The MC/MC-C Engine

Whether the freight rates rise or fall, an attractive payback time for newbuildings starts with low investment cost. Once in operation, the ease and flexibility in assigning engineers to operate the engine plant are together with low consumption rates of fuels, lubes, parts and service among the important functional issues which contribute to the cost benefit. The MAN B&W MC/MC-C engine meets both requirements.

The world market-leading two-stroke MC/MC-C engine programme from MAN Diesel has evolved since the early 1980s to embrace bore sizes from 260 mm to 980 mm for propelling ocean-going ships of all types and sizes. In fact, low-speed two-stroke main engines of the MC/MC-C type have become industry standard in a huge number of ship types. Also land-based applications (power plants mainly) have found the MC/MC-C engine types attractive.

The MC/MC-C engine features chain driven camshaft, camshaft controlled fuel injection timing and exhaust valve opening as well as a conventional fuel oil pumps, all well-known and proven technology familiar to marine engineers all over the world.

To conclude, the MAN B&W MC/MC-C engine combines classic virtues of commonly known, well-proven technology continuously upgraded and up-rated to suit the requirements to modern prime movers. Consequently, our latest cutting edge design and manufacturing features are built into each component.

Concept of the MC/MC-C engine

The engine concept is based on a mechanical camshaft system for activation of the fuel injection and the exhaust valves. The engine is provided with a pneumatic/electric manoeuvring system and the engine speed is controlled by an electronic/hydraulic type governor.

Each cylinder is equipped with its own fuel injection pump, which consists of a simple plunger activated by the fuel cam directly. Fuel economy at part load is optimized by means of the Variable Injection Timing (VIT) incorporated in the fuel pumps (optional on certain MC-C engines).

The cam controlled exhaust valve is opened hydraulically and closed by means of an air spring.

Lubrication is either by means of a uni-lube oil system serving both crankshaft, chain drive, piston cooling and camshaft or a combination of a main lubricating oil system and a separate camshaft lube oil system.

Cylinder lubrication is accomplished by electronically controlled Alpha lubricators, securing a low lube oil consumption, or timed mechanical lubricators alternatively.

The starting valves are opened pneumatically by control air from the starting air distributor(s) and closed by a spring.

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

The main features of the MC engine are described in the following pages.

For further information about the application of MC/MC-C engines based on ship particulars and power demand, please refer to our publications titled:

Propulsion Trends in Container Vessels

Propulsion Trends in Bulk Carriers

Propulsion Trends in Tankers

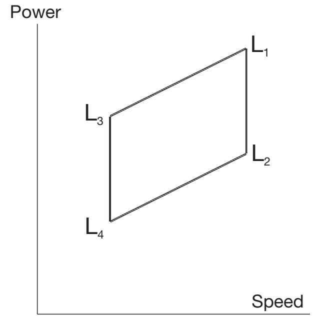
The publications are available at:
www.mandiesel.com under
‘Quicklinks’ → ‘Technical Papers’

Engine Type Designation

6 S 70 M C-C 7

- **7** – Mark version
- **C** – Design
  - **C** Compact engine
- **C** – Concept
  - **C** Camshaft controlled
- **M** – Engine programme
- **70** – Diameter of piston in cm
- **S** – Stroke/bore ratio
  - **S** Super long stroke
  - **L** Long stroke
  - **K** Short stroke
- **6** – Number of cylinders

Power, Speed, Fuel and Lubricating Oil Consumption



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MAN B&W S70MC-C8
Bore: 700 mm
Stroke: 2,800 mm

Power and speed

Layout points	Engine speed r/min	Mean effective pressure bar	Power kW			
			Number of cylinders			
			5	6	7	8
L_1	91	20.0	16,350	19,620	22,890	26,160
L_2	91	16.0	13,050	15,660	18,270	20,880
L_3	77	20.0	13,850	16,620	19,390	22,160
L_4	77	16.0	11,050	13,260	15,470	17,680

Fuel and lubricating oil consumption

	Specific fuel oil consumption g/kWh				Lubricating oil consumption		
	With high efficiency turbocharger		With conventional turbocharger		System oil Approximate g/kWh	Cylinder oil g/kWh	
At load Layout point	100%	80%	100%	80%		Mechanical cyl. lubricator	MAN B&W Alpha cyl. lubricator
L_1	169	166	171	168	0.15	1.0-1.5	0.7
L_2	162	159	164	161			
L_3	169	166	171	168			
L_4	162	159	164	161			

Fig. 1.03.01 Power, speed, fuel and lubricating oil consumption

Engine Power Range and Fuel Oil Consumption

Engine Power

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

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

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

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

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

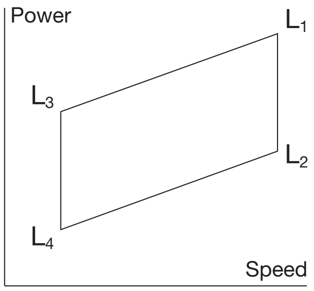

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Fig. 1.04.01: Layout diagram for engine power and speed

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

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

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

Specific fuel oil consumption (SFOC)

Specific fuel oil consumption values refer to brake power, and the following reference conditions:

ISO 3046/1-2002:
Blower inlet temperature................................. 25°C
Blower inlet pressure ....1000 mbar
Charge air coolant temperature..................... 25 °C
Fuel oil lower calorific value .... 42,700 kJ/kg
(~10,200 kcal/kg)

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

SFOC guarantee

The figures given in this project guide represent the values obtained when the engine and turbocharger are matched with a view to obtaining the lowest possible SFOC values and fulfilling the IMO NO_x emission limitations.

The Specific Fuel Oil Consumption (SFOC) is guaranteed for one engine load (power-speed combination), this being the one in which the engine is optimised.

The guarantee is given with a margin of 5%.

As SFOC and NO_x are interrelated parameters, an engine offered without fulfilling the IMO NO_x limitations is subject to a tolerance of only 3% of the SFOC.

Lubricating oil data

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

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

Performance Curves

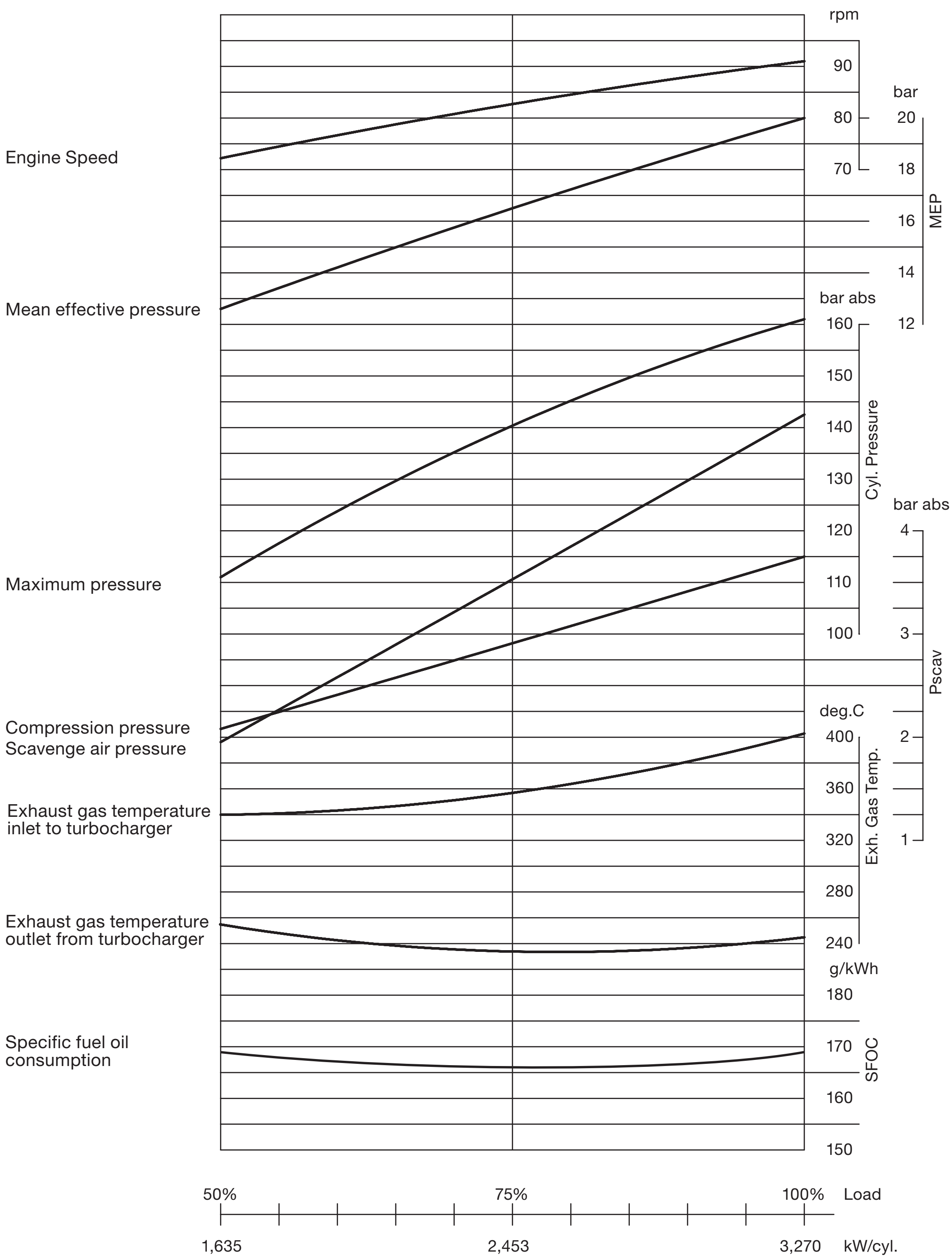

Fig. 1.05.01: Performance curves

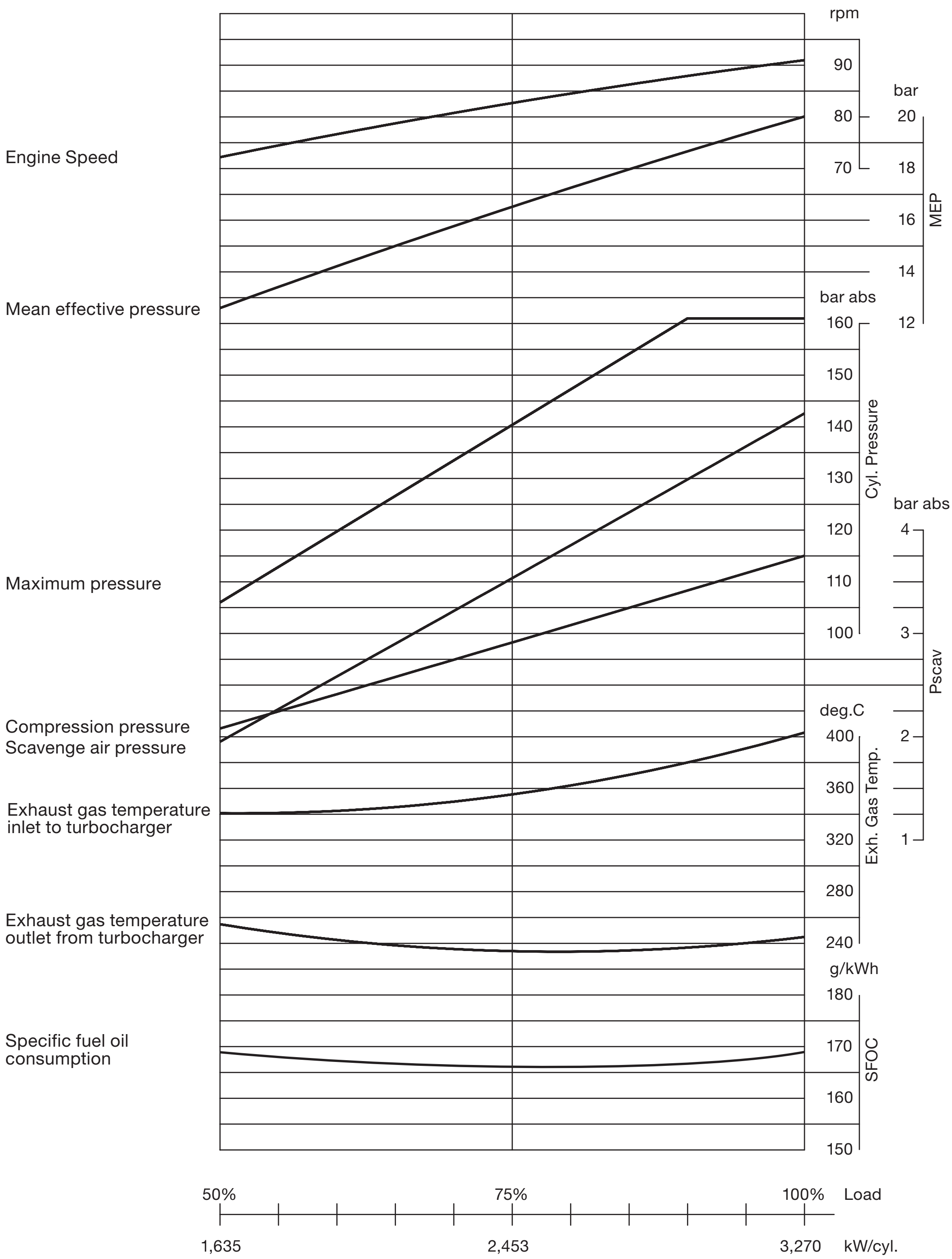

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Fig. 1.05.01: Performance curves with VIT

MC Engine Description

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

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

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

Bedplate and Main Bearing

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

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

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

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

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

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

Frame Box

The frame box is of triangular plate welded design on new and a range of recent engine types. Table 1.06.01 lists current engine types not yet updated from the rib to the triangular plate framebox design. On the exhaust side, it is provided with relief valves for each cylinder while, on the manoeuvring side, it is provided with a large hinged door for each cylinder. The crosshead guides are welded onto the frame box.

