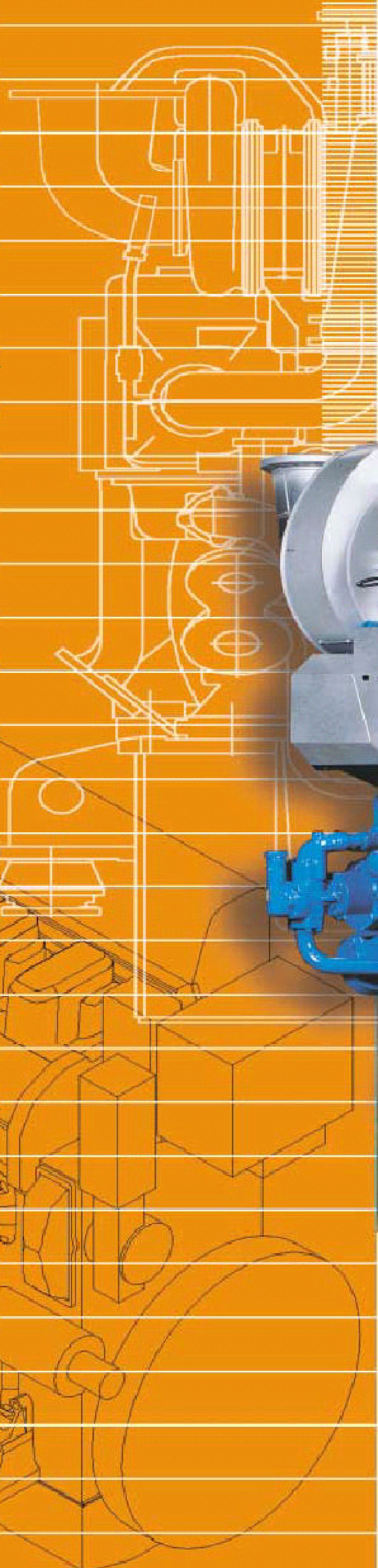
WÄRTSILÄ 20

Project guide for

Marine Applications



WÄRTSILÄ

Introduction

This Project Guide provides engine 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 1/2002 issue replaces all previous issues of the Wärtsilä 20 Project Guides. Numerous revisions have been made. Also the structure of this Project Guide has been amended.

Wärtsilä Finland Oy
Marine & Licensing
Application Technology

Vaasa, January 2002

Table of Contents

1. General 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, 5, 6, 8, 9, in-line
Direction of rotation	Clockwise, counter-clockwise on request

1.2. Maximum continuous output

The mean effective pressure Pe can be calculated as follows:

$$P_e\,[\text{bar}] = \frac{P[\text{kW}] \cdot c \cdot 1.2 \cdot 10^9}{D^2 \cdot L \cdot n \cdot cyl \cdot \pi}$$

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]

cyl = number of cylinders

c = operating cycle (4)

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	720	980
5L20	825	1120
6L20	1080	1470
8L20	1440	1960
9L20	1620	2200

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 alternator outputs are calculated for an efficiency of 0.95 and a power factor of 0.8.

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 the ISO standard.

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	680	810	720	855
5L20					775	920	825	980
6L20	780	930	810	960	1020	1210	1080	1280
8L20	1040	1240	1080	1280	1360	1615	1440	1710
9L20	1170	1390	1215	1440	1530	1815	1620	1925

The specific fuel consumption is stated in the chapter for Technical data with the reference for the engine driven equipment and the effect they have on the specific fuel consumption. The statement applies to engines operating in ambient conditions according to ISO 3046-1 : 1995(E).

- total barometric pressure 100 kPa
- air temperature 25°C
- relative humidity 30%
- charge air coolant temperature 25°C

For other than ISO 3046-1 conditions the same standard gives correction factors on the fuel oil consumption.

1.3.1. Fuel characteristics

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

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.

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 fuel specification “HFO 2" is base on the ISO 8217:1996(E) standard and covers the fuel categories IS-F-RMA10 - RMK55. Additionally, the engine manufacturer has specified the fuel specification ”HFO 1". This tighter specification is an alternative and by using this specification, longer overhaul intervals of specific engine components are possible. See table in the chapter for Description of the engine.

Table 1.4. HFO Specifications

Property	Unit	Limit HFO 1	Limit HFO 2	Test method ref.
Viscosity, max.	cSt at 50°C cSt at 100°C Redwood No. 1 s at 100°F	55 730 7200	55 730 7200	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

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.

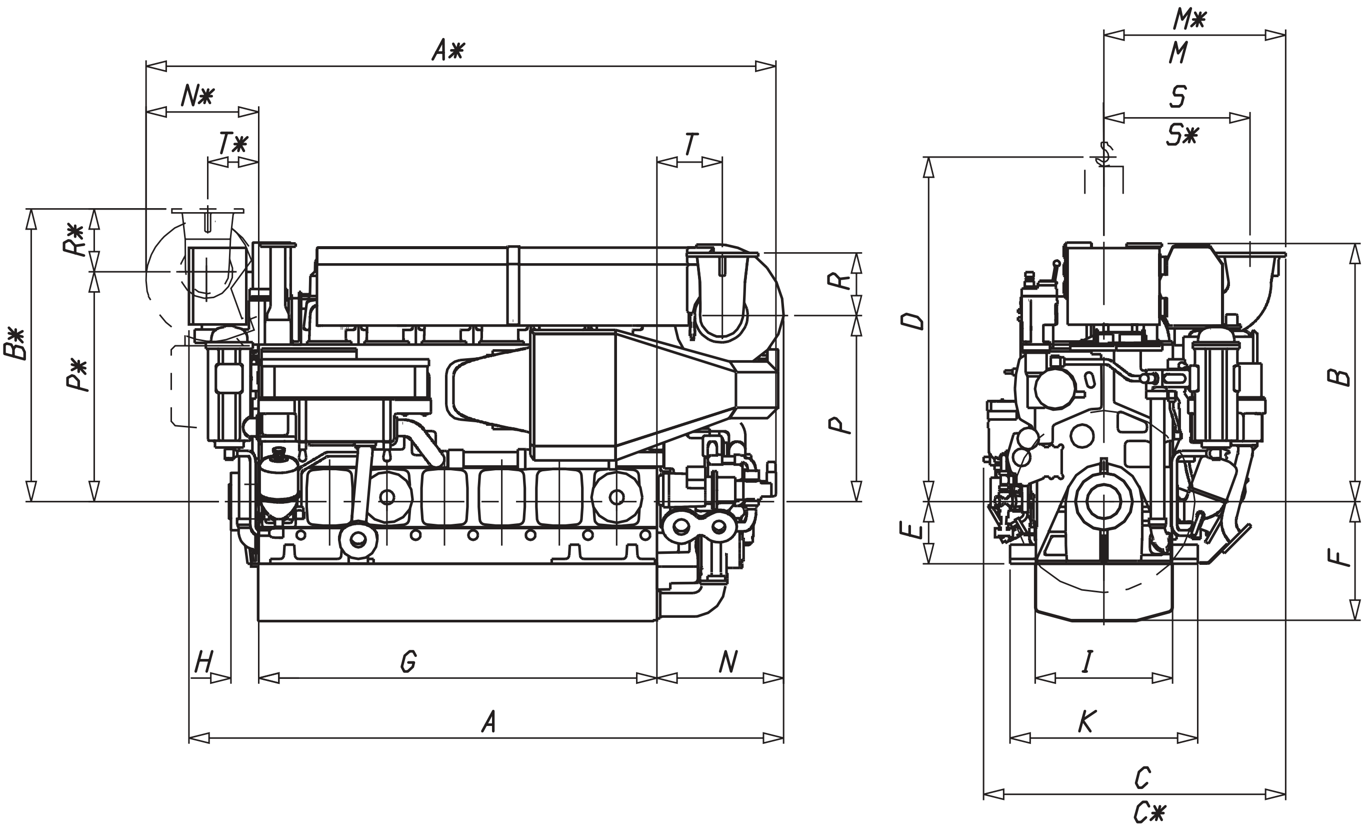
Lubricating oil, foreign substances or chemical waste, hazardous to the safety of the installation or detrimental to the performance of the engines, should not be contained in the fuel.

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

1.4. Principal dimensions and weights

Main engines (3V92E0068b)

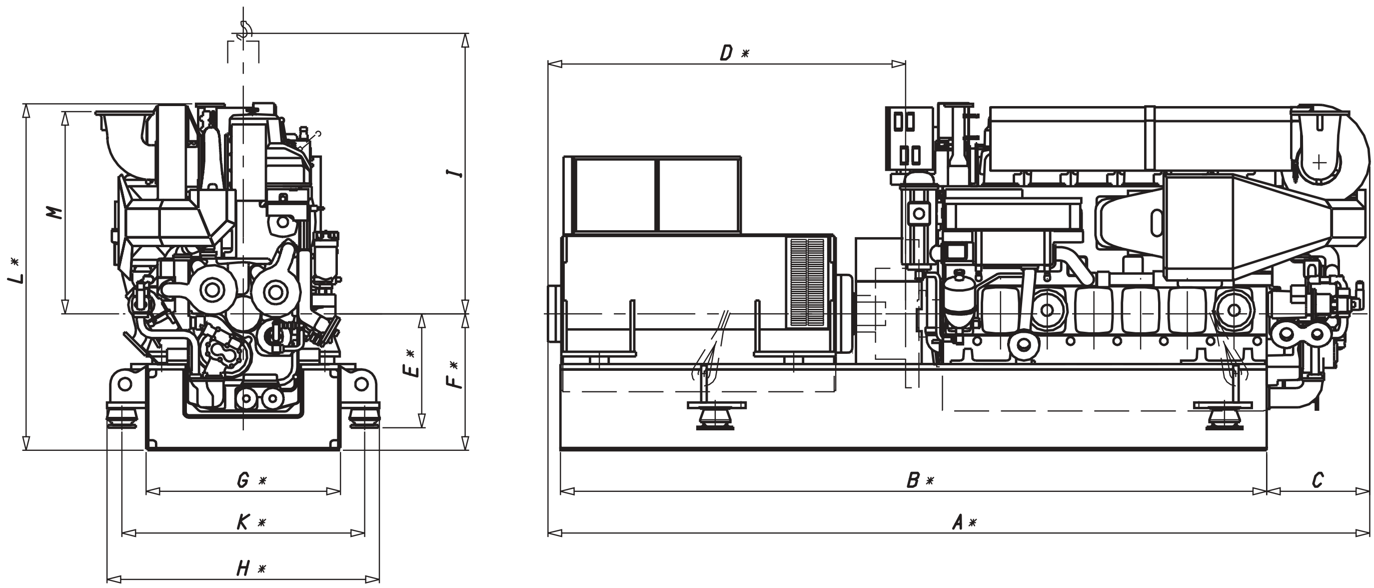


Engine	A*	A	B*	B	C*	C	D	E	F	G	H	I	K
4L20		2510		1348		1483	1800	325	725	1480	155	718	980
5L20		2833		1423		1567	1800	325	725	1780	155	718	980
6L20	3254	3108	1528	1348	1580	1579	1800	325	624	2080	155	718	980
8L20	3973	3783	1614	1465	1756	1713	1800	325	624	2680	155	718	980
9L20	4261	4076	1614	1449	1756	1713	1800	325	624	2980	155	718	980

Engine	M*	M	N*	N	P*	P	R*	R	S*	S	T*	T	Weight **
4L20		854		665		920		248		694		349	7.2
5L20		938		688		1001		328		750		370	7.8
6L20	951	950	589	663	1200	971	328	328	762	763	273	343	9.3
8L20	1127	1084	708	738	1224	1000	390	390	907	863	325	339	11
9L20	1127	1084	696	731	1224	1000	390	390	907	863	325	339	11.6

* Turbocharger at flywheel end
** Weights (in Metric tons) with liquids (wet sump) but without flywheel

Auxiliary engines (3V58E0576a)



ENGINE	A*	B*	C	D*	E*	F*	G*	H*	I	K*	L*	M	Weight [ton]
4L20	4910	4050	665	2460	728	990	1270	1770	1800	1580	2338	1168	14.0
5L20	5190	3945	688	2430	728	1075	1270	1770	1800	1580	2458	1329	15.1
6L20	5290	4540	663	2300	728	895/ 975	1270/ 1420	1770/ 1920	1800	1580/ 1730	2243/ 2323	1299	16.8
8L20	6010	5080	731	2310	728	1025	1420 / 1570	1920/ 2070	1800	1730/ 1880	2474	1390	20.7
9L20	6550	5415	731	2580	728	1075/ 1125	1570/ 1800	2070/ 2300	1800	1880/ 2110	2524/ 2574	1390	23.8

* Values are based on standard alternator, whose type (water or air cooled) and size affects to width, length, height and weight. Weight is based on wet sump engine with engine liquids.

2. Operating ranges

2.1. General

The available 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, and consequently also the cooling effect on the combustion chamber. 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. Some margin is required to permit some reasonable wear and fouling of the turbocharging system and different ambient conditions (e.g. suction air temperature).

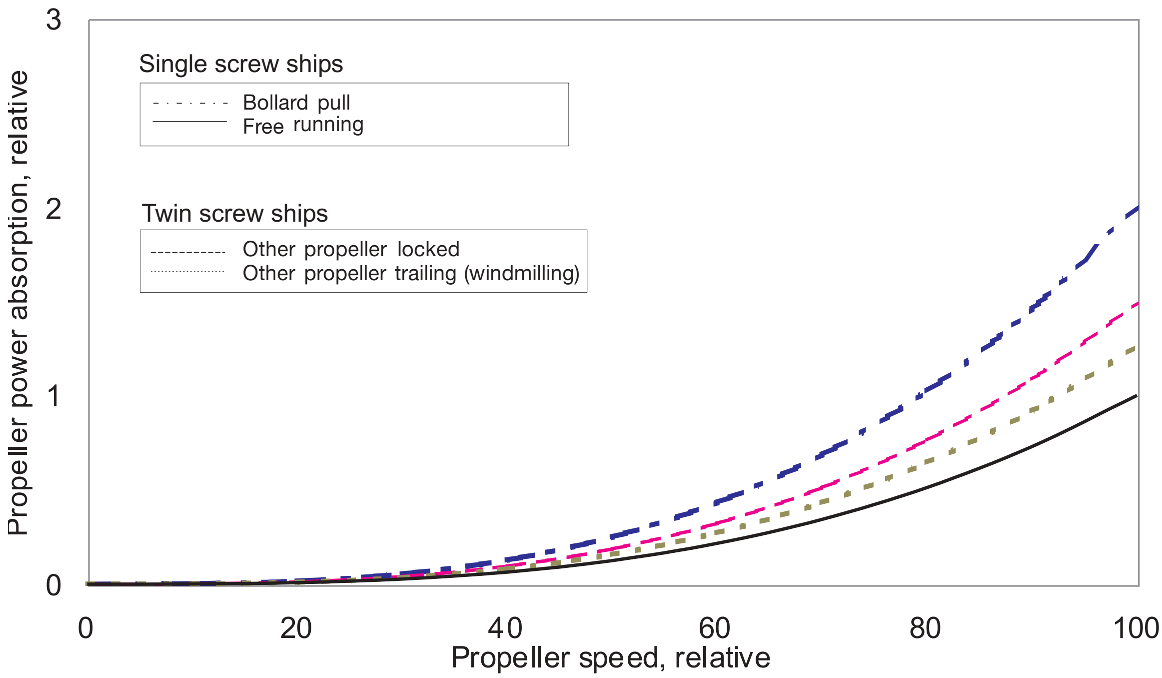
As a rule, the higher the specified mean effective pressure the narrower is the permitted engine operating range. There is a trend in the industry to specify higher and higher outputs, unfortunately on the expense of the width of the operating field. This is the reason why separate operating fields may 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 also a risk for surging of the turbocharger at a certain part load (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.

An FP-propeller should never be specified for a twin-in/single-out 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 issues like 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 optimisation (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 optimise 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).

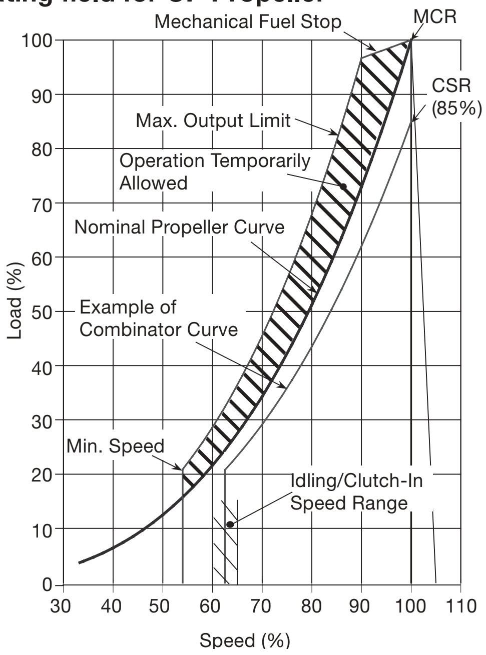
At a given propeller speed and pitch, the ship's speed affects the power absorption of the propeller. This effect is to some extent ship-type specific, being more pronounced on ships with a small block coefficient. The power absorption of the propeller can sometimes be almost twice as high during acceleration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

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.

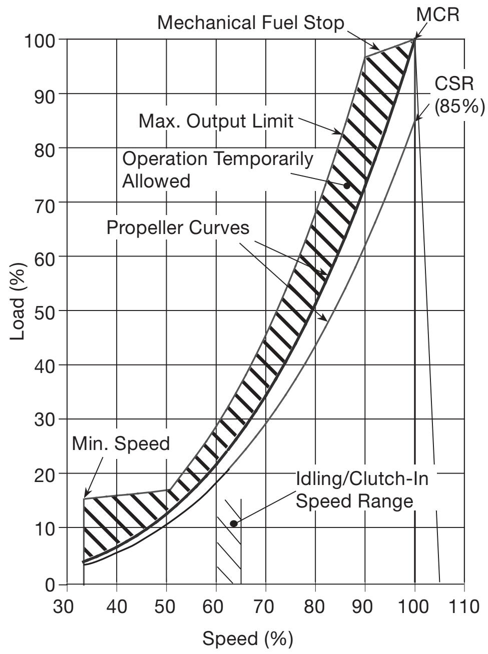
2.2.2. FP-propeller

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.

Operating field for FP Propeller



The engine is non-reversible, so the gear box 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.

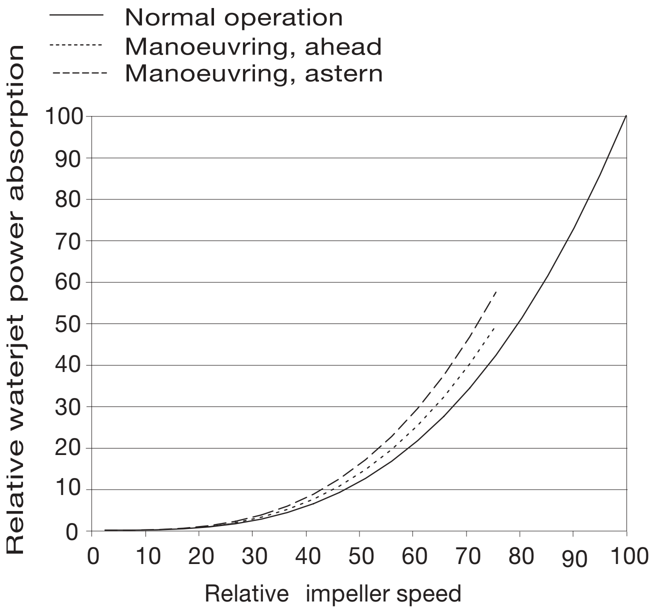
FP-propellers in single-screw ships typically rotate clockwise, requiring a counter clockwise sense of rotation of the engine with a typically single-stage (in the ahead mode) reverse reduction gear.

2.2.3. Water jets

Water jets also requires 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.

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 optimised 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.

This configuration, when the dredging pumps are mechanically driven e.g. from the free end of the crankshaft, some dredging modes may require a capability to run a constant and full torque down to 70 or 80 % of the nominal speed. This kind of torque requirement is difficult to meet with a standard diesel engine and normally de-rating of the main engines is required. The trend in the industry to specify higher and higher outputs has also lead to narrower operating fields, so this configuration is becoming less and less feasible.

2.2.6. Generators

Generators are typically operated at nominal speed. Modern generators are typically 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 gear box has to be selected in steps of complete teeth. The result is illustrated in the example below. In this example, the nominal frequency of the shaft generator is obtained inside the speed range of the main engine.

Table 2.2. Example of relative speeds between engine and PTO-generator.

	Engine speed	Propeller		PTO generator		
		output	speed	speed	frequency	
	%	%	%	rpm	%	Hz
Engine MCR	100	100	100	1512	100.8	50.4
Nominal frequency of PTO generator	99.2	97.6	99.2	1500	100	50
95 % frequency of PTO generator	94.2	83.7	94.2	1425	95	47.5
85 % of engine MCR, propeller law	94.7	85	94.7	1432	95.5	47.7
85 % of engine MCR, constant speed	100	85	100	1512	100.8	50.4

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. The load should always be applied gradually in normal operation.

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. In an emergency situation the engine can be loaded in three equal steps in accordance with class requirement.

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, alternator 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. In an emergency situation the engine can be loaded in three equal steps with minimum 5 seconds between each step. 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. Ambient conditions

2.4.1. 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.

2.4.2. 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 for Combustion air system.

2.4.3. 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.

2.4.4. 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:

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.

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.

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

No restrictions.

3. Technical data tables

Diesel engine Wärtsilä 4L20			ME	AE	AE	AE	AE
Engine speed		RPM	1000	720	750	900	1000
Engine output		kW	720	520	540	680	720
Engine output		HP	980	710	730	920	980
Cylinder bore		mm			200		
Stroke		mm			280		
Swept volume		dm^3			35,2		
Compression ratio					15		
Compression pressure, max.		bar	167	150	150	167	167
Firing pressure, max.		bar	185	170	170	185	185
Charge air pressure at 100% load		bar			0,3		
Mean effective pressure		bar	24,6	24,6	24,6	25,8	24,6
Mean piston speed		m/s	9,3	6,7	7	8,4	9,3
Idling speed		RPM	350				
Combustion air system							
Flow of air at 100% load		kg/s	1,42	0,94	0,99	1,25	1,42
Ambient air temperature, max.		°C			45		
Air temperature after air cooler		°C			45…60		
Air temperature after air cooler, alarm		°C			75		
Exhaust gas system							
Exhaust gas flow (100% load)	3)	kg/s	1,46	0,97	1,02	1,39	1,46
Exhaust gas flow (85% load)	3)	kg/s	1,25	0,84	0,89	1,21	1,28
Exhaust gas flow (75% load)	3)	kg/s	1,1	0,76	0,81	1,08	1,15
Exhaust gas flow (25% load)	3)	kg/s	0,73	0,55	0,59	0,77	0,84
Exhaust gas temp. after turbocharger (100% load)	1) 3)	°C	350	360	360	340	350
Exhaust gas temp. after turbocharger (85% load)	1) 3)	°C	365	360	360	340	350
Exhaust gas temp. after turbocharger (75% load)	1) 3)	°C	370	360	365	340	350
Exhaust gas temp. after turbocharger (50% load)	1) 3)	°C	390	370	370	350	360
Exhaust gas back pressure drop, max.		kPa			3		
Diameter of turbocharger connection		mm			200		
Exhaust gas pipe diameter, min.		mm	300	250	250	300	300
Calculated dia for 35 m/s		mm	305	251	257	295	305
Heat balance	2) 3)						
Jacket water		kW	161	127	132	149	161
Charge air		kW	220	157	161	206	220
Lubricating oil		kW	85	67	69	78	85
Exhaust gases		kW	523	356	374	490	523
Radiation		kW	42	31	31	37	42
Fuel system							
Pressure before injection pumps		kPa (bar)			600(6)		
Pump capacity, MDF, engine driven		m^3/h	0,41	0,87	0,9	0,41	0,41
Fuel consumption (100% load)	3)	g/kWh	195	194	194	193	195
Fuel consumption (85% load)	3)	g/kWh	192	195	195	193	195
Fuel consumption (75% load)	3)	g/kWh	193	197	197	194	196
Fuel consumption (50% load)	3)	g/kWh	200	204	204	200	201
Leak fuel quantity, clean MDF fuel (100% load)		kg/h	0,5	0,4	0,4	0,5	0,5
Lubricating oil system							
Pressure before engine, nom.		kPa (bar)			450 (4,5)		
Pressure before engine, alarm		kPa (bar)			300 (3)		
Pressure before engine, stop		kPa (bar)			200 (2)		

Priming pressure, nom.		kPa (bar)			80 (0,8)		
Priming pressure, alarm		kPa (bar)			50 (0,5)		
Temperature before engine, nom.		°C			63		
Temperature before engine, alarm		°C			80		
Temperature after engine, abt.		°C			78		
Pump capacity (main), engine driven		m^3/h			28		
Pump capacity (main), separate		m^3/h			18		
Pump capacity (priming)	4)	m^3/h			6,9/8,4		
Oil volume, wet sump, nom.		m^3			0,27		
Oil volume in separate system oil tank, nom.		m^3	1	0,7	0,7	0,9	1
Filter fineness, nom.		microns/60%	15	15	15	15	15
Filter difference pressure, alarm		kPa (bar)			150 (1,5)		
Oil consumption (100% load), abt.	5)	g/kWh			0,6		

Cooling water system

High temperature cooling water system

Pressure before engine, nom.		kPa (bar)			200 (2,0) + static		
Pressure before engine, alarm		kPa (bar)			100 (1,0) + static		
Pressure before engine, max.		kPa (bar)			350 (3,5)		
Temperature before engine, abt.		°C			83		
Temperature after engine, nom.		°C			91		
Temperature after engine, alarm		°C			105		
Temperature after engine, stop		°C			110		
Pump capacity, nom.		m^3/h	18,5	18	18,5	18,5	20
Pressure drop over engine		kPa (bar)			50 (0,5)		
Water volume in engine		m^3			0,09		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		

Low temperature cooling water system

Pressure before charge air cooler, nom.		kPa (bar)			200 (2) + static		
Pressure before charge air cooler, alarm		kPa (bar)			100 (1) + static		
Pressure before charge air cooler, max.		kPa (bar)			350 (3,5)		
Temperature before charge air cooler, max.		°C			38		
Temperature before charge air cooler, min.		°C			25		
Pump capacity, nom.		m^3/h	20	19	20	20	24
Pressure drop over charge air cooler		kPa (bar)			30 (0,3)		
Pressure drop over oil cooler		kPa (bar)			30 (0,3)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		

Starting air system

Air supply pressure before engine (max.)		Mpa (bar)			3 (30)		
Air supply pressure, alarm		Mpa (bar)			1,8 (18)		
Air consumption per start (20°C)	6)	Nm^3			0,4		

1) At an ambient temperature of 25°C.
2) The figures are at 100% load and include the 5% tolerance on sfoc and engine driven pumps.
3) According to ISO 3046/1, lower calorific value 42 700 kJ/kg, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.
4) Capacities at 50 and 60 Hz respectively.
5) Tolerance + 0.3 g/kWh
6) At remote and automatic starting, the consumption is 1.2 Nm^3

Subject to revision without notice.

