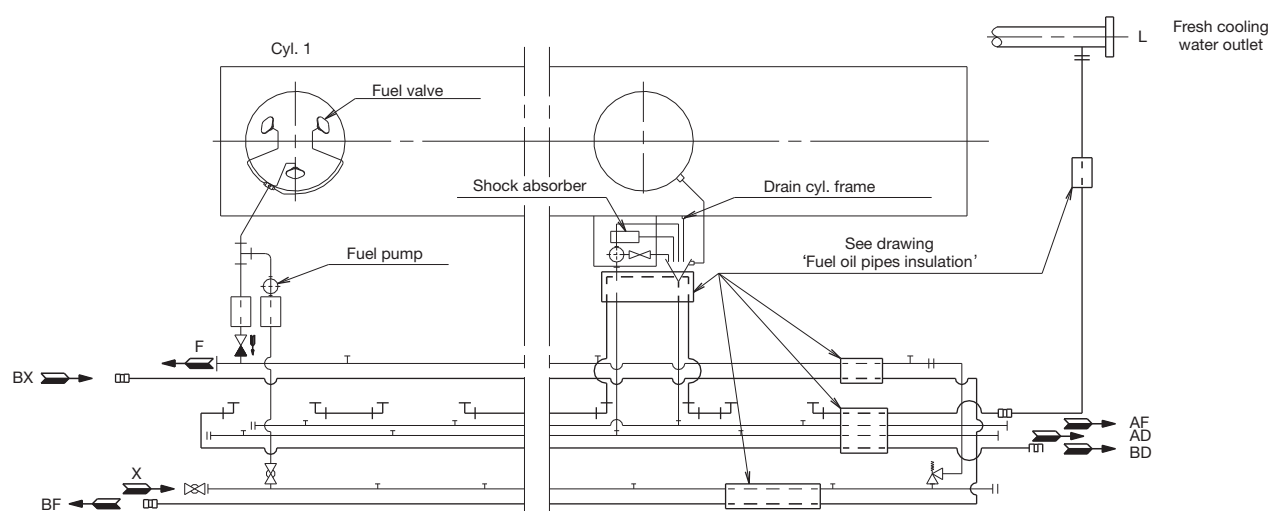
Fig. 7.04.02: Heat loss/Pipe cover

Fuel Oil Pipe Heat Tracing

The steam tracing of the fuel oil pipes is intended to operate in two situations:

1. When the circulation pump is running, there will be a temperature loss in the piping, see Fig. 7.04.02. This loss is very small, therefore tracing in this situation is only necessary with very long fuel supply lines.
2. When the circulation pump is stopped with heavy fuel oil in the piping and the pipes have cooled down to engine room temperature, as it is not possible to pump the heavy fuel oil. In this situation the fuel oil must be heated to pumping temperature of about 50 °C.

To heat the pipe to pumping level we recommend to use 100 watt leaking/meter pipe.



The letters refer to list of 'Counterflanges'

178 50 62-5.0

Fig. 7.04.03: Fuel oil pipe heat tracing

Fuel Oil and Lubricating Oil Pipe Spray Shields

In order to fulfil IMO regulations, fuel oil and lubricating oil pipe assemblies are to be enclosed by spray shields as shown in Fig. 7.04.04a and b.

To avoid leaks, the spray shields are to be installed after pressure testing of the pipe system.

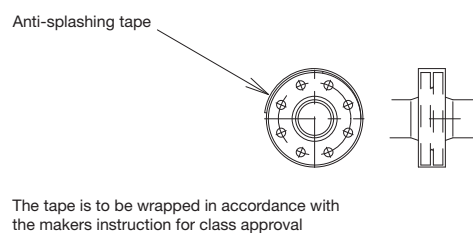


Fig. 7.04.04a: Spray Shields by anti-splashing tape

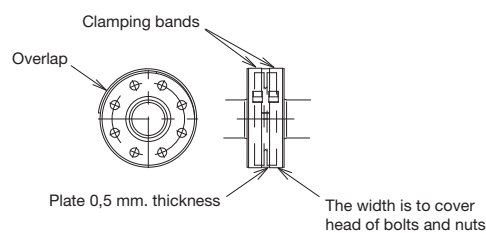


Fig. 7.04.04b: Spray Shields by clamping bands

178 52 55-5.2

Components for Fuel Oil System

Fuel oil centrifuges

The manual cleaning type of centrifuges are not to be recommended, neither for attended machinery spaces (AMS) nor for unattended machinery spaces (UMS). Centrifuges must be self-cleaning, either with total discharge or with partial discharge.

Distinction must be made between installations for:

- Specific gravities < 0.991 (corresponding to ISO 8217 and British Standard 6843 from RMA to RMH, and CIMAC from A to H-grades)
- Specific gravities > 0.991 and (corresponding to CIMAC K-grades).

For the latter specific gravities, the manufacturers have developed special types of centrifuges, e.g.:

Alfa Laval	Alcap
Westfalia	Unitrol
Mitsubishi	E-Hidens II

The centrifuge should be able to treat approximately the following quantity of oil:

0.23 litres/kWh

This figure includes a margin for:

- Water content in fuel oil
- Possible sludge, ash and other impurities in the fuel oil
- Increased fuel oil consumption, in connection with other conditions than ISO standard condition
- Purifier service for cleaning and maintenance.

The size of the centrifuge has to be chosen according to the supplier's table valid for the selected viscosity of the Heavy Fuel Oil. Normally, two centrifuges are installed for Heavy Fuel Oil (HFO), each with adequate capacity to comply with the above recommendation.

A centrifuge for Marine Diesel Oil (MDO) is not a must. However, MAN Diesel recommends that at least one of the HFO purifiers can also treat MDO.

If it is decided after all to install an individual purifier for MDO on board, the capacity should be based on the above recommendation, or it should be a centrifuge of the same size as that for HFO.

The *Nominal MCR* is used to determine the total installed capacity. Any derating can be taken into consideration in border-line cases where the centrifuge that is one step smaller is able to cover *Specified MCR*.

Fuel oil supply pump

This is to be of the screw or gear wheel type.

Fuel oil viscosity, specified.... up to 700 cSt at 50 °C
Fuel oil viscosity maximum 1000 cSt
Pump head 4 bar
Fuel oil flow see 'List of Capacities'
Delivery pressure 4 bar
Working temperature 100 °C
Minimum temperature 50 °C

The capacity stated in 'List of Capacities' is to be fulfilled with a tolerance of: $\pm 0\%$ to $+15\%$ and shall also be able to cover the back-flushing, see 'Fuel oil filter'.

Fuel oil circulating pump

This is to be of the screw or gear wheel type.

Fuel oil viscosity, specified.... up to 700 cSt at 50 °C
Fuel oil viscosity normal 20 cSt
Fuel oil viscosity maximum 1000 cSt
Fuel oil flow see 'List of Capacities'
Pump head 6 bar
Delivery pressure 10 bar
Working temperature 150 °C

The capacity stated in 'List of Capacities' is to be fulfilled with a tolerance of: $\pm 0\%$ to $+15\%$ and shall also be able to cover the back-flushing, see 'Fuel oil filter'.

Pump head is based on a total pressure drop in filter and preheater of maximum 1.5 bar.

Fuel Oil Heater

The heater is to be of the tube or plate heat exchanger type.

The required heating temperature for different oil viscosities will appear from the 'Fuel oil heating chart', Fig. 7.05.01. The chart is based on information from oil suppliers regarding typical marine fuels with viscosity index 70-80.

Since the viscosity after the heater is the controlled parameter, the heating temperature may vary, depending on the viscosity and viscosity index of the fuel.

Recommended viscosity meter setting is 10-15 cSt.

Fuel oil viscosity specified ... up to 700 cSt at 50°C
Fuel oil flow see capacity of fuel oil circulating pump
Heat dissipation see 'List of Capacities'
Pressure drop on fuel oil side maximum 1 bar
Working pressure 10 bar
Fuel oil inlet temperature approx. 100 °C
Fuel oil outlet temperature 150 °C
Steam supply, saturated 7 bar abs

To maintain a correct and constant viscosity of the fuel oil at the inlet to the main engine, the steam supply shall be automatically controlled, usually based on a pneumatic or an electrically controlled system.

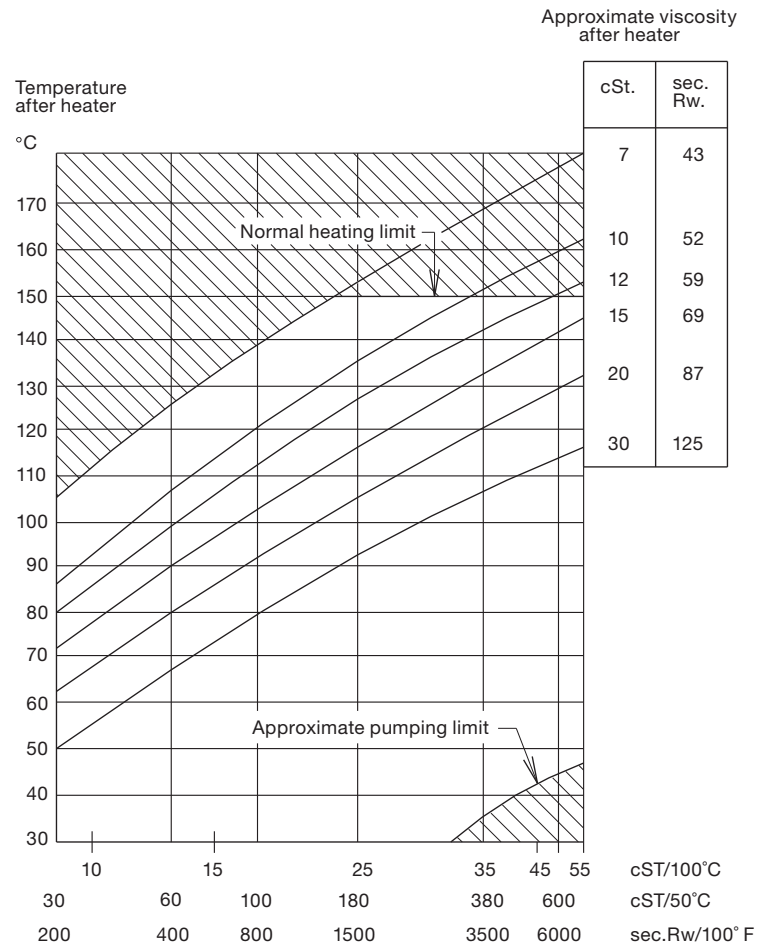


Fig. 7.05.01: Fuel oil heating chart

178 06 28-0.1

Fuel oil filter

The filter can be of the manually cleaned duplex type or an automatic filter with a manually cleaned bypass filter.

If a **double filter** (duplex) is installed, it should have sufficient capacity to allow the specified full amount of oil to flow through each side of the filter at a given working temperature with a max. 0.3 bar pressure drop across the filter (clean filter).

If a **filter with backflushing** arrangement is installed, the following should be noted. The required oil flow specified in the 'List of capacities', i.e. the delivery rate of the fuel oil supply pump and the fuel oil circulating pump, should be increased by the amount of oil used for the backflushing, so that the fuel oil pressure at the inlet to the main engine can be maintained during cleaning.

In those cases where an **automatically cleaned filter** is installed, it should be noted that in order to activate the cleaning process, certain makers of filters require a greater oil pressure at the inlet to the filter than the pump pressure specified. Therefore, the pump capacity should be adequate for this purpose, too.

The fuel oil filter should be based on heavy fuel oil of: 130 cSt at 80 °C = 700 cSt at 50 °C = 7000 sec Redwood I/100 °F.

Fuel oil flow see 'List of capacities'
 Working pressure 10 bar
 Test pressure according to class rule
 Absolute fineness 50 µm
 Working temperature maximum 150 °C
 Oil viscosity at working temperature 15 cSt
 Pressure drop at clean filter maximum 0.3 bar
 Filter to be cleaned at a pressure drop of maximum 0.5 bar

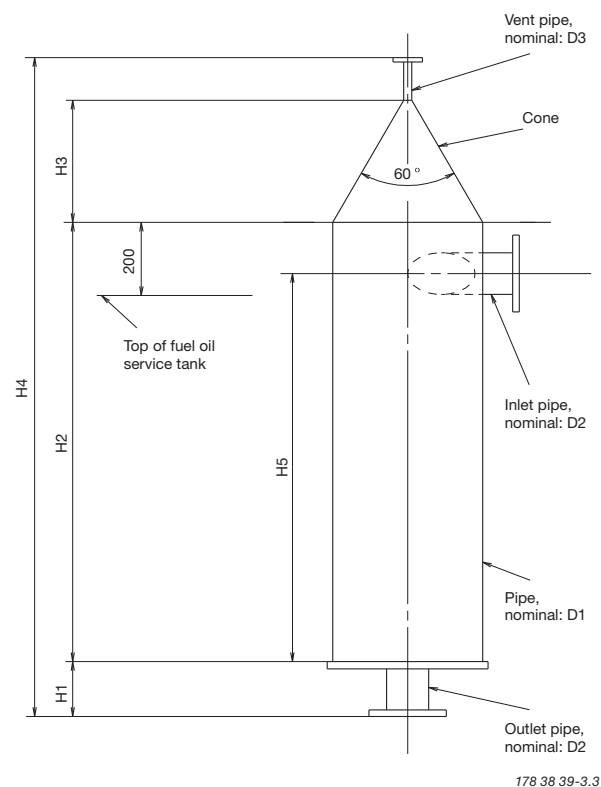
Note:

Absolute fineness corresponds to a nominal fineness of approximately 35 µm at a retaining rate of 90%.

The filter housing shall be fitted with a steam jacket for heat tracing.

Fuel oil venting box

The design of the Fuel oil venting box is shown in Fig. 7.05.02. The size is chosen according to the maximum flow of the fuel oil circulation pump, which is listed in section 6.03.



Flow m³/h Q (max.)*	Dimensions in mm							
	D1	D2	D3	H1	H2	H3	H4	H5
1.3	150	32	15	100	600	171.3	1,000	550
2.1	150	40	15	100	600	171.3	1,000	550
5.0	200	65	15	100	600	171.3	1,000	550
8.4	400	80	15	150	1,200	333.5	1,800	1,100
11.5	400	90	15	150	1,200	333.5	1,800	1,100
19.5	400	125	15	150	1,200	333.5	1,800	1,100
29.4	500	150	15	150	1,500	402.4	2,150	1,350
43.0	500	200	15	150	1,500	402.4	2,150	1,350

* The maximum flow of the fuel oil circulation pump

Fig. 07.05.02: Fuel oil venting box

Flushing of the fuel oil system

Before starting the engine for the first time, the system on board has to be flushed in accordance with MAN Diesel's recommendations 'Flushing of Fuel Oil System' which is available on request.

Water In Fuel Emulsification

The emulsification of water into the fuel oil reduces the NO_x emission with about 1% per 1% water added to the fuel up to about 20% without modification of the engine fuel injection equipment.

A Water In Fuel emulsion (WIF) mixed for this purpose and based on Heavy Fuel Oil (HFO) is stable for a long time, whereas a WIF based on Marine Diesel Oil is only stable for a short period of time unless an emulsifying agent is applied.

As both the MAN B&W two-stroke main engine and the MAN Diesel GenSets are designed to run on emulsified HFO, it can be used for a common system.

It is supposed below, that both the main engine and GenSets are running on the same fuel, either HFO or a homogenised HFO-based WIF.

Special arrangements are available on request for a more sophisticated system in which the GenSets can run with or without a homogenised HFO-based WIF, if the main engine is running on that.

Please note that the fuel pump injection capacity shall be confirmed for the main engine as well as the GenSets for the selected percentage of water in the WIF.

Temperature and pressure

When water is added by emulsification, the fuel viscosity increases. In order to keep the injection viscosity at 10-15 cSt and still be able to operate on up to 700 cSt fuel oil, the heating temperature has to be increased to about 170 °C depending on the water content.

The higher temperature calls for a higher pressure to prevent cavitation and steam formation in the system. The inlet pressure is thus set to 13 bar.

In order to avoid temperature chock when mixing water into the fuel in the homogeniser, the water inlet temperature is to be set to 70-90 °C.

Safety system

In case the pressure in the fuel oil line drops, the water homogenised into the Water In Fuel emulsion will evaporate, damaging the emulsion and creating supply problems. This situation is avoided by installing a third, air driven supply pump, which keeps the pressure as long as air is left in the tank 'S', see Fig. 7.06.01.

Before the tank 'S' is empty, an alarm is given and the drain valve is opened, which will drain off the WIF and replace it with HFO or diesel oil from the service tank.

The drain system is kept at atmospheric pressure, so the water will evaporate when the hot emulsion enters the safety tank. The safety tank shall be designed accordingly.

Impact on the auxiliary systems

Please note that if the engine operates on Water In Fuel emulsion (WIF), in order to reduce the NO_x emission, the exhaust gas temperature will decrease due to the reduced air / exhaust gas ratio and the increased specific heat of the exhaust gas.

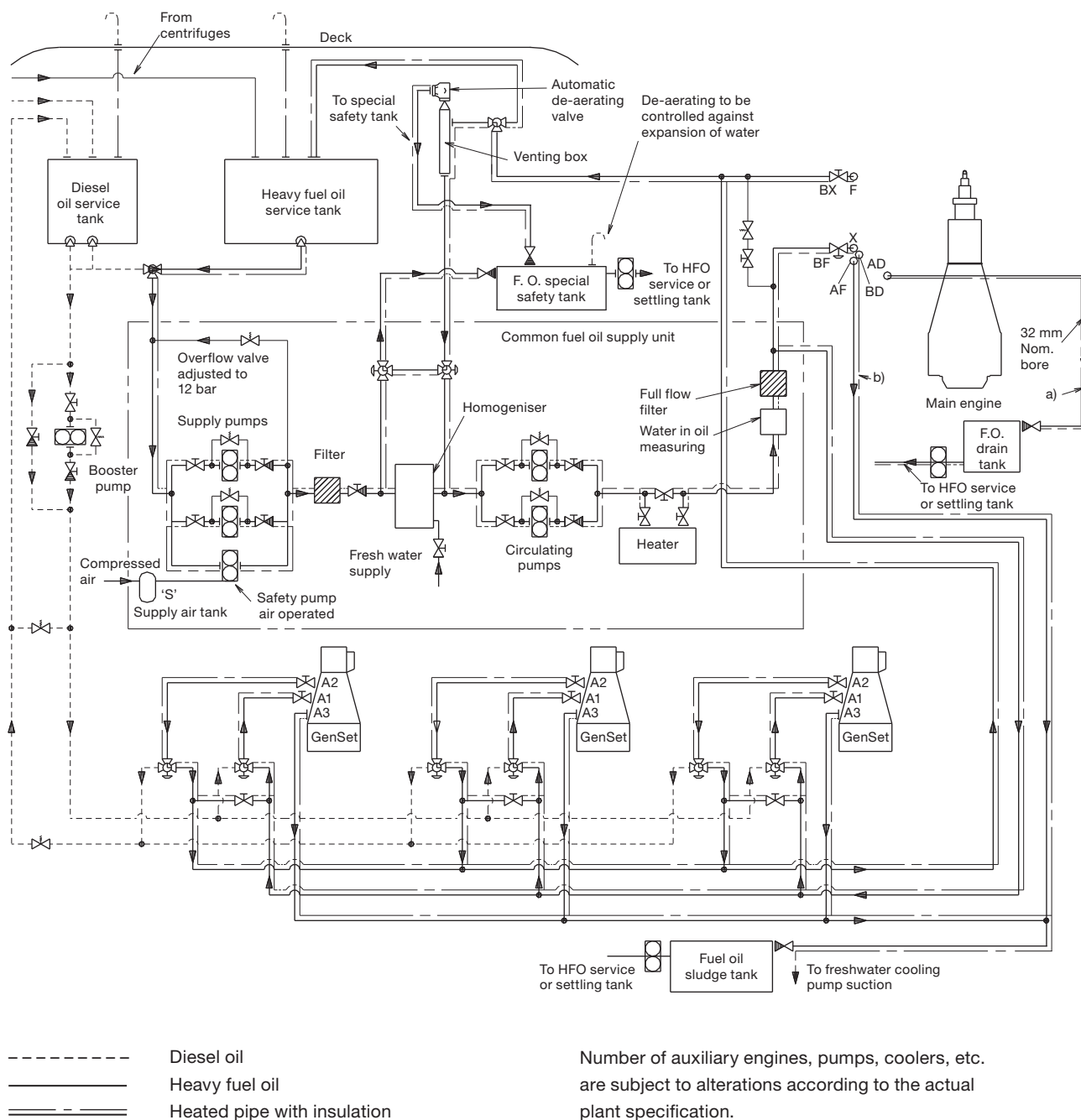
Depending on the water content, this will have an impact on the calculation and design of the following items:

- Freshwater generators
- Energy for production of freshwater
- Jacket water system
- Waste heat recovery system
- Exhaust gas boiler
- Storage tank for freshwater

For further information about emulsification of water into the fuel and use of Water In Fuel emulsion (WIF), please refer to our publication titled:

Exhaust Gas Emission Control Today and Tomorrow

The publication is available at: www.mandiesel.com under 'Quicklinks' → 'Technical Papers'



198 99 01-8.3

Fig. 7.06.01: System for emulsification of water into the fuel common to the main engine and MAN Diesel GenSets

Lubricating Oil

8

Lubricating and Cooling Oil System

The lubricating oil is pumped from a bottom tank by means of the main lubricating oil pump to the lubricating oil cooler, a thermostatic valve and, through a full-flow filter, to the engine inlet RU, Fig. 8.01.01.

RU lubricates main bearings, thrust bearing, axial vibration damper, piston cooling, crosshead bearings, crankpin bearings. It also supplies oil to the Hydraulic Power Supply unit and to moment compensator and torsional vibration damper.

From the engine, the oil collects in the oil pan, from where it is drained off to the bottom tank, see Fig. 8.06.01a and b 'Lubricating oil tank, with cofferdam'. By class demand, a cofferdam must be placed underneath the lubricating oil tank.

The engine crankcase is vented through 'AR' by a pipe which extends directly to the deck. This pipe

has a drain arrangement so that oil condensed in the pipe can be led to a drain tank, see details in Fig. 8.07.01.

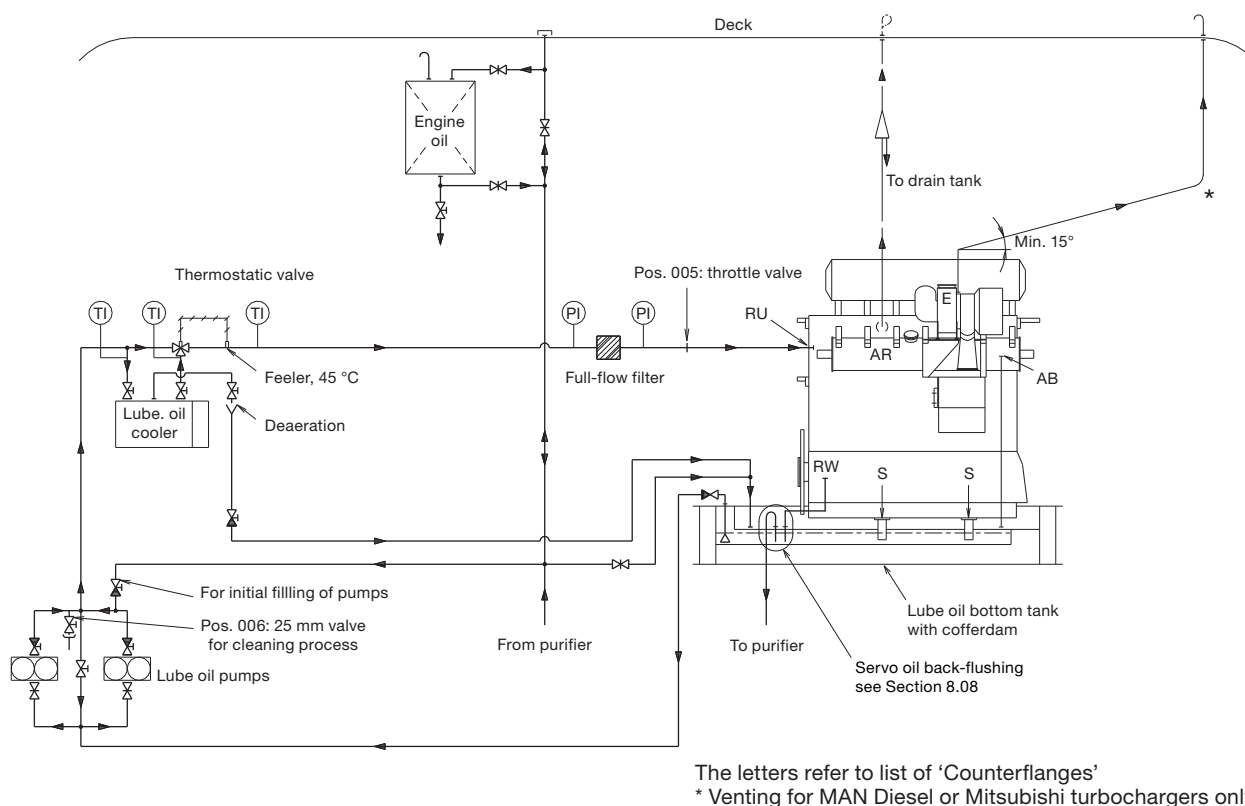
Drains from the engine bedplate 'AE' are fitted on both sides, see Fig. 8.07.02 'Bedplate drain pipes'.

For external pipe connections, we prescribe a maximum oil velocity of 1.8 m/s.

Lubrication of turbochargers

Turbochargers with slide bearings are normally lubricated from the main engine system. AB is outlet from the turbocharger, see Figs. 8.03.01 to 8.03.04, which are shown with sensors for UMS.

Figs. 8.03.01 to 8.03.04 show the lube oil pipe arrangements for different turbocharger makes.



198 99 84-4.5

Fig. 8.01.01 Lubricating and cooling oil system

Hydraulic Power Supply Unit

Internally on the engine RU is connected to the Hydraulic Power Supply unit (HPS) which supplies the hydraulic oil to the Hydraulic Cylinder Units (HCUs). The HPS unit can be either mounted onto the engine and engine driven (EoD 4 40 160) or delivered separately electrically driven, option 4 40 660. See figs. 16.01.02 and 16.01.03 respectively.

The hydraulic power supply unit shown in Fig. 8.02.01, consists of:

- an automatic main filter with a redundancy filter, in parallel
- two electrically driven pumps
- three engine driven pumps
- an safety and accumulator block

RW is the oil outlet from the automatic backflushing filter.

At start one of the two electrically driven start-up pumps is activated, and it is stopped as soon as the three engine driven pumps have taken over the hydraulic oil supply.

The hydraulic oil is supplied to the Hydraulic Cylinder Units (HCU) located at each cylinder, where it is diverted to the electronic Fuel Injection system, and to the electronic exhaust Valve Activation (FIVA) system, which perform the fuel injection and opens the exhaust valve. The exhaust valve is closed by the conventional 'air spring'.

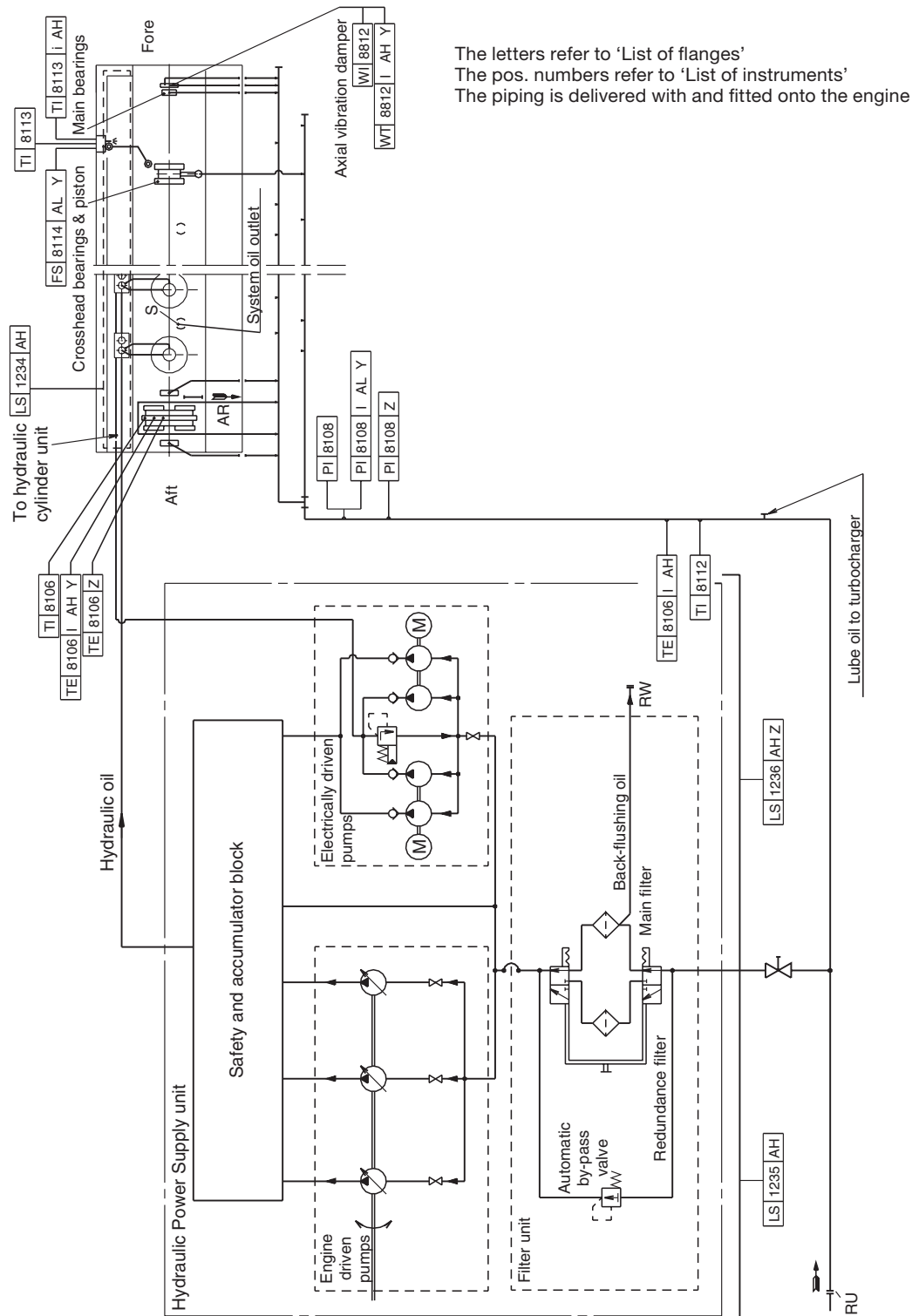
The electronic signals to the FIVA valves are given by the Engine Control System, see Chapter 16, Engine Control System (ECS).

The Hydraulic power supply is available in 2 versions

The standard version, EoD 4 40 660, is the classic ME power supply where the hydraulic power is generated by engine driven pumps and start up pressure is created by electric driven start pumps. The capacity of the start up pumps is only sufficient to make the start up pressure. The engine can not run with the engine driven pumps out of operation.

The optional version, EoD 4 40 661 is similar to the standard version, but the electric driven start up pumps have a capacity sufficient to give Take Home power at least 15% engine power. The electric power consumption should be taken into consideration in the specification of the auxiliary machinery capacity.

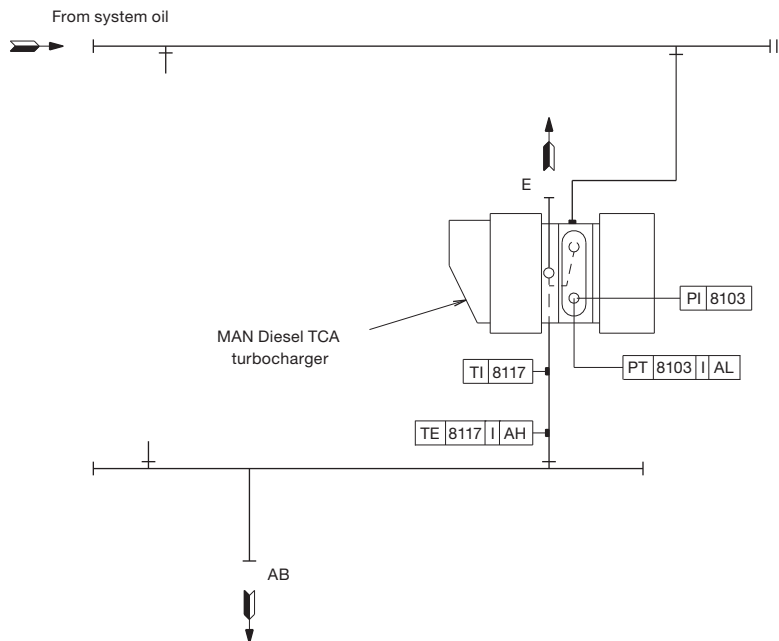
Hydraulic power supply unit, Engine Driven



178 48 13-4.1

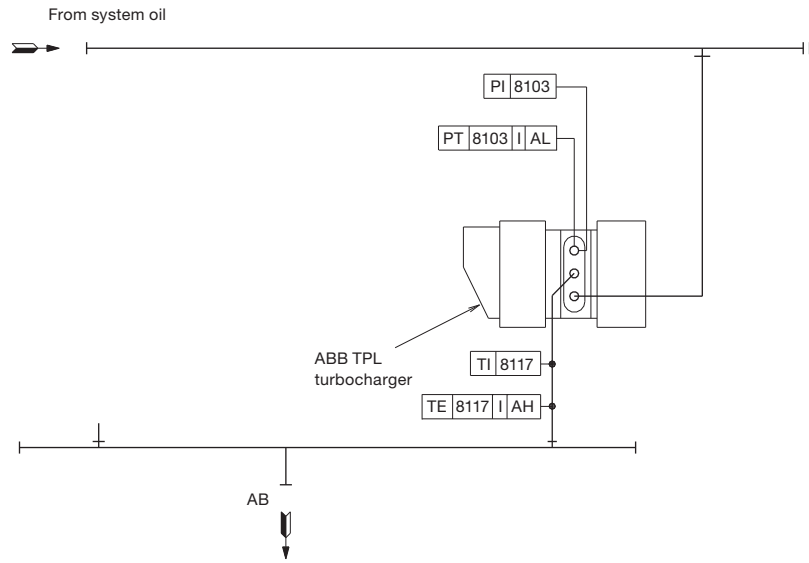
Fig. 8.02.01: Engine driven hydraulic power supply unit

Lubricating Oil Pipes for Turbochargers



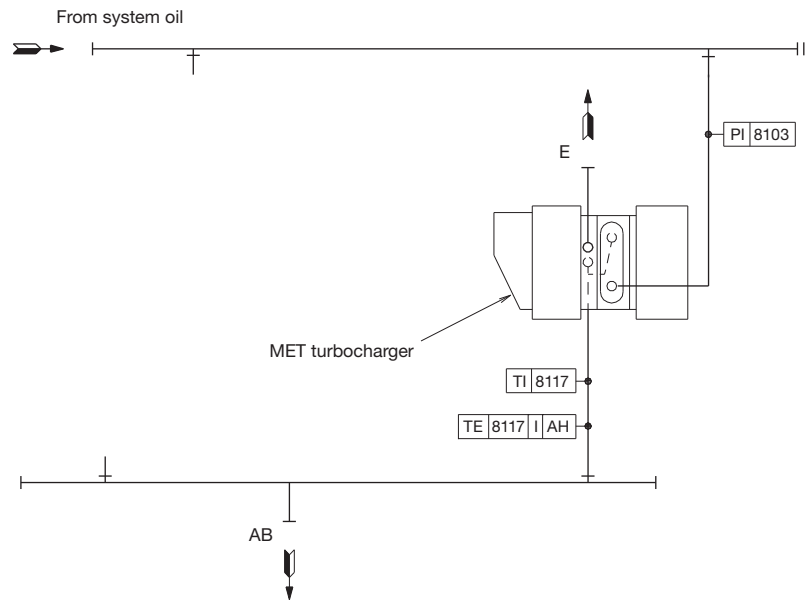
121 14 96-6.1.0

Fig. 8.03.01: MAN Diesel turbocharger type TCA



126 40 85-8.3.0

Fig. 8.03.02: ABB turbocharger type TPL



126 40 87-1.2.0

Fig. 8.03.03: Mitsubishi turbocharger type MET

Lubricating Oil Centrifuges and List of Lubricating Oils

For Unattended Machinery Spaces (UMS), automatic centrifuges with total discharge or partial discharge are to be used. Manual cleaning centrifuges can only be used for Attended Machinery Spaces (AMS).

The nominal capacity of the centrifuge is to be according to the supplier's recommendation for lubricating oil, based on the figure:

0.136 litre/kWh

The Nominal MCR is used as the total installed power.

List of lubricating oils

The circulating oil (lubricating and cooling oil) must be of the rust and oxidation inhibited type of oil of SAE 30 viscosity grade.

In order to keep the crankcase and piston cooling spaces clean of deposits, the oil should have adequate dispersion and detergent properties.

Alkaline circulating oils are generally superior in this respect.

The oils listed below have all given long-term satisfactory service in MAN B&W engine installations:

Company	Circulating oil SAE 30, BN 5-10
BP	Energol OE-HT 30
Castrol	CDX 30
Chevron *)	Veritas 800 Marine 30
ExxonMobil	Mobilgard 300
Shell	Melina 30 / S 30
Total	Atlanta Marine D 3005

*) Includes Caltex, Chevron and Texaco

Also other brands have been used with satisfactory results.

Components for Lubricating Oil System

Lubricating oil pump

The lubricating oil pump can be of the displacement wheel, or the centrifugal type:

Lubricating oil viscosity, specified...75 cSt at 50 °C
 Lubricating oil viscosity..... maximum 400 cSt *
 Lubricating oil flow see 'List of capacities'
 Design pump head 4.6 bar
 Delivery pressure 4.6 bar
 Max. working temperature 70 °C

** 400 cSt is specified, as it is normal practice when starting on cold oil, to partly open the bypass valves of the lubricating oil pumps, so as to reduce the electric power requirements for the pumps.*

The flow capacity must be within a range from 100 to 112% of the capacity stated.

The pump head is based on a total pressure drop across cooler and filter of maximum 1 bar.

Referring to Fig. 8.01.01, the bypass valve shown between the main lubricating oil pumps may be omitted in cases where the pumps have a built-in bypass or if centrifugal pumps are used.

If centrifugal pumps are used, it is recommended to install a throttle valve at position '005' to prevent an excessive oil level in the oil pan if the centrifugal pump is supplying too much oil to the engine.

During trials, the valve should be adjusted by means of a device which permits the valve to be closed only to the extent that the minimum flow area through the valve gives the specified lubricating oil pressure at the inlet to the engine at full normal load conditions. It should be possible to fully open the valve, e.g. when starting the engine with cold oil.

It is recommended to install a 25 mm valve (pos. 006), with a hose connection after the main lubricating oil pumps, for checking the cleanliness of the lubricating oil system during the flushing procedure. The valve is to be located on the underside of a horizontal pipe just after the discharge from the lubricating oil pumps.

Lubricating oil cooler

The lubricating oil cooler must be of the shell and tube type made of seawater resistant material, or a plate type heat exchanger with plate material of titanium, unless freshwater is used in a central cooling water system.

Lubricating oil viscosity, specified...75 cSt at 50 °C
 Lubricating oil flow see 'List of capacities'
 Heat dissipation see 'List of capacities'
 Lubricating oil temperature, outlet cooler..... 45 °C
 Working pressure on oil side..... 4.6 bar
 Pressure drop on oil side maximum 0.5 bar
 Cooling water flow..... see 'List of capacities'
 Cooling water temperature at inlet:
 seawater 32 °C
 freshwater..... 36 °C
 Pressure drop on water side..... maximum 0.2 bar

The lubricating oil flow capacity must be within a range from 100 to 112% of the capacity stated.

The cooling water flow capacity must be within a range from 100 to 110% of the capacity stated.

To ensure the correct functioning of the lubricating oil cooler, we recommend that the seawater temperature is regulated so that it will not be lower than 10 °C.

The pressure drop may be larger, depending on the actual cooler design.

Lubricating oil temperature control valve

The temperature control system can, by means of a three-way valve unit, by-pass the cooler totally or partly.

Lubricating oil viscosity, specified....75 cSt at 50 °C
 Lubricating oil flow see 'List of capacities'
 Temperature range, inlet to engine 40 - 47 °C

Lubricating oil full flow filter

Lubricating oil flow see 'List of capacities'
 Working pressure 4.6 bar
 Test pressure according to class rules
 Absolute fineness 50 μm *
 Working temperature approximately 45 °C
 Oil viscosity at working temp. 90 - 100 cSt
 Pressure drop with clean filter maximum 0.2 bar
 Filter to be cleaned
 at a pressure drop maximum 0.5 bar

* *The absolute fineness corresponds to a nominal fineness of approximately 35 μm at a retaining rate of 90%.*

The flow capacity must be within a range from 100 to 112% of the capacity stated.

The full-flow filter should be located as close as possible to the main engine.

If a double filter (duplex) is installed, it should have sufficient capacity to allow the specified full amount of oil to flow through each side of the filter at a given working temperature with a pressure drop across the filter of maximum 0.2 bar (clean filter).

If a filter with a back-flushing arrangement is installed, the following should be noted:

- The required oil flow, specified in the 'List of capacities', should be increased by the amount of oil used for the back-flushing, so that the lubricating oil pressure at the inlet to the main engine can be maintained during cleaning.
- If an automatically cleaned filter is installed, it should be noted that in order to activate the cleaning process, certain makes of filter require a higher oil pressure at the inlet to the filter than the pump pressure specified. Therefore, the pump capacity should be adequate for this purpose, too.

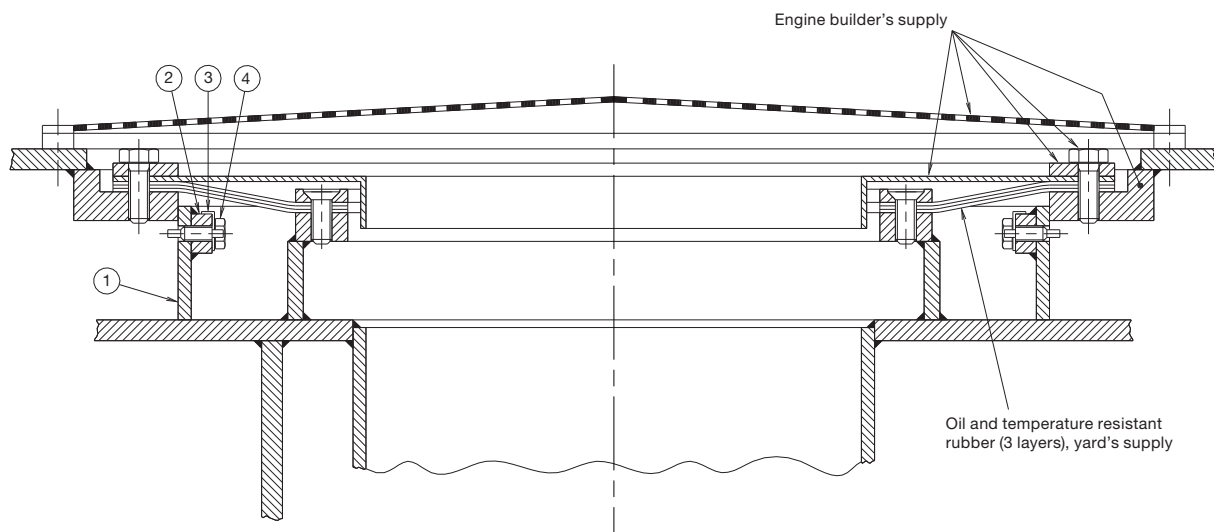
Flushing of lube oil system

Before starting the engine for the first time, the lubricating oil system on board has to be cleaned in accordance with MAN Diesel's recommendations: 'Flushing of Main Lubricating Oil System', which is available on request.

Lubricating oil outlet

A protecting ring position 1-4 is to be installed if required, by class rules, and is placed loose on the tanktop and guided by the hole in the flange.

In the vertical direction it is secured by means of screw position 4, in order to prevent wear of the rubber plate.