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

Cylinder Frame and Stuffing Box

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

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

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

Cylinder Liner

The cylinder liner is made of alloyed cast iron and is suspended in the cylinder frame with a low-situated flange. The top of the cylinder liner is fitted with a cooling jacket.

The cylinder liner has scavenge ports, drilled holes for cylinder lubrication and is prepared for installation of temperature sensors, if required.

Cylinder Cover

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

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

Crankshaft

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

At the aft end, the crankshaft is provided with the collar for the thrust bearing, and the flange for the turning wheel and for the coupling bolts to an intermediate shaft.

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

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

Thrust Bearing

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

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

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

Turning Gear and Turning Wheel

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

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

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

Axial Vibration Damper

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

Engine Layout and Load Diagrams

Introduction

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

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

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

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

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

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

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

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

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

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

$$\log (P) = i \times \log (n) + \log (c)$$

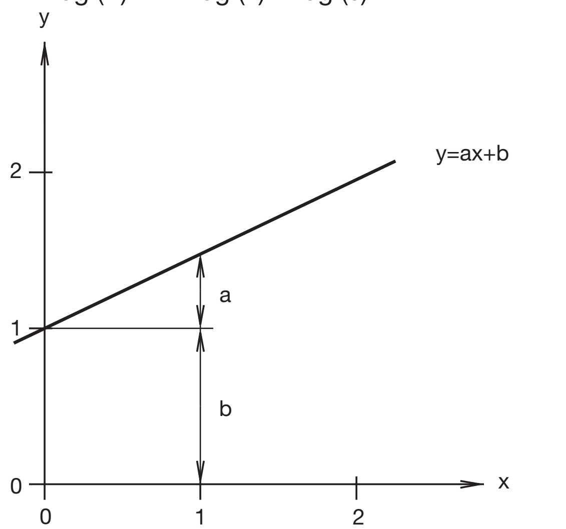


Fig. 2.01.01: Straight lines in linear scales

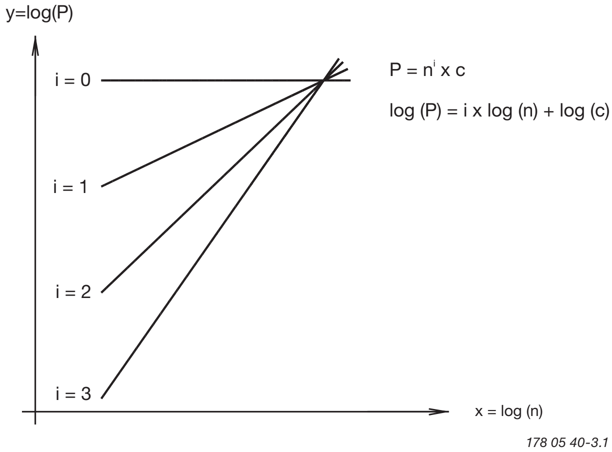


Fig. 2.01.02: Power function curves in logarithmic scales

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

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

Propulsion and Engine Running Points

Propeller curve

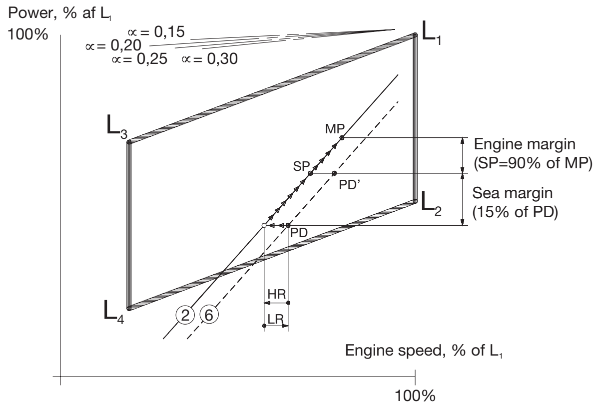
The relation between power and propeller speed for a fixed pitch propeller is as mentioned above described by means of the propeller law, i.e. the third power curve:

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

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

Propeller design point

Normally, estimates of the necessary propeller power and speed are based on theoretical calculations for loaded ship, and often experimental tank tests, both assuming optimum operating conditions, i.e. a clean hull and good weather. The combination of speed and power obtained may be called the ship's propeller design point (PD), placed on the light running propeller curve 6. See below figure. On the other hand, some shipyards, and/or propeller manufacturers sometimes use a propeller design point (PD) that incorporates all or part of the so-called sea margin described below.



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

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Fig. 2.01.03: Ship propulsion running points and engine layout

Fouled hull

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

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

Sea margin and heavy weather

If, at the same time the weather is bad, with head winds, the ship's resistance may increase compared to operating in calm weather conditions. When determining the necessary engine power, it is normal practice to add an extra power margin, the so-called sea margin, which is traditionally about 15% of the propeller design (PD) power.

Engine layout (heavy propeller)

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

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

Engine margin

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

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

Constant ship speed lines

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

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

Propeller diameter and pitch, influence on the optimum propeller speed

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

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

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

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

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

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

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

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

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

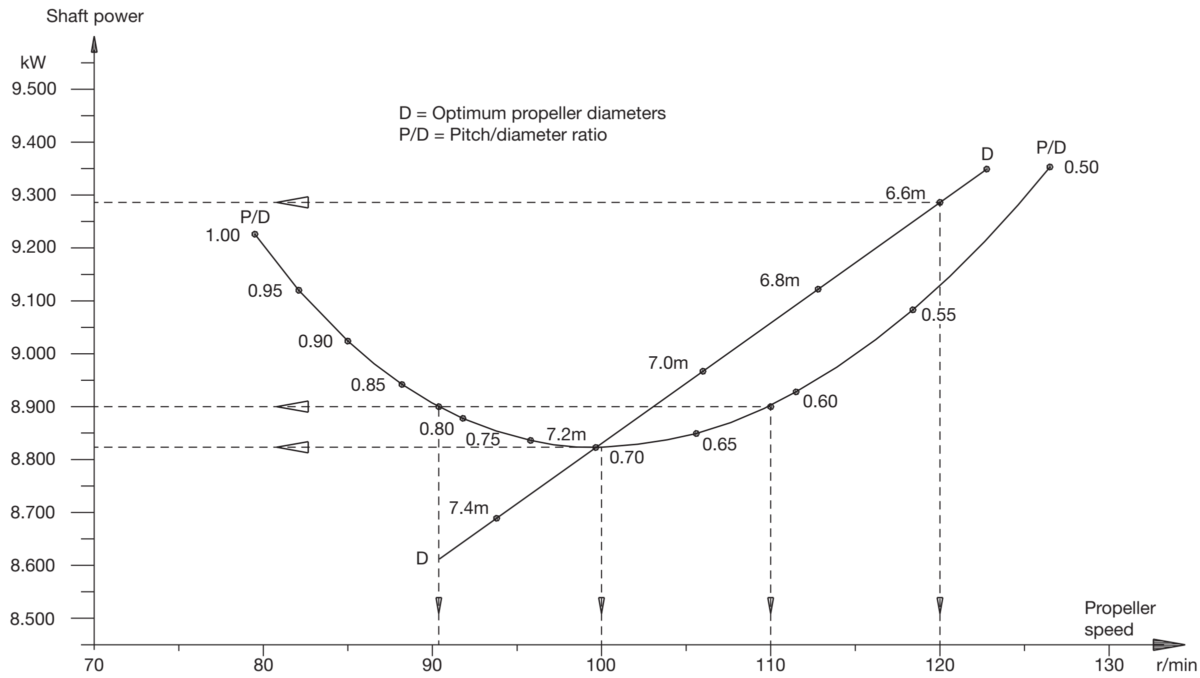

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Fig. 2.02.01: Influence of diameter and pitch on propeller design

Constant ship speed lines

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

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

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

where:
P = Propulsion power
n = Propeller speed, and
∝= the constant ship speed coefficient.

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

When such a constant ship speed line is drawn into the layout diagram through a specified propulsion MCR point 'MP_1', selected in the layout area and parallel to one of the ∝-lines, another specified propulsion MCR point 'MP_2' upon this line can be chosen to give the ship the same speed for the new combination of engine power and speed.

Fig. 2.02.02 shows an example of the required power speed point MP_1, through which a constant ship speed curve ∝= 0.25 is drawn, obtaining point MP_2 with a lower engine power and a lower engine speed but achieving the same ship speed.

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

for general cargo, bulk carriers and tankers
∝= 0.25 -0.30

and for reefers and container vessels
∝= 0.15 -0.25

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

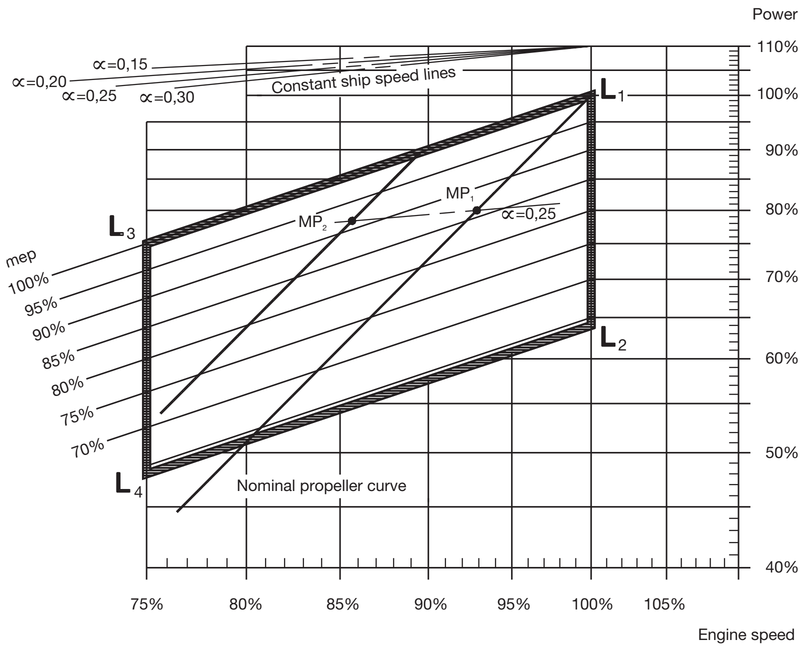

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Fig. 2.02.02: Layout diagram and constant ship speed lines

Engine Layout and Load Diagram

Engine Layout Diagram

An engine's layout diagram is limited by two constant mean effective pressure (mep) lines $L_1 – L_3$ and $L_2 – L_4$, and by two constant engine speed lines $L_1 – L_2$ and $L_3 – L_4$. The L_1 point refers to the engine's nominal maximum continuous rating, see Fig. 2.01.03.

In the layout area, the engine's specified MCR point M can be set freely to suit the ship's demand for propeller power and speed.

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

Specified maximum continuous rating (M)
Based on the propulsion and engine running points, found previously, the layout diagram of a relevant main engine may be drawn-in. The specified MCR point (M) must be inside the limitation lines of the layout diagram; if it is not, the propeller speed must be changed or another main engine type chosen. Yet, in special cases, point M may be located to the right of the line L_1-L_2, see 'Optimising point' below.

