Technical Documentation

Project guide

G80ME-C10.5-LGIM

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G80ME-C10.5-LGIM

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Preface

MAN B&W G80ME-C10.5-LGIM

Project Guide

Electronically Controlled

Dual Fuel 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 Update

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.man-es.com --> 'Two stroke'.

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 the internet at: www.man-es.com --> 'Two stroke'. where they can be downloaded.

Edition 0.5

September 2023

All data provided in this document is non-binding. This data serves informational purposes only and is especially not guaranteed in any way.

Depending on the subsequent specific individual projects, the relevant data may be subject to changes and will be assessed and determined individually for each project. This will depend on the particular characteristics of each individual project, especially specific site and operational conditions.

If this document is delivered in another language than English and doubts arise concerning the translation, the English text shall prevail.

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ME-LGI dual-fuel engine

The development in fuel prices in combination with emission control regulations has created a need for dual-fuel engines.

The MAN B&W ME-LGI engine is the dual-fuel solution for methanol and LPG injected on liquid form into the engine. The LGI functionality is designed as an add-on to the electronically controlled ME engine.

In this project guide, second fuel is used as the common denomination for methanol and liquefied petroleum gas (LPG).

The fuel oil system has not been altered significantly on an LGI engine compared to a standard ME engine.

The ME-LGI engine can be delivered in different versions depending on the second fuel type used, the ME-LGIP engine for LPG fuels and the ME-LGIM engine for methanol.

ME-LGI injection system

Dual-fuel operation requires injection of pilot fuel (to start the combustion) and second fuel into the combustion chamber.

Different types of valves are used for the injection of second fuel and pilot fuel. Auxiliary media required for dual-fuel operation and fuel-oil-only operation are:

- Second fuel (methanol or LPG)
- Pilot fuel (fuel oil supplied by the existing ME fuel oil system)
- Control oil for the LGI control block and for fuel booster injection valve (FBIV) operation
- Sealing oil for lubricating moving parts in the FBIV and for separating second fuel and control oil.

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ME-LGI vs ME engine design

Although few technical differences separate fuel oil and dual-fuel engines, the ME-LGI engine provides optimal fuel flexibility. Fig. 1.00.01 shows components that are modified and added to the engine, allowing it to operate in dual-fuel modes.

The added systems/components and related functions for dual-fuel operation are:

- Fuel valve train (FVT) providing a block-and-bleed function between the second fuel supply system and the engine
- Double-walled piping for distribution of second fuel to individual cylinders
- Ventilation and leakage detection system which vents the space between the inner and outer pipes of the double-walled piping, a hydrocarbon (HC) analyser checks the hydrocarbon content of the vented air. The inlet air is taken from a non-hazardous area and exhausted outside the engine room
- Fuel booster injection valves (FBIV) for injection of second fuel into the combustion chamber.

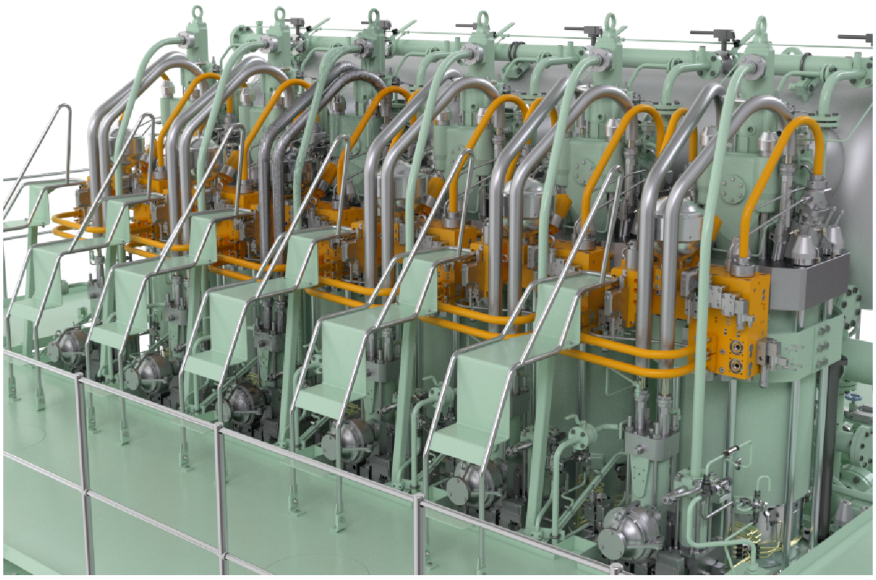

Fig. 1.00.01: An ME-LGI engine with double-walled second fuel pipes, control blocks and FBIVs

- Hydraulic control systems which control the fuel booster valve operation.
- Sealing oil supply unit fully integrated on the engine delivers sealing oil to FBIVs. The shipyard does not need to consider this installation.
- Draining and inert gas purging system for quick and reliable removal of second fuel from the engine.
- In addition to the engine control system (ECS), a safety system monitors the second fuel injection and combustion and ensures that the engine reverts to diesel oil operation in case of alarms.

Engine operating modes

One main advantage of the ME-LGI engine is its fuel flexibility. The control concept comprises three different fuel modes:

- Dual-fuel operation with minimum pilot oil amount
- Specified dual-fuel operation (SDF) with injection of a fixed second fuel amount
- Compliant fuel-oil-only mode

The dual-fuel operating mode is used for second fuel operation. It can only be started manually by an operator on the main operating panel (MOP) in the control room.

The specified dual-fuel operating mode gives the operator full fuel flexibility and the option to admit a fixed amount of second fuel. The ME control system adds fuel oil until the required engine load is reached.

Fuel-oil-only mode is known from the ME engine. Operating the engine in this mode can only be done on fuel oil.

Safety

The ME-LGI control and safety system is designed to fail to safe condition. All failures detected during dual-fuel running result in a stop of second fuel supply and a changeover to fuel oil operation. This condition applies also to failures of the control system itself.

Following the changeover, the second fuel pipes and the complete second supply system are blown-out and freed from fuel by purging.

The changeover to fuel oil mode is always done without any power loss of the engine.

Second fuel supply systems

Different applications call for different second fuel supply systems.

The supply system can generate the second fuel pressure in different ways depending on the chosen technology. The number of coolers, stages, and type of pumps can vary from maker to maker.

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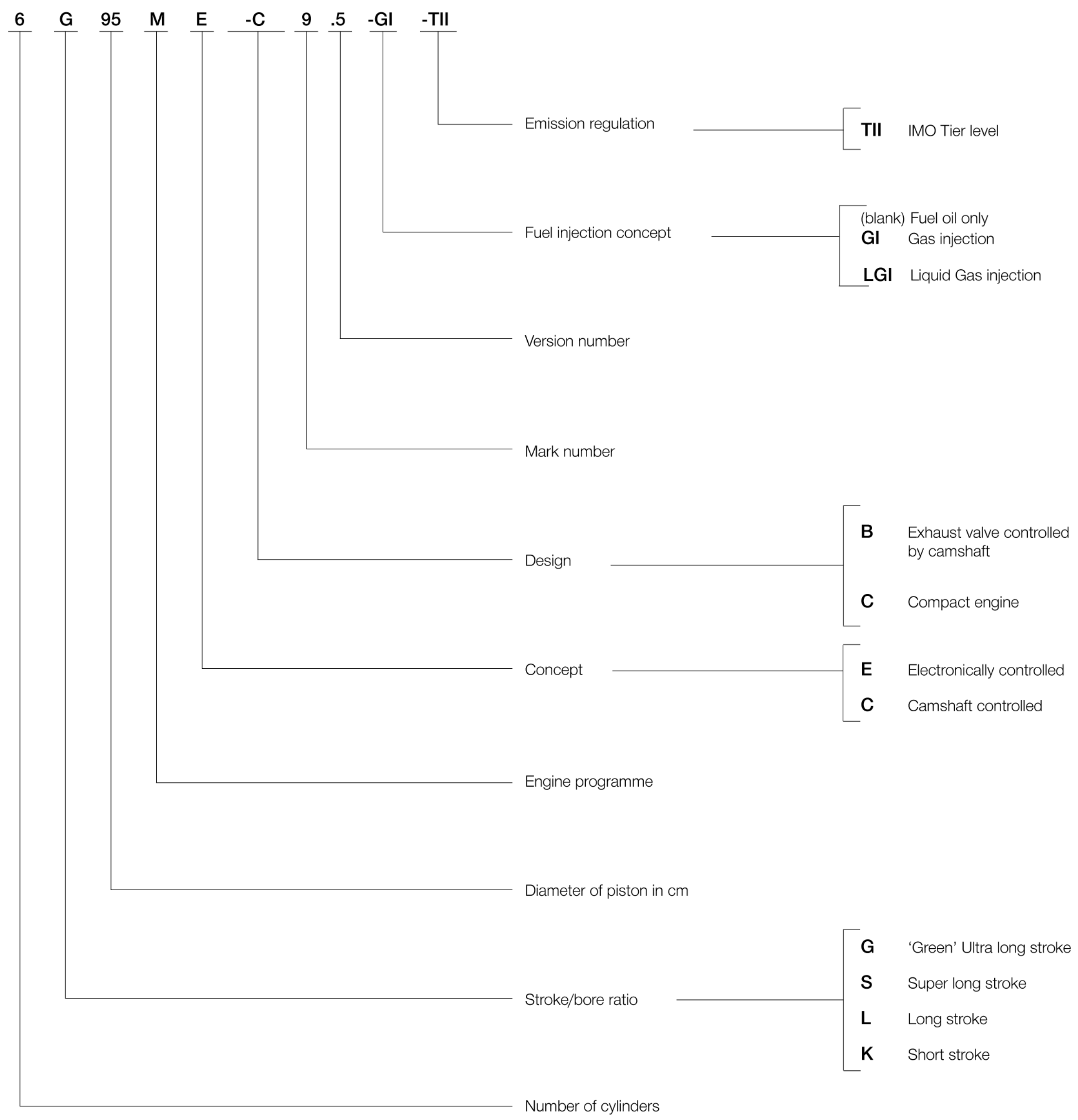
Engine type designation



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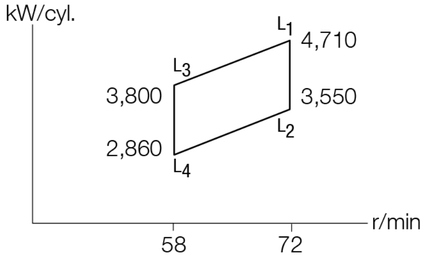
Power, speed and fuel oil

Tier II Tier III

MAN B&W G80ME-C10.5-LGIM

Cyl.	L_1 kW
6*	28,260
7	32,970
8	37,680
9	42,390

Stroke: 3,720 mm/**L_1 MEP:** 21.0 bar



MAN B&W G80ME-C10.5-LGIM (standard methanol tuning)

L_1 dual fuel mode (SMC+SPOC (5.0%))/fuel oil mode (SFOC) [g/kWh]	50%	75%	100%
Standard tuned	307.9+13.0/156.5	320.1+9.9/159.0	339.7+8.2/166.5

MAN B&W G80ME-C10.5-LGIM-EGRTC (standard methanol tuning)

L_1 dual fuel mode (SMC+SPOC (5.0%))/fuel oil mode (SFOC) [g/kWh]	50%	75%	100%
Tier II mode	307.9+13.0/156.5	320.1+9.9/159.0	338.6+8.2/166.0
Tier III mode	320.8+13.0/162.5	327.6+9.9/162.5	342.9+8.2/168.0

* 6-cylinder engines can be ordered with reduced or external moment compensation depending on rating and ship dynamics. Evaluation is made on request.

Fig. 1.03.01: Power, speed and fuel oil

SFOC for derated engines can be calculated in the CEAS engine application at www.man-es.com

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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.

For conversions between kW and metric horsepower, please note that 1 BHP = 75 kpm/s = 0.7355 kW.

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.

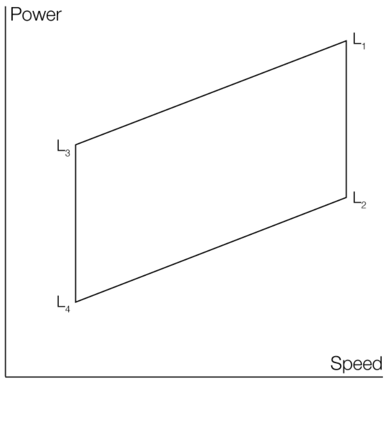

Fig. 1.04.01: Layout diagram for engine power and speed

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 ....1,000 mbar
Seawater temperature .... 32°C
Relative humidity 60%

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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 NO_X 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

Specific fuel oil consumption varies with ambient conditions and fuel oil lower calorific value. For calculation of these changes, see Chapter 2.

Gas consumption

The energy consumption (heat rate) for the -GI engine is lower when running on gas in dual fuel mode (heat rate in kJ/kWh) compared to fuel only mode.

When a given amount of oil is known in g/kWh, and after deducting the pilot fuel oil the additional gas consumption can be found by converting the energy supplied as gas into cubic metre per hour according to the LCV of the gas.

In the following sections, the energy consumption is calculated as related equivalent fuel consumption, i.e. with all our usual figures.

Example:

Related equivalent SFOC og gas .... 169 g/kWh
Ref. LCV .... 42,700 kJ
Heat rate ....0.169 x 42,700 = 7,216 kJ/kWh

The heat rate is also referred to as the ‘Guiding Equivalent Energy Consumption’.

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 can be increased. This is explained in Section 9.02.

Performance curves

Updated engine and capacities data is available from the CEAS program on www.marine.man-es.com --> 'Two-Stroke' --> 'CEAS Engine Calculations'.

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ME-LGI engine description

General

Engines built by MAN Energy Solutions' licensees are in accordance with our drawings and standards. In certain cases, local standards may be applied but all spare parts are interchangeable with parts designed by MAN Energy Solutions.

Some components may differ from MAN Energy Solutions' design because of local production facilities, or the application of local standard components.

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

Bedplate and main bearing

The bedplate consists of high welded longitudinal girders and welded cross girders with cast steel bearing supports. Long elastic holding-down bolts and hydraulic tightening tools are used to fit the bedplate to the engine seating in the ship. The bedplate is made with the thrust bearing in the aft end.

For engines mounted on epoxy chocks, the bedplate is made without taper.

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

The main bearings consist of thin-walled steel shells lined with white metal. The main bearing bottom shell can be rotated out and in with special tools and hydraulic tools for lifting the crankshaft. A bearing cap keeps the shells in position.

Frame box

The frame box is welded. On the exhaust side of the frame box, a relief valve is mounted for each cylinder. On the manoeuvring side, each cylinder has a large hinged door. Crosshead guides are welded onto the frame box. The frame box is bolted to the bedplate. Stay bolts tighten together bedplate, frame box, and cylinder frame.

The framebox is of the well-proven triangular guide-plane design with twin staybolts giving excellent support for the guide shoe forces.

Cylinder frame and stuffing box

For the cylinder frame, two possibilities are available:

- Nodular cast iron
- Welded design with integrated scavenge air receiver.

The cylinder frame has access covers for cleaning the scavenge air space, if required, and for inspection of scavenge ports and piston rings from the manoeuvring side. The cylinder frame and cylinder liner forms the scavenge air space.

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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 placed on the cylinder frame. The bottom of the cylinder frame contains the piston rod stuffing box with sealing rings for scavenge air, and oil scraper rings preventing crankcase oil from entering the scavenge air space.

Scavenge air space and piston rod stuffing box drains are located in the bottom of the cylinder frame.

Cylinder liner

A cylinder liner made of alloyed cast iron 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, drilled holes for cylinder lubrication and gas admission.

Cylinder liners prepared for installation of temperature sensors are available as an option.

Cylinder cover

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

The side of the cylinder cover facing the hydraulic cylinder unit (HCU) block has a face for mounting a special valve block, the gas control block, see later description.

In addition, the cylinder cover is provided with one set of bores and pipes for supplying and returning SF from the control block to each injection valve.

The bore for the indicator valve is also used for PMI Auto-tuning.

Crankshaft

The crankshaft is of the semi-built type made from forged or cast steel throws. Depending on the number of cylinders, the crankshaft may be supplied in two parts.

At the aft end, the crankshaft is provided with:

- a collar for the thrust bearing
- a flange for fitting the gear wheel for the step-up gear to the hydraulic power supply unit
- a flange for the turning wheel and for the coupling bolts to an intermediate shaft.

At the front end, the crankshaft is fitted with a collar for the axial vibration damper and a flange for fitting a tuning wheel. The flange can also be used for power take off.

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 thrust collar, segments, and bedplate to end chocks and engine seating, and to the ship’s hull.

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

Engines with nine cylinders or more are specified with a 360-degree type thrust bearing, while a 240-degree type is used for all other engines. The flexible thrust cam design of MAN Energy Solutions' 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

For a mechanically engine-driven hydraulic power supply, the crankshaft drives the main hydraulic oil pumps via a step-up gear. The main engine lubricating oil system lubricates the step-up gear.

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 arrangement mounted on the bedplate. The turning gear is driven by an electric motor with a built-in brake.

A blocking device prevents the main engine from starting when the turning gear is engaged. Engagement and disengagement of the turning gear is done manually by moving the pinion.

The basic design includes a control device for the turning gear, consisting of starter and manual control box.

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.

The vibration damper has a mechanical guide to enable a functional check, an optionally electronic vibration monitor can be supplied.

Engines Mk. 9 and higher require an axial vibration monitor, which indicates condition checks of the axial vibration damper, and terminals for alarm and slowdown.

Tuning wheel / torsional vibration damper

Depending on the final torsional vibration calculations, it may be necessary to order a tuning wheel or a torsional vibration damper, separately.

Connecting rod

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

The crosshead and crank pin bearing caps are secured to the connecting rod with studs and nuts tightened with 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 crank pin bearing is provided with thin-walled steel shells, lined with bearing metal. Lubricating oil is supplied through ducts in the crosshead and connecting rod.

Piston

The piston consists of a piston crown and a 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 topland.

The piston has three or four ring grooves which are hard-chrome plated on both the upper and lower surfaces of the grooves. Three or four piston rings are fitted depending on the engine type.

The uppermost piston ring is always a controlled pressure relief (CPR) ring type, whereas the other two or three piston rings are either of the CPR type, or have an oblique cut. Depending on the engine type, the uppermost piston ring is higher than the others. All rings are alu-coated on the outer surface for running-in.

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

Piston rod

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

Crosshead

A crosshead of forged steel is provided with cast steel guide shoes with white metal on the running surface. The guide shoe is of the low-friction type, and the crosshead bearings are of the wide pad design. The telescopic pipe for oil inlet and the pipe for oil outlet are mounted on the guide shoes.

Scavenge air system

The turbocharger draws air 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. The scavenge air receiver has a D-shape design.

Scavenge air cooler

Each turbocharger has a scavenge air cooler of the mono-block type.

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The scavenge air cooler is most commonly cooled by freshwater from a central cooling system. Alternatively, it can be cooled by seawater from a seawater cooling system, or from a combined cooling system with separate seawater and freshwater pumps. The working pressure is up to 4.5 bar.

The scavenge air cooler is designed to keep the temperature difference between the scavenge air and the water inlet at about 12°C, at specified MCR.

Auxiliary blower

The engine is provided with electrically-driven scavenge air blowers integrated in the scavenge air cooler. 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.

To obtain a safe start, the auxiliary blowers start consecutively before the engine is started to ensure sufficient scavenge air pressure.

Find more information 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 insulated and 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, Accelleron or MHI turbochargers.

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

Reversing

The engine is reversed electronically by the engine control system which changes the timing of fuel injection, exhaust valve activation, and starting air valves.

2nd order moment compensators

In general, 2nd order moment compensators are relevant only for 5- and 6-cylinder engines of 50 and 45 bore sizes.. When needed, an external electrically driven moment compensator type RotComp or similar can be installed in the steering room.

Section 17.02 describes 2nd order moment compensators as well as the basic design and options.

The hydraulic power supply

The hydraulic power supply (HPS) filters and pressurises the lube oil for the hydraulic system. The HPS consists of either mechanically driven (by the engine) main pumps with electrically driven start-up pumps or electrically driven combined main and start-up pumps. The hydraulic pressure is 300 bar.

The engine driven HPS is mounted aft on the engine, for engines with the chain drive placed aft (8 cylinders or less). For engines with the chain drive located in the middle (9 cylinders or more), the HPS is placed in the middle. Usually, the electrically driven HPS is mounted aft on the engine.

A combined HPS is available as an option, it is mechanically driven with electrically driven start-up/back-up pumps with backup capacity.

Hydraulic cylinder unit

The hydraulic cylinder unit (HCU), one per cylinder, consists of a distributor block mounted on a base plate. The distributor block has one or more accumulators to ensure the necessary peak flow of hydraulic oil during the electronically controlled fuel injection.

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

Fuel oil injection

The engine has one hydraulically activated fuel oil pressure booster for each cylinder. Fuel oil high-pressure pipes are of the double-wall type with built-in conical support. The pipes are insulated, but not heated. A ‘fuel oil leakage’ system on detects leakages, and if a leakage is detected, it immediately stops the injection on the affected 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 (ECS).

An automatic vent slide in the fuel valve allows the fuel oil to circulate through the fuel valve and high-pressure pipes when the engine is stopped. If the valve spindle sticks, the vent slide also prevents the compression chamber from being filled with fuel oil. Oil from the vent slide and other drains is led away in a closed system. Chapter 7 contains further information.

Section 7.01 contains further information.

SF pipes

A chain pipe system is fitted for distribution of SF to each adapter block. The chain pipes are connected to the SF control block via the adapter block.

SF pipes are double walled, with the outer shielding pipe designed to prevent SF outflow to the machinery space if leakages occur, or if the inner gas pipe ruptures.

A separate mechanical ventilation system with a capacity of 30 air changes per hour vents the intervening SF pipe space, including the space around valves, flanges, etc. Any SF leakage is led to the ventilated part of the double-walled piping system where hydrocarbon (HC) sensors detect the leakage.

The pressure in the intervening gas pipe space is kept below the engine room pressure. The extractor fan motor is placed outside the duct and the machinery space. Ventilation inlet air must be drawn from a SF safe area and exhausted to a safe place.

The gas pipes on the engine are designed for, and pressure tested at a pressure, which is 50% higher than the normal working pressure. SF pipes are supported to avoid mechanical vibrations, and they should be protected against drops of heavy items as well.

The chain piping to the individual cylinders is flexible to cope with mechanical stress from the thermal expansion of the engine when going from cold to hot conditions.

The SF pipe system is designed to avoid excessive SF pressure fluctuations during operation.

The SF pipes must be connected to an inert gas purging system.

SF control block

The SF control block consists of a square steel block bolted to the HCU side of the cylinder cover.

The control block incorporates one or more accumulators, ensuring stable pressure for the supply of SF to the SF injection valves.

Minute volumes around the SF injection valves in the cylinder cover are kept under vacuum by the venting air in the double-walled pipes.

Internal bores in the control block connects hydraulic oil, sealing oil, low-pressure oil and SF to the cylinder cover and the various valves mounted on the cover.

Depending on design, a combination of electronic window valve - liquid (ELWI-L), electronic fuel injection - liquid (ELFI-L), electronic block injection (ELBI) and electronic gas Injection (ELGI) valves are also incorporated in the control block.

These valves are controlled electrically and they control the activation of the SF injection valve and the window valve by means of control oil.

The pressure in the channel between the window valve and the SF injection valve is measured to monitor the function of the window valve and the SF injection valve and to detect if any of these are leaking. A large pressure increase indicates a severe leakage in the window valve and a pressure decrease indicates a severe leakage from the SF injection valve seats or in the connected pipes. The safety system will detect the pressure changes and shut down SF injection.

From the accumulator, the SF passes through a bore in the SF control block to the window valve, which in SF mode is opened and closed by hydraulic oil in each cycle. From the window valve, the SF is led via bores in the SF control block and in the cylinder cover to the SF injection valve.

Fuel valves, fuel booster injection valves and starting air valve

The cylinder cover is equipped with two or three conventional fuel valves, two or three fuel booster injection valves (FBIV) for injection of LPG or methanol, a starting air valve and an 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.

The opening of the FBIVs is controlled by the ELGI or ELFI-L valve, which operates on control oil taken from the system oil.

Each FBIV has an integrated booster unit, which ensures correct injection pressure.

An automatic vent slide allows circulation of fuel oil through the valve and the 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.

Supply of starting air is provided by one solenoid valve per cylinder, controlled by the CCUs of the engine control system (ECS).

The starting valve is opened by control air, timed by the ECS, and is closed by a spring.

Slow turning before starting is a program incorporated in the basic ECS.

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

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 Wide-seat design.

The exhaust valve spindle is a DuraSpindle, a spindle made of Nimonic is available as an option. 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 closed by an air spring.

The exhaust valve is of the low-force design. Operation of the exhaust valve is controlled by a multi-way valve (ELVA or FIVA). In operation, the valve spindle rotates slowly, driven by the exhaust gas acting on a vane wheel fixed to the spindle.

Sealing of the exhaust valve spindle guide is obtained with an oil bath, or controlled oil level (COL), in the bottom of the air cylinder above the sealing ring. This oil bath lubricates the exhaust valve spindle guide and sealing ring.

Indicator cock

The engine has an indicator cock for connecting the PMI pressure transducer.

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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 positions of the brackets are carefully chosen to provide the best possible overhauling and inspection conditions.

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

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

Piping arrangements

The engine is delivered with piping arrangements for:

- Fuel oil
- High-pressure gas supply
- Heating of fuel oil
- Lubricating oil, piston cooling oil, hydraulic oil and sealing oil for gas valves
- 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 (required only for Visatron VN 215/93, make Schaller Automation)
- Various drain pipes.

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

The piping has sockets for local instruments, alarm and safety equipment, and several sockets for supplementary signal equipment.

Chapter 18 contains more information about instrumentation.

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Engine cross section - TIII

General

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Propeller layout and engine matching with margins

The combination of speed and power obtained for the design condition of the ship may be called the propeller design point (PD). PD is placed on the light running propeller curve, in Fig. 2.01.01.

Some shipyards, and/or propeller manufacturers occasionally use an alternative propeller design point (PD’) for the propeller design. To reflect average conditions at sea, PD’ incorporates all or part of the so-called sea margin described later.

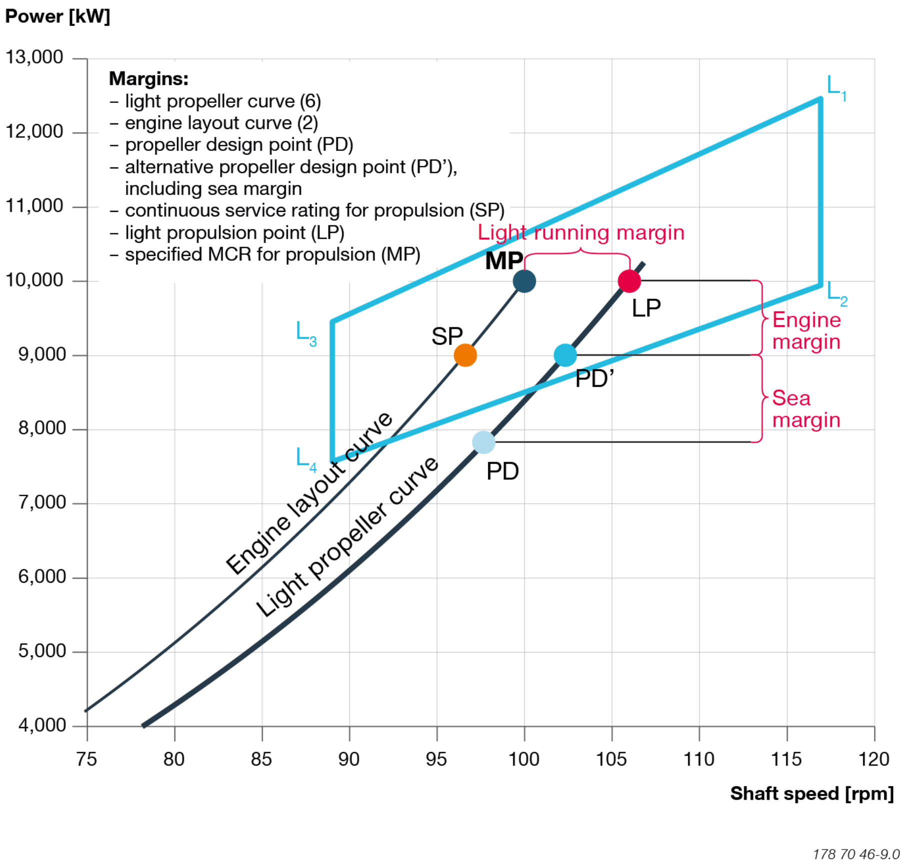

Fig. 2.01.01: Light propeller curve with margins added to establish the specified MCR for propulsion (MP) within a layout area for a specific engine

The service propulsion point (SP) on the engine layout curve is obtained by adding the sea margin, and the light running margin to the propeller design point (PD).

The specified MCR for propulsion, attained by including all margins described in the following sections, must be placed within the layout area. See the later section "Engine layout limitations" for further information.

Sea margin

As the sea is rarely completely calm and a wind often blows, the ship will experience increased resistance from wind and waves in average conditions. When determining the necessary engine power it is normal practice to add a power margin, the so-called sea margin. The sea margin ensures that the ship can maintain the design speed in average conditions at sea.

Traditionally, the sea margin has been approximately 15% of the power required to achieve the design speed with a clean hull in calm weather (PD). As ship design speeds reduce, it can be necessary to increase the sea margin since the resistance experienced by the ship is not reduced. If allowed by the EEDI regulation, it can be sensible to use a larger margin for ships often operating in heavy weather.

Engine margin

Often an engine owner will not permit 100% utilisation of engine power for normal operation due to the increase of the fuel consumption and the reduction of the power reserve. Therefore an engine margin of 10 to 15% is included to operate the engine at 90 or 85% load at the service propulsion point. Higher margins have been experienced for specific trades or reasons. If allowed by the EEDI regulation, a high engine margin is typically preferred for ships in scheduled traffic to make it possible to catch up with delays.

The engine margin can be increased for ships with a shaft generator, see the later section 2.03.

Light running margin

When the ship has sailed for some time, the hull and propeller becomes fouled. This will increase the hull resistance, and reduce the ship speed unless the engine delivers more power to the propeller to maintain the speed. As a result, the propeller will require more torque and the engine will be heavier loaded and heavy running (HR), which requires an increased torque from the engine.

When determining the necessary engine layout speed, the influence of a heavy running propeller and operation with increased vessel resistance must be considered. Therefore it is recommended to use a heavier propeller curve for the layout of the engine. The propeller curve for clean hull, and calm weather may then represent a "light running" (LR) propeller, whereas the engine layout curve represents a propeller curve in heavy weather, or with heavy fouling.

MAN Energy Solutions recommends using a light running margin (LRM) of normally 4.0–7.0%, however, for special cases up to 10%. It means that for a given engine power, the light running propeller rpm is 4.0–10.0% higher than the rpm given on the engine layout curve. Or, conversely, the engine is specified to a speed 4.0–7.0% lower than given on the design light propeller curve.

The point attained in the engine layout diagram, after shifting the propeller curve (containing the sea and engine margins) by the light running margin, is termed the “specified MCR for propulsion” point (MP).

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The high end of the range (7–10%) is primarily intended for vessels often operating in adverse conditions with a heavy running propeller. Low-powered EEDI ships such as tankers, and bulkers with blunt bows may also experience an operational benefit from a relatively high light-running margin.

Vessels with shaft generators, or vessels with high ice classes can also benefit from a light running margin in the high range, or in special cases even beyond 10%. It makes it possible to keep the shaft generator and power take-off (PTO) in operation for longer periods at sea. See the later guidance on PTO layout limits.

The SMCR values of engine power (SMCRpower) and speed (SMCRspeed) when including the margins can be calculated using equations 2.01.01 and 2.01.02.

Eg. 2.01.01

$$SMCR_{power} = PD_{power} \times \frac{\frac{100 + SM}{100}}{\frac{100 - EM}{100}}$$

178 70 80-3.0

Eg. 2.01.02

$$SMCR_{speed} = PD_{speed} \times \left(\frac{SMCR_{power}}{PD_{power}}\right)^{1/3} \times \left(1 - \frac{LRM}{100}\right)$$

178 70 81-5.0

The recommendation towards LRM is applicable to all draughts at which the ship is intended to operate, whether ballast, design or scantling draught. The recommendation is applicable to engine loads from 50 through 100%. If an average of the measured values between 50% and 100% load is used for verification of the attained light running margin, this will smoothen out the effect of measurement uncertainty and other variations.

Note on LRM

Light and heavy running, fouling, and sea margin are partially overlapping terms. Light and heavy running of the propeller refers to hull and propeller deterioration, as well as heavy weather and the resulting shift of the propeller curve towards the left in the load diagram. See for example Fig. 2.01.01. This shift stems from the increased torque required by the propeller during encounters of added resistance on the hull, that is, a lower rpm output at the same power.

The light running margin gives a margin towards engine limitations, see section 2.03. This margin ensures that the ship can deliver maximum power in conditions not as ideal as sea trial conditions. If the light running margin was not included, this might not be the case.

The sea margin gives the power margin necessary to maintain the service speed during average sea conditions with added wave and wind resistance. The light running margin ensures that the necessary power is available.

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Within the recommendations for the light running margin, the degree of light running must be decided based on experience from the actual trade, and the hull design of the vessel. In general, slender designs with sharp bows require smaller margins than full-body ships with blunt bows. The latter design results in an increase of the added resistance in adverse weather.

For further information on the effects of engine heavy running, see the later section: "Engine power and speed limits".

Optimum propeller speed

Propeller diameter and pitch

In general, the following applies: the larger the propeller diameter, D, the lower the optimum propeller speed and the power required for a certain design draught and ship speed, see curve D in Fig. 2.02.01.

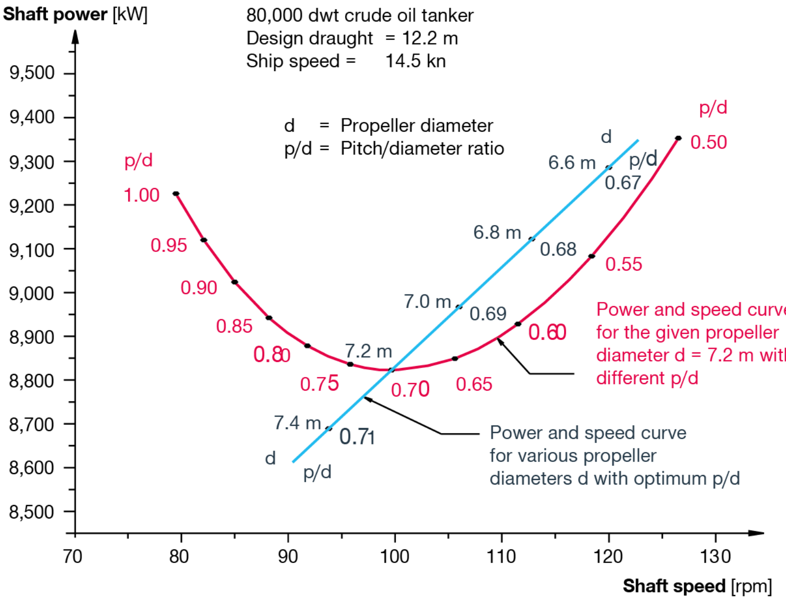

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Fig. 2.02.01: Influence of propeller diameter (D) and pitch on propeller design

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

Fig. 2.02.01 shows an example with 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 propeller diameter is increased from 6.6 m to 7.2 m, the power demand is reduced from approximately 9,290 kW to 8,820 kW. The optimum propeller speed is reduced from 120 rpm to 100 rpm. This corresponds to the constant ship speed coefficient α = 0.28. See the definition of α in the later section “Definition of constant ship speed lines”.

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

However, if the optimum propeller speed of 100 r/min does not match the preferred or selected main engine speed, a change of pitch away from the optimum will only cause a relatively small extra power demand. It will then be possible to keep the same maximum propeller diameter:

- To increase engine revolutions from 100 rpm to 110 rpm (P/D = 0.62) requires 8,900 kW, that is, an extra power demand of 80 kW
- To decrease engine revolutions from 100 rpm to 91 rpm (P/D = 0.81) requires 8,900 kW, that is, an extra power demand of 80 kW.

In both cases, the extra power demand is only 0.9%, and the corresponding "equal speed curves" are α = +0.1, and α = -0.1, respectively. So there is a certain interval of propeller speeds where the "power penalty" is limited. An interval that can be utilised to accommodate the most fuel-efficient engine.

Definition of constant ship speed lines

Fig. 2.02.02 shows the constant ship speed lines, α. These lines indicate the power required at various propeller speeds to keep the same ship speed, when an optimum pitch/diameter ratio is used.

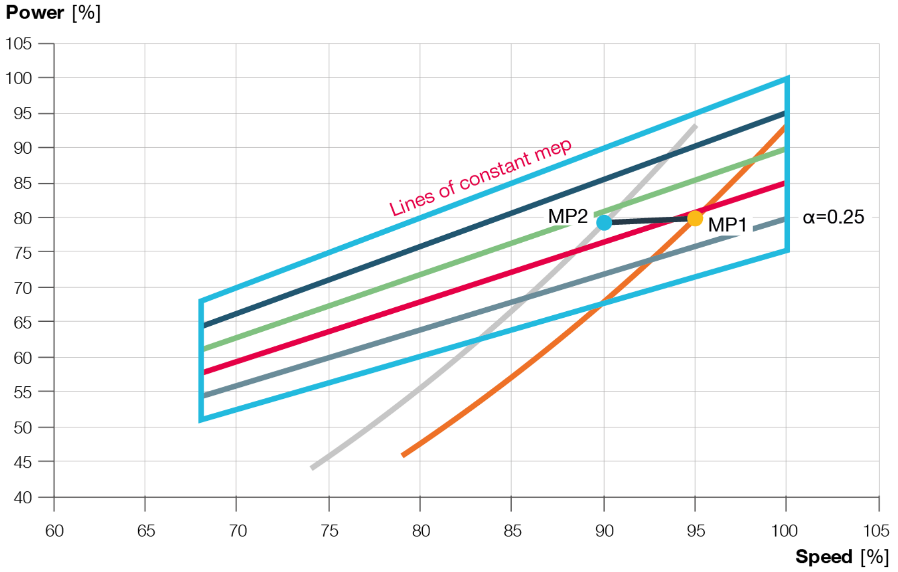

Fig. 2.02.02: Layout diagram, and constant ship speed lines

Normally, if propellers with an optimum pitch are used, the following relation between the necessary power, P, and the propeller speed, n, can be assumed:

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

where:

P is the propulsion power

n is the propeller speed

α is the constant ship speed coefficient.

For any combination of power and speed, points on lines which are parallel to the ship speed lines give the same ship speed. If a constant ship speed line is drawn into the layout area through the specified propulsion MCR point, MP1, then another specified propulsion MCR point, MP2, on this line will give the same ship speed.

2.02 Optimum propeller speed

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Fig. 2.02.02 shows an example of the required power-speed point MP1 through which a constant ship speed curve α = 0.25 is drawn. Thereby, point MP2 is obtained with a lower engine power, and a lower engine speed but with the same ship speed.

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

- For general cargo, bulk carriers, and tankers: α = 0.20–0.30
- For container vessels, and roll-on/roll-off cargo: α = 0.15–0.25.

When changing the propeller speed by changing the pitch, the α-constant will be different.

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Engine layout and load diagram

Engine layout limitations

As Fig. 2.03.01 shows, an engine's layout diagram is limited by:

1. Constant mean effective pressure (MEP) lines (L_1– L_3 and L_2– L_4)
2. Constant engine speed lines (L_1– L_2 and L_3– L_4).

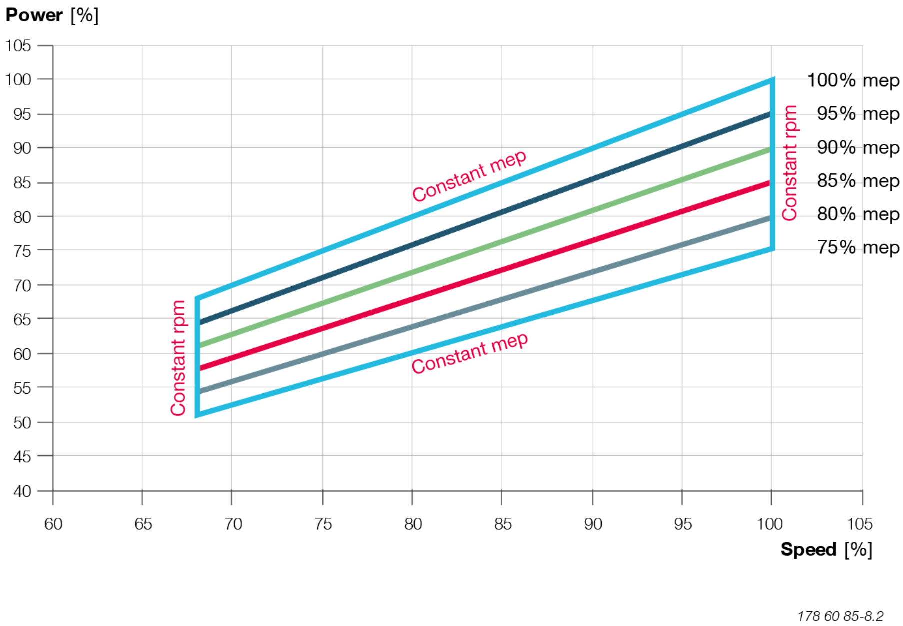

Fig. 2.03.01: Engine layout diagram with limits

Within the layout area, there is complete freedom to select the engine's specified maximum continuous rating (SMCR), point MP, which suits the ship's demand for power and speed.

The nominal maximum continuous rating (NMCR) of an engine design is equivalent to L_1 in the layout area.

The effective power, P, of a combustion engine is proportional to the mean effective pressure, pe, and engine speed, n. The expression for P, where c is a constant, is:

$P = c \times p_e \times n$

For constant mean effective pressure (MEP), the power is proportional to the speed:

$P = c \times n^1$ (for constant MEP)

When running with a fixed pitch propeller (FPP), the power can be expressed according to the propeller law as:

$P = c \propto n^3$ (propeller law)

Although the proportionality, $P \propto k \times V^3$. between the required power and the cubic, i.e. $n_i = 3$, of the speed is often referred to as a law, it is an assumption valid only for frictional resistance. If the ship has sufficient engine

2.03 Engine layout and load diagram

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power for operating at elevated speeds, the wave making resistance must be considered. At elevated ship speeds, V, the exponent can be higher, for example, $P \propto k \times V^4$.

Normally, estimates of the necessary propeller power and speed, n, are based on theoretical calculations for the design condition of the ship, and often also on computer simulations along with experimental tank tests. Calculations and simulations are based on optimum operating conditions, that is, a clean hull, good weather and calm seas.

Specified maximum continuous rating within the layout area

Based on the propulsion and engine running points, the layout diagram of the relevant main engine can be drawn in a power-speed diagram like in Fig. 2.03.01. The SMCR, point MP, must be placed inside the limitation lines of the layout diagram. Otherwise, the propeller speed has to be changed, or another main engine type must be chosen. The selected SMCR influences the mechanical design of the engine, for example, turbocharger, piston shims, liners, and fuel valve nozzles.

Once the specified SMCR has been chosen, the engine design, and the capacities of the auxiliary equipment will be adapted to the SMCR, as reflected in CEAS reports.

If the SMCR is changed later on, it may involve a change of:

- shafting system
- vibrational characteristics
- pump and cooler capacities
- fuel valve nozzles
- piston shims
- cylinder liner cooling, and lubrication.

Furthermore, it may be necessary to re-match the turbocharger, or even to change to a different size of turbocharger. Sometimes, such a change also requires a piping system of larger dimensions. If the specification has to be prepared for a later change of SMCR, it is important to consider this already at the project stage. It can be an option to design the ship with a derated engine, and auxiliaries (coolers, pumps and pipe dimensions, shafting, and so on) that are sufficient for a later uprating of the engine. This engine is termed a dual-rated engine, which must be indicated in the Extent of Delivery (EoD). Note, that EEDI regulations must permit this.

If a dual-rated engine is ordered, it is beneficial to carry out the testing necessary to get the IMO technical file for the alternative SMCR during shop testing of the engine. When testing is done before ship delivery, the more expensive in-ship testing of the engine is avoided. For all fuel variants of the ME-C engines, the timing of fuel injection and the timing of exhaust valve activation are electronically optimised over a wide operating range of the engine. 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.

Various tunings are available for pure Tier II engines. These tunings allow optimisation of an engine for the specific needs of a project. There are no tunings available for Tier III engines. See the later section “Example of SFOC curves”.

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Engine load diagram and limitations

Definitions

A load diagram defines the power and speed limits for continuous and overload operation of the installed engine, see Fig. 2.03.02.

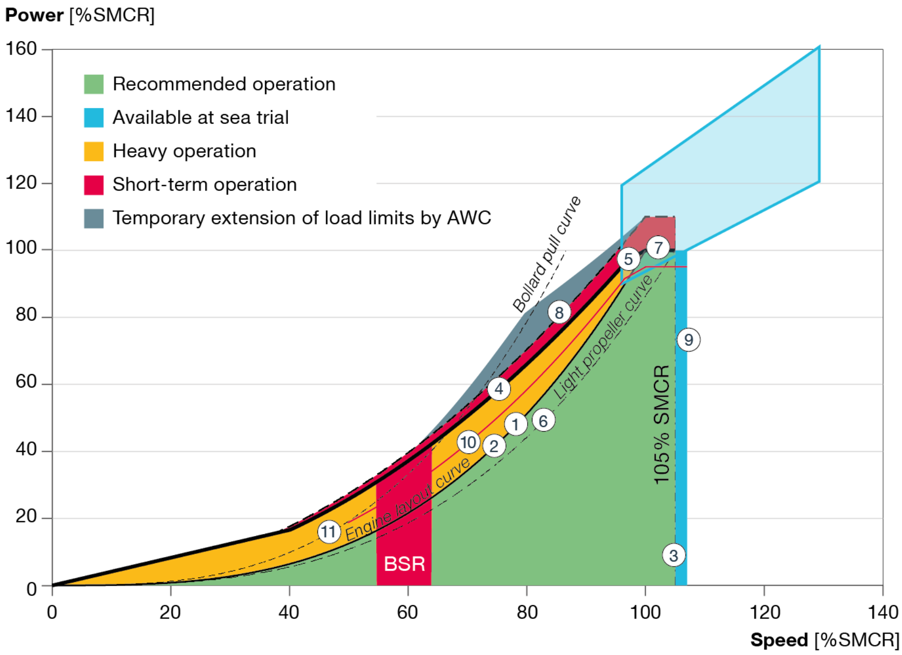

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Fig. 2.03.02: Engine load diagram with adverse weather condition (AWC) function for SMCR placed within the layout diagram (light blue)

The specified MCR, point MP, of the engine corresponds to the ship specification. The service points of the installed engine incorporate the engine power required for ship propulsion, and by the shaft generator.

Operating curves and limits

The lines on Fig. 2.03.02 describe the service range of the engine. We will refer to these lines throughout section 2.03. The location of the SMCR point within the layout diagram does not affect the appearance of the load diagram. The SMCR point alone defines the load diagram, since all specifications utilise the maximum capabilities of the engine design. See the later section on "Derating for lower specific fuel oil consumption".

Line 1:
Engine layout curve, per definition passing through 100% SMCR rpm, and 100% SMCR power. This curve coincides with the "heavy propeller curve", line 2. An engine without PTO will typically operate to the right of this curve about 95% of the time.

Line 2:
Heavy propeller curve, the light propeller curve (line 6) shifted with the light running margin to account for heavy weather, and fouled hull.

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Line 3:
Maximum rpm for continuous operation. For engines with an SMCR on the line L_1-L_2 in the layout diagram, up to 105% of L_1-rpm can be utilised. If the SMCR is sufficiently speed derated, 110% of SMCR rpm, but no more than 105% of L_1-rpm, can be utilised for standard engines. Torsional vibration conditions must permit the rpm values.

If the SMCR (MP) is sufficiently speed derated, and if torsional vibration conditions permit it, more than 110% speed is possible by choosing the “extended load diagram”. The extended load diagram is described later in this chapter.

Line 4:
Torque/speed limit of the engine, limited mainly by the thermal load on the engine.

Line 5:
Represents the maximum mean effective pressure (MEP) level acceptable for continuous operation. Note, that this is only a limit at high loads, and engine speeds. At lower speeds, line 4 is a stricter limit.

Line 6:
Light propeller curve for clean hull, and calm weather, often used for propeller layout. The light running margin is the rpm margin (in percent) between the engine layout curve (lines 1 and 2) and the light propeller curve.

Line 7:
Represents the maximum power for continuous operation. Note that when increasing rpm towards lines 3 and 9, the maximum power for continuous operation cannot exceed 100%.

Line 8:
The area between lines 4, 5, 7 and line 8 represents the overload operation limit of the engine. Overload running is possible only for limited periods, 1 hour out of every 12 hours, as the resulting thermal load on the engine is high.

Line 9:
Maximum acceptable rpm at sea trial conditions with clean hull and propeller in calm water. 110% of SMCR rpm, but no more than 107% of L_1-rpm if permitted by torsional vibrations.

If point M / the SMCR of the engine is sufficiently speed derated, more than 110% speed is possible by choosing “Extended load diagram” which is described later in this chapter.

Line 10:
PTO layout limit. This curve describes the maximum combined power required by the light propeller curve and the PTO at a given rpm with a shaft generator/PTO. The PTO layout limit is to be considered when dimensioning the system. This layout limit ensures operational margin to line 4, see the subsequent section “Shaft generator/PTO layout, governor stability and integration”.

Line 11: Bollard pull propeller curve. The heaviest propeller curve possible at zero advance speed. The bollard pull curve is typically 15 to 20% heavier than the light propeller curve, values may vary for individual designs.

AWC area
Extended overload operation limits of engines equipped with the adverse weather conditions function, the AWC function. The AWC function increases the percentage of engine power that can be developed with a heavy propeller, as long as required in an emergency.

When the function is activated, the electronic control of the ME engine alters the cyclic process of the combustion to reduce the negative effects of developing a high engine torque at low rpm. It is done at the cost of an increased specific fuel oil consumption. Due to the resulting SFOC increase, AWC is not to be considered a replacement for an adequate light running margin. See the later section “AWC function” for a further description of this function.

Limits for low-speed running

As the fuel injection for ME engines is automatically controlled over the entire power range, the engine is able to operate down to approximately 15-20% of the nominal L_1 speed, depending on the actual propulsion system. Absolute values of minimum speed must be determined at the sea trial.

Recommendation for operation

The area between lines 1, 3 and 7 is available for continuous operation without limitation.

The area between lines 1, 4 and 5 is available for operation in shallow waters, in heavy weather, and during acceleration, that is for non-steady operation without any strict time limitation.

The area between lines 4, 5, 7 and 8 is available for overload operation for 1 out of every 12 hours.

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

In calm weather conditions, the extent of heavy running of the propeller can indicate the need for cleaning the hull, and/or polishing the propeller.

Passage of a barred speed range

If the engine and shaft line has a barred speed range (BSR), it is usually a class requirement that it can be passed quickly. The quickest way to pass the BSR is the following:

1. Set the rpm setting to a value just below the BSR
2. Wait while the ship accelerates to a ship speed corresponding to the rpm setting
3. Increase the rpm setting to a value above the BSR.

Sometimes, for example in certain manoeuvring situations inside a harbour, or at sea in adverse conditions, it may not be possible to follow the procedure for passing the BSR. Either because there is no time to wait for the ship speed to build up, or because high ship resistance makes it impossible to achieve a ship speed corresponding to the engine rpm setting. In such cases, it can be necessary to pass the BSR at low ship speed.

The most basic guidance on avoiding slow passing of the BSR is to avoid a BSR that extends higher than 60% engine rpm, while specifying a light running margin within the recommendation. If so, it is normally possible to achieve a sufficiently quick passage of the BSR in relevant conditions.

A more detailed approach is to ensure a BSR power margin, BSR_{PM}, of at least 10% in the design.

$BSR_{PM} = ((P_L - P_P)/P_P*100$

Here, P_P is the power required by the bollard pull propeller curve at the upper end of the BSR, whereas P_L is the power limit for continuous operation within the engine load diagram, line 4 in Fig. 2.03.02. As such, the BSR_{PM} expresses the excess engine power in the upper range of the BSR, and hereby the ship's capability to pass it.

If a ship faces challenges with passing the BSR, a special function termed the "dynamic limiter function" (DLF) may be specified for diesel cycle engines. The DLF can for shorter periods increase the index limits of the engine, and ensure a faster passage of the BSR.

For 5- and 6- cylinder engines with short shaft lines, such as on many bulkers and tankers, the BSR may extend high up in the rpm range. Special attention must be given to ensure that the BSR can be passed quickly. 5- and 6- cylinder engines are as standard delivered with the DLF functionality.

For support regarding passage of the BSR, contact MAN Energy Solutions, Copenhagen at MarineProjectEngineering2s@man-es.com.

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Shaft generator/PTO layout, governor stability and integration

For the dimensioning and layout of a two-stroke propulsion plant with power take-off (PTO)/shaft generator, two aspects are important to consider for the design:

- PTO layout limit: prevention of engine overload
- Governor stability evaluation: prevention of engine speed hunting, and potentially also engine overspeed

Both aspects and their interactions will be clarified in detail in the following sections. See the flowchart in Fig. 2.03.03 for an overview of the considerations needed for a specific plant.

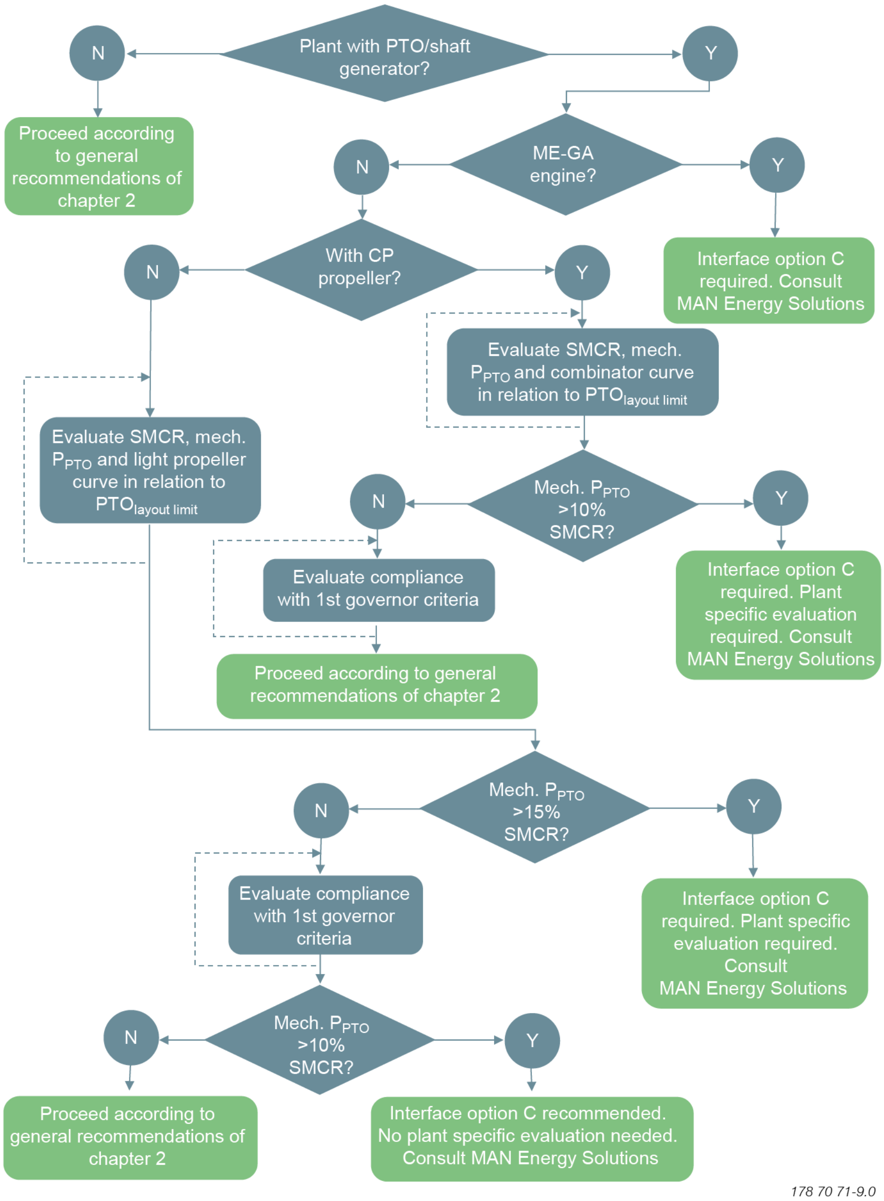

Fig. 2.03.03: Flowchart for determining evaluations needed, depending on plant characteristics

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2.03 Engine layout and load diagram

It is a prerequisite for a successful governor stability evaluation that the criteria of the PTO layout limit is fulfilled, and conversely. It is important to consider both aspects as illustrated by the flowchart. At the end of this section, considerations for the protection of the engine towards overload in service is given.

PTO layout limit

In the load diagram in Fig. 2.03.02, line 10 represents the "$PTO_{layout\ limit}$". Line 10 describes the maximum combined power required by the light propeller curve and the PTO at a given speed, *n*, when a shaft generator/PTO is installed. The minimum value of the following three equations governs the "$PTO_{layout\ limit}$". Observe the speed interval for which the second equation is applicable:

$$\mathrm{PTO}_{\mathrm{layout\ \ limit}} = \begin{cases} P_{SMCR} \times \left(\dfrac{n}{n_{SMCR}}\right)^{2.4}, & \text{for } n \in 50\% \text{ to } 96.2\% \text{ of } SMCR \\ P_{SMCR} \times 0.95 \times \left(\dfrac{n}{n_{SMCR}}\right), & \text{for } n \in 96.2\% \text{ to } 100\% \text{ of } SMCR \\ P_{SMCR} \times 0.95, & \text{for } n > 100\% \text{ of } SMCR \end{cases}$$

As marked in Fig. 2.03.02, the maximum design PTO power at a given speed is the vertical difference between line 6 (the light propeller/combinator curve of a propeller) and line 10 (the $PTO_{layout\ limit}$). PTO operation is not possible below 50% of SMCR-speed. Table 2.03.01 shows the relative PTO power available when sea conditions allow operation along the light propeller curve. At engine speeds above 50% of SMCR, the relative PTO power is given as a function of the light running margin.

Designing the combined power of the PTO and propeller according to the $PTO_{layout\ limit}$ ensures that the PTO can be operated in conditions less ideal than sea trial conditions. Note that neither the torque/speed limit (line 4) nor the MEP limit (line 5) is used for the layout of the PTO capacity.
With increased heavy running, the electric power taken off with the PTO must be decreased gradually not to push the operational point outside the engine limits. In severe cases, fouling and sea conditions alone are enough to shift the propeller curve to line 4. It these cases, the PTO cannot be utilised without overloading the engine, and the auxiliary engines must deliver all the electric energy.

It can be beneficial to increase the SMCR power and/or the light running margin for ships with a large electrical consumption, which often operate at high speeds/engine loads, or in areas with frequent encounters of adverse weather conditions. This will increase the margin from the light propeller curve to the $PTO_{layout\ limit}$ and ensure a higher availability of the PTO.

Increasing the SMCR power of the engine, while maintaining a constant propeller pitch, results in an increase of the light running margin. See Table 2.03.01 at the end of this section. When evaluating the possibility for an increase of SMCR power, compliance with EEDI regulations must be considered. See the application examples in main section “Examples of the use of load diagrams”.

For information on PTO power available at an given propeller speed as a function of the propeller light running margin, see the later section "PTO layout table".

Governor stability for plants with PTO

A PTO connected to the electric grid will deliver constant power, and introduce negative damping of the engine speed. If a vessel encounters, for example, large waves, the speed through water and the engine speed will drop. Therefore, the frequency drive will load the PTO with a higher torque to deliver the same electrical power. The torque increase enhances the speed drop experienced, and conversely for a speed increase.
To ensure a stable engine speed during operation, MAN Energy Solutions has established guidelines on governor stability for the maximum PTO power allowed.
These guidelines must be considered along with the PTO layout limit. It means that the maximum allowable PTO power to design for at any engine speed is determined as the minimum of the two aspects.

First governor stability criteria

The first governor stability criteria depicted on Fig. 2.03.04 represents a minimum with respect to governor stability at lower speeds. The criteria are valid for plants where the mechanical power of the PTO does not exceed 15% SMCR power for fixed pitch propeller (FPP) plants and 10% SMCR power for controllable pitch propeller (CPP) plants.

Fig. 2.03.04 shows three limits, which establish the first governor stability criteria for:

- FPP plants (blue solid curve)
- CPP plants, and engines with bore sizes smaller than 80 cm (orange solid curve)
- CPP, and 80-bore engines or larger (grey solid curve)

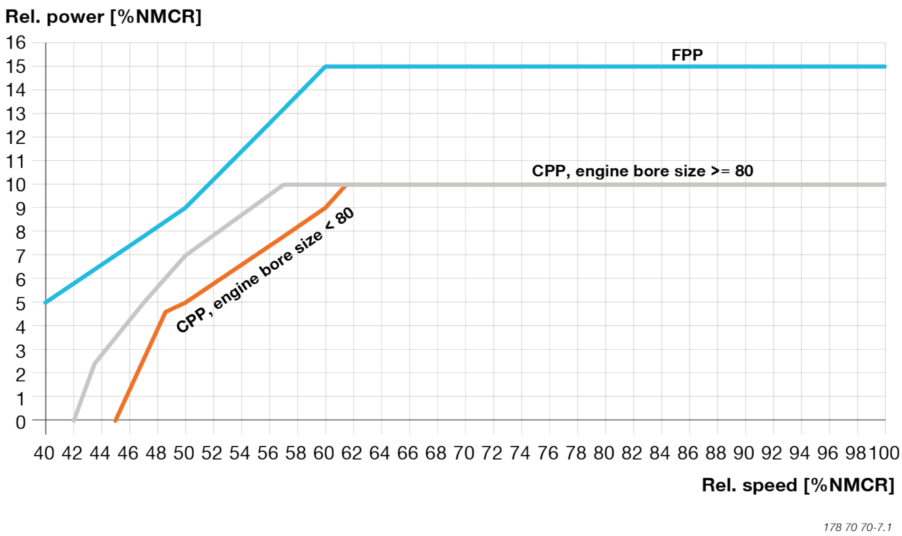

Fig. 2.03.04: First governor stability criteria for maximum mechanical PTO which ensure an acceptable governor performance and stability without interface option C. The limits relate to NMCR and are independent of the choice of SMCR

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2.03 Engine layout and load diagram

FPP		CPP < 80		CPP >= 80	
[%NMCR]	[%NMCR]	[%NMCR]	[%NMCR]	[%NMCR]	[%NMCR]
100	15.0	100.0	10.0	100.0	10.0
60	15.0	61.5	10.0	60.0	10.0
57	13.2	60.0	9.0	57.0	10.0
50	9.0	57.0	7.8	50.0	7.0
47	7.8	50.0	5.0	47.0	5.0
40	5.0	48.6	4.6	45.0	3.5
		47.0	2.6	43.5	2.4
		45.0	0.0	42.0	0.0

Table 2.03.01: First governor stability criteria for maximum mechanical PTO ensuring acceptable governor performance and stability without interface option C. The limits relate to NMCR and FPP plants, CPP plants and engines with bore sizes smaller than 80 cm, and CPP plants with 80-bore engines or larger

The limits in Fig. 2.03.04 are based on NMCR power and speed, not SMCR power and speed. This is because stability is related closely to the actual speed and the physical parameters of the engine, i.e., power and inertia, rather than the choice of SMCR.

MAN Energy Solutions must be consulted for a plant-specific PTO layout and design evaluation, if the maximum mechanical PTO load on the shaft is higher than:

- 15% of SMCR power for FPP plants,
- or 10% for CPP plants,
- or if the plant does not fulfil the first governor stability criteria.

Interface option C described below is mandatory for these plants.

For FPP plants, where the mechanical PTO power exceeds 10% of SMCR power, Interface option C is recommended.

PTO operation is in any case not possible below 50% of SMCR speed.

Interface option C - integration between power management and engine control system

For plants with a shaft generator more powerful than ordinary, Interface option C can be installed between the engine control system (ECS) and the power management system (PMS) to increase the maximum PTO power. This interface improves the integration of ECS and PMS, and enhances governor stability. A plant-specific evaluation is performed for each application of interface option C. See also the next sections for other benefits of Interface option C.

The plant-specific PTO layout and design evaluation may lead to changes in the control equipment. For example, an increase of signals from the plant and requirements to the design of engine-driven mechanical components in the form of turning and tuning wheels. The evaluation may also lead to changes in the use of the PTO or set restrictions for the rotational speed while taking out maximum power.

Fixed pitch propellers
For plants with FPP, interface option C is:

- Required if the mechanical power of the PTO exceeds 15% of SMCR
- Recommended if the mechanical power of the PTO exceeds 10% of SMCR

Application of Interface option C requires consultation of MAN Energy Solutions for a plant-specific PTO layout and design evaluation. By applying Interface option C, at least 20% of SMCR power is available for PTO within certain speed limitations. An even larger ratio may be available for PTO and can be investigated as part of a plant-specific PTO layout and design evaluation.

Controllable pitch propellers
For plants with CPP, further considerations are necessary regarding governor stability, as when the propeller pitch is reduced. For CPP plants, Interface option C is:

- Required if the mechanical power of the PTO exceeds 10% of SMCR.
- Recommended if the mechanical power of the PTO exceeds 5% of SMCR

Based on a plant-specific evaluation, specific limits may be set to the minimum engine speed at which manoeuvring can be performed while sustaining the maximum PTO power.

For plants, where the mechanical PTO power exceeds 10% of SMCR, a plant-specific evaluation includes the risk of overspeed if a total electric load loss occurs on the PTO during engine operation in constant speed mode.

Load sharing and overload protection in service

Designing according to the PTO layout limit does not protect the engine from overload during PTO operation in combination with a fouled hull, or an encounter of heavy seas. With the standard interfaces between the engine and remote control, it is the responsibility of the crew to balance the load between PTO and gensets to avoid overload of the engine during conditions with increased hull resistance.

Besides ensuring governor stability, interface option C provides signals for the PMS to automate load sharing between the main engine PTO and gensets. The extended interface will help ensuring higher utilisation rates of the PTO, thus reducing genset running hours. If supplying power solely by the PTO, it will also reduce the risk of blackout without overloading the engine.

For support regarding layout of PTO/PTI and plant-specific evaluations, contact MAN Energy Solutions, Copenhagen at MarineProjectEngineering2s@man-es.com.

AWC functionality

The AWC functionality is only available for single fuel diesel engines equipped with high-efficiency turbochargers. The AWC function is introduced for ME-C 10.5 and 9.7 engines. If the AWC function is installed, it can be activated by pushing the “Increase limitation”-button, found on all ME-C engines.

There is no limitation on the duration of engine operation in the area of the AWC function. As such, the increased power produced may be utilised when evaluating a ship designs compliance with IMO minimum propulsion power requirements.

Ice-classed ships are designed to operate in ice, and ice operation is therefore not an emergency running condition. The AWC functionality is therefore not applicable for compliance with ice-class power requirements, or similar requirements that the ship is designed for (not emergency). For ice-classed ships, the standard, or if selected, the rpm-extended load diagram, should be applied as usual.

Based on the same argument as for ice-classed ships, the AWC functionality does not increase the power available for PTO. The reason is that the operation of a PTO is not an emergency running condition. A PTO installation must still comply with the PTO layout limits, and the governor stability criteria.

As a countermeasure to the temperature increase from heavy running, the AWC functionality alters the cyclic process of the combustion by changing the fuel injection timing and the exhaust valve timing. This improves thermal conditions in the combustion chamber at a cost of an increased SFOC. The SFOC penalty depends on the specific load conditions. Due to the increased SFOC, the AWC functionality should not be considered a replacement for an adequate light running margin.

When the engine is not running heavier than the normal load diagram, the AWC functionality has no effect and does not affect the SFOC or emissions.

For ships frequently operating in adverse weather conditions, an increased light running margin combined with an extended load diagram will ensure a lower SFOC during (such) encounters of adverse weather than the AWC function. See the later section about extended load diagrams. For specific enquiries towards the availability of the AWC functionality, contact MarineProjectEngineering2s@man-es.com. For specific projects applying AWC, send the torsional vibration calculation (TVC) to MAN Energy Solutions, Copenhagen at RDCPH@man-es.com.

Rpm-extended load diagram

When a ship with fixed pitch propeller operates in normal sea service, it will in general operate around the design/light propeller curve (line 6) as shown on the standard load diagram in Fig. 2.03.05.

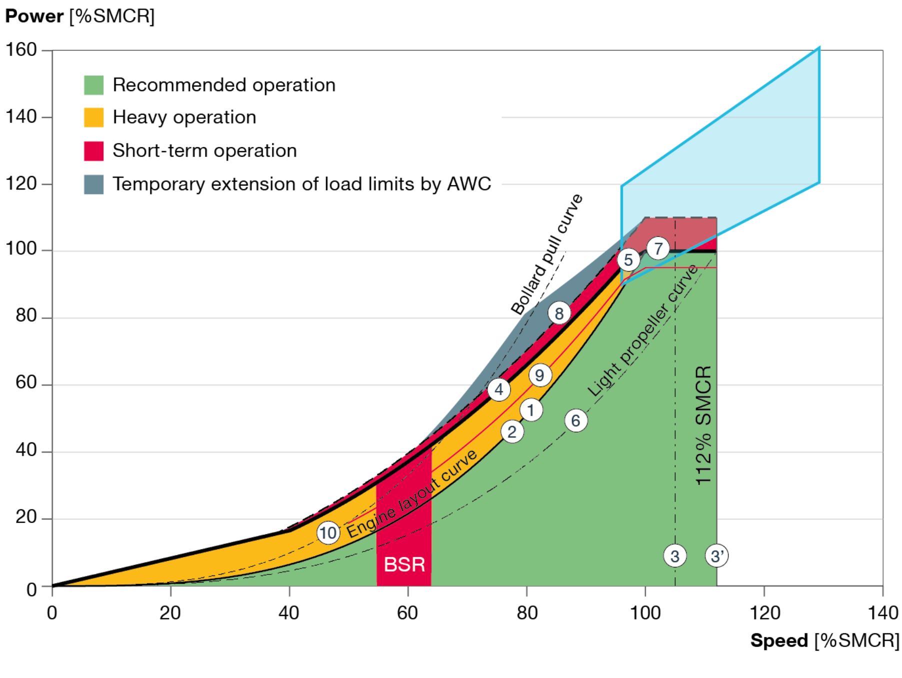

Fig. 2.03.05: Rpm-extended load diagram for a speed derated engine with an extreme increase of the light running margin

Sometimes, when operating in heavy weather or ice, the performance of the fixed pitch propeller will be characterised as more heavy running. 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 a fixed pitch propeller 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. It requires that the light running margin is within recommendations, see Fig. 2.03.02.

For some ships and operating conditions, it would be an advantage – when occasionally needed – to have a maximum margin for the torque increase from the light propeller curve (line 6) to the torque/speed limit (line 4).

If the vessel has a fixed pitch propeller which requires a high light running margin, an rpm-extended load diagram is relevant. Torsional vibration conditions must permit this, and the classification society in question must approve the solution.

The high light running margin, and rpm-extended load diagram is especially relevant when at least two of the listed cases apply to the ship:

- Sailing in areas with frequent encounters of heavy weather, especially for low-powered ships with blunt bows
- Sailing for long periods in shallow or otherwise restricted waters
- A high ice class
- Two fixed pitch propellers/two main engines, where one propeller/one engine is declutched/stopped for some reason
- Large electric loads and according to PTO capacity.

See the examples in the following section about application of the rpm-extended load diagram.

Combinator curves for CPP propulsion plants

In principle, a controllable pitch propeller (CPP) can load the engine in any point within the load diagram. It means that the engine can operate along a combinator curve with optimised pitch settings and propeller speed, making it possible to operate the total propulsion system with optimum efficiency.

There are three modes for operating a ship with a CPP, as reflected in Fig. 2.03.06:

- Constant engine speed (generator mode) – red line
- Fixed combinator curve – black curve (6)
- Adaptive combinator curve - this mode continuously adapts/controls pitch and rpm, typically based on a combinator curve

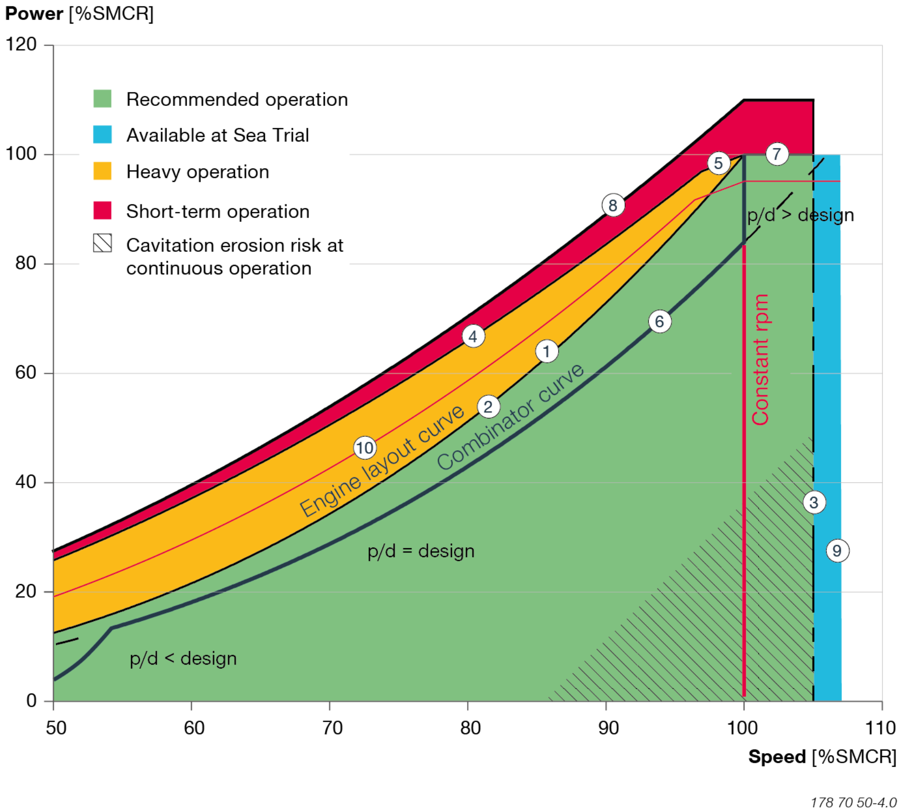

Fig. 2.03.06: Combinator curve and engine load diagram. The constant rpm curve can also be referred to as the generator curve. The exact speed of the generator curve will depend on the gear ratio of the PTO

Recommendations will be given regarding:

- Two-stroke engines coupled directly with a CPP without PTO
- Additional recommendation for two-stroke engines coupled directly with a CPP and PTO

Controllable pitch propeller operating without PTO

Constant engine speed mode

For plants without PTO, the constant speed mode is typically used for, but not limited to, manoeuvring as the ship approaches port. Operating the engine at high speed and low pitch ensures the maximum margin to the load limitations of the engine and the fastest response to any load change.

It is recommended avoiding continuous operation in the low-load area of the engine and at high propeller speed close to SMCR speed, due to:

- Potential risk of erosive pressure side cavitation
- Relatively higher losses of propeller and engine

Fixed combinator curve

A fixed combinator curve implies that the engine operates at a certain speed and the CPP load controller sets an associated pitch, depending on the setting of the machinery telegraph. When operating along a combinator curve, the shaft speed is reduced at lower loads. This will reduce the losses of the propeller and the engine at lower loads and increase the propulsion plant efficiency.

Fig. 2.03.07 shows a typical fixed combinator curve, which consists of two constant speed parts and one combinator part

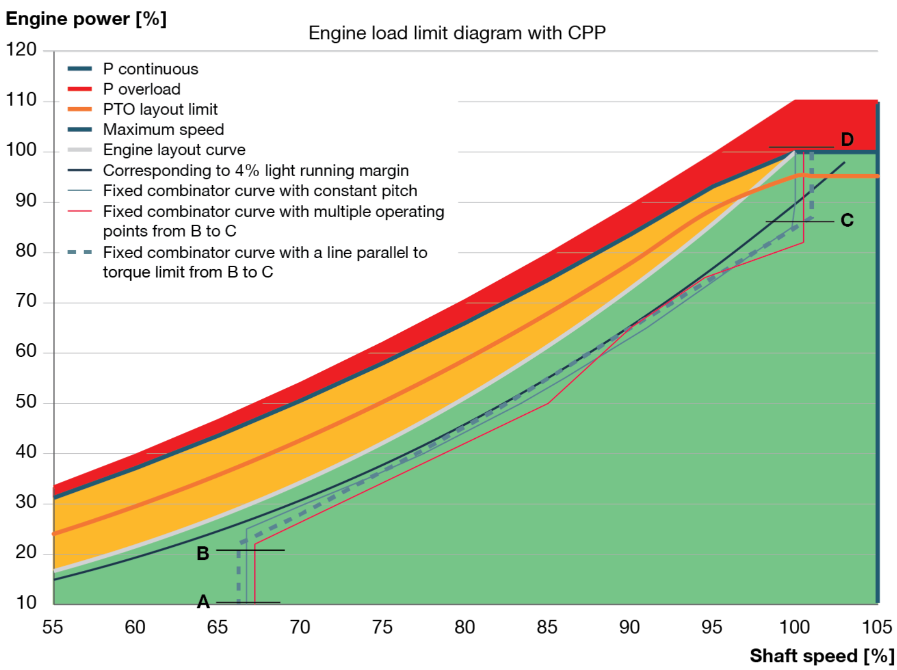

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Fig. 2.03.07: Typical fixed combinator curves for controllable pitch propellers without PTO

The minimum constant speed part (point A to B) is typically placed after the upper end of a barred speed range (BSR), if any. The maximum constant speed part (point C to D) with an increased pitch is typically laid out as the

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SMCR speed, so only the pitch is increased to attain 100% load. However, up to 105% of the SMCR speed is available, as for an engine connected to a fixed pitch propeller (FPP).

The combinator part (point B to C), is the part of the combinator curve that connects the constant speed parts. Usually, the propeller design point is within the range of the combinator part. The pitch can be constant along this part, or it can follow a preset combination of speeds and different pitch settings.

During the encounter of heavy weather, or fouling of the hull, the response from a fixed combinator curve is similar to that of an FPP propeller curve.

The torque required from the propeller increases when the resistance of the hull increases, which leads to a heavier running engine and higher SFOC. The result is higher thermal loading of the engine. To achieve an adequate margin, it is recommended that any arbitrary point along the fixed combinator curve follows the recommendation of the FPP light running margin (LRM). This margin is 4–7%, and in special cases up to 10%, except for the pitch-in at SMCR speed.

When reaching the engine limits for continuous operation, the propeller pitch must be reduced, if the engine speed is not to be reduced.

Adaptive combinator curve

Typically, adaptive or dynamic combinator curves are based on fixed combinator curves, and they can be applied to vessels experiencing resistance variations.

For long-term operation of plants with an adaptive combinator curve/propeller pitch, it is recommended loading the engine no heavier than the engine layout curve (line 1 or 2).

Short-term loading of the engine beyond the engine layout curve and up to the limits of continuous operation (line 4) is available for acceleration, peak wave resistance, etc., without adjusting the pitch. When reaching the limit for continuous operation, the pitch must be reduced to maintain the engine speed.

The pitch adjustment capabilities of a CPP together with the adaptive combinator curve enable an operating curve where the engine is not operated heavier than given by the engine layout curve. Furthermore, the benefits are that the SFOC does not increase as a result of engine heavy running, and that the thermal load of the engine is reduced.

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Controllable pitch propeller operating with PTO

Constant engine speed mode (generator mode)

A constant engine speed is mainly relevant when operating a synchronous PTO connected to a grid without a frequency converter, or for manoeuvring.

For plants with a PTO exceeding 10% of the SMCR power, the margin towards engine overspeed during a total loss of shaft generator load at zero or low pitch should be considered. See the section on governor stability, and for further information contact MarineProjectEngineering2S@man-es.com.

Fixed combinator curve

For CPP propulsion plants with PTO and a fixed combinator curve, it is recommended designing the combinator curve so that the combined load of propulsion power and maximum PTO output is within the PTO layout limit (line 10 in Fig. 2.03.06) at any speed.

For any combinator curve, the margin between the propulsion power design points and the PTO layout limit should be large enough to cover the maximum PTO mechanical power within the PTO layout limit at any speed. See Fig. 2.03.08.

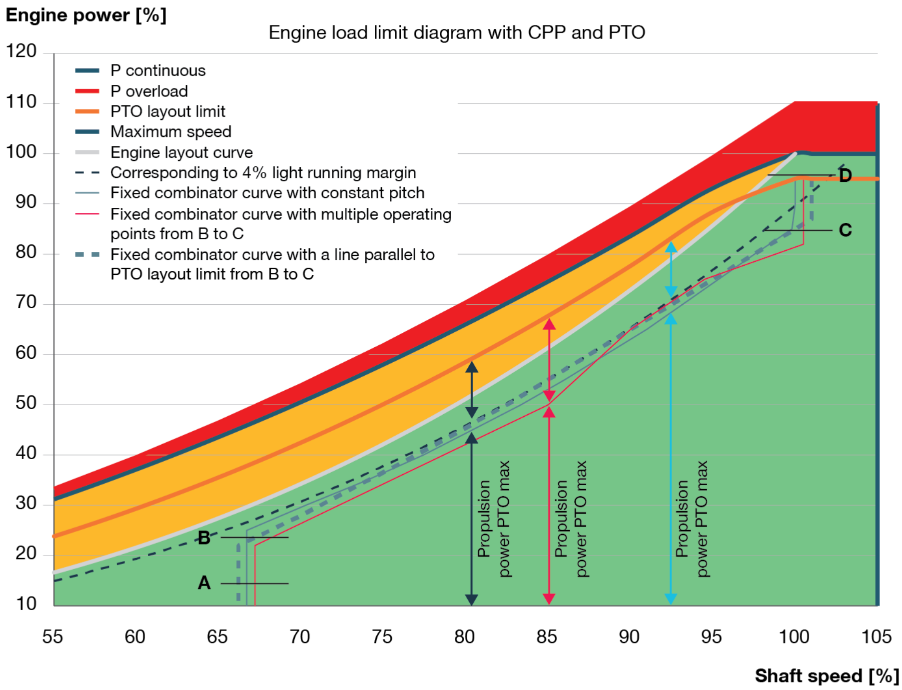

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Fig. 2.03.08: Typical fixed combinator curve for controllable pitch propellers with PTO

Although the power of the intended PTO is within the PTO layout limit for a combinator curve heavier than 4% LRM, the recommendation of a combinator curve corresponding to at least 4% LRM prevails, for any point on the combinator curve. Some CPP plants have a PTO capacity larger than required at sea because the main engine driven PTO supplies power to the thrusters during manoeuvring. In such cases, the maximum PTO power required at sea can be considered the margin between the combinator curve and the PTO layout limit (line 10). However, the general recommendation of 4–7% LRM, up to 10% in special cases, prevails.

Adaptive combinator curve

Assuming that the full PTO capacity at a given speed is not utilised for plants with an adaptive combinator curve/propeller pitch. In this case, it is recommended not to operate with a combined load of propeller and PTO, which is heavier than the engine layout curve (line 1, 2 in Fig. 2.03.06).

Short-term loading of the engine beyond the layout curve and up to the limits of continuous operation (line 4) is available for acceleration, start of heavy electric consumers, etc., without adjusting the pitch.

The dynamic capabilities of the CP propeller can reduce the level of heavy running by reducing the propeller torque to counter a torque increase of driving a PTO. This will reduce the SFOC not only for the power taken out for the grid, but also for the power used to propel the vessel.

Examples of the use of load diagrams

In the following, various examples illustrate the flexibility of the layout and load diagrams.

Example 1: Engine coupled to FPP without PTO

In this example, which represents a typical application of marine two-stroke engines, typical values of 7%, engine margin, and 15% sea margin are included. A light running margin in the higher range of the recommended 4–7% (up to 10% in special cases) is displayed for the sake of the example.

The maximum speed acceptable for an engine with a standard load diagram is the minimum value of 110% SMCR-speed or 105% L_1-speed. If torsional vibration conditions permit it, 107% of the SMCR-speed will be available for continuous operation, even if the light propeller curve extends beyond the layout diagram, see Fig. 2.03.09.

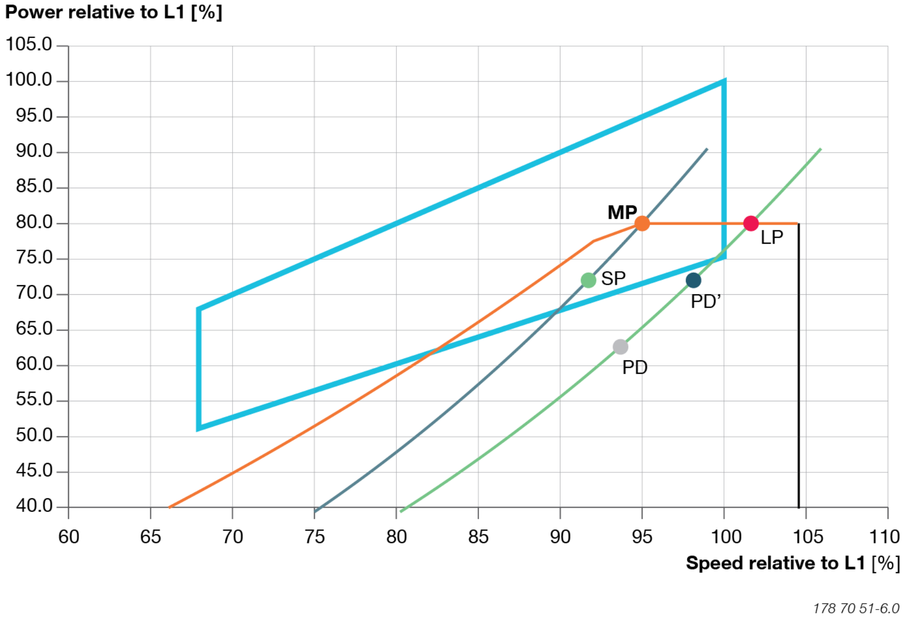

Fig. 2.03.09: Engine coupled to a fixed pitch propeller without shaft generator. The load diagram is the result of selecting the MP/SMCR within the layout area, 15% sea margin, 10% engine margin, and 7% light running margin.

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Example 2: Engine coupled to FPP with PTO

In this example, the SMCR is determined by using the regular method for determining PD and adding margins. Thereafter, it is investigated whether the desired PTO of 9% of the SMCR can be accommodated within the PTO layout limit.

As seen on Fig. 2.03.10, the power of the light propeller curve plus the power of the PTO lies well within the PTO layout limit (line 10 in Fig. 2.03.06), which is up to 102% of the SMCR speed (96% relative to L_1 on Fig. 2.03.10). At a shaft speed above this, the PTO output must be reduced.

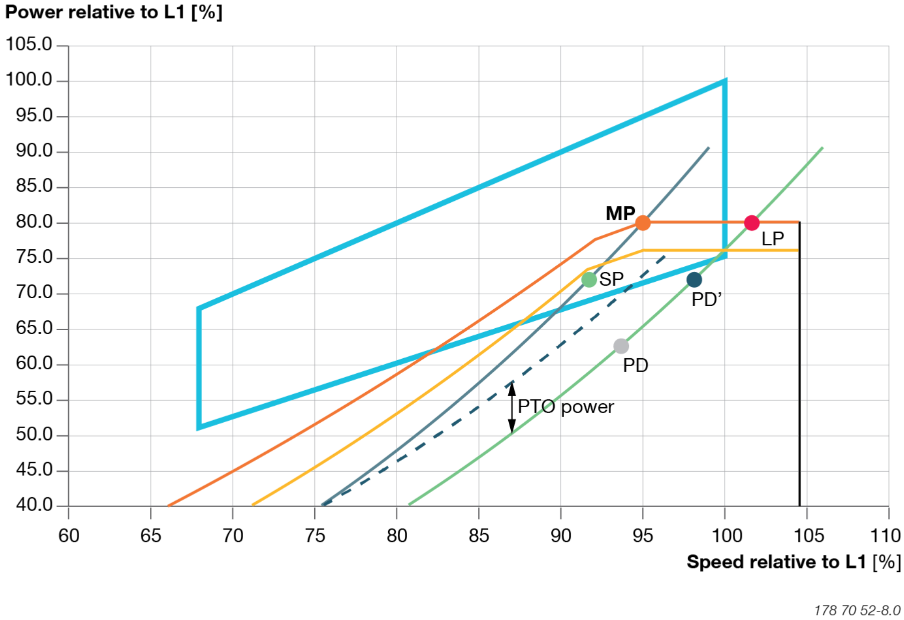

Fig. 2.03.10: Engine coupled to a fixed pitch propeller, and shaft generator. The load diagram is the result of selecting the MP/SMCR within the layout area, the PTO layout limit (line 10), and light propeller curve plus PTO power (dashed).

Since the PTO power is less than 10% of the SMCR power, only first governor stability criteria should be considered. There is no need for further considerations about the impact of the PTO towards governor stability. As an example of evaluating according to the first governor stability criteria, consider a 6S60ME-C10 engine and:

- NMCR (L1) of 14,940 kW at 105 rpm (80% of L1 as on Fig. 2.03.10)
- SMCR of 11,950 kW at 99.8 rpm (95% of L1 as on Fig. 2.03.10).

  For a mechanical PTO power of:

- 9% of SMCR, which corresponds to: 0.09 x 11,905 = 1,070 kW
- available in a speed range from 80 - 102% of SMCR
  which is: 80% x 95% = 76% to 102% x 95% = 97% of NMCR.

Absolute		Relative to SMCR		Relative to NMCR	
Speed [rpm]	Power [kW]	Speed [%]	Power [%]	Speed [%]	Power [%]
101.8	1070	102	9.0	97	7.2
79.8	1070	80	9.0	76	7.2
59.9	800	60	6.7	57	5.4

Table 2.03.01: Values of PTO realtive to an SMCR of 11950 kW at 99.8 rpm relative to NMCR of 14940 kW at 105 rpm.

This is plotted relative to the first governor stability criteria, as this PTO power represents 7.2% of the power at NMCR. The PTO is available down to 60% of SMCR speed, and it delivers a constant torque between 60% and 80% of SMCR speed. It means that the PTO delivers a reduced power proportional to the speed reduction, compared to the speed at which the nominal PTO power is available: 60%/80% x 1070 = 800 kW. This corresponds to 5.4% of NMCR power and 60% x 95% = 57% of the NMCR speed. This is depicted in Fig. 2.3.11.

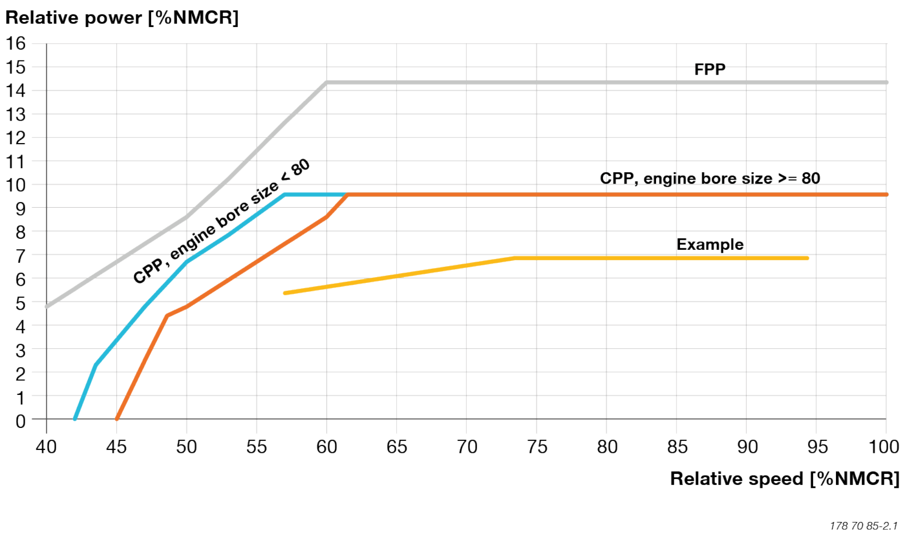

Fig. 2.03.11:

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2.03 Engine layout and load diagram

Example 3: Engine coupled to FPP with PTO, and increased light running margin

In this example, a PTO of 18% of the SMCR is desired, which represents a larger percentage of the SMCR power than considered in example 2.

To accommodate the larger PTO in a desired range of 80–100% of the SMCR-speed (75–95% on Fig. 2.03.12), the light running margin is increased to 9%. For the present SMCR located at 95% of the L_1-speed, a 9% light running margin is still within the limit given by the minimum value of 110% SMCR-speed, or 105% L_1-speed.

As the PTO power exceeds 15% of the SMCR power, interface option C between the power management system and the engine control system is a prerequisite for applying the PTO to ensure sufficient governor stability. A plant specific evaluation of the governor stability is part of the application of interface option C.

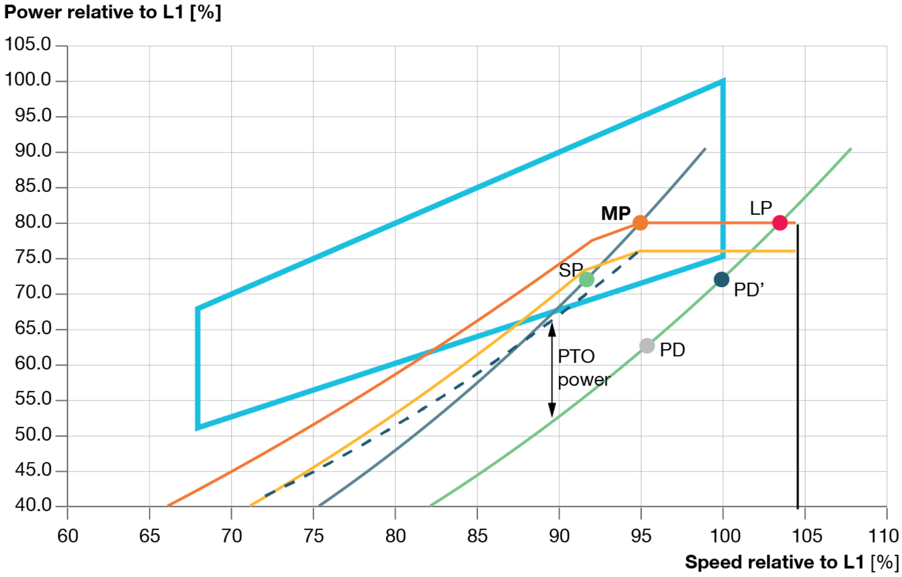

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Fig. 2.03.12: Engine coupled to a fixed pitch propeller and shaft generator. The load diagram is the result of selecting the MP/SMCR within the layout area, the PTO layout limit (line 10), and the light propeller curve plus the PTO power (dashed).

Example 4: Engine coupled to FPP with PTO, increased SMCR power, and rpm-extended load diagram

In this case, an increase of the PTO power to 24% of the SMCR power is considered. If considering the same absolute propeller curve as in example 3, the power cannot be accommodated within the PTO layout limit. For the sake of example, it is not desirable to increase the propeller light running margin further by decreasing the propeller pitch since it affects the propeller efficiency negatively.

To accommodate the higher power of the PTO, the SMCR power is increased by 7% while the SMCR-speed is maintained. This results in an engine that delivers a higher torque, see Fig. 2.03.13. The SMCR increase has the consequence that the propeller light running margin at 100% of SMCR power now corresponds to 11.5% - without changing the propeller pitch.

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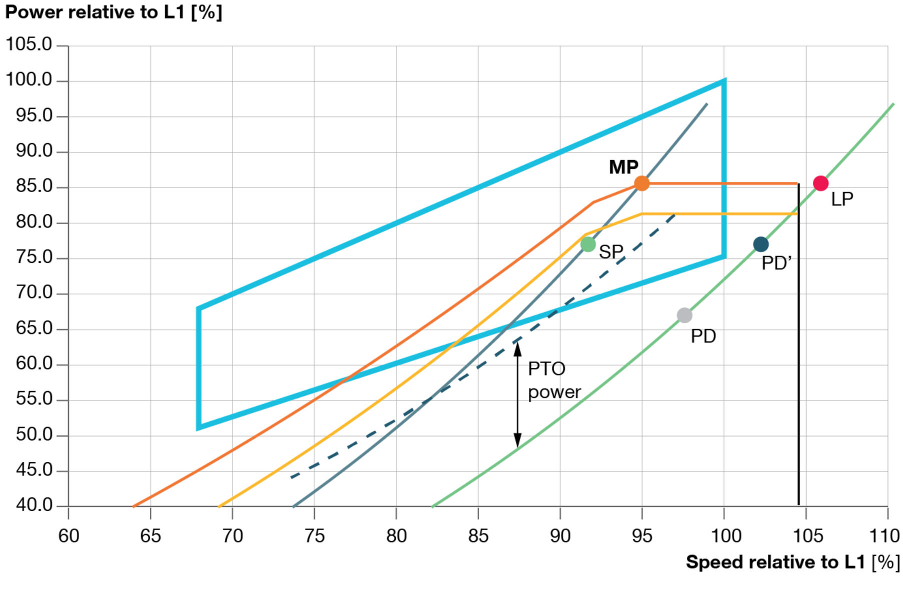

Fig. 2.03.13: Engine coupled to a fixed pitch propeller, and a very large shaft generator of 24% of SMCR. The load diagram is the result of selecting the MP/SMCR within the layout area, the PTO layout limit (line 10), and the light propeller curve plus the PTO power (dashed line).

In this example, the light propeller curve will deliver 100% power at 111.5% of SMCR speed, or 106% of L_1-speed. This is beyond the criteria of the minimum value of 110% of SMCR-speed, or 105% of L_1-speed.

For speed-derated engines, it is possible to extend the maximum speed limit to maximum 105% of the engine's L_1/L_2 speed (line 3 in Fig. 2.03.06), but only if the torsional vibration conditions permit this. Thus, with respect to torsional vibrations, the shafting has to be approved by the classification society in question, based on the selected extended maximum speed limit.

When choosing an increased light running margin, the load diagram area may be extended from line 3 to line 3', as shown in Fig. 2.03.05.

The increased light propeller curve (line 6), may have a correspondingly increased light running margin before exceeding the torque/speed limit (line 4).

In this example, the rpm extension of the load diagram will have limited effect. Relative to the SMCR, 105% of L_1-speed corresponds to 110.5%. Thereby, 100% power will not be available for continuous operation only by loading the engine with the propeller and with the hull as in sea trial condition.

For further speed-derated engines, the effects of the rpm-extended load diagram will be greater.

As for sea trial, 107% of L_1-speed is available for demonstrating full power, if torsional vibration conditions permits, see Fig. 2.03.02. In this example, it corresponds to 112.5% of the SMCR. If 100% power is to be available only for propeller load for continuous operation, the SMCR must be further speed derated, that is, the SMCR point will be moved further to the left in the layout diagram in Fig. 2.03.13.

As the PTO power exceeds 15% of the SMCR power, interface option C between the power management system and the engine control system is a prerequisite for applying the PTO to ensure sufficient governor stability. A plant specific evaluation of the governor stability is part of the application of interface option C.

Example 5: Engine coupled to CPP without PTO

If a controllable pitch propeller (CPP) is applied, the combinator curve (of the propeller) will normally be selected for a loaded ship including sea margin. For a given propeller speed, the combinator curve may have a given propeller pitch, and it may be heavy running in heavy weather like for a fixed pitch propeller.

Therefore, it is recommended using a light running combinator curve observing the recommendations for the light running margin for a fixed pitch propeller. For plants equipped with dynamic combinator curves adapting the propeller pitch continuously, it is recommended to seeking an operational point corresponding to the recommendations for the light running margin for fixed pitch propellers.

See Fig. 2.03.14 for a typical combinator curve corresponding to 5% light running margin along the constant pitch part of the combinator curve. Even without a PTO, some combinator curves are designed not to exceed 100% of the SMCR-speed, as represented by the dashed line. The engine does not prevent the combinator curve from continuing at constant pitch beyond 100% of the SMCR-speed. However, the general speed limits of the engine must be observed and torsional vibration conditions must permit it.

Sea and engine margins can in general be considered the same as for FP propellers. However, as the pitch can be reduced in an encounter of adverse weather conditions, there are no reasons for applying the AWC functionality.

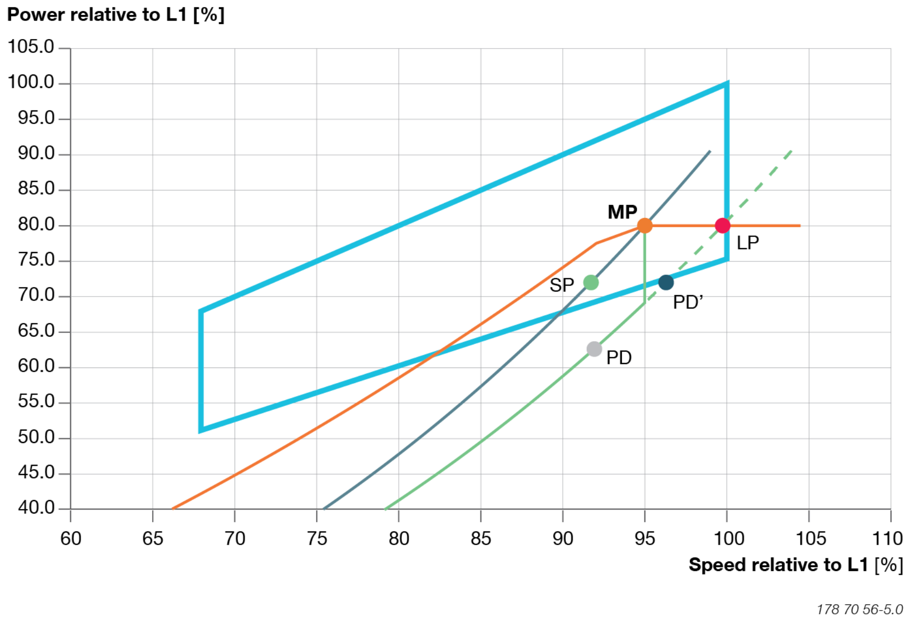

Fig. 2.03.14: Engine coupled to a controllable pitch propeller without shaft generator. The load diagram is the result of selecting the MP/SMCR within the layout area.

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Example 6: Engine coupled to CPP with PTO

In this example, a PTO of 15% of the SMCR is desired which requires a combinator curve corresponding to 7% light running margin to accommodate the PTO within the PTO layout limit, see Fig. 2.03.15.

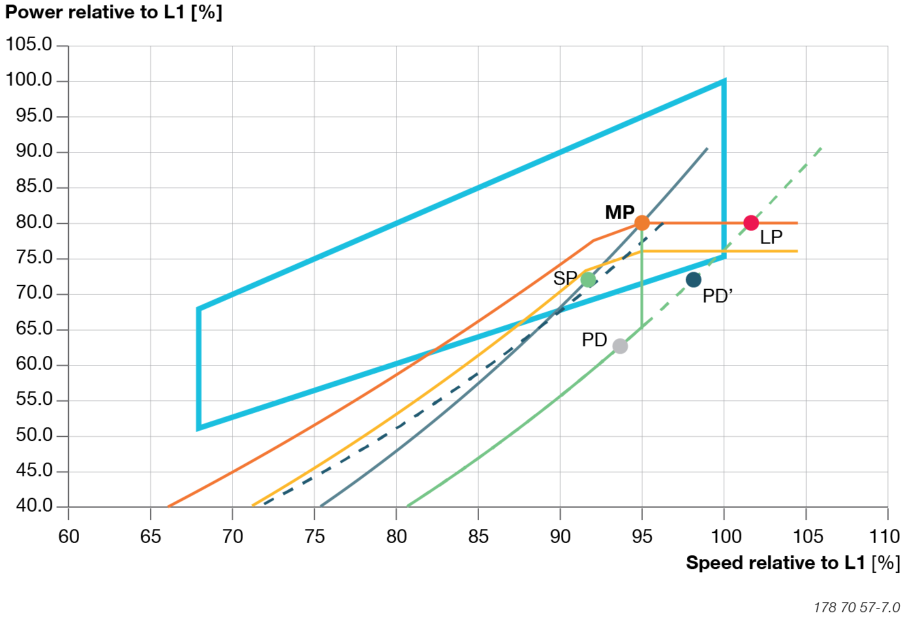

Fig. 2.03.15: Engine coupled to a controllable pitch propeller with a shaft generator corresponding to 15% of the SMCR power. The load diagram is the result of selecting the MP/SMCR within the layout area.

As for a fixed pitch propeller, the combined load of the combinator curve and PTO power must lie within the PTO layout limit.

Even if the pitch of the CP propeller can be reduced to accommodate the PTO, when full PTO power is needed, it may be an advantage to increase the SMCR power as in example 4, if high ratios of PTO power are to be available. Especially if full utilisation of the PTO power is foreseen for the major part of the operational time.

As the PTO power exceeds 10% of the SMCR power, interface option C between the power management system and the engine control system is a prerequisite for applying the PTO to ensure sufficient governor stability for a CPP plant. A plant-specific evaluation of the governor stability is part of the application of interface option C. For CPP plants, this evaluation also considers the margin against overspeed if a total load loss takes place on the PTO, while the propeller is at zero pitch. This scenario can take place during manoeuvring if the PTO drives the thrusters.

Contact Marine Project Engineering2S@man-es.com for enquires and assistance with the layout of the engine.

PTO layout table

Maximum (mechanical) PTO power [% of SMCR power] as a function of engine speed and propeller light running margin							
Engine speed [% of SMCR]	Propeller light running margin [%]						
	4%	5%	6%	7%	8%	9%	10%
50%	7.8	8.1	8.5	8.7	9.0	9.3	9.6
51%	8.1	8.4	8.7	9.0	9.3	9.6	9.9
52%	8.3	8.7	9.0	9.3	9.7	10.0	10.3
53%	8.6	8.9	9.3	9.6	10.0	10.3	10.6
54%	8.8	9.2	9.6	9.9	10.3	10.6	11.0
55%	9.0	9.4	9.8	10.2	10.6	11.0	11.3
56%	9.3	9.7	10.1	10.5	10.9	11.3	11.7
57%	9.5	10.0	10.4	10.8	11.2	11.6	12.0
58%	9.7	10.2	10.7	11.1	11.6	12.0	12.4
59%	9.9	10.4	10.9	11.4	11.9	12.3	12.8
60%	10.1	10.7	11.2	11.7	12.2	12.7	13.1
61%	10.4	10.9	11.5	12.0	12.5	13.0	13.5
62%	10.6	11.2	11.7	12.3	12.8	13.3	13.8
63%	10.8	11.4	12.0	12.6	13.1	13.7	14.2
64%	11.0	11.6	12.3	12.9	13.5	14.0	14.6
65%	11.1	11.8	12.5	13.1	13.8	14.4	14.9
66%	11.3	12.1	12.8	13.4	14.1	14.7	15.3
67%	11.5	12.3	13.0	13.7	14.4	15.0	15.6
68%	11.7	12.5	13.2	14.0	14.7	15.3	16.0
69%	11.8	12.7	13.5	14.2	15.0	15.7	16.4
70%	12.0	12.9	13.7	14.5	15.3	16.0	16.7
71%	12.1	13.0	13.9	14.7	15.5	16.3	17.1
72%	12.3	13.2	14.1	15.0	15.8	16.6	17.4
73%	12.4	13.4	14.3	15.2	16.1	16.9	17.8
74%	12.5	13.5	14.5	15.5	16.4	17.3	18.1
75%	12.6	13.7	14.7	15.7	16.6	17.6	18.4
76%	12.7	13.8	14.9	15.9	16.9	17.9	18.8
77%	12.8	14.0	15.1	16.1	17.2	18.2	19.1
78%	12.9	14.1	15.2	16.3	17.4	18.4	19.4
79%	13.0	14.2	15.4	16.5	17.7	18.7	19.8

Maximum (mechanical) PTO power [% of SMCR power] as a function of engine speed and propeller light running margin							
80%	13.0	14.3	15.5	16.7	17.9	19.0	20.1
81%	13.1	14.4	15.7	16.9	18.1	19.3	20.4
82%	13.1	14.5	15.8	17.1	18.3	19.5	20.7
83%	13.1	14.5	15.9	17.3	18.6	19.8	21.0
84%	13.1	14.6	16.0	17.4	18.8	20.0	21.3
85%	13.1	14.7	16.1	17.6	19.0	20.3	21.6
86%	13.1	14.7	16.2	17.7	19.1	20.5	21.8
87%	13.0	14.7	16.3	17.8	19.3	20.7	22.1
88%	13.0	14.7	16.4	18.0	19.5	21.0	22.4
89%	12.9	14.7	16.4	18.1	19.6	21.2	22.6
90%	12.8	14.7	16.4	18.1	19.8	21.4	22.9
91%	12.8	14.6	16.5	18.2	19.9	21.6	23.1
92%	12.6	14.6	16.5	18.3	20.0	21.7	23.4
93%	12.5	14.5	16.5	18.4	20.2	21.9	23.6
94%	12.4	14.5	16.5	18.4	20.3	22.1	23.8
95%	12.2	14.4	16.4	18.4	20.4	22.2	24.0
96%	12.0	14.2	16.4	18.4	20.4	22.3	24.2
97%	11.0	13.3	15.5	17.6	19.7	21.7	23.6
98%	9.4	11.8	14.1	16.3	18.4	20.4	22.4
99%	7.8	10.2	12.6	14.8	17.0	19.1	21.2
100%	6.1	8.6	11.0	13.4	15.6	17.8	19.9
101%	3.4	6.0	8.5	10.9	13.2	15.4	17.6
102%	0.7	3.3	5.9	8.4	10.8	13.1	15.3
103%	0.0	0.6	3.3	5.8	8.3	10.6	12.9
104%	0.0	0.0	0.6	3.2	5.7	8.1	10.5
105%	0.0	0.0	0.0	0.5	3.1	5.6	8.0
106%	0.0	0.0	0.0	0.0	0.5	3.0	5.5
107%	0.0	0.0	0.0	0.0	0.0	0.4	3.0
108%	0.0	0.0	0.0	0.0	0.0	0.0	0.4
109%	0.0	0.0	0.0	0.0	0.0	0.0	0.0
110%	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Table 2.03.01: Maximum (mechanical) PTO power (a percentage of SMCR power) as a function of engine speed, and propeller light running margin

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Load diagram for an actual project

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SFOC guarantee conditions

The specific fuel oil consumption (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.

The SFOC is based on fuels with lower calorific values (LCV) as specified in Table 2.05.01.

Fuel type (Engine type)	LCV [kj/kg]
Diesel	42,700
Methane (GI)	50,000
Ethane (GIE)	47,500
Methanol (LGIM)	19,900
LPG (LGIP)	46,000

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Table 2.05.01: Lower calorific values of fuels

For ambient conditions that are different from the ISO reference conditions, the SFOC will be adjusted according to the conversion factors in Table 2.05.02.

		With P_{max} adjusted	Without P_{max} adjusted
Parameter	Condition change	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, lower calorific value	per 1%	-1.00%	-1.00%

178 69 18-8.0.0

Table 2.05.02: Specific fuel oil consumption conversion factors

A 1°C increase of the scavenge air coolant temperature, results in a corresponding 1°C increase of the scavenge air temperature, and if pmax is unchanged SFOC increases 0.06%.

SFOC guarantee

The SFOC guarantee refers to the above ISO reference conditions, the lower calorific values, and is valid for one running point only.

The energy efficiency design index (EEDI) has increased the focus on partload SFOC. We therefore offer the option of selecting the SFOC guarantee at a load point in the range between 50% and 100%, EoD: 4 02 002.

