Marine Installation Manual

X62

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List of Changes

The following table reflects the changes and updates to the contents of this document.
Minor changes in layout or language are not taken into consideration.

Revision: 01	Date of issue: 2018-11
Location of change	Subject
1.2 Primary engine data	Table 1-1: guide feed rate of cylinder oil stated more precisely
2.1 Pressure and temperature ranges	Paragraph rewritten and link name updated
2.2 Engine rating field and power range	Whole section updated and restructured
3.10 Fire protection	Table 3-4: table head rewritten for clarification
4.1 Twin-engine installation	Section restructured; more detailed information added
4.2.1 Central freshwater cooling system components High-temperature circuit	Table 4-3: design pressure for automatic temp. control valve changed from 10 bar to 5 bar
4.2.2 Cooling water treatment	Link to external document: document name and link name updated
4.3.1 Lubricating oil requirements	Link to external document: document name and link name updated
4.3.5 Cylinder lubricating oil system Alternatives to finished cylinder oils	Link to external document: document name and link name updated
4.4.6 Fuel oil specification	Paragraph updated
5.4.2 Recommended manoeuvring characteristics FPP manoeuvring steps and warm-up times CPP manoeuvring steps and warm-up times	Paragraph reworded for clarification Table 5-2: table caption changed Paragraph reworded for clarification Table 5-3: table caption changed
5.6.2 Signal processing	Link to external document changed
6.7 Countermeasures for dynamic effects	New link to external document replaces former tables 6-1 & 6-2
7.1.2 Selective catalytic reduction High-pressure SCR	Design group no. corrected
9 Appendix 9.2 List of acronyms	Restructured Updated

2018-11

Revision: --	Date of issue: 2018-06
Location of change	Subject
Entire document	The present Marine Installation Manual (MIM) is published in a completely new version with a new layout. It supersedes former MIM version ‘a7’ dated 18 April 2017. All future changes and updates (revisions) will be tracked and described based on the present Manual.

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0 Preface

Introduction

The present Marine Installation Manual (MIM) is for use by project and design personnel. Each chapter contains detailed information for design engineers and naval architects, enabling them to optimise plant items and machinery space, and to carry out installation design work.

The manual is not to be considered as a specification. The build specification is subject to the laws of the legislative body of the country of registration and the rules of the classification society selected by the owners.
Furthermore, system components are not the responsibility of WinGD. Guidelines for installation and operation from the makers' side must be observed. Additionally, the engine requirements and any third-party maker requirements must be fulfilled.

The content of this document is subject to the understanding that we have prepared the data and information herein with care and to the best of our knowledge.
However, these data and information are subject to revision without notice. We do not assume any liability with regard to unforeseen variations in accuracy thereof or for any consequences arising therefrom.

The MIM is only designed for persons dealing with this engine.

Attention is drawn to the following:

— All data are related to engines compliant with the regulations of:
- Revised MARPOL Annex VI
- NO_x Technical code 2008

— Engine performance data (rating R1+) refer to General Technical Data (GTD).

— You can obtain the engine performance data (BSEC, BSEF and tEaT) and other data from the GTD application, which can be downloaded from the WinGD Customer Portal or from the corporate webpage.

Tier II certified

The engine is Tier II certified and operates with heavy fuel oil (HFO) that has a viscosity of up to 700 cSt, or with distillate fuels (MDO or MGO) in accordance with the ISO 8217:2017 specification.

Marine Installation Drawing Set

The Marine Installation Drawing Set (MIDS) is part of the documentation for licensees, shipyards and operators.
It includes drawings and guidelines for engine installation and operation, providing:
- engine-ship interface specifications
- general installation / system proposals

Engine design groups

The MIDS covers design groups (DG) **97xx**:

9707	Engine Alignment Record Sheets
9709	Engine Alignment
9710	Engine Seating / Foundation
9710-01	Tool Engine Alignment
9715	Engine Stays
9721	Cooling Water Systems
9722	Lubricating Oil Systems
9723	Fuel Oil System
9724	Leakage Collection
9725	Starting and Control Air System
9726	Exhaust / Ventilation System
9730	Various Installation Items

The drawings which are part of the MIDS have to be delivered to the shipyard by the engine builder (licensee).

Links to complete drawing packages

The latest versions of drawing packages relevant for the present MIM are provided on the WinGD corporate webpage under the following links:

- Marine installation drawings:
  MIDS - Complete package

- Shipyard installation instructions and system concept guidance:
  Concept guidance and instructions - Complete package

Explanation of symbols used in this manual

Cross references

Cross references are written in blue. They lead to another section or a table or figure in this manual and can be activated by mouseclick.
They consist of the number of the respective figure or table, or the section title, followed by the page symbol introducing the page number.
Example: Table 4-5, 4-26

Notes

They give additional information considered important, or they draw your attention to special facts.
Example:

NOTE	The illustration does not necessarily represent the actual configuration or the stage of development of your engine.

Weblinks

Weblinks are written in blue italics. They are preceded by the following symbols and refer to:

— Drawings of the Marine Installation Drawing Set **MIDS**, which is provided on the WinGD corporate webpage.
Example: *MIDS*

— Documents like **concept guidance**, **instructions**, which are provided on the WinGD corporate webpage.
Example: *Fuel oil treatment*

— General Technical Data **GTD**. This is an application provided on the WinGD corporate webpage.
Link: *GTD*

1 Engine Description

The WinGD X62 engine is a camshaftless low-speed, reversible and rigidly direct-coupled two-stroke engine featuring common-rail injection.

Bore:	620 mm
Stroke:	2,658 mm
Number of cylinders:	5 to 8
Power (MCR):	2,660 kW/cyl
Speed (MCR):	77-103 rpm
Mean effective pressure (R1):	20.5 / 19.3 bar
Stroke/bore ratio:	4.29

This engine type is designed for running on a wide range of fuels, from marine diesel oil (MDO) to heavy fuel oils (HFO) of different qualities.

UNIC Engine Control System

Electronic control of the key engine functions such as exhaust valve drives, engine starting and cylinder lubrication is effected by the UNIC Engine Control System. UNIC also ensures volumetric control of the fuel injection.

1.1 Power/speed range

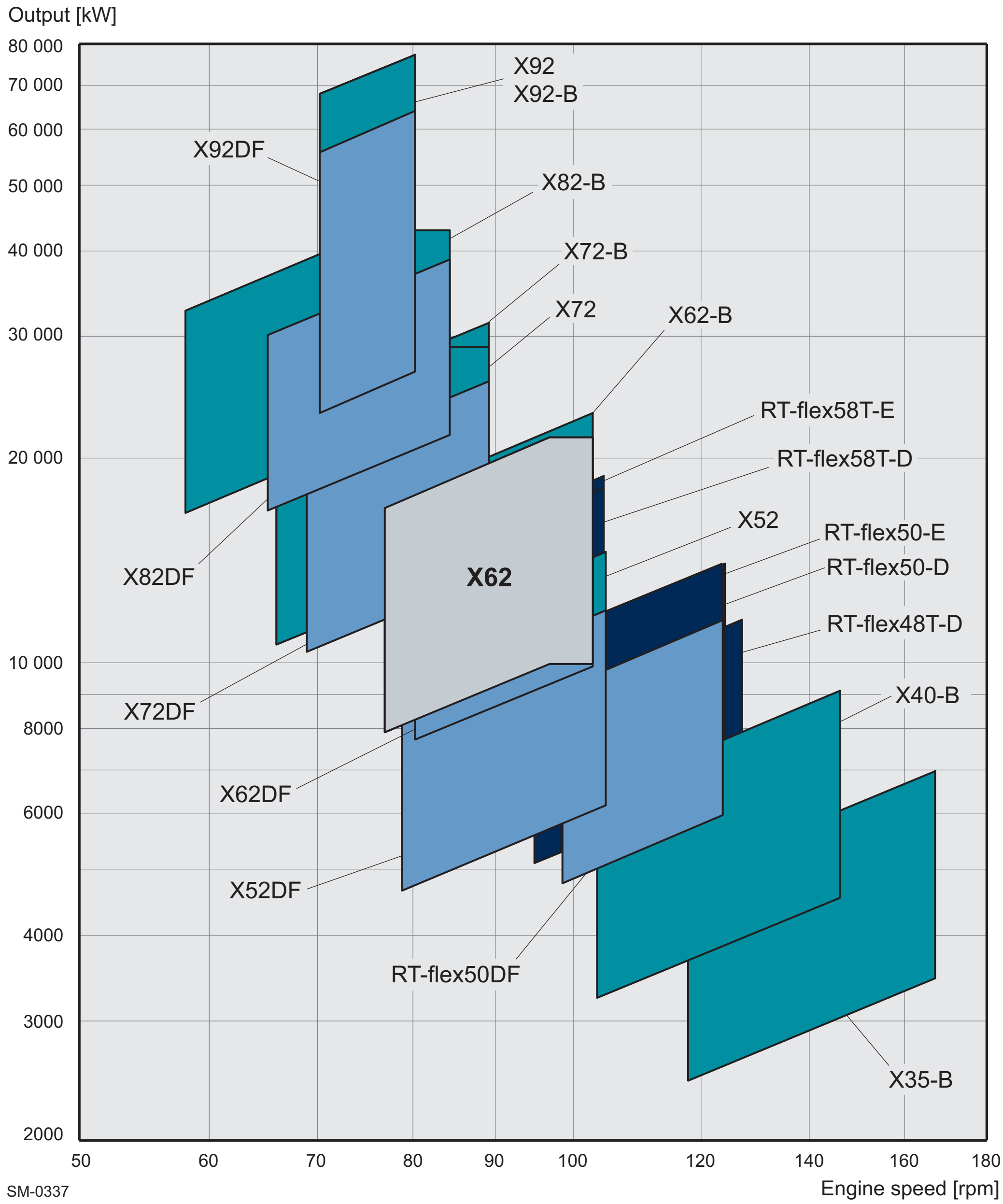


Figure 1-1 Power/speed range of WinGD engines complying with IMO regulations

1.2 Primary engine data

Table 1-1 Rating points

Bore x stroke: 620 x 2,658 [mm]				
No. of cyl.	R1 / R1+	R2 / R2+	R3	R4
	Power [kW]			
5	13,300	10,000	10,550	7,950
6	15,960	12,000	12,660	9,540
7	18,620	14,000	14,770	11,130
8	21,280	16,000	16,880	12,720
Speed [rpm]				
All cyl.	97 / 103	97 / 103	77	77
Brake specific diesel fuel consumption (BSFC) [g/kWh] 100% power				
All cyl.	167.0 / 166.0	160.0 / 160.0	167.0	160.0
Mean effective pressure (MEP) [bar]				
All cyl.	20.5 / 19.3	15.4 / 14.5	20.5	15.4
Lubricating oil consumption (for fully run-in engines under normal operating conditions)				
System oil	approx. 6 kg/cyl per day			
Cylinder oil	guide feed rate 0.6 g/kWh (for low sulphur content only)			
BSFC data are quoted for fuel of lower calorific value 42.7 MJ/kg. All other reference conditions refer to ISO standard (ISO 3046-1). For BSFC the following tolerances are to be taken into account: +5% for 100-85% engine power +6% for 84-65% engine power +7% for 64-50% engine power				

1.3 Components and sizes of the engine

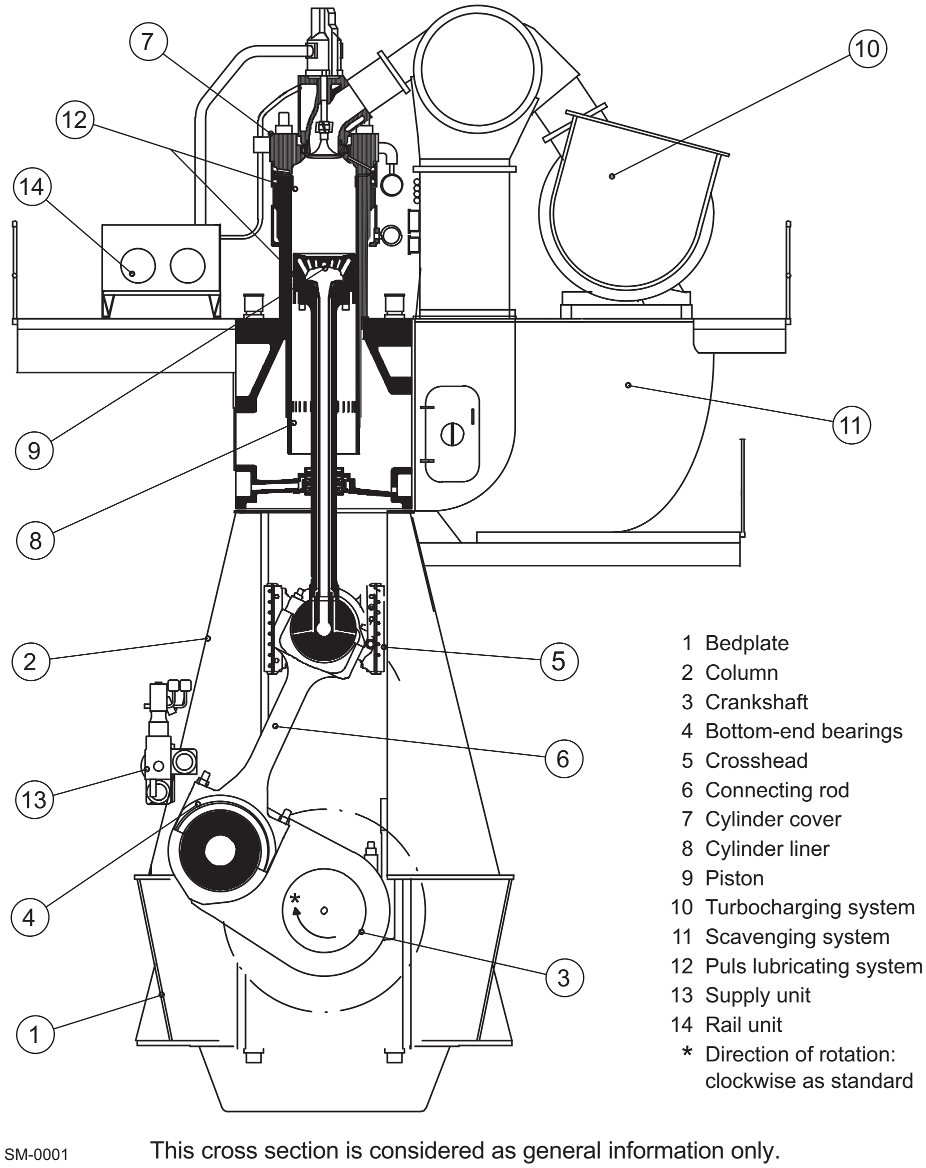


Figure 1-2 Cross section

Table 1-2 Overall sizes and masses

No. of cyl.	Length [mm]	Piston dismantling height F1 [a] (crank centre - crane hook) [mm]	Dry weight [t]
5	7,000	11,670	325
6	8,110		377
7	9,215		435
8	10,320		482

a) For F2 and F3 (piston removal with double-jib crane) see Table 3-1, 3-1.

Design features

- Welded bedplate with integrated thrust bearing and main bearings designed as thin-shell white metal bearings
- Sturdy engine structure with stiff thin-wall box type columns and cast iron cylinder blocks attached to the bedplate by pre-tensioned vertical tie rods
- Semi-built crankshaft
- Thin-shell aluminium bottom-end bearings
- Crosshead with crosshead pin and single-piece large white-metal surface bearings
- Rigid cast iron cylinder monoblock
- Special grey-cast iron cylinder liners, water cooled
- Pulse Jet Lubricating System for high-efficiency cylinder lubrication
- Cylinder cover of high-grade material with a bolted exhaust valve cage containing a Nimonic 80A exhaust valve
- Piston with crown, cooled by combined jetshaker oil cooling
- Constant-pressure turbocharging system comprising high-efficiency turbochargers and auxiliary blowers for low-load operation
- Latest piston running concept for excellent piston running and extended TBO up to 5 years
- Supply unit: high-efficiency fuel pumps feeding the 1,000 bar fuel rail
- Rail unit (common rail): common rail injection and exhaust valve actuation controlled by quick-acting solenoid valves

1.4 Tuning options

As the Flex system (see section 1.5, 1-14) allows free selection of injection and exhaust valve control parameters — specifically variable injection timing (VIT) and variable exhaust closing (VEC) — it can be used in special tuning regimes to optimise the brake specific fuel consumption (BSFC) at individual engine loads. The reduction of BSFC is achieved by changing software parameters, without modifying any engine parts.

Compliance with IMO Tier II

All tuning regimes comply with the IMO Tier II regulations for NO_X emissions. Data for the tuning regimes are available from the *GTD* program.

The following table gives an overview of the available engine tuning methods with their application and the required engine components.

Table 1-3 Available tuning regimes

Tuning	Description	Application	Additional components
Standard tuning (Std)	High-load tuning	When ship operates most of the time above 90 % engine load	None
Delta tuning (Delta)	Part-load tuning	When ship operates most of the time between 75 and 90 % engine load	None
Delta bypass tuning (DBT)	Part-load tuning with increased steam power production	For increased steam production between 50 and 100 % engine power Allows reducing economiser size and minimising use of auxiliary boiler	Exhaust gas waste gate
Low load tuning (LLT)	Lowest possible BSFC in the operating range of 40-70 % engine load	When ship operates most of the time at less than 75 % engine load	Exhaust gas waste gate & Turbo LLT kit

Data for these tuning regimes as well as de-rating and part-load performance data are obtainable from the *GTD* application.

The following figure shows the BSFC curves for the available tuning options.

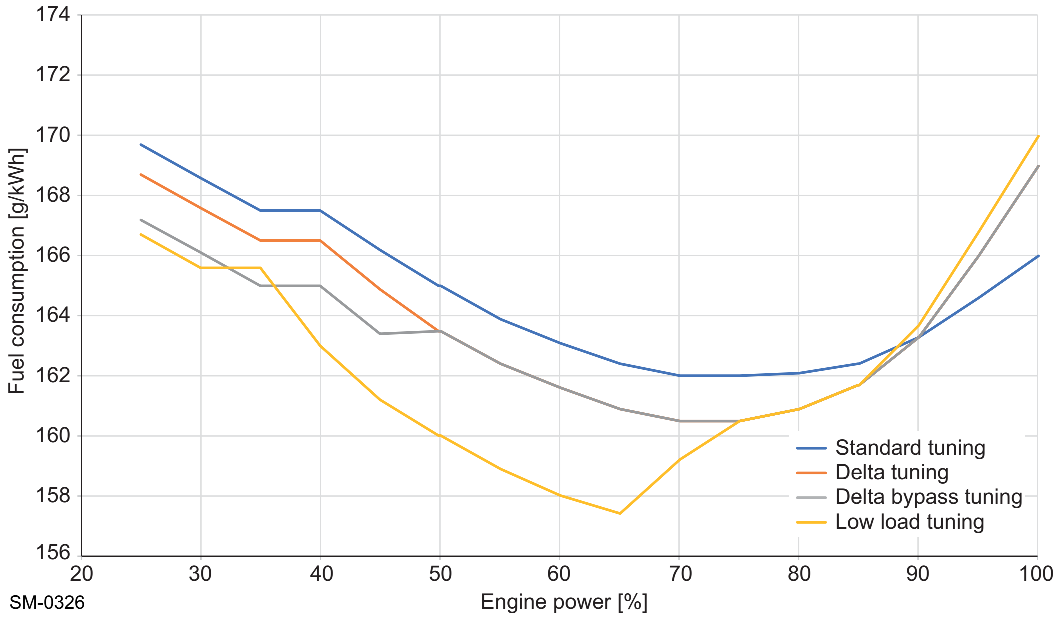


Figure 1-3 Typical BSFC curves in relation to engine load

BSFC data for Standard tuning is given in Table 1-1, 1-3.
BSFC data for the other tuning options can be obtained from the *GTD* application.

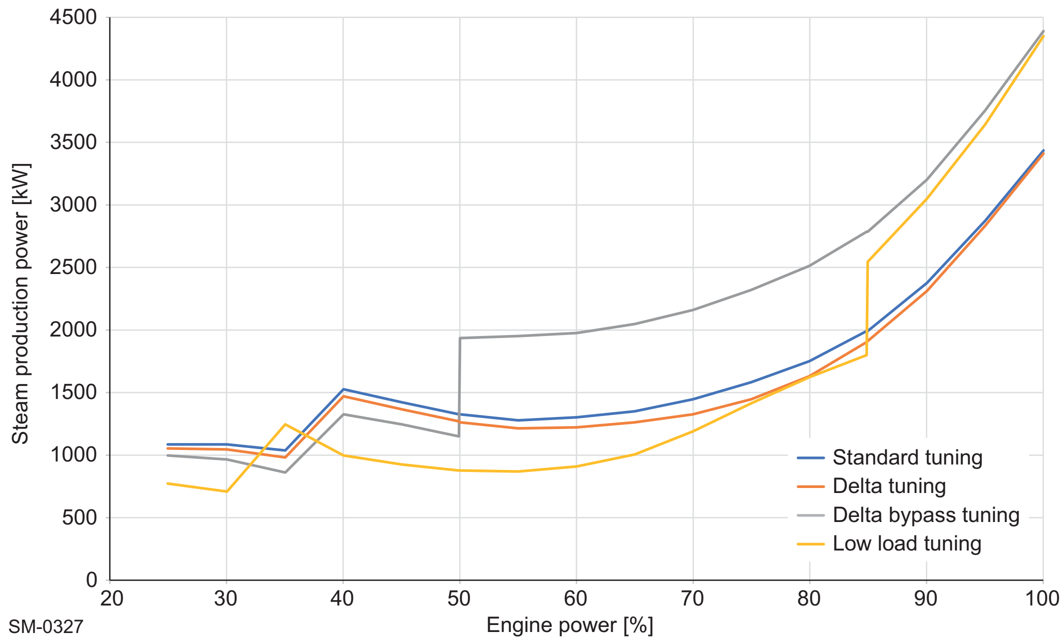


Figure 1-4 Steam production power diagram

1.4.1 BSFC and NO_X emission

The engine parameters controlling the fuel injection and exhaust valve operational characteristics have to be selected appropriately to allow the full potential of the respective tuning, while ensuring compliance with the applicable NO_X limit value.
Due to the trade-off between BSFC and NO_X emissions, the associated increase in NO_X emissions at part load must be compensated by a corresponding decrease at full load. In this process, the same design related limitations with respect to these two quantities are applied as in Standard tuning. However, there is also a slight increase in full-load BSFC to maintain compliance of the engine with the IMO regulations.

1.4.2 Impact from engine dynamics

The Flex system allows application of the low torsional vibration tuning option (LowTV tuning) on 5-, 6- and 7-cylinder engines. The reduction of the tangential gas excitations in the main torsional criticals of orders 5/6/7 by a software adaptation avoids in many cases the use of a costly torsional vibration damper (see 6.4 Torsional vibration, 6-7). This option can be combined with all other available tuning possibilities.

LowTV tuning shall be considered during the torsional vibration calculation. WinGD offers assistance in checking whether LowTV tuning can be applied in specific projects.

The following figure shows a comparison in regard to torsional vibration between Standard tuning and LowTV tuning during sea trials of a WinGD engine. LowTV tuning reduced the measured torsional vibration amplitudes by nearly 30%, hence the use of a torsional vibration damper could be avoided.

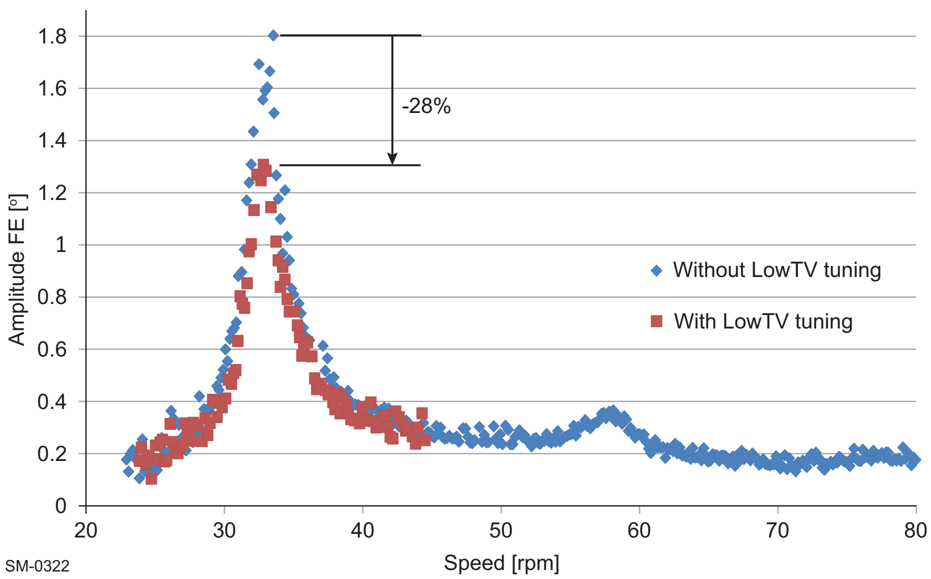


Figure 1-5 Vibration amplitudes - Comparison between general & LowTV tuning

1.4.3 Project specification

Delta, DBT and LLT need to be specified at a very early stage of the project, as they also have an impact on the following aspects of engine and system design:

- The layout of the ancillary systems is to be based on the correct specifications.
- To prepare the software for Flex system control, the parameters are to be known in due time before commissioning of the engine.

NOTE	The reliability of the engine is by no means impaired by applying the tuning options, since all existing limitations to mechanical stresses and thermal load are observed.

1.4.4 Delta tuning

This tuning regime needs to be specified at a very early stage of the project. It is used to reduce the BSFC in the part-load range of less than 90%.

The concept is based on tailoring the firing pressure and firing ratio for maximum efficiency in the range up to 90% load and then reducing them again towards full load. In this process, the same design related limitations with respect to these two quantities are applied as in the specification of Standard tuning.

1.4.5 Delta bypass tuning

DBT is an engine tuning method designed to increase the exhaust gas temperature and steam production power (SPP), while reducing the use of auxiliary boilers. This is achieved at a load of more than 50%, without any penalty to performance or the BSFC, while still complying with all existing emission legislations.

In particular, DBT is achieved by:
- optimising the ECS software parameters (different from Delta tuning)
- increasing the firing pressure and changing injection timing (compared to Delta tuning)
- implementing a specifically designed turbocharging system setup
- adding one (1) exhaust gas waste gate

Functionality of exhaust gas waste gate

DBT requires the fitting of an exhaust gas waste gate (a pneumatically operated valve, see Figure 1-6) on the exhaust gas receiver before the turbocharger turbine. Exhaust gas blown off through the waste gate is bypassed to the main exhaust uptake.

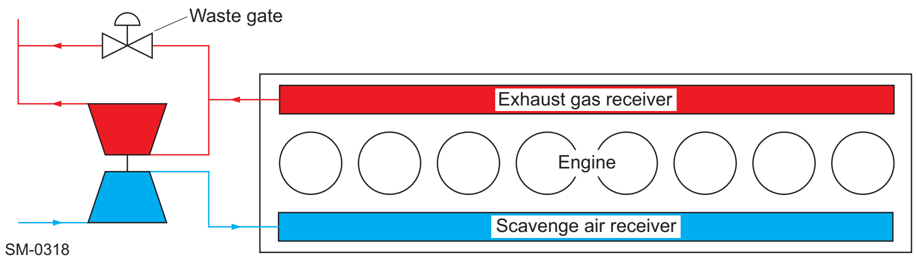


Figure 1-6 Schematic functional principle of an exhaust gas waste gate

The exhaust gas waste gate works in the following two positions:
- Waste gate closed at less than 49% engine load —> increased combustion pressure due to increased scavenge air pressure and higher air flow; as a consequence, the BSFC at low load is reduced.
- Waste gate opened at more than 50% engine load —> higher exhaust gas temperature, as part of the exhaust gas flow is blown off before the turbocharger; this allows increasing the steam production by means of an economiser.

NOTE	Since the exhaust gas waste gate is controlled by the scavenge air pressure, the indicated load is an approximation only.

Exhaust gas temperature

The exhaust gas temperature with DBT is significantly higher than with Delta; see Figure 1-7.

tEaT and tEbE

In particular the tEaT (temperature exhaust gas after turbocharger) is about 25 °C higher at 70 % engine load than with Delta tuning.
The tEbE (temperature exhaust gas before economiser) will increase further (about 5 °C) due to the mixing of the bypassed exhaust gas.

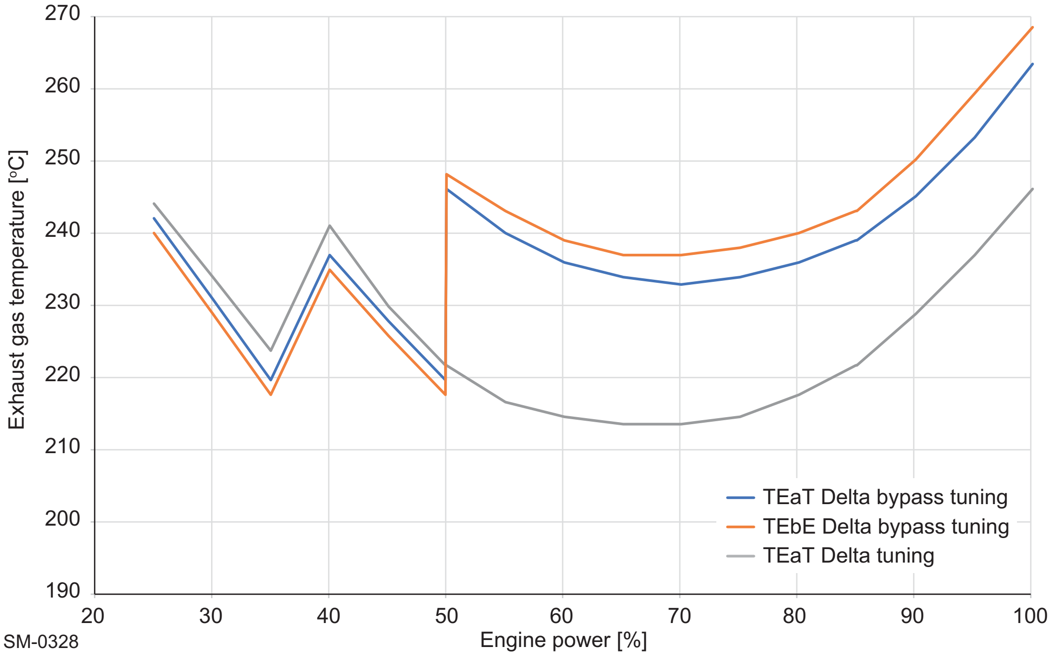


Figure 1-7 Exhaust gas temperature increase with DBT

Steam production

In such condition DBT is the most economical tuning option; see Figure 1-4, 1-7

Steam production by increased exhaust gas temperature and exhaust gas economiser is an efficient way to recover waste heat from main engine exhaust gas. Within certain engine power ranges this can be achieved without running any auxiliary boiler. Such a solution is commonly used on board handysize / max bulk carriers.

For the calculation of steam production through economiser the tEbE and the relevant mass flow shall be considered in the output of *GTD*.

1.4.6 Low load tuning

With LLT, WinGD engines with Flex system can be operated continuously and reliably at less than 75% engine load. Operating at less than 60% CMCR for extended periods requires ongoing attention to ensure satisfactory operation.

Besides the appropriately adjusted engine parameters related to fuel injection and exhaust valve control, the LLT concept combines a specifically designed turbocharging system setup with the use of an exhaust gas waste gate (see Functionality of exhaust gas waste gate, 1-10).

The reduced part-load BSFC is achieved in LLT by optimising the turbocharger match for part-load operation. This is done by increasing the combustion pressure at less than 75% load through an increased scavenge air pressure and higher air flow (waste gate closed), and by blowing off part of the exhaust gas flow at engine loads above 85% (waste gate open).

The higher scavenge air pressure in part load results in lower thermal load and better combustion over the entire part-load range.

1.4.7 Tuning for de-rated engines

The tuning options are applicable over the entire rating field as illustrated in Figure 1-8.

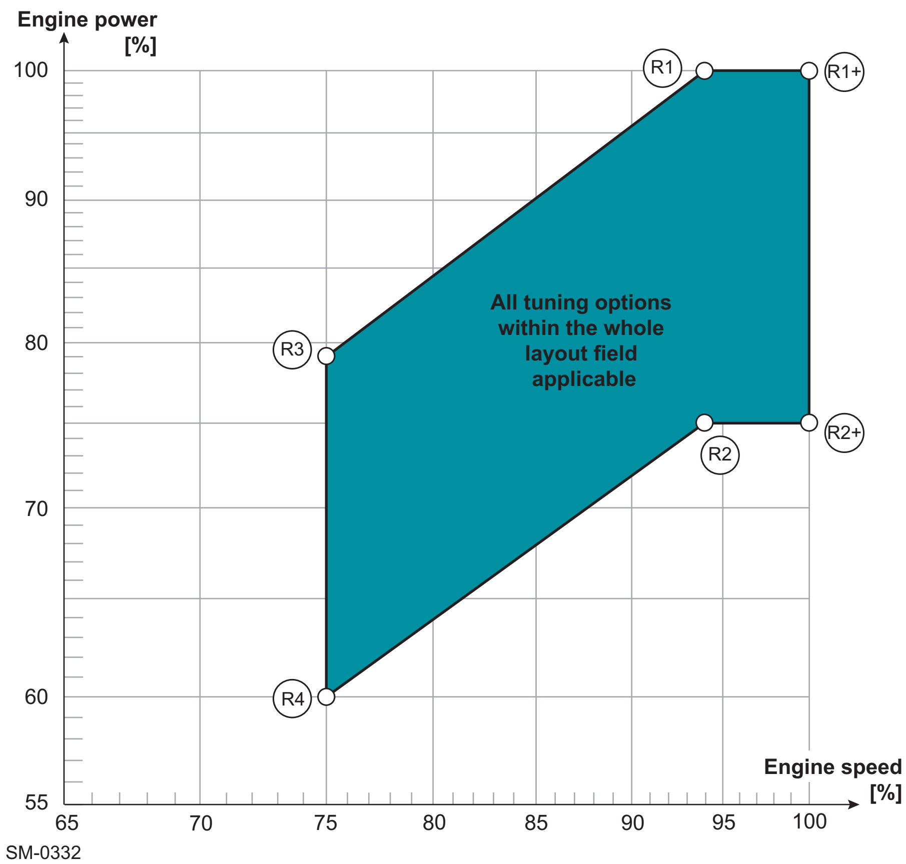


Figure 1-8 Application area for tuning options

1.4.8 Dual tuning

The WinGD 2-stroke engines can be built and certified with ‘dual tuning’, i.e. Delta and LLT or DBT and LLT.
Each tuning method has its own advantages in terms of specific fuel consumption or exhaust gas flow and temperatures.

Changeover between tuning regimes

Changing over from one tuning to the other when the engine is in service is a long-term consideration, since the following modifications are to be carried out on the engine:

- Exchange of turbocharger nozzle ring (and diffuser)
- ECS software parameter change
- Installation/removal of blind flange for exhaust gas bypass (not needed for DBT and LLT)
- Change of orifice size in exhaust gas bypass

An engine cannot be operated with both tuning regimes at the same time, as switching from one tuning to the other when the engine is in operation is not in accordance with the IMO MARPOL Annex VI NO_X regulation. Since for NO_X certification the Technical Files and EIAPP certificates will be approved separately for each tuning, the NO_X emissions need to be measured on the testbed for both tuning regimes.

Considerations to be made when choosing dual tuning

The following must be considered before ordering an engine with dual tuning:

- GTD ancillary system data must be selected for the tuning with higher requirements concerning pump and cooler capacity.
- The torsional vibration calculation (TVC) must be carried out for both tunings. However, only the calculation for the tuning showing worse torsional stresses in the shafting shall be submitted for Class approval.
- The engine interface drawings must correspond to the tuning method with exhaust gas bypass (LLT or DBT)
- The sea trial program (engine related tests) must be discussed with the shipyard. It should be defined beforehand with which tuning the speed trial of the vessel is to be performed.

1.5 The Flex system

The X62 engine is equipped with WinGD's common rail fuel injection technology, allowing flexible fuel injection. The flexibility provided by this technology is reflected in the naming *Flex system*.

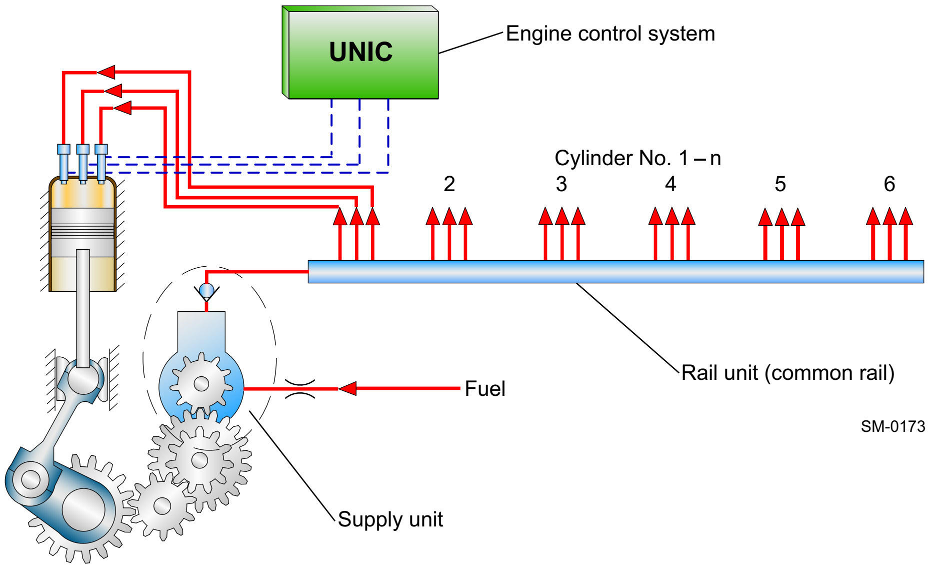


Figure 1-9 Flex system parts

Major benefits

- Adaptation to different operating modes
- Adaptation to different fuels
- Optimised part-load operation
- Optimised fuel consumption
- Precise speed regulation, in particular at very slow steaming
- Smokeless operation at all engine loads
- Benefits in terms of operating costs, maintenance requirement and compliance with emissions regulations

2 General Engine Data

Selecting a suitable main engine to meet the power demands of a given project involves proper tuning in respect of load range and influence of operating conditions which are likely to prevail throughout the entire life of the ship.
This chapter explains the main principles in selecting a WinGD 2-stroke marine diesel engine.

Data for 5-8X62

The data provided in this manual are applicable to the **nominal maximum continuous rating R1+**.

These data refer to the following engine conditions / features:
- Design (tropical) conditions
- Delta tuning
- Central freshwater cooling system with single-stage scavenge air cooler and separate HT circuit
- ABB A100-L / A200-L turbochargers
- Turbochargers lubricated from the engine's lubricating system

This chapter also outlines the pressure and temperature ranges and electrical power requirement for the engine. Besides the design parameters for the ancillary systems you will also find information about the General Technical Data (GTD) program.

2.1 Pressure and temperature ranges

Please refer to the list **'Usual values and safeguard function setting'**, which is provided on the WinGD corporate webpage under the following link:
Usual values & safeguard function setting

For signal processing see also 5.6.2 Signal processing, 🗎 5-11.

2.2 Engine rating field and power range

2.2.1 Introduction

It is critical that a ship's propulsion system is correctly matching the main engine characteristics to ensure reliable operation in a variety of conditions including design and off design situations. The below sections outline the specifics to aid in this process.

2.2.2 Engine rating field

The rating field shown in Figure 2-1 is the area of selectable engine design power and engine design speed. In this area, the contracted maximum continuous rating (CMCR) of an engine can be positioned individually to give the desired combination of propulsive power and rotational speed. Engines within this layout field are tuned for maximum firing pressure and best efficiency.