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

Optimising point (O)
The optimising point O is the rating at which the turbocharger is matched, and at which the engine timing and compression ratio are adjusted.

Optimising point (O) = specified MCR (M) for engine without VIT
In its basic design the engine type is not fitted with VIT fuel pumps, so the specified MCR is the point at which the engine is optimised – point M coincides with point O.

Optimising point (O) for engine with VIT
The engine can be fitted with VIT fuel pumps, option: 4 35 104, in order to improve the SFOC.

The optimising point O is placed on line 1 of the load diagram, see below, and the optimised power can be from 85% to 100% of point M's power, when turbocharger(s) and engine timing are taken into consideration. When optimising between 85% and 100% of point M's power, overload running will still be possible (110% of M).

The optimising point O is to be placed inside the layout diagram. In fact, the specified MCR point M can, in special cases, be placed outside the layout diagram, but only by exceeding line L_1-L_2, and of course, only provided that the optimising point O is located inside the layout diagram, and that the MCR power is not higher than the L_1 power.

Engine Load Diagram

Definitions

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

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

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

Limits for continuous operation

The continuous service range is limited by four lines:

Line 3 and line 9:
Line 3 represents the maximum acceptable speed for continuous operation, i.e. 105% of A.

If, in special cases, A is located to the right of line L_1-L_2, the maximum limit is, however, 105% of L_1.

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

The above limits may in general be extended to 105%, and during trial conditions to 107%, of the nominal L_1 speed of the engine, if permitted by the torsional vibration conditions.

The overspeed set-point is 109% of the speed in A, however, it may be moved to 109% of the nominal speed in L_1, if permitted by that torsional vibration conditions.

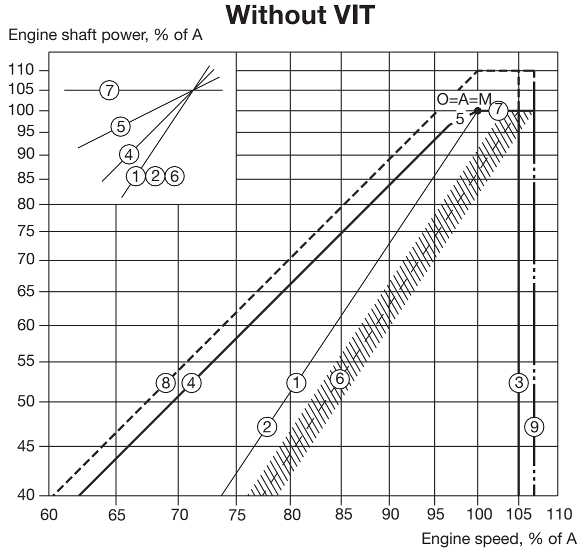
Running above 100% of the nominal L_1 speed at a load lower than about 65% specified MCR should, however, be avoided for extended periods. Only plants with controllable pitch propellers can reach this light running area.

Line 4:
Represents the limit at which an ample air supply is available for combustion and imposes a limitation on the maximum combination of torque and speed.

Line 5:
Represents the maximum mean effective pressure level (mep), which can be accepted for continuous operation.

Line 7:
Represents the maximum power for continuous operation.

Without VIT



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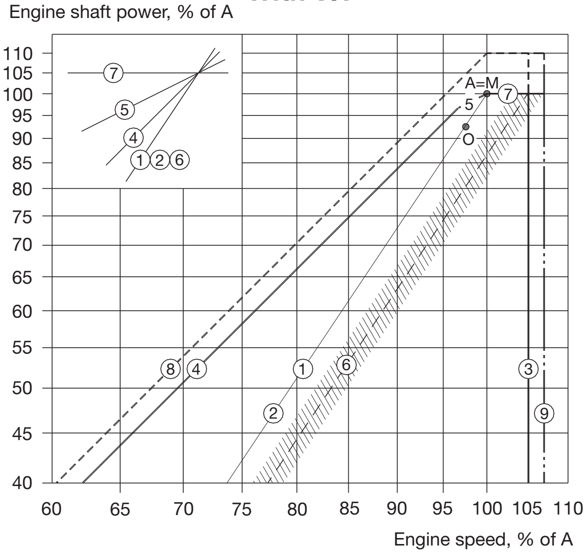
With VIT



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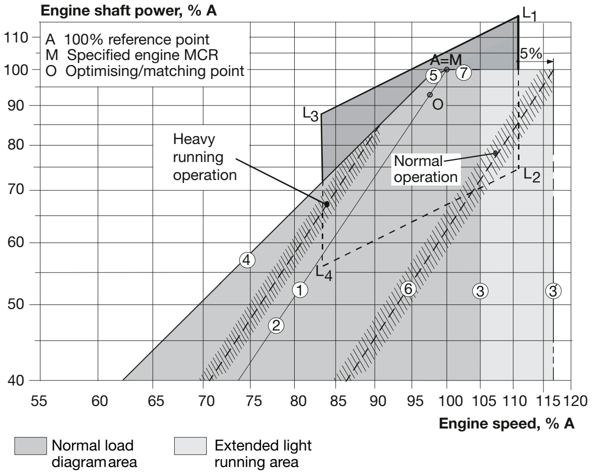
A	100% reference point
M	Specified MCR point
O	Optimising point

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

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

Fig. 2.04.01: Standard engine load diagram



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

178 52 25-6.0

Fig. 2.04.02: Extended load diagram for speed derated engine with increased light running

Examples of the use of the Load Diagram

In the following, some examples are illustrating the flexibility of the layout and load diagrams and the significant influence of the choice of the optimising point O.

The upper diagrams of the examples show engines **without** VIT fuel pumps, i.e. point A = O, the lower diagrams show engines **with** VIT fuel pumps for which the optimising point O is normally different from the specified MCR point M as this can improve the SFOC at part load running.

- Example 1 shows how to place the load diagram for an engine without a shaft generator coupled to a fixed pitch propeller.
- Example 2 comprises diagrams for the same configuration, here with the optimising point on the left of the heavy running propeller curve (2) providing an extra engine margin for heavy running.
- Example 3 shows the same layout for an engine with fixed pitch propeller (Example 1), but with a shaft generator.
- Example 4 shows a special case with a shaft generator. In this case, the shaft generator is cut off, and the gensets used when the engine runs at specified MCR. This makes it possible to choose a smaller engine with a lower power output.
- Example 5 shows diagrams for an engine coupled to a controllable pitch propeller, with or without a shaft generator, (constant speed or combinator curve operation).
- Example 6 shows where to place the optimising point for an engine coupled to a controllable pitch propeller, and operating at constant speed.

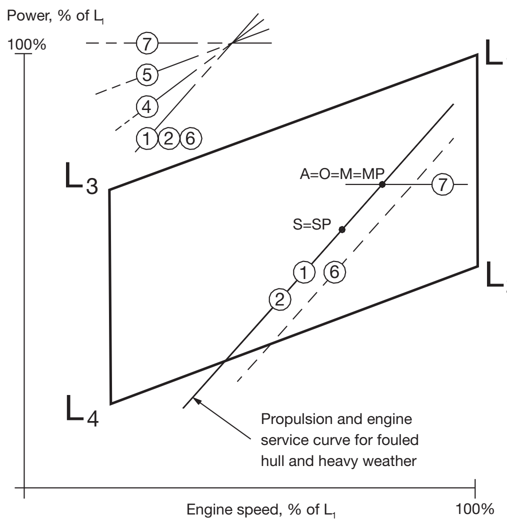
For a specific project, the layout diagram for the actual project shown later in this chapter may be used for drawing of the actual load diagram.

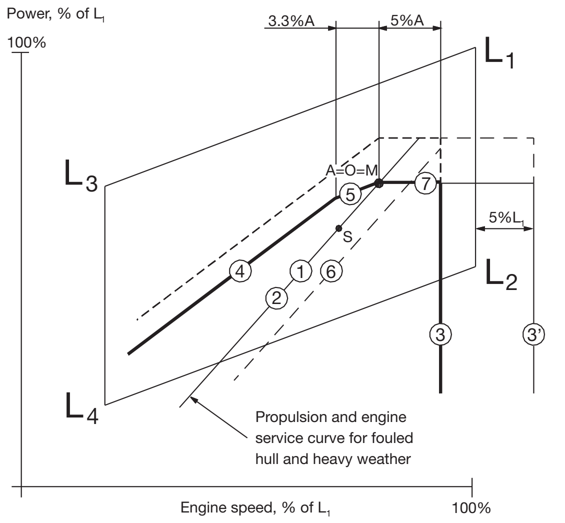
Example 1: Normal running conditions.
Engine coupled to a fixed pitch propeller (FPP) and without a shaft generator

Layout diagram

Load diagram

Without VIT





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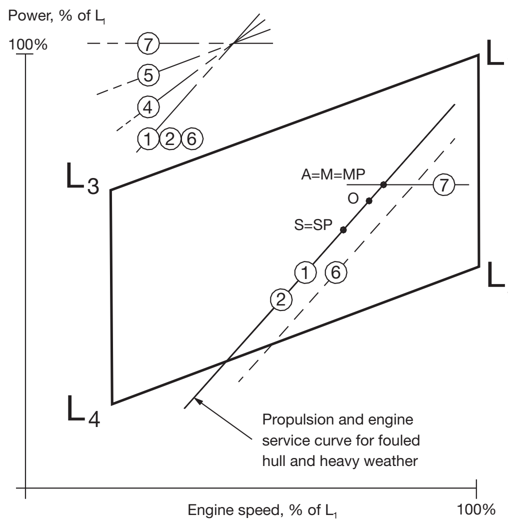
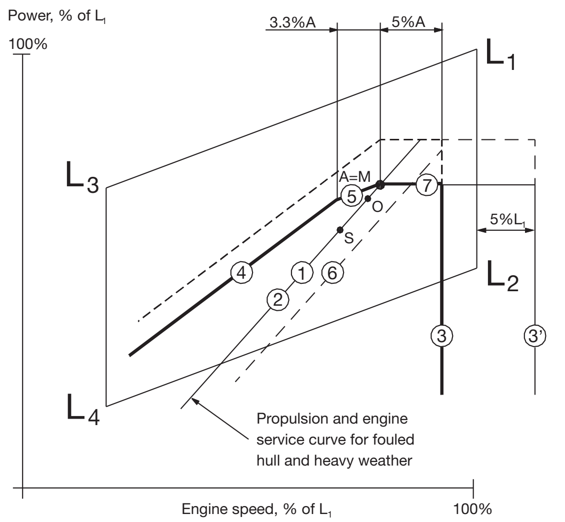
With VIT





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

Point A of the load diagram is found:

Line 1	Propeller curve through optimising point (O) is equal to line 2
Line 7	Constant power line through specified MCR (M)
Point A	Intersection between line 1 and 7

For engines without VIT, the optimising point O will have the same power as point M, and its propeller curve 1 for engine layout will normally be selected on the engine service curve 2 (for fouled hull and heavy weather), as shown in the upper diagram.

For engines with VIT, the optimising point O and its propeller curve 1 will normally be selected on the engine service curve 2, see the lower diagram.

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

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

Fig. 2.04.03: Normal running conditions. Engine coupled to a fixed pitch propeller (FPP) and without a shaft generator

SFOC for High Efficiency/Conventional Turbochargers

All engine types are as standard fitted with high efficiency turbochargers (EoD option: 4 59 104) but can alternatively use conventional turbochargers, option: 4 59 107.

The high efficiency turbocharger is applied to the engine in the basic design with the view to obtaining the lowest possible Specific Fuel Oil Consumption (SFOC) values, see curve A, Fig. 2.07.01.

With a conventional turbocharger the amount of air required for combustion purposes can, however, be adjusted to provide a higher exhaust gas temperature, if this is needed for the exhaust gas boiler.

The matching of the engine and the turbocharging system is then modified, thus increasing the exhaust gas temperature by 20 °C.

This modification will lead to a 7-8% reduction in the exhaust gas amount, and involve an SFOC penalty of 2 g/kWh, see curve B, Fig. 2.07.01.

