

Marine Installation Manual

Document ID	DBAB646817-
Date of issue	28 February 2017
Product	WinGD X92
Version	a3

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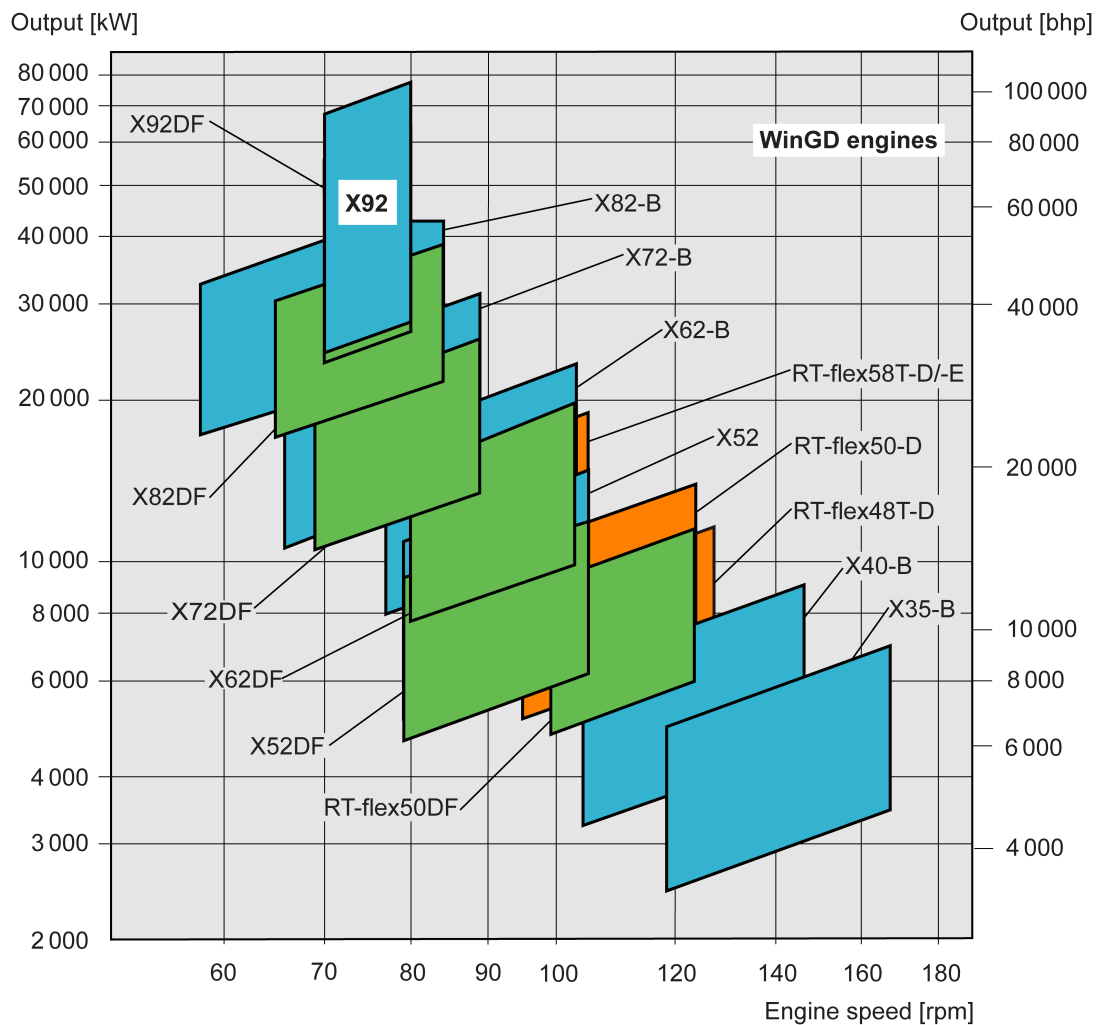
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Preface

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

This manual is only designed for persons dealing with this engine.



This manual provides the information required for the layout of marine propulsion plants. It 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.

Its content is subject to the understanding that any data and information herein have been prepared with care and to the best of our knowledge. We do not, however, assume any liability with regard to unforeseen variations in accuracy thereof or for any consequences arising therefrom.

NOTE



Attention is drawn to the following:

- All data are related to engines compliant with **the regulations according to the Revised Marpol Annex VI and NO_x Technical code 2008 IMO Tier II.**
- The engine performance data (rating R1) refer to *GTD*.
- The engine performance data (BSFC, BSEF and TEaT) and other data can be obtained from the *GTD*. The *GTD* can be downloaded from our Portals.

Winterthur Gas & Diesel Ltd.
Product Information
Schützenstrasse 1-3
PO Box 414
CH-8401 Winterthur
Switzerland
Tel: +41 52 262 07 14
Fax: +41 52 262 07 18
www.wingd.com

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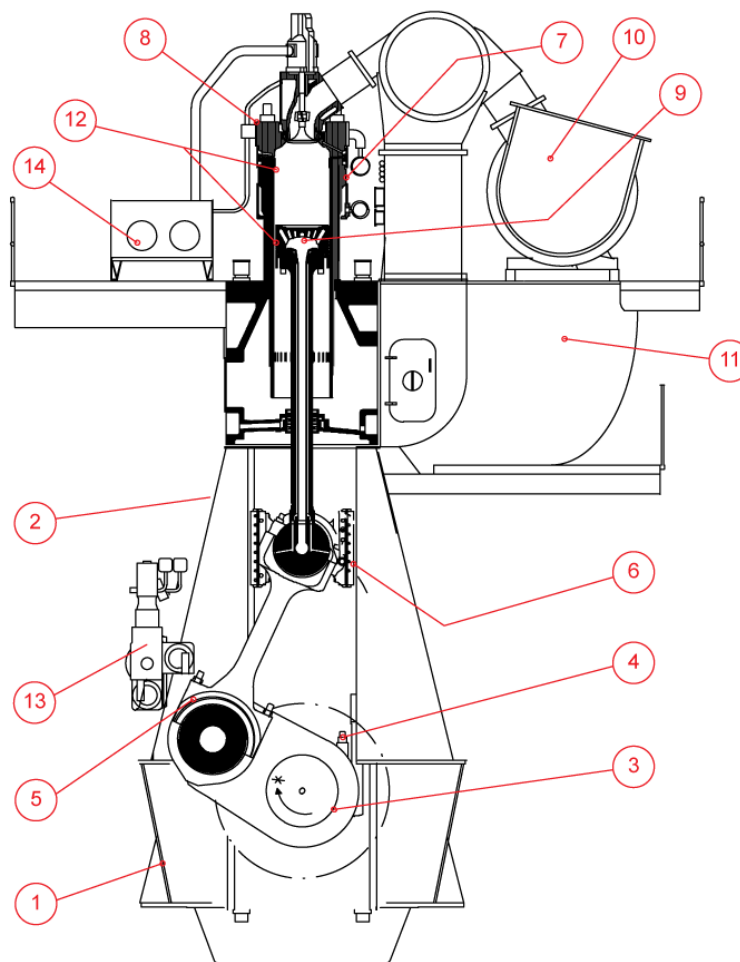
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1. Engine Characteristics

The WinGD X92 engine is a camshaftless low-speed, reversible and rigidly direct-coupled two-stroke engine featuring common rail injection.

Electronic control of the key engine functions such as exhaust valve drives, engine starting and cylinder lubrication is effected by the WECS-9520 Engine Control System. WECS-9520 also ensures volumetric control of fuel injection.

The WinGD X92 is designed for running on a wide range of fuels, from marine diesel oil (MDO) to heavy fuel oils (HFO) of different qualities.



Remark: * Direction of rotation: clockwise as standard (viewed from the propeller towards the engine).

Note: This cross section is considered as general information only

Fig 1.1 Cross section

1	Bedplate	2	Column	3	Crankshaft	4	Main bearing elastic studs
5	Bottom-end bearings	6	Crosshead	7	Cylinder liner	8	Cylinder cover
9	Piston	10	Turbocharging system	11	Scavenging system	12	Pulse Lubricating System (PLS)
13	Supply unit	14	Rail unit (common rail)				

1.1 Primary engine data

Table 1.1.1 Rating points

Bore x stroke: 920 x 3468 [mm]				
No. of cyl.	R1	R2	R3	R4
	Power [kW]			
6	38700	27900	33900	24420
7	45150	32550	39550	28490
8	51600	37200	45200	32560
9	58050	41850	50850	36630
10	64500	46500	56500	40700
11	70950	51150	62150	44770
12	77400	55800	67800	48840
Speed [rpm]				
All cyl.	80	80	70	70
Brake specific diesel fuel consumption (BSFC) [g/kWh] 100% power Brake specific fuel consumption (BSFC) data are quoted for fuel of lower calorific value 42.7 MJ/kg.				
All cyl.	165.8	158.8	165.8	158.8
mep [bar]				
All cyl.	21.0	15.1	21.0	15.1
Lubricating oil consumption (for fully run-in engines under normal operating conditions)				
System oil	approximately 9 kg/cyl. per day			
Cylinder oil	guide feed rate 0.6 g/kWh			
All other reference conditions refer to ISO standard (ISO 3046-1). The following tolerances for BSFC are to be taken into account: + 5% for 100-85% engine power, + 6% for 84-65% engine power and + 7% for 64-50% engine power.				

1.2 Tuning options

As the Flex system 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.

All tuning regimes comply with the IMO Tier II regulations for NO_x emissions.

Data for the tuning regimes are available from the General Technical Data program.

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

Tuning	Description	Application	Additional components
Standard Tuning (ST)	High-load tuning	When ship operates most of the time above 90 % engine load	None
Delta Tuning (DT)	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

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

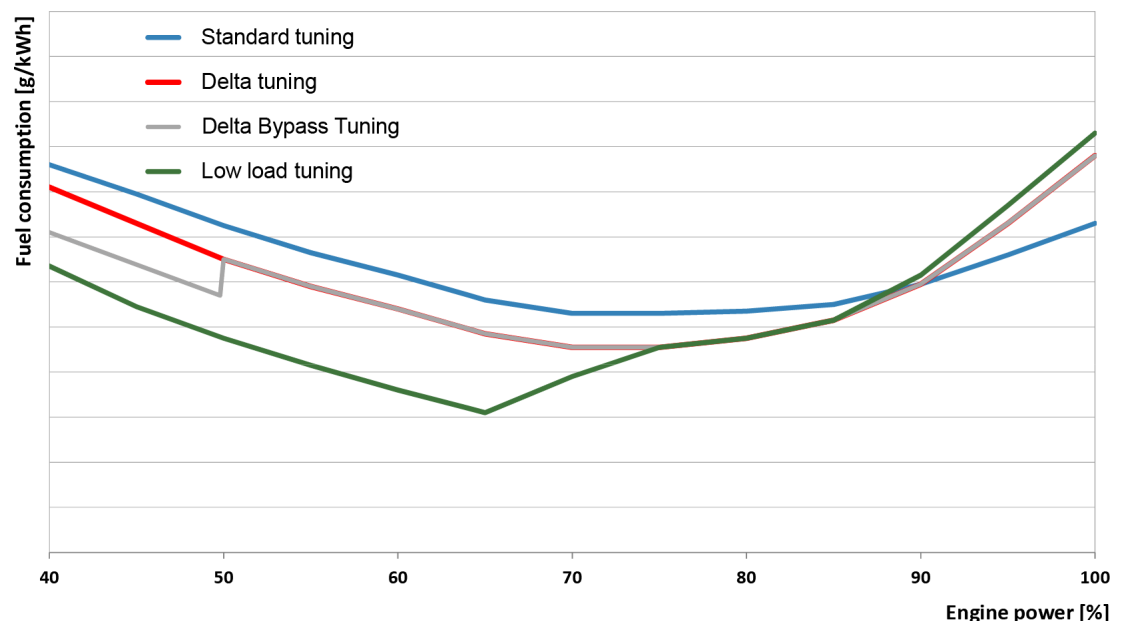


Fig 1.2.1 Typical BSFC curves in relation to engine load

BSFC data for Standard Tuning is given in section Primary engine data. Data for the other tuning options can be obtained from the *GTD* application.

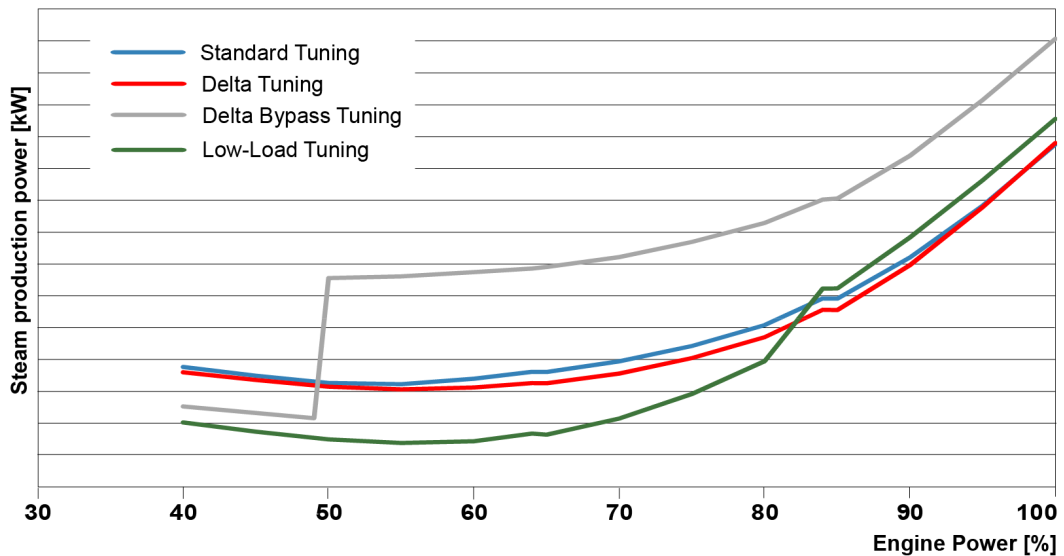


Fig 1.2.2 Steam production power diagram

1.2.1 BSFC and NOx emission

The engine parameters controlling the fuel injection and exhaust valve operational characteristics have to be selected appropriately to allow realising the full potential of the respective tuning, while ensuring compliance with the applicable NOx limit value. Due to the trade-off between BSFC and NOx emissions, the associated increase in NOx 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.

1.2.2 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. This option can be combined with all other available tuning possibilities.

LowTV Tuning shall be considered during the torsional vibration calculation. Winterthur Gas & Diesel Ltd. offers assistance to check if LowTV Tuning can be applied in specific projects.

The following figure shows a comparison of 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.

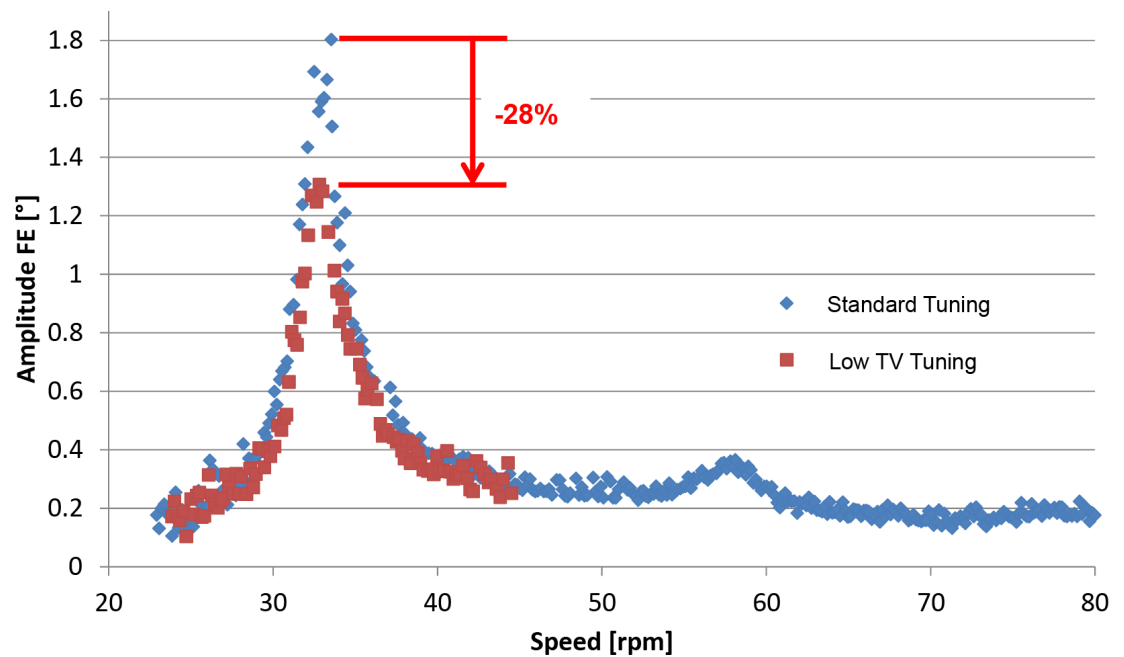


Fig 1.2.2.1 Comparison of torsional vibration amplitudes between Standard Tuning and LowTV tuning

1.2.3

Project specification

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

- The layout of the ancillary systems has to be based on the correct specifications.
- To prepare the software for the Flex system control, the parameters have to be known in due time before commissioning of the engine.
- A waste heat recovery (WHR) solution is available on application basis. To provide the most energy-efficient solution Winterthur Gas & Diesel Ltd. offers customised technical support on demand, considering various aspects of the specific installation like steam pressure, single/double exhaust gas bypass, steam and power turbine configuration, combustion air suction, etc.

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.

1.2.4 Delta Tuning

In realising Delta Tuning, the flexibility of the Flex system in terms of free selection of injection and exhaust valve control parameters, specifically variable injection timing (VIT) and variable exhaust closing (VEC), is used to reduce the BSFC in the part-load range of less than 90 %.

The reduction of BSFC with Delta Tuning is achieved by changing software parameters without modifying any engine parts. The Delta Tuning option needs to be specified at a very early stage of the project.

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. Hence, there is also a slight increase in full load BSFC to maintain compliance of the engine with the IMO regulations.

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.

1.2.5 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 engine specific fuel consumption (BSFC), while still complying with all existing emission legislations.

In particular, DBT is achieved by:

- optimising the engine control system software parameters (different from DT)
- increasing the firing pressure and changing injection timing (compared to DT)
- implementing a specifically designed turbocharging system setup
- adding one (1) exhaust gas waste gate.

Functionality of exhaust gas waste gate

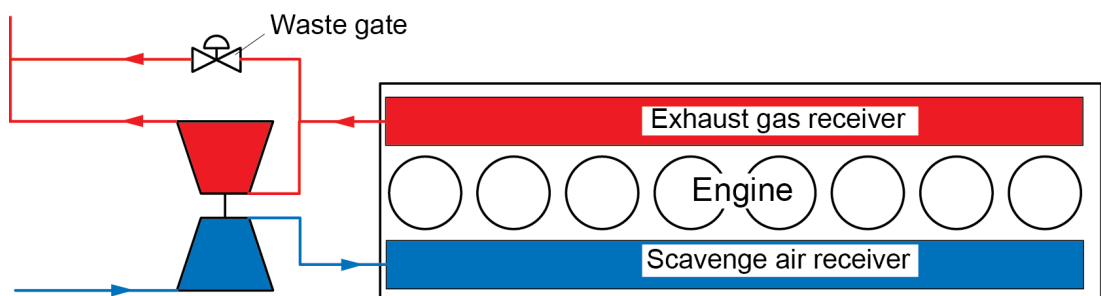


Fig 1.2.5.1 Schematic functional principle of the exhaust gas waste gate

DBT requires the fitting of an exhaust gas waste gate (a pneumatically operated valve, see figure 1.2.5.1) on the exhaust gas receiver before the turbocharger turbine. Exhaust gas blown off through the waste gate is bypassed to the main exhaust uptake.

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 DT. See figure 1.2.5.2.

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.

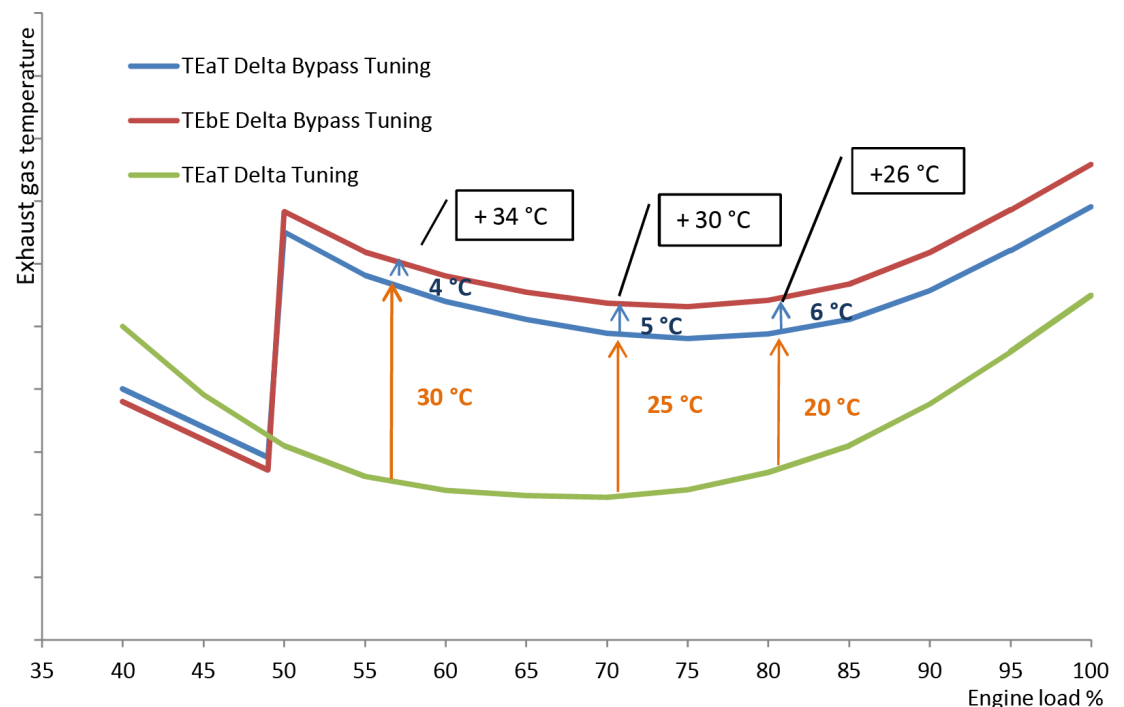


Fig 1.2.5.2 Exhaust gas temperature increase with DBT

Steam production

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.

In such a condition DBT is the most economical tuning option, see figure 1.2.2.

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

1.2.6 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.

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.

NOTE



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

1.2.7 Tuning for de-rated engines

The tuning options are applicable over the entire rating field as illustrated on figure 1.2.7.1.

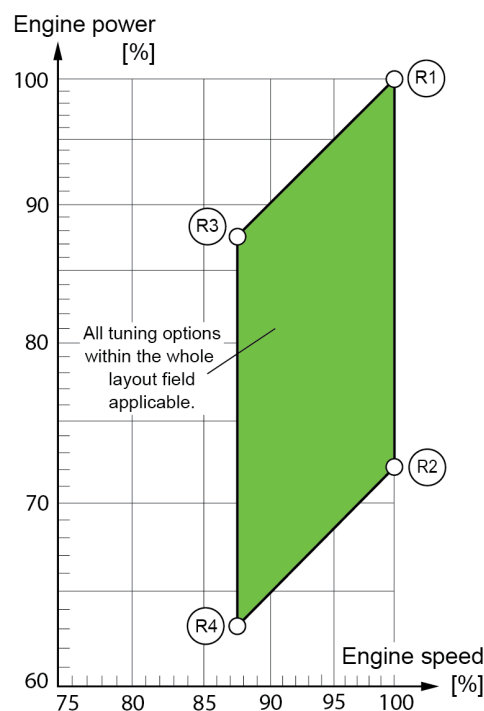


Fig 1.2.7.1 Application area for tuning options

1.2.8 Dual Tuning

The WinGD 2-stroke engines can be built and certified with 'dual tuning', i.e. DT 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. Changing over from one tuning to the other when the engine is in service is a long-term consideration, since the following modifications have to be carried out on the engine:

- Exchange of turbocharger nozzle ring (and diffuser)
- WECS-9520 software, parameter change
- Installation/removal of exhaust gas bypass blind flange (not needed for DBT and LLT)
- Change of orifice size in exhaust gas bypass

An engine cannot be operated with both tunings 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 testbed for both tunings.

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 done 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.

1.3 Main features and parameters

Bore	920 mm
Stroke	3468 mm
Number of cylinders	6 to 12
Power (MCR)	6450 kW/cyl
Speed (MCR)	80 rpm
Mean effect. press.	21.0 bar
Mean piston speed	9.2 m/s

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 white metal bottom-end bearings
- Crosshead with crosshead pin and single-piece large white-metal surface bearings lubricated by the engine lubricating system
- Rigid cast iron cylinder monoblock
- Special grey-cast iron cylinder liners, water cooled
- Pulse 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 1000 bar fuel rail
- Rail unit (common rail): both, common rail injection and exhaust valve actuation, are controlled by quick acting solenoid valves

The WinGD X92 is available with 6 to 12 cylinders rated at 6450 kW/cyl to provide a maximum output of 77400 kW for the 12-cylinder engine (see section 1.1).

Table 1.3.1.1 Overall sizes of engine

No. cyl.	Overall sizes of engine		
	Length [mm]	Piston dismantling height F1 *1) (crank centre – crane hook) [mm]	Dry weight [t]
6	11570	15420	1120
7	13160		1260
8	14750		1380
9	17780		1630
10	19370		1790
11	21030		1960
12	22700		2140
*1) For F2 and F3 (piston removal with double-jib crane) see section 17.1 'Engine dimensions and masses'.			

1.4 The Flex system

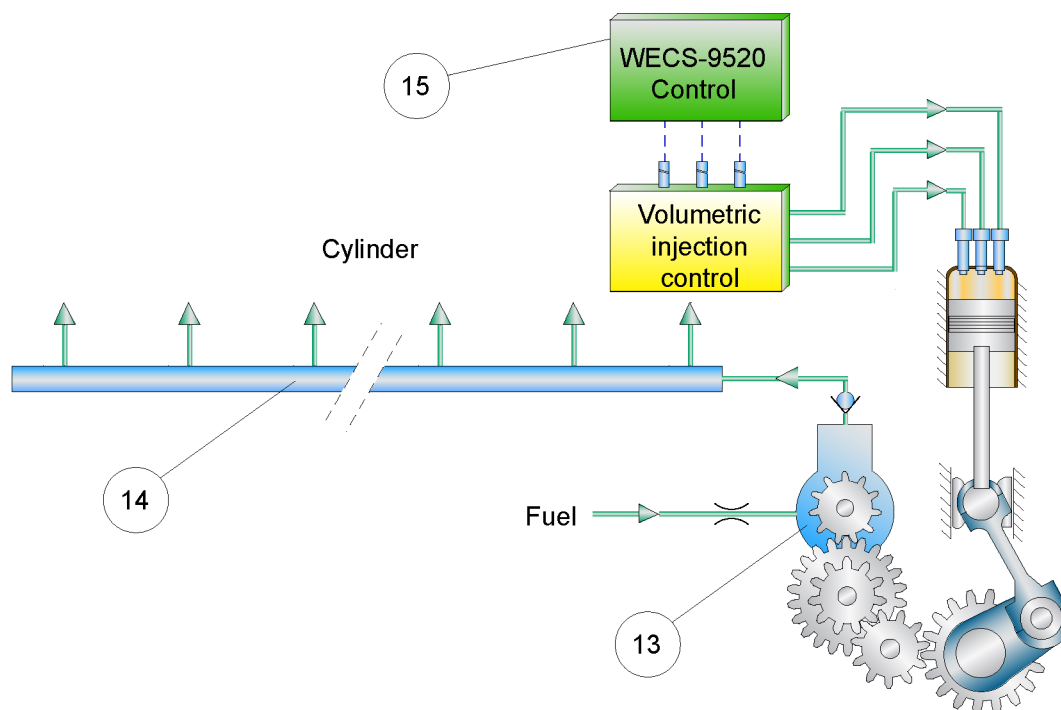


Fig 1.4.1 Flex system parts

13	Supply unit	14	Rail unit (common rail)	15	Control system
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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. Engine Data

2.1 Pressure and temperature ranges

The following table represents a summary of the required pressure and temperature ranges at continuous service rating (CSR). The pump delivery head is obtained by adding the pressure losses in piping system, filters, coolers, valves, etc. and the vertical level pressure difference between pump suction and pressure gauge to the values in the table.

Table 2.1.1 Pressure and temperature ranges

System	Location of measurement	Gauge pressure limit values [bar]		Temperature limit values [°C]		
		Min.	Max.	Min.	Max.	Difference
Freshwater						
Cylinder cooling	Inlet	4.0	5.0	70	-	max. 15
	Outlet each cyl.	-	-	set point 90 *1)		
SAC LT circuit (single-stage SAC)	Inlet Cooler	2.0	4.0	25	36	*2)
	Outlet Cooler	-	-	-	80	
Cylinder lubrication						
Cyl. lubricating oil, static pressure	Inlet Engine	0.4	-	35	50	-
Crosshead lubrication Crosshead LO booster pump optional.						
Crosshead bearing oil	Inlet Engine *3)	10.0	12.0	40	50	-
	Inlet *4)	4.0	5.0	40	50	-
	Outlet	-	-	-	65	-
External fuel pump lubrication Optional						
Bearing oil	Inlet Engine *5)	4.0	5.0	40	50	-
	Inlet Engine *6)			45	55	-
Main lubrication						
Servo oil	Inlet Servo oil pump	4.0 *7)	5.0	-	-	-
Main bearing oil	Supply	4.0	5.0	40	50	-
	Outlet	-	-	-	65	-
Piston cooling oil	Inlet	4.0	5.0	40	50	max. 30
	Outlet	-	-	-	80	
Thrust bearing pads	Pads AHEAD	-	-	-	80	-
Torsional vibration damper (steel spring damper)	Supply	4.0	5.0	-	-	-
	Inlet Damper *8)	2.8	5.0	-	-	-
Axial vibration damper (chamber pressure)	Supply	4.0	5.0	-	-	-
	Monitoring	1.7	-	-	-	-

System	Location of measurement	Gauge pressure limit values [bar]		Temperature limit values [°C]		
		Min.	Max.	Min.	Max.	Difference
TC bearing oil (on engine lub. oil system), ABB A100-L/200-L	Inlet	1.0	2.5	-	-	-
	Outlet	-	-	-	110	-
TC bearing oil (with separate lub. oil system), ABB A100-L/200-L	Inlet	1.3	2.5	-	85	-
	Outlet	-	-	-	130	-
TC bearing oil MHI MET MB	Inlet	0.7	1.5	-	-	-
	Outlet	-	-	-	85	-
Fuel oil						
Booster (injection pump)	Inlet	7.0 *9)	10.0 *10)	-	150	-
After pressure retaining valve	Return	3.0	5.0	-	-	-
Scavenge air						
Intake from engine room (pressure drop, max)	Air filter / silencer	max. 10 mbar		-	-	-
Intake from outside (pressure drop, max)	Ducting and filter	max. 20 mbar		-	-	-
Cooling (pressure drop)	New SAC	max. 30 mbar		-	-	-
	Fouled SAC	max. 50 mbar		-	-	-
Air						
Starting air	Inlet Engine	12	25/30	-	-	-
Control air	Engine inlet (engine internal)	7.0	9.0	-	-	-
Air spring air for exhaust valve	Main distributor (engine internal)	6.0	7.5	-	-	-
		normal 6.5				
Exhaust gas						
Receiver	After each cylinder	-	-	-	515	Dev. ±50 *11)
	Before each TC	-	-	-	515	-
Back pressure in manifold after turbocharger	Design maximum	30 mbar		-	-	-
	Fouled maximum	50 mbar		-	-	-
*1) Tolerance steady state condition: ±2 °C. Tolerance transient condition: ±4 °C. *2) The water flow has to be within the prescribed limits. *3) Pressure range for engines with Crosshead LO booster pump (optional). *4) Pressure range for engines without Crosshead LO booster pump (standard). *5) Temperature range for lubricating oil of properties SAE30. *6) Temperature range for lubricating oil of properties SAE40. *7) Pressure can be 0.8 bar lower than indicated due to the specified max. allowable pressure difference over the fine filter. *8) The alarm value can be different. For the applicable setting value, refer to the specification of the damper manufacturer. *9) At 100 % engine power. *10) In stand-by condition; during commissioning of the fuel oil system the fuel oil pressure is adjusted to 10 bar. *11) Maximum temperature deviation among the cylinders.						

2.2 Operating Conditions

The engine can be operated at full load in the ambient condition range between **reference conditions** and **design (tropical) conditions**.

2.2.1 Reference conditions

The engine performance data, like **BSFC**, **BSEF**, **tE_{at}** and others, are based on **reference conditions**. They 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 temp.	25 °C
Coolant temp. before SAC	25 °C
Barometric pressure	1000 mbar
Relative air humidity	30 %

2.2.2 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 temp.	45 °C
Coolant temp. before SAC	36 °C
Barometric pressure	1000 mbar
Relative air humidity	60 %

2.3 Ancillary system design parameters

The layout of the ancillary systems of the engine 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 water outlet temp.	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. The cylinder water outlet temperature and the oil temperature before engine are system-internally controlled and have to remain at the specified level.

2.4 Engine performance data

The calculation of the performance data (BSFC, BSEF, TEaT, etc.) for any engine power is done with the help of the *GTD* program, which provides characteristic parameters in the whole rating field of the engine.

2.5 Turbocharger and scavenge air cooler

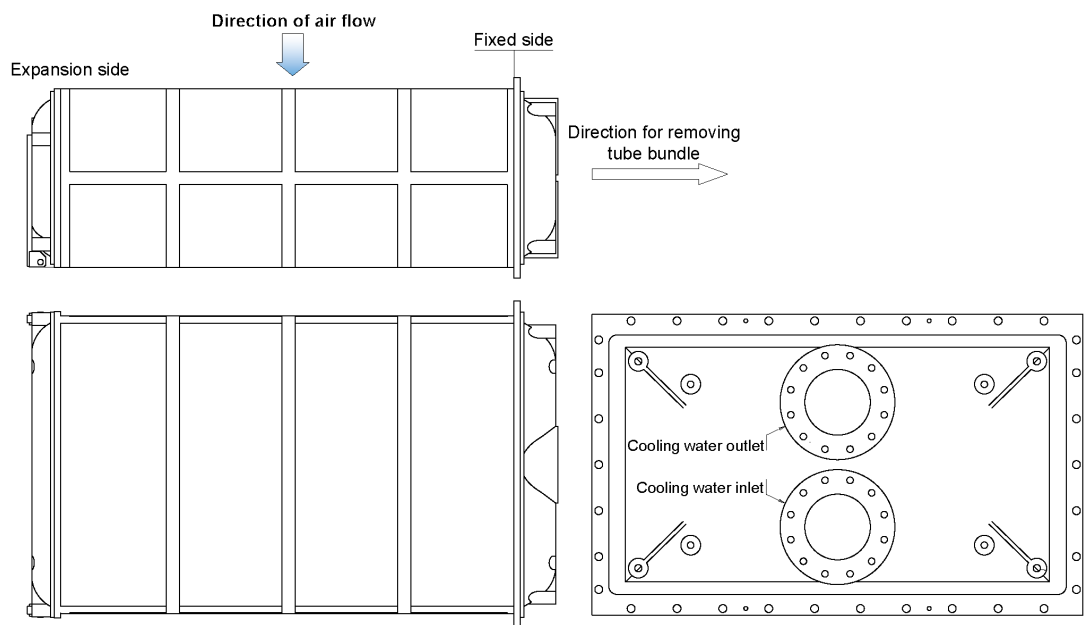


Fig 2.5.1 Scavenge air cooler

2.5.1 Turbocharger weights

Table 2.5.1.1 Turbocharger and scavenge air cooler selection at MCR

Number of cylinders	Turbocharger		Scavenge air cooler
	Number and type	Mass per unit [kg]	Number and type
6	2 x A185L	10500	2 x SAC-B21-SF
	2 x A280-L	8700	2 x SAC-B21-SF
7	2 x A280-L	8700	2 x SAC-B23-SF
8	3 x A180-L	7500	3 x SAC-B20-SF
	2 x A285-L *1)	--	2 x SAC-B25-SF
	2 x A275-L	6300	2 x SAC-B20-SF
9	3 x A185-L	10500	3 x SAC-B21-SF
	2 x A285-L *1)	--	2 x SAC-B26-SF
	3 x A280-L	8700	3 x SAC-B21-SF
10	3 x A280-L	8700	3 x SAC-B22-SF
11	3 x A285-L *1)	--	3 x SAC-B24-SF
12	3 x A285-L *1)	--	3 x SAC-B25-SF
For other ratings than MCR and other makes than ABB turbochargers see GTD. *1) Availability not approved, please contact WinGD.			

2.5.2 Air filtration

In the event that the air supply to the machinery spaces has a dust content exceeding 0.5 mg/m³, 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 the 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.

The necessity for installing a dust filter and the choice of filter type depends mainly on the concentration and composition of the dust in the suction air. Where the suction air is expected to have a dust content of 0.5 mg/m³ or more, the engine must be protected by filtering this air before entering the engine, e.g. on coastal vessels or vessels frequenting ports having high atmospheric dust or sand content.

Winterthur Gas & Diesel Ltd. advises to install a filtration unit for the air supplies to the diesel engines and general machinery spaces on vessels regularly transporting dust creating cargoes, such as iron ore and bauxite.

Table 2.5.2.1 Guidance for air filtration

Dust concentration in ambient air			
Normal Most frequent particle sizes	Normal shipboard requirement Short period < 5 % of running time, < 0.5 mg/m ³	Alternatives necessary in very special circumstances	
		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.	

