Marine Installation Manual

RT-flex58T-E

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List of Changes

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

Revision: 01 Date of issue: 2020-06	
Location of change	**Subject**
0 Preface Marine Installation Drawing Set	 Remark added to DG 9730
1.3 Components and sizes of the engine Design features	 This section removed, overview table added on page 1-2 instead
1.4 Engine tuning	Section renamed, updated and restructured
2.2.4 Power range	Section rewritten and updated
2.2.5 Power range limits	Section rewritten and updated
2.2.6 Power range limits with ME driven generator for FPP	Section title changed; section rewritten and updated
2.2.7 Prohibited operation area	Section title changed; section rewritten and updated
2.2.8 CPP requirements for Propulsion Control System	New section added with updated content from former section 2.2.7
3.5.1 Assembly of subassemblies	Note added
3.9.2 Earthing device Position of earthing device on shaft	 Note added
4.2 Cooling water system	Whole section restructured; content updated
4.3.2 Main lubricating oil system Lubrication of crosshead bearings Main lubricating oil system components	 New paragraph specifies application of booster pump for crosshead lubrication Full-flow filter specification: mesh size changed from max. 0.035 mm to max. 0.050 mm
4.4 Fuel oil system	Section partly revised
4.5.2 System specification Starting air compressors Starting air receivers	 Delivery gauge pressure defined more precisely Working gauge pressure defined more precisely
4.8 Engine room ventilation	Whole section restructured; content updated
4.10 PTO, PTI, PTH and primary generator applications	Whole section updated
5.4.2 Recommended manoeuvring characteristics FPP manoeuvring steps and warm-up times CPP manoeuvring steps and warm-up time	 Figure 5-6 added Figure 5-7 added
5.6 Alarm sensors and safety functions	Section rewritten
5.7 WinGD Integrated Digital Expert	New section added
6 Engine Dynamics	Whole chapter restructured; content updated
7.2 Engine noise	Introduction updated
9.1 Classification societies	Table 9-1: footnote removed
9.2 List of acronyms	Updated

Revision:	--	Date of issue:	**2019-01**	
Location of change				**Subject**
Entire document				The present Marine Installation Manual (MIM) is published in a completely new version with a new layout. It supersedes former MIM version ‘a5’ dated 6 April 2017. All future changes and updates (revisions) will be tracked and described based on the present Manual.

Table of Contents

List of Changes 0-1

0 **Preface** 0-1
Introduction 0-1
Marine Installation Drawing Set 0-2
Explanation of symbols used in this manual 0-3

1 **Engine Description** 1-1
1.1 Power/speed range 1-3
1.2 Primary engine data 1-4
1.3 Components and sizes of the engine 1-5
1.4 Engine tuning 1-6
1.4.1 BSFC and NOx emission 1-7
1.4.2 Standard tuning 1-8
1.4.3 Delta tuning 1-8
1.4.4 Delta bypass tuning 1-8
Exhaust gas waste gate 1-9
Exhaust gas temperature 1-10
Steam production 1-10
1.4.5 Low load tuning 1-11
1.4.6 Steam production control (SPC) 1-11
1.4.7 Low torsional vibration tuning (LowTV) 1-12
1.4.8 Tuning for de-rated engines 1-13
1.4.9 Dual tuning 1-14
1.5 The Flex system 1-15

2 **General Engine Data** 2-1
2.1 Pressure and temperature ranges 2-1
2.2 Engine rating field and power range 2-2
2.2.1 Introduction 2-2
2.2.2 Engine rating field 2-2
Rating points 2-3
2.2.3 Propeller diameter and influence of propeller revolutions 2-3
2.2.4 Power range 2-4
Propeller curves and operational points 2-4
Sea trial power 2-5
Sea margin (SM) 2-5
Light running margin (LR) 2-6
Continuous service rating (CSR) 2-6
Engine margin (EM) 2-6
Contracted maximum continuous rating (CMCR) 2-6
2.2.5 Power range limits 2-7
2.2.6 Power range limits with main-engine driven generator for FPP 2-10
PTO incorporation of Method 1 2-10
PTO incorporation of Method 2 2-12
2.2.7 Prohibited operation area 2-13
Prohibited operation area for different speed rated engines 2-15
2.2.8 CPP requirements for Propulsion Control System 2-18
2.3 Operating conditions 2-19

2.3.1 Reference conditions 2-19
2.3.2 Design conditions 2-19
2.4 Ancillary system design parameters 2-20
2.5 Electrical power requirement 2-21
2.6 GTD - General Technical Data 2-22

3 Engine Installation 3-1
3.1 Dimensions and masses 3-1
3.1.1 Dismantling heights for piston and cylinder liner 3-2
3.1.2 Crane requirements 3-2
3.1.3 Thermal expansion at turbocharger expansion joints 3-3
3.1.4 Content of fluids in the engine 3-4
3.2 Engine outline views 3-5
3.3 Platform arrangement 3-6
3.3.1 Drawings 3-6
3.3.2 Minimum requirements for escape routes 3-6
3.4 Seating 3-7
3.5 Assembly 3-8
3.5.1 Assembly of subassemblies 3-8
3.5.2 Installation of a complete engine 3-9
3.5.3 Installation of an engine from assembled subassemblies 3-9
3.5.4 Installation of an engine in ship on slipway 3-9
3.6 Engine and shaft alignment 3-10
3.6.1 Instructions and limits 3-10
3.6.2 Tools 3-10
3.7 Engine coupling 3-11
3.7.1 Design 3-11
3.7.2 Machining and fitting of coupling bolts 3-11
3.7.3 Tightening 3-11
3.7.4 Installation drawing 3-11
3.8 Engine stays 3-12
3.9 Propulsion shaft earthing 3-13
3.9.1 Preventive action 3-13
3.9.2 Earthing device 3-13
3.10 Fire protection 3-16

4 Ancillary Systems 4-1
4.1 Twin-engine installation 4-2
4.2 Cooling water system 4-5
4.2.1 Central freshwater cooling system components 4-7
Low-temperature circuit 4-7
High-temperature circuit 4-9
4.2.2 Cooling water treatment 4-10
4.2.3 General recommendations for design 4-11
4.2.4 Freshwater generator 4-11
4.2.5 Pre-heating 4-12
Pre-heating from cooling water systems 4-12
Pre-heating by direct water circulation 4-12
4.3 Lubricating oil systems 4-14
4.3.1 Lubricating oil requirements 4-14
4.3.2 Main lubricating oil system 4-14
Main lubricating oil system components 4-15

System oil 4-17
4.3.3 Flushing the lubricating oil system 4-17
4.3.4 Lubrication for turbochargers 4-17
4.3.5 Cylinder lubricating oil system 4-17
Service tank and storage tank 4-18
Electrical trace heating for cylinder lubricating oil piping 4-18
4.3.6 Maintenance and treatment of lubricating oil 4-19
4.3.7 Drain tank. 4-20
4.4 Fuel oil system 4-25
4.4.1 Fuel oil system components 4-25
Feed pump — Low-pressure fuel oil 4-26
Pressure regulating valve 4-27
Mixing unit 4-27
Booster pump — High-pressure fuel oil 4-29
End-heater 4-29
Viscometer 4-30
MDO/MGO heat exchanger 4-30
Fuel oil filters — Arrangement ‘A’ 4-31
Fuel oil filter — Arrangement ‘B’ 4-36
4.4.2 Flushing the fuel oil system 4-36
4.4.3 Fuel oil treatment 4-37
Settling tanks 4-37
Service tanks 4-37
Centrifugal fuel oil separators 4-37
4.4.4 Pressurised fuel oil system 4-39
4.4.5 Fuel oil specification. 4-39
4.4.6 Fuel oil viscosity-temperature dependency 4-40
4.5 Starting and control air system 4-41
4.5.1 Capacities of air compressor and receiver. 4-42
4.5.2 System specification 4-42
Starting air compressors 4-42
Starting air receivers 4-42
4.5.3 Control air 4-43
4.5.4 Service and working air 4-43
4.6 Leakage collection system and washing devices 4-44
4.6.1 Draining of exhaust uptakes 4-45
4.6.2 Air vents 4-45
4.7 Exhaust gas system 4-46
4.8 Engine room ventilation 4-47
4.8.1 Ventilation requirements 4-47
4.8.2 Ventilation arrangement. 4-48
Arrangement 1 — Engine room ventilation system 4-49
Arrangement 2 — Direct engine ventilation system. 4-50
4.8.3 Air intake quality. 4-51
4.8.4 Outside ambient air temperature 4-53
4.9 Piping 4-54
4.9.1 Pipe connections 4-54
4.9.2 Flow rates and velocities 4-54
4.10 PTO, PTI, PTH and primary generator applications 4-55
4.10.1 Requirements. 4-55
4.10.2 Arrangements for PTO, PTI, PTH and primary generator 4-55
4.10.3 Application constraints 4-57

4.10.4 Service conditions 4-59

5 Engine Automation 5-1
5.1 DENIS-9520 5-1
5.2 DENIS-9520 concept 5-2
5.2.1 Interface definition 5-2
5.2.2 Approved Propulsion Control Systems 5-2
5.3 DENIS-9520 Specification 5-3
5.3.1 DENIS-9520 Interface Specification 5-3
5.3.2 DENIS-9520 Propulsion Control Specification 5-3
5.4 Propulsion Control Systems 5-4
5.4.1 PCS functions 5-6
Remote Control System 5-6
Electronic Speed Control System 5-6
Safety System 5-7
Telegraph System 5-7
Local manual control 5-7
ECR manual control panel 5-7
Options 5-7
5.4.2 Recommended manoeuvring characteristics 5-9
5.5 Alarm and Monitoring System 5-12
5.5.1 Integrated solution 5-12
5.5.2 Split solution 5-12
5.6 Alarm sensors and safety functions 5-13
5.6.1 Signal processing 5-13
5.6.2 Requirements from WinGD and classification societies 5-13
5.7 WinGD Integrated Digital Expert 5-14
5.7.1 Data Collection and Monitoring 5-14
5.7.2 Engine Diagnostic System 5-15
5.7.3 WiDE installation process 5-16

6 Engine Dynamics 6-1
6.1 External mass forces and moments 6-2
6.1.1 Balancing of mass forces and moments 6-2
6.1.2 Countermeasure for second order vertical mass moments 6-3
Electrically driven compensator (external compensator) 6-3
Power related unbalance 6-4
6.2 External lateral forces and moments 6-5
6.2.1 Lateral vibration types 6-6
H-type vibration 6-6
X-type vibration 6-6
6.2.2 Reduction of lateral vibration 6-7
Lateral hydraulic stays 6-7
Electrically driven compensator 6-8
6.3 Longitudinal vibration (pitching) 6-9
Reduction of longitudinal vibration (5-cylinder engines) 6-9
6.4 Torsional vibration 6-11
6.4.1 Reduction of torsional vibration 6-11
Low-energy vibrations 6-12
High-energy vibrations 6-12
6.4.2 PTO/PTI systems effect on torsional vibration 6-13
6.5 Axial vibration 6-14

Reduction of axial vibration 6-14
6.6 Whirling vibration 6-15
6.7 Hull vibration 6-16
6.8 Countermeasures for dynamic effects 6-17
6.8.1 External mass moments and vibrations 6-17
6.8.2 Synchro-Phasing System in twin engines 6-18
Concept 6-18
Components and control 6-19
Operating modes and restrictions 6-20
6.9 Order forms for vibration calculation & simulation 6-21

7 Engine Emissions 7-1
7.1 Exhaust gas emissions 7-1
7.1.1 Regulation regarding NOx emissions 7-1
7.1.2 Selective catalytic reduction 7-2
Low-pressure SCR 7-2
High-pressure SCR 7-3
7.2 Engine noise 7-4
7.2.1 Air-borne noise 7-4
7.2.2 Exhaust noise 7-6
7.2.3 Structure-borne noise 7-8

8 Engine Dispatch 8-1
8.1 Engines to be transported as part assemblies 8-1
8.2 Protection of disassembled engines 8-1
8.3 Removal of rust preventing oils after transport 8-1
8.3.1 Internal parts 8-1
8.3.2 External parts 8-1

9 Appendix 9-1
9.1 Classification societies 9-1
9.2 List of acronyms 9-2
9.3 SI dimensions for internal combustion engines 9-4
9.4 Approximate conversion factors 9-6

List of Tables

1-1 Principal engine features 1-2
1-2 Rating points 1-4
1-3 Overall sizes and masses 1-5
1-4 Available tuning options 1-6
2-1 Line 5 coefficients 2-8
2-2 Line 6 coefficients 2-9
2-3 Line 10 coefficients 2-11
2-4 Electrical power requirement 2-21
3-1 Engine dimensions and masses 3-1
3-2 Fluid quantities in the engine 3-4
3-3 Recommended quantities of fire extinguishing medium 3-16
4-1 Common and independent systems in twin-engine installations 4-2
4-2 Recommended parameters for raw water 4-10
4-3 Minimum inclination angles for full operability of the engine (1) 4-22
4-4 Minimum inclination angles for full operability of the engine (2) 4-23
4-5 Minimum inclination angles for full operability of the engine (3) 4-24
4-6 Specification of automatic self-cleaning filter in feed system 4-33
4-7 Specification of automatic self-cleaning filter in booster system 4-34
4-8 Specification of duplex filter in booster system 4-35
4-9 Control air flow capacities 4-43
4-10 Guidance for air filtration 4-51
4-11 PTO/PTI/PTH arrangements for RT-flex58T-E 4-57
4-12 Possible options for RT-flex58T-E 4-57
4-13 Influence of options on engineering 4-58
5-1 Suppliers of RCS and Electronic Speed Control System 5-4
5-2 Recommended manoeuvring steps and warm-up times for FPP 5-10
5-3 Recommended manoeuvring steps and warm-up times for CPP 5-11

5-4 **Additional Class requirements for alarm sensors and safety functions** . 5-13

6-1 **Countermeasures for external mass moments** 6-17

6-2 **Countermeasures for lateral and longitudinal vibrations** 6-17

6-3 **Countermeasures for torsional and axial vibrations of the shafting** 6-17

9-1 **List of classification societies** 9-1

9-2 **List of acronyms** 9-2

9-3 **SI dimensions** 9-4

9-4 **Conversion factors** 9-6

List of Figures

1-1 Positions of supply unit on RT-flex58T-E 1-1

1-2 Power/speed range of WinGD diesel engines 1-3

1-3 Cross section 1-5

1-4 Typical BSFC curves in relation to engine power. 1-7

1-5 Steam production power diagram 1-8

1-6 Functional principle of an exhaust gas waste gate 1-9

1-7 Exhaust gas temperature increase with DBT 1-10

1-8 Steam production of Delta bypass tuning with variable bypass. 1-12

1-9 Vibration amplitudes - Achievements with default LowTV tuning 1-13

1-10 Application area for tuning options 1-13

1-11 Flex system parts. 1-15

2-1 Rating field for RT-flex58T-E. 2-2

2-2 Propeller curves and operational points 2-5

2-3 Power range limits 2-7

2-4 Power range diagram of an engine with main-engine driven generator. . 2-10

2-5 Power range limits for PTO operation — Method 1 2-11

2-6 Power range limits for PTO operation — Method 2 2-12

2-7 Prohibited engine operation area (CMCR speed = R1—R2) 2-13

2-8 Calculating prohibited operation area for CMCR speed 2-15

2-9 Prohibited engine operation area (CMCR speed = 85% of R1—R2). 2-16

2-10 Prohibited engine operation area (CMCR speed = R3—R4) 2-17

3-1 Engine dimensions 3-1

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

3-3 Minimum requirements for headroom 3-6

3-4 Typical shaft earthing arrangement 3-14

3-5 Typical shaft earthing with condition monitoring facility 3-15

4-1 LT cooling water system for twin-engine installation 4-3

4-2 Cylinder LO system for twin-engine installation 4-4
4-3 Scheme of cooling water system 4-5
4-4 Principal layouts of HT cooling water circuit 4-6
4-5 Pre-heating power requirement per cylinder 4-13
4-6 Lubricating oil system 4-14
4-7 Dimensioning and filling process of lubricating oil drain tank 4-20
4-8 Arrangement of vertical lubricating oil drains for 6-cylinder engines . . . 4-21
4-9 Fuel oil system 4-25
4-10 Mixing unit 4-28
4-11 Mesh size difference between absolute and nominal 4-31
4-12 Fuel oil filter arrangement ‘A’ 4-32
4-13 Fuel oil filter arrangement ‘B’ 4-36
4-14 Fuel oil viscosity-temperature diagram 4-40
4-15 Starting and control air system 4-41
4-16 Sludge oil trap 4-44
4-17 Arrangement of automatic water drain 4-45
4-18 Determination of exhaust pipe diameter 4-46
4-19 Ventilation system arrangement 1 — Engine room ventilation system . . 4-49
4-20 Ventilation system arrangement 2 — Direct engine ventilation system . . 4-50
4-21 Air filter size (example for 8-cyl. engine) 4-52
4-22 Arrangements for PTO, PTI, PTH 4-56
4-23 FPP with mandatory frequency converter 4-59
4-24 CPP in combination with an optional frequency converter 4-60
4-25 CPP in constant speed operation without frequency converter 4-60
4-26 CPP with two fixed operation speeds without frequency converter 4-61
5-1 Engine automation architecture 5-1
5-2 Engine management and automation concept 5-2
5-3 Remote Control System 5-5

5-4 **Propulsion Control** 5-8

5-5 **Manoeuvring speed/power settings for FPP/CPP installations** 5-9

5-6 **Full sea load steps in FPP load-up program** 5-10

5-7 **Full sea load steps in CPP load-up program** 5-11

5-8 **WiDE system** 5-14

5-9 **WiDE installation process map** 5-16

6-1 **External mass forces and moments** 6-2

6-2 **Locating an electrically driven compensator** 6-3

6-3 **Forces through the engine** 6-5

6-4 **Lateral vibration — X-type and H-type** 6-6

6-5 **General arrangement of hydraulic stays for one-side installation** 6-7

6-6 **General arrangement of hydraulic stays for both-side installation** 6-8

6-7 **Arrangement of longitudinal stays** 6-9

6-8 **Vibration dampers (spring type and viscous type)** 6-12

6-9 **Example of axial vibration damper** 6-14

6-10 **Resulting vibration from SPS combinations** 6-18

6-11 **Synchro-Phasing system** 6-19

7-1 **Speed dependent maximum allowable average of NOx emissions** 7-1

7-2 **Low-pressure SCR — Arrangement** 7-2

7-3 **High-pressure SCR — Arrangement** 7-3

7-4 **Sound pressure level at 1m distance from engine** 7-5

7-5 **Exhaust noise reference point** 7-6

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

7-7 **Structure-borne noise level at engine feet vertical** 7-8

0 Preface

Introduction

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

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

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

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

Attention is drawn to the following:

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

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

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

Tier II certified

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

Marine Installation Drawing Set

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

Engine design groups

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

9707	Engine Alignment Record Sheets
9709	Engine Alignment
9710	Engine Seating / Foundation
9710-01	Tool Engine Alignment
9715	Engine Stays
9721	Cooling Water Systems
9722	Lubricating Oil Systems
9723	Fuel Oil System
9724	Leakage Collection / Washing System
9725	Starting and Control Air System
9726	Exhaust System
9730	Various Installation Items [1)]

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

Links to complete drawing packages

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

— Marine installation drawings:
  MIDS - Complete package

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

1) A key for MIDS *Piping Symbols* is included in the design group ‘Various Installation Items’ (DG 9730) for reference.

Explanation of symbols used in this manual

Cross references

Cross references are written in blue. They lead to another section or a table or figure in this manual and can be activated by mouseclick.
They consist of the number of the respective figure or table, or the section title, followed by the page symbol introducing the page number.
Example: Table 4-4, 🗎 4-23

Notes

They give additional information considered important, or they draw the reader’s attention to special facts.
Example:

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

Weblinks

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

Docu WinGD

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

Docu WinGD

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

GTD

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

1 Engine Description

The WinGD RT-flex58T-E engine is a camshaftless low-speed, reversible and rigidly direct-coupled two-stroke engine featuring common-rail injection.

Bore:	580 mm
Stroke:	2,416 mm
Number of cylinders:	5 to 8
Power (MCR):	2,350 kW/cyl
Speed (MCR):	105 rpm
Mean effective pressure:	21.0 bar
Stroke/bore ratio:	4.17

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

WECS-9520 Engine Control System

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

Position of supply unit

With the RT-flex58T-E engine, the position of the supply unit depends on the number of cylinders (see Figure 1-1):

- 5 and 6 cylinders: supply unit on aft end
- 7 and 8 cylinders: supply unit on exhaust side

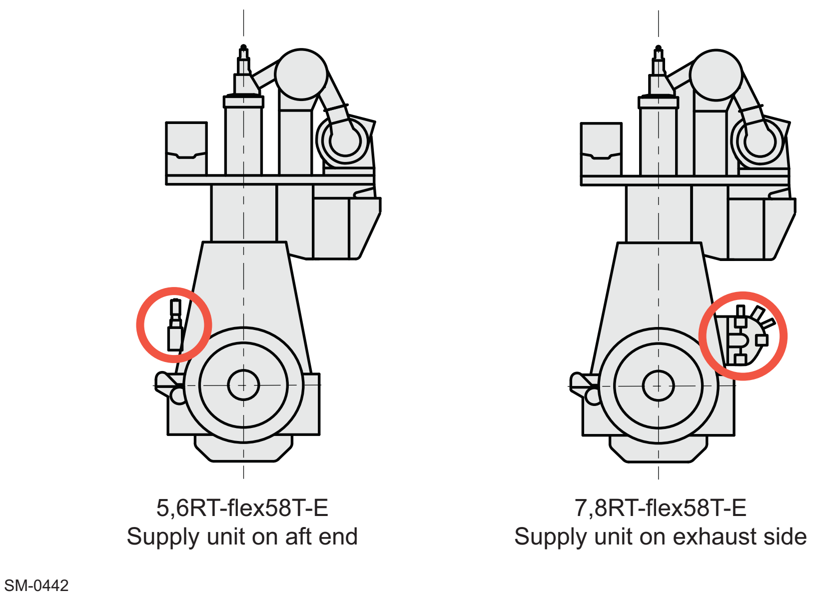


Figure 1-1 Positions of supply unit on RT-flex58T-E

Engine performance data are the same, irrespective of the supply unit position.

Table 1-1 Principal engine features

Engine features	MIM section
Tier III compliance available with low and high-pressure SCR	7.1.2
Different engine tunings available, best fitting to operating profile, incl. steam production optimisation option	1.4
Low torsional vibration tuning available	1.4.8
Rail unit: Common rail injection and exhaust valve actuation controlled by quick-acting solenoid valves	1.3
Supply unit: High-efficiency fuel pumps feeding the high-pressure fuel rail. Adaptive injector cut-off at lower loads optimises the combustion and fuel consumption at lower engine loads	4.4.1
Pulse Jet Lubricating System (PLS) for high-efficiency cylinder lubrication with optimised cylinder lubricating oil consumption	4.3
Piston with crown, cooled by combined jet-shaker oil cooling	4.3.2
Fully electronically controlled engine	5
Data collection and monitoring (WiDE)	5.7
Synchro-Phasing System (SPS) for twin-engine installations available, if contracted	6.8.2

1.1 Power/speed range

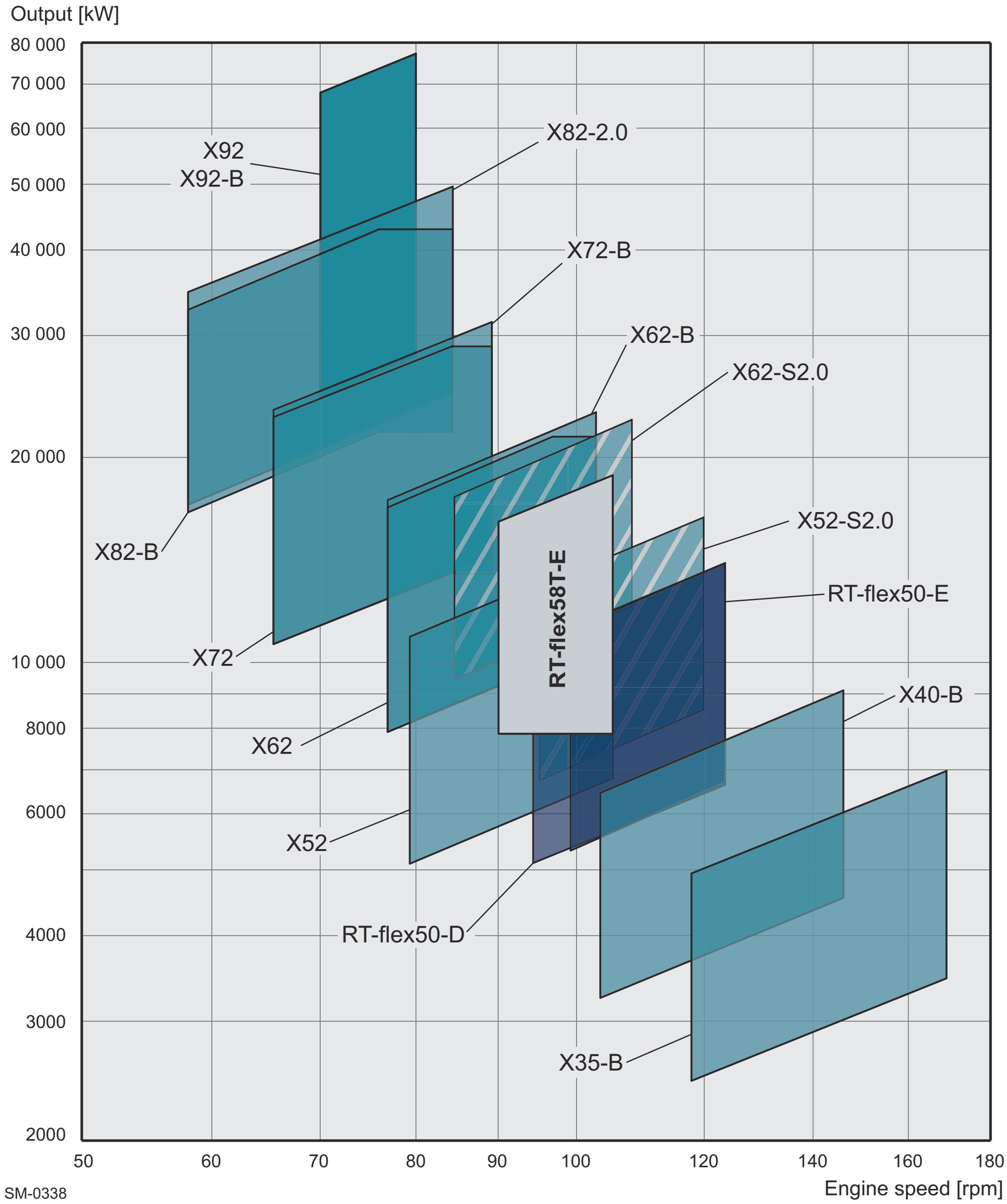


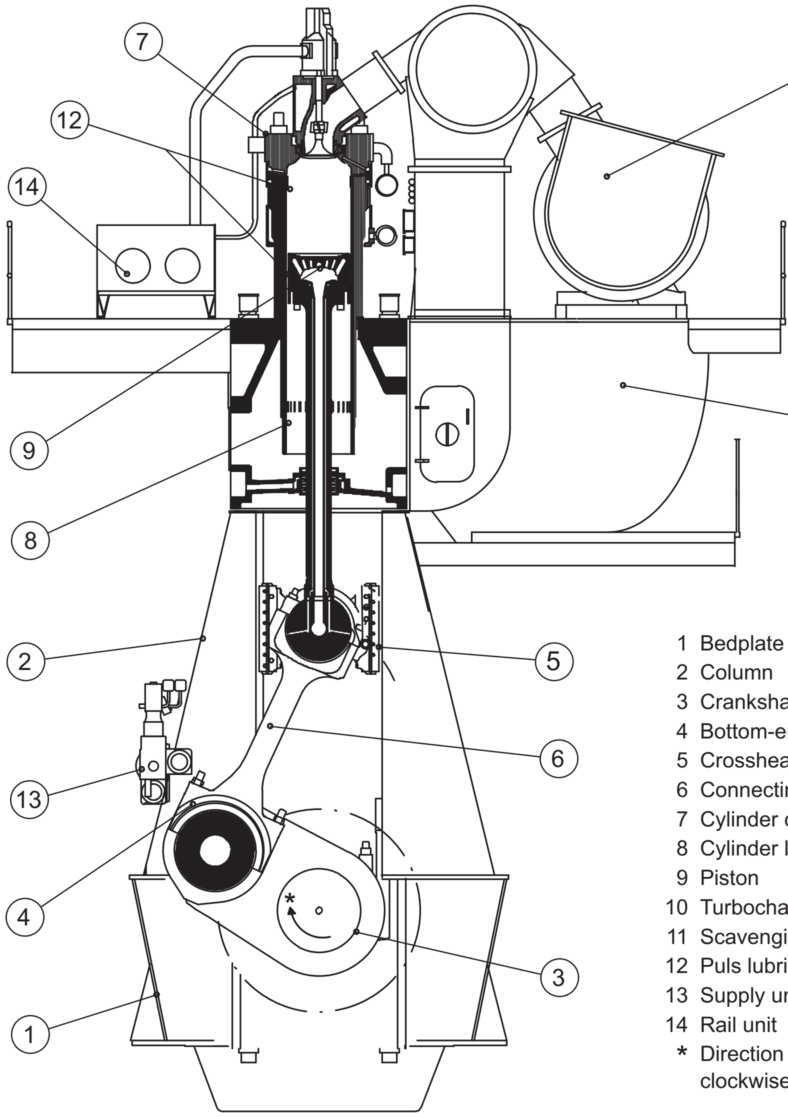
Figure 1-2 Power/speed range of WinGD diesel engines

1.2 Primary engine data

Table 1-2 Rating points

Bore x stroke: 580 x 2,416 [mm]				
No. of cyl.	**R1**	**R2**	**R3**	**R4**
	Power [kW]			
5	11,750	7,900	10,075	7,900
6	14,100	9,480	12,090	9,480
7	16,450	11,060	14,105	11,060
8	18,800	12,640	16,120	12,640
Speed [rpm]				
All cyl.	105	105	90	90
Brake specific diesel fuel consumption (BSFC) [g/kWh] 100 % power				
All cyl.	167.8	161.8	167.8	161.8
Mean effective pressure (MEP) [bar]				
All cyl.	21.0	14.1	21.0	16.5
Lubricating oil consumption (for fully run-in engines under normal operating conditions)				
System oil	approx. 6 kg/cyl per day			
Cylinder oil	guide feed rate 0.6 g/kWh (for low sulphur content only)			
BSFC data are quoted for fuel of lower calorific value 42.7 MJ/kg. All other reference conditions refer to ISO standard (ISO 3046-1). For BSFC the following tolerances are to be taken into account: +5 % for 100-85 % engine power +6 % for <85-65 % engine power +7 % for <65-50 % engine power				
The data given in this table refer to Standard tuning.				

1.3 Components and sizes of the engine



1 Bedplate
2 Column
3 Crankshaft
4 Bottom-end bearings
5 Crosshead
6 Connecting rod
7 Cylinder cover
8 Cylinder liner
9 Piston
10 Turbocharging system
11 Scavenging system
12 Puls lubricating system
13 Supply unit
14 Rail unit
* Direction of rotation: clockwise as standard

SM-0001 This cross section is considered as general information only.

Figure 1-3 Cross section

Table 1-3 Overall sizes and masses

No. of cyl.	Length [mm]	Piston dismantling height F1[a] (crank centre - crane hook) [mm]	Dry weight [t]
5	6,381	10,960	281
6	7,387		322
7	8,393		377
8	9,399		418

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

1.4 Engine tuning

As the Flex system (see section 1.5, 1-15) allows 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 options to optimise the brake specific fuel consumption (BSFC) at individual engine loads.

Compliance with IMO Tier II and III

All tuning options comply with the IMO Tier II regulations for NO_x emissions. For Tier III emission compliance, an exhaust gas treatment is required as described in 7.1.2 Selective catalytic reduction, 7-2.

GTD

Combinations of tuning and exhaust gas treatment methods can be obtained from the *GTD* application.

Engine tuning options

The following table gives an overview of the available tuning options with their application and the required engine components. Tuning options need to be specified at a very early stage of the project.

Table 1-4 Available tuning options

Tuning	Description	Application	Additional components
Standard tuning (Std)	High-load tuning	When ship operates most of the time above 90 % engine power	None
Delta tuning (Delta)	Part-load tuning	When ship operates most of the time between 75 and 90 % engine power	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 power	When ship operates most of the time at less than 75 % engine power	Exhaust gas waste gate

GTD

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

NOTE	The tuning options must be predefined along with any engine order.

LowTV tuning
see section 1.4.7, 1-12

Low torsional vibration tuning (LowTV) can be applied when vibrations arise with 5-, 6-, and 7-cylinder engines (see 6.4 Torsional vibration, 6-11). This tuning method is combined with the available tuning options listed in Table 1-4.

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

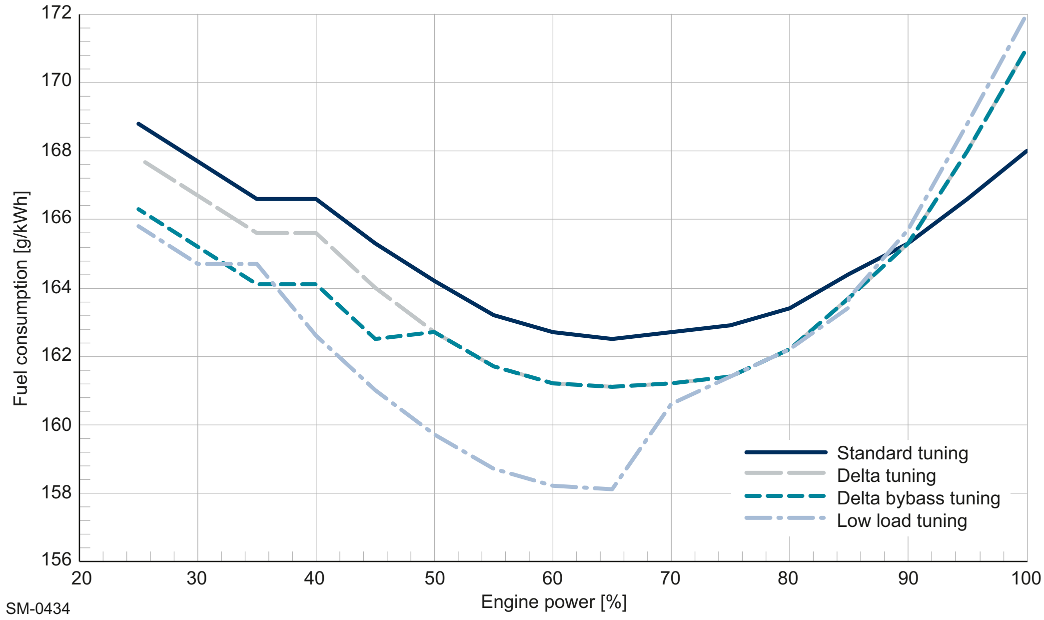


Figure 1-4 Typical BSFC curves in relation to engine power

BSFC data for Standard tuning are given in Table 1-2, 1-4.

GTD

BSFC data for the other tuning options can be obtained from the *GTD* application.

NOTE	The reliability of the engine is by no means impaired by applying a tuning option. All mechanical stresses and thermal loads are well within limits irrespective of engine tuning.

1.4.1 BSFC and NO_x emission

The parameters controlling the fuel injection and exhaust valve timing are modified with the engine tuning process. This ensures full tuning potential by suitably balancing the design related limitations, BSFC and NO_x.

There is a trade-off between BSFC and NO_x emissions, where low BSFC results in high NO_x emissions and vice versa. To ensure that IMO regulations are met, any associated increase in NO_x emissions at specific load ranges must be compensated with a reduction in other load ranges.

1.4.2 Standard tuning

Standard tuning is based on camshaft controlled engines. Although the Flex technology seldom uses the Standard tuning option, it is still used as a reference for the more advantageous Delta, DBT and LLT.

1.4.3 Delta tuning

The Delta tuning option is used to reduce the BSFC in the part-load range by tailoring the firing pressure and the firing compression ratio of the engine to maximum efficiency below 90% load. However, this is offset with a reduction in efficiency towards full load.

1.4.4 Delta bypass tuning

Delta bypass tuning is an engine tuning option designed to increase the exhaust gas temperature and steam production power (SPP), therefore allowing for a reduction in auxiliary boilers use. This increase occurs at loads of more than 50%, while still complying with all existing emission legislations.

