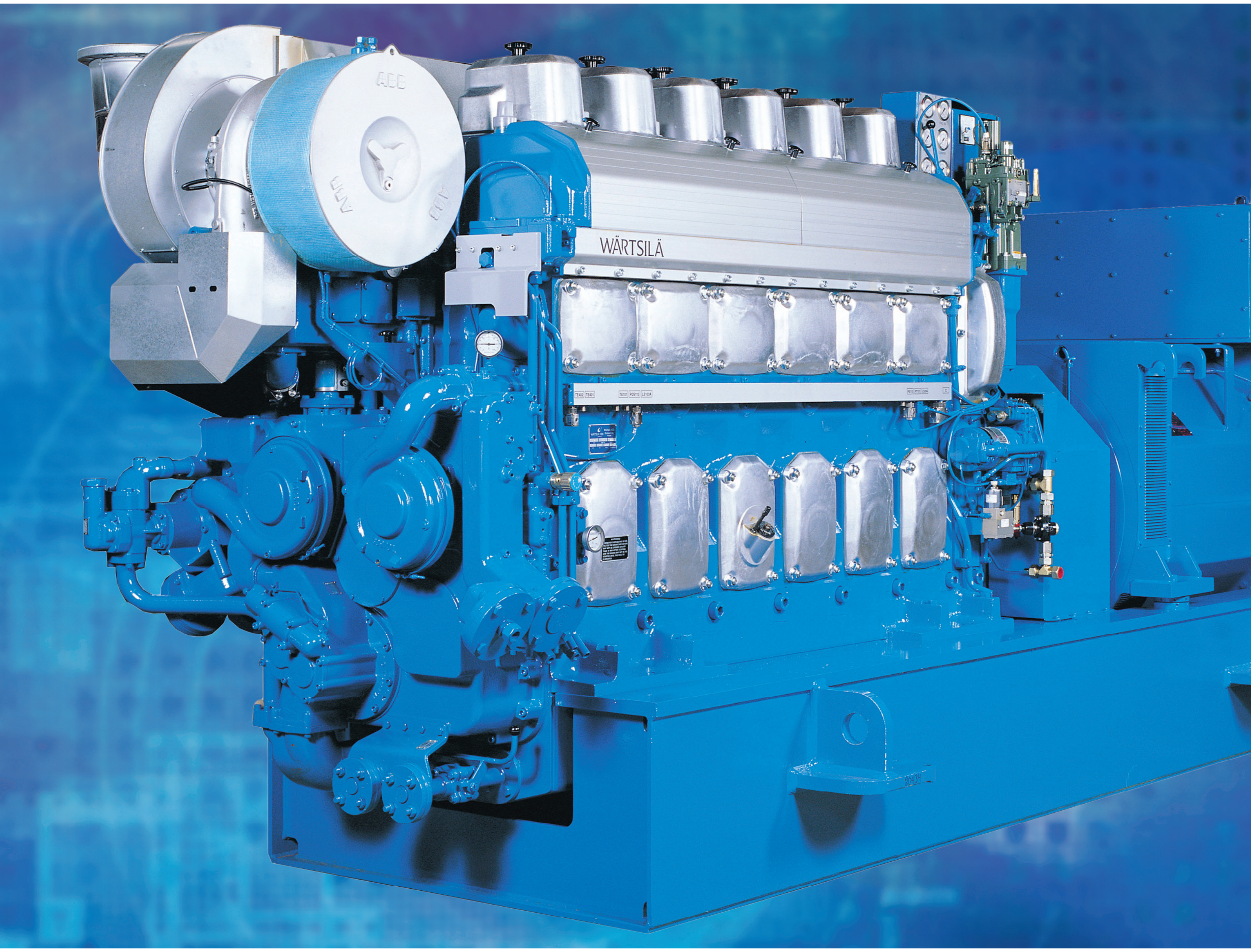
WÄRTSILÄ 20

Project guide



WÄRTSILÄ

Introduction

This Project Guide provides data and system proposals for the early design phase of marine engine installations. For contracted projects specific instructions for planning the installation are always delivered. Any data and information herein is subject to revision without notice. This 1/2005 issue replaces all previous issues of the Wärtsilä 20 Project Guides.

WÄRTSILÄ FINLAND OY
Ship Power
Application Technology

Vaasa, August 2005

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1. Main data and outputs

1.1 Technical main data

The Wärtsilä 20 is a 4-stroke, non-reversible, turbocharged and intercooled diesel engine with direct injection of fuel.

Cylinder bore	200 mm
Stroke	280 mm
Piston displacement	8.8 l/cyl
Number of valves	2 inlet valves and 2 exhaust valves
Cylinder configuration	4, 6, 8, 9, in-line
Direction of rotation	Clockwise, counterclockwise on request

1.2 Maximum continuous output

The mean effective pressure p_e can be calculated as follows:

$$P_e = \frac{P \times c \times 1.2 \times 10^9}{D^2 \times L \times n \times \pi}$$

where:

P_e = Mean effective pressure [bar]
P = Output per cylinder [kW]
n = Engine speed [r/min]
D = Cylinder diameter [mm]
L = Length of piston stroke [mm]
c = Operating cycle (2)

Note! The minimum nominal speed is 1000 rpm both for installations with controllable pitch and fixed pitch propellers.

Table 1.1 Rating table for main engines

Engine	Output in kW (BHP) at 1000 rpm	
	kW	(BHP)
4L20	800	1080
6L20	1200	1630
8L20	1600	2170
9L20	1800	2440

The maximum fuel rack position is mechanically limited to 100% of the continuous output for main engines.

The permissible overload is 10% for one hour every twelve hours. The maximum fuel rack position is mechanically limited to 110% continuous output for auxiliary engine. The generator outputs are calculated for an efficiency of 0.95 and a power factor of 0.8.

Table 1.2 Rating table for auxiliary engines

Engine	Output at							
	720 rpm / 60 Hz		750 rpm / 50 Hz		900 rpm / 60 Hz		1000 rpm / 50 Hz	
	Engine (kW)	Generator (kVA)	Engine (kW)	Generator (kVA)	Engine (kW)	Generator (kVA)	Engine (kW)	Generator (kVA)
4L20	520	620	540	640	740	880	800	950
6L20	780	930	810	960	1110	1320	1200	1420
8L20	1040	1240	1080	1280	1360	1610	1440	1710
9L20	1170	1390	1215	1440	1665	1980	1800	2140

1.3 Reference Conditions

The output is available up to a charge air coolant temperature of max. 38°C and an air temperature of max. 45°C. For higher temperatures, the output has to be reduced according to the formula stated in ISO 3046-1:2002 (E).

The specific fuel consumption is stated in the chapter *Technical data*. The stated specific fuel consumption applies to engines without engine driven pumps, operating in ambient conditions according to ISO 3046-1.

- Total barometic pressure	100 kPa
- Air temperature	25°C
- Relative humidity	30%
- Charge air coolant temperature	25°C

1.3.1 High air temperature

The maximum inlet air temperature is + 45ºC. Higher temperatures would cause an excessive thermal load on the engine, and can be permitted only by de-rating the engine (permanently lowering the MCR) 0.35 % for each 1ºC above + 45ºC.

1.3.2 Low air temperature

When designing ships for low temperatures the following minimum inlet air temperature shall be taken into consideration:

- For starting + 5ºC
- For idling: - 5ºC
- At high load: - 10ºC

At high load, cold suction air with a high density causes high firing pressures. The given limit is valid for a standard engine.

For temperatures below 0ºC special provisions may be necessary on the engine or ventilation arrangement.

Other guidelines for low suction air temperatures are given in the chapter *Combustion air system*.

1.3.3 High water temperature

The maximum inlet LT-water temperature is + 38ºC. Higher temperatures would cause an excessive thermal load on the engine, and can be permitted only if de-rating the engine (permanently lowering the MCR) 0.3 % for each 1ºC above + 38ºC.

1.4 Fuel characteristics

The Wärtsilä 20 engine is designed to operate on fuels with properties listed in the tables below. The fuel specifications are based on the ISO 8217:1996 (E) standard, but some additional properties are defined in the tables.

Distillate fuel grades are ISO-F-DMX, DMA, DMB and DMC. These fuel grades are all referred to as MDF (Marine Diesel Fuel) in this publication.

Residual fuel grades are referred to as HFO (Heavy Fuel Oil). The fuel specification HFO2 covers the categories ISO-F-RMA 10 to RMK 55. Fuels fulfilling the specification HFO1 permit longer overhaul intervals of specific engine components than HFO2. See table in the chapter *Description of the engine*.

Table 1.3 MDF Specifications

Property	Unit	ISO-F-DMX	ISO-F-DMA	ISO-F-DMB	ISO-F-DMC 1)	Test method ref.
Viscosity, min., before injection pumps 2)	cSt	1.8	1.8	1.8	1.8	ISO 3104
Viscosity, max.	cSt at 40°C	5.5	6	11	14	ISO 3104
Viscosity, max, before injection pumps 2)		24	24	24	24	ISO 3104
Density, max.	kg/m³ at 15°C	3)	890	900	920	ISO 3675 or 12185
Cetane number		45	40	35	—	ISO 5165 or 4264
Water, max.	% volume	—	—	0.3	0.3	ISO 3733
Sulphur, max.	% mass	1	1.5	2	2	ISO 8574
Ash, max.	% mass	0.01	0.01	0.01	0.05	ISO 6245
Vanadium, max.	mg/kg	—	—	—	100	ISO 14597
Sodium before engine, max. 2)	mg/kg	—	—	—	30	ISO 10478
Aluminium + Silicon, max.	mg/kg	—	—	—	25	ISO 10478
Aluminium + Silicon before engine, max. 2)	mg/kg	—	—	—	15	ISO 10478
Carbon residue (micro method, 10 % vol dist. bottoms), max.	% mass	0.30	0.30	—	—	ISO 10370
Carbon residue (micro method), max.	% mass	—	—	0.30	2.50	ISO 10370
Flash point (PMCC), min. 2)	°C	60	60	60	60	ISO 2719
Pour point, max. 4)	°C	—	-6 - 0	0–6	0–6	ISO 3016
Sediment	% mass	—	—	0.07	—	ISO 3735
Total sediment potential, max.	% mass	—	—	—	0.1	ISO 10307-1

1) Use of ISO-F-DMC category fuel is allowed provided that the fuel treatment system is equipped with a fuel centrifuge.

2) Additional properties specified by the engine manufacturer, which are not included in the ISO specification or differ from the ISO specification.

3) In some geographical areas there may be a maximum limit.

4) Different limits specified for winter and summer qualities.

Table 1.4 HFO Specifications

Property	Unit	Limit HFO 1	Limit HFO 2	Test method ref.
Viscosity, max.	cSt at 100°C cSt at 50°C Redwood No. 1 s at 100°F	55 730 7200	55 730 7200	ISO 3104
Viscosity, min/max. before engine 4)	cSt	16/24	16/24	ISO 3104
Density, max.	kg/m³ at 15°C	991 1) / 1010	991 1) / 1010	ISO 3675 or 12185
CCAI, max.4)		850	870 2)	ISO 8217
Water, max.	% volume	1.0	1.0	ISO 3733
Water before engine, max.4)	% volume	0.3	0.3	ISO 3733
Sulphur, max.	% mass	2.0	5.0	ISO 8754
Ash, max.	% mass	0.05	0.20	ISO 6245
Vanadium, max.	mg/kg	100	600 3)	ISO 14597
Sodium, max.4)	mg/kg	50	100 3)	ISO 10478
Sodium before engine, max.4)	mg/kg	30	30	ISO 10478
Aluminium + Silicon, max.	mg/kg	30	80	ISO 10478
Aluminium + Silicon before engine, max.4)	mg/kg	15	15	ISO 10478
Conradson carbon residue, max.	% mass	15	22	ISO 10370
Asphaltenes, max.4)	% mass	8	14	ASTM D 3279
Flash point (PMCC), min.	°C	60	60	ISO 2719
Pour point, max.	°C	30	30	ISO 3016
Total sediment potential, max.	% mass	0.10	0.10	ISO 10307-2

1) Max. 1010 kg/m³ at 15°C provided the fuel treatment system can remove water and solids.

2) Straight run residues show CCAI values in the 770 to 840 range and are very good ignitors. Cracked residues delivered as bunkers may range from 840 to - in exceptional cases - above 900. Most bunkers remain in the max. 850 to 870 range at the moment.

3) Sodium contributes to hot corrosion on exhaust valves when combined with high sulphur and vanadium contents. Sodium also contributes strongly to fouling of the exhaust gas turbine blading at high loads. The aggressiveness of the fuel depends not only on its proportions of sodium and vanadium but also on the total amount of ash constituents. Hot corrosion and deposit formation are, however, also influenced by other ash constituents. It is therefore difficult to set strict limits based only on the sodium and vanadium content of the fuel. Also a fuel with lower sodium and vanadium contents that specified above, can cause hot corrosion on engine components.

4) Additional properties specified by the engine manufacturer, which are not included in the ISO specification.

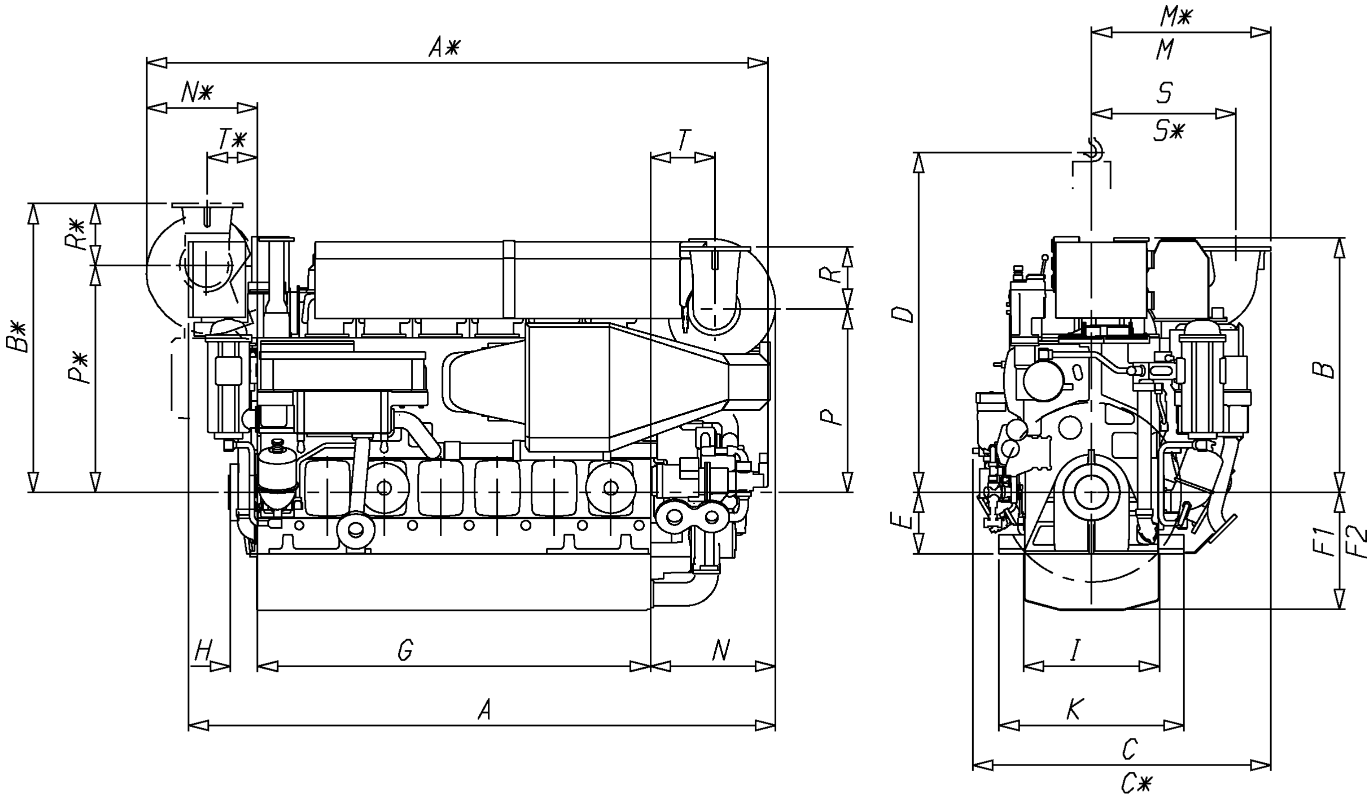
Lubricating oil, foreign substances or chemical waste, hazardous to the safety of the installation or detrimental to the performance of the engines, should not be contained in the fuel.

The limits above also correspond to the demands of the following standards. The properties marked with 4) are not specifically mentioned in the standards but should also be fulfilled.

- BS MA 100: 1996, RMH 55 and RMK 55
- CIMAC 1990, Class H55 and K55
- ISO 8217: 1996(E), ISO-F-RMH 55 and RMK 55

1.5 Principal dimensions and weights

Figure 1.1 Main engines (3V92E0068c)



Engine	A*	A	B*	B	C*	C	D	E	F1	F2	H	H	I	K
4L20		2510		1348		1483	1800	325	725	725	1480	155	718	980
6L20	3292	3108	1528	1348	1580	1579	1800	325	624	824	2080	155	718	980
8L20	4011	3783	1614	1465	1756	1713	1800	325	624	824	2680	155	718	980
9L20	4299	4076	1614	1449	1756	1713	1800	325	624	824	2980	155	718	980

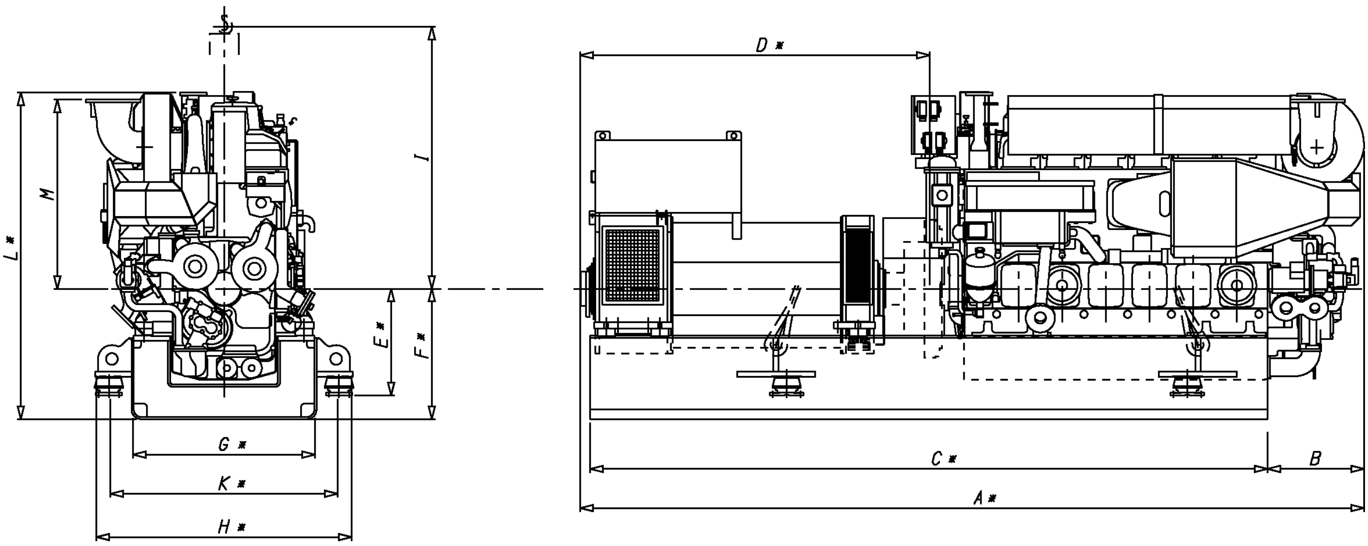
F1 for dry sump and F2 for deep wet sump

Engine	M*	M	N*	N	P*	P	R*	R	S*	S	T*	T	Weight
4L20		854		665		920		248		694		349	7.2
6L20	951	950	589	663	1200	971	328	328	762	763	266	343	9.3
8L20	1127	1084	708	738	1224	1000	390	390	907	863	329	339	11.0
9L20	1127	1084	696	731	1224	1000	390	390	907	863	329	339	11.6

** Turbocharger at flywheel end*

Dimensions in mm. Weight in tons.

Figure 1.2 Auxiliary engines (3V58E0576d)



Engine	A*	B	C*	D*	E*	F*	G*	H*	I	K*	L*	M	Weight [ton] *
4L20	4910	665	4050	2460	725	990	1270/1420	1770/1920	1800	1580/1730	2338	1168	14.0
6L20	5325	663	4575	2300	725	895/975/1025	1270/1420/1570	1770/1920/2070	1800	1580/1730/1880	2243/2323/2373	1299	16.8
8L20	6030	731	5100	2310	725	1025/1075	1420/1570	1920/2070	1800	1730/1880	2474/2524	1390	20.7
9L20	6535	731	5400	2580	725	1075/1125	1570/1800	2070/2300	1800	1880/2110	2524/2574	1390	23.8

** Dependent on generator type and size.*
Dimensions in mm. Weght in tons.

2. Operating ranges

2.1 General

The operating field of the engine depends on the required output, and these should therefore be determined together. This applies to both FPP and CPP applications. Concerning FPP applications also the propeller matching must be clarified.

A diesel engine can deliver its full output only at full engine speed. At lower speeds the available output and also the available torque are limited to avoid thermal overload and turbocharger surging. This is because the turbocharger is less efficient and the amount of scavenge air supplied to the engine is low. Often e.g. the exhaust valve temperature can be higher at low load (when running according to the propeller law) than at full load. Furthermore, the smallest distance to the so-called surge limit of the compressor typically occurs at part load. Margin is required to permit reasonable wear and fouling of the turbocharging system and different ambient conditions (e.g. suction air temperature).

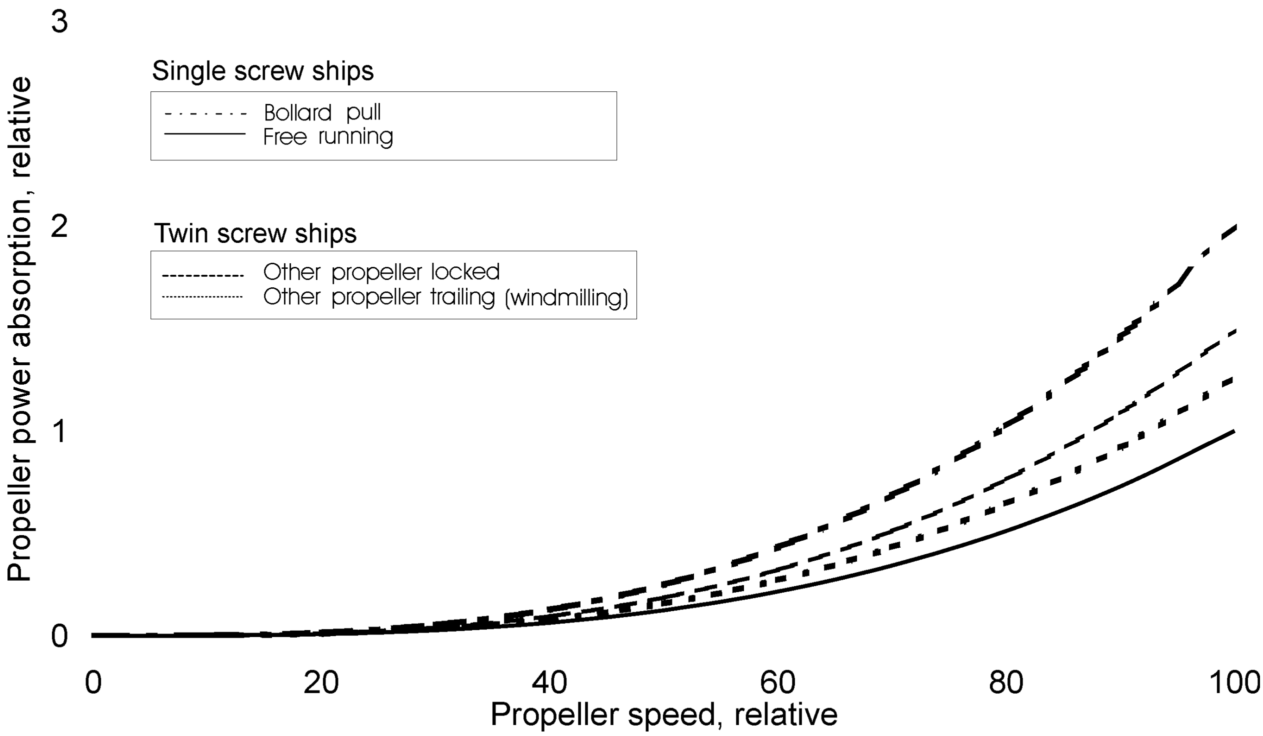
As a rule, the higher the specified mean effective pressure the narrower is the permitted engine operating range. This is the reason why separate operating fields may be specified for different output stages, and the available output for FP-propellers may be lower than for CP-propellers. Today's development towards lower emissions, lower fuel consumption and SCR compatibility also contribute to the restriction of the operating field.

A matter of high importance is the matching of the propeller and the engine. Weather conditions, acceleration, the loading condition of the ship, draught and trim, the age and fouling of the hull, and ice conditions all play an important role.

With a FP propeller these factors all contribute to moving the power absorption curve towards higher thermal loading of the engine. There is a risk for surging of the turbocharger (when moving to the left in the power-rpm diagram). On the other hand, with a new and clean hull in ballast draft the power absorption is lighter and full power will not be absorbed as the maximum engine speed limits the speed range upwards. These drawbacks are avoided by specifying CP-propellers.

A similar problem is encountered on twin-screw (or multi-screw) ships with fixed-pitch propellers running with only one propeller. If one propeller is wind-milling (rotating freely), the other propeller will feel an increased power absorption, and even more so, if the other propeller is blocked. The phenomenon is more pronounced on ships with a small block coefficient. The issue is illustrated in the diagram below.

Figure 2.1 Propeller power absorption in different conditions - example



The figure also indicates the magnitude of the so-called bollard pull curve, which means the propeller power absorption curve at zero ship speed. It is a relevant condition for some ship types, such as tugs, trawlers and icebreakers. This diagram is valid for open propellers. Propellers running in nozzles are less sensitive to the speed of advance of the ship.

The bollard pull curve is also relevant for all FPP applications since the power absorption during acceleration is always somewhere between the free running curve and the bollard pull curve! If the free sailing curve is very close to the 100% engine power curve and the bollard pull curve at the same time is considerably higher than the 100% engine power curve, then the acceleration from zero ship speed will be very difficult. This is because the propeller will require such a high torque at low speed that the engine is not capable of increasing the speed. As a consequence the propeller will not develop enough thrust to accelerate the ship.

Heavy overload will also occur on a twin-screw vessel with FP propellers during manoeuvring, when one propeller is reversed and the other one is operating forward. When dimensioning FP propellers for a twin screw vessel, the power absorption with only one propeller in operation should be max. 90% of the engine power curve, or alternatively the bollard pull curve should be max 120% of the engine power curve. Otherwise the engine must be de-rated 20-30% from the normal output for FPP applications. This will involve extra costs for non-standard design and separate EIAPP certification. For this reason it is recommended to select CP-propellers for twin-screw ships with mechanical propulsion.

FP propeller should never be specified for a twin-in / single-out reduction gear as one engine is not capable of driving a propeller designed for the power of two engines.

For ships intended for operation in heavy ice, the additional torque of the ice should furthermore be considered.

For selecting the machinery, typically a sea margin of 10…15 % is applied, sometimes even 25…30 %. This means the relative increase in shaft power from trial conditions to typical service conditions (a margin covering increase in ships resistance due to fouling of hull and propeller, rough seas, wind, shallow water depth etc). Furthermore, an engine margin of 10…15 % is often applied, meaning that the ship’s specified service speed should be achieved with 85…90 % of the MCR. These two independent parameters should be selected on a project specific basis.

The minimum speed of the engine is a project specific issue, involving torsional vibrations, elastic mounting, built-on pumps etc.

In projects where the standard operating field, standard output or standard nominal speed do not satisfy all project specific demands, the engine maker should be contacted.

2.2 Matching the engines with driven equipment

2.2.1 CP-propeller

Controllable pitch propellers are normally dimensioned and classified to match the Maximum Continuous Rating of the prime mover(s). In case two (or several) engines are connected to the same propeller it is normally dimensioned corresponding to the total power of all connected prime movers. This is also the case if the propeller is driven by prime movers of different types, as e.g. one diesel engine and one electric motor (which may work as a shaft generator in some operating modes). In case the total power of all connected prime movers will never be utilised, classification societies can approve a dimensioning for a lower power in case the plant is equipped with an automatic overload protection system. The rated power of the propeller will affect the blade thickness, hub size and shafting dimensions.

Designing a CP-propeller is a complex issue, requiring compromises between efficiency, cavitation, pressure pulses, and limitations imposed by the engine and a possible shaft generator, all factors affecting the blade geometry. Generally speaking the point of optimization (an optimum pitch distribution) should correspond to the service speed and service power of the ship, but the issue may be complicated in case the ship is intended to sail with various ship speeds, and even with different operating modes. Shaft generators or generators (or any other equipment) connected to the free end of the engine should be considered in case these will be used at sea.

The propeller efficiency is typically highest when running along the propeller curve defined by the design pitch, in other word requiring the engine at part load to run slowly and heavily. Typically also the efficiency of a diesel engine running at part load is somewhat higher when running at a lower speed than the nominal.

Pressure side cavitation may easily occur when running at high propeller speed and low pitch. This is a noisy type of cavitation and it may also be erosive. However the pressure side cavitation behaviour can be improved a lot by a suitable propeller blade design. Also cavitation at high power may cause increased pressure pulses, which can be reduced by increased skew angle and optimized blade geometry.

It is of outmost importance that the propeller designer has information about all the actual operation conditions for the vessel. Often the main objective is to minimise the extent and fluctuation of the suction side cavitation to reduce propeller-induced hull vibrations and noise at high power, while simultaneously avoiding noisy pressure side cavitation and a large drop in efficiency at reduced propeller pitch and power.

The propeller may enter the pressure side cavitation area already when reducing the power to less than half, maintaining nominal speed. In twin-in/single-out installations the plant cannot be operated continuously with one engine and a shaft generator connected, if the shaft generator requires operation at nominal propeller speed.

Many solutions are possible to solve this problem:

- The shaft generator (connected to the secondary side of the clutch) is used only when sailing with high power.
- The shaft generator (connected to the secondary side of the clutch) is used only when manoeuvring with low or moderate power, the transmission ratio being selected to give nominal frequency at reduced propeller speed.
- The shaft generator is connected to the primary side of the clutch of one of the engines, and can be used independently from the propeller, e.g. to produce power for thrusters during manoeuvring.
- No shaft generator is installed.

This type of issues are not only operational of nature, they have to be considered at an early stage when selecting the machinery configuration. For all these reasons it is essential to know the ship's operating profile when designing the propeller and defining the operating modes.

In normal applications no more than two engines should be connected to the same propeller.

CP-propellers typically have the option of being operated at variable speed. To avoid the above mentioned pressure side cavitation the propeller speed should be kept sufficiently below the cavitation limit, but not lower than necessary. On the other hand, there are also limitations on the engine's side, such as avoiding thermal overload at lower speeds.

To optimize the operating performance considering these limitations CP-propellers are typically operating along a preset combinator curve, combining optimum speed and pitch throughout the whole power range, controlled by one single control lever on the bridge. Applications with two engines connected to the same propeller must have separate combinator curves for one engine operation and twin engine operation. This applies similarly to twin-screw vessels. Two or several combinator curves may be foreseen in complicated installations for different operating modes (one-engine, two-engines, manoeuvring, free running etc).

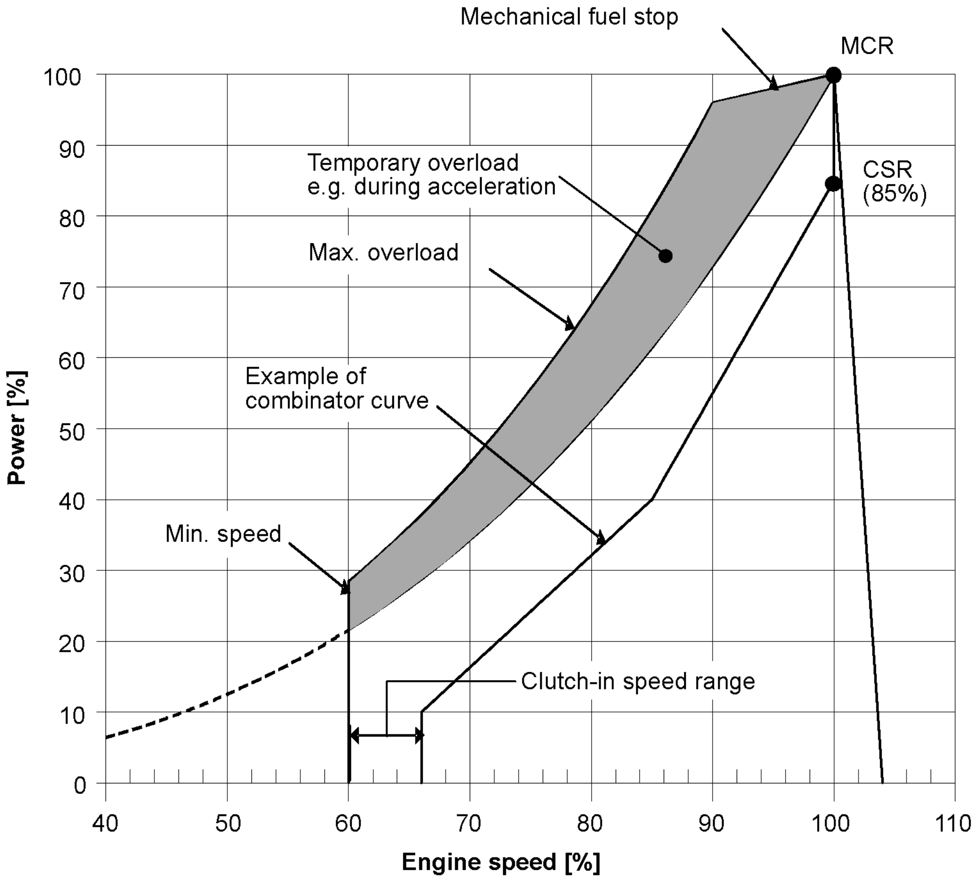
At a given propeller speed and pitch, the ship's speed affects the power absorption of the propeller. This effect is to some extent ship-type specific, being more pronounced on ships with a small block coefficient. The power absorption of the propeller can sometimes be almost twice as high during acceleration than during free steady-state running. Navigation in ice can also add to the torque absorption of the propeller.

An engine can deliver power also to other equipment like a pump, which can overload the engine if used without prior load reduction of the propeller.

For the above mentioned reasons an automatic load control system is required in all installations running at variable speed. The purpose of this system is to protect the engine from thermal load and surging of the turbocharger. With this system the propeller pitch is automatically reduced when a pre-programmed load versus speed curve (the "load curve") is exceeded, overriding the combinator curve if necessary. The load information must be derived from the actual fuel rack position and the speed should be the actual speed (and not the demand). A so-called overload protection, which is active only at full fuel pump settings, is not sufficient in variable speed applications.

The diagrams below show the operating ranges for CP-propeller installations. The design range for the combinator curve should be on the right hand side of the nominal propeller curve. Operation in the shaded area is permitted only temporarily during transients.

Figure 2.2 Operating field for CP Propeller



The clutch-in speed is a project specific issue. From the engine point of view, the clutch-in speed should be high enough to have a sufficient torque available, but not too high. The slip time on the other hand should be as long as possible. In practise longer slip times than 5 seconds are exceptions, but the clutch should typically be dimensioned so that it allows a slip time of at least 3 seconds. From the clutch point of view, a high clutch-in speed causes a high thermal load on the clutch itself, which has to be taken into account when specifying the clutch. A reasonable compromise is to select the idle speed as clutch-in speed. In applications with two engines connected to the same propeller (CP), it might be necessary to select a slightly higher clutch-in speed. In case the engine has to continue driving e.g. a pump or a generator (connected on the primary side of the clutch) during the clutch-in process a higher clutch-in speed may be necessary, but then also some speed drop has to be permitted.

CP-propellers in single-screw ships typically rotate counter-clockwise, requiring a clockwise sense of rotation of the engine with a typically single-stage reduction gear. The sense of rotation of propellers in twin-screw ships is a project specific issue.

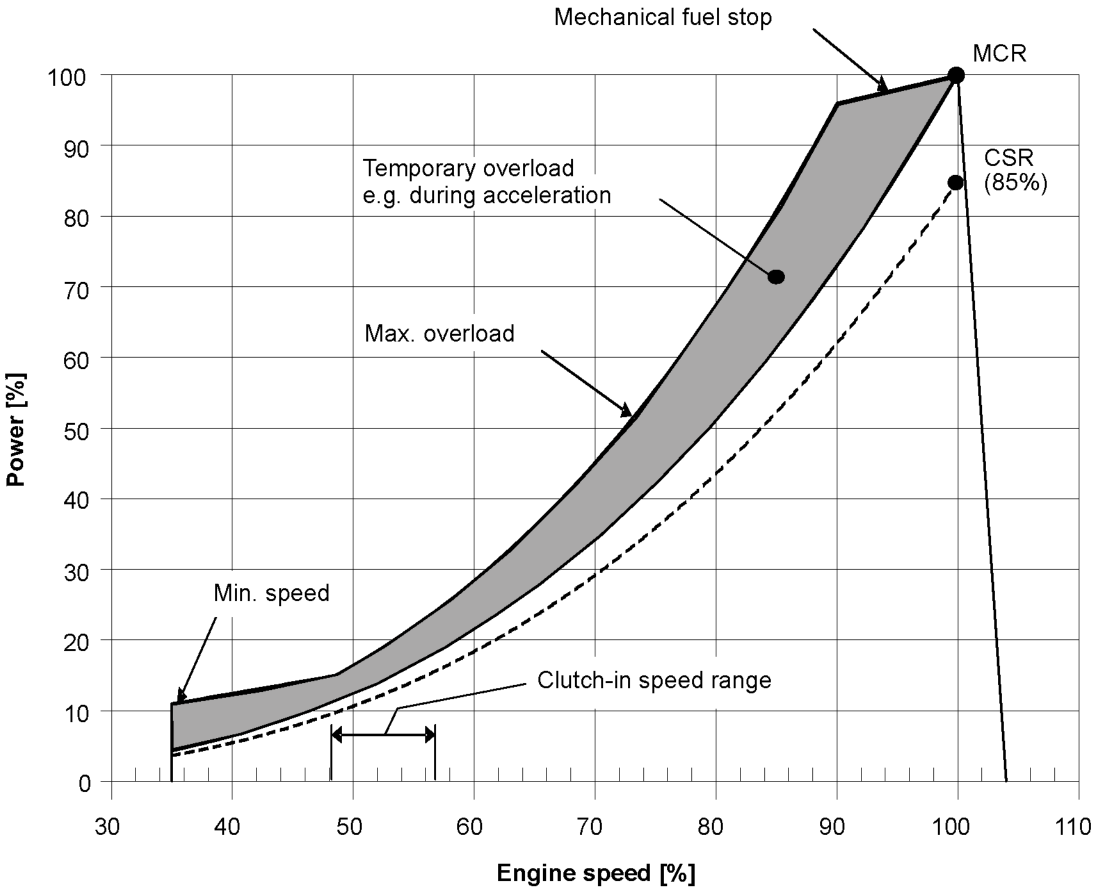
2.2.2 FP-propeller

The fixed pitch propeller needs a very careful matching, as explained above. The operational profile of the ship is very important (acceleration requirements, loading conditions, sea conditions, manoeuvring, fouling of hull and propeller etc).

The FP-propeller should normally be designed to absorb maximum 85 % of the maximum continuous output of the main engine (power transmission losses included) at nominal speed when the ship is on trial. Typically this corresponds to 81 – 82 % for the propeller itself (excluding power transmission losses). This is typically referred to as the “light running margin”, a compensation for expected future drop in revolutions for a constant given power, typically 5-6 %.

For ships intended for towing, the bollard pull condition needs to be considered as explained earlier. The propeller should be designed to absorb not more than 95 % of the maximum continuous output of the main engine at nominal speed when operating in towing or bollard pull conditions, whichever service condition is relevant. In order to reach 100 % MCR it is allowed to increase the engine speed to 101.7 %. The speed does not need to be restricted to 100 % after bollard pull tests have been carried out. The absorbed power in free running and nominal speed is then relatively low, e.g. 50 – 65 % of the output at service conditions.

Figure 2.3 Operating field for FP Propeller



The engine is non-reversible, so the reduction gear has to be of the reversible type. A shaft brake should also be installed.

A Robinson diagram (= four-quadrant diagram) showing the propeller torque ahead and astern for both senses of rotation is needed to determine the parameters of the crash stop.

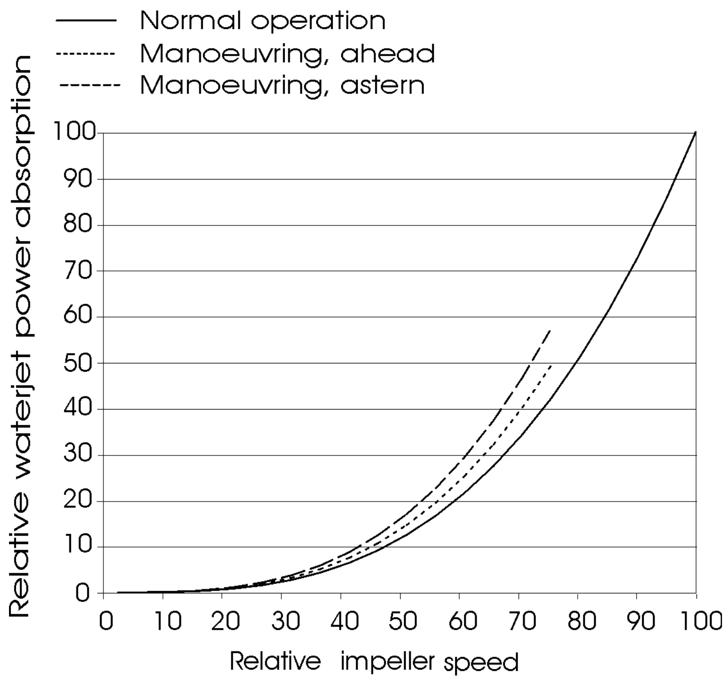
FP-propellers in single-screw ships typically rotate clockwise, requiring a counter clockwise sense of rotation of the engine with a typically single-stage (in the ahead mode) reverse reduction gear.

2.2.3 Water jets

Water jets also require a careful matching with the engine, similar to that of the fixed pitched propeller. However, there are some distinctive differences between the dimensioning of a water jet compared to that of a fixed pitch propeller.

Water jets operate at variable speed depending on the thrust demand. The power absorption vs. rpm of a water jet follows a cubic curve under normal operation. The power absorption vs. rpm is higher when the ship speed is reduced, with the maximum torque demand occurring when manoeuvring astern. The power absorption vs. revolution speed for a typical water jet is illustrated in the diagram below.

Figure 2.4 Water jet power absorption



The reversal of the thrust from the water jet is achieved by a reversing bucket. Moving the bucket into the jet stream and thereby deflecting it forward, towards the bow, reverses the thrust from the jet. The bucket can be gradually inserted in the water jet, so that only part of the jet is deflected. This way the thrust can be controlled continuously from full ahead to full astern just by adjusting the position of the bucket. The reversing bucket is typically operated at part speed only.

The speed of the ship has only a small influence on the revolution speed of water jet, unlike the case for a fixed pitched propeller. This means that there will only be a very small change in water jet speed when the ship speed drops. Increased resistance, due to fouling of the hull, rough seas, wind or shallow water depth, will therefore not affect the torque demand on an engine coupled to a water jet in the same degree as on an engine coupled to a fixed pitched propeller. This means that the water jet can be matched closer to the MCR than a fixed pitched propeller. In fact, the water jet power absorption should be dimensioned close to 100% MCR to get out as much power as possible. However, some margin should be left, due to tolerances in the power estimates of the jet and the small, but still present, increase in torque demand due to a possible increase in ship resistance.

The torque demand at lower speeds should also be carefully compared to the operating field of the engine. Engines with highly optimized turbo chargers can have an operating field that does not cover the water jet power demand over the entire speed range. Also the lower efficiency of the transmission and the reduction gear at part load should be accounted for in the estimation of the power absorption. The time spent at manoeuvring should be considered as well, if the power absorption in manoeuvring mode exceeds the operating field for continuous operation for the engine. In projects where the standard operating field does not satisfy all project specific demands, the engine maker should be contacted.

2.2.4 Other propulsors

Azimuth thrusters

Azimuth thrusters can be equipped with fixed-pitch or controllable-pitch propellers. Most of the above given instructions for CP- and FP-propellers are valid also in case of azimuth thrusters, however with some specific features. The azimuth thrusters offer a good manoeuvrability by turning the propulsor. During slow manoeuvring in harbour the propeller works close to the bollard pull curve, which therefore has to be properly considered especially when matching azimuth thrusters with FP-propeller with the engine. Reversing and crash stop are also performed by turning the FP-thrusters (rather than changing the sense of rotation), causing a heavy propeller curve but in a different way than with an ordinary shaft line.

Tunnel thrusters

Tunnel thrusters are typically driven by electric motors, but can also be driven by diesel engines. Tunnel thrusters can be equipped with fixed-pitch or controllable-pitch propellers. Tunnel thrusters with CP-propellers can be operated at constant speed, which may be feasible to get the quickest possible response, or according to a combinator curve. A load control system is required. A non-reversible diesel engine driving a tunnel thruster with FP-propeller is typically not a feasible solution, as an extra reversible gear box would be needed.

Voith-Schneider propellers

This type of propulsor is operated at variable speed and pitch. It is important to have some kind of load control system to prevent overload over the whole speed range, as described in previous chapters.

2.2.5 Dredgers

The power generation plant of a dredger can be of different configurations:

- Diesel-electric. Propulsors and dredging pumps are electrically driven. This is a good and flexible solution, but also the most expensive.
- Mechanically driven main propellers, and electrically driven dredging pumps and thrusters. The main engines and generators driven e.g. from the free end of the crankshaft are running at constant speed, and the dredging pumps can be operated at variable speed with a frequency converter. This is a good, flexible and cost-effective solution.

The configuration with the main engine running at constant speed has proved to be a good solution, also capable of taking the typical load transients coming from the dredging pumps.

Mechanically driven main propellers and dredging pumps. The main engines have to operate at variable speed. This may appear to be the cheapest solution, but it has operational limitations.

- In this configuration, when the dredging pumps are mechanically driven, dredging requires a capability to run a constant torque down to 70 or 80 % of the nominal speed. This kind of torque requirement results in normally significant de-rating of the main engines.

2.2.6 Generators

Generators are typically operated at nominal speed. Modern generators are synchronous AC machines, producing a frequency equalling the number of pole pairs times the rotational speed. The synchronous speed of such generators is listed below.

Table 2.1 Synchronous speed of generators

Number of pole pairs	Number of poles	Synchr.speed, rpm	
		50 Hz	60 Hz
1	2	3000	3600
2	4	1500	1800
3	6	1000	1200
4	8	750	900
5	10	600	720
6	12	500	600
7	14	428.6	514.3
8	16	375	450
9	18	333.3	400
10	20	300	360
11	22	272.7	327.3

In some rare installations, shaft generators or diesel-generators may be operated at variable frequency, sometimes referred to as floating frequency. This may be the case with a shaft generator supplying the ship’s service electricity, when it may be clearly feasible to operate the propulsion plant at variable speed for reasons of propeller efficiency or cavitation.

Desired transmission ratios between main engines and shaft generators cannot always be exactly found, as the number of teeth in the reduction gear has to be selected in steps of complete teeth.

This is also the case when the generator nominal speed is a multiple of the nominal speed of the engine. The number of teeth is selected to permit all teeth being in contact with all teeth of the other gear wheel, to avoid uneven wear. To achieve this target, gear wheels with a multiple number of teeth compared with its smaller pair should be avoided. This is valid for the main power transmission from the engine to the propeller, as well as for PTOs for shaft generators. In other words cases where a combination of tooth numbers giving exactly the desired transmission ratio can be found, it is not feasible to use them.

The maximum output of diesel engines driving auxiliary generators and diesel engines driving generators for propulsion is 110 % of the MCR.

2.3 Loading capacity

The loading rate of a highly supercharged diesel engine must be controlled, because the turbocharger needs time to accelerate before it can deliver the required amount of air. In an emergency situation the engine can be loaded in three equal steps in accordance with class requirement. However in normal operation the load should always be applied gradually.

This can be obtained if the loading speed (rate) does not exceed the curves given below.

2.3.1 Diesel-mechanical propulsion

The loading is to be controlled by a load increase programme, which is included in the propeller control system.

2.3.2 Diesel-electric propulsion

Class rules regarding load acceptance capability should not be interpreted as guidelines on how to apply load on the engine in normal operation. The class rules only determine what the engine must be capable of, if an emergency situation occurs.

The electrical system onboard the ship must be designed so that the diesel generators are protected from load steps that exceed the limit. Normally system specifications must be sent to the classification society for approval and the functionality of the system is to be demonstrated during the ship’s trial.

The loading performance is affected by the rotational inertia of the whole generating set, the speed governor adjustment and behaviour, generator design, generator excitation system, voltage regulator behaviour and nominal output.

Loading capacity and overload specifications are to be developed in co-operation between the plant designer, engine manufacturer and classification society at an early stage of the project. Features to be incorporated in the power management systems are presented in the Chapter for electrical power generation.

2.3.3 Auxiliary engines driving generators

The load should always be applied gradually in normal operation. This will prolong the lifetime of engine components. The class rules only determine what the engine must be capable of, if an emergency situation occurs. Provided that the engine is preheated to a HT-water temperature of 60…70ºC the engine can be loaded immediately after start.

The fastest loading is achieved with a successive gradual increase in load from 0 to 100 %. It is recommended that the switchboards and the power management system are designed to increase the load as smoothly as possible.

The electrical system onboard the ship must be designed so that the diesel generators are protected from load steps that exceed the limit. Normally system specifications must be sent to the classification society for approval and the functionality of the system is to be demonstrated during the ship’s trial.

2.4 Operation at low load and idling

The engine can be started, stopped and operated on heavy fuel under all operating conditions. Continuous operation on heavy fuel is preferred rather than changing over to diesel fuel at low load operation and manoeuvring. The following recommendations apply:

2.4.1 Absolute idling (declutched main engine, disconnected generator)

Maximum 5 minutes (recommended about 1 min for post cooling), if the engine is to be stopped after the idling.

2.4.2 Operation at < 20 % load on HFO or < 10 % on MDF

Maximum 100 hours continuous operation. At intervals of 100 operating hours the engine must be loaded to minimum 70 % of the rated load.

2.4.3 Operation at > 20 % load on HFO or > 10 % on MDF

No restrictions.

3. Technical data

3.1 Wärtsilä 4L20

Wärtsilä 4L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Engine output	kW	800	520	540	740	800
Engine output	HP	1088	708	735	1007	1088
Cylinder bore	mm	200	200	200	200	200
Stroke	mm	280	280	280	280	280
Mean effective pressure	MPa	2.73	2.46	2.46	2.8	2.73
Mean piston speed	m/s	9.3	6.7	7.0	8.4	9.3
Minimum speed, FPP-installation	rpm	350				
Combustion air system						
Flow of air at 100% load	kg/s	1.57	0.94	0.99	1.46	1.57
Ambient air temperature, max.	°C	45	45	45	45	45
Air temperature after air cooler	°C	45...60	45...60	45...60	45...60	45...60
Air temperature after air cooler, alarm	°C	75	75	75	75	75
Exhaust gas system (Note 1)						
Exhaust gas flow (100% load)	kg/s	1.62	0.97	1.02	1.5	1.62
Exhaust gas flow (85% load)	kg/s	1.39	0.84	0.89	1.31	1.42
Exhaust gas flow (75% load)	kg/s	1.22	0.76	0.81	1.17	1.28
Exhaust gas flow (50% load)	kg/s	0.81	0.55	0.59	0.82	0.93
Exhaust gas temperature after turbocharger (100% load)	°C	360	360	360	350	360
Exhaust gas temperature after turbocharger (85% load)	°C	365	360	360	345	355
Exhaust gas temperature after turbocharger (75% load)	°C	375	360	365	345	350
Exhaust gas temperature after turbocharger (50% load)	°C	390	370	370	355	360
Exhaust gas temperature after cylinder, alarm	°C	450	400	400	400	400
Exhaust gas back pressure, rec. max.	kPa	3.0	3.0	3.0	3.0	3.0
Exhaust gas pipe diameter, min	mm	300	250	250	300	300
Heat balance (Note 2)						
Jacket water	kW	183	127	132	170	183
Charge air	kW	244	157	161	224	244
Lubricating oil	kW	115	67	69	103	115
Exhaust gases	kW	570	356	374	512	570
Radiation, etc.	kW	33	31	31	32	33
Fuel system (Note 3)						
Pressure before injection pumps	kPa	600	600	600	600	600
Pump capacity (MDF), engine driven	m^3/h	0.87	0.87	0.9	0.78	0.87
Fuel consumption (100% load)	g/kWh	196	195	195	194	196
Fuel consumption (85% load)	g/kWh	193	196	196	194	196
Fuel consumption (75% load)	g/kWh	194	198	198	195	197
Fuel consumption (50% load)	g/kWh	200	204	204	200	201
Leak fuel quantity, clean fuel (100% load)	kg/h	2.7	2.0	2.0	3.2	3.2
Lubricating oil system						
Pressure before engine, nom.	kPa	450	450	450	450	450
Pressure before engine, alarm	kPa	300	300	300	300	300
Pressure before engine, stop	kPa	200	200	200	200	200
Priming pressure, nom.	kPa	80	80	80	80	80
Priming pressure, alarm	kPa	50	50	50	50	50
Temperature before engine, nom.	°C	63	63	63	63	63
Temperature before engine, alarm	°C	80	80	80	80	80
Temperature after engine, abt.	°C	78	78	78	78	78
Pump capacity (main), direct driven	m^3/h	35	28	28	28	28
Pump capacity (main), separate	m^3/h	18	18	18	18	18
Pump capacity (priming), 50Hz/60Hz	m^3/h	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4
Oil volume, wet sump, nom.	m^3	0.27	0.27	0.27	0.27	0.27
Oil volume in separate system oil tank, nom.	m^3	1.1	0.7	0.7	1.0	1.1
Filter fineness, nom.	microns	15	15	15	15	15
Filters difference pressure, alarm	kPa	150	150	150	150	150
Oil consumption (100% load), abt.	g/kWh	0.5	0.5	0.5	0.5	0.5

Wärtsilä 4L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Cooling water system						
High temperature cooling water system						
Pressure before engine, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before engine, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before engine, max.	kPa	350	350	350	350	350
Temperature before engine, abt.	°C	83	83	83	83	83
Temperature after engine, nom.	°C	91	91	91	91	91
Temperature after engine, alarm	°C	105	105	105	105	105
Temperature after engine, stop	°C	110	110	110	110	110
Pump capacity, nom.	m^3/h	20	18	19	20	20
Pressure drop over engine	kPa	50	50	50	50	50
Pressure drop over central cooler, max	kPa	60	60	60	60	60
Water volume in engine	m^3	0.09	0.09	0.09	0.09	0.09
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Low temperature cooling water system						
Pressure before charge air cooler, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before charge air cooler, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before charge air cooler, max.	kPa	350	350	350	350	350
Temperature before charge air cooler	°C	25...38	25...38	25...38	25...38	25...38
Pump capacity, nom.	m^3/h	24	19	20	23	24
Pressure drop over charge air cooler	kPa	30	30	30	30	30
Pressure drop over oil cooler	kPa	30	30	30	30	30
Pressure drop over central cooler, max.	kPa	60	60	60	60	60
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Compressed air system (Note 4)						
Air pressure, nom.	kPa	3000	3000	3000	3000	3000
Air pressure, max.	kPa	3000	3000	3000	3000	3000
Air pressure, alarm	kPa	1800	1800	1800	1800	1800
Starting air consumption, start (manual)	Nm^3	0.4	0.4	0.4	0.4	0.4
Starting air consumption, start (remote)	Nm^3	1.2	1.2	1.2	1.2	1.2

Notes:

Note 1 At an ambient temperature of 25°C.

Note 2 ISO-optimized engine at ambient conditions according to ISO 3046/1. With engine driven pumps. Fuel net caloric value: 42700 kJ/kg. The figures are at 100% load and include engine driven pumps.

Note 3 According to ISO 3046/1, lower calorofic value 42 700 kJ/kg at constant engine speed, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.

Note 4 At remote and automatic starting, the consumption is 1.2 Nm3.

Subject to revision without notice.