178 07 41-6.1

Fig. 8.05.01: Lubricating oil outlet

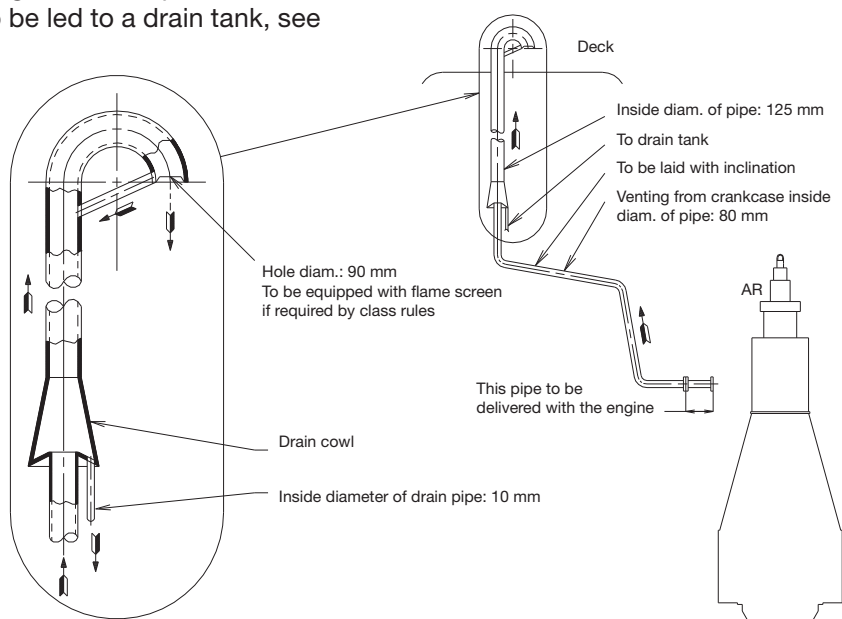
Lubricating Oil Tank

This section is available on request

Crankcase Venting and Bedplate Drain Pipes

Crankcase venting

The engine crankcase is vented through 'AR' through a pipe extending directly to the deck. This pipe has a drain arrangement that permits oil condensed in the pipe to be led to a drain tank, see Fig. 8.01.01.



198 97 10-1.4a

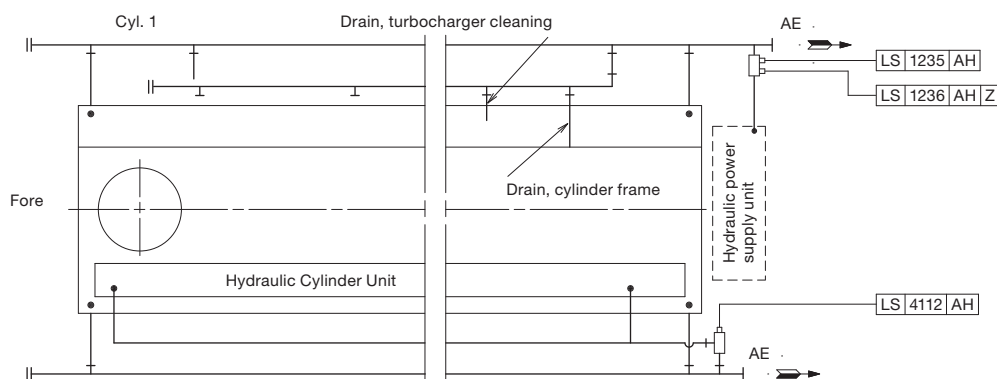
Fig. 8.07.01: Crankcase venting

Drains

Drains from the engine bedplate 'AE' are fitted on both sides of the engine, see Fig. 8.08.01.

For external pipe connections, we specify a maximum oil velocity of 1.8 m/s.

From the engine the oil collects in the oil pan from where it is drained off to the bottom tank.



121 15 35-1.2.0

Fig. 8.07.02: Bedplate drain pipes

Hydraulic Oil Back-flushing

The special suction arrangement for purifier suction in connection with the ME engine (Integrated system).

The back-flushing oil from the self cleaning 6 μm hydraulic control oil filter unit built onto the engine is contaminated and it is therefore not expedient to lead it directly into the lubricating oil sump tank.

The amount of back-flushed oil is large, and it is considered to be too expensive to discard it. Therefore, we suggest that the lubricating oil sump tank is modified for the ME engines in order not to have this contaminated lubricating hydraulic control oil mixed up in the total amount of lubricating oil. The lubricating oil sump tank is designed with a small 'back-flushing hydraulic control oil drain tank' to which the back-flushed hydraulic control oil is led and from which the lubricating oil purifier can also suck.

This is explained in detail below and the principle is shown in Fig. 8.08.01. Three suggestions for the arrangement of the drain tank in the sump tank are shown in Fig. 8.08.02 illustrates another suggestion for a back-flushing oil drain tank.

The special suction arrangement for the purifier is consisting of two connected tanks (lubricating oil sump tank and back-flushing oil drain tank) and of this reason the oil level will be the same in both tanks, as explained in detail below.

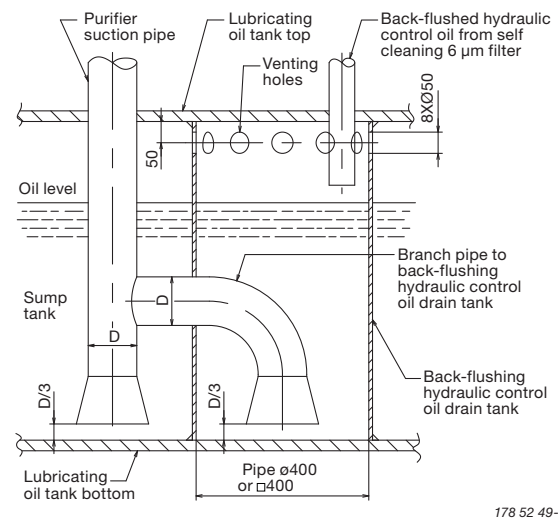
The oil level in the two tanks will be equalizing through the 'branch pipe to back-flushing oil drain tank', see Fig. 8.08.01. As the pipes have the same diameters but a different length, the resistance is larger in the 'branch pipe to back-flushing oil drain tank', and therefore the purifier will suck primarily from the sump tank.

The oil level in the sump tank and the back-flushing oil drain tank will remain to be about equal because the tanks are interconnected at the top.

When hydraulic control oil is back-flushed from the filter, it will give a higher oil level in the back-flushing hydraulic control oil drain tank and the purifier will suck from this tank until the oil level is the same in both tanks. After that, the purifier will suck from the sump tank, as mentioned above.

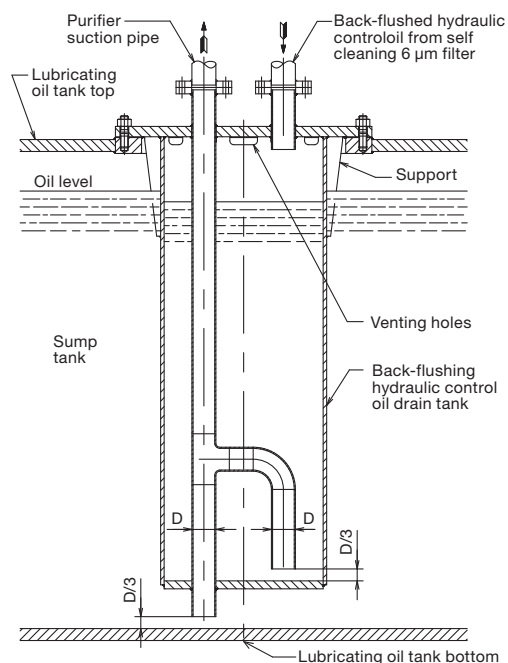
This special arrangement for purifier suction will ensure that a good cleaning effect on the lubrication oil is obtained.

If found profitable the back-flushed lubricating oil from the main lubricating oil filter (normally a 50 or 40 μm filter) can also be returned into the special back-flushing oil drain tank.



178 52 49-6.2

Fig. 8.08.01: Back-flushing servo oil drain tank



178 52 51-8.2

Fig. 8.08.02: Alternative design for the back-flushing servo oil drain tank

Separate System for Hydraulic Control Unit

As an option, the engine can be prepared for the use of a separate hydraulic control oil system Fig. 8.09.01.

The separate hydraulic control oil system can be built as a unit, or be built streamlined in the engine room with the various components placed and fastened to the steel structure of the engine room.

The design and the dimensioning of the various components are based on the aim of having a reliable system that is able to supply low-pressure oil to the inlet of the engine-mounted high-pressure hydraulic control oil pumps at a constant pressure, both at engine stand-by and at various engine loads. The quality of the hydraulic control oil must fulfil the same grade as for our standard integrated lube/cooling/hydraulic-control oil system, i.e. ISO 4406 XX/16/13 equivalent to NAS 1638 Class 7.

The hydraulic control oil system comprises:

- 1 Hydraulic control oil tank
- 2 Hydraulic control oil pumps (one for stand-by)
- 1 Pressure control valve
- 1 Hydraulic control oil cooler, water-cooled by the low temperature cooling water
- 1 Three-way valve, temperature controlled
- 1 Hydraulic control oil filter, duplex type or automatic self-cleaning type
- 1 Hydraulic control oil fine filter with pump
- 1 Temperature indicator
- 1 Pressure indicator
- 2 Level alarms
 - Valves and cocks
 - Piping.

Hydraulic control oil tank

The tank can be made of mild steel plate or be a part of the ship structure.

The tank is to be equipped with flange connections and the items listed below:

- 1 Oil filling pipe
- 1 Outlet pipe for pump suction
- 1 Return pipe from engine
- 1 Drain pipe
- 1 Vent pipe.

The hydraulic control oil tank is to be placed at least 1 m below the hydraulic oil outlet flange, RZ.

Hydraulic control oil pump

The pump must be of the displacement type (e.g. gear wheel or screw wheel pump).

The following data is specified in Fig. 8.09.02:

- Pump capacity
- Pump head
- Delivery pressure
- Working temperature
- Oil viscosity range.

Pressure control valve

The valve is to be of the self-operating flow controlling type, which bases the flow on the pre-defined pressure set point. The valve must be able to react quickly from the fully-closed to the fully-open position ($t_{\max} = 4$ sec), and the capacity must be the same as for the hydraulic control oil low-pressure pumps. The set point of the valve has to be within the adjustable range specified on a separate drawing.

The following data is specified in Fig. 8.09.02:

- Flow rate
- Adjustable differential pressure range across the valve
- Oil viscosity range.

Hydraulic control oil cooler

The cooler must be of the plate heat exchanger or shell and tube type.

The following data is specified in Fig. 8.09.02:

- Heat dissipation
- Oil flow rate
- Oil outlet temperature
- Maximum oil pressure drop across the cooler
- Cooling water flow rate
- Water inlet temperature
- Maximum water pressure drop across the cooler.

Temperature controlled three-way valve

The valve must act as a control valve, with an external sensor.

The following data is specified in Fig. 8.09.02:

- Capacity
- Adjustable temperature range
- Maximum pressure drop across the valve.

Hydraulic control oil filter

The filter is to be of the duplex full flow type with manual change over and manual cleaning or of the automatic self cleaning type.

A differential pressure gauge is fitted onto the filter

The following data is specified in Fig. 8.09.02:

- Filter capacity
- Maximum pressure drop across the filter
- Filter mesh size (absolute)
- Oil viscosity
- Design temperature.

Off-line hydraulic control oil fine filter or purifier

Fig. 8.09.01

The off-line fine filter unit or purifier must be able to treat 15-20% of the total oil volume per hour.

The fine filter is an off-line filter and removes metallic and non-metallic particles larger than 0,8 µm as well as water and oxidation. The filter has a pertaining pump and is to be fitted on the top of the hydraulic control oil tank.

A suitable fine filter unit is:

Make: CJC, C.C. Jensen A/S, Svendborg,
Denmark - www.cjc.dk.

For oil volume <10,000 litres:

HDU 27/-MZ-Z with a pump flow of 15-20% of the total oil volume per hour.

For oil volume >10,000 litres:

HDU 27/-GP-DZ with a pump flow of 15-20% of the total oil volume per hour.

Temperature indicator

The temperature indicator is to be of the liquid straight type.

Pressure indicator

The pressure indicator is to be of the dial type.

Level alarm

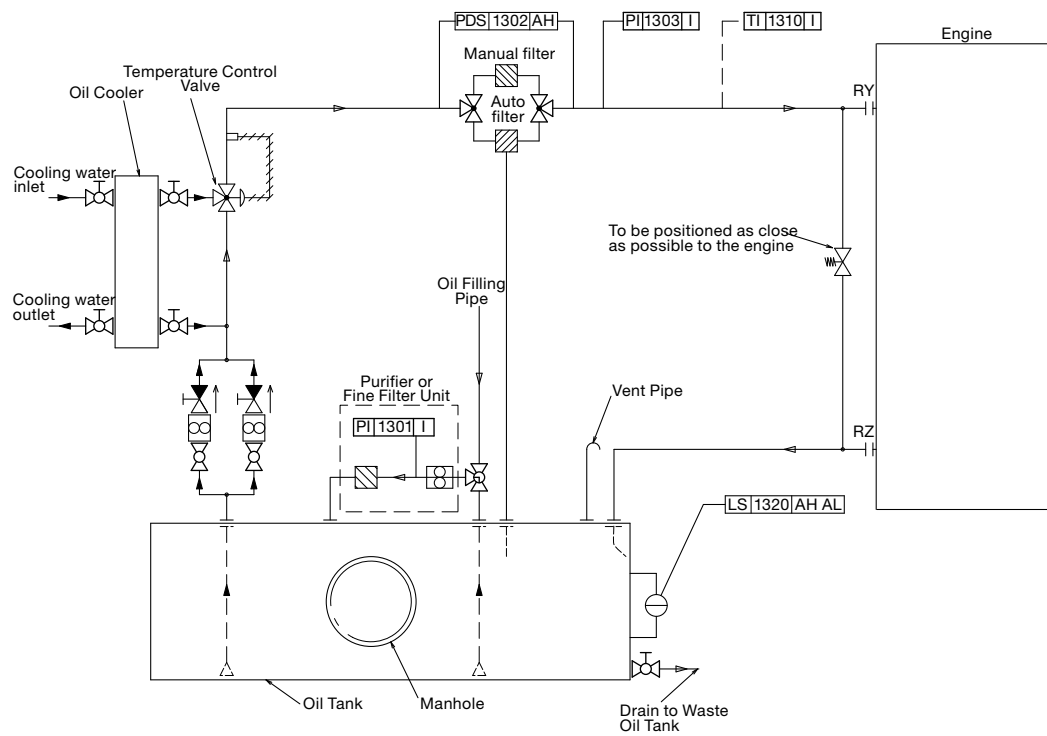
The hydraulic control oil tank has to have level alarms for high and low oil level.

Piping

The pipes can be made of mild steel.

The design oil pressure is to be 10 bar.

The return pipes are to be placed vertical or laid with a downwards inclination of minimum 15°.



178 53 39-5.0

Fig. 8.09.01: Hydraulic control oil system, manual filter

Hydraulic Control Oil System

This section is available on request

Cylinder Lubrication

Cylinder Lubricating Oil System

The cost of the cylinder lubricating oil is one of the largest contributions to total operating costs, next to the fuel oil cost. Another aspect is that the lubrication rate has a great influence on the cylinder condition, and thus on the overhauling schedules and maintenance costs.

It is therefore of the utmost importance that the cylinder lubricating oil system as well as its operation is optimised.

Cylinder oils

Cylinder oils should, preferably, be of the SAE 50 viscosity grade.

Modern high-rated two-stroke engines have a relatively great demand for detergency in the cylinder oil. Therefore cylinder oils should be chosen according to the below list.

A BN 70 cylinder oil is to be used as the default choice of oil and it may be used on all fuel types. However, in case of the engine running on fuel with sulphur content lower than 1.5% for more than 1 to 2 weeks, we recommend to change to a lower BN cylinder oil such as BN 40-50.

The cylinder oils listed below have all given long-term satisfactory service during heavy fuel operation in MAN B&W engine installations:

Company	Cylinder oil SAE 50, BN 60-80	Cylinder oil SAE 50, BN 40-50
BP	Energol CLO 50 M Energol CL 605	Energol CL 505 Energol CL-DX 405
Castrol	Cyltech 70 / 80AW	Cyltech 40 SX / 50 S
Chevron *)	Taro Special HT 70	Taro Special HT LS 40
ExxonMobil	Mobilgard 570	Mobilgard L540
Shell	Alexia 50	Alexia LS
Total	Talusia Universal Talusia HR 70	Talusia LS 40
*) Includes Caltex, Chevron and Texaco		

Also other brands have been used with satisfactory results.

Cylinder oil feed rate (dosage)

Adjustment of the cylinder oil dosage to the sulphur content in the fuel being burnt is further explained in Section 9.02.

MAN B&W Alpha Cylinder Lubrication System

The MAN B&W Alpha cylinder lubrication system, see Figs. 9.02.02a and 9.02.02b, is designed to supply cylinder oil intermittently, e.g. every four engine revolutions with electronically controlled timing and dosage at a defined position.

The cylinder lubricating oil is pumped from the cylinder oil storage tank to the service tank, the size of which depends on the owner's and the yard's requirements, - it is normally dimensioned for minimum two days' cylinder lubricating oil consumption.

Cylinder lubricating oil is fed to the Alpha cylinder lubrication system by gravity from the service tank.

The storage tank and the service tank may alternatively be one and the same tank.

The oil fed to the injectors is pressurised by means of the Alpha Lubricator which is placed on the HCU and equipped with small multi-piston pumps.

The oil pipes fitted on the engine is shown in Fig. 9.02.04.

The whole system is controlled by the Cylinder Control Unit (CCU) which controls the injection frequency on the basis of the engine-speed signal given by the tachometer signal and the fuel index.

Prior to start-up, the cylinders can be pre-lubricated and, during the running-in period, the operator can choose to increase the lubricating oil feed rate to a max. setting of 200%.

The MAN B&W Alpha Cylinder Lubricator is preferably to be controlled in accordance with the Alpha ACC (Adaptive Cylinder oil Control) feed rate system.

The yard supply should be according to the items shown in Fig. 9.02.02a within the broken line. With regard to the filter and the small box, please see Fig. 9.02.05.

Alpha Adaptive Cylinder Oil Control (Alpha ACC)

It is a well-known fact that the actual need for cylinder oil quantity varies with the operational conditions such as load and fuel oil quality. Consequently, in order to perform the optimal lubrication – cost-effectively as well as technically – the cylinder lubricating oil dosage should follow such operational variations accordingly.

The Alpha lubricating system offers the possibility of saving a considerable amount of cylinder lubricating oil per year and, at the same time, to obtain a safer and more predictable cylinder condition.

Working principle

The basic feed rate control should be adjusted in relation to the actual fuel quality and amount being burnt at any given time. The sulphur percentage is a good indicator in relation to wear, and an oil dosage proportional to the sulphur level will give the best overall cylinder condition.

The following two criteria determine the control:

- The cylinder oil dosage shall be proportional to the sulphur percentage in the fuel
- The cylinder oil dosage shall be proportional to the engine load (i.e. the amount of fuel entering the cylinders).

The implementation of the above two criteria will lead to an optimal cylinder oil dosage, proportional to the amount of sulphur entering the cylinders.

Basic and minimum setting with Alpha ACC

The recommendations are valid for all plants, whether controllable pitch or fixed pitch propellers are used.

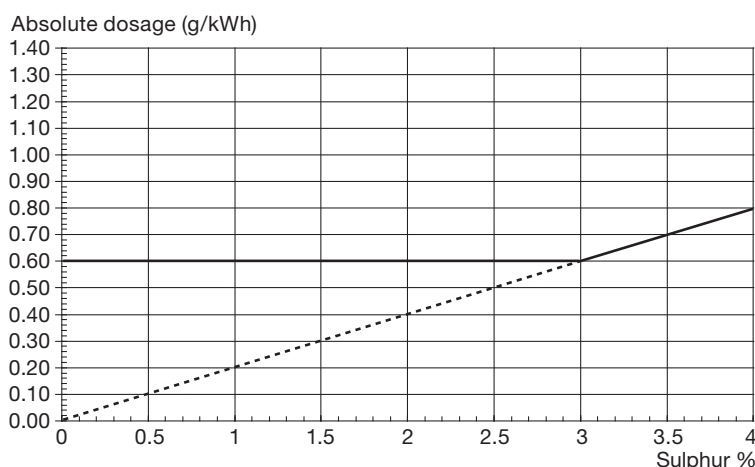
Safe and very lubricating-economical control after running-in is obtained with a basic setting according to the formula:

Basic lubricating oil setting = $0.20 \text{ g/kWh} \times \text{S}\%$

with a minimum setting of 0.60 g/kWh , i.e. the setting should be kept constant from about 3% sulphur and downwards.

Due to the sulphur dependency, the average cylinder oil dosages rely on the sulphur distribution in worldwide fuel bunkers. Based on deliveries all over the world, the resulting yearly specific cylinder oil dosage is close to 0.65 g/kWh .

Further information on cylinder oil as a function of fuel oil sulphur content and alkalinity of lubricating oil is available from MAN Diesel.



178 61 19-6.0

Fig 9.02.01: Cylinder lubricating oil dosage with Alpha ACC at all loads (BN 70 cylinder oil) after running-in

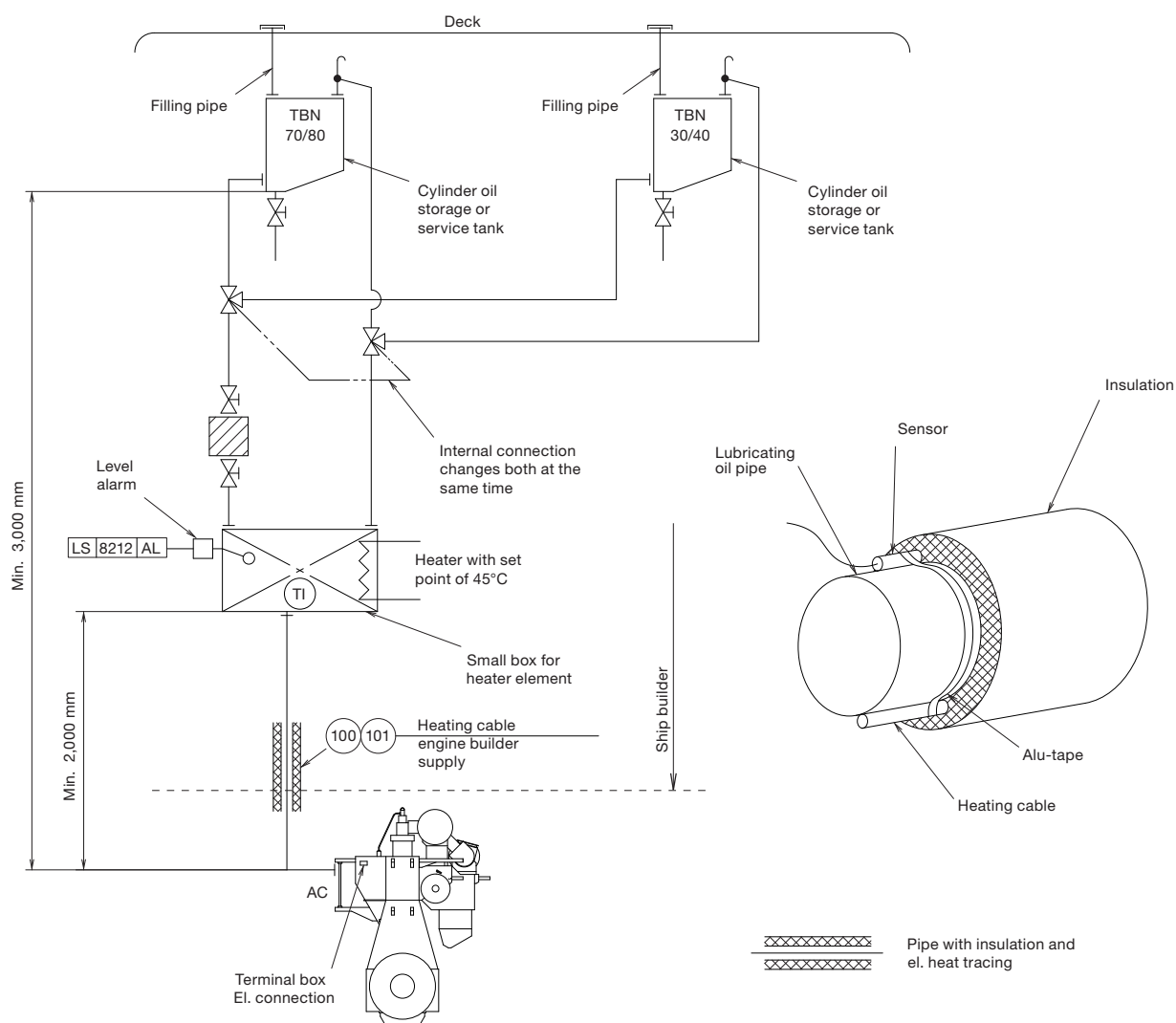
Cylinder Oil Pipe Heating

In case of low engine room temperature, it can be difficult to keep the cylinder oil temperature at 45 °C at the MAN B&W Alpha Lubricator, mounted on the hydraulic cylinder.

Therefore the cylinder oil pipe from the small tank, see Figs. 9.02.02a and 9.02.02b, in the vessel and of the main cylinder oil pipe on the engine is insulated and electrically heated.

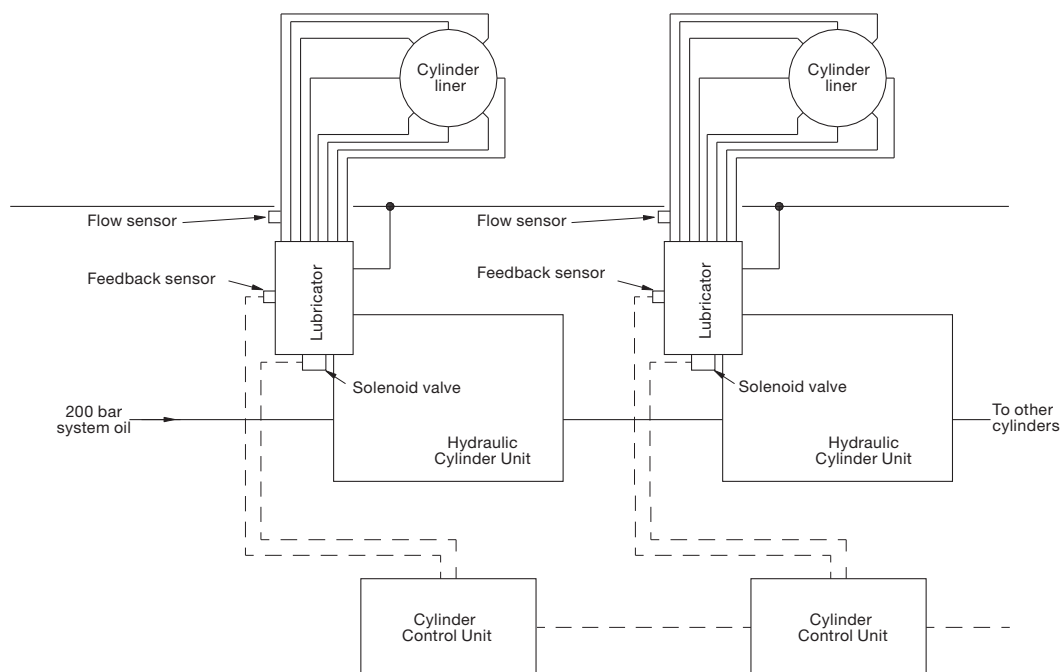
The engine builder is to make the insulation and heating on the main cylinder oil pipe on the engine. Moreover, the engine builder is to mount the junction box and the thermostat on the engine. See Fig. 9.02.03.

The ship yard is to make the insulation of the cylinder oil pipe in the engine room. The heating cable supplied by the engine builder is to be mounted from the small tank to the junction box on the engine. See Figs. 9.02.02a and 9.02.02b.



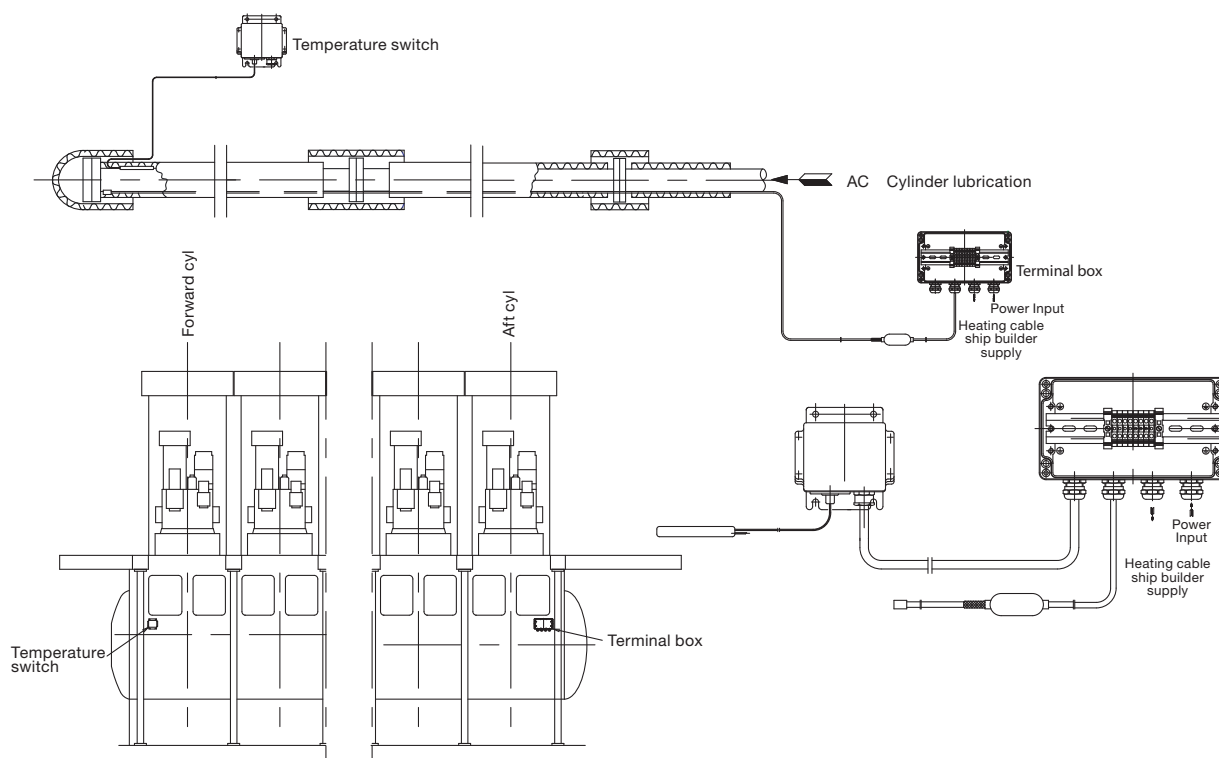
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Fig. 9.02.02a: Cylinder lubricating oil system with dual service tanks for two different TBN cylinder oils



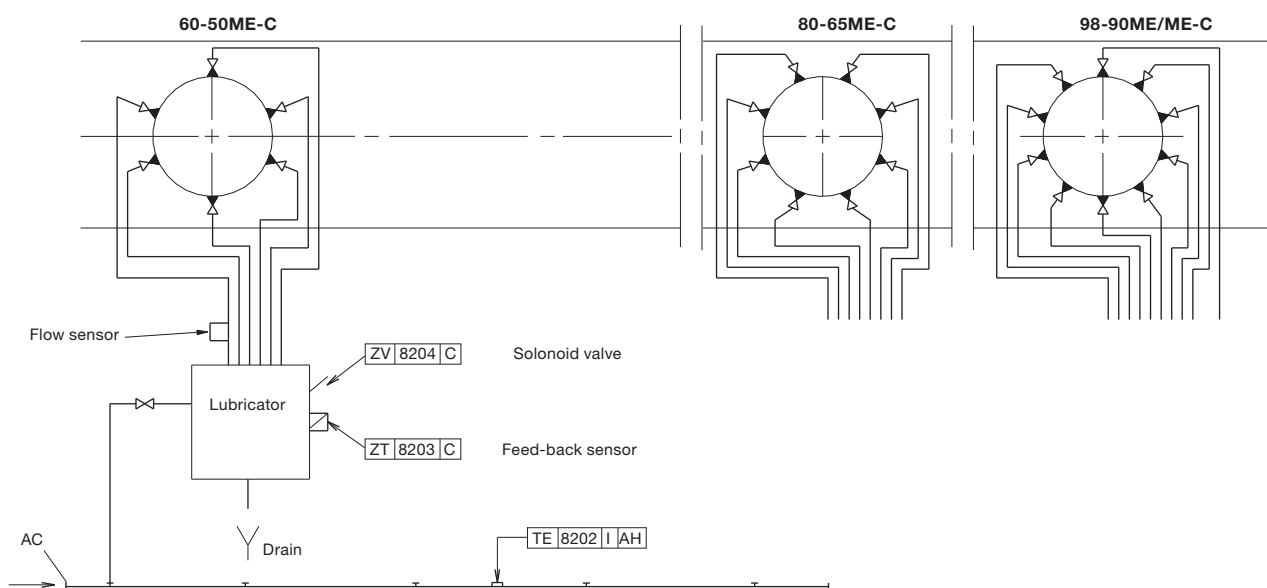
178 49 83-4.6b

Fig. 9.02.02b: Cylinder lubricating oil system. Example from 80/70/65ME-C engines



178 53 71-6.0

Fig. 9.02.03: Electric heating of cylinder oil pipes

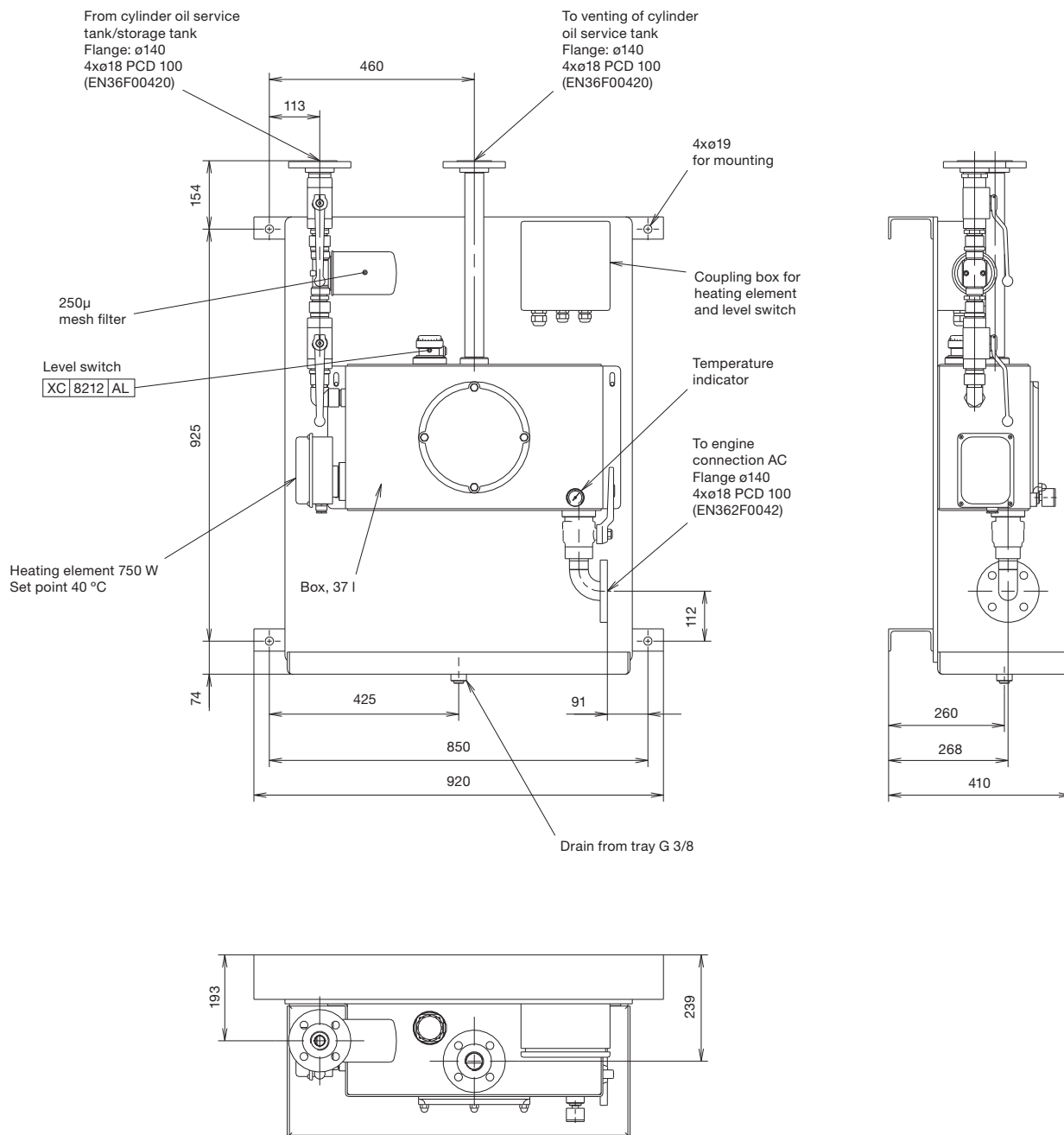


The letters refer to list of 'Counterflanges'

The item No refer to 'Guidance Values Automation'

178 54 68-8.3

Fig. 9.02.04: Cylinder lubricating oil pipes



178 52 75-8.1

Fig. 9.02.05: Suggestion for small heating box with filter

Piston Rod Stuffing
Box Drain Oil

10

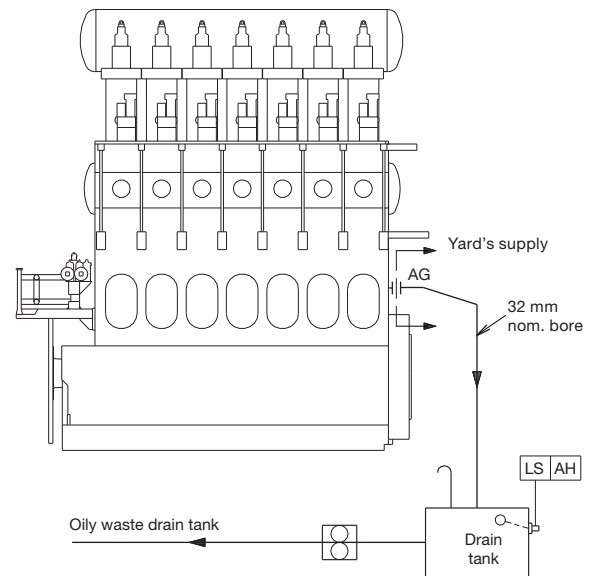
Stuffing Box Drain Oil System

For engines running on heavy fuel, it is important that the oil drained from the piston rod stuffing boxes is not led directly into the system oil, as the oil drained from the stuffing box is mixed with sludge from the scavenge air space.

The performance of the piston rod stuffing box on the engines has proved to be very efficient, primarily because the hardened piston rod allows a higher scraper ring pressure.

The amount of drain oil from the stuffing boxes is about 5 - 10 litres/24 hours per cylinder during normal service. In the running-in period, it can be higher.

The relatively small amount of drain oil is led to the general oily waste drain tank or is burnt in the incinerator, Fig. 10.01.01. (Yard's supply).



198 97 44-8.1

Fig. 10.01.01: Stuffing box drain oil system

Central Cooling Water System

11

Central Cooling Water System

The water cooling can be arranged in several configurations, the most common system choice being a Central cooling water system.

Advantages of the central cooling system:

- Only one heat exchanger cooled by seawater, and thus, only one exchanger to be overhauled
- All other heat exchangers are freshwater cooled and can, therefore, be made of a less expensive material
- Few non-corrosive pipes to be installed
- Reduced maintenance of coolers and components
- Increased heat utilisation.

Disadvantages of the central cooling system:

- Three sets of cooling water pumps (seawater, central water and jacket water.
- Higher first cost.

For information on the alternative Seawater Cooling System, see Chapter 12.

An arrangement common for the main engine and MAN Diesel auxiliary engines is available on request.

For further information about common cooling water system for main engines and auxiliary engines please refer to our publication:

Uni-concept Auxiliary Systems for Two-stroke Main

The publication is available at www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

Central Cooling Water System

The central cooling water system is characterised by having only one heat exchanger cooled by seawater, and by the other coolers, including the jacket water cooler, being cooled by central cooling water.

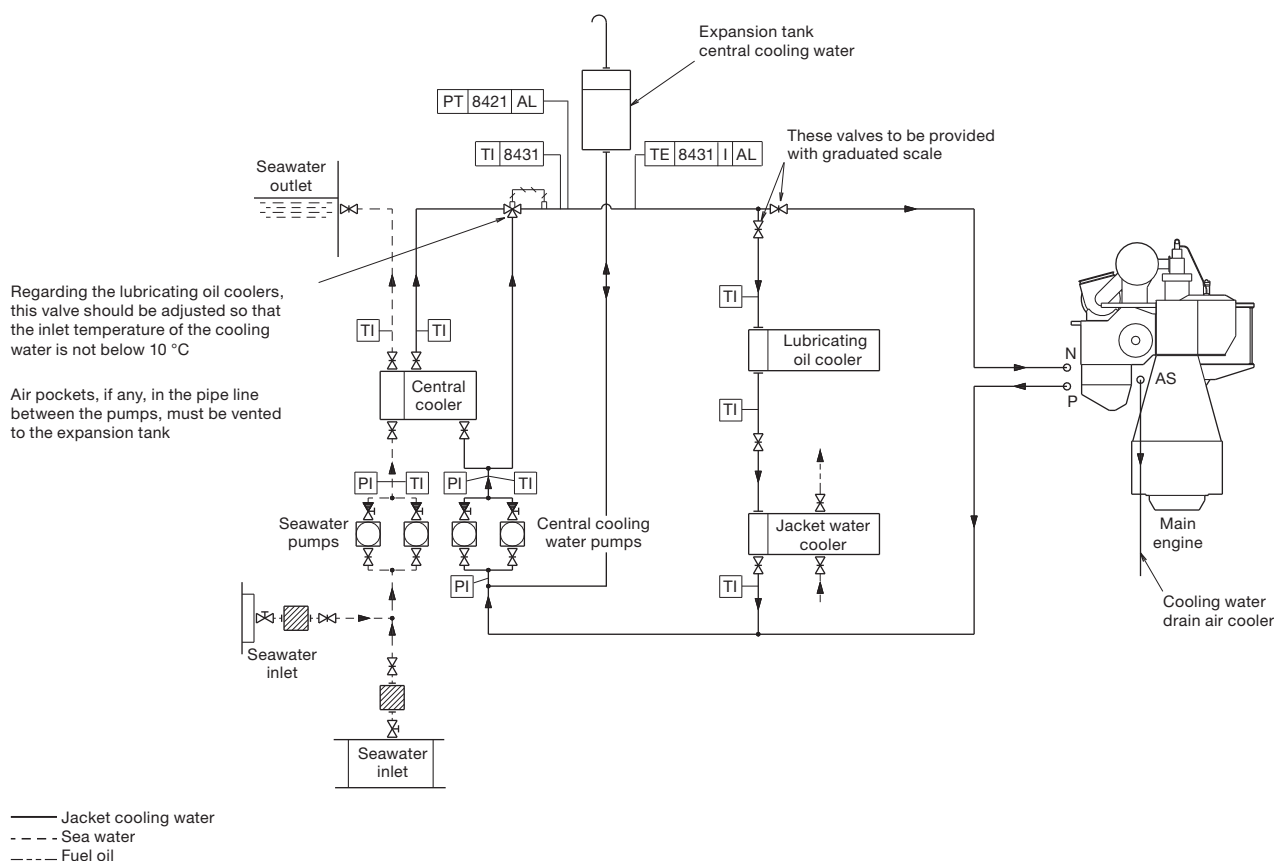
In order to prevent too high a scavenge air temperature, the cooling water design temperature in the central cooling water system is normally 36 °C, corresponding to a maximum seawater temperature of 32 °C.

Our recommendation of keeping the cooling water inlet temperature to the main engine scavenge

air cooler as low as possible also applies to the central cooling system. This means that the temperature control valve in the central cooling water circuit is to be set to minimum 10 °C, whereby the temperature follows the outboard seawater temperature when central cooling water temperature exceeds 10 °C.

For external pipe connections, we prescribe the following maximum water velocities:

Jacket water	3.0 m/s
Central cooling water	3.0 m/s
Seawater	3.0 m/s



The letters refer to list of 'Counterflanges', Fig. 5.10.01
 The item No. refer to 'Guidance values automation'

178 52 77-1.1

Fig. 11.02.01: Central cooling water system

Components for Central Cooling Water System

Seawater cooling pumps

The pumps are to be of the centrifugal type.

Seawater flow see 'List of Capacities'
 Pump head 2.5 bar
 Test pressure according to class rules
 Working temperature, normal 0-32 °C
 Working temperature maximum 50 °C

The flow capacity must be within a range from 100 to 110% of the capacity stated.

The differential pressure of the pumps is to be determined on the basis of the total actual pressure drop across the cooling water system.

Central cooler

The cooler is to be of the shell and tube or plate heat exchanger type, made of seawater resistant material.

Heat dissipation see 'List of Capacities'
 Central cooling water flow see 'List of Capacities'
 Central cooling water temperature, outlet 36 °C
 Pressure drop on central cooling side max. 0.2 bar
 Seawater flow see 'List of Capacities'
 Seawater temperature, inlet 32 °C
 Pressure drop on seawater side maximum 0.2 bar

The pressure drop may be larger, depending on the actual cooler design.

The heat dissipation and the seawater flow figures are based on MCR output at tropical conditions, i.e. a seawater temperature of 32 °C and an ambient air temperature of 45 °C.

Overload running at tropical conditions will slightly increase the temperature level in the cooling system, and will also slightly influence the engine performance.

Central cooling water pumps

The pumps are to be of the centrifugal type.