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

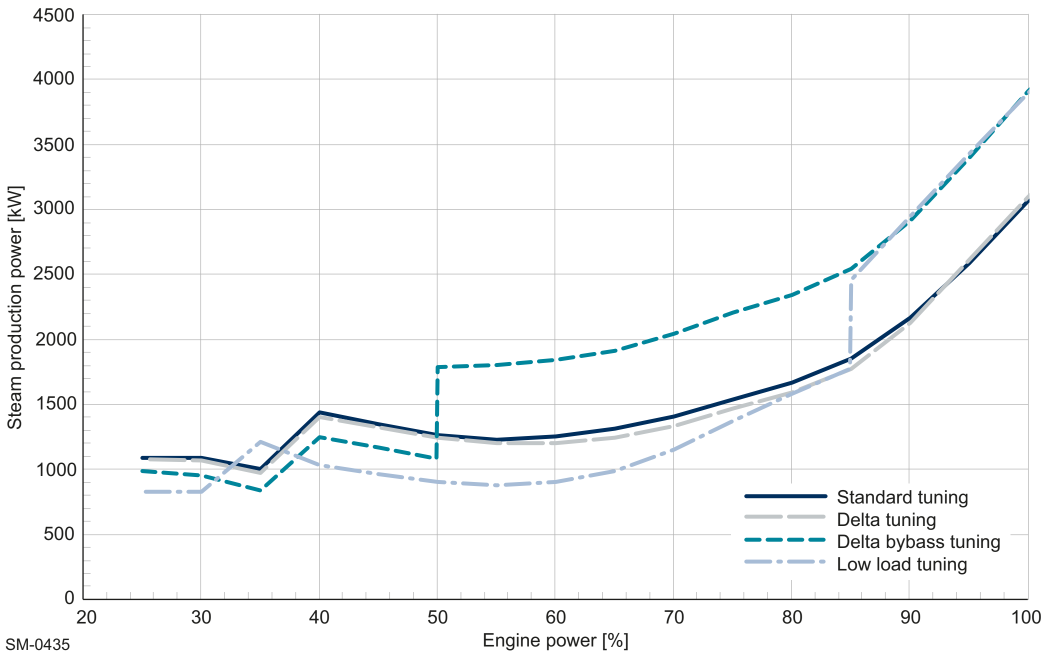


Figure 1-5 Steam production power diagram

Besides the appropriately adjusted engine parameters related to fuel injection and exhaust valve control, the DBT concept combines a specifically designed turbocharger system setup with the use of an exhaust gas waste gate (with a 50% power switch-point).

Exhaust gas waste gate

DBT requires the fitting of an exhaust gas waste gate on the exhaust gas receiver before the turbocharger turbine (as seen in Figure 1-6). Exhaust gas passing through this valve bypasses the turbocharger, flowing to the main exhaust uptake.

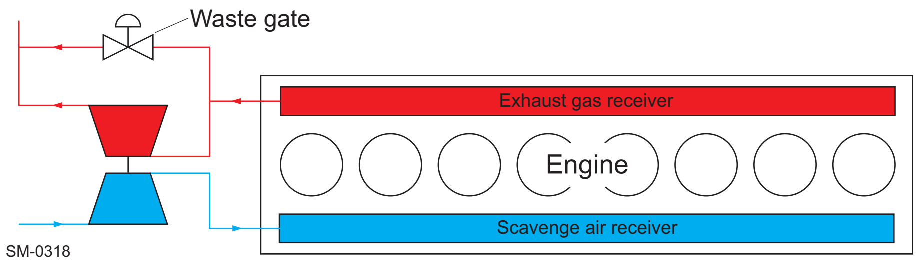


Figure 1-6 Functional principle of an exhaust gas waste gate

Working range

The exhaust gas waste gate works as so:

- Below 50% engine power → Waste gate is closed
  All exhaust gas flows into the turbocharger, this increases combustion pressure due to increased scavenge air pressure. As a consequence, the BSFC is reduced at low load compared to Delta tuning.

- Above 50% engine power → Waste gate is open
  A small percentage of the exhaust gas bypasses the turbocharger. This reduces the mass flow rate of the turbocharger and the pressure of the scavenge air. As a consequence, the exhaust temperature rises, allowing for an increase in the steam production by means of an economiser.

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

Exhaust gas temperature

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

tEaT and tEbE

In particular the tEaT (temperature exhaust gas after turbocharger) is approximately 20 °C higher at 70% engine power than with Delta tuning. This increase is principally due to the slowing of the turbocharger. The open waste gate bypass reduces the mass flow rate of exhaust gas, resulting in a relative reduction of the scavenge air.

The tEbE (temperature exhaust gas before economiser) is further increased (about 5 °C) due to the mixing of exhaust gas from the waste gate bypass.

As seen in Figure 1-7, the Delta tuning exhaust gas temperature does not change from the turbocharger to the economiser, as there isn't this mixing of additional bypassed exhaust gas.

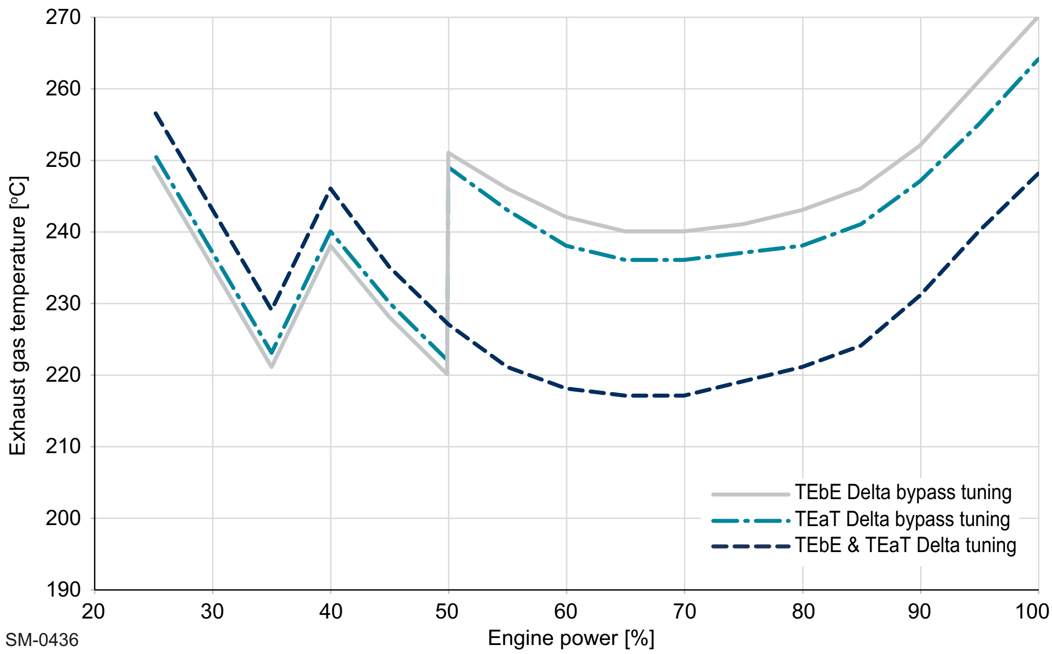


Figure 1-7 Exhaust gas temperature increase with DBT

Steam production

Increasing the exhaust gas temperature to produce more steam by way of the economiser is an efficient way of powering on-board steam services and using waste heat from main engine exhaust gas.

In such condition DBT is the most economical tuning option; see Figure 1-5, 1-8. Within certain engine power ranges it may be possible to run without any auxiliary boiler.

GTD

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

1.4.5 Low load tuning

The Low load tuning option is used to reduce the BSFC in the lower part-load range by optimising the engine and turbocharger to match for this low load operation. However, this is offset with a reduction in efficiency towards full load.

Like DBT, LLT must consider engine parameters related to fuel injection and exhaust valve control, combining a specifically designed turbocharger system setup with the use of an exhaust gas waste gate (with a 85% power switch-point); see Exhaust gas waste gate, 🗎 1-9.

Working range

The exhaust gas waste gate works as so:

- Below 85% engine power → Waste gate is closed
  All exhaust gas flows into the turbocharger, this increases combustion pressure due to increased scavenge air pressure. As a consequence, the BSFC is reduced at low load.

- Above 85% engine power → Waste gate is open
  As the turbocharger is optimised for lower part-load operation, at higher loads there is a surplus of available exhaust gas energy. This needs to be released via the open waste gate to protect against turbocharger overspeed.

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

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

1.4.6 Steam production control (SPC)

The SPC system consists of an analogue controlled valve that enables the opening and closing of the exhaust gas waste gate (see Exhaust gas waste gate, 🗎 1-9), regulating the bypass of the turbocharger from the main engine. By increasing the bypass rate it reduces the mass flow rate of the turbocharger, this in turn increases the exhaust gas heat, which is used to produce steam as needed.

The SPC option can be applied to DBT and LLT, as the tuning options are already equipped with an exhaust waste gate (see Exhaust gas waste gate, 🗎 1-9). Without the SPC this waste gate valve is either open or closed according to a set engine power percentage. The SPC constantly reacts, restricting the bypass flow to an optimum level. This is achieved by adjusting the valve according to real time steam pressure values, enabling the SPC system to maintain a set steam requirement.

Connection to external systems

The SPC is connected to and receives inputs from external systems, such as the exhaust gas economiser and auxiliary boiler control systems. The additional systems work together with the engine to manage the valve. The system's automation and optimisation ensures steam requirement without over production, as defined by the user. This is true regardless of the engine power (as seen in Figure 1-8), where a minimum steam production requirement is set and maintained across the engine power range. With the availability of increased steam, the SPC is more efficient than switching on an auxiliary boiler, with overall fuel and cost saving.

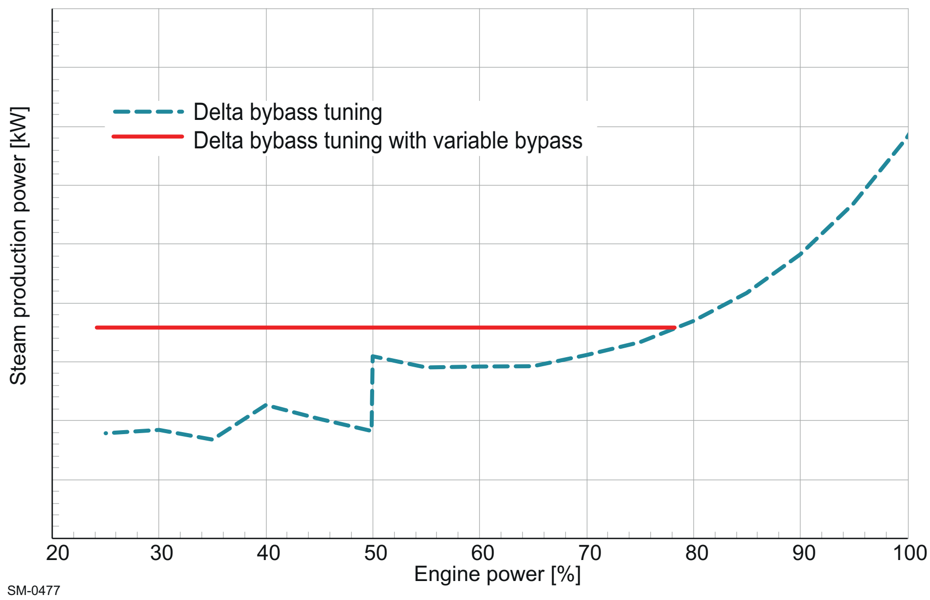


Figure 1-8 Steam production of Delta bypass tuning with variable bypass

As well as a fully integrated steam production control system, user operated waste gate control is also available. Such an arrangement remains restricted to within defined engine limitations, however does not ensure optimised efficiency.

Performance data referring to the use of the SPC in conjunction with WinGD engines can be obtained using the *GTD* application.

1.4.7 Low torsional vibration tuning (LowTV)

If required LowTV tuning is applied to the RT-flex58T-E on the 5-, 6-, and 7-cylinder engines, in many cases negating the need for a costly torsional vibration damper.

Figure 1-9, 🗎 1-13 shows a comparison in regard to torsional vibration when LowTV tuning is applied. At a certain engine speed, the measured torsional vibration amplitudes decreased by nearly 30%.

NOTE	LowTV tuning does not impair the engine performance.

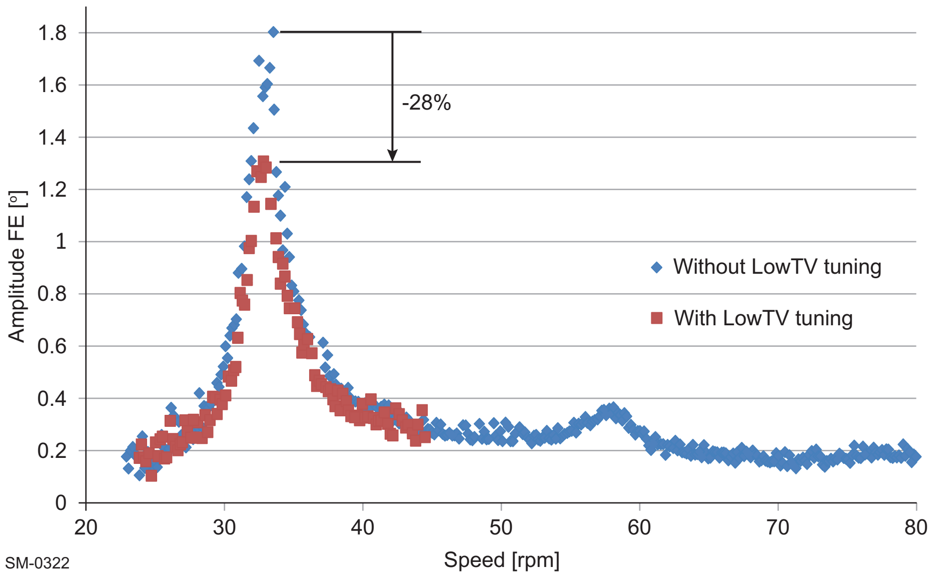


Figure 1-9 Vibration amplitudes - Achievements with default LowTV tuning

1.4.8 Tuning for de-rated engines

- Standard and Delta bypass tuning are applicable over the whole rating field.
- Delta and Low load tuning are applied as illustrated in Figure 1-10.

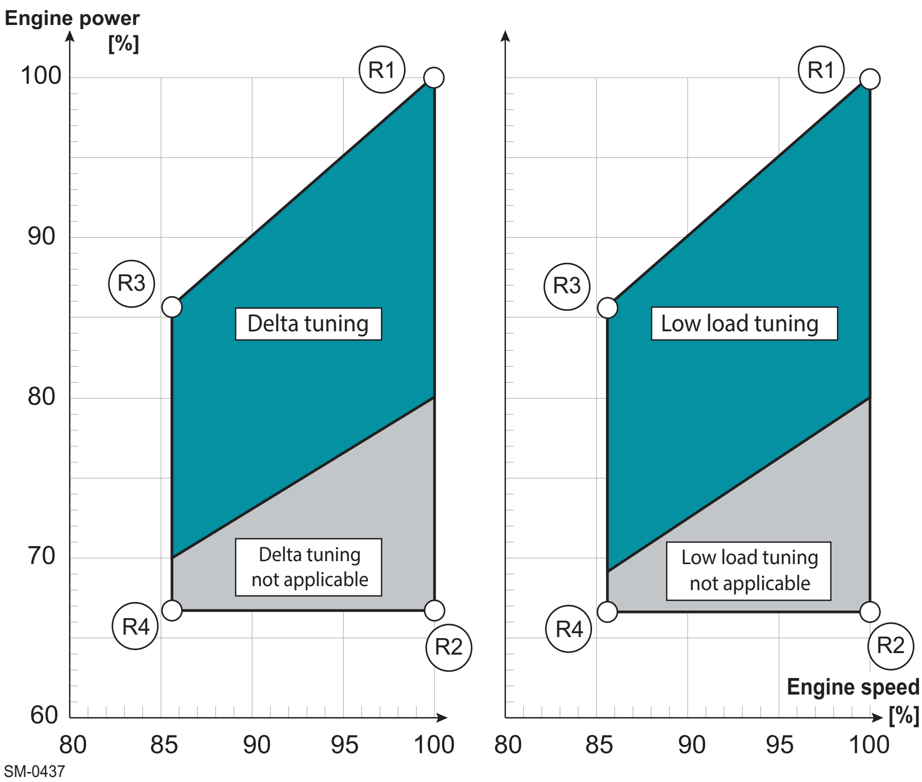


Figure 1-10 Application area for tuning options

1.4.9 Dual tuning

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

Changeover between tuning regimes

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

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

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

Considerations to be made when choosing dual tuning

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

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

1.5 The Flex system

The RT-flex58T-E engine is equipped with WinGD's common rail fuel injection technology, allowing flexible fuel injection. The flexibility provided by this technology is reflected in the naming *Flex system*.

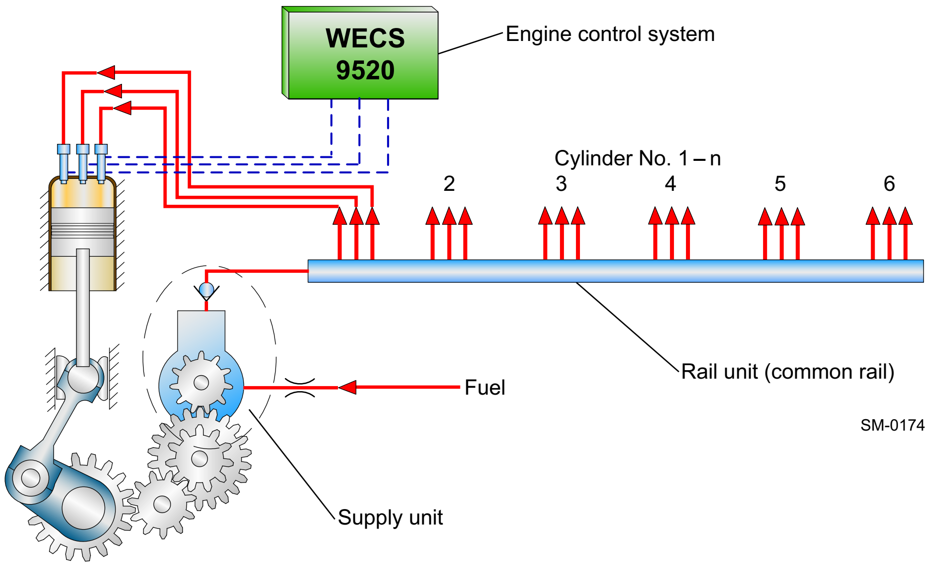


Figure 1-11 Flex system parts

Major benefits

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

2 General Engine Data

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

2.1 Pressure and temperature ranges

Please refer to the document **'Usual values and safeguard settings'**, which is provided by WinGD under the following link:
Usual values and safeguard settings

For signal processing see also 5.6.1 Signal processing, 5-13.

2.2 Engine rating field and power range

2.2.1 Introduction

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

2.2.2 Engine rating field

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

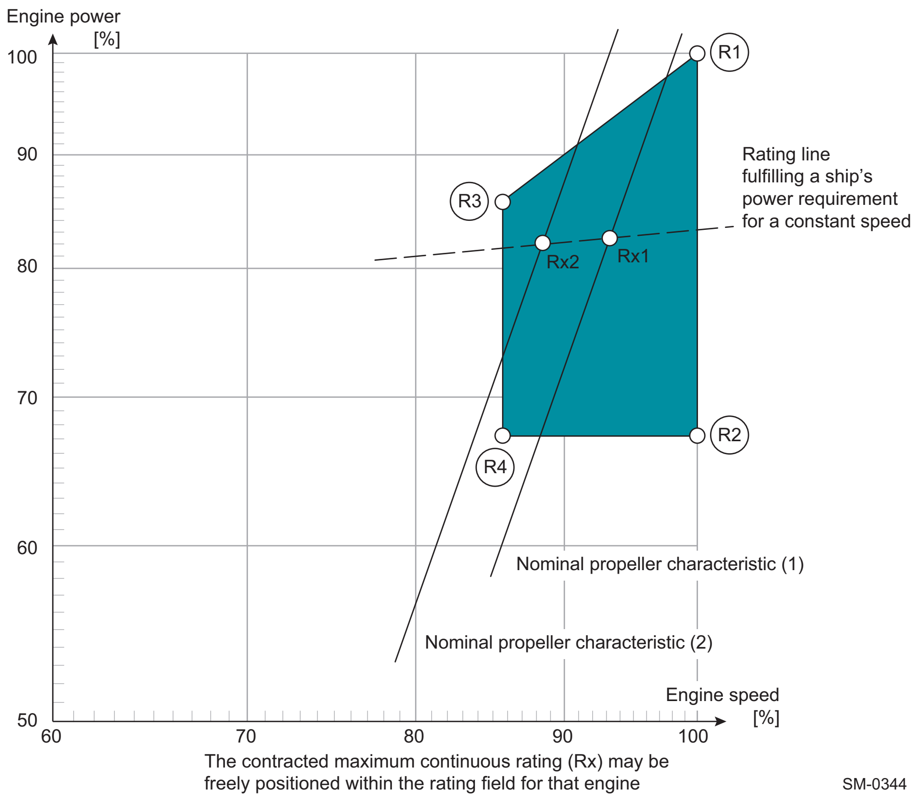


Figure 2-1 Rating field for RT-flex58T-E

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

Percentage values

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

Rating points

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

The point R1 represents the nominal maximum continuous rating (MCR). It is the maximum power/speed combination which is available for a particular engine.

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

2.2.3 Propeller diameter and influence of propeller revolutions

Influence of propeller revolutions on the power requirement

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

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

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

Formula 2-1

where:

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

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

Maximum propeller diameter

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

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

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

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

2.2.4 Power range

Propeller curves and operational points

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

Determining power/propeller speed relationships

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

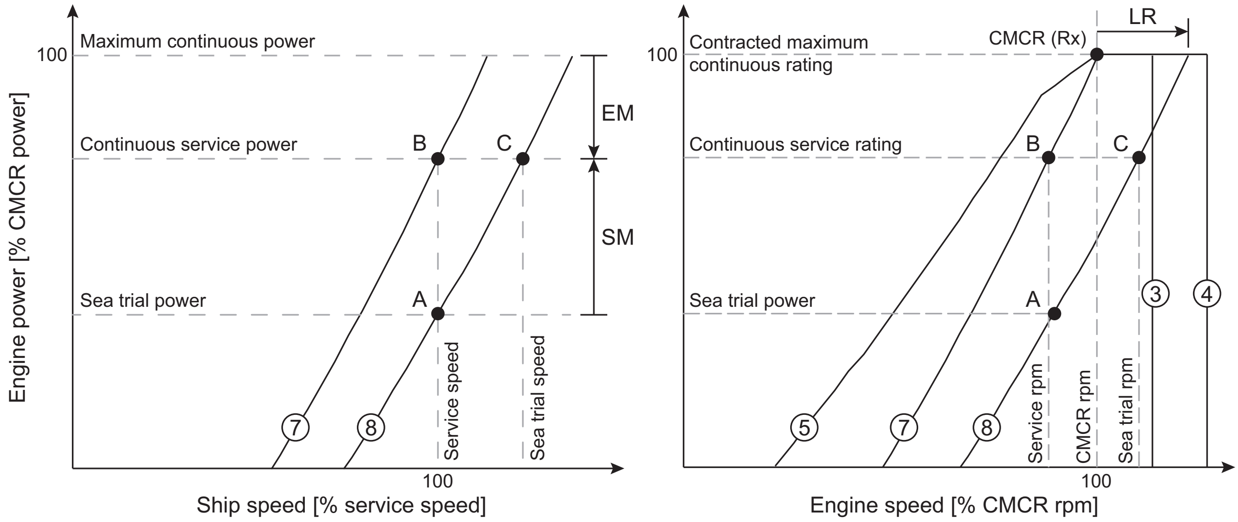
The relation between absorbed power and propeller speed for a fixed pitch propeller (FPP) can be approximated by the following cubic relation:

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

Formula 2-2

where:

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



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

SM-0026

Figure 2-2 Propeller curves and operational points

Figure 2-2 outlines the various engine limits, propeller curves and margins required for engine optimisation. By incorporating the margins listed below, the various operational points and subsequently the CMCR point can be determined. For detailed descriptions of the various line limits refer to section 2.2.5, 2-7.

Sea trial power

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

Sea margin (SM)

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

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

Light running margin (LR)

The light running margin ('LR' in Figure 2-2, 🗎 2-5) is the expected change in speed to relative power, caused by fouling and deterioration of the vessel over time. The light running margin is agreed between shipyard and ship owner, depending on hull and propeller cleaning interval and operation route (that will affect the rate of deterioration e.g. speed, location, shallow water, etc.).

Typically, the light running margin is specified in the range of 4 to 7%. However, additional power/engine speed allowance must be provided for shaft generator/ PTO installations (see section 2.2.6, 🗎 2-10).

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

Continuous service rating (CSR)

- Also known as NOR or NCR

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

Engine margin (EM)

Most owners specify the contractual ship's loaded service speed at 85 to 90% power of the contracted maximum continuous rating. Different selections are possible. The remaining e.g. 10 to 15% power can then be used to catch up with delays in schedule.
This margin is deducted from the CMCR. Therefore, the 100% power line is found by dividing the power at point B by the selected CSR power percentage, e.g. 85 to 90%. The graphic approach to find the level of CMCR is illustrated in Figure 2-2, 🗎 2-5.

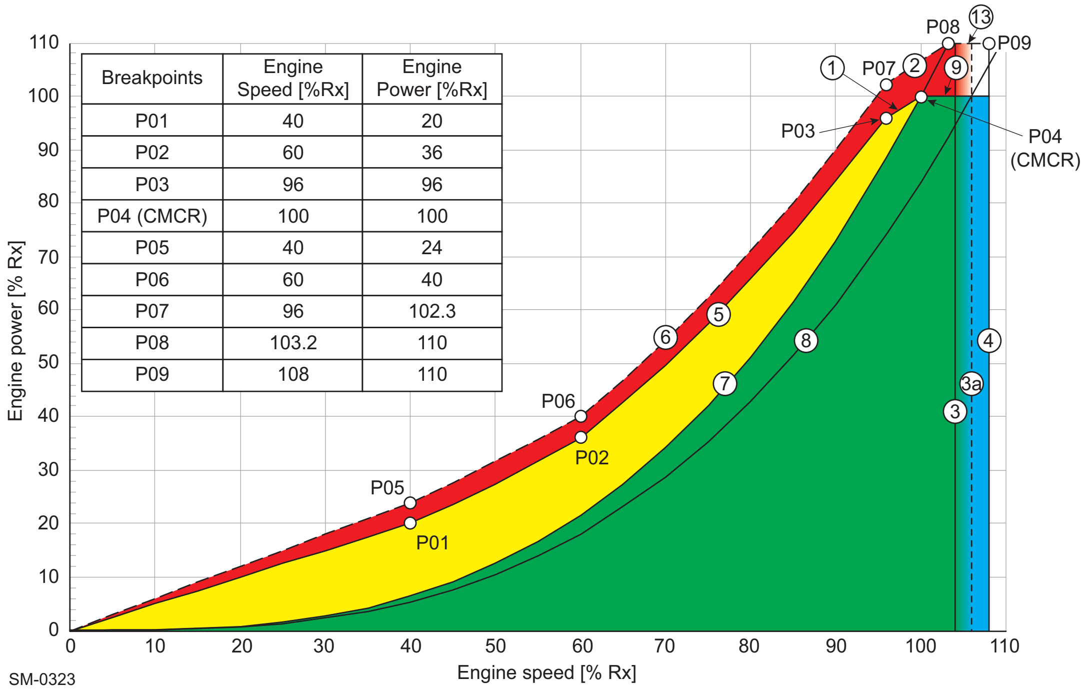
Contracted maximum continuous rating (CMCR)

- Also known as Rx or SMCR

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

2.2.5 Power range limits

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



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

Figure 2-3 Power range limits

Line 1
100% Torque Limit

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

Line 2
Overload Limit

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

Line 3
Speed Limit

Maximum speed limit for continuous engine running, set to 104% of CMCR speed. However, for a reduced Rx speed ($n_{CMCR} \leq 0.98\, n_{MCR}$) this limit can be extended to 106% (Line 3a) provided that the specified torsional vibration limits are not exceeded.

Line 4
Overspeed Limit

The overspeed range between Line 3 (104% or 106%) and 108% speed is only permissible during sea trials if needed for demonstration. This may only be carried out in the presence of authorised engine builder representatives, with the ship's speed at CMCR power and a light running propeller. However, the specified torsional vibration limits must not be exceeded.

Line 5
Engine Operation Power Limit

Admissible power limit for engine operation. The line is separated by the breakpoints listed in Figure 2-3.

Line 6 Transient Operation Power Limit

Maximum power limit for transient operation. The line is separated by the breakpoints listed in Figure 2-3, 2-7.

Line 7 Nominal Engine Characteristic

Nominal engine characteristic curve that passes through the CMCR point is defined by the propeller law:

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

Formula 2-3

Line 8 Light Running Propeller Curve

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

$$\frac{P_{LR}}{P_{CMCR}} = C \cdot \left(\frac{n}{n_{CMCR}}\right)^3 \qquad C = \left(\frac{1}{1 + LR}\right)^3$$

Formula 2-4

where:

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

Line 9 CMCR Power

Maximum power for continuous operation.

Line 13 110% CMCR Power

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

Engine Operation Power Range

Line 5, Line 1 and Line 9 form the curve for the engine's operation power range limit, as defined by Formula 2-5, 2-9. Each component is governed by different coefficients, as seen in Table 2-1.

Table 2-1 Line 5 coefficients

Line no.	Range (n/n_{CMCR})	C2	C1	C0
Line 5	0.00 - 0.40	0.000	0.500	0.000
	0.40 - 0.60	0.500	0.300	0.000
	0.60 - 0.96	1.111	-0.067	0.000
Line 1	0.96 - 1.00	0.000	1.000	0.000
Line 9	1.00 - 1.08	0.000	0.000	1.000

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

Formula 2-5

where:

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

The area formed by Lines 1, 3, 5 and 9 is the range within which the engine should be operated.

The area limited by Line 7, Line 9 and Line 3 is recommended for continuous operation.

The area between Line 7 and Line 5 is reserved for acceleration, shallow water and normal operational flexibility. If a main-engine driven generator (PTO) is installed, then the operating characteristics of the engine will differ. Refer to section 2.2.6, 2-10 for further details regarding PTO characteristics.

Overload Power Range

Line 6, Line 2 and Line 13 form the curve for the engine's overload power limit, as defined by Formula 2-5. Each component is governed by different coefficients as seen in Table 2-2.

Table 2-2 Line 6 coefficients

Line no.	Range (n/n_{CMCR})	C2	C1	C0
Line 6	0.00 - 0.40	0.000	0.600	0.000
	0.40 - 0.60	0.330	0.468	0.000
	0.60 - 0.96	1.110	0.000	0.000
Line 2	0.96 - 1.032	0.000	1.066	0.000
Line 13	1.032 - 1.08	0.000	0.000	1.100

The area above Lines 1 and 9 is the overload range. It is only allowed to operate engines in that range for a maximum duration of one hour in diesel mode, during sea trials in the presence of authorised representatives of the engine builder.
The area between Lines 1, 5 and 6 (Figure 2-3, 2-7), called 'service range with operational time limit', is only applicable to transient conditions in diesel mode, i.e. sea trial or during emergency fast acceleration. The engine can only be operated in this area for limited periods of time, in particular 1 hour per 24 hours.

2.2.6 Power range limits with main-engine driven generator for FPP

The addition of a main-engine driven generator (PTO) alters the working range and operating characteristics of the engine. Two methods of incorporating the PTO are outlined in the following sections. WinGD recommend to follow Method 1.

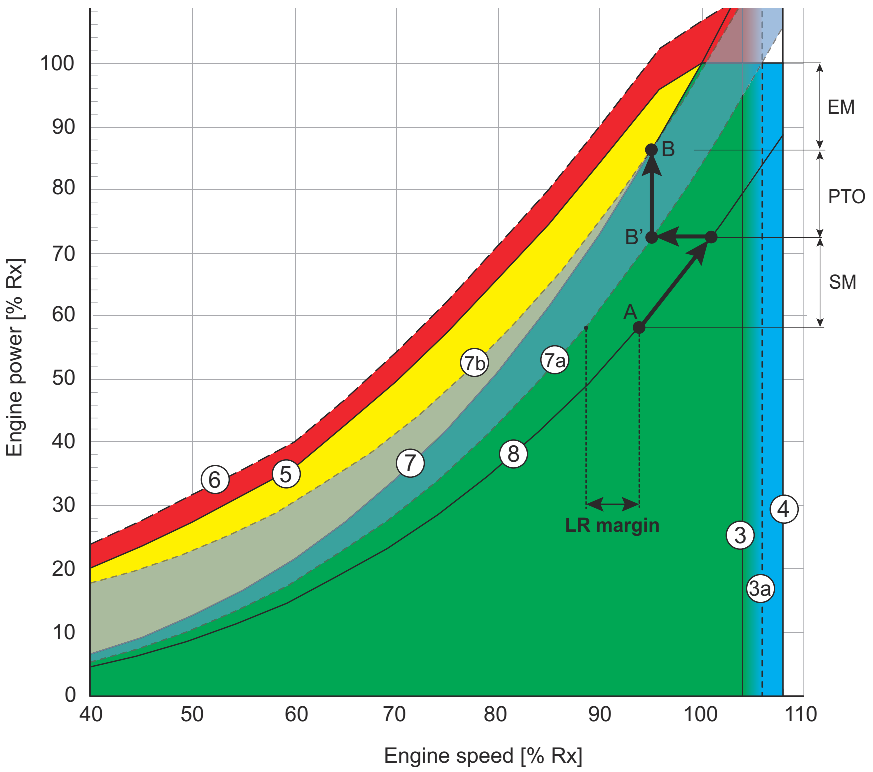
PTO considerations

- The PTO is used for generating the navigation electric power.
- The PTO is connected with frequency converter system.
- The PTO is not engaged in heavy sea conditions.
- Mechanical power absorption of the PTO must be considered.

PTO incorporation of Method 1

CMCR - Method 1

This first method considers the PTO as an addition to the previously defined propeller power requirements, therefore increasing the CMCR of the engine.



(Line 7) = Nominal engine characteristic
(Line 7a) = Nominal propeller characteristic without PTO
(Line 7b) = Nominal propeller characteristic with PTO
Point A = Light running in ideal sea conditions (sea trial)
Point B = CSR
Point B' = Continuous service without PTO power

SM-0029

Figure 2-4 Power range diagram of an engine with main-engine driven generator

Line 7a in Figure 2-4 shows the power needed for the propeller, where Line 7b shows the power needed for the propeller combined with a constant power requirement from the main-engine driven generator, the PTO.

With the addition of a constant nominal generator power across the engine load range, the engine curve is optimised, so no longer directly related to a propeller characteristic. In the example of Figure 2-4, 2-10, the main-engine driven generator is assumed to absorb 15% of nominal engine power. The CSR point includes the PTO power demand, this is shown in Figure 2-4, 2-10 between points B' and B. The CMCR point is selected from this propeller curve. This curve defines the nominal engine characteristic.

This approach allows for practical and flexible PTO operation, only limited in the lower engine speed range by the PTO required minimum speed (as defined by the PTO device supplier) and the PTO layout limit Line 10 (only relevant if a significant percentage of the installed engine power is utilised for PTO).

Line 10
PTO Layout Limit

The PTO layout limit line (Line 10 in Figure 2-5) defines the power limit for the resulting combination of the propeller and PTO.
Considering Line 10 as PTO layout limit provides a margin for normal power load fluctuation and acceleration.

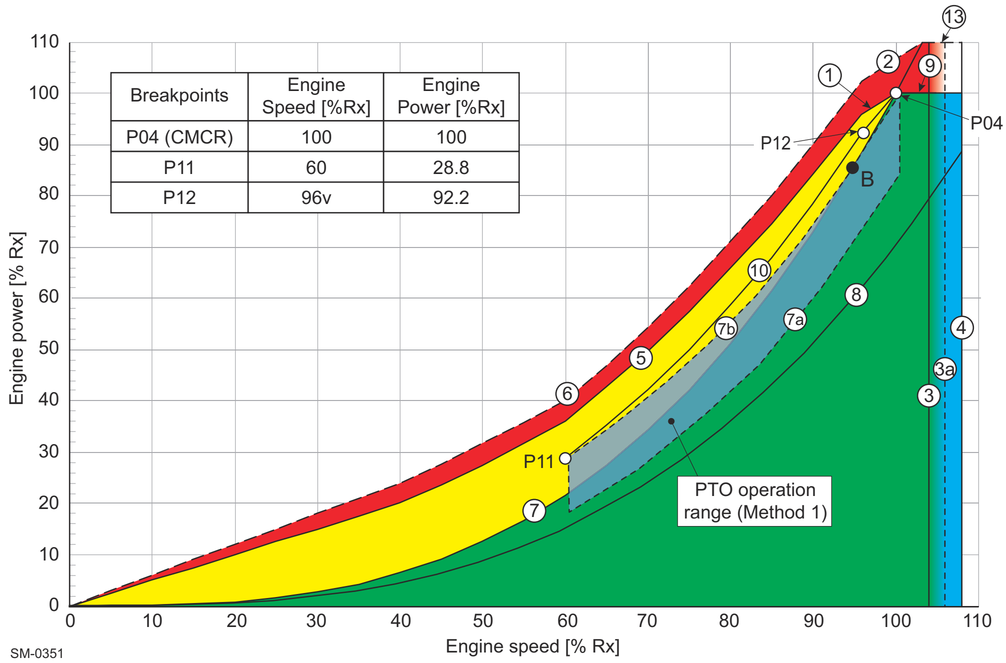


Figure 2-5 Power range limits for PTO operation — Method 1

The breakpoints of Line 10 are listed in Figure 2-5. Line 10 is a curve defined by Formula 2-5, 2-9. The different components have a different coefficient, forming the entire curve. Refer to Table 2-3 for the individual coefficients.