All engine design criteria, for example heat load, bearing load and mechanical stresses on the construction, are defined at 100% load, independent of the guarantee point selected. This means that turbocharger matching, engine adjustment and engine load calibration must also be performed at 100% load, independent of the guarantee point. At 100% load, the tolerances are compensated for by matching, adjustment and calibration, the SFOC tolerance is 5%.

When choosing an SFOC guarantee below 100%, the tolerances will affect engine running at the lower SFOC guarantee load point. This includes tolerances on measurement equipment, engine process control, and turbocharger performance.

Consequently, the SFOC guarantee depends on the selected guarantee point, and it s given with a tolerance of:

Engine load (% of SMCR)	SFOC tolerance
100 - 85%	5%
<85 - 65%	6%
<65 - 50%	7%

Table 2.05.03: SFOC tolerance depending on engine load

Please note that the SFOC guarantee can only be given in one (1) load point.

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 and improve the engine performance.

When operating with 36°C cooling water temperature setpoint instead of, for example, 10°C (to the air coolers), the SFOC will increase by approx. 2 g/kWh.

With a lower cooling water temperature, the air cooler and water mist catcher will remove more water from the compressed scavenge air. This has a positive effect on the cylinder condition as the humidity level in the combustion gasses is lowered, and the tendency to condensation of acids on the cylinder liner is thereby reduced.

Derating for lower specific fuel oil consumption

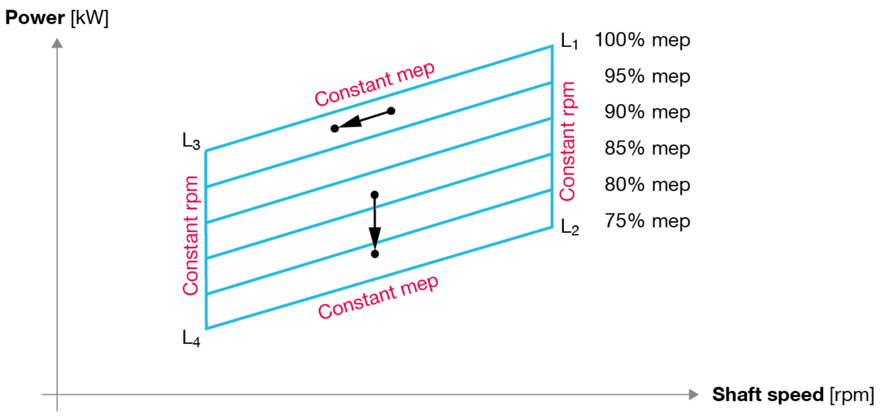

Fig. 2.05.01: Layout diagram showing MEP derating along L1-L2 (reduced SFOC) and power and speed derating along L1-L3 (SFOC is unchanged)

The ratio between the maximum firing pressure (P_{max}) and the mean effective pressure (MEP) is influences the efficiency of a combustion engine. If the P_{max}/MEP ratio is increased, the SFOC will be reduced.

The engine is designed to withstand a certain P_{max} and this P_{max} is utilised by the engine control system when other constraints do not apply.

The maximum MEP can be chosen in a range of values defined by the layout diagram of the engine, and it is therefore possible to specify a reduced MEP to achieve a reduced SFOC. This concept is known as MEP derating or simply derating, see Fig. 2.05.01.

If the layout point is moved parallel to the constant MEP lines, SFOC is not reduced, see Fig. 2.05.01.

Engine choices when derating

Due to requirements to ship speed and possibly shaft generator power output, derating is often not achieved by reducing MCR power. Instead, if allowed under the EEDI, a larger engine is selected to be able to choose a lower MEP rating, for example an engine of the same type but with an extra cylinder.

Derating reduces the overall SFOC level. The actual SFOC for a project will also depend on other parameters such as:

- Engine tuning method
- Engine running mode (Tier II, Tier III)
- Operating curve (fixed pitch propeller, controllable pitch propeller)
- Actual engine load
- Ambient conditions.

The actual SFOC for an engine can be found using the CEAS application available at www.man-es.com --> 'Planning Tools & Downloads' --> 'CEAS Engines Calculations'

It is possible to use CEAS to investigate the effect of derating for a particular engine by running CEAS for different engine ratings, for example the L_1 rating (not MEP derated) and the L_2 rating (fully MEP derated). This information can be used in the initial design work where the basic layout of the propulsion plant is decided.

Example of SFOC curves

Fig. 2.05.02 shows examples of SFOC curves for high-load tuning, part-load (EGB-PL) and low-load (EGB-LL) exhaust gas bypass tuning for a tier II engine operating with a fixed pitch propeller.

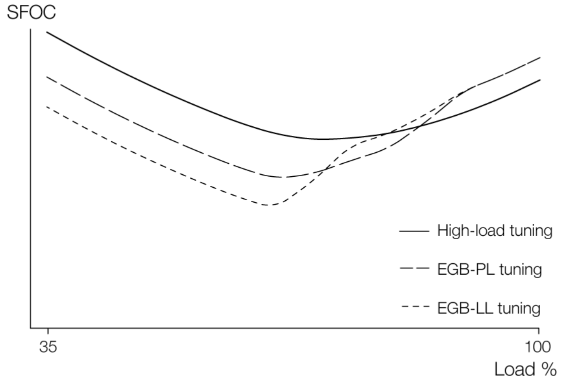

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Fig. 2.05.02: Influence on SFOC from engine tuning method and actual engine load

As an example, Fig. 2.05.02 illustrates the relative changes in SFOC due to the engine tuning method and the engine load. The graphs in this figure are only examples. Use CEAS to get actual project values.

Tier III engines do not offer the option for load tuning while in tier II mode, as the parameters controlling the combustion process are already fixed in order to meet both Tier II and Tier III demands.

Fuel consumption in an arbitrary operating point

Once the specified MCR (M) of the engine has been chosen, the specific fuel oil consumption in an arbitrary point S_1, S_2 or S_3, can be estimated based on the SFOC in points ‘1’ and ‘2’. See Fig. 2.06.01.

The SFOC values in points ‘1’ and ‘2’ can be found by using our CEAS application, for the propeller curve I and for the constant speed curve II, giving the SFOC in points 1 and 2, respectively. See section 20.02.

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.

The SFOC curve through points S_2, on the left of point 1, is symmetric with respect to point 1. It means that 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. This is a service which is available to our customers on request. Contact MAN Energy Solutions, Copenhagen at MarineProjectEngineering2S@man-es.com.

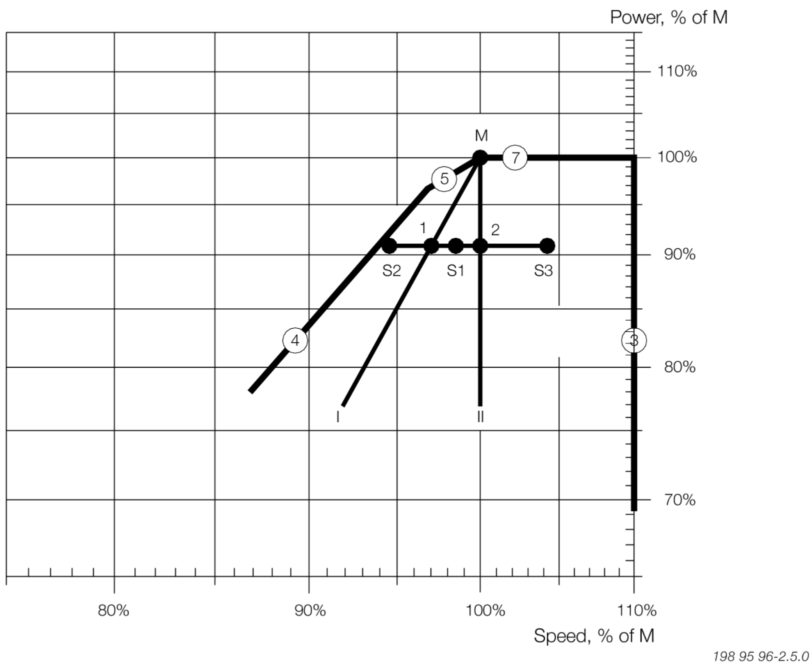

Fig. 2.06.01: SFOC at an arbitrary load

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Turbocharger selection

Updated turbocharger data based on the latest information from the turbocharger makers are available from the Turbocharger Selection program on www.man-es.com --> 'Turbocharger Selection'.

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

The MAN B&W engines are designed for the application of either MAN, Accelleron 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 Table 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’ program 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 G80ME-C10.5-LGIM engines - L_1 output			
Cyl.	MAN	Accelleron	MHI
6		1 x A280-L + 1 x A265-L	1 x MET71MBII + 1 x MET53MBII
7	1 x TCA88-24 + 1 x TCT50-ML	1 x A280-L + 1 x A270-L	1 x MET83MBII + 1 x MET53MBII
8			1 x MET83MBII + 1 x MET60MBII
9			1 x MET83MBII + 1 x MET60MBII

Table 3.01.01: High efficiency turbochargers

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3.01 Turbocharger selection

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Climate conditions and exhaust gas bypass

Extreme ambient conditions

As mentioned in Chapter 1, the engine power figures are valid for tropical conditions at sea level: 45 °C air at 1,000 mbar and 32 °C seawater, whereas the reference fuel consumption is given at ISO conditions: 25 °C air at 1,000 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.man-es.com → 'Marine' → 'Products' → 'Planning Tools and Downloads' → '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 bypass

Option: 4 60 118
Compensation for low ambient temperature can be obtained by using exhaust gas bypass 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 bypass is applied, the turbocharger size and specification has to be determined by other means than stated in this Chapter.

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Emergency Running Condition

Exhaust gas receiver with total bypass flange and blank counterflange

Option: 4 60 119

Bypass of the total amount of exhaust gas round the turbocharger is only used for emergency running in the event of turbocharger failure on engines, see Fig. 3.02.01.

This enables the engine to run at a higher load with only one turbocharger under emergency conditions. The engine's exhaust gas receiver will in this case be fitted with a bypass flange of approximately the same diameter as the inlet pipe to the turbocharger. The emergency pipe is yard's supply.

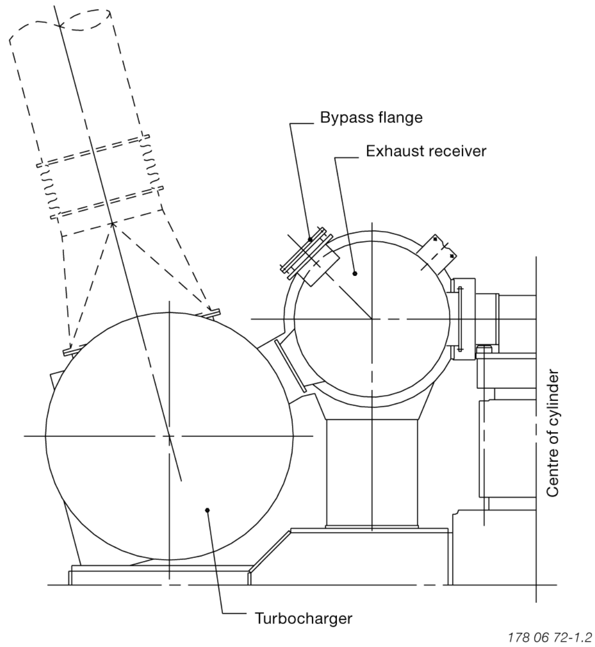

Fig. 3.02.01: Total bypass of exhaust for emergency running

Emission control

IMO Tier II NO_x emission limits

As standard, all ME, ME-B and ME-C/-GI/-GA engines fulfill the IMO Tier II NO_x emission requirements, a speed-dependent NO_x limit measured according to ISO 8178 Test Cycles E2/E3 for Heavy Duty Diesel Engines.

The E2/E3 test cycles are referred to in the Extent of Delivery as EoD: 4 06 200 Economy mode with the options: 4 06 201 Engine test cycle E3, or 4 06 202 Engine test cycle E2.

NO_x reduction methods for IMO Tier III

As adopted by IMO for future enforcement, the engine must fulfil the more restrictive IMO Tier III NO_x requirements when sailing in a NO_x emission control area (NECA).

The Tier III NO_x requirements can be met by exhaust gas recirculation (EGR), a method which directly affects the combustion process by lowering the generation of NO_x.

Alternatively, the required NO_x level could be met by installing a selective catalytic reduction (SCR) system, an aftertreatment system that reduces the emission of NO_x already generated in the combustion process.

ME-GA engines which operate on gas supplied at low pressure during the compression (Otto process) also comply with Tier III NO_x requirements. The engine is configured as an EGR engine which utilises the EGR process to prevent knocking and to suppress the methane slip. When operating in fuel oil mode, the engine operates according to the diesel process and applies the EGR for Tier II and III compliance.

Details of MAN Energy Solutions' NO_x reduction methods for IMO Tier III can be found in our publication:

Emission Project Guide

The publication is available at www.man-es.com 'Project Guides' -->'Other Guides'.

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04 Electricity Production

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Electricity production and hybrid solutions

Introduction

Hotel load and other electric consumptions are significant fuel consumers on a vessel, second only to propulsion power. It is consistently necessary to produce most, if not all, of the electricity on board due to the long voyages. The following machinery produces, running either alone or in parallel, the required electricity:

1. Auxiliary diesel or dual-fuel generating sets
2. Main engine driven generators
3. Exhaust gas or steam driven turbo generators using exhaust gas waste heat
4. Emergency diesel generating sets
5. Marine battery systems
6. Solar cells

The machinery installed should be selected based on the environmental impact, and an economic evaluation of first cost, operating costs, spare part costs, and the demand for work hours for maintenance.

The following sections give technical information about main engine driven generators (power take-off), different PTO configurations with exhaust gas and steam driven turbo generators, and auxiliary diesel generating sets produced by MAN Energy Solutions

Power take-off

A generator coupled to a power take-off (PTO) from the main engine enables production of electrical power based on the main engine's low specific fuel oil and gas consumption (SFOC/SGC). As of 2020, the use of a PTO solution has a positive effect on the attained EEDI, which can be evaluated according to IMO rules. The PTO maximum service power should be evaluated regarding two independent aspects:

1. Thermal overload of the engine and
2. Rpm stability/load change capability of the engine.

MAN Energy Solutions has developed guidelines to assist you with the layout of systems with PTO. See the next paragraph 'PTO maximum service power' to assess thermal overload. Refer to section 17.06 'Governor stability evaluation for special propulsion plants' for evaluation of the rpm stability/load change capability of the engine. Several standardised PTO systems are available; see the paragraph 'Designation' later in this chapter.

PTO maximum service power

This document refers to the electrical output of the PTO as PTO_E, the mechanical input to the PTO from the engine as PTO_M and the PTO efficiency as η_{PTO}. They are related according to the following law:

$$PTO_M = \frac{PTO_E}{\eta_{PTO}}$$

Fig. 4.01.01 shows an example of maximum PTO service power (not necessarily rated power) provided by the PTO maker, that is, how much power is available for the specific machinery in the speed range. P_M is the maximum selected power of the PTO, and n_M the maximum selected speed. P_M cannot coincide with the SMCR power of the engine, but n_M can, though it is not always the case.

The designed maximum service power must observe the guideline of MAN Energy Solutions. It means that the maximum service power must be within the boundaries given by the light propeller curve and the PTO layout limit, see section 2.03 ‘Engine Layout and Load Diagram’.

Due to the general shape of the PTO layout limit and the PTO characteristics, it is sufficient to verify each corner of the operating range with equation 1, equation 2, and equation 3 to avoid thermal overload. In Fig. 4.01.01, those corners are located at 52.5%, 70%, and 100% of the engine speed.

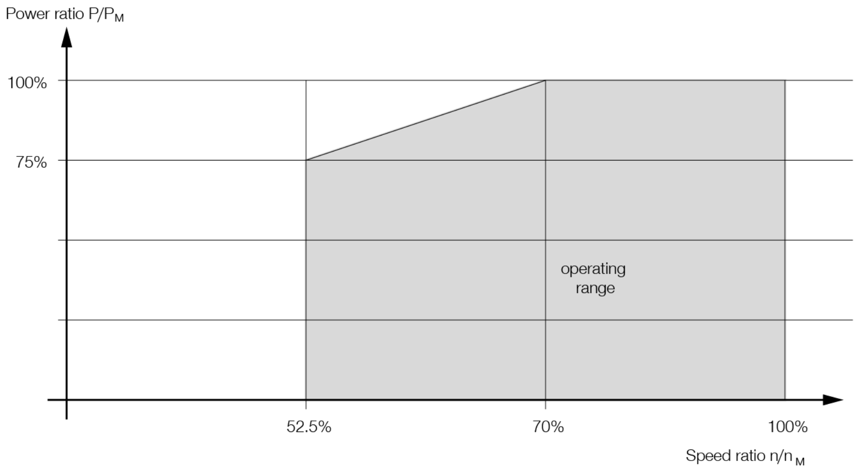

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Fig. 4.01.01: Example of PTO maximum service power

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$$PTO_M(n) \le P_{SMCR} \times \left[\left(\frac{n}{n_{SMCR}}\right)^{2.4} - \left(\frac{n}{(1+LRM\%)\times n_{SMCR}}\right)^3\right] \quad (1)$$

$$PTO_M(n) \le P_{SMCR} \times \left[1 - \frac{EMP\%}{100\%} - \left(\frac{n}{(1+LRM\%)\times n_{SMCR}}\right)^3\right] \quad (2)$$

$$PTO_M(n) \le P_{SMCR} \times \left[\frac{95\%}{100\%} \times \left(\frac{n}{n_M}\right) - \left(\frac{n}{(1+LRM\%)\times n_{SMCR}}\right)^3\right] \quad (3)$$

P_{SMCR} and n_{SMCR} is the power, and the engine speed of rotation at the SMCR point, respectively. n is the specific engine speed of rotation at which the mechanical PTO power is generated, LRM is the propeller light running margin, and EMP the engine margin for PTO operation (the minimum recommended margin is 5%).

Designation

A PTO system can be designed in different way. MAN Energy Solutions categorises a design according to two classifications: engine-to-generator and generator-to-grid.

The engine-to-generator classification relates to the position of the PTO system, and the connection between the engine and the system. The generator-to-grid classification relates to the frequency of the power fed to the grid, and the systems between the generator and the grid.

The generator-to-grid classification relates to:

1. The frequency of the power fed to the grid but also to
2. The type of/composition of the systems between the generator and the grid.

Two positions are available on the engine for the installation of a PTO system: the aft end (towards the propeller) and the front end. Side-mounted systems are currently not available. Front-end mounted generators can be mounted on-engine or on-tank-top.

They can be connected either using an elastic coupling or directly coupled to the crankshaft. Aft-end mounted generators are mounted either on the shaft or through a tunnel-gear. Fig. 4.01.02 illustrates the options. It is important to note that some of the PTO solutions might not be commercially available depending on engine bore size

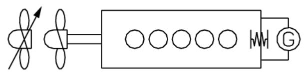
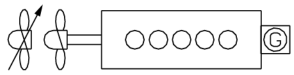
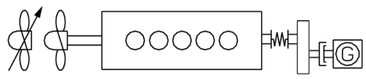
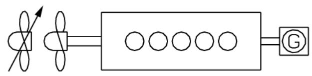
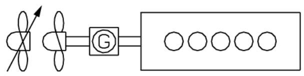
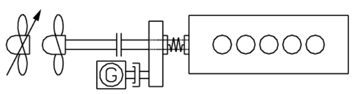
Position	Seating	Connection		Abbreviation
Front-end	On engine	Geared (with elastic coupling)		FEG
Front-end	On engine	Direct (no gear)		FED
Front-end	On tank top	Geared (with elastic coupling)		FTG
Front-end	On tank top	Direct (no gear)		FTD
Aft-end	Shaft-mounted	Direct (no gear)		ASM
Aft-end	Tunnel-geared	Geared (with elastic coupling)		ATG

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Fig. 4.01.02: PTO layout

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The fluctuations in the frequency of the output electric current generated by the shaft generator are proportional to the variation of the engine speed of rotation that occurs during normal operation. Most of the electric equipment cannot withstand large fluctuations of the current's frequency. As a result, the system is either limited to a range in which the PTO can be operated, or a frequency control system must be included.

Therefore, MAN Energy Solutions has defined two categories for PTO frequency types, or generator-to-grid systems (Fig. 4.01.03). The first category is systems with a frequency converter, and the second category is systems with synchronous frequency. The latter has a possible sub-designation of floating frequency (Fig. 4.01.04), which is further explained in ‘Floating Frequency System’.

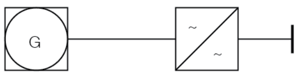
System		Abbreviation
Frequency Coverter		FC
Synchronous Frequency*		SF

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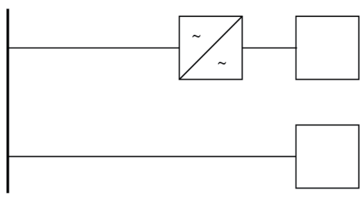
Fig. 4.01.03: PTO frequency type

System		Abbreviation
*Floating Frequency Possible system with synchronous frequency		FF

178 70 10-9.0.0

Fig. 4.01.04: PTO frequency type (optional)

On the design specification order (DSO), there are two spaces to fill in. The PTO type, which refers to Fig. 4.01.02 and the PTO frequency type, which consists of the three options displayed on Figs. 4.01.03 and Fig. 4.01.04.

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Example

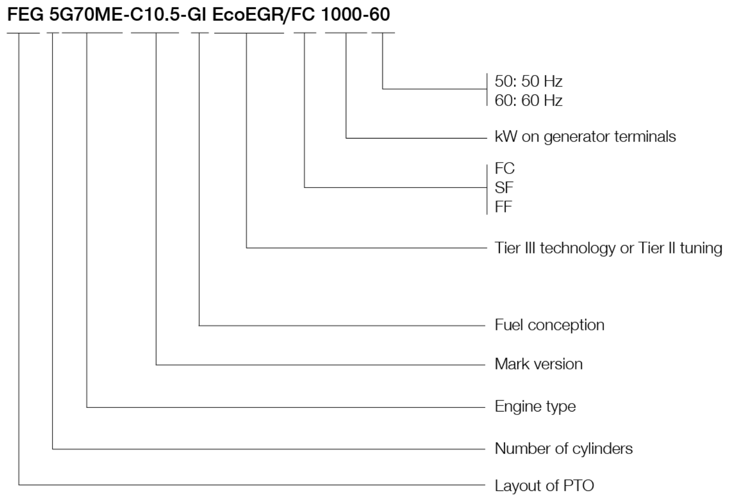

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Fig. 4.01.05: Example of designation

Similarity between old and new PTO designations

MAN Energy Solutions has implemented a new way of designating PTO systems. For this reason, Fig. 4.01.06 shows the relation between old and new designations. The figure also shows how RENK's systems are designated. Side-mounted PTO and RCF (RENK Constant Frequency) solutions are discontinued, and therefore no longer offered.

New designation	Old designation		RENK
FEG FC	Not available		IFPS
FEG SF	BW I	GCR RCF*	RCF*
FED FC	DMG	CFE	
FED SF	Not available		
FTG FC	Not available		
FTG SF	BW II	GCR RCF*	RCF*
FTD FC	Not available		
FTD SF	Not available		
ASM FC	SMG	CFE	
ASM SF	Not available		
ATG FC	Not available		
ATG SF	BW IV	GCR RCF*	RCF*
N/A	BW III*	GCR RCF*	RCF*

* Discontinued

Table 4.01.01: Equivalence between old and new PTO designations

Floating frequency systems

If most of the electrical equipment is suited for a supply with varying frequency, for example between 50 - 60 Hz, a PTO/SF configuration can be the economically and technically best solution. It consists of a constant gear ratio, that is, the frequency follows the engine speed. However, the PTO will be able to generate the required electricity within an engine load range of approximately 52% to 90% (80% to 97% of the SMCR speed). For the limited part of equipment, which requires a fixed frequency, a smaller frequency converter can be used.

Bulk carriers, tankers, and other vessels with low variations in cruise speed will obtain the following advantages:

- It is simple and thereby reliable
- Simple electrical system
- Highest possible efficiency (approximately 95%)
- Relatively cheap.
- Lower electrical power consumption at part load due to the lower speed. This will also function as an optimisation of auxiliary systems, since the required power automatically reduces at lower engine loads, whereas in a normal system, the power required by auxiliary systems is constant.

On the other hand, parallel running of an auxiliary engine and the PTO system is not possible with such systems, since it requires a frequency converter.

The installation of a PTO/SF system is restricted to a certain speed range, but most likely, a speed range can be chosen according to the most typical engine operating range. The electric equipment must be evaluated to establish the correct dimensioning. As an example, the main engine lube oil pumps must be dimensioned for the required flow at 50 Hz. Therefore, it is recommended to use a centrifugal lube oil pump.

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Power take-off solutions supplied by RENK

PTO maker and supplier, RENK, provides two specific solutions: An integrated front-end solution and a tunnel gearbox solution. These systems have different layouts and characteristics. This chapter uses these as examples of what is to be expected in the extent of delivery (EoD).

RENK IFPS© (Integrated Front-end Power System)

Fig. 1 shows the RENK IFPS. It features from one up to four generators connected to the crankshaft via a single-step gearbox. The high-ratio gearbox is designed for limited engine room space, and mounted on the engine fore end without an additional foundation.

The IFPS system has been developed in cooperation with the German gearbox manufacturer RENK. As standard, it is available for PTO powers of 500, 1000, 1500 and 2000 kW. The intermediate shaft is mounted directly on the crankshaft and its gearbox housing is bolted on to a strengthened front-end cover. The system is even compatible with a tuning wheel if required by torsional vibration conditions. As an option, an angle encoder can be mounted on the gearbox front side.

Extent of delivery for IFPS

- Gearbox incl. intermediate shaft
- Highly elastic coupling
- One to four generators
- IGBT active infeed frequency converter
- Transformer

1. Tuning wheel
2. Pinion
3. Generator
4. Gear wheel
5. High elastic coupling
6. Crankshaft
7. Intermediate shaft

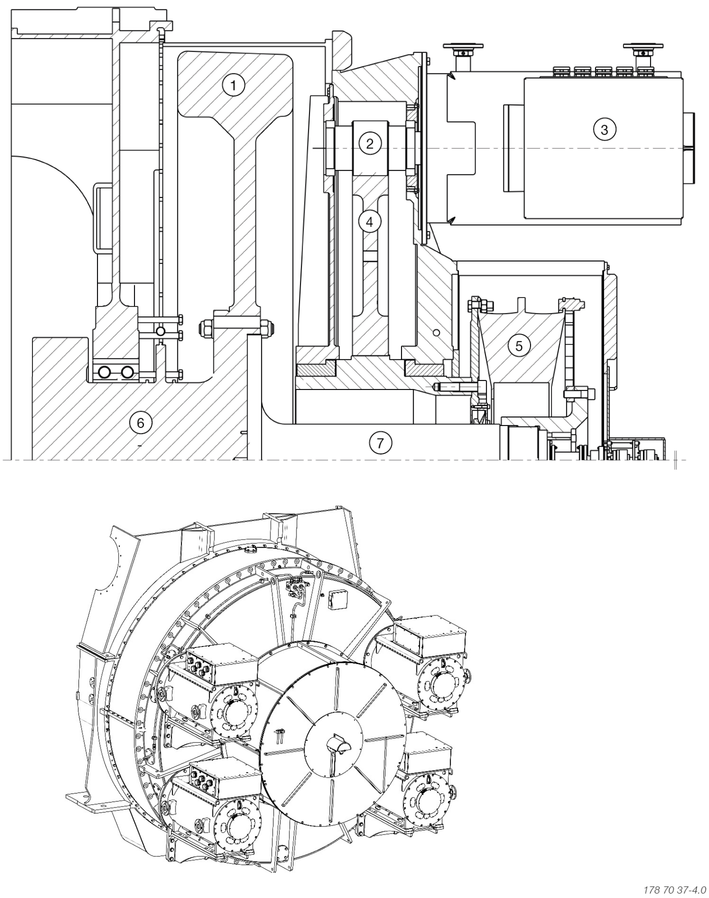

Fig. 1: Components of the IFPS

The IFPS is designed for operation with full output PTO-power between 100% and 70% specified speed, and with reduced power down to 50% of the engine speed at specified MCR (see Fig. 2).

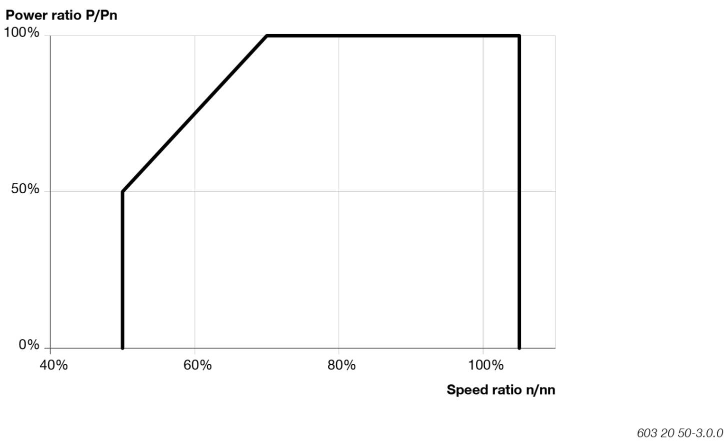

Fig. 2: Operating range of the IFPS

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An IGBT active infeed converter provides the constant grid frequency. The system can supply reactive power to the electric mains without a synchronous condenser, and causes negligible harmonics to the grid.

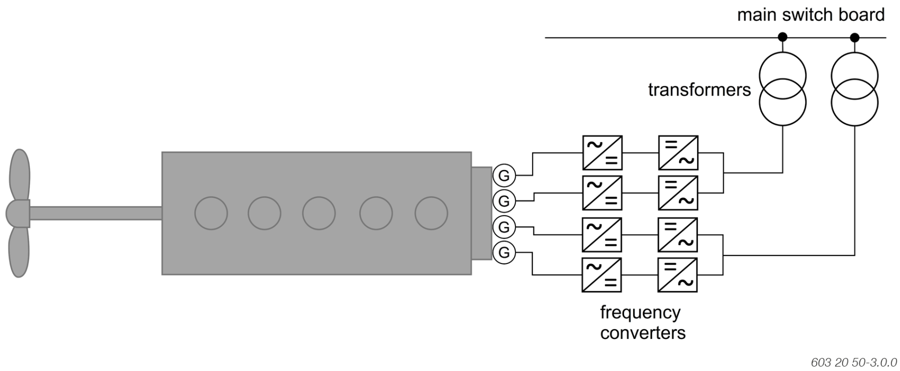

Fig. 3: Diagram of IFPS with active infeed converters and transformer

With a small modification, the IFPS can be operated both as a generator (PTO) and as a motor (PTI).

RENK MARHY© as auxiliary propulsion system/take-home system

Special applications require an auxiliary propulsion system/take-home system capable of driving the propeller by using the shaft generator as an electric motor.

MAN Energy Solutions can provide a solution where the propeller is driven by the alternator via a tunnel gearbox (RENK MARHY©). A number of gensets produce the electric power for the propulsion mode.

The main engine is disengaged by a clutch (RENK PSC) which is made as an integral part of the shafting. The clutch is installed between the tunnel gearbox and the main engine. Torque transmission will be switched by a hydraulically engaged backlash-free conical toothing. Equipped with a fully automatic PSC, the RENK MARHY© system can quickly establish auxiliary propulsion, and switch back to PTO operation from the engine control room and/or the bridge, even with an unmanned engine room. The built-in thrust bearing transfers the propulsion propeller thrust to the engine thrust bearing in both operating modes. In disengaged condition, a pair of sleeve bearings transmit the bending moments and shear forces within the propeller shaft line.

To minimise the required power for auxiliary propulsion mode, a two-speed tunnel gearbox can be used, which provides lower propeller speed, and thus higher propeller efficiency in auxiliary propulsion mode.

Extent of delivery for RENK MARHY©

Included in the MARHY© system are:

- PSC clutch
- Tunnel gearbox
- Highly elastic coupling for gearbox input and output
- Generator/motor
- IGBT active infeed converter frequency converter
- Transformer

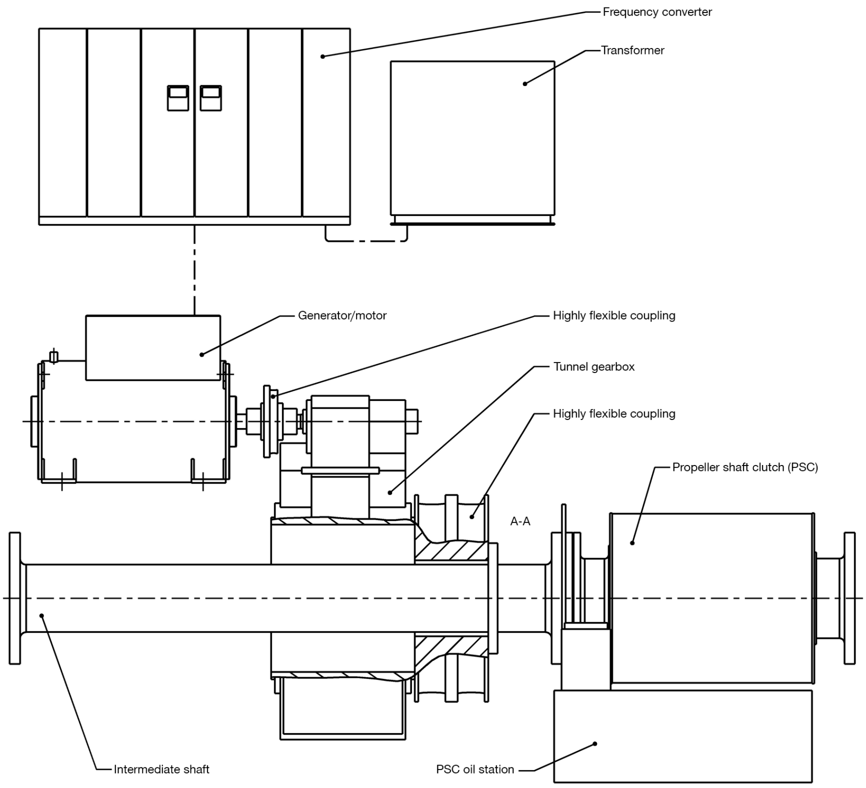

Fig. 4: Generic outline of the auxiliary propulsion system

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Steps for obtaining approval of a PTO solution

MAN Energy Solutions frequently receives enquiries about the different PTO solutions described in Section 4.01. To facilitate the process of getting a project approved, follow the necessary steps described here:

1. PTO layout guidance.

   The sizing of the PTO generator must adhere to the layout guide from MAN Energy Solutions. The guide stipulates the maximum PTO mechanical power from the main engine in relation to the SMCR chosen. Sections 2.03 and 4.01 describe this in detail.

2. Governor stability evaluation for special propulsion plants (17.06)

   When a PTO generator is connected to the ship's electrical grid, it becomes important to evaluate whether the power management system and the main engine can cope with the frequency converter's inhered negative damping on rpm stability. The sizing of the PTO generator must fulfil the criteria of MAN Energy Solutions to prevent insufficient stability of the system. Furthermore, MAN Energy Solutions can provide interface documentation for the connection between the main engine control system and the ship's power management system. For more information, see Section (17.06).

3. Torsional vibration.

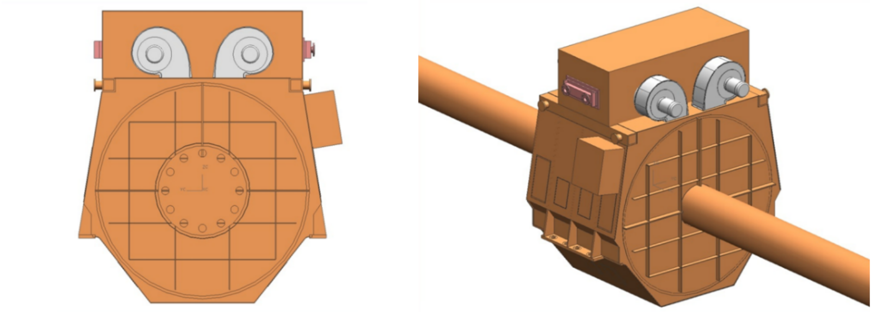
   The entire propulsion train: Main engine, shafting, propeller, and PTO, must fulfil class rules concerning torsional (and axial) vibrations. This must be evaluated in each individual case, because every case deviates from the other. If required, MAN Energy Solutions can assist with the analysis. Otherwise, it is normally a matter of shipyard responsibility. It might be required, sometimes that a torsional vibration damper is installed.

4. Specifically for main engine front-end mounted PTO solutions.

   For front-end mounted PTO solutions such as RENK IFPS or HHI EMG, the interface between the main engine front end and the PTO system must be identified to determine the scope of delivery together with that of the main engine, and the PTO solution. The result of the torsional analysis can involve installation of large tuning wheels and/or torsional dampers, which will affect the scope of delivery. This can necessitate the installation of an extra bearing on the main engine side to cope with the weight of these parts. Be aware that such cases differ from one another, and that MAN Energy Solutions can assist in the evaluation of whether direct mounted PTO solutions can be accepted or not.

We strongly recommend that shipyards involve MAN Energy Solutions in the design phase of a new ship with a PTO solution. It is important to choose the correct PTO size compared to the main engine size, and to comply with the IMO EEDI rules. Figs. 1–4 show examples of different PTO solutions.

Figs. 1–4 show examples of different PTO solutions.



178 70 34-9.0

Fig. 1: Intermediate shaft mounted generator. A solution mostly used on large container ships using high-power PTO generator sizes. Image of intermediate shaft mounted generator for the EEE class synchronous motor/generator (by Ccourtesy of Siemens).

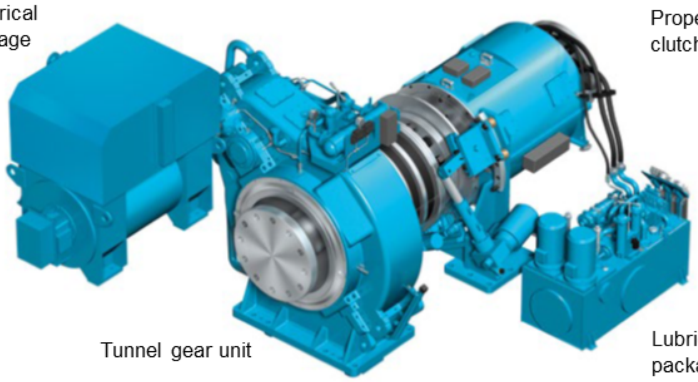


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Fig. 2: An example of a tunnel-geared PTO generator solution – RENK MAR-HY®. The solution is used for many different ship types, but in particular for ro-ro ships and container feeders (Courtesy of RENK).

RENK/Shipyard scope

1. Gearbox housing
2. Bull gear
3. PM generators
4. Intermediate shaft
5. Coupling cover
6. Highly flexible cover

Bolt connection between intermediate shaft and crankshaft

Bolt connection between intermediate shaft and crankshaft

MAN-ES/Engine maker scope

7. Front-end cover
8. Tuning wheel or TV damper
9. Crankshaft
10. M/E angle encoder

Bolt connection between crankshaft and tuning wheel or TV damper

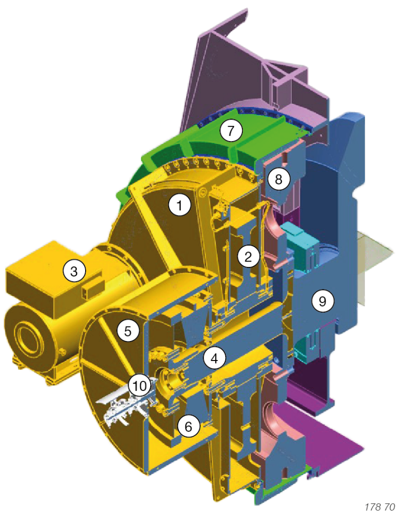

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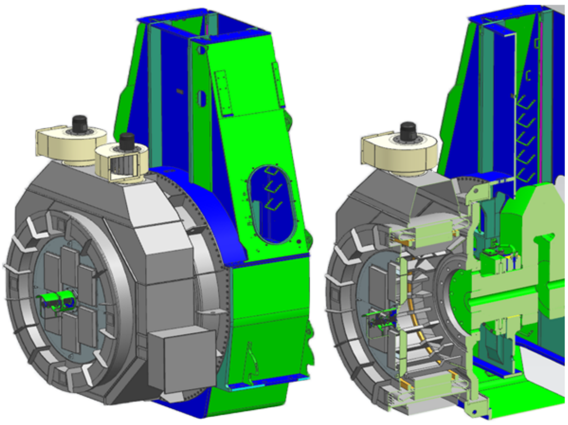
Fig. 3: Main engine front-end mounted PTO – RENK IFPS. The new PTO solution allows for a short engine room due to the short total length of main engine and PTO. The solution is particularly useful for tankers and bulk carriers (. Courtesy of RENK).



178 70 33-7.0

Fig. 4: Main engine front-end mounted PTO – HHI EMG (Engine Mounted Generator). The new PTO solution allows the maximum cargo capacity with its short installation space, especially for tankers, LNGC carriers, and bulk carriers (Courtesy of HHI).

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Power take off/gear constant ratio

This section is not applicable

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Waste heat recovery systems (WHRS)

General

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 Energy Solutions terms is designated as WHRS (Waste Heat Recovery Systems).

WHRS can be divided into different types of subsystems, 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 Energy Solutions 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. 3.5% 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.

The WHRS output depends on the main engine rating and whether service steam consumption must be deducted or not.

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 WHRS. The main engine will then be unloaded, or it will be possible to increase the speed of the ship, without penalising the fuel bill.

Because the energy from WHRS 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 152 g/kWh (ref. LCV 42,700 kJ/kg).

Power turbine generator (PTG)

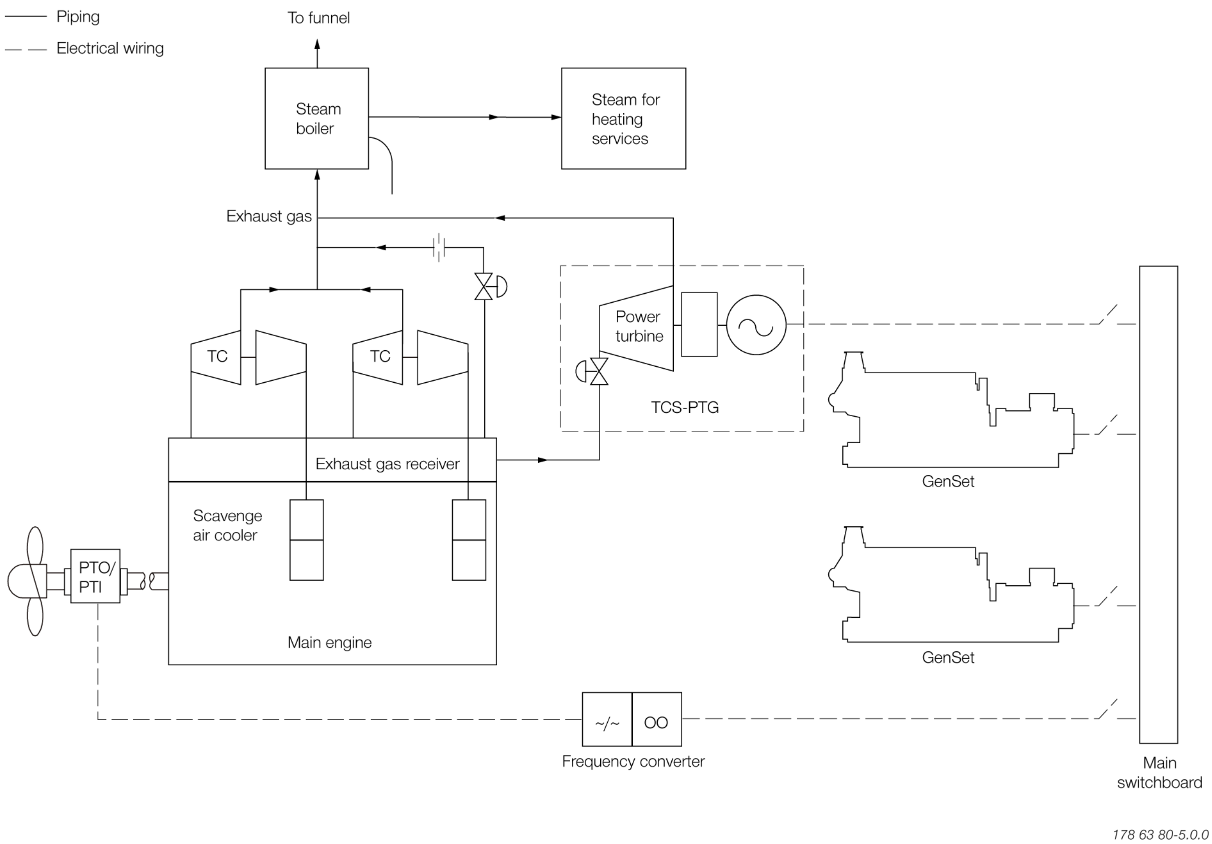

Fig. 4.05.01: PTG diagram

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

MAN Energy Solutions offers PTG solutions called TCS-PTG in the range from approx. 1,000 kW to 5,000 kW, see Fig. 4.05.02.

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 diagram of the PTG arrangement is shown.

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 3.5% of the main engine's SMCR, when the engine is running at SMCR.

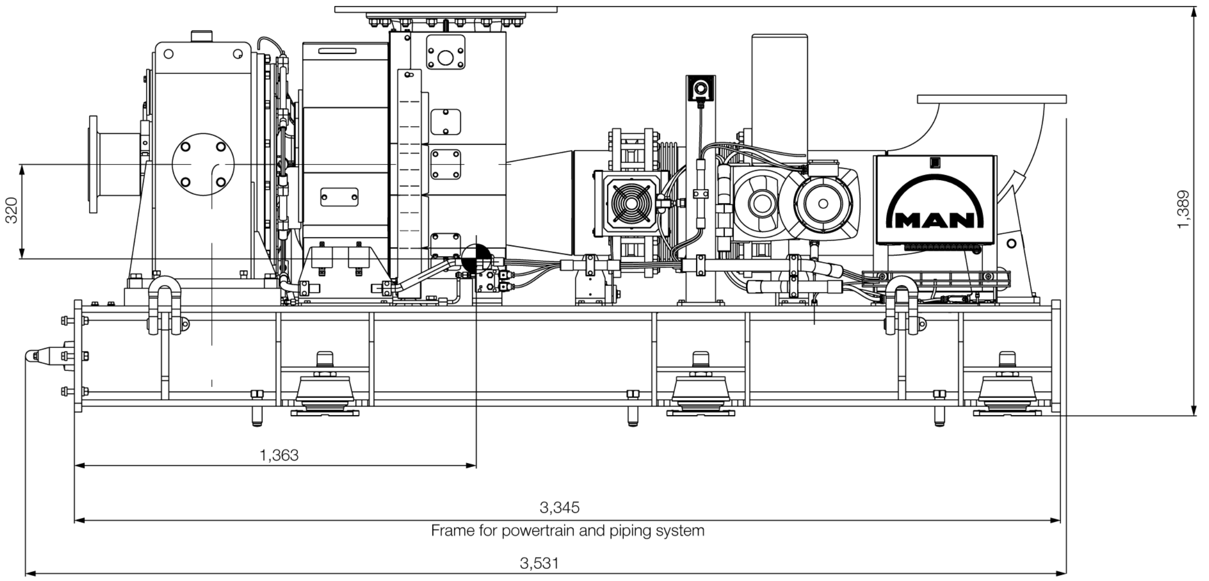

Fig. 4.05.02: MAN Energy Solutions 1,500 kW TCS-PTG solution

Steam turbine generator (STG)

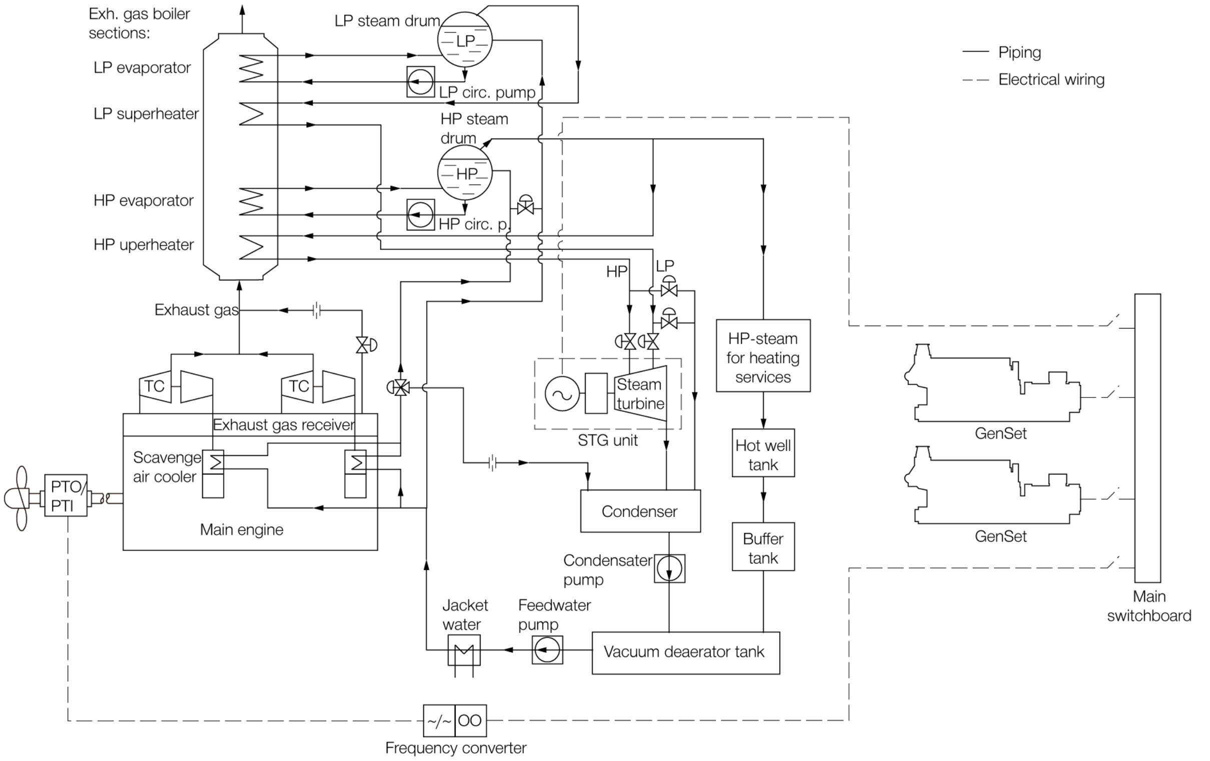

Fig. 4.05.03: STG system diagram

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 Energy Solutions designates this system STG. Fig. 4.05.03 shows an example of the STG diagram.

For WHR matching the engine, a bypass is installed to increase the temperature of the exhaust gas and improve the boiler output. The bypass valve is controlled by the engine control system.

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. A 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 bar, and for the dual pressure system the high-pressure cycle will be 9 to 10 bar and the low-pressure cycle will be 4 to 5 bar.

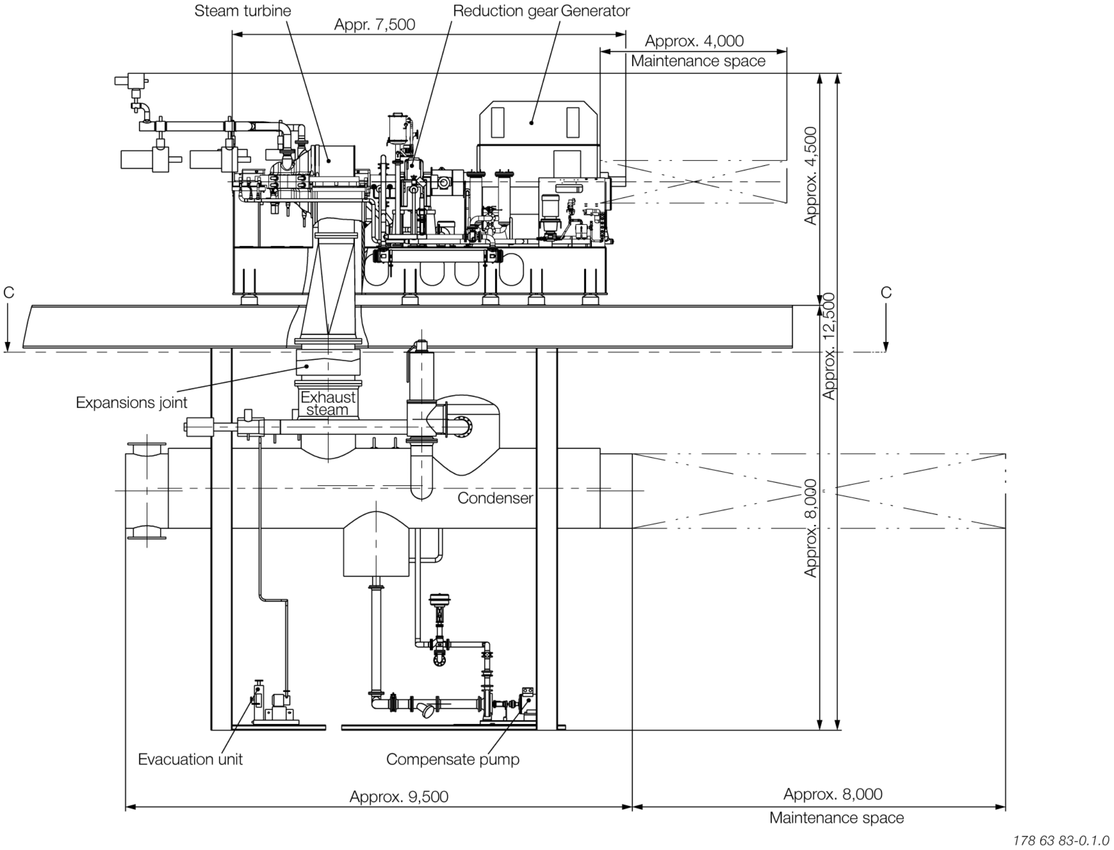

Fig. 4.05.04: STG steam turbine generator arrangement with condenser - typical arrangement

Full WHRS steam and power turbines combined

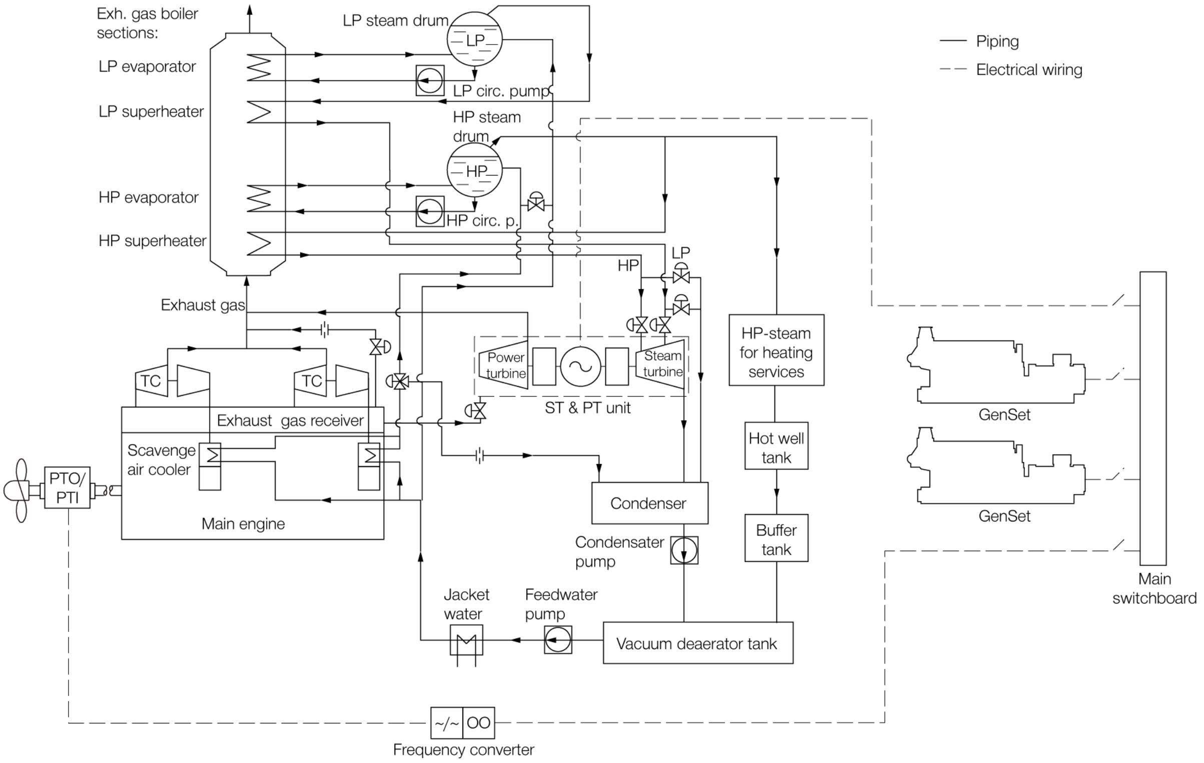

Fig. 4.05.05: Full WHRS with both steam and power turbines

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 full WHRS 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 considerably bigger than the size of an ordinary boiler system, and the actual boiler size has to be calculated from case to case. Casing space for the exhaust boiler must be reserved in the initial planning of the ship's machinery spaces.

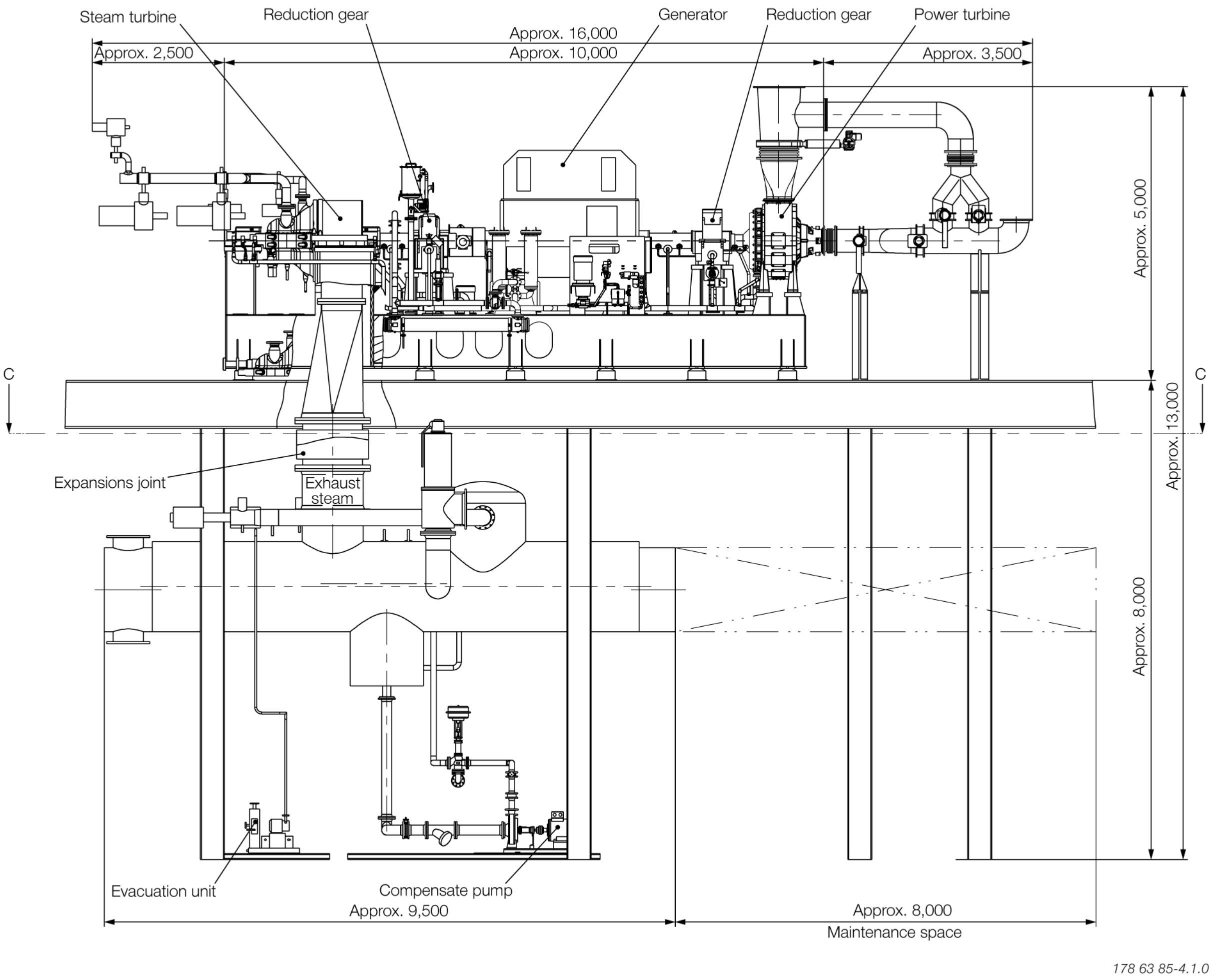

Fig. 4.05.06: Full ST & PT full waste heat recovery unit arrangement with condenser - typical arrangement

WHRS Generator output

This section is available on request

Waste heat recovery element and safety valve

The boiler water or steam for power generator is preheated in the Waste Heat Recovery (WHR) element, also called the first-stage air cooler.

The WHR element is typically built as a high-pressure water/steam heat exchanger which is placed on top of the scavenge air cooler, see Fig. 4.05.07.

Full water flow must be passed through the WHR element continuously when the engine is running. This must be considered in the layout of the steam feed water system (the WHR element supply heating). Refer to our ‘WHR element specification’ which is available from MAN Energy Solutions, Copenhagen.

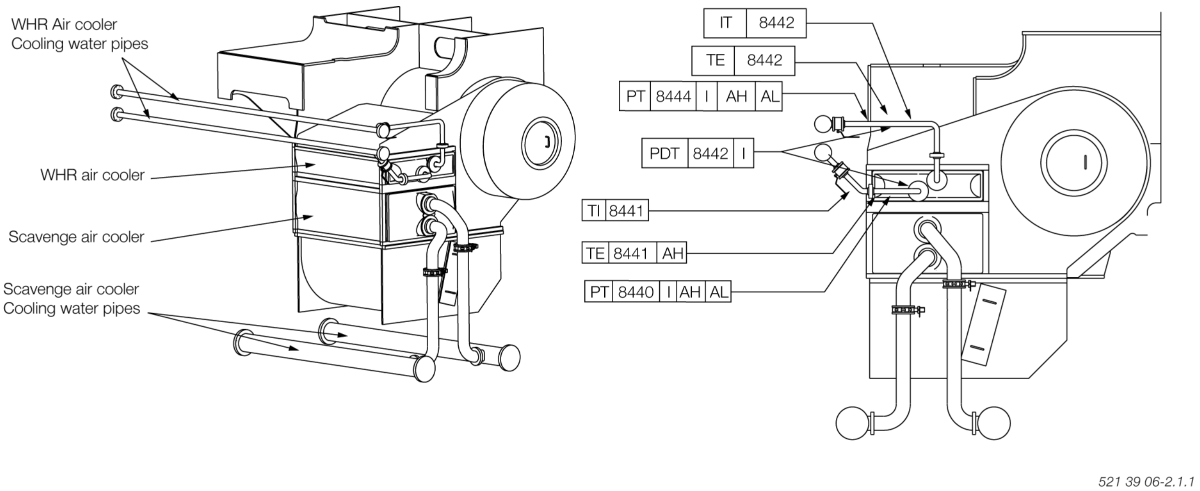

Fig. 4.05.07: WHR element on Scavenge air cooler

Safety valve and blow-Off

In normal operation, the temperature and pressure of the WHR element is in the range of 140-150°C and 8-21 bar respectively.

In order to prevent leaking components from causing personal injuries or damage to vital parts of the main engine, a safety relief valve will blow off excess pressure. The safety relief valve is connected to an external connection, ‘W’, see Fig. 4.05.08.

Connection ‘W’ must be passed to the funnel or another free space according to the class rules for steam discharge from safety valve.

As the system is pressurised according to class rules, the safety valve must be type approved.

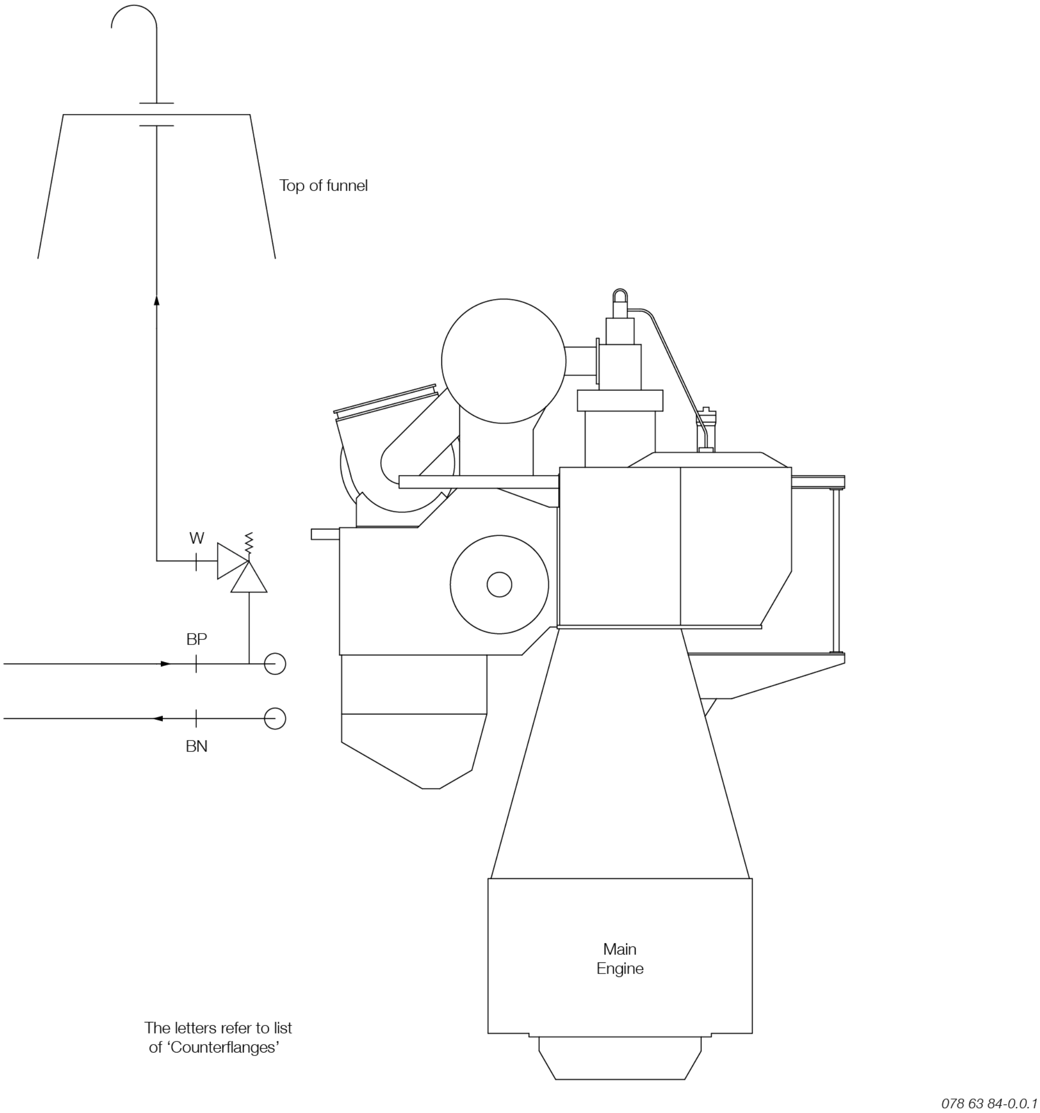

Fig. 4.05.08: WHR safety valve blow-off through connection ‘W’ to the funnel

L16/24 GenSet Data

Engine ratings

Engine type No of cylinders	1000 rpm		1200 rpm	
	1000 rpm	Available turning direction	1200 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]
5L16/24	450	Yes	500	Yes
6L16/24	570	Yes	660	Yes
7L16/24	665	Yes	770	Yes
8L16/24	760	Yes	880	Yes
9L16/24	855	Yes	990	Yes

[1] CW clockwise

Table 1: Engine ratings for emission standard - IMO Tier II

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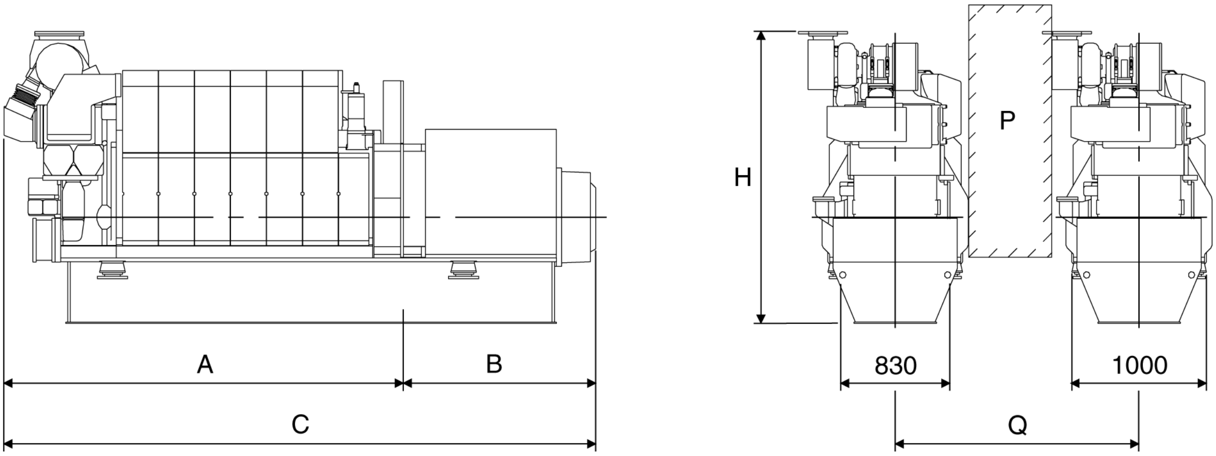
General



Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (1000 rpm)	2807	1400	4207	2337	9.5
5 (1200 rpm)	2807	1400	4207	2337	9.5
6 (1000 rpm)	3082	1490	4572	2337	10.5
6 (1200 rpm)	3082	1490	4572	2337	10.5
7 (1000 rpm)	3557	1585	5142	2337	11.4
7 (1200 rpm)	3557	1585	5142	2415	11.4
8 (1000 rpm)	3832	1680	5512	2415	12.4
8 (1200 rpm)	3832	1680	5512	2415	12.4
9 (1000 rpm)	4107	1680	5787	2415	13.1
9 (1200 rpm)	4107	1680	5787	2415	13.1

P Free passage between the engines, width 600 mm and height 2000 mm.
Q Min. distance between engines: 1800 mm.

* Depending on alternator
** Weight included a standard alternator

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

Capacities

5L:90 kW/cyl., 6L-9L: 95 kW/cyl. at 1000 rpm		5	6	7	8	9
Engine output	kW	450	570	665	760	855
Speed	rpm	1000	1000	1000	1000	1000
Heat to be dissipated [3]						
Cooling water cylinder	kW	107	135	158	181	203
Charge air cooler; cooling water HT	kW	138	169	192	213	234
Charge air cooler; cooling water LT	kW	56	69	80	91	102
Lubricating oil cooler	kW	98	124	145	166	187
Heat radiation engine	kW	15	19	23	26	29
Flow rates [4]						
Internal (inside engine)						
HT circuit (cylinder + charge air cooler HT stage)	m^3/h	10.9	12.7	14.5	16.3	18.1
LT circuit (lub. oil + charge air cooler LT stage)	m^3/h	15.7	18.9	22	25.1	28.3
Lubrication oil	m^3/h	18	18	30	30	30
External (from engine to system)						
HT water flow (at 40°C inlet)	m^3/h	5.2	6.4	7.4	8.3	9.2
LT water flow (at 38°C inlet)	m^3/h	15.7	18.9	22	25.1	28.3
Air data						
Temperature of charge air at charge air cooler outlet	°C	49	51	52	54	55
Air flow rate	m^3/h [5]	2721	3446	4021	4595	5169
	kg/kWh	6.62	6.62	6.62	6.62	6.62
Charge air pressure	bar	4.13	4.13	4.13	4.13	4.13
Air required to dissipate heat radiation (eng.) (t_2-t_1= 10°C)	m^3/h	4860	6157	7453	8425	9397
Exhaust gas data [6]						
Volume flow (temperature turbocharger outlet)	m^3/h [7]	5710	7233	8438	9644	10849
Mass flow	t/h	3.1	3.9	4.5	5.2	5.8
Temperature at turbine outlet	°C	375	375	375	375	375
Heat content (190°C)	kW	170	216	252	288	324
Permissible exhaust back pressure	mbar	< 30	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)	mbar	< 50	< 50	< 50	< 50	< 50
Pumps						
External pumps [8]						
Diesel oil pump (5 bar at fuel oil inlet A1)	m^3/h	0.32	0.40	0.47	0.54	0.60
Fuel oil supply pump (4 bar discharge pressure)	m^3/h	0.15	0.19	0.23	0.26	0.29
Fuel oil circulating pump [9] (8 bar at fuel oil inlet A1)	m^3/h	0.32	0.40	0.47	0.54	0.60
Starting air data						
Air consumption per start, incl. air for jet assist (IR/TDI)	Nm^3	0.47	0.56	0.65	0.75	0.84
Air consumption per start, incl. air for jet assist (Gali)	Nm^3	0.80	0.96	1.12	1.28	1.44

Capacities

5L:100 kW/cyl., 6L-9L: 110 kW/cyl. at 1200 rpm		5	6	7	8	9
Engine output	kW	500	660	770	880	990
Speed	rpm	1200	1200	1200	1200	1200

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5L:100 kW/cyl., 6L-9L: 110 kW/cyl. at 1200 rpm			5	6	7	8	9
Heat to be dissipated [3]							
Cooling water cylinder		kW	100	132	154	177	199
Charge air cooler; cooling water HT		kW	149	187	211	234	255
Charge air cooler; cooling water LT		kW	66	83	96	109	122
Lubricating oil cooler		kW	113	149	174	199	224
Heat radiation engine		kW	17	23	26	30	34
Flow rates [4]							
Internal (inside engine)							
HT circuit (cylinder + charge air cooler HT stage)		m^3/h	13.1	15.2	17.4	19.5	21.6
LT circuit (lube oil + charge air cooler LT stage)		m^3/h	19.3	20.7	24.2	27.7	31.1
Lubrication oil		m^3/h	21	21	35	35	35
External (from engine to system)							
HT water flow (at 40°C inlet)		m^3/h	5.7	7.3	8.4	9.4	10.4
LT water flow (at 38°C inlet)		m^3/h	19.1	20.7	24.2	27.7	31.1
Air data							
Temperature of charge air at charge air cooler outlet		°C	51	53	55	56	57
Air flow rate		m^3/h [5]	3169	4183	4880	5578	6275
		kg/kWh	6.94	6.94	6.94	6.94	6.94
Charge air pressure		bar	3.92	3.92	3.92	3.92	3.92
Air required to dissipate heat radiation (eng.) (t_2-t_1= 10°C)		m^3/h	5509	7453	8425	9721	11017
Exhaust gas data [6]							
Volume flow (temperature turbocharger outlet)		m^3/h [7]	6448	8511	9929	11348	12766
Mass flow		t/h	3.6	4.7	5.5	6.3	7.1
Temperature at turbine outlet		°C	356	356	356	356	356
Heat content (190°C)		kW	178	235	274	313	352
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)		mbar	< 50	< 50	< 50	< 50	< 50
Pumps							
External pumps [8]							
Diesel oil pump	(5 bar at fuel oil inlet A1)	m^3/h	0.35	0.47	0.54	0.62	0.70
Fuel oil supply pump	(4 bar discharge pressure)	m^3/h	0.17	0.22	0.26	0.30	0.34
Fuel oil circulating pump [9]	(8 bar at fuel oil inlet A1)	m^3/h	0.35	0.47	0.54	0.62	0.70
Starting air data							
Air consumption per start, incl. air for jet assist (IR/TDI)		Nm^3	0.47	0.56	0.65	0.75	0.84
Air consumption per start, incl. air for jet assist (Gali)		Nm^3	0.80	0.96	1.12	1.28	1.44

Remarks to capacities

1) HT cooling water flows first through HT stage charge air cooler, then through water jacket and cylinder head, water temperature outlet engine regulated by mechanical thermostat.
2) LT cooling water flows first through LT stage charge air cooler, then through lube oil cooler, water temperature outlet engine regulated by mechanical thermostat.
3) Tolerance: + 10% for rating coolers, - 15% for heat recovery.
4) Basic values for layout of the coolers.
5) Under above mentioned reference conditions.
6) Tolerance: quantity +/- 5%, temperature +/- 20°C.
   Exhaust gas flow are calculated from given exhaust gas temperature at the Tropic reference condition.

7) Tolerance of the pumps' delivery capacities must be considered by the manufactures.
8) In order to ensure sufficient flow through the engine fuel system the capacity of the fuel oil circulation pumps
9) must be minimum 3 times the full load consumption of the installed engines

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L21/31 GenSet Data

Engine ratings

Engine type No of cylinders	900 rpm		1000 rpm	
	900 rpm	Available turning direction	1000 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]
5L21/31	1000	Yes	1000	Yes
6L21/31	1320	Yes	1320	Yes
7L21/31	1540	Yes	1540	Yes
8L21/31	1760	Yes	1760	Yes
9L21/31	1980	Yes	1980	Yes
[1] CW clockwise				

Table 1: Engine ratings for emission standard

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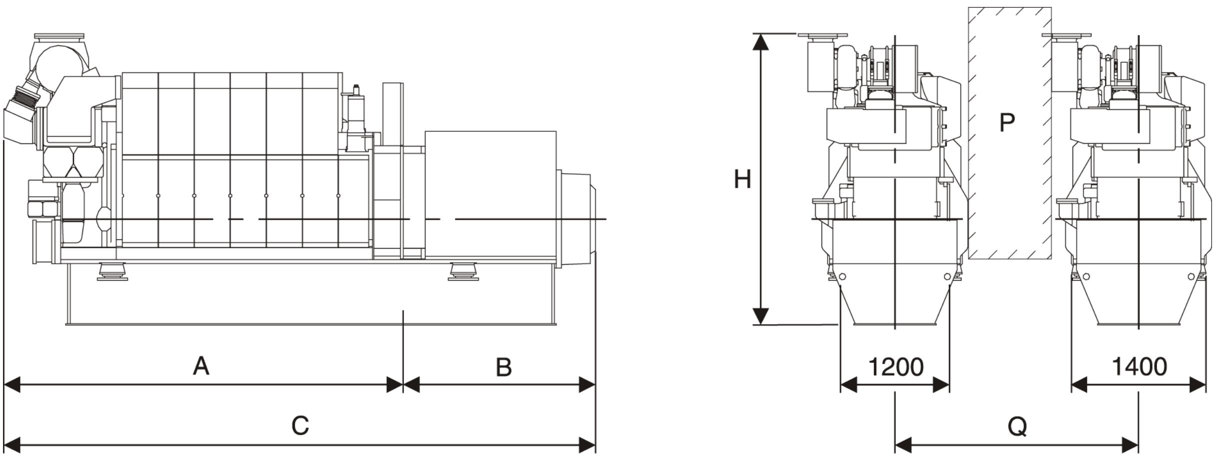
General



1 bearing

Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry WEight GenSet (t)
5 (900 rpm)	3959	1820	5779	3183	22.5
5 (1000 rpm)	3959	1870	5829	3183	22.5
6 (900 rpm)	4314	1870	6184	3183	26.0
6 (1000 rpm)	4314	2000	6314	3183	26.0
7 (900/1000 rpm)	4669	1970	6639	3289	29.5

2 bearings

Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (900/1000 rpm)	4507	2100	6607	3183	22.5
6 (900/1000 rpm)	4862	2100	6962	3183	26.0
7 (900/1000 rpm)	5217	2110	7327	3289	29.5
8 (900/1000 rpm)	5572	2110	7682	3289	33.0
9 (900/1000 rpm)	5927	2135	8062	3289	36.5

P Free passage between the engines, width 600 mm and height 2000 mm.
Q Min. distance between engines: 2400 mm (without gallery) and 2600 mm (with gallery)

* Depending on alternator
** Weight included a standard alternator

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

Capacities

5L: 200 kW/cyl., 6L-9L: 220kW/cyl. at 900 rpm, 1-String		5	6	7	8	9
Engine output	kW	1000	1320	1540	1760	1980
Speed	rpm	900	900	900	900	900
External (from engine to system)						
1-string cooling water (mix)	°C	52.4	56.4	59.1	61.6	64.2
Heat to be dissipated [3]						
Cooling water cylinder	kW	208	289	347	405	464
Charge air cooler; cooling water HT	kW	346	435	490	542	590
Charge air cooler; cooling water LT	kW	198	244	274	303	332
Lubricating oil cooler	kW	176	238	281	324	368
Heat radiation engine	kW	49	65	76	87	98
Flow rates [4]						
Internal (inside engine)						
HT circuit (cylinder + charge air cooler HT stage)	m^3/h	55	55	55	55	55
LT circuit (lube oil + charge air cooler LT stage)	m^3/h	55	55	55	55	55
Lubrication oil	m^3/hh	31	31	41	41	41
External (from engine to system)						
HT water flow (at 40°C inlet)	m^3/h	11.1	14.1	16.0	17.8	19.5
LT water flow (at 38°C inlet)	m^3/h	55	55	55	55	55
Air data						
Temperature of charge air at charge air cooler outlet	°C	52	56	58	60	62
Air flow rate	m^3/h [5]	6656	8786	10250	11714	13178
	kg/kWh	7.28	7.28	7.28	7.28	7.28
Charge air pressure	bar	4.58	4.61	4.63	4.64	4.66
Air required to dissipate heat radiation (eng.) (t_2-t_1= 10°C)	m^3/h	17980	23800	27600	31500	35300
Exhaust gas data [6]						
Volume flow (temperature turbocharger outlet)	m^3/h [7]	13484	17918	20981	24055	27130
Mass flow	t/h	7.5	9.9	11.5	13.2	14.8
Temperature at turbine outlet	°C	353	357	360	362	363
Heat content (190°C)	kW	366	496	587	679	771
Permissible exhaust back pressure	mbar	< 30	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)	mbar	< 50	< 50	< 50	< 50	< 50
Pumps						
External pumps [8]						
Diesel oil pump (5 bar at fuel oil inlet A1)	m^3/h	0.89	1.18	1.37	1.57	1.76
Fuel oil supply pump (4 bar discharge pressure)	m^3/h	0.30	0.39	0.46	0.52	0.59
Fuel oil circulating pump [9] (8 bar at fuel oil inlet A1)	m^3/h	0.89	1.18	1.37	1.57	1.76
Starting air data						
Air consumption per start, incl. air for jet assist (TDI)	Nm^3	1.0	1.2	1.4	1.6	1.8
Air consumption per start, incl. air for jet assist (Gali)	Nm^3	1.8	2.1	2.4	2.7	3.0

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2021-11-12 - en

Capacities

5L:200 kW/cyl., 6L-9L: 220 kW/cyl. at 1000 rpm, 1-String			5	6	7	8	9
External (from engine to system) 1-String coding water (mix)		°C	50.6	54.1	56.4	58.6	60.8
Engine output		kW	1000	1320	1540	1760	1980
Speed		rpm	1000	1000	1000	1000	1000
Heat to be dissipated [3]							
Cooling water cylinder		kW	206	285	342	399	456
Charge air cooler; cooling water HT		kW	321	404	455	503	548
Charge air cooler; cooling water LT		kW	192	238	266	294	321
Lubricating oil cooler		kW	175	236	279	322	365
Heat radiation engine		kW	49	65	76	87	98
Flow rates [4]							
Internal (inside engine)							
HT circuit (cylinder + charge air cooler HT stage)		m^3/h	61	61	61	61	61
LT circuit (lube oil + charge air cooler LT stage)		m^3/h	61	61	61	61	61
Lubrication oil		m^3/h	34	34	46	46	46
External (from engine to system)							
HT water flow (at 40°C inlet)		m^3/h	10.7	13.5	15.4	17.1	18.8
LT water flow (at 38°C inlet)		m^3/h	61	61	61	61	61
Air data							
Temperature of charge air at charge air cooler outlet		°C	51	55	57	59	60
Air flow rate		m^3/h [5]	6647	8774	10237	11699	13161
		kg/kWh	7.27	7.27	7.27	7.27	7.27
Charge air pressure		bar	4.25	4.28	4.29	4.30	4.31
Air required to dissipate heat radiation (eng.) (t_2-t_1= 10°C)		m^3/h	17980	23800	27600	31500	35300
Exhaust gas data [6]							
Volume flow (temperature turbocharger outlet)		m^3/h [7]	13730	18235	21348	24468	27594
Mass flow		t/h	7.5	9.9	11.5	13.2	14.8
Temperature at turbine outlet		°C	365	369	372	373	375
Heat content (190°C)		kW	394	532	628	725	823
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)		mbar	< 50	< 50	< 50	< 50	< 50
Pumps							
External pumps [8]							
Diesel oil pump	(5 bar at fuel oil inlet A1)	m^3/h	0.89	1.18	1.37	1.57	1.76
Fuel oil supply pump	(4 bar)	m^3/h	0.30	0.39	0.46	0.52	0.59
Fuel oil circulating pump [9]	(8 bar)	m^3/h	0.89	1.18	1.37	1.57	1.76
Starting air data							
Air consumption per start, incl. air for jet assist (TDI)		Nm^3	1.0	1.2	1.4	1.6	1.8
Air consumption per start, incl. air for jet assist (Gali)		Nm^3	1.8	2.1	2.4	2.7	3.0

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L23/30H Mk2 genset data

Engine ratings

Engine type No of cylinders	720 rpm		750 rpm		900 rpm	
	720 rpm	Available turning direction	750 rpm	Available turning direction	900 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]	kW	CW [1]
5L23/30H Mk2	650/710	Yes	675/740	Yes	–	–
6L23/30H Mk2	852	Yes	888	Yes	1050	Yes
7L23/30H Mk2	994	Yes	1036	Yes	1225	Yes
8L23/30H Mk2	1136	Yes	1184	Yes	1400	Yes
[1] CW clockwise						

Table 1: Engine ratings for emission standard IMO Tier II.

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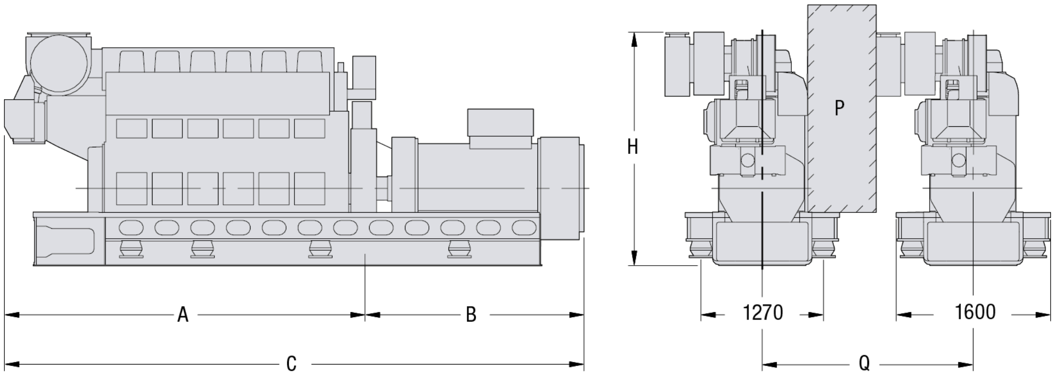
General



Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (720 rpm)	3369	2155	5524	2402	18.0
5 (750 rpm)	3369	2155	5524	2402	17.6
6 (720 rpm)	3738	2265	6004	2402	19.7
6 (750 rpm)	3738	2265	6004	2402	19.7
6 (900 rpm)	3738	2265	6004	2466	21.0
7 (720 rpm)	4109	2395	6504	2466	21.4
7 (750 rpm)	4109	2395	6504	2466	21.4
7 (900 rpm)	4109	2395	6504	2466	22.8
8 (720 rpm)	4475	2480	6959	2466	23.5
8 (750 rpm)	4475	2480	6959	2466	22.9
8 (900 rpm)	4475	2340	6815	2466	24.5

P Free passage between the engines, width 600 mm and height 2000 mm.
Q Min. distance between engines: 2250 mm

* Depending on alternator
** Weight included a standard alternator

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

2022-03-02 - en

Capacities

5-8L23/30H Mk 2: 142 kW/Cyl., 720 rpm			5	6	7	8
Engine output		kW	710	852	994	1136
Speed		rpm	720	720	720	720
Heat to be dissipated [3)]						
Cooling water cylinder		kW	217	262	302	347
Charge air cooler; cooling water HT						
(1 stage cooler: no HT-stage)		kW	-	-	-	-
Charge air cooler; cooling water LT		kW	351	407	505	563
Lubricating oil cooler		kW	67	81	94	107
Heat radiation engine		kW	30	36	42	48
Air data						
Charge air temp. at charge air cooler outlet, max.		°C	54	56	53	54
Air flow rate		m^3/h [5)]	5430	6516	7602	8688
		kg/kWh	8.03	8.03	8.03	8.03
Charge air pressure		bar (abs)	3.39	3.40	3.39	3.39
Air required to dissipate heat radiation (eng.)						
$(t_2\text{-}t_1=10°C)$		m^3/h	9756	11708	13659	15610
Exhaust gas data [6)]						
Volume flow (temperature turbocharger outlet)		m^3/h [7)]	10086	12069	14138	16121
Mass flow		t/h	5.85	7.00	8.20	9.35
Temperature at turbine outlet		°C	324	325	323	324
Heat content (190°C)		kW	234	283	325	374
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)		mbar	< 50	< 50	< 50	< 50
Pumps						
Engine driven pumps [4)]						
HT cooling water pump	1-2.5 bar	m^3/h	36	36	36	36
LT cooling water pump	1-2.5 bar	m^3/h	55	55	55	55
Lubrication oil	3-5 bar	m^3/h	16	16	16	16
External pumps [8)]						
Diesel oil pump	4 bar at fuel oil inlet A1	m^3/h	0.52	0.62	0.73	0.83
Fuel oil supply pump	4 bar discharge pressure	m^3/h	0.26	0.31	0.36	0.41
Fuel oil circulating pump [9)]	8 bar at fuel oil inlet A1	m^3/h	0.51	0.62	0.72	0.82
Cooling water pumps						
"Internal cooling water system 1"						
LT cooling water pump	1-2.5 bar	m^3/h	35	42	48	55
"Internal cooling water system 2"						
HT cooling water pump	1-2.5 bar	m^3/h	20	24	28	32
LT cooling water pump	1-2.5 bar	m^3/h	35	42	48	55
Lubricating oil pump	3-5 bar	m^3/h	14	15	16	17
Starting air system						
Air consumption per start (10 bar starter)		Nm^3	1.40	1.43	1.50	1.54

1) Tolerance: + 10 % for rating coolers, - 15 % for heat recovery

2) LT cooling water flows in parallel through one-stage charge air cooler and lube oil cooler HT cooling water flows only through water jacket and cylinder head, water temperature outlet engine regulated by mechanical thermostat

3) Basic values for layout of the coolers

4) Under above mentioned reference conditions

5) Tolerance: quantity +/- 5%, temperature +/- 20°C

6) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions

7) Tolerance of the pumps delivery capacities must be considered by the manufactures

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

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Capacities

6-8L23/30H Mk 2: 175 kW/Cyl., 900 rpm			6	7	8
Engine output Speed		kW rpm	1050 900	1225 900	1400 900
Heat to be dissipated [3]					
Cooling water cylinder		kW	298	349	402
Charge air cooler; cooling water HT					
1 stage cooler: no HT-stage		kW	-	-	-
Charge air cooler; cooling water LT		kW	441	604	672
Lubricating oil cooler		kW	122	143	164
Heat radiation engine		kW	42	49	56
Air data					
Temp. of charge air at charge air cooler outlet, max.		°C	54	55	56
Air flow rate		m^3/h [5]	8020	9357	10693
		kg/kWh	8.02	8.02	8.02
Charge air pressure		bar (abs)	3.60	3.61	3.61
Air required to dissipate heat radiation (eng.) (t_2-t_1=10°C)		m^3/h	13669	15947	18225
Exhaust gas data [6]					
Volume flow (temperature turbocharger outlet)		m^3/h [7]	15636	18364	20909
Mass flow		t/h	8.60	10.10	11.50
Temperature at turbine outlet		°C	354	355	356
Heat content (190°C)		kW	423	497	571
Permissible exhaust back pressure		mbar	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)		mbar	< 50	< 50	< 50
Pumps					
Engine driven pumps [4]					
HT cooling water pump	1-2.5 bar	m^3/h	45	45	45
LT cooling water pump	1-2.5 bar	m^3/h	69	69	69
Lubrication oil	3-5 bar	m^3/h	20	20	20
External pumps [8]					
Diesel oil pump	4 bar at fuel oil inlet A1	m^3/h	0.78	0.91	1.04
Fuel oil supply pump	4 bar discharge pressure	m^3/h	0.38	0.45	0.51
Fuel oil circulating pump [9]	8 bar at fuel oil inlet A1	m^3/h	0.77	0.90	1.03
Cooling water pumps					
"Internal cooling water system 1"					
LT cooling water pump	1-2.5 bar	m^3/h	52	61	70
"Internal cooling water system 2"					
HT cooling water pump	1-2.5 bar	m^3/h	30	35	40
LT cooling water pump	1-2.5 bar	m^3/h	52	61	70
Lubricating oil pump	3-5 bar	m^3/h	17	18	19
Starting air system					
Air consumption per start (10 bar starter)		Nm^3	1.43	1.50	1.54

1) Tolerance: + 10 % for rating coolers, - 15 % for heat recovery

2) LT cooling water flows in parallel through one-stage charge air cooler and lube oil cooler HT cooling water flows only through water jacket and cylinder head, water temperature outlet engine regulated by mechanical thermostat

2022-03-02 - en

4.08 L23/30H Mk2 genset data

3) Basic values for layout of the coolers

4) Under above mentioned reference conditions

5) Tolerance: quantity +/- 5%, temperature +/- 20°C

6) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions

7) Tolerance of the pumps delivery capacities must be considered by the manufactures

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

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2022-03-02 - en

L27/38 GenSet Data

Engine ratings

Engine type No of cylinders	720 rpm		750 rpm		720/750 MGO	
	720 rpm	Available turning direction	750 rpm	Available turning direction	720/750 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]	kW	CW [1]
5L27/38	1,500	Yes	1,600	Yes	-	-
6L27/38	1,980	Yes	1,980	Yes	2,100	Yes
7L27/38	2,310	Yes	2,310	Yes	2,450	Yes
8L27/38	2,640	Yes	2,640	Yes	2,800	Yes
9L27/38	2,970	Yes	2,970	Yes	3,150	Yes

[1] CW clockwise

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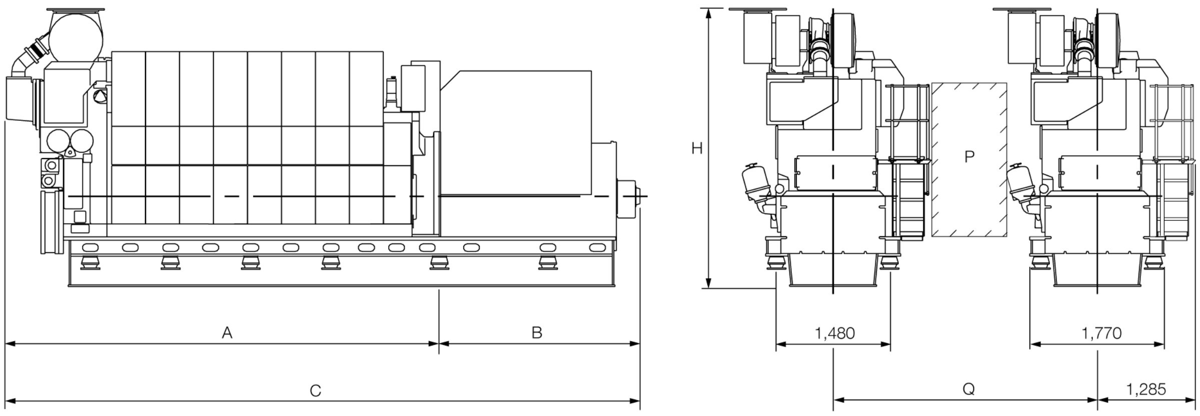
General



Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (720 mm) 5 (750 mm)	4,346 4,346	2,486 2,486	6,832 6,832	3,712 3,712	40.0 40.0
6 (720 mm 6 (750 mm)	4,791 4,791	2,766 2,766	7,557 7,557	3,712 3,712	44.5 44.5
7 (720 mm) 7 (750 mm)	5,236 5,236	2,766 2,766	8,002 8,002	3,899 3,899	50.4 50.4
8 (720 mm) 8 (750 mm)	5,681 5,681	2,986 2,986	8,667 8,667	3,899 3,899	58.2 58.2
9 (720 mm) 9 (750 mm)	6,126 6,126	2,986 2,986	9,112 9,112	3,899 3,899	64.7 64.7

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 included a standard alternator

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

Capacities

5L27/38: 300 kW/cyl., 720 rpm, 6-9L27/38: 330 kW/cyl., 720 rpm						
Reference condition : Tropic						
Air temperature	°C	45				
LT-water temperature inlet engine (from system)	°C	38				
Air pressure	bar	1				
Relative humidity	%	50				
Temperature basis:						
Setpoint HT cooling water engine outlet [1]	°C	79°C nominal (Range of mech. thermostatic element 77-85°C)				
Setpoint LT cooling water engine outlet [2]	°C	35°C nominal (Range of mech. thermostatic element 29°-41°C)				
Setpoint Lube oil inlet engine	°C	66°C nominal (Range of mech. thermostatic element 63-72°C)				
Number of cylinders		5	6	7	8	9
Engine output	kW	1,500	1,980	2,310	2,640	2,970
Speed	rpm	720				
Heat to be dissipated [3]						
Cooling water (C.W.) Cylinder	kW	256	330	385	440	495
Charge air cooler; cooling water HT	kW	466	594	675	750	820
Charge air cooler; cooling water LT	kW	178	216	242	268	297
Lube oil (L.O.) cooler	kW	224	279	325	372	418
Heat radiation engine	kW	63	83	97	111	125
Flow rates[4] Internal (inside engine)						
HT circuit (cylinder + charge air cooler HT stage)	m^3/h	58	58	58	58	58
LT circuit (lube oil + charge air cooler LT stage)	m^3/h	58	58	58	58	58
Lube oil	m^3/h	64	64	92	92	92
External (from engine to system)						
HT water flow (at 40°C inlet)	m^3/h	16	20.2	23	25.5	28
LT water flow (at 38°C inlet)	m^3/h	58	58	58	58	58

Air data	

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Temperature of charge air at charge air cooler outlet		°C	50	53	55	56	57
Air flow rate		m^3/h 5)	9,137	12,061	14,071	16,082	18,092
		kg/kWh	6.67	6.67	6.67	6.67	6.67
Charge air pressure		bar	4.01	4.01	4.01	4.01	4.01
Air required to dissipate heat radiation (eng.)(t_2-t_1= 10°C)		m^3/h	20,414	26,895	31,431	35,968	40,504
Exhaust gas data 6)							
Volume flow (temperature turbocharger outlet)		m^3/h 7)	19,203	25,348	29,572	33,797	38,021
Mass flow		t/h	10.3	13.6	15.9	18.1	20.4
Temperature at turbine outlet		°C	376	376	376	376	376
Heat content (190°C)		k/W	575	759	886	1,012	1,139
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30	< 30
Pumps							
External pumps 8)							
Diesel oil pump	(5 bar at fuel oil inlet A1)	m^3/h	1.06	1.40	1.63	1.87	2.10
Fuel oil supply pump	(4 bar discharge pressure)	m^3/h	0.51	0.67	0.79	0.90	1.01
Fuel oil circulating pump	(8 bar at fuel oil inlet A1)	m^3/h	1.06	1.40	1.63	1.87	2.10
Starting air data							
Air consumption per start, incl. air for jet assist (IR/TDI)		Nm^3	2.5	2.9	3.3	3.8	4.3

1) HT cooling water flows first through HT stage charge air cooler, then through water jacket and cylinder head, water temperature outlet engine regulated by mechanical thermostat.

2) LT cooling water flows first through LT stage charge air cooler, then through lube oil cooler, water temperature outlet engine regulated by mechanical thermostat.

3) Tolerance: + 10% for rating coolers, - 15% for heat recovery.

4) Basic values for layout of the coolers.

5) Under above mentioned reference conditions.

6) Tolerance: quantity +/- 5%, temperature +/- 20°C.

7) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions.

8) Tolerance of the pumps delivery capacities must be considered by the manufactures.

D10050_1689471-7.3

Capacities

5L27/38: 320 kW/cyl., 750 rpm, 6-9L27/38: 330 kW/cyl., 750 rpm						
Reference condition : Tropic						
Air temperature	°C	45				
LT-water temperature inlet engine (from system)	°C	38				
Air pressure	bar	1				
Relative humidity	%	50				
Temperature basis:						
Setpoint HT cooling water engine outlet 1)	°C	79°C nominal (Range of mech. thermostatic element 77-85°C)				
Setpoint LT cooling water engine outlet 2)	°C	35°C nominal (Range of mech. thermostatic element 29°-41°C)				
Setpoint Lube oil inlet engine	°C	66°C nominal (Range of mech. thermostatic element 63-72°C)				
Number of cylinders		5	6	7	8	9
Engine output	kW	1,600	1,980	2,310	2,640	2,970
Speed	rpm	750				
Heat to be dissipated 3)						
Cooling water (C.W.) Cylinder	kW	263	330	385	440	495
Charge air cooler; cooling water HT	kW	488	587	666	741	811
Charge air cooler; cooling water LT	kW	194	225	252	280	307
Lube oil (L.O.) cooler	kW	230	279	325	372	418
Heat radiation engine	kW	67	83	97	111	125
Flow rates 4) Internal (inside engine)						
HT circuit (cylinder + charge air cooler HT stage)	m^3/h	69	69	69	69	69
LT circuit (lube oil + charge air cooler LT stage)	m^3/h	69	69	69	69	69
Lube oil	m^3/h	66	66	96	96	96
External (from engine to system)						
HT water flow (at 40°C inlet)	m^3/h	16.8	20.3	23	25.7	28.2
LT water flow (at 38°C inlet)	m^3/h	69	69	69	69	69
Air data						
Temperature of charge air at charge air cooler outlet	°C	51	53	55	56	57
Air flow rate	m^3/h 5) kg/kWh	9,951 6.81	12,314 6.81	14,367 6.81	16,419 6.81	18,472 6.81
Charge air pressure	bar	4.04	4.04	4.04	4.04	4.04
Air required to dissipate heat radiation (eng.)(t_2-t_1= 10°C)	m^3/h	21,710	26,895	31,431	35,968	40,504
Exhaust gas data 6)						

Volume flow (temperature turbocharger outlet)		m³/h 7)	20,546	25,426	29,664	33,901	38,139
Mass flow		t/h	11.2	13.9	16.2	18.5	20.8
Temperature at turbine outlet		°C	365	365	365	365	365
Heat content (190°C)		k/W	589	729	850	972	1,093
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30	< 30
Pumps							
External pumps 8)							
Diesel oil pump	(5 bar at fuel oil inlet A1)	m³/h	1.13	1.40	1.63	1.87	2.10
Fuel oil supply pump	(4 bar discharge pressure)	m³/h	0.54	0.67	0.79	0.90	1.01
Fuel oil circulating pump	(8 bar at fuel oil inlet A1)	m³/h	1.13	1.40	1.63	1.87	2.10
Starting air data							
Air consumption per start, incl. air for jet assist (IR/TDI)		Nm³	2.5	2.9	3.3	3.8	4.3

1) HT cooling water flows first through HT stage charge air cooler, then through water jacket and cylinder head, water temperature outlet engine regulated by mechanical thermostat.

2) LT cooling water flows first through LT stage charge air cooler, then through lube oil cooler, water temperature outlet engine regulated by mechanical thermostat.

3) Tolerance: + 10% for rating coolers, - 15% for heat recovery.

4) Basic values for layout of the coolers.

5) Under above mentioned reference conditions.

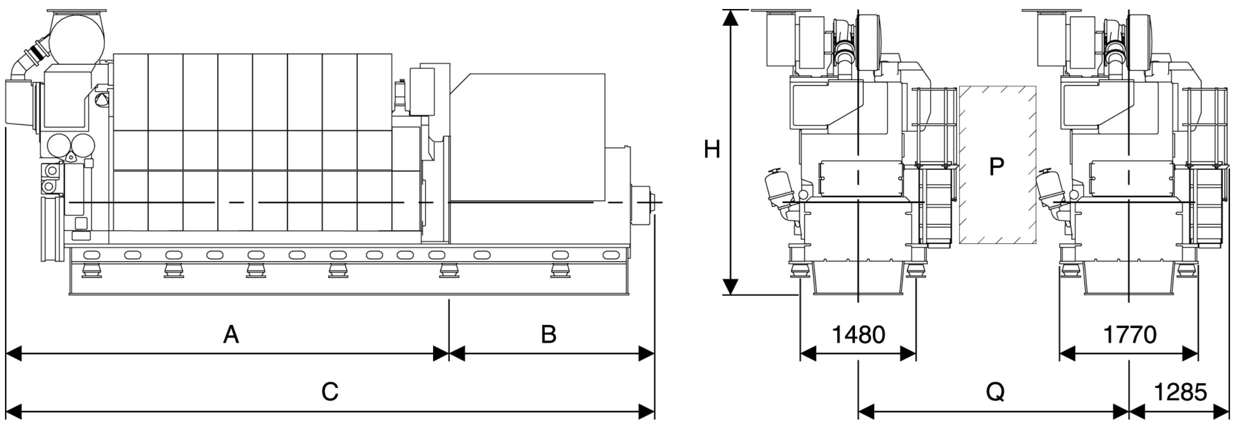
6) Tolerance: quantity +/- 5%, temperature +/- 20°C.

7) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions.

8) Tolerance of the pumps delivery capacities must be considered by the manufactures.

D10050_1689472-9.3

General



Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (720 rpm)	4346	2486	6832	3705	42.0
5 (750 rpm)	4346	2486	6832	3705	42.3
6 (720 rpm)	4791	2766	7557	3717	45.8
6 (750 rpm)	4791	2766	7557	3717	46.1
7 (720 rpm)	5236	2766	8002	3717	52.1
7 (750 rpm)	5236	2766	8002	3717	52.1
8 (720 rpm)	5681	2986	8667	3717	56.5
8 (750 rpm)	5681	2986	8667	3717	58.3
9 (720 rpm)	6126	2986	9112	3797	61.8
9 (750 rpm)	6126	2986	9112	3797	63.9

P Free passage between the engines, width 600 mm and height 2000 mm.
Q Min. distance between engines: 2900 mm (without gallery) and 3100 mm (with gallery)

* Depending on alternator
** Weight included a standard alternator

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

Engine ratings

Engine type No of cylinders	720 rpm		750 rpm		720/750 MGO	
	720 rpm	Available turning direction	750 rpm	Available turning direction	720/750 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]	kW	CW [1]
5L27/38	1500	Yes	1600	Yes	–	–
6L27/38	1980	Yes	1980	Yes	2100	Yes
7L27/38	2310	Yes	2310	Yes	2450	Yes
8L27/38	2640	Yes	2640	Yes	2800	Yes
9L27/38	2970	Yes	2970	Yes	3150	Yes
[1] CW clockwise						

Table 1: Engine ratings for emission standard - IMO Tier II.

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4.09 L27/38 GenSet Data

L28/32H Genset data

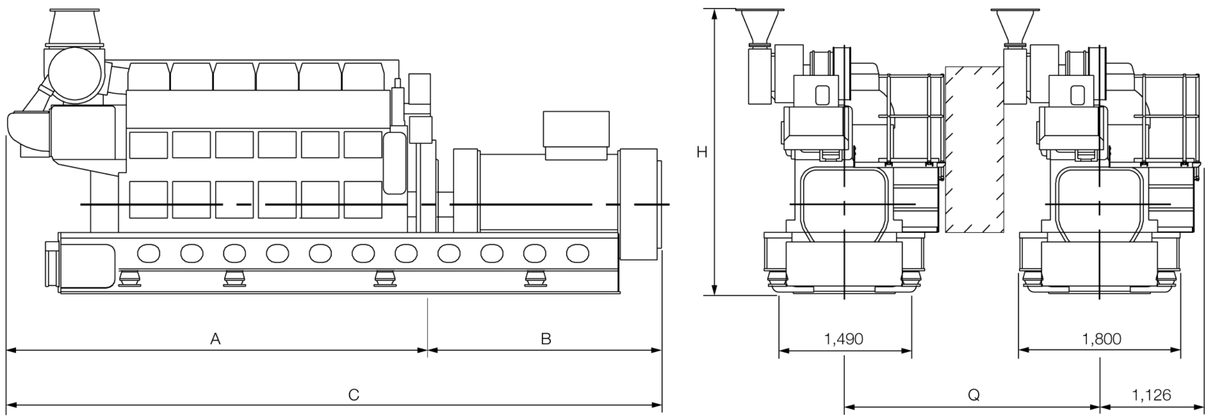
Engine Ratings

Engine type No of cylinders	720 rpm		750 rpm	
	720 rpm	Available turning direction	750 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]
5L28/32	1,050	Yes	1,100	Yes
6L28/32	1,260	Yes	1,320	Yes
7L28/32	1,470	Yes	1,540	Yes
8L28/32	1,680	Yes	1,760	Yes
9L28/32	1,890	Yes	1,980	Yes

[1] CW clockwise

B10011-3700014-9.0

General



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Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (720 mm) 5 (750 mm)	4,279 4,279	2,400 2,400	6,679 6,679	3,184 3,184	32.6 32.6
6 (720 mm 6 (750 mm)	4,759 4,759	2,510 2,510	7,269 7,269	3,184 3,184	36.3 36.3
7 (720 mm) 7 (750 mm)	5,499 5,499	2,680 2,680	8,179 8,179	3,374 3,374	39.4 39.4
8 (720 mm) 8 (750 mm)	5,979 5,979	2,770 2,770	8,749 8,749	3,374 3,374	40.7 40.6
9 (720 mm) 9 (750 mm)	6,199 6,199	2,690 2,690	8,889 8,889	3,534 3,534	47.1 47.1

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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 included a standard alternator

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

Capacities

5L-9L: 210 kW/Cyl. at 720 rpm						
Reference condition : Tropic						
Air temperature	°C	45				
LT water temperature inlet engine (from system)	°C	38				
Air pressure	bar	1				
Relative humidity	%	50				
Number of cylinders		5	6	7	8	9
Engine output	kW	1,050	1,260	1,470	1,680	1,890
Speed	rpm	720	720	720	720	720
Heat to be dissipated [1]						
Cooling water (C.W.) Cylinder	kW	234	281	328	375	421
Charge air cooler; cooling water HT	kW	0	0	0	0	0
(Single stage charge air cooler)						
Charge air cooler; cooling water LT	kW	355	397	500	553	592
Lube oil (L.O.) cooler	kW	191	230	268	306	345
Heat radiation engine	kW	26	31	36	42	47
	kW					
Flow rates [2] Internal (inside engine)						
HT cooling water cylinder	m^3/h	37	45	50	55	60
LT cooling water lube oil cooler *	m^3/h	7.8	9.4	11	12.7	14.4
LT cooling water lube oil cooler **	m^3/h	28	28	40	40	40
LT cooling water charge air cooler	m^3/h	37	45	55	65	75
Air data						
Temperature of charge air at charge air cooler outlet	°C	51	52	51	52	53
Air flow rate	m^3/h [3] kg/kWh	7,355 7.67	8,826 7.67	10,297 7.67	11,768 7.67	13,239 7.67
Charge air pressure	bar	2.97	2.97	2.97	2.97	2.97
Air required to dissipate heat radiation (engine) (t_2-t_1= 10°C)	m^3/h	8,425	10,045	11,665	13,609	15,230

Exhaust gas data [4]

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Volume flow (temperature turbocharger outlet)	m^3/h^5	14,711	17,653	20,595	23,537	26,479
Mass flow	t/h	8.3	9.9	11.6	13.2	14.9
Temperature at turbine outlet	°C	347	347	347	347	347
Heat content (190°C)	kW	389	467	545	623	701
Permissible exhaust back pressure	mbar	< 30	< 30	< 30	< 30	< 30
Starting air system						
Air consumption per start	Nm^3	2.5	2.5	2.5	2.5	2.5
Pumps Engine driven pumps						
Fuel oil feed pump (5.5-7.5 bar)	m^3/h	1.4	1.4	1.4	1.4	1.4
HT circuit cooling water (1.0-2.5 bar)	m^3/h	45	45	60	60	60
LT circuit cooling water (1.0-2.5 bar)	m^3/h	45	60	75	75	75
Lube oil (3.0-5.0 bar)	m^3/h	24	24	34	34	34
External pumps [6)]						
Diesel oil pump (4 bar at fuel oil inlet A1)	m^3/h	0.74	0.89	1.04	1.19	1.34
Fuel oil supply pump (4 bar discharge pressure)	m^3/h	0.36	0.43	0.50	0.57	0.64
Fuel oil circulating pump (8 bar at fuel oil inlet A1)	m^3/h	0.74	0.89	1.04	34	34
HT circuit cooling water (1.0-2.5 bar)	m^3/h	37	45	50	55	60
LT circuit cooling water (1.0-2.5 bar) *	m^3/h	45	54	65	77	89
LT circuit cooling water (1.0-2.5 bar) **	m^3/h	65	73	95	105	115
Lube oil (3.0-5.0 bar)	m^3/h	22	23	35	27	28

1) Tolerance: + 10% for rating coolers, - 15% for heat recovery.