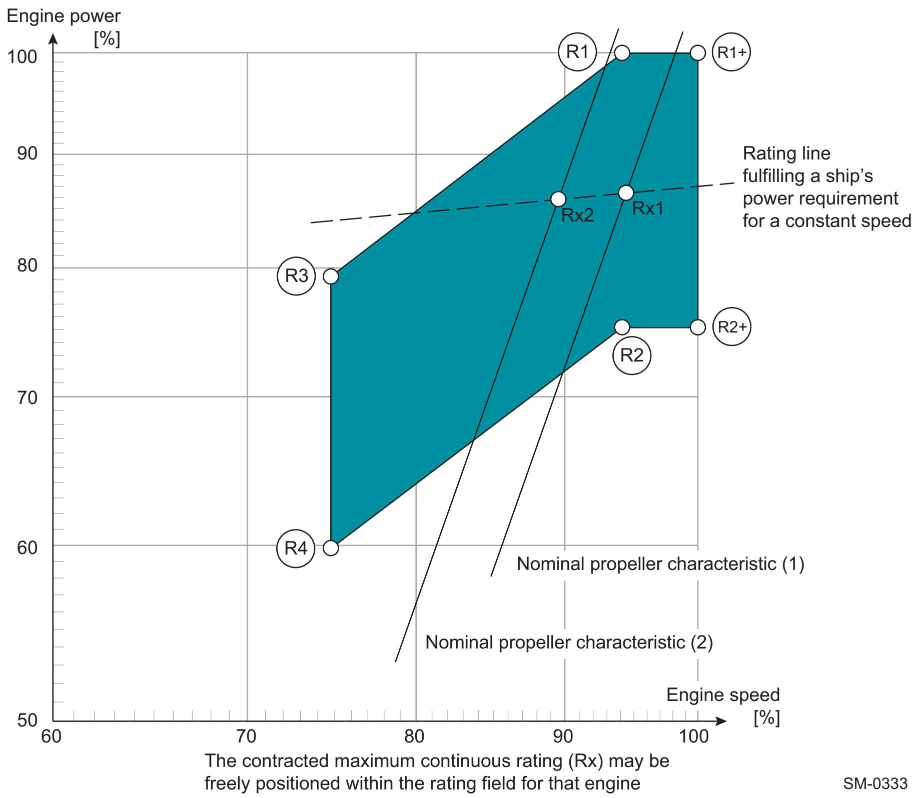


Figure 2-1 Engine rating field for X62

The rating field serves to determine the specific fuel consumption, exhaust gas flow and temperature, fuel injection parameters, turbocharger and scavenge air cooler specifications at the selected rating.

Percentage values

The engine speed is given on the horizontal axis and the engine power on the vertical axis of the rating field. Both are expressed as a percentage [%] of the respective engine's nominal R1+ parameters. Percentage values are being used so that the same diagram can be applied to various engine arrangements.

Rating points

The rating points (R1, R1+, R2, R2+, R3 and R4) for WinGD engines are the corner points of the engine rating field (Figure 2-1, 2-3). The rating field is limited by two constant MEP (mean effective pressure) lines R1 — R3 and R2 — R4 and by two constant engine speed lines R1+ — R2+ and R3 — R4.

The point R1 represents the nominal maximum continuous rating (MCR).

Any rating point (Rx) can be selected within the entire rating field to meet the requirements of each particular project. Such rating points require specific engine adaptations.

2.2.3 Propeller diameter and influence of propeller revolutions

Influence of propeller revolutions on the power requirement

At constant ship speed and for a given propeller type, a lower propeller speed combined with a larger propeller diameter increases the total propulsive efficiency. Less power is needed to propel the vessel at a given speed.

The relative change of required power in function of the propeller revolutions can be approximated by the following relation:

$$\frac{PX_2}{PX_1} = \left(\frac{n_2}{n_1}\right)^{\alpha}$$

Formula 2-1

where:

PX_j .... = propulsive power at propeller revolution n_j
n_j .... = propeller speed corresponding with propulsive power PX_j
α .... = 0.15 for tankers and general cargo ships up to 10,000 dwt, or
= 0.20 for tankers and bulk carriers from 10,000 to 30,000 dwt, or
= 0.25 for tankers and bulk carriers larger than 30,000 dwt, or
= 0.17 for reefers and container ships up to 3,000 TEU, or
= 0.22 for container ships larger than 3,000 TEU

This relation is used in the engine selection procedure to compare different engine alternatives and to select an optimum propeller speed within the selected engine rating field. Usually, the number of revolutions depends on the maximum permissible propeller diameter.

Maximum propeller diameter

The maximum propeller diameter is often determined by operational requirements, such as:

- Design draught and ballast draught limitations
- Class recommendations concerning propeller/hull clearance (pressure impulse induced on the hull by the propeller)

The selection of a main engine in combination with the optimum propeller (efficiency) is an iterative procedure where also commercial considerations (engine and propeller prices) are playing an important role.

According to the above approximation, when a required power/speed combination is known — for example point Rx1 in Figure 2-1, 2-3 — a CMCR line can be drawn which fulfils the ship's power requirement for a constant speed. The slope of this line depends on the ship's characteristics (coefficient α). Any other point on this line represents a new power/speed combination, for example Rx2, and requires a specific propeller adaptation.

2.2.4 Power range

Propeller curves and operational points

To establish the proper propeller curves, it is necessary to know the ship's speed to power response.

Determining power/propeller speed relationships

Normally, the curves can be determined by using full-scale trial results from similar ships, algorithms developed by maritime research institutes, or model tank results. With this information and by applying propeller series, the power/speed relationships can be established and characteristics developed.

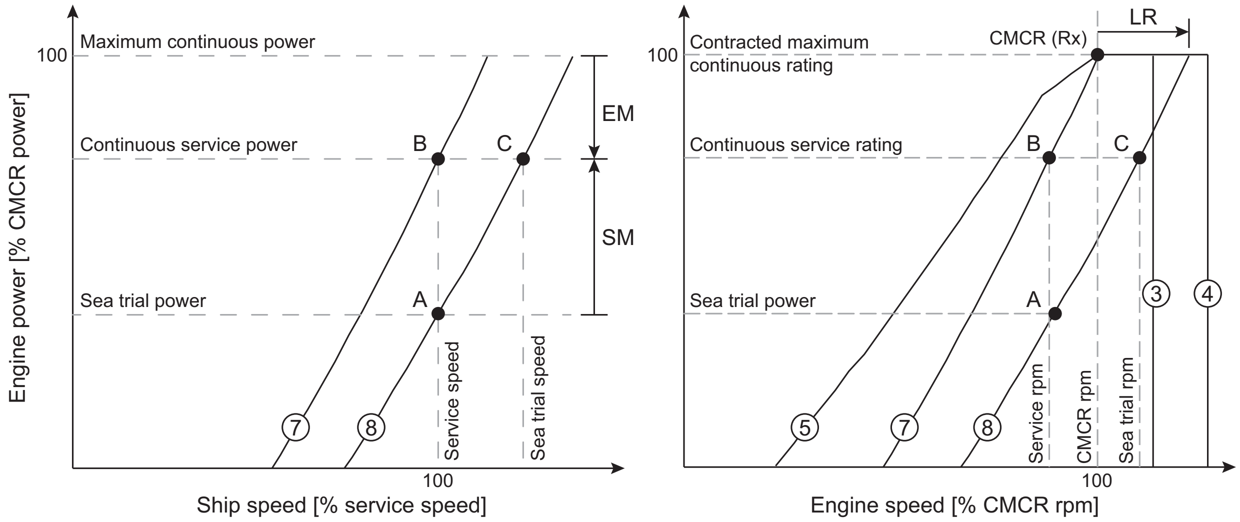
The relation between absorbed power and propeller speed for a fixed pitch propeller (FPP) can be approximated by the following cubic relation:

$$\frac{P}{P_{CMCR}} = \left(\frac{n}{n_{CMCR}}\right)^3$$

Formula 2-2

where:

P .... = propeller power
n .... = propeller speed



A	Point of contractual power and ship speed during sea-trials
B	Hydrodynamic design point / Continuous service rating (CSR)
C	Recommended point for adaptation of propeller pitch under sea-trial condition
EM	Engine margin
LR	Light running margin
SM	Sea margin
Line 3	Maximum engine speed limit for continuous operation
Line 4	Maximum engine overspeed limit during sea-trials
Line 5	Admissible torque limit
Line 7	Nominal propeller characteristic curve
Line 8	Propeller curve with a light running margin

SM-0026

Figure 2-2 Propeller curves and operational points

Figure 2-2 outlines the various engine limits, propeller curves and margins required for engine optimisation. By incorporating the margins listed below, the various operational points and subsequently the CMCR point can be determined. For detailed descriptions of the various line limits refer to section 2.2.5, 2-8.

Sea trial power

The sea trial power must be specified. Figure 2-2 shows the sea trial power to be the power required for reaching service speed, marked as point A, on the propeller curve with a light running margin (Line 8).

Sea margin

The increase in power to maintain a given ship's speed achieved in calm weather (point A in Figure 2-2) under average service condition (point B) is defined as 'sea margin' (SM). This margin can vary depending on owner's and charterer's expectations, routes, season and schedules of the ship.

The location of reference point A and the magnitude of the sea margin are part of the new building contract and are determined between shipbuilder and owner. Typically, the sea margin is specified in the range of 10 to 25% of the sea trial power.

Light running margin

The light running margin (LR in Figure 2-2, 🗎 2-6) is the margin in propeller revolutions with a new ship (i.e. under sea-trial condition) to attain or maintain any power up to 100% in future continuous service. An additional power/engine speed allowance must be provided for shaft generator/PTO installations (see section 2.2.6, 🗎 2-11).

The magnitude of the margin is generally determined by the engine builder and/ or the shipbuilder and varies with specific ship designs, speeds, dry-docking intervals and trade routes. Typically, the light running margin is specified in the range of 4 to 7%.

NOTE	It is the shipbuilder's responsibility to determine a light running margin large enough so that the power range limits on the left side of the nominal propeller characteristic (Line 7) are not reached in any service condition (see Figure 2-3, 🗎 2-8).

Continuous service rating (CSR = NOR = NCR)

Point A represents the power and propeller speed of a ship operating at contractual service speed in calm seas with a new clean hull and propeller. On the other hand, the same ship at same speed under service condition with aged hull and under average weather conditions requires a power/speed combination according to point B. In that case, B is the CSR point.

Engine margin (EM) / operational margin

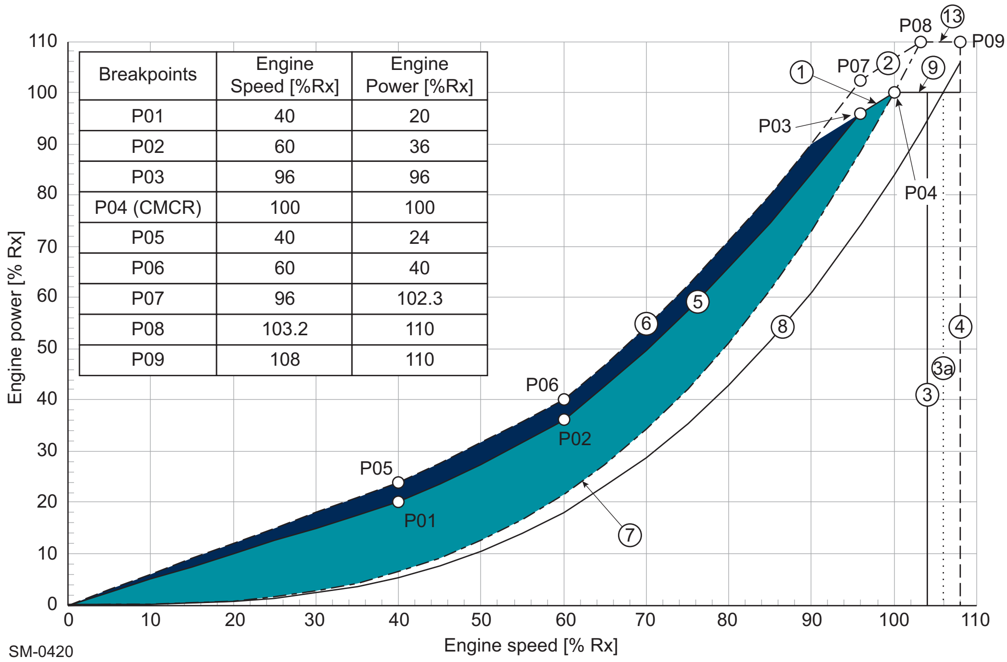
Most owners specify the contractual ship's loaded service speed at 85 to 90% power of the contracted maximum continuous rating. Different selections are possible. The remaining e.g. 10 to 15% power can then be used to catch up with delays in schedule.
This margin is deducted from the CMCR. Therefore, the 100% power line is found by dividing the power at point B by the selected CSR power percentage, e.g. 85 to 90%. The graphic approach to find the level of CMCR is illustrated in Figure 2-2, 🗎 2-6.

Contracted maximum continuous rating (CMCR = Rx = SMCR)

The contracted maximum continuous rating is the point obtained by applying the margins (SM and EM) to the propeller curves. The calculated CMCR point can be selected freely within the entire engine rating field.

2.2.5 Power range limits

Once an engine is optimised at CMCR (Rx), the working range of the engine is limited by the following border lines (refer to Figure 2-3).



Breakpoints	Engine Speed [%Rx]	Engine Power [%Rx]
P01	40	20
P02	60	36
P03	96	96
P04 (CMCR)	100	100
P05	40	24
P06	60	40
P07	96	102.3
P08	103.2	110
P09	108	110

Figure 2-3 Power range limits

Line 1: 100% Torque Limit

Constant mean effective pressure (MEP) or torque line through CMCR from 100% speed and power down to 96% speed and power.

Line 2: Overload Limit

Available for testbed operation and emergency operation according to SOLAS Regulation II-1/3.6. It is a constant MEP line, reaching from 102.3% power and 96% speed (point P07) to 110% power and 103.2% speed (point P08). P08 is the point of intersection between Line 7 and 110% power.

Line 3: Speed Limit

Maximum engine speed limit where an engine can run continuously. It is 104% of CMCR speed. For Rx with reduced speed ($n_{CMCR} \leq 0.98\, n_{MCR}$) this limit can be extended to 106% (Line 3a), while the specified torsional vibration limits must not be exceeded.

Line 4: Overspeed Limit

The overspeed range between 104% (106%) and 108% speed is only permissible during sea trials if needed to demonstrate, in the presence of authorised representatives of the engine builder, the ship's speed at CMCR power with a light running propeller. However, the specified torsional vibration limits must not be exceeded.

Line 5: Continuous Operation Power Limit

Admissible power limit for continuous operation. The line is separated by the breakpoints listed in Figure 2-3, 2-8.
Line 5 is a curve defined by Formula 2-3 and is separated into five components to form the entire curve. Each component is governed by different coefficients. Refer to Table 2-1 for the individual coefficients.

$$\frac{P}{P_{CMCR}} = C2\left(\frac{n}{n_{CMCR}}\right)^2 + C1\left(\frac{n}{n_{CMCR}}\right) + C0$$

Formula 2-3

where:

P .... = selected engine power [kW]
P_{CMCR} .... = CMCR engine power [kW]
n .... = selected engine speed [rpm]
n_{CMCR} .... = CMCR engine speed [rpm]
$C2 / C1 / C0$ .. = coefficients / constants

Table 2-1 Line 5 coefficients

Line no.	Range (n/n_{CMCR})	C2	C1	C0
Line 5	0.00 - 0.40	0.000	0.500	0.000
	0.40 - 0.60	0.500	0.300	0.000
	0.60 - 0.96	1.111	-0.067	0.000
	0.96 - 1.00	0.000	1.000	0.000
	1.00 - 1.08	0.000	0.000	1.000

The area formed by Lines 1, 3, 5 and 9 is the range within which the engine should be operated.
The area limited by Line 7, Line 9 and Line 3 is recommended for continuous operation.
The area between Line 7 and Line 5 is reserved for acceleration, shallow water and normal operational flexibility. If a main-engine driven generator (PTO) is installed, then the operating characteristics of the engine will differ. Refer to section 2.2.6, 2-11 for further details regarding PTO characteristics.

Line 6: Transient Condition Power Limit

Maximum power limit in transient conditions. The line is separated by the breakpoints listed in Figure 2-3, 2-8.
Line 6 is a curve defined by Formula 2-3 and is separated into five components to form the entire curve. Each component is governed by different coefficients. Refer to Table 2-2, 2-10 for the individual coefficients.

Table 2-2 Line 6 coefficients

Line no.	Range (n/n_{CMCR})	C32	C1	C0
Line 6	0.00 - 0.40	0.000	0.600	0.000
	0.40 - 0.60	0.330	0.468	0.000
	0.60 - 0.96	1.110	0.000	0.000
	0.96 - 1.032	0.000	1.066	0.000
	1.032 - 1.08	0.000	0.000	1.100

The area above Line 1 and Line 9 is the overload range. It is only allowed to operate engines in that range for a maximum duration of one hour during sea trials in the presence of authorised representatives of the engine builder.
The area between Line 1, Line 5 and Line 6 (Figure 2-3, 2-8), called ‘service range with operational time limit’, is only applicable to transient conditions, i.e. sea trial or during emergency fast acceleration. The engine can only be operated in this area for limited periods of time, in particular 1 hour per 24 hours.

Line 7: Nominal Propeller Characteristic

Nominal propeller characteristic curve that passes through the CMCR point. The curve is defined by the 100% propeller law:

$$\frac{P}{P_{CMCR}} = \left(\frac{n}{n_{CMCR}}\right)^3$$

Formula 2-4

Line 8: Light Running Propeller Curve

Propeller curve with a light running margin (typically between 4% and 7%). The curve is defined by the propeller law with a constant, governed by the selected light running margin (Formula 2-5).

$$\frac{P_{LR}}{P_{CMCR}} = C \times \left(\frac{n}{n_{CMCR}}\right)^3$$

$$C = \left(\frac{1}{1+LR}\right)^3$$

Formula 2-5

where:

P_{LR} .... = propeller power at selected light running margin [kW]
P_{CMCR} .... = CMCR engine power [kW]
n .... = selected engine speed [rpm]
n_{CMCR} .... = CMCR engine speed [rpm]
C .... = constant
LR .... = light running margin [%]

Line 9: CMCR power

Maximum power for continuous operation.

Line 13: 110% CMCR power

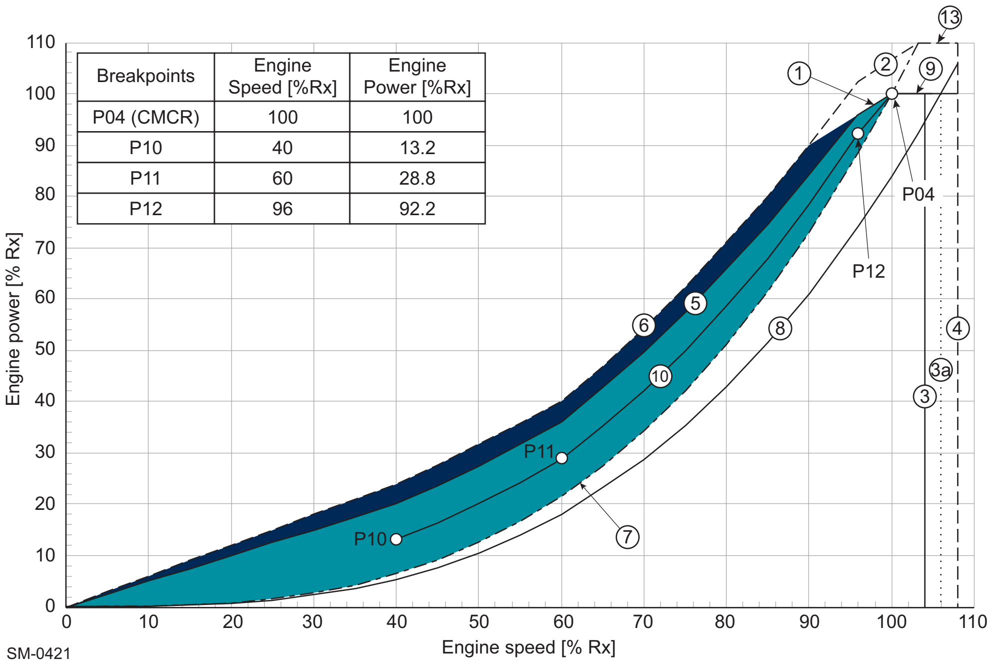
Constant power overload limit, available for testbed operation and emergency operation according to SOLAS Regulation II-1/3.6.

2.2.6 Power range limits with main-engine driven generator

The addition of a main-engine driven generator (PTO) alters the working range and operating characteristics of the engine. To generate the relevant curves, multiple approaches can be used to incorporate the PTO limits. One such approach is outlined in the following.

Line 10: PTO Layout Limit

The PTO layout limit line (Line 10 in Figure 2-5, 2-12) defines the layout limit for the power demanded by the propeller and PTO.
Considering Line 10 as PTO layout limit provides the margin for normal power load fluctuation and acceleration.



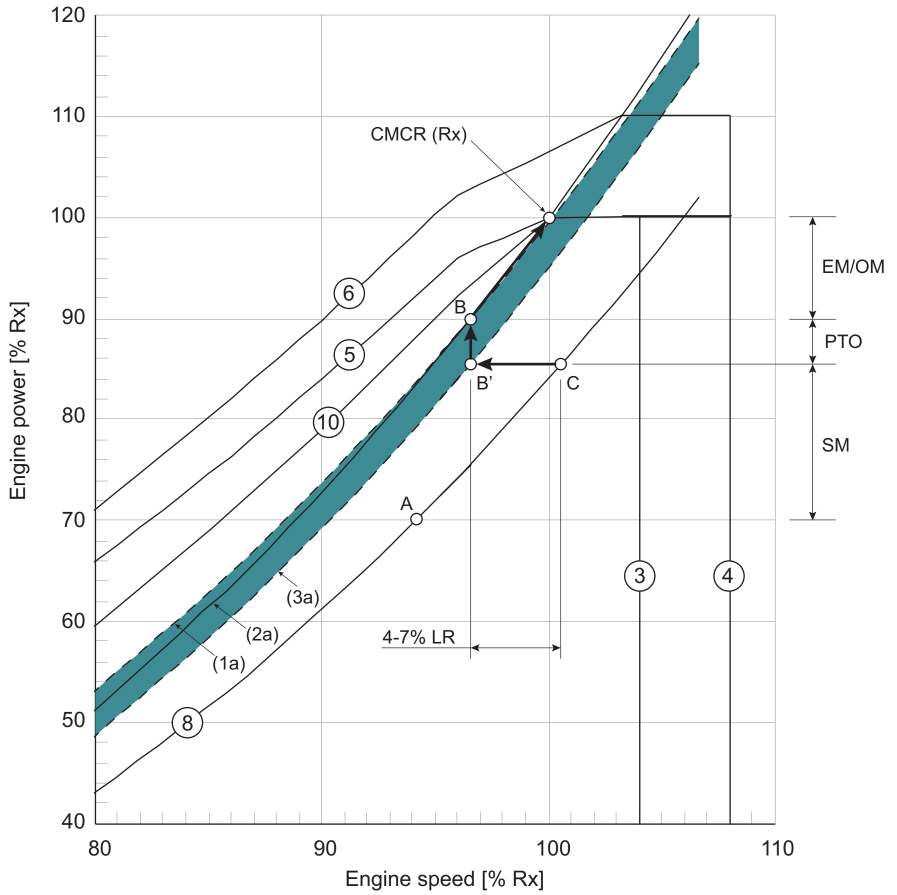
Breakpoints	Engine Speed [%Rx]	Engine Power [%Rx]
P04 (CMCR)	100	100
P10	40	13.2
P11	60	28.8
P12	96	92.2

Figure 2-4 Power range limits for PTO operation

The breakpoints of Line 10 are listed in Figure 2-4. Line 10 is a curve defined by Formula 2-3, 2-9. It is separated into three components to form the entire curve. Each component is governed by different coefficients. Refer to Table 2-3, 2-12 for the individual coefficients.

Table 2-3 Line 10 coefficients

Line no.	Range (n/n_{CMCR})	C2	C1	C0
Line 10	0.40 - 0.60	0.750	0.030	0.000
	0.60 - 0.96	1.336	-0.321	0.000
	0.96 - 1.00	0.000	1.941	-1.941
	1.00 - 1.08	0.000	0.000	1.000



(1a) Nominal engine operation characteristic with PTO
(2a) Nominal engine characteristic
(3a) Nominal propeller characteristic without PTO

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Figure 2-5 Power range diagram of an engine with main-engine driven generator

Curve 1a in Figure 2-5 shows the power range with main-engine driven generator (PTO)[1]. The latter can be a shaft generator (SG) which is either directly mounted on the intermediate shaft, or driven by a power take-off gear (PTO–G) mounted on the intermediate shaft or on engine free end side.

Due to the addition of constant nominal generator power over the major range of engine load, the curve does not directly relate to a propeller characteristic.

1) without specification of installation type

In the example of Figure 2-5, 2-12, the main-engine driven generator is assumed to absorb 5 % of nominal engine power. The CMCR point is selected on a propeller curve which includes the PTO power demand at the CSR point. This curve defines the nominal engine characteristic.
This approach allows a practically unlimited flexible PTO operation, just limited in the lower engine speed range by the PTO required minimum speed (as defined by the PTO device supplier) and the PTO layout limit Line 10, which is only relevant if a significant percentage of the installed engine power is utilised for PTO.

2.2.7 Power range limits with controllable pitch propeller (CPP)

For manoeuvring at nominal engine speed, a certain minimum engine power output is required, depending on the engine's rating. The minimum required power output depending on engine speed (point F) is shown in Figure 2-6.

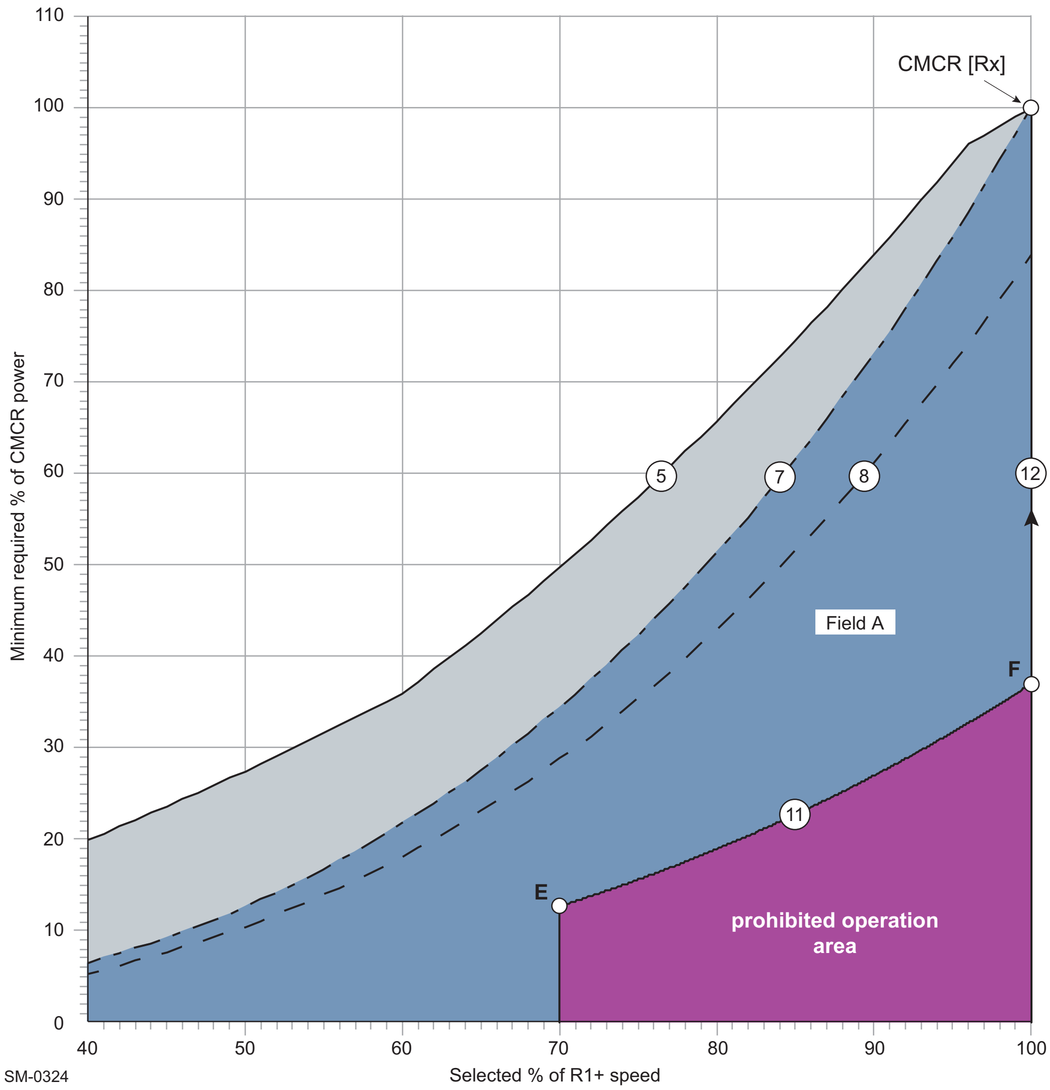


Figure 2-6 Speed dependent minimum required engine power

The minimum required CMCR engine power can be calculated as follows:

$P_{min} = 37\% \cdot n_{rel}^3$

where:

P_{min} .... = percentage of minimum required **CMCR** engine power
n_{rel} .... = percentage of engine speed, referring to **R1+** engine speed ($n_{rel} = n_{CMCR}/n_{R1+}$)

For engine speeds (n_{rel}) with less than 70% of R1+ speed, no minimum power is required.

The additional border lines and the different ranges shown in Figure 2-6, 2-13 are described below. For the existing border lines (Lines 5, 7 and 8) refer to section 2.2.5, 2-8 for detailed descriptions.

Line 11 Lower load limit between 70% of R1+ speed and 100% CMCR speed. The pitch position is such that at 100% speed, a minimum power as calculated by the formula above is reached (point F).

Line 12 For PTO operation without frequency converter, the engine power can vary between the minimum required power (point F) — as calculated by above formula — and 100% (CMCR), while running at 100% speed. By this the electrical sea load can be provided with constant frequency.

Field A Available design range for combinator operation as defined by Line 7, Line 11 and Line 12. The barred speed range due to torsional vibration limits must be respected.
For test purposes, the engine may be run at rated speed and low load for a one-time period of 30 minutes during dock trials (e.g. for shaft generator adjustment), in the presence of authorised representatives of the engine builder.
Further requests must be agreed by WinGD.

Example 1: Power range with CPP at 85% R1+ rated speed

For engines rated at e.g. 85% of R1+ speed, minimum 22.7% CMCR power is required for 100% CMCR propeller speed operation (see Figure 2-7). A zero pitch propeller line is shown, considering a typical characteristic. It is for reference only and might deviate project-specifically.

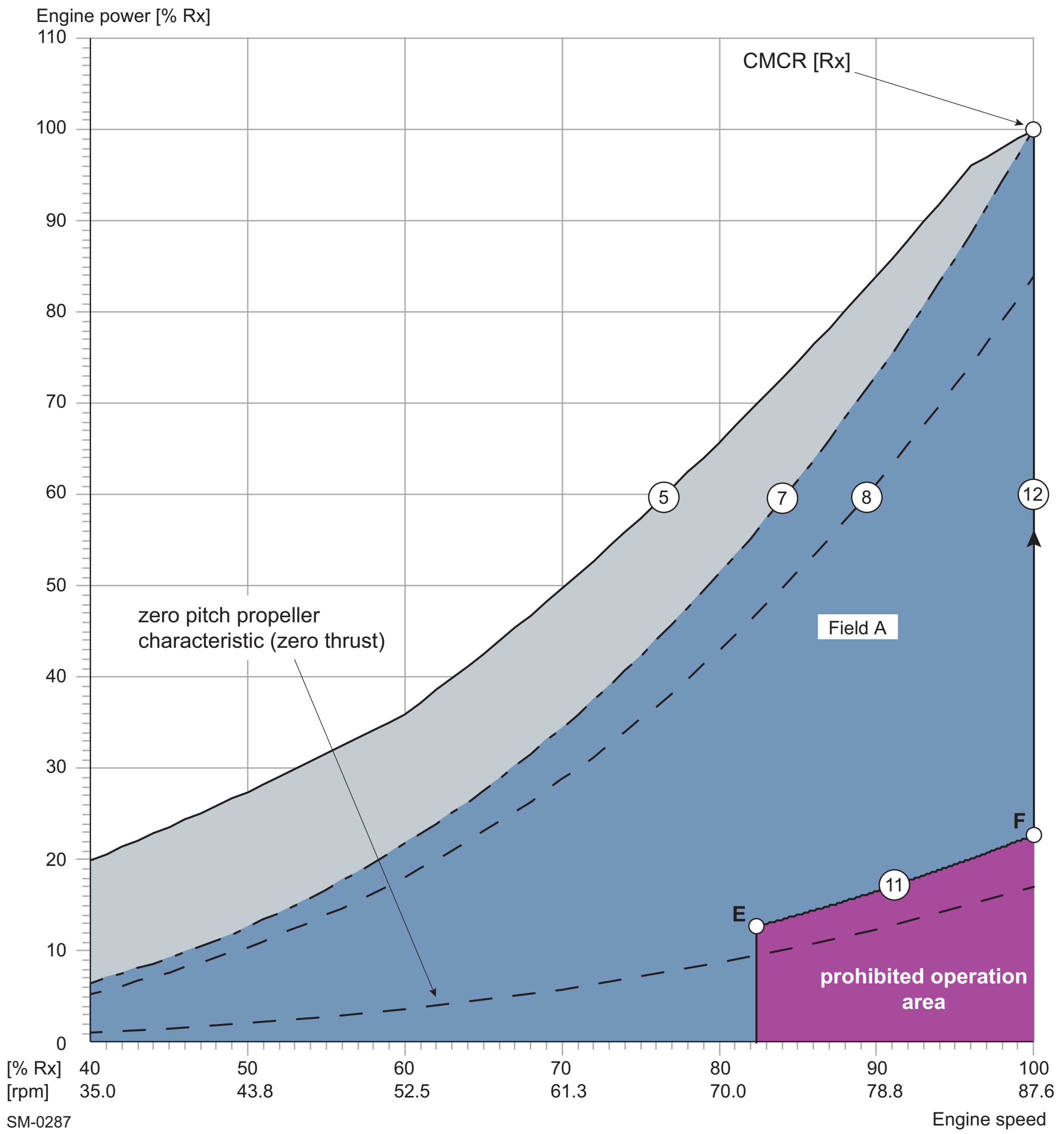


Figure 2-7 Power range diagram for engines rated with 85% of R1+ speed

Example 2: Power range with CPP at R1+ — R2+ rated speed

For engines rated at R1+ — R2+ speed, continuous operation in the marked Field A in Figure 2-6, 🗎 2-13 is possible. For 100% propeller speed operation, a minimum engine power of 37% CMCR is required (see Figure 2-8).

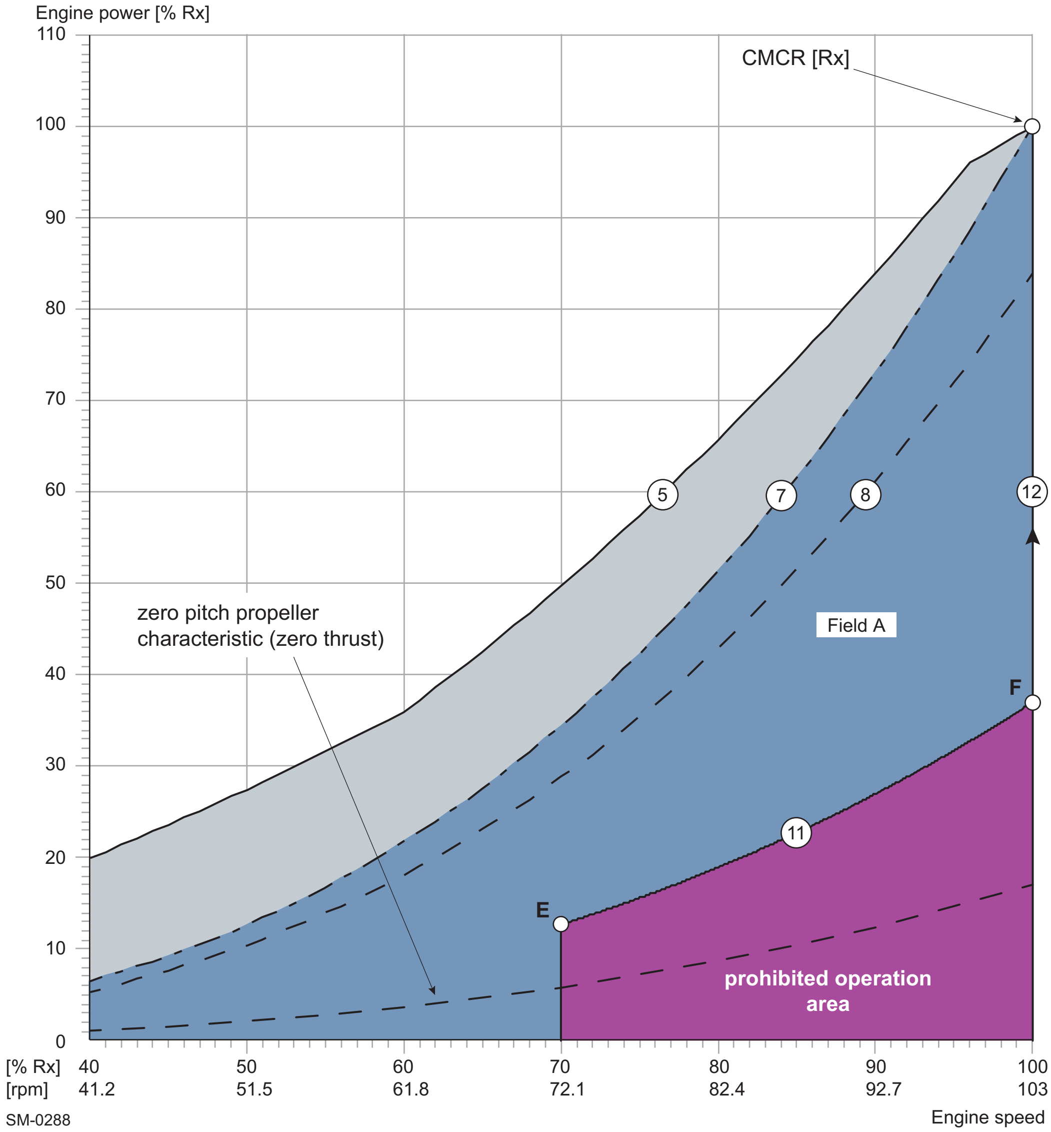


Figure 2-8 Power range diagram for engines rated on the R1+ — R2+ line

Example 3: Power range with CPP at R3 — R4 rated speed

For engines rated at R3 — R4 speed, a minimum 15.7% CMCR power is required for 100% CMCR propeller speed operation (see Figure 2-9). Considering the typical zero pitch propeller characteristic, unlimited engine operation is available as long as the propeller is coupled.

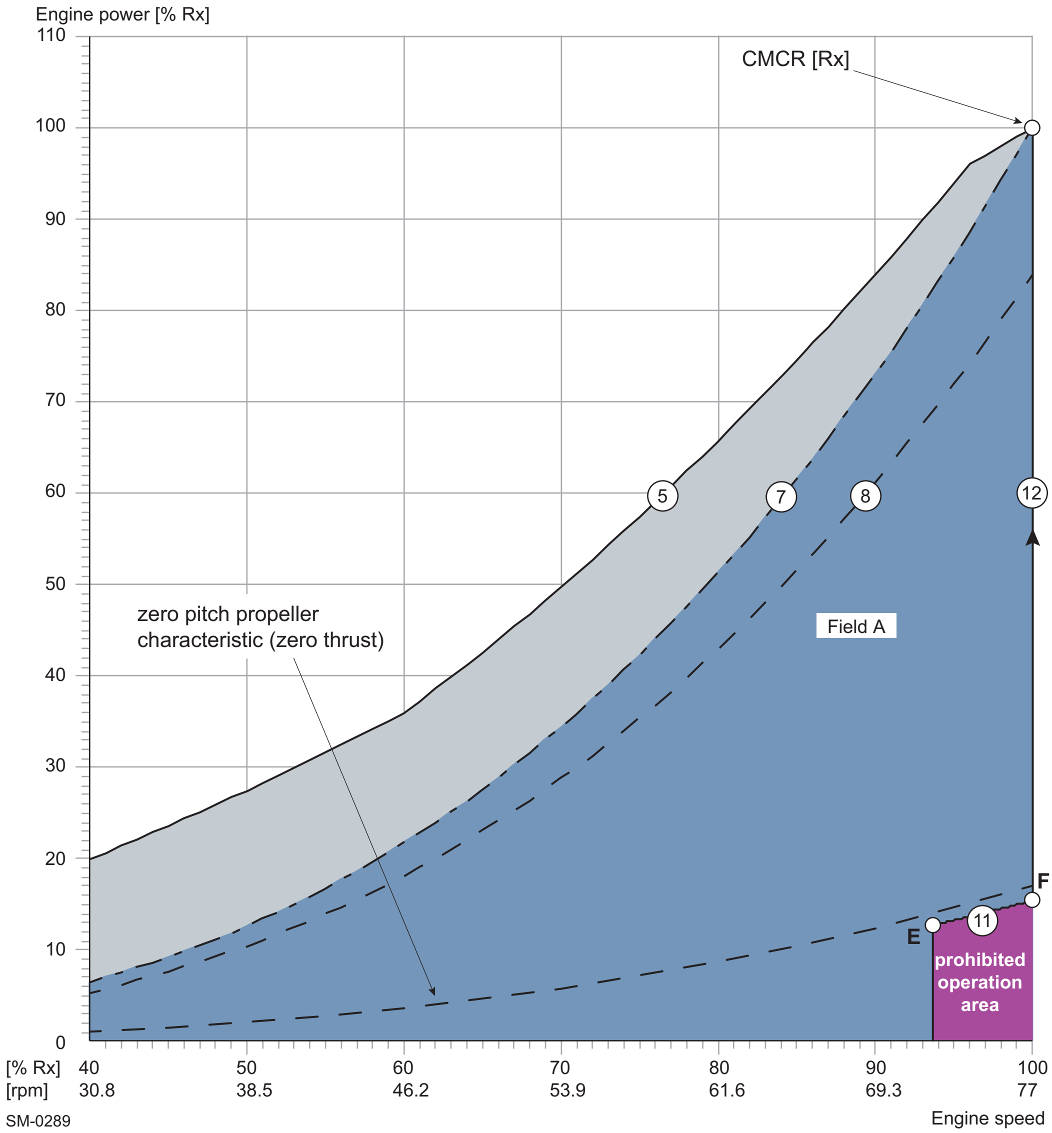


Figure 2-9 Power range diagram for engines rated on the R3 — R4 line

Requirements for control system with CPP

WinGD advises to include CPP control functions in an engine remote control system from an approved supplier. This ensures, amongst others, that the requirements of the engine builder are strictly followed.