Diesel engine Wärtsilä 5L20

			ME	AE	AE
Engine speed		RPM	1000	900	1000
Engine output		kW	825	775	825
Engine output		HP	1120	1050	1120
Cylinder bore		mm		200	
Stroke		mm		280	
Swept volume		dm^3		44	
Compression ratio				15	
Compression pressure, max.		bar		155	
Firing pressure, max.		bar		175	
Charge air pressure at 100% load		bar		0,3	
Mean effective pressure		bar	22,5	23,5	22,5
Mean piston speed		m/s	9,3	8,4	9,3
Idling speed		RPM	350		
Combustion air system					
Flow of air at 100% load		kg/s	1,5	1,42	1,5
Ambient air temperature, max.		°C		45	
Air temperature after air cooler		°C		45…60	
Air temperature after air cooler, alarm		°C		75	
Exhaust gas system					
Exhaust gas flow (100% load)	3)	kg/s	1,55	1,55	1,55
Exhaust gas flow (85% load)	3)	kg/s	1,33	1,37	1,37
Exhaust gas flow (75% load)	3)	kg/s	1,19	1,25	1,25
Exhaust gas flow (25% load)	3)	kg/s	0,82	0,94	0,94
Exhaust gas temp. after turbocharger (100% load)	1) 3)	°C	360	360	360
Exhaust gas temp. after turbocharger (85% load)	1) 3)	°C	365	350	350
Exhaust gas temp. after turbocharger (75% load)	1) 3)	°C	385	360	360
Exhaust gas temp. after turbocharger (50% load)	1) 3)	°C	395	360	360
Exhaust gas back pressure drop, max.		kPa		3	
Diameter of turbocharger connection		mm		250	
Exhaust gas pipe diameter, min.		mm		350	
Calculated dia for 35 m/s		mm	317	317	317
Heat balance	2) 3)				
Jacket water		kW	189	173	189
Charge air		kW	240	226	240
Lubricating oil		kW	101	91	101
Exhaust gases		kW	602	558	602
Radiation		kW	49	43	49
Fuel system					
Pressure before injection pumps		kPa (bar)		600(6)	
Pump capacity, MDF, engine driven		m^3/h	0,58	0,58	0,58
Fuel consumption (100% load)	3)	g/kWh	195	195	195
Fuel consumption (85% load)	3)	g/kWh		194	
Fuel consumption (75% load)	3)	g/kWh	195	196	196
Fuel consumption (50% load)	3)	g/kWh	202	208	208
Leak fuel quantity, clean MDF fuel (100% load)		kg/h	0,7	0,7	0,7
Lubricating oil system					
Pressure before engine, nom.		kPa (bar)	450 (4,5)	450 (4,5)	450 (4,5)
Pressure before engine, alarm		kPa (bar)	300 (3)	300 (3)	300 (3)
Pressure before engine, stop		kPa (bar)	200 (2)	200 (2)	200 (2)
Priming pressure, nom.		kPa (bar)	80 (0,8)	80 (0,8)	80 (0,8)
Priming pressure, alarm		kPa (bar)	50 (0,5)	50 (0,5)	50 (0,5)
Temperature before engine, nom.		°C		63	

Temperature before engine, alarm		°C		80	
Temperature after engine, abt.		°C		78	
Pump capacity (main), engine driven		m³/h		28	
Pump capacity (main), separate		m³/h		19,5	
Pump capacity (priming)	4)	m³/h		6,9/8,4	
Oil volume, wet sump, nom.		m³		0,32	
Oil volume in separate system oil tank, nom.		m³	1,1	1	1,1
Filter fineness, nom.		microns/60%	15	15	15
Filter difference pressure, alarm		kPa (bar)	150 (1,5)	150 (1,5)	150 (1,5)
Oil consumption (100% load), abt.	5)	g/kWh		0,6	
Cooling water system					
High temperature cooling water system					
Pressure before engine, nom.		kPa (bar)		200 (2,0) + static	
Pressure before engine, alarm		kPa (bar)		100 (1,0) + static	
Pressure before engine, max.		kPa (bar)		350 (3,5)	
Temperature before engine, abt.		°C		83	
Temperature after engine, nom.		°C		91	
Temperature after engine, alarm		°C		105	
Temperature after engine, stop		°C		110	
Pump capacity, nom.		m³/h	25	25	25
Pressure drop over engine		kPa (bar)		50 (0,5)	
Water volume in engine		m³	0,105	0,09	0,09
Pressure from expansion tank		kPa (bar)		70…150 (0,7…1,5)	
Pressure drop over central cooler, max.		kPa (bar)		60 (0,6)	
Delivery head of stand-by pump		kPa (bar)		200 (2)	
Low temperature cooling water system					
Pressure before charge air cooler, nom.		kPa (bar)		200 (2) + static	
Pressure before charge air cooler, alarm		kPa (bar)		100 (1) + static	
Pressure before charge air cooler, max.		kPa (bar)		350 (3,5)	
Temperature before charge air cooler, max.		°C		38	
Temperature before charge air cooler, min.		°C		25	
Pump capacity, nom.		m³/h	30	30	30
Pressure drop over charge air cooler		kPa (bar)		30 (0,3)	
Pressure drop over oil cooler		kPa (bar)		30 (0,3)	
Pressure drop over central cooler, max.		kPa (bar)		60 (0,6)	
Pressure from expansion tank		kPa (bar)		70…150 (0,7…1,5)	
Delivery head of stand-by pump		kPa (bar)		200 (2)	
Starting air system					
Air supply pressure before engine (max.)		Mpa (bar)		3 (30)	
Air supply pressure, alarm		Mpa (bar)		1,8 (18)	
Air consumption per start (20°C)	6)	Nm³		0,4	

1) At an ambient temperature of 25°C.
2) The figures are at 100% load and include the 5% tolerance on sfoc and engine driven pumps.
3) According to ISO 3046/1, lower calorific value 42 700 kJ/kg, with engine driven pumps. Tolerance 5%.
Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.
4) Capacities at 50 and 60 Hz respectively.
5) Tolerance + 0.3 g/kWh
6) At remote and automatic starting, the consumption is 1.2 Nm³

Subject to revision without notice.

Diesel engine Wärtsilä 6L20

			ME	AE	AE	AE	AE
Engine speed		RPM	1000	720	750	900	1000
Engine output		kW	1080	780	810	1020	1080
Engine output		HP	1470	1060	1100	1390	1470
Cylinder bore		mm			200		
Stroke		mm			280		
Swept volume		dm^3			52,8		
Compression ratio					15		
Compression pressure, max.		bar	167	150	150	167	167
Firing pressure, max.		bar	185	170	170	185	185
Charge air pressure at 100% load		bar			0,3		
Mean effective pressure		bar	24,6	24,6	24,6	25,8	24,6
Mean piston speed		m/s	9,3	6,7	7	8,4	9,3
Idling speed		RPM	350				
Combustion air system							
Flow of air at 100% load		kg/s	2,2	1,36	1,43	2,1	2,2
Ambient air temperature, max.		°C			45		
Air temperature after air cooler		°C			45…60		
Air temperature after air cooler, alarm		°C			75		
Exhaust gas system							
Exhaust gas flow (100% load)	3)	kg/s	2,26	1,4	1,48	2,16	2,26
Exhaust gas flow (85% load)	3)	kg/s	1,93	1,21	1,29	1,86	1,97
Exhaust gas flow (75% load)	3)	kg/s	1,69	1,09	1,16	1,68	1,78
Exhaust gas flow (25% load)	3)	kg/s	1,11	0,79	0,84	1,2	1,29
Exhaust gas temp. after turbocharger (100% load)	1) 3)	°C	330	370	370	330	330
Exhaust gas temp. after turbocharger (85% load)	1) 3)	°C	335	380	380	330	330
Exhaust gas temp. after turbocharger (75% load)	1) 3)	°C	345	380	380	330	330
Exhaust gas temp. after turbocharger (50% load)	1) 3)	°C	385	390	390	340	330
Exhaust gas back pressure drop, max.		kPa			3		
Diameter of turbocharger connection		mm			250		
Exhaust gas pipe diameter, min.		mm	350	300	300	350	350
Calculated dia for 35 m/s		mm	374	304	312	365	374
Heat balance	2) 3)						
Jacket water		kW	226	189	194	212	226
Charge air		kW	327	201	223	305	327
Lubricating oil		kW	143	106	108	133	143
Exhaust gases		kW	727	530	549	685	727
Radiation		kW	59	44	47	53	59
Fuel system							
Pressure before injection pumps		kPa (bar)			600(6)		
Pump capacity, MDF, engine driven		m^3/h	1,49	0,87	0,9	1,34	1,49
Fuel consumption (100% load)	3)	g/kWh	190	190	191	190	190
Fuel consumption (85% load)	3)	g/kWh	188	191	192	189	190
Fuel consumption (75% load)	3)	g/kWh	188	193	193	190	190
Fuel consumption (50% load)	3)	g/kWh	195	203	203	197	197
Leak fuel quantity, clean MDF fuel (100% load)		kg/h	0,9	0,6	0,7	0,7	0,9
Lubricating oil system							
Pressure before engine, nom.		kPa (bar)			450 (4,5)		
Pressure before engine, alarm		kPa (bar)			300 (3)		
Pressure before engine, stop		kPa (bar)			200 (2)		
Priming pressure, nom.		kPa (bar)			80 (0,8)		
Priming pressure, alarm		kPa (bar)			50 (0,5)		
Temperature before engine, nom.		°C			63		

Temperature before engine, alarm		°C			80		
Temperature after engine, abt.		°C			78		
Pump capacity (main), engine driven		m³/h	35	35	35	35	35
Pump capacity (main), separate		m³/h			21		
Pump capacity (priming)	4)	m³/h			6,9/8,4		
Oil volume, wet sump, nom.		m³			0,38		
Oil volume in separate system oil tank, nom.		m³	1,5	1,1	1,1	1,4	1,5
Filter fineness, nom.		microns/60%	25	25	25	25	25
Filter difference pressure, alarm		kPa (bar)			150 (1,5)		
Oil consumption (100% load), abt.	5)	g/kWh			0,6		

Cooling water system

High temperature cooling water system

Pressure before engine, nom.		kPa (bar)			200 (2,0) + static		
Pressure before engine, alarm		kPa (bar)			100 (1,0) + static		
Pressure before engine, max.		kPa (bar)			350 (3,5)		
Temperature before engine, abt.		°C			83		
Temperature after engine, nom.		°C			91		
Temperature after engine, alarm		°C			105		
Temperature after engine, stop		°C			110		
Pump capacity, nom.		m³/h	30	27	28	29	30
Pressure drop over engine		kPa (bar)			50 (0,5)		
Water volume in engine		m³			0,12		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		

Low temperature cooling water system

Pressure before charge air cooler, nom.		kPa (bar)			200 (2) + static		
Pressure before charge air cooler, alarm		kPa (bar)			100 (1) + static		
Pressure before charge air cooler, max.		kPa (bar)	28		350 (3,5)		
Temperature before charge air cooler, max.		°C			38		
Temperature before charge air cooler, min.		°C			25		
Pump capacity, nom.		m³/h	36	29	30	34	36
Pressure drop over charge air cooler		kPa (bar)			30 (0,3)		
Pressure drop over oil cooler		kPa (bar)			30 (0,3)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		

Starting air system

Air supply pressure before engine (max.)		Mpa (bar)			3 (30)		
Air supply pressure, alarm		Mpa (bar)			1,8 (18)		
Air consumption per start (20°C)	6)	Nm³			0,4		

1) At an ambient temperature of 25°C.
2) The figures are at 100% load and include the 5% tolerance on sfoc and engine driven pumps.
3) According to ISO 3046/1, lower calorific value 42 700 kJ/kg, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.
4) Capacities at 50 and 60 Hz respectively.
5) Tolerance + 0.3 g/kWh
6) At remote and automatic starting, the consumption is 1.2 Nm³

Subject to revision without notice.

Diesel engine Wärtsilä 8L20			ME	AE	AE	AE	AE
Engine speed		RPM	1000	720	750	900	1000
Engine output		kW	1440	1040	1080	1360	1440
Engine output		HP	1960	1410	1470	1850	1960
Cylinder bore		mm			200		
Stroke		mm			280		
Swept volume		dm^3			70,4		
Compression ratio					15		
Compression pressure, max.		bar	167	150	150	167	167
Firing pressure, max.		bar	185	170	170	185	185
Charge air pressure at 100% load		bar			0,3		
Mean effective pressure		bar	24,6	24,6	24,6	25,8	24,6
Mean piston speed		m/s	9,3	6,7	7	8,4	9,3
Idling speed		RPM	350				
Combustion air system							
Flow of air at 100% load		kg/s	2,86	1,96	2,04	2,79	2,98
Ambient air temperature, max.		°C			45		
Air temperature after air cooler		°C			45…60		
Air temperature after air cooler, alarm		°C			75		
Exhaust gas system							
Exhaust gas flow (100% load)	3)	kg/s	2,94	2,02	2,1	2,87	3,06
Exhaust gas flow (85% load)	3)	kg/s	2,5	1,74	1,81	2,48	2,67
Exhaust gas flow (75% load)	3)	kg/s	2,18	1,57	1,62	2,24	2,41
Exhaust gas flow (25% load)	3)	kg/s	1,44	1,11	1,15	1,61	1,76
Exhaust gas temp. after turbocharger (100% load)	1) 3)	°C	350	360	360	350	340
Exhaust gas temp. after turbocharger (85% load)	1) 3)	°C	355	360	360	340	340
Exhaust gas temp. after turbocharger (75% load)	1) 3)	°C	360	360	360	340	340
Exhaust gas temp. after turbocharger (50% load)	1) 3)	°C	390	370	370	350	350
Exhaust gas back pressure drop, max.		kPa			3		
Diameter of turbocharger connection		mm			300		
Exhaust gas pipe diameter, min.		mm	400	350	350	400	400
Calculated dia for 35 m/s		mm	433	362	369	428	438
Heat balance	2) 3)						
Jacket water		kW	330	244	254	307	330
Charge air		kW	442	306	322	407	442
Lubricating oil		kW	219	162	167	204	219
Exhaust gases		kW	1057	684	708	890	943
Radiation		kW	82	55	57	74	76
Fuel system							
Pressure before injection pumps		kPa (bar)			600(6)		
Pump capacity, MDF, engine driven		m^3/h	1,92	1,48	1,54	1,73	1,92
Fuel consumption (100% load)	3)	g/kWh	199	192	192	191	192
Fuel consumption (85% load)	3)	g/kWh	198	193	193	190	191
Fuel consumption (75% load)	3)	g/kWh	198	194	194	190	192
Fuel consumption (50% load)	3)	g/kWh	203	202	202	199	199
Leak fuel quantity, clean MDF fuel (100% load)		kg/h	1,2	0,8	0,9	1,1	1,2
Lubricating oil system							
Pressure before engine, nom.		kPa (bar)			450 (4,5)		
Pressure before engine, alarm		kPa (bar)			300 (3)		
Pressure before engine, stop		kPa (bar)			200 (2)		
Priming pressure, nom.		kPa (bar)			80 (0,8)		
Priming pressure, alarm		kPa (bar)			50 (0,5)		
Temperature before engine, nom.		°C			63		

Temperature before engine, alarm		°C			80		
Temperature after engine, abt.		°C			78		
Pump capacity (main), engine driven		m^3/h	50	50	50	50	50
Pump capacity (main), separate		m^3/h			27		
Pump capacity (priming)	4)	m^3/h			6,9/8,4		
Oil volume, wet sump, nom.		m^3			0,49		
Oil volume in separate system oil tank, nom.		m^3	1,9	1,4	1,5	1,8	1,9
Filter fineness, nom.		microns/60%	25	25	25	25	25
Filter difference pressure, alarm		kPa (bar)			150 (1,5)		
Oil consumption (100% load), abt.	5)	g/kWh			0,6		

Cooling water system

High temperature cooling water system

Pressure before engine, nom.		kPa (bar)			200 (2,0) + static		
Pressure before engine, alarm		kPa (bar)			100 (1,0) + static		
Pressure before engine, max.		kPa (bar)			350 (3,5)		
Temperature before engine, abt.		°C			83		
Temperature after engine, nom.		°C			91		
Temperature after engine, alarm		°C			105		
Temperature after engine, stop		°C			110		
Pump capacity, nom.		m^3/h	40	35	37	39	40
Pressure drop over engine		kPa (bar)			50 (0,5)		
Water volume in engine		m^3			0,15		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		

Low temperature cooling water system

Pressure before charge air cooler, nom.		kPa (bar)			200 (2) + static		
Pressure before charge air cooler, alarm		kPa (bar)			100 (1) + static		
Pressure before charge air cooler, max.		kPa (bar)			350 (3,5)		
Temperature before charge air cooler, max.		°C			38		
Temperature before charge air cooler, min.		°C			25		
Pump capacity, nom.		m^3/h	48	38	40	45	48
Pressure drop over charge air cooler		kPa (bar)			30 (0,3)		
Pressure drop over oil cooler		kPa (bar)			30 (0,3)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		

Starting air system

Air supply pressure before engine (max.)		Mpa (bar)			3 (30)		
Air supply pressure, alarm		Mpa (bar)			1,8 (18)		
Air consumption per start (20°C)	6)	Nm^3			0,4		

1) At an ambient temperature of 25°C.
2) The figures are at 100% load and include the 5% tolerance on sfoc and engine driven pumps.
3) According to ISO 3046/1, lower calorific value 42 700 kJ/kg, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.
4) Capacities at 50 and 60 Hz respectively.
5) Tolerance + 0.3 g/kWh
6) At remote and automatic starting, the consumption is 1.2 Nm^3

Subject to revision without notice.

Diesel engine Wärtsilä 9L20

			ME	AE	AE	AE	AE
Engine speed		RPM	1000	720	750	900	1000
Engine output		kW	1620	1170	1215	1530	1620
Engine output		HP	2200	1590	1650	2080	2200
Cylinder bore		mm			200		
Stroke		mm			280		
Swept volume		dm³			79,2		
Compression ratio					15		
Compression pressure, max.		bar	167	150	150	167	167
Firing pressure, max.		bar	185	170	170	185	185
Charge air pressure at 100% load		bar			0,3		
Mean effective pressure		bar	24,6	24,6	24,6	25,8	24,6
Mean piston speed		m/s	9,3	6,7	7	8,4	9,3
Idling speed		RPM	350				
Combustion air system							
Flow of air at 100% load		kg/s	3,43	1,98	2,11	3,09	3,43
Ambient air temperature, max.		°C			45		
Air temperature after air cooler		°C			45…60		
Air temperature after air cooler, alarm		°C			75		
Exhaust gas system							
Exhaust gas flow (100% load)	3)	kg/s	3,52	2,05	2,18	3,17	3,52
Exhaust gas flow (85% load)	3)	kg/s	3,05	1,79	1,91	2,76	3,08
Exhaust gas flow (75% load)	3)	kg/s	2,67	1,62	1,73	2,47	2,79
Exhaust gas flow (25% load)	3)	kg/s	1,74	1,18	1,27	1,76	2,05
Exhaust gas temp. after turbocharger (100% load)	1) 3)	°C	340	360	360	340	340
Exhaust gas temp. after turbocharger (85% load)	1) 3)	°C	350	360	360	340	340
Exhaust gas temp. after turbocharger (75% load)	1) 3)	°C	350	370	370	340	340
Exhaust gas temp. after turbocharger (50% load)	1) 3)	°C	370	380	380	350	330
Exhaust gas back pressure drop, max.		kPa			3		
Diameter of turbocharger connection		mm			300		
Exhaust gas pipe diameter, min.		mm	450	350	350	450	450
Calculated dia for 35 m/s		mm	470	365	376	446	470
Heat balance	2) 3)						
Jacket water		kW	380	280	291	353	380
Charge air		kW	495	342	355	458	495
Lubricating oil		kW	244	177	183	229	244
Exhaust gases		kW	1044	771	800	985	1044
Radiation		kW	79	63	66	75	79
Fuel system							
Pressure before injection pumps		kPa (bar)			600(6)		
Pump capacity, MDF, engine driven		m³/h	1,92	1,48	1,54	1,73	1,92
Fuel consumption (100% load)	3)	g/kWh	191	192	192	190	191
Fuel consumption (85% load)	3)	g/kWh	189	192	192	190	190
Fuel consumption (75% load)	3)	g/kWh	190	193	193	190	191
Fuel consumption (50% load)	3)	g/kWh	195	202	202	198	199
Leak fuel quantity, clean MDF fuel (100% load)		kg/h	1,3	0,9	1	1,2	1,3
Lubricating oil system							
Pressure before engine, nom.		kPa (bar)			450 (4,5)		
Pressure before engine, alarm		kPa (bar)			300 (3)		
Pressure before engine, stop		kPa (bar)			200 (2)		
Priming pressure, nom.		kPa (bar)			80 (0,8)		
Priming pressure, alarm		kPa (bar)			50 (0,5)		
Temperature before engine, nom.		°C			63		

Temperature before engine, alarm		°C			80		
Temperature after engine, abt.		°C			78		
Pump capacity (main), engine driven		m³/h	50	50	50	50	50
Pump capacity (main), separate		m³/h			30		
Pump capacity (priming)	4)	m³/h			6,9/8,4		
Oil volume, wet sump, nom.		m³			0,55		
Oil volume in separate system oil tank, nom.		m³	2,2	1,6	1,6	2,1	2,2
Filter fineness, nom.		microns/60%	25	25	25	25	25
Filter difference pressure, alarm		kPa (bar)			150 (1,5)		
Oil consumption (100% load), abt.	5)	g/kWh			0,6		
Cooling water system							
High temperature cooling water system							
Pressure before engine, nom.		kPa (bar)			200 (2,0) + static		
Pressure before engine, alarm		kPa (bar)			100 (1,0) + static		
Pressure before engine, max.		kPa (bar)			350 (3,5)		
Temperature before engine, abt.		°C			83		
Temperature after engine, nom.		°C			91		
Temperature after engine, alarm		°C			105		
Temperature after engine, stop		°C			110		
Pump capacity, nom.		m³/h	45	40	42	44	45
Pressure drop over engine		kPa (bar)			50 (0,5)		
Water volume in engine		m³			0,16		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		
Low temperature cooling water system							
Pressure before charge air cooler, nom.		kPa (bar)			200 (2) + static		
Pressure before charge air cooler, alarm		kPa (bar)			100 (1) + static		
Pressure before charge air cooler, max.		kPa (bar)			350 (3,5)		
Temperature before charge air cooler, max.		°C			38		
Temperature before charge air cooler, min.		°C			25		
Pump capacity, nom.		m³/h	54	43	45	50	54
Pressure drop over charge air cooler		kPa (bar)			30 (0,3)		
Pressure drop over oil cooler		kPa (bar)			30 (0,3)		
Pressure drop over central cooler, max.		kPa (bar)			60 (0,6)		
Pressure from expansion tank		kPa (bar)			70…150 (0,7…1,5)		
Delivery head of stand-by pump		kPa (bar)			200 (2)		
Starting air system							
Air supply pressure before engine (max.)		Mpa (bar)			3 (30)		
Air supply pressure, alarm		Mpa (bar)			1,8 (18)		
Air consumption per start (20°C)	6)	Nm³			0,4		

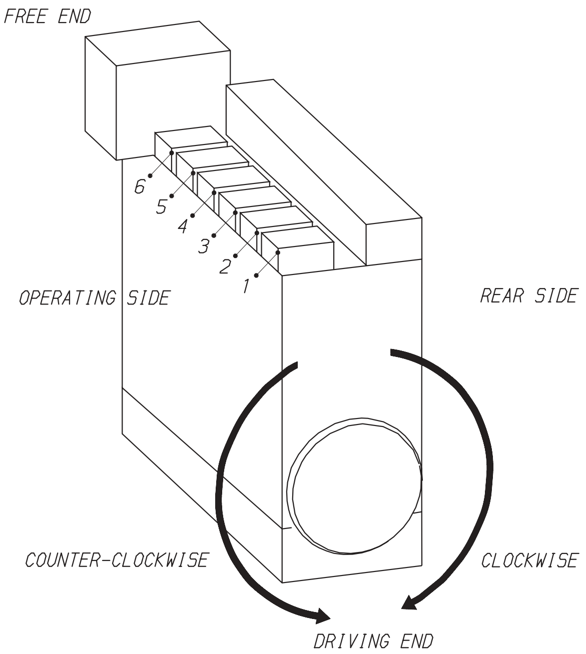
1) At an ambient temperature of 25°C.
2) The figures are at 100% load and include the 5% tolerance on sfoc and engine driven pumps.
3) According to ISO 3046/1, lower calorific value 42 700 kJ/kg, with engine driven pumps. Tolerance 5%. Constant speed applications are Auxiliary and DE. Mechanical propulsion variable speed applications according to propeller law.
4) Capacities at 50 and 60 Hz respectively.
5) Tolerance + 0.3 g/kWh
6) At remote and automatic starting, the consumption is 1.2 Nm³

Subject to revision without notice.

4. Description of the engine

4.1. Definitions

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 main bearing caps, made of nodular cast iron, 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. The crankshaft satisfies the requirements of all classification societies.

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

The camshaft is built of one piece for each cylinder cam piece with separate bearing pieces in between. The cam and bearing pieces are held together with two hydraulically tightened centre screws. 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 bearings are installed and removed by means of a hydraulic tool. The original installation in the factory in done with cooling of the bearing. 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 make the valve mechanism 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 turbocharger offers the ideal combination of high-pressure ratios and good efficiency at full and part load. The charge air cooler is single stage type and cooled by LT-water.

4.2.12. Injection equipment

The injection pumps are one-cylinder pumps located in the “multi-housing”, 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-flown to enable heavy fuel operation. They are also equipped with a stop cylinder, which is connected to the electro-pneumatic overspeed protection system.

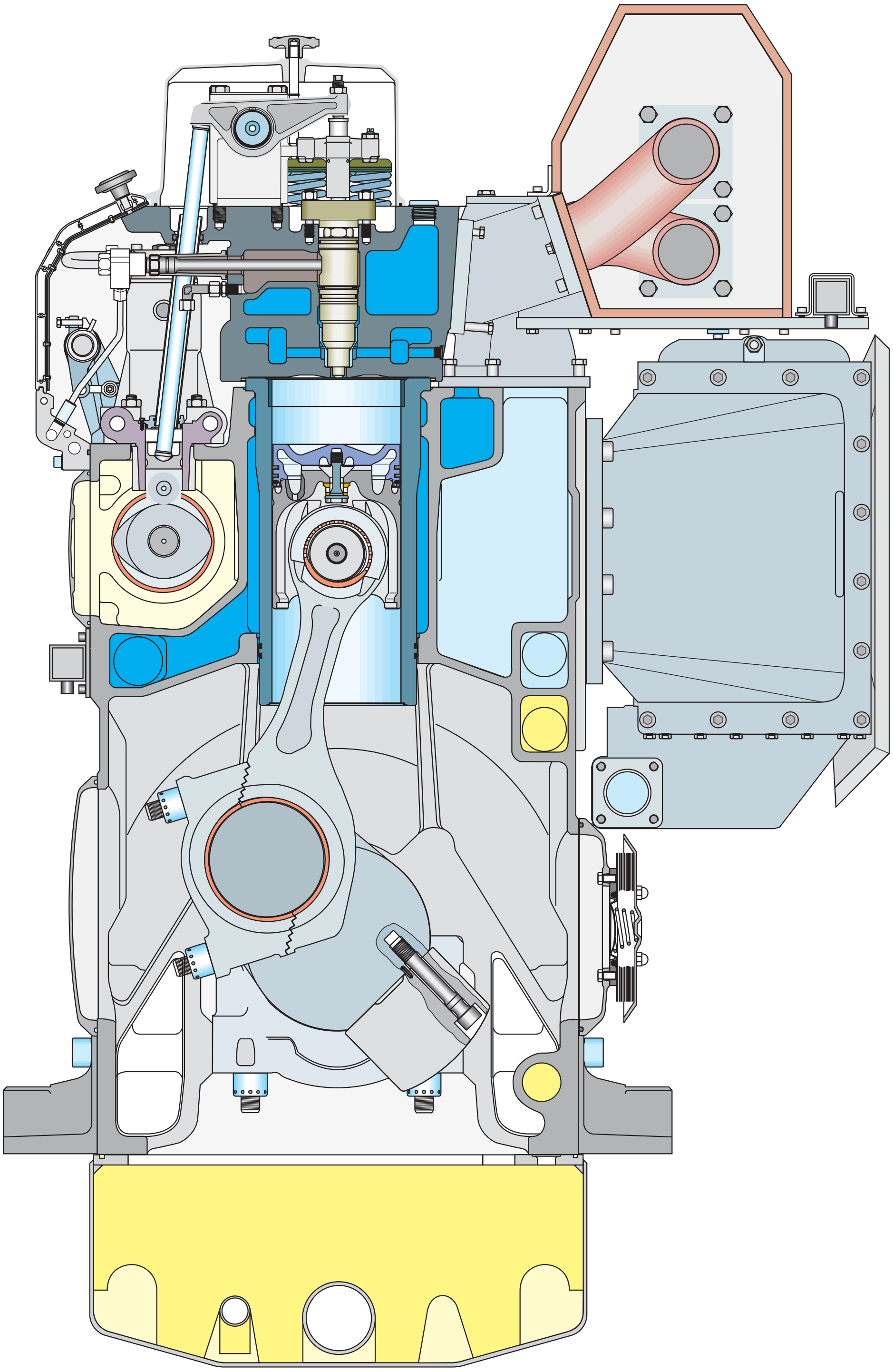
The injection valve is centrally located in the cylinder head and the fuel is admitted sideways through a high pressure connection screwed in the nozzle holder. The injection pipe between the injection pump and the high pressure connection is well protected inside the hot box. The high pressure side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

4.2.13. Exhaust pipes

The exhaust manifold pipes are made of special heat resistant nodular cast iron alloy.

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

4.3. 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	MDO	HFO	MDO
	Time between overhauls (h)	Time between overhauls (h)	Expected comp. lifetimes (h)	Expected comp. 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	24000	32000
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). Sea-water piping should be in Cunifer or hot dip galvanized steel. Plastic pipes for part of the seawater piping require a special approval process.

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 double hoses of approved type. If flexible hoses are used in the compressed air system an outlet valve shall be fitted in front of the hose(s).

As it is already in the design phase necessary to know in addition to how the system is supposed to work also how the system most feasibly can be built a fitting order plan shall be done prior to construction. The following aspects but not necessarily limited to these 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
- the plan shall include the time schedule and manpower needed
- 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 apparatuses can be made with reasonable effort. The estimated need of service during the ship's lifetime shall be taken into consideration when deciding the "open-inspect" priority order. This determines the accepted amount of dismantling and refitting work.