Central cooling water flow... see 'List of Capacities'
 Pump head 2.5 bar
 Delivery pressure depends on location of expansion tank
 Test pressure according to class rules
 Working temperature 80 °C
 Design temperature 100 °C

The flow capacity must be within a range from 100 to 110% of the capacity stated.

The 'List of Capacities' covers the main engine only. The differential pressure provided by the pumps is to be determined on the basis of the total actual pressure drop across the cooling water system.

Central cooling water thermostatic valve

The low temperature cooling system is to be equipped with a three-way valve, mounted as a mixing valve, which by-passes all or part of the fresh water around the central cooler.

The sensor is to be located at the outlet pipe from the thermostatic valve and is set so as to keep a temperature level of minimum 10 °C.

Jacket water system

Due to the central cooler the cooling water inlet temperature is about 4 °C higher for this system compared to the seawater cooling system. The input data are therefore different for the scavenge air cooler, the lube oil cooler and the jacket water cooler.

The heat dissipation and the central cooling water flow figures are based on an MCR output at tropical conditions, i.e. a maximum seawater temperature of 32 °C and an ambient air temperature of 45 °C.

Jacket water cooling pump

The pumps are to be of the centrifugal type.
 Jacket water flow see 'List of Capacities'
 Pump head 3.0 bar
 Delivery pressure depends on location of expansion tank
 Test pressure according to class rules
 Working temperature 80 °C
 Design temperature 100 °C

The flow capacity must be within a range from 100 to 110% of the capacity stated.

The stated of capacities cover the main engine only. The pump head of the pumps is to be determined on the basis of the total actual pressure drop across the cooling water system.

Scavenge air cooler

The scavenge air cooler is an integrated part of the main engine.

Heat dissipation see 'List of Capacities'
 Central cooling water flow see 'List of Capacities'
 Central cooling temperature, inlet 36 °C
 Pressure drop on FW-LT water side approx. 0.5 bar

Lubricating oil cooler

See Chapter 8 'Lubricating Oil'.

Jacket water cooler

The cooler is to be of the shell and tube or plate heat exchanger type.

Heat dissipation see 'List of Capacities'
 Jacket water flow see 'List of Capacities'
 Jacket water temperature, inlet 80 °C
 Pressure drop on jacket water side max. 0.2 bar
 Central cooling water flow ... see 'List of Capacities'
 Central cooling water temperature, inlet approx. 42 °C
 Pressure drop on Central cooling water side max. 0.2 bar

The other data for the jacket cooling water system can be found in chapter 12.

For further information about a common cooling water system for main engines and MAN Diesel auxiliary engines, please refer to our publication:

Uni-concept Auxiliary Systems for Two-stroke Main

The publication is available at www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

Seawater Cooling System

12

Seawater Systems

The water cooling can be arranged in several configurations, the most simple system choices being seawater and central cooling water system:

- A **seawater cooling system** and a jacket cooling water system
- The advantages of the seawater cooling system are mainly related to first cost, viz:
- Only two sets of cooling water pumps (seawater and jacket water)
- Simple installation with few piping systems.

Whereas the disadvantages are:

- Seawater to all coolers and thereby higher maintenance cost
- Expensive seawater piping of non-corrosive materials such as galvanised steel pipes or Cu-Ni pipes.

Seawater Cooling System

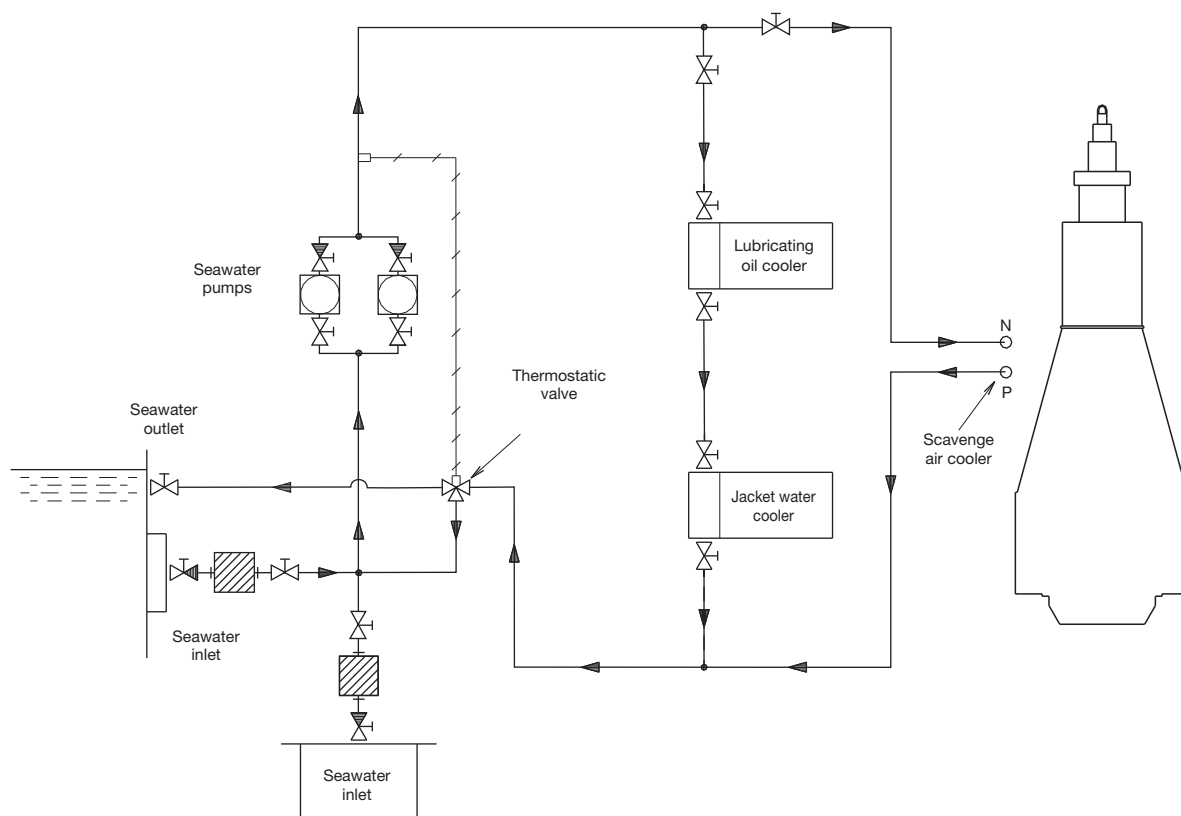
The seawater cooling system is used for cooling, the main engine lubricating oil cooler, the jacket water cooler and the scavenge air cooler, see Fig. 12.02.01.

The lubricating oil cooler for a PTO step-up gear should be connected in parallel with the other coolers. The capacity of the seawater pump is based on the outlet temperature of the seawater being maximum 50 °C after passing through the coolers – with an inlet temperature of maximum 32 °C (tropical conditions), i.e. a maximum temperature increase of 18 °C.

The valves located in the system fitted to adjust the distribution of cooling water flow are to be provided with graduated scales.

The inter-related positioning of the coolers in the system serves to achieve:

- The lowest possible cooling water inlet temperature to the lubricating oil cooler in order to obtain the cheapest cooler. On the other hand, in order to prevent the lubricating oil from stiffening in cold services, the inlet cooling water temperature should not be lower than 10 °C
- The lowest possible cooling water inlet temperature to the scavenge air cooler, in order to keep the fuel oil consumption as low as possible.

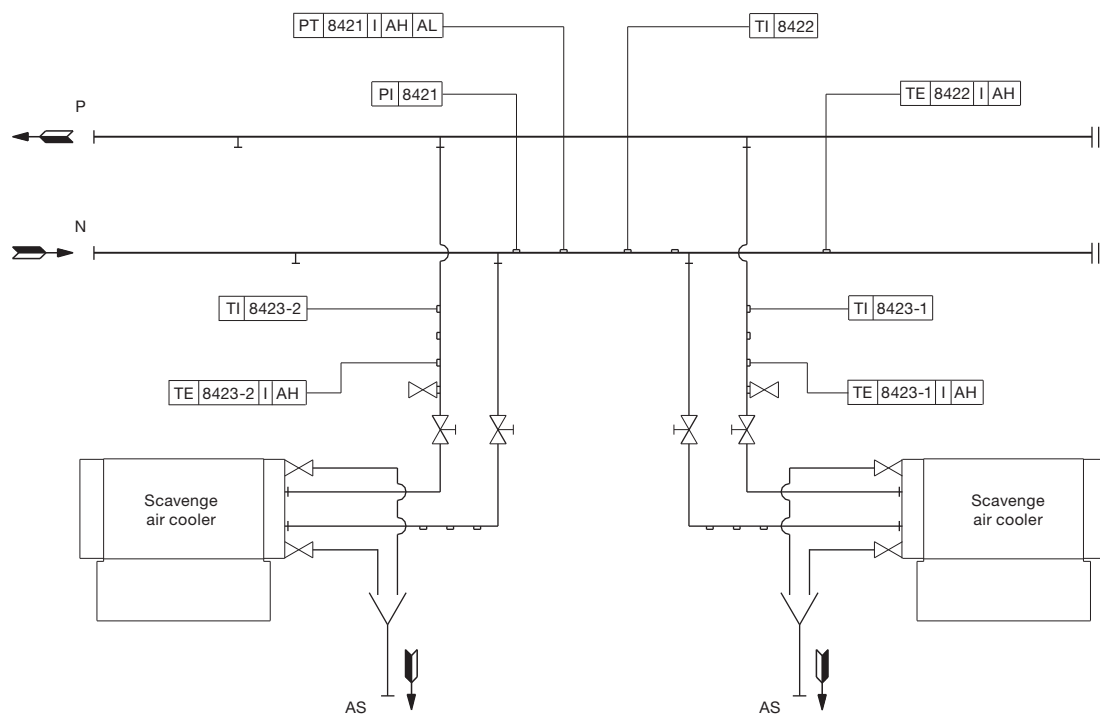


198 98 13-2.5

The letters refer to list of 'Counterflanges'

Fig. 12.02.01: Seawater cooling system

Seawater Cooling Pipes



178 50 37-5.1

The letters refer to list of 'Counterflanges'
 The item No. refer to 'Guidance values automation'

Fig. 12.03.01: Seawater cooling pipes for engines with two or more turbochargers

Components for Seawater Cooling System

Seawater cooling pump

The pumps are to be of the centrifugal type.

Seawater flow see 'List of Capacities'
 Pump head 2.5 bar
 Test pressure according to class rule
 Working temperature maximum 50 °C

The flow capacity must be within a range from 100 to 110% of the capacity stated.

Lubricating oil cooler

See Chapter 8 'Lubricating Oil'.

Jacket water cooler

The cooler is to be of the shell and tube or plate heat exchanger type, made of seawater resistant material.

Heat dissipation see 'List of Capacities'
 Jacket water flow see 'List of Capacities'
 Jacket water temperature, inlet 80 °C
 Pressure drop
 on jacket water side maximum 0.2 bar
 Seawater flow see 'List of Capacities'
 Seawater temperature, inlet 38 °C
 Pressure drop on
 seawater side maximum 0.2 bar

The heat dissipation and the seawater flow are based on an MCR output at tropical conditions, i.e. seawater temperature of 32 °C and an ambient air temperature of 45 °C.

Scavenge air cooler

The scavenge air cooler is an integrated part of the main engine.

Heat dissipation see 'List of Capacities'
 Seawater flow see 'List of Capacities'
 Seawater temperature,
 for seawater cooling inlet, max. 32 °C
 Pressure drop on
 cooling water side between 0.1 and 0.5 bar

The heat dissipation and the seawater flow are based on an MCR output at tropical conditions, i.e. seawater temperature of 32 °C and an ambient air temperature of 45 °C.

Seawater thermostatic valve

The temperature control valve is a three-way valve which can recirculate all or part of the seawater to the pump's suction side. The sensor is to be located at the seawater inlet to the lubricating oil cooler, and the temperature level must be a minimum of +10 °C.

Seawater flow see 'List of Capacities'
 Temperature range,
 adjustable within +5 to +32 °C

Jacket Cooling Water System

The jacket cooling water system is used for cooling the cylinder liners, cylinder covers and exhaust valves of the main engine and heating of the fuel oil drain pipes, see Fig. 12.05.01.

The jacket water pump) draws water from the jacket water cooler outlet and delivers it to the engine.

At the inlet to the jacket water cooler there is a thermostatically controlled regulating valve, with a sensor at the engine cooling water outlet, which keeps the main engine cooling water outlet at a temperature of 80 °C.

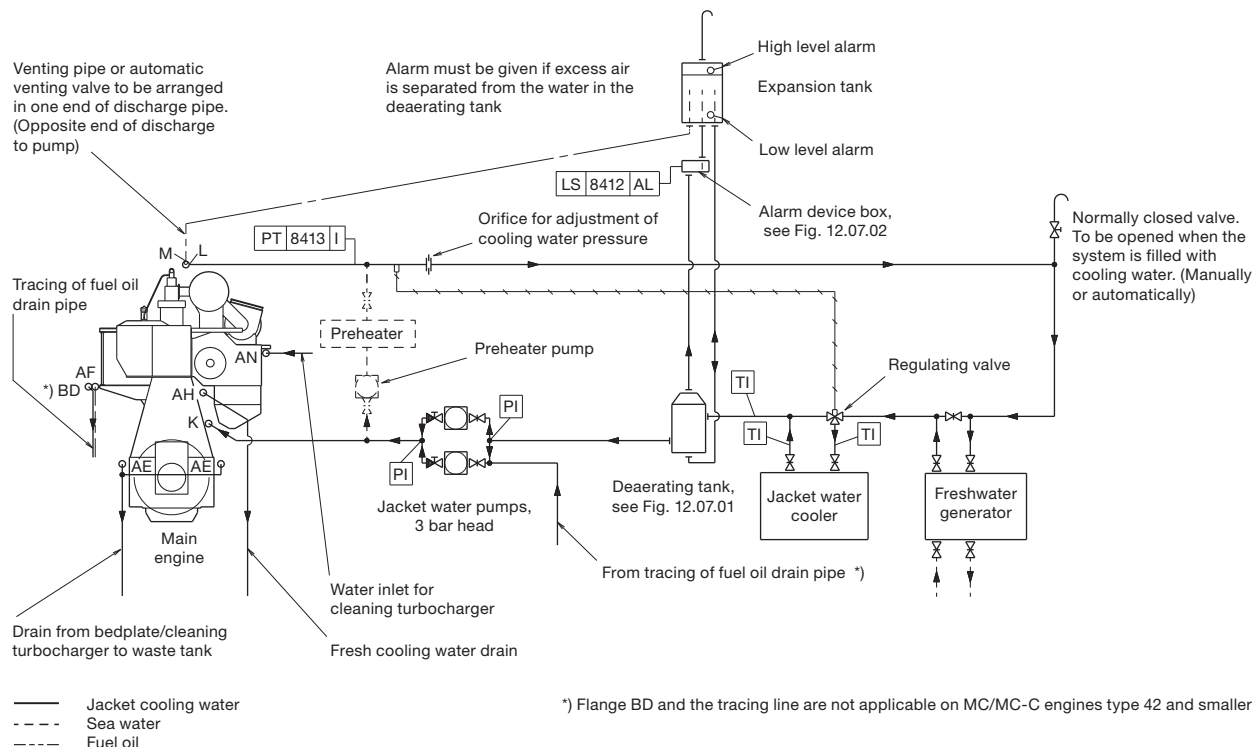
The engine jacket water must be carefully treated, maintained and monitored so as to avoid corrosion, corrosion fatigue, cavitation and scale formation. It is recommended to install a preheater if preheating is not available from the auxiliary engines jacket cooling water system.

The venting pipe in the expansion tank should end just below the lowest water level, and the expansion tank must be located at least 5 m above the engine cooling water outlet pipe.

The freshwater generator, if installed, may be connected to the seawater system if the generator does not have a separate cooling water pump. The generator must be coupled in and out slowly over a period of at least 3 minutes.

For external pipe connections, we prescribe the following maximum water velocities:

Jacket water	3.0 m/s
Seawater	3.0 m/s

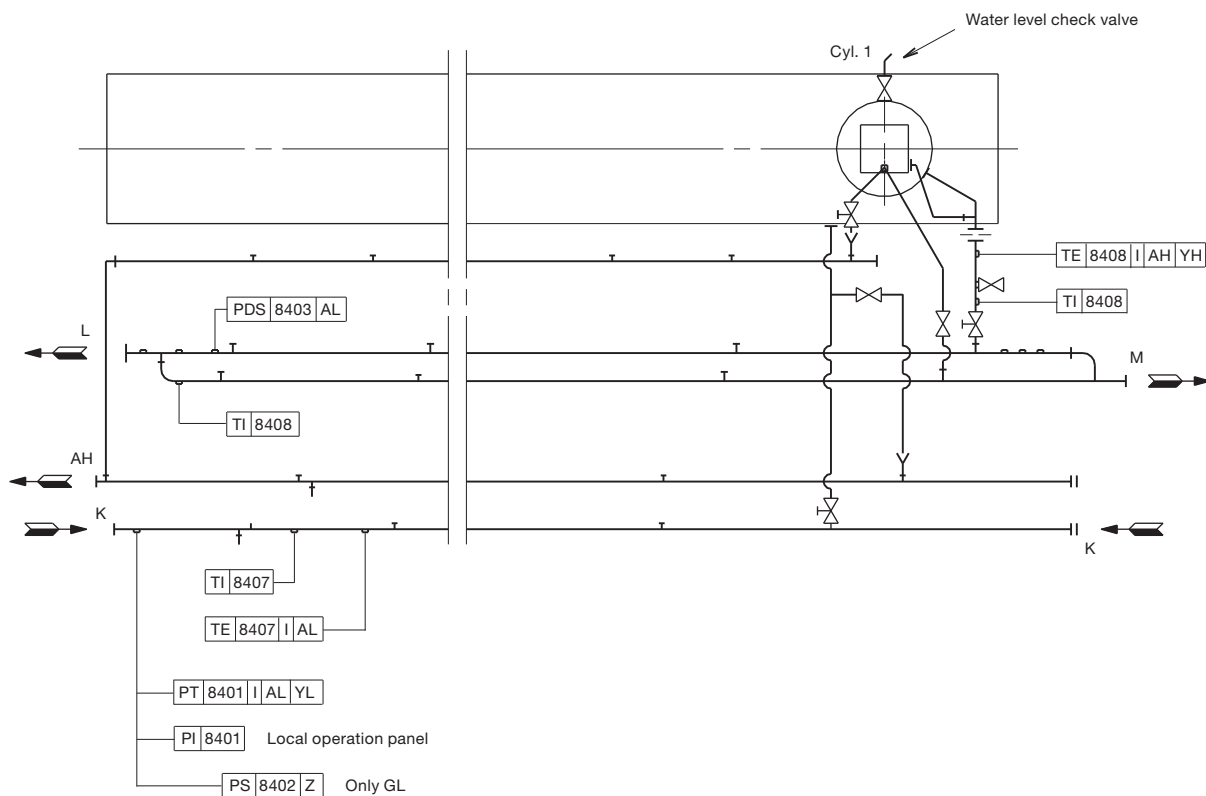


The letters refer to list of 'Counterflanges', Fig. 5.10.01

178 50 17-2.5

Fig. 12.05.01: Jacket cooling water system

Jacket Cooling Water Pipes



178 50 44-6.1

The letters refer to list of 'Counterflanges'
The item No. refer to 'Guidance values automation'

Fig. 12.06.01: Jacket cooling water pipes for engines with MAN Diesel turbochargers, type TCA, ABB turbochargers, type TPL, Mitsubishi turbochargers, type MET

Components for Jacket Cooling Water System

Jacket water cooling pump

The pumps are to be of the centrifugal type.

Jacket water flow see 'List of Capacities'

Pump head.....3.0 bar

Delivery pressuredepends on position
of expansion tank

Test pressure..... according to class rule

Working temperature, 80 °C, max. 100 °C

The flow capacity must be within a range from 100 to 110% of the capacity stated.

The stated capacities cover the main engine only. The pump head of the pumps is to be determined based on the total actual pressure drop across the cooling water system.

Freshwater generator

If a generator is installed in the ship for production of freshwater by utilising the heat in the jacket water cooling system it should be noted that the actual available heat in the jacket water system is lower than indicated by the heat dissipation figures given in the 'List of Capacities'. This is because the latter figures are used for dimensioning the jacket water cooler and hence incorporate a safety margin which can be needed when the engine is operating under conditions such as, e.g. overload. Normally, this margin is 10% at nominal MCR.

The calculation of the heat actually available at specified MCR for a derated diesel engine is stated in Chapter 6 'List of Capacities'.

For illustration of installation of fresh water generator see Fig. 12.05.01.

Jacket water thermostatic valve

The temperature control system is equipped with a three-way valve mounted as a diverting valve, which by-pass all or part of the jacket water around the jacket water cooler.

The sensor is to be located at the outlet from the main engine, and the temperature level must be adjustable in the range of 70-90 °C.

Jacket water preheater

When a preheater, see Fig. 12.05.01, is installed in the jacket cooling water system, its water flow, and thus the preheater pump capacity, should be about 10% of the jacket water main pump capacity.

Based on experience, it is recommended that the pressure drop across the preheater should be approx. 0.2 bar. The preheater pump and main pump should be electrically interlocked to avoid the risk of simultaneous operation.

The preheater capacity depends on the required preheating time and the required temperature increase of the engine jacket water. The temperature and time relations are shown in Fig. 12.08.01.

In general, a temperature increase of about 35 °C (from 15 °C to 50 °C) is required, and a preheating time of 12 hours requires a preheater capacity of about 1% of the engine's nominal MCR power.

Deaerating tank

Design and dimensions of the deaerating tank are shown in Fig. 12.07.01 'Deaerating tank' and the corresponding alarm device is shown in Fig. 12.07.02 'Deaerating tank, alarm device'.

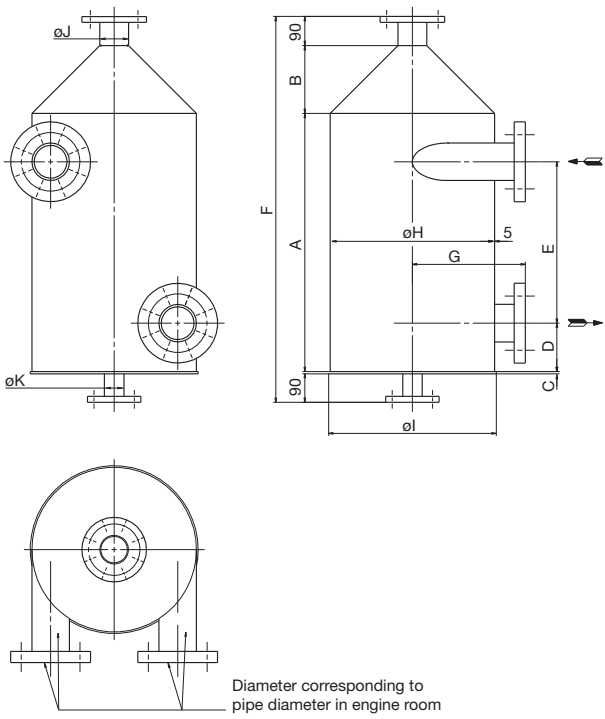
Expansion tank

The total expansion tank volume has to be approximate 10% of the total jacket cooling water amount in the system.

Fresh water treatment

The MAN Diesel recommendations for treatment of the jacket water/freshwater are available on request.

Deaerating tank



178 06 27-9.2

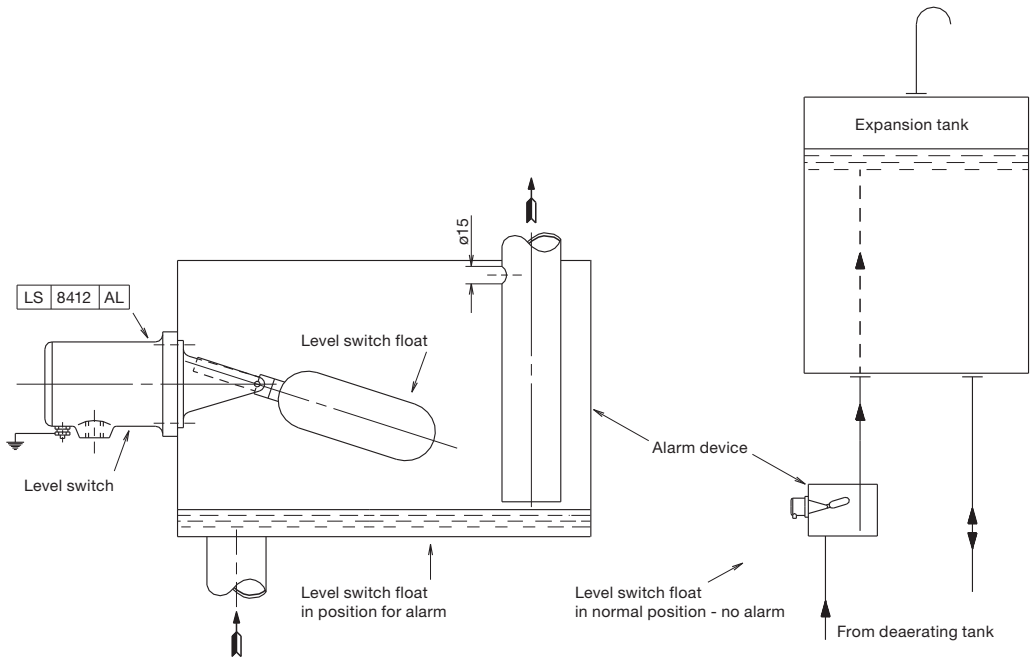
Fig. 12.07.01: Deaerating tank, option: 4 46 640

Deaerating tank dimensions		
Tank size	0.16 m ³	0.70 m ³
Max. jacket water capacity	300 m ³ /h	700 m ³ /h
Dimensions in mm		
Max. nominal diameter	200	300
A	800	1,200
B	210	340
C	5	8
D	150	200
E	500	800
F	1,195	1,728
G	350	550
ϕH	500	800
ϕI	520	820
ϕJ	ND 80	ND 100
ϕK	ND 50	ND 80

ND: Nominal diameter

Working pressure is according to actual piping arrangement.

In order not to impede the rotation of water, the pipe connection must end flush with the tank, so that no internal edges are protruding.



198 97 09-1.1

Fig. 12.07.02: Deaerating tank, alarm device, option: 4 46 645

Temperature at Start of Engine

In order to protect the engine, some minimum temperature restrictions have to be considered before starting the engine and, in order to avoid corrosive attacks on the cylinder liners during starting.

Normal start of engine

Normally, a minimum engine jacket water temperature of 50 °C is recommended before the engine is started and run up gradually to 90% of specified MCR speed.

For running between 90% and 100% of specified MCR speed, it is recommended that the load be increased slowly – i.e. over a period of 30 minutes.

Start of cold engine

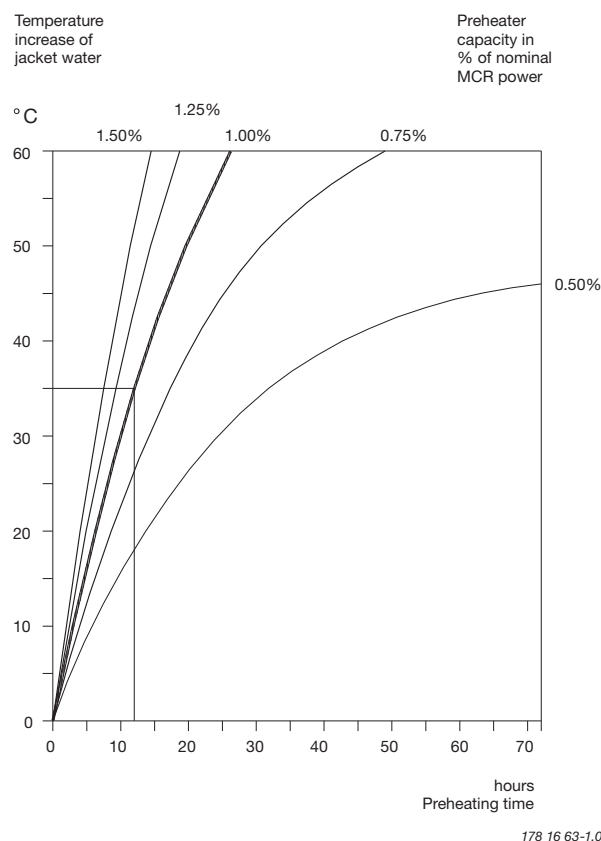
In exceptional circumstances where it is not possible to comply with the above-mentioned recommendation, a minimum of 20 °C can be accepted before the engine is started and run up slowly to 90% of specified MCR speed.

However, before exceeding 90% specified MCR speed, a minimum engine temperature of 50 °C should be obtained and, increased slowly – i.e. over a period of at least 30 minutes.

The time period required for increasing the jacket water temperature from 20 °C to 50 °C will depend on the amount of water in the jacket cooling water system, and the engine load.

Note:

The above considerations are based on the assumption that the engine has already been well run-in.



178 16 63-1.0

Fig. 12.08.01: Jacket water preheater

Preheating of diesel engine

Preheating during standstill periods

During short stays in port (i.e. less than 4-5 days), it is recommended that the engine is kept preheated, the purpose being to prevent temperature variation in the engine structure and corresponding variation in thermal expansions and possible leakages.

The jacket cooling water outlet temperature should be kept as high as possible and should – before starting-up – be increased to at least 50 °C, either by means of cooling water from the auxiliary engines, or by means of a built-in preheater in the jacket cooling water system, or a combination.

Starting and Control Air

13

Starting and Control Air Systems

The starting air of 30 bar is supplied by the starting air compressors to the starting air receivers and from these to the main engine inlet 'A'.

Through a reduction station, filtered compressed air at 7 bar is supplied to the control air for exhaust valve air springs, through engine inlet 'B'

Through a reduction valve, compressed air is supplied at 10 bar to 'AP' for turbocharger cleaning (soft blast), and a minor volume used for the fuel valve testing unit.

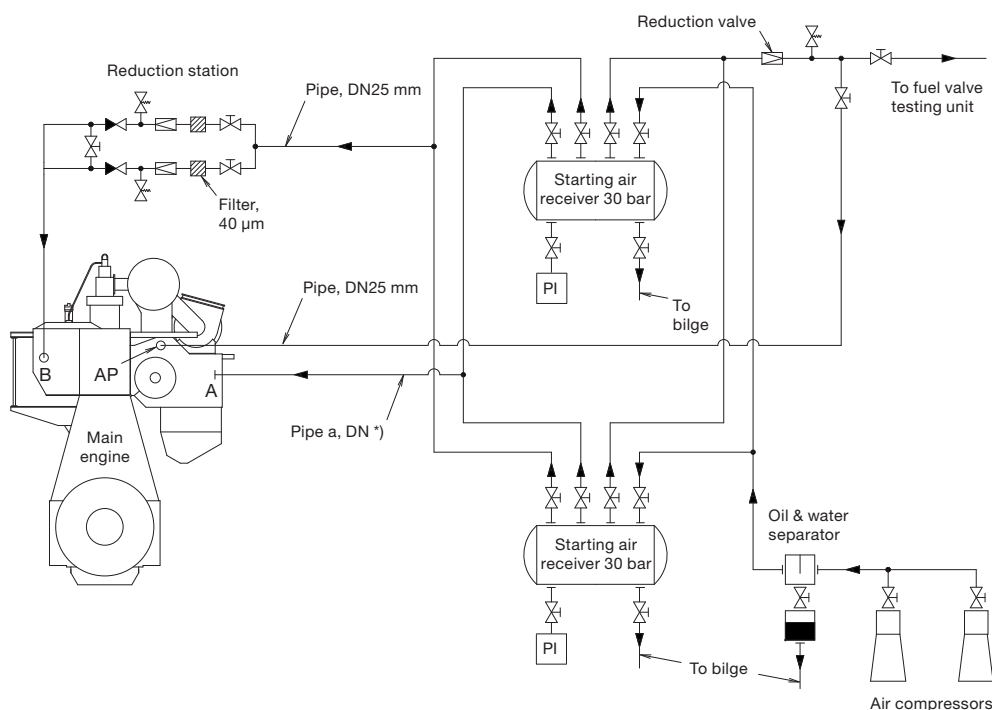
Please note that the air consumption for control air, safety air, turbocharger cleaning, sealing air for exhaust valve and for fuel valve testing unit are momentary requirements of the consumers.

The components of the starting and control air systems are further described in Section 13.02.

For information about a common starting air system for main engines and MAN Diesel auxiliary engines, please refer to our publication:

Uni-concept Auxiliary Systems for Two-Stroke Main Engines and Four-Stroke Auxiliary Engines

The publication is available at www.mandiesel.com under 'Quicklinks' → 'Technical Papers'



The letters refer to list of 'Counterflanges'

*) Pipe a nominal dimension: DN175 mm

078 83 76-7.2.0

Fig. 13.01.01: Starting and control air systems

Components for Starting Air System

Starting air compressors

The starting air compressors are to be of the water-cooled, two-stage type with intercooling.

More than two compressors may be installed to supply the total capacity stated.

Air intake quantity:

Reversible engine,

for 12 starts see 'List of capacities'

Non-reversible engine,

for 6 starts see 'List of capacities'

Delivery pressure 30 bar

Starting air receivers

The volume of the two receivers is:

Reversible engine,

for 12 starts see 'List of capacities' *

Non-reversible engine,

for 6 starts see 'List of capacities' *

Working pressure 30 bar

Test pressure according to class rule

* The volume stated is at 25 °C and 1,000 mbar

Reduction station for control and safety air

In normal operating, each of the two lines supplies one engine inlet. During maintenance, three isolating valves in the reduction station allow one of the two lines to be shut down while the other line supplies both engine inlets, see Fig. 13.01.01.

Reduction from 30-10 bar to 7 bar
(Tolerance $\pm 10\%$)

Flow rate, free air 2,100 Normal liters/min
equal to 0.035 m³/s

Filter, fineness 40 μm

Reduction valve for turbocharger cleaning etc

Reduction from 30-10 bar to 7 bar
(Tolerance $\pm 10\%$)

Flow rate, free air 2,600 Normal liters/min
equal to 0.043 m³/s

The consumption of compressed air for control air, exhaust valve air springs and safety air as well as air for turbocharger cleaning and fuel valve testing is covered by the capacities stated for air receivers and compressors in the list of capacities.

Starting and control air pipes

The piping delivered with and fitted onto the main engine is shown in the following figures in Section 13.03:

Fig. 13.03.01 Starting air pipes

Fig. 13.03.02 Air spring pipes, exhaust valves

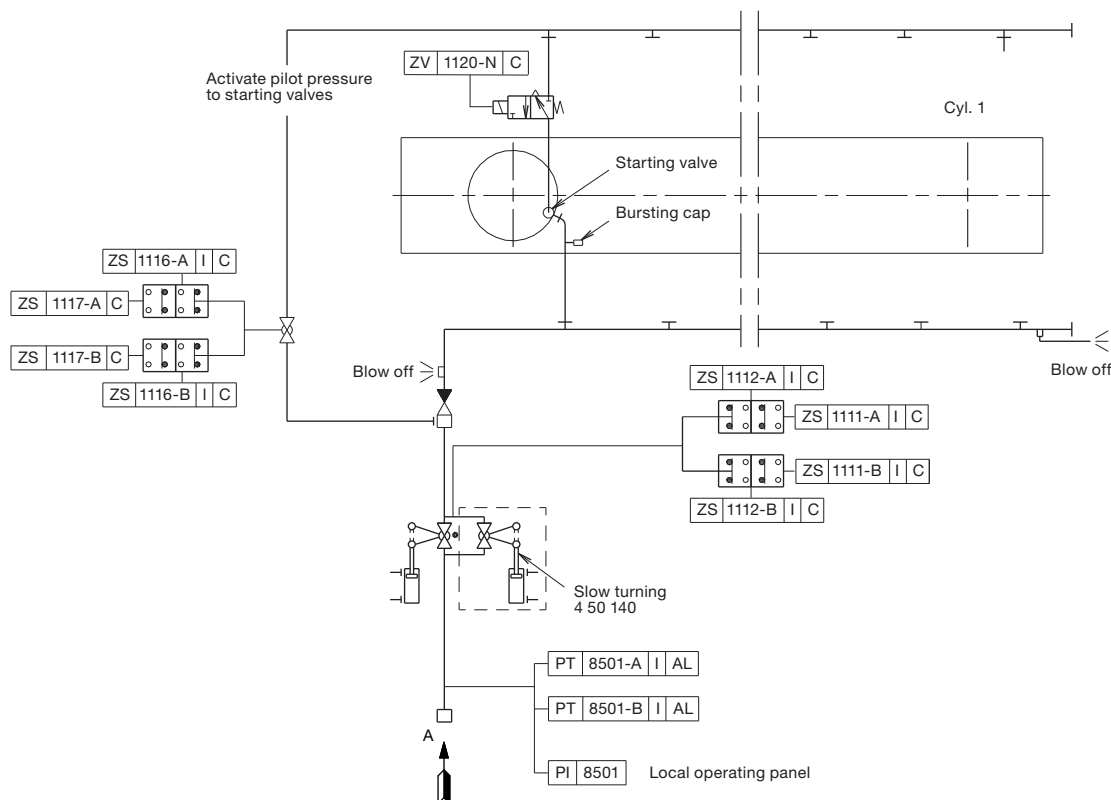
Turning gear

The turning wheel has cylindrical teeth and is fitted to the thrust shaft. The turning wheel is driven by a pinion on the terminal shaft of the turning gear, which is mounted on the bedplate.

Engagement and disengagement of the turning gear is effected by displacing the pinion and terminal shaft axially. To prevent the main engine from starting when the turning gear is engaged, the turning gear is equipped with a safety arrangement which interlocks with the starting air system.

The turning gear is driven by an electric motor with a built-in gear and brake. Key specifications of the electric motor and brake are stated in Section 13.04.

Starting and Control Air Pipes



The letters refer to list of 'Counterflanges'
 The item Nos. refer to 'Guidance values automation'
 The piping is delivered with and fitted onto the engine

198 98 21-5.3

Fig. 13.03.01: Starting air pipes

The starting air pipes, Fig. 13.03.01, contain a main starting valve (a ball valve with actuator), a non-return valve, a solenoid valve and a starting valve. The main starting valve is controlled by the Engine Control System. Slow turning before start of engine (4 50 140) is included in the basic design.

The Engine Control System regulates the supply of control air to the starting valves in accordance with the correct firing sequence and the timing.

Please note that the air consumption for control air, turbocharger cleaning and for fuel valve testing unit are momentary requirements of the consumers. The capacities stated for the air receivers

and compressors in the 'List of Capacities' cover all the main engine requirements and starting of the auxiliary engines.

For information about a common starting air system for main engines and auxiliary engines, please refer to the Engine Selection Guide or to our publication:

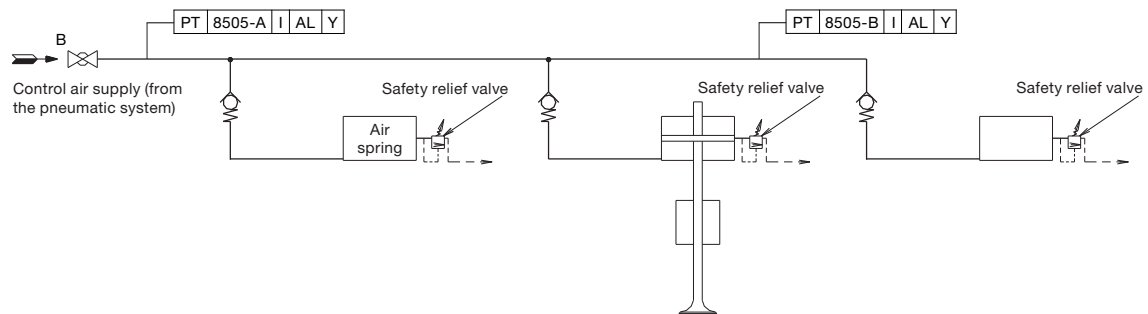
Uni-concept Auxiliary Systems for Two-stroke Main

The publication is available at www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

Exhaust Valve Air Spring Pipes

The exhaust valve is opened hydraulically by the Fuel Injection Valve Actuator (FIVA) system which is activated by the Engine Control System, and the closing force is provided by an 'air spring' which leaves the valve spindle free to rotate.

The compressed air is taken from the control air supply, see Fig. 13.03.02.



The item Nos. refer to 'Guidance values automation'
The piping is delivered with and fitted onto the engine

121 36 87-1.1.0c

Fig. 13.03.02: Air spring pipes for exhaust valves

Electric Motor for Turning Gear

MAN Diesel delivers a turning gear with built-in disc brake, option 40 80 101. Two basic executions are available for power supply frequencies of 60 and 50 Hz respectively. Nominal power and current consumption of the motors are listed below.

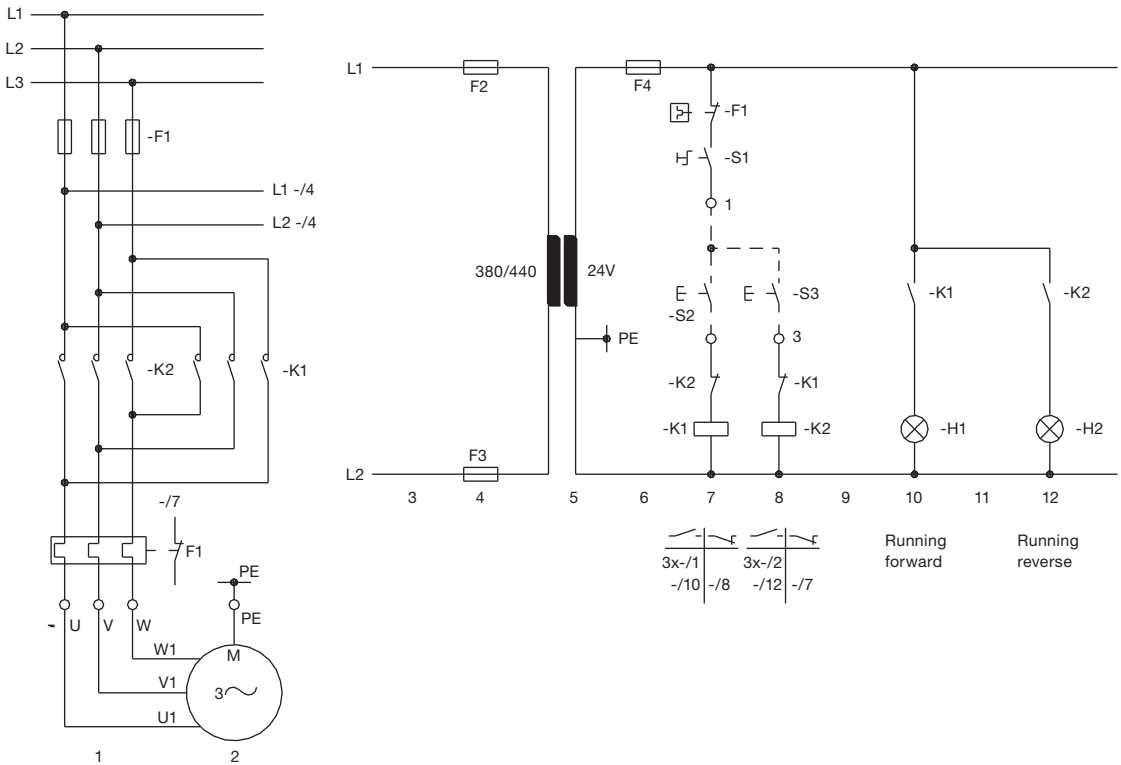
Electric motor and brake, voltage..... 3 x 440 V
 Electric motor and brake, frequency60 Hz
 Protection, electric motor / brake IP 55 / IP 54
 Insulation class F

Number of cylinders	Electric motor	
	Nominal power, kW	Normal current, A
6-8	9.0	14.8
9-12	13.2	22.0

Turning gear with electric motor of other protection or insulation classes can be ordered, option 40 80 103. Information about the alternative executions is available on request.