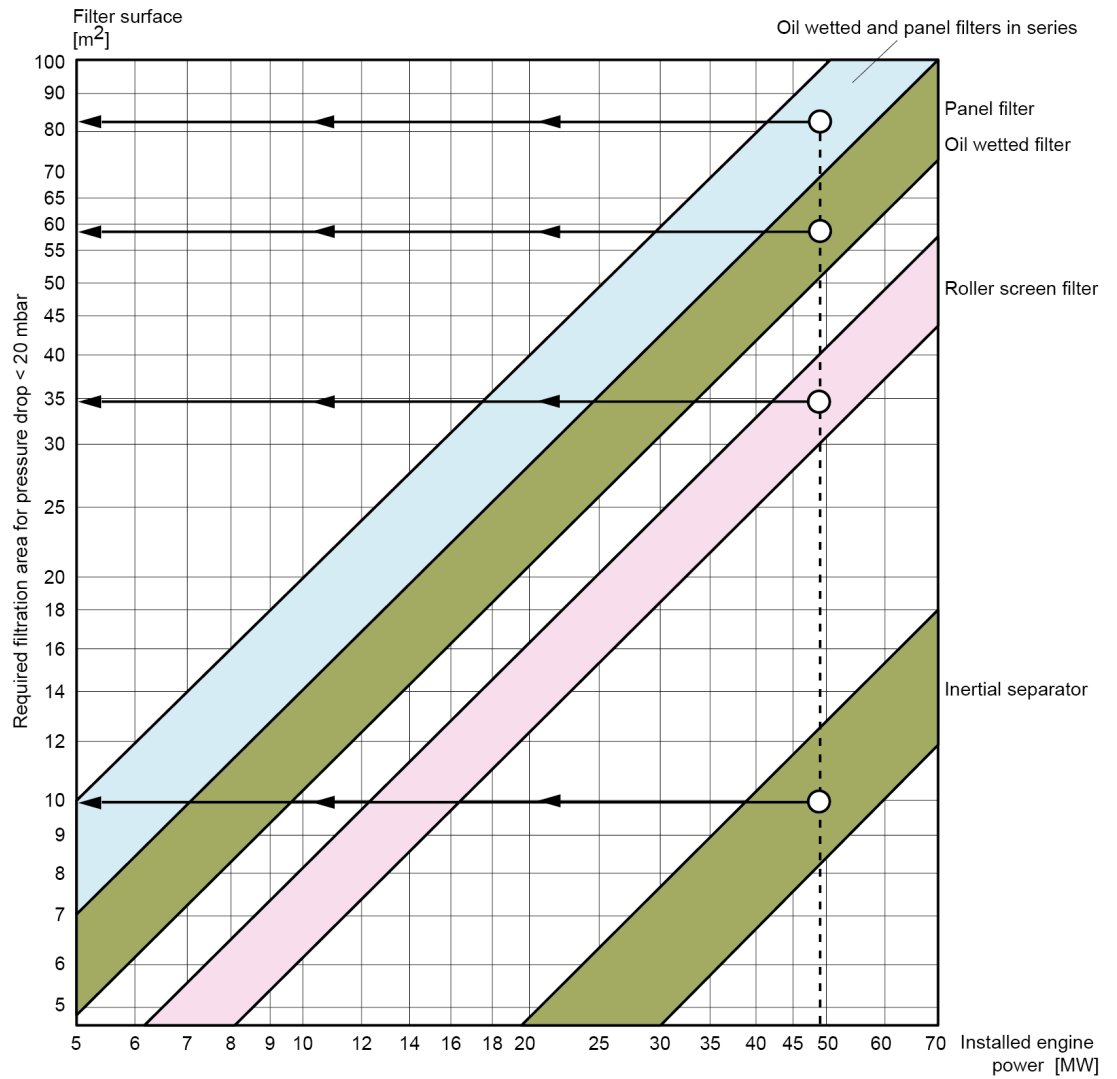


Fig 2.5.2.1 Air filter size (example for 8-cyl. engine)

2.6 Electrical power requirement

Table 2.6.1 Electrical power requirement

Device	No. cyl.	Supply	Power requirement
Auxiliary blower *1)	6	460 V / 60 Hz	2 x 113 kW
	7		2 x 142 kW
	8		2 x 181 kW
	9		2 x 181 kW
	10		2 x 229 kW
	11		2 x 229 kW
	12		2 x 229 kW
Turning gear	6	460 V / 60 Hz	11 kW
	7		11 kW
	8		11 kW
	9		15 kW
	10		15 kW
	11		15 kW
	12		15 kW
WECS power supply	6	230 V / 60 Hz	1.4 kW
	7		1.6 kW
	8		1.8 kW
	9		2.0 kW
	10		
	11		
	12		
Propulsion control system	All	24 VDC UPS	acc. to maker's specifications
Additional monitoring devices (e.g. oil mist detector etc.)	All	acc. to maker's specifications	
*1) Minimal electric motor power (shaft output) is indicated. The <i>actual</i> electric power requirement depends on the size, type and voltage/frequency of the installed electric motor. Direct starting or Star-Delta starting to be specified when ordering.			

3. Engine Rating and Load Range

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.

Every engine has a rating field within which the combination of power and speed (= rating) can be selected.

The 'load range' is the admissible area of operation once the **Contract Maximum Continuous Rating (CMCR)** has been determined. To define the CMCR, various parameters need to be considered, such as propulsive power, propeller efficiency, operational flexibility, power and speed margins, possibility of a main-engine driven generator, and the ship's trading patterns. Selecting the most suitable engine is vital to achieving an efficient cost/benefit response to a specific transport requirement.

3.1 Rating field

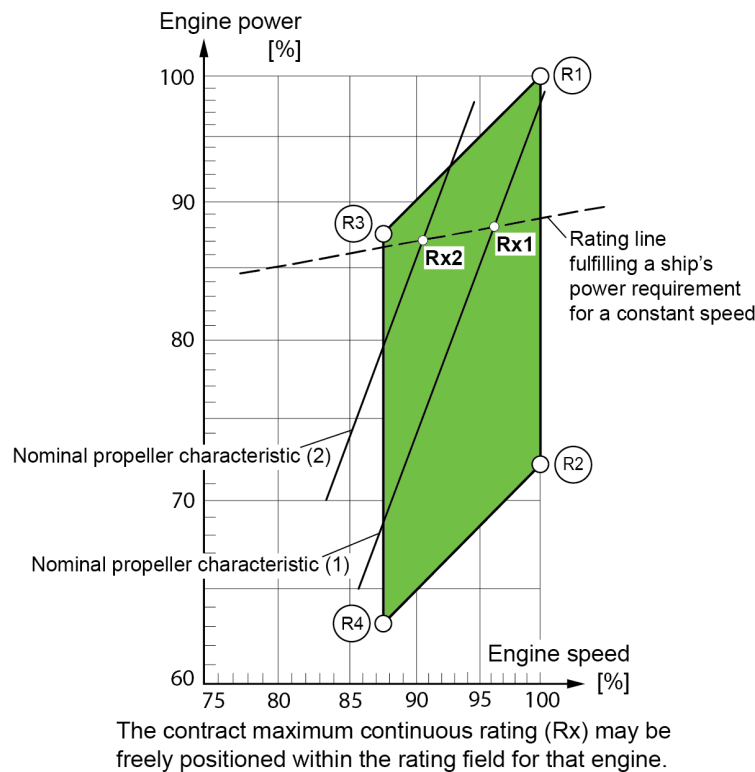


Fig 3.1.1 Rating field

The rating field shown in fig. 3.1.1 is the area of power and engine speed. In this area the contract maximum continuous rating of an engine can be positioned individually to give the wanted combination of propulsive power and rotational speed. Engines within this rating field will be tuned for maximum firing pressure and best efficiency.

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 models. The scales are logarithmic so that exponential curves, such as propeller characteristics (cubic power) and mean effective pressure (mep) curves (first power), are straight lines.

The rating field serves to determine the specific fuel oil consumption, exhaust gas flow and temperature, fuel injection parameters, turbocharger and scavenge air cooler specifications for a given engine.

Calculations for specific fuel consumption, exhaust gas flow and temperature after turbine are explained in further chapters.

3.1.1 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: $PX_2/PX_1 = (n_2/n_1)^\alpha$

PX_j = propulsive power at propeller revolution n_j

n_j = propeller speed corresponding with propulsive power PX_j

0.15 for tankers and general cargo ships up to 10,000 dwt

0.20 for tankers and bulk carriers from 10,000 dwt to 30,000 dwt

$\alpha =$ 0.25 for tankers and bulk carriers larger than 30,000 dwt

0.17 for reefers and container ships up to 3,000 TEU

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.

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 - 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.

3.2 Load range

3.2.1 Propeller curves

To establish the proper location of propeller curves, it is necessary to know the ship's speed to power response. The propeller curve without sea margin (see *Sea margin (SM)*) is, for a ship with a new and clean hull in calm water and weather, often referred to as 'trial condition'.

The curves can be determined by using full-scale trial results from similar ships, algorithms developed by maritime research institutes, or model tank results. Furthermore, it is necessary to define the maximum reasonable diameter of the propeller which can be fitted to the ship. With this information and by applying propeller series such as the 'Wageningen', 'SSPA' (Swedish Maritime Research Association), 'MAU' (Modified AU), etc., the power/speed relationships can be established and characteristics developed.

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

$$P/P_{\text{CMCR}} = (n/n_{\text{CMCR}})^3 \text{ (in which } P = \text{propeller power, } n = \text{propeller speed)}$$

The propeller curve without sea margin is often called the 'light running curve'. The nominal characteristic is a cubic curve through the CMCR-point. (For additional information, refer to section 3.2.4).

The load range diagram shown in figure 3.2.1.1 defines the power/speed limits for the operation of the engine. Percentage values are given as explained in section *Rating field 3.1*; in practice absolute figures might be used for a specific installation project.

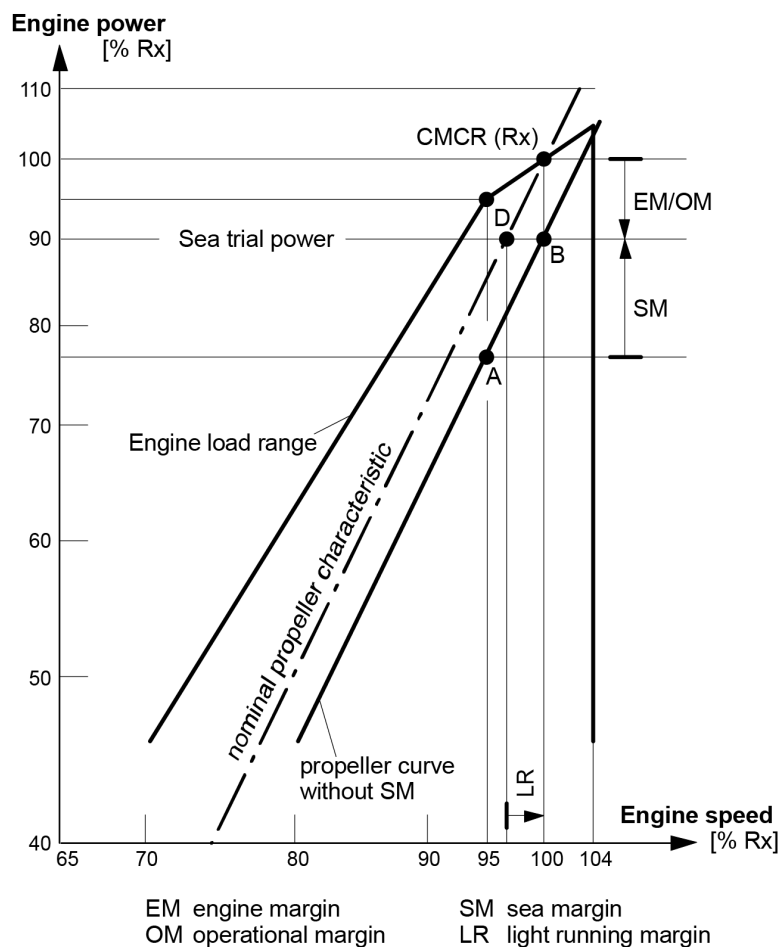


Fig 3.2.1.1 Load range

3.2.2 Sea trial power

The sea trial power must be specified. Figure 3.2.1.1 shows the sea trial power to be the power required for point B on the propeller curve. Often and alternatively, the power required for point A on the curve is referred to as 'sea trial power'.

3.2.3 Sea margin (SM)

The increase in power to maintain a given ship's speed achieved in calm weather (point A in figure 3.2.1.1) under average service condition (point D) is defined as the 'sea margin'. This margin can vary depending on owner's and charterer's expectations, routes, season and schedules of the ship. The location of the reference point A and the magnitude of the sea margin are determined between the shipbuilder and the owner. They are part of the new building contract.

With the help of effective antifouling paints, dry-docking intervals have been prolonged to 4 or 5 years. Therefore, it is still realistic to provide an average sea margin of about 15 % of the sea trial power (refer to 3.2.1.1), unless, as mentioned above, the actual ship type and service route dictate otherwise.

3.2.4 Light running margin (LR)

The sea trial performance (curve 'a') in Fig. 3.2.4.1 should allow for a 4 to 7 % light running of the propeller when compared to the nominal characteristic.

This margin provides a sufficient torque reserve whenever full power must be attained under unfavourable conditions.

Normally, the propeller is hydrodynamically optimised for a point 'B'. The trial speed found for 'A' is equal to the service speed at 'D' stipulated in the contract at 90 % of CMCR.

The recommended light running margin originates from past experience. It varies with specific ship designs, speeds, dry-docking intervals, and trade routes.

NOTE



It is the shipbuilder's responsibility to determine the light running margin large enough so that the load range limits on the left side of the nominal propeller characteristic line are not reached in any service condition (see Fig. 3.2.6.1).

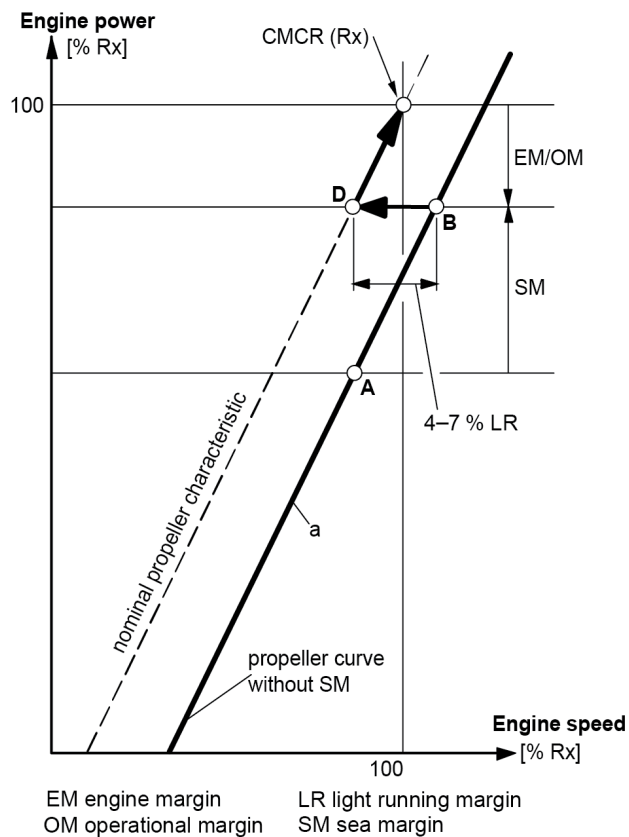


Fig 3.2.4.1 Load diagram for a specific engine, showing the corresponding power and speed margins

Assuming, for example, the following:

- Dry-docking intervals of the ship: 5 years
- Time between engine overhauls: 2 years or more
- Full service speed must be attainable, without surpassing the torque limit, under less favorable conditions and without exceeding 100 % mep

Therefore the required light running margin will be between 4 and 7 %.

This is the sum of the following factors:

- **1.5-2 %** influence of wind and weather with adverse effect on the intake water flow of the propeller. Difference between Beaufort 2, sea trial condition, and Beaufort 4-5, average service condition. For vessels with a pronounced wind sensitivity, i.e. container ships or car carriers, this value will be exceeded.
- **1.5-2 %** increase of ship's resistance and mean effective wake brought about by:
 - Rippling of hull (frame to frame)
 - Fouling of local, damaged areas, i.e. boot top and bottom of the hull
 - Formation of roughness under paint
 - Influence on wake formation due to small changes in trim and immersion of bulbous bow, particularly in ballast condition
- **1 %** frictional losses due to increase in propeller blade roughness and consequent drop in efficiency, e.g. aluminium bronze propellers:
 - New: surface roughness = 12 micron
 - Aged: rough surface but no fouling = 40 micron
- **1 %** deterioration in engine efficiency such as:
 - Fouling of scavenge air coolers
 - Fouling of turbochargers
 - Condition of piston rings
 - Fuel injection system (condition and timing)
 - Increase of back pressure due to fouling of the exhaust gas boiler, etc.

3.2.5 Engine margin (EM) or operational margin (OM)

Most owners specify the contractual ship's loaded service speed at 85 to 90 % of the contract maximum continuous rating. The remaining 10-15 % power can then be used to catch up with delays in schedule or for the timing of dry-docking intervals. This margin is deducted from the CMCR. Therefore, the 100 % power line is found by dividing the power at point D by 0.85-0.90. The graphic approach to find the level of CMCR is illustrated in fig 3.2.1.1.

In the examples two current methods are shown. Figure 3.2.1.1 presents the method of fixing point B and CMCR at 100 % speed, thus obtaining automatically a light running margin B-D of 3.5 %. Figures show the method of plotting the light running margin from point B to point D or D' and then along the nominal propeller characteristic to obtain the CMCR-point. In the examples, the engine power at point B was chosen to be at 90 % and 85 % respectively.

Continuous service rating (CSR=NOR=NCR)

Point A represents power and speed of a ship operating at contractual 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 average weather requires a power/speed combination according to point D. In that case D is the CSR-point.

Contract maximum continuous rating (CMCR = Rx)

By dividing, in our example, the CSR (point D) by 0.90, the 100 % power level is obtained and an operational margin of 10 % is provided. The found point Rx, also designated as CMCR, can be selected freely within the rating field defined by the corner points R1, R2, R3 and R4.

3.2.6 Load range limits

Once an engine is optimised at CMCR (Rx), the working range of the engine is limited for operation by the border lines as shown in Fig. 3.2.6.1.

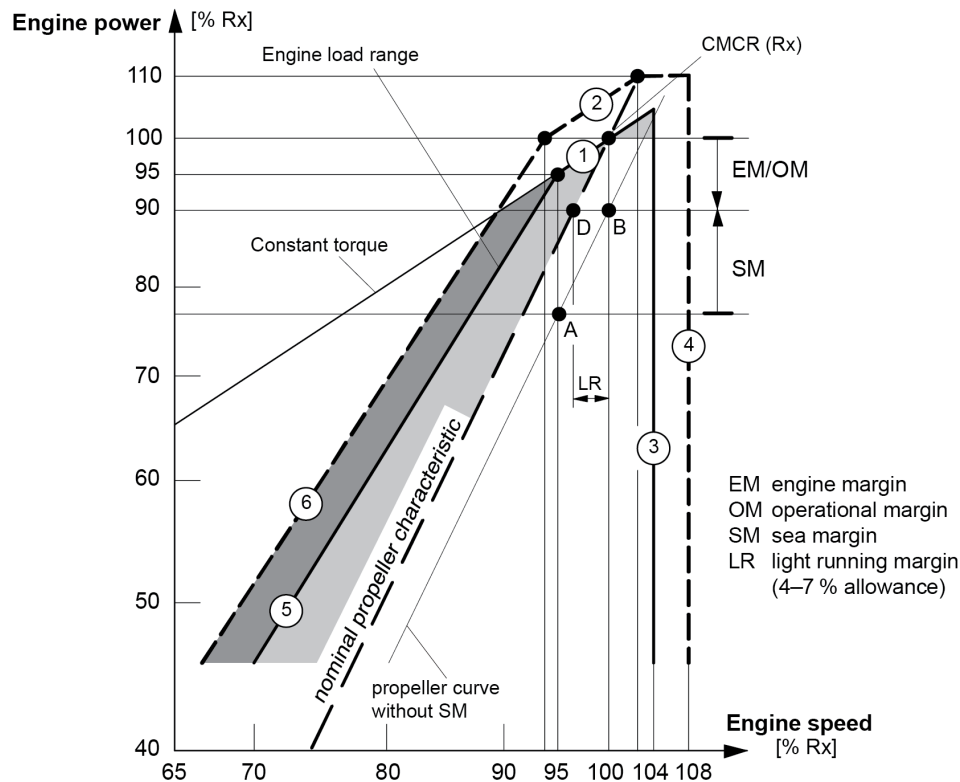


Fig 3.2.6.1 Load range limits

- Line 1 Constant mep or torque line through CMCR from 100 % speed and power down to 95 % power and speed.
 - Line 2 Oerload limit. It is a constant mep line reaching from 100 % power and 93.8 % speed to 110 % power and 103.2 % speed. The latter is the point of intersection between the nominal propeller characteristic and 110 % power.
 - Line 3 104 % speed limit where an engine can run continuously. For Rx with reduced speed ($N_{CMCR} \leq 0.98 N_{MCR}$) this limit can be extended to 106 %, however, 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 Admissible torque limit: it reaches from 95 % power and 95 % speed to 45 % power and 70 % speed and is defined by $P_2/P_1 = (N_2/N_1)^{2.45}$.
- The area formed by lines 1, 3 and 5 is the range within which the engine should be operated. The area limited by the nominal propeller characteristic, 100 % power and line 3 is recommended for continuous operation. The area between the nominal propeller characteristic and line 5 is reserved for acceleration, shallow water and normal operational flexibility.

Line 6 Maximum torque limit in transient conditions defined by $P_2/P_1 = (N_2/N_1)^{2.45}$ through 100 % power and 93.8 % speed.

The area above line 1 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 authorized representatives of the engine builder.

The area between lines 5 and 6 and constant torque line (dark area of Fig. 3.2.6.1), called 'service range with operational time limit', is only applicable for transient conditions, i.e. sea trial or during emergency fast acceleration.

3.2.7 Load range with main-engine driven generator

The load range with main-engine driven generator (PTO), whether it is a shaft generator (S/G) mounted on the intermediate shaft or driven through a power take-off gear (PTO-G) mounted on the intermediate shaft or at engine free end side, is shown by curve 'c' in figure 3.2.7.1.

This curve is not parallel to the propeller characteristic without main-engine driven generator, due to the addition of a constant generator power over most of the engine load. In the example of figure 3.2.7.1, the main-engine driven generator is assumed to absorb 5 % of the nominal engine power.

The CMCR-point is selected on a curve which is parallel to the nominal propeller characteristic without PTO. This curve defines the nominal engine characteristic.

	PTO	S/G	PTO-G
Abbreviations	P ower T ake O ff generator – general, without specification of installation type	PTO, directly mounted on the intermediate shaft, named as S haft G enerator	P TO with power take-off G ear

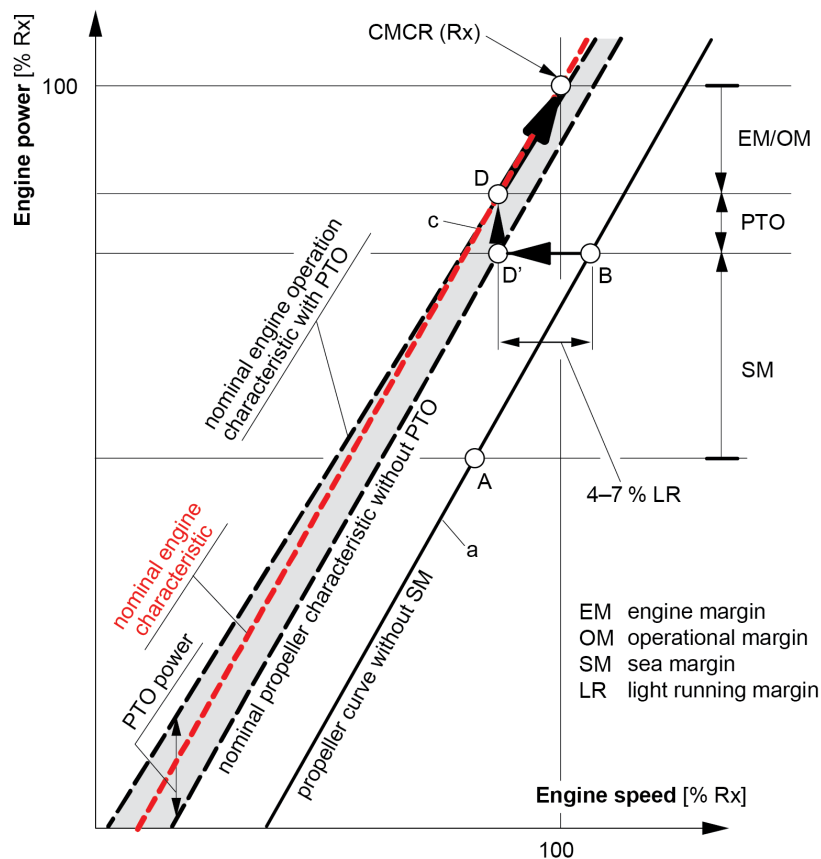


Fig 3.2.7.1 Load range diagram of an engine with main-engine driven generator

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4. GTD

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

Beside the output of characteristic parameters in the whole rating field of an engine, the program provides information on the power requirement for ancillary systems and layout of cooler capacities. 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 program is available as download from the Customer Portal or on the WinGD corporate webpage under the following link:

www.wingd.com/en/media/general-technical-data

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5. Engine Dynamics

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

For successful design, the vibration behaviour needs to be calculated over the whole operating range of the engine and 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

5.1 External 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 minimized.

However, 6-cylinder engines generate second order unbalanced 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 5.1.1 shows the external forces and moments acting on the engine.

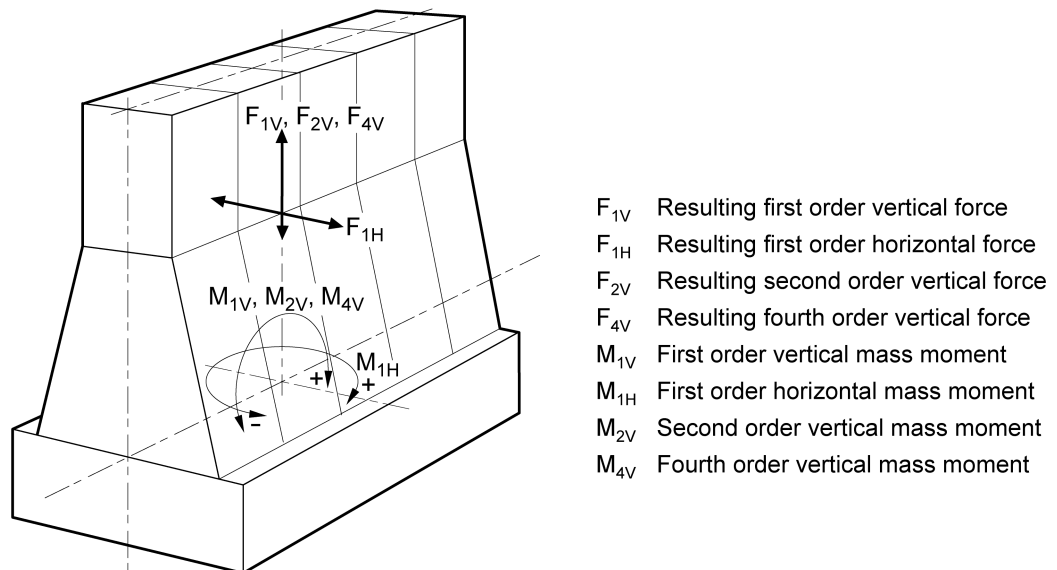


Fig 5.1.1 External forces and moments

Dynamic characteristics data

The latest version of the Dynamic characteristics data such as 'External forces and moments' is provided on the WinGD corporate webpage under the following link:

External mass forces and moments

5.1.1 Balancing free 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} .

5.1.2 Balancing free second order moments

The second order vertical moment (M_{2V}) is higher on 6-cylinder engines compared with 7-12-cylinder engines, the second order vertical moment being negligible for the 7-12-cylinder engines.

For the reduction of the effects of second order moments to acceptable values Winterthur Gas & Diesel Ltd. recommends for 6-cylinder engines the following countermeasure:

- The installation of an electrically driven compensator on the ship's structure (Fig. 5.1.2.1). If no experience is available from a sister ship, it is advisable to establish at the design stage of what kind the ship's vibration will be.

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

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

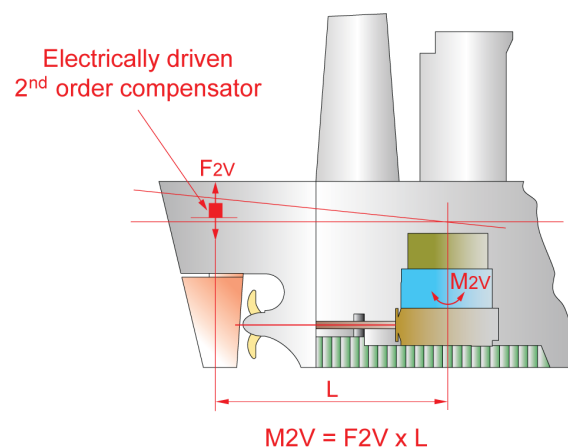


Fig 5.1.2.1 Locating electrically driven compensator

5.1.3 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.

The external mass moments M_1 and M_2 given in section 5.1 are related to R1 speed. For other engine speeds, the corresponding external mass moments are calculated with the following formula: $M_{Rx} = M_{R1} \times (n_{Rx}/n_{R1})^2$

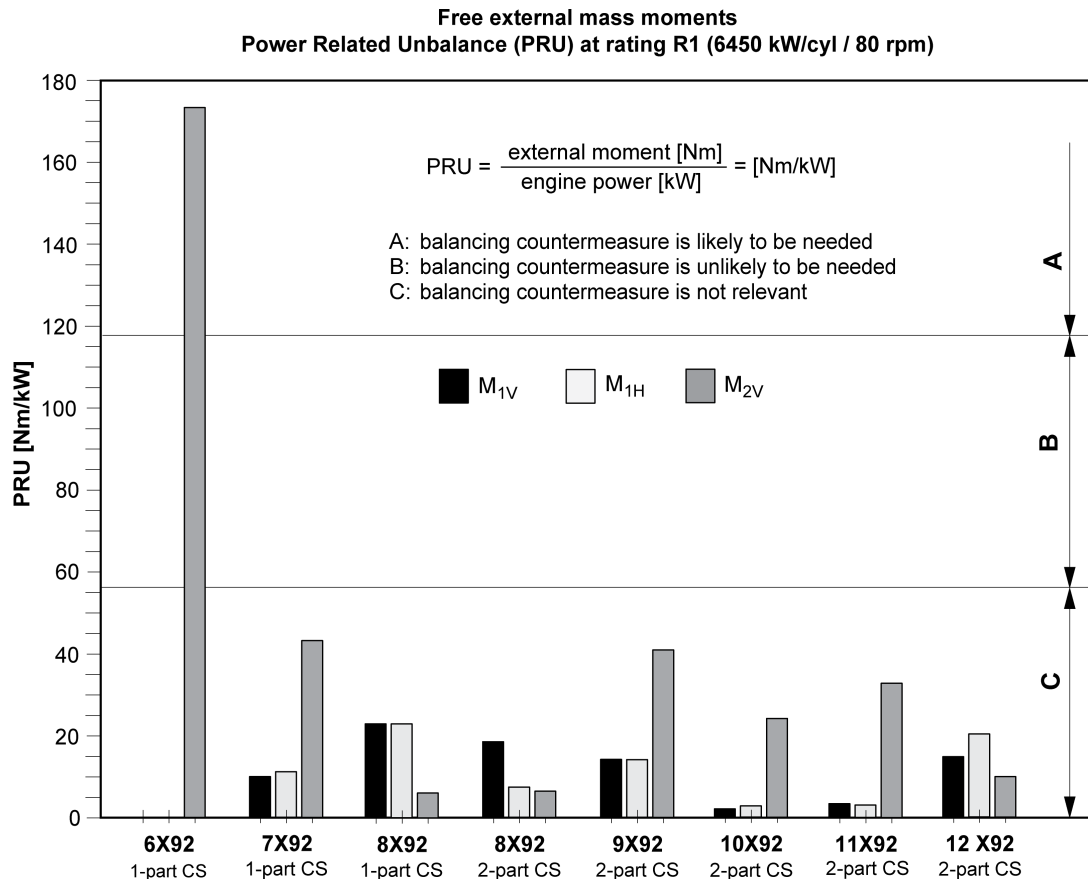


Fig 5.1.3.1 Power related unbalance (PRU)

5.2 Lateral engine vibration (rocking)

The lateral components of the forces acting on the crosshead induce lateral rocking, depending on the number of cylinders and firing order. 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'; refer to Fig. 5.2.1.

The '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} .

The '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.

The table in section 5.1 gives the values of resulting lateral guide forces and moments of the relevant orders.

The amplitudes of the vibrations transmitted to the hull depend on the design of the 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.

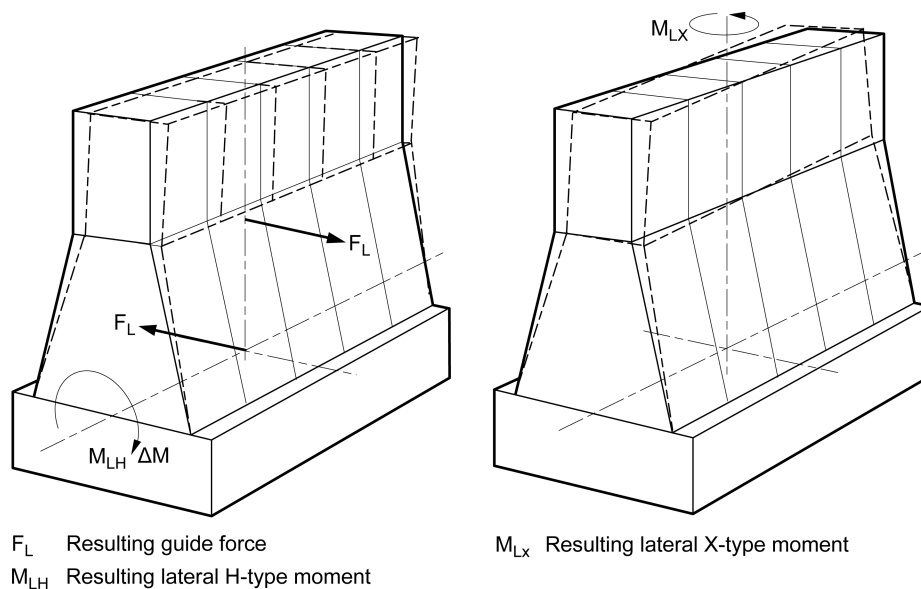


Fig 5.2.1 Lateral vibration

5.3 Reduction of lateral vibration

5.3.1 Engine stays

Fitting of lateral stays between the upper platform level and the hull reduces transmitted vibration and lateral rocking. Two different arrangements can be considered: single-acting hydraulic type or double-acting type.

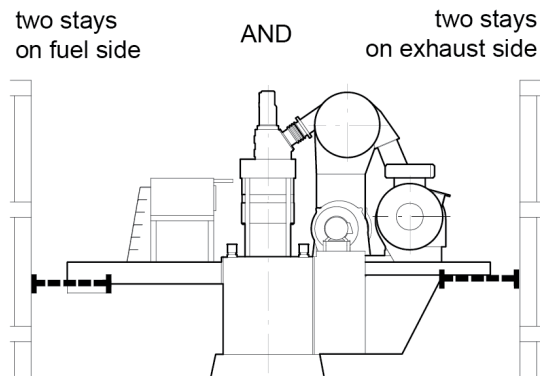


Fig 5.3.1.1 General arrangement of lateral stays (single-acting hydraulic type)

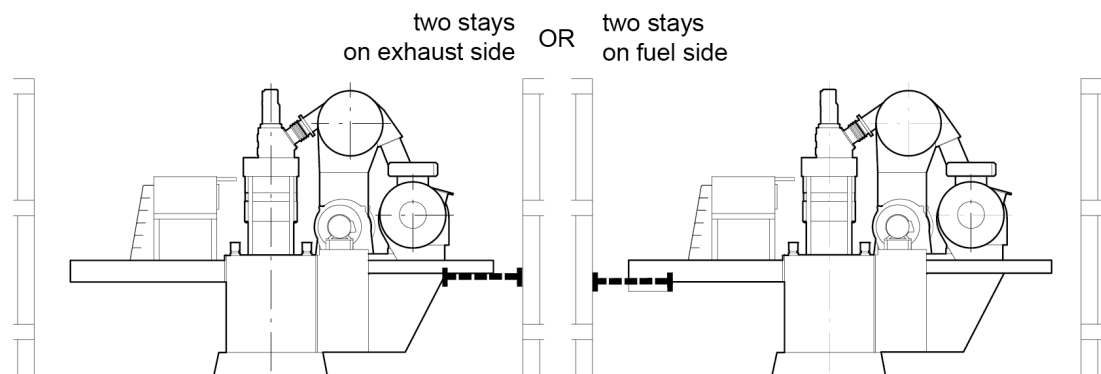


Fig 5.3.1.2 General arrangement of lateral stays (double-acting hydraulic type)

5.3.2 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 can be compensated at a time, and in case of an 'X-type' vibration mode, two compensators, one fitted at each end of the engine top, are necessary.

5.4 Longitudinal engine vibration (pitching)

Longitudinal engine vibration of this engine type is insignificant since the number of cylinders is 6 to 12. Countermeasures are not needed.

5.5 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 causes additional dynamic stresses in the shafting.

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

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, e.g. optimizing 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.

The amplitudes and frequencies of torsional vibration must be calculated at 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 at an early design stage (see section 5.10).

5.5.1 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. Where low energy torsional vibrations have to be reduced, a viscous damper can be installed; refer to Fig. 5.5.1.1. 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.

For high energy vibrations, e.g. for higher additional torque levels that can occur with 5- and 6-cylinder engines, a spring damper with its higher damping effect may have to be considered; refer to Fig. 5.5.1.2.

This 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 150 kW energy (depends on number of cylinders).

The oil flow to the damper should be approx. 45 m³/h, but an accurate value will be given after the results of the torsional vibration calculation are known.

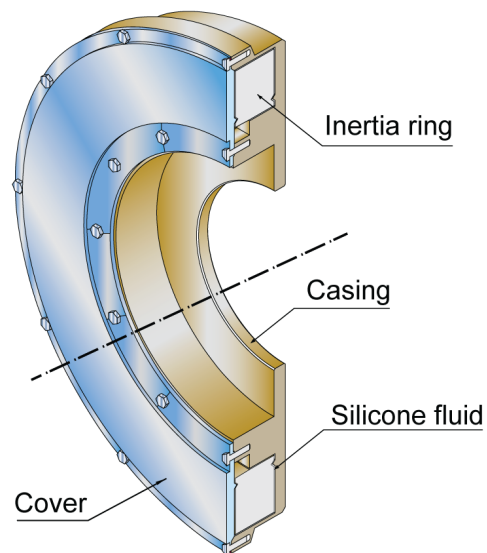


Fig 5.5.1.1 Torsional vibration damper (viscous type)

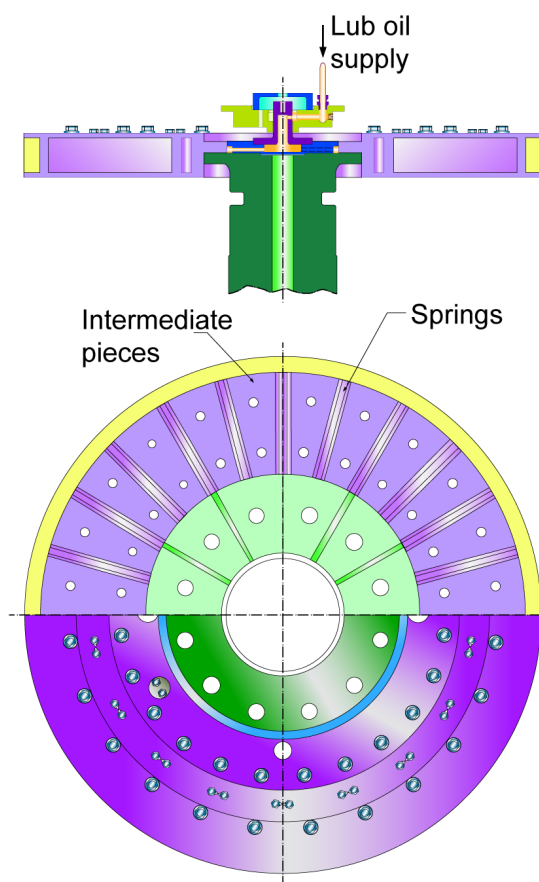


Fig 5.5.1.2 Torsional vibration damper (spring type)

5.6 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 5.5. The system, made up of masses and elasticities, will feature several resonant frequencies. These will result in axial vibration causing excessive stresses in the crankshaft, if no countermeasures are taken. Strong axial vibration of the shafting can also lead to excessive axial (or longitudinal) vibration of the engine, particularly at its upper part.

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 to carry out axial vibration calculations at the same time as the torsional vibration calculation. To consider the coupling effect of the torsional vibrations on the axial vibrations, it is necessary to use a suitable coupled axial vibration calculation method.