5.2. Pipe dimensions

Table 5.1. Recommended maximum fluid velocities and flow rates for pipework*

Nominal pipe diameter	Flow rate [m/sec] Flow amount [m³/h]									
(Media –>	Sea-water		Fresh water		Lubricating oil		Marine diesel oil		Heavy fuel oil	
Pipe material –>	Steel galvanized		Mild steel		Mild steel		Mild steel		Mild steel	
Pump side –>)	suction	delivery	suction	delivery	suction	delivery	suction	delivery	suction	delivery
32	1 2.9	1.4 4.1	1.5 4.3	1.5 4.3	0.6 1.7	1 2.9	0.9 2.6	1.1 3.2	0.5 1.4	0.6 1.7
40	1.2 5.4	1.6 7.2	1.7 7.7	1.7 7.7	0.7 3.2	1.2 5.4	1 4.5	1.2 5.4	0.5 2.3	0.7 3.2
50	1.3 9.2	1.8 12.7	1.9 13.4	1.9 13.4	0.8 5.7	1.4 9.9	1.1 7.8	1.3 9.2	0.5 3.5	0.8 5.7
65	1.5 17.9	2 23.9	2.1 25.1	2.1 25.1	0.8 9.6	1.5 17.9	1.2 14.3	1.4 16.7	0.6 7.2	0.9 10.8
80	1.6 29	2.1 38	2.2 39.8	2.2 39.8	0.9 16.3	1.6 29	1.3 23.5	1.5 27.1	0.6 10.9	1 18.1
100	1.8 50.9	2.2 62.2	2.3 65	2.3 65	0.9 25.5	1.6 45.2	1.4 39.6	1.6 45.2	0.7 19.8	1.2 33.9
125	2 88.4	2.3 101.6	2.4 106	2.4 110.4	1.1 48.6	1.7 75.1	1.5 66.3	1.7 75.1	0.8 35.3	1.4 61.9
150	2.2 140	2.4 152.7	2.5 159	2.6 165.4	1.3 82.7	1.8 114.5	1.5 95.4	1.8 114.5	0.9 57.3	1.6 108.2
200	2.3 260.2	2.5 282.8	2.6 294.1	2.7 305.4	1.3 147	1.8 203.6	— —	— —	— —	— —
Aluminium brass	2.6 294									
250	2.5 441.8	2.6 459.5	2.7 477.2	2.7 477.2	1.3 229.8	1.9 335.8	— —	— —	— —	— —
Aluminium brass	2.7 447.2									
300	2.6 661.7	2.6 661.7	2.7 687.2	2.7 687.2	1.3 330.9	1.9 483.6	— —	— —	— —	— —
Aluminium brass	2.8 712.5									
350	2.6 900.5	2.6 900.5	2.7 935.2	2.7 935.2	1.4 484.9	2 692.7	— —	— —	— —	— —
Aluminium brass	2.8 969.8									
400	2.6 1176.2	2.7 1221.5	2.7 1221.5	2.7 1221.5	1.4 633.3	2 904.8	— —	— —	— —	— —
Aluminium brass	2.8 1266.7									
450	2.6 1488.6	2.7 1545.9	2.7 1545.9	2.7 1545.9	1.4 801.6	2 1145.1	— —	— —	— —	— —
Aluminium brass	2.9 1660.4									
500	2.6 1837.8	2.7 1908.5	2.7 1908.5	2.7 1908.5	1.5 1060.4	2.1 1484.6	— —	— —	— —	— —
Aluminium brass	2.9 2049.9									

* The velocities given in the above table are guidance figures only. National standards can also be applied.

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. pre-heating 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 categorisation 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 on 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
Fuel oil	> 16	or > 150	< 16	and < 150	< 7	and < 60
Other media	> 40	or > 300	< 40	and < 300	< 16	and < 200

5.6. Insulation

In addition to the operational aspects of the different piping systems requiring insulation the fire risk aspect shall be given attention (e.g. Insulating and/or shielding of hot surfaces). 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.

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
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

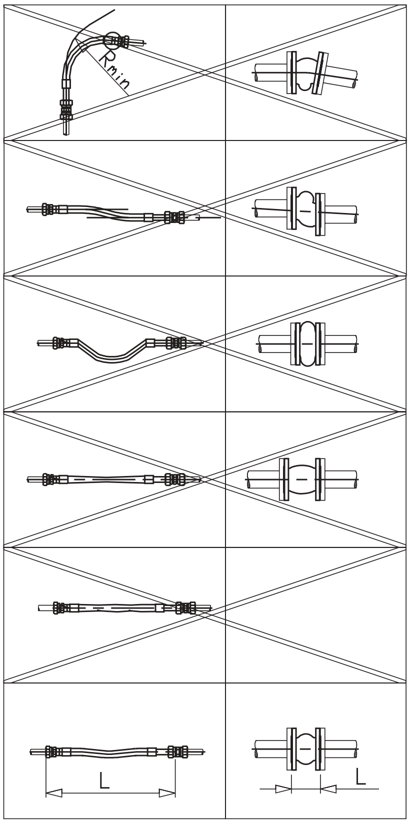
More detailed recommendations on flushing procedures are when necessary described under the relevant chapters concerning the fuel oil system and the lubricating oil system. Provisions are to be made to ensure that necessary temporary bypasses can be arranged and that flushing hoses, filters and pumps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

5.9. Flexible pipe connections

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.

Flexible hoses (4V60B0100)



Bending radius

Radially not aligned

Too short inst. length

Stretched

Twisted

Correctly installed

6. Fuel oil system

6.1. General

Fuel characteristics of the fuels are presented under heading Fuel characteristics in the Chapter for General data and outputs.

This refers to HFO usage unless otherwise indicated.

6.1.1. Operating principals

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 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

In installations where the stand-by engines can be fed from the diesel fuel day tank, sufficient fuel pressure for a safe start must also be ensured in the 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

If the engines are equipped with engine driven fuel feed pumps, see heading for MDF installations.

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. Local gauges

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

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

6.2. MDF installations

6.2.1. General

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

6.2.2. Internal fuel system

The standard system comprises the following built-on equipment:

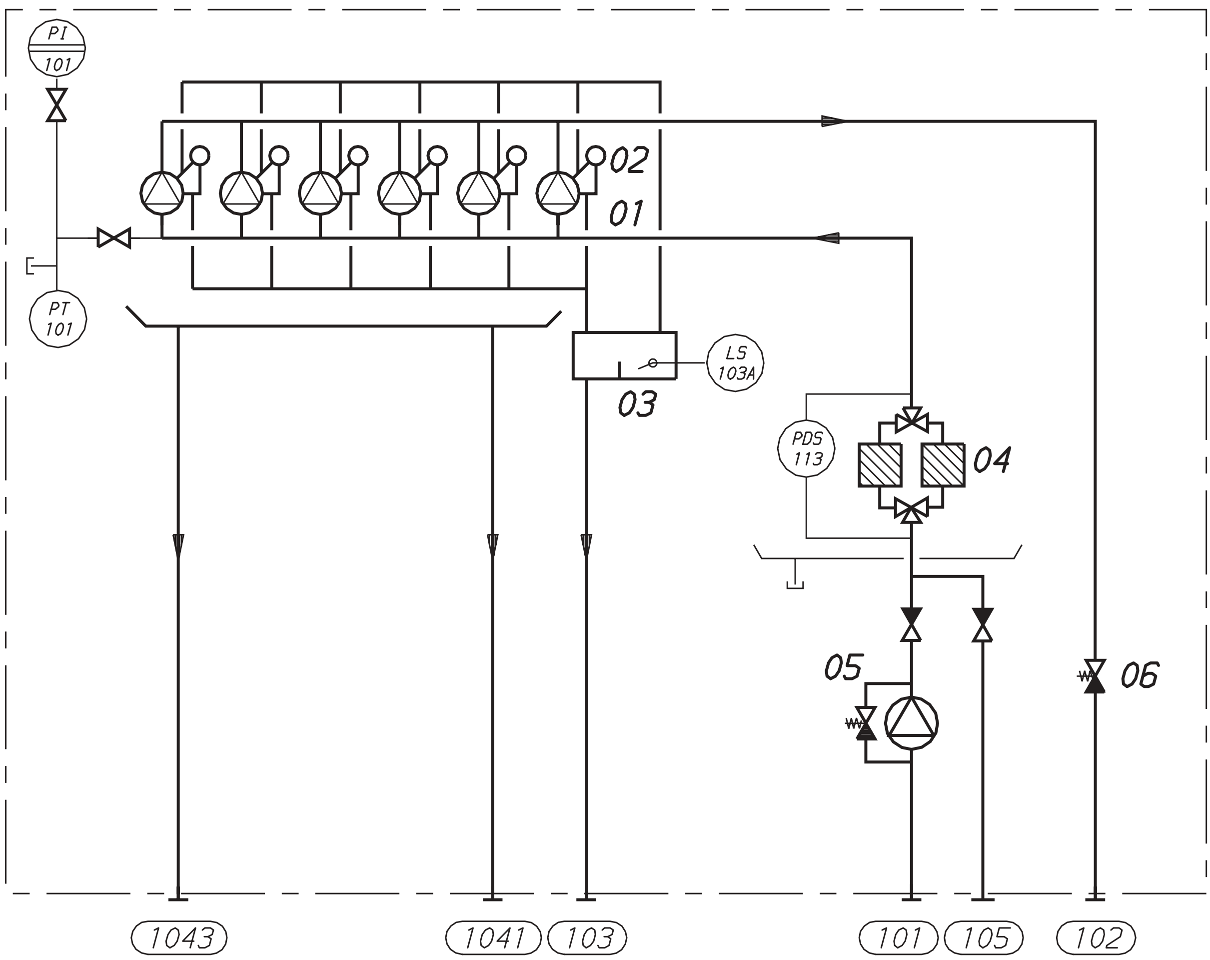
- fuel injection pumps
- injection valves
- pressure control valve in the outlet pipe

Controlled 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 for 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).

Dimensions of fuel pipe connections on the engine

Code	Description	Size	Pressure class	Standard
101	Fuel inlet, HFO	OD18	PN250	DIN 2353
101	Fuel inlet, MDF	OD28	PN250	DIN 2353
102	Fuel outlet, HFO	OD18	PN250	DIN 2353
102	Fuel outlet, MDF	OD28	PN250	DIN 2353
103	Leak fuel drain, clean fuel	OD18	-	DIN 2391
1041	Leak fuel drain, dirty fuel	OD22	-	DIN 2391
1043	Leak fuel drain, dirty fuel	OD18	-	DIN 2391
105	Fuel stand-by connection	OD22	PN250	DIN 2353

Internal fuel system, MDF (4V76F5881)



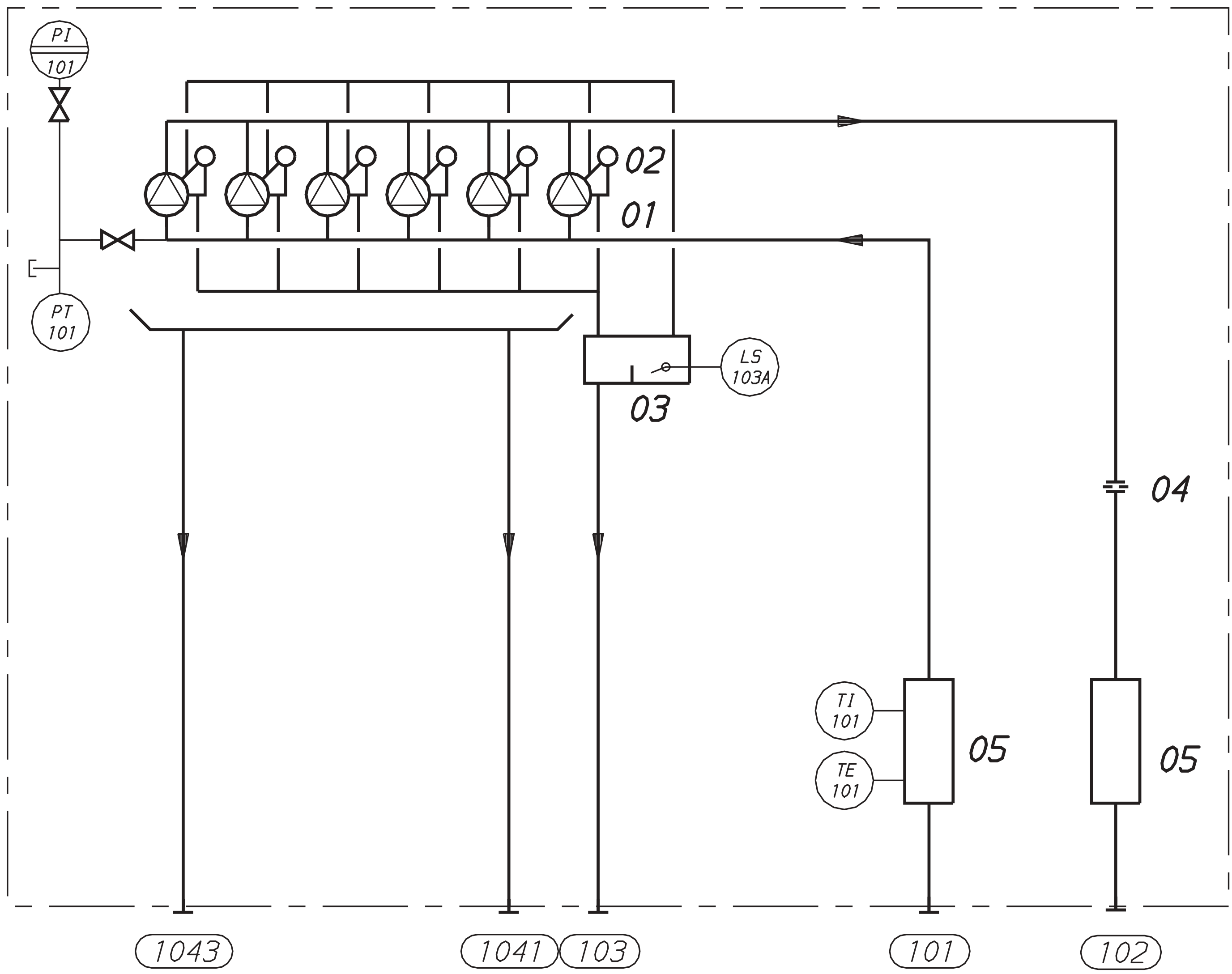
System components

01	Injection pump
02	Injection valve
03	Leak fuel oil system with level alarm
04	Duplex fine filter
05	Engine driven fuel feed pump
06	Pressure regulating valve

Pipe connections, engine

101	Fuel inlet
102	Fuel outlet
103	Leak fuel drain, clean fuel
104	Leak fuel drain, dirty fuel
105	Fuel stand-by connection

Internal fuel system, HFO (4V76F5880)



System components		Pipe connections	
01	Injection pump	101	Fuel inlet
02	Injection valve	102	Fuel outlet
03	Leak fuel oil system with level alarm	103	Leak fuel drain, clean fuel
04	Adjustable orifice	104	Leak fuel drain, dirty fuel
05	Pulse damper		

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 above the top of the engine. 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.

6.2.3. 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.

Main and auxiliary engines are recommended to be connected to separate circuits.

Filling, transfer and storage

The filling methods of the bunker tanks depend on the off board facilities available.

The ship must have means to transfer the fuel from bunker tanks to setting 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 between the bunker tanks and the settling tank or day tanks, 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. 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 filter for separator feed pump (1F02)

A suction filter shall be fitted to protect the feed pump. The filter can be either a duplex filter with change over valves or two separate simplex filters. The design of the filter should be such that air suction cannot occur.

- 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

Preheater, separator (1E01)

Fuels having a viscosity higher than 5 mm^2/s (cSt) at 50°C need preheating before the separator. For MDF the preheating temperature should be according to the separator supplier.

MDF separator (1N05)

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

For max viscosity 11 mm^2/s (cSt) at 50°C fuels a flow rate of 80% and a preheating temperature of 45°C are recommended.

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

For marine diesel fuel (MDF) and fuels having a viscosity of less than 115 mm^2/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.

General

It has to be possible to shut-off the heating of the pipes when running with MDF (the tracing pipes to be grouped 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 mm^2/s(cSt) and not drop below 2.0 mm^2/s(cSt) even during short transient conditions. In certain applications a cooler may be necessary.

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 anyhow that SOLAS Chapter II-1 Part C Regulation 26 states that "Two fuel oil service tanks for each type of fuel used on board necessary for propulsion and vital systems or equivalent arrangements shall be provided on each new ship, with the capacity of at least 8 h at maximum continuos rating of the propulsion plant and normal operating load at sea of the generator plant. This paragraph applies only to ships constructed on or after 1 July 1998."

Suction strainer, MDF (1F03)

A suction strainer with a fineness of 0.5 mm should be installed for protecting the feed pumps. The strainer may be either of duplex type with change over valves or simplex strainers in parallel. The design should be such that air suction is prevented.

Circulation pump, MDF (1P03)

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

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)

Fuel consumption meter

If a fuel consumption meter is required, it should be fitted in the day tank feed line. In case of individual engine fuel consumption metering is required, two meters per engine need to be installed.

They should then be located in the fuel feed line before the automatic filter and in the return line after the engine. An automatically opening by-pass line around the consumption meter is recommended in case of possible clogging.

Cooler/Heater

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. Cooler is needed where long periods of low load operation is 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 re-used 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.

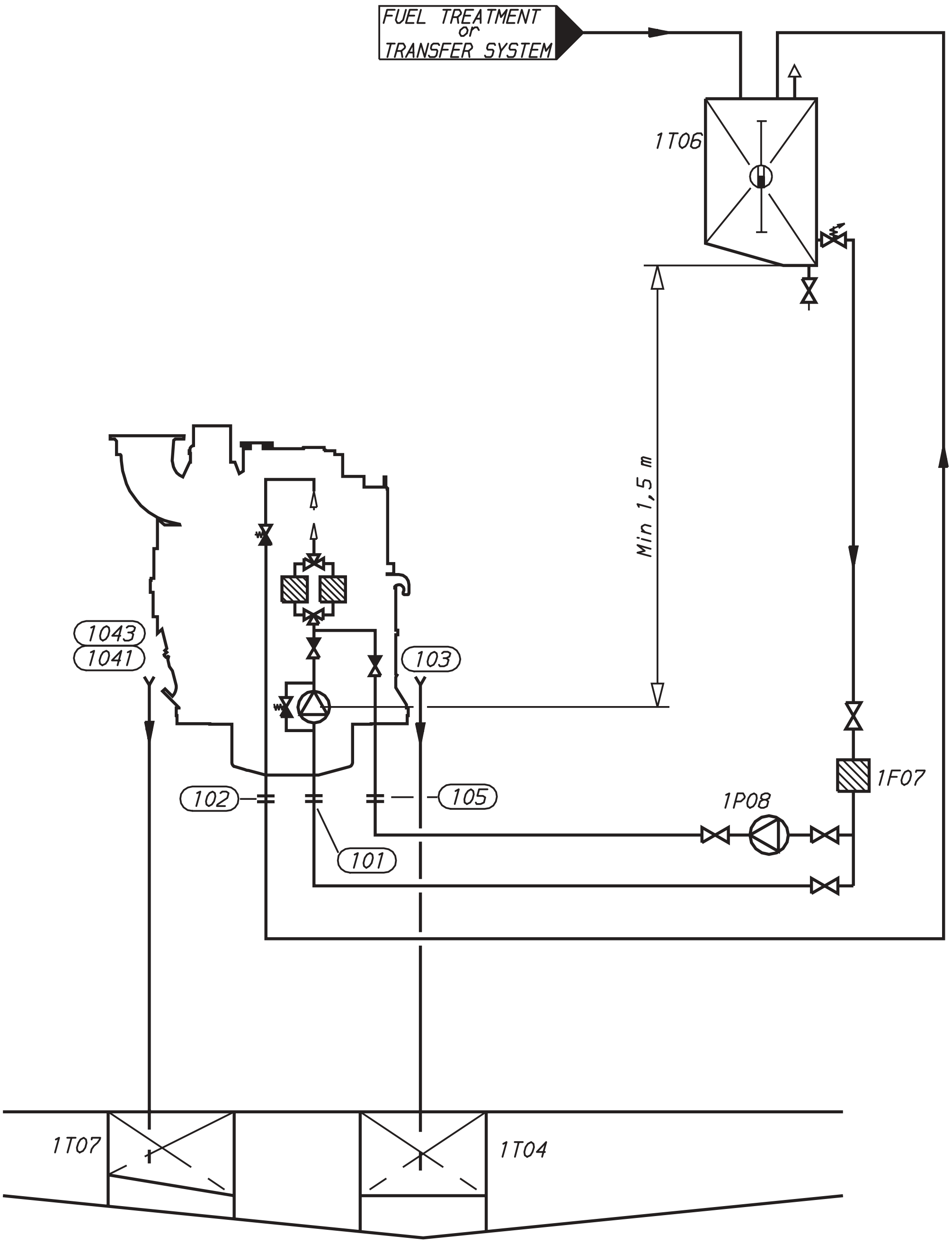
Leak fuel tank, dirty fuel (1T07)

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.

Fuel feed unit

Fuel feed equipment can also be combined to form a unit.

Fuel feed system, main engine (3V76F5884)



System components	
1F07	Suction strainer, MDF
1P08	MDF stand by pump
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 flywheel end
105	Fuel stand-by connection

6.3. HFO installations

6.3.1. General

In ships intended for operation on heavy fuel, steam or thermal oil heating coils must be installed in the bunker, settling and day tanks, so that it is possible to maintain a temperature of 40 - 50°C (or even higher temperature, depending on the pour point and viscosity of the heavy fuel used).

Normally the heating coils are dimensioned on basis of the heat transfer required for raising the temperature of the tank to the above temperature in a certain time, e.g. 1°C/h, as well as on the heat losses when maintaining the tank at that temperature.

All tanks, from, which heavy fuel is pumped, are to be kept 5 - 10°C above the pour point. Max. allowed pour point for allowed fuels is +30°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 big problems 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. When operating on heavy fuel the dimensioning of the separators is of greatest importance and therefore the recommendations of the separator designer should be closely followed.

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

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 all 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.

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 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 steel pipe.

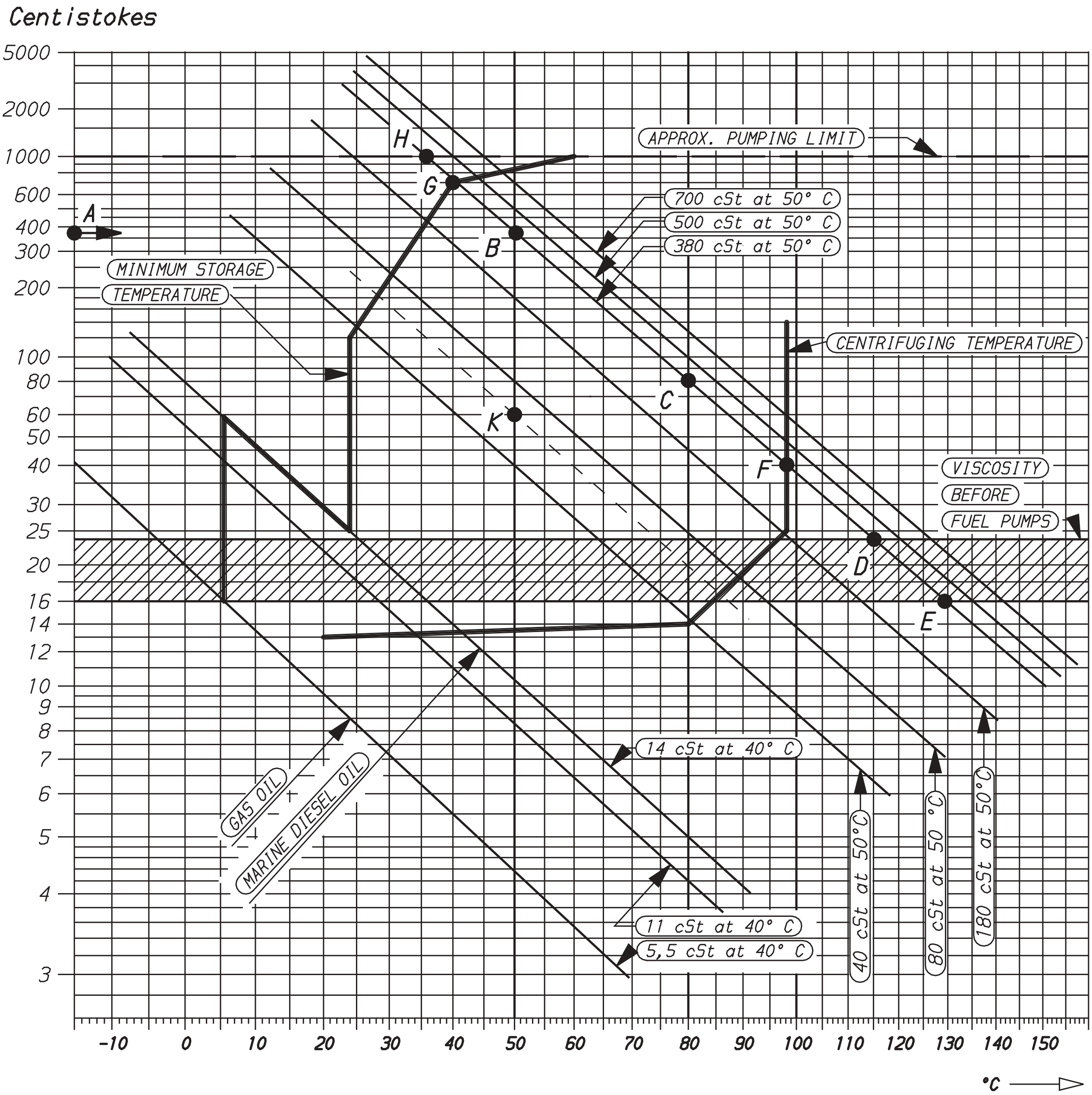
Fuel separators require typically 7 kW/installed engine MW and booster units 30 kW/installed engine MW. See also formulas presented later in this chapter.

Example: 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 centrifuge 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: 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, centrifuging temperature 86°C, minimum storage tank temperature 28°C.

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



6.3.2. Internal fuel system

The standard system comprises the following built-on equipment:

- heavy fuel injection pumps
- injection valves
- pressure control orifice in the outlet pipe

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 for 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).

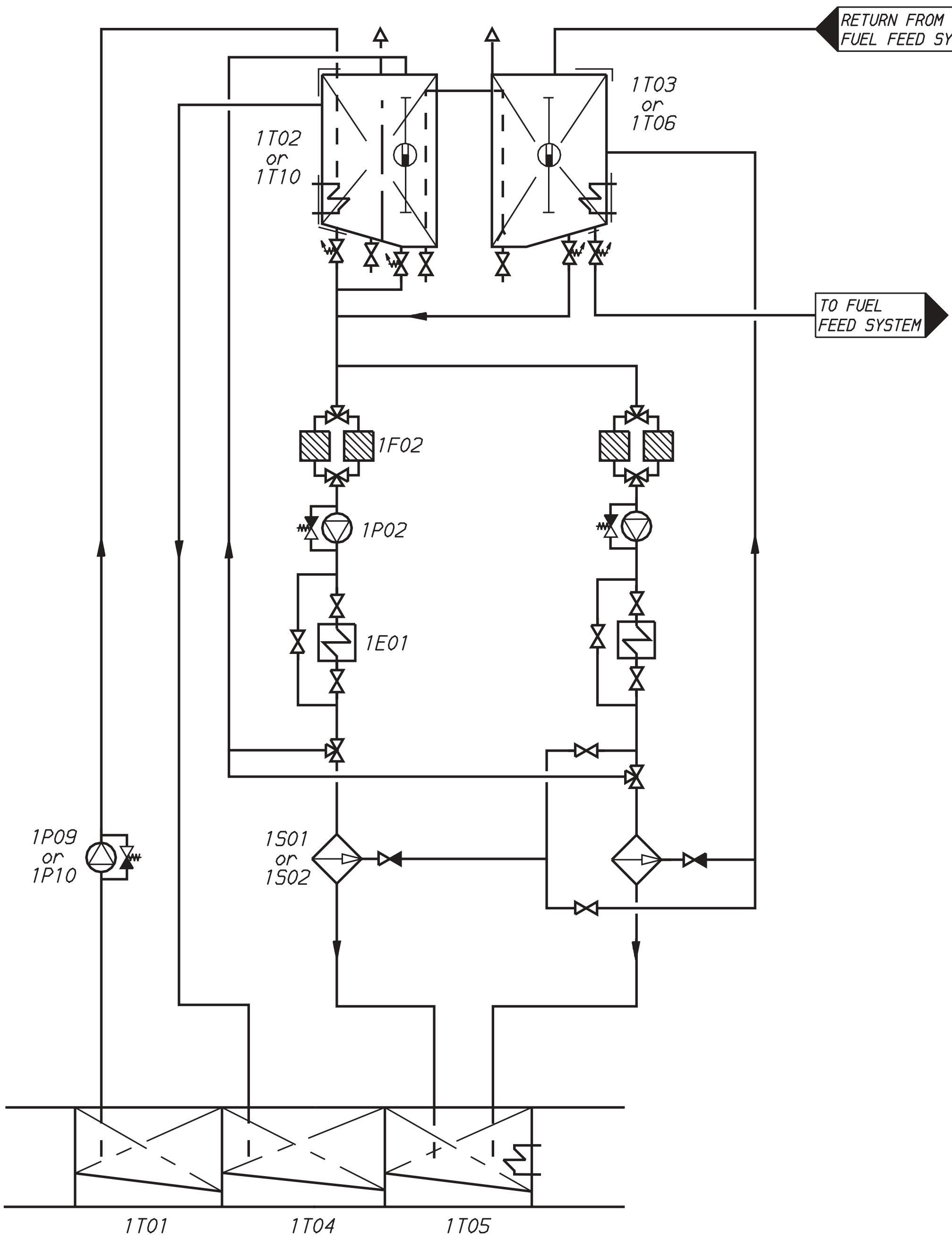
6.3.3. External fuel system

General

The engine is designed for continuous heavy fuel operation. It is, however, possible to operate the engine on 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 MDF for start and stop is not recommended.

Fuel transfer and separating system (3V76F5882)



System components

1E01	Heater	1T01	Bunker tank
1F02	Suction filter	1T02	Settling tank, HFO
1P02	Feed pump	1T03	Day tank, HFO
1P09	Transfer pump, HFO	1T04	Overflow tank
1P10	Transfer pump, MDF	1T05	Sludge tank
1S01	Separator, HFO	1T06	Day tank, MDF
1S02	Separator, MDF	1T10	Settling tank, MDF

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 filling methods of the bunker tanks depend on the off board facilities available.

The ship must have means to transfer the fuel from bunker tanks to setting 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

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 arrangement can be used for fuels with a density up to max. abt. 991 kg/m³ at 15°C.

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 min. 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 (1N05)

Suction filter for separator feed pump (1F02)

A suction filter shall be fitted to protect the feed pump. The filter should be equipped with a heating jacket in case the installation place is cold. The filter can be either a duplex filter with change over valves or two separate simplex filters. The design of the filter should be such that air suction cannot occur.

- fineness 0.5 mm

Feed pump, separator (1P02)

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.