The following operating modes shall be included in the control system:

- **Combinator mode 1:**
  Combinator mode for operation without shaft generator, or with shaft generator and frequency control system. Any combinator curve including a suitable light running margin can be set in field A.

- **Combinator mode 2:**
  Optional mode used in connection with shaft generators. During manoeuvring the combinator curve is freely selected in Field A. At sea, the engine is operated at constant speed between point F and Line 12.

For manual and emergency operation, separate set-points for speed and pitch are usually provided. At any location allowing such operation, a plate must be placed with the following warning:

> Engine must not be operated continuously at a pitch lower than xx% at any speed above xx rpm.

The values (xx) are to be defined according to installation data.

Operation in prohibited area

In addition, if the engine is operated for more than 3 minutes in the prohibited operation area, an alarm must be provided in either the ME safety system or the vessel's Alarm and Monitoring System.
If the engine is operated for more than 5 minutes in the prohibited operation area, then the speed must be reduced to idle speed (less than 70% speed).

2.3 Operating conditions

The engine can be operated without any restrictions in the ambient condition range 'winter', as permissible by *GTD*, up to design conditions. Operation outside these limits is possible, but further measures might need to be taken and the power output might be limited. For project-specific support please contact WinGD.

2.3.1 Reference conditions

Engine performance data — like BSEC, BSEF, tEaT and others — are based on reference conditions. These conditions are specified in ISO standard 15550 (core standard) and, for marine application, in ISO standard 3046 (satellite standard) as follows:

Air temperature before blower:	25 °C
Engine room ambient air temperature:	25 °C
Coolant temperature before SAC:	25 °C
Barometric pressure:	1,000 mbar
Relative humidity:	30 %

2.3.2 Design conditions

The capacities of ancillaries are specified according to ISO standard 3046-1 (clause 11.4) following the International Association of Classification Societies (IACS) and are defined as design conditions:

Air temperature before blower:	45 °C
Engine room ambient air temperature:	45 °C
Coolant temperature before SAC:	36 °C
Barometric pressure:	1,000 mbar
Relative humidity:	60 %

2.4 Ancillary system design parameters

The layout of the engine's ancillary systems is based on the rated performance (rating point Rx, CMCR). The given design parameters must be considered in the plant design to ensure a proper function of the engine and its ancillary systems:

Cylinder cooling water outlet temperature:	90 °C
Oil temperature before engine:	45 °C
Exhaust gas back pressure at rated power (Rx):	30 mbar

The engine power is independent of ambient conditions as found in marine applications. The cylinder water outlet temperature and the oil temperature before engine are system-internally controlled and have to remain at the specified level.

2.5 Electrical power requirement

Table 2-4 Electrical power requirement

No. cyl.	Power requirement [kW]	Power supply
Auxiliary blowers a)		
5	2 x 46	460 V / 60 Hz
6	2 x 58	
7	2 x 58	
8	2 x 73	
Turning gear		
5	5.5	440 V / 60 Hz
6	5.5	
7	5.5	
8	5.5	
UNIC Engine Control System		
5	1.0	230 V / 60 Hz
6	1.1	
7	1.2	
8	1.4	
Propulsion Control System		
all	acc. to maker's specifications	24 VDC UPS
Additional monitoring devices (e.g. oil mist detector, etc.)		
all	acc. to maker's specifications	

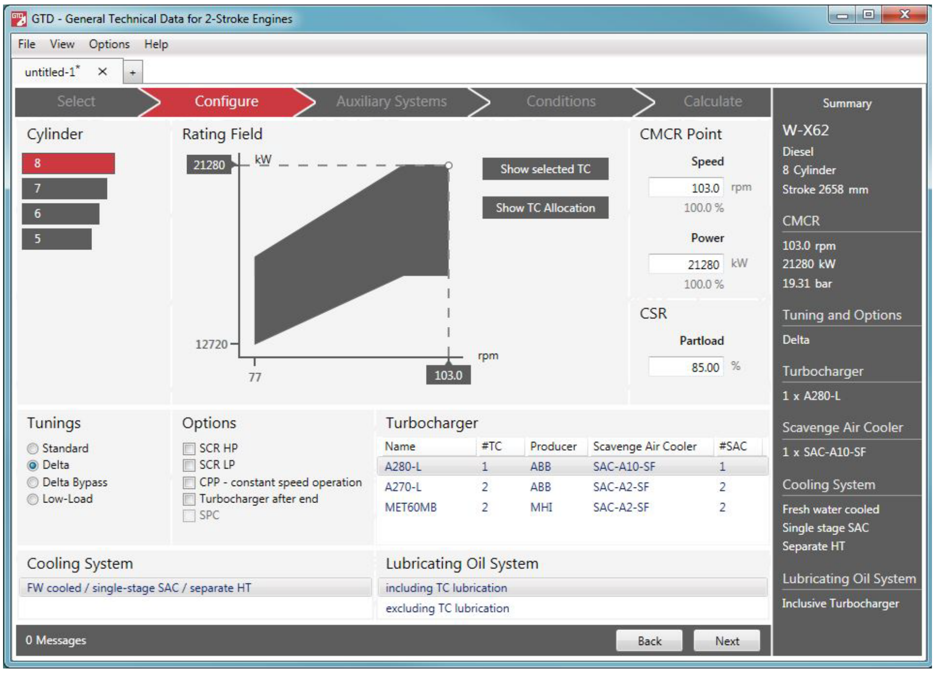
a) Minimal electric motor power (shaft) is indicated. Actual electric power requirement depends on size, type and voltage / frequency of installed electric motor. Direct starting or Star-Delta starting to be specified when ordering.

2.6 GTD - General Technical Data

GTD is a program for the calculation and output of general technical data which are relevant for planning a marine propulsion plant. All data in this program are relating to the entire 2-stroke engine portfolio.

Engine performance data

The GTD program allows calculation of the performance data (BSEC, BSEF, tEaT, etc.) for any engine power.



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GTD output

Beside the output of characteristic parameters in the whole rating field of an engine, the GTD application delivers data on the capacities of coolers, pumps, starting air bottles and air compressors. It also provides information on engine radiation, on the power requirement for ancillary systems, and outputs data suitable for estimating the size of ancillary equipment.
Furthermore, data about the available components and options depending on specification and engine rating can be output. In addition to the standard output for ISO reference and design conditions, further operating conditions for which information is required can be defined.

The GTD application is accessible on Internet at the WinGD Customer Portal or from the WinGD corporate webpage using the following link:
https://www.wingd.com/en/media/general-technical-data

3 Engine Installation

The purpose of this chapter is to provide information to assist in the installation of the engine. It is for guidance only and does not supersede current instructions.

3.1 Dimensions and masses

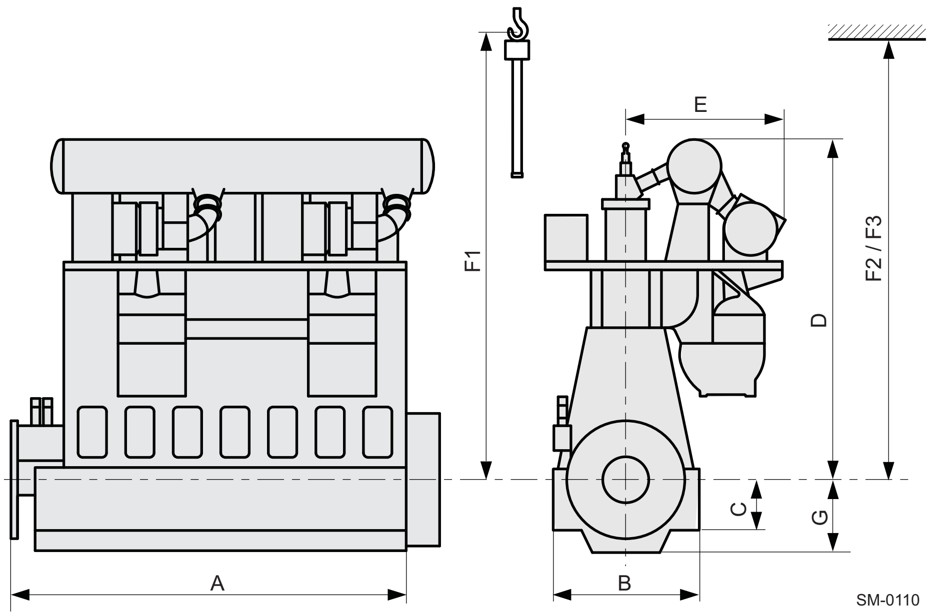


Figure 3-1 Engine dimensions

Table 3-1 Engine dimensions and masses

No. cyl.	Dimension in mm with a tolerance of approx. ±10 mm									Net eng. mass a)
	A	B	C	D	E b)	F1 c)	F2 d)	F3 e)	G	[tonnes]
5	7,000	4,200	1,360	9,750	3,915	11,670	11,670	10,800	2,110	325
6	8,110									377
7	9,215									435
8	10,320									482
Min. crane capacity: 5,000 kg										

a) Without oil / water; net engine mass estimated according to nominal dimensions given in drawings, including turbocharger and SAC, piping and platforms
b) Dimension depending on turbocharger type
c) Min. height for vertical removal of piston
d) Min. height for vertical piston removal with double-jib crane
e) Min. height for tilted piston removal with double-jib crane

NOTE	The dimensions given in above table are not binding. For prevailing data refer to the relevant drawings, which are updated on a regular basis.

Table 3-2 Dimensions and masses of main components

Component		Number of cylinders				Unit	Dimens./ Mass
		5	6	7	8		
Bedplate, including bearing girders		7,578	8,684	9,790	10,896	[mm]	length
		45,330	54,888	64,446	74,004	[kg]	mass
Crankshaft		7,497	8,603		10,815	[mm]	length
	Crankshaft types: — 5-6 cyl.: FCV2 [a] / FCV3 [b] — 8 cyl.: FCV1	72,000 [a] 70,000 [b]	84,000 [a] 82,000 [b]		114,000	[kg]	mass
Flywheel		3,460				[mm]	diameter
	— light — medium — heavy	4,058 7,623 11,003				[kg]	mass
Monoblock column		5,873	6,979	8,085	9,191	[mm]	length
		34,893	40,505	46,117	51,729	[kg]	mass
Tie rod, without assembly parts		6,208				[mm]	length
		318				[kg]	mass
Cylinder block		5,810	6,916	8,022	9,128	[mm]	length
		32,062	37,124	42,186	47,248	[kg]	mass
Cylinder liner, without assembly parts		2,976				[mm]	length
		3,535				[kg]	mass
Cylinder cover, without assembly parts		1,102				[mm]	diameter
		1,487				[kg]	mass
Connecting rod, without top-/bottom-end bearing shells / covers		2,658				[mm]	length
		2,540				[kg]	mass
Crosshead	Pin	860				[mm]	length
		1,454				[kg]	mass
	Guide shoe	1,096				[mm]	width
		349				[kg]	mass
Piston complete, with rod, without rings		3,649				[mm]	length
		1,695				[kg]	mass

Component	Number of cylinders				Unit	Dimens./ Mass
	5	6	7	8		
Gland box housing	460				[mm]	diameter
	72				[kg]	mass
Scavenge air receiver complete, without assembly parts	5,770	6,876			[mm]	length
	13,500	14,500			[kg]	mass
Exhaust valve complete	approx. 1,460				[mm]	height
	2,160				[kg]	mass
Supply unit	1,825	1,825	1,825	2,195	[mm]	length
	1,121	1,121	1,121	1,143	[kg]	mass
The accuracy of all given figures is ± 10 %.						

3.1.1 Dismantling heights for piston and cylinder liner

Dimensions F1, F2, F3 in Figure 3-1, 🗎 3-1 and the corresponding table are for guidance only and may vary depending on crane dimension, handling tools and dismantling tolerances.
However, please contact WinGD or any of its representatives if these values cannot be maintained or if more detailed information is required.

For details see also drawings **'Dismantling Dimensions'** (DG 0812) provided on the WinGD corporate webpage under the following links:
5-cyl. engine
6-cyl. engine
7-cyl. engine
8-cyl. engine

3.1.2 Crane requirements

- An overhead travelling crane is to be provided for normal engine maintenance. (Crane capacity see Table 3-1, 🗎 3-1.)
- The crane is to conform to the requirements of the classification society.

NOTE	As a general guidance WinGD recommends a two-speed hoist with pendent control, which allows selecting either high or low speed, i.e. high speed 6.0 m/minute, low speed 0.6-1.5 m/minute.

3.1.3 Thermal expansion at turbocharger expansion joints

Before making expansion pieces, enabling connections between the engine and external engine services, the thermal expansion of the engine has to be taken into account. The expansions are defined (from ambient temperature 20 °C to service temperature 55 °C) as follows (see also Figure 3-2):

Expansion	Distance from ...
Transverse expansion (X)	... crankshaft centreline to centre of gas outlet flange
Vertical expansion (Y)	... bottom edge of bedplate to centre of gas outlet flange
Longitudinal expansion (Z)	... engine bedplate aft edge to centre of gas outlet flange

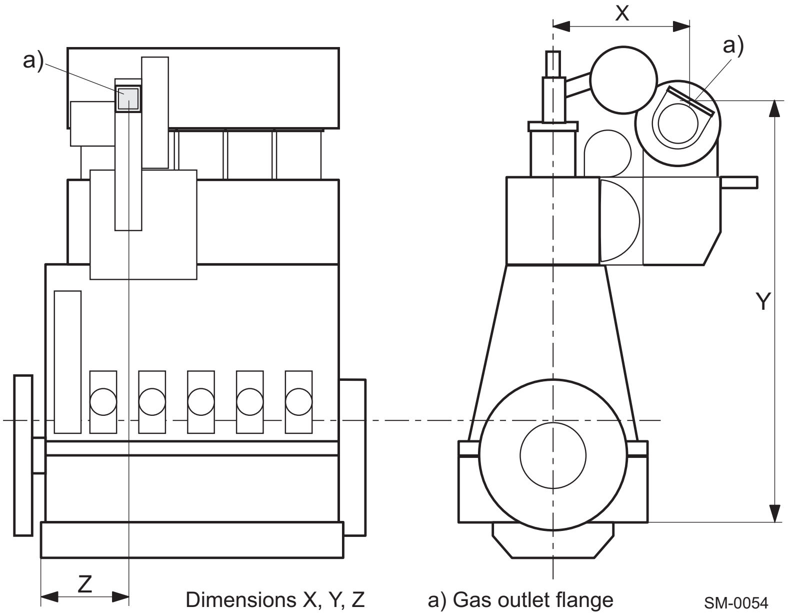


Figure 3-2 Thermal expansion, dim. X, Y, Z

Calculating thermal expansion

$\Delta x\ (\Delta y, \Delta z) = X\ (Y, Z) \cdot \alpha \cdot \Delta T$

where:

Δx, Δy, Δz .. = thermal expansion
X, Y, Z ...... = distance as per relevant pipe connection plan and outline drawing
α .... = $1.15 \cdot 10^{-5}$ (coefficient of thermal expansion)
ΔT .... = difference between service temp. and ambient temp. [°C]

3.1.4 Content of fluids in the engine

Table 3-3 Fluid quantities in the engine

No. of cyl.	Lubricating oil	Fuel oil	Cylinder cooling water	Freshwater in SAC [a]
	[kg]	[kg]	[kg]	[kg]
5	2,100	20	650	600 [b]
6	2,400	20	775	600 [b] / 1,050 [c]
7	2,700	20	925	700 [b] / 1,300 [c]
8	2,950	28	1,050	700 [b] / 1,400 [c]

a) The given water content is approximate.
b) Values for executions with 1 scavenge air cooler.
c) Values for executions with 2 scavenge air coolers.

3.2 Engine outline views

The latest versions of the **Engine Outline Drawings** (DG 0812) are provided on the WinGD corporate webpage under the following links:

5-cyl. engine
6-cyl. engine
7-cyl. engine
8-cyl. engine

3.3 Platform arrangement

3.3.1 Drawings

For platform arrangement see the links given in section 3.2, 🗎 3-6.

3.3.2 Minimum requirements for escape routes

The platforms shown in the relevant drawings are arranged in such a way as to ensure safe escape routes for the crew. The minimum sizes required by the classification societies are met.

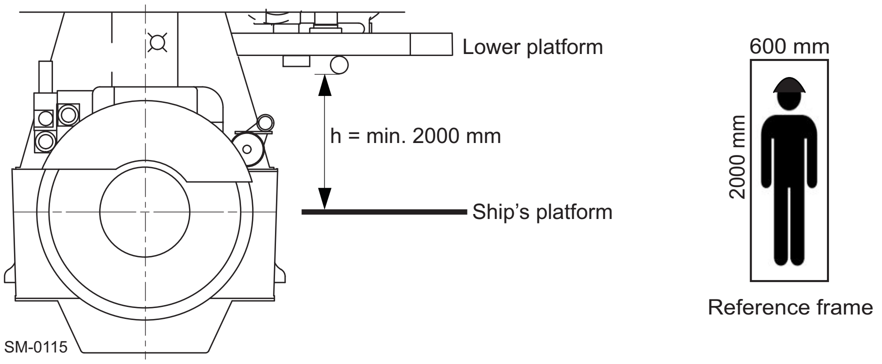


Figure 3-3 Minimum requirements for headroom

Important!

- The minimum sizes are to be taken into account when installing the engine. Special attention is to be given to the minimum distance between the ship's platform and the lower engine platform, to ensure sufficient headroom (see Figure 3-3).
- No dead ends may be created on the platforms by shipboard installations. If a dead end cannot be avoided, then a passage leading to the ship's platform has to be cleared before the dead end (distance from dead end: max. 2,000 mm).

See also the links to drawings in section 3.2, 🗎 3-6.

3.4 Seating

Engine seating is integral with the double-bottom structure and has to be of sufficient strength to support the weight of the engine, transmit the propeller thrust and withstand external couples and stresses related to propeller and engine resonance.

- Before any seating work can be performed, make sure the engine is aligned with the intermediate propeller shaft.
- The longitudinal beams situated under the engine are to protrude from the engine room bulkhead by at least half the length of the engine, and aft as far as possible.
- The maximum allowable rake is 3° to the horizontal.

More details about engine seating can be found in the relevant **Fitting Instruction** (DG 9710) on the WinGD corporate webpage under the following link:
Fitting instruction - Engine seating and foundation

The latest version of the **Marine Installation Drawing Set** relevant for engine seating and foundation (DG 9710) is provided on the WinGD corporate webpage under the following link:
MIDS

3.5 Assembly

Engines may be installed as complete units or assembled from subassemblies in the vessel, which may be afloat, in dry dock, or on the slipway.

3.5.1 Assembly of subassemblies

When the engine seating has been approved, the bedplate is lowered onto blocks placed between the chocking points. The thickness of the blocks depends on the final alignment of the engine. Engine bedplates comprise fabricated sections with tapped holes for the jacking screws for engine alignment, and drilled holes to allow the passing of the holding-down bolts.

For checking the dimensions optical devices or lasers may be used

- Proceed with the preliminary alignment of the bedplate using wedges or jacking screws.
- Position the engine coupling flange to the intermediate shaft coupling flange.
- Ensure that the gap between both flanges is close to the calculated figures and that both flanges are exactly parallel on the horizontal plane (max. deviation 0.05 mm).
- In the vertical plane, set the engine coupling flange 0.4-0.6 mm higher than the calculated figures.
- Place the bearing caps in position and install the turning gear.
- Ensure that the crankshaft deflections are as recorded in the ‘Engine Assembly Records’.
- Check the bedplate level in longitudinal and diagonal directions with a taut-wire measuring device provided by the engine builder.
- Compare the readings with those recorded at works.

All final dimensions are to be witnessed by the representatives of the engine builder and the classification society, and recorded on appropriate log sheets. Crankshaft deflections at this stage are to correspond with the values recorded at works.

- Temporarily secure the bedplate against unexpected movement.
- Continue engine assembly by mounting the columns, cylinder blocks, running gear and scavenge air receiver.
- Ensure that the bearing caps are loose before tensioning the tie rods.
- Make periodic checks of the crankshaft deflections to observe and correct any possible engine distortions.
- Carry out careful adjustments of the wedges or the jacking screws to re-establish the preliminary alignment setting.

Once the engine assembly is completed, the final alignment and chocking is carried out with the vessel afloat.

3.5.2 Installation of a complete engine

In the event that the engine is shipped in part deliveries and assembled at the shipyard before installation in the vessel, the shipyard is to undertake assembly work in accordance with the demands of a representative of the engine builder and the classification society.

NOTE	
	• Strict attention is to be paid to the removal of anti-corrosion coatings and the subsequent application of rust preventing oil where required. • The alignment tools are to be clean and ready for use.

Please observe:

- Engine mounting is to be carried out systematically.
- The measurement readings have to be recorded on appropriate log sheets and compared for correctness with the data in the ‘Engine Assembly Records’ completed after test run in the manufacturer’s works.
- The engine is to be lowered onto blocks placed between the chocking points.
- The blocks are to be set in such a manner that the engine is slightly higher than the final position, because less effort is required to lower the engine than to raise it for alignment.
- For movements in the horizontal plane, both in lateral or longitudinal directions, the shipyard is to construct appropriate anchor points for the use of hydraulic jacks. Such movements have to be carried out with great care to avoid stresses and distortions to the bedplate.
- Regular crankshaft deflection readings have to be taken to observe the effects, and any noticed deviation has to be rectified immediately.

3.5.3 Installation of an engine from assembled subassemblies

Subassemblies of the engine may be assembled ashore before installation in the ship. One such assembly may comprise the following components:

- Bedplate
- Main and thrust bearings
- Crankshaft
- Turning gear
- Flywheel

The placing on blocks and alignment to shafting is analogue to that described in section 3.5.1, 🗎 3-9.

3.5.4 Installation of an engine in ship on slipway

Installing a complete or partially assembled engine in a ship under construction on an inclined slipway is possible when careful attention is paid to the following:

- Large components suspended to take account of the incline
- Tie rods centred and exactly perpendicular to the bedplate before tightening.
- Side, fore and aft arresters temporarily fitted to prevent the engine from moving during launching of the ship
- Additional temporary stays attached at upper platform level to steady the engine during launching

3.6 Engine and shaft alignment

Alignment and chocking of the engine should be carried out in accordance with our recommendations and is subject to test and inspection by the relevant classification society.

Each stage of engine mounting is to be checked by qualified personnel and the measurements cross-checked with the design figures. In the event of discrepancies the responsible parties (e.g. shipyard) are to advise the representative of the engine builder or WinGD.

3.6.1 Instructions and limits

Alignment can be done with either jacking screws or wedges.

For detailed alignment procedures refer to the latest version of **Engine Alignment Documents** (DG 9709) provided on the WinGD corporate webpage under the following link:
Engine alignment

3.6.2 Tools

For **Engine Alignment Tools** (DG 9710-01) refer to the latest version of the respective drawings, which are provided on the WinGD corporate webpage under the following link:
Tool engine alignment

3.7 Engine coupling

3.7.1 Design

The design of coupling bolts and holes for the flange connection of crankshaft / propulsion shafts as provided by design group 3114 is included in the engine design approval by all major classification societies.

3.7.2 Machining and fitting of coupling bolts

- Before fitting the coupling bolts ensure that the mating flanges are concentric. Close the gap between the flanges completely by means of min. 4 temporary (non-fitted) bolts evenly distributed over the pitch hole diameter.
- Carry out drilling and reaming of engine and shaft couplings by means of a computer controlled drilling machine or an accurately centred jig.
- In case of non-matching holes or damaged holes apply joint cylindrical reaming to an oversize hole and then fit an individually machined bolt.
- The bolts have to be available for inserting in the holes on completion of reaming. Each bolt is to be stamped with its position in the coupling, with the same mark stamped adjacent to the hole. The following tolerances have to be met:
  — bolt hole tolerance: H7
  — bolt tolerance: g6 (clearance fit)
- If there is any doubt that a fitted bolt is too slack or too tight, refer to the classification society surveyor and a representative of the engine builder.

3.7.3 Tightening

- When tightening the coupling bolts it is essential to work methodically. Perform crosswise tightening, taking up the threads on opposite bolts to hand-tight, followed by sequential torque tightening. Finally ensure the same proper tightening for all bolts.
- Mark each bolt head in turn (1, 2, 3, etc.) and tighten opposite nuts in turn according to Tightening Instructions, making sure that the bolt head is securely held and unable to rotate with the nut.
- Lock castellated nuts according to Class requirements with either locking wires or split pins. Use feeler gauges during the tightening process to ensure that the coupling faces are properly mated with no clearance.

3.7.4 Installation drawing

The latest version of the drawing, relevant for the **Connection Crank / Propeller Shaft** (DG 3114), is provided on the WinGD corporate webpage under the following link:
Connection crank / propeller shaft

3.8 Engine stays

Ship vibrations and engine rocking caused by the engine behaviour (see chapter 6 Engine Dynamics, 6-1) are reduced by fitting lateral stays (see 6.2, 6-4) and longitudinal stays (see 6.3, 6-6).

The latest version of the **Marine Installation Drawing Set** relevant for engine stays (DG 9715) is provided on the WinGD corporate webpage under the following link:
MIDS

3.9 Propulsion shaft earthing

Electric current flows when a potential difference exists between two materials. The creation of a potential difference is associated with *thermoelectric* by the application of heat, *tribo-electric* between interactive surfaces, *electrochemical* when an electrolytic solution exists, and *electromagnetic induction* when a conducting material passes through a magnetic field.

Tracking or leakage currents are created in machinery by any of the above means and — if they are not adequately directed to earth — can lead to component failures, or in some cases result in fires and interference with control and monitoring instrumentation.

3.9.1 Preventive action

Using earthing brushes in contact with slip rings and bonding the chassis by braided copper wire are common ways of protecting electric machines. Where operating loads and voltages are comparatively low, the supply is isolated from the machine by an 'isolating transformer', often with handheld power tools. The build specification dictates the earthing procedure to be followed and the classification society is to approve the final installation.
On vessels with star-wound alternators the neutral is considered to be earth, and electrical devices are protected by automatic fuses.

Isolation of instrument wiring

Ensure that instrument wiring meets the building and classification society specifications and that it is shielded and isolated to prevent induced signal errors and short circuits.
In certain cases large items of machinery are isolated from their foundations, and couplings are isolated to prevent current flow, for instance when electric motors are connected to a common gear box.

Retrospective fitting of earthing devices is not uncommon, but due consideration is to be given at design stage to adequate shielding of control equipment and earthing protection where tracking and leakage currents are expected. Magnetic induction and polarisation are to be avoided and degaussing equipment incorporated if there is likely to be a problem.

3.9.2 Earthing device

Figure 3-4, 🗎 3-15 shows a typical shaft earthing device.
The slip ring (1) is supplied as matched halves to suit the shaft, and secured by two tension bands (2) using clamps (12). The slip ring mating faces are finished flush and butt jointed with solder. The brushes (4) are housed in the twin holder (3) clamped to a stainless steel spindle (6), and there is a monitoring brush (11) in a single holder (10) clamped to an insulated spindle (9). Both spindles are attached to the mounting bracket (8).

Conducting material for slip rings

Different combinations of conducting material are available for the construction of slip rings. However, alloys with a high silver content are found to be efficient and hard wearing.

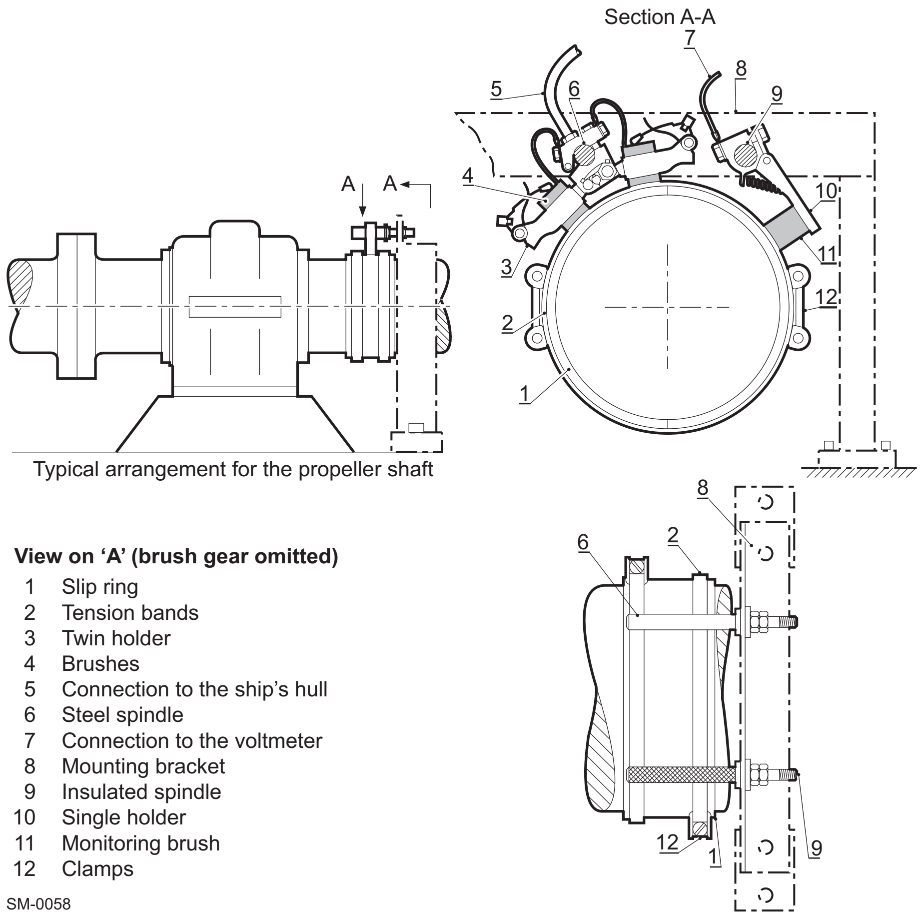


Figure 3-4 Shaft earthing arrangement

Position of earthing device on shaft

The earthing device has to be arranged as close as possible to the engine. In case a shaft generator / motor is installed, the earthing device has to be arranged on the front side of the generator / motor, as close a possible to the engine.

Connecting electric cables

The electric cables are connected as shown in Figure 3-5, 3-16 with the optional voltmeter. This instrument is at the discretion of the owner, but it is useful to observe that the potential to earth does not rise above 100 mV.

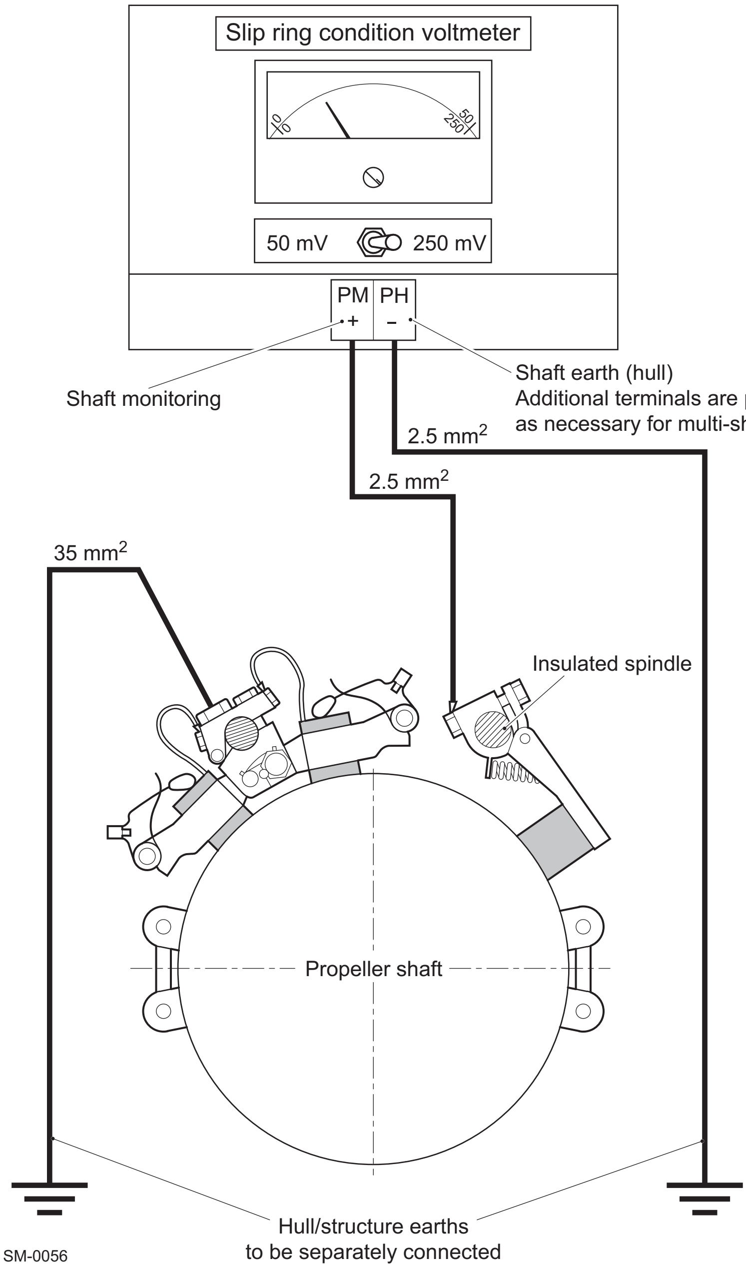


Figure 3-5 Shaft earthing with condition monitoring facility

3.10 Fire protection

Fires may develop in areas such as scavenge air receiver / piston underside. The engine is fitted with a piping system which leads the fire extinguishing agent into the critical areas.

Where fire protection is required, the final arrangement of the fire extinguishing system is to be submitted for approval to the relevant classification society.

Extinguishing agents

Various extinguishing agents can be considered for fire fighting purposes. They are selected either by the shipbuilder or the shipowner in compliance with the rules of the classification society involved.

Steam as an alternative fire extinguishing medium is permissible for the scavenge air spaces of the piston underside, but may cause corrosion if countermeasures are not taken immediately after its use.
These countermeasures comprise:
— Opening scavenge spaces and removing oil and carbon deposits
— Drying all unpainted surfaces and applying rust protection (i.e. LO)

NOTE	If steam is used for the scavenge spaces, a water trap is recommended to be installed at each entry to the engine and assurance obtained that steam shut-off valves are tight when not in use.

Table 3-4 Recommended quantities of fire extinguishing medium

Piston underside and scavenge air receiver		Bottle		Number of cylinders			
				5	6	7	8
Volume [m^3/cyl]	Mass [kg/cyl]	Size [kg]	Extinguishing medium	Quantity of fire extinguishing bottles			
6	22	45	Carbon dioxide (CO_2)	3	3	4	4

4 Ancillary Systems

Sizing the ancillary systems of the engine, i.e. for freshwater cooling, lubricating oil, fuel oil, etc., depends on the contracted maximum engine power. If the expected system design is out of the scope of this manual, then contact our representative or WinGD directly.

GTD

The *GTD* program provides data for estimating the size of ancillary equipment and enables all engine and system data at any Rx rating within the engine rating field to be obtained. However, for convenience or final confirmation when optimising the plant, WinGD provides a computerised calculation service.

All pipework systems to be flushed and proved clean before commissioning!

All pipework systems and fittings are to conform to the requirements laid down by the legislative council of the vessel's country of registration and the classification society selected by the owners. They are to be designed and installed to accommodate the quantities, velocities, flow rates and contents identified in this manual, set to work in accordance with the build specification as approved by the classification society and protected at all times from ingress of foreign bodies.

4.1 Twin-engine installation

A vessel equipped with two separate main propulsion engines is considered a twin-engine installation. The installation of two WinGD 2-stroke engines allows combining individual engine auxiliary systems.

In Table 4-1 WinGD provides information based on engines' requirements. Class and other binding rules might overrule.

Table 4-1 Common and independent systems in twin-engine installations

System	Independent system for each engine required	Common system possible	Remarks
LT cooling water system (see Figure 4-1, 4-3)		X	Please note: Parallel independent LT cooling water supply per engine to the scavenge air coolers from common LT cooling water circuit
		X	Please note: Parallel independent LT cooling water supply per engine to the LO cooler and HT cooling water cooler from common LT cooling water circuit
HT cooling water system	X		
Main LO system	X		
Cylinder LO system (see Figure 4-2, 4-4)		X	Day tanks for high- resp. low BN lubricating oil
		X	Rising pipe
	X		Separate distribution to each engine
Fuel oil system		X	Feed system
	X		Booster circuit systems
Starting air system	X		
Control air		X	Supply system
Leakage collection system and washing devices	X		
Exhaust gas system	X		
Engine venting pipes	X		

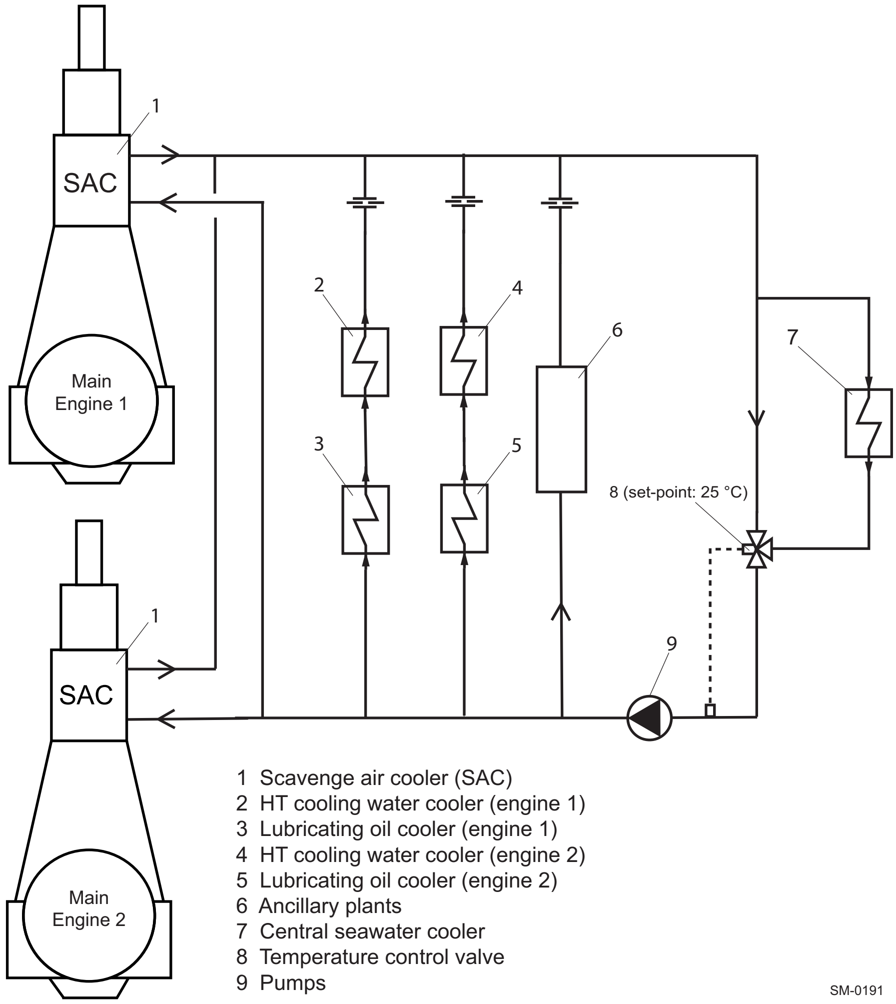


Figure 4-1 LT cooling water system layout for twin-engine installation

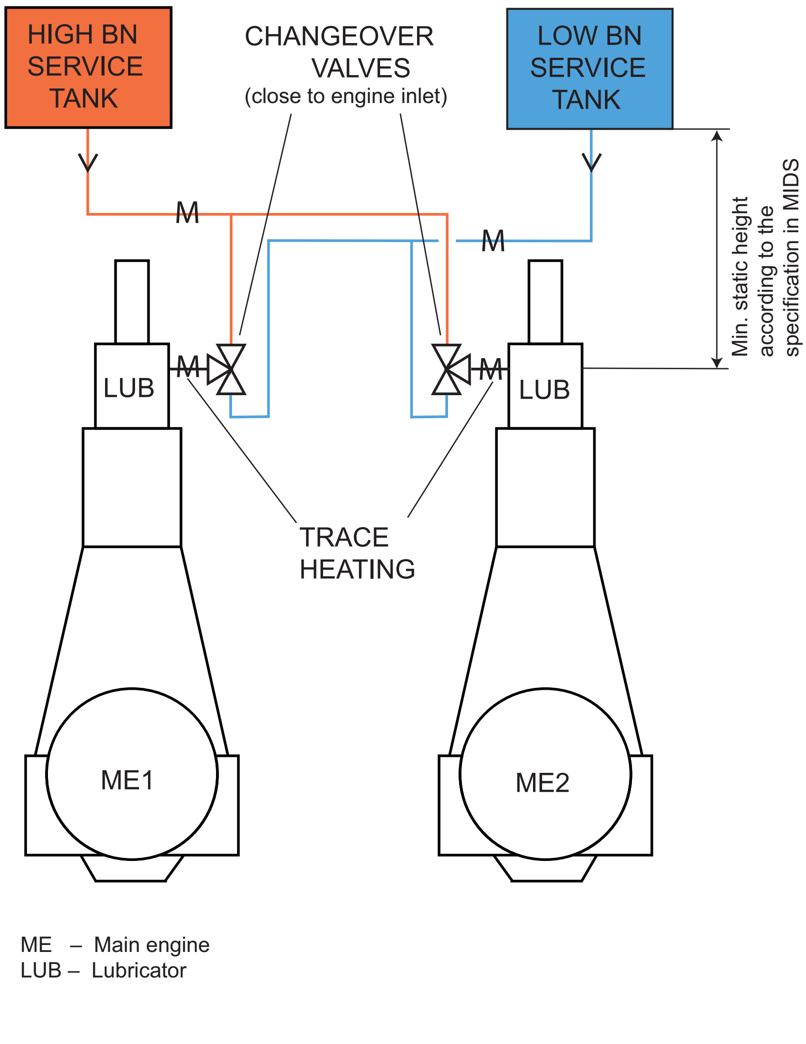


Figure 4-2 Cylinder LO system layout for twin-engine installation

4.2 Cooling water system

The latest version of the **Marine Installation Drawing Set** relevant for the cooling water system (DG 9721) is provided on the WinGD corporate webpage under the following link:
MIDS

Freshwater cooling system

As freshwater is the standard cooling medium of scavenge air coolers, it involves the use of a central freshwater cooling system.