3.2 Wärtsilä 6L20

Wärtsilä 6L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Engine output	kW	1200	780	810	1110	1200
Engine output	HP	1633	1061	1102	1510	1633
Cylinder bore	mm	200	200	200	200	200
Stroke	mm	280	280	280	280	280
Mean effective pressure	MPa	2.73	2.46	2.46	2.8	2.73
Mean piston speed	m/s	9.3	6.7	7.0	8.4	9.3
Minimum speed, FPP-installation	rpm	350				
Combustion air system						
Flow of air at 100% load	kg/s	2.42	1.36	1.43	2.27	2.42
Ambient air temperature, max.	°C	45	45	45	45	45
Air temperature after air cooler	°C	50...70	45...60	45...60	50...70	50...70
Air temperature after air cooler, alarm	°C	75	75	75	75	75
Exhaust gas system (Note 1)						
Exhaust gas flow (100% load)	kg/s	2.49	1.4	1.48	2.33	2.49
Exhaust gas flow (85% load)	kg/s	2.16	1.21	1.29	1.99	2.16
Exhaust gas flow (75% load)	kg/s	1.89	1.09	1.16	1.8	1.95
Exhaust gas flow (50% load)	kg/s	1.23	0.79	0.84	1.28	1.4
Exhaust gas temperature after turbocharger (100% load)	°C	335	370	370	335	335
Exhaust gas temperature after turbocharger (85% load)	°C	340	380	380	335	340
Exhaust gas temperature after turbocharger (75% load)	°C	350	380	380	340	340
Exhaust gas temperature after turbocharger (50% load)	°C	390	390	390	345	355
Exhaust gas temperature after cylinder, alarm	°C	450	400	400	400	400
Exhaust gas back pressure, rec. max.	kPa	3.0	3.0	3.0	3.0	3.0
Exhaust gas pipe diameter, min	mm	400	300	300	400	400
Heat balance (Note 2)						
Jacket water	kW	245	189	194	220	245
Charge air	kW	385	201	223	345	385
Lubricating oil	kW	157	106	108	142	157
Exhaust gases	kW	810	530	549	758	810
Radiation, etc.	kW	49	44	47	45	49
Fuel system (Note 3)						
Pressure before injection pumps	kPa	600	600	600	600	600
Pump capacity (MDF), engine driven	m³/h	1.49	0.87	0.9	1.34	1.49
Fuel consumption (100% load)	g/kWh	191	191	192	190	191
Fuel consumption (85% load)	g/kWh	188	192	193	189	190
Fuel consumption (75% load)	g/kWh	188	194	194	190	190
Fuel consumption (50% load)	g/kWh	196	203	203	198	198
Leak fuel quantity, clean fuel (100% load)	kg/h	5.0	3.0	3.5	4.5	5.0
Lubricating oil system						
Pressure before engine, nom.	kPa	450	450	450	450	450
Pressure before engine, alarm	kPa	300	300	300	300	300
Pressure before engine, stop	kPa	200	200	200	200	200
Priming pressure, nom.	kPa	80	80	80	80	80
Priming pressure, alarm	kPa	50	50	50	50	50
Temperature before engine, nom.	°C	63	63	63	63	63
Temperature before engine, alarm	°C	80	80	80	80	80
Temperature after engine, abt.	°C	78	78	78	78	78
Pump capacity (main), direct driven	m³/h	50	35	35	35	35
Pump capacity (main), separate	m³/h	21	21	21	21	21
Pump capacity (priming), 50Hz/60Hz	m³/h	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4
Oil volume, wet sump, nom.	m³	0.38	0.38	0.38	0.38	0.38
Oil volume in separate system oil tank, nom.	m³	1.6	1.1	1.1	1.5	1.6
Filter fineness, nom.	microns	25	25	25	25	25
Filters difference pressure, alarm	kPa	150	150	150	150	150
Oil consumption (100% load), abt.	g/kWh	0.5	0.5	0.5	0.5	0.5
Cooling water system						
High temperature cooling water system						
Pressure before engine, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before engine, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before engine, max.	kPa	350	350	350	350	350
Temperature before engine, abt.	°C	83	83	83	83	83

Wärtsilä 6L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Temperature after engine, nom.	°C	91	91	91	91	91
Temperature after engine, alarm	°C	105	105	105	105	105
Temperature after engine, stop	°C	110	110	110	110	110
Pump capacity, nom.	m³/h	30	27	28	29	30
Pressure drop over engine	kPa	50	50	50	50	50
Pressure drop over central cooler, max	kPa	60	60	60	60	60
Water volume in engine	m³	0.12	0.12	0.12	0.12	0.12
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Low temperature cooling water system						
Pressure before charge air cooler, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before charge air cooler, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before charge air cooler, max.	kPa	350	350	350	350	350
Temperature before charge air cooler	°C	25...38	25...38	25...38	25...38	25...38
Pump capacity, nom.	m³/h	36	29	30	34	36
Pressure drop over charge air cooler	kPa	30	30	30	30	30
Pressure drop over oil cooler	kPa	30	30	30	30	30
Pressure drop over central cooler, max.	kPa	60	60	60	60	60
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Compressed air system (Note 4)						
Air pressure, nom.	kPa	3000	3000	3000	3000	3000
Air pressure, max.	kPa	3000	3000	3000	3000	3000
Air pressure, alarm	kPa	1800	1800	1800	1800	1800
Starting air consumption, start (manual)	Nm³	0.4	0.4	0.4	0.4	0.4
Starting air consumption, start (remote)	Nm³	1.2	1.2	1.2	1.2	1.2

Notes:

Note 1 At an ambient temperature of 25°C.

Note 2 ISO-optimized engine at ambient conditions according to ISO 3046/1. With engine driven pumps. Fuel net caloric value: 42700 kJ/kg. The figures are at 100% load and include engine driven pumps.

Note 3 According to ISO 3046/1, lower calorofic value 42 700 kJ/kg at constant engine speed, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.

Note 4 At remote and automatic starting, the consumption is 1.2 Nm3.

Subject to revision without notice.

3.3 Wärtsilä 8L20

Wärtsilä 8L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Engine output	kW	1600	1040	1080	1360	1440
Engine output	HP	2176	1415	1469	1850	1959
Cylinder bore	mm	200	200	200	200	200
Stroke	mm	280	280	280	280	280
Mean effective pressure	MPa	2.73	2.46	2.46	2.58	2.46
Mean piston speed	m/s	9.3	6.7	7.0	8.4	9.3
Minimum speed, FPP-installation	rpm	350				
Combustion air system						
Flow of air at 100% load	kg/s	3.28	1.96	2.04	2.79	2.85
Ambient air temperature, max.	°C	45	45	45	45	45
Air temperature after air cooler	°C	50...70	45...60	45...60	45...60	45...60
Air temperature after air cooler, alarm	°C	75	75	75	75	75
Exhaust gas system (Note 1)						
Exhaust gas flow (100% load)	kg/s	3.42	2.02	2.1	2.8	2.93
Exhaust gas flow (85% load)	kg/s	2.84	1.74	1.81	2.45	2.53
Exhaust gas flow (75% load)	kg/s	2.48	1.57	1.62	2.2	2.29
Exhaust gas flow (50% load)	kg/s	1.62	1.11	1.15	1.6	1.64
Exhaust gas temperature after turbocharger (100% load)	°C	345	360	360	345	340
Exhaust gas temperature after turbocharger (85% load)	°C	345	360	360	340	340
Exhaust gas temperature after turbocharger (75% load)	°C	355	360	360	340	340
Exhaust gas temperature after turbocharger (50% load)	°C	385	370	370	350	350
Exhaust gas temperature after cylinder, alarm	°C	450	400	400	400	400
Exhaust gas back pressure, rec. max.	kPa	3.0	3.0	3.0	3.0	3.0
Exhaust gas pipe diameter, min	mm	450	350	350	400	400
Heat balance (Note 2)						
Jacket water	kW	335	244	254	300	330
Charge air	kW	480	306	322	400	442
Lubricating oil	kW	230	162	167	200	219
Exhaust gases	kW	1167	684	708	941	969
Radiation, etc.	kW	66	55	57	58	61
Fuel system (Note 3)						
Pressure before injection pumps	kPa	600	600	600	600	600
Pump capacity (MDF), engine driven	m³/h	1.92	1.48	1.54	1.73	1.92
Fuel consumption (100% load)	g/kWh	192	193	193	192	193
Fuel consumption (85% load)	g/kWh	189	194	194	191	192
Fuel consumption (75% load)	g/kWh	190	195	195	191	193
Fuel consumption (50% load)	g/kWh	196	202	202	200	200
Leak fuel quantity, clean fuel (100% load)	kg/h	6.2	4.0	4.5	5.5	6.0
Lubricating oil system						
Pressure before engine, nom.	kPa	450	450	450	450	450
Pressure before engine, alarm	kPa	300	300	300	300	300
Pressure before engine, stop	kPa	200	200	200	200	200
Priming pressure, nom.	kPa	80	80	80	80	80
Priming pressure, alarm	kPa	50	50	50	50	50
Temperature before engine, nom.	°C	63	63	63	63	63
Temperature before engine, alarm	°C	80	80	80	80	80
Temperature after engine, abt.	°C	78	78	78	78	78
Pump capacity (main), direct driven	m³/h	65	50	50	50	50
Pump capacity (main), separate	m³/h	27	27	27	27	27
Pump capacity (priming), 50Hz/60Hz	m³/h	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4
Oil volume, wet sump, nom.	m³	0.49	0.49	0.49	0.49	0.49
Oil volume in separate system oil tank, nom.	m³	2.2	1.4	1.5	1.8	1.9
Filter fineness, nom.	microns	25	25	25	25	25
Filters difference pressure, alarm	kPa	150	150	150	150	150
Oil consumption (100% load), abt.	g/kWh	0.5	0.5	0.5	0.5	0.5
Cooling water system						
High temperature cooling water system						
Pressure before engine, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before engine, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before engine, max.	kPa	350	350	350	350	350
Temperature before engine, abt.	°C	83	83	83	83	83

Wärtsilä 8L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Temperature after engine, nom.	°C	91	91	91	91	91
Temperature after engine, alarm	°C	105	105	105	105	105
Temperature after engine, stop	°C	110	110	110	110	110
Pump capacity, nom.	m³/h	40	35	37	39	40
Pressure drop over engine	kPa	50	50	50	50	50
Pressure drop over central cooler, max	kPa	60	60	60	60	60
Water volume in engine	m³	0.15	0.15	0.15	0.15	0.15
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Low temperature cooling water system						
Pressure before charge air cooler, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before charge air cooler, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before charge air cooler, max.	kPa	350	350	350	350	350
Temperature before charge air cooler	°C	25...38	25...38	25...38	25...38	25...38
Pump capacity, nom.	m³/h	48	38	40	45	48
Pressure drop over charge air cooler	kPa	30	30	30	30	30
Pressure drop over oil cooler	kPa	30	30	30	30	30
Pressure drop over central cooler, max.	kPa	60	60	60	60	60
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Compressed air system (Note 4)						
Air pressure, nom.	kPa	3000	3000	3000	3000	3000
Air pressure, max.	kPa	3000	3000	3000	3000	3000
Air pressure, alarm	kPa	1800	1800	1800	1800	1800
Starting air consumption, start (manual)	Nm³	0.4	0.4	0.4	0.4	0.4
Starting air consumption, start (remote)	Nm³	1.2	1.2	1.2	1.2	1.2

Notes:

Note 1 At an ambient temperature of 25°C.

Note 2 ISO-optimized engine at ambient conditions according to ISO 3046/1. With engine driven pumps. Fuel net caloric value: 42700 kJ/kg. The figures are at 100% load and include engine driven pumps.

Note 3 According to ISO 3046/1, lower calorofic value 42 700 kJ/kg at constant engine speed, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.

Note 4 At remote and automatic starting, the consumption is 1.2 Nm3.

Subject to revision without notice.

3.4 Wärtsilä 9L20

Wärtsilä 9L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Engine output	kW	1800	1170	1215	1665	1800
Engine output	HP	2448	1592	1653	2265	2448
Cylinder bore	mm	200	200	200	200	200
Stroke	mm	280	280	280	280	280
Mean effective pressure	MPa	2.73	2.46	2.46	2.8	2.73
Mean piston speed	m/s	9.3	6.7	7.0	8.4	9.3
Minimum speed, FPP-installation	rpm	350				
Combustion air system						
Flow of air at 100% load	kg/s	3.74	1.98	2.11	3.3	3.74
Ambient air temperature, max.	°C	45	45	45	45	45
Air temperature after air cooler	°C	50...70	45...60	45...60	50...70	50...70
Air temperature after air cooler, alarm	°C	75	75	75	75	75
Exhaust gas system (Note 1)						
Exhaust gas flow (100% load)	kg/s	3.85	2.05	2.18	3.4	3.85
Exhaust gas flow (85% load)	kg/s	3.2	1.79	1.91	2.95	3.36
Exhaust gas flow (75% load)	kg/s	2.79	1.62	1.73	2.66	3.06
Exhaust gas flow (50% load)	kg/s	1.82	1.18	1.27	1.88	2.22
Exhaust gas temperature after turbocharger (100% load)	°C	345	360	360	345	345
Exhaust gas temperature after turbocharger (85% load)	°C	345	360	360	345	345
Exhaust gas temperature after turbocharger (75% load)	°C	355	370	370	345	345
Exhaust gas temperature after turbocharger (50% load)	°C	385	380	380	355	360
Exhaust gas temperature after cylinder, alarm	°C	450	400	400	400	400
Exhaust gas back pressure, rec. max.	kPa	3.0	3.0	3.0	3.0	3.0
Exhaust gas pipe diameter, min	mm	500	350	350	450	500
Heat balance (Note 2)						
Jacket water	kW	350	280	291	340	350
Charge air	kW	540	342	355	515	540
Lubricating oil	kW	233	177	183	220	233
Exhaust gases	kW	1294	771	800	1142	1294
Radiation, etc.	kW	74	63	66	68	74
Fuel system (Note 3)						
Pressure before injection pumps	kPa	600	600	600	600	600
Pump capacity (MDF), engine driven	m³/h	1.92	1.48	1.54	1.73	1.92
Fuel consumption (100% load)	g/kWh	192	193	193	191	192
Fuel consumption (85% load)	g/kWh	189	193	193	190	191
Fuel consumption (75% load)	g/kWh	190	194	194	191	192
Fuel consumption (50% load)	g/kWh	196	202	202	199	200
Leak fuel quantity, clean fuel (100% load)	kg/h	6.5	4.5	5.0	6.0	6.5
Lubricating oil system						
Pressure before engine, nom.	kPa	450	450	450	450	450
Pressure before engine, alarm	kPa	300	300	300	300	300
Pressure before engine, stop	kPa	200	200	200	200	200
Priming pressure, nom.	kPa	80	80	80	80	80
Priming pressure, alarm	kPa	50	50	50	50	50
Temperature before engine, nom.	°C	63	63	63	63	63
Temperature before engine, alarm	°C	80	80	80	80	80
Temperature after engine, abt.	°C	78	78	78	78	78
Pump capacity (main), direct driven	m³/h	65	50	50	50	50
Pump capacity (main), separate	m³/h	30	30	30	30	30
Pump capacity (priming), 50Hz/60Hz	m³/h	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4	6.9 / 8.4
Oil volume, wet sump, nom.	m³	0.55	0.55	0.55	0.55	0.55
Oil volume in separate system oil tank, nom.	m³	2.4	1.6	1.6	2.2	2.4
Filter fineness, nom.	microns	25	25	25	25	25
Filters difference pressure, alarm	kPa	150	150	150	150	150
Oil consumption (100% load), abt.	g/kWh	0.5	0.5	0.5	0.5	0.5
Cooling water system						
High temperature cooling water system						
Pressure before engine, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before engine, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before engine, max.	kPa	350	350	350	350	350
Temperature before engine, abt.	°C	83	83	83	83	83

Wärtsilä 9L20		ME	AE/DE	AE/DE	AE/DE	AE/DE
Engine speed	**RPM**	**1000**	**720**	**750**	**900**	**1000**
Temperature after engine, nom.	°C	91	91	91	91	91
Temperature after engine, alarm	°C	105	105	105	105	105
Temperature after engine, stop	°C	110	110	110	110	110
Pump capacity, nom.	m³/h	45	40	42	44	45
Pressure drop over engine	kPa	50	50	50	50	50
Pressure drop over central cooler, max	kPa	60	60	60	60	60
Water volume in engine	m³	0.16	0.16	0.16	0.16	0.16
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Low temperature cooling water system						
Pressure before charge air cooler, nom.	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Pressure before charge air cooler, alarm	kPa	100 + static	100 + static	100 + static	100 + static	100 + static
Pressure before charge air cooler, max.	kPa	350	350	350	350	350
Temperature before charge air cooler	°C	25...38	25...38	25...38	25...38	25...38
Pump capacity, nom.	m³/h	54	43	45	50	54
Pressure drop over charge air cooler	kPa	30	30	30	30	30
Pressure drop over oil cooler	kPa	30	30	30	30	30
Pressure drop over central cooler, max.	kPa	60	60	60	60	60
Pressure from expansion tank	kPa	70...150	70...150	70...150	70...150	70...150
Delivery head of stand-by pump	kPa	200 + static	200 + static	200 + static	200 + static	200 + static
Compressed air system (Note 4)						
Air pressure, nom.	kPa	3000	3000	3000	3000	3000
Air pressure, max.	kPa	3000	3000	3000	3000	3000
Air pressure, alarm	kPa	1800	1800	1800	1800	1800
Starting air consumption, start (manual)	Nm³	0.4	0.4	0.4	0.4	0.4
Starting air consumption, start (remote)	Nm³	1.2	1.2	1.2	1.2	1.2

Notes:

Note 1 At an ambient temperature of 25°C.

Note 2 ISO-optimized engine at ambient conditions according to ISO 3046/1. With engine driven pumps. Fuel net caloric value: 42700 kJ/kg. The figures are at 100% load and include engine driven pumps.

Note 3 According to ISO 3046/1, lower calorofic value 42 700 kJ/kg at constant engine speed, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.

Note 4 At remote and automatic starting, the consumption is 1.2 Nm3.

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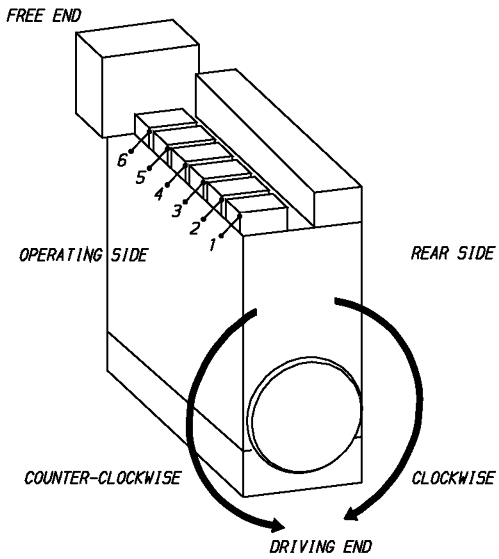
4. Description of the engine

4.1 Definitions

Figure 4.1 In-line engine (1V93C0029)



4.2 Main components

The dimensions and weights of engine parts are shown in the chapter for dimensions and weights.

4.2.1 Engine block

The engine block is a one piece nodular cast iron component with integrated channels for lubricating oil and cooling water.

The main bearing caps are fixed from below by two hydraulically tensioned screws. They are guided sideways by the engine block at the top as well as at the bottom. Hydraulically tightened horizontal side screws at the lower guiding provide a very rigid crankshaft bearing.

4.2.2 Crankshaft

The crankshaft is forged in one piece and mounted on the engine block in an under-slung way.

4.2.3 Connecting rod

The connecting rod is of forged alloy steel. All connecting rod studs are hydraulically tightened. Oil is led to the gudgeon pin bearing and piston through a bore in the connecting rod.

4.2.4 Main bearings and big end bearings

The main bearings and the big end bearings are of the Al based bi-metal type with steel back.

4.2.5 Cylinder liner

The cylinder liners are centrifugally cast of a special grey cast iron alloy developed for good wear resistance and high strength. They are of wet type, sealed against the engine block metallically at the upper part and by O-rings at the lower part. To eliminate the risk of bore polishing the liner is equipped with an anti-polishing ring.

4.2.6 Piston

The piston is of composite design with nodular cast iron skirt and steel crown. The piston skirt is pressure lubricated, which ensures a well-controlled oil flow to the cylinder liner during all operating conditions. Oil is fed through the connecting rod to the cooling spaces of the piston. The piston cooling operates according to the cocktail shaker principle. The piston ring grooves in the piston top are hardened for better wear resistance.

4.2.7 Piston rings

The piston ring set consists of two directional compression rings and one spring-loaded conformable oil scraper ring. All rings are chromium-plated and located in the piston crown.

4.2.8 Cylinder head

The cylinder head is made of grey cast iron. The thermally loaded flame plate is cooled efficiently by cooling water led from the periphery radially towards the centre of the head. The bridges between the valves cooling channels are drilled to provide the best possible heat transfer.

The mechanical load is absorbed by a strong intermediate deck, which together with the upper deck and the side walls form a box section in the four corners of which the hydraulically tightened cylinder head bolts are situated. The exhaust valve seats are directly water-cooled.

All valves are equipped with valve rotators.

4.2.9 Camshaft and valve mechanism

There is one cam piece for each cylinder with separate bearing in between. The drop forged completely hardened camshaft pieces have fixed cams. The camshaft bearing housings are integrated in the engine block casting and are thus completely closed. The camshaft covers, one for each cylinder, seal against the engine block with a closed O-ring profile.

The valve tappets are of piston type with self-adjustment of roller against cam to give an even distribution of the contact pressure. The valve springs ensure that the valve mechanism is dynamically stable.

4.2.10 Camshaft drive

The camshafts are driven by the crankshaft through a gear train.

4.2.11 Turbocharging and charge air cooling

The selected turbo charger offers the ideal combination of high-pressure ratios and good efficiency.

The charge air cooler is single stage type and cooled by LT-water.

4.2.12 Fuel injection equipment

The injection pumps are one-cylinder pumps located in the “hot-box”, which has the following functions:

- Housing for the injection pump element
- Fuel supply channel along the whole engine
- Fuel return channel from each injection pump
- Lubricating oil supply to the valve mechanism
- Guiding for the valve tappets

The injection pumps have built-in roller tappets and are through-flow type to enable heavy fuel operation. They are also equipped with a stop cylinder, which is connected to the electro-pneumatic overspeed protection system.

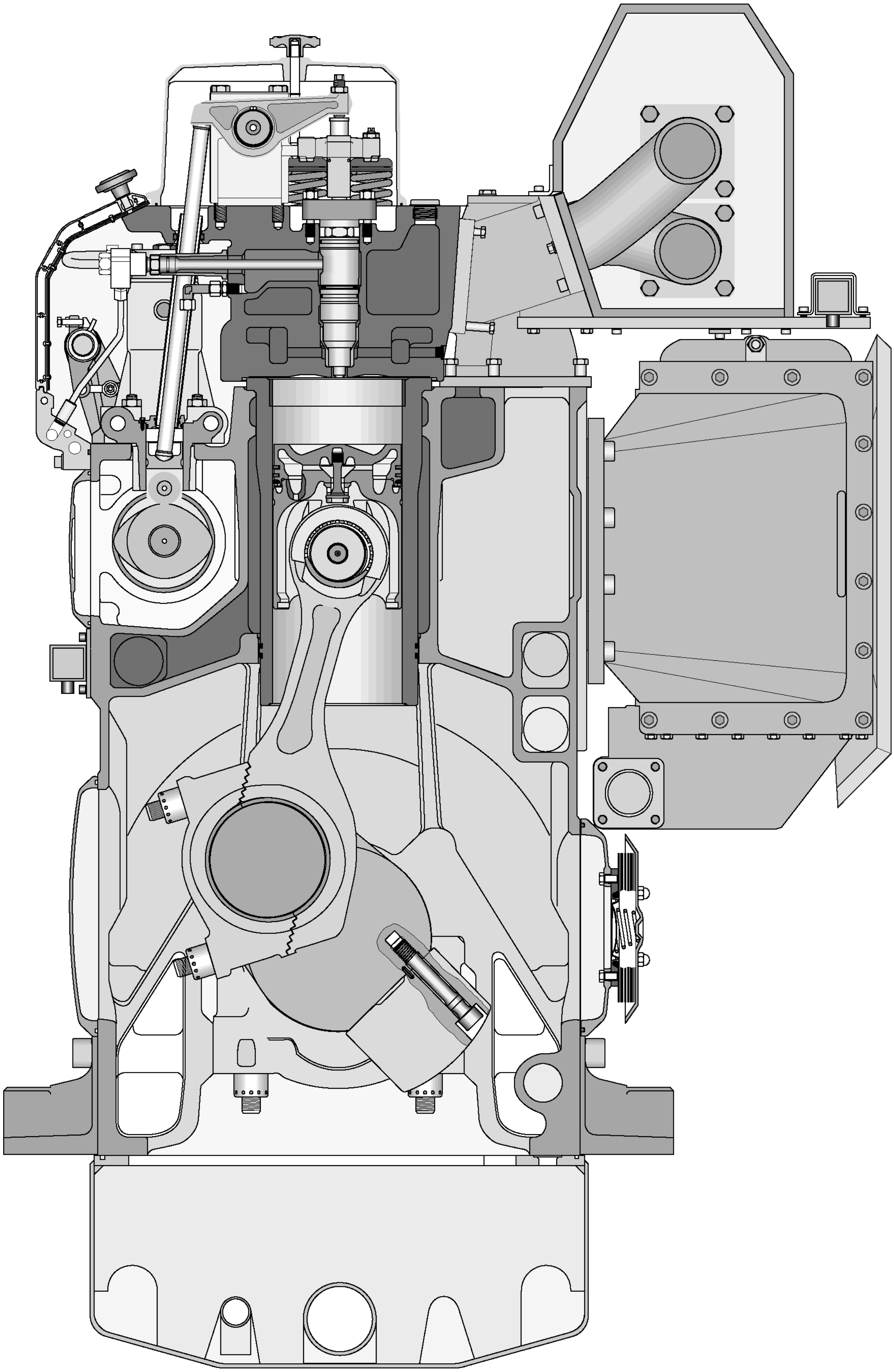
The injection valve is centrally located in the cylinder head and the fuel is admitted sideways through a high pressure connection screwed in the nozzle holder. The injection pipe between the injection pump and the high pressure connection is well protected inside the hot box. The high pressure side of the injection system is completely separated from the hot parts of the exhaust gas components.

4.2.13 Exhaust pipes

The complete exhaust gas system is enclosed in an insulated box consisting of easily removable panels. Mineral wool is used as insulating material.

4.3 Cross sections of the engine

Figure 4.2 Cross sections of the engine



4.4 Overhaul intervals and expected life times

The following overhaul intervals and lifetimes are for guidance only. Actual figures will be different depending on service conditions. Expected component lifetimes have been adjusted to match overhaul intervals.

In this list HFO is based on HFO2 specification stated in the chapter for general data and outputs.

Table 4.1 Time between overhauls and expected component lifetimes

	HFO	MDF	HFO	MDF
	Time between overhauls (h)	**Time between overhauls (h)**	**Expected components lifetimes (h)**	**Expected components lifetimes (h)**
Main bearing	12000	16000	36000	48000
Big end bearing	12000	16000	24000	32000
Gudgeon pin bearing	12000	16000	48000	48000
Camshaft bearing bush	16000	16000	32000	32000
Camshaft intermed. gear bearing	16000	16000	32000	32000
Balancing shaft bearing, 4L20	12000	16000	24000	32000
Cylinder head	12000	16000		
Inlet valve	12000	16000	36000	32000
Inlet valve seat	12000	16000	36000	32000
Exhaust valve	12000	16000	24000	32000
Exhaust valve seat	12000	16000	36000	32000
Valve guide, EX	12000	16000	24000	32000
Valve guide, IN	12000	16000	36000	48000
Piston crown	12000	16000	24000	48000
Piston rings	12000	16000	12000	16000
Cylinder liner	12000	16000	48000	64000
Antipolishing ring	12000	16000	12000	16000
Connecting rod	12000	16000		
Connecting rod screws	12000	16000	24000	32000
Valve tappet and roller			24000	32000
Injection pump tappet and roller			24000	32000
Injection element	12000	16000	24000	32000
Injection valve	6000	8000		
Injection nozzle	6000	8000	6000	8000
Water pump shaft seal	12000	12000	12000	12000
Water pump bearing			24000	24000
Turbocharger	24000	24000		
Governor	12000	12000		
Vibration damper	Acc. to manuf.	Acc. to manuf.		

5. Piping design, treatment and installation

5.1 General

This chapter provides general guidelines for the design, construction and installation of piping systems, however, not excluding other solutions of at least equal standard.

Fuel, lubricating oil, fresh water and compressed air piping is usually made in seamless carbon steel (DIN 2448) and seamless precision tubes in carbon or stainless steel (DIN 2391), exhaust gas piping in welded pipes of corten or carbon steel (DIN 2458). The pipes on the freshwater side of the cooling water system must not be galvanized. Sea-water piping should be in Cunifer or hot dip galvanized steel.

Attention shall be given to the fire risk aspects. The fuel supply and return lines shall be designed so that they can be fitted without tension. When flexible hoses are used, they shall be of class approved type. If flexible hoses are used in the compressed air system an outlet valve shall be fitted in front of the hose(s).

It is recommended to make a fitting order plan prior to construction.

The following aspects shall be taken into consideration:

- In the tank top sections (blocks) larger pipes shall be installed prior to smaller and if/when the deck sections are upside down the large pipes comes closer to the underside of the deck.
- The main lines shall be installed before the branches
- Technically more difficult systems to be built before simpler systems
- Pockets shall be avoided and when not possible equipped with drain plugs and air vents
- Leak fuel drain pipes shall have continuous slope
- Vent pipes shall be continuously rising
- Flanged connections shall be used, cutting ring joints for precision tubes

Maintenance access and dismounting space of valves, coolers and other devices shall be taken into consideration. Flange connections and other joints shall be located so that dismounting of the equipment can be made with reasonable effort.

5.2 Pipe dimensions

When selecting the pipe dimensions, take into account:

- The relevant pipe material and its resistance to against corrosion/erosion
- Allowed pressure loss in the circuit vs delivery head of the pump
- Required net positive suction head (NPSH) for pumps (suction lines)
- In small pipe sizes the max acceptable velocity is usually somewhat lower than in large pipes of equal length
- Due to increased risk of fouling and pitting velocities below 1 m/s should not be designed for in sea water piping
- To ensure a sufficient NPSH velocities in suction pipes are typically about 2/3 of those in delivery pipes

Recommended maximum fluid velocities are given as guidance in table *5.1*.

Table 5.1 Recommended maximum velocities for guidance

Piping	Pipe material	Max velocity [m/s]
Fuel piping (MDF and HFO)	Black steel	1.0
Lubricating oil piping	Black steel	1.5
Fresh water piping	Black steel	2.5

Piping	Pipe material	Max velocity [m/s]
Sea water piping	Galvanized steel	2.5
	Aluminium brass	2.5
	10/90 copper-nickel-iron	3.0
	70/30 copper-nickel	4.5
	Rubber lined pipes	4.5

NOTE! The diameter of gas fuel and compressed air piping depends only on the allowed pressure loss in the piping, which has to be calculated project specifically.

5.3 Trace heating

The following pipes shall be equipped with trace heating (steam, thermal oil or electrical). It shall be possible to shut off the trace heating.

- All heavy fuel pipes
- All leak fuel and filter flushing pipes carrying heavy fuel

5.4 Pressure class

The pressure class of the piping should be higher than or equal to the design pressure, which should be higher than or equal to the highest operating (working) pressure. The highest operating (working) pressure is equal to the setting of the safety valve in a system.

The pressure in the system can:

- Originate from a positive displacement pump
- Be a combination of the static pressure and the pressure on the highest point of the pump curve for a centrifugal pump
- Rise in an isolated system if the liquid is heated e.g. preheating of a system

Within this Project Guide there are tables attached to drawings, which specify pressure classes of connections. The pressure class of a connection can be higher than the pressure class required for the pipe.

Example 1:

The fuel pressure before the engine should be 7 bar. The safety filter in dirty condition may cause a pressure loss of 1.0 bar. The viscosimeter, automatic filter, preheater and piping may cause a pressure loss of 2.5 bar. Consequently the discharge pressure of the circulating pumps may rise to 10.5 bar, and the safety valve of the pump shall thus be adjusted e.g. to 12 bar.

- A design pressure of not less than 12 bar has to be selected.
- The nearest pipe class to be selected is PN16.
- Piping test pressure is normally 1.5 x the design pressure = 18 bar.

Example 2:

The pressure on the suction side of the cooling water pump is 1.0 bar. The delivery head of the pump is 3.0 bar, leading to a discharge pressure of 4.0 bar. The highest point of the pump curve (at or near zero flow) is 1.0 bar higher than the nominal point, and consequently the discharge pressure may rise to 5.0 bar (with closed or throttled valves).

- Consequently a design pressure of not less than 5.0 bar shall be selected.
- The nearest pipe class to be selected is PN6.
- Piping test pressure is normally 1.5 x the design pressure = 7.5 bar.

Standard pressure classes are PN4, PN6, PN10, PN16, PN25, PN40, etc.

5.5 Pipe class

The principle of categorization of piping systems in classes (e.g. DNV) or groups (e.g. ABS) by the classification societies can be used for choosing of:

- Type of joint to be used
- Heat treatment
- Welding procedure
- Test method

Systems with high design pressures and temperatures and hazardous media belong to class I (or group I), others to II or III as applicable. Quality requirements are highest in class I.

Examples of classes of piping systems as per DNV rules are presented in the table below.

Table 5.2 Classes of piping systems as per DNV rules.

Media	Class I		Class II		Class III	
	bar	°C	bar	°C	bar	°C
Steam	> 16	or > 300	< 16	and < 300	< 7	and < 170
Flammable fluid	> 16	or > 150	< 16	and < 150	< 7	and < 60
Other media	> 40	or > 300	< 40	and < 300	< 16	and < 200

5.6 Insulation

The following pipes shall be insulated:

- All trace heated pipes
- Exhaust gas pipes

Insulation is also recommended for:

- Pipes between engine or system oil tank and lubricating oil separator
- Pipes between engine and jacket water preheater
- For personnel protection work safety any exposed parts of pipes at walkways, etc., to be insulated to avoid excessive temperatures and risks for personnel injury.

In addition to the operational aspects of the different piping systems requiring insulation the risks of fire and personnel injury due to hot surfaces shall be given attention by insulating and/or shielding of hot surfaces.

5.7 Local gauges

Local thermometers should be installed wherever a new temperature occurs, i.e. before and after heat exchangers, etc.

Pressure gauges should be installed on the suction and discharge side of each pump.

5.8 Cleaning procedures

Instructions shall be given to manufacturers and/or fitters of how different piping systems shall be treated, cleaned and protected before and during transportation and before block assembly or assembly in the hull. All piping should be checked to be clean from debris before installation and joining. All piping should be cleaned according to the procedures listed below.

Table 5.3 Pipe cleaning

System	Methods
Fuel oil	A,B,C,D,F
Lubricating oil	A,B,C,D,F
Starting air	A,B,C
Cooling water	A,B,C

System	Methods
Exhaust gas	A,B,C
Charge air	A,B,C

A = Washing with alkaline solution in hot water at 80°C for degreasing (only if pipes have been greased)

B = Removal of rust and scale with steel brush (not required for seamless precision tubes)

C = Purging with compressed air

D = Pickling

F = Flushing

5.8.1 Pickling

Pipes are pickled in an acid solution of 10% hydrochloric acid and 10% formaline inhibitor for 4-5 hours, rinsed with hot water and blown dry with compressed air.

After the acid treatment the pipes are treated with a neutralizing solution of 10% caustic soda and 50 grams of trisodiumphosphate per litre of water for 20 minutes at 40...50°C, rinsed with hot water and blown dry with compressed air.

5.8.2 Flushing

More detailed recommendations on flushing procedures are when necessary described under the relevant chapters concerning the fuel oil system and the lubricating oil system. Provisions are to be made to ensure that necessary temporary bypasses can be arranged and that flushing hoses, filters and pumps will be available when required.

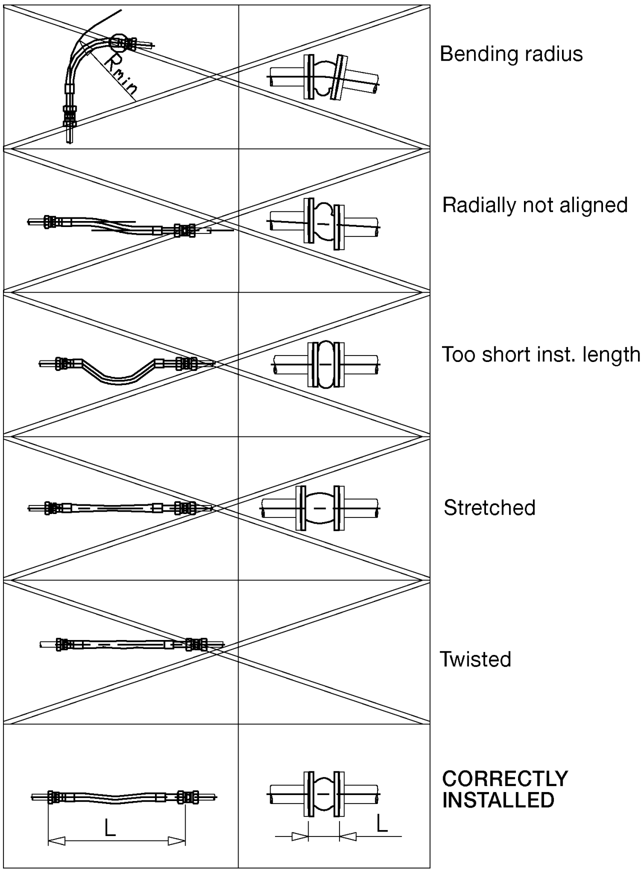
5.9 Flexible pipe connections

Legislative bodies and Classification Societies have stated in their recent rules that pressurized flexible connections carrying flammable fluids or compressed air need to be type approved.

Great care must be taken to ensure the proper installation of flexible pipe connections between resiliently mounted engines and ship's piping.

- Flexible pipe connections must not be twisted
- Installation length of flexible pipe connections must be correct
- Minimum bending radius must respected
- Piping must be concentrically aligned
- When specified the flow direction must be observed
- Mating flanges shall be clean from rust, burrs and anticorrosion coatings
- Bolts are to be tightened crosswise in several stages
- Flexible elements must not be painted
- Rubber bellows must be kept clean from oil and fuel
- The piping must be rigidly supported close to the flexible piping connections.

Figure 5.1 Flexible hoses (4V60B0100)



5.10 Clamping of pipes

It is important to correctly support piping around flexible connections to prevent damage caused by vibration. The following guidelines should be applied:

- The pipe clamps and supports next to engine must be very rigid and welded to the steel structure of the foundation.
- First support should be located as close as possible to the flexible connection. Next support should be 0.3-0.5 m from the first support.
- First three supports closest to the engine or generating set should be fixed supports. Where necessary, sliding supports can be used after these three fixed supports to allow thermal expansion of the pipe.
- Supports should never be welded directly to the pipe. Either pipe clamps or flange supports should be used for flexible connection.
- Holes through structures should be machined (flame cutting should not be used).

Examples of flange support structures are shown in drawing 4V60L0796.

A typical pipe clamp for a fixed support is shown in drawing 4V61H0842. Pipe clamps must be made of steel; plastic clamps or similar may not be used.

Figure 5.2 Flange supports of flexible pipe connections (4V60L0796)

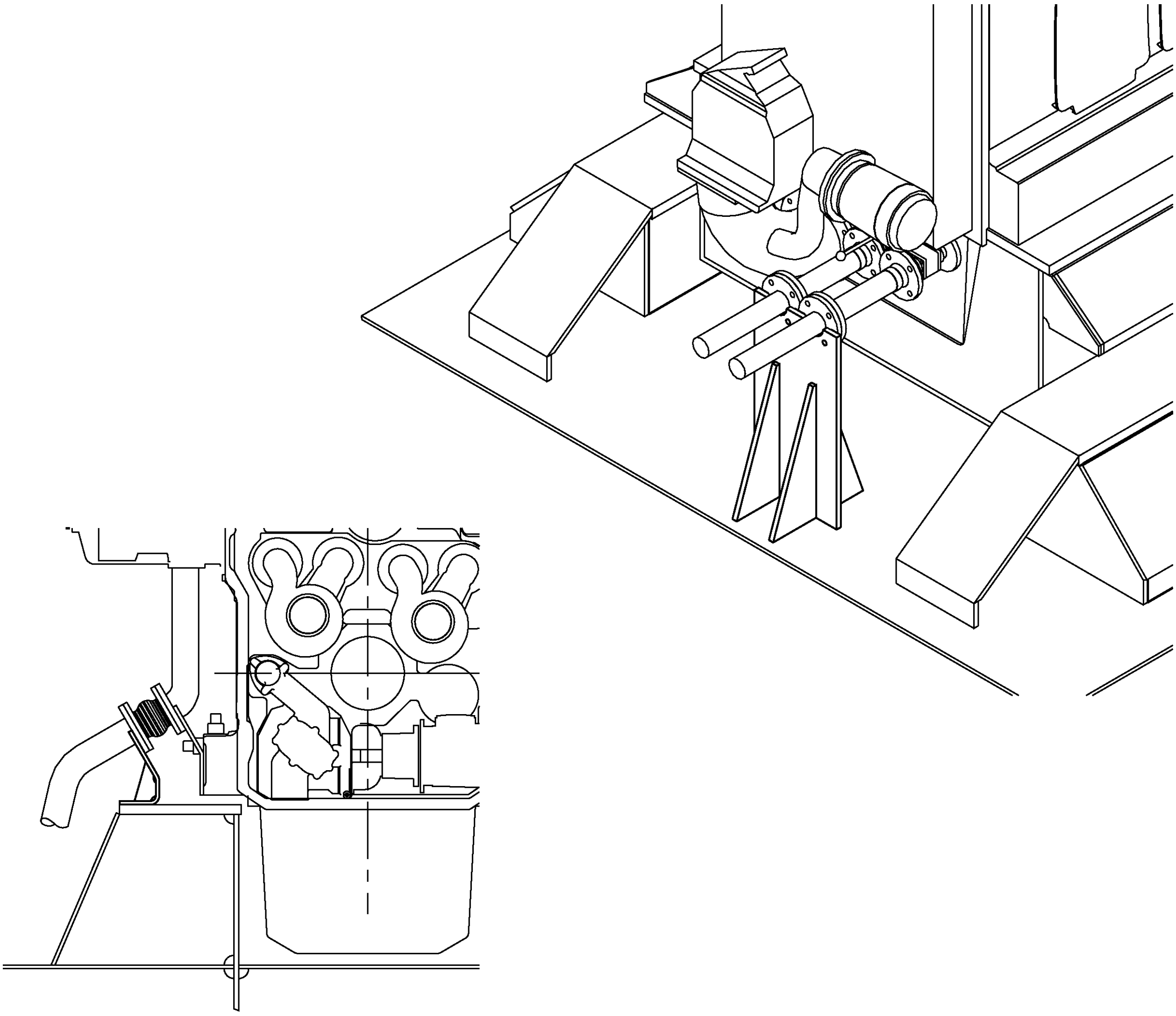
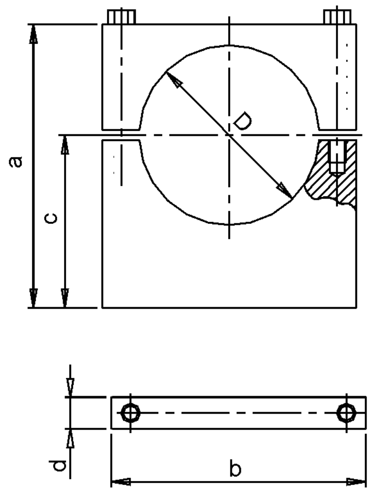


Figure 5.3 Pipe clamp for fixed support (4V61H0842)



DN	d_u [mm]	D [mm]	a [mm]	b [mm]	c [mm]	d [mm]	BOLTS
25	33.7	35	150	80	120	25	M10x50
32	42.4	43	150	75	120	25	M10x50
40	48.3	48	154.5	100	115	25	M12x60
50	60.3	61	185	100	145	25	M12x60
65	76.1	76.5	191	115	145	25	M12x70
80	88.9	90	220	140	150	30	M12x90
100	114.3	114.5	196	170	121	25	M12x100
125	139.7	140	217	200	132	30	M16x120
150	168.3	170	237	240	132	30	M16x140
200	219.1	220	295	290	160	30	M16x160
250	273.0	274	355	350	190	30	M16x200

d_u = Pipe outer diameter

6. Fuel oil system

6.1 General

Fuel characteristics of the fuels are presented under heading *Fuel characteristics* in the chapter *General data and outputs.*

6.1.1 Operating principles

The engine needs regulated fuel system before and after the engine to control viscosity and temperature of the fuel. Fuel systems are recommended to be closed due to better control of viscosity and temperature and conservation of the heating energy.

Fuel heating and cooling

The fuel temperature has to be controlled so that the viscosity of the fuel before injection pumps is stable and according to the limits specified in chapter *General data and outputs.*

6.1.2 Black out starting

For stand-by generating set engines sufficient fuel pressure for a safe start must be ensured in a case of a black out. This can be done with:

- A gravity tank min. 15 m above the engine centerline
- A pneumatic emergency pump (1P11)
- An electric motor driven pump (1P11) fed from an emergency supply

6.1.3 Number of engines

In multi-engine installations, the following main principles should be followed when dimensioning the fuel system:

- A separate fuel feed circuit is recommended for each propeller shaft (two-engine installations); in four-engine installations so that one engine from each shaft is fed from the same circuit.
- Main and auxiliary engines are recommended to be connected to separate fuel feed circuits.

6.1.4 Pulse damping

The fuel pipes must be amply sized and firmly clamped in order to withstand pressure pulsation normally occurring in fuel systems of diesel engines.

6.2 Internal fuel oil system

The standard system comprises the following built-on equipment:

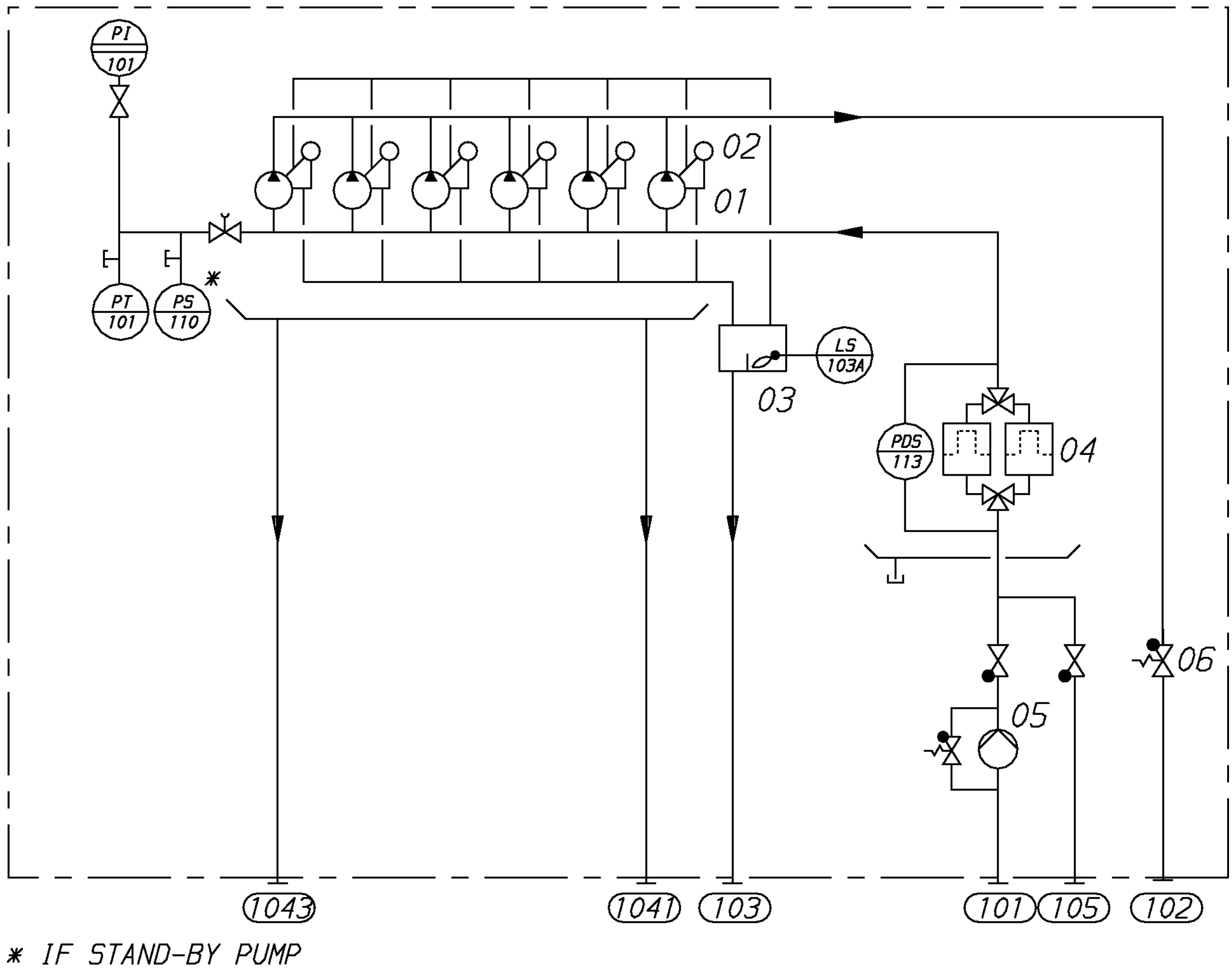
- Fuel injection pumps
- Injection valves
- Inlet and outlet fuel damper with pressure control valve on the outlet damper

Leak fuel from the injection valves and the injection pumps is drained to atmospheric pressure (Clean leak fuel system). The clean leak fuel can be reconducted to the system without treatment. The quantity of leak fuel is given in chapter *Technical data*. Possible uncontrolled leak fuel and spilled water and oil is separately drained from the hot-box and shall be led to a sludge tank (“Dirty” leak fuel system).

Table 6.1 Dimensions of fuel pipe connections on the engine

Pipe connections		Size	Pressure class	Standard
101	Fuel inlet, HFO	OD18	PN160	DIN 2353
101	Fuel inlet, MDF	OD28	PN100	DIN 2353
102	Fuel outlet, HFO	OD18	PN160	DIN 2353
102	Fuel outlet, MDF	OD22	PN100	DIN 2353
103	Leak fuel drain, clean fuel	OD18	-	ISO 3304
1041	Leak fuel drain, dirty fuel	OD22	-	ISO 3304
1043	Leak fuel drain, dirty fuel	OD18	-	ISO 3304
105	Fuel stand-by connection	OD22	PN160	DIN 2353

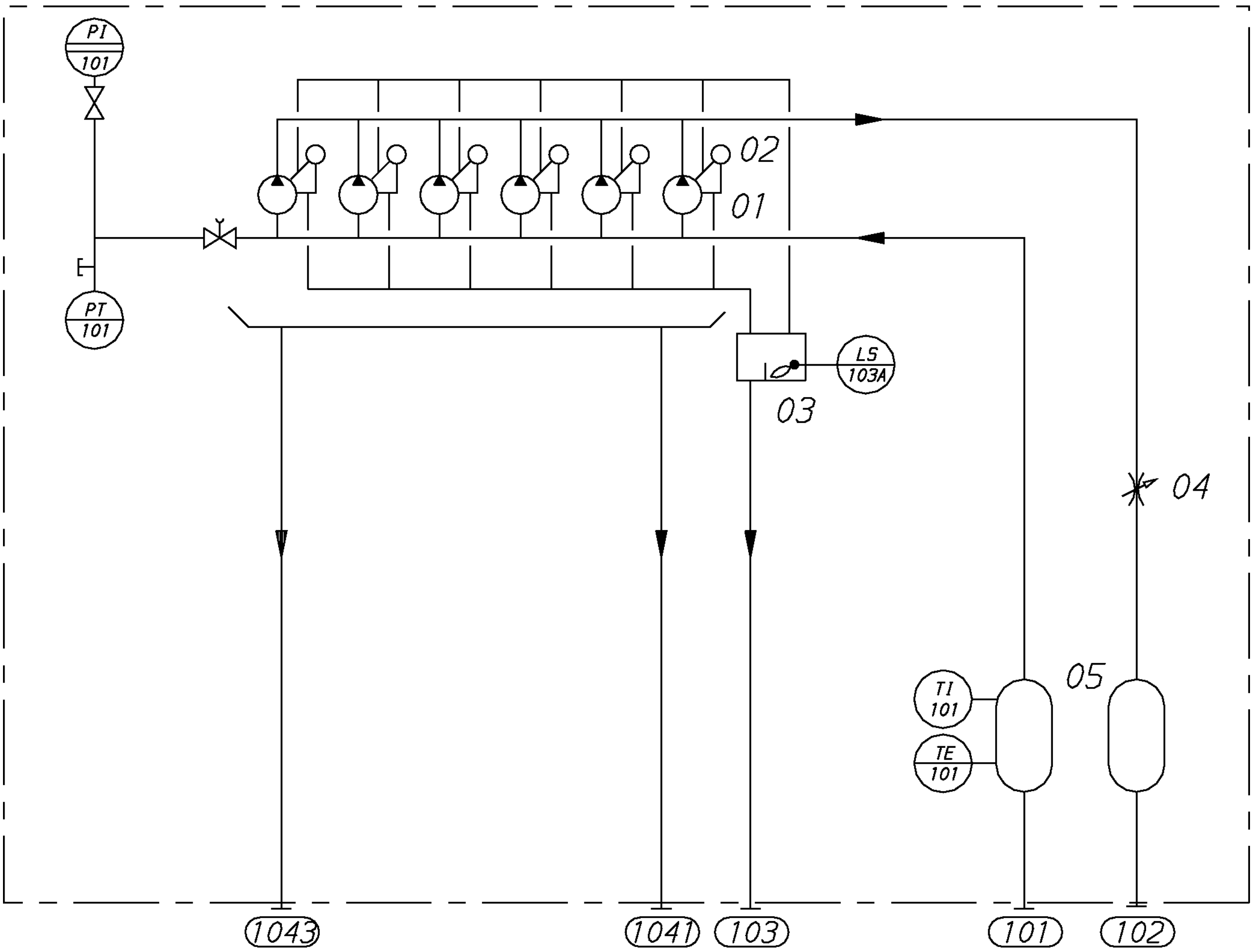
Figure 6.1 Internal fuel system, MDF (4V76F6655a)



System components

01	Injection pump
02	Injection valve
03	Level alarm for leak fuel oil from injection pipes
04	Duplex fine filter
05	Engine driven fuel feed pump
06	Pressure regulating valve

Figure 6.2 Internal fuel system, HFO (4V76F6654a)



System components		Pipe connections	
01	Injection pump	101	Fuel inlet
02	Injection valve	102	Fuel outlet
03	Level alarm for leak oil from injection pipes	103	Leak fuel drain, clean fuel
04	Adjustable orifice	1041	Leak fuel drain, dirty fuel free end
05	Pulse damper	1043	Leak fuel drain, dirty fuel FW-end

6.2.1 Engine driven fuel feed pump

If the engine is equipped with an engine driven gear type fuel feed pump, the day tank shall be arranged so that the minimum level always remains 0.5 m above the crankshaft centerline. This arrangement enables deaeration of the circuit and minimizes the risk of sucking air into the system, if there is a leakage e.g. in a pipe joint. Special measures for black-out start are not required.

6.2.2 Pressure control valve adjustment

For auxiliary engines installed in parallel, the pressure drop differences around engines shall be compensated with the control valve on the outlet pulse damper. It is recommended to install pressure gauges at suitable places to be able to verify the pressure drop. If pressures can not be measured onboard, the temperatures can be used for a rough estimation.

The adjustment on the control valves should be carried out after the pressure regulating valve in the fuel system has been adjusted to approximately 6-7 bar with all control valves fully open. The adjustment must be tested in different loading situations including the cases with one or more of the engines being in stand by mode. If the main engine is connected to the same fuel booster unit the circulation/temperatures must also be checked with and without the main engine being in operation.

Matters other than piping geometry that can influence on the circulation/temperatures are:

- Overflow valve adjustment
- Fuel pipe insulation
- Trace heating efficiency
- Booster pump and/or heater sizing
- Set point on the feeder and/or booster pump safety valve

6.3 MDF installations

6.3.1 General

When running on MDF the fuel oil inlet temperature should be kept at +45°C ±5°C.When running long periods with low load this requires an external MDF cooler (1E04) to be installed.

6.3.2 External fuel system

General

The design of the external fuel system may vary from ship to ship but every system should provide well cleaned fuel with the correct temperature and pressure to each engine. See *figure* "*6.5* " for recommended fuel transfer and separating system.

Filling, transfer and storage

The ship must have means to transfer the fuel from bunker tanks to settling tanks and between the bunker tanks in order to balance the ship.

The amount of fuel in the bunker tanks depends on the total fuel consumption of all consumers onboard, maximum time between bunkering and the decided margin.

Separation

Even if the fuel to be used is marine diesel fuel or gas oil only, it is recommended to install a separator as there should be some means of separating water from the fuel.

Settling tank, MDF (1T10)

In case where MDF is the only fuel onboard the settling tank should normally be dimensioned to ensure fuel supply for min. 24 operating hours when filled to maximum. The tank should be designed to provide the most efficient sludge and condensed water rejecting effect. The bottom of the tank should have slope to ensure good drainage. The MDF settling tank does not need heating coils or insulation.

The temperature in the MDF settling tank should be between 20 - 40°C.

Separator unit, MDF (1N05)

Suction strainer for separator feed pump (1F02)

A suction strainer shall be fitted to protect the feed pump.

Design data:	
fineness	0.5 mm

Feed pump, separator (1P02)

The use of a screw pump is recommended. The pump should be separate from the separator and electrically driven.

Design data:

The pump should be dimensioned for the actual fuel quality and recommended throughput through the separator. The flow rate through the separator should not exceed the maximum fuel consumption by more than 10%. No control valve should be used to reduce the flow of the pump.

Operating pressure, max.	0.5 MPa (5 bar)
Operating temperature	40°C
Viscosity for dimensioning the electric motor	100 cSt

Preheater, separator (1E01)

Fuels having a viscosity higher than 5 mm²/s (cSt) at 50°C need preheating before the separator. For MDF the preheating temperature should be according to the guidelines of the separator supplier.

MDF separator (1S02)

The fuel oil separator should be sized according to the recommendations of the separator supplier.

Sludge tank, separator (1T05)

The sludge tank should be placed below the separators and as close as possible. The sludge pipe should be continuously falling without any horizontal parts.

Fuel feed system

General

For marine diesel fuel (MDF) and fuels having a viscosity of less than 11.5 mm²/s(cSt)/50°C and if the tanks can be located high enough to prevent cavitation in the fuel feed pump, a system with an open de-aeration tank may be installed.

Day tank, MDF (1T06)

The diesel fuel day tank is dimensioned to ensure fuel supply for 12...14 operating hours when filled to maximum*.

*Note that according to SOLAS 1974 Chapter II-1 Part C Regulation 26.11 (as amended in 1981 and 1996), ships are to be fitted with two separate service tanks for fuel to propulsion and vital systems such as main engines (ME), auxiliary engines (AE) and auxiliary boilers (AB). Settling tanks must not be considered en lieu of service tanks.

Acceptable arrangements acc. to SOLAS are:

For MDO operation:		**For HFO operation:**	
TANK	CAPACITY FOR	TANK	CAPACITY FOR
MDO 1 service	ME+AE+AB 8 hours	HFO 1 service	ME+AE+AB 8 hours
MDO 2 service	ME+AE+AB 8 hours	HFO 2 service	ME+AE+AB 8 hours
		MDO service	cold start and repairs
		or	
		HFO service	ME+AE+AB 8 hours
		MDO service	ME+AE+AB 8 hours

For ME and AB operating on HFO and AE operating on MDO:	
TANK	CAPACITY FOR
HFO 1 service	ME+AB 8 hours
HFO 2 service	ME+AB 8 hours
MDO 1 service	AE 8 hours
MDO 2 service	AE 8 hours
or	
HFO service	ME+AB 8 hours
MDO 1 service	The greater of ME+AE+AB 4 hours or ME+AB 8 hours
MDO 2 service	The greater of ME+AE+AB 4 hours or ME+AB 8 hours

Suction strainer, MDF (1F07)

A suction strainer with a fineness of 0.5 mm should be installed for protecting the feed pumps.