5.6.1 Reduction of axial vibration

To limit the influence of axial excitations and reduce the level of vibration, the standard WinGD X92 engine is equipped with an integrated axial vibration damper mounted at the free end of the crankshaft.

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 preset 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.

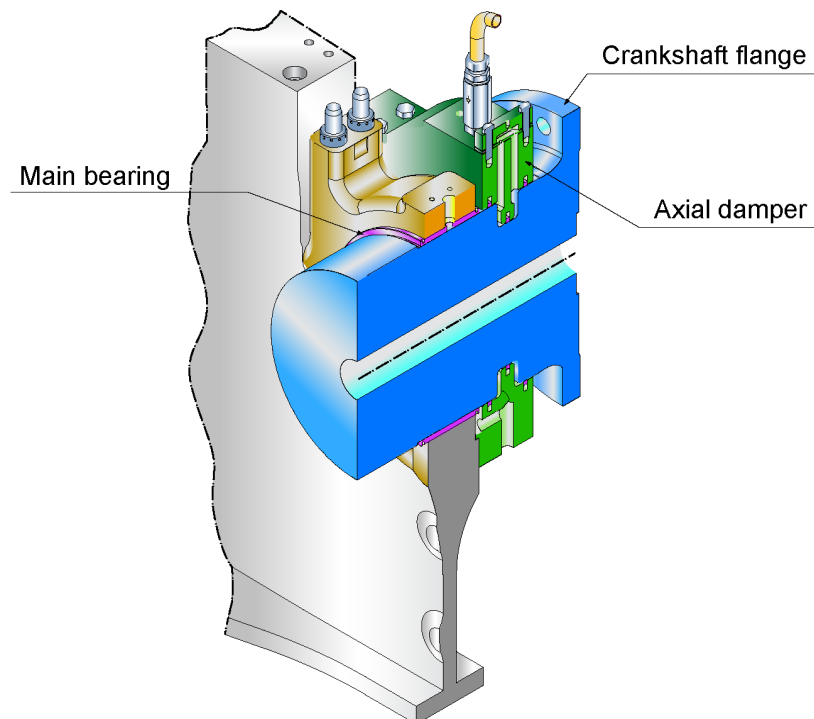


Fig 5.6.1.1 Example of an axial vibration damper

5.7 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 fitting mass moment compensators, lateral stays, torsional vibration damper and axial vibration damper. Avoiding disturbing hull vibration requires a close cooperation between the propeller manufacturer, naval architect, shipyard and engine builder. To enable Winterthur Gas & Diesel Ltd. to provide the most accurate information and advice on protecting the installation and vessel from the effects of plant vibration, complete the order forms as given in section 5.10 and send it to the address given.

5.8 Summary of countermeasures for dynamic effects

The following tables indicate where special attention is to be given to dynamic effects and the countermeasures required to reduce them. Where installations incorporate PTO arrangements, further investigation is required, and Winterthur Gas & Diesel Ltd. should be contacted.

Table 5.8.1 Countermeasures

No. cyl.	Countermeasures for external mass moments	Countermeasures for lateral and longitudinal vibrations	
	2 nd order compensator	Lateral stays	Longitudinal stays
6	Balancing countermeasure is likely to be needed *1)	B *2) / A *3)	C
7	Balancing countermeasure is not relevant	B *2) / A *3)	C
8	Balancing countermeasure is not relevant	A	C
9	Balancing countermeasure is not relevant	A	C
10	Balancing countermeasure is not relevant	A	C
11	Balancing countermeasure is not relevant	A	C
12	Balancing countermeasure is not relevant	A	C

A: The countermeasure indicated is needed.
 B: The countermeasure indicated may be needed and provision for the corresponding countermeasure is recommended.
 C: The countermeasure indicated is not needed.
 *1) No engine fitted 2nd order balancer available. If reduction of M2v needed, an external 2nd order compensator has to be applied.
 *2) 'B' for $n_{cmcr} \leq 76$ rpm.
 *3) 'A' for $n_{cmcr} > 76$ rpm.

5.8.1 Torsional and axial vibrations

Table 5.8.1.1 Countermeasures for torsional and axial vibration

No. cyl.	Torsional vibrations	Axial vibrations
6 to 12	Detailed calculations have to be carried out for every installation, countermeasures to be selected accordingly (shaft diameters, critical or barred speed range, flywheel, tuning wheel, TV damper).	An integrated axial vibration damper is fitted as standard to reduce the axial vibration in the crankshaft. However, the effect of the coupled axial vibration on the propulsion shafting components should be checked by calculation.

5.9 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).

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. This program allows to check, for example, if an elastic coupling will be overloaded during engine start, or to optimize 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.

Winterthur Gas & Diesel Ltd. 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 given in the next section 5.10. We will provide an answer as soon as possible.

5.10 Order forms for vibration calculations and simulation

For system dynamics and vibration analysis the following forms are available on the Licensee Portal. They can be filled in and submitted directly to Winterthur Gas & Diesel Ltd.

Marine installation:	Torsional Vibration Calculation	PDF available on request
Testbed installation:	Torsional Vibration Calculation	
Marine installation:	Coupled Axial Vibration Calculation	
	Whirling/Bending Vibration Calculation	

If you have no access to the Licensee Portal, order these forms from Winterthur Gas & Diesel Ltd. Send a PDF or fax a copy of the completed relevant forms to the following address:

Winterthur Gas & Diesel Ltd.
 Dept. 21347 Engine and System Dynamics
 Schützenstrasse 1-3
 PO Box 414
 CH-8401 Winterthur
 E-mail: dynamics.ch@wingd.com
 Fax: +41-52-262 07 25

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6. Auxiliary Power Generation

This chapter covers a number of auxiliary power arrangements for consideration. However, if your requirements are not fulfilled, contact our representative or consult Winterthur Gas & Diesel Ltd. directly. Our aim is to provide flexibility in power management, reduce overall fuel consumption and maintain uni-fuel operation.

The sea load demand for refrigeration compressors, engine and deck ancillaries, machinery space auxiliaries and hotel load can be met by using a main-engine-driven generator, a steam turbine driven generator utilising waste heat from the engine exhaust gas, or simply by applying auxiliary generator sets. The waste heat option is a practical proposition for high-powered engines employed on long voyages.

The electrical power required when loading and discharging cannot be met with a main-engine driven generator or with the waste heat recovery system, and for vessels employed on comparatively short voyages the waste heat system is not viable. Standby diesel generator sets (GenSets) burning heavy fuel oil or marine diesel oil, available for use in port, when manoeuvring or at anchor, provide the required flexibility when the main-engine power cannot be used.

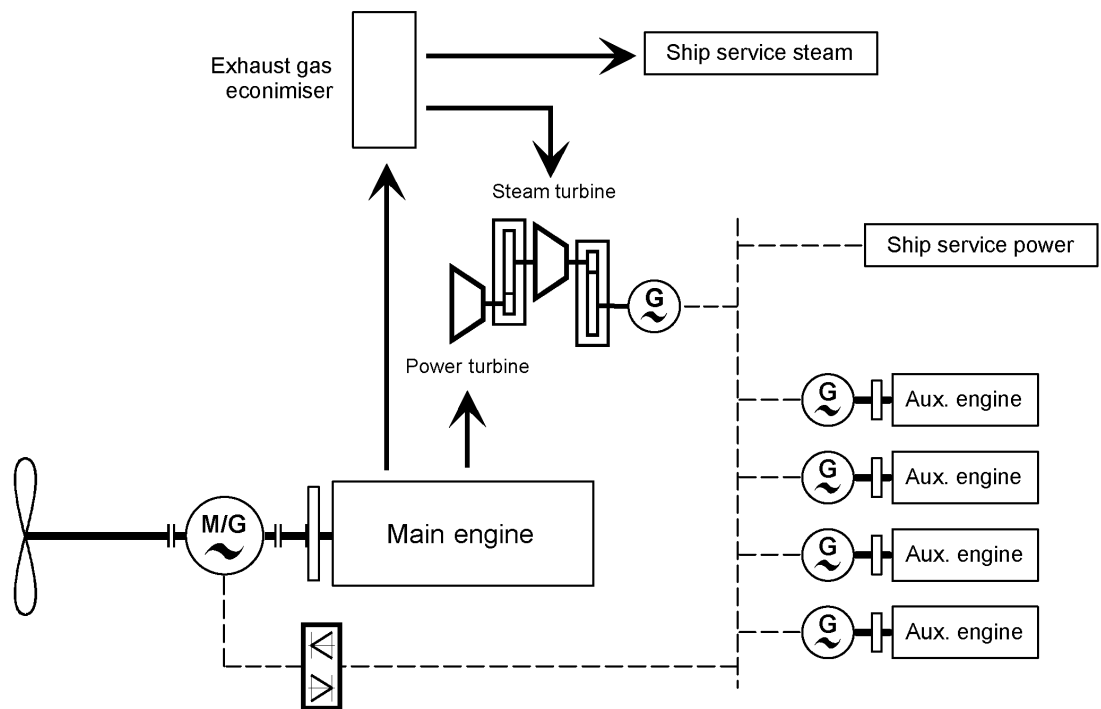


Fig 6.1 Heat recovery, typical system layout

Although initial installation costs for a heat recovery plant are relatively high, these are recovered by fuel savings if maximum use is made of the steam output, i.e. electrical power and domestics, space heating, heating of tank, fuel and water.

6.1 Waste heat recovery (WHR)

About 50 % of the fuel input energy is not used productively by diesel engines. Wasted engine heat can be recovered with the Rankine cycle (thermodynamic cycle converting heat into work). Heat is supplied to a closed loop with water as a working fluid.

A new concept for high-efficiency waste heat recovery combined with Wärtsilä RT-flex common-rail low-speed engines allows up to 12 % of the main engine shaft power to be recovered as electrical power for use as additional ship propulsion power and for shipboard services.

6.1.1 How to recover waste energy

Heat is extracted from exhaust gas, scavenge air and jacket cooling water and converted into electric power. This happens by means of a power turbine or a steam turbine or a combination of both and operating a generator.

- Power turbine

Part of the exhaust gas delivered by the engine bypasses the turbocharger through the power turbine. Even with reduced exhaust gas flow, today's modern high-efficiency turbochargers are able to supply sufficient scavenge air to the engine as they have a surplus of efficiency within the upper load range.

- Steam turbine

The steam turbine can be in multi-stage, single or dual pressure execution. Various alternatives regarding steam production and operating concept may be involved in the WHR plant (see section 6.1.5).

The engine's exhaust gas energy generates steam for the turbine in the economiser by evaporating feed water. In a single pressure steam system the exhaust gas economiser consists of an evaporator and a superheater with a drum. In dual pressure (low / high) steam systems the economiser may contain various combinations of evaporators, superheaters and LP + HP drums.

To increase the efficiency, the feed water is preheated by cylinder jacket water energy and scavenge air energy.

Special engine tuning in combination with direct ambient scavenge air suction enables an elevated exhaust gas temperature to achieve increased efficiency in heat recovery.

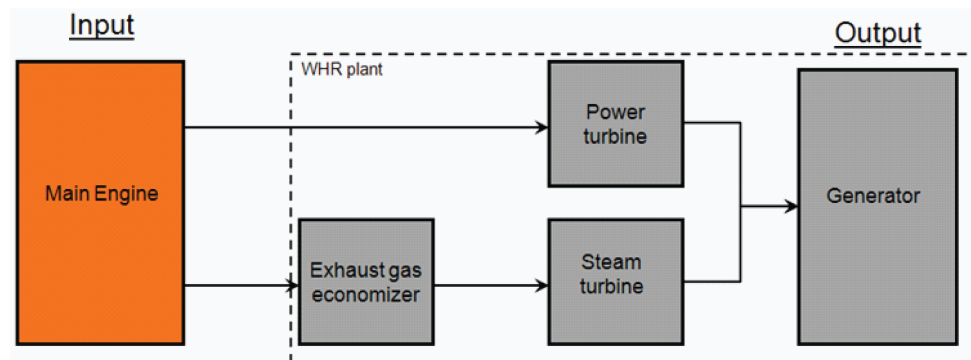


Fig 6.1.1.1 WHR system

6.1.2 Benefits of waste heat recovery:

- The operator benefits from lower annual fuel costs.
- The operator contributes to reducing emissions, such as CO₂, NO_x and SO_x.
- The operator profits by increased competitiveness in the freight market.
- With a WHR system the vessel's EEDI can be reduced.

6.1.3 Installation aspects

Air suction

The engine can be tuned for engine room air suction or direct outside air suction.

With outside air suction, it is assumed that the outside air temperature does not exceed 35 °C. With that the turbochargers are tuned at 15 °C air temperature. At ISO conditions direct outside air suction increases the exhaust gas temperature by 15 °C and reduces the exhaust gas flow by 0.2 kg/kWh. The turbochargers have a working range of +/- 20 °C from the reference point.

With outside air suction the BSFC increases by 1 g/kWh .

The engine needs to be equipped with an air waste gate to avoid excessive cylinder pressure at very low outside air temperature.

Thermal loading of the engine does not exceed the limit defined for a conventional engine.

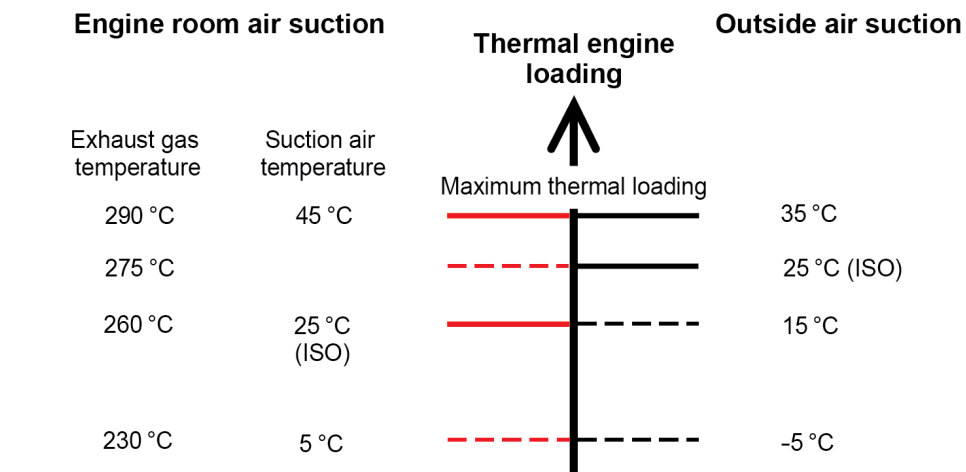
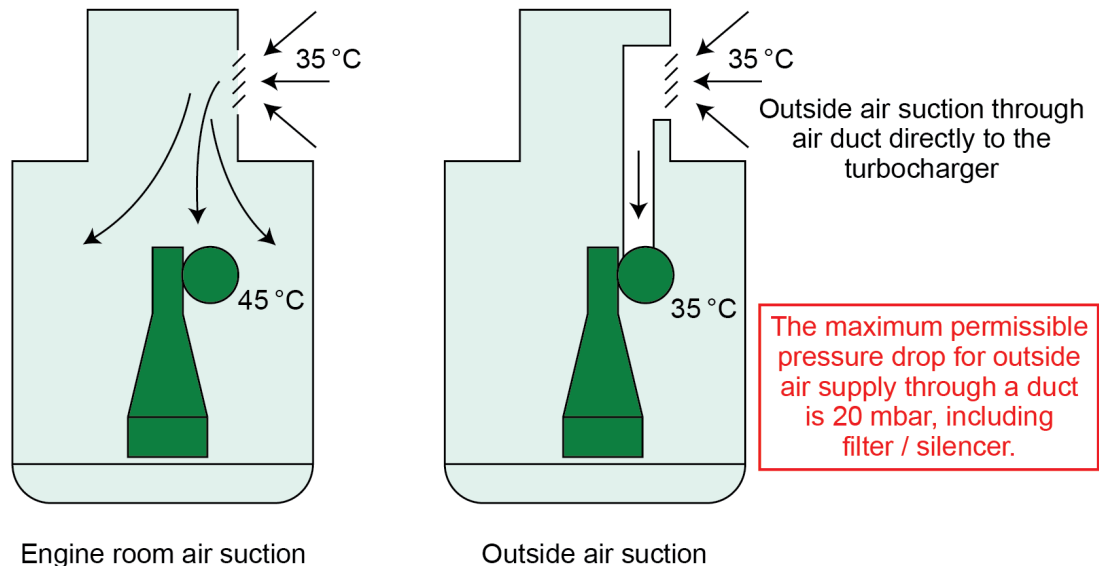


Fig 6.1.3.1 Tuning for air suction

**Fig 6.1.3.2 Air suction**

The standard filter / silencer on the turbocharger is used. A separate casing is built around the filter / silencer to guide the outside air to the turbocharger. Therefore no additional filter or silencer is required in the air duct.

Waste gate

Many of the WHR projects use outside air suction. In case of outside air temperatures lower than minus 5 °C, the engine has to be protected from overpressure in the scavenge air receiver causing excessive cylinder pressure which in turn results in turbocharger overspeeding caused by the high specific density of the cold air. This will be achieved by applying a waste gate (blow-off valve) for either scavenge air or exhaust gas.

Exhaust gas bypass

An exhaust gas bypass works in such a way that the gas flow is used either to increase the exhaust gas temperature at the boiler inlet or to drive a power turbine. The exhaust gas temperature after power turbine is the same as after turbocharger. The working range of the WHR system is above approximately 55 % engine load with the exhaust gas bypass valve open. The exhaust gas bypass valve must be closed at around 50 % engine load and lower.

The power turbine output depends on the exhaust gas bypass rate. The exhaust gas bypass rate depends on the engine type.

Feed water heating

In a first stage the feed water is heated to about 80 °C with jacket cooling water. In a second stage, the feed water is further heated up with scavenge air. Each scavenge air cooler module is equipped with a feed water heating section, which is not needed for scavenge air cooling. Cooling of the scavenge air is ensured even if the feed water heating section runs dry.

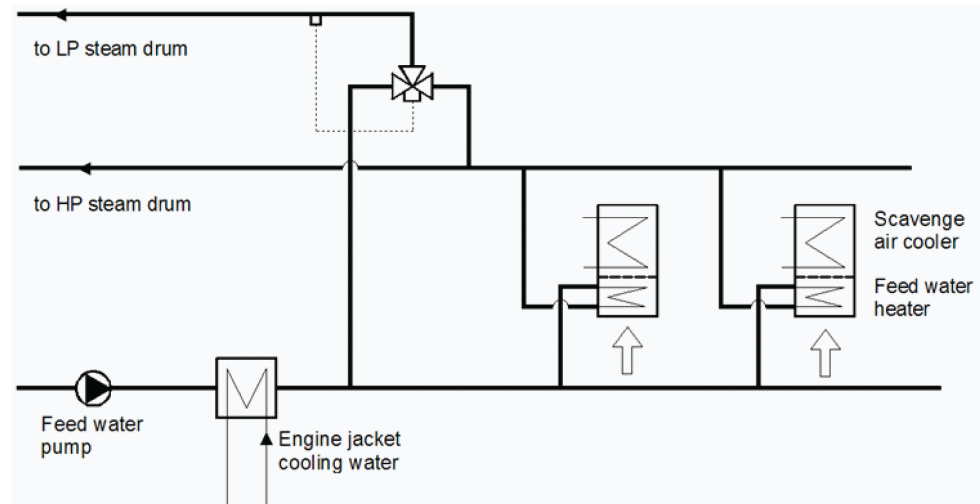


Fig 6.1.3.3 Feed water heating

The feed water temperature at the heater outlet varies with the engine load. Since the feed water temperature after heater may be higher than the water temperature in the low-pressure steam drum, the temperature of the water being fed to the low-pressure steam drum must be controlled to avoid steam flashing.

The amount of feed water is equivalent to the total amount of steam generated in the exhaust gas economiser.

Power take-in (PTI) and power take-off (PTO)

For the additional benefit of operational flexibility of the ship, two different systems can be combined with the WHR system: the shaft motor system (PTI) or the shaft generator system (PTO).

In the PTI mode, surplus electric power generated by the turbocharger is utilised. In the PTO mode, the main engine balances the shortage of electric power (at low engine load). With that the gensets are not in operation with the ship at sea.

Engine tuning

The following tuning options are available for engines with Flex system:

- Standard Tuning
- Delta Tuning
- Low-Load Tuning

The exhaust gas temperature and flow values for the different tuning options are illustrated by the following graphs:

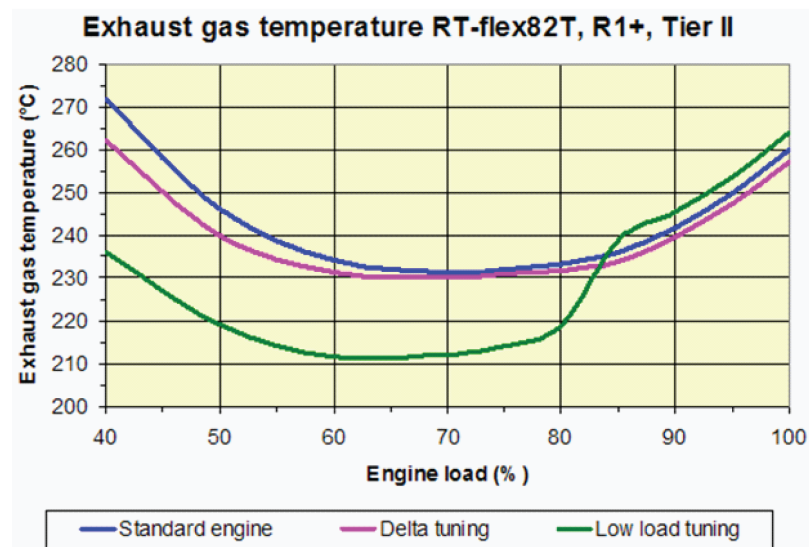


Fig 6.1.3.4 Exhaust gas temperature

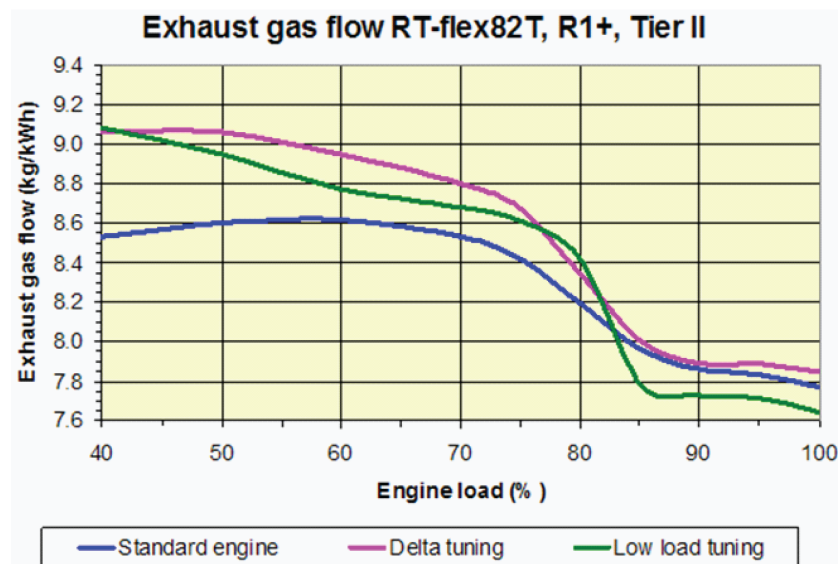


Fig 6.1.3.5 Exhaust gas flow

As the exhaust gas temperature is low at less than 85 % load, it is recommended that Low-Load Tuning is not used in combination with WHR. Usually the Delta Tuning option is selected for WHR applications.

6.1.4 Heat recovery concepts

Heat recovery with generator only

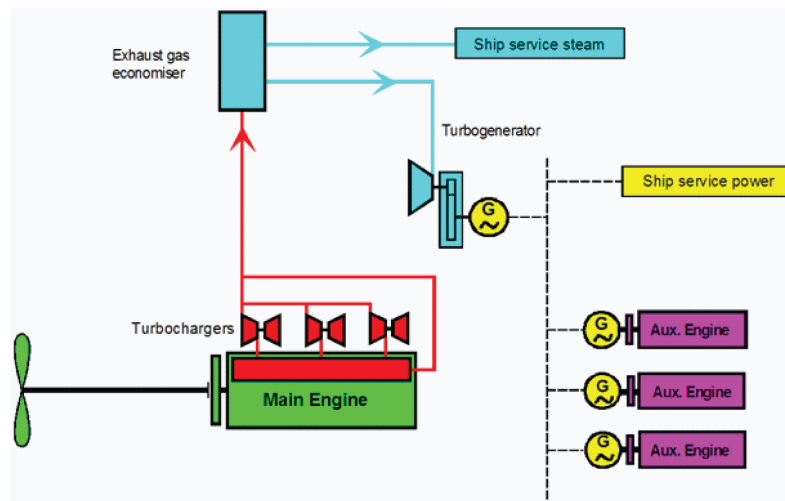


Fig 6.1.4.1 Heat recovery with generator only

Heat recovery with power and steam turbines

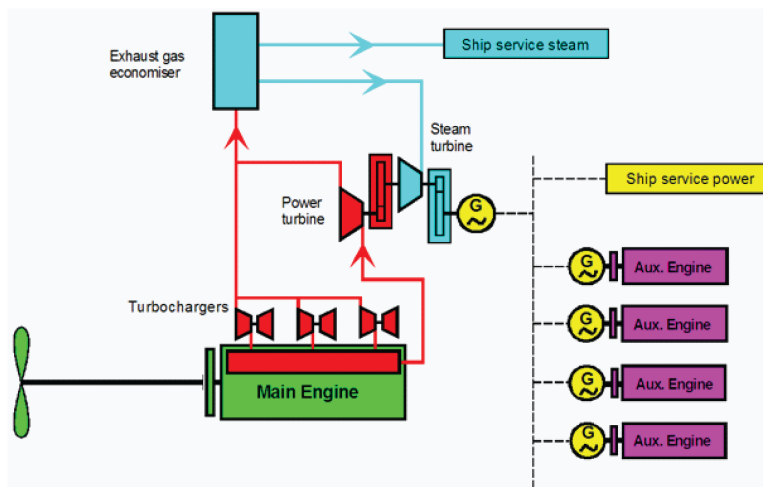


Fig 6.1.4.2 Heat recovery with power and steam turbines

There are two alternative operating modes for this concept:

a High electric power mode

Both the steam turbine and the power turbine generate electric power for ship service. The generator output is controlled by the steam flow to the steam turbine.

b Low electric power mode

The power turbine is disconnected. Electric power is generated by the steam turbine only. The generator output is controlled by the steam flow to the steam turbine.

Heat recovery with power & steam turbines and PTI / PTO

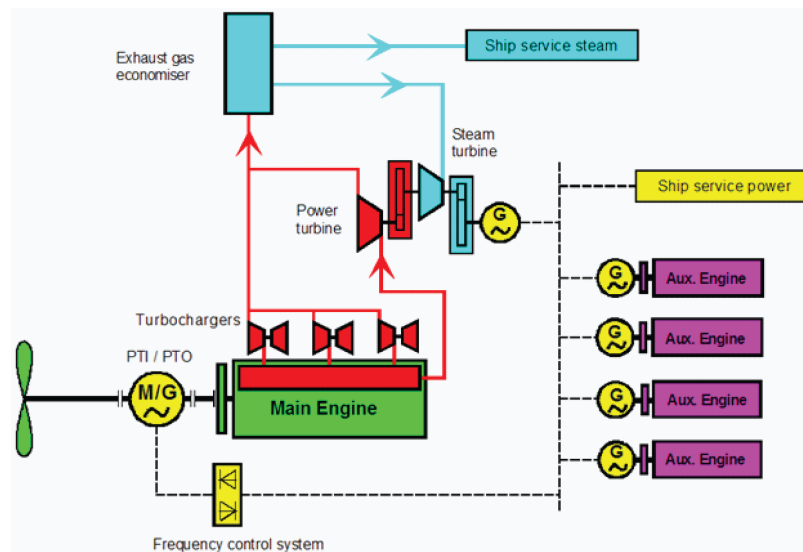


Fig 6.1.4.3 Heat recovery with power & steam turbines and PTI / PTO

Alternative operating modes:

Motor mode	The heat recovery system generates more electrical power than needed for shipboard service. The surplus electric power is fed to a shaft motor to support propulsion.
Alternator mode	The heat recovery system generates less power than needed for shipboard service. The shortage of electric power is compensated by using the PTO solution.
Booster mode	More propulsion power is needed than is available from the main engine. The PTI system acts as a motor with the electric power being generated by the heat recovery system. The electricity for shipboard service is supplied by auxiliary engines.
Emergency propulsion mode (optional)	The main engine is disconnected from the propeller shaft. The ship is propelled by the PTI system with power supplied by auxiliary diesel engines.

Heat recovery with power turbine only

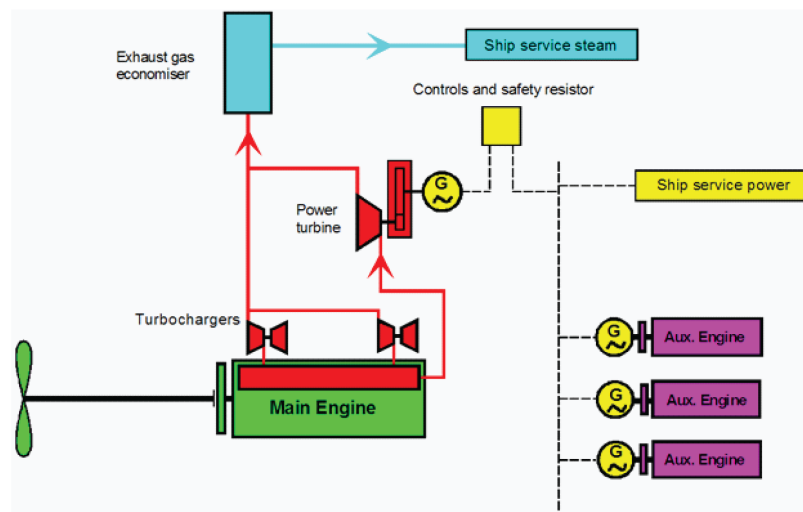


Fig 6.1.4.4 Heat recovery with power turbine only

6.1.5 WHR steam systems

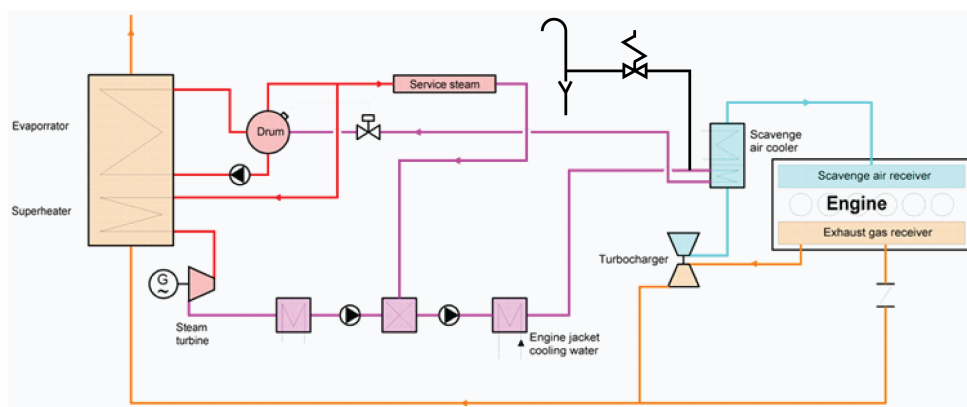


Fig 6.1.5.1 Single pressure steam system with evaporator and superheater

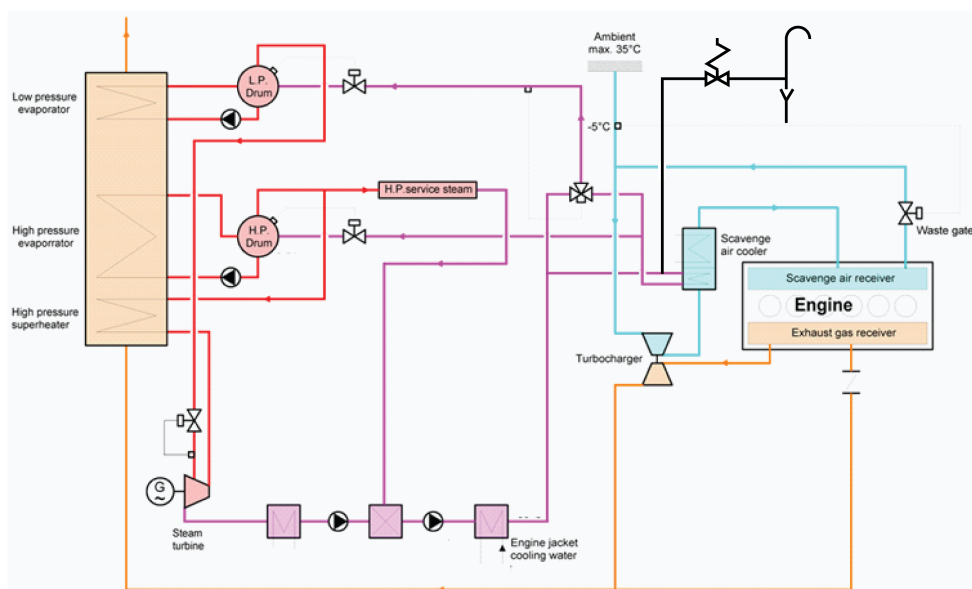


Fig 6.1.5.2 Dual pressure steam system with HP superheater

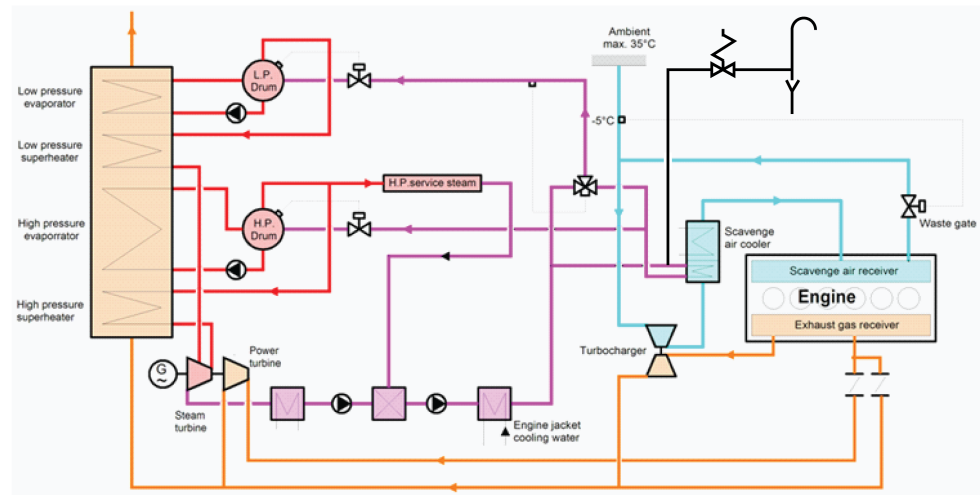


Fig 6.1.5.3 Dual pressure steam system with HP and LP superheater

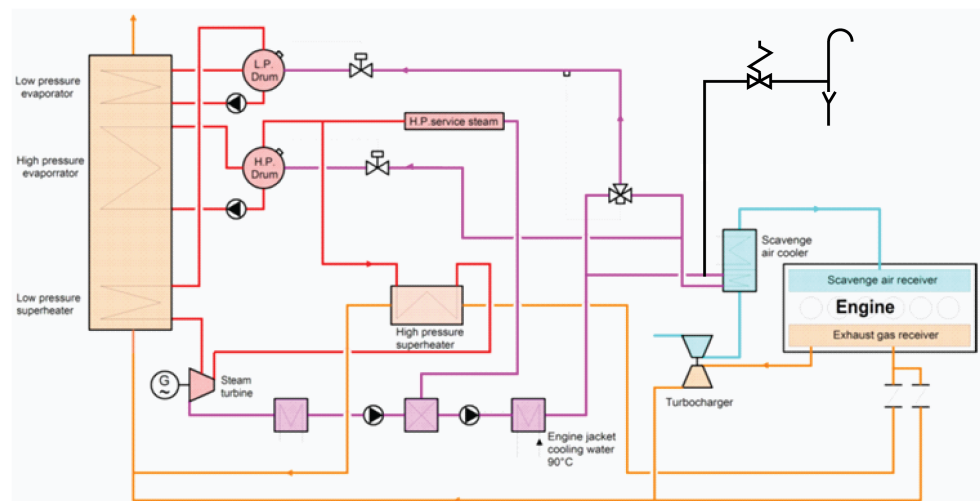


Fig 6.1.5.4 Dual pressure steam system with LP superheater and separate HP superheater

NOTE



All given alternatives are subject to a detailed project-specific study and definition. Please consult Winterthur Gas & Diesel Ltd. via its licensee.

6.2 Power take-off, take-in, and take-home applications

Winterthur Gas & Diesel Ltd. proposes various power take-off (PTO) and power take-in (PTI) options improving 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.

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

NOTE

All given alternatives are subject to a detailed project-specific study and definition. Please consult Winterthur Gas & Diesel Ltd. via its licensee.

6.2.1 Requirements

At first an engine selection has to be made, the shaft power and shaft speed have to be defined and the demand for electric power as a function of time has to be estimated. Then one evaluates which of the PTO / PTI systems is the most economical for producing the required electrical power. Furthermore one needs to select the suitable components enabling the vessel to consume electricity under foreseen operating conditions.

The type of PTO / PTI system has an influence on the execution of the main engine. Therefore you may change from one system type to the other in the project stage, but not after having ordered the engine.

6.2.2 Options

The different options are illustrated in the following. In any case their application for the selected engine type needs to be checked via the licensee with Winterthur Gas & Diesel Ltd. Project dependent options can also be considered.