2) Basic values for layout of the coolers.

3) Under above mentioned reference conditions.

4) Tolerance: quantity +/- 5%, temperature +/- 20°C.

5) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions.

6) Tolerance of the pumps delivery capacities must be considered by the manufactures.

* Only valid for engines equipped with internal basic cooling water system no. 1 and 2.

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

D10050_3700075-9.0

Capacities

5L-9L: 220 kW/Cyl. at 750 rpm						
Reference condition : Tropic						
Air temperature	°C	45				
LT water temperature inlet engine (from system)	°C	38				
Air pressure	bar	1				
Relative humidity	%	50				
Number of cylinders		5	6	7	8	9
Engine output	kW	1,100	1,320	1,540	1,760	1,980
Speed	rpm	750	750	750	750	750
Heat to be dissipated [1)]						
Cooling water (C.W.) Cylinder	kW	245	294	343	392	442
Charge air cooler; cooling water HT	kW	0	0	0	0	0
(Single stage charge air cooler)						
Charge air cooler; cooling water LT	kW	387	435	545	587	648
Lube oil (L.O.) cooler	kW	201	241	281	321	361
Heat radiation engine	kW	27	33	38	44	49
Flow rates [2)] Internal (inside engine)						
HT cooling water cylinder	m^3/h	37	45	50	55	60
LT cooling water lube oil cooler *	m^3/h	7.8	9.4	11	12.7	14.4
LT cooling water lube oil cooler **	m^3/h	28	28	40	40	40
LT cooling water charge air cooler	m^3/h	37	45	55	65	75
Air data						
Temperature of charge air at charge air cooler outlet	°C	52	54	52	52	55
Air flow rate	m^3/h [3)] kg/kWh	7,826 7.79	9,391 7.79	10,956 7.79	12,521 7.79	14,087 7.79
Charge air pressure	bar	3.07	3.07	3.07	3.07	3.07
Air required to dissipate heat radiation (engine) (t_2-t_1= 10°C)	m^3/h	8,749	10,693	12,313	14,257	15,878
Exhaust gas data [4)]						
Volume flow (temperature turbocharger outlet)	m^3/h [5]	15,520	18,624	21,728	24,832	27,936
Mass flow	t/h	8.8	10.5	12.3	14.1	15.8
Temperature at turbine outlet	°C	342	342	342	342	342
Heat content (190°C)	kW	401	481	561	641	721
Permissible exhaust back pressure	mbar	< 30	< 30	< 30	< 30	< 30

Starting air system						
Air consumption per start	Nm^3	2.5	2.5	2.5	2.5	2.5

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Pumps Engine driven pumps						
Fuel oil feed pump (5.5-7.5 bar)	m^3/h	1.4	1.4	1.4	1.4	1.4
HT circuit cooling water (1.0-2.5 bar)	m^3/h	45	45	60	60	60
LT circuit cooling water (1.0-2.5 bar)	m^3/h	45	60	75	75	75
Lube oil (3.0-5.0 bar)	m^3/h	24	24	34	34	34
External pumps [6)]						
Diesel oil pump (4 bar at fuel oil inlet A1)	m^3/h	0.78	0.93	1.09	1.24	1.40
Fuel oil supply pump (4 bar discharge pressure)	m^3/h	0.37	0.45	0.52	0.60	0.67
Fuel oil circulating pump (8 bar at fuel oil inlet A1)	m^3/h	0.78	0.93	1.09	1.24	1.40
HT circuit cooling water (1.0-2.5 bar)	m^3/h	37	45	50	55	60
LT circuit cooling water (1.0-2.5 bar) *	m^3/h	45	54	65	77	89
LT circuit cooling water (1.0-2.5 bar) **	m^3/h	65	73	95	105	115
Lube oil (3.0-5.0 bar)	m^3/h	22	23	25	27	28

1) Tolerance: + 10% for rating coolers, - 15% for heat recovery.

2) Basic values for layout of the coolers.

3) Under above mentioned reference conditions.

4) Tolerance: quantity +/- 5%, temperature +/- 20°C.

5) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions.

6) Tolerance of the pumps delivery capacities must be considered by the manufactures.

* Only valid for engines equipped with internal basic cooling water system no. 1 and 2.

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

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GenSet Data

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L23/30DF GenSet Data

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L28/32DF genset data

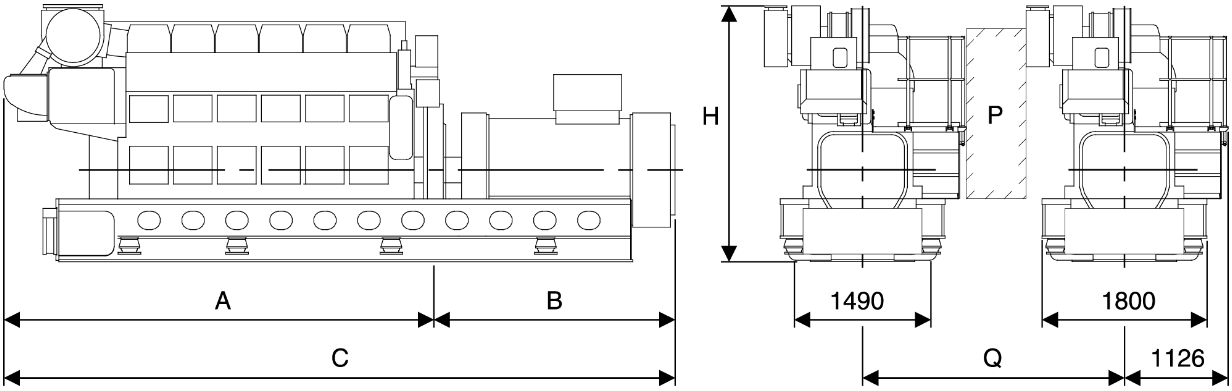
Engine ratings

Engine type No of cylinders	720 rpm	Available turning direction	750 rpm	Available turning direction
	kW	CW [1]	kW	CW [1]
5L28/32DF	1000	Yes	1000	Yes
6L28/32DF	1200	Yes	1200	Yes
7L28/32DF	1400	Yes	1400	Yes
8L28/32DF	1600	Yes	1600	Yes
9L28/32DF	1800	Yes	1800	Yes

[1] CW clockwise

B10011_3700283-2.1

General



Cyl. no	A (mm)	* B (mm)	* C (mm)	H (mm)	** Dry weight GenSet (t)
5 (720 rpm)	4321	2400	6721	2835	32.6
5 (750 rpm)	4321	2400	6721	2835	32.3
6 (720 rpm)	4801	2510	7311	3009	36.3
6 (750 rpm)	4801	2510	7311	3009	36.3
7 (720 rpm)	5281	2680	7961	3009	39.4
7 (750 rpm)	5281	2680	7961	3009	39.4
8 (720 rpm)	5761	2770	8531	3009	40.7
8 (750 rpm)	5761	2770	8531	3009	40.6
9 (720 rpm)	6241	2690	8931	3009	47.1
9 (750 rpm)	6241	2690	8931	3009	47.1

P Free passage between the engines, width 600 mm and height 2000 mm.
Q Min. distance between engines: 2655 mm (without gallery) and 2850 mm (with gallery).

* Depending on alternator
** Weight included a standard alternator

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

B10011_3700300-1.0

2022-03-02 - en

Capacities

5L-9L: 200 kW/Cyl. at 720 rpm			5	6	7	8	9
Engine output		kW	1000	1200	1400	1600	1800
Speed		rpm	720	720	720	720	720
Heat to be dissipated [3]							
Cooling water cylinder		kW	234	281	328	375	421
Charge air cooler; cooling water HT		kW	0	0	0	0	0
(Single stage charge air cooler)							
Charge air cooler; cooling water LT		kW	355	397	500	553	592
Lubricating oil cooler		kW	191	230	268	306	345
Heat radiation engine		kW	26	31	36	42	47
Flow rates [4]							
Internal (inside engine)							
HT cooling water cylinder		m^3/h	37	45	50	55	60
LT cooling water lube oil cooler		m^3/h	7.8	9.4	11	12.7	14.4
LT cooling water charge air cooler		m^3/h	37	45	55	65	75
Air data							
Temperature of charge air at charge air cooler outlet		°C	51	52	51	52	53
Air flow rate		m^3/h [5]	7355	8826	10297	11768	13239
		kg/kWh	7.67	7.67	7.67	7.67	7.67
Charge air pressure		bar	2.97	2.97	2.97	2.97	2.97
Air required to dissipate heat radiation (engine) (t_2-t_1= 10°C)		m^3/h	8425	10045	11665	13609	15230
Exhaust gas data [6]							
Volume flow (temperature turbocharger outlet)		m^3/h [7]	14711	17653	20595	23537	26479
Mass flow		t/h	8.3	9.9	11.6	13.2	14.9
Temperature at turbine outlet		°C	347	347	347	347	347
Heat content (190°C)		kW	389	467	545	623	701
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)		mbar	< 50	< 50	< 50	< 50	< 50
Starting air system							
Air consumption per start		Nm^3	2.5	2.5	2.5	2.5	2.5
Pumps							
Engine driven pumps							
Fuel oil feed pump	(5.5-7.5 bar)	m^3/h	1.4	1.4	1.4	1.4	1.4
HT circuit cooling water	(1.0-2.5 bar)	m^3/h	45	45	60	60	60
LT circuit cooling water	(1.0-2.5 bar)	m^3/h	45	60	75	75	75
Lube oil	(3.0-5.0 bar)	m^3/h	24	24	34	34	34
External pumps [8]							
Diesel oil pump	(4 bar at fuel oil inlet A1)	m^3/h	0.74	0.89	1.04	1.19	1.34
Fuel oil supply pump	(4 bar discharge pressure)	m^3/h	0.36	0.43	0.50	0.57	0.64
Fuel oil circulating pump [9]	(8 bar at fuel oil inlet A1)	m^3/h	0.74	0.89	1.04	1.19	1.34
HT circuit cooling water	(1.0-2.5 bar)	m^3/h	37	45	50	55	60
LT circuit cooling water	(1.0-2.5 bar)	m^3/h	45	54	65	77	89
Lube oil	(3.0-5.0 bar)	m^3/h	22	23	25	27	28

1) Tolerance: + 10 % for rating coolers, - 15 % for heat recovery

2) Basic values for layout of the coolers

3) Under above mentioned reference conditions

4) Tolerance: quantity +/- 5%, temperature +/- 20°C

5) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions

6) Tolerance of the pumps delivery capacities must be considered by the manufactures

* Only valid for engines equipped with internal basic cooling water system no. 1 and 2.

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

D10050_3700325-3.0

Capacities

5L-9L: 200 kW/Cyl. at 750 rpm			5	6	7	8	9
Engine output		kW	1000	1200	1400	1600	1800
Speed		rpm	750	750	750	750	750
Heat to be dissipated [3]							
Cooling water cylinder		kW	245	294	343	392	442
Charge air cooler; cooling water HT		kW	0	0	0	0	0
(Single stage charge air cooler)							
Charge air cooler; cooling water LT		kW	387	435	545	587	648
Lubricating oil cooler		kW	201	241	281	321	361
Heat radiation engine		kW	27	33	38	44	49
Flow rates [4]							
Internal (inside engine)							
HT cooling water cylinder		m^3/h	37	45	50	55	60
LT cooling water lube oil cooler		m^3/h	7.8	9.4	11	12.7	14.4
LT cooling water charge air cooler		m^3/h	37	45	55	65	75
Air data							
Temperature of charge air at charge air cooler outlet		°C	52	54	52	52	55
Air flow rate		m^3/h [5]	7826	9391	10956	12521	14087
		kg/kWh	7.79	7.79	7.79	7.79	7.79
Charge air pressure		bar	3.07	3.07	3.07	3.07	3.07
Air required to dissipate heat radiation (engine) (t_2-t_1= 10°C)		m^3/h	8749	10693	12313	14257	15878
Exhaust gas data [6]							
Volume flow (temperature turbocharger outlet)		m^3/h [7]	15520	18624	21728	24832	27936
Mass flow		t/h	8.8	10.5	12.3	14.1	15.8
Temperature at turbine outlet		°C	342	342	342	342	342
Heat content (190°C)		kW	401	481	561	641	721
Permissible exhaust back pressure		mbar	< 30	< 30	< 30	< 30	< 30
Permissible exhaust back pressure (SCR)		mbar	< 50	< 50	< 50	< 50	< 50
Starting air system							
Air consumption per start		Nm^3	2.5	2.5	2.5	2.5	2.5
Pumps							
Engine driven pumps							
Fuel oil feed pump	(5.5-7.5 bar)	m^3/h	1.4	1.4	1.4	1.4	1.4
HT circuit cooling water	(1.0-2.5 bar)	m^3/h	45	45	60	60	60
LT circuit cooling water	(1.0-2.5 bar)	m^3/h	45	60	75	75	75
Lubrication oil	(3.0-5.0 bar)	m^3/h	24	24	34	34	34
External pumps [8]							
Diesel oil pump	(4 bar at fuel oil inlet A1)	m^3/h	0.78	0.93	1.09	1.24	1.40
Fuel oil supply pump	(4 bar discharge pressure)	m^3/h	0.37	0.45	0.52	0.60	0.67
Fuel oil circulating pump [9]	(8 bar at fuel oil inlet A1)	m^3/h	0.78	0.93	1.09	1.24	1.40
HT circuit cooling water	(1.0-2.5 bar)	m^3/h	37	45	50	55	60
LT circuit cooling water	(1.0-2.5 bar)	m^3/h	45	54	65	77	89
Lubrication oil	(3.0-5.0 bar)	m^3/h	22	23	25	27	28

1) Tolerance: + 10 % for rating coolers, - 15 % for heat recovery

2) Basic values for layout of the coolers

3) Under above mentioned reference conditions

4) Tolerance: quantity +/- 5%, temperature +/- 20°C

5) Under below mentioned temperature at turbine outlet and pressure according above mentioned reference conditions

6) Tolerance of the pumps delivery capacities must be considered by the manufactures

* Only valid for engines equipped with internal basic cooling water system no. 1 and 2.

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

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Space requirements and overhaul

General

The latest version of the Installation Drawings of this section is available for download at www.marine.man-es.com--> 'Two -Stroke' --> 'Installation Drawings'. Specify engine and accept the 'Conditions for use' before clicking on 'Download Drawings'.

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.

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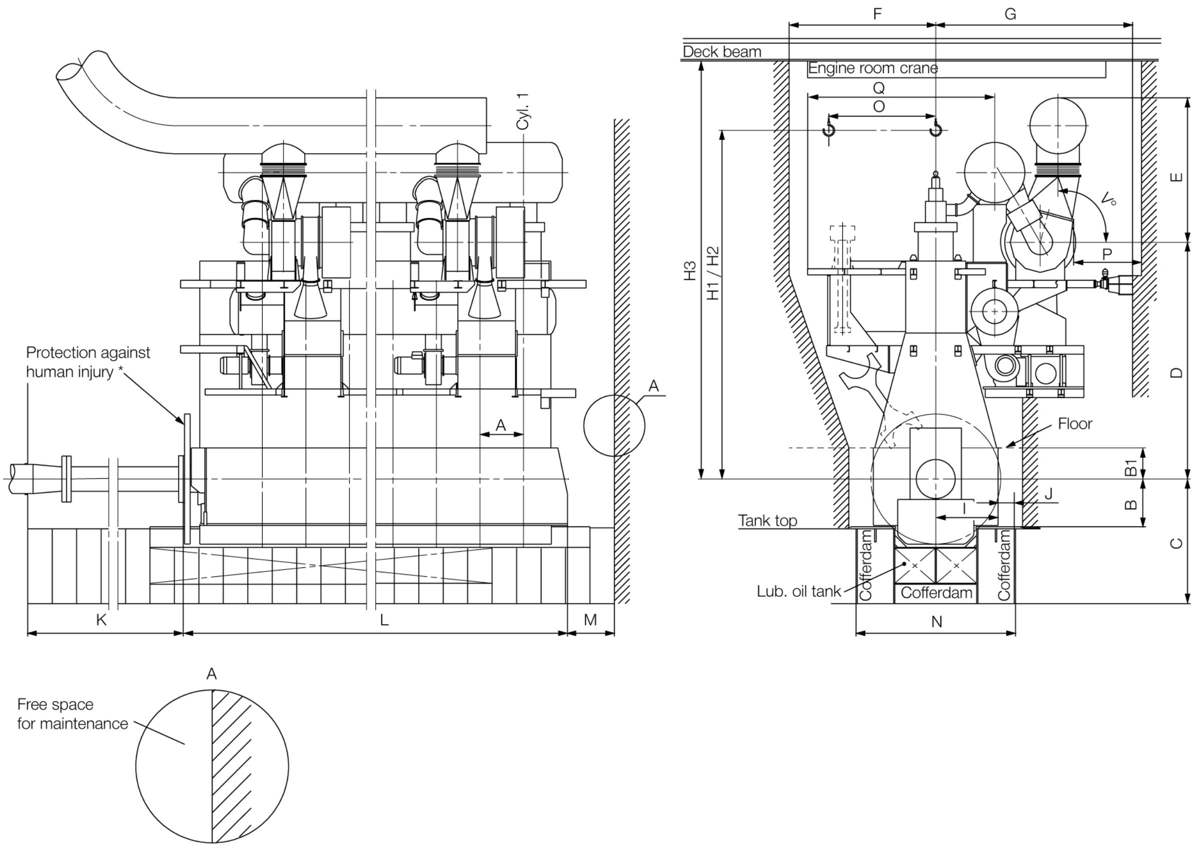
Space requirement

Engine space requirement turbocharger(s) mounted on the exhaust side



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Minimum access conditions around the engine to be used for an escape route is 600 mm.

The dimensions are given in mm, and are for guidance only. If the dimensions cannot be fulfilled, please contact MAN Energy Solutions or our local representative.

* To avoid human injury from rotating turning wheel, the turning wheel has to be shielded or access protected (Yard supply).

Fig. 5.02.01: Space requirement for the engine, turbocharger(s) mounted on the exhaust side, 4 59 122

5.02 Space requirement

Dimensions of exhaust side

Cylinder No.	6	7	8	9		
A	1,400					Cylinder distance. See drawing 'Outline drawing'
B	2,010					Distance from crankshaft centre line to foundation. See drawing 'Engine seating'
B1	966					Distance from crankshaft centre line to floor. See drawing 'Outline drawing'
C	4,207	4,272	4,337	4,447		The dimension includes a cofferdam of 600 mm and must fulfil minimum height to tank top according to classification rules. See drawing 'Lubricating oil bottom tank'
D *)	10,080	9,635	9,635	10,080	MAN TCA/TCR/TCT	Dimensions according to turbocharger choice at nominal MCR in Tier II mode. See drawing 'Outline drawing' for the specified dimensions in Tier II or III mode.
	9,220	9,220	9,322	9,322	Accelleron A100/A200	
	10,040	9,350	9,350	9,350	MHI MET	
E *)	See text					Height of exhaust pipe is according to engine room design
F	See text					See drawing: 'Engine Top Bracing', if top bracing fitted on camshaft side
G		6,075	6,075		MAN TCA/TCR/TCT	The required space to the engine room casing includes hydraulic top bracing. Dimensions according to turbocharger choice at nominal MCR in Tier II mode. See drawing 'Top bracing' for the specified dimensions in Tier II or III mode.
	5,875	5,875	6,075	6,075	Accelleron A100/A200	
		6,075	6,075	6,075	MHI MET	
H1 *)	15,750					Minimum overhaul height, normal lifting procedure. See drawing 'Engine room crane'
H4 *)	15,775					Minimum overhaul height, normal lifting procedure, with MAN B&W Double Jib Crane. See drawing 'Engine room crane'
I	2,460					Length from crankshaft centre line to outer side bedplate. See drawing 'Engine seating'
J	510					Space for tightening control of holding down bolts. See drawing 'Engine seating'
K	See text					K must longer than the propeller shaft, if the propeller shaft is to be drawn into the engine room
L *)	12,410	12,629	15,045	16,445		Minimum length of a basic engine, without 2nd order moment compensators. See drawing 'Outline drawing'
M	≈ 800					Free space in front of engine
N	5,910					Distance between outer foundations girders. See drawing 'Engine seating'
O	2,800					Minimum crane operation area. See drawing 'Outline drawing'

Cylinder No.	6	7	8	9	
P	See text				See drawing 'Crane beam for Turbocharger' for overhaul of turbocharger
Q	See text				Recommended crane operation area. See drawing 'Outline drawing'
V	0°, 15°, 30°, 45°, 60°, 75°				Maximum 30° when engine room has minimum headroom above the turbocharger

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*) The min. engine room crane height is ie. dependent on the choice of crane, see the actual heights “H1” or “H4”
The min. engine room height is dependent on “H1”, “H4” or “E+D”. ..
Max. length of engine see the engine outline drawing.
Length of engine with PTO see corresponding space requirement.
For the description H1 and H4, see drawing 5887764-6

Table 5.02.01: Space requirement for the engine, turbocharger(s) mounted on the exhaust side

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Crane beam requirements - turbocharger and air cooler

General

If the travelling area of the engine room crane covers the recommended area in the Engine Room Crane drawing, Fig. 5.04.01, crane beams can be omitted for the overhaul of turbocharger. If not, a crane beam with trolleys is required at each end of the turbocharger(s).

Crane beam and trolleys

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

as indicated in Figs. 5.03.01 and 5.03.02.

Lifting capacity

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).

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 table 5.03.01 for the various turbocharger makes and types.

The crane beam shall be dimensioned for lifting the weight 'W' with a deflection of some 5 mm only.

Relative position of the crane hook

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: 4 59 122, the letter 'a' indicates the distance between vertical centrelines of the engine and the turbocharger.

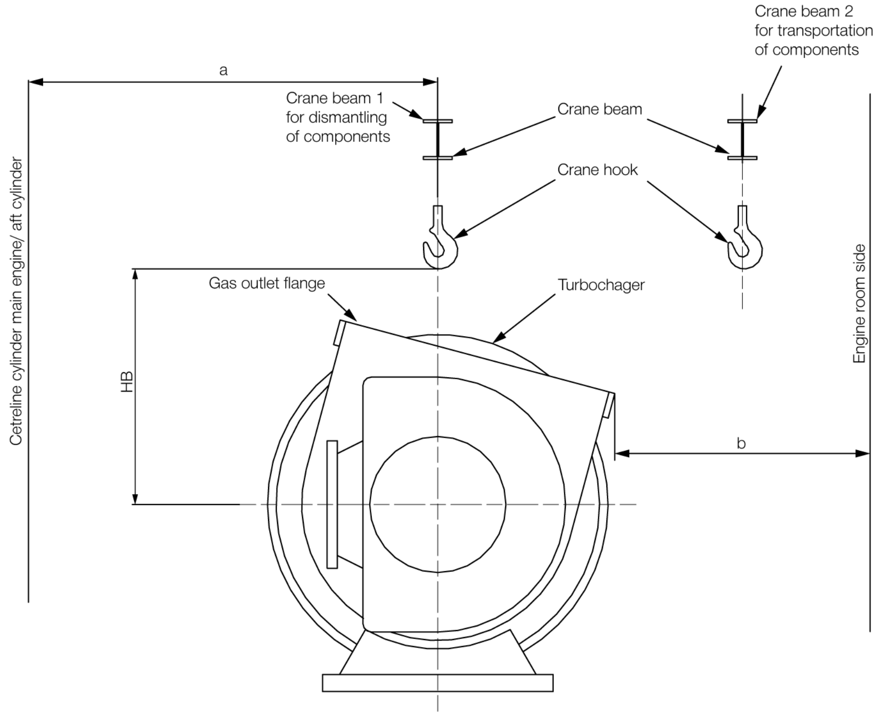

Fig. 5.03.01: Required height and distance

MAN

Turbocharger	W kg	HB mm	b mm
TCR18	1,500	760	500
TCR20	1,500	1,000	500
TCR22	1,500	1,200	500
TCA44	1,000	1,200	500
TCA55	1,000	1,384	600
TCA66	1,200	1,608	700
TCA77	2,000	1,700	800
TCA88	3,080	2,040	1,000
TCT30	500	1,200	700
TCT40	500	1,300	800
TCT50	750	1,450	900
TCT60	1,000	1,600	1,000
TCT70	1,500	1,750	1,100
TCT80	2,000	1,950	1,250

Accelleron

Turbocharger	W kg	HB mm	b mm
A165-L	750	1,250	800
A170-L	1,000	1,450	950
A175-L	1,470	1,750	1,100
A180-L	1,950	2,000	1,250
A185-L	2,550	2,200	1,400
A255-L	530	1,025	700
A260-L	530	1,200	700
A265-L	750	1,475	800
A270-L	1,000	1,750	950
A275-L	1,470	2,000	1,100
A280-L	1,950	2,225	1,250

Mitsubishi (MHI)

Turbocharger	W kg	HB mm	b mm
MET18	1,000	1,000	500
MET22	1,000	1,000	500
MET26	1,000	1,500	500
MET30	1,000	1,500	500
MET33	1,000	1,500	600
MET37	1,000	1,500	600
MET42	1,000	1,500	600
MET48	1,000	1,500	700
MET53	1,000	1,500	700
MET60	1,000	1,600	700
MET66	1,500	1,800	800
MET71	1,800	1,800	800
MET83	2,700	2,000	1,000
MET90	3,500	2,200	1,000

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The figures ‘a’ are stated in the ‘Engine and gallery outline’ drawing, Section 5.06.

Table 5.03.01: Required height, distance and weight

Crane beam for turbochargers

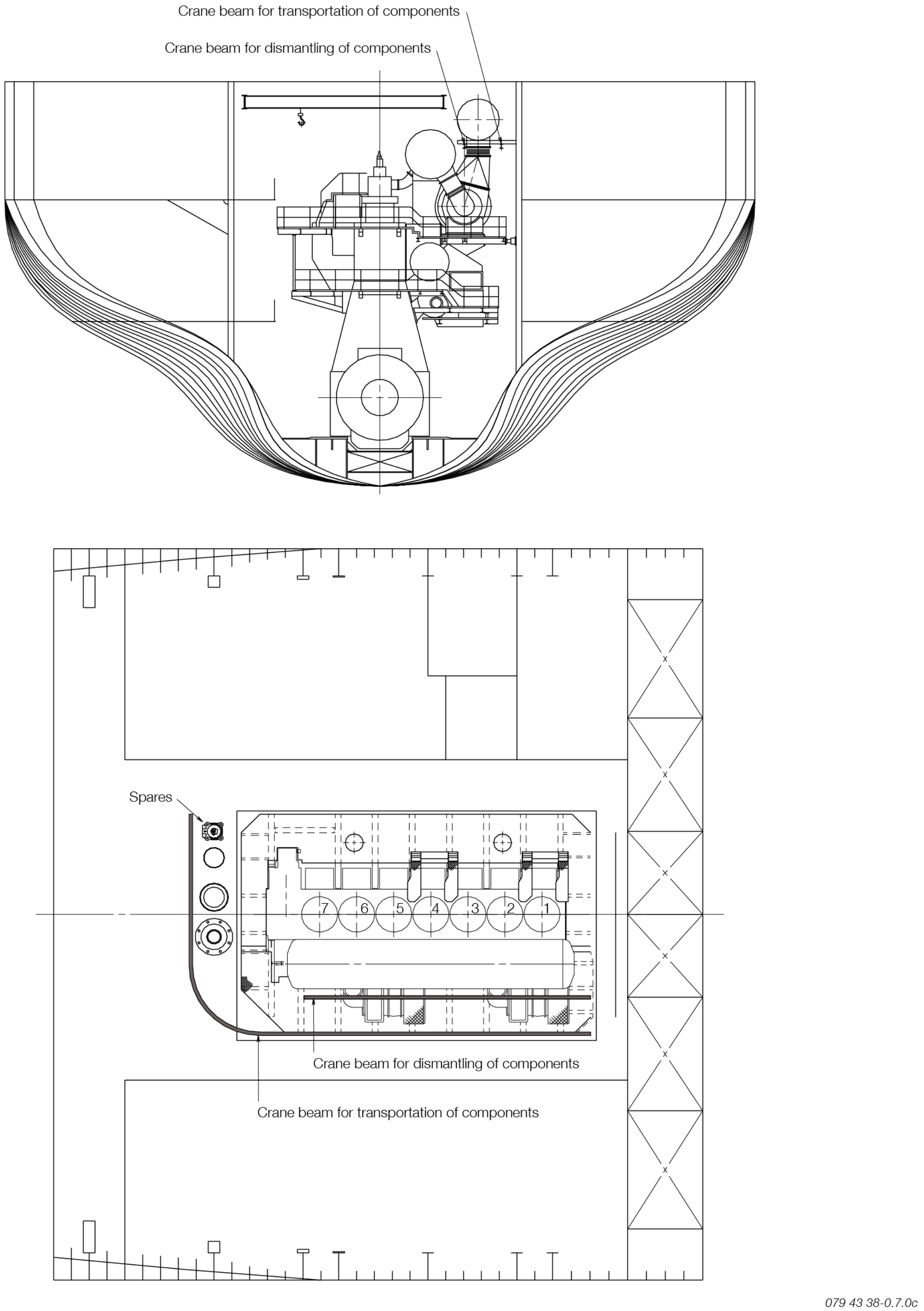

Fig. 5.03.02: Crane beam for turbocharger

Crane beam for overhaul of EGR cooler, turbocharger on exhaust side

Overhaul/exchange of scavenge air and EGR 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 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 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 cooler insert can be lifted out of the engine room.

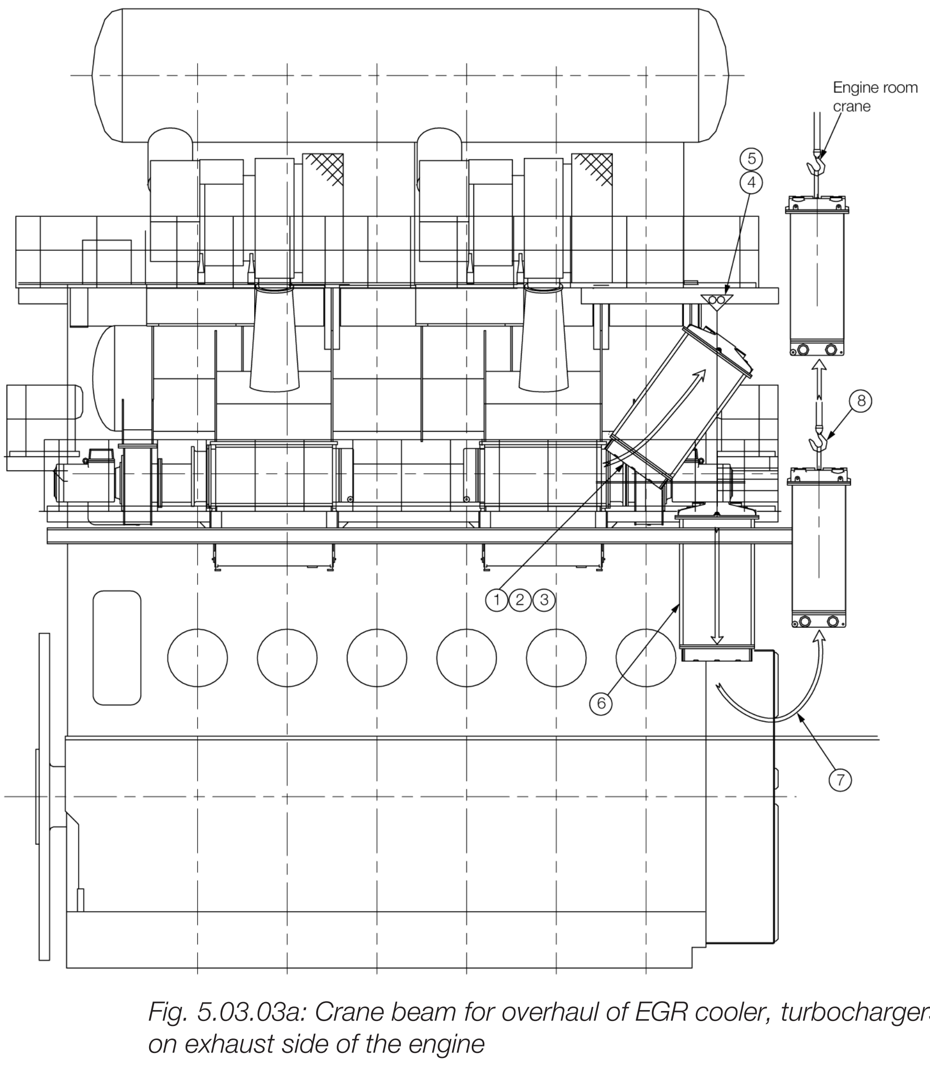

Fig. 5.03.03a: Crane beam for overhaul of EGR cooler, turbochargers located on exhaust side of the engine

Overhaul/exchange of scavenge air and EGR cooler.

The text and figure are for guidance only.

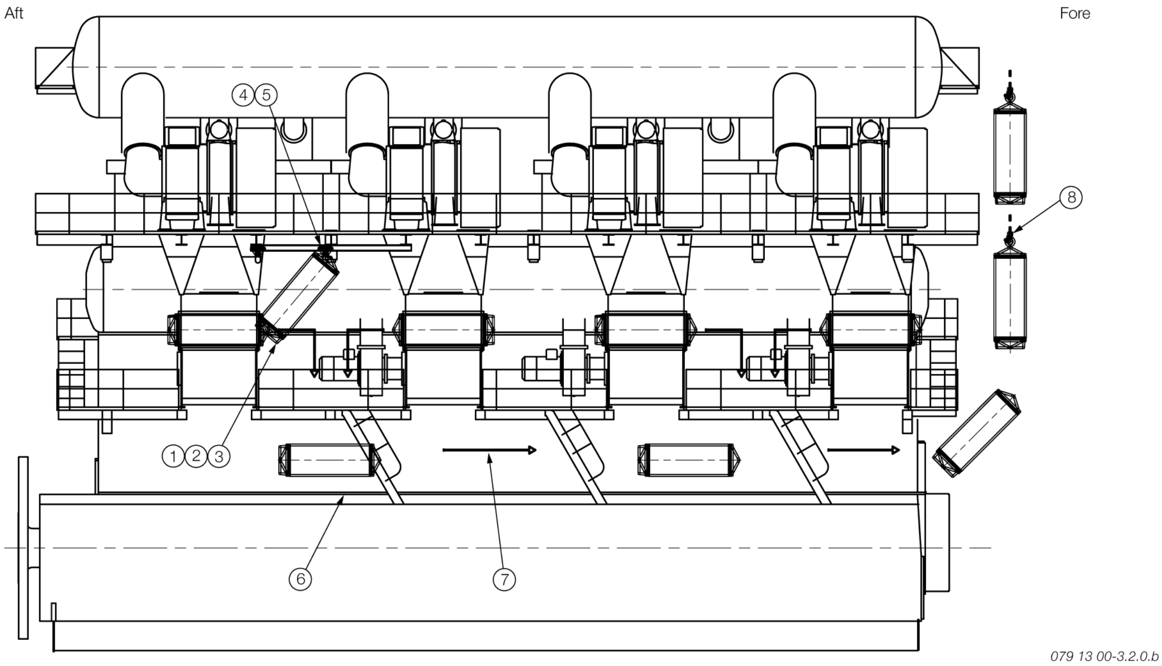

Fig. 5.03.03b: Crane beam for overhaul of EGR cooler, turbochargers located on exhaust side of the engine

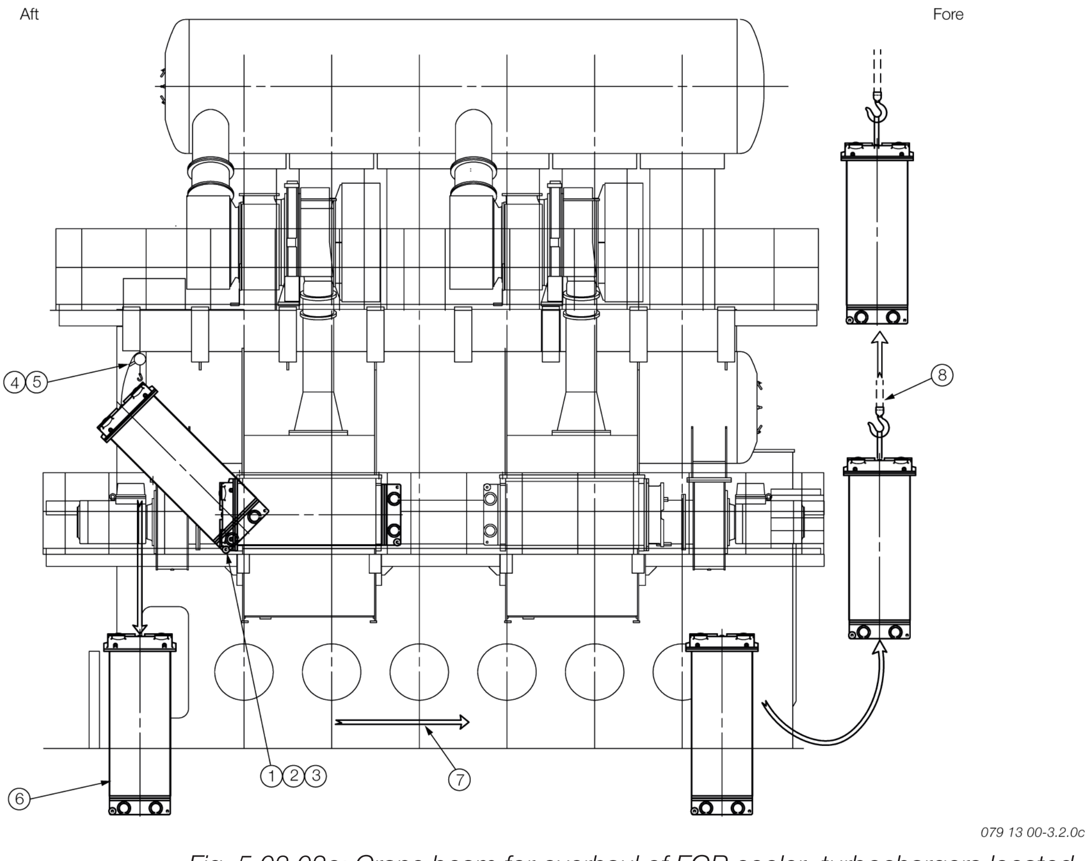

Fig. 5.03.03c: Crane beam for overhaul of EGR cooler, turbochargers located on exhaust side of the engine

Overhaul/exchange of scavenge air and EGR cooler.

The text and figure are for guidance only.

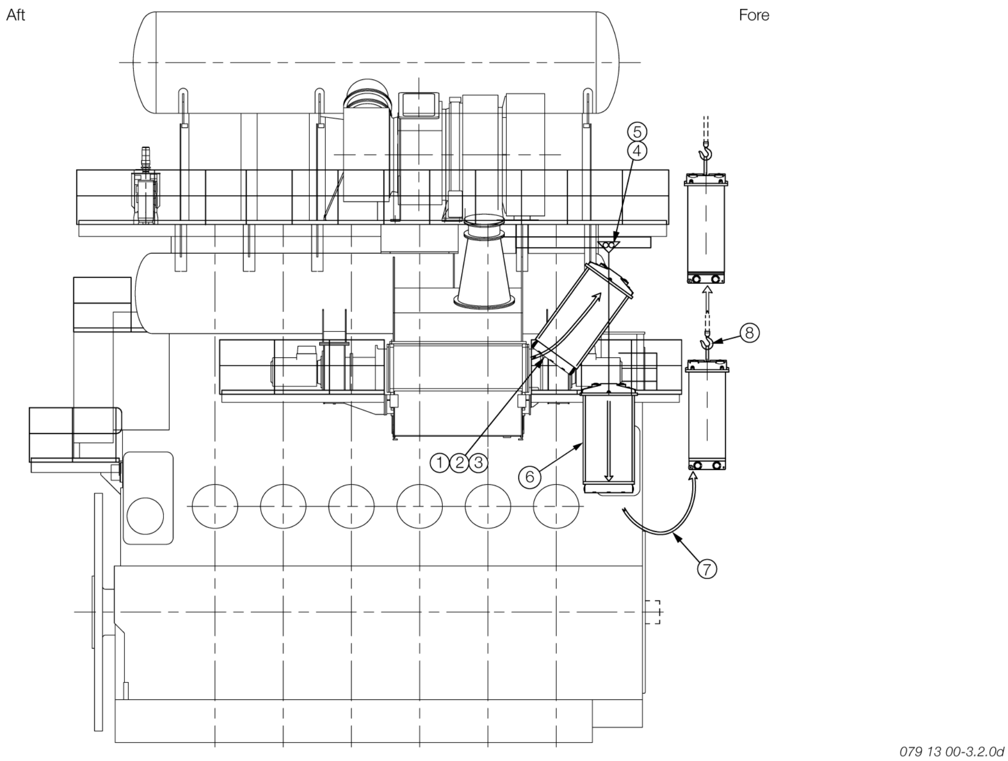

Fig. 5.03.03d: Crane beam for overhaul of EGR cooler, turbochargers located on exhaust side of the engine

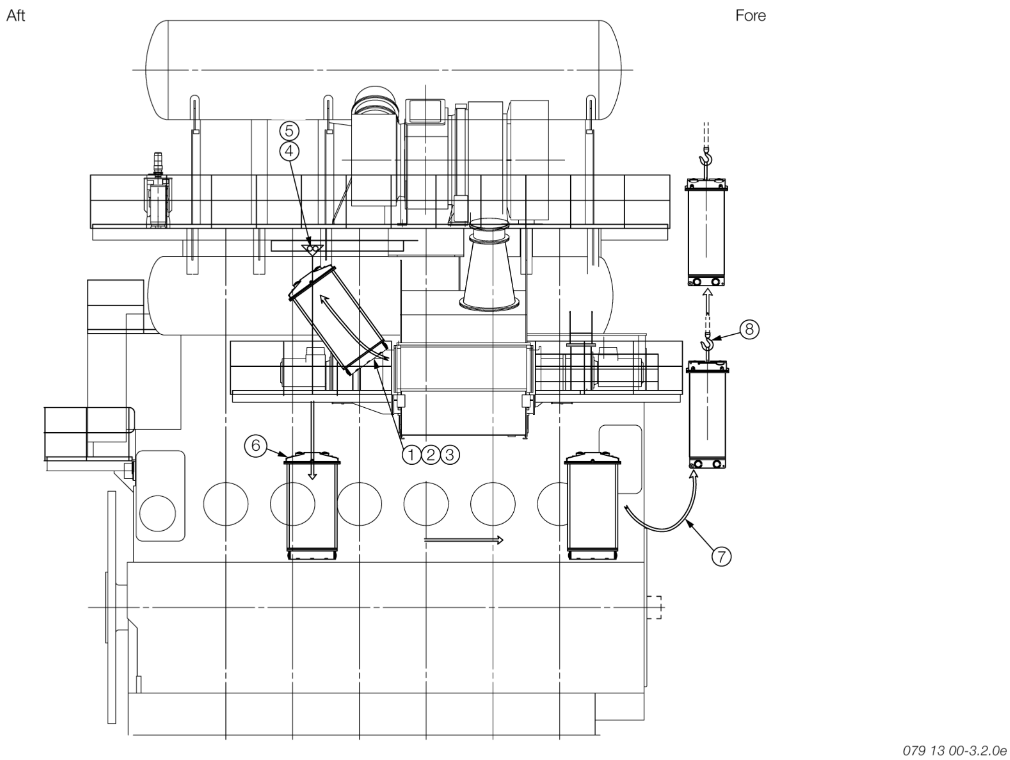

Fig. 5.03.03e: Crane beam for overhaul of EGR cooler, turbochargers located on exhaust side of the engine

Crane beam for overhaul of air cooler, turbocharger on aft end

This section is not applicable for 95-60 bore engines.

Engine room cranes - requirements and applications

General

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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.

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Engine and gallery outline - TIII

General

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Centre of gravity - TIII

General

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Mass of water and oil - TIII

General

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Engine pipe connections - TIII

General

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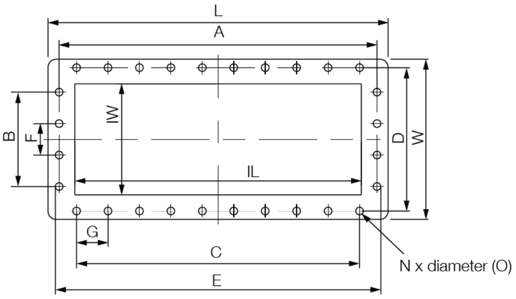
Counterflanges, Connections D and E

MAN Type TCA44-88



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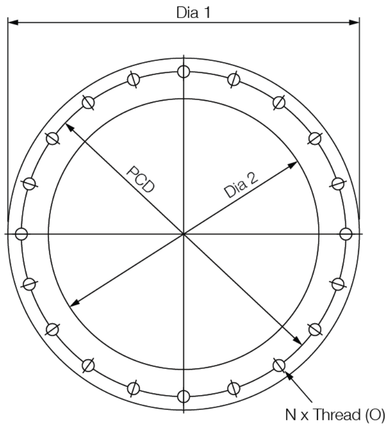
Type TCA series – Rectangular type													
TC	L	W	IL	IW	A	B	C	D	E	F	G	N	O
TCA44	1,054	444	949	340	1,001	312	826	408	1,012	104	118	24	ø13.5
TCA55	1,206	516	1,080	390	1,143	360	1,000	472	1,155	120	125	26	ø17.5
TCA66	1,433	613	1,283	463	1,358	420	1,200	560	1,373	140	150	26	ø17.5
TCA77	1,694	720	1,524	550	1,612	480	1,440	664	1,628	160	160	28	ø22
TCA88	2,012	855	1,810	653	1,914	570	1,710	788	1,934	190	190	28	ø22
TCA99	2,207	938	1,985	717	2100	624	1,872	866	2,120	208	208	28	ø22

MAN Type TCR



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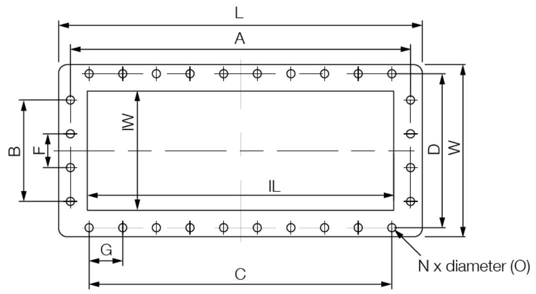
Type TCR series – Round type					
TC	Dia 1	Dia 2	PCD	N	O
TCR18	425	310	395	12	ø22
TCR20	540	373	495	16	ø22
TCR22	703	487	650	20	ø22

Fig. 5.10.01a and b: Turbocharger MAN TCA and TCR, exhaust outlet, connection D

ABB Type A100/A200-L



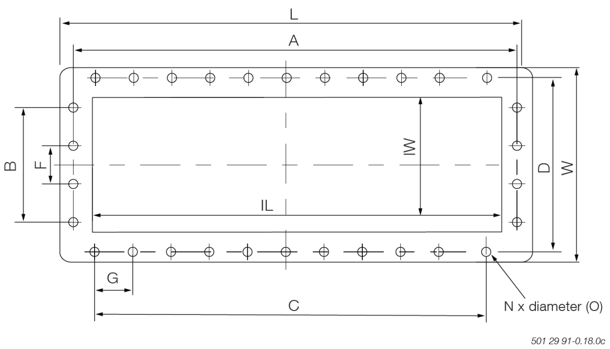
Type A100/200-L series – Rectangular type												
TC	L	W	IL	IW	A	B	C	D	F	G	N	O
A255-L	Available on request											
A260-L												
A165/A265-L	1,114	562	950	404	1,050	430	900	511	86	100	32	ø22
A170/A270-L	1,280	625	1,095	466	1,210	450	1,080	568	90	120	32	ø22
A175/A275-L	1,523	770	1,320	562	1,446	510	1,260	710	170	140	28	ø30
A180/A280-L	1,743	856	1,491	634	1,650	630	1,485	786	150	135	36	ø30
A185-L	1,955	958	1,663	707	1,860	725	1,595	886	145	145	36	ø30

Fig. 5.10.01c: Turbocharger ABB A100/200-L, exhaust outlet, connection D

MHI Type MET

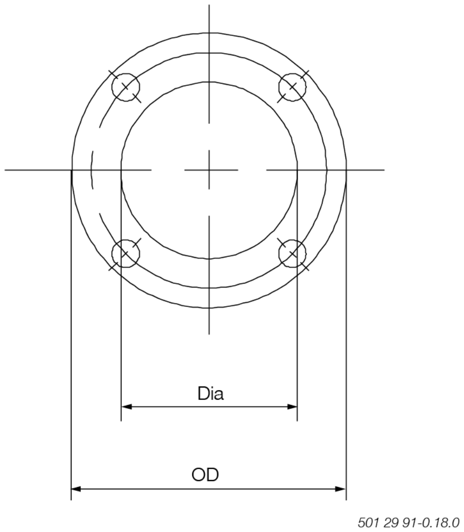
Type MET – Rectangular type											
TC	L	W	IL	IW	A	B	C	D	F	G	N	O
Series MB												
MET33	Available on request											
MET37	999	353	909	263	969	240	855	323	80	95	28	ø15
MET42	1,094	381	1,004	291	1,061	261	950	351	87	95	30	ø15
MET48	1,240	430	1,140	330	1,206	300	1,070	396	100	107	30	ø15
MET53	1,389	485	1,273	369	1,340	330	1,200	440	110	120	30	ø20
MET60	1,528	522	1,418	410	1,488	330	1,320	482	110	110	34	ø20
MET66	1,713	585	1,587	459	1,663	372	1,536	535	124	128	34	ø20
MET71	1,837	617	1,717	497	1,792	480	1,584	572	120	132	36	ø20
MET83	2,163	731	2,009	581	2,103	480	1,920	671	160	160	34	ø24
MET90	2,378	801	2,218	641	2,318	525	2,100	741	175	175	34	ø24
Series MA												
MET33	700	310	605	222	670	180	550	280	90	110	18	ø15
MET42	883	365	793	275	850	240	630	335	80	90	24	ø15
MET53	1,122	465	1,006	349	1,073	300	945	420	100	105	28	ø20
MET60	1,230	500	1,120	388	1,190	315	1,050	460	105	105	30	ø20
MET66	1,380	560	1,254	434	1,330	345	1,200	510	115	120	30	ø20
MET71	1,520	600	1,400	480	1,475	345	1,265	555	115	115	34	ø20
MET83	1,740	700	1,586	550	1,680	450	1,500	640	150	150	30	ø24
MET90	1,910	755	1,750	595	1,850	480	1,650	695	160	165	30	ø24

Fig. 5.10.01d: Turbocharger MHI MET MB and MA, exhaust outlet, connection D

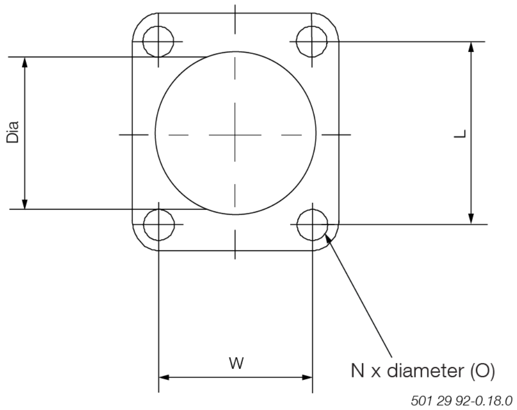


Counterflanges, Connection E

MAN Type TCA

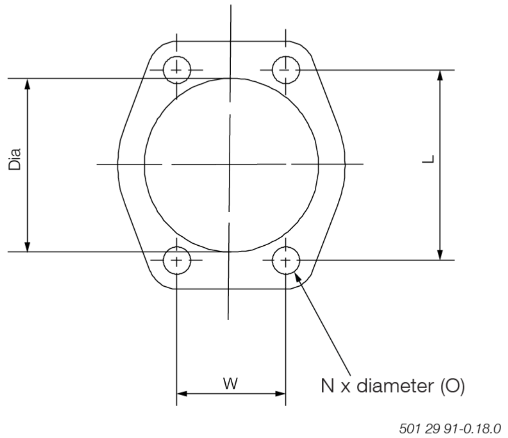


TC	Dia/ISO	Dia/JIS	OD	PCD	N	O	Thickness of flanges
TCA44	61	77	120	90	4	ø14	14



TC	Dia/ISO	Dia/JIS	L	W	N	O	Thickness of flanges
TCA55	61	77	86	76	4	ø14	16
TCA66	90	90	110	90	4	ø18	16

Fig. 5.10.01e and f: Turbocharger MAN TCA, venting of lube oil discharge pipe, connection E



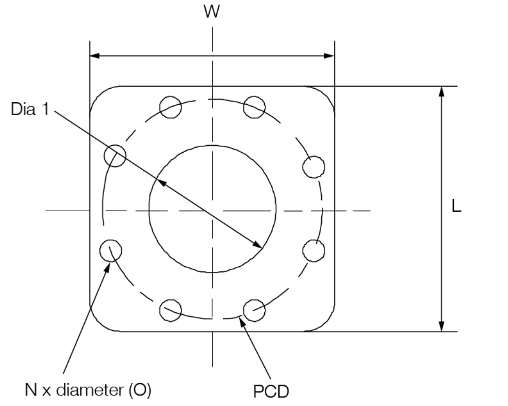
TC	Dia/ISO	Dia/JIS	L	W	N	O	Thickness of flanges
TCA77	115	115	126	72	4	ø18	18
TCA88	141	141	150	86	4	ø18	18
TCA99	141	141	164	94	4	ø22	24

Fig. 5.10.01g: Turbocharger MAN TCA, venting of lube oil discharge pipe, connection E

ABB Type A100/A200-L

TC	Dia 1	PCD	L = W	N	O	Thickness of flanges
A255-L	Available on request					
A260-L						
A165/A265-L	43	100	106	8	ø8.5	18
A170/A270-L	77	100	115	8	ø11	18
A175/A275-L	77	126	140	8	ø11	18
A180/A280-L	90	142	158	8	ø13	18
A185-L	115	157	178	8	ø13	18

Fig. 5.10.01h: Turbocharger ABB A100/200-L, venting of lube oil discharge pipe, connection E



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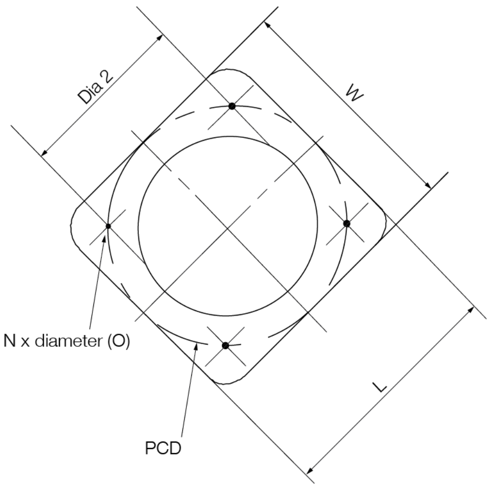
MHI Type MET MB



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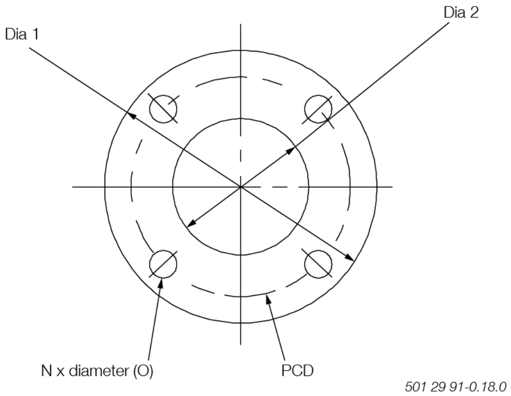
TC	L = W	Dia 2	PCD	N	O	Thickness of flanges (A)
MET33MB	Available on request					
MET42MB	105	61	105	4	ø14	14
MET48MB	125	77	130	4	ø14	14
MET53MB	125	77	130	4	ø14	14
MET60MB	140	90	145	4	ø18	14
MET66MB	140	90	145	4	ø18	14



TC	Dia 1	Dia 2	PCD	N	O	Thickness of flanges (A)
MET71MB	180	90	145	4	ø18	14
MET83MB	200	115	165	4	ø18	16
MET90MB	200	115	165	4	ø18	16

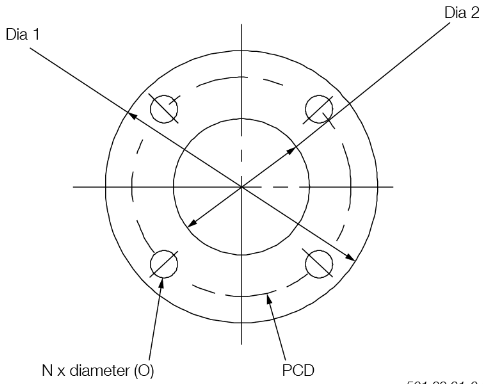
Fig. 5.10.01i and j: Turbocharger MHI MET MB, venting of lube oil discharge pipe, connection E

MHI Type MET MA

TC	L=W	Dia 2	PCD	N	O	Thickness of flanges (A)
MET33MA	Available on request					
MET42MB	105	61	105	4	ø14	14
MET53MB	125	77	130	4	ø14	14
MET60MB	140	90	145	4	ø18	14
MET66MB	140	90	145	4	ø18	14
MET71MB	140	90	145	4	ø18	14
MET90MB	155	115	155	4	ø18	14

TC	Dia 1	Dia 2	PCD	N	O	Thickness of flanges (A)
MET83MB	180	90	145	4	ø18	148

Fig. 5.10.01k and l: Turbocharger MHI MET MA, venting of lube oil discharge pipe, connection E



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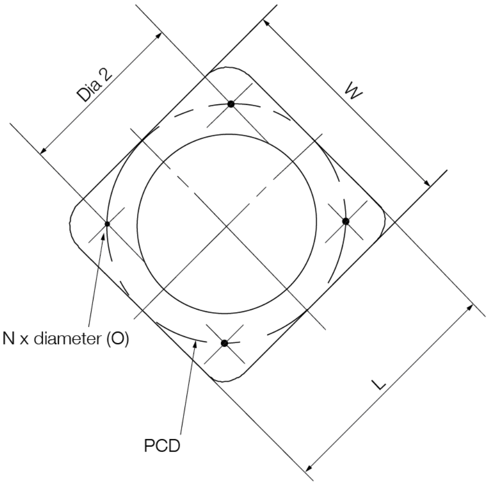

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Counterflanges, connection EB

MHI Type MET MB

TC	Dia 1	Dia 2	PCD	N	O	Thickness of flanges (A)
MET42MB	95	43	75	4	ø12	10
MET60MB	120	49	95	4	ø14	12
MET66MB	120	49	95	4	ø14	12
MET71MB	120	49	95	4	ø14	12
MET83MB	120	49	95	4	ø14	12

TC	L = W	Dia 2	PCD	N	O	Thickness of flanges (A)
MET48MB	95	49	95	4	ø14	12
MET53MB	95	49	95	4	ø14	12
MET90MB	125	77	130	4	ø14	14

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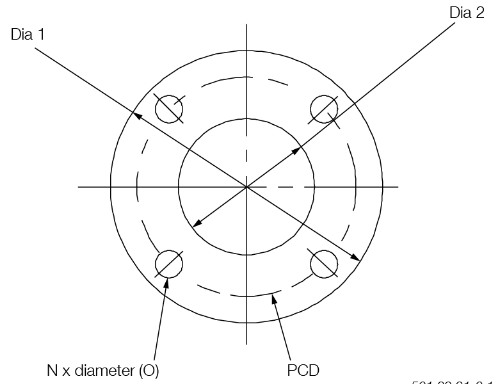
Fig. 5.10.01m and n: Turbocharger MHI MB, cooling air, connection EB



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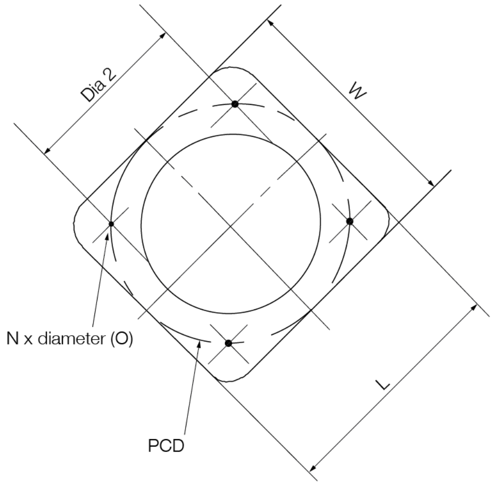

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Engine seating and arrangement of holding down bolts

General

The latest version of the Installation Drawings of this section is available for download at www.marine.man-es.com --> 'Two-Stroke' --> Installation Drawings'. Specify engine and accept the 'Conditions for use' before clicking on 'Download Drawings'.

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 Energy Solutions are:

- 'Chockfast Orange PR 610 TCF'
  and 'Epocast 36' from ITW Philadelphia Resins Corporation, USA.
- 'Durasin' from
  Daemmstoff Industrie Korea Ltd.
- 'EPY' from
  Marine Service Jaroszewicz S.C., Poland.
- 'Loctite Fixmaster Marine Chocking', Henkel.
- 'CMP Liner Blue' from
  Chugoku Marine Paints Ltd, Japan.

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Epoxy chocks arrangement - TIII

General

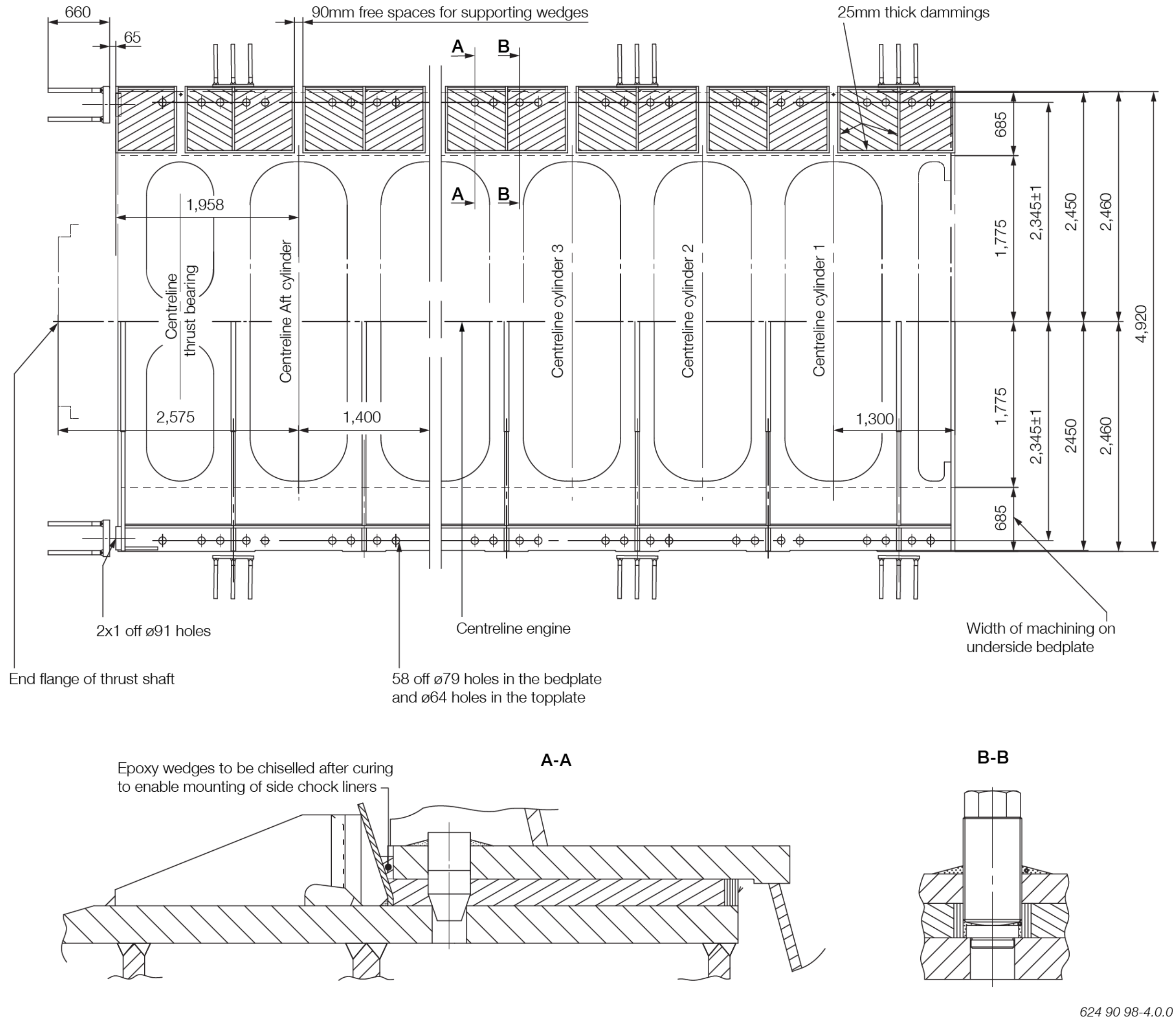


Fig. 5.12.01: Arrangement of epoxy chocks and holding down bolts

For details of chocks and bolts see special drawings. For securing of supporting chocks see special drawing.

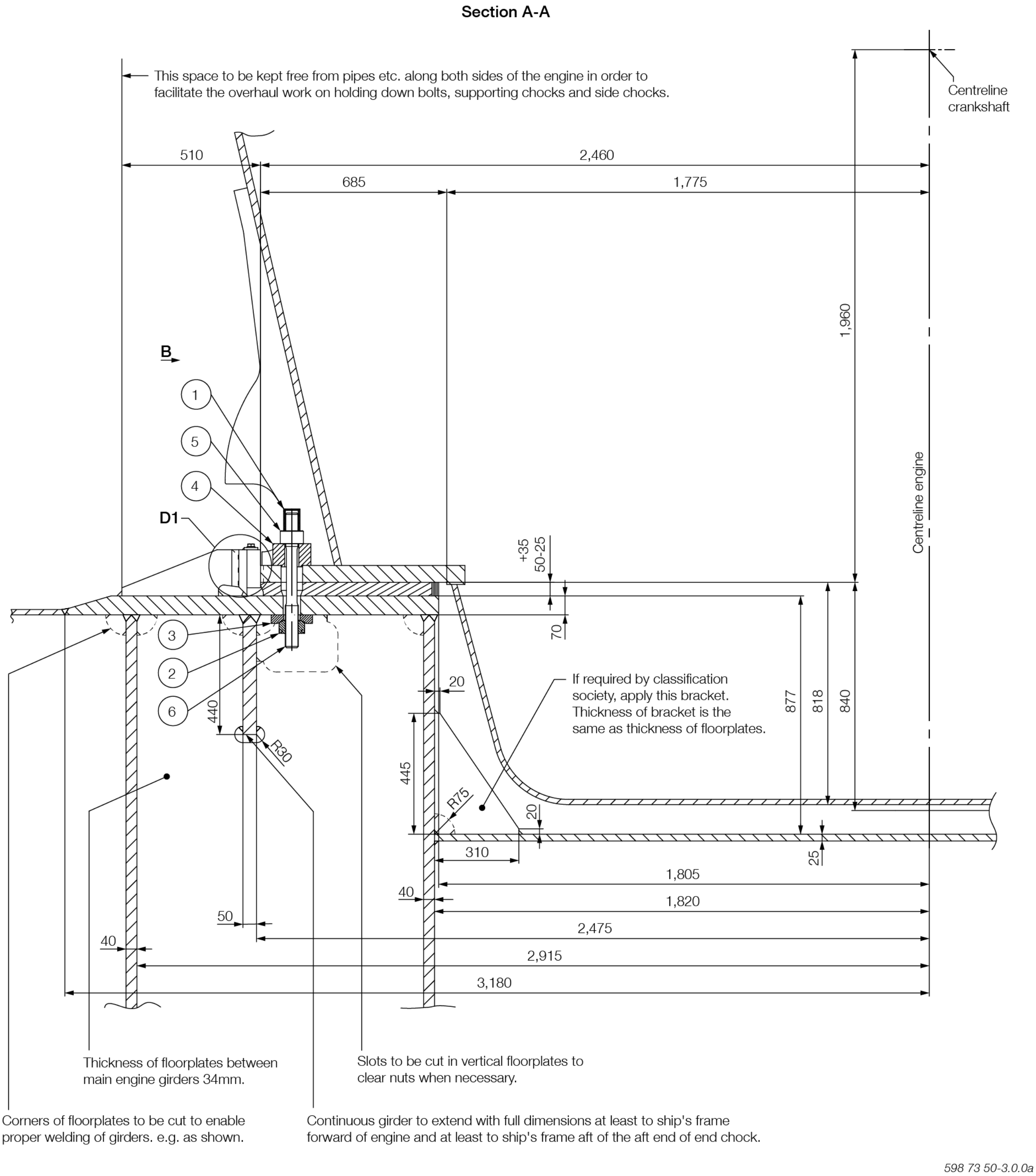
Preparing holes for holding down bolts

1) The engine builder drills the holes for holding down bolts in the bedplate while observing the toleranced locations indicated on MAN Energy Solutions' drawings for machining the bedplate

2) The shipyard drills the holes for holding down bolts in the top plates while observing the toleranced locations given on the present drawing

3) The holding down bolts must be made in accordance with MAN Energy Solutions' drawings of these bolts.

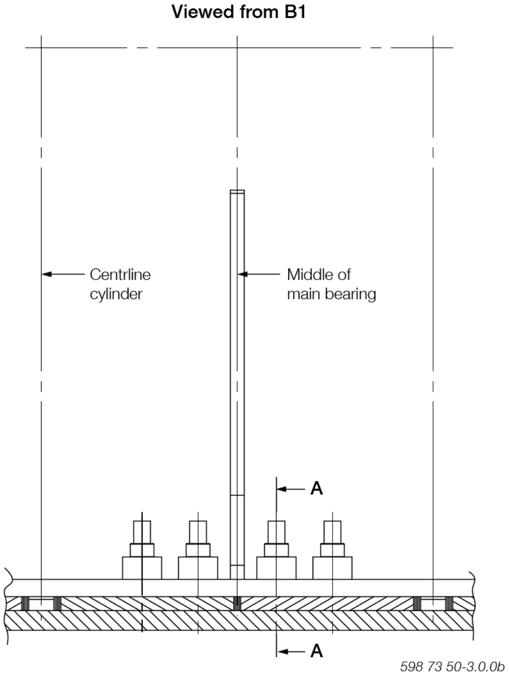
Engine seating profile



Holding down bolts, option: 4 82 602 include:

1. Protecting cap
2. Spherical nut
3. Spherical washer
4. Distance pipe
5. Round nut
6. Holding down bolt

Fig. 5.12.02a: Profile of engine seating with vertical lubricating oil outlet



End chock bolts, option: 4 82 610 includes:

1. Stud for end chock bolt
2. Round nut
3. Round nut
4. Spherical washer
5. Spherical washer
6. Protecting cap

End chock liner, option: 4 82 612 includes:

7. Liner for end chock

End chock brackets, option: 4 82 614 includes:

8. End chock bracket

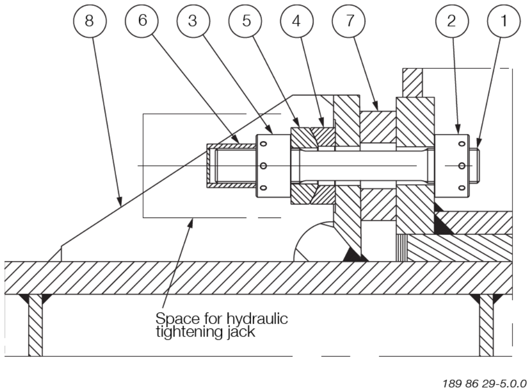


Fig. 5.12.02b: Profile of engine seating, end chocks, option: 4 82 610

Side chock brackets, option: 4 82 622 includes:

1. Side chock brackets

   Side chock liners, option: 4 82 620 includes:

2. Liner for side chock
3. Lock plate
4. Washer
5. Hexagon socket set screw

Detail D1

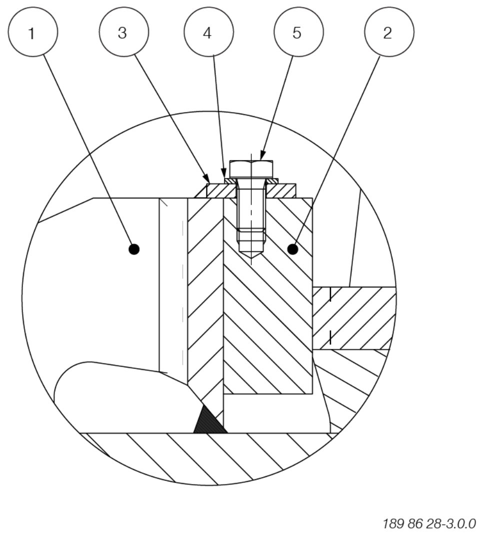


Fig. 5.12.02c: Profile of engine seating, side view, side chocks, option: 4 82 620

Engine top bracing

General

The so-called guide force moments are caused by the transverse reaction forces acting on the crossheads 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 countermeasures are not taken.

As a detailed calculation of this system is normally not available, MAN Energy Solutions 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 system is installed either as a mechanical top bracing (typically on smaller engine types) or a hydraulic top bracing (typically on larger engine types). Both systems are described below.

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

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 Energy Solutions instructions.

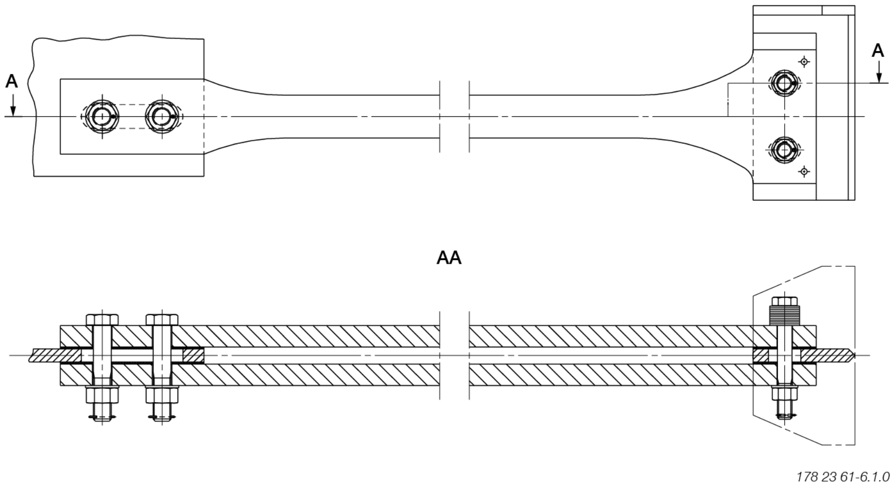


Fig. 5.13.01: Mechanical top bracing stiffener.
Option: 4 83 112

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.

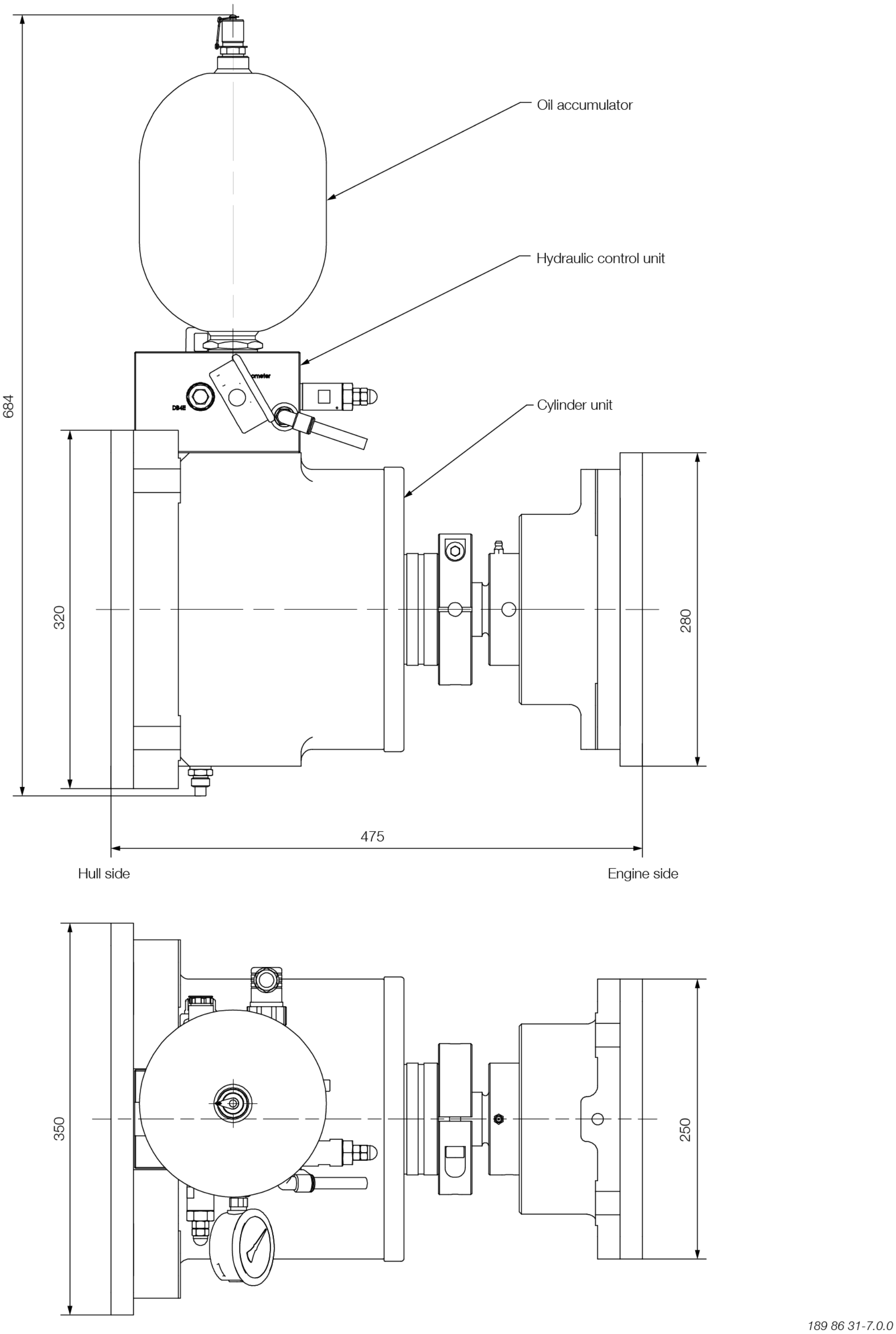


Fig. 5.13.02: Outline of a hydraulic top bracing unit. The unit is installed with the oil accumulator pointing either up or down. Option: 4 83 123

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Mechanical top bracing - T III

General

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Hydraulic top bracing arrangement - TIII

General

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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):

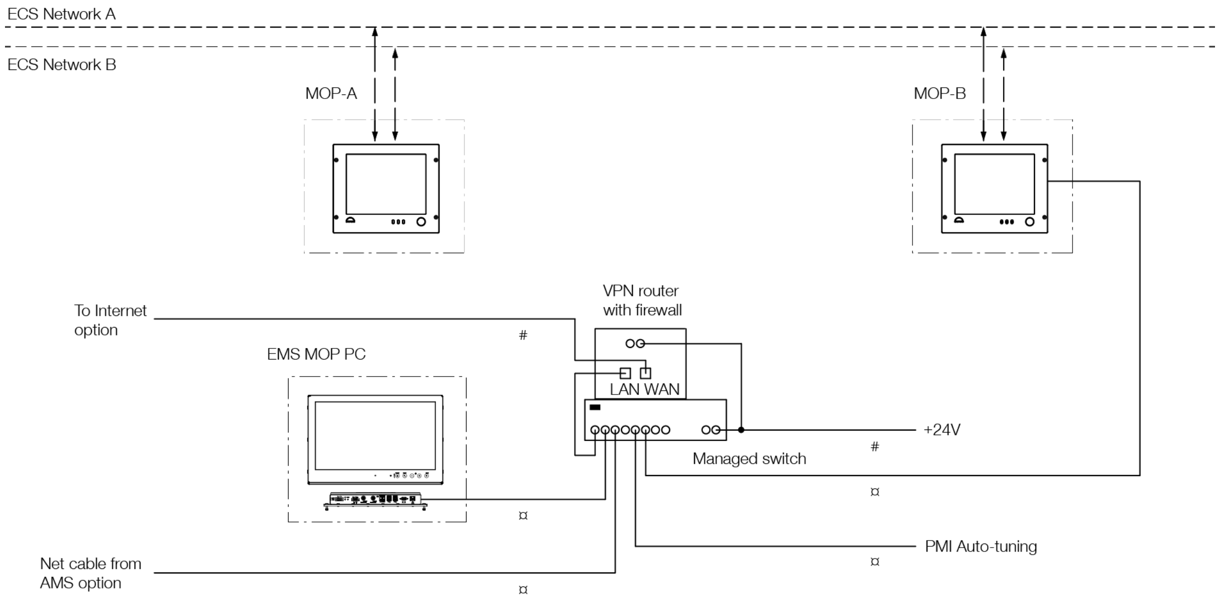
- 2 pcs EICU (Engine Interface Control Unit) (1 pcs only for ME-B engines)
- 1 pcs ECS MOP-A (Main Operating Panel) EC-MOP with touch display, 15”
- 1 pcs ECS MOP-B EC-MOP with touch display, 15”
- 1 pcs EMS MOP with system software Display, 24” marine monitor PC unit
- 1 pcs Managed switch and VPN router with firewall

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.

MOP-A and -B are redundant and are the operator’s interface to the ECS. Via both MOPs, the operator can control and view the status of the ECS. Via the EMS MOP PC, the operator can view the status and operating history of both the ECS and the engine, EMS is decribed in Section 18.01.

The PMI Auto-tuning application is run on the EMS MOP PC. PMI Auto-tuning is used to optimize the combustion process with minimal operator attendance and improve the efficiency of the engine. See Section 18.01.

CoCoS-EDS ME Basic is included as an application in the Engine Management Services as part of the standard software package installed on the EMS MOP PC. See Section 18.01.



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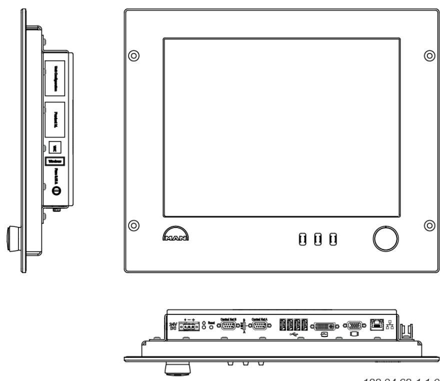
5.16 Components for engine control system

Abbreviations:		# Yard Supply
AMS:	Alarm Monitoring Systems	
EICU:	Engine Interface Control Unit	¤ Ethernet, 10 m patch cable supplied with switch. Type:
EMS:	Engine Management Services	RJ45, STP (Shielded Twisted Pair), CAT 5.
MOP:	Main Operating Panel	In case 10 m cable is not enough, this becomes Yard supply.

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

EC-MOP

- Integrated PC unit and touch display, 15”
- Direct dimming control (0-100%)
- USB connections at front
- IP20 resistant front
- Dual Arcnet



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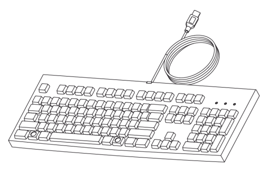
Pointing Device

- Keyboard model
  - UK version, 104 keys
  - USB connection
- Trackball mouse
  - USB connection



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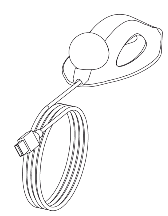

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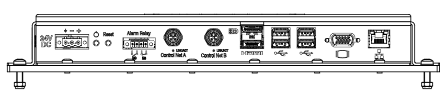
EMS MOP PC

- Standard industry PC with MS Windows
  operating system, UK version



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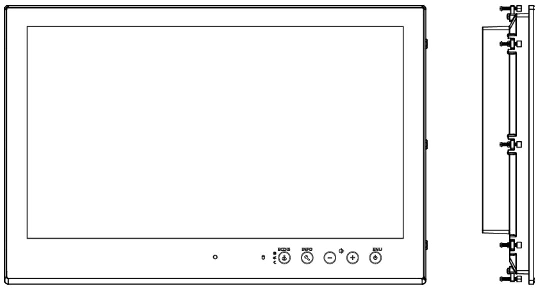
Marine Monitor for EMS MOP PC

- LCD (MVA) monitor 24”
    - Projected capacitive touch
    - Resolution 1,920x1,080, WSXGA+
    - Direct dimming control (0-100%)
    - IP54 resistant front
    - For mounting in panel



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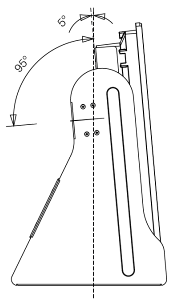
- Bracket for optional mounting on desktop, with hinges (5° tilt, adjustable 95°) or without hinges (10° tilt, not adjustable



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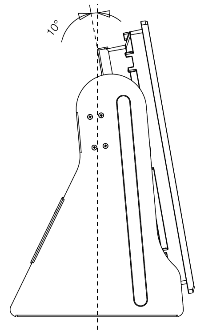

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Network Components

- Managed switch and VPN router with firewall

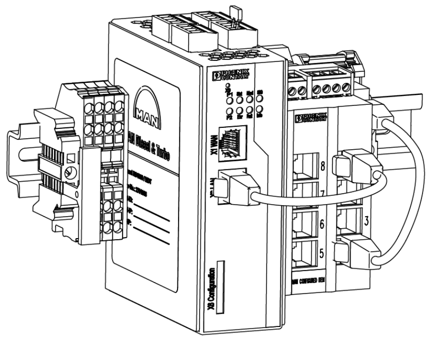

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Fig. 5.16.02 MOP PC equipment for the ME/ME-B Engine Control System

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5.16 Components for engine control system

EICU Cabinet

- Engine interface control cabinet for ME-ECS for installation in ECR (recommended) or ER

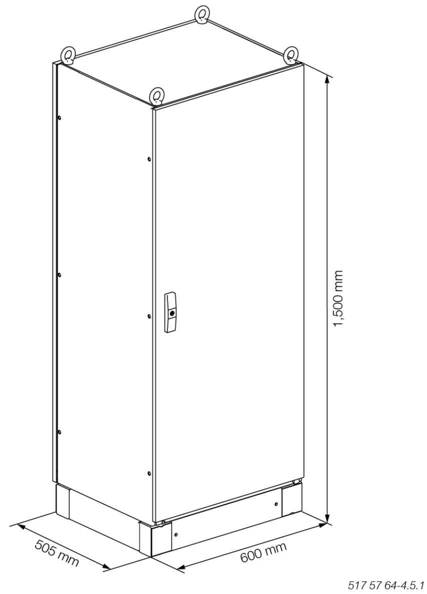
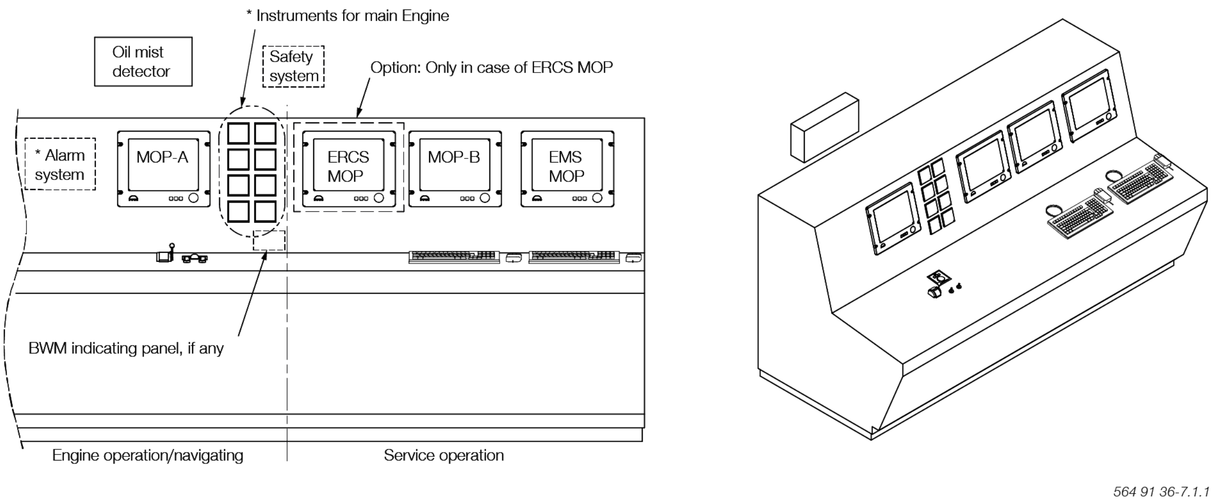


Fig. 5.16.03: The network printer and EICU cabinet unit for the ME Engine Control System

Engine Control Room Console

- Recommended outline of Engine Control Room console with ME equipment



* Yard supply

Oil mist detector equipment depending on supplier/maker
BWM: Bearing Wear Monitoring

Fig. 5.16.04: Example of Engine Control Room console

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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.

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 0.1µ Ohm x 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.

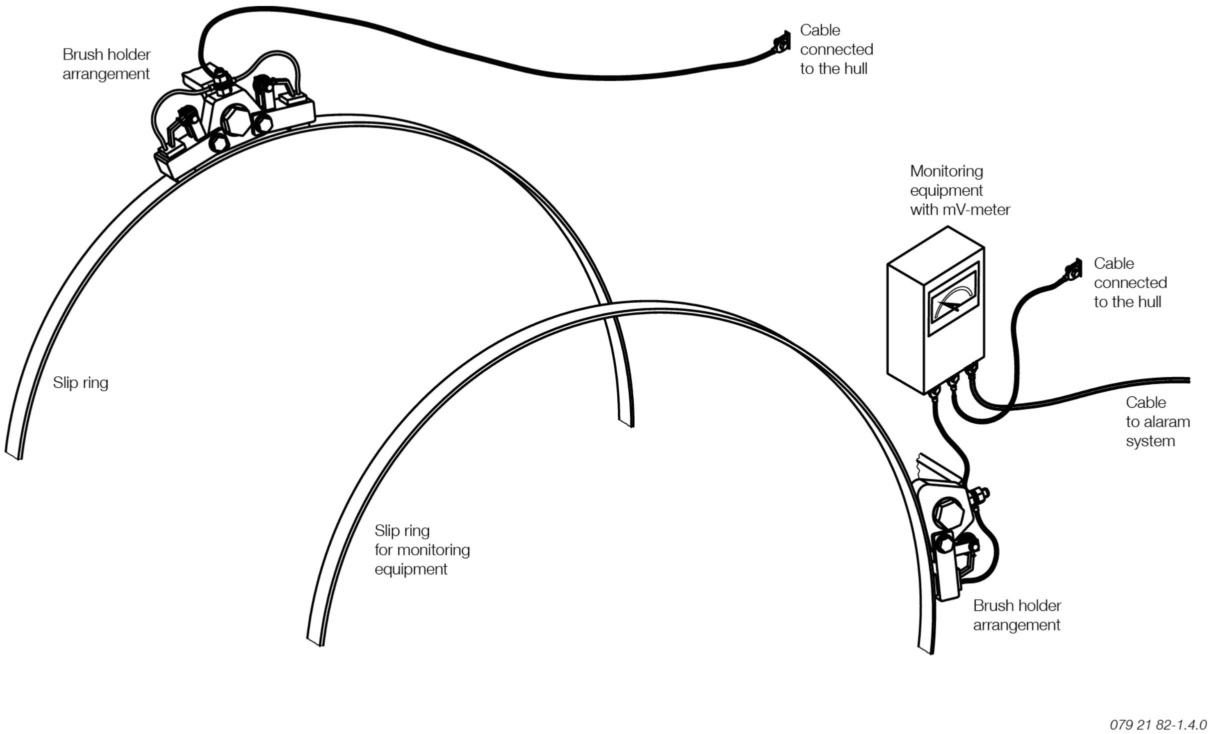

Fig. 5.17.01: Connection of cables for the shaftline earthing device

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

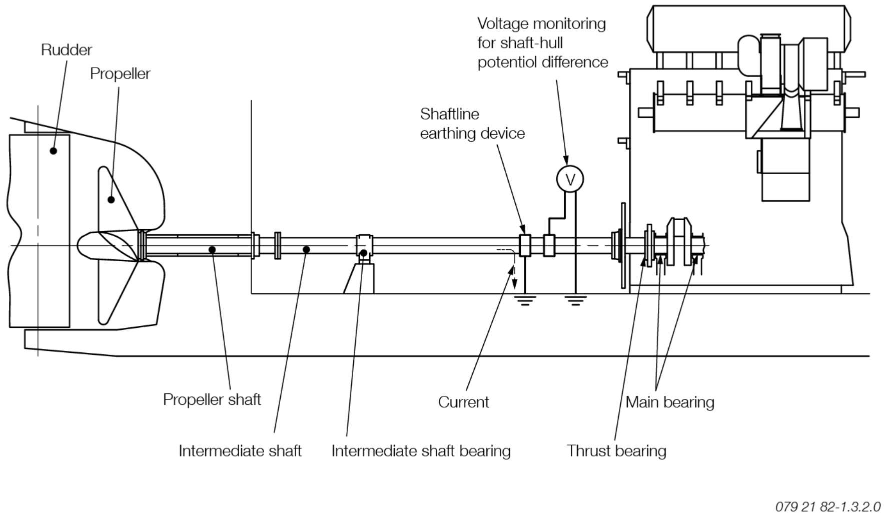

Fig. 5.17.02: Installation of shaftline earthing device in an engine plant without shaft-mounted generator

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

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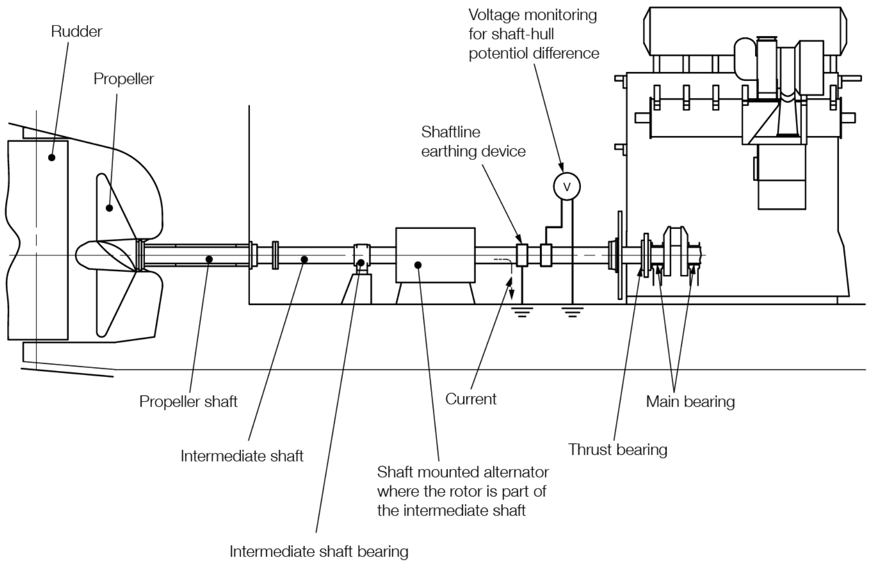

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Fig. 5.17.03: Installation of shaftline earthing device in an engine plant with shaft-mounted generator

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MAN Alpha CPP and Alphatronic propulsion control

General

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Calculation of List of Capacities

Updated engine and capacities data is available from the CEAS application at www.marine.man-es.com -->' Two-Stroke' --> 'CEAS engine calculations'.

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 a derated engine, i.e. with a specified MCR 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.

For a derated engine, calculations of:

- Derated capacities
- Available heat rate, for example for freshwater production
- Exhaust gas amounts and temperatures

can be made in the CEAS application available at the above link.

Nomenclature

In the following description and examples of the auxiliary machinery capacities in Section 6.02, the below nomenclatures are used:

Engine ratings	Point / Index	Power	Speed
Nominal maximum continuous rating (NMCR)	L_1	P_{L1}	n_{L1}
Specified maximum continuous rating (SMCR)	M	P_M	n_M
Normal continuous rating (NCR)	S	P_S	n_S

Table. 6.01.01: Nomenclature of basic engine ratings

Parameters	Cooler index	Flow index
M = Mass flow	air = scavenge air cooler	exh = exhaust gas

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

Engine Configurations Related to SFOC

The engine type is available in the following version with respect to the efficiency of the turbocharger(s) alone:

High efficiency turbocharger, the basic engine design (EoD: 4 59 104)

Conventional turbocharger, (option: 4 59 107) for both of which the lists of capacities Section 6.03 are calculated.

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List of capacities for 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 NMCR. 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 turbocharger running at NMCR for:

- Seawater cooling system,
  See diagram, Fig. 6.02.01 and nominal capacities in Fig. 6.03.01

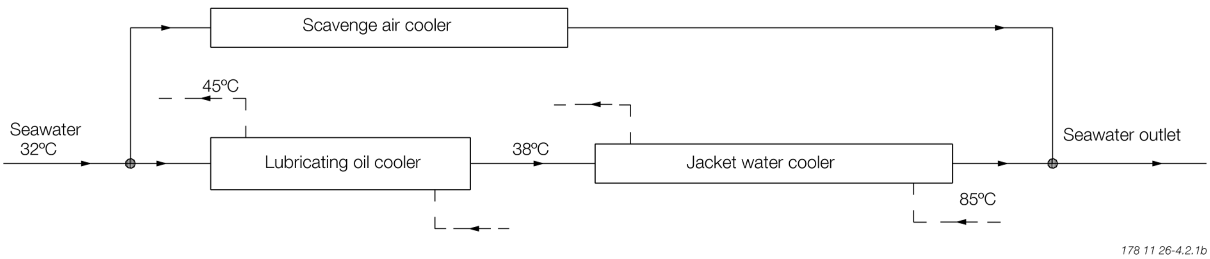

Fig. 6.02.01: Diagram for seawater cooling system

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

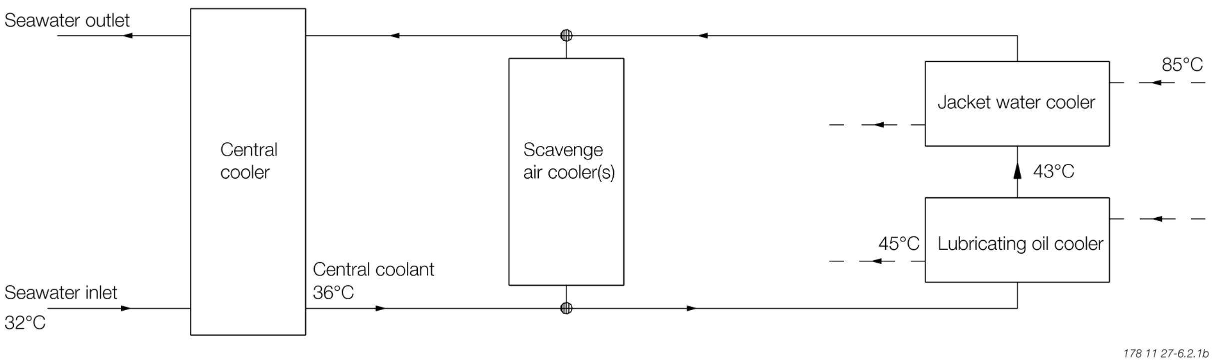

Fig. 6.02.02: Diagram for central cooling water system

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

Heat Radiation

The radiation and convection heat losses to the engine room is around 1% of the engine power at NMCR.

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 ‘Counter flanges’; both can be found in Chapter 5.

The diagrams use the ‘Basic symbols for piping’, the symbols for instrumentation are according to ‘ISO 1219-1’ / ‘ISO 1219-2’ and the instrumentation list both found in Appendix A.

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List of capacities

Download an engine report with capacities for pumps, coolers, auxiliary system, etc., for your specific engine type by using our online engine calculation tool CEAS at:

https://www.man-es.com/marine/products/planning-tools-and-downloads/ceas-engine-calculations

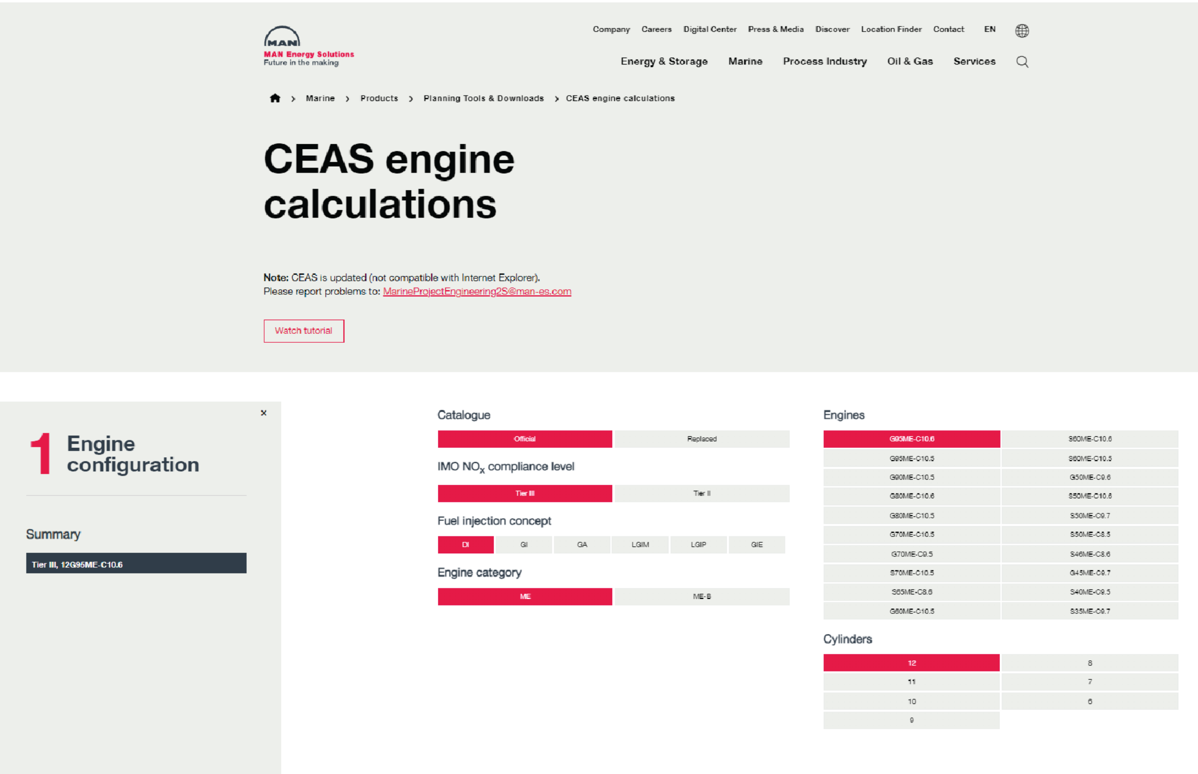

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Fig. 6.03.01: The browser-based CEAS calculation system

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Auxiliary machinery capacities derated engines

Further to the auxiliary machinery capacities for a nominally rated engine shown in Section 6.03, the dimensioning of heat exchangers (coolers) and pumps for derated engines as well as calculating the:

- List of capacities for derated engine
- Available heat to be removed, for example for freshwater production
- Exhaust gas amounts and temperatures

can be made in the CEAS application descibed in Section 20.02.

The CEAS application is available at www.marine.man-es.com --> 'Two stroke' --> 'CEAS Engine Calculations' .

Pump Pressure and Temperatures

The pump heads stated in the table below are for guidance only and depend on the actual pressure drop across coolers, filters, etc. in the systems.

	Pump head, bar	Max. working temp. °C
Fuel oil supply pump	4	100
Fuel oil circulating pump	6	150
Lubricating oil pump	4.8	70
Seawater pump, for seawater cooling system	2.5	50
Seawater pump, for central cooling water system	2.0	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

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Centrifugal Pump Selection

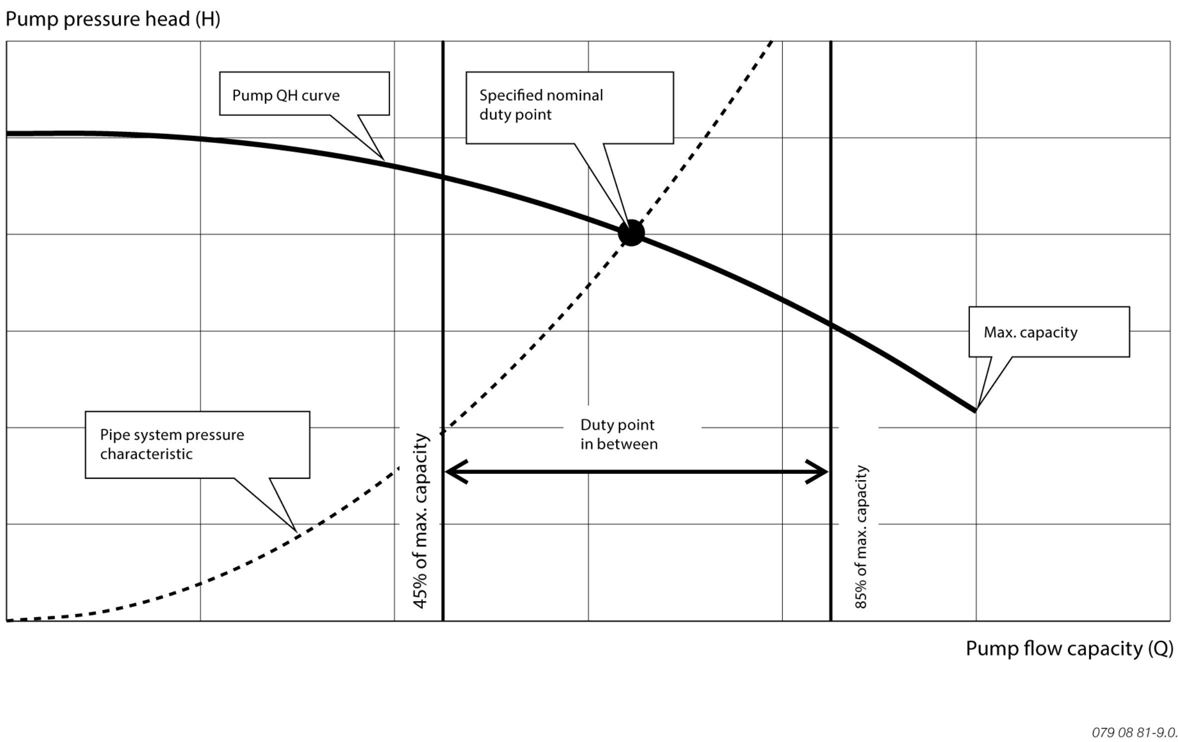

Fig. 6.04.01: Location of the specified nominal duty point (SNDP) on the pump QH curve

When selecting a centrifugal pump, it is recommended to carefully evaluate the pump QH (capacity/ head) curve in order for the pump to work properly both in normal operation and under changed conditions. But also for ensuring that the maximum pipe design pressure is not exceeded.

The following has to be evaluated:

- Location of the specified nominal duty point (SNDP) on the pump QH curve
- Pump QH curve slope
- Maximum available delivery pressure from the pump.

Location of the Duty Point on the Pump QH

Particularly important is the location of the specified nominal duty point (SNDP) on the pump QH curve: the SNDP is equal to the intersection of the pump QH curve and the pipe system pressure characteristic, which is defined at the design stage.

The SNDP must be located in the range of 45 to 85% of the pump's maximum capacity, see Fig. 6.04.01.

Thus, the pump will be able to operate with slightly lower or higher pipe system pressure characteristic than specified at the design stage, without the risk of cavitation or too big variations in flow.

Pump QH Curve Slope

At the location of the SNDP, the pump capacity should not decrease by more than 10% when the pressure is increased by 5%, see Fig. 6.04.02.

This way, the flow stays acceptable even if the pipe system pressure is higher than expected and the flow does not change too much, for example when a thermostatic valve changes position.

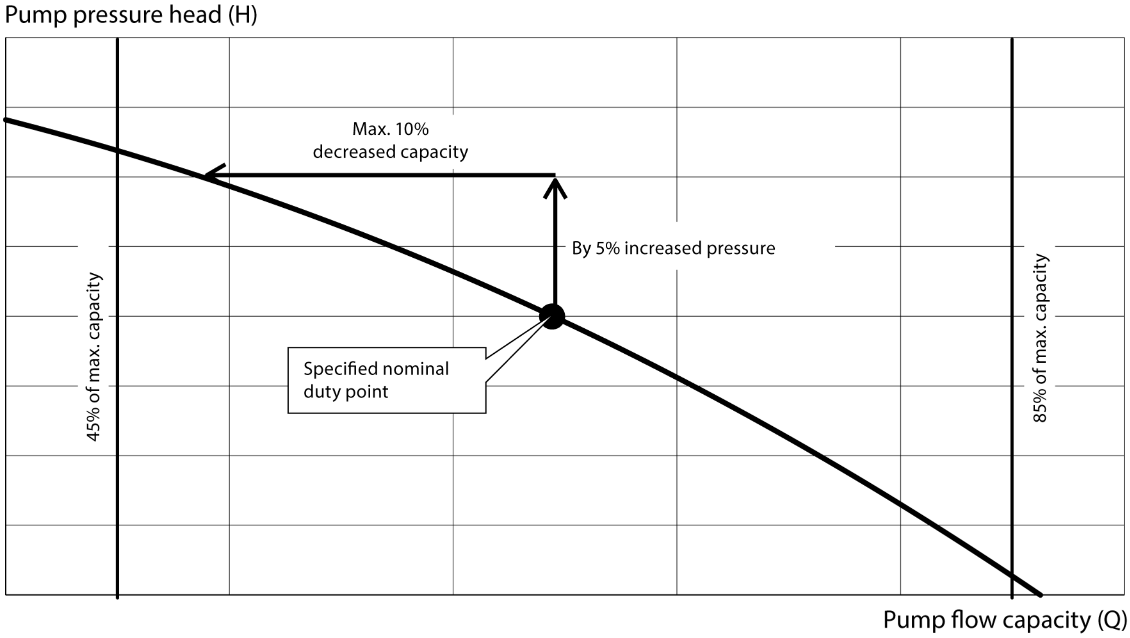

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Fig. 6.04.02: Pump QH curve slope

Maximum Available Pump Delivery Pressure

It is important to evaluate, if the maximum available delivery pressure from the pump contributes to exceeding the maximum allowable design pressure in the pipe system.