Circulation pump, MDF (1P03)

The circulation pump maintains the pressure before the engine. It is recommended to use a screw pump as circulation pump.

Design data:

- Capacity constant (see below) times the total consumption of the engines and the flush quantity of a possible automatic filter
- Capacity constant 6
- The pumps should be placed so that a positive static pressure of about 30 kPa is obtained on the suction side of the pumps.

Stand-by pump, MDF (1P08)

The stand-by pump is required in case of a single main engine equipped with an engine driven pump. It is recommended to use a screw pump as stand-by pump.

Design data:

- Capacity constant (see below) times the total consumption of the engines and the flush quantity of a possible automatic filter
- Capacity constant 6
- The pump should be placed so that a positive static pressure of about 30 kPa is obtained on the suction side of the pump.

Pressure control (overflow) valve, MDF (1V02)

The pressure control valve maintains the pressure in the feed line directing the surplus flow to the suction side of the feed pump.

set point	0.4 MPa (4 bar)

Flow meter, MDF (1I03)

If a totalizer fuel consumption meter is required, it should be fitted in the day tank feed line. In case of continuous engine fuel consumption indication is required, it can be arranged either using two meters per engine or by installing only one meter between the day tank and a fuel return tank.

Cooler/Heater (1E04)

Since the viscosity before the engine must stay between the allowed limits stated in the Chapter for General data and outputs, a heater might be necessary in case the day tank temperature is low. A cooler is needed where long periods of low load operation are expected since fuel gets heated in the engine during the circulation. The cooler is located in the return line after the engine(s). LT-water is normally used as cooling medium.

Leak fuel tank, clean fuel (1T04)

Clean leak fuel drained from the injection pumps can be reused without repeated treatment. The fuel should be collected in a separate clean leak fuel tank and, from there, be pumped to the settling tank. The pipes from the engine to the drain tank should be arranged continuously sloping.

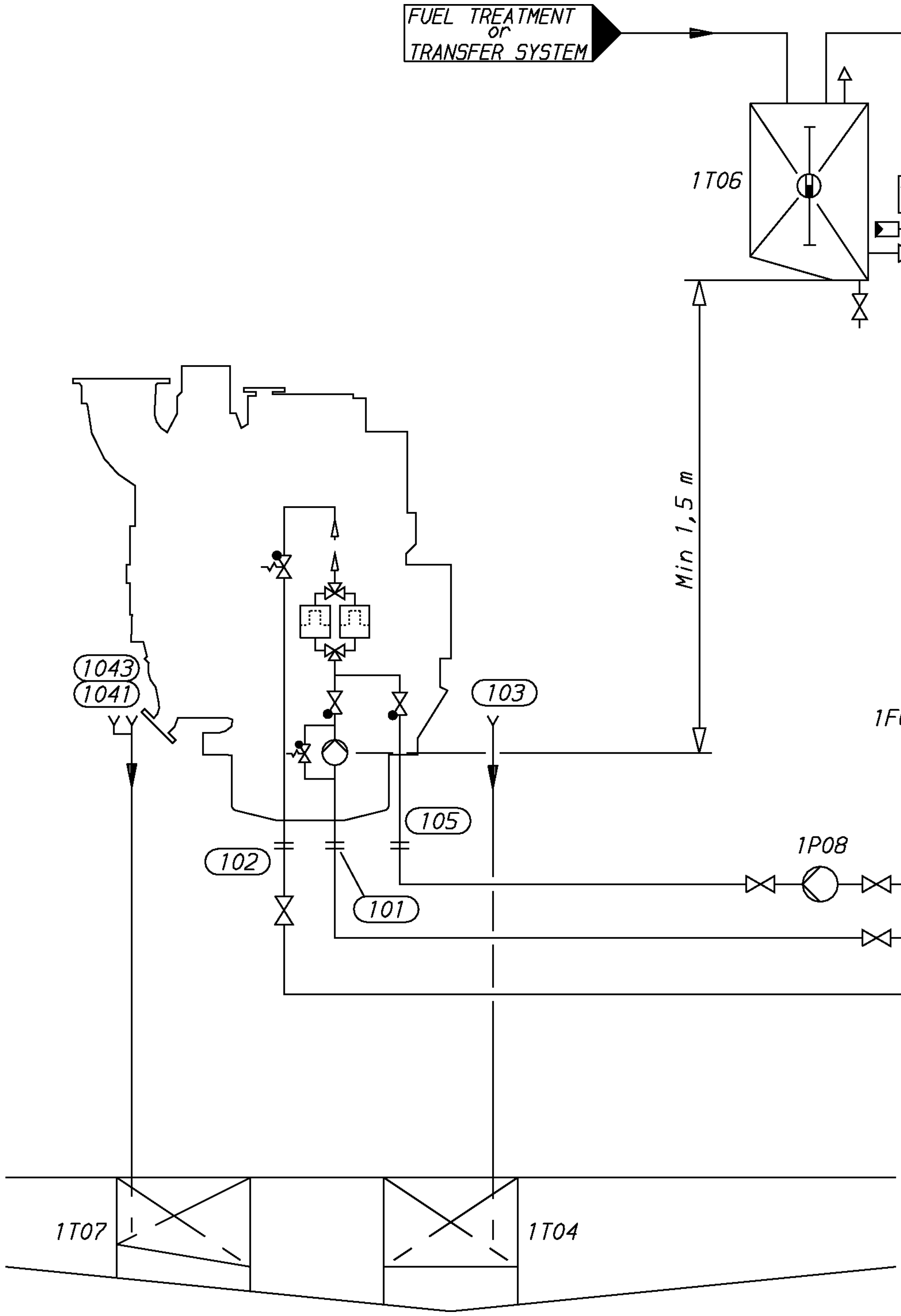
In order to prevent dirt entering the system the leak line(s) should be connected via a closed system.

Leak fuel tank, dirty fuel (1T07)

Under normal operation no fuel should leak out of the dirty system. Fuel, water and oil are drained only in the event of unattended leaks or during maintenance. Dirty leak fuel pipes shall be led to a sludge tank.

6.4 Example system diagrams (MDF)

Figure 6.3 Fuel feed system, main engine (3V76F6657a)



System components	
1F07	Suction strainer, MDF
1P08	Stand-by pump, MDF
1T04	Leak fuel tank, clean fuel
1T06	Day tank, MDF
1T07	Leak fuel tank, dirty fuel

Pipe connections	
101	Fuel inlet
102	Fuel outlet
103	Leak fuel drain, clean fuel
1041	Leak fuel drain, dirty fuel free end
1043	Leak fuel drain, dirty fuel FW-end
105	Fuel stand-by connection

6.5 HFO installations

6.5.1 General

For pumping, the temperature of fuel storage tanks must always be maintained 5 - 10°C above the pour point - typically at 40 - 50°C. The heating coils can be designed for a temperature of 60°C.

The design of the external fuel system may vary from ship to ship, but every system should provide well cleaned fuel with the correct temperature and pressure to each engine. When using heavy fuel it is most important that the fuel is properly cleaned from solid particles and water. In addition to the harm poorly centrifuged fuel will do to the engine, a high content of water may cause damage to the heavy fuel feed system. For the feed system, well-proven components should be used.

The fuel treatment system should comprise at least one settling tank and two (or several) separators to supply the engine(s) with sufficiently clean fuel. Dimensioning of the HFO separators is of greatest importance and therefore the recommendations of the separator designer should be closely followed.

The vent pipes of all tanks containing heavy fuel oil must be continuously upward sloping.

Remarks:

- When dimensioning the pipes of the fuel oil system common known rules for recommended fluid velocities must be followed.
- The fuel oil pipe connections on the engine can be smaller than the pipe diameter on the installation side.

Fuel heating

In ships intended for operation on heavy fuel, steam or thermal oil heating coils must be installed in the bunker tanks.

All heat consumers should be considered:

- Bunker tanks
- Day and settling tanks
- Trace heating
- Fuel separators
- Fuel booster modules

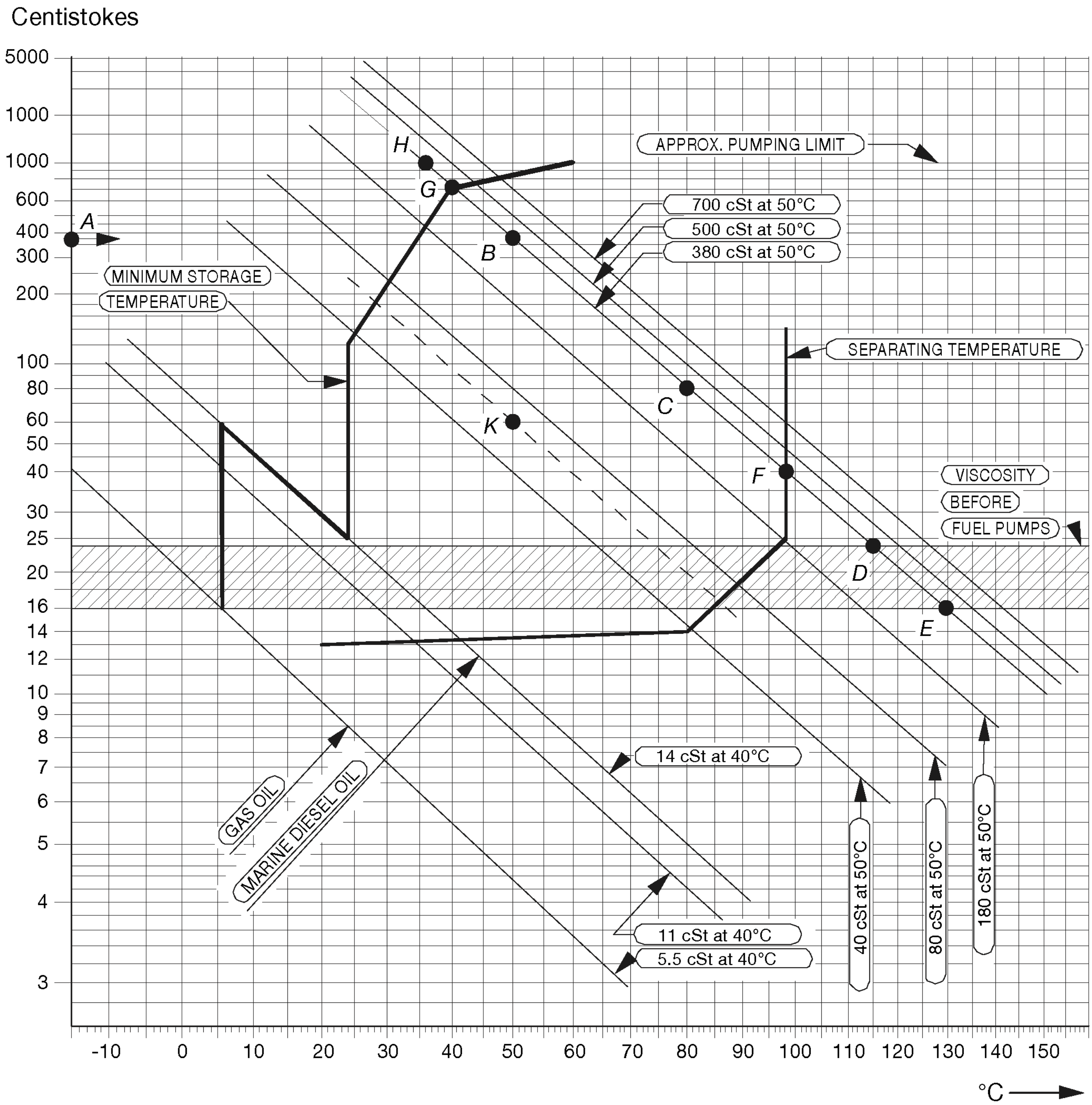
The heating requirement of tanks is calculated from the maximum heat losses from the tank and from the requirement of raising the temperature by typically 1°C/h. The heat loss can be assumed to be 15 W/m²°C between tanks and shell plating against the sea and 3 W/m²°C between tanks and cofferdams. The heat capacity of fuel oil can be taken as 2 kJ/kg°C.

The day and settling tank temperatures are usually in the range 50 - 80°C. A typical heating capacity is 12 kW each.

Trace heating of insulated fuel pipes requires about 1.5 W/m²°C. The area to be used is the total external area of the fuel pipe.

Fuel separators require typically 7 kW/installed engine MW and booster units 30 kW/installed engine MW. See also formulas presented later in this chapter.

Figure 6.4 Fuel oil viscosity-temperature diagram for determining the preheating temperatures of fuel oils (4V92G0071a)



Example 1: A fuel oil with a viscosity of 380 mm²/s (cSt) (A) at 50°C (B) or 80 mm²/s (cSt) at 80°C (C) must be preheated to 115 - 130°C (D-E) before the fuel injection pumps, to 98°C (F) at the separator and to minimum 40°C (G) in the storage tanks. The fuel oil may not be pumpable below 36°C (H).

To obtain temperatures for intermediate viscosities, draw a line from the known viscosity/temperature point in parallel to the nearest viscosity/temperature line in the diagram.

Example 2: Known viscosity 60 mm²/s (cSt) at 50°C (K). The following can be read along the dotted line: viscosity at 80°C = 20 mm²/s (cSt), temperature at fuel injection pumps 74 - 87°C, separating temperature 86°C, minimum storage tank temperature 28°C.

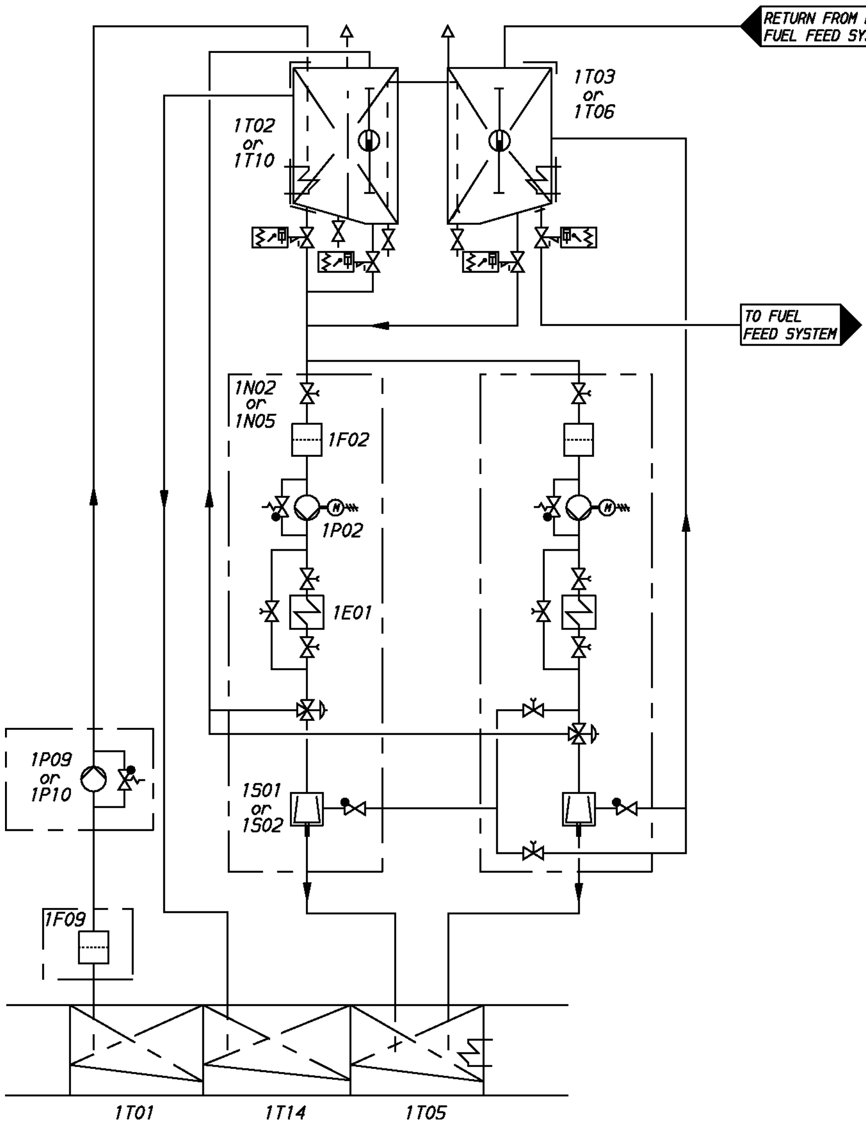
6.5.2 External fuel system

General

The engine is designed for continuous heavy fuel operation. It is, however, possible to operate the engine on marine diesel fuel without making any alterations.

The engine can be started and stopped on heavy fuel provided that the engine and the fuel system are preheated to operating temperature. Switch-over from HFO to Leak fuel drain, MDF for start and stop is not recommended.

Figure 6.5 Fuel transfer and separating system (3V76F6626b)



System components

1N02	Separator unit (HFO)	1F09	Suction strainer
1F02	Suction strainer	1P09	Transfer pump (HFO)
1P02	Feed pump	1P10	Transfer pump (MDF)
1E01	Heater	1T01	Bunker tank
1S01	Separator (HFO)	1T02	Settling tank (HFO)
1N05	Separator unit (MDF)	1T03	Day tank (HFO)
1F02	Suction strainer	1T05	Sludge tank
1P02	Feed pump	1T06	Day tank (MDF)
1E01	Heater	1T10	Settling tank (MDF)
1S02	Separator (MDF)	1T14	Overflow tank

Note that settling and day tanks have been drawn separate in order to show overflow pipe. They normally have common intermediate wall and insulation.

Filling, transfer and storage

The ship must have means to transfer the fuel from bunker tanks to settling tanks and between the bunker tanks in order to balance the ship.

The amount of fuel in the bunker tanks depends on the total fuel consumption of all consumers onboard, maximum time between bunkering and the decided margin.

Separation

Heavy fuel (residual, and mixtures of residuals and distillates) must be cleaned in an efficient centrifugal separator before entering the day tank.

Separator mode of operation

At least two separators, both of the same size, should be installed. The capacity of one separator to be sufficient for the total fuel consumption. The other (stand-by) separator should also be in operation all the time.

It is recommended that conventional separators with gravity disc are arranged for operation in series, the first as a purifier and the second as a clarifier. This arrangment can be used for fuels with a density up to max. abt. 991 kg/m³ at 15°C.

Alternatively, the main and stand-by separators may be run in paralell, but this makes heavier demands on the choice of correct gravity disc and on constant flow and temperature control to achieve optimum results.

Separators with controlled discharge of sludge (without gravity disc) operating on a continuous basis can handle fuels with densities exceeding 991 kg/m³ at 15°C. In this case the main and stand-by separators should be run in parallel.

Settling tank, HFO (1T02)

The settling tank should normally be dimensioned to ensure fuel supply for min. 24 operating hours when filled to maximum. The tank should be designed to provide the most efficient sludge and water rejecting effect. The bottom of the tank should have slope to ensure good drainage. The tank is to be provided with a heating coil and should be well insulated.

To ensure constant fuel temperature at the separator, the settling tank temperature should be kept stable. The temperature in the settling tank should be between 50...70°C.

The minimum level in the settling tank should be kept as high as possible. In this way the temperature will not decrease too much when filling up with cold bunker.

Separator unit (1N02)

Suction strainer for separator feed pump (1F02)

A suction strainer shall be fitted to protect the feed pump. The strainer should be equipped with a heating jacket in case the installation place is cold.

Design data:	
Fineness	0.5 mm

Feed pump, separator (1P02)

The pump should be dimensioned for the actual fuel quality and recommended throughput through the separator. No control valve should be used to reduce the flow of the pump.

Design data:	
operating pressure (max.)	0.2 MPa (2 bar)
operating temperature	100°C
viscosity for dimensioning electric motor	1000 mm²/s (cSt)

Preheater, separator (1E01)

The preheater is normally dimensioned according to the feed pump capacity and a given settling tank temperature. The heater surface temperature must not be too high in order to avoid cracking of the fuel.

The heater should be controlled to maintain the fuel temperature within ± 2°C. The recommended preheating temperature for heavy fuel is 98°C.

Design data:

The required minimum capacity of the heater is:

$$P = \frac{m \times \Delta t}{1700}$$

where:

P = heater capacity (kW)
m = capacity of the separator feed pump [l/h]
Δt = temperature rise in heater [°C]

For heavy fuels Δt = 48°C can be used, i.e. a settling tank temperature of 50°C.

The heaters to be provided with safety valves with escape pipes to a leakage tank (so that the possible leakage can be seen).

HFO separator (1S01)

The fuel oil separator should be sized according to the recommendations of the separator supplier.

Based on a separation time of 23 or 23.5 h/day, the nominal capacity of the separator can be estimated acc. to the following formula:

$$Q = \frac{P \times b \times 24}{\rho \times t}$$

where:

P = max. continuous rating of the diesel engine [kW]
b = specific fuel consumption + 15% safety margin [g/kWh]
ρ = density of the fuel [kg/m^3]
t = daily separating time for selfcleaning separator [h] (usually = 23 h or 23.5 h)

The flow rates recommended for the separator and the grade of fuel in use must not be exceeded. The lower the flow rate the better the separation efficiency.

Sample valves must be placed before and after the separator.

Sludge tank, separator (1T05)

The sludge tank should be placed below the separators and as close as possible. The sludge pipe should be continuously falling without any horizontal parts.

Fuel feed system

General

The fuel feed system for HFO shall be of the pressurized type in order to prevent foaming in the return lines and cavitation in the circulation pumps.

The heavy fuel pipes shall be properly insulated and equipped with trace heating, if the viscosity of the fuel is 180 mm²/s (cSt)/50°C or higher. It shall be possible to shut-off the heating of the pipes when running MDF (the tracing pipes to be grounded together according to their use).

Any provision to change the type of fuel during operation should be designed to obtain a smooth change in fuel temperature and viscosity, e.g. via a mixing tank. When changing from HFO to MDF, the viscosity at the engine should be above 2.8 mm²/s (cSt) and not drop below 2.0 mm²/s (cSt) even during short transient conditions. In certain applications a cooler may be necessary.

Day tank, HFO (1T03)

The heavy fuel day tank is usually dimensioned to ensure fuel supply for about 24 operating hours when filled to maximum (see note for MDF day tanks). The design of the tank should be such that water and dirt particles do not accumulate in the suction pipe. The tank has to be provided with a heating coil and should be well insulated.

Maximum recommended viscosity in the day tank is 140 mm²/s (cSt). Due to the risk of wax formation, fuels with a viscosity lower than 50 mm²/s (cSt)/50°C must be kept at higher temperatures than what the viscosity would require.

Fuel viscosity (mm²/s (cSt) at 100°C)	Minimum day tank temperature (°C)
55	80
35	70
25	60

Feeder/booster unit (1N01)

A completely assembled fuel feed unit can be supplied as an option.

This unit normally comprises the following equipment:

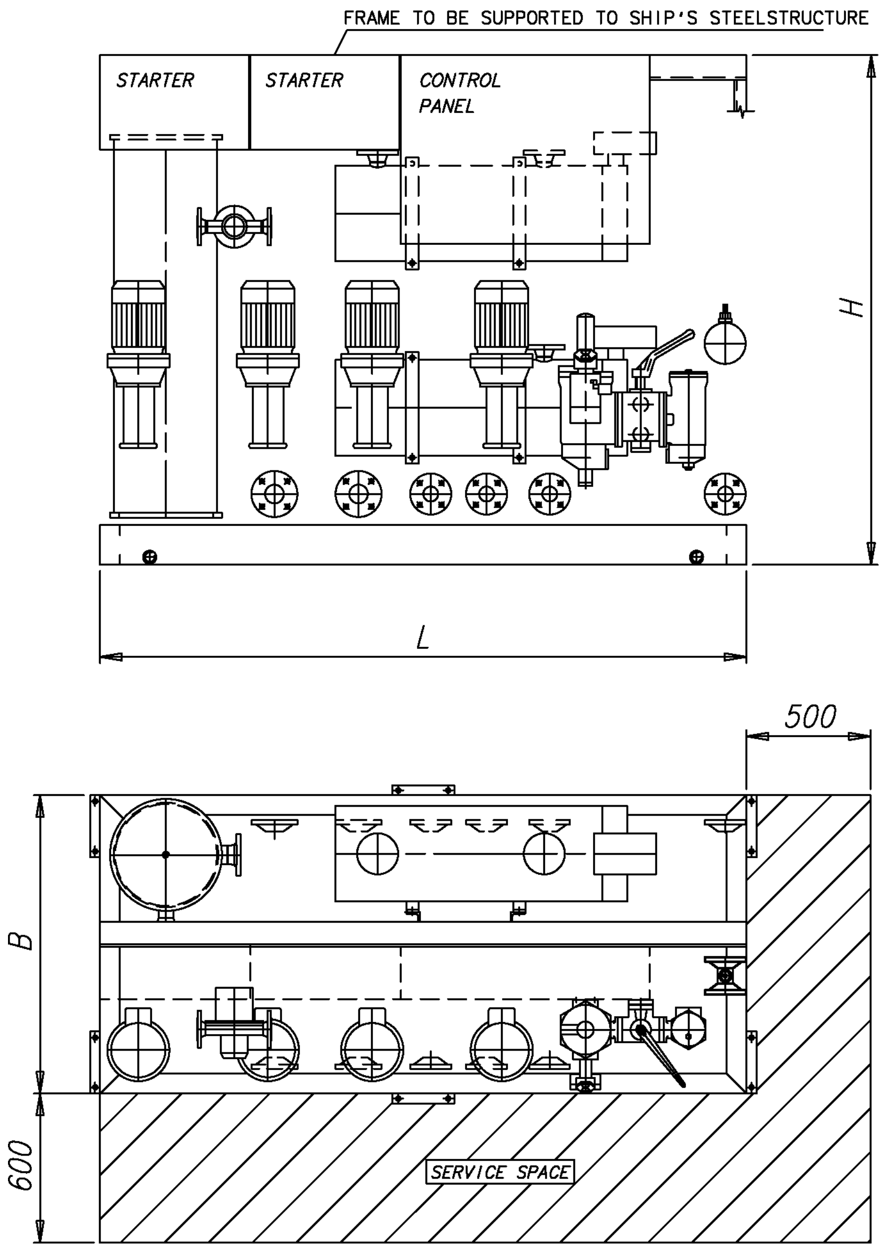
- Two suction strainers
- Two fuel feed pumps of screw type, equipped with built-on safety valves and electric motors
- One pressure control/overflow valve
- One pressurized de-aeration tank, equipped with a level switch operated vent valve
- Two booster pumps, same type as above
- Two heaters, steam, electric or thermal oil (one in operation, the other as spare)
- One automatic back-flushing filter with by-pass filter
- One viscosimeter for the control of the heaters
- One steam or thermal oil control valve or control cabinet for electric heaters
- One thermostat for emergency control of the heaters
- One control cabinet with starters for pumps, automatic filter and viscosimeter
- One alarm panel

The above equipment is built on a steel frame, which can be welded or bolted to its foundation in the ship. All heavy fuel pipes are insulated and provided with trace heating.

When installing the unit, only power supply, group alarms and fuel, steam and air pipes have to be connected.

It is recommended to supply not more than two engines from the same system. Alternatively, an individual circulation pump (and a stand-by pump if required) is recommended to obtain the correct and sufficient flow to the engine, ensuring that nothing is "lost" in pressure control valves, safety valves, overflow valves, etc.

Figure 6.6 Feeder/booster unit, example (DAAE006659)



	H	L	B
STEAM	1800	2500	1100
ELECTRICAL	2050	2600	1200

Suction strainer HFO (1F06)

A suction strainer with a fineness of 0.5 mm should be installed for protecting the feed pumps. The strainer should be equipped with a heating jacket.

Feeder pump, HFO (1P04)

The feeder pump maintains the pressure in the fuel feed system. It is recommended to use a high temperature resistant screw pump as feeder pump.

Design data:

Capacity to cover the total consumption of the engines and the flush quantity of a possible automatic filter.

The pumps should be placed so that a positive static pressure of about 30 kPa is obtained on the suction side of the pumps.

delivery pressure	0.6 MPa (6 bar
operating temperature	100°C
viscosity for dimensioning electric motor	1000 mm²/s (cSt)

Pressure control valve HFO (1V03)

The pressure control valve maintains the pressure in the de-aeration tank directing the surplus flow to the suction side of the feed pump.

Design data:	
set point	0.3…0.5 MPa (3...5 bar)

Automatically cleaned fine filter, HFO (1F08)

The use of automatic back-flushing filters is recommended, installed between the feeder pumps and the deaeration tank in parallel with an insert filter as the stand-by half.

For back-flushing filters the feed pump capacity should be sufficient to prevent pressure drop during the flushing operation.

Design data:	
Fuel oil	acc. to specification
Operating temperature	0...100°C
Preheating	from 25 mm²/s (cSt)/100°C
Flow	feed pump capacity
Operating pressure	1 MPa (10 bar
Design pressure	1.6 MPa (16 bar)
Test pressure	Fuel side 2 MPa (20 bar)
	Heating jacket 1 MPa (10 bar)
Fineness:	
Back-flushing filter	35 µm (absolute mesh size)
Insert filter	35 µm (absolute mesh size)

Maximum recommended pressure drop for normal filters at 14 mm²/s (cSt):

clean filter	20 kPa (0.2 bar)
dirty filter	60 kPa (0.6 bar)
alarm	80 kPa (0.8 bar)

Fuel consumption meter (1I01)

If a fuel consumption meter is required, it should be fitted between the feed pumps and the de-aeration tank. An automatically opening by-pass line around the consumption meter is recommended in case of possible clogging.

If the meter is provided with a prefilter,it is recommendable to install an alarm for high pressure difference across the filter.

De-aeration tank (1T08)

The volume of the tank should be about 50 l. It shall be equipped with a vent valve, controlled by a level switch. It shall also be insulated and equipped with a heating coil. The vent pipe should, if possible, be led downwards, e.g. to the overflow tank.

Booster pump, HFO (1P06)

The purpose of this pump is to circulate the fuel in the system and to maintain the pressure stated in the chapter for Technical data at the injection pumps. It also circulates the fuel in the system to maintain the viscosity, and keeps the piping and injection pumps at operating temperature.

The feed pump capacity should be sufficient to prevent pressure drop during the flushing of the automatic filter if installed on the pressure side of this pump.

Design data:

capacity constant (see below) times the total consumption of the engines and the flushing of the automatic filter

Capacity constant	4
Pressure	1 MPa (10 bar)
Temperature	150°C
Viscosity for dimensioning the electric motor	500 mm²/s (cSt)

Booster unit heater (1E02)

The heater(s) is normally dimensioned to maintain an injection viscosity of 14 mm²/s (cSt) according to the maximum fuel consumption and a given day tank temperature.

To avoid cracking of the fuel the surface temperature in the heater must not be too high. The surface power of electric heaters must not be higher than 1.5 W/cm². The output of the heater shall be controlled by a viscosimeter. As a reserve a thermostat control may be fitted.

The set point of the viscosimeter shall be somewhat lower than the required viscosity at the injection pumps to compensate for heat losses in the pipes.

Design data:

The required minimum capacity of the heater is:

$$P = \frac{m \times \Delta t}{1700}$$

where:

P = heater capacity (kW)

m = evaluated by multiplying the specific fuel consumption of the engines by the total max. output of the engines [l/h]

Δt = temperature rise, higher with increased fuel viscosity [°C]

To compensate for heat losses due to radiation the above power should be increased with 10% + 5 kW.

The following values can be used:

Fuel viscosity (mm²/s (cSt) at 100°C)	Temperature rise in heater (°C)
55	65 (80 in day tank)
35	65 (70 in day tank)
25	60 (60 in day tank)

Viscosimeter (1I02)

For the control of the heater(s) a viscosimeter has to be installed. A thermostatic control shall be fitted, to be used as safety when the viscosimeter is out of order. The viscosimeter should be of a design, which stands the pressure peaks caused by the injection pumps of the diesel engine.

Design data:

operation range	0...50 mm²/s (cSt)
design temperature	180°C
design pressure	4 MPa (40 bar)

Safety filter (1F03)

Since no fuel filters are built on the engine, one duplex type safety filter with an alarm contact for high differential pressure is installed between the booster module and the engine. The filter should be located as close to the engine as possible. A common filter can be used for all engines after each booster module.

min. fineness	50 μm

Overflow valve (1V05)

In multiple engine installations an overflow valve between in- and outlet line is recommended to limit the pressure in fuel line to the engine by relieving the pressure to the return line when the fueloil lines to one engine are closed for maintenance purposes.

Leak fuel tank, clean fuel (1T04)

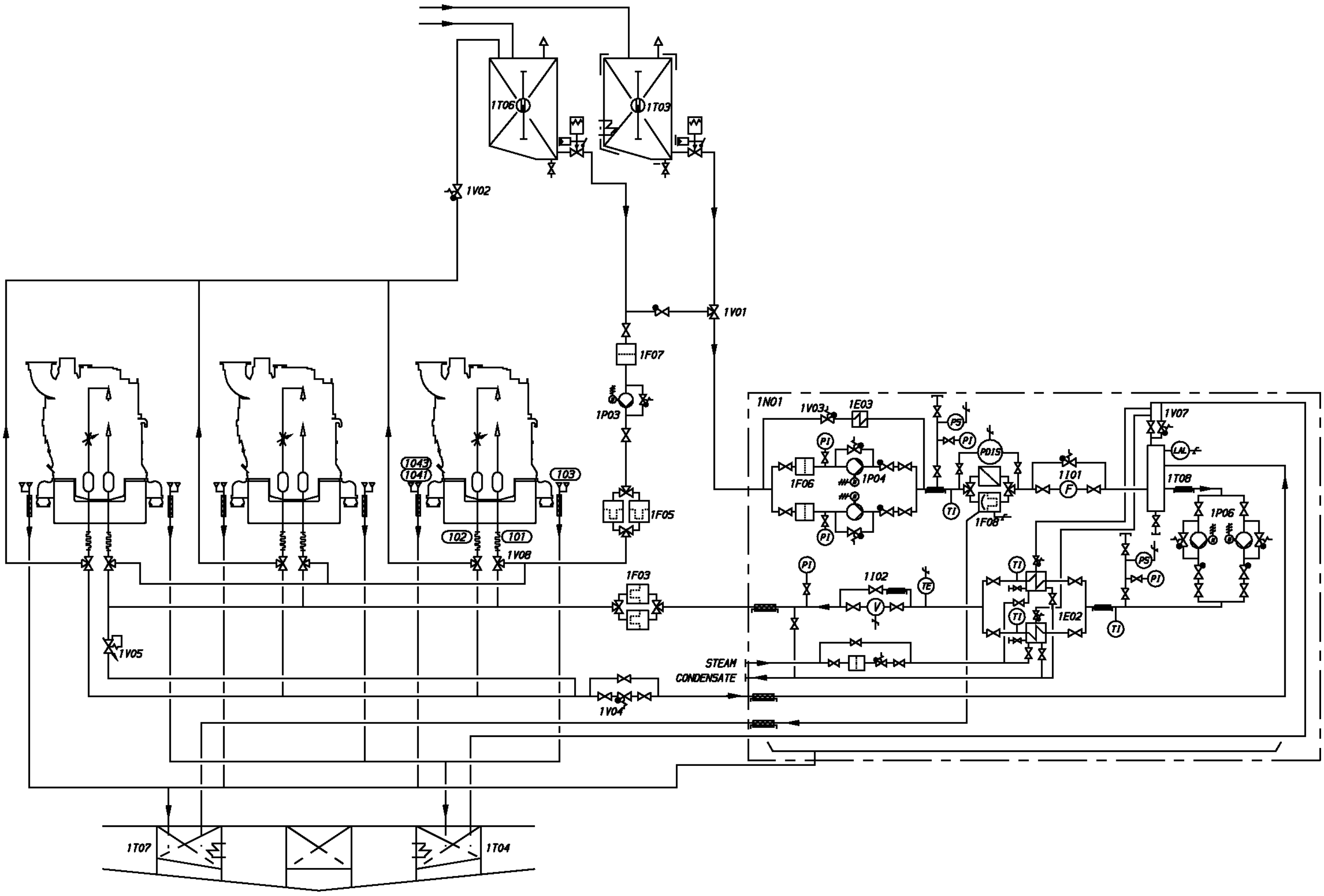
Clean leak fuel drained from the injection pumps can be reused without repeated treatment. The fuel should be collected in a separate clean leak fuel tank and, from there, be pumped to the settling tank. The pipes from the engine to the drain tank should be arranged continuously sloping and should be provided with heating and insulation. To prevent dirt entering the system the leak line(s) should be connected via a closed system.

Leak fuel tank, dirty fuel (1T07)

Under normal operation no fuel should leak out of the dirty system. Fuel, water and oil is drained only in the event of unattended leaks or during maintenance. Dirty leak fuel pipes shall be led to a sludge tank and be trace heated and insulated.

6.6 Example system diagrams (HFO)

Figure 6.7 Fuel feed system, auxiliary engines (3V76F6656b)



System components

1E02	Heater	1T03	Day tank, HFO
1E03	Cooler	1T04	Leak fuel tank, clean fuel
1F03	Safety filter, HFO	1T06	Day tank, MDF
1F05	Safety filter, MDF	1T07	Leak fuel tank, dirty fuel
1F06	Suction filter, HFO	1T08	De-aeration tank
1F07	Suction strainer, MDF	1V01	Change over valve
1F08	Automatic filter	1V02	Pressure control valve, MDF
1I01	Flowmeter	1V03	Pressure control valve, Booster unit
1I02	Viscosity meter	1V04	Pressure control valve, HFO
1N01	Feeder/Booster unit	1V05	Overflow valve, HFO
1P03	Circulation pump, MDF	1V07	Venting valve
1P04	Fuel feed pump, HFO	1V08	Change over valve
1P06	Booster pump		

Pipe connections

101	Fuel inlet	1041	Leak fuel drain, dirty fuel free end
102	Fuel outlet	1043	Leak fuel drain, dirty fuel FW-end
103	Leak fuel drain, clean fuel		

7. Lubricating oil system

7.1 General

Each engine should have a lubricating oil system of its own. The lubricating oil must not be mixed between different systems. Engines operating on heavy fuel should have continuous separation of the lubricating oil.

The standard system comprises the following built on equipment:

- Engine driven lubricating oil pump
- Pre-Lubricating oil pump
- Lubricating oil cooler
- Thermostatic valve
- Automatic filter
- Centrifugal filter
- Pressure control valve

The following equipment can be mounted on the engine as optional:

- Stand by pump connections

The engine sump is normally wet but dry sump is available upon request.

7.2 Lubricating oil quality

7.2.1 Engine lubricating oil

The system oil should be of viscosity class SAE 40 (ISO VG 150).

It is recommended to use in the first place BN 50-55 lubricants when operating on heavy fuel. This recommendation is valid especially for engines having wet lubricating oil sump and using heavy fuel with sulphur content above 2.0% mass. BN 40 lubricants can be used when operating on heavy fuel as well if experience shows that the lubricating oil BN equilibrium remains at an acceptable level.

BN 30 lubricants are recommended to be used only in special cases, such as installations equipped with an SCR catalyst. Lower BN products eventually have a positive influence on cleanliness of the SCR catalyst. With BN 30 oils lubricating oil change intervals may be rather short, but lower total operating costs may be achieved because of better plant availability provided that the m aintenance intervals of the SCR catalyst can be increased.

BN 30 oils are also a recommended alternative when operating on crude oil having low sulphur content. Though crude oils many times have low sulphur content, they can contain other acid compounds and thus an adequate alkali reserve is im portant. With crude oils having higher sulphur content BN 40 – 55 lubricating oils should be used.

If both distillate fuel and residual fuel are used periodically as fuel, lubricating oil quality has to be chosen according to instructions being valid for residual fuel operation, i.e. BN 30 is the minimum . Optimum BN in this kind of operation depends on the length of operating periods on both fuel qualities as well as of sulphur content of fuels in question. Thus in particular cases BN 40 or even higher BN lubricating oils should be used.

The intervals between lubricating oil changes may be extended by adding oil daily to keep the oil level constantly close to the maximum level.

Additives

The oils should contain additives that give good oxidation stability, corrosion protection, load carrying capacity, neutralisation of acid com bustion and oxidation residues and should prevent deposit form ation on internal engine parts (piston cooling gallery, piston ring zone and bearing surfaces in particular).

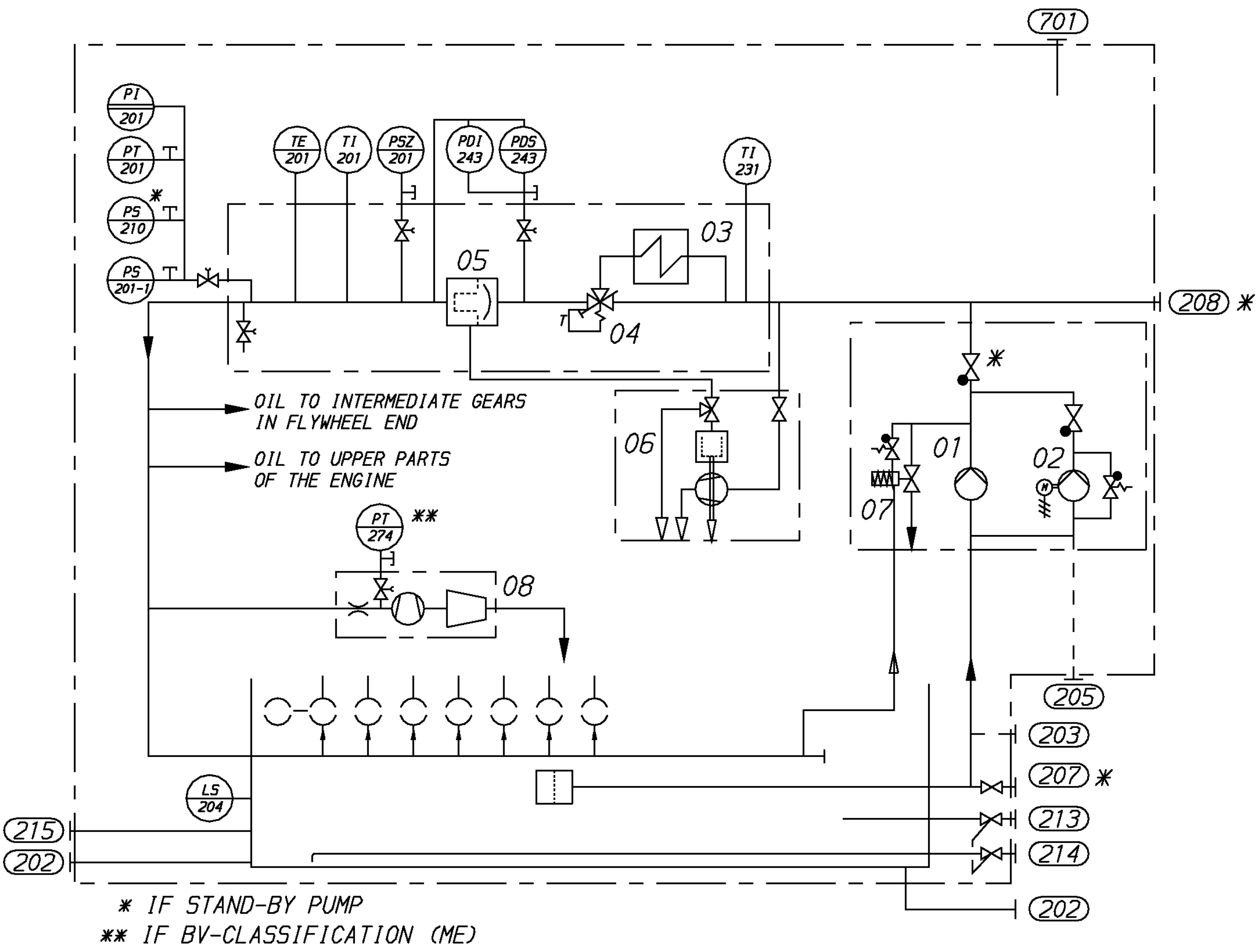
7.3 Internal lubricating oil system

Depending on the type of application the lubricating oil system built on the engine can vary somewhat in design.

Table 7.1 Dimensions of lubricating oil pipe connections on the engine

Pipe connections		Size	Pressure class	Standard
202	Lubrication oil outlet (If dry sump)	DN100	See figure 7.2	
203	Lubrication oil to engine driven pump (If dry sump)	DN100	See figure 7.2	
205	Lubrication oil to priming pump (If dry sump)	DN32	PN40	ISO 7005-1
207	Lubrication oil to electric driven pump	DN100	PN16	ISO 7005-1
208	Lubrication oil from electric driven pump	DN80	PN16	ISO 7005-1
213	Lubrication oil from separator and filling	DN32	PN40	ISO 7005-1
214	Lubrication oil to separator and drain	DN32	PN40	ISO 7005-1
215	Lubrication oil filling	M48*2		ISO 7005-1
701	Crankcase air vent	DN65		

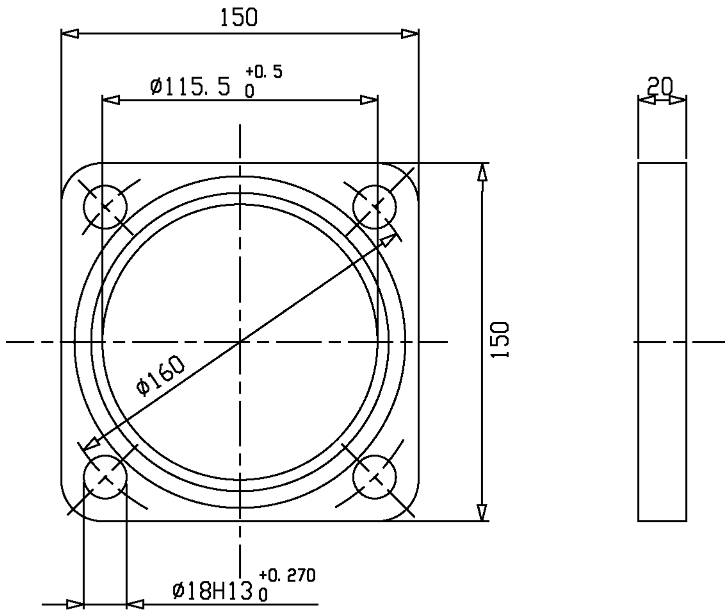
Figure 7.1 Internal lubricating oil system (4V76E4589b)



System components

01	Lubricating oil main pump	05	Automatic filter
02	Prelubricating oil pump	06	Centrifugal filter
03	Lubricating oil cooler	07	Pressure control valve
04	Thermostatic valve	08	Turbocharger

Figure 7.2 Flange for lubricating oil pump (4V32A0506a)



7.3.1 Lubricating oil pump

The direct driven lubricating oil pump is of the gear type. The pump is dimensioned to provide sufficient flow even at low speeds and is equipped with an overflow valve which is controlled from the oil inlet pipe. If necessary, the engine is provided with pipe connections for a separate, electric motor driven stand-by pump.

Concerning flow rates and pressures, see *Technical Data*. The suction height of the pump should not exceed 4 m.

7.3.2 Prelubricating oil pump

The pre-lubricating pump is an electric motor driven constant volume pump equipped with a safety valve.

The pump is of gear wheel type.

The pump is used for:

- Filling of the engine lubricating oil system before starting, e.g. when the engine has been out of operation for a long time
- Continuous pre-lubrication of a stopped engine through which heated heavy fuel is circulating
- Continuous pre-lubrication of a stopped diesel engine(s) in a multi-engine installation always when any one engine is running

Concerning flow and pressures, see chapter *Technical Data*. The suction height of the built-on pre-lubricating pump should not exceed 3.5 m.

7.3.3 Lubricating oil cooler

The lubricating oil cooler is a brazed plate cooler and it is integrated in the lubricating oil module.

7.3.4 Thermostatic valve

The thermostatic valve is integrated in the lubricating oil module.

7.3.5 Lubricating oil automatic filter

The lubricating oil fine filter is a back flushing automatic filter with sludge discharge to the centrifugal filter.

Design data:	
Full flow fine filter	25 µm
Full flow safety filter	100 µm

7.3.6 Centrifugal filter

The centrifugal filter is powered by oil flow and filters the back flush of fine filter.

7.3.7 Lubricating oil module

The lubricating oil module, consisting of filters, thermostatic valve and oil cooler, is supported directly on the engine block.

7.4 External lubricating oil system

When designing the piping diagram, the procedure to flush the system should be clarified and presented in the diagram.

7.4.1 System oil tank (2T01)

The dry engine sump has two drain outlets at the flywheel end and two at the free end. Two of the drains shall be connected. The pipe connection between the sump and the system oil tank should be arranged flexible enough to allow thermal expansion.

Recommendations for the tank design are given in the drawing of the engine room arrangement. The tank must not be placed so that the oil is cooled so much that the recommended lubricating oil temperature cannot be obtained. If there is space enough a cofferdam below the tank is recommended.

Design data:	
Oil volume	1.2...1.5 l/kW
Tank filling	75...80%

7.4.2 Suction strainer (2F06)

A suction strainer complemented with magnetic rods can be fitted in the suction pipe to protect the lubricating oil pump.

The suction strainer as well as the suction pipe diameter should be amply dimensioned to minimize the pressure loss. The suction strainer should always be provided with alarm for high differential pressure.

Design data:	
Fineness	0.5...1.0 mm

7.4.3 Lubricating oil pump, stand-by (2P04)

The stand-by lubricating oil pump is normally of screw type and should be provided with an overflow valve.

Design data:	
Capacity	see *Technical data*
Operating pressure, max	0.8 MPa (8 bar)
Operating temperature, max.	100°C
Lubricating oil viscosity	SAE 40

7.5 Separation system

7.5.1 Separator (2S01)

For HFO operation the Lubricating oil separator should be dimensioned for continuous separating. For MDF intermittent separating might be sufficient. Each lubricating oil system should have a separator unit of its own.

Each main engine operating on heavy fuel shall have a dedicated separator.

Auxiliary engines operating on a fuel having a viscosity of max. 35 mm²/s (cSt) / 100°C may have a common separator unit.

Two engines may have a common separator unit. In installations with four or more engines two separator units should be installed.

The separators should preferably be of a type with controlled discharge of the bowl to minimize the lubricating oil losses.

Design data:

Separating temperature	90...95°C

Capacity according to the following formula:

$$Q = \frac{1.36 \times P \times n}{t}$$

where:

Q =	volume flow [l/h]
P =	total engine output [kW]
n =	number of through-flows of dry sump system oil tank volume n/day: 5 for HFO, 4 for MDF
t =	operating time [h/day]: 24 for continuos separator operation, 23 for normal dimensioning

NOTE! Det Norske Veritas states in their class rules of July 2001 that come into force 1.1.2002 the following:

(Pt.4 Ch.6 Sec.5 C 203) "For diesel engines burning residual oil fuel, cleaning of the Lubricating oil by means of purifiers are to be arranged. These means are additional to filters."

7.5.2 Separator pump (2P03)

The separator pump can be directly driven by the separator or separately driven by an electric motor. The flow should be adapted to achieve the above mentioned optimal flow.

7.5.3 Separator preheater (2E02)

The preheater can be a steam, thermal oil or an electric heater. The surface temperature of the heater must not exceed 150°C in order to avoid decomposition of the oil.

- For separators centrifuging during operation, the heater should be dimensioned for this operating condition. The temperature in the separate system oil tank in the ship's bottom is normally 65 - 75°C.
- For separators centrifuging when the engine is stopped, the heater should be large enough to allow centrifuging at optimal rate of the separator without heat supply from the diesel engine.

NOTE! The heaters are to be provided with safety valves with escape pipes to a leakage tank so that the possible leakage can be seen.

7.5.4 Renovating oil tank

In case of wet sump engines the oil sump content can be drained to this tank prior to separation.

7.5.5 Renovated oil tank

This tank contains renovated oil ready to be used as a replacement of the oil drained for separation.

7.6 Filling, transfer and storage

7.6.1 New oil tank (2T03)

In engines with wet sump, the lubricating oil may be filled into the engine, using a hose or an oil can, through the crankcase cover or through the separator pipe. The system should be arranged so that it is possible to measure the filled oil volume.

7.7 Crankcase ventilation system

A crankcase vent pipe shall be provided for each engine. If the engine has a dry sump and there is a system oil tank, this tank shall have its own vent pipe. Vent pipes of several engines and vent pipes of engine crankcases and tanks should not be joined together.

The connection between the engine and the vent pipe is to be flexible.

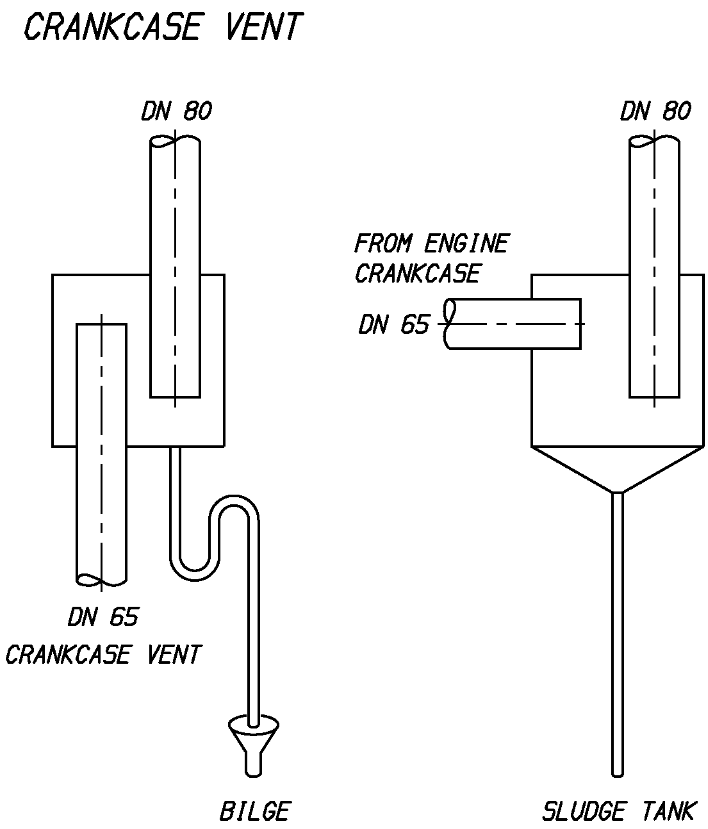
A condensate trap shall be fitted on all vent pipes within 1 - 2 meters of the engine, see figure *7.3*.

Recommended size of the vent pipe after the condensate trap is DN 80.

Pipe connection engine:

701 Crankcase air vent DN65, DIN2576, PN10

Figure 7.3 Crankcase ventilation (4V76E2522)



7.8 Flushing instructions

If the engine is equipped with a wet oil sump and the complete lubricating oil system is built on the engine, flushing is not required. The system oil tank should be carefully cleaned and the oil separated to remove dirt and welding slag.

If the engine is equipped with a dry sump and parts of the lubricating oil system are off the engine, these must be flushed in order to remove any foreign particles before start up.

If an electric motor driven stand-by pump is installed, this should be used for the flushing. In case only an engine driven main pump is installed, the ideal is to use for flushing a temporary pump of equal capacity as the main pump.

The circuit is to be flushed drawing the oil from the sump tank pumping it through the off-engine lubricating oil system and a flushing oil filter with a mesh size of 34 microns or finer and returning the oil through a hose and a crankcase door to the engine sump.

The flushing pump should be protected by a suction strainer. Automatic lubricating oil filters, if installed, must be bypassed during the first hours of flushing.

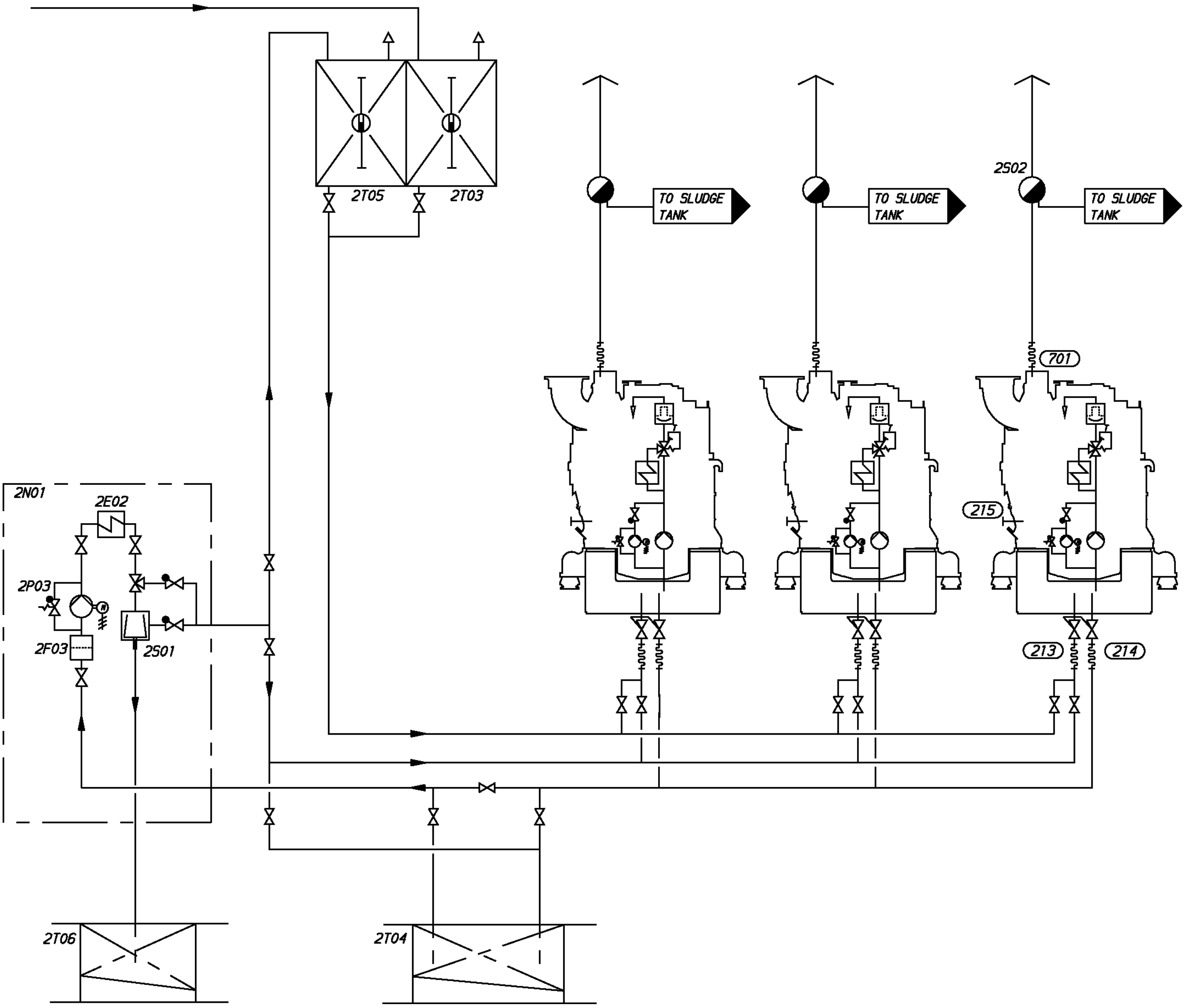
The flushing is more effective if a dedicated heated low viscosity flushing oil is used or if the engine oil is heated. Furthermore, lubricating oil separators should be in operation prior to and during the flushing.