Design data:

- delivery 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:

$P(kW) = m(l/h)\ \Delta t(°C) / 1700$

P(kW) = heater capacity

m = capacity of the separator feed pump

Δt = temperature rise in heater

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

Fuels having a viscosity higher than 5 mm²/s (cSt) at 50°C need preheating before the separator.

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:

$$Q\ [l/h] = \frac{P[kW] \cdot b \cdot 24[h]}{\rho \cdot t[h]}$$

where:

P = max. continuous rating of the diesel engine

b = specific fuel consumption + 15% safety margin

ρ = density of the fuel

t = daily separating time for selfcleaning separator (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.

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 grouped 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 tanks, HFO (1T03)

The heavy fuel day tank is usually dimensioned to ensure fuel supply for about 24 operating hours when filled to maximum*. 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

*Note anyhow that SOLAS Chapter II-1 Part C Regulation 26 states that "Two fuel oil service tanks for each type of fuel used on board necessary for propulsion and vital systems or equivalent arrangements shall be provided on each new ship, with the capacity of at least 8 h at maximum continuos rating of the propulsion plant and normal operating load at sea of the generator plant. This paragraph applies only to ships constructed on or after 1 July 1998."

Fuel feed unit (1N01)

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

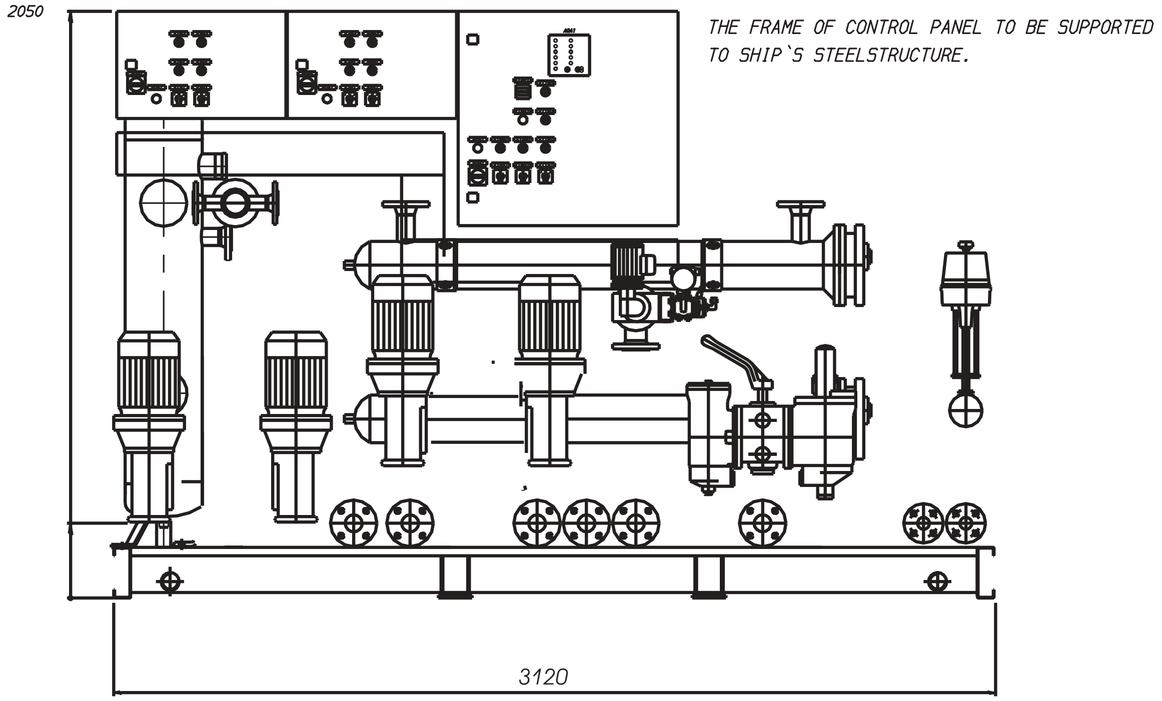
This unit normally comprises the following equipment:

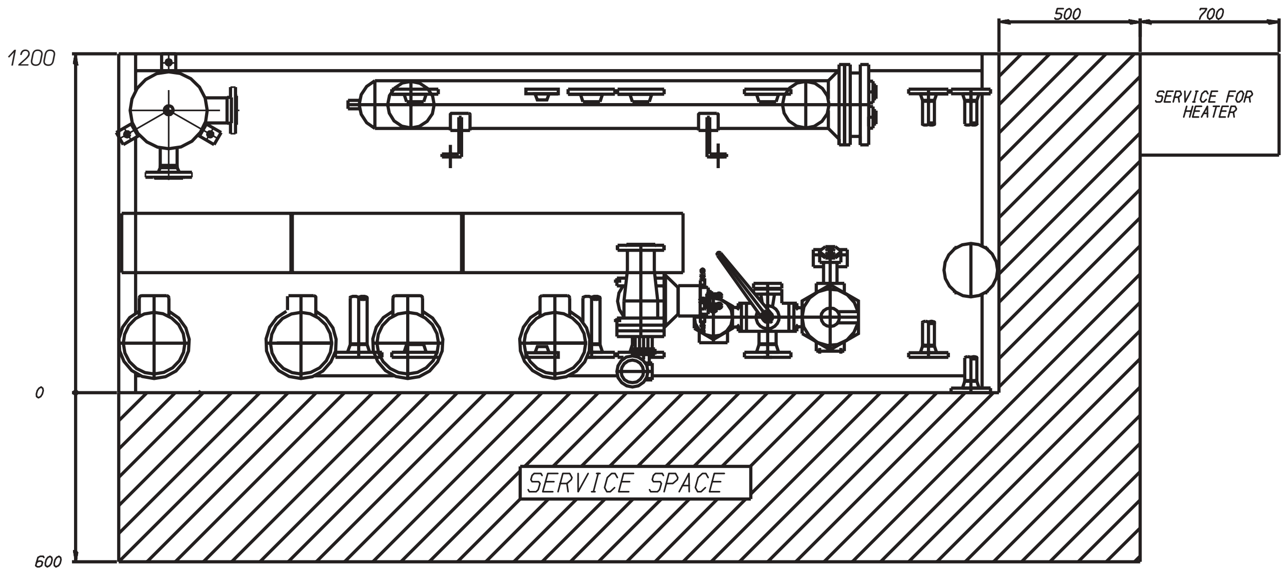
- two suction strainers
- two booster pumps of screw type, equipped with built-on safety valves and electric motors
- one pressure control/overflow valve
- one pressurized de-aeration tank, equipped with a level switch operated vent valve
- two circulation pumps, same type as above
- two heaters, steam, 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.

Fuel feed unit, example (4V76F5613)





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. The strainer may be either of duplex type with change over valves or simplex strainers in parallel. The design should be such that air suction is prevented.

Feed pump, HFO (1P04)

The feed pump maintains the pressure in the fuel feed system. It is recommended to use a high temperature resistant screw pump as feed 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 (overflow) valve HFO

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

- 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	according to spec.
• operating temperature	0...150°C
• preheating	from 25 mm²/s (cSt)/100°C
• flow	circulation pump capacity
• operating pressure	0.1 MPa (10 bar)
• design pressure	0.16 Mpa (16 bar)
• test pressure	fuel side 0.2MPa(20 bar) heating jacket 0.1 Mpa (10 bar)
• fineness:	
- back-flushing filter	25 μm (absolute mesh size)
- insert filter	25 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

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.

De-aeration tank

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.

Circulation 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
  - operating pressure 1 MPa (10 bar)
  - operating temperature 150°C
  - viscosity (for dimensioning the el. motor) 500 mm²/s (cSt)

Heater

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:

$P(kW) = m(l/h)\ \Delta t(°C) / 1700$

P(kW) = heater capacity

m = evaluated by multiplying the specific fuel consumption of the engines by the total max. output of the engines

Δt = temperature rise, higher with increased fuel viscosity

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

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:

- viscosity range (at injection pumps) 12...24 mm²/s (cSt)
- operating temperature 180°C
- operating pressure 4 MPa (40 bar)

Overflow valve (1V05)

This valve limits the maximum pressure in fuel line to the engine by relieving the pressure to the return line.

Pressure control valve on the return line (1V04)

This valve controls the pressure in the return line from the engine.

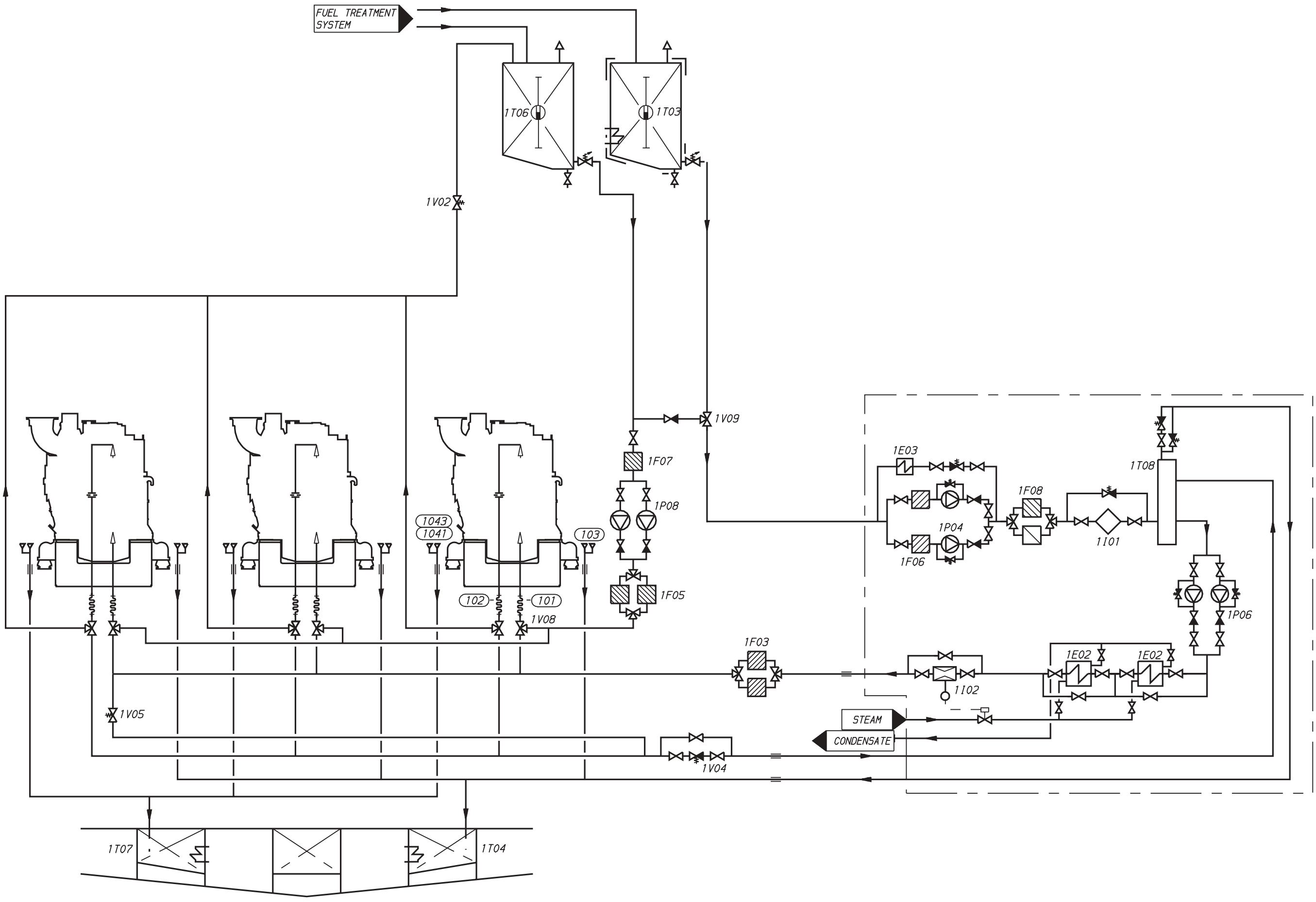
Leak fuel tank, clean fuel (1T04)

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.

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.

Fuel feed system, auxiliary engines (3V76F5883)



System components

1E02	Heater
1E03	Radiator
1F03	Fine filter, HFO
1F05	Fine filter, MDF
1F06	Suction filter, HFO
1F07	Suction strainer, MDF
1F08	Automatic filter
1I01	Flow meter
1I02	Viscosimeter
1P04	Fuel feed pump, HFO
1P06	Circulation pump
1P08	MDF pump
1T03	Day tank, HFO
1T04	Leak fuel tank, clean fuel
1T06	Day tank, MDF
1T07	Leak fuel tank, dirty fuel
1T08	De-aeration tank

1V02	MDF pressure control valve
1V04	Pressure regulating valve
1V05	Overflow valve
1V08	3-way change over valve
1V09	Change over valve

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 flywheel end

7. Lubricating oil system

7.1. General

Each engine should have a separate lubricating oil system of its own. Engines operating on heavy fuel should have continuous centrifuging of the lubricating oil.

The following equipment is built on the engine as standard:

- Engine driven lubricating oil pump
- Prelubricating 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

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.

7.2. Lubricating oil quality

Engine lubricating oil

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

The alkalinity, BN, of the system oil should be 30 - 55 mg/KOH/g in heavy fuel use; higher at higher sulphur content of the fuel. It is recommended to use BN 40 lubricants with category C fuels. The use of high BN (50 - 55) lubricants in heavy fuel installations is recommended, if the use of BN 40 lubricants causes short oil change intervals.

Today's modern trunk piston diesel engines are stressing the lubricating oils heavily due to a.o. low specific lubricating oil consumption. Also ingress of residual fuel combustion products into the lubricating oil can cause deposit formation on the surface of certain engine components resulting in severe operating problems. Due to this many lubricating oil suppliers have developed new lubricating oil formulations with better fuel and lubricating oil compatibility.

If MDF is used as fuel, a lubricating oil with a BN of 10 - 22 is recommended. However, an approved lubricating oil with a BN of 24 - 30 can also be used, if the desired lower BN lubricating oil brand is not included in table below.

Table 7.1. Approved system oils - recommended in the first place, in gas oil (A) or marine diesel oil (B) installations

Supplier	Brand name	Viscosity	BN	Fuel category
BP	Energol HPDX40	SAE 40	12	A
Castrol	TLX 154	SAE 40	15	A, B
	TLX 204	SAE 40	20	A, B
Mobil	Mobilgard ADL 40	SAE 40	15	A, B
	Mobilgard 412	SAE 40	15	A, B
Shell	Gadinia Oil 40 (SL0391)	SAE 40	12	A
	Sirius FB Oil 40	SAE 40	13	A

The lubricating oils mentioned in table below are representing a new detergent/dispersant additive chemistry and have shown good performance in Wärtsilä engines. These lubricating oils are recommended in the first place in order to reach full service intervals.

Table 7.2. Approved system oils: lubricating oils with improved detergent/dispersant additive chemistry - heavy fuel (C), recommended in the first place

Supplier	Brand name	Viscosity	BN	Fuel category
BP	Energol IC-HFX 304	SAE 40	30	A, B, C
	Energol IC-HFX 404	SAE 40	40	A, B, C
	Energol IC-HFX 504	SAE 40	50	A, B, C
Caltex	Delo 3000 Marine SAE 40	SAE 40	30	A, B, C
	Delo 3400 Marine SAE 40	SAE 40	40	A, B, C
	Delo 3550 Marine SAE 40	SAE 40	55	A, B, C
Castrol	TLX 304	SAE 40	30	A, B, C
	TLX 404	SAE 40	40	A, B, C
	TLX 504	SAE 40	50	A, B, C
	TLX 554	SAE 40	55	A, B, C
Chevron	Delo 3000 Marine 40	SAE 40	30	A, B, C
	Delo 3400 Marine 40	SAE 40	40	A, B, C
	Delo 3550 Marine 40	SAE 40	55	A, B, C
Elf Lub Marine	Aurelia 4030	SAE 40	30	A, B, C
	Aurelia XT 4040	SAE 40	40	A, B, C
	Aurelia XT 4055	SAE 40	55	A, B, C
Esso	Exxmar 30 TP 40 PLUS	SAE 40	30	A, B, C
	Exxmar 40 TP 40 PLUS	SAE 40	40	A, B, C
	Exxmar 50 TP 40 PLUS	SAE 40	50	A, B, C
Fina	Stellano S 430	SAE 40	30	A, B, C
	Stellano S 440	SAE 40	40	A, B, C
	Stellano S 455	SAE 40	55	A, B, C
Mobil	Mobilgard 430	SAE 40	30	A, B, C
	Mobilgard 440	SAE 40	40	A, B, C
	Mobilgard 50 M	SAE 40	50	A, B, C
	Mobilgard SP 55	SAE 40	55	A, B, C
Shell	Argina T 40	SAE 40	30	A, B, C
	Argina X 40	SAE 40	40	A, B, C
	Argina XL 40	SAE 40	50	A, B, C
Texaco	Taro 30 DP 40	SAE 40	30	A, B, C
	Taro 40 XL 40	SAE 40	40	A, B, C
	Taro 50 XL 40	SAE 40	50	A, B, C

The lubricating oils in table below, representing conventional additive technology, are also approved for use. However, with these lubricating oils, the service intervals will most likely be shorter.

NB! Different oil brands not to be blended unless approved by oil supplier and, during guarantee time, by engine manufacturer.

Table 7.3. Approved system oils: lubricating oils with conventional detergent/dispersant additive chemistry

Supplier	Brand name	Viscosity	BN	Fuel category
Esso	Exxmar 30 TP 40	SAE 40	30	A, B, C
	Exxmar 40 TP 40	SAE 40	40	A, B, C
Repsol	Neptuno 3000 SAE 40	SAE 40	30	A, B, C
	Neptuno 4000 SAE 40	SAE 40	40	A, B, C

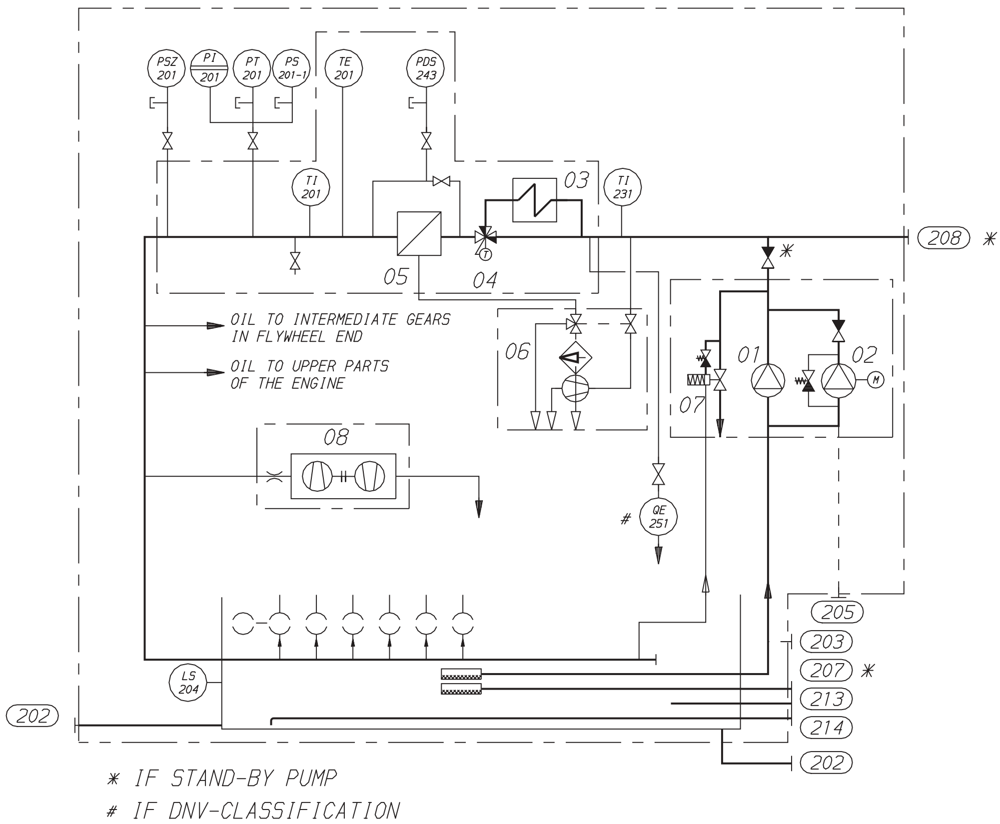
7.3. Internal lubricating oil system

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

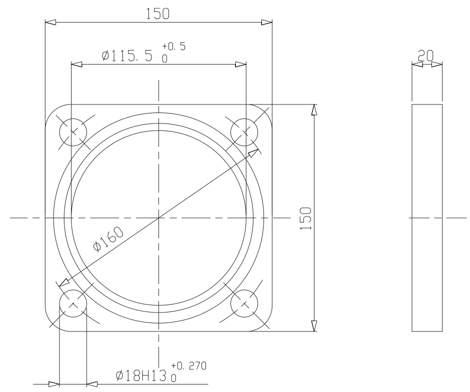
Dimensions of lubricating oil pipe connections on the engine

Pipe connections		Size	Pressure class	Standard
202	Lubrication oil outlet (dry sump)	DN100	see 4V32A0506	
203	Lubrication oil to engine driven pump (dry sump)	DN100	see 4V32A0506	
205	Lubrication oil to priming pump (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

Internal lubricating oil system (4V76E3854)



Flange for lubricating oil pump (4V32A0506a)



System components

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

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 pump

The prelubricating pump is an electric motor driven constant volume pump equipped with a safety valve.

The pump is of srew 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 prelubrication of a stopped engine through which heated heavy fuel is circulating
- Continuous prelubrication of a stopped diesel engine(s) in a multi-engine installation always when any one engine is running

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

7.3.3. Lubricating oil cooler

Lubricating oil cooler is brazed plate cooler and integrated in the lubricating oil module.

7.3.4. Thermostatic valve

Thermostatic valve is integrated in the lubricating oil module.

7.3.5. Lubricating oil automatic fine filter

Lubricating oil fine filter is back flushing automatic filter whose back flush is led to the centrifugal filter.

- Fine filter (full flow) 25 μm
- Safety net (full flow) 100 μm

7.3.6. Centrifugal filter

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

7.3.7. Lubricating oil module

Lubricating oil module, consisting of filters, thermostatic valve and oil cooler, is supported directly from the engine block.

7.4. External circulating 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

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

If necessary, 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.

- 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	8 bar
Operating temperature, max.	100°C
Lubricating oil viscosity	SAE 40

7.4.4. Lubrication oil gravity tank

In installations without an engine driven pump it is recommended to have a lubricating oil gravity tank arrangement for black-out situations.

7.5. Separation system

7.5.1. Separator (2N01)

For HFO the separator should be dimensioned for continuous centrifuging. For MDF intermittent centrifuging might be sufficient. Each lubricating oil system should have a separator of its own. The separator system must not be designed for water mixing when centrifuging.

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. In installations with four or more auxiliary engines two separators should be installed.

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

The separators should be dimensioned for continuous operation.

Design data:

- Centrifuging temperature 90 - 95°C

Capacity:

$Q = 1.36\ P\ n\ /\ t$

Where:

Q = volume flow [l / h]

P = total engine output

n = number of through-flows of dry sump system oil tank volume [1 / 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 lubrication 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 coking of the oil.

Design data

- For engines with 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 engines with centrifuging stopped engine, 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(2T04)

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

7.5.5. Renovated oil tank (2T05)

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. Lubricating oil storage 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.

7.6.2. Used oil handling tank (1T05)

Sludge tank can be used for the storage of used lubrication oil.

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.

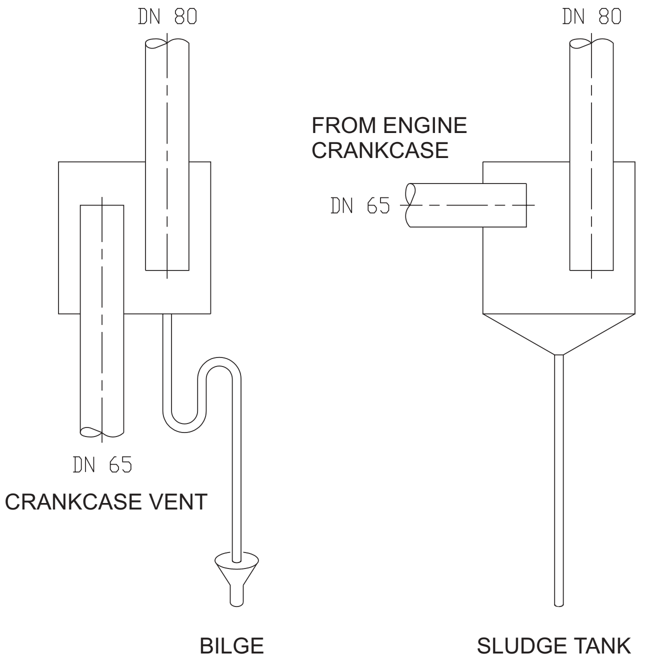
A condensate trap shall be fitted on all vent pipes within 1 - 2 meters of the engine, see drawing 4V76E2522.

Recommended size of the vent pipe after the condensate trap is NS 80

Pipe connection engine:

701 Crankcase air vent DN65, ISO 7005-1, NP16

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.

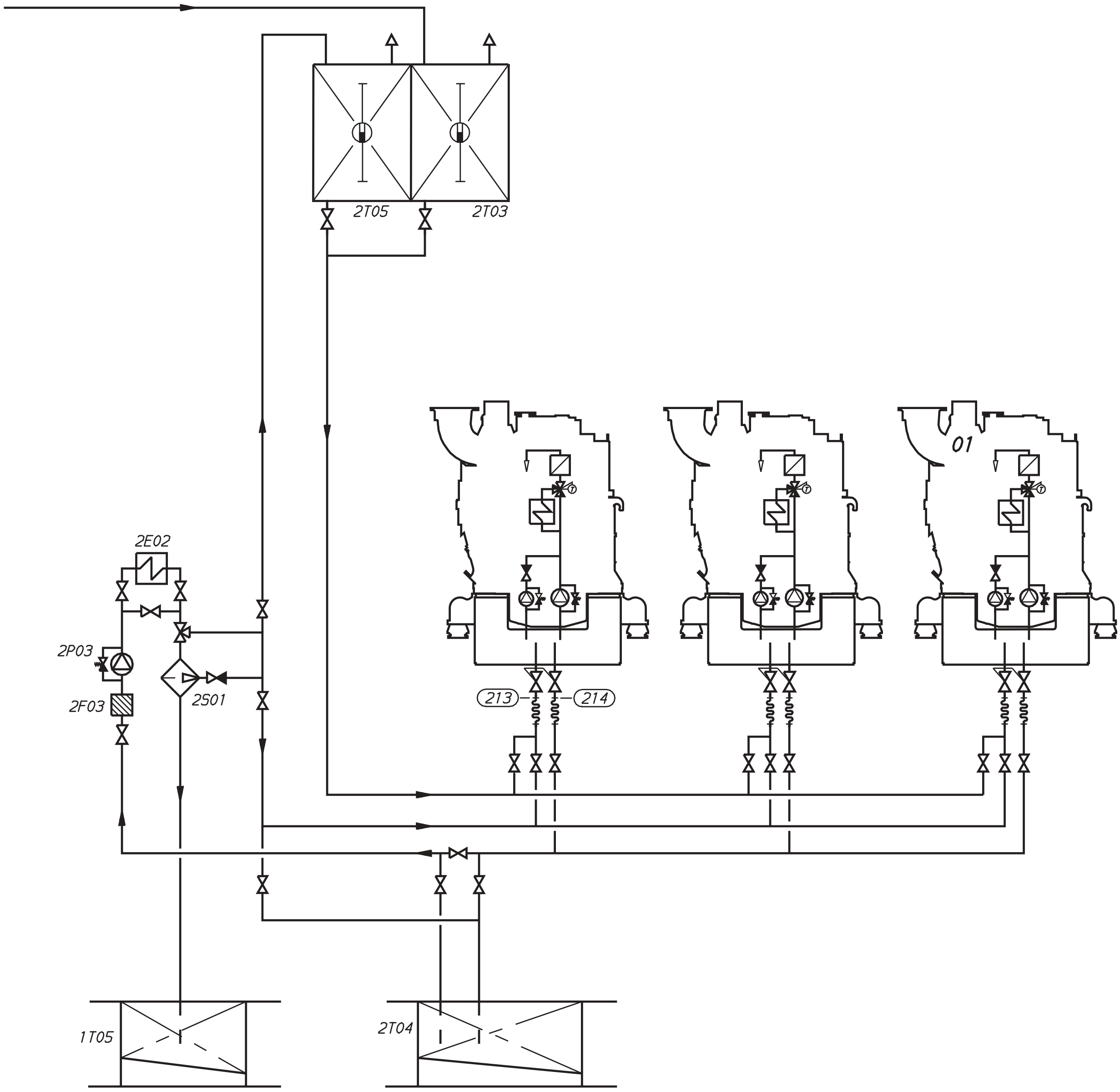
The flushing is more effective if the lubricating oil is heated. Furthermore, lubricating oil separators should be in operation prior to and during the flushing.

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. If an approved engine oil is used, it can be maintained provided that it is separated 4 - 5 times over after the flushing has been terminated and the filter inserts remain clean from any visible contamination.