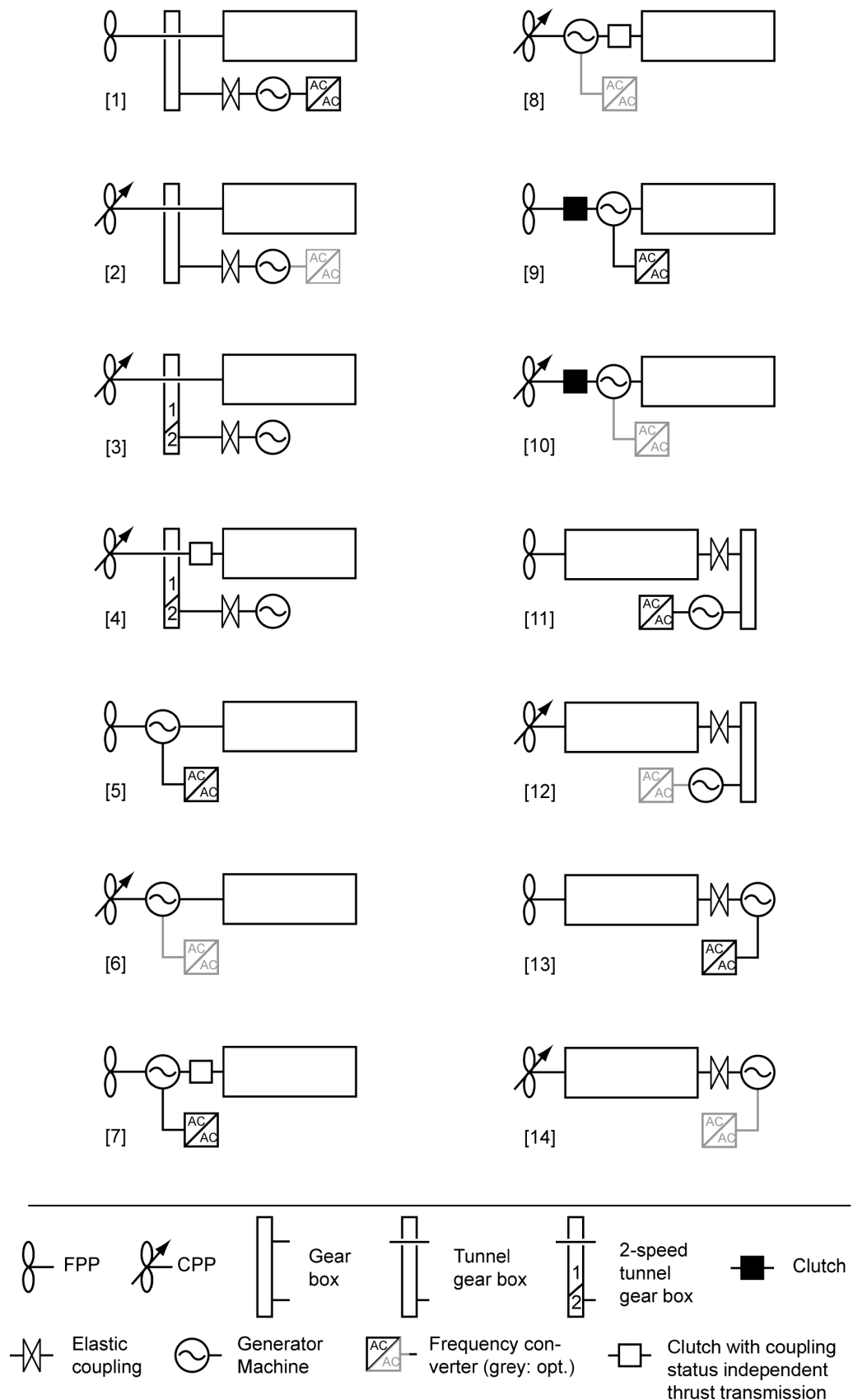


Fig 6.2.2.1 Options for PTI, PTO, PTH

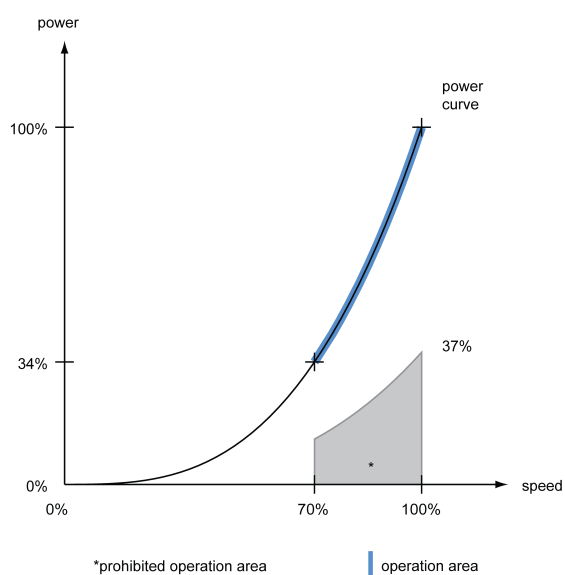
In the following table the options which can be applied to WinGD X92 are marked with 'X'.

Options for WinGD X92													
[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]
X				X		X				X		X	

6.2.3 Service conditions

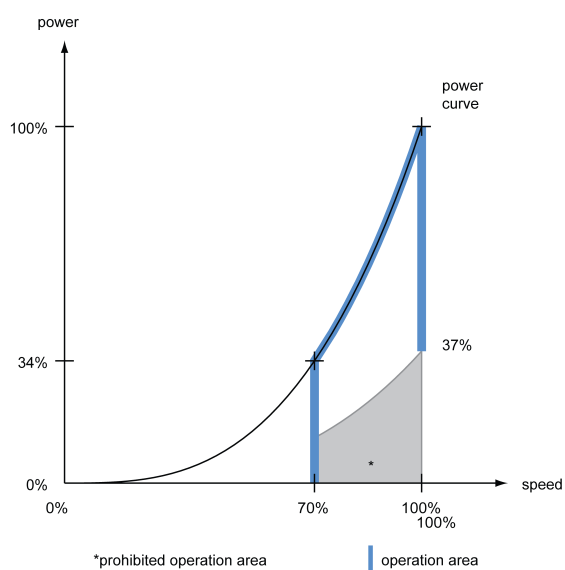
The service condition depends on the selected option. Depending on the engine type there are one or several cases, which are illustrated below.

The blue line in the illustration represents how the engine generator unit can be run.



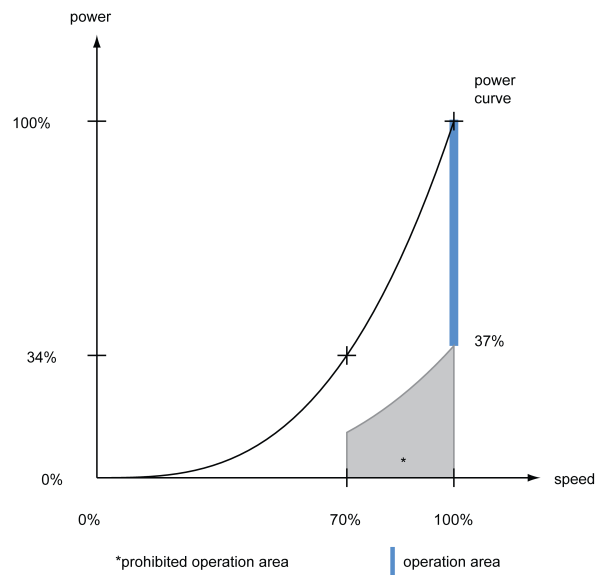
Valid for FPP with a mandatory frequency converter with option 1, 5, 7, 9, 11, and 13

Fig 6.2.2 FPP with a mandatory frequency converter



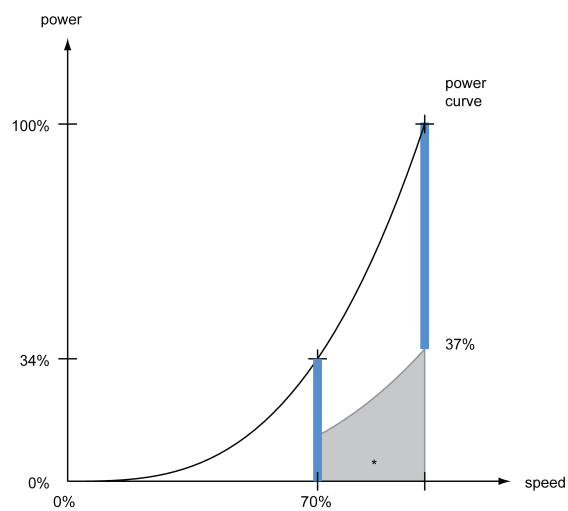
Valid for CPP in combination with an optional frequency converter with option 2, 6, 8, 10, 12, and 14

Fig 6.2.3 CPP in combination with an optional frequency converter



Valid for CPP applications with option 2, 6, 8, 10, 12, and 14

Fig 6.2.4 CPP



Valid for CPP in combination with a two-speed tunnel gear with option 3 and 4

Fig 6.2.5 CPP with a two-speed tunnel gear

6.2.4 PTO / PTI application constraints

In any case the feasibility of project-specific PTO / PTI needs to be studied. An overview about impacts is given in the following table.

Table 6.2.4.1 Features

Features	Options													
	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]
Power take-off (PTO)	X				X		X				X		X	
Power take-in (PTI)	X				X						X		X	
Power take-home (PTH)														
Primary generator														
Power take-off / power take-in / power take-home / primary generator 'X' means that this option is offered for the described feature. In addition to the PTO / PTI solutions there is the PTH option, enabling to run the propeller by electric power, and the use of the engine plus generator as primary generator. The latter enables the vessel to generate electric power by the main engine without running the propeller.														

Table 6.2.4.2 Engineering

Engineering	Options													
	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]
Extended TVC	X				X		X				X		X	
Misfiring detection	X				O		X				X		X	
Impact on engine control system	(X)				(X)		X				(X)		(X)	
Additional alignment effort	(X)				X		X				X		X	
Bending moment study	X				X		X				X		X	
Bearing load due to ext. load	X				X	X	X	X	X	X	X	X	X	X
Dynamic condition due to ext. load	X				X		X				X		X	
Influence of the options on engineering														
X = the option has an influence (X) = the option has limited influence O = the option has no influence An empty field indicates that the option is not applicable for this engine type. - Extended TVC (torsional vibration calculation) The concept adds components to the propulsion line which need 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 lower than/equal to 2 Hz, a misfiring detection is needed to protect the elastic coupling. - Impact on engine control system The position of the clutch with coupling status independent thrust transmission has to be detected for e.g. using another parameter set in the speed control. The PTO / PTI / PTH application has to be analysed via the licensee with concerned (propulsion) control system suppliers and Winterthur Gas & Diesel Ltd. for the engine control system. - Additional alignment effort As long as a tunnel gear is applied in the shaft line, the additional alignment effort is moderate (indicated by (X)). However, as soon as additional elements like a clutch with coupling status independent thrust transmission are added, the alignment effort increases (indicated by X). One needs to ensure that the bearings inside the engine as well as those in the shaft line are properly selected and adjusted. - Bending moment study Additional elements like a clutch with coupling status independent thrust transmission or the generator / motor itself bring additional bending moments to the shaft line. As a countermeasure additional and suitable bearing positions need to be defined. This is also valid in case the generator / motor is applied at the engine's free end. - Bearing load due to external load The additional elements are not only increasing the bending moment but also the related bearing load. Therefore they need 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.														

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7. Ancillary Systems

Sizing engine ancillary systems, i.e. freshwater cooling, lubricating oil, fuel oil, etc., depends on the contract maximum engine power. If the expected system design is out of the scope of this manual, contact our representative or Winterthur Gas & Diesel Ltd. directly.

The *GTD* enable all engine and system data at any Rx rating within the engine rating field to be obtained. However, for convenience or final confirmation when optimizing the plant, Winterthur Gas & Diesel Ltd. provide a computerized calculation service.

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. **All pipework systems are to be flushed and proved clean before commissioning.**

The data given in section 7.1 are applicable to the **nominal maximum continuous rating R1** of the 6 to 12-cylinder engines and suitable for estimating the size of ancillary equipment.

These data refer to engines with the following conditions/features:

- At design (tropical) conditions
- Delta Tuning
- Central freshwater cooling system with single-stage scavenge air cooler (SAC) and separate HT circuit
- ABB A100-L / A200-L turbochargers
- Turbochargers lubricated from the engine's lubricating system

Furthermore the following data are obtainable from the *GTD* or on request from Winterthur Gas & Diesel Ltd.:

- Data for engines fitted with ABB A200-L and MHI MET turbochargers
- Standard Tuning
- Derating and part-load performance data
- Delta Bypass Tuning
- Data for Low-Load Tuning

7.1 Ancillary systems data at MCR

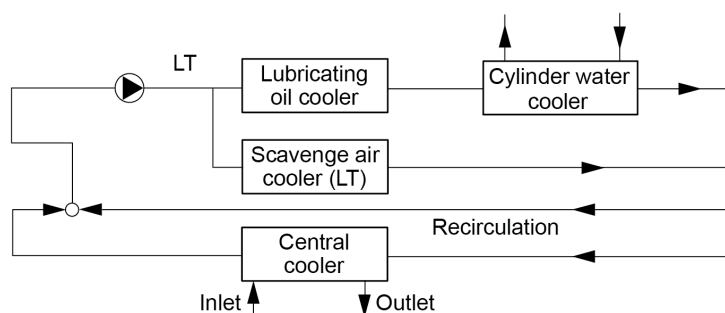


Fig 7.1.1 Central freshwater cooling system with separate HT circuit

Table 7.1.1 Cylinder cooling, Scavenge air cooler, Exhaust gas

Cyl	Cylinder cooling (HT)		Scavenge air cooler (LT)		Exhaust gas	
6	Heat dissipation [kW]	4645	Heat dissipation [kW]	19225	Steam Production Power [kW] *1)	9430
	FW flow [m ³ /h]	227	FW flow [m ³ /h]	501	Mass flow [t/h]	282
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 69.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	275		
7	Heat dissipation [kW]	5430	Heat dissipation [kW]	22422	Steam Production Power [kW] *1)	11002
	FW flow [m ³ /h]	266	FW flow [m ³ /h]	722	Mass flow [t/h]	328
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 63.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	320		
8	Heat dissipation [kW]	6215	Heat dissipation [kW]	25632	Steam Production Power [kW] *1)	12573
	FW flow [m ³ /h]	304	FW flow [m ³ /h]	621	Mass flow [t/h]	375
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 72.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	366		
9	Heat dissipation [kW]	7001	Heat dissipation [kW]	28834	Steam Production Power [kW] *1)	14145
	FW flow [m ³ /h]	343	FW flow [m ³ /h]	682	Mass flow [t/h]	422
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 72.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	412		
10	Heat dissipation [kW]	7787	Heat dissipation [kW]	32035	Steam Production Power [kW] *1)	15717
	FW flow [m ³ /h]	381	FW flow [m ³ /h]	992	Mass flow [t/h]	469
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 64.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	458		

Cyl	Cylinder cooling (HT)		Scavenge air cooler (LT)		Exhaust gas	
11	Heat dissipation [kW]	8575	Heat dissipation [kW]	35243	Steam Production Power [kW] *1)	17289
	FW flow [m ³ /h]	420	FW flow [m ³ /h]	902	Mass flow [t/h]	515
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 70.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	503		
12	Heat dissipation [kW]	9364	Heat dissipation [kW]	38448	Steam Production Power [kW] *1)	18860
	FW flow [m ³ /h]	458	FW flow [m ³ /h]	932	Mass flow [t/h]	562
	FW temp. engine in/out [°C]	75.0 / 90.0	FW temp. engine in/out [°C]	36.0 / 72.0	Temp. after turbine [°C]	289
			SAC mass flow [t/h]	549		
*1) Steam Production Power for a constant temperature after economizer of 180 °C. Tolerances: SPP ± 7 %, cylinder heat dissipation $+0/-15$ %.						

Table 7.1.2 Lubricating oil cooler, Central cooler

Cyl	Lubricating oil cooler		Central cooler	
6	Heat dissipation *2) [kW]	2937	Heat dissipation [kW]	26808
	Oil flow *2) [m ³ /h]	335	FW flow [m ³ /h]	757
	Oil temp. cooler in/out [°C]	63.0 / 45.0	FW temp. cooler in/out [°C]	67.0 / 36.0
	Water flow [m ³ /h]	255	SW flow [m ³ /h]	1289
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0
7	Heat dissipation *2) [kW]	3390	Heat dissipation [kW]	31243
	Oil flow *2) [m ³ /h]	382	FW flow [m ³ /h]	945
	Oil temp. cooler in/out [°C]	63.0 / 45.0	FW temp. cooler in/out [°C]	63.0 / 36.0
	Water flow [m ³ /h]	294	SW flow [m ³ /h]	1502
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0
8	Heat dissipation *2) [kW]	3895	Heat dissipation [kW]	35744
	Oil flow *2) [m ³ /h]	427	FW flow [m ³ /h]	962
	Oil temp. cooler in/out [°C]	64.0 / 45.0	FW temp. cooler in/out [°C]	68.0 / 36.0
	Water flow [m ³ /h]	338	SW flow [m ³ /h]	1719
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0
9	Heat dissipation *2) [kW]	4350	Heat dissipation [kW]	40188
	Oil flow *2) [m ³ /h]	477	FW flow [m ³ /h]	1139
	Oil temp. cooler in/out [°C]	64.0 / 45.0	FW temp. cooler in/out [°C]	69.0 / 36.0
	Water flow [m ³ /h]	377	SW flow [m ³ /h]	1932
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0

Cyl	Lubricating oil cooler		Central cooler	
10	Heat dissipation *2) [kW]	4875	Heat dissipation [kW]	44699
	Oil flow *2) [m ³ /h]	522	FW flow [m ³ /h]	1314
	Oil temp. cooler in/out [°C]	64.0 / 45.0	FW temp. cooler in/out [°C]	63.0 / 36.0
	Water flow [m ³ /h]	423	SW flow [m ³ /h]	2149
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0
11	Heat dissipation *2) [kW]	5408	Heat dissipation [kW]	49229
	Oil flow *2) [m ³ /h]	571	FW flow [m ³ /h]	1327
	Oil temp. cooler in/out [°C]	64.0 / 45.0	FW temp. cooler in/out [°C]	67.0 / 36.0
	Water flow [m ³ /h]	469	SW flow [m ³ /h]	2367
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0
12	Heat dissipation *2) [kW]	5863	Heat dissipation [kW]	53678
	Oil flow *2) [m ³ /h]	615	FW flow [m ³ /h]	1444
	Oil temp. cooler in/out [°C]	65.0 / 45.0	FW temp. cooler in/out [°C]	68.0 / 36.0
	Water flow [m ³ /h]	508	SW flow [m ³ /h]	2581
	Water temp. cooler in/out [°C]	36.0 / 46.0	SW temp. cooler in/out [°C]	32.0 / 50.0
*2) Excluding heat and oil flow for damper and PTO gear.				

Table 7.1.3 Pump capacities and delivery heads

Cyl	Lubricating oil *4)		HT circuit		LT circuit		Fuel oil booster		Fuel oil feed		Seawater	
	m³/h	bar *3)	m³/h	bar *3)	m³/h	bar *3)	m³/h	bar *3)	m³/h	bar *3)	m³/h	bar *3)
6	332	6.9	270	3.0	756	2.8	17.4	7.0	9.7	5.0	1289	2.0
7	376	6.9	316	3.0	1015	2.8	20.3	7.0	11.3	5.0	1502	2.0
8	424	6.9	362	3.0	959	2.8	23.2	7.0	12.9	5.0	1719	2.0
9	468	6.9	408	3.0	1059	2.8	26.1	7.0	14.5	5.0	1932	2.0
10	518	6.9	454	3.0	1415	2.8	29.0	7.0	16.1	5.0	2149	2.0
11	567	6.9	499	3.0	1371	2.8	31.9	7.0	17.7	5.0	2367	2.0
12	611	6.9	545	3.0	1440	2.8	34.8	7.0	19.4	5.0	2581	2.0

*3) Pressure difference across pump. Final delivery head must be according to the actual piping layout.

*4) Oil flow is specified for CMCR speed range C "without crosshead LO pump" and without flow for damper and PTO gear. Speed range A and B "with crosshead LO" require a slightly higher oil flow. See values in the GTD.

Table 7.1.4 Engine radiation

No. of cyl.	Engine radiation	
6	kW	322
7		361
8		400
9		438
10		474
11		509
12		544

Table 7.1.5 Starting air

Cyl	Bottles, pressure [bar]	Bottles, capacity each [m³]	Air compressors, capacity each [m³/h]
6	30	2 x 15.5	2 x 465
7		2 x 16	2 x 480
8		2 x 16.5	2 x 495
9		2 x 16.5	2 x 495
10		2 x 17	2 x 510
11		2 x 17	2 x 510
12		2 x 17.5	2 x 525

Values calculated for 12 starts and a refilling time of 1 hour, when Jrel = 2.0.

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8. Cooling Water System

The cooling system runs on the following standard layout:

Central freshwater cooling system with single-stage scavenge air cooler and **separate** HT circuit.

As freshwater is the standard cooling medium of the scavenge air cooler(s), this involves the use of a central freshwater cooling system.

The central freshwater cooling system comprises 'low-temperature' (LT) and 'high-temperature' (HT) circuits. Freshwater cooling systems reduce the amount of seawater pipework and its attendant problems and provide for improved cooling control. Optimizing central freshwater cooling results in lower overall running costs compared to the conventional seawater cooling system.

8.1 Central freshwater cooling system components

The high-temperature circuit may also be completely separated from the low-temperature circuit. In this case the high-temperature circuit has its own cooler with freshwater from the low-temperature circuit as cooling medium. The necessary data for this arrangement can be obtained from the *GTD* program.

8.1.1 Low-temperature circuit

Table 8.1.1.1 Main components

Main components	
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) to prevent passage of large particles and debris damaging the pumps and impairing heat transfer across the coolers.
Seawater pump	
Pump type	Centrifugal
Pump capacity	The given seawater flow capacity covers the need of the engine only and is to be within a tolerance of 0 to +10 %.
	Refer to tables in section 7.1 Ancillary systems data at MCR.
Delivery head	Is determined by the layout of the system.
Central cooler	
Cooler type	Plate or tubular
Cooling medium	Seawater
Cooled medium	Freshwater
Margin for fouling	10-15 % to be added.
Heat dissipation	Refer to tables in section 7.1 Ancillary systems data at MCR.
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 between 25 °C and 36 °C.
Freshwater pumps for LT circuit	
Pump type	Centrifugal
Pump capacity	The given capacity of freshwater flow covers the need of the engine only and is to be within a tolerance of 0 to +10 %.
	Refer to tables in section 7.1 Ancillary systems data at MCR.
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 coolers is within the range of the summarized data.

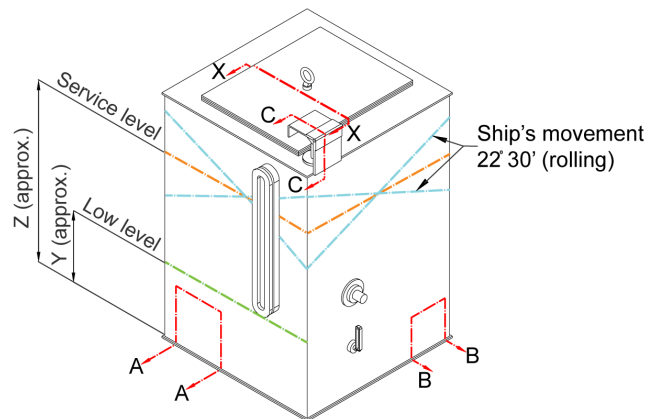
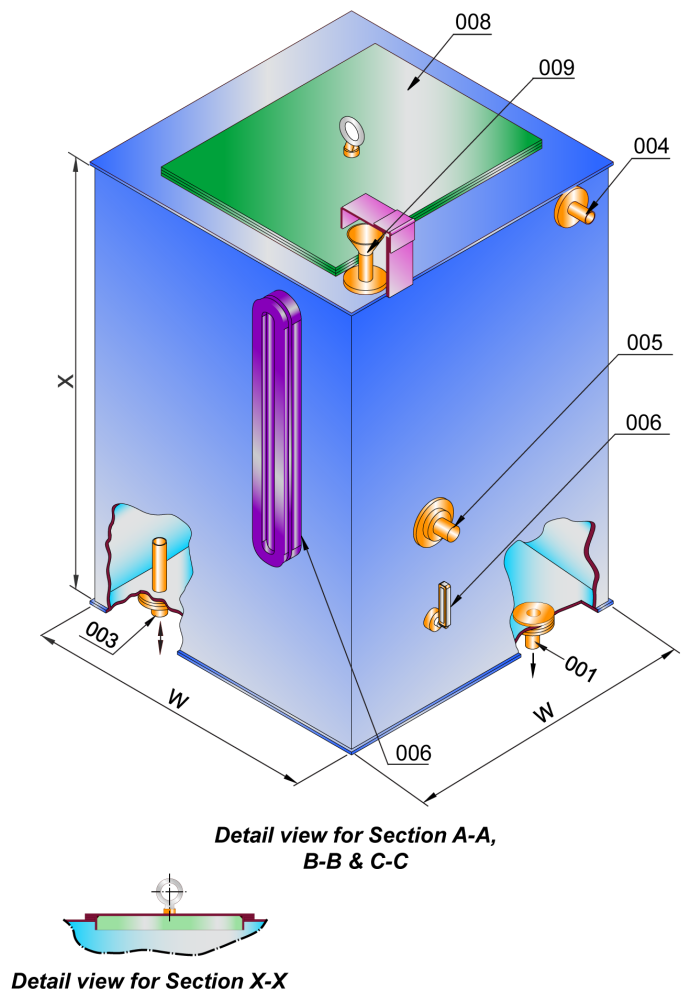


Fig 8.1.1.1 Central cooling water system, expansion tank (LT circuit)

001	Drain from LT circuit	006	Level indicator *1)
		007	Thermometer
003	Balance pipe from circuit	008	Inspection cover *2)
004	Overflow / air vent	009	Filling pipe / inlet chemical treatment *2)
005	Low-level alarm		

*1) Level indicator can be omitted if an alternative is fitted.

*2) Other designs (like hinged covers, etc.) are possible.

8.1.2 High-temperature circuit

Table 8.1.2.1 Main components

HT cooling water pump	
Pump type	Centrifugal, with a steep headcurve is to be given preference. 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 %.
Pump capacity	The flow capacity is to be within a tolerance of -10 to +20 %. <i>Refer to tables in section 7.1 Ancillary systems data at MCR.</i>
Delivery head	Determined by system layout
Working temperature	95 °C
Pump delivery head (p_p)	
The system pressure (p_{ei}) at the engine inlet will be: $p_{ei} = p_p + p_{st} - d_p - h/10.2$ [bar] where : p_p : pump delivery head [bar] p_{st}: static pressure at pump inlet [bar] d_p: 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: - Minimum = 4.5 bar - Maximum = 5.5 bar The delivery head has to be determined according to the total pressure losses (resistance) of the actual arrangement of the piping installation, as for a non-pressurized system.	
Buffer Unit	
The required static water pressure at the pump inlet is obtained by the use of a buffer unit in which an air cushion with constant and controlled pressure is maintained. The buffer unit acts as volume-compensating device, the water volume can expand or contract without altering the system pressure. In case the cooling water pumps should stop due to power failure, the cooling system remains pressurized thus avoiding vapour formation in the system. The initial filling of the buffer unit should be just above the low-level alarm, i.e. at approx. 30 % of its total capacity. The large air cushion, available in the buffer unit can so partly compensate the expansion or contraction of the water volume without bringing the control air pressure unit into operation. In this way, continuous operation of this unit is avoided. The final design must meet the requirements of the concerned classification societies. The working pressure indicated on this figure can vary, depending on the location of the main water pumps with regard to the engine cooling water inlet. The buffer unit contains the following main equipment:	

- Control air pressure unit (DN15, pressure range adjustable 3–5 bar), which reduces the supply air pressure (7 bar, filtered air) to the required static pressure and automatically maintains it constant.
- Solenoid valve, fitted upstream of the control air pressure unit, which interlocks the air inlet with the minimum water level in the buffer unit.
- Relief valve DN32, adjusted to approx. 5.5 bar.
- High- and low-level switch to control the supply pump. The low-level switch is set at approx. 35 % of the total content of the buffer unit. The difference between the high and the low level should correspond to a volume of approx. 150 l.
- Low- and high-level alarm. The low level is set at approx. 30 % of the total capacity of the buffer unit.

The compensation pipe should be led by the shortest route close to the pump suction. With the buffer unit close to the cooling water pumps, a module including all main components of the cooling system can easily be realized. For pipe sizes and tank capacities see drawings in 'List of Documents' at the end of the chapter.

Supply pump

A positive displacement supply pump with a capacity of 0.5 m³/h, 7 bar, is provided to replace the leakage water losses in the cylinder cooling water system. This pump is automatically controlled by the water level in the buffer unit as described under paragraph headed 'Buffer unit'. It would also be advisable to monitor the running period of the supply pump. A monitoring of the pump running period will warn when the running period exceeds a preset value, indicating unusual water losses in the system. Spare parts for the supply pump must be available according to classification societies' requirements.

Automatic temp. control valve

Electrically or electro-pneumatically actuated three-way type (butterfly valves are not adequate) having a linear characteristic	
Design pressure	10 bar
Test pressure	Refer to the specification laid down by the 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
Temp. sensor	According to the control valve manufacturer's specification, fitted in the engine outlet pipe

8.2 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 the different branches of the cylinder cooling water system is to be avoided by installing and setting throttling discs at the commissioning stage, but not by adjusting the valves.

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 the cylinder cooling water is to be examined carefully before being added. Uncontrolled increases or decreases in cylinder cooling water temperature may lead to thermal shock of the engine components and scuffing of the 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 at the 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.

8.2.1 Cooling water treatment

Correct treatment of the cooling freshwater is essential for safe engine operation. Only totally demineralized water or condensate must be used. 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 demineralized water.

Recommended parameters for raw water:

min. pH	6.5
max. dH	10°dH (corresponds to 180 mg/l CaCO ₃) *1)
max. chloride	80 mg/l
max. sulphates	150 mg/l

*1) In case of higher values the water is to be softened.

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

8.3 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. It is crucial at 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, the use of valves, e.g. butterfly valves at the freshwater generator inlet and in the bypass line which are linked and actuated with a large reduction ratio, will be of advantage. The following installations are given as examples and we recommend that the freshwater generator valves (7 and 8) be operated by progressive servomotors and a warning sign be displayed on the freshwater generator to remind engine room personnel of the possibility of thermal shocking if automatic start-up is overridden.

CAUTION



Avoid thermal shock to your main engine. The freshwater generator inlet and outlet valves to be opened and closed slowly and progressively.

The bypass with valve (8) 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 to interlock valves 7 and 8.

Figures 8.3.1.1 and 8.3.2.1 provide two systems designed to use up to 50 % of available heat (alternative 'A') and up to 85 % of available heat (alternative 'B').

8.3.1 Alternative 'A'

Freshwater generators, with an evaporator heat requirement not in excess of 50 % of the heat available to be dissipated from the cylinder cooling water at full load (CMCR) and only for use at engine loads above 50 %, can be connected in series as shown in figure *Freshwater generator installation, alternative 'A'*. The throttling disc (6) serves to correct the water flow rate if the pressure drop in the cooling circuit is less than that in the freshwater generator circuit. It is to be adjusted so that the cylinder cooling water pressure at the engine inlet is maintained within the pressure range of the summarized data in table 2.1 2.1 when the freshwater generator is started up and shut down.

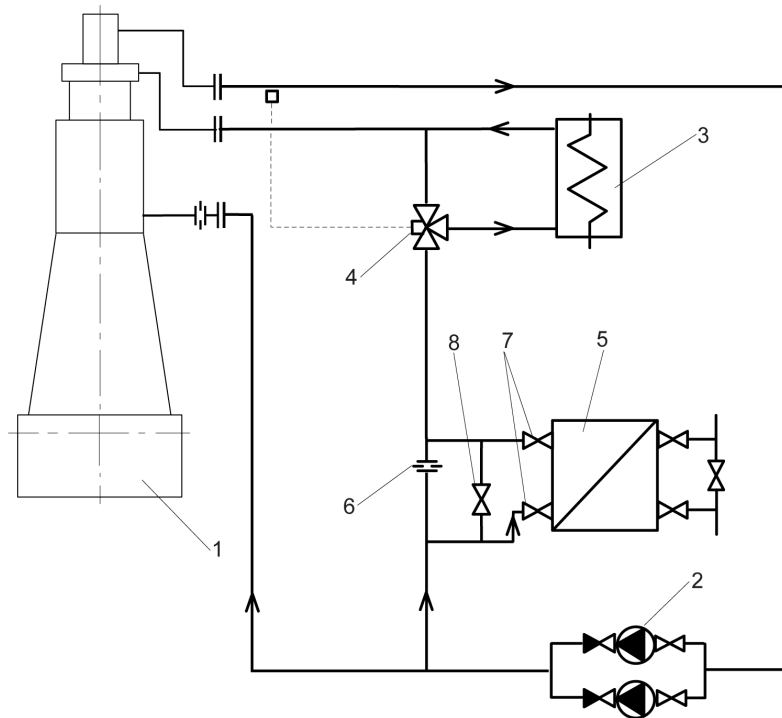


Fig 8.3.1.1 Freshwater generator installation, alternative 'A'

1	Main engine	5	Freshwater generator
2	Cylinder cooling water pump	6	Throttling disc
3	Cylinder cooling water cooler	7	Freshwater valves
4	Automatic temperature control valve	8	Freshwater generator bypass valve

Example:

12-cyl. engine - R1 specification with 77400 kW at 80 rpm

The available heat (from section 7.1 *Ancillary systems data at MCR*) is 6215 kW. Alternative 'A' utilises up to 50 % of the available heat, hence 3107 kW of heat is available. Substitute this value in the equation:

- FW produced in t/day = constant x available heat
- FW produced in t/day = $32 \times 10^{-3} \times 3107$
- FW produced in t/day = **99**

8.3.2 Alternative 'B'

A freshwater generator, with an evaporator heat requirement not in excess of 85 % of the heat available to be dissipated from the cylinder cooling water at full load (CMCR), can be connected in series as shown in figure 8.3.2.1. This arrangement requires the provision of an additional automatic temperature control valve (4A), connected in cascade control with the cylinder cooling water cooler temperature control valve (4B), and controlled by the step controller (9) sensing the outlet cylinder cooling water temperature from the engine. If the cylinder cooling water outlet temperature is falling below the set point, valve (4A) reduces the flow of cylinder cooling water to the freshwater generator for compensation. A part of the cylinder cooling water 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 cylinder cooling water, valve (4A) is fully opened across connections 1 and 2, and a valve travel limit switch changes the regulation of cylinder cooling water temperature over to temperature control valve (4B). This in turn passes water to the cylinder cooling water cooler (3) to maintain the cylinder water outlet at the required temperature. If in this condition the cylinder cooling water temperature falls below the set point and the cooler (3) is fully by-passed, valve (4B) is fully opened across connections 1 and 3, and a valve travel limit switch transfers the regulation of cylinder cooling water temperature back to temperature control valve (4A). As an alternative to the single-step controller (9), two controllers can be installed, one for each valve, making sure that there is a 3 °C difference in the set point between (4A) and (4B) to avoid both controllers acting at the same time.

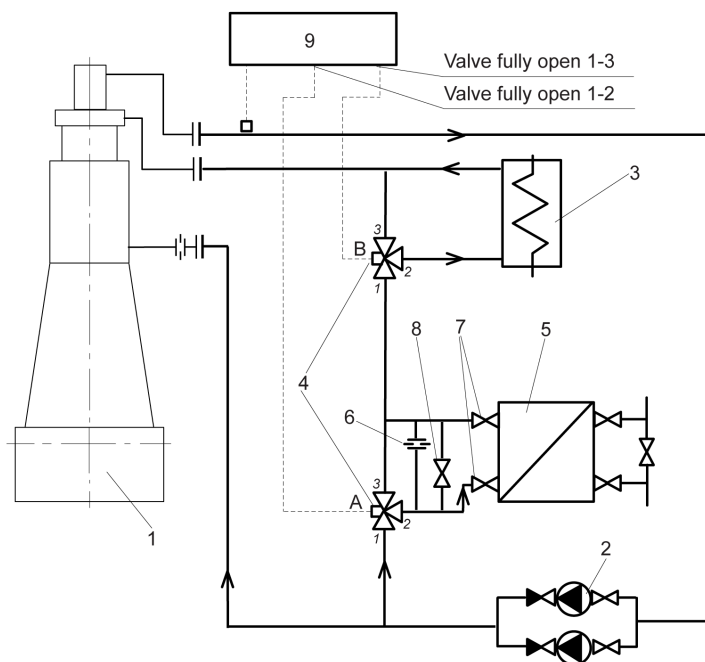


Fig 8.3.2.1 Freshwater generator installation, alternative 'B'

1	Main engine	6	Throttling disc
2	Cylinder cooling water pump	7	Freshwater valves
3	Cylinder cooling water cooler	8	Freshwater generator bypass valve
4	Automatic temperature control valve	9	Controller
5	Freshwater generator		

The quantity of freshwater produced by a single-effect vacuum (flash) evaporator can be estimated for guidance purposes as follows:

- FW produced in t/day = $32 \times 10^{-3} \times Q_{FW}$

where Q_{FW} is the heat in kW available from the cylinder cooling water, estimated from the derating table 7.1 *Ancillary systems data at MCR*.

Example:

12-cyl. engine - R1 specification with 77400 kW at 80 rpm

The available heat (from section 7.1 *Ancillary systems data at MCR*) is 6215 kW. Alternative 'B' utilises up to 85 % of the available heat, hence 5282 kW of heat is available. Substitute this value in the equation:

- FW produced in t/day = constant x available heat
- FW produced in t/day = $32 \times 10^{-3} \times 5282$
- FW produced in t/day = **169**

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 (e.g. feed water pre-heater for waste heat recovery, etc.).

8.3.3

Concept guidance for freshwater generator installation

The latest version of the concept guidance (DG 9721), is provided on the WinGD corporate webpage under the following link:

https://www.wingd.com/media/1832/concept-guidance_wingd-2s_fresh-water-generation.pdf

8.4 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.

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 cylinder water system.

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

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 cylinder cooling water 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.

The recommended temperature to start and operate the engine is 60 °C at cylinder cooling water 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 8.4.1 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.

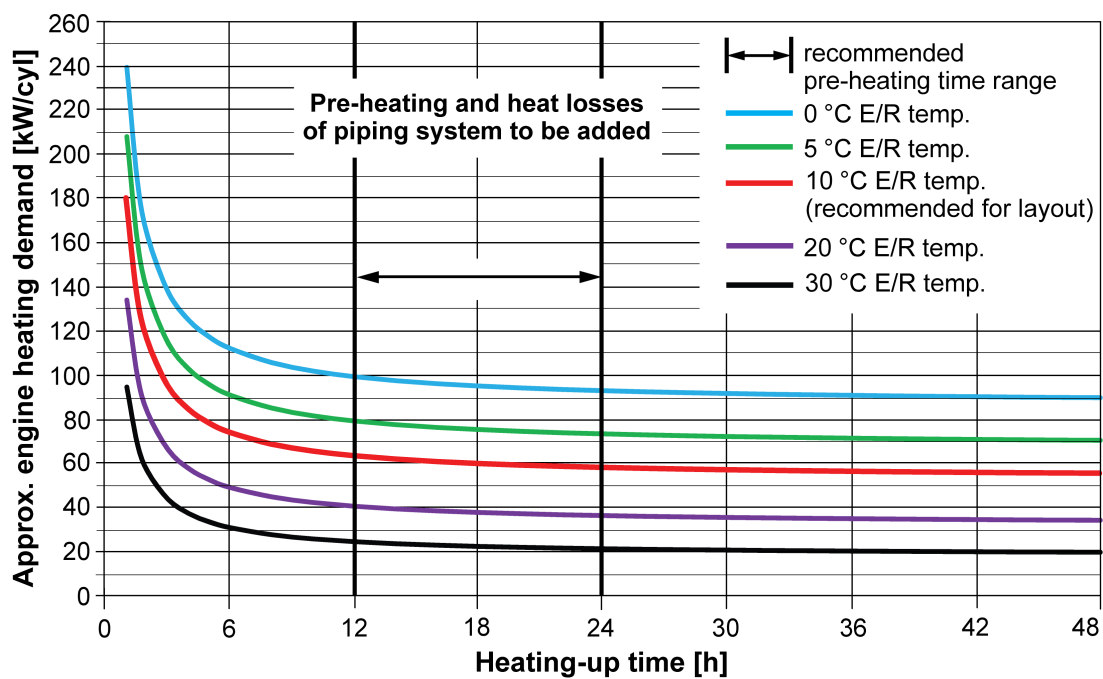


Fig 8.4.1 Pre-heating power requirement per cylinder

8.5 Installation drawings and concept guidance

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

https://www.wingd.com/media/1906/mids_wingd-x92_cooling-water-system.pdf

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9. Lubricating Oil Systems

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 as well as to lubricate and cool the torsional vibration damper and the axial vibration damper.

The consumption of system oil is given in table 1.1.1 *Rating points*.