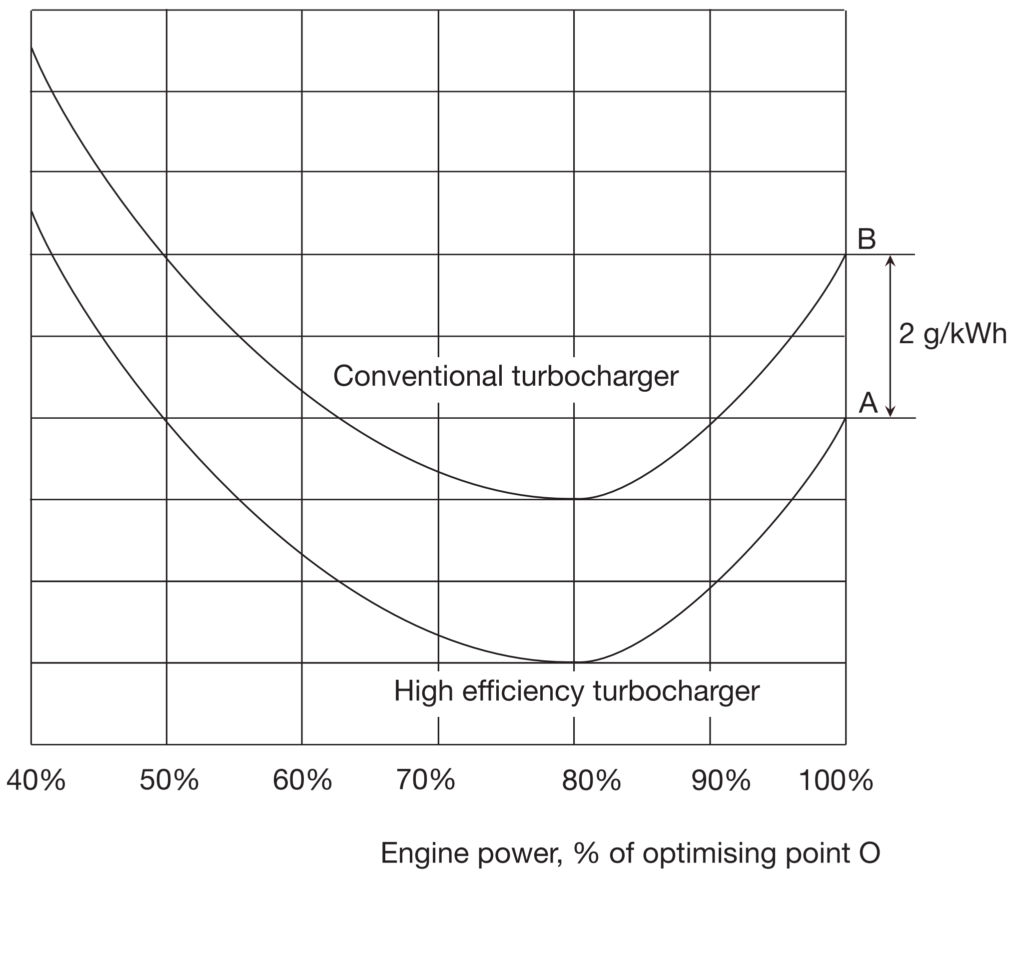

Fig. 2.07.01: Example of part load SFOC curves for high efficiency and conventional turbochargers

SFOC reference conditions and guarantee

SFOC at reference conditions

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

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

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

Any discrepancies between g/kWh and g/BHPh are a result of the rounding of numbers for the latter.

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

		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 oil lower calorific value	rise 1% (42,700 kJ/kg)	-1.00%	- 1.00%

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

SFOC guarantee

The SFOC guarantee refers to the above ISO reference conditions and lower calorific value. It is guaranteed for the power-speed combination in the optimising point (O) and the engine running 'fuel economy mode' in compliance with IMO NO_x emission limitations.

The SFOC guarantee is given with a tolerance of 5%

With or without VIT fuel pumps

In its basic design this engine type is fitted with fuel pumps without Variable Injection Timing (VIT), so the optimising point 'O' then has to be at the specified MCR power 'M'.

VIT fuel pumps can, however, be fitted as an option: 4 35 104, and, in that case, they can be optimised between 85-100% of the specified MCR, point 'M', as for the other large MC engine types. Engines with VIT fuel pumps can be part-load optimised between 85-100% of the specified MCR.

To facilitate the graphic calculation of SFOC, we use the same diagrams b and c for guidance in both cases, the only difference is the location of the optimising point.

In the part load area from approx. 60-95%, the exact SFOC calculated by our computer program will result in a slightly improved SFOC compared to engines without VIT fuel pumps.

Examples of graphic calculation of SFOC

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

The solid lines are valid at 100%, 80% and 50% of the optimising point (O).

Point O is drawn into the above-mentioned b or c diagram. A straight line along the constant mep curves (parallel to L_1-L_3) is drawn through point O. The intersections of this line and the curves indicate the reduction in specific fuel oil consumption at 100%, 80% and 50% of the optimising point, related to the SFOC stated for the nominal MCR L_1 rating.

An example of the calculated SFOC curves for an engine with fixed pitch propeller is shown in Diagram a, and is valid for two alternative engine optimising points:

- Optimising point O_1 at 100% of M

- Optimising point O_2 at 90% of M

See Fig. 2.10.01.

The optmising point typically chosen is 90%, randomly chosen between 85-100% in order to reduce SFOC at part load running.

SFOC Calculation for S70MC-C8

Data at nominel MCR (L_1)			SFOC at nominal MCR (L_1)	
			High efficiency TC	**Conventional TC**
Engine	**kW**	**r/min**	**g/kWh**	**g/kWh**
5 S70MC-C8	16,350	91	169	171
6 S70MC-C8	19,620			
7 S70MC-C8	22,890			
8 S70MC-C8	26,160			

Data optimising point (O):

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

Diagram a

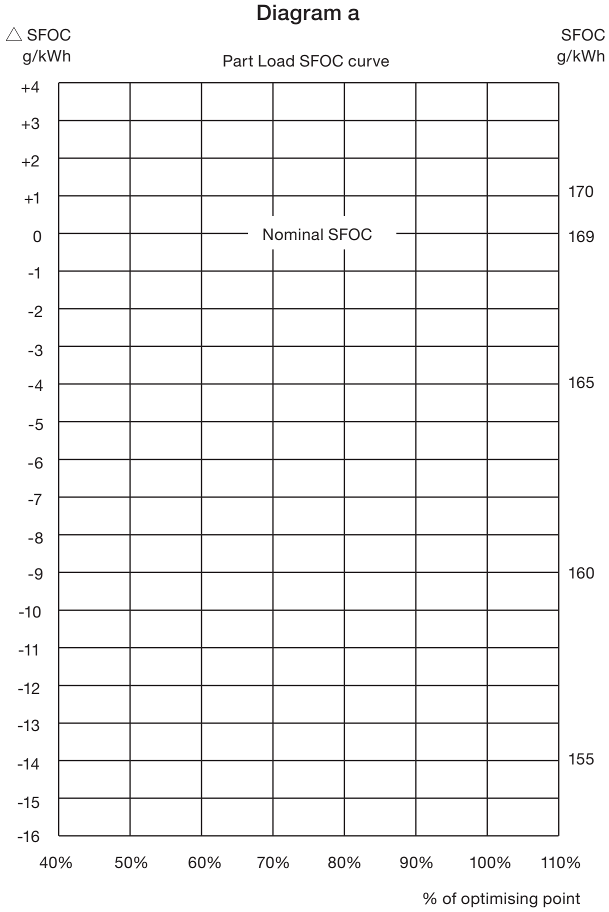

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Fig. 2.09.01

SFOC calculations, example

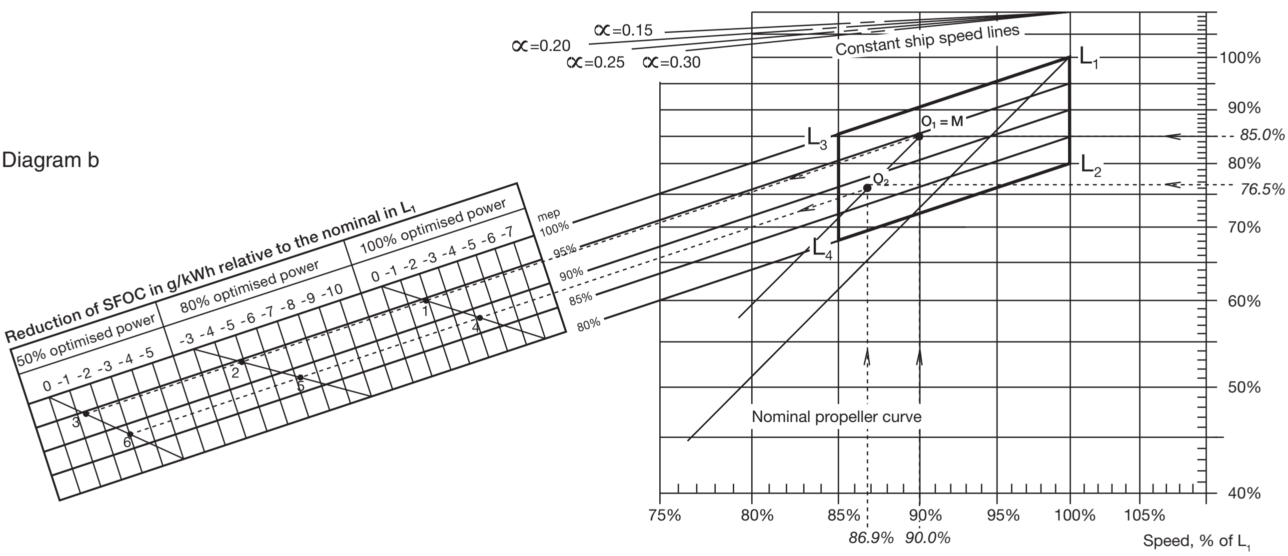
Data at nominel MCR (L_1): 6S70MC-C8	
Power 100%	19,620 kW
Speed 100%	91 r/min
Nominal SFOC:	
• High efficiency turbocharger	169 g/kWh
• Conventional turbocharger	171 g/kWh

Example of specified MCR = M		
Power	16,677 kW (85.0% L_1)	
Speed	81.9 r/min (90.0% L_1)	
Turbocharger type	High efficiency	
Optimising point (O)	O_1	O_2
Two alternatives	100% SMCR	90% SMCR
Power of O	16,677 kW	15,009 kW
Speed of O	81.9 r/min	79.1 r/min
SFOC found in O	167.1 g/kWh	164.8 g/kWh

Two alternative optimising points, O_1 and O_2 are used in the above example for the SFOC calculations:

O_1 = 100% M = 85.0% L_1 power and 90.0% L_1 speed
O_2 = 90% M = 76.5% L_1 power and 86.9% L_1 speed

Diagram b



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

Power in	Part load points		SFOC g/kWh	SFOC g/kWh
100% O_1	1	100% M	-1.9	167.1
80% O_1	2	80% M	-4.9	164.1
50% O_1	3	50% M	-1.4	167.6
100% O_2	4	90% M	-4.2	164.8
80% O_2	5	72% M	-7.2	161.8
50% O_2	6	45% M	-3.0	166.0

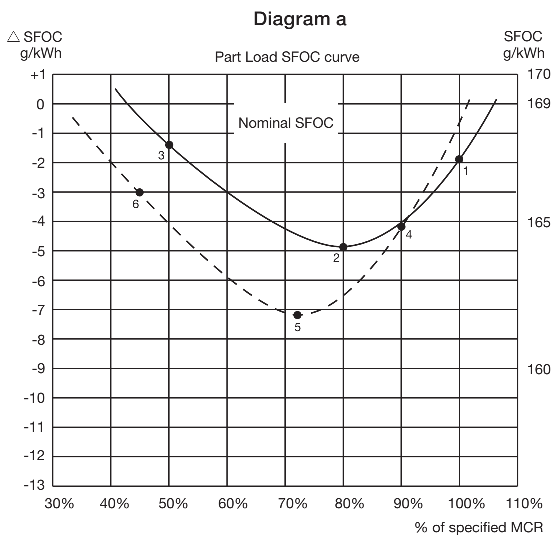

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Fig. 2.10.01: Example of SFOC for derated S70MC-C8 with fixed pitch propeller and high efficiency turbocharger

Fuel Consumption at an Arbitrary Load

Once the optimising point (O) of the engine has been chosen, the specific fuel oil consumption at an arbitrary point S_1, S_2 or S_3 can be estimated based on the SFOC at point '1' and '2'.

These SFOC values can be calculated by using the graphs for the relevant engine type for the propeller curve I and for the constant speed curve II, giving the SFOC at points 1 and 2, respectively.