7.9. System diagrams

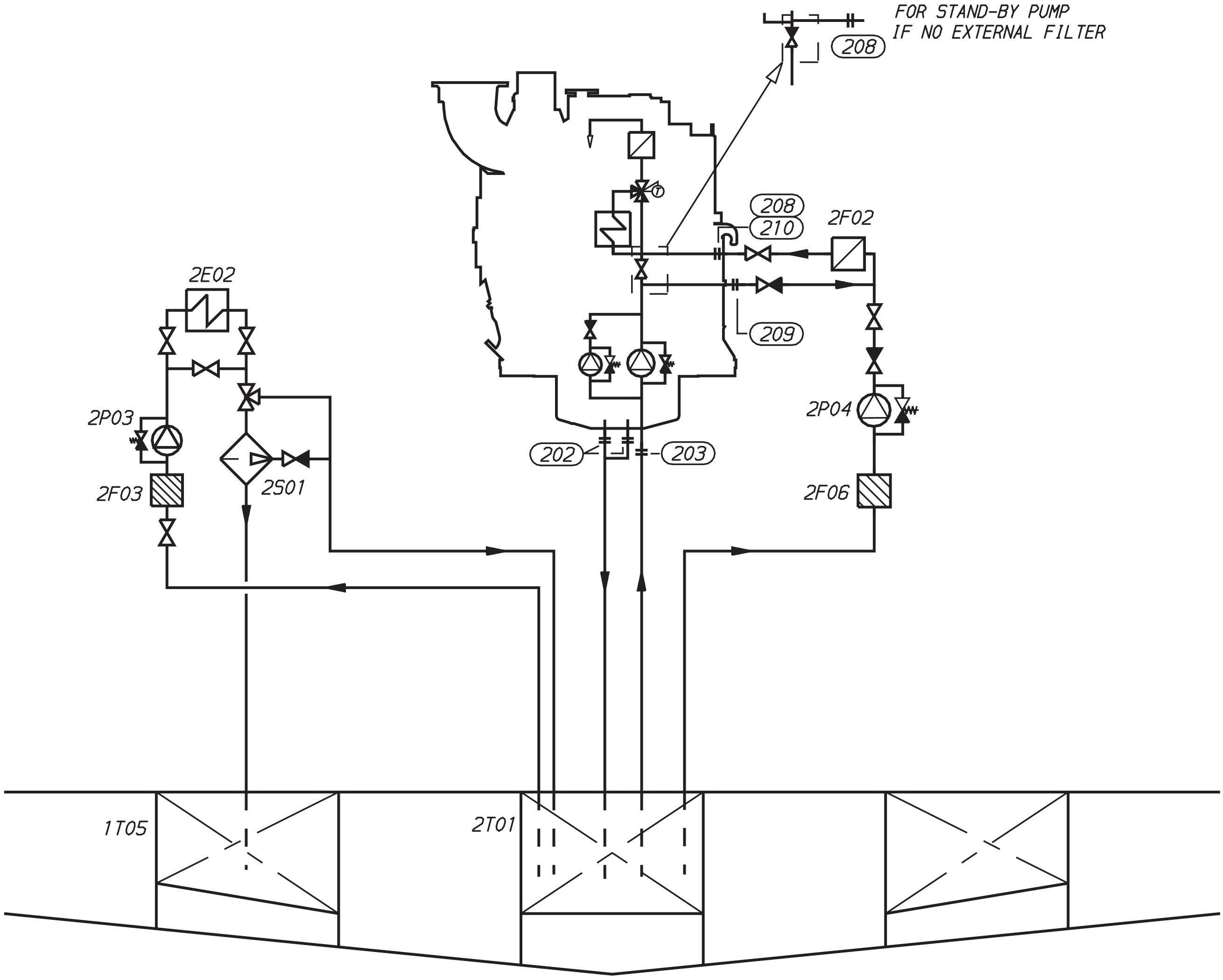
Lubricating oil system, auxiliary engines (3V76E3855)



System components	
1T05	Sludge tank
2E02	Heater
2F03	Suction strainer
2P03	Separator pump
2S01	Separator
2T03	New oil tank
2T04	Renovating oil tank
2T05	Renovated oil tank

Pipe connections	
213	Lubrication oil from separator and filling
214	Lubrication oil to separator and drain

Lubricating oil system, main engine (3V76E3856)



System components	
1T05	Sludge tank
2E02	Heater
2F02	Automatic filter
2F03	Suction strainer
2F06	Suction strainer
2P03	Separator pump
2P04	Stand-by lubrication oil pump
2S01	Separator
2T01	System oil tank

Pipe connections	
202	Lubrication oil outlet (from oil sump)
203	Lubrication oil to engine driven pump
208	Lubrication oil from electric driven pump
209	Lubrication oil to external filter
210	Lubrication oil from external filter

8. Compressed air system

8.1. General

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 in the classication regulations.

8.2. Compressed air quality

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.

8.3. Internal starting air system

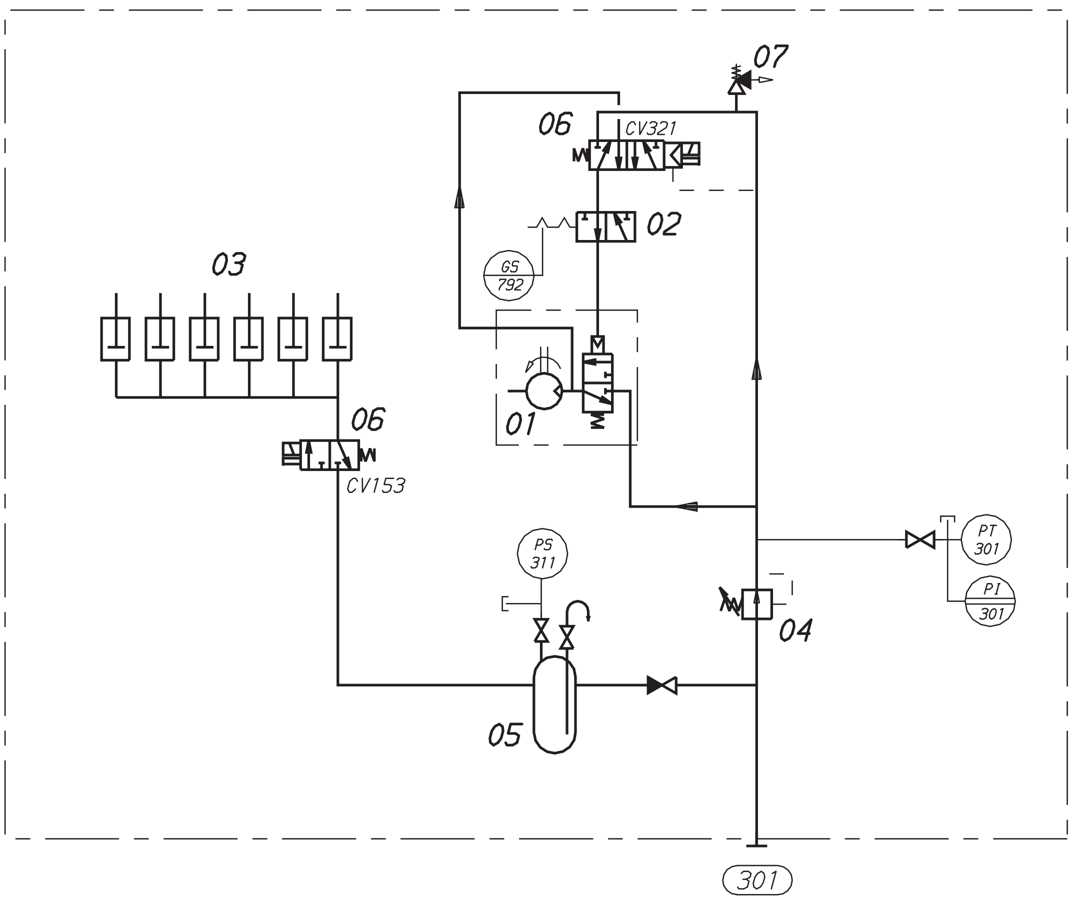
All engines are equipped with a pneumatic starting motor driving the engine through a gear rim on the flywheel.

Table 8.1. Dimensions of starting air pipe connections on the engine

Code	Description	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.

Internal starting air system (4V76H3460)



System components

01	Turbine starter with pneumatic actuator
02	Blocking valve, turning gear engaged
03	Pneumatic cylinder at each injection pump
04	Pressure regulator
05	Air container
06	Solenoid valve
07	Safety valve

Pipe connections

301	Starting air inlet

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.

8.4. 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 test-bed the starting air consumption per start value has to be increased in relation to total (engine included) inertial masses involved.

It should be noted that the minimum pressures stated in the chapter for technical data assume that this pressure is available at engine inlet.

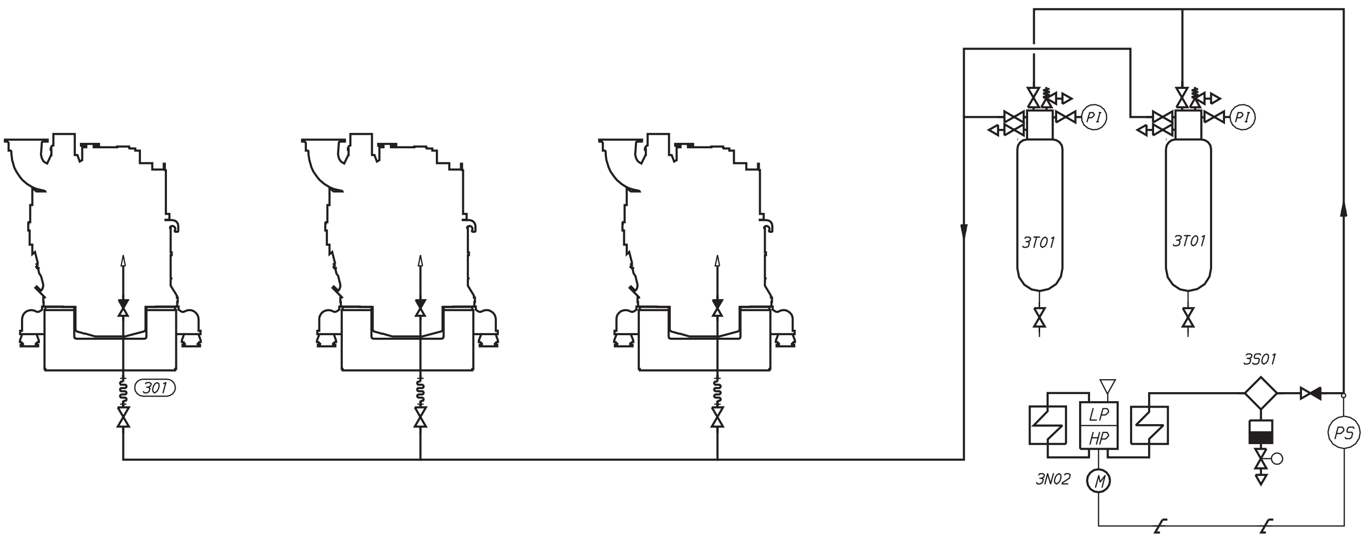
The rule requirements of some classification societies are not precise for multiple engine installations.

Starting air receiver (3T01)

The starting air receiver should be dimensioned for a nominal pressure of 30 bar.

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

External starting air system (3V76H3461)



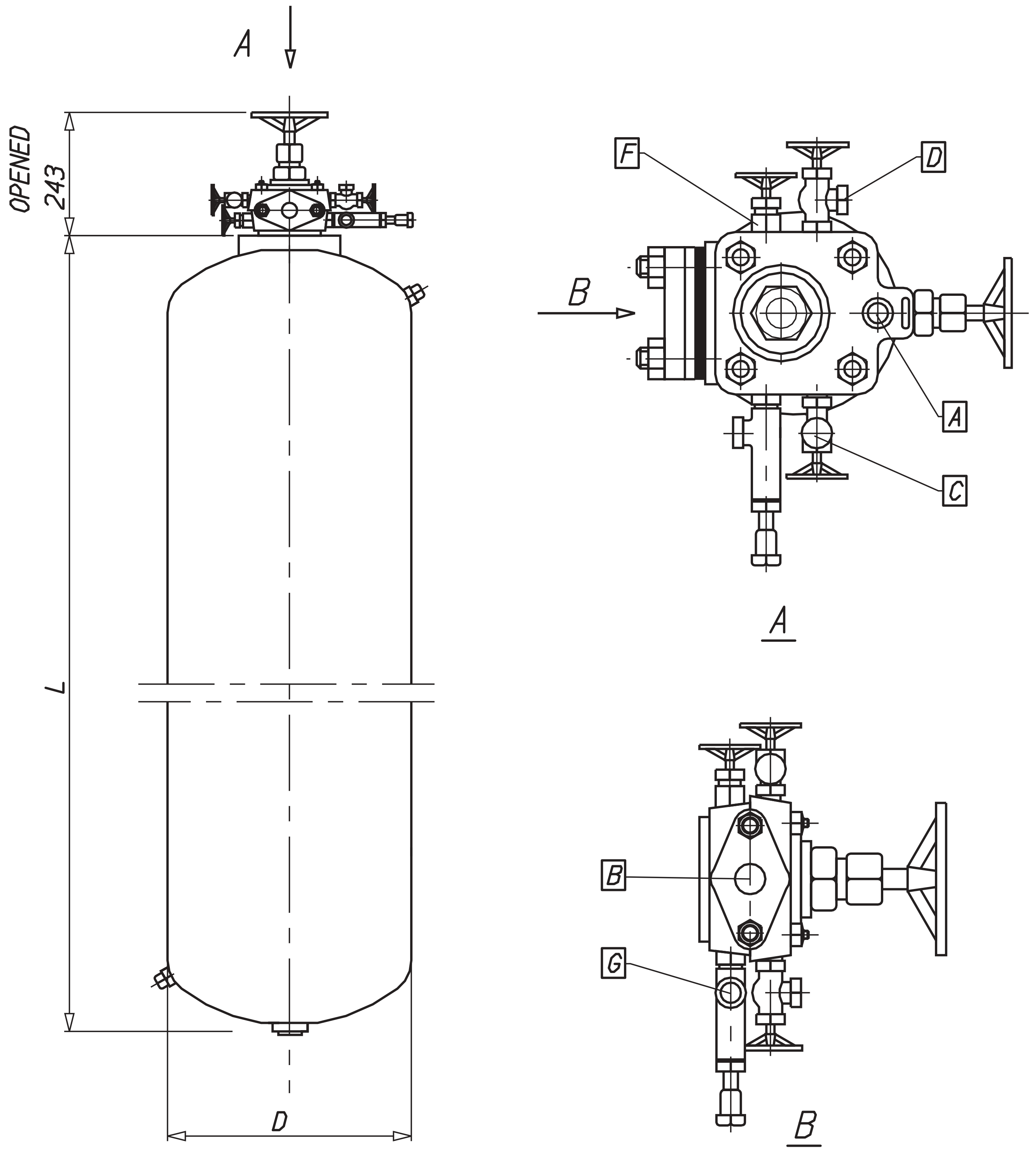
System components

3T01	Starting air vessel
3S01	Oil and water separator
3N02	Starting air compressor

Pipe connections

301	Starting air inlet

Starting air receiver (3V49A0097)



Connections		
A	Inlet	Ø 38
B	Outlet	R ¾ in
C	Pressure gauge	R ¼ in
D	Drain	R ¼ in
E	Drain	R ¼ in
F	Venting	
G	Safety valve	R ½ in

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

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

Recommended min. volumes of starting air vessels are:

- Single main engine driving CPP 2 x 125 l
- Single main engine driving FPP 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

Oil and water separator

- 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.

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). To prevent rust formation in the cooling water system, the use of corrosion inhibitors is mandatory. See the instructions in the Instruction Manual.

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).

Sea-water will cause severe corrosion and deposits formation even in small amounts.

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

Product	Supplier
Drewgard 4109 Maxigard DEWT-NC powder Liquidewt Vecom CWT Diesel QC-2	Drew Ameroid Marine Division, Ashland Chemical Company Boonton, USA
Dearborn 547	Grace Dearborn Ltd. Widnes, Chesire, U.K
Cooltreat 651	Houseman Ltd. Burnham, Slough, U.K.
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
Texaco ETX6282	S.A. Arteco N.V. Belgium
Ruostop XM	Tampereen Prosessi-Insinöörit Tampere, Finland
Dieselguard NB Rocor NB liquid	Unitor A/S Kolbotn, Norway
Vecom CWT Diesel	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. Charge air cooler

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

Design data:

- See Technical data

9.2.2. 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.

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

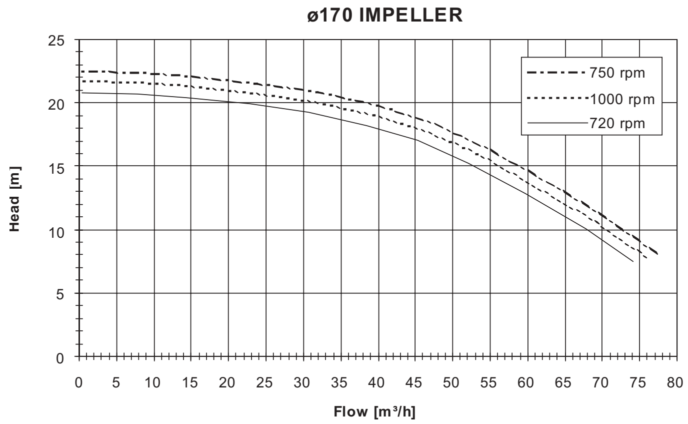
Material:

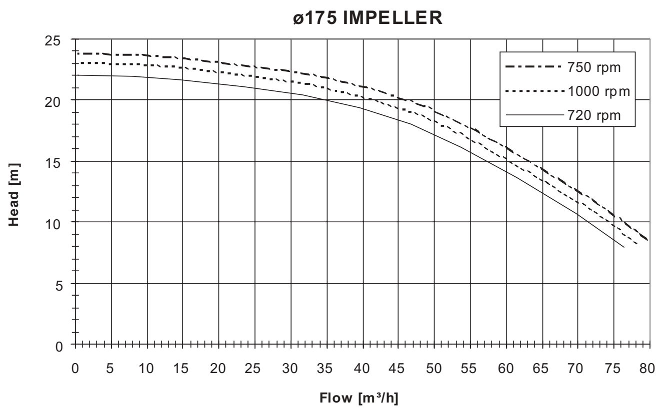
- housing cast iron
- impeller bronze
- shaft stainless steel
- sealing mechanical

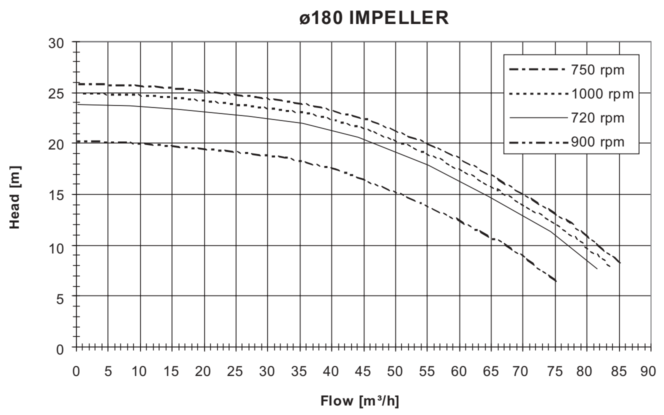
Capacities are according to Chapter for Technical data and the pump curves below.

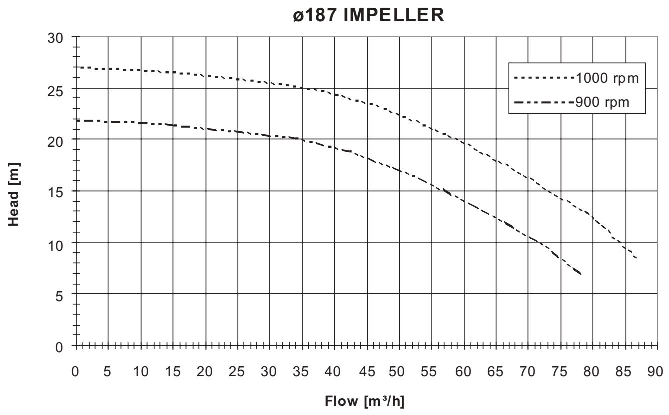
Table 9.1. Impeller diameters and nominal flows of engine driven HT & LT pumps

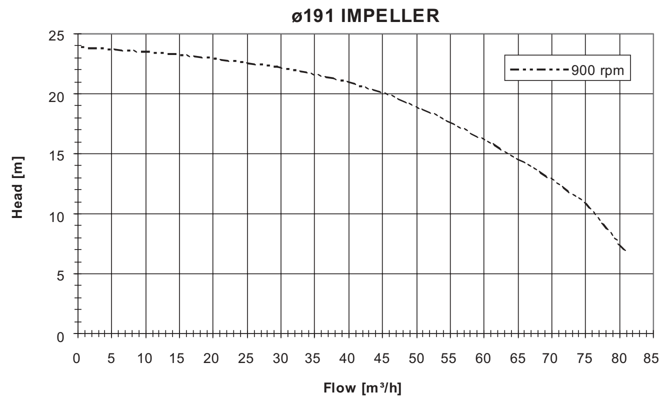
Engine	Engine speed [RPM]	HT impeller [Ø mm]	LT impeller [Ø mm]
4L20	720	170	170
	750	170	170
	900	180	187
	1000	170	170
5L20	900	187	187
	1000	170	170
6L20	720	175	175
	750	175	175
	900	187	187
	1000	175	175
8L20	720	180	180
	750	180	180
	900	191	197
	1000	180	187
9L20	720	180	180
	750	180	180
	900	191	197
	1000	180	187

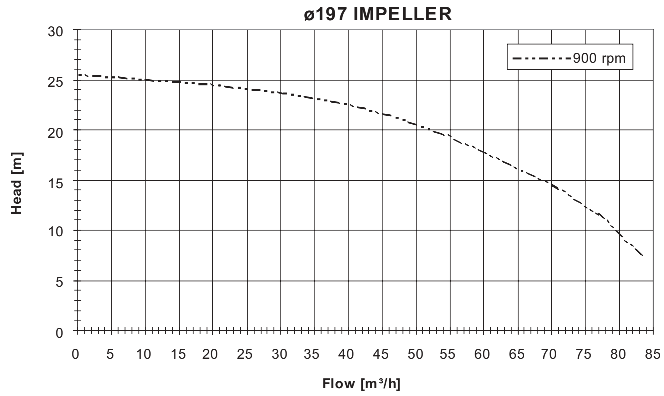












9.2.3. Engine driven sea water pump (4P11)

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

Capacity [m³/h]:

4L20:	40
5L20:	60
6L20:	60
8L20:	104
9L20:	104

Head about 20 meters water column

9.2.4. Thermostatic valve LT-circuit (4V03)

The thermostatic valve for the LT-circuit is arranged to control the outlet temperature of the water on engines. The thermostatic valve has one fixed set point of 49°C with 38°C as fully closed and 50°C as fully open and it is of the direct acting type.

9.2.5. Thermostatic valve HT-circuit (4V01)

The thermostatic valve for the HT-circuit is arranged to control the outlet temperature of the water. It is of the direct acting type.

- set point of the HT-thermostatic valve 91°C

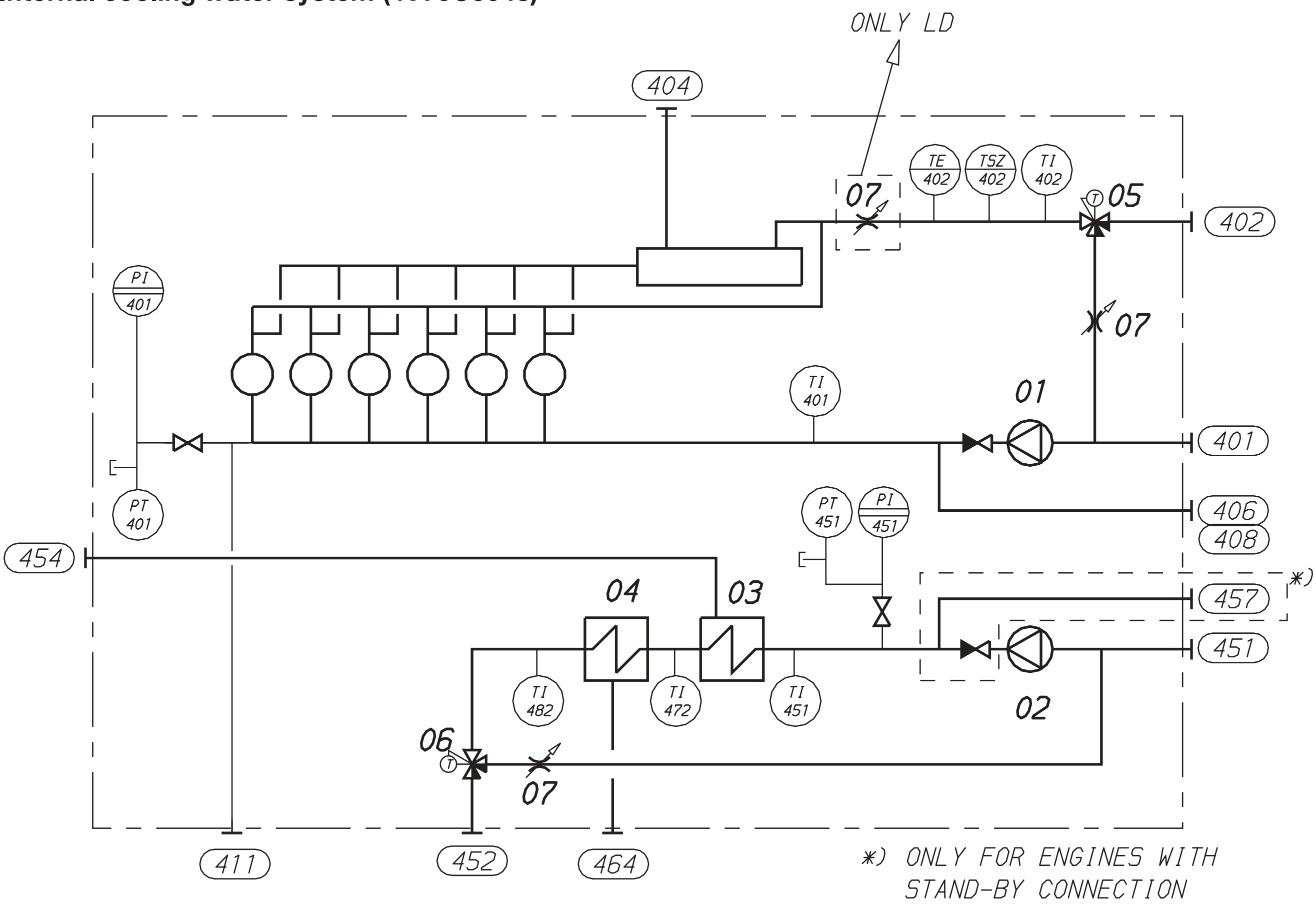
9.2.6. Lubricating oil cooler (2E01)

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

Dimensions of water pipe connections on the engine

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*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*1.5	-	Plug

Internal cooling water system (4V76C5048)



System components

01	HT-cooling water pump
02	LT-cooling water pump
03	Charge air cooler
04	Lubrication oil cooler
05	HT-thermostatic valve
06	LT-thermostatic valve
07	Adjustable orifice

9.3. External cooling water system

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.

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 are usually electrically driven. The capacity of the pumps is determined by the type of coolers used 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 is used.

Design data:

- Fresh water flow see Technical Data

In case of fresh water central cooler is used for both LT and HT water the fresh water flow can be calculated with the following formula:

$$q = q_{LT} + \frac{3.6 \cdot \Phi}{4.19 \cdot (T_{out} - T_{in})}$$

where:

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

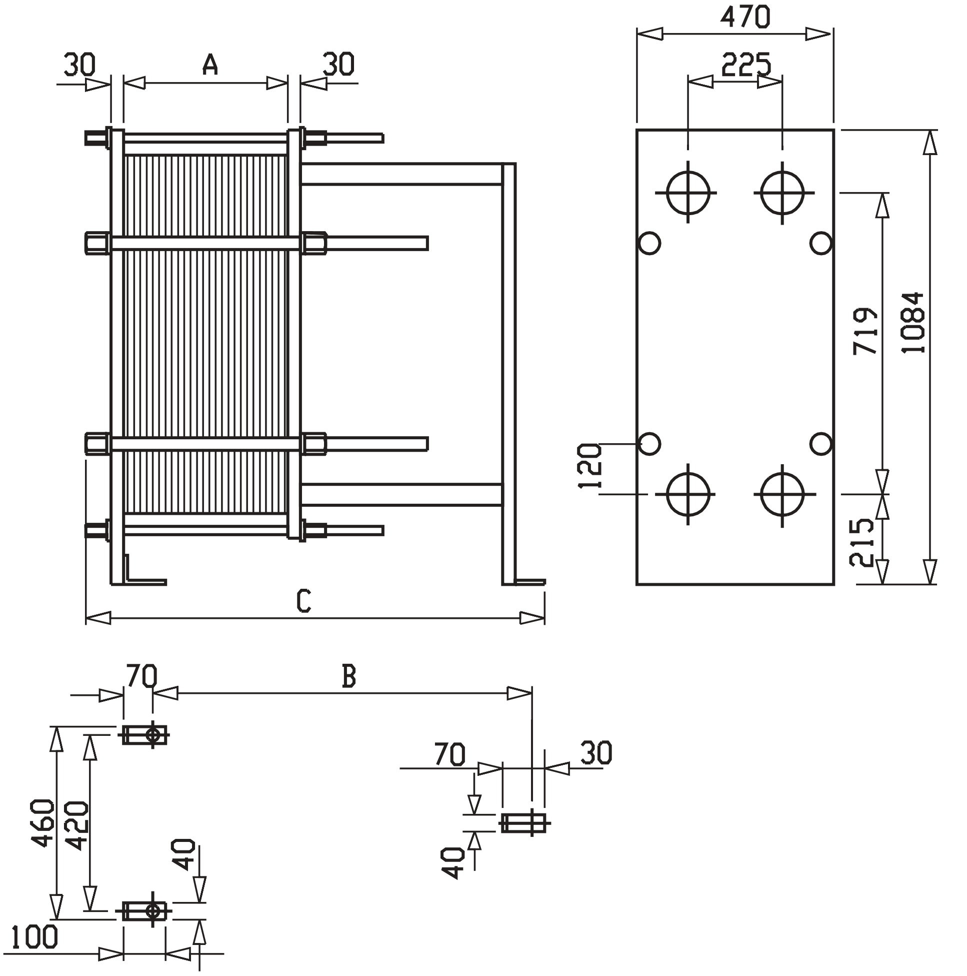
- Pressure drop on fresh water side, max. 60 kPa (0.6 bar)

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 cooler (before engine), max. 38°C. see Technical Data
- Safety margin to be added 15% + margin for fouling

See also the table showing example coolers with calculation data.