9.1 Flushing the lubricating oil system

The latest version of the flushing instruction, relevant for the lubricating oil system (DG 9722), is provided on the WinGD corporate webpage under the following link:

https://www.wingd.com/media/1835/flushing-instruction_wingd-2s_lubricating-oil-system.pdf

9.2 Main lubricating oil system components

9.2.1 Lubricating oil systems for turbochargers

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

9.2.2 Lubricating oil pump

Positive displacement screw pumps with built-in overpressure relief valves, or centrifugal pumps (pump capacity see to tables in section 7.1 *Ancillary systems data at MCR*).

Centrifugal pump:	The given flow rate is to be within a tolerance of -10 to +10 % plus back-flushing flow of automatic filter, if any. Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Delivery head:	The final delivery head to be determined is subject to the actual piping layout. Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Oil type:	SAE30, 50 cSt at working temperature; maximum viscosity to be allowed for when sizing of pump motor is 400 cSt
Positive displacement screw pump:	The given flow rate is to be within a tolerance of 0 to +10 % plus back-flushing flow of automatic filter, if any. Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Working temperature:	60 °C

9.2.3 Lubricating oil cooler

Cooling medium:	Freshwater
Cooling water flow:	Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Cooling water temp.:	Freshwater 36 °C
Heat dissipation:	Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Margin for fouling:	10-15 % to be added
Oil flow:	Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Oil temperature at inlet:	Approx. 60 °C
Oil temperature at outlet:	45 °C
Oil viscosity at cooler inlet:	50 cSt at 60 °C
Type:	Plate or tubular
Working press. oil side:	6 bar
Working press. water side:	Approx. 3 bar

9.2.4 Lubricating oil full-flow filters

Diff. pressure, alarm:	Max. 0.8 bar
Diff. pressure, clean filter:	Max. 0.2 bar
Diff. pressure, dirty filter:	Max. 0.6 bar
Filter inserts bursting press.: ...	Min. 8 bar (= differential pressure across the filter inserts)
Filter material:	Stainless steel mesh
Mesh size:	Sphere passing max. 0.050 mm
Oil flow:	Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Test pressure:	Specified by classification society
Type*:	Change-over duplex filter designed for in-service cleaning, with differential pressure gauge and high differential pressure alarm contacts
Working pressure:	6 bar
Working viscosity:	95 cSt, at working temperature

*** Alternatively:** 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. The drain from the filter is to be sized and fitted to allow free flow into the lubricating oil drain tank. The output required by 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.

9.2.5 Crosshead LO booster pumps

NOTE



Optional for W-X92

Delivery head	See table in section 7.1 <i>Ancillary systems data at MCR</i>
Oil type	SAE 30, 95 cSt (at working temperature, maximum viscosity to be allowed for when sizing the pump motor is 400 cSt).
Pump capacity	The given flow rate is to be within a tolerance of 0 % to 10 % (see table in section 7.1 <i>Ancillary systems data at MCR</i>).
Pump type	Positive displacement screw or gear types having built-in overpressure relief valves.
Working temperature	Approx. 45 °C

9.3 Cylinder lubricating oil system

Cylinder lubrication is carried out by a separate system, working on the once-through principle normally using a high-alkaline oil of SAE 50 grade. The cylinder lubricating oil is fed to the surface of the cylinder liner by a hydraulically actuated dosage pump through quills in the cylinder liner. The oil supply rate is adjustable and metered to suit the age and running condition of the piston rings and liners.

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 is to be located at the same height as the service tank to provide the necessary head. Furthermore, it has to be of similar design, ensuring a sloping tank floor.

For cylinder lubricating oil consumption refer to table 1.1.1 *Rating points*.

9.4 Lubricating oil maintenance and treatment

It is essential that the 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.

9.4.1 Lubricating oil separator

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

9.5 Lubricating oil requirements

The products listed in section 9.5.1 were selected in co-operation with the oil suppliers and in their respective product lines are considered as appropriate lubricants for the application indicated. Winterthur Gas & Diesel Ltd. does not accept any liability for the quality of the supplied lubricating oil or its performance in actual service. In addition to the oils shown in the mentioned list, there are other brands which might be suitable for use in WinGD two-stroke diesel engines. Information concerning such brands may be obtained on request from Winterthur Gas & Diesel Ltd.

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

For the WinGD X92 engines designed with oil-cooled pistons, the crankcase oils typically used as **system oil** have the following properties (see also section 9.5.1):

- SAE 30
- Minimum BN of 5.0 mgKOH/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

*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.

9.5.1 Validated lubricating oils

The following tables with the listed suppliers of validated system oils and cylinder oils are prepared according to Appendix 1 (RT-138_A1) which is part of the Data & Specifications bulletin RT-138, entitled 'Lubricating oils'. For an up to date information: these documents and their updates can be downloaded from the WinGD Customer Portal.

<https://my.wingd.com/idp/Authn/UserPassword>

Table 9.5.1.1 Validated system oils

Oil supplier	System oil
Aegean	Alfasys 305 *2)
BP	Energol OE-HT 30
Castrol	CDX 30
Chevron	Veritas 800 Marine 30
CPC	Marilube Oil AC-30 *2)
ENI	Cladium 50
ExxonMobil	Mobilgard 300 Mobilgard 300 HD *1) Mobilgard 300C
FL Selenia	MESYS 3006
Gulf Oil Marine	GulfSea Superbear 3006 GulfSea Superbear 3008
IOC	Servo Marine 0530
JX Nippon Oil & Energy	Marine S30
LUKOIL	Navigo 6 SO Navigo 6 CO
Pertamina	Medripal 307
Petrobras	Marbrax CAD-308
PetroChina	KunLun DCC3008 KunLun DCC3005H *2)
Shell	Melina S30 Melina 30
SINOPEC	Marine System Oil 3005 Marine System Oil 3006 Marine System Oil 3008
SK	Supermar AS *2)
Total	Atlanta Marine D 3005

*1) For RT-flex and X engines built after February 2012.
*2) These system oils are under development & testing and are not yet applicable.

Table 9.5.1.2 Validated cylinder oils

Oil supplier	15 ≤ BN ≤ 25	BN 40	50 ≤ BN ≤ 60	70 ≤ BN ≤ 80	BN 100
Aegean	Alfacylo 525 DF (BN 25)			Alfacyclo 570 (BN 70) *5)	Alfacyclo 100 HS *5)
Bardahl				Naval 50 (BN 70)	
Castrol	Cyltech ACT (BN 16)	Cyltech 40 SX		Cyltech 70 (BN 70)	Cyltech 100
Chevron	Taro Special HT LF (BN 25)	Taro Special HT LS 40	Taro Special HT 55 (BN 55) *3)	Taro Special HT 70 (BN 70) Taro Special 70 (BN 70) *4)	Taro Special HT 100
CPC				Marilub Oil CO-700 Plus (BN 70) *5)	
ENI	Punica 525 (BN 25)			Punica 570 (BN 70)	
ExxonMobil	Mobilgard 525 (BN 25)	Mobilgard L 540	Mobilgard 560VS (BN 60) *1)	Mobilgard 570 (BN 70)	Mobilgard 5100
FL Selenia				MECO 5070 (BN 70)	
Gdanska				Marinol RG 7050 (BN 70)	
Gulf Oil Marine	GulfSea Cylcare ECA 50 (BN 15)			GulfSea Cylcare DCA5070H (BN 70)	GulfSea Cylcare 50100 *5)
IOC				Servo Marine 7050 (BN 70)	
JX Nippon Oil & Energy	Marine C255 (BN 25)	Marine C405 Marine C405Z		Marine C705 (BN 70)	Marine C1005 *5)
LUKOIL	Navigo MCL Ultra (BN 20) *5)	Navigo 40 MCL		Navigo 70 MCL (BN 70)	Navigo 100 MCL
Mexicana de Lubricantes				Marinelub 7050 (BN 70) *4)	
Pertamina				Medripal 570 (BN 70)	
Petrobras		Marbrax CID-54-APN	Marbrax CID-55 (BN 50) *2)	Marbrax CID-57 (BN 70)	
PetroChina				KunLun DCA 5070H (BN 70)	
Premier six		Opt-Max BoB 300 additives (BN 70 - BN 120) *5)			
Shell	Alexia S3 (BN 25)		Alexia S4 (BN 60) *1)	Alexia 50 (BN 70) Alexia S5 (BN 80)	Alexia S6
SINOPEC	Cylinder Oil 5025 (BN 25) *5)	Cylinder Oil 5040 *5)		Cylinder Oil 5070 (BN 70) *4) Cylinder Oil 5070S (BN 70) Cylinder Oil 5080S (BN 80)	Cylinder Oil 50100 *5)

Oil supplier	15 ≤ BN ≤ 25	BN 40	50 ≤ BN ≤ 60	70 ≤ BN ≤ 80	BN 100
SK	Supermar CYL 25 (BN 25)	Supermar CYL 40 Supermar CYL 40L		Supermar Cyl 70 plus (BN 70)	Supermar Cyl 100 100 *5)
Total	Talusia LS 25 (BN 25)	Talusia LS 40	Talusia Universal (BN 57) *1)	Talusia HR 70 (BN 70)	Talusia Universal 100 Talusia Optima *5)
*1) These BN 57 and BN 60 cylinder lubricants may be used with fuel in the sulphur range 0.5 % to 2.5 % m/m *2) This BN 50 cylinder lubricant can be used up to 2.5 % m/m sulphur in the fuel for engines built before 2000.			*3) This BN 55 cylinder lubricant can be used for engines built before 2011 in the sulphur range: 1.5 % to 2.5 % m/m for continuous and 0.5 % to 1.5 % m/m for intermittent operation up to 10 days. *4) Limited to engines built before 1995. *5) These cylinder lubricants are under development & testing and are not yet applicable		

NOTE



- BN refers to the lubricant Base Number expressed in mg KOH/g as determined by test method ASTM D 2896.
- Intermediate BN cylinder lubricants (50 < BN < 60 mg KOH/g) may be used the performance is regularly monitored and lubricating the oil feed rate is adjusted to avoid piston underside BN which is too low. Residual BN which is too low can lead to excessive corrosive wear and scuffing. See Data & Specifications bulletin RT-138 for further instructions.

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

9.6 Lubricating oil 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. There is to maintain adequate drainage under sea conditions resulting in pitching and rolling.

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

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. The amount of lubricating oil required for an initial charge of the drain tank is indicated in fig. 9.6.4.1. The total tank size is normally 5-10 % greater than the amount of lubricating oil required for an initial filling (see fig. 9.6.4.1).

9.6.1 Arrangement of vertical lubricating oil drains

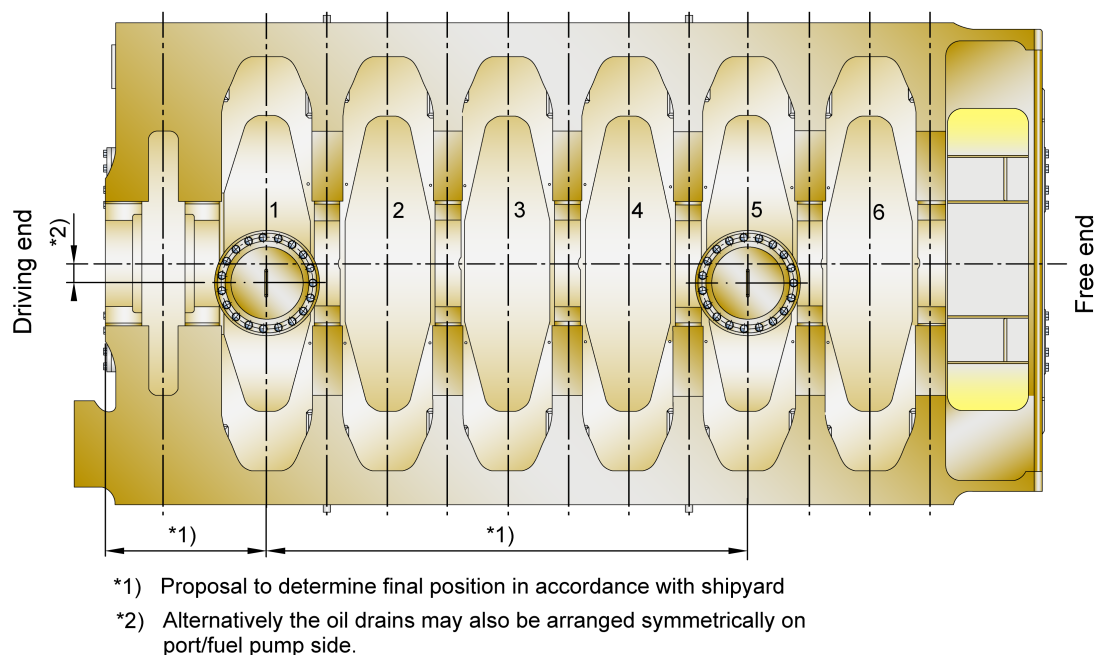


Fig 9.6.1.1 Arrangement of vertical lubricating oil drains

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 the latest version of the Marine Installation Drawing Set (MIDS) is provided on the WinGD corporate webpage. See link at the end of this chapter.

9.6.2 Inclination angles

Table 9.6.2.1 Minimum inclination angles at which the engine is to remain fully operational (1)

	Classification societies Year of latest update						
	ABS 2017	BV 2016	CCS 2015	CRS 2013	DNV 2016	DNV-GL 2016	GL 2016
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	4/1/3/B 200	4/1/2/2.2/2.2.1	I-1- 2/1/C/C.1.1
Heel to each side	15°	15°	15°	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°	22.5°	22.5°	22.5°
Trim by the head *1)	5°	5°	5°	5°	5°	5°	5°
Trim by the stern *1)	5°	5°	5°	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±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	4/1/3/B 200	4/1/2/2.2/2.2.1	I-1- 2/1/C/C.1.1
Heel to each side	22.5°	22.5°	22.5° *3)	22.5°	22.5° *3)	22.5° *3)	22.5° *3)
Rolling to each side	22.5°	22.5°	22.5° *3)	22.5°	22.5° *3)	22.5° *3)	22.5° *3)
Trim	10°	10°	10°	10°	10° *1)	10° *1)	10°
Pitching	±10°	±10°	±10°	±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	4/8/3/B 100	4/1/2/2.2/2.2.1	I-1- 2/1/C/C.1.1
Heel to each side	22.5° *2)	22.5° *2)*3)	15°	22.5° *2)	22.5°	22.5° *2)*3)	22.5° *3)
Rolling to each side	22.5° *2)	22.5° *2)*3)	22.5°	22.5° *2)	22.5°	22.5° *2)*3)	22.5° *3)
Trim	10°	10°	5°	10° *2)	10° *1)	10° *1)*2)	10°
Pitching	±10°	±10°	±7.5°	±10° *2)	±10°	±10° *2)	±10°
*1) Where the ship's length exceeds 100 m, the fore-and-aft static angle of inclination may be taken as: 500/L degrees. L = length of ship in meters. *2) Up to an inclination angle of 45°, switches and controls are to remain in their last set position as no undesired switching operations or operational changes may occur. *3) For ships carrying liquefied gases or chemicals the arrangement is to be such that the emergency power supply must also remain operable with the ship flooded to a final athwartships inclination up to 30°.							

NOTE



The data in the tables of this section represent the state of data as of the year 2017 and earlier. If you want to obtain the latest data please address yourself to the relevant classification society.

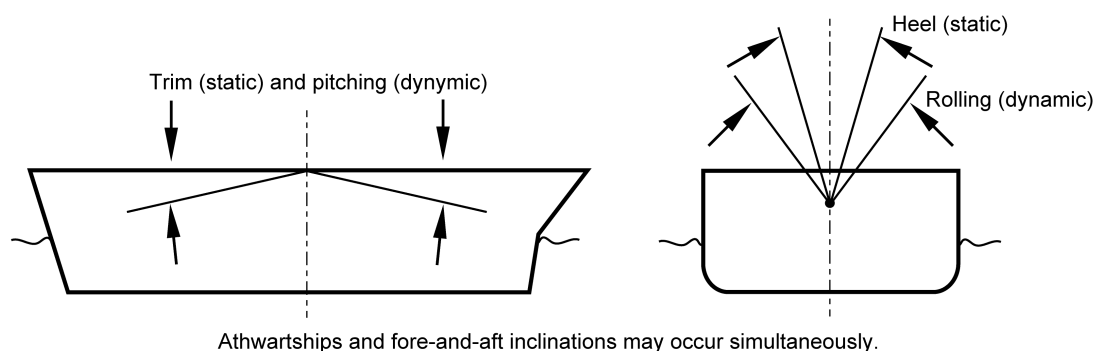


Fig 9.6.2.1 Inclination angles

Table 9.6.2.2 Minimum inclination angles at which the engine is to remain fully operational (2)

	Classification societies Year of latest update						
	IRS 2016	KR 2016	LR 2016	NK 2016	PRS 2016	RINA 2016	RS 2015
Main and auxiliary engine							
Abbreviation	4/1/1/1.7	5/1/103/1.	5/1/3/3.7	D/1.3.1/6	VI/1/1.6.1	C/1/1/2.4	2/VII/2/2.3
Heel to each side	15°	15°	15°	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°	22.5°	22.5°	22.5°
Trim by the head *1)	5°	5°	5°	5°	5°	5°	5°
Trim by the stern *1)	5°	5°	5°	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±7.5°	±7.5°	±7.5°	±7.5°
Emergency sets							
Abbreviation	4/1/1/1.7	5/1/103/1.	5/1/3/3.7.	D/1.3.1/6	VI/1/1.6.1	C/1/1/2.4	2/VII/2/2.3
Heel to each side	22.5°	22.5° *3)	22.5° *3)	22.5° *3)	22.5° *3)	22.5° *3)	22.5° *3)
Rolling to each side	22.5°	22.5° *3)	22.5° *3)	22.5° *3)	22.5° *3)	22.5° *3)	22.5° *3)
Trim	10°	10°	10°	10°	10°	10°	10°
Pitching	±10°	±10°	±10°	±10°	±10°	±10°	±10°
Electrical installation							
Abbreviation	4/1/1/1.7	5/1/103/1.	6/2/1/1.10	H/1/1.1.7	VIII/2/2.1.2.2	C/2/2/1.6	2/XI/2/2.1.2.2
Heel to each side	22.5° *2)*3)	22.5° *2)*3)	15°	15° *2)	15°	15° *2)	15°
Rolling to each side	22.5° *2)*3)	22.5° *2)*3)	22.5°	22.5° *2)	22.5°	22.5° *2)	22.5°
Trim	10° *2)	10° *2)	5°	5° *2)	5°	5° *2)	5°
Pitching	±10° *2)	±10° *2)	±7.5°	±7.5° *2)	±10°	±7.5° *2)	±10°
*1) Where the ship's length exceeds 100 m, the fore-and-aft static angle of inclination may be taken as: 500/L degrees. L = length of ship in meters. *2) Up to an inclination angle of 45°, switches and controls are to remain in their last set position as no undesired switching operations or operational changes may occur. *3) For ships carrying liquefied gases or chemicals the arrangement is to be such that the emergency power supply must also remain operable with the ship flooded to a final athwartships inclination up to 30°.							

9.6.3 Classification societies

IACS	International Association of Classification Societies		
ABS	American Bureau of Shipping	IRS	Indian Register of Shipping
BV	Bureau Veritas	KR	Korean Register of Shipping
CCS	Chinese Classification Society	LR	Lloyd's Register of Shipping
CRS	Croatian Register of Shipping	NK	Nippon Kaiji Kyokai
DNV	Det Norske Veritas	PRS	Polski Rejestr Statkow
DNV-GL	Det Norske Veritas - Germanischer Lloyd *1)	RINA	Registro Italiano Navale
GL	Germanischer Lloyd	RS	Russian Maritime Register of Shipping

*1) The rule books of DNV and GL are still valid until further notice.

9.6.4 Dimensioning guidelines and filling process

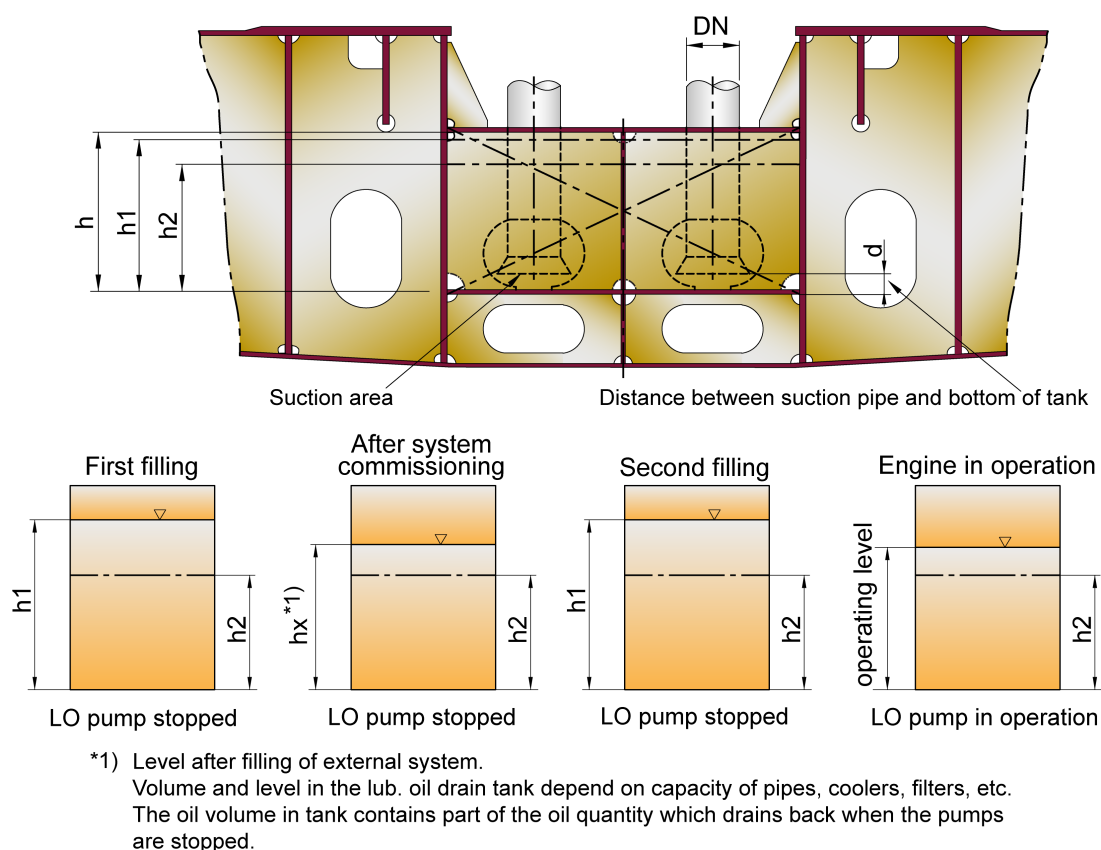


Fig 9.6.4.1 Filling process of lubricating oil tank

9.7 Installation drawings

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

https://www.wingd.com/media/1907/mids_wingd-x92_lubricating-oil-system.pdf

10. Diesel Fuel System

10.1 Flushing the fuel oil system

The instruction for flushing the fuel oil system (DG 9723), is provided on the WinGD corporate webpage under the following link:

https://www.wingd.com/media/1834/flushing-instruction_wingd-2s_fuel-oil-system.pdf

10.2 Concept guidance for operation on distillate fuels

The latest version of the concept guidance (DG 9723) is provided on the WinGD corporate webpage under the following link:

[https://www.wingd.com/media/1830/
concept-guidance_wingd-2s_operation-on-distillate-fuels.pdf](https://www.wingd.com/media/1830/concept-guidance_wingd-2s_operation-on-distillate-fuels.pdf)

10.3 Fuel oil treatment

10.3.1 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.

10.3.2 Service tanks

Most of the service tank design features are similar to the settling tank, having 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. The recirculation pipe reaches to the lower part of the service tank to guide water which may be present in the fuel after the separators (e.g. due to condensation or coil leakage) into the settling tank. A pipe to the separators should be provided to reclean the fuel in case of dirty water contamination. This line should be connected just above the drain valve at the service tank bottom.

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.

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.

10.3.3 Centrifugal separators

Separator type - self-cleaning: It is advisable to use fuel oil separators without gravity discs to meet the process requirements of the marine diesel oil and 700 cSt heavy fuel oils. These separators are self-adjusting and do not require gravity discs to be changed for different fuel densities. The manufacturers claim extended periods between overhaul and greatly improved reliability, enabling unattended onboard operation. The required minimum effective throughput capacity of the separators is determined as shown in the following example. The nominal separator capacity and the installation are to comply with the recommendations of the separator manufacturer.

Throughput capacity = $1.2 \times \text{CMCR} \times \text{BSFC} \times 10^{-3}$ [litres/hour]

Example

8-cyl. engine with CMCR R1:

- BSFC: 165.8 g/kWh

CMCR: 51600 kWh (see table 1.1.1 *Rating points*)

Throughput = $1.2 \times 51600 \times 165.8 \times 10^{-3} = 14653$ litres/hour

The marine diesel oil (MDO) separator capacity can be estimated using the same formula.

Separator without gravity disc:

One of the main features of the self-adjusting separators is that only a single unit is required. This unit operates as a combined purifier/clarifier. However, as it is usual to install a standby separator as a back-up, it is of advantage to use the separator to improve the separation result. For the arrangement of the separators, parallel or in series, refer to the manufacturer's instructions.

Separator with gravity disc:

These types are running in series with the fuel being purified in one and clarified in the other; thus two separators are required. The clarifier improves the separation result and acts as a safety device in case the purifier is not properly adjusted. When processing heavy fuel oils it is indispensable to strictly adhere to the separator manufacturer's instructions. If using these separators it will be advantageous to install an extra separator for marine diesel oil, only to avoid the changing of gravity discs when changing over from HFO to MDO separation.

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:

- $n = 100 \cdot (1 - C_{out}/C_{in})$

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

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

10.3.4 Concept guidance for fuel oil treatment

The latest version of the concept guidance, relevant for the fuel oil treatment (DG 9723), is provided on the WinGD corporate webpage under the following link:

[https://www.wingd.com/media/1831/
concept-guidance_wingd-2s_fuel-oil-treatment.pdf](https://www.wingd.com/media/1831/concept-guidance_wingd-2s_fuel-oil-treatment.pdf)

10.4 Fuel specification

The following specification is prepared according to 'Data & specifications' RT-126, Issue 2. For an up to date information: this document and their updates can be downloaded from the WinGD Customer Portal.

<https://my.wingd.com/idp/Authn/UserPassword>

Parameter	Unit	Bunker limit	Test method	Required fuel quality at engine inlet
Kinematic viscosity at 50 °C	mm ² /s [cSt] *1)	max. 700	ISO 3104	12 to 20 *2)
Density at 15 °C	kg/m ³	max. 1010 *3)	SO 3675/12185	max. 1010
CCAI	--	max. 870	Calculated	max. 870
Sulphur *4)	m/m [%]	Statutory requirements	ISO 8754/14596	max. 3.5
Flash point	°C	min. 60	ISO 2719	min. 60
Hydrogen sulphide *5)	mg/kg	max. 2	IP 570	max. 2
Acid number	mg KOH/g	max. 2.5	ASTM D 664	max. 2.5
Total sediment aged	m/m [%]	max. 0.1	ISO 10307-2	max. 0.1
Carbon residue micro method	m/m [%]	max. 20	ISO 10370	max. 20
Pour point (upper) *6)	°C	max. 30	ISO 3016	max. 30
Water	v/v [%]	max. 0.5	ISO 3733	max. 0.2
Ash	m/m [%]	max. 0.15	ISO 6245	max. 0.15
Vanadium	mg/kg [ppm]	max. 450	ISO 14597 / IP 501/470	max. 450
Sodium	mg/kg [ppm]	max. 100	IP 501 / IP 470	max. 30
Aluminium + Silicon	mg/kg [ppm]	max. 60	ISO 10478 / IP 501 / 470	max. 15
Used lubricating oils (ULO) may not be present: Calcium and zinc Calcium and phosphorous	mg/kg	ULO present if: Ca>30 and Zn>15 or Ca>30 and P>15	IP 501 or IP 470 IP500	Do not use if: Ca>30 and Zn>15 or Ca>30 and P>15

*1) 1 mm²/s = 1 cSt (Centistroke).

*2) 13 to 17 mm²/s range is recommended for the operation with HFO.

*3) Limited to max. 991 kg/m³ if the fuel treatment plant cannot remove water from high-density fuel oil.

*4) ISO 8217:2012, RMK700. Note that lower sulphur limits can apply based on statutory requirements and sulphur limits are not defined in ISO 8217:2012.

*5) The hydrogen sulphide limit is applied since 1st July 2012.

*6) Purchasers must make sure that the equipment on board is capable of maintaining the fuel at a temperature above the Pour Point at all times, especially in cold climates.

NOTE



The maximum allowed temperature of the fuel at engine inlet for all WinGD X engines is 150 °C.

10.5 Heavy fuel oil system components

10.5.1 Fuel oil feed pump

Type: **positive displacement screw pump with built-in overpressure relief valve**

Capacity: the given capacity is to be within a tolerance of 0 to +20 %. Refer to tables in section 7.1 *Ancillary systems data at MCR*.

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 following formula and example:

$$\text{Delivery gauge pressure} = p_v + 1 + \Delta p_1 + \Delta p_2 \text{ [bar]}$$

Electric motor: the electric motor driving the fuel oil feed pumps 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

Fuel type: marine diesel oil and heavy fuel oil, up to 700 cSt at 50 °C

Working temperature: ambient to 90 °C

p_v = water vapour gauge pressure at the required system temperature [bar] (see viscosity / temperature diagram in 'Data & specifications' RT-126, as indicated in 10.4 *Fuel specification*)

Δp_1 = maximum pressure losses between the feed pumps and the mixing unit [bar]

Δp_2 = maximum pressure change difference across the pressure regulating valve of the feed system between minimum and maximum flow (see 10.5.2)

Example

HFO of 700 cSt at 50 °C

Required system temperature: approx. 145 °C

Water vapour gauge pressure at 145 °C: $p_v = 3.2$ bar

Press. losses betw. feed pump and mixing unit: ... $\Delta p_1 = 0.5$ bar

Pressure change difference across the pressure regulating valve: $\Delta p_2 = 0.6$ bar

Substituting these values in the formula: delivery pressure = $3.2 + 1 + 0.5 + 0.6 = 5.3$ bar

10.5.2 Pressure regulating valve

The pressure regulating valve maintains the inlet pressure to the booster system practically constant, irrespective of the actual amount of fuel consumed by the main engine and the auxiliaries. It 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.**

Fuel oil viscosity: 100 cSt, at working temp. (HFO 700 cSt at 50 °C)

Inlet pressure change: ≤ 0.8 bar, between 20 and 100 % flow (upstream pressure build-up over the valve capacity; between the minimum and maximum flow capacity)

Maximum capacity: refer to feed pump capacity. Refer to tables in section 7.1 *Ancillary systems data at MCR*.

Minimum capacity: approx. 20 % of that of the feed pump

Pressure setting range: 2-6 bar

Service pressure: max. 10 bar

Working temperature: ambient to 90 °C

10.5.3 Mixing unit

Due to the small amount of fuel consumed, 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.

The mixing unit equalizes the temperature between the hotter fuel oil returning from the engine and the cooler fuel oil from the service tank, particularly when changing over from heavy fuel oil to marine diesel oil and vice versa.

Type: **cylindrical steel fabricated pressure vessel See figure 10.5.3.1**

Capacity: Refer to tables in section 7.1 *Ancillary systems data at MCR*.

Dimensions: See 'List of Documents' at the end of the chapter

Service pressure: 10 bar

Test pressure: according to the classification society

Working temperature: ambient to 150 °C

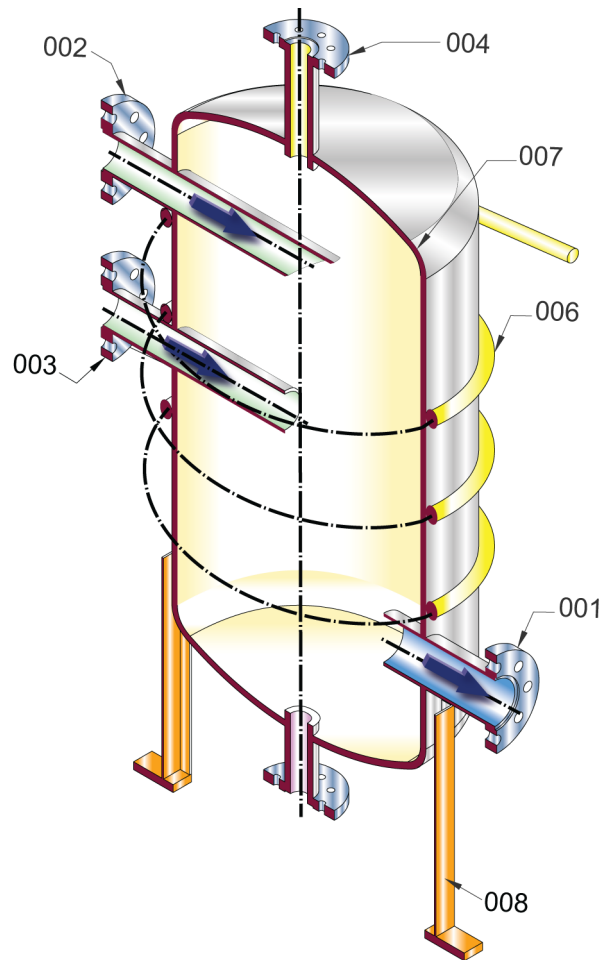


Fig 10.5.3.1 Fuel oil system mixing unit

001	Outlet	003	Inlet, from feed pump	005	Drain	007	Insulation
002	Inlet, return pipe	004	Vent	006	Heating coil	008	Mounting bracket *1)

Configuration and dimensioning of the mixing unit have to comply with the relevant classification society/rules.
 *1) Mounting brackets for fixation on floor plate. The mixing unit must not, under any circumstances, be fitted unsupported.

10.5.4 High-pressure booster pump

Type: **positive displacement screw pump with built-in overpressure relief valve**

Capacity: the given flow rate is to be within a tolerance of 0 to +20 %. Refer to tables in section 7.1 *Ancillary systems data at MCR*.

Delivery head: final delivery pressure according to the actual piping layout. Refer to tables in section 7.1 *Ancillary systems data at MCR*.

Electric motor: the electric motor driving the high-pressure 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

Inlet pressure: up to 6 bar

Working temperature: ambient to 150 °C

10.5.5 Fuel oil end heater

Type: **steam, electric or thermal oil, tubular or plate type heat exchanger suitable for heavy oils up to 700 cSt at 50 °C**

Consumption of saturated steam: $= 1.32 \times 10^{-6} \times \text{CMCR} \times \text{BSFC} \times (T_1 - T_2)$ (at 7 bar gauge pressure [kg/h])
 where:

- BSFC is the brake specific fuel consumption at the contract maximum continuous rating (CMCR).
- T_1 is the temperature of the fuel oil at the viscosimeter.
 T_2 is the temperature of the fuel oil from the service tank.

Heating capacity [kW]: $= 0.75 \times 10^{-6} \times \text{CMCR} \times \text{BSFC} \times (T_1 - T_2)$

Working pressure: max. 12 bar, pulsating on fuel oil side

Working temperature: ambient to 150 °C, outlet temperature on fuel oil side

The viscosimeter monitors the fuel viscosity before the supply unit and transmits signals to the heater controls to maintain the viscosity by regulating the fuel temperature after the end heater.

10.5.6 Fuel oil filter

In the fuel oil system, between service tank and engine inlet, fine filtration by a **10 micron absolute filter** (sphere passing mesh size) is **required**. In addition a fuel filter close to the engine inlet is required for protecting the engine against any foreign particles. These requirements can be fulfilled by the following two system arrangements:

Arrangement A: fine filter in the feed system (recommended)

10 micron fine filter in the feed line

The 10 micron fine filter is installed in the 'cold' feed system. This installation position is recommended as the flow rate in the feed system is less compared to the booster system and the risk for filter clogging is limited due to the lower fuel oil temperature. This filter is used to protect the engine against serious damages as it should catch the catalytic fines which were not removed from the fuel by the separator. In addition such a filter provides a good indication, if the separator efficiency is sufficient or if improvements are required.

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

Working viscosity:	100 cSt, for HFO of 700 cSt at 50 °C
Flow rate:	The given capacities cover the needs of the engine only. The feed pump capacity must be increased by the quantity needed for the back-flushing of the filter. Refer to tables in section 7.1 <i>Ancillary systems data at MCR</i> .
Service press. at filter inlet, after feed pumps:	10 bar
Test pressure:	Specified by classification society
Permitted differential press. at 100 cSt:	• clean filter: max. 0.2 bar • dirty filter: 0.6 bar • alarm setting: max. 0.8 bar
Min. bursting press. of filter insert:	max. 8 bar differential across filter
Working temperature:	Ambient to 90 °C
Mesh size:	Max. 10 micron absolute (sphere passing mesh)
Mesh size for bypass filter:	Max. 25 micron absolute (sphere passing mesh)
Filter insert material:	Stainless steel mesh (CrNiMo)

60 micron filter in the booster circuit

A filter with maximum 60 micron (absolute sphere passing mesh size) is installed in the booster system close to the engine inlet. Its function is to protect the engine against foreign particles coming from the system. A duplex filter is sufficient as most particles are already kept by the fine filter in the feed line.