The maximum available delivery pressure from the pump will occur e.g. when a valve in the system is closed, see Fig. 6.04.03.

The maximum allowable pipe system design pressure must be known in order to make the pressure rate sizing for equipment and other pipe components correctly.

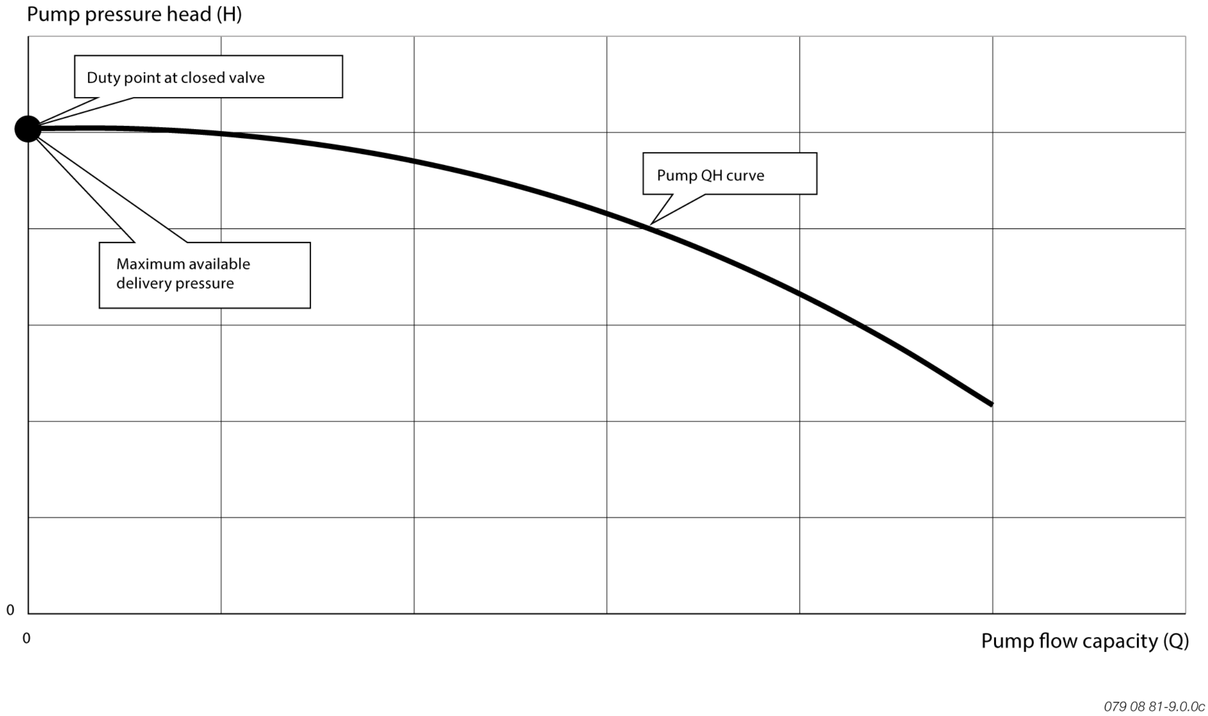

Fig. 6.04.03: Maximum available pump delivery pressure

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Second fuel system – ME-LGIM engine

The MAN B&W ME-LGIM engine is the methanol-burning version of our dual-fuel solution for liquid injection of fuels, the ME-LGIM engine. The ME-LGIM engine is a dual-fuel engine combusting methanol ignited by pilot oil in dual-fuel mode, and fuel oil when running in fuel oil mode.

In this section, the term second fuel denotes methanol. Methanol is a fuel which can be stored in the liquid phase at atmospheric pressure.

Methanol from a low-flashpoint fuel supply system (LFSS) passes through the main supply pipe and a fuel valve train (FVT) before it is distributed to each cylinder.

The components and piping on the engine for methanol operation are described in the present Section 7.00, the conventional fuel oil system and pilot oil system are described in 7.01–7.05, the LFSS and auxiliary systems for dual-fuel operation in Sections 7.07 and 7.09.

ME-LGIM specific engine parts

Fig. 7.00.01. shows LGIM components and piping on the engine for injection of methanol.

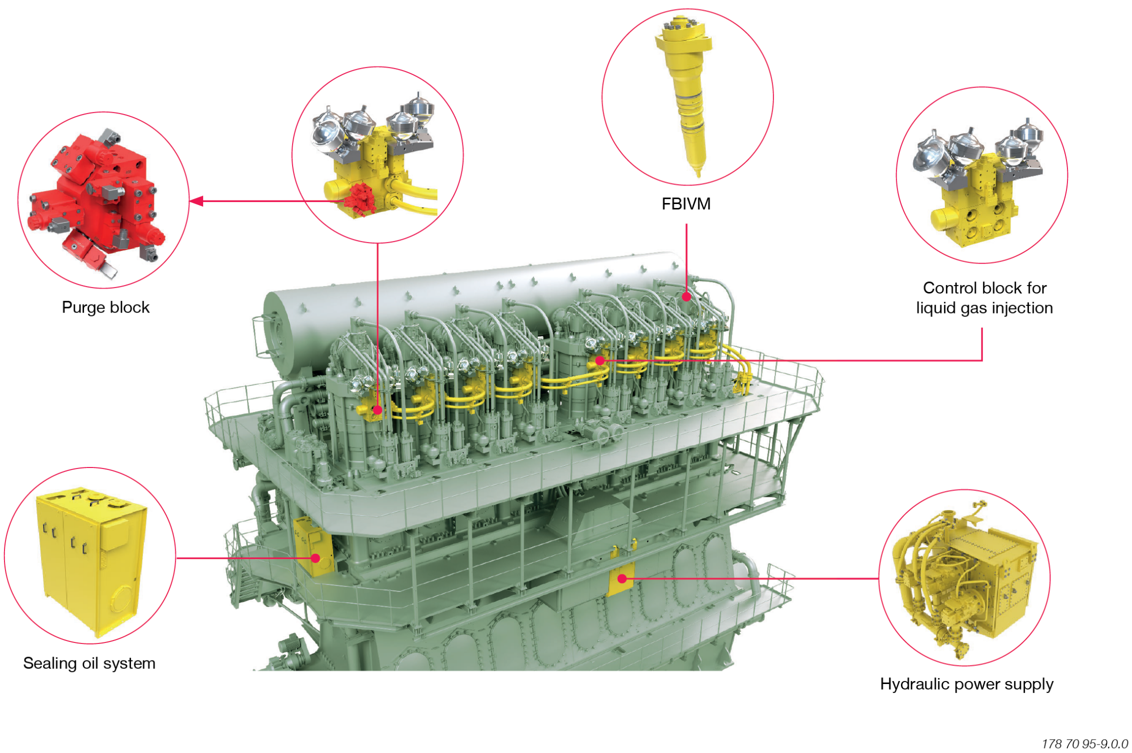

Fig. 7.00.01 Second fuel components and piping on the engine

The injection system consists of a control block for liquid gas injection (LGI), fuel booster injection valves for methanol (FBIVM), and a hydraulic power supply unit (HPS). Methanol is supplied to each FBIVM via drillings in the cylinder cover.

A sealing oil system delivers sealing oil to the FBIVM to keep control oil and methanol separate.

Apart from these systems on the engine, the engine auxiliaries comprise:

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- Fully automated methanol supply system with an embedded purge system
- The ME-LGI engine control system (ME-ECS)
- Conventional fuel oil system for pilot oil injection
- Drain and purge return system for quick and reliable removal of methanol from the engine
- Leakage detection and ventilation system, which ventilates the outer pipe in the double-wall piping completely, and incorporates leakage detection
- Inert gas system for purging of the second fuel supply system and the second fuel system on the engine, two separate inert gas supply systems (A and B supply)
- FVT provides a block-and-bleed function between the fuel system and the engine
- Heat traced and insulated second fuel supply pipes
- Service tank with two compartments: fuel compartment and return compartment
- Sealing ventilation
- Flow switches detect dry-air supply and monitor air flow.

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Second fuel piping on the engine

Fig. 7.00.02 shows the double-walled supply and return piping distributing methanol to a cylinder.

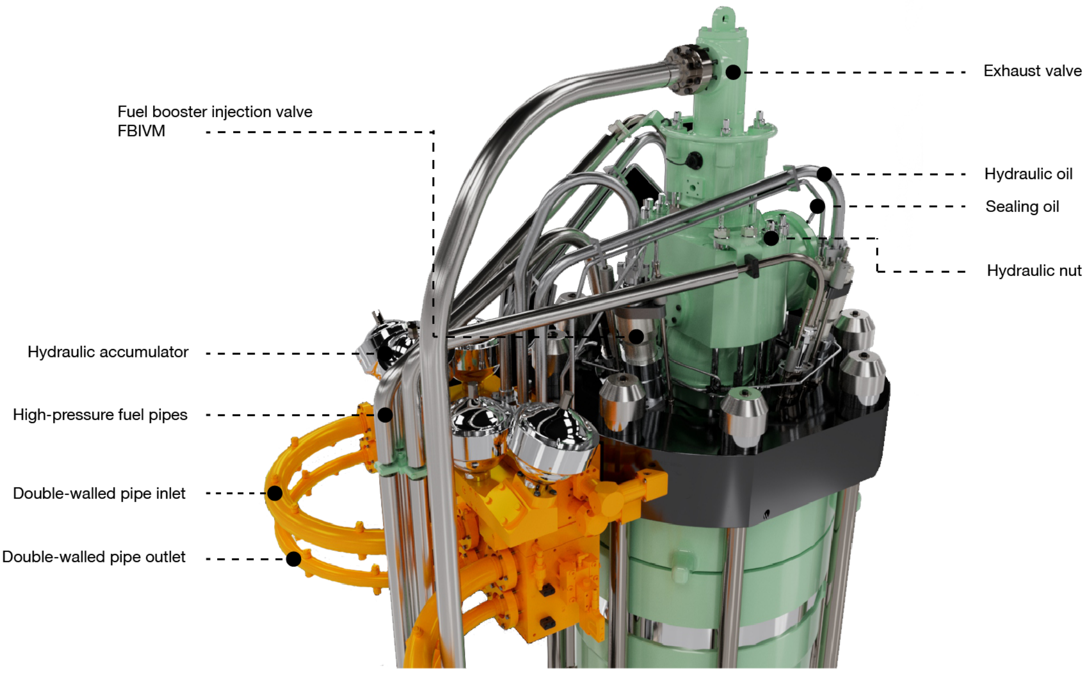

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Fig. 7.00.02: Second fuel components and piping on the cylinder

This double-walled design is similar to the concept used for other dual-fuel engines of MAN Energy Solutions. The double-walled design ensures that methanol is contained within a ventilated pipe if leakages occur. The ventilated pipe has methanol detectors and vents any leakage to outside the engine room. When switching from dual-fuel mode to fuel-oil-only mode, the methanol piping will be purged with an inert gas purging system (nitrogen). At the lowest point on the return piping, a level switch is placed to detect any leftover methanol. Purging will ensure that any leftover methanol is returned to the service tank.

Second fuel injection system

The LGI control block in Fig. 7.00.01 includes high-speed analogue control valves for regulation of the methanol injection pattern and timing by regulating the high-pressure hydraulic oil supply to the FBIVMs. The engine control system (ECS) control the valves.

The design features hydraulic oil accumulators to ensure an adequate hydraulic oil supply during injection.

The amount of liquefied fuel in dual-fuel mode depends on the amount available from the LFSS. The ECS receives information about the available amount from the LFSS and calculates the needed amount of pilot oil.

When operating the engine in dual-fuel mode, a small amount of pilot oil (fuel oil) is always necessary to ensure combustion.

The ECS controls the operation in three different modes:

- Dual-fuel mode with a minimum pilot oil amount
- Specified dual-fuel operation (SDF) with injection of a fixed second fuel amount
- Fuel-oil-only mode.

Pilot oil injection amount versus engine load

Dual-fuel operation is possible down to 10% load.

The minimum pilot oil amount in dual-fuel mode is 5% at MCR (in L1), see Fig. 7.00.03.

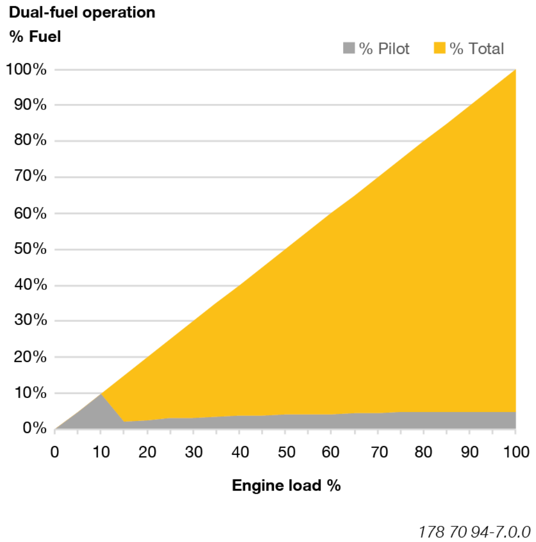

Fig. 7.00.03: Fuel index in dual-fuel operation

If the engine is derated, the pilot oil amount is relatively higher as calculated in CEAS.

CEAS can be found here:

https://www.man-es.com/marine/products/planning-tools-and-downloads/ceas-engine-calculations

Condition of the second fuel supply to the engine

The following data is based on methanol.

Pressure

Operating pressure	12.5–13.5 bar
Supply pressure	13 bar
Safety relief valve	16 bar
Pulsation limit	±0.5 bar

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Flow

The maximum flow requirement is specified at 100% SMCR, 13 bar, with reference to a lower calorific value (LCV) of 19,900 kJ/kg.

Maximum / minimum requirements	See CEAS
Minimum flow requirement in standby	0 kg/h

The maximum flow requirement must also be achievable close to the overhaul interval of the second fuel supply system.

If there is a specific LCV requirement, inform MAN Energy Solutions. Under certain circumstances, modification of the FBIVM may be required to accommodate a special LCV.

Temperature

Temperature inlet to engine	25–50°C
Alarm, low / high	25°C / 55°C
Stop, low / high	15°C / 60°C

The temperature specification takes the following into account:

- Reduce condensation on the outer wall of the inner pipe for a double-walled piping
- Performance of the engine is not adversely affected
- Reduces thermal loads on the second fuel piping itself
- Obtain a uniform second fuel density.

The second fuel temperature during blow-off will still be within the temperature limits of the materials selected for piping and components.

Methanol is a clear, colourless, water soluble liquid which boils at 65°C. It is also known as methyl alcohol and is often abbreviated MeOH. It is most commonly produced from natural gas but can also be made from bio-feedstocks and gasification of coal.

Methanol is flammable and can form explosive mixtures with air. It burns with a non-luminous flame and is highly toxic.

The values in the guiding fuel specification for ME-LGIM engines, Table 7.00.01, refer to the fuel as delivered to the ship.

Designation	Unit	Limit	Value	Test method reference* (latest edition to be applied)
Lower calorific value (LCV)	MJ/kg	Min.	19	
Methanol (CH_3OH)	% w/w	Min.	95	IMPCA 001-14
Ethanol (C_2H_5OH)	% w/w	Max.	5	IMPCA 001-14
Water (H_2O)	% w/w	Max.	5*	ASTM E1064-12**
Acetone (CH_3COCH_3)	mg/kg	Max.	30	IMPCA 001-14
Chloride as Cl^-	mg/kg	Max.	0.5	IMPCA 002-98

Designation	Unit	Limit	Value	Test method reference* (latest edition to be applied)
Acidity as acetic acid	mg NaOH/ kg sample	Max.	30	ASTM D1613-17
Sulphur (S)	mg/kg	Max.	0.5	ASTM D3961-98 or ASTM D5453-12
Appearance	N/A	Clear, uncoloured and free of suspended solids		IMPCA 003-98
Sampling	N/A	IMPCA Methanol Sampling Methods, International Methanol Producers & Consumers Association (IMPCA), www.impca.eu, October 2014.		
*A water content up to 30% w/w can be accepted under certain conditions and after agreement with MAN Energy Solutions. Note that this will affect the possibility to reach max. power. **ASTM E1064 is valid up to max 2% water.				

Table 7.00.01: Guiding methanol fuel specification for MAN B&W ME-LGIM engines. Values refer to methanol as delivered to the ship.

Bunkering

Liquid or solid contaminants such as metal shavings, welding debris, insulation (perlite), sand, wood, cloth, and oil must be removed from the methanol.

It is considered as good engineering and operating practice to have methanol cargo strainers in loading and discharge lines to minimise particulate contamination of methanol and subsequent tanks and equipment. It is recommended that the filter is inspected after bunkering to establish the contamination degree.

It is important to note that the quality and impurity degree can vary among the suppliers due to production and handling differences and the type of bunkering/transfer process (for example: terminal tank to vessel, truck to vessel).

Sealing oil system

The sealing oil system is a pressurised hydraulic oil system with a constant differential pressure kept at a higher level than the second fuel pressure, which prevents second fuel from entering the hydraulic oil system.

The sealing oil is applied to the FBIVMs and the window/shutdown valve in the space between second fuel on one side and hydraulic oil on the other side.

The sealing oil pump unit consists of a pump and a safety block with an accumulator. The sealing oil pump unit is connected to one of the second fuel adapter blocks with double-walled pipes. Interconnecting pipes between the second fuel adapter blocks distribute sealing oil to all units.

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The sealing oil system uses the low-pressure supply oil from the hydraulic system. The oil is pressurised to an operating pressure which is 20–25 bar higher than the second fuel pressure for methanol to prevent that the hydraulic oil is polluted with second fuel.

The sealing oil system also has a ventilated pipe similar to the second fuel system but only a single-walled pipe which terminates outside the engine room. This is done as a precaution to remove any methanol vapours from the sealing oil tank.

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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.

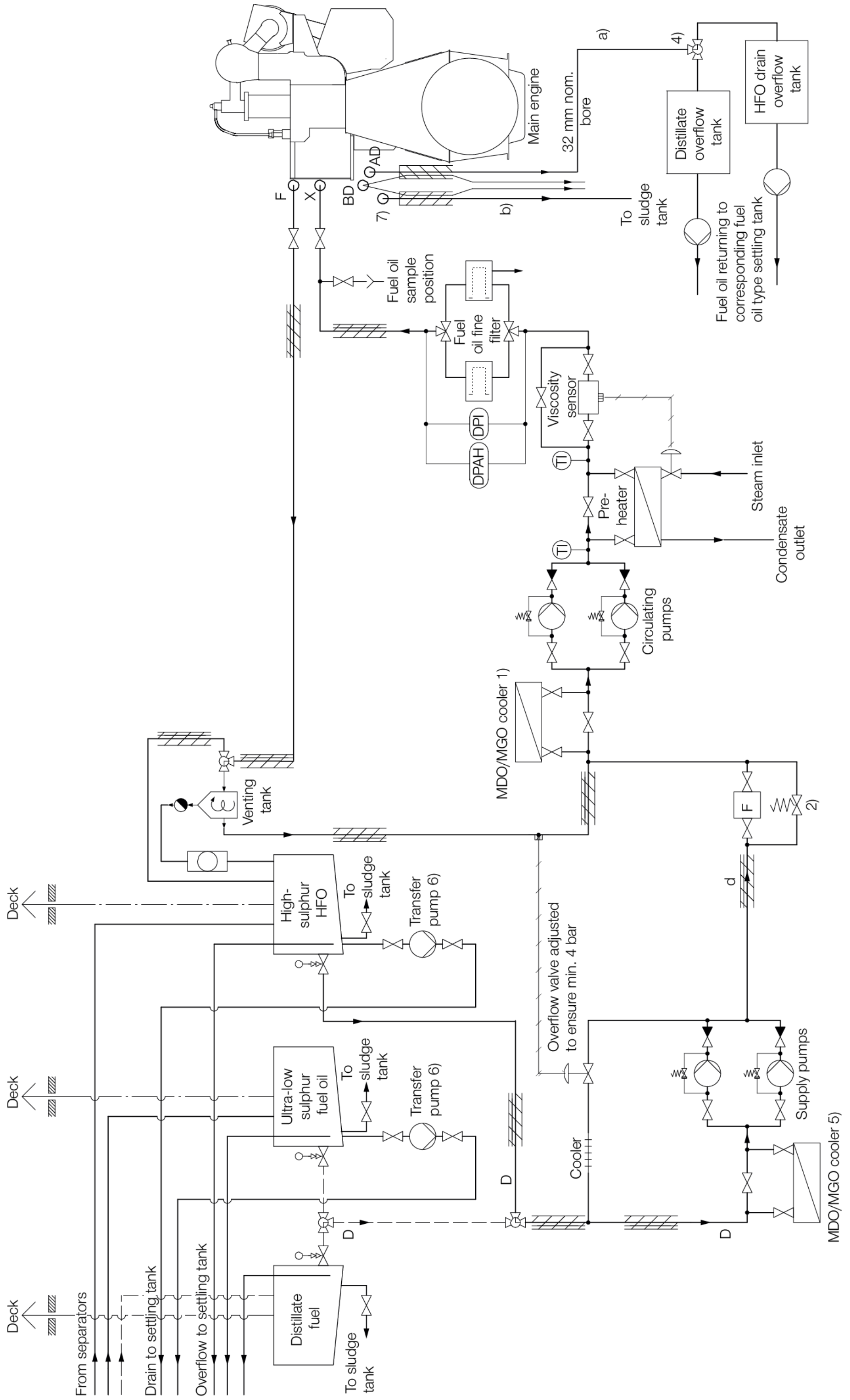
In special circumstances a change-over to diesel oil may become necessary – and this can be performed at any time, even when the engine is not running. Such a change-over may become necessary if, for instance, the vessel is expected to be inactive for a prolonged period with cold engine e.g. due to:

- 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



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7.01 Fuel oil system

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1) MDO/MGO Cooler

For low-viscosity distillate fuels like marine gas oil (MGO), it is necessary to have a cooler to ensure that the viscosity at engine inlet is above 2 cSt.

Location of cooler: As shown or, alternatively, anywhere before inlet to engine.

2) Fuel oil flowmeter (Optional)

Flow rate: See ‘List of Capacities’ (same as fuel supply pump).
Type: In case a damaged flow meter can block the fuel supply, a safety bypass valve is to be placed across the flowmeter.

3) 0.23 litre/kWh in relation to certified Flow Rate (CFR); the engine SMCR can be used to determine the capacity. The separators should be capable of removing cat fines (Al+Si) from 80 ppm to a maximum level of 15 ppm Al+Si but preferably lower.
Inlet temperature: Min. 98°C.

4) Valve in engine drain pipe

Valve in engine drain pipe is not acceptable. If the drain is blocked, the pressure booster top cover seal will be damaged.
In case a valve between the engine connection AD and the drain tank is required, the valve should be locked in open position and marked with a text, indicating that the valve must only be closed in case of no fuel oil pressure to the engine. In case of non-return valve, the opening pressure for the valve has to be below 0.2 bar.

5) MDO/MGO Cooler (Optional)

For protection of supply pumps against too warm oil and thus too low viscosity.

6) Transfer pump (Optional)

The transfer pump has to be able to return part of the content of the service tank to the settling tank to minimize the risk of supplying fuel to the engine with a high content of settled particles, e.g. cat fines, if the service tank has not been used for a while.

7) Name of flange connection

AF for engines with a bore of 60 cm and above
AE for engines with a bore of 50 cm and below

a) Tracing, fuel oil lines: By jacket cooling water

b) Tracing, drain lines: By jacket cooling water
– only for engines with bore of 60 cm and above

*) Optional installation

The letters refer to the list of ‘Counterflanges’

Heavy Fuel Oil
Distillate fuel or ultra-low sulphur fue oil
Heated pipe with insulation

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Fig. 7.01.01: Fuel oil system

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Heavy Fuel Oil Tank

This type of tank should be used for any residual fuel usage. (It can also be used for distillate fuel). The tank must be designed as high as possible and equipped with a sloping bottom in order to collect the solid particles settling from the fuel oil.

The tank outlet to the supply pumps must be placed above the slope to prevent solid particles to be drawn into the heavy fuel oil supply pumps. An overflow pipe must be installed inside the tank below the pump outlet pipe to ensure that only ‘contaminated’ fuel is pumped back to settling tank.

A possibility of returning the day tank content to the settling tank must be installed for cases where the day tank content have not been used for some time.

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 drain amount in litres per cylinder per hour is approximately as listed in Table 7.01.02.

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.

Engine bore, ME/ME-C, ME-B (incl. -GI & -LGI versions)	Flow rate, litres/cyl./hr.
98	On request
95, 90	1.7
80	2.1
70, 65	1.5
60	1.2

Table 7.01.02: Drain amount from fuel oil pump umbrella seal, figures for guidance

Leakage Oil Amount Dependencies

Due to tolerances in the fuel pumps, the table figures may vary and are therefore for guidance only. In fact, the leakage amount relates to the clearance between plunger and barrel in the third power. Thus, within the drawing tolerances alone, the table figures can vary quite a lot.

The engine load, however, has little influence on the drain amount because the leakage does not originate from the high-pressure side of the fuel pump. For the same reason, the varying leakage amount does not influence the injection itself.

The figures in Table 7.01.02 are based on fuel oil with 12 cSt viscosity. In case of distillate fuel oil, the figures can be up to 6 times higher due to the lower viscosity.

Fuel Oil Drains in Service and for Overhaul

The main purpose of the drain 'AD' is to collect fuel oil from the fuel pumps.

The drain oil is led to an overflow tank and can be pumped to the heavy fuel oil (HFO) tank or to the settling tank. In case of ultra low sulphur (ULSFO) or distillate fuel oil, the piping should allow the fuel oil to be pumped to the ultra low sulphur or distillate fuel oil tank.

As a safety measure for the crew during maintenance, an overhaul drain from the umbrella leads clean fuel oil from the umbrella directly to drain 'AF' and further to the sludge tank. Also washing water from the cylinder cover and the baseplate is led to drain 'AF'.

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', 'AF' and the drain for overhaul 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 (ME only), as well as turbocharger cleaning water etc. 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 oil system' as flange 'BD'. (Flange BD and the tracing line are not applicable on MC/ MC-C engines type 42 and smaller).

Fuel Oil Flow Velocity and Viscosity

For external pipe connections, we prescribe the following maximum flow velcities:

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.

Cat Fines

Cat fines is a by-product from the catalytic cracking used in fuel distillation. Cat fines is an extremely hard material, very abrasive and damaging to the engine and fuel equipment. It is recommended always to purchase fuel with as low cat fines content as possible.

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Cat fines can to some extent be removed from the fuel by means of a good and flexible tank design and by having optimum conditions for the separator in terms of flow and high temperature.

Further information about fuel oil specifications and other fuel considerations is available in our publications:

Guidelines for Fuels and Lubes Purchasing

0.50% S fuel operation - 2020

The publications are available at www.man-es.com → ‘Marine’ → ‘Products’ → ‘Planning Tools and Downloads’ → ’Technical Papers’.

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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 8217:2012, 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.

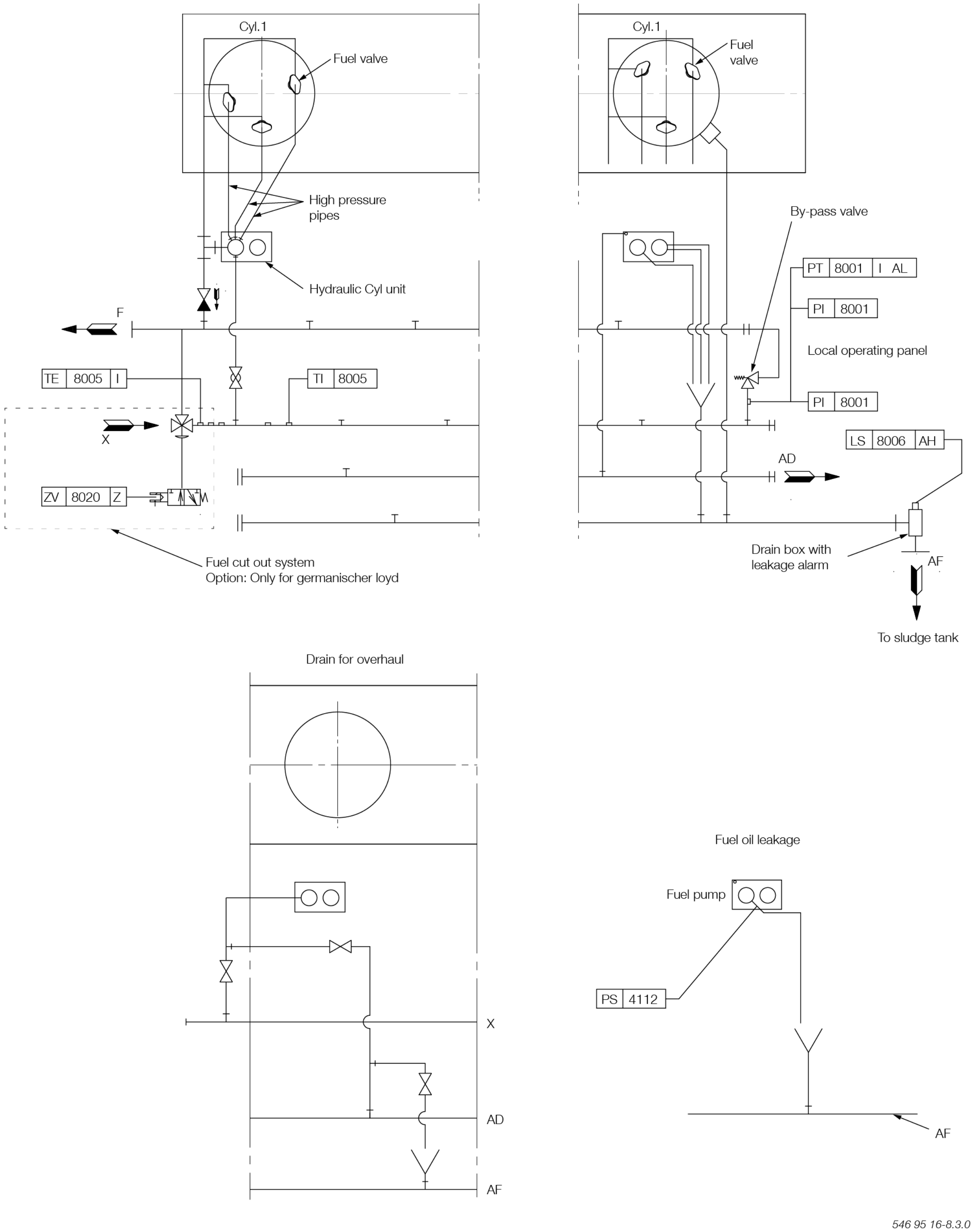
The data refers to the fuel as supplied i.e. before any on-board cleaning.

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)	≤ 20
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	≤ 450
Aluminum + Silicon	mg/kg	≤ 60
Equal to ISO 8217:2010 - 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 nos. refer to 'Guidance values automation'

Fig. 7.03.01: Fuel oil and drain pipes

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Insulation and heat tracing of fuel oil piping

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/m^3 and covered with glass cloth of minimum 400 g/m^2.

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/m^3 covered with glass cloth of minimum 400 g/m^2.

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/m^2, containing mineral wool stuffed to minimum 150 kg/m^3.

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.

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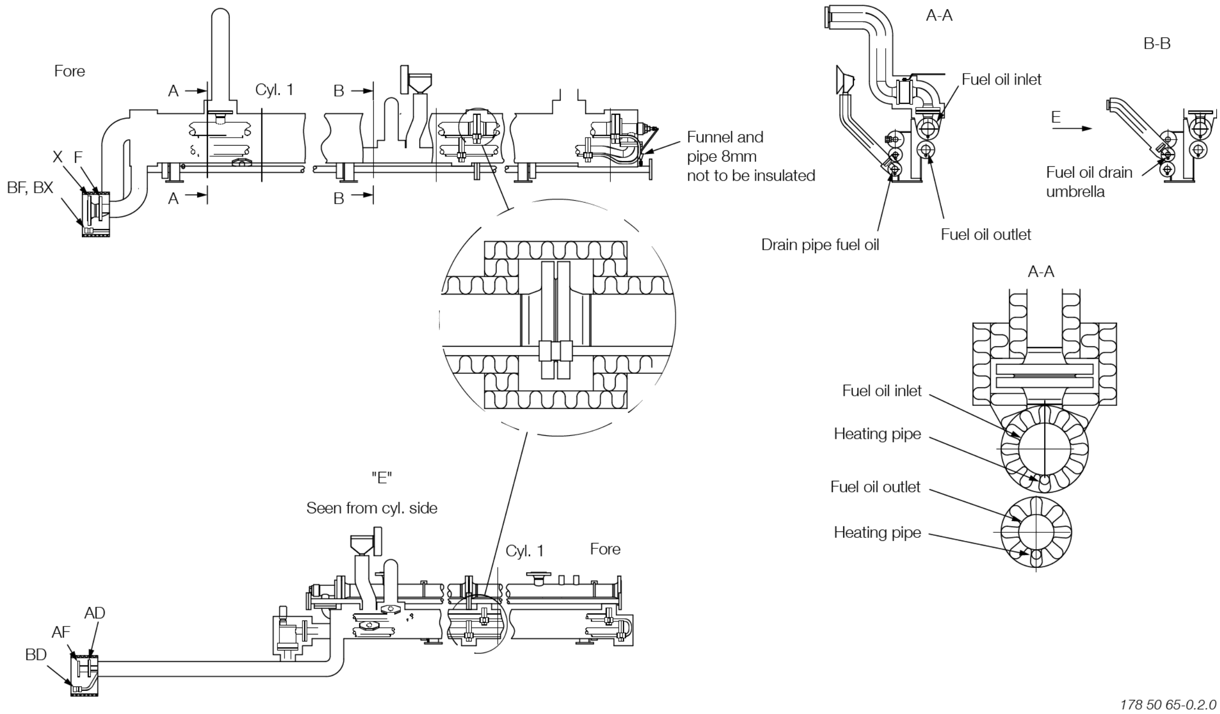

Fig. 7.04.01: Details of fuel oil pipes insulation, option: 4 35 121. Example from 98-50 MC engine

Heat Loss in Piping

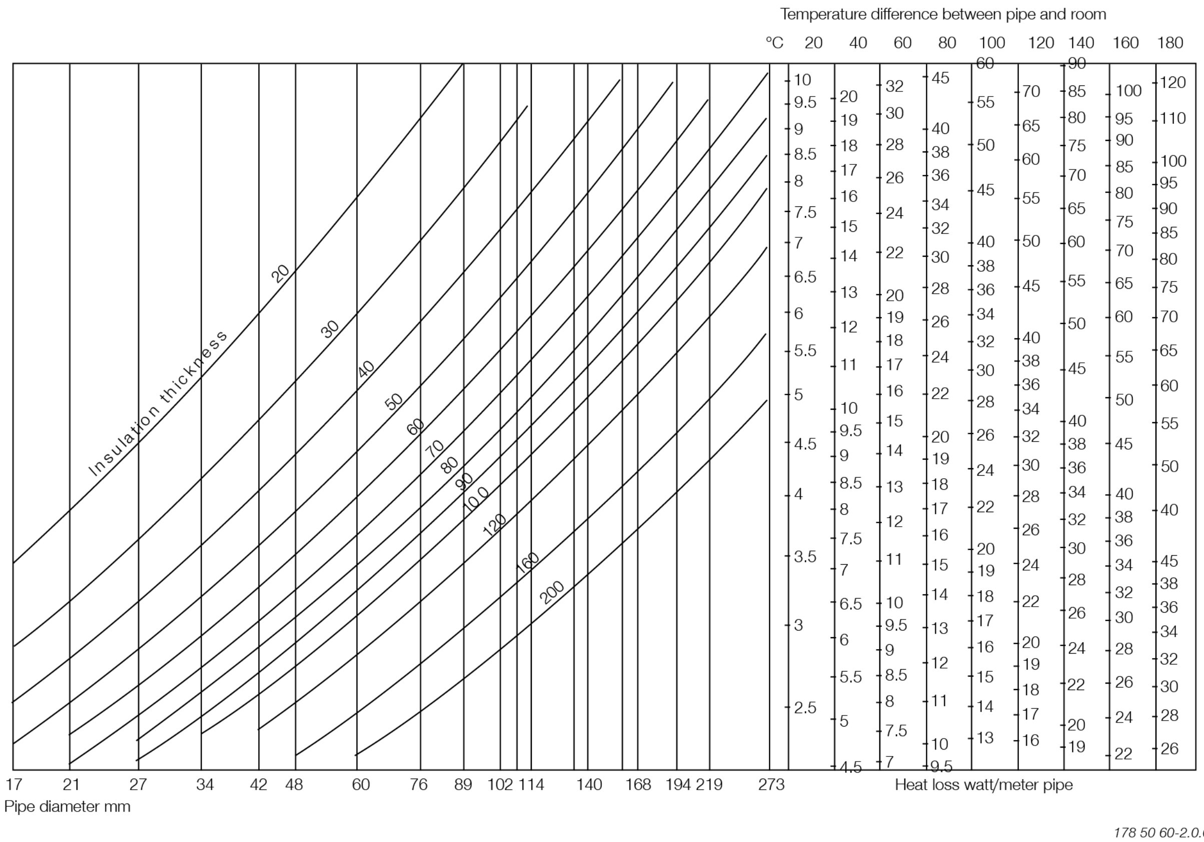

Fig. 7.04.02: Heat loss/Pipe cover

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.

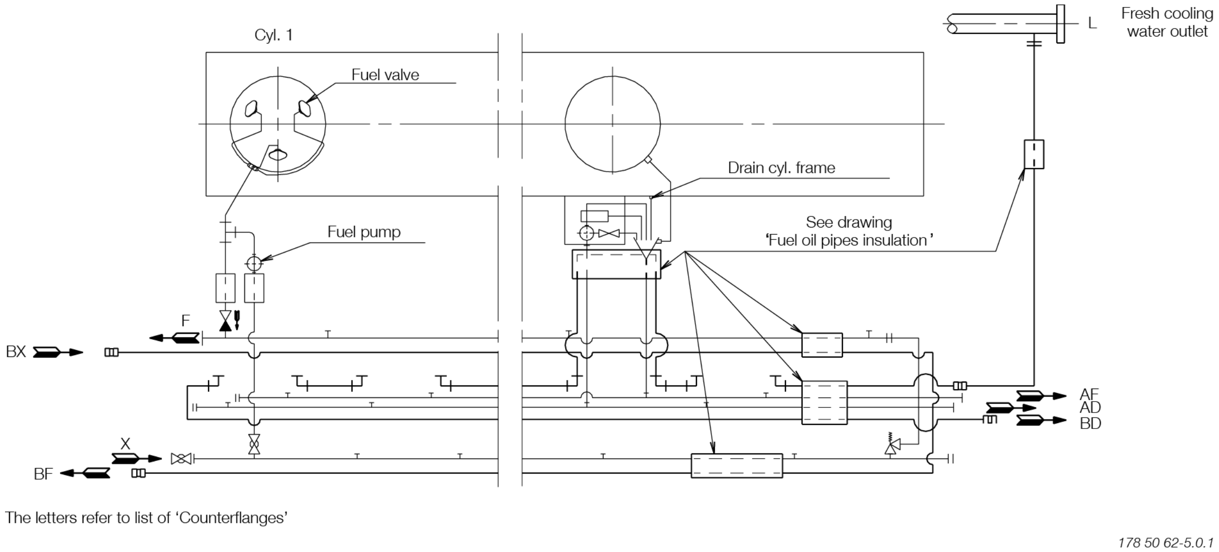

Fig. 7.04.03: Fuel oil pipe heat tracing

Fuel Oil and Lubricating Oil Pipe Spray Shields

To fulfill IMO regulations, fuel and oil pipe assemblies are to be secured by spray shields.

The shields can be made either by a metal flange cover according to IMO MSC/Circ.647 or antisplashing tape wrapped according to makers instruction for Class approval, see examples shown in Fig. 7.04.04a and b.

To ensure tightness, the spray shields are to be applied after pressure test of the pipe system.

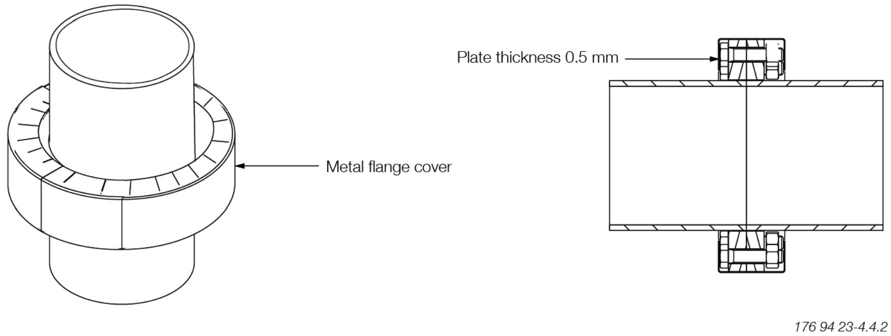

Fig. 7.04.04a: Metal flange cover and clamping band

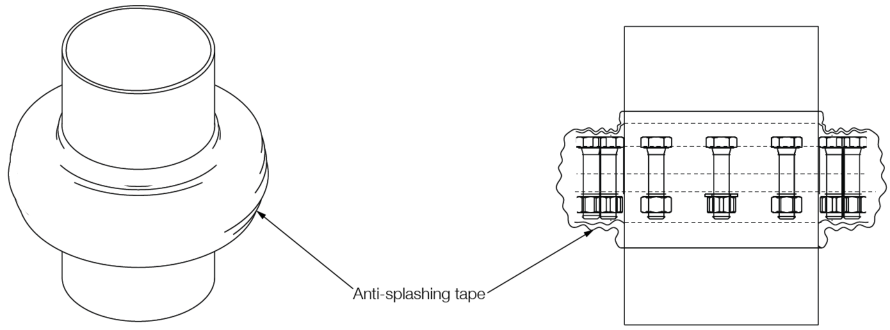

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Fig. 7.04.04b: Anti-splashing tape (FN tape)

Components for Fuel Oil System

Fuel Oil Separator

The manual cleaning type of separators are not to be recommended. Separators must be self-cleaning, either with total discharge or with partiadischarge.

Distinction must be made between installations for:

- Specific gravities < 0.991 (corresponding to ISO 8217: RMA-RMD grades and British Standard 6843 from RMA to RMH, and CIMAC from A to H-grades)
- Specific gravities > 0.991 (corresponding to ISO 8217: RME-RMK grades and CIMAC K-grades).

For the latter specific gravities, the manufacturers have developed special types of separators, e.g.:

Alfa Laval ....Alcap
Westfalia ....Unitrol
Mitsubishi ....E-Hidens II

MAN Energy Solutions also recommends using high-temperature separators, which will increase the efficiency.

The separator should be able to treat approximately the following quantity of oil:

0.23 litres/kWh in relation to CFR (certified flow rate)

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 Specified MCR can be used to determine the capacity. The separator capacity must always be higher than the calculated capacity.

Inlet temperature to separator, minimum ....98°C

CFR according to CEN, CWA 15375

The size of the separator has to be chosen according to the supplier's table valid for the selected viscosity of the Heavy Fuel Oil and in compliance with CFR or similar. Normally, two separators are installed for Heavy Fuel Oil (HFO), each with adequate capacity to comply with the above recommendation.

A separator for Marine Diesel Oil (MDO) is not a must. However, MAN Energy Solutions recommends that at least one of the HFO separators 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 separator of the same size as that for HFO.

It is recommended to follow the CIMAC Recommendation 25:

Recommendations concerning the design of heavy fuel treatment plants for diesel engines.

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Fuel Oil Supply Pump

Fuel oil viscosity, specified ....up to 700 cSt at 50°C
Fuel oil viscosity, maximum .... 700 cSt
Fuel oil viscosity, minimum .... 2 cSt
Pump head .... 4 bar
Fuel oil flow .... see ‘List of Capacities’
Delivery pressure .... 4 bar
Working temperature, maximum .... 110°C *)

*) If a high temperature separator is used, higher working temperature related to the separator must be specified.

The capacity stated in ‘List of Capacities’ is to be fulfilled with a tolerance of: –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 .... 700 cSt
Fuel oil viscosity, minimum .... 2 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: –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.

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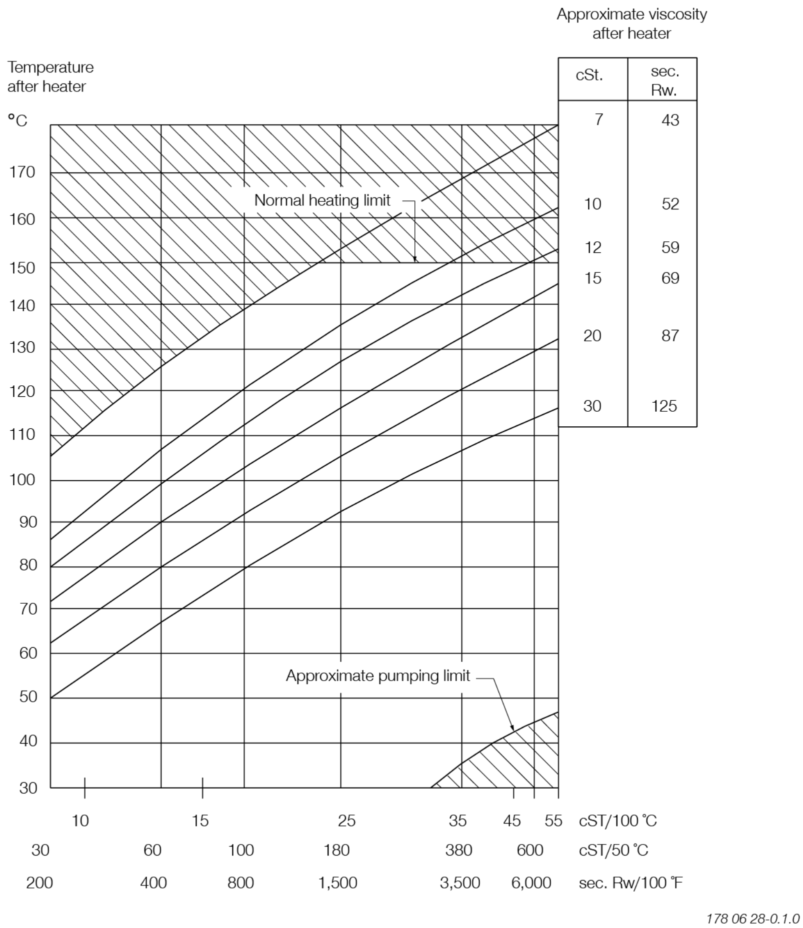

Fuel Oil Heater

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.

Fig. 7.05.01: Fuel oil heating chart

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 20 cSt at 150°C fuel oil circulating pump

Heat dissipation .... see ‘List of Capacities’ Pressure drop on fuel oil side, maximum .... 1 bar at 15 cSt

Working pressure ....10 bar

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.

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.

Alternatively positioned in the supply circuit after the supply pumps, the filter has the same flow rate as the fuel oil supply pump. In this case, a duplex safety filter has to be placed in the circulation circuit before the engine. The absolute fineness of the safety filter is recommended to be maximum 60 µm and the flow rate the same as for the circulation oil pump.

The fuel oil filter should be based on heavy fuel oil of: 130 cSt at 80°C = 700 cSt at 50°C = 7,000 sec Redwood I/100°F.

Fuel oil flow ....see 'Capacity of fuel oil circulating pump'
Working pressure ....10 bar
Test pressure .... according to Class rule
Absolute fineness, maximum ....10 µm
Working temperature, maximum ....150°C
Oil viscosity at working temperature, Oil viscosity at working temperature, maximum ....20 cSt
Pressure drop at clean filter,
maximum ....0.3 bar
Filter to be cleaned at a pressure
drop of ....0.5 bar

Note:
Some filter makers refer the fineness of the filters to be 'nominal fineness'. Thus figures will be approximately 40% lower than the 'absolute fineness' (6 µm nominal).

The filter housing shall be fitted with a steam jacket for heat tracing.

Further information about cleaning heavy fuel oil and other fuel oil types is available in MAN Energy Solutions' most current Service Letters on this subject.

The Service Letters are available at www.marine.man-es.com --> 'Service Letters'.

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Fuel Oil Filter (Option)

Located as shown in drawing or alternatively in the supply circuit after the supply pumps. In this case, a duplex safety filter has to be placed in the circulation circuit before the engine, with an absolute fineness of maximum 60 µm.

Pipe Diameter ‘D’ & ‘d’

The pipe (D) between the service tank and the supply pump is to have minimum 50% larger passage area than the pipe (d) between the supply pump and in the circulating pump. This ensures the best suction conditions for the supply pump (small pressure drop in the suction pipe).

Overflow Valve

See ‘List of Capacities’ (fuel oil supply oil pump).

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 Energy Solutions recommendations:

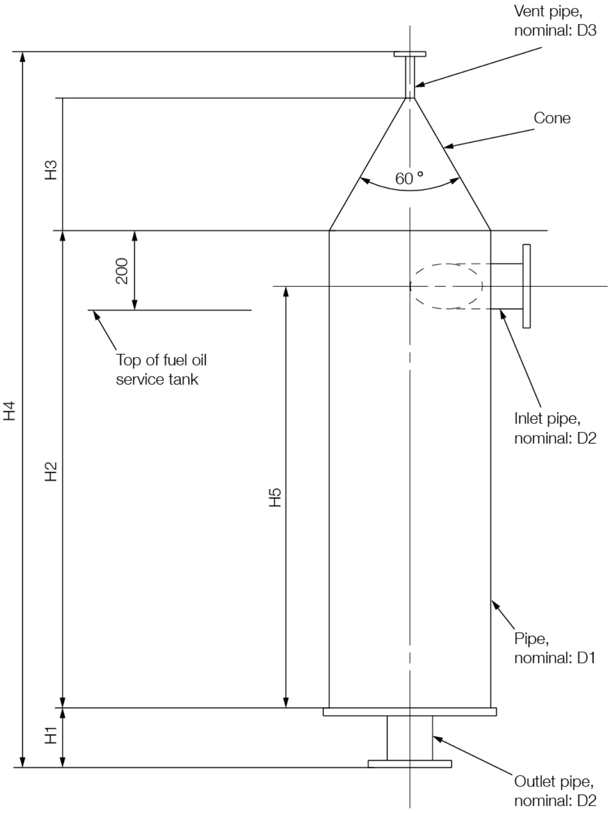
Flushing of Fuel Oil System

which is available from MAN Energy Solutions, Copenhagen.

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.

The venting tank has to be placed at the top service tank. If the venting tank is placed below the top of the service tank, the drain pipe from the automatic venting valve has to be led to a tank placed lower than the venting valve. The lower tank can be a ‘Fuel oil over flow tank’, if this tank has venting to deck.



Flow m^3/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

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Cooling of Distillate Fuels

The external fuel systems (supply and circulating systems) have a varying effect on the heating of the fuel and, thereby, the viscosity of the fuel when it reaches the engine inlet.

Today, external fuel systems on-board are often designed to have an optimum operation on HFO, which means that the temperature is kept high.

For low-viscosity distillate fuels like marine diesel oil (MDO) and marine gas oil (MGO), however, the temperature must be kept as low as possible in order to ensure a suitable viscosity at engine inlet.

Fuel Oil Viscosity at Engine Inlet

The recommended fuel viscosity range for MAN B&W two-stroke engines at engine inlet is listed in Table 7.05.03.

The lower fuel viscosity limit is 2 cSt

However, 3 cSt or higher is preferable as this will minimise the risk of having problems caused by wear for instance.

For low-viscosity fuel grades, care must be taken not to heat the fuel too much and thereby reduce the viscosity.

Range	Fuel viscosity at engine inlet, cSt
Minimum	2
Normal, distillate	3 or higher
Normal, HFO	10-15
Maximum	20

Table 7.05.03: Recommended fuel viscosity at engine inlet

Information about temperature – viscosity relationship of marine fuels is available in our publication:

Guidelines for Operation on Fuels with less than 0.1% Sulphur, SL2014-593

The publication is available at www.marine.man-es.com -->'Service Letters'.

Impact of Fuel Viscosity on Engine Operation

Many factors influence the actually required minimum viscosity tolerance during start-up and lowload operation:

- engine condition and maintenance
- fuel pump wear
- engine adjustment (mainly starting index)
- actual fuel temperature in the fuel system

Although achievable, it is difficult to optimize all of these factors at the same time. This situation complicates operation on fuels in the lowest end of the viscosity range.

Fuel Oil Cooler

To build in some margin for safe and reliable operation and to maintain the required viscosity at engine inlet, installation of a cooler will be necessary as shown in Fig. 7.01.01.

Viscosity Requirements of Fuel Pumps etc.

The fuel viscosity does not only affect the engine. In fact, most pumps in the external system (supply pumps, circulating pumps, transfer pumps and feed pumps for the separator) also need viscosities above 2 cSt to function properly.

MAN Energy Solutions recommends contacting the actual pump maker for advice.

Water in fuel emulsification

General

This section is not applicable

Second fuel supply – ME-LGIM engine

The ME-LGIM engine requires a supply of methanol at the load-dependent pressure and temperature specified in Section 7.00.

A second fuel supply system consisting of the following components and systems can meet these requirements.

- Low-flashpoint fuel supply system (LFSS)
- Fuel valve train (FVT) for controlling the fuel flow to the engine
- Auxiliary systems for leakage detection, ventilation and purging with inert gas, see Section 7.09
- Return piping to the service tank.

Fig. 7.07.01 shows the systems placed outside the engine room for a single-engine plant.

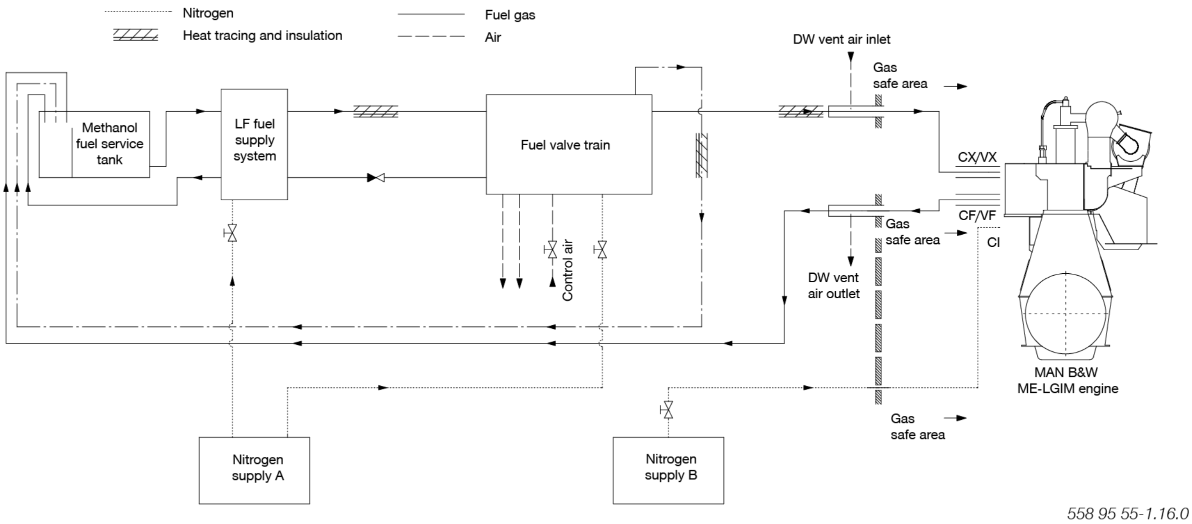

Fig. 7.07.01: Fuel supply system placed outside the engine room for ME-LGIM engine plants

Normally, the individual shipyard/contractor carries out the detailed design of LFSS and auxiliary systems. Therefore, the second fuel supply system is not subject to the type approval of the engine.

Key components

The second fuel supply system consists of key components that ensure the engine consumes a correctly specified fuel. Moreover, safety components ensure a safe engine operation and a possibility for engine shutdown.

Fuel valve train

The FVT is the ME-LGI interface to external systems.

The FVT contains valves in a double-block-and-bleed arrangement for safe isolation of the ME-LGIM engine. The main function of the FVT is to isolate the engine from the LFSS during shutdown in dual-fuel mode.

The engine control system (ECS) controls the FVT and they are closely linked.

Figs. 7.07.01 and 7.07.02 illustrate the working principles of the FVT and Fig. 7.07.02 shows the FVT with shutdown valves, redundant blow-off valves, connection for purging, and pressure testing functions.

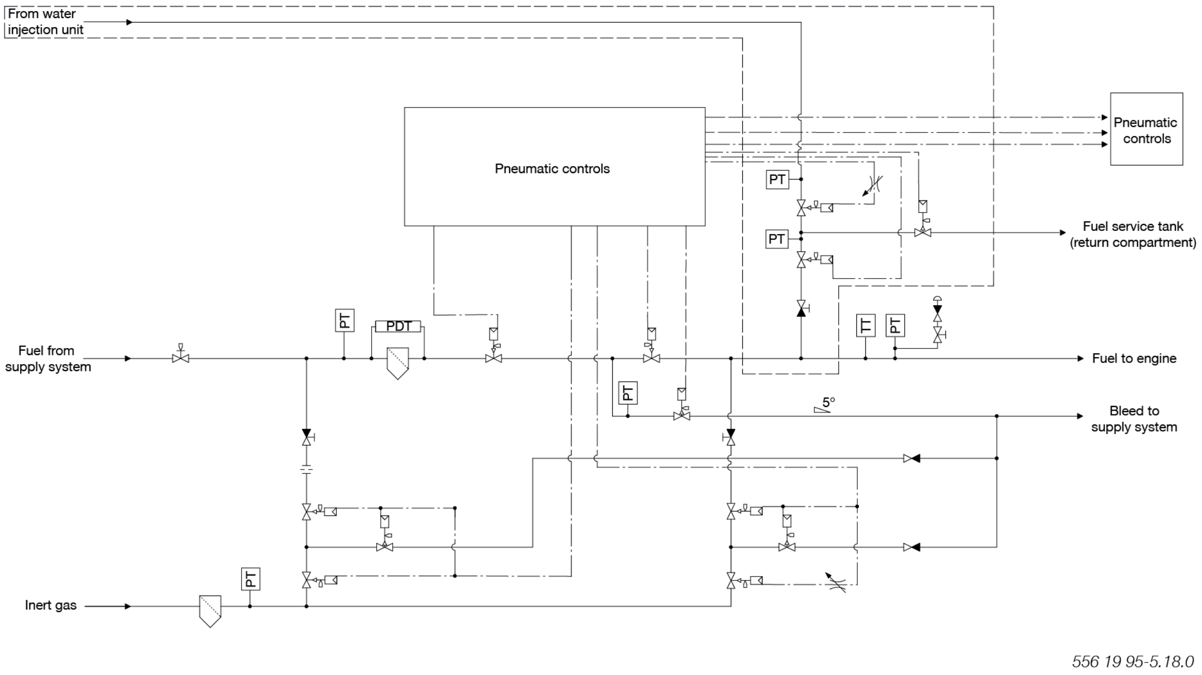

Fig. 7.07.02: Fuel valve train schematic for LGIM engines

As an option, and subject to class approval, the FVT can also function as master fuel valve.

The FVT contains various functions required by the International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk (IGC) and the International Code of Safety for Ships Using Gases or Other Low-Flashpoint Fuels (IGF). Furthermore, additional functions are required for the safe and reliable operation of the MAN B&W ME-LGIM engine.

The FVT is a controlled make of MAN Energy Solutions and an integral part of the engine. It can be configured and ordered from MAN Energy Solutions based on the engine configuration.

Location of the FVT

The expected area needed for the FVT skid is: 6 m x 3 m x 3 m (length x width x height) but it can vary for different FVT makes and engine configurations. The precise sizing must be determined on a project basis.

Careful consideration must be given to the installation of the FVT. Preferably, it should be placed outside the engine room but as close as possible to the engine to ensure the shortest piping distance.

If the FVT is installed on deck, a single-walled piping can be used from the LFSS to the FVT. From the FVT to the ME-LGIM engine below deck, a double-walled piping must be installed. In this case, it is possible to begin the double-walled pipe ventilation just after the FVT.

If it is preferred to install the FVT below deck, it is recommended to install it in a room next to the engine room.

The room with the FVT requires a separate ventilation system providing 30 air changes per hour and a hydrocarbon (HC) sensor as required by classification societies.

Filters

As standard, a filtration setup has to be included in the LFSS. A filter system of the duplex type has to be placed at the outlet of the LFSS. Each filter must have a capacity enabling the specified maximum fuel flow at the working pressure and temperature required by the engine.

Filters must be fitted with differential pressure transmitters for monitoring and alarm tripping.

At the FVT inlet, a safety filter is fitted with the purpose to protect the FVT and the ME-LGIM engine from foreign particles that could damage the sealing of the fuel valves.

For further information about filters for the LFSS and FVT, contact MAN Energy Solutions, Copenhagen at MarineInstallation2S@man-es.com

Fuel piping

For delivery of low-pressure liquid fuel (methanol) to the ME-LGIM main engine, double-walled fuel pipes can be used in both open and enclosed spaces, and for interior piping it is a requirement. Moreover, the double-walled fuel pipe requires ventilation of the annular space as described in Section 7.09.

The single-walled fuel piping can only be used in areas/rooms classified as a hazardous zone 0 or 1. In all other locations, a double-walled piping is required.

Outer pipe of double-walled piping

The outer pipe must be designed to withstand local peak pressures and low temperatures arising from a rupture of the inner pipe. The tangential membrane stress of a straight pipe should not exceed the tensile strength divided by 1.5 (Rm/1.5) when subjected to the critical pressure. The pressure ratings of all other piping components must reflect the same level of strength as the straight pipe.

Temperature range	-25°C to 60°C

The maximum total pressure loss must comply with the ventilation specification of MAN Energy Solutions, see Section 7.09.

Inner pipe maximum working pressure	16 barg
Design/test pressure	10 barg

Material

Recommended outer pipe materials are stainless steel 304L (EN 1.4306) or 316L (EN 1.4404). The selection of these materials is based on sufficient corrosion resistance, required strength, low-temperature fracture toughness, and resistance to stress corrosion chloride cracking. By using these materials, long maintenance intervals and a long service life can be offered.

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The piping should be cold-worked in order to reduce the internal surface roughness.

Maximum surface roughness	280 μm

Sizing

Outer pipe sizes are selected based on common standard sizes as per ASME B36.10 and B36.19.

Tables 7.07.01a and b provide pipe dimension guidelines for the ME-LGIM engine based on standard pipe sizes for stainless steel (EN 1.4306 and EN 1.4404).

SMCR power	NPS		OD	Sch	t	Test pressure
	DN	Inch				
[MW]	*[mm]*	*[-]*	*[mm]*	*[-]*	*[mm]*	*[barg]*
0-15.5	80	3	88.9	5	2.11	10
15.5-55	100	4	114.3	5	2.11	10
55-82.5	125	5	141.3	5	2.77	10

Table 7.07.01a: Pipe dimension guideline – supply, outer pipe recommendations for ME-LGIM engines

SMCR power	NPS		OD	Sch	t	Test pressure
	DN	Inch				
[MW]	*[mm]*	*[-]*	*[mm]*	*[-]*	*[mm]*	*[barg]*
0-55	32	1.25	42.2	5	1.65	10

Table 7.07.01b: Pipe dimension guideline – return, outer pipe recommendations for ME-LGIM engines

Single-walled and inner pipe

The following specification applies to the single-walled pipe and the inner pipe of the double-walled fuel piping which delivers methanol to the ME-LGIM main engine.

Design pressure	16 bar
Temperature range	-25°C to 50°C
Total pressure loss (max)	2 bar*
Lower calorific value (LCV)	19 MJ/kg

**This refers to the maximum allowable pressure loss in the total length of the supply piping from the LFSS to the main engine connection.*

Design calculations for the pipe are performed using the above design assumptions, and the formula specified in chapter 5.11 of the IGC code for calculation of the pipe thickness. Pipe strengths for different pipe sizes are selected based on manufacturer information according to ASME B31.3.

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The demand made on the maximum total pressure loss also applies to projects using methanol with a specific LCV. It means that for a higher flow, a larger pipe diameter is required to comply with the demand on maximum pressure loss.

Material

Recommended materials for single-walled pipes and inner pipes are stainless steel 304L (EN 1.4306) and 316L (EN 1.4404).

The selection of these materials is based on sufficient corrosion resistance, required strength, low-temperature fracture toughness, resistance to stress corrosion chloride cracking. Using these materials gives a long maintenance interval and a long service life.

The piping should be cold-worked to reduce internal surface roughness.

Maximum surface roughness	20 µm

Sizing

The guidelines in Tables 7.07.02a to b can be used for dimensioning the single-walled pipe and the inner pipe.

The pressure loss calculation is based on the length of the piping from the LFSS to the main engine inlet flange. It is shown for the fuel supply piping for the ME-LGIM engine in table 7.07.02a.

It is recommended using a design with welded bends and a minimum radius as per ASME B16.9.

Power range	Max. flow	NPS		Pipe OD	Sch	t	Test pressure	Pressure loss	
		DN	Inch					50 m	100 m
[MW]	*[kg/h]*	*[mm]*	*[-]*	*[mm]*	*[-]*	*[mm]*	*[barg]*	*[bar]*	
0-8.5	3,600	25	1	33.4	10	2.77	24	0.65	1.30
8.5-15.5	6,500	32	$1^{1/4}$	42.2	10	2.77	24	0.50	1.01
15.5-25	10,500	40	$1^{1/2}$	48.3	10	2.77	24	0.58	1.15
25-25	23,000	50	2	60.3	10	2.77	24	0.74	1.49
55-82.5	34,500	65	$2^{1/2}$	73	10	3.05	24	0.59	1.18

Table 7.07.02a: Dimension guidelines , EN150 HB (304L), EN150YB (316L) supply piping recommendations for ME-LGIM engines

SMCR power	NPS		OD	Sch	t	Test pressure
	DN	Inch				
[MW]	*[mm]*	*[-]*	*[mm]*	*[-]*	*[mm]*	*[barg]*
0-82.5	15	½	21.3	10	2.11	24

Table 7.07.02b: Dimension guidelines , EN150 HB (304L), EN150YB (316L) return/recirculation piping recommendations for ME-LGIM engines

Fuel supply pressure range

The LFSS can generate the methanol fuel pressure in different ways depending on the chosen technology. The number of coolers, stages, and type of pumps can vary from maker to maker.

MAN Energy Solutions only requires that the LFSS must be designed and manufactured to operate within the margins of the fuel supply pressure range of 13 bar with a margin of ±0.5 bar.

The supply pressure must be kept constant from zero to maximum fuel flow.

Control of second fuel supply systems

Section 16.00 gives a description of the ME-LGI engine control system.

The fuel pressure is specified by a fixed setpoint and the engine load on second fuel, or in dual-fuel mode (methanol), is specified by the LGI-ECS.

Safety standards for second fuel supply system

All equipment must comply with but not necessarily be limited to the following:

1. Full class requirements for UMS notation and ACCU notation, etc. (ABS, LRS, and DNV)
2. IGF Code and/or IGC Code as applicable

   IGF code: International Code of Safety for Ships Using Gases or other Low-Flashpoint Fuels

   IGC Code: International Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk
3. SOLAS and Flag State requirements for fire safety and detection systems
4. Other standards to be fulfilled:

   DNV Rules Part 6 Chapter 13 Gas Fuelled Engine Installations

   ABS applicable sections in their guidelines for propulsion and auxiliary for gas-fuelled ships

   ALPEMA SE 2000 or latest, Standards for Plate-fin Heat Exchangers

   ASME VIII div 1 Plate-fin Heat Exchangers

   ASME BPVC-VIII-3 Construction of High Pressure Vessels

   IEC 60092 Electrical installations in ships Certified according to ATEX directives.

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Second fuel supply system

See Section 7.07

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Auxiliary systems for second fuel supply system

Auxiliary systems for ME-LGIM second fuel (methanol) supply include:

- Leakage detection and ventilation system which ventilates the outer pipe of the double-walled piping completely and incorporates leakage detection
- Ventilation of single-walled pipe in the sealing oil system, similar to the second fuel system
- Nitrogen system, which purges the second fuel supply system on the engine and the second fuel supply piping from the main engine to the fuel valve train (FVT). Nitrogen purging moves any leftover methanol back to the service tank. Furthermore, the nitrogen system can carry out methanol freeing of the low-flashpoint fuel supply system (LFSS) and return methanol to the service tank
- Second fuel return piping to the service tank.

Fig. 7.07.01 shows the ME-LGIM second fuel supply auxiliary systems and connecting piping.

Capacities

The capacities of ME-LGIM second fuel supply auxiliary systems are listed in the CEAS report for the actual project, see Section 20.02.

Ventilation system and leakage detection

The purpose of the leakage detection and ventilation system is to ensure that the outer pipe of the double-walled second fuel pipe system is constantly ventilated by air. At least 30 air changes per hour are needed according to requirements set by the applicable codes and regulations. The ME-LGIM engine and all double-walled second fuel piping require ventilation.

The sealing oil system also has a ventilated pipe similar to the second fuel system but only a single-walled pipe which is exhausted outside the engine room. This is done as a precaution to remove any methanol vapours that could occur in the sealing oil tank.

General data for ventilation systems

MAN Energy Solutions recommends that ventilation air for venting the double-walled piping is service or working air from the ship's compressed air system. Alternatively, starting air from the main engine starting air receiver can be used if air consumption and starting air compressor running conditions are examined.

Other air supply systems can be installed if they comply with the specification in Table 7.09.01.

A reduction valve regulates the supply inlet pressure to slightly above atmospheric pressure to maintain a small overflow of air at the ventilation air box, see Section 13.00.

The design of the air box ensures for all operating conditions that there cannot be an overpressure at the ventilation air inlet, according to existing rules and legislations.

Description	Value
Medium	Air
Ventilation air supply quality	According to ISO 8573-1: 1–4
Particle size	Class 7 according to ISO 8573-1, maximum allowed particle size: 40 µm
Oil content	Class 4 according to ISO 8573-1, maximum allowed content: 50 mg/m^3
Dewpoint temperature	Should be ≤ +3°C (condition should apply for tropical ambiance conditions at atmospheric pressure)
Pressure range	-20 mbarg to less than 1 atm
Air changes	Minimum 30 air changes per hour Maximum 45 air changes per hour

Table 7.09.01: General data for ventilation systems

Temperature of ventilation air

Take ambient temperatures into consideration when designing the ventilation system. Temperatures below the prescribed dewpoint will lead to frost formation and should be avoided.

For specific projects, the ventilation piping might require insulation or heat tracing to ensure that the ventilation air is always within the specification.

Ventilation air fan capacity

To decide the necessary capacity for the fan, the volume of intermediate spaces of the pipe system must be calculated. The complete volume consists of:

- The volume of annular spaces in the second fuel double-walled piping
- The vented volume in the second fuel control block.

Contact MAN Energy Solutions for further information regarding pipe sizes and venting volume in the second fuel block for a specific engine type.

Based on the calculated volume, the capacity must comply with the requirements for air changes in Table 7.09.01.

Fan requirements and installation guidelines

The fan must comply with requirements from class rules. A redundant fan may be implemented to reduce downtime in dual-fuel running mode, since a running fan is required in this mode. Fig. 7.09.01 shows the leakage detection and ventilation system for the double-walled piping.

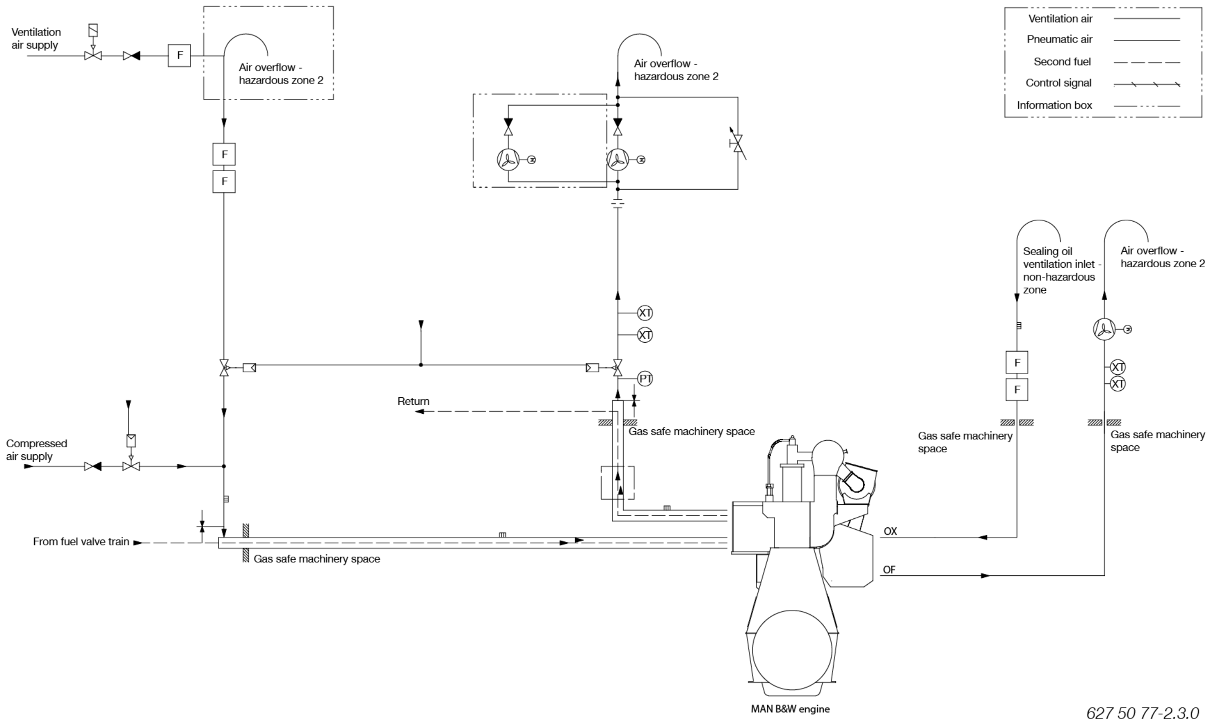

Fig. 7.09.01: Leakage detection and ventilation system for double-walled piping

The ventilation inlet must be located in open air away from ignition sources, and away from the ventilation outlet. Inlet and outlet must be considered as hazardous zone 1.

Venting air fan control

Flow switches and hydrocarbon (HC) detectors send/receive signals to/from the ECS system.

Two flow switches must be installed in the venting air intake to monitor a sufficient air flow. Fig. 7.09.01 shows the locations of the flow switches.

Leakage detection

To detect any second fuel leaks into the annular space of the double-walled piping, two HC sensors must be installed in the outlet of the ventilation system. Fig. 7.09.01 shows the locations of the HC sensors.

Safety standards for leakage detection and ventilation system

All equipment and designs must comply with but not necessarily be limited to the following:

- Current International Gas Code (IGC) and current International Gas Fuel for ships (IGF) requirements
- Classification requirements from the specified classification society
- SOLAS and flag state requirements for fire safety and detection systems
- IEC 60092 Electrical installations in Ships.

Further information about ventilation systems and a calculation of venting fan capacity are available from MAN Energy Solutions, Copenhagen.

Nitrogen system

A purging sequence is performed to dilute the remaining second fuel in the second fuel system after an engine stop. The dilution is done by pressurising the pipe volume with nitrogen and then releasing the methanol/nitrogen-mixture to the service tank.

The sequence is performed a consecutive number of times to reach a safe second fuel concentration level in the system (below the lower explosion limit (LEL)). The sequential procedure will be continued for a predefined number of purges determined by parameter settings in the ECS. See the diagram of second fuel supply auxiliary systems in Fig. 7.07.01.

The second fuel supply system also requires nitrogen for purging, which can either be supplied from a common nitrogen system, or a separate standalone system. This depends on the individual installation. For the methanol start-up sequence, nitrogen is filled into the return piping from the FBIVM to the service tank. Detection of a leakage can then be done in the return system.

A sufficient quantity of nitrogen must be available on-board before the engine is operated on second fuel.

For further information about the nitrogen system, contact MAN Energy Solutions, Copenhagen.

Control of nitrogen system

After a second fuel shutdown, the ECS initiates purging of the second fuel systems. As a minimum, the ECS will perform the needed purging sequences, the actual number depends on the nitrogen system layout. Before a start-up in dual-fuel mode, the nitrogen will be used for tightness verification test and leakage detection of the return system.

Purging volume and storage capacity

The purging storage volume must be designed for a number of consecutive starts on second fuel, each including several purging sequences, and for purging before and after engine operation on second fuel.

To calculate the purging volume, the total volume of piping being purged must be calculated. The purging sequence is described in the following.

The complete system purge volume includes:

- Nitrogen system and nitrogen supply piping
- Second fuel supply pipe from FVT to main engine
- Second fuel return pipe from main engine to service tank
- Second fuel volume on the engine.

In addition, any volume between service tank and FVT (including LFSS) should be considered, as recommended by the LFSS supplier.

Buffer tank volume

The buffer tank volume for nitrogen has to be calculated by the ship designer. The volume needs to be sufficient for tightness verification test, start-up procedures and shutdown. The number of consecutive second fuel starts are to be decided by the ship designer. Calculation method and recommendations can be provided by MAN Energy Solutions.

For further information, contact MAN Energy Solutions at MarineInstallation2S@man-es.com

Nitrogen generator

Since the nitrogen buffer tank pressure decreases when nitrogen is consumed, the nitrogen generator must be able to uphold a minimum capacity.

The minimum capacity depends on the nitrogen buffer tank volume, and the nitrogen consumption per purging sequence. The designer should evaluate the overall nitrogen system setup, and assess the most optimal balance between nitrogen buffer tank size and nitrogen generator capacity, while complying with MAN Energy Solutions' minimum requirements for the nitrogen system.

Nitrogen booster requirements

A nitrogen booster/compressor (N_2 booster) should be fitted for tightness verification testing of the second fuel system.

During commissioning and after reassembling components in the second fuel system, a tightness verification test must be performed by pressurising the engine with nitrogen to the maximum working pressure. This is done to verify that the components have been assembled properly and that second fuel pressure can be safely applied.

The requirements for the tightness verification test are:

Tightness testing pressure:	13 bar

The nitrogen boost should be designed to increase the nitrogen pressure to the testing pressure within a reasonable time limit. The time it takes the system to reach the required tightness testing pressure, after opening the nitrogen buffer tank, determines the necessary capacity of the N_2 booster.

An uninterrupted and sufficient nitrogen supply should be provided to ensure that the N_2 booster operation complies with manufacturer specifications.

MAN Energy Solutions does not demand Ex-approved N_2 boosters but the equipment must comply with hazardous zone requirements of the room where it is installed, if any.

MAN Energy Solutions recommends an electrically driven N_2 booster type, if the N_2 booster is pneumatically driven, special attention should be paid to the drop in outlet capacity when the outlet pressure increases.

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Lubricating and cooling oil system

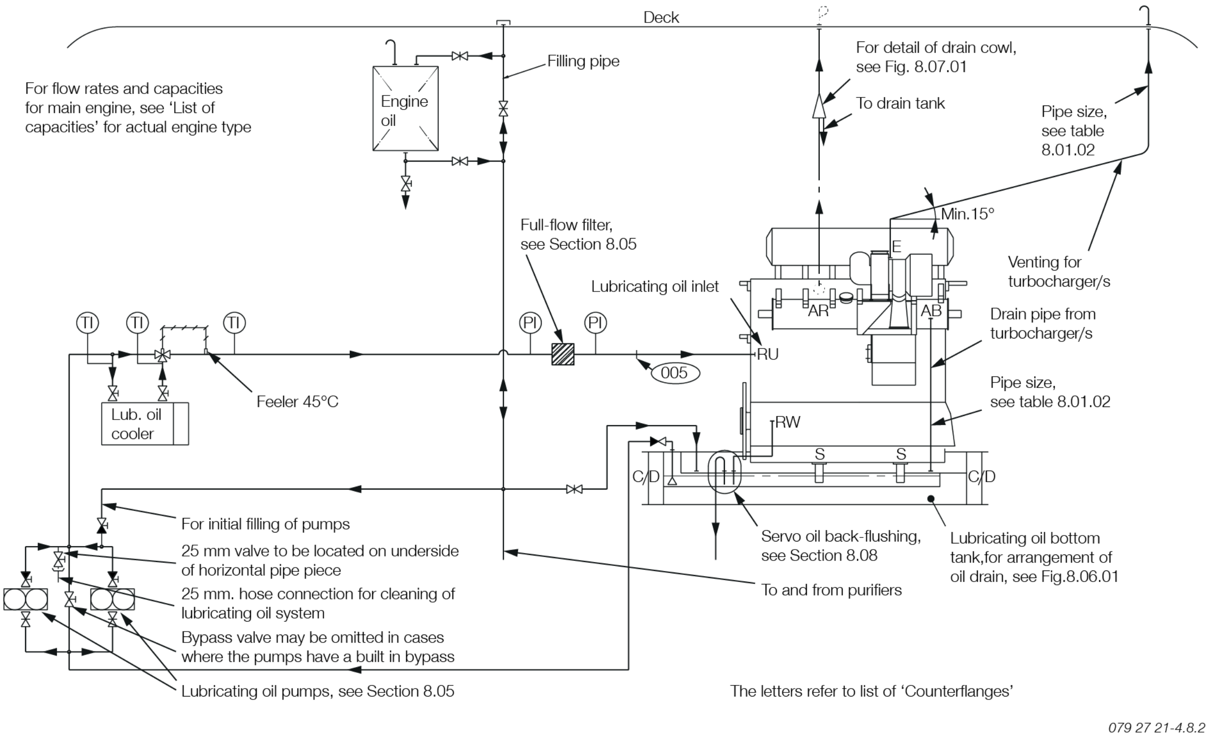
The lubricating oil is pumped from a bottom tank by means of the main lubricating oil pump to the lubricating oil cooler, a thermostatic valve and, through a full-flow filter, to the engine inlet RU, Fig. 8.01.01.

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, moment compensator, torsional vibration damper, exhaust valve, Hydraulic Cylinder Unit and gas control block.

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.



The letters refer to list of 'Counterflanges'
* Venting for MAN or Mitsubishi turbochargers only

Lubrication of Turbochargers

Turbochargers with slide bearings are normally lubricated from the main engine system. AB is outlet from the turbocharger, see Fig. 8.03.01 to Fig. 8.03.04.

Fig. 8.03.01 to Fig. 8.03.04 show the lube oil pipe arrangements for various turbocharger makes.

Turbocharger Venting and Drain Pipes

MAN

Type	No. of TC	Venting pipe		Drain
		Each TC DN	Collect TC DN	Pipe from TC DN
TCR22	1	50	50	65
TCA44	1	65	65	65
	2	65	100	100
TCA55	1	65	65	65
	2	65	100	100
TCA66	1	80	80	80
	2	80	125	125
TCA77	1	100	100	100
	2	100	125	125
TCA88	1	125	125	125
	2	125	150	150
	3	125	200	200
	4	125	250	250

Accelleron

Type	No. of TC	Venting pipe		Drain
		Each TC DN	Collect TC DN	Pipe from TC DN
A165-L A265-L	1	60	65	65
	2	60	80	80
A170-L A270-L	1	65	65	65
	2	65	90	90
A175-L A275-L	1	65	65	65
	2	65	100	100
	3	65	125	125
A180-L A280-L	1	80	80	80
	2	80	100	100
	3	80	125	125
A185-L	1	80	80	80

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Mitsubishi (MHI)

Type	No. of TC	Venting pipe		Drain
		Each TC DN	Collect TC DN	Pipe from TC DN
MET33	1	40	40	65
	2	40	80	90
MET42	1	50	50	80
	2	50	65	125
MET53	1	65	65	90
	2	65	80	125
	3	65	100	150
MET66	1	80	80	100
	2	80	100	150
	3	80	125	175
	4	80	150	225
MET71	1	80	80	125
	2	80	100	175
	3	80	125	225
	4	80	150	300
MET83	1	100	100	125
	2	100	125	175
	3	100	150	225
	4	100	175	300
MET90	1	100	100	125
	2	100	125	175
	3	100	150	225
	4	100	175	300

Table. 8.01.02: Turbocharger venting and drain pipes

For size of turbocharger inlet pipe see ‘List of capacities’

079 27 21-4.8.1

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Hydraulic power supply unit

Hydraulic power for the ME hydraulic-mechanical system for activation of the fuel injection and the exhaust valve is supplied by the Hydraulic Power Supply (HPS) unit.

As hydraulic medium, normal lubricating oil is used, as standard taken from the engine's main lubricating oil system and filtered in the HPS unit.

HPS Connection to Lubrication Oil System

Internally on the engine, the system oil inlet RU is connected to the HPS unit which supplies the hydraulic oil to the Hydraulic Cylinder Units (HCUs). See Figs. 16.01.02a and 16.01.02b.

RW is the oil outlet from the automatic backflushing filter.

The hydraulic oil is supplied to the Hydraulic Cylinder Units (HCU) located at each cylinder. From here the hydraulic oil is diverted to the multi-way valves, which perform the fuel injection and open the exhaust valve respectively: Electronic Fuel Injection (ELFI) and Proportional Exhaust Valve Actuator (PEVA). The exhaust valve is closed by the conventional 'air spring'.

The electronic signals to the multi-way valves are given by the Engine Control System, see Chapter 16, Engine Control System (ECS).

HPS Configurations

The HPS pumps are driven either mechanically by the engine (via a step-up gear from the crankshaft) or electrically.

The HPS unit is mounted on the engine no matter how its pumps are driven.

With mechanically driven pumps, the HPS unit consists of:

- an automatic and a redundant filter
- three to five engine driven main pumps
- two electrically driven start-up pumps
- a safety and accumulator block

as shown in Fig. 8.02.01.

With electrically driven pumps, the HPS unit differs in having a total of three pumps which serve as combined main and start-up pumps.

Motor Start Method

Direct Online Start (DOL) is required for all the electric motors for the pumps for the Hydraulic Power Supply (HPS) to ensure proper operation under all conditions, including the start up against maximum pressure in the system.

HPS unit types

Altogether, three HPS configurations are available:

- STANDARD mechanically driven HPS, EoD: 4 40 160, with mechanically driven main pumps and start-up pumps with capacity sufficient to deliver the start-up pressure only. The engine cannot run with all engine driven main pumps out of operation, whereas 66% engine load is available in case one main pump is out.

- COMBINED mechanically driven HPS unit, EoD: 4 40 167 with electrically driven start-up pumps with back-up capacity. In this case, at least 15% engine power is available as back-up power if all engine driven pumps are out
- electrically driven HPS, EoD: 4 40 161, with 66% engine load available in case one pump is out.

The electric power consumption of the electrically driven pumps should be taken into consideration in the specification of the auxilliary machinery capacity.

Hydraulic Power Supply Unit, Engine Driven, and Lubricating Oil Pipes

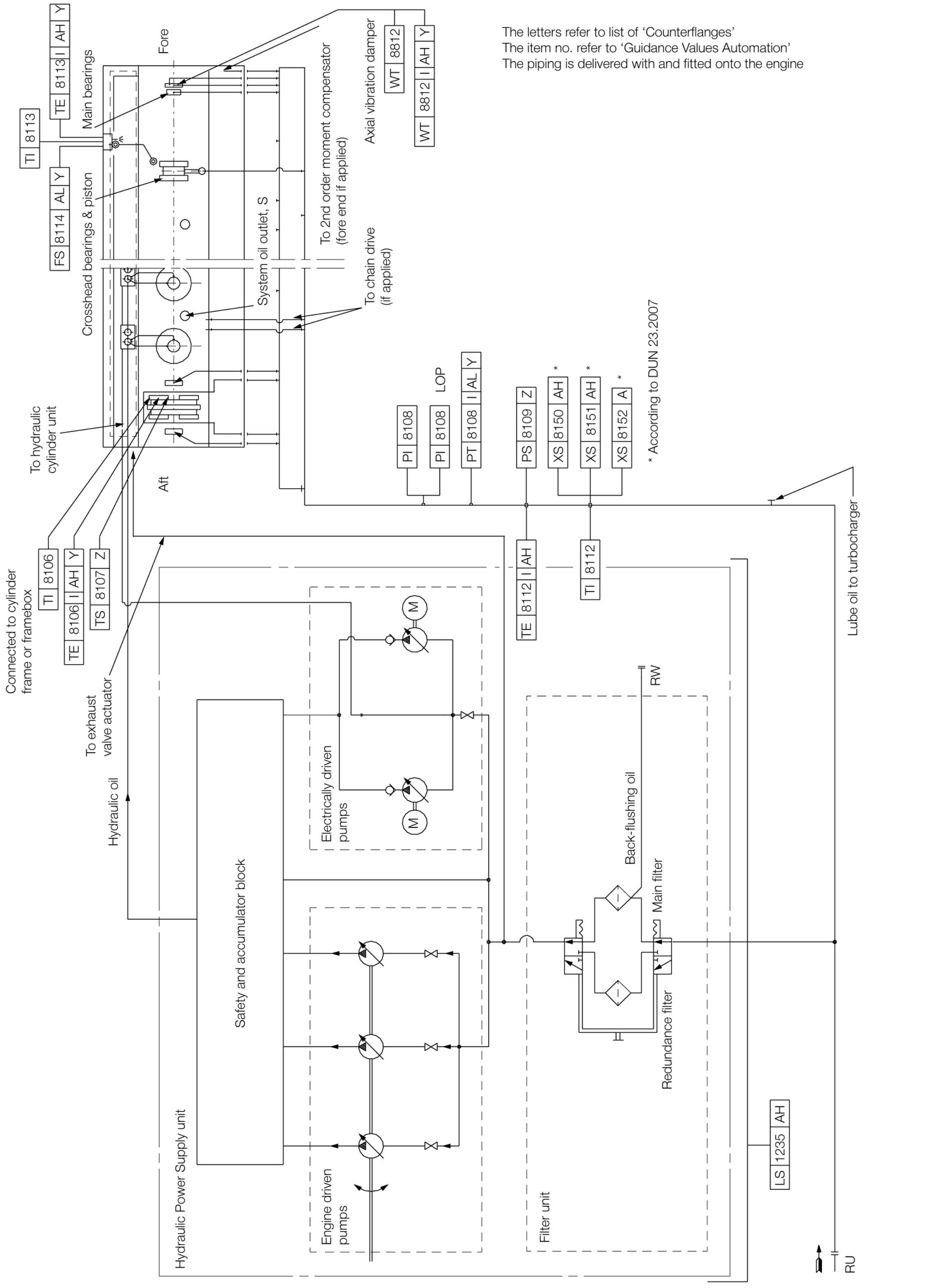

Fig. 8.02.01: Engine driven hydraulic power supply unit and lubricating oil pipes

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Lubricating oil pipes for turbochargers

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System oil list, consumption and centrifuges

System oil – a versatile lubricating oil used for many purposes

System oil is the common designation of a lubricating oil used for different purposes in systems in and on the engine. The system oil is used as:

1. Circulating oil
   - Lubrication of crosshead bearings, crankshaft bearings, main and thrust bearings
   - Cooling of pistons
   - Turbocharger(s)
2. Hydraulic oil
   - ME-system, hydraulic block
3. Control oil
   - Activates (control) valves, etc.
4. Sealing oil in dual-fuel engines
   - Confines low-flashpoint (LF) fuel in LF-systems (various systems and types).

List of system oils

MAN Energy Solutions recommends using system oils (circulating oil) with the following main properties:

- SAE 30 viscosity grade
- BN level 5 - 10
- high corrosion protection
- good anti-oxidant properties
- high detergency and dispersancy.

Adequate dispersion and detergent properties will keep the crankcase and piston cooling spaces clean of deposits.

Table 1 lists major international system oil brands tested in service, and which have passed the testing procedure and obtained a No Objection Letter (NOL). Do not consider this list to be complete, as other system oils with NOLs from MAN Energy Solutions can be equally suitable. System oils are recommended for all MAN B&W two-stroke engines, independent of Mark number.

Company	Circulating oil SAE 30, BN 5 - 10
Castrol	Castrol CDX 30
Chevron Lubricants	Veritas 800 Marine 30
ENEOS Corporation	Marine S30
ExxonMobil	Mobilgard 300 C
Gulf Oil Marine	GulfSea Superbear 3006
Shell	Shell Melina S 30
Sinopec Lubricant co.	System Oil 3005
SK Lubricants	SK Supermar AS
TotalEnergies Lubmarine	Atlanta Marine D3005

Table 1: Examples of international system oil brands that have an NOL from MAN Energy Solutions.

Do not consider the list complete, as oils from other companies can be equally suitable. Further information can be obtained from the engine builder or MAN Energy Solutions, Copenhagen.

System oil consumption

The system oil consumption depends on factors like backflushing from centrifuges and drain from stuffing boxes.

Furthermore, the consumption varies for different engine sizes as well as operational and maintenance patterns.

System oil centrifuges

Automatic centrifuges with total or partial discharge must be used.

The nominal capacity of the centrifuge must be according to the supplier's recommendation for system oil, based on the figure:

0.136 litre/kWh

The nominal MCR is used as the total installed power.

Recommendations of engine lubrication are available in the most current Service Letters on this subject at www.marine.man-es.com --> 'Two-Stroke' --> 'Service Letters'.

Notwithstanding the foregoing, it remains the responsibility of the owner/operator of an engine to ensure that suitable fuels and lubes are conditioned and used in order to prevent damage to the engine and other equipment on board. MAN Energy Solutions disclaims any and all liability and cannot be held responsible for any damage to the engine, engine components or other equipment on board that may be caused by the use of the mentioned lubricants.

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Components and installation

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 ....4.8 bar
Delivery pressure ....4.8 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 25mm 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 ....4.8 bar
Pressure drop on oil side ....maximum 0.5 bar
Cooling water flow .... see ‘List of capacities’
Cooling water temperature at inlet:

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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 ....4.8 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.

LPS booster pump

The hydraulic system is equipped with a booster pump for the low pressure supply (LPS) oil inside of the flange RU, Fig. 8.05.00.

The purpose of the pump is to ensure proper deaerating of the oil which enters the HCUs and the gas control blocks by increasing the LPS oil pressure from 2 to 5 bar (approximately).

The LPS booster pump must be run in parallel with the main lubricating oil pumps and therefore started together. For the LPS booster pump to start up, the oil pressure on the inlet side must be sufficient as monitored by the pressure switch on the inlet side. The purpose of the switch is to protect the LPS booster pump against dry-running operation.

At maintenance of the booster pump and other components, the LPS oil can be drained to waste tank via a drain plug. Otherwise the LPS oil is drained to the main tank. A bypass valve opens for flow of the approx. 2 bar low pressure oil to the HCU's in case the LPS booster pump stops.

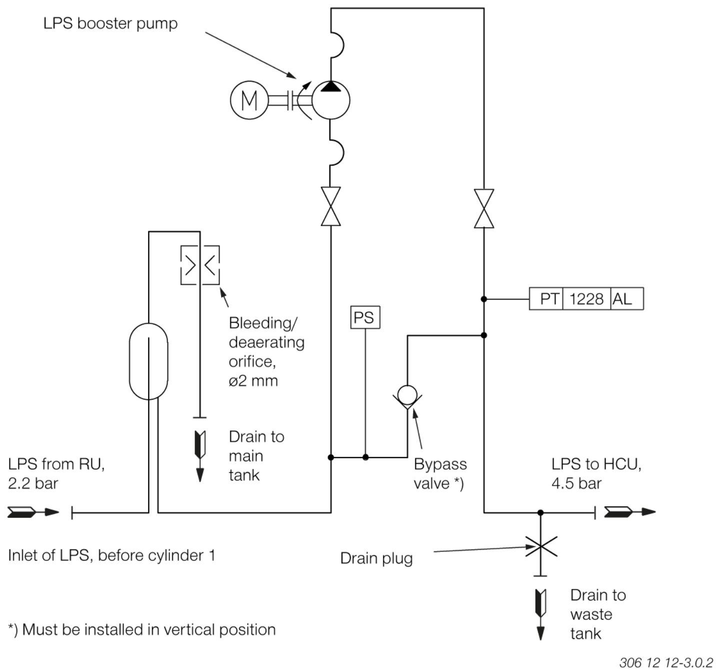

Fig. 8.05.00 LPS booster pump circuit

Flushing of Lubricating Oil Components and Piping System at the Shipyard

During installation of the lubricating oil system for the main engine, it is important to minimise or eliminate foreign particles in the system. This is done as a final step onboard the vessel by flushing the lubricating oil components and piping system of the MAN B&W main engine types ME/ME-C/ME-B/-GI/-GA before starting the engine.

At the shipyard, the following main points should be observed during handling and flushing of the lubricating oil components and piping system:

- Before and During Installation

Components delivered from subsuppliers, such as pumps, coolers and filters, are expected to be clean and rust protected. However, these must be spot-checked before being connected to the piping system.

All piping must be 'finished' in the workshop before mounting onboard, i.e. all internal welds must be ground and piping must be acid-treated followed by neutralisation, cleaned and corrosion protected.

Both ends of all pipes must be closed/sealed during transport.

Before final installation, carefully check the inside of the pipes for rust and other kinds of foreign particles.

Never leave a pipe end uncovered during assembly.

- Bunkering and Filling the System

Tanks must be cleaned manually and inspected before filling with oil.

When filling the oil system,
MAN Energy Solutions recommends that new oil is bunkered through 6 µm fine filters, or that a purifier system is used. New oil is normally delivered with a cleanliness level of XX/23/19 according to ISO 4406 and, therefore, requires further cleaning to meet our specification.

- Flushing the Piping with Engine bypass

When flushing the system, the first step is to bypass the main engine oil system. Through temporary piping and/or hosing, the oil is circulated through the vessel's system and directly back to the main engine oil sump tank.

If the system has been out of operation, unused for a long time, it may be necessary to spot-check for signs of corrosion in the system. Remove end covers, bends, etc., and inspect accordingly.

It is important during flushing to keep the oil warm, approx 60°C, and the flow of oil as high as possible. For that reason it may be necessary to run two pumps at the same time.

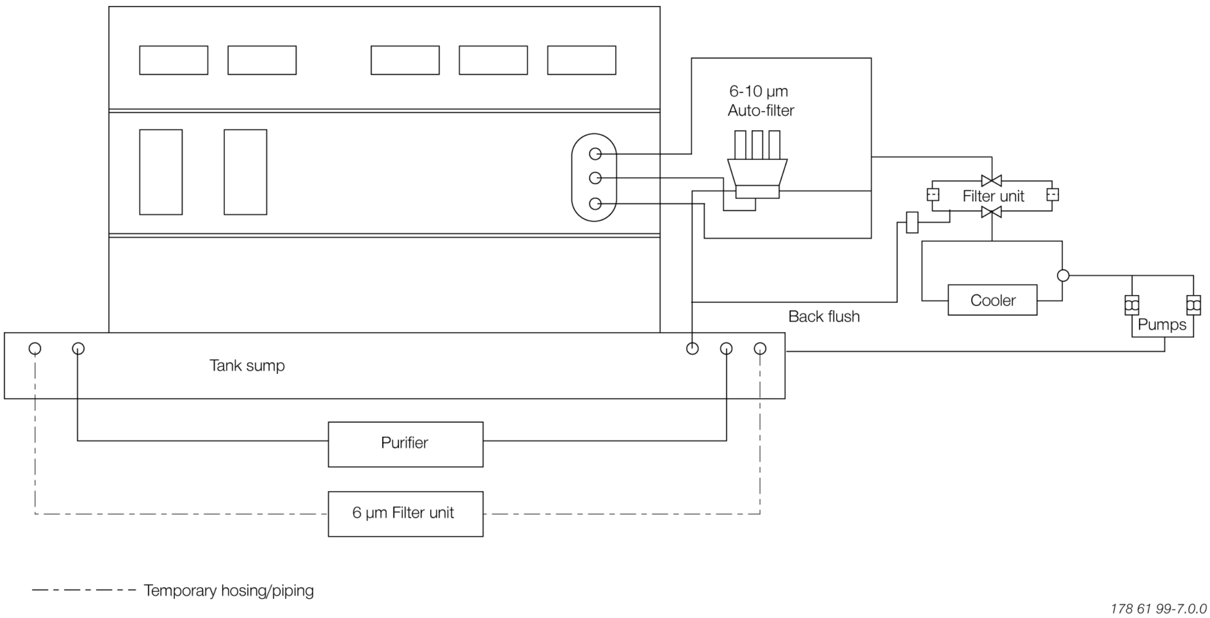

Fig. 8.05.01: Lubricating oil system with temporary hosing/piping for flushing at the shipyard

- Filtering and Removing Impurities

In order to remove dirt and impurities from the oil, it is essential to run the purifier system during the complete flushing period and/or use a bypass unit with a 6 μm fine filter and sump-to-sump filtration, see Fig. 8.05.01.

Furthermore, it is recommended to reduce the filter mesh size of the main filter unit to 10-25 μm (to be changed again after sea trial) and use the 6 μm fine filter already installed in the auto-filter for this temporary installation, see Fig. 8.05.01. This can lead to a reduction of the flushing time.

The flushing time depends on the system type, the condition of the piping and the experience of the yard. (15 to 26 hours should be expected).

- Cleanliness Level, Measuring Kit and Flushing Log

MAN Energy Solutions specifies ISO 4406 XX/16/13 as accepted cleanliness level for the ME/ME-C/ME-B/-GI/-GA hydraulic oil system, and ISO 4406 XX/19/15 for the remaining part of the lubricating oil system.

The amount of contamination contained in system samples can be estimated by means of the Pall Fluid Contamination Comparator combined with the Portable Analysis Kit, HPCA-Kit-0, which is used by MAN Energy Solutions. This kit and the Comparator included is supplied by Pall Corporation, USA, www.pall.com.

It is important to record the flushing condition in statements to all inspectors involved. The MAN Energy Solutions Flushing Log form, which is available on request, or a similar form is recommended for this purpose.

- Flushing the Engine Oil System

The second step of flushing the system is to flush the complete engine oil system. The procedure depends on the engine type and the condition in which the engine is delivered from the engine builder. For detailed information we recommend contacting the engine builder or MAN Energy Solutions.

- Inspection and Recording in Operation

Inspect the filters before and after the sea trial.

During operation of the oil system, check the performance and behavior of all filters, and note down any abnormal condition. Take immediate action if any abnormal condition is observed. For instance, if high differential pressure occurs at short intervals, or in case of abnormal back flushing, check the filters and take appropriate action.

Further information and recommendations regarding flushing, the specified cleanliness level and how to measure it, and how to use the NAS 1638 oil cleanliness code as an alternative to ISO 4406, are available in our publication:

Filtration Handbook, Filtration and Flushing Strategy

The publication is available at www.man-es.com → 'Marine' → 'Products' → 'Planning Tools and Downloads' → 'Technical Papers'.

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.

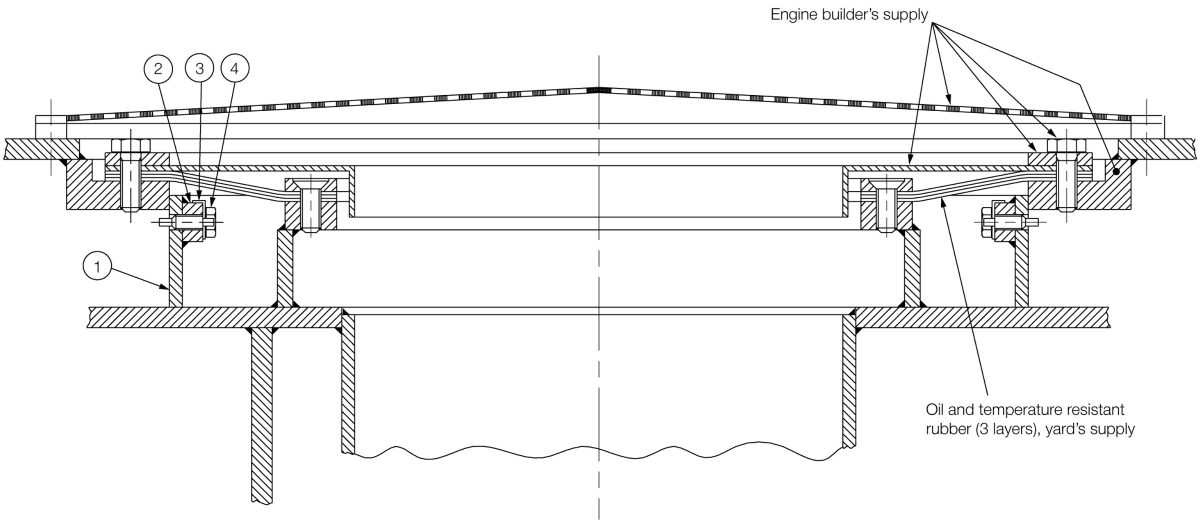

Fig. 8.05.02: Lubricating oil outlet

Lubricating oil tank

General

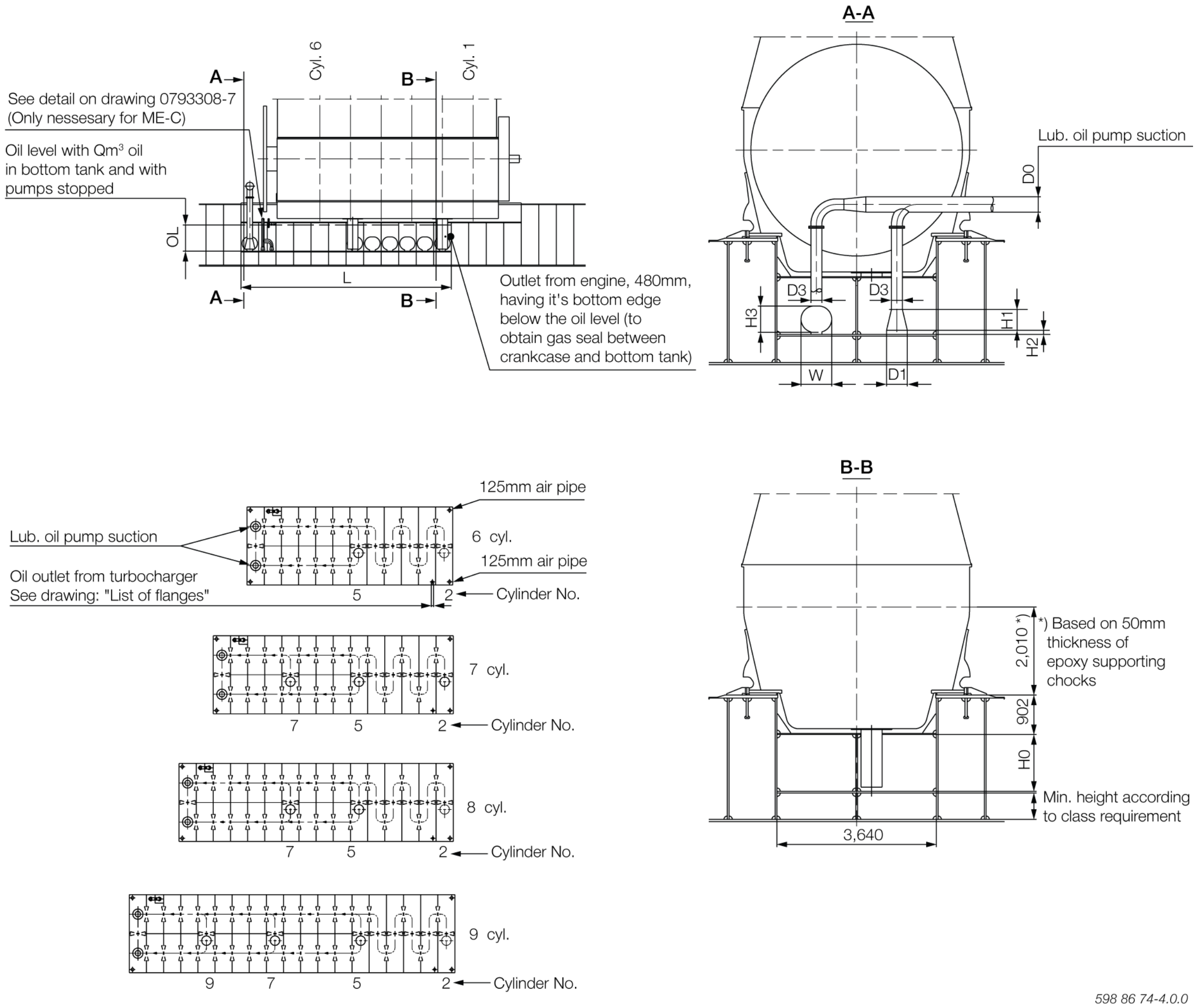

Fig. 8.06.01: Lubricating oil tank, with cofferdam

Note:
When calculating the tank heights, allowance has not been made for the possibility that a part of the oil quantity from the system outside the engine may - when the pumps are stopped - be returned to the bottom tank.

Provided that the system outside the engine is so executed that a part of the oil quantity is drained back to the tank, when the pumps are stopped, the height of the bottom tank indicated on the drawing is to be increased corresponding to this quantity.

Cyl. No.	Drain at Cyl. No.	D0	D1	D3	H0	H1	H2	H3	W	L	OL	Qm^3
6	2-5	350	2x475	2x250	1,295	475	95	600	700	9,600	1,195	44.1
7	2-5-7	375	2x550	2x275	1,360	550	110	600	700	11,200	1,260	54.2
8	2-5-7	400	2x550	2x275	1,425	550	110	600	700	12,800	1,325	65.1
9	2-5-7-9	400	2x600	2x300	1,535	600	110	600	700	16,000	1,435	88.2

Table 8.06.01: Lubricating oil tank, with cofferdam

If space is limited, however, other solutions are possible. Minimum lubricating oil bottom tank volume (m^3) is:

6 cyl.	7 cyl.	8 cyl.	9 cyl.
30.6	35.6	40.0	45.0

Lubricating oil tank operating conditions

The lubricating oil bottom tank complies with the rules of the classification societies by operation under the following conditions:

Angle of inclination, degrees			
Athwartships		**Fore and aft**	
Static	Dynamic	Static	Dynamic
15	22.5	5	7.5

8.06 Lubricating oil tank

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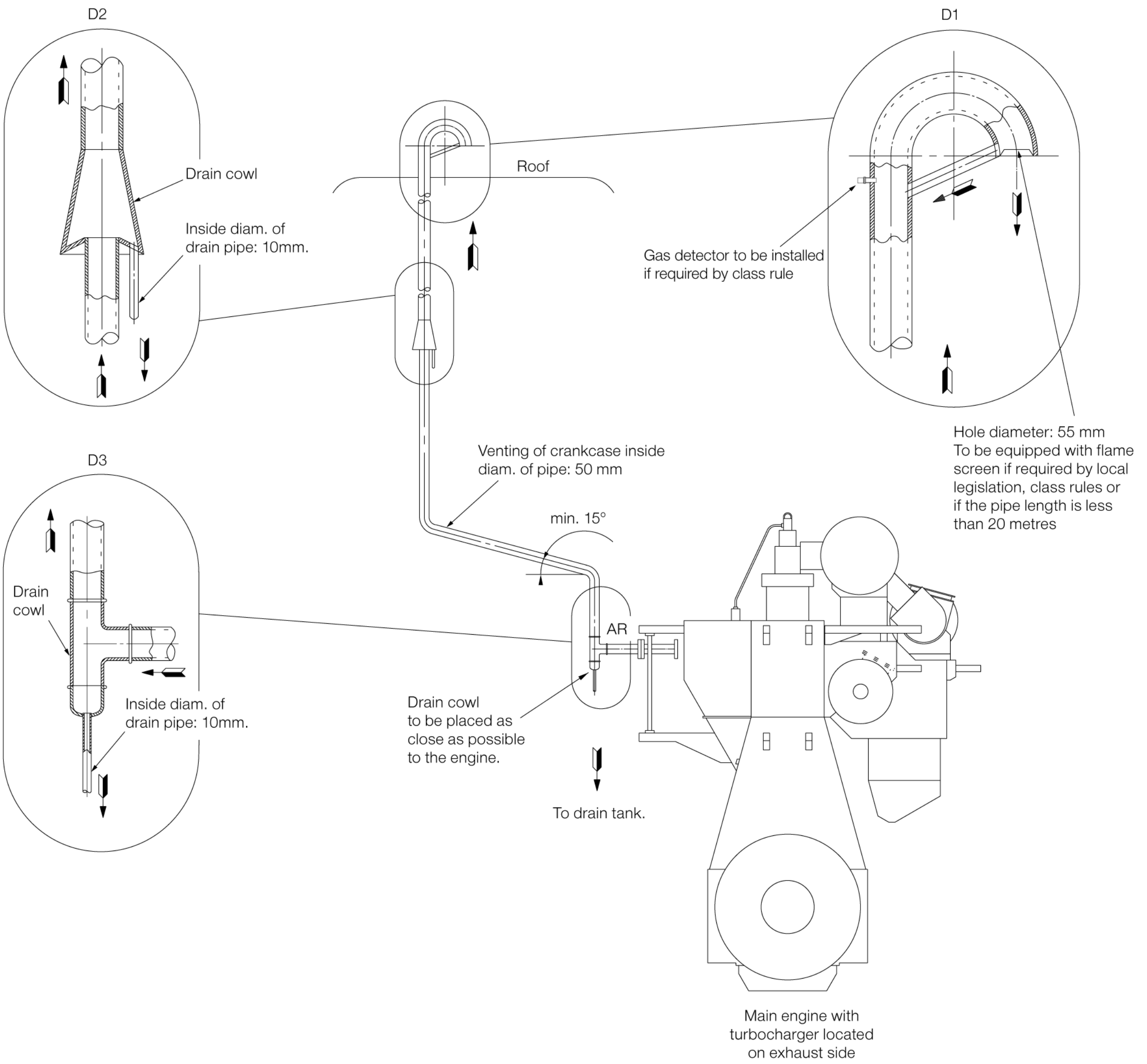
Venting and drain pipes



The venting pipe has to be equipped with a drain cowl as shown in detail D2 and D3.
Note that only one of the above solutions should be chosen.