Electric motor and brake, voltage..... 3 x 380 V
 Electric motor and brake, frequency50 Hz
 Protection, electric motor / brake IP 55 / IP 54
 Insulation class F

Number of cylinders	Electric motor	
	Nominal power, kW	Normal current, A
6-8	7.5	14.8
9-12	11.0	22.0



178 31 30-9.1

Fig. 13.04.01: Electric motor for turning gear, option: 40 80 101

Scavenge Air

14

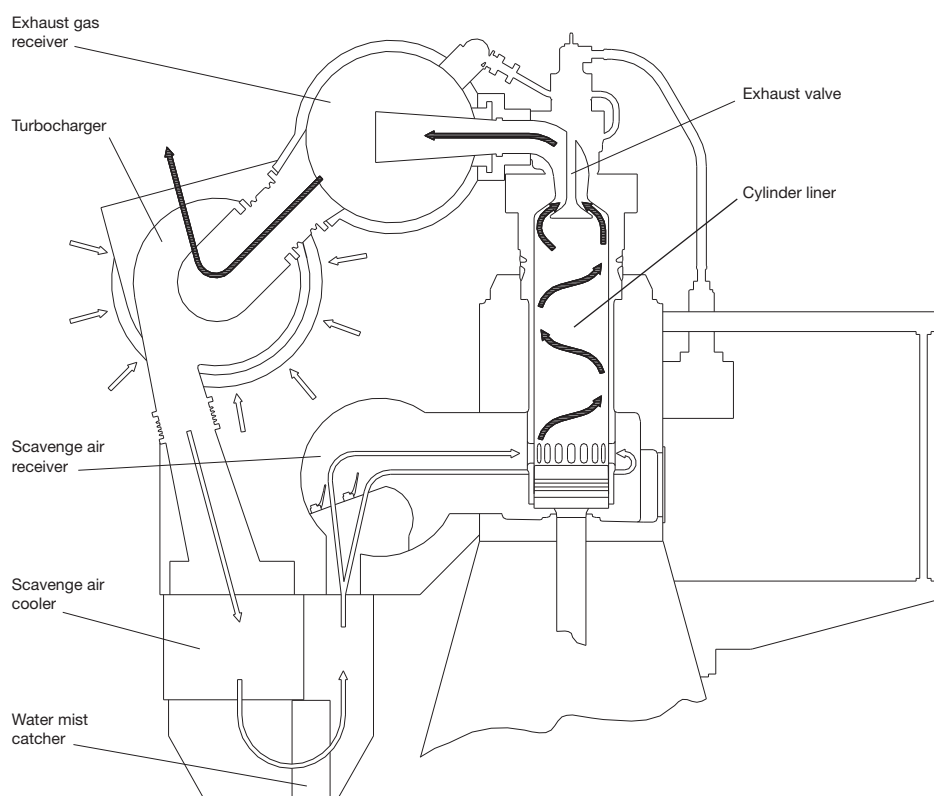
Scavenge Air System

Scavenge air is supplied to the engine by two or more turbochargers, located on the exhaust side of the engine.

The compressor of the turbocharger draws air from the engine room, through an air filter, and the compressed air is cooled by the scavenge air cooler, one per turbocharger. The scavenge air cooler is provided with a water mist catcher, which prevents condensate water from being carried with the air into the scavenge air receiver and to the combustion chamber.

The scavenge air system (see Figs. 14.01.01 and 14.02.01) is an integrated part of the main engine.

The engine power figures and the data in the list of capacities are based on MCR at tropical conditions, i.e. a seawater temperature of 32 °C, or freshwater temperature of 36 °C, and an ambient air inlet temperature of 45 °C.



178 25 18-8.1

Fig. 14.01.01: Scavenge Air System

Auxiliary Blowers

The engine is provided with a minimum of two electrically driven auxiliary blowers, the actual number depending on the number of cylinders as well as the turbocharger make and amount.

The auxiliary blowers are fitted onto the main engine. Between the scavenge air cooler and the scavenge air receiver, non-return valves are fitted which close automatically when the auxiliary blowers start supplying the scavenge air.

Auxiliary blower operation

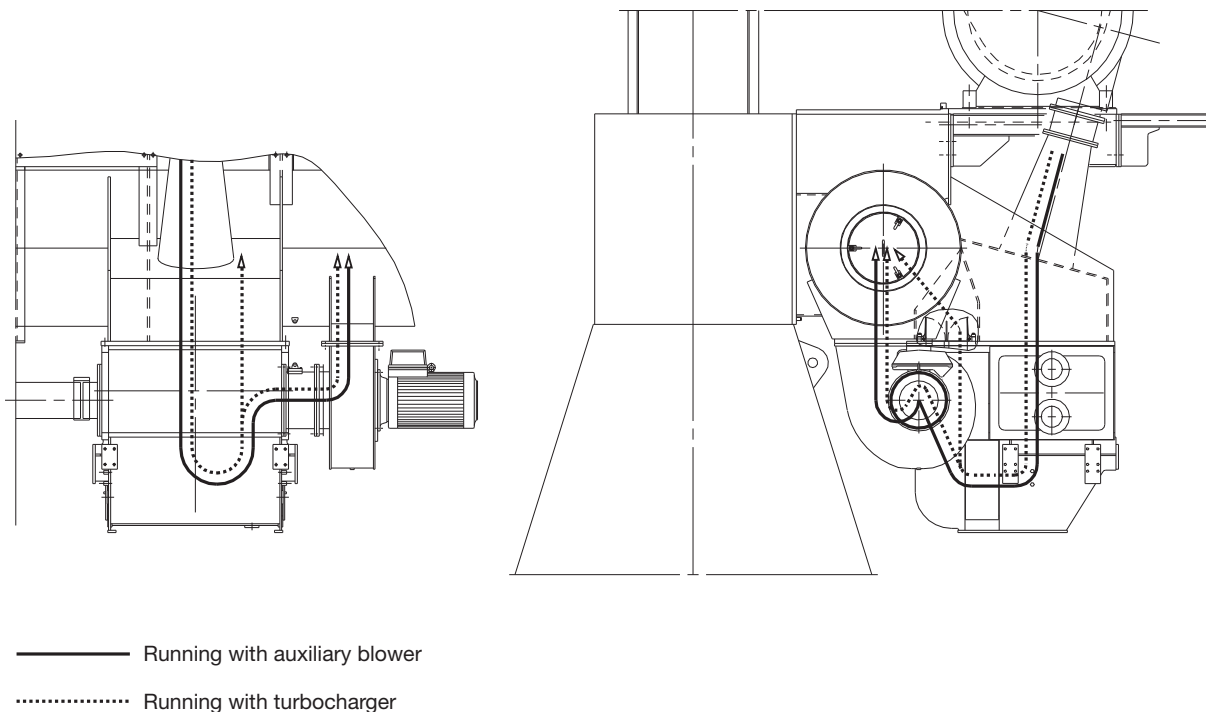
The auxiliary blowers start operating consecutively before the engine is started and will ensure complete scavenging of the cylinders in the starting phase, thus providing the best conditions for a safe start.

During operation of the engine, the auxiliary blowers will start automatically whenever the blower inlet pressure drops below a preset pressure, corresponding to an engine load of approximately 25-35%.

The blowers will continue to operate until the blower inlet pressure again exceeds the preset pressure plus an appropriate hysteresis (i.e. taking recent pressure history into account), corresponding to an engine load of approximately 30-40%.

Emergency running

If one of the auxiliary blowers is out of function, the other auxiliary blower will function in the system, without any manual adjustment of the valves being necessary.



178 44 70-5.1

Fig. 14.02.01: Scavenge air system

Control of the Auxiliary Blowers

The control system for the auxiliary blowers is integrated in the Engine Control System. The auxiliary blowers can be controlled in either automatic (default) or manual mode.

In automatic mode, the auxiliary blowers are started sequentially at the moment the engine is commanded to start. During engine running, the blowers are started and stopped according to preset scavenge air pressure limits.

When the engine stops, the blowers are stopped after 10 minutes to prevent overheating of the blowers. When a start is ordered, the blower will be started in the normal sequence and the actual start of the engine will be delayed until the blowers have started.

In manual mode, the blowers can be controlled individually from the ECR (Engine Control Room) panel irrespective of the engine condition.

Referring to Fig. 14.02.02, the Auxiliary Blower Starter Panels control and protect the Auxiliary Blower motors, one panel with starter per blower.

The starter panels with starters for the auxiliary blower motors are not included, they can be ordered as an option: 4 55 653. (The starter panel design and function is according to MAN Diesel's diagram, however, the physical layout and choice of components has to be decided by the manufacturer).

Heaters for the blower motors are available as an option: 4 55 155.

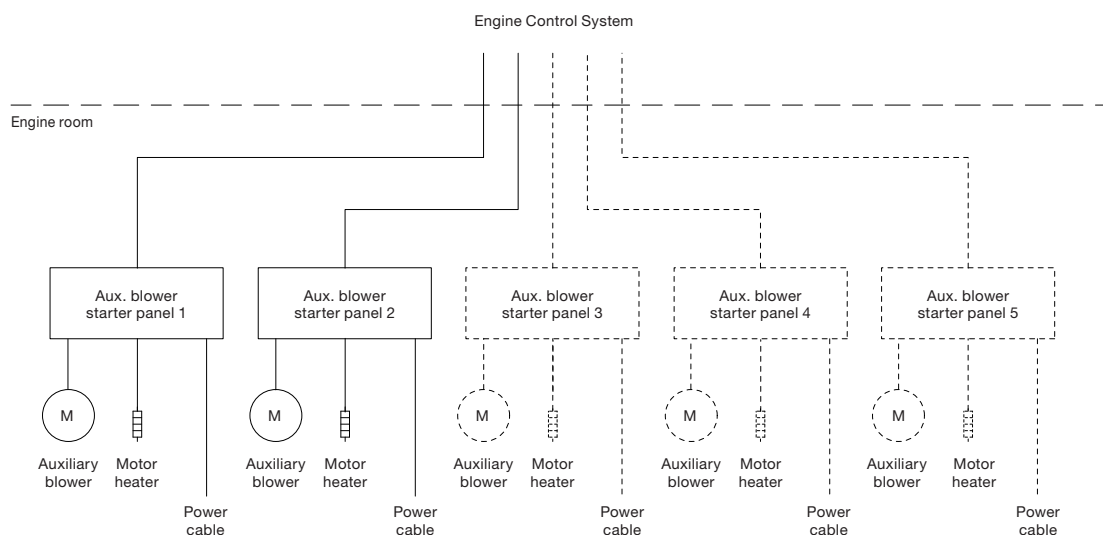
Scavenge air cooler requirements

The data for the scavenge air cooler is specified in the description of the cooling water system chosen.

For further information, please refer to our publication titled:

Influence of Ambient Temperature Conditions

The publication is available at: www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

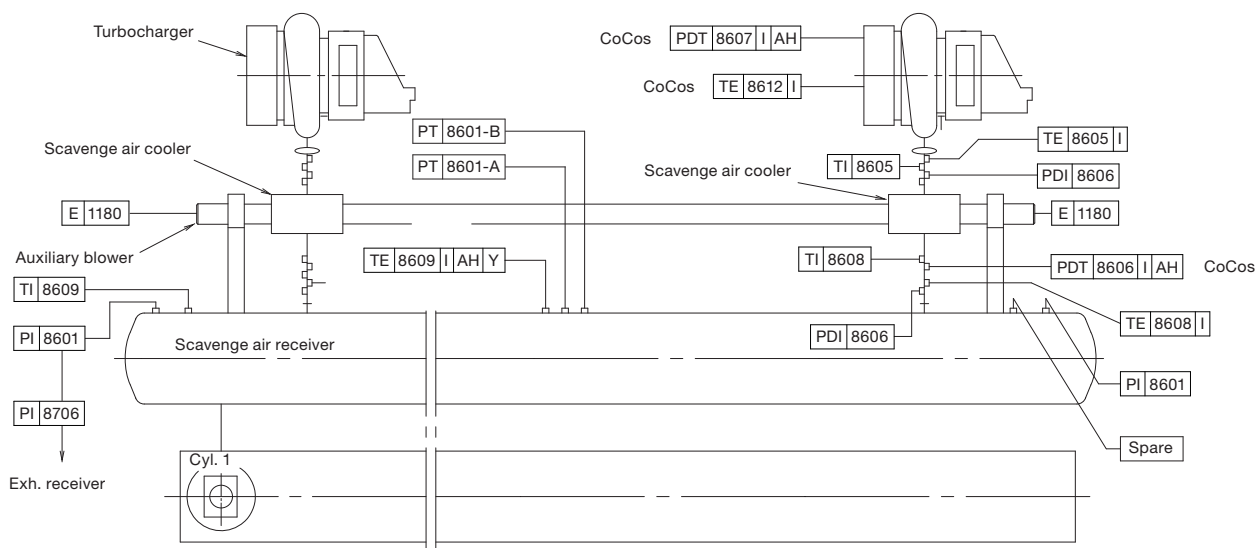


178 61 30-2.0

Fig. 14.02.02: Diagram of auxiliary blower control system

Scavenge Air Pipes

Two Turbochargers or more

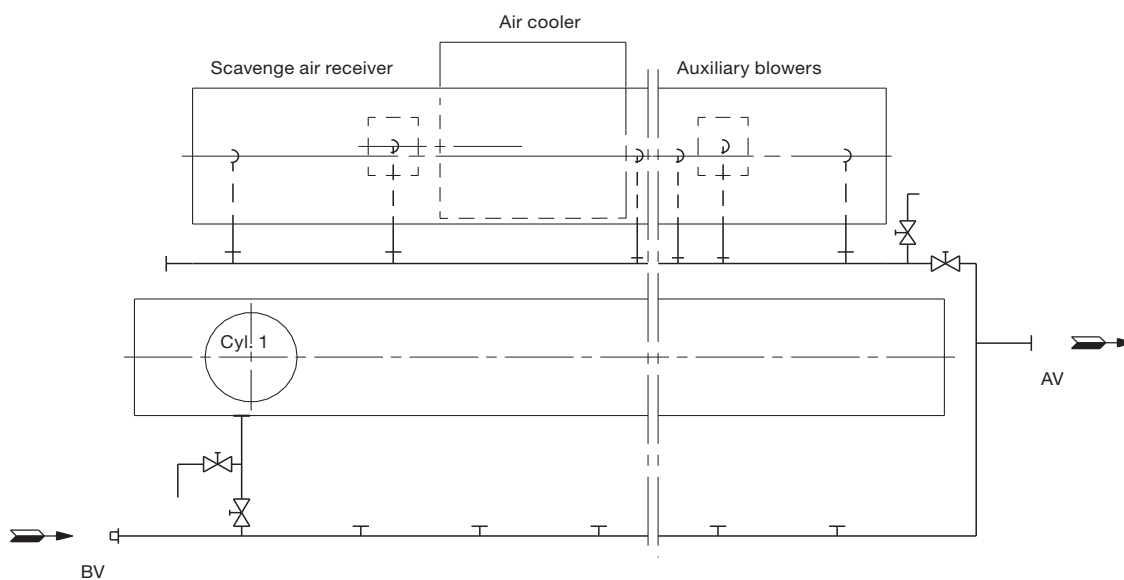


The item No. refer to 'Guidance Values Automation'

121 15 25-5.6.0

Fig. 14.03.01: Scavenge air pipes

Scavenge air space, drain pipes



The letters refer to list of 'Counterflanges'

121 36 91-7.2.0

Fig. 14.03.02: Scavenge air space, drain pipes

Electric Motor for Auxiliary Blower

The number of auxiliary blowers in a propulsion plant may vary depending on the actual amount of turbochargers as well as space requirements.

For typical engine configurations, the required power of the auxiliary blowers as well as the installed size of the electric motors are listed in Table 14.04.01.

Number of cylinders	Number of auxiliary blowers	Required power/blower kW	Installed power/blower kW
6	Data is available on request		
7			
8			
9			
10			
11			
12			

The installed power of the electric motors are based on a voltage supply of 3x440V at 60Hz.

The electric motors are delivered with and fitted onto the engine.

Table 14.04.01: Electric motor for auxiliary blower

Scavenge Air Cooler Cleaning System

The air side of the scavenge air cooler can be cleaned by injecting a grease dissolving media through 'AK' to a spray pipe arrangement fitted to the air chamber above the air cooler element.

The system is equipped with a drain box with a level switch, indicating any excessive water level.

The piping delivered with and fitted on the engine is shown in Fig 14.05.01.

Drain from water mist catcher

Sludge is drained through 'AL' to the drain water collecting tank and the polluted grease dissolvent returns from 'AM', through a filter, to the chemical cleaning tank. The cleaning must be carried out while the engine is at standstill.

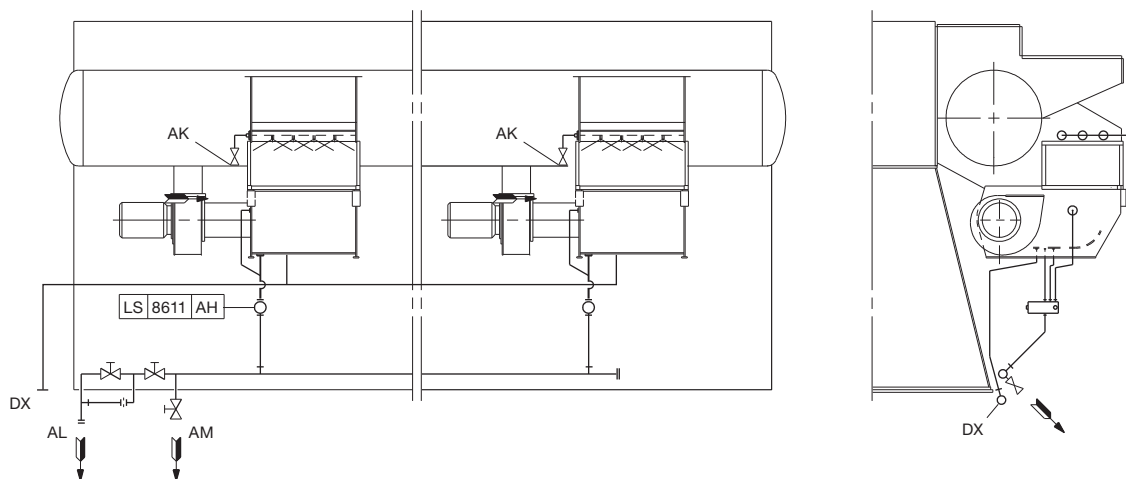
Dirty water collected after the water mist catcher is drained through 'DX' and led to the bilge tank via an open funnel, see Fig. 14.05.02.

The 'AL' drain line is, during running, used as a permanent drain from the air cooler water mist catcher. The water is led through an orifice to prevent major losses of scavenge air.

Auto Pump Overboard System

It is common practice on board to lead drain water directly overboard via a collecting tank. Before pumping the drain water overboard, it is recommended to measure the oil content. If above 15ppm, the drain water should be lead to the clean bilge tank / bilge holding tank.

If required by the owner, a system for automatic disposal of drain water with oil content monitoring could be built as outlined in Fig. 14.05.02.

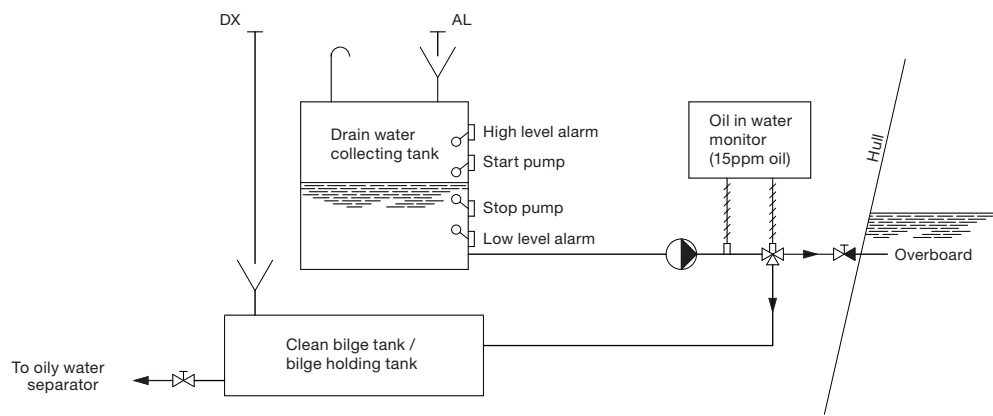


The letters refer to list of 'Counterflanges'
The item no refer to 'Guidance values automation'

178 56 35-4.2

Fig. 14.05.01: Air cooler cleaning pipes

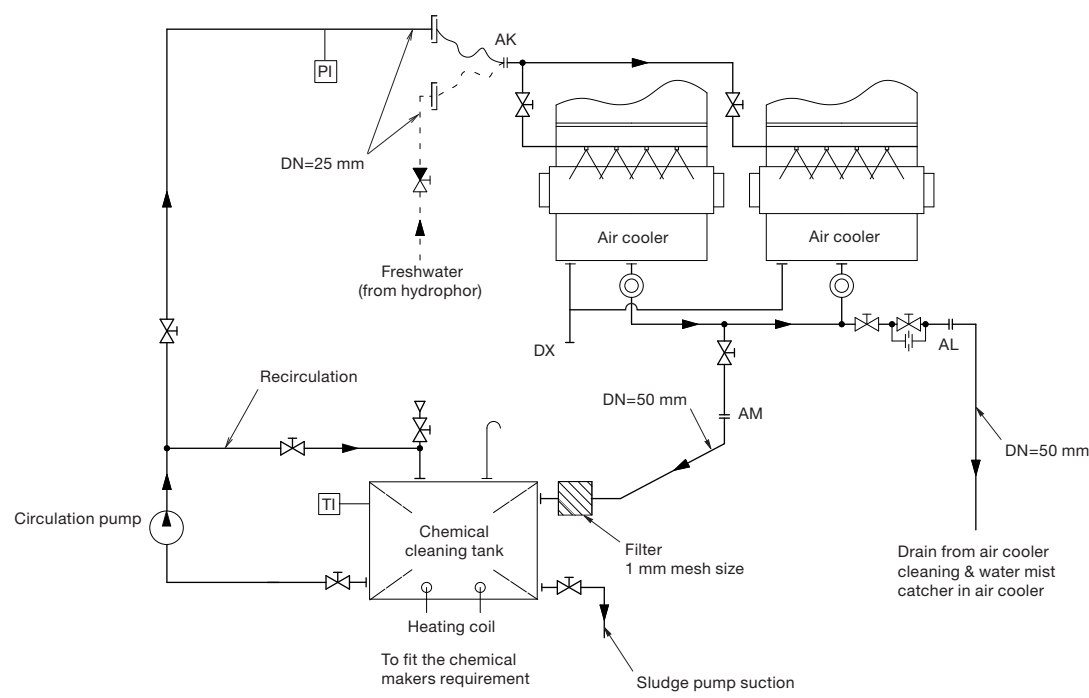
Auto Pump Overboard System



079 21 94-1.0.0c

Fig. 14.05.02: Suggested automatic disposal of drain water, if required by owner (not a demand from MAN Diesel)

Air Cooler Cleaning Unit



The letters refer to list of 'Counterflanges'

	No. of cylinders	
	6-8	9-12
Chemical tank capacity	0.9 m ³	1.5 m ³
Circulation pump capacity at 3 bar	3 m ³ /h	5 m ³ /h

079 21 94-1.0.0a

Fig. 14.05.03: Air cooler cleaning system with Air Cooler Cleaning Unit, option: 4 55 665

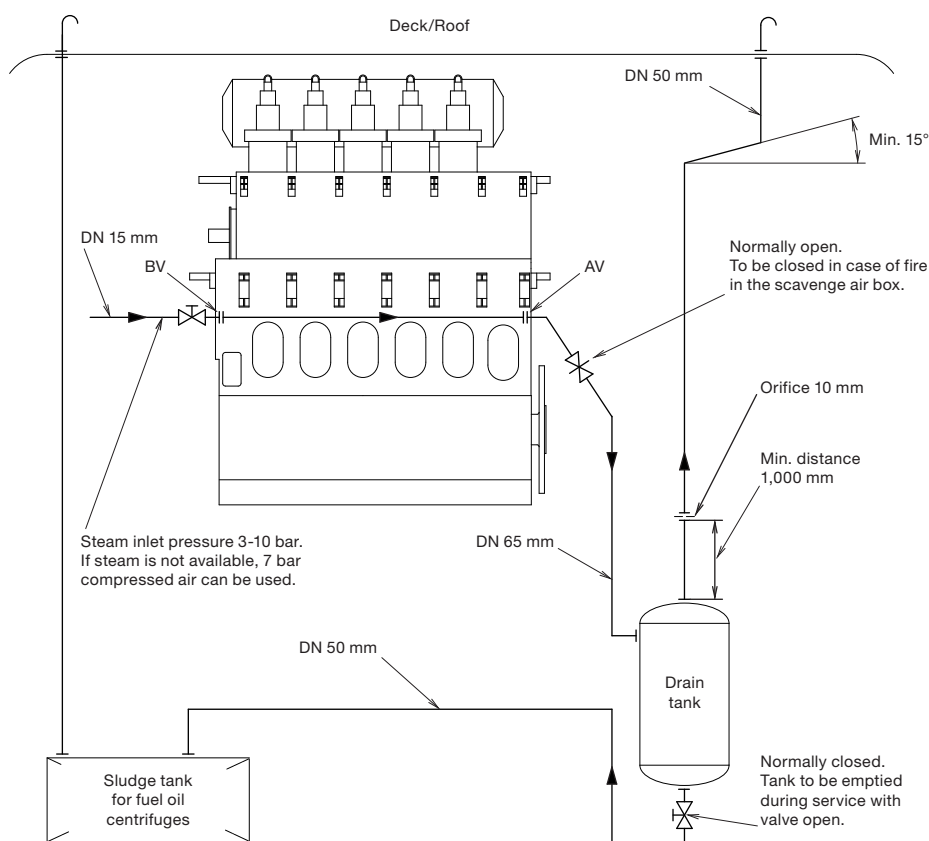
Scavenge Air Box Drain System

The scavenge air box is continuously drained through 'AV' to a small pressurised drain tank, from where the sludge is led to the sludge tank. Steam can be applied through 'BV', if required, to facilitate the draining. See Fig. 14.06.01.

The continuous drain from the scavenge air box must not be directly connected to the sludge tank owing to the scavenge air pressure.

The pressurised drain tank must be designed to withstand full scavenge air pressure and, if steam is applied, to withstand the steam pressure available.

The system delivered with and fitted on the engine is shown in Fig. 14.03.02 Scavenge air space, drain pipes.



	No. of cylinders		
	6	7-9	10-12
Drain tank capacity	0.8 m ³	1.1 m ³	1.5 m ³

The letters refer to list of 'Counterflanges'

079 61 03-0.2.0

Fig. 14.06.01: Scavenge air box drain system

Fire Extinguishing System for Scavenge Air Space

Fire in the scavenge air space can be extinguished by steam, this being the basic solution, or, optionally, by water mist or CO₂.

The external system, pipe and flange connections are shown in Fig. 14.07.01 and the piping fitted onto the engine in Fig. 14.07.02.

In the Extent of Delivery, the fire extinguishing system for scavenge air space is selected by the fire extinguishing agent:

- basic solution: 4 55 140 Steam
- option: 4 55 142 Water mist
- option: 4 55 143 CO₂

The key specifications of the fire extinguishing agents are:

Steam fire extinguishing for scavenge air space

Max. test pressure: 15 bar

Steam quantity, approx.: 7.8 kg/cyl.

Water mist fire extinguishing for scavenge air space

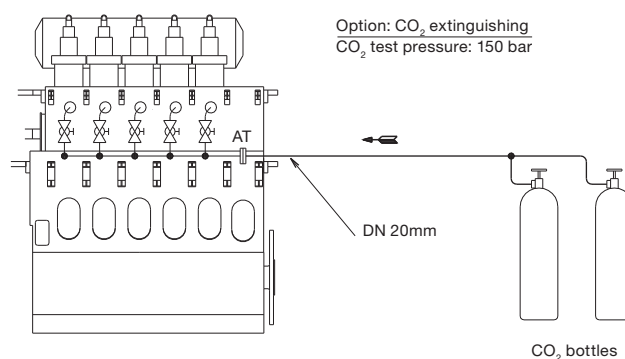
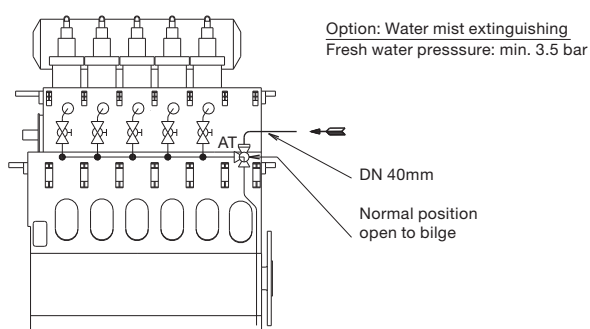
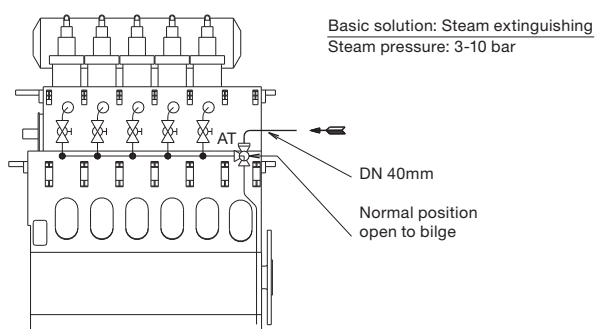
Max. test pressure: 10 bar

Freshwater quantity, approx.: 6.3 kg/cyl.

CO₂ fire extinguishing for scavenge air space

Max. test pressure: 150 bar

CO₂ quantity, approx.: 15.7 kg/cyl.



CO₂

At least two bottles ought to be installed. In most cases, one bottle should be sufficient to extinguish fire in three cylinders, while two or more bottles would be required to extinguish fire in all cylinders.

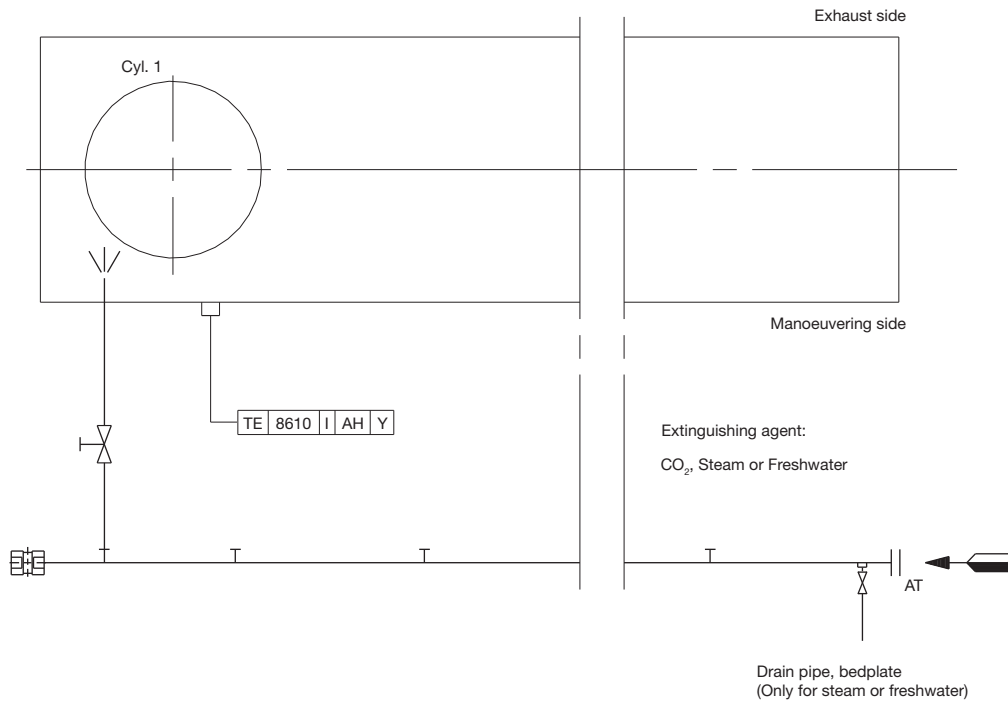
To prevent the fire from spreading to the next cylinder(s), the ball-valve of the neighbouring cylinder(s) should be opened in the event of fire in one cylinder.

079 61 02-9.0.0a

The letters refer to list of 'Counterflanges'

Fig. 14.07.01: Fire extinguishing system for scavenge air space

Fire Extinguishing Pipes in Scavenge Air Space



126 40 81-0.6.0a

The letters refer to list of 'Counterflanges'

Fig. 14.07.02: Fire extinguishing pipes in scavenge air space

Exhaust Gas

Exhaust Gas System

The exhaust gas is led from the cylinders to the exhaust gas receiver where the fluctuating pressures from the cylinders are equalised and from where the gas is led further on to the turbocharger at a constant pressure. See fig. 15.01.01.

Compensators are fitted between the exhaust valve housings and the exhaust gas receiver and between the receiver and the turbocharger. A protective grating is placed between the exhaust gas receiver and the turbocharger. The turbocharger is fitted with a pick-up for monitoring and remote indication of the turbocharger speed.

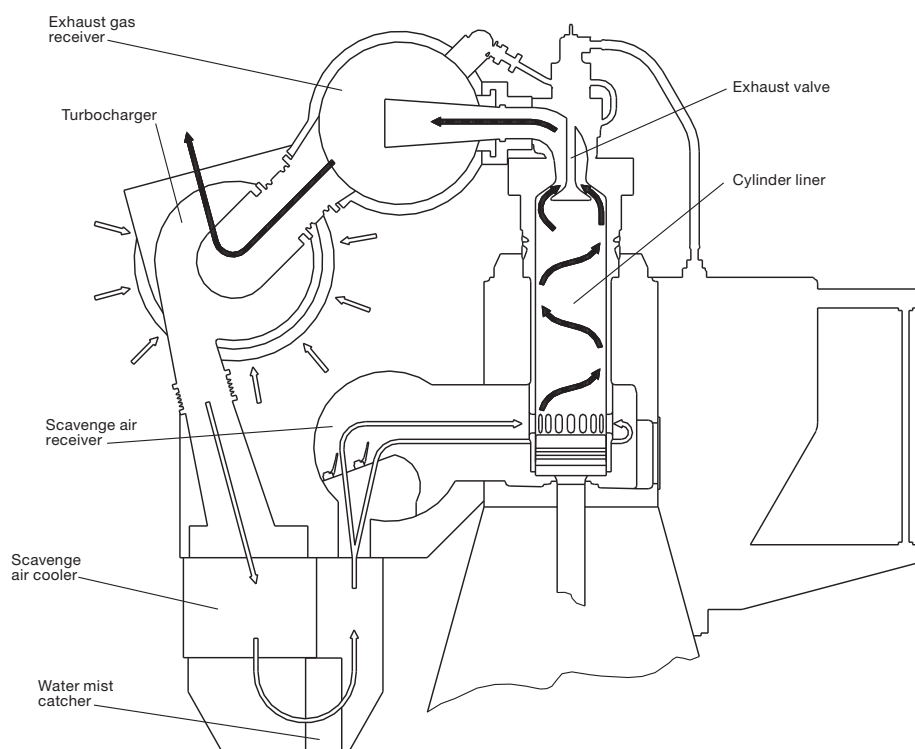
The exhaust gas receiver and the exhaust pipes are provided with insulation, covered by steel plating.

Turbocharger arrangement and cleaning systems

The turbochargers are located on the exhaust side of the engine.

The engine is designed for the installation of the MAN Diesel turbocharger type TCA, option: 4 59 101, ABB turbocharger types TPL or A100, option: 4 59 102, or MHI turbocharger type MET, option: 4 59 103.

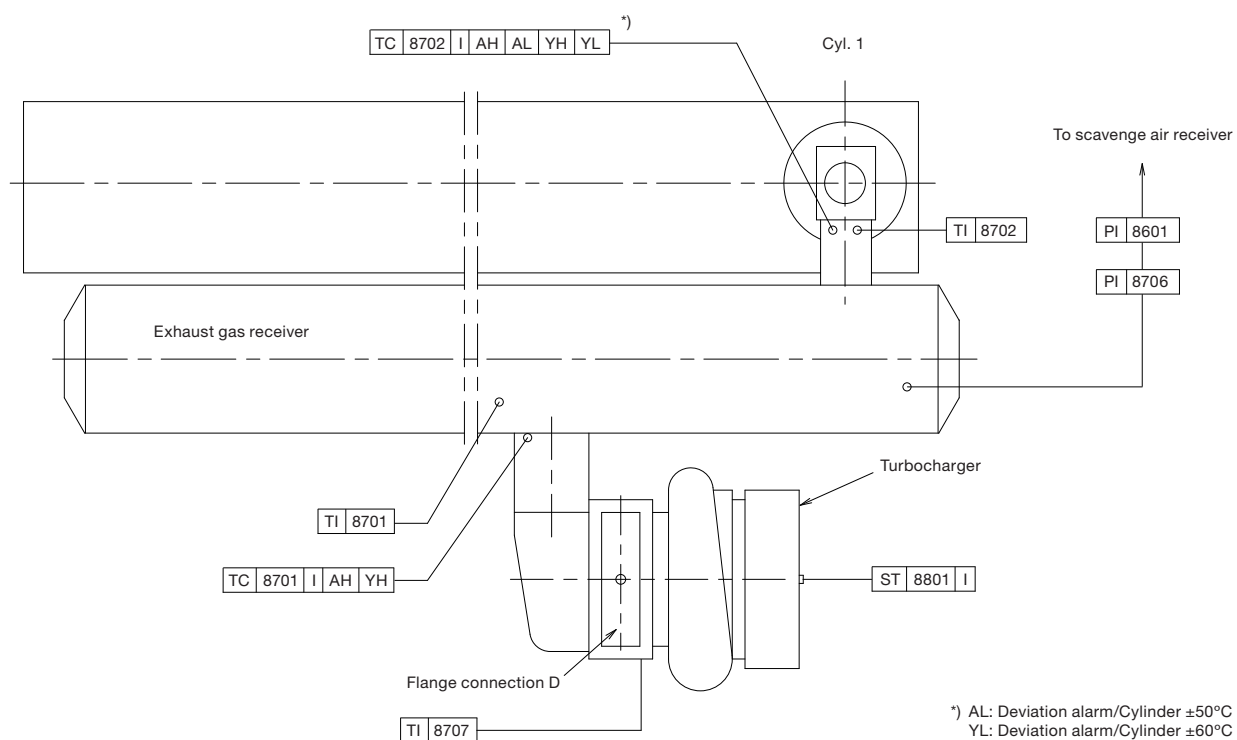
All makes of turbochargers are fitted with an arrangement for water washing of the compressor side, and soft blast cleaning of the turbine side, see Figs. 15.02.02, 15.02.03 and 15.02.04. Washing of the turbine side is only applicable on MAN Diesel and ABB turbochargers.



178 07 27-4.1

Fig. 15.01.01: Exhaust gas system on engine

Exhaust Gas Pipes

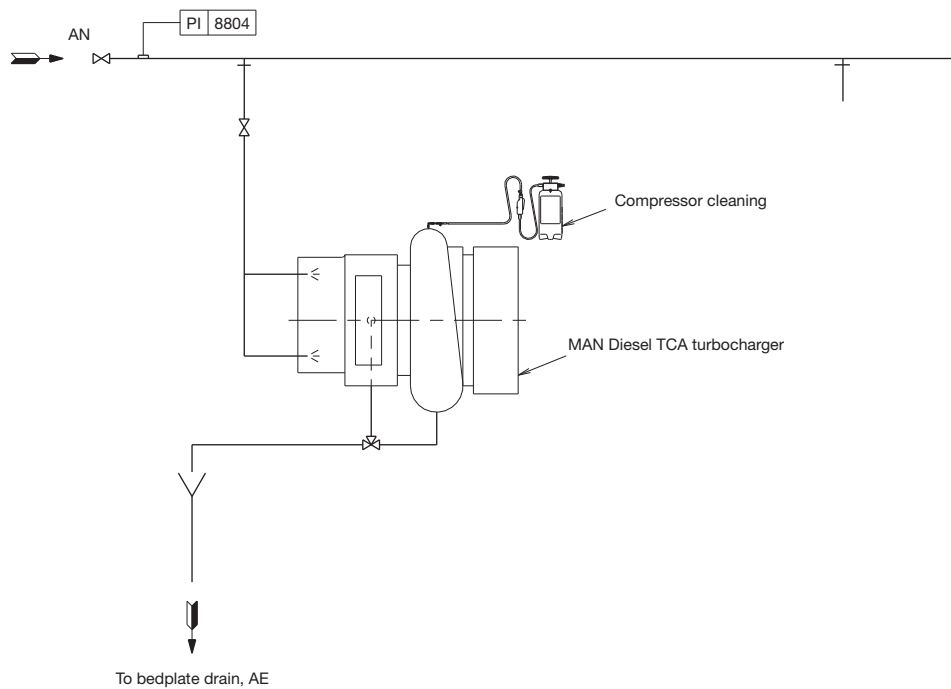


The letters refer to list of 'Counterflanges'
The item no. refer to 'Guidance Values Automation'

121 15 27-9.2.0

Fig. 15.02.01: Exhaust gas pipes

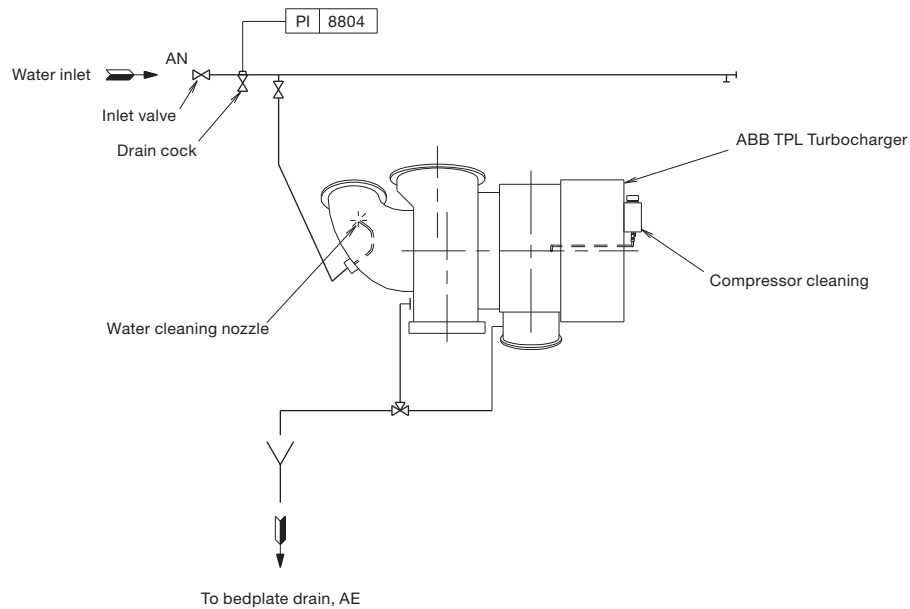
Cleaning Systems



121 15 21-8.0.0

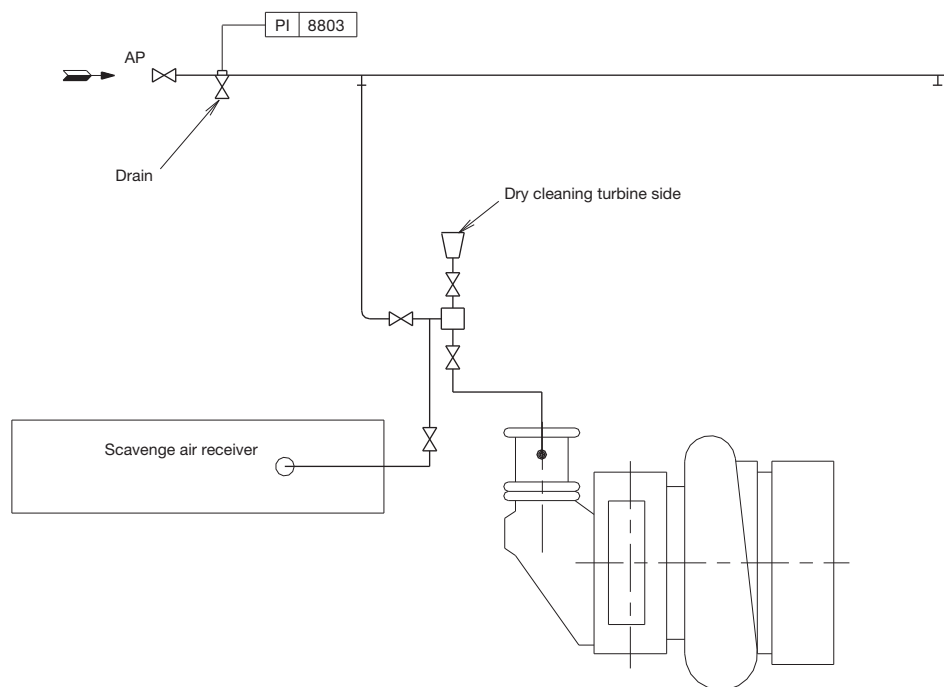
Fig. 15.02.02: MAN Diesel TCA turbocharger, water washing of turbine side

Cleaning Systems



121 36 75-1.0.0

Fig. 15.02.03: Water washing of turbine and compressor sides for ABB, TPL turbochargers



121 36 88-3.2.0

Fig. 15.02.04: Soft blast cleaning of turbine side

Exhaust Gas System for Main Engine

At the specified MCR of the engine, the total back-pressure in the exhaust gas system after the turbocharger (as indicated by the static pressure measured in the piping after the turbocharger) must not exceed 350 mm WC (0.035 bar).

In order to have a back-pressure margin for the final system, it is recommended at the design stage to initially use a value of about 300 mm WC (0.030 bar).

The actual back-pressure in the exhaust gas system at specified MCR depends on the gas velocity, i.e. it is proportional to the square of the exhaust gas velocity, and hence inversely proportional to the pipe diameter to the 4th power. It has by now become normal practice in order to avoid too much pressure loss in the pipings to have an exhaust gas velocity at specified MCR of about 35 m/sec, but not higher than 50 m/sec.

For dimensioning of the external exhaust pipe connections, see the exhaust pipe diameters for 35 m/sec, 40 m/sec, 45 m/sec and 50 m/sec respectively, shown in Table 15.07.02.