Freshwater cooling systems reduce the amount of seawater pipework and its attendant problems like scaling and corrosion. They provide for more efficient cooling as they allow a higher heat load, i.e. freshwater can be heated up to a higher temperature level than seawater and together with a lower flow rate allows the same cooling effect to be obtained. Thereby the overall running costs are reduced.

Figure 4-3 shows the general installation principle.

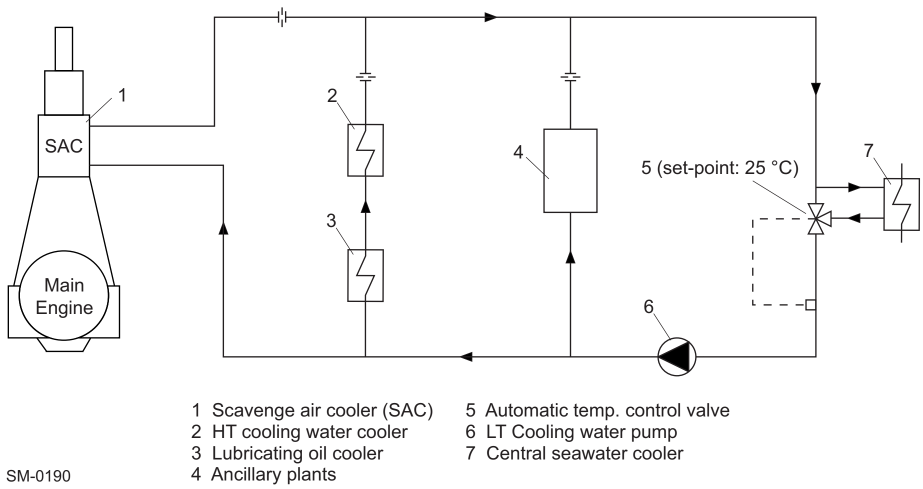


Figure 4-3 Scheme of cooling water system

The central freshwater cooling system comprises a low-temperature (LT) and a high-temperature (HT) circuit and runs with single-stage scavenge air cooler and separate HT circuit.

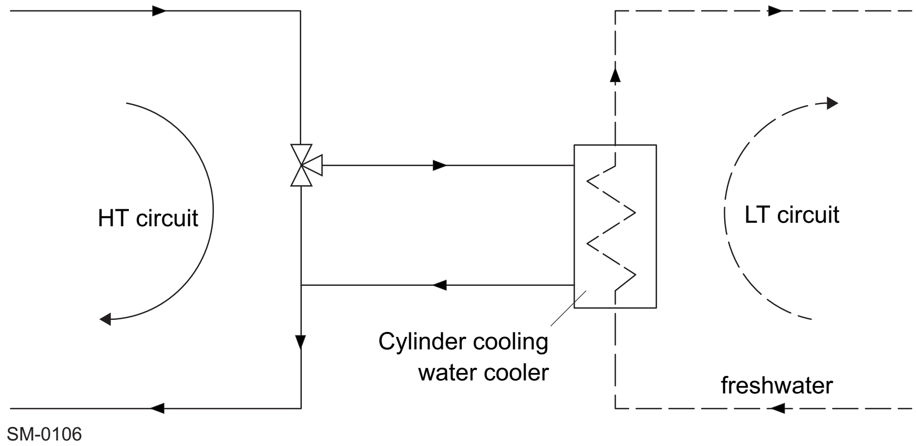


Figure 4-4 Separate HT cooling water circuit

Separate HT circuit with own cooler

The high-temperature circuit must be completely separated from the low-temperature circuit. In this case the HT circuit has its own separate cooler with freshwater from the low-temperature circuit as cooling medium.

GTD

To obtain the necessary data for this arrangement refer to the *GTD* application.

4.2.1 Central freshwater cooling system components

Low-temperature circuit

Table 4-2 Main components LT circuit

Seawater strainer	
Simplex or duplex to be fitted at each sea chest and arranged to enable manual cleaning without interrupting the flow. The strainer perforations are to be sized (no more than 6 mm) such that the passage of large particles and debris damaging the pumps and impairing heat transfer across the coolers is prevented.	
Seawater pump	
Pump type	Centrifugal
Pump capacity	According to GTD: The seawater flow capacity covers the need of the engine only and is to be within a tolerance of 0 to + 10 % of the GTD value.
Delivery head	Determined by system layout
Central cooler	
Cooler type	Plate or tubular
Cooling medium	Seawater
Cooled medium	Freshwater
Design criterion	Keeping max. 36 °C LT while seawater temp. is 32 °C
Margin for fouling	10-15 % to be added
Heat dissipation	Refer to GTD.
Freshwater flow	
Seawater flow	
Temperatures	
Temperature control	
The central freshwater cooling system is to be capable of maintaining the inlet temperature to the scavenge air cooler(s) between 10 and 36 °C. It is recommended to set the controller to 25 °C.	
Freshwater pumps for LT circuit	
Pump type	Centrifugal
Pump capacity	According to GTD: The freshwater flow capacity covers the need of the engine only and is to be within a tolerance of 0 to + 10 % of the GTD value.
Delivery head	The final delivery head is determined by the layout of the system and is to ensure that the inlet pressure to the scavenge air cooler(s) is within the range of summarised data.

NOTE	WinGD recommends that the LT cooling water controller is set to 25°C (set-point) as this has a positive influence on the engine's performance.

High-temperature circuit

Table 4-3 Main components HT circuit

HT cooling water pump	
Pump type	Centrifugal, preferably with a steep headcurve [a)]
Pump capacity	According to *GTD*: The flow capacity is to be within a tolerance of -10 to +20% of the GTD value.
Delivery head [b)]	The delivery heat must be determined according to the total pressure losses (resistance) of the actual piping installation arrangement, as for a non-pressurised system.
Working temperature	95°C
Buffer unit	
The required static water pressure at pump inlet is generated by a buffer unit. The buffer unit pressure is maintained at a constant setting by a controlled air supply, effectively creating an air cushion in the top section of the unit. The buffer unit acts as volume-compensating device. The water volume can expand or contract without altering the system pressure. Subsequently, if the cooling water pumps should fail, the cooling system remains pressurised, thus avoiding vapour formation in the system. The initial filling of the buffer unit should be above the low-level alarm, i.e. at approximately 30% of its total capacity. The large air cushion available in the buffer unit can partly compensate the expansion or contraction of the water volume without activating the control air pressure unit. In this way, continuous operation of this air pressure unit is avoided. The final design must meet the requirements of the concerned classification societies. The working pressure can vary, depending on the location of the main water pumps to the engine cooling water inlet. The buffer unit contains the following main equipment: — Control air valve (DN15, pressure range adjustable to 3-5bar), which automatically reduces and maintains the supply air pressure (7-8bar, filtered air) to the required constant static pressure. — Solenoid valve, fitted upstream of the control air valve, which interlocks the air inlet with the minimum water level in the buffer unit. — Relief valve DN32, adjusted to approximately 5.5bar. — High- and low-level switches to control the supply pump. The low-level switch is set at approximately 35% of the total content of the buffer unit. The difference between the high and the low level should correspond to a volume of approximately 150l. — Low- and high-level alarms. The low level is set at approximately 30% of the total capacity of the buffer unit. The compensation pipe route, which connects the buffer unit to the pump suction, should be kept to a minimum. With the buffer unit close to the cooling water pumps, a module including all main components of the cooling system can easily be realised. For pipe sizes and tank capacities see *MIDS*.	

Supply pump	
Pump type	Positive displacement pump; to replace the water leakage losses in the cylinder cooling water system. This pump is automatically controlled by the water level in the buffer unit. It would also be advisable to monitor the running period of the supply pump. Such monitoring will warn when the running period exceeds a pre-set value, indicating unusual water losses in the system. Spare parts for the supply pump must be available according to classification societies' requirements.
Pump capacity	$0.5\,m^3/h$
Design pressure	7 bar
Automatic temperature control valve	
Valve type	Electrically or electro-pneumatically actuated three-way type (butterfly valves are not adequate) having a linear characteristic
Design pressure	5 bar
Test pressure	Refer to specification laid down by classification society.
Press. drop across valve	Max. 0.5 bar
Controller	Proportional plus integral (PI); also known as proportional plus reset for steady state error of max. ±2 °C and transient condition error of max. ±4 °C
Temperature sensor	According to control valve manufacturer's specification; fitted in engine outlet pipe

a) As a guide, the minimum advisable curve steepness can be defined as follows:
For a pressure increase from 100 to 107 %, the pump capacity should not decrease by more than 10 %.

b) The pump delivery head (p_p) will be:

$$p_p = p_{ei} - p_{st} + \Delta p + \frac{h}{10.2} \text{ [bar]}$$

where:
p_{ei} = pressure at engine inlet [bar]
p_{st} = static pressure at pump inlet [bar]
Δp = pressure losses [bar] between pump outlet and engine inlet
h = height difference [m] between pump outlet and engine inlet

The pressure (p_{ei}) related to liner top has to be between 4.0 bar and 5.0 bar.

4.2.2 Cooling water treatment

Correct treatment of the cooling freshwater is essential for safe engine operation. Only demineralised water or condensate according to the following specification must be used as raw water. In the event of an emergency, tap water may be used for a limited period, but afterwards the entire cylinder cooling water system is to be drained off, flushed, and recharged with demineralised water.

Recommended parameters for raw water:

min. pH:	6.5
max. dH:	10° dH (corresponds to 180 mg/l $CaCo_3$)[a]
max. chloride:	80 mg/l
max. sulphates:	150 mg/l

a) In case of higher values the water has to be softened.

Corrosion inhibitors

In addition, the water used must be treated with a suitable corrosion inhibitor to prevent corrosive attack, sludge formation and scale deposits. (For details refer to the chemical supply companies.) Monitoring the level of the corrosion inhibitor and water softness is essential to prevent down-times due to component failures resulting from corrosion or impaired heat transfer.

NOTE	No internally galvanised steel pipes should be used in connection with treated freshwater, since most corrosion inhibitors have a nitrite base. Nitrites attack the zinc lining of galvanised piping and create sludge.

For further information about permissible cooling water additives please refer to the document **Cooling water and additives**, which is provided on the WinGD corporate webpage under the following link:
Cooling water and additives

4.2.3 General recommendations for design

- The number of valves in the system is to be kept to a minimum to reduce the risk of incorrect setting.
- Valves are to be locked in the set position and labelled to eliminate incorrect handling.
- The possibility of manual interference with the cooling water flow in different branches of the cylinder cooling water system is to be avoided — not by adjusting the valves — but by installing and setting throttling discs at commissioning stage.
- Under normal operation of the cylinder cooling water system the pump delivery head and the total flow rate are to remain constant, even when the freshwater generator is started up or shut down.
- The cylinder cooling water system is to be totally separated from steam systems. Under no circumstances must there be any possibility of steam entering the cylinder cooling water system, e.g. via a freshwater generator.
- The installation of equipment affecting the controlled temperature of cylinder cooling water (CCW) is to be examined carefully before being added. Uncontrolled increases or decreases in CCW temperature may lead to thermal shock of engine components and scuffing of pistons. Thermal shock is to be avoided, and the temperature gradient of the cooling water when starting and shutting down additional equipment is not to exceed two degrees per minute (2 °C/min) at engine inlet.
- The design pressure and temperature of all the component parts such as pipes, valves, expansion tank, fittings, etc. are to meet the requirements of the classification society.

4.2.4 Freshwater generator

A freshwater generator, using heat from the cylinder cooling system to distil seawater, can be used to meet the demand for washing and potable water. The capacity of the freshwater generator is limited by the amount of heat available, which in turn is dependent on the service power rating of the engine.

Risk of thermal shock

It is crucial in the design stage to ensure that there are sufficient safeguards to protect the main engine from thermal shock when the freshwater generator is started.
To reduce such risk, it will be of advantage to use valves (for instance butterfly valves), which are linked and actuated with a large reduction ratio, at the freshwater generator inlet and in the bypass line.

See installation examples:

Figure 4-5, 4-12
Figure 4-6, 4-13

To remind engine room personnel of the possibility of thermal shocking if automatic start-up is overridden, WinGD recommends that freshwater generator valves (6) and (7) are operated by progressive servomotors and that the freshwater generator is labelled with a sign stating the following warning:

> Avoid thermal shock to your main engine! Freshwater generator inlet and outlet valves to be opened and closed slowly and progressively.

The bypass with valve (7) must have the same pressure drop as the freshwater generator. The valve must be open when the freshwater generator is not in operation and closed when the freshwater generator is operating. To avoid any wrong manipulation we recommend interlocking of valves (6) and (7).

Estimation for freshwater production

The quantity of freshwater produced by a single-effect vacuum (flash) evaporator can be estimated for guidance purposes as follows (in t/day):
$32 \cdot 10^{-3} \cdot Q_{FW}$
where Q_{FW} is the heat in kW available from the cylinder cooling water, estimated from the data in the *GTD* program.

- Figure 4-5, 4-12 shows alternative ‘A’, a freshwater generator system designed to use up to 50 % of available heat.
- Figure 4-6, 4-13 represents alternative ‘B’, a system laid out to use up to 85 % of available heat.

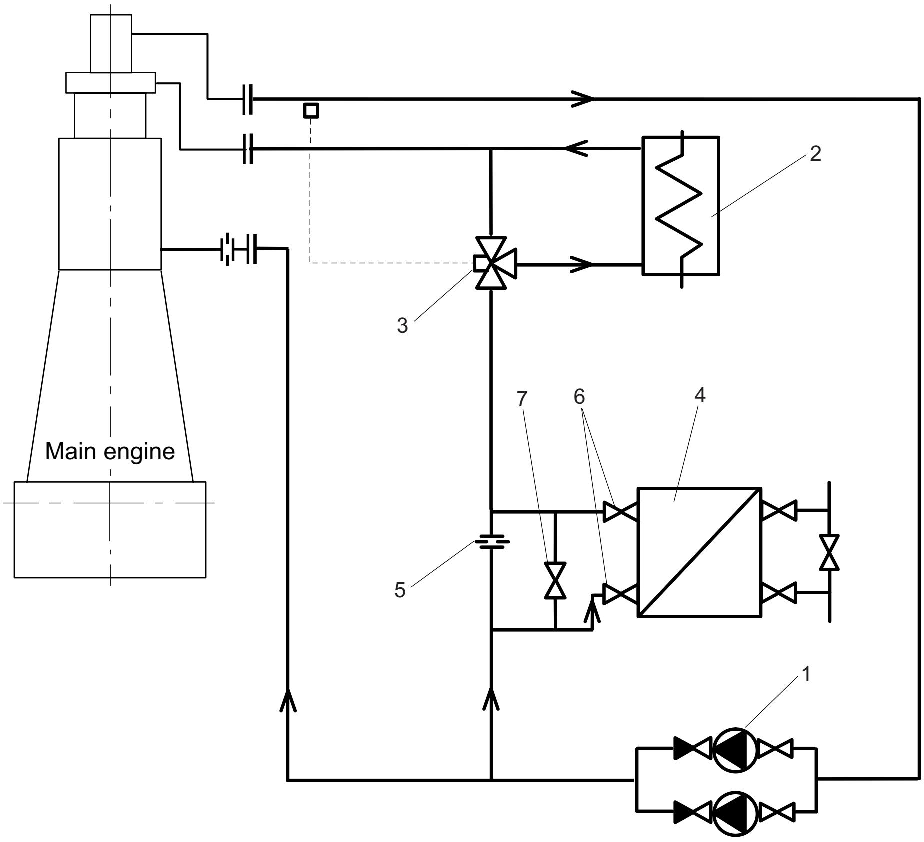
The latest version of the **Concept Guidance** for freshwater generator installation (DG 9721) is provided on the WinGD corporate webpage under the following link:
Freshwater generator installation

Alternative 'A'

Freshwater generators arranged as alternative 'A'

- can be connected in series (see Figure 4-5)
- have an evaporator heat requirement which does not exceed 50% of the heat available to be dissipated from the CCW at full load (CMCR)
- are only used at engine loads above 50%

Throttling disc (5) serves to correct the water flow rate if the pressure drop in the cooling circuit is less than that in the freshwater generator circuit. The throttling disc is to be adjusted so that, when the freshwater generator is started up or shut down, the pressure of CCW at engine inlet is maintained.



1 Cylinder cooling water pump
2 Cylinder cooling water cooler
3 Automatic temperature control valve
4 Freshwater generator
5 Throttling disc
6 Freshwater valves
7 Freshwater generator bypass valve

SM-0063

Figure 4-5 Freshwater generator installation — alternative 'A'

Example

8-cyl. engine — R1 with 21,280 kW at 97 rpm:
The available heat is approx. 2,075 kW (depending on engine configuration).
Alternative 'A' utilises up to 50% of the available heat, hence 1,038 kW of heat is available. Substitute this value in the equation:

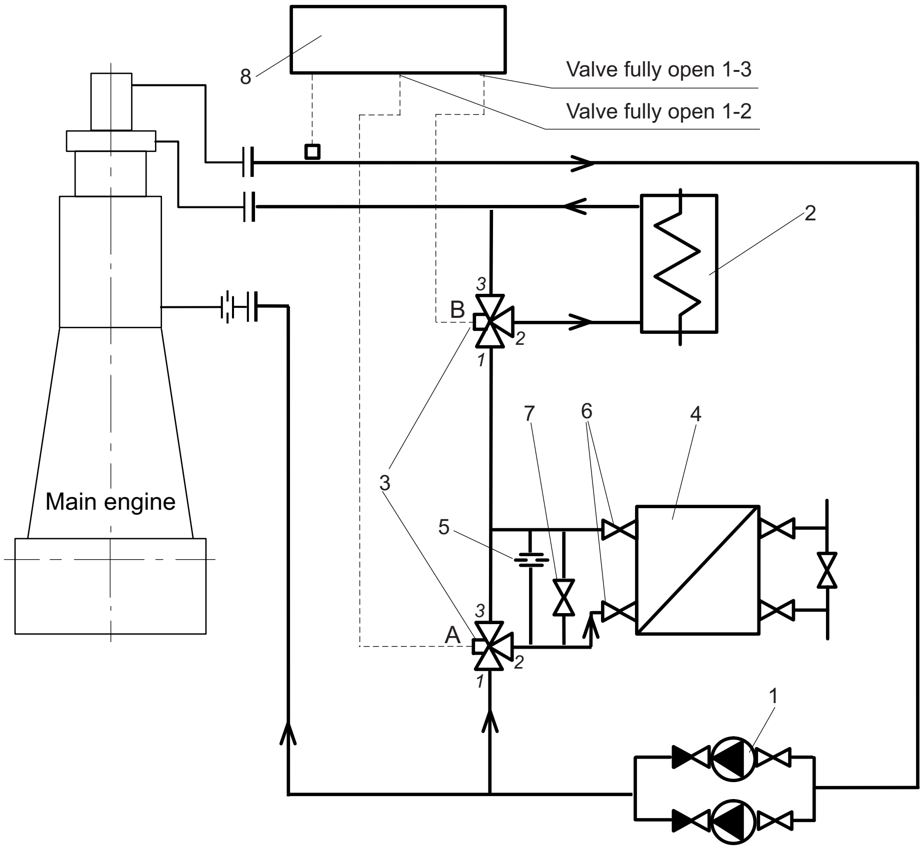
FW produced in t/day = constant $(32 \cdot 10^{-3})$ · available heat
$= 32 \cdot 10^{-3} \cdot 1{,}038$
= 33 (approx.)

Alternative ‘B’

Freshwater generators arranged as alternative ‘B’

- can be connected in series (see Figure 4-6)
- have an evaporator heat requirement which does not exceed 85% of the heat available to be dissipated from the CCW at full load (CMCR)

This arrangement requires an additional automatic temperature control valve (3A), connected in cascade control with the CCW cooler temperature control valve (3B), and controlled by step controller (8) sensing the outlet CCW temperature from the engine.



1 Cylinder cooling water pump
2 Cylinder cooling water cooler
3 Automatic temperature control valve
4 Freshwater generator
5 Throttling disc
6 Freshwater valves
7 Freshwater generator bypass valve
8 Controller
SM-0064

Figure 4-6 Freshwater generator installation — alternative ‘B’

Functional description

If the CCW outlet temperature is falling below the set-point, then valve (3A) reduces the flow of CCW to the freshwater generator for compensation. An amount of CCW is then routed directly to the engine inlet connection until the normal temperature is attained.
This means that the freshwater generator can be kept in continuous operation, although the generated freshwater volume decreases due to the reduced flow of hot water to the evaporator.

When the freshwater generator cannot dissipate all the heat in the CCW, valve (3A) is fully opened across connections 1 and 2, and a valve travel limit switch changes the regulation of CCW temperature over to temperature control valve (3B).

Valve (3B) in turn passes water to the CCW cooler (2) to maintain the CCW outlet at the required temperature. If in this condition the CCW temperature falls below the set-point and the cooler (2) is fully bypassed, then valve (3B) is fully opened across connections 1 and 3, and a valve travel limit switch transfers the regulation of CCW temperature back to temperature control valve (3A).

Two controllers possible

As an alternative to the single-step controller (8), two controllers can be installed, one for each valve, making sure that there is a 3 °C difference in the set-point between (3A) and (3B) to avoid both controllers acting at the same time.

Example

8-cyl. engine — R1 with 21,280 kW at 97 rpm:
The available heat is approx. 2,075 kW (depending on engine configuration).
Alternative ‘B’ utilises up to 85 % of the available heat, hence 1,764 kW of heat is available. Substitute this value in the equation:

FW produced in t/day = constant ($32 \cdot 10^{-3}$) · available heat
$= 32 \cdot 10^{-3} \cdot 1{,}764$
= 56 (approx.)

NOTE	The indicated values for evaporator heat requirement and load in alternatives ‘A’ and ‘B’ (i.e. 50 % and 85 % respectively) are only applicable if there are **no additional heat consumers** installed.

4.2.5 Pre-heating

To prevent corrosive liner wear when not in service or during short stays in port, it is important that the main engine is kept warm. Warming-through can be provided by a dedicated heater, using boiler raised steam or hot water from the diesel auxiliaries, or by direct circulation from the diesel auxiliaries.

Pre-heating from cooling water systems

If the requirement for warming-up is from the cooling water systems of the diesel auxiliaries, it is essential that the amount of heat available at normal load is sufficient to warm the main engine.
If the main and auxiliary engines have a cooling water system which can be cross-connected, it has to be ensured that, when the cross-connection is made, any pressure drop across the main engine does not affect the cooling water pressure required by the auxiliaries.
If the cooling water systems are apart, then a dedicated heat exchanger is required to transfer the heat to the main CCW system.

Pre-heating by direct water circulation

Use of main cylinder cooling water pump

If the main CCW pump is to be used to circulate water through the engine during pre-heating, the heater is to be arranged parallel with the CCW system, and on/off control is to be provided by a dedicated temperature sensor at the CCW outlet of the engine. The flow through the heater is set by throttling discs, but not by valves.

Use of separate pre-heating pump

If the requirement is for a separate pre-heating pump, a small unit with 10% of the main pump capacity and an additional non-return valve between the CCW pump and the heater are to be installed. In addition, the pumps are to be electrically interlocked to prevent two pumps running at the same time.

Recommended temperature

The recommended temperature to start and operate the engine is 60 °C at CCW outlet. If the engine has to be started below the recommended temperature, engine power is not to exceed 80% of CMCR until the water temperature has reached 60 °C.

The ambient engine room temperature and warm-up time are key parameters to estimate the heater power capacity required to achieve the target temperature of 60 °C. The shipyard or ship designer should determine the ambient engine room temperature and the warm-up time (which may also be specified by the shipowner) on the basis of their own experience.

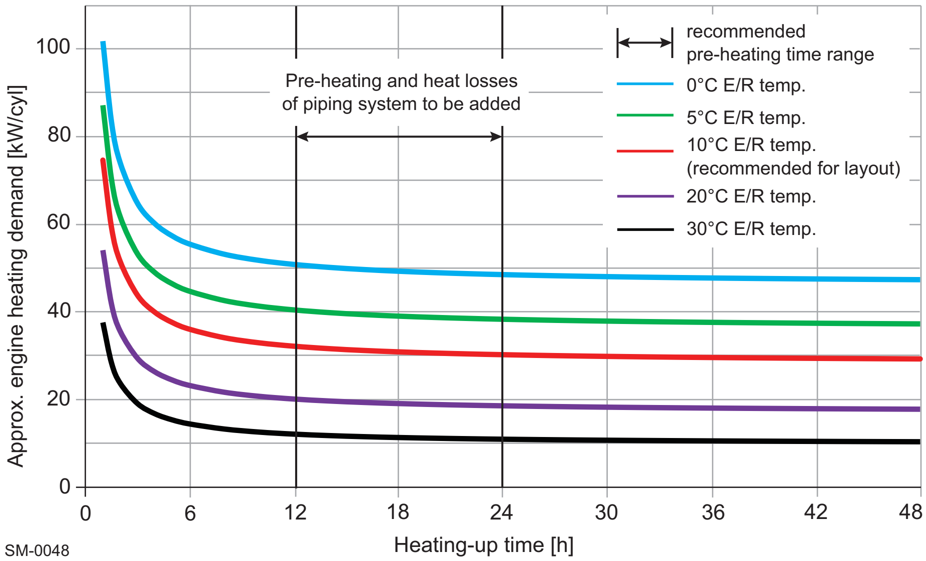


Figure 4-7 Pre-heating power requirement per cylinder

Warm-up time

The graph in Figure 4-7 shows the warm-up time needed in relation to the ambient engine room temperature to arrive at the heat amount required per cylinder. The graph covers the warming-up of engine components per cylinder, taking also the radiation heat into account.
The readable figure is then multiplied by the number of cylinders to show the heater capacity required for the engine.

All figures are related to requirements of the engine and should only be used for a first rough layout of the heater capacity. However, during pre-heater selection the shipyard or ship designer must also consider further aspects such as heat losses in the external piping system, water volume inside the system, pipe lengths, volume of ancillary equipment, etc.

4.3 Lubricating oil systems

The latest version of the **Marine Installation Drawing Set** relevant for the lubricating oil system (DG 9722) is provided on the WinGD corporate webpage under the following link:
MIDS

4.3.1 Lubricating oil requirements

The validated lubricating oils were selected in co-operation with the oil suppliers. In their respective product lines the products are considered as appropriate lubricants for the application indicated.
WinGD does not accept any liability for the quality of the supplied lubricating oil or its performance in actual service.

The validated cylinder and system oils are published in the document **Lubricants** provided on the WinGD corporate webpage under the following link:
Lubricants

4.3.2 Main lubricating oil system

Lubrication of the main bearings, thrust bearings, bottom-end bearings, crosshead bearings, together with piston cooling, is carried out by the main lubricating oil system. The main bearing oil is also used to cool the piston crown and to lubricate and cool the torsional and axial vibration dampers.

Figure 4-8 shows the general installation principle.

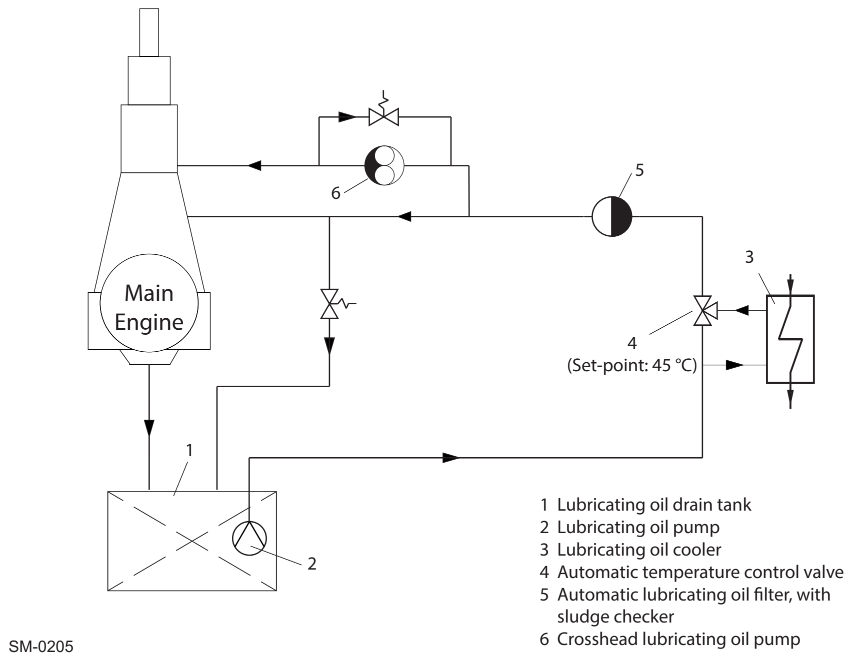


Figure 4-8 Scheme of lubricating oil system

Lubricating oil pump

Positive displacement screw pumps with built-in overpressure relief valves, or centrifugal pumps (for pump capacities refer to *GTD*):

Type: Positive displacement screw pump	The flow rate is to be within a tolerance of 0 to +10% of the GTD value, plus: - back-flushing flow of automatic filter, if any - torsional vibration damper, if any
Type: Centrifugal pump	The flow rate is to be within a tolerance of -10 to +10% of the GTD value, plus: - back-flushing flow of automatic filter, if any - torsional vibration damper, if any
Delivery head	The final delivery head to be determined is subject to the actual piping layout.
Working temperature	60 °C
Oil type	SAE30, 50 cSt at working temperature; when sizing the pump motor the maximum viscosity to be allowed for is 400 cSt.

Lubricating oil cooler

Type	Plate or tubular
Cooling medium	Freshwater
Cooling water flow	Refer to *GTD*.
Cooling water temperature	36 °C
Heat dissipation	Refer to *GTD*.
Margin for fouling	10-15 % to be added
Oil flow	Refer to *GTD*.
Oil viscosity at cooler inlet	50 cSt at 60°C
Oil temperature at inlet	Approx. 60 °C
Oil temperature at outlet	45 °C
Working pressure oil side	6 bar
Working pressure water side	Approx. 3 bar

Full-flow filter

The drain from the filter is to be sized and fitted to allow free flow into the lubricating oil drain tank.
The output required for the main lubricating oil pump to ‘back-flush’ the filter without interrupting the flow is to be taken into account when estimating the pump capacity (see Lubricating oil pump, 4-18).

Type[a]	Automatic back-flushing filter with differential pressure gauge and high-differential pressure alarm contacts. Designed to clean itself automatically using reverse flow or compressed air techniques. Back-flushing oil treatment by sludge checker.
Oil flow	Refer to *GTD*.
Working viscosity	95 cSt, at working temperature
Working pressure	6 bar
Test pressure	Specified by classification society
Diff. pressure, clean filter	Max. 0.2 bar
Diff. pressure, dirty filter	Max. 0.6 bar
Diff. pressure, alarm	Max. 0.8 bar Note: real operational settings could be less according to filter maker’s recommendation.
Mesh size	Sphere passing max. 0.035 mm
Filter material	Stainless steel mesh
Filter inserts bursting press.	Max. 3 bar differential across filter

a) Optional: change-over duplex filter designed for in-service cleaning, with differential pressure gauge and high-differential pressure alarm contacts

Booster pump for crosshead lubrication

Type	Positive displacement screw or gear types with built-in overpressure relief valves
Capacity	According to *GTD*: The flow rate is to be within a tolerance of 0 to 10 % of the GTD value.
Delivery head	Refer to *GTD*.
Working temperature	Approx. 45 °C
Oil type	SAE 30, 95 cSt at working temperature; when sizing the pump motor the maximum viscosity to be allowed for is 400 cSt.

System oil

For WinGD X6B engines designed with oil-cooled pistons, the crankcase oils used as system oil are specified as follows:

- SAE 30
- Minimum BN of 5.0 mg KOH/g and detergent properties
- Load carrying performance in FZG gear machine test method A/8, 3/90 according to ISO 14635-1, failure load stage 11 as a minimum[1)]
- Good thermal stability
- Antifoam properties
- Good demulsifying performance

The consumption of system oil is given in Table 1-1, 1-3.

4.3.3 Flushing the lubricating oil system

For flushing of the lubricating oil system refer to the latest version of the relevant **Instruction** (DG 9722), which is provided on the WinGD corporate webpage under the following link:
Instruction for flushing - Lubricating oil system

4.3.4 Lubrication for turbochargers

For lubricating oil for turbochargers equipped with separate lubricating oil systems the recommendations given by the supplier must be observed.

4.3.5 Cylinder lubricating oil system

Cylinder lubrication is carried out by a separate system, working with the once-through principle. A hydraulically actuated dosage pump feeds cylinder lubricating oil to the surface of the cylinder liner through quills in the liner. The oil supply rate is adjustable and metered to suit the age and running condition of piston rings and liners.

For cylinder lubricating oil consumption refer to Table 1-1, 1-3.

Cylinder oil

For normal operating conditions, a high-alkaline marine cylinder oil of SAE 50 viscosity grade with a minimum kinematic viscosity of 18.5 cSt (mm^2/s) at 100 °C is recommended. The alkalinity of the oil is indicated by its Base Number (BN)[2)].

Cylinder lubricants of intermediate BN (50 < BN < 60 mg/KOH/g) may be used if the performance is regularly monitored and the lubricating oil feed rate is adjusted to avoid a low piston underside BN. Residual BN which is too low can lead to excessive corrosive wear and scuffing.

1) The FZG gear machines located at the FZG Institute, Munich/Germany shall be the reference test apparatus and will be used in the event of any uncertainty about test repeatability and reproducibility.
2) The Base Number is expressed in mg KOH/g as determined by test method ASTMD 2896.

Recommended residual BN

The following values are recommended when operating on fuel with a sulphur content in the range of 0.5 to 3.5% m/m:

- The **safe** piston underside residual BN to avoid piston ring and liner corrosion is higher than **25 mg KOH/g** but lower than **50 mg KOH/g**
- The **alert** limit for piston underside residual BN to avoid excessive corrosion is between **10** and **25 mg KOH/g**
- The **danger** limit is less than **10 mg KOH/g** piston underside residual BN and is likely to lead to excessive corrosion and early piston ring and liner wear if not corrected. It often leads to scuffing, premature failure of piston rings and excessive corrosive liner wear.

Base number of cylinder lubricating oil

The base number (BN) of the cylinder lubricating oil must be selected depending on the total sulphur content of the fuel burnt. The higher the sulphur content in the fuel, the higher BN for cylinder lubricating oil is required.
Consequently, for low-sulphur fuel operation, low BN cylinder lubricating oil needs to be supplied, whereas high BN cylinder lubricating oil is required when the engine is running on HFO.

Alternatives to finished cylinder oils

The cylinder lubricating oil can also be blended/mixed on board. Multiple concepts for blending/mixing cylinder oil on board are available.

The validated additives and oils which can be used for this purpose can be found in the document **Lubricants**, which is provided on the WinGD corporate webpage under the following link:
Lubricants
For additional information please contact the oil supplier.

Another solution to have the needed BN value available is to mix lubricating oils of different BN values.

Service tank and storage tank

The arrangement of service tank and storage tank can be changed by locating the storage tank in place of the service tank. If this arrangement is preferred, the storage tank must be placed at the same height as the service tank to provide the necessary head. Furthermore, the storage tank must be of similar design, with a sloping floor.

Electrical trace heating for cylinder lubricating oil piping

The cylinder lubricating oil piping on ship side shall be electrically trace heated and insulated to ensure an oil temperature of approx. 40 °C at main engine inlet.
WinGD has introduced a trace heating cable and insulation for the ME internal cylinder LO piping and provided a power connection box on the engine. The shipyards can arrange the trace heating cable on the piping on ship side and connect the cable to the ME power connection box.
For details of the power connection box and trace heating cable please refer to the drawings of the relevant design group.

4.3.6 Maintenance and treatment of lubricating oil

It is essential that engine lubricating oil is kept as clean as possible. Water and solid contaminants held in suspension are to be removed using centrifugal separators which operate in bypass to the engine lubricating system.
Great care has to be taken of the separators and filters to ensure that they work correctly. The separators are to be set up as purifiers and completely isolated from the fuel oil treatment systems; there must be no possibility of cross-contamination.

Oil separator

Type	Self-cleaning centrifugal separator
Min. throughput capacity [l/h]	Refer to *GTD*.
Rated separator capacity	The rated or nominal capacity of the separator is to be according to the separator manufacturer's recommendations.
Separation temperature	90-95 °C; refer to manufacturer's instructions.

Oil samples

To ensure that representative samples of lubricating oil can be taken, dedicated sampling points (cocks) are provided on engine side. Such cocks need also to be installed on system side according to the relevant system proposal drawing in *MIDS*.

4.3.7 Drain tank

The engine is designed to operate with a dry sump: the oil returns from the bearings, flows to the bottom of the crankcase and through strainers into the lubricating oil drain tank. The drain connections from the crankcase to the drain tank are arranged vertically.

The drain tank is to be located beneath the engine and equipped with the following:

- Depth sounding pipe
- Pipe connections for lubricating oil purifiers
- Heating coil adjacent to pump suction
- Air vents with flame protection

NOTE	The classification societies require that all drain pipes from the crankcase to the drain tank are taken as low as possible below the free surface of the oil to prevent aeration and foaming; they have to remain below the oil surface at all times. **Strict attention has to be paid to this specification.**

There is to maintain adequate drainage under sea conditions resulting in pitching and rolling. The amount of lubricating oil required for an initial charge of the drain tank is indicated in Figure 4-9. The total tank size is normally 5-10% greater than the amount of lubricating oil required for an initial filling.