The minimum recommended flushing time is 24 hours. During this time the welds in the lubricating oil piping should be gently knocked at with a hammer to release slag and the flushing filter inspected and cleaned at regular intervals.

Either a separate flushing oil or the approved engine oil can be used for flushing. Even if an approved engine oil is used, it cannot further be used as engine oil.

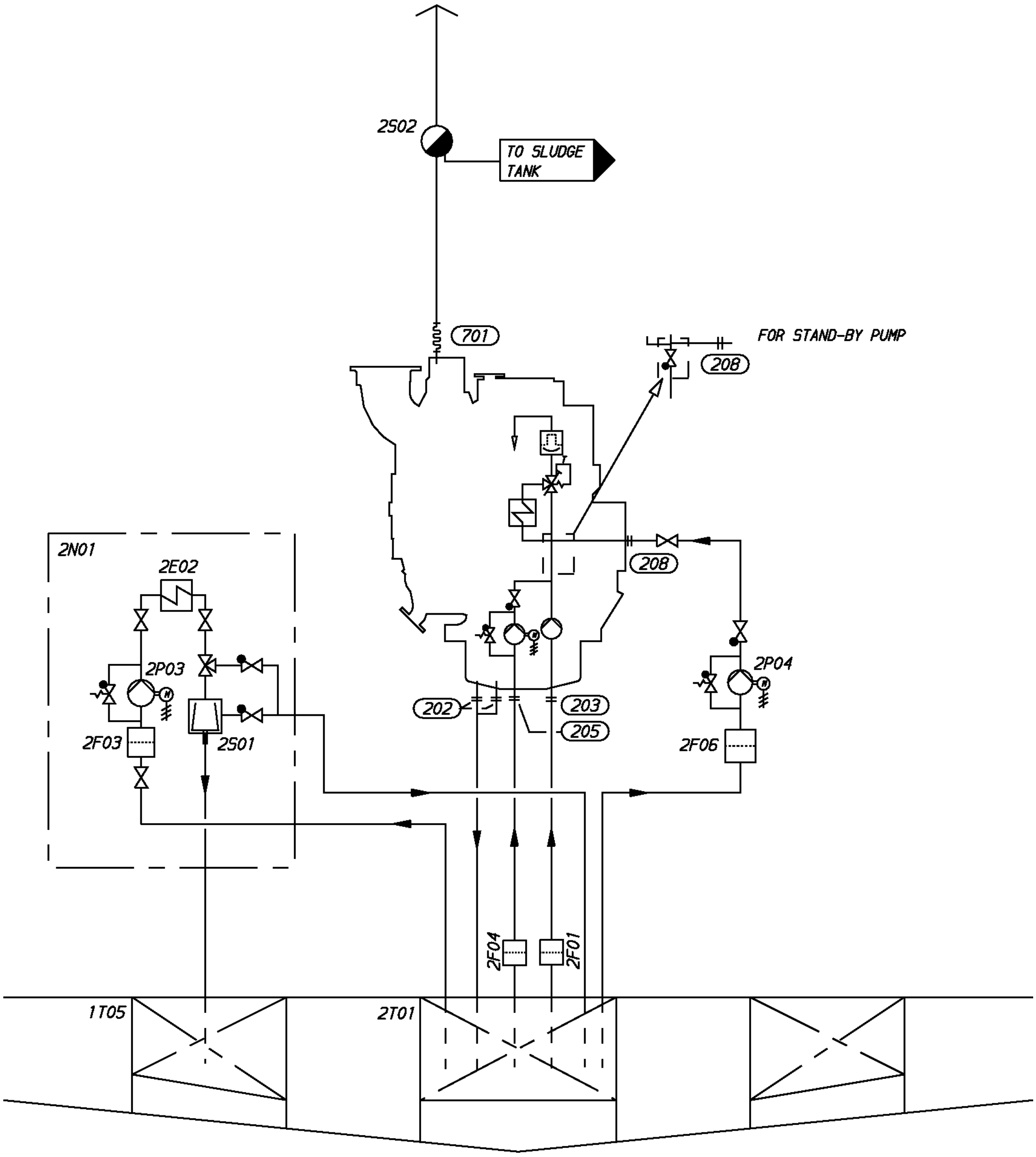
7.9 Example system diagrams

Figure 7.4 Lubricating oil system, auxiliary engines (3V76E4590a)



System components		Pipe connections		
2E02	Heater (Separator unit)	213	Lubricating oil from separator and filling	DN32
2F03	Suction filter (Separator unit)	214	Lubricating oil to separator and drain	DN32
2N01	Separator unit	215	Lubricating oil filling	M48*2
2P03	Separator pump (Separator unit)	701	Crankcase air vent	DN65
2S01	Separator			
2S02	Condensate trap			
2T03	New oil tank			
2T04	Renovating oil tank			
2T05	Renovated oil tank			
2T06	Sludge tank			

Figure 7.5 Lubricating oil system, main engine (3V76E4591a)



System components		Pipe connections		
2E02	Heater (Separator unit)	202	Lubricating oil outlet (from oil sump)	DN100
2F01	Suction strainer (Main lubricating oil pump)	203	Lubricating oil to engine driven pump	DN100
2F03	Suction filter (Separator unit)	205	Lubricating oil to priming pump	DN32
2F04	Suction strainer (Prelubricating oil pump)	208	Lubricating oil from electric driven pump	DN80
2F06	Suction strainer (Stand-by pump)	701	Crankcase air vent	DN65
2N01	Separator unit			
2P03	Separator pump (Separator unit)			
2P04	Stand-by pump			
2S01	Separator			
2S02	Condensate trap			
2T01	System oil tank			
2T06	Sludge tank			

8. Compressed air system

Compressed air is used to start engines and to provide actuating energy for safety and control devices. Compressed air is used onboard also for other purposes with different pressures. The use of starting air supply for these other purposes is limited by the classification regulations.

To ensure the functionality of the components in the compressed air system, the compressed air has to be dry and clean from solid particles and oil.

8.1 Internal starting air system

The engine is equipped with a pneumatic starting motor driving the engine through a gear rim on the flywheel.

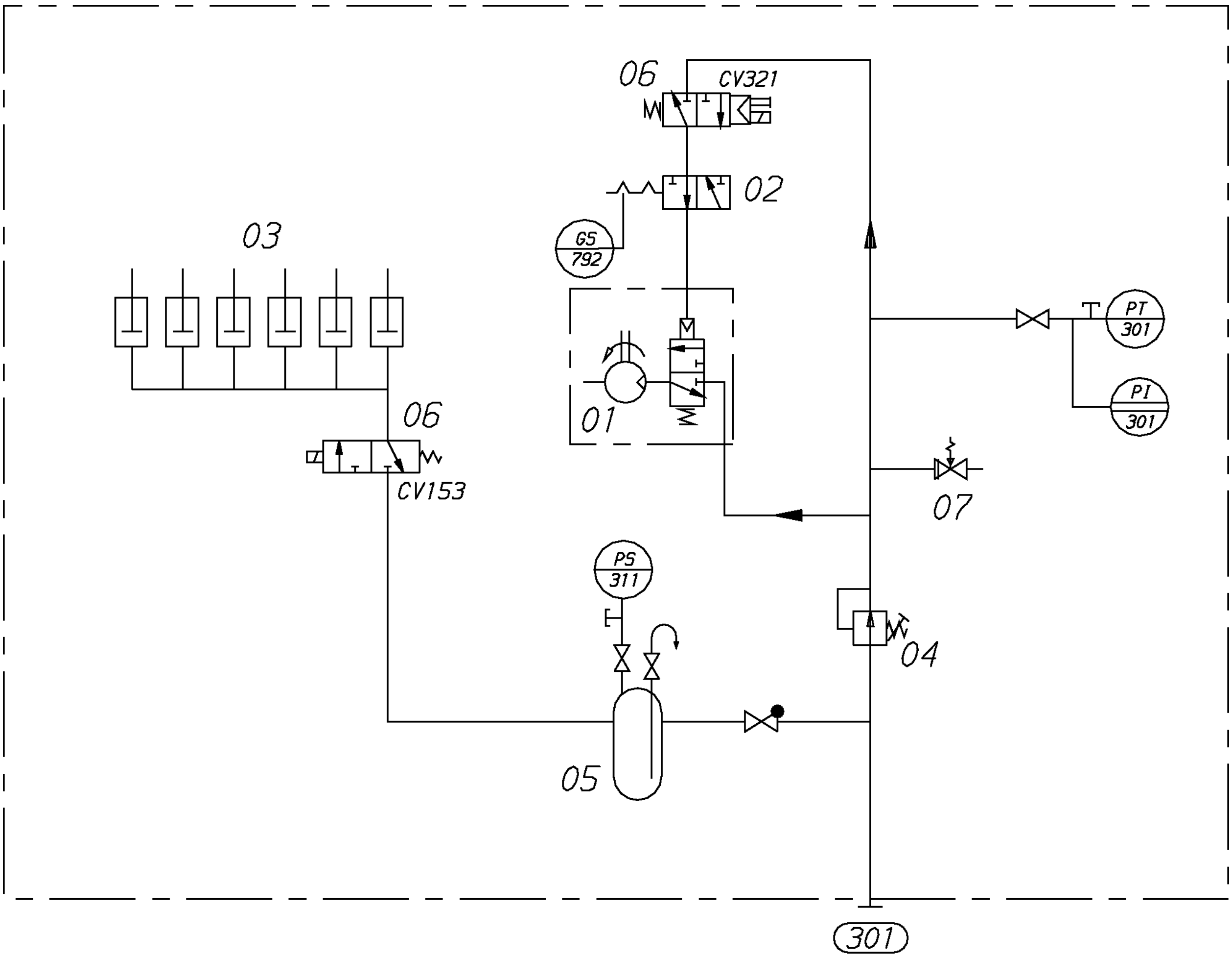
The compressed air system of the electro-pneumatic overspeed trip is connected to the starting air system. For this reason, the air supply to the engine must not be closed during operation.

Table 8.1 Dimensions of starting air pipe connections on the engine

Pipe connections		Size	Pressure class	Standard
301	Starting air inlet	OD28	PN100	DIN 2353

The nominal starting air pressure of 30 bar is reduced to 10 bar with a pressure regulator mounted on the engine.

Figure 8.1 Internal starting air system (4V76H4163b)



System components

01	Turbine starter
02	Blocking valve, turning gear engaged
03	Pneumatic cylinder at each injection pump

System components	
04	Pressure regulator
05	Air container
06	Solenoid valve
07	Safety valve

The compressed air system of the electro- pneumatic overspeed trip is connected to the starting air system. For this reason, the air supply to the engine must not be closed during operation.

8.2 External starting air system

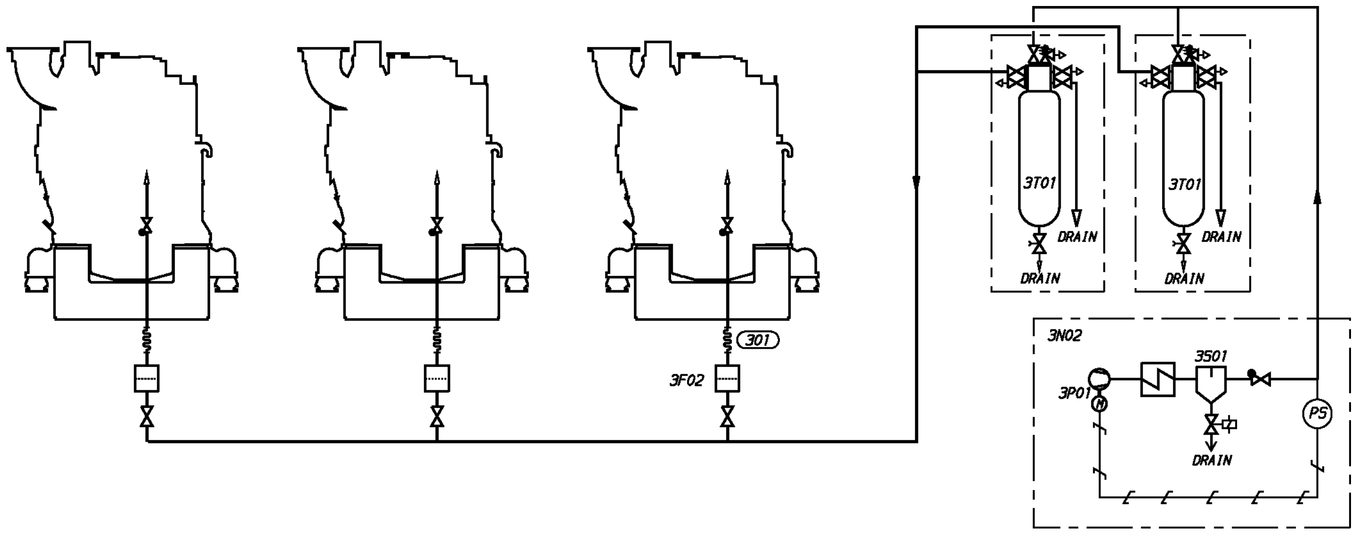
The design of the starting air system is partly determined by the rules of the classification societies. Most classification societies require the total capacity to be divided over two roughly equally sized starting air receivers and starting air compressors.

If the inertia of the directly coupled equipment is much larger than the normal reference equipment used on the testbed, the starting air consumption per start value has to be increased in relation to total (engine included) inertial masses involved.

The minimum pressures stated in the chapter *Technical data* assume that this pressure is available at engine inlet.

The rule requirements of some classification societies are not precise for multiple engine installations.

Figure 8.2 External starting air system (3V76H4164a)



System components		Pipe connections	
3F02	Air filter (Starting air inlet)	301	Starting air inlet
3N02	Starting air compressor unit		
3P01	Compressor (Starting air compressor unit)		
3S01	Separator (Starting air compressor unit)		
3T01	Starting air vessel		

8.2.1 Starting air filter (3F02)

Condense formation after the water separator (between starting air compressor and starting air receivers) has been experienced in tropical areas. This can, depending of the materials and surface treatments used, create and loosen abrasive rust from the piping, fittings and receivers.

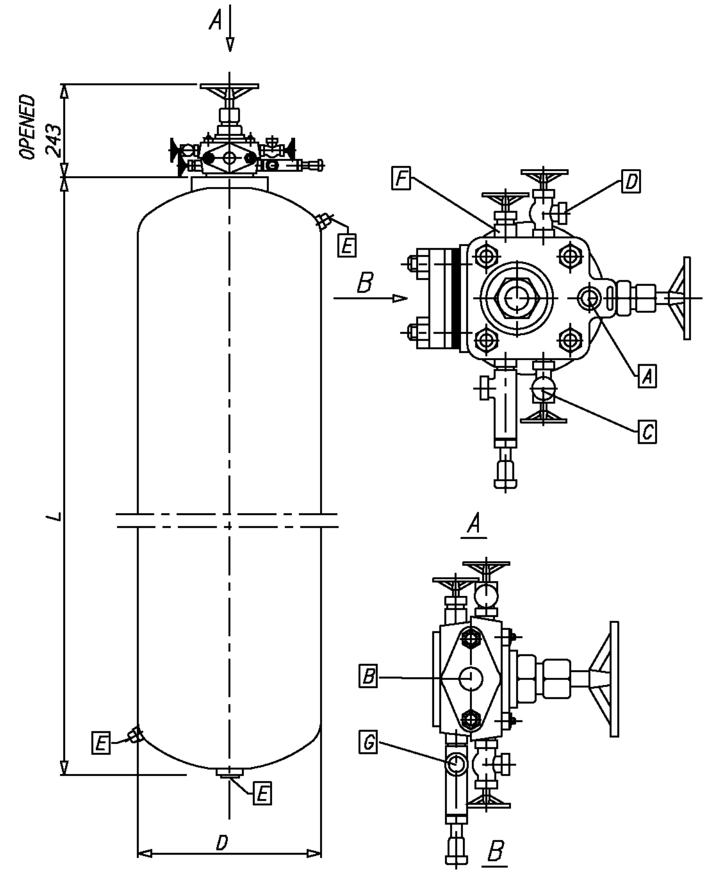
Therefore, it might be needed to install a filter in the external starting air system just before the engine to prevent particles to enter the starting air equipment if a high condense formation is expected.

8.2.2 Starting air receiver (3T01)

The starting air receiver should be dimensioned for a nominal pressure of 3 MPa.

The number and the capacity of the air receivers for propulsion engines depends on the requirements of the classification societies and the type of installation.

Figure 8.3 Starting air receiver (3V490097a)



	Function:	Size:		Function:	Size:
A	Inlet	Ø38	E	Drain	R1/4in
B	Outlet	R3/4in	F	Venting	
C	Pressure gauge	R1/4in	G	Safety valve	R1/2in
D	Drain	R1/4in			

Size [litres]	L [mm]	D [mm]	Weight [kg]
125	1807	320	150
250	1767	480	270
500	3204	480	480

The starting air receivers are to be equipped with at least a manual valve for condensate drainage. If the air receivers are mounted horizontally, there must be an alignment of 3-5° towards the drain valve to ensure efficient draining.

Recommended minimum number of starts for starting air vessel sizing:

Single main engine	2 x 125 l
Multiple main engines	2 x 250 l
1 - 3 auxiliary engines	2 x 125 l
> 3 auxiliary engines	2 x 250 l

8.2.3 Oil and water separator (3S01)

An oil and water separator should always be installed in the pipe between the compressor and the air receiver. Depending on the operation conditions of the installation, an oil and water separator may be needed in the pipe between the air receiver and the engine.

The starting air pipes should always be drawn with slope and be arranged with manual or automatic draining at the lowest points.

8.2.4 Starting air compressor (3N02)

At least two starting air compressors must be installed. It is recommended that the compressors are capable of filling the starting air receiver from minimum to maximum pressure in 15 - 30 minutes. For exact determination of the minimum capacity, the rules of the classification societies must be followed.

9. Cooling water system

9.1 General

Only treated fresh water may be used for cooling the engines.

To allow start on heavy fuel, the cooling water system has to be preheated to a temperature as near to the operating temperature as possible.

9.1.1 Water quality

The cylinder, turbocharger, charge air and oil are all cooled with fresh water. The pH-value and hardness of the water should be within normal values (hardness < 10°dH, pH > 6.5). The chloride and sulphate contents should be as low as possible (chlorides < 80 mg/l, sulphates max. 150mg/l). To prevent rust formation in the cooling water system, the use of corrosion inhibitors is mandatory.

Shore water is not always suitable. The hardness of shore water may be too low, which can be compensated by additives, or too high, causing scale deposits even with additives.

Fresh water generated by a reverse osmosis plant onboard often has a high chloride content (higher than the permitted 80 mg/l) causing corrosion.

For ships with a wide sailing area a safe solution is to use fresh water produced by an evaporator (onboard), using additives according to the Instruction Manual (important).

Rain water is unsuitable as cooling water due to a high oxygen and carbon dioxide content, causing a great risk for corrosion.

9.1.2 Approved cooling water treatment products

Only products from approved suppliers may be used as additives. The use of emulsion oils phosphates and borate (sole) is not recommended.

All the approved cooling water treatment products are given in the following table.

Product	Supplier
CorrShield NT 4293 CorrShield NT 4200	GE Betz Europe, Belgium GE Betz, Trevose, USA
DEWT-NC powder Drewgard 4109 Liquidewt Maxigard Vecom CWT Diesel QC-2	Drew Ameroid Marine Division Boonton, USA
Cooltreat 651	Houseman Ltd. Burnham, Slough, U.K.
Q8 Corrosion Inhibitor Long-Life	Kuwait Petroleum (Danmark) AS Virum, Denmark
Marisol CW	Maritech AB Kristianstad, Sweden
Nalco 39 L Nalcool 2000	Nalco Chemical Company Naperville, Illinois, USA
Nalfleet EWT 9-108 Nalfleet CWT 9-131C Nalcool 2000	Nalfleet Marine Chemicals Nortwich, Cheshire, U.K.
RD11 RD11M RD25	Rohm & Haas Paris, France
Korrostop KV	RRS-Yhtiöt Jäppilä, Finland
Ruostop XM	Tampereen Prosessi- Insinöörit Pirkkala, Finland
Havoline XLi	S.A. Arteco N.V., Belgium

Product	Supplier
Havoline XLi	Texaco Global Products, LLC Houston, USA
WT Supra	TotalFinaElf Paris, France
Dieselguard NB Rocor NB liquid Cooltreat AL	Unitor ASA Oslo, Norway
Vecom CWT Diesel QC-2	Vecom Holding B.V. Maassluis, Holland

Glycol

Use of glycol in the cooling water is not recommended. It is however possible to use up to 10% glycol without engine derating. For higher concentrations the engine shall be derated 0.67% for each percentage unit exceeding 10.

9.2 Internal cooling water system

9.2.1 Engine driven circulating cooling water pumps

The LT and HT circuit circulating pumps are always engine driven. The pumps are centrifugal pumps driven by the engine crankshaft through a gear transmission.

The HT and LT water pump impeller diameters and corresponding pump curves are presented in the following tables.

The HT and LT circuit circulating pumps are always engine driven. The pumps are centrifugal pumps driven by the engine crankshaft through a gear transmission.

The HT and LT water pump curves are presented in the pump diagrams:

On request, connections for electric motor driven stand-by pumps can be provided.

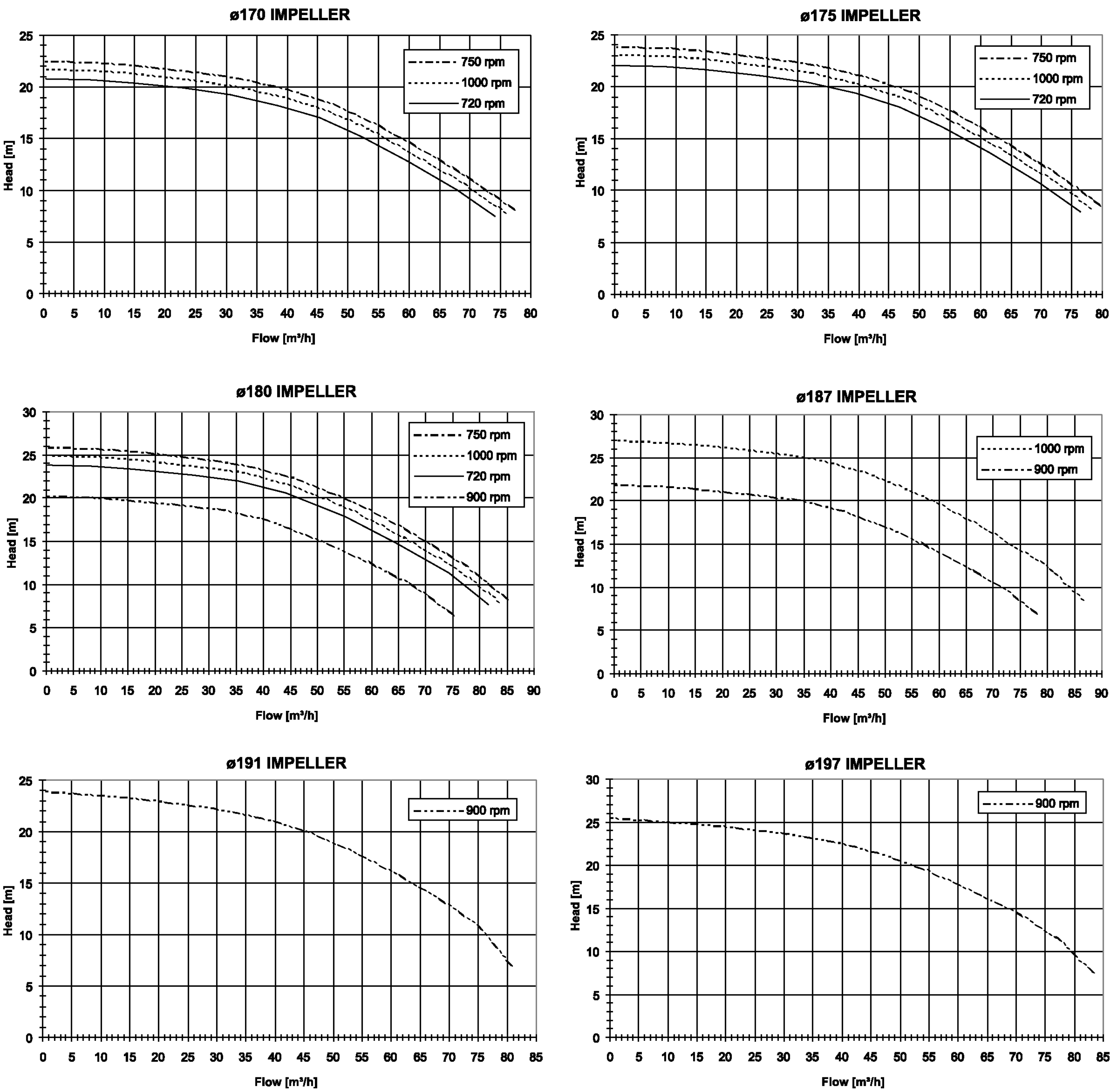
Pump materials:	
housing	cast iron
impeller	bronze
shaft	stainless steel
sealing	mechanical

Capacities are according to chapter *Technical data*.

Table 9.1 Impeller diameters and nominal flows of engine driven HT & LT pumps

Engine	Engine speed [rpm]	HT impeller [Ø mm]	LT impeller [Ø mm]
4L20	720 750 900 1000	170 170 180 170	170 170 187 170
6L20	720 750 900 1000	175 175 187 175	175 175 187 175
8L20	720 750 900 1000	180 180 191 180	180 180 197 187
9L20	720 750 900 1000	180 180 191 180	180 180 197 187

Figure 9.1 Impeller curves



9.2.2 Charge air cooler

The charge air cooler built on the engine is of the insert type with removable cooler insert.

For design data see chapter *Technical data*.

The charge air is cooled by fresh water and connected in series with the lubricating oil cooler. The water temperature after the lubricating oil cooler is controlled by a thermostatic valve.

9.2.3 Engine driven sea water pump

For main engines (only) an engine driven sea-water pump is available:

Figure 9.2 Engine driven pump, 900 rpm

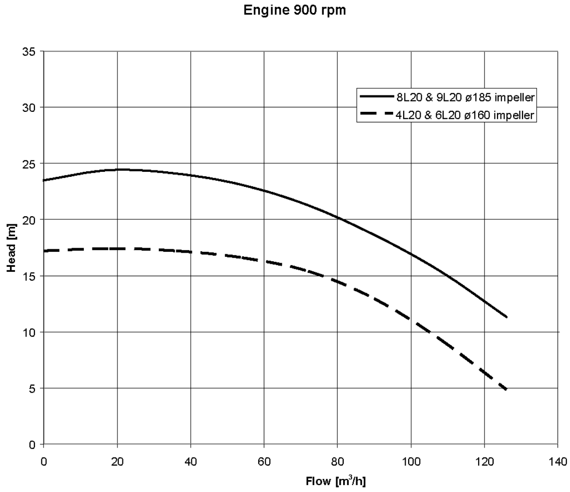
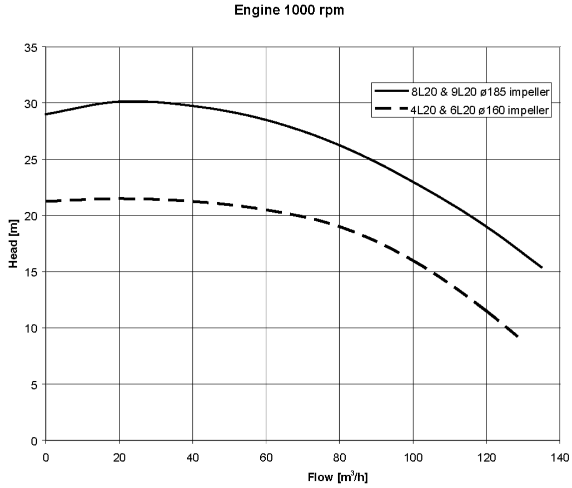


Figure 9.3 Engine driven pump, 1000 rpm



9.2.4 Thermostatic valve LT-circuit

The direct acting thermostatic valve controls the water outlet temperature from the lubricating oil cooler.

set point	49°C (43...54°C)

9.2.5 Thermostatic valve HT-circuit

The direct acting thermostatic valve controls the engine outlet temperature.

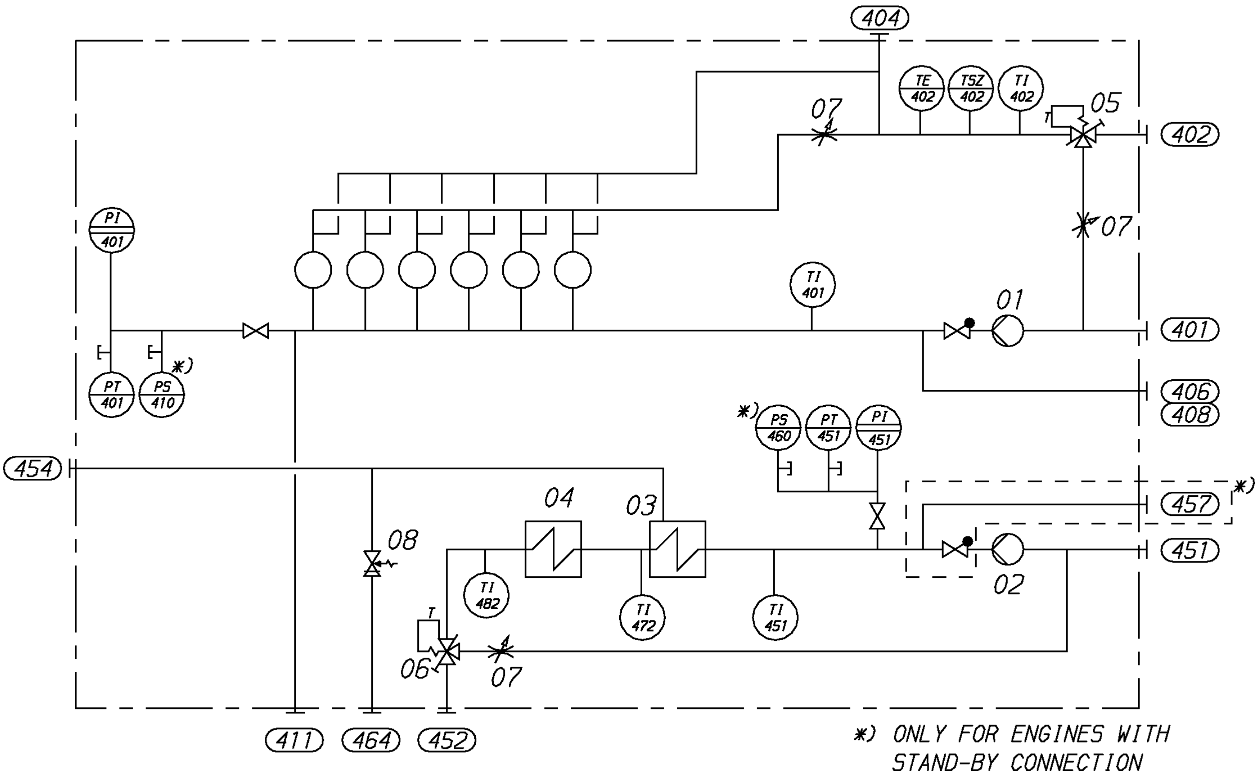
set point	91°C (87...98°C)

9.2.6 Lubricating oil cooler

The lubricating oil cooler is cooled by fresh water and connected in series with the charge air cooler.

Pipe connections		Size	Pressure class	Standard
401	HT-water inlet	DN65	PN16	ISO 7005-1
402	HT-water outlet	DN65	PN16	ISO 7005-1
404	HT-water air vent	OD12	PN250	DIN 2353
406	Water from preheater to HT-circuit	OD28	PN100	DIN 2353
408	HT-water from stand-by pump	DN65	PN16	ISO 7005-1
411	HT-water drain	M10 x 1	-	Plug
451	LT-water inlet	DN80	PN16	ISO 7005-1
452	LT-water outlet	DN80	PN16	ISO 7005-1
454	LT-water air vent from air cooler	OD12	PN250	DIN 2353
457	LT-water from stand-by pump	DN80	PN16	ISO 7005-1
464	LT-water drain	M18 x 1.5	-	Plug

Figure 9.4 Internal cooling water system (4V76C5822a)



System components	
01	HT-cooling water pump
02	LT-cooling water pump
03	Charge air cooler
04	Lubricating oil cooler
05	HT-thermostatic valve
06	LT-thermostatic valve
07	Adjustable orifice
08	Safety valve

9.3 External cooling water system

The external cooling water system shall provide the engine with de-aerated cooling water of correct quality, flow, pressure and temperature.

The fresh water pipes should be designed to minimize the flow resistance in the external piping. Galvanized pipes should not be used for fresh water.

The use of pipes having galvanized inner surfaces is not allowed in the external cooling water system. This is because some cooling water additives react with zinc coating generating harmful sludge, and at elevated temperatures zinc becomes nobler than cast iron leading to heavy corrosion of engine components.

Ships (with ice class) designed for cold sea-water should have temperature regulation with a recirculation back to the sea chest:

- For heating of the sea chest to melt ice and slush, to avoid clogging the sea-water strainer
- To increase the sea-water temperature to enhance the temperature regulation of the LT-water

9.3.1 Sea water pump (4P11)

The sea water pumps have to be electrically driven. The capacity of the pumps is determined by the type of the coolers and the heat to be dissipated.

9.3.2 Fresh water central cooler (4E08)

The fresh water cooler can be of either tube or plate type. Due to the smaller dimensions the plate cooler is normally used. The fresh water cooler can be common for several engines, also one independent cooler per engine can be used.

Design data:

- Fresh water flow, see chapter *Technical Data*
- Pressure drop on fresh water side, max. 60 kPa (0.6 bar)

In case where fresh water central cooler is used for combined LT and HT water flows in a parallel system the total flow can be calculated with the following formula:

$$q = q_{LT} + \frac{3.6 \times \Phi}{4.19 \times (T_{OUT} - T_{IN})}$$

where:

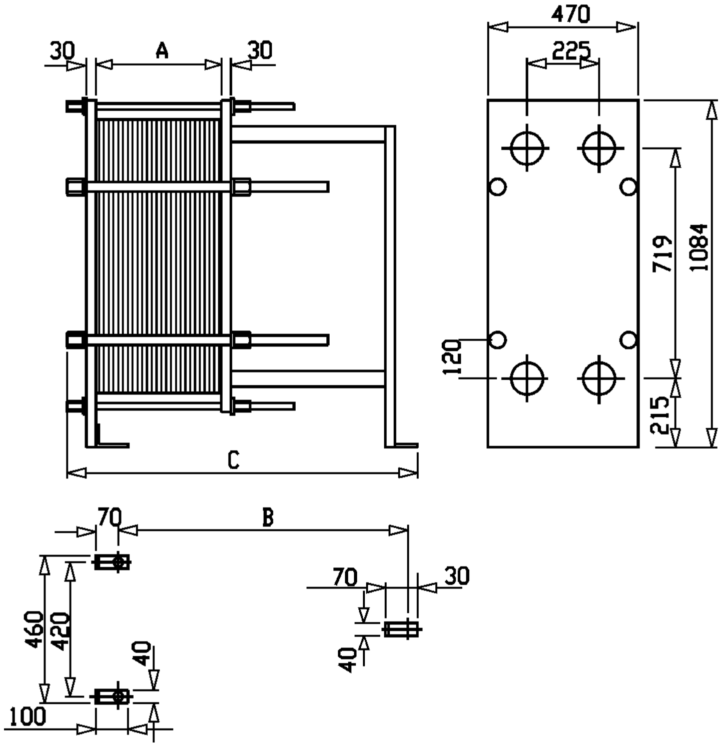
q =	total fresh water flow [m³/h]
q_{LT} =	nominal LT pump capacity[m³/h]
Φ =	heat dissipated to HT water [kW]
T_{out} =	HT water temperature after engine (91°C)
T_{in} =	HT water temperature after cooler (38°C)

If the flow resistance in the external pipes is high it should be observed when designing the cooler.

Sea-water flow	acc. to cooler manufacturer, normally 1.2 - 1.5 x the fresh water flow
Pressure drop on sea-water side, norm.	80 - 140 kPa (0.8 - 1.4 bar)
Fresh water temperature after LT cooler max.	38°C
Fresh water temperature after HT cooler max.	60-65°C
Margin for safety and fouling	15%

See also the table showing example coolers with calculation data.

Figure 9.5 Central cooler, main dimensions (4V47E0188b)



Engine		Cooling water			Sea-water			Measures			Weight	
Type	**Engine speed**	**Flow [m³/h]**	**Tcw, in [°C]**	**Tcw, out [°C]**	**Flow [m³/h]**	**Tsw, in [°C]**	**Tsw, out [°C]**	**A [mm]**	**B [mm]**	**C [mm]**	**Dry [kg]**	**Wet [kg]**
4L20	750	22	52.1	38	30	32	42.6	80	505	695	270	287
-	1000	27	54.3	38	36	32	44.3	106	505	695	275	298
6L20	750	33	52	38	44	32	42.6	121	655	845	280	306
-	1000	40	53.3	38	53	32	43.5	150	655	845	288	321
8L20	750	44	52	38	59	32	42.6	156	655	845	289	323
-	1000	53	53.6	38	71	32	43.8	198	655	845	298	341
9L20	750	49	52.1	38	67	32	42.7	186	655	845	293	336
-	1000	59	53.7	38	80	32	43.8	221	905	1095	305	354

As an alternative for the central coolers of the plate or of the tube type a box cooler can be installed. The principle of box cooling is very simple. Cooling water is forced through a U-tube-bundle, which is placed in a sea-chest having inlet- and outlet-grids. Cooling effect is reached by natural circulation of the surrounding water. The outboard water is warmed up and rises by its lower density, thus causing a natural upward circulation flow which removes the heat.

Box cooling has the advantage that no raw water system is needed, and box coolers are less sensitive for fouling and therefor well suited for shallow or muddy waters.

9.3.3 Stand-by circulating cooling water pumps

The pumps should be centrifugal pumps driven by an electric motor. Capacities according to chapter *Technical data*.

HT-pump (4P03)

HT cooling water circuit must have individual pumps for each circuit.

LT-pump (4P05)

LT cooling water circuit can have either dedicated pump for each engine, or alternatively several engines can share one pump.

NOTE! Some classification societies require spare pumps to be carried on board also in case of a multiple engine installation. By installing standby pumps also for multiple engine installation this requirement can be fulfilled.

9.3.4 Expansion tank (4T05)

The expansion tank should compensate for volume changes in the cooling water system, serve as venting arrangement and provide sufficient static pressure for the cooling water circulating pumps.

Design data:	
pressure from the expansion tank	70 - 150 kPa (0.7...1.5 bar)
volume	min. 10% of the system

Concerning engine water volumes, see chapter *Technical data*.

The tank should be equipped so that it is possible to dose water treatment agents.

The vent pipe of each engine should be drawn to the tank separately, continuously rising, and so that mixing of air into the water cannot occur (the outlet should be below the water level).

The expansion tank is to be provided with inspection devices.

9.3.5 Drain tank (4T04)

It is recommended to provide a drain tank to which the engines and coolers can be drained for maintenance so that the water and cooling water treatment can be collected and reused. For the water volume in the engine, see chapter *Technical data*.

Most of the cooling water in the engine can be recovered from the HT-circuit, whereas the amount of water in the LT circuit is small.

9.3.6 Preheating

Engines started and stopped on heavy fuel and all engines on which high load will be applied immediately after start (stand-by generating sets) have to be preheated as close to the actual operating temperature as possible, or minimum 60°C. Preheating is however, recommended for all engines, also main engines running on MDF only.

The energy required for heating of the HT-cooling water in the main and auxiliary engines can be taken from a running engine or a separate source. In both cases a separate circulating pump should be used to ensure the circulation. If the cooling water systems of the main and auxiliary engines are separated from each other in other respects, the energy is recommended to be transmitted through heat exchangers.

For installations with several engines the preheater unit can be chosen for heating up only part of the engines. The heat from a running engine can be used and therefore the power consumption of the heater will be less than the nominal capacity.

Heater (4E05)

Steam, electrical or thermal oil heaters can be used.

Design data:	
Preheating temperature, min.	60°C
Capacity to keep hot engine warm	1 kW/cyl

Capacity needed for heating up engine to 60°C, see formula below:

$$P = \frac{(60 - t_0)(m_{eng} \times 0.14 + V_{LO} \times 0.48 + V_{FW} \times 1.16)}{T}$$

where:

P = Preheater output [kW]

t_0 = Ambient temperature [°C]

where:

m_{eng} = Engine weight [ton]

V_{LO} = Lubricating oil volume [m^3] (wet sump engines only)

V_{FW} = HT water volume [m^3]

T= Preheating time [h]

Preheating pump (4P04)

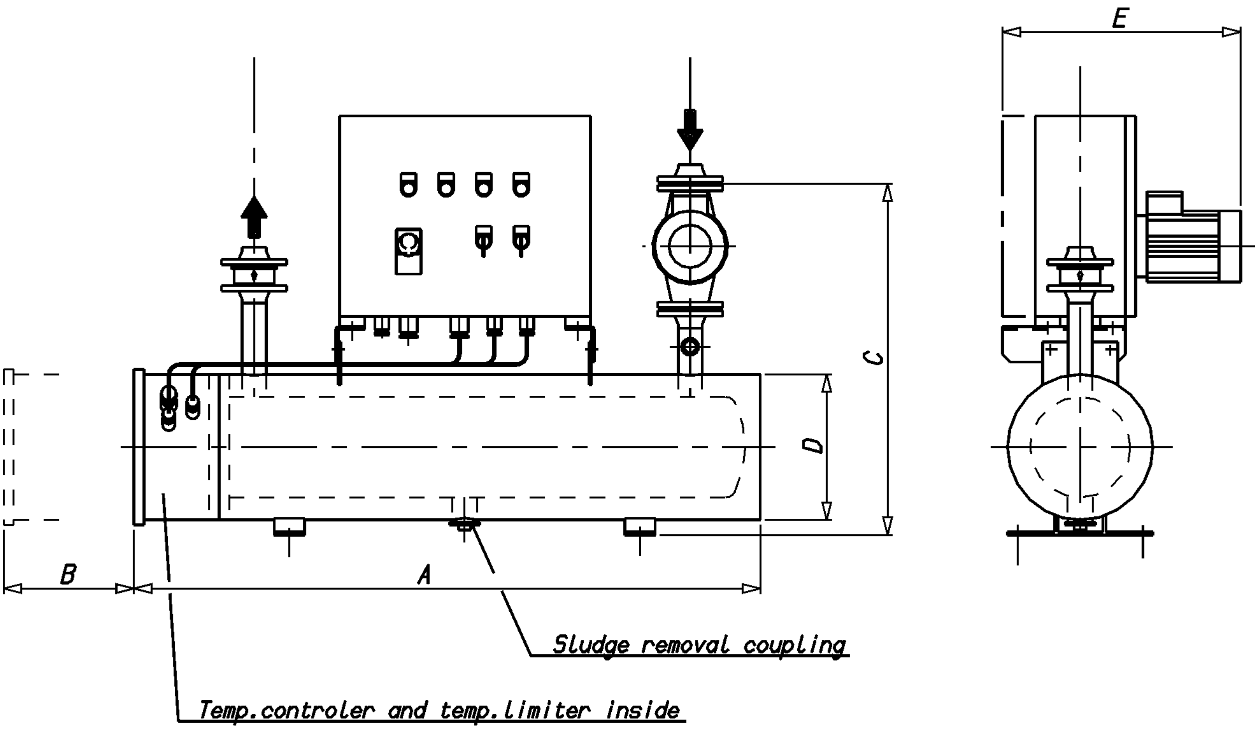
Design data:	
capacity	0.3 m^3/h x cyl.
pressure	abt. 80 kPa (0.8 bar)

Preheating unit (4N01)

A complete preheating unit can be supplied as option. The unit comprises:

- Electric or steam heaters
- Circulating pump
- Control cabinet for heaters and pump
- One set of thermometers

Figure 9.6 Preheating unit, electric (3V60L0653a)



Heater capacity	Pump capacity	Weight	Pipe connections	Dimensions				
kW	m^3 / h	kg	Inlet / Outlet	A	B	C	D	E
7.5	3	75	DN40	1050	720	610	190	425
12	3	93	DN40	1050	550	660	240	450
15	3	93	DN40	1050	720	660	240	450
18	3	95	DN40	1250	900	660	240	450
22.5	8	100	DN40	1050	720	700	290	475

Heater capacity	Pump capacity	Weight	Pipe connections	Dimensions				
kW	m^3 / h	kg	Inlet / Outlet	A	B	C	D	E
27	8	103	DN40	1250	900	700	290	475
30	8	105	DN40	1050	720	700	290	475
36	8	125	DN40	1250	900	700	290	475
45	8	145	DN40	1250	720	755	350	505
54	8	150	DN40	1250	900	755	350	505

9.3.7 Air venting (4S01)

Air and gas may be entrained in the piping after overhaul, centrifugal pump seals may leak, or air or gas may leak from in any equipment connected the HT- or LT-circuit, such as diesel engine, water cooled starting air compressor etc.

As presented in the external cooling diagrams, it is recommended that either of the following air venting equipment is installed:

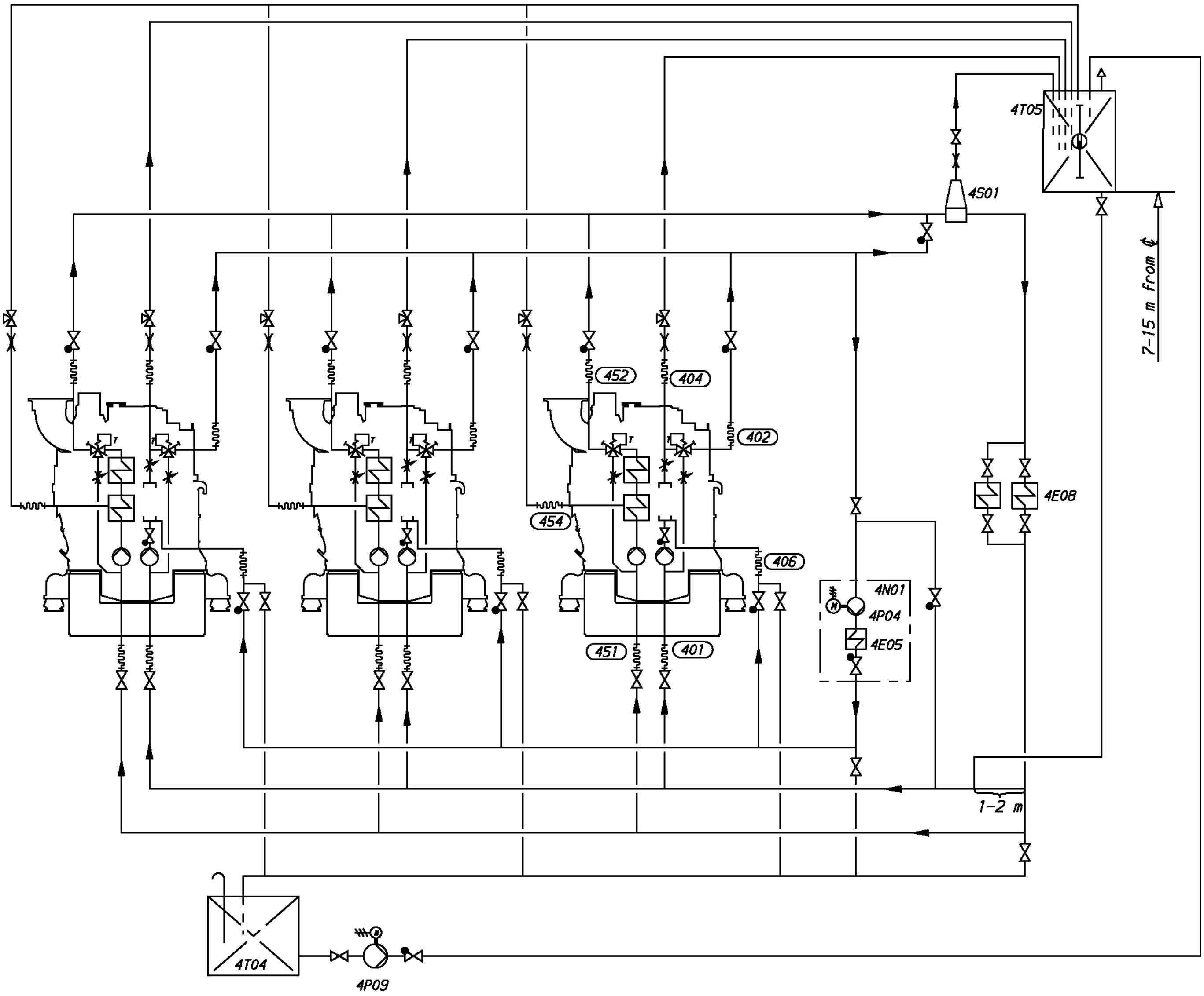
1. At the HT-outlet from the engine. This is necessary for a quick venting after starting the engine, especially after overhaul when entrained air may remain in the system, and especially at departures at low load, when the HT thermostatic valve recirculates all water. At higher load when a part of the HT-water goes to the cooler, any possible air or gas bubbles may still be recirculated depending on the geometry and position of the HT thermostatic valve. If the branch to the cooler is vertically down the bubbles may be conducted to the by-pass line and back into circulation.
2. One in the LT system line for venting of any entrained air.

9.3.8 Waste heat recovery

The waste heat of the HT-circuit may be used for fresh water production, central heating, tank heating etc. In such cases the piping system should permit by-passing of the central cooler. With this arrangement the HT-water flow through the heat recovery can be increased.