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 symmetrical about point 1, i.e. at speeds lower than that of point 1, the SFOC will also increase.

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

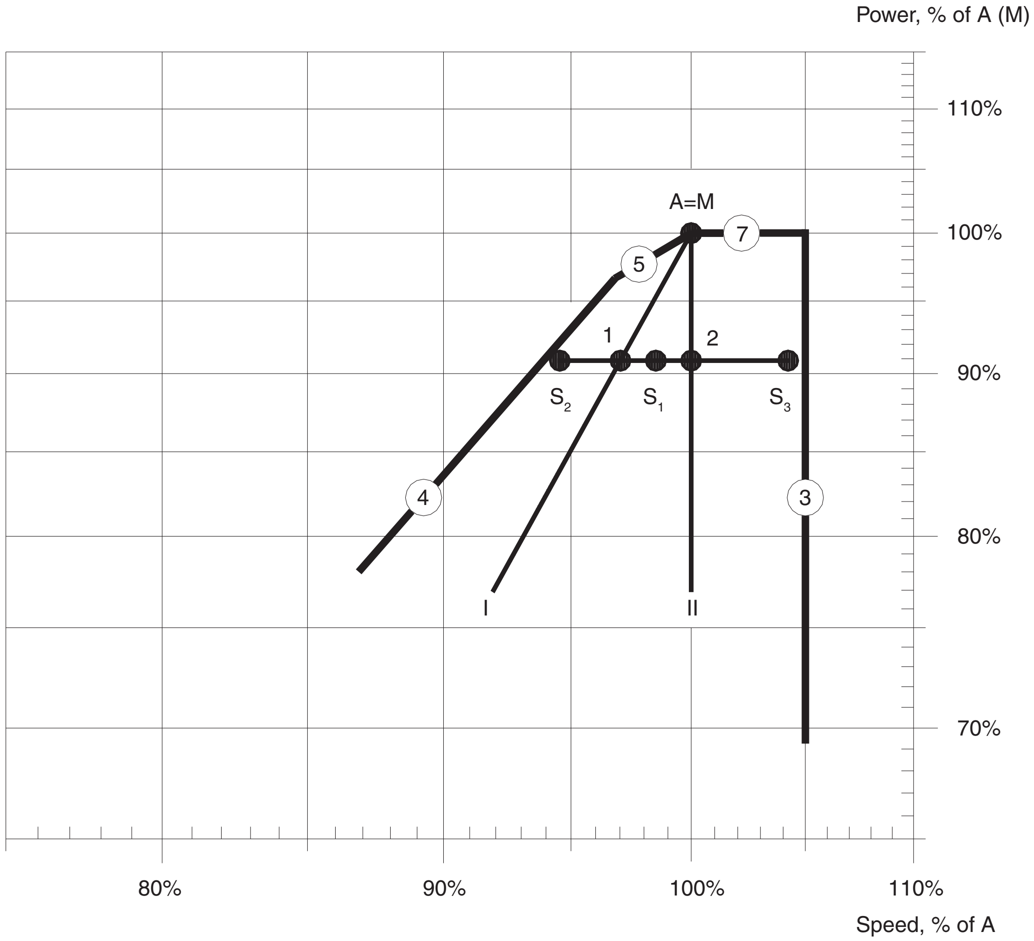
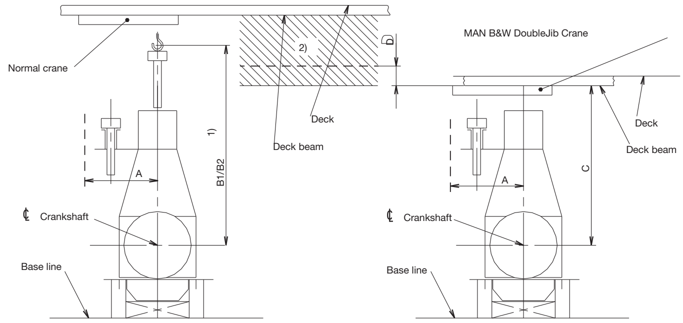
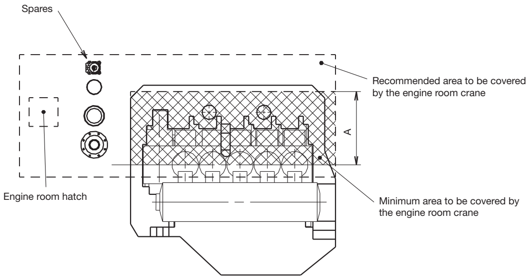


Fig. 2.11.01: SFOC at an arbitrary load

Engine room crane





1) The lifting tools for the engine are designed to fit together with a standard crane hook with a lifting capacity in accordance with the figure stated in the table. If a larger crane hook is used, it may not fit directly to the overhaul tools, and the use of an intermediate shackle or similar between the lifting tool and the crane hook will affect the requirements for the minimum lifting height in the engine room (dimension B)

2) The hatched area shows the height where an MAN B&W Double-Jib Crane has to be used.

Fig. 5.04.01: Engine room crane

Weight in kg including lifting tools			**Crane capacity in tons selected in accordance with DIN and JIS standard capacities**		**Crane operating width in mm**	**Normal crane Height to crane hook in mm for:**		**MAN B&W Double-Jib Crane**	
						Normal lifting procedure	**Reduced height lifting procedure involving tilting of main components (option)**	**Building-in height in mm**	
Cylinder cover complete with exhaust valve	Cylinder liner with cooling jacket	Piston with piston rod and stuffing box	Normal crane	MAN B&W Double-Jib Crane	A Minimum distance	B1 Minimum height from centre line crankshaft to centre line crane hook	B1 Minimum height from centre line crankshaft to underside deck beam	C Minimum height from centre line crankshaft to underside deck beam	D Additional height required for removal of exhaust valve without removing any exhaust valve stud
4,600	5,425	2,550	6.3	2x3.0	2,850	11,475	11,675	11,425	425

The crane hook travelling area must cover at least the full length of the engine and a width in accordance with dimension A given on the drawing, see cross-hatched area.

It is furthermore recommended that the engine room crane can be used for transport of heavy spare parts from the engine room hatch to the spare part stores and to the engine. See example on this drawing.

The crane hook should at least be able to reach down to a level corresponding to the centre line of the crankshaft.

For overhaul of the turbocharger(s), trolley mounted chain hoists must be installed on a separate crane beam or, alternatively, in combination with the engine room crane structure, see 'Crane beam for overhaul of turbochargers' with information about the required lifting capacity for overhaul of turbocharger(s).

Mass of Water and Oil

No. of cylinders	Mass of water and oil in engine in service					
	Mass of water			Mass of oil		
	Jacket cooling water kg	Scavenge air cooling water kg	Total kg	Engine system kg	Oil pan kg	Total kg
5	537	50	587	682	783	1,465
6	634	378	1,012	942	1,100	2,042
7	858	291	1,149	1,146	1,123	2,269
8	862	713	1,575	1,286	1,258	2,544

Fig. 5.08.01: Water and oil in engine

Calculation of List of Capacities and Exhaust Gas Data

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

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

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

Nomenclature

In the following description and examples of the auxiliary machinery capacities, freshwater generator pro-

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

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

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

Engine configurations related to SFOC

The engine type is available in the following two versions with respect to the efficiency of the turbocharger:

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

- **B) With conventional turbocharger,:** Which is an optional design (EoD No. 4 59 107) if a higher exhaust gas temperature is required for the exhaust gas boiler. This modification will lead to a 7-8% reduction in the exhaust gas amount and a temperature increase of about 20°C. The SFOC penalty will be 2 g/kWh. The corresponding lists of capacities are shown in Section 6.03.

List of Capacities and Cooling Water Systems

The List of Capacities contain data regarding the necessary capacities of the auxiliary machinery for the main engine only, and refer to a nominally rated engine. Complying with IMO Tier I 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 nominal MCR (L_1) for:

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

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

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

Heat radiation and air consumption

The radiation and convection heat losses to the engine room is around 1% of the engine nominal power (kW in L_1).

The air consumption is approximately 98.2% of the calculated exhaust gas amount, ie. $M_{air} = M_{exh} \times 0.982$.

Flanges on engine, etc.

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

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

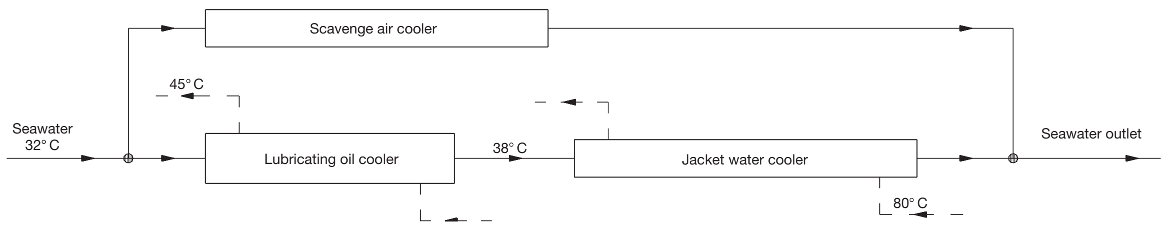


Fig. 6.02.01: Diagram for seawater cooling system

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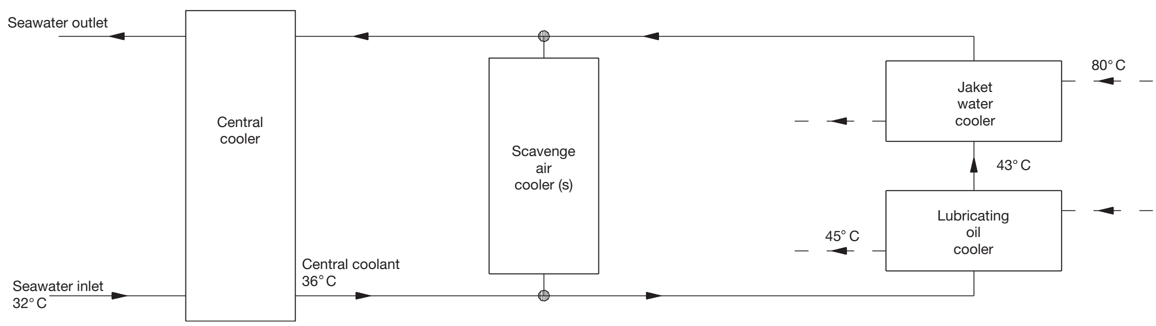

Fig. 6.02.02: Diagram for central cooling water system

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List of Capacities for 6S70MC-C8 at NMCR - IMO NO_x Tier I compliance

		Seawater cooling						Central cooling					
		Conventional TC			High eff. TC			Conventional TC			High eff. TC		
		1 x TCA88-20	1 x TPL85-B14	1 x MET83MA	1 x TCA88-20	1 x TPL85-B15	1 x MET83MA	1 x TCA88-20	1 x TPL85-B14	1 x MET83MA	1 x TCA88-20	1 x TPL85-B15	1 x MET83MA
Pumps													
Fuel oil circulation	m³/h	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6
Fuel oil supply	m³/h	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9
Jacket cooling	m³/h	165.0	165.0	165.0	165.0	165.0	165.0	165.0	165.0	165.0	165.0	165.0	165.0
Seawater cooling *	m³/h	610.0	620.0	610.0	630.0	630.0	630.0	590.0	590.0	590.0	610.0	610.0	610.0
Main lubrication oil *	m³/h	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0
Central cooling *	m³/h	-	-	-	-	-	-	475	475	475	485	490	485
Scavenge air cooler(s)													
Heat diss. app.	kW	7,610	7,610	7,610	8,000	8,000	8,000	7,560	7,560	7,560	7,950	7,950	7,950
Central water flow	m³/h	-	-	-	-	-	-	264	264	264	276	276	276
Seawater flow	m³/h	396	396	396	414	414	414	-	-	-	-	-	-
Lubricating oil cooler													
Heat diss. app. *	kW	1,530	1,590	1,550	1,530	1,590	1,550	1,530	1,590	1,550	1,530	1,590	1,550
Lube oil flow *	m³/h	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0	400.0
Central water flow	m³/h	-	-	-	-	-	-	211	211	211	209	214	209
Seawater flow	m³/h	214	224	214	216	216	216	-	-	-	-	-	-
Jacket water cooler													
Heat diss. app.	kW	2,840	2,840	2,840	2,840	2,840	2,840	2,840	2,840	2,840	2,840	2,840	2,840
Jacket water flow	m³/h	165	165	165	165	165	165	165	165	165	165	165	165
Central water flow	m³/h	-	-	-	-	-	-	211	211	211	209	214	209
Seawater flow	m³/h	214	224	214	216	216	216	-	-	-	-	-	-
Central cooler													
Heat diss. app. *	kW	-	-	-	-	-	-	11,930	11,990	11,950	12,320	12,380	12,340
Central water flow	m³/h	-	-	-	-	-	-	475	475	475	485	490	485
Seawater flow	m³/h	-	-	-	-	-	-	590	590	590	610	610	610
Starting air system, 30.0 bar g, 12 starts. Fixed pitch propeller - reversible engine													
Receiver volume	m³	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0	2 x 8.0
Compressor cap.	m³	480	480	480	480	480	480	480	480	480	480	480	480
Starting air system, 30.0 bar g, 6 starts. Controllable pitch propeller - non-reversible engine													
Receiver volume	m³	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5	2 x 4.5
Compressor cap.	m³	270	270	270	270	270	270	270	270	270	270	270	270
Other values													
Fuel oil heater	kW	225	225	225	225	225	225	225	225	225	225	225	225
Exh. gas temp.	°C	265	265	265	245	245	245	265	265	265	245	245	245
Exh. gas amount	kg/h	168,000	168,000	168,000	181,800	181,800	181,800	168,000	168,000	168,000	181,800	181,800	181,800
Air consumption	kg/h	45.7	45.7	45.7	49.6	49.6	49.6	45.7	45.7	45.7	49.6	49.6	49.6