Fig. 8.07.01: Crankcase venting

Bedplate drain pipes

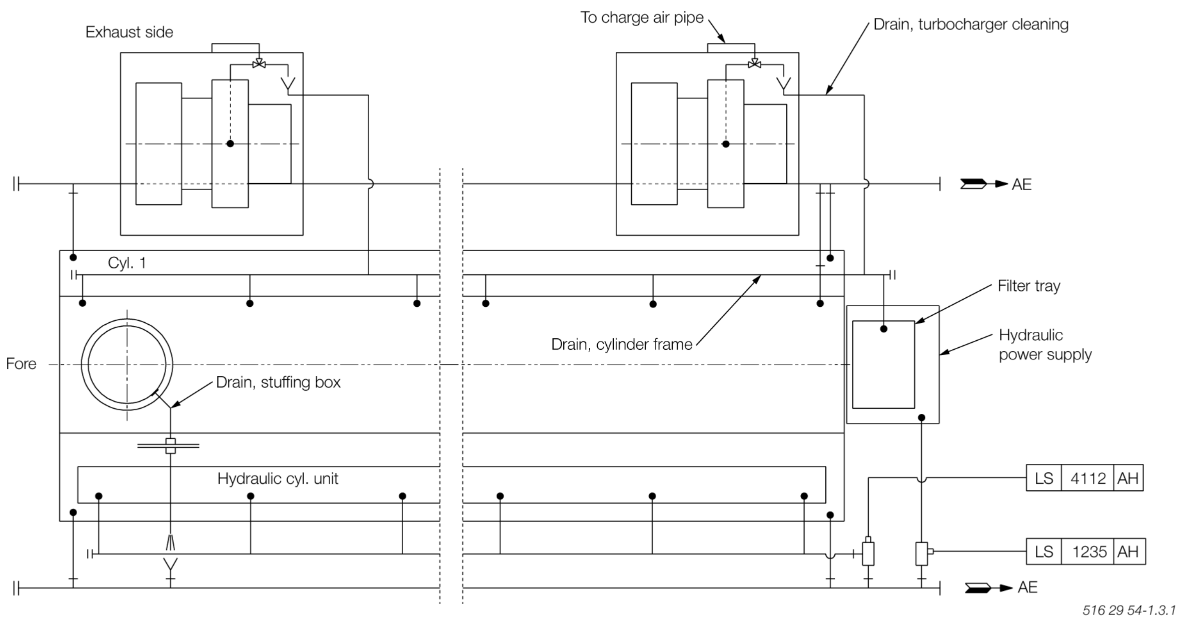


Fig. 8.07.02: Bedplate drain pipes, aft-mounted HPS

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Engine and tank venting to the outside air

Venting of engine plant equipment separately

The various tanks, engine crankcases and turbochargers should be provided with sufficient venting to the outside air.

MAN Energy Solutions recommends to vent the individual components directly to outside air above deck by separate venting pipes as shown in Fig. 8.07.03a.

It is not recommended to join the individual venting pipes in a common venting chamber as shown in Fig. 8.07.03b.

In order to avoid condensed oil (water) from blocking the venting, all vent pipes must be vertical or laid with an inclination.

Additional information on venting of tanks is available from MAN Energy Solutions, Copenhagen.

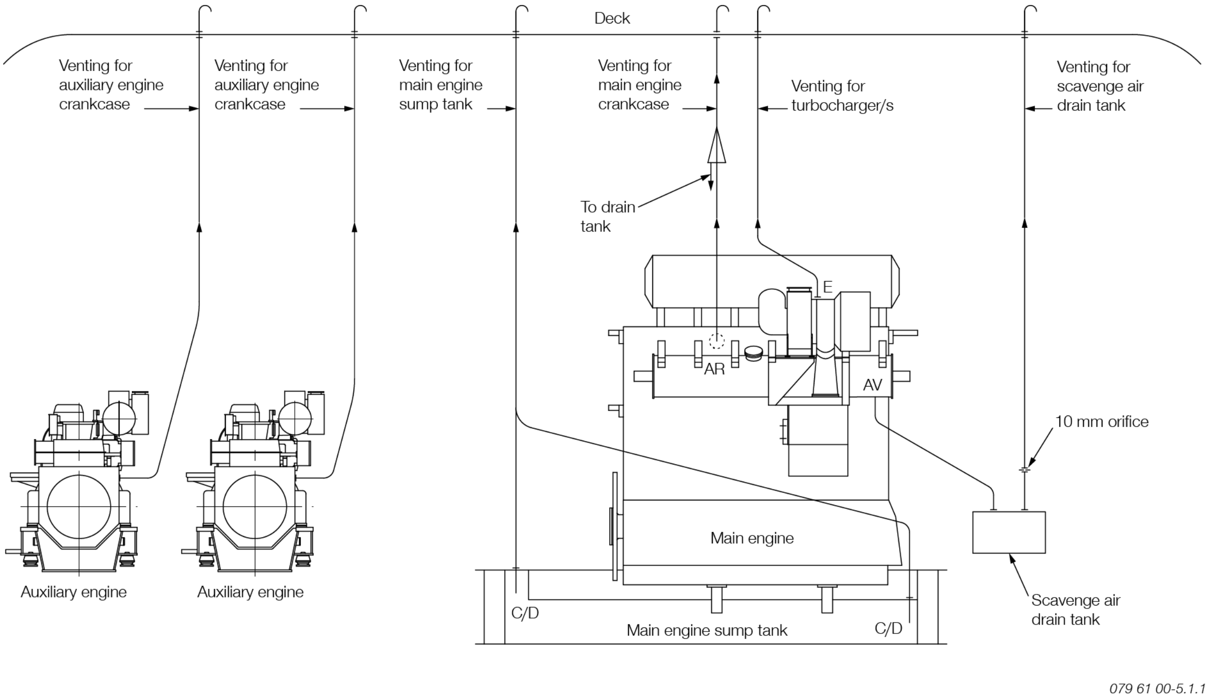

Fig. 8.07.03a: Separate venting of all systems directly to outside air above deck

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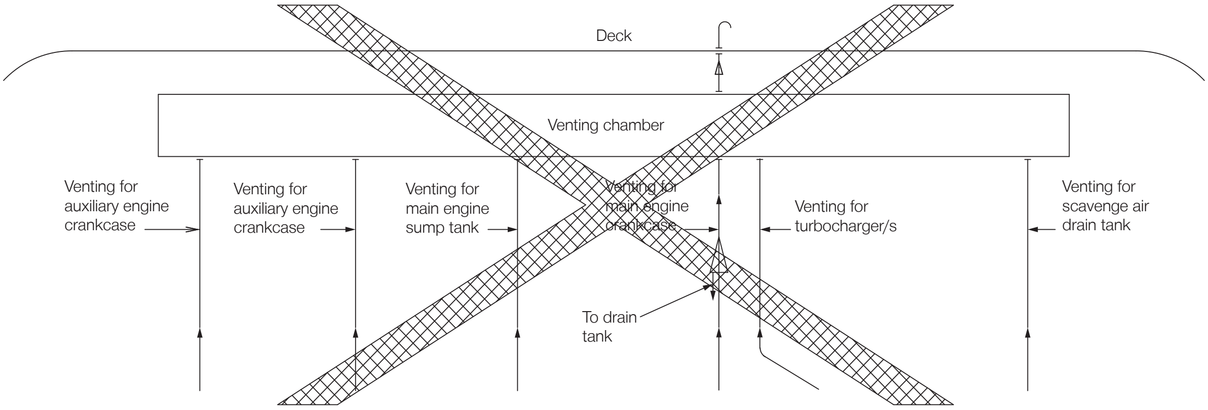

Fig. 8.07.03b: Venting through a common venting chamber is ***not recommended***

Hydraulic oil back-flushing

This section is not applicable

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Hydraulic control oil system

As an option, the engine can be prepared for a separate hydraulic control oil system, see Fig. 8.09.01.

The hydraulic control oil system can be a separate unit, or it can be integrated in the engine room with the various components placed and fastened to the steel structure of the engine room.

The design and dimensioning of the various components gives a reliable system capable of supplying 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.

Cleanliness of the hydraulic control oil

The hydraulic control oil must fulfil the same cleanliness level as our standard integrated lube/cooling/hydraulic-control oil system, i.e. ISO 4406 XX/16/13 equivalent to NAS 1638 Class 7.

MAN Energy Solutions can supply information and recommendations for:

- Flushing
- The specified cleanliness level and how it is measured
- Applying the NAS 1638 oil cleanliness code as an alternative to ISO 4406.

Control oil system components

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 integrated in the ships' structure.

The tank must be equipped with flange connections and the items listed below:
1 oil filling pipe
1 outlet pipe for pump suctions
1 return pipe from engine
1 drain pipe
1 vent pipe.
The hydraulic control oil tank must be placed at least 1 m below the hydraulic oil outlet flange, RZ.

Hydraulic control oil pump

The pump must be a displacement type (for example, a gearwheel or a screw wheel pump).

Use the browser-based CEAS application to specify the following data, see 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.

Use the browser-based CEAS application to specify the following data, see Fig. 8.09.02.
• Flow rate
• Adjustable differential pressure range across the valve
• Oil viscosity range.

Hydraulic control oil cooler

The cooler must be a plate heat exchanger or a shell and tube type cooler.

Use the browser-based CEAS application to specify the following data, see 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.

Use the browser-based CEAS application to specify the following data, see Fig. 8.09.02.
• Capacity
• Adjustable temperature range
• Maximum pressure drop across the valve.

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Hydraulic control oil filter

The filter must be a duplex full-flow type with manual change-over and manual cleaning, or an automatic self-cleaning type.

A differential pressure gauge is fitted onto the filter.

Use the browser-based CEAS application to specify the following data, see 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 / purifier

The off-line fine filter unit, or purifier must be able to treat 15-20% of the total oil volume per hour, see Fig. 8.09.01.

The fine filter is an off-line filter which removes metallic and non-metallic particles larger than 0.8 μm, as well as water and oxidation residues. The filter has a pertaining pump and it must be fitted on 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 must be a liquid straight type indicator.

Pressure indicator

The pressure indicator must be a dial type indicator.

Level Alarm

The hydraulic control oil tank must have level alarms for high and low oil levels.

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Piping

The pipes can be made of mild steel.

The design oil pressure is 10 bar.

The return pipes must be placed vertically, or with a downwards inclination of minimum 15°.

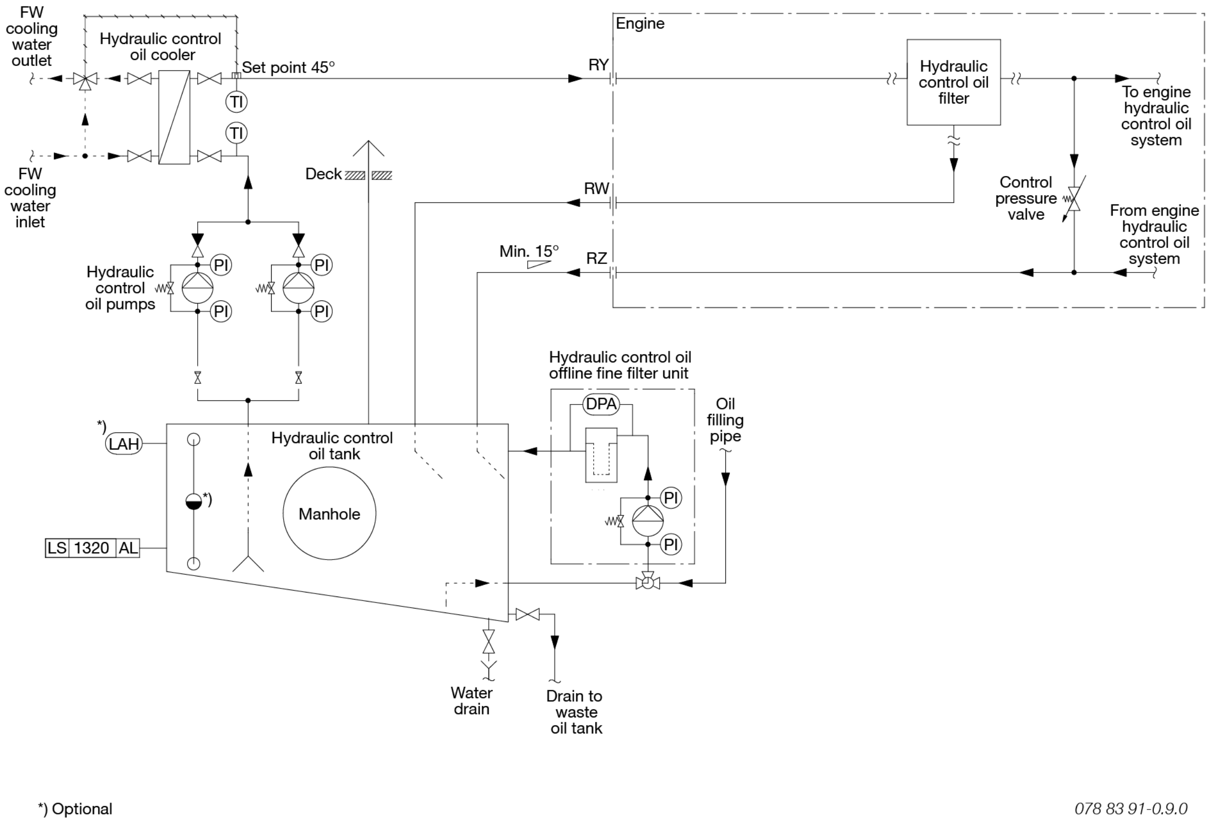

Fig. 8.09.01: Hydraulic control oil system, manual filter

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Hydraulic control oil system capacities

Calculate the capacities of the hydraulic control oil system in the CEAS application

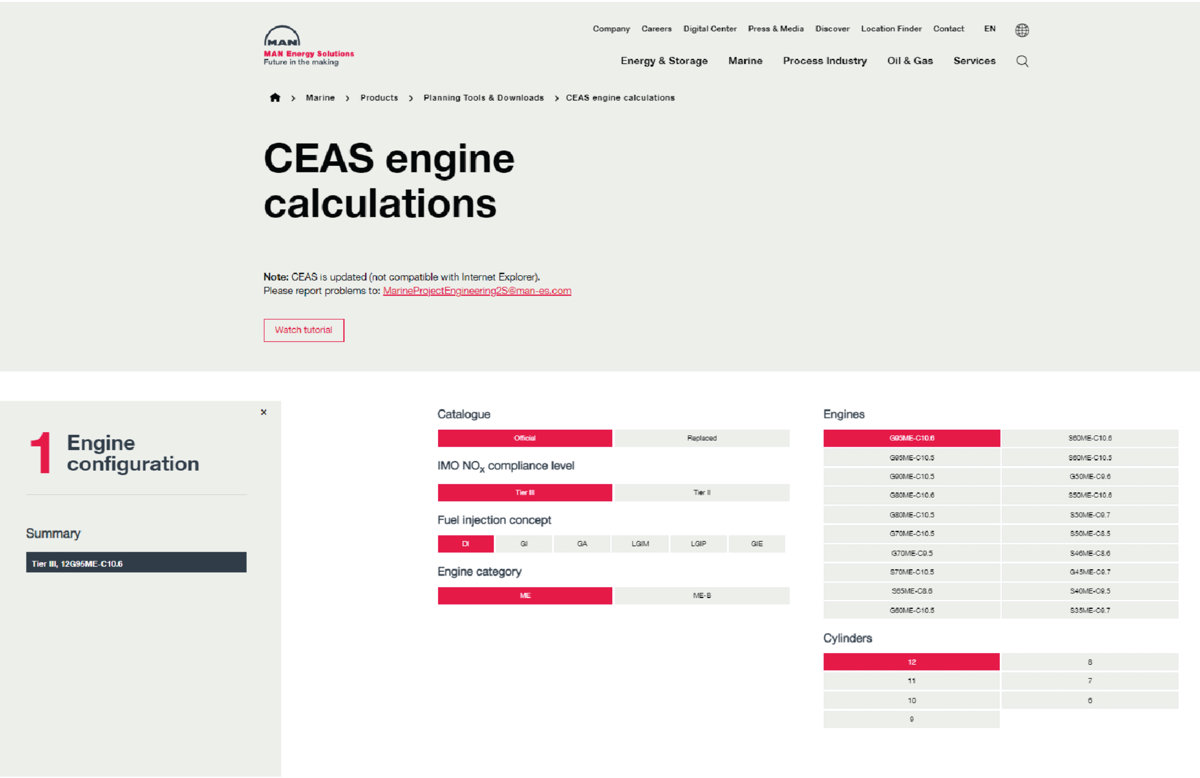

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Fig. 8.09.02: The browser-based CEAS calculation system

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Cylinder oil specification and system description

Cylinder lubricators

Each cylinder liner has a number of lubricating quills through which oil is introduced by MAN B&W Alpha Cylinder Lubricators, see Section 9.02.

The oil is pumped into the cylinder (via non-return valves) when the piston rings pass the lubricating orifices during the upward stroke.

The control of the lubricators is integrated in the engine control system (ME-ECS). Fig. 9.02.02b shows an overview of the cylinder lubricating oil control system.

Cylinder lubrication strategy

The general lubrication strategy is to match the cylinder oil with the fuel type and its sulphur content.

When operating on liquid natural gas (LNG), ethane liquid petroleum gas (LPG) or methanol, the same cylinder oil is used as for <0.10 ULSFO (ultra-low-sulphur fuel oil) to <0.5 VLSFO (very-low-sulphur fuel oil) operation.

For specific lubrication guidelines, please refer to the most recent lubrication guideline for your specific engine type, for example service letters and circular letters. Service letters are publicly available.

Circular letters are only distributed to customers with specific engine types.

Cylinder oil

MAN Energy Solutions is continously improving the engine design. Highly fuel-efficient engines with higher pressures and higher temperatures require lubricants with matching performance. Cylinder oils must be designed to:

- lubricate pistons and liners
- reduce friction
- introduce wear protection
- minimise risk of seizures
- neutralise acids and oxidation products in accordance with the engine requirement,
- and keep pistons, piston rings, ring grooves, ringlands, and liners clean.

The cylinder oils are divided into two performance categories, Cat. I and Cat. II, of which Cat. II is the overall higher performing category.

MAN Energy Solutions recommends using cylinder oils with the following main properties:

kinematic viscoity

- kinematic viscoity
  - minimum 18.5 cSt at 100°C
  - maximum 21.9 cSt at 100°C
- viscosity index (VI): min. 95
- high detergency
- alkalinity or base number (BN).

Category II cylinder oils – all MAN B&W engines and recommended for Mk. 9 and higher

Cat. II cylinder oils have excellent overall performance with a special focus on cleaning ability. In order to receive this status, the cylinder oil must complete extensive testing. The lubricants awarded Cat. II status are 40 BN, 100 BN and 140 BN cylinder oils. Cylinder oils with 100 and 140 BN are mainly used for high-sulphur fuel applications. The Cat. II 40 BN oils can be used for operation on <0.10- 0.50%S fuels and LNG, ethane, methanol, and LPG.

Table 9.01.01 lists major international system oil brands tested in service with acceptable results, and which have passed the testing procedure and obtained an No Objection Letter (NOL). Do not consider these lists to be complete, as other Cat. II cylinder oils with NOLs from MAN Energy Solutions can be equally suitable.

Category II cylinder oils as stated in SL2022-728*			
Company	Cylinder oils		
	140 BN	100 BN	40 BN
Castrol	Cyltech 140	Cyltech 100	Cyltech 40 XDC
Chevron	Taro Ultra 140	Taro Ultra 100	Taro Ultra Advanced 40
ExxonMobil	Mobilgard 5145	Mobilgard 5100	Mobilgard 540 AC
Gulf Oil Marine	Gulfsea Cylcare 50140X	Gulfsea Cylcare 50100X	Gulfsea Cylcare XP 5040X
ENEOS Corporation		ENEOS Marine C1005S	
Shell	Shell Alexia 140	Shell Alexia 100	Shell Alexia 40XC
Sinopec Lubricant co.		Sinopec Marine Cylinder Oil 50100	
TotalEnergies Lubmarine	Talusia HR 140	Talusia Universal 100	Talusia HD 40

*Cat. II cylinder oils suitable for all MAN B&W two-stroke engines and recommended for Mark 9 and higher. Examples of international cylinder oils for which an NOL has been granted Cat. II status by MAN Energy Solutions.

Table 9.01.01: Cat. II cylinder oils.

Category I cylinder oils – MAN B&W engines Mk. 8 and lower

Table 9.01.02 lists major international cylinder oil brands tested in service with acceptable results, and which have passed the testing procedure and obtained an NOL. Do not consider these lists to be complete, as other Cat. I cylinder oils with NOLs from MAN Energy Solutions can be equally suitable.

Category I cylinder oils as stated in SL2022-728**			
Company	Cylinder oils		
	100BN - 140BN	70 BN	40 BN
Castrol		Cyltech 70	Cyltech 40SX* Cyltech 40
Chevron Lubricants		Taro Ultra 70	Taro Ultra 40
ExxonMobil		Mobilgard 570	Mobilgard 540 Mobilgard 540 X*
Gulf Oil Marine		Gulfsea Cylcare DCA 5070X	Gulfsea Cylcare DCA 5040X
ENEOS Corp.		Marine C705	Marine 405Z
Shell		Shell Alexia 70	Shell Alexia 40
Sinopec Lubricant co.	Sinopec Marine Cylinder Oil 50140	Sinopec Marine Cylinder OIl 5070S	Sinopec Marine Cylinder Oil 5040
SK lubricants	SK Supermar Cyl 140 / SK Supermar Cyl 100	SK Supermar Cyl 70 Plus	SK Supermar Cyl 40 Plus
TotalEnergies Lubmarine		Talusia HR 70	Talusia LS 40

* Also tested according the new Cat. I requirements.

** Cat. I cylinder oils suitable for MAN B&W two-stroke engines Mk. 8 and lower. Examples of international cylinder oils with an NOL from MAN Energy Solutions

Table 9.01.02: Cat. I cylinder oils.

Notwithstanding the foregoing, it remains the responsibility of the owner/operator of an engine to ensure that suitable fuels and lubes are conditioned and used in order to prevent damage to the engine and other equipment on board. MAN Energy Solutions disclaims any and all liability and cannot be held responsible for any damage to the engine, engine components or other equipment on board that may be caused by the use of the mentioned lubricants.

Description of cylinder oil systems

The general cylinder lubrication strategy for MAN B&W engines is to match the cylinder oil with the fuel type, its sulphur content, and the dependency on detergent. This match can be achieved by the two options available of the cylinder oil lubricating system, both of which require access to two cylinder oils.

The two options are – options 1 and 2:

- Option 1 - Automated cylinder oil switching system (ACOS2) is considered the standard system for most engines, whereas
- Option 2 - Automated cylinder oil mixing system (ACOM) is a more advanced system designed for engine operation which places demands on the change of cylinder oil.

Engine type	ME, ME-B, ME-C, ME-GI, ME-LGI, ME-GA			
Fuel type	low-sulphur (LS)	high-sulphur (HS)	HS Tier II LS Tier III	HS and specified dual-fuel (SDF) operation
Option 1-ACOS2	standard	standard	standard	N/A
Option 2 - ACOM	option	option	option	standard

Table 9.01.01: Lubrication systems for all engines

Option 1 - Automated cylinder oil switching system (ACOS2)

ACOS2 is a simple system securing fully acceptable conditions when a longer changeover period is available. Fig. 9.01.01 shows Option 1 – the ACOS2 system.

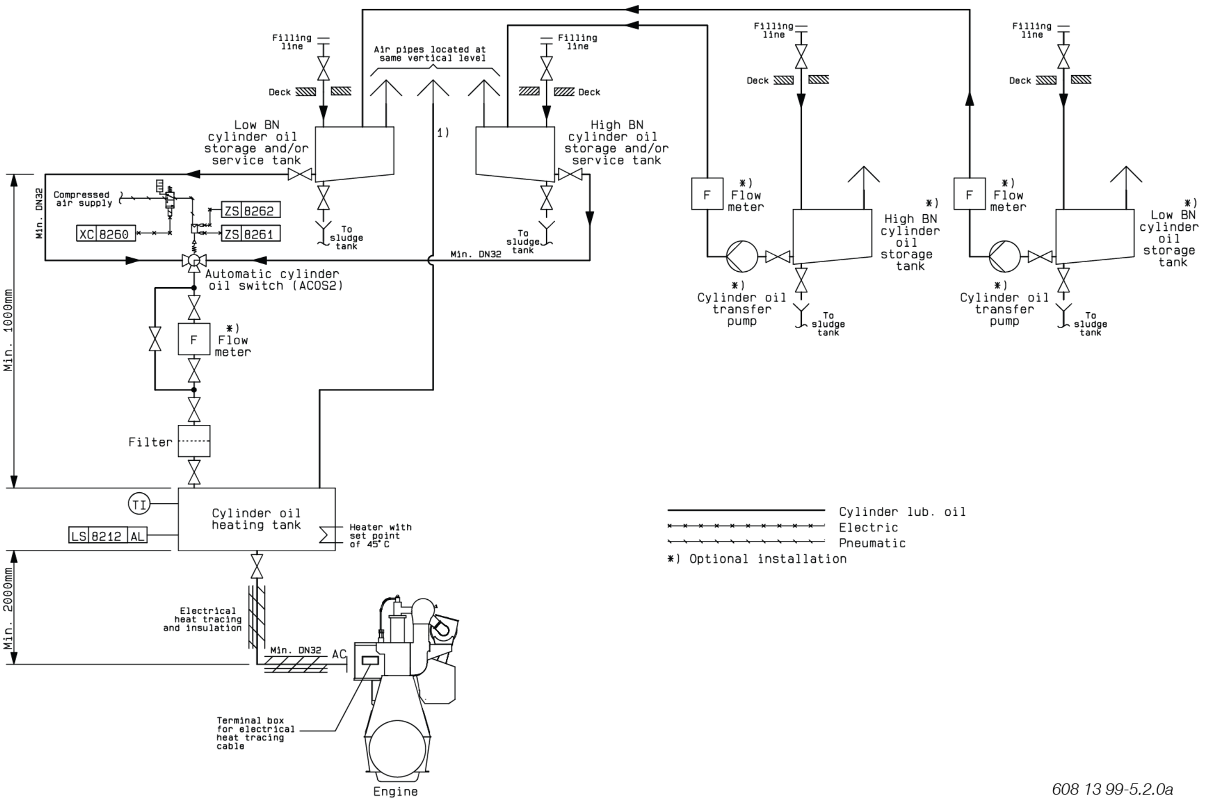

Fig. 9.01.01: Option 1 - ACOS2 system - standard system for most engines

Option 2 - Automated cylinder oil mixing system (ACOM)

ACOM is specified for engine operation and the need for a faster responding system when switching fuels. Fig. 9.01.02 shows Option 2 – the ACOM system.

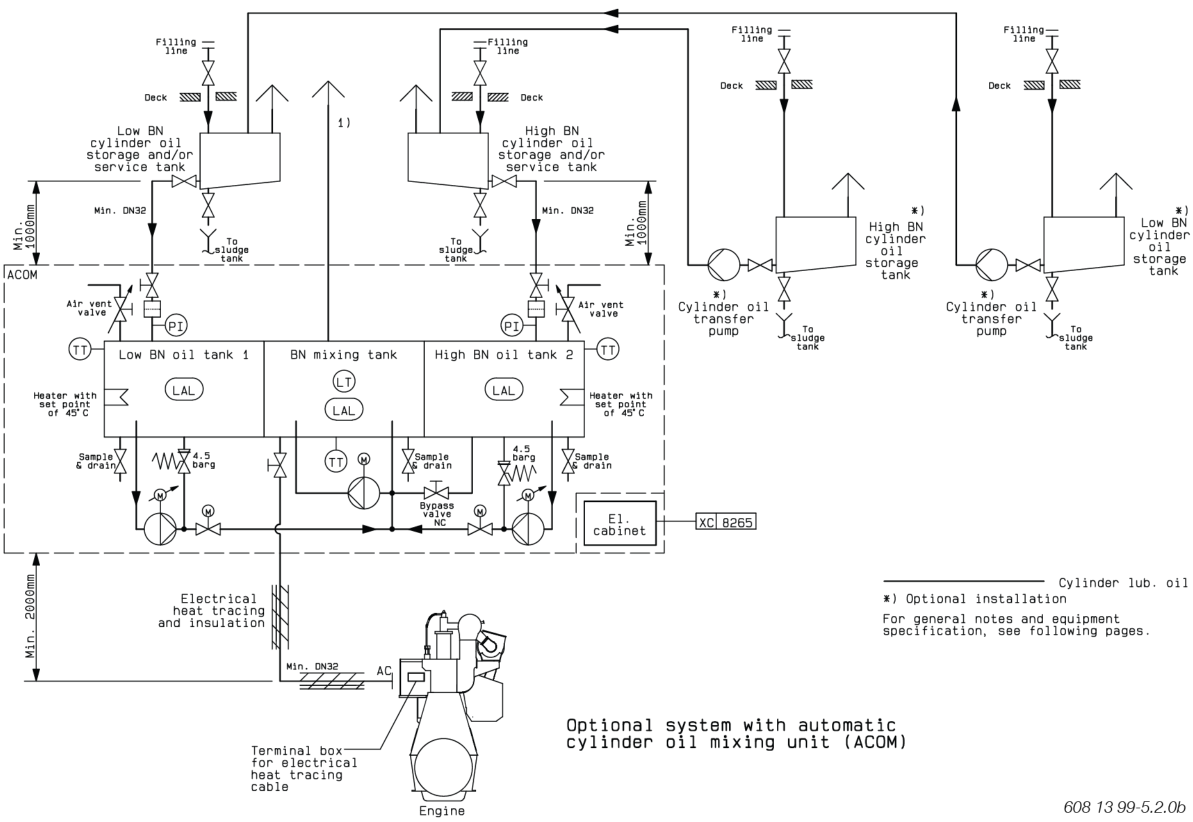

Fig. 9.01.02: Option 2 - ACOM system

Cylinder oil feed rate (dosage)

The minimum cylinder oil feed rate can be found in the latest official guideline issued in our most current Service Letter on this subject. Download Service Letters from our website here. The minimum feed rate is the amount of oil needed to lubricate all parts sufficiently.

Continuously monitoring the cylinder condition and analysing drain oil samples are important to secure:

1. optimised cylinder oil feed rate and consumption, and
2. safeguard the engine against wear.

Further information about cylinder lubrication is available in MAN Energy Solutions' most current Service Letters on this subject.

Option 1 - Automated cylinder oil switching system (ACOS2)

The ACOS2 system automatically switches between two grades of cylinder oil, for example, cylinder lubricating oils with a high and a low base number (BN). In general, the cylinder oil storage and/or service tanks should be arranged separately for the two cylinder oils.

In dual-fuel operation, the sulphur content of the combusted fuel depends on the:

- engine load
- amount of pilot oil in the combusted fuel
- sulphur content in the pilot fuel.

The sulphur content in the combusted fuel is called the sulphur equivalent. The ACOS2 system automatically calculates the sulphur equivalent based on input about the sulphur content of the pilot fuel, which the crew enters on the main operating panel (MOP).

The engine control system (ME-ECS) controls the ACOS2 system and the mixing of cylinder oils via a three-way valve, see Fig. 9.01.01. The three-way valve switches between, for example, a high- and a low-BN oil, when the value of the sulphur equivalent changes from high to low.

The three-way ball valve is activated pneumatically by control air and in fail-safe position, a spring forces the valve to switch to the high-BN oil.

Option 2 - Automated cylinder oil mixing system (ACOM)

The automated cylinder oil mixing (ACOM) is a cylinder oil delivery system which automatically mixes to two fully formulated cylinder oils to the optimum BN, depending on the sulphur content of the fuel.

In general, the cylinder oil storage and/or service tanks should be arranged separately for the two cylinder oils.

MAN Energy Solutions' ACOM system mixes commercially available cylinder oils to the BN value required. The resulting BN of the cylinder oil supplied to the liners is in the range of the BN values of the two cylinder oils stored on board.

The basic principle is to mix a cylinder lubricating oil with an optimal BN, as illustrated in Fig. 9.01.03. At a certain sulphur content level, the engine has to run on a high-BN cylinder oil.

Time based switch? – waiting for JAP

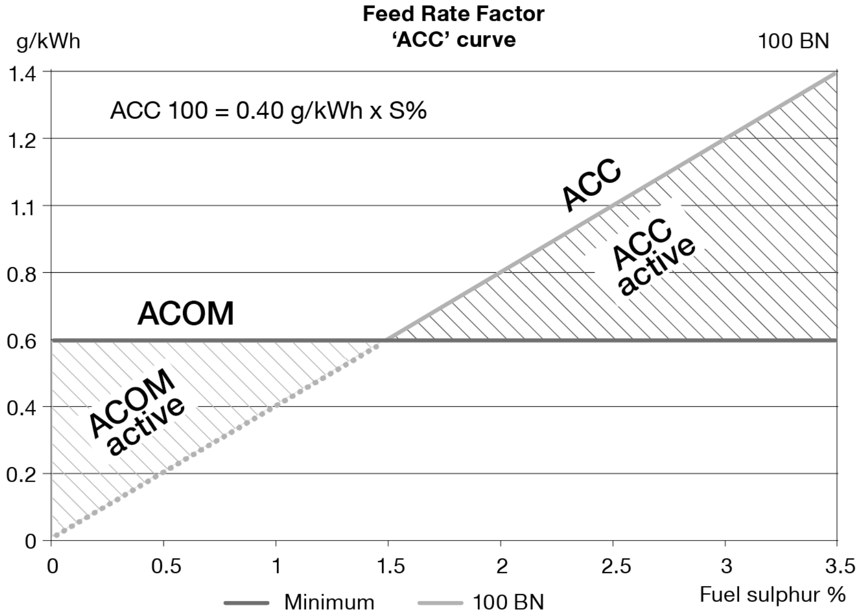

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Fig. 9.01.03: Mixing principles, ACOM (wating for JAP)

ACOM working principle

The ME-ECS controls the ACOM system and the mixing of cylinder oils based on an input about the sulphur content of the fuel combusted. In dual-fuel operation, the sulphur content of the fuel combusted depends on:

- engine load
- amount of pilot oil in the combusted fuel
- sulphur content in the pilot fuel.

The sulphur content of fuel is called the sulphur equivalent. The ACOM system automatically calculates the sulphur equivalent.

The engine control systems of ME-C/-GI/-LGI/-GA and ME-B-GI/-LGI engines contain the ACOM functionality, and the crew must input the sulphur content of the pilot fuel on the MOP.

On the ME-B engine, ACOM is a stand-alone installation controlled from an ACOM operating panel separate to the engine control system (ME-B ECS). The ship's alarm system handles alarms.

The cylinder oil mixing volumes are kept small to enable a fast changeover from one cylinder oil BN to another.

Two-tank cylinder lubrication system with ACOM

The ACOM design enables measurement of the daily consumption of cylinder lubricating oil, which eliminates the need for two cylinder oil service/day tanks.

Furthermore, compared to Option 1 – the ACOS2 system in Fig. 9.01.01, the ACOM system also eliminates the need for a heating tank.

The cylinder lubricating oil is fed from the storage tanks to the ACOM system by gravity. The ACOM system is located in the engine room near to and above the cylinder lubricating oil inlet flange, AC, in a vertical distance of minimum 2 m. Fig. 9.01.04 shows the layout of the ACOM system.

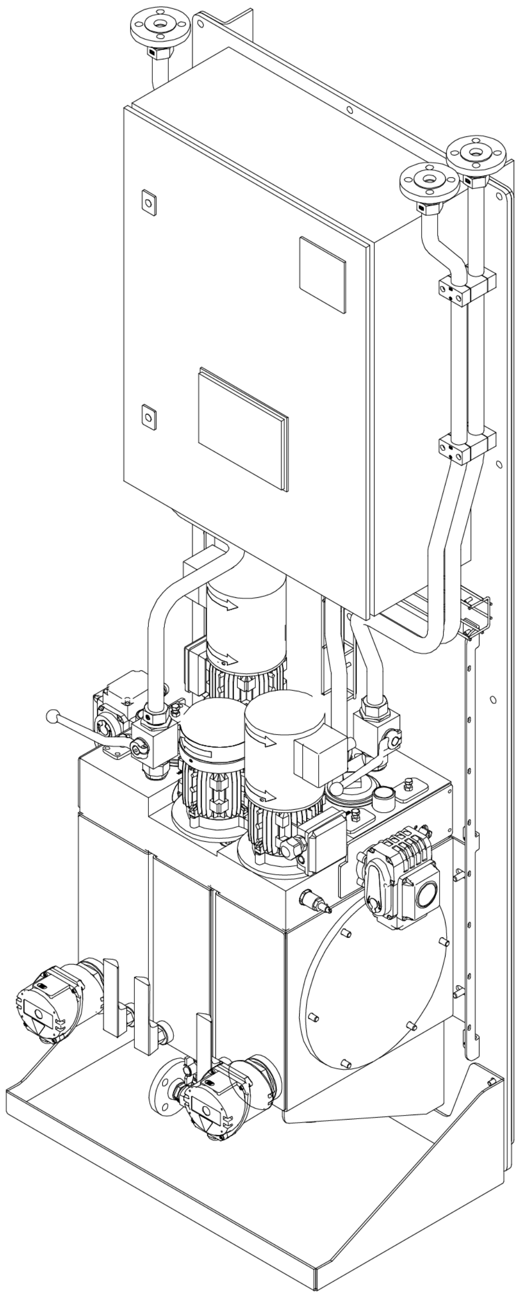

Fig. 9.01.04: Automated cylinder oil mixing system (ACOM) in a single-rack version for installation in the engine room

Components for the cylinder oil lubricating systems

The next section describes the recommendations and requirements to the components of the cylinder lubricating oil systems.

1. Cylinder oil storage tanks and/or service tanks

a) Combined storage and service tank design

It is recommended that the tank design is tall and slim to optimise settling in the tank. The tank bottom must be sloped for easy water drainage.

b) Separate service tank design

It is recommended that the tank design is tall and slim to optimise settling in the tank and the accuracy of readings for calculating the consumption. The recommended length, width, and height aspect ratio is 1:1:3. The tank bottom must be sloped for easy water drainage.

Capacity

The tank capacities must be determined based on hourly cylinder oil consumption and the needed operating storage time. The storage time will differ from project to project according to the time between bunkering of cylinder oil estimated by the vessel operator. If separate cylinder oil storage tanks are installed, MAN Energy Solutions normally recommends that the capacity for the service tanks is defined as one day.

Hourly consumption [m^3/h] = Feed rate [g/kWh] x $Power_{SMCR}$ [kW] x f_{safe}/(1000 [g/kg] x Density [kg/m^3])

- Feed rate [g/kWh] = ACC x S [%], the feed rate used must not be less than 0.6 g/kWh.
- ACC [g/kWh] = 0.4, as the design (normally between 0.2–0.4)
- S [%] is the sulphur content of the fuel oil
- $Power_{SMCR}$ [kW] is engine power at SMCR
- f_{safe} is a safety factor, f_{safe} = 1.2
- Density [kg/m^3] = 900

Location

The supply line must be located minimum 1000 mm above the top of the cylinder oil heating tanks and the ACOM system.

Minimum cylinder oil temperature in the tank

The design minimum cylinder oil temperature in the tank is 25°C.

If the holding temperature is lower, the size of the heating element in the heating tank and the pipe dimension from the service tank to the heating tank must be increased.

Outlet pipe dimension

Min. DN32 (min. DN40 if 0°C is the minimum cylinder oil temperature design criteria)

2. Filter

Fineness (absolute): 250 µm

3. Cylinder oil heating tanks and ACOM tanks

Capacity: Approx. 40 litres

Location

The bottom of the heating tank must be minimum 2000 mm above the engine connection AC.

Heating element size

Can be calculated as: Heat [W] = $Power_{SMCR}$ [kW] x f_{heat} [W/kW]

- $Power_{SMCR}$ [kW] is engine power at SMCR
- f_{heat25} [W/kW] = 0.0078 for low-sulphur (0.5% S) engines (Based on service tank holding temperature, T_h = 25°C)
- f_{heat25} [W/kW] = 0.0183 for high-sulphur (3.5% S) engines (Based on service tank holding temperature, T_h = 25°C)

If a lower holding temperature is considered, f_{heat} must be:

$f_{heat} = (45-T_{h\ act}) \times f_{heat25}/20$

Where $T_{h\ act}$ [°C] is the actual service tank holding temperature

- Inlet pipe dimension: Min. DN32 (min. DN40, if 0°C service tank holding temperature is the design criteria)
- Outlet pipe dimension: Min. DN32
- Vent pipe dimension: Min. DN32
- Level alarm low switch: Guidance values for automation (GVA) tag no. LS 8212 AL

4. Electrical heat tracing and insulation

Location

The entire pipeline from heating tank to engine must be fitted with insulation and electrical heat tracing.

Electrical tracing cable connection

The electrical tracing cable (shipyard supply) must be connected to the terminal box installed internally at the engine. Fig. 9.01.05 shows the arrangement.

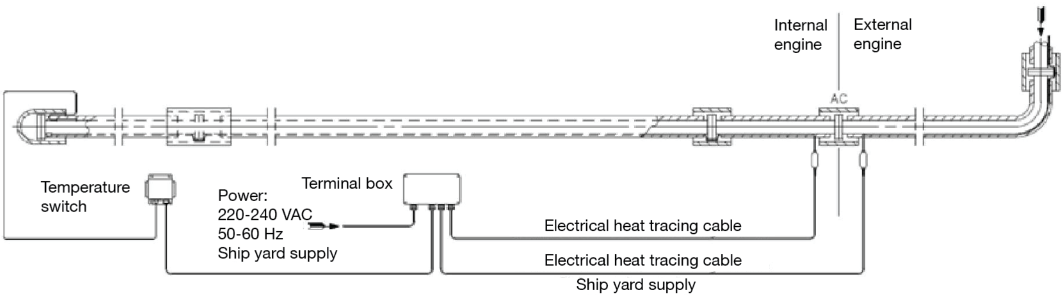

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Fig 9.01.05: Electrical tracing cable arrangement

5. Automatic cylinder oil switch (ACOS2)

- Type: 3-way ball valve (L-bored)
- Actuator: Pneumatic single acting type with spring return to position "High BN cylinder oil"
- Solenoid valve: 3/2 type with silencer
- Controlled by: Engine Control System – control signal XC8260 (24 V DC), closed contact = "Low BN cylinder oil"
- Valve position feed-back to: engine control system - feed-back signal ZS8261 (potential free contact) - "Low BN cylinder oil"

- Engine control system - feed-back signal ZS8262 (potential free contact) - “High BN cylinder oil”
- Valve location: Must be located as close as possible to the heating tank.

6. Cylinder oil storage tanks (Optional)

Purpose

If for some reason, for example owner requirement, or if it is not possible to locate the needed storage capacity in the service tanks at the required vertical elevation level, independent storage tanks can be used.

Tank design

It is recommended that the tank design is tall and slim to optimise settling in the tank. The tank bottom must be sloped for easy water drainage.

Capacity

The capacities of the tanks must be determined based on hourly cylinder oil consumption and the needed operating storage time. The needed storage time will differ from project to project according to the time between bunkering of cylinder oil estimated by the vessel operator.

For calculating hourly consumption [m^3/h], reference is made to section “Cylinder oil storage tanks and/or service tanks” – “Capacity”.

7. Cylinder oil transfer pumps (Optional)

Purpose

To transfer the cylinder oil between storage tanks and service tanks.

Option

If one transfer pump is used for pumping both cylinder oils, it has to be secured that the pump suction/discharge valves are interlocked with each other to avoid unintended blending of the two cylinder oils with different BN values.

8. Flowmeter (Optional)

Location

Installation options:

- Located between service tank and cylinder oil heating tank *)
- Located between cylinder oil heating tank and engine. Only a Coriolis type can be used *)
- Located between storage tanks and service tanks.

*) These flowmeter locations can result in unreliable measurements because of the very low flow.

Type

A mass flowmeter of the Coriolis type can be recommended based on: low differential pressure, no built-in non-return function, and it provides mass flow results (not a volumetric result).

Note 1

Vent pipes for twin engine plants:

- ACOS2 system: A separate vent pipe must be used from each engine heating tank
- ACOM system: A separate vent pipe must be used from each engine ACOM BN mixing tank.

Alpha ACC cylinder lubrication system

The MAN B&W Alpha cylinder lubrication system, see Figs. 9.02.02a, 02b and 02c, is designed to supply cylinder oil intermittently, for instance every 2, 4 or 8 engine revolutions with electronically controlled timing and dosage at a defined position.

Traditional two-tank cylinder lubrication system

Separate storage and service tanks are installed for each of the different Base Number (BN) cylinder oils used onboard ships operating on both high- and low-sulphur fuels, see Fig. 9.02.02a.

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 about one week's cylinder lubricating oil consumption.

Oil feed to the Alpha cylinder lubrication system

Cylinder lubricating oil is fed to the Alpha cylinder lubrication system by gravity from the service tank or ACOM.

The oil fed to the injectors is pressurised by the Alpha Lubricator which is placed on the hydraulic cylinder unit (HCU) and equipped with small multi-piston pumps.

The oil pipes fitted on the engine are shown in Fig. 9.02.04.

The whole system is controlled by the Cylinder Control Unit (CCU) which controls the injection frequency based on the engine-speed signal given by the tacho 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 (Adaptable 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.

Regarding the filter and the small tank for heater, please see Fig. 9.02.05.

Alpha Lubricator variants

Since the Alpha Lubricator on ME and ME-B engines are controlled by the engine control system, it is also referred to as the ME lubricator on those engines.

A more advanced version with improved injection flexibility, the Alpha Lubricator Mk 2, is being introduced on the G95/50/45/40ME-C9 and S50MEC9 including their GI dual fuel variants.

Further information about the Alpha Lubricator Mk 2 is available in our publication:

Service Experience MAN B&W Two-stroke Engines

The publication is available at www.man-es.com → 'Marine' → 'Products' → 'Planning Tools and Downloads' → 'Technical Papers'.

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.

Alpha ACC (Adaptive Cylinder-oil Control) is the lubrication mode for MAN B&W two-stroke engines, i.e. lube oil dosing proportional to the engine load and proportional to the sulphur content in the fuel oil being burnt.

Working Principle

The feed rate control should be adjusted in relation to the actual fuel quality and amount being burnt at any given time.

The following 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 actual feed rate is dependent of the operating pattern and determined based on engine wear, cylinder condition and BN of the cylinder oil.

The implementation of the above criteria will lead to an optimal cylinder oil dosage.

Specific Minimum Dosage with Alpha ACC

The recommendations are valid for all plants, whether controllable pitch or fixed pitch propellers are used. The specific minimum dosage at lowersulphur fuels is set at 0.6 g/kWh.

After a running-in period of 500 hours, the feed rate sulphur proportional factor is 0.20 - 0.40 g/kWh × S%. The actual ACC factor will be based on cylinder condition, and preferably a cylinder oil feed rate sweep test should be applied. The ACC factor is also referred to as the Feed Rate Factor (FRF).

Examples of average cylinder oil consumption based on calculations of the average worldwide sulphur content used on MAN B&W two-stroke engines are shown in Fig. 9.02.01a and b.

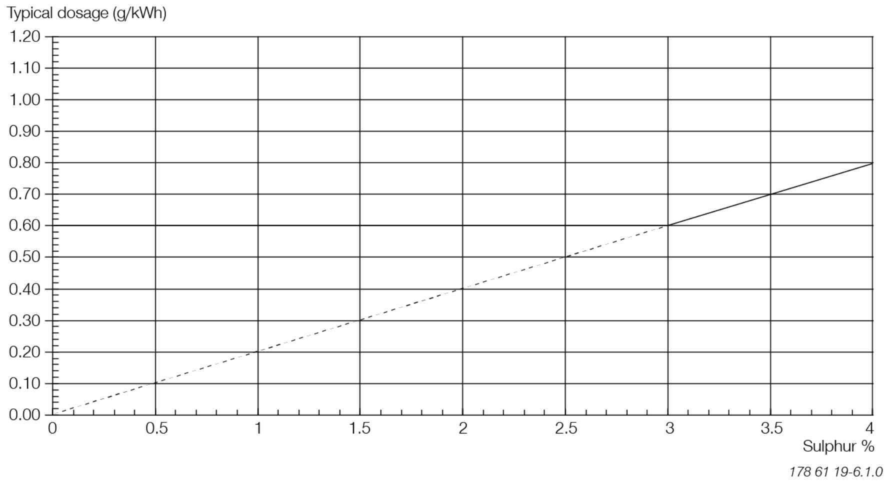

Fig. 9.02.01a: FRF = 0.20 g/kWh × S% and BN 100 cylinder oil – average consumption less than 0.65 g/kWh

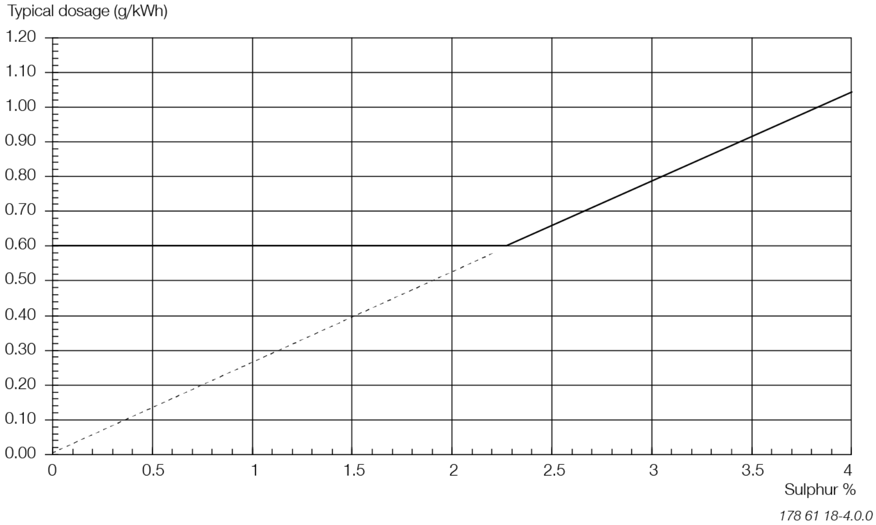

Fig. 9.02.01b: FRF = 0.26 g/kWh × S% and BN 100 cylinder oil – average consumption less than 0.7 g/kWh

Further information about cylinder oil dosage is available in MAN Energy Solutions' most current Service Letters on this subject available at www.marine.man-es.com --> 'Two-Stroke' --> 'Service Letters'.

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 two small tanks for heater element in the vessel, Fig. 9.02.02a, or from the ACOM, Fig. 9.02.02b, and the main cylinder oil pipe on the engine is insulated and electricallly heated.

The engine builder is to make the insulation and heating of the main cylinder oil pipe on the engine. Moreover, the engine builder is to mount the terminal box and the thermostat on the engine, see Fig. 9.02.03.

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The ship yard is to make the insulation of the cylinder oil pipe in the engine room. The heating cable is to be mounted from the small tank for heater element or the ACOM to the terminal box on the engine, see Figs. 9.02.02a and 02b.

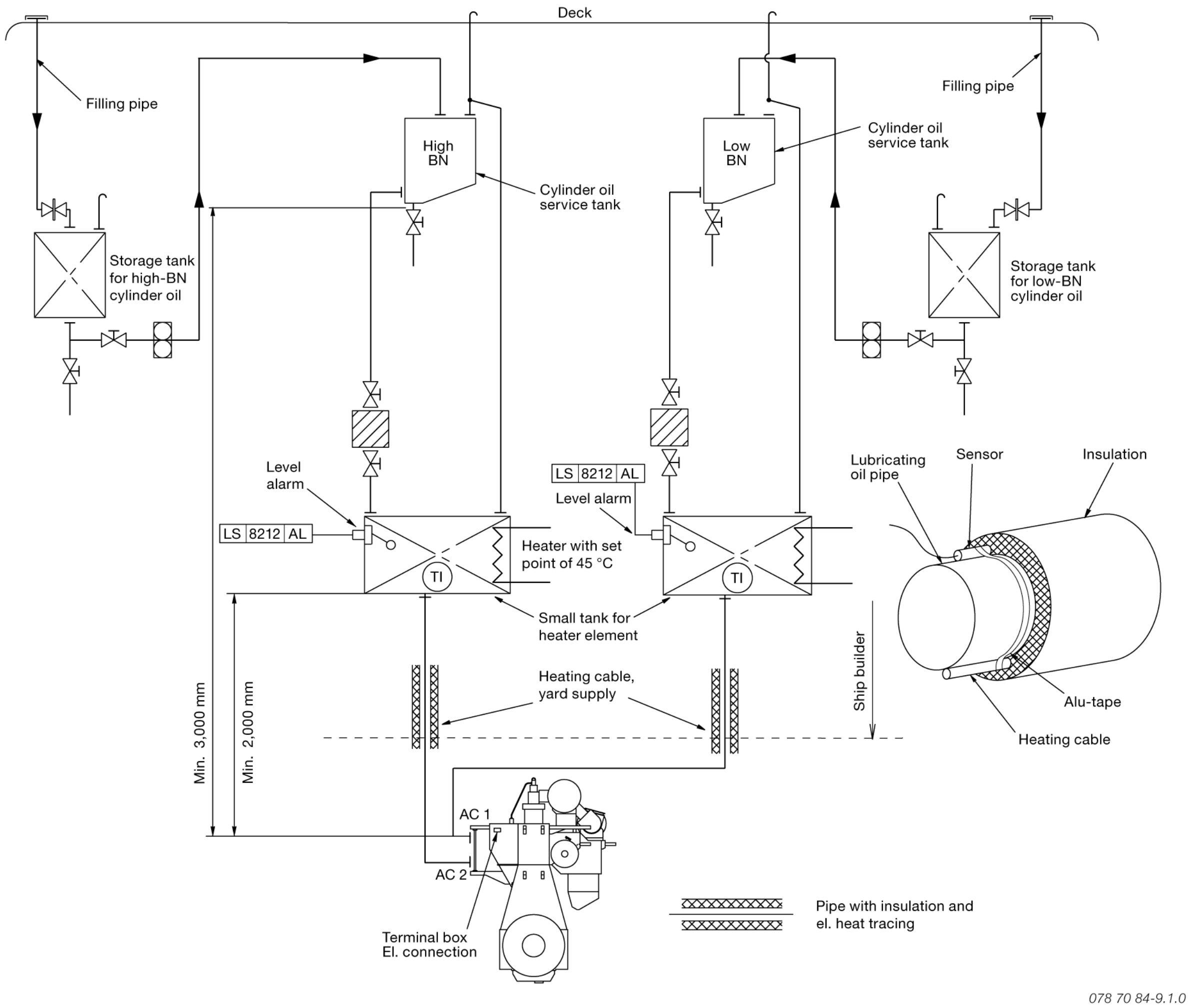

Fig. 9.02.02a: Cylinder lubricating oil system with dual storage and service tanks and ACOS (behind AC1 and AC2)

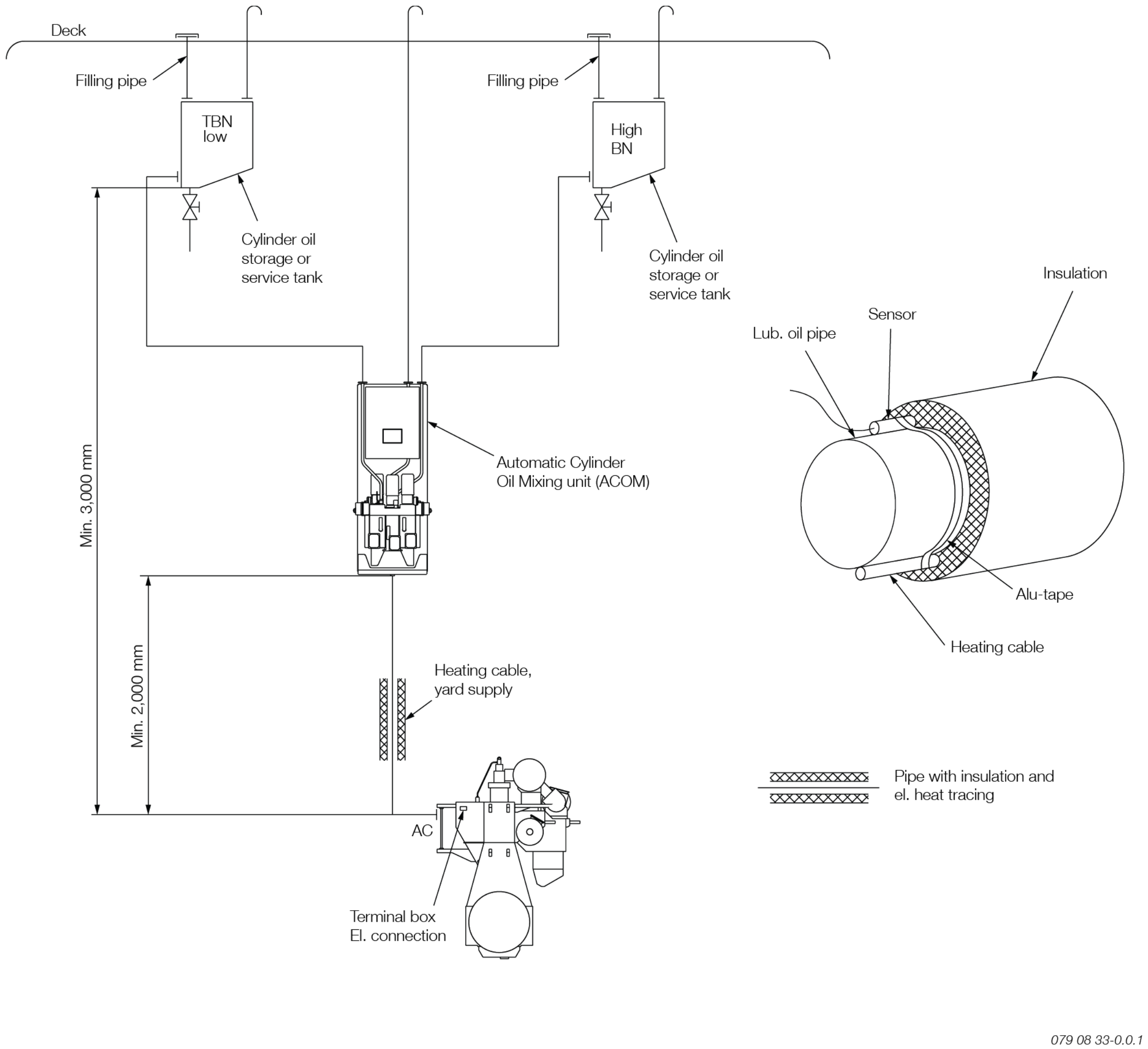

Fig. 9.02.02b: Cylinder lubricating oil system with dual storage or service tanks and ACOM

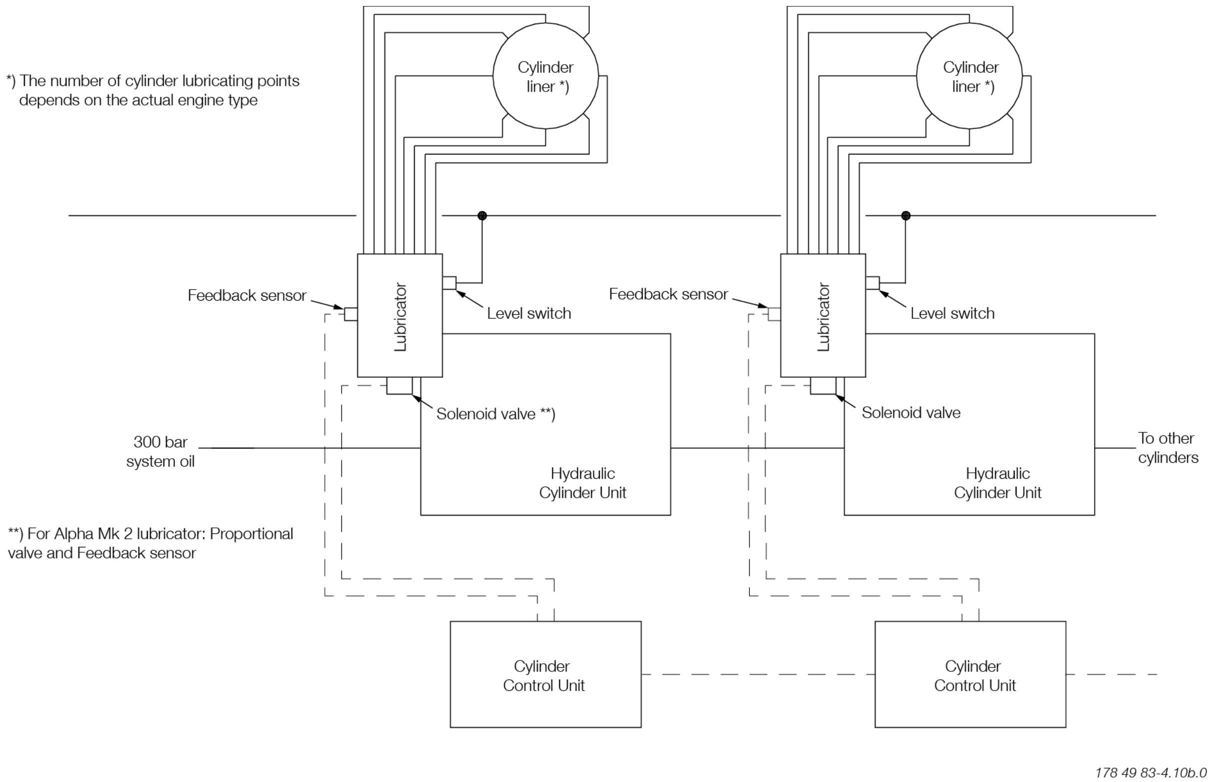

Fig. 9.02.02c: Cylinder lubricating oil system. Example from 80/70/65ME-C/-GI/-LGI engines

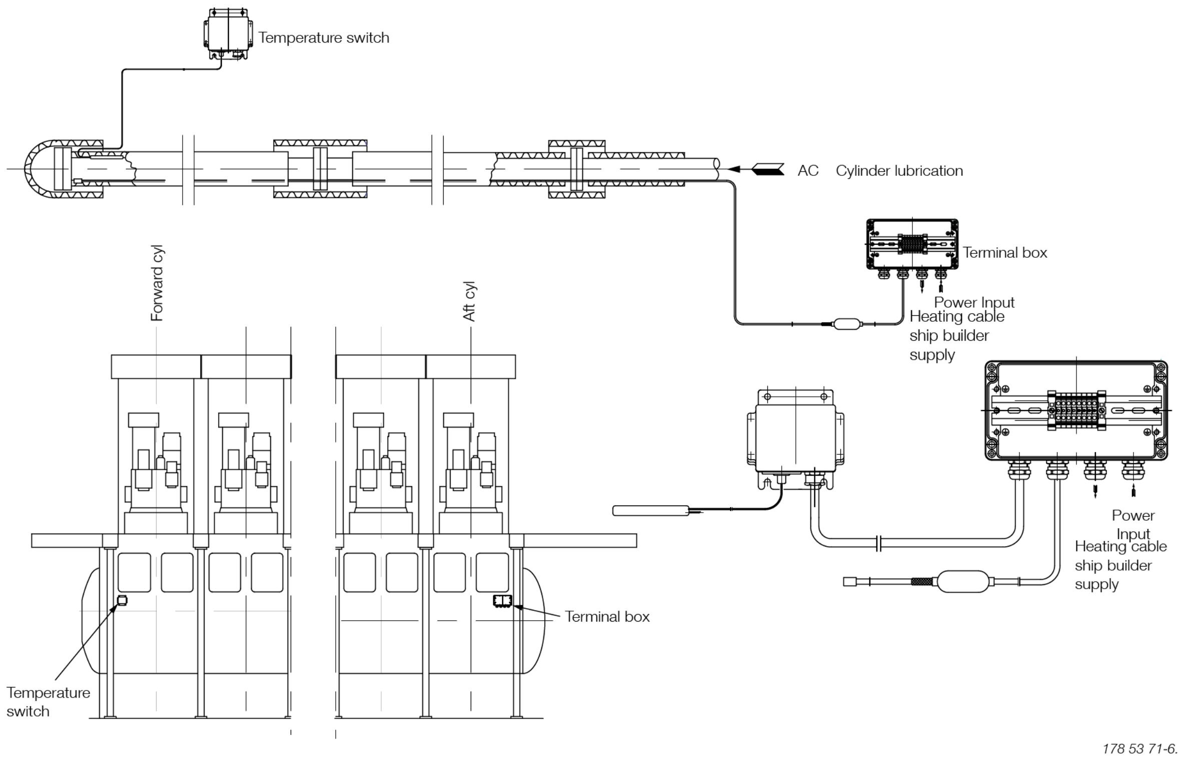

Fig. 9.02.03: Electric heating of cylinder oil pipes

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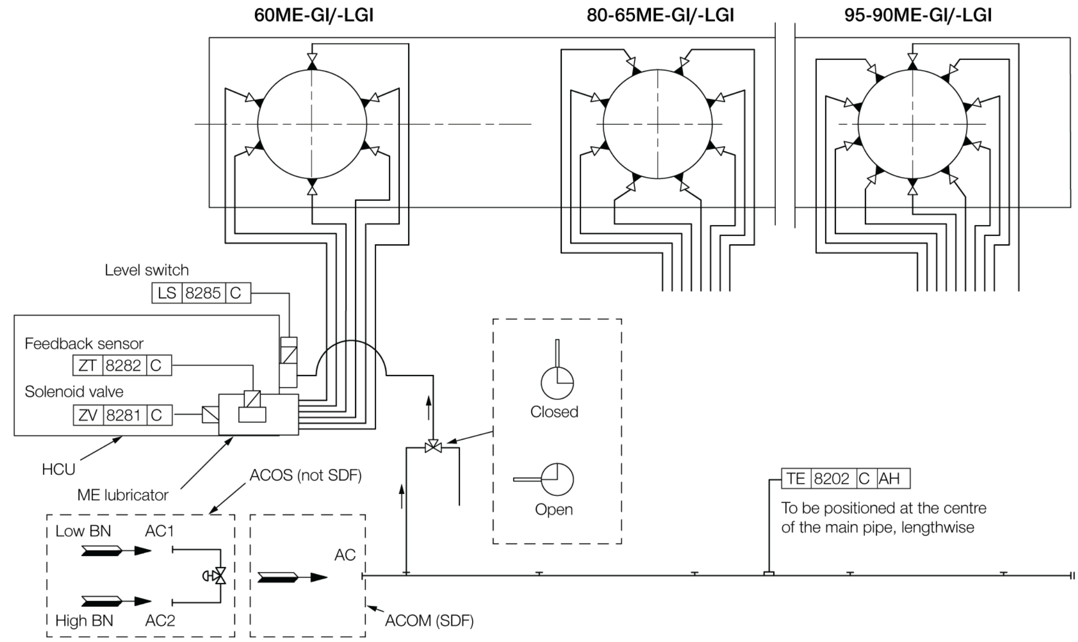

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Fig. 9.02.04a: Cylinder lubricating oil pipes, Alpha/ME lubricator

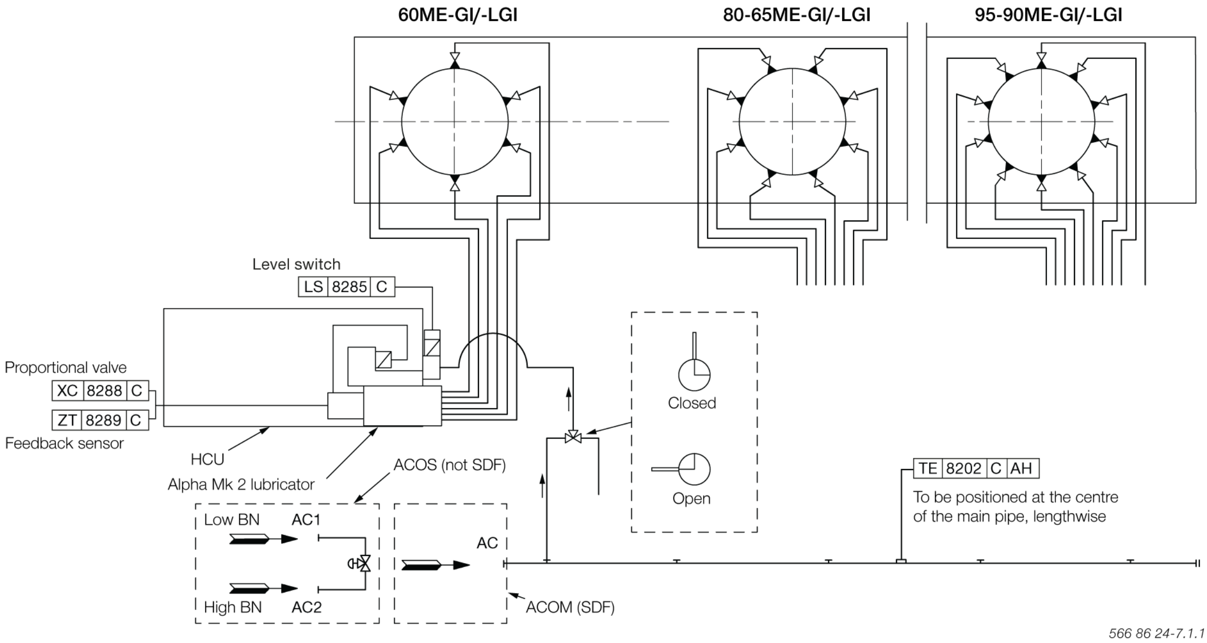

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Fig. 9.02.04b: Cylinder lubricating oil pipes, Alpha Mk 2 lubricator

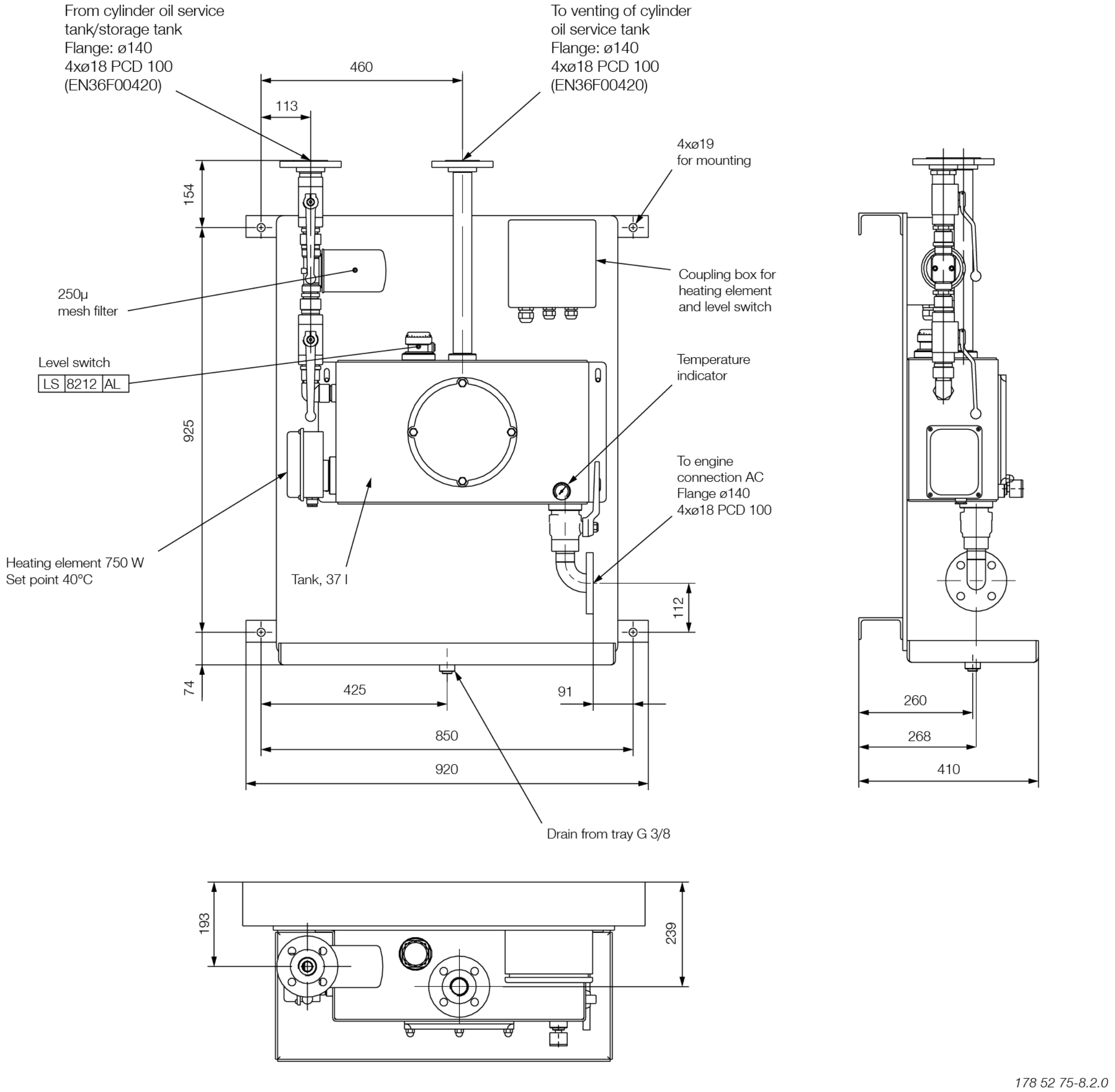

Fig. 9.02.05: Suggestion for small heating tank with filter (for engines without ACOM)

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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 typically about 10 - 15 litres/ 24 hours per cylinder during normal service. In the running-in period, it can be higher. The drain oil is a mixture of system oil from the crankcase, used cylinder oil, combustion residues and water from humidity in the scavenge air.

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).

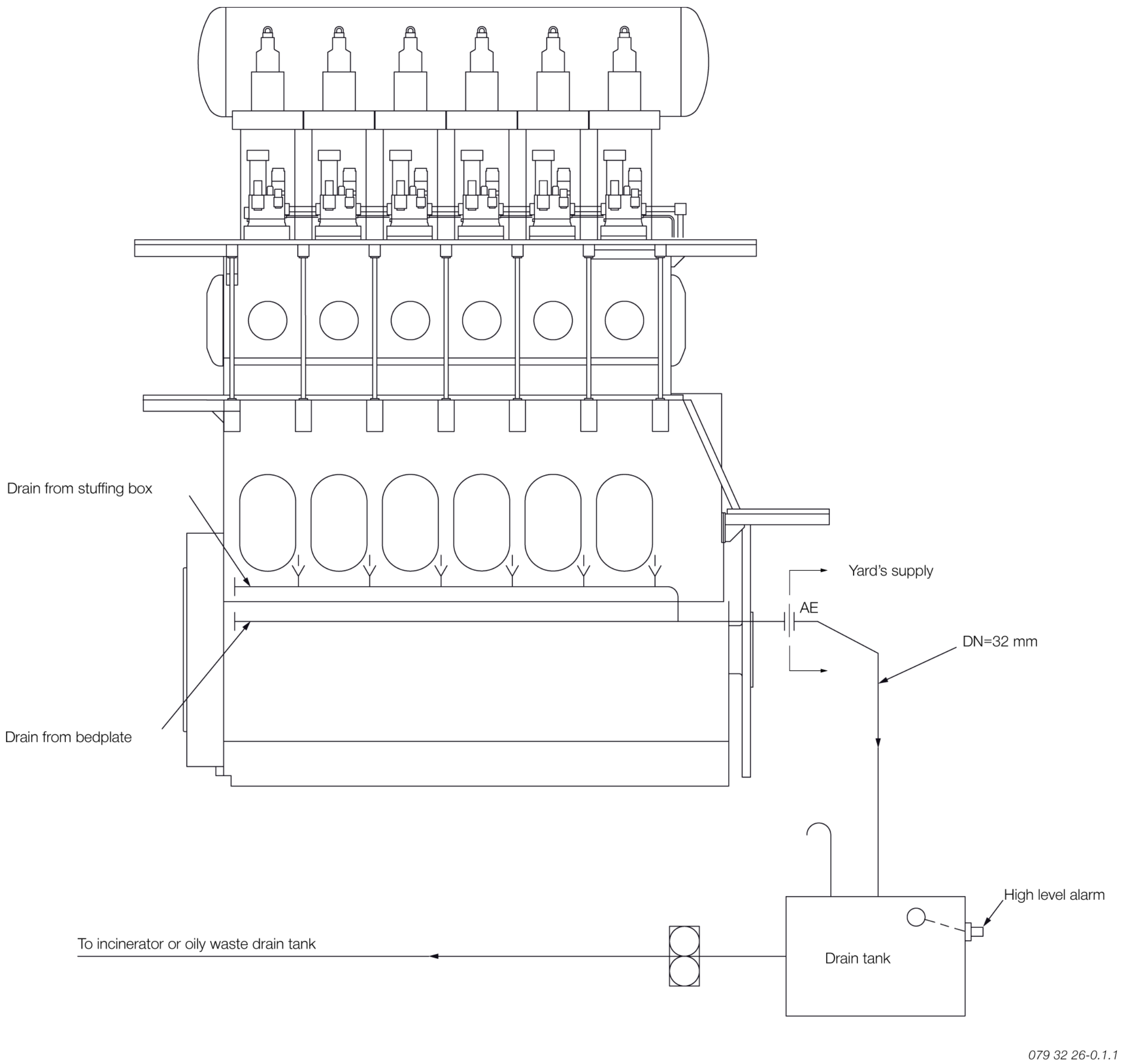

Fig. 10.01.01: Stuffing box drain oil system

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Low-temperature Cooling Water System

The low-temperature (LT) cooling water system supplies cooling water for the lubricating oil, jacket water and scavenge air coolers.

The LT cooling water system can be arranged in several configurations like a:

- Central cooling water system being the most common system choice and the basic execution for MAN B&W engines, EoD: 4 45 111
- Seawater cooling system being the most simple system and available as an option: 4 45 110
- Combined cooling water system with seawater cooled scavenge air cooler but freshwater cooled jacket water and lubricating oil cooler, available as an option: 4 45 117.

Principle diagrams of the above LT cooling water systems are shown in Fig. 11.01.01a, b and c and descriptions are found later in this chapter.

Further information and the latest recommendations concerning cooling water systems are found in MAN Energy SolutionsService Letters available at www.marine.man-es.com --> 'Two-Stroke' --> 'Service Letters'.

Chemical Corrosion Inhibition

Various types of inhibitors are available but, generally, only nitrite-borate based inhibitors are recommended.

Where the inhibitor maker specifies a certain range as normal concentration, we recommend to maintain the actual concentration in the upper end of that range.

MAN Energy Solutions recommends keeping a record of all tests to follow the condition and chemical properties of the cooling water and notice how it develops. It is recommended to record the quality of water as follows:

- Once a week:

  Take a sample from the circulating water during running, however not from the expansion tank nor the pipes leading to the tank. Check the condition of the cooling water. Test kits with instructions are normally available from the inhibitor supplier.
- Every third month:
  Take a water sample from the system during running, as described above in ‘Once a week’. Send the sample for laboratory analysis.
- Once a year:
  Empty, flush and refill the cooling water system. Add the inhibitor. For further information please refer to our recommendations for treatment of the jacket water/ freshwater. The recommendations are available from MAN Energy Solutions, Copenhagen.

Cooling System for Main Engines with EGR

For main engines with exhaust gas recirculation (EGR), a central cooling system using freshwater as cooling media will be specified.

Further information about cooling water systems for main engines with EGR is available from MAN Energy Solutions, Copenhegan.

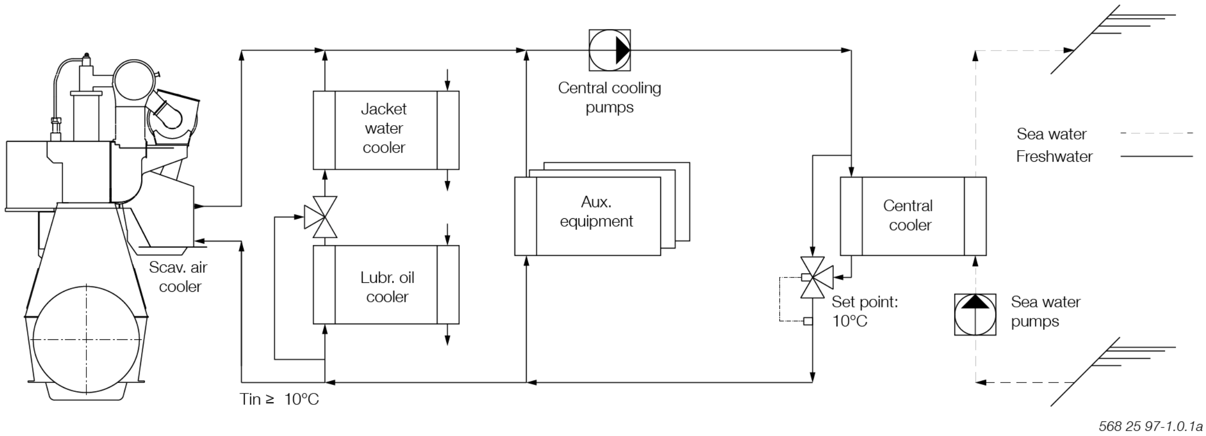


Fig. 11.01.01a: Principle diagram of central cooling water system

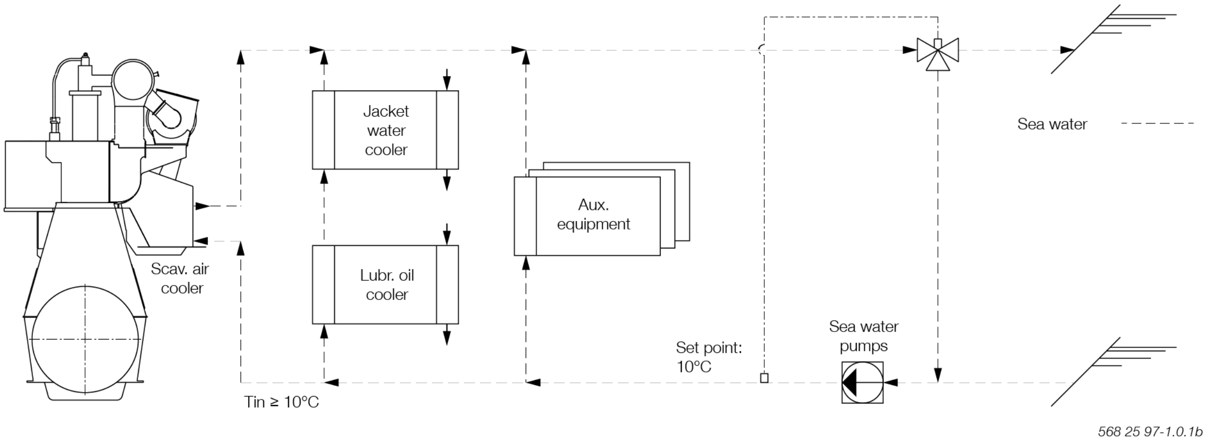


Fig. 11.01.01b: Principle diagram of seawater cooling system

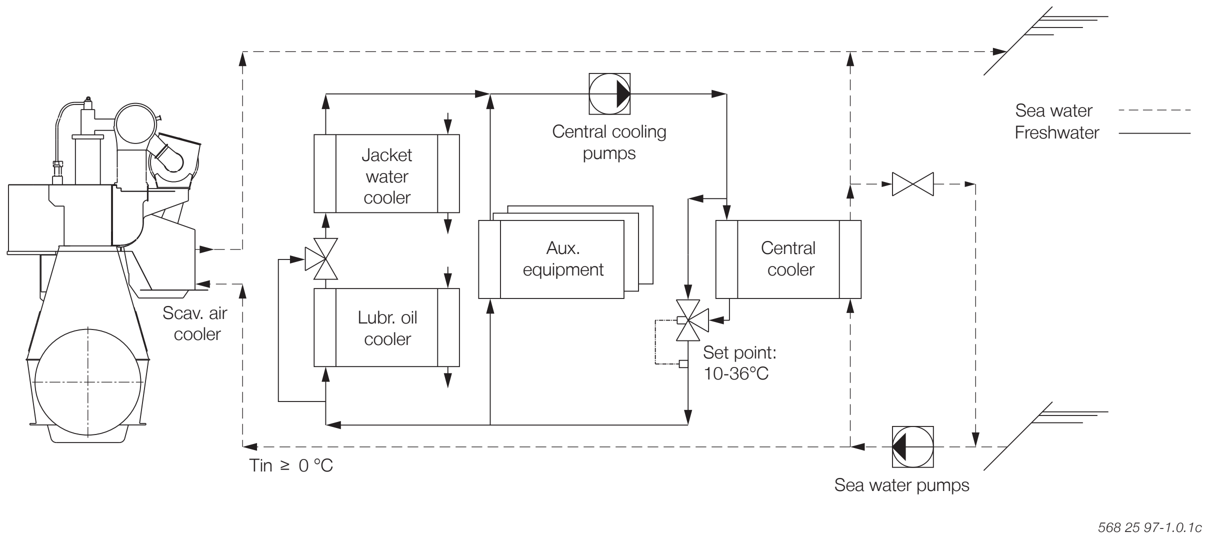


Fig. 11.01.01c: Principle diagram of combined cooling water system

Central Cooling Water System

The central cooling water system is characterized by having only one heat exchanger cooled by seawater. The other coolers, including the jacket water cooler, are then cooled by central cooling water.

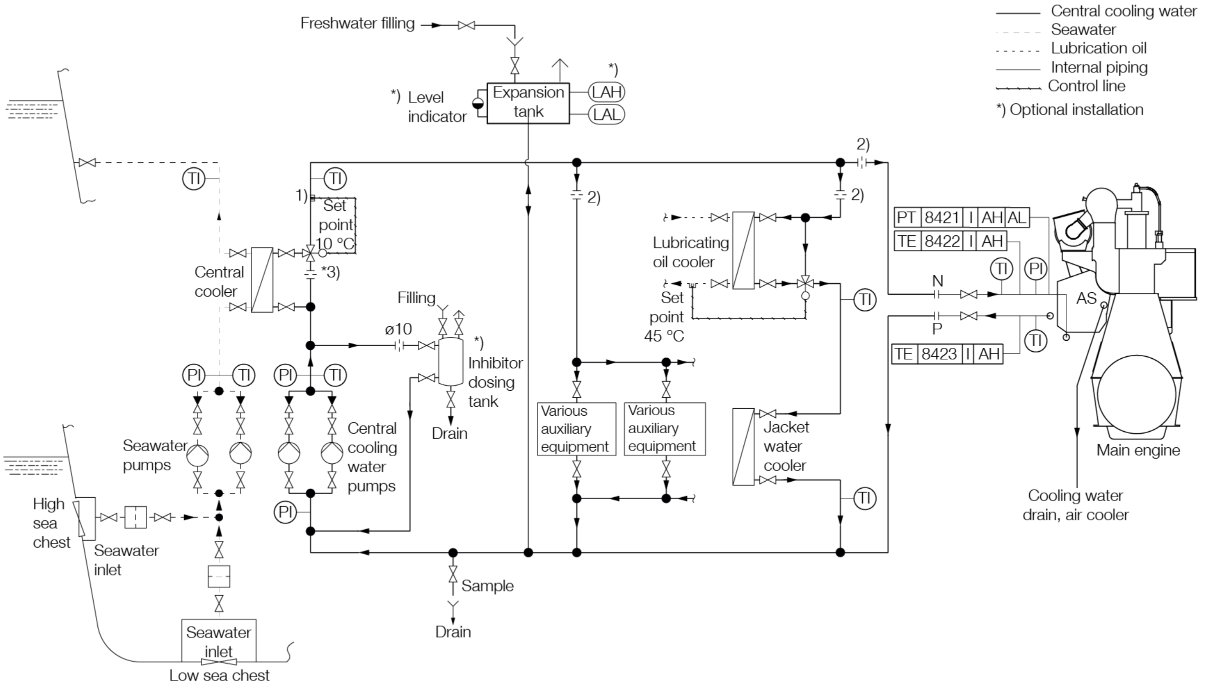
Cooling Water Temperature

The capacity of the seawater pumps, central cooler and freshwater pumps are based on the outlet temperature of the freshwater being maximum 54°C after passing through the main engine lubricating oil cooler. With an inlet temperature of maximum 36°C (tropical conditions), the maximum temperature increase is 18°C.

To achieve an optimal engine performance regarding fuel oil consumption and cylinder condition, it is important to ensure the lowest possible cooling water inlet temperature at the scavenge air cooler.

MAN Energy Solutions therefore requires that the temperature control valve in the central cooling water circuit is to be set to minimum 10°C. In this way, the temperature follows the outboard seawater temperature when the central cooling water temperature exceeds 10°C, see note 1 in Fig. 11.02.01.

Alternatively, in case flow control of the seawater pumps is applied, the set point is to be approximately 4°C above the seawater temperature but not lower than 10°C.



*) Optional installation

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Fig. 11.02.01: Central cooling water system

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Cooling Water Pump Capacities

The pump capacities listed by MAN Energy Solutions cover the requirement for the main engine only.

For any given plant, the specific capacities have to be determined according to the actual plant specification and the number of auxiliary equipment. Such equipment include GenSets, starting air compressors, provision compressors, air-conditioning compressors, etc.

A guideline for selecting centrifugal pumps is given in Section 6.04.

Cooling Water Piping

Orifices (or lockable adjustable valves for instance) must be installed in order to create:

- the proper distribution of flow between each of the central cooling water consumers, see note 2)
- a differential pressure identical to that of the central cooler at nominal central cooling water pump capacity, see note 3).

References are made to Fig. 11.02.01.

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

Central cooling water ....3.0 m/s
Seawater ....3.0 m/s

Expansion Tank Volume

The expansion tank shall be designed as open to atmosphere. Venting pipes entering the tank shall terminate below the lowest possible water level i.e. below the low level alarm.

The expansion tank volume has to be 10% of the total central cooling water amount in the system.

The 10% expansion tank volume is defined as the volume between the lowest level (at the low level alarm sensor) and the overflow pipe or high level alarm sensor.

If the pipe system is designed with possible air pockets, these have to be vented to the expansion tank.

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 ....2.0 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 pump head of the pumps is to be determined based on the total actual pressure drop across the seawater cooling water system.

A guideline for selecting centrifugal pumps is given in Section 6.04.

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. 0.7 bar
Seawater flow ....see ‘List of Capacities’
Seawater temperature, inlet ....32°C
Pressure drop on seawater side ....maximum 1.0 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 pump head of the pumps is to be determined based on the total actual pressure drop across the central cooling water system.

A guideline for selecting centrifugal pumps is given in Section 6.04.

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 bypasses all or part of the freshwater around the central cooler.

The sensor is to be located at the outlet pipe from the thermostatic valve and is set to keep a temperature of 10°C.

Lubricating Oil Cooler Thermostatic Valve

The lubricating oil cooler is to be equipped with a three-way valve, mounted as a mixing valve, which bypasses all or part of the freshwater around the lubricating cooler.

The sensor is to be located at the lubricating oil outlet pipe from the lubricating oil cooler and is set to keep a lubricating oil temperature of 45°C.

Chemical Corrosion Inhibitor and Dosing Tank

In order to properly mix the inhibitor into the central cooling water system circuit, the tank shall be designed to receive a small flow of jacket cooling water through the tank from the jacket water pumps. The tank shall be suitable for mixing inhibitors in form of both powder and liquid.

Recommended tank size 0.3 m3

Design pressure ....max. central cooling water system pressure

Suggested inlet orifice size ....ø10 mm

Lubricating Oil Cooler

See Chapter 8 ‘Lubricating Oil’.

Jacket Water Cooler

See Chapter 12 ‘High-temperature Cooling Water’.

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 ..... 0.3-0.8 bar

Cooling Water Pipes for Air Cooler

Diagrams of cooling water pipes for scavenge air cooler are shown in Figs. 11.08.01.

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Seawater Cooling System

The seawater cooling system is an option for cooling the main engine lubricating oil cooler, the jacket water cooler and the scavenge air cooler by seawater, see Fig. 11.04.01. The seawater system consists of pumps and a thermostatic valve.

Cooling Water Temperature

The capacity of the seawater pump is based on the outlet temperature of the seawater being maximum 50°C after passing through the main engine lubricating oil cooler, the jacket water cooler and the scavenge air cooler.

With an inlet temperature of maximum 32°C (tropical conditions), the maximum temperature increase is 18°C.

In order to prevent the lubricating oil from stiffening during cold services, a thermostatic valve is to be installed. The thermostatic valve recirculates all or part of the seawater to the suction side of the pumps. A set point of 10°C ensures that the cooling water to the cooling consumers will never fall below this temperature, see note 1 in Fig. 11.04.01.

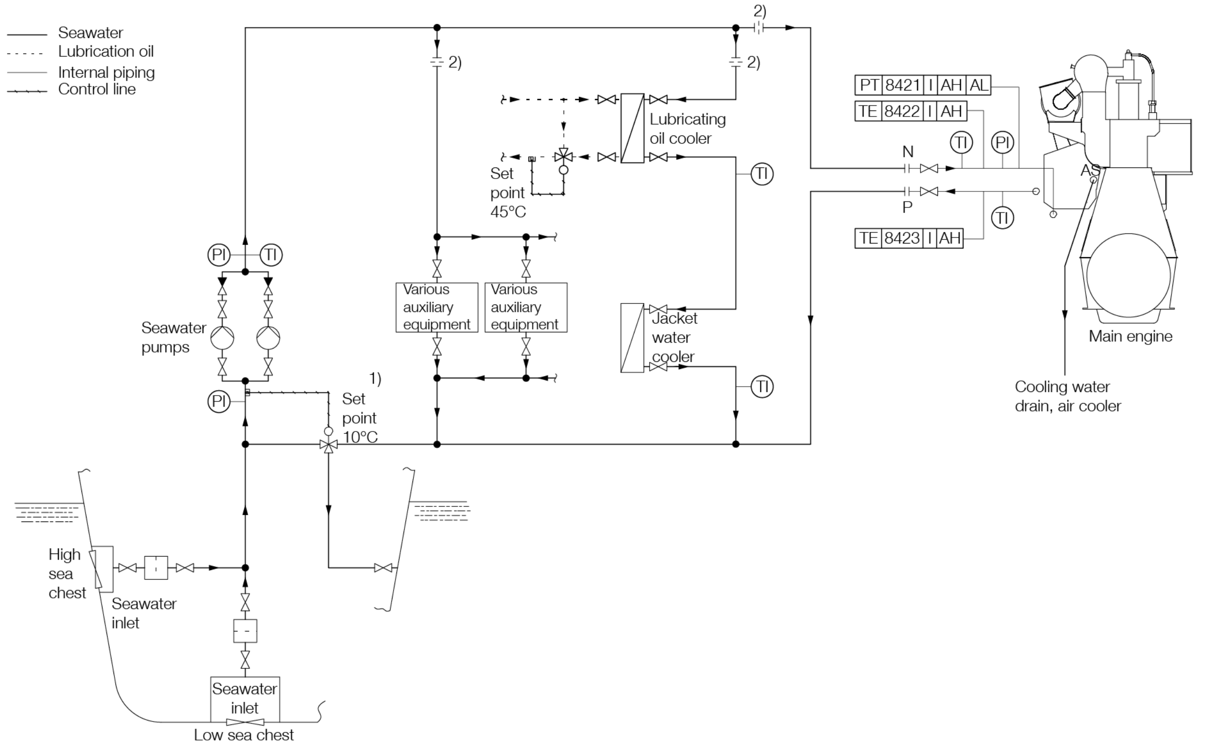

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Fig. 11.04.01: Seawater cooling system

Cooling Water Pump Capacities

The pump capacities listed by MAN Energy Solutions cover the requirement for the main engine only.

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For any given plant, the specific capacities have to be determined according to the actual plant specification and the number of auxiliary equipment. Such equipment include GenSets, starting air compressors, provision compressors, airconditioning compressors, etc.

A guideline for selecting centrifugal pumps is given in Section 6.04.

Cooling Water Piping

In order to create the proper distribution of flow between each of the cooling water consumers, orifices (or lockable adjustable valves for instance) must be installed, see note 2) in Fig. 11.04.01.

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

Seawater ....3.0 m/s

If the pipe system is designed with possible air pockets, these have to be vented to the expansion tank.

Components for Seawater Cooling 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 rule
Working temperature ....maximum 50°C

The flow capacity must be within a range from 100 to 110% of the capacity stated.

The pump head of the pumps is to be determined based on the total actual pressure drop across the seawater cooling water system.

A guideline for selecting centrifugal pumps is given in Section 6.04.

Seawater Thermostatic Valve

The temperature control valve is a three-way mixing valve. The sensor is to be located at the seawater inlet to the lubricating oil cooler, and the temperature set point must be +10 °C.

Seawater flow ....see ‘List of Capacities’
Temperature set point ....+10°C

Lubricating Oil Cooler

See Chapter 8 ‘Lubricating Oil’.

Jacket Water Cooler

See Chapter 12 ‘High-temperature Cooling Water’.

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 .... 0.3-0.8 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.

Cooling Water Pipes for Air Cooler

Diagrams of cooling water pipes for scavenge air cooler are shown in Figs. 11.08.01.

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Combined Cooling Water System

The combined cooling water system is characterized by having one heat exchanger and the scavenge air cooler cooled by seawater. The other coolers, including the jacket water cooler, are then cooled by central cooling water.

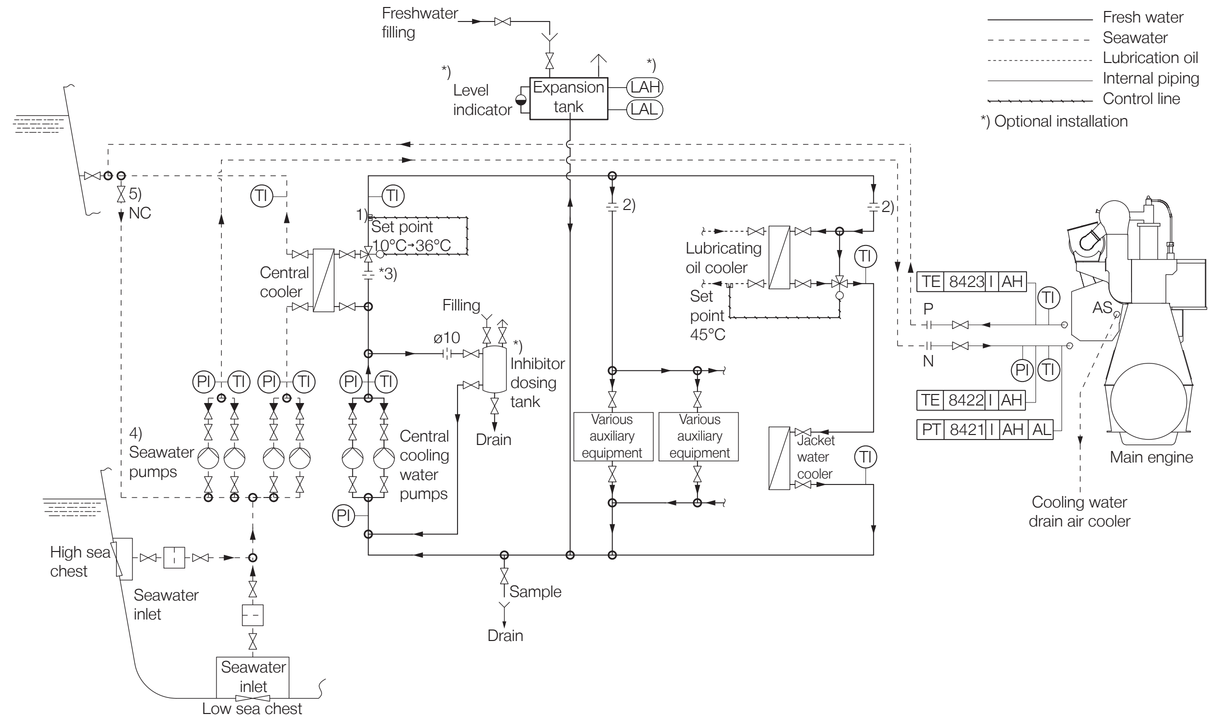
In this system, the cooling water to the scavenge air cooler will always be approx. 4°C lower than in a central cooling water system.

Cooling Water Temperature

The capacity of the seawater pumps, central cooler pumps are based on the outlet temperature of the freshwater being maximum 54°C after passing through the main engine lubricating oil cooler.

With an inlet temperature of maximum 36°C (tropical conditions), the maximum temperature increase is 18°C.

The temperature control valve in the central cooling water circuit can be set to minimum 10°C and maximum 36°C, see note 1 in Fig. 11.06.01.



*) optional installation
The letters refer to list of 'Counterflanges'
The item no. refer to 'Guidance Values Automation'

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Fig. 11.06.01: Combined cooling water system

Alternatively, in case flow control of the seawater pumps is applied, the set point is to be approximately 4°C above the seawater temperature but not lower than 10°C.

In order to avoid seawater temperatures below 0°C at the scavenge air cooler inlet, a manual by-pass valve is installed in the seawater circuit, see note 5) in Fig. 11.06.01. The valve recirculates all or part of the seawater to the suction side of the pumps.

Cooling Water Pump Capacities

The pump capacities listed by MAN Energy Solutions cover the requirement for the main engine only.

For any given plant, the specific capacities have to be determined according to the actual plant specification and the number of auxiliary equipment. Such equipment include GenSets, starting air compressors, provision compressors, airconditioning compressors, etc.

In fig. 11.06.01, note 4 both seawater pumps for main engine scavenge air cooler and for central cooling water system are shown. Alternative common seawater pumps serving both systems can be installed.

A guideline for selecting centrifugal pumps is given in Section 6.04.

Cooling Water Piping

Orifices (or lockable adjustable valves for instance) must be installed in order to create:

- the proper distribution of flow between each of the central cooling water consumers, see note 2)
- a differential pressure identical to that of the central cooler at nominal central cooling water pump capacity, see note 3).

References are made to Fig. 11.08.01.

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

Central cooling water ....3.0 m/s
Seawater ....3.0 m/s

Expansion Tank Volume

The expansion tank shall be designed as open to atmosphere. Venting pipes entering the tank shall terminate below the lowest possible water level i.e. below the low level alarm.

The expansion tank volume has to be 10% of the total central cooling water amount in the system. The 10% expansion tank volume is defined as the volume between the lowest level (at the low level alarm sensor) and the overflow pipe or high level alarm sensor.

If the pipe system is designed with possible air pockets, these have to be vented to the expansion tank.

Components for Combined Cooling Water System

Seawater Cooling Pumps

The pumps are to be of the centrifugal type.

Seawater flow ....see 'List of Capacities'
Pump head ....2.0 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 pump head of the pumps is to be determined based on the total actual pressure drop across the seawater cooling water system.

A guideline for selecting centrifugal pumps is given in Section 6.04.

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. 0.7 bar
Seawater flow ....see 'List of Capacities'
Seawater temperature, inlet ....32°C
Pressure drop on
seawater side ....maximum 1.0 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 pump head of the pumps is to be determined based on the total actual pressure drop across the central cooling water system.

A guideline for selecting centrifugal pumps is given in Section 6.04.

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 bypasses all or part of the freshwater around the central cooler.

The sensor is to be located at the outlet pipe from the thermostatic valve and is set to keep a temperature of minimum 10 °C and maximum 36°C.

Lubricating Oil Cooler Thermostatic Valve

The lubricating oil cooler is to be equipped with a three-way valve, mounted as a mixing valve, which bypasses all or part of the freshwater around the lubricating cooler.

The sensor is to be located at the lubricating oil outlet pipe from the lubricating oil cooler and is set to keep a lubricating oil temperature of 45°C.

Chemical Corrosion Inhibitor and Dosing Tank

In order to properly mix the inhibitor into the combined cooling water system circuit, the tank shall be designed to receive a small flow of jacket cooling water through the tank from the jacket water pumps. The tank shall be suitable for mixing inhibitors in form of both powder and liquid.

Recommended tank size 0.3 m3
Design pressure ......max. combined cooling water system pressure
Suggested inlet orifice size ....ø10 mm

Lubricating Oil Cooler

See Chapter 8 ‘Lubricating Oil’.

Jacket Water Cooler

See Chapter 12 ‘High-temperature Cooling Water’.

Scavange Air Cooler

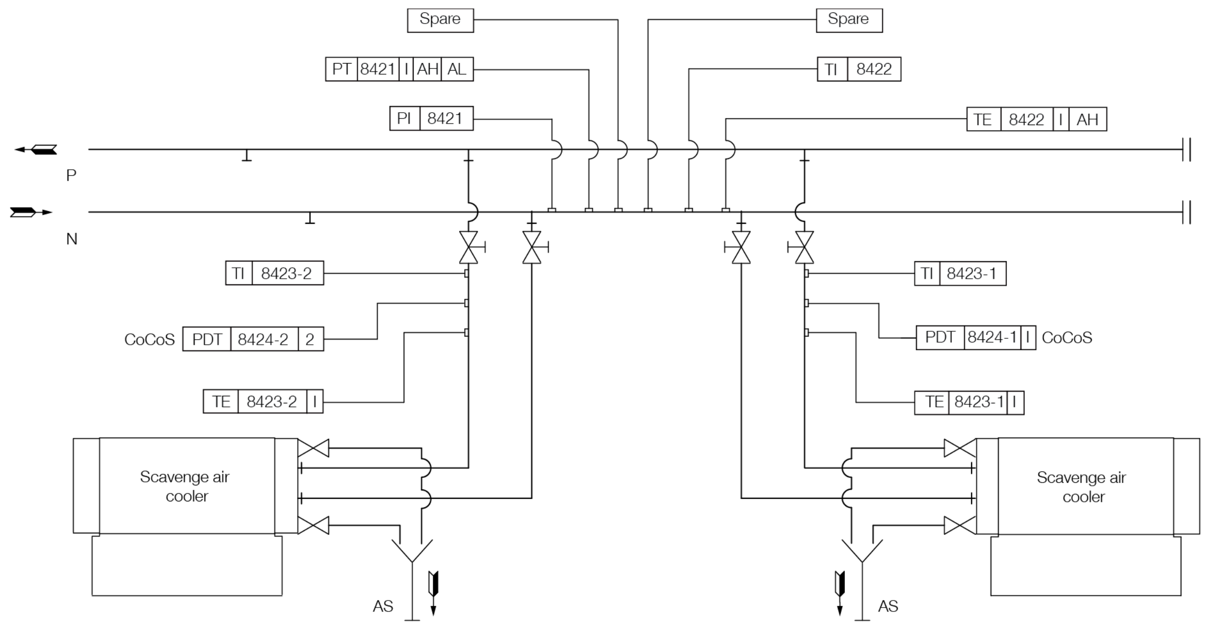
The scavenge air cooler is an integrated part of the main engine.

Heat dissipation ....see ‘List of Capacities’
Seawater flow ....see ‘List of Capacities’
Central cooling temperature, inlet ....32°C
Pressure drop on seawater side .... 0.3-0.8 bar

Cooling Water Pipes for Air Cooler

Diagrams of cooling water pipes for scavenge air cooler are shown in Figs. 11.08.01.

Cooling Water Pipes for Scavenge Air Cooler



The letters refers to list of 'Counterflanges' . The item no. refer to 'Guidance Values Automation'

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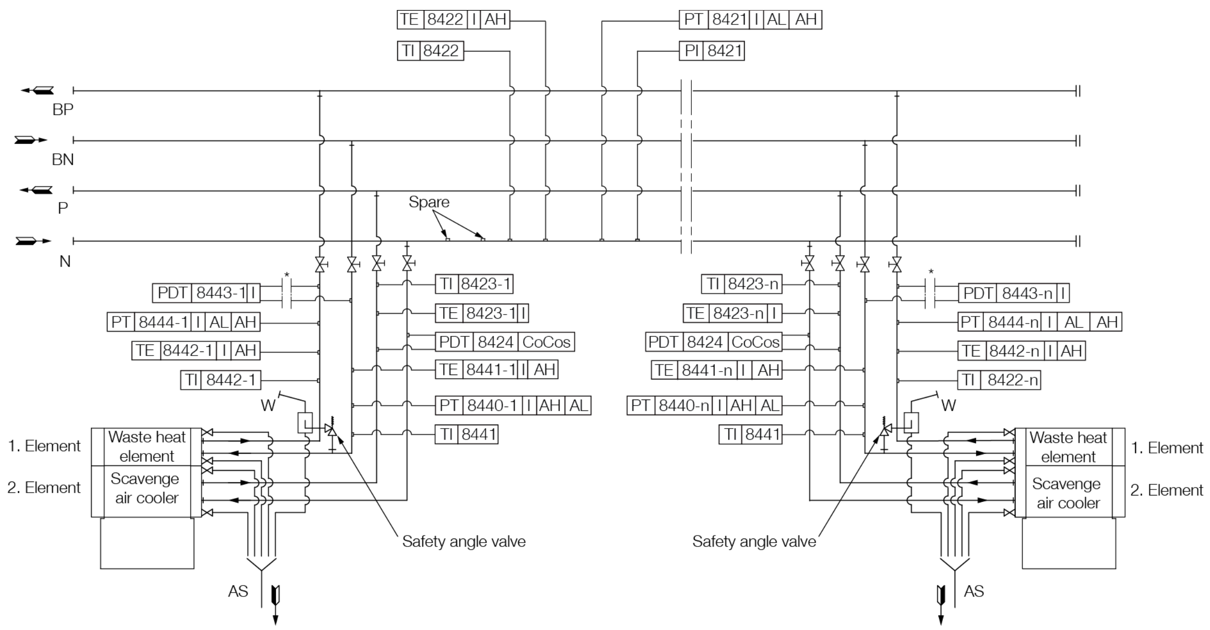
Fig. 11.08.01a: Cooling water pipes for engines with two or more turbochargers



The letters refer to list of 'Counterflanges'. The item no. refer to 'Guidance Values Automation'
* Calculated value: PT8444-n subtracted from PT8440-n, if possible
n Refer to number of air coolers

521 21 78-2.3.1

Fig. 11.08.01b: Cooling water pipes with waste heat recovery for engines with two or more turbochargers

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High-temperature cooling water system

The high-temperature (HT) cooling water system, also known as the jacket cooling water (JCW) system, is used for cooling the cylinder unit components of the main engine and heating of the fuel oil drain pipes, see Fig. 12.01.01 to 12.01.04.