Table 2-3 Line 10 coefficients

Line no.	Range (n/n_{CMCR})	C2	C1	C0
Line 10	0.60 - 0.96	1.336	-0.321	0.000
	0.96 - 1.00	0.000	1.941	-0.941

PTO incorporation of Method 2

CMCR - Method 2

With this second method, the engine's CMCR is determined by the propeller power only. The PTO uses the available engine power that isn't absorbed by the propeller.

As seen in Figure 2-6, the available power for PTO application is between Line 8 and Line 10. Therefore, the available PTO power depends on the available Light Running margin at the current time.

When considering this method, a LR margin in the range of 8% is recommended.

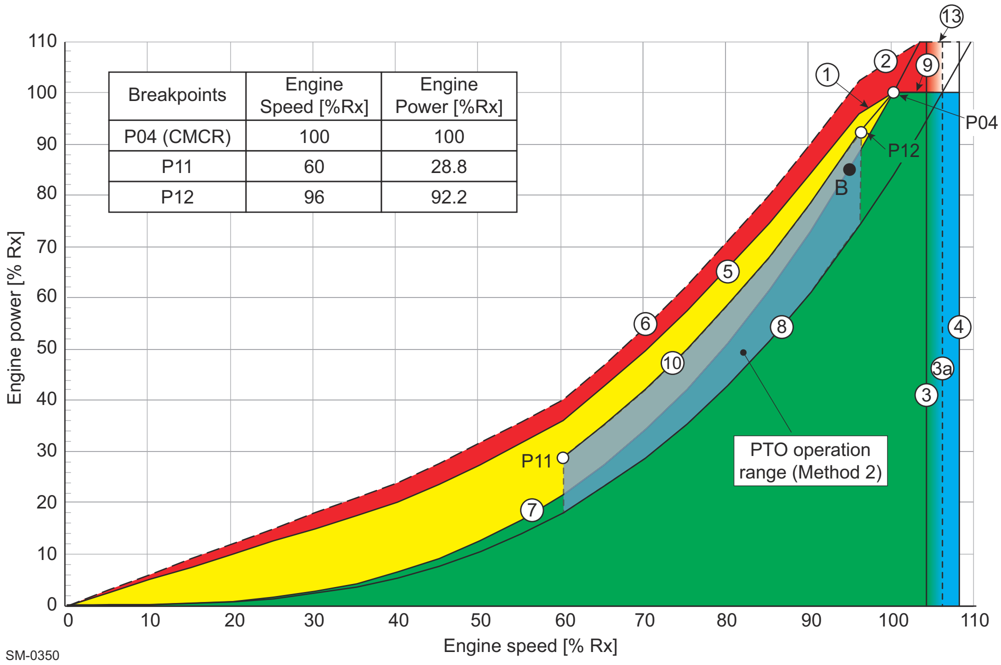


Figure 2-6 Power range limits for PTO operation — Method 2

The PTO power must be controlled by the Ship Power Management System, which ensures that the engine operating point will not exceed Line 10.

Further information

The following disadvantages must be observed for Method 2:

- With the reduction of the LR margin (as a consequence of the ageing hull and propeller) the available PTO power will be reduced and must be limited by the Ship Power Management System.
- The PTO is typically engaged between approx. 60% - 96.5% of engine speed. Final lower limit must be defined with the supplier of the generator. Final upper limit must be set to the project related CSR engine speed.
- Operation above the engine characteristic Line 7 can lead to increased DCC activation.

If the requested PTO power is higher than the limits defined by Method 2, an increase in CMCR power is necessary.

2.2.7 Prohibited operation area

Within the higher speed range of the engine there is a prohibited operation area defined by a minimum engine power requirement. During normal operation, including CPP at zero pitch operation, the engine will not enter this prohibited area. However, if the propeller is disconnected from the engine it would be capable of entering the prohibited zone; this is strictly forbidden.

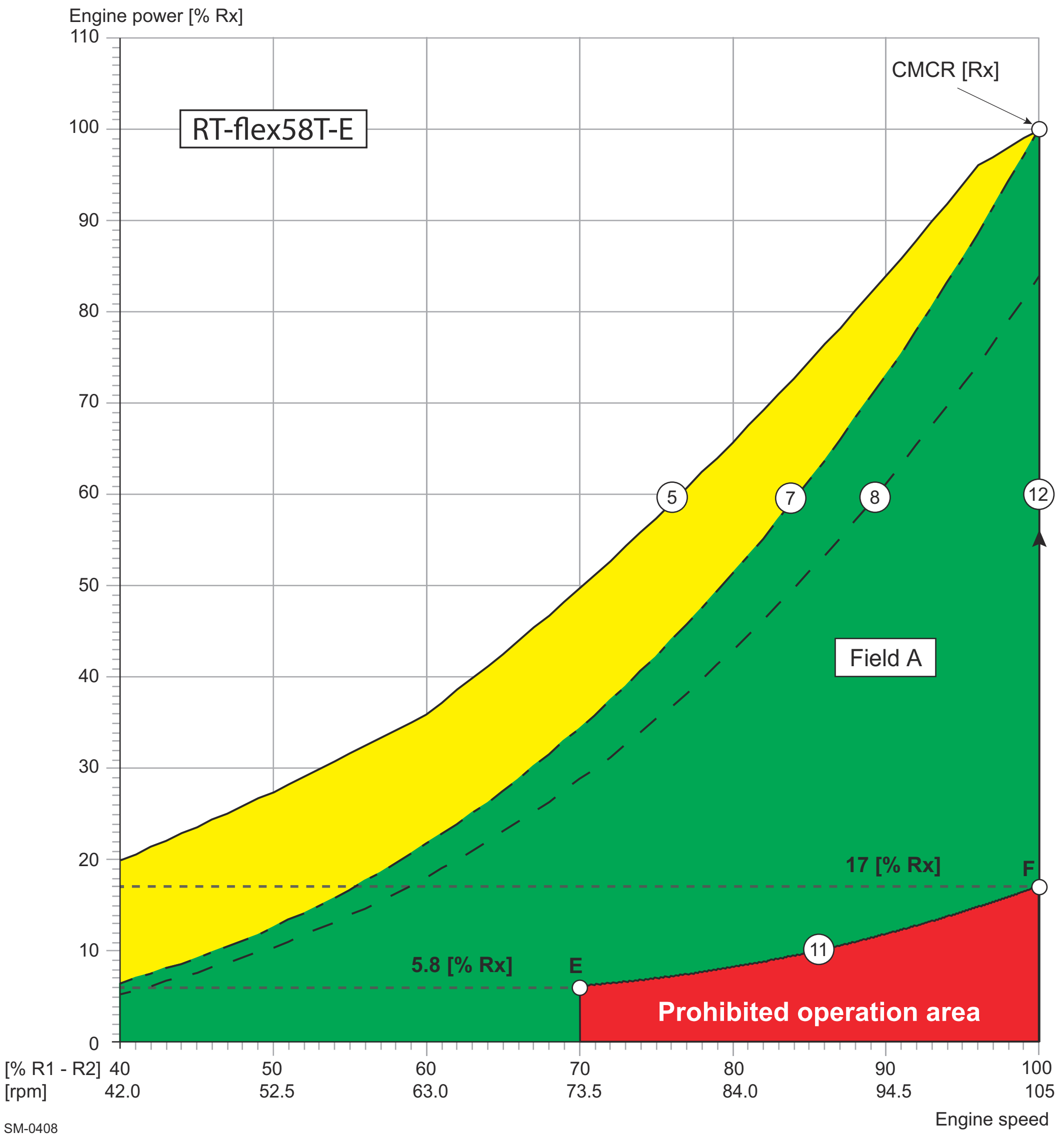


Figure 2-7 Prohibited engine operation area (CMCR speed = R1—R2)

As seen in Figure 2-7, 🗎 2-13, the prohibited operation area of an engine is defined by an engine tuned to an R1—R2 speed. At this speed rating the restriction exists between 70% and 100% speed, with a required minimum engine power at these points of 5.8% and 17% respectively. These values are governed by the Formula 2-6 for Line 11.

If the CMCR speed rating of the engine is less than the R1—R2 speed, the required minimum power at this point is also calculated by the Line 11 equation.

Line 11 The lowest operational power limit, between 70% of R1—R2 speed and 100% CMCR speed, is defined with the following equation:

$$Line11 = 0.17 \bullet \left(\frac{n_{CMCR}}{n_{R1-R2}}\right)^3$$

Formula 2-6

As defined by this equation and shown in Figure 2-7, 2-13, at 70% of R1—R2 speed the minimum engine power equals approximately 6% of CMCR power (point E). The minimum power requirement at 100% CMCR speed (point F) must be calculated depending on the engine rating. Examples of this calculation are shown further on.

Line 12 While operating at 100% CMCR speed, the allowed engine power can range between the minimum required power (point F) and 100% CMCR power.

Field A The available design range of the engine is defined by Line 7, Line 11 and Line 12.
For test purposes (e.g. for shaft generator adjustment), the engine may run at low load, within the prohibited area. This is for a maximum time period of 30 minutes during testing and sea trials, and only in the presence of authorised representatives of the engine builder. Further requests must be agreed by WinGD.

NOTE	The operational design range must respect the BSR limits from torsional vibration.

Prohibited operation area for different speed rated engines

As the prohibited operating area of the engine is between 70% and 100% of the R1—R2 speed, the prohibited area is smaller when the speed rating of the engine is lowered.

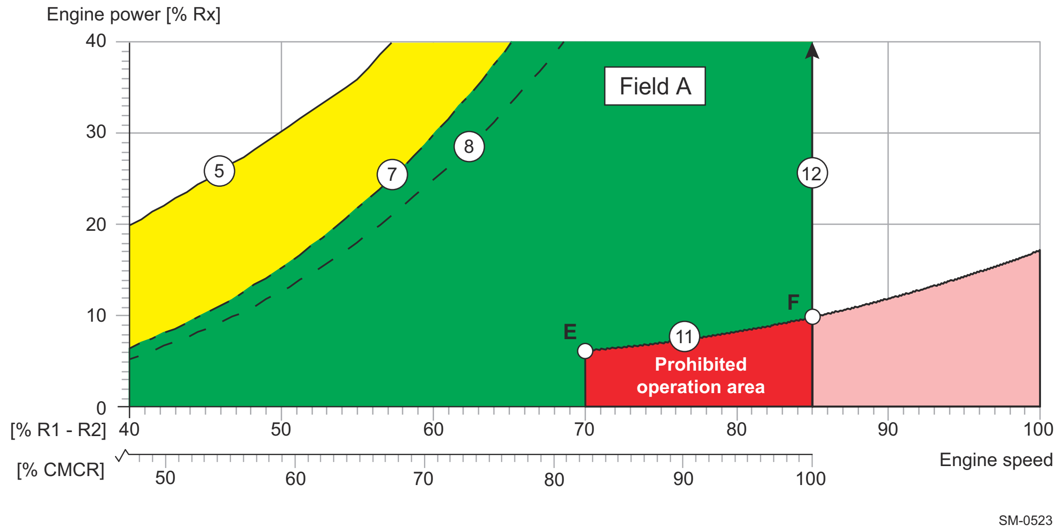


Figure 2-8 Calculating prohibited operation area for CMCR speed

Figure 2-8 shows an engine with a CMCR rated at 85% of R1—R2 speed and therefore is only affected by a portion of the prohibited area of the R1—R2 speed range. The final graph for a CMCR at this speed is shown in Figure 2-9, 2-16.

Power / speed range for CMCR [Rx] = 85% R1—R2 speed

In Figure 2-9, the engine's CMCR speed is rated at 85% of the R1—R2 speed. At this speed, a minimum engine power (point F) of 10.4% is required. Below this is the prohibited area of operation for a disconnected propeller.

Point E is always defined at 70% of the R1—R2 speed and has a minimum power of 5.8%, however in Figure 2-9 this equates to approximately 82% CMCR speed.

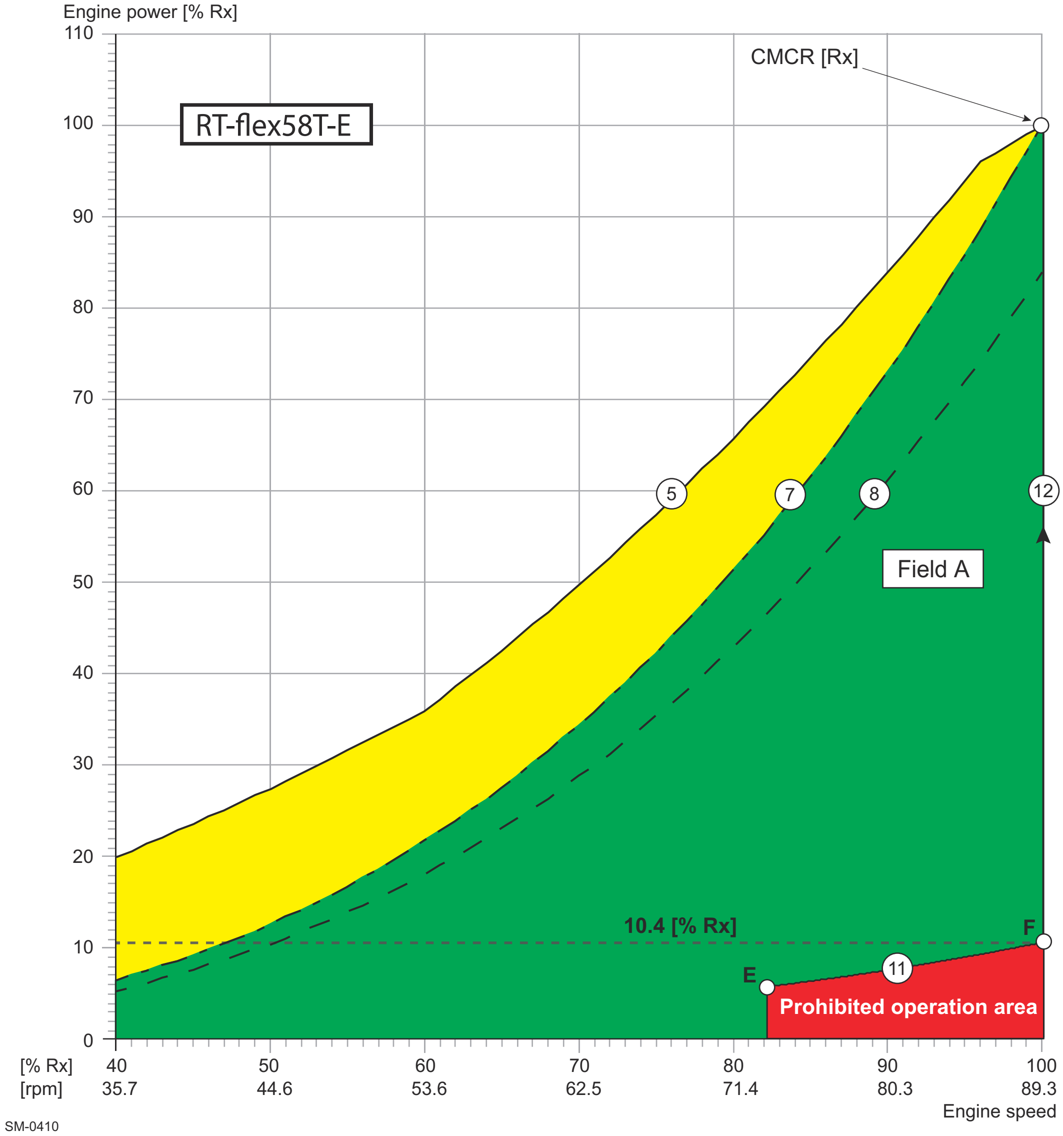


Figure 2-9 Prohibited engine operation area (CMCR speed = 85% of R1—R2)

Power / speed range for CMCR [Rx] = R3—R4 speed

In Figure 2-10, the engine's CMCR speed is rated at the R3—R4 speed. At this speed, a minimum engine power (point F) of 10.7% is required. Below this is the prohibited area of operation for a disconnected propeller.

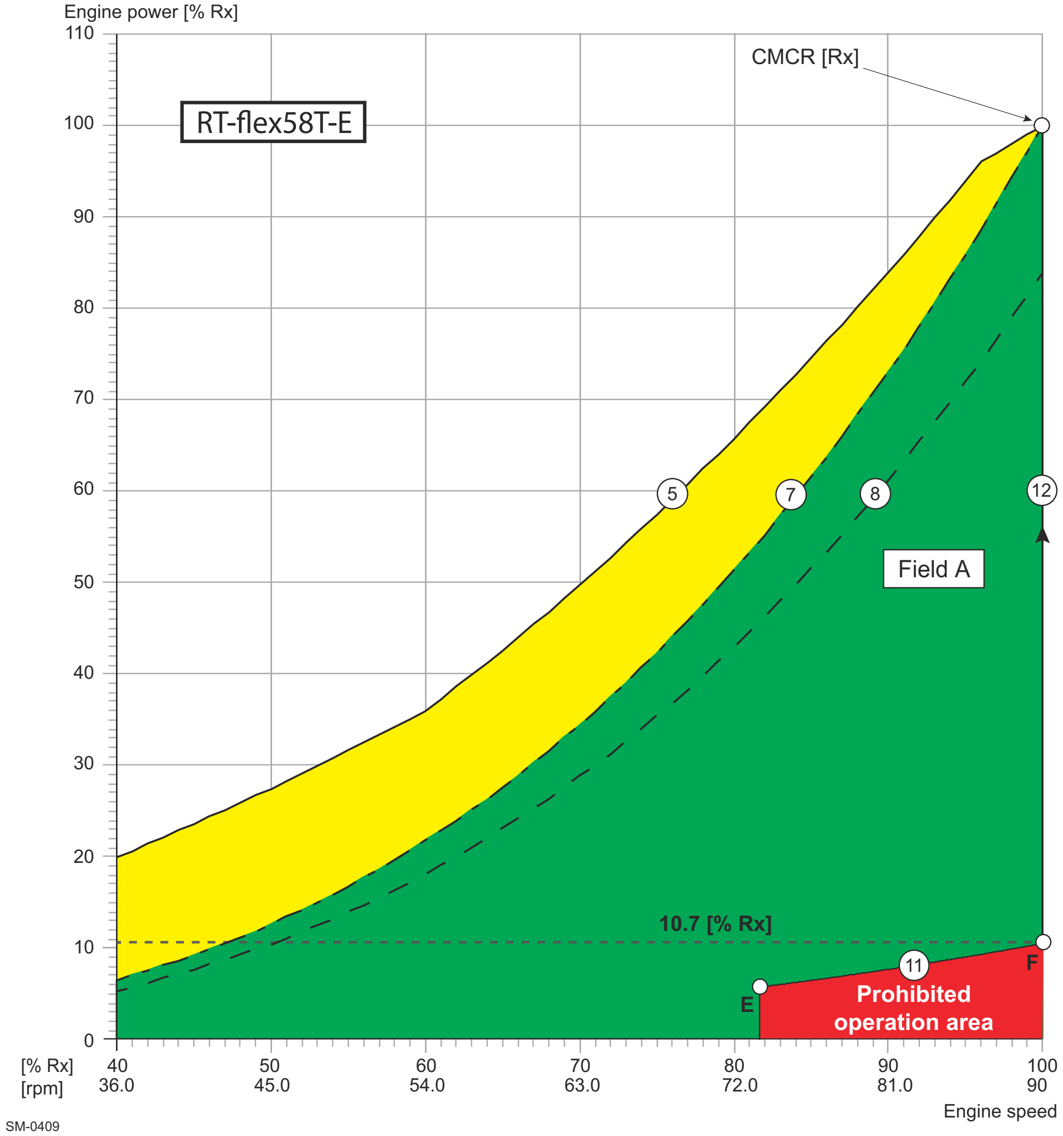


Figure 2-10 Prohibited engine operation area (CMCR speed = R3—R4)

2.2.8 CPP requirements for Propulsion Control System

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

The following operating modes shall be included in the Propulsion Control System:

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

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

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

Engine must not be operated continuously at a pitch lower than xx % at any speed above 73 rpm.

The pitch value (xx) is to be defined according to installation data.

For additional information about CPP application in the Propulsion Control System, refer to section 5.4.2 Recommended manoeuvring characteristics, 5-9.

Operation in prohibited area

In addition, if the engine is operated for more than 3 minutes in the prohibited operation area, an alarm must be provided in either the ME safety system or the vessel's Alarm and Monitoring System.
If the engine is operated for more than 5 minutes in the prohibited operation area, then the speed must be reduced below 73 rpm.

2.3 Operating conditions

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

2.3.1 Reference conditions

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

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

2.3.2 Design conditions

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

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

2.4 Ancillary system design parameters

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

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

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

2.5 Electrical power requirement

Table 2-4 Electrical power requirement

No. cyl.	Power requirement [kW]	Power supply
Auxiliary blowers [a]		
5	2 x 36	440 V / 60 Hz
6	2 x 46	
7	2 x 46	
8	2 x 58	
Turning gear		
5	3.7	440 V / 60 Hz
6	3.7	
7	3.7	
8	3.7	
Engine Control System		
5	1.2	230 V / 60 Hz
6	1.4	
7	1.7	
8	1.8	
Propulsion Control System		
all	acc. to maker's specifications	24 VDC UPS
Additional monitoring devices (e.g. oil mist detector, etc.)		
all	acc. to maker's specifications	

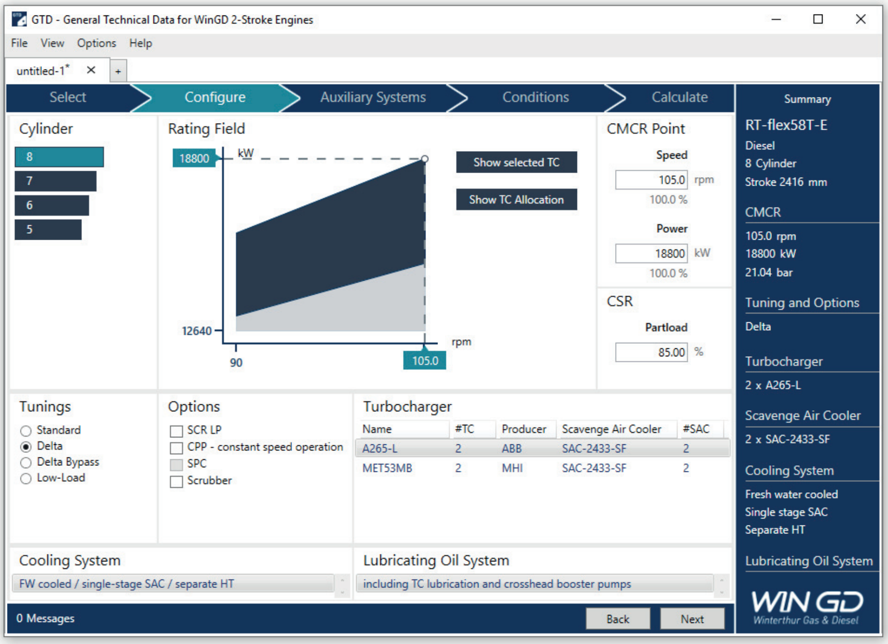
a) Minimal electric motor power (shaft) is indicated. Actual electric power requirement depends on size, type and voltage/frequency of installed electric motor. Direct starting or Star-Delta starting to be specified when ordering.

2.6 GTD - General Technical Data

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

Engine performance data

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



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GTD output

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

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

3 Engine Installation

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

3.1 Dimensions and masses

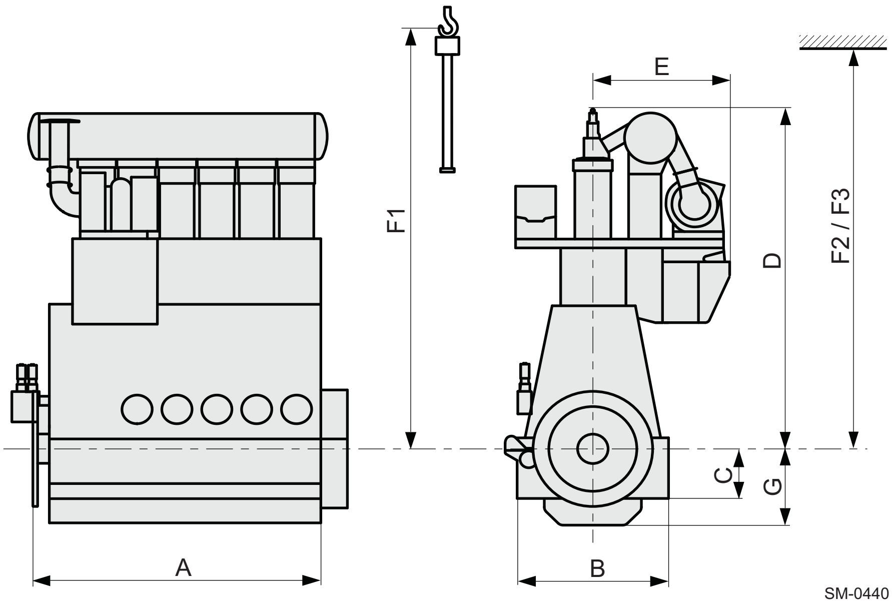


Figure 3-1 Engine dimensions

Table 3-1 Engine dimensions and masses

<table>
<tr><th rowspan="2">No. cyl.</th><th colspan="9">Dimension in mm with a tolerance of approx. ±10 mm</th><th>Net eng. mass a)</th></tr>
<tr><th>A</th><th>B</th><th>C</th><th>D</th><th>E</th><th>F1 b)</th><th>F2 c)</th><th>F3 d)</th><th>G</th><th>[tonnes]</th></tr>
<tr><td>5</td><td>6,381</td><td rowspan="4">3,820</td><td rowspan="4">1,300</td><td rowspan="4">8,971</td><td rowspan="4">Dim. depending on TC type</td><td rowspan="4">10,960</td><td rowspan="4">11,000</td><td rowspan="4">10,400</td><td rowspan="4">2,000</td><td>281</td></tr>
<tr><td>6</td><td>7,387</td><td>322</td></tr>
<tr><td>7</td><td>8,393</td><td>377</td></tr>
<tr><td>8</td><td>9,399</td><td>418</td></tr>
<tr><td colspan="11">Min. capacity of bridge crane: 4,350 kg
Min. capacity of double-jib crane: 2 x 2,325 kg e)</td></tr>
</table>

a) Without oil / water; net engine mass estimated according to nominal dimensions given in drawings, including turbocharger and SAC, piping and platforms
b) Min. height for vertical removal of piston with standard crane
c) Min. height for vertical removal of piston with double-jib crane
d) Min. height for tilted removal of piston with double-jib crane
e) In cases of double-jib crane application, both hooks are used in parallel; special lifting tools are required
When selecting the double-jib lifting method, it must be considered that maintenance work will demand additional time and effort, especially for tilted removal (F3), compared to standard procedure (F1). Availability of the special lifting tools needs to be considered in the project schedule.

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

Project-specific dimensions and masses for main components to be requested from engine builder.

3.1.1 Dismantling heights for piston and cylinder liner

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

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

3.1.2 Crane requirements

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

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

3.1.3 Thermal expansion at turbocharger expansion joints

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

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

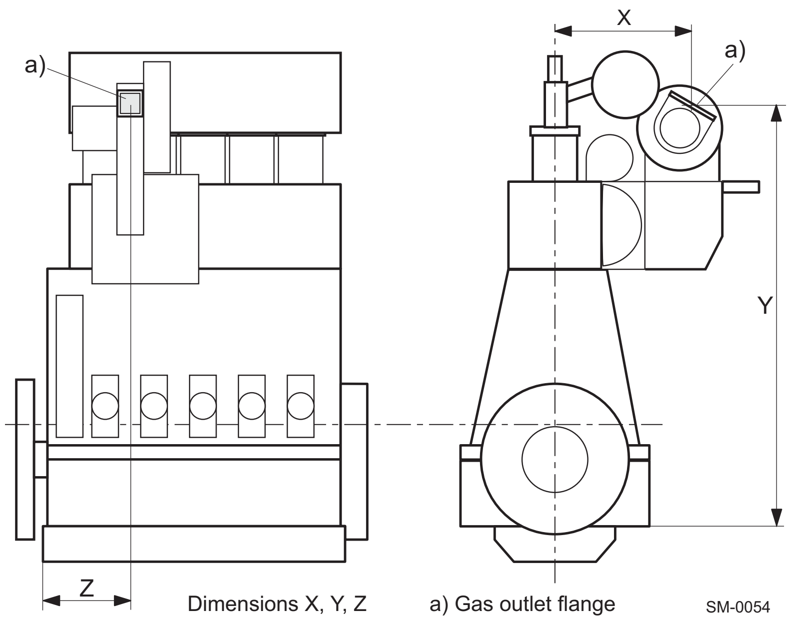


Figure 3-2 Thermal expansion, dim. X, Y, Z

Calculating thermal expansion

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

where:

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

3.1.4 Content of fluids in the engine

Table 3-2 Fluid quantities in the engine

No. of cyl.	Lubricating oil	Fuel oil	Cylinder cooling water	Freshwater in SAC [a]
	[kg]	[kg]	[kg]	[kg]
5	1,300	10	675	410
6	1,500	10	800	410 [b] / 510 [c]
7	1,700	10	925	540 [b] / 510 [c]
8	1,900	10	1,050	710

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

3.2 Engine outline views

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

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

3.3 Platform arrangement

3.3.1 Drawings

For platform arrangement see the links given in section 3.2, 3-5.

3.3.2 Minimum requirements for escape routes

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

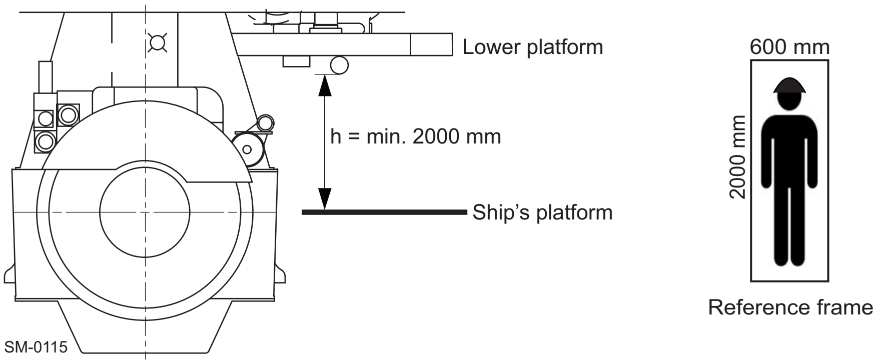


Figure 3-3 Minimum requirements for headroom

Important!

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

See also the links to drawings in section 3.2, 3-5.

3.4 Seating

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

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

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

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

3.5 Assembly

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

3.5.1 Assembly of subassemblies

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

For checking the dimensions optical devices or lasers may be used

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

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

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

NOTE	The process of using jacking screws and wedges is defined in *MIDS* and must be followed.

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

3.5.2 Installation of a complete engine

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

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

Please observe:

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

3.5.3 Installation of an engine from assembled subassemblies

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

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

The placing on blocks and alignment to shafting is analogue to that described in section 3.5.1, 3-8.

3.5.4 Installation of an engine in ship on slipway

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

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

3.6 Engine and shaft alignment

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

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

3.6.1 Instructions and limits

Alignment can be done with either jacking screws or wedges.

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

3.6.2 Tools

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

3.7 Engine coupling

3.7.1 Design

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

3.7.2 Machining and fitting of coupling bolts

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

3.7.3 Tightening

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

3.7.4 Installation drawing

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

3.8 Engine stays

Ship vibrations and engine rocking caused by the engine behaviour (see chapter 6 Engine Dynamics, 🗎 6-1) are reduced by fitting lateral stays (refer to section 6.2 External lateral forces and moments, 🗎 6-5) and longitudinal stays (refer to 6.3 Longitudinal vibration (pitching), 🗎 6-9).

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

3.9 Propulsion shaft earthing

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

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

3.9.1 Preventive action

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

Isolation of instrument wiring

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

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

3.9.2 Earthing device

Figure 3-4, 🗎 3-14 shows a typical shaft earthing device.
The slip ring (1) is supplied as matched halves to suit the shaft, and secured by two tension bands (2) using clamps (12). The slip ring mating faces are finished flush and butt jointed with solder. The brushes (4) are housed in the twin holder (3) clamped to a stainless steel spindle (6), and there is a monitoring brush (11) in a single holder (10) clamped to an insulated spindle (9). Both spindles are attached to the mounting bracket (8).

Conducting material for slip rings

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

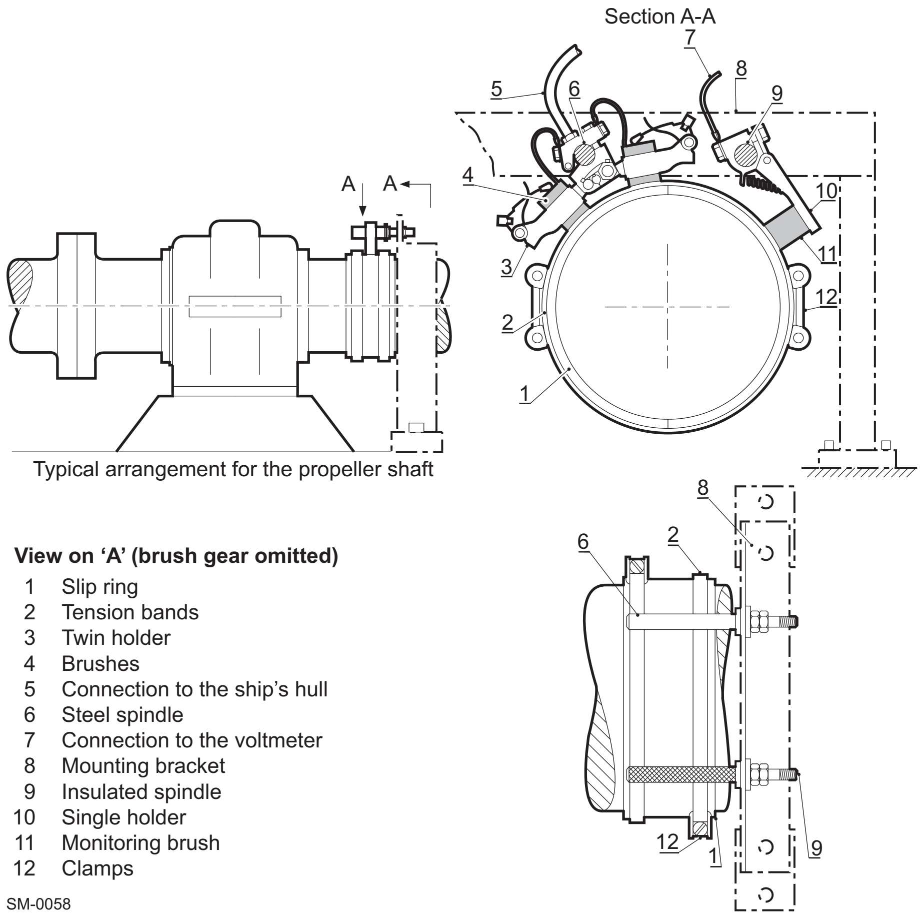


Figure 3-4 Typical shaft earthing arrangement

Position of earthing device on shaft

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

NOTE	For detailed information please contact the earthing device supplier.

Connecting electric cables

The electric cables are connected as shown in Figure 3-5, 🗎 3-15 with the optional voltmeter. This instrument is at the discretion of the owner, but it is useful to observe that the potential to earth does not rise above 100 mV.