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

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

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

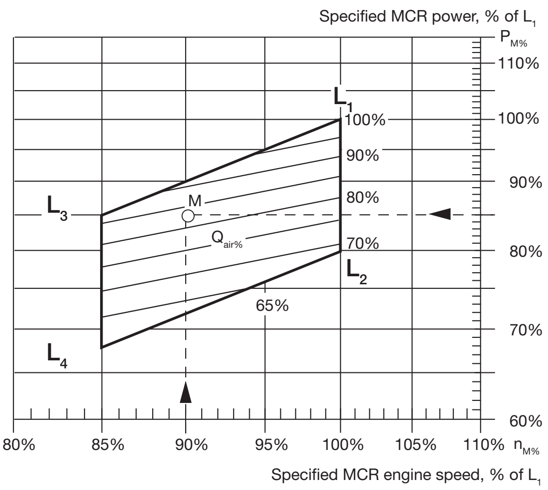
Auxiliary Machinery Capacities

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

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

Cooler heat dissipations

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



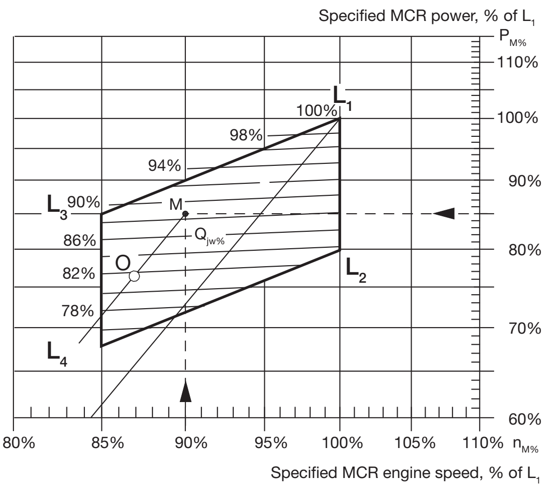
178 53 75-3.1

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

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

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

The percentage power ($P_{M\%}$) and speed ($n_{M\%}$) of L_1
ie: $P_{M\%} = P_M/P_{L1} \times 100\%$
$n_{M\%} = n_M/n_{L1} \times 100\%$
for specified MCR (M) of the derated engine is used as input in the above-mentioned diagrams, giving the % heat dissipation figures relative to those in the 'List of Capacities',



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

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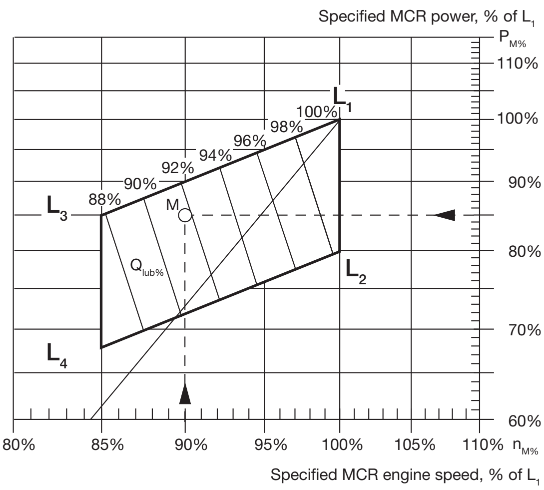
Fig. 6.04.02: Jacket water cooler, heat dissipation $Q_{jw\%}$ in point M, in % of the L_1 value $Q_{jw, L1}$



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

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

Calculation of List of Capacities for Derated Engine

Example 1:

Pump and cooler capacities for a *derated 6S70MC-C8 with high efficiency MAN Diesel turbocharger, type TCA with fixed pitch propeller and central cooling water system.*

Nominal MCR, (L_1) P_{L1}: 19,620 kW (100.0%) and 91.0 r/min (100.0%)

Specified MCR, (M) P_M: 16,677 kW (85.0%) and 81.9 r/min (90.0%)

Optimising point, (O) P_O: 15,009 kW (76.5%) and 79.1 r/min (86.9%), P_O = 90.0% of P_M

The method of calculating the reduced capacities for point M ($n_{M\%}$ = 90.0% and $P_{M\%}$ = 85.0%) is shown below.

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

Heat dissipation of scavenge air cooler
Fig. 6.04.01 which approximately indicates a $Q_{air\%}$ = 83.1% heat dissipation, and corrected for optimising point O lower than M, by applying correcting factor k_O, equal 83.1 x (1 + 0.27 x (1-0.900)) = 85.3%, i.e.:

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

$Q_{air,M}$ = 7,950 x 0.853 = 6,781 kW

Heat dissipation of jacket water cooler
Fig. 6.04.02 indicates a $Q_{jw\%}$ = 88.5% heat dissipation; i.e.:

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

$Q_{jw,M}$ = 2,840 x 0.885 = 2,513 kW

Heat dissipation of lube oil cooler
Fig. 6.04.03 indicates a $Q_{lub\%}$ = 91.7% heat dissipation; i.e.:

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

$Q_{lub,M}$ = 1,530 x 0.917 = 1,403 kW

Heat dissipation of central water cooler

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

$Q_{cent,M}$ = 6,781 + 2,513 + 1,403 = 10,697 kW

Total cooling water flow through scavenge air coolers

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

$V_{cw,air,M}$ = 276 x 0.853 = 235 m^3/h

Cooling water flow through lubricating oil cooler

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

$V_{cw,lub,M}$ = 209 x 0.917 = 192 m^3/h

Cooling water flow through central cooler (Central cooling water pump)

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

$V_{cw,cent,M}$ = 235 + 192 = 427 m^3/h

Cooling water flow through jacket water cooler (as for lube oil cooler)

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

$V_{cw,jw,M}$ = 192 m^3/h

Seawater pump for central cooler
As the seawater pump capacity and the central cooler heat dissipation for the nominal rated engine found in the 'List of Capacities' are 970 m^3/h and 19,730 kW the derated seawater pump flow equals:

Seawater pump:

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

= 610 x 10,697 / 12,320 = 530 m^3/h

		Nominal rated engine (L_1) High efficiency turbocharger (TCA)	Example 1 Specified MCR (M)
Shaft power at MCR		19,620 kW	16,677 kW
Engine speed at MCR		at 91.0 r/min	at 81.9 r/min
Power of optimising point %MCR		100%	90%
Pumps:			
Fuel oil circulating pump	m^3/h	8.6	8.6
Fuel oil supply pump	m^3/h	4.9	4.9
Jacket cooling water pump	m^3/h	165	165
Central cooling water pump	m^3/h	485	427
Seawater pump	m^3/h	610	530
Lubricating oil pump	m^3/h	400	400
Coolers:			
Scavenge air cooler			
Heat dissipation	kW	7,950	6,781
Central water quantity	m^3/h	276	235
Lub. oil cooler			
Heat dissipation	kW	1,530	1,403
Lubricating oil quantity	m^3/h	400	400
Central water quantity	m^3/h	209	192
Jacket water cooler			
Heat dissipation	kW	2,840	2,513
Jacket cooling water quantity	m^3/h	165	165
Central water quantity	m^3/h	209	192
Central cooler			
Heat dissipation	kW	12,320	10,697
Central water quantity	m^3/h	485	427
Seawater quantity	m^3/h	610	530
Fuel oil heater:	kW	225	225
Gases at ISO ambient conditions*			
Exhaust gas amount	kg/h	181,800	154,900
Exhaust gas temperature	°C	245	234.8
Air consumption	kg/sec.	49.6	42.2
Starting air system: 30 bar (gauge)			
Reversible engine			
Receiver volume (12 starts)	m^3	2 x 8.0	2 x 8.0
Compressor capacity, total	m^3/h	480	480
Non-reversible engine			
Receiver volume (6 starts)	m^3	2 x 4.5	2 x 4.5
Compressor capacity, total	m^3/h	270	270
Exhaust gas tolerances: temperature –/+ 15 °C and amount +/– 5%			

The air consumption and exhaust gas figures are expected and refer to 100% specified MCR, ISO ambient reference conditions and the exhaust gas back pressure 300 mm WC
The exhaust gas temperatures refer to after turbocharger
* Calculated in example 3, in this chapter

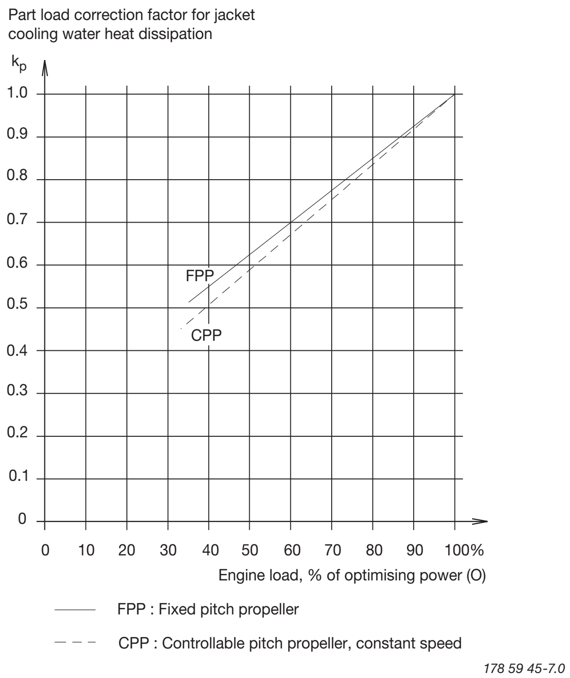
Example 1 – Capacities of derated 6S70MC-C8 with high efficiency MAN Diesel turbocharger type TCA and central cooling water system.

Freshwater Generator

If a freshwater generator is installed and is utilising the heat in the jacket water cooling system, it should be noted that the actual available heat in the jacket cooling water system is **lower** than indicated by the heat dissipation figures valid for nominal MCR (L_1) given in the List of Capacities. This is because the latter figures are used for dimensioning the jacket water cooler and hence incorporate a safety margin which can be needed when the engine is operating under conditions such as, e.g. overload. Normally, this margin is 10% at nominal MCR.

Calculation Method

For a derated diesel engine, i.e. an engine having a specified MCR (M) and/or a optimising point (O) different from L_1, the relative jacket water heat dissipation for point M and O may be found, as previously described, by means of Fig. 6.04.02.



FPP : $k_p = 0.742 \times \frac{P_S}{P_O} + 0.258$

CPP : $k_p = 0.822 \times \frac{P_S}{P_O} + 0.178$

Fig. 6.04.04: Correction factor 'kp' for jacket cooling water heat dissipation at part load, relative to heat dissipation at optimising power

At part load operation, lower than optimising power, the actual jacket water heat dissipation will be reduced according to the curves for fixed pitch propeller (FPP) or for constant speed, controllable pitch propeller (CPP), respectively, in Fig. 6.04.04.