Central cooler, main dimensions (4V47E0188b)



Engine		Cooling water			Sea-water			Measures			Weight	
Type	[RPM]	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
5L20	1000	33	52.9	38	44	32	43.2	121	655	845	280	306
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

9.3.3. Stand-by circulating cooling water pumps

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

9.3.4. Expansion tank

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 0.7...1.5 bar
- volume min. 10% of the system

Concerning engine water volumes, see Chapter for 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 Technical data (HT-circuit).

Most of the cooling water in the engine can be recovered from the HT-circuit.

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 two 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
- required heating power 2 kW/cyl.

Preheating pump (4P04)

Design data of the pump:

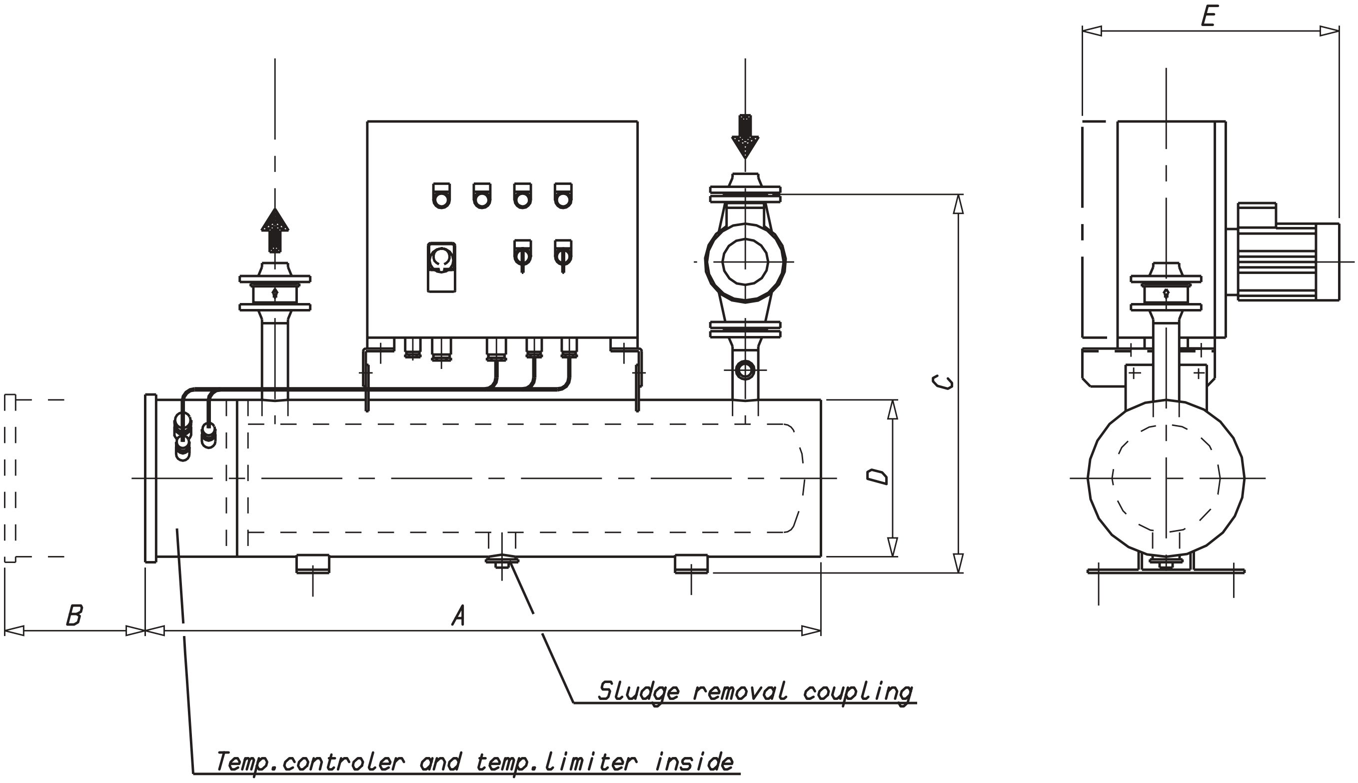
- capacity 0.3 m³/h x cyl.
- pressure abt. 80 kPa (0.8 bar)

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

Preheating unit, electric (3V60L0653a)



Heater capacity	Pump capacity	Weight	Pipe conn.	Dimensions				
kW	m3/h	kg	In / Outlet	**A**	**B**	**C**	**D**	**E**
7.5	3	75	DN40	1050	720	610	790	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
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

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 air venting equipment is installed in the LT system line for venting of any entrained air.

9.3.8. Local thermometers

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

9.3.9. Pressure gauges

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

9.3.10. Orifices

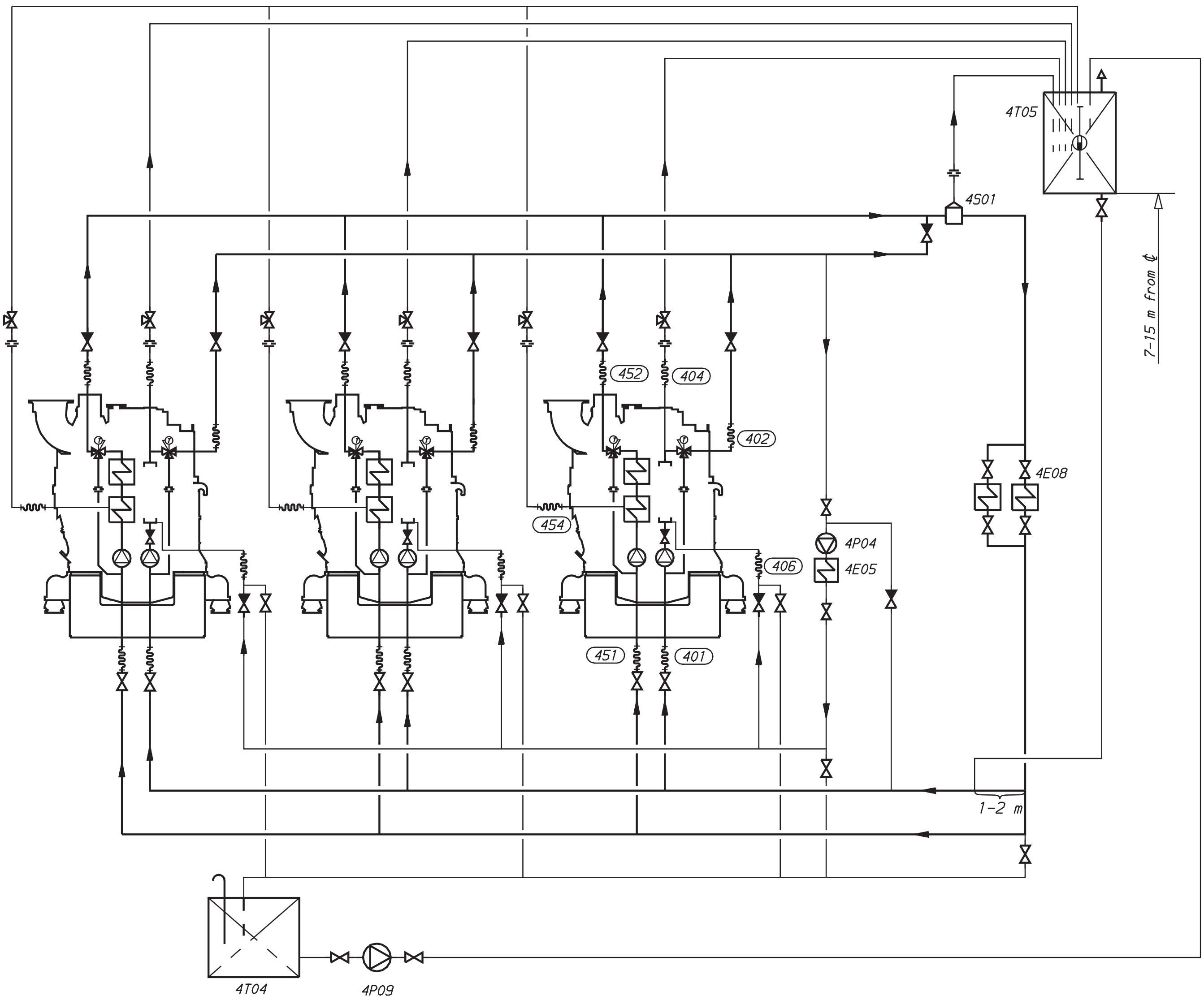
Orifices must be mounted after the HT outlet, after lubricating oil cooler and in all by-pass lines in order to adjust the circulations pumps and to balance the pressure drop when the water is not flowing through the cooler.

9.3.11. 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

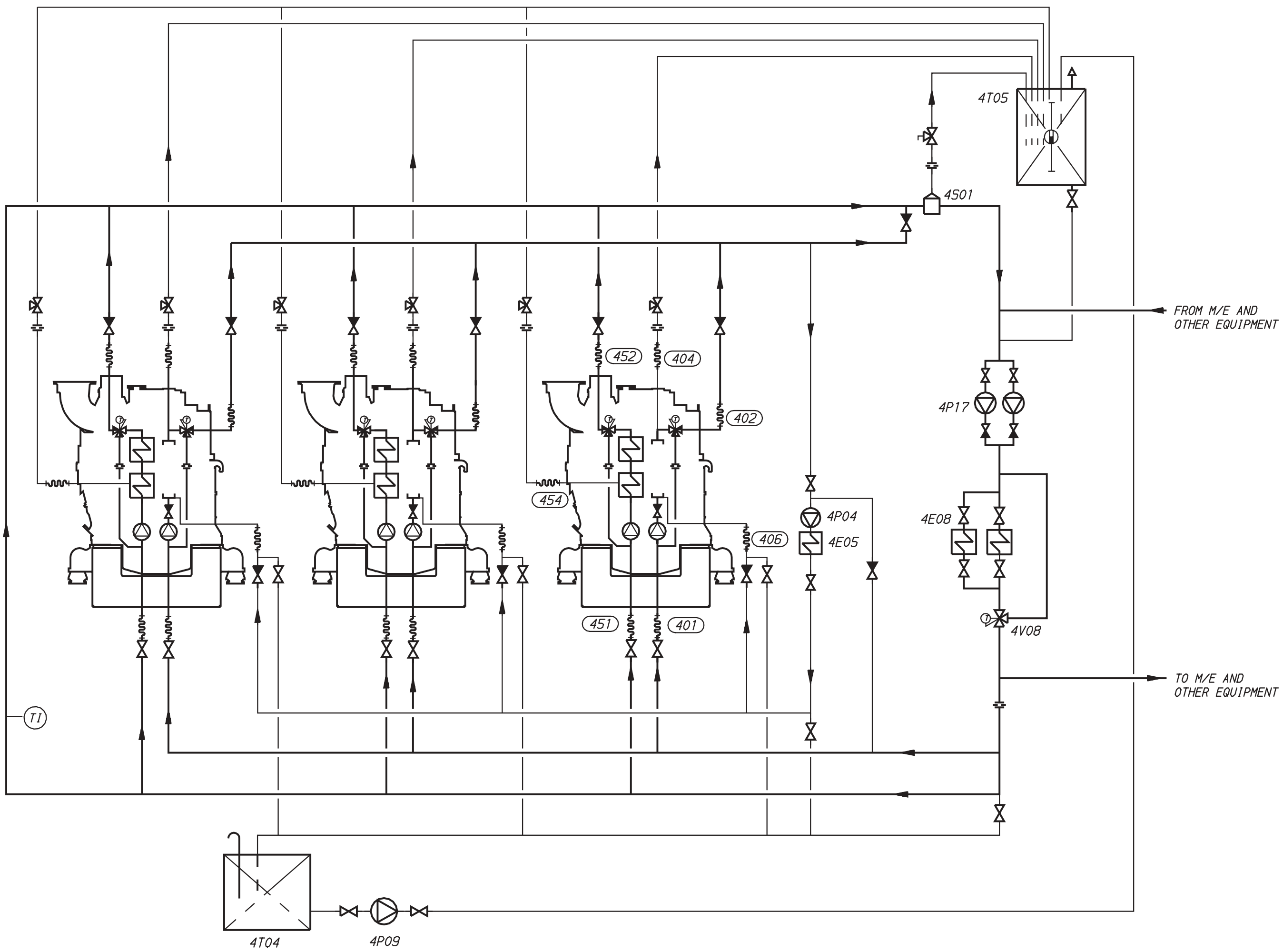
Cooling water system, auxiliary engines (3V76C5049)



System components	
4E05	Preheater
4E08	Central cooler
4P04	Preheating pump
4P09	Transfer pump
4S01	Air venting
4T04	Drain tank
4T05	Expansion tank

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

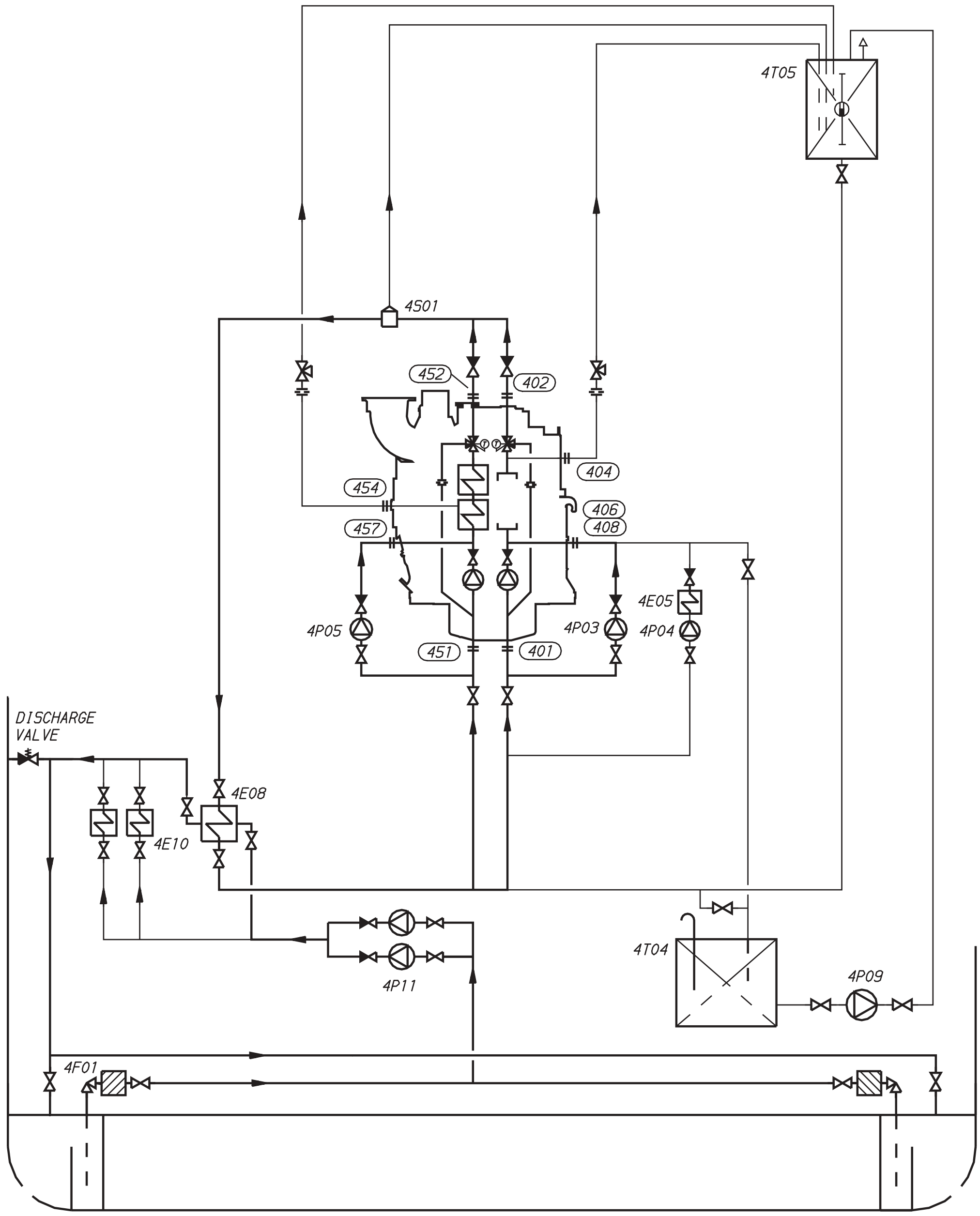
Cooling water system, auxiliary engines and main engine (3V76C5050)



System components	
4E05	Preheater
4E08	Central cooler
4P04	Preheating pump
4P09	Transfer pump
4P17	LP-pump
4S01	Air venting
4T04	Drain tank
4T05	Expansion tank
4V08	Thermostatic valve

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

Cooling water system, main engine (3V76C5051)



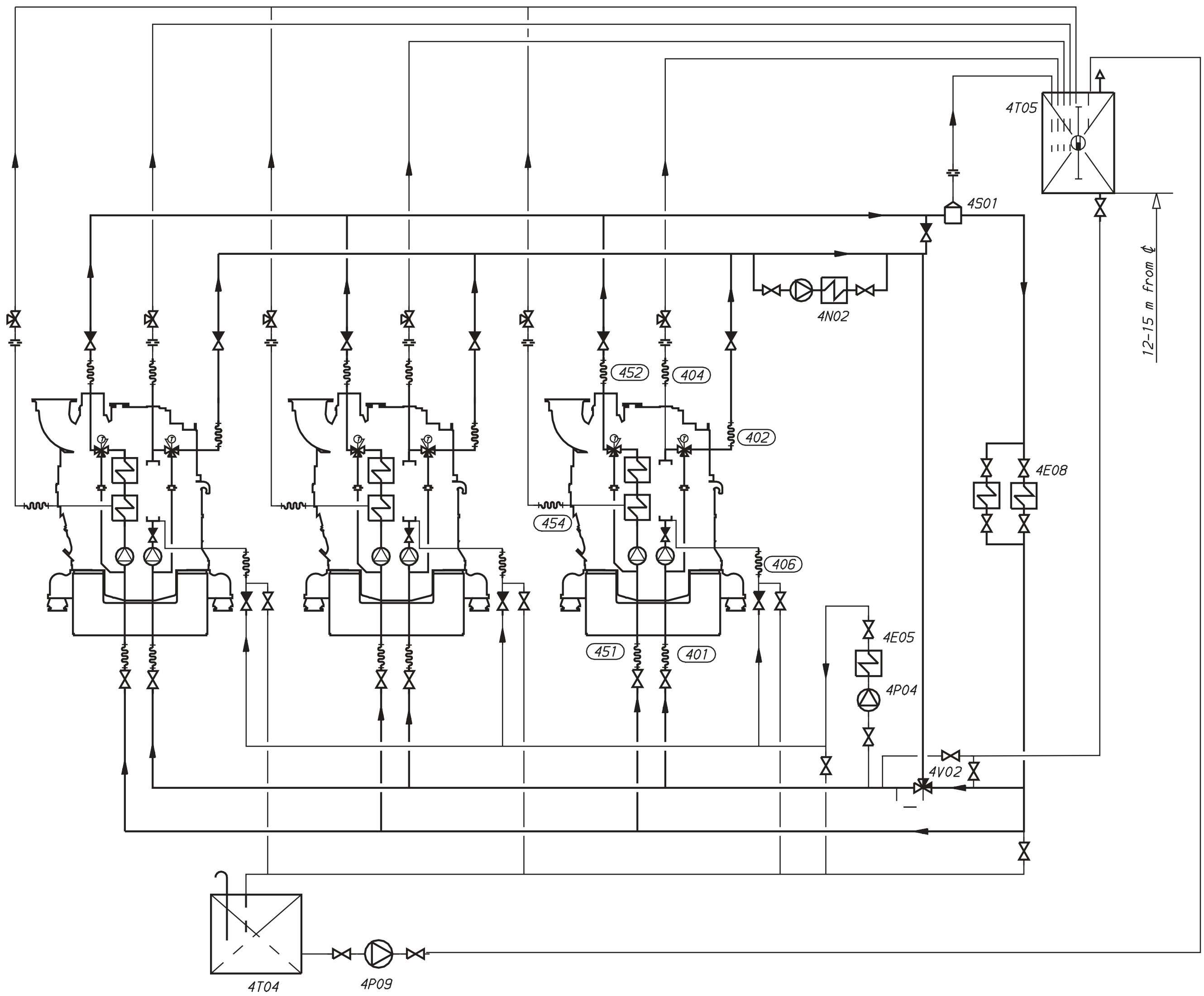
System components

4E05	Preheater
4E08	Central cooler
4E10	Gear cooler
4F01	Sea water filter
4P03	HT-stand-by pump
4P04	Preheating pump
4P05	LT-stand-by pump
4P09	Transfer pump
4P11	Sea-water pump
4S01	Air venting
4T04	Drain tank
4T05	Expansion tank

Pipe connections

401	HT-water inlet
402	HT-water outlet
404	HT-air vent
406	Water from preheater to HT circuit
408	HT-water from stand-by pump
451	LT-water inlet
452	LT-water outlet
454	LT-water air vent from air cooler

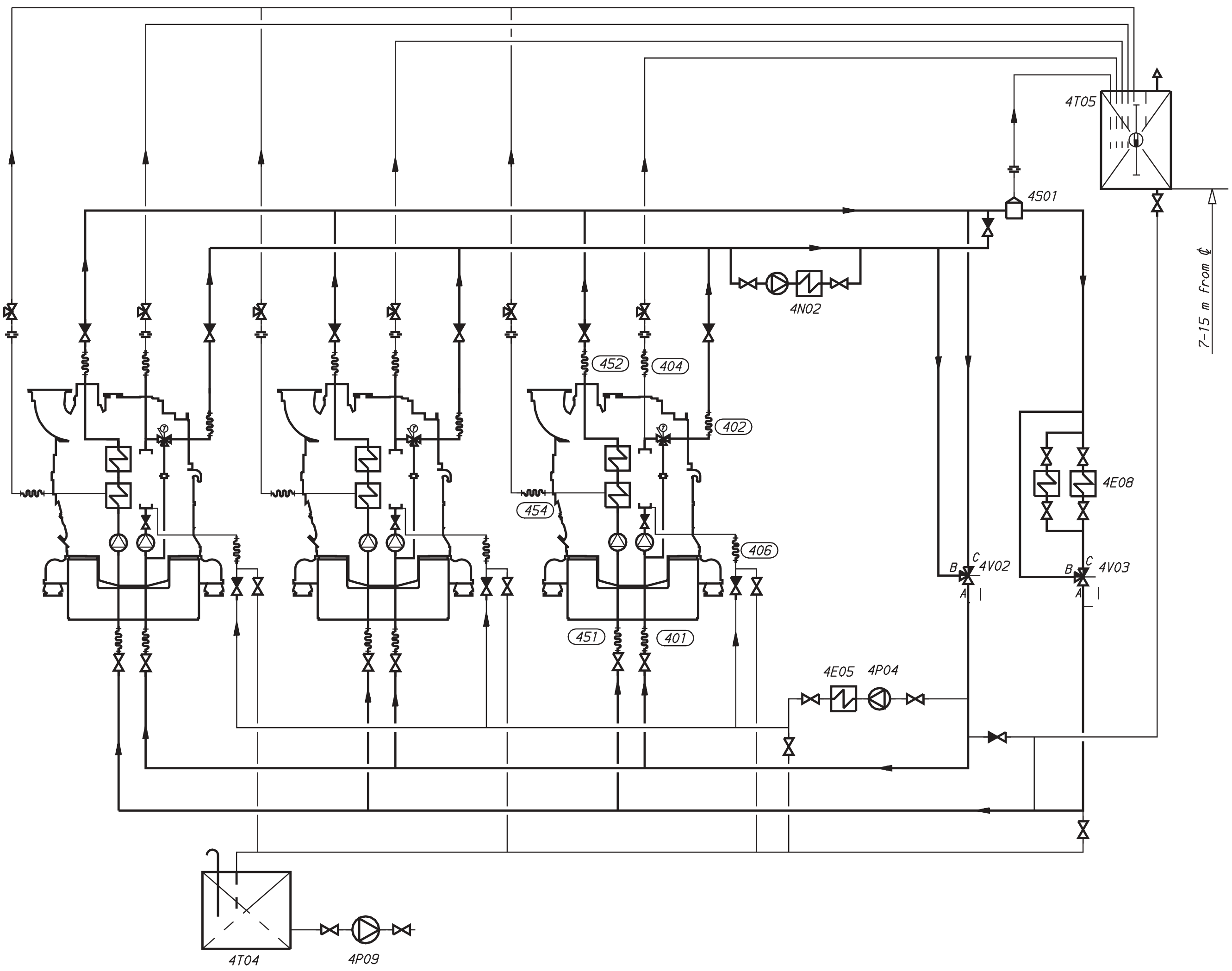
Cooling water system, HFO engines with evaporator (3V76C5052)



System components	
4E05	Preheater
4E08	Central cooler
4N02	Evaporator
4P04	Preheating pump
4P09	Transfer pump
4S01	Air venting
4T04	Drain tank
4T05	Expansion tank
4V02	Thermostatic valve

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

Cooling water system, MDO engines with evaporator (3V76C5053)



System components	
4E05	Preheater
4E08	Central cooler
4N02	Evaporator
4P04	Preheating pump
4P09	Transfer pump
4S01	Air venting
4T04	Drain tank
4T05	Expansion tank
4V02	Thermostatic valve
4V03	LT-thermostatic valve

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

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.

Ventilation

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
- Alternators
- Electric appliances and lighting
- Boilers
- Steam and condensate piping
- Tanks

It is recommended to consider an outside air temperature of not 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 from the formula:

$$Q_v = \frac{\Phi}{\rho \cdot \Delta t \cdot c}$$

where:

Q_v = amount of ventilation air [m³/s]

Φ = total heat emission to be evacuated [kW]

ρ = density of ventilation air 1.15 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 the chapter for Technical data.

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.

10.2. Combustion air quality

During normal operating conditions the air temperature at the turbocharger inlet should be kept between 15ºC and 35ºC. Temporarily max. 45ºC is allowed.

10.3. Combustion air system design

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.

The combustion air should be delivered through a dedicated duct close to the turbocharger, directed towards the turbocharger air intake. Also auxiliary engines shall be served by dedicated combustion air ducts.

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

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 mmWC.

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 also regulated in the classification rules for Offshore Units as mandatory.

10.3.1. Cold operating conditions

In installations intended for operation in cold air conditions, restrictions for operation at low air temperature must be considered. See Chapter for Operation ranges for details.

Combustion air for engines

- Each engine has its own combustion air fan, with a capacity slightly higher than the maximum air consumption. The fan should have a two-speed electric motor (or variable speed) for enhanced flexibility. In addition to manual control, the fan speed can be controlled by the engine load.

- The combustion air is conducted close to the turbocharger, the outlet being equipped with a flap for controlling the direction and amount of air.

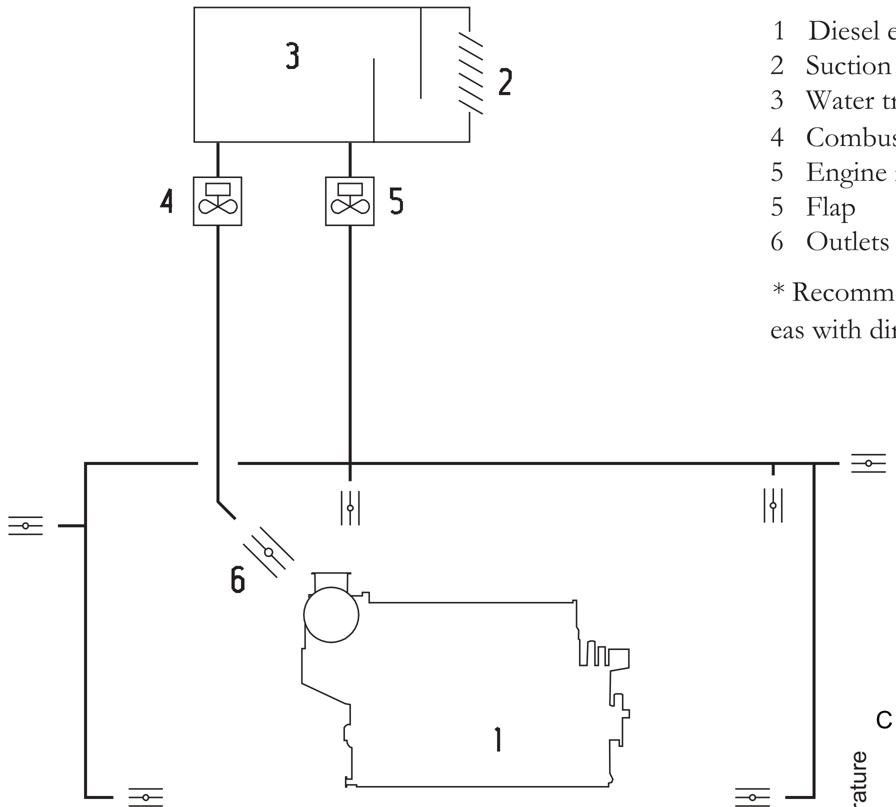
With these arrangements the normally required minimum air temperature to the main engine, see Chapter for operation ranges, can typically be maintained. For lower temperatures special provisions are necessary.