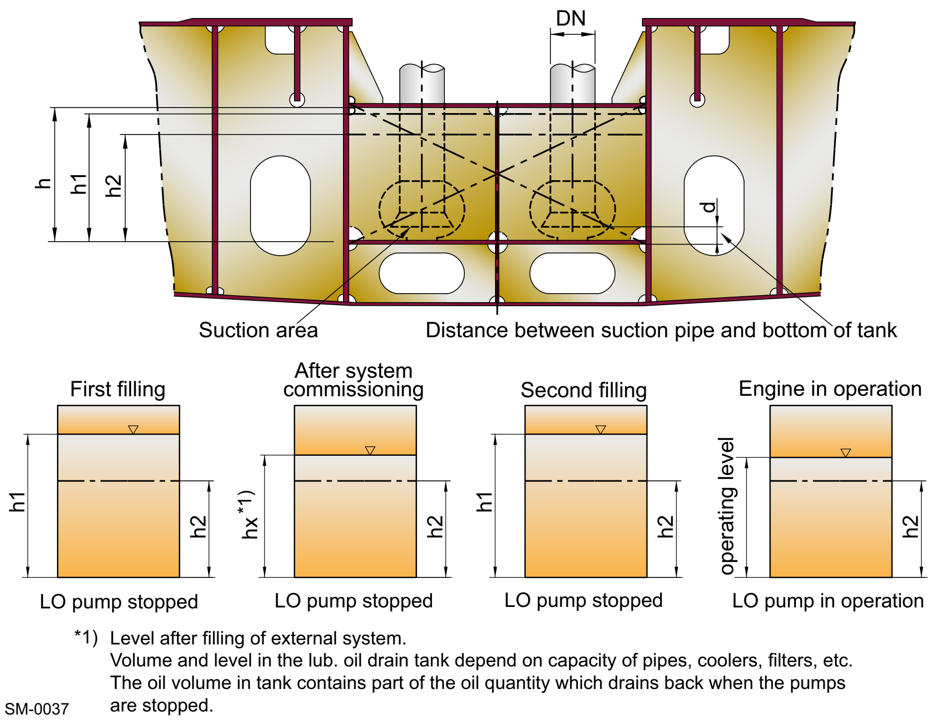


Figure 4-9 Dimensioning and filling process of lubricating oil drain tank

Arrangement of vertical lubricating oil drains

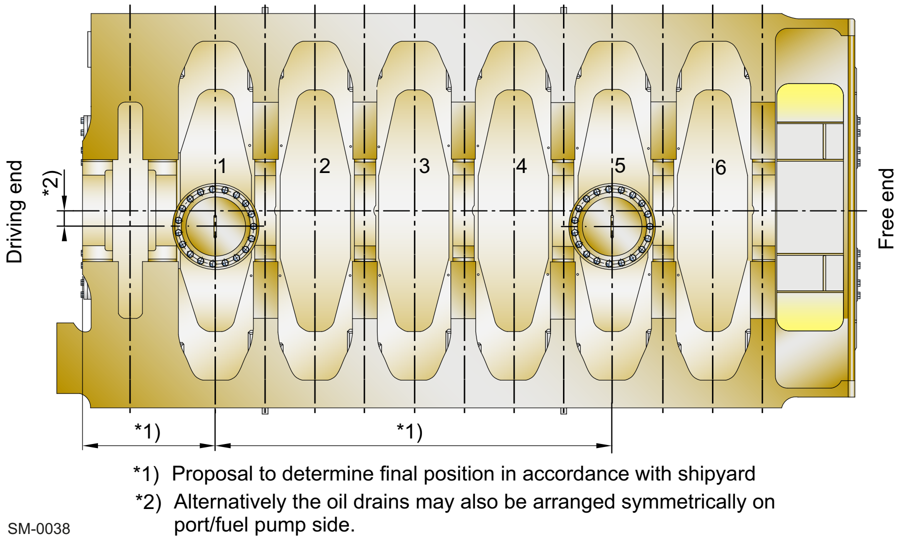
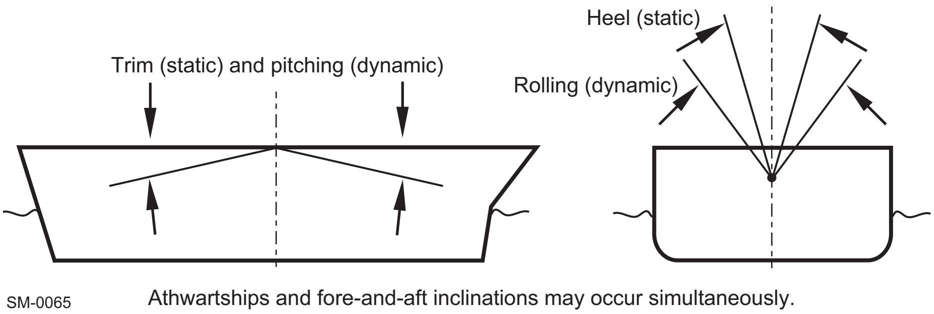


Figure 4-10 Arrangement of vertical lubricating oil drains for 6-cylinder engines

NOTE	The illustration above does not necessarily represent the actual configuration or the stage of development, nor the type of your engine. For all relevant and prevailing information see MIDS drawings, 4-17.

Inclination angles



NOTE	The data in the following tables represent the state of data as of the year 2018 and earlier. If you want to obtain the latest data please address yourself to the relevant classification society.

Table 4-4 Minimum inclination angles for full operability of the engine (1)

Classification societies (overview see Appendix, 9.1, 9-1)				
Year of latest update by Class	ABS 2018	BV 2017	CCS 2015	CRS 2018
Main and auxiliary engine				
Abbreviation	4/1/1/7.9	C/1/1/2.4	3/1/1/1.2.1	7/1/1.6/1.6.2
Heel to each side	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°
Trim by the head a)	5°	5°	5°	5°
Trim by the stern a)	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±7.5°
Emergency sets				
Abbreviation	4/1/1/7.9	C/1/1/2.4	3/1/1/1.2.1	7/1/1.6/1.6.2
Heel to each side	22.5° c)	22.5°	22.5° c)	22.5° c)
Rolling to each side	22.5° c)	22.5°	22.5° c)	22.5° c)
Trim	10°	10°	10°	10°
Pitching	±10°	±10°	±10°	±10°
Electrical installation				
Abbreviation	4/1/1/7.9	C/1/1/2.4	4/1/2/1.2.1	7/1/1.6/1.6.2
Heel to each side	22.5° b)	22.5° b) c)	15° c)	22.5° b)
Rolling to each side	22.5° b)	22.5° b) c)	22.5° c)	22.5° b)
Trim	10°	10° b)	5°	10° b)
Pitching	±10°	±10° b)	±7.5°	±10° b)

a)
Where the ship's length exceeds 100 m, the fore-and-aft static angle of inclination may be taken as 500/L degrees. (where L = length of ship in metres)

b)
Up to an inclination angle of 45 degrees, switches and controls are to remain in their last set position as no undesired switching operations or operational changes may occur.

c)
For ships carrying liquefied gases or chemicals the arrangement is to be such that the emergency power supply also remains operable with the ship flooded to a final athwartships inclination up to 30 degrees.

Table 4-5 Minimum inclination angles for full operability of the engine (2)

Classification societies (overview see Appendix, 9.1, 9-1)					
Year of latest update by Class	DNV 2016	DNV-GL 2018	GL 2016	IRS 2017	KR 2017
Main and auxiliary engine					
Abbreviation	4/1/3/B 200	4/1/3/2.2/2.2.1	I-1-2/1/C/C.1.1	4/1/1/1.7/1.7.1	5/1/103./1.
Heel to each side	15°	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°	22.5°
Trim by the head a)	5°	5°	5°	5°	5°
Trim by the stern a)	5°	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±7.5°	±7.5°
Emergency sets					
Abbreviation	4/1/3/B 200	4/1/3/2.2/2.2.1	I-1-2/1/C/C.1.1	4/1/1/1.7/1.7.1	5/1/103./1.
Heel to each side	22.5° c)	22.5° c)	22.5° c)	22.5° c)	22.5° c)
Rolling to each side	22.5° c)	22.5° c)	22.5° c)	22.5° c)	22.5° c)
Trim	10° a)	10° a)	10°	10°	10°
Pitching	±10°	±10°	±10°	±10°	±10°
Electrical installation					
Abbreviation	4/8/3/B 100	4/1/3/2.2/2.2.1	I-1-2/1/C/C.1.1	4/1/1/1.7/1.7.1	5/1/103./1.
Heel to each side	22.5° b) c)	22.5° b) c)	22.5° b) c)	22.5° b) c)	22.5° b) c)
Rolling to each side	22.5° b) c)	22.5° b) c)	22.5° b) c)	22.5° b) c)	22.5° b) c)
Trim	10° a) b)	10° a) b)	10° b)	10° b)	10° b)
Pitching	±10° b)	±10° b)	±10° b)	±10° b)	±10° b)

a)
Where the ship's length exceeds 100 m, the fore-and-aft static angle of inclination may be taken as 500/L degrees.
(where L = length of ship in metres)

b)
Up to an inclination angle of 45 degrees, switches and controls are to remain in their last set position as no undesired switching operations or operational changes may occur.

c)
For ships carrying liquefied gases or chemicals the arrangement is to be such that the emergency power supply also remains operable with the ship flooded to a final athwartships inclination up to 30 degrees.

Table 4-6 Minimum inclination angles for full operability of the engine (3)

Classification societies (overview see Appendix, 9.1, 9-1)					
Year of latest update by Class	LR 2017	NK 2017	PRS 2018	RINA 2018	RS 2018
Main and auxiliary engine					
Abbreviation	5/1/3/3.7	D/1.3.1/6	VI/1/1.6.1	C/1/1/2.4	VII/2/2.3
Heel to each side	15°	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°	22.5°
Trim by the head [a]	5°	5°	5°	5°	5°
Trim by the stern [a]	5°	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±7.5°	±7.5°
Emergency sets					
Abbreviation	5/1/3/3.7	D/1.3.1/6	VI/1/1.6.1	C/1/1/2.4	VII/2/2.3
Heel to each side	22.5° [c]	22.5° [b] [c]	22.5° [c]	22.5° [c]	22.5° [c]
Rolling to each side	22.5° [c]	22.5° [b] [c]	22.5° [c]	22.5° [c]	22.5° [c]
Trim	10°	10° [b]	10°	10°	10°
Pitching	±10°	±10° [b]	±10°	±10°	±10°
Electrical installation					
Abbreviation	6/2/1/1.10	H/1/1.1.7	VIII/2/2.1.2.2	C/2/2/1.6	XI/2/2.1.2.2
Heel to each side	15°	15° [c]	15°	22.5° [b]	15° [c]
Rolling to each side	22.5°	22.5° [c]	22.5°	22.5° [b]	22.5° [c]
Trim	5° [a]	5° [a]	5°	10° [b]	5° [c]
Pitching	±7.5°	±7.5°	±10°	±10° [b]	±10° [c]

a)
Where the ship's length exceeds 100 m, the fore-and-aft static angle of inclination may be taken as 500/L degrees. (where L = length of ship in metres)

b)
Up to an inclination angle of 45 degrees, switches and controls are to remain in their last set position as no undesired switching operations or operational changes may occur.

c)
For ships carrying liquefied gases or chemicals the arrangement is to be such that the emergency power supply also remains operable with the ship flooded to a final athwartships inclination up to 30 degrees.

4.4 Fuel oil system

The latest version of the **Marine Installation Drawing Set** relevant for the fuel oil system (DG 9723) is provided on the WinGD corporate webpage under the following link:
MIDS

Figure 4-11 shows the general installation principle.

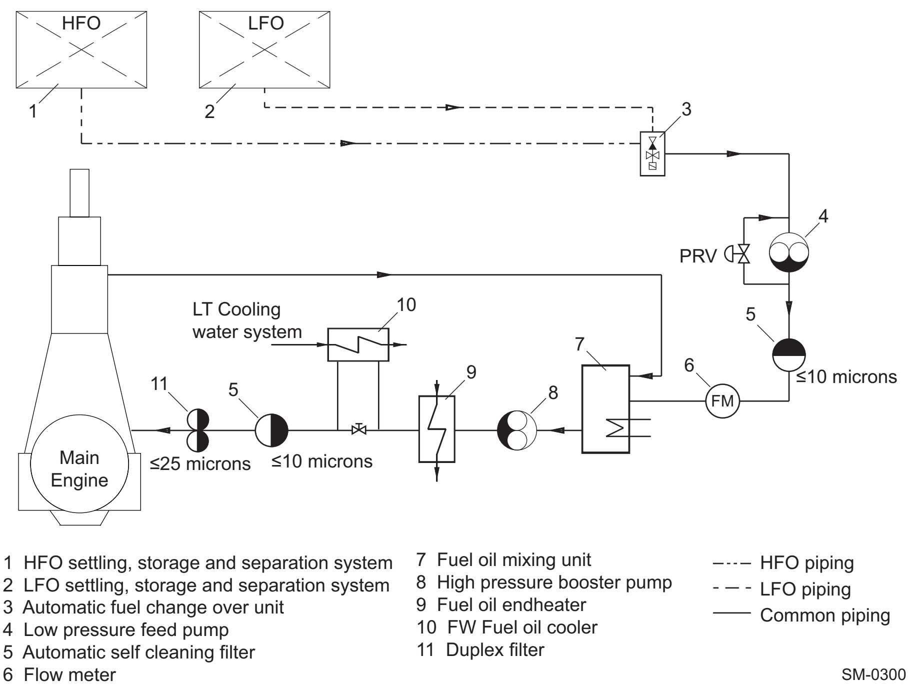


Figure 4-11 Scheme of fuel oil system

4.4.1 Fuel oil system components

Fuel oil feed pump

Type	Positive displacement screw pump with built-in overpressure relief valve
Capacity	According to *GTD*: The capacity is to be within a tolerance of 0 to +20% of the GTD value, plus back-flushing flow of automatic filter, if any.
Delivery pressure	The delivery pressure is to take into account the system pressure drop and prevent entrained water from flashing off into steam by ensuring that the pressure in the mixing unit is at least 1 bar above the water vapour pressure, and no lower than 3 bar. The water vapour pressure is a result of the system temperature and pressure for a given fuel type. Heavier oils need more heat and higher temperatures to maintain them at the correct viscosity than lighter oils. (Refer to the formula and example below.)
Electric motor	The electric motor driving the fuel oil feed pump shall be sized large enough for the power absorbed by the pump at maximum pressure head (difference between inlet and outlet pressure), maximum fuel oil viscosity (700 cSt), and the required flow.
Working temp.	Up to 90 °C
Fuel type	Marine diesel oil and heavy fuel oil, up to 700 cSt at 50 °C

Formula for delivery gauge pressure

$p_v + 1 + \Delta p_1 + \Delta p_2$ [bar]

where:

p_v .... = water vapour gauge pressure at the required system temp. [bar] (see viscosity-temperature diagram in section 4.4.7, 4-42)

Δp_1 .... = max. pressure losses between feed pumps and mixing unit [bar]

Δp_2 .... = max. pressure change difference across the pressure regulating valve of the feed system between min. and max. flow (see Pressure regulating valve, 4-30)

Example

HFO of 700 cSt at 50 °C, required system temperature 145 °C:

p_v .... = 3.2 bar

Δp_1 .... = 0.5 bar

Δp_2 .... = 0.6 bar

Delivery gauge pressure = 3.2 + 1 + 0.5 + 0.6 = **5.3 bar**

Pressure regulating valve

To prevent entrained water from flashing off into steam, the pressure regulating valve controls the delivery of the low-pressure feed pump by returning excessive supply back to the pump's suction side, ensuring that the discharge pressure is 1 bar above the evaporation pressure of water. To avoid heating-up of the fuel by recirculation, the return pipe is designed with cooling ribs.

The pressure regulating valve should have a flat steady-state characteristic across the fuel oil recirculation flow range.

Type	Self- or pilot-operated which senses the upstream pressure to be maintained through an external line. It is to be pneumatically or direct hydraulically actuated with an additional manual control for emergency operation. When using a pneumatic type, use a combined spring type to close the valve in case of air supply failure.
Maximum capacity	According to *GTD*: Refer to feed pump capacity.
Minimum capacity	Approx. 20 % of that of the fuel oil feed pump
Service pressure	Max. 10 bar
Pressure setting range	2-6 bar
Inlet pressure change	The inlet pressure may vary by up to 0.8 bar depending on the flow in the range of 20 % to 100 %.
Working temperature	Up to 90 °C
Fuel oil viscosity	100 cSt, at working temperature (HFO 700 cSt at 50 °C)

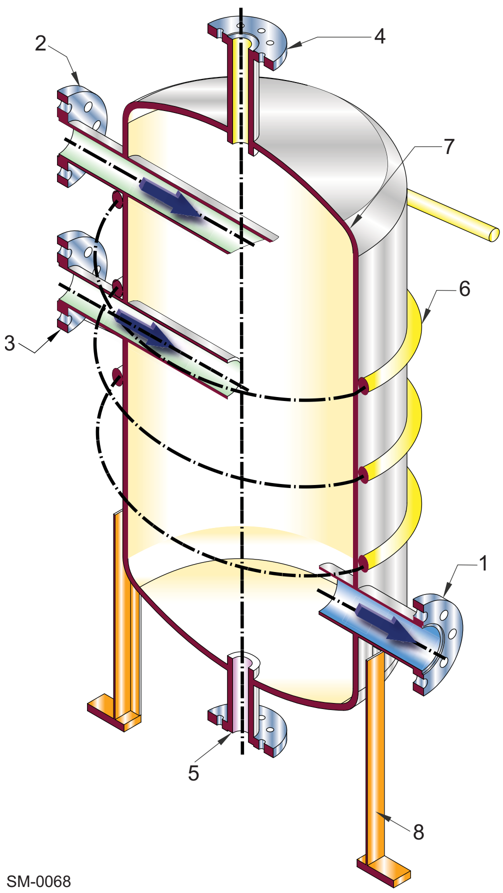
Mixing unit

The mixing unit equalises the temperature between the hotter fuel oil returning from the engine and the colder fuel oil from the service tank, particularly when changing over from HFO to MDO and vice versa.

Due to the small amount of fuel consumed, especially in part-load operation, only a small mixing unit is required. It is recommended that the tank contains no more than approx. 100 litres. This is to avoid the changeover from HFO to MDO or vice versa taking too long.

For changing over between heavy fuel oil and marine diesel oil (MGO/MDO) and vice versa, as well as for operation on distillate fuel, refer to the separate **Concept Guidance** (DG 9723), which is provided on the WinGD corporate webpage under the following link:
Operation on distillate fuels

Type	Cylindrical steel fabricated pressure vessel as shown in Figure 4-12
Capacity	Refer to *GTD*.
Dimensions	See *MIDS*.
Service pressure	10 bar
Test pressure	According to classification society
Working temperature	Up to 150 °C



1 Outlet
2 Inlet, return pipe
3 Inlet from feed pump
4 Vent
5 Drain
6 Heating coil
7 Insulation
8 Mounting brackets

Figure 4-12 Mixing unit

Fuel oil booster pump

The fuel oil booster pump delivers the fuel to the engine via a fuel oil end heater for HFO operation.

Type	Positive displacement screw pump with built-in overpressure relief valve
Capacity	According to *GTD*: The flow rate is to be within a tolerance of 0 to +20 % of the GTD value, plus back-flushing flow of automatic filter, if any.
Inlet pressure	Up to 6 bar
Delivery head	Final delivery pressure according to actual piping layout. Refer to *GTD*.
Electric motor	The electric motor driving the HP booster pump shall be sized large enough for the power absorbed by the pump at maximum pressure head (difference between inlet and outlet pressure), maximum fuel oil viscosity (600 cSt), and the required flow.
Working temperature	Up to 150 °C

End heater

Operates either temperature- or fuel oil viscosity controlled (default mode). The viscosity is measured by the viscosimeter.

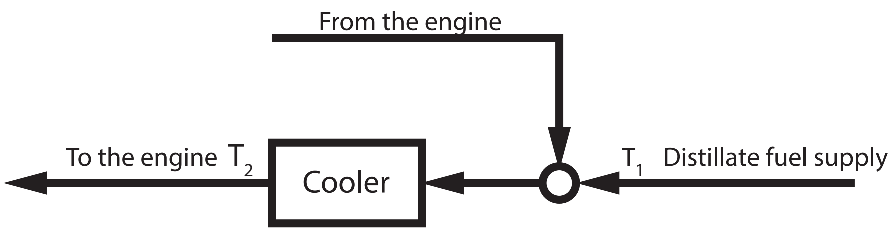
Type	Tubular- or plate type heat exchanger, suitable for heavy oils up to 700 cSt at 50 °C
Heating source	Steam, electricity, or thermal oil
Consumption of saturated steam	At 7 bar gauge pressure [kg/h]: $1.32 \cdot 10^{-6} \cdot CMCR \cdot BSFC \cdot (T_1 - T_2)$ where: — *BSFC* = brake specific fuel consumption at contracted maximum continuous rating (*CMCR*) — T_1 = temperature of fuel oil at viscosimeter a) — T_2 = temperature of fuel oil from service tank
Heating capacity [kW]	$0.75 \cdot 10^{-6} \cdot CMCR \cdot BSFC \cdot (T_1 - T_2)$
Working pressure	Max. 12 bar, pulsating on fuel oil side
Working temperature	Up to 150 °C, outlet temperature on fuel oil side

a) The viscosity is maintained by regulating the fuel temperature after the end heater in that the viscosimeter monitors the fuel viscosity before the supply unit and transmits the signals to the heater controls.

Diesel oil cooler

For diesel oil operation the fuel might need to be cooled to keep a minimum viscosity of 2 cSt at engine inlet. A chiller unit is not required if the fuel properties are in line with the latest ISO 8217 specification; such a unit would only be needed for off-spec fuels that are not supported by WinGD.

Type	Tubular- or plate type heat exchanger, suitable for diesel oils
Cooling medium	LT cooling water Alternatively: glycol-water mixture delivered from chiller unit
Cooling capacity [kW]	$Q = \frac{0.34 \cdot BSFC \cdot P \cdot (T_1 - T_2 + 25.65)}{10^6}$ where: Q [kW] = cooler heat dissipation at 100 % engine load $BSFC$ [g/kWh] = specific fuel consumption at design conditions and 100 % engine load P [kW] = engine power at 100 % CMCR T_1 [°C] = temp. of distillate fuel supplied to engine T_2 [°C] = temp. of distillate fuel required at engine inlet
Working pressure	Max. 12 bar, pulsating on fuel oil side



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Fuel oil filter

Two arrangements for the fuel oil filters can be applied in the fuel oil system, including:

- Arrangement **‘A’**: (see Figure 4-13, 🗎 4-35)
  - a maximum 10 micron fine filter installed in either the feed ‘cold’ system or booster ‘hot’ system
  - a second, manually cleaned duplex filter of recommended maximum 25 micron installed upstream of the engine inlet booster system
- Arrangement **‘B’**: (see Figure 4-14, 🗎 4-38)
  - a maximum 10 micron fine filter installed in the booster ‘hot’ system

NOTE	WinGD recommends arrangement **‘A’**.

Arrangement ‘A’ (recommended)

A manually cleaned 25 micron (absolute sphere passing mesh size) duplex filter is installed in the booster system close to engine inlet. This arrangement is a best practice recommendation. However, a coarser filter is acceptable (arrangement ‘B’ does not include secondary duplex filtration and lacks the indication of fuel oil treatment system overall performance).
A duplex filter is sufficient, as most particles are already removed by the fine filter outlined in option 1 or option 2 below.

Table 4-7 Specification of duplex filter in booster system

Working viscosity	10-20 cSt required for HFO (13-17 cSt recommended)
Flow rate	According to *GTD*. The capacities cover the needs of the engine only. If a filter of automatic back-flushing type is installed, the feed and booster pump capacities must be increased by the quantity needed for back-flushing of the filter.
Service pressure	Max. 12 bar at filter inlet
Test pressure	Specified by classification society
Permitted differential press. at 17 and 20 cSt	— clean filter: max. 0.2 bar — dirty filter: max. 0.6 bar — alarm setting: max. 0.8 bar Note: real operational settings could be less according to filter maker’s recommendation.
Minimum bursting press. of filter insert	Max. 3 bar differential across filter
Mesh size	Recommended max. 25 micron (absolute sphere passing mesh)
Filter insert material	Stainless steel mesh (CrNiMo)
Working temperature	Up to 150 °C

The maximum 10 micron fine filter can be installed in two locations:

- Option 1:feed system
- Option 2:booster system

The filter is used to protect the engine against serious damage. It captures the catalytic fines which were not removed by the fuel oil separator. In addition, the filter provides a good indication of the separator's efficiency.

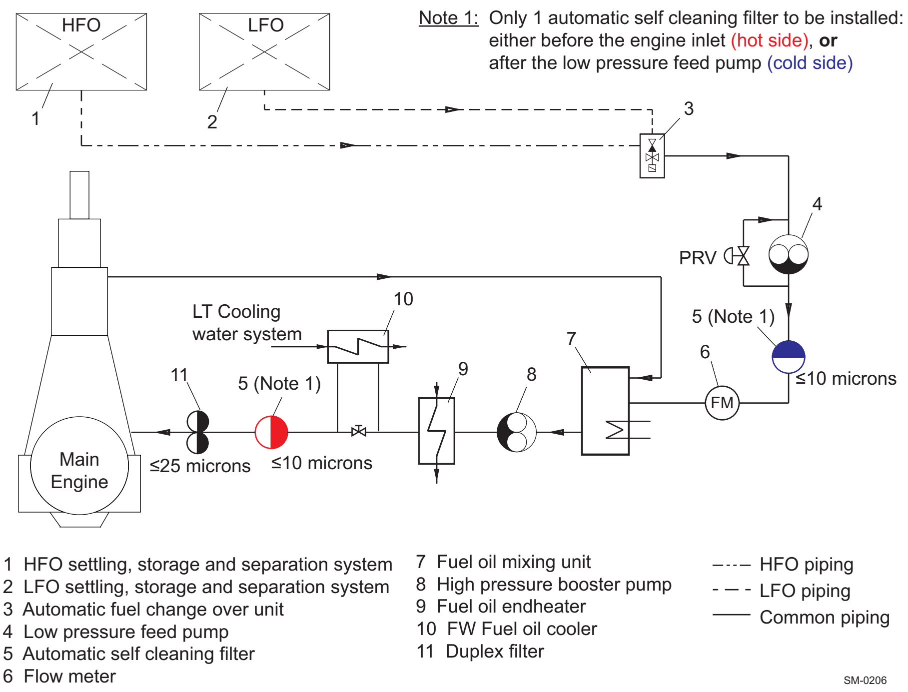


Figure 4-13 Fuel oil filter arrangement 'A'

NOTE	Under consideration of the filter fineness an automatic filter with good self-cleaning performance must be selected.

Option 1

10 micron fine filter in feed line:
The maximum 10 micron (absolute sphere passing mesh size) fine filter is installed in the 'cold' feed system. In this position the filter can be designed for a lower flow rate compared to the booster system. However, higher resistance due to higher fuel viscosity needs to be considered.

This filter position has the following advantage and disadvantage:

Advantage	Booster pump is protected against abrasive catfines
Disadvantage	Engine is not optimally protected against booster pump wear particles

Table 4-8 Specification of automatic filter in feed system

Working viscosity	100 cSt, for HFO of 700 cSt at 50 °C
Flow rate	According to *GTD*. The capacities cover the needs of the engine only. The feed pump capacity must be increased by the quantity needed for back-flushing of the filter.
Service pressure after feed pumps	10 bar at filter inlet
Test pressure	Specified by classification society
Permitted differential press. at 100 cSt	— clean filter: max. 0.2 bar — dirty filter: max. 0.6 bar — alarm setting: max. 0.8 bar Note: real operational settings could be less according to filter maker's recommendation.
Minimum bursting press. of filter insert	Max. 3 bar differential across filter
Mesh size	Max. 10 micron absolute (sphere passing mesh)
Mesh size bypass filter	Max. 25 micron absolute (sphere passing mesh)
Filter insert material	Stainless steel mesh (CrNiMo)

Option 2

10 micron fine filter in the booster circuit:
The maximum 10 micron (absolute sphere passing mesh size) fine filter is installed in the booster circuit close to engine inlet. The filter needs to be laid out for a maximum working temperature of 150 °C.

This filter position has the following advantage and disadvantage:

Advantage	Optimum engine protection from fuel oil catfines and other abrasive particles from system wear
Disadvantage	Booster pump is not ideally protected against catfines

Table 4-9 Specification of automatic filter in booster system

Working viscosity	10-20 cSt required for HFO (13-17 cSt recommended)
Flow rate	According to *GTD*. The capacities cover the needs of the engine only. If a filter of automatic back-flushing type is installed, the feed and booster pump capacities must be increased by the quantity needed for back-flushing of the filter.
Service pressure	Max. 12 bar at filter inlet
Test pressure	Specified by classification society
Permitted differential press. at 17 and 20 cSt	— clean filter: max. 0.2 bar — dirty filter: max. 0.6 bar — alarm setting: max. 0.8 bar Note: real operational settings could be less according to filter maker's recommendation.
Minimum bursting press. of filter insert	Max. 3 bar differential across filter
Mesh size	Max. 10 micron absolute (sphere passing mesh)
Mesh size bypass filter	Max. 25 micron absolute (sphere passing mesh)
Filter insert material	Stainless steel mesh (CrNiMo)
Working temperature	Up to 150 °C

Arrangement 'B'

The 10 micron (absolute sphere passing mesh size) fine filter is installed in the booster circuit close to engine inlet. The filter needs to be laid out for a maximum working temperature of 150 °C. With this arrangement, no indication is available if the automatic filter fails.

NOTE	Under consideration of the filter fineness an automatic filter with good self-cleaning performance must be selected.

Same filter specification as provided by Table 4-9.

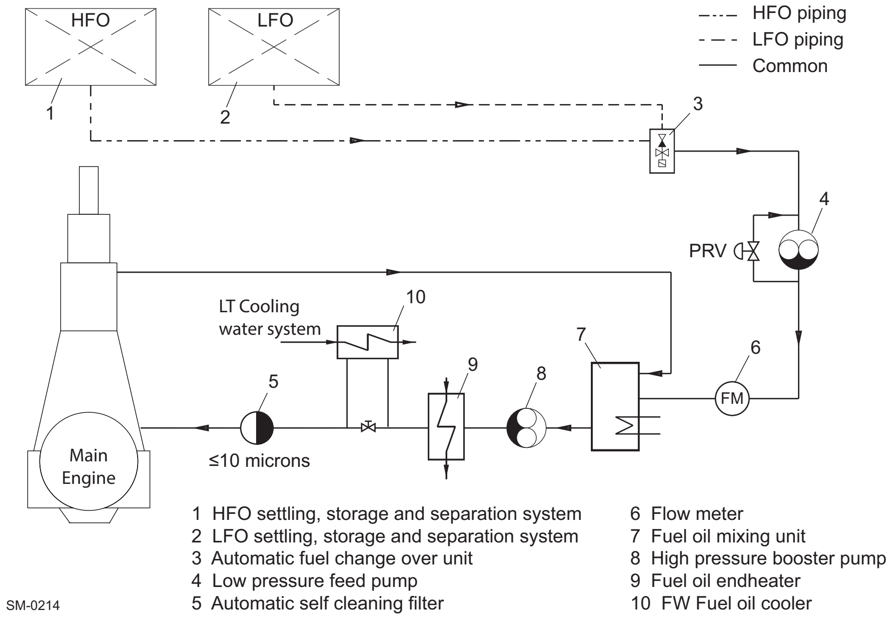


Figure 4-14 Fuel oil filter arrangement ‘B’

4.4.2 Fuel oil system components for installations without HFO

The layout of the system without HFO is defined project-specifically. Significant system simplifications are possible. Please consult WinGD via its licensee.

4.4.3 Flushing the fuel oil system

For flushing of the fuel oil system refer to the latest version of the relevant **Instruction** (DG 9723), which is provided on the WinGD corporate webpage under the following link:
Instruction for flushing - Fuel oil system

4.4.4 Fuel oil treatment

The latest version of the **Concept Guidance** for fuel oil treatment (DG 9723) is provided on the WinGD corporate webpage under the following link:
Fuel oil treatment

Settling tanks

Gravitational settling of water and sediment from modern heavy fuel oils is an extremely slow process due to the small difference in densities. The settling process is a function of the fuel surface area of the tank to the viscosity, temperature and density difference. Heated large-surface area tanks enable better separation than heated small-surface area tanks.

Service tanks

Diesel oil service tanks are similar to heavy oil service tanks, with the possible exception of tank heating, although this may be incorporated for vessels constantly trading in cold climates.
Most of the service tank design features are similar to those of settling tanks, comprising a self-closing sludge cock, level monitoring device and remote closing discharge valves to the separator(s) and engine systems. The service tank is to be equipped with a drain valve arrangement at its lowest point, an overflow to the overflow tank, and recirculating pipework to the settling tank.

Water in fuel

Due to condensation or coil leakage, water may be present in the fuel after the separators. The recirculation pipe, which reaches to the lower part of the service tank, leads the water into the settling tank. A pipe to the separators should be provided to re-clean the fuel in case of dirty water contamination. This line should be connected just above the drain valve at the service tank bottom.

Cleaning of fuel

The fuel is cleaned either from the settling tank to the service tank or recirculating the service tank. Ideally, when the main engine is operating at CMCR, the fuel oil separator(s) should be able to maintain a flow from the settling tank to the service tank with a continual overflow back to the settling tank. The sludge cock is to be operated at regular intervals to observe the presence of water, a significant indication for the condition of the separator(s) and heating coils.

Centrifugal fuel oil separators

There are two types of oil separators:

- Type 1 — Separators with gravity discs
- Type 2 — Separators without gravity discs

NOTE	Separators with gravity discs represent outdated technology and are therefore not supported by WinGD.

Separators without gravity discs

These separators are self-adjusting to the fuel properties and self-cleaning. Separators without gravity discs operate as combined purifiers-clarifiers; thus water and sediment separation is integrated in one unit. The manufacturers claim extended periods between overhaul. Compared to the outdated separators with gravity discs the reliability is greatly improved, enabling unattended onboard operation. As it is usual to install a standby separator as a back-up, it is of advantage to use both units in parallel to improve the separation result.

For further details and information regarding the separators please refer to the manufacturer's instructions.

Separation efficiency

The separation efficiency is a measure of the separator's capability to remove specified test particles. The separation efficiency is defined as follows:

$$n = 100 \cdot \left(1 - \frac{C_{out}}{C_{in}}\right)$$

where:

n .... = separation efficiency [%]

C_{out} .... = number of test particles in cleaned test oil

C_{in} .... = number of test particles in test oil before separator

Certified Flow Rate

To express the performance of separators according to a common standard, the term Certified Flow Rate (CFR) has been introduced. CFR is defined as the flow rate in litres/hour, 30 minutes after sludge discharge, at which the separation efficiency is 85% when using defined test oils and test particles. CFR is defined for equivalent fuel oil viscosities of 380 and 700 cSt at 50 °C.
More information can be found in the CEN document (European Committee for Standardization) CWA 15375:2005 (E).

Throughput capacity

The required minimum effective throughput capacity (litres/hour) of the separators is determined by the formula $1.2 \cdot CMCR \cdot BSFC \cdot 10^{-3}$ [litres/hour] as shown in the following example. The nominal separator capacity and the installation are to comply with the recommendations of the separator manufacturer.
(The MDO separator capacity can be estimated using the same formula.)

Example

- 8-cyl. engine
- CMCR / R1+: 21,280 kW
- BSFC / R1+: 166.0 g/kWh
- **Throughput:** $1.2 \cdot 21{,}280 \cdot 166.0 \cdot 10^{-3}$ = **4,239 litres/hour**

Oil samples

To ensure that representative samples of fuel oil can be taken, dedicated sampling points (cocks) are provided on engine side. Such cocks need also to be installed on system side according to the relevant system proposal drawing in *MIDS*.

4.4.5 Pressurised fuel oil system

The fuel is supplied from the heated heavy fuel oil service tank or the unheated diesel oil service tank to the low-pressure feed system.

Fuel changeover

For changing over from one fuel type to the other it was common to have a simple, manually operated three-way valve. This arrangement is not recommended any longer, as with the introduction of different Emission Control Areas (ECA), fuel changeover is quite frequently required, even at high engine load.
(In the past it was needed in rare cases only, for instance due to maintenance or before stopping the engine, i.e. at relatively low loads.)

Automatic changeover unit

Consequently, a well proven automatic changeover unit is nowadays recommended, which ensures:

- A maximum temperature gradient of 2 K/min during changeover
- A maximum viscosity of 20 cSt
- A minimum viscosity of 2 cSt; this minimum limit is most challenging during changeover from HFO to distillate fuel.
  Attention: not all changeover units guarantee keeping the minimum viscosity limit, as viscosity is not controlled.
- A best-practice automatic control of diesel oil cooler activation

4.4.6 Fuel oil specification

The validated fuel oil qualities are published in the document **Diesel engine fuels** provided on the WinGD Corporate Webpage under the following link:
Fuel qualities

4.4.7 Fuel oil viscosity-temperature dependency

The fuel oil viscosity depends on its temperature. This dependency is shown as graph in Figure 4-15.

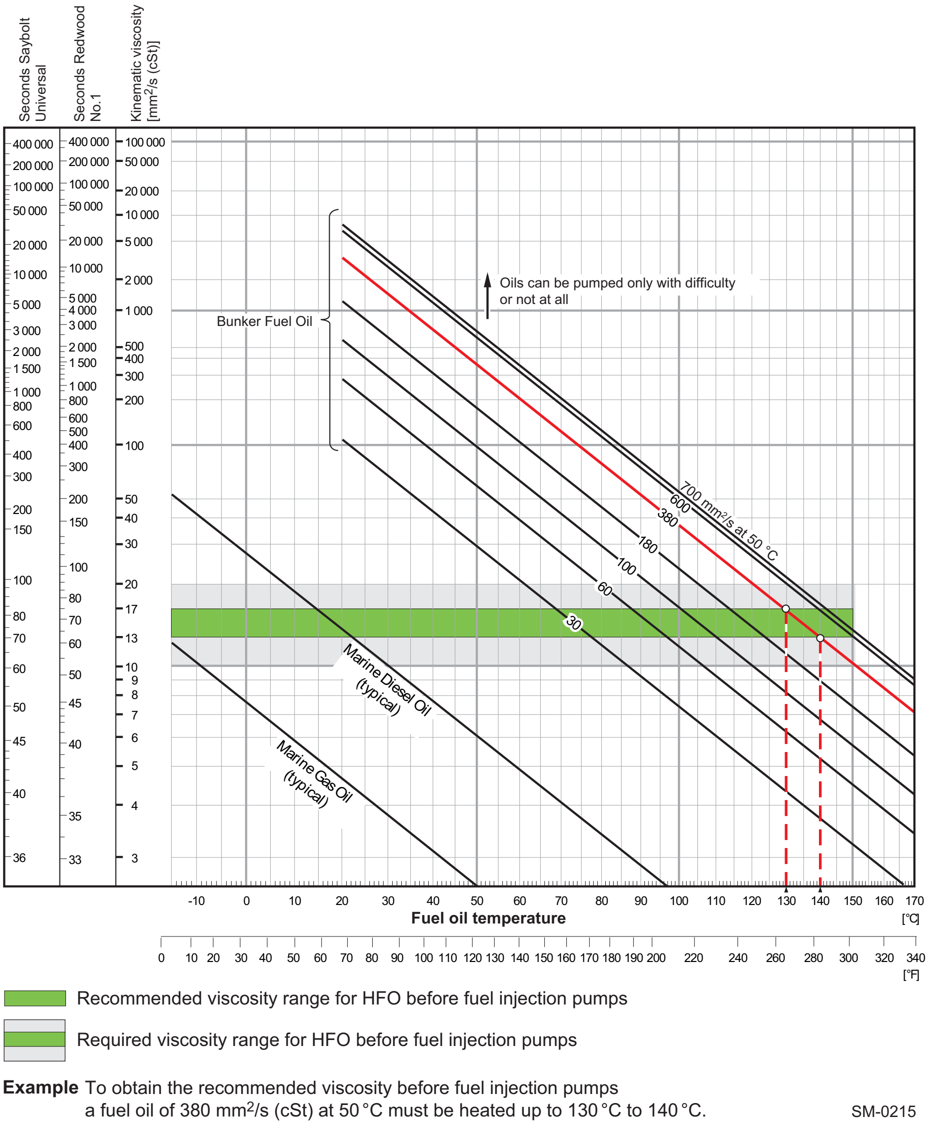


Figure 4-15 Fuel oil viscosity-temperature diagram

4.5 Starting and control air system

The latest version of the **Marine Installation Drawing Set** relevant for the starting air system (DG 9725) is provided on the WinGD corporate webpage under the following link:
MIDS

Compressed air is required for engine starting and control, exhaust valve air springs, the washing plant for scavenge air coolers, and general services.

The starting and control air system shown in Figure 4-16 comprises two air compressors, two air receivers, and systems of pipework and valves connected to the engine starting air manifold.