The jacket water pump draws water from the jacket water cooler outlet through a deaerating tank, and delivers it to the engine.

A thermostatically controlled regulating valve is located at the inlet, or alternatively at the outlet from the cooler. The regulating valve keeps the main engine cooling water outlet at a fixed temperature level, independent of the engine load.

A deaerating tank alarm device is installed between the deaerating tank and the expansion tank. The purpose of the alarm device is to sound an alarm if a large amount of gas is detected in the JCW circuit, for example caused by a cylinder liner rupture.

An expansion tank is installed to create a sufficient static pressure in the JCW system and to provide space for the water to expand and contract. The expansion tank must be placed at least 15 m above the top of the main engine exhaust gas valves.

The engine jacket water must be carefully treated, maintained and monitored to avoid corrosion, corrosion fatigue, cavitation and scale formation. Therefore, it is recommended installing a chemical corrosion inhibitor dosing tank and a means to take water samples from the JCW system.

Chemical corrosion inhibition

Various types of inhibitors are available but, generally, only nitrite-borate based inhibitors are recommended.

Where the inhibitor maker specifies the normal concentration as a specific range, MAN Energy Solutions recommends maintaining the actual concentration in the upper end of that range.

MAN Energy Solutions recommends keeping a record of all tests to follow the condition and chemical properties of the cooling water, and notice how it develops. It is recommended recording the quality of water as follows:

- Once a week:

  Take a sample from the circulating water during running, however, not from the expansion tank or the pipes leading to the tank. Check the condition of the cooling water. Test kits with instructions are normally available from the inhibitor supplier.
- Every third month:

  Take a water sample from the system during running, as described above in "Once a week". Send the sample for laboratory analysis.
- Once a year:

  Empty, flush and refill the cooling water system. Add the inhibitor.

For further information, refer to our recommendations for treatment of the jacket water/freshwater. The recommendations are available from MAN Energy Solutions, Copenhagen.

Cooling water drain for maintenance

For maintenance of the main engine, a drain arrangement is installed at the engine. By this drain arrangement, the jacket cooling water can be drained to, for example, a freshwater drain tank for possible reuse of the chemical corrosion inhibitor-treated water.

Preheater system

During short stays in port (that is, less than 4-5 days), it is recommended that the engine is kept preheated. The purpose is to prevent temperature variations in the engine structure and corresponding variations in thermal expansion 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. Preheating could be provided by a built-in preheater in the jacket cooling water system, by means of cooling water from the auxiliary engines, or a combination of the two.

Preheating procedure

To protect the engine and to avoid corrosive attacks on the cylinder liners during starting, minimum temperature restrictions must be considered before starting the engine.

Normal engine start, fixed pitch propeller

A minimum engine jacket water temperature of 50°C is recommended before starting the engine and gradually running it up to 80-90% SMCR speed (SMCR rpm) during a 30 minute period.

To run up the engine to 90-100% SMCR rpm, it is recommended increasing the speed slowly during a period of 60 minutes.

Start of cold engine, fixed pitch propeller

In exceptional circumstances where it is not possible to comply with the above recommendation, a minimum temperature of 20°C can be accepted before the engine is started and slowly run up to 80% SMCR rpm.

Before exceeding 80% SMCR rpm, a minimum jacket water temperature of 50°C should be obtained before the above described normal load-up procedure for starting the engine can be continued.

Note:

The above considerations for starting a cold engine are based on the assumption that the engine has already been well run in.

For further information, see our publication:

Influence of Ambient Temperature Conditions

Freshwater generator

A freshwater generator can be installed in the JCW circuit for utilising the heat radiated to the jacket cooling water from the main engine.

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Jacket cooling water piping - ME/-LGI engines (fuel sulphur content <= 0.5%)

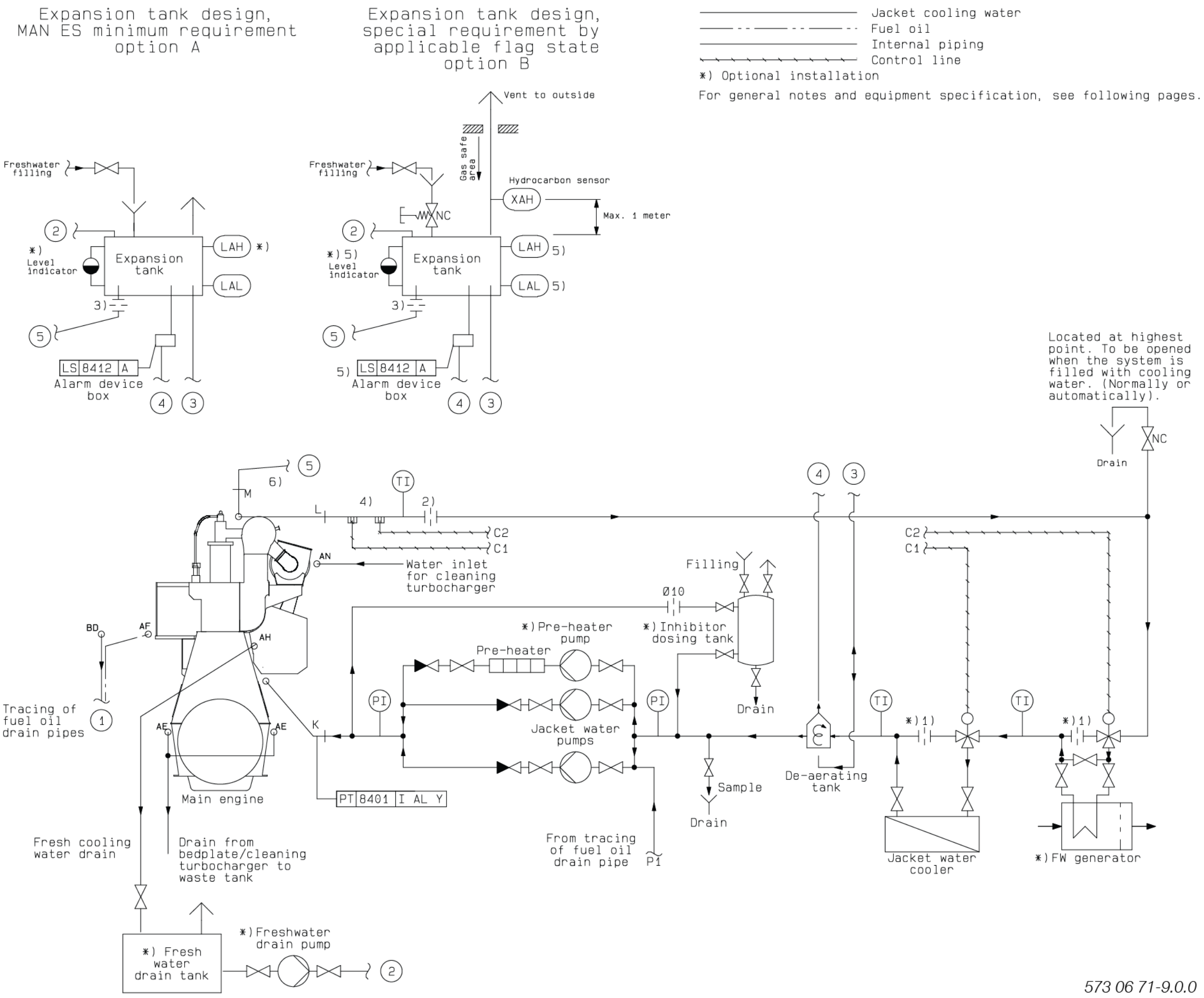

Fig. 12.01.01: Jacket cooling water system for ME/-LGI engines (fuel sulphur content <=0.5%)

Notes:

1. Orifices (or lockable adjustable valves) to be installed to create a differential pressure identical to that of the jacket water cooler/freshwater generator at nominal jacket water pump capacity.
2. (Optional) Install orifices to create a minimum inlet pressure at engine inlet connection "K" (at sensor PT 8401) which is higher than the minimum pressure stated in "Guidance Values Automation" (GVA).
3. Flow restriction: Install an orifice with a small hole to avoid jacket water flow through the expansion tank.
4. Temperature setpoint for thermostatic regulating valves
   Thermostatic regulating valve for jacket water cooler: Setpoint: 87°C if a thermostatic regulating valve is installed for the freshwater generator, otherwise 85°C.
   Thermostatic regulating valve for freshwater generator cooler: Setpoint: 83°C.

5. IEC Ex proof equipment classification
   Sensor type: Type according to applicable classification society requirement.
6. Deaeration of cooling water manifold of exhaust gas valve
   Engine connection M: Venting pipe has to be arranged in fore or aft end of the cooling water manifold discharge pipe from the exhaust gas valve (opposite end of discharge to jacket water pump). An automatic deaerating valve located at the engine for expansion tank design option A can replace the vent pipe.

The letters refer to list of "Counterflanges"

For external pipe connections, we prescribe the following maximum water velocity:

Jacket cooling water .... 3.0 m/s

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Jacket cooling water piping - ME/-LGI engines (fuel sulphur content >0.5%)

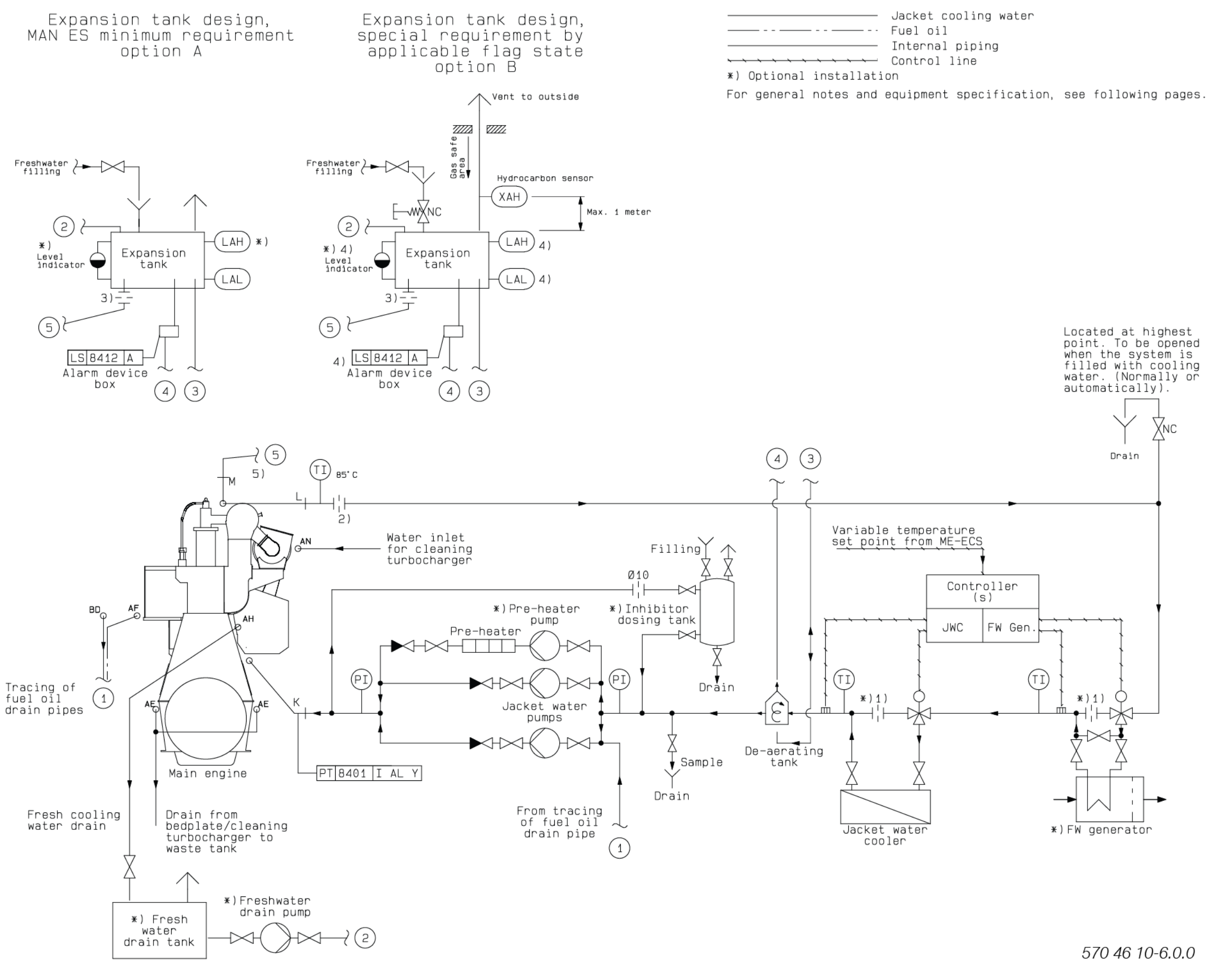

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Fig. 12.01.02: Jacket cooling water system for ME/-LGI engines (fuel sulphur content >0.5%)

Notes:

1. Orifices (or lockable adjustable valves) to be installed to create a differential pressure identical to that of the jacket water cooler/freshwater generator at nominal jacket water pump capacity.
2. (Optional) Install orifices to create a minimum inlet pressure at engine inlet connection "K" (at sensor PT 8401) which is higher than the minimum pressure stated in "Guidance Values Automation" (GVA).
3. Flow restriction: Install an orifice with a small hole to avoid jacket water flow through the expansion tank.
4. IEC Ex proof equipment classification
   Sensor type: Type according to applicable classification society requirement.
5. Deaeration of cooling water manifold of exhaust gas valve
   Engine connection M: Venting pipe has to be arranged in fore or aft end of the cooling water manifold discharge pipe from the exhaust gas valves

(opposite end of discharge to jacket water pump). An automatic deaerating valve located at the engine for expansion tank design option A can replace the vent pipe.

The letters refer to list of "Counterflanges"

For external pipe connections, we prescribe the following maximum water velocity:

Jacket cooling water .... 3.0 m/s

Jacket cooling water piping - ME-GI engines

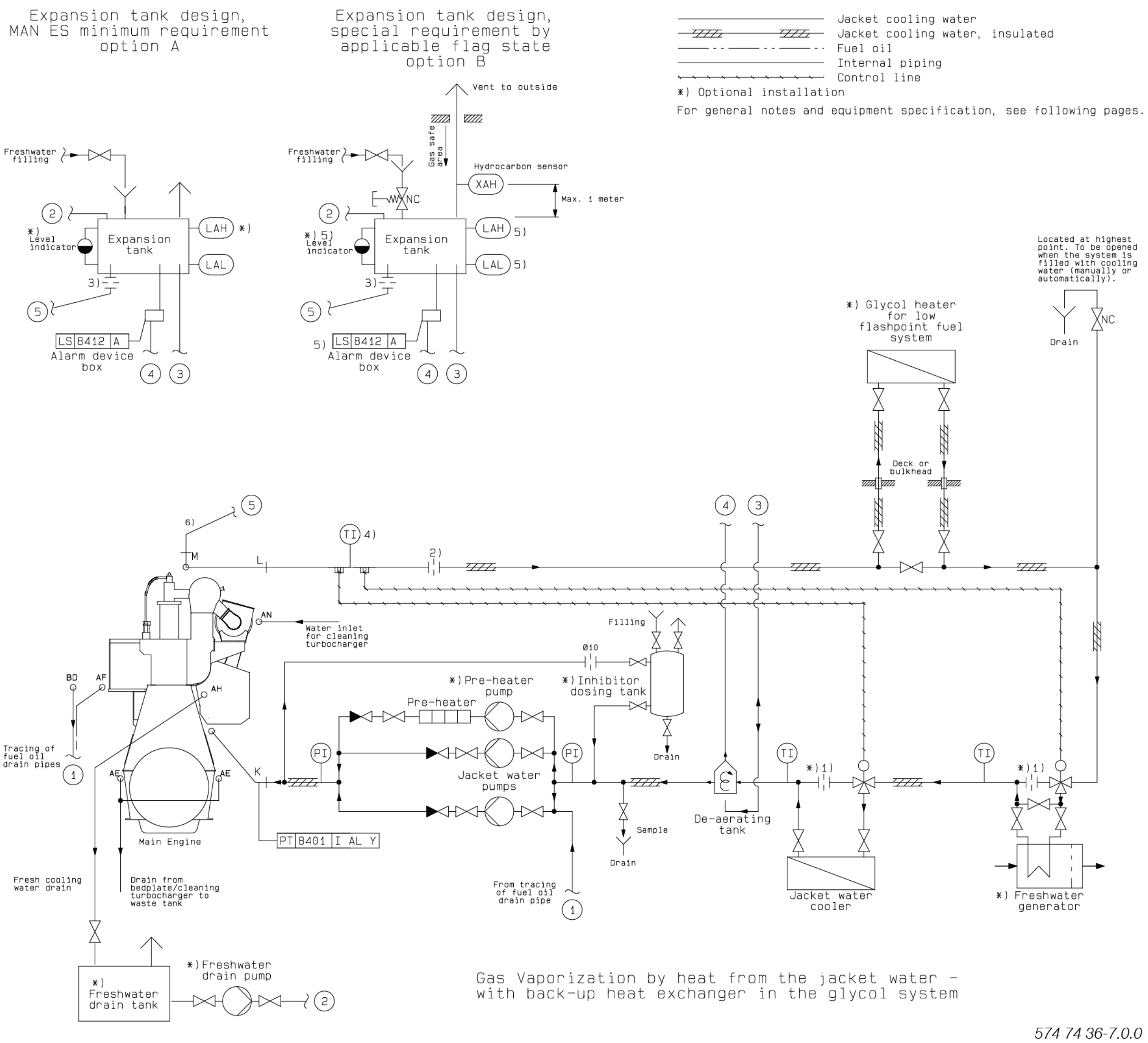

Fig. 12.01.03: Jacket cooling water system for ME-GI engines

Notes:

1. (Optional) Orifices (or lockable adjustable valves) to be installed to create a differential pressure identical to that of the jacket water cooler/freshwater generator at nominal jacket water pump capacity.
2. (Optional) Install orifices to create a minimum inlet pressure at engine inlet connection "K" (at sensor PT 8401), which is higher than the minimum pressure stated in "Guidance Values Automation" (GVA).
3. Flow restriction: Install an orifice with a small hole to avoid jacket water flow through the expansion tank.
4. Temperature setpoint for thermostatic regulating valves
   Thermostatic regulating valve for jacket water cooler: Setpoint: 87°C if a thermostatic regulating valve is installed for freshwater generator, other-

wise 85°C.
Thermostatic regulating valve for freshwater generator cooler: Setpoint: 83°C.

5. IEC Ex proof equipment classification
   Sensor type: Type according to applicable classification society requirement.
6. Deaeration of cooling water manifold of exhaust gas valve
   Engine connection M: Venting pipe has to be arranged in fore or aft end of cooling water manifold discharge pipe from the exhaust gas valve (opposite end of discharge to jacket water pump). An automatic deaerating valve located at the engine for expansion tank design option A can replace the vent pipe..

The letters refer to list of "Counterflanges"

For external pipe connections, we prescribe the following maximum water velocity:

Jacket cooling water .... 3.0 m/s

Jacket cooling water piping - ME-GA engines

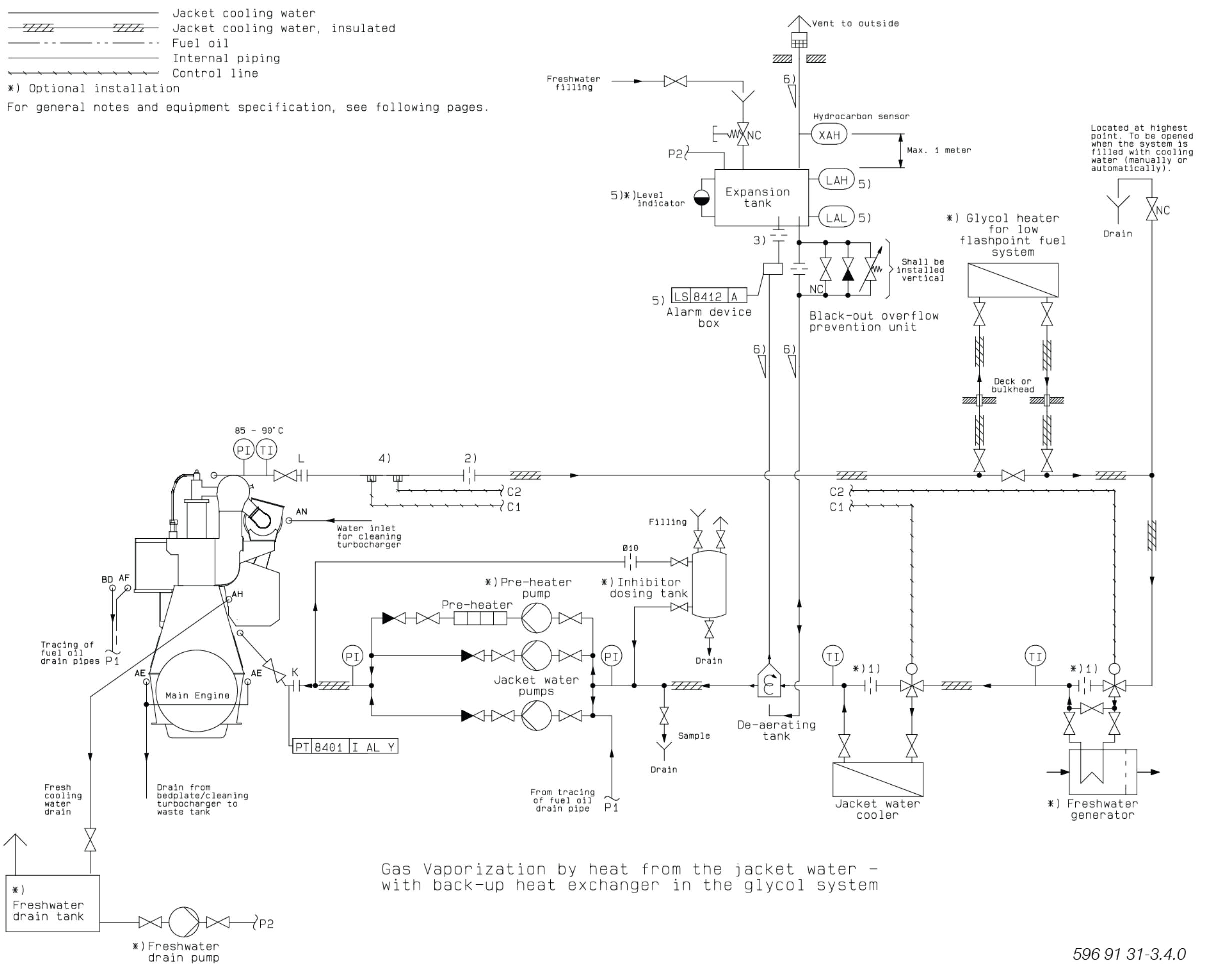

Fig. 12.01.04: Jacket cooling water system for ME-GA engines

Notes:

1. Orifices (or lockable adjustable valves) to be installed to create a differential pressure identical to that of the jacket water cooler / freshwater generator at nominal jacket water pump capacity.
2. (Optional) Install orifices to create a minimum inlet pressure at engine inlet connection "K" (at sensor PT 8401), which is higher than the minimum pressure stated in "Guidance Values Automation" (GVA).
3. Flow restriction: Install an orifice with a small hole to avoid jacket water flow through the expansion tank.
4. Temperature setpoint
   Thermostatic regulating valve for jacket water cooler: Setpoint: 87°C if a thermostatic regulating valve is installed for the freshwater generator, otherwise 85°C.
   Freshwater generating cooler thermostatic regulation valve: Setpoint: 83°C.
5. IEC Ex proof equipment classification
   Sensor type: IECEx approved

6. Vent pipe routing inclination
   Vent pipes has to be routed steadily ascending so possible air in the system easily can escape according to the vessel static longitudinal and transverse inclination. In general a longitudinal inclination shall be taken as min. 5° (might be smaller based on length of vessel and applicable classification society rules) and a transverse inclination shall be taken as min. 15°.

The letters refer to list of "Counterflanges"

For external pipe connections, we prescribe the following maximum water velocity:

Jacket cooling water .... 3.0 m/s

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Components

Jacket water cooling pump

The pumps should be of the centrifugal type.

Pump flow rate/jacket water flow	see 'List of capacities'
Pump head (see below note)	3.0 bar
Delivery pressure	depends on the location of the expansion tank
Test pressure	according to Class rules
Working temperature	85°C
Max. temperature (design purpose)	100°C

The flow capacity must be within the range from 100 to 110% of the capacity stated.

The pump head of the pumps should be determined based on the actual total pressure drop across the cooling water system, i.e. the pressure drop across main engine, jacket water cooler, three-way valve, valves, and other pipe components.

Section 6.04 contains a guideline for selecting centrifugal pumps.

Jacket water cooler

Often the jacket water cooler is of the plate heat exchanger type, but it could also be of the shell and tube type.

Heat dissipation	see 'List of capacities'
Jacket water flow	see 'List of capacities'
Jacket water temperature, inlet	85°C
Max. working temperature	up to 100°C
Max. pressure drop on jacket water side	0.5 bar
Cooling water flow	see 'List of capacities'
Cooling water inlet temp., SW cooled	~38°C
Cooling water inlet temp., FW cooled	~43°C
Max. pressure drop on cooling side	0.5 bar

The following materials should be used for the cooler:

Seawater cooled	SW resistant (for example, titanium or copper alloy for tube coolers)
Freshwater cooled	stainless steel

The heat dissipation and flow are based on SMCR output at tropical conditions, i.e. a seawater temperature of 32°C and an ambient air temperature of 45°C.

Jacket water thermostatic regulating valve

The main engine cooling water outlet temperature should be kept at a fixed temperature of 85°C, independent of the engine load. This is done by a three-way thermostatic regulating valve.

The controller of the thermostatically controlled regulating valve must be able to receive a remote, variable set-point given by the main engine control system (ECS). The variable set-point corresponds to the main engine jacket water inlet temperature required for keeping the main engine outlet temperature at the specified 85°C.

The reference measurement temperature sensor must be located after the water has been mixed, i.e. between the cooler/cooler bypass and the jacket water pumps as indicated in Fig. 12.01.01.

Jacket water flow	see 'List of capacities'
Max. working temperature	up to 100°C
Max. pressure drop	~0.3 bar
Actuator type	electric or pneumatic is recommended
Leak rate	less than 0.5% of nominal flow

Note:
A low valve leak rate specified for the valve port against the cooler will provide better utilisation of the heat available for the freshwater production.

Valve controller specification:

Remote set-point signal standard	4-20 mA
Range	0-4 mA = 65°C; 20 mA = 95°C

Expansion tank

The expansion tank must be designed as an open tank towards the atmosphere. Venting pipes entering the tank must terminate below the lowest possible water level, i.e. below the low-level alarm.

The expansion tank must be located at least 15 m above the top of the main engine exhaust gas valves.

The expansion tank volume has to be at least 10% of the total jacket cooling water (JWC) amount in the system.

The 10% expansion tank volume is defined as the volume between the lowest level (at the low-level alarm sensor) and the overflow pipe or high-level alarm sensor.

De-aerating tank and alarm device

Design and dimensions of the de-aerating tank are shown in Fig. 12.02.01 and the corresponding alarm device is shown in Fig. 12.02.02.

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Chemical corrosion inhibitor and dosing tank

In order to properly mix the inhibitor into the JCW system, the tank must be designed to receive a small flow of JCW through the tank from the jacket water pumps. The tank must be suitable for mixing inhibitors in both powder and liquid form.

Recommended tank size	0.3 m^3
Design pressure	max. JCW system pressure
Suggested inlet orifice size	ø10 mm

Other dosing point options, besides the above dosing tank proposal, are available. If the following requirements are met, the expansion tank can be used, for example.

- The expansion tank must be designed as an open tank towards the engine room
- A continuous small jacket water flow is established through the tank. This means that there is a pipe connection from the jacket water pump discharge side via the expansion tank to the suction side of the jacket water pump.

De-aerating tank

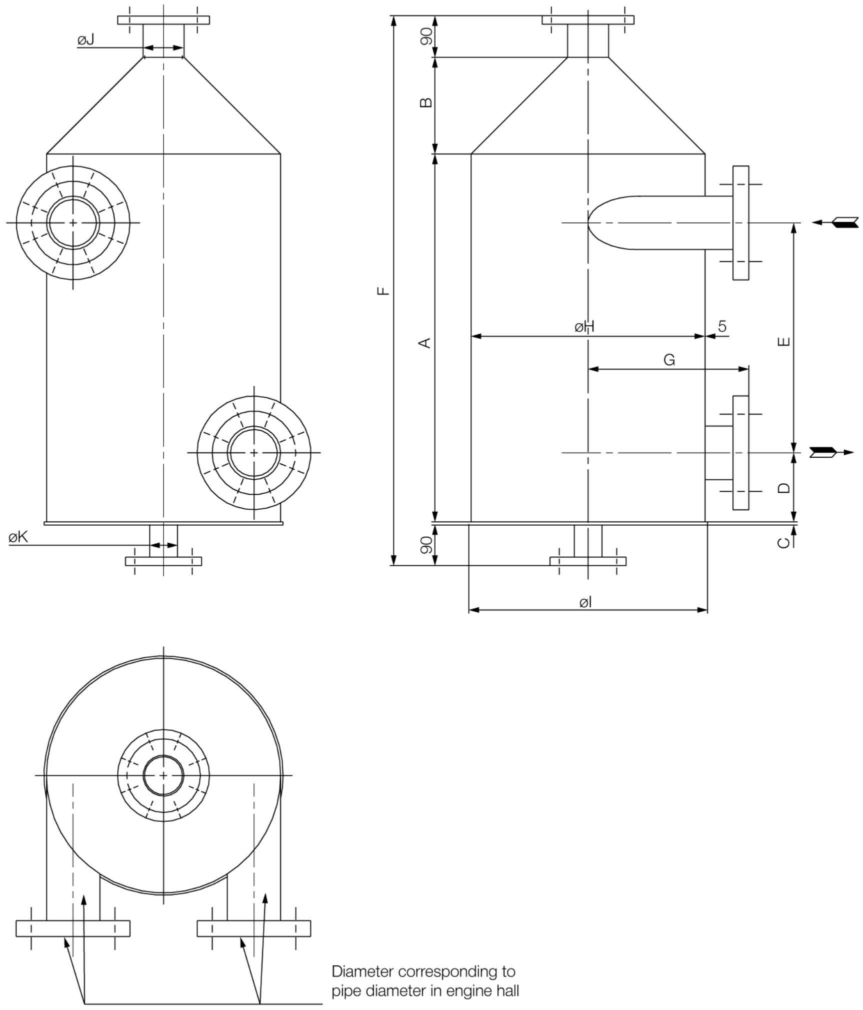

Fig. 12.02.01: De-aerating tank, option: 4 46 640

De-aerating 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
øH	500	800
øI	520	820
øJ	ND 80	ND 100
øK	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.

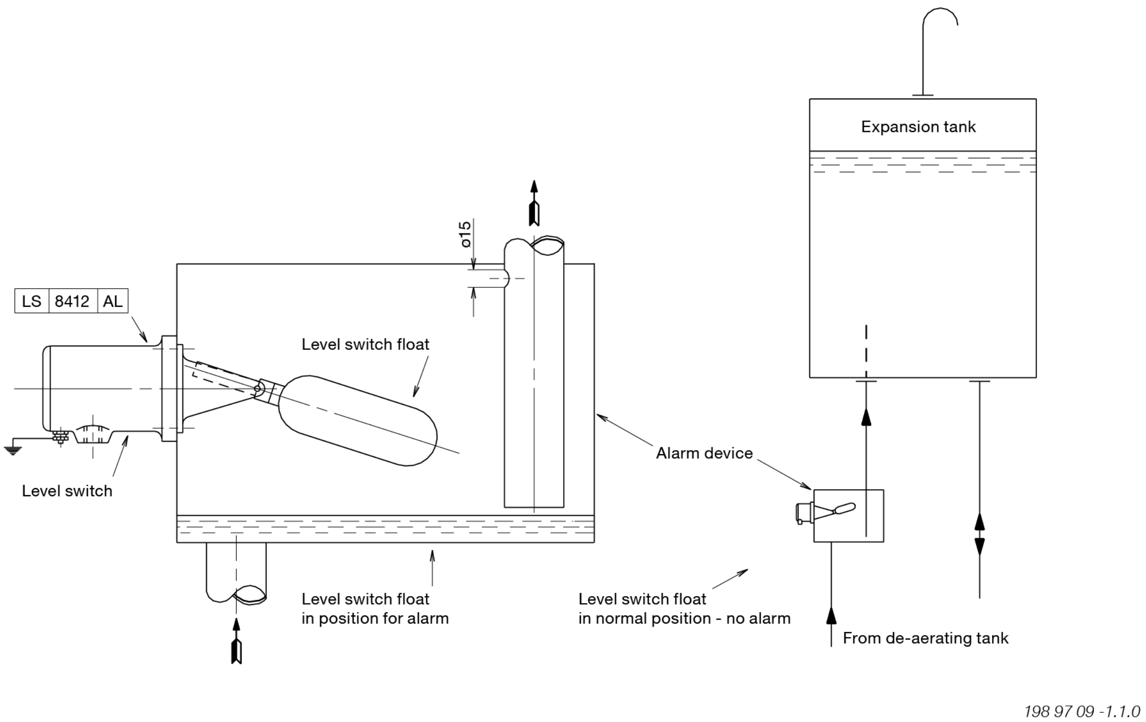

Fig. 12.02.02: De-aerating tank, alarm device, option: 4 46 645

Preheater components

When a preheater system is installed like in Fig. 12.01.01, the components must be specified as follows.

Preheater pump (Optional)

The pump should be of the centrifugal type.

Pump flow rate	10% of the JCW flow, see 'List of capacities'
Working temperature	50-85°C
Pump flow rate	10% of the JCW flow, see 'List of capacities'
Max. working temperature	up to 100°C

Section 6.04 contains a guideline for selcting centrifugal pumps.

The preheater must be relocated if no preheater pump is installed.

Preheater

Heating flow rate	10% of the JCW flow, see 'List of capacities'
Heating capacity	see the note below *)
Preheater type	steam, thermal, oil, or electrical
Working temperature	50-85°C
Max. working temperature	up to 100 °C
Max. pressure drop on jacket water side	~0.2 bar

*) The preheater heating capacity depends on the required preheating time and the required temperature increase of the engine jacket water. Fig. 12.02.03 shows the temperature and time relations. 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 NMCR power.

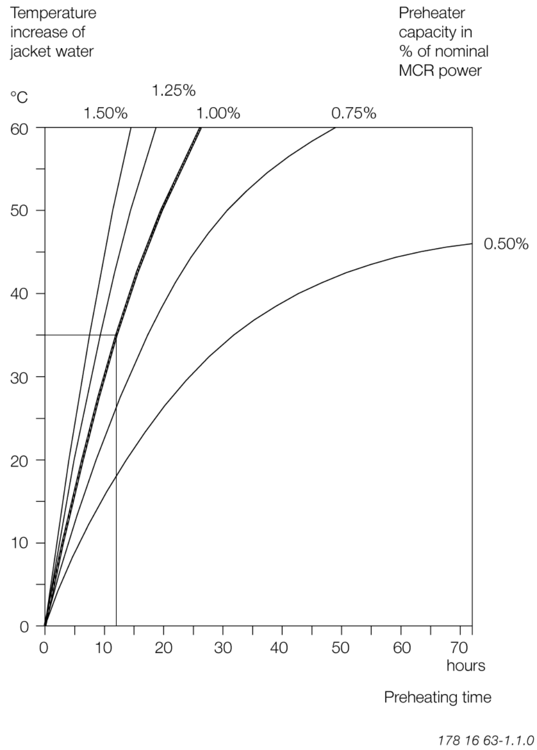

Fig. 12.02.03: Jacket water preheater, example

Freshwater generator installation

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 in 'List of capacities'.

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The reason is that the latter figure is used for dimensioning the jacket water cooler and therefore incorporates a safety margin which can be needed when the engine is operating under conditions such as overload. Normally, this margin is 10% at SMCR.

The calculation of the heat actually available at SMCR for a derated diesel engine can be made in the CEAS application described in Section 20.02.

A freshwater generator installation is shown in Fig. 12.01.01.

Calculation method

When using a normal freshwater generator of the single-effect vacuum evaporator type, the freshwater production (based on the available JCW heat for design purpose, $Q_{d\text{-}jw}$) can, for guidance, be estimated as 0.03 t/24h/kW heat:

$M_{fw} = 0.03 \times Q_{d\text{-}jw}$ t/24h

Where:
M_{fw}: Freshwater production (tonnes per 24 hours)
$Q_{d\text{-}jw} = Q_{jw50\%} \times Tol._{-15\%}$ (kW)

Where
Q_{jw50}[%]: Jacket water heat at 50% SMCR engine load at ISO condition (kW)
$Tol._{-15}$[%]: Minus tolerance of 15% = 0.85

If more heat is utilised than the heat available at 50% SMCR and/or when using the freshwater generator below 50% engine load, a special temperature control system must be incorporated. The purpose is to ensure that the JWC temperature at the outlet from the engine does not fall below a certain level.

Such a temperature control system may consist of a thermostatic three-way valve as shown in Fig. 12.01.01 or a special built-in temperature control in the freshwater generator, e.g. an automatic start/stop function, or similar.

If more heat is utilised than the heat available at 50% SMCR, the freshwater production may for guidance be estimated as:

M_{fw} = 0.03 × Qd-jw t/24h

Where

M_{fw} : Freshwater production (tonness per 24 hours)
$Q_{d\text{-}jw} = Q_{jwNCR} \times Tol._{-15\%}$ (kW)

Where

Q_{jwNCR}: Jacket water heat at NCR engine load at ISO condition (kW)
$Tol._{-15\%}$: Minus tolerance of 15% = 0.85

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12.02 Components

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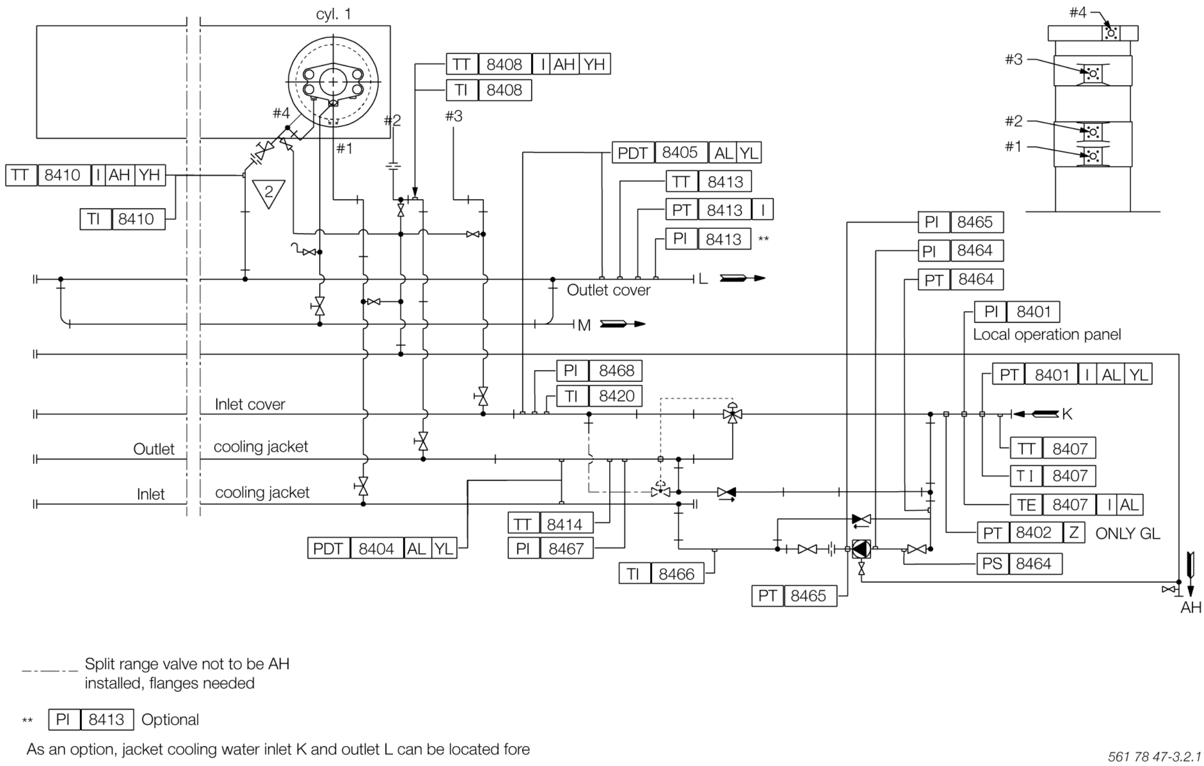
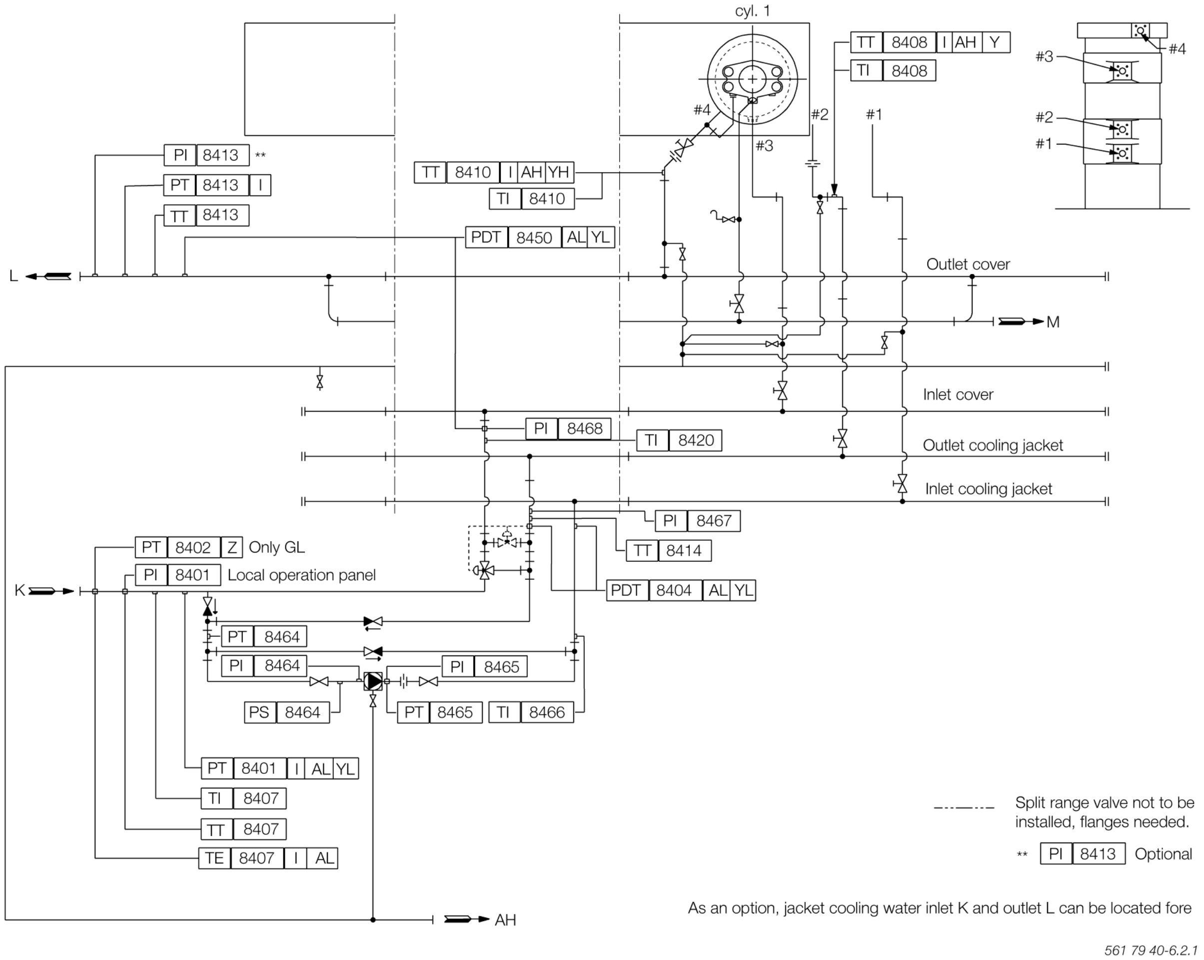
Jacket cooling water pipes



The letters refer to list of 'Counterflanges'
The item no. refer to 'Guidance values automation'

Fig. 12.06.01a: Jacket cooling water pipes, 5-7G80ME-C9/-GI



The letters refer to list of 'Counterflanges'
The item no. refer to 'Guidance values automation'

Fig. 12.06.01b: Jacket cooling water pipes, 8-9G80ME-C9/-GI

Liquid fuel gas vaporisation

A liquid gas vaporising system is normally designed as a closed glycol circulation system. It basically consists of a glycol circulation pump, a glycol heat exchanger and a gas vaporiser. The glycol system is considered as part of the fuel gas supply system.

Most likely the heat available from the jacket water system is used for the glycol heat exchanger because this energy is free. Other heat options are also available such as steam and thermal oil.

When utilising the heat in the jacket water cooling system, the following should be noted:

- A backup heat exchanger shall be incorporated in either the glycol system or the jacket water system when the jacket water heat is used as the main heat source. The purpose is to ensure that the high-pressure gas temperature will be higher than the minimum required inlet temperature to the engine during operation under very low load and during engine load change (increasing the engine load)
- In the jacket water system showed in Fig. 12.04.01, the backup heat exchanger is located in the glycol circuit. Where as in Fig. 12.04.02, the backup heat exchanger is located in the jacket water system as a preheater/ hot water loop heater.
- Utilising both the jacket water heat for gas vaporization and freshwater production, special attendance has to be made when calculating the freshwater production. The calculation shall be based on the excess of heat available after the necessary heat for gas vaporization has been used.
- The actual available heat in the jacket water system is lower than indicated by the heat dissipation figures given in the 'List of capacities'. The reason is that the latter figure is used for dimensioning the jacket water cooler.

  The figure therefore incorporates a safety margin which could be needed when the engine is operating under conditions such as for instance overload. Normally, this margin is 10% at SMCR
- The calculation of the heat actually available at SMCR and part loads for a derated diesel engine can be made in the CEAS application described in Section 20.02

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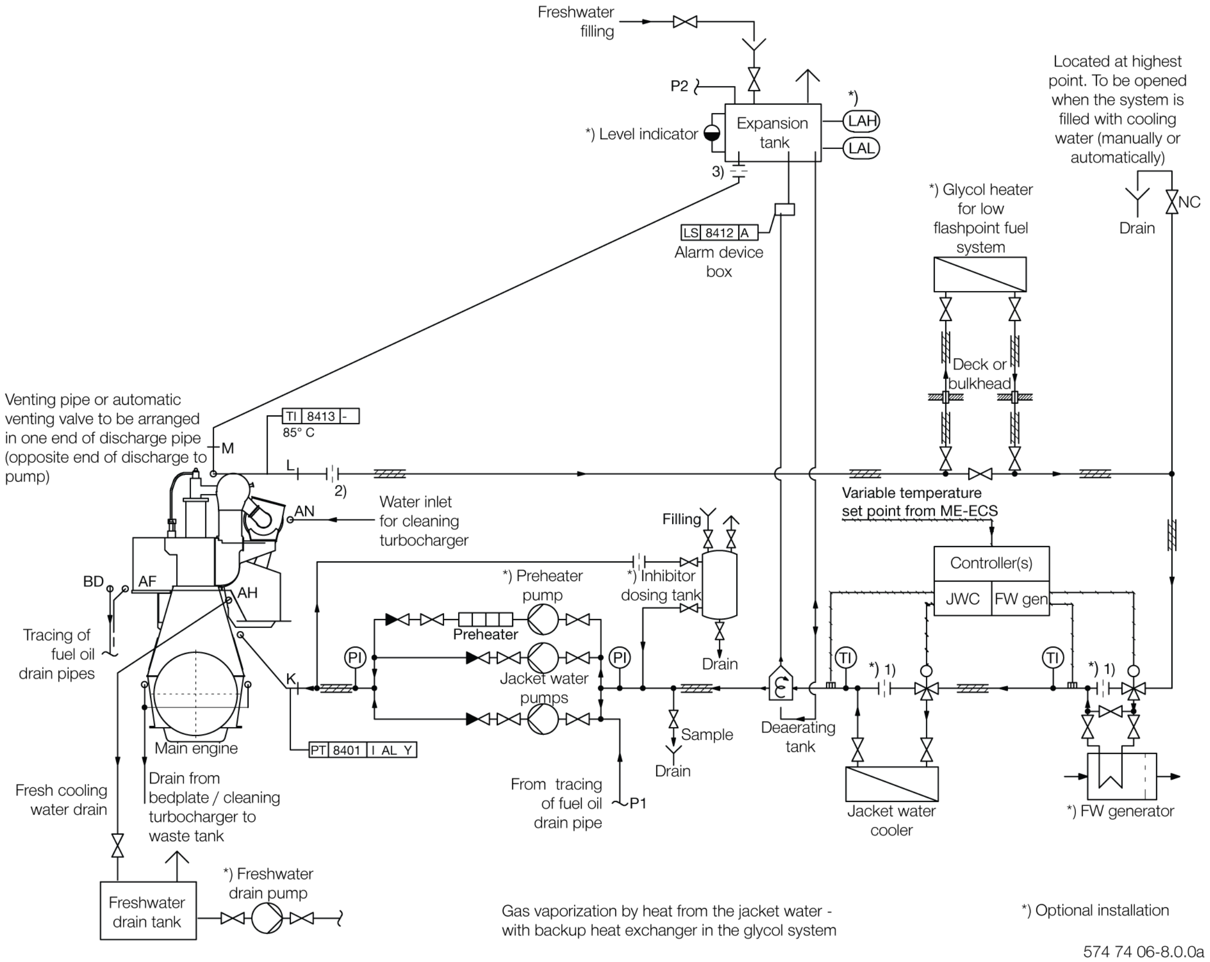
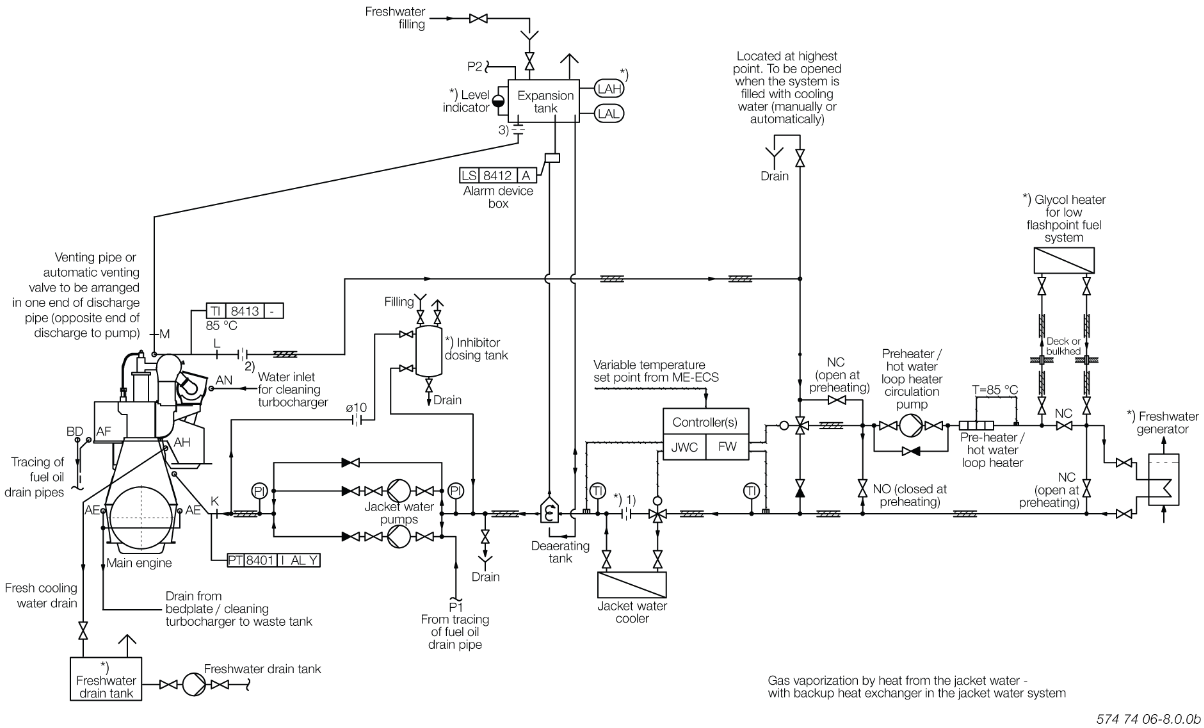


1) (Optional) Orifices (or lockable adjustable valves) to be installed in order to create a differential pressure indentical to that of the jacket water cooler / freshwater generator at nominal jacket water pump capacity.

2) (Optional) Orifices (or lockable adjustable valves to be installed in order to create a min. inlet pressure indicated at sensor PT 8401 above the min. pressure stated in the Guidance Values Automation (GVA) at engine inlet connection “K” and besides ensuring a correct flow from the jacket water pump.

3) Orifices with small size hole to be installed for avoiding jacket water flow through the expansion tank.

Fig. 12.04.01: Gas vaporization by heat from the jacket water – with backup heat exchanger in the glycol system



1) (Optional) Orifices (or lockable adjustable valves) to be installed in order to create a differential pressure indentical to that of the jacket water cooler / freshwater generator at nominal jacket water pump capacity.

2) (Optional) Orifices (or lockable adjustable valves to be installed in order to create a min. inlet pressure indicated at sensor PT 8401 above the min. pressure stated in the Guidance Values Automation (GVA) at engine inlet connection “K” and besides ensuring a correct flow from the jacket water pump.

3) Orifices with small size hole to be installed for avoiding jacket water flow through the expansion tank.

Fig. 12.04.02: Gas vaporization by heat from the jacket water – with backup heat exchanger in the jacket water system

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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 approx. 7 bar to ‘AP’ for turbocharger cleaning (soft blast), and a minor volume used for the fuel valve testing unit. The specific air pressure required for turbocharger cleaning is subject to make and type of turbocharger.

Through a reduction valve, compressed air is supplied at 1.5 bar to the leakage detection and ventilation system for the double-wall gas piping.

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 Energy Solutions 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 [man-es.com/](man-es.com/) --> 'Two-Stroke --> 'Technical Papers'.

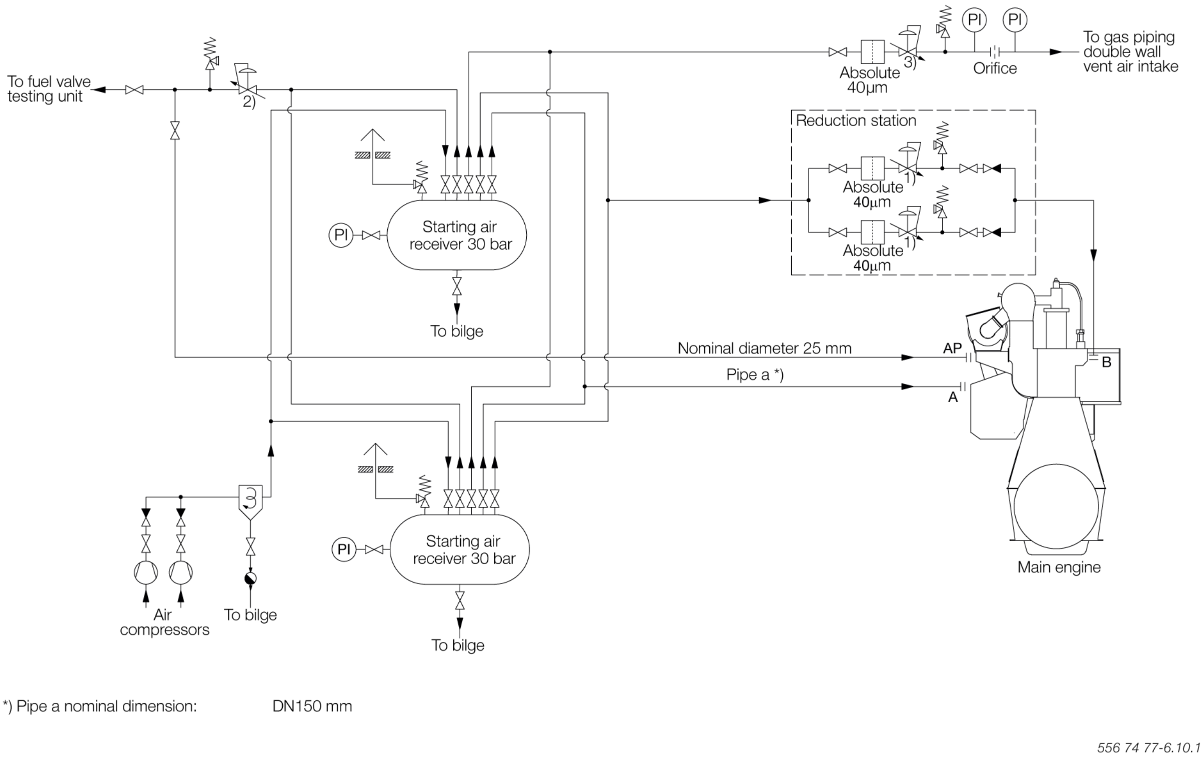

Fig. 13.01.01: Starting and control air systems

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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’
Nonreversible 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’ *)
Nonreversible 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 ± 10%)

Flow rate, free air .... 2,100 Normal liters/min equal to 0.035 m^3/s

Filter, fineness .... 40 µm

Reduction Valve for Turbocharger Cleaning etc

Reduction............from 30-10 bar to approx. 7 bar *)
*) Subject to make and type of TC (Tolerance ±10%)

Flow rate, free air.............2,600 Normal liters/min equal to 0.043 m^3/s

Reduction Valve for Venting Air for Gas Piping

Reduction .... from 30-10 bar to 1.5 bar (Tolerance ±10%)

Flow rate, free air .... 1,000 Normal liters/min equal to 0.015 m^3/s

The consumption of compressed air for control air, exhaust valve air springs and safety air as well as air for turbocharger cleaning, fuel valve testing and venting of gas piping 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.

Piping

The starting air pipes, Fig. 13.03.01, contain a main starting valve (a ball valve with actuator), a nonreturn 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, EoD: 4 50 141, is included in the basic design.

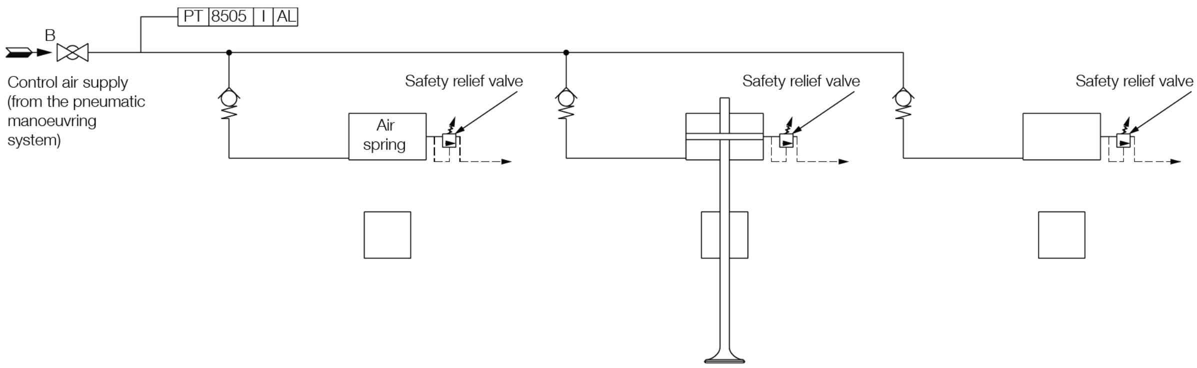
The Engine Control System regulates the supply of control air to the starting valves in accordance with the correct firing sequence and the timing.

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 our publication:

Uni-concept Auxiliary Systems for Two-Stroke Main Engines and Four-Stroke Auxiliary Engines

The publication is available at man-es.com/marine --> 'Two-Stroke' --> 'Technical Papers'.



The item nos. refer to ‘Guidance values automation’
The piping is delivered with and fitted onto the engine

517 15 68-2.1.1

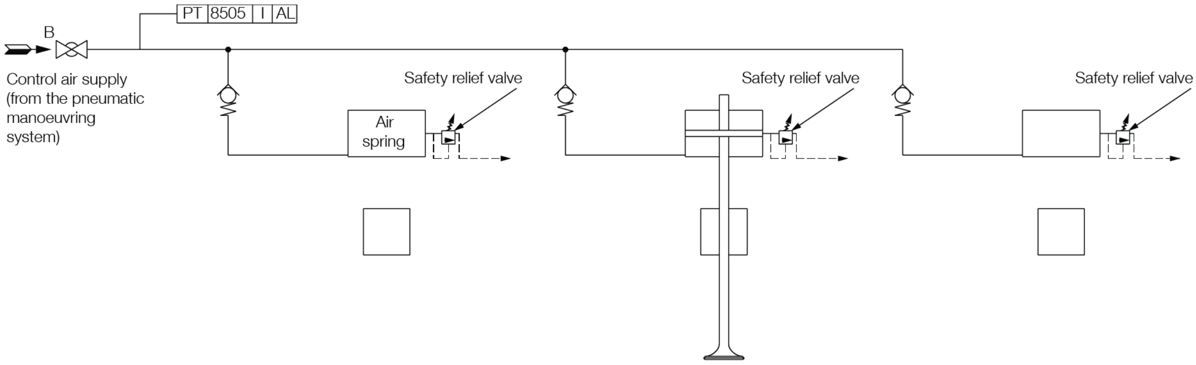
Fig. 13.03.01: Starting air pipes

Exhaust Valve Air Spring Pipes

The exhaust valve is opened hydraulically by the Proportional Exhaust Valve Actuator (PEVA) valve, which is activated by the Engine Control System.

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

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The item nos. refer to 'Guidance values automation'
The piping is delivered with and fitted onto the engine

Fig. 13.03.02: Air spring pipes for exhaust valves

Electric Motor for Turning Gear

General

MAN Energy Solutions delivers a turning gear with built-in disc brake, option 4 80 101.

A turning gear with an electric motor of another protection or insulation class can be ordered, option 4 80 103. Information about the alternative executions is available on request.

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 440-480V
Electric motor and brake, frequency	60 Hz
Protection, electric motor and brake	IP 44
Insulation class	F

Number of cylinders	Electric motor	
	Nominal power, kW	Nominal current, A
6-9	9	13.4

Electric motor and brake, voltage	3 x 380-415V
Electric motor and brake, frequency	50 Hz
Protection, electric motor and brake	IP 44
Insulation class	F

Number of cylinders	Electric motor	
	Nominal power, kW	Nominal current, A
6-9	7.5	15.5

533 12 49-6.3.0

Table 13.04.01: Electric motor for turning gear, option: 4 80 103

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Scavenge Air System

Scavenge air is supplied to the engine by one 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.

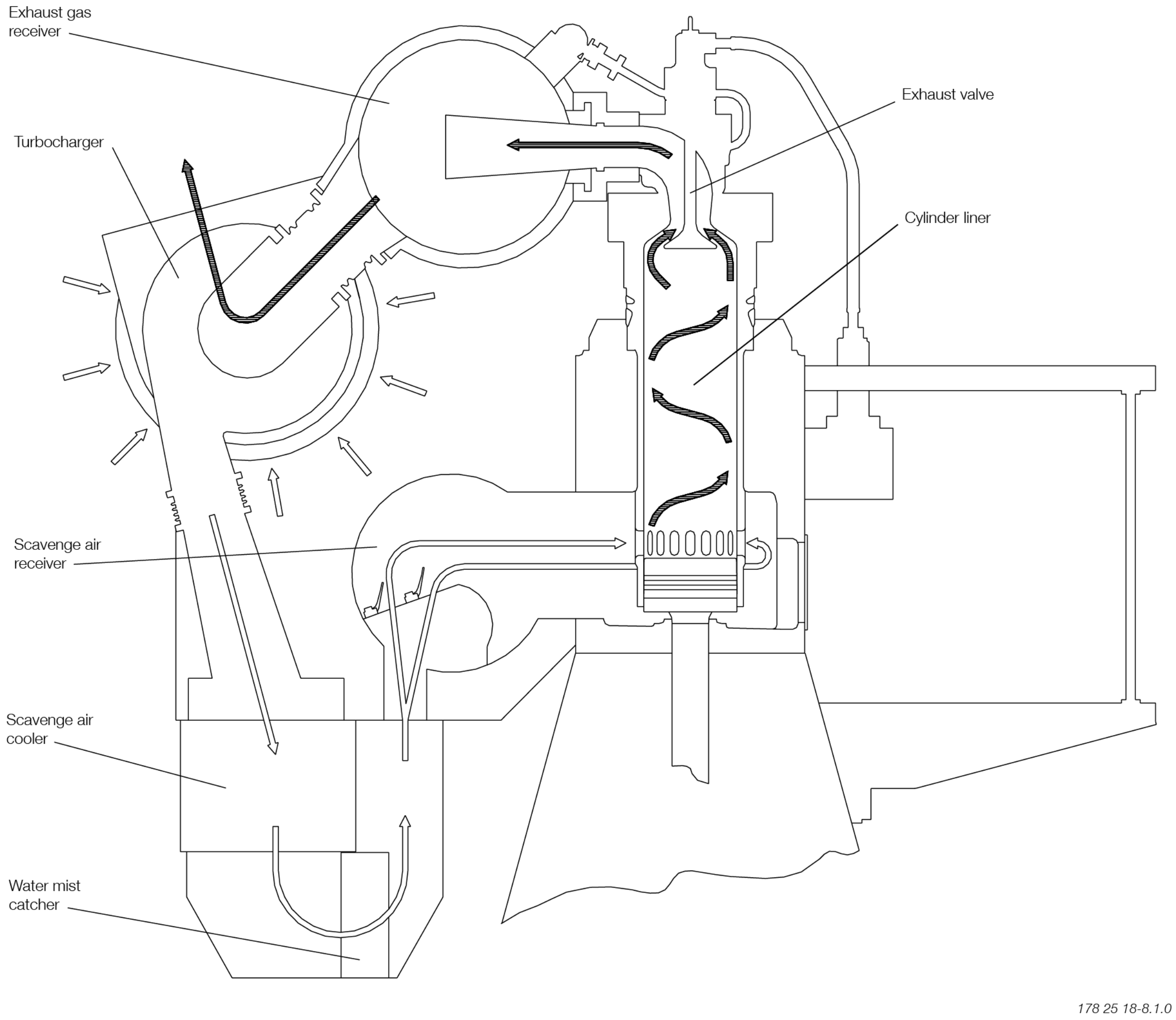

Fig. 14.01.01: Scavenge Air System

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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 integrated in the reversing chamber below the scavenge air cooler. 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

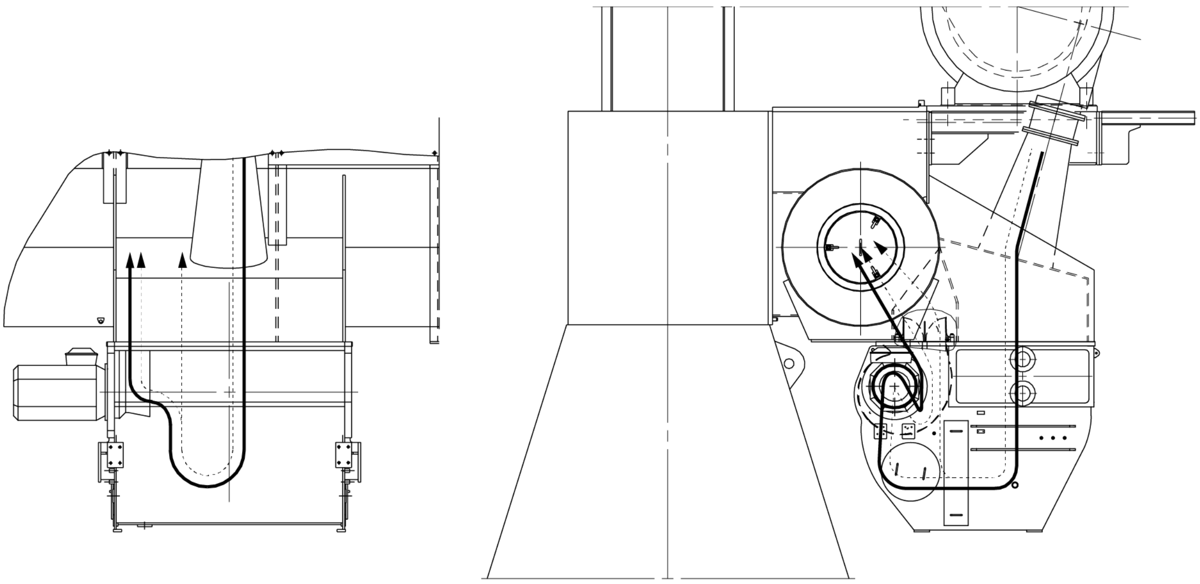
The auxiliary blowers start operating consecutively before the engine is started and will ensure complete scavenging of the cylinders in the starting phase, thus providing the best conditions for a safe start.

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.



Running with auxiliary blower ------- Running with turbocharger

178 63 77-1.0.0a

Fig. 14.02.01: Scavenge air system, integrated blower

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 30 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 Energy Solutions'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: *MAN Energy Solutions Influence of Ambient Temperature Conditions*

The publication is available at man-es.com/marine → 'Two-Stroke' → 'Technical Papers'.

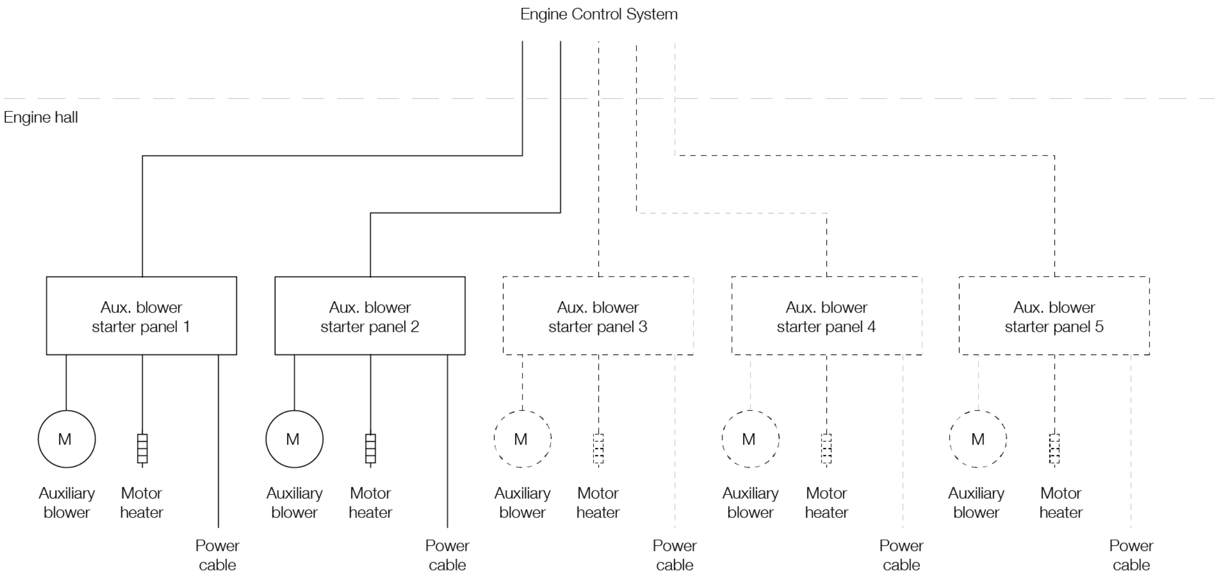


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Fig. 14.02.02: Diagram of auxiliary blower control system

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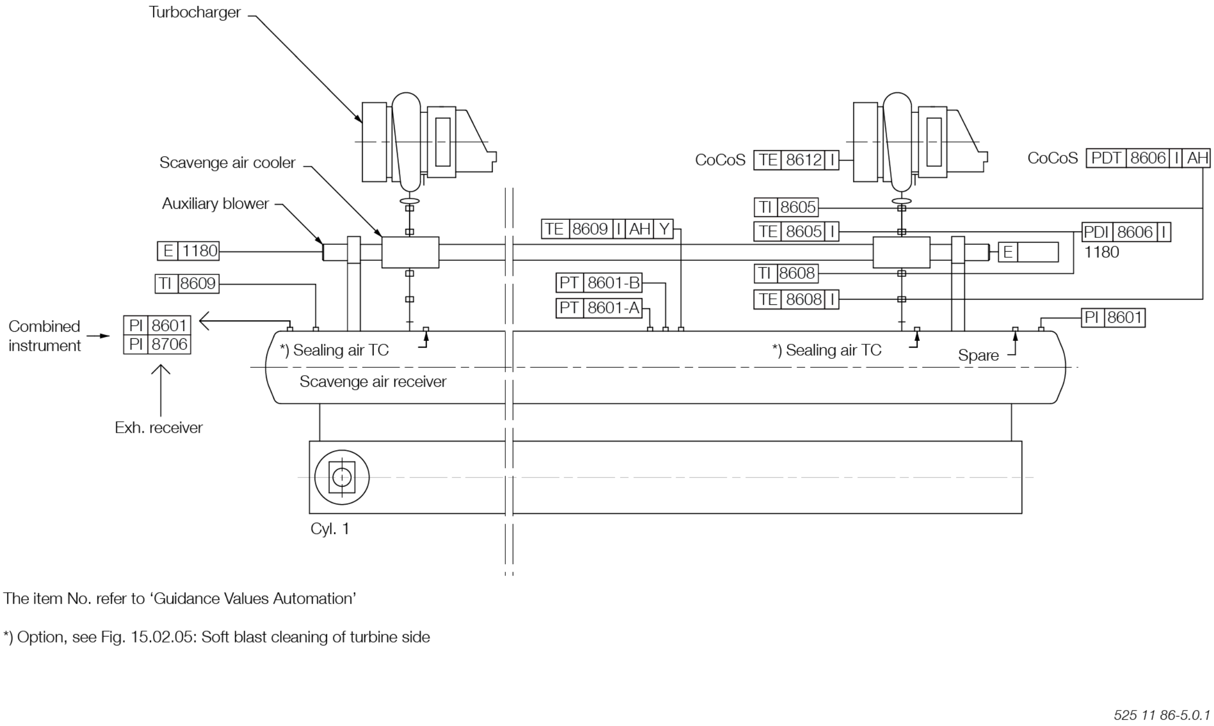
Scavenge air pipes



The item No. refer to 'Guidance Values Automation'

*) Option, see Fig. 15.02.05: Soft blast cleaning of turbine side

Fig. 14.03.01: Scavenge air pipes

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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.

Motor start method and size

Direct Online Start (DOL) is required for all auxiliary blower electric motors to ensure proper operation under all conditions.

For typical engine configurations, the installed size of the electric motors for auxiliary blowers are listed in Table 14.04.01.

Special operating conditions

For engines with Dynamic Positioning (DP) mode in manoeuvring system, option: 4 06 111, larger electric motors are required. This is in order to avoid start and stop of the blowers inside the load range specified for dynamic positioning. The actual load range is to be decided between the owner and the yard.

Engine plants with waste heat recovery exhaust gas bypass and engines with low-and part-load exhaust gas bypass may require less blower capacity, please contact MAN Energy Solutions, Copenhagen.

Number of cylinder	Number of turbochargers	Number of auxiliary blowers	Installed power/ blower kW
6	1	2	90
6	2	2	90
7	2	2	105
8	2	2	125
9	3	3	90

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

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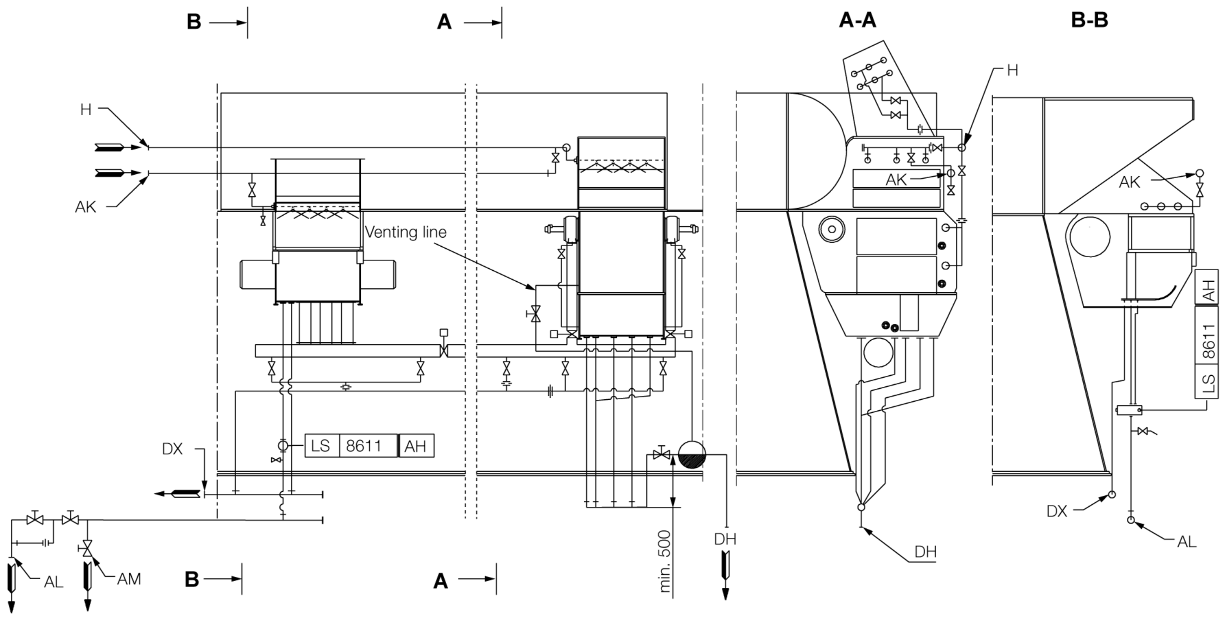
Scavenge air cooler cleaning system

General



The letters refer to list of 'Counterflanges'. The item nos. refer to 'Guidance values automation'.

Fig. 14.05.01: Air cooler cleaning pipes, two or more air coolers, for EGR

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.

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.

The system is equipped with a drain box with a level switch, indicating any excessive water level.

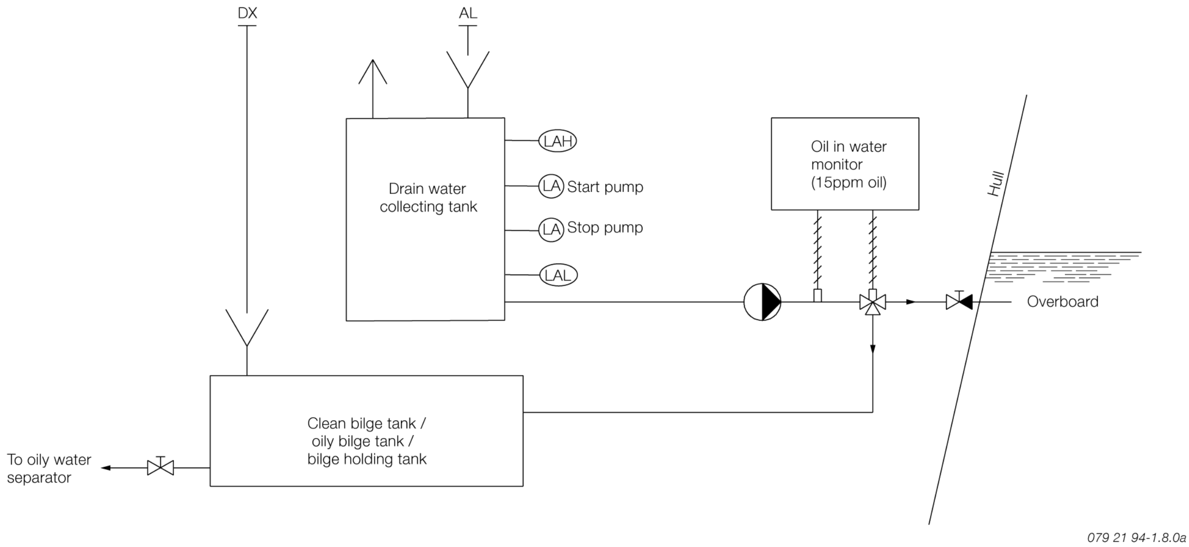
The piping delivered with and fitted on the engine is shown in Fig 14.05.01.

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.

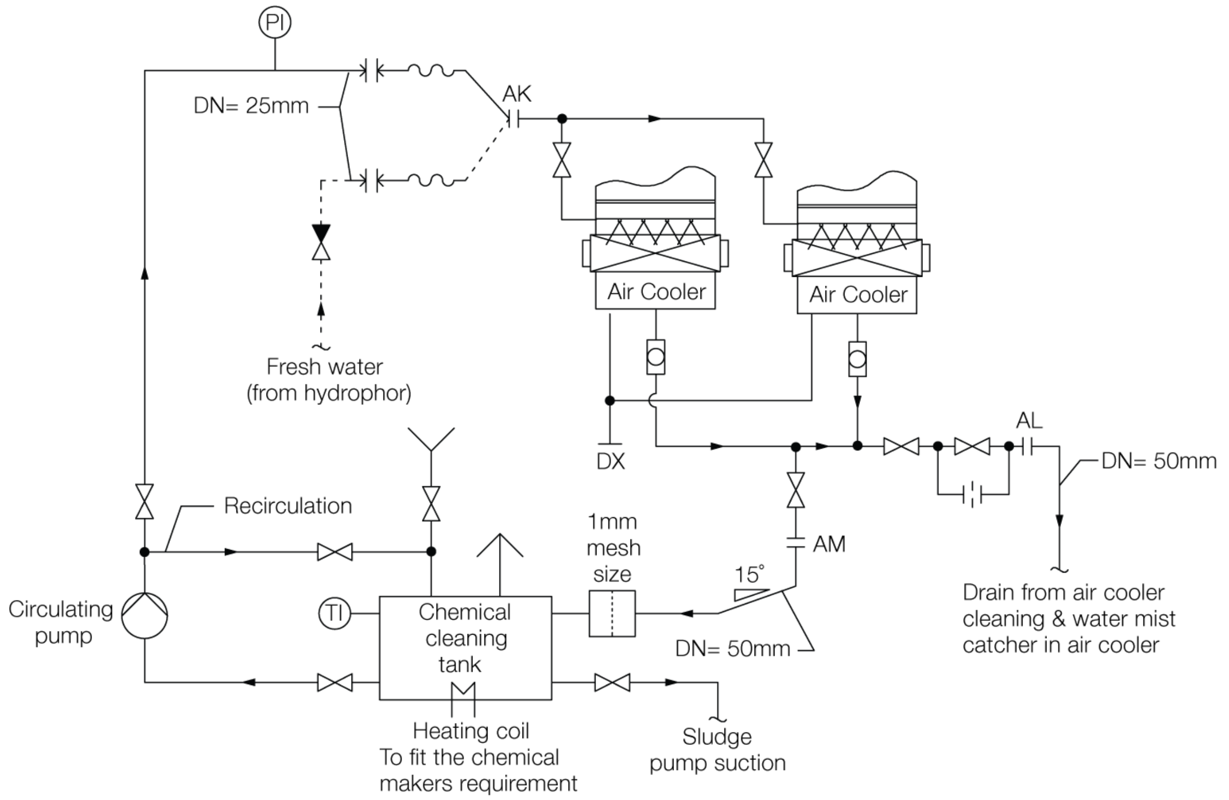
198 76 84-9.2



The letters refer to list of 'Counterflanges'.

Fig. 14.05.02: Suggested automatic disposal of drain water, if required by owner (not a demand from MAN Energy Solutions)

Air cooler cleaning unit



Engine type	No. of cylinders	Chemical tank capacity, m^3	Circulation pump capacity at 3 bar, m^3/h
G60ME-C	5	0.3	1
	6-8	0.6	2
S60ME-C	5	0.3	1
	6-8	0.6	2
G70ME-C	5-6	0.6	2
S70ME-C	5-7	0.6	2
	8	0.9	3
G80ME-C	6-7	0.6	2
	8-9	0.9	3
G90ME-C	6-8	0.9	3
	9-12	1.5	5
G95ME-C	6-8	0.9	3
	9-12	1.5	5

Fig. 14.05.03: Air cooler cleaning system with air cooler cleaning unit, option: 4 55 665

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Scavenge air box drain system

General

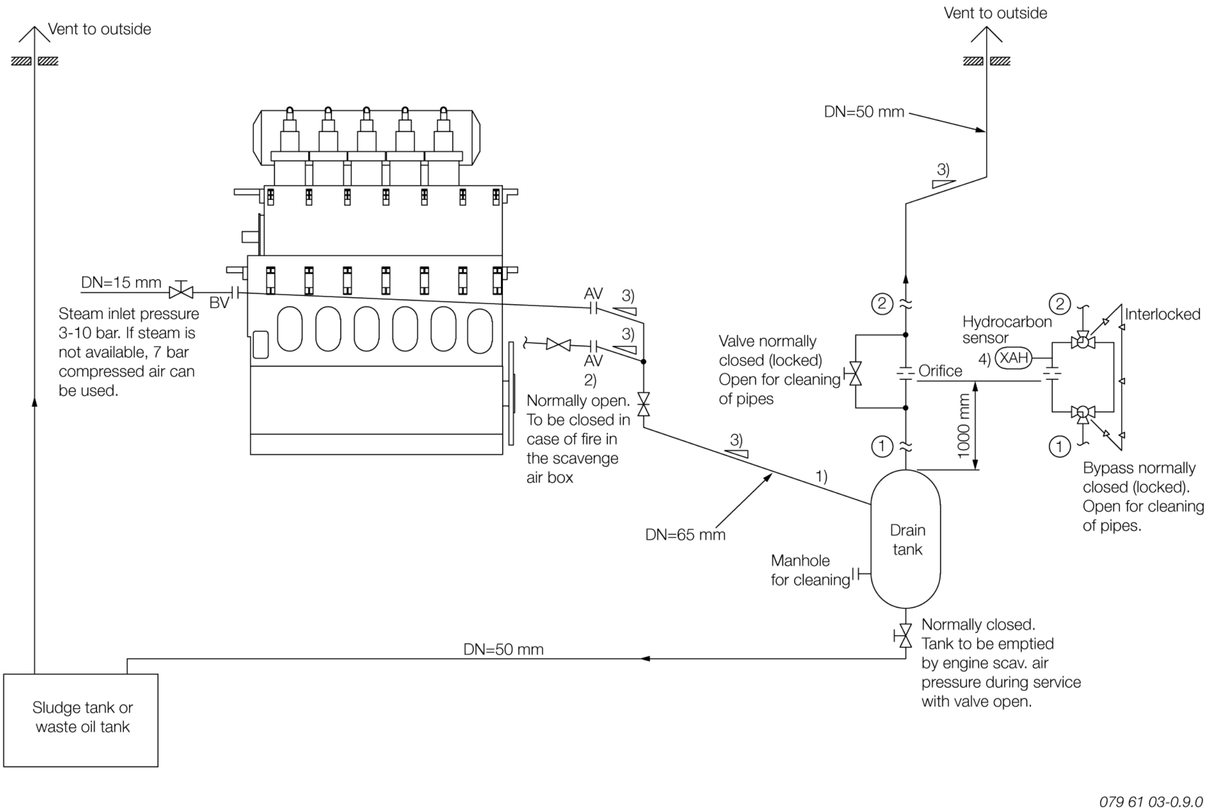
The scavenge air box is continuously drained through 'AV' to a small pressurised drain tank, from where the sludge is led to the sludge tank. Steam can be applied through 'BV', if required, to facilitate the draining. See Fig. 14.06.01.

The continuous drain from the engine scavenge air area must not be directly connected to the sludge tank due to the pressure level.

The drain tank shall be designed according to the pressurised system connected to the BV connection as one of the following:

- Steam maximum working pressure
- Compressed air maximum working pressure

It is recommended that the drain tank is placed close to the engine to avoid lon piping between engine and drain tank and thereby minimize the risk of the pipe being blocked by sludge.



The letters refer to list of 'Counterflanges'

1) Drain line
2) AV1 connection
3) Vent pipe routing
4) Hydrocarbon sensor and special vent outlet location

No. of cylinders:	5-6	7-9	10-12
Drain tank capacity, m^3	0.8	1.1	1.5

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Fig. 14.06.01: Scavenge air box drain system

Fire Extinguishing Systems for Scavenge Air Space

General

Fire in the scavenge air space can be extinguished by steam, this being the basic solution, or, optionally, by water mist or CO_2.

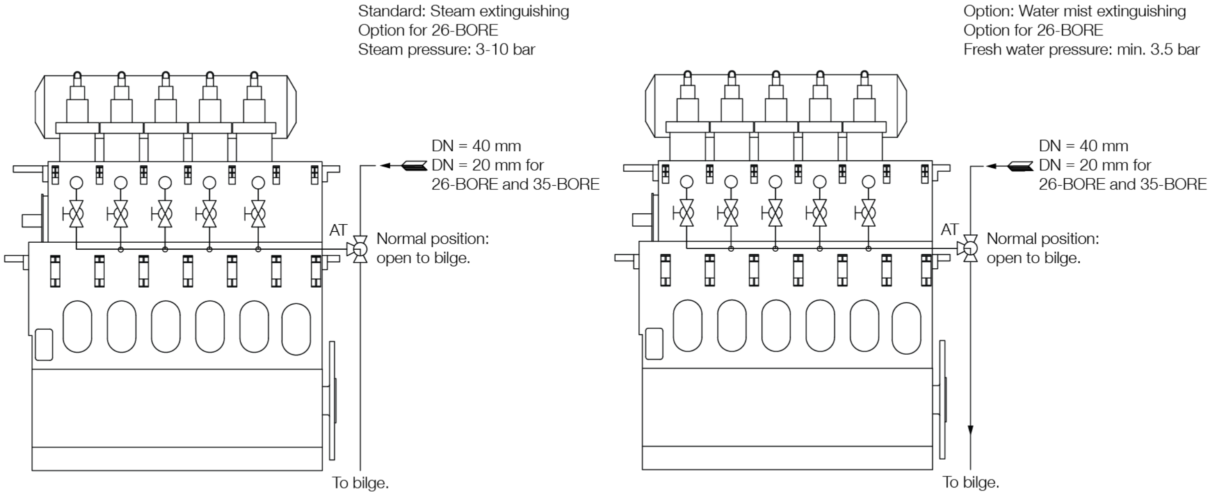
The external system, pipe and flange connections are shown in Fig. 14.07.01 and the piping fitted onto the engine in Fig. 14.07.02.

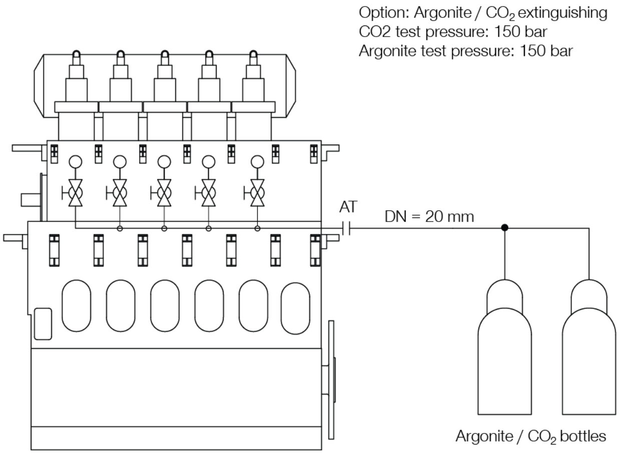
In the Extent of Delivery, the fire extinguishing system for scavenge air space is selected by the fire extinguishing agent:

- basic solution: 4 55 140 Steam
- option: 4 55 142 Water mist
- option: 4 55 143 CO_2

The key specifications of the fire extinguishing agents are:

Steam fire extinguishing for scavenge air space	
Steam pressure:	3-10 bar
Steam quantity, approx.:	5.8 kg/cyl.
Water mist fire extinguishing for scavenge air space	
Freshwater pressure:	min. 3.5 bar
Freshwater quantity, approx.:	4.7 kg/cyl.
CO_2 fire extinguishing for scavenge air space	
CO_2 test pressure:	150 bar
CO_2 quantity, approx.:	11.07 kg/cyl.





CO_2

At least two bottles ought to be installed.
In most cases, one bottle should be sufficient to extinguish fire in three cylilnders, 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.

Argonite

At least two bottles ought to be installed.
One as spare.

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.

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Fig. 14.07.01: Fire extinguishing system for scavenge air space

Scavenge air space, drain pipes

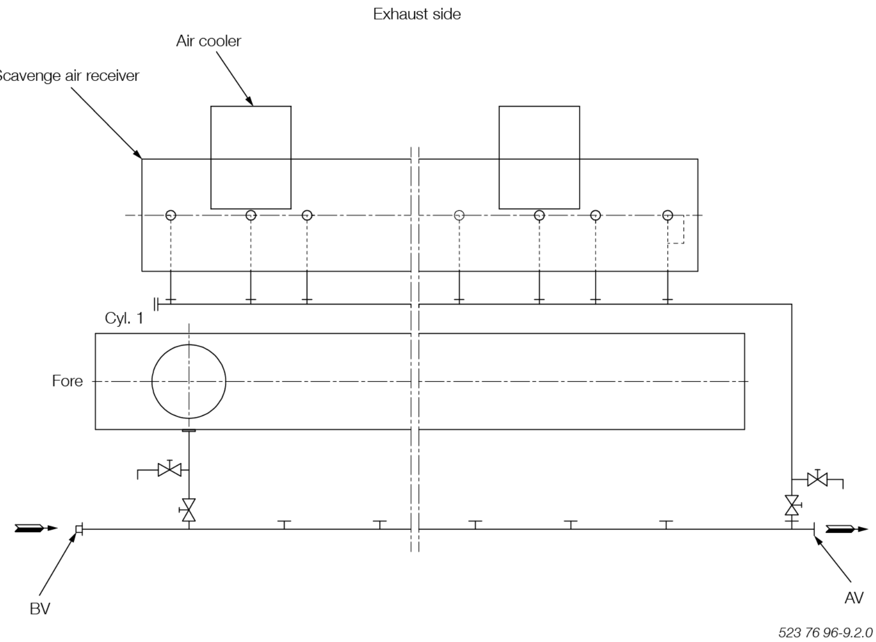

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Fig. 14.07.03: Scavenge air space, drain pipes

Fire extinguishing pipes in scavenge air space

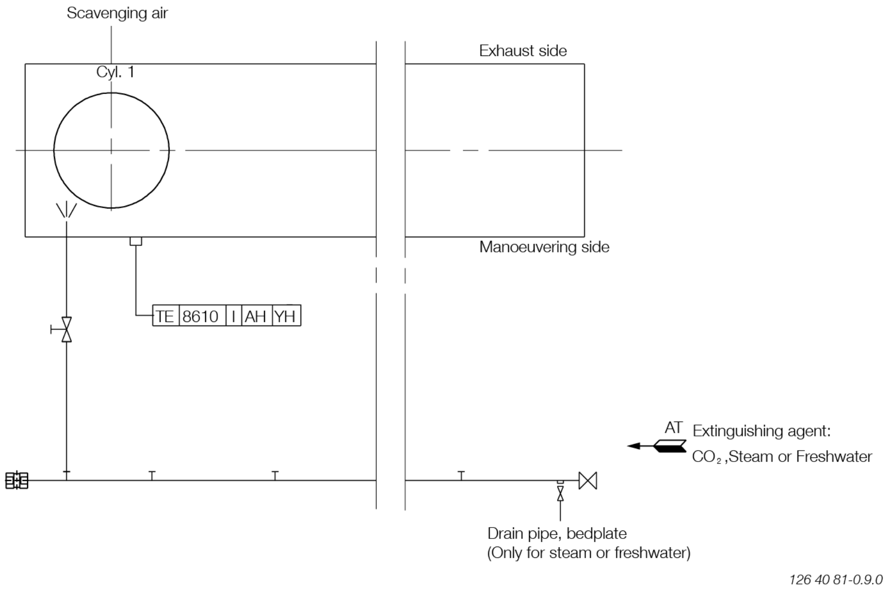

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Fig. 14.07.02: Fire extinguishing pipes in scavenge air space

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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 System

The turbochargers are located on the exhaust side of the engine.

The engine is designed for the installation of the MAN turbocharger type TCA, option: 4 59 101, Accelleron turbocharger type A-L, 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 turbochargers, though not for dual fuel engines.

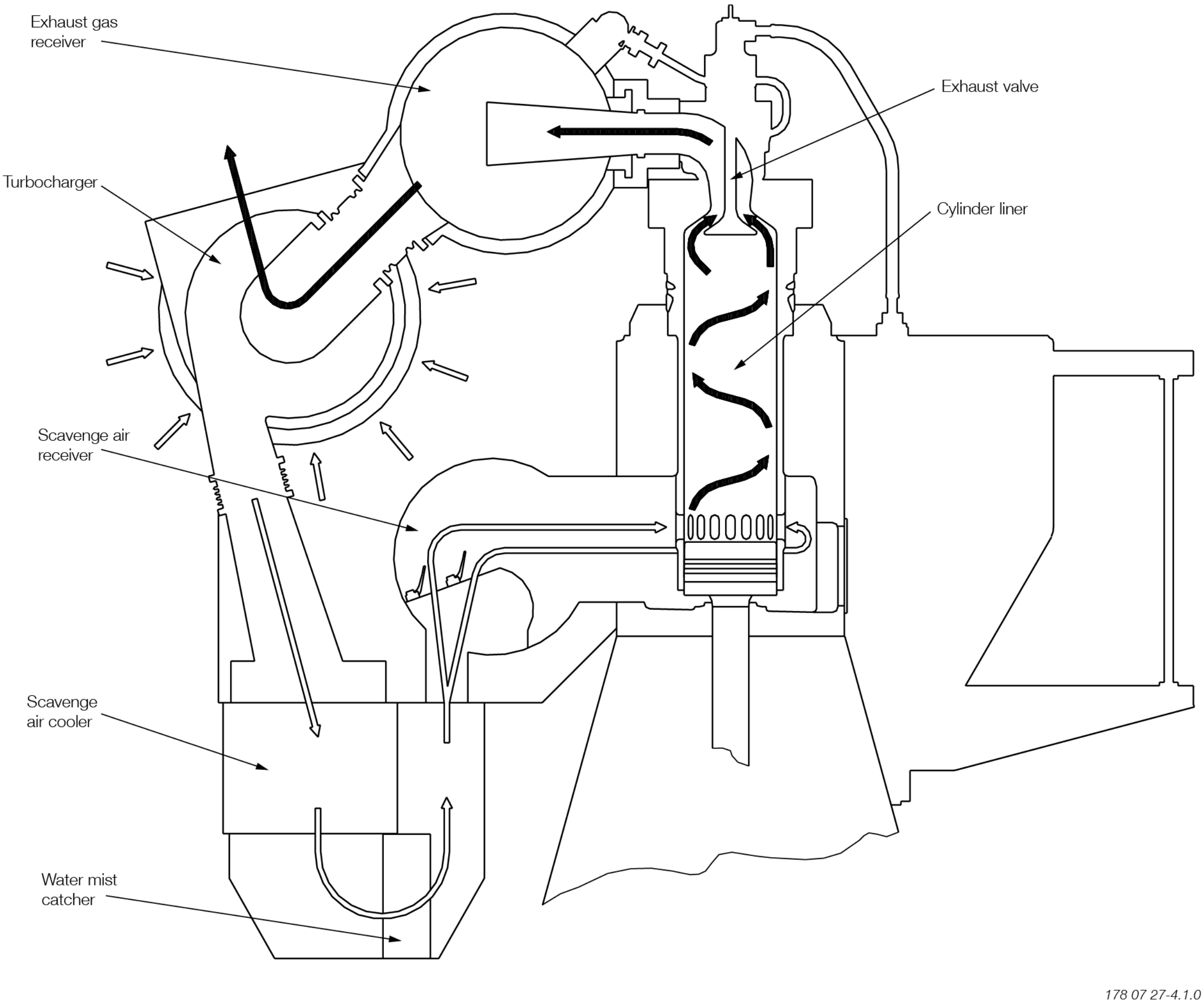
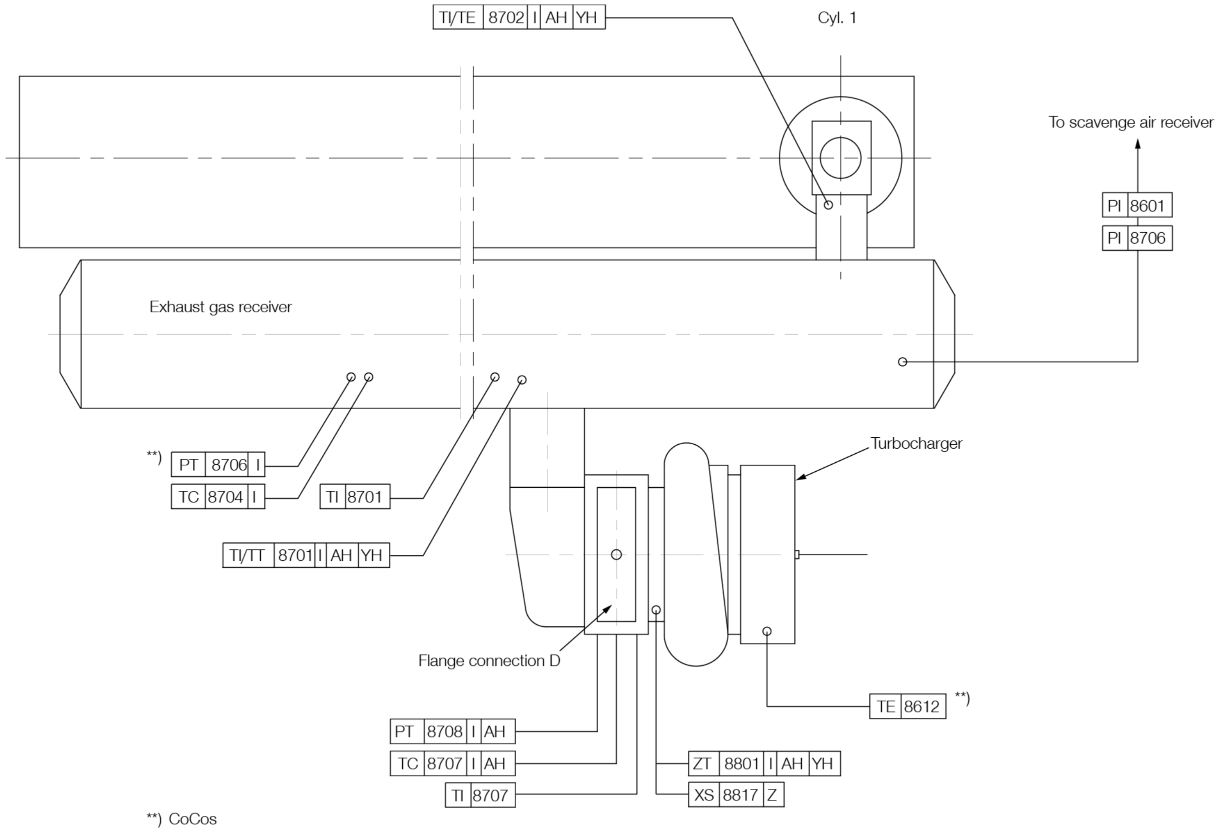


Fig. 15.01.01: Exhaust gas system on engine

Piping and cleaning systems



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The item nos. refer to 'Guidance Values Automation'

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Fig. 15.02.01: Exhaust gas pipes

Cleaning Systems

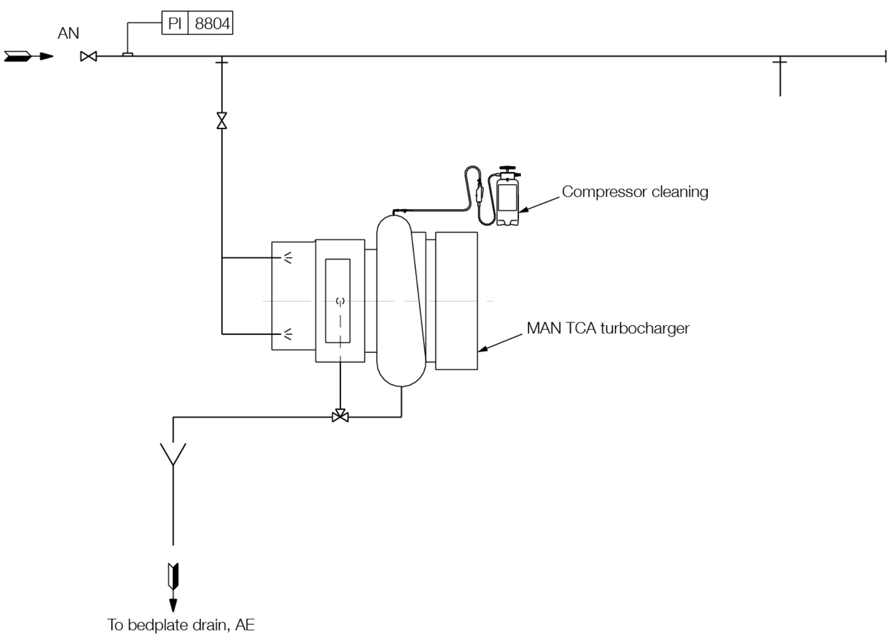

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Fig. 15.02.02: MAN TCA turbocharger, water washing of turbine side

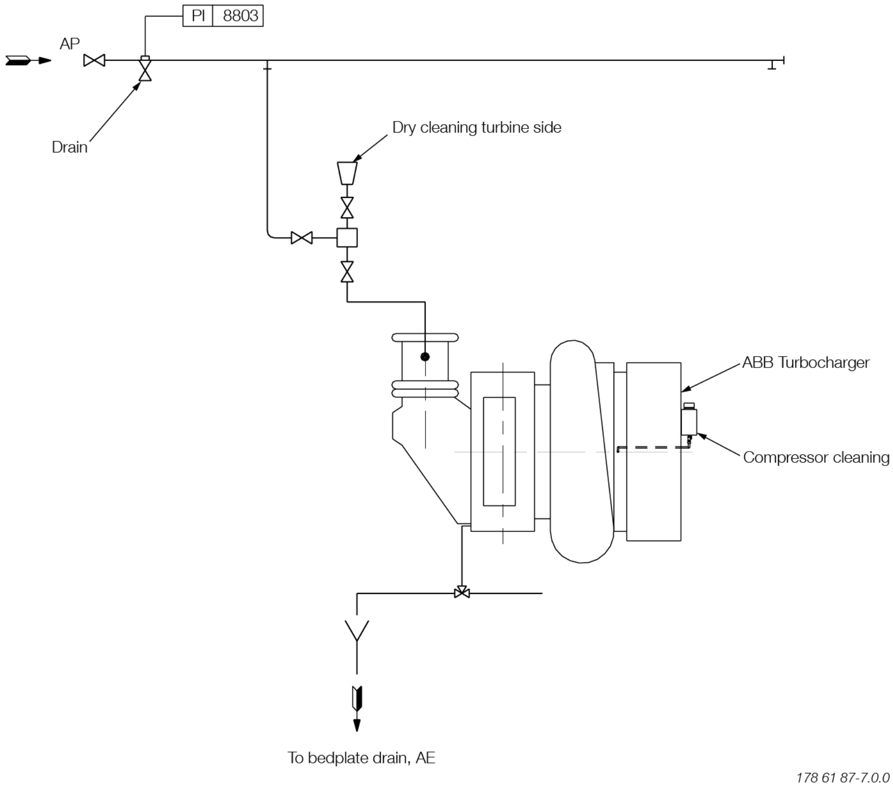

Fig. 15.02.03: Soft blast cleaning of turbine side and water washing of compressor side for ABB turbochargers

Soft Blast Cleaning Systems

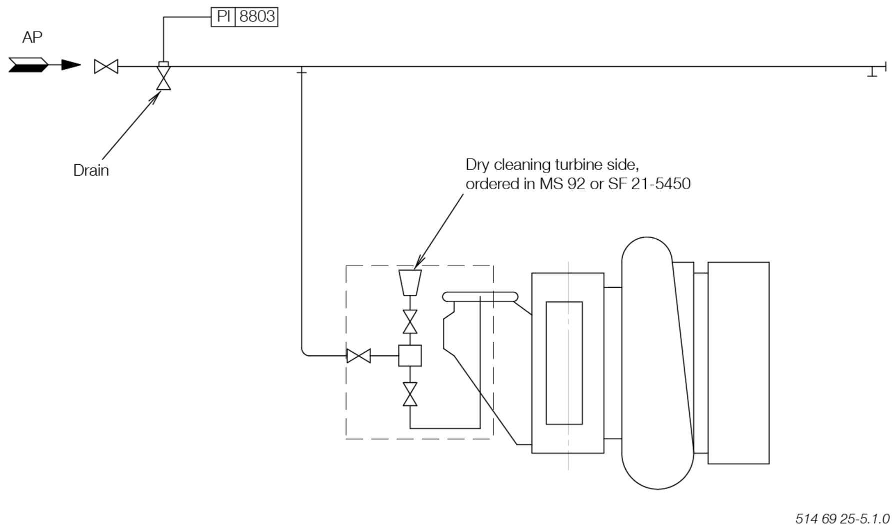

Fig. 15.02.04: Soft blast cleaning of turbine side, basic

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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.

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System components

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 15.06.01 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 15.06.02. The orientation of the maximum permissible forces and moments on the gas outlet flange of the turbocharger is shown in Fig. 15.06.02.

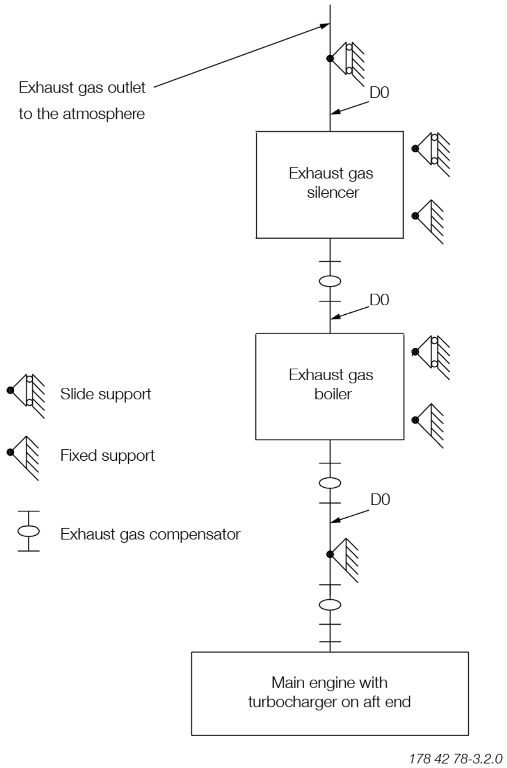

Fig. 15.04.01a: Exhaust gas system, one turbocharger

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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.

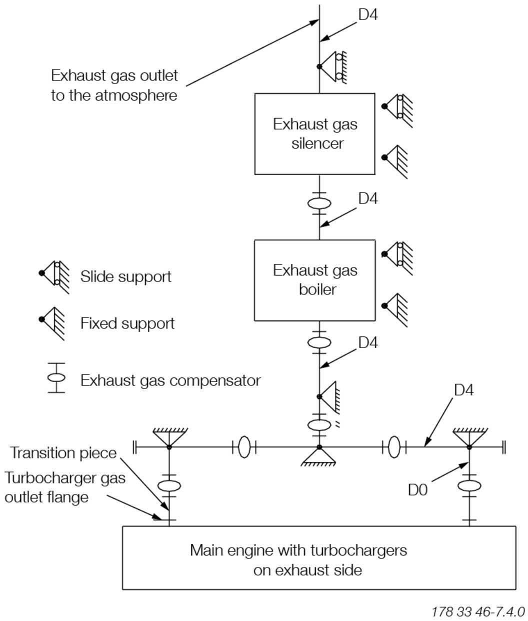

Fig. 15.04.01b: Exhaust gas system, two or more TCs

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Exhaust gas silencer with L_1 at 72 rpm

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 meter 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.