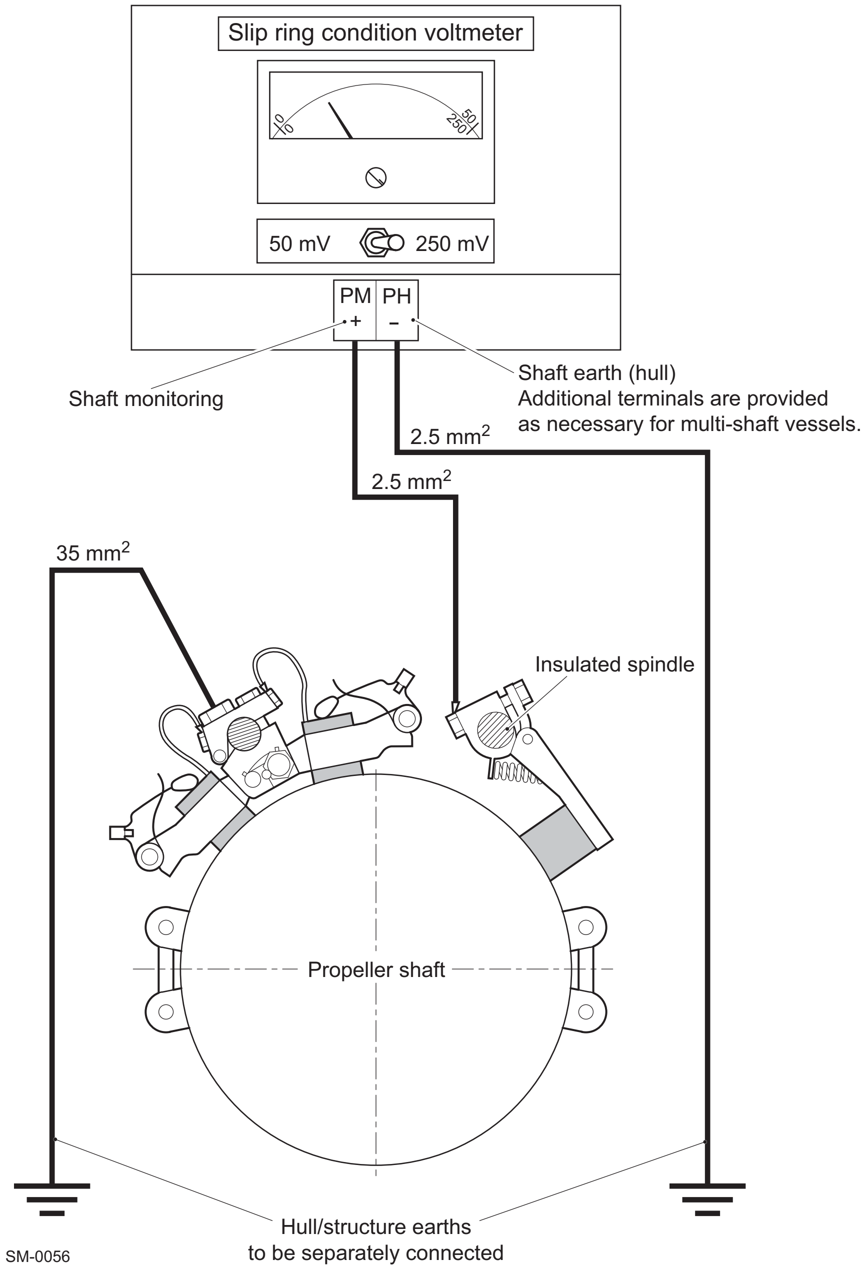


Figure 3-5 Typical shaft earthing with condition monitoring facility

3.10 Fire protection

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

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

Extinguishing agents

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

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

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

Table 3-3 Recommended quantities of fire extinguishing medium

Piston underside and scavenge air receiver		Bottle		Number of cylinders			
				5	6	7	8
Volume [m^3/cyl]	Mass [kg/cyl]	Size [kg]	Extinguishing medium	Quantity of fire extinguishing bottles			
3.5	13	45	Carbon dioxide (CO_2)	1	2	2	2

4 Ancillary Systems

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

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

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

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

4.1 Twin-engine installation

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

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

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

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

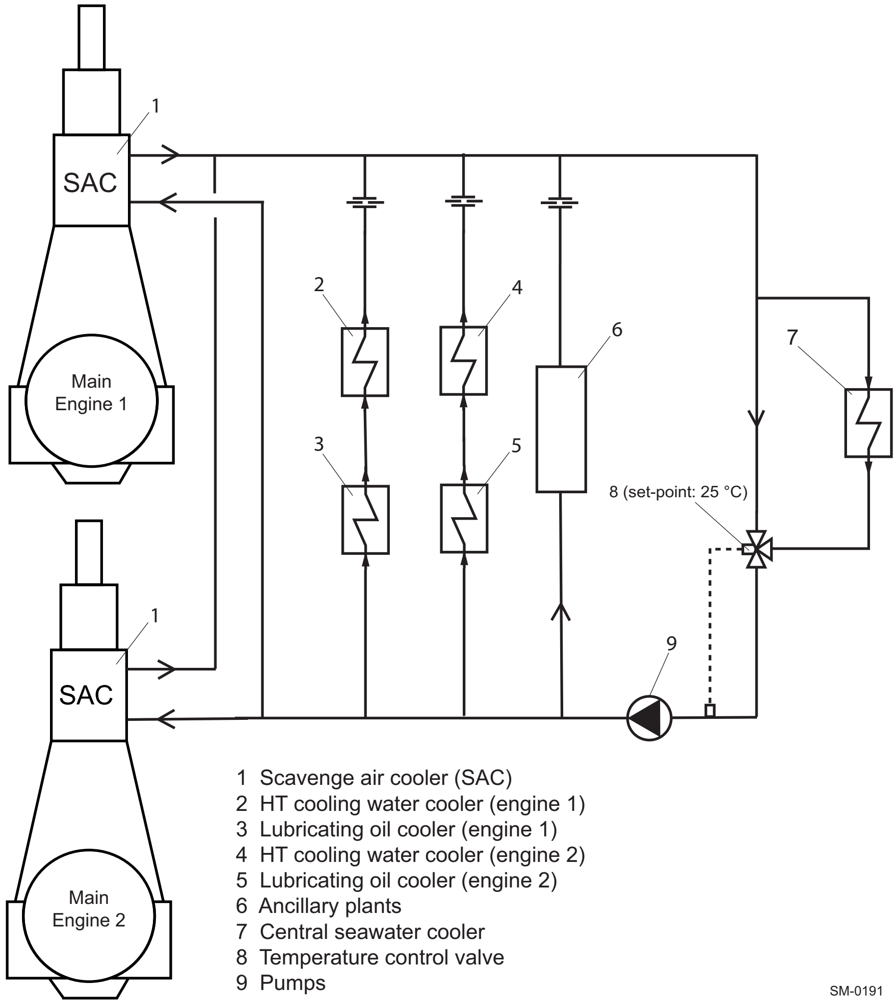
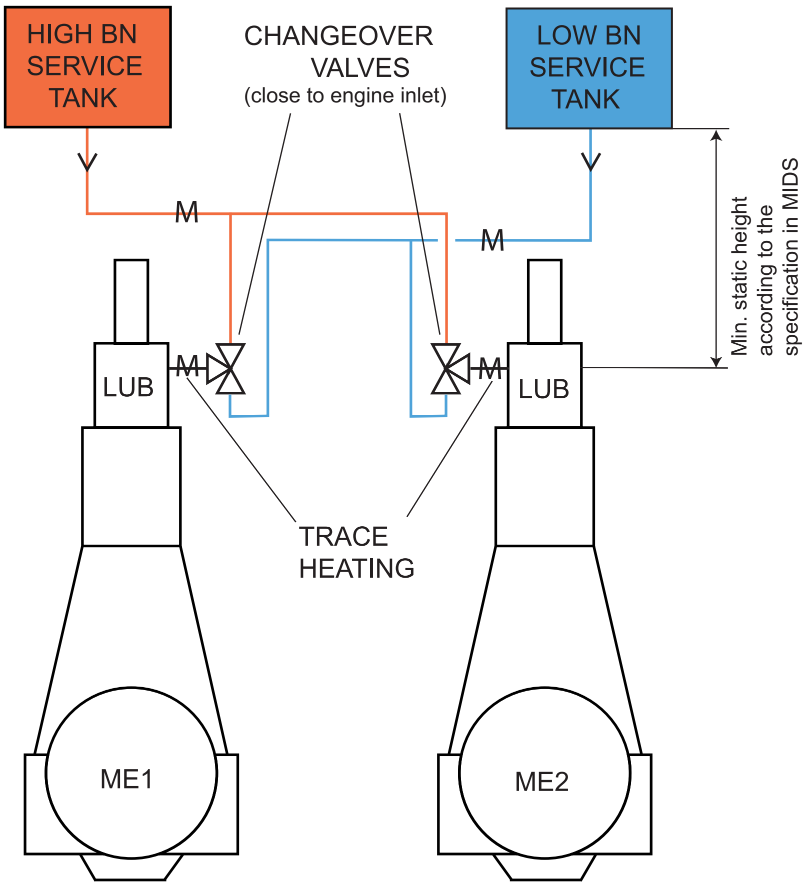


Figure 4-1 LT cooling water system for twin-engine installation



ME – Main engine
LUB – Lubricator

SM-0193

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

4.2 Cooling water system

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

Freshwater cooling system

The main engine high-temperature (HT) and low-temperature (LT) cooling circuits use freshwater as their cooling medium. As such, the HT and LT circuits are integrated in the ship's central freshwater cooling system.

Advantage of freshwater over seawater

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

Figure 4-3 shows the general installation principle.

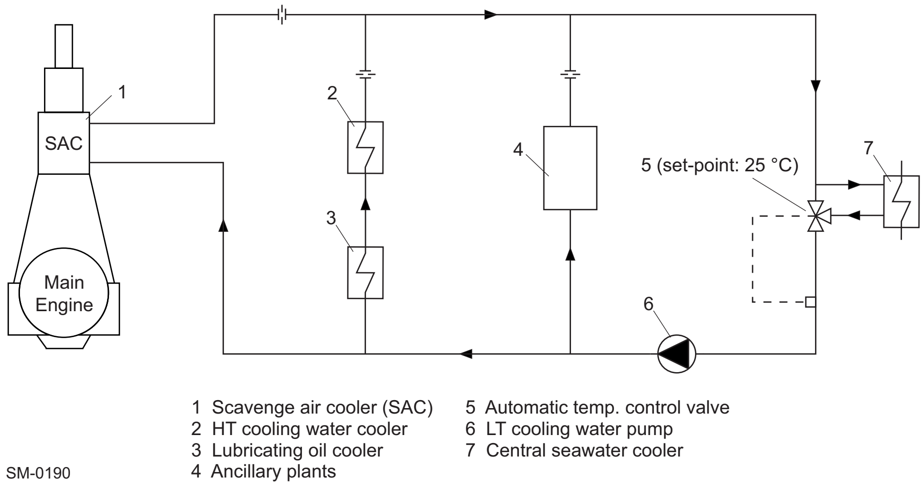


Figure 4-3 Scheme of cooling water system

Two standard layouts

The central freshwater cooling system for RT-flex58T-E runs on either of the following two standard layouts:

Layout **A**: with single-stage scavenge air cooler and **separate** HT circuit
Layout **B**: with single-stage scavenge air cooler and **integrated** HT circuit

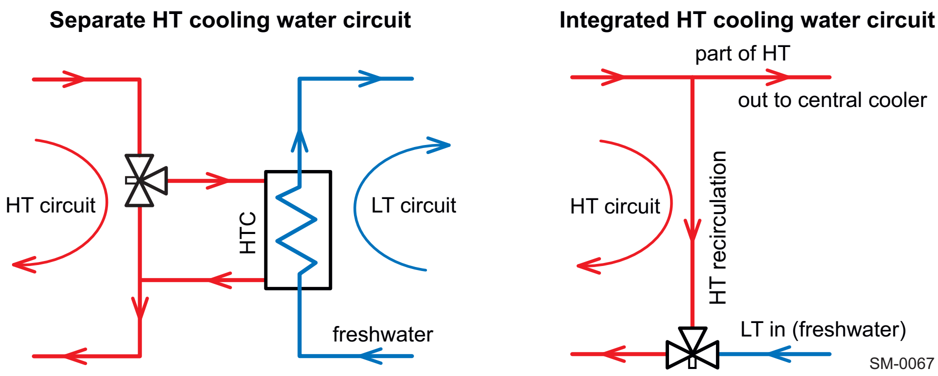


Figure 4-4 Principal layouts of HT cooling water circuit

Separate HT circuit with own cooler

In the case of layout **A** the HT circuit must be completely separate from the LT circuit. A dedicated HT water cooler is applied for heat exchange between HT and LT circuit without flow medium mixing.

GTD

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

4.2.1 Central freshwater cooling system components

Low-temperature circuit

Seawater circulating pump

The seawater circulating pump delivers seawater from the high and low sea chests to the central seawater cooler.

Pump type	Centrifugal
Capacity	According to *GTD*: The seawater flow capacity covers the need of the engine only and is to be within a tolerance of 0 to +10% of the GTD value
Delivery pressure	Determined by system layout
Working temperature	According to ship specification

Seawater strainer

Simplex or duplex strainers to be fitted at each sea chest and arranged to enable manual cleaning without interrupting the flow. The strainer mesh size (max. 6mm) must prevent the passage of large particles and debris that could damage the pumps and impair the heat transfer across the coolers.

Central seawater cooler

Cooler type	Plate or tubular
Cooling medium	Seawater
Cooled medium	Freshwater
Design criterion	Keeping max. 36°C LT while seawater temp. is 32°C
Margin for fouling	10-15% to be added
Heat dissipation	Refer to *GTD*
Freshwater flow	
Seawater flow	
Temperatures	

Automatic temperature control valve

The central freshwater cooling system is to be capable of maintaining the inlet temperature to the scavenge air cooler(s) between 10 and 36 °C. WinGD recommend that the controller is set to 25 °C (set-point) as this has a positive influence on the engine's performance.

Valve type	Electrically or electro-pneumatically actuated three-way type (butterfly valves are not adequate) having a linear characteristic
Design pressure	5 bar
Test pressure	Refer to specification laid down by classification society
Press. drop across valve	Max. 0.5 bar
Controller	Proportional plus integral (PI)
Temperature sensor	According to control valve manufacturer's specification; fitted in engine outlet pipe

Freshwater pumps

Pump type	Centrifugal
Capacity	According to *GTD*: The freshwater flow capacity covers the need of the engine only and is to be within a tolerance of 0 to +10 % of the GTD value
Delivery head	The final delivery head is determined by the layout of the system and must ensure that the inlet pressure to the scavenge air cooler(s) is within the range of summarised data
Working temperature	According to ship specification

High-temperature circuit

Cooling water pump

Pump type	Centrifugal, preferably with a steep head curve[a)]
Pump capacity	According to *GTD*: The flow capacity is to be within a tolerance of -10 to +20% of the GTD value
Delivery head[b)]	Determined by system layout
Working temperature	95°C

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

b) The required pump delivery head (p_p) can be calculated as follows:

$$p_p \geq \Sigma\Delta p \geq p_0 - \frac{h}{10.2} + d_p \text{ [bar]}$$

where:
$\Sigma\Delta p$ = system pressure losses
p_0 = required pressure at engine inlet
d_p = pressure drop between pump inlet and engine inlet
$h/10.2$ = constant

Automatic temperature control valve

Valve type	Electrically or electro-pneumatically actuated three-way type (butterfly valves are not adequate) having a linear characteristic
Design pressure	5bar
Test pressure	Refer to specification laid down by classification society
Press. drop across valve	Max. 0.5bar
Controller	Proportional plus integral (PI), known as proportional plus reset for steady state error of max. ±2°C and transient condition error of max. ±4°C
Temperature sensor	According to control valve manufacturer's specification; fitted in engine outlet pipe

Expansion tank

To ensure that the required static head is applied to the cylinder cooling water (CCW) system, the expansion tank is to be fitted at least 3.5m above the highest point of the engine's cooling water piping. The tank is to be connected by a balance pipe to the CCW pump suction.

4.2.2 Cooling water treatment

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

Table 4-2 Recommended parameters for raw water

Parameter	Value
Min. pH	6.5
Max. dH	10° (corresponds to 180 mg/l $CaCO_3$) [a)]
Max. chloride	80 mg/l
Max. sulphates	150 mg/l

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

Corrosion inhibitors

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

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

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

4.2.3 General recommendations for design

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

4.2.4 Freshwater generator

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

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

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

4.2.5 Pre-heating

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

Pre-heating from cooling water systems

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

Pre-heating by direct water circulation

Use of main cylinder cooling water pump

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

Use of separate pre-heating pump

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

Recommended temperature

The recommended temperature to start and operate the engine is 60 °C at CCW outlet. If the engine is started below the recommended temperature, engine power must not 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 ship owner) on the basis of their own experience.

Warm-up time

The graph in Figure 4-5, 🗎 4-13 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.

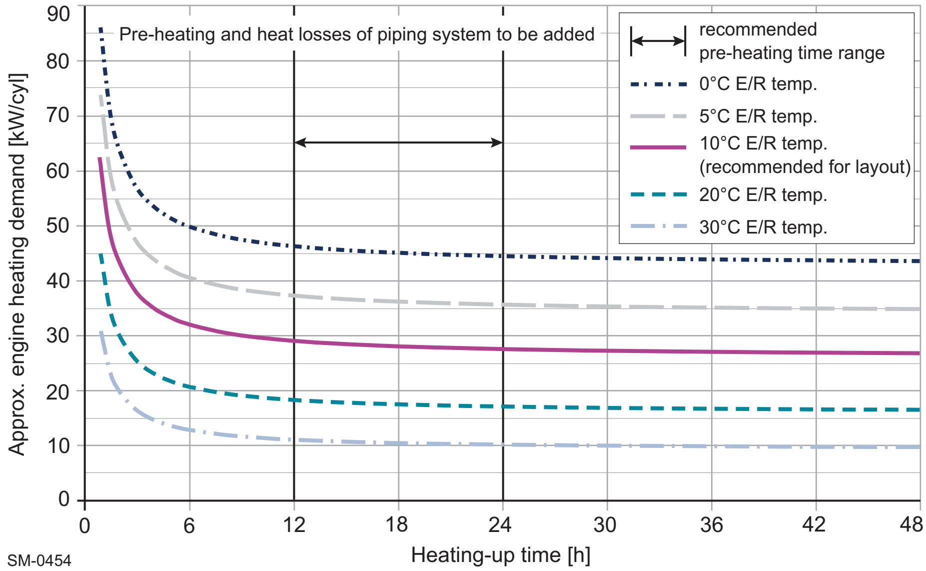


Figure 4-5 Pre-heating power requirement per cylinder

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

4.3 Lubricating oil systems

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

4.3.1 Lubricating oil requirements

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

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

4.3.2 Main lubricating oil system

Field of application

Lubrication of the main bearings, thrust bearings, bottom-end bearings, together with piston cooling, is carried out by the main lubricating oil system.
The main bearing oil is also used to cool the piston crown and to lubricate and cool the torsional and axial vibration dampers.
Figure 4-6 shows the general installation principle.

Lubrication of crosshead bearings

The crosshead bearings are lubricated by an additional crosshead pump (specification see Booster pump for crosshead lubrication, 4-16).

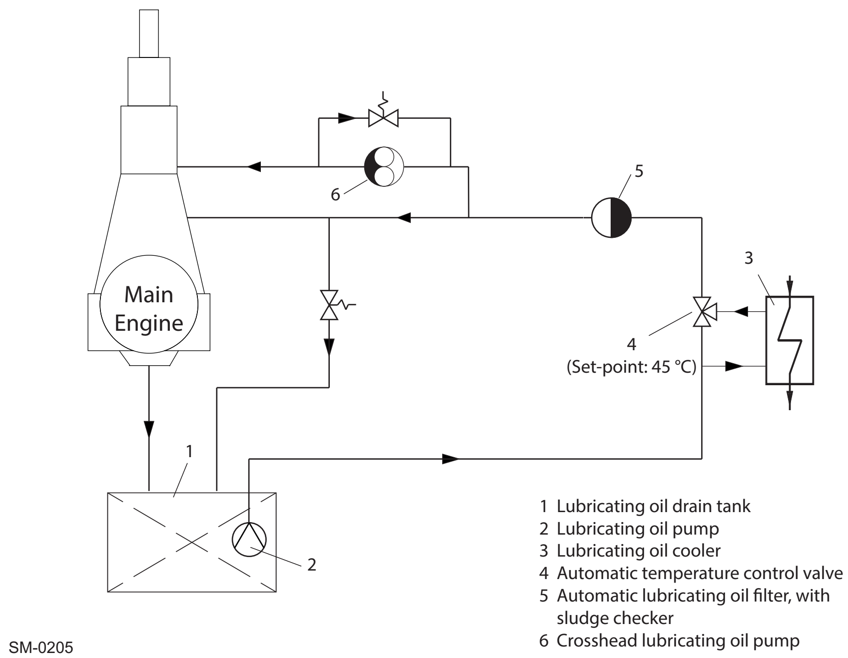


Figure 4-6 Lubricating oil system

Main lubricating oil system components

Lubricating oil pump

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

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

Lubricating oil cooler

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

Full-flow filter

The drain from the filter is to be sized and fitted to allow free flow into the lubricating oil drain tank.
The output required for the main lubricating oil pump to 'back-flush' the filter without interrupting the flow is to be taken into account when estimating the pump capacity (see Lubricating oil pump, 4-15).

Type [a]	Automatic back-flushing filter with differential pressure gauge and high-differential pressure alarm contacts. Designed to clean itself automatically using reverse flow or compressed air techniques. Back-flushing oil treatment by sludge checker.
Oil flow	Refer to *GTD*.
Working viscosity	95 cSt, at working temperature
Working pressure	6 bar
Test pressure	Specified by classification society
Diff. pressure, clean filter	Max. 0.2 bar
Diff. pressure, dirty filter	Max. 0.6 bar
Diff. pressure, alarm	Max. 0.8 bar Note: Real operational settings could be less according to filter maker's recommendation.
Mesh size	Sphere passing max. 0.050 mm
Filter material	Stainless steel mesh
Filter inserts bursting press.	Max. 3 bar differential across filter

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

Booster pump for crosshead lubrication

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

System oil

The engine pistons feature highly efficient jet-shaker oil cooling. The crankcase oils used as system oil are specified as follows:

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

The consumption of system oil is given in Table 1-2, 1-4.

4.3.3 Flushing the lubricating oil system

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

4.3.4 Lubrication for turbochargers

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

4.3.5 Cylinder lubricating oil system

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

For cylinder lubricating oil consumption refer to Table 1-2, 1-4.

Cylinder oil

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

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

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1) The FZG gear machines located at the FZG Institute, Munich/Germany shall be the reference test apparatus and will be used in the event of any uncertainty about test repeatability and reproducibility.
2) The Base Number is expressed in mg KOH/g as determined by test method ASTM D2896.

Recommended residual BN

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

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

Base number of cylinder lubricating oil

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

Alternatives to finished cylinder oils

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

Docu WinGD

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

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

Service tank and storage tank

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

Electrical trace heating for cylinder lubricating oil piping

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

4.3.6 Maintenance and treatment of lubricating oil

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

Oil separator

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

Oil samples

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

4.3.7 Drain tank

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

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

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

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

There is to maintain adequate drainage under sea conditions resulting in pitching and rolling. The amount of lubricating oil required for an initial charge of the drain tank is indicated in Figure 4-7. The total tank size is normally 5-10% greater than the amount of lubricating oil required for an initial filling.

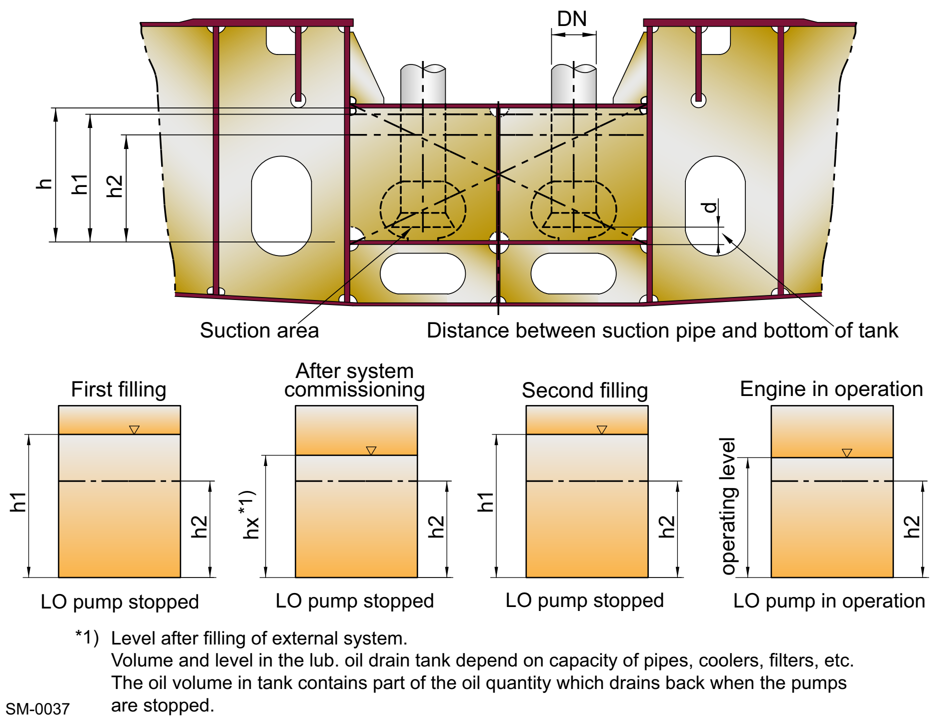


Figure 4-7 Dimensioning and filling process of lubricating oil drain tank

Arrangement of vertical lubricating oil drains

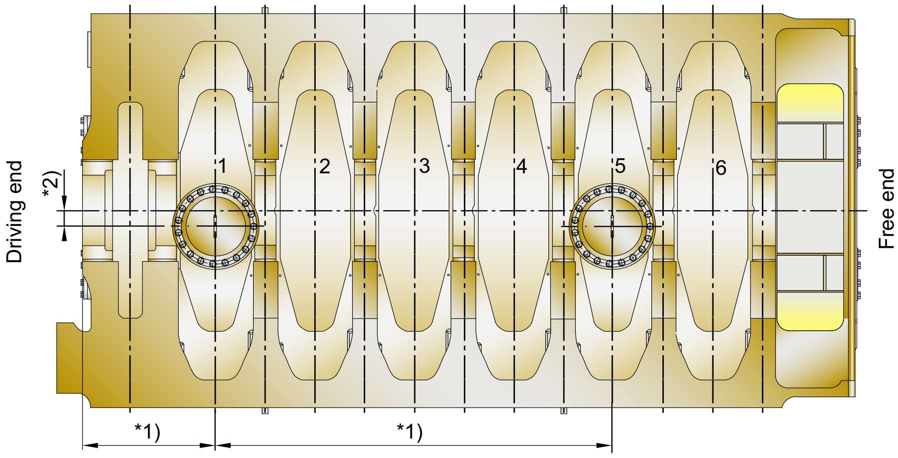
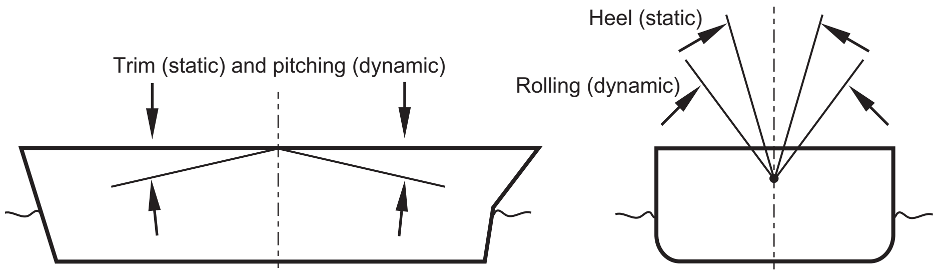


Figure 4-8 Arrangement of vertical lubricating oil drains for 6-cylinder engines

NOTE	The illustration above does not necessarily represent the actual configuration or the stage of development, nor the type of the engine concerned. For all relevant and prevailing information see MIDS drawings, 4-14.

Inclination angles



Athwartships and fore-and-aft inclinations may occur simultaneously.

NOTE	The data in the following tables represent the state of data as of the year 2019 and earlier. To obtain the latest data please contact the relevant classification society.

Table 4-3 Minimum inclination angles for full operability of the engine (1)

Classification societies (overview see Appendix, Table 9-1, 9-1)				
Year of latest update by Class	**ABS 2019**	**BV 2018**	**CCS 2018**	**CRS 2018**
Main and auxiliary engine				
Abbreviation	4/1/1/7.9	C/1/1/2.4	3/1/1/1.2.1	7/1/1.6/1.6.2
Heel to each side	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°
Trim by the head a)	5°	5°	5°	5°
Trim by the stern a)	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±7.5°
Emergency sets				
Abbreviation	4/1/1/7.9	C/1/1/2.4	3/1/1/1.2.1	7/1/1.6/1.6.2
Heel to each side	22.5° c)	22.5°	22.5° c)	22.5° c)
Rolling to each side	22.5° c)	22.5°	22.5° c)	22.5° c)
Trim	10°	10°	10°	10°
Pitching	±10°	±10°	±10°	±10°
Electrical installation				
Abbreviation	4/1/1/7.9	C/1/1/2.4	4/1/2/1.2.1	7/1/1.6/1.6.2
Heel to each side	22.5° b)	22.5° b) c)	15° c)	22.5° b)
Rolling to each side	22.5° b)	22.5° b) c)	22.5° c)	22.5° b)
Trim	10°	10° b)	5°	10° b)
Pitching	±10°	±10° b)	±7.5°	±10° b)

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

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

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

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

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

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

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

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

Table 4-5 Minimum inclination angles for full operability of the engine (3)

Classification societies (overview see Appendix, Table 9-1, 9-1)					
Year of latest update by Class	**LR 2018**	**NK 2018**	**PRS 2019**	**RINA 2018**	**RS 2019**
Main and auxiliary engine					
Abbreviation	5/1/3/3.7	D/1.3.1/6	VI/1/1.6.1	C/1/1/2.4	VII/2/2.3
Heel to each side	15°	15°	15°	15°	15°
Rolling to each side	22.5°	22.5°	22.5°	22.5°	22.5°
Trim by the head a)	5°	5°	5°	5°	5°
Trim by the stern a)	5°	5°	5°	5°	5°
Pitching	±7.5°	±7.5°	±7.5°	±7.5°	±7.5°
Emergency sets					
Abbreviation	5/1/3/3.7	D/1.3.1/6	VI/1/1.6.1	C/1/1/2.4	VII/2/2.3
Heel to each side	22.5° c)	22.5° b) c)	22.5° c)	22.5° c)	22.5° c)
Rolling to each side	22.5° c)	22.5° b) c)	22.5° c)	22.5° c)	22.5° c)
Trim	10°	10° b)	10°	10°	10°
Pitching	±10°	±10° b)	±10°	±10°	±10°
Electrical installation					
Abbreviation	6/2/1/1.10	H/1/1.1.7	VIII/2/2.1.2.2	C/2/2/1.6	XI/2/2.1.2.2
Heel to each side	15°	15° c)	15°	22.5° b)	15° c)
Rolling to each side	22.5°	22.5° c)	22.5°	22.5° b)	22.5° c)
Trim	5° a)	5° a)	5°	10° b)	5° c)
Pitching	±7.5°	±7.5°	±10°	±10° b)	±10° c)

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

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

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

4.4 Fuel oil system

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

Figure 4-9 shows the installation principle for maximum fuel flexibility.

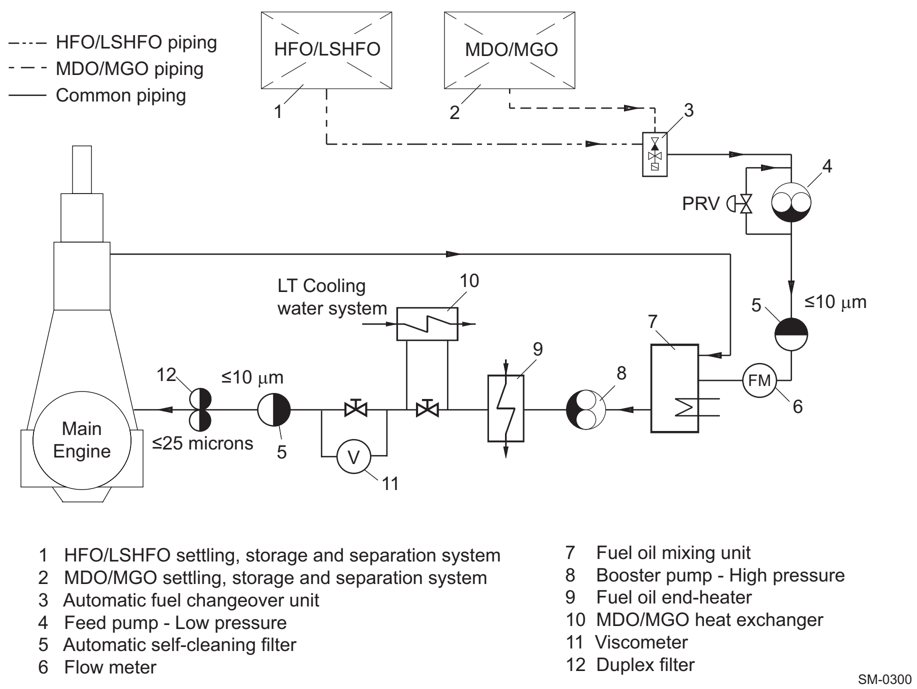


Figure 4-9 Fuel oil system

Further information about MDO/MGO fuels is available in the separate **Concept Guidance** (DG 9723). This considers additional design options for the fuel oil system, as well as optional heat exchangers for better viscosity regulation when changing between HFO/LSHFO and MDO/MGO.
This is provided on the WinGD corporate webpage under the following link:
Concept Guidance Distillate Fuels

Fuel consumption

Data of fuel consumption should be taken from the project-specific *GTD* data sheet.

4.4.1 Fuel oil system components

The following components are associated with a fuel oil system of maximum fuel flexibility, i.e. operation on heavy fuel oils and distillates, as indicated in Figure 4-9. Therefore, the following section considers a fuel oil viscosity of 700 cSt at 50 °C.

Feed pump — Low-pressure fuel oil

Type	Positive displacement screw pump with built-in safety valve
Capacity	According to *GTD*: The capacity is to be within a tolerance of 0 to +20 % of the GTD value, plus back-flushing flow of automatic self-cleaning filter, if such filter is installed.
Delivery pressure	The feed pump must provide a required pressure in the downstream mixing unit to prevent water in the system from vaporising into steam. The pump, whilst considering system pressure drop, must provide a minimum of 1 bar above the water vapour pressure and always be above a 3 bar value. 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 compared to lighter oils. (Refer to the formula and example below.)
Electric motor	The electric motor driving the fuel oil feed pump shall be sized for the maximum pressure head (difference between inlet and outlet pressure), maximum fuel oil viscosity (100 cSt), and the maximum required flow.
Working temp.	Below 100 °C
Fuel oil viscosity	Depending on the fuel oil system's heat control, viscosity at working temperature will often differ. It will not be more than 100 cSt, however can be considerably less (as low as 2 cSt with lower viscosity fuel like MDO/MGO or possibly LSHFO, see viscosity-temperature diagram - Figure 4-14, 4-40). The manufacturer's specification must comply with the fuel viscosity range. For system options with additional temperature regulation, see *Concept Guidance Distillate Fuels*.

Formula for delivery gauge pressure

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

where:

p_v .... = water vapour gauge pressure at the required system temp. [bar] (see viscosity-temperature diagram in section 4.4.6, 4-40)

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

Δp_2 .... = max. pressure change difference across the pressure regulating valve of the feed system between min. and max. flow (see Pressure regulating valve, 4-27)

Example

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

p_v .... = 3.2 bar

Δp_1 .... = 0.5 bar

Δp_2 .... = 0.6 bar

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

Pressure regulating valve

The pressure regulating valve returns the excess fuel oil that is not required by the main engine, recirculating more when the engine is at lower power. To avoid heating-up of the fuel by recirculation, the return pipe is designed with cooling ribs.
It also works to ensure that the pressure downstream of the low-pressure feed pump remains 1 bar above the evaporation pressure of the water and there prevents entrained water within the fuel oil system from flashing off into steam.

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

Type	Self-operated or pilot-operated, with a manual emergency control. Either direct hydraulically or pneumatically actuated. However, when using a pneumatically actuated valve, use a combined spring type to close the valve in case of air supply failure.
Maximum capacity	According to *GTD*: Refer to feed pump capacity.
Minimum capacity	Approx. 20 % of that of the fuel oil feed pump
Service pressure	Max. 10 bar
Pressure setting range	2-6 bar
Inlet pressure change	The inlet pressure may vary by up to 0.8 bar depending on the flow in the range of 20 % to 100 %.
Working temperature	Below 100 °C
Fuel oil viscosity	Depending on the fuel oil system's heat control, viscosity at working temperature will often differ. It will not be more than 100 cSt, however can be considerably less (as low as 2 cSt with lower viscosity fuel like MDO/MGO or possibly LSHFO, see viscosity-temperature diagram - Figure 4-14, 4-40). The manufacturer's specification must comply with the fuel viscosity range. For system options with additional temperature regulation, see *Concept Guidance Distillate Fuels*.