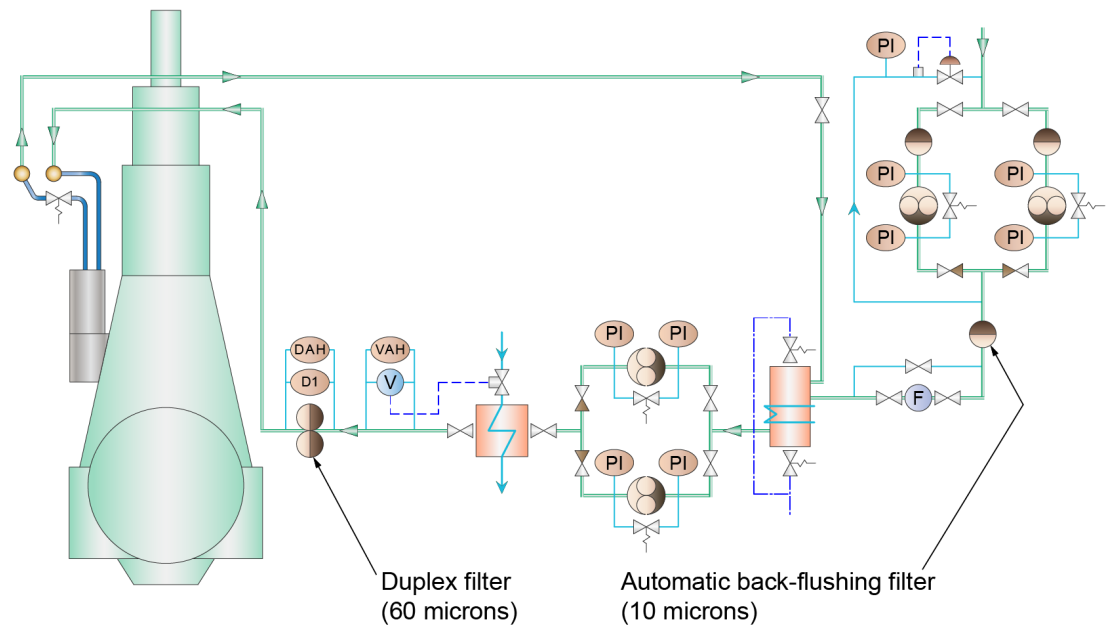
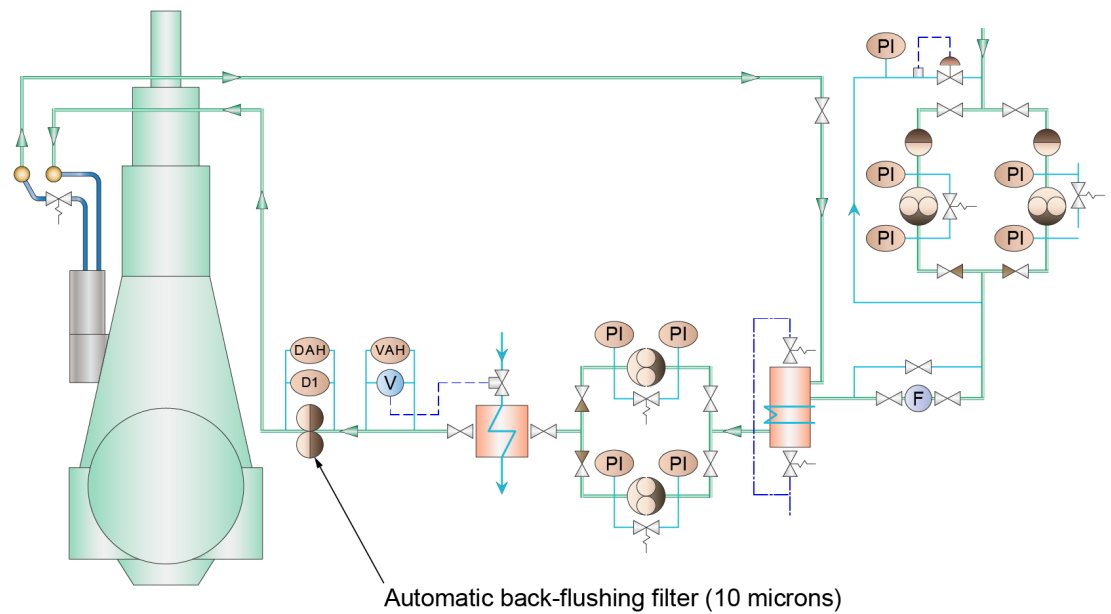
Working viscosity:	100 cSt, for HFO of 700 cSt at 50 °C
Flow rate:	The given capacities cover the needs of the engine only. The feed pump capacity must be increased by the quantity needed for the back-flushing of the filter (see tables in section 7.1 <i>Ancillary systems data at MCR</i>).
Service pressure at filter inlet:	10 bar
Test pressure:	Specified by classification society

Permitted differential pressure at 100 cSt	• clean filter: max. 0.2 bar • dirty filter: 0.6 bar • alarm setting: max. 0.8 bar
Minimum bursting pressure of filter insert:	Max. 8 bar differential across filter
Working temperature:	150 °C
Mesh size:	Max. 60 micron, sphere passing mesh
Filter insert material:	Stainless steel mesh (CrNiMo)

Arrangement B: fine filter in the booster circuit

The 10 micron (absolute sphere passing mesh size) fine filter is installed in the booster circuit close to the engine inlet. The high temperature in this circuit might increase the risk of filter clogging due to agglomeration of asphaltenes. The filter needs to be laid out for a maximum working temperature of 150 °C (in general special high-temperature gaskets are required).

Working viscosity:	10-20 cSt (mm ² /s) for HFO required, but 13-17 cSt (mm ² /s) recommended
Flow rate:	The given capacities cover the needs of the engine only. If an automatic back-flushing filter type is installed, the feed and booster pump capacities must be increased by the quantity needed for back-flushing of the filter (see tables in section 7.1 <i>Ancillary systems data at MCR</i>).
Service pressure at filter inlet:	Max. 12 bar at filter inlet
Test pressure:	Specified by classification society
Permitted differential pressure: at 17 cSt	• clean filter: max. 0.2 bar
and at 20 cSt	• dirty filter: 0.6 bar • alarm setting: max. 0.8 bar
Minimum bursting pressure of filter insert:	Max. 8 bar differential across filter
Working temperature:	150 °C
Mesh size:	Max. 10 micron absolute (sphere passing mesh)
Mesh size for bypass filter:	Max. 25 micron absolute (sphere passing mesh)
Filter insert material:	Stainless steel mesh (CrNiMo)

Arrangement A: Back-flushing filter in the feed system (recommended)**Arrangement B: Back-flushing or duplex filter before engine inlet (minimum requirement)****Fig 10.5.6.1 Fuel oil filter arrangements A and B**

10.6 Installation drawings and concept guidance

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

https://www.wingd.com/media/1908/mids_wingd-x92_fuel-oil-system.pdf

11. Starting and Control Air Systems

Compressed air is required for engine starting, engine control, exhaust valve air springs, washing plant for the scavenge air coolers, and general services. The starting and control air system shown in figure 11.1 comprises two air compressors, two air receivers, and systems of pipework and valves connected to the engine starting air manifold.

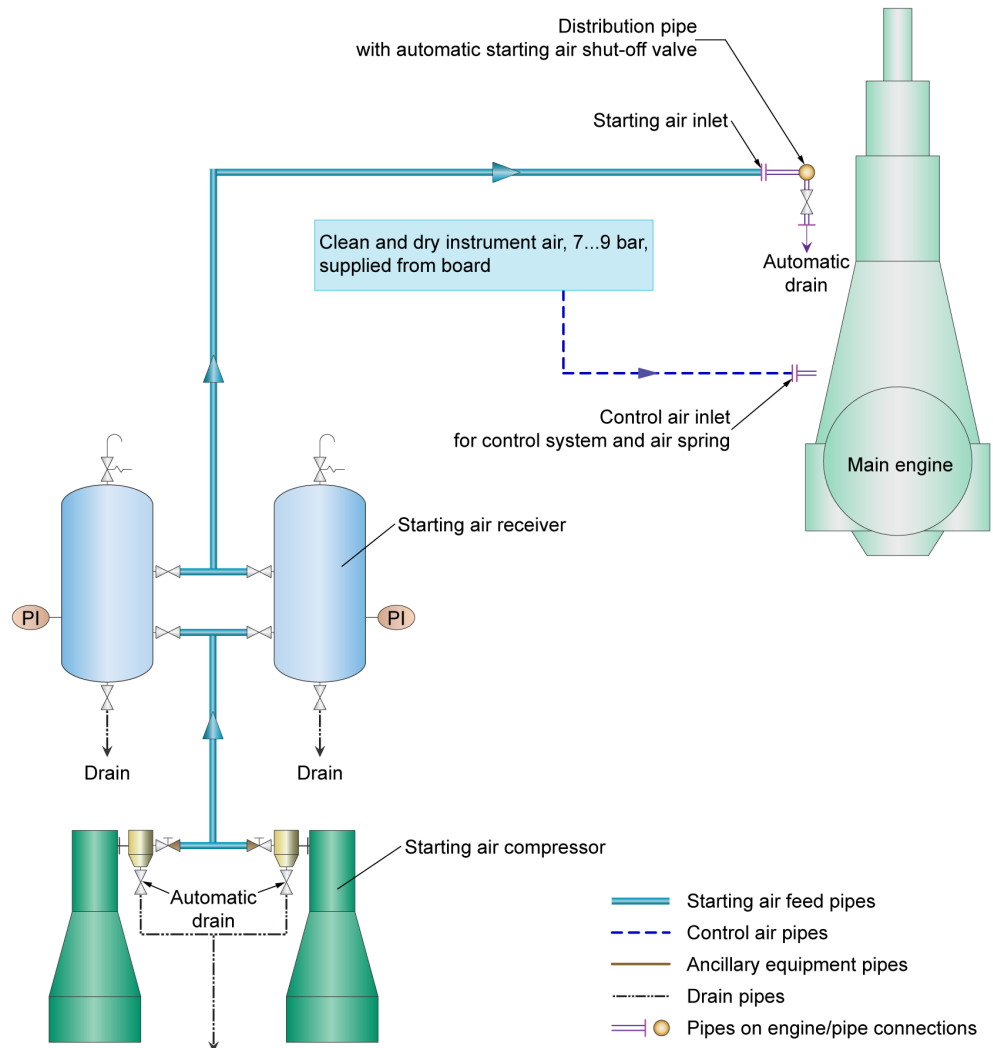


Fig 11.1 Starting and control air system

11.1 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 => (J_{Tot}) = (J_{Eng}) + (J_{S+P})
- Propeller inertia includes the part of entrained water
- Engine inertia (J_{Eng}) see section 11.1.1
- Relative inertia $J_{Rel} = J_{Tot} / J_{Eng}$

The air receiver and compressor capacities of section 11.1.1 refer to a relative inertia ($J_{Rel} = 2.0$).

For other values than 2.0, the air receiver and compressor capacities have to be calculated with the *GTD*. It provides data on the capacity of air compressor and receiver for relative inertia values (J_{Rel}).

Section 11.1.1 outlines the basic requirements for a system similar to figure 11.1 for maximum engine rating. The *GTD* program enables to optimise the capacities of the compressors and air receivers for the contract maximum continuous rating (CMCR).

11.1.1 Air receiver and air compressor capacities

Table 11.1.1.1 Air receiver and air compressor capacities

No. cyl.	Air receivers *1) Max. air pressure: 30 bar	Air compressors *1) Free air delivery at 30 bar	J_{Eng} *2) [kgm ²]
	Number x volume [m ³]	Number x capacity [Nm ³ /h]	
6	2 x 15.5	2 x 465	401600
7	2 x 16	2 x 480	466650
8	2 x 16.5	2 x 495	560000
9	2 x 16.5	2 x 495	628650
10	2 x 17	2 x 510	710000
11	2 x 17	2 x 510	790000
12	2 x 17.5	2 x 525	875000

For reversible engines the classification societies require 12 consecutive starts, alternating between ahead and astern.

*1) Data for air pressure 25 bar available from *GTD* program

*2) Data for engines without damper and front disc on crankshaft, but smallest flywheel included

11.2 Starting air compressors

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

Capacity: see section 11.1.1

Delivery gauge pressure: 30 bar

11.3 Starting air receivers

Type: fabricated steel pressure vessels having domed ends and integrated pipe fittings for isolating valves, automatic drain valves, pressure reading instruments and pressure relief valves

Capacity: see section 11.1.1

Working gauge pressure: 30 bar

11.4 Control air

Control air system supply

The control air is supplied from the board instrument air supply system (see figure 11.1) 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 the following table. These data can be used for sizing the relevant engine external piping and facilities.

No. of cyl.	Control air flow capacity [Nm ³ /h]
6	14.4
7	16.8
8	19.2
9	21.6
10	24.0
11	26.4
12	28.8

11.5 General service and working air

General 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.

11.6 Installation drawings

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

https://www.wingd.com/media/1910/mids_wingd-x92_starting-air-system.pdf

12. Leakage Collection System

Dirty oil collected from the piston underside is led under a pressure of approximately 2.8 bar to the sludge oil trap and then to the sludge oil tank. The purpose of the sludge oil trap is to retain the large amount of solid parts contained in the 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. The dirty oil from the piston rod stuffing box, which consists of waste system oil, cylinder oil, metallic particles and small amounts of combustion products, is led directly to the sludge tank. Condensate from scavenge air is formed when the vessel is operating in a humid climate and is to be continually drained from the scavenge air receiver to avoid excessive piston ring and liner wear.

12.1 Sludge oil trap

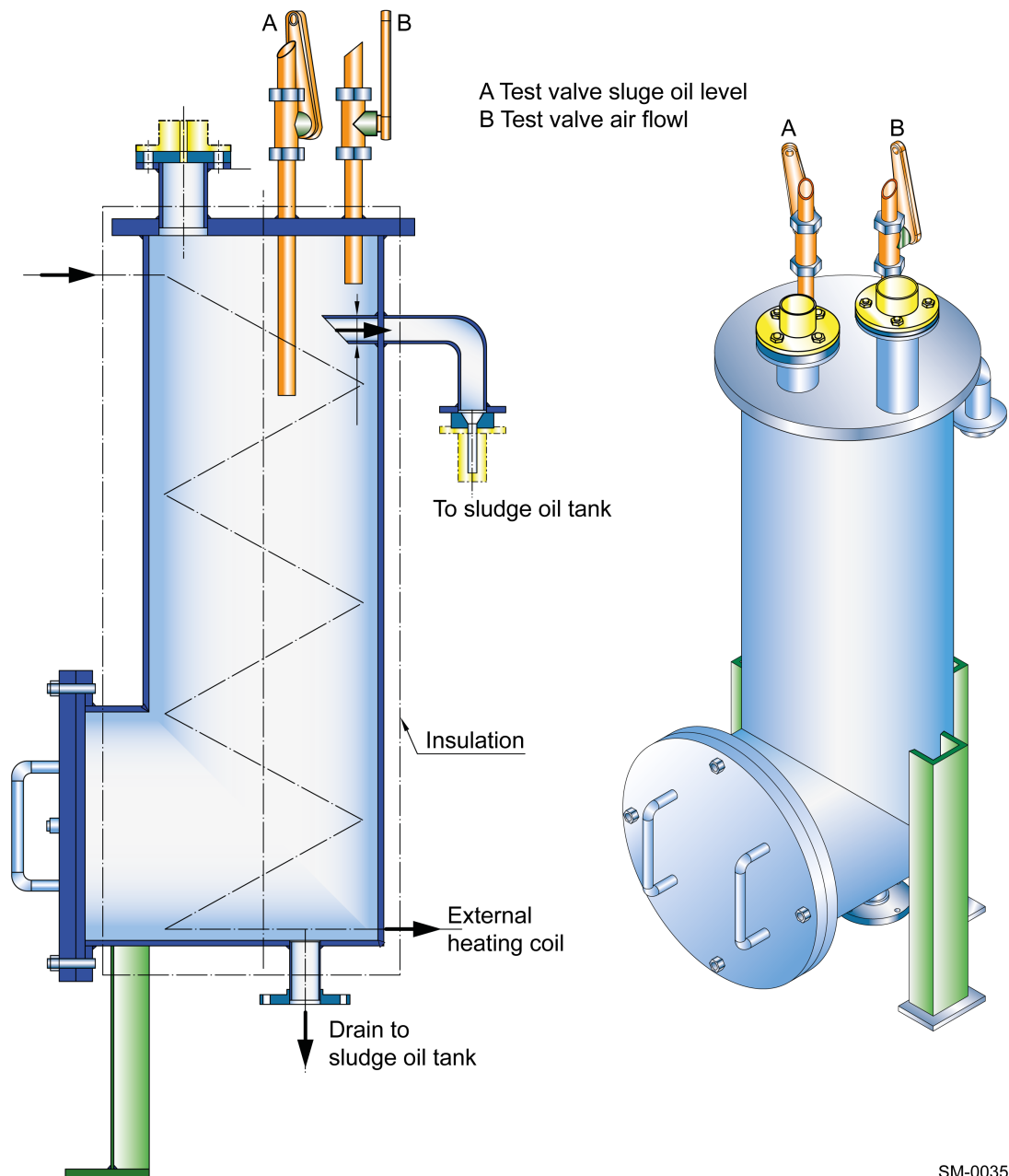


Fig 12.1.1 Sludge oil trap

For design and dimensions see drawing of the Marine Installation Drawing Set (MIDS) which can be downloaded by the link at the end of this chapter.

Engine exhaust uptakes can be drained automatically using a system as shown in figure 12.1.2.

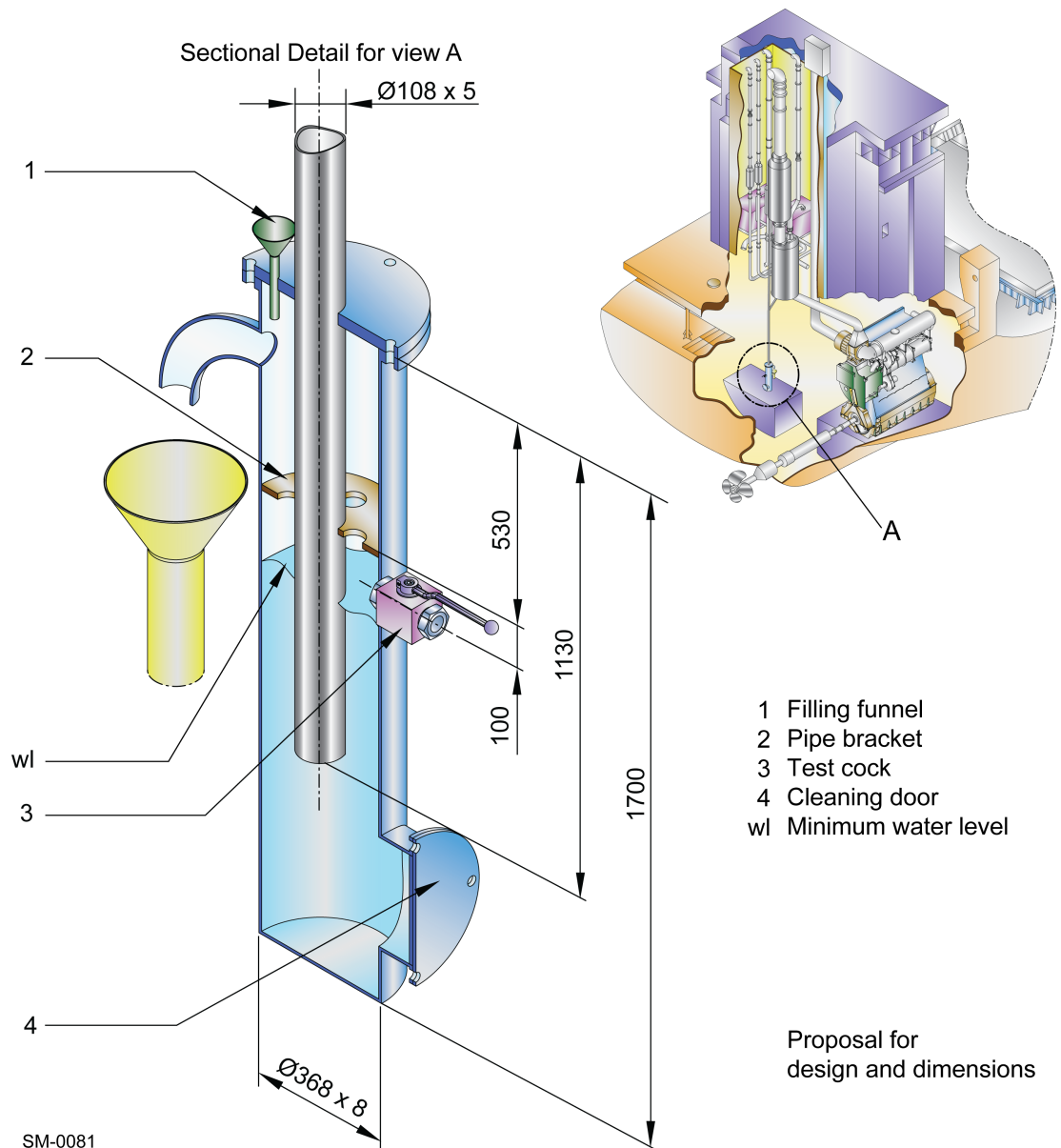


Fig 12.1.2 Arrangement of automatic water drain

12.2 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 an arrangement enables the vapour to separate into its air and fluid components, discharging the air to atmosphere and returning the fluid to its source.

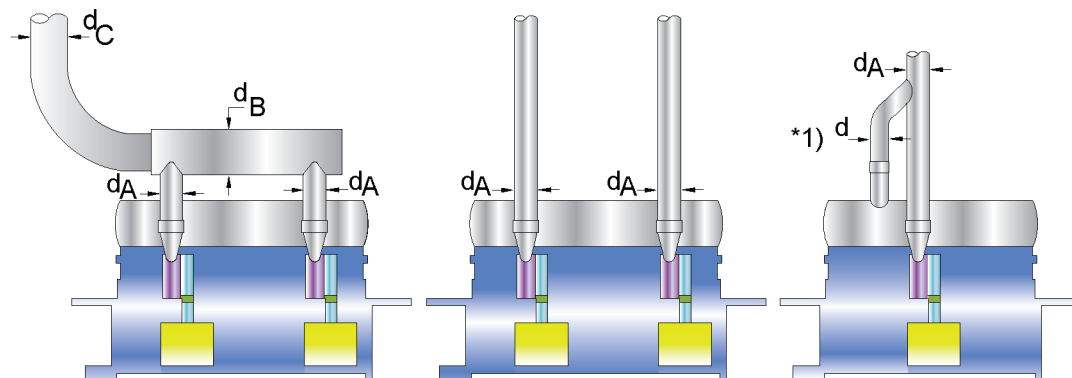
12.3 Installation drawings

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

[https://www.wingd.com/media/1909/
mids_wingd-x92_leakage-collection_and_washing-system.pdf](https://www.wingd.com/media/1909/mids_wingd-x92_leakage-collection_and_washing-system.pdf)

13. Exhaust Gas System

The following gas velocities are indicated as a guideline for an optimised exhaust gas system.



- Remarks:**
- *1) The reason for this bypass pipe is to allow engine operation after a turbocharger failure, during normal operation it is blinded off. The bypass can be omitted if agreed with the classification society and the owner.
 - 2) The flange connection of the compensator can be designed as follows:

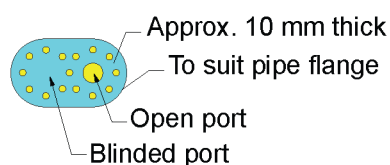


Fig 13.1 Determination of exhaust pipe diameter

13.1 Recommended gas velocities

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

For the values of the pipe diameters see GTD.

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14. Engine Room Ventilation

The 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'.

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

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

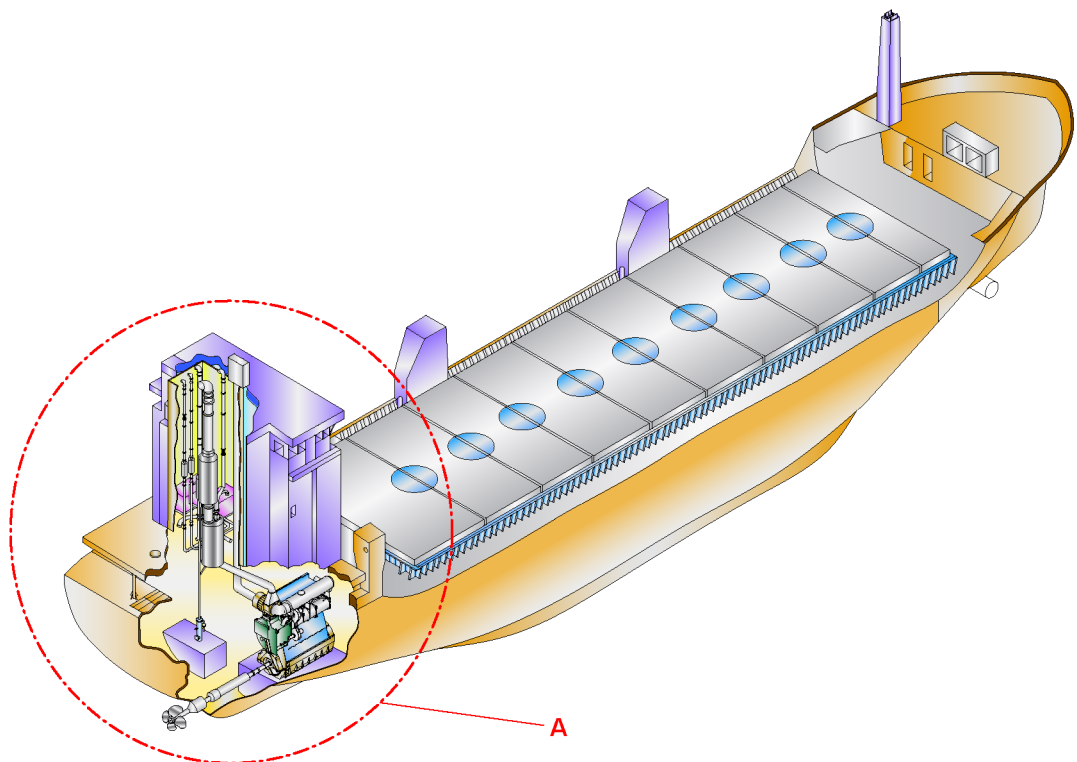


Fig 14.1 Direct suction of combustion air - main and auxiliary engine

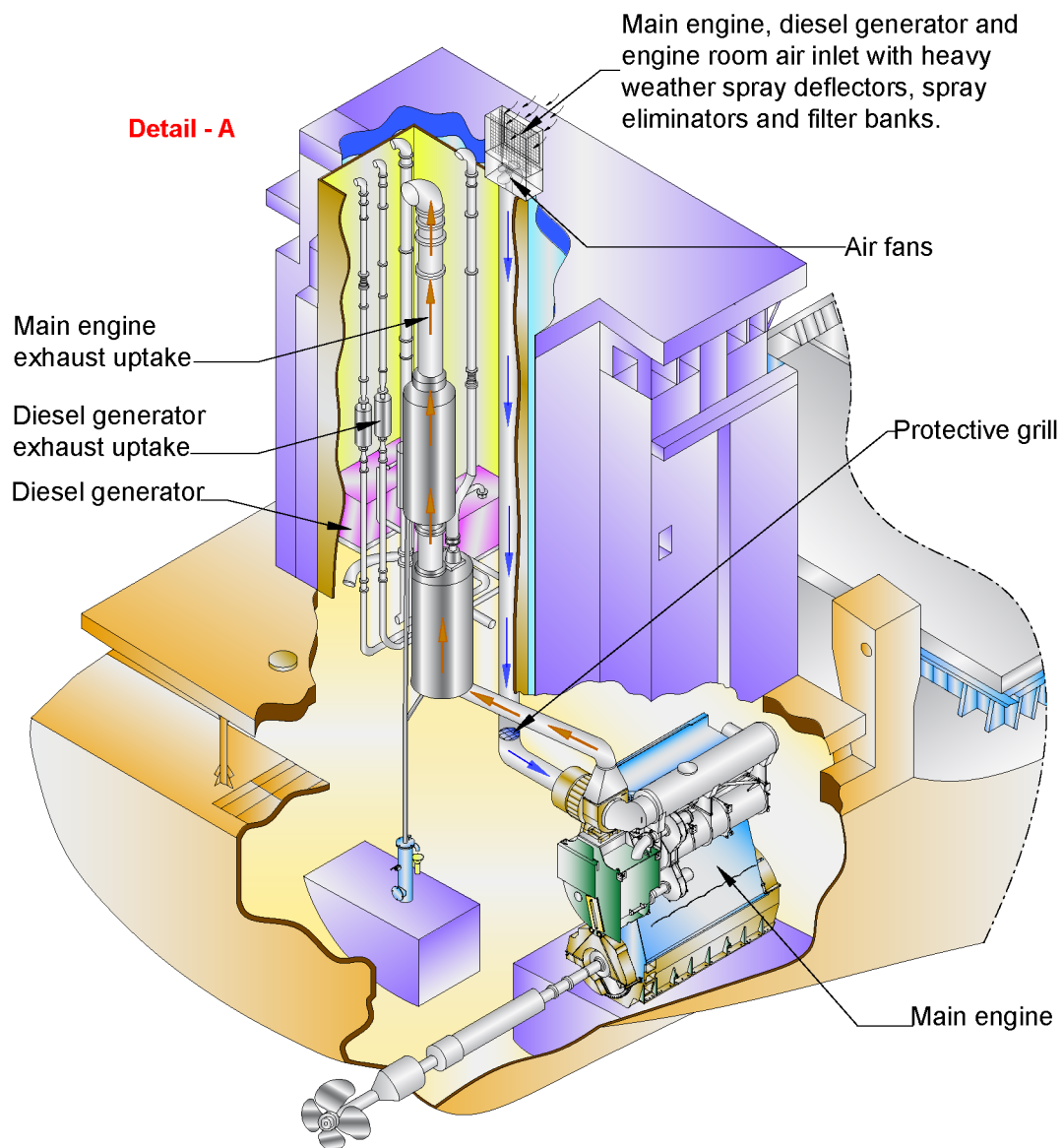


Fig 14.2 Direct suction of combustion air - main and auxiliary engine

14.1 Engine air inlet - Operating temperatures of 45 to 5 °C

The WinGD X92 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 not lower than 25 °C.

This means:

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

14.2 Arctic conditions at operating temperatures of less than 5 °C

Under arctic conditions the ambient air temperatures can meet levels of more than minus 50 °C. 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).

To avoid the need of providing an expensive combustion air pre-heater, a system has been developed that enables the engine to be operated directly with cold air from outside.

If the air inlet temperature drops to less than 5 °C, the air density in the cylinders increases to such an extent that the maximum permissible cylinder pressure is exceeded. This can be compensated by blowing off a certain amount of the scavenge air through a blow-off device as shown in figure 14.2.1.

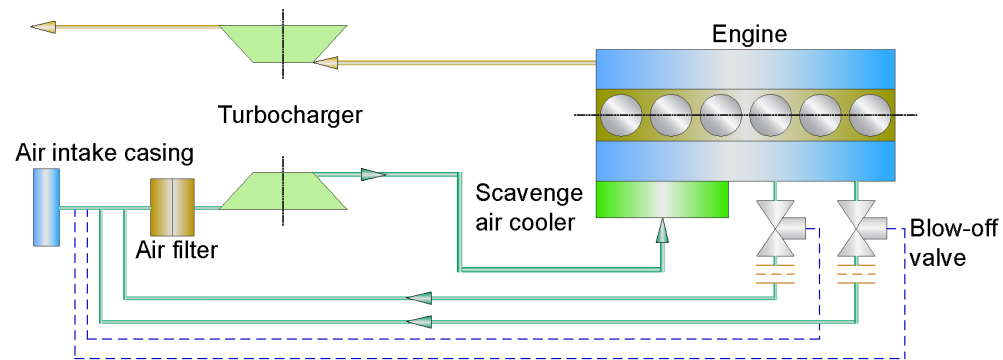


Fig 14.2.1 Scavenge air system for arctic conditions

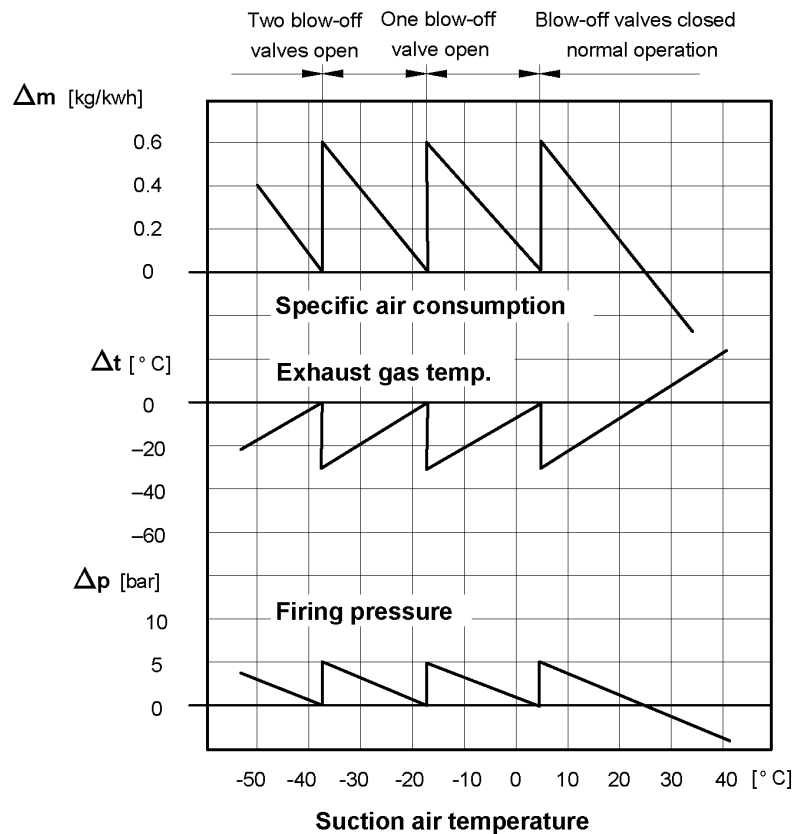


Fig 14.2.2 Blow-off effect under arctic conditions

There are up to three blow-off valves fitted on the scavenge air receiver. In the event that the air inlet temperature to the turbocharger is less than +5 °C the first blow-off valve vents. For each actuated blow-off valve, a higher suction air temperature is simulated by reducing the scavenge air pressure and thus the air density. The second blow-off valve vents automatically as required to maintain the wanted relationship between scavenge and firing pressures. Figure 14.2.2 shows the effect of the blow-off valves on the air flow, the exhaust gas temperature after turbine and the firing pressure.

Control of the blow-off valves is effected by means of a signal generated by the temperature sensors in the inlet piping. Care is to be taken that no foreign particles in the form of ice gain access to the turbocharger compressor in any way, because they could lead to its destruction. Reduction of the pipe's cross sectional area by snow is also to be prevented.

NOTE



The scavenge air cooling water inlet temperature is to be maintained at min. 25 °C. This means that the scavenge air cooling water will have to be pre-heated in case of low-power operation. The required heat is obtained from the lubricating oil cooler and the engine cylinder cooling.

15. Pipe Connections

15.1 Drawings

The latest version of the pipe connection plans (DG 8020) is provided on the WinGD corporate webpage. Tab 'Drawings' is accessible under the following link:

Link is under preparation. Please ask WinGD Switzerland directly.

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16. Engine Automation

Developments in Automation & Controls at Winterthur Gas & Diesel Ltd. are focussed on the latest trends in ship automation that tend to still higher integration levels.

The standard electrical interface, designated DENIS-9520 (**D**iesel **E**ngine **C**ontrol and opt**I**mizing **S**pecification), assures a perfect match with approved remote control systems, while the WECS-9520 (WinGD Engine Control System) takes care of all Flex system-specific control functions.

All those systems provide data bus connection to the ship automation to facilitate installation and make specific data available wherever required. Complete ship automation systems provided by one of the leading suppliers approved by Winterthur Gas & Diesel Ltd. offer the degree of integration demanded in modern shipbuilding while being perfectly adapted to the engine's requirements.

Applying a single supplier strategy for the entire ship automation shows many other advantages in terms of full responsibility, ease in operation and maintenance.

DENIS The DENIS family contains specifications for the engine management systems of all modern types of WinGD two-stroke marine diesel engines. The diesel engine interface specification applicable is DENIS-9520.

WECS-9520 Under the designation of WECS-9520, Winterthur Gas & Diesel Ltd. provides a computerised control system for all Flex system functions. As such it is a component of the Flex system and includes all necessary interfaces to the engine as well as to the remote control and electronic speed control system.

16.1 Concept

The concept of DENIS-9520 meets the requirements of increased flexibility and higher integration in modern ship automation and provides the following advantages for shipowners, shipyards and engine builders:

Clear interface definition

The well defined and documented interface results in a clear separation of the responsibilities between engine builder and automation supplier. It allows authorised suppliers to adapt their systems to the common rail system engines with reduced engineering effort. The clear signal exchange simplifies troubleshooting.

Approved propulsion control systems

Propulsion control systems including remote control, speed control, safety and telegraph systems are available from suppliers approved by Winterthur Gas & Diesel Ltd. This cooperation ensures that these systems fully comply with the specifications of the engine designer.

Easy integration in ship management system

Providing data bus communication between WECS-9520, the propulsion control and the vessel's alarm and monitoring system enables an easy integration of the different systems. The man-machine interface (MMI) of the vessel's automation can therefore also handle the additional MMI functions attributed to the WECS-9520.

Ship automation from one supplier - integrated solution

Automation suppliers approved by Winterthur Gas & Diesel Ltd. can handle all ship board automation tasks. Complete automation systems from one supplier show advantages like easier engineering, standardisation, easier operation, less training, fewer spare parts, etc.

The WECS-9520 is well suited to support this integrated automation concept by providing redundant data bus lines that deliver all necessary information for propulsion control, alarm / monitoring system and man-machine interface. The MMI of the WECS-9520 can provide additional features when using such an integrated solution.

Ship automation from different suppliers - split solution

In case the propulsion control and alarm / monitoring systems are from different suppliers, the WECS-9520 also supports such a split solution by providing two separate redundant data bus lines, one each for the propulsion control and the alarm / monitoring system. In that case the MMI functions are also split within propulsion control and alarm / monitoring system.

DENIS-9520 describes the signal interface between the common rail system engine including its flex engine control system (WECS-9520) and the ship automation.

The DENIS-9520 specification does not include any hardware. It summarises all the data exchanged and defines the control functions required by the engine. The DENIS-9520 specification is presented in two sets of documents:

DENIS-9520 engine specification

This file contains the specification of the signal interface on the engine and is made available to engine builders and shipyards.

- It consists basically of the control diagram of the engine, the signal list including a minimum of functional requirements, and gives all information related to the electrical wiring on the engine
- It lists also the necessary alarm and display functions to be realised in the vessel's alarm and monitoring system.
- The DENIS-9520 engine specification covers the engine-built components for control, alarm and indication.
- With the replacement of the previous camshaft controlled function by the WECS-9520, the engine-built control components are reduced to a minimum.

remote control specification

This file contains the detailed functional specification of the remote control system.

- The intellectual property on this remote control specification remains with Winterthur Gas & Diesel Ltd. Therefore this file is licensed to remote control partners of Winterthur Gas & Diesel Ltd. only. The companies offer systems which are built exactly according to the engine designer's specifications, tested and approved by Winterthur Gas & Diesel Ltd.

16.2 Propulsion control system

The propulsion control system is divided in the following sub-systems:

- Remote control system
- Safety system
- Electronic speed control system
- Telegraph system

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

Winterthur Gas & Diesel Ltd. has an agreement with each of the following marine automation suppliers concerning the development, production, sales and servicing of remote control, electronic speed control and safety systems for their engines. All approved propulsion control systems listed in the following table comprise the same functionality specified by Winterthur Gas & Diesel Ltd..

Table 16.1 Suppliers of remote control systems and electronic speed control systems

Supplier		Remote control system	Electronic speed control system
Kongsberg Maritime			
Kongsberg Maritime AS P.O. Box 1009 N-3194 Horten/Norway	km.sales@kongsberg.com Tel. +47 81 57 37 00 www.km.kongsberg.com	AutoChief 600	DGS C20
NABTESCO Corporation			
NABTESCO corp., Marine Control Systems Company 1617-1, Fukuyoshi-dai 1-chome Nishi-ku Kobe, 651-22413/Japan	newbuilding@nabtesco.com Tel. +81 78 967 5361 www.nabtesco.com	M-800-V	MG-800 FLEX
SAM Electronics GmbH / Lyngsø Marine			
SAM Electronics GmbH Behringstrasse 120 D-22763 Hamburg/Germany	anc@sam-electronics.de Tel. +49 40 88 25 0000 www.sam-electronics.de	DMS2200 / EMS2200	EGS2200RTf
Lyngsø Marine AS 2, Lyngsø Allé DK-2970 Hørsholm/Denmark	info@lyngsoe.com Tel. +45 45 16 62 00 www.lyngsoe.com		

Modern remote control systems consist of electronic modules and operator panels for display and order input for engine control room and bridge. The different items normally communicate via serial bus connections. The engine signals described in the DENIS-9520 specification are usually connected via the terminal boxes on the engine to the electronic modules placed in the engine control room.

These electronic modules are in most cases built to be located 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.