With reference to the above, the heat actually available for a derated diesel engine may then be found as follows:

1. Engine power between optimising and specified power.

   For powers between specified MCR (M) and optimising power (O), the diagram Fig. 6.04.02 is to be used, i.e. giving the percentage correction factor '$Q_{jw\%}$' and hence for optimising power P_O:

   $$Q_{jw,O} = Q_{jw,L1} \times \frac{Q_{jw\%}}{100} \times 0.9 \ (0.88) \qquad [1]$$

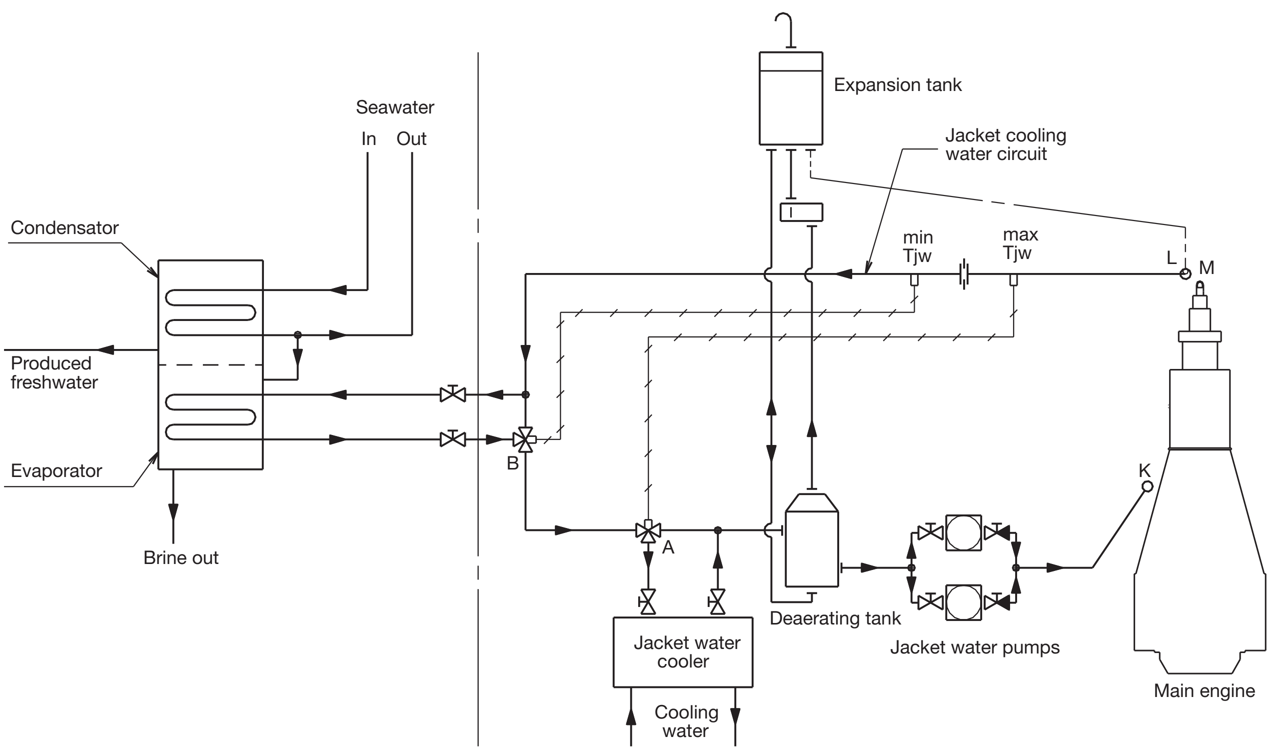
2. Engine power lower than optimising power.

   For powers lower than the optimising power, the value $Q_{jw,O}$ found for point O by means of the above equation [1] is to be multiplied by the correction factor k_p found in Fig. 6.04.04 and hence

   $$Q_{jw} = Q_{jw,O} \times k_p \ \text{-15\%/0\%} \qquad [2]$$

where
Q_{jw} = jacket water heat dissipation
$Q_{jw,L1}$ = jacket water heat dissipation at nominal MCR (L_1)
$Q_{jw\%}$ = percentage correction factor from Fig. 6.04.02
$Q_{jw,O}$ = jacket water heat dissipation at optimising power (O), found by means of equation [1]
k_p = part load correction factor from Fig. 6.04.04
0.9 = factor for safety margin of cooler, tropical ambient conditions

The heat dissipation is assumed to be more or less independent of the ambient temperature conditions, yet the safety margin/ambient condition factor of about 0.88 instead of 0.90 will be more accurate for ambient conditions corresponding to ISO temperatures or lower. The heat dissipation tolerance from -15% to 0% stated above is based on experience.



Valve A: ensures that T_{jw} < 85° C
Valve B: ensures that T_{jw} > 85 – 5° C = 80° C
Valve B and the corresponding by-pass may be omitted if, for example, the freshwater generator is equipped with an automatic start/stop function for too low jacket cooling water temperature
If necessary, all the actually available jacket cooling water heat may be utilised provided that a special temperature control system ensures that the jacket cooling water temperature at the outlet from the engine does not fall below a certain level

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Fig. 6.04.05: Freshwater generators. Jacket cooling water heat recovery flow diagram

Jacket Cooling Water Temperature Control

When using a normal freshwater generator of the single-effect vacuum evaporator type, the freshwater production may, for guidance, be estimated as 0.03 t/24h per 1 kW heat, i.e.:

M_{fw} = 0.03 x Q_{jw} t/24h -15%/0% [3]

where

M_{fw} is the freshwater production in tons per 24 hours

and

Q_{jw} is to be stated in kW

If necessary, all the actually available jacket cooling water heat may be used provided that a special temperature control system ensures that the jacket cooling water temperature at the outlet from the engine does not fall below a certain level. Such a temperature control system may consist, e.g., of a special by-pass pipe installed in the jacket cooling water system, see Fig. 6.04.05, or a special built-in temperature control in the freshwater generator, e.g., an automatic start/stop function, or similar.

If such a special temperature control is not applied, we recommend limiting the heat utilised to maximum 50% of the heat actually available at specified MCR, and only using the freshwater generator at engine loads above 50%. Considering the cooler margin of 10% and the minus tolerance of -15%, this heat corresponds to 50 x(1.00-0.15)x0.9 = 38% of the jacket water cooler capacity $Q_{jw,M}$ used for dimensioning of the jacket water cooler.

Calculation of Freshwater Production for Derated Engine

Example 2:

Freshwater production *from a derated 6S70MC-C8 with high efficiency MAN Diesel turbocharger of TCA type and with fixed pitch propeller.*

Based on the engine ratings below, this example will show how to calculate the expected available jacket cooling water heat removed from the diesel engine, together with the corresponding freshwater production from a freshwater generator.

The calculation is made for the service rating (S) of the diesel engine being 80% of the specified MCR.

Nominal MCR, (L_1) P_{L1}: 19,620 kW (100.0%) and 91.0 r/min (100.0%)

Specified MCR, (M) P_M: 16,677 kW (85.0%) and 81.9 r/min (90.0%)

Optimising point, (O) P_O: 15,009 kW (76.5%) and 79.1 r/min (86.9%), P_O = 90.0% of P_M

Service rating, (S) P_S: 13,342 kW and 76.0 r/min, P_S = 80.0% of P_M and P_S = 88.9% of P_O

Ambient reference conditions: 20° C air and 18° C cooling water.

The expected available jacket cooling water heat at service rating is found as follows:

$Q_{jw,L1}$ = 2,840 kW from List of Capacities
$Q_{jw\%}$ = 81.5% using 76.5% power and 86.9% speed for O in Fig. 6.04.02

By means of equation [1], and using factor 0.88 for actual ambient condition the heat dissipation in the optimising point (O) is found:

$$Q_{jw,O} = Q_{jw,L1} \times \frac{Q_{jw\%}}{100} \times 0.88$$

$$= 2{,}840 \times \frac{81.5}{100} \times 0.88 = 2{,}037 \text{ kW}$$

By means of equation [2], the heat dissipation in the service point (S) i.e. for 88.9% of optimising power, is found:

k_p = 0.918 using 88.9% in Fig. 6.04.04
Q_{jw} = $Q_{jw,O} \times k_p$ = 2,037 x 0.918 = 1,870 kW -15%/0%

For the service point the corresponding expected obtainable freshwater production from a freshwater generator of the single effect vacuum evaporator type is then found from equation [3]:

M_{fw} = 0.03 x Q_{jw} = 0.03 x 1,870 = 56.1 t/24h -15%/0%

Jacket water system

Due to the central cooler the cooling water inlet temperature is about 4 °C higher for for this system compared to the seawater cooling system. The input data are therefore different for the scavenge air cooler, the lube oil cooler and the jacket water cooler.

The heat dissipation and the central cooling water flow figures are based on an MCR output at tropical conditions, i.e. a maximum seawater temperature of 32 °C and an ambient air temperature of 45 °C.

Jacket water cooling pump

The pumps are to be of the centrifugal type.
Jacket water flow .... see 'List of capacities'
Pump head .... 3.0 bar
Delivery pressure .... depends on location of expansion tank
Test pressure.................... according to class rules
Working temperature .... 80 °C
Design temperature .... 100 °C

The flow capacity is to be within a tolerance of 0% to +10%.

The stated of capacities cover the main engine only. The pump head of the pumps is to be determined on the basis of the total actual pressure drop across the cooling water system.

Scavenge air cooler

The scavenge air cooler is an integrated part of the main engine.

Heat dissipation .... see 'List of capacities'
Central cooling water flow .... see 'List of capacities'
Central cooling temperature, inlet .... 36 °C
Pressure drop on FW-LT water side.... approx. 0.5 bar

Lubricating oil cooler

See chapter 8 'Lubricating Oil'.

Jacket water cooler

The cooler is to be of the shell and tube or plate heat exchanger type.

Heat dissipation .... see 'List of capacities'
Jacket water flow .... see 'List of capacities'
Jacket water temperature, inlet .... 80 °C
Pressure drop on jacket water side ... max. 0.2 bar
Central cooling water flow .... see 'List of capacities'
Central cooling water temperature, inlet .... approx. 42 °C
Pressure drop on Central cooling water side................................ max. 0.2 bar

The other data for the jacket cooling water system can be found in chapter 12.

For further information about a common cooling water system for main engines and MAN Diesel auxiliary engines, please refer to our publication:

Uni-concept Auxiliary Systems for Two-stroke Main

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

Jacket Cooling Water System

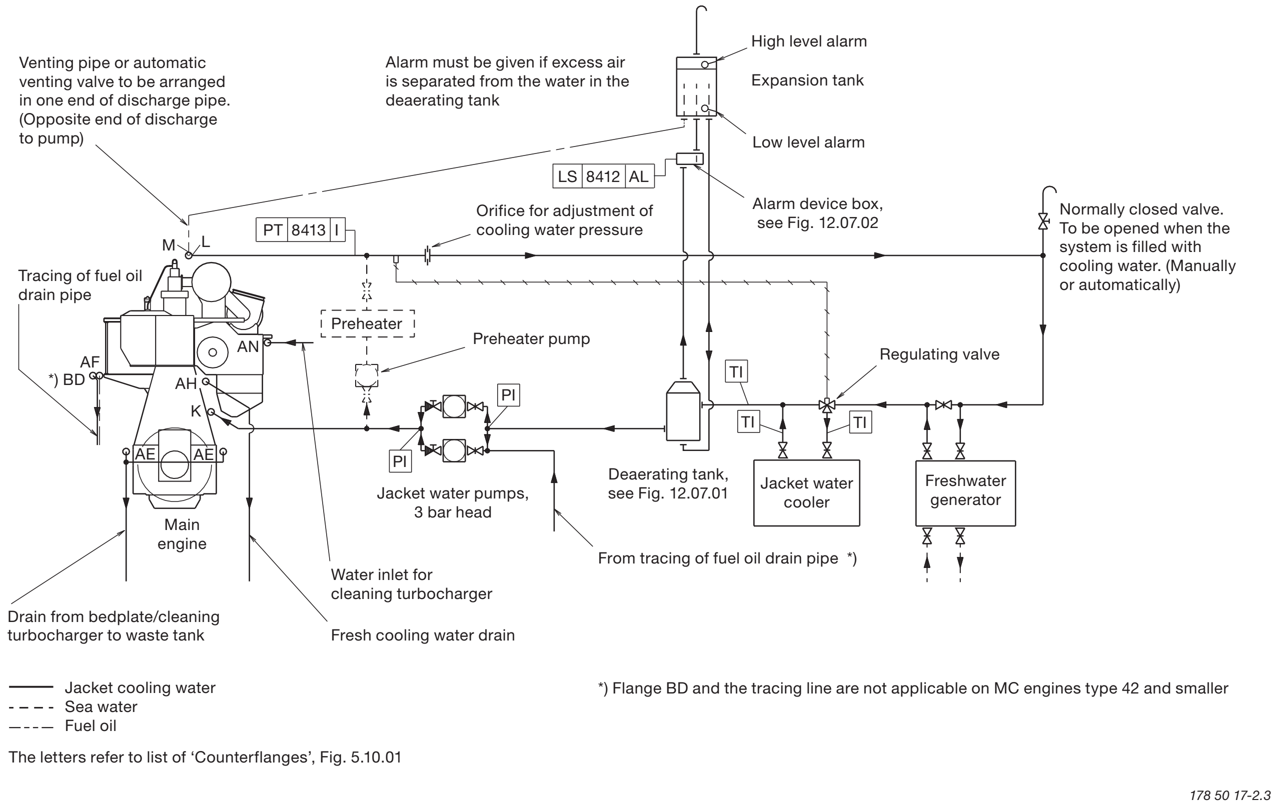

Fig. 12.05.01: Jacket cooling water system

The jacket cooling water system is used for cooling the cylinder liners, cylinder covers and exhaust valves of the main engine and heating of the fuel oil drain pipes, see Fig. 12.05.01.