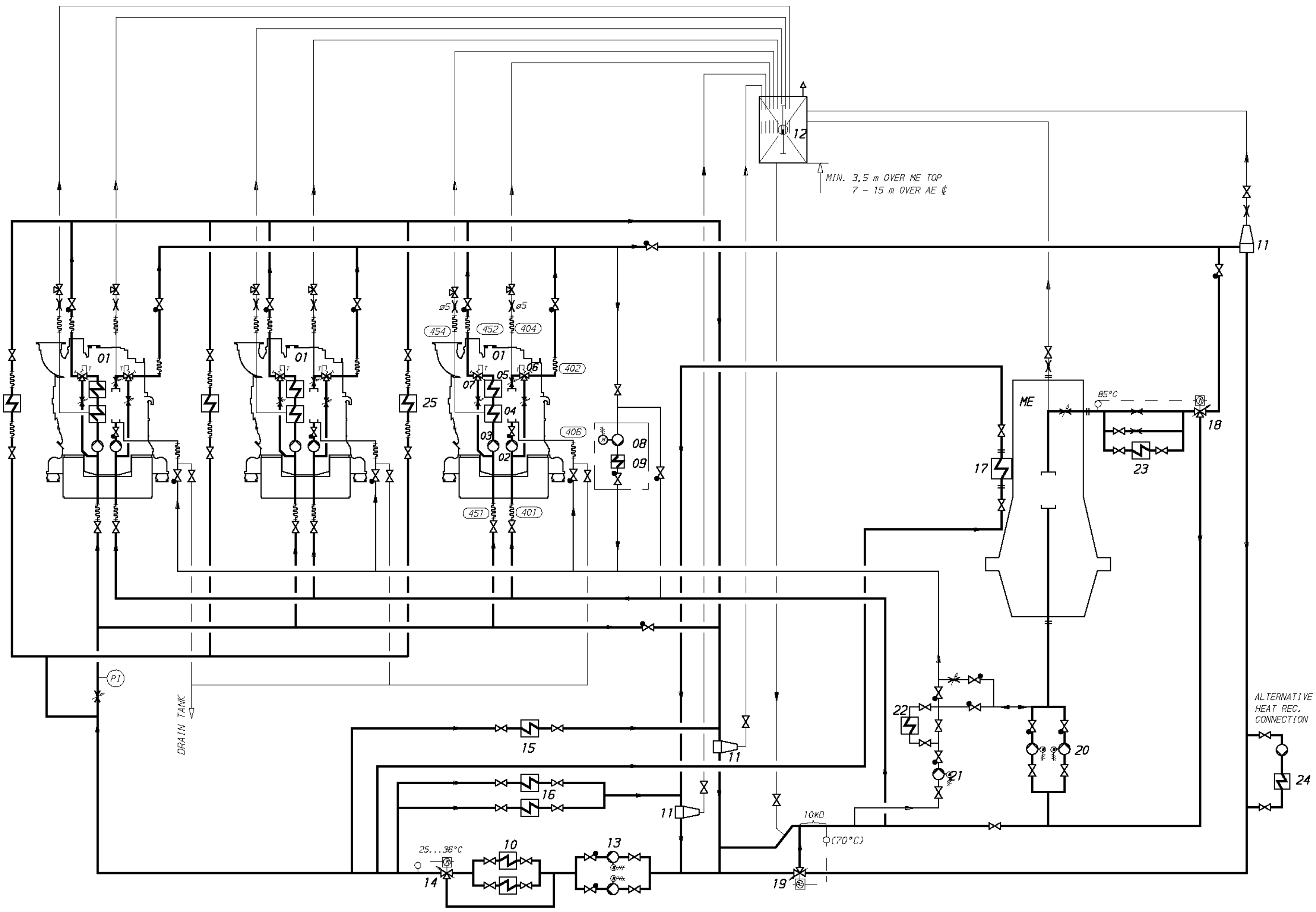
9.4 Example system diagrams

Figure 9.7 Cooling water system, auxiliary engines (3V76C5823)



System components		Pipe connections	
4E05	Heater (Preheating unit)	401	HT-water inlet
4E08	Central cooler	402	HT-water outlet
4N01	Preheating unit	404	HT-air vent
4P04	Circulating pump (Preheating unit)	406	Water from preheater to HT-circuit
4P09	Transfer pump	451	LT-water inlet
4S01	Air venting	452	LT-water outlet
4T04	Drain tank	454	LT-water air vent from air cooler
4T05	Expansion tank		

Figure 9.8 Cooling water system for mixed LT and HT-circuit (DAAE018225a)



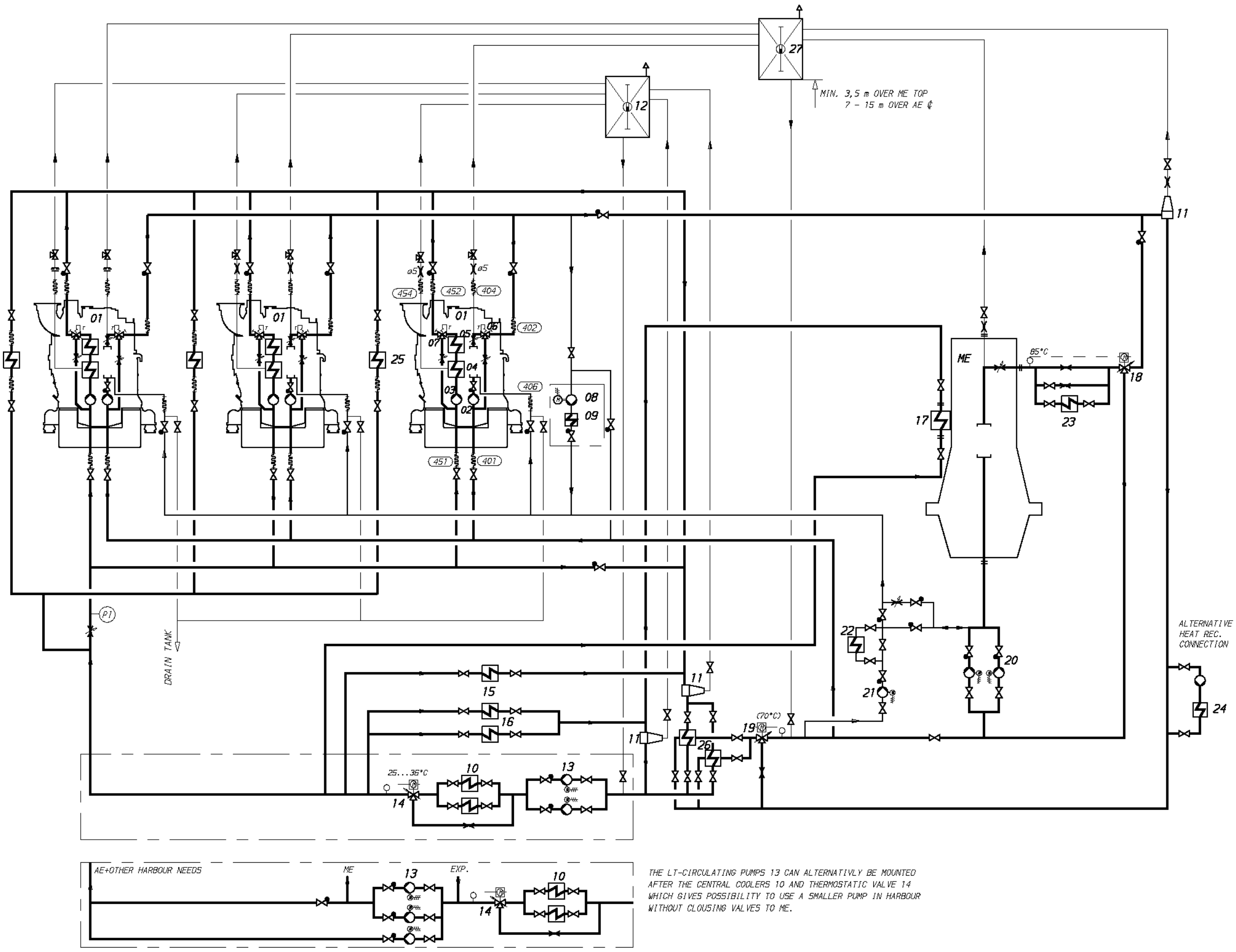
System components

01	L20 genset	14	LT-Thermostatic valve
02	HT-Circulating pump	15	Lubricating oil cooler, Main engine
03	LT-Circulating pump	16	Ancillary plants
04	Air cooler	17	Scavenge air cooler, Main engine
05	Oil cooler	18	Thermostatic valve HT
06	HT-Thermostatic valve	19	Thermostatic valve HT, heat recovery/preheating
07	LT-Thermostatic valve	20	HT-Circulating pump, Main engine
08	Preheating pump, Auxiliary engine	21	HT-Preheating pump, Main engine + Auxiliary engine
09	Preheater, Auxiliary engine	22	HT-Preheater
10	Central cooler	23	Heat recovery from HT-circuit, Main engine
11	Air venting	24	Heat recovery from LT-circuit, Main engine + Auxiliary engine
12	Expansion tank	25	Alternator cooler
13	LT-Circulating pump		

Pipe connections

401	HT-Water inlet	451	LT-Water inlet
402	HT-Water outlet	452	LT-Water outlet
404	HT-Air vent	454	LT-Water air vent from air cooler
406	Water from preheater to HT-circuit		

Figure 9.9 Cooling water system for split LT and HT-circuit (DAAE018600)



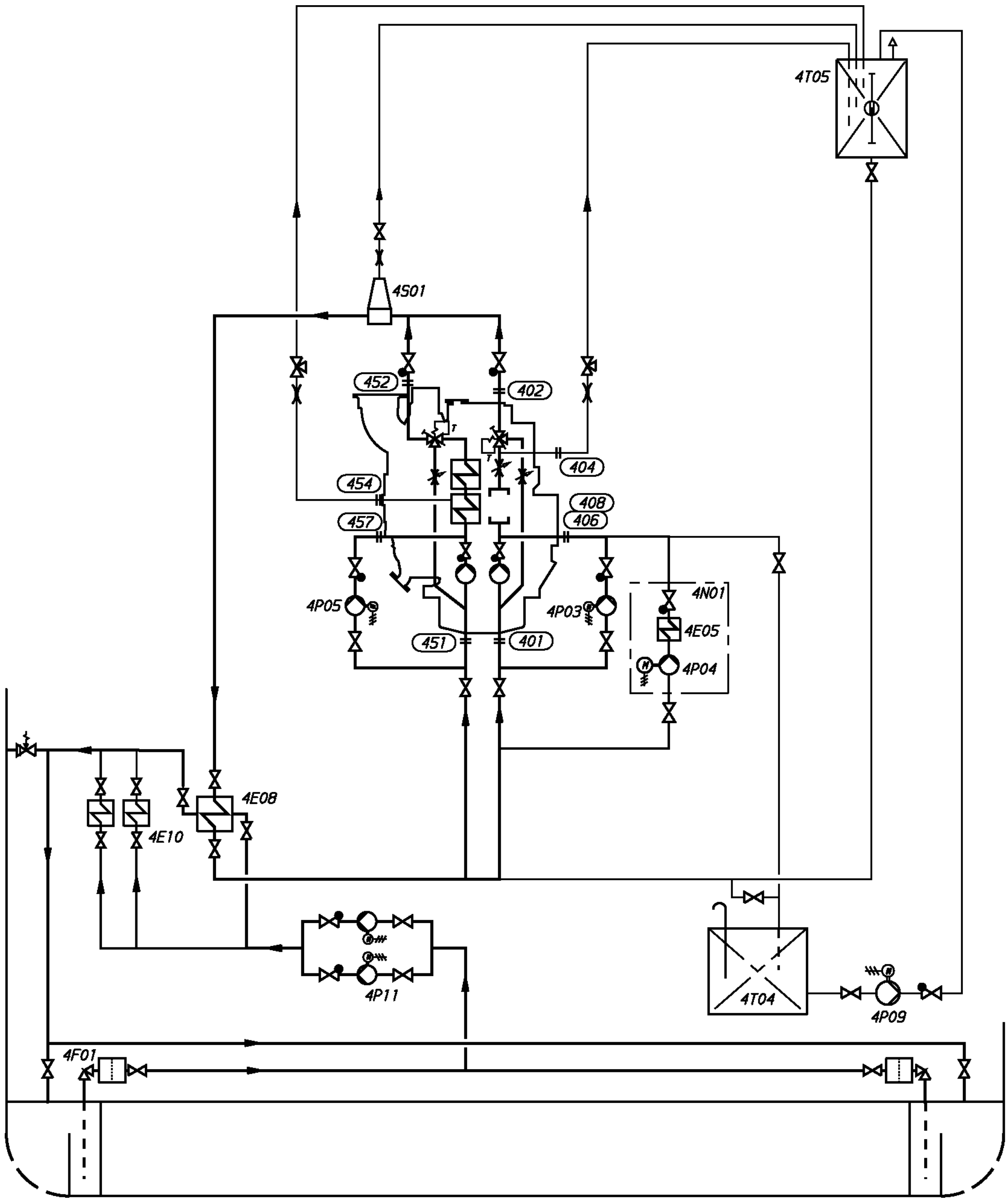
System components

01	L20 Genset	15	Lubricating oil cooler, Main engine
02	HT-Circulating pump	16	Ancilliary plants
03	LT-Circulating pump	17	Scavenge air cooler, Main engine
04	Air cooler	18	Thermostatic valve HT
05	Oil cooler	19	Thermostatic valve HT, heat recovery/preheating
06	HT-Thermostatic valve	20	HT-Circulating pump, Main engine
07	LT-Thermostatic valve	21	HT-Preheating pump, Main engine + auxiliary engine
08	Preheating pump, Auxiliary engine	22	HT-Preheater
09	Preheater, Auxiliary engine	23	Heat recovery form HT-circuit, Main engine
10	Central cooler	24	Heat recovery from HT-circuit, Main engine + auxiliary engine
11	Air venting	25	Alternator cooler
12	Expansion tank	26	HT-Cooler
13	LT-Circulating pump	27	HT-Expansion tank
14	LT-Thermostatic valve		

Pipe connections

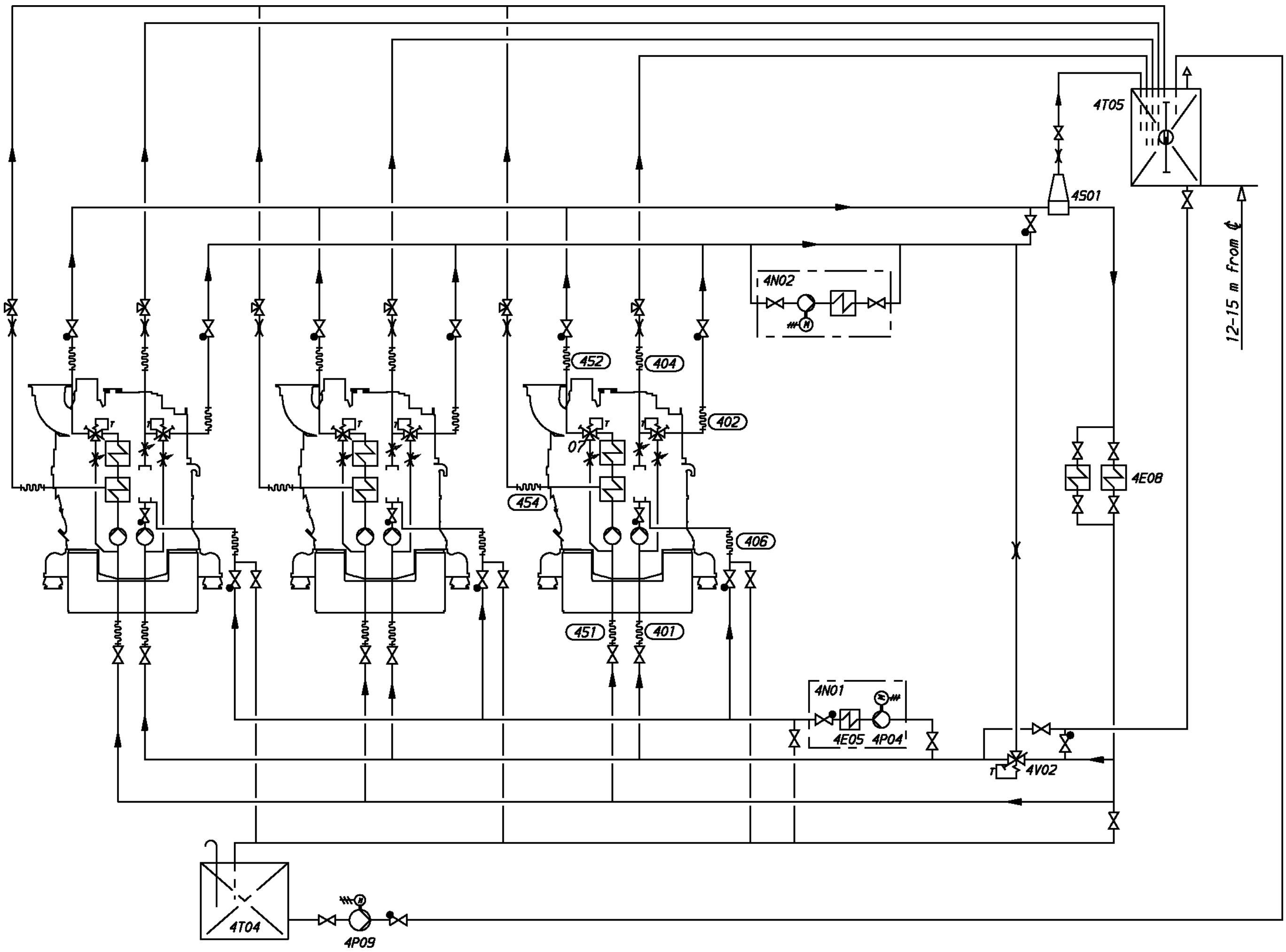
401	HT-Water inlet	451	LT-Water inlet
402	HT-Water outlet	452	LT-Water
404	HT-Air vent	454	LT-Water air vent from air cooler
406	Water from preheater to HT-circuit		

Figure 9.10 Cooling water system, main engine (3V76C5825)



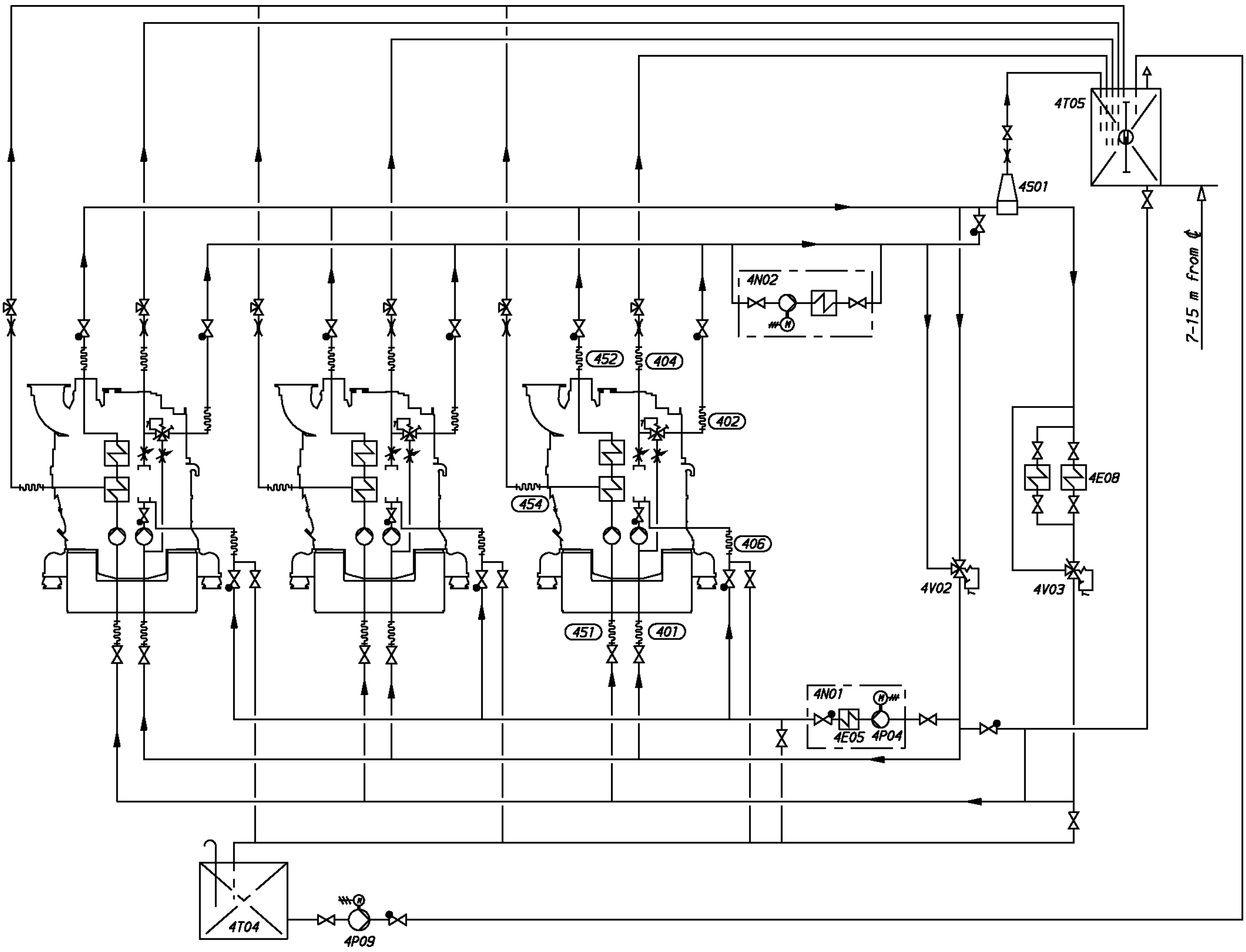
System components		Pipe connections	
4E05	Heater (Preheating unit)	401	HT-water inlet
4E08	Central cooler	402	HT-water outlet
4E10	Cooler (Reduction gear)	404	HT-air vent
4F01	Suction strainer (Sea water)	406	Water from preheater to HT-circuit
4N01	Preheating unit	408	HT-water from stand-by pump
4P03	Stand-by pump (HT)	451	LT-water inlet
4P04	Circulating pump (Preheating unit)	452	LT-water outlet
4P05	Stand-by pump (LT)	454	LT-water air vent from air cooler
4P09	Transfer pump	457	LT-water from stand-by pump
4P11	Circulating pump (Sea water)		
4S01	Air venting		
4T04	Drain tank		
4T05	Expansion tank		

Figure 9.11 Cooling water system, HFO engines with evaporator (3V76C5826)



System components		Pipe connections	
4E05	Heater (Preheating unit)	401	HT-water inlet
4E08	Central cooler	402	HT-water outlet
4N01	Preheating unit	404	HT-air vent
4N02	Evaporator unit	406	Water from preheater to HT-circuit
4P04	Circulating pump (Preheating unit)	451	LT-water inlet
4P09	Transfer pump	452	LT-water outlet
4S01	Air venting	454	LT-water air vent from air cooler
4T04	Drain tank		
4T05	Expansion tank		
4V02	Thermostatic valve (Heat recovery)		

Figure 9.12 Cooling water system, MDO engines with evaporator (3V76C5827)



System components		Pipe connections	
4E05	Heater (Preheating unit)	401	HT-water inlet
4E08	Central cooler	402	HT-water outlet
4N02	Preheating unit	404	HT-air vent
4P04	Evaporator unit	406	Water from preheater to HT-circuit
4P09	Circulating pump (Preheating unit)	451	LT-water inlet
4S01	Transfer pump	452	LT-water outlet
4T04	Air venting	454	LT-water air vent from air cooler
4T05	Drain tank		
4P11	Expansion tank		
4V02	Thermostatic valve (Heat recovery)		
4V03	Thermostatic valve (LT)		

10. Combustion air system

10.1 Engine room ventilation

To maintain acceptable operating conditions for the engines and to ensure trouble free operation of all equipment, attention shall be paid to the engine room ventilation and the supply of combustion air.

The air intakes to the engine room must be so located that water spray, rain water, dust and exhaust gases cannot enter the ventilation ducts and the engine room.

The dimensioning of blowers and extractors should ensure that an overpressure of about 5 mmWC is maintained in the engine room in all running conditions.

For the minimum requirements concerning the engine room ventilation and more details, see applicable standards, such as ISO 8861.

The amount of air required for ventilation is calculated from the total heat emission Φ to evacuate. To determine Φ, all heat sources shall be considered, e.g.:

- Main and auxiliary diesel engines
- Exhaust gas piping
- Generators
- Electric appliances and lighting
- Boilers
- Steam and condensate piping
- Tanks

It is recommended to consider an outside air temperature of no less than 35°C and a temperature rise of 11°C for the ventilation air.

The amount of air required for ventilation is then calculated using the formula:

$$Q_V = \frac{\Phi}{\rho \times \Delta t \times c}$$

where:

Qv = amount of ventilation air [m³/s]

Φ = total heat emission to be evacuated [kW]

ρ = density of ventilation air 1.13 kg/m³

ΔT = temperature rise in the engine room [°C]

c = specific heat capacity of the ventilation air 1.01 kJ/kgK

The heat emitted by the engine is listed in chapter *Technical data*.

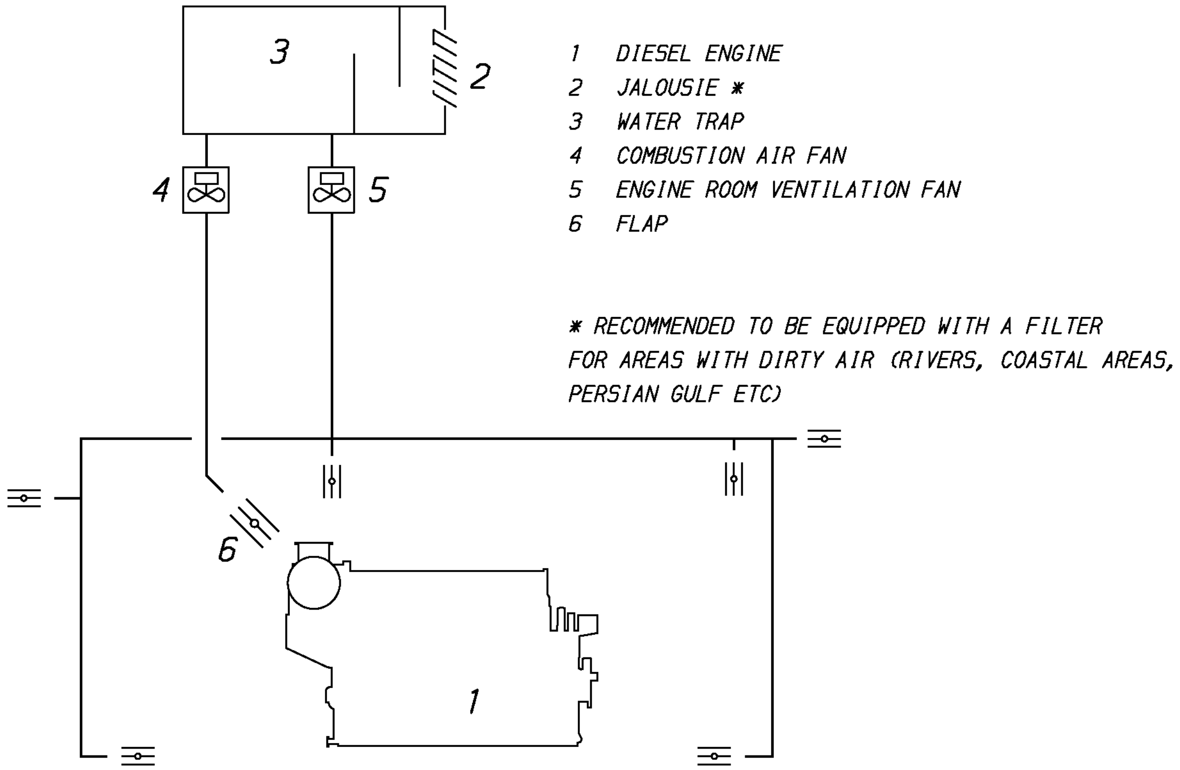
The engine room ventilation has to be provided by separate ventilation fans. These fans should preferably have two-speed electric motors (or variable speed). Thus flexible operation is possible, e.g. in port the capacity can be reduced during overhaul of the main engine when it is not preheated (and therefore not heating the room).

The ventilation air is to be equally distributed in the engine room considering air flows from points of delivery towards the exits. This is usually done so that the funnel serves as an exit for the majority of the air. To avoid stagnant air, extractors can be used.

It is good practice to provide areas with significant heat sources, such as separator rooms with their own air supply and extractors.

For very cold conditions a preheater in the system should be considered. Suitable media could be thermal oil or water/glycol to avoid the risk for freezing. If steam is specified as a heating system for the ship the preheater should be in a secondary circuit.

Figure 10.1 Engine room ventilation (4V69E8169a)



10.2 Combustion air system design

The combustion air shall be supplied by separate combustion air fans, with a capacity slightly higher than the maximum air consumption.

For the required amount of combustion air, see the chapter *Technical data*.

The fans should preferably have a two-speed electric motors (or variable speed) for enhanced flexibility.

In multi-engine installations each main engine should preferably have its own combustion air fan. Thus the air flow can be adapted to the number of engines in operation.

The combustion air should be delivered through a dedicated duct close to the turbocharger, directed towards the turbocharger air intake. The outlet of the duct should be equipped with a flap for controlling the direction and amount of air.

Also other combustion air consumers like other engines, gas turbines and boilers shall be served by dedicated combustion air ducts.

Usually, the air required for combustion is taken from the engine room through a filter fitted on the turbocharger. This reduces the risk for too low temperatures and contamination of the combustion air. It is imperative that the combustion air is free from sea water, dust, fumes, etc.

If necessary, the combustion air duct can be directly connected to the turbocharger with a flexible connection piece. To protect the turbocharger a filter must be built into the air duct. The permissible pressure drop in the duct is max. 100 mm WC.

With these arrangements the normally required minimum air temperature to the main engine, see chapter *Operation ranges*, can typically be maintained. For lower temperatures special provisions are necessary. In special cases the duct can be connected directly to the turbocharger, with a stepless change-over flap to take the air from the engine room or from outside depending on engine load.

10.2.1 Charge air shut-off valve

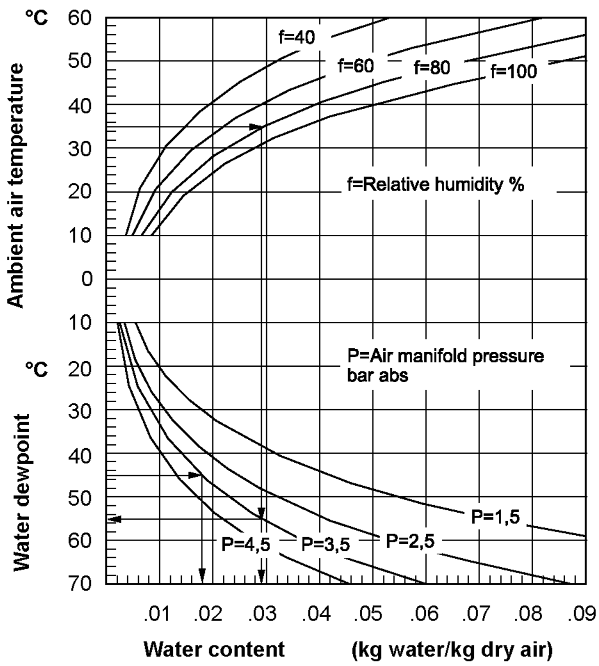
In installations where it is possible that the combustion air includes combustible gas or vapour the engines can be equipped with charge air shut-off valve. This is regulated mandatory where ingestion of flammable gas or fume is possible.

10.2.2 Condensation in charge air coolers

Example, according to the diagram:

At an ambient air temperature of 35°C and a relative humidity of 80%, the content of water in the air is 0.029 kg water/ kg dry air. If the air manifold pressure (receiver pressure) under these conditions is 2.5 bar (= 3.5 bar absolute), the dew point will be 55°C. If the air temperature in the air manifold is only 45°C, the air can only contain 0.018 kg/kg. The difference, 0.011 kg/kg (0.029 - 0.018) will appear as condensed water.

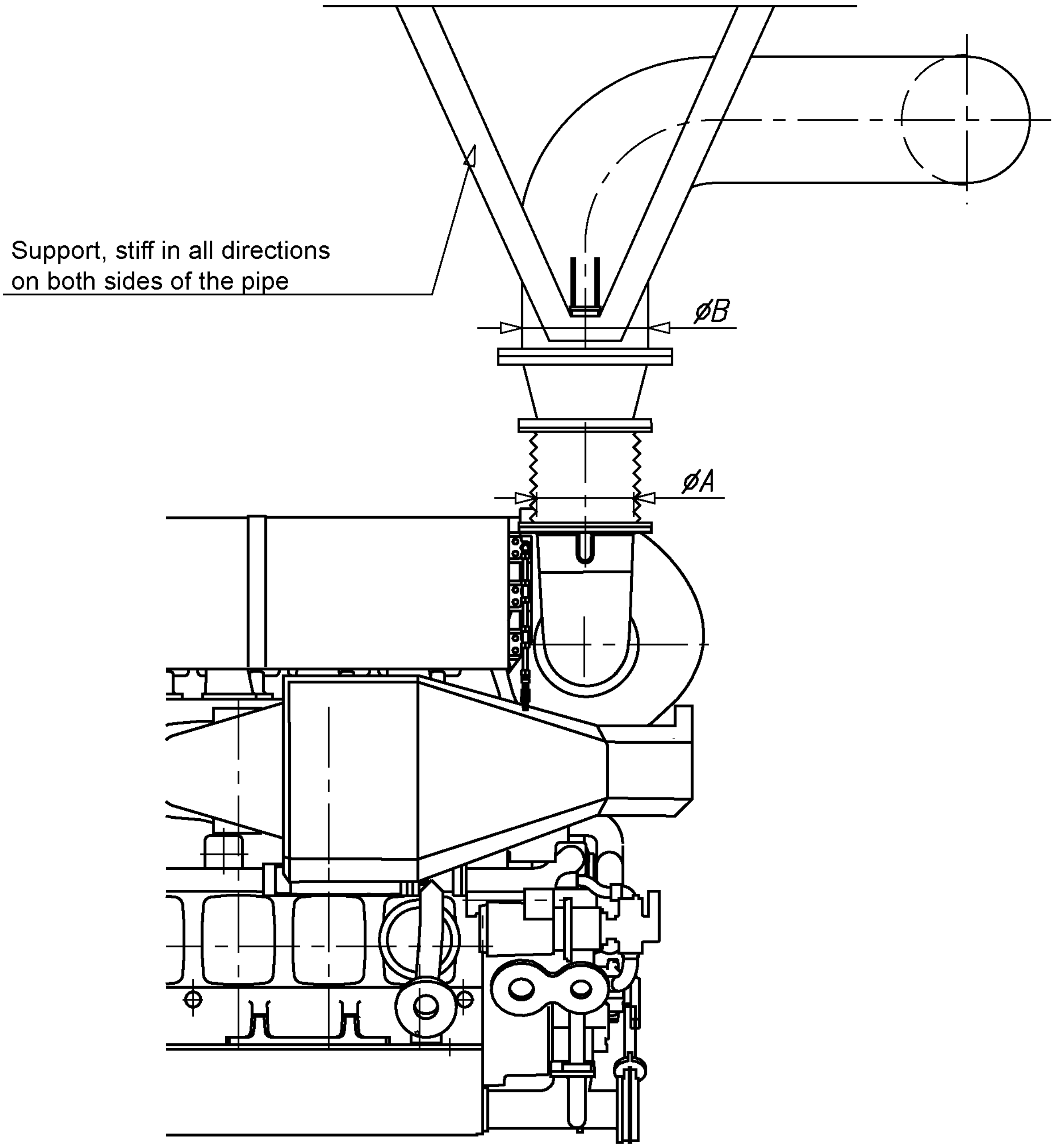
Figure 10.2 Condensation in charge air coolers



11. Exhaust gas system

11.1 Exhaust gas outlet

Figure 11.1 Exhaust gas outlet (4V76A2679)



Engine	Exhaust gas bellows inner diameter (ØA [mm])	Exhaust gas pipe inner diameter (ØB [mm])
4L20	200	300
6L20	250	350-400
8L20	300	400-450
9L20	300	450

The exhaust gas outlet from the turbocharger can be rotated to several positions, the positions depending on the number of cylinders. Other directions can be arranged by means of the adapter at the turbocharger outlet.

11.2 External exhaust gas system

11.2.1 General

Each engine should have its own exhaust pipe into open air.

Flexible bellows have to be mounted directly to the turbocharger outlet, to compensate for thermal expansion and prevent damages on the turbocharger due to vibrations. The pipe outside these bellows must be properly fixed. An exhaust gas ventilation system is installed in the engine room. Exhaust gas boilers (if any) and silencers are installed outside the engine room.

It is very important that the exhaust pipe is properly fixed to a support rigid in all directions directly after the bellows. There should be a fixing point on both sides of the pipe at the support. Absolutely rigid mounting between the pipe and the support is recommended at the first fixing point after the turbocharger. Resilient mounts can be accepted for resiliently mounted engines with long bellows, provided that the mounts are self-captive; maximum deflection at total failure being less than 2 mm radial and 4 mm axial with regards to the bellows. The natural frequencies of the mounting should be on a safe distance from the running speed, the firing frequency of the engine and the blade passing frequency of the propeller. The resilient mounts can be rubber mounts of conical type, or high damping stainless steel wire pads. Adequate thermal insulation must be provided to protect the rubber mounts from high temperatures. When using resilient mounting, the alignment of the exhaust bellows must be checked on a regular basis and corrected when necessary.

The piping should be as short and straight as possible.

The bends should be made with the largest possible bending radius, minimum radius used should be 1.5 D. The exhaust pipe must be insulated all the way from the turbocharger and the insulation is to be protected by a covering plate or similar to keep the insulation intact.Closest to the turbocharger the insulation should consist of a hook on padding to facilitate maintenance. It is especially important to prevent the airstream to the turbocharger detaching insulation, which will then clog the filters.

The exhaust gas pipes and/or silencers should be provided with water separating pockets and drainage.

Recommended flow velocity is 35...40 m/s. Lower velocities might be needed with long piping or if there are many resistance factors in the piping.

The exhaust gas mass flow given in chapter *Technical data* can be translated to velocity using the formula:

$$v = \frac{4 \times m}{1.3 \times \left(\frac{273}{273 + t}\right) \times \pi \times D^2}$$

Where:

v = gas velocity [m/s]

m = exhaust gas mass flow [kg/s]

t = exhaust gas temperature [°C]

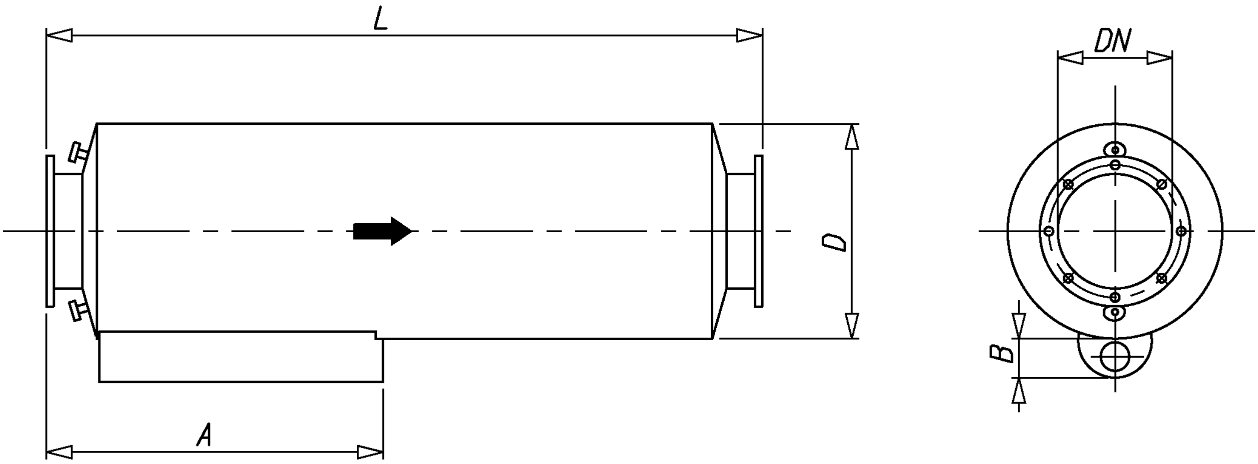
D = exhaust gas pipe diameter [m]

11.2.2 Exhaust gas silencer (5R02)

When included in the scope of supply, the standard silencer is of the absorption type, equipped with a spark arrester. It is also provided with a soot collector and a water drain, but is without mounting brackets and insulation. The silencer can be mounted either horizontally or vertically.

The noise attenuation of the standard silencer is either 25 or 35 dB(A).

Figure 11.2 Exhaust silencer (4V49E0137b)



Attenuation							
				25dB (A)		35 dB (A)	
DN	**D**	**A**	**B**	**L**	**Weight [kg]**	**L**	**Weight [kg]**
300	860	1250	150	2530	360	3530	455
350	950	1405	115	2780	440	3780	580
400	1060	1500	150	3280	570	4280	710
450	1200	1700	180	3430	685	4280	855
500	1200	1700	200	3430	690	4280	860

11.2.3 Exhaust gas boiler

If exhaust gas boilers are installed, each engine should have a separate exhaust gas boiler. Alternatively, a common boiler with separate gas sections for each engine is acceptable.

For dimensioning the boiler, the exhaust gas quantities and temperatures given in chapter *Technical data* may be used.

11.2.4 Exhaust gas bellows (5H01)

Bellows must be used in exhaust gas piping where thermal expansion or ship's structural deflections have to be segregated in order to limit stress levels.

11.2.5 Supporting

The number of mounting supports should always be kept to a minimum and positioned at stiffened locations within the ship's structure, e.g. decklevels, webframes or specially constructed supports.

The supporting must allow thermal expansion and ship's structural deflections during construction and operation.

11.2.6 Back pressure

The maximum permissible exhaust gas back pressure is 3 kPa (300 mm WC) at full load, which should be verified by a calculation, made by the shipyard. The back pressure should also be measured on the sea trial. A measuring connection should be provided on each exhaust piping system during the construction.

12. Turbocharger cleaning

12.1 Turbine cleaning system

Periodic water cleaning of the turbine reduces the formation of deposits and extends the interval between overhauls. Only fresh water should be used and the cleaning instructions in the operation manual must be carefully followed.

For washing of the turbine side of the turbocharger, fresh water with a pressure of not less than 300 kPa (3.0 bar) is required.

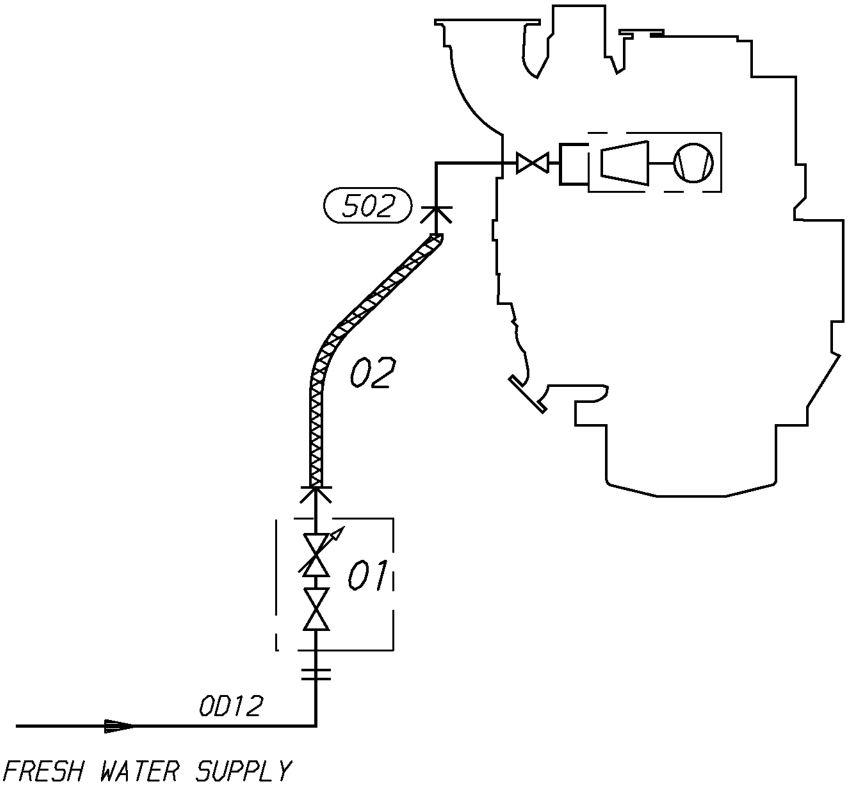
The washing is carried out during operation at regular intervals, depending on the quality of the heavy fuel, 100 – 500 h.

The water flow required for each turbine washing depends on the final turbocharger selection. The following typical values can be given (for guidance) for engines with a nominal speed of 900 or 1000 rpm:

4L20:	6 l/min
6L20:	8 l/min
8L20, 9L20:	10 l/min

The washing time is three times 30 seconds with 10 minutes between washings.

Figure 12.1 Turbocharger cleaning system (DAAE003884)



System components		Pipe connections		
01	Shut-off and flow adjusting unit	502	Cleaning water to turbine and compressor	Quick coupling
02	Rubber hose			

NOTE!

1. Shut-off and flow adjusting unit bulkhead mounted in the engineroom
2. Rubber hose length ~10 m

13. Exhaust emissions

13.1 General

Exhaust emissions from the diesel engine mainly consist of nitrogen, carbon dioxide (CO_2) and water vapour with smaller quantities of carbon monoxide (CO), sulphur oxides (SO_x) and nitrogen oxides (NO_x), partially reacted and non-combusted hydrocarbons and particulates. Emission control of large diesel engines means primarily the control of the NO_x emissions.

13.2 Diesel engine exhaust components

Due to the high efficiency of the diesel engines, the emissions of carbon dioxide (CO_2), carbon monoxide (CO) and hydrocarbons (HC) are low. The same high combustion temperatures that give thermal efficiency in the diesel engine also cause high emissions of nitrogen oxides (NO_x). The emissions of sulphur oxides (SO_x) and particulates are formed in the combustion process out of the sulphur, ash and asphaltenes that are always present in heavy fuel oil.

13.2.1 Nitrogen oxides (NO_x)

Nitric oxide (NO) and Nitrogen dioxide (NO_2) are usually grouped together as NO_x emissions. Predominant oxide of nitrogen found inside the diesel engine cylinder is NO, which forms mainly in the oxidation of atmospheric (molecular) nitrogen in the high temperature gas regions. NO can also be formed through oxidation of the nitrogen in fuel and through chemical reactions with fuel radicals. The amount of NO_2 emissions is approximately 5 %.

All standard Wärtsilä engines meet the NOx emission level set by the IMO (International Maritime Organisation) and most of the local emission levels without any modifications. Wärtsilä has also developed solutions to significantly reduce NOx emissions when this is required. For Wärtsilä 26, the Selective Catalytic Reduction (SCR) is an optional NOx reduction method.

13.2.2 Sulphur Oxides (SO_x)

Sulphur oxides (SO_x) are direct result of the sulphur content of the fuel oil. During the combustion process the fuel bound sulphur is rapidly oxidized to sulphur dioxide (SO_2). A small fraction of SO_2 may be further oxidized to sulphur trioxide (SO_3). The SO_x emission controls are directed mainly at limiting the sulphur content of the fuel.

13.2.3 Particulates

The particulate fraction of the exhaust emissions represents a complex mixture of inorganic and organic substances mainly comprising soot (elemental carbon), fuel oil ash (together with sulphates and associated water), nitrates, carbonates and a variety of non or partially combusted hydrocarbon components of the fuel and lubricating oil.

The main parameters affecting the particulate emissions are the fuel oil injection and fuel oil parameters. The use of fuel oil with good ignition and combustion properties and low content of ash and sulphur will reduce the formation of particulates. For marine diesel engines the particulate removal systems, because of their size and high cost, are not for the time being economically or practically potential solutions.

13.2.4 Smoke

Although smoke is usually the visible indication of particulates in the exhaust, the correlations between particulate emissions and smoke is not fixed. The lighter and more volatile hydrocarbons will not be visible nor will the particulates emitted from a well maintained and operated diesel engine.

Smoke can be black, blue, white, yellow or brown in appearance. Black smoke is mainly comprised of carbon particulates (soot). Blue smoke indicates the presence of the products of the incomplete combustion of the fuel or lubricating oil. White smoke is usually condensed water vapour. Yellow smoke is caused by NO_x emissions. When the exhaust gas is cooled significantly prior to discharge to the atmosphere, the condensed NO_2 component can have a brown appearance.

13.3 Marine exhaust emissions legislation

The increasing concern over the air pollution has resulted in the introduction of exhaust emission controls to the marine industry. To avoid the growth of uncoordinated regulations, the IMO (International Maritime Organization) has developed the Annex VI of MARPOL 73/78, which represents the first set of regulations on the marine exhaust emissions.

There is yet no legislation concerning the particulate emissions from the marine diesel engines, although the authorities are planning to set strict limits to the particulates in the near future. Smoke is regulated in some countries or regions based on its visibility.

13.3.1 MARPOL Annex VI

MARPOL 73/78 Annex VI includes regulations for example on such emissions as nitrogen oxides, sulphur oxides, volatile organic compounds and ozone depleting substances. The Annex VI will enter into force on the 19th of May 2005. The most important regulation of the MARPOL Annex VI is the control of NO_x emissions.

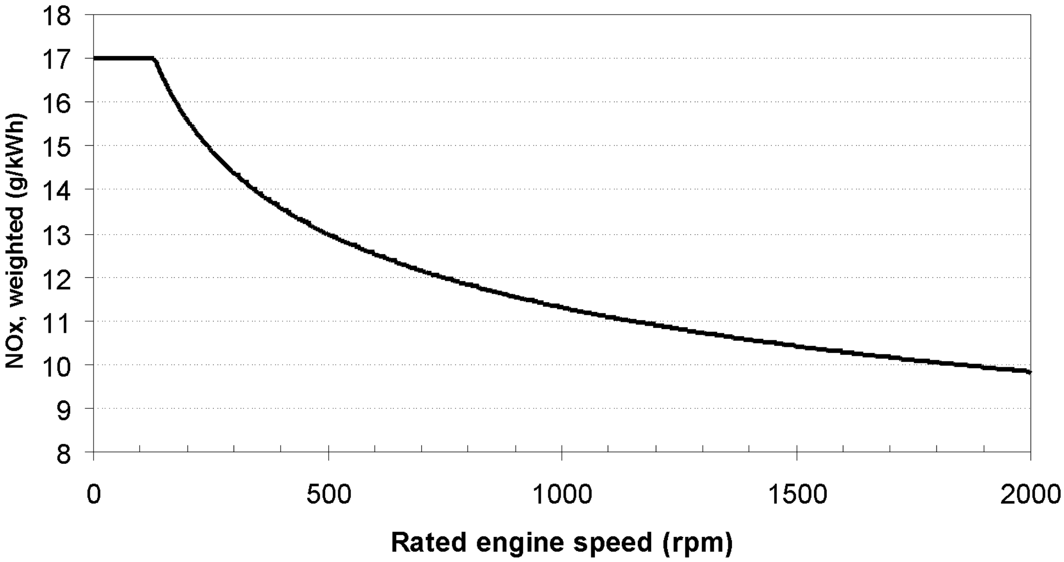
The engines comply with the NO_x levels set by the IMO in the MARPOL Annex VI. The NO_x controls apply to diesel engines over 130 kW installed on ships built (defined as date of keel laying or similar stage of construction) on or after January 1, 2000 along with engines which have undergone a major conversion on or after January 1, 2000.

For Wärtsilä 20 with a rated speed of 720 rpm, the NO_x level is below 12.1 g/kWh and with 750 rpm the level is below 12.0 g/kWh. With a rated speed of 900 rpm the NO_x level is below 11.5 g/kWh and with 1000 rpm the level is below 11.3 g/kWh. The tests are done according to IMO regulations (NO_x Technical Code).

The IMO NO_x limit is defined as follows:

NOx [g/kWh] = 17 when rpm < 130
= $45 \times \text{rpm}^{-0.2}$ when 130 < rpm < 2000
= 9.8 when rpm > 2000

Figure 13.1 IMO NO_x emission limit



13.3.2 EIAPP Certificate

An EIAPP (Engine International Air Pollution Prevention) certificate will be issued for each engine showing that the engine complies with the NO_x regulations set by the IMO.

When testing the engine for NO_x emissions, the reference fuel is Marine Diesel Fuel (distillate) and the test is performed according to ISO 8178 test cycles. Subsequently, the NO_x value has to be calculated using different weighting factors for different loads that have been corrected to ISO 8178 conditions. The most commonly used ISO 8178 test cycles are presented in the following table.

Table 13.1 ISO 8178 test cycles.

E2: Diesel electric propulsion, variable pitch	Speed (%)	100	100	100	100	
	Power (%)	100	75	50	25	
	Weighting factor	0.2	0.5	0.15	0.15	
E3: Propeller law	Speed (%)	100	91	80	63	
	Power (%)	100	75	50	25	
	Weighting factor	0.2	0.5	0.15	0.15	
D2: Auxiliary engine	Speed (%)	100	100	100	100	100
	Power (%)	100	75	50	25	10
	Weighting factor	0.05	0.3	0.3	0.3	0.1

For EIAPP certification, the “engine family” or the “engine group” concepts may be applied. This has been done for the Wärtsilä 20 diesel engine. The engine families are represented by their parent engines and the certification emission testing is only necessary for these parent engines. Further engines can be certified by checking documents, components, settings etc., which have to show correspondence with those of the parent engine.

All non-standard engines, for instance over-rated engines, non-standard-speed engines etc. have to be certified individually, i.e. “engine family” or “engine group” concepts do not apply.

According to the IMO regulations, a Technical File shall be made for each engine. This Technical File contains information about the components affecting NO_x emissions, and each critical component is marked with a special IMO number. Such critical components are injection nozzle, injection pump, camshaft, cylinder head, piston, connecting rod, charge air cooler and turbocharger. The allowable setting values and parameters for running the engine are also specified in the Technical File.

The marked components can later, on-board the ship, be identified by the surveyor and thus an IAPP (International Air Pollution Prevention) certificate for the ship can be issued on basis of the EIAPP certificate and the on-board inspection.

13.4 Methods to reduce exhaust emissions

Diesel engine exhaust emissions can be reduced either with primary or secondary methods. The primary methods limit the formation of specific emissions during the combustion process. The secondary methods reduce emission components after formation as they pass through the exhaust gas system.

13.4.1 Combustion Air Saturation System (CASS)

With the CASS (Combustion Air Saturation System) NO_x emissions can be reduced by 30-50%, depending on engine load and application.

Water is injected after the turbocharger, carried to the cylinders with the air and decreasing the maximum temperatures in the cylinders, thus reducing NO_x.

The CASS system includes a pump unit and a control unit outside the engine and group(s) of spraying nozzles with solenoid valves built on the charge air duct. The pump unit should be placed somewhere close to the engine.

Figure 13.2 System overview

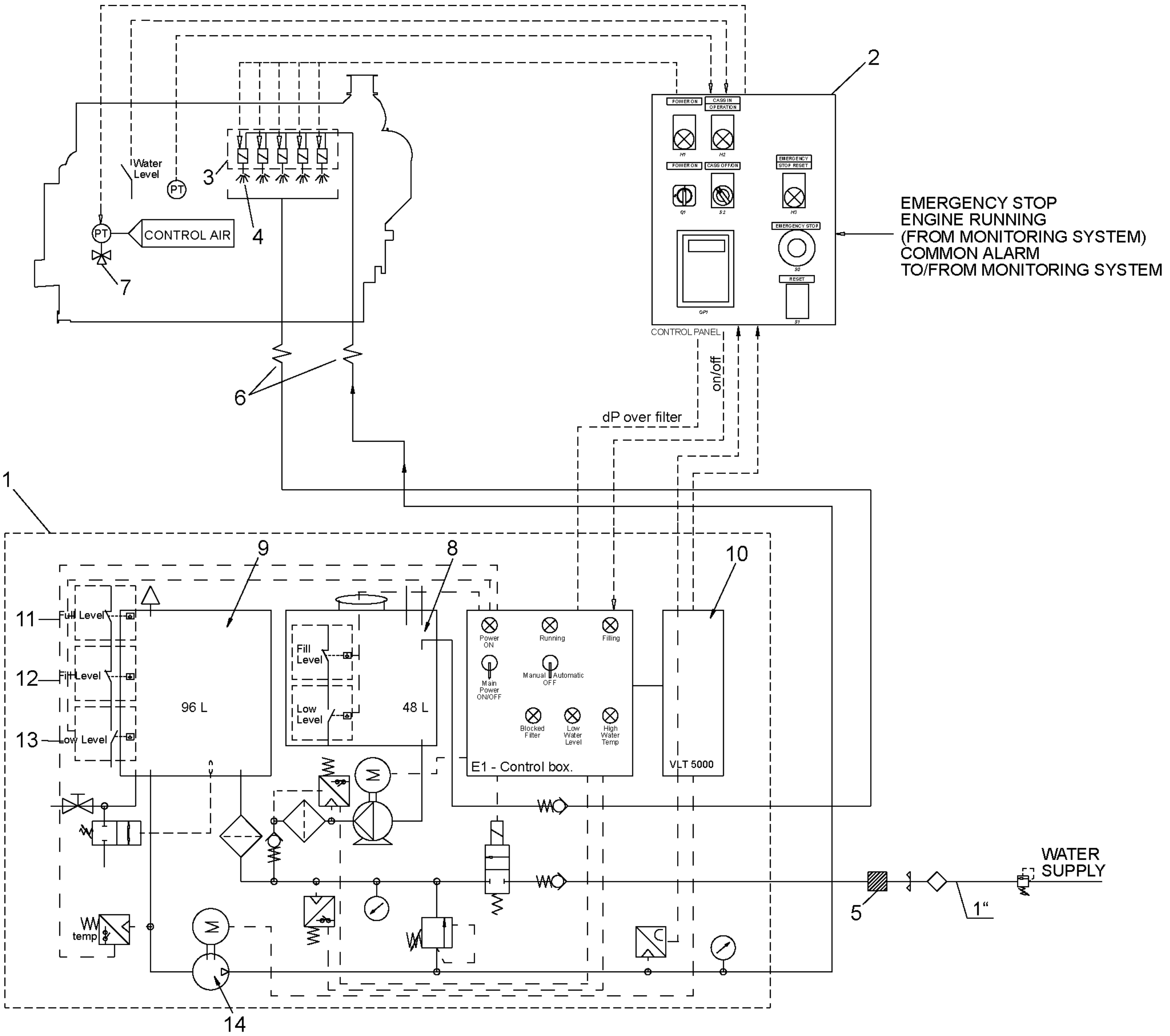


Table 13.2 Main components

1	High pressure unit	8	Recovery tank
2	CASS control panel	9	Water tank
3	Solenoid block	10	Control panel
4	Injector(s) & nozzles	11	Full level
5	Pre-filter	12	Fill level
6	Flexible hose	13	Low level
7	3-way valve	14	High pressure pump

The system is designed and built so that the amount of water injected can be adjusted and the CASS system can also be shut off if necessary. In case of failure the system will automatically go in off-mode.

The amount of water needed varies with the load and with the NO_X level the operator wants to achieve. Typically the water amount needed is 1.5...2 times the engine fuel oil consumption for full NO_X reduction. The water used for CASS has to be of very good quality, preferably evaporated, see table "*13.3 Requirements for CASS -water*".

Please note that when CASS is in operation the heat balance of the engine will change, always ask for project specific information.

Water quality used for CASS has to fulfill the following requirements:

Table 13.3 Requirements for CASS -water

Property	Unit	Maximum value
PH		6...8
Hardness	°dH	0.4
Chlorides as Cl	mg/l	5
Suspended solids	mg/l	5

In order to achieve a safe operation of the CASS system and the engine, water produced with a fresh water generator / distiller has to be used. The water must not be contaminated by oil, grease, surfactants or similar impurities. These kinds of impurities may cause blocking of the filters or other malfunctions in the CASS system.

Table 13.4 CASS water consumption

Engine type (900/1000 rpm)	Water consumption l/hours, at max NOx reduction and 85% load of the engine
Wärtsilä 4L20	280
Wärtsilä 6L20	420
Wärtsilä 8L20	560
Wärtsilä 9L20	630

13.4.2 Selective Catalytic Reduction (SCR)

Selective Catalytic Reduction (SCR) is the only way to reach a NO_x reduction level of 85-95%.

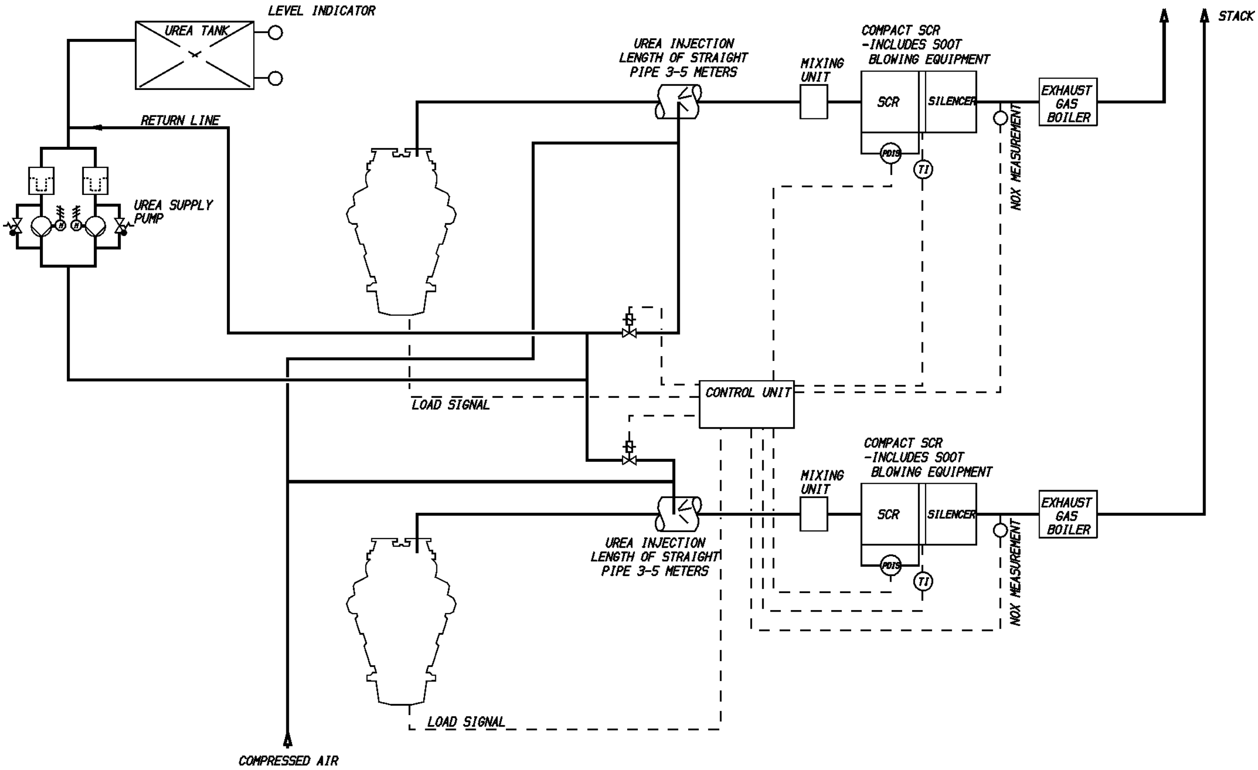
General system description

The reducing agent, aqueous solution of urea (40 wt-%), is injected into the exhaust gas directly after the turbocharger. Urea decays immediately to ammonia (NH_3) and carbon dioxide. The mixture is passed through the catalyst where NO_x is converted to harmless nitrogen and water, which are normally found in the air that we breathe. The catalyst elements are of honeycomb type and are typically of a ceramic structure with the active catalytic material spread over the catalyst surface.

The injection of urea is controlled by feedback from a NO_x measuring device after the catalyst. The rate of NO_x reduction depends on the amount of urea added, which can be expressed as NH_3/NO_x ratio. The increase of the catalyst volume can also increase the reduction rate.

When operating on HFO, the exhaust gas temperature before the SCR must be at least 330°C, depending on the sulphur content of the fuel. When operating on MDF, the exhaust gas temperature can be lower. If an exhaust gas boiler is specified, it should be installed after the SCR.

Figure 13.3 Typical P&ID for Compact SCR (3V28A0006b).



The disadvantages of the SCR are the large size and the relatively high installation and operation costs. To reduce the size, Wärtsilä has together with subsuppliers developed the Compact SCR, which is a combined silencer and SCR. The Compact SCR will require only a little more space than an ordinary silencer.

The lifetime of the catalyst is mainly dependent on the fuel oil quality and also to some extent on the lubricating oil quality. The lifetime of a catalyst is typically 3-5 years for liquid fuels and slightly longer if the engine is operating on gas. The total catalyst volume is usually divided into three layers of catalyst, and thus one layer at time can be replaced, and remaining activity in the older layers can be utilised.

Urea consumption and replacement of catalyst layers are generating the main running costs of the catalyst. The urea consumption is about 15 g/kWh of 40 wt-% urea. The urea solution can be prepared mixing urea granulates with water or the urea can be purchased as a 40 wt-% solution. The urea tank should be big enough for the ship to achieve the required autonomy.

14. Automation System

14.1 General

The engine automation system consists of local and remote control of the running parameters, local and remote monitoring of the sensors and automatic safety operations.

14.2 Power supply

The power requirement of the automation system is about 150 W (24 V DC).

14.3 Safety System

14.3.1 General

The safety system can be split into five major parts: starting, stopping, start blocking, shutdowns and load reduction requests.

14.3.2 Starting (8N08)

Start of an auxiliary engine

See figure 14.7 Start stop system for an auxiliary engine (DAAE011244-1a) showing a principle diagram of a start/stop system.

The described relay automation is usually not included in the scope of supply, but can as an option be supplied in a separate cabinet. The control, safety and sequencing functions of this system can also be incorporated in the power management or automation system of the vessel.

Start sequence

The engine is equipped with a pneumatic starting motor, which drives the engine through a gear rim on the flywheel. The starting motor is controlled by a solenoid valve. The engine can be started by activating the solenoid valve, locally by a start button on the engine or remotely e.g. from the diesel automation system.

Agenerating set reaches the nominal speed in 6...8 seconds after the starting solenoid has been activated.

Start blocking (8N08)

Starting is inhibited by the following functions:

- Turning device engaged
- Prelubricating pressure low. In case of black-out, starting is allowed within 30 minutes after the pressure has dropped below the set point of 0.5 bar.
- Engine start blocking selector switch turned into “Blocked” position
- Engine running (300 rpm)
- External start blocking activated
- Stop signal to engine activated (safety shut-down, emergency stop, normal stop)
- Stop lever in stop position

In an emergency case, the engine can always be started by manually operating the main starting valve. This by-passes start blocking due to low prelubricating pressure.

Starting air cut off

The start signal is cut off by:

- Speed switch in SPEMOS (115 rpm)
- Any start blocking activated

Start fuel limiter

The speed governor is provided with a start fuel limiter function in order to optimize the fuel injection during the acceleration period. This is controlled by a speed switch in the speed measuring system.