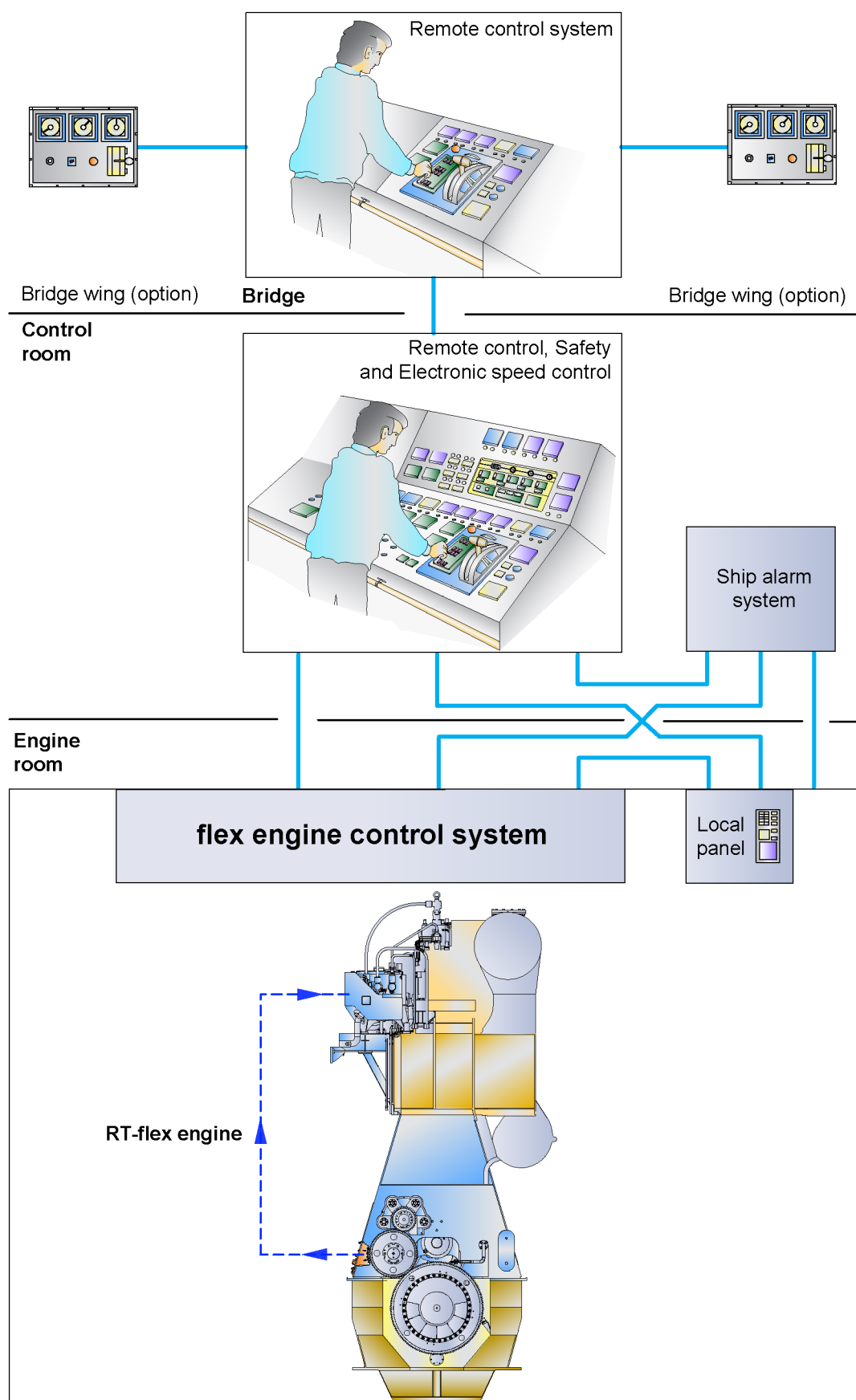


Fig 16.1 DENIS-9520 remote control system layout

16.3 Functions of the propulsion control system

Approved propulsion control systems comprise the following independent sub-systems:

Remote control system

Main functions:

- Start, stop, reversing
- Cylinder pre-/post-lubrication
- Automatic slow turning
- Auxiliary blower control
- Control transfer
- Speed setting
- Automatic speed program
- Load-dependent cylinder lubrication (Pulse Lubricating System)

Indications:

- The remote control system is delivered with control panels for local, control room 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 instruments for remote indication in the control room are specified in the DENIS-9520 standard as a minimum:
 - Starting air pressure
 - Engine speed
 - Revolution counter
 - Running hour counter
 - Load indicator
 - Turbocharger speed
 - Scavenge air pressure in air receiver
- The following instruments for remote indication on the bridge are specified in the DENIS-9520 standard as a minimum:
 - Starting air pressure
 - Engine speed
 - In addition to these indications, the remote control system applied to the common rail system engine includes displaying of the most important values of the flex engine control system (WECS-9520), like fuel pressure, servo oil pressure, etc.

Electronic speed control system

- Keeps engine speed at the setpoint given by the remote control system.
- Sends fuel command to the WECS-9520.
- Limits fuel amount in function of charge air and measured speed for proper engine protection.
- To avoid compatibility problems and increased engineering efforts, Winterthur Gas & Diesel Ltd. recommends to apply remote control systems and speed control systems of the same supplier.
- Traditionally the electronic speed control system was considered as part of the main engine and was therefore usually delivered together with the engine. With the introduction of WECS-9520 and DENIS-9520, the electronic speed control system is assigned to the propulsion control system and shall therefore be delivered together with the corresponding remote control system and other components of the propulsion control package by the party responsible for the complete propulsion control system, i.e. in most cases the shipyard.
- The details concerning system layout, mechanical dimensions of components as well as information regarding electrical connections have to be gathered from the technical documentation of the respective supplier.

Safety system (Main functions)

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

Telegraph system

- Order communication between different control locations

ECR manual control panel

- A manual control panel delivered together with the propulsion control system 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 equal to the control functions on the local control panel at the engine side.

Local manual control

- Local manual control of the engine is performed from a control panel located on the engine. This panel includes elements for manual order input and indication for safety system, telegraph system and WECS-9520.
- The local control box with the local manual control panel is included in the package delivered by approved remote control system suppliers.

Options

- Bridge wing control
- Order recorder

16.4 Recommended manoeuvring characteristics

Recommended values for the manoeuvring positions are given in figure 16.4.1.

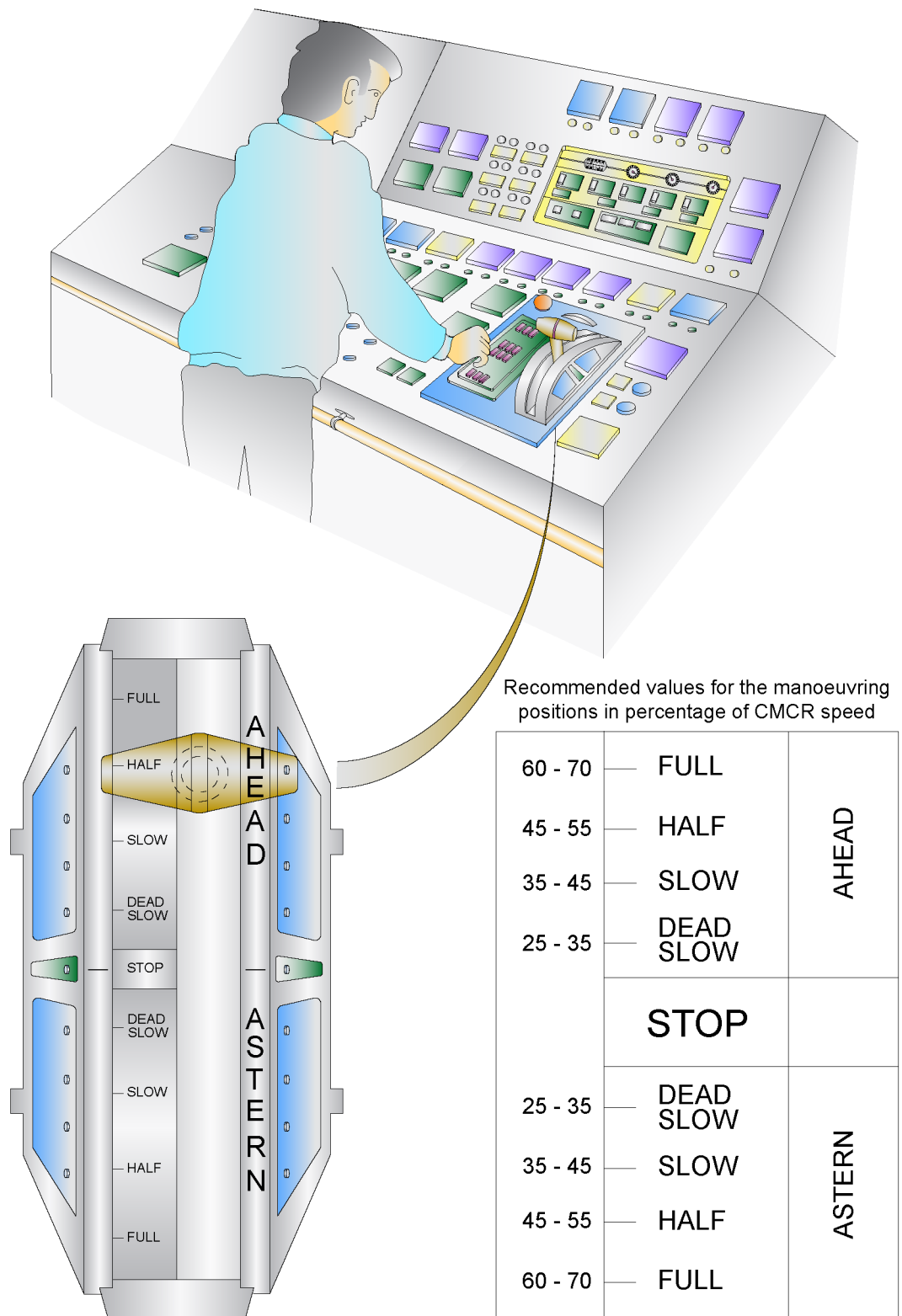


Fig 16.4.1 Recommended manoeuvring characteristics

16.5 Interface to alarm and monitoring systems

General layout - operator interface (OPI)

To monitor common rail system-specific circuits of the engine, sensors with hardwired connections are fitted. In addition to that, the flex engine control system (WECS-9520) provides alarm values and analogue indications via data bus connection to the ship's alarm and monitoring system as part of the operator interface of the common rail system engine. Connection from WECS-9520 to engine automation can be made in two ways.

Integrated solution

Propulsion control system and alarm / monitoring system from same supplier:

- This allows connecting both the propulsion control system and the alarm / monitoring system through one redundant bus line only (CANopen or Modbus, depending on automation maker) to the WECS-9520.
- With the integrated solution an extended presentation of relevant parameters is possible, as well as comfortable access to changeable user parameters, taking full profit of the graphical user interface functions available in the alarm / monitoring system.
- A further integration step is possible when using a DataChief C20 alarm / monitoring system of Kongsberg Maritime. In this case also all the conventional sensors and the additional flex sensors can be connected via data bus lines. The design allows mounting the data acquisition units straight on the engine in the same boxes used as terminal boxes for any other alarm / monitoring system.
- These boxes, which are part of the alarm / monitoring system usually provided by the shipyard, have to be delivered to the engine builder for mounting on the engine and connecting to the sensors.
- Commissioning and testing of the complete set of alarm signals already on the engine maker's testbed is thus facilitated, and the wiring at the shipyard is limited to a few power cables and bus communication.

Split solution

Propulsion control system and alarm / monitoring system from different suppliers:

- The propulsion control system is connected through one redundant bus line (CANopen or Modbus, depending on automation maker) to the WECS-9520.
- For the separate alarm / monitoring system an additional redundant Modbus connection is available.

Also the operator interface is split in this case:

- Changing of parameters accessible to the operator and display of parameters relevant for engine operation is included in the remote control system.
- The alarm / monitoring system must include the display of:
 - some flex system indications, like fuel pressure, servo oil pressure, etc.
 - the flex system alarms provided by the WECS-9520.
- Winterthur Gas & Diesel Ltd. provides modbus lists specifying the display values and alarm conditions as part of the DENIS-9520 engine specification.

Requirements for any alarm / monitoring system 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 from WECS-9520 to the standard alarm list (100-200 alarms depending on engine type and number of cylinders).

16.6 Alarm sensors and safety functions

The classification societies require different alarm and safety functions, depending on the class of the vessel and its degree of automation. These requirements are listed together with a set of sensors defined by Winterthur Gas & Diesel Ltd. in the tables below.

The time delays for the slow-down and shut-down functions given in tables below are maximum values. They may be reduced at any time according to operational requirements. When decreasing the values for the slow-down delay times, the delay times for the respective shut-down functions are to be adjusted accordingly. The delay values are not to be increased without written consent of Winterthur Gas & Diesel Ltd.

Included in the standard scope of supply are the minimum of safety sensors as required by Winterthur Gas & Diesel Ltd. for attended machinery space (AMS). If the option of unattended machinery space (UMS) was chosen, the respective sensors have to be added according to the requirements issued by Winterthur Gas & Diesel Ltd.

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

The extent of delivery of alarm and safety sensors has to cover the requirements of the respective classification society, Winterthur Gas & Diesel Ltd., the shipyard and the owner. The sensors delivered with the engine are basically connected to terminal boxes mounted on the engine. Signal processing has to be performed in a separate alarm and monitoring system usually provided by the shipyard.

Table 16.6.1 Alarm sensors and safety functions (part 1)

Alarm and safety functions					Values		
Medium / System	Phys. value	Location of sensor	Signal no.	Function	Level	Setting	Delay [s]
Cylinder cooling water							
	Pressure	Inlet Engine	PT1101A	ALM	L	4.0 bar	0
				SLD	L	3.8 bar	60
			PS1101S	SHD	L	3.5 bar	60
	Differential pressure	Inlet Engine	PT1102A	ALM	(L)	< 0.25 bar	0
				SLD	(L)	< 0.20 bar	60
	Temperature	Inlet Engine	TE1111A	ALM	L	70 °C	0
		Outlet each cylinder	TE1121-32A	ALM	H	95 °C	0
				SLD	H	97 °C	60
Scavenge air cooling water							
Freshwater single-stage	Pressure	Inlet Cooler	PT1361A	ALM	L	2.0 bar	0
	Temperature	Inlet Cooler	TE1371A	ALM	L	25 °C	0
		Outlet Cooler	TE1381-83A	ALM	H	80 °C	0
Main bearing oil							
	Pressure	Inlet Engine (Supply)	PT2001A	ALM	L	4.0 bar	0
				SLD	L	3.8 bar	60
			PS2002S	SHD	L	3.3 bar	10

Alarm and safety functions					Values		
Medium / System	Phys. value	Location of sensor	Signal no.	Function	Level	Setting	Delay [s]
Main bearing oil							
	Temperature	Inlet Engine (Supply)	TE2011A	ALM	H	50 °C	0
				SLD	H	55 °C	60
	Temperature	Outlet Bearings1-15 *7)	TE2101-15A	ALM	H	65 °C	0
				SLD	H	70 °C	60
External fuel pump lubrication (optional)							
	Pressure	Inlet Engine	PT2012A	ALM	L	4.0 bar	0
				SLD	L	3.8 bar	60
			PS2012S	SHD	L	3.3 bar	10
	Temperature	Inlet Engine	TE2012A	ALM	H	*9)	0
SLD				H	*9)	60	
Servo oil							
	Flow *1)	Inlet Pump	FS2061-63A	ALM	L	low flow	0
	Failure	Automatic filter	PS2053A	ALM	F	--	0
Leakage monitoring	Level	Supply pump	LS2055A	ALM	H	max.	0
Thrust bearing pads							
	Temperature	Fore side	TE4521A	ALM	H	80 °C	0
				SLD	H	85 °C	60
			TS4521S	SHD	H	90 °C	60
Crank pin bearing oil							
	Temperature	Outlet Bearing 1-12	TE2201-12A *7)	ALM	H	65 °C	0
				SLD	H	70 °C	60
Crosshead bearing oil							
	Temperature	Outlet Bearing 1-12 *11)	TE2301-12A *7)	ALM	H	65 °C	0
				SLD	H	70 °C	60
Crosshead bearing oil booster pump (optional)							
	Pressure	Inlet Engine	PT2021A *12)	ALM	L	10 bar	0
				SLD	L	9 bar	60
Oil mist concentration							
	Concentration	Crankcase	AE2401-12A	ALM	H	--	0
			AS2401-02A	ALM	H	--	0
			AS2401S	SLD	H	--	60
		In Supply unit	AE2421-22A	ALM	H	--	0
		Crankcase gear case	AE2415A	ALM	H	--	0
	Failure	Detection unit	XS2411A	ALM	F	--	30

Alarm and safety functions					Values		
Medium / System	Phys. value	Location of sensor	Signal no.	Function	Level	Setting	Delay [s]
Piston cooling oil							
	Temperature	Outlet each cylinder	TE2501-12A	ALM	H	80 °C	0
				SLD	H	85 °C	60
	Flow	Inlet each cylinder	FS2521-32S	SHD	L	no flow	15
Turbocharger oil							
ME bearing oil supply ABB A100-L/200-L	Pressure	Inlet each TC	PT2611-13A *2)	ALM	L	1.0 bar	5
				SLD	L	0.8 bar	60
			PS2611-13S	SHD	L	0.6 bar	5
	Temperature	Outlet TC	TE2601-03A	ALM	H	110 °C	0
				SLD	H	120 °C	60
	ME bearing oil supply MHI MET MB	Pressure	Inlet each TC	PT2611-13A *2)	ALM	L	0.7 bar
SLD					L	0.6 bar	60
PS2611-13S				SHD	L	0.4 bar	5
Temperature		Outlet TC	TE2601-03A	ALM	H	85 °C	0
				SLD	H	90 °C	60
Separate TC lub. oil supply ABB A100-L/200-L		Pressure	Inlet each TC	PT2611-13A *2)	ALM	L	1.3 bar
	SLD				L	1.1 bar	60
	PS2611-13S			SHD	L	0.9 bar	5
	Temperature	Outlet TC	TE2601-03A	ALM	H	130 °C	0
				SLD	H	140 °C	60
	Additional requirement when separate TC lub. oil supply	Temperature	Inlet TC ABB A100-L/200-L	TE2621A	ALM	H	85 °C
SLD					H	90 °C	60
Inlet TC MHI MET MB			TE2621A	ALM	H	60 °C	0
				SLD	H	65 °C	60
Turbocharger overspeed							
	Speed	TC casing	ST5201-03C	ALM	H	*6)	0
Torsional vibration damper oil							
	Pressure *10)	Inlet Damper	PT2711A	ALM	L	2.2 bar	0
Axial vibration damper oil							
	Pressure	Chamber rear	PT2721A	ALM	L	1.7 bar	60
		Chamber front	PT2722A	ALM	L	1.7 bar	60

Alarm and safety functions					Values		
Medium / System	Phys. value	Location of sensor	Signal no.	Function	Level	Setting	Delay [s]
Fuel oil							
	Viscosity	Before fuel pump unit	*8)	ALM	H	20 cSt	0
				ALM	L	10 cSt	0
	Temperature	Before fuel pump unit	TE3411A *3)	ALM	H	50-160 °C	0
				ALM	L	20-130 °C	0
	Pressure	Before supply unit	PT3421A	ALM	L	7 bar	0
Leakage	Level	Fuel pumps & pipes	LS3426A	ALM	H	max.	0
		Rail unit DE/FE	LS3444/45A	ALM	H	max.	0
		Fuel oil pipe	LS3446-47A	ALM	H	max.	0
Fuel heating	Failure	Pipe	XS3463A	ALM	F	--	0
Exhaust gas							
	Temperature	After each cylinder	TE3701-12A	ALM	H	515 °C	0
				ALM	D	± 50 °C	0
				SLD	H	530 °C	60
				SLD	D	± 70 °C	60
	Temperature	Before each turbocharger	TE3721-23A	ALM	H	515 °C	0
				SLD	H	530 °C	60
		After each turbocharger	TE3731-33A	ALM	H	480 °C	0
				SLD	H	500 °C	60
Scavenge air							
	Temperature	After each cooler	TE4031-33A *4)	ALM	L	25 °C	0
				ALM	H	60 °C	0
				SLD	H	70 °C	60
		Each piston underside	TE4081-92A *5)	ALM	H	80 °C	0
				SLD	H	120 °C	60
Condensation water	Level	Water separator	LS4071-73A	ALM	H	max.	0
				SLD	H	max.	60
		Before water separator	LS4075-77A	ALM	H	max.	0
				SLD	H	max.	60
Starting air							
	Pressure	Inlet Engine	PT4301-02C	ALM	L	12.0 bar	0

Alarm and safety functions					Values		
Medium / System	Phys. value	Location of sensor	Signal no.	Function	Level	Setting	Delay [s]
Air spring air							
	Pressure	Distributor	PT4341A	ALM	H	7.5 bar	0
				ALM	L	5.5 bar	0
				SLD	L	5.0 bar	60
			PS4341S	SHD	L	4.5 bar	0
Leakage oil	Level	Exh. valve air	LS4351-52A	ALM	H	max.	0
Control air							
Supply	Pressure	Inlet Engine	PT4401A	ALM	L	6.0 bar	0
Standby supply			PT4411A	ALM	L	5.5 bar	0
			PT4421A	ALM	L	5.0 bar	0
WECS-9520 control system							
	Pwr. failure	Power supply box	XS5056A	ALM	F	--	0
Engine Speed							
Overspeed	Speed	Crankshaft	ST5111-12S	SHD	H	110%	0
*1) ALM to be suppressed at low load (≤ 30 % of estimated power).							
*2) The indicated alarm and slow-down values are minimum settings allowed by the TC maker. To obtain an earlier warning, the ALM and SLD values may be increased up to 0.4 bar below the minimum effective pressure measured within the entire engine operation range. The final ALM/SLD setting shall be determined during commissioning / sea trial of the vessel.							
*3) ALM value depending on fuel viscosity.							
*4) For water separators made of plastic material the sensor must be placed right after the separator.							
*5) Acts as fire monitor.							
*6) ALM value depending on turbocharger type. Optional SLD for turbocharger available on customer's request.							
*7) Optional sensors.							
*8) Recommended: 13 to 17 cSt							
*9) Values depending on the oil type applied. SAE30: ALM L 50 °C, SLD L 55 °C. SAE40: ALM L 55 °C, SLD L 60 °C							
*10) The alarm value can be different. For the applicable setting value, refer to the specification of the damper manufacturer.							
*11) Suppressed at engine stop.							
*12) ALM & SLD to be suppressed below 40 % engine load.							

Table 16.6.2 Alarm sensors and safety functions (part 2)

Alarm and safety functions			min. WinGD requirements			Requirements of classification societies for UMS											
	Signal no.	Function	for AMS	add. to AMS for UMS	Flex sig-nals	IACS	ABS	BV	CCS	DNV	GL	KR	LR	MRS	NK	PRS	RINA
Cylinder cooling water																	
	PT1101A	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•
		SLD	•			•	•	•	•	•	•	•	•	•	•	•	•
	PS1101S	SHD	•								•						
	PT1102A	ALM															
		SHD															
	TE1111A	ALM		•							•						•
	TE1121-32A	ALM		•		•	•	•	■	•	•	•	•	•	•	•	•
		SLD		•		•	•	•	•	•	•	•	•	•	•	•	•
Scavenge air cooling water																	
	PT1361A	ALM		•													
	TE1371A	ALM		•													
	TE1381-83A	ALM		•													
Main bearing oil																	
	PT2001A	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•
		SLD	•			•			•		•		•		•	•	•
	PS2002S	SHD	•			•	•	•	•	•	•	•	•	•	•	•	•
	TE2011A	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•
		SLD		•			•	•	•		•			•		•	•
	TE2101-15A	ALM				•	A	A	A	A	A	A	A	A	A	A	A
		SLD										A					
Bering oil Fuel pump																	
	PT2012A	ALM															
		SLD															
	PS2012S	SHD															
	TE2012A	ALM															
		SLD															
Servo oil																	
	FS2061-63A	ALM	•														
	PS2053A	ALM	•														
	LS2055A	ALM	•														

Alarm and safety functions			min. WinGD requirements			Requirements of classification societies for UMS											
	Signal no.	Function	for AMS	add. to AMS for UMS	Flex signals	IACS	ABS	BV	CCS	DNV	GL	KR	LR	MRS	NK	PRS	RINA
Thrust bearing pads																	
	TE4521A	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•
		SLD		•		•	•	•	•	•	•				•	•	
	TS4521S	SHD				•	•		•							•	
Crank bearing oil																	
	TE2201-12A	ALM				•	A	A	A	A	A	A	A	A	A	A	A
		SLD							•		A						
Crosshead bearing oil																	
	TE2301-12A	ALM				•	A	A	A	A	A	A	A	A	A	A	A
		SLD							•		A						
Crosshead bearing oil booster pump (optional)																	
	PT2021A	ALM		•					•								
		SLD	•														
Oil mist concentration																	
	AE2401-12A	ALM	•														
	AS2401-02A	ALM	•			•	B	B	B	B	B	B	B	B	B	B	B
	AS2401S	SLD	•			•	B	B	B	B	B	B	B	B	B	B	B
	AE2421-22A	ALM	•														
	AE2415A	ALM	•														
	XS2411A	ALM	•								•		•				
Piston cooling oil																	
	TE2501-12A	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•
		SLD		•		•	•	•	•	•	•	•	•	•	•	•	•
	FS2521-32S	SHD	•						•		•						
Turbocharger oil																	
	ME bearing oil supply ABB																
	PT2611-13A	ALM		•					■		•					•	
		SLD		•													
	PS2611-13S	SHD	•														
	TE2601-03A	ALM		•					▲		•					•	
SLD		•															

Alarm and safety functions			min. WinGD requirements			Requirements of classification societies for UMS												
	Signal no.	Function	for AMS	add. to AMS for UMS	Flex sig-nals	IACS	ABS	BV	CCS	DNV	GL	KR	LR	MRS	NK	PRS	RINA	
	ME bearing oil supply MHI MET																	
	PT2611-13A	ALM		●					■		●					●		
		SLD		●														
	PS2611-13S	SHD	●															
	TE2601-03A	ALM		●					▲		●					●		
		SLD	●															
	Sep. lub. oil supply ABB A100-L/200-L																	
	PT2611-13A	ALM		●					■		●					●		
		SLD		●														
	PS2611-13S	SHD	●															
	TE2601-03A	ALM		●					▲		●					●		
		SLD	●															
	Additional requirement when separate TC lub. oil supply																	
	TE2621A	ALM		●												●		
		SLD		●											●		●	
	TE2621A	ALM		●												●		
		SLD		●											●		●	
	Turbocharger overspeed																	
	ST5201-03C	ALM								●								
Torsional vibration damper oil																		
	PT2711A	ALM		●														
Axial vibration damper oil																		
	PT2721A	ALM		●														
	PT2722A	ALM		●														
Fuel oil																		
		ALM		●		D	D	D	D	D	D	D	D	D	D	D	D	
		ALM		●		●	●	●	●	●	●	●	●	●	●	●	●	
	TE3411A	ALM		●							C							
		ALM		●		C	C	C	C	C	C	C	C	C	C	C	C	C
	PT3421A	ALM		●		●	●	●	●	●	●	●	●	●	●	●	●	●

Alarm and safety functions			min. WinGD requirements			Requirements of classification societies for UMS											
	Signal no.	Function	for AMS	add. to AMS for UMS	Flex signals	IACS	ABS	BV	CCS	DNV	GL	KR	LR	MRS	NK	PRS	RINA
Fuel oil																	
	Leakage																
	LS3426A	ALM			•		•	•	•	•	•	•	•	•	•	•	•
	LS3444/45A	ALM			•												
	LS346-47A	ALM			•												
	Fuel heating																
	XS3463A	ALM															
Exhaust gas																	
	TE3701-12A	ALM		•		•	•	•	•	•	•	E	•	•	•	•	•
		ALM		•		•	•	•	•	•	•	F	•	•	•	•	•
		SLD		•		•	•	•	•	•	•	•	•		•	•	G
		SLD		•												•	•
	TE3721-23A	ALM				•			•		•				•	•	•
		SLD															H
	TE3731-33A	ALM				•			•		•		•		•	•	
		SLD									•						
Scavenge air																	
	TE4031-33A	ALM		•			I				I	•		○	•	•	I
		ALM		•			•		I		•	•		○	•	•	•
		SLD							I								
	TE4081-92A	ALM		•		•	•	•	K	•	•	•	•	•	•	•	•
		SLD		•		•	•	•	K	•	•	•	•		•	•	•
	Condensation water																
	LS4071-73A	ALM		•		•	K	•	•		K		•		•		K
		SLD		•													
	LS4075-77A	ALM		•		•	K	•	•		K				•		K
		SLD		•													
Starting air																	
	PT4301C	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•

Alarm and safety functions			min. WinGD requirements			Requirements of classification societies for UMS											
	Signal no.	Function	for AMS	add. to AMS for UMS	Flex sig-nals	IACS	ABS	BV	CCS	DNV	GL	KR	LR	MRS	NK	PRS	RINA
Air spring air																	
	PT4341A	ALM		•		•											
		ALM	•			•			•								
		SLD	•			•											
	PS4341S	SHD	•			•											
Leakage oil																	
	LS4351-52A	ALM	•			•											
Control air																	
Supply																	
	PT4401A	ALM		•		•	•	•	•	•	•	•	•	•	•	•	•
Standby supply																	
	PT4411A	ALM		•					•								
	PT4421A	ALM		•					•		•						
WECS-9520 control system																	
	XS5056A	ALM			•												
Engine																	
	ST5111-12S	SHD	•			•	•	•	•	•	•	•	•	•	•	•	•

16.6.1 Classification societies

IACS	International Association of Classification Societies		
ABS	American Bureau of Shipping	IRS	Indian Register of Shipping
BV	Bureau Veritas	KR	Korean Register of Shipping
CCS	Chinese Classification Society	LR	Lloyd's Register of Shipping
CRS	Croatian Register of Shipping	NK	Nippon Kaiji Kyokai
DNV	Det Norske Veritas	PRS	Polski Rejestr Statkow
DNV-GL	Det Norske Veritas - Germanischer Lloyd *1)	RINA	Registro Italiano Navale
GL	Germanischer Lloyd	RS	Russian Maritime Register of Shipping

*1) The rule books of DNV and GL are still valid until further notice.

16.6.2 Requirements of classification societies

Table 16.6.2.1 Requirements of classification societies

Requirements of classification societies for UMS:				Special requirement for AMS:	
•	Requirement	A or B	are required altern- atively	▲	Requirement for AMS only
○	Recommendation	C or D		■	Additional requirement to UMS for AMS
AMS	Attended machinery space	E or F			
UMS	Unattended machinery space	G or H			
		I or K			

16.6.3 Functions and level

Table 16.6.3.1 Functions and level

Function:		Level:	
ALM	Alarm	H	High
SLD	Slow down	L	Low
SHD	Shut down	D	Deviation
		F	Failure

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17. General Installation Aspects

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

17.1 Engine dimensions and masses

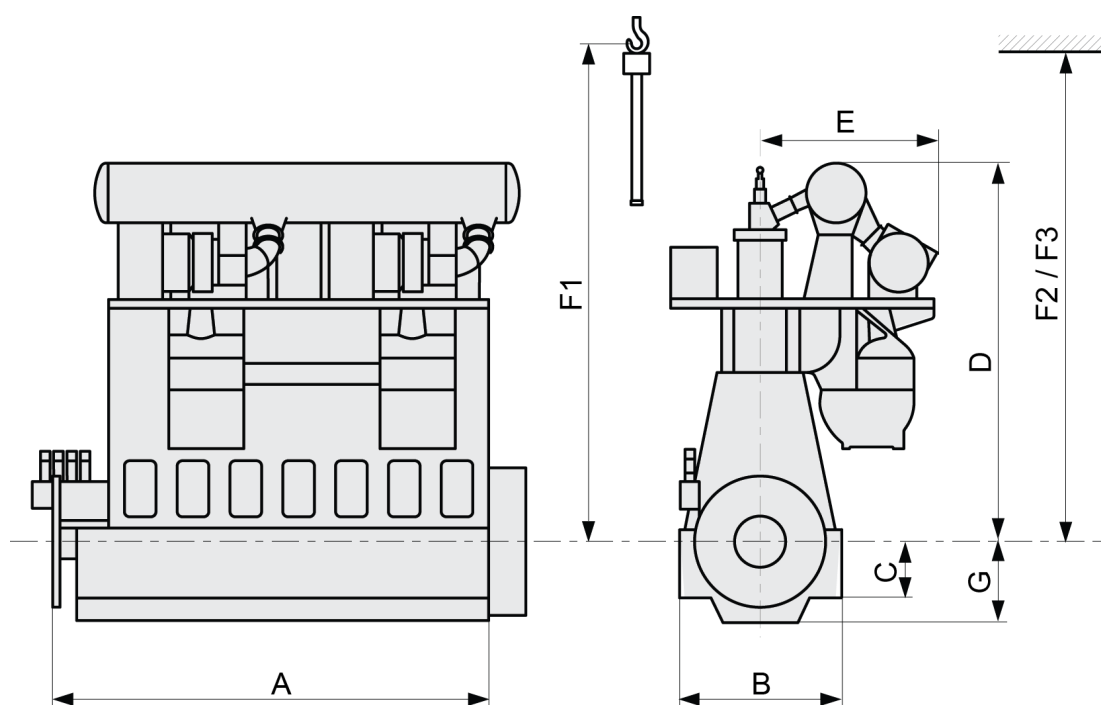


Fig 17.1.1 Engine dimensions

No. cyl.	Dimensions in mm with a tolerance of approx. ±10 mm									Net engine mass *1)
	A	B	C	D	E	F1	F2	F3	G	[tonnes]
6	11570	5550	1900	12950	6050	15420	15450	14240	2930	1120
7	13160									1260
8	14750									1380
9	17780									1630
10	19370									1790
11	21030									1960
12	22700									2140
Min. crane capacity: 12500 kg										

No. cyl.	Dimensions in mm with a tolerance of approx. ± 10 mm									Net engine mass *1)
	A	B	C	D	E	F1	F2	F3	G	[tonnes]
*1) without oil/water. Net engine mass estimated according to nominal dimensions given in drawings, including turbocharger and SAC, piping and platforms E: dimension depends on turbocharger type F1: min. height for vertical removal of piston F2: min. height for vertical removal with double-jib crane F3: min. height for tilted removal with double-jib crane - For the actual dimensions: See latest version of the drawings (DG 0812), provided on the WinGD corporate webpage.										

17.1.1 Dimensions and masses of main components

Table 17.1.1.1 Dimensions and masses of main components (6-9 cyl.)

Main parts		No. cylinders		6	7	8	9
Bedplate including bearing girders		Length	[mm]				
		Mass	[kg]				
Crankshaft, one-piece execution		Length	[mm]				
		Mass	[kg]				
Flywheel		Diameter	[mm]				
		Mass	[kg]				
Monoblock column		Length	[mm]				
		Mass	[kg]				
Tie rod, without assembly parts		Length	[mm]				
		Mass	[kg]				
Cylinder jacket (A), complete with studs (B)		Height	[mm]				
		Mass	[kg]				
Cylinder liner, without any assembly parts		Length	[mm]				
		Mass	[kg]				
Cylinder cover with starting valve		Height	[mm]				
		Mass	[kg]				
Connecting rod, without top-/bottom-end bearing shells/covers		Length	[mm]				
		Mass	[kg]				
Crosshead	Pin	Length	[mm]				
		Mass	[kg]				
	Guide shoe, middle part	Width	[mm]				
		Mass	[kg]				
Piston, complete with rod, without rings		Length	[mm]				
		Mass	[kg]				
Gland box		Diameter	[mm]				
		Mass	[kg]				
Scavenge air receiver, complete, without assembly parts		Length	[mm]				
		Mass	[kg]				
Exhaust valve, complete		Height	[mm]				
		Mass	[kg]				
Supply unit		Length	[mm]				
		Mass	[kg]				

Table 17.1.1.2 Dimensions and masses of main components (6-9 cyl.)

Main parts		No. cylinders		10	11	12
Bedplate including bearing girders		Length	[mm]			
		Mass	[kg]			
Crankshaft, one-piece execution		Length	[mm]			
		Mass	[kg]			
Flywheel		Diameter	[mm]			
		Mass	[kg]			
Monoblock column		Length	[mm]			
		Mass	[kg]			
Tie rod, without assembly parts		Length	[mm]			
		Mass	[kg]			
Cylinder jacket (A), complete with studs (B)		Height	[mm]			
		Mass	[kg]			
Cylinder liner, without any assembly parts		Length	[mm]			
		Mass	[kg]			
Cylinder cover with starting valve		Height	[mm]			
		Mass	[kg]			
Connecting rod, without top-/bottom-end bearing shells/covers		Length	[mm]			
		Mass	[kg]			
Crosshead	Pin	Length	[mm]			
		Mass	[kg]			
	Guide shoe, middle part	Width	[mm]			
		Mass	[kg]			
Piston, complete with rod, without rings		Length	[mm]			
		Mass	[kg]			
Gland box		Diameter	[mm]			
		Mass	[kg]			
Scavenge air receiver, complete, without assembly parts		Length	[mm]			
		Mass	[kg]			
Exhaust valve, complete		Height	[mm]			
		Mass	[kg]			
Supply unit		Length	[mm]			
		Mass	[kg]			

17.1.2 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 fig. 17.1.2.1):

Expansion	Distance from
-----------------	---------------

Transverse expansion (X):	crankshaft centerline 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
-----------------------------------	---

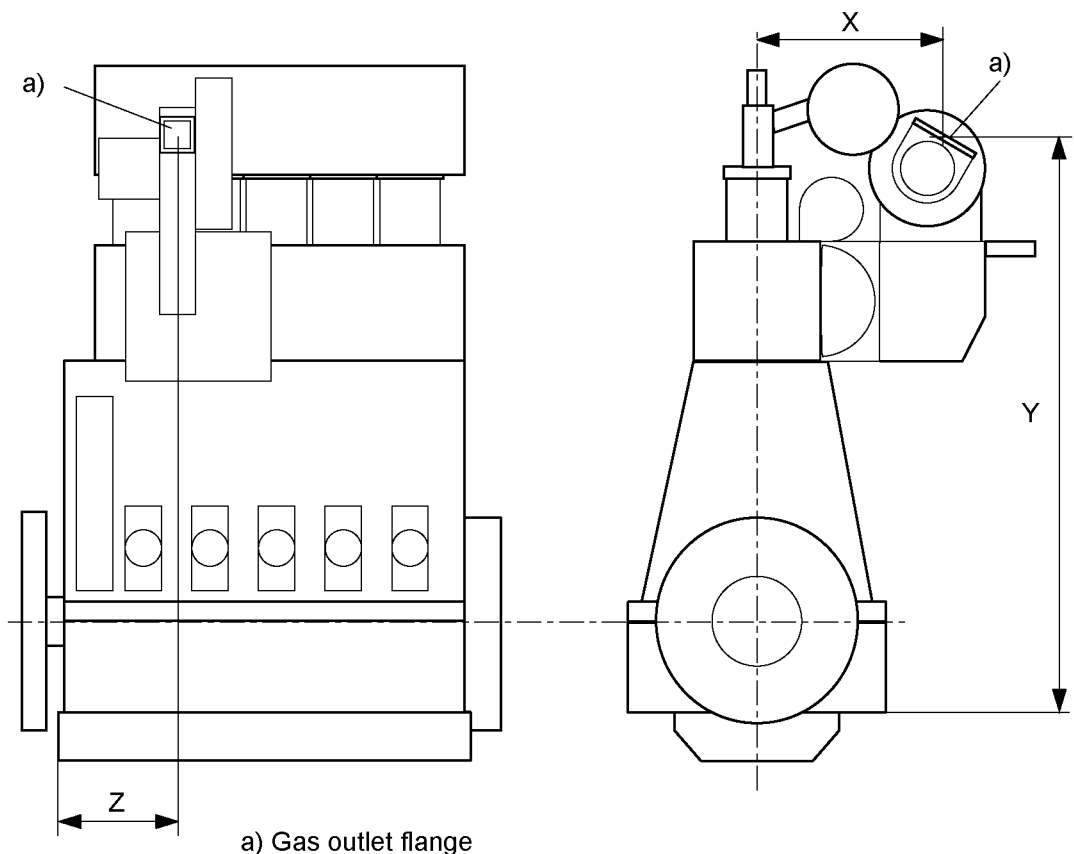


Fig 17.1.2.1 Thermal expansion, dim. X, Y, Z

Instruction for calculating thermal expansion:

Formula	$\Delta x (\Delta y, \Delta z) = X (Y, Z) \times \alpha \times \Delta T$
X, Y, Z	= distance as per relevant pipe connection plan and outline drawing
0.0000115	= coefficient of thermal expansion (α)
ΔT	= difference between service temperature and ambient temperature [°C]
$\Delta x, \Delta y, \Delta z$	= thermal expansion

17.1.3 Content of fluids in the engine

Table 17.1.3.1 Fluid quantities in the engine

No. of cyl.	Lubricating oil	Cylinder cooling water	Freshwater in scavenge air cooler(s)	Total of water and oil in engine
	[kg]	[kg]	[kg]	[kg]
6	5850	4550	1700	12100
7	6500	5300	2150	13950
8	6500	6100	2050 *1) / 2350 *2)	14650 *1) / 14950 *2)
9	7150	6850	2450 *1) / 2550 *2)	16450 *1) / 16550 *2)
10	8100	7600	2400 *1) / 3100 *2)	18100 *1) / 18800 *2)
11	8800	8350	3000	20150
12	9500	9150	3150	21800

*1) Values for executions with 2 scavenge air coolers

*2) Values for executions with 3 scavenge air coolers

17.1.4 Crane requirements

- An overhead travelling crane is to be provided for normal engine maintenance. (Crane capacity see table 17.1.1.)
- The crane is to conform to the requirements of the classification society.