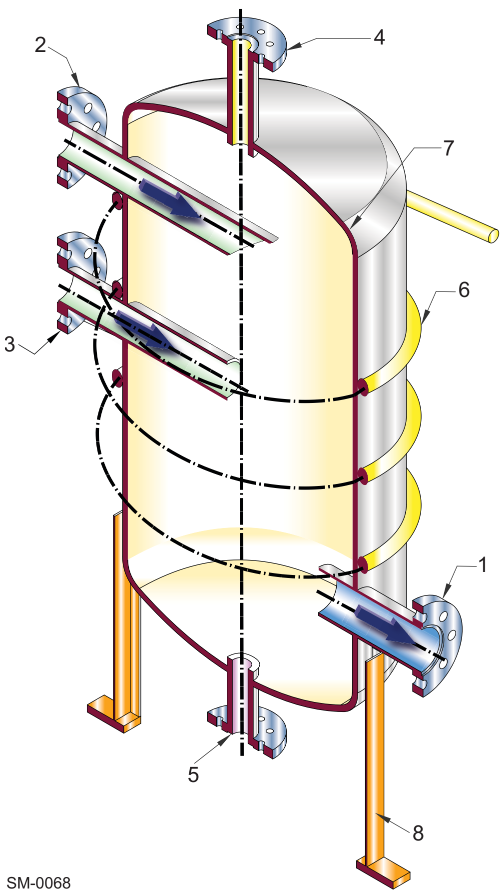
Mixing unit

The mixing unit equalises the temperature between the hotter fuel oil returning from the engine and the colder fuel oil from the service tank. The temperature difference between these sources is particularly high when changing over from HFO to MDO/MGO and vice versa.

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

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

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



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

Figure 4-10 Mixing unit

Booster pump — High-pressure fuel oil

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

Type	Positive displacement screw pump with built-in safety valve
Capacity	According to *GTD*: The flow rate is to be within a tolerance of 0 to +20% of the GTD value, plus back-flushing flow of automatic self-cleaning filter, if such filter is installed.
Inlet pressure	Up to 6 bar
Delivery head	Final delivery pressure according to actual piping layout. Refer to *GTD*.
Electric motor	The electric motor driving the HP booster pump shall be sized large enough for the power absorbed by the pump at maximum pressure head (difference between inlet and outlet pressure), maximum fuel oil viscosity (70 cSt), and the required flow.
Working temperature	Up to 150 °C

End-heater

Operation is regulated with either the temperature or the viscosity (default mode) of the fuel oil. The viscosity is measured by the viscometer.

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

a) The viscosity is maintained by regulating the fuel temperature after the end-heater.

Viscometer

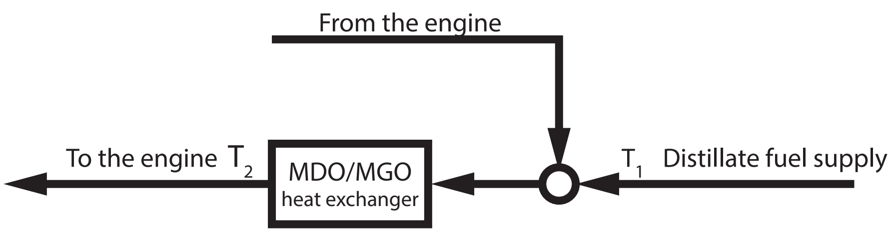
The viscometer regulates the fuel oil end-heater, enabling the fuel oil viscosity to be kept at preferable engine conditions. When using HFO/LSHFO the ideal is 13-17 cSt, and between 10-20 cSt is acceptable. Although there is no active cooling required to keep the minimum limit of 10 cSt, the absolute minimum viscosity is 2 cSt.
When using low-viscosity fuels, the end-heater must not be active and the MDO/MGO heat exchangers will be required to maintain these limits.

MDO/MGO heat exchanger

For MDO/MGO operation, the fuel might need to be cooled to stay above a minimum viscosity of 2 cSt at engine inlet. This fuel oil heat exchanger (and any optional heat exchanger that is included, as shown in *Concept Guidance Distillate Fuels*) uses low-temperature cooling water.

A chiller unit (cooling from refrigeration) is not required if the fuel properties are in line with the latest ISO 8217:2017 specification. Such a unit would only be needed for off-spec fuels that are not supported by WinGD.

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



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Fuel oil filters — Arrangement 'A'

Filtration grading

The grade of filter mesh is relative to the size of particles it captures, however, there are different ratings for filtration efficiencies and the rating parameters vary among manufacturers.

For simplicity, it is assumed that particles are spherical, therefore the size is defined by an equivalent diameter. A filter's grading size is associated with this equivalent diameter but can vary depending on whether an *absolute* or *nominal* grading system is used.

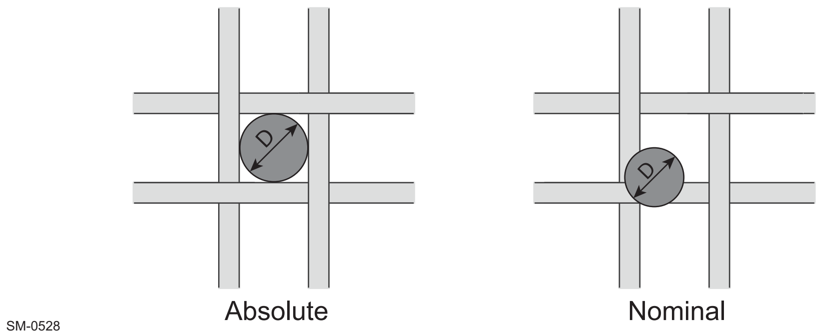


Figure 4-11 Mesh size difference between absolute and nominal

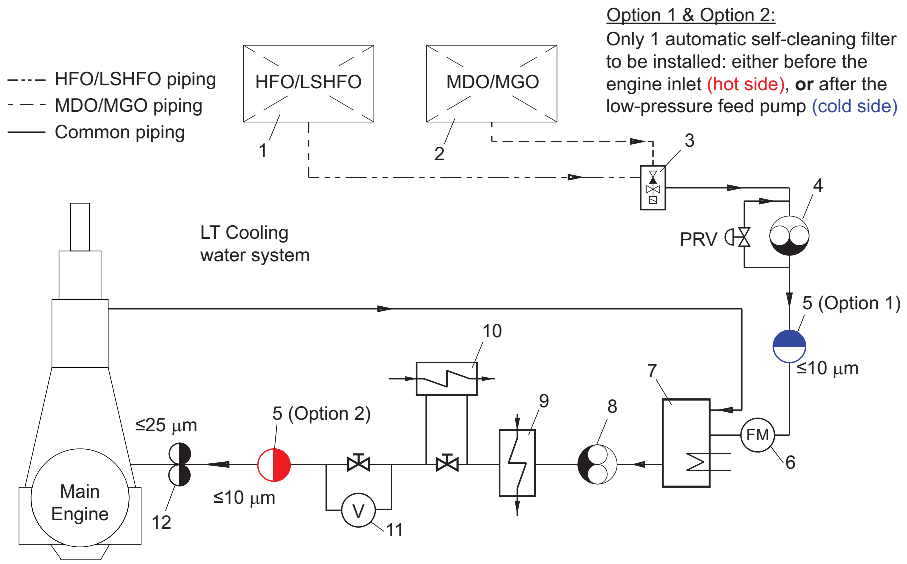
Absolute Filtration Grade
This value indicates the largest equivalent diameter capable of passing through the filter. This value equals the size of the mesh coarseness and can therefore be referred to as the *absolute sphere passing mesh size*. Approximately all particles larger than the specified size will be trapped on or within the filter.

Nominal Filtration Grade
This value is typically smaller than the absolute value and refers to an equivalent diameter where most particles will be trapped. Generally, this is interpreted as a 85% to 90% filtration for particles larger than the nominal value but this can range (from 60% to 90% approximately). Because of the complication this variation can cause, the nominal grade for filtration is not used in the following.

NOTE	WinGD provides all filter mesh sizes in absolute (abs.) values.

Arrangement 'A' of fuel oil filters (see Figure 4-12, 4-32) comprises:

- An automatic self-cleaning filter of maximum 10 µm abs., installed either in the 'cold' feed system (see Option 1, 4-33) or in the 'hot' booster system close to engine inlet (see Option 2, 4-34).
- A duplex filter of recommended maximum 25 µm abs., installed downstream of the engine inlet booster system (see Duplex filter, 4-35).



1 HFO/LSHFO settling, storage and separation system
2 MDO/MGO settling, storage and separation system
3 Automatic fuel changeover unit
4 Feed pump - Low pressure
5 Automatic self-cleaning filter
6 Flow meter
7 Fuel oil mixing unit
8 Booster pump - High pressure
9 Fuel oil end-heater
10 MDO/MGO heat exchanger
11 Viscometer
12 Duplex filter

SM-0206

Figure 4-12 Fuel oil filter arrangement 'A'

The automatic self-cleaning filter of maximum 10 µm abs. is used to protect the engine against serious damage. It captures the catalytic fines which were not removed by the fuel oil separator. In addition, the filter provides a good indication of the separator's efficiency.

NOTE	The self-cleaning performance must be suitable for the required filter fineness.

The 10 µm abs. filter can be installed in two different locations:

Option 1

Filter installation in the feed system:
In this position the maximum 10 µm abs. filter can be designed for a lower flow rate compared to the installation in the booster system. However, higher resistance due to higher fuel viscosity needs to be considered.

Advantage and disadvantage of this filter position:

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

Table 4-6 Specification of automatic self-cleaning filter in feed system

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

Option 2 Filter installation in the booster circuit:
The maximum 10 μm abs. filter is installed in the booster circuit close to engine inlet. The filter needs to be laid out for a maximum working temperature of 150 °C.

Advantage and disadvantage of this filter position:

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

Table 4-7 Specification of automatic self-cleaning filter in booster system

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

Duplex filter

The second filter in Arrangement 'A' is a duplex filter of recommended maximum 25 µm abs. A coarser filter is also acceptable. The duplex filter is of manual cleaning type and is installed in the booster system close to engine inlet. This filter type is sufficient as most particles are already removed by the 10 µm filter as outlined in Option 1, 🗎 4-33 and Option 2, 🗎 4-34.

Table 4-8 Specification of duplex filter in booster system

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

Fuel oil filter — Arrangement ‘B’

A maximum 10 µm abs. filter of automatic self-cleaning type is installed in the 'hot' booster system close to engine inlet. The filter needs to be laid out for a maximum working temperature of 150 °C.

NOTE	The self-cleaning performance must be suitable for the required filter fineness.

Same filter specification as provided by Table 4-7, 4-34.

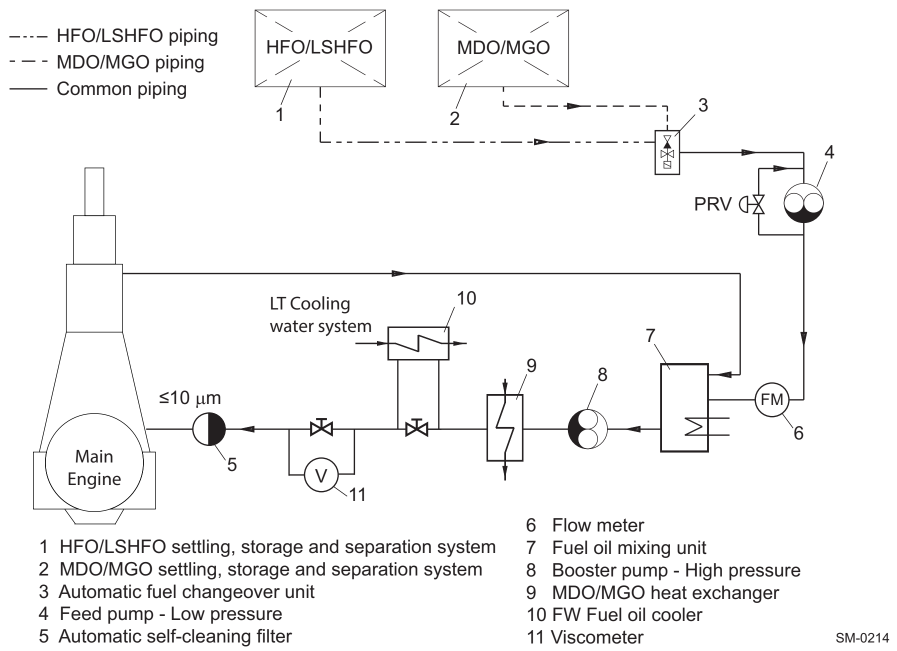


Figure 4-13 Fuel oil filter arrangement ‘B’

Arrangement ‘B’ does not include secondary duplex filtration. It lacks the indication of overall performance of the fuel oil treatment system and gives no indication when the automatic self-cleaning filter fails.

NOTE	WinGD recommend Arrangement ‘A’, as this is a best practice solution.

4.4.2 Flushing the fuel oil system

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

4.4.3 Fuel oil treatment

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

Settling tanks

Gravitational settling of water and sediment from modern heavy fuel oils is an extremely slow process due to the small difference in densities. The time required for the settling to occur depends on the depth of the tank, as well as on the viscosity, temperature and density difference. Tanks that are shallower with a wider diameter enable better separation than thinner, taller tanks.
Further design features consider a level monitoring device and remote closing discharge valves to the separator(s) and engine systems, a connection to an overflow tank, and a self-closing cock just above the bottom of the tank for removal of the sludge and water.

Service tanks

Most of the service tank design features are similar to the settling tanks, however, there is no direct connection to the overflow tank. Instead a recirculating line, with an inlet just above the bottom of the service tank, leads back to the settling tanks.
Diesel oil service tanks are similar to heavy oil service tanks, with the possible exception of tank heating, although this may be incorporated for vessels constantly trading in cold climates.

Water in fuel

Due to condensation or heating coil leakage, water may be present in the fuel after the separators. This can be manually removed by a self-closing cock. In addition, the recirculation connection close to the bottom of the tank ensures that contaminated fuel is recirculated to the settling tank.

Cleaning of fuel

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

Centrifugal fuel oil separators

There are two types of oil separators:
- Type 1 — Separators with gravity discs
- Type 2 — Separators without gravity discs

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

Separators without gravity discs

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

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

Separation efficiency

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

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

where:

n .... = separation efficiency [%]

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

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

Certified Flow Rate

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

Throughput capacity

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

Example

- 8-cyl. engine
- CMCR / R1: 18,800 kW
- BSFC / R1: 167.8 g/kWh
- **Throughput:** $1.2 \cdot 18{,}800 \cdot 167.8 \cdot 10^{-3}$ = **3,786 litres/hour**

Oil samples

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

4.4.4 Pressurised fuel oil system

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

Fuel changeover

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

Automatic changeover unit

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

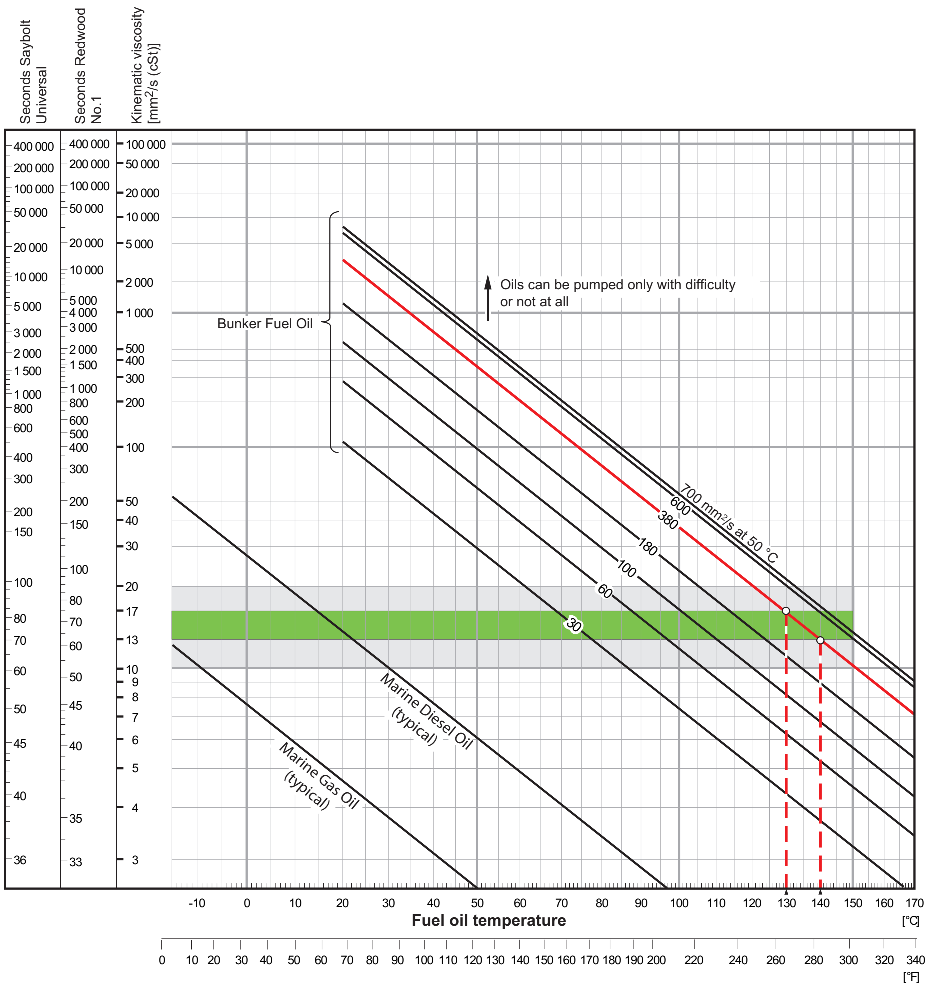
- A maximum temperature gradient of 2K/min during changeover
- A maximum viscosity of 20cSt
- A minimum viscosity of 2cSt; this minimum limit is most challenging during changeover from HFO to distillate fuel.
  Attention: Not all changeover units guarantee keeping the minimum viscosity limit, as viscosity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

4.4.5 Fuel oil specification

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

4.4.6 Fuel oil viscosity-temperature dependency

The fuel oil viscosity depends on its temperature. This dependency is shown as graph in Figure 4-14.



Recommended viscosity range for HFO before fuel injection pumps

Required viscosity range for HFO before fuel injection pumps

Example To obtain the recommended viscosity before fuel injection pumps a fuel oil of 380 mm²/s (cSt) at 50 °C must be heated up to 130 °C to 140 °C.

SM-0215

Figure 4-14 Fuel oil viscosity-temperature diagram

4.5 Starting and control air system

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

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

The starting and control air system shown in Figure 4-15 comprises two air compressors, two air receivers, and systems of pipework and valves connected to the engine starting air manifold.

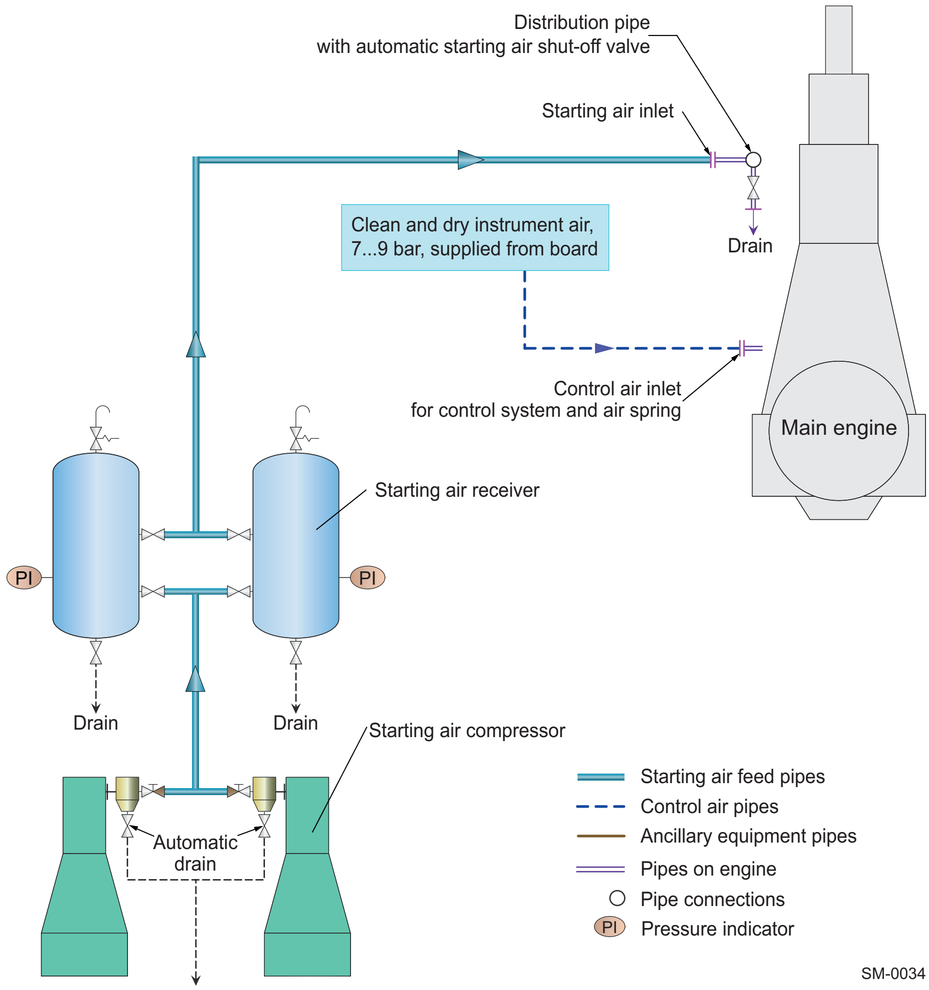


Figure 4-15 Starting and control air system

4.5.1 Capacities of air compressor and receiver

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

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

4.5.2 System specification

Starting air compressors

Capacity	Refer to *GTD*.
Delivery gauge pressure	25 or 30 bar

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

Starting air receivers

Type	Fabricated steel pressure vessels with domed ends and integrated pipe fittings for isolating valves, automatic drain valves, pressure reading instruments and safety valves
Capacity	Refer to *GTD*.
Working gauge pressure	25 or 30 bar

1) Propeller inertia includes the part of entrained water.
2) The *GTD* application enables the capacities of compressors and air receivers to be optimised for the inertia of the engine and shaft line.

4.5.3 Control air

Control air system supply

Control air is supplied from the board instrument air supply system (see Figure 4-15, 4-41) providing air at 8 bar gauge pressure (within a range of 7.0-9.0 bar). The air quality should comply with the compressed air purity class **2-4-2** according to ISO 8573-1 (2010-04-15).

Control air consumption

With the development of engine technology the WinGD RT-flex and X/X-DF engines consume much less control air than conventional engines. The required control air flow capacities are shown in Table 4-9. These data can be used for sizing the relevant engine external piping and facilities.

Table 4-9 Control air flow capacities

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

4.5.4 Service and working air

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

4.6 Leakage collection system and washing devices

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

Sludge oil trap

Dirty oil collected from the piston underside is led under a pressure of approx. 2.8 bar to the sludge oil trap and then to the sludge oil tank.
The purpose of the sludge oil trap (see Figure 4-16) is to retain the large amount of solid parts contained in dirty oil and to reduce the pressure by means of an orifice or throttling disc fitted at its outlet, so that the sludge oil tank is under atmospheric pressure.
Design and dimensions of the sludge oil trap are given in the MIDS.

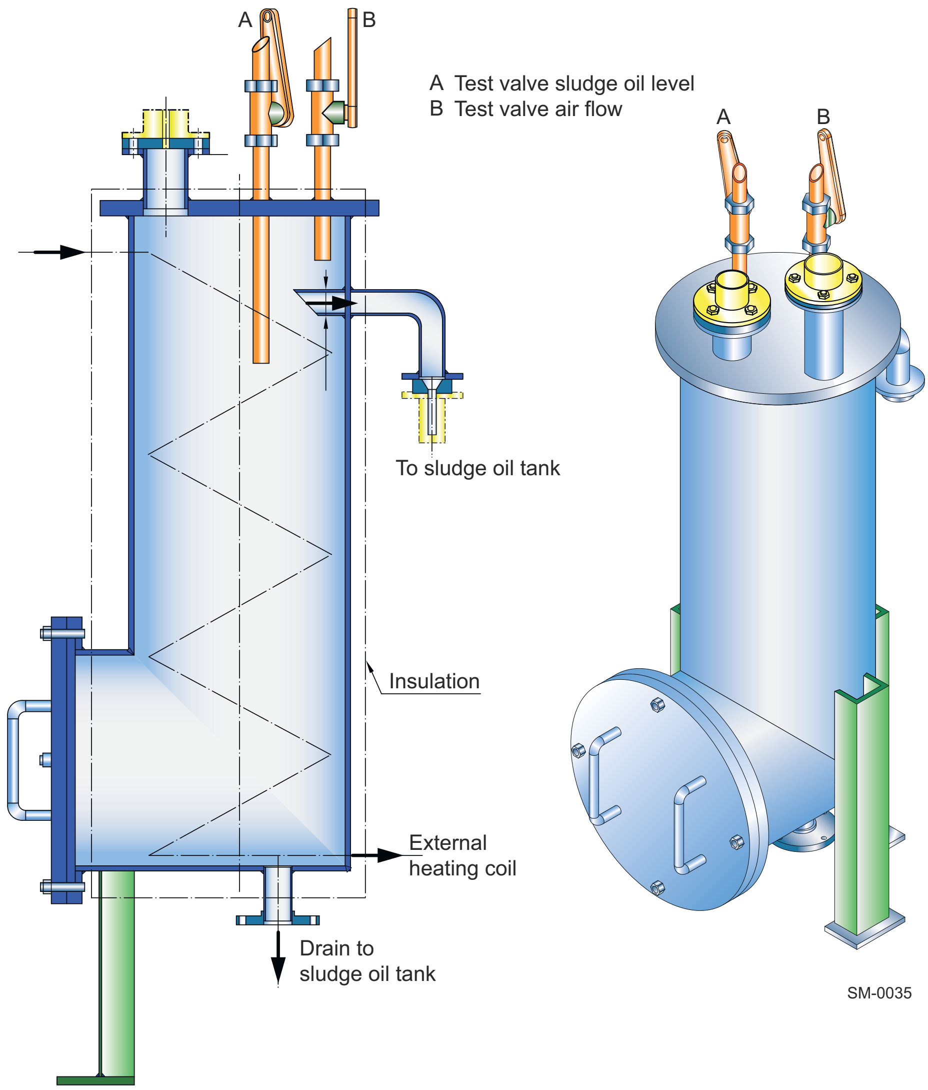


Figure 4-16 Sludge oil trap

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

4.6.1 Draining of exhaust uptakes

Engine exhaust uptakes can be drained automatically using a system as shown in Figure 4-17.

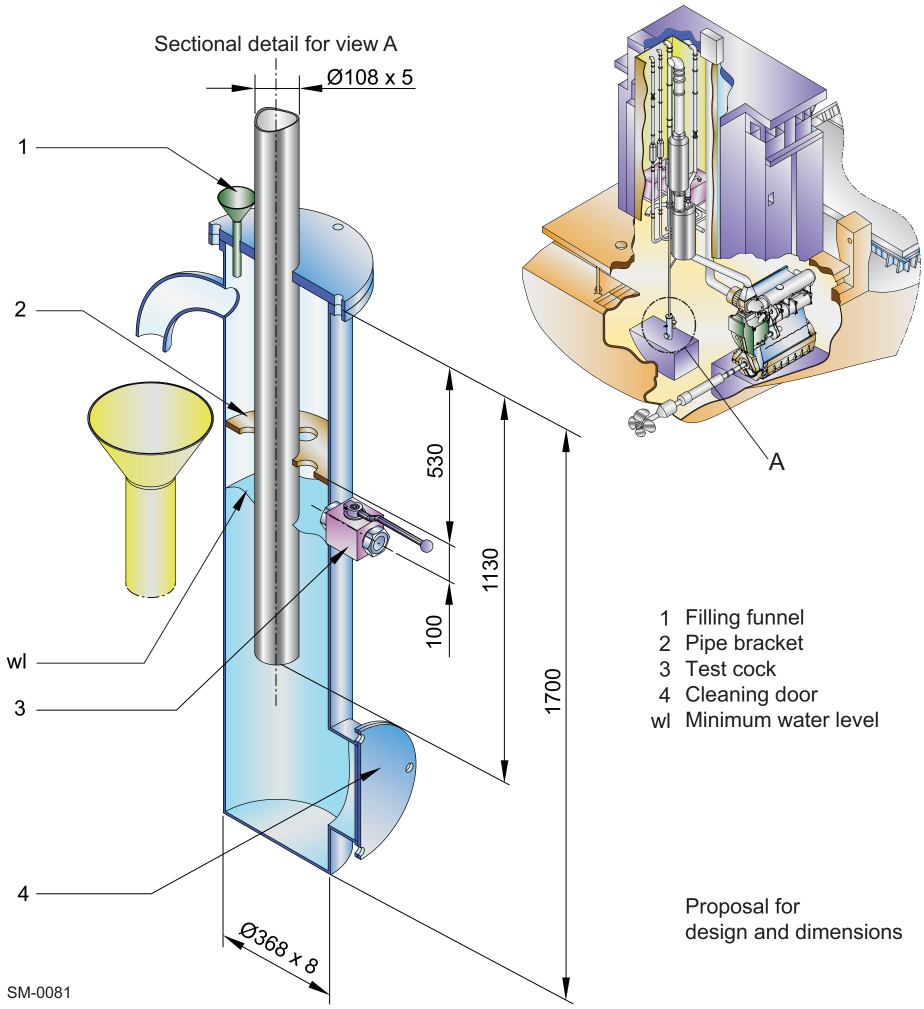


Figure 4-17 Arrangement of automatic water drain

4.6.2 Air vents

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

4.7 Exhaust gas system

The drawings relevant for the exhaust system (DG 9726) are provided on the WinGD corporate webpage under the following link:
MIDS

Flow velocities

For an optimised exhaust gas system the following velocities are recommended for pipes A, B and C shown in Figure 4-18:

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

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

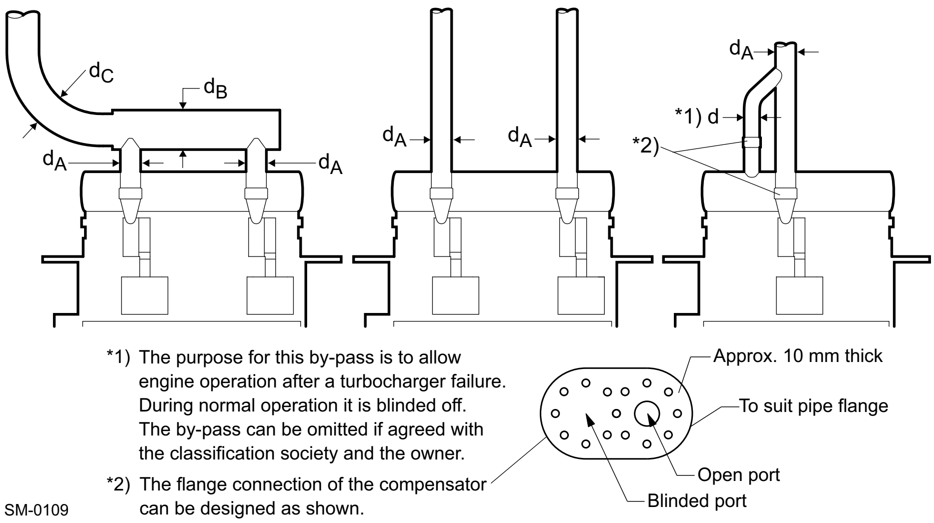


Figure 4-18 Determination of exhaust pipe diameter

4.8 Engine room ventilation

Special attention for the engine room ventilation is essential to ensure trouble-free operation of all equipment. It is important that the ventilation requirements, ventilation arrangement, air quality and outside ambient air temperature are taken into consideration.

4.8.1 Ventilation requirements

Engine room ventilation is to conform to the requirements specified by the legislative council of the vessel's country of registration and the classification society selected by the ship owner.

Calculation methods for combustion air flow requirements and for avoiding excessive heating of the machinery spaces are provided in the international standard ISO 8861 'Shipbuilding — Engine-room ventilation in diesel-engined ships — Design requirements and basis of calculations'.

The engine's combustion air is considered, and typically provided, as part of the overall engine room ventilation system. Approximately 50% of the overall engine room ventilation air is for the main engine's combustion, while the other half is used for the auxiliary engines, the boilers, and to provide sufficient cooling for equipment in the engine room. It is therefore vitally important that the ventilation system for the engine room has sufficient capacity to supply the necessary air flow for all engine room needs.

GTD

The heat emissions, required air flow and estimated power for the layout of engine room ventilation can be obtained from the *GTD* application. These values consider the ISO 8861 standard, however, in some circumstances the results are different from the standard calculations. In these cases, WinGD has provided the specific engine values and these should be considered before ISO 8861.

It should be considered that the engine requires less combustion air when not running at full load. This then provides a potential energy save, by reducing the frequency of the ventilation fans when demand is low. This process can be automated, interfacing with the engine, if requested.

4.8.2 Ventilation arrangement

It is important to follow the best practice methods for supplying the combustion air for main engine as described in this section. However, the final layout of the engine room ventilation is at the discretion of the shipyard.

Two different ventilation arrangements

Experience shows that the air flow in the engine room, from the ventilation system outlet to the turbocharger inlets, should be as direct as possible. This increases the amount of air directly supplied to the turbocharger, limiting heat transfer to the air flow and therefore providing the best possible engine performance, especially during tropical conditions.
Alternatively, a ventilation system with a direct air suction layout can be arranged, where the ventilation system connects the outside ambient air directly to the engine.

These two different arrangements are discussed as follows:

- **Arrangement 1** — Engine room ventilation system (Figure 4-19, 4-49)
  The ventilation system draws air from the outside ambient air into the engine room, where it is sucked into the turbocharger inlet.

- **Arrangement 2** — Direct engine ventilation system (Figure 4-20, 4-50)
  The ventilation system outlet is connected to the turbocharger inlet. Therefore, the outside ambient air is sucked directly into the turbocharger without passing through the engine room.

NOTE	In both arrangements, the ventilation inlets must be located and designed to ensure that water spray, rain water, dust and exhaust gases cannot enter the system nor the engine room.

Arrangement 1 — Engine room ventilation system

Functional principle

The ventilation system draws air from outside the vessel using ventilation fans at the inlet. Ventilation inlets are typically protected with a weather hood and louvres to minimise the amount of water and other particles entering the system. The air travels to the engine room where it leaves the ventilation outlets and enters the engine.

Layout

The engine room ventilation should be arranged in such a way that the main engine combustion air is **delivered directly to the turbocharger inlet**, locating the ventilation outlet and turbocharger inlet as close as possible, and directly facing to each other, ensuring a smooth and direct flow of air.

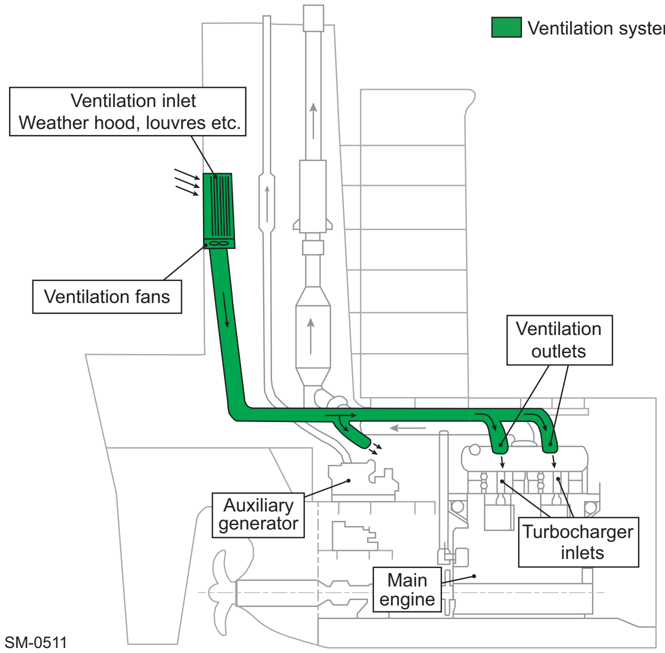


Figure 4-19 Ventilation system arrangement 1 — Engine room ventilation system

By ensuring that the air flow from the ventilation outlet to the turbocharger inlet is as direct as possible, the air intake is at its coolest. The ventilation arrangement should limit air from the engine room, mixing with the inlet air and entering the turbocharger. Limiting this heating effect will keep the engine's specific fuel consumption low as a consequence.

TC with filter

The turbocharger is fitted with a filter silencer that reduces noise and prevents large items from entering the turbocharger. Most turbocharger manufacturers supply an optional secondary fibre or mat filter that can further remove particles and oil mist. This can reduce the effort required for scavenge air cooler cleaning.

NOTE	WinGD recommend selecting the optional secondary filter to further assist with removing fine particles and oil mist that may be present in the engine room.

Arrangement 2 — Direct engine ventilation system

Layout

In this arrangement, the ventilation outlets are coupled with the turbocharger inlets. As the turbochargers directly receive all the outside ambient air drawn via the ventilation system, there is little chance for the temperature to increase. As a result, this arrangement delivers cooler air to the engine than in ‘Arrangement 1’, reducing the engine’s brake specific fuel consumption.