Override of lubricating oil pressure shut-down

To enable start of the engine, the automatic shut-down for low lubricating oil pressure is disabled during the starting sequence. This is done using the “engine running” contact. Further, a time delay of about 10 seconds is arranged in order to allow the engine driven lubricating oil pump to establish sufficient pressure.

Start of a main engine

The principle diagram of a start/stop system is shown in drawing *See figure 14.9 Start/stop system for a main engine (DAAE003936-1b)*.

A main engine can be started and stopped in the same way as an auxiliary engine. Some minor differences should, however, be noted:

- The 30 min. time delay of prelubricating pressure described for black-out start is not recommended.
- For some installations start blocking may be required from clutch position, pitch NOT zero and reduction gear lubricating oil pressure. Autostop of the engine can also be required for low lubricating oil pressure in the reduction gear.

14.3.3 Stopping (8N08)

Normal stop of the engine

The engine is stopped remotely via the ‘remote stop’ input or in local control by the stop button on the engine.

Manual stop can also be done by turning the stop lever into the stop position.

There are two stop solenoids on the engine. One is built into the speed governor. The other one is controlling compressed air, which is fed to pneumatic cylinders at each fuel injection pump, forcing the pumps to no-fuel when activated. This system is independent of the governor. The engine can be stopped by activating one or both of the solenoids for at least 60 seconds. Emergency and safety shut-down should activate both.

When two engines are connected to a common reduction gear it is recommended that the clutches are blocked in the “OUT” position when the engine is not running. When an engine is stopped, the clutch should open to prevent the engine from being driven through the gear. At an overspeed shutdown signal the clutch should remain closed.

‘Engine stop/shutdown' output contact is always closed when the stop signal is active.

14.3.4 Shutdowns (8N08)

The engine shall be automatically shut down in the following cases:

- Lubricating oil pressure low (pressure switch)
- Cooling water temperature high (temperature switch)
- Overspeed (speed switch in SPEMOS)

The shutdown is latching, and a shutdown reset has to be given before it is possible to re-start. Naturally, before this the reason of the shutdown must be investigated.

For a single main engine installation a shutdown override input is available. This gives the possibility to override auto-stop-signals (except for overspeed) from the bridge and prevent the engine from stopping in a critical manoeuvring situation.

Overspeed protection

A main engine is equipped with two independently adjustable switches for overspeed.

- The lower speed switch with the set point (nom. rpm + 15%).
- The higher speed switch with the set point (nom. rpm + 18%).

Both the lower and the higher speedswitches keeps a latching function in order to ensure shut-down of the engine.

14.4 Speed Measuring (8N03)

An electronic speed measuring and monitoring system (SPEMOS) is built into the engine junction box.

The system monitors the engine speed with two pick-ups and the turbocharger speed with a single pick-up. A 24 V DC power supply is required for the SPEMOS.

Table 14.1 Speed measuring & monitoring system signals

Function	Signal	Usage	Remark
Start fuel limiter	contact	internal	100 rpm below idling speed
Starting air cut off	contact	internal	115 rpm
Engine running	volt. free contact	external	300 rpm
Overspeed	volt. free contact	external	nominal + 15% speed
Engine speed	0 - 10 V DC	internal & external	0 - 1500 rpm
Turbocharger speed	0 - 10 V DC	internal & external	0 - 60000 rpm
Power/tacho failure	volt. free contact	external	

14.5 Sensors & signals

Figures 14.2-6 Typical wiring diagram (4V50L5692-1b) shows a typical engine wiring diagram with a standard set of sensors for monitoring, alarm and safety.

Table 14.2 Standard sensors for remote monitoring and alarm

Sensor	Code	Signal	Alarm	Remark
Fuel oil pressure	PT101	4 - 20 mA	0.4 MPa (4 bar)	HFO
Lubricating oil pressure	PT201	4 - 20 mA	0.3 MPa (3 bar)	
Starting air pressure	PT301	4 - 20 mA	0.8 MPa (8 bar)	
HT water pressure	PT401	4 - 20 mA	0.2 MPa (2 bar)	installation specific
LT water pressure	PT451	4 - 20 mA	0.2 MPa (2 bar)	installation specific
Exhaust gas temperature after each cylinder and turbocharger	TE501	4 - 20 mA	500°C	NiCrNi + amplifier
Lubricating oil temperature	TE201	Pt 100	80°C	
HT water temperature	TE402	Pt 100	105°C	
Charge air temperature	TE622	Pt 100	75°C	
Injection pipe leakage	LS103A	volt. free contact	—	
Lubricating oil level in wet oil sump low	LS204	volt. free contact	—	
Lubricating oil filter pressure drop	PDS243	volt. free contact	0.15 MPa (1.5 bar)	
Pneumatic overspeed trip pressure low	PS311	volt. free contact	1.8 MPa (18 bar)	
Overload	GS166	volt. free contact	—	main engines only

Table 14.3 Standard sensors/signals for engine safety

Sensor	Code	Signal	Set point	Remark
Lubricating oil pressure	PSZ201	volt. free contact	0.2 MPa (2 bar)	shut-down*
Cooling water temperature high	TSZ402	volt. free contact	110°C	shut-down*
Overspeed	SSZ173	volt. free contact	nom. rpm + 15%	shut-down
Turning gear enganged	GS792	volt. free contact		start block
Prelubricating pressure low	PS201-1	volt. free contact	50 kPa (0.5 bar)	start block
Selector switch blocked, local, remote	HS724	volt. free contact		start block

** for DNV single main engine only load reduction*

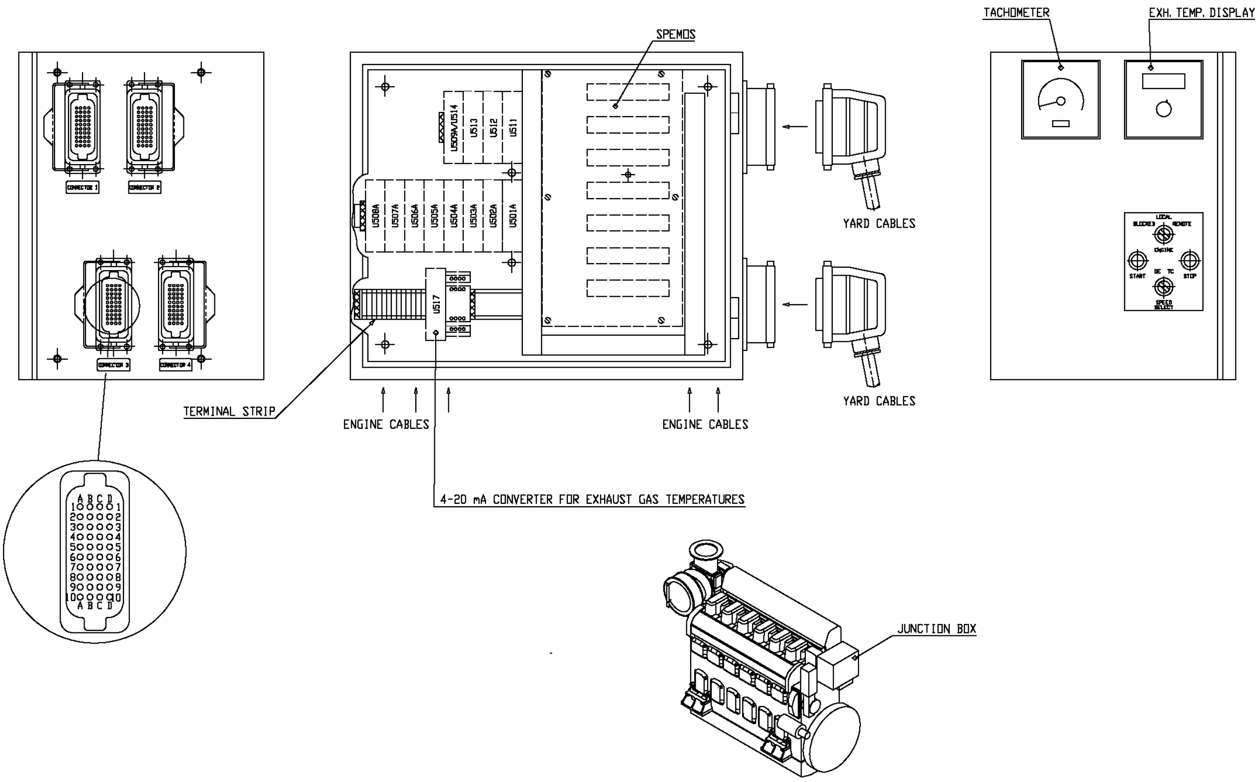
The analogue temperature sensors are of type Pt-100. (100 Ω at 0°C).

The exhaust gas temperature sensors (NiCr/Ni thermocouples) are connected to converters with an output signal of 4 - 20 mA, two wire connection.

The analogue pressure transmitters have an output signal of 4 - 20 mA, two wire connection.

All sensors are wired to a common junction box, see *figure 14.1 Junction box with multi pin connectors (1V50L5433c)*, on the engine. Most sensor cables plug into connection rails located on either side of the engine. Multicore cables connect the rails to the junction box. Cables furnished by the yard for monitoring, alarm and control shall be connected to the junction box with multi pin connectors.

Figure 14.1 Junction box with multi pin connectors (1V50L5433c)



14.6 Local instrumentation

The engine is equipped with the following set of instruments for local reading of pressures, temperatures and other parameters.

14.6.1 Pressure gauges in panel on engine

- Lubricating oil pressure
- Fuel oil pressure
- Cooling water pressure (HT)
- Cooling water pressure (LT)
- Charge air pressure
- Starting air pressure

14.6.2 Thermometers

- Fuel oil before engine
- Lubricating oil before lubricating oil cooler
- Lubricating oil after lubricating oil cooler

- Cooling water (HT) before engine
- Cooling water (HT) after engine
- Cooling water (LT) before charge air cooler
- Cooling water (LT) after charge air cooler
- Cooling water (LT) after lubricating oil cooler
- Charge air after charge air cooler

14.6.3 Electrical instruments

- Tachometer for engine speed and turbocharger speed
- Running hour counter
- Digital display for exhaust gas temperature with selector switch for temperature after each cylinder and after turbocharger

14.7 Control of auxiliary equipment

14.7.1 Stand-by pumps

Stand-by pumps are required for single main engine.

If the pressure drops below a pre-set level when the engine is running, the stand-by pump should be started. The stand-by pump starter shall include an interposing relay controlling the main contactor.

Latching must be done in the standby starter and alarm system respectively. The reason for the pressure drop should be investigated as soon as possible.

Stop of the standby pump should always be a manual operation. Before stopping the standby pump, the reason for the pressure drop must have been investigated and rectified.

Monitoring signals can be used to initiate the start of stand-by pumps.

14.7.2 Pre-lubricating oil pump (9N03)

The engine is equipped with an electric pre-lubricating pump.

The pre-lubricating pump is used for filling of the lubricating oil system, pre-lubricate a stopped engine before start and for preheating by circulating warm lubricating oil. The colder the engine is, the earlier the pump should be started before the engine is started.

The pump may also be run continuously when the engine is stopped and must run in multiple engine installations when other engines are running.

For continuous pre-lubrication of a stopped engine, through which heavy fuel is circulating.

To ensure continuous pre-lubrication of a stopped engine, automatic starting and stopping of the pre-lubricating pump is recommended. This can be achieved using the 300 rpm speed switch.

For dimensioning the pre-lubricating pump starter, the values indicated below can be used. For different voltages, the values may differ slightly. The starter is not included in the standard delivery of the engine.

- 400V / 50Hz 2.2kW, In = 4.7A
- 440V / 60Hz 2.5kW, In = 4.7A

14.7.3 Cooling water pre-heater & circulation pump

In order to get the engine up to and maintain a cooling water temperature > 70°C, pre-heating has to be arranged for the engine. Pre-heating is preferably done by an electric pre-heater with a required heating power depending of the engine type. The pre-heater is not included in the standard delivery of the engine.

The temperature control should be automatic.

For automatic starting and stopping of the circulating pump to circulate cooling water through the stopped engine(s), the ‘Engine running’ signal can be used as reference.

In some main engine installations the circulating pump needs to be running at prolonged idling. For these cases special instructions are given.

14.8 Speed control (8I03)

14.8.1 Main engine speed control

Mechanical-hydraulic governors

The engines have hydraulic-mechanical governors with pneumatic speed setting. These governors are usually provided with a shut-down solenoid as the only electrical equipment.

The idling speed is selected for each installation based on calculations, for CP-propeller installations at 55 - 65% of the nominal speed and for FP-propeller installations at about 35%.

The standard control air pressure for pneumatically controlled governors is:

$$p = 0.5 * n$$

p = control air pressure [kPa]

n = engine speed [rpm]

Governors for engines in FP-propeller installations are provided with a smoke limiting function, which limits the fuel injection as a function of the charge air pressure.

Governors for engines connected to a common reduction gear are specially adapted and adjusted for the same speed droop, normally about 4%, to obtain basic load sharing. In addition, it is recommended to arrange external load sharing based on the fuel rack position transducer.

Governors are, as standard, equipped with a built-in delay of the speed change rate so that the time for speed acceleration from idle to rated speed and vice versa is preset.

In special cases speed governors of the electronic type can be used.

14.8.2 Generating set speed control

Mechanical-hydraulic governors

Auxiliary generator sets are normally provided with mechanical- hydraulic governors for remote electric speed setting from e.g. a Power Management System (PMS).

The governor is equipped with a speed setting motor for synchronizing, load sharing and frequency control.

The governor is also equipped with a shutdown solenoid and an electrically controlled start fuel limiter. The synchronizing is operated by ON/OFF control as “increase” or “decrease” by polarity switching. Normal speed change rate is about 0.3 Hz/s.

Engines, which are to be run in parallel have governors specially adapted for the same speed droop, about 4%, to obtain basic load sharing. During load sharing and frequency control, the external load sharing system (PMS) must have a control deadband implemented, allowing for an uneven load or frequency drift of 1 - 2%.

14.8.3 Electronic speed governor

An Electronic speed control, comprising a separately mounted electronic speed control unit and a built-on actuator, offers efficient tools for filtering speed and load signals. This is often required in order to achieve good stability without sacrificing the transient response. Further the dynamic response can be adjusted and optimised for the particular installation, or even for different operating modes of the same engine. An electronic speed control is also capable of isochronous load sharing. In isochronous mode, there is no need for external load sharing, frequency adjustment, or engine loading/unloading control in the external control system. Both isochronous load sharing and traditional speed droop are standard features in all electronic speed controllers and either mode can be selected.

Speed droop means that the governor speed reference automatically decreases as the engine load increases. The speed droop is normally adjusted to about 4%. This is to ensure proper load sharing between parallelling units. To compensate for the speed decrease of the plant when the load increases, and vice versa when the load decreases, the PMS must in an outer (cascade) loop correct for the frequency drift.

Isochronous load sharing means that the governor speed reference stays the same, regardless of the load level. A shielded twisted pair cable between the speed controllers is necessary for isochronous load sharing.

If the ship has two or more switchboard sections, which can be either connected or separated, there must be a breaker also for the load sharing lines between each speed control.

Electronic speed control for Main Engines

An electronic speed control is recommended for more demanding installations, e.g. main engine installations with two engines connected to the same reduction gear, in particular if there is a shaft generator on the reduction gear.

The remote speed setting can be either an increase/decrease signal, or an analog 4-20 mA speed reference, both from e.g. a propulsion control system. The rate at which the speed changes is adjustable in the speed controller.

Actuators with mechanical backup are only recommended for single main engines. The actuator should in case of a single main engine be reverse acting, so that the change over to the mechanical backup takes place automatically.

Electronic speed control for Diesel electric/Generator set

An electronic speed control is always recommended for diesel electric installations due to the sometimes strongly fluctuating power demand from the dominant consumer (propulsion).

For an auxiliary generating set, an electronic speed control can be specified as an option.

Actuators with mechanical backup are not recommended for multi-engine installations.

14.9 Microprocessor based engine control system (WECS) (8N01)

As an alternative to the conventional way of cabling the sensor signals wire by wire from the engine to the external alarm, monitoring and control systems, an Engine Control System (WECS) can be provided. The WECS is a microprocessor based monitoring and control system.

14.9.1 Components

The system for one engine consists of one main control unit (MCU) and several distributed control units (DCU) or sensor multiplexer units (SMU) depending on the amount of sensors which are connected to the DCU:s and SMU:s. The SMU is used only to collect sensor data, while the DCU also is used for distributing processing power from the MCU.

14.9.2 Functions of the MCU

The MCU collects measuring data from the sensors on the engine, converts the information into digital form and communicates by serial link with the external monitoring, alarm and control systems. The MCU also handles the functions described in the previous paragraphs, i.e. speed measuring, start/stop sequences and automatic shut-downs. Vital functions, such as lubricating oil pressure and overspeed shut-down, are also handled by external switches independent of the MCU.

14.9.3 Communications

By having field bus connection via multi-standard RS-ports for communication with external systems, cabling work furnished by the yard will be minimized. In addition, installation work, service and maintenance will be easier.

The RS-interface is 4-wire RS485. The communication follows the Modbus RTU protocol specifications with the MCU as a slave in the Modbus network. A typical setup is a monitoring and alarm system functioning as a master and each MCU (one per engine) functioning as slaves.

14.10 Typical wiring diagrams

Figure 14.2 Typical wiring diagram (4V50L5692-1b)

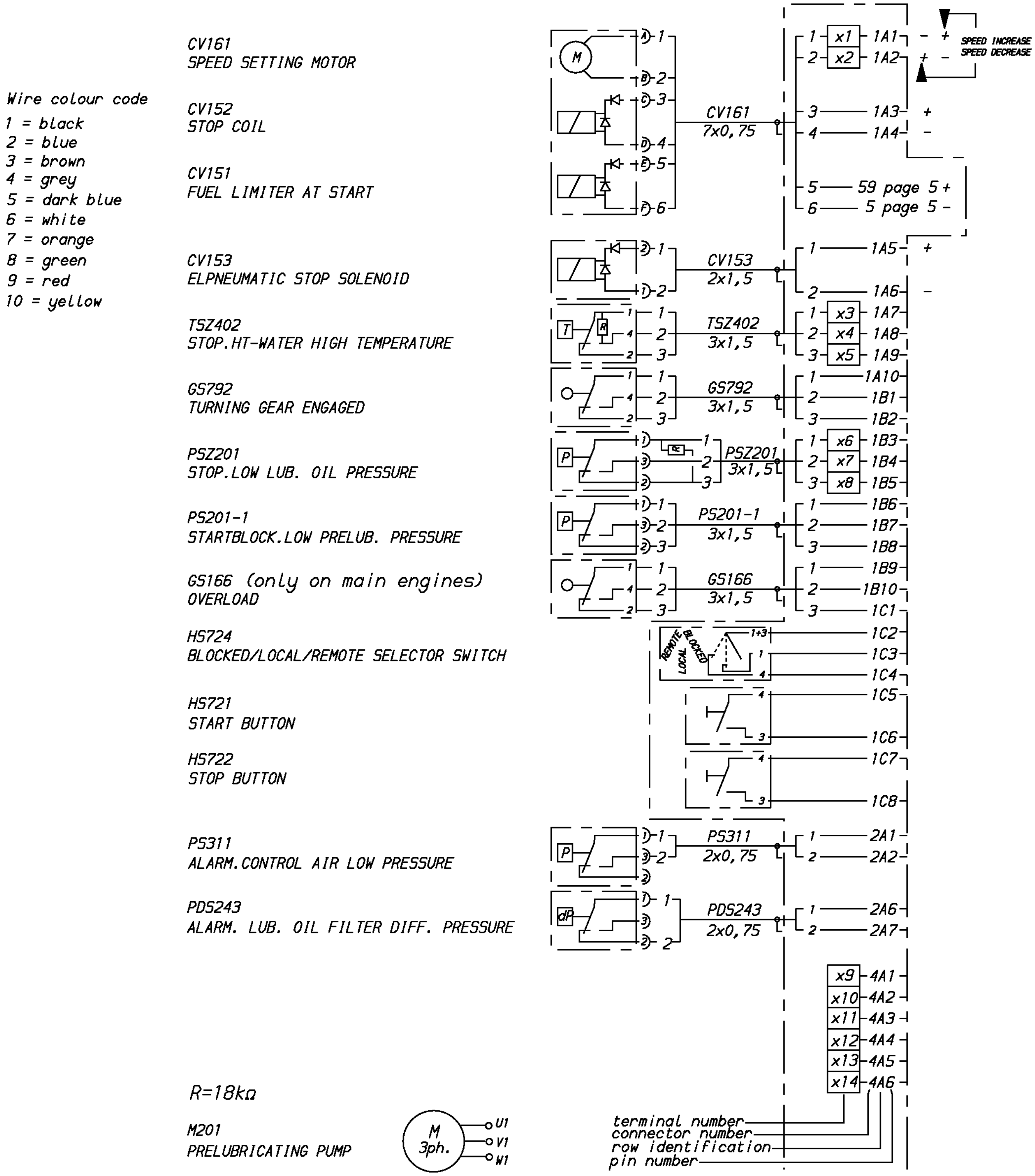


Figure 14.3 Typical wiring diagram (4V50L5692-2b)

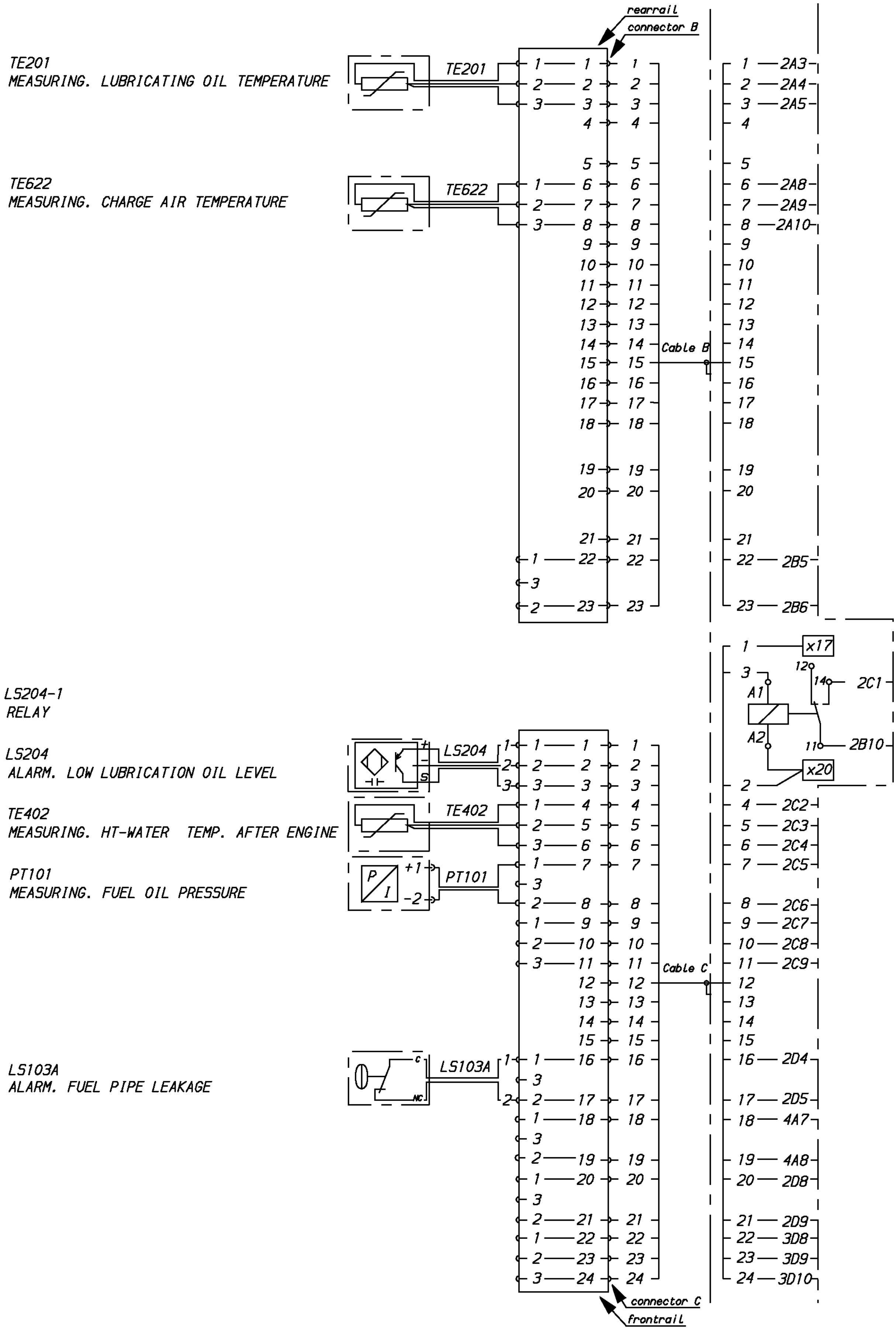


Figure 14.4 Typical wiring diagram (4V50L5692-3b)

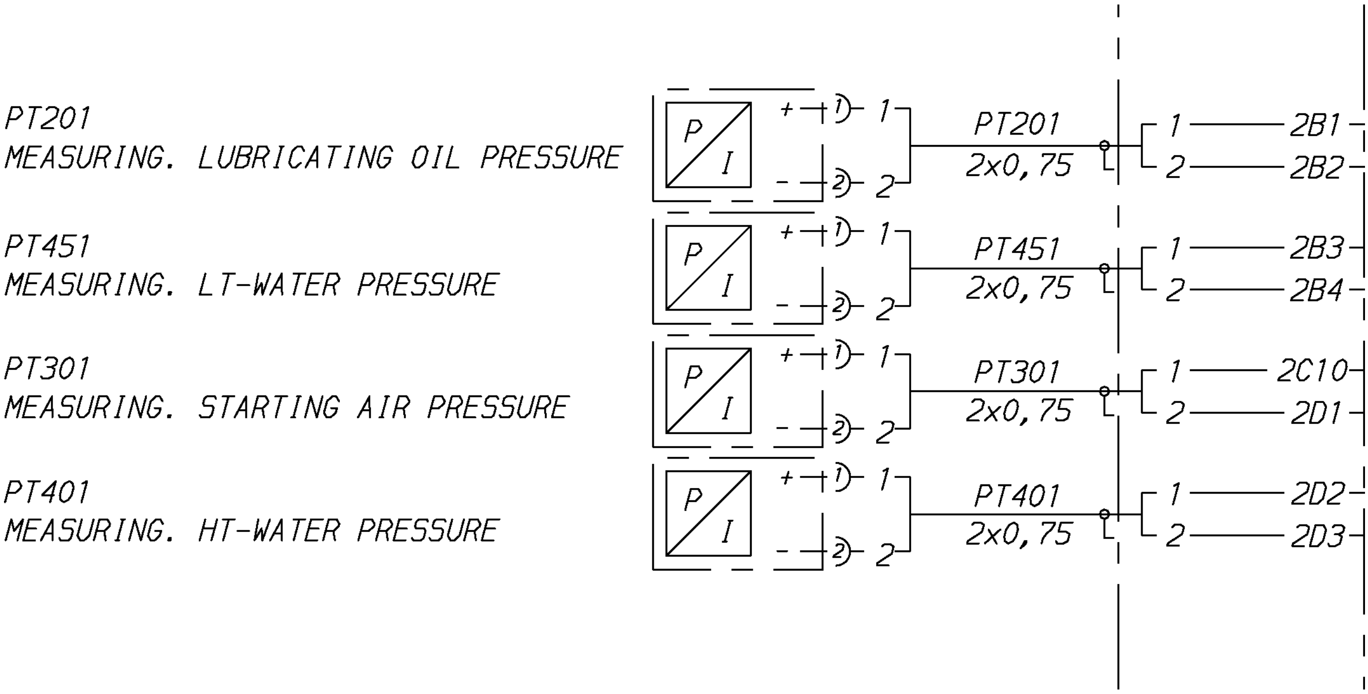


Figure 14.5 Typical wiring diagram (4V50L5692-4b)

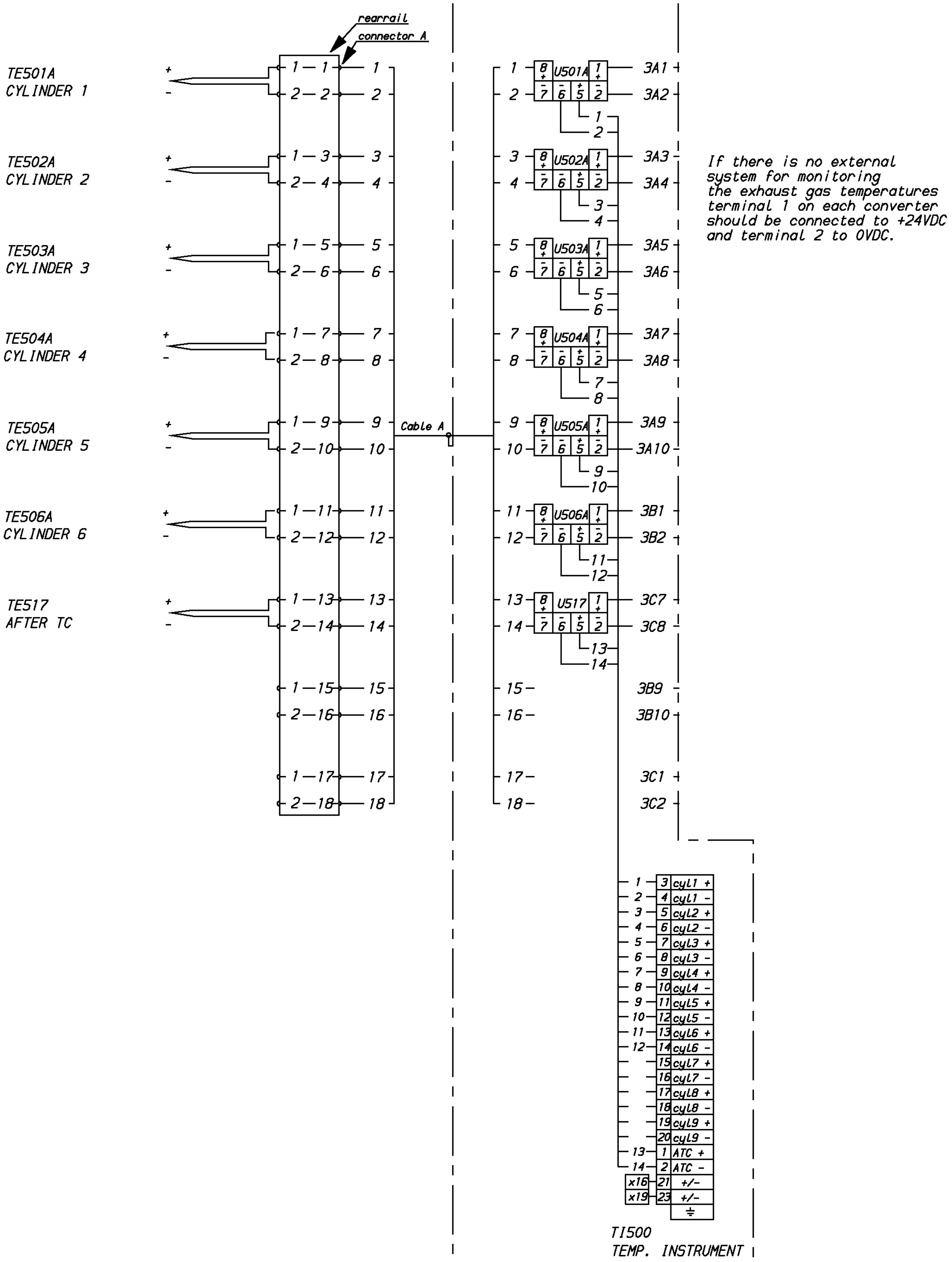


Figure 14.6 Typical wiring diagram (4V50L5692-5b)

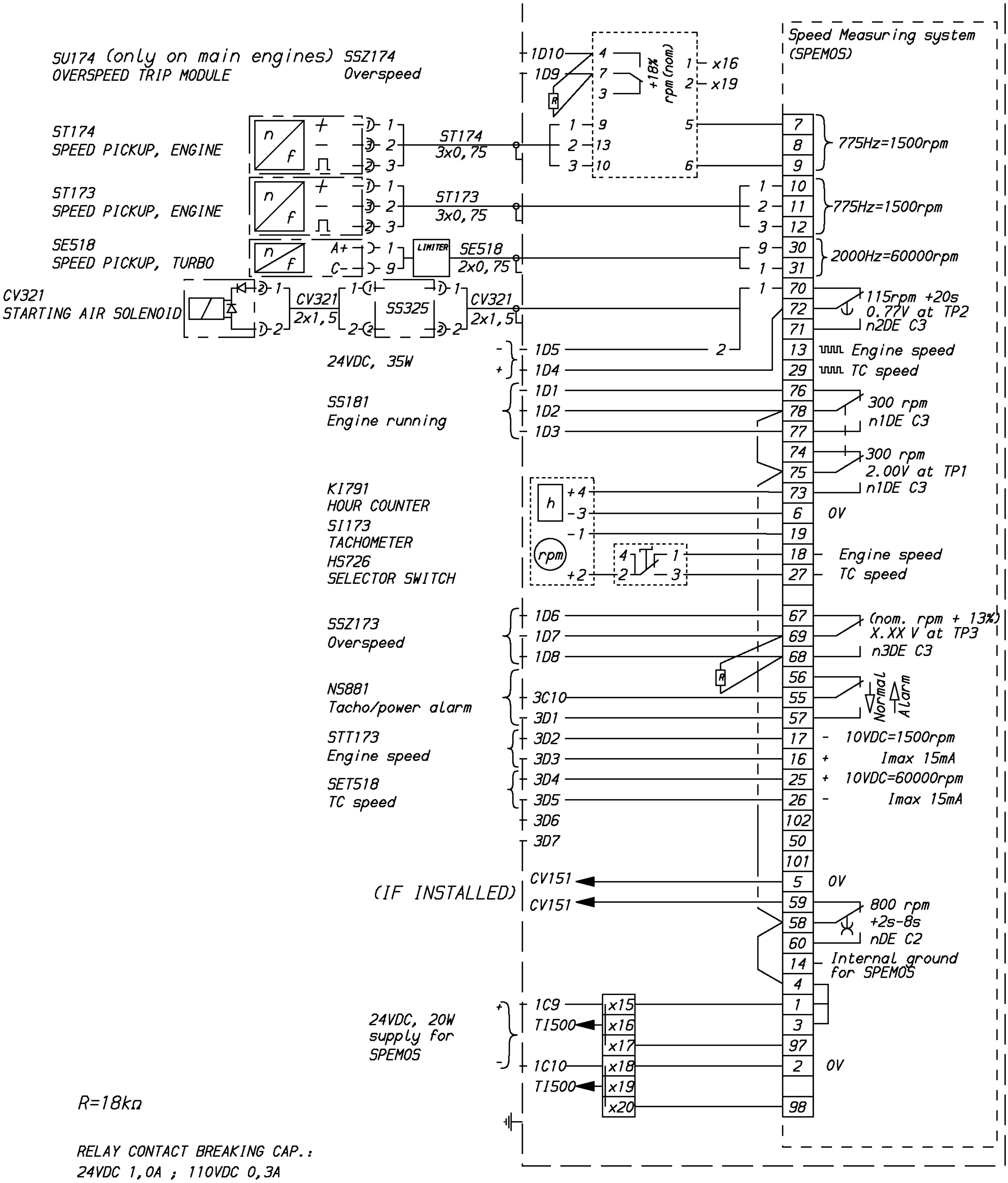
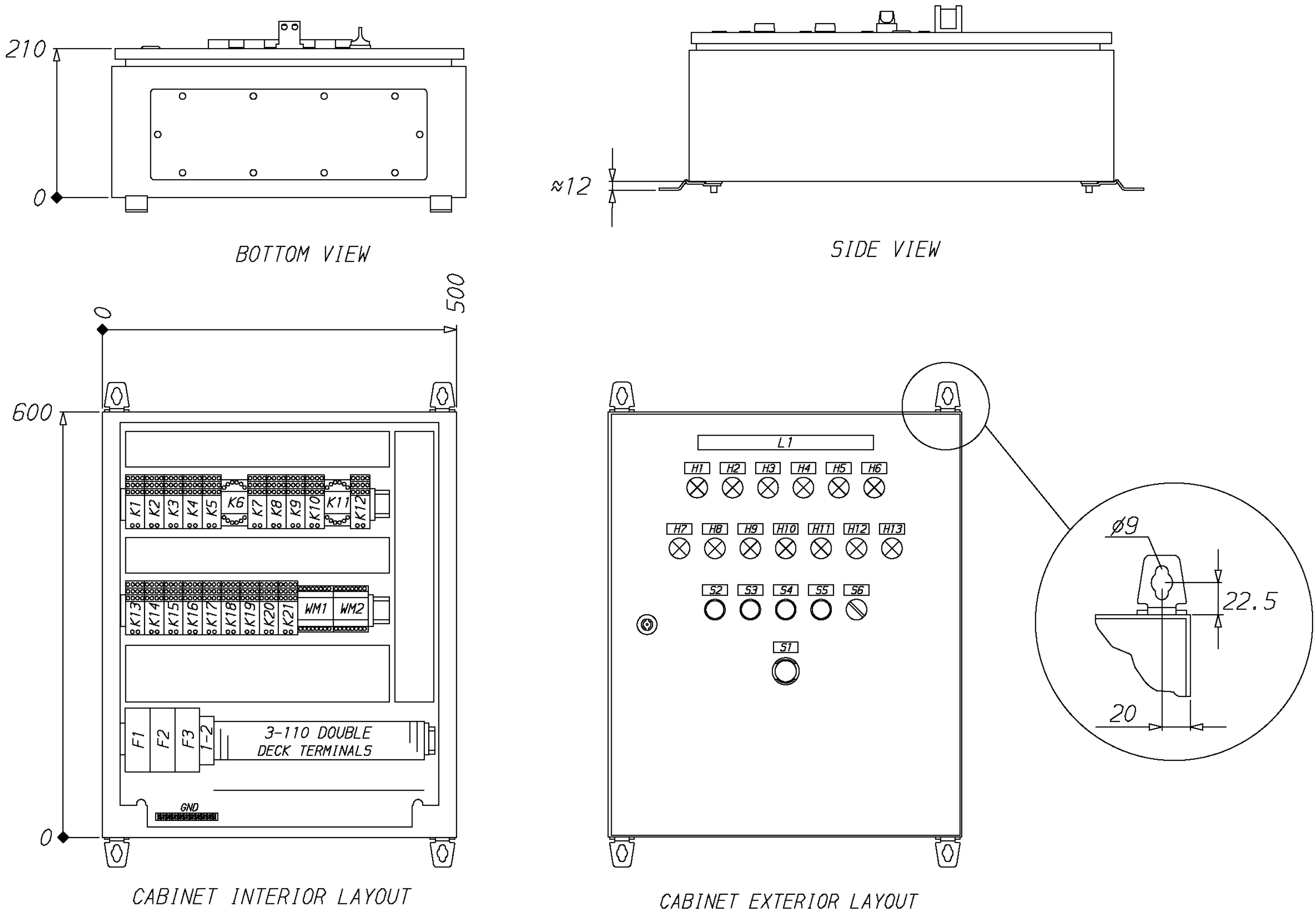


Figure 14.7 Start stop system for an auxiliary engine (DAAE011244-1a)



Indication lamps + Labels:		Colors:	Indication lamps + Labels:		Colors:
H1	Power on	White	H8	Start failure	Red
H2	Control on engine	White	H9	HT-water temperature high	Red
H3	Ready for start	White	H10	Lubrication oil pressure low	Red
H4	Engine running	Green	H11	Overspeed	Red
H5	Turning gear engaged	Orange	H12	Remote emergency stop	Red
H6	Pre-lubricating oil pressure low	Orange	H13	External shutdown	Red
H7	External start blocking	Orange			

Pushbuttons & Selectors + Labels:		
S1	Emergency stop	Degree of protection: IP44
S2	Stop	Colour: RAL7035
S3	Start	
S4	Reset	
S5	Lamp test	
S6	Local/Remote mode of control	

Figure 14.8 Start/stop system for an auxiliary engine (DAAE012244-7a)

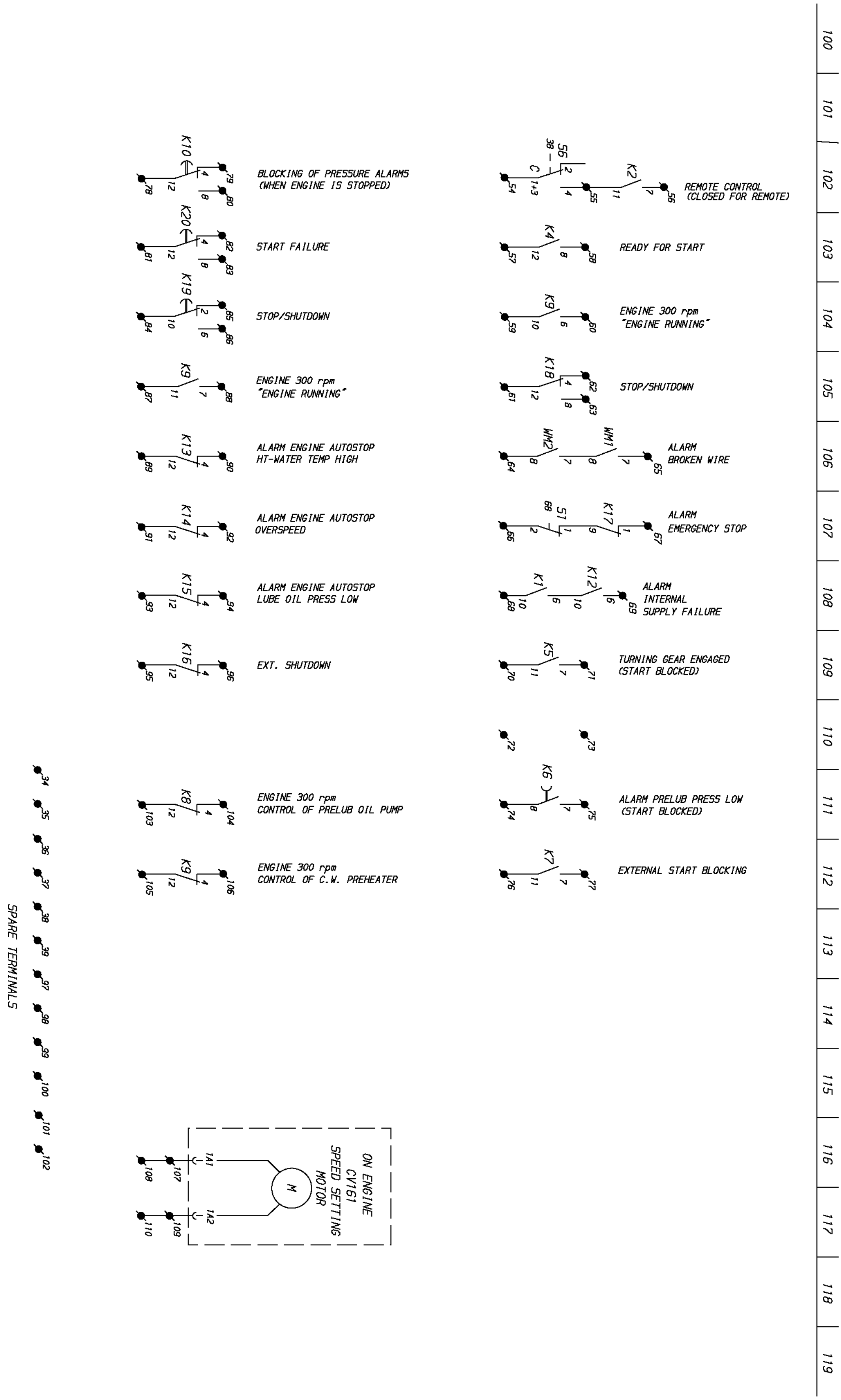
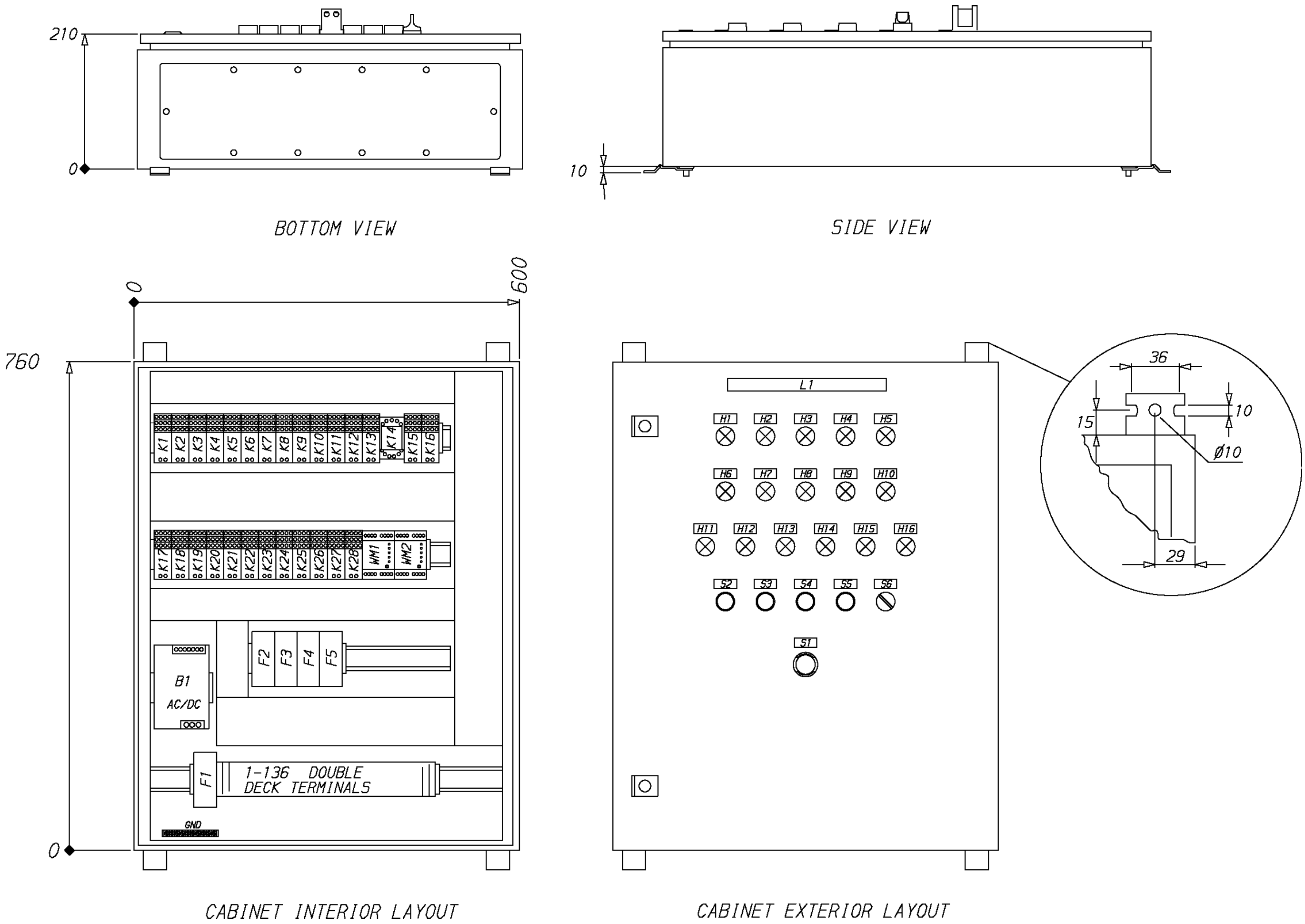


Figure 14.9 Start/stop system for a main engine (DAAE003936-1b)

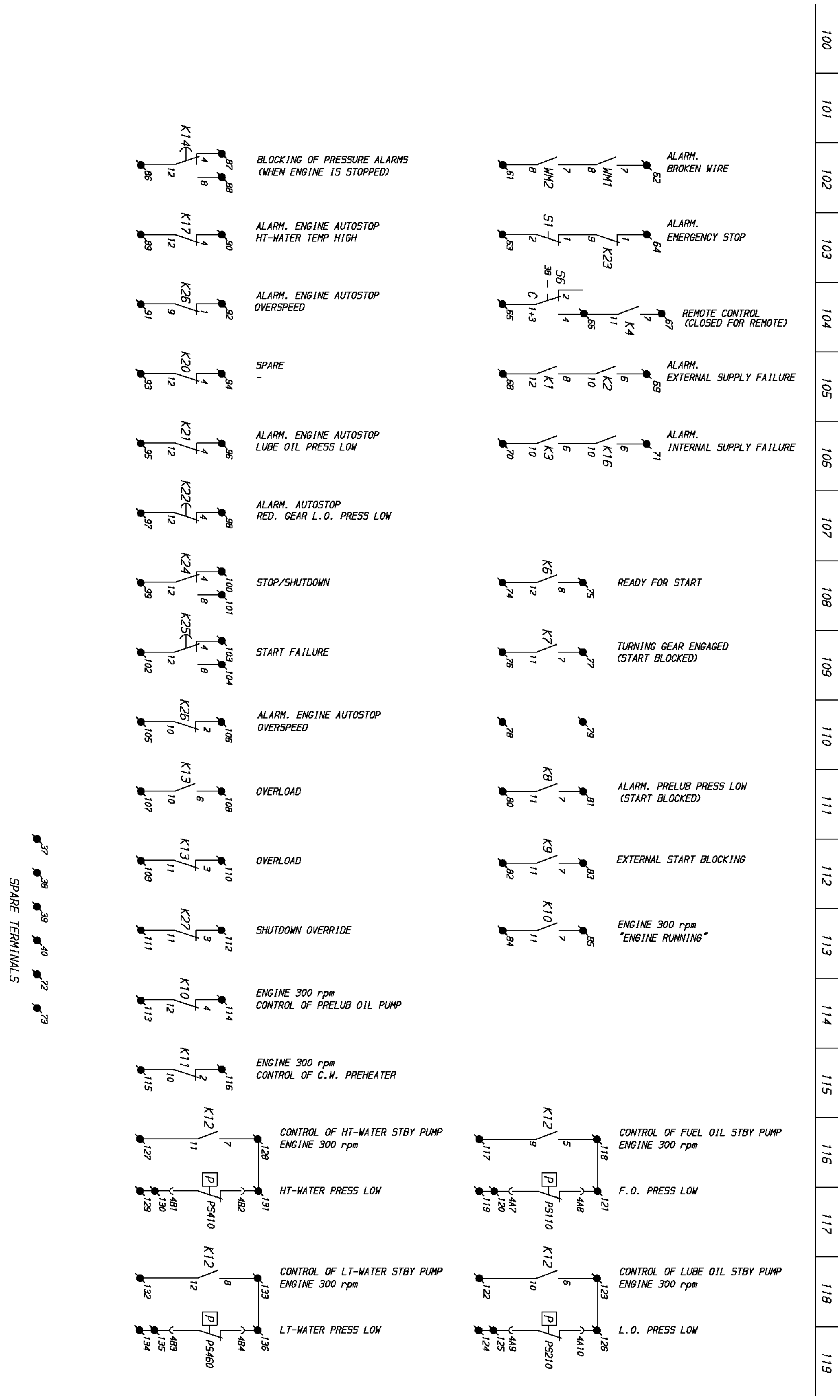


Indication lamps + Lables:		Colors:	Indication lamps + Labels:		Colors:
H1	Power on	White	H9	Start failure	Red
H2	Control on engine	White	H10	HT-water temperature high	Red
H3	Ready for start	White	H11	Lubrication oil pressure low	Red
H4	Engine running	Green	H12	Overspeed	Red
H5	Turning gear engaged	Orange	H13	Spare	Red
H6	Pre-lubricating oil pressure low	Orange	H14	Reduction gear lubricating oil pressure low	Red
H7	External start blocking	Orange	H15	Remote emergency stop	Red
H8	Shutdown override	Orange	H16	Overload	Red

Pushbuttons & Selectors + Lables:

S1	Emergency stop	Degree of protection: IP44
S2	Stop	Colour: RAL7032
S3	Start	
S4	Reset	
S5	Lamp test	
S6	Local/Remote mode of control	

Figure 14.10 Start/stop system for a main engine (DAAE003936-7b)



15. Electrical power generation and management

15.1 General

15.1.1 Definitions

The marine vessel's electric supply system is an alternating current (a.c.) three-phase, three–wire insulated system. The engine produced mechanical energy is converted into electrical energy by a generator, which usually is of the synchronous type and intended for continuous operation.

The voltage of the network and generator is low voltage (LV) up to 1000 V and medium voltage (MV) from 1 kV and up.

Ordinary low voltages are 400 V (50 Hz), 450 V (60 Hz) and 690 V (50 or 60 Hz).

Nominal medium system voltages are 3 kV, 3.3 kV, 6 kV, 6.6 kV, 10 kV and 11 kV for 50 Hz or 60 Hz.

Low voltage is normally used in installations with total power up to about 10 MVA due to short circuit current restrictions in the switchgear.

The common network frequency (f) is 50 Hz or 60 Hz and the generator synchronous rated speed nrG [rpm] is calculated from:

$$nrG = \frac{60 \times f}{p}$$

where p = pole pairs, and subsequently the number of poles = 2 x p

Generator power definitions:

S_r = rated output, rated apparent power in kilovolt-amperes kVA

$$S_r = \frac{P_r}{\cos \phi_r}$$

P_r = rated active power in kilowatts kW

$\cos \phi_r$ = rated power factor

$$\cos \phi_r = \frac{P_r}{S_r}$$

The generator rated active power limit P_r should match with the diesel rated output power P_{DIESEL} taking into account the efficiency η_{GEN} of the generator.

$$P_r = \eta_{GEN} \times P_{DIESEL}$$

Generator η_{GEN} is typically 95…97 % at full load and $\cos \phi = 0.8$

15.1.2 Electric load demand at consumers and generators

The load demand analysis (electric load balance) listing the various loads and modes onboard ship is usually evaluated in the concept design phase and made available to the generator set supplier as the basis for dimensioning the generator sets.

The generator feeds power to the consumers in the network including all electrical transmission losses. If only the consumer power consumption is advised, the total required power supplied by the generator shall be increased with the network losses, which typically could be 5…9 % depending on type, size and quality of electrical components.

15.1.3 Operation modes

The generators shall be capable of operating in parallel.

The operation modes of the vessel have different demands of electric power and number of generating sets in operation. Important factors are:

- Operation profile
- Actual operation mode and maximum expected load

- Operational practice (e.g. at least 2 generating sets running)
- Redundancy requirements
- Accepted loading practice of the generating sets (e.g. 90 % of P_r)

15.1.4 Basic requirements

For a.c. generating sets used onboard ships and offshore installations which have to comply with rules of classification society (Class), the specific requirements of the Class shall be observed.

The main source of electrical power consists of at least two generating sets, and a shaft generator may be considered to be one of the required generators if capable of operating in parallel. The capacity of the generating sets shall be such that in the event of any one set being stopped it will still be possible to supply those services necessary to provide normal operational conditions of propulsion, safety and minimum comfortable conditions of habitability.

In the following there are some common basic requirements of the generating set performance.

Frequency and voltage variations in a.c. installations:

Load condition:	Steady state	Transient state
Freq./speed regulation	95...105 %	90...110 %
A.c. voltage regulation	97.5...102.5 %	85...120 %

Although the Class sets requirements for sudden load changes, the general recommendation is to apply electrical loads in a ramp function rather than in sudden load steps. See also chapter *Operating Ranges*.

15.2 Electric power generation

15.2.1 General dimensioning criteria

The generator voltage, capacity and number of units are basically defined from the operation mode with the highest expected electric load. The most demanding operation mode is usually manoeuvring or cargo handling, while max speed at sea in a diesel-electric ship may require more power.

It should be considered that at least one generating set should be stand-by offering flexibility to perform maintenance work on any other generating set.

For example, in an uncomplicated vessel the generator capacity could be selected in a way that one unit is suitable for port and sea conditions, and two units for manoeuvring conditions having a 3rd unit as a stand-by.

General dimensioning criteria with respect to power, among others:

- Type of vessel
- Operation mode and application
- Requirements of the connected load
- Load power factor cos ϕ
- Cost efficient loading level, optimum specific fuel consumption
- Redundancy requirements
- Starting characteristics of high power motors

Due consideration is to be given to the transient frequency and voltage characteristics of the generating set during and after a sudden load change. Any particular requirement of the load acceptance shall be subject to agreement between the customer and Wärtsilä.

15.2.2 Power factor

Rated power factor cos ϕ_r of the generator shall be selected in accordance with the network load cos ϕ, which usually is 0.8 … 0.85.

In a diesel electric drive vessel e.g.: with cyclo converters and/or low loading of propulsors, the power factor is usually 0.7…0.8 and the generators are to be dimensioned accordingly.

The most common power factor for generators is 0.8.

15.2.3 Generator reactances

An important issue with regard to short circuit figures and starting capacity in the network is the generators' subtransient reactance x_d”. The x_d” is typically 15…20 (up to 25) %.

Generally a high x_d” causes a lower short circuit current but reduces the starting capacity of high power motors in the network due to an excessive voltage drop.

A very low x_d” increases the generator size in comparison to a high x_d”, but the possibility to choose a specific x_d” is somewhat restricted.

A compromise between high starting capacity and low short circuit level of the network, and low distortion level of the distorted voltage waveform in a ‘polluted’ vessel, is to be done when deciding the generator reactances.

15.2.4 Generator protection and switchgear

Generator set switchgear, control gear and monitoring equipment is usually mounted off the generating set. All components incorporated in the switchgear shall be adequately rated to suit the generating set and the specified mains operation, including the prospective fault current.

The generator is basically protected by the generator breaker and protection devices, usually being tripped by the following protection functions:

- Short circuit
- Overload
- Time delayed over-current
- Reverse-power
- Differential-current
- Voltage protections (over and under voltage release)
- Earth fault
- Stator RTD temperature HI/HI

Generating set protection systems mainly related to the engine are set in the chapter for *Automation system*, and comprise among others:

- Load shedding
- Overspeed
- Engine shutdown
- Emergency stop
- Major alarm from the speed governor

15.2.5 Motor starting capacity of the network

The starting capacity of the electrical network depends mainly on the connected spare generator capacity, generator x_d”, x_d’ and allowed voltage drop. The maximum allowed transient voltage drop is 15 %, which in some cases is too much for sensitive equipment.

The starting characteristics of the most power consuming motor or consumer is to be carefully checked. The generator manufacturer is to be informed (preferably at the offering stage) on the motor characteristics, operation and starting method in order to evaluate the expected voltage drop.

An excessive voltage drop causes generator dimensioning adjustments and/or means of alternative motor starting methods, e.g. soft starting device.

15.2.6 Speed Governor

The speed governor is a device, which senses the speed of the engine and controls the fuel flow to the engine to maintain the speed at the desired level to meet changes in load output. Governor types are mainly hydraulic/mechanical or electronic, which are used in more complex projects.