As long as the total back-pressure of the exhaust gas system (incorporating all resistance losses from pipes and components) complies with the above-mentioned requirements, the pressure losses across each component may be chosen independently, see proposed measuring points (M) in Fig. 15.05.01. The general design guidelines for each component, described below, can be used for guidance purposes at the initial project stage.

Exhaust gas piping system for main engine

The exhaust gas piping system conveys the gas from the outlet of the turbocharger(s) to the atmosphere.

The exhaust piping is shown schematically in Fig. 15.04.01.

The exhaust system for the main engine comprises:

- Exhaust gas pipes
- Exhaust gas boiler
- Silencer
- Spark arrester (if needed)
- Expansion joints (compensators)
- Pipe bracings.

In connection with dimensioning the exhaust gas piping system, the following parameters must be observed:

- Exhaust gas flow rate
- Exhaust gas temperature at turbocharger outlet
- Maximum pressure drop through exhaust gas system
- Maximum noise level at gas outlet to atmosphere
- Maximum force from exhaust piping on turbocharger(s)
- Sufficient axial and lateral elongation ability of expansion joints
- Utilisation of the heat energy of the exhaust gas.

Items that are to be calculated or read from tables are:

- Exhaust gas mass flow rate, temperature and maximum back pressure at turbocharger gas outlet
- Diameter of exhaust gas pipes
- Utilisation of the exhaust gas energy
- Attenuation of noise from the exhaust pipe outlet
- Pressure drop across the exhaust gas system
- Expansion joints.

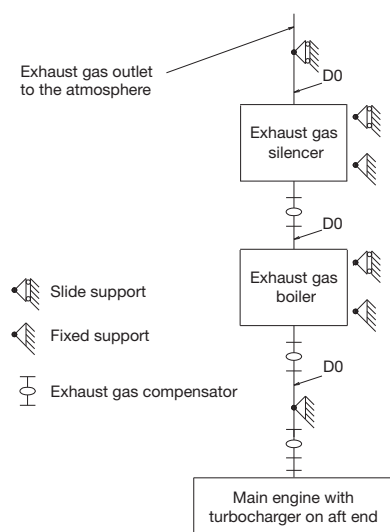
Components of the Exhaust Gas System

Exhaust gas compensator after turbocharger

When dimensioning the compensator, option: 4 60 610, for the expansion joint on the turbocharger gas outlet transition piece, option: 4 60 601, the exhaust gas piece and components, are to be so arranged that the thermal expansions are absorbed by expansion joints. The heat expansion of the pipes and the components is to be calculated based on a temperature increase from 20 °C to 250 °C. The max. expected vertical, transversal and longitudinal heat expansion of the engine measured at the top of the exhaust gas transition piece of the turbocharger outlet are indicated in Fig. 15.06.01 and Table 15.06.02 as DA, DB and DC.

The movements stated are related to the engine seating, for DC, however, to the engine centre. The figures indicate the axial and the lateral movements related to the orientation of the expansion joints.

The expansion joints are to be chosen with an elasticity that limits the forces and the moments of the exhaust gas outlet flange of the turbocharger as stated for each of the turbocharger makers in Table 15.06.04. The orientation of the maximum permissible forces and moments on the gas outlet flange of the turbocharger is shown in Fig. 15.06.03.



178 42 78-3.2

Fig. 15.04.01a: Exhaust gas system, one turbocharger

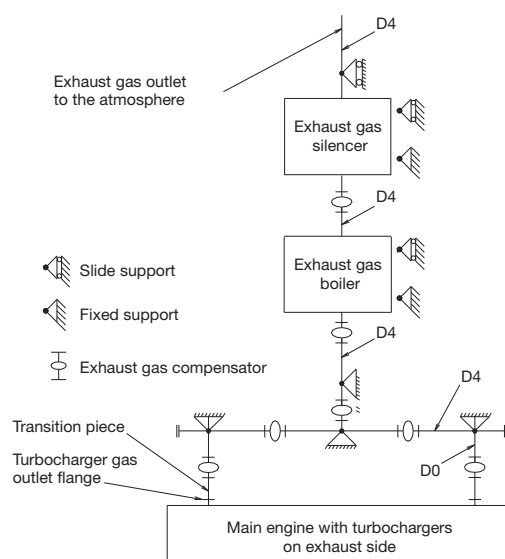
Exhaust gas boiler

Engine plants are usually designed for utilisation of the heat energy of the exhaust gas for steam production or for heating the thermal oil system. The exhaust gas passes an exhaust gas boiler which is usually placed near the engine top or in the funnel.

It should be noted that the exhaust gas temperature and flow rate are influenced by the ambient conditions, for which reason this should be considered when the exhaust gas boiler is planned. At specified MCR, the maximum recommended pressure loss across the exhaust gas boiler is normally 150 mm WC.

This pressure loss depends on the pressure losses in the rest of the system as mentioned above. Therefore, if an exhaust gas silencer/spark arrester is not installed, the acceptable pressure loss across the boiler may be somewhat higher than the max. of 150 mm WC, whereas, if an exhaust gas silencer/spark arrester is installed, it may be necessary to reduce the maximum pressure loss.

The above mentioned pressure loss across the exhaust gas boiler must include the pressure losses from the inlet and outlet transition pieces.



178 33 46-7.4

Fig. 15.04.01b: Exhaust gas system, two or more TCs

Exhaust gas silencer

The typical octave band sound pressure levels from the diesel engine's exhaust gas system – at a distance of one meter from the top of the exhaust gas uptake – are shown in Fig.15.04.02.

The need for an exhaust gas silencer can be decided based on the requirement of a maximum permissible noise level at a specific position.

The exhaust gas noise data is valid for an exhaust gas system without boiler and silencer, etc.

The noise level is at nominal MCR at a distance of one metre from the exhaust gas pipe outlet edge at an angle of 30° to the gas flow direction.

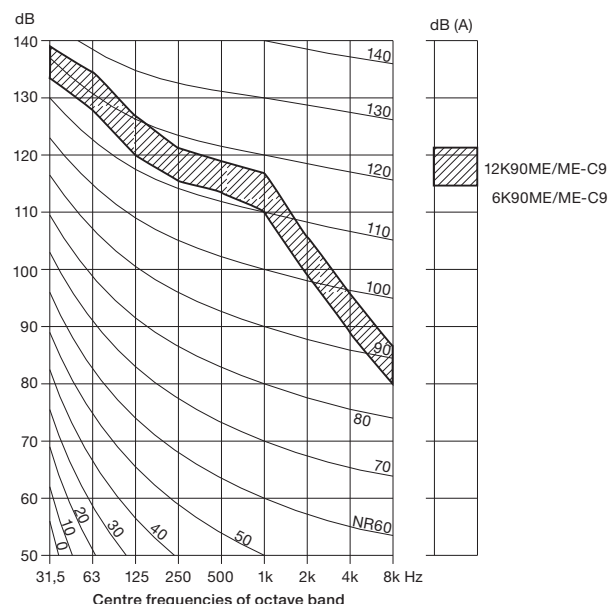
For each doubling of the distance, the noise level will be reduced by about 6 dB (far-field law).

When the noise level at the exhaust gas outlet to the atmosphere needs to be silenced, a silencer can be placed in the exhaust gas piping system after the exhaust gas boiler.

The exhaust gas silencer is usually of the absorption type and is dimensioned for a gas velocity of approximately 35 m/s through the central tube of the silencer.

An exhaust gas silencer can be designed based on the required damping of noise from the exhaust gas given on the graph.

In the event that an exhaust gas silencer is required – this depends on the actual noise level requirement on the bridge wing, which is normally maximum 60-70 dB(A) – a simple flow silencer of the absorption type is recommended. Depending on the manufacturer, this type of silencer normally has a pressure loss of around 20 mm WC at specified MCR.



178 61 92-4.0

Fig. 15.04.02: ISO's NR curves and typical sound pressure levels from the engine's exhaust gas system. The noise levels at nominal MCR and a distance of 1 metre from the edge of the exhaust gas pipe opening at an angle of 30 degrees to the gas flow and valid for an exhaust gas system – without boiler and silencer, etc.

Spark arrester

To prevent sparks from the exhaust gas being spread over deck houses, a spark arrester can be fitted as the last component in the exhaust gas system.

It should be noted that a spark arrester contributes with a considerable pressure drop, which is often a disadvantage.

It is recommended that the combined pressure loss across the silencer and/or spark arrester should not be allowed to exceed 100 mm WC at specified MCR. This depends, of course, on the pressure loss in the remaining part of the system, thus if no exhaust gas boiler is installed, 200 mm WC might be allowed.

Calculation of Exhaust Gas Back-Pressure

The exhaust gas back pressure after the turbo-charger(s) depends on the total pressure drop in the exhaust gas piping system.

The components, exhaust gas boiler, silencer, and spark arrester, if fitted, usually contribute with a major part of the dynamic pressure drop through the entire exhaust gas piping system.

The components mentioned are to be specified so that the sum of the dynamic pressure drop through the different components should, if possible, approach 200 mm WC at an exhaust gas flow volume corresponding to the specified MCR at tropical ambient conditions. Then there will be a pressure drop of 100 mm WC for distribution among the remaining piping system.

Fig. 15.05.01 shows some guidelines regarding resistance coefficients and back-pressure loss calculations which can be used, if the maker's data for back-pressure is not available at an early stage of the project.

The pressure loss calculations have to be based on the actual exhaust gas amount and temperature valid for specified MCR. Some general formulas and definitions are given in the following.

Exhaust gas data

M: exhaust gas amount at specified MCR in kg/sec.
T: exhaust gas temperature at specified MCR in °C

Please note that the actual exhaust gas temperature is different before and after the boiler. The exhaust gas data valid after the turbocharger may be found in Chapter 6.

Mass density of exhaust gas (ρ)

$$\rho \cong 1.293 \times \frac{273}{273 + T} \times 1.015 \text{ in kg/m}^3$$

The factor 1.015 refers to the average back-pressure of 150 mm WC (0.015 bar) in the exhaust gas system.

Exhaust gas velocity (v)

In a pipe with diameter D the exhaust gas velocity is:

$$v = \frac{M}{\rho} \times \frac{4}{\pi \times D^2} \text{ in m/s}$$

Pressure losses in pipes (Δp)

For a pipe element, like a bend etc., with the resistance coefficient ζ , the corresponding pressure loss is:

$$\Delta p = \zeta \times \frac{1}{2} \rho v^2 \times \frac{1}{9.81} \text{ in mm WC}$$

where the expression after ζ is the dynamic pressure of the flow in the pipe.

The friction losses in the straight pipes may, as a guidance, be estimated as :

1 mm WC per 1 diameter length

whereas the positive influence of the up-draught in the vertical pipe is normally negligible.

Pressure losses across components (Δp)

The pressure loss Δp across silencer, exhaust gas boiler, spark arrester, rain water trap, etc., to be measured/ stated as shown in Fig. 15.05.01 (at specified MCR) is normally given by the relevant manufacturer.

Total back-pressure (Δp_M)

The total back-pressure, measured/stated as the static pressure in the pipe after the turbocharger, is then:

$$\Delta p_M = \Sigma \Delta p$$

where Δp incorporates all pipe elements and components etc. as described:

Δp_M has to be lower than 350 mm WC.

(At design stage it is recommended to use max. 300 mm WC in order to have some margin for fouling).

Measuring Back Pressure

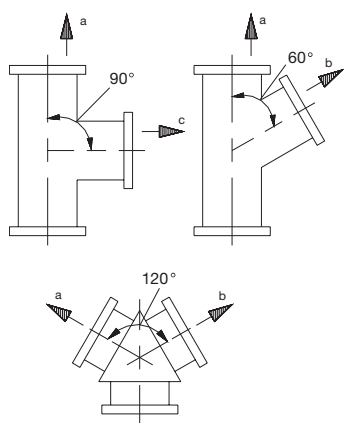
At any given position in the exhaust gas system, the total pressure of the flow can be divided into dynamic pressure (referring to the gas velocity) and static pressure (referring to the wall pressure, where the gas velocity is zero).

At a given total pressure of the gas flow, the combination of dynamic and static pressure may change, depending on the actual gas velocity. The measurements, in principle, give an indication of the wall pressure, i.e., the static pressure of the gas flow.

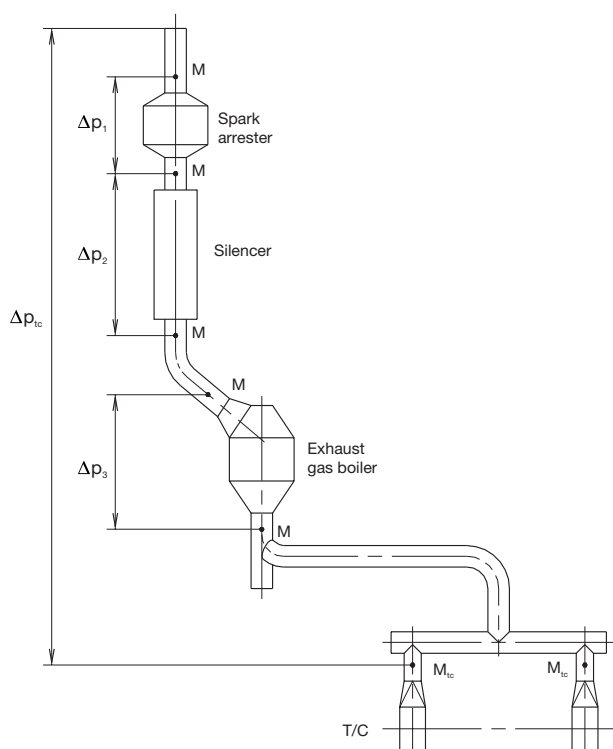
It is, therefore, very important that the back pressure measuring points are located on a straight part of the exhaust gas pipe, and at some distance from an 'obstruction', i.e. at a point where the gas flow, and thereby also the static pressure, is stable. Taking measurements, for example, in a transition piece, may lead to an unreliable measurement of the static pressure.

In consideration of the above, therefore, the total back pressure of the system has to be measured after the turbocharger in the circular pipe and not in the transition piece. The same considerations apply to the measuring points before and after the exhaust gas boiler, etc.

Pressure losses and coefficients of resistance in exhaust pipes

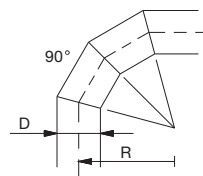
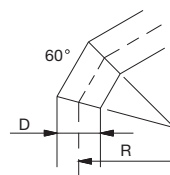
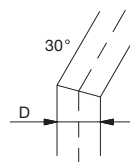
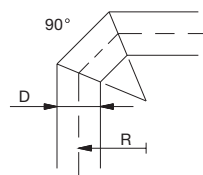
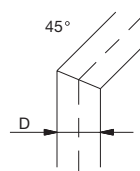


Change-over valves

Change-over valve
of type with constant
cross section $\zeta_a = 0.6$ to 1.2 $\zeta_b = 1.0$ to 1.5 $\zeta_c = 1.5$ to 2.0 Change-over valve
of type with volume $\zeta_a = \zeta_b = \text{about } 2.0$ 

M: Measuring points

178 32 09-1.0

 $R = D$ $\zeta = 0.28$ $R = 1.5D$ $\zeta = 0.20$ $R = 2D$ $\zeta = 0.17$  $R = D$ $\zeta = 0.16$ $R = 1.5D$ $\zeta = 0.12$ $R = 2D$ $\zeta = 0.11$  $\zeta = 0.05$  $R = D$ $\zeta = 0.45$ $R = 1.5D$ $\zeta = 0.35$ $R = 2D$ $\zeta = 0.30$  $\zeta = 0.14$ Outlet from
top of exhaust
gas uptake $\zeta = 1.00$ Inlet (from
turbocharger) $\zeta = -1.00$

178 06 85-3.0

Fig. 15.05.01: Pressure losses and coefficients of resistance in exhaust pipes

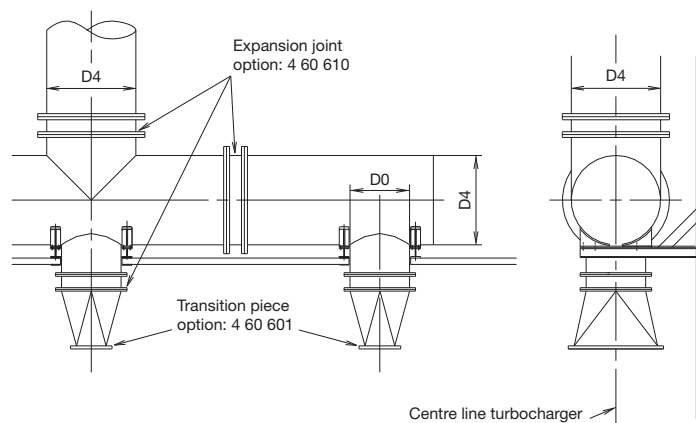
Forces and Moments at Turbocharger

This section is available on request

Diameter of Exhaust Gas Pipes

The exhaust gas pipe diameters listed in Table 15.07.02 are based on the exhaust gas flow capacity according to ISO ambient conditions and an exhaust gas temperature of 250 °C.

The exhaust gas velocities and mass flow listed apply to collector pipe D4. The table also lists the diameters of the corresponding exhaust gas pipes D0 for various numbers of turbochargers installed.



178 09 39-5.2

Fig. 15.07.01: Exhaust pipe system, with turbocharger located on exhaust side of engine

Gas velocity				Exhaust gas pipe diameters				
35 m/s	40 m/s	45 m/s	50 m/s	D0				D4
Gas mass flow				1 T/C	2 T/C	3 T/C	4 T/C	
kg/s	kg/s	kg/s	kg/s	[DN]	[DN]	[DN]	[DN]	[DN]
74.2	84.8	95.4	106.0	N.A.	1,400	1,150	1,000	2,000
81.8	93.5	105.2	116.9	N.A.	1,500	1,200	1,050	2,100
89.8	102.6	115.5	128.3	N.A.	1,600	1,300	1,100	2,200
98.1	112.2	126.2	140.2	N.A.	1,600	1,300	1,150	2,300
106.9	122.1	137.4	152.7	N.A.	1,700	1,400	1,200	2,400
116.0	132.5	149.1	165.6	N.A.	1,800	1,400	1,300	2,500
125.4	143.3	161.2	179.2	N.A.	1,800	1,500	1,300	2,600
135.2	154.6	173.9	193.2	N.A.	N.A.	1,600	1,400	2,700
145.5	166.2	187.0	207.8	N.A.	N.A.	1,600	1,400	2,800
156.0	178.3	200.6	222.9	N.A.	N.A.	1,700	1,500	2,900
167.0	190.8	214.7	238.5	N.A.	N.A.	1,700	1,500	3,000
178.3	203.8	229.2	254.7	N.A.	N.A.	1,800	1,600	3,100
190.0	217.1	244.3	271.4	N.A.	N.A.	1,800	1,600	3,200

Table 15.07.02: Exhaust gas pipe diameters and exhaust gas mass flow at various velocities

Engine Control System

16

Engine Control System ME

The Engine Control System for the ME engine is prepared for conventional remote control, having an interface to the Bridge Control system and the Local Operating Panel (LOP).

A Multi-Purpose Controller (MPC) is applied as control unit for specific tasks described below: ACU, CCU, ECU, and EICU. The control units are all built on the same identical piece of hardware and differ only in the software installed.

The layout of the Engine Control System is shown in Figs. 16.01.01a and b, the mechanical-hydraulic system is shown in Figs. 16.01.02a and b, and the pneumatic system, shown in Fig. 16.01.03.

The ME system has a high level of redundancy. It has been a requirement to its design that no single failure related to the system may cause the engine to stop. Furthermore, the ME system has been designed so that a single failure in most cases will not, or only slightly, affect the performance or power availability.

Main Operating Panel (MOP)

In the engine control room a MOP screen is located, which is a Personal Computer with a touch screen as well as a trackball from where the engineer can carry out engine commands, adjust the engine parameters, select the running modes, and observe the status of the control system.

A conventional marine approved PC is also located in the engine control room serving as a back-up unit for the MOP.

Engine Control Unit (ECU)

For redundancy purposes, the control system comprises two ECUs operating in parallel and performing the same task, one being a hot stand-by for the other. If one of the ECUs fail, the other unit will take over the control without any interruption.

The ECUs perform such tasks as:

- Speed governor functions, start/stop sequences, timing of fuel injection, timing of exhaust valve activation, timing of starting valves, etc.
- Continuous running control of auxiliary functions handled by the ACUs
- Alternative running modes and programs.

Cylinder Control Unit (CCU)

The control system includes one CCU per cylinder. The CCU controls the electronic exhaust Valve Activation (FIVA) and the Starting Air Valves (SAV), in accordance with the commands received from the ECU.

All the CCUs are identical, and in the event of a failure of the CCU for one cylinder only this cylinder will automatically be put out of operation.

It should be noted that any electronic part could be replaced without stopping the engine, which will revert to normal operation immediately after the replacement of the defective unit.

Auxiliary Control Unit (ACU)

The control of the auxiliary equipment on the engine is normally divided among three ACUs so that, in the event of a failure of one unit, there is sufficient redundancy to permit continuous operation of the engine.

The ACUs perform the control of the auxiliary blowers, the control of the electrically and engine driven hydraulic oil pumps of the Hydraulic Power Supply (HPS) unit, etc.

Engine Interface Control Unit (EICU)

The two EICUs perform such tasks as interface with the surrounding control systems, see Fig. 16.01.01a and b. The two redundant EICU units operate in parallel.

The EICUs are located either in the Engine Control Room (ECR) or in the engine room.

In basic execution, the EICUs are placed in the Cabinet for EICUs, EoD: 4 65 601. Optionally, the EICUs can be placed in the ECS Common Control Cabinet, option: 4 65 602, with the ACUs, CCUs and ECUs. See Figs. 16.01a and b.

Local Operating Panel (LOP)

In normal operating the engine can be controlled from either the bridge or from the engine control room.

Alternatively, the LOP can be activated. This redundant control is to be considered as a substitute for the previous Engine Side Control console mounted directly onto the MC engine.

The LOP is as standard placed on the engine.

From the LOP, the basic functions are available, such as starting, engine speed control, stopping, reversing, and the most important engine data are displayed.

Control Network

The MOP, the backup MOP and the MPCs are interconnected by means of the doubled Control Network, A and B respectively.

The maximum length of Control Network cabling between the furthestmost units on the engine and in the Engine Control Room (an EICU or a MOP) is 160 meter.

Should the layout of the ship make longer Control Network cabling necessary, a Control Network Repeater must be inserted to amplify the signals and divide the cable into segments no longer than 160 meter. For instance, where the Engine Control Room and the engine room are located far apart.

Power Supply

Supply voltage, nominal	24 V DC
Supply voltage, operational limits	20 V - 30 V
Supply voltage, max. ripple voltage	± 1 Vpp or 1 Vrms, whichever is lowest

Hydraulic Power Supply (HPS)

The purpose of the HPS unit is to deliver the necessary high pressure hydraulic oil flow to the Hydraulic Cylinder Units (HCU) on the engine at the required pressure (approx. 300 bar) during start-up as well as in normal service.

As hydraulic medium, normal lubricating oil is used, and it is in the standard execution taken from the main lubricating oil system of the engine.

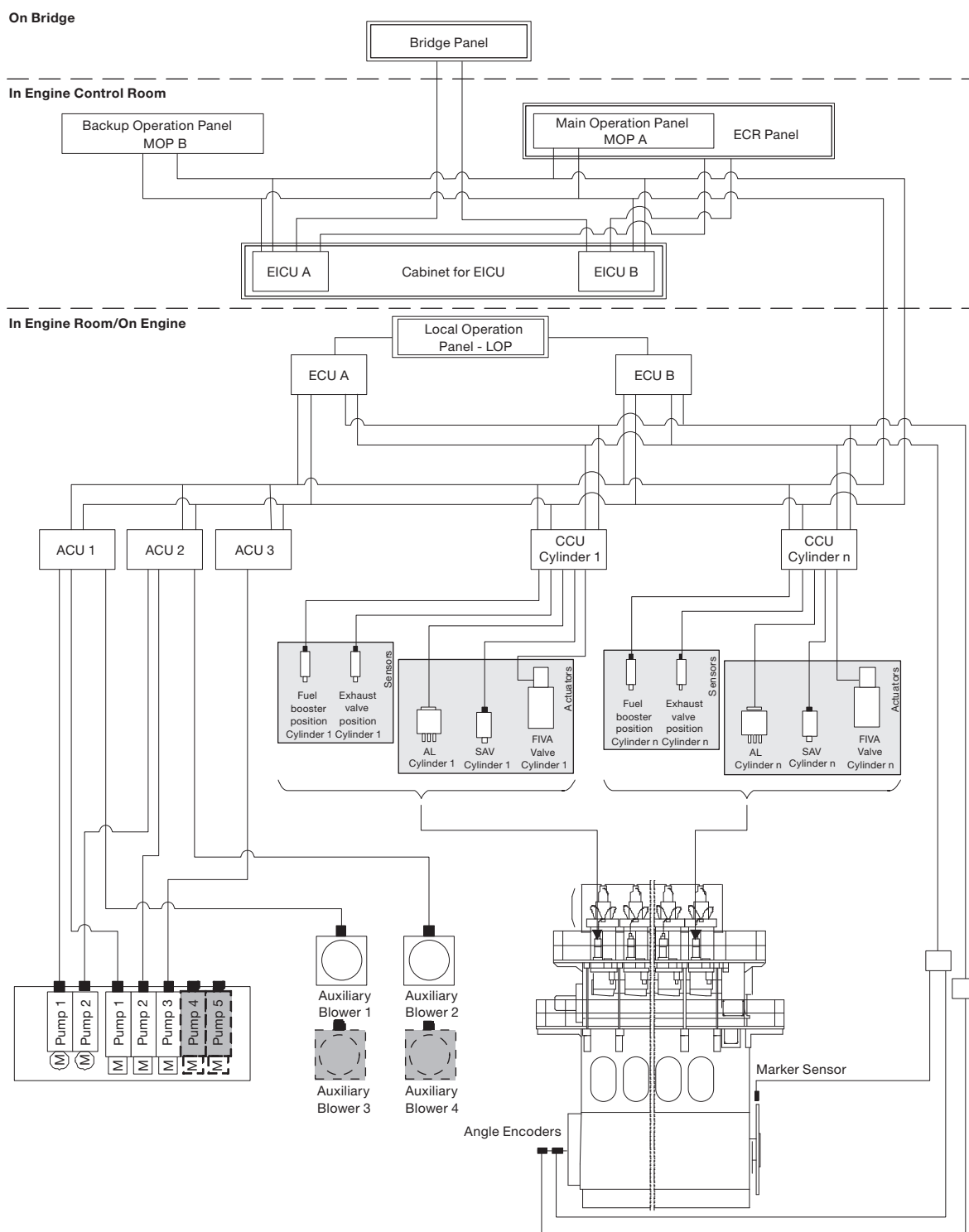
The HPS unit can be driven either mechanically from the engine crankshaft, see Fig. 16.01.02.

The multiple pump configuration with standby pumps ensures redundancy with regard to the hydraulic power supply. The control of the engine driven pumps and electrical pumps are divided between the three ACUs.

The high pressure pipes between the HPS unit and the HCU are of the double walled type, having a leak detector. Emergency running is possible using the outer pipe as pressure containment for the high pressure oil supply.

The sizes and capacities of the HPS unit depend on the engine type. Further details about the HPS and the lubricating oil/hydraulic oil system can be found in Chapter 8.

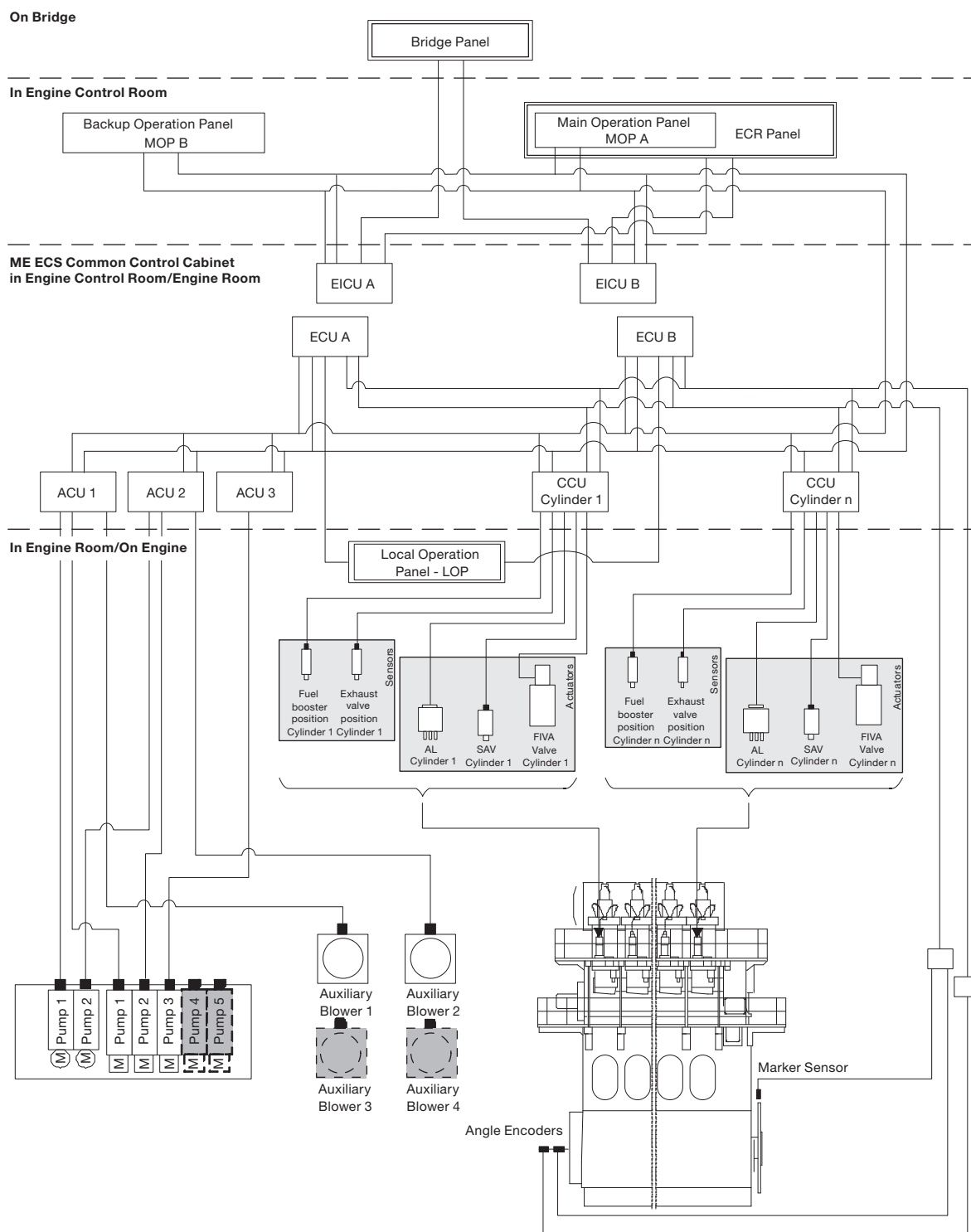
Engine Control System Layout with Cabinet for EICU



178 61 91-2.0

Fig. 16.01.01a: Engine Control System layout with cabinet for EICU for mounting in ECR or ER, EoD: 4 65 601

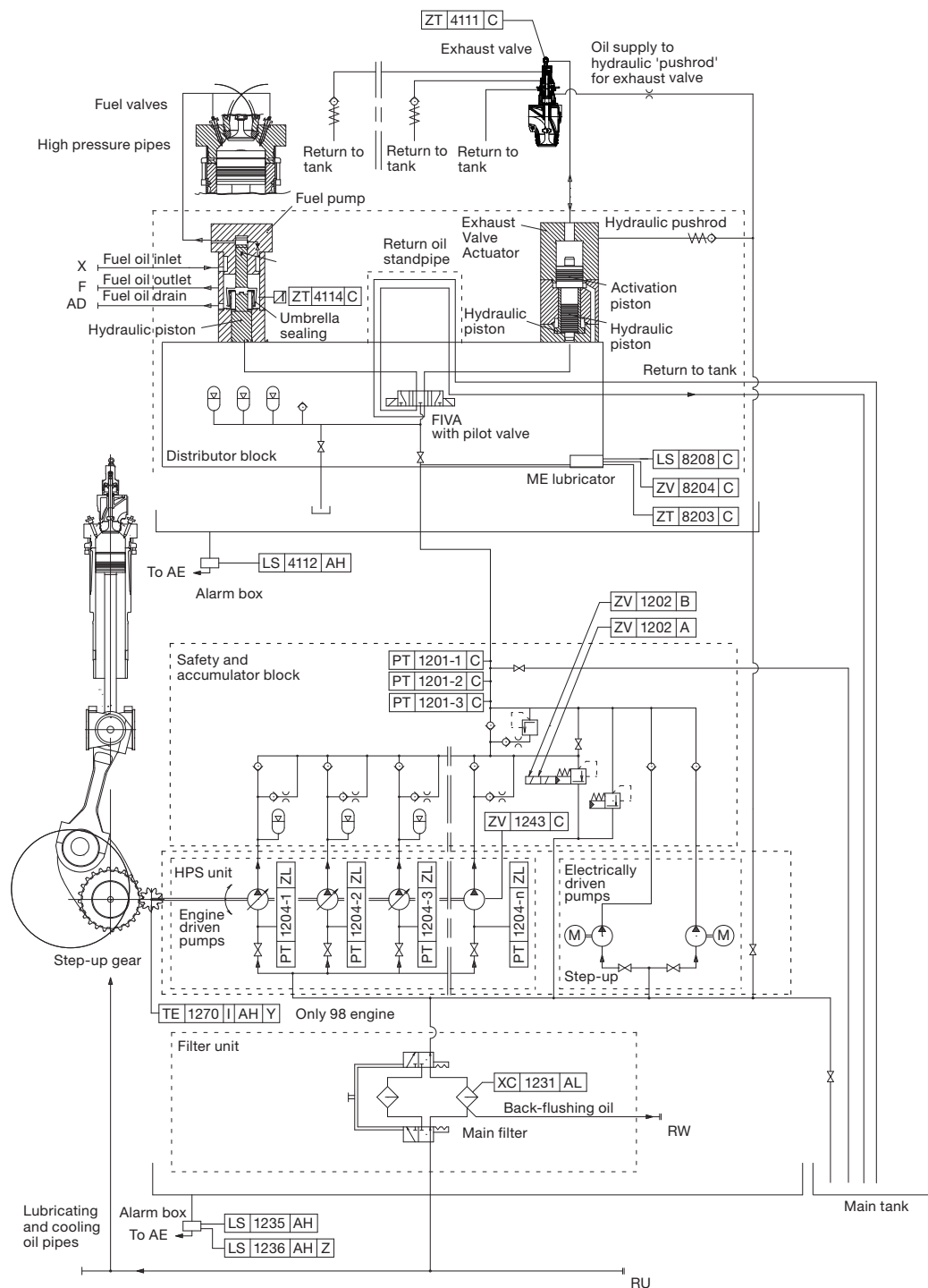
Engine Control System Layout with Common Control Cabinet



178 61 76-9.1

Fig. 16.01.01b: Engine Control System layout with ECS Common Control Cabinet for mounting in ECR or ER, option: 4 65 602

Mechanical-hydraulic System with Hydraulic Power Supply Unit on Engine

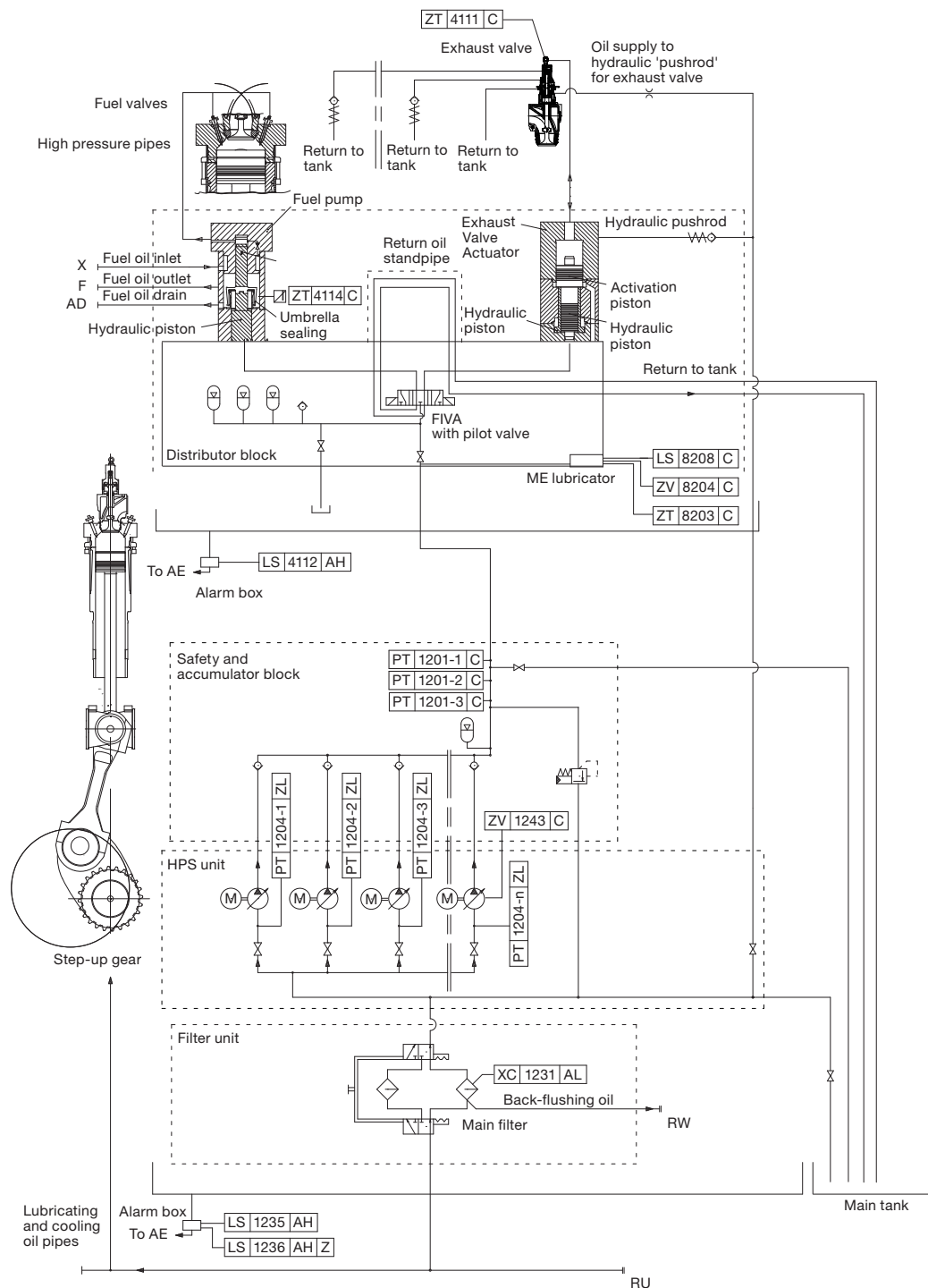


The letters refer to list of 'Counterflanges'
Th item No. refer to 'Guidance Values Automation'

515 75 30-9.2.0

Fig. 16.01.02: Mechanical-hydraulic System with Hydraulic Power Supply Unit on Engine, 300 bar, common supply

Mechanical-hydraulic System with Hydraulic Power Supply Unit in Ship



The letters refer to list of 'Counterflanges'
Th item No. refer to 'Guidance Values Automation'

515 75 49-1.1.0

Fig. 16.01.02b: Mechanical-hydraulic System with Hydraulic Power Supply Unit in ship, 300 bar, common supply.
Example from S90/80ME-C engine

Engine Control System Interface to Surrounding Systems

To support the navigator, the vessels are equipped with a ship control system, which includes subsystems to supervise and protect the main propulsion engine.

Alarm system

The alarm system has no direct effect on the ECS. The alarm alerts the operator of an abnormal condition.

The alarm system is an independent system, in general covering more than the main engine itself, and its task is to monitor the service condition and to activate the alarms if a normal service limit is exceeded.

The signals from the alarm sensors can be used for the slow down function as well as for remote indication.

Slow down system

Some of the signals given by the sensors of the alarm system are used for the 'Slow down request' signal to the ECS of the main engine.

Safety system

The engine safety system is an independent system with its respective sensors on the main engine, fulfilling the requirements of the respective classification society and MAN Diesel.

If a critical value is reached for one of the measuring points, the input signal from the safety system must cause either a cancellable or a non-cancellable shut down signal to the ECS.

The safety system must be compatible with the remote control system.

Telegraph system

This system enables the navigator to transfer the commands of engine speed and direction of rotation from the Bridge, the engine control room or the Local Operating Panel (LOP), and it provides signals for speed setting and stop to the ECS.

The engine control room and the LOP are provided with combined telegraph and speed setting units.

Remote Control system

The remote control system normally has two alternative control stations:

- the bridge control
- the engine control room control

The remote control system is to be delivered by an approved supplier and it must be compatible with the safety system.

Power Management System

The system handles the supply of electrical power onboard, i. e. the starting and stopping of the generating sets as well as the activation / deactivation of the main engine Shaft Generator (SG), if fitted.

The normal function involves starting, synchronising, phasing-in, transfer of electrical load and stopping of the generators based on the electrical load of the grid on board.

The activation / deactivation of the SG is to be done within the engine speed range which fulfils the specified limits of the electrical frequency.

Auxiliary equipment system

The input signals for 'Auxiliary system ready' are given partly through the Remote Control system based on the status for:

- fuel oil system
- lube oil system
- cooling water systems

and partly from the ECS itself:

- turning gear disengaged
- main starting valve 'open'
- control air valve for sealing air 'open'
- control air valve for air spring 'open'
- auxiliary blowers running
- hydraulic power supply ready.

Monitoring systems

In addition to the PMI system type PT/S off-line required for the installation of the ME engine, PMI on-line and CoCoS-EDS can be used to improve the monitoring of the engine.

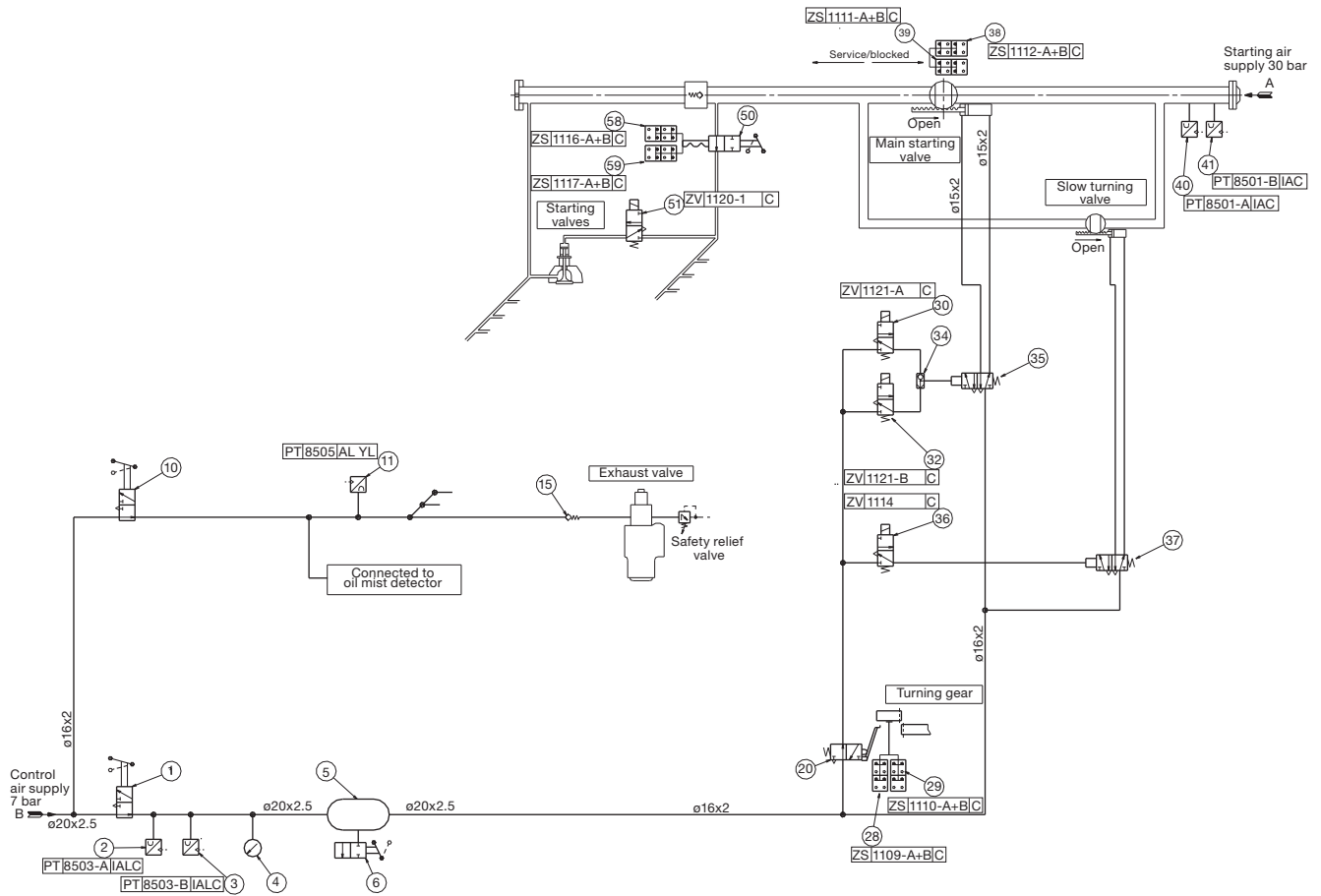
A description of the systems can be found in Chapter 18 of this project guide.