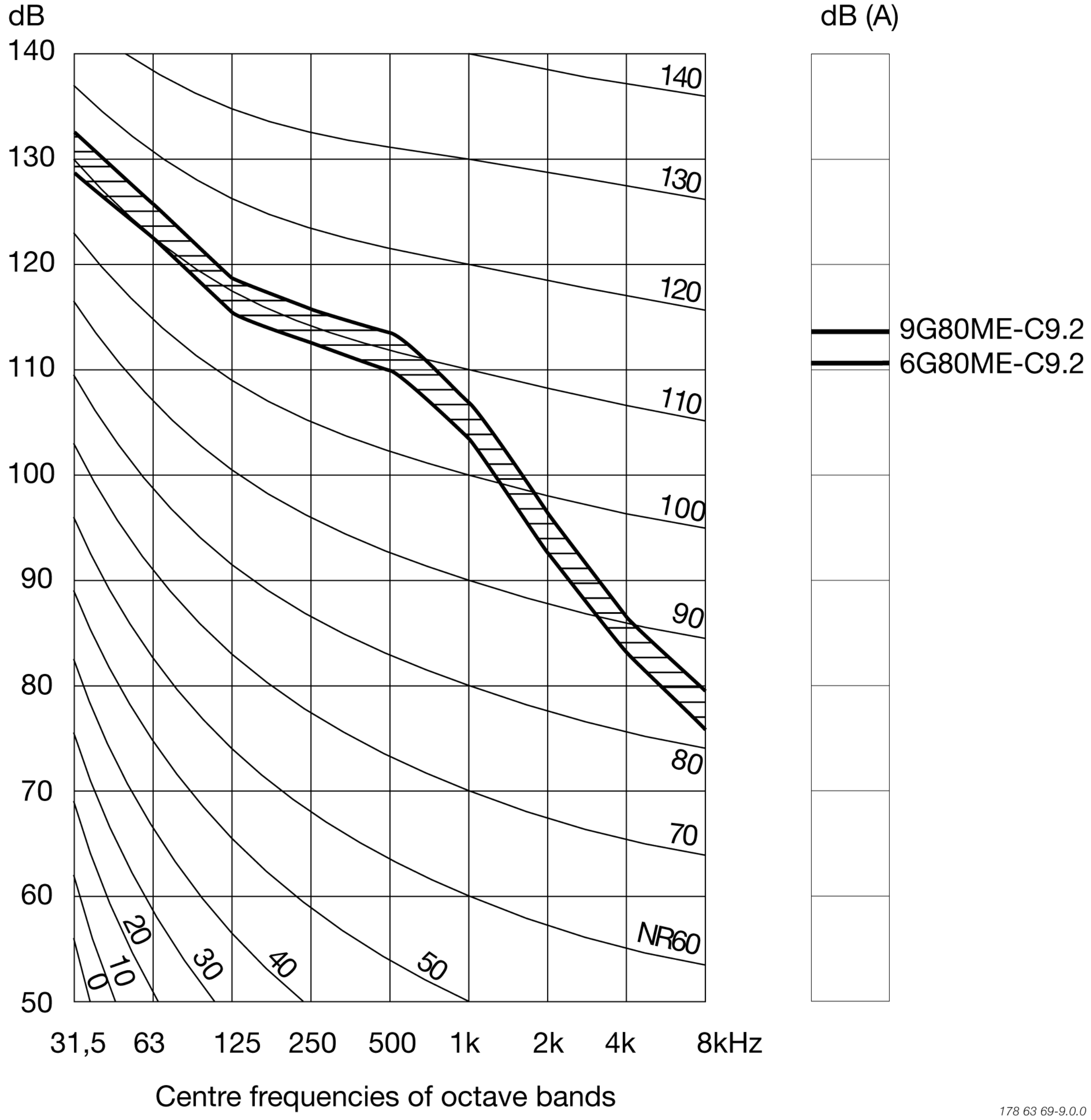

Fig. 15.04.02: ISO's NR curves and typical sound pressure levels from the engine's exhaust gas system. The noise levels at nominal MCR and a distance of 1 meter from the edge of the exhaust gas pipe opening at an angle of 30 degrees to the gas flow and valid for an exhaust gas system – without boiler and silencer, etc. Data for a specific engine and cylinder no. is available on request.

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.

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Calculation of exhaust gas back-pressure

The exhaust gas back pressure after the turbocharger(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 Destiny of Exhaust Gas (ρ)

$$\rho \cong 1.293 \times (273 / [\,273 + T\,]) \times 1.015 \text{ in kg/m}^3$$

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:

$$V = M/\rho \times (4 / [\,\pi \times D2]\,) \text{ in m/s}$$

Pressure Losses in Pipes (Δp)

For a pipe element, like a bend etc., with the resistance coefficient ζ, the corresponding pressure loss is:

$$\Delta p = (\zeta \times \tfrac{1}{2}\,\rho\,[v2 \times 1/9.81]) \text{ in mm WC}$$

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 (ΔpM)

The total back-pressure, measured/stated as the static pressure in the pipe after the turbocharger, is then:

$$\Delta pM = \Sigma \Delta p$$

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).

Mesuring Back Pressure

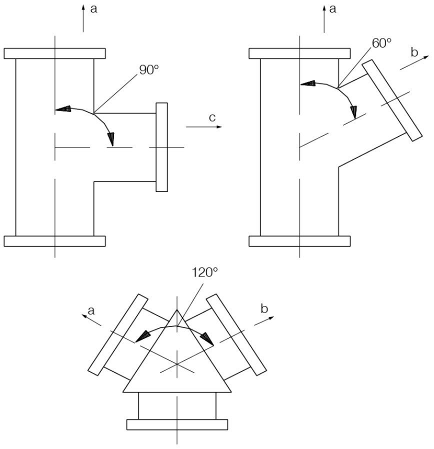
At any given position in the exhaust gas system, the total pressure of the flow can be divided into dynamic pressure (referring to the gas velocity) and static pressure (referring to the wall pressure, where the gas velocity is zero).

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



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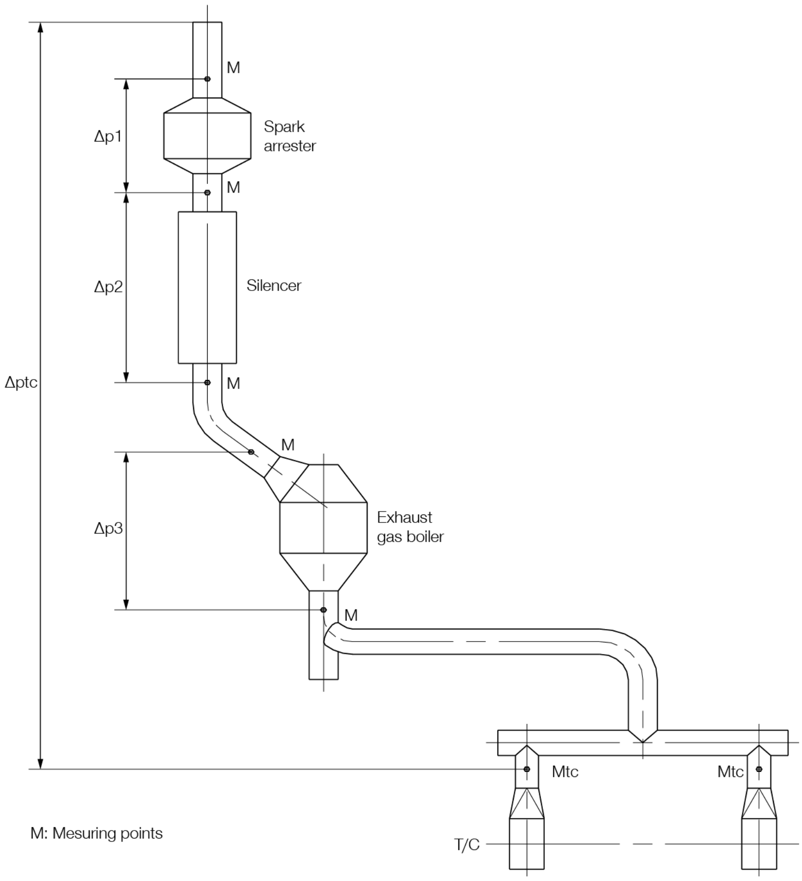
Change-over valves

Change-over valves of type with constant cross section

ζa = 0.6 to 1.2
ζb = 1.0 to 1.5
ζc = 1.5 to 2.0

Change-over valve of type with volume

$\zeta a = \zeta b$ = about 2.0



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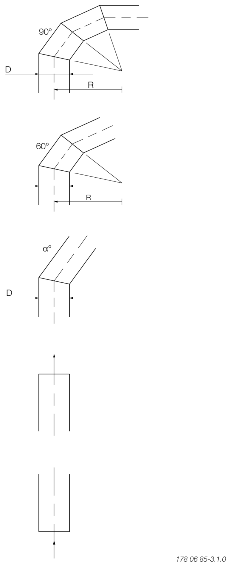
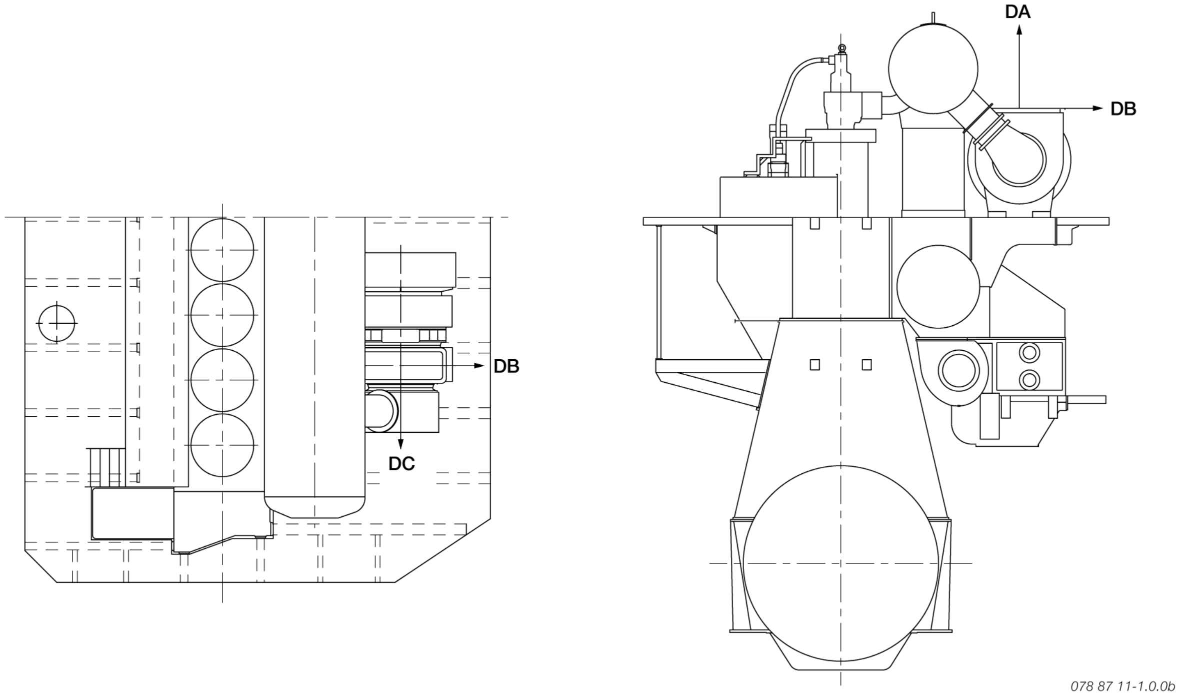


R = D	ζ = 0.60
R = 1.5D	ζ = 0.41
R = 2D	ζ = 0.27
R = D	ζ = 0.30
R = 1.5D	ζ = 0.15
R = 2D	ζ = 0.10
α°	
15°	ζ = 0.06
30°	ζ = 0.15
45°	ζ = 0.29
Outlet from top of exhaust gas uptake	ζ = 1.00
Inlet (from turbocharger)	ζ = - 1.00

Fig. 15.05.01: Pressure losses and coefficients of resistance in exhaust pipes

Forces and Moments at Turbocharger



DA: Max. movement of the turbocharger flange in the vertical direction
DB: Max. movement of the turbocharger flange in the transversal direction
DC: Max. movement of the turbocharger flange in the longitudinal direction

Fig. 15.06.01: Vectors of thermal expansion at the turbocharger(s) exhaust gas outlet flange

No. of cylinders		6-9		6	7	8	9
Turbocharger		DA mm	DB mm	DC mm	DC mm	DC mm	DC mm
Make	Type						
MAN	TCA66	8.8	1.7	2.2	2.5	2.8	3.1
	TCA77	10.1	1.8	2.2	2.5	2.8	3.1
	TCA88	10.6	1.9	2.2	2.5	2.8	3.1
ABB	A170 / A270	7.1	1.7	2.2	2.5	2.8	3.1
	A175 / A275	7.7	1.7	2.2	2.5	2.8	3.1
	A180 / A280	8.7	1.7	2.2	2.5	2.8	3.1
	A185 / A285	9.5	1.8	2.2	2.5	2.8	3.1
	A190	10.4	1.8	2.2	2.5	2.8	3.1
MHI	MET53	7.7	1.7	2.2	2.5	2.8	3.1
	MET60	8.2	1.7	2.2	2.5	2.8	3.1
	MET66	8.7	1.7	2.2	2.5	2.8	3.1
	MET71	9.1	1.7	2.2	2.5	2.8	3.1
	MET83	9.8	1.7	2.2	2.5	2.8	3.1

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	MET90	10.3	1.8	2.2	2.5	2.8	3.1

Table 15.06.01: Max. expected movements of the exhaust gas flange resulting from thermal expansion

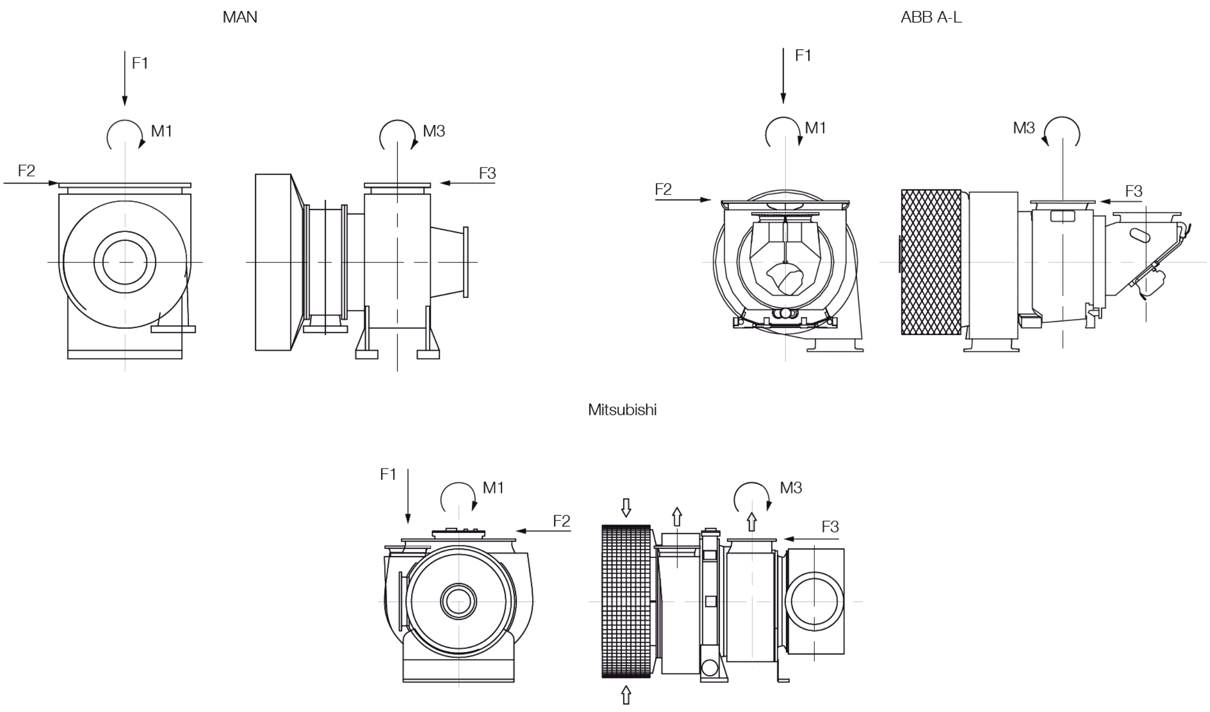

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Fig. 15.06.02: Forces and moments on the turbocharger(s) exhaust gas outlet flange

Turbocharger Make	Turbocharger Type	M1 Nm	M3 Nm	F1 N	F2 N	F3 N
MAN	TCA66	3,700	7,500	9,900	9,900	4,900
	TCA77	4,100	8,200	10,900	10,900	5,400
	TCA88	4,500	9,100	12,000	12,000	5,900
ABB	A170 / A270	1,900	1,900	3,600	2,400	2,400
	A175 / A275	3,300	3,300	5,400	3,500	3,500
	A180 / A280	4,600	4,600	6,800	4,400	4,400
	A185 / A285	6,600	6,600	8,500	5,500	5,500
	A190	8,700	8,700	10,300	6,700	6,700
MHI	MET53	4,900	2,500	7,300	2,600	2,300
	MET60	6,000	3,000	8,300	2,900	3,000
	MET66	6,800	3,400	9,300	3,200	3,000
	MET71	7,000	3,500	9,600	3,300	3,100
	MET83	9,800	4,900	11,700	4,100	3,700
	MET90	11,100	5,500	12,700	4,400	4,000

Table 15.06.02: The max. permissible forces and moments on the turbocharger's gas outlet flanges

Table 15.06.02 indicates the maximum permissible forces (F1, F2 and F3) and moments (M1 and M3), on the exhaust gas outlet flange of the turbocharger(s). Reference is made to Fig. 15.06.02.

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Diameter of exhaust gas pipes

General

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.

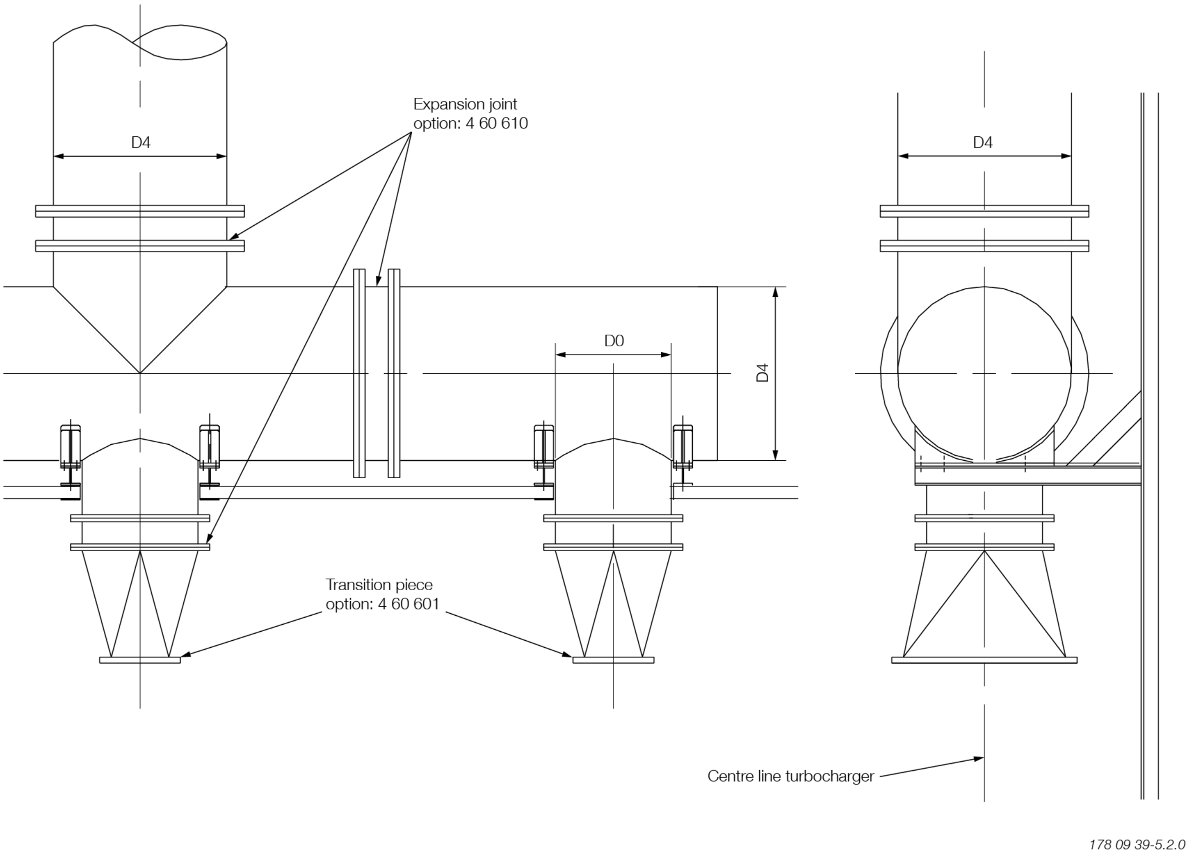

Fig. 15.07.01: Exhaust pipe system, with turbocharger located on exhaust side of engine

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	
kg/s	kg/s	kg/s	kg/s	[DN]	[DN]	[DN]	[DN]
47.5	54.3	61.1	67.8	1,600	1,150	900	1,600
53.6	61.3	68.9	76.6	1,700	1,200	1,000	1,700
60.1	68.7	77.3	85.9	1,800	1,300	1,050	1,800
67.0	76.5	86.1	95.7	N.A.	1,300	1,100	1,900
74.2	84.8	95.4	106.0	N.A.	1,400	1,150	2,000
81.8	93.5	105.2	116.9	N.A.	1,500	1,200	2,100
89.8	102.6	115.5	128.3	N.A.	1,600	1,300	2,200
98.1	112.2	126.2	140.2	N.A.	1,600	1,300	2,300
106.9	122.1	137.4	152.7	N.A.	1,700	1,400	2,400

Table 15.07.01: Exhaust gas pipe diameters and exhaust gas mass flow at various velocities

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Dual-fuel engine control system

The dual-fuel engine control system (ME-ECS) is a control system consisting of the ME-ECS core and the second fuel (SF) extension. It controls all functions known from the ME-engine as well as SF injection and the additional functionality and auxiliary systems related to handling SF on the engine and in the engine room.

The control includes:

- Fuel oil injection
- Electronically controlled exhaust valve actuation
- Governor/speed control
- Start and reverse sequencing
- Cylinder lubrication
- Variable turbocharging (if applied)
- Electronically controlled SF injection
- Sequencing changeover between fuel oil and dual-fuel operation
- SF combustion monitoring and SF shutdown
- Double-pipe ventilation and leakage monitoring
- Sealing oil control
- Purging of SF piping with inert gas
- Interface to SF control system.

For safety reasons, many functions are duplicated and as a result the SF extension is divided into two main parts: SF control system and SF safety system.

Fig. 16.00.01 illustrates how the ME-ECS core controls both pilot oil and SF injection. The SF extension handles SF safety and plant control, including the interface to the SF control system.

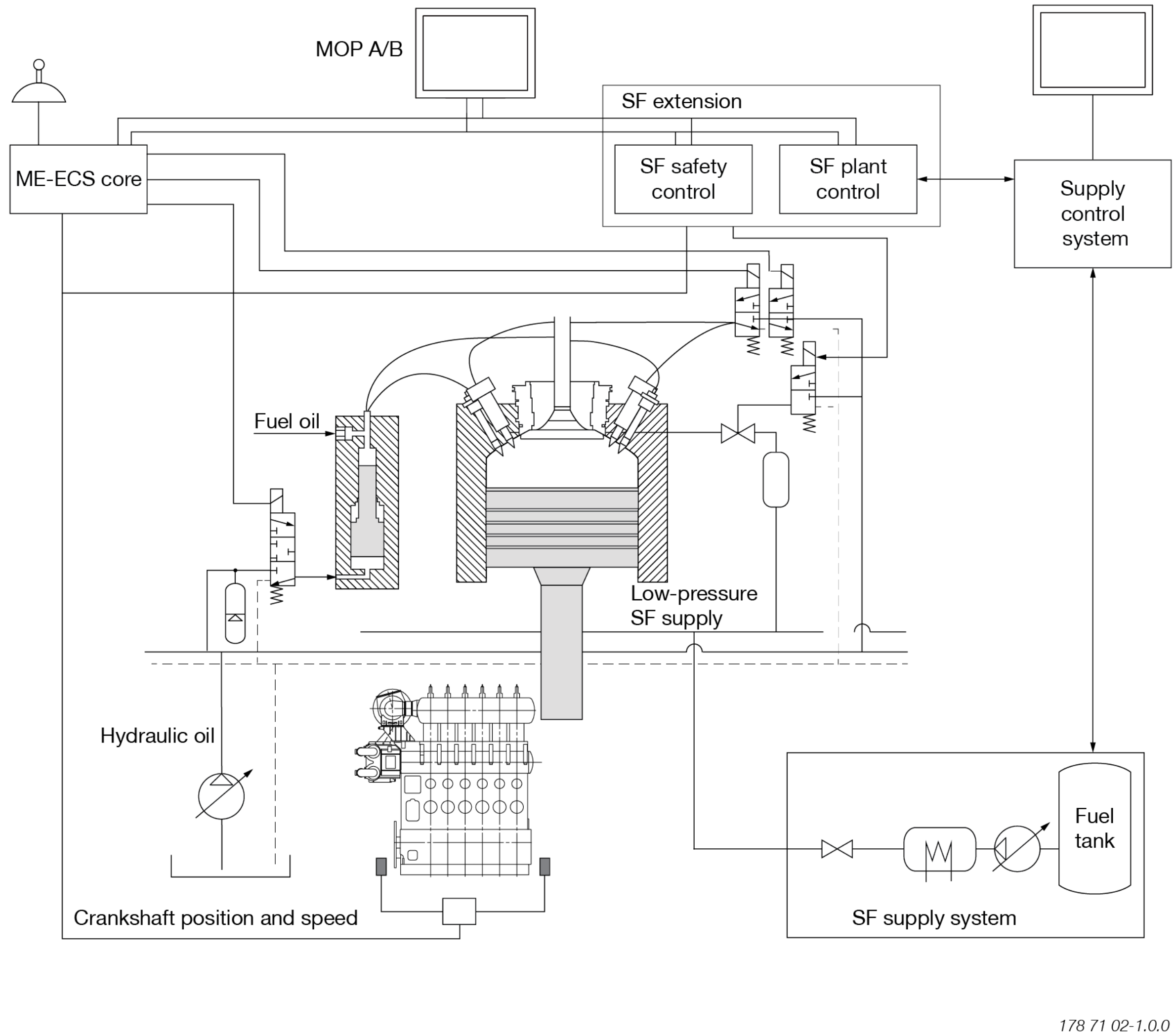

Fig. 16.00.01: Overview of ME-ECS core and SF extension

SF injection components

SF injection is controlled by two valves in series. A window valve which sets up a timing window during which SF can be injected and limits the maximum injection. A SF injection valve which controls the precise timing and injection amount.

The SF injection valve and the window valve are both controlled by the SF plant control and the SF safety control, respectively, as is the case for several other systems, see Fig. 16.00.02.

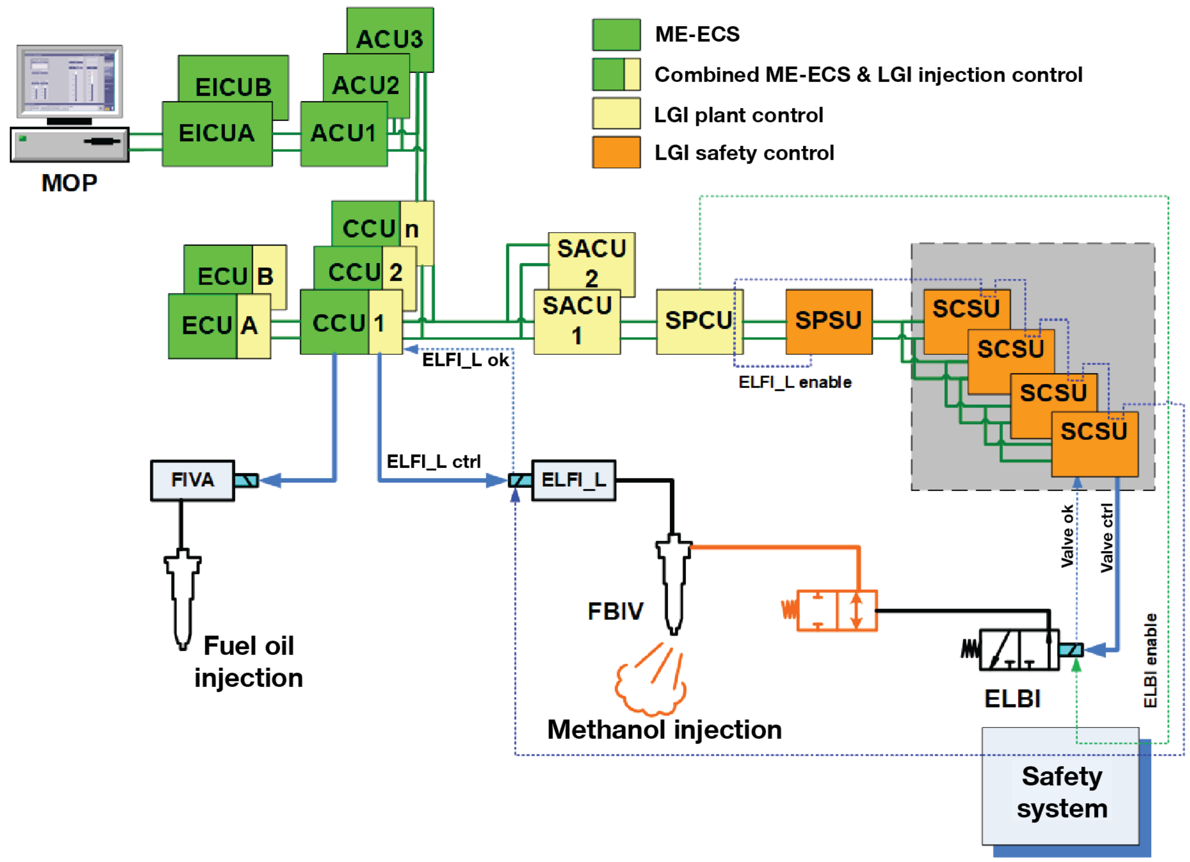

Fig. 16.00.02: ME-ECS and LGIM extension

Triton control system

All control units in the ME control system are based on the Triton platform.

Control units

Second fuel plant control unit (SPCU) and second fuel auxiliary control unit (SACU) perform the change of the SF system from 'no SF on engine' to 'SF running' and back again.

Safety units

The second fuel plant safety unit (SPSU) monitors specific second fuel plant safety sensors and, in case of a failure, it carries out a gas shutdown.

The second fuel cylinder safety unit (SCSU) monitors the cylinder pressure on all cylinders and evaluates for every cycle whether compression, injection and combustion processes are as expected.

If the SCSU detects a failure, a SF shutdown is carried out and the SF injection valve is disabled immediately.

See Fig. 16.00.02

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Units, layout and interfaces

The Engine Control System (ECS) for the ME engine is prepared for conventional remote control, having an interface to the Bridge Control system and the Local Operating Panel (LOP).

Control units for specific tasks described below are based on the available module types in the Triton hardware platform.

The layout of the Engine Control System is shown in Figs. 16.01.01, 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. In most cases, a single failure will not affect the performance or power availability, or only partly do so by activating a slow down.

It should be noted that any controller could be replaced without stopping the engine, which will revert to normal operation immediately after the replacement of the defective unit.

Main operating panel

Two redundant main operating panel (MOP) screens are available for the engineer to carry out engine commands, adjust the engine parameters, select the running modes, and observe the status of the control system. Both MOP screens are located in the engine control room (ECR), one serving as back-up unit in case of failure or to be used simultaneously, if preferred.

Both MOP screens consist of a marine approved personal computer with a touch screen and pointing device as shown in Fig. 5.16.02.

Engine control unit

For redundancy purposes, the control system comprises two engine control units (ECU) operating in parallel and performing the same tasks, 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

The control system includes one cylinder control unit (CCU) per cylinder. The CCU controls the electronic control valves for fuel injection and exhaust valve actuation as well as 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 cut out of operation.

Auxiliary control unit

The control of the auxiliary equipment on the engine is normally divided among four auxiliary control units (ACU) 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 and the scavenge air control (SCU).

If applied, the ACU also controls cooling water system (LDCL/JWA) and exhaust gas bypass (EGB)

Engine interface control unit

The engine interface control unit (EICC) perform such tasks as interface with the surrounding control systems, see Fig. 16.01.01a and b.

The EICC is located either in the engine control room (recommended) or in the engine room.

Control network

The main operating panels (MOP-A & MOP-B) are connected to the control system controllers via a redundant ehternet based control network. Due to limitations in cobber based ethernet, the maximum cable routing distance between the engine and the engine room is 100m.

Power supply for engine control system

The engine control system requires two separate power supplies with battery backup, power supply A and B.

The ME-ECS power supplies must be separated from other DC systems, i.e. only ME-ECS components must be connected to the supplies.

Power supply A	
System	IT (Floating), DC system w. individually isolated outputs
Voltage	Input 100-240V AC, 45-65 Hz, output 24V DC
Protection	Input over current, output over current, output high/low voltage
Alarms as potential free contacts	AC power, UPS battery mode, Batteries not available (fuse fail)

Power supply B	
System	IT (Floating), DC system w. individually isolated outputs
Voltage	Input 110-240 VAC, output 24V DC
Protection	Input over current, output over current, output high/low voltage

Power supply B	
Alarms as potential free contacts	AC power, UPS battery mode, Batteries not available (fuse fail)

Local operating panel

In normal operating the engine can be controlled from either the bridge or from the engine control room.

Alternatively, the local operating panel (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.

Hydraulic power supply

The purpose of the hydraulic power supply (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 during start-up as well as in normal service.

In case of the standard mechanically driven HPS unit, at start, one of the two electrically driven start-up pumps is activated.

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 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

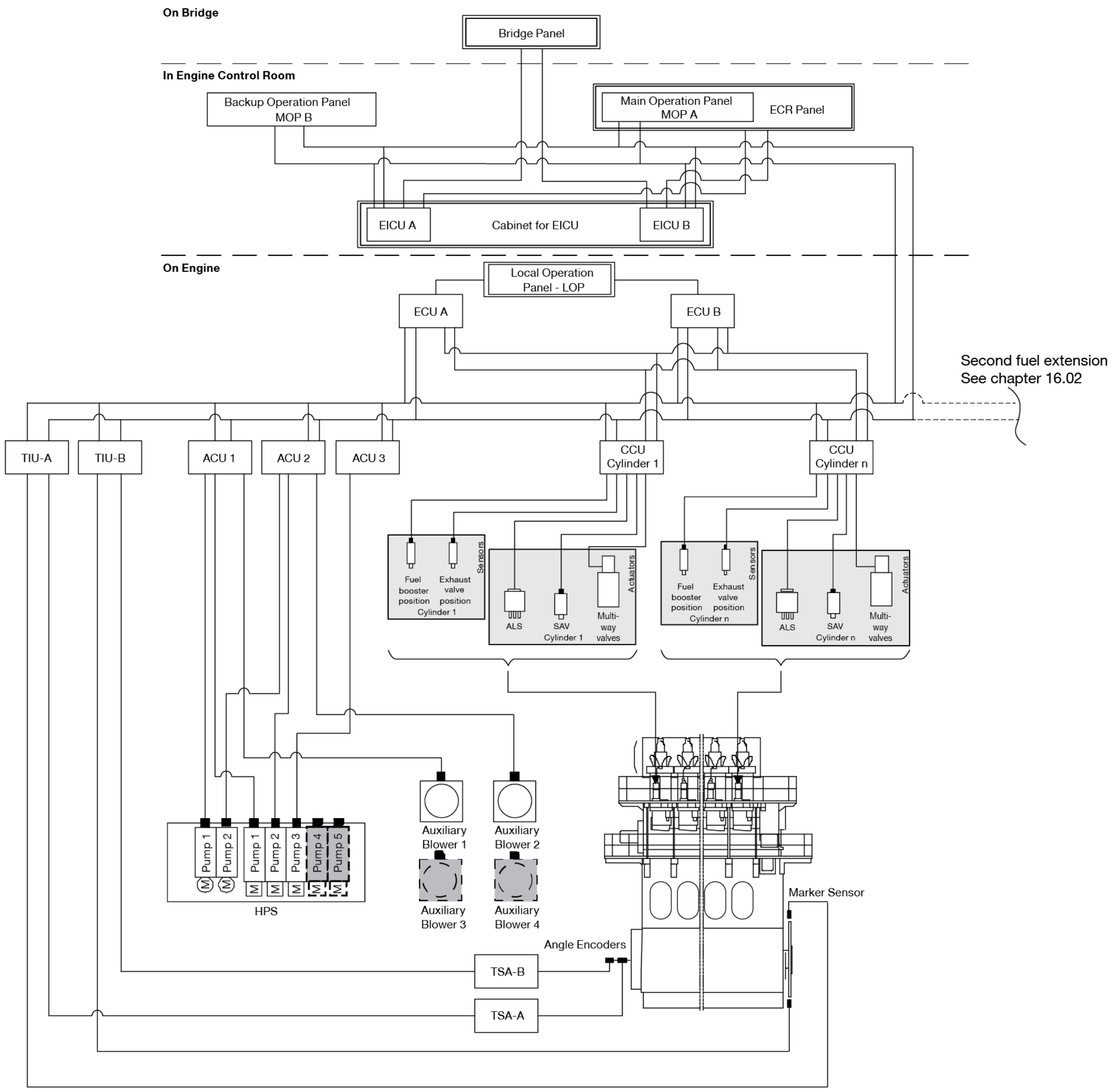

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Fig. 16.01.01: Engine control system layout with cabinet for EICU for mounting in ECR or on engine, EoD: 4 65 601

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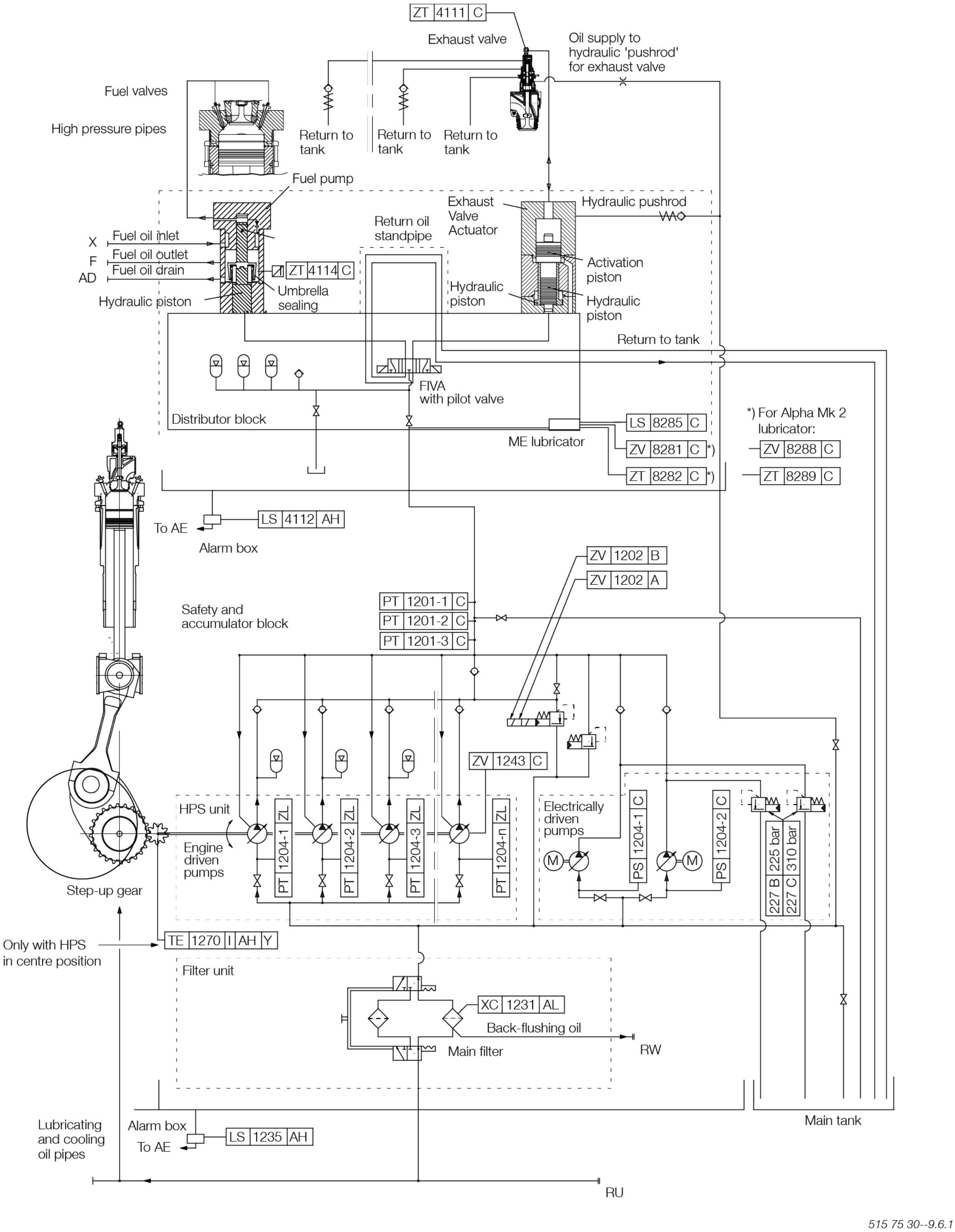
16.01 Units, layout and interfaces

Mechanical-hydraulic system with mechanically driven HPS



The letters refer to list of ‘counterflanges’
The item no. refer to ‘guidance values automation’

Fig. 16.01.02a: Mechanical-hydraulic system with mechanically driven hydraulic power supply, 300 bar, common supply

16.01 Units, layout and interfaces

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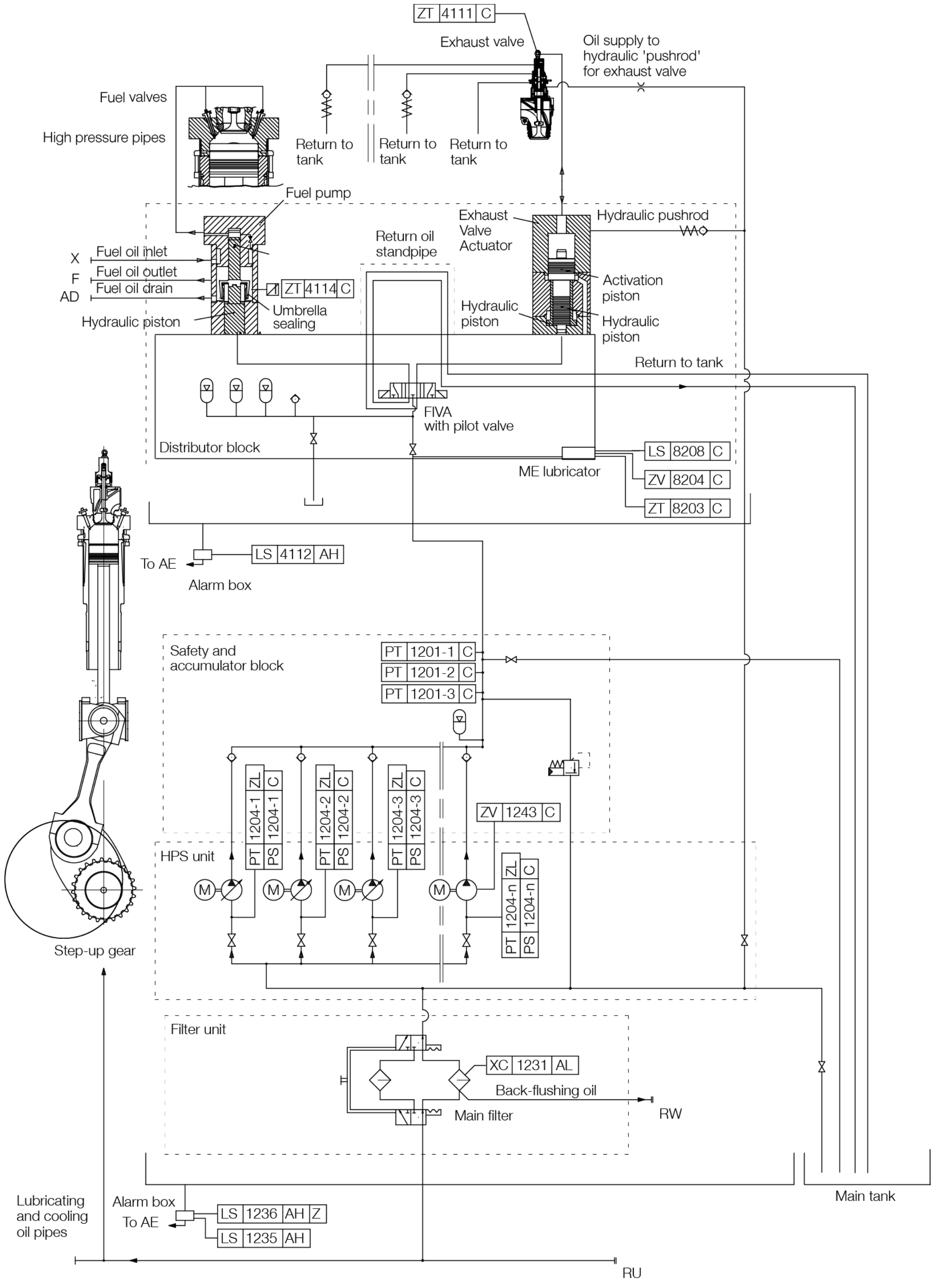
Mechanical-hydraulic system with electrically driven HPS



The letters refer to list of 'counterflanges'
The item no. refer to 'guidance values automation'

Fig. 16.01.02b: Mechanical-hydraulic system with electrically driven hydraulic power supply, 300 bar, common supply. Example from S90/80ME-C engine

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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.

Slow Down System

The Slowdown system receives slowdown requests from either ECS or AMS, and if the request is not cancelled, a slowdown command will be sent to ECS.

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 Energy Solutions. 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.

For the safety system, combined shut down and slow down panels approved by MAN Energy Solutions are available. The following options are listed in the Extent of Delivery:

4 75 631 Lyngsø Marine

4 75 632 Kongsberg Maritime

4 75 633 Nabtesco

4 75 636 Mitsui Zosen Systems Research.

Where separate shut down and slow down panels are installed, only panels approved by MAN Energy Solutions must be used.

In any case, the remote control system and the safety system (shut down and slow down panel) must be compatible.

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 three alternative control stations:

- the bridge control

- the engine control room control
- local control.

The remote control system is to be delivered by a supplier approved by MAN Energy Solutions.

Bridge control systems from suppliers approved by MAN Energy Solutions are available. The Extent of Delivery lists the following options:

- for fixed pitch propeller plants, e.g.:
  4 95 703 Lyngsø Marine
  4 95 704 Mitsui Zosen Systems Research
  4 95 705 Nabtesco
  4 95 715 Kongsberg Maritime
- and for controllable pitch propeller plants, e.g.:
  4 95 701 Lyngsø Marine
  4 95 716 Kongsberg Maritime
  4 95 719 MAN Alphatronic.

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.

ECS offers a number of interface options for safe connection/dis-connection of shaft generators

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 system

The engine control system (ECS) is supported by the engine management services (EMS), which includes the PMI auto-tuning and the CoCoS-EDS (Computer Controlled Surveillance-Engine Diagnostics System) applications.

A description of the EMS is found in chapter 18 of this project guide.

Instrumentation

The following lists of instrumentation are included in chapter 18:

- the class requirements and MAN Energy Solutions' requirements for alarms, slow down and shut down for unattended machinery spaces
- local instruments
- control devices.

Pneumatic manoeuvring diagram

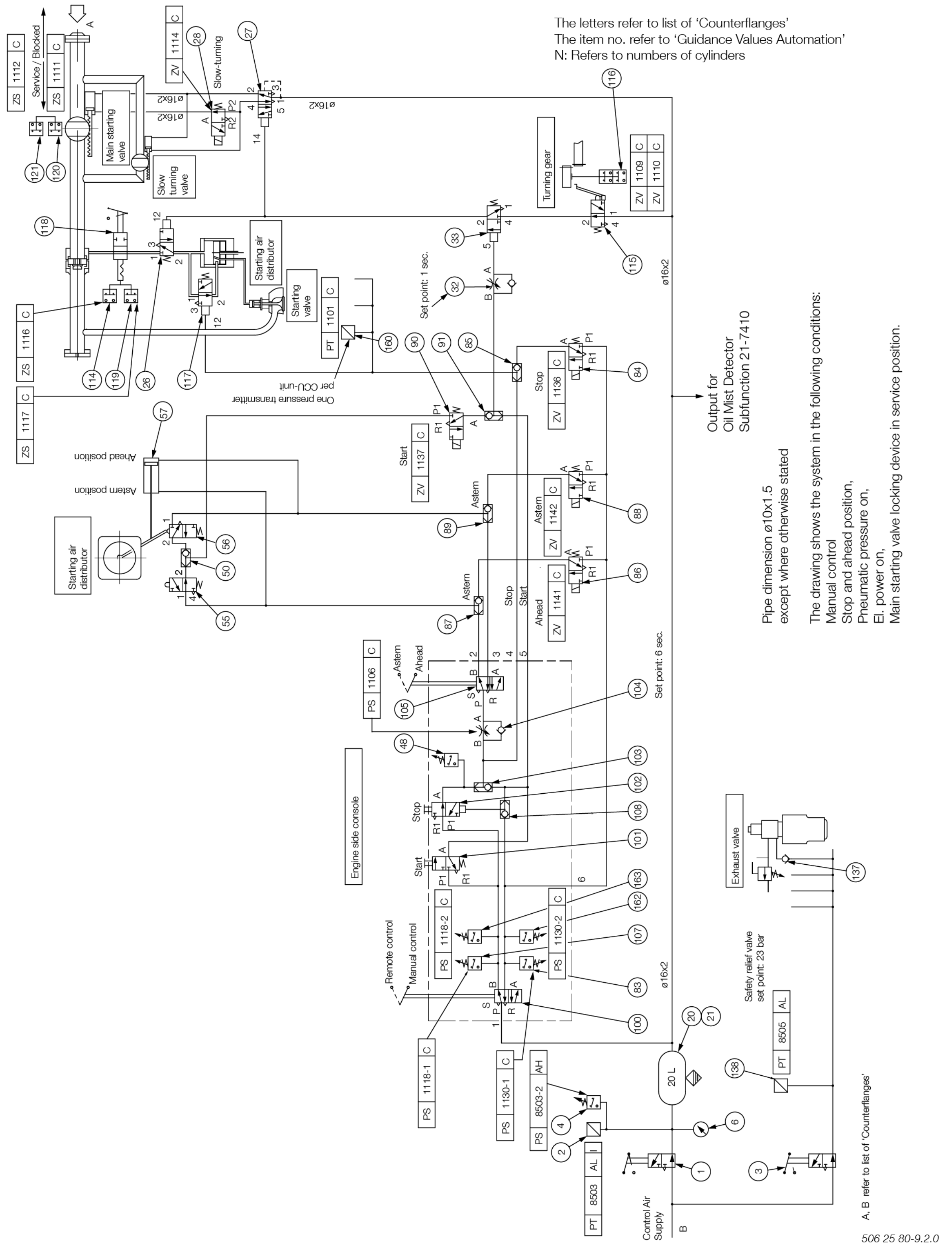

Fig. 16.01.03: Pneumatic manoeuvring diagram, FPP

Engine control system – second fuel extensions and interfaces

General

In addition to the ME-ECS core, a dual fuel extension is installed to control the gas supply and to monitor safety issues when the engine is operating on alternative fuels, see Fig. 16.02.01.

As mentioned, the dual fuel extension is designed as an add,on to the standard ME control system. The bridge panel, the main operating panel (MOP) and the local operating panel (LOP) is equipped with a dual-fuel running indication lamp. All operations in gas mode are performed solely from the engine room, while the operation from the bridge is exactly the same whether in dual fuel or fuel oil mode.

Dual-fuel control

The dual fuel control system consists of three parts:

- fuel injection control
- plant control
- safety control.

Dual fuel injection control is an additional functionality added to ECUs and CCUs of the MEECS, while plant control and safety control are handled by additional units: the SPCU (second fuel plant control unit) and SACU (second fuel auxiliary control unit), respectively SPSU (second fuel plant safety unit) and SCSUs (second fuel cylinder safety units).

Dual-fuel injection control

The task of the dual fuel injection control is to determine the fuel gas index and the pilot oil index when running in the different modes.

Dual-fuel plant control

The dual fuel plant control has the functions:

Function A – Controls the supply of dual fuel from the fuel gas supply system (FGSS) to the engine in a safe way.

Function B – Closes down the supply of gas to the engine after end of gas operation.

Function A:

- purges the gas pipes and gas volumes for atmospheric air before gas is allowed
- starts the double-pipe ventilation system and turns on double-pipe leakage detection
- applies gas to the engine in steps and checks for leakage and correct valve function, while the gas pressure builds up
- starts the sealing oil system, when SF enters the cylinder cover.

Function B:

- closes the GVT (gas valve train) block valves
- releases the dual fuel pressure

- stops the sealing oil system
- starts purging the gas pipes and volumes
- stops the double-pipe ventilation.

Furthermore, the task of the plant control is to handle the changeover between the two stable states:

- Fuel oil mode (HFO only)
- Dual fuel mode.

The plant control can operate all the fuel gas equipment. For the plant control to operate, it is required that the SF safety control allows it to work, otherwise the safety control will overrule and return to a gas safe condition.

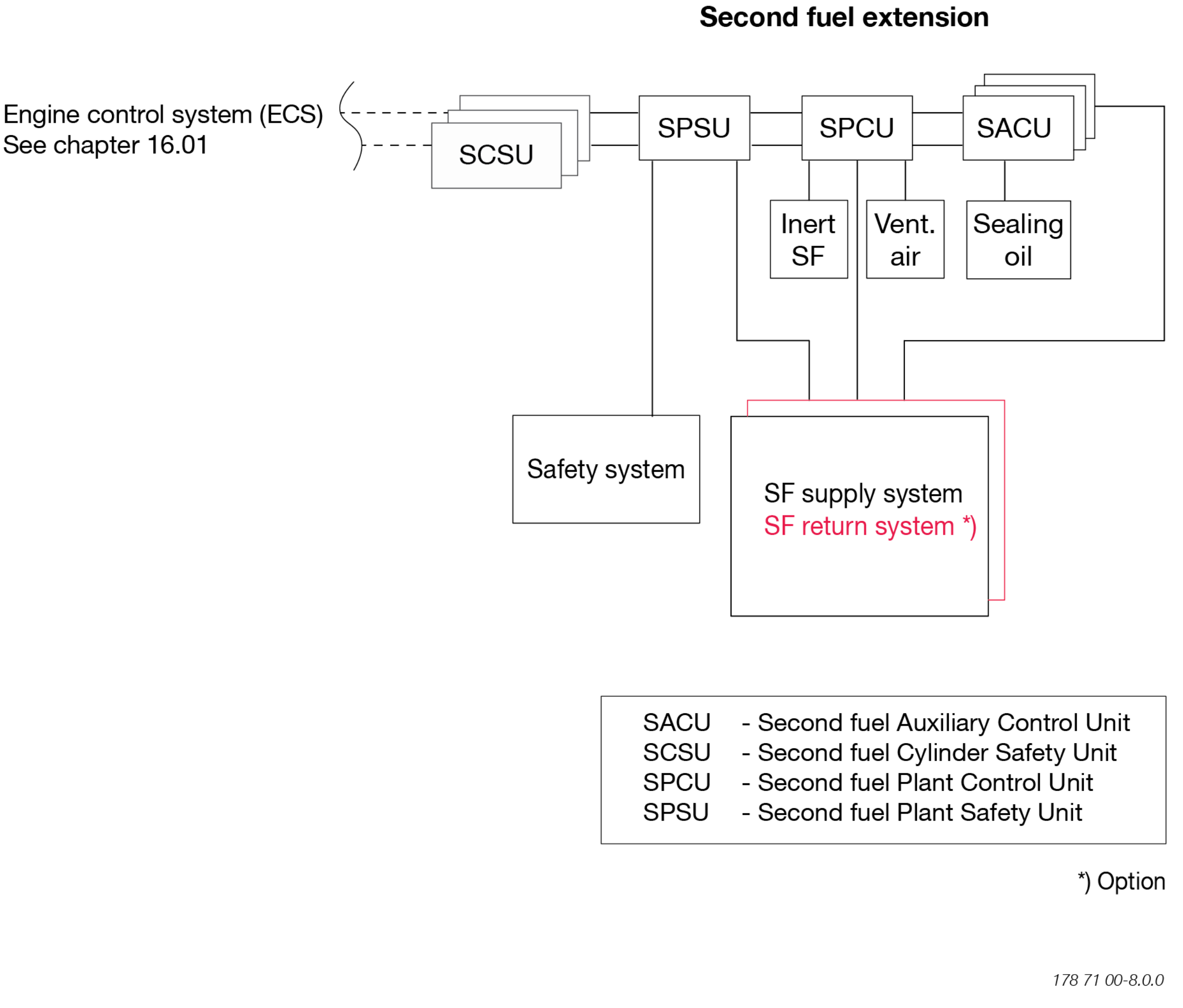

Fig. 16.02.01: The second fuel engine control system

Dual-fuel safety control

The task of the safety system is to monitor:

- manual and external automatic SF shut down
- engine shut down signal from the engine safety system
- double-pipe ventilation and leakage

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- sealing oil pressure
- SF pressure
- combustion pressure within normal values
- SF injection valve and window valve leakage.

If one of the above mentioned failures is detected, the SF safety control releases the gas shut down sequence:

The GVT, main gas valve and window valve will close. The electronic gas injection (ELGI) valves will be disabled. The fuel gas will be blown out by opening the SF bleed valve, the blow-off valves and purge valves, and finally the gas pipe system will be purged with inert fuels. See Fig. 7.00.01.

Safety principles of the dual fuel control system

SF mode running is not essential for the manoeuvrability of the ship, as the engine will continue to run on fuel oil if an unintended fuel gas stop occurs. The two fundamental safety principles of the SF equipment are, in order of priority:

- Safety to personnel must be at least on the same level as for a conventional diesel engine
- A fault in the dual fuel equipment must cause stop of SF operation and change over to fuel oil mode

which to some extent complement each other.

The dual fuel control system is designed to ‘fail to safe condition’. All failures detected during fuel gas running and failures of the control system itself will result in a fuel gas stop or shut down and change over to fuel oil operation.

Subsequently, the control system initiates blow out and purging of high pressure fuel gas pipes which releases all gas from the entire gas supply system of the engine room.

If the failure relates to the purging system, it may be necessary to carry out purging manually before engine repair is carried out.

The dual fuel control system itself is in general a single system without redundancy or manual back-up control.

Main operating panel (MOP)

The MOP is common to both the ME-ECS core and the SF extension. All the manual operations can be initiated from the MOP.

The MOP functions include the facilities to manually start up or to stop SF operation.

Additionally, the change between the different running modes can be done and the operator has the possibility to manually initiate purging of the SF piping with inert gas.

Dual-fuel function of the ECU and CCU

The dual fuel injection control is part of the ECU which includes all facilities required for calculating the fuel gas injection and the pilot oil injection based on the command from the ME governor function and the actual active mode.

Based on these data and information about the fuel gas pressure, the dual fuel injection control calculates the start and duration time of the injection, then sends the signal to the CCU which effectuates the injection by controlling both the electronic fuel injection valve (ELFI) (or the fuel injection valve actuation (FIVA) if applied) and the ELGI valve.

Second fuel plant and second fuel auxiliary control unit

When ‘Dual Fuel Mode Start’ is initiated manually by the operator, the second fuel plant (SPCU) will start the automatic start sequence.

The SPCU and second fuel auxiliary control unit (SACU) contain functionalities necessary to control and monitor auxiliary systems.

The SPCU and SACU controls:

- start/stop of pumps, fans, and of the SF supply system
- sealing oil pressure set points
- pressure set points for the SF supply system
- the purging with inert gas
- the ACOS, if applied.

The SPCU monitors the condition of the following:

- SF supply system
- sealing oil system
- double-pipe ventilation
- inert gas system

and, if a failure does occur, the SPCU will automatically interrupt SF mode start operation and return the plant to fuel oil mode.

Second fuel plant safety unit

The central Second fuel plant safety unit (SPSU) performs safety monitoring of the SF system and controls the SF shut down.

The SPSU monitors the following:

- SF leakage to the outer pipe of the double pipe
- pipe ventilation of the double-wall piping
- sealing oil pressure
- fuel gas pressure
- SCSU ready signal.

If one of the above parameters (referring to the relevant fuel gas state) differs from normal service value, the SPSU overrules any other signals and SF shut down will be released.

After the cause of the SF shut down has been corrected, the SF operation can be manually restarted.

The SPCU main state diagram is shown in Fig. 16.02.02.

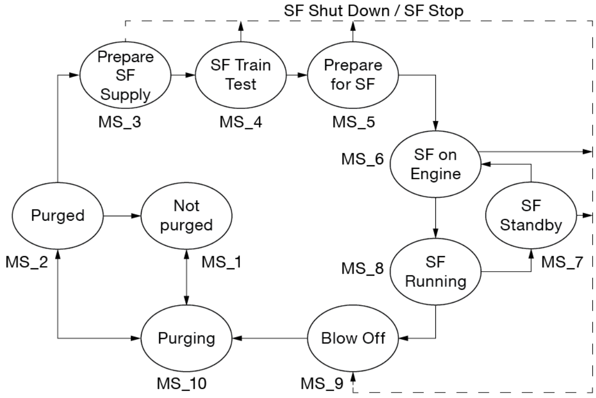

Fig. 16.02.02: SPCU main state diagram

Second fuel cylinder safety unit

The purpose of second fuel cylinder safety (SCSU) is to monitor if the cylinders are ready for the injection of SF.

The following events are monitored by the SCSU:

- SF pressure variations in SF block
  - SF injection valve leakage and malfunction
  - window valve leakage and malfunction
  - blow-off valve leakage
  - resume valve leakage and malfunction (if present)
  - sensor supervision
- Cylinder pressure
  - low compression pressure - too high maximum pressure
  - low expansion pressure/misfiring
  - too fast combustion pressure increase
- Fuel gas accumulator pressure drop during injection.

If one of the parameters is abnormal, the ELWI valve is closed and a shut down of SF running is activated by the SPSU.

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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.

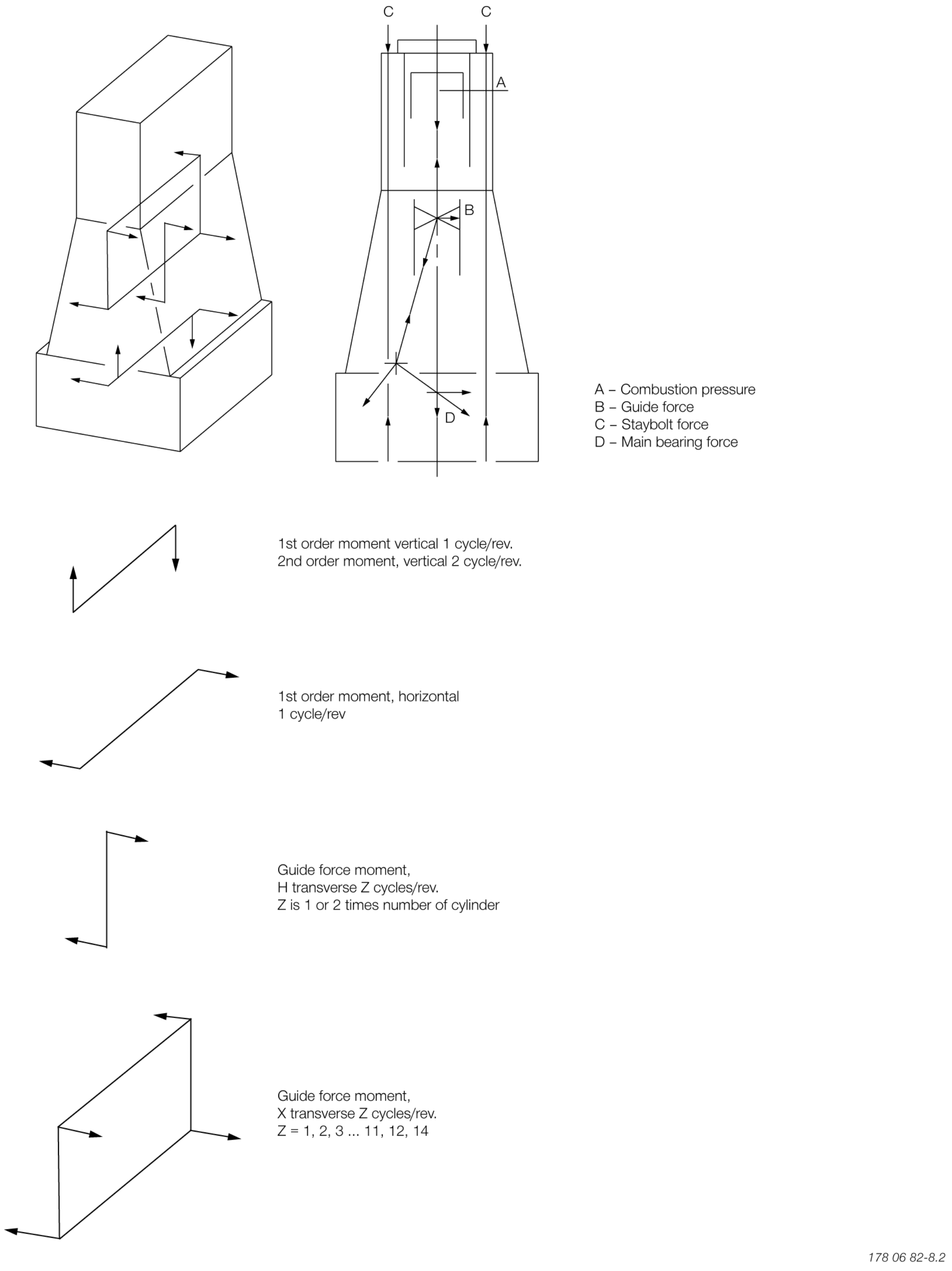

Fig. 17.01.01: External unbalanced moments and guide force moments

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First and second order moments

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.

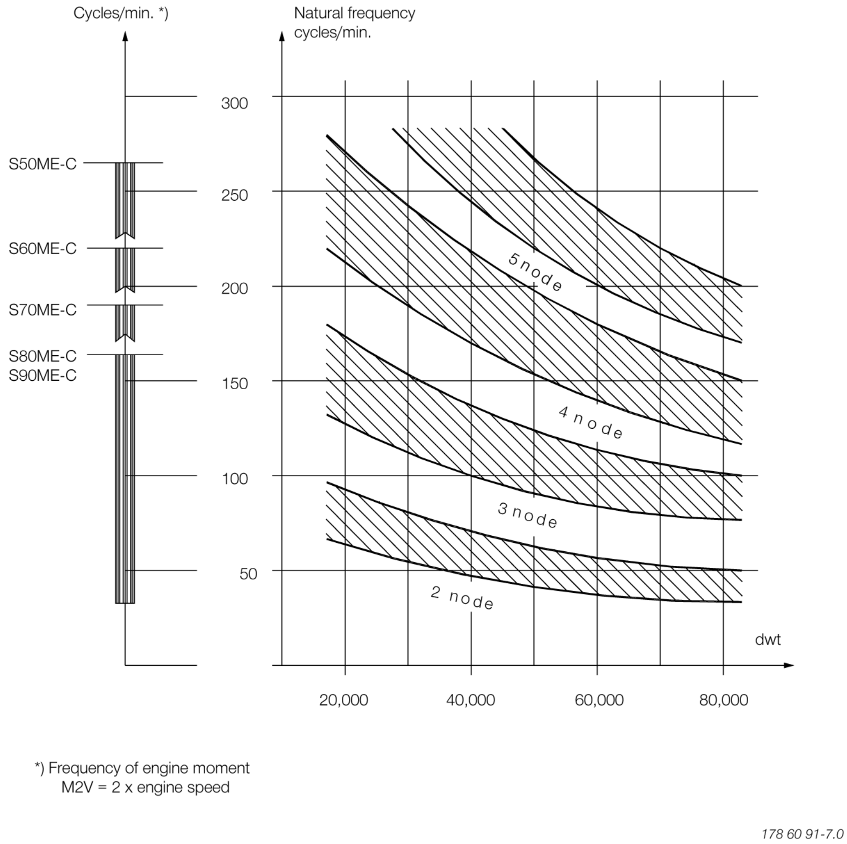

Fig. 17.02.01: Statistics of vertical hull vibrations in tankers and bulk carriers

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.

Determine the Need

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 MAN B&W external electrically driven moment compensator, RotComp, EoD: 4 31 255
- Prepared for compensator fore, EoD: 4 31 212

The available options are listed in the Extent of Delivery.

1st Order Moments on 4-Cylinder Engines

This section is not applicable.

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 external electrically driven moment compensator can neutralise the excitation, synchronised to the correct phase relative to the external force or moment.

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.

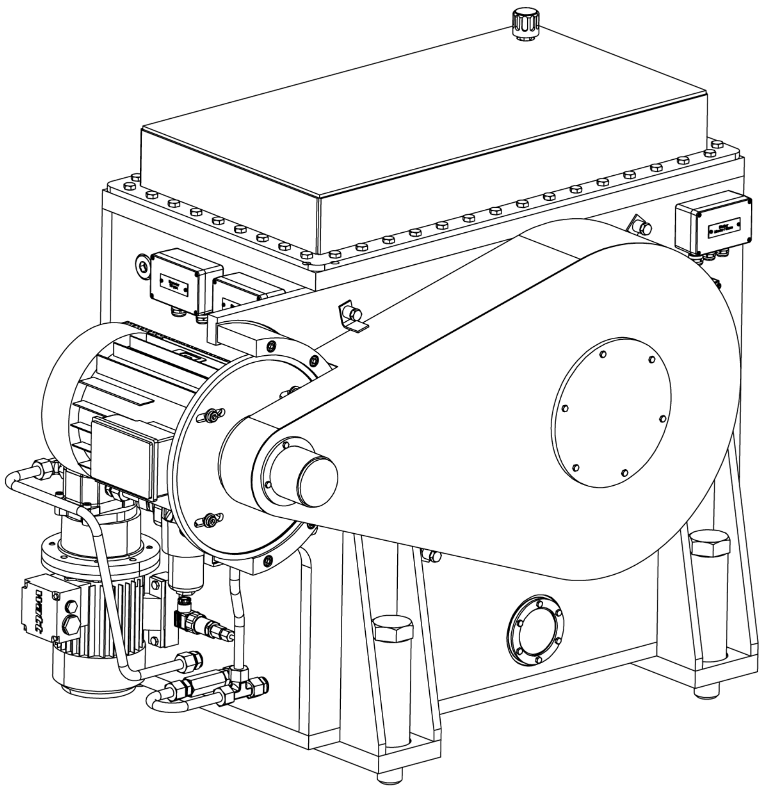
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Fig. 17.03.01: MAN B&W external electrically driven moment compensator, RotComp, option: 4 31 255

• 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.

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 reused 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.

Balancing Other Forces and Moments

Further to compensating 2nd order moments, electrically driven balancers are also available for balancing other forces and moments. The available options are listed in the Extent of Delivery.

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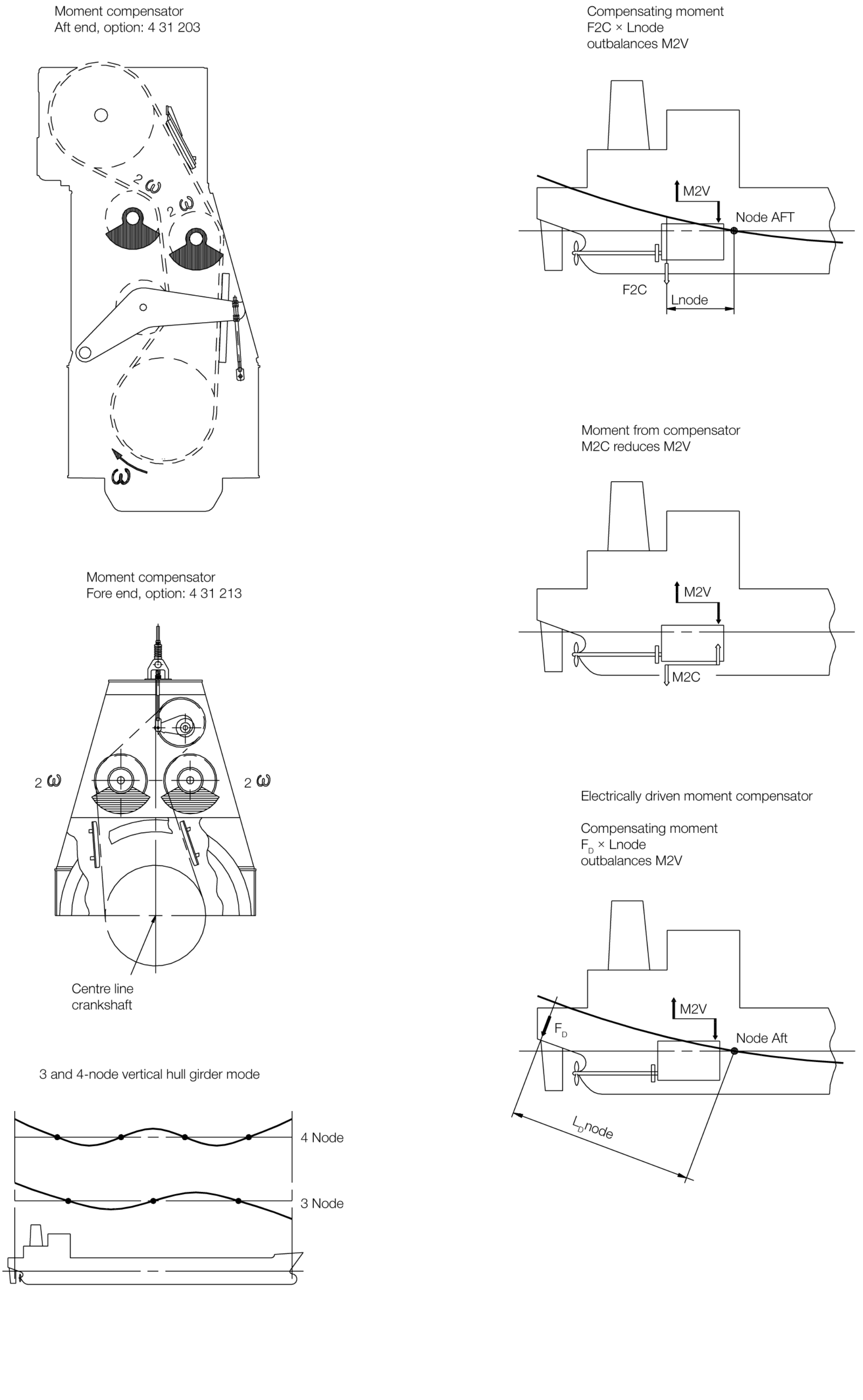

Fig. 17.03.02: Compensation of 2nd order vertical external moments

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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.

PRU = (External moment/ Engine power) 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

G80ME-C10.5/-GI – 4,710 kW/cyl at 72 r/min

	6 cyl.	7 cyl.	8 cyl.	9 cyl.
PRU acc. to 1st order, Nm/kW	0	8	2	21
PRU acc. to 2nd order, Nm/kW	138	34	0	30

Based on external moments in layout point L_1

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 L_1 of the layout diagram. For other speeds (n_A), the corresponding external moments (M_A) are calculated by means of the formula:

$$M_A = M_1 \times \{ n_A / n_1 \}^2 \text{ kNm}$$

(The tolerance on the calculated values is 2.5%).

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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_1) 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 Energy Solutions 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 Energy Solutions 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 proper 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.

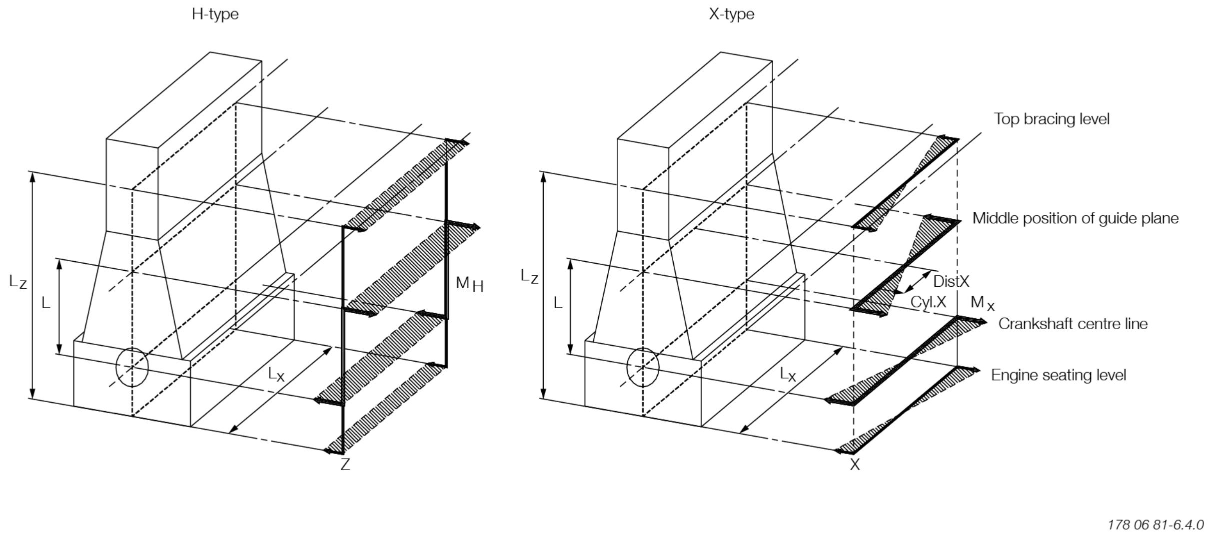

Fig. 17.05.01: H-type and X-type guide force moments

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:

$N \times M_{H(one\ cylinder)}$

For modelling purposes, the size of the forces in the force couple is:

$Force = M_H/L$ [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:

$Force_Z = M_H/L_Z$ [kN]

Any other vertical distance may be applied so as to accomodate 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.

$Force_{Z,\ one\ point} = Force_{Z,\ total}/N_{top\ bracing,\ total}$ [kN]

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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 contributes much.

A so-called ‘Bi-moment’ can be calculated (Fig. 17.05.01):

‘Bi-moment’ = ∑[force-couple(cyl.X) × distX] in kNm^2

The X-type guide force moment is then defined as:

M_X = ‘Bi-Moment’/L kNm

For modelling purpose, the size of the four (4) forces can be calculated:

Force = M_X/L_X [kN]

where:

L_X is the horizontal length between ‘force points’.

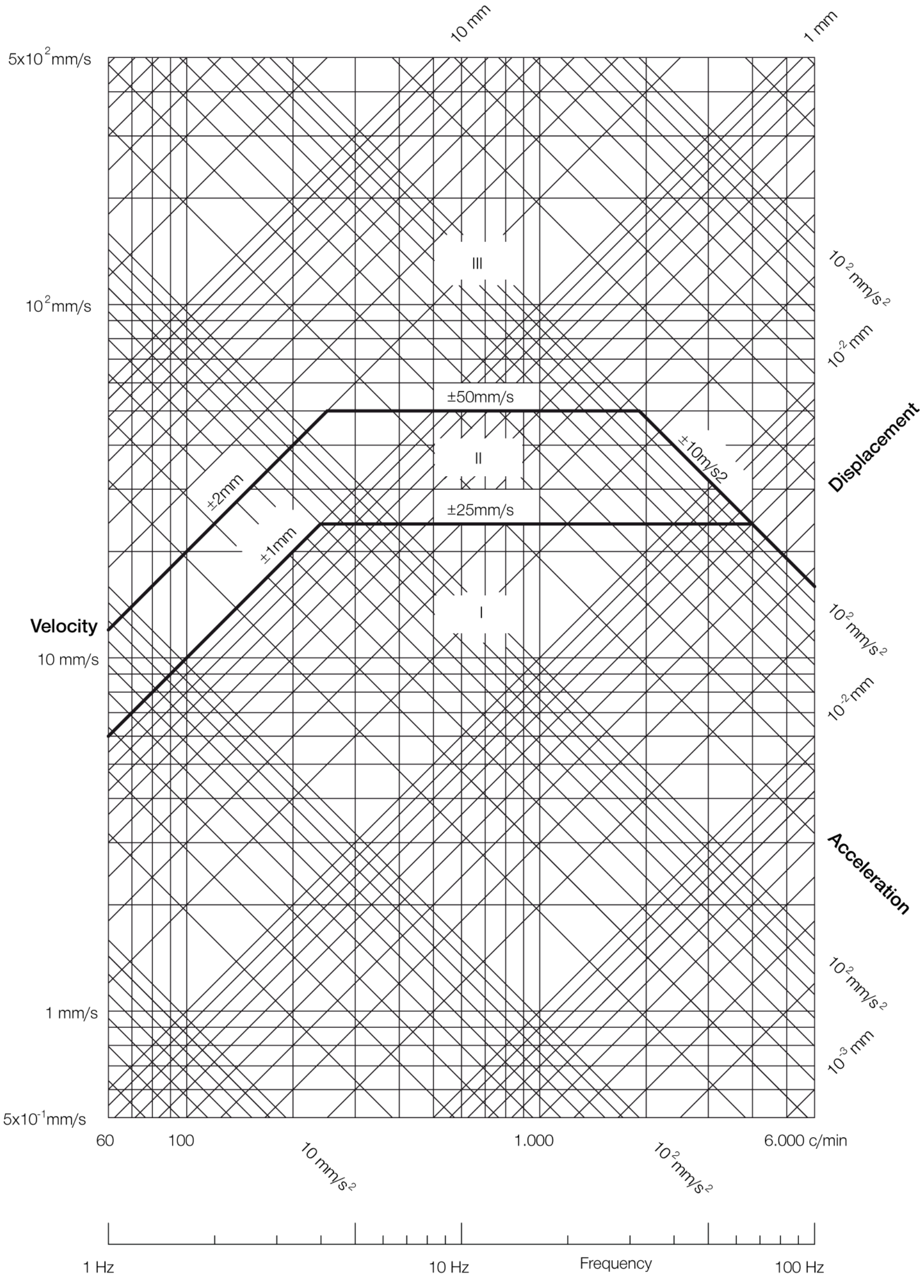
Similar to the situation for the H-type guide force moment, the forces may be applied in positions suitable for the FEM model of the hull. Thus the forces may be referred to another vertical level L_Z above the crankshaft centre line. These forces can be calculated as follows:

$Force_{Z,\ one\ point} = (M_x \text{ X } L) / (L_z \text{ X } L_x)$ [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
G95ME-C10/-GI/-LGI	3,720
G90ME-C10/-GI/-LGI	3,342
S90ME-C9/10/-GI/-LGI	3,600
G80ME-C10/-GI/-LGI	3,530
S80ME-C9/-GI/-LGI	3,450
G70ME-C9/10/-GI/-LGI	3,256
S70ME-C10/-GI/-LGI	2,700
S70ME-C7/8/-GI/-LGI	2,870
S65ME-C8/-GI/-LGI	2,730
G60ME-C10/-GI/-LGI	2,790
S60ME-C10/-GI/-LGI	2,310
G50ME-C9/-GI/-LGI	2,500
S50ME-C9/-GI/-LGI	2,214
S50ME-C8/-GI/-LGI	2,050
S46ME-C8/-GI/-LGI	1,980
G45ME-C9/-GI/-LGI	2,250
S40ME-C9/-GI/-LGI	1,770
S35ME-C9/-GI/-LGI	1,550

Vibration limits valid for single order harmonics



Zone I: Acceptable
Zone II: Vibration will not damage the main engine, however, under adverse conditions, annoying/harmful vibration responses may appear in the connected structures
Zone III: Not acceptable

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Fig. 17.05.02: Vibration limits

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Axial and torsional 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 consideraed.

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 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-cylinder engines in the bore range 98-80 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 Energy Solutions 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 at about 30-60% of 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 ±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.

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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.

Governor stability evaluation for special plants

On rare occasions, plant layouts are prone to engine speed instability or “engine hunting”. These plant layouts usually fall within one of the three categories mentioned below. MAN Energy Solutions may require that a governor stability evaluation is carried out for plants in the risk of encountering engine speed instability. The purpose of the governor stability evaluation is to identify the potential risks of engine speed instability and suggest corresponding countermeasures to the plant design. MAN Energy Solutions offers a governor stability evaluation against a fee.

1. Torsional vibrations at low frequency

A low torsional vibration frequency of a slender shaft line may interfere with the engine speed measurement of the engine control system (ECS). The ECS uses the speed measurement to calculate the fuel index, and therefore it directly affects the engine torque. The interference can in worst-case lead to amplification of the vibrations and hunting phenomena.

If the governor stability analysis reveals a risk of interference, the solution is to install an extra speed measurement device on the shaft line for the ECS, a so-called dual tacho.

2. Large electrical power take-off

If the power take-off (PTO) solution includes a variable frequency drive (VFD), which is the norm today, the PTO control system strategy is to deliver the required electrical power independent of the actual engine speed. This control strategy reduces the stability of the engine speed, because a drop in engine speed results in an increase in shaft generator torque to maintain a constant electrical power output. If the destabilisation effect is large, the engine control system is not able to compensate and properly stabilise the engine speed, this could result in engine hunting.

The destabilisation effect is largest at low speeds and at a high level of PTO.

If the governor stability analysis shows an inadequate stability margin, the PTO has to be limited in specific speed ranges, or turning and tuning wheels of higher inertia could be installed.

Usually, it is possible to mitigate the instability by applying our extended interface to the power management system (interface option C). See section 2.03 of the project guide for more information about the use of a PTO and PTO limits.

3. Operation without propeller

The propeller adds inertia and damping to the system which both increases engine speed stability and torque disturbance rejection. If the propeller is detached, for example by a clutch, the reduced system becomes more volatile. The result is an increased risk of critical speed overshoot and a reduced engine speed stability margin.

Normally, an engine is only allowed to operate in a narrow speed range without a propeller. The speed range is determined as part of the governor stability evaluation.

Plant criteria requiring a governor stability evaluation

If a plant fulfils one of the below criteria, MAN Energy Solutions must be contacted for further analysis, in case a plant fulfils one of the below mentioned criteria. In that case, we will conduct a governor stability evaluation against a fee. These special plants must be handled on an individual basis, preferably at an early stage of the design.

- PTO output is higher than specified in project guide section 2.03
- The engine is operated with PTO at test bed
- 1st node torsional vibration frequency in the propeller shaft line is lower than:
  - 3 Hz for FPP plants
  - 5 Hz for CPP plants
- The design includes a clutch for disconnecting the propeller
- The engine is for other reasons operated without a propeller, for example during PTO tests
- The design deviates from known “standard” plant designs.

The governor stability evaluation can lead to changes in the control equipment. This could, for example, be an increase of signals from the plant and requirements to the design of engine-driven mechanical components. The evaluation can also result in changes in the use of the PTO.

Direct questions regarding the governor stability evaluation at RDCPH@man-es.com.

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External forces and moments, G80ME-C10.5/-GI layout point L_1

No of cylinder :	6	7	8	9
Firing type :	1-5-3-4-2-6	1-7-2-5-4-3-6	1-6-5-2-7-4-3-8	1-6-7-3-5-8-2-4-9
External forces [kN] :				
1. Order : Horizontal	0	0	0	0
1. Order : Vertical	0	0	0	0
2. Order : Vertical	0	0	0	0
4. Order : Vertical	0	0	0	0
6. Order : Vertical	36	0	0	0
External moments [kNm] :				
1. Order : Horizontal a)	0	264	92	910
1. Order : Vertical a)	0	264	92	910
2. Order : Vertical	3,895	1,131	0	1,271
4. Order : Vertical	316	897	365	455
6. Order : Vertical	0	2	0	150
Guide force H-moments in [kNm] :				
1 x No. of cyl.	3,530	2,869	2,293	1,720
2 x No. of cyl.	265	91	61	94
3 x No. of cyl.	-	-	-	-

Guide force X-moments in [kNm] :				
1. Order :	0	216	75	745
2. Order :	432	126	0	141
3. Order :	655	717	1,093	1,312
4. Order :	1,148	3,262	1,326	1,653
5. Order :	0	342	5,097	1,694
6. Order :	0	62	0	4,204
7. Order :	0	0	15	184
8. Order :	394	30	0	105
9. Order :	682	76	7	104
10. Order :	174	494	0	46
11. Order :	0	300	458	34
12. Order :	0	18	70	316
13. Order :	0	3	157	52
14. Order :	18	0	0	26
15. Order :	23	1	0	47

16. Order :	10	3	0	3

a) 1st order moments are, as standard, balanced so as to obtain equal values for horizontal and vertical moments for all cylinder numbers.
c) 5 and 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

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Monitoring systems and instrumentation

The Engine Control System (ECS) is supported by the Engine Management Services (EMS), which manages software, data and applications for engine monitoring and operation.

The EMS includes the PMI and the CoCoS-EDS (Computer Controlled Surveillance-Engine Diagnostics System) as applications.

In its basic design, the ME/ME-B engine instrumentation consists of:

- Engine Control System (ECS), see Section 16.01
- Shut-down sensors, EoD: 4 75 124
- EMS including PMI and CoCoS-EDS software and support for LAN-based interface to the AMS, EoD: 4 75 217, see Section 18.02
- Sensors for alarm, slow down and remote indication according to the classification society's and MAN Energy Solutions' requirements for UMS, EoD: 4 75 127, see Section 18.04.

All instruments are identified by a combination of symbols and a position number as shown in Section 18.07.

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Engine Management Services

Engine Management Services Overview

The Engine Management Services (EMS) is used on MAN B&W engines from MAN Energy Solutions for condition monitoring, data logging & data distribution. EMS is integrated with the ECS (Engine Control System) to allow for continuous performance tuning.

EMS is executed on the EMS MOP, an industrial type PC designed by MAN Energy Solutions. EMS is implemented as a hardened platform, robust to virus threats and other unauthorized use and ac-cess.

The EMS network topology is shown in Fig. 18.02.01.

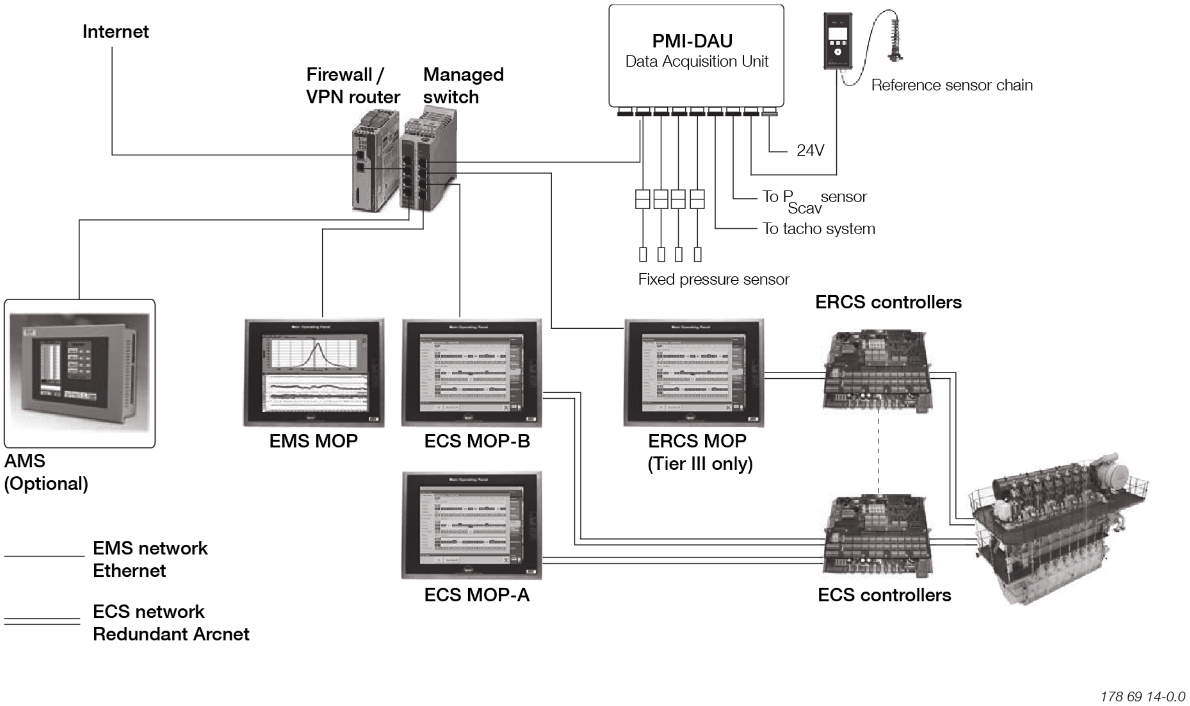

Fig 18.02.01: Engine Management Services, EMS, EoD: 4 75 217

EMS Applications

EMS includes the applications PMI Auto-tuning, CoCoS-EDS and EMS manager.

PMI Auto-tuning

- Online cylinder pressure monitoring
- Input to engine control system for closed-loop performance tuning
- Engine power estimation.

PMI Auto-tuning continuously measures the cylinder pressures using online sensors mounted on each cylinder cover. Pressure measurements are presented continuously in real time and the corresponding key performance values are transferred to the Engine Control System.

The Engine Control System constantly monitors and compares the measured combustion pressures to a reference value. As such, the control system automatically adjusts the fuel injection and valve timing to reduce the deviation between the measured values and the reference. This, in turn, facilitates the optimal combustion pressures for the next firing. Thus, the system ensures that the engine is running at the desired maximum pressure, p(max).

CoCoS-EDS

- Data logging
- Engine condition monitoring and reporting
- Engine operation troubleshooting.

With CoCoS-EDS, early intervention as well as preventive maintenance, the engine operators are able to reduce the risk of damages and failures.

CoCoS-EDS further allows for easier troubleshooting in cases where unusual engine behavior is observed.

EMS Manager

- Installation and supervision of EMS applications
- Network and interface monitoring
- Optional interface for data exchange with AMS (Alarm Monitoring System).

The EMS manager provides a process for integrated installation, commissioning and maintenance of PMI Auto-tuning and CoCoS-EDS.

Further, the EMS Manager includes status information and functionality, e.g. for network status, internal and external interfaces and EMS application execution.

Condition Monitoring System CoCoS-EDS

This section is not applicable

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Slow down and shut down

The shut down system must be electrically separated from other systems by using independent sensors, or sensors common to the alarm system and the monitoring 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 B&W engine are designed for Unattended Machinery Space (UMS) and comprises:

- the temperature sensors and pressure sensors that are specified in the 'MAN Energy Solutions' 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 B&W engine are designed for Unattended Machinery Space (UMS) and comprises:

- The sensors for alarm and slow down.

These sensors are included in the basic Extent of Delivery, EoD: 4 75 127.

The shut down and slow down panels can be ordered as options: 4 75 630, 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 Energy Solutions 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 is the sum of requirements of the classification society, the Buyer and MAN Energy Solutions.

If further analogue sensors are required, they can be ordered as option: 4 75 128.

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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.

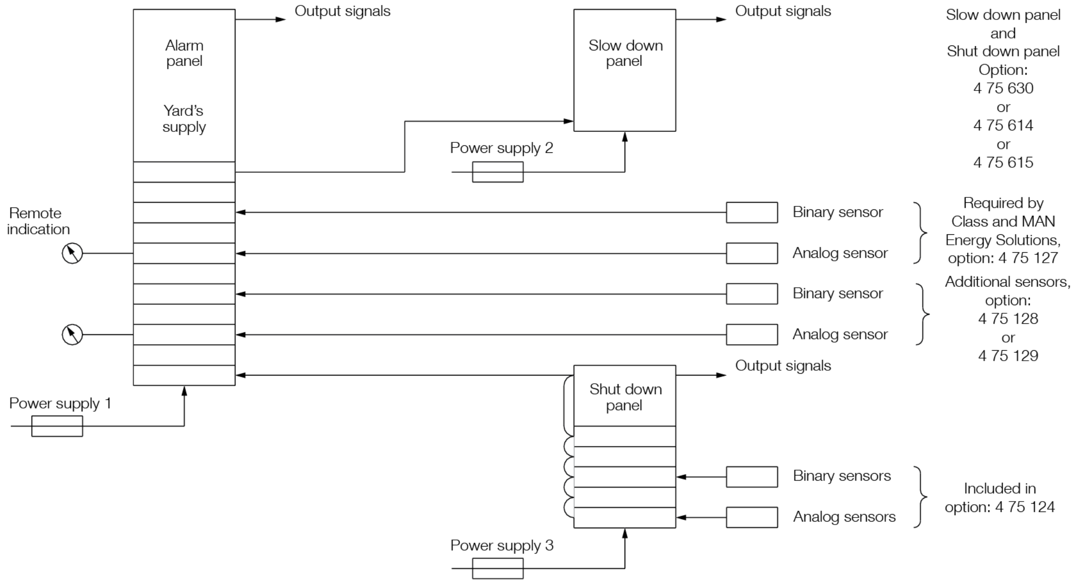
Electrical System, General Outline

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.



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Alarms for UMS – Class and MAN Energy Solutions Requirements

ABS	BV	CCS	DNV	KR	LR	NK	RINA	RS	IACS	MAN ES	Sensor and function	Point of location
												Fuel oil
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	LS 8006 AH	Leakage from high pressure pipes
												Lubricating oil
1	1	1	1	1	1	1	1	1	1	1	TT 8106 AH	Thrust bearing segment
1	1	1	1	1	1	1	1	1	1	1	PT 8108 AL	Lubricating oil inlet to main en-gine
1	1	1	1	1		1	1	1	1	1	TT 8113 AH	Piston cooling oil outlet/cylinder
1	1	1	1	1		1	1	1	1	1	FS 8114 AL	Piston cooling oil outlet/cylinder
										1	TT 8117 AH	Turbocharger lubricating oil out-let/turbocharger
										1	TT 8123 AH	Main bearing oil outlet temperat-ure/main bearing (S40/35ME-B9 only)
										1	XC 8126 AH	Bearing wear (All types except S40/35ME-B9); sensor common to XC 8126/27
										1	XS 8127 A	Bearing wear detector failure (All types except S40/ 35ME-B)
		1			1	1				1	PDS 8140 AH	Lubricating oil differential pres-sure - cross filter
										1	XS 8150 AH	Water in lubricating oil; sensor common to 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	TE 8202 AH	Cylinder lubricating oil temperat-ure
										1	LS 8212 AL	Small tank for heating element, low level (Not for ACOM)
										1	XC 8265 AL	ACOM common alarm (Only for ACOM)

1. Indicates that the sensor is required.
The sensors in the MAN ESS and relevant Class columns are included in the basic Extent of Delivery, EoD: 4 75 127.

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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 Energy Solutions' requirements

ABS	BV	CCS	DNV	KR	LR	NK	RINA	RS	IACS	MAN ES	Sensor and function	Point of location
												Hydraulic Power Supply
										1	PT 1228 AL	LPS booster, oil presure after pump (Only if LPS pump)
										1	PDS 1231 A	ME (Auto) filter differential pressure across filter
										1	TT 1310 AH	Lubrication oil inlet (Only for ME/-GI with separate oil system to HPS installed)
												Cooling water
1	1	1	1	1	1	1	1	1	1	1	PT 8401 AL	Jacket cooling water inlet
										1	PDT 8403 AL	Jacket cooling water across engine; to be calculated in alarm system from sensor no. 8402 and 8413 3)
										1	PDT 8404 AL	Jacket cooling water across cylinder liners 2)
										1	PDT 8405 AL	Jacket cooling water across cylinder covers and exhaust valves 2)
										1	TT 8407 AL	Jacket cooling water inlet
1	1	1	1	1	1	1	1	1	1	1	1 TT 8408 AH	Jacket cooling water outlet, cylinder
										1	TT 8410 AH	Cylinder cover cooling water outlet, cylinder 2)
										1	PT 8413 I	Jacket cooling water outlet, common pipe
1	1	1		1	1	1	1	1	1	1	PT 8421 AL	Cooling water inlet air cooler

										1	TT 8422 AH	Cooling water inlet air cooler/air cooler
												Compressed air
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	PT 8503 AL	Control air inlet and finished with engine
			1							1	PT 8505 AL	Air inlet to air cylinder for exhaust valve
												Scavenge air
								1		1	PS 8604 AL	Scavenge air, auxiliary blower, failure (Only ME-B)
		1				1÷				1	TT 8609 AH	Scavenge air receiver
1	1	1	1	1	1	1	1	1	1	1	TT 8610 AH	Scavenge air box – fire alarm, cylinder/cylinder
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 ES and relevant Class columns are included in the basic Extent of Delivery, EoD: 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 only for engines wirh LDCL cooling water system.

3. Not applicable for engines with LDCL cooling water system.

⎡ 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 Energy Solutions' requirements

ABS	BV	CCS	DNV	KR	LR	NK	RINA	RS	IACS	MAN ES	Sensor and function	Point of location
												Exhaust gas
1	1	1	1	1	(1)	1	1	1	1	1	TT8701 AH	Exhaust gas before turbocharger/turbocharger
1	1		1	1	1	1	1	1	1	1	TT8702 AH	Exhaust gas after exhaust valve, cylinder/cylinder
1	1	1	1	1	1	1	1	1	1		TT8707 AH	Exhaust gas outlet turbocharger/turbocharger (Yard's supply)
												Miscellaneous
										1	ZT8801 AH	Turbocharger speed/ turbocharger
										1	WT8812 AH	Axial vibration monitor 2)
1	1		1	1	1	1	1	1	1	1	XS8813 AH	Oil mist in crankcase/ cylinder; sensor common to XS 8813/14
	1									1	XS8814 AL	Oil mist detector failure
										1	XC8816 AH	Shaftline earthing device
										1	TT8820 AH	Cylinder liner monitoring/cylinder 3)
												Engine Control System
1	1	1	1	1	1	1	1	1	1	1	XC2201 A	Power failure
1	1		1		1	1	1	1	1	1	XC2202 A	ME common failure
										1	XC2202-A A	ME common failure (ME-GI only)
										1	XC2213 A	Double-pipe HC alarm (ME-GI only)
												Power Supply Units to Alarm System
										1	XC2909	Main supply failure
										1	XC2910	Battery discharging
										1	XC2911	Battery protective device tripped

1. Indicates that the sensor is required.
The sensors in the MAN ES and relevant Class columns are included in the basic Extent of Delivery, EoD: 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 certain engines only, see the list in Section 18.06, Axial Vibration Monitor.

3) Required for: K98ME/ME-C, S90ME-C, K90ME-C and K80ME-C9 engines incl. ME-GI variants.

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 Energy Solutions' requirements

ABS	BV	CCS	DNV	KR	LR	NK	RINA	RS	IACS	MAN -ES	Sensor & function	Point of location
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	PT 8108 YL	Lubricating oil inlet to main engine
1	1	1	1	1		1	1	1	1	1	PT 8113 YH	Piston cooling oil outlet/cylinder
1	1	1	1	1	1	1	1	1	1	1	PT 8114 YH	Piston cooling oil outlet/cylinder
1	1	1		1	1		1	1	1	1	PT 8117 YL	Turbocharger lubricating oil outlet/ TC
										1	PT 8123 YH	Main bearing oil outlet temp./main bearing
										1	XC 8126 YH	Bearing wear
1	1	1	1	1	1	1	1	1	1	1	PT 8401 YL	Jacket cooling water inlet
										1	PDT 8403 YL	Jacket cooling water across engine (not LDCL)
										1	PDT 8404 YL	Jacket cooling water across en-gine(only LDCL)
										1	PDT 8405 YL	Jacket cooling water across cyl. covers and exhaust valves (Only LDCL)
1	1	1	1	1	1	1	1	1	1	1	TE 8408 YH	Jacket cooling water outlet
		1				1				1	TE 8609 YH	Scavenge air receiver
1	1	1	1	1	1	1	1	1	1	1	TE 8610 YH	Scavenge air box fire alarm
		1	1					1		1	TC 8701 YH	Exhaust gas before TC
1	1		1	1	1	1	1	1	1	1	TC 8702 YH	Exhaust gas after exhaust valve
			1							1	TC 8702 YH	Exhaust gas after exhaust valve, cylinder deviation from average
										1	ZT 8801 YH	Turbocharger overspeed
										1	WT 8812 YH	Axial vibration monitor 2)
1	1	1	1	1	1	1	1	1	1	1	WT 8813 YH	Oil mist in crankcase/cylinder
										1	TE 1310 YH	Lubricating oil inlet (only ME-GI w. separate HPS)

1. Indicates that the sensor is required.
The sensors in the MAN ES and relevant Class columns are included in the basic Extent of Delivery, EoD: 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 certain engines only, see the list in Section 18.06, Axial Vibration Monitor.

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Table 18.04.03: Slow down functions for UMS

Shut down for AMS and UMS - Class and MAN Energy Solutions' requirements

ABS	BV	CCS	DNV	KR	LR	NK	RINA	RS	IACS	MAN -ES	Sensor & function	Point of location
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	ZT 4020 Z	Engine overspeed
1	1	1		1			1	1	1	1	TE/TS 8107 Z	Thrust bearing segment
			☐							1	XS 8813 Z	Oil mist in crankcase/cylinder
										1	XS 8817 Z	Turbocharger overspeed (only in case of EGR or EGB, VT TC, power turbine/hybrid TC, TC cut-out and system handshake, see table 18.06.03

1. Indicates that the sensor is required.

- The sensors in the MAN ES and relevant Class columns are included in the basic Extent of Delivery, EoD: 4 75 124.

  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 or slow down, option: 4 75 134

Se also table 18.04.03: Slow down functions for UMS.

Table 18.04.04: Shut down functions for AMS and UMS, option 4 75 124

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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
KR	Korean Register
LR	Lloyd's Register
NK	Nippon Kaiji Kyokai
RINA	Registro Italiano Navale
RS	Russian Maritime Register of Shipping

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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 ME/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
TI 8410	TT 8410	Cylinder cover cooling water outlet, cylinder (Only for LDCL)
		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/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	
		Fuel oil
PI 8001	PT 8001	Fuel oil, inlet engine
		Lubricating oil
PI 8103	PT 8103	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
		High temperature jacket cooling water, jacket cooling water
PI 8401	PT 8401	Jacket cooling water inlet
	PS/PT 8402	Jacket cooling water inlet (Only Germanischer Lloyd)
	PDT 8403	Jacket cooling water across engine (or PT 8401 and PT 8413) (Not for LDCL)
	PDT 8404	Jacket cooling water across cylinder liners (Only for LDCL)
	PDT 8405	Jacket cooling water across cylinder covers and exhaust valves (Only for LDCL)
	PT 8413	Jacket cooling water outlet, common pipe
		Low temperature cooling water, seawater or freshwater for central cooling
PI 8421	PT 8421	Cooling water inlet, air cooler
		Compressed air
PI 8501	PT 8501	Starting air inlet to main starting valve
PI 8503	PT 8503	Control air inlet
	PT 8505	Air inlet to air cylinder for exhaust valve (Only ME-B)
		Scavenge air
PI 8601	PT 8601	Scavenge air receiver (PI 8601 instrument same as PI 8706)
PDI 8606	PDT 8606	Pressure drop of air across cooler/air cooler
		Exhaust gas
PI 8706		Exhaust gas receiver/Exhaust gas outlet turbocharger
		Miscellaneous functions
PI 8803		Air inlet for dry cleaning of turbocharger
PI 8804		Water inlet for cleaning of turbocharger (Not applicable for MHI turbochargers) Table

Table 18.05.01b: Local pressure gauges on engine, options: 4 70 119, and remote indication sensors, option: 4 75 127

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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
		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 8212	Small tank for heating element, low level (Not for ACOM)
	XC 8265	ACOM (Only for ACOM)
	LS 8285	Level switch
		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
	XS/XT 8817	Turbocharger overspeed (Only in case of EGB, VT TC, power turbine/ hybridTC, TC Cut-out, see Table 18.06.03)

Table 18.05.01c: Other indicators on engine, options: 4 70 119, and remote indication sensors, option: 4 75 127

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Engine protection systems and alarms

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 in the basic design of MAN B&W engines.

Bearing Condition Monitoring

Based on our experience, we decided in 1990 that all plants must include an oil mist detector specilfied by MAN Energy SolutionsSince 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 Bear-ing 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 Energy Solutions requires that the oil mist detector is connected to the shut down system.

For personnel safety, the oil mist detectors and related equipment are located on the manoeuvring side of the engine.

The following oil mist detectors are available:

4 75 162	Graviner Mk 7, make: Kidde Fire Protection
4 75 163	Visatron VN 215/93, make: Schaller Automation GmbH & Co. KG *)
4 75 166	MD-SX, make: Daihatsu Diesel Mfg. Co., Ltd.
4 75 167	Vision III C, make: Specs Corporation
4 75 168	GDMS-OMDN09, make: MSS AG

4 75 271	Triton, make: Heinzmann GmbH & Co. KG
4 75 272	Visatron VN301^{plus}, make: Schaller Automation GmbH & Co. KG
4 75 273	MOT-2R5M7R5MP, make: Meiyo Electric Co., Ltd.
*) Only applicable for S50ME-C8/-GI as well as MC-C and ME-B/-GI/-LGI types 50 and smaller	

Examples of piping diagrams (for Visatron VN 215/93 only) and wiring diagrams (for all other detectors) are shown for reference in Figs. 18.06.01a and 18.06.01b.