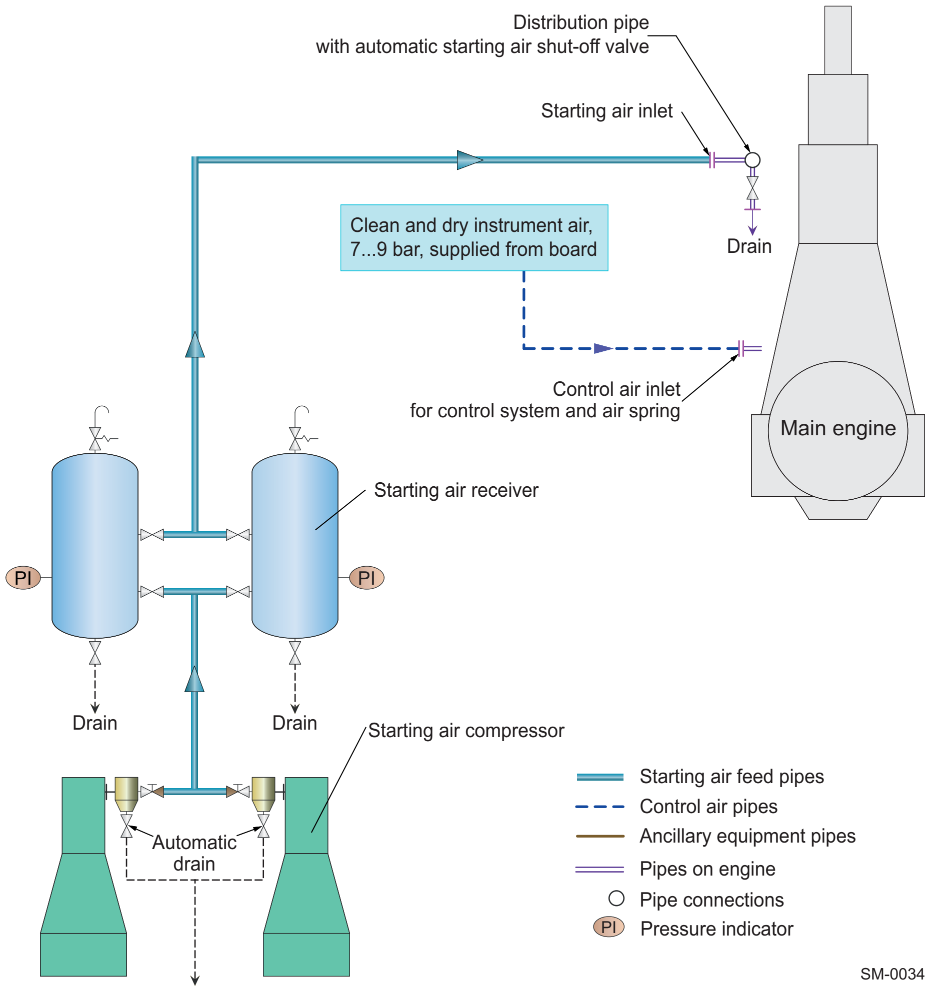


Figure 4-16 Starting and control air system

4.5.1 Capacities of air compressor and receiver

The capacity of the air compressor and receiver depends on the total inertia (J_{tot}) of the propulsion system's rotating parts.

- Total inertia = engine inertia + shafting and propeller inertia [1]:
  $J_{tot} = J_{eng} + J_{S+P}$
- Engine inertia (J_{eng}): refer to *GTD* [2]
- Relative inertia:
  $J_{rel} = \frac{J_{tot}}{J_{eng}}$

4.5.2 System specification

Starting air compressors

Capacity	Refer to *GTD*.
Delivery gauge pressure	30 bar

The discharge air temperature must not exceed 90 °C and the air supply to the compressors is to be as clean as possible without oil vapour.

Starting air receivers

Type	Fabricated steel pressure vessels with domed ends and integrated pipe fittings for isolating valves, automatic drain valves, pressure reading instruments and pressure relief valves
Capacity	Refer to *GTD*.
Working gauge pressure	30 bar

1) Propeller inertia includes the part of entrained water.
2) The *GTD* application enables to optimise the capacities of compressors and air receivers for CMCR.

4.5.3 Control air

Control air system supply

Control air is supplied from the board instrument air supply system (see Figure 4-16, 4-43) providing air at 8 bar gauge pressure (within a range of 7.0-9.0 bar). The air quality should comply with the compressed air purity class **2-4-2** according to ISO 8573-1 (2010-04-15).

Control air consumption

With the development of engine technology the WinGD RT-flex and X/X-DF engines consume much less control air than conventional engines. The required control air flow capacities are shown in Table 4-10. These data can be used for sizing the relevant engine external piping and facilities.

Table 4-10 Control air flow capacities

No. of cyl.	Control air flow capacity [Nm^3/h]
5	12.0
6	14.4
7	16.8
8	19.2

4.5.4 Service and working air

Service and working air for driving air powered tools and assisting in the cleaning of scavenge air coolers is also provided by the board instrument air supply system.

4.6 Leakage collection system and washing devices

The latest version of the **Marine Installation Drawing Set** relevant for the leakage collection and washing system (DG 9724) is provided on the WinGD corporate webpage under the following link:
MIDS

Sludge oil trap

Dirty oil collected from the piston underside is led under a pressure of approx. 2.8 bar to the sludge oil trap and then to the sludge oil tank.
The purpose of the sludge oil trap (see Figure 4-17) is to retain the large amount of solid parts contained in dirty oil and to reduce the pressure by means of an orifice or throttling disc fitted at its outlet, so that the sludge oil tank is under atmospheric pressure.
Design and dimensions of the sludge oil trap are given in the MIDS.

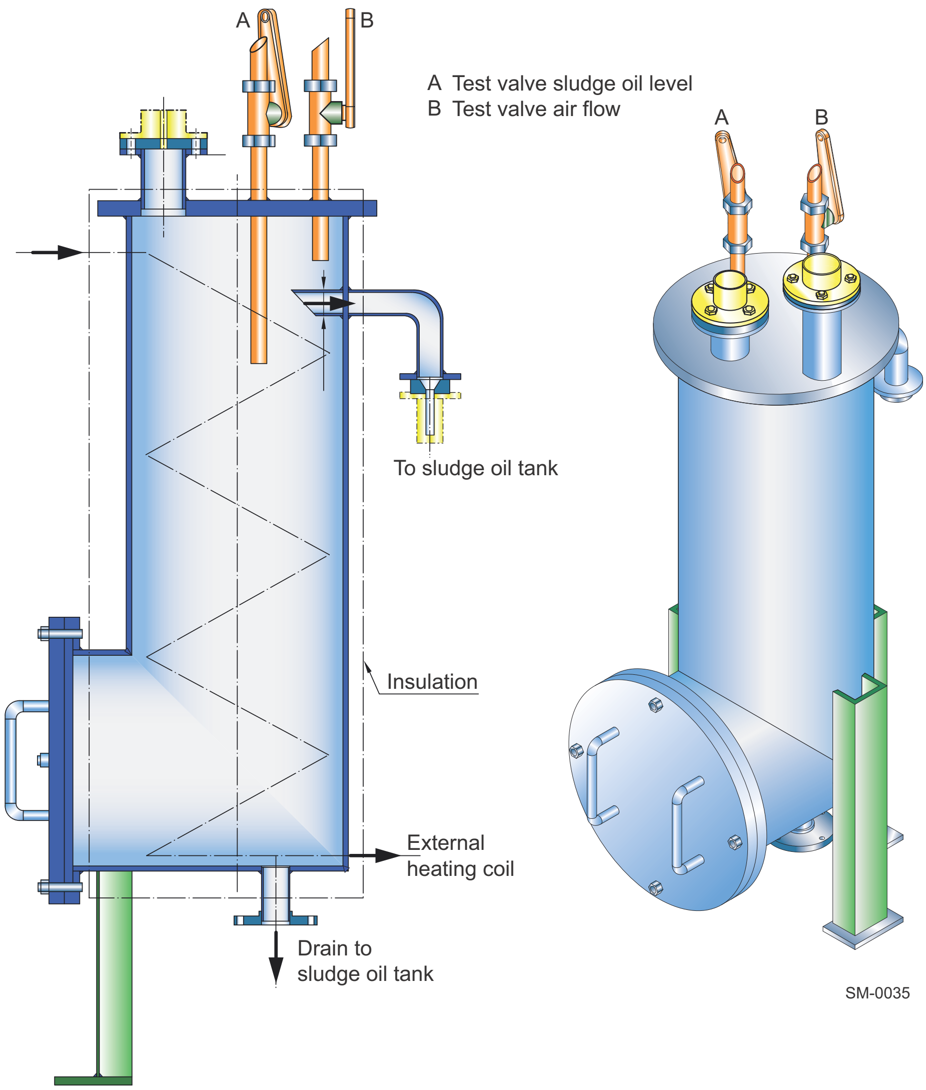


Figure 4-17 Sludge oil trap

From the piston rod stuffing box, dirty oil consisting of waste system oil, cylinder oil, metallic particles and small amounts of combustion products is led directly to the sludge oil tank.
Condensate from scavenge air is formed when the vessel is operating in a humid climate. To avoid excessive piston ring and liner wear, the condensate is to be continually drained from the scavenge air receiver.

4.6.1 Draining of exhaust uptakes

Engine exhaust uptakes can be drained automatically using a system as shown in Figure 4-18.

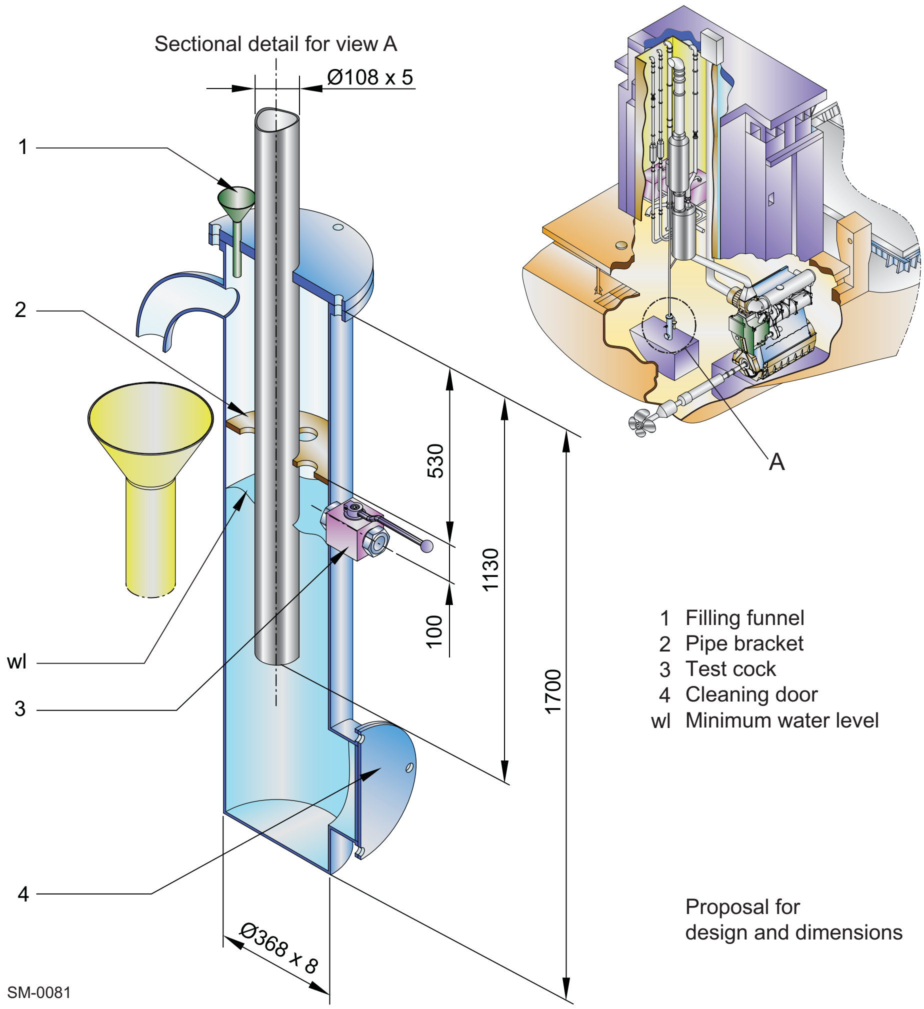


Figure 4-18 Arrangement of automatic water drain

4.6.2 Air vents

The air vent pipes of the ancillary systems have to be fully functional at all inclination angles of the ship at which the engine must be operational. This is normally achieved if the vent pipes have an uninterrupted inclination of min. 5%.
Such arrangement enables the vapour to separate into its air and fluid components, discharging the air to atmosphere and returning the fluid to its source.

4.7 Exhaust gas system

For an optimised exhaust gas system the following velocities are recommended for pipes A, B and C shown in Figure 4-19.

Pipe A: 40 m/s
Pipe B: 25 m/s
Pipe C: 35 m/s

For the pipe diameters please refer to the *GTD* application.

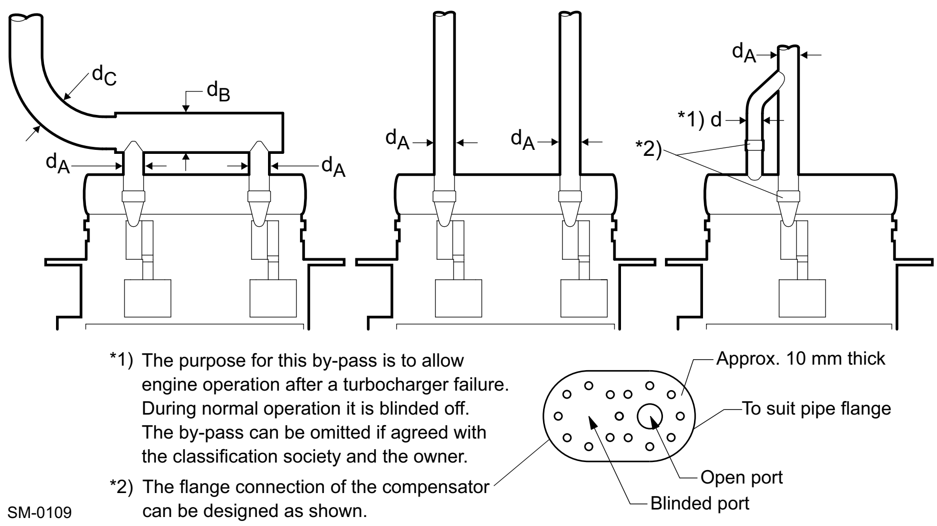


Figure 4-19 Determination of exhaust pipe diameter

4.8 Engine room ventilation

4.8.1 Requirements

Engine room ventilation is to conform to the requirements specified by the legislative council of the vessel's country of registration and the classification society selected by the shipowners.
Calculation methods for the air flows required for combustion and keeping the machinery spaces cool are given in the international standard ISO 8861 'Shipbuilding — Engine-room ventilation in diesel engined ships; Design requirements and basis of calculations'.

GTD

Based on ISO 8861, the radiated heat, required air flow and power for the layout of engine room ventilation can be obtained from the *GTD* program.

The final layout of the engine room ventilation is, however, at the discretion of the shipyard.

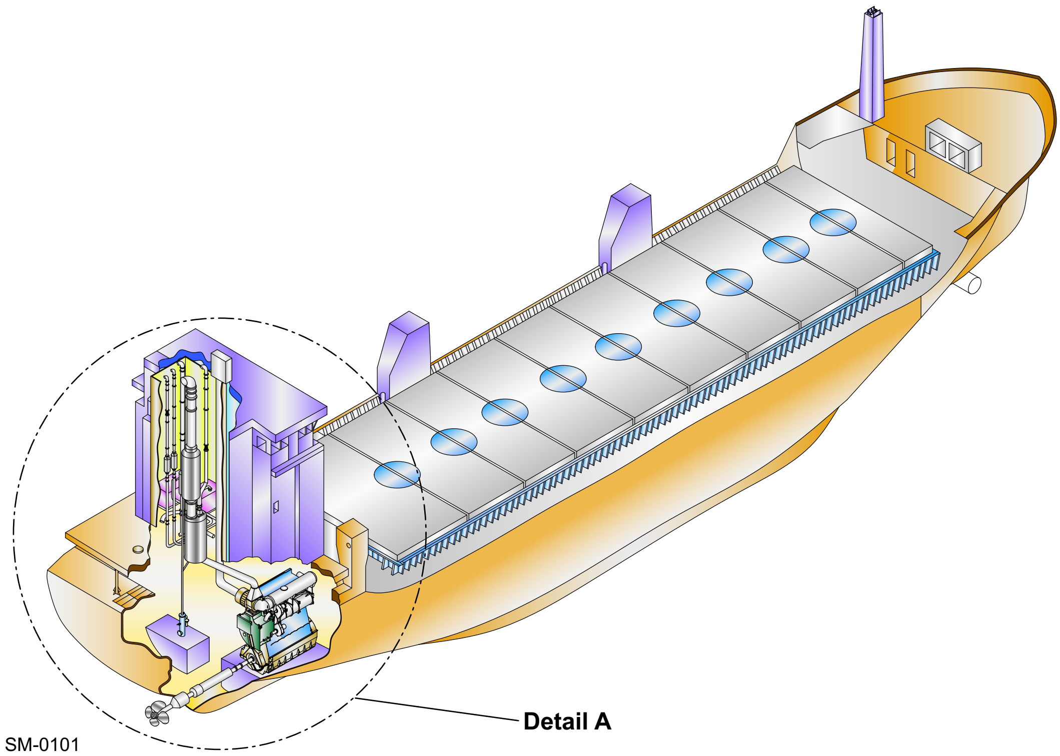


Figure 4-20 Direct suction of combustion air — main and auxiliary engine

4.8.2 Air intake

If the combustion air is drawn directly from outside, the engine may operate over a wide range of ambient air temperatures between arctic condition and tropical (design) condition (45 °C).

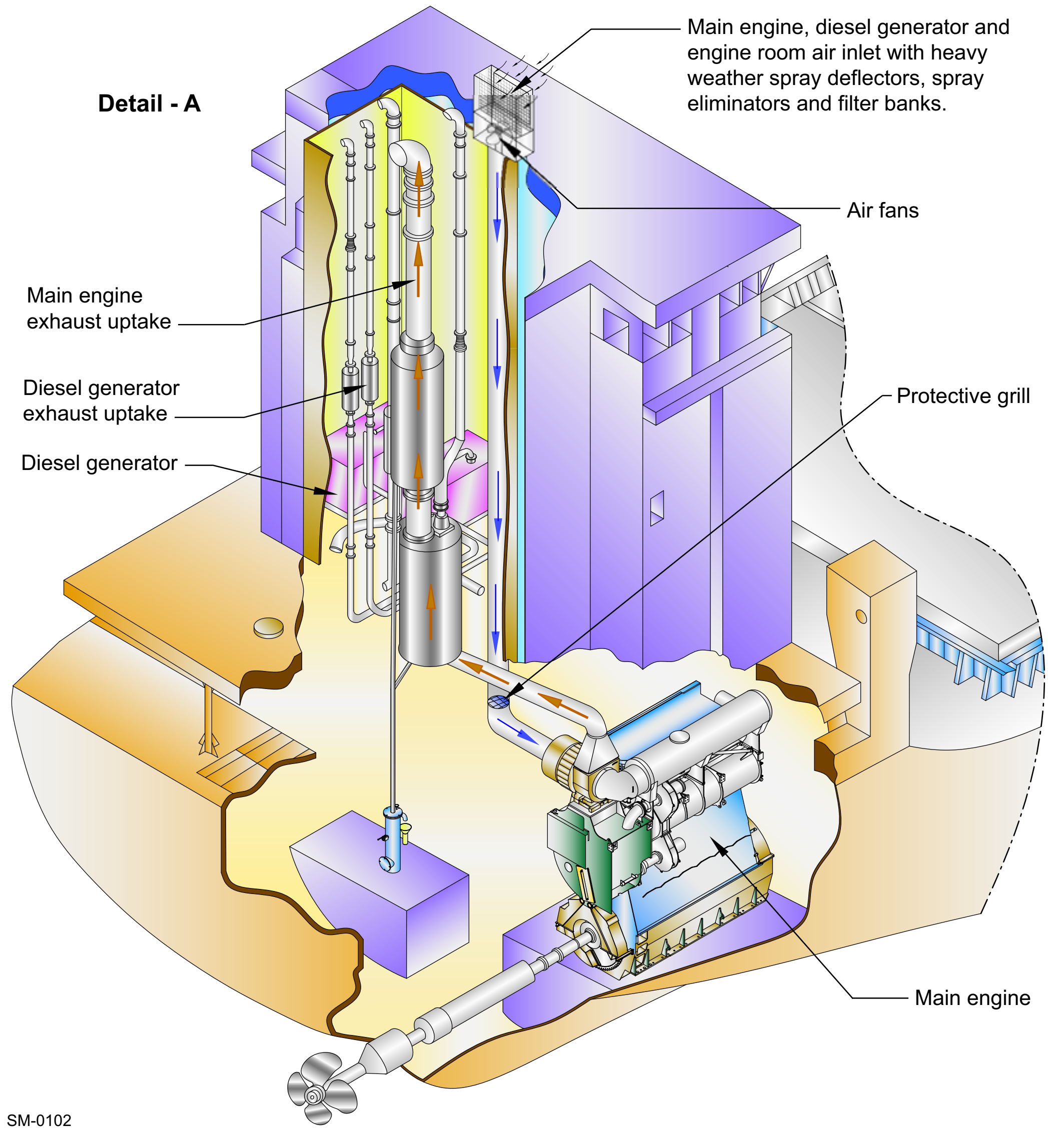


Figure 4-21 Direct suction of combustion air (detail)

Operating temperatures between 45 and 5 °C

The WinGD X62 engine does not require any special measures, such as pre-heating the air at low temperatures, even when operating on heavy fuel oil at part load, idling and starting up. The only condition which must be fulfilled is that the water inlet temperature to the scavenge air cooler is no lower than 25 °C.

This means:

- When combustion air is drawn directly from the engine room, no pre-heating of the combustion air is necessary.
- When combustion air is ducted in from outside the engine room and the air suction temperature does not fall below 5 °C, no measures need to be taken.

The central freshwater cooling system allows recovering the heat dissipated from the engine and maintains the required scavenge air temperature after the scavenge air cooler by recirculating part of the warm water through the low-temperature system.

Operating temperatures of less than 5 °C (arctic conditions)

- For Standard and Delta tuning: not available.
- For Delta bypass and Low load tuning: no further requirements are needed, as the engine integrated exhaust gas bypass adjusts the scavenge air flow to the engine.

NOTE	The scavenge air cooling water inlet temperature is to be maintained at min. +25 °C. In case of low-power operation this means that the scavenge air cooling water will have to be pre-heated. For that purpose, heat dissipation from other ancillary equipment, including lubricating oil and cylinder cooling water cooler, is utilised. Consequently no additional heater is required.

4.8.3 Air filtration

The necessity for installing a dust filter and the choice of filter type depends mainly on the concentration and composition of dust in the suction air. Where suction air is expected to have a dust content of 0.5 mg/m^3 or higher — for instance on coastal vessels or vessels frequenting ports where the air has a high atmospheric dust or sand content — the air must be filtered before it enters the engine.

Wear on piston rings and cylinder liners

In the event that the air supply to machinery spaces has a dust content exceeding 0.5 mg/m^3, which can be the case for ships trading in coastal waters, desert areas or transporting dust creating cargoes, there is a risk of increased wear to piston rings and cylinder liners. The normal air filters fitted to the turbochargers are intended mainly as silencers but not to protect the engine against dust.

NOTE	WinGD advises to install a filtration unit for the air supplies to main engines and general machinery space on vessels regularly transporting dust creating cargoes, such as iron ore and bauxite.

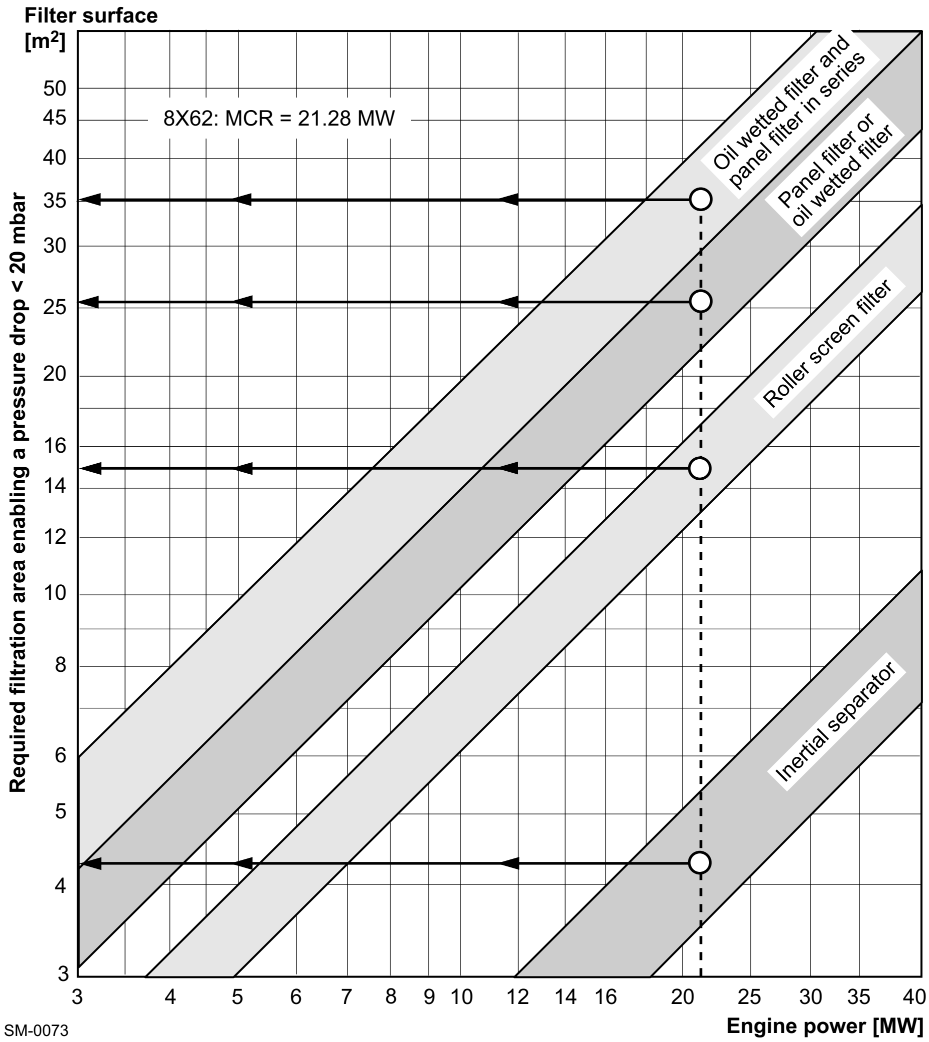


Figure 4-22 Air filter size (example for 8-cyl. engine)

Table 4-11 Guidance for air filtration

Dust concentration in ambient air			
Normal	Normal shipboard requirement	Alternatives necessary in very special circumstances	
Most frequent particle sizes	Short period < 5 % of running time, < 0.5 mg/m^3	Frequently to permanently ≥ 0.5 mg/m^3	Permanently > 0.5 mg/m^3
> 5 µm	Standard TC filter sufficient	Oil wetted or roller screen filter	Inertial separator and oil wetted filter
< 5 µm	Standard TC filter sufficient	Oil wetted or panel filter	Inertial separator and oil wetted filter
---	Normal requirement for the vast majority of installations	These alternatives apply most likely to only very few extreme cases, e.g. ships carrying bauxite or similar dusty cargoes, or ships routinely trading along desert coasts.	

4.9 Piping

4.9.1 Pipe connections

The latest versions of the **Pipe Connection Plan**s (DG 8020) are provided on the WinGD corporate webpage under the following links:
5-cyl. engine
6-cyl. engine
7-cyl. engine
8-cyl. engine

4.9.2 Flow rates and velocities

For the different media in piping, WinGD recommends flow rates and velocities as stated in the document 'Fluid velocities and flow rates'.
Note that the given values are guidances figures only and that national standards may also be applied.

The latest version of the document '**Fluid Velocities and Flow Rates**' (DG 9730) is provided on the WinGD corporate webpage under the following link:
Recommended fluid flow rates and velocities

4.10 PTO, PTI, PTH and primary generator applications

WinGD proposes various power take-off (PTO) and power take-in (PTI) options that improve the efficiency and usability of the vessel's propulsion chain. Some of the proposals are even suitable as power take-home devices (PTH), which enable the vessel to immobilise the main engine while staying capable to move. Furthermore, the primary generator enables the vessel to generate electric power by the main engine without running the propeller.

Depending on engine design the PTO solution can be applied either in the shaft line or at engine's free end.

NOTE	All given alternatives are subject to a detailed project-specific study and definition. Please consult WinGD via its licensee.

4.10.1 Requirements

After selecting the engine:
1) Define the shaft power and the shaft speed.
2) Estimate the electric power demand for propulsion purpose.
3) Evaluate which of the PTO / PTI systems is the most suitable.
4) Select suitable electrical components like frequency converter, etc.

NOTE	The type of the PTO/PTI system has an influence on the execution of the main engine. Thus, if you want to change from one system type to another, you may do so in the project stage but not after having ordered the engine.

4.10.2 Options

Figure 4-23, 4-55 illustrates the different options for PTO, PTI and PTH.

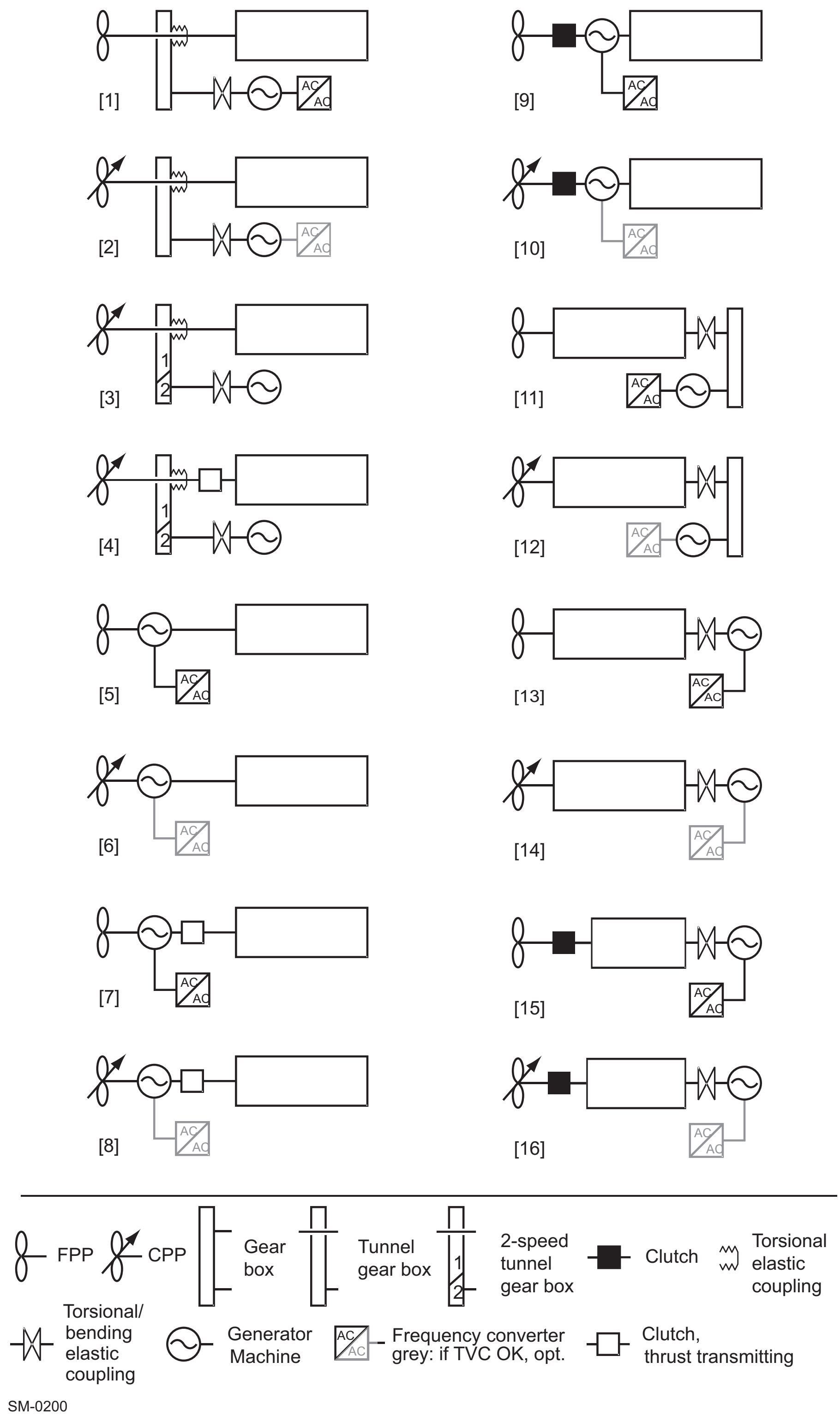


Figure 4-23 Options for PTO, PTI, PTH

The following table itemises the options corresponding to the numbers in Figure 4-23, 4-55.

Table 4-12 PTO / PTI / PTH options for WinGD X62

[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]	[15]	[16]
X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
'X' means that the option can be applied to the WinGD X62 engine.															

NOTE	In any case please check the application of options for the selected engine with WinGD via its licensee. Project dependent options can also be considered.

4.10.3 Application constraints

The feasibility of project-specific PTO / PTI / PTH and primary generator needs to be studied in any case. An overview about impacts is given in Table 4-14, 4-57.

Table 4-13 Features

	Options (see Figure 4-23, 4-55)							
Features	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]
Power take-off (PTO)	X	(X) a)/X b)	(X)	(X)	X	X	X	X
Power take-in (PTI)	X	(X) a)/X b)	(X)	(X)	X	X	X	X
Power take-home (PTH)							X	X
Primary generator								
Features	[9]	[10]	[11]	[12]	[13]	[14]	[15]	[16]
Power take-off (PTO)	X	X	X	(X) a)/X b)	X	(X) a)/X b)	X	(X) a)/X b)
Power take-in (PTI)	X	X	X	(X) a)/X b)	X	(X) a)/X b)	X	(X) a)/X b)
Power take-home (PTH)								
Primary generator	X	X					X	X
'X' means that the option is offered for the described feature. '(X)' means that the damping capability of couplings needs to be checked and if required increased to ensure sufficient speed stability.								

a) Installation without frequency converter
b) Installation with frequency converter

Table 4-14 Influence of options on engineering

Engineering	Options (see Figure 4-23, 4-55)															
	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]	[15]	[16]
Extended TVC	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
Misfiring detection	(X)	(X)	(X)	(X)	O	O	O	O	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)
Impact on ECS	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)
Shaft alignment study	(X)	(X)	(X)	(X)	X	X	X	X	X	X	(X)	(X)	(X)	(X)	X	X
Bearing load due to external load	(X)	(X)	(X)	(X)	X	X	X	X	X	X	(X)	(X)	(X)	(X)	X	X
Dynamic condition due to external load	---	---	---	---	---	---	---	---	---	---	X	X	X	X	X	X

X = the option has an influence
(X) = the option might have an influence
O = the option has no influence
--- = no external load

Extended TVC

The concept adds components to the propulsion line. This factor needs to be considered in the related project-specific TVC. Proper case dependent countermeasures need to be taken.

Misfiring detection

- In case the natural frequency in the shaft line is approx. 2 Hz or higher, a misfiring detection is needed to protect the elastic coupling.
- In case the natural frequency in the shaft line is lower than 2 Hz, a misfiring detection is needed to protect the elastic coupling and avoid speed stability problems.

Impact on ECS

The PTO / PTI / PTH application has to be analysed via the licensee with concerned (propulsion) control system suppliers and WinGD for the Engine Control System.

Shaft alignment study

If additional elements like a clutch with coupling status independent thrust transmission or a tunnel gear are added, the alignment effort increases. One needs to ensure that the bearings inside the engine as well as those in the shaft line are properly selected and adjusted.

Bearing load due to external load

The additional elements are not only increasing the bending moment but also the related bearing load. Therefore their application needs to be checked carefully.

Dynamic conditions due to external load

With the generator / motor at the engine's free end, the dynamic movement of the extension shaft has to be checked. This includes the proper location of the elastic coupling.

4.10.4 Service conditions

The service condition depends on the selected PTO/PTI/PTH option. Depending on engine type there are one or several cases, which are illustrated below.

Operation area and prohibited area

The green fields and respectively the green lines in the following illustrations indicate how the engine generator unit can operate. The red field represents the prohibited operation area, which is defined as described in section 2.2 Engine rating field and power range, 🗎 2-3 and shown in Figure 2-6, 🗎 2-13.

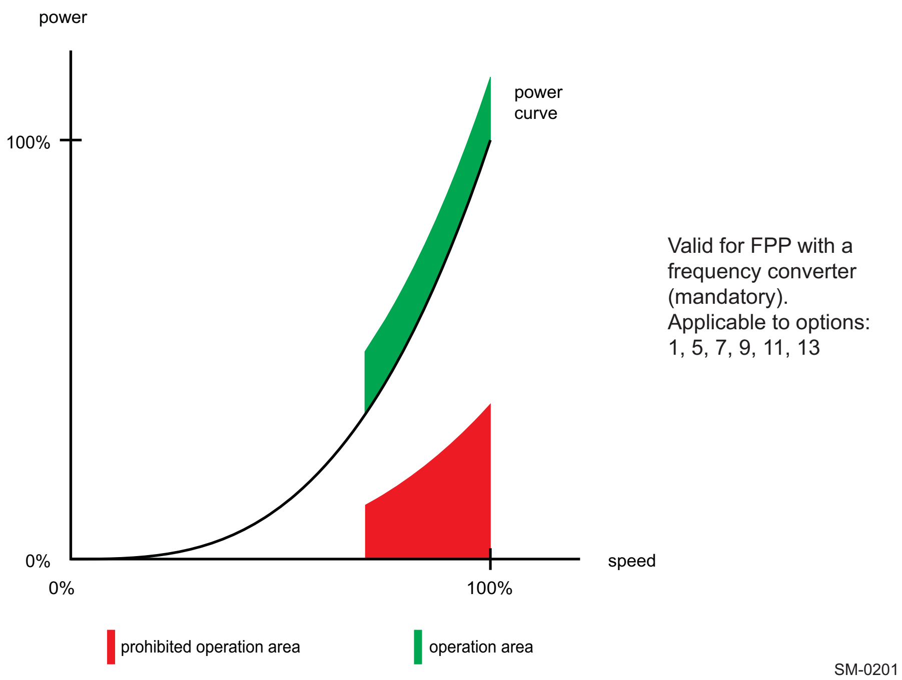


Figure 4-24 FPP with mandatory frequency converter

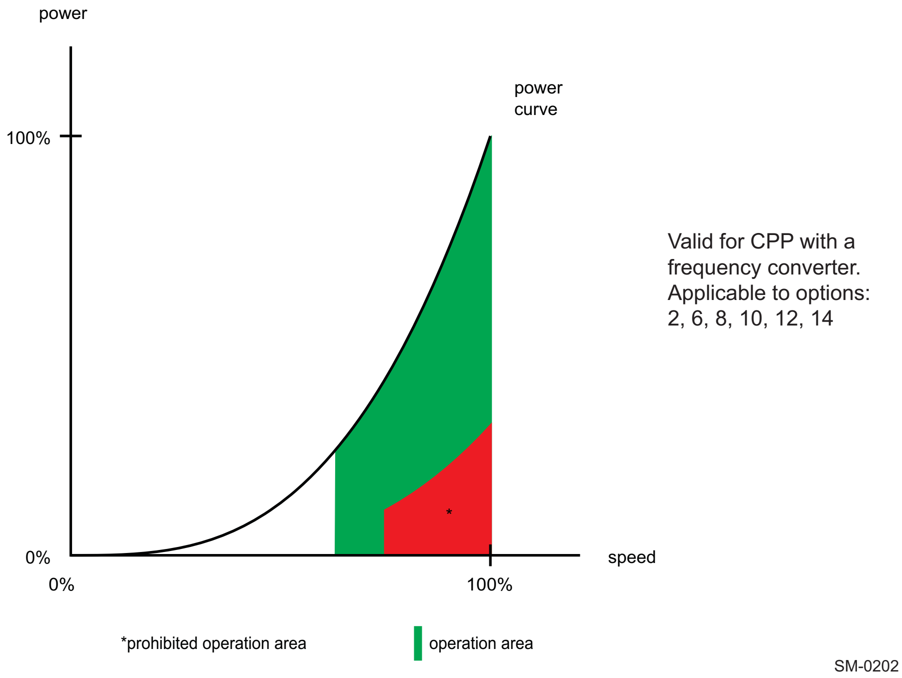


Figure 4-25 CPP in combination with an optional frequency converter

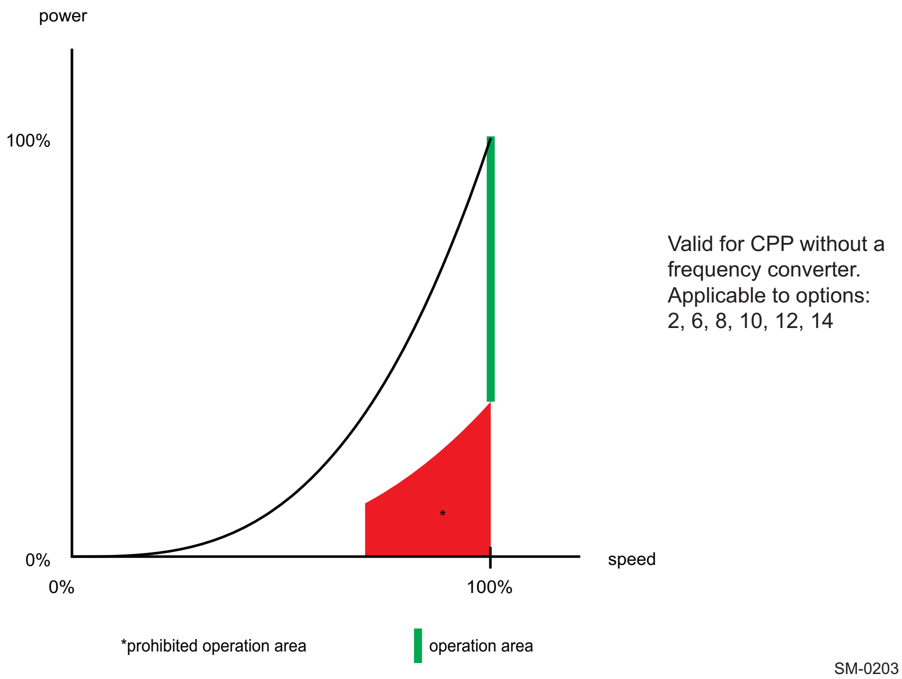
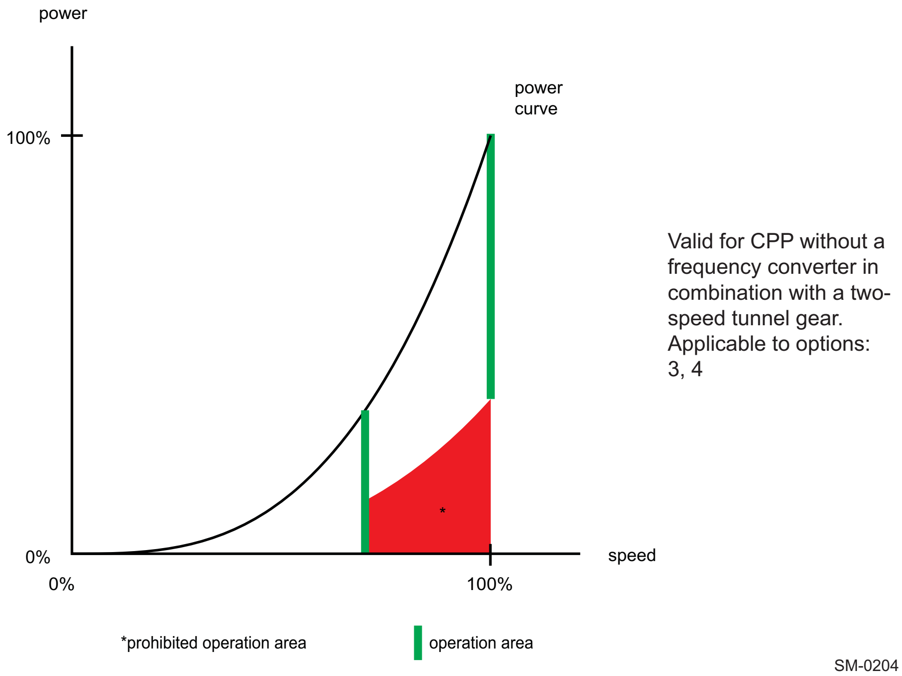


Figure 4-26 CPP in constant speed operation without frequency converter



Valid for CPP without a frequency converter in combination with a two-speed tunnel gear. Applicable to options: 3, 4

Figure 4-27 CPP with two fixed operation speeds without frequency converter

5 Engine Automation

The Engine Control System (ECS) provides data bus connection to the Propulsion Control System (PCS) and the Alarm and Monitoring System (AMS). The AMS is usually provided by the shipyard.