Instrumentation

Chapter 18 in the Project Guide for the specific engine type includes lists of instrumentation for:

- The CoCoS-EDS on-line system
- The class requirements and MAN Diesel's requirements for alarms, slow down and shut down for Unattended Machinery Spaces.

Pneumatic Manoeuvring Diagram



178 49 73-8.2

Fig. 16.01.03: Pneumatic Manoeuvring Diagram

Vibration Aspects

17

Vibration Aspects

The vibration characteristics of the two-stroke low speed diesel engines can for practical purposes be split up into four categories, and if the adequate countermeasures are considered from the early project stage, the influence of the excitation sources can be minimised or fully compensated.

In general, the marine diesel engine may influence the hull with the following:

- External unbalanced moments
These can be classified as unbalanced 1st and 2nd order external moments, which need to be considered only for certain cylinder numbers
- Guide force moments
- Axial vibrations in the shaft system
- Torsional vibrations in the shaft system.

The external unbalanced moments and guide force moments are illustrated in Fig. 17.01.01.

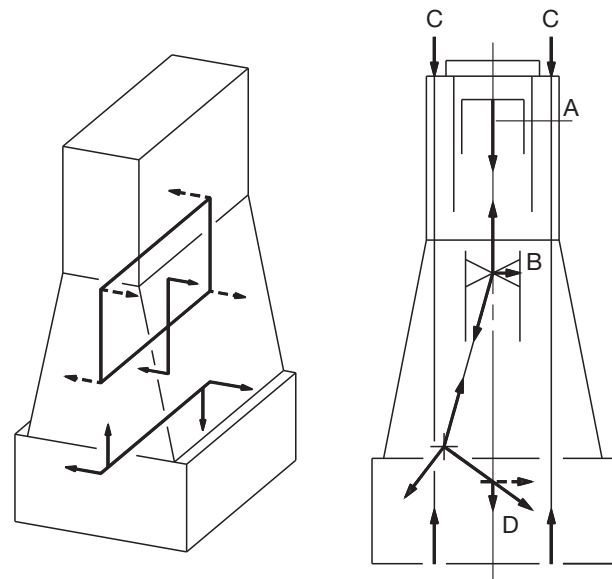
In the following, a brief description is given of their origin and of the proper countermeasures needed to render them harmless.

External unbalanced moments

The inertia forces originating from the unbalanced rotating and reciprocating masses of the engine create unbalanced external moments although the external forces are zero.

Of these moments, the 1st order (one cycle per revolution) and the 2nd order (two cycles per revolution) need to be considered for engines with a low number of cylinders. On 7-cylinder engines, also the 4th order external moment may have to be examined. The inertia forces on engines with more than 6 cylinders tend, more or less, to neutralise themselves.

Countermeasures have to be taken if hull resonance occurs in the operating speed range, and if the vibration level leads to higher accelerations and/or velocities than the guidance values given by international standards or recommendations (for instance related to special agreement between shipowner and shipyard). The natural frequency of the hull depends on the hull's rigidity and distribution of masses, whereas the vibration level at resonance depends mainly on the magnitude of the external moment and the engine's position in relation to the vibration nodes of the ship.



A – Combustion pressure
B – Guide force
C – Staybolt force
D – Main bearing force



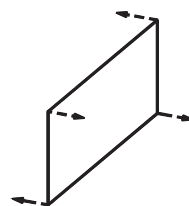
1st order moment vertical 1 cycle/rev.
2nd order moment, vertical 2 cycle/rev.



1st order moment, horizontal
1 cycle/rev.



Guide force moment,
H transverse Z cycles/rev.
Z is 1 or 2 times number of cylinder



Guide force moment,
X transverse Z cycles/rev.
Z = 1, 2, 3 ... 11, 12, 14

178 06 82-8.2

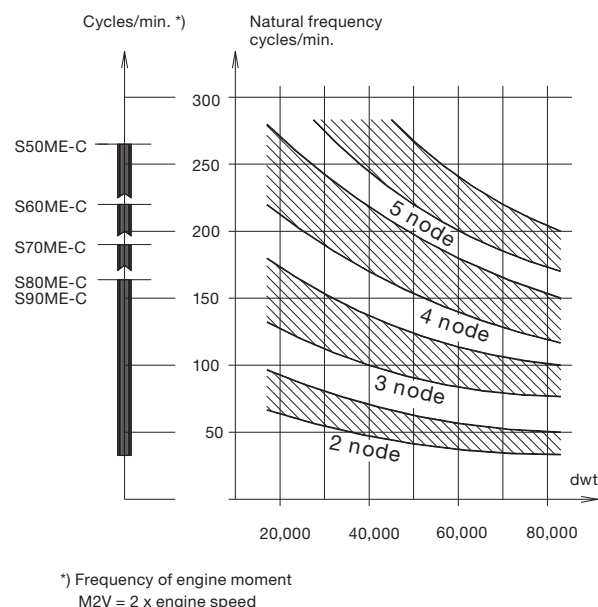
Fig. 17.01.01: External unbalanced moments and guide force moments

2nd Order Moments on 6-cylinder Engines

The 2nd order moment acts only in the vertical direction. Precautions need only to be considered for 6-cylinder engines in general.

Resonance with the 2nd order moment may occur in the event of hull vibrations with more than 3 nodes. Contrary to the calculation of natural frequency with 2 and 3 nodes, the calculation of the 4 and 5-node natural frequencies for the hull is a rather comprehensive procedure and often not very accurate, despite advanced calculation methods.

A 2nd order moment compensator comprises two counter-rotating masses running at twice the engine speed.



178 60 91-7.0

Fig. 17.02.01: Statistics of vertical hull vibrations in tankers and bulk carriers

Compensator solutions

Several solutions are available to cope with the 2nd order moment, as shown in Fig. 17.03.02, out of which the most cost efficient one can be chosen in the individual case, e.g.:

- 1) No compensators, if considered unnecessary on the basis of natural frequency, nodal point and size of the 2nd order moment.
- 2) A compensator mounted on the aft end of the engine, driven by chain, option: 4 31 203.
- 3) A compensator mounted on the fore end, driven from the crankshaft through a separate chain drive, option: 4 31 213.

As standard, the compensators reduce the external 2nd order moment to a level as for a 7-cylinder engine or less.

Briefly speaking, solution 1) is applicable if the node is located far from the engine, or the engine is positioned more or less between nodes. Solution 2) or 3) should be considered where one of the engine ends is positioned in a node or close to it, since a compensator is inefficient in a node or close to it and therefore superfluous.

A decision regarding the vibrational aspects and the possible use of compensators must be taken at the contract stage. If no experience is available from sister ships, which would be the best basis for deciding whether compensators are necessary or not, it is advisable to make calculations to determine which of the solutions should be applied.

Preparation for compensators

If compensator(s) are initially omitted, the engine can be delivered prepared for compensators to be fitted on engine fore end later on, but the decision to prepare or not must be taken at the contract stage, option: 4 31 212. Measurements taken during the sea trial, or later in service and with fully loaded ship, will be able to show if compensator(s) have to be fitted at all.

If no calculations are available at the contract stage, we advise to make preparations for the fitting of a compensator in the steering compartment, see Section 17.03.

Basic design regarding compensators

For 6-cylinder engines with mechanically driven HPS, the basic design regarding 2nd order moment compensators is:

- With compensator aft, EoD: 4 31 203
- Prepared for compensator fore, EoD: 4 31 212

For 6-cylinder engines with electrically driven HPS, the basic design regarding 2nd order moment compensators is:

- With electric balancer RotComp, EoD: 4 31 255
- Prepared for compensator fore, EoD: 4 31 212

The available options are listed in the Extent of Delivery.

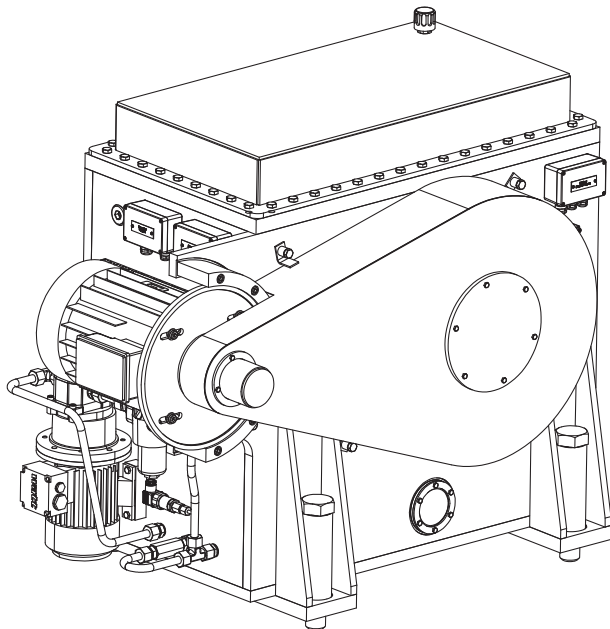
Electrically Driven Moment Compensator

If it is decided not to use chain driven moment compensators and, furthermore, not to prepare the main engine for compensators to be fitted later, another solution can be used, if annoying 2nd order vibrations should occur: An electrically driven moment compensator synchronised to the correct phase relative to the external force or moment can neutralise the excitation.

This type of compensator needs an extra seating fitted, preferably, in the steering gear room where vibratory deflections are largest and the effect of the compensator will therefore be greatest.

The electrically driven compensator will not give rise to distorting stresses in the hull, but it is more expensive than the engine-mounted compensators. It does, however, offer several advantages over the engine mounted solutions:

- When placed in the steering gear room, the compensator is not as sensitive to the positioning of the node as the compensators 2) and 3) mentioned in Section 17.02.
- The decision whether or not to install compensators can be taken at a much later stage of a project, since no special version of the engine structure has to be ordered for the installation.
- No preparation for a later installation nor an extra chain drive for the compensator on the fore end of the engine is required. This saves the cost of such preparation, often left unused.
- Compensators could be retrofit, even on ships in service, and also be applied to engines with a higher number of cylinders than is normally considered relevant, if found necessary.
- The compensator only needs to be active at speeds critical for the hull girder vibration. Thus, it may be activated or deactivated at specified speeds automatically or manually.
- Combinations with and without moment compensators are not required in torsional and axial vibration calculations, since the electrically driven moment compensator is not part of the mass-elastic system of the crankshaft.



178 57 45-6.0

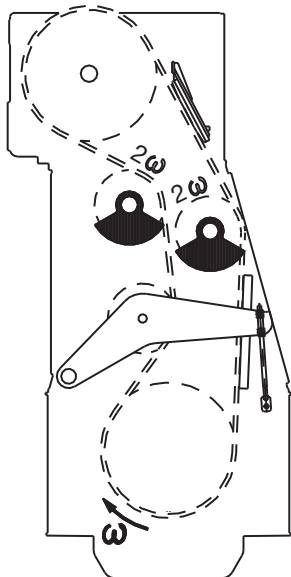
Furthermore, by using the compensator as a vibration exciter a ship's vibration pattern can easily be identified without having the engine running, e.g. on newbuildings at an advanced stage of construction. If it is verified that a ship does not need the compensator, it can be removed and re-used on another ship.

It is a condition for the application of the rotating force moment compensator that no annoying longitudinal hull girder vibration modes are excited. Based on our present knowledge, and confirmed by actual vibration measurements onboard a ship, we do not expect such problems.

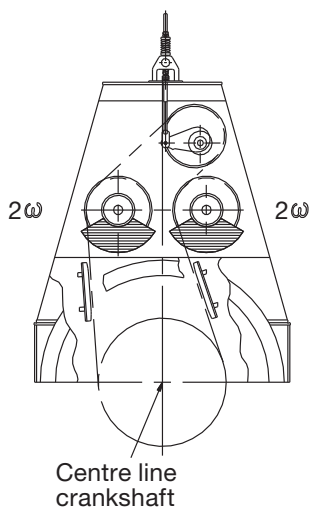
Further to compensating 2nd order moments, electrically driven moment compensators are also available for balancing other forces and moments. The available options are listed in the Extent of Delivery.

Fig. 17.03.01: MAN Diesel 2nd order electrically driven moment compensator, separately mounted, option: 4 31 255

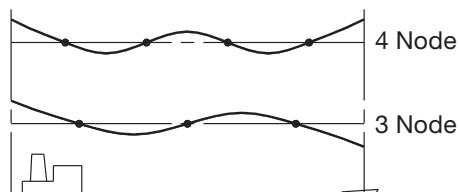
Moment compensator
Aft end, option: 4 31 203



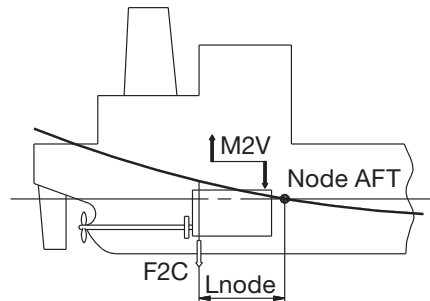
Moment compensator
Fore end, option: 4 31 213



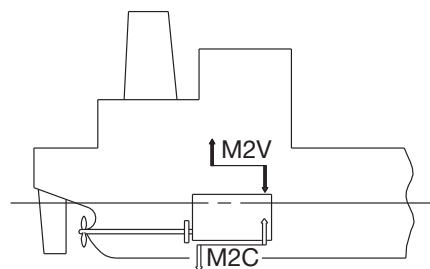
3 and 4-node vertical hull girder mode



Compensating moment
 $F_{2C} \times L_{node}$
outbalances M_{2V}

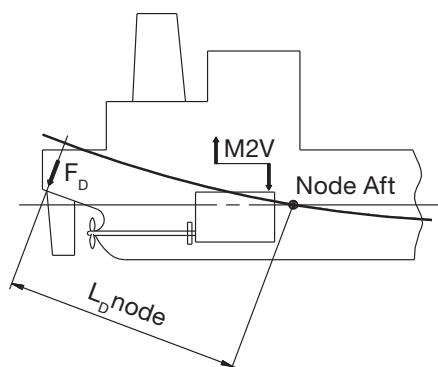


Moment from compensator
 M_{2C} reduces M_{2V}



Electrically driven moment compensator

Compensating moment
 $F_D \times L_{node}$
outbalances M_{2V}



178 27 10-4.1

Fig. 17.03.02: Compensation of 2nd order vertical external moments

Power Related Unbalance

To evaluate if there is a risk that 1st and 2nd order external moments will excite disturbing hull vibrations, the concept Power Related Unbalance (PRU) can be used as a **guidance**, see Table 17.04.01 below.

$$\text{PRU} = \frac{\text{External moment}}{\text{Engine power}} \text{ Nm/kW}$$

With the PRU-value, stating the external moment relative to the engine power, it is possible to give an estimate of the risk of hull vibrations for a specific engine.

Based on service experience from a great number of large ships with engines of different types and cylinder numbers, the PRU-values have been classified in four groups as follows:

PRU Nm/kW	Need for compensator
0 - 60	Not relevant
60 - 120	Unlikely
120 - 220	Likely
220 -	Most likely

K90ME9 – 5,720 kW/cyl at 94 r/min

	5 cyl.	6 cyl.	7 cyl.	8 cyl.	9 cyl.	10 cyl.	11 cyl.	12 cyl.	14 cyl.
PRU acc. to 1st order, Nm/kW	N.a.	0.0	10.5	1.4	28.1	18.6	1.4	0.0	N.a.
PRU acc. to 2nd order, Nm/kW	N.a.	143.8	35.8	0.0	31.3	5.8	2.0	0.0	N.a.

Based on external moments in layout point L₁

N.a. Not applicable

Table 17.04.01: Power Related Unbalance (PRU) values in Nm/kW

Calculation of External Moments

In the table at the end of this chapter, the external moments (M_1) are stated at the speed (n_1) and MCR rating in point L₁ of the layout diagram. For other speeds (n_A), the corresponding external moments (M_A) are calculated by means of the formula:

$$M_A = M_1 \times \left\{ \frac{n_A}{n_1} \right\}^2 \text{ kNm}$$

(The tolerance on the calculated values is 2.5%).

Guide Force Moments

The so-called guide force moments are caused by the transverse reaction forces acting on the crossheads due to the connecting rod/crankshaft mechanism. These moments may excite engine vibrations, moving the engine top athwartships and causing a rocking (excited by H-moment) or twisting (excited by X-moment) movement of the engine as illustrated in Fig. 17.05.01.

The guide force moments corresponding to the MCR rating (L_y) are stated in Table 17.07.01.

Top bracing

The guide force moments are harmless except when resonance vibrations occur in the engine/double bottom system.

As this system is very difficult to calculate with the necessary accuracy, MAN Diesel strongly recommend, as standard, that top bracing is installed between the engine's upper platform brackets and the casing side.

The vibration level on the engine when installed in the vessel must comply with MAN Diesel vibration limits as stated in Fig. 17.05.02.

We recommend using the hydraulic top bracing which allow adjustment to the loading conditions of the ship. Mechanical top bracings with stiff connections are available on request.

With both types of top bracing, the above-mentioned natural frequency will increase to a level where resonance will occur above the normal engine speed. Details of the top bracings are shown in Chapter 05.

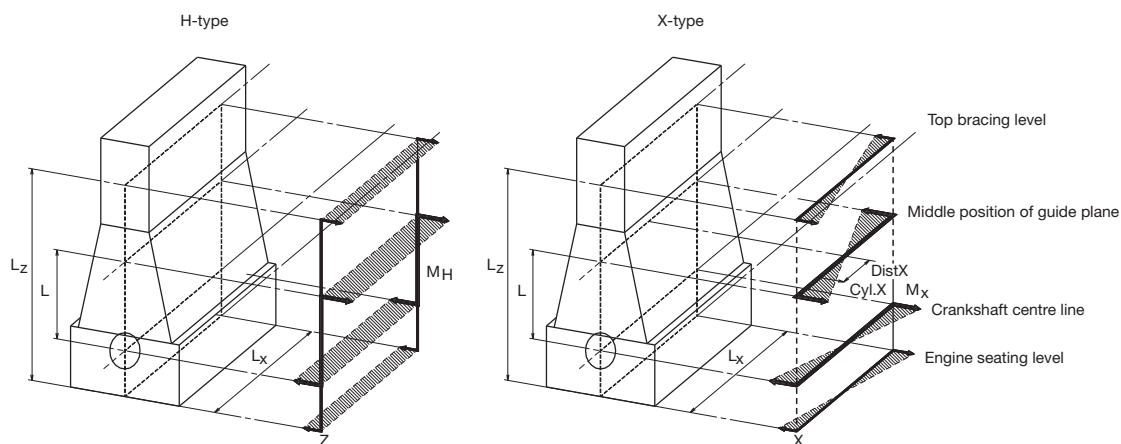
Definition of Guide Force Moments

Over the years it has been discussed how to define the guide force moments. Especially now that complete FEM-models are made to predict hull/engine interaction, the propeller definition of these moments has become increasingly important.

H-type Guide Force Moment (M_H)

Each cylinder unit produces a force couple consisting of:

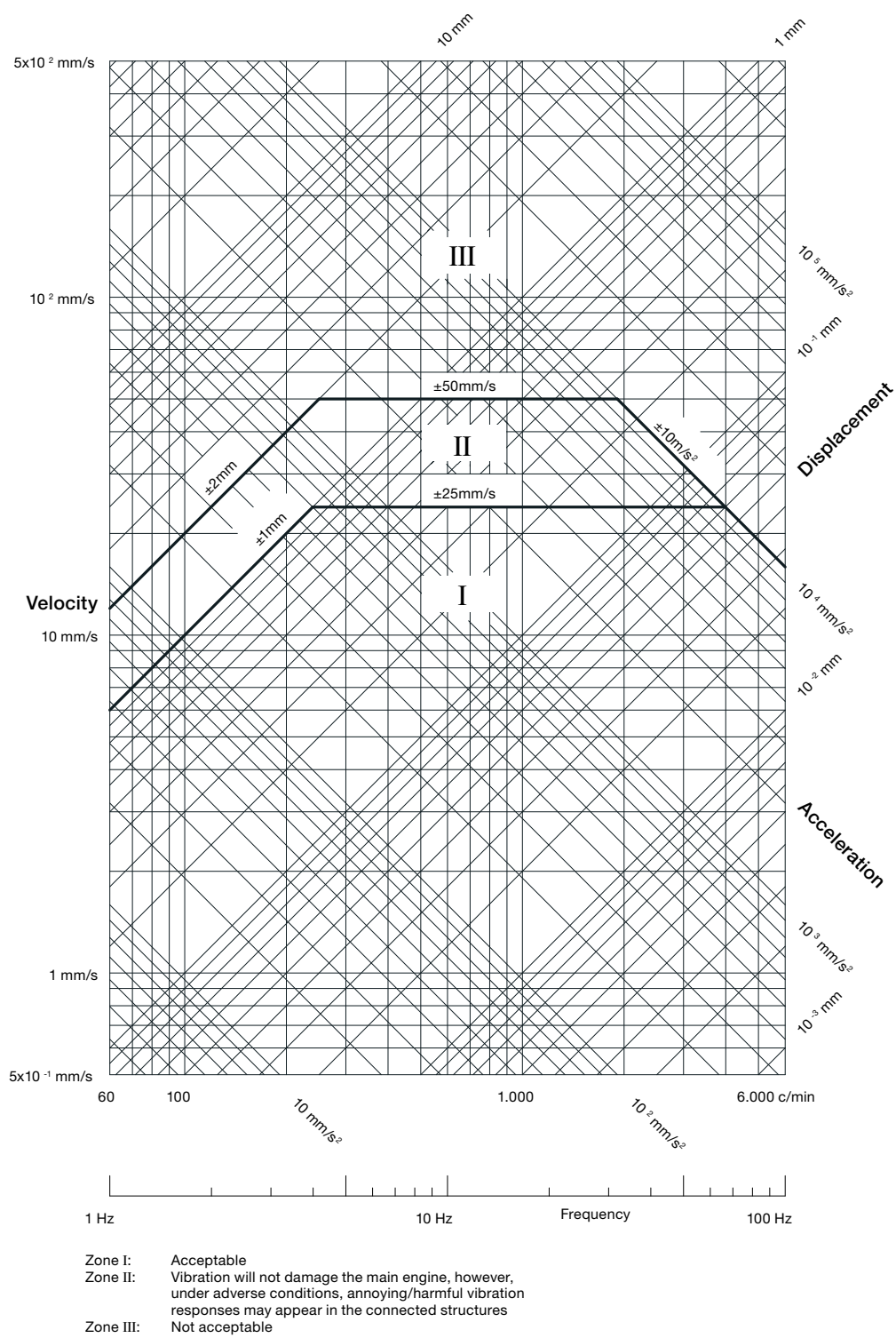
1. A force at crankshaft level
2. Another force at crosshead guide level. The position of the force changes over one revolution as the guide shoe reciprocates on the guide.



178 06 81-6.4

Fig. 17.05.01: H-type and X-type guide force moments

Vibration Limits Valid for Single Order Harmonics



078 81 27-6.1

Fig.17.05.02: Vibration limits

As the deflection shape for the H-type is equal for each cylinder, the Nth order H-type guide force moment for an N-cylinder engine with regular firing order is:

$$N \times M_{H(\text{one cylinder})}$$

For modelling purposes, the size of the forces in the force couple is:

$$\text{Force} = M_H/L \text{ [kN]}$$

where L is the distance between crankshaft level and the middle position of the crosshead guide (i.e. the length of the connecting rod).

As the interaction between engine and hull is at the engine seating and the top bracing positions, this force couple may alternatively be applied in those positions with a vertical distance of (L_z). Then the force can be calculated as:

$$\text{Force}_z = M_H/L_z \text{ [kN]}$$

Any other vertical distance may be applied so as to accommodate the actual hull (FEM) model.

The force couple may be distributed at any number of points in the longitudinal direction. A reasonable way of dividing the couple is by the number of top bracing and then applying the forces at those points.

$$\text{Force}_{Z, \text{ one point}} = \text{Force}_{Z, \text{ total}} / N_{\text{top bracing, total}} \text{ [kN]}$$

X-type Guide Force Moment (M_x)

The X-type guide force moment is calculated based on the same force couple as described above. However, as the deflection shape is twisting the engine, each cylinder unit does not contribute with an equal amount. The centre units do not contribute very much whereas the units at each end contribute much.

A so-called 'Bi-moment' can be calculated (Fig. 17.05.01):

$$\text{'Bi-moment'} = \sum [\text{force-couple(cyl.X)} \times \text{distX}] \text{ in kNm}^2$$

The X-type guide force moment is then defined as:

$$M_x = \text{'Bi-Moment'}/L \text{ kNm}$$

For modelling purpose, the size of the four (4) forces can be calculated:

$$\text{Force} = M_x/L_x \text{ [kN]}$$

where:

L_x is the horizontal length between 'force points'.

Similar to the situation for the H-type guide force moment, the forces may be applied in positions suitable for the FEM model of the hull. Thus the forces may be referred to another vertical level L_z above the crankshaft centre line. These forces can be calculated as follows:

$$\text{Force}_{Z, \text{ one point}} = \frac{M_x \times L}{L_x \times L_z} \text{ [kN]}$$

In order to calculate the forces, it is necessary to know the lengths of the connecting rods = L, which are:

Engine Type	L in mm	Engine Type	L in mm
K98ME6/7	3,220	S65ME-C8	2,730
K98ME-C6/7	3,090	S65ME-GI8	2,730
S90ME-C7/8	3,270	S60ME-C7/8	2,460
K90ME9	3,320	S60ME-GI8	2,460
K90ME-C9	3,120	S60ME-B8	2,460
K90ME-C6	3,159	L60ME-C7/8	2,280
S80ME-C9	3,450	S50ME-C7/8	2,050
S80ME-C7/8	3,280	S50ME-B9	2,114
K80ME-C9	2,975	S50ME-B8	2,050
K80ME-C6	2,920	S46ME-B8	1,980
S70ME-C7/8	2,870	S40ME-B9	1,770
S70ME-GI8	2,870	S35ME-B9	1,550
L70ME-C7/8	2,660		

Axial Vibrations

When the crank throw is loaded by the gas pressure through the connecting rod mechanism, the arms of the crank throw deflect in the axial direction of the crankshaft, exciting axial vibrations. Through the thrust bearing, the system is connected to the ship's hull.

Generally, only zero-node axial vibrations are of interest. Thus the effect of the additional bending stresses in the crankshaft and possible vibrations of the ship's structure due to the reaction force in the thrust bearing are to be considered.

An axial damper is fitted as standard on all engines, minimising the effects of the axial vibrations, EoD: 4 31 111.

Torsional Vibrations

The reciprocating and rotating masses of the engine including the crankshaft, the thrust shaft, the intermediate shaft(s), the propeller shaft and the propeller are for calculation purposes considered as a system of rotating masses (inertias) interconnected by torsional springs. The gas pressure of the engine acts through the connecting rod mechanism with a varying torque on each crank throw, exciting torsional vibration in the system with different frequencies.

In general, only torsional vibrations with one and two nodes need to be considered. The main critical order, causing the largest extra stresses in the shaft line, is normally the vibration with order equal to the number of cylinders, i.e., six cycles per revolution on a six cylinder engine. This resonance is positioned at the engine speed corresponding to the natural torsional frequency divided by the number of cylinders.

The torsional vibration conditions may, for certain installations require a torsional vibration damper, option: 4 31 105.

Plants with 11 or 12 cylinders require a torsional vibration damper.

Based on our statistics, this need **may arise** for the following types of installation:

- Plants with controllable pitch propeller
- Plants with unusual shafting layout and for special owner/yard requirements
- Plants with 8-cylinder engines.

The so-called QPT (Quick Passage of a barred speed range Technique), is an alternative to a torsional vibration damper, on a plant equipped with a controllable pitch propeller. The QPT could be implemented in the governor in order to limit the vibratory stresses during the passage of the barred speed range.

The application of the QPT, option: 4 31 108, has to be decided by the engine maker and MAN Diesel based on final torsional vibration calculations.

Six-cylinder engines, require special attention. On account of the heavy excitation, the natural frequency of the system with one-node vibration should be situated away from the normal operating speed range, to avoid its effect. This can be achieved by changing the masses and/or the stiffness of the system so as to give a much higher, or much lower, natural frequency, called undercritical or overcritical running, respectively.

Owing to the very large variety of possible shafting arrangements that may be used in combination with a specific engine, only detailed torsional vibration calculations of the specific plant can determine whether or not a torsional vibration damper is necessary.

Undercritical running

The natural frequency of the one-node vibration is so adjusted that resonance with the main critical order occurs about 35-45% above the engine speed at specified MCR.

Such undercritical conditions can be realised by choosing a rigid shaft system, leading to a relatively high natural frequency.

The characteristics of an undercritical system are normally:

- Relatively short shafting system
- Probably no tuning wheel
- Turning wheel with relatively low inertia
- Large diameters of shafting, enabling the use of shafting material with a moderate ultimate tensile strength, but requiring careful shaft alignment, (due to relatively high bending stiffness)
- Without barred speed range.

Critical Running

When running undercritical, significant varying torque at MCR conditions of about 100-150% of the mean torque is to be expected.

This torque (propeller torsional amplitude) induces a significant varying propeller thrust which, under adverse conditions, might excite annoying longitudinal vibrations on engine/double bottom and/or deck house.

The yard should be aware of this and ensure that the complete aft body structure of the ship, including the double bottom in the engine room, is designed to be able to cope with the described phenomena.

Overcritical running

The natural frequency of the one-node vibration is so adjusted that resonance with the main critical order occurs about 30-70% below the engine speed at specified MCR. Such overcritical conditions can be realised by choosing an elastic shaft system, leading to a relatively low natural frequency.

The characteristics of overcritical conditions are:

- Tuning wheel may be necessary on crankshaft fore end
- Turning wheel with relatively high inertia
- Shafts with relatively small diameters, requiring shafting material with a relatively high ultimate tensile strength
- With barred speed range, EoD: 4 07 015, of about $\pm 10\%$ with respect to the critical engine speed.

Torsional vibrations in overcritical conditions may, in special cases, have to be eliminated by the use of a torsional vibration damper.

Overcritical layout is normally applied for engines with more than four cylinders.

Please note:

We do not include any tuning wheel or torsional vibration damper in the standard scope of supply, as the proper countermeasure has to be found after torsional vibration calculations for the specific plant, and after the decision has been taken if and where a barred speed range might be acceptable.

For further information about vibration aspects, please refer to our publications:

An Introduction to Vibration Aspects

Vibration Characteristics of Two-stroke Engines

The publications are available at www.mandiesel.com under 'Quicklinks' → 'Technical Papers'

External Forces and Moments, K90ME9 Layout point L₁ - SFOC

No of cylinder :	6	7	8	9	10	11	12
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Firing type :	1-5-3-4-2-6	1-7-2-5-4-3-6	1-8-3-4-7-2-5-6	1-6-7-3-5-8-2-4-9	1-8-7-3-5-9-4-2-10-6	1-5-11-6-2-7-9-4-3-8-10	1-8-12-4-2-9-10-5-3-7-11-6
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External forces [kN] :							
1. Order : Horizontal	0	0	0	0	113	2	0
1. Order : Vertical	0	0	0	0	113	2	0
2. Order : Vertical	0	0	0	0	52	3	0
4. Order : Vertical	0	0	0	0	53	68	0
6. Order : Vertical	17	0	0	0	5	4	0
External moments [kNm] :							
1. Order : Horizontal a)	0	419	62	1,445	1,062	88	0
1. Order : Vertical a)	0	419	62	1,445	1,062	88	0
2. Order : Vertical	4,935 c)	1,433	0	1,611	333	123	0
4. Order : Vertical	254	722	294	366	564	662	508
6. Order : Vertical	0	1	0	73	36	6	0
Guide force H-moments in [kNm] :							
1 x No. of cyl.	2,400	1,772	1,259	738	239	188	208
2 x No. of cyl.	104	121	126	-	-	-	-
3 x No. of cyl.	-	-	-	-	-	-	-
Guide force X-moments in [kNm] :							
1. Order :	0	260	38	898	660	55	0
2. Order :	363	105	0	118	24	9	0
3. Order :	1,209	1,322	1,998	2,402	2,914	3,415	4,269
4. Order :	1,185	3,368	1,368	1,706	2,631	3,085	2,370
5. Order :	0	301	4,458	1,494	188	2,794	0
6. Order :	0	48	0	3,189	1,564	233	0
7. Order :	0	0	4	128	2,258	245	0
8. Order :	243	19	0	65	225	1,453	486
9. Order :	329	37	1	47	77	200	1,162
10. Order :	73	208	0	19	1	152	0
11. Order :	0	121	183	14	66	204	0
12. Order :	0	8	31	138	50	63	0
13. Order :	0	2	104	35	112	6	0
14. Order :	27	0	0	38	132	107	0
15. Order :	68	2	0	135	35	138	240
16. Order :	24	7	0	8	98	136	49

a) 1st order moments are, as standard, balanced so as to obtain equal values for horizontal and vertical moments for all cylinder numbers.

c) 6-cylinder engines can be fitted with 2nd order moment compensators on the aft and fore end, reducing the 2nd order external moment.

Table 17.07.01

Monitoring Systems and Instrumentation

18

Monitoring Systems and Instrumentation

The Engine Control System (ECS) can be supported by the computerised PMI system and the CoCoS-EDS on-line (Computer Controlled Surveillance-Engine Diagnostics System), both of which have been in service since 1994.

The monitoring system measures the main parameters of the engine and makes an evaluation of the general engine condition, indicating the countermeasures to be taken. This ensures that the engine performance is kept within the prescribed limits throughout the engine's lifetime.

In its basic design the ME-engine instrumentation consists of:

- Engine Control System
- Shut-down sensors, option: 4 75 124
- PMI system type PT/S off-line, option: 4 75 208

The optional extras are:

- CoCoS system
type EDS on-line, option: 4 09 660
- PMI system, on-line, option: 4 75 215

As most engines are sold for Unattended Machinery Spaces (UMS), the following option is normally included:

- Sensors for alarm, slow down and remote indication according to the classification society's and MAN Diesel's requirements for UMS, option: 4 75 127, see Section 18.04.

Sensors for CoCoS can be ordered, if required, as option: 4 75 129. They are listed in Section 18.03.

All instruments are identified by a combination of symbols and a position number as shown in Section 18.07.

PMI System, Type PT/S Off-line

On the ME-engines, the mechanical indicator system is replaced by a Pressure Analyser System for measurement of the cylinder combustion pressure.

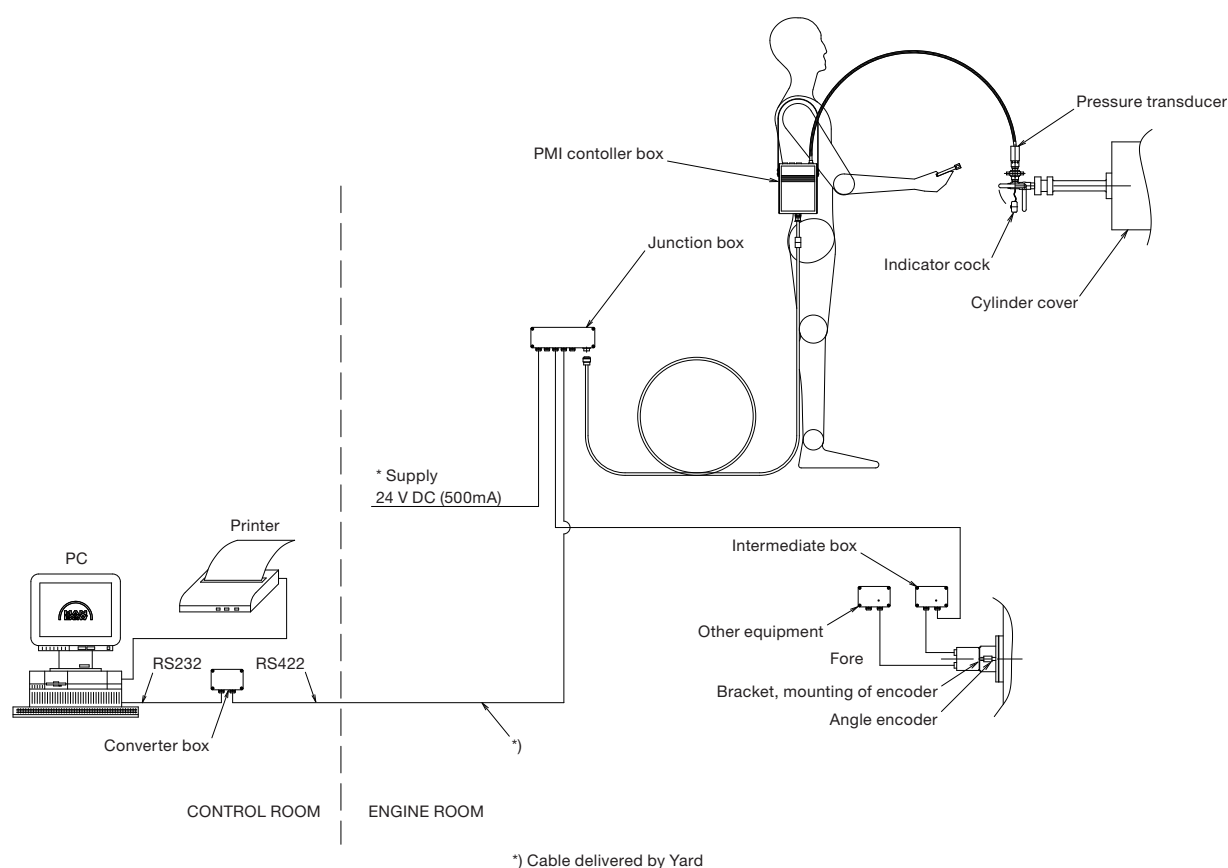
The PMI pressure analyser systems measures the engine's main parameters, such as cylinder pressure, scavenge air pressure, engine speed etc. enabling the engineer to run the diesel engine at its optimum performance.

This system gets its data from a high performance piezo-electric pressure transducer which is to be

mounted on the indicator valve. The transducer is moved from one cylinder to another in order to complete measurements on all cylinders.

The crankshaft position is determined by means of the same trigger system as for the engine control system.

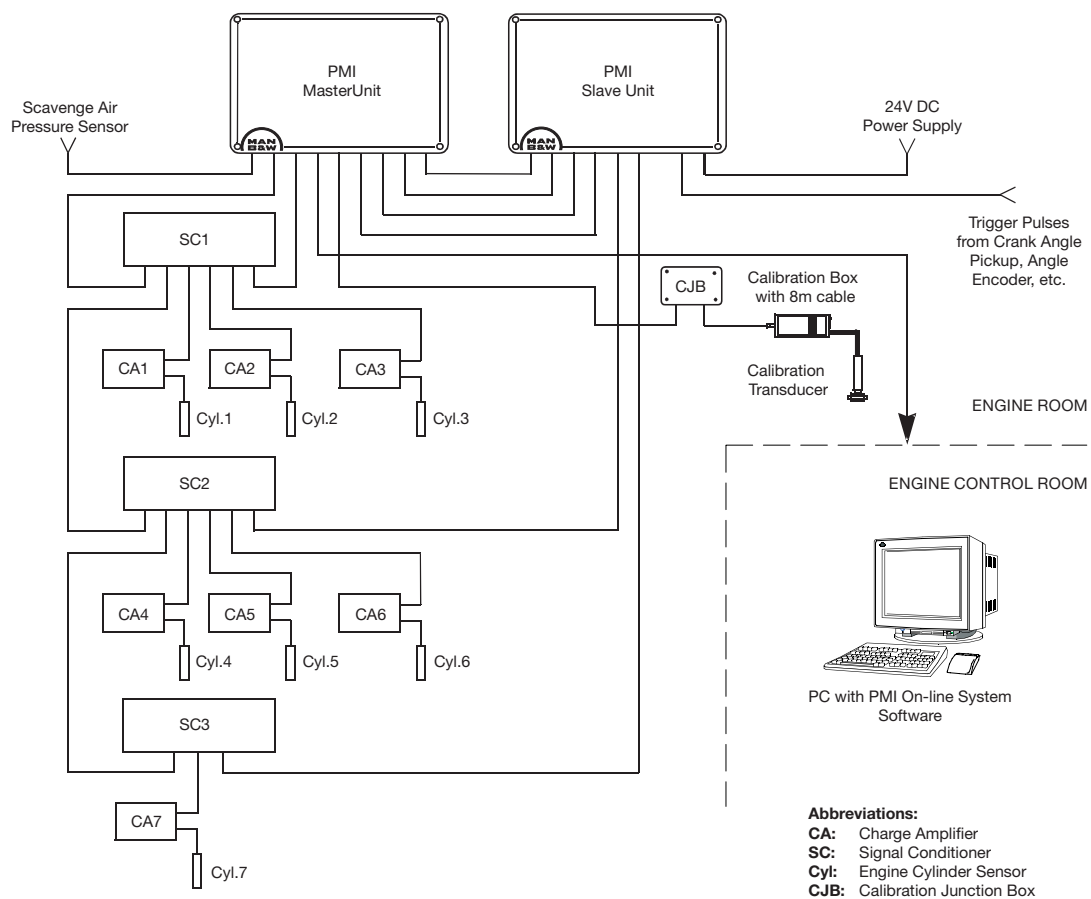
The PMI system compensates automatically for the twisting experienced by each section of the crankshaft due to the torque generated at different loads.



178 59 57-7.0

Fig. 18.02.01: PMI type PT/S off-line, 4 75 208

PMI System, Type On-line



178 51 47-7.0

Fig. 18.02.02: PMI type on-line, 4 75 215

CoCoS Systems

The **C**omputer **C**ontrolled **S**urveillance system is the family name of the software application products from the MAN Diesel group.

In order to obtain an easier, more versatile and continuous diagnostics system, the Engine Control System and the PMI System is recommended extended by the CoCoS-EDS products.

CoCoS-EDS

CoCoS-EDS, option: 4 09 660, assists in engine performance evaluation and provides detailed engine operation surveillance.

Key features are: online data logging, monitoring, trending, diagnostics and reporting.

Table 18.03.01 lists the sensors required to enable online diagnostics for the CoCoS-EDS, option: 4 75 129.

CoCoS-EDS Sensor List

Sensors required for the CoCoS-EDS online engine performance analysis, option: 4 75 129, see Table 18.03.01. All pressure gauges are measuring relative pressure, except for 'PT 8802 Ambient pressure'.

Sensor	Parameter name	No. sensors	Recommended range	Resolution 3)	Remark
Fuel oil system data					
PT 8001	Inlet pressure	1	0 - 10 bar	0.1 bar	
TE 8005	Inlet temperature	1	0 - 200 °C	0.1 °C	
Cooling water system					
PT 8421	Pressure air cooler inlet	A/C	0 - 4 bar	0.1 bar	
TE 8422	Temperature air cooler inlet	1	0 - 100 °C	0.1 °C	
TE 8423	Temperature air cooler outlet	A/C	0 - 100 °C	0.1 °C	
PDT 8424	dP cooling water across air cooler	A/C	0 - 800 mbar	0.1 mbar	
Scavenging air system					
PT 8601	Scavenge air receiver pressure	Rec.	0 - 4 bar	1 mbar	1)
TE 8605	Scavenge air cooler air inlet temperature	A/C	0 - 200 °C	0.1 °C	
PDT 8606	dP air across scavenge air cooler	A/C	0 - 100 mbar	0.1 mbar	
PDT 8607	dP air across T/C air intake filter	T/C	0 - 100 mbar	0.1 mbar	
TE 8608	Scavenge air cooler air outlet temperature	A/C	0 - 100 °C	0.1 °C	Optional if one T/C
TE 8609	Scavenge air receiver temperature	Rec.	0 - 100 °C	0.1 °C	
TE 8612	T/C air intake temperature	T/C	0 - 100 °C	0.1 °C	
Exhaust gas system					
TC 8701	Exhaust gas temperature at turbine inlet	T/C	0 - 600 °C	0.1 °C	
TC 8702	Exhaust gas temperature after exhaust valve	Cyl.	0 - 600 °C	0.1 °C	
PT 8706	Exhaust gas receiver pressure	Rec.	0 - 4 bar	0.01 bar	
TC 8707	Exhaust gas temperature at turbine outlet	T/C	0 - 600 °C	0.1 °C	
PT 8708	Turbine back pressure	T/C	0 - 100 mbar	0.1 mbar	
General data					
ZT 8801	Turbocharger speed	T/C	rpm	1 rpm	
PT 8802	Ambient pressure	1	900 - 1,100 mbar	1 mbar	Absolute!
ZT 4020	Engine speed	1	rpm	0.1 rpm	1)
XC 8810	Governor index (relative)	1	%	0.1 %	1)
-	Power take off/in from main engine shaft (PTO/PTI)	1	kW	1 kW	With option installed
Pressure measurement					
XC1401	Mean Indicated Pressure, MIP	Cyl.	bar	0.01 bar	2)
XC1402	Maximum Pressure, Pmax	Cyl.	bar	0.1 bar	2)
XC1403	Compression Pressure, Pcomp	Cyl.	bar	0.1 bar	2)
-	PMI online engine speed	Cyl.	rpm	0.1 rpm	2)

1) Signal acquired from Engine Control System (ECS)

2) In case of MAN Diesel PMI system: signal from PMI system. Other MIP systems: signal from manual input

3) Resolution of signals transferred to CoCoS-EDS (from the Alarm Monitoring System).