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.

Engine room ventilation

- The rest of the engine room ventilation is provided by separate ventilation fans. These fans should preferably have two-speed electric motors (or variable speed) for enhanced flexibility.
- 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.
- This system permits flexible operation, 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).

Engine room ventilation (4V69E8169)



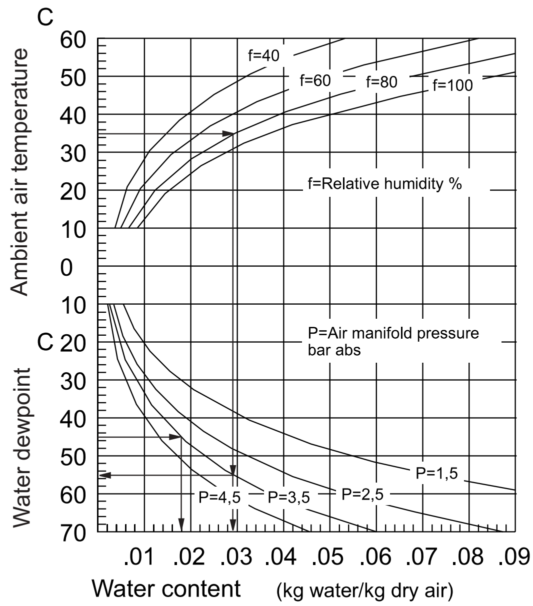
1 Diesel engine
2 Suction louver *
3 Water trap
4 Combustion air fan
5 Engine room ventilation fan
5 Flap
6 Outlets with flaps

* Recommended to be equipped with a filter for areas with dirty air (rivers, coastal areas, etc.)

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.

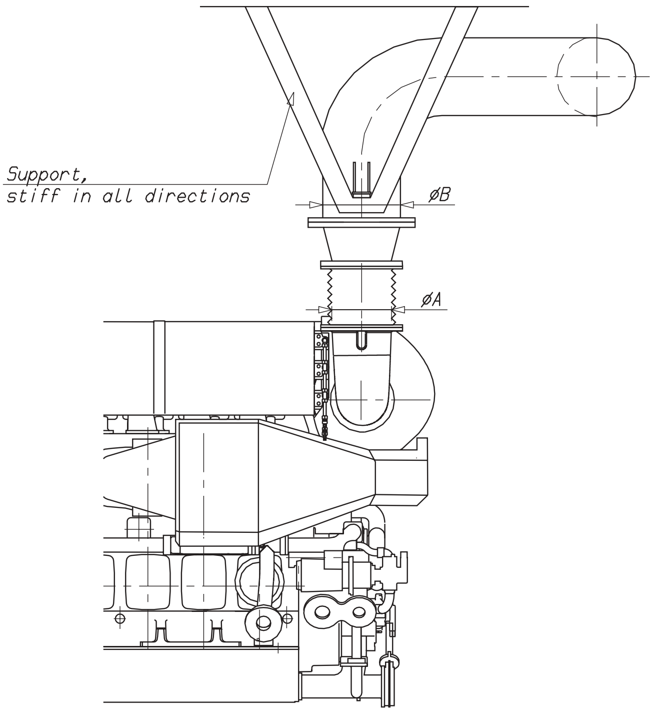


11. Exhaust gas system

11.1. Internal exhaust gas system

11.1.1. Exhaust gas outlet

Exhaust gas outlet (4V76A2679)



Engine	Bellows A (inner dia)	Piping B (inner dia)
4L20	200	250-300
5L20	250	300-350
6L20	250	300-350
8, 9L20	300	350-450

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

11.2. External exhaust gas system

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.

It is very important that the exhaust pipe is properly fixed to a rigid support directly after the bellows. Resilient mounts are acceptable at the fixing points between the exhaust pipe and the rigid support. The mounts must however be stiff enough to prevent dynamic deflections in excess of 1 mm peak to peak. Conical rubber mounts similar to the mounts that are installed under generating sets can be used. Adequate thermal insulation must be provided to protect the rubber mounts from high exhaust gas temperatures.

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 the Chapter for Technical data can be translated to velocity using the formula:

$$v\,[m/s] = \frac{4 \cdot m}{1.3 \cdot \left(\frac{273}{273 + t}\right) \cdot \pi \cdot D^2}$$

Where:

v[m/s] = gas velocity

m [kg/s] = exhaust gas mass flow

t [°C] = exhaust gas temperature

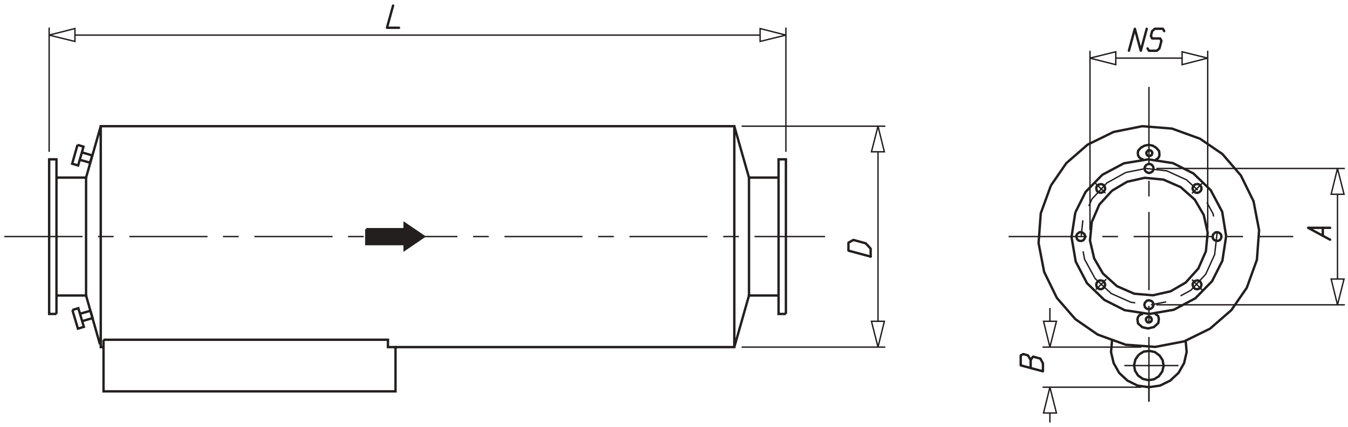
D [m] = exhaust gas pipe diameter

11.2.1. Silencer (5R01)

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).

Exhaust silencer (4V49E0137a)



				Attenuation			
				25 dB (A)		35 dB (A)	
DN	D	A	B	L	Weight (kg)	L	Weight (kg)
250	700	335	120	2070	230	2870	340
300	700	395	150	2600	280	3600	400
350	850	445	180	2640	340	3640	490
400	950	495	205	3180	500	4180	670
450	1100	550	230	3440	600	4440	780

11.2.2.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 the Chapter for Technical data may be used.

11.2.3.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.

11.2.4.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, framewebs or specially constructed supports.

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

11.2.5.Back pressure

The maximum permissible exhaust gas back pressure is 3 kPa 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 and air cooler cleaning

12.1. Turbine cleaning system (5Z03)

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.

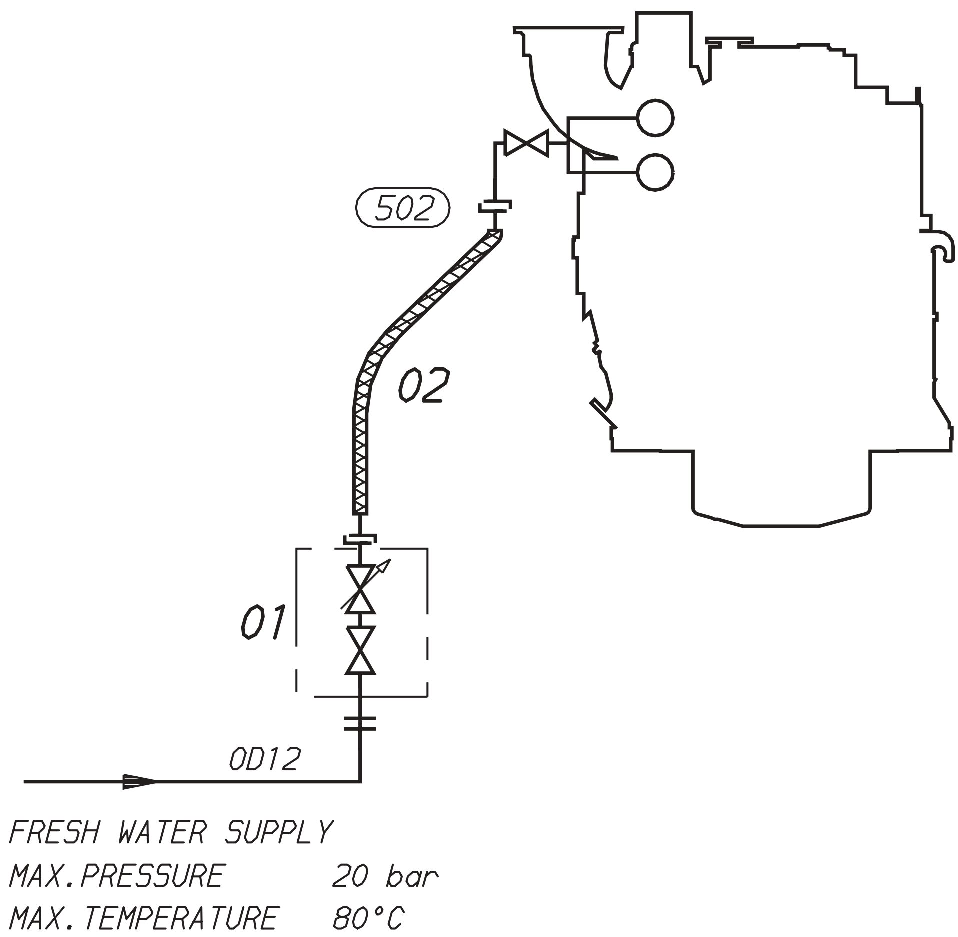
The washing is carried out during operation at regular intervals, depending on the quality of the heavy fuel, 100 – 500 h.

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
- 5L20, 6L20: 8 l/min
- 8L20, 9L20: 10 l/min

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

Turbocharger cleaning system (4V76A2709)



System components		Pipe connections	
01	Shut of and flow adjusting unit, bulkhead mounted	502	Cleaning water to turbine
02	Rubber hose about 10 m		Quick coupling

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 (SOx) and nitrogen oxides (NOx), partially reacted and non-combusted hydrocarbons and particulates. Emission control of large diesel engines means primarily the control of the NOx emissions.

To improve the combustion process and reduce the emissions, especially NOx emissions, Wärtsilä has developed a Low NOx combustion process that substantially reduces the NOx level without compromising thermal efficiency. The Low NOx combustion concept has been implemented in all Wärtsilä engines.

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 (NOx). The emissions of sulphur oxides (SOx) 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 (NOx)

Nitric oxide (NO) and Nitrogen dioxide (NO_2) are usually grouped together as NOx 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 10-30 %.

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 it is required. For Wärtsilä 20, the Selective Catalytic Reduction (SCR) is an optional NOx reduction method.

13.2.2. Sulphur Oxides (SOx)

Sulphur oxides (SOx) are direct result of the sulphur content of the fuel oil. During the combustion process the fuel bound sulphur is rapidly oxidised to sulphur dioxide (SO_2). A small fraction of SO_2 may be further oxidised to sulphur trioxide (SO_3). The SOx 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 NOx 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 Organisation) 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 has yet to be ratified. The regulations will enter into force 12 months after the date on which at least 15 states, constituting not less than 50 % of the gross tonnage of the world's merchant shipping, have signed the protocol. The most important regulation of the MARPOL Annex VI is the control of NOx emissions.

All Wärtsilä engines are Low NOx engines and comply with the proposed NOx levels set by the IMO in the MARPOL Annex VI. The NOx controls apply only 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.

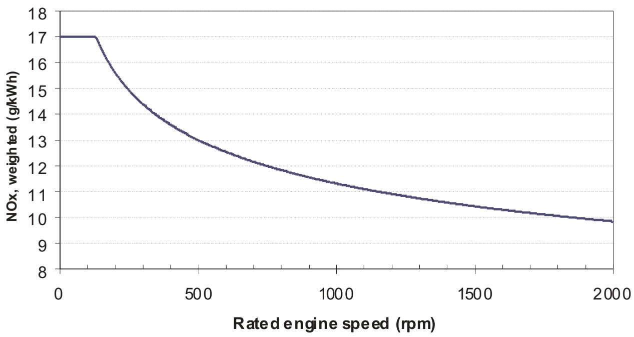
For Wärtsilä 20 with a rated speed of 720 rpm, the NOx level is below 12.1 g/kWh and with 750 rpm the level is below 12.0 g/kWh. With a rated speed of 900 rpm the NOx level is below 11.5 g/kWh and with 1000 rpm the level is below 11.3 g/kWh. The tests are done according to IMO regulations (NOx Technical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

The IMO NOx limit is defined as follows:

NOx [g/kWh]

= 17	rpm < 130
= 45 x rpm-0.2	130 < rpm < 2000
= 9.8	rpm > 2000

IMO NOx emission limit



13.3.2. EIAPP Statement of Compliance

An EIAPP (Engine International Air Pollution Prevention) Statement of Compliance will be issued for each engine showing that the engine complies with the NOx regulations set by the IMO. For the time being only a Statement of Compliance can be issued, because the regulation is not yet in force.

When testing the engine for NOx emissions, the reference fuel is Marine Diesel Fuel (distillate) and the test is performed according to ISO 8178 test cycles. Subsequently, the NOx 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 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	50	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 NOx 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 easily identified by the surveyor and thus an IAPP (International Air Pollution Prevention) Statement of Compliance for the ship can be issued on basis of the EIAPP Statement of Compliance 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. Selective Catalytic Reduction (SCR)

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

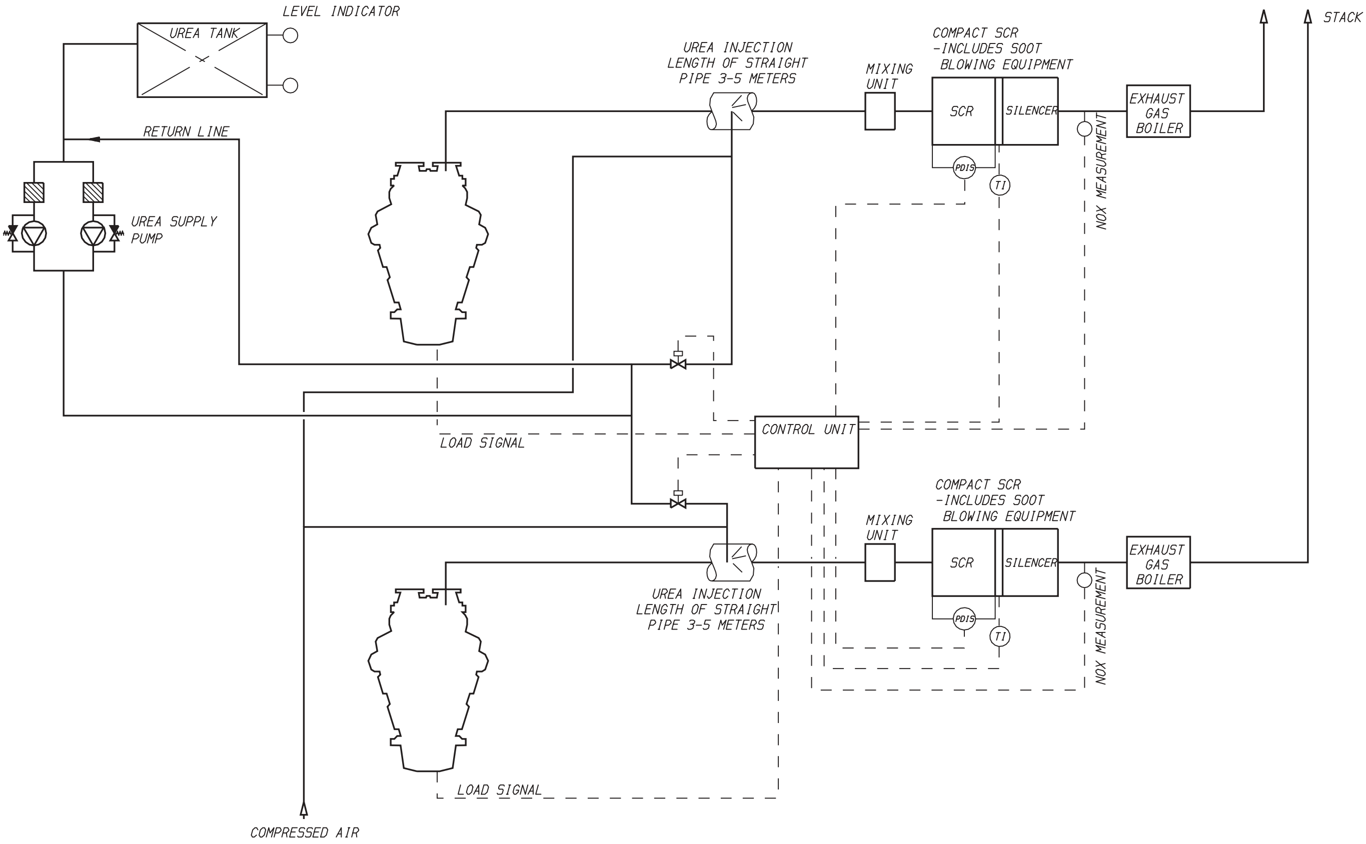
General system description

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 NOx 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 NOx measuring device after the catalyst. The rate of NOx reduction depends on the amount of urea added, which can be expressed as NH_3/NOx 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.

Typical P&ID for Compact SCR (3V28A0006a)



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 15-20 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 relative autonomy.

14. Automation system

14.1. General

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

Automation system requires depending on the scope 24 VDC about 150 W.

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

The principle diagram of a start/stop system is shown in drawing 4V50G3472.

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.

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

Start blocking (8N08)

Starting shall be inhibited by the following functions:

- Turning device engaged
- Prelubricating pressure low. In case of black-out, starting is allowed within 5 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)
- Stop signal to engine activated (safety shut-down, emergency stop, normal stop)
- Stop lever on 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 to be cut off by:

- Speed switch in SPEMOS (115 RPM)
- A time relay function, which allows the start signal to be activated about 5 seconds. The time between consecutive starting attempts shall be about 30 seconds.
- Stop signal to engine 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 must be disabled during the starting sequence. This is most conveniently done using the “engine running” contact. Further, a time delay of about 10 seconds is to be 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 4V50G3619.

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

- The 5 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' is always closed when the stop signal is active.

For a single main engine installation it might be necessary to arrange a 5 sec delay on the autostop functions (except for overspeed) to give the possibility of overriding the autostop signal from the bridge and prevent the engine from stopping in a critical manoeuvring situation.

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 it might be necessary to arrange a 5 sec delay on the autostop functions (except for overspeed) to give the possibility of overriding the autostop signal 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 speed switch with the lower set point (nom. RPM + 15%) can be connected for momentary activation of the electro-pneumatic stop solenoid. The speed switch is activated and the stop solenoid is energized only as long as the speed is above the set point. When the speed has decreased, the stop solenoid is de-energized and the speed is again controlled by the governor.

The speed switch with the higher set point (nom. RPM + 18%) shall be connected with latching function in order to ensure shut-down of the engine.

14.3.5. Charge air shut-off valve

If gas detector senses combustible gas or vapour in the engine room the charge air shut-off valve must be automatically closed and engine shutdown activated. Also overspeed of the engine should automatically close this valve and activate shutdown. Since this is optional equipment most commonly used in offshore installations the construction varies with engine type and installation and the instructions in manuals must be followed.

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

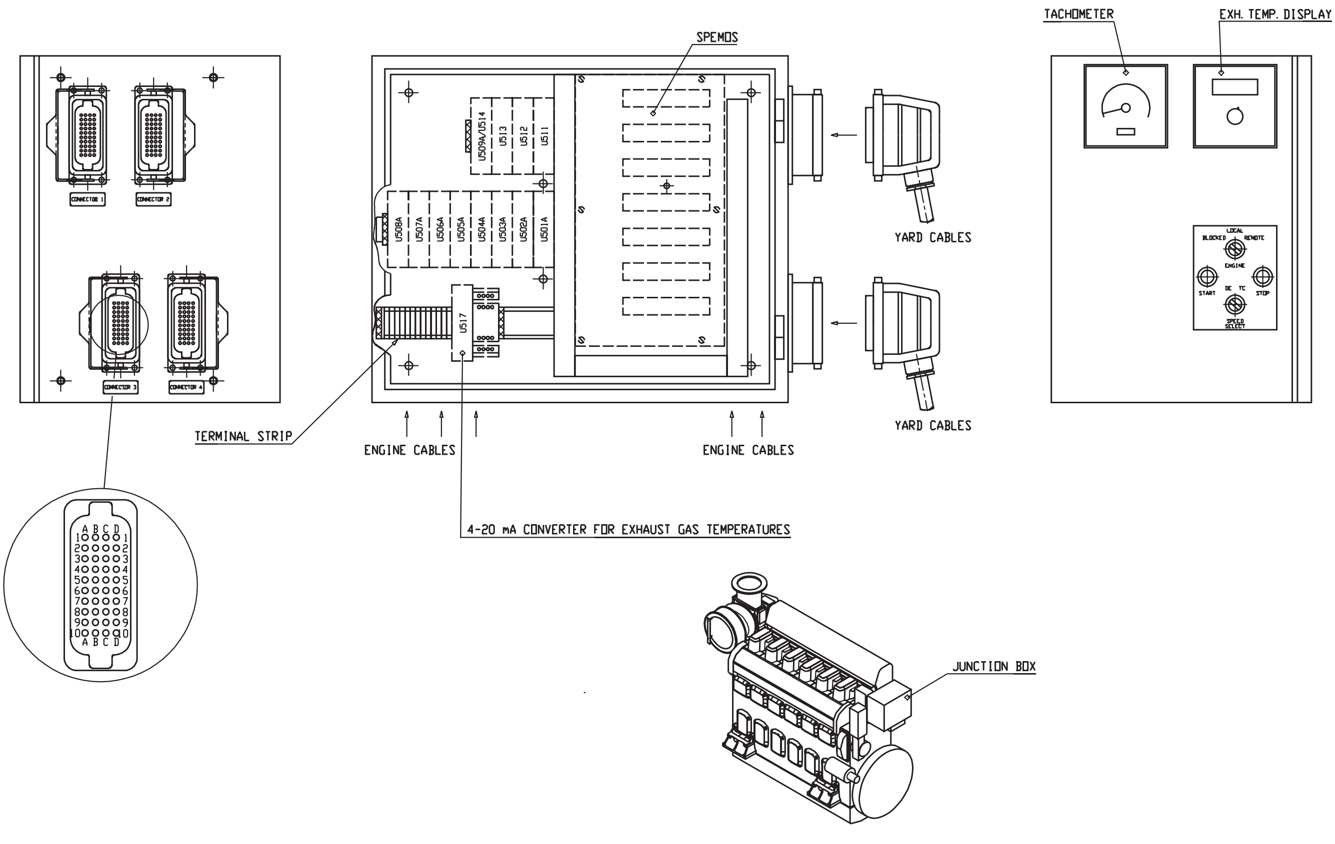
Function	Signal	Usage	Remark
Start fuel limiter	contact	internal	100 RPM below idling speed
Starting air cut off	contact	internal	115 RPM
Engine running	volt. free contact	external	300 RPM
Overspeed	volt. free contact	external	nominal + 15% speed
Engine speed	0 - 10 V DC	internal & external	0 - 1500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

14.5. Sensors & signals

Drawing 4V50L5692 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


SPEMOS
TACHOMETER
EXH. TEMP. DISPLAY
U509A/U514
U513
U512
U511
U508A
U507A
U506A
U505A
U504A
U503A
U502A
U501A
U517
YARD CABLES
YARD CABLES
TERMINAL STRIP
ENGINE CABLES
ENGINE CABLES
4-20 mA CONVERTER FOR EXHAUST GAS TEMPERATURES
JUNCTION BOX

14.6. Local instrumentation

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

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

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

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 prelubrication of a stopped engine, through which heavy fuel is circulating.

To ensure continuous prelubrication of a stopped engine, automatic starting and stopping of the prelubricating 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, preheating has to be arranged for the engine. Preheating is preferably done by an Electric preheater with a required heating power depending of the engine type. The preheater 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 60 - 70% of the nominal speed and for FP-propeller installations at about 35%.

The standard control air pressure for pneumatically controlled governors is:

p = 0.514 * n - 14.3

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 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 easily 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 easily 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 4-20mA speed reference, both from e.g. a PCS. 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

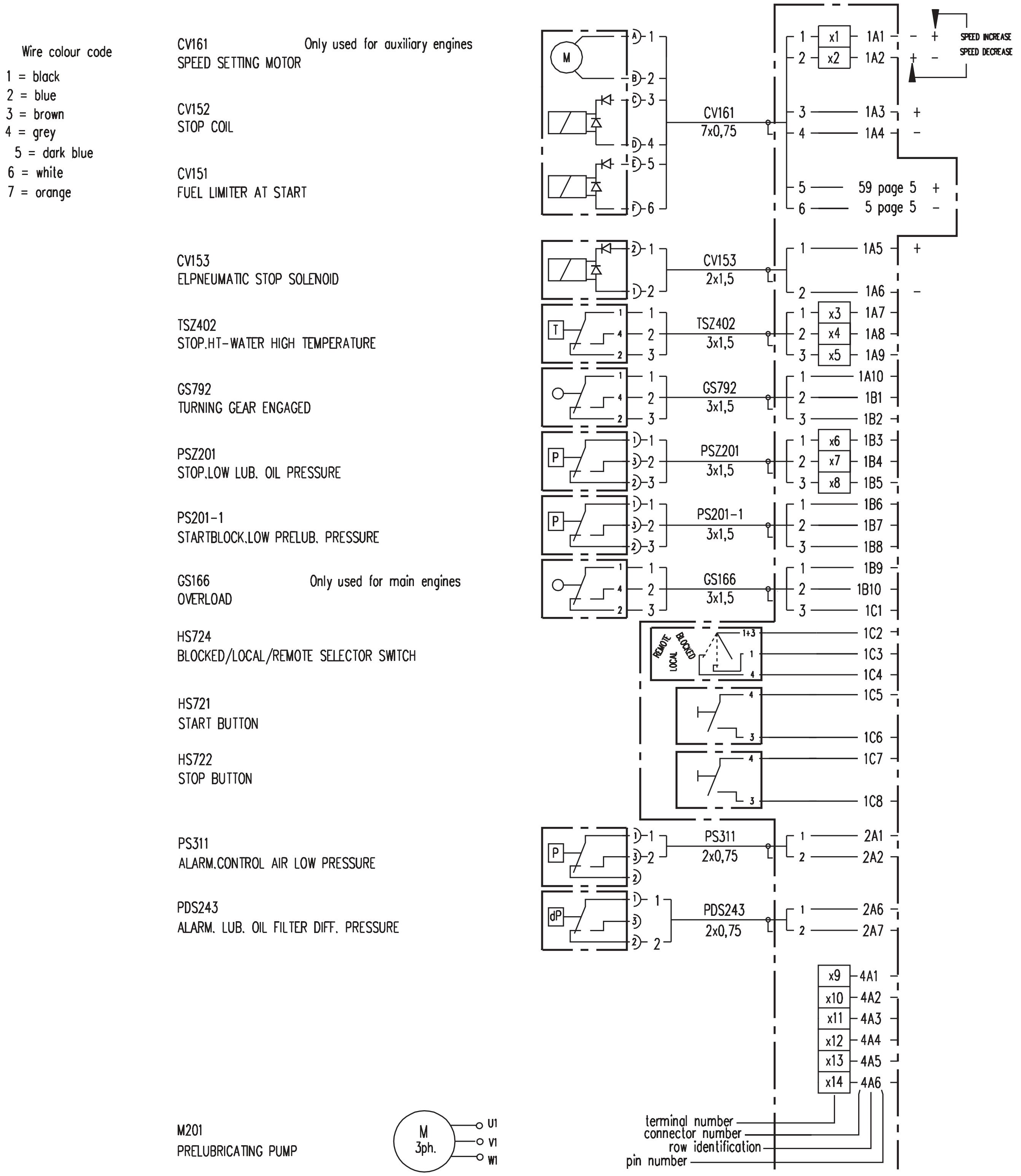
The MCU collects measuring data from the sensors on the engine, converts the information into digital form and communicates by serial link with the external monitoring, alarm and control systems. The MCU also handles the functions described in the previous paragraphs, i.e. speed measuring, start/stop 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

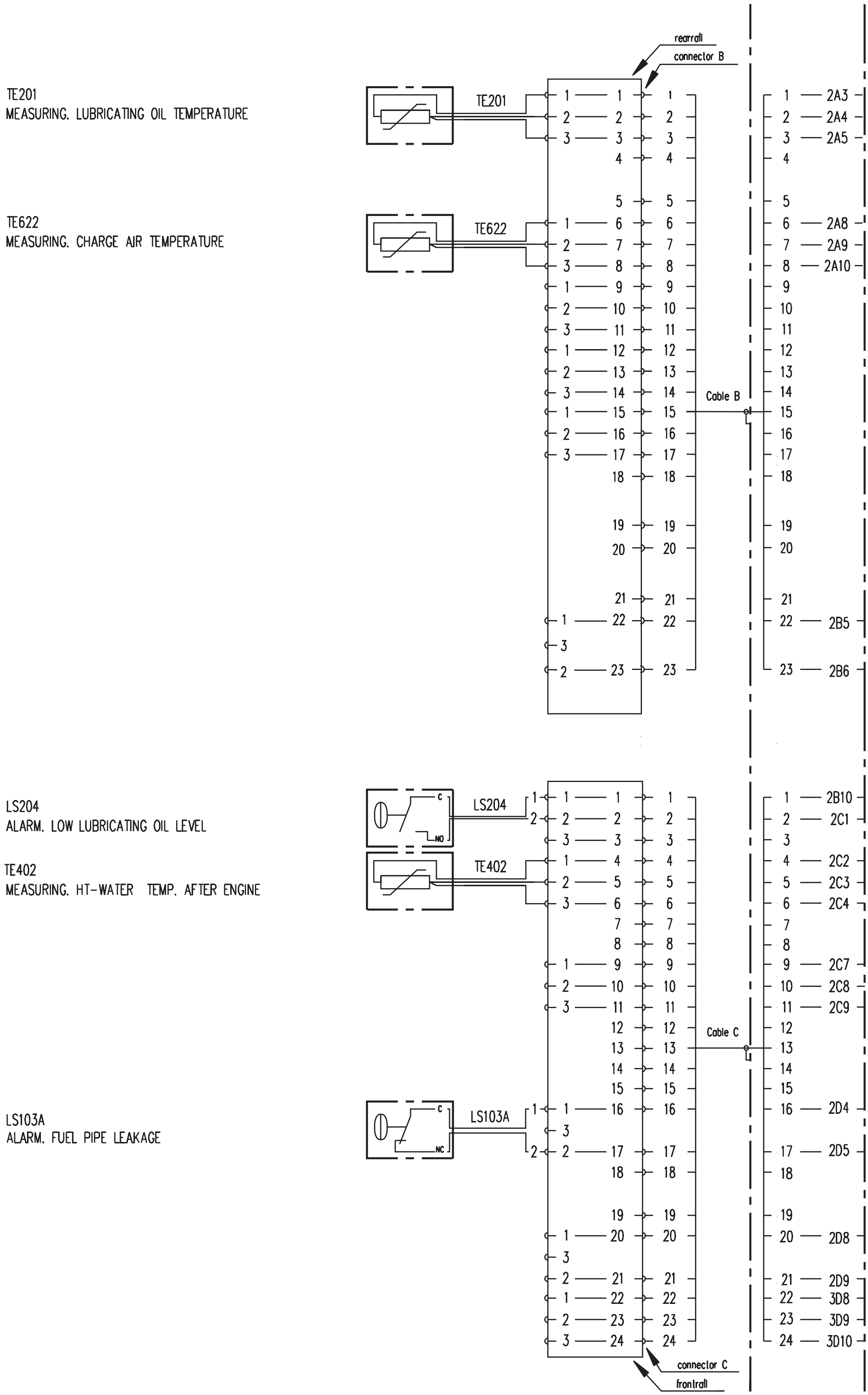
By having field bus connection via multi-standard RS-ports for communication with external systems, cabling work furnished by the yard will be minimized. In addition, installation work, service and maintenance will be easier.