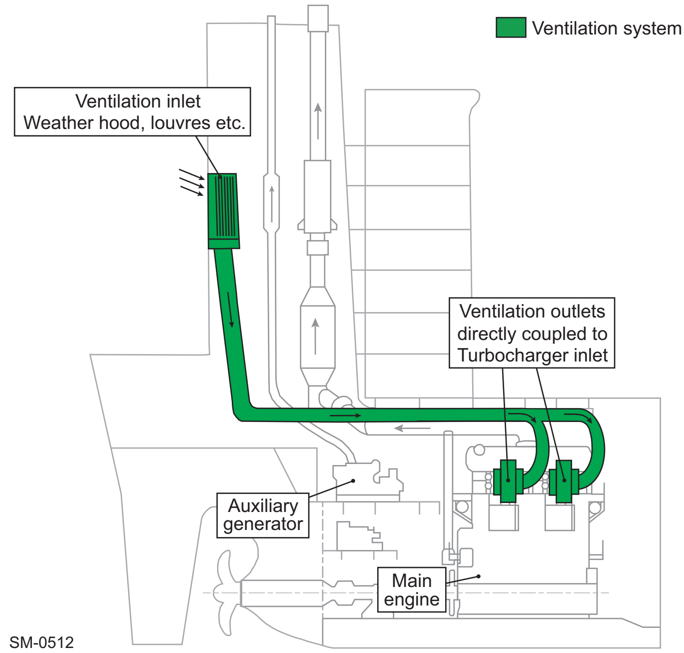


Figure 4-20 Ventilation system arrangement 2 — Direct engine ventilation system

The outside ambient air is drawn through the ventilation system by the turbochargers, and therefore there is no need for ventilation fans in this arrangement. However, it is still essential that the ventilation inlet is protected, typically with a weather hood and louvres. A separate filter unit, if required, can also be fitted here, within the inlet housing.

Requirements

The engine room ventilation system is provided with separate ventilation fans and ducting. It should be appropriately sized to provide comfortable working conditions in the engine room, supply the necessary combustion air for auxiliary generator(s) and the boiler, and to prevent heat-sensitive apparatus from overheating.

As the main engine combustion air is no longer provided for by the engine room ventilation system, the energy demand of the ventilation fans is reduced compared to ‘Arrangement 1’.

4.8.3 Air intake quality

Air intake quality can vary depending on the circumstances of the vessel. For example, suction air is expected to have a dust content of 0.5 mg/m^3 or higher if a vessel is carrying dusty or dust creating cargoes, such as iron ore and bauxite, or if it is often in port, trading in coastal waters and desert areas. In these cases, the air must be filtered before it enters the engine (see Table 4-10).

Dust filters

The necessity for installing a dust filter and the choice of filter type depends mainly on the concentration and composition of dust in the suction air. The normal air filters fitted as standard to the turbochargers are intended mainly as silencers but not to protect the engine against dust. If the air supply to machinery spaces has a dust content exceeding 0.5 mg/m^3, there is a risk of increased wear to the piston rings and cylinder liners.

NOTE	WinGD advise to install a filtration unit on vessels regularly transporting dust creating cargoes, or trading in areas of atmospheric dust.

Table 4-10 Guidance for air filtration

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

All filters' surfaces must be sized correctly to ensure full functionality of the filtration. This is dependent on the engine's maximum power output as is seen below in Figure 4-21, 4-52.

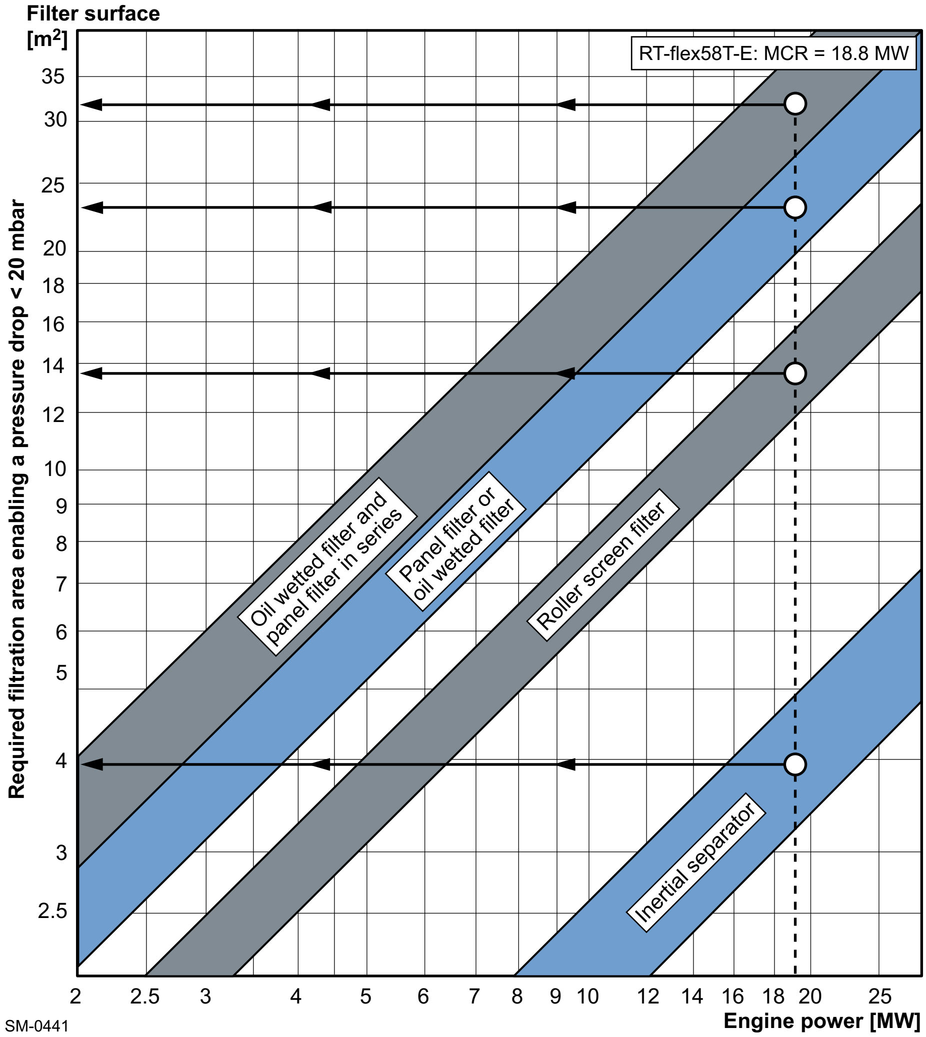


Figure 4-21 Air filter size (example for 8-cyl. engine)

4.8.4 Outside ambient air temperature

The intake air temperature can vary greatly depending on the area of operation, and as the engine may operate over a wide range of ambient air temperatures, hot and cold limits should be considered.

GTD

Depending on the engine tuning option, the acceptable range of 'Air temperature before compressor' (as titled by the GTD application) will vary. Please consult the condition values provided by the *GTD* for the maximum and minimum normal operating temperatures.

When operating within the normal range (as defined by the GTD application) the engine does not require any special measures. This includes pre-heating at low temperature, even when operating on heavy fuel oil at part load, idling and start-up. Because of this, there is no need for a separate scavenge air heater.

If the engine is required to operate outside of the temperature range defined by the GTD application, contact WinGD for special measures and requirements.

NOTE	No special measures are required for engine operation within the wide temperature range as defined in the GTD application.

4.9 Piping

4.9.1 Pipe connections

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

4.9.2 Flow rates and velocities

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

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

4.10 PTO, PTI, PTH and primary generator applications

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

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

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

4.10.1 Requirements

After selecting the engine:

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

NOTE	The type of the PTO/PTI system has an influence on the execution of the main engine. Thus, changing from one system type to another is possible in the project stage but not after having ordered the engine.

4.10.2 Arrangements for PTO, PTI, PTH and primary generator

Figure 4-22, 🗎 4-56 illustrates the different arrangements for PTO, PTI, PTH and primary generator.

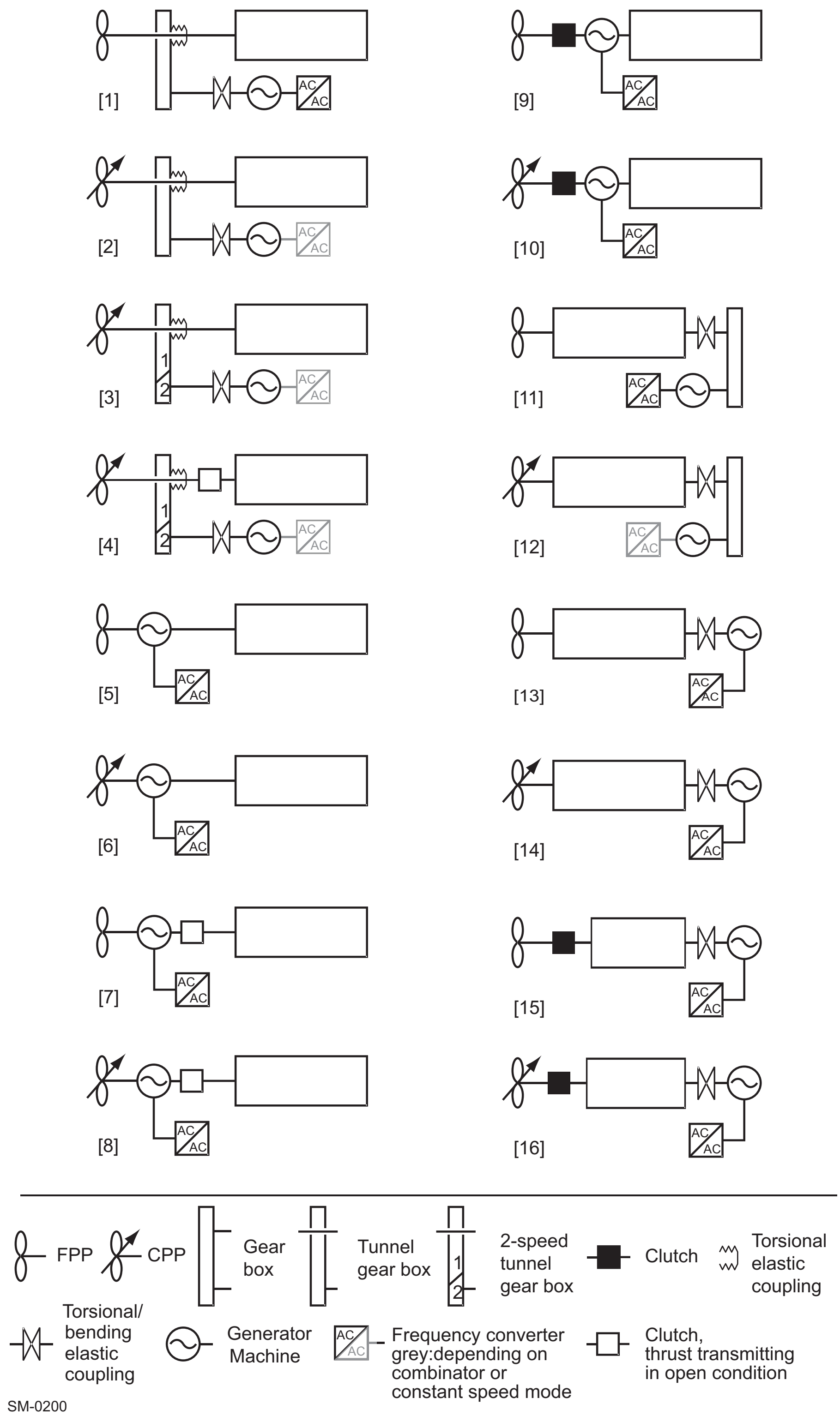


Figure 4-22 Arrangements for PTO, PTI, PTH

The following table itemises the arrangements corresponding to the numbers in Figure 4-22, 4-56.

Table 4-11 PTO/PTI/PTH arrangements for RT-flex58T-E

[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]	[15]	[16]
X	X	X	X	X	X	X	X	X	X	O	O	O	O	O	O

X = the arrangement is possible
O = the arrangement is not possible or plausible

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

4.10.3 Application constraints

The feasibility of project-specific PTO / PTI / PTH and primary generator needs to be studied in any case. An overview about impacts is given in Table 4-13, 4-58.

Table 4-12 Possible options for RT-flex58T-E

	Arrangements (see Figure 4-22, 4-56)															
Option	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]	[15]	[16]
PTO	X	X	X	X	X	X	X	X	X	X	--	--	--	--	--	--
PTI	X	X	X	X	X	X	X	X	X	X	--	--	--	--	--	--
PTH	O	O	O	X	O	O	X	X	O	O	--	--	--	--	--	--
Primary generator	O	O	O	O	O	O	O	O	X	X	--	--	--	--	--	--
Remarks	a)	a) b)	a) b)	a) b)					c)	c)						

X = the option is possible
O = the option is not possible
-- = the arrangement is not possible for RT-flex58T-E

a) If the lowest torsional natural frequency is < 1.5 Hz, special care has to be taken regarding possible engine speed fluctuations.
b) In case the electric generator/motor is operated at variable speed (CPP combinator mode), a frequency converter is needed.
c) With de-clutched propeller and pure generator operation, the minimum engine load requirement has to be obeyed.

Permanent Magnet

In cases where ‘Permanent Magnet’ type generators or electric motors are installed, special attention must be given to the alignment issue. Due to the low rotor mass in relation to the magnetic forces, a potential risk of unloaded shaft bearings may exist. In addition, bearing load measurements may be falsified by the influence of the permanent-magnetic pull force.

Table 4-13 Influence of options on engineering

Engineering	Arrangements (see Figure 4-22, 4-56)															
	[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]	[12]	[13]	[14]	[15]	[16]
Extended TVC	X	X	X	X	X	X	X	X	X	X	--	--	--	--	--	--
Misfiring detection	(X)	(X)	(X)	(X)	O	O	O	O	O	O	--	--	--	--	--	--
Impact on ECS	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	(X)	--	--	--	--	--	--
Shaft alignment study	(X)	(X)	(X)	(X)	X	X	X	X	X	X	--	--	--	--	--	--
Bearing load due to external load	(X)	(X)	(X)	(X)	X	X	X	X	X	X	--	--	--	--	--	--
Dynamic condition due to external load	O	O	O	O	O	O	O	O	O	O	--	--	--	--	--	--

X = the arrangement has an influence on this engineering aspect
(X) = the arrangement might have an influence on this engineering aspect
O = the arrangement has no influence on this engineering aspect
-- = the arrangement is not possible for RT-flex58T-E

Extended TVC

The added components have a considerable influence on the related project-specific torsional vibration calculation. Proper case dependent countermeasures need to be taken depending on the results of the detailed TVC. For further details, refer to section 6.4.2 PTO / PTI systems effect on torsional vibration, 6-13.

Misfiring detection

Depending on the results of the TVC, a misfiring detection device (MFD) might be needed to protect the elastic coupling and the gear-train (if present) from inadmissible torsional vibrations in case of misfiring.

Impact on ECS

The PTO / PTI / PTH application has to be analysed via the licensee with the Propulsion Control System supplier and with WinGD for the Engine Control System.

Shaft alignment study

The added components can have an influence on the alignment layout. The shaft bearing layout has to be properly selected and adjusted to comply with the given alignment rules. For further details, refer to 3.6 Engine and shaft alignment, 3-10.

Bearing load due to external load

The added components increase the bending moment and the related bearing loads. The bearing loads have to be checked for compliance with the given rules.

Dynamic conditions due to external load

The components attached to the free end have to be checked for any influence on the axial and radial movements of the extension shaft caused by the dynamics of the engine.

4.10.4 Service conditions

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

Operation area and prohibited area

The following illustrations indicate how the engine generator unit can be operated. The prohibited operation area is defined in section 2.2.7, 2-13.

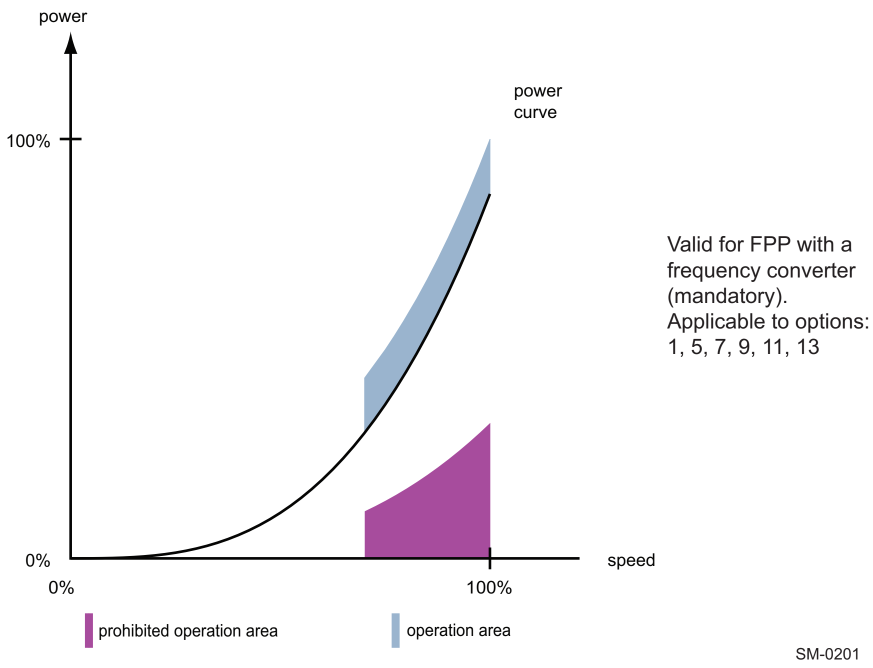


Figure 4-23 FPP with mandatory frequency converter

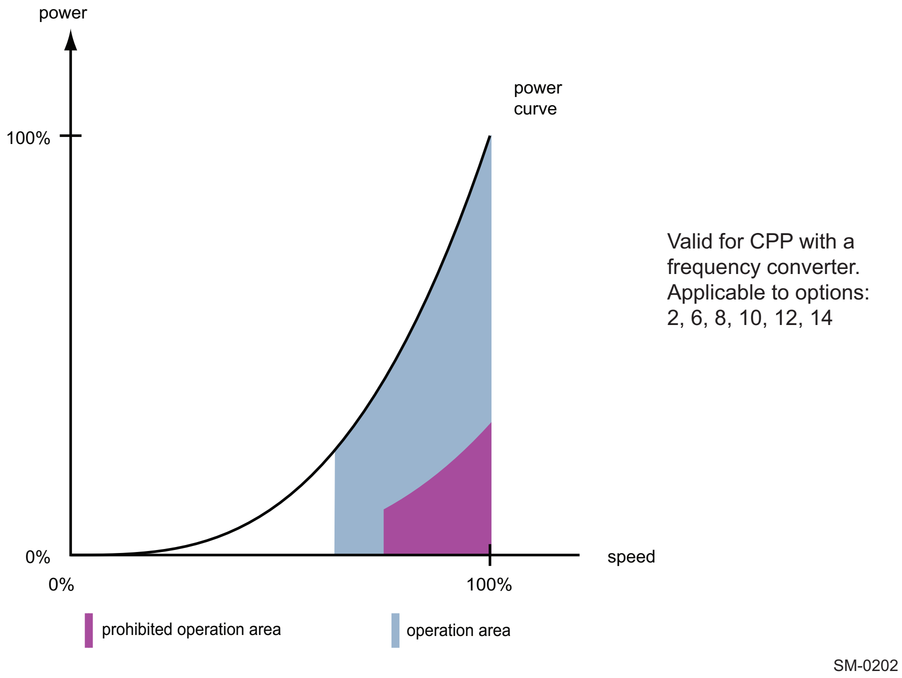


Figure 4-24 CPP in combination with an optional frequency converter

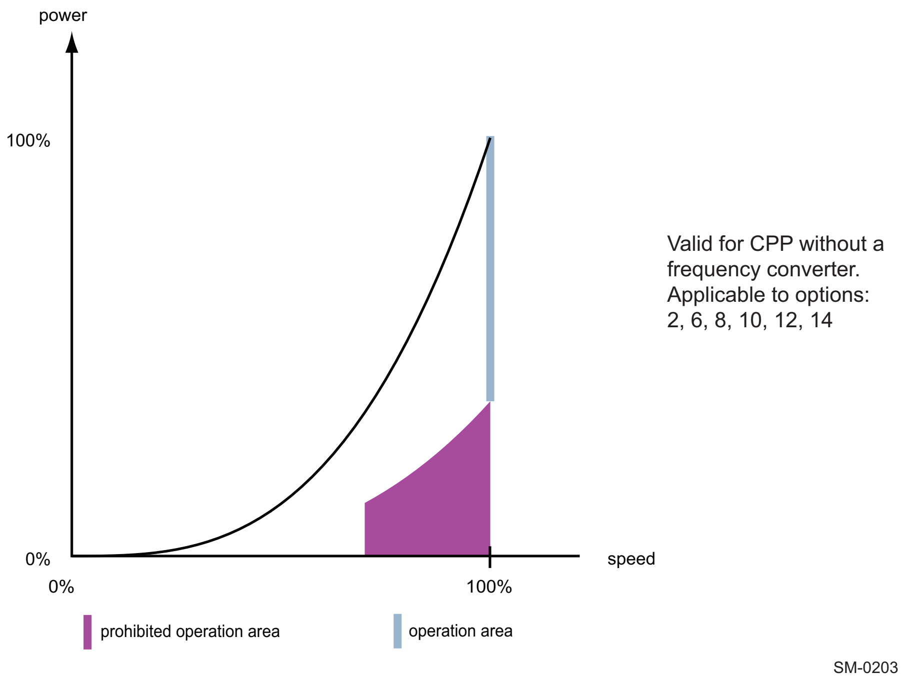


Figure 4-25 CPP in constant speed operation without frequency converter

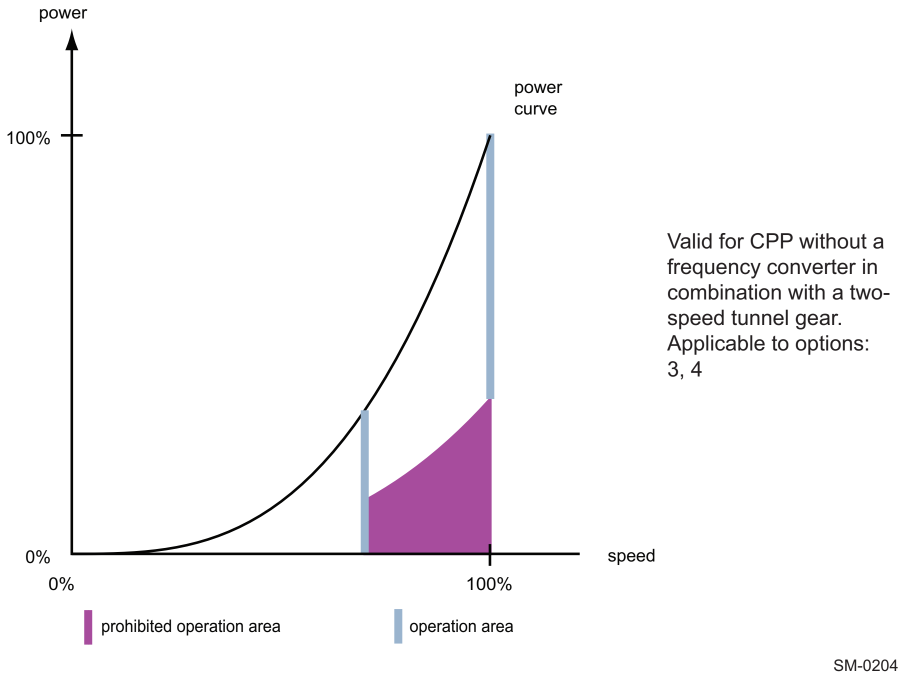


Figure 4-26 CPP with two fixed operation speeds without frequency converter

5 Engine Automation

WinGD provide a fully integrated Engine Control System named **WECS-9520**. This is connected via data bus to the Propulsion Control System (PCS) and the Alarm and Monitoring System (AMS). The AMS is usually provided by the shipyard.

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

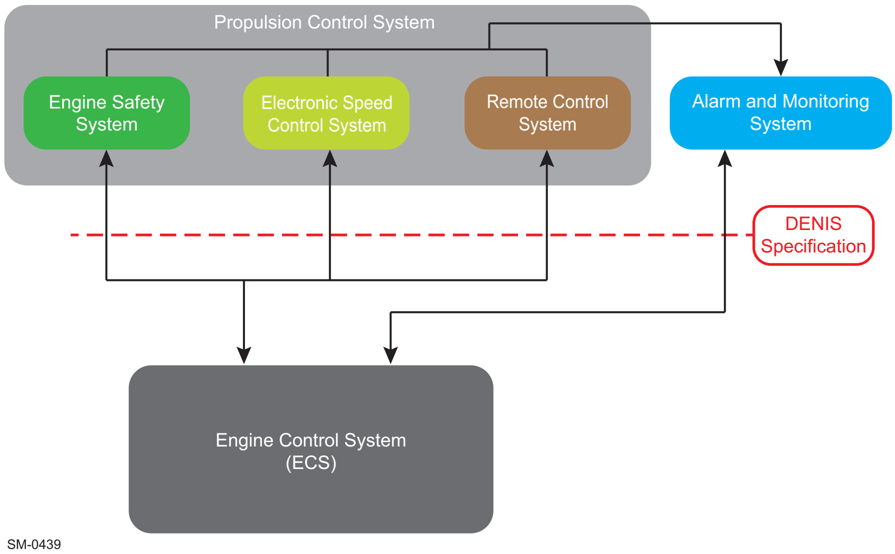


Figure 5-1 Engine automation architecture

5.1 DENIS-9520

WinGD's standard electrical interface is **DENIS-9520**, which is in line with approved Propulsion Control Systems.

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

ECS The Engine Control System (ECS) takes care of all Flex system-specific control functions, e.g. fuel injection, exhaust valve control, cylinder lubrication, crank angle measurement, and speed/load control. The system uses modern bus technologies for safe transmission of sensor and other signals.

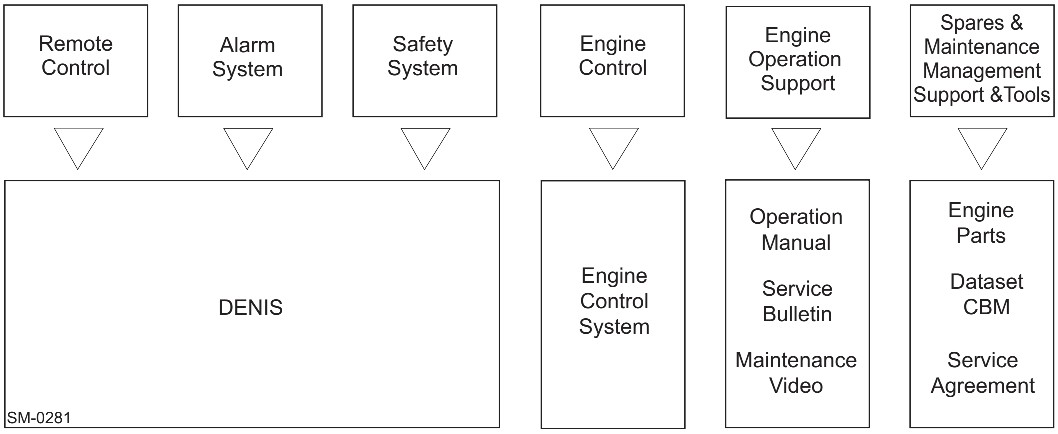


Figure 5-2 Engine management and automation concept

5.2 DENIS-9520 concept

The concept of DENIS-9520 offers the following features to ship owners, shipyards and engine builders:

5.2.1 Interface definition

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

5.2.2 Approved Propulsion Control Systems

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

5.3 DENIS-9520 Specification

The DENIS-9520 Specification describes the signal interface between the Engine Control System and the PCS and AMS. It does not include any hardware, but summarises all data exchanged and defines the control functions required by the engine.
The DENIS-9520 Specification consists of two sets of documents:

5.3.1 DENIS-9520 Interface Specification

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

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

5.3.2 DENIS-9520 Propulsion Control Specification

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

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

5.4 Propulsion Control Systems

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

- Remote Control System (RCS)
- Electronic Speed Control System
- Safety System
- Telegraph System

The Safety and the Telegraph Systems work independently and are fully operative even with the RCS out of order.

Approved RCS and Electronic Speed Control System

WinGD have an agreement with the marine automation suppliers listed in Table 5-1 concerning development, production, sale and servicing of the RCS and the Electronic Speed Control and Safety Systems. All approved control systems listed in this table comprise the same functionality specified by WinGD.

Table 5-1 Suppliers of RCS and Electronic Speed Control System

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 Phone +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 Phone +81 78 967 5361 www.nabtesco.com	**M-800-V**	**MG-800 FLEX**
Wärtsilä Lyngsø Marine A/S			
Wärtsilä SAM Electronics GmbH Behringstrasse 120 D-22763 Hamburg / Germany	www.sam-electronics.de	**Wärtsilä NACOS PCS Platinum**	**EGS2200RTf**
Wärtsilä Lyngsø Marine A/S 2, Lyngsø Allé DK-2970 Hørsholm / Denmark	jan.overgaard@wartsila.com Phone +45 45 16 62 00 www.wartsila.com/lyngsoe		

Modern Remote Control Systems consist of electronic modules and operator panels for display and order input in the Engine Control Room (ECR) and on the bridge (see Figure 5-3, 5-5). The different items normally communicate via serial bus connections. The engine signals described in the DENIS-9520 Specification are usually connected via terminal boxes on the engine with the electronic modules placed in the ECR.

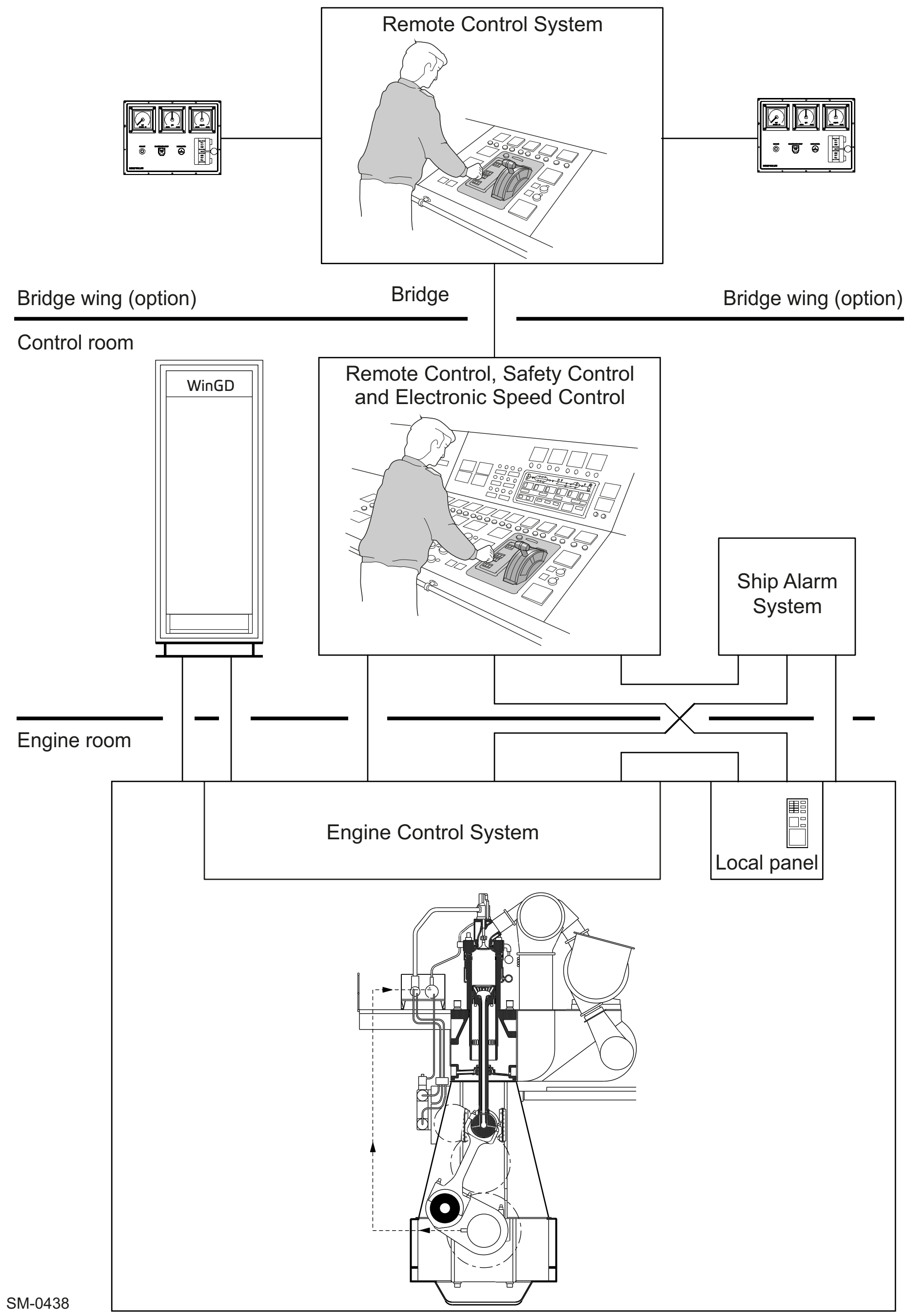


Figure 5-3 Remote Control System

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

5.4.1 PCS functions

Remote Control System

Main functions

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

Indications

- The RCS is delivered with control panels for local, ECR and bridge control, including all necessary order input elements and indications, e.g. push buttons / switches and indication lamps or alternatively a respective display.
- The following conditions in the engine are specified by the DENIS-9520 standard to be indicated as a minimum:
  - In the control room:
    - Starting air pressure
    - Engine speed
    - Revolutions
    - Operating hours
    - Load
    - Turbocharger speed
    - Scavenge air pressure in air receiver
  - On the bridge:
    - Starting air pressure
    - Engine speed
  - In addition to these indications, the RCS applied to the common rail system engine includes displaying the primary values from the ECS, like fuel pressure, servo oil pressure, etc.

Electronic Speed Control System

Main functions

- Keeps the engine speed at the set-point given by the RCS.
- Sends fuel command to the ECS.
- Limits fuel amount in function of charge air and measured speed for proper engine protection.

To avoid compatibility problems and increased engineering effort, WinGD recommends to apply RCS and Speed Control Systems of the same supplier.

Traditionally, the Electronic Speed Control System was considered a part of the main engine and was therefore usually delivered together with the engine. With the introduction of the ECS and DENIS-9520, the Electronic Speed Control System is assigned to the PCS and shall therefore be delivered along with the corresponding RCS and other components of the propulsion control package by the party responsible for the complete PCS, i.e. in most cases the shipyard.

The details concerning system layout, mechanical dimensions of components and information regarding electrical connections have to be gathered from the technical documentation of the respective supplier.

Safety System

Main functions

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

Telegraph System

- Order communication between the different control locations

Local manual control

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

ECR manual control panel

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

Options

- Bridge wing control
- Command recorder

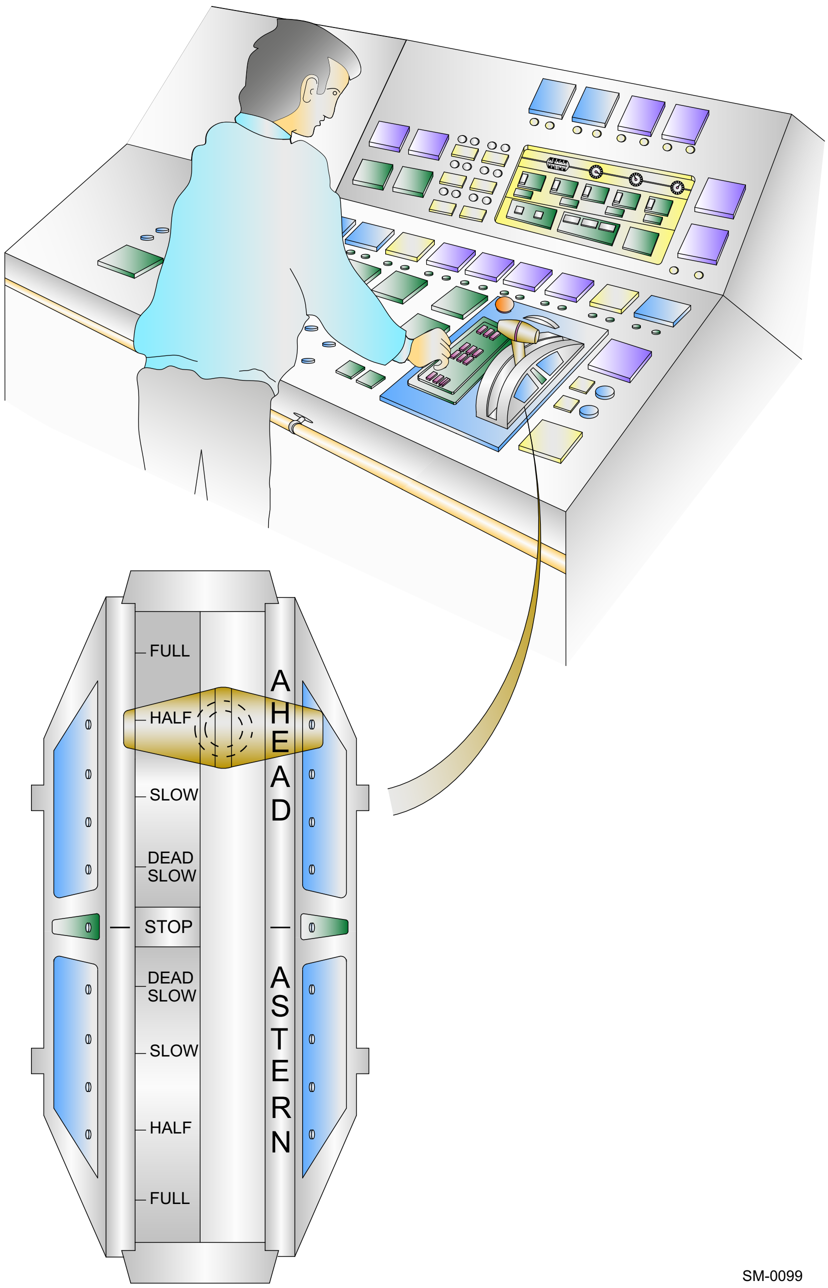


Figure 5-4 Propulsion Control

5.4.2 Recommended manoeuvring characteristics

The vessel speed is adjusted by the engine telegraph. Manoeuvring, e.g. for leaving a port, is available from full astern to full ahead. For regular full sea operation, the engine power can be further increased up to 100% CMCR power.