Table 18.03.01: List of sensors for CoCoS-EDS

Alarm – Slow Down and Shut Down System

The shut down system must be electrically separated from other systems by using independent sensors, or sensors common for the alarm system but with galvanically separated electrical circuits, i.e. one sensor with two sets of electrically independent terminals. The list of sensors are shown in Table 18.04.04.

Basic safety system design and supply

The basic safety sensors for a MAN Diesel engine are designed for Unattended Machinery Space (UMS) and comprises:

- the temperature sensors and pressure sensors that are specified in the 'MAN Diesel' column for shut down in Table 18.04.04.

These sensors are included in the basic Extent of Delivery, EOD: 4 75 124.

Alarm and slow down system design and supply

The basic alarm and slow down sensors for a MAN Diesel engine are designed for Unattended Machinery Space (UMS) and comprises:

- the sensors for alarm and slow down, option: 4 75 127.

The shut down and slow down panels can be ordered as options: 4 75 610, 4 75 614 or 4 75 615 whereas the alarm panel is yard's supply, as it normally includes several other alarms than those for the main engine.

For practical reasons, the sensors for the engine itself are normally delivered from the engine supplier, so they can be wired to terminal boxes on the engine.

The number and position of the terminal boxes depends on the degree of dismantling specified in the Dispatch Pattern for the transportation of the engine based on the lifting capacities available at the engine maker and at the yard.

Alarm, slow down and remote indication sensors

The International Association of Classification Societies (IACS) indicates that a common sensor can be used for alarm, slow down and remote indication.

A general view of the alarm, slow down and shut down systems is shown in Fig. 18.04.01.

Tables 18.04.02 and 18.04.03 show the requirements by MAN Diesel for alarm and slow down and for UMS by the classification societies (Class), as well as IACS' recommendations.

The number of sensors to be applied to a specific plant for UMS is the sum of requirements of the classification society, the Buyer and MAN Diesel.

If further analogue sensors are required, they can be ordered as option: 4 75 128.

Slow down functions

The slow down functions are designed to safeguard the engine components against overloading during normal service conditions and to keep the ship manoeuvrable if fault conditions occur.

The slow down sequence must be adapted to the actual plant parameters, such as for FPP or CPP, engine with or without shaft generator, and to the required operating mode.

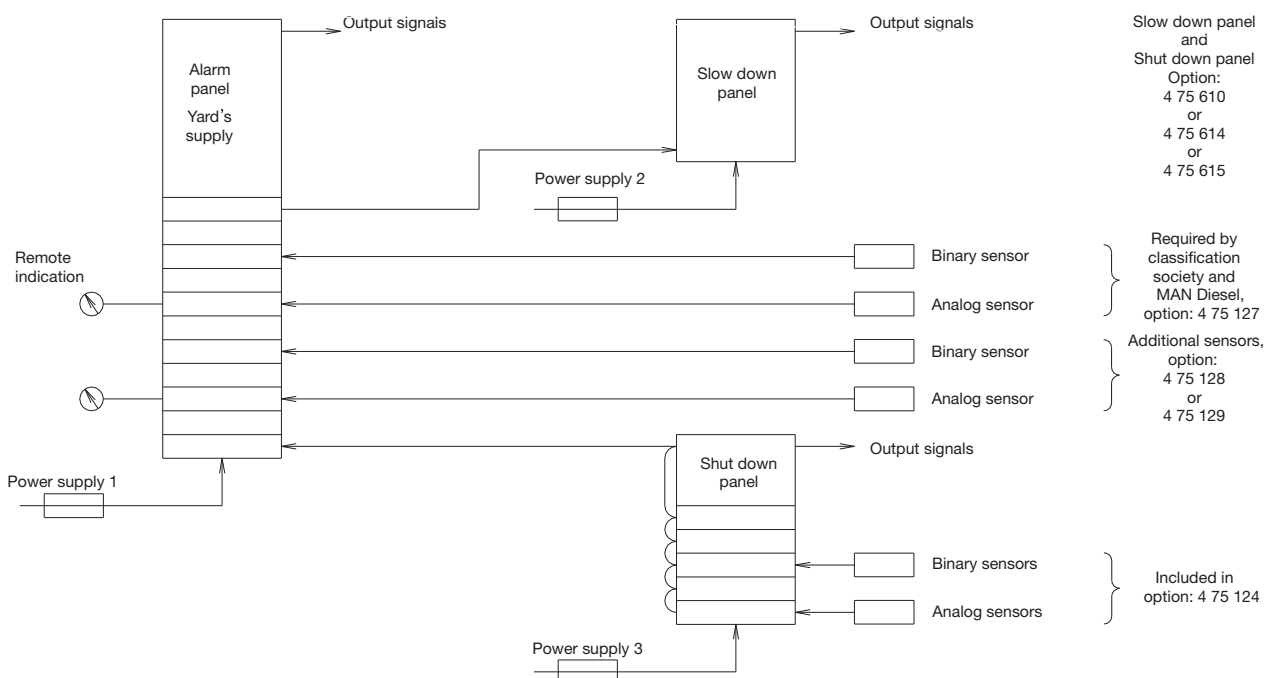
General outline of the electrical system

The figure shows the concept approved by all classification societies.

The shut down panel and slow down panel can be combined for some makers.

The classification societies permit having common sensors for slow down, alarm and remote indication.

One common power supply might be used, instead of the three indicated, provided that the systems are equipped with separate fuses.



178 30 10-0.5

Fig. 18.04.01: Panels and sensors for alarm and safety systems

Alarms for UMS – Class and MAN Diesel requirements

ABS	BV	CCS	DNV	GL	KR	LR	NK	RINA	RS	IACS	MAN Diesel	Sensor and function	Point of location
Fuel oil													
1	1	1	1	1	1	1	1	1	1	1	1	PT 8001 AL	Fuel oil, inlet engine
1	1	1	1	1	1	1	1	1	1	1	1	LS 8006 AH	Leakage from high pressure pipes
Lubricating oil													
1	1	1	1	1	1	1	1	1	1	1	1	TE 8106 AH	Thrust bearing segment
1	1	1	1	1	1	1	1	1	1	1	1	PT 8108 AL	Lubricating oil inlet to main engine
1	1	1	1	1	1	1	1	1	1	1	1	TE 8112 AH	Lubricating oil inlet to main engine
1	1	1	1	1	1		1	1	1	1	1	TE 8113 AH	Piston cooling oil outlet/cylinder
1	1	1	1	1	1		1	1	1	1	1	FS 8114 AL	Piston cooling oil outlet/cylinder
1	1	1		1	1	1		1	1	1	1	TE 8117 AH	Turbocharger lubricating oil outlet from turbocharger/turbocharger
											1	TE 8123 AH	Main bearing oil outlet temperature/main bearing (S40/35ME-B9 only)
											1	XC 8126 AH	Bearing wear (All types except S40/35ME-B9); sensor common for XC 8126/27
											1	XS 8127 A	Bearing wear detector failure (All types except S40/35ME-B)
		1		1		1	1				1	PDS 8140 AH	Lubricating oil differential pressure – cross filter
											1	XS 8150 AH	Water in lubricating oil; sensor common for XS 8150/51/52
											1	XS 8151 AH	Water in lubricating oil – too high
											1	XS 8152 A	Water in lubricating oil sensor not ready
MAN B&W Alpha Lubrication													
											1	LS 8212 AL	Small box for heating element, low level

1 Indicates that the sensor is required.

The sensors in the MAN Diesel column are included for Unattended Machinery Spaces (UMS), option: 4 75 127.

The sensor identification codes and functions are listed in Table 18.07.01.

The tables are liable to change without notice, and are subject to latest class requirements.

Table 18.04.02a: Alarm functions for UMS

Alarms for UMS – Class and MAN Diesel requirements

ABS	BV	CCS	DNV	GL	KR	LR	NK	RINA	RS	IACS	MAN Diesel	Sensor and function	Point of location
													Hydraulic Power Supply
												1 XC 1231 A	Automatic main lube oil filter, failure (Boll & Kirch)
													Cooling water
1	1	1	1	1	1	1	1	1	1	1	1	PT 8401 AL	Jacket cooling water inlet
												1 PDS/PDT 8403 AL	Jacket cooling water across engine; to be calculated in alarm system from sensor no. 8402 and 8413
												1 TE 8407 AL	Jacket cooling water inlet
1	1	1	1	1	1	1	1	1	1	1	1	TE 8408 AH	Jacket cooling water outlet, cylinder
												1 PT 8413 I	Jacket cooling water outlet, common pipe
1	1	1		1	1	1	1	1	1	1	1	PT 8421 AL	Cooling water inlet air cooler
												1 TE 8422 AH	Cooling water inlet air cooler/air cooler
													Compressed air
1	1	1		1	1	1	1	1	1	1	1	PT 8501 AL	Starting air inlet to main starting valve
1	1	1	1	1	1	1	1	1+	1	1	1	PT 8503 AL	Control air inlet and finished with engine
												1 PT 8505 AL	Air inlet to air cylinder for exhaust valve
													Scavenge air
												1 PS 8604 AL	Scavenge air, auxiliary blower, failure (Only ME-B)
												1 TE 8609 AH	Scavenge air receiver
1	1	1	1	1	1	1	1	1	1	1	1	TE 8610 AH	Scavenge air box – fire alarm, cylinder/cylinder
1	1	1		1	1	1	1	1	1	1	1	LS 8611 AH	Water mist catcher – water level

1 Indicates that the sensor is required.

The sensors in the MAN Diesel column are included for Unattended Machinery Spaces (UMS), option: 4 75 127.
The sensor identification codes and functions are listed in Table 18.07.01.

The tables are liable to change without notice, and are subject to latest class requirements.

- ☐ Select one of the alternatives
 + Alarm for high pressure, too
 ÷ Alarm for low pressure, too

Table 18.04.02b: Alarm functions for UMS

Alarms for UMS – Class and MAN Diesel requirements

ABS	BV	CCS	DNV	GL	KR	LR	NK	RINA	RS	IACS	MAN Diesel	Sensor and function	Point of location	
Exhaust gas														
1	1	1	1	1	1	(1)	1	1	1	1	1	TC 8701 AH	Exhaust gas before turbocharger/turbocharger	
1	1		1		1	1	1	1	1	1	1	TC 8702 AH	Exhaust gas after exhaust valve, cylinder/cylinder	
1	1	1	1	1	1	1	1	1	1	1		TC 8707 AH	Exhaust gas outlet turbocharger/turbocharger (Yard's supply)	
Miscellaneous														
			1									ZT 8801 AH	Turbocharger overspeed	
			1									WT 8805 AH	Vibration of turbocharger	
											1	WT 8812 AH	Axial vibration monitor 2)	
1	1	<input type="checkbox"/>	1	1	1	1	1	1	1	1	1	XS 8813 AH	Oil mist in crankcase/cylinder; sensor common for XS 8813/14	
	1											1	XS 8814 AL	Oil mist detector failure
												1	XC 8816 I	Shaftline earthing device
												1	TE 8820 AH	Cylinder liner monitoring/cylinder 3)
Engine Control System														
1	1	1	1	1	1	1	1	1	1	1	1	XC 2201 A	Power failure	
1	1		1	1		1	1	1	1	1	1	XC 2202 A	ME common failure	
Power Supply Units to Alarm System														
												1	XC 2901 A	Low voltage ME power supply A
												1	XC 2902 A	Low voltage ME power supply B
												1	XC 2903 A	Earth failure ME power supply

1 Indicates that the sensor is required.

The sensors in the MAN Diesel column are included for Unattended Machinery Spaces (UMS), option: 4 75 127.

The sensor identification codes and functions are listed in Table 18.07.01.

The tables are liable to change without notice, and are subject to latest class requirements.

(1) May be combined with TC 8702 AH where turbocharger is mounted directly on the exhaust manifold.

2) Required for: K-ME-C6/7 and K98ME6/7 engines with 11 and 14 cylinders.
S-ME-C7/8, S-ME-GI7/8, and L-ME-C7/8 engines with 5 and 6 cylinders.
S-ME-B8/9 engines with 5 and 6 cylinders mainly.
(For K90ME9, K/S-ME-C9, and S50ME-B9 data is available on request).

3) Required for: K98ME/ME-C6/7, S90ME-C7/8, K90ME/ME-C9 and K80ME-C9 engines

☐ Alarm for overheating of main, crank and crosshead bearings, option: 4 75 134.

Table 18.04.02c: Alarm functions for UMS

Slow down for UMS – Class and MAN Diesel requirements

ABS	BV	CCS	DNV	GL	KR	LR	NK	RINA	RS	IACS	MAN Diesel	Sensor and function	Point of location
1	1	1	1	1	1	1	1	1	1	1	1	TE 8106 YH	Thrust bearing segment
1	1	1	1*	1	1	1	1	1	1	1	1	PT 8108 YL	Lubricating oil inlet to main engine
				1		1						TE 8112 YH	Lubricating oil inlet to main engine
1	1	1	1	1	1		1	1	1	1	1	TE 8113 YH	Piston cooling oil outlet/cylinder
1	1	1	1	1	1		1	1	1	1	1	FS 8114 YL	Piston cooling oil outlet/cylinder
											1	TE 8123 YH	Main bearing oil outlet temperature/main bearing (S40/35ME-B9 only)
											1	XC 8126 YH	Bearing wear (All except S40/35ME-B9)
1	<input type="checkbox"/>	<input type="checkbox"/>	1		1	<input type="checkbox"/>	1	1	1	1	1	PT 8401 YL	Jacket cooling water inlet
1	1	1	1	1	1	1	1	1	1	1	1	TE 8408 YH	Jacket cooling water outlet, cylinder/cylinder
		<input type="checkbox"/>	<input type="checkbox"/>				<input type="checkbox"/>					TE 8609 YH	Scavenge air receiver
1	1	<input type="checkbox"/>	1	1	1	1	<input type="checkbox"/>	1	1	1	1	TE 8610 YH	Scavenge air box fire-alarm, cylinder/cylinder
		1	1						1			TC 8701 YH	Exhaust gas before turbocharger/turbocharger
1	1		1	<input type="checkbox"/>	1	1	1	1	1	1	1	TC 8702 YH	Exhaust gas after exhaust valve, cylinder/cylinder
			1	<input type="checkbox"/>								TC 8702 YH	Exhaust gas after exhaust valve, cylinder/cylinder, deviation from average
											1	WT 8812 YH	Axial vibration monitor 2)
<input type="checkbox"/>	<input type="checkbox"/>	<input type="checkbox"/>	<input type="checkbox"/>			<input type="checkbox"/>	<input type="checkbox"/>	<input type="checkbox"/>	<input type="checkbox"/>	<input type="checkbox"/>	1	XS 8813 YH	Oil mist in crankcase/cylinder

1 Indicates that the sensor is required.

The sensors in the MAN Diesel column are included for Unattended Machinery Spaces (UMS), option: 4 75 127.

The sensor identification codes and functions are listed in Table 18.07.01.

The tables are liable to change without notice, and are subject to latest class requirements.

2) Required for: K-ME-C6/7 and K98ME6/7 engines with 11 and 14 cylinders.

S-ME-C7/8, S-ME-GI7/8, and L-ME-C7/8 engines with 5 and 6 cylinders.

S-ME-B8/9 engines with 5 and 6 cylinders mainly.

(For K90ME9, K/S-ME-C9, and S50ME-B9 data is available on request).

☐ Select one of the alternatives

* Or shut down

☐ Or alarm for low flow

☐ Or shut down

☐ Or alarm for overheating of main, crank and crosshead bearings, option: 4 75 134.
See also Table 18.04.04: Shut down functions for AMS and UMS

Table 18.04.03: Slow down functions for UMS

Shut down for AMS and UMS – Class and MAN Diesel requirements

ABS	BV	CCS	DNV	GL	KR	LR	NK	RINA	RS	IACS	MAN Diesel	Sensor and function	Point of location
1	1	1	1*	1	1	1	1	1	1	1	1	PS/PT 8109 Z	Lubricating oil inlet to main engine and thrust bearing
1	1	1	1*	1	1	1	1	1	1	1	1	ZT 4020 Z	Engine overspeed
1	1	1			1			1	1	1	1	TE/TS 8107 Z	Thrust bearing segment
				1								PS/PT 8402 Z	Jacket cooling water inlet
			(*)	(1)								XS 8813 Z	Oil mist in crankcase/cylinder

1 Indicates that the sensor is required.

The sensors in the MAN Diesel column are included for Unattended Machinery Spaces (UMS), option: 4 75 127.

The sensor identification codes and functions are listed in Table 18.07.01.

The tables are liable to change without notice, and are subject to latest class requirements.



Or alarm for overheating of main, crank and crosshead bearings, option: 4 75 134.

See also Table 18.04.03: Slow down functions for UMS



Or slow down

International Association of Classification Societies

The members of the International Association of Classification Societies, IACS, have agreed that the stated sensors are their common recommendation, apart from each class' requirements.

The members of IACS are:

ABS	American Bureau of Shipping
BV	Bureau Veritas
CCS	China Classification Society
DNV	Det Norske Veritas
GL	Germanischer Lloyd
KR	Korean Register
LR	Lloyd's Register
NK	Nippon Kaiji Kyokai
RINA	Registro Italiano Navale
RS	Russian Maritime Register of Shipping

and the associated member is:

IRS	Indian Register of Shipping
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Table 18.04.04: Shut down functions for AMS and UMS, option: 4 75 124

Local Instruments

The basic local instrumentation on the engine, options: 4 70 119 comprises thermometers, pressure gauges and other indicators located on the piping or mounted on panels on the engine. The tables 18.05.01a, b and c list those as well as sensors for slow down, alarm and remote indication, option: 4 75 127.

Local instruments	Remote sensors	Point of location
Thermometer, stem type	Temperature element/switch	
		Hydraulic power supply
	TE 1270	HPS bearing temperature (Only K98ME/ME-C with HPS in centre position)
		Fuel oil
TI 8005	TE 8005	Fuel oil, inlet engine
		Lubricating oil
TI 8106	TE 8106	Thrust bearing segment
	TE/TS 8107	Thrust bearing segment
TI 8112	TE 8112	Lubricating oil inlet to main engine
TI 8113	TE 8113	Piston cooling oil outlet/cylinder
TI 8117	TE 8117	Lubricating oil outlet from turbocharger/turbocharger (depends on turbocharger design)
	TE 8123	Main bearing oil outlet temperature/main bearing (S40/35ME-B9 only)
		Cylinder lubricating oil
	TE 8202	Cylinder lubricating oil inlet
	TS 8213	Cylinder lubricating heating
		High temperature cooling water, jacket cooling water
TI 8407	TE 8407	Jacket cooling water inlet
TI 8408	TE 8408	Jacket cooling water outlet, cylinder/cylinder
TI 8409	TE 8409	Jacket cooling water outlet/turbocharger
		Low temperature cooling water, seawater or freshwater for central cooling
TI 8422	TE 8422	Cooling water inlet, air cooler
TI 8423	TE 8423	Cooling water outlet, air cooler/air cooler
		Scavenge air
TI 8605	TE 8605	Scavenge air before air cooler/air cooler
TI 8608	TE 8608	Scavenge air after air cooler/air cooler
TI 8609	TE 8609	Scavenge air receiver
	TE 8610	Scavenge air box – fire alarm, cylinder/cylinder
Thermometer, dial type	Thermo couple	
		Exhaust gas
TI 8701	TC 8701	Exhaust gas before turbocharger/turbocharger
TI 8702	TC 8702	Exhaust gas after exhaust valve, cylinder/cylinder
	TC 8704	Exhaust gas inlet exhaust gas receiver
TI 8707	TC 8707	Exhaust gas outlet turbocharger

Table 18.05.01a: Local thermometers on engine, options 4 70 119, and remote indication sensors, option: 4 75 127

Local instruments	Remote sensors	Point of location
Pressure gauge (manometer)	Pressure transmitter/switch	
PI 8001	PT 8001	Fuel oil Fuel oil, inlet engine
PI 8103	PT 8103	Lubricating oil Lubricating oil inlet to turbocharger/turbocharger
PI 8108	PT 8108	Lubricating oil inlet to main engine
	PS/PT 8109	Lubricating oil inlet to main engine and thrust bearing
	PDS 8140	Lubricating oil differential pressure – cross filter
PI 8401	PT 8401	High temperature jacket cooling water, jacket cooling water Jacket cooling water inlet
	PS/PT 8402	Jacket cooling water inlet (Only Germanischer Lloyd)
	PDS/PDT 8403	Jacket cooling water across engine
	PT 8413	Jacket cooling water outlet, common pipe
PI 8421	PT 8421	Low temperature cooling water, seawater or freshwater for central cooling Cooling water inlet, air cooler
PI 8501	PT 8501	Compressed air Starting air inlet to main starting valve
PI 8503	PT 8503	Control air inlet
	PT 8505	Air inlet to air cylinder for exhaust valve
PI 8601	PT 8601	Scavenge air Scavenge air receiver (PI 8601 instrument same as PI 8706)
PDI 8606	PDT 8606	Pressure drop of air across cooler/air cooler
	PDT 8607	Pressure drop across blower filter of turbocharger (ABB turbochargers only)
PI 8613		Pressure compressor spiral housing/turbocharger
PDI 8614		Pressure drop across compressor spiral housing
PI 8706		Exhaust gas Exhaust gas receiver/Exhaust gas outlet turbocharger
PI 8803		Miscellaneous functions Air inlet for dry cleaning of turbocharger
PI 8804		Water inlet for cleaning of turbocharger

Table 18.05.01b: Local pressure gauges on engine, options: 4 70 119, and remote indication sensors, option: 4 75 127

Local instruments	Remote sensors	Point of location
Other indicators	Other transmitters/ switches	
		Hydraulic power supply
	XC 1231	Automatic main lube oil filter, failure (Boll & Kirch)
	LS 1235	Leakage oil from hydraulic system
	LS 1236	Leakage oil from hydraulic system
		Engine cylinder components
	LS 4112	Leakage from hydraulic cylinder unit
		Fuel oil
	LS 8006	Leakage from high pressure pipes
		Lubricating oil
	FS 8114	Piston cooling oil outlet/cylinder
	XC 8126	Bearing wear (All types except S40/35ME-B9)
	XS 8127	Bearing wear detector failure (All types except S40-35ME-B9)
	XS 8150	Water in lubricating oil
	XS 8151	Water in lubricating oil – too high
	XS 8152	Water in lubricating oil sensor not ready
		Cylinder lube oil
	LS 8208	Level switch
	LS 8212	Small box for heating element, low level
		Scavenge air
	LS 8611	Water mist catcher – water level
		Miscellaneous functions
	ZT 8801 I	Turbocharger speed/turbocharger
WI 8812	WT 8812	Axial vibration monitor (For certain engines only, see note in Table 18.04.04) (WI 8812 instrument is part of the transmitter WT 8812)
	XS 8813	Oil mist in crankcase/cylinder
	XS 8814	Oil mist detector failure
	XC 8816	Shaftline earthing device

Table 18.05.01c: Other indicators on engine, options: 4 70 119, and remote indication sensors, option: 4 75 127

Other Alarm Functions

Drain Box for Fuel Oil Leakage Alarm

Any leakage from the fuel oil high pressure pipes of any cylinder is drained to a common drain box fitted with a level alarm. This is included for both Attended Machinery Space (AMS) and Unattended Machinery Space (UMS).

Bearing Condition Monitoring

Based on our experience we decided in 1990 that all plants, whether constructed for AMS or for UMS, must include an oil mist detector specified by MAN Diesel. Since then an Oil Mist Detector (OMD) and optionally some extent of Bearing Temperature Monitoring (BTM) equipment have made up the warning arrangements for prevention of crankcase explosions on two-stroke engines. Both warning systems are approved by the classification societies.

In order to achieve a response to damage faster than possible with Oil Mist Detection and Bearing Temperature Monitoring alone we introduce Bearing Wear Monitoring (BWM) systems. By monitoring the actual bearing wear continuously, mechanical damage to the crank-train bearings (main-, crank- and crosshead bearings) can be predicted in time to react and avoid damaging the journal and bearing housing.

If the oil supply to a main bearing fails, the bearing temperature will rise and in such a case a Bearing Temperature Monitoring system will trigger an alarm before wear actually takes place. For that reason the ultimate protection against severe bearing damage and the optimum way of providing early warning, is a combined bearing wear and temperature monitoring system.

For all types of error situations detected by the different bearing condition monitoring systems applies that in addition to damaging the components, in extreme cases, a risk of a crankcase explosion exists.

Oil Mist Detector

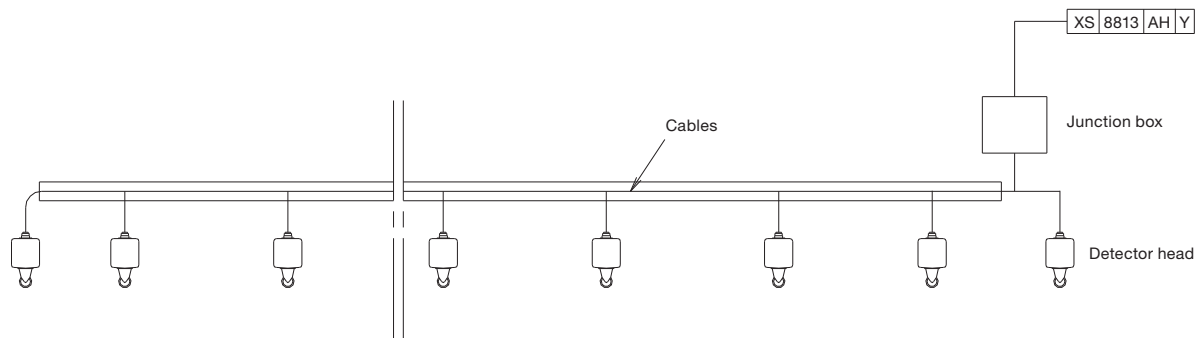
The oil mist detector system constantly measures samples of the atmosphere in the crankcase compartments and registers the results on an optical measuring track, where the opacity (degree of haziness) is compared with the opacity of the atmospheric air. If an increased difference is recorded, a slow down is activated (a shut down in case of Germanischer Lloyd).

Furthermore, for shop trials only MAN Diesel requires that the oil mist detector is connected to the shut down system.

Four alternative oil mist detectors are available:

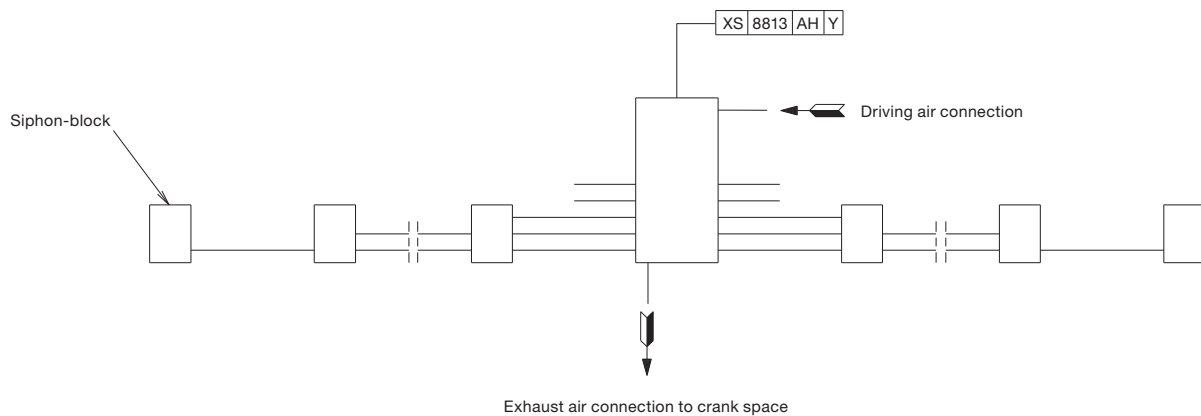
4 75 161	Oil mist detector Graviner MK6. Make: Kidde Fire Protection
4 75 163	Oil mist detector Visatron VN 215/93. Make: Schaller Automation
4 75 165	Oil mist detector QMI. Make: Quality Monitoring Instruments Ltd.
4 75 166	Oil mist detector MD-SX. Make: Daihatsu Diesel Mfg. Co., Ltd.
4 75 167	Oil mist detector Vision III C. Make: Specs Corporation

Diagrams of the two of them are shown for reference in Figs. 18.06.01a and 18.06.01b.



178 49 80-9.3

Fig. 18.06.01a: Oil mist detector pipes on engine, type Graviner MK6 from Kidde Fire Protection (4 75 161)



178 49 81-0.3

Fig. 18.06.01b: Oil mist detector pipes on engine, type Visatron VN215/93 from Schaller Automation (4 75 163)

Bearing Wear Monitoring System

The Bearing Wear Monitoring (BWM) system monitors all three principal crank-train bearings using two proximity sensors forward/aft per cylinder unit and placed inside the frame box.

Targeting the guide shoe bottom ends continuously, the sensors measure the distance to the crosshead in Bottom Dead Center (BDC). Signals are computed and digitally presented to computer hardware, from which a useable and easily interpretable interface is presented to the user.

The measuring precision is more than adequate to obtain an alarm well before steel-to-steel contact in the bearings occur. Also the long-term stability of the measurements has shown to be excellent.

In fact, BWM is expected to provide long-term wear data at better precision and reliability than the manual vertical clearance measurements normally performed by the crew during regular service checks.

For the above reasons, we consider unscheduled open-up inspections of the crank-train bearings to be superfluous, given BWM has been installed.

Two BWM 'high wear' alarm levels including deviation alarm apply. The first level of the high wear / deviation alarm is indicated in the alarm panel only while the second level also activates a slow down.

The Extent of Delivery lists four Bearing Wear Monitoring options of which the two systems from Dr. E. Horn and Kongsberg Maritime could also include Bearing Temperature Monitoring:

4 75 142	Bearing Wear Monitoring System XTS-W. Make: AMOT
4 75 143	Bearing Wear Monitoring System BDMS. Make: Dr. E. Horn
4 75 144	Bearing Wear Monitoring System PS-10. Make: Kongsberg Maritime
4 75 147	Bearing Wear Monitoring System OPEN-predictor. Make: Rovsing Dynamics

ME, ME-C and ME-GI engines are as standard specified with Bearing Wear Monitoring for which any of the above mentioned options could be chosen.

Bearing Temperature Monitoring System

The Bearing Temperature Monitoring (BTM) system continuously monitors the temperature of the bearing. Some systems measure the temperature on the backside of the bearing shell directly, other systems detect it by sampling a small part of the return oil from each bearing in the crankcase.

In case a specified temperature is recorded, either a bearing shell/housing temperature or bearing oil outlet temperature alarm is triggered.

In main bearings, the shell/housing temperature or the oil outlet temperature is monitored depending on how the temperature sensor of the BTM system, option: 4 75 133, is installed.

In crankpin and crosshead bearings, the shell/housing temperature or the oil outlet temperature is monitored depending on which BTM system is installed, options: 4 75 134 or 4 75 135.

For shell/housing temperature in main, crankpin and crosshead bearings two high temperature alarm levels apply. The first level alarm is indicated in the alarm panel while the second level activates a slow down.

For oil outlet temperature in main, crankpin and crosshead bearings two high temperature alarm levels including deviation alarm apply. The first level of the high temperature / deviation alarm is indicated in the alarm panel while the second level activates a slow down.

In the Extent of Delivery, there are three options:

4 75 133	Temperature sensors fitted to main bearings
4 75 134	Temperature sensors fitted to main bearings, crankpin bearings, crosshead bearings and for moment compensator, if any
4 75 135	Temperature sensors fitted to main bearings, crankpin bearings and crosshead bearings

Water In Oil Monitoring System

In case the lubricating oil becomes contaminated with an amount of water exceeding our limit of 0.2%, acute corrosive wear of the crosshead bearing overlayer may occur. The higher the water content, the faster the wear rate.

To prevent water from accumulating in the lube oil and, thereby, causing damage to the bearings, the oil should be monitored manually or automatically by means of a Water In Oil (WIO) monitoring system connected to the engine alarm and monitoring system. In case of water contamination the source should be found and the equipment inspected and repaired accordingly.

The WIO system should trigger an alarm when the water content exceeds 0.2%, and preferably again when exceeding 0.35% measured as absolute water content.

Some WIO systems measure water activity, ie the relative availability of water in a substance expressed in 'aw' on a scale from 0 to 1. Here, '0' indicates oil totally free of water and '1' oil fully saturated by water. The correlation to absolute water content in normal running as well as alarm condition is as follows:

Engine condition	Abs. water content, %	Water activity, aw
High alarm level	0.2	0.5
High High alarm level	0.35	0.9

ME, ME-C and ME-GI engines are as standard specified with Water In Oil monitoring system.

Please note: Corrosion of the overlayer is a potential problem only for crosshead bearings, because only crosshead bearings are designed with an overlayer. Main and crankpin bearings may also suffer irreparable damage from water contamination, but the damage mechanism would be different and not as acute.

Liner Wall Monitoring System

The Liner Wall Monitoring (LWM) system monitors the temperature of each cylinder liner. It is to be regarded as a tool providing the engine room crew the possibility to react with appropriate countermeasures in case the cylinder oil film is indicating early signs of breakdown.

In doing so, the LWM system can assist the crew in the recognition phase and help avoid consequential scuffing of the cylinder liner and piston rings.

Signs of oil film breakdown in a cylinder liner will appear by way of increased and fluctuating temperatures. Therefore, recording a preset max allowable absolute temperature for the individual cylinder or a max allowed deviation from a calculated average of all sensors will trigger a cylinder liner temperature alarm.

The LWM system includes two sensors placed in the manoeuvring and exhaust side of the liners, near the piston skirt TDC position. The sensors are interfaced to the ship alarm system which monitors the liner temperatures.

For each individual engine, the max and deviation alarm levels are optimised by monitoring the temperature level of each sensor during normal service operation and setting the levels accordingly.

The temperature data is logged on a PC for one week at least and preferably for the duration of a round trip for reference of temperature development.

All types 98 and 90 ME and ME-C engines as well as K80ME-C9 are as standard specified with Liner Wall Monitoring system. For all other engines, the LWM system is available as an option: 4 75 136.

Control Devices

The control devices mainly include a position switch (ZS) or a position transmitter (ZT) and solenoid valves (ZV) which are listed in Table 18.06.02 below. The sensor identification codes are listed in Table 18.07.01.

Sensor	Point of location
Manoeuvring system	
ZS 1109-A/B C	Turning gear – disengaged
ZS 1110-A/B C	Turning gear – engaged
ZS 1111-A/B C	Main starting valve – blocked
ZS 1112-A/B C	Main starting valve – in service
ZV 1114 C	Slow turning valve
ZS 1116-A/B C	Start air distribution system – in service
ZS 1117-A/B C	Start air distribution system – blocked
ZV 1120 C	Activate pilot press air to starting valves
ZS 1121-A/B C	Activate main starting valves - open
E 1180	Electric motor, auxiliary blower
E 1181	Electric motor, turning gear
E 1185 C	LOP, Local Operator Panel
Hydraulic power supply	
PT 1201-1/2/3 C	Hydraulic oil pressure, after non-return valve
ZV 1202-A/B C	Force-driven pump by-pass
PS/PT 1204-1/2/3 C	Lubricating oil pressure after filter, suction side
Tacho/crankshaft position	
ZT 4020	Tacho for safety
Engine cylinder components	
XC 4108 C	ELVA NC valve
ZT 4111 C	Exhaust valve position
ZT 4114 C	Fuel plunger, position 1
Fuel oil	
ZV 8020 Z	Fuel oil cut-off at engine inlet (shut down), Germanischer Lloyd only
Cylinder lubricating oil	
ZT 8203 C	Confirm cylinder lubricator piston movement, cyl/cyl
ZV 8204 C	Activate cylinder lubricator, cyl/cyl
Scavenge air	
PS 8603 C	Scavenge air receiver, auxiliary blower control

Table 18.06.02: Control devices on engine

Identification of Instruments

The instruments and sensors are identified by a position number which is made up of a combination of letters and an identification number:

Measured variables

First letters:

DS	Density switch
DT	Density transmitter
FT	Flow transmitter
FS	Flow switch
GT	Gauging transmitter (Index, load)
LI	Level indication, local
LS	Level switch
LT	Level transmitter
PDI	Pressure difference indication, local
PDS	Pressure difference switch
PDT	Pressure difference transmitter
PI	Pressure indication, local
PS	Pressure switch
PT	Pressure transmitter
ST	Speed transmitter
TC	Thermo couple (NiCr-Ni)
TE	Temperature element (Pt 100)
TI	Temperature indication, local
TS	Temperature switch
VS	Viscosity switch
VT	Viscosity transmitter
WI	Vibration indication, local
WS	Vibration switch
WT	Vibration transmitter
XC	Unclassified control
XS	Unclassified switch
XT	Unclassified transmitter
ZS	Position switch
ZT	Position transmitter (proximity switch)
ZV	Position valve (solenoid valve)

Location of measuring point

Ident. number:

11xx	Manoeuvring system
12xx	Hydraulic power supply system
14xx	Combustion pressure supervision
20xx	ECS to/from safety system
21xx	ECS to/from remote control system
22xx	ECS to/from alarm system
30xx	ECS miscellaneous input/output
40xx	Tacho/crankshaft position system
41xx	Engine cylinder components
50xx	VOC, supply system
51xx	VOC, sealing oil system
52xx	VOC, control oil system
53xx	VOC, other related systems

54xx	VOC, engine related components
80xx	Fuel oil system
81xx	Lubricating oil system
82xx	Cylinder lube oil system
83xx	Stuffing box drain system
84xx	Cooling water systems
85xx	Compressed air systems
86xx	Scavenge air system
87xx	Exhaust gas system
88xx	Miscellaneous functions
90xx	Project specific functions

xxxx-A	Alternative redundant sensors
xxxx-1	Cylinder/turbocharger numbers

ECS: Engine Control System
VOC: Volatile Organic Compound

Functions

Secondary letters:

A	Alarm
AH	Alarm, high
AL	Alarm, low
C	Control
H	High
I	Indication
L	Low
R	Recording
S	Switching
X	Unclassified function
Y	Slow down
Z	Shut down

Repeated signals

Signals which are repeated for example for each cylinder or turbocharger are provided with a suffix number indicating the location, '1' for cylinder 1, etc.

If redundant sensors are applied for the same measuring point, the suffix is a letter: A, B, C, etc.

Examples:

TI 8005 indicates a local temperature indication (thermometer) in the fuel oil system.

ZS 1112-A C and **ZS 1112-B C** indicate that there are two position switches in the manoeuvring system, A and B for control of the main starting air valve position.

PT 8501 I I AL Y indicates a pressure transmitter located in the control air supply for remote indication, alarm for low pressure and slow down for low pressure.

Table 18.07.01: Identification of instruments

Dispatch Pattern, Testing, Spares and Tools

19

Dispatch Pattern, Testing, Spares and Tools

Painting of Main Engine

The painting specification, Section 19.02, indicates the minimum requirements regarding the quality and the dry film thickness of the coats of, as well as the standard colours applied on MAN B&W engines built in accordance with the 'Copenhagen' standard.

Paints according to builder's standard may be used provided they at least fulfil the requirements stated.

Dispatch Pattern

The dispatch patterns are divided into two classes, see Section 19.03:

- A: Short distance transportation and short term storage
- B: Overseas or long distance transportation or long term storage.

Short distance transportation (A) is limited by a duration of a few days from delivery ex works until installation, or a distance of approximately 1,000 km and short term storage.

The duration from engine delivery until installation must not exceed 8 weeks.

Dismantling of the engine is limited as much as possible.

Overseas or long distance transportation or long term storage require a class B dispatch pattern.

The duration from engine delivery until installation is assumed to be between 8 weeks and maximum 6 months.

Dismantling is effected to a certain degree with the aim of reducing the transportation volume of the individual units to a suitable extent.

Note:

Long term preservation and seaworthy packing are always to be used for class B.

Furthermore, the dispatch patterns are divided into several degrees of dismantling in which '1' comprises the complete or almost complete engine. Other degrees of dismantling can be agreed upon in each case.

When determining the degree of dismantling, consideration should be given to the lifting capacities and number of crane hooks available at the engine maker and, in particular, at the yard (purchaser).

The approximate masses of the sections appear in Section 19.04. The masses can vary up to 10% depending on the design and options chosen.

Lifting tools and lifting instructions are required for all levels of dispatch pattern. The lifting tools, options: 4 12 110 or 4 12 111, are to be specified when ordering and it should be agreed whether the tools are to be returned to the engine maker, option: 4 12 120, or not, option: 4 12 121.

MAN Diesel's recommendations for preservation of disassembled / assembled engines are available on request.

Furthermore, it must be considered whether a drying machine, option: 4 12 601, is to be installed during the transportation and/or storage period.

Shop trials/Delivery Test

Before leaving the engine maker's works, the engine is to be carefully tested on diesel oil in the presence of representatives of the yard, the shipowner and the classification society.

The shop trial test is to be carried out in accordance with the requirements of the relevant classification society, however a minimum as stated in Section 19.05.

MAN Diesel's recommendations for shop trial, quay trial and sea trial are available on request.

In connection with the shop trial test, it is required to perform a pre-certification survey on engine plants with FPP or CPP, options: 4 06 060a Engine test cycle E3 or 4 06 060b Engine test cycle E2 respectively.

Spare Parts

List of spare parts, unrestricted service

The tendency today is for the classification societies to change their rules such that required spare parts are changed into recommended spare parts.

MAN Diesel, however, has decided to keep a set of spare parts included in the basic extent of delivery, EoD: 4 87 601, covering the requirements and recommendations of the major classification societies, see Section 19.06.

This amount is to be considered as minimum safety stock for emergency situations.

Additional spare parts recommended by MAN Diesel

The above-mentioned set of spare parts can be extended with the 'Additional Spare Parts Recommended by MAN Diesel', option: 4 87 603, which facilitates maintenance because, in that case, all the components such as gaskets, sealings, etc. required for an overhaul will be readily available, see Section 19.07.

Wearing parts

The consumable spare parts for a certain period are not included in the above mentioned sets, but can be ordered for the first 1, 2, up to 10 years' service of a new engine, option: 4 87 629, a service year being assumed to be 6,000 running hours.

The wearing parts that, based on our service experience, are estimated to be required, are divided into groups and listed with service hours in Tables 19.08.01 and 19.08.02.

Large spare parts, dimensions and masses

The approximate dimensions and masses of the larger spare parts are indicated in Section 19.09. A complete list will be delivered by the engine maker.

Tools

List of standard tools

The engine is delivered with the necessary special tools for overhauling purposes. The extent, dimensions and masses of the main tools is stated in Section 19.10. A complete list will be delivered by the engine maker.

Tool Panels

Most of the tools are arranged on steel plate panels, EoD: 4 88 660, see Section 19.11 'Tool Panels'.

It is recommended to place the panels close to the location where the overhaul is to be carried out.

Specification for painting of main engine

Components to be painted before shipment from workshop	Type of paint	No. of coats/ Total dry film thickness μm	Colour: RAL 840HR DIN 6164 MUNSELL
Component/surfaces, inside engine, exposed to oil and air 1. Unmachined surfaces all over. However cast type crankthrows, main bearing cap, crosshead bearing cap, crankpin bearing cap, pipes inside crankcase and chainwheel need not to be painted but the cast surface must be cleaned of sand and scales and kept free of rust.	Engine alkyd primer, weather resistant Oil and acid resistant alkyd paint. Temperature resistant to minimum 80 °C.	2/80 1/30	Free White: RAL 9010 DIN N:0:0.5 MUNSELL N-9.5
Components, outside engine 2. Engine body, pipes, gallery, brackets etc. Delivery standard is in a primed and finally painted condition, unless otherwise stated in the contract.	Engine alkyd primer, weather resistant. Final alkyd paint resistant to salt water and oil, option: 4 81 103.	2/80 1/30	Free Light green: RAL 6019 DIN 23:2:2 MUNSELL 10GY 8/4
Heat affected components: 3. Supports for exhaust receiver Scavenge air-pipe outside. Air cooler housing inside and outside.	Paint, heat resistant to minimum 200 °C.	2/60	Alu: RAL 9006 DIN N:0:2 MUNSELL N-7.5
Components affected by water and cleaning agents 4. Scavenge air cooler box inside.	protection of the components exposed to moderately to severely corrosive environment and abrasion.	2/75	Free
5. Gallery plates topside.	Engine alkyd primer, weather resistant.	2/80	Free
6. Purchased equipment and instruments painted in makers colour are acceptable unless otherwise stated in the contract.			
Tools Unmachined surfaces all over on handtools and lifting tools. Purchased equipment painted in makers colour is acceptable, unless otherwise stated in the contract/drawing.	Oil resistant paint.	2/60	Orange red: RAL 2004 DIN:6:7:2 MUNSELL N-7.5r 6/12
Tool panels	Oil resistant paint.	2/60	Light grey: RAL 7038 DIN:24:1:2 MUNSELL N-7.5

Note: All paints are to be of good quality. Paints according to builder's standard may be used provided they at least fulfil the above requirements.