The leading suppliers of Propulsion Control Systems approved by WinGD ensure complete adaption to engine requirements.

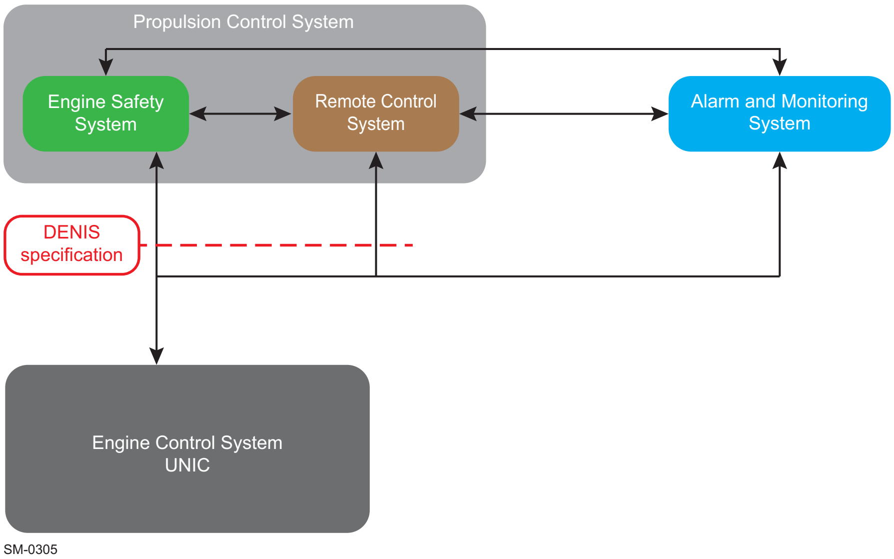


Figure 5-1 ECS layout

5.1 DENIS-UNIC

WinGD’s standard electrical interface is DENIS-UNIC, which is in line with approved Propulsion Control Systems.

DENIS The DENIS (**D**iesel **E**ngine Co**N**trol and opt**I**mizing **S**pecification) interface contains specifications for the engine management of all WinGD two-stroke marine diesel engines.

UNIC Under the designation of UNIC (**UNI**fied **C**ontrols), WinGD provides a fully integrated Engine Control System, which takes care of all Flex system-specific control functions, e.g. fuel injection, exhaust valve control, cylinder lubrication, crank angle measurement, and speed/load control. The system uses modern bus technologies for safe transmission of sensor- and other signals.

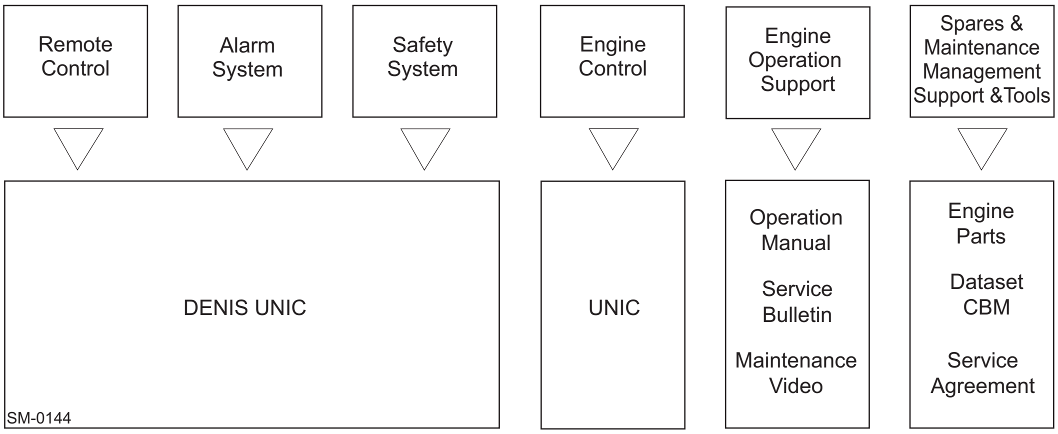


Figure 5-2 Engine management and automation concept with DENIS and UNIC

5.2 Concept

The concept of DENIS-UNIC offers the following features to shipowners, shipyards and engine builders:

5.2.1 Interface definition

The WinGD interface defines the division of responsibilities between engine builder and PCS and AMS supplier, enabling the authorised suppliers to adapt their systems to the common rail system engines. The data bus connection provides clear signal exchange.

5.2.2 Approved Propulsion Control Systems

Propulsion Control Systems including remote control, safety and telegraph systems are available from suppliers approved by WinGD (see Table 5-1, 5-4). This cooperation ensures that the systems fully comply with the specifications of the engine designer.

5.3 DENIS-UNIC Specification

The DENIS-UNIC Specification describes the signal interface between the UNIC Engine Control System and the PCS and AMS. It does not include any hardware, but summarises all data exchanged and defines the control functions required by the engine.
The DENIS-UNIC Specification consists of two sets of documents:

5.3.1 DENIS-UNIC Interface Specification

This signal interface specification is made available to engine builders and shipyards. Besides the description of engine-built components for control, alarm and indication the specification contains the following:

- List of alarm and display functions to be realised in the vessel's Alarm and Monitoring System
- Control diagram of the engine
- Signal list including a minimum of functional requirements
- Information related to the electrical wiring on the engine

5.3.2 DENIS-UNIC Propulsion Control Specification

This document contains a detailed functional specification of the Propulsion Control System (PCS).

The intellectual property rights of this specification remain with WinGD. Hence the document is licensed only to the partner companies of WinGD developing Propulsion Control Systems. These companies offer systems which are built exactly according to the engine designer's specifications and are finally tested and approved by WinGD.

5.4 Propulsion Control Systems

Approved Propulsion Control Systems (PCS) comprise the following independent sub-systems:

- Remote Control System (RCS)
- Safety system
- Telegraph system

The safety and the telegraph systems work independently and are fully operative even with the RCS out of order.

WinGD has an agreement with the marine automation suppliers listed in Table 5-1 concerning development, production, sale and servicing of the RCS and the safety system. All approved control systems listed in this table comprise the same functionality specified by WinGD.

Table 5-1 Suppliers of Remote Control Systems

Supplier		RCS
Kongsberg Maritime		
Kongsberg Maritime AS P.O. Box 1009 N-3194 Horten / Norway	km.sales@kongsberg.com Phone +47 81 57 37 00 www.km.kongsberg.com	AutoChief 600
NABTESCO Corporation		
NABTESCO corp., Marine Control Systems Company 1617-1, Fukuyoshi-dai 1-chome Nishi-ku Kobe, 651-22413 / Japan	newbuilding@nabtesco.com Phone +81 78 967 5361 www.nabtesco.com	M-800-V
SAM Electronics / Lyngsø Marine		
SAM Electronics GmbH Behringstrasse 120 D-22763 Hamburg / Germany	anc@sam-electronics.de Phone +49 40 88 25 0000 www.sam-electronics.de	DMS2200/EMS2200
Lyngsø Marine AS 2, Lyngsø Allé DK-2970 Hørsholm / Denmark	info@lyngsoe.com Phone +45 45 16 62 00 www.lyngsoe.com	

Modern Remote Control Systems consist of electronic modules and operator panels for display and order input in the ECR (Engine Control Room) and on the bridge (see Figure 5-3, 🗎 5-5). The different items normally communicate via serial bus connections. The engine signals described in the DENIS-UNIC Specification are usually connected via terminal boxes on the engine with the electronic modules placed in the ECR.

The electronic modules are in most cases built to be placed either inside the ECR console, or in a separate cabinet to be located in the ECR. The operator panels are to be inserted in the ECR console's surface.

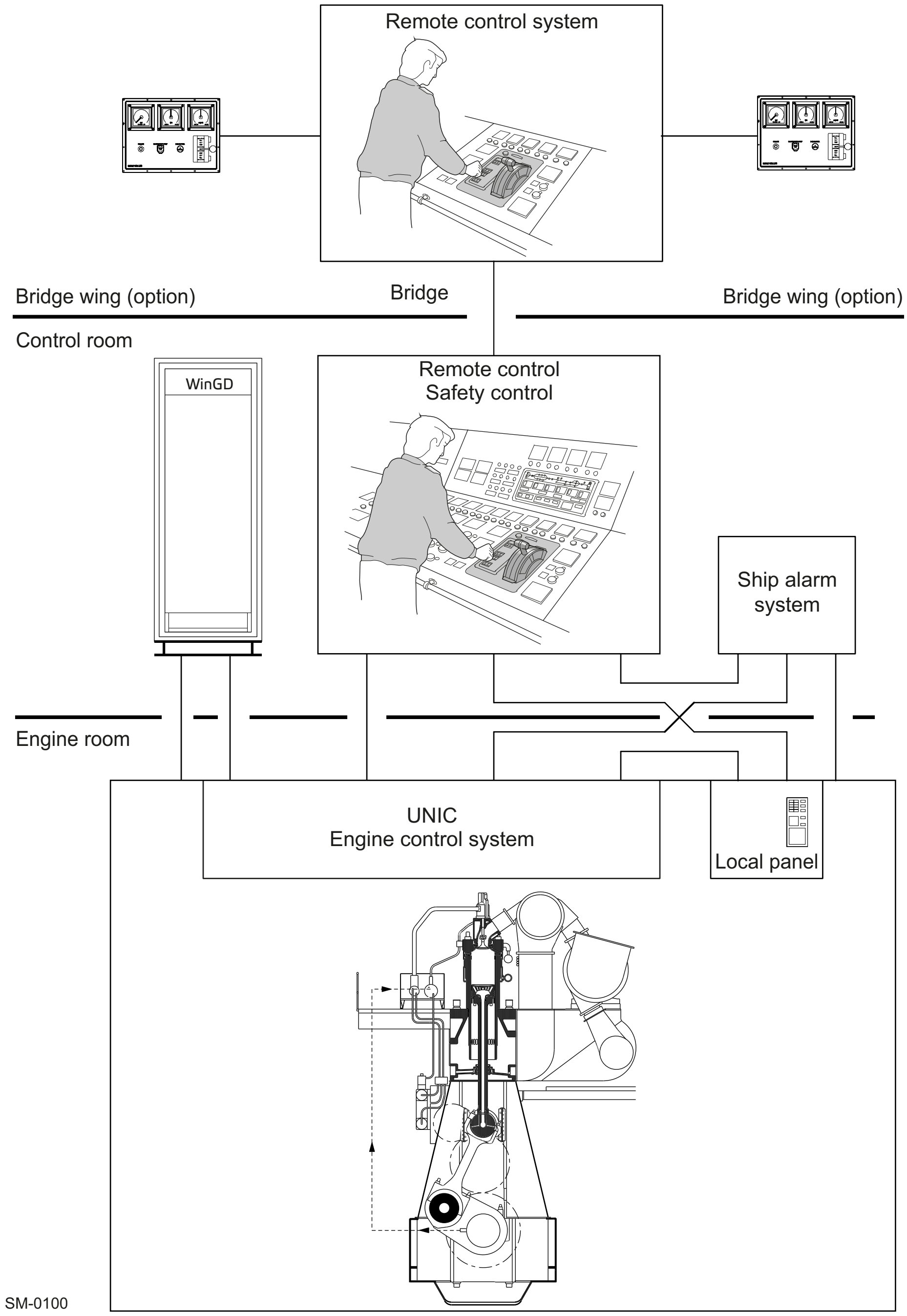


Figure 5-3 Remote Control System layout

5.4.1 PCS functions

Remote Control System

Main functions

- Start, stop, reversing
- Speed setting
- Automatic speed program

Indications

- The RCS is delivered with control panels for local, ECR and bridge control, including all necessary order input elements and indications, e.g. push buttons / switches and indication lamps or alternatively a respective display.
- The following conditions in the engine are specified by the DENIS-UNIC standard to be indicated as a minimum:
  - In the control room:
    - Starting air pressure
    - Engine speed
    - Revolutions
    - Operating hours
    - Load
    - Turbocharger speed
    - Scavenge air pressure in air receiver
  - On the bridge:
    - Starting air pressure
    - Engine speed
  - In addition to these indications, the RCS applied to the common rail system engine includes displaying the primary values from UNIC, like fuel pressure, servo oil pressure, etc.

Safety system

Main functions

- Emergency stop
- Overspeed protection
- Automatic shut-down
- Automatic slow-down

Telegraph system

- Order communication between the different control locations

Local manual control

- Local manual control of the engine is performed from a control panel located on the engine. The panel includes elements for manual order input and indication for the safety system, telegraph system and UNIC.
- The local control box with the local manual control panel is included in the package delivered by approved RCS suppliers.

ECR manual control panel

- A manual control panel delivered together with the PCS and fitted in the ECR console allows operating the engine manually and independently of the Remote Control System.
- The functions of the ECR manual control are identical to the control functions on the engine's local control panel.

Options

- Bridge wing control
- Command recorder

5.4.2 Recommended manoeuvring characteristics

The vessel speed is adjusted by the engine telegraph. Manoeuvring, e.g. for leaving a port, is available from full astern to full ahead. For regular full sea operation, the engine power can be further increased up to 100% CMCR power. Depending on the load step, a certain time is needed for the requested power increase or decrease to protect the engine.

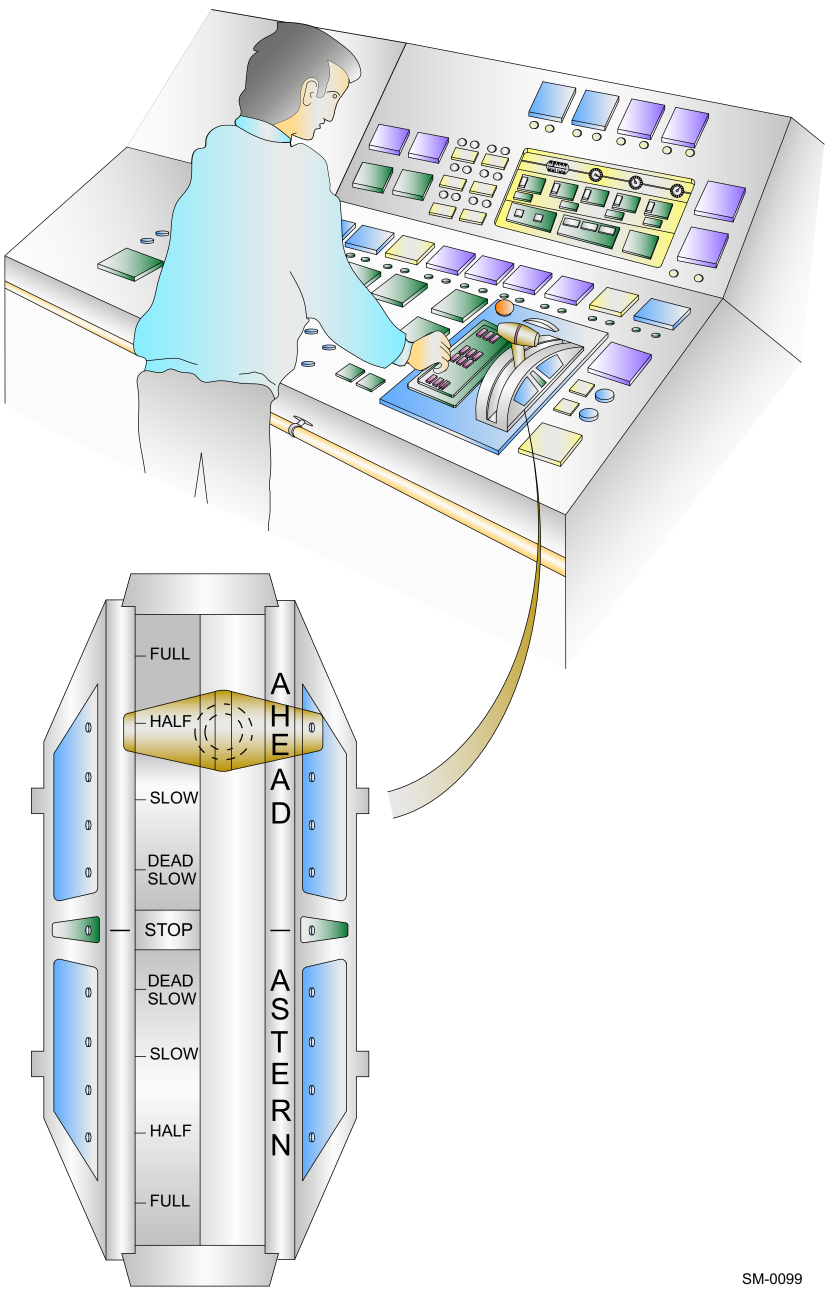


Figure 5-4 Propulsion control

FPP manoeuvring steps and warm-up times

The recommended manoeuvring steps and warm-up times for engine speed increase are indicated in Table 5-2. The engine speed-up/down program is included in the ECS.

Table 5-2 Recommended manoeuvring steps and warm-up times for FPP

Manoeuvring position	Point	Recommended CMCR speed [%]	Corresponding power [%]	Recommended warm-up time [min]	Min. warm-up time [min]
DEAD SLOW	1	25 - 35	1.5 - 4.5	0	0
SLOW	2	35 - 45	4 - 9	0	0
HALF	3	45 - 55	9 - 17	0.1	0.1
FULL	4	60 - 70	22 - 34	0.5	0.5
FULL SEA 1	5	92	78	30	23
FULL SEA 2	6	100	100	40	30

Load reduction is possible in half time of values mentioned in Table 5-2.

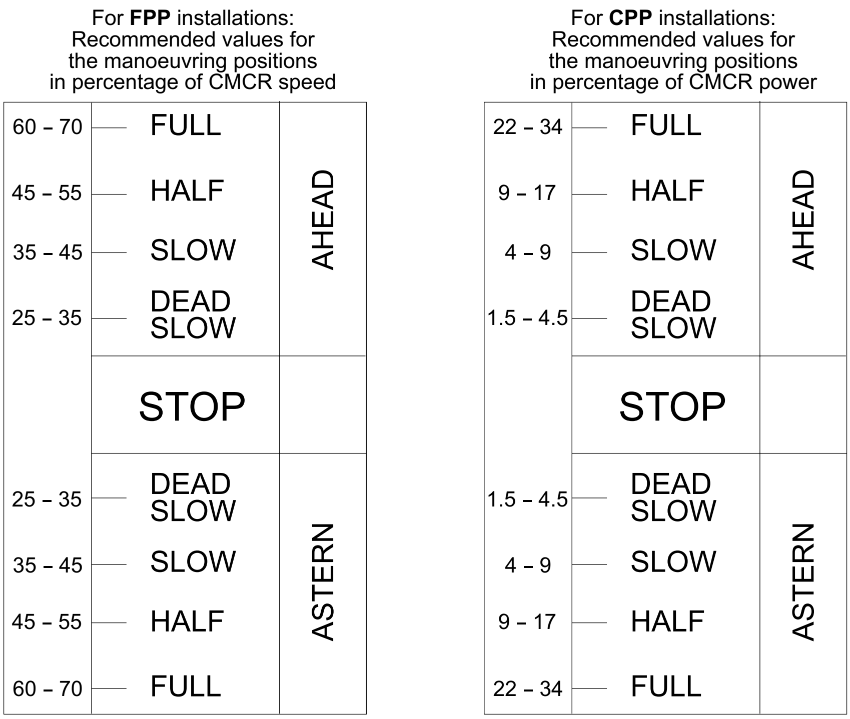


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Figure 5-5 Manoeuvring speed/power settings for FPP/CPP installations

CPP manoeuvring steps and warm-up times

The recommended manoeuvring steps and warm-up times for engine power increase are shown in Table 5-3. The shipyard needs to include the engine power-up/down program in the PCS.

Table 5-3 Recommended manoeuvring steps and warm-up times for CPP

Manoeuvring position	Point	Recommended CMCR power [%]	Recommended warm-up time [min]	Minimum warm-up time [min]
DEAD SLOW	1	1.5 - 4.5	0	0
SLOW	2	4 - 9	0	0
HALF	3	9 - 17	0.1	0.1
FULL	4	22 - 34	0.5	0.5
FULL SEA 1	5	78	30	23
FULL SEA 2	6	100	40	30

Load reduction is possible in half time of values mentioned in Table 5-3.

5.5 Alarm and Monitoring System

To monitor common rail system-specific circuits of the engine, sensors with hardwired connections are fitted. In addition to that, the UNIC Engine Control System provides alarm values and analogue indications via data bus connection to the ship's Alarm and Monitoring System (AMS).

5.5.1 Integrated solution

PCS and AMS from same supplier

- PCS and AMS are connected to the UNIC system through one redundant bus line (CANopen or Modbus, depending on automation maker).
- The integrated solution allows an extended presentation of relevant parameters and easy access to alterable user parameters backed by the graphical user interface functions available in the AMS.
- With the AutoChief 600 Alarm and Monitoring System by Kongsberg Maritime even the conventional sensors and additional Flex system-specific sensors can be connected via data bus lines, and the data acquisition units can be mounted on the engine in the same boxes used as terminal boxes for any other AMS. These boxes are usually provided by the shipyard and have to be delivered to the engine builder for mounting on the engine and connecting to the sensors.
- The integrated solution facilitates commissioning and testing of the alarm signals set on the engine maker's testbed and limits the wiring at the shipyard to a few power cables and bus communication.

5.5.2 Split solution

PCS and AMS from different suppliers

- The PCS is connected to the UNIC system through two redundant bus lines (CANopen or Modbus, depending on automation maker).
- For the separate AMS an additional redundant Modbus connection is available.
  Requirements for any AMS to be fulfilled in a split solution:
  - Possibility to read values from a redundant Modbus line according to standard Modbus RTU protocol
  - Ability to display analogue Flex system values (typically 20 values) and add alarm values provided by UNIC to the standard alarm list (300-800 alarms depending on engine type and number of cylinders)

With this solution the HMI is split as well:

- The Remote Control System includes the following functions:
  - Changing of parameters accessible to the operator
  - Displaying the parameters relevant for engine operation
- The Alarm and Monitoring System includes the display of:
  - Flex system parameters, like fuel pressure, servo oil pressure, etc.
  - Flex system alarms provided by UNIC
- WinGD provides Modbus lists specifying the display values and alarm conditions as part of the DENIS-UNIC Specification.

5.6 Alarm sensors and safety functions

To ensure safe operation the engine is provided with alarm sensors and safety functions.

5.6.1 Scope of delivery

The scope of delivery of alarm and safety sensors has to cover the requirements of the respective classification society, WinGD, the shipyard and the owner. WinGD requires a minimum of safety sensors for attended machinery space (AMS) and the addition of respective sensors in case the option of unattended machinery space (UMS) was chosen.

There are also some additional sensors defined for monitoring the Flex system-specific engine circuits.

The sensors are delivered with the engine and basically connected to terminal boxes mounted on the engine.

5.6.2 Signal processing

Signal processing has to be performed in the AMS. An alarms and safeguards table in the X62 Operation Manual lists the signal levels at which an alarm, a slow-down or a shut-down is triggered and mentions the corresponding response times at a specific setting.

The **Alarms and Safeguards** table is provided as excerpt from the X62 Operation Manual on the WinGD corporate webpage under the following link:
Usual values & safeguard function setting

Please note that the signalling time delays given in the table are maximum values. They may be reduced at any time according to operational requirements. When decreasing the values for slow-down times, the delay times for the respective shut-down functions are to be adjusted accordingly.
The delay values are not to be increased without the written consent of WinGD.

5.6.3 Requirements of classification societies

The different alarm and safety functions required by the classification societies depend on the class of the vessel and the degree of automation.

(List of classification societies see Appendix, section 9.1, 🗎 9-1.)

Table 5-4 Legend to Table 5-5

Requirements of classification societies for unattended machinery space (UMS)		
Required: ●	Recommended: ○	Required alternatively: - either A or B - either C or D - either E or F - either G or H - either I or K
Special requirement for attended machinery space (AMS)		
Required for AMS only: ▲		Additional requirement to UMS for AMS: ■

Table 5-5 Alarm and safety functions: Class and WinGD requirements

An update of this table is under preparation.

6 Engine Dynamics

As a leading designer and licensor we are concerned that vibrations are minimised in our engine installations. The assessment and reduction of vibration is subject to continuing research. To deal with this subject we have developed extensive computer software, analytical procedures and measuring techniques.

For successful design, the vibration behaviour needs to be calculated over the whole operating range of the engine and the propulsion system. The following vibration types and their causes are to be considered:

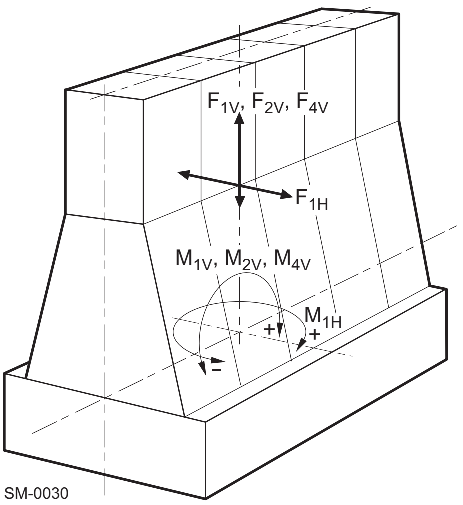
- External mass forces and moments
- Lateral engine vibration
- Longitudinal engine vibration
- Torsional vibration of the shafting
- Axial vibration of the shafting

6.1 External mass forces and moments

In the design of the engine, free mass forces are eliminated and unbalanced external moments of first, second and fourth order are minimised.

However, 5- and 6-cylinder engines generate unbalanced second order vertical moments of a magnitude greater than those encountered with higher numbers of cylinders. Depending on the ship's design, the moments of fourth order have to be considered, too.

Under unfavourable conditions, depending on hull structure, type, distribution of cargo and location of the main engine, the unbalanced moments of first, second and fourth order may cause unacceptable vibrations throughout the ship and thus call for countermeasures. Figure 6-1 shows the external forces and moments acting on the engine.



F_{1V} Resulting first order vertical mass force
F_{1H} Resulting first order horizontal mass force
F_{2V} Resulting second order vertical mass force
F_{4V} Resulting fourth order vertical mass force
M_{1V} First order vertical mass moment
M_{1H} First order horizontal mass moment
M_{2V} Second order vertical mass moment
M_{4V} Fourth order vertical mass moment

Figure 6-1 External forces and moments

Dynamic characteristics

The latest version of the **Dynamic Characteristics Data** is provided on the WinGD corporate webpage under the following link:
External mass forces and moments

6.1.1 Balancing first order moments

Standard counterweights fitted to the ends of the crankshaft reduce the first order mass moments to acceptable limits. However, in special cases non-standard counterweights can be used to reduce either M_{1V} or M_{1H}.

6.1.2 Balancing second order moments

The second order vertical moment (M_{2V}) is higher on 5- and 6-cylinder engines compared with 7- and 8-cylinder engines, the second order vertical moment being negligible for 7- and 8-cylinder engines.

To reduce the effects of second order moments to acceptable values, WinGD recommends one of the following countermeasures for 5- and 6-cylinder engines:

- Install engine-fitted second order balancers (iELBA) at free end and driving end.
- Install an electrically driven compensator on the ship's structure (Figure 6-2, 6-3). If no experience is available from a sister ship, it is advisable to establish in the design stage of what kind the ship's vibration will be.

External compensator

However, when the ship's vibration pattern is not known at an early stage, an external electrically driven compensator can be installed later, should disturbing vibrations occur. Such a compensator is usually installed in the steering compartment. It is tuned to the engine operating speed and controlled accordingly.

Suppliers of electrically driven compensators		
Gertsen & Olufsen AS	Savsvinget 4 DK-2970 Hørsholm Denmark	Phone: +45 45 76 36 00 Fax: +45 45 76 17 79 www.gertsen-olufsen.dk
Nishishiba Electric Co., Ltd	Shin Osaka Iida Bldg. 5th Floor 1-5-33, Nishimiyahara, Yodogawa-ku Osaka 532-0004 Japan	Phone: +81 6 6397 3461 Fax: +81 6 6397 3475 www.nishishiba.co.jp

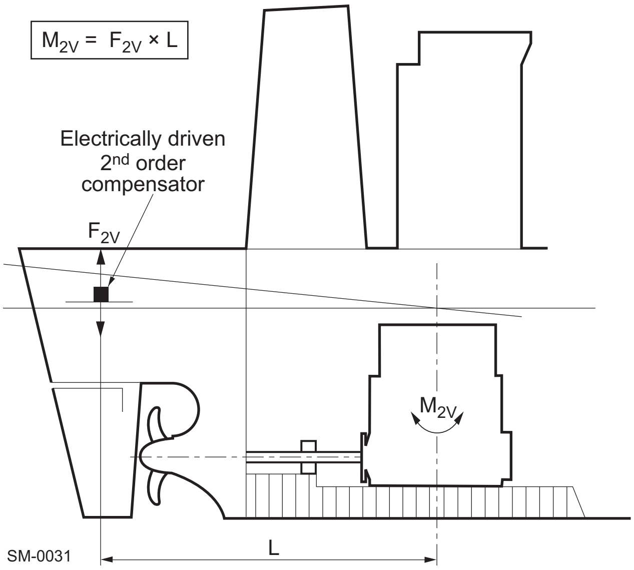


Figure 6-2 Locating an electrically driven compensator

6.1.3 Power related unbalance

The so-called power related unbalance (PRU) values can be used to evaluate if there is a risk that free external mass moments of first and second order cause unacceptable hull vibrations. See the graphs for R1+ and R1 on page 5 and 10 respectively in the linked document:
External mass forces and moments

The external mass moments M_1 and M_2 given in the table 'External forces and moments' (see Dynamic characteristics, 6-2) are related to R1 / R1+ speed. For other engine speeds, the corresponding external mass moments are calculated with the following formula:

$$M_{Rx} = M_{R1} \cdot \left(\frac{n_{Rx}}{n_{R1}} \right)^2$$

or

$$M_{Rx} = M_{R1+} \cdot \left(\frac{n_{Rx}}{n_{R1+}} \right)^2$$

6.2 Lateral vibration (rocking)

Depending on the number of cylinders and firing order, the lateral components of the forces acting on the crosshead induce lateral rocking. These forces may be transmitted to the engine room bottom structure. From there, hull resonance or local vibrations in the engine room may be excited.

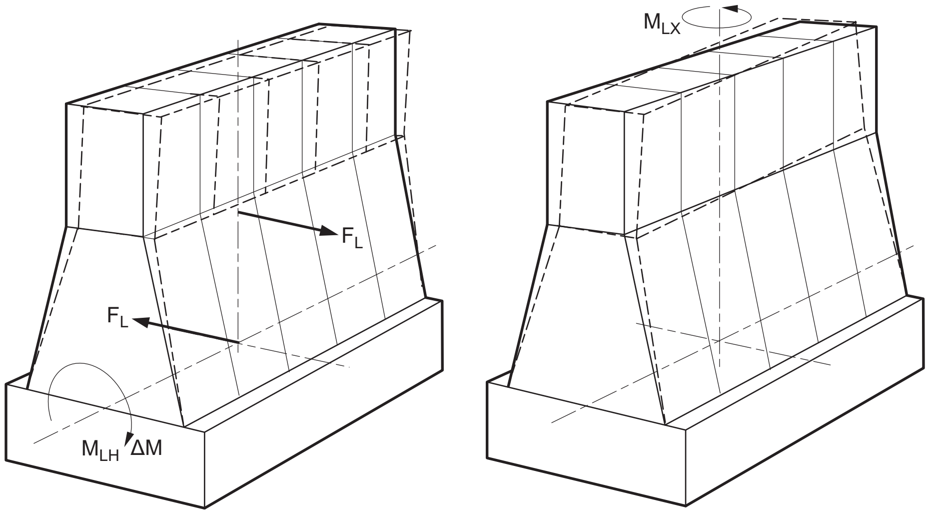
There are two different modes of lateral engine vibration, the so-called ‘H-type’ and ‘X-type’ vibration; refer to Figure 6-3.

H-type vibration

H-type lateral vibrations are characterised by a deformation where the driving and free end sides of the engine top vibrate in phase as a result of the lateral guide force F_L and the lateral H-type moment. The torque variation (ΔM) is the reaction moment to M_{LH}.

X-type vibration

X-type lateral vibrations are caused by the resulting lateral guide force moment M_{LX}. The driving and free end sides of the engine top vibrate in counterphase.



F_L Resulting guide force
M_{LH} Resulting lateral H-type moment
M_{LX} Resulting lateral X-type moment
SM-0032

Figure 6-3 Lateral vibration

The table ‘External forces and moments’ (see Dynamic characteristics, 🗎 6-2) gives the values of resulting lateral guide forces and moments of the relevant orders.

Amplitudes of vibrations

The amplitudes of the vibrations transmitted to the hull depend on the design of engine seating, frame stiffness and exhaust pipe connections. As the amplitude of the vibrations cannot be predicted with absolute accuracy, the support to the ship’s structure and space for installation of lateral stays should be considered in the early design stages of the engine room structure.

Reduction of lateral vibration

Lateral hydraulic stays

Lateral stays fitted between the upper platform level and the hull reduce transmitted vibration and rocking. Such stays must be of single- or double-acting hydraulic type and are installed in either of the two following general arrangements:

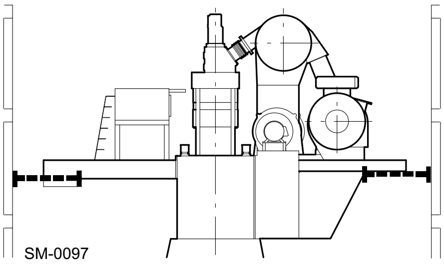


Figure 6-4 General arrangement of single-acting hydraulic stays

The latest version of the **Assembly Instruction** for single-acting hydraulic stays (DG 9715) is provided on the WinGD corporate webpage under the following link:
Assembly instruction - Single-acting hydraulic type stays

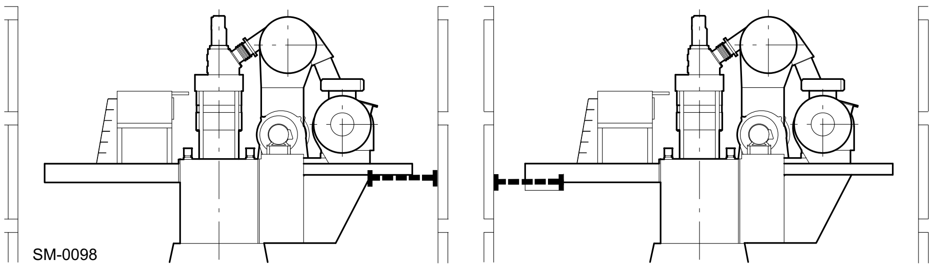


Figure 6-5 General arrangement of double-acting hydraulic stays

Electrically driven compensator

If for some reason it is not possible to fit lateral stays, an electrically driven compensator can be installed, which reduces the lateral engine vibrations and their effect on the ship's superstructure.

It has to be noted that only one harmonic excitation at a time can be compensated, and in case of an 'X-type' vibration, two compensators — one fitted at each end of the engine top — are necessary.

6.3 Longitudinal vibration (pitching)

In some cases with 5-cylinder engines, specially those coupled to very stiff intermediate shafts and propeller shafts, the engine foundation can be excited at a frequency close to full-load speed range resonance, leading to increased axial (longitudinal) vibration at the engine top, and as a result to vibrations in the ship's superstructure (refer to section 6.5, 6-8). To prevent such vibration, the stiffness of the double-bottom structure should be as strong as possible.

Reduction of longitudinal vibration on 5-cylinder engines

Depending on the results of vibration measurements it may be necessary to install longitudinal engine stays. They are arranged as shown in Figure 6-6.

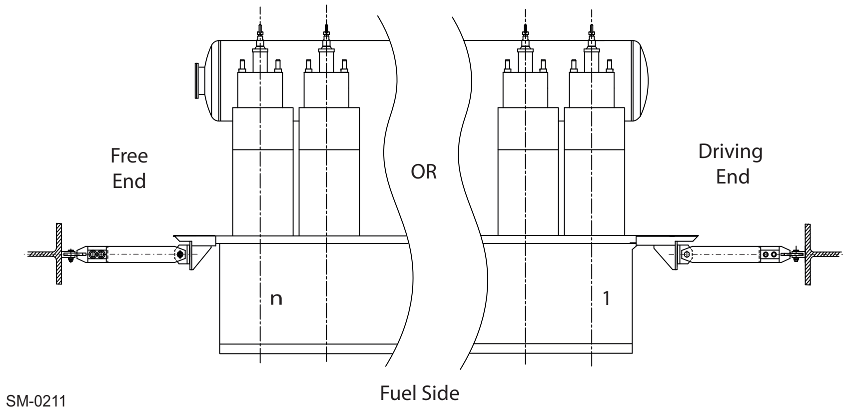


Figure 6-6 Arrangement of longitudinal stays

Types of longitudinal engine stays

The following types of longitudinal stays can be applied:

- either friction type stays according to WinGD design (see below link to MIDS drawing) or third-party maker design, to be installed on engine free end or driving end side
- or hydraulic stays according to WinGD design (see below link to MIDS drawing) or third-party maker design, to be installed on engine free end or driving end side

The layout of friction type and hydraulic type stays has to be as shown in the drawing 'Engine stays' in the *MIDS* (DG 9715).

For the assembly of friction type stays please observe the latest version of the **Assembly Instruction** for WinGD friction type stays (DG 9715), which is provided on the WinGD corporate webpage under the following link:
Assembly instruction - Friction type stays

6.4 Torsional vibration

Torsional vibrations are generated by gas and inertia forces as well as by the irregularity of the propeller torque. It does not cause hull vibration (except in very rare cases) and is not perceptible in service, but produces additional dynamic stresses in the shafting.

The shafting system comprising crankshaft, propulsion shafting, propeller, engine running gear, flexible couplings and power take-off (PTO) has resonant frequencies, as any system capable of vibrating.