To protect the engine, any increase or decrease in power is limited by a rate of change, considering the warm-up respectively cool-down times. Therefore, depending on the magnitude of any change in power, it takes time to reach the required engine output; see the following graphs and tables.

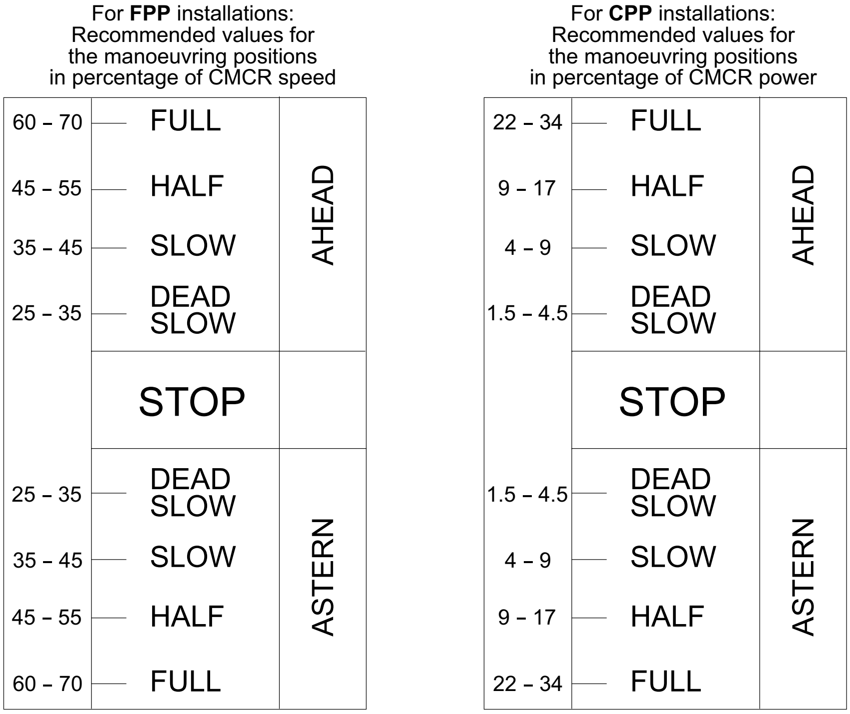


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Figure 5-5 Manoeuvring speed/power settings for FPP/CPP installations

FPP manoeuvring steps and warm-up times

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

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

Manoeuvring position	Recommended CMCR speed [%]	Corresponding power [%]	Recommended warm-up time per load step [min]	Min. warm-up time per load step [min]
DEAD SLOW	25 - 35	1.5 - 4.5	0	0
SLOW	35 - 45	4 - 9	0	0
HALF	45 - 55	9 - 17	0.1	0.1
FULL	60 - 70	22 - 34	0.5	0.5
FULL SEA 1	92	78	15	12
FULL SEA 2	100	100	32	24

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

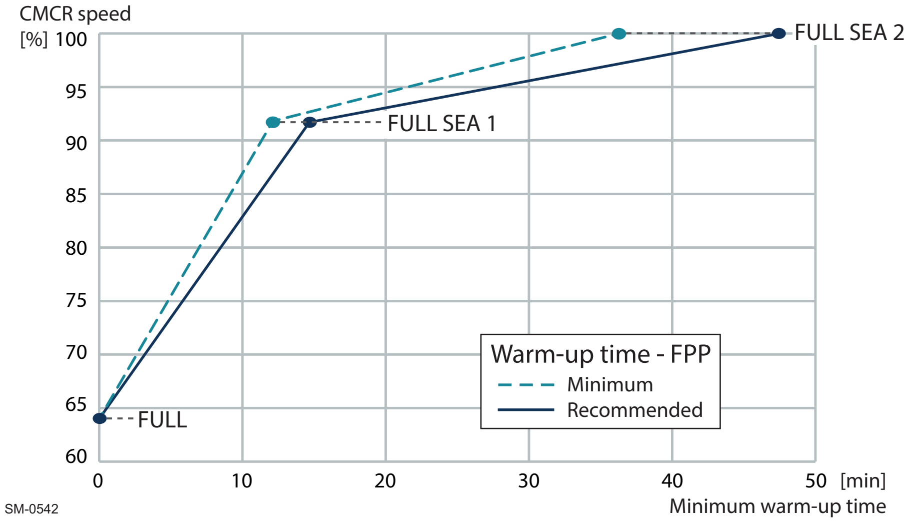


Figure 5-6 Full sea load steps in FPP load-up program

CPP manoeuvring steps and warm-up times

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

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

Manoeuvring position	Recommended CMCR power [%]	Recommended warm-up time per load step [min]	Min. warm-up time per load step [min]
DEAD SLOW	1.5 - 4.5	0	0
SLOW	4 - 9	0	0
HALF	9 - 17	0.1	0.1
FULL	22 - 34	0.5	0.5
FULL SEA 1	78	15	12
FULL SEA 2	100	32	24

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

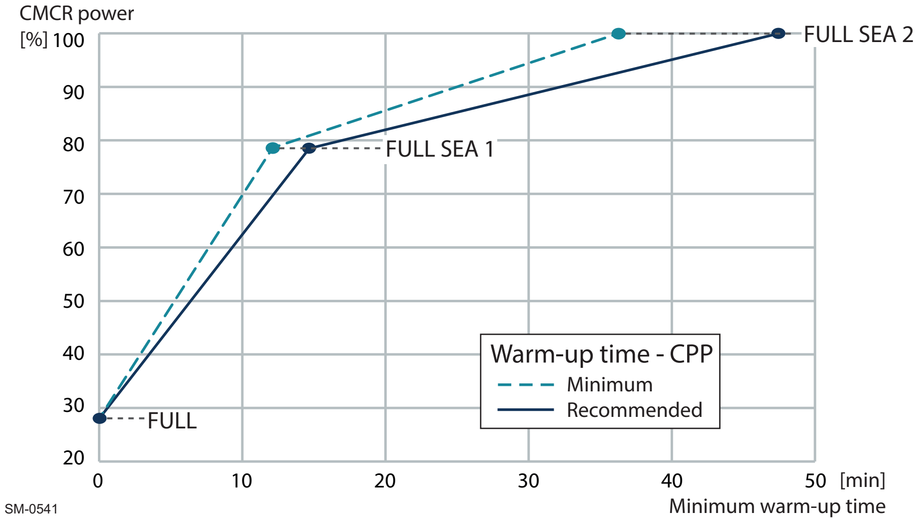


Figure 5-7 Full sea load steps in CPP load-up program

5.5 Alarm and Monitoring System

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

5.5.1 Integrated solution

PCS and AMS from same supplier

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

5.5.2 Split solution

PCS and AMS from different suppliers

- The PCS is connected to the ECS through two redundant bus lines (CANopen or Modbus, depending on automation maker).

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

With this solution the HMI is split as well:

- The Remote Control System includes the following functions:
  - Changing of parameters accessible to the operator
  - Displaying the parameters relevant for engine operation

- The Alarm and Monitoring System includes the display of:
  - Flex system parameters, like fuel pressure, servo oil pressure, etc.
  - Flex system alarms provided by the ECS

- WinGD provide Modbus lists specifying the display values and alarm conditions as part of the DENIS-9520 Specification.

5.6 Alarm sensors and safety functions

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

5.6.1 Signal processing

Signal processing has to be performed in the Alarm and Monitoring System. WinGD provide a separate document named 'Usual values and safeguard settings', which lists the signal indication values. This includes the alarm function and alarm level with corresponding setting and response time.

The document **Usual values and safeguard settings** for RT-flex58T-E can be found under the following link:
Usual values and safeguard settings

Please note that the signalling time delays given in this document are maximum values. They may be reduced at any time according to operational requirements. When decreasing the values for slow-down times, the delay times for the respective shut-down functions are to be adjusted accordingly.

NOTE	The delay values are not to be increased without the written consent of WinGD.

5.6.2 Requirements from WinGD and classification societies

The scope of delivery of alarm and safety sensors has to cover the requirements of the respective classification society, WinGD, the shipyard and the owner. (List of classification societies see Appendix, section 9.1, 9-1.)

The alarm sensors and safety functions listed in the document *Usual values and safeguard settings* are mandatory for an unattended machinery space and reflect the minimum requirements by WinGD. Additional requirements from the classification societies see Table 5-4.

NOTE	In case the engine has been configured for attended machinery space, please consult the respective classification society.

Table 5-4 Additional Class requirements for alarm sensors and safety functions

Table under preparation

5.7 WinGD Integrated Digital Expert

The WinGD Integrated Digital Expert (WiDE) is a comprehensive system that enables engine monitoring and assistance.

The DCM (Data Collection Monitoring) system collects large amounts of data from engine's onboard sensors and signals, while EDS (Engine Diagnostic System) analyses these data, monitoring the engine performance, predicting component malfunctions and supporting the crew with live troubleshooting and diagnostic.

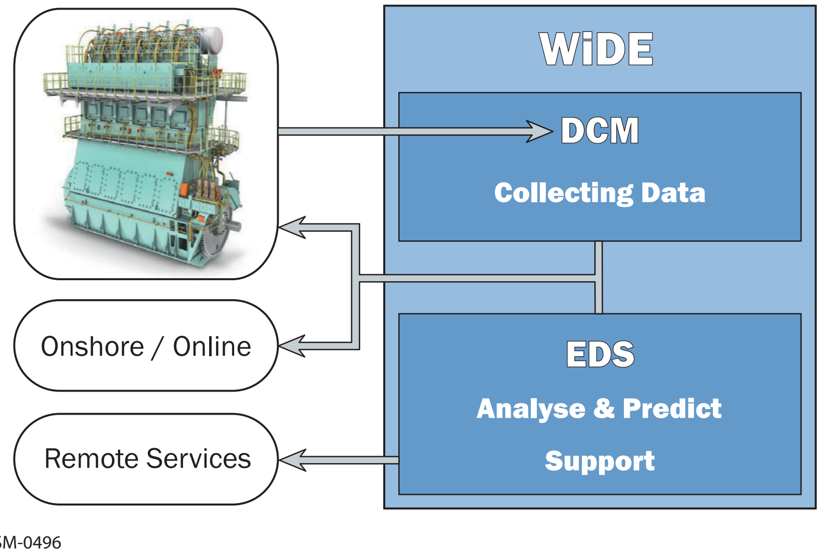


Figure 5-8 WiDE system

5.7.1 Data Collection and Monitoring

WiDE uses the DCM unit to collect data from the engine and other ship systems. These data are displayed and stored, both on board the ship and remotely. Using a secure communication channel, DCM allows data to be sent to an external server of the customer's choice. All the data are stored for the engine lifetime and are accessible for the customer by a protected user account.

With the permission of the customer, DCM enables WinGD to review ship company data when required, allowing the first step towards remote support collaboration.

The DCM system is included in the standard engine scope.

5.7.2 Engine Diagnostic System

WiDE uses the EDS to analyse and compare data against predefined references, identifying any discrepancies between 'reference' and 'actual' data. The findings are then consolidated, and any predicted issues or required actions are highlighted to the crew. This allows for improved planning and automation of subsequent supporting processes like troubleshooting, maintenance, spare parts handling and identification.

Analysis

EDS analyses the DCM data to obtain a full engine diagnostic by using several methods:

- The 'digital twin', an engine thermodynamic model which is calibrated at the engine's shop test and sea trial, receives and simulates real-time engine parameter inputs. This provides an ongoing performance assessment by measuring any deviations between the simulated model and the real engine.
- An algorithm rule set is used to monitor, analyse and diagnose the subsystem components. It encapsulates WinGD's expert knowledge by considering calculations, documentation and investigations.
- Data trends are recognised from a combination of DCM monitoring and historical sample data that is uploaded to the EDS during installation, enabling progressive performance improvement.

Prediction and troubleshooting

This ongoing analysis is consolidated to a final output, allowing for EDS to make predictions. If a potential 'critical' fault is diagnosed the crew is notified, enabling them to decide on immediate action. However, if no action is taken EDS will proceed with the troubleshooting process in the following sequence:

- At the initial stage all proposed actions largely follow the directions of the engine Operating Manual.
- Followed by recommendation procedures defined by the Maintenance Manual, EDS may also connect to the Planned Maintenance System (PMS), updating the Maintenance Plan and the Spare Parts List.
- If required, EDS will connect to external help and support, potentially engaging with human experts through on-line and remote troubleshooting.
- Once the risk is resolved, EDS collects all the available feedback and creates relevant reports archiving it for future references.

Support

As well as the actions mentioned above, when EDS responses to single-case instances or potential issues, it also provides ongoing supports other ways:

- The troubleshooting module keeps track of current issues, collecting and displaying data.

- The remaining life expectancy of components is tracked from performance, operation and maintenance data. This means that the maintenance scheduling becomes dynamic, updating and preparing before failure.

- Planned maintenance and associated Spare Parts Lists are clearly displayed in the spare parts modules. With the integrated WinGD Spare Parts Catalogue, component information and creating orders is simplified.

Software availability WinGD provide the EDS software for a limited 'free-trial' period on all their engines. If there are any questions regarding this, please contact the WinGD representative.

5.7.3 WiDE installation process

The process map below shows the steps towards installation of DCM and EDS against various testing points.

The process order map seen in Figure 5-9 ensures effective installation. The DCM is installed before shop test as the data it collects provides the information needed for the EDS tuning. This process order ensures that the WiDE system is fully operational by the time of the vessel's maiden voyage.

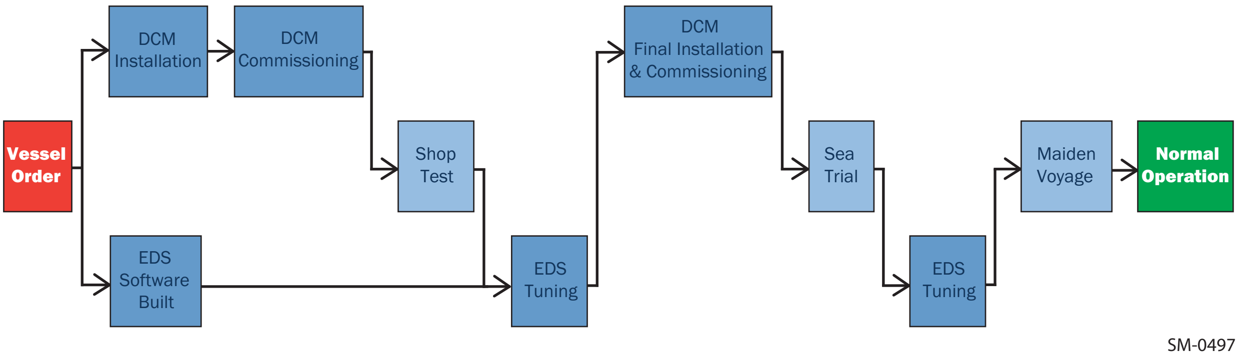


Figure 5-9 WiDE installation process map

6 Engine Dynamics

It is critical that vibration is minimised throughout the design and construction stage of any engine installations. The assessment and reduction of vibration are subject to continuous development and research, requiring expert knowledge. For successful design, vibration behaviour calculations are required over the whole operating range of the engine and the propulsion system. As such, WinGD have developed extensive computer software, analytical procedures and measuring techniques.

NOTE	WinGD provide additional support services to assist with system dynamics and vibration analysis. See section 6.9, 6-21 for info about forms and links.

Forces and moments causing vibrations

Within the engine, various forces and moments are generated by the reciprocating and rotating masses. Often these cyclical forces and moments are neutralised by counterbalancing within the engine. However, if this is not achieved the engine will experience the sum of these forces and moment as external responses, reacting around its own axis and causing vibrations outside of the engine. Vibrations are problematic, especially if a vibration frequency forces a resonance, causing an amplitude to pass acceptable limits. This section highlights the importance of dynamic consideration, the causes and relevance.

After considering the external forces and moments types, this section explores the resulting vibration, along with recommended considerations and countermeasures relevant to engine type and other associated systems and design features.

Types of vibration

The vibration types considered in this section are as follows:

- External mass forces and moments
- External lateral forces and moments (Lateral engine vibration or 'rocking')
- Longitudinal engine vibration
- Torsional vibration of the shafting
- Axial vibration of the shafting
- Whirling vibration of the shafting
- Hull vibration

Dynamic characteristics data

The external forces and moments generated by a specific engine defines its dynamic characteristics. These must be considered throughout the design process of the vessel to avoid adverse impact on the vessel.

In the document **External forces and moments** WinGD provide a complete list of the external forces and moments for each engine type. The latest version of this document is provided on the WinGD corporate webpage under the following link:
External forces and moments

The external forces and moments for engines not tuned to an R1 rating are available on request.

6.1 External mass forces and moments

The external mass forces and moments are the resulting forces and moments produced by reciprocating and rotating masses of the running gear (i.e. the engine's main oscillating masses) that are transmitted to the surrounding vessel via the foundation. This therefore doesn't consider forces and moments that are produced by combustion forces — see section 6.2, 6-5. The external mass forces and moments depend on the design of a specific engine and the engine speed. The engine power and tuning has no influence on the external mass forces and moments.

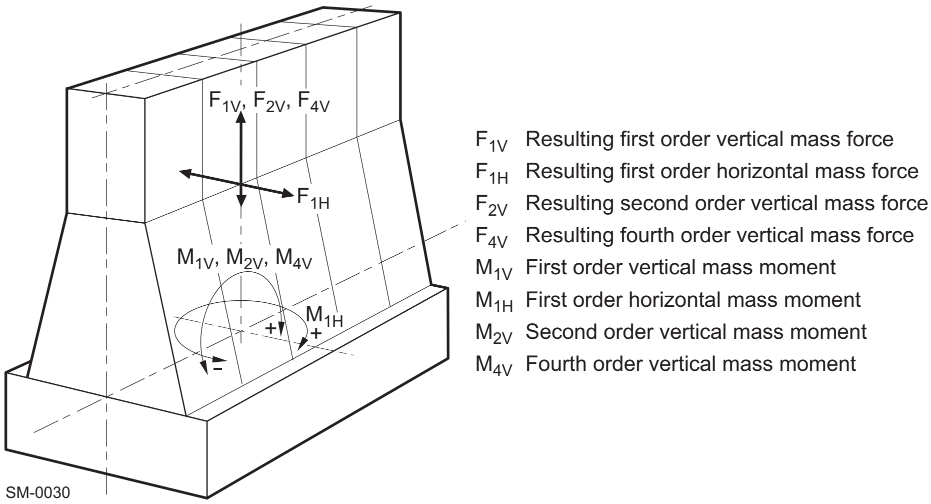


Figure 6-1 External mass forces and moments

Figure 6-1 shows the mass forces and moments generated by the engine. However, where possible these are neutralised. If not, cyclical generation of the external mass forces and moments may lead to unwanted and disturbing vibrations throughout the vessel. This highlights the importance of using countermeasures that balance out the generated mass forces and moments where possible.

6.1.1 Balancing of mass forces and moments

Forces

With a regular firing order of evenly distributed crank angles, an engine will inherently balance the summation of all vertical (F_V) and horizontal (F_H) free forces. Sometimes the firing order is designed to be irregular, i.e. unevenly distributed crank angles, to optimise the overall vibration characteristic of a specific engine type. Regardless, the resulting mass forces are considered to be negligible.

First order moments

First order mass moments (M_{1V} and M_{1H}) can be reduced to acceptable levels by introducing standard counterweights, fitted to the ends of the crankshaft. In special cases non-standard counterweights can be used to reduce either vertical (M_{1V}) or horizontal (M_{1H}) first order mass moments as required.

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}.

Second and fourth order moments

Second (M_{2V}) and fourth (M_{4V}) order vertical mass moments are also generated, although these magnitudes will vary depending on engine type, tuning, and number of cylinders. Unless a problematic vessel design leads to unfavourable vibration, there is normally no cause for concern for engines with 7 cylinders or more. However, 5 and 6-cylinder engines are known to generate high magnitudes of unbalanced second order vertical mass moments (M_{2V}) and should therefore be carefully considered. Consequently, for 5 and 6-cylinder engines WinGD strongly recommend that the impact of the second order vertical mass moment on the vessel is carefully checked. In cases where the investigation reveals a possible problem, WinGD recommend to consider the installation of one of the following countermeasures, designed to reduce the effects of second order vertical mass moments to acceptable values.

6.1.2 Countermeasure for second order vertical mass moments

Electrically driven compensator (external compensator)

If disturbing second order vibrations occur on 5 and 6-cylinder engines, WinGD strongly recommend that an electrically driven compensator is fitted or retrofitted to the ship's structure. As seen in Figure 6-2, such a compensator is usually installed in the steering gear compartment. It is tuned to the engine operating speed and controlled accordingly.

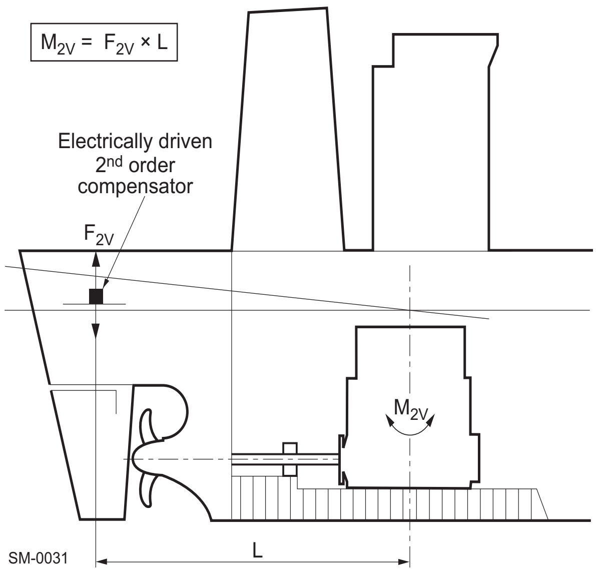


Figure 6-2 Locating an electrically driven compensator

This countermeasure should also be considered for other cylinder number engines if the second order vertical mass moments (M_{2V}) surpass the necessary limits. However, suitability will vary for different engines and vessel design.

Power related unbalance

The power related unbalance (PRU) values can be used to estimate the risk of unacceptable levels of hull vibrations caused by external mass moments of first and second order. The PRU is calculated with the following formula:

$$PRU = \frac{M_x\,(Nm)}{Engine\ Power\,(kW)}$$

Formula 6-1 Power related unbalance calculation

where:

PRU .... = power related unbalance

M_x .... = M_{2V} (typically), M_{1H} & M_{1V} (considered as well)

The M_x and the resulting PRU values of an engine are dependent on the number of cylinders and tuning option.

The resulting PRU values are graphically displayed in the link below, along with the likelihood of needing a compensator.

External forces and moments

The link provides dynamic characteristics data for engines tuned at R1 rating. For other engine speeds, the corresponding external mass moments are calculated with the following formula:

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

Formula 6-2 External mass moments calculation for R_x rating

where:

$M_{x(CMCR)}$ ... = resulting moments for a specific engine's CMCR

$M_{x(R1)}$ .... = moments for engine at R1 rating

n_{CMCR} .... = speed of engine for a specific engine's CMCR

n_{R1} .... = speed of engine at R1 rating

6.2 External lateral forces and moments

The external lateral forces and moments (lateral engine vibrations resulting in ‘rocking’) are generated by the combustion process and to a small extent by the reciprocating masses of the running gear. The lateral forces depend on the CMCR, tuning, and engine speed.

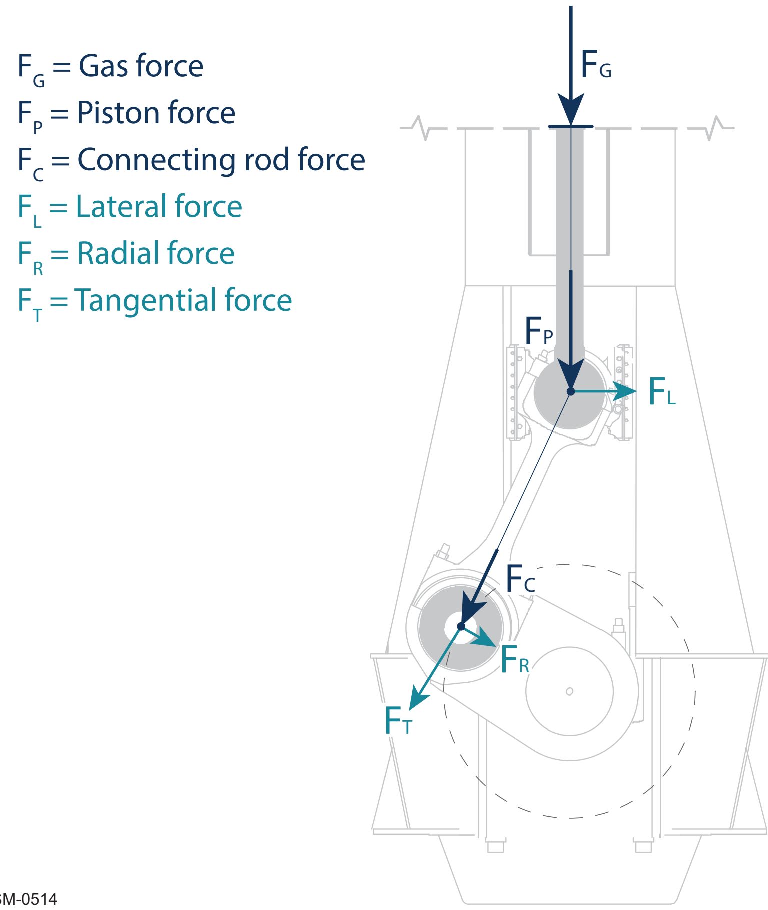


Figure 6-3 Forces through the engine

The forces between the piston and the connecting rod reaction cause a lateral force to act on the crosshead guide rails. The lateral forces at the guide rails are transmitted to the engine block and to the foundation.

The resulting lateral forces and moments may excite resonances of the combined engine and foundation system. In addition, hull resonances or local vibrations in the engine room may be generated.

6.2.1 Lateral vibration types

The resulting lateral forces and moments generate two different modes of lateral engine vibration, the H-type and X-type vibration; refer to Figure 6-4.

The table of H-type and X-type vibration values is also provided in the link below:
External forces and moments

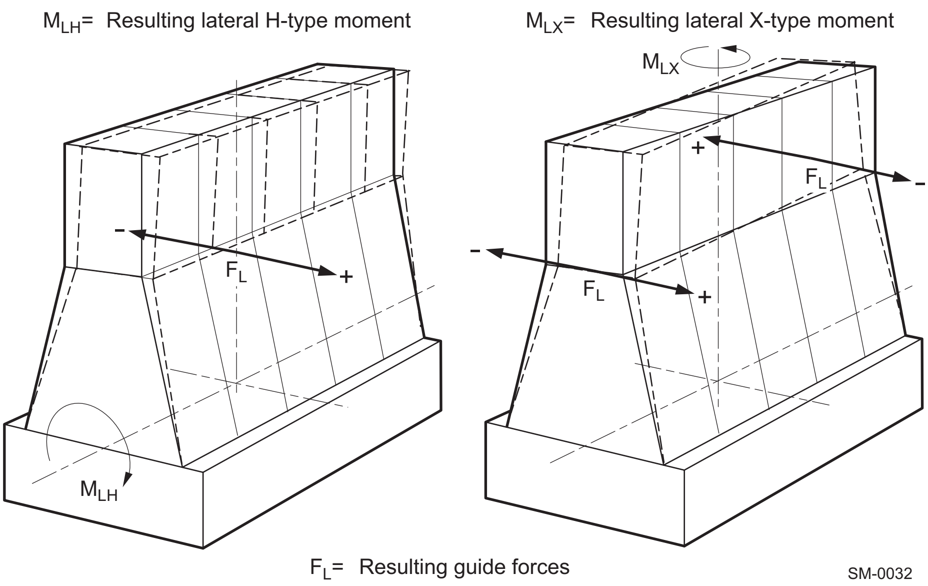


Figure 6-4 Lateral vibration — X-type and H-type

H-type vibration

H-type lateral vibrations are characterised by a mode shape where both sides of the top of the engine, the driving and free end, vibrate together, in phase. The lateral guide forces (F_L) result in a lateral moment, expressed as resulting lateral H-type moment (M_{LH}).

X-type vibration

X-type lateral vibrations are characterised by a mode shape where at the top of the engine, the driving and free ends vibrate in counter-phase to each other. As these resulting lateral guide forces create opposing axial moments at the two ends of the engine, the X-type lateral vibrations are expressed as a moment around the vertical axis, the resulting lateral X-type moment (M_{LX}).

6.2.2 Reduction of lateral vibration

The amplitudes of the vibrations transmitted to the hull depend on the design of engine seating, frame stiffness and exhaust pipe connections. As the amplitude of the vibrations cannot be predicted with absolute accuracy, the support to the ship's structure and the space needed to install stays should be considered in the early design stages of the engine room structure. This is true for both lateral and longitudinal vibrations, which is further discussed along with relative reduction methods in the following sections.

NOTE	WinGD recommend a stiff engine foundation design in the longitudinal and lateral directions, as this is always best practice in minimising hull vibrations.

Lateral hydraulic stays

If needed, lateral stays must be fitted between the upper engine platform and the ship hull to avoid harmful resonance conditions. The main effect of lateral stays is to shift the resonance frequency sufficiently above nominal speed. In addition, some damping effect is provided by the hydraulic stays. Such hydraulic stays can be either for both-side or one-side installation.

- Hydraulic stays for one-side installation have two oil chambers (one on each side of the piston) and provide in this regard a 'damping effect' in both directions.
- Hydraulic stays for both-side installation (defined by WinGD design) have an oil chamber on one side of the piston and an air chamber on the other side. The air chamber provides little to no damping effect.

The two general arrangements for both stay types are shown in Figure 6-5 below and Figure 6-6, 6-8.

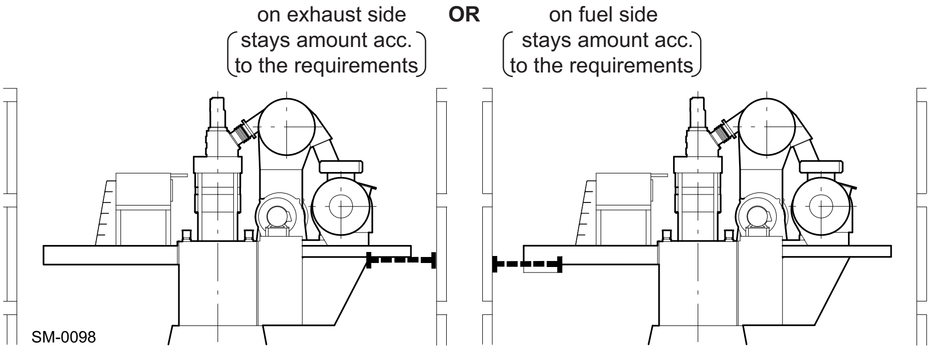


Figure 6-5 General arrangement of hydraulic stays for one-side installation

NOTE	The shipyard must have confirmation from the hydraulic stay maker acknowledging its suitability for one-side installation on the engine.

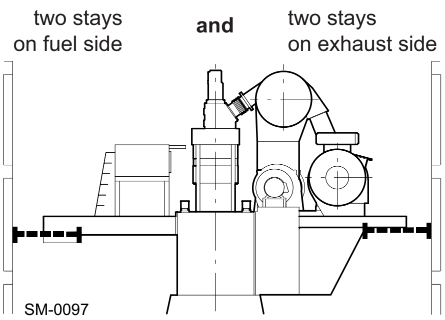


Figure 6-6 General arrangement of hydraulic stays for both-side installation

Hydraulic stays of WinGD design

WinGD provide instructions for both-side installation when using WinGD type stays. Please refer to the **Assembly Instruction** (DG 9715), which can be found on the WinGD corporate webpage under the following link:

Assembly instruction - Hydraulic lateral device

NOTE	The use of friction type lateral stays is no longer supported by WinGD.

Electrically driven compensator

If lateral stays cannot be installed, the following can be used to reduce lateral engine vibrations:

- For H-type mode, one electrically driven compensator can be installed on the upper platform in the longitudinal centre point of the engine. This reduces the lateral engine vibrations and the effect on the ship's superstructure. Alternatively, two compensators, one fitted at each end of the engine upper platform, can be applied, synchronised in phase.

- For X-type vibration, two compensators, one fitted at each end of the engine upper platform, are necessary and are synchronised in counter-phase.

It must be noted that electrically driven compensators can only compensate one harmonic excitation frequency at a time.

6.3 Longitudinal vibration (pitching)

Strong axial vibrations in the shafting, transmitted from the thrust bearing to the engine structure and foundation, can excite inadmissible longitudinal vibrations at the engine top and as a consequence in the superstructure (refer to section 6.5, 6-14). In any case, to prevent such vibrations, the double-bottom structure should be as stiff as possible in the longitudinal direction.

Reduction of longitudinal vibration (5-cylinder engines)

In general, longitudinal stays are only especially needed on 5-cylinder engines when a shafting layout has a main torsional critical speed above the nominal speed, resulting in strong longitudinal engine vibrations at the top of the engine block. This tends to only occur by using intermediate shafts with oversized diameters, which is no longer common today.

Longitudinal stays to prevent vibration in superstructure

However, there can be installations where it might be beneficial to install longitudinal stays. This is not because of inadmissible longitudinal vibrations at the engine top but because of possible disturbing longitudinal vibrations in the superstructure, close to nominal speed. By fitting longitudinal stays the disturbing resonance can be shifted above nominal speed.

The decision if longitudinal stays are needed or not has to be made by the shipyard based on a global ship vibration investigation, or on vibration measurements taken at the top of the engine block and in the superstructure (on the first vessel of a series).
They are arranged as shown in Figure 6-7.

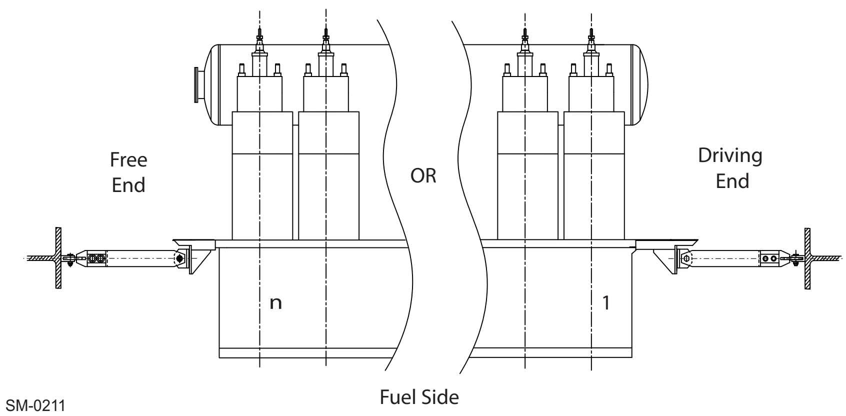


Figure 6-7 Arrangement of longitudinal stays

The following types of longitudinal stays can be applied:

Hydraulic type stays

Hydraulic stays can be installed to either the free end or the driving end side of the engine according to the design and requirements of WinGD or third-party suppliers, as defined in the MIDS drawing (DG 9715) seen below.

Friction type stays

Friction stays can be installed according to WinGD design or third-party maker design, to either the engine free end or driving end side.
The layout of friction type stays must conform to the drawing ‘Engine stays / friction type’ and the associated friction stays drawings (see below link to MIDS drawing). Deviations are not acceptable, especially the friction coefficient of the shim and the disc spring properties, which must follow exact specification.

Docu WinGD

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

Docu WinGD

The layout of friction type and hydraulic type stays must be as shown in the drawing ‘Engine stays’ in the *MIDS* (DG 9715).

6.4 Torsional vibration

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

The shafting system comprises of the crankshaft, propulsion shafting, propeller, engine running gear, flexible couplings, and power take-off (PTO). The complete assembly of the shafting system must be considered when determining the torsional loads in the system components.