In electrical terms, the speed governor controls the generator's and network's frequency and the active load sharing by speed droop feedback or an 'isochronous' (zero droop) mode.

The steady state frequency characteristics depend mainly on the performance of the engine speed governor, while the transient frequency characteristics depend on the combined behaviour of all engine system components.

Basic definition of speed droop:

A decrease in speed reference for an increase in load, i.e. the % of the current speed reference by which the speed reference is drooped (decreased) from zero to full load.

An external speed setting from the power management system compensates the speed droop effect keeping the frequency stable in long term steady state conditions.

Speed droop based load sharing is possible with both a hydraulic/mechanical and an electronic governor. For most applications a droop of 3…5 % is recommended. The droop setting, as well as the dynamical performances of the governor, shall be equal for all paralleling generators in order to have a proportional load sharing.

An isochronous load sharing for paralleling generators is possible with an electronic governor. All paralleling generators are to have the same maker and type of electronic governor. The isochronous mode governor will maintain a constant speed up to 100 % load.

15.2.7 Automatic Voltage Regulator (AVR)

The AVR controls the generator voltage and the reactive load sharing. The brushless exciter-AVR system is to detect changes in terminal voltage (e.g. caused by a sudden load change) and to vary the field excitation as required to restore the terminal voltage of the generator.

The AVR, including the spare AVR where applicable, shall be tested and approved by the Class together with the generator forming a unit.

The exciter and AVR are normally supplied from the generator (shunt excitation) or sometimes from a shaft-mounted external Permanent Magnet Generator (PMG), which is used on generators, e.g. in a network with notable voltage distortion.

In order to maintain a possible network short-circuit current, high enough (at least 3 * I_N) to trip the generator or achieve selectivity in the distribution, a booster (short-circuit excitation) circuit is provided for the shunt excitation.

The reactive load sharing of paralleling generators is provided by the AVR using paralleling compensation circuits called:

- Voltage droop compensation
- Crosscurrent compensation

The droop compensation is the most commonly used circuit for reactive load sharing and is possible with an analogue or a digital AVR. The voltage droop depends on the reactive load, i.e. a decrease in voltage for an increase in reactive load.

The crosscurrent compensation is a more complex method for reactive load sharing. The voltage is maintained constant without 'droop', and the reactive load is balanced.

Manual voltage control in the main switchboard as a back-up is generally provided only on the request of the customer.

15.2.8 Shaft generators

A shaft generator (SG) is driven by a main propulsion unit, which usually is intended to operate at constant speed in a CPP installation.

Shaft generators are normally connected to:

- A secondary PTO from a step-up gear (generator runs parallel to the propeller shaft)
- A primary PTO from a step-up gear (generator runs parallel to the engine)
- An engine free end

A constant frequency shaft generator may be an alternative in a vessel with a diesel driving a FPP.

It is recommended to provide the main engines with electronic speed governors when shaft generator installations are applied in multi engine installations (twin-in/single-out).

The SG is dimensioned with regard to the operating mode, electric load at sea and thruster (or other high power consumer) sizes.

In the case with secondary PTO the shaft generator speed nrG and the gear ratio is to correspond to a suitable high speed of the main engine, in order to have power enough to run both shaft generator and CPP at a constant speed at sea. In the manoeuvring mode the propeller cavitation can be reduced, by selecting a 2-stage (speed) PTO gear enabling a lower main engine and propeller speed.

15.2.9 Earthed neutral

The vessels' generation and distribution systems are ordinarily insulated in low voltage installations as well as for tankers.

The network in medium voltage installations is mostly earthed via a high resistance connected to the generators' neutral. The rating of the earthed neutral system shall be defined taking into account the ratings of all components of electrical equipment in the generation circuit.

Earthed neutral options are e.g. a separate earthing transformer with a resistance, a low resistance earthed neutral or a direct earthed neutral.

The earthed neutral cabinet is normally delivered by the switchgear supplier and co-ordinated with the generator supplier.

15.2.10 Emergency diesel generator

The emergency source of electrical power shall be self-contained independently from engine room systems with more stringent requirements as to operability when heeling and listing as well as location, starting arrangements and load acceptance.

The emergency diesel generator (EDG), supplying the emergency consumers required by statutory requirements, is basically dimensioned according to worst loading case of fire fighting, flooding and blackout start.

The starting capacity of the emergency network shall be specially considered, as the most power consuming emergency electrical consumer (motor) often determines the size of EDG. Allowance is also recommended for possible future additional emergency loads.

The emergency consumers comprises e.g.: emergency lighting, navigational and communication equipment, fire alarm systems, fire and sprinkler pumps, bilge pump, water-tight doors, person lifts, steering gear.

Many shipowners have additional requirements with regard to EDG-supplied services as precautionary measures against blackout, e.g.: essential (non-emergency) auxiliaries for electric power generation and propulsion. This further loading of EDG shall of course be reflected in the EDG size, and a shedding system for non-emergency consumers to be provided and trip, in case the EDG should be overloaded.

It is not recommended to use the EDG as a harbour generator, ref. Solas Ch. II-1 Part D Reg. 42. 1.4 and Reg. 43. 1.4.

15.3 Electric power management system (PMS)

15.3.1 General

The main task of an electric power management system (PMS) is to control the generation plant and to ensure the availability of electrical power in the network as well as to avoid blackout situations.

The PMS controls the starting/stopping and synchronizing of a generator to the network, frequency monitoring, steady state load sharing between on-line generators, blackout starting, shaft generator, gear clutches and executes load tripping when the power plant is overloaded (load shedding).

The main busbar is normally subdivided into at least two parts connected by bus tie circuit breakers, and the connection of generating sets and other duplicated equipment shall be equally divided between the parts.

15.3.2 Control modes

The PMS is to have redundant hierarchy of control modes, the following provisions being typical:

- Automatic, independently derived signals without manual intervention

- Remote control, manually initiated
- Local control, e.g. hand or electric

The automatic mode is the normal operation mode. It is recommended that means are to be provided to start an engine locally and to synchronize manually at the main switchboard in case of PMS failure. The back-up system is recommended to be an independent operating system, hard wired and with galvanic isolation from the main system.

Monitoring of the generating set operation to verify correct functioning by measurement or protection and supervisory control parameters in accordance to Class and requirements are set in the chapter *Automation system*.

15.3.3 Main breaker control

The following main breakers in the main switchboard are typically controlled from the PMS:

- Diesel generator
  Shaft generator
- Bus tie breaker
- Shore connection
- High power consumers, e.g.: bow thruster, AC-compressor
- Emergency switchboard connection

15.3.4 Blackout start and precautionary measures

In case of blackout in the main switchboard (MSB) the related generating sets get a starting order and the first available generating set to ‘run up’ will connect to the MSB. The following units are to be automatically synchronized.

Precautions against failing blackout start are:

- Booster and fuel supply pumps connected to emergency switchboard (ES)
  pre lubricating pump connected to ES
- Sequential re-start of essential pumps, fans and heavy consumers to achieve a loading ramp rather than big loading steps

Precautions against total loss of propulsion (diesel mechanical concepts) in a blackout situation could be following measures:

- Essential ME pumps are engine driven
- Essential propulsion train pumps are gear driven
- Essential electrical pumps and fans for propulsion are connected to ES
- Operate with split network

15.3.5 Paralleling of generators, load sharing

The PMS provides automatic synchronizing of auxiliary diesel generators i.e. voltage and frequency adjustment to bring the incoming set into synchronism and phase with the existing system, considering possible restrictions (e.g.: short circuit level) regarding max number of generators allowed to be connected to the MSB.

The PMS controls the active (kW) load sharing over the speed governor:

- Droop control, characteristics about 4 %
- Isochronous load sharing, possible by means of an electronic speed governor taking care of ramping up, load sharing and ramping down; PMS only connects the set and after allowance by the governor disconnects the set.

Active load sharing between diesel generators is normally proportional (balanced). The droop setting shall be equal for all paralleling generators in order to have a proportional load sharing.

Some feature mode options could promote an economical and environment-friendly operation of the engines, e.g.:

- Master-topping up, i.e. master(s) with constant optimal load and a topping up set taking care of the load variations
- Sequencing of the master-topping up units

15.3.6 Shaft generator load transfer

The PMS controls the main engine in shaft generator (SG) applications giving priority to the electric generation, including possible propulsion load reduction where applicable.

Operating with SG supplying the main switchboard (MSB) in parallel with the connected propulsion line, the frequency may be unstable in rough sea, etc. It is recommended to use the SG independently supplying the MSB or part of it. If 2 SG are available e.g. in a twin-screw vessel, the MSB should be split into 2 parts, each part being supplied by a dedicated SG.

The load transfer from/to the auxiliary diesel generator(s) should normally be on a short time basis, i.e. paralleling only for the time of unloading the generator(s) followed by generator breaker opening.

The shaft generator is typically supplying thruster(s) in a separate network during the manoeuvring mode.

In the following a typical example of load transfer at sea to a running shaft generator when the thrusters have been disconnected:

- Assure that the main engine load is stable and that the constant speed mode is selected
- Synchronize the SG-section and the MSB (i.e. the auxiliary diesel engine(s) are usually synchronized to the main engine) and close the SG-section bustie breaker
- Transfer load to SG by unloading the auxiliary diesel generator(s)
- Open the auxiliary diesel generator's breaker(s) when unloading trip level is reached
- Stop the auxiliary diesel engine(s) after the cooling down time

15.3.7 Load dependent start/stop

The PMS includes functions for automatic load dependent start/stop of diesel generation sets.

The start/stop limits and start order in an installation with several paralleling generating sets are set to achieve an optimal loading of the engines in the specific operation mode of the vessel. The PMS calculates the network's P_N and total generator load over a defined period of time and compares that against the load dependent autostart/autostop limits. The objective is to ensure that the actual load is supplied by an appropriate number of generating sets to achieve best possible energy efficiency and fuel economy.

15.3.8 Power reservation for heavy consumers

Heavy consumers may be connected to a power reservation system in the PMS, which checks if there is enough reserve power capacity in the network upon a start request from the heavy consumer. If necessary the PMS will start and synchronize the next standby unit, and gives the start permission to the heavy consumer when the needed starting capacity is available.

15.3.9 Load shedding (preference tripping)

Auto start function is not fast enough as blackout prevention after rapid and large loss of power generating capacity, e.g. after tripping of a generator.

In order to protect the generator(s) against sustained overload, and to ensure the integrity of supplies to services required for propulsion and steering as well as the safety of the ship, suitable load shedding arrangements shall be arranged.

Typical consumers that may be tripped are:

- Galley consumers
- AC-compressors
- Accommodation ventilation
- Reduction of propulsion power

15.3.10 Special applications, e.g.: Auxiliary Propulsion Drive (APD)

A special application providing limited redundancy with respect to increased availability of the vessel's propulsion system is the so-called Auxiliary Propulsion Drive (APD). The principle idea of this solution is that the ship can be propelled by the auxiliary generating sets, by using the shaft generator as an electric motor, in case the main engine (ME) is not available.

The benefit of the combined shaft generator and APD is an increase of safety when it is used as back-up propulsion in e. g. following operating modes:

- Booster mode, both ME and PTO are driving the propeller
- Standby mode, ME disconnected for maintenance and APD is connected if manoeuvring is required
- Emergency mode (take me home), APD is used to propel the ship if ME fails

15.3.11 Typical one line main diagrams

Figure 15.1 Diesel electric ship, medium voltage network

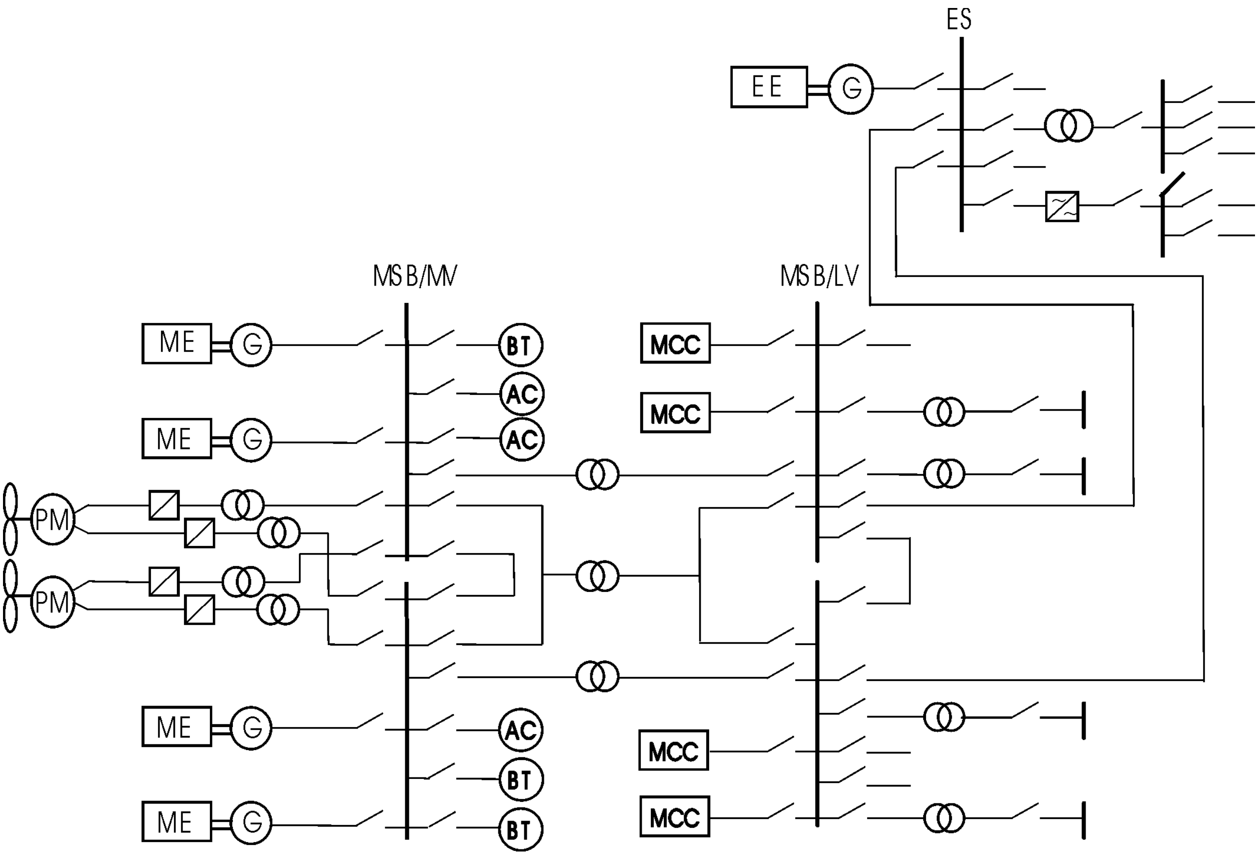


Figure 15.2 4 ADG low voltage network

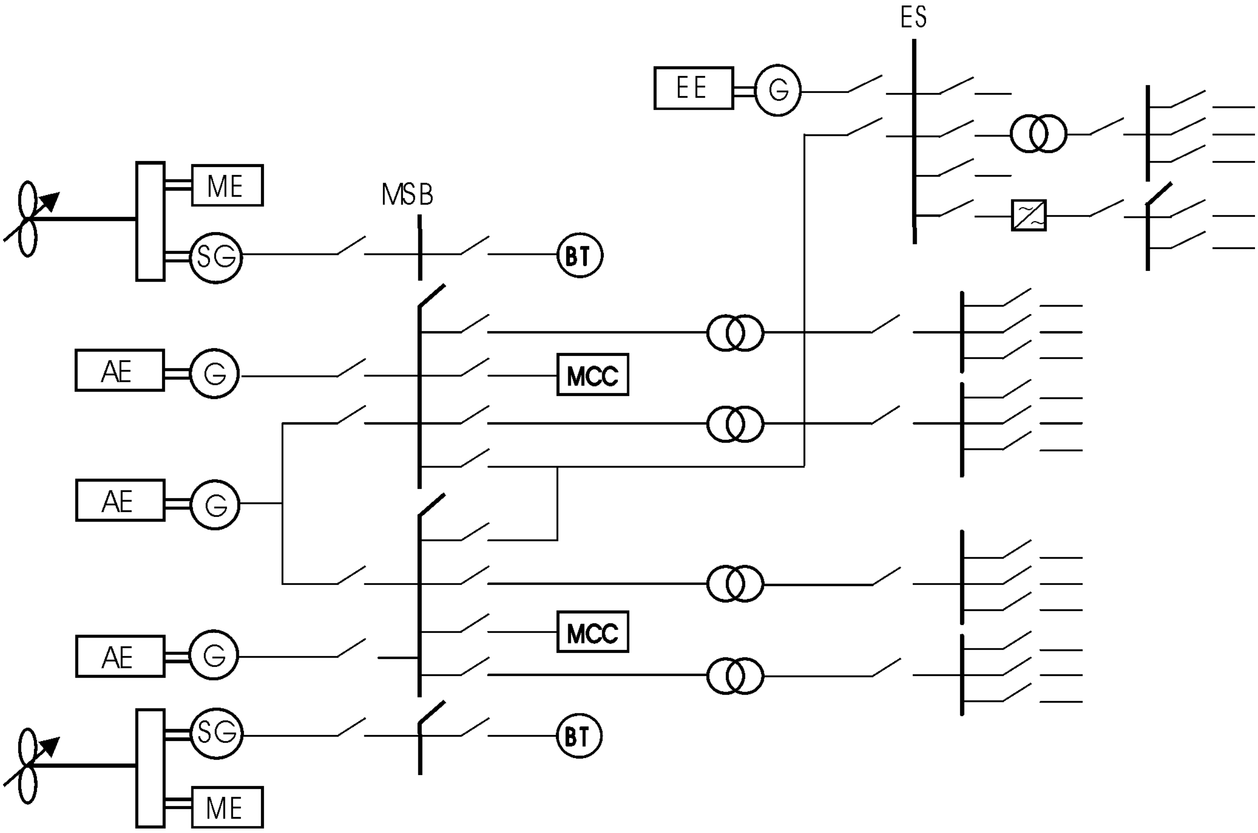
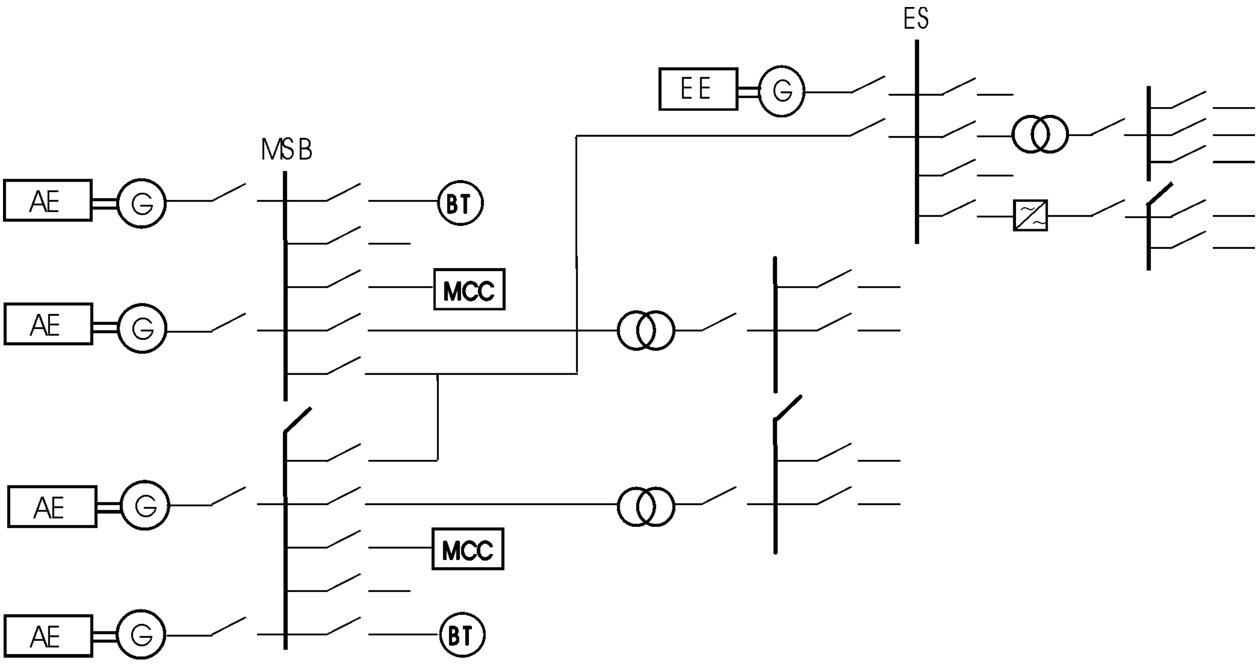


Figure 15.3 3 ADG + 2 SG low voltage network



AC	Air conditioning	**G**	Generator	**MSB**	Main switch board
AE	Auxiliary engine	**LV**	Low voltage	**MV**	Medium voltage
BT	Bow thruster	**MCC**	Motor control center	**PM**	Propulsion motor
EE	Emergency engine	**ME**	Main Engine	**SG**	Shaft generator
ES	Emergency switch board				

16. Foundation

16.1 Engine seating

Engines can be either rigidly mounted on chocks, or resiliently mounted on rubber elements. Wärtsilä should be informed about existing excitations (other than Wärtsilä supplied engine excitations) and natural hull frequencies, especially if resilient mounting is considered.

The foundation should be as stiff as possible in all directions to absorb the dynamic forces caused by the engine, reduction gear and thrust bearing. The foundation should be dimensioned and designed so that harmful deformations are avoided.

Dynamic forces caused by the engine are given in the *Vibration and noise* chapter.

16.1.1 System oil tank

The system oil tank should not extend under the reduction gear, if the engine is of dry sump type and the oil tank is located beneath the engine foundation. Neither should the tank extend under the support bearing, in case there is a PTO arrangement in the free end. The oil tank must be symmetrically located in transverse direction under the engine.

16.2 Reduction gear foundation

The engine and the reduction gear must have common foundation girders.

16.3 Free end PTO driven equipment foundations

The foundation of the driven equipment must be integrated with the engine foundation.

16.4 Rigid mounting of engine

Main engines can be rigidly mounted to the foundation either on steel chocks or resin chocks.

Prior to installation the shipyard must send detailed plans and calculations of the chocking arrangement to the classification society and to Wärtsilä for approval.

The engine has four feet integrated to the engine block. There are two Ø22 mm holes for M20 holding down bolts and a threaded M16 hole for a jacking screw in each foot. The Ø22 holes in the seating top plate for the holding down bolts can be drilled though the holes in the engine feet. In order to avoid bending stress in the bolts and to ensure proper fastening, the contact face underneath the seating top plate should be counterbored.

16.4.1 Holding down bolts

Holding down bolts are through-bolts with lock nuts. Selflocking nuts are acceptable, but hot dip galvanized bolts should not be used together with selflocking (nyloc) nuts. Two of the holding down bolts are fitted bolts and the rest are clearance (fixing) bolts. The fixing bolts are M20 8.8 bolts according DIN 931, or equivalent. The two Ø23 H7/m6 fitted bolts are located closest to the flywheel, one on each side of the engine. The fitted bolts must be designed and installed so that a sufficient guiding length in the seating top plate is achieved, if necessary by installing a distance sleeve between the seating top plate and the lower nut. The guiding length in the seating top plate should be at least equal to the bolt diameter. The fitted bolts should be made from a high strength steel, e.g. 42CrMo4 or similar and the bolt should have a reduced shank diameter above the guiding part in order to ensure a proper elongation. The recommended shank diameter for the fitted bolts is 17 mm.

The tensile stress in the bolts is allowed to be max. 80% of the material yield strength and the equivalent stress during tightening should not exceed 90% of the yield strength.

The elongation of holding down bolts can be calculated from the formula:

$$\Delta L = 6.18 \times 10^{-6} \times F \sum_{i=1}^{n} \frac{L_i}{D_i^2}$$

where:

ΔL = bolt elongation [mm]

F = tensile forece in bolt [N]

L_i = part length of bolt with diameter D_i [mm]

D_i = part diameter of bolt with length L_i [mm]

16.4.2 Lateral supports

Lateral supports as shown in *Figure 16.3 Chocking of main engines (2V69A0238b)* shall be fitted against the engine block. If the engine is to be installed on resin chocks, the supports should be welded to the foundation before casting the chocks. The wedges in the supports are to be installed without clearance, when the engine has reached normal operating temperature. The wedges are then secured in position with welds. An acceptable contact surface against the engine block must be obtained on the wedges of the supports.

16.4.3 Steel chocks

The top plates of the engine foundation are normally inclined outwards with regard to the centre line of the engine. The inclination of the supporting surface should be 1/100. The seating top plate should be designed so that the wedge-type steel chocks can easily be fitted into their positions. The supporting surface of the seating top plate should be machined so that a contact surface of at least 75% is obtained.

The size of the chocks should be approximately 115 x 170 mm at the position of the fitted bolts (2 pieces) and approx. 115 x 190 mm at the position of the fixing bolts (6 pieces). The design should be such that the chocks can be removed, when the lateral supports are welded to the foundation and the engine is supported by the jacking screws. The chocks should have an inclination of 1:100 (inwards with regard to the engine centre line). The cut out in the chocks for the fixing bolts shall be 24 - 26 mm (M20 bolts), while the hole in the chocks for the fitted bolts shall be drilled and reamed to the correct size (ø23 H7) when the engine is finally aligned to the reduction gear. The chocks are preferably made of steel but also cast iron chocks are permitted.

The M20 holding down bolts are to be tightened to 430 Nm (lightly oiled); assuming grade 8.8 clearance bolts and fitted bolts having a tensile strength of at least 640 N/mm2 and a minimum shank diameter of 17 mm. In case a low-friction lubricant is used, such as MoS2, the maximum permissible tightening torque must be verified.

16.4.4 Resin chocks

The top plates of the engine foundation can be horizontal when resin chocks are used.

Engine installation on resin chocks is possible provided that the requirements of the classification societies are fulfilled. During normal conditions, the support face of the engine feet has a maximum temperature of about 75°C, which should be considered when selecting the type of resin.

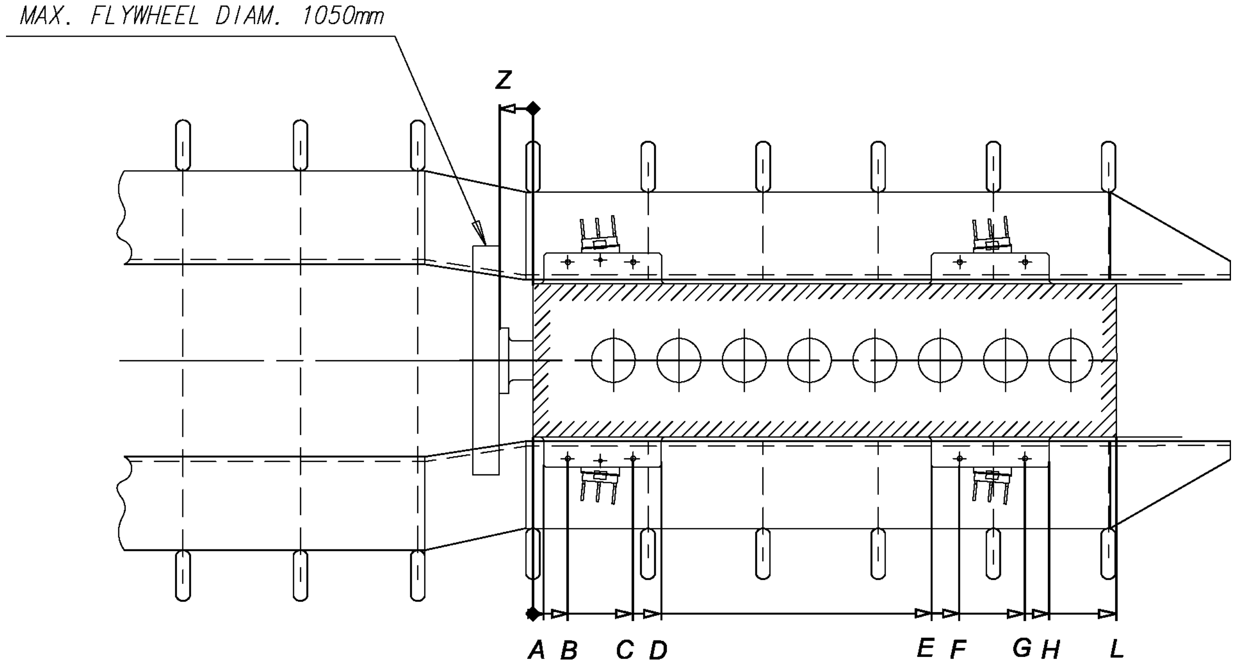
The size of the resin chocks should be 105 x 540 mm. The total surface pressure on the resin must not exceed the maximum value, which is determined by the type of resin and the requirements of the classification society. It is recommended to select a resin that has a type approval from the relevant classification society for a total surface pressure of 5 N/mm^2. (A typical conservative value is p_{tot} 3.5 N/mm^2).

For a given bolt diameter the permissible bolt tension is limited either by the strength of the bolt material, or by the maximum permissible surface pressure on the resin. For 105 x 540 mm chocks, 8.8 grade M20 fixing bolts and fitted bolts having a tensile strength of at least 640 N/mm^2 and a minimum shank diameter of 17 mm the tightening torques (lightly oiled) are

* Total surface pressure < 3.5 N/mm^2 => 300 Nm

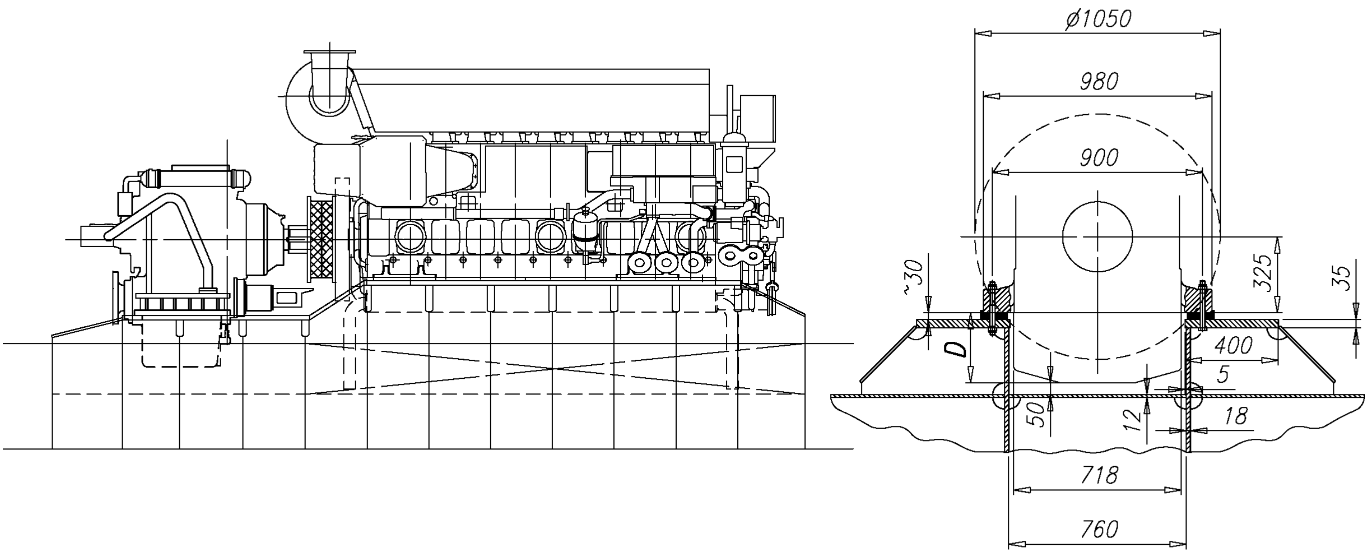
* Total surface pressure approx. 4.5 N/mm^2 => 400 Nm

Figure 16.1 Main engine seating, view from above (DAAE017514a)



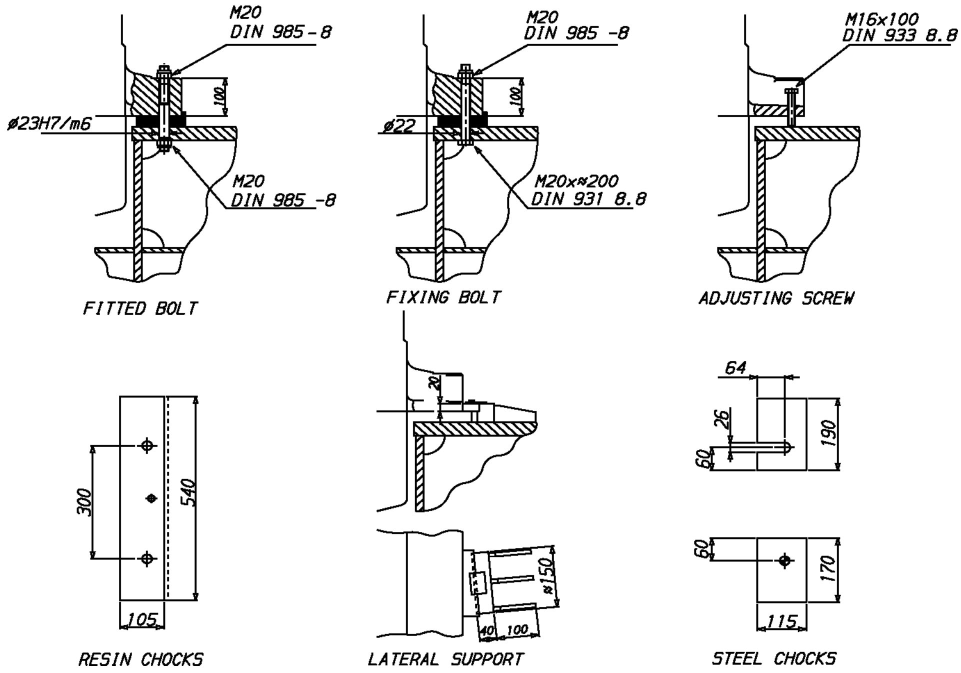
Engine type	A [mm]	B [mm]	C [mm]	D [mm]	E [mm]	F [mm]	G [mm]	H [mm]	L [mm]	Z [mm]
4L20	50	160	460	590	930	1060	1360	1470	1480	155
6L20	50	160	460	590	1530	1660	1960	2070	2080	155
8L20	50	160	460	590	1830	1960	2260	2370	2380	155
9L20	50	160	460	590	2130	2260	2560	2670	2680	155

Figure 16.2 Main engine seating, side and end view (DAAE017514a)



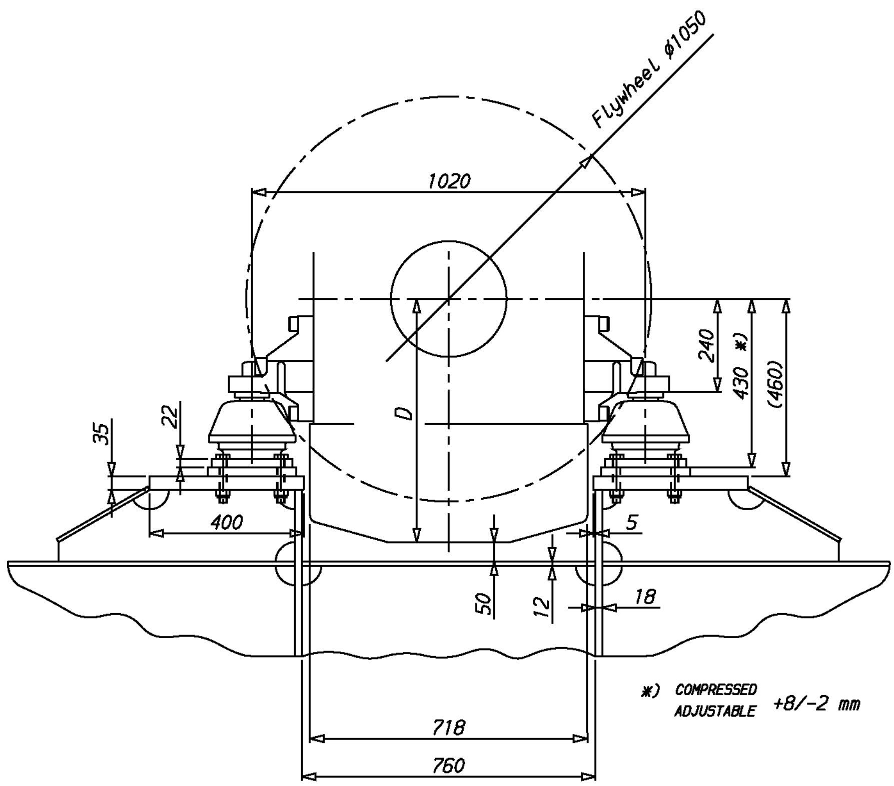
Engine type	(D) Deep sump [mm]	(D) Wet sump [mm]	(D) Dry sump [mm]
4L20	-	400	-
6, 8, 9L20	500	300	300

Figure 16.3 Chocking of main engines (2V69A0238b)



16.5 Resilient mounting of engine

Figure 16.4 Resilient mounting (DAAE003263)



Engine type	(D) Deep sump [mm]	(D) Wet sump [mm]	(D) Dry sump [mm]
4L20	-	725	-
6, 8, 9L20	825	625	625

In order to reduce vibrations and structure borne noise, main engines can be resiliently mounted on rubber mounts. The transmission of forces emitted by a resiliently mounted engine is 10-20% compared to a rigidly mounted engine.

For resiliently mounted engines a speed range of 750-1000 rpm is generally available.

Conical rubber mounts are used in the normal mounting arrangement and additional buffers are thus not required. A different mounting arrangement can be required for wider speed ranges (e.g. FPP installations).

Resilient mounting is not available for 4 cylinder engines.

16.6 Mounting of generating sets

Generating sets, comprising engine and generator mounted on a common base plate, are installed on resilient mounts on the foundation in the ship.

The resilient mounts reduce the structure borne noise transmitted to the ship and also serve to protect the generating set bearings from possible fretting caused by hull vibration.

The number of mounts and their location is calculated to avoid resonance with excitations from the generating set engine, the main engine and the propeller.

Note! To avoid induced oscillation of the generating set, the following data must be sent by the shipyard to Wärtsilä at the design stage:

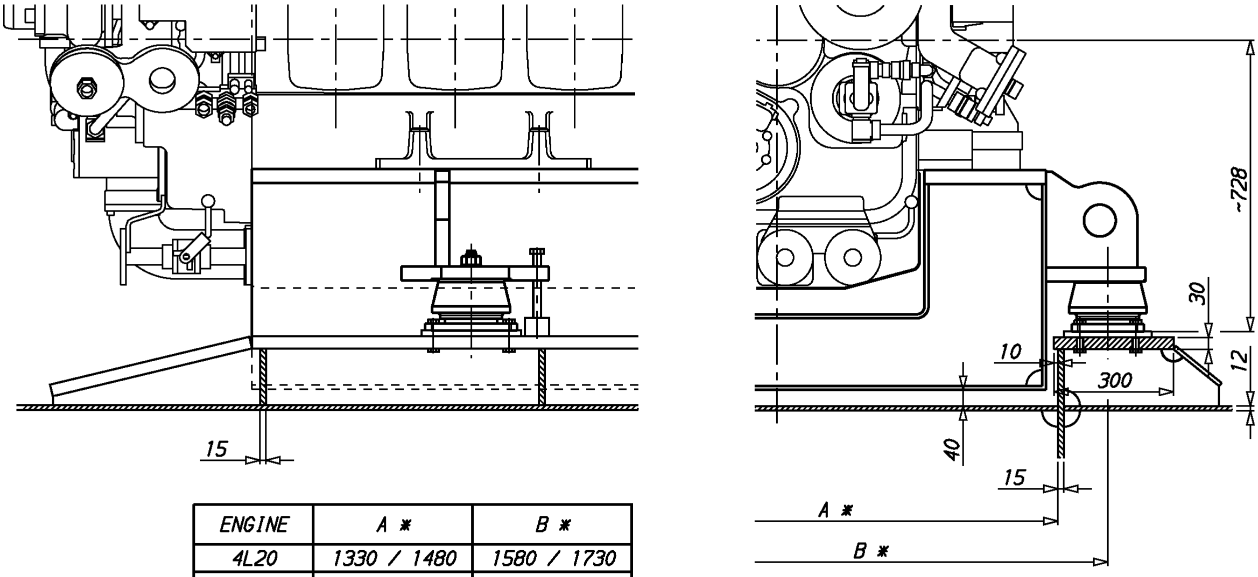
- Main engine speed [rpm] and number of cylinders
- Propeller shaft speed [rpm] and number of propeller blades

The selected number of mounts and their final position is shown in the generating set drawing.

16.6.1 Seating

The seating for the common base plate must be rigid enough to carry the load from the generating set. The recommended seating design is shown in the figure below.

Figure 16.5 Recommended design of the generating set seating (3V46L0720d)



ENGINE	A *	B *
4L20	1330 / 1480	1580 / 1730
6L20	1330/1480/1630	1580/1730/1880
8L20	1480 / 1630	1730 / 1880
9L20	1630 / 1860	1880 / 2110

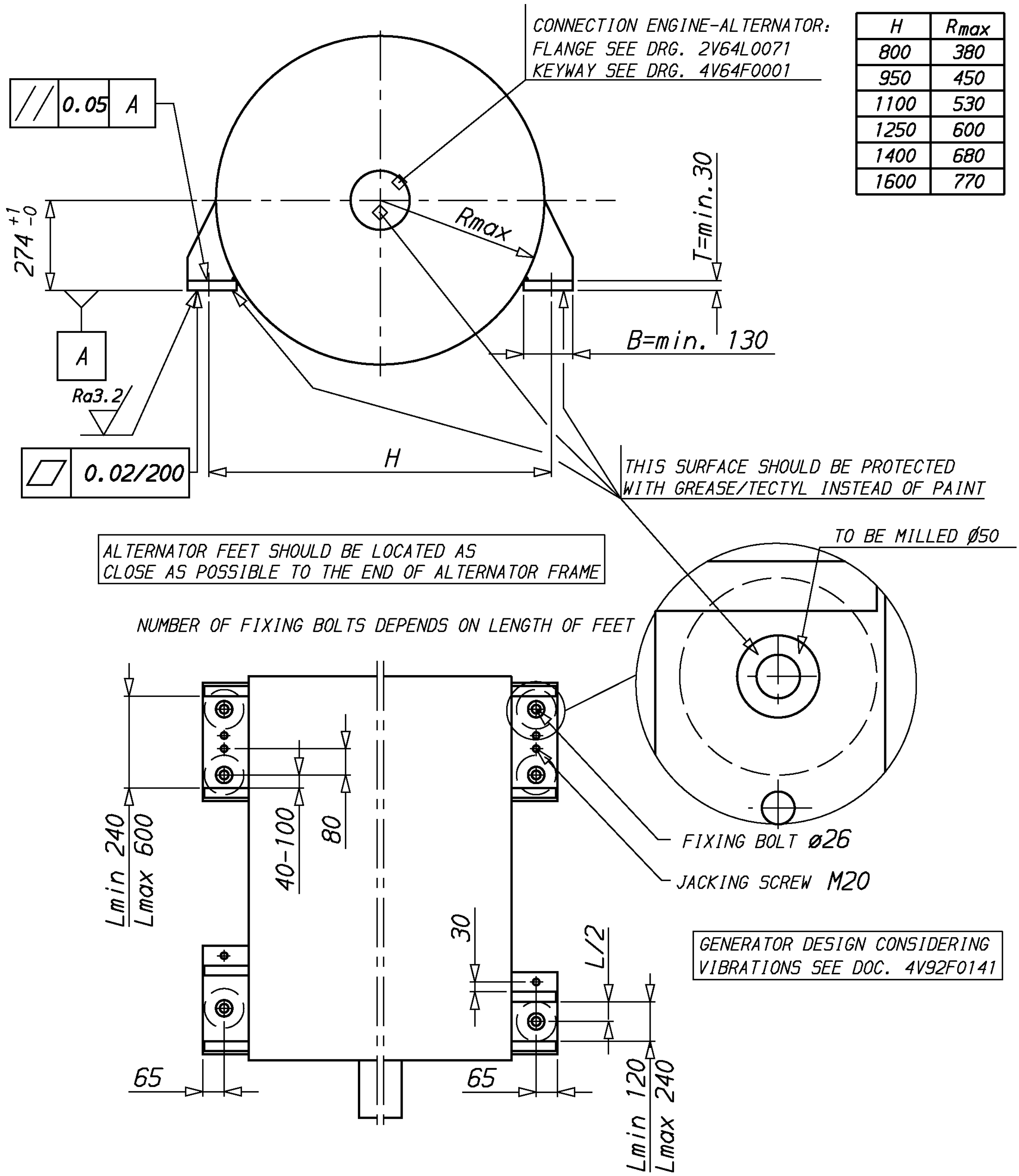
* DEPENDENT ON ALTERNATOR WIDTH

If the generating set will be installed directly on a deck or on the tank top, the alternative design shown in drawing 4V46L0296 can be permitted.

The lateral distance between the mounts varies between 1340 mm and 1640 mm depending on the number of cylinders of the engine and the type of generator.

16.6.2 Generator feet design

Figure 16.6 Instructions for designing the feet of the generator and the distance between its holding down bolt (4V92F0134c)



16.6.3 Rubber mounts

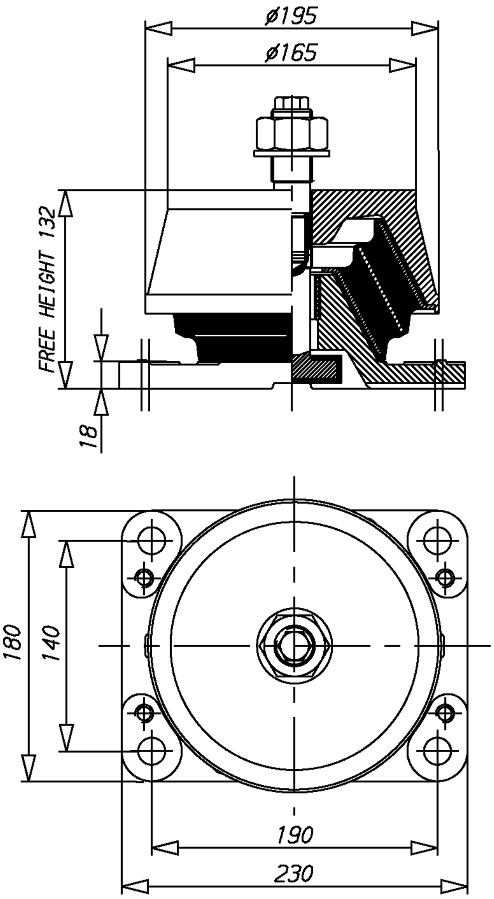
The generating set is mounted on conical resilient mounts, which are designed to withstand both compression and shear loads. In addition the mounts are equipped with an internal buffer to limit movements of the generating set due to ship motions. Hence, no additional side or end buffers are required.

The rubber in the mounts is natural rubber and it must therefore be protected from oil, oily water and fuel.

The mounts should be evenly loaded, when the generating set is resting on the mounts. The maximum permissible variation in compression between mounts is 2,0 mm. If necessary, chocks or shims should be used to compensate for local tolerances. Only one shim is permitted under each mount.

The transmission of forces emitted by the engine is 10-20% when using conical mounts.

Figure 16.7 Rubber mounts (3V46L0706a)



Type	Rubber hardness	Load capacity
RD 314	45	2600 kg
RD 314	50	3200 kg
RD 314	55	3700 kg
RD 314	60	4300 kg
RD 314	65	4900 kg
RD 315	45	1400 kg
RD 315	50	1900 kg
RD 315	55	2300 kg
RD 315	60	2700 kg
RD 315	65	3100 kg

16.7 Flexible pipe connections

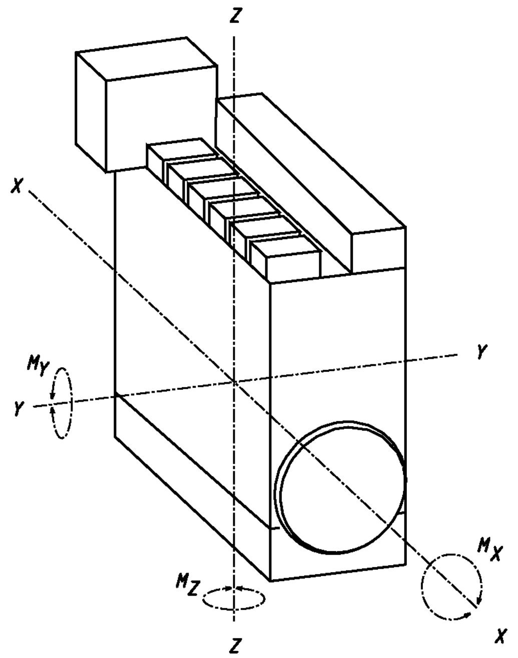
When the engine or the generating set is resiliently installed, all connections must be flexible and no grating nor ladders may be fixed to the generating set. When installing the flexible pipe connections, unnecessary bending or stretching should be avoided. The external pipe must be precisely aligned to the fitting or flange on the engine. It is very important that the pipe clamps for the pipe outside the flexible connection must be very rigid and welded to the steel structure of the foundation to prevent vibrations, which could damage the flexible connection.

17. Vibration and noise

17.1 General

Dynamic forces and moments caused by the engine appear from the table. Due to manufacturing tolerances some variation of these values may occur.

Figure 17.1 Coordinate system of the external torques (1V93C0029)



17.2 External forces and couples

Table 17.1 External forces

Engine	Speed [rpm]	Frequency [Hz]	Fz [kN]
4L20	720 750 900 1000	48 50 60 66.7	0.7 0.8 1.1 1.4
8L20	720 750 900 1000	48 50 60 66.7	1.4 1.5 2.2 2.7

$F_Z = 0$, $F_Y = 0$ and $F_X = 0$ for 5, 6 and 9 cylinder engines

Table 17.2 External couples

Engine	Speed [rpm]	Frequency [Hz]	MY [kNm]	MZ [kNm]
9L20	720 - 750 - 900 - 1000 	12 24 12.5 25 15 30 16.7 33.3	4.5 3.1 4.9 3.3 7 4.8 8.6 5.9	4.5 — 4.9 — 7 — 8.6 —

$M_Z = 0$, $M_Y = 0$ for 4, 6 and 8 cylinder engines

Table 17.3 Rolling moments

Engine	Speed [rpm]	Frequency [Hz]	Full load (M_X) [kNm]	Zero load (M_X) [kNm]	Frequency [Hz]	Full load (M_X) [kNm]	Zero load (M_X) [kNm]
4L20	720	24	10	4.8	48	7.5	1.6
	750	25	9.4	5.6	50	7.5	1.5
	900	30	4.8	10	60	7.4	1.4
	1000	33.3	1.5	13	66.7	7.4	1.3
6L20	720	36	13	1.4	72	4.2	1.2
	750	37.5	12	1.9	75	4.2	1.2
	900	45	9.8	4.7	90	4.7	1.3
	1000	50	7.8	6.8	100	4.7	1.3
8L20	720	48	15	3.1	96	1.7	0.7
	750	50	15	3.1	100	1.7	0.7
	900	60	15	2.8	120	2.2	0.7
	1000	66.7	15	2.6	133.3	2.2	0.7
9L20	720	54	13	3.5	108	1.1	0.5
	750	56.3	13	3.5	112.5	1.1	0.5
	900	67.5	14	3.6	135	1.6	0.5
	1000	75	14	3.6	150	1.6	0.5

17.3 Mass moments of inertia

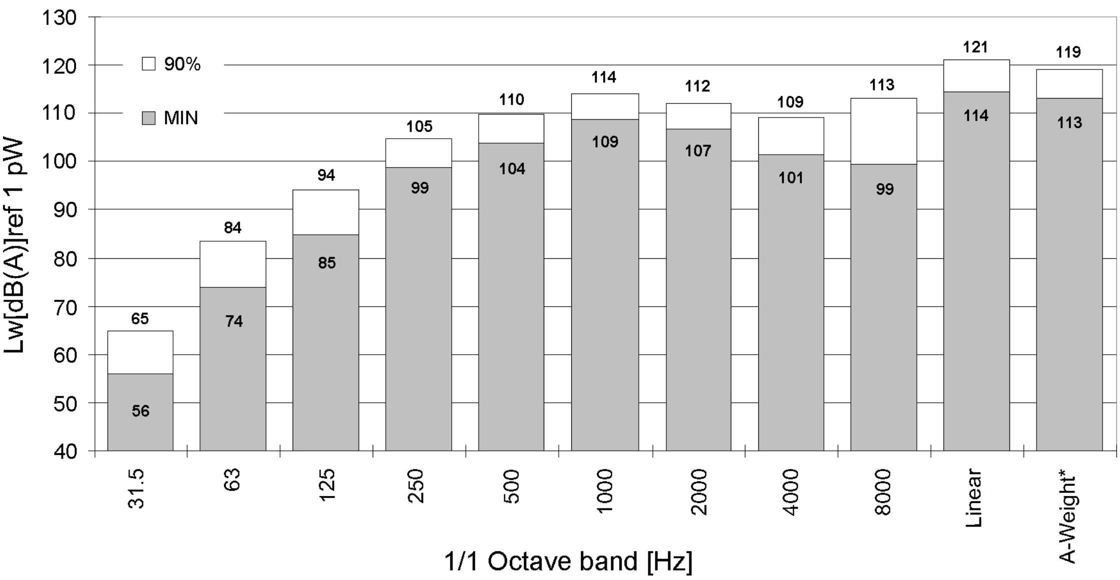
The mass-moments of inertia of the propulsion engines (including flywheel, coupling outer part and damper) are typically as follows:

Engine	J[kgm²]
4L20	90 - 120
6L20	90 - 150
8L20	110 - 160
9L20	100 - 170

17.4 Air borne noise

The airborne noise of the engine is measured as a sound power level according to ISO 9614-2. The results are presented with A-weighing in octave bands, reference level 1 pW. Two values are given; a minimum value and a 90% value. The minimum value is the smallest sound power level found in the measurements. The 90% level is such that 90% of all measured values are below this figure.

Figure 17.2 Sound power levels



18. Power transmission

18.1 General

The full engine power can be taken from both ends of the engine. At the flywheel end there is always a flywheel for the management of the torsional vibration characteristics of the system and to facilitate manual turning of the engine.The flywheel creates a natural flange connection. At the free end a shaft connection as a power take off can be provided.

18.2 Connection to generator

Figure 18.1 Connection engine/single bearing generator (2V64L0071)

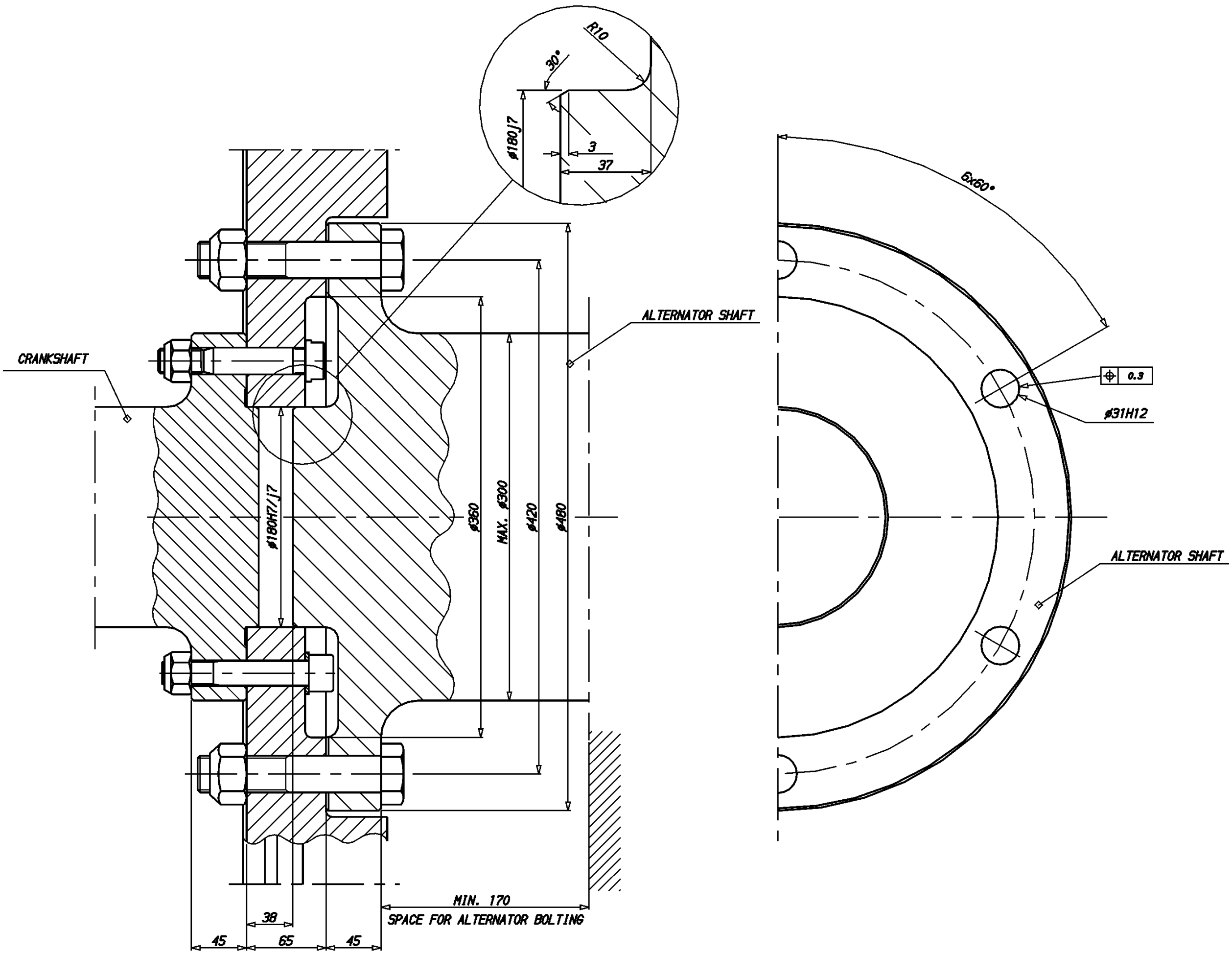
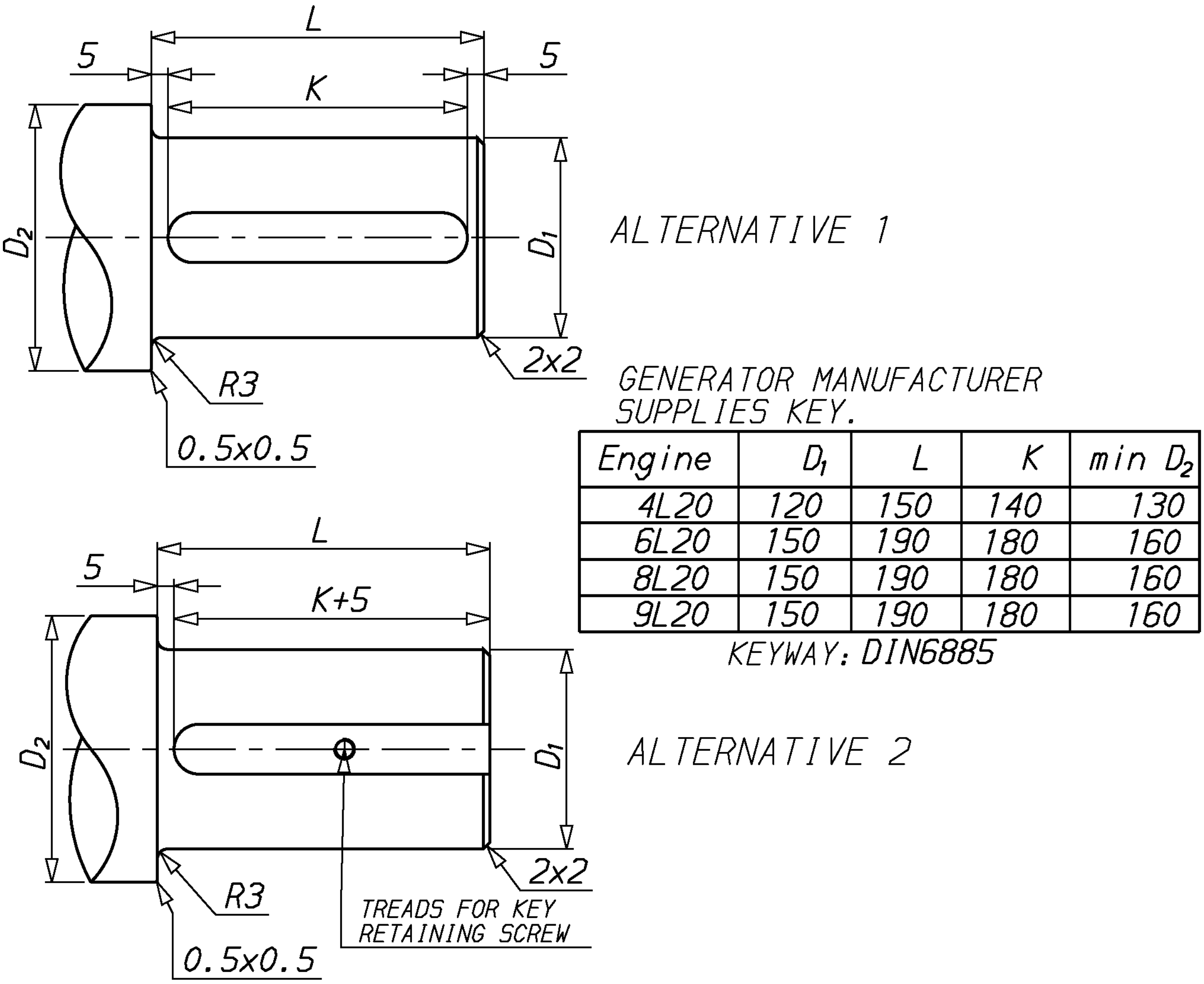


Figure 18.2 Connection engine/two-bearing generator (4V64F0001a)



Engine	D_1	L	K	min D_2
4L20	120	150	140	130
6L20	150	190	180	160
8L20	150	190	180	160
9L20	150	190	180	160

18.3 Flexible coupling

The power transmission of propulsion engines is accomplished through a flexible coupling or a combined flexible coupling and clutch mounted on the flywheel. The crankshaft is equipped with an additional shield bearing at the flywheel end. Therefore also a rather heavy coupling can be mounted on the flywheel without intermediate bearings.