Dangerous resonances

If any source generates excitation at resonant frequencies, the torsional loads in the system reach maximum values. These torsional loads have to be limited, if possible by design, for example by optimising shaft diameters and flywheel inertia. If the resonance still remains dangerous, its frequency range (critical speed) has to be passed through rapidly (barred speed range), provided that the corresponding limits for this transient condition are not exceeded, otherwise other appropriate countermeasures have to be taken.

Torsional vibration calculation (TVC)

The amplitudes and frequencies of torsional vibration must be calculated in the design stage for every engine installation. The calculation normally requires approval by the relevant classification society and may require verification by measurement on board ship during sea trials. All data required for torsional vibration calculations should be made available to the engine supplier in an early design stage (see section 6.9, 6-11).

Reduction of torsional vibration

Excessive torsional vibration can be reduced, shifted or even avoided by installing a heavy flywheel at the driving end and/or a tuning wheel at the free end, or a torsional vibration damper at the free end of the crankshaft. Such dampers reduce the level of torsional stresses by absorbing part of the energy.

Low-energy vibrations

Where low-energy torsional vibrations have to be reduced, a viscous damper can be installed; refer to Figure 6-7, 6-8. In some cases the torsional vibration calculation shows that an additional oil-spray cooling for the viscous damper is needed. In such cases the layout has to be in accordance with the recommendations of the damper manufacturer and our design department.

High-energy vibrations

For high-energy vibrations — e.g. for higher additional torque levels that may occur with 5- and 6-cylinder engines — a spring damper with its higher damping effect may have to be considered; refer to Figure 6-7, 6-8.

The spring damper has to be supplied with oil from the engine's lubricating oil system. Depending on the torsional vibration energy to be absorbed, it can dissipate up to 120 kW energy (depends on number of cylinders).

The oil flow to the damper should be approx. 25 m^3/h. An accurate value will be given after the results of the torsional vibration calculation are known.

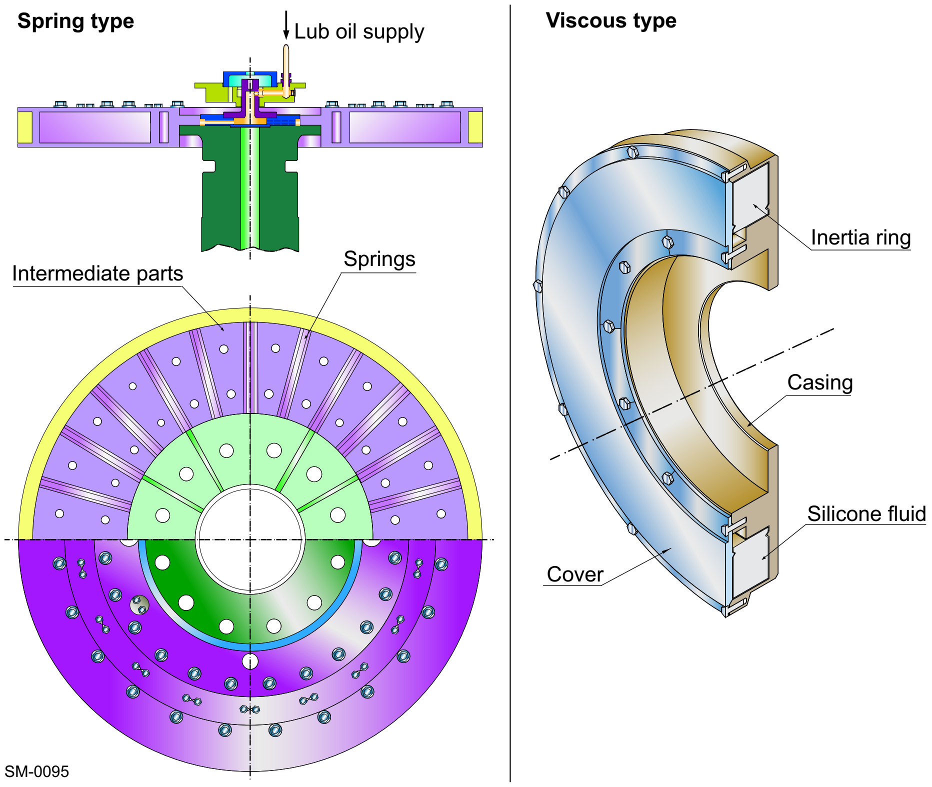


Figure 6-7 Vibration dampers (spring type and viscous type)

6.5 Axial vibration

The shafting system, formed by the crankshaft and propulsion shafting, can vibrate in axial direction, the basic principle being the same as described in section 6.4, 6-7. The system, made up of masses and elasticities, will feature several resonant frequencies. If no counter-measures are taken, these frequencies will result in axial vibration causing excessive stresses in the crankshaft. Strong axial vibration of the shafting can also lead to excessive axial (or longitudinal) vibration of the engine, particularly in its upper part.

Coupling effect

The axial vibrations of installations mainly depend on the dynamical axial system of the crankshaft, the mass of the torsional vibration damper, free-end gear (if any) and flywheel fitted to the crankshaft. Additionally, axial vibrations can be considerably influenced by torsional vibrations. This influence is called *coupling effect of torsional vibrations*.

It is recommended that axial vibration calculations are carried out at the same time as torsional vibration calculations. To consider the coupling effect of torsional vibrations on axial vibrations, it is necessary to apply a suitable coupled axial vibration calculation method.

Reduction of axial vibration

To limit the influence of axial excitations and reduce the level of vibration, the standard WinGD X62 engine is equipped with an integrated axial vibration damper mounted at the free end of the crankshaft.

Axial vibration damper

The axial vibration damper reduces the axial vibrations in the crankshaft to acceptable values. No excessive axial vibrations should then occur, neither in the crankshaft, nor in the upper part of the engine.

The effect of the axial vibration damper can be adjusted by an adjusting throttle. However, the throttle is pre-set by the engine builder, and there is normally no need to change the setting.

The integrated axial vibration damper does not affect the external dimensions of the engine. It is connected to the main lubricating oil circuit.
An integrated monitoring system continuously checks the correct operation of the axial vibration damper.

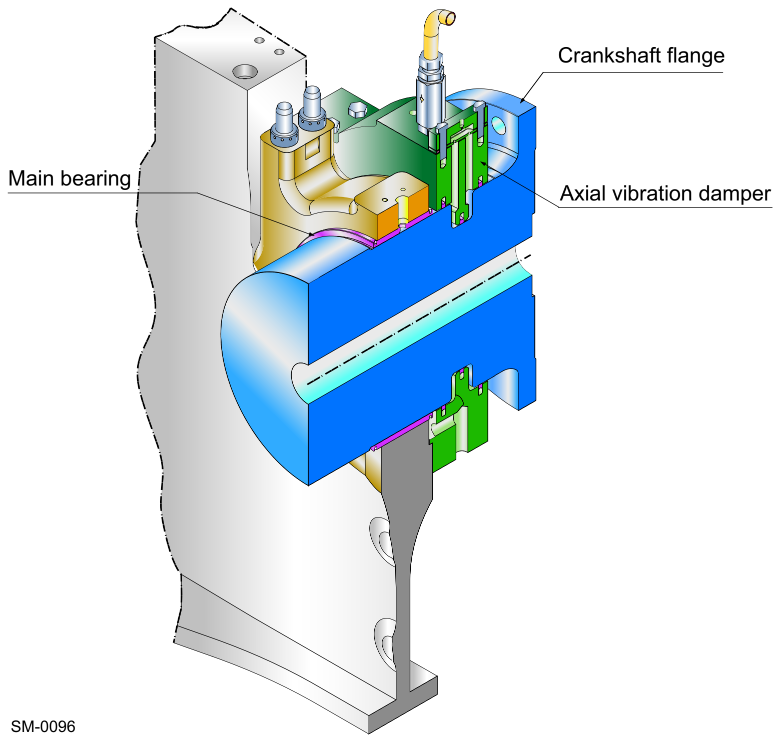


Figure 6-8 Example of axial vibration damper

6.6 Hull vibration

The hull and accommodation area are susceptible to vibration caused by the propeller, machinery and sea conditions. Controlling hull vibration is achieved by a number of different means and may require the fitting of mass moment compensators, lateral stays, torsional vibration dampers and axial vibration dampers.

Avoiding disturbing hull vibration requires a close cooperation between the propeller manufacturer, naval architect, shipyard, and engine builder.
To enable WinGD to provide the most accurate information and advice on protecting the installation and vessel from the effects of plant vibration, please complete the order forms as given in section 6.9, 🗎 6-11 and send it to the address stated.

6.7 Countermeasures for dynamic effects

The following document provided on the WinGD corporate webpage specifies where special attention is to be given to dynamic effects and the countermeasures required to reduce them:
Countermeasures for dynamic effects

Where installations incorporate PTO arrangements (see Figure 4-23, 🗎 4-55), further investigation is required and WinGD should be contacted.

6.8 System dynamics

A modern propulsion plant may include a main-engine driven generator. This element is connected by clutches, gears, shafts and elastic couplings. Under transient conditions, heavy perturbations — due to changing the operating point, loading or unloading generators, engaging or disengaging a clutch — cause instantaneous dynamic behaviour, which weakens after a certain time (or is transient). Usually the transfer from one operating point to another is monitored by a control system to allow the plant to adapt safely and rapidly to the new operating point (engine speed control and propeller speed control).

Analysis of dynamic behaviour

Simulation is an opportune method for analysing the dynamic behaviour of a system subject to heavy perturbations or transient conditions. Mathematical models of several system components such as clutches and couplings have been determined and programmed as library blocks to be used with a simulation program. Such program allows to check, for example, if an elastic coupling will be overloaded during engine start, or to optimise a clutch coupling characteristic (engine speed before clutching, slipping time, etc.), or to adjust the speed control parameters.

This kind of study should be requested at an early stage of the project if some special specification regarding speed deviation and recovery time, or any special speed and load setting programs have to be fulfilled.

WinGD would like to assist if you have any questions or problems relating to the dynamics of the engine. Please describe the situation and send or fax the completed relevant order form listed in the table in section 6.9, 🗎 6-11. We will provide an answer as soon as possible.

6.9 Order forms for vibration calculation & simulation

The following forms for system dynamics and vibration analysis are available on the Licensee Portal. (PDF format available on request.) They can be filled in and submitted directly to WinGD.

Marine installation	Testbed installation
Torsional Vibration Calculation	Torsional Vibration Calculation
Coupled Axial Vibration Calculation	
Whirling/Bending Vibration Calculation	

If you have no access to the Licensee Portal, you can order the forms from WinGD and e-mail a PDF of the completed relevant forms to the following address: dynamics.ch@wingd.com.

Winterthur Gas & Diesel Ltd.
Dept. 21336 Engine Dynamics & Structural Analysis
Schützenstrasse 1-3
PO Box 414
CH-8401 Winterthur

7 Engine Emissions

In 1973 an agreement on the International Convention for the Prevention of Pollution from Ships was reached. It was modified in 1978 and is now known as MARPOL 73/78.

Annex VI to MARPOL 73/78, entered into force in 2005, contains regulations limiting or prohibiting certain types of emissions from ships, including limitations with respect to air pollution. Following the entry into force of the annex, a review process was started, resulting in an amended Annex IV, which was adopted by the International Maritime Organization (IMO) in October 2008 and entered into force in July 2010.
This amended Annex IV includes provisions for the further development of emissions regulations until 2020.

7.1 Exhaust gas emissions

7.1.1 Regulation regarding NO_X emissions

Regulation 13 of Annex IV specifies a limit for the nitrogen oxides (NO_X) emissions of engines installed on ships, which has a direct implication on the design of propulsion engines.
Depending on the rated speed of the engine and the date of keel-laying of the vessel, the weighted average NO_X emission of that engine must not exceed the maximum allowable value as indicated by the respective curves in the following diagram.

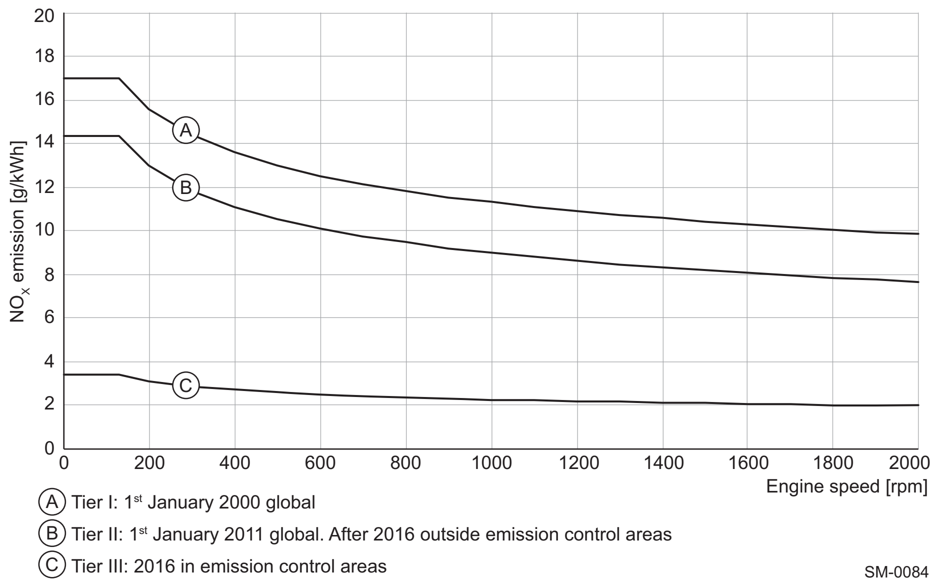


Figure 7-1 Speed dependent maximum allowable average of NO_X emissions

NO_X Technical Code

The rules and procedures for demonstrating and verifying compliance with this regulation are laid down in the NO_X Technical Code, which is part of Annex VI and is largely based on the latest revision of ISO 8178.

7.1.2 Selective catalytic reduction

Selective catalytic reduction systems (SCR) are used on board ships to ensure that the exhaust gas emissions comply with the Tier III NO_X regulations stipulated by the International Maritime Organization (IMO).

SCR technology is based on the reduction of nitrogen oxides (NO_X) by means of a reductant (typically ammonia, generated from urea) at the surface of a catalyst situated in a reactor.

The drawings relevant for the SCR system (DG 9726) are provided on the WinGD corporate webpage under the following link:
Exhaust and ventilation system

Low-pressure SCR
The SCR reactor is located on the low-pressure side, after the turbine.
For low-pressure SCR applications WinGD has developed a 2-stroke engine interface specification that complies with the known low-pressure SCR system providers. Low-pressure SCR is typically larger in volume than high-pressure SCR, but more flexible in installation position, as any after-turbocharger position is acceptable.

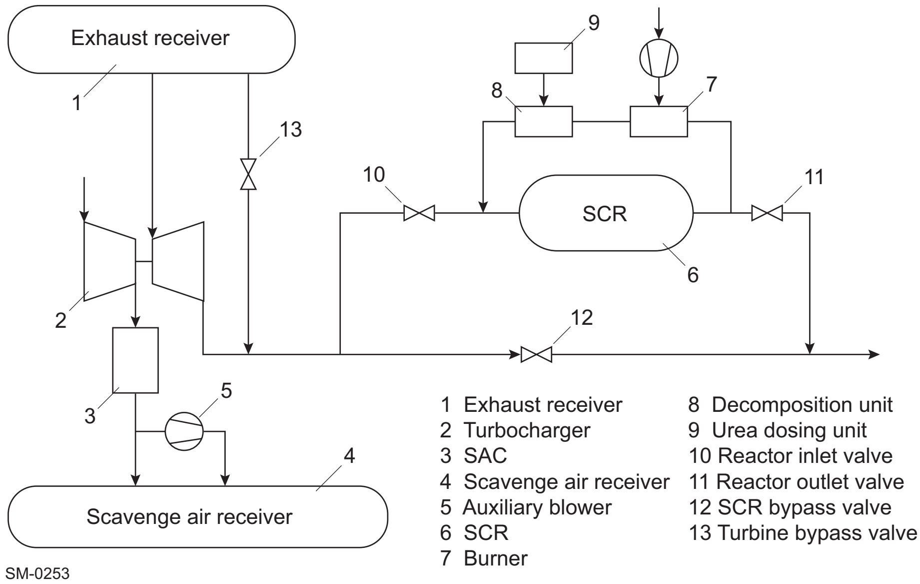


Figure 7-2 Low-pressure SCR — arrangement

High-pressure SCR

The SCR reactor is located on the high-pressure side, before the turbine. Integrating the SCR reactor before the turbine allows the reactor to be designed in the most compact way due to the higher density of the exhaust gas.

WinGD has developed and is systematically deploying high-pressure SCR solutions for the complete 2-stroke engine portfolio with single- and multi-turbocharger applications. Furthermore, WinGD allows high-pressure SCR suppliers to interface third-party branded products to the engine, provided that interface specifications are met.

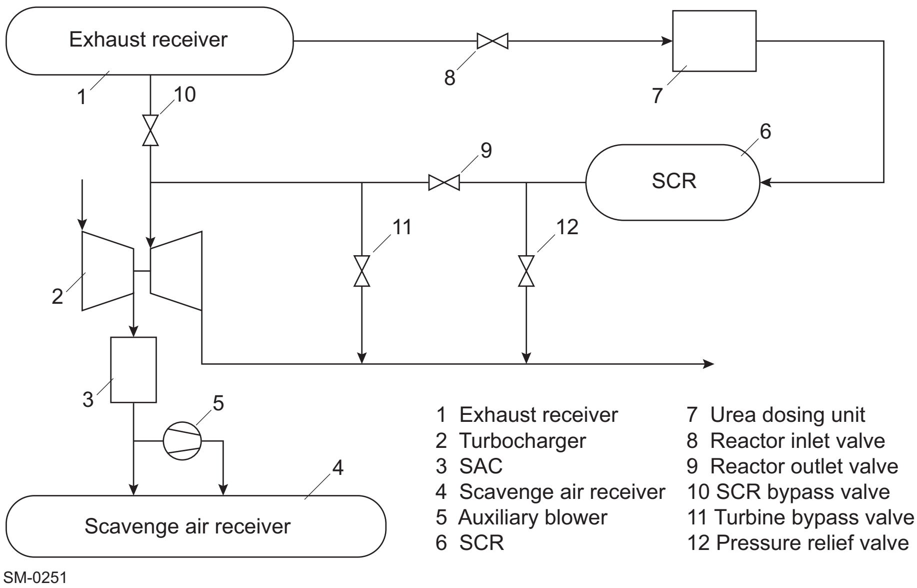


Figure 7-3 High-pressure SCR — arrangement

The **Concept Guidance** for HP-SCR installation (DG 8159) is provided on the WinGD corporate webpage under the following link:
SCR piping guide

7.2 Engine noise

As the ship's crew / passengers must be protected from the effects of machinery space noise, the maximum acceptable noise levels are defined by rules. In general, for new building projects, the latest SOLAS Resolution MSC.337 'Code of Noise Levels Onboard Ships' is applied. This resolution is stepwise introduced since 2014 and fully in force beginning of 2018.

One major change of the new SOLAS MSC.337 compared to the previous Resolution A468(XII) is that the maximum noise level is fixed for a single point, whereas the former Resolution allowed the calculation of average noise levels.

NOTE	The noise level graphs in Figure 7-4, 🗎 7-5, Figure 7-6, 🗎 7-7 and Figure 7-7, 🗎 7-8 show typical values for MCR. As the rating and tuning dependency is marginal, the values can be used for all ratings.

7.2.1 Air-borne noise

Figure 7-4, 🗎 7-5 shows the average surface sound pressure level. The data in the graph are related to:

- Distance of 1 m from engine
- Average values Lp in dB, in comparison with ISO NR-Curves
- Overall average values LpA in dB(A) and expected maximal overall single point values
- Free field conditions

Near the turbocharger (air intake), the maximum measured noise level will normally be 3-5 dB(A) higher than the average noise level of the engine.

Standard noise reduction & additional noise reduction

The present document includes the expected maximum overall value for a single point. Figure 7-4, 🗎 7-5 distinguishes between standard noise reduction and additional noise reduction on turbocharger air side. The turbocharger suppliers are currently developing different silencer solutions to comply with the new noise limit regulation of 110 dB(A) for single point.

NOTE	The single point noise limit of 110 dB(A) for machinery spaces may be exceeded if standard silencers are applied.

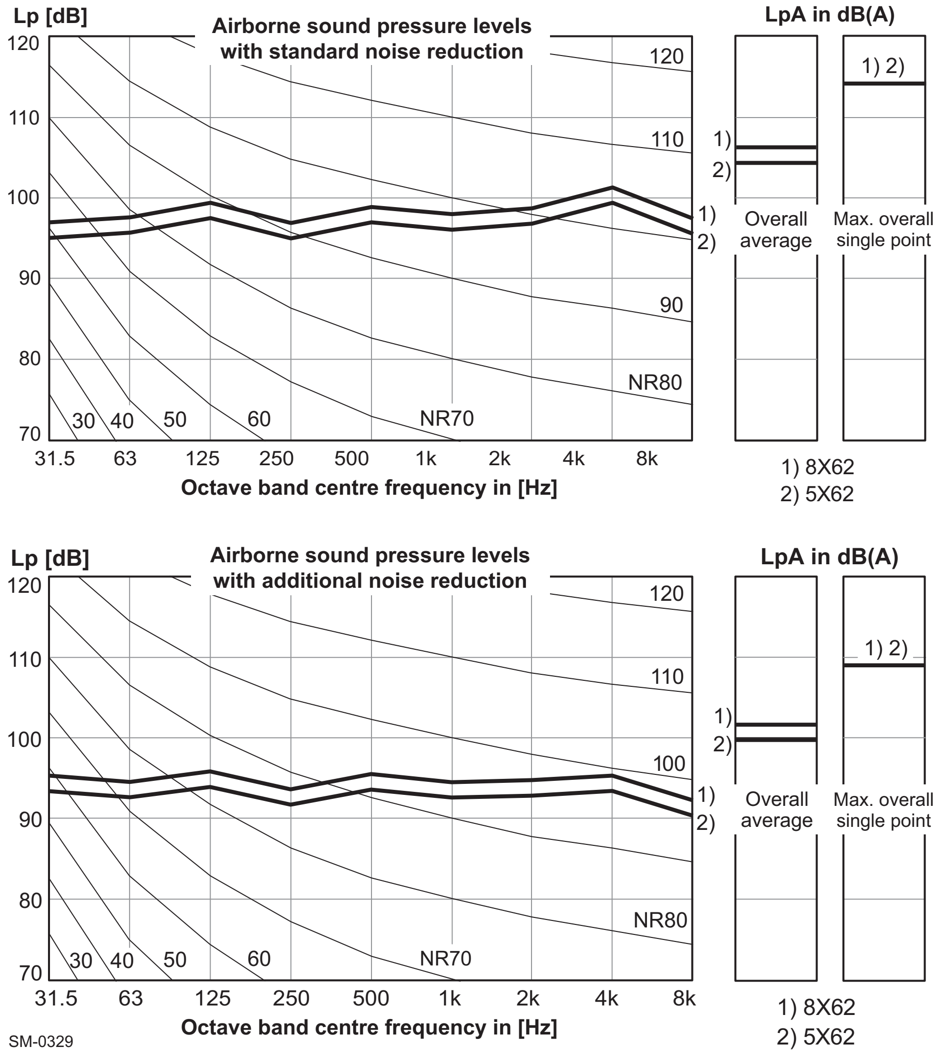


Figure 7-4 Sound pressure level at 1 m distance from engine

7.2.2 Exhaust noise

In the engine exhaust gas system, the sound pressure level at funnel top (see Figure 7-6, 7-7) is related to:

- Distance of 1 m from edge of exhaust gas pipe opening (uptake)
- Angle of 30° to gas flow direction (see Figure 7-5)
- Average values Lp in dB, in comparison with ISO NR-Curves
- Overall average values LpA in dB(A)
- Without boiler, silencer, exhaust gas bypass

Each doubling of the distances from the centre of the duct reduces the noise level by about 6 dB.

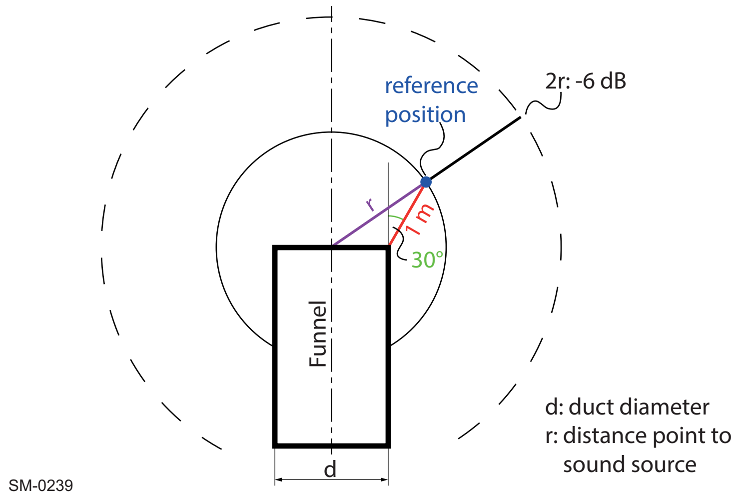


Figure 7-5 Exhaust noise reference point

Depending on the actual noise level allowed on the bridge wing — which is normally between 60 and 70 dB(A) — a simple flow silencer of the absorption type may be placed after the exhaust gas boiler, if the noise reduction of the boiler is not sufficient.
For installations with exhaust gas bypass, a silencer in the main engine exhaust line may be considered.

Dimensioning

The silencers are to be dimensioned for a gas velocity of approx. 35 m/s with a pressure loss of approx. 2 mbar at specified CMCR.

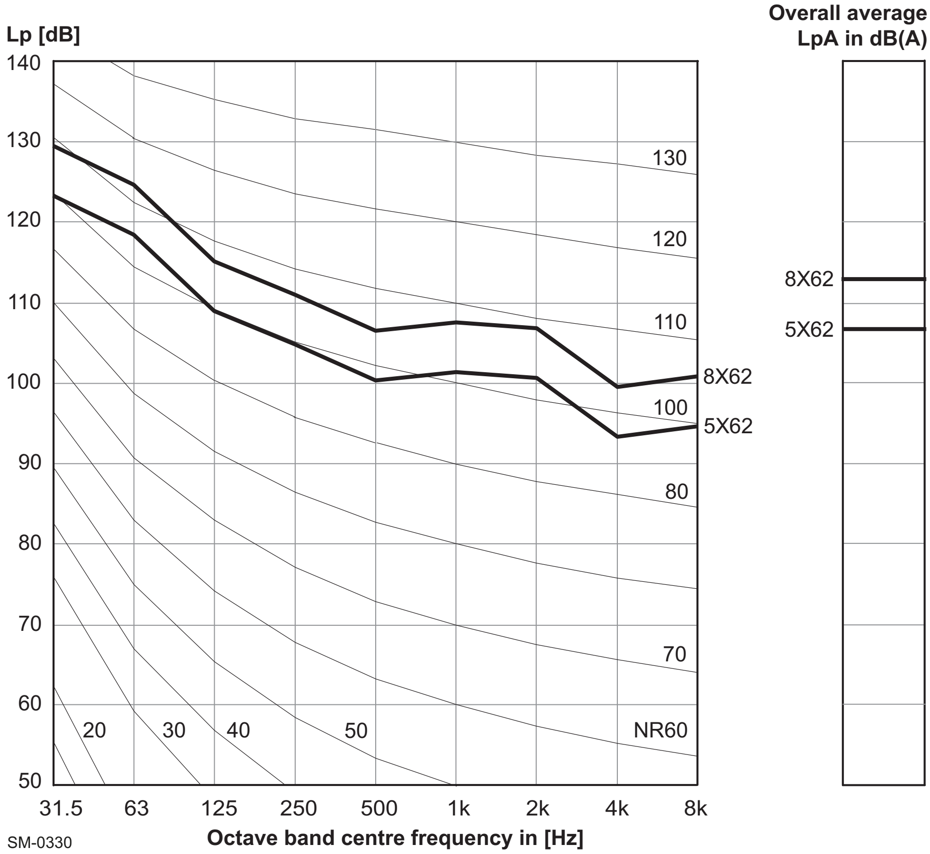


Figure 7-6 Sound pressure level at funnel top of exhaust gas system

7.2.3 Structure-borne noise

The vibrational energy is propagated via engine structure, bedplate flanges and engine foundation to the ship’s structure, which starts to vibrate and thus emits noise.

The sound pressure levels in the accommodations can be estimated with the aid of standard empirical formulas and the vibration velocity levels.

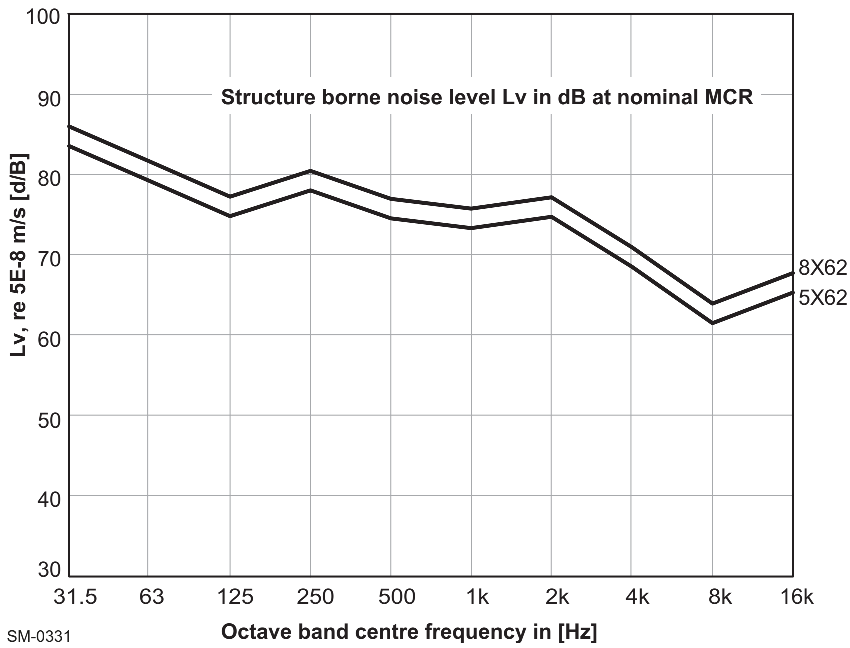


Figure 7-7 Structure-borne noise level at engine feet vertical

8 Engine Dispatch

This chapter describes the provisions to be made for transporting the engine from the engine builder to the shipyard or final destination.

Engines are transported complete or as part assemblies, depending on the terms of contract.

8.1 Engines to be transported as part assemblies

- Engines to be transported as part assemblies have to be systematically disassembled and cleaned using dry cloths.
- Each item is to be clearly identified with ‘paint ball’ pen, similar indelible marker ink, or figure and letter stamps.
- To ensure correct reassembly and eliminate the risk of parts from one cylinder unit being fitted to another by mistake, it is indispensable that bearings and running gear are clearly marked cylinder by cylinder.

8.2 Protection of disassembled engines

All parts have to be protected against damage by careful crating and from corrosion by applying rust preventing oils or paper.

For further details refer to the latest version of the relevant **Guideline** (DG 0345), which is provided on the WinGD corporate webpage under the following link: *Guideline for engine protection*

8.3 Removal of rust preventing oils after transport

8.3.1 Internal parts

The rust preventing oils applied to the internal parts of an assembled engine have properties similar to lubricating oils. As they do not contain thickening agents of wax type they will wash off easily and mix without causing harm to the engine or its systems.

8.3.2 External parts

Wax type rust preventing oils applied to exposed surfaces of engine components contain thickening agents of wax, forming an anti-corrosion coating when applied. This coating has to be washed off with gas oil, kerosene or white spirit.

9 Appendix

The Appendix gives an overview of the relevant classification societies and lists acronyms mentioned throughout this document in alphabetical order. Tables of SI dimensions and conversion factors can also be found here.

9.1 Classification societies

Table 9-1 List of classification societies

IACS	International Association of Classification Societies		
ABS	American Bureau of Shipping	KR	Korean Register
BV	Bureau Veritas	LR	Lloyd’s Register
CCS	Chinese Classification Society	NK	Nippon Kaiji Kyokai
CRS	Croatian Register of Shipping	PRS	Polski Rejestr Statkow
DNV-GL	DNV-GL[a)]	RINA	Registro Italiano Navale
IRS	Indian Register of Shipping	RS	Russian Maritime Register of Shipping

a) The rule books of Det Norske Veritas and Germanischer Lloyd are still valid until further notice.

9.2 List of acronyms

Table 9-2 List of acronyms

ALM	Alarm	EM	Engine margin
AMS	Alarm and Monitoring System Attended machinery space	EMA	Engine management & automation
BFO	Bunker fuel oil	FCM	Flex control module
BN	Base number	FIA	Fuel ignition analysis
BSEC	Brake specific energy consumption	FPP	Fixed pitch propeller
BSEF	Brake specific exhaust gas flow	FQS	Fuel quality setting
BSFC	Brake specific fuel consumption	FW	Freshwater
CCAI	Calculated carbon aromaticity index	GTD	General Technical Data (program)
CCR	Conradson carbon	HFO	Heavy fuel oil
CCW	Cylinder cooling water	HMI	Human-machine interface
CEN	European Committee for Standardization www.cen.eu	HP	High pressure
CFR	Certified flow rate	HT	High temperature
CMCR	Contracted maximum continuous rating (Rx)	IACS	Int. Association of Classification Societies www.iacs.org.uk
CPP	Controllable pitch propeller	iELBA	Integrated electrical balancer
CSR	Continuous service rating (also designated NOR or NCR)	IMO	International Maritime Organization www.imo.org
DAH	Differential pressure alarm, high	IPDLC	Integrated power-dependent liner cooling
DBT	Delta bypass tuning	ISO	International Organization for Standardization www.iso.org
Delta	Delta tuning	LAH	Level alarm, high
DENIS	Diesel engine control and optimising specifica- tion	LAL	Level alarm, low
DFO	Diesel fuel oil, covering MGO and MDO, i.e. DMA, DMB, DMZ	LCV	Lower calorific value
DG	Design group	LDU	Local display unit
DMA, DMB, DMZ	Diesel oil quality grades as per ISO 8217	LFO	Light fuel oil
ECA	Emission control area	LHV	Lower heating value
ECR	Engine Control Room	LLT	Low load tuning
ECS	Engine Control System	LO	Lubricating oil
EEDI	Energy efficiency design index	LP	Low pressure
EIAPP	Engine International Air Pollution Prevention	LR	Light running margin

LSL	Level switch, low	SAE	Society of Automotive Engineers www.sae.org
LT	Low temperature	SCR	Selective catalytic reduction
MARPOL	International Convention for the Prevention of Pollution from Ships	SG	Shaft generator
MCR	Maximum continuous rating (R1)	SHD	Shut-down
MDO	Marine diesel oil	SIB	Shipyard interface box
MEP	Mean effective pressure	SLD	Slow-down
MGO	Marine gas oil	SM	Sea margin
MIDS	Marine Installation Drawing Set	SMCR	Specified maximum continuous rating
MIM	Marine Installation Manual	SOLAS	Int. Convention for the Safety of Life at Sea
NAS	National Aerospace Standard	SPP	Steam production power
NCR	Nominal continuous rating	SSU	Saybolt seconds, universal
NOR	Nominal operation rating	Std	Standard tuning
NO_X	Nitrogen oxides	SW	Seawater
NR-Curve	ISO noise rating curve	TBO	Time between overhauls
OM	Operational margin	TC	Turbocharger
PAL	Pressure alarm, low	tEaT	Temperature exhaust gas after turbocharger
PCS	Propulsion Control System	tEbE	Temperature exhaust gas before economiser
PI	Proportional plus integral	TVC	Torsional vibration calculation
PLS	Pulse Lubricating System (cylinder liner)	ULO	Used lubricating oil
PRU	Power related unbalance	UMS	Unattended machinery space
PTH	Power take-home	UNIC	Unified Controls
PTI	Power take-in	VEC	Variable exhaust closing
PTO	Power take-off	VI	Viscosity index
PTO-G	Power take-off gear	VIT	Variable injection timing
PUR	Rigid polyurethane	WECS	WinGD Engine Control System
RCS	Remote Control System	WHR	Waste heat recovery
SAC	Scavenge air cooler	WinGD	Winterthur Gas & Diesel Ltd.

9.3 SI dimensions for internal combustion engines

Table 9-3 SI dimensions

Symbol	Definition	SI-Units	Other units
a	Acceleration	m/s^2	
A	Area	m^2, cm^2, mm^2	
BSFC	Brake specific fuel consumption	kg/J, kg/(kWh), g/(kWh)	
c	Specific heat capacity	J/(kgK)	
C, S	Heat capacity, entropy	J/K	
e	Net calorific value	J/kg, J/m^3	
E	Modulus of elasticity	N/m^2, N/mm^2	
F	Force	N, MN, kN	
f, v	Frequency	Hz, 1/s	
I	Current	A	
I, J	Moment of inertia (radius)	kgm^2	
l, L	Length	m, cm, mm	
I_a, I_p	Second moment of area	m^4	
K	Coefficient of heat transfer	$W/(m^2K)$	
L	Angular momentum	Nsm	
$L_{(A)TOT}$	Total A noise pressure level	dB	
$L_{(LIN)TOT}$	Total LIN noise pressure level	dB	
L_{OKT}	Average spatial noise level over octave band	dB	
m	Mass	t, kg, g	
M, T	Torque moment of force	Nm	
N, n	Rotational frequency	1/min, 1/s	rpm
p	Momentum	Nm	
p	Pressure	N/m^2, bar, mbar, kPa	1 bar = 100 kPa 100 mmWG = 1 kPa
P	Power	W, kW, MW	
q_m	Mass flow rate	kg/s	
q_v	Volume flow rate	m^3/s	
t	Time	s, min, h, d	
T, Θ, t, θ	Temperature	K, °C	
U	Voltage	V	
V	Volume	m^3, dm^3, l, cm^3	
v, c, w, u	Velocity	m/s, km/h	Kn

Symbol	Definition	SI-Units	Other units
W, E, A, Q	Energy, work, quantity of heat	J, kJ, MJ, kWh	
Z, W	Section modulus	m^3	
ΔT, ΔΘ, ...	Temperature interval	K, °C	
α	Angular acceleration	rad/s^2	
α	Linear expansion coefficient	1/K	
α, β, γ, δ, φ	Angle	rad, °	
γ, σ	Surface tension	N/m	
η	Dynamic viscosity	Ns/m^2	
λ	Thermal conductivity	W/(mK)	
ν	Kinematic viscosity	m^2/s	cSt, RW1
ρ	Density	kg/m^3, kg/dm^3, g/cm^3	
σ, τ	Stress	N/m^2, N/mm^2	
ω	Angular velocity	rad/s	

9.4 Approximate conversion factors

Table 9-4 Conversion factors

Length	1 in			=	25.4 mm
	1 ft	=	12 in	=	304.8 mm
	1 yd	=	3 feet	=	914.4 mm
	1 statute mile	=	1760 yds	=	1609.3 m
	1 nautical mile	=	6080 feet	=	1853 m
Mass	1 oz			=	0.0283 kg
	1 lb	=	16 oz	=	0.4536 kg
	1 long ton			=	1016.1 kg
	1 short ton			=	907.2 kg
	1 tonne			=	1000 kg
Volume (fluids)	1 Imp. pint			=	0.568 l
	1 U.S. pint			=	0.473 l
	1 Imp. quart			=	1.136 l
	1 U.S. quart			=	0.946 l
	1 Imp. gal			=	4.546 l
	1 U.S. gal			=	3.785 l
	1 Imp. barrel	=	36 Imp. gal	=	163.66 l
	1 barrel petroleum	=	42 U.S. gal	=	158.98 l

Force	1 lbf (pound force)	=	4.45 N
Pressure	1 psi (lb/sq in)	=	6.899 kPa (0.0689 bar)
Velocity	1 mph	=	1.609 km/h
	1 knot	=	1.853 km/h
Acceleration	1 mphps	=	0.447 m/s^2
Temperature	1 °C	=	0.55 x (°F -32)
Energy	1 BTU	=	1.06 kJ
	1 kcal	=	4.186 kJ
Power	1 kW	=	1.36 bhp
	1 kW	=	860 kcal/h
Volume	1 in^3	=	16.4 cm^3
	1 ft^3	=	0.0283 m^3
	1 yd^3	=	0.7645 m^3
Area	1 in^2	=	6.45 cm^2
	1 ft^2	=	929 cm^2
	1 yd^2	=	0.836 m^2
	1 acre	=	4047 m^2
	1 sq mile (of land) = 640 acres	=	2.59 km^2

Winterthur Gas & Diesel in brief

Winterthur Gas & Diesel Ltd. (WinGD) is a leading developer of two-stroke low-speed gas and diesel engines used for propulsion power in merchant shipping. WinGD's target is to set the industry standard for reliability, efficiency and environmental friendliness. WinGD provides designs, licences and technical support to manufacturers, shipbuilders and ship operators worldwide. The engines are manufactured under licence in four shipbuilding countries. WinGD has its headquarters in Winterthur, Switzerland, where its activities were founded in 1898.

www.wingd.com

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