The jacket water pump) draws water from the jacket water cooler outlet and delivers it to the engine.

At the inlet to the jacket water cooler there is a thermostatically controlled regulating valve, with a sensor at the engine cooling water outlet, which keeps the main engine cooling water outlet at a temperature of 80 °C.

The engine jacket water must be carefully treated, maintained and monitored so as to avoid corrosion, corrosion fatigue, cavitation and scale formation. It is recommended to install a preheater if preheating is not available from the auxiliary engines jacket cooling water system.

The venting pipe in the expansion tank should end just below the lowest water level, and the expansion tank must be located at least 5 m above the engine cooling water outlet pipe.

The freshwater generator, if installed, may be connected to the seawater system if the generator does not have a separate cooling water pump. The generator must be coupled in and out slowly over a period of at least 3 minutes.

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

Jacket water .... 3.0 m/s
Seawater .... 3.0 m/s

Components for Jacket Cooling Water System

Jacket water cooling pump

The pumps are to be of the centrifugal type.

Jacket water flow ....see 'List of capacities'
Pump head ....3.0 bar
Delivery pressure ....depends on position of expansion tank
Test pressure...................... according to class rule
Working temperature, ... 80 °C, max. 100 °C
The capacity must be met at a tolerance of 0% to +10%.

The stated capacities cover the main engine only. The pump head of the pumps is to be determined based on the total actual pressure drop across the cooling water system.

Freshwater generator

If a generator is installed in the ship for production of freshwater by utilising the heat in the jacket water cooling system it should be noted that the actual available heat in the jacket water system is lower than indicated by the heat dissipation figures given in the 'List of capacities.' This is because the latter figures are used for dimensioning the jacket water cooler and hence incorporate a safety margin which can be needed when the engine is operating under conditions such as, e.g. overload. Normally, this margin is 10% at nominal MCR.

The calculation of the heat actually available at specified MCR for a derated diesel engine is stated in chapter 6 'List of capacities'.

For illustration of installation of fresh water generator see Fig. 12.05.01.

Jacket water thermostatic valve

The temperature control system is equipped with a three-way valve mounted as a diverting valve, which by-pass all or part of the jacket water around the jacket water cooler.

The sensor is to be located at the outlet from the main engine, and the temperature level must be adjustable in the range of 70-90 °C.

Jacket water preheater

When a preheater, see Fig. 12.05.01, is installed in the jacket cooling water system, its water flow, and thus the preheater pump capacity, should be about 10% of the jacket water main pump capacity.

Based on experience, it is recommended that the pressure drop across the preheater should be approx. 0.2 bar. The preheater pump and main pump should be electrically interlocked to avoid the risk of simultaneous operation.

The preheater capacity depends on the required preheating time and the required temperature increase of the engine jacket water. The temperature and time relations are shown in Fig. 12.08.01.

In general, a temperature increase of about 35 °C (from 15 °C to 50 °C) is required, and a preheating time of 12 hours requires a preheater capacity of about 1% of the engine`s nominal MCR power.

Deaerating tank

Design and dimensions of the deaerating tank are shown in Fig. 12.07.01 'Deaerating tank' and the corresponding alarm device is shown in Fig. 12.07.02 'Deaerating tank, alarm device'.

Expansion tank

The total expansion tank volume has to be approximate 10% of the total jacket cooling water amount in the system.

Fresh water treatment

The MAN Diesel recommendations for treatment of the jacket water/freshwater are available on request.

Deaerating tank

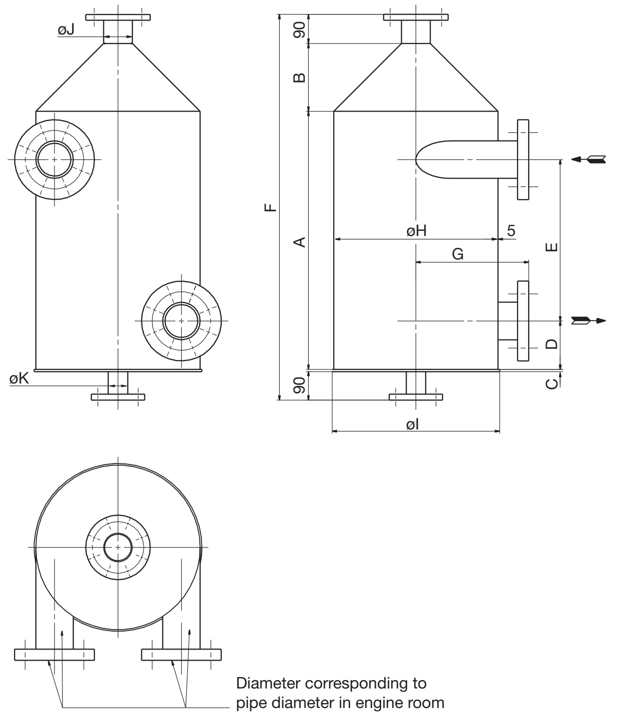

Fig. 12.07.01: Deaerating tank, option: 4 46 640

Deaerating tank dimensions		
Tank size	0.05 m^3	0.16 m^3
Max. jacket water capacity	120 m^3/h	300 m^3/h
	Dimensions in mm	
Max. nominal diameter	125	200
A	600	800
B	125	210
C	5	5
D	150	150
E	300	500
F	910	1,195
G	250	350
øH	300	500
øI	320	520
øJ	ND 50	ND 80
øK	ND 32	ND 50

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.

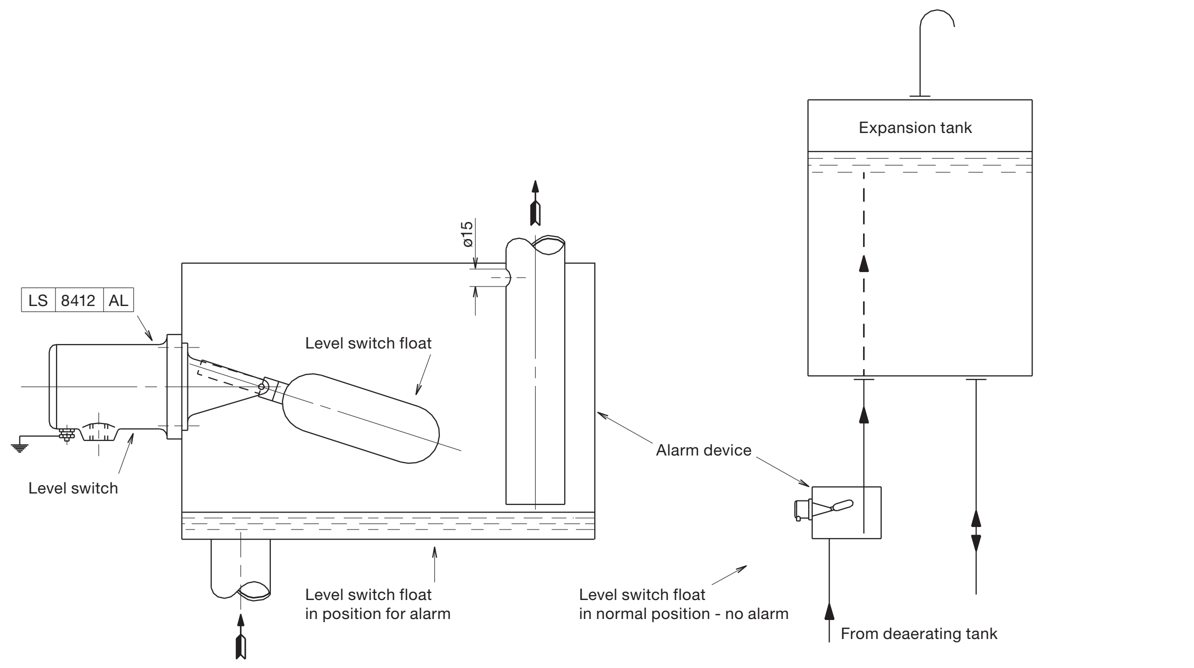

Fig. 12.07.02: Deaerating tank, alarm device, option: 4 46 645

Temperature at Start of Engine

In order to protect the engine, some minimum temperature restrictions have to be considered before starting the engine and, in order to avoid corrosive attacks on the cylinder liners during starting.

Normal start of engine

Normally, a minimum engine jacket water temperature of 50 °C is recommended before the engine is started and run up gradually to 90% of specified MCR speed.

For running between 90% and 100% of specified MCR speed, it is recommended that the load be increased slowly – i.e. over a period of 30 minutes.

Start of cold engine

In exceptional circumstances where it is not possible to comply with the above-mentioned recommendation, a minimum of 20 °C can be accepted before the engine is started and run up slowly to 90% of specified MCR speed.

However, before exceeding 90% specified MCR speed, a minimum engine temperature of 50 °C should be obtained and, increased slowly – i.e. over a period of at least 30 minutes.

The time period required for increasing the jacket water temperature from 20 °C to 50 °C will depend on the amount of water in the jacket cooling water system, and the engine load.

Note:
The above considerations are based on the assumption that the engine has already been well run-in.

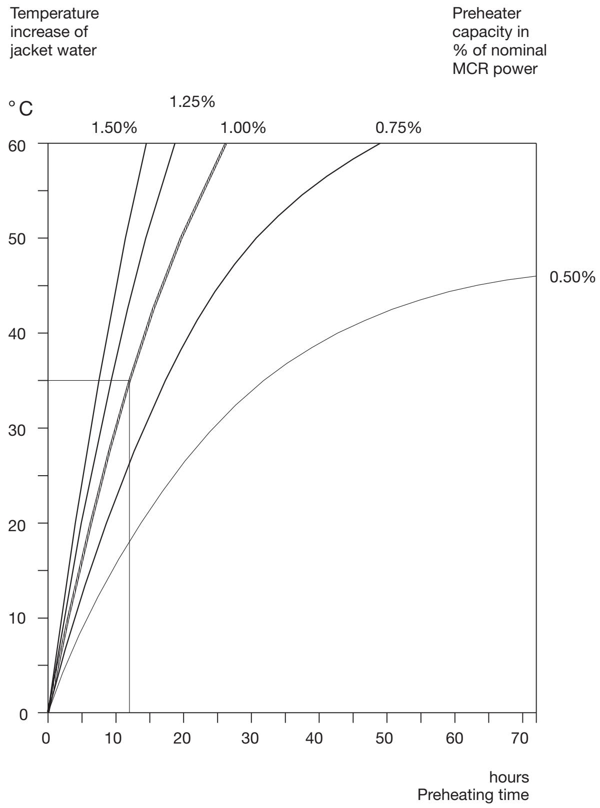

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Fig. 12.08.01: Jacket water preheater

Preheating of diesel engine

Preheating during standstill periods

During short stays in port (i.e. less than 4-5 days), it is recommended that the engine is kept preheated, the purpose being to prevent temperature variation in the engine structure and corresponding variation in thermal expansions and possible leakages.

The jacket cooling water outlet temperature should be kept as high as possible and should – before starting-up – be increased to at least 50 °C, either by means of cooling water from the auxiliary engines, or by means of a built-in preheater in the jacket cooling water system, or a combination.