The data stated are only to be considered as guidelines. Preparation, number of coats, film thickness per coat, etc. have to be in accordance with the paint manufacturer's specifications.

178 30 20-7.4

Fig. 19.02.01: Painting of main engine: option 4 81 101, 4 81 102 or 4 81 103

Dispatch Pattern

The relevant engine supplier is responsible for the actual execution and delivery extent. As differences may appear in the individual suppliers' extent and dispatch variants.

Class A (option 4 12 020):

Short distance transportation limited by duration of transportation time within a few days or a distance of approximately 1000 km and short term storage.

Duration from engine delivery to installation must not exceed eight weeks.

Dismantling must be limited.

Class B (option 4 12 030):

Overseas and other long distance transportation, as well as long-term storage.

Dismantling is effected to reduce the transport volume to a suitable extent.

Long-term preservation and seaworthy packing must always be used.

Classes A + B comprise the following basic variants:

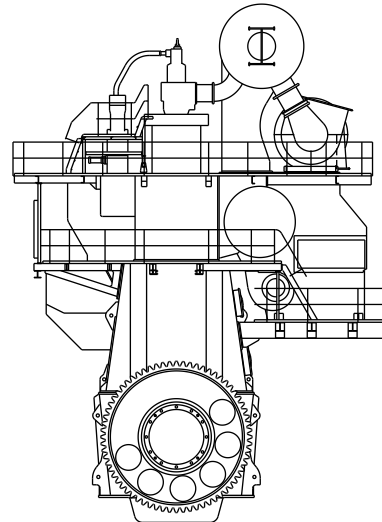
A1 + B1 (option 4 12 021 + 4 12 031)

Engine complete, i.e. not disassembled

A2 + B2 (option 4 12 022 + 4 12 032)

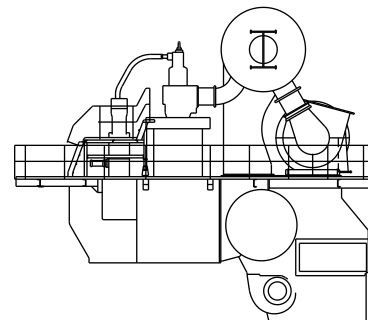
- Top section including cylinder frame complete, cylinder covers complete, scavenge air receiver including cooler box and cooler insert, turbocharger(s), piston complete and galleries with pipes, HCU units and oil filter
- Bottom section including bedplate complete, frame box complete, connecting rods, turning gear, crankshaft complete and galleries
- Remaining parts including stay bolts, chains, FIVA valves etc.

A1 + B1

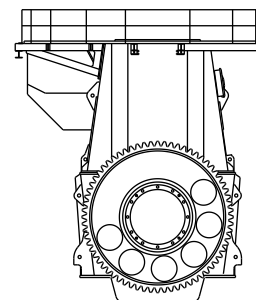


Engine complete

A2 + B2



Top section



Bottom section

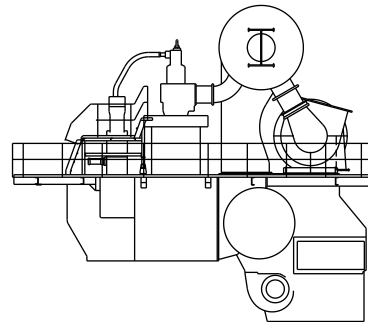
074 27 27-7.0.0a

Fig. 19.03.01: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

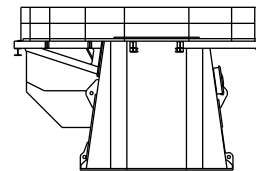
A3 + B3 (option 4 12 023 + 4 12 033)

- Top section including cylinder frame complete, cylinder covers complete, scavenge air receiver including cooler box and cooler insert, turbocharger(s), piston complete and galleries with pipes, HCU Units
- Frame box section including frame box complete, chain drive, connecting rods and galleries, gearbox for hydraulic power supply, hydraulic pump station and oil filter
- Bedplate/crankshaft section including bedplate complete, crankshaft complete with chain-wheels and turning gear
- Remaining parts including stay bolts, chains FIVA valves, etc.

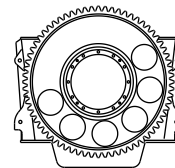
A3 + B3



Top section



Frame box section



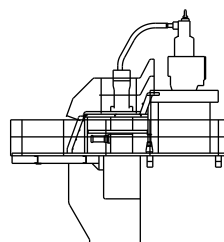
Bedplate/crankshaft section

074 27 27-7.0.0b

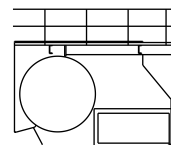
Fig. 19.03.02: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

A4 + B4 (option 4 12 024 + 4 12 034)

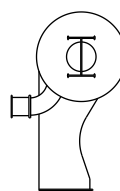
- Top section including cylinder frame complete, cylinder covers complete, piston complete and galleries with pipes on manoeuvre side, HCU units
- Exhaust receiver with pipes
- Scavenge air receiver with galleries and pipes
- Turbocharger
- Air cooler box with cooler insert
- Frame box section including frame box complete, chain drive, connecting rods and galleries, gearbox for hydraulic power supply, hydraulic power station and oil filter
- Crankshaft with chain wheels
- Bedplate with pipes and turning gear
- Remaining parts including stay bolts, auxiliary blowers, chains FIVA valves etc.



Top section



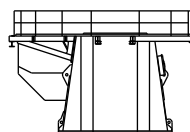
Scavenge air receiver



Exhaust receiver



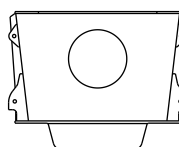
Turbocharger



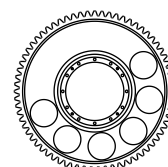
Frame box section



Air cooler box



Bedplate section



Crankshaft section

Note

The engine supplier is responsible for the necessary lifting tools and lifting instructions for transportation purposes to the yard. The delivery extent of lifting tools, ownership and lend/lease conditions are to be stated in the contract. (Options: 4 12 120 or 4 12 121)

Furthermore, it must be stated whether a drying machine is to be installed during the transportation and/or storage period. (Option: 4 12 601)

074 27 27-7.0.0c

Fig. 19.03.03: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

Dispatch Pattern, List of Masses and Dimensions

This section is available on request

Shop Test

Minimum delivery test

The minimum delivery test, EoD: 4 14 001, involves:

- Starting and manoeuvring test at no load
- Load test
Engine to be started and run up to 50% of Specified MCR (M) in 1 hour

Followed by:

- 0.50 hour running at 25% of specified MCR
- 0.50 hour running at 50% of specified MCR
- 0.50 hour running at 75% of specified MCR
- 1.00 hour running at 100% of specified MCR
- 0.50 hour running at 110% of specified MCR

Only for Germanischer Lloyd:

- 0.75 hour running at 110% of specified MCR

Governor tests, etc:

- Governor test
- Minimum speed test
- Overspeed test
- Shut down test
- Starting and reversing test
- Turning gear blocking device test
- Start, stop and reversing from the Local Operating Panel (LOP)

Before leaving the factory, the engine is to be carefully tested on diesel oil in the presence of representatives of Yard, Shipowner, Classification Society, and MAN Diesel.

At each load change, all temperature and pressure levels etc. should stabilise before taking new engine load readings.

Fuel oil analysis is to be presented.

All tests are to be carried out on diesel or gas oil.

EIAPP certificate

All marine engines are required by IMO to have an 'Engine International Air Pollution Prevention' (EIAPP) Certificate. Therefore, a pre-certification survey is to be carried out for all engines according to the performance parameters recorded in the engine's Unified Technical File (UTF), which is prepared by MAN Diesel.

The EIAPP certificate documents that the specific engine meets the international NO_x emission limitations specified in Regulation 13 of MARPOL Annex VI. The basic engine 'Economy running mode', EoD: 4 06 060, complies with these limitations.

The pre-certification survey for a 'Parent' or an 'Individual' engine includes NO_x measurements during the delivery test. For 'Member' engines, a parameter check according to the UTF for the engine group, based on the delivery test, is needed.

The tests, if required, are:

- E3, marine engine, propeller law for FPP, option: 4 06 060a
- or
- E2, marine engine, constant speed for CPP, option: 4 06 060b.

For further information and options regarding shop test, see Extent of Delivery.

Fig. 9.05.01: Shop trial running/delivery test: 4 14 001

List of Spare Parts, Unrestricted Service

Spare parts are **requested** by the following Classes only: GL, KR, NK and RS, while just **recommended** by: ABS, DNV and LR, but neither requested nor recommended by: BV, CCS and RINA.

Cylinder cover, plate 901 and others

- 1 Cylinder cover with fuel, exhaust and starting valves, indicator valve and sealing rings (disassembled)
- ½ set Studs for 1 cylinder cover

Piston, plate 902

- 1 Piston complete (with cooling pipe), piston rod, piston rings and stuffing box, studs and nuts
- 1 set Piston rings for 1 cylinder

Cylinder liner, plate 903

- 1 Cylinder liner inclusive of sealing rings and gaskets.

Cylinder lubricating oil system, plate 903 ¹⁾

- 1 set Spares for lubricating oil system for 1 cyl.
- 2 Lubricator backup cable

Connecting rod, and crosshead bearing, plate 904

- 1 Telescopic pipe with bushing for 1 cylinder
- 1 Crankpin bearing shells in 2/2 with studs and nuts
- 1 Crosshead bearing shell lower part with studs and nuts
- 2 Thrust pieces

Thrust block, plate 905

- 1 set Thrust pads for 'ahead'
- For NK also one set 'astern' if different from 'ahead'

HPS - Hydraulic Power Supply, plate 906 ^{1 and 2)}

- 1 Proportional valve for hydraulic pumps
- 1 Leak indicator
- 1 Safety coupling for hydraulic pump
- 1 Accumulator
- 6 Chain links. Only for ABS, LR and NK
- 1 set Flex pipes, one of each size
- 1 Electric motor

Engine control system, plate 906 ²⁾

- 1 Multi Purpose Controller
- 1 Amplifier for Auxiliary Control Unit
- 1 Position Amplifier
- 1 Trigger sensor for tacho system, only if trigger ring
- 1 Marker sensor for tacho system
- 1 Tacho signal amplifier
- 1 ID-key

- 1 Encoder

- 1 Fuse kit

Starting valve, plate 907

- 1 Starting valve, complete
- 1 Solenoid valve ²⁾

Hydraulic cylinder unit, plate 907 ^{1 and 2)}

- 1 Fuel booster barrel, complete with plunger
- 1 FIVA valve complete
- 1 Suction valve complete
- 1 set Flex pipes, one of each size
- 1 High-pressure pipe kit
- 1 Packing kit

Exhaust valve, plate 908

- 2 Exhaust valves complete. 1 only for GL
- 1 High-pressure pipe from actuator to exhaust valve
- 1 Exhaust valve position sensor

Fuel valve, plate 909

- 1 set Fuel valves for all cylinders on one engine for BV, CCS, DNV, GL, KR, NK, RINA, RS and IACS
- 1 set Fuel valves for half the number of cylinders on the engine for ABS
- 1 High-pressure pipe, from fuel oil pressure booster to fuel valve

Turbocharger, plate 910

- 1 Set of maker's standard spare parts
- 1 a) Spare rotor for one turbocharger, including compressor wheel, rotor shaft with turbine blades and partition wall, if any

Scavenge air blower, plate 910

- 1 set Rotor, rotor shaft, gear wheel or equivalent working parts
- 1 set Bearings for electric motor
- 1 set Bearing for blower wheel
- 1 Belt, if applied
- 1 set Packing for blower wheel

Bedplate, plate 912

- 1 Main bearing shell in 2/2 of each size
- 1 set Studs and nuts for 1 main bearing

¹⁾ MD required spare parts.

²⁾ All spare parts are requested by all Classes.

a) Only required for RS. To be ordered separately as option: 4 87 660 for other classification societies.

Please note: Plate number refers to Instruction Book, Vol. III containing plates with spare parts

Fig. 19.06.01: List of spare parts, unrestricted service: 4 87 601

Additional Spares

Beyond class requirements or recommendation, for easier maintenance and increased security in operation.

Cylinder cover, section 90101

- 4 Studs for exhaust valve
- 4 Nuts for exhaust valve
- ½ set O-rings for cooling jacket
- 1 Cooling jacket
- ½ set Sealing between cylinder cover and liner
- 4 Spring housings for fuel valve

Hydraulic tool for cylinder cover, section 90161

- 1 set Hydraulic hoses with protection hose complete with couplings
- 8 pcs O-rings with backup rings, upper
- 8 pcs O-rings with backup rings, lower

Piston and piston rod, section 90201

- 1 box Locking wire, L=63 m
- 5 Piston rings of each kind
- 2 D-rings for piston skirt
- 2 D-rings for piston rod

Piston rod stuffing box, section 90205

- 15 Self-locking nuts
- 5 O-rings
- 5 Top scraper rings
- 15 Pack sealing rings
- 10 Cover sealing rings
- 120 Lamellas for scraper rings
- 30 Springs for top scraper and sealing rings
- 20 Springs for scraper rings

Cylinder frame, section 90301

- ½ set Studs for cylinder cover for one cyl.
- 1 Bushing

Cylinder liner and cooling jacket, section 90302

- 1 Cooling jacket of each kind
- 4 Non return valves
- 1 set O-rings for one cylinder liner
- ½ set Gaskets for cooling water connection
- ½ set O-rings for cooling water pipes
- 1 set Cooling water pipes between liner and cover for one cylinder

Cylinder Lubricating Oil System, section 90306

- 1 set Spares for MAN B&W Alpha lubricating oil system for 1cyl.
- 1 Lubricator
- 2 Feed back sensor, complete
- 1 Complete sets of O-rings for lubricator (depending on number of lubricating nozzles per cylinder)

Connecting rod and crosshead, section 90401

- 1 Telescopic pipe
- 2 Thrust piece

HPS Hydraulic Power Supply, section 906

- 1 Delivery pump
- 1 Start up pump
- 1 Pressure relief valve
- 1 Pumps short cutting valve
- 1 set Check valve Cartridge (3 pcs)

Engine Control System, section 906

- 1 set Fuses for MPC, TSA, CNR
- 1 Segment for triggerring

HCU Hydraulic Cylinder Unit, section 906

- 1 set Packings

Main starting valve, section 90702

- 1 Repair kit for main actuator
- 1 Repair kit for main ball valve
- 1 *) Repair kit for actuator, slow turning
- 1 *) Repair kit for ball valve, slow turning

*) if fitted

Starting valve, section 90704

- 2 Locking plates
- 2 Piston
- 2 Spring
- 2 Bushing
- 1 set O-ring
- 1 Valve spindle

Fig. 19.07.01a: Additional spare parts beyond class requirements or recommendation, option: 4 87 603

Exhaust valve, section 90801

- 1 Exhaust valve spindle
- 1 Exhaust valve seat
- 1/2 set O-ring exhaust valve/cylinder cover
- 4 Piston rings
- 1/2 set Guide rings
- 1/2 set Sealing rings
- 1/2 set Safety valves
- 1 set Gaskets and O-rings for safety valve
- 1 Piston complete
- 1 Damper piston
- 1 set O-rings and sealings between air piston and exhaust valve housing/spindle
- 1 Liner for spindle guide
- 1 set Gaskets and O-rings for cooling water connection
- 1 Conical ring in 2/2
- 1 set O-rings for spindle/air piston
- 1 set Non-return valve

Exhaust valve, section 90802

- 1 Sealing oil control unit

Exhaust valve actuator, section 90805

- 1 Hydraulic exhaust valve actuator complete for 1 cylinder
- 1 Electronic exhaust valve control valve

Cooling water outlet, section 90810

- 2 Ball valve
- 1 Butterfly valve
- 1 Compensator
- 1 set Gaskets for butterfly valve and compensator

Fuel injection system, section 90901

- 1 Fuel oil pressure booster complete, for 1 cyl.
- 1 Hydraulic cylinder unit
- 1 set Gaskets and sealings
- 1 Electronic fuel injection control valve

Fuel valve, section 90910

- 1 set Fuel nozzles
- 1 set O-rings for fuel valve
- 3 Spindle guides, complete
- 1/2 set Springs
- 1/2 set Discs, +30 bar
- 3 Thrust spindles
- 3 Non return valve (if mounted)

Fuel oil high pressure pipes, section 90913

- 1 High pressure pipe, from fuel oil pressure booster to fuel valve
- 1 High pressure pipe from actuator to exhaust valve
- 1 set O-rings for high pressure pipes

Overflow valve, section 90915

- 1 Overflow valve, complete
- 1 O-rings of each kind

Turbocharger, section 91000

- 1 Spare rotor, complete with bearings
- 1 Spare part set for turbocharger

Scavenge air receiver, section 91001

- 2 Non-return valves complete
- 1 Compensator

Exhaust pipes and receiver, section 91003

- 1 Compensator between TC and receiver
- 2 Compensator between exhaust valve and receiver
- 1 set Gaskets for each compensator

Air cooler, section 91005

- 16 Iron blocks (Corrosion blocks)

Safety valve, section 91101

- 1 set Gasket for safety valve
- 2 Safety valve, complete

Arrangement of safety cap, section 91104

- 1 set Bursting disc

Engine Lubricating System, section 912

- 1 set 6 μ filter

Note: Section numbers refer to Instruction Book, Vol. III containing plates with spare parts

Fig. 19.07.01b: Additional spare parts beyond class requirements or recommendation, option: 4 87 603

Wearing parts

The wearing parts are divided into 20 groups, each including the components stated in **Table A**.

The average expected consumption of spare parts is stated in **Table B** for 1, 2, 3... 10 years' service of a new engine, a service year being assumed to be of 6000 hours.

In order to find the expected consumption of spare parts:

Multiply the quantity stated in Table A with the factor in Table B for a given number of service hours.

Table A:

Group No.	Section	Quantity	Descriptions	
1	90101	½	set	O-rings and gaskets for 1 cylinder
2		¼	set	Spring housing, complete for 1 cylinder
	90103	¼	set	Indicator valves, O-rings and gaskets for 1 cylinder
3	90161	½	set	O-ring W / Back-up ring for 1 cylinder
4		½	set	Hose with union for 1 cylinder
5	90201	1	box	Locking wire 1,0MM L=63
		1	set	Piston rings for 1 cylinder
		1	set	O-rings for 1 cylinder
6	90205	1	set	O-rings for 1 cylinder
		1	set	Lamella rings 3/3 for 1 cylinder
		½	set	Top scraper rings 4/4 for 1 cylinder
		½	set	Pack Sealing rings 4/4 for 1 cylinder
		½	set	Cover Sealing rings 4/4 for 1 cylinder
		½	set	Springs of each kind for 1 cylinder
7	90302	½	set	O-rings / Sealing rings for Cylinder liner
		1	set	O-rings, Packings and Gaskets for cooling water connections
8		1	pcs	Cylinder liner
		1	pcs	Piston cleaning ring (if Mounted)
10	90635-45	1	set	Packings and Gaskets for 1 Engine
12	90702	½	set	Repair Kit for each type of valve for 1 Engine
13	90704	1	set	O-rings, Packings and Gaskets for 1 Engine
14	90801	¼	set	Exhaust valve spindle for 1 Engine
		¼	set	Exhaust valve W-bottom piece for 1 Engine
15		1	set	Piston rings for exhaust valve air piston and oil piston for 1 Engine
		1	set	O-rings for water connections for 1 Engine
		1	set	Gasket for cooling for water connections for 1 Engine
		1	set	O-rings for oil connections for 1 Engine
		1	pcs	Spindle guide
		2	pcs	Air sealing ring
		½	set	Guide sealing rings
		1	set	O-rings for bottom piece for 1 Engine
17	90910	½	set	Fuel valve nozzle for 1 cylinder
		¼	set	Spindle guide complete and non-return valve for 1 cylinder
		2	set	O-rings for 1 cylinder
18	90917	¼	set	Plunger and housing for fuel oil booster for 1 Engine
		½	set	Suction valve complete for 1 Cylinder
		1	set	Sealing rings, O-rings and Gaskets for 1 cylinder
19	91000	1		Slide bearing for turbocharger for 1 engine (roller bearings)
		1		Guide bearing for turbocharger for 1 engine (roller bearings)
20	91000	1		Slide bearing for turbocharger for 1 engine (slide bearings)
		1		Guide bearing for turbocharger for 1 engine (slide bearings)

Note: Section numbers refers to Instruction Book, Vol. III containing plates with spare parts

Fig. 19.08.01: Table A

Table B:

		Service hours:	0-	0-	0-	0-	0-	0-	0-	0-	0-	0-
			6000	12000	18000	24000	3000	36000	42000	48000	54000	60000
Group.	Section											
No.	No.	Description	Factor for number of cylinders									
1	90101	O-rings and gaskets	1	2	3	4	5	6	7	8	9	10
2		Spring housing	0	1	1	1	2	1	1	1	1	1
	90103	Packing and Gaskets	1	2	3	4	5	6	7	8	9	10
3	90161	O-ring W / Back-up ring	1	2	3	4	5	6	7	8	9	10
4		Hose with union	0	0	1	1	1	2	1	2	1	2
5	90201	Set of piston rings	0	1	1	2	3	4	3	4	4	4
6	90205	St. box, lamella / sealing rings	0	1	1	2	2	3	3	4	3	4
7	90302	O-rings / Sealing rings Cyl. liner	0	1	1	2	1	2	2	4	1	2
8		Cylinder liners	0	0	0	0	0	0	0	0	0	0
9	90610	Bearing Shells and Guide Disc	0	0	0	1	1	2	1	2	1	2
10	90635-45	Packings and Gaskets	1	2	3	4	5	6	7	8	7	8
12	90702	Repair Kit for each type of valve	0	1	1	2	3	4	3	4	3	4
13	90704	O-rings, Packings and Gaskets	1	2	3	4	5	6	7	8	9	10
14	90801	Exhaust valve spindles / bottom pieces	0	0	1	1	1	2	1	2	1	2
15		Exhaust valve guide bushings	0	1	1	2	2	4	2	4	2	4
		O-rings for exhaust valve	1	2	3	4	5	6	7	8	9	10
17	90910	Fuel valve guides and nozzles	0	1	1	2	4	4	5	5	3	3
18	90917	Plunger and housing for fuel oil booster	0	0	0	0	0	1	1	1	1	1
		Suction/puncture valves, Sealing rings										
		and Gaskets	0	1	1	2	2	3	3	4	3	3
19	91000	Set bearings per TC (roller bearings) *)	0	0	1 set	2 set	2 set	3 set	3 set	4 set	4 set	5 set
20	91000	Set bearings per TC (slide bearings) *)	0	0	0	1 set	1 set	1 set	1 set	2 set	2 set	2 set

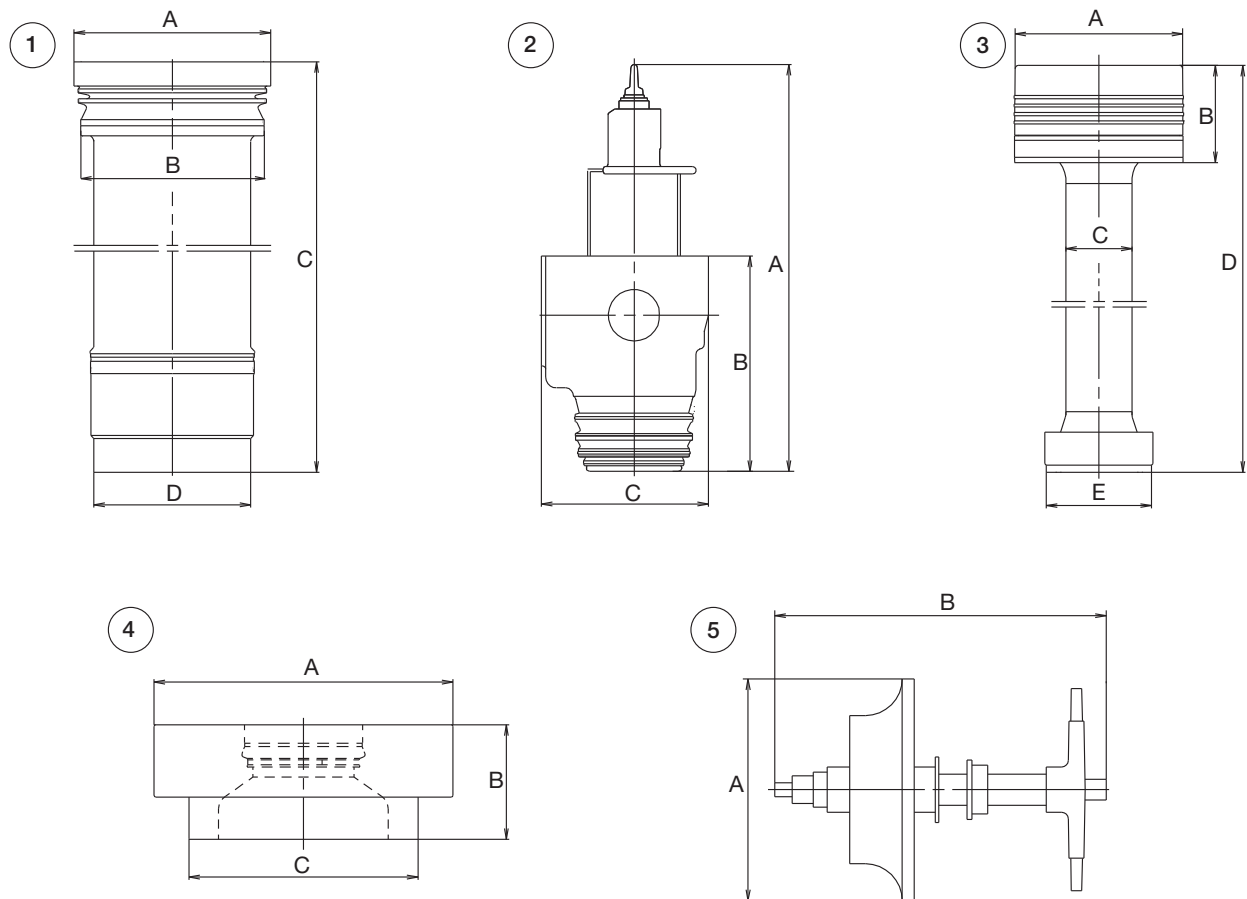
*) Not depending on number of cylinders.

Note:

Section numbers refers to Instruction Book, Vol. III containing plates with spare parts

Fig. 19.08.02: Table B

Large spare parts, dimensions and masses



178 51 59-7.0

Pos	Sec. Description	Mass	Dimensions (mm)				
		(kg)	A	B	C	D	E
1	Cylinder liner, incl. cooling jacket	7,900	ø1,270	ø1,210	3,270	ø1,044	
2	Exhaust valve	2,372	2,271	1,155	808		
3	Piston complete, with piston rod	4,400	ø900	625	ø350	4,044	566
4	Cylinder cover, incl. valves	5,690	ø1,630	725	ø1,220		
5	Rotor for turbocharger, TCA 77-21	360	ø750	1,360			
5	Rotor for turbocharger, TCA 88-21	610	ø890	1,630			
5	Rotor for turbocharger, TCA 88-25	750	ø940	1,630			
5	Rotor for turbocharger, A185-L34/L35	Available on request					
5	Rotor for turbocharger, A190-L34/L35						
5	Rotor for turbocharger, MET83MA	600	ø924	1,555			
5	Rotor for turbocharger, MET90MA	850	ø1,020	1,723			

Fig. 19.09.01: Large spare parts, dimensions and masses

List of Standard Tools for Maintenance

This section is available on request

Tool Panels

This section is available on request

Project Support and Documentation

20

Project Support and Documentation

The selection of the ideal propulsion plant for a specific newbuilding is a comprehensive task. However, as this selection is a key factor for the profitability of the ship, it is of the utmost importance for the end-user that the right choice is made.

MAN Diesel is able to provide a wide variety of support for the shipping and shipbuilding industries all over the world.

The knowledge accumulated over many decades by MAN Diesel covering such fields as the selection of the best propulsion machinery, optimisation of the engine installation, choice and suitability of a Power Take Off for a specific project, vibration aspects, environmental control etc., is available to shipowners, shipbuilders and ship designers alike.

Part of this information can be found in the following documentation:

- Installation Drawings
- CEAS - Engine Room Dimensioning
- Project Guides
- Extent of Delivery (EOD)
- Technical Papers

The publications are available at:
www.mandiesel.com → 'Marine' → 'Low Speed'

Engine Selection Guides

The 'Engine Selection Guides' are intended as a tool to provide assistance at the very initial stage of the project work. The guides give a general view of the MAN B&W two-stroke Programme for MC as well as for ME engines and include information on the following subjects:

- Engine data
- Engine layout and load diagrams
specific fuel oil consumption
- Turbocharger selection
- Electricity production, including power take off
- Installation aspects
- Auxiliary systems
- Vibration aspects.

After selecting the engine type on the basis of this general information, and after making sure that the engine fits into the ship's design, then a more detailed project can be carried out based on the 'Project Guide' for the specific engine type selected.

Project Guides

For each engine type of MC or ME design a 'Project Guide' has been prepared, describing the general technical features of that specific engine type, and also including some optional features and equipment.

The information is general, and some deviations may appear in a final engine documentation, depending on the content specified in the contract and on the individual licensee supplying the engine. The Project Guides comprise an extension of the general information in the Engine Selection Guide, as well as specific information on such subjects as:

- Engine Design
- Engine Layout and Load Diagrams, SFOC
- Turbocharger Selection & Exhaust Gas By-pass
- Electricity Production
- Installation Aspects
- List of Capacities: Pumps, Coolers & Exhaust Gas
- Fuel Oil
- Lubricating Oil
- Cylinder Lubrication
- Piston Rod Stuffing Box Drain Oil
- Central Cooling Water System
- Seawater Cooling
- Starting and Control Air
- Scavenge Air
- Exhaust Gas
- Engine Control System
- Vibration Aspects
- Monitoring Systems and Instrumentation
- Dispatch Pattern, Testing, Spares and Tools
- Project Support and Documentation.

Computerised Engine Application System (CEAS)

Further customised information can be obtained from MAN Diesel as project support and, for this purpose, we have developed a 'Computerised Engine Application System' (CEAS), by means of which specific calculations can be made during the project stage, such as:

- Estimation of ship's dimensions
- Propeller calculation and power prediction
- Selection of main engine
- Main engines comparison
- Layout/load diagrams of engine
- Maintenance and spare parts costs of the engine
- Total economy – comparison of engine rooms
- Steam and electrical power – ships' requirement
- Auxiliary machinery capacities for derated engine
- Fuel and lube oil consumption – exhaust gas data
- Heat dissipation of engine
- Utilisation of exhaust gas heat
- Water condensation separation in air coolers
- Noise – engine room, exhaust gas, structure borne
- Preheating of diesel engine
- Utilisation of jacket cooling water heat, fresh water production
- Starting air system
- Exhaust gas back pressure
- Engine room data: pumps, coolers, tanks.

For further information, please refer to www.mandiesel.com under 'Marine' → 'Low speed' → 'CEAS Engine Room Dimensions'.

Extent of Delivery

The 'Extent of Delivery' (EoD) sheets have been compiled in order to facilitate communication between owner, consultants, yard and engine maker during the project stage, regarding the scope of supply and the alternatives (options) available for MAN B&W two-stroke engines.

We provide four different EoDs:

EoD 98 - 50 MC Type Engine

EoD 46 - 26 MC Type Engines

EoD 98 - 50 ME Type Engines

EoD 60 - 35 ME-B Type Engines

These publications are available at:
www.mandiesel.com under 'Marine' → 'Low speed'
 → 'Project Guides and Extent of Delivery (EOD)'

Content of Extent of Delivery

The 'Extent of Delivery' includes a list of the basic items and the options of the main engine and auxiliary equipment and, it is divided into the systems and volumes stated below:

General information

4 00 xxx	General information
4 02 xxx	Rating
4 03 xxx	Direction of rotation
4 06 xxx	Rules and regulations
4 07 xxx	Calculation of torsional and axial vibrations
4 09 xxx	Documentation
4 11 xxx	Voltage on board for electrical consumers
4 12 xxx	Dismantling, packing and shipping of engine
4 14 xxx	Testing of diesel engine
4 17 xxx	Supervisors and advisory work
4 20 xxx	Propeller
4 21 xxx	Propeller hub
4 22 xxx	Stern tube
4 23 xxx	Propeller shaft
4 24 xxx	Intermediate shaft
4 25 xxx	Propeller shaftline
4 26 xxx	Propeller, miscellaneous

Diesel engine

4 30 xxx	Diesel engine
4 31 xxx	Torsional and axial vibrations
4 35 xxx	Fuel oil piping
4 40 xxx	Lubricating oil piping
4 42 xxx	Cylinder lubricating oil piping
4 43 xxx	Piston rod stuffing box drain piping
4 45 xxx	Low temperature cooling water piping
4 46 xxx	Jacket cooling water piping
4 50 xxx	Starting and control air piping
4 54 xxx	Scavenge air cooler
4 55 xxx	Scavenge air piping
4 59 xxx	Turbocharger
4 60 xxx	Exhaust gas piping
4 65 xxx	Engine control system
4 70 xxx	Local instrumentation
4 75 xxx	Monitoring, safety, alarm and remote indication
4 78 xxx	Electrical wiring on engine

Miscellaneous

4 80 xxx	Miscellaneous
4 81 xxx	Painting
4 82 xxx	Engine seating
4 83 xxx	Galleries
4 85 xxx	Power Take Off
4 87 xxx	Spare parts
4 88 xxx	Tools

Remote control system

4 95 xxx	Bridge control system
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Description of the 'Extent of Delivery'

The 'Extent of Delivery' (EoD) is the basis for specifying the scope of supply for a specific order.

The list consists of 'Basic' and 'Optional' items.

The 'Basic' items define the simplest engine, designed for attended machinery space (AMS), without taking into consideration any specific requirements from the classification society, the yard, the owner or any specific regulations.

The 'Options' are extra items that can be alternatives to the 'Basic', or additional items available to fulfil the requirements/functions for a specific project.

Copenhagen Standard Extent of Delivery

We base our first quotations on a 'mostly required' scope of supply, which is the so called 'Copenhagen Standard EoD', which are marked with an asterisk *.

This includes:

- Items for Unattended Machinery Space
- Minimum of alarm sensors recommended by the classification societies and MAN Diesel
- Moment compensator for certain numbers of cylinders
- MAN Diesel turbochargers
- The basic Engine Control System
- CoCoS-EDS online
- Spare parts either required or recommended by the classification societies and MAN Diesel
- Tools required or recommended by the classification societies and MAN Diesel.

The filled-in EoD is often used as an integral part of the final contract.

Installation Documentation

When a final contract is signed, a complete set of documentation, in the following called 'Installation Documentation', will be supplied to the buyer by the engine maker.

The 'Installation Documentation' is normally divided into the 'A' and 'B' volumes mentioned in the 'Extent of Delivery' under items:

4 09 602 Volume 'A':

Mainly comprises general guiding system drawings for the engine room

4 09 603 Volume 'B':

Mainly comprises specific drawings for the main engine itself

Most of the documentation in volume 'A' are similar to those contained in the respective Project Guides, but the Installation Documentation will only cover the order-relevant designs. These will be forwarded within 4 weeks from order.

The engine layout drawings in volume 'B' will, in each case, be customised according to the buyer's requirements and the engine manufacturer's production facilities. The documentation will be forwarded, as soon as it is ready, normally within 3-6 months from order.

As MAN Diesel and most of our licensees are using computerised drawings UniGraphics, Cadam and TIFF format, the documentation forwarded will normally be in size A4 or A3. The maximum size available is A1.

The drawings of volume 'A' are available on CD ROM.

The following list is intended to show an example of such a set of Installation Documentation, but the extent may vary from order to order.

Engine-relevant documentation

Main Section 901 Engine data

- External forces and moments
- Guide force moments
- Water and oil in engine
- Centre of gravity
- Basic symbols for piping
- Instrument symbols for piping
- Balancing

Main Section 915 Engine connections

- Scaled engine outline
- Engine outline
- List of flanges/counterflanges
- Engine pipe connections
- Gallery outline

Main Section 921 Engine instrumentation

- List of instruments
- Connections for electric components
- Guidance values for automation

Main Section 923 Engine Control System

- Engine Control System, description
- Engine Control System, diagrams
- Pneumatic system
- Speed correlation to telegraph
- List of components
- Sequence diagram

Main Section 924 Oil mist detector

- Oil mist detector

Main Section 925 Control equipment for auxiliary blower

- Electric wiring diagram
- Auxiliary blower
- Starter for electric motors

Main Section 932 Shaft line

- Crankshaft driving end
- Fitted bolts

Main Section 934 Turning gear

- Turning gear arrangement
- Turning gear, control system
- Turning gear, with motor

Main Section 939 Engine paint

Specification of paint

Main Section 940 Gaskets, sealings, O-rings

Instructions

Packings

Gaskets, sealings, O-rings

Main Section 950 Engine pipe diagrams

Engine pipe diagrams

Bedplate drain pipes

Instrument symbols for piping

Basic symbols for piping

Lube oil, cooling oil and hydraulic oil piping

Cylinder lube oil pipes

Stuffing box drain pipes

Cooling water pipes, air cooler

Jacket water cooling pipes

Fuel oil drain pipes

Fuel oil pipes

Control air pipes

Starting air pipes

Turbocharger cleaning pipe

Scavenge air space, drain pipes

Scavenge air pipes

Air cooler cleaning pipes

Exhaust gas pipes

Steam extinguishing, in scav.box

Oil mist detector pipes

Pressure gauge pipes

Engine room-relevant documentation**Main Section 901 Engine data**

List of capacities

Basic symbols for piping

Instrument symbols for piping

Main Section 902 Lube and cooling oil

Lube oil bottom tank

Lubricating oil filter

Crankcase venting

Lubricating and hydraulic oil system

Lube oil outlet

Main Section 904 Cylinder lubrication

Cylinder lube oil system

Main Section 905 Piston rod stuffing box

Stuffing box drain oil cleaning system

Main Section 906 Seawater cooling

Seawater cooling system

Main Section 907 Jacket water cooling

Jacket water cooling system

Deaerating tank

Deaerating tank, alarm device

Main Section 909 Central cooling system

Central cooling water system

Deaerating tank

Deaerating tank, alarm device

Main Section 910 Fuel oil system

Fuel oil heating chart

Fuel oil system

Fuel oil venting box

Fuel oil filter

Main Section 911 Compressed air

Starting air system

Main Section 912 Scavenge air

Scavenge air drain system

Main Section 913 Air cooler cleaning

Air cooler cleaning system

Main Section 914 Exhaust gas

Exhaust pipes, bracing

Exhaust pipe system, dimensions

Main Section 917 Engine room crane

Engine room crane capacity, overhauling space

Main Section 918 Torsiograph arrangement

Torsiograph arrangement

Main Section 919 Shaft earthing device

Earthing device

Main Section 920 Fire extinguishing in scavenge air space

Fire extinguishing in scavenge air space

Main Section 921 Instrumentation

Axial vibration monitor

Main Section 926 Engine seating

Profile of engine seating

Epoxy chocks

Alignment screws

Main Section 927 Holding-down bolts

Holding-down bolt

Round nut

Distance pipe

Spherical washer

Spherical nut

Assembly of holding-down bolt

Protecting cap

Arrangement of holding-down bolts

Main Section 928 Supporting chocks

Supporting chocks

Securing of supporting chocks

Main Section 929 Side chocks

Side chocks

Liner for side chocks, starboard

Liner for side chocks, port side

Main Section 930 End chocks

Stud for end chock bolt

End chock

Round nut

Spherical washer, concave

Spherical washer, convex

Assembly of end chock bolt

Liner for end chock

Protecting cap

Main Section 931 Top bracing of engine

Top bracing outline

Top bracing arrangement

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Main Section 932 Shaft line

Static thrust shaft load

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Main Section 933 Power Take-Off

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PTO/RCF arrangement, if fitted

Main Section 936 Spare parts dimensions

Connecting rod studs

Cooling jacket

Crankpin bearing shell

Crosshead bearing

Cylinder cover stud

Cylinder cover

Cylinder liner

Exhaust valve

Exhaust valve bottom piece

Exhaust valve spindle

Exhaust valve studs

Fuel valve

Main bearing shell

Main bearing studs

Piston complete

Starting valve

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Thrust block segment

Turbocharger rotor

Main Section 940 Gaskets, sealings, O-rings

Gaskets, sealings, O-rings

Main Section 949 Material sheets

MAN B&W Standard Sheets Nos:

- S19R
- S45R
- S25Cr1
- S34Cr1R
- C4

Engine production and installation-relevant documentation**Main Section 935 Main engine production records, engine installation drawings**

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Reference sag line for piano wire
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Piano wire measurement of bedplate
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Inspection after shop trials
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Main Section 941 Shop trials

Shop trials, delivery test
Shop trial report

Main Section 942 Quay trial and sea trial

Stuffing box drain cleaning
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Adjustment of fuel pump
Heavy fuel operation
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Main Section 945 Flushing procedures

Lubricating oil system cleaning instruction

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Hydraulic jack for holding down bolts
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List of tools
Outline dimensions, main tools

Main Section 938 Tool panel

Tool panels

Auxiliary equipment

980 Fuel oil supply unit, if ordered
990 Exhaust silencer, if ordered
995 Other auxiliary equipment



















Appendix

A

Symbols for Piping

No.	Symbol	Symbol designation	No.	Symbol	Symbol designation
1	General conventional symbols		2.14		Spectacle flange
1.1		Pipe	2.15		Bulkhead fitting water tight, flange
1.2		Pipe with indication of direction of flow	2.16		Bulkhead crossing, non-watertight
1.3		Valves, gate valves, cocks and flaps	2.17		Pipe going upwards
1.4		Appliances	2.18		Pipe going downwards
1.5		Indicating and measuring instruments	2.19		Orifice
2	Pipes and pipe joints		3	Valves, gate valves, cocks and flaps	
2.1		Crossing pipes, not connected	3.1		Valve, straight through
2.2		Crossing pipes, connected	3.2		Valves, angle
2.3		Tee pipe	3.3		Valves, three way
2.4		Flexible pipe	3.4		Non-return valve (flap), straight
2.5		Expansion pipe (corrugated) general	3.5		Non-return valve (flap), angle
2.6		Joint, screwed	3.6		Non-return valve (flap), straight, screw down
2.7		Joint, flanged	3.7		Non-return valve (flap), angle, screw down
2.8		Joint, sleeve	3.8		Flap, straight through
2.9		Joint, quick-releasing	3.9		Flap, angle
2.10		Expansion joint with gland	3.10		Reduction valve
2.11		Expansion pipe	3.11		Safety valve
2.12		Cap nut	3.12		Angle safety valve
2.13		Blank flange	3.13		Self-closing valve

No.	Symbol	Symbol designation	No.	Symbol	Symbol designation
3.14		Quick-opening valve	4	Control and regulation parts	
3.15		Quick-closing valve	4.1		Hand-operated
3.16		Regulating valve	4.2		Remote control
3.17		Kingston valve	4.3		Spring
3.18		Ballvalve (cock)	4.4		Mass
3.19		Butterfly valve	4.5		Float
3.20		Gate valve	4.6		Piston
3.21		Double-seated changeover valve	4.7		Membrane
3.22		Suction valve chest	4.8		Electric motor
3.23		Suction valve chest with non-return valves	4.9		Electro-magnetic
3.24		Double-seated changeover valve, straight	5	Appliances	
3.25		Double-seated changeover valve, angle	5.1		Mudbox
3.26		Cock, straight through	5.2		Filter or strainer
3.27		Cock, angle	5.3		Magnetic filter
3.28		Cock, three-way, L-port in plug	5.4		Separator
3.29		Cock, three-way, T-port in plug	5.5		Steam trap
3.30		Cock, four-way, straight through in plug	5.6		Centrifugal pump
3.31		Cock with bottom connection	5.7		Gear or screw pump
3.32		Cock, straight through, with bottom conn.	5.8		Hand pump (bucket)
3.33		Cock, angle, with bottom connection	5.9		Ejector
3.34		Cock, three-way, with bottom connection	5.10		Various accessories (text to be added)

No.	Symbol	Symbol designation	No.	Symbol	Symbol designation
5.11		Piston pump	7	Indicating instruments with ordinary symbol designations	
6	Fittings		7.1		Sight flow indicator
6.1		Funnel	7.2		Observation glass
6.2		Bell-mounted pipe end	7.3		Level indicator
6.3		Air pipe	7.4		Distance level indicator
6.4		Air pipe with net	7.5		Counter (indicate function)
6.5		Air pipe with cover	7.6		Recorder
6.6		Air pipe with cover and net			
6.7		Air pipe with pressure vacuum valve			
6.8		Air pipe with pressure vacuum valve with net			
6.9		Deck fittings for sounding or filling pipe			
6.10		Short sounding pipe with selfclosing cock			
6.11		Stop for sounding rod			

The symbols used are in accordance with ISO/R 538-1967, except symbol No. 2.19

178 30 61-4.1

Fig. A.01.01: Symbols for piping