As a general guidance Winterthur Gas & Diesel Ltd. recommends a two-speed hoist with pendent control, allowing to select high or low speed, i.e. high speed 6.0 m/minute, low speed 0.6-1.5 m/minute.

17.1.5 Piston and cylinder liner dismantling heights

Dimensions F1, F2, F3 in Fig. 17.1.1 and the corresponding table are for guidance only and may vary depending on crane dimension, handling tools and dismantling tolerances. However, please contact Winterthur Gas & Diesel Ltd. or any of their representatives if these values cannot be maintained or if more detailed information is required.

For details see also drawing 'Dismantling dimensions' (DG 0812), provided on the WinGD corporate webpage. Tab 'Drawings' is accessible under the following link:

Link is under preparation. Please ask WinGD Switzerland directly.

17.2 Outline views

The latest version of the drawings (DG 0812) is provided on the WinGD corporate webpage. Tab 'Drawings' is accessible under the following link:

Link is under preparation. Please ask WinGD Switzerland directly.

17.3 Platform arrangement

The latest version of the drawings (DG 76XX) is provided on the WinGD corporate webpage. Tab 'Drawings' is accessible under the following link:

Link is under preparation. Please ask WinGD Switzerland directly.

17.4 Engine 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.

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.

Before any engine seating work can be performed, make sure the engine is aligned with the intermediate propeller shaft.

17.4.1 Installation drawings and concept guidance

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

https://www.wingd.com/media/1904/mids_wingd-x92_engine-seating_and_foundation.pdf

17.5 Engine coupling

17.5.1 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.

17.5.2 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.

17.5.3 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 to an angle of ° (or see Tightening Instructions, design group 0352), 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.

17.5.4 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:

Link is under preparation. Please ask WinGD Switzerland directly.

17.6 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.

17.6.1 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, then 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. 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, e.g. 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.

17.6.2 Earthing device

Figure 17.6.2.1 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).

The electric cables are connected as shown in figure 17.6.2.2 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. Different combinations of conducting material are available for the construction of the slip-rings. However, alloys with a high silver content are found to be efficient and hard wearing.

Position of the earthing device on the shaft

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

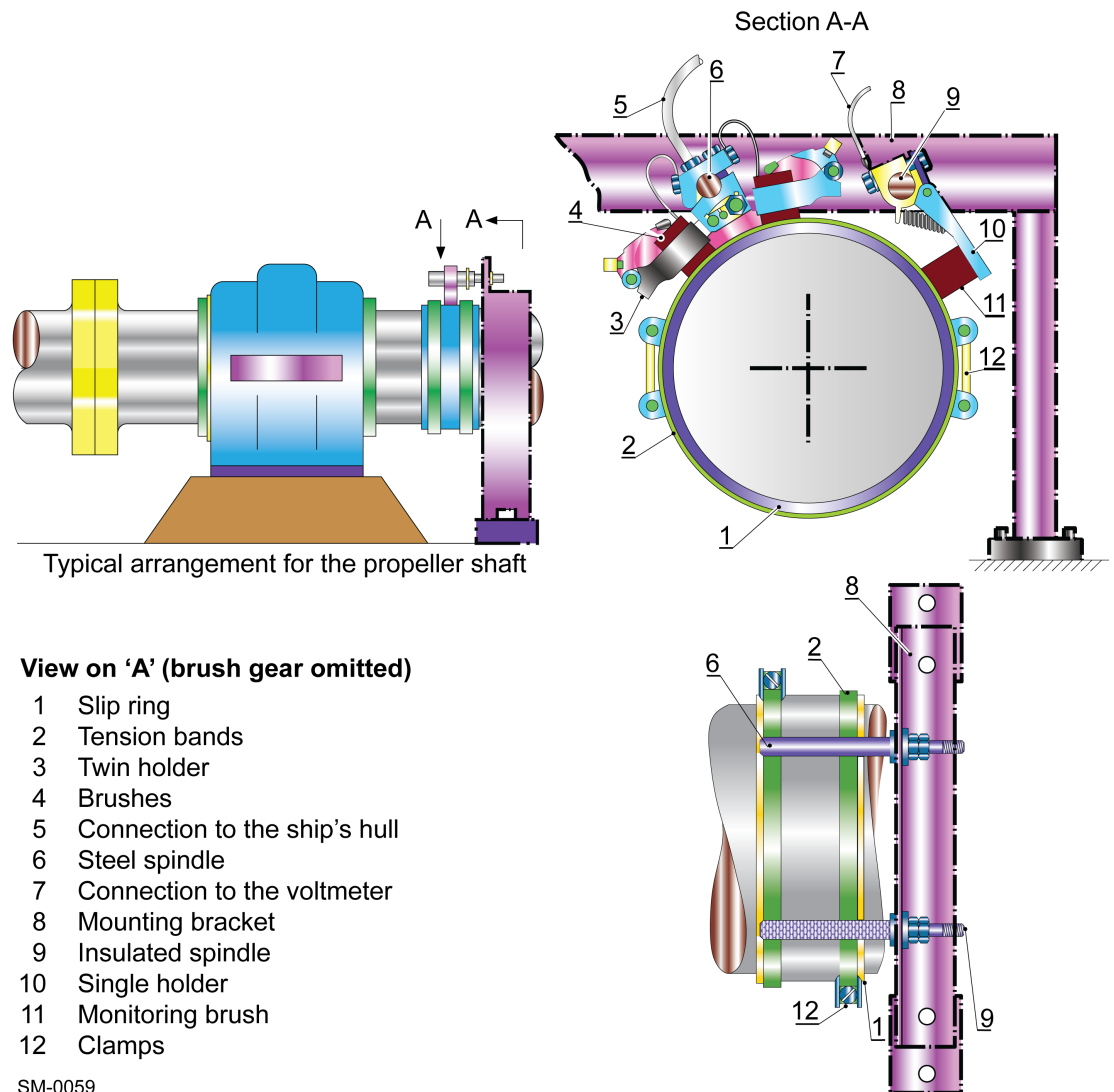


Fig 17.6.2.1 Shaft earthing arrangement

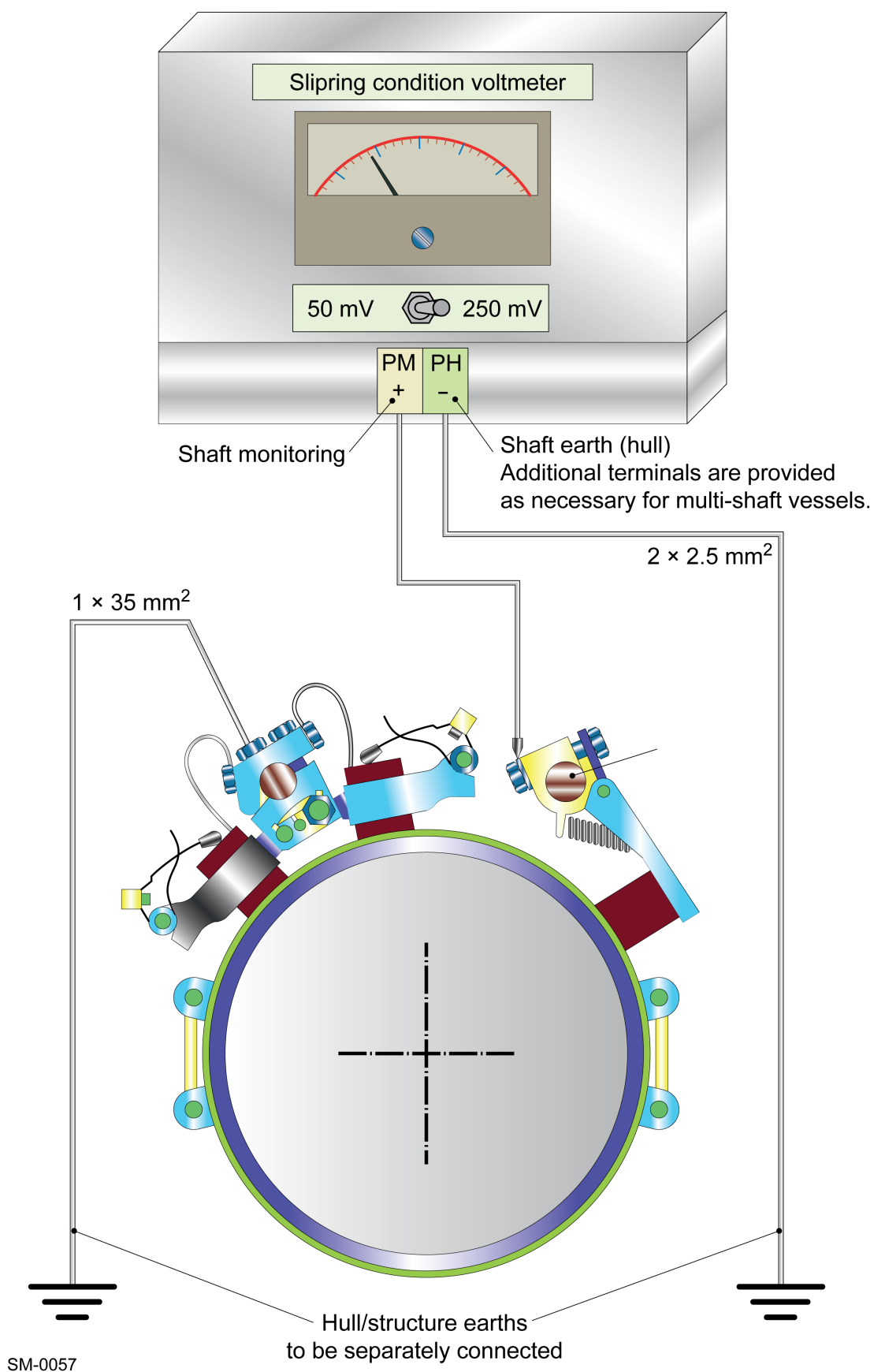


Fig 17.6.2.2 Shaft earthing with condition monitoring facility

17.7 Engine stays

Ship vibrations and engine rocking caused by the engine behaviour are reduced by fitting lateral and longitudinal stays.

17.7.1 Stays arrangement

Lateral stays: WinGD recommends the installation of hydraulic type stays (single-acting or double-acting).

Longitudinal stays: WinGD recommends the installation of friction type stays.

17.7.2 Installation drawings and concept guidance

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

https://www.wingd.com/media/1905/mids_wingd-x92_engine-stays.pdf

17.8 Fire protection

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

17.8.1 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. lubricating oil)

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.

Recommended quantities of fire extinguishing medium

Piston underside at bottom dead centre		Bottle		Number of cylinders							
				6	7	8	9	10	11	12	
Volume [m³/cyl.]	Mass [kg/cyl.]	Size [kg]	Extinguishing medium	Number of fire extinguishing bottles							
11	40	45	Carbon dioxide	6	7	8	8	9	10	11	

18. Exhaust gas emissions

The International Maritime Organisation (IMO) is a specialized agency of the United Nations (UN), dealing with technical aspects of shipping. For more information see <http://www.imo.org>.

18.1 Establishment of emission limits for ships

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**.

The **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 the allowed 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 IMO in October 2008 and entered into force in July 2010. This amended Annex IV includes provisions for the further development of the emissions regulations until 2020.

18.2 Regulation regarding NO_x emissions

Regulation 13 of Annex IV specifies a limit for the nitrogen oxide (NO_x) emissions of engines installed on ships, which has a direct implication on propulsion engine design. 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.

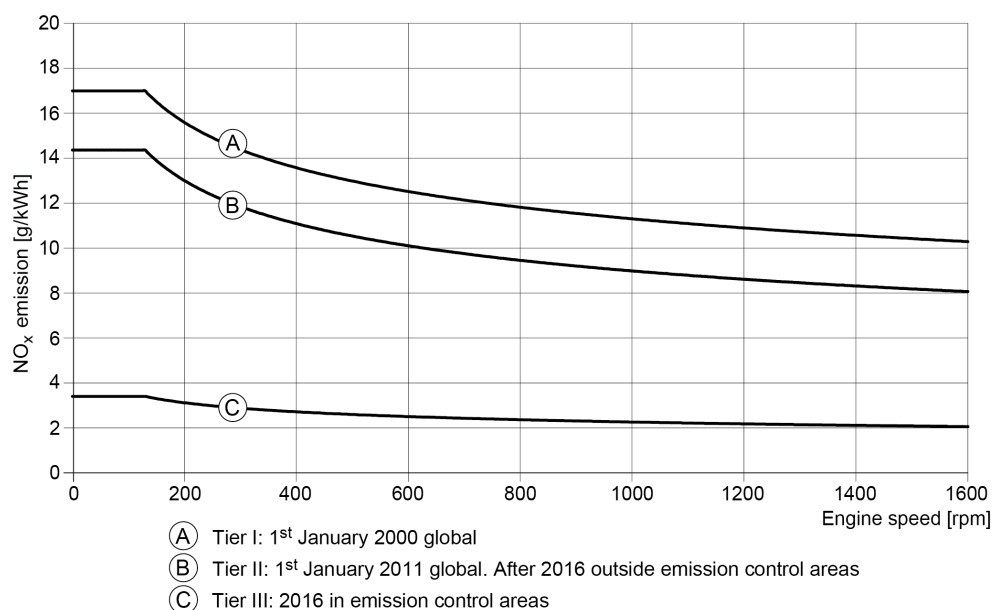


Fig 18.2.1 Speed dependent maximum allowable average of NO_x emissions

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.

18.3 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 Organisation.

18.3.1 SCR solutions

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.

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, but more flexible in installation position, as any after-turbo charger position is acceptable.

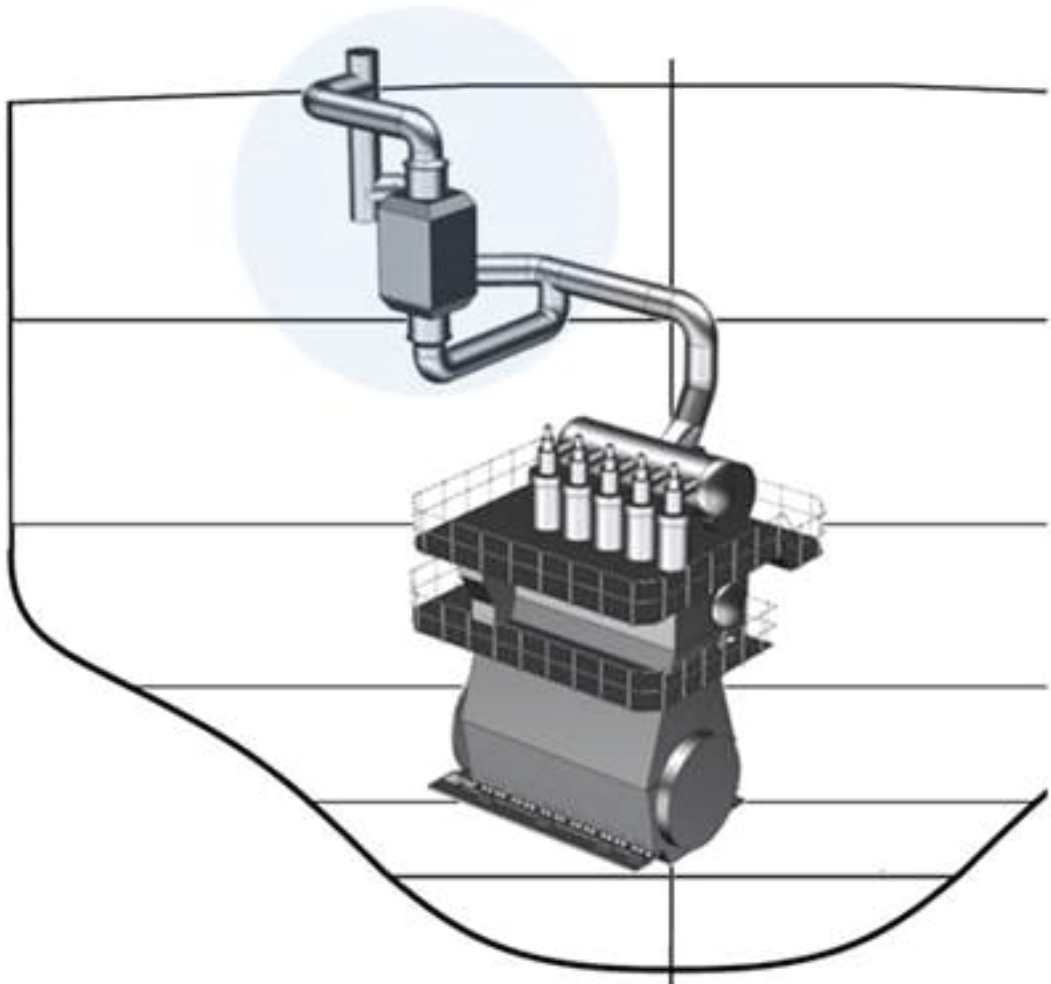


Fig 18.3.1.1 LP SCR components (principle layout)

High-pressure SCR

Available only for engines with one and two turbochargers.

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. Further, WinGD allows high-pressure SCR suppliers to interface third-party branded products to the engine, provided that interface specifications are met.

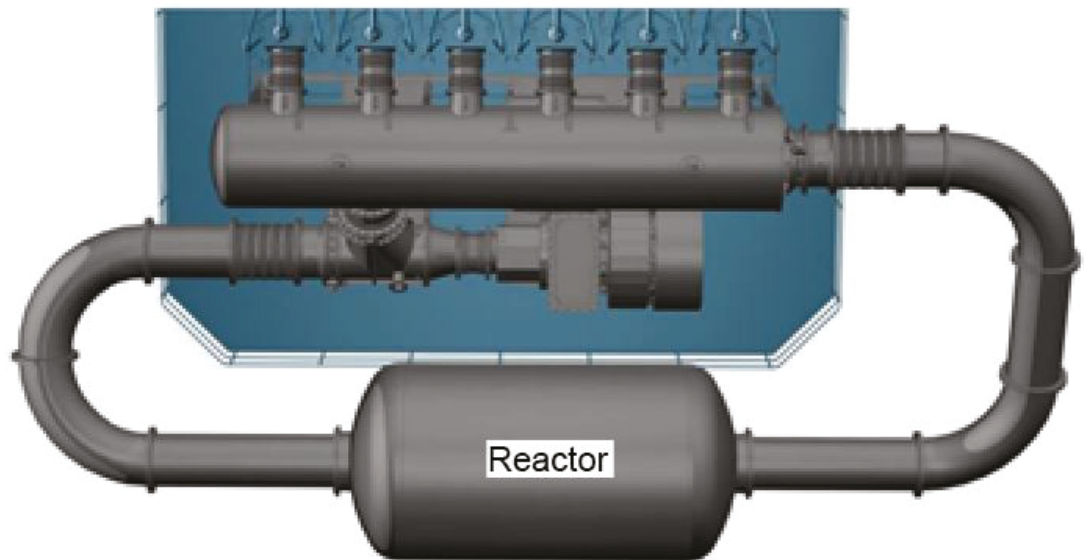


Fig 18.3.1.2 HP SCR components (principle layout)

Documents and specifications

The latest version of the documents, relevant for the SCR system (DG 8159 and 9726), is provided on the WinGD corporate webpage. Tab 'Drawings' is accessible under the following link:

Link is under preparation. Please ask WinGD Switzerland directly.

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19. Engine noise

It is necessary to protect the ship's crew/passengers from the effects of machinery space noise. Therefore the scavenge air ducts and the exhaust duct system (both expansion joints of turbocharger gas outlet and inlet) should be equipped with the standard insulation and the turbocharger with the standard intake silencer.

19.1 Engine surface sound pressure level

Figure 19.1.1 shows the average air borne noise level, measured at 1 m distance and at nominal MCR. Near the turbocharger (air intake), the maximum measured noise level will normally be about 3-5 dB(A) higher than the average noise level of the engine.

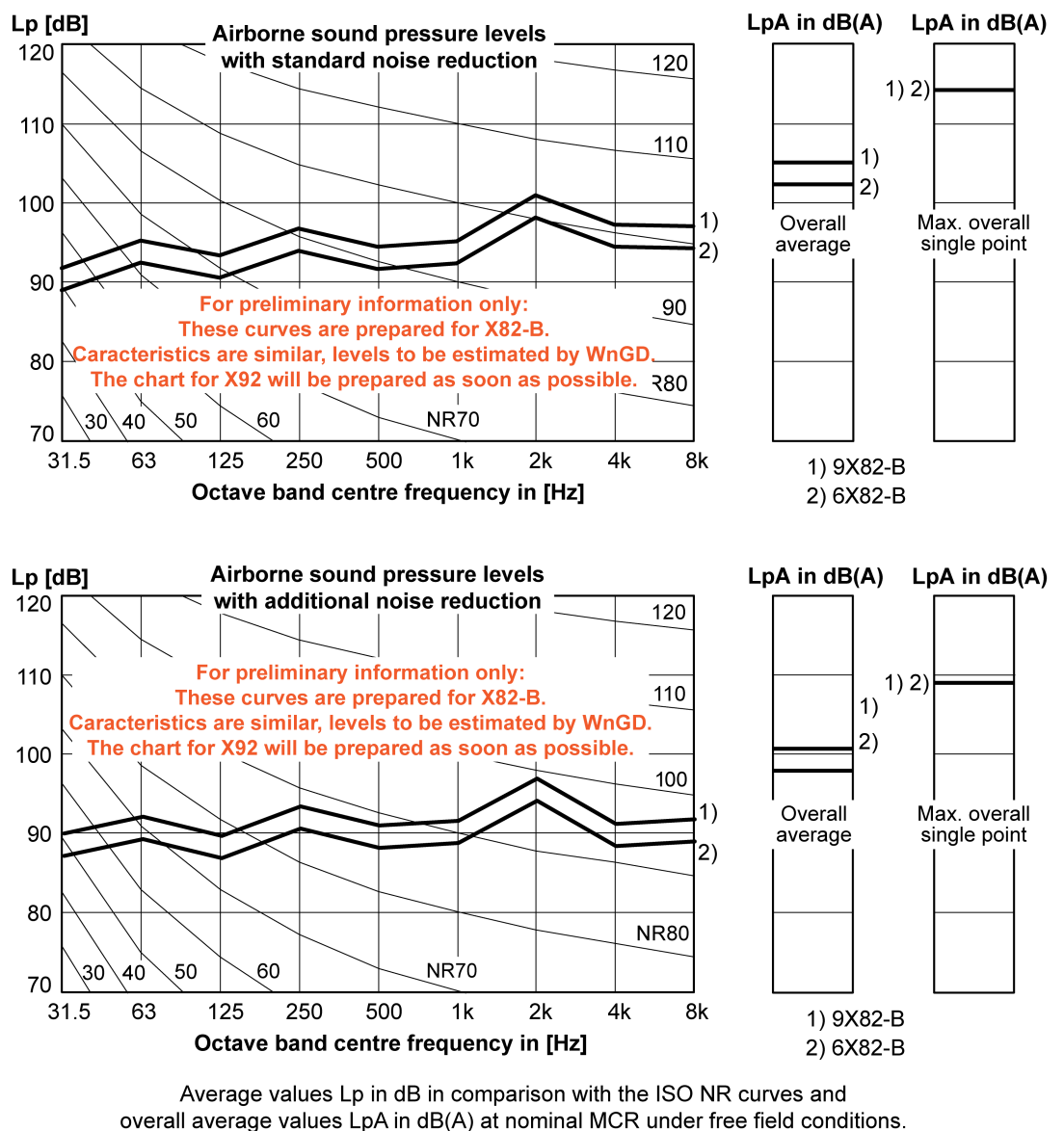


Fig 19.1.1 Engine sound pressure level at 1 m distance

19.2 Engine exhaust sound pressure level at funnel top

The sound pressure level from the engine exhaust gas system without boiler and silencer (given in figure 19.2.1) is related to:

- a distance of one metre from the edge of the exhaust gas pipe opening (uptake)
- an angle of 30° to the gas flow direction
- nominal MCR

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

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 necessary and placed after the exhaust gas boiler.

For installations with exhaust gas bypass a silencer in the main engine exhaust line may be considered.

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

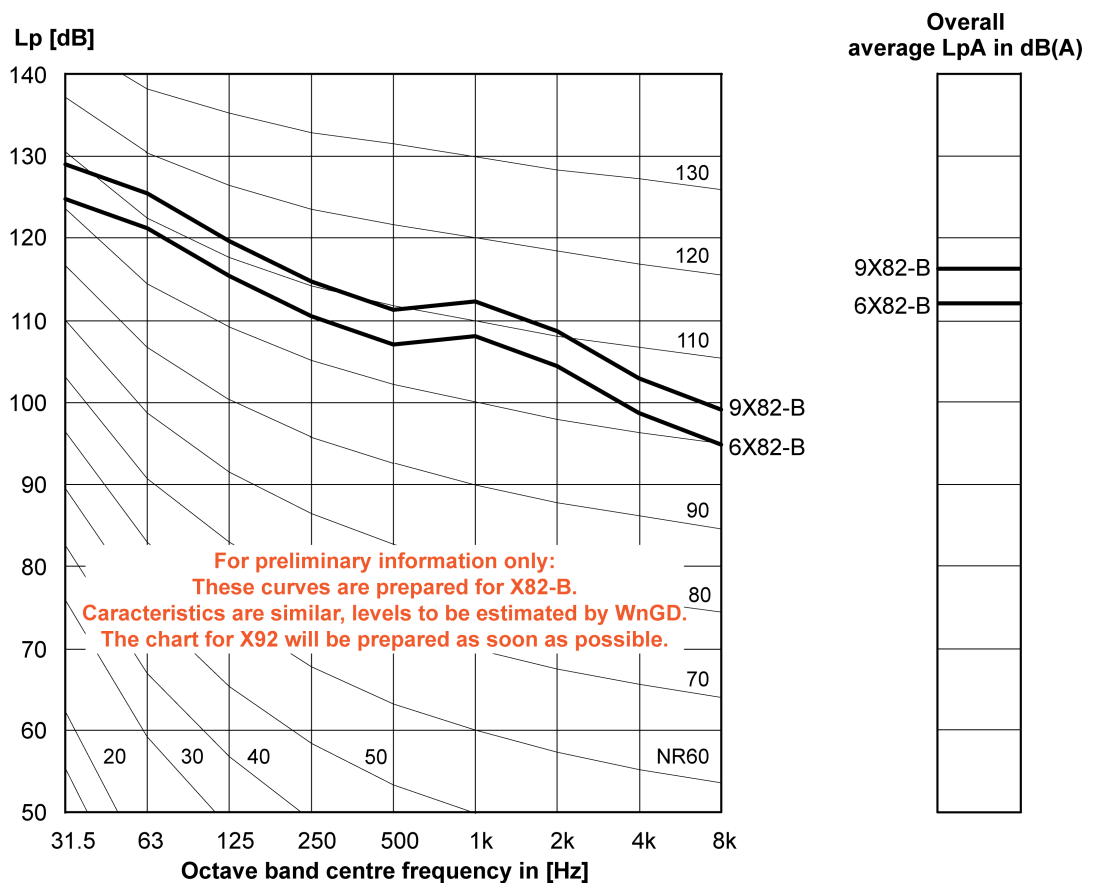


Fig 19.2.1 Sound pressure level at funnel top of engine exhaust gas system

19.3 Engine 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.

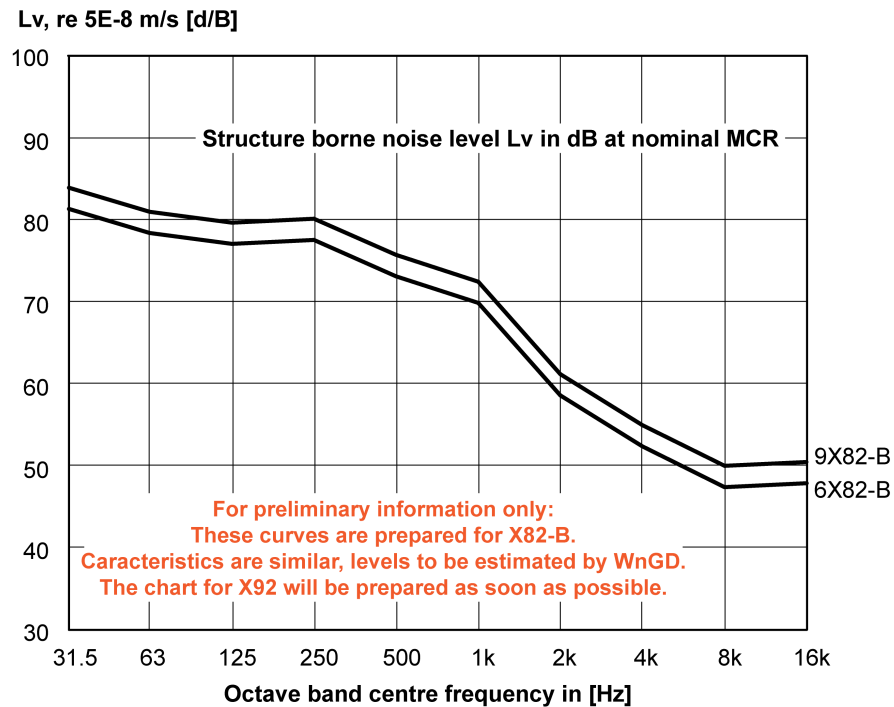


Fig 19.3.1 Structure borne noise level at engine feet vertical

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20. Engine Dispatch and Installation

Engines are transported as complete or part assemblies. They have to be protected from corrosion by rust preventing oils or vapour-phase corrosion inhibitor papers (VCI), and from damage by using wooden crates lined with jute reinforced bituminous paper.

20.1 Engine dismantling

Engines transported as part assemblies are 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, and protected from damage by careful crating as well as corrosion protected by rust preventing oils or paper. 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. Use a paint brush to apply high-viscosity rust preventing oil to the piston and connecting rods, crosshead guides, gear wheels, camshaft and rollers. Air powered spray guns are to be used only if the air is absolutely free of water. Crankshaft and crosshead pins are to be protected with an anti-corrosive coating of Tectyl 506 or a similar product.

20.2 Removing rust preventing oils

Rust preventing oils applied to the internal parts of an assembled engine do not contain thickening agents of wax or bitumen. These oils have properties similar to the engine lubricating oils, will wash off easily and mix without causing harm to the engine or its systems.

Rust preventing oils of the wax-type applied to exposed surfaces of the engine components do contain thickening agents of wax or bitumen forming an anti-corrosion coating when applied, which has to be washed off using a proprietary 'cold cleaner'. It is not sufficient to use gas oil, kerosene or white spirit on their own as solvents; they are to be mixed with 2 to 3 parts of a 'cold cleaner', such as 'Magnusol', 'Agitol' or 'Emultan'.

20.3 Engine installation

The 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 the engine mounting is to be checked by qualified personnel and measurements cross-checked with the design figures. The responsible parties (e.g. shipyard) are to advise the representative of the engine builder or Winterthur Gas & Diesel Ltd. directly in the event of any discrepancies. Engines may be installed as complete units or assembled from sub-assemblies in the vessel, which may be afloat, in dry dock, or on the slipway. The engine alignment can be done with either jacking screws or wedges.

20.3.1 Installation and 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 drilled holes to allow the passing of the holding-down bolts and tapped holes for the jacking screws for engine alignment.

Proceed with the preliminary alignment of the bedplate using one of the methods mentioned in section 20.3 to 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, the engine coupling flange is to be set 0.4 to 0.6 mm higher than the calculated figures. Place bearing caps in position, install turning gear and check that crankshaft deflections are as recorded on the "Engine Assembly Records". To check the bedplate level in longitudinal and diagonal direction a taut-wire measuring device will be provided by the engine builder. Compare the readings with those recorded at works. Optical devices or lasers may also be used.

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

Continue engine assembly by mounting the columns, cylinder blocks, running gear and scavenge air receiver, but 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. Careful adjustments of the wedges or of the jacking screws are necessary 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.

20.3.2 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 the assembly work in accordance with the demands of a representative of the engine builder and the classification society. The engine mounting is to be carried out systematically and measurement readings taken, recorded on appropriate log sheets and compared for correctness with the data of the "Engine Assembly Records" completed after test run in the works of the manufacturer.

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 engine is to be lowered onto blocks placed between the chocking points. The alignment tools are to be clean and ready for use. Set the blocks 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 deviations have to be rectified immediately.

20.3.3 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 bedplate, main and thrust bearings, crankshaft, turning gear, and flywheel. The placing on blocks and alignment to shafting is analogue to that described in section 20.3.1.

20.3.4 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:

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

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21. Engine and Shaft alignment

21.1 Procedure

21.1.1 Drawings

The latest version of the documents, relevant for the engine alignment (DG 9709), is provided on the WinGD corporate webpage under the following link:

https://www.wingd.com/media/1827/instruction_and_limits_wingd-2s_engine-alignment.pdf

21.2 Tools

21.2.1 Drawings

The latest version of the drawing, relevant for the tool engine alignment (DG 9710-01), is provided on the WinGD corporate webpage under the following link:

https://www.wingd.com/media/1903/mids_wingd-x92_tool-engine-alignment.pdf

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22. Appendix

22.1 Links to complete drawing packages

The latest version of drawing packages, relevant for this MIM, is provided on the WinGD corporate webpage under the following links:

Marine Installation Set (MIDS), DG 97XX

https://www.wingd.com/media/1911/mids_wingd-x92_dg97xx_complete.zip

Shipyard Installation Instructions and System Concept Guidance

[https://www.wingd.com/media/1856/
concept-guidance_and_instructions_wingd-2s_complete.zip](https://www.wingd.com/media/1856/concept-guidance_and_instructions_wingd-2s_complete.zip)

22.2 Abbreviations

ABB	ASEA Brown Boveri	MAPEX	Monitoring and maintenance performance enhancement with expert knowledge
ALM	Alarm	MCR	Maximum continuous rating (R1)
AMS	Attended machinery space	MDO	Marine diesel oil
BFO	Bunker fuel oil	mep	Mean effective pressure
BN	Base Number	METxxMB	Turbocharger (Mitsubishi manufacture)
BSEF	Brake specific exhaust gas flow	MHI	Mitsubishi Heavy Industries
BSFC	Brake specific fuel consumption	MIM	Marine installation manual
CCAI	Calculated Carbon Aromaticity Index	MMI	Man-machine interface
CCR	Conradson carbon	N, n	Rotational speed
CCW	Cylinder cooling water	NAS	National Aerospace Standard
CMCR	Contract maximum continuous rating (Rx)	NCR	Nominal continuous rating
CPP	Controllable pitch propeller	NOR	Nominal operation rating
CSR	Continuous service rating (also designated NOR and NCR)	OM	Operational margin
cSt	centi-Stoke (kinematic viscosity)	OPI	Operator interface
DAH	Differential pressure alarm, high	P	Power
DENIS-9520	WinGD Engine Control System	PAL	Pressure alarm, low
EM	Engine margin	PI	Pressure indicator
EMA	Engine Management & Automation	PLS	Pulse Lubricating System (cylinder liner)
FCM	Flex control module	ppm	Parts per million
FPP	Fixed pitch propeller	PRU	Power related unbalance
FQS	Fuel quality setting	PTO	Power take-off
FW	Freshwater	RCS	Remote control system
GEA	Scavenge air cooler (GEA manufacture)	RW1	Redwood seconds No. 1 (kinematic viscosity)
HFO	Heavy fuel oil	S/G	Shaft generator
HT	High temperature	SAC	Scavenge air cooler
IMO	International Maritime Organisation	SAE	Society of Automotive Engineers
IND	Indication	SHD	Shut down
IPDLC	Integrated power-dependent liner cooling	SIB	Shipyard interface box
ISO	International Standard Organisation	SLD	Slow down
kW	Kilowatt	SM	Sea margin
kWe	Kilowatt electrical	SSU	Saybolt second universal
kWh	Kilowatt hour	SW	Seawater
LAH	Level alarm, high	TBO	Time between overhauls
LAL	Level alarm, low	TC	Turbocharger
LCV	Lower calorific value	tEaT	Temperature of exhaust gas after turbine

LI	Level indicator	TI	Temperature indicator
LLT	Low-Load Tuning	A1xx / A2xx	Turbocharger (ABB manufacture)
LR	Light running margin	UMS	Unattended machinery space
LSL	Level switch, low	VI	Viscosity index
LT	Low temperature	WinGD	Winterthur Gas & Diesel Ltd.
M	Torque	WECS-9520	WinGD Engine Control System
M1H	External moment 1 st order horizontal	GTD	General Technical Data program
M1V	External moment 1 st order vertical	ΔM	Torque variation
M2V	External moment 2 nd order vertical		

22.3 SI dimensions for internal combustion engines

Symbol	Definition	SI-Units	Other units
a	Acceleration	m/s ²	
A	Area	m ² , mm ² , cm 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	
I, L	Length	m, mm, mm	
I _a , I _p	Second moment of area	m ⁴	
K	Coefficient of heat transfer	W/(m ² K)	
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	kg, t, g	
M, T	Torque moment of force	Nm	
N, n	Rotational frequency	1/s, 1/min	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	

Symbol	Definition	SI-Units	Other units
q_m	Mass flow rate	kg/s	
q_v	Volume flow rate	m ³ /s	
t	Time	s, d, h, min	
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
W, E, A, Q	Energy, work, quantity of heat	J, MJ, kJ, kWh	
Z, W	Section modulus	m ³	
ΔT , $\Delta \Theta$, ...	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 , g/cm ³ , kg/dm ³	
σ , τ	Stress	N/m ² , N/mm ²	
ω	Angular velocity	rad/s	

22.4 Approximate 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 US. 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 · (°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

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Winterthur Gas & Diesel Ltd. 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 sold under the WinGD brand name and are manufactured under licence in four shipbuilding countries. WinGD has its headquarters in Winterthur, Switzerland, where its activities were founded in 1898.

See also www.wingd.com