The type of flexible coupling to be used has to be decided separately in each case on the basis of the torsional vibration calculations.

In case of two bearing type generator installations a flexible coupling between the engine and the generator is required.

18.4 Clutch

In many installations the propeller shaft can be separated from the diesel engine using a clutch. The use of multiple plate hydraulically actuated clutches built into the reduction gear is recommended.

A clutch is required when two or more engines are connected to the same driven machinery such as a reduction gear.

To permit maintenance of a stopped engine clutches must be installed in twin screw vessels which can operate on one shaft line only. A shaft line locking device should also be fitted to be able to secure a propeller shaft in position so that windmilling is avoided as also an open hydraulic clutch can transmit a small torque.

18.5 Shaftline locking device and brake

18.5.1 Locking device

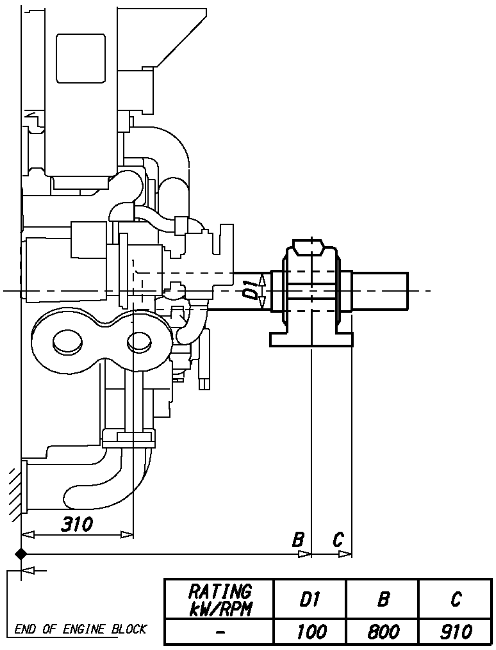
A shaftline locking device is needed when the operation of the ship makes it possible to turn the shafting by the water flow in the propeller.

18.5.2 Brake

A shaftline brake is needed when the shaftline needs to be actively stopped. This is the case when the direction of rotation needs to be reversed.

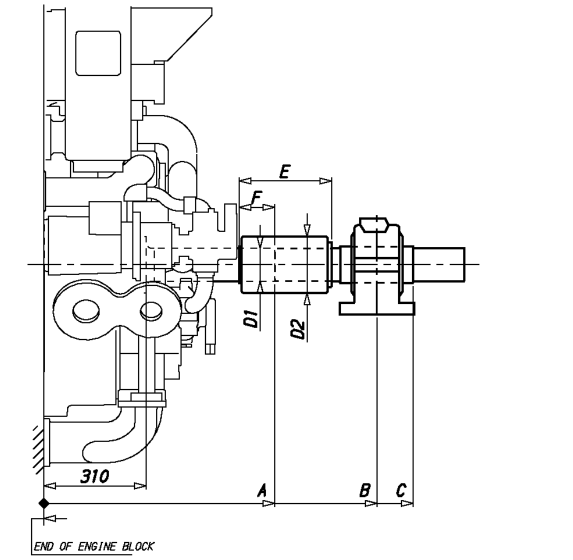
18.6 Power-take-off from the free end

Figure 18.3 Power take off at free end alternative 1 (4V62L0931)



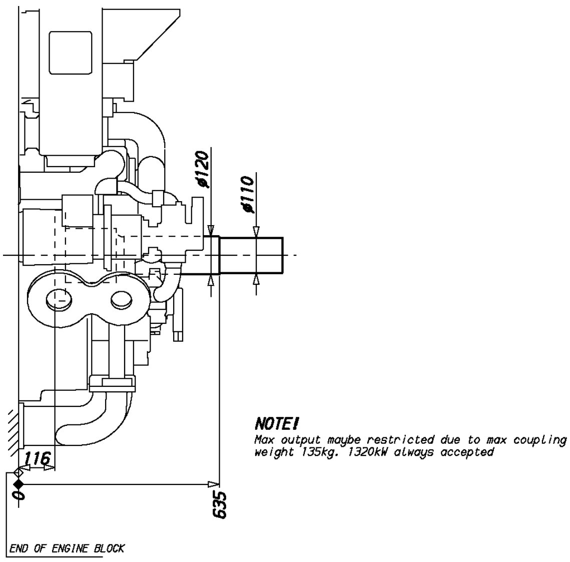
RATING kW/RPM	D1	B	C
-	100	800	910

Figure 18.4 Alternative 2 (4V52L0932a)



RATING kW/RPM	D1	D2	A	B	C	E	F
0.97	100	170	700	1010	1120	280	108
1.29	110	185	755	1070	1180	300	118
1.67	120	200	765	1100	1210	325	130
2.13	130	215	775	1120	1230	350	140

Figure 18.5 Alternative 3 (4V62L1158)



18.7 Input data for torsional vibration calculations

A torsional vibration calculation is made for each installation. For this purpose exact data of all components included in the shaft system are required. See list below.

18.7.1 Installation

- Classification society
- Ice class
- Operating modes

18.7.2 Reduction gear

A mass-elastic diagram showing:

- All clutching possibilities
- Sense of rotation of all shafts
- Dimensions of all shafts
- Mass moment of inertia of all rotating parts including shafts and flanges
- Torsional stiffness of shafts between rotating masses
- Material of shafts including tensile strength and modulus of rigidity
- Gear ratios
- Drawing and revision number of the diagram

18.7.3 Propeller and shafting

A mass-elastic diagram or propeller shaft drawing showing:

- Mass moment of inertia of all rotating parts including the rotating part of the OD-box, SKF couplings and rotating parts of the bearings.
- Mass moment of inertia of the propeller at full pitch and zero pitch in water
- Torsional stiffness or exact dimensions of the shaft
- Material of the shaft including tensile strength and modulus of rigidity
- Drawing and revision number of the diagram or drawing

18.7.4 Main generator or shaft generator

A mass-elastic diagram or an generator shaft drawing showing:

- Generator output, speed and sense of rotation
- Mass moment of inertia of all rotating parts or a total inertia value of the rotor, including the shaft
- Torsional stiffness or dimensions of the shaft
- Material of the shaft including tensile strength and modulus of rigidity
- Drawing and revision number of the diagram or drawing

18.7.5 Flexible coupling/clutch

If not supplied by Wärtsilä the following data must be informed:

- Mass moments of inertia of all parts of the coupling
- Number of flexible elements
- Linear, progressive or degressive torsional stiffness per element
- Dynamic magnification or relative damping
- Nominal torque, permissible vibratory torque and permissible power loss

- Drawing of the coupling showing make, type and drawing and revision number

18.7.6 Operational data

- Operational profile (load distribution over time)
- Clutch-in speed
- Power distribution between the different users
- Power speed curve of the load

18.8 Turning gear

The engine can be turned with a manual ratchet tool after engaging a gear wheel on the flywheel gear rim. The ratchet tool is provided with the engine.

19. Engine room layout

19.1 Crankshaft distances

Minimum crankshaft distances have to be followed in order to provide sufficient space between engines for maintenance and operation.

Figure 19.1 Minimum crankshaft distances, main engine (DAAE006291)

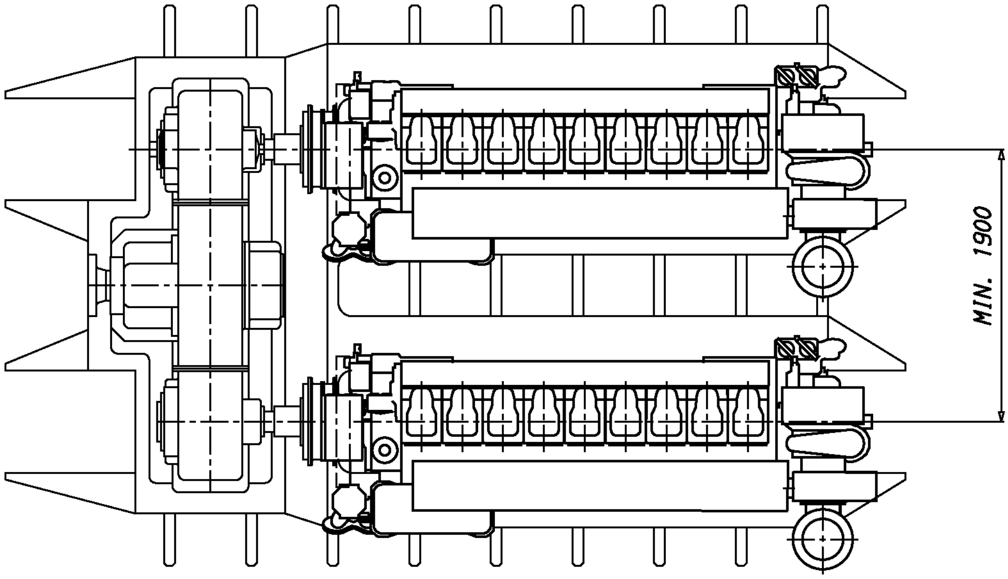
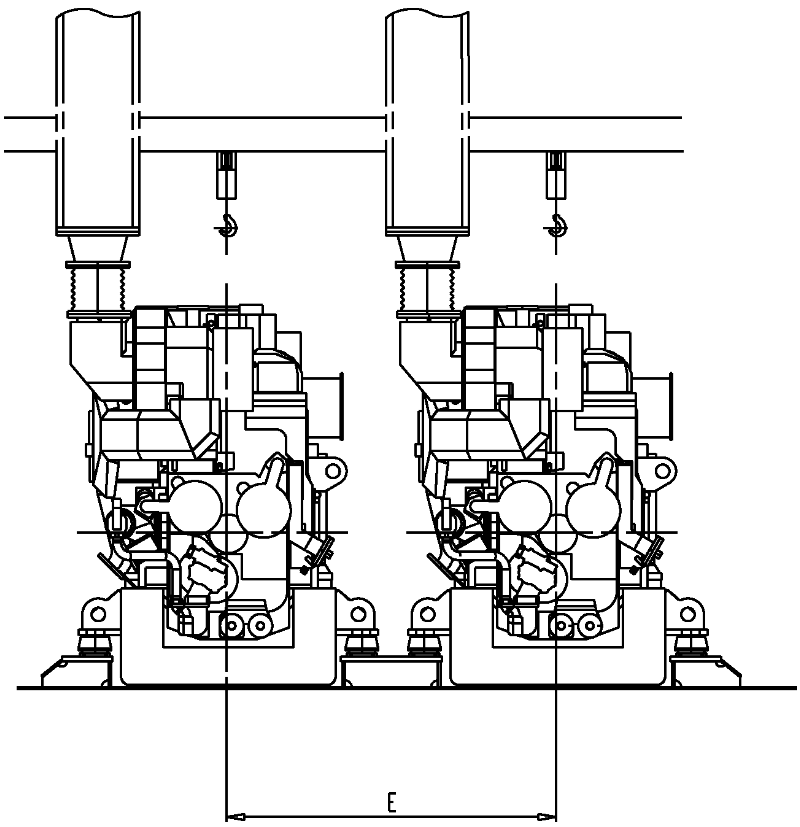


Figure 19.2 Minimum crankshaft distances, generating sets (DAAE007434)



Engine	E
4L20	1970 / 2020
6L20	1970 / 2020
8L20	2020
9L20	2020

19.2 Space requirements for maintenance

19.2.1 Working space reservation

The required working space around the engine is mainly determined by the dismounting dimensions of some engine components, as well as space requirement of some special tools. It is especially important that no obstructive structures are built next to engine driven pumps, as well as camshaft and crankcase doors.

However, also at locations where no space is required for any engine part dismounting, a minimum of 1000 mm free space everywhere around the engine is recommended to be reserved for maintenance operations.

19.2.2 Lifting equipment

It is essential for efficient and safe working conditions that the lifting equipment are applicable for the job and they are correctly dimensioned and located.

The required engine room height depends on space reservation of the lifting equipment and also on the lifting and transportation arrangement. The minimum engine room height can be achieved if there is enough transversal and longitudinal space, so that there is no need to transport parts over insulation box or rocker covers.

Separate lifting arrangement for overhauling turbocharger is required (unless overhead travelling crane, which also covers the turbocharger is used). Turbocharger lifting arrangement is usually best handled with a chain block on a rail located above the turbocharger axis.

See Chapter *General data and outputs* for the necessary hook heights.

19.3 Handling of spare parts and tools

Transportation arrangement from engine room to storage and workshop has to be prepared for heavy engine components. This can be done with several chain blocks on rails or alternatively utilising pallet truck or trolley. If transportation must be carried out using several lifting equipment, coverage areas of adjacent cranes should be as close as possible to each other.

Engine room maintenance hatch has to be large enough to allow transportation of main components to/from engine room.

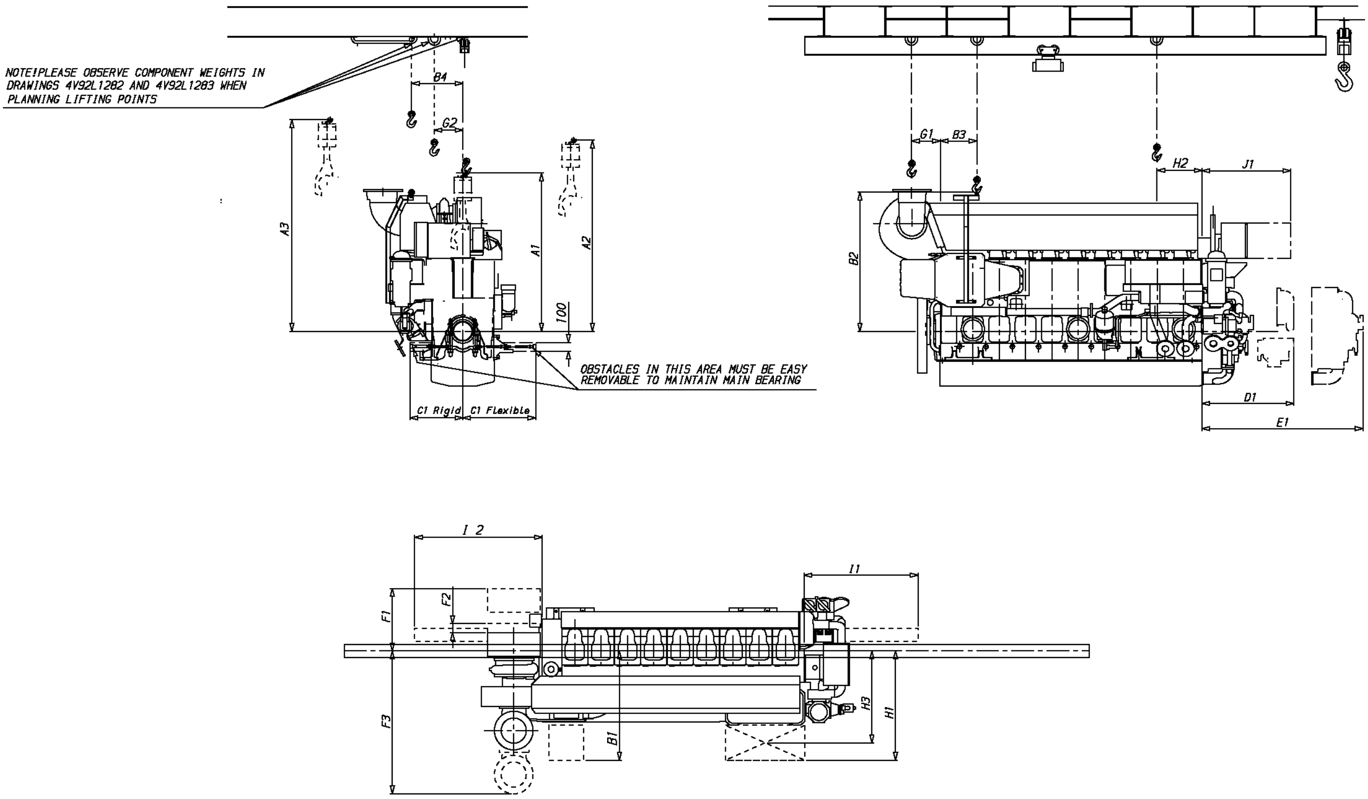
It is recommended to store heavy engine components on slightly elevated adaptable surface e.g. wooden pallets. All engine spare parts should be protected from corrosion and excessive vibration.

On single main engine installations it is important to store heavy engine parts close to the engine to make overhaul as quick as possible in an emergency situation.

19.4 Required deck area for service work

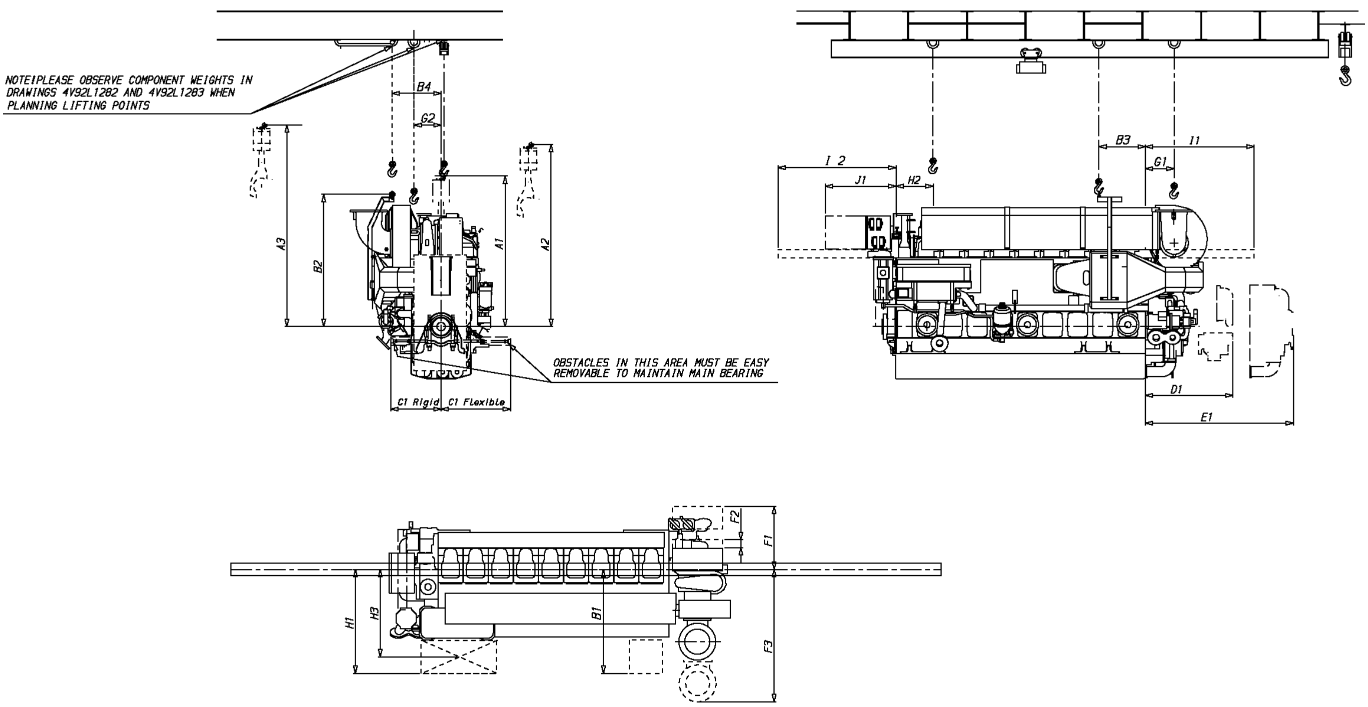
During engine overhaul some deck area is required for cleaning and storing dismantled components. Size of the service area is dependent of the overhauling strategy chosen, e.g. one cylinder at time, one bank at time or the whole engine at time. Service area should be plain steel deck dimensioned to carry the weight of engine parts.

Figure 19.3 Service space for W20 engine with turbocharger in driving end (1V69C0301a)



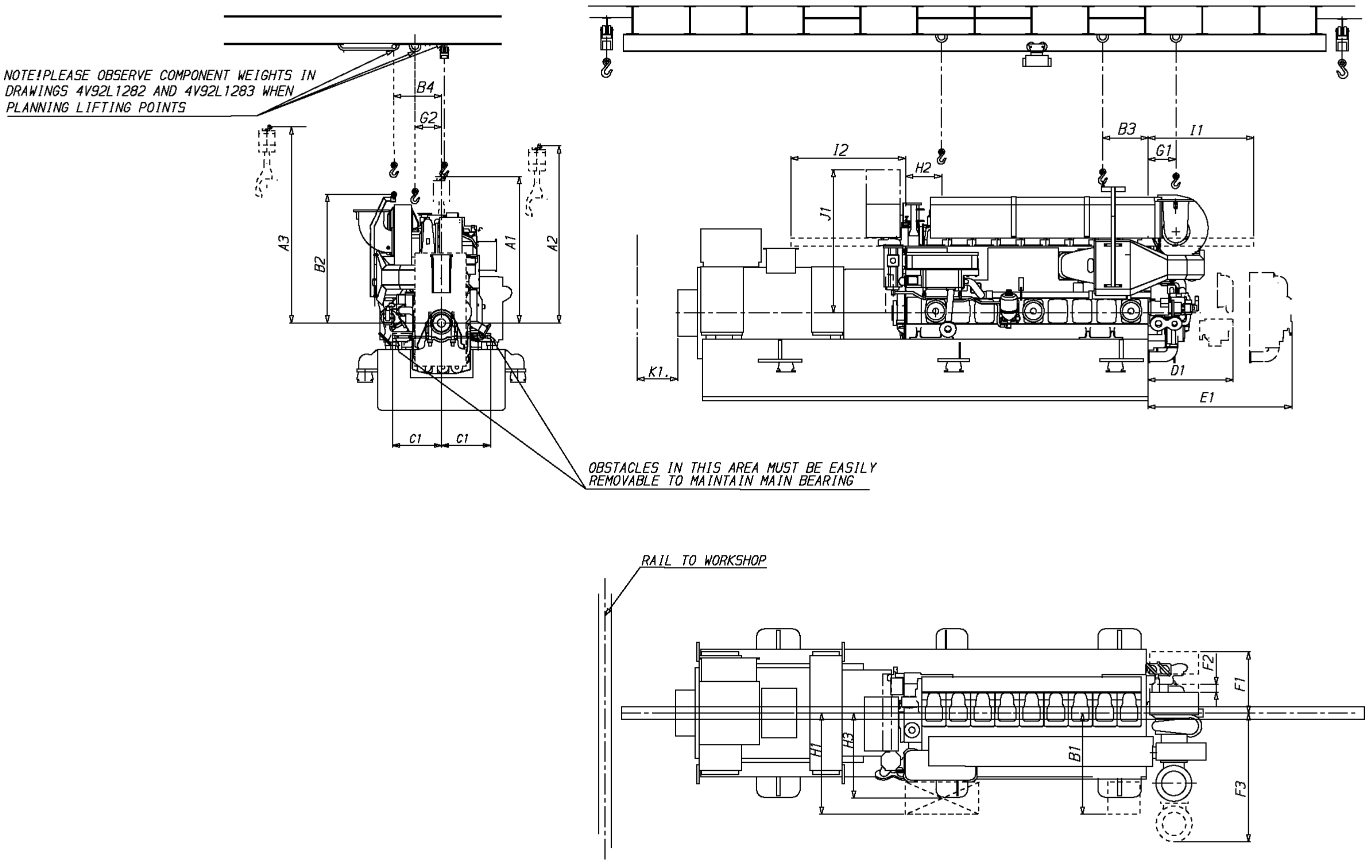
Engine		6L20	8L20	9L20
A1	Height for overhauling piston and connecting rod	1800		
A2	Height for transporting piston and connecting rod freely over adjacent cylinder head covers	2300		
A3	Height for transporting piston and connecting rod freely over exhaust gas insulation box	2300	2400	2400
B1	Width for dismantling charge air cooler and air inlet box sideways by using lifting tool	1200		
B2	Height of the lifting eye for the charge air cooler lifting tool	1580		
B3	Recommended lifting point for charge air cooler lifting tool	390		
B4	Recommended lifting point for charge air cooler lifting tool	590		
C1	Removal of main bearing side screw, flexible / rigid mounting	800 / 560		
D1	Distance needed for dismantling lubricating oil and water pumps	635		
E1	Distance needed for dismantling pump cover with fitted pumps	With PTO: lenght + 515 mm Without PTO: 650 mm		
F1	The recommended axial clearance for dismantling and assembly of silencers. Minimum axial clearance: 100 mm (F2)	650	710	710
F3	Recommended distance for dismantling the gas outlet elbow	990	1170	1170
G1	Recommended lifting point for the turbocharger	300		
G2	Recommended lifting point sideways for the turbocharger	345		
H1	Width for dismantling lubricating oil module and/or plate cooler	1250		
H2	Recommended lifting point for dismantling lubricating oil module and/or plate cooler	445		
H3	Recommended lifting point sideways for dismantling lubricating oil module and/or plate cooler	1045		
I1	Camshaft overhaul distance (free end)	1000	1300	1300
I2	Camshaft overhaul distance (flywheel end)	1000	1300	1300
J1	Space necessery for access to the connection box	1010		

Figure 19.4 Service space for W20 engine with turbocharger in free end (1V69C0302a)



<table>
<tr><th colspan="2">Engine</th><th>4L20</th><th>6L20</th><th>8L20</th><th>9L20</th></tr>
<tr><td>A1</td><td>Height for overhauling piston and connecting rod</td><td colspan="4">1800</td></tr>
<tr><td>A2</td><td>Height for transporting piston and connecting rod freely over adjacent cylinder head covers</td><td colspan="4">2300</td></tr>
<tr><td>A3</td><td>Height for transporting piston and connecting rod freely over exhaust gas insulation box</td><td>2230</td><td>2300</td><td>2400</td><td>2400</td></tr>
<tr><td>B1</td><td>Width for dismantling charge air cooler and air inlet box sideways by using lifting tool</td><td colspan="4">1200</td></tr>
<tr><td>B2</td><td>Height of the lifting eye for the charge air cooler lifting tool</td><td colspan="4">1580</td></tr>
<tr><td>B3</td><td>Recommended lifting point for charge air cooler lifting tool</td><td>260</td><td>550</td><td>550</td><td>550</td></tr>
<tr><td>B4</td><td>Recommended lifting point for charge air cooler lifting tool</td><td colspan="4">560</td></tr>
<tr><td>C1</td><td>Removal of main bearing side screw, flexible / rigid mounting</td><td colspan="4">800 / 560</td></tr>
<tr><td>D1</td><td>Distance for dismantling lubricating oil and water pump</td><td colspan="4">635</td></tr>
<tr><td>E1</td><td>Distance for dismantling pump cover with fitted pumps</td><td colspan="4">With PTO: lenght + 515 mm
Without PTO: 650 mm</td></tr>
<tr><td>F1</td><td>The recommended axial clearance for dismantling and assembly of silencers. Minimum axial clearance: 100 mm (F2)</td><td>590</td><td>650</td><td>750</td><td>750</td></tr>
<tr><td>F3</td><td>Recommended distance for dismantling the gas outlet elbow</td><td>890</td><td>990</td><td>1120</td><td>1120</td></tr>
<tr><td>G1</td><td>Recommended lifting point for the turbocharger</td><td colspan="4">350</td></tr>
<tr><td>G2</td><td>Recommended lifting point sideways for the turbocharger</td><td colspan="4">320</td></tr>
<tr><td>H1</td><td>Width for dismantling lubricating oil module and/or plate cooler</td><td colspan="4">1250</td></tr>
<tr><td>H2</td><td>Recommended lifting point for dismantling lubricating oil module and/or plate cooler</td><td colspan="4">445</td></tr>
<tr><td>H3</td><td>Recommended lifting point sideways for dismantling lubricating oil module and/or plate cooler</td><td colspan="4">1045</td></tr>
<tr><td>I1</td><td>Camshaft overhaul distance (free end)</td><td>700</td><td>1000</td><td>1300</td><td>1300</td></tr>
<tr><td>I2</td><td>Camshaft overhaul distance (flywheel end)</td><td>700</td><td>1000</td><td>1300</td><td>1300</td></tr>
<tr><td>J1</td><td>Space necessery for access to the connection box</td><td colspan="4">845</td></tr>
</table>

Figure 19.5 Service space for W20 based gensets (DAAE006367)



Engine		4L20	6L20	8L20	9L20
A1	Height for overhauling piston and connecting rod	1800			
A2	Height for transporting piston and connecting rod freely over adjacent cylinder head covers	2300			
A3	Height for transporting piston and connecting rod freely over exhaust gas insulation box	2230	2300	2400	2400
B1	Width for dismantling charge air cooler and air inlet box sideways by using lifting tool	1200			
B2	Height of the lifting eye for the charge air cooler lifting tool	1580			
B3	Recommended lifting point for charge air cooler lifting tool	260	550	550	550
B4	Recommended lifting point for charge air cooler lifting tool	560			
C1	Width for removing main bearing side screw	560			
D1	Distance needed to dismantle lube oil and water pump	635			
E1	Distance needed to dismantle pump cover with fitted pumps	650			
F1	The recommended axial clearance for dismantling and assembly of silencers Minimum axial clearance: 100 mm (F2)	590	650	750	750
F3	Recommended distance for dismantling the gas outlet elbow	890	990	1120	1120
G1	Recommended lifting point for the turbocharger	350			
G2	Recommended lifting point sideways for the turbocharger	320			
H1	Width for dismantling lube oil module	1250 mm (and/or plate cooler)			
H2	Recommended lifting point for dismantling lube oil module	445 mm (and/or plate cooler)			
H3	Recommended lifting point sideways for dismantling lube oil module	1045 mm (and/or plate cooler)			
I1	Camshaft overhaul distance (free end)	700	1000	1300	1300
I2	Camshaft overhaul distance (flywheel end)	700	1000	1300	1300
J1	Space necessary for access to the connection box	1762			
K1	Service space for generator	500			

20. Transport dimensions and weights

20.1 Lifting of engines

Figure 20.1 Lifting of generating sets (3V83D0300c)

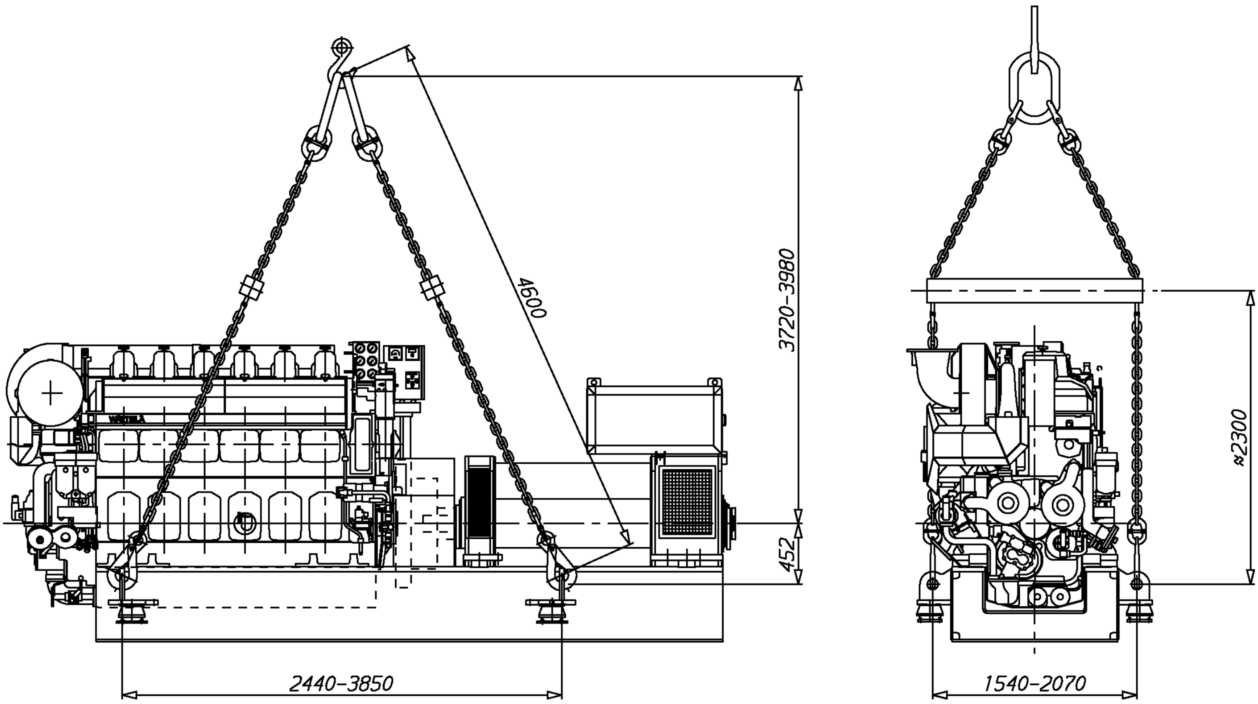
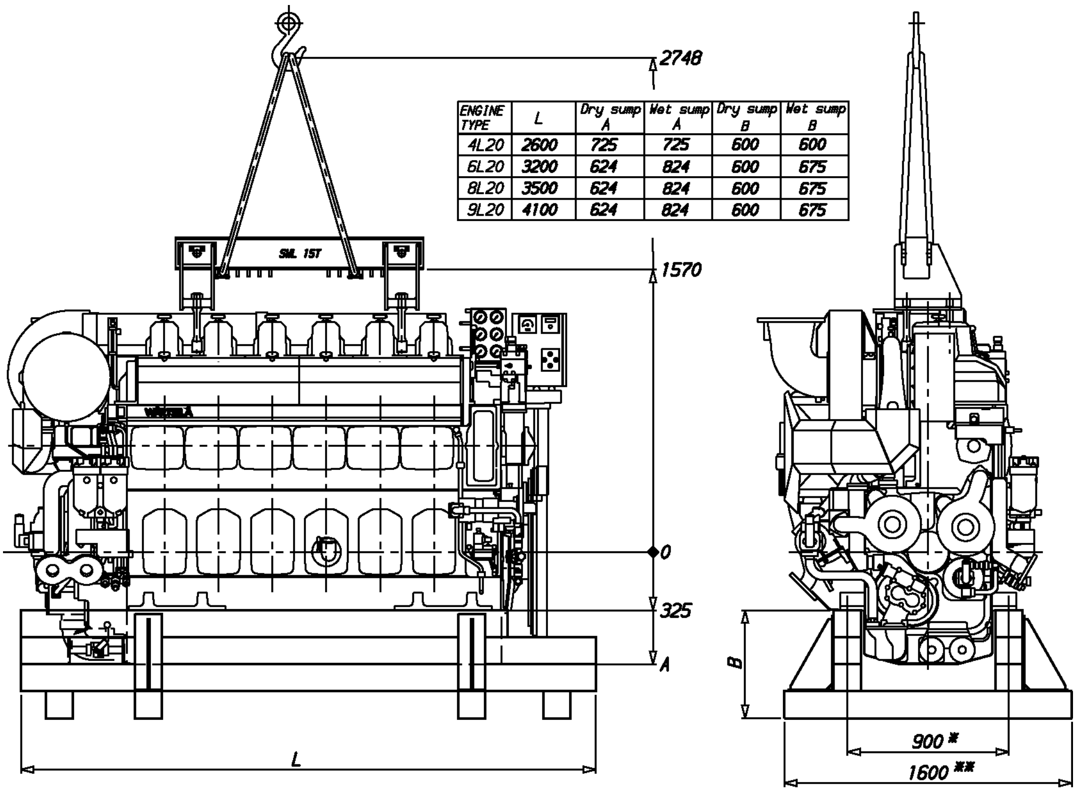


Figure 20.2 Lifting of main engines (3V83D0285c)



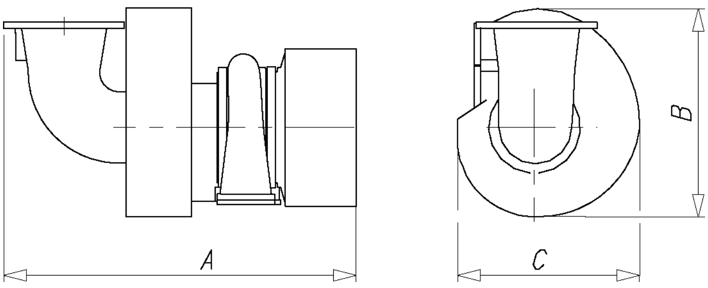
ENGINE TYPE	L	Dry sump A	Wet sump A	Dry sump B	Wet sump B
4L20	2600	725	725	600	600
6L20	3200	624	824	600	675
8L20	3500	624	824	600	675
9L20	4100	624	824	600	675

*) 1020 FOR FLEXIBLE MOUNTED ENGINE
**) 1720 FOR FLEXIBLE MOUNTED ENGINE

20.2 Weights of engine components

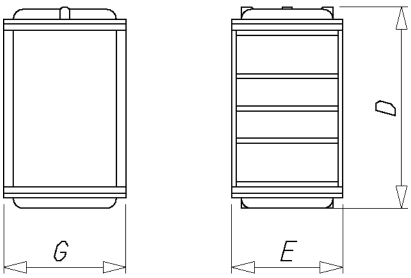
20.2.1 Turbocharger and cooler inserts

Turbocharger



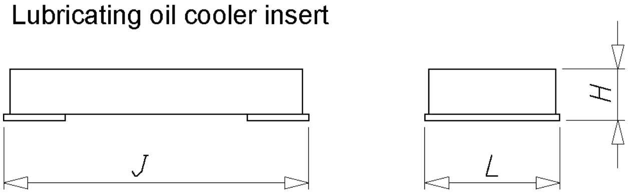
Engine type	A [mm]	B [mm]	C [mm]	Weight [kg]
4L20	945	576	480	150
6L20	1097	636	568	215
8L20	1339	760	675	340
9L20	1339	773	667	340

Charge air cooler



Engine type	D [mm]	E [mm]	G [mm]	Weight [kg]
4L20	616	285	340	120
6L20	626	345	380	160
8L20	626	345	380	160
9L20	626	345	380	160

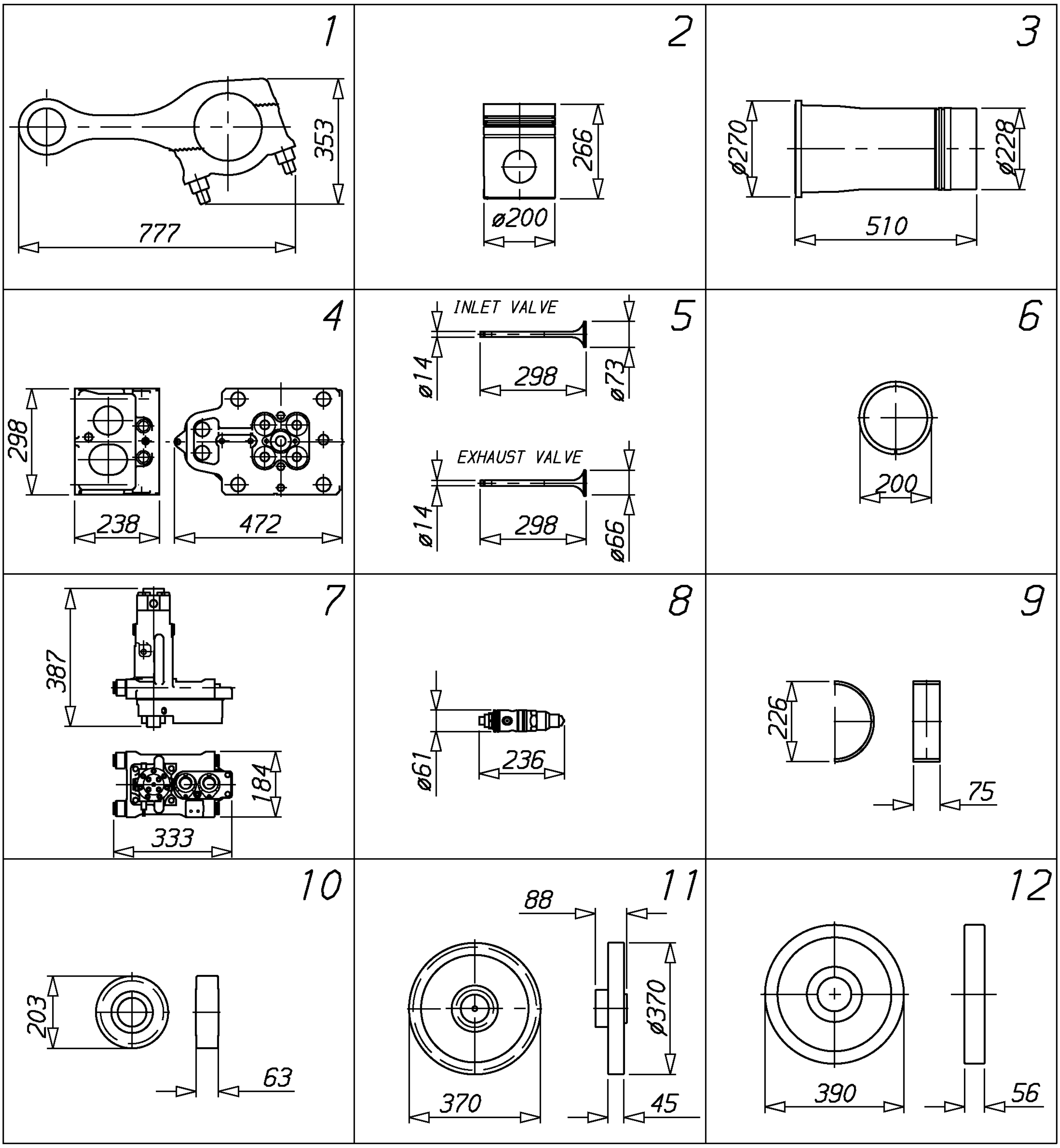
Lubricating oil cooler insert



Engine type	H [mm]	J [mm]	L [mm]	Weight [kg]
4L20	150	896	396	80
6L20	196	896	396	100
8L20	247	896	396	125
9L20	275	896	396	140

20.2.2 Major spare parts

Figure 20.3 Major spare parts (4V92L1283)



Item	Weight/kg	Item	Weight/kg
1 Connecting rod	39	7 Injection pump	27
2 Piston	21	8 Injection valve	3.2
3 Cylinder liner	41	9 Main bearing shell	1.4
4 Cylinder head	94	10 Smaller intermediate gear	11.4
5 Valve	0.8	11 Bigger intermediate gear	23.5
6 Piston ring	0.2	12 Camshaft drive gear	25

21. Dimensional drawings

Dimensional drawings can be found in the CD-ROM included in the back cover pocket of this project guide. The drawing formats are Adobe portable document file (.pdf) and AutoCAD (.dxf).

List of the drawings:	
3V58E0584a	4L20 Generating set, Turbocharger in flywheel end
3V58E0553c	6L20 Generating set, Turbocharger in flywheel end
3V58E0552c	8L20 Generating set, Turbocharger in flywheel end
3V58E0549c	9L20 Generating set, Turbocharger in flywheel end
4V58E0541f	6L20 Main engine, Turbocharger in driving end
4V58E0533d	8L20 Main engine, Turbocharger in driving end
4V58E0534e	9L20 Main engine, Turbocharger in driving end
4V58E0564	4L20 Main engine, Turbocharger in flywheel end
4V58E0560c	6L20 Main engine, Turbocharger in flywheel end
4V58E0559c	8L20 Main engine, Turbocharger in flywheel end
4V58E0561b	9L20 Main engine, Turbocharger in flywheel end

21.1 Notes for the CD-ROM

Hardware requirements:

- CD-ROM drive

Software requirements:

- Adobe Acrobat Reader 4.0 or later or other application capable of reading the files
- AutoCAD 13 or later or other application capable of reading the files.

The files are organized in folders according to the engine types.

22. ANNEX

22.1 Ship inclination angles

Table 22.1 Inclination angels at which main and auxiliary machinery is to operate satisfactorily

Classification society	Lloyd's Register of Shipping	Det Norske Veritas	American Bureau of Shipping	Germanischer Lloyd	Bureau Veritas
Rules referred	2002	2003	2003	2003	2003
Paragraphs where referenced	Pt.5 Ch.1 Sec.3 Par.3.6 Pt.6 Ch.2 Sec1 Par.1.9	Pt.4 Ch.1 Sec.3 Par.B200 Pt.4 Ch.4 Sec.2 Par.A101	Pt.4 Ch.1 Sec.1 Par.7.9	Pt.1 Ch.2 Sec.1 Par.C1.1 Pt.1 Ch.2 Sec.1 Par.E1.1	Pt.C Ch.1 Sec.1 Par.2.4.1 Pt.C Ch.2 Sec.2 Par.1.6.1

Classification society	Russian Maritime Reg. of Shipping	Registro Italiano Navale	China Classification Society	Indian Register of Shipping
Rules referred	2000	2001	2002	1999
Paragraphs where referenced	VII-2.3	Pt.C Ch.2 Sec.2.1.6.1	Pt.III Ch.1 Sec.1.1.3.1	Pt.4 Ch.1.7.1

Main and auxiliary engines			
Heel to side	15		
Rolling to each side	22.5 ****		
Ship length, L	L < 100	L > 100	Ship length is used for LR, DNV and CCS
Trim	5	500/L	Other Classes have constant trim of 5 degrees
Pitching	7.5 ****		

Emergency sets	
Heel to each side	22.5 *
Rolling to each side	22.5
Trim	10
Pitching	10

Electrical installation **			
Heel to each side	15		
Rolling to each side	22.5 ***		
Ship length, L	L < 100	L > 100	Ship length is used for LR, DNV and CCS
Trim	5	500/L	Other Classes have constant trim of 5 degrees
Pitching	7.5		

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

** In ships for the carriage of liquefied gases and of chemicals the emergency power supply must also remain operable to a final inclination up to a maximum of 30 degrees.*

*** Not emergency equipment*

**** DNV, ABS, RINA, GL and BV stipulate that up to an angle of 45 degrees no undesired switching or functional operations may occur. IRS stipulates that no undesired switching or functional operations may occur without stating an angle.*

***** RINA states that the rolling period of 10 s and pitching period of 5 s.*

22.2 Unit conversion tables

Table 22.2 Length conversion table, equations are accurate.

Length	m	in	ft	mile	nautical mile
m	1	1/0.0254	1/(12*0.0254)	1/(0.0254*63360)	1/1852
in	0.0254	1	1/12	1/(63360)	0.0254/1852
ft	0.0254*12	12	1	1/(5280)	12*0.0254/1852
mile	0.0254*63360	63360	5280	1	63360*0.0254/1852
nautical mile	1852	1852/0.0254	1852/(12*0.0254)	1852/(63360*0.0254)	1

Table 22.3 Area conversion, equations are accurate.

Area	square m	square inch	square foot
square m	1	$1/0.0254^2$	$1/(12*0.0254)^2$
square inch	0.0254^2	1	1/144
square foot	$(12*0.0254)^2$	144	1

Table 22.4 Volume conversion, equations are accurate but some of them are reduced in order to limit the number of decimals.

Volume	cubic m	l (liter)	cubic inch	cubic foot	Imperial gallon	US gallon
cubic m	1	1000	$1/0.0254^3$	$1/(12*0.0254)^3$	1/0.00454609	$1/(231*0.0254^3)$
l (liter)	0.001	1	$1/0.254^3$	$1/(12*0.254)^3$	1/4.54609	$1/(231*0.254^3)$
cubic inch	0.0254^3	0.254^3	1	$1/12^3$	$0.254^3/4.54609$	1/231
cubic foot	$(12*0.0254)^3$	$(12*0.254)^3$	12^3	1	$(12*0.254)^3/4.54609$	$12^3/231$
Imperial gallon	0.00454609	4.54609	$4.54609/0.254^3$	$4.54609/(12*0.0254)^3$	1	$4.54609/(231*0.254^3)$
US gallon	$231*0.0254^3$	$231*0.254^3$	231	$231/12^3$	$231*0.254^3/4.54609$	1

Table 22.5 Energy conversion, values are rounded to five significant digits where not accurate.

Energy	J	BTU	cal	lbf ft
J	1	$9.4781*10^{-4}$	0.23885	0.73756
BTU	1055.06	1	252.00	778.17
cal	4.1868	$3.9683*10^{-3}$	1	0.32383
lbf ft	1.35582	$1.2851*10^{-3}$	3.0880	1

Table 22.6 Mass conversion, values are rounded to five significant digits where not accurate.

Mass	kg	lb	oz
kg	1	2.2046	35.274
lb	0.45359	1	16
oz	0.028350	0.0625	1

Table 22.7 Density conversion, values are rounded to five significant digits where not accurate.

Density	kg / cubic m	lb / US gallon	lb / Imperial gallon	lb / cubic ft
kg / cubic m	1	0.0083454	0.010022	0.062428
lb / US gallon	119.83	1	0.83267	0.13368
lb / Imperial gallon	99.776	1.2009	1	0.16054
lb / cubic ft	16.018	7.4805	6.2288	1

Table 22.8 Power conversion table, values are rounded to five significant digits where not accurate.

Power	W	hp	US hp
W	1	0.0013596	0.0013410
hp	735.499	1	1.0136
US hp	745.7	0.98659	1

Table 22.9 Pressure conversion, values are rounded to five significant digits where not accurate.

Pressure	Pa	bar	mmWG	psi
Pa	1	0.00001	0.10197	0.00014504
bar	100000	1	10197	14.504
mmWG	9.80665	9.80665×10^{-5}	1	0.0014223
psi	6894.76	0.0689476	703.07	1

Table 22.10 Mass flow conversion, values are rounded to five significant digits where not accurate.

Mass flow	kg / s	lb / s
kg / s	1	2.2046
lb / s	0.45359	1

Table 22.11 Volume flow conversion, values are rounded to five significant digits where not accurate.

Volume flow	cubic m / s	l / min	cubic m / h	cubic in / s	cubic ft / s	cubic ft / h	USG / s	USG / h
cubic m / s	1	60000	3600	61024	35.315	127133	264.17	951019
l / min	1.6667×10^{-5}	1	0.06	0.98322	1699.0	0.47195	227.12	0.063090
cubic m / h	0.00027778	16.667	1	0.058993	101.94	0.028317	13.627	0.0037854
cubic in / s	1.6387×10^{-5}	1.0171	16.951	1	1728	0.48	231	0.064167
cubic ft / s	0.028317	0.00058858	0.0098096	0.00057870	1	0.00027778	0.13368	3.7133×10^{-5}
cubic ft / h	7.8658×10^{-6}	2.1189	35.315	2.0833	3600	1	481.25	0.13368
USG / s	0.0037854	0.0044029	0.073381	0.0043290	7.4805	0.0020779	1	0.00027778
USG / h	1.0515×10^{-6}	15.850	264.17	15.584	26930	7.4805	3600	1

Temperature

Below are the most common temperature conversion formulas:

- °C = value[K] - 273.15
- °C = 5 / 9 * (value[F] - 32)
- K = value[°C] + 273.15
- K = 5 / 9 * (value[F] - 32) + 273.15
- F = 9 / 5 * value[°C] + 32
- F = 9 / 5 * (value[K] - 273.15) + 32

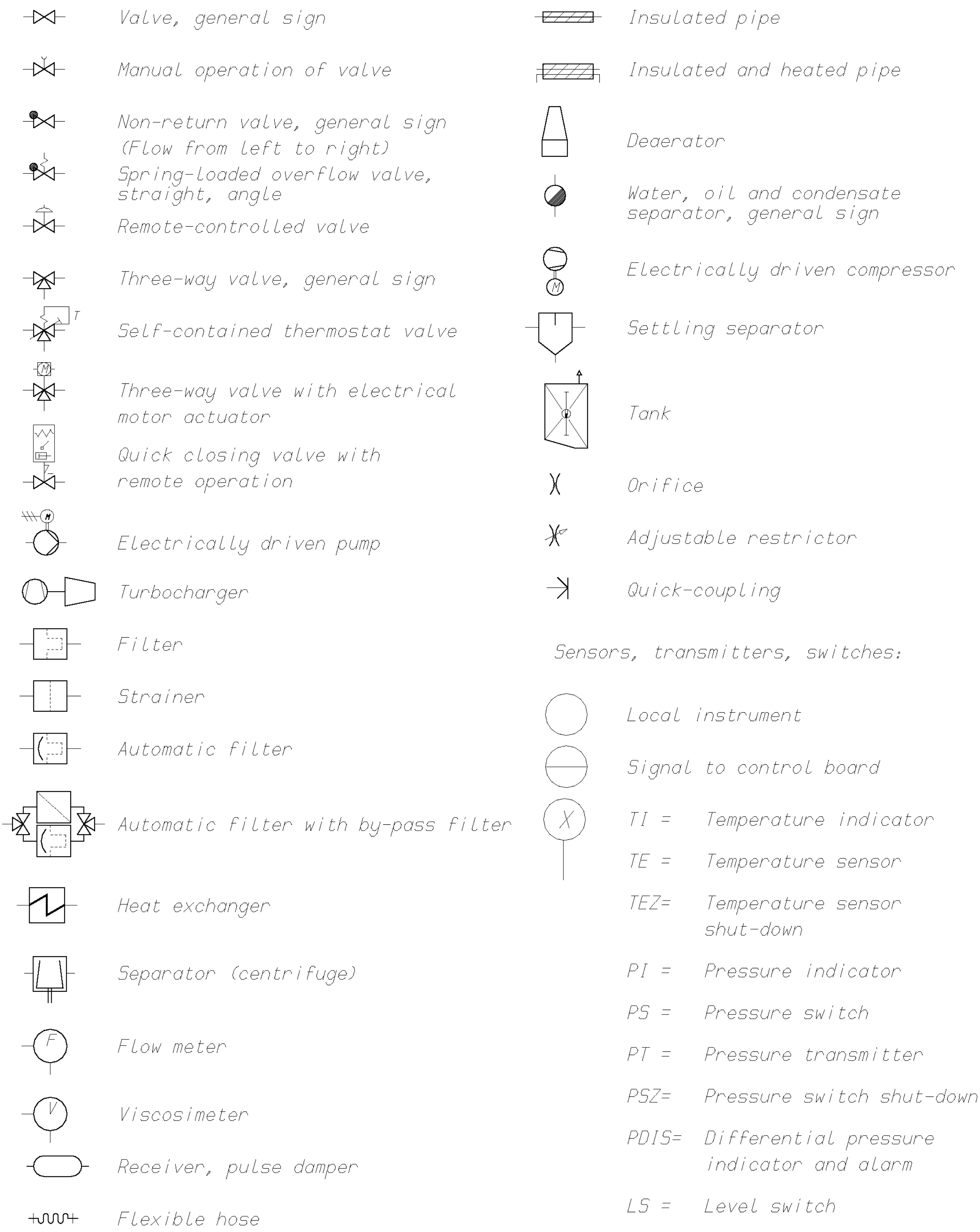
Prefix

Below are the most common prefix multipliers:

Name	Symbol	Factor
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}

22.3 Collection of drawing symbols used in drawings

Figure 22.1 List of symbols (4V92A1402a)



Wärtsilä Finland Oy
Ship Power
P.O Box 252
65101 Vaasa, Finland

Tel: +358 10 709 0000
Fax: +358 6 356 7188
www.wartsila.com