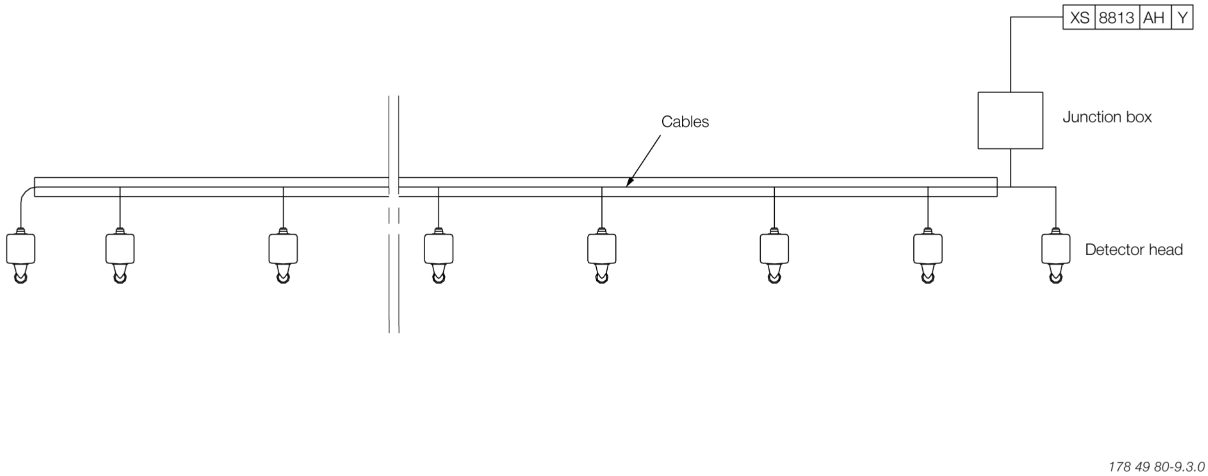

Fig. 18.06.01a: Example of oil mist detector wiring on engine

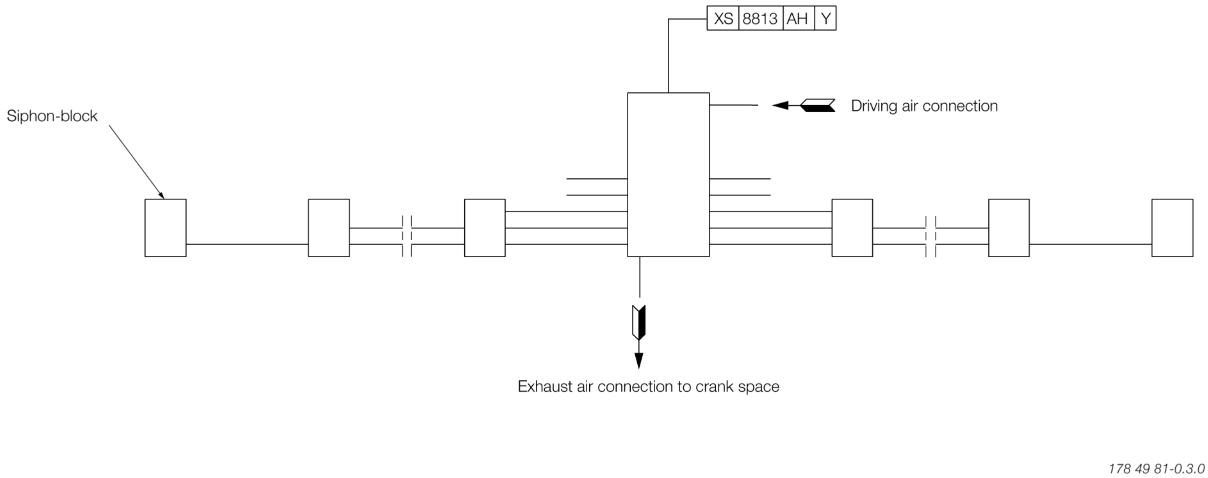

Fig. 18.06.01b: Oil mist detector pipes on engine, type Visatron VN215/93 from Schaller Automation, option: 4 75 163

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.

he Extent of Delivery lists the following Bearing Wear Monitoring options:

4 75 261	XTS–W (BWM), make: AMOT
4 75 262	BDMS (BW&TMS), make: Dr. E. Horn
4 75 263	BWCM, make: Kongsberg Maritime
4 75 265	B-WACS, make: Doosan Engine Co., Ltd.
4 75 266	BWCMS, make: KOMECO
4 75 267	BCM-1, make: Mitsui Zosen Systems Research Inc.

ME, ME-C/-GI/-LGI 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

All MAN B&W engines are as standard specified with Water In Oil monitoring system in order to detect and avoid free water in the lubricating oil.

In case the lubricating oil becomes contaminated with an amount of water exceeding our limit of 50% of the saturation point (corresponding to approx. 0.2% water content), 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 saturation point of the water content in the lubricating oil varies depending on the age of the lubricating oil, the degree of contamination and the temperature. For this reason, we have chosen to specify the water activity measuring principle and the aw-type sensor. Among the available methods of measuring the water content in the lubricating oil, only the aw-type sensor measures the relationship between the water content and the saturation point regardless of the properties of the lubricating oil.

WIO systems with aw-type sensor measure water activity 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.

Alarm levels are specified as follows:

Engine condition	Water activity, aw
High alarm level	0.5
High High alarm level	0.9

The aw = 0.5 alarm level gives sufficient margin to the satuartion point in order to avoid free water in the lubricating oil. If the aw = 0.9 alarm level is reached within a short time after the aw = 0.5 alarm, this may be an indication of a water leak into the lubricating oil 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, thrust 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.

Axial Vibration Monitor

For functional check of the vibration damper a mechanical guide is fitted, while an electronic vibration monitor can be supplied as an option.

An Axial Vibration Monitor (AVM) with indication for condition check of the axial vibration damper and terminals for alarm and slow down ia available as an option: 4 31 117. It is required for the following engines:

- All ME-C9/10 engines incl. their -GI and -LGI variants
- All ME-C7/8 engines with 5 and 6 cylinders incl. their -GI and -LGI variants
- K-ME-C6/7 and K98ME6/7 engines with 11 and 14 cylinders incl. their -GI and -LGI variants.

The requirement for AVM on 4-cylinder engines is available on request.

The alarm and slow down system should include the filtration necessary to prevent the AVM from unintentionally activating the alarm and slow down functions at torsional vibration resonances, i.e. in the barred speed range, and when running Astern.

In the low speed range and when running Astern, the alarm and slow down functions are to be disabled so that the AVM only gives an indication of the vibration level.

The AVM alarm and slow down functions shall be enabled when the engine is running Ahead and at speeds above the barred range.

To prevent rapid hunting of the engine speed in a slow down situation, a holding time function has been introduced in order to delay the automatic re-setting of the slow down function.

The specification of the AVM interface to the alarm and slow down system is available from MAN Energy Solutions Copenhagen.

(LDCL) Cooling Water System

With the Load Dependent Cylinder Liner (LDCL) cooling water system, the cooling water outlet temperature from the cylinder liner is controlled relative to the engine load, independent of the cooling water outlet from the cylinder cover.

The interval for the liner outlet may be wide, for instance from 70 to 130 degree Celsius. The cooling water outlet temperature is measured by one sensor for each cylinder liner of the engine.

For monitoring the LDCL cooling water system the following alarm and slow down functionality must be fulfilled:

The Alarm system must be able, from one common analog sensor, to detect two alarm limits and two slow down limits as follows:

• Upper slow down limit
• Upper alarm limit
• Load dependent slow down limit
• Load dependent alarm limit.

An example of the limits is shown in Fig. 18.06.02.

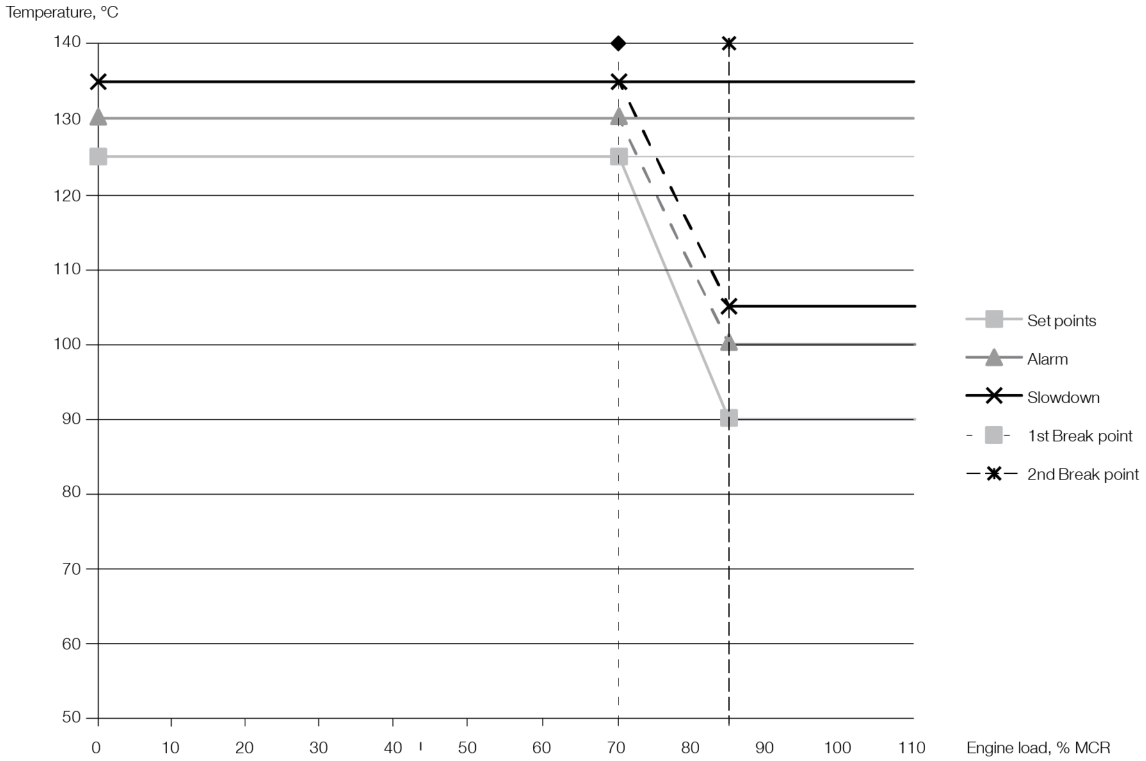

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Fig. 18.06.02: Example of set points versus slow down and alarm limits for LDCL cooling water system

The load dependent limits must include at least one break point to allow cut-in/-out of the lower limits. The upper limits are fixed limits without breakpoints.

The values of the load dependent limits are defined as a temperature difference (DT) to actual cooling water temperature (which vary relative to the engine load).

The cooling water temperature is plant dependent and consequently, the actual values of both upper limits and load dependent limits are defined during commissioning of the engine.

All 95-50ME-C10/9/-GI dot 2 and higher as well as G50ME-B9.5/.3 and S50ME-B9.5 are as standard specified with LDCL Cooling Water Monitoring System while S50ME-B9.3 and G45ME-C9.5/-GI are prepared for the installation of it.

Motor Start Method

Direct Online Start (DOL) is required for all the electric motors for the pumps for the Load Dependent Cylinder Liner (LDCL) to ensure proper operation under all conditions

Turbocharger Overspeed Protection

All engine plants fitted with turbocharger cut-out, exhaust gas bypass (EGB), power turbine / turbo generator (PT), hybrid turbocharger or variable tur - bocharger (VT) run the risk of experiencing turbo - charger overspeed.
To protect the turbocharger, such plants must be equipped with a turbocharger overspeed alarm and slow-down function.

However, the handshake interface between the ship's power management system and a waste heat recovery system (WHRS) or a shaft generator (SG) may delay the slowdown for up to 120 seconds.
Therefore, the slow-down function must be upgraded to a non-cancellable shutdown for engine plants with handshake interface.

On engine plants designed with exhaust gas recirculation (EGR), a sudden increase of energy to the turbocharger(s) will occur if the EGR system trips.
As protection, turbocharger overspeed alarm and non-cancellable slowdown must be fitted.

Consequently, the turbocharger speed must be monitored by the ship alarm system and the safety system(s), triggering slowdown or non-cancellable shutdown if the turbocharger speed exceeds the defined alarm levels.

The protection applicable for individual engine plant and power management configurations is summarised in Table 18.06.03.

Turbocharger overspeed protection		
Engine plant configuration	No power management system handshake	Engine with WHR or shaft generator with power management system handshake
Traditional exhaust gas train and turbocharger	No monitoring of turbocharger overspeed	No monitoring of turbocharger overspeed
Exhaust gas bypass, variable turbo charger, power turbine or hybrid turbocharger	Turbocharger overspeed slowdown	Turbocharger overspeed shutdown
Exhaust gas recirculation	Turbocharger overspeed slowdown	Turbocharger overspeed shutdown
Turbocharger cut-out	Turbocharger overspeed slowdown	Turbocharger overspeed shutdown

Table 18.06.03: Turbocharger overspeed protection for individual engine plant configurations

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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 18.06.04 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 bypass
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, Alpha/ME lubricator
ZV 8281 C	Solenoid valve, lubricator activation
ZT 8282 C	Feedback sensor, lubricator feedback
	Cylinder lubricating oil, Alpha Mk 2 lubricator

XC 8288 C	Propoprtional valve
ZT 8289 C	Feedback sensor
	Scavenge air
PS 8603 C	Scavenge air receiver, auxiliary blower control
	ME-GI alarm system (ME-GI only)
XC 2212	External gas shut down (request)
	ME-GI safety system (ME-GI only)
XC 2001	Engine shut down (command)
XC 6360	Gas plant shut down (command)

Table 18.06.04b: Control devices on engine

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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 or Indicating Variables

First letters:

DS	Density switch
DT	Density transmitter
E	Electrical component
FS	Flow switch
FT	Flow transmitter
GT	Gauging transmitter, index/load transmitter
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
TT	Temperature transmitter
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 (limit switch)
ZT	Position transmitter (proximity sensor)
ZV	Position valve (solenoid valve)

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Location of Measuring Point

Ident. number; first two digits indicate the measurement point and xx the serial number:

11xx	Manoeuvring system
12xx	Hydraulic power supply system (HPS)
13xx	Hydraulic control oil system, separate oil to HPS
14xx	Combustion pressure supervision
15xx	Top bracing pressure, stand alone type
16xx	Exhaust Gas Recirculation (EGR)
20xx	ECS to/from safety system
21xx	ECS to/from remote control system
22xx	ECS to/from alarm system
24xx	ME ECS outputs
29xx	Power supply units to 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
60xx	GI-ECS to Fuel Gas Supply System (FGSS)
61xx	GI-ECS to Sealing Oil System
62xx	GI-ECS to Control Air System
63xx	GI-ECS to other GI related systems
64xx	GI engine related components
66xx	Selective Catalytic Reduction (SCR) related component. Stand alone
80xx	Fuel oil system
81xx	Lubricating oil system
82xx	Cylinder lubricating oil system
83xx	Stuffing box drain system
84xx	Cooling water systems, e.g. central, sea and jacket cooling water
85xx	Compressed air supply systems, e.g. control and starting air
86xx	Scavenge air system
87xx	Exhaust gas system

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88xx	Miscellaneous functions, e.g. axial vibration
90xx	Project specific functions
A0xx	Temporary sensors for projects
xxxx-A	Alternative redundant sensors
xxxx-1	Cylinder/turbocharger numbers
	ECS: Engine Control System
	GI: Gas Injection engine
	VOC: Volatile Organic Compound

Table 18.07.01a: Identification of instruments

Functions

Secondary letters:

A	Alarm
C	Control
H	High
I	Indication, remote
L	Low
R	Recording
S	Switching
X	Unclassified function
Y	Slow down
Z	Shut down

Repeated Signals

Signals which are repeated, for example measurements 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 two redundant position switches in the manoeuvring system, A and B, for control of the main starting air valve position.

PT 8501 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.

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Table 18.07.01b: Identification of instruments

ME-LGIM safety aspects

General

The normal safety systems incorporated in the fuel oil systems are fully retained also during dual fuel operation. However, additional safety devices will be incorporated in order to prevent situations which might otherwise lead to failures.

Safety devices – external systems

Leaking SF components in engine room

All components in the engine room containing SF like valves and pipes are encapsulated to prevent leakage to engine room and the duct is ventilated 30 times pr. hour. At the ventilation outlet two Hydro Carbon (HC) sensors detects if there is a leakage. If HC level on just one sensor exceeds 30% (or 20%) LEL an alarm is released and if it exceeds 60% (or 40%) a SF shutdown is released with alarm.

Assembly test before filling SF system on engine

To improve safety in the engine room, an assembly test is performed by pressurizing the SF system on engine with inert gas followed by pressure drift monitoring. The test has to pass before opening the SFVT and sending SF to the engine.

Lack of outer pipe ventilation

As mentioned above a SF leakage to outer pipe will be detected at the outlet of the ventilation of the outer pipe. To make sure the detection will be reasonably fast it is required to exchange the ventilated volume within 2 min (or 30 times pr. hour). By having flow switches at the inlet of the ventilation adjusted for the required exchange of volume it is possible to detect lack of flow. If flow switches show insufficient flow a SF shutdown is performed and an alarm is released.

Sealing oil pressure failure

Sealing oil prevents SF from entering the control oil by sealing between the control oil and SF pressure inside the SF FBIV on the moving parts with a higher pressure than the SF supply pressure. A certain sealing oil pressure above SF supply pressure is required. If just one of two sealing oil pressure sensors drop below the minimum sealing oil pressure SF running is stopped and an alarm is released. If the sealing oil pressure drops further below a SF shutdown limit a SF shutdown and alarm is released.

Beside sealing inside the FBIV a separate lower sealing oil pressure with separate pressure monitoring (two pressure sensors) and evaluation is used for the ELBI valve to prevent SF to enter the ELBI control oil. If the sealing oil pressure drops below a minimum pressure a SF Stop and associated alarm is released and if it drop below a SF shutdown limit a SF shutdown and associated alarm is released.

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Safety devices – internal systems

The SF fuel booster injection (FBIV) can fail by:

- not opening when commanded or delayed opening due to seize of the sliding valve inside the FBIV
- not closing completely or hanging fully open when commanded to close
- clogged atomizer will reduce the expected amount of SF injected
- malfunctioning ELFI-L valve.

The FO FBIV injection can fail by:

- reduced or no FO pilot injection or delayed injection due to wear or seized sliding valve inside the FBIV
- malfunctioning ELFI/FIVA

All failures mentioned above will affect the combustion of the FO and SF injected into the combustion chamber. The cylinder pressure is monitored and evaluated at every engine revolution to detect for compression failure, too high max. pressure, combustion failure, misfiring etc. One single failure results in SF shutdown.

A malfunctioning injection system often results in pure combustion which will affect the exhaust gas temperature. The exhaust gas temperatures are monitored as explained in the following section.

ME-LGIM SF control and safety, cause-and-effect table

ME-LGIM Mk. 1.2 SF control and safety, cause and effect is presented in Table 18.08.01.

		Alarm	SF Stan dby	SF Stop	SF Shut-down	
A	SF plant safety functions					
1	SF pressure supervision:					
	Min inlet SF pressure	x			x	2 bar
	Max inlet SF pressure	x			x	15 bar
2	Sealing oil pressure difference to SF pressure, low	x			x	SF pressure +10 bar
3	External SF shutdown, causes:	x			x	
	a) Exhaust temperature high deviation (MPC control platform only)					normally +/- 60° C, shall be equal to the engine slowdown level
	b) Exhaust temperature high (individually per cylinder) (MPC control platform only)					normally 470° C, shall be equal to the engine slowdown level
	c) Engine room (and other rooms containing SF equipment) fire detection					
	d) Shutdown of SF supply system					
	e) Engine room HC detector (if present)					

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4	Double-wall pipe HC sensors, concentration (one out of two)					
	a) Double-wall pipe HC sensors high concentration	x				20 - 30% LEL (level dependent of class requirement)
	b) Double-wall pipe HC sensors high-high concentration	x			x	40 - 60% LEL (level dependent of class requirement)
5	Double-wall pipe ventilation flow switch, low	x			x	
6	Manual SF shutdown from:	x			x	
7	a) Bridge	x			x	
	b) ECR	x			x	
	c) Machinery space (LOP)	x			x	
	d) External to machinery space	x			x	
8	Engine shutdown	x			x	
9	Hydraulic control oil pressure low	x			x	180 bar
10	Feedback error of SF train safety block valve (801)	x			x	
11	Feedback error of SF train bleed valve (802)	x			x	
12	Feedback error of SF return safety valve	x			x	Return pipe test valve (836)
13	Sensor failure (wire breakage) of SF plant safety unit	x			x	SF pressure, seal oil pressure, LGI SD, double wall pipe flow switch, double-wall pipe HC sensor
14	Controller failure of SF plant safety control	x			x	Power failure and Internal failure
15	Seal oil tank:					
	Low-pressure seal oil outside range	x			x	Min. pressure =12 bar, maximum pressure = 20 bar
	High-pressure seal oil outside range	x			x	OK when high-pressure seal oil pressure > SF inlet pressure + 15 bar
16	Sensor failure sealing oil tank	x			x	Sealing oil press., tank temp. tank level min./max., flowswitch
B	SF cylinder safety unit functionality					
1	Cylinder pressure abnormal	x			x	Max. pressure high, compression pressure low, expansion pressure low
2	ELBI valve not OK	x			x	Valve status failure (power and internal failure)
3	Tacho system failure	x			x	
4	Sensor failure (wire breakage) of SF cylinder safety unit	x			x	Cylinder pressure, cylinder SF pressure, ELBI, tacho system

5	SCSU controller failure of SF cylinder safety control	x			x	Power failure and Internal failure
C	SF plant control units functionality					
1	Triton platform: exhaust gas monitoring wired to ME-ECS	x		x		
	a) Exhaust temperature high deviation					Normallly +/- 50° C.
	b) Exhaust temperature high (individually per cylinder)					Normally 470° C
2	SF temperature supervision:					
	a) SF temperature out of range	x				Alarm: Outside range: 25 - 55°C, Alarm: During starting period below: 10-25° C (600s)
	b) SF temperature out of range	x		x		SF stop: Outside range: 15 - 60° C, SF stop: During starting period below: 0-15° C (600s)
3	SF supply system signals:					
	a) SF supply pressure too low compared to setpoint	x		x		
	b) Local/fail (= not ready)	x		x		Ready signal off
4	Inert gas pressure low	x				5 bar
5	High sealing oil pressure difference to SF pressure	x		x		High deal oil pressure setpoint = SF press.+35bar. Max seal pressure above setpoint: 5 bar max seal pressure Below setpoint: 15 bar
6	Double wall pipe ventilation flow switch, low	x		x		
7	Hydraulic control oil pressure low	x		x		180 bar
8	Valve control air pressure low	x		x		6 - 7 bar
9	Dry air flow low	x		x		
10	ELGI valve not OK	x		x		Valve status failure (power and internal failure)
11	Feedback error of SF valve train valves	x		x		Plant main block valve, bleed valve
12	Feedback error of SF return pipe manual valve	x		x		Return pipe manual valve (856)
13	Feedback error of Inert gas valves	x		x		Inert gas supply valve (806), Inert gas bleed valve (808), Inert gas block valve (809)
14	Return pipe purge block failure	x		x		Return pipe purge block is in failure state
15	Sensor failure (wire breakage) of control unit	x		x		Control wire for ventilation, SFSS, Inert gas system, sealing oil pump and SFVT signals, Sealing oil unit (SOU)
16	Controller failure of SF plant control	x			x	Power failure and Internal failure

D	SF Stop					
1	Activate dual fuel stop button on MOP			x		
E	SF Standby					
1	Engine load < minimum LGI running load		x			

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Table 18.08.01: Second fuel control and safety, cause-and-effect table

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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 Energy Solutions' 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 Energy Solutions' 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 precertification survey on engine plants with FPP or CPP, options: 4 06 201 Engine test cycle E3 or 4 06 202 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 Energy Solutions, 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 Energy Solutions

The aboveœmentioned set of spare parts can be extended with the 'Additional Spare Parts Recommended by MAN Energy Solutions', 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.

The wearing parts that, based on our service experience, are estimated to be required, are 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.

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Tools

List of standard tools

The engine is delivered with the necessary special tools for overhauling purposes. The extent, dimen sions 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 pan els, 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.

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Specification for Painting of main Engine

Components to be painted before shipment from work-shop	Type of paint	No. of coats Total Nominal Dry Film Thickness (NDFT) µm	Colour: RAL 840 HR, DIN 6164, MUNSELL
1. Components/surfaces, inside engine, exposed to oil and air: Unmachined surfaces all over. However, cast type crank throws, main bearing cap, crosshead bearing caps, crankpin bearing cap, pipes inside crankcase and chain-wheel need not be painted but the cast surface must be cleaned of sand and scales and kept free of rust.	In accordance with corrosivity categories C2 Medium ISO 12944-5		Free
	Engine alkyd primer, weather resistant.	1 - 2 layer Total NDFT 80 µm	
	Oil and acid resistant alkyd paint. Temperature resist-ant to min. 80°C.	1 layer Total NDFT 40 µm	White: RAL 9010 DIN N:0:0,5 MUNSELL N-9.5
		Total NDFT: 120 µm	
2. Components, outside engine: Engine body, pipes, gallery brackets, etc. Delivery standard is in a primed and finally painted condition, unless otherwise stated in the con-tract. See also MAN ES EN690D for colour marking of piping systems	In accordance with corrosivity categories C2 Medium ISO 12944-5		Free
	Engine alkyd primer, weather resistant.	1 - 2 layer Total NDFT 80 µm	
	Final alkyd paint, resistant to salt water and oil	1 layer Total NDFT 40 µm	Light green: RAL 6019 DIN 23:2:2 MUNSELL 10GY 8/4
		Total NDFT: 120 µm	
3. Low flashpoint fuels: See also MAN ES EN690D for colour marking of piping systems. Chain pipe, supply pipe. Spool piece.	In accordance with corrosivity categories C2 Medium ISO 12944-5		Free
	Engine alkyd primer, weather resistant.	1 - 2 layer Total NDFT 80 µm	
	Final alkyd paint, resistant to salt water and oil	1 layer Total NDFT 40 µm	Yellow: RAL 1021 MUNSELL 2.5Y 8/14
		Total NDFT: 120 µm	

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Components to be painted before shipment from workshop	Type of paint	No. of coats Total Nominal Dry Film Thickness (NDFT) µm	Colour: RAL 840 HR, DIN 6164, MUNSELL
4. Heat affected components:	In accordance with corrosivity categories C3 Medium ISO 12944-5		
Supports for exhaust receiver. Air cooler housing inside and outside. No surface may be left unpainted in the cooler housing. Exhaust valve housing (exhaust flange), (Non water cooled only)	Ethyl silicate based zincrich paint, heat resistant to min. 300 ºC.	1 layer Total NDFT 80 µm	
5. Components affected by water, cleaning agents, and acid fluid below neutral Ph:	In accordance with corrosivity categories C5-M High ISO 12944-5		Free
Scavenge air cooler box inside. (Reversing chamber). Preparation, actual number of coats, film thickness per coat, etc. has to be according to the paint manufacturer specifications. Air flow reversing chamber inside and outside. No surfaces may be left unpainted Supervision from manufacturer is recommended, in the phase of introduction of the paint system.	Two component epoxy phenolic	3 layer Total NDFT 350 µm See specifications from product data sheet	
6. Gallery plates topside:	Engine alkyd primer, weather resistant	C2 medium 1 - 2 layer Total NDFT 80 µm	

Components to be painted before shipment from workshop	Type of paint	No. of coats Total Nominal Dry Film Thickness (NDFT) µm	Colour: RAL 840 HR, DIN 6164, MUNSELL
7. EGR-system - Mixing chamber *) To be applied after Water Mist Catcher (WMC) to Non-return Valve at scavenge air reciever. See figure 2 for details. Optional: EGR paint can be applied from Air cooler outlet, (reversing chamber). See figure 2 for details. *) Only for engines specified with EGR or prepared for EGR installation	Vinyl ESTER Acrylic copolymer	Total NDFT 500 -1200 µm	Free
8. Purchased equipment and instruments painted in makers colour are acceptable, unless otherwise stated in the contract:			
Tools are to be surface treated according to specifications stated on the drawings. Purchased equipment painted in markers colour is acceptable, unless otherwise stated in the contract/ drawing.	Electro(-) galvanised	See specifications from product data sheet	
Tool panels	Oil resistant paint	1 - 2 layer Total NDFT 80 µm	Light grey: RAL 7038 DIN 24:1:2 MUNSELL N-7.5
9. Lifting points: Pad eyes, wholes, clamps, threaded wholes, eye screws, eye nuts and other lifting points	Alkyd paint, resistant to water, lubricants, hydraulic oil and degreaser.	1 layer Total NDFT 80 ym (my)	Yellow: RAL 1021 MUNSELL 2.5y 8/14

All paints must 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., must be in accordance with the paint manufacturer's specifications.

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Fig. 19.02.01: Painting of main engine, option: 4 81 101, 4 81 102 or 4 81 103

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Dispatch pattern

General

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.

Dispatch pattern A – short:

Short distance transportation limited by duration of transportation time within a few days or a distance of approximately 1,000 km and short term storage.

Duration from engine delivery to installation must not exceed eight weeks.

Dismantling must be limited.

Dispatch pattern B – long:

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.

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)

Dispatch pattern variants

A1 + B1 (option 4 12 021 + 4 12 031)
Engine complete, i.e. not disassembled

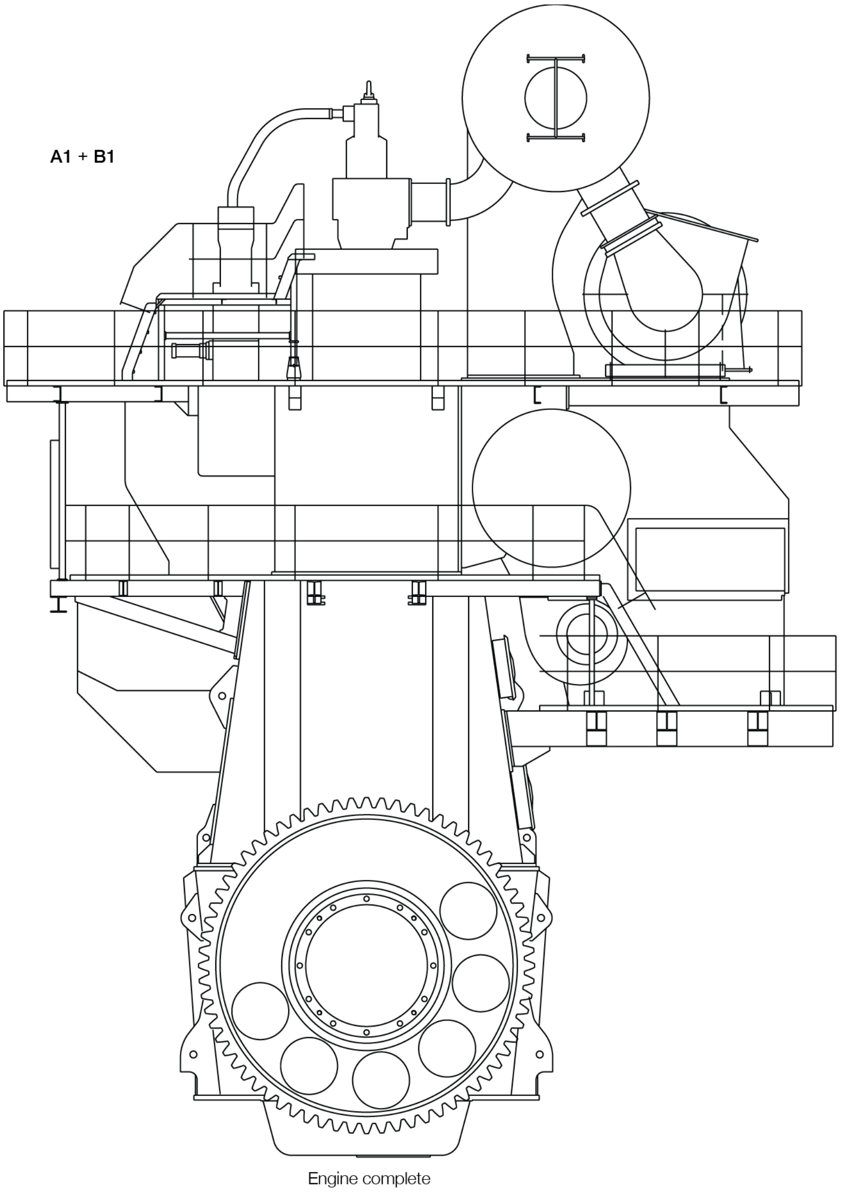
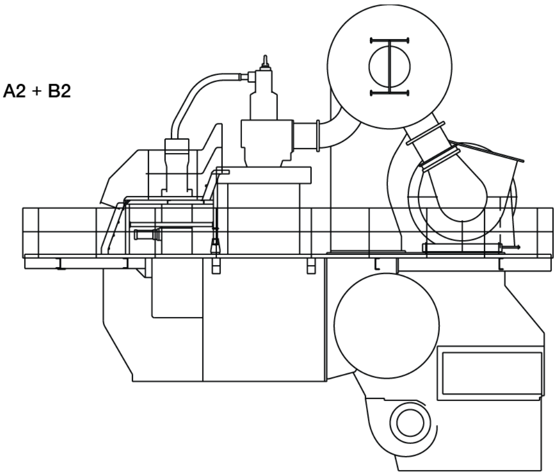


Fig. 19.03.01: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

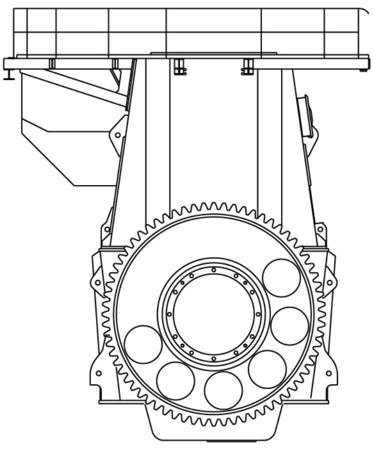
A2 + B2 (option 4 12 022 + 4 12 032)

- Top section including cylinder frame complete, cylinder covers complete, scavenge air receiver including cooler box and cooler insert, turbochar-ger(s), piston complete and galleries with pipes, HCU units, oil filter, gas control blocks, gas chain pipes and sealing oil pump unit
- Bottom section including bedplate complete, frame box complete, con-necting rods, turning gear, crankshaft complete and galleries
- Remaining parts including stay bolts, chains, multi-way valves, etc

A2 + B2



Top section



Bottom section

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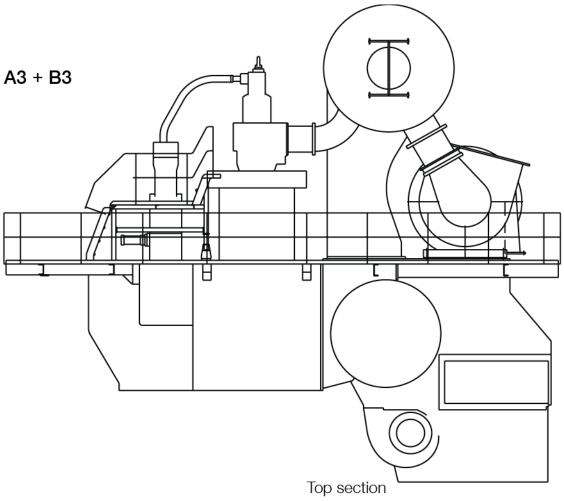
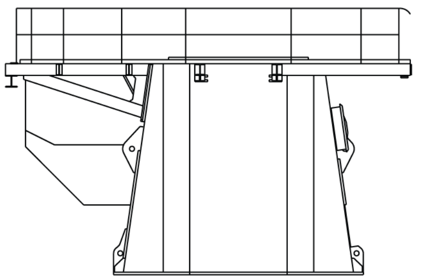
Fig. 19.03.02: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

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, gas control block, gas chain pipes, and sealing oil pump unit
- Frame box section including frame box complete, chain drive, connecting rods and galleries, gearbox for hydraulic power supply, hydraulic pump station and oil flter
- Bedplate/crankshaft section including bedplate complete, crankshaft complete with chainwheels and turning gear
- Remaining parts including stay bolts, chains, multi-way valves, etc.

A3 + B3





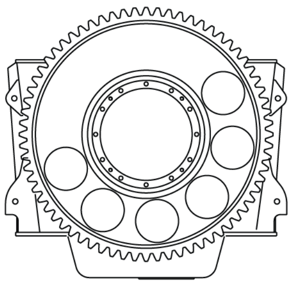


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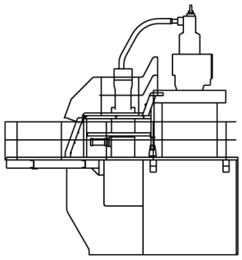
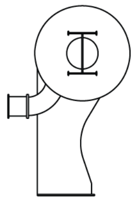
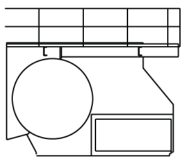
Fig. 19.03.03: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

A4 + B4 (option 4 12 024 + 4 12 034)

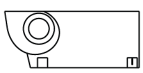
- Top section including cylinder frame complete, cylinder covers complete, piston complete and galleries with pipes on manoeuvring side, HCU units, gas control block, gas chain pipe, and sealing oil pump unit
- Exhaust receiver with pipes
- Scavenge air receiver with galleries and pipes
- Turbocharger
- 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 flter
- Crankshaft with chain wheels
- Bedplate with pipes and turning gear
- Remaining parts including stay bolts, auxiliary blowers, chains, multi-way valves, etc.

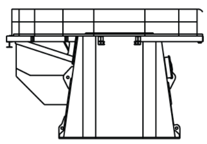

Top section

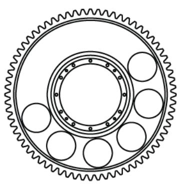

Exhaust receiver


Scavenge air receiver


Turbocharger


Air cooler box


Frame box section


Crankshaft section

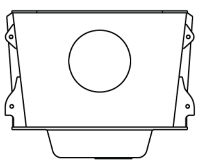

Bedplate section

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Fig. 19.03.04: Dispatch pattern, engine with turbocharger on exhaust side (4 59 123)

Dispatch Pattern, List of Masses and Dimensions

This section is available on request

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Shop test

The minimum delivery test for MAN B&W two-stroke engines, EoD: 4 14 001, involves:

- Starting
- Load test
- Engine to be started and run up to 50% of Specified MCR (M) in 1 hour.

and is followed by the below mentioned tests.

Load test at specific load points

The engine performance is recorded running at:

- 25% of specified MCR
- 50% of specified MCR
- 75% of specified MCR
- 90% of specified MCR or at NCR
- 100% of specified MCR *)
- 110% of specified MCR

Records are to be taken after 15 minutes or after steady conditions have been reached, whichever is shorter.

*) Two sets of recordings are to be taken at a minimum interval of 30 minutes.

Governor test and more:

- Integration test of ECS
- 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).

Second fuel test

Further to the minimum delivery test, in the shop test on second fuel, EoD: 4 14 041 (-GI) and 4 00 042 (-LGI), the engine is to be carefully tested on both diesel oil and second fuel. The test is carried out in the presence of representatives of Yard, Shipowner, Classification Society, and MAN Energy Solutions.

Fuel oil and second fuel analyses are to be presented. All load point measurements are to be carried out on diesel or gas oil as well as on second fuel and must include:

- test of auto change-over to second fuel from second fuel standby condition when engine load exceeds the lowest limit for second fuel operation
- test of auto change-over to fuel oil when engine load falls below the lowest limit for second fuel operation
- demonstration of change-over between fuel oil and second fuel at each load stop

- second fuel shut down at 110% load, resulting in continued running of main engine on primary fuel.

The shop tests are all carried out according to:

Factory Acceptance Test and Shipboard Trials of I.C. Engines, UR M51

by International Association of Classification Societies LTD. (IACS), www.iacs.org.uk

Vibration aspects

Torsional vibration

The installation aspects in a shop test and on a vessel are different. As a result, the torsional vibration characteristics are also different, and special countermeasures may apply during the shop test.

To reduce the stress amplitudes in the shafting system at the main critical resonance, a tuning wheel is commonly applied on 5-7 cylinder engines. In a shop test, the shaft between the engine and water brake is typically short and stiff compared to the shafting system connecting the propeller to the engine on a vessel. Due to the different installation aspects, a torsional vibration calculation of the shop test conditions is always made to ensure acceptable vibrations. It is common that the tuning wheel is omitted at the shop test to avoid excessive crankshaft stresses.

Regarding SFOC, the power absorbed in the tuning wheel and main bearing is very small. Based on elasto-hydrodynamic simulations of the foremost main bearing, the power loss with a large tuning wheel is 0.033% of engine power and 0.032% without a tuning wheel. Therefore, this power loss can be ignored in terms of the SFOC measurement conducted in a shop test.

In some cases, a torsional vibration damper has to be applied on the water brake side to achieve acceptable vibration conditions.

Structural vibration

In most cases, the vibration level and behaviour of the main engine is quite different when comparing test bed trials with sea trials. The main reason for this is the strong influence and dynamic interaction with the surroundings, the most important being:

1. The engine seating stiffness (foundation) is lower for the test bed compared to a vessel's double bottom. This will shift vibration resonances to lower engine speeds, which results in significantly different vibration levels and resonance speeds.
2. If top bracings are installed (vessel installation), this additional stiffness will not be present at the shop test (due to lack of stiffness/strength of the surrounding building). Some licensees may mount an electrically driven moment compensator (EMC) temporarily to reduce vibration levels during shop tests.

The global vibration behaviour of an engine erected on a test bed cannot, and should not, be compared to the vibration levels of a vessel-installed engine. Local vibrations of turbochargers and exhaust gas receivers are also strongly influenced by the dynamic interaction with the surroundings, be it a "soft test bed" or a "stiff ship hull".

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EIAPP certificate

Most marine engines installed on ocean going vessels are required to have an ‘Engine International Air Pollution Prevention’ (EIAPP) Certificate, or similar. Therefore, a pre-certification survey is to be carried out for all engines according to the survey method described in the engine’s NOx Technical File, which is prepared by the engine manufacturer. For MAN B&W engines, the Unified Technical File (UTF) format is recommended.

The EIAPP certificate documents that the specific engine meets the international NOx emission limitations specified in Regulation 13 of MARPOL Annex VI. The basic engine ‘Economy running mode’, EoD: 4 06 200, complies with these limitations.

The pre-certification survey for a ‘Parent’ or an ‘Individual’ engine includes NOx measurements during the delivery test. For ‘Member’ engines, a survey according to the group definition for the engine group is needed. This survey should be based on the delivery test.

The applicable test cycles are:

- E3, marine engine, propeller law for FPP, option: 4 06 201

  or

- E2, marine engine, constant speed for CPP, option: 4 06 202

For further options regarding shop test, see Extent of Delivery.

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List of spare parts, unrestricted service

Spare parts are requested by the following Classification Society only: NK, while just recommended by: ABS, DNV, CRS, KR, LR and RS, but neither requested nor recommended by: BV, CCS and RINA.

The final scope of spare parts is to be agreed between the owner and engine builder/yard.

LGI Cylinder Cover, Plate 2272-0300 (901 and more)

1 Cylinder cover complete with extra holes for LGI equipment and Inconel cladding.
Including fuel, exhaust and starting valves, indicator valve, cooling jacket and sealing rings (disassembled)
½ set Studs for 1 cylinder cover.

LFL Pipe Supply Chain, Plate 4272-2800 (901 and more)

1 set Double-wall LFL (Low flash point liquid) pipes for 1 cylinder.
1 set Repair kit for high-pressure LFL pipes including gaskets and packings.

Piston and Piston Rod, Plates 2272-0400/0420/0500 (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 2272-0600 (903)

1 Cylinder liner complete, including cooling jacket, non-return valves, sealing rings and gaskets (assembled)

Cylinder lubricating Oil System, Plates 3072-0600,6670-0100 (903) 1)

1 Lubricator complete
1 Inductive sensor
1 set O-rings and seals
2 Lubricator backup cable

Connecting Rod, and Crosshead Bearing, Plates 1472-0300,2572-0300/0200 (904)

1 Telescopic pipe with bushing for 1 cylinder
1 Crankpin bearing shell (1 upper and 1 lower part) with studs and nuts
1 Crosshead bearing shell lower part with studs and nuts

Thrust block, plate 2572-0600 (905)

1 set Thrust pads, complete FWD set for ‘Ahead’
1 set For KR and NK also 1 set ‘Astern’ if different from ‘Ahead’

HPS , Hydraulic Power Supply, Plates 4572-1000/0750,4572-1100/1200/1250 (906) 1 and 2)

1 Proportional valve for hydraulic pumps
1 Leak indicator
1 Drive shaft for hydraulic pump, of each type (length)
1 Membrane plus seals for accumulator
1 set Minimess for accumulator
1 Compensator, fluid type
6 Chain links. Only for ABS, LR and NK
1 set Flexible hoses, one of each size and length
1 set High-pressure gasket kit
1 Coupling for start-up pump

ME Filter, Plate 4572-0800

1 set Filter cartridges for oil filters.
Cartridge filtration ability, minimum $Beta_6$=16.

Engine Control System, Plates 4772-1500/1550,7072-0800/1100/1250 [2]

Triton control system

1. 1 set Controller spares:
   1 Base module
   1 CPU module
   1 Power supervision module
   1 Modbus module
   1 Digital I/O module (DI, DO, DRO)
   1 Analog I/O module
   1 Digital I/O module (DI, DO)
   1 Combi I/O module (AI/DO) Only GI
   1 ID-key for Triton

   1 Trigger sensor for tacho system. (if trigger ring and no angular encoder on fore end
   1 Encoder, steel compensator and bearing set (if angular encoder on fore end and no trigger ring)
   1 Marker sensor for tacho system
   1 Tacho signal amplifier (TCA)

Starting Valve, Plates 3472-0200/0250 (907)

1 Starting valve, complete [2])
(Included in the Cylinder cover complete)
1 Solenoid valve [1])

Hydraulic Cylinder Unit, Plates 4572-0500/0550/0100/0900,4272-0500 (906, 907) 1 and 2)

1 Fuel booster top cover, complete with plunger
1 ELFI + ELVA valves complete, or FIFA if applied.
1 Suction valve complete
1 20 %* Flexible high-pressure hoses, one of each size and length. *) Only for DNV
1 High-pressure pipe kit, one of each size and length
1 set Membrane plus seals for accumulator, 1 set for 1 HCU

1 Packing kit (O-rings, square seals and bonded seals)
1 Fuel booster position sensor
1 Exhaust actuator complete incl. non-return valve

Exhaust Valve, Plates 2272-0200/0210/0240 (908)

2 Exhaust valves complete
(The 2nd exhaust valve is included in the Cylinder cover complete)
1 High-pressure pipe from actuator to exhaust valve
1 Exhaust valve position sensor
1 Safety valve

Fuel Valve, Plates 4272-0200 (909)

1 set Fuel valves of each size and type fitted, complete with all fittings, for 1 engine
a) engines with 1 or 2 fuel valves: 1 set of fuel valves for all cylinders on the engine
b) engines with 3 and more fuel valves per cylinder: 2 fuel valves complete per cylinder, and a sufficient number of valve parts, excluding the body, to form, with those fitted in the complete valve, a full engine set

Fuel Oil high-pressure Pipes, Plate 4272-0100 (909)

1 set High-pressure pipe, from fuel oil pressure booster to fuel valve

Turbocharger, Plate 5472-0700 (910)

1 set Maker's standard spare parts
1 a) Spare rotor for 1 turbocharger, including compressor wheel, rotor shaft with turbine blades and partition wall, if any

Auxiliary blower, Plate 5472-0500 (910)

1 set Bearing for blower wheel
1 set Packing for blower wheel

Bedplate, Plates 1072-0400, 2572-0400 (912)

1 Main bearing shell (1 upper and 1 lower part) of each size
1 set Studs and nuts for 1 main bearing

1) MAN ES 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

Note: Plate numbers refer to the Instruction Manual containing plates with spare parts (older three-digit numbers are included for reference)
Fig. 19.06.01b: List of spare parts, unrestricted service: 4 87 601

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Additional spares

Additional spares beyond class requirements or recommendation, for easier maintenance and increased operating security.

The final scope of spare parts is to be agreed between the owner and engine builder/yard.

Cylinder cover, plate 2272-0300 (901)

4 studs for exhaust valve
4 nuts for exhaust valve
½ engine O-rings for cooling jacket
½ engine sealing between cylinder cover and liner
1 set spring housings for fuel valve for 1 cylinder
1 set sleeves for FBIVP for 1 cylinder

Hydraulic tool for cylinder cover, plates 2270-0310/0315 (901)

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, plates 2272-0400/0420 (902)

1 box locking wire, L=63 m
5 piston rings of each type
2 D-rings for piston skirt
2 D-rings for piston rod

Piston rod stuffing box, plate 2272-0500 (902)

15 self-locking nuts
5 O-rings
5 top scraper rings
15 sets 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, plate 1072-0710 (903)

½ set studs for cylinder cover for 1 cylinder
1 bushing for stuffing box

Cylinder liner and cooling jacket, plates 2272-0600/0660/0665 (903)

4 non-return valves for two cylinders
1 engine O-rings for cylinder liner
½ engine gaskets for cooling water connection
½ engine O-rings for cooling water pipes
1 set cooling water pipes with blocks between liner and cover for 1 cylinder

Cylinder lubricating oil system, plate 3072-0600 (903)

1 solenoid valve
1 level switch for lubricator

Hydaulic power supply (HPS), plates 4572-1000/0750, 4572-1100/1200/1250 (906)

1 electric motor for start-up pump
1 pressure relief valve for start-up pump
2 pressure reducers for pump inlet
25% plug screws for hydraulic system (HPS and HCU)
1 accumulator, complete
1 proportional valve
1 swashplate transducer
1 rubber compensator for inlet

LGIP control block, plates 4272-2000/2100/2200 (909)

1 accumulator for LGIP
1 accumulator for sealing oil
1 accumulator for hydraulic oil
1 repair kit for 1 LGIP control block, including sealing for the intermediate piece, gaskets and packings
1 ELGI valve
1 proportional valve for ELGI
1 ELWI valve
2 in/outlet valves
1 engine sealings and packings for ELWI valve/accumulators and in/outlet valves
1 return oil valve
1 reduction valve for sealing oil
1 solenoid valve for sealing oil unit
1 repair kit for by-pass valve
½ engine of sealings for FBIVP valves
1 constant oil pressure sensor (for 35-bore LGIP only)
½ engine FBIVPs

LGIM control block, plates 4272-2000/2100/2200 (909)

1 accumulator for LGIM
1 accumulator for sealing oil
1 accumulator for hydraulic oil
1 repair kit for 1 LGIM control block, including sealing for the intermediate piece, gaskets and packings
1 ELBI valve
1 proportional valve for ELBI
1 ELFI-L valve
2 blow-off valves
1 set hydraulic valves for return purge block
1 return oil valve
1 repair kit for by-pass valve
½ engine of sealings for FBIVM valves
1 repair kit for sealing oil accumulator
1 solenoid valve for high- and low-pressure sealing oil line

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2 sets of sealings for LGIM gas block pressure sensor
½ engine FBIVMs
1 engine sealings and packings for ELFI-L/ELBI valve/accumulators and blow-off valves

Control and sealing oil for FBIVP or FBIVM valves, plate 4272-2100 (909)

1 set sealing oil pipes for 1 cylinder
1 set control oil pipes for 1 cylinder
1 set constant oil pipe for 1 cylinder (for 35-bore LGIP only)

Engine control system (ECS), plates 4772-1550, 7072-1250 (906)

1 set fuses for MPC, TSA and CNR
1 set sensors for gas system

Hydraulic cylinder unit (HCU), plates 4572-0500, 4272-2300 (906)

1 set packings for booster and actuator, complete set
1 ELFI/1 ELVA valve, or 1 FIVA if applied
1 ball valve, pos. 420
1 ball valve DN10
1 set accumulator, complete
25% plug screws, shared with HPS and accumulator block
1 set piping for activation of gas injection valves for 1 cylinder

Accumulator/safety block, plate 4572-0700 (906)

1 pressure transducer, pos. 320
25% plug screws, shared with HPS and HCU
1 ball valve DN10
1 solenoid valve for valve pos. 310 (shut down)

Alarm and safety system, plates 4772-0100/0200 (906)

1 pressure sensor for scavenge air receiver, PT 8601
1 pressure switch for lubricating oil inlet, PS 8109
1 thrust bearing temperature sensor, TS 8107 (sensor only)
1 pressure switch for jacket cooling water inlet, PS 8402

LGIP control, plates 4772-1500, 7072-0800 (906)

1 DASU computer
1 cylinder pressure sensor
1 LGIP pressure sensor, according to maker's recommendation (yard supply)
1 pressure sensor for sealing oil pressure
1 temperature sensor for LGIP

Main starting valve, plate 3472-0300 (907)

The below main starting valve parts are all to be in accordance with the supplier's recommendation:

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, plate 3472-0200 (907)

2 locking plates
2 pistons
2 springs
2 bushing
1 set O-rings
1 valve spindle

Exhaust valve, plates 2272-0200/0210 (908)

1 exhaust valve spindle
1 exhaust valve seat
½ engine sealing rings between exhaust valve and cylinder cover
4 piston rings
½ engine guide rings for air piston
½ engine sealing rings
½ engine safety valves
1 engine gaskets and O-rings for safety valve
1 piston complete
1 opening damper piston
1 engine O-rings and sealings between air piston and exhaust valve housing/spindle
1 spindle guide
1 engine gaskets and O-rings for cooling water connection
1 conical ring in 2/2 (only for low-force design)
1 engine O-rings for spindle/air piston
1 engine non-return valve
1 sealing oil unit (only for engines without low-force design/COL)
1 inductive sensor for exhaust valve positioning

Exhaust valve pipe, plate 2272-0240 (908)

1 high-pressure pipe from actuator to exhaust valve

Cooling water outlet, plate 5072-0100 (908)

2 ball valves
1 butterfly valve
1 gasket for butterfly valve
1 engine packings for cooling water compensator (only for S50ME-LGI)

Fuel injection system, plate 4272-0500 (909)

1 fuel oil pressure booster, complete for 1 cylinder

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Fuel valve, plate 4272-0200 (909)

½ engine springs
½ engine discs, +30 bar
3 thrust spindles
3 non-return valves, if mounted

Fuel oil high-pressure pipes, plate 4272-0100 (909)

1 set high-pressure pipe, from fuel oil pressure booster to fuel valve
1 set O-rings for high-pressure pipes

Fuel oil regulating valve, plate 4272-0030 (909)

1 fuel oil regulating valve, complete
1 O-ring of each type

Turbocharger, plate 5472-0700 (910)

1 set spare parts for 1 turbocharger in accordance with the supplier's recommendation

Scavenge air receiver, plates 5472-0400/0630 (910)

1 set non-return valves for turbocharger, complete
1 compensator between turbocharger and air cooler

Exhaust pipes and receiver, plates 5472-0750/0900(910)

1 compensator between turbocharger and receiver
2 compensators between exhaust valve and receiver
1 set gaskets for each compensator
1 compensator between FWD and AFT part, if any

Safety valve, plate 2272-0330 (911)

1 set gaskets for safety valve (only for CR and NK)
2 safety valves complete (only for CR and NK)

Air cooler, plate 5472-0100 (910)

1 set anodes (corrosion blocks)
1 set packings (only for cooler type LKMY)

Auxiliary blower, plate 5472-0500 (910)

1 set bearings for electric motor
1 set shaft sealings
1 set bearings/belt/sealings for gearbox (only for belt-driven blowers)

Arrangement of safety cap, plate 3472-0900 (911)

1 engine bursting disc

ME filter, plate 4572-0800 (912)

1 set filter cartridges for redundancy filter
Cartridge filtration ability, minimum Beta6=16
Only for filter make Boll & Kirch

Notes: In the pcs/set column, 'engine' means 'engine set', i. e. a set for one engine, whereas 'set' means a set for the specific component(s).

Section numbers refer to Instruction Book, Vol. III containing plates with spare parts.

Fig. 19.07.01c: Additional spare parts beyond class requirements or recommendation, option: 4 87 603.

Wearing Parts

MAN Energy Solutions Service Letter SL2023-744 provides guiding overhaul intervals and expected service life for key engine components.

See the latest Service Letter on
https://www.man-es.com/docs/default-source/service-letters/sl2023-744.pdf

Service Letter SL2023-744/SRJ

MAN Energy Solutions

Action code: WHEN CONVENIENT

Guiding overhaul intervals
Updated tables
Replaces SL2019-681/SRJ

SL2023-744/SRJ
August 2023

Concerns
Owners and operators of MAN B&W two-stroke, low-speed marine engines.
Types: ME/ME-C, ME-B, LGIM, LGIP, GI, GIE, and ME-GA

Summary
Guiding overhaul intervals and expected service life of engine components on two-stroke low-speed engines.
This Service Letter replaces SL2019-681/SRJ.

Dear Sir or Madam

Based on the latest service experience and engine development we are pleased to issue a revised version of the guiding overhaul intervals tables. The guiding overhaul intervals apply to electronically controlled ME and dual-fuel type engines.

Longer overhaul intervals can be obtained with a condition-based overhaul strategy. The means to obtain and document this are described in SL07-483/HRR.

In addition, it must be noted that the application of, for example, WHR, EGB, EGR and SCR will affect the heat load on the combustion chamber components. Similarly, a more frequent heavy propeller running caused by the Energy Efficiency Design Index (EEDI) condition and the Adverse Weather Condition (AWC) software can have an influence. The above factors as well as fuel qualities and slow steaming will most probably have an impact on the overhaul intervals of

178 70 45-7.1.0

Fig.19.08.01: Example from Service Letter

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Large spare parts, dimensions and masses

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List of standard tools for maintenance

General

This section is available on request

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Tool Panels

This section is available on request

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Tools and special tools

Standard tools

The engine is delivered with a comprehensive and extensive set of tools. These enable normal maintenance work to be carried out.

Special tools

A wide range of special tools for on-board maintenance are available upon the customer's request. The optional tool serve as a supplement to the standard set of tools specified for each engine. These are available via MAN Energy Solutions, or directly via our co-operation agreement holders.

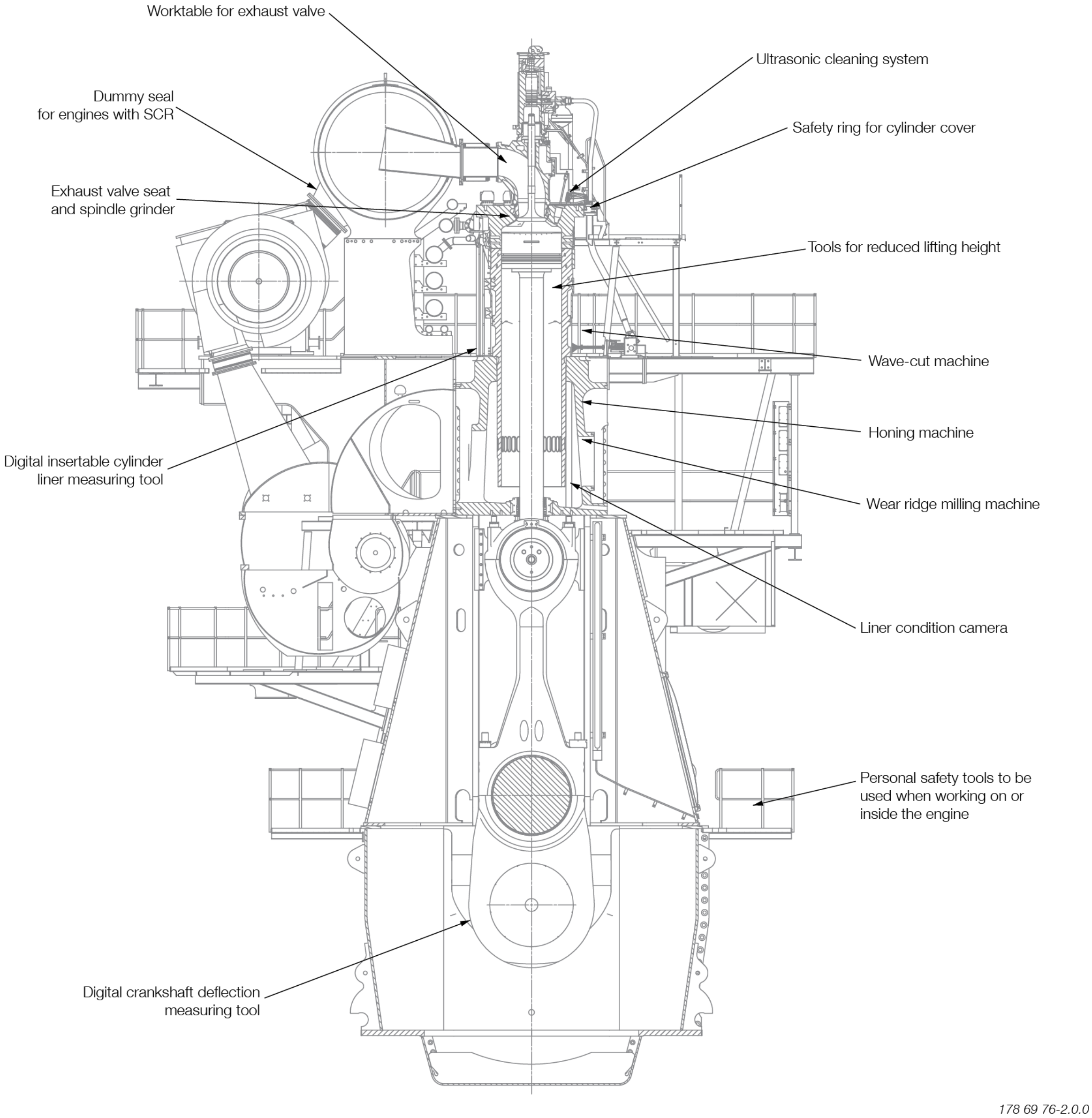

Fig. 19.12.01 Outline of tools

Optional inspection tools to assist condition-based overhaul (CBO)

The objective of the Condition-Based Overhaul (CBO) strategy is to obtain the highest number possible of safe running hours. Overhauling should only be carried out when necessary and the most important factor in a CBO strategy is the evaluation of the actual condition of the engine. This is done by regular scavenge port inspections and logging of wear and hot corrosion.

MAN Energy Solutions recommends the following optional measuring tools to assist with the CBO strategy.

Digital insertable cylinder liner measuring tool

The insertable tool facilitates liner condition monitoring. The cylinder liner trend wear profile can be revealed and predicted by taking measurements regularly during scavenge port inspections at engine stand-stills. The measurements are done without removing the cylinder cover or exhaust valve housing, which is very time saving and allows for more frequent measurements than using the standard version of the tool. The output is digital, see Fig. 19.12.02.

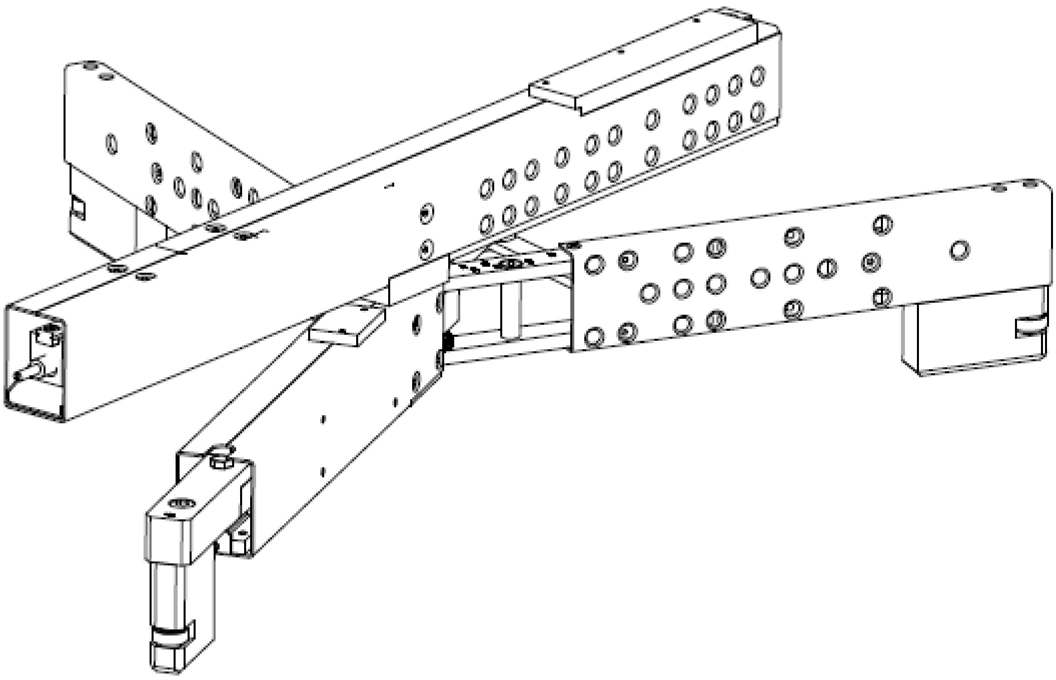

Fig. 19.12.02 The insertable tool is available from different makers, starting from bore size 40

Liner condition camera

The liner condition camera is used for in situ photography of the cylinder liner walls and piston ring packs in 2-stroke engines.

The photos can be used for evaluating cylinder condition parameters such as cleanliness of ring land, size of cylinder wear edge, cylinder honing mark and wave-cut groove extension, black lacquering from corrosive wear and bore polish. See Fig. 19.12.03.

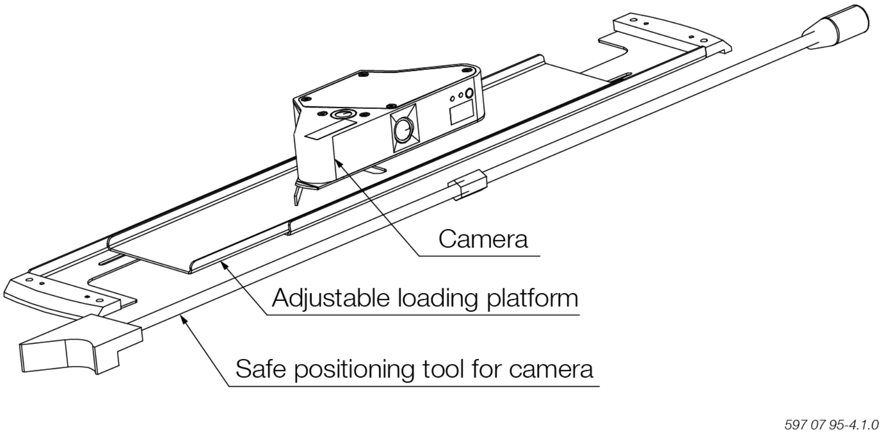

Fig. 19.12.03 Liner condition camera

Digital crankshaft deflection measuring tool

Crank web deflection measurements are used to assess the alignment of the crank throws and the main bearings. The digital crankshaft deflection tool has a higher accuracy, easier reading and is able to store data compared to the standard tool with mechanical dial gauge, see Fig. 19.12.04.

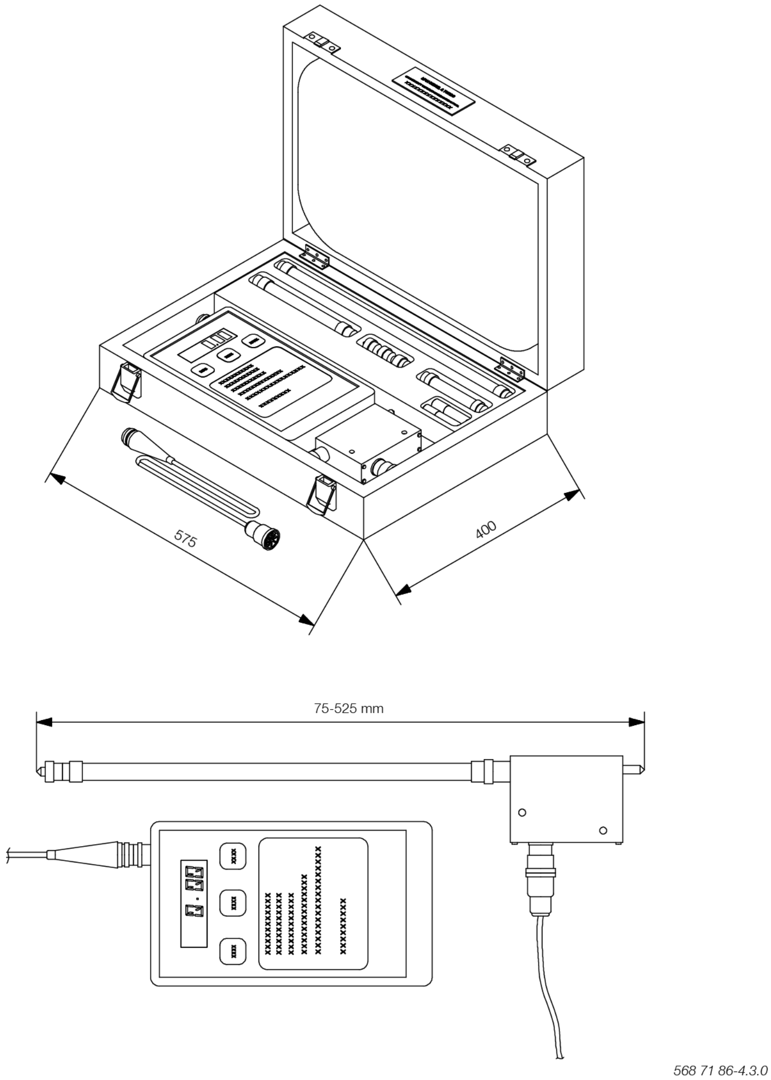

Fig. 19.12.04 The digital measuring tool is available for all engine bore sizes

Optional handling, grinding or refreshing tool

The following tools are recommended for best practice overhaul.

Wear ridge milling machine

The machine is used to remove the wear ridge created in the top of the cylinder liner. This edge must be removed before the pulling of piston, to prevent damage of the piston and/or lifting of the cylinder liner. The risk of damage to piston rings when pulling piston without removing the wear ridge is especially high for CPR rings. The wear ridge milling machine minimizes the risk of liner damage due to un-controlled grinding. The wear ridge milling machine can also be used to create a groove in the top of the liner to prevent the buildup of a new wear ridge.

The wear ridge milling machine is available in two different versions: A standard milling machine and a customized milling machine for which a Hot Works Permit is not necessary. see Fig. 19.12.05.

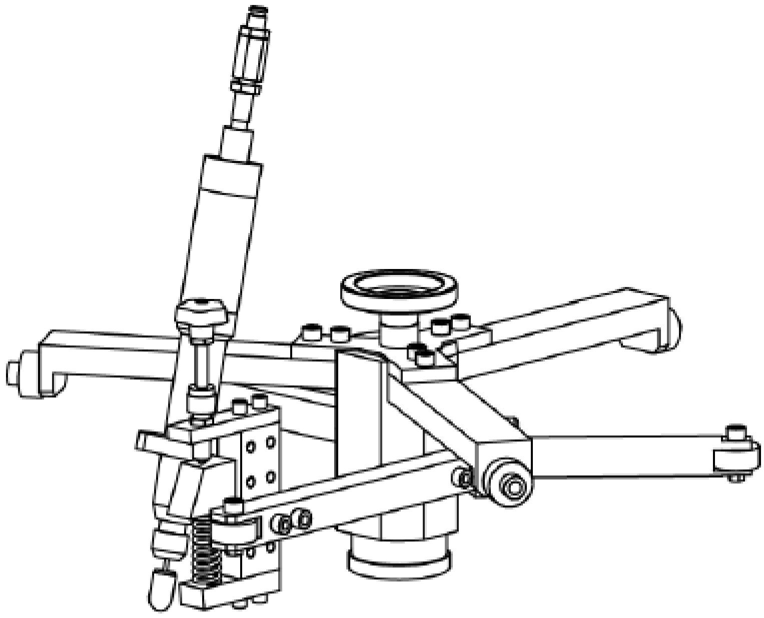

Fig. 19.12.05 The wear ridge milling machine is available for all engine bore sizes

Honing machine

Honing is the best method to remove liner ovality, which cause premature ring breakage. Honing will also remove liner surface hardening and re-establish a normal wear rate of a hardened liner.

The honing machine can be used on its own or combined with the wave-cut machine, see Fig. 19.12.06.

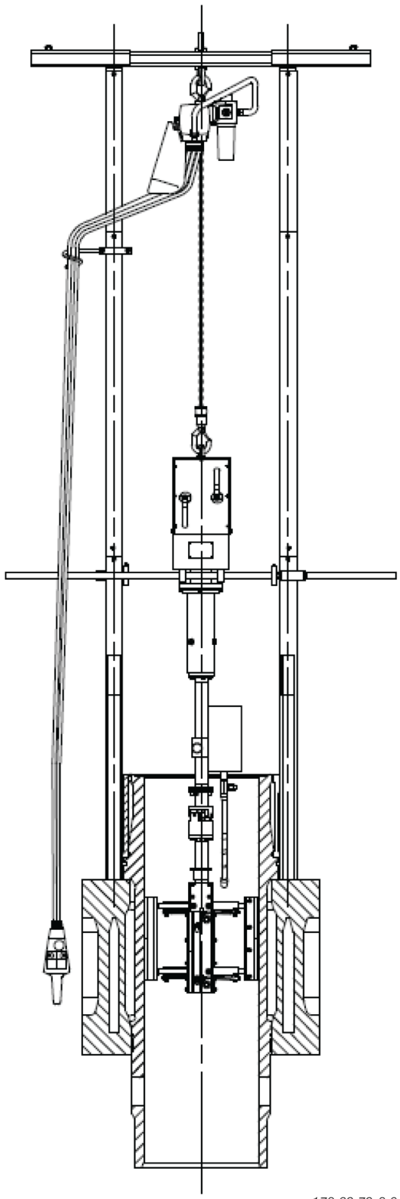

Fig. 19.12.06 Cylinder liner with honing machine

Wave-cut machine

The purpose of the wave-cut machine is to reestablish the wave-cut pattern of the cylinder liner wall, which retains oil and facilitates the running-in of new piston rings. Wave-cutting does not compensate for liner ovality. The wave-cut machine can be used on its own or combined with honing, see Fig. 19.12.07.

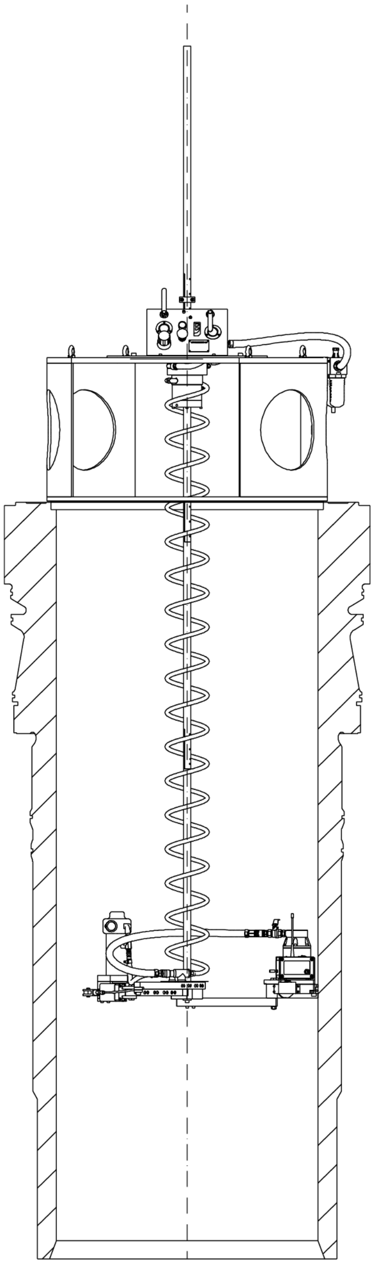

Fig. 19.12.07 Cylinder with wave-cut machine and wave-cut pattern

Worktable for exhaust valve

The worktable for exhaust valve ensures that maintenance of the complete exhaust valve or the exhaust valve housing can be carried out safe and easy, see Fig. 19.12.08.

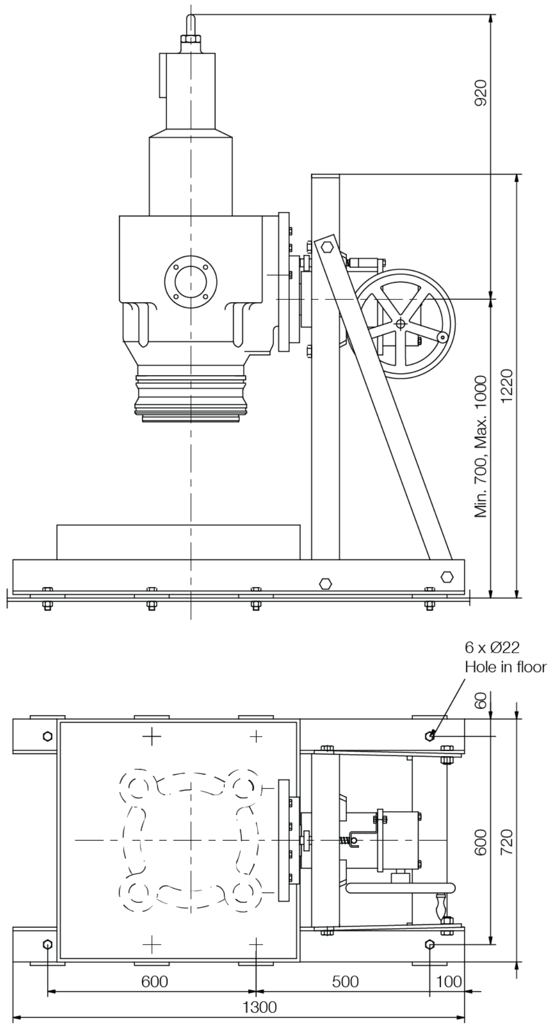

Fig. 19.12.08 Worktable is available for all bore sizes

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Exhaust valve seat and spindle grinder

The exhaust valve seat and spindle grinder provides easy and accurate on-board grinding of the exhaust valve spindle and bottom piece sealing surfaces. See fig. 19.12.09.

Different versions of the grinder exist and can be ordered for all engine bore sizes.

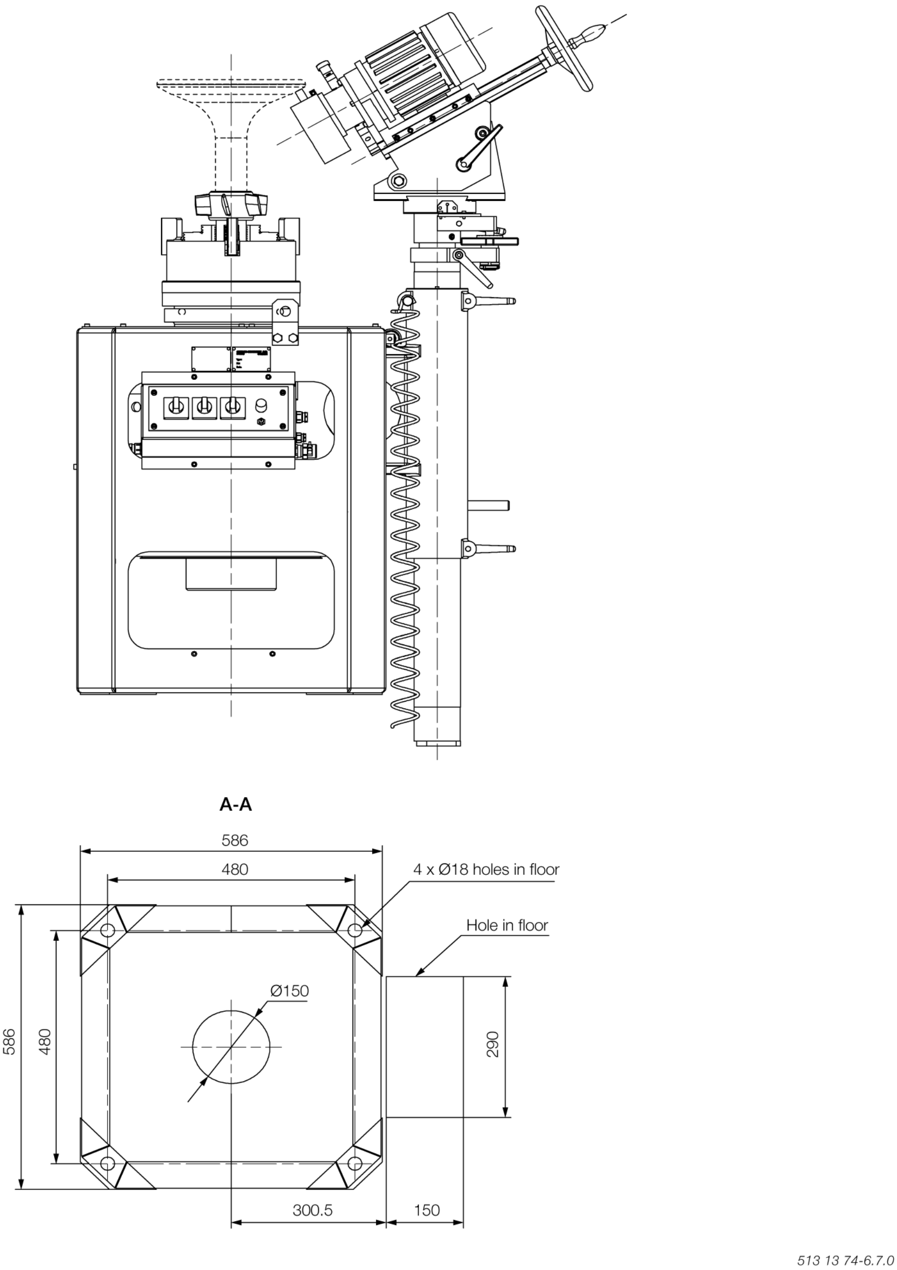
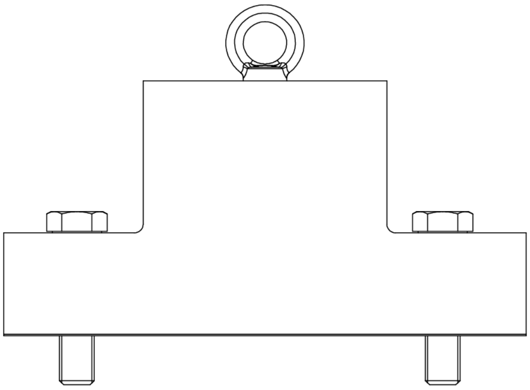


Fig. 19.12.09. Exhaust valve seat and spindle grinder

Dummy seal for SCR



587 41 89-8.1.0

Fig. 19.12.10 Dummy seal for SCR

Optional safety tools

Personal safety tools

Working on or inside two-stroke engines are very often connected with risk of falling down. The risk of falling down can be mitigated by using fall protection equipment.

The supplementary fall arrest safety equipment is CE or ANSI approved and includes a safety harness and lanyards

Full body harness, lightweight 1,3 kg. Attack point in the back. Easy donning and adjustment. Certified to North American, European and Russian standards. It is available in three sizes, see Fig. 19.12.11.

178 69 80-8.0.0

a)

Full body harness, lightweight 1.3 kg. Attack point in the back. Easy donning and adjustment. American

n, European and Russian standards It is available in three sizes.

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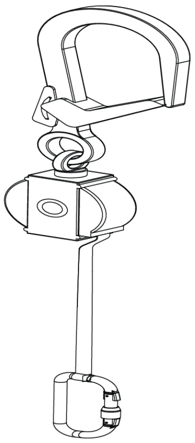
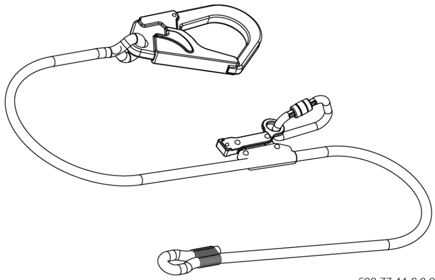

598 77 54-2.0.0

b)

Mini fall arrest block with steel hook. Extracted up to 2 m max. gives the flexibility. Th brake will arrest a free fall within few inches. Connect it directly to harness attach point at the hip and hook it to a safe anchor point. It is designed to avoid falling and must not be used as.


598 77 44-6.0.0

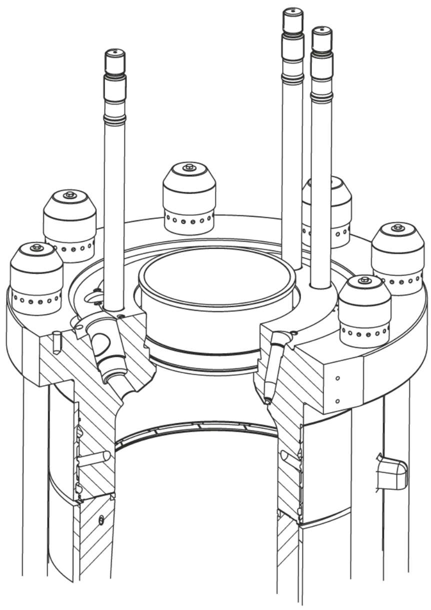
c)

Working positioning lanyard 2 m, adjustable length and a one- hand operated steel hook. Connect it to harness attack point at the hip and hook it to a safe anchor point. It is designed to avoid falling and must not be used as.

Fig. 19.12.11 Full body harness (a), mini fall arrest block (b) and working positioning lanyard (c)

Safety ring for cylinder cover

The safety ring is used for allowing access to the cylinder liner through the exhaust valve bore of the cylinder cover. This option is available for bore size 90 and above. For bore sizes below the the cylinder cover are to be dismantled, see Fig. 19.12.12.


178 69 84-5.0.0

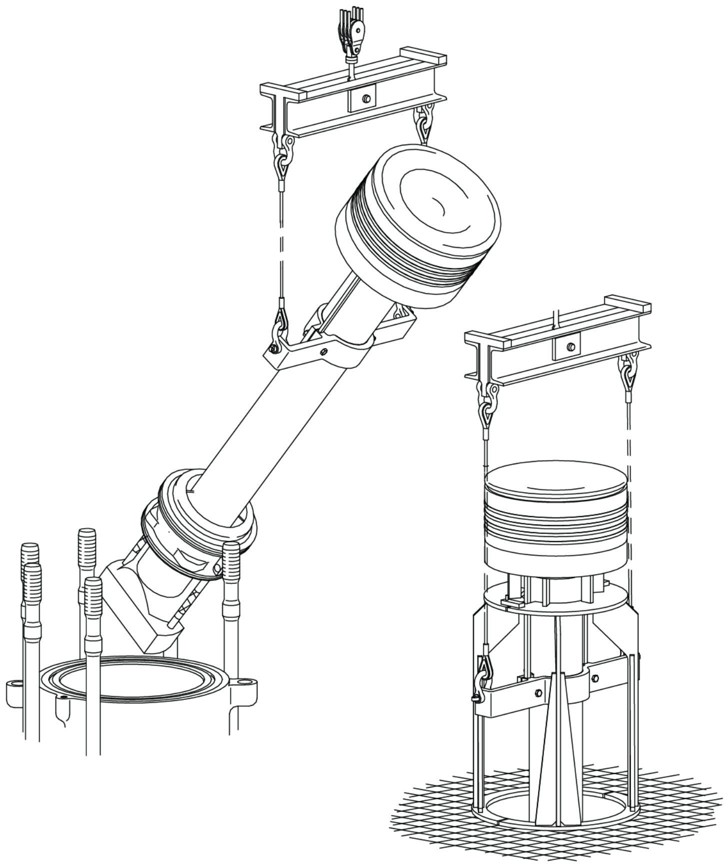
Optional tools for special design

Tools for reduced lifting height

The optional tool package for reduced lifting height includes a collar ring which makes it possible to tilt the piston, hence reducing the required lifting height of the engine room.

Engines with bore size less than 70 can be ordered with reduced lifting height in the engine room. Engines with bore size from 70 can receive reduced lifting height upon special request.

Tilted lift is not allowed for the cylinder liner, but by using the MAN B&W double jib engine room crane the necessary lifting height is greatly reduced compared to the single crane standard solution. Lifting screws for the double jib crane is included in the tool package for reduced lifting height, see Fig. 19.12.13 and 19.12.14.



178 69 85-7.0.0

Fig. 19.12.13 Piston tool for reduced lifting height

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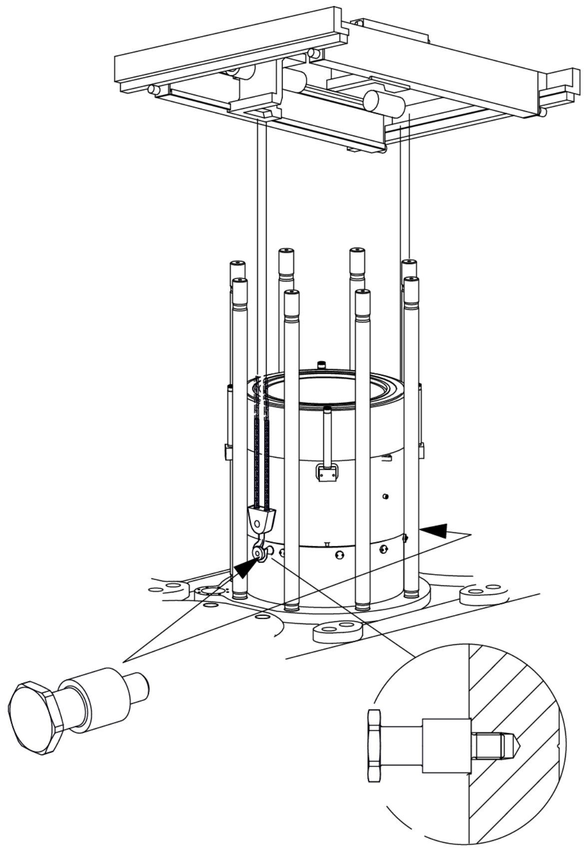

Fig. 19.12.14 Cylinder liner tool for reduced lifting height

Ultrasonic cleaning system

Ultrasonic cleaning system can be installed onboard to give the possibility of cleaning of smaller engine parts such as fuel and spray nozzles, filters etc. using ultrasound waves in a specific cleaning chemical bath. The system is available in different sizes. See Fig. 19.12.15.

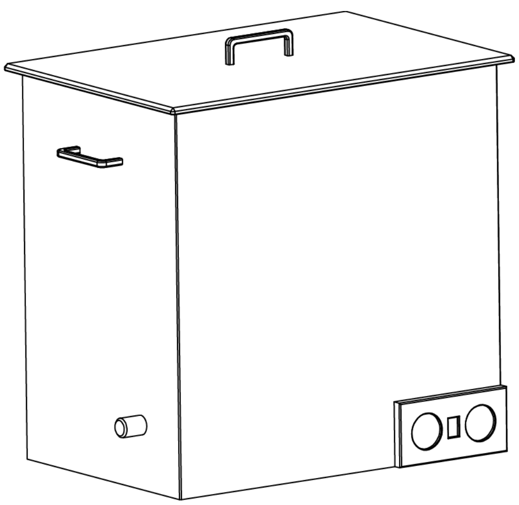

Fig. 19.12.15 Onboard ultrasonic cleaning system

Rig for large fuel valve

The working rig facilitates overhaul work on large injection valves e.g. dismantling and cleaning.

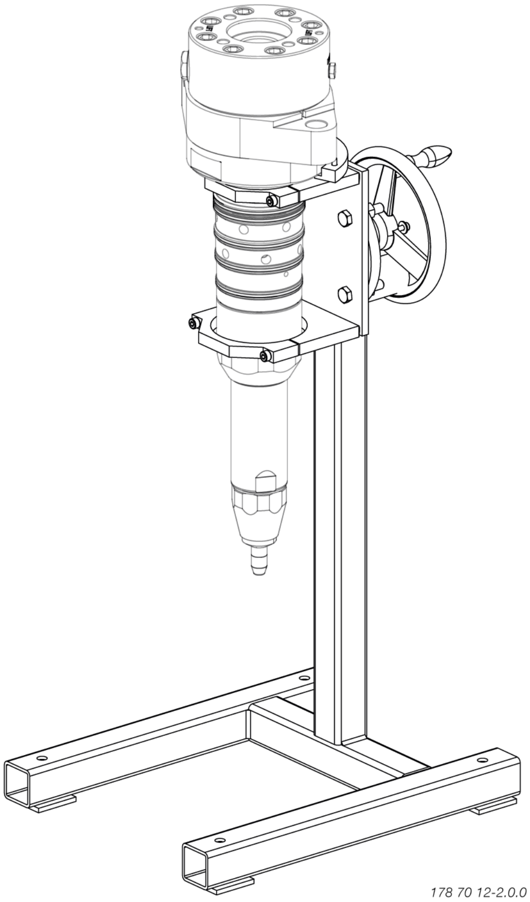

Fig. 19.12.16 Rig for large fuel valve.

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Tools for oil testing

Clean lubricating oil is the basis for trouble free operation of a MAN B&W engine. MAN Energy Solutions recommend testing oil samples of the cylinder lubricating oil for metal particles and determining the level of corrosive elements in order to take appropriate steps to avoid excessive cylinder liner wear.

Contaminants in the lubricating oil may also damage the hydraulic components in the ME system. For testing the lubricating oil, the following equipment is available.

- Ferrous wear test equipment for detecting metal particles from samples of cylinder lubricating oil
- Additional consumables pack for ferrous wear test equipment containing 500 test tubes and sampling pipettes.

178 70 58-9.0.0

Fig. 19.12.17 Ferrous wear test equipment for detecting metal particles from samples of cylinder lubricating oil.

- Cold corrosion test equipment for obtaining the level of corrosive elements in the cylinder lubricating oil including reagent pack for 100 tests
- Additional reagent pack for cold corrosion test kit containing reagents for 100 tests.

178 70 59-0.0.0

Fig. 19.12.18 Cold corrosion test eqiupment for obtaining the level of corrosive elements in the cylinder liner lubricating oil.

- Total base number test kit for determining viscosity, total base number, total acid number, soot and water in oil
- Additional reagent pack for total base number test kit containing reagents for 50 tests.

178 70 60-0.0.0

Fig. 19.12.19 Total base number test kit for determining viscosity, total base number, total acid number, soot and water in oil.

- Drip oil analyzer for measuring the total iron content in the cylinder lubricating oil which indicates the corrosive and abrasive wear.

178 70 61-2.0.0

Fig. 19.12.20 Drip analizer for measuring the total iron content in the cylinder lubricating oil.

- Rack including 12 pcs. Iron testing units for drip oil analizer

178 70 62-4.0.0

Fig. 19.12.21 Rack including 12 pcs. Iron testing units for drip oil analizer.

- Pipette for drip oil analizer.

178 70 63-6.0.0

Fig. 19.12.22 Pipette for drip oil analyzer

Appendix: tool plates for optional

The tool plates of the optional tools described in 'Optional tools' are included below. For ordering, please contact MAN PrimeServ and state the IMO no., engine type, plate no. and item no. of the tool in question.

Plate No.	Description
2270-0610	Digital insertable cylinder liner measuring tool
1070-1035	Digital crankshaft deflection measuring tool
2270-0620	Wear ridge milling machine
2270-0630	Honing machine
2270-0630	Wave-cut machine
2270-0220	Worktable for exhaust valve
2270-0210	Exhaust valve seat and spindle grinder
0570-0100	Personal safety package
2270-0335	Safety ring for cylinder liner
2270-0460	Tools for reduced lifting height, bore size 50 and below
2270-0460	Tools for reduced lifting height, bore size 60 and above
4270-0240	Rig for large fuel valve
3070-0600	Ferrous wear meter test kit (FWM)
3070-0600	Consumables pac for FWM
3070-0600	Cold corrosion test kit (CCTK)
3070-0600	Reagent pack for CCTK
3070-0600	Total base number test kit (TBN)
3070-0600	Reagent pack TBN
3070-0600	DOT FAST drip oil analyzer
3070-0600	DispoRack including 12 pcs. iron testing units
3070-0600	MicroMan pipette

Table. 19.12.01 Tool plates for optional tools.

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Project support and documentation

General

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 Energy Solutions 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 Energy Solutions covering such fields as the selection of the best propulsion machinery, optimization 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:

- Marine Engine Programme
- Turbocharger Selection
- Installation Drawings
- CEAS - Engine Room Dimensioning
- Project Guides
- Extent of Delivery (EOD)
- Technical Papers

The publications are available at:
www.marine.man-es.com--> Two-stroke.

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 and ME-B 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, ME or ME-B 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.

CEAS application

General

Additional customised information can be obtained from MAN Energy Solutions as project support. For this purpose, we have developed the CEAS application, by means of which specific calculations can be made during the project stage.

The CEAS Application

The CEAS application is found at
https://www.man-es.com-->CEAS En-gine Calculation.

On completion of the CEAS application, a report is generated covering the following:

• Main engine room data
• Specified main engine and ratings
• Ambient reference conditions
• Expected SFOC, lube oil consumption, air and exhaust gas data
• Necessary capacities of auxiliary machinery (SMCR)
• Starting air system, engine dimensions, tanks, etc.
• Tables of SFOC and exhaust gas data
• Heat dissipation of engine
• Water condensation separation in air coolers
• Noise – engine room, exhaust gas, structure borne
• Preheating of diesel engine
• Alternative engines and turbochargers, further reading
.

Links to related MAN Energy Solutions publications and papers are provided, too.

Supplementary Project Data on Request

Further to the data generated by the CEAS application, the following data are available on request at the project stage:

• Estimation of ship's dimensions
• Propeller calculation and power prediction
• Selection of main engine
• Main engines comparison
• Maintenance and spare parts costs of the engine
• Total economy – comparison of engine rooms
• Steam and electrical power – ships' requirement
• Utilisation of exhaust gas heat
• Utilisation of jacket cooling water heat, fresh water production
• Exhaust gas back pressure
• Layout/load diagrams of engine.

Contact MAN Energy Solutions Copenhagen in this regard.

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Extent of delivery

General

MAN Energy Solutions' ‘Extent of Delivery’ (EoD) is provided to facilitate negotiations between the yard, the engine maker, consultants and the customer in specifying the scope of supply for a specific project involving MAN B&W two-stroke engines.

We provide two different EoDs:

EoD 95-40 ME-C/-GI/-LGI Tier II Engines
EoD 50-30 ME-B/-GI/-LGI Tier II Engines

These publications are available in print and at: www.marine.man-es.com --> Two-Stroke --> Extemt of Delivery (EoD).

Basic Items and Options

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 unattended machinery space (UMS), without taking into consideration any further 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

At MAN Energy Solutions Copenhagen, we base our first quotations on a ‘mostly required’ scope of supply. This is the so-called ‘Copenhagen Standard Extent of Delivery’, which is made up by options marked with an asterisk * in the far left column in the EoD.

The Copenhagen Standard Extent of Delivery includes:

• Minimum of alarm sensors recommended by the classification societies and MAN Energy Solutions
• Moment compensator for certain numbers of cylinders
• MAN turbochargers
• The basic engine control system
• Engine Management Services (EMS) incl. PMI software and LAN-based interface to AMS
• Spare parts either required or recommended by the classification societies and MAN Energy Solutions
• Tools required or recommended by the classification societies and MAN Energy Solutions

MAN Energy Solutions licencees may select a differ-ent extent of delivery as their standard.

EoD and the Final Contract

The filled-in EoD is often used as an integral part of the final contract.

The final and binding extent of delivery of MAN B&W two-stroke engines is to be supplied by our licensee, the engine maker, who should be contacted in order to determine the execution for the actual project.

Installation documentation

General

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 extent of Installation Documentation is decided by the engine maker and may vary from order to order.

As an example, for an engine delivered according to MAN Energy Solutions 'Copenhagen Standard Extent of Delivery', the Installation Documentation is divided into the volumes 'A' and 'B':

- 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.

The engine layout drawings in volume 'B' will, in each case, be customised according to the buyer's requirements and the engine maker's production facilities.

A typical extent of a set of volume 'A' and B' drawings is listed in the following.

For questions concerning the actual extent of Installation Documentation, please contact the engine maker.

Engine-relevant Documentation

Engine Data, on Engine

External forces and moments
Guide force moments
Water and oil in engine
Centre of gravity
Basic symbols for piping
Instrument symbols for piping
Balancing

Engine Connections

Engine outline
List of flanges/counterflanges
Engine pipe connections

Engine Instrumentation

List of instruments
Connections for electric components
Guidance values automation, engine
Electrical wiring

Engine Control System

Engine Control System, description
Engine Control System, diagrams
Pneumatic system
Speed correlation to telegraph
List of components
Sequence diagram

Control Equipment for Auxiliary Blower

Electric wiring diagram
Auxiliary blower
Starter for electric motors

Shaft line, on engine

Crankshaft driving end
Fitted bolts

Turning Gear

Turning gear arrangement
Turning gear, control system
Turning gear, with motor

Spare Parts

List of spare parts

Engine Paint

Specification of paint

Gaskets, Sealings, O-rings

Instructions
Packings
Gaskets, sealings, O-rings

Engine Pipe Diagrams

Engine pipe diagrams
Bedplate drain pipes
Instrument symbols for piping
Basic symbols for piping
Lubricating oil, cooling oil and hydraulic oil piping

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Cylinder lubricating 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 scavenge air box
Oil mist detector pipes, if applicable
Pressure gauge pipes

Engine Room-relevant Documentation

Engine data, in engine room

List of capacities
Basic symbols for piping
Instrument symbols for piping

Lubricating and cooling oil

Lubricating oil bottom tank
Lubricating oil filter
Crankcase venting
Lubricating and hydraulic oil system
Lubricating oil outlet

Cylinder Lubrication

Cylinder lubricating oil system

Piston Rod Stuffing Box

Stuffing box drain oil cleaning system

Seawater Cooling

Seawater cooling system

Jacket Water Cooling

Jacket water cooling system
Deaerating tank
Deaerating tank, alarm device

Central Cooling System

Central cooling water system
Deaerating tank
Deaerating tank, alarm device

Fuel Oil System

Fuel oil heating chart
Fuel oil system
Fuel oil venting box
Fuel oil filter

Compressed Air

Starting air system

Scavenge Air

Scavenge air drain system

Air Cooler Cleaning

Air cooler cleaning system

Exhaust Gas

Exhaust pipes, bracing
Exhaust pipe system, dimensions

Engine Room Crane

Engine room crane capacity, overhauling space

Torsiograph Arrangement

Torsiograph arrangement

Shaft Earthing Device

Earthing device

Fire Extinguishing in Scavenge air Space

Fire extinguishing in scavenge air space

Instrumentation

Axial vibration monitor

Engine Seating

Profile of engine seating
Epoxy chocks
Alignment screws

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

Side Chocks

Side chocks
Liner for side chocks, starboard
Liner for side chocks, port side

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

Engine Top Bracing

Top bracing outline
Top bracing arrangement
Friction materials
Top bracing instructions
Top bracing forces
Top bracing tension data

Shaft Line, in Engine Room

Static thrust shaft load
Fitted bolt

Power Take-Off

List of capacities
PTO/RCF arrangement,if fitted

Large 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
Telescope pipe
Thrust block segment
Turbocharger rotor

Gaskets, Sealings, O-rings

Gaskets, sealings, O-rings

Material Sheets

MAN Energy Solutions Standard Sheets Nos.:
- S19R
- S45R
- S25Cr1
- S34Cr1R
- C4

Engine Production and Installation-relevant Documentation

Main Engine Production Records, Engine Installation drawings

Installation of engine on board
Dispatch pattern 1, or
Dispatch pattern 2
Check of alignment and bearing clearances
Optical instrument or laser
Reference sag line for piano wire
Alignment of bedplate
Piano wire measurement of bedplate
Check of twist of bedplate
Crankshaft alignment reading
Bearing clearances
Check of reciprocating parts
Production schedule Inspection after shop trials
Dispatch pattern, outline
Preservation instructions

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Shop Trials

Shop trials, delivery test
Shop trial report

Quay Trial and Sea Trial

Stuffing box drain cleaning
Fuel oil preheating chart
Flushing of lubricating oil system
Fresh water system treatment
Fresh water system preheating
Quay trial and sea trial
Adjustment of control air system
Adjustment of fuel pump
Heavy fuel operation
Guidance values automation

Flushing Procedures

Lubricating oil system cleaning instruction

Tools

Engine Tool

List of tools
Outline dimensions, main tools

List of Tools

Tool panels

Engine Seating Tools

Hydraulic jack for holding down bolts
Hydraulic jack for end chock bolts

Auxiliary Equipment

Ordered auxiliary equipment

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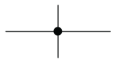
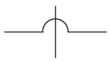
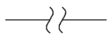
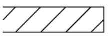
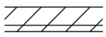
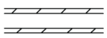
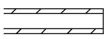
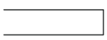
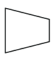
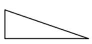
Symbols for piping

Lines, Pipes etc.

	Line, primary process
	Line, secondary process
	Control line, general type
	Control line, capilar type
	Lines connected
	Crossing lines, connected
	Crossing lines, not connected
	Interruption of pipe line
	Interruption of pipe line with reference indication
	Pipeline or duct with thermal insulation
	Pipeline with thermal insulation, heated or cooled by a separate circuit, end
	Pipeline with thermal insulation, heated or cooled by a separate circuit
	Pipeline with thermal insulation, heated or cooled by a separate circuit, end
	Jacketed (sleeved) pipeline with thermal insulated
	Jacketed (sleeved) pipeline with thermal insulated, end
	Jacketed (sleeved) pipeline
	Jacketed (sleeved) pipeline, end
	Change of pipe diameter, pipe reducer
	Pipe slope, located above pipe

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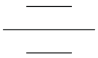
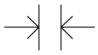
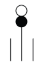
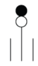
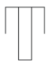
	Interlocked, located in interlocked line
NO	Indication of valve normally open. With the symbol 'function' field
NO	Indication of valve normally closed. With the symbol 'function' field

Flanges, Connections and Other in-line Pipe Fittings

	Flange, single
	Flange coupling, flange pair, blind flange
	Flange coupling, clamped
	Screw joint
	Quick release coupling
	Quick release coupling, with automatically closing when uncoupled
	End cap, threaded
	End cap
	Orifice
	Swing blind, closed
	Swing blind, open
	Rupture disc
	Siphon
	Boss
	Boss with intersection pipe
	Spray nozzle, single
	Spray nozzle, multiple

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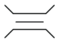
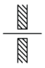
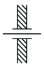
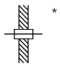
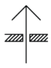
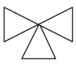
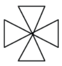
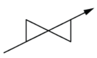
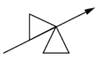
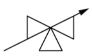
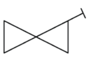
Pipe Supports

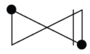
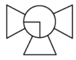
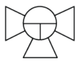
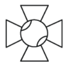
	Pipe support, fixation type
	Pipe support, sliding type

Wall Penetrations, Drains and Vents

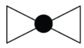
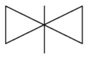
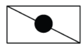
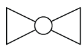
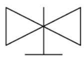
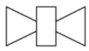
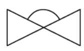
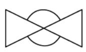
	Wall or roof penetration, general
	Wall or roof penetration, general, jacketed (sleeved) pipeline
	Wall or roof penetration, sealed. * Shall be replaced with a designation for the type of seal, e.g. fire
	Drain, funnel etc.
	Drain pan
	Vent, outlet to atmosphere
	Vent, outlet to atmosphere outside enclosure

Valve Symbols

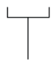
	2-way on-off valve, straight type, general
	2-way on-off valve, angle type, general
	3-way valve, general
	4-way valve, general
	Non-return function, check function, flow left to right
	Control valve, straight type, general
	Control valve, angle type, general
	Control valve, 3-way type, general
	Pre-set control valve, e.g. flow balancing valve

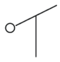
	Safety function, straight type general, inlet / internal side to the left
	Safety function, angle type general, inlet / internal side bottom
	Breather valve, straight type general, with safety function, e.g. tank overpressure / vacuum function
	Breather valve, angle type general, with safety function, e.g. tank overpressure / vacuum function
	3-way plug valve, L-bore, general
	3-way plug valve, T-bore, general
	4-way plug valve, double L-bore, general

Supplementary Valve Symbols

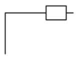
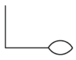
	Valve, globe type
	Valve, gate type
	Valve, butterfly type
	Valve, ball type
	Valve, piston or plunger type
	Valve, plug type
	Valve, diaphragm type
	Valve, hose type
	Valve, needle type

Manual Operators

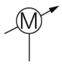
	Manually operated
	Manually operated, by pushing
	Manually operated, by pulling

	Manually operated, by pulling and pushing
	Manually operated, by a lever
	Manually operated, by a pedal
	Manually operated, incl. locking device

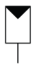
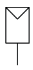
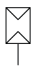
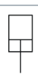
Mechanical Operators

	Mechanically operated, by weight
	Mechanically operated, by float
	Mechanically operated, by spring

Electric Drives

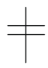
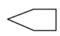
	Electrical motor
	Electrical motor, adjustable

Automatic Operators

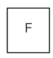
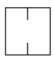
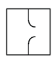
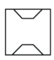
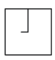
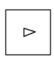
	Actuator, without indication of type
	Single-acting hydraulic actuator
	Double-acting hydraulic actuator
	Single-acting pneumatic actuator
	Double-acting pneumatic actuator
	Single-acting diaphragm actuator
	Double-acting diaphragm actuator
	Single- or double-acting fluid actuator. (For double- acting, two pilot lines are needed)

	Electromagnetic actuator
	Self-operated pressure sustaining control diaphragm. Upstream to valve, right side
	Self-operated pressure reducing control diaphragm. Upstream to valve, right side

Spindle Information, e.g. Safety Operators

	Fail to close
	Fail to open
	Quick-closing
	Quick-opening
	Double-acting, fail freeze
	Double-acting, fail freeze, drifting against open position
	Double-acting, fail freeze, drifting against closed position
	Limit switch, mechanical type

Flow Meters

	Flow meter, general
	Flow meter, propeller, turbine and screw type
	Flow meter, orifice type
	Flow meter, flow nozzle type
	Flow meter, venturi type
	Flow meter, pitot tube type
	Flow meter, vortex type

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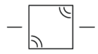
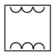
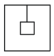
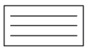
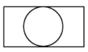
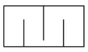
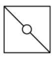
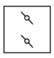
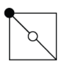
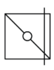
	Flow meter, ultrasonic in-line type
	Flow meter, ultrasonic clamp-on type
	Flow meter, magnetic type
	Flow meter, Coriolis type
	Flow switch, paddle type

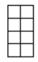
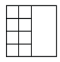
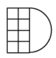
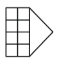
Various

	Air release valve
	Condensate release valve
	Restrictor, multistage type
	Flow straightener
	Viewing glass
	Silencer
	Flow restriction
	Flow restriction, adjustable

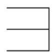
Dampers

	2-way on-off damper, general
	Multi-leaf damper, louvre type
	3-way on-off damper, general
	Non return damper, general
	Safety damper, general

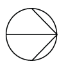
Safety Devices Other than Valves

	Flame arrester, general
	Flame arrester, explosion- proof
	Flame arrester, fireresistant
	Flame arrester, detonation- proof

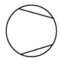
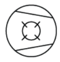
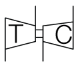
Expansions

	Expansion loop
	Expansion sleeve
	Expansion joint / compensator bellow
	Flexible pipe, hose

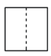
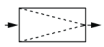
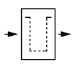
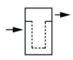
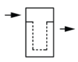
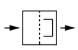
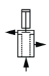
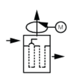
Liquid Pumps

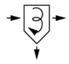
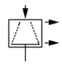
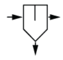
	Pump, general
	Pump, positive displacement type
	Pump, centrifugal type
	Pump, hand type
	Pump, gear type
	Pump, screw type
	Pump, piston type
	Pump, piston radial type
	Pump, membrane type
	Pump, ejector type

Fans, Ventilators and Compressors

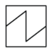
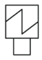
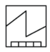
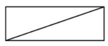
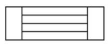
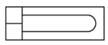
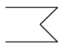
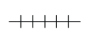
	Fan, ventilator, blower, compressor. General
	Fan, ventilator, blower, compressor. Rotary vane type
	Fan, ventilator, blower, compressor. Impeller type
	Fan, ventilator, blower, compressor. Screw type
	Fan, ventilator, blower, compressor. Reciprocating piston type
	Fan, ventilator, blower, compressor. Rotary reciprocating piston type
	Fan, ventilator, blower, compressor. Reciprocating diaphragm type
	Fan, ventilator, blower, compressor. Turbo type
	Fan, ventilator, blower, compressor. Rotary piston type, e.g. Roots type
	Turbocharger

Filters, Separators

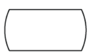
	Screen, strainer, general
	Screen, strainer with pit for draining
	In-line strainer, general
	Cartridge filter, bag filter etc, general
	Cartridge filter, bag filter etc flow direction outside - in, general
	Cartridge filter, bag filter etc, flow direction inside - out, general
	Permanent magnet filter
	Filter with backflush, Bernoulli type
	Filter with continuous backflush

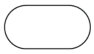
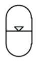
	Separator, cyclone type
	Separator, centrifuge type
	Separator, impact type

Heat Exchangers

	Heat exchanger, general
	Condenser with hot well
	Electrical heater, superheater
	Heat exchanger with plate
	Heat exchanger with tubes
	Heat exchanger with ubend tubes
	Heat exchanger with coil-shaped tubes
	Electrical heating element
	Heating or cooling coil
	Finned tube

Tanks

	Open tank, basin
	Closed tank
	Closed tank with sloped bottom
	Tank with flat bottom and conical roof
	Tank with flat bottom and dished roof
	Vessel with dished ends

	Vessel with spherical ends
	Accumulator
	Gas cylinder

Instrumentation, General

	Instrument with two letters, e.g. PI
	Instrument with three letters, e.g. DPI

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Fig. A.01.01: Basic symbols for pipe plants according to MAN Energy Solutions

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