Torsional vibration calculation (TVC)

The torsional loads in the system components are determined by performing a torsional vibration calculation (TVC). The TVC must be done in the early stage for every project.
Across the engine's speed range, all system components must remain within their corresponding torsional vibration load limits. If in a component, the torsional loads exceed the corresponding limit, appropriate countermeasures have to be applied.
The calculation normally requires approval by the relevant classification society and may require verification by measurement on board ship during sea trials. All data required for torsional vibration calculations should be made available to the engine supplier in an early design stage (see section 6.9, 6-21).

Barred speed range (BSR)

At a certain speed range the torsional vibration stresses in the shafting may exceed the limits for continuous operation. If this occurs, a barred speed range (BSR) must be defined. The width of the BSR is defined by the classification society. The BSR must be passed through rapidly and some classification societies have defined rules about the maximum permissible passage time through the BSR. In general, the target is to have a maximum passage time of 30 seconds.
In order to guarantee a rapid passage of the BSR, a minimum power margin of 10% at the upper boundary of the BSR must be present. The power margin is defined as the margin between the bollard pull curve and the engine torque limit.
In any case, within the BSR the torsional shaft stress must not exceed the transient limit, otherwise other appropriate countermeasures have to be taken.

6.4.1 Reduction of torsional vibration

Excessive torsional vibration can be reduced by optimising the shaft diameters, selecting a different (heavier) flywheel, adding a front disc (tuning wheel) to the free end of the crankshaft or adding a torsional vibration damper to the free end of the crankshaft. A torsional vibration damper reduces the torsional stresses by absorbing part of the vibration energy.

Low-energy vibrations

Viscous damper

Where low-energy torsional vibrations have to be reduced, a viscous damper can be installed; refer to Figure 6-8. In some cases, the torsional vibration calculation shows that an additional oil-spray cooling for the viscous damper is needed. In such cases the layout must be in accordance with the recommendations of the damper manufacturer and WinGD design department. The viscosity of the silicone oil in the viscous damper must be checked periodically. The interval is specified by the damper manufacturer. For more information, refer to the Operation Manual.

High-energy vibrations

For high-energy torsional vibrations that may occur e.g. on 5 and 6-cylinder engines, a spring type damper with its tuning and damping effect may be considered; refer to Figure 6-8.

Spring damper

The spring type damper must 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 75 kW energy (depends on number of cylinders).

The oil flow to the damper is 15-24 m^3/h. An accurate value will be given after the results of the torsional vibration calculation are known.

For spring type damper installation, the application of a damper monitoring system is mandatory.

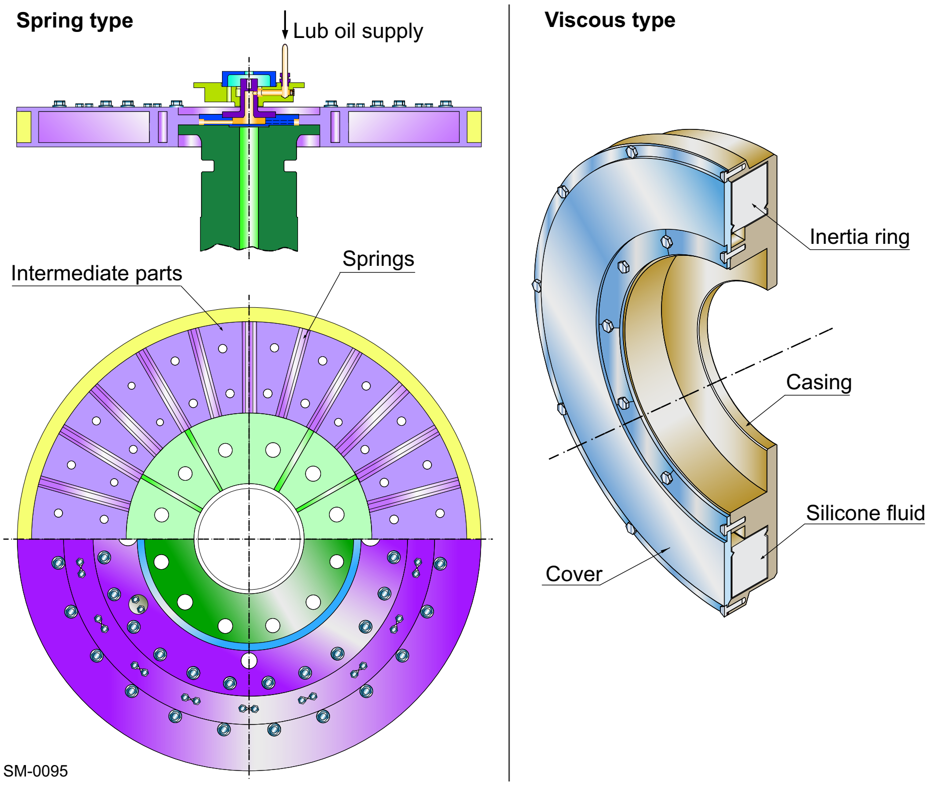


Figure 6-8 Vibration dampers (spring type and viscous type)

6.4.2 PTO/PTI systems effect on torsional vibration

A propulsion plant may include a main-engine driven generator (PTO, power take-off), a shaft-line connected electric motor (PTI, power take-in), or both. These elements are connected to the engine or shafting by clutches, gears, shafts and/or elastic couplings. See Figure 4-22, 4-56 for the different arrangements.

Installations with PTO or PTI require special attention in the early stages of a project. These systems may cause torsional vibrations and alignment challenges.

Risk of instable engine speed

For many PTO / PTI systems that use elastic couplings, the lowest torsional natural frequency can be problematic if it is below approximately 1.5 Hz. Here, there is a risk of engine speed instability where the engine constantly adjusts its speed to compensate the rotating vibration; this must be considered and compensated for in the engine speed control system.

Installation of MFD

In addition, such PTO / PTI systems are very sensitive to misfiring as varying firing loads can cause inadmissible torsional vibrations. To protect the elastic couplings and gears from any misfiring, a misfiring detection device (MFD) must be installed. This indicates either partial or total misfiring, allowing for appropriate countermeasures (e.g. speed reduction, de-clutching of PTO / PTI branch) to be applied automatically, protecting the PTO / PTI components.

For additional consideration about PTO / PTI application refer to section 4.10, 4-55, and for support regarding system layout, please contact WinGD.

6.5 Axial vibration

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

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

As the shafting system is made up of masses and elastic connections, it is capable of vibrating and resonating at several frequencies. This would result in excessive stress in the crankshaft and in some cases can lead to excessive vibration of the upper part of the engine.

Reduction of axial vibration

Axial vibration damper

To limit the influence of axial excitations and reduce the level of vibration, all present WinGD engines are equipped with an integrated axial vibration damper. In most cases, this lowers the axial vibrations in the crankshaft to acceptable values, meaning no further countermeasures are required. No excessive axial vibrations occur, neither in the crankshaft, nor in the upper part of the engine.

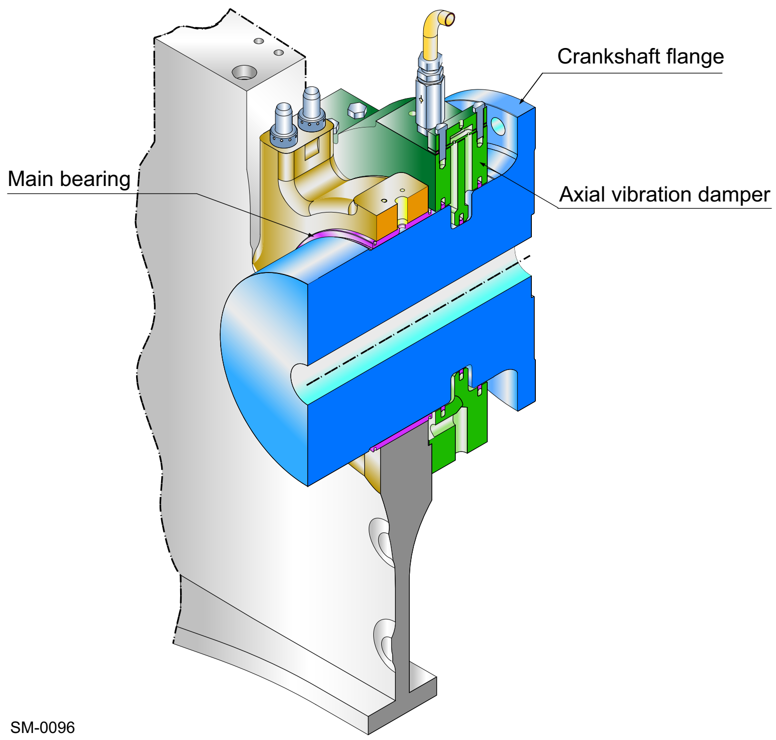


Figure 6-9 Example of axial vibration damper

The integrated axial vibration damper is mounted at the free end of the crankshaft. It is connected to the main lubricating oil circuit. An integrated oil pressure monitoring system continuously checks the correct operation of the axial vibration damper.

6.6 Whirling vibration

Whirling vibrations are generated when the shaft rotates and goes into transverse oscillations. If the shaft is out of balance, the resulting centrifugal forces will induce the shaft to vibrate. This vibration is commonly known as *whirling vibration*, *bending vibration* or *lateral shaft vibration*.

Whirling vibrations are in most cases not relevant in propulsion shafting with directly coupled low-speed 2-stroke engines. Typically, whirling vibrations are only relevant in 2-stroke installations having a very long shaft line (longer than 60 m).

Many classification societies do not require whirling vibration calculations for installations with low-speed 2-stroke engines. In general, only the natural whirling frequencies are calculated. The number and position of the shaft bearings have a significant influence on the natural frequencies. As such, the whirling vibration calculation must be performed after or together with the alignment calculation.
Alignment Guidelines for Layout Calculation

6.7 Hull vibration

The hull and accommodation area are susceptible to vibration caused by the propeller, machinery and sea conditions. Controlling hull vibration from engine excitation is achieved by a number of different means and may require the fitting of second order mass moment compensators, lateral / longitudinal stays, electrical H-type or X-type compensators and/or torsional vibration dampers.

Avoiding problematic hull vibrations cannot be achieved in isolation and requires consideration and cooperation from propeller manufacturer, naval architect, shipyard, and engine builder.
WinGD can provide, on request, a simplified FE engine model to enable the shipyard or design institute to predict the influence of the engine forces and moments on the ship hull.

6.8 Countermeasures for dynamic effects

6.8.1 External mass moments and vibrations

The following tables indicate where dynamic effects and the countermeasures required to reduce them are to be given special attention.

Where installations incorporate PTO arrangements (see Figure 4-22, 4-56), further investigation is required and WinGD should be contacted.

Table 6-1 Countermeasures for external mass moments

No. of cyl.	Second order compensator
5	Balancing countermeasure is likely to be needed[a)]
6	Balancing countermeasure is unlikely to be needed[a)]
7-8	Balancing countermeasure is not relevant

a) No engine fitted second order balancer available. If reduction in M_{2V} is needed, then an external second order compensator has to be applied.

Table 6-2 Countermeasures for lateral and longitudinal vibrations

No. of cyl.	Lateral stays	Longitudinal stays
5	A	B
6	B	C
7	C	C
8	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.

Table 6-3 Countermeasures for torsional and axial vibrations of the shafting

No. of cyl.	Torsional vibration	Axial vibration
5-8	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, torsional vibration 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.

6.8.2 Synchro-Phasing System in twin engines

An available countermeasure for vibration reduction in twin engine vessels is WinGD's Synchro-Phasing System (SPS). By changing the relative phase difference of the two engines operating with the same speed, it is possible to neutralise vibrations of a selected frequency and the resulting resonance on the ship's hull or structure.
Synchro-Phasing can be an effective way of reducing vibrations without modifications to the ship's structure and with limited cost.

Concept

As discussed previously in this section, it is important that vibrations of the engine or propeller are minimised. Some specific vibrations will be problematic if they cause resonance in the surrounding vessel structure.

Alteration of phase angles

With twin-engine vessels the resulting vibration amplitude is equal to the sum of both engines and is therefore dependent on the phase difference of the engines. The phase difference of any two vibrations is the relative difference (of angle or time) between any matching points of their vibrational cycles.

It is shown in Figure 6-10 that if the vibrations are in phase (0° phase difference) the resultant amplitude can reach double that of a single vibration. However, towards a phase difference of 180° the amplitude is reduced from the vibrations neutralising each other.

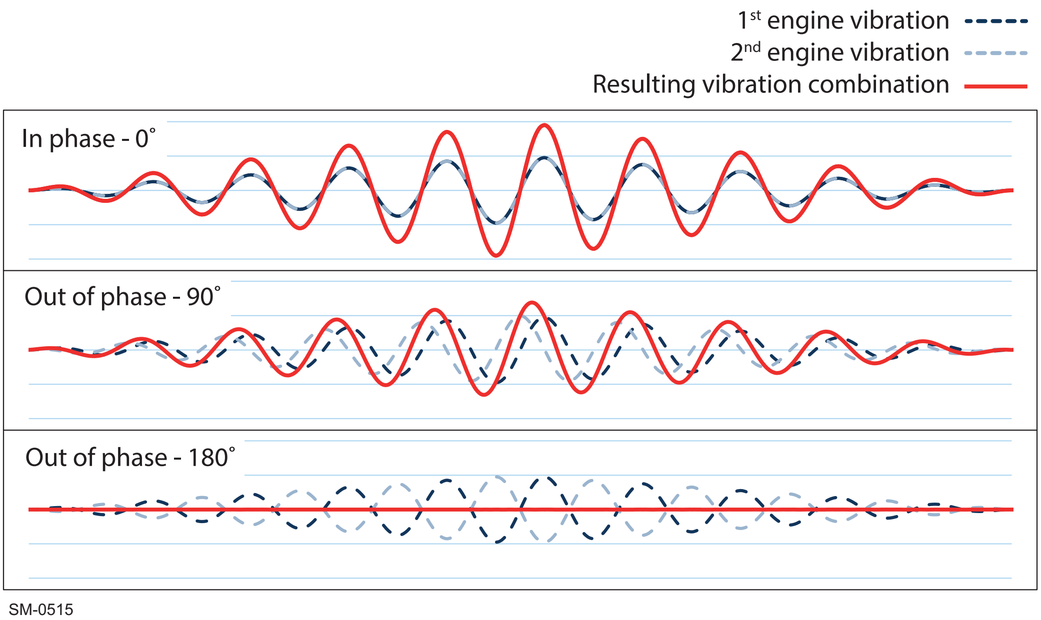


Figure 6-10 Resulting vibration from SPS combinations

By correctly altering the phase angles between two crankshafts, a vibration can be reduced and possibly eliminated, limiting vibrations distribution in the ship's hull and superstructure.

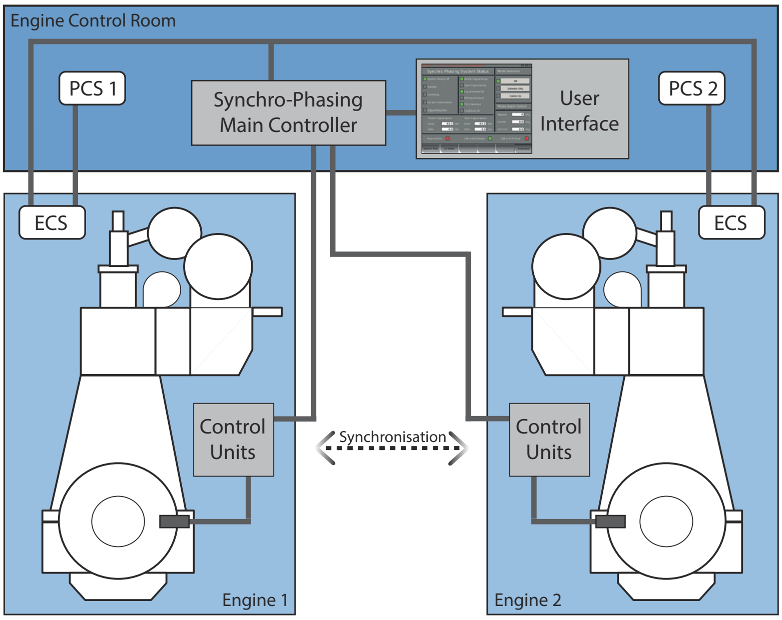
SPS is used to compensate one of the following:

- Second order vertical mass moments (M_{2V}) discussed in section 6.1, 6-2
- Lateral H-type guide moments discussed in section 6.2, 6-5
- Excitations generated by the blade frequency of the propellers

NOTE	The compensation is only capable of neutralising one selected frequency at a time.

Components and control

The speed and the crank angle of both engines are measured by control units, which are installed on the engines. These signals are continuously transmitted to the Synchro-Phasing main controller, where the relative position and speed between the two engines is measured.



SM-0516

Figure 6-11 Synchro-Phasing system

Main controller and user interface in ECR

The main controller is installed in the engine control room along with the user interface, where the relative phase difference angle (provided by WinGD vibration experts) can be entered. This enables the system to implement a closed loop control of the set-points, which are a function of the difference between the reference phase angle and current phase angle. To adjust the phase angle between the two engines, the speed of only one (slave) engine is adapted.

The additional components required are:

- Control unit (x 2)
- Synchro-Phasing main controller (x 1)
- User interface (x 1)

NOTE	Any phase angle value entered into the user interface must be previously approved by WinGD's Dynamics experts, as incorrect settings can lead to excessive vibrations.

Operating modes and restrictions

There are three operating modes:

- **Control On**
  Fully operational. This can be selected when the release conditions are fulfilled (see in the following).

- **Estimate Only**
  The speed of each engine and the relative phase angle between them is indicated and continuously updated, but not controlled. This can be selected when the release conditions are fulfilled (see below).

- **Off**
  All control parameters of Synchro-Phasing control are reduced and reset to initial value (zero). This can be selected at any time.

Release conditions

The following conditions must be fulfilled to activate the **Control On** and **Estimate Only** modes. These conditions are:

- Both engines are running in ahead direction in normal operational condition and unrestricted operation conditions (no protective actions of the engine or major failures)
- Both engines are operated within the speed conditions of the Synchro-Phasing system
- Heavy Sea Mode is not active on either engine
- No major failure of phase angle control system is active
- Both engines have the same speed set-point and have completed the load up / down programs to reach this speed set-point

Deactivation of the **Control On** and **Estimate Only** modes will occur if required. Synchro-Phasing is also overruled and reset if engine limitations are reached. After deactivation, **Off** will be automatically active.

6.9 Order forms for vibration calculation & simulation

WinGD provide additional support services to assist with system dynamics and vibration analysis. All questionnaires and forms can be downloaded from the WinGD corporate webpage under the following link:
Questionnaires for shaft calculations

Forms should be filled in and submitted via e-mail to the following e-mail address, along with any further questions:
dynamics.ch@wingd.com

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

7 Engine Emissions

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

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

7.1 Exhaust gas emissions

7.1.1 Regulation regarding NO_x emissions

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

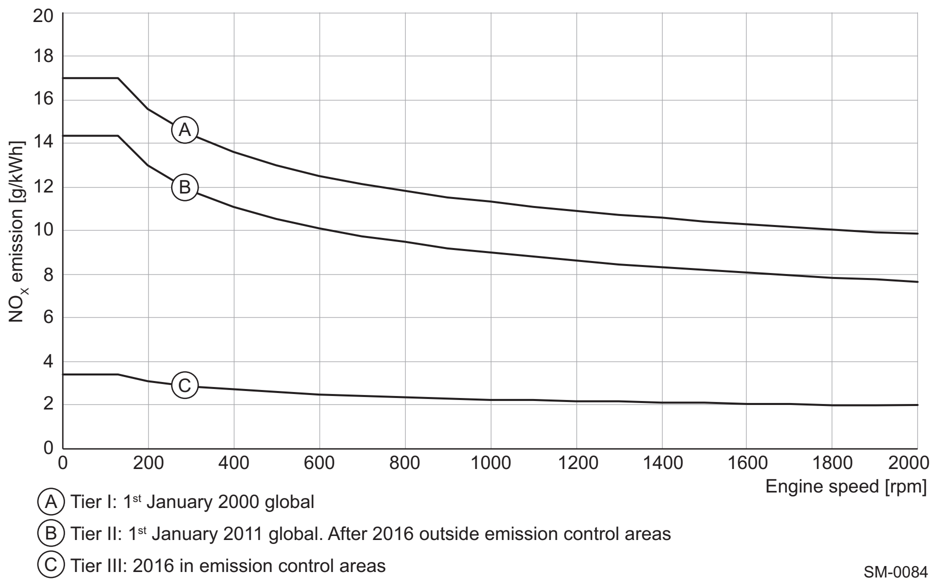


Figure 7-1 Speed dependent maximum allowable average of NO_x emissions

NO_x Technical Code

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

7.1.2 Selective catalytic reduction

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

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

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

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

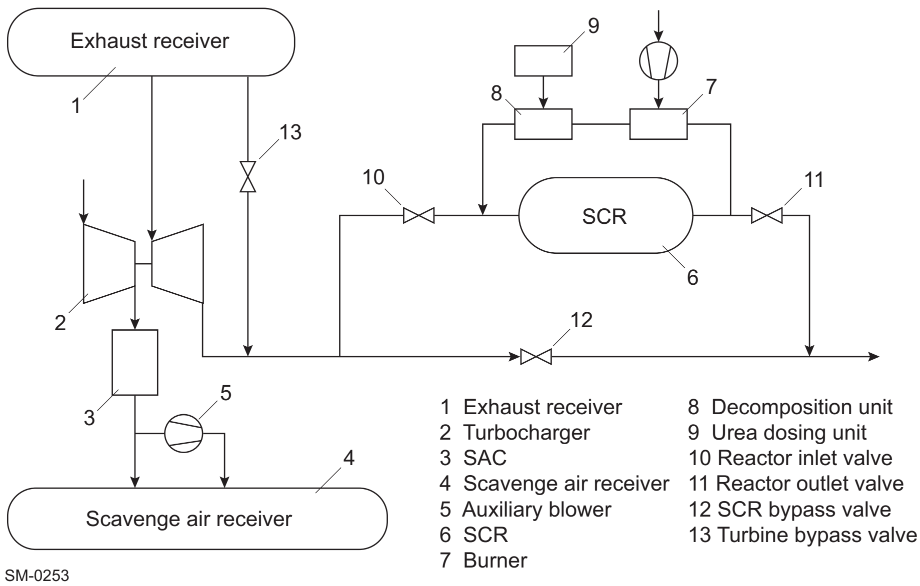


Figure 7-2 Low-pressure SCR — Arrangement

High-pressure SCR

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

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

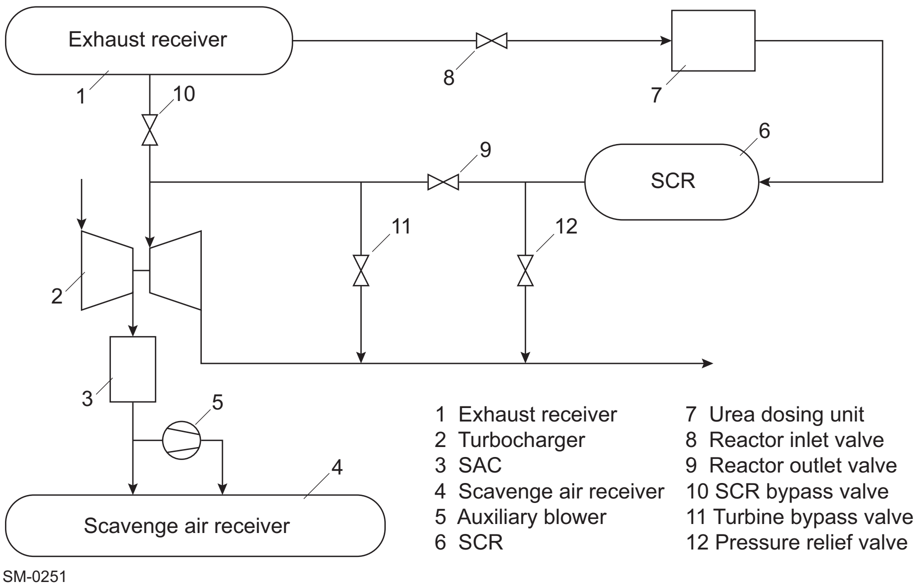


Figure 7-3 High-pressure SCR — Arrangement

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

7.2 Engine noise

As the ship's crew / passengers must be protected from the effects of machinery space noise, the maximum acceptable noise levels are defined by rules. In general, for new building projects, the latest IMO Resolution MSC.337 'Code of Noise Levels Onboard Ships' is applied.

The main change introduced by the new IMO MSC.337, compared to the previous Resolution A468(XII), is that in large rooms with many measurement positions, the individual positions must be compared to the maximum admissible limit.

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

7.2.1 Air-borne noise

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

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

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

Standard noise reduction & additional noise reduction

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

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

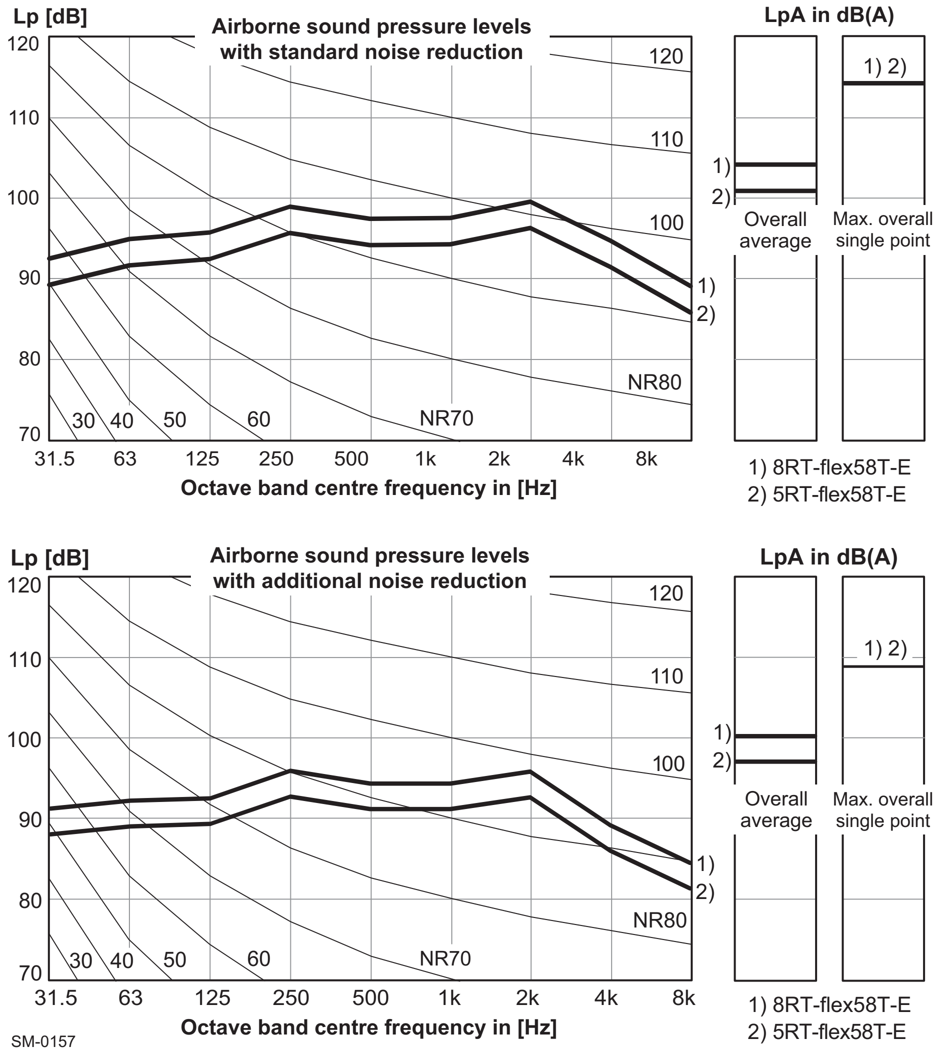


Figure 7-4 Sound pressure level at 1 m distance from engine

7.2.2 Exhaust noise

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

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

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

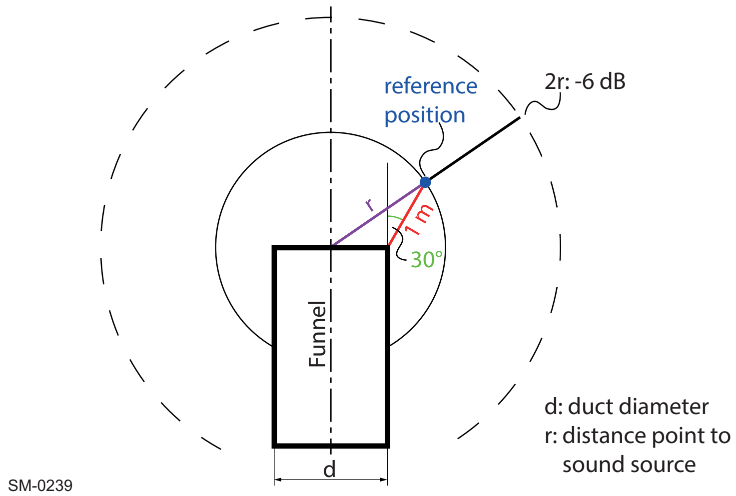


Figure 7-5 Exhaust noise reference point

Installation of silencer

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

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

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

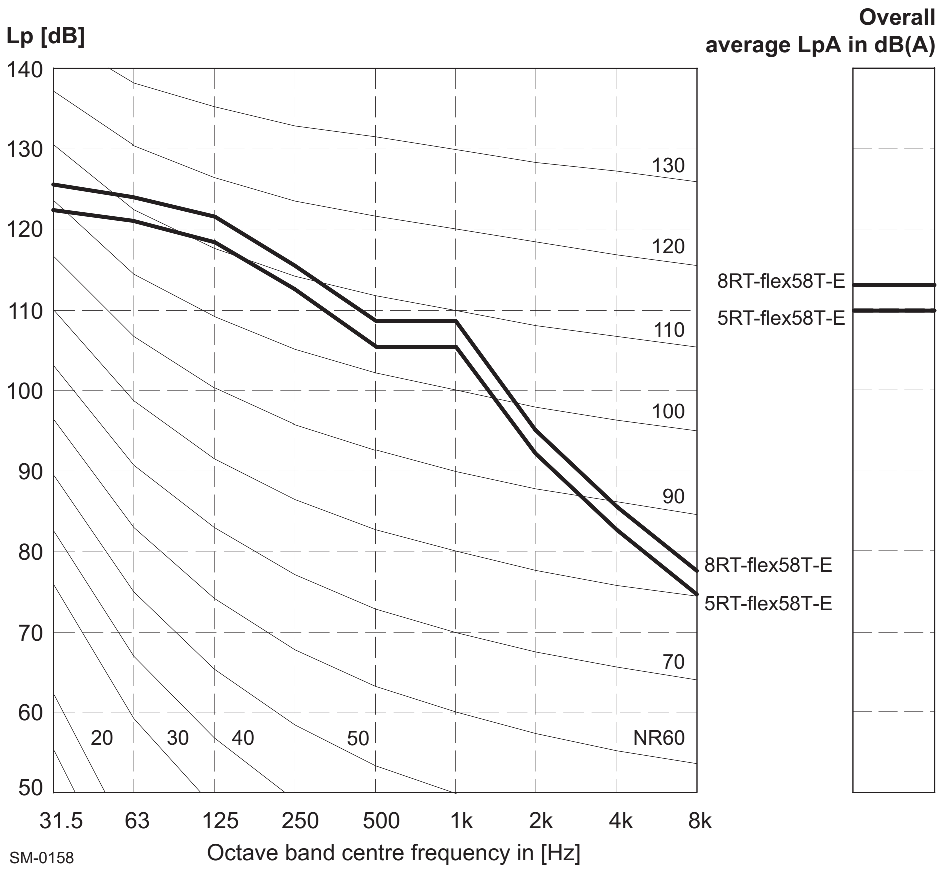


Figure 7-6 Sound pressure level at funnel top of exhaust gas system

7.2.3 Structure-borne noise

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

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

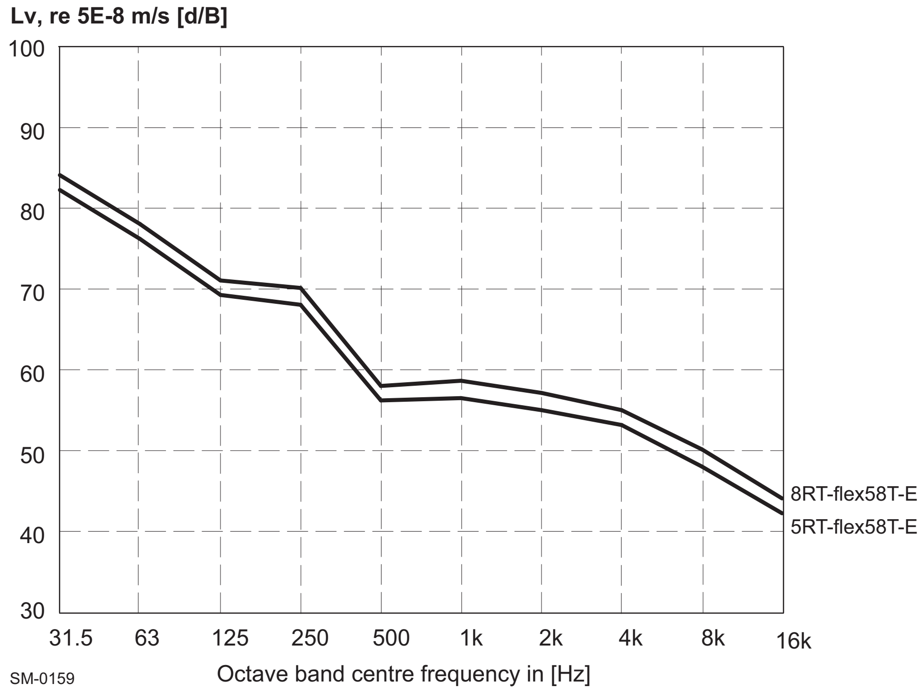


Figure 7-7 Structure-borne noise level at engine feet vertical

8 Engine Dispatch

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

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

8.1 Engines to be transported as part assemblies

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

8.2 Protection of disassembled engines

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

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

8.3 Removal of rust preventing oils after transport

8.3.1 Internal parts

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

8.3.2 External parts

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

9 Appendix

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

9.1 Classification societies

Table 9-1 List of classification societies

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

9.2 List of acronyms

Table 9-2 List of acronyms

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

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

9.3 SI dimensions for internal combustion engines

Table 9-3 SI dimensions

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

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

9.4 Approximate conversion factors

Table 9-4 Conversion factors

Length	1 in			=	25.4 mm
	1 ft	=	12 in	=	304.8 mm
	1 yd	=	3 feet	=	914.4 mm
	1 statute mile	=	1760 yds	=	1609.3 m
	1 nautical mile	=	6080 feet	=	1853 m
Mass	1 oz			=	0.0283 kg
	1 lb	=	16 oz	=	0.4536 kg
	1 long ton			=	1016.1 kg
	1 short ton			=	907.2 kg
	1 tonne			=	1000 kg
Volume (fluids)	1 Imp. pint			=	0.568 l
	1 U.S. pint			=	0.473 l
	1 Imp. quart			=	1.136 l
	1 U.S. quart			=	0.946 l
	1 Imp. gal			=	4.546 l
	1 U.S. gal			=	3.785 l
	1 Imp. barrel	=	36 Imp. gal	=	163.66 l
	1 barrel petroleum	=	42 U.S. gal	=	158.98 l
Force	1 lbf (pound force)			=	4.45 N
Pressure	1 psi (lb/sq in)			=	6.899 kPa (0.0689 bar)
Velocity	1 mph			=	1.609 km/h
	1 knot			=	1.853 km/h
Acceleration	1 mphps			=	0.447 m/s^2
Temperature	1 °C			=	0.55 x (°F -32)
Energy	1 BTU			=	1.06 kJ
	1 kcal			=	4.186 kJ
Power	1 kW			=	1.36 bhp
	1 kW			=	860 kcal/h
Volume	1 in^3			=	16.4 cm^3
	1 ft^3			=	0.0283 m^3
	1 yd^3			=	0.7645 m^3

Area	1 in^2		=	6.45 cm^2
	1 ft^2		=	929 cm^2
	1 yd^2		=	0.836 m^2
	1 acre		=	4047 m^2
	1 sq mile (of land)	= 640 acres	=	2.59 km^2

Winterthur Gas & Diesel in brief

Winterthur Gas & Diesel Ltd. (WinGD) is a leading developer of low-speed gas and diesel engines used for propulsion power in merchant shipping. WinGD sets the industry standard for environmental sustainability, reliability, efficiency and safety. WinGD provides designs, training and technical support to engine manufacturers, shipbuilders and ship operators worldwide. Headquartered in Winterthur, Switzerland, since its inception as the Sulzer Diesel Engine business in 1893, it carries on the legacy of excellence in design.

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