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.

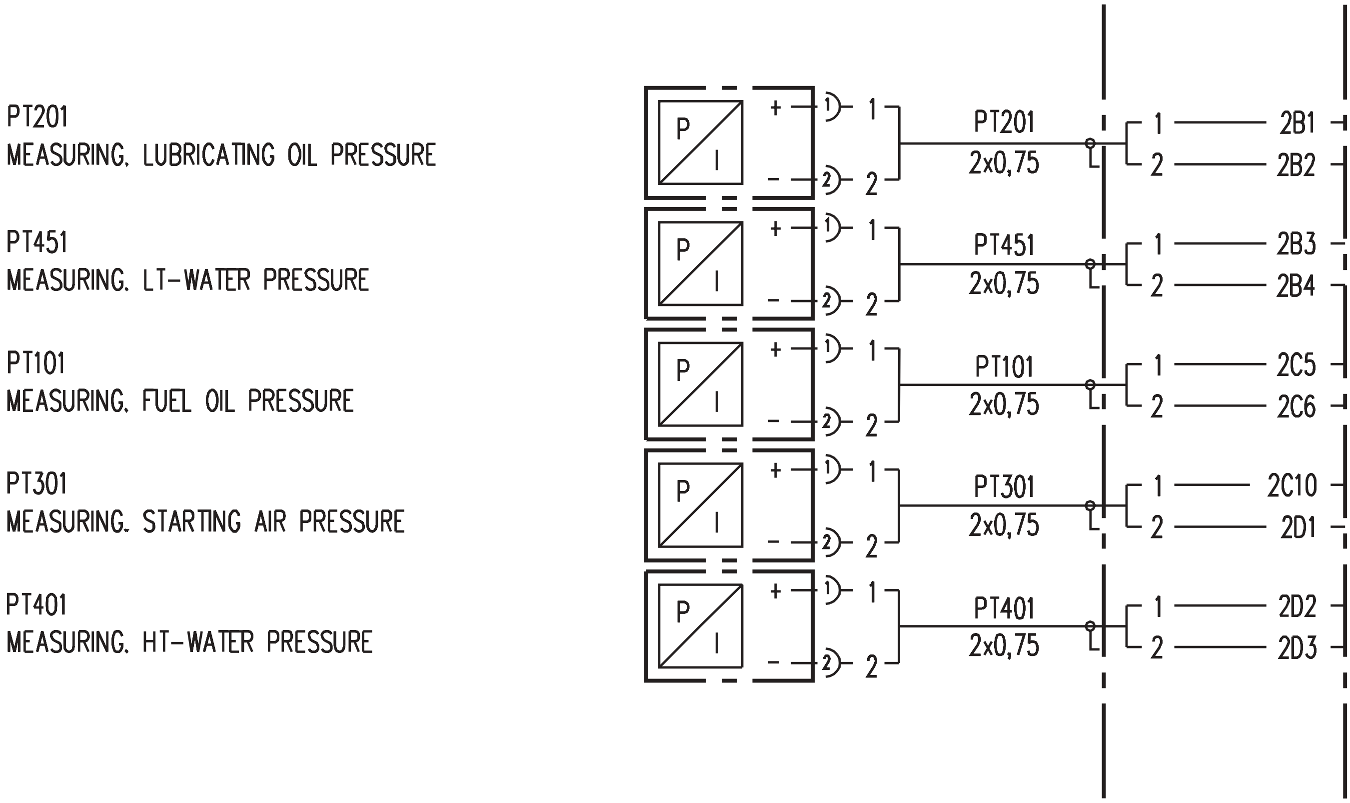
Typical wiring diagram (4V50L5692-1a)



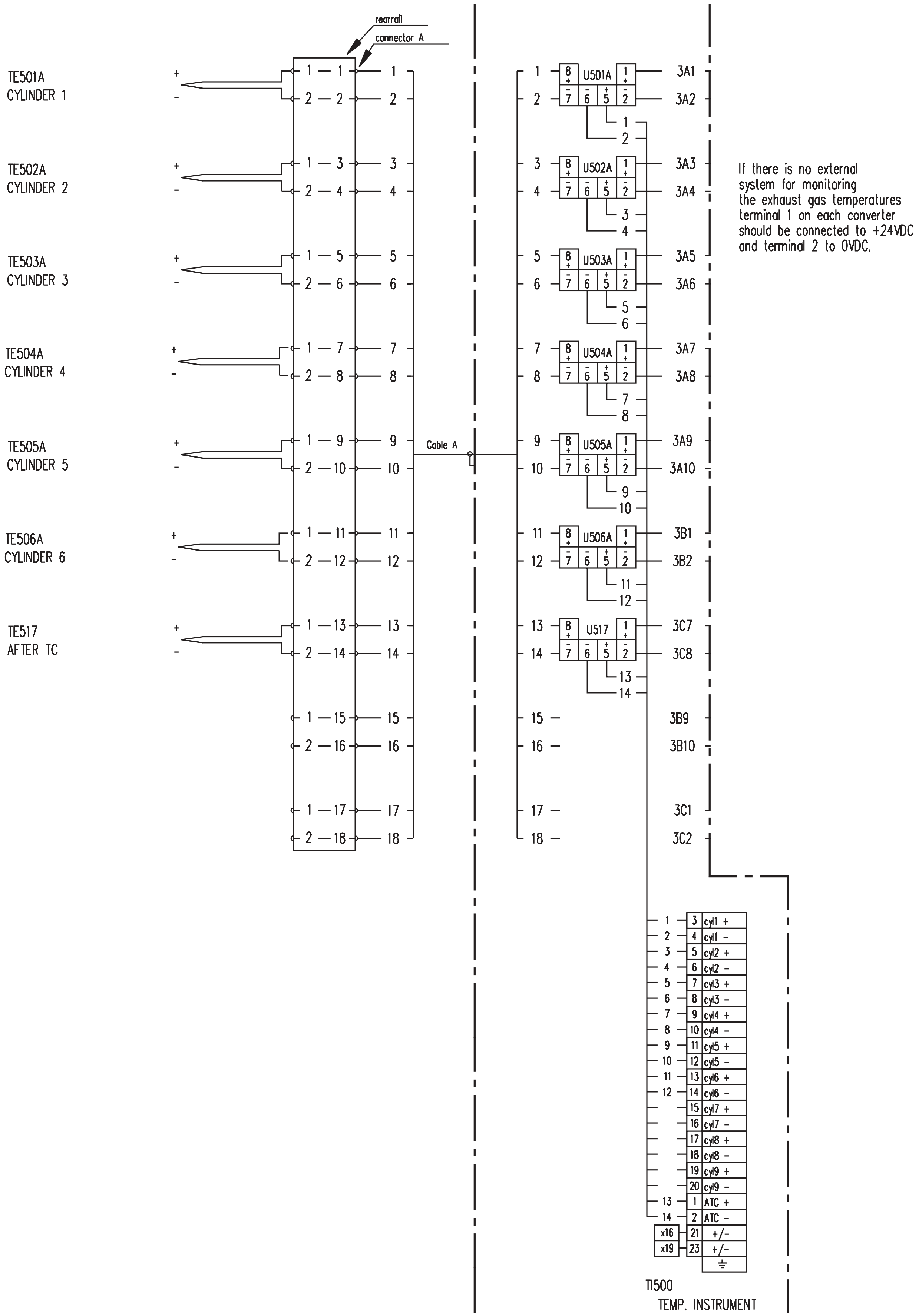
Typical wiring diagram (4V50L5692-2a)



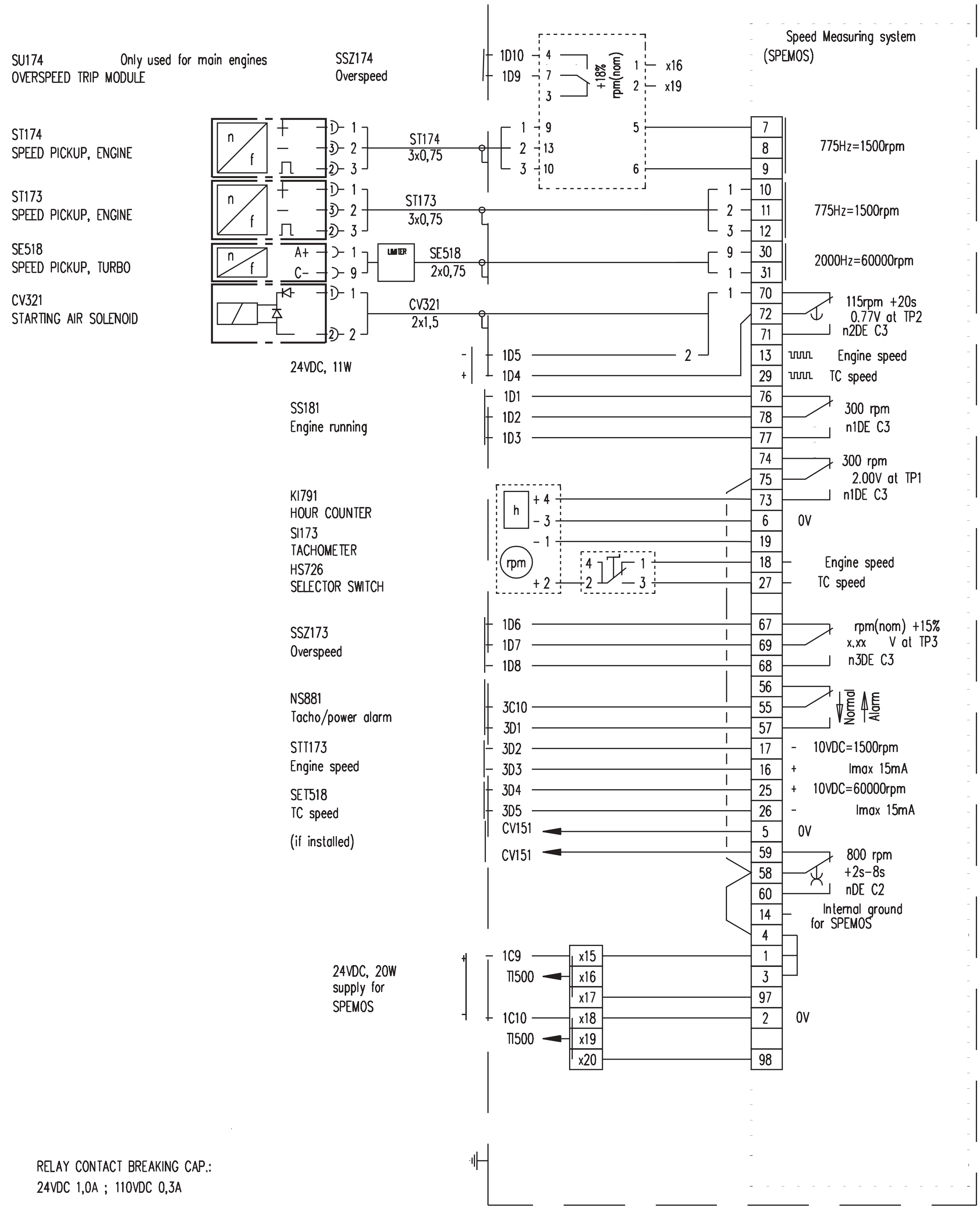
Typical wiring diagram (4V50L5692-3a)



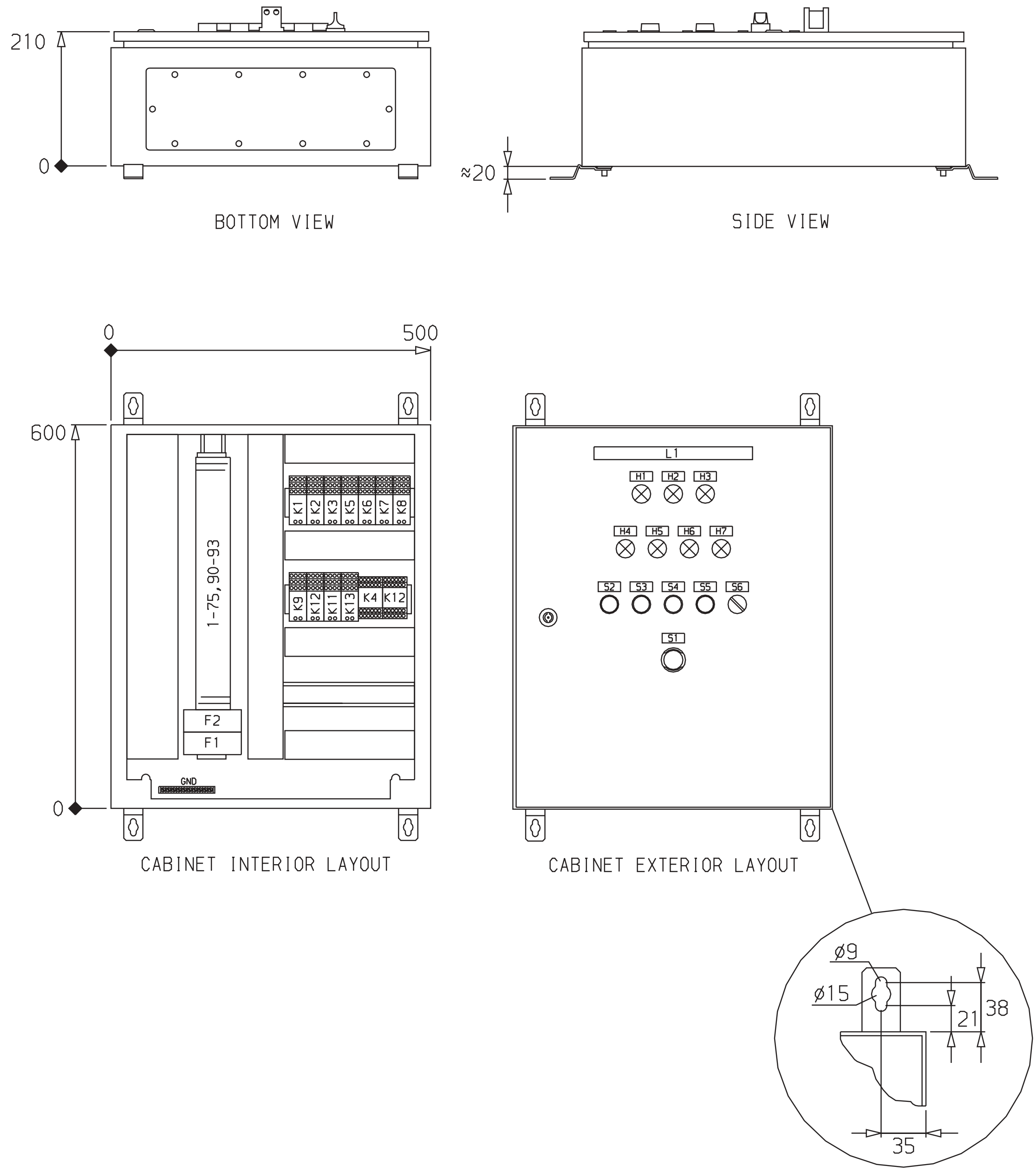
Typical wiring diagram (4V50L5692-4a)



Typical wiring diagram (4V50L5692-5a)



Start stop system for an auxiliary engine (4V50G3472-1d)



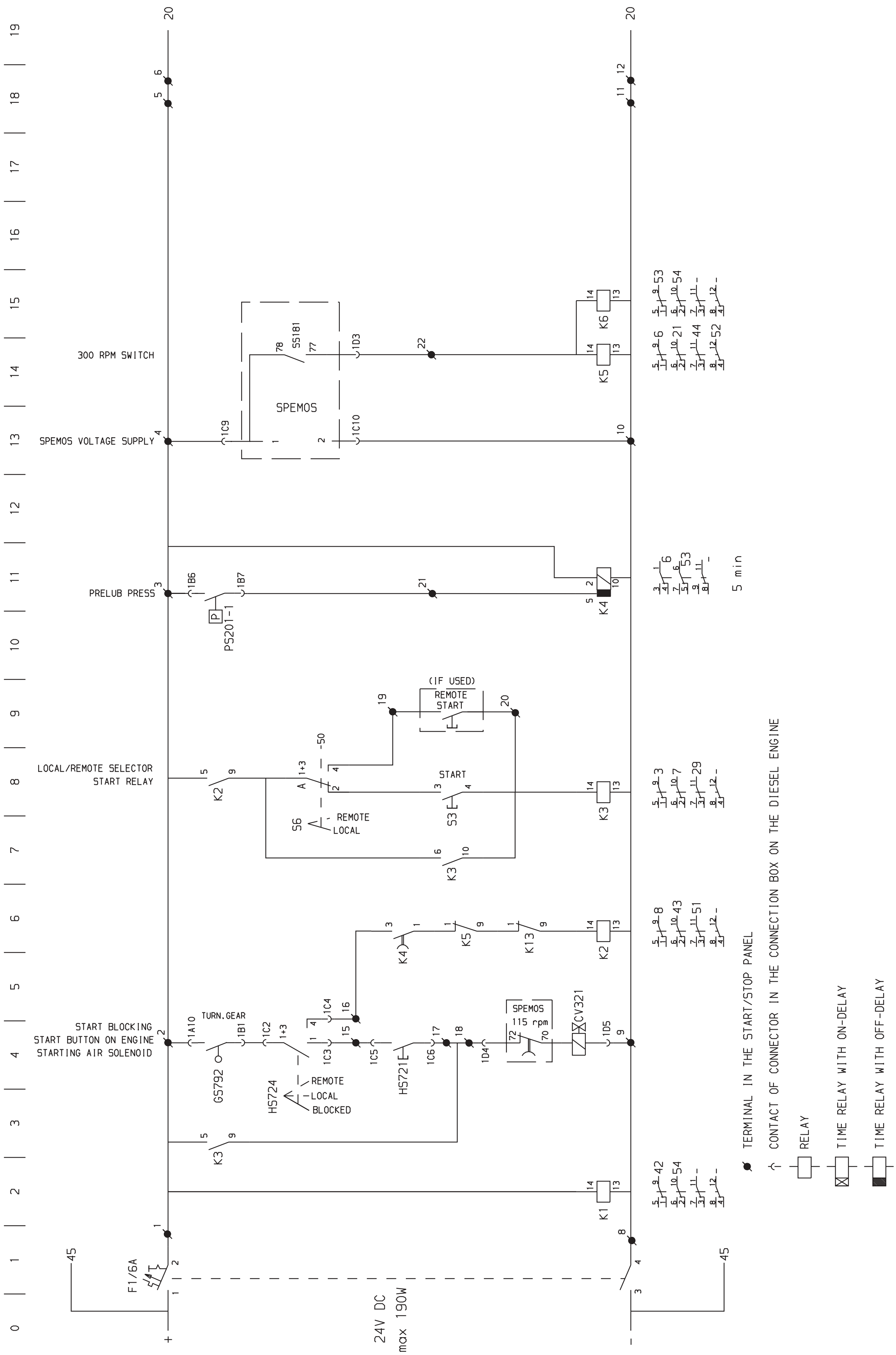
Indication lamps

H1	Power on
H2	Ready for start
H3	Engine running
H4	Cooling water temperature high
H5	Lubrication oil pressure low
H6	Overspeed
H7	Start failure

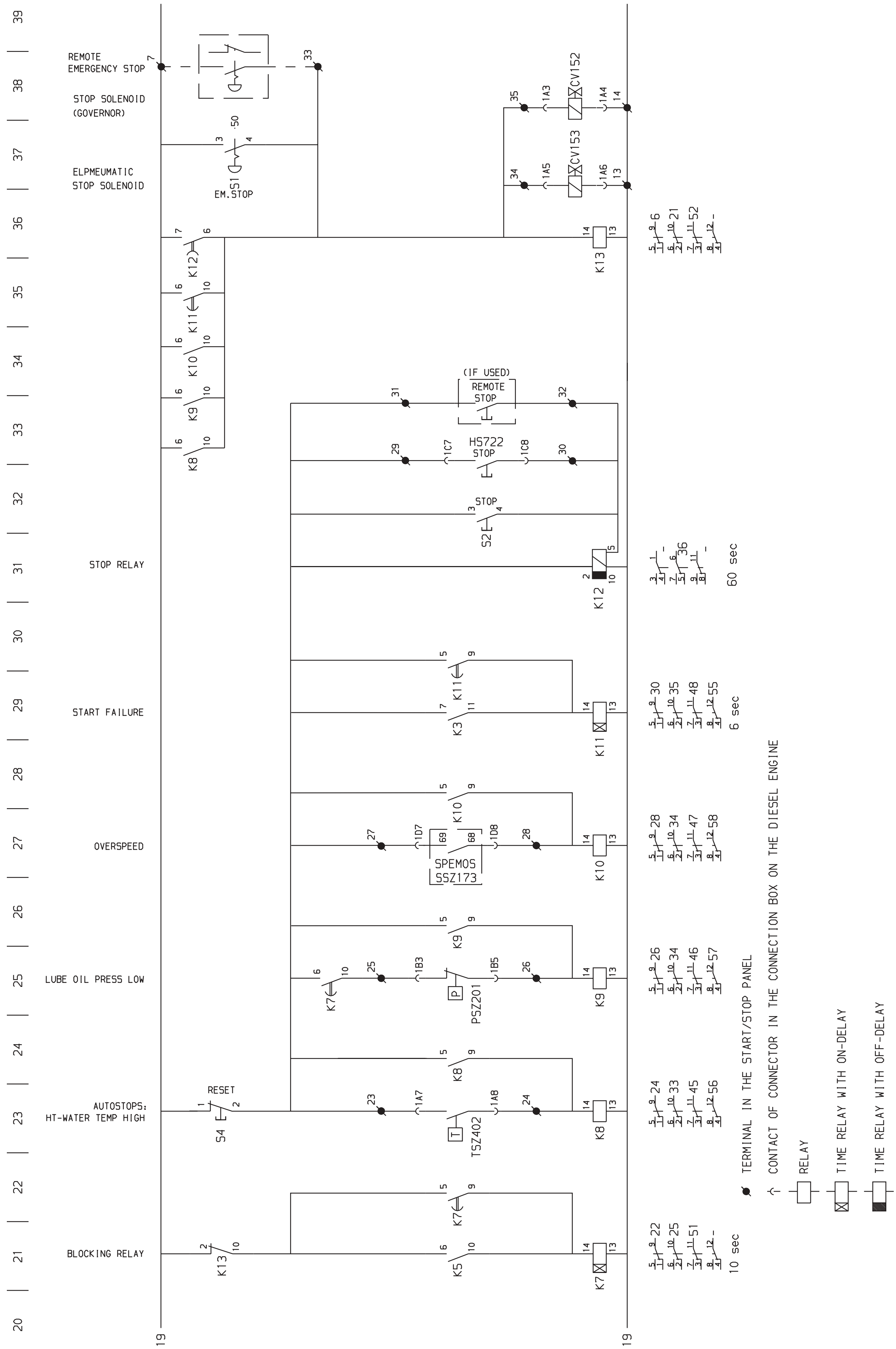
Pushbuttons and selectors

S1	Emergency stop
S2	Stop
S3	Start
S4	Reset
S5	Lamp test
S6	Local / Remote mode of start

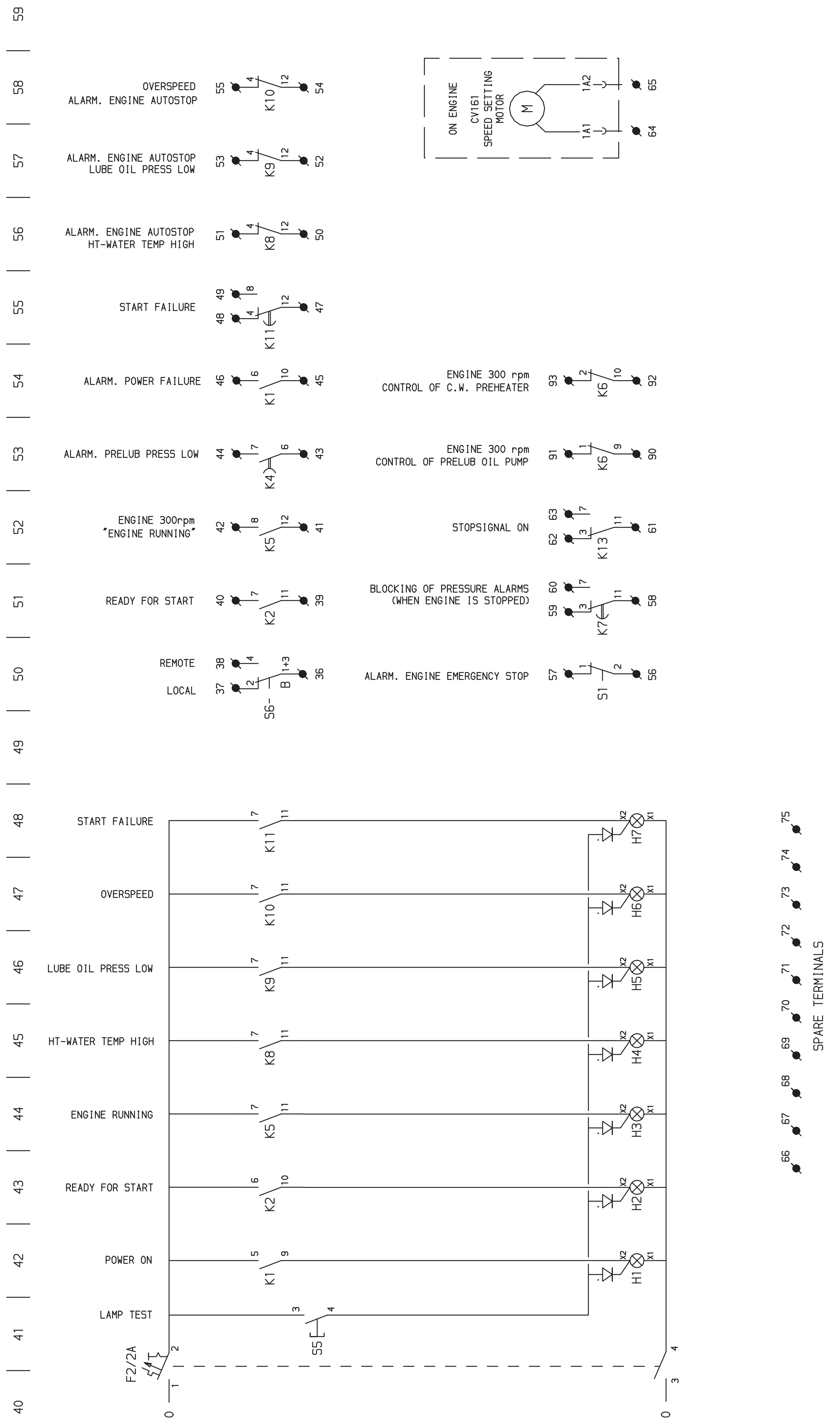
Start stop system for an auxiliary engine (4V50G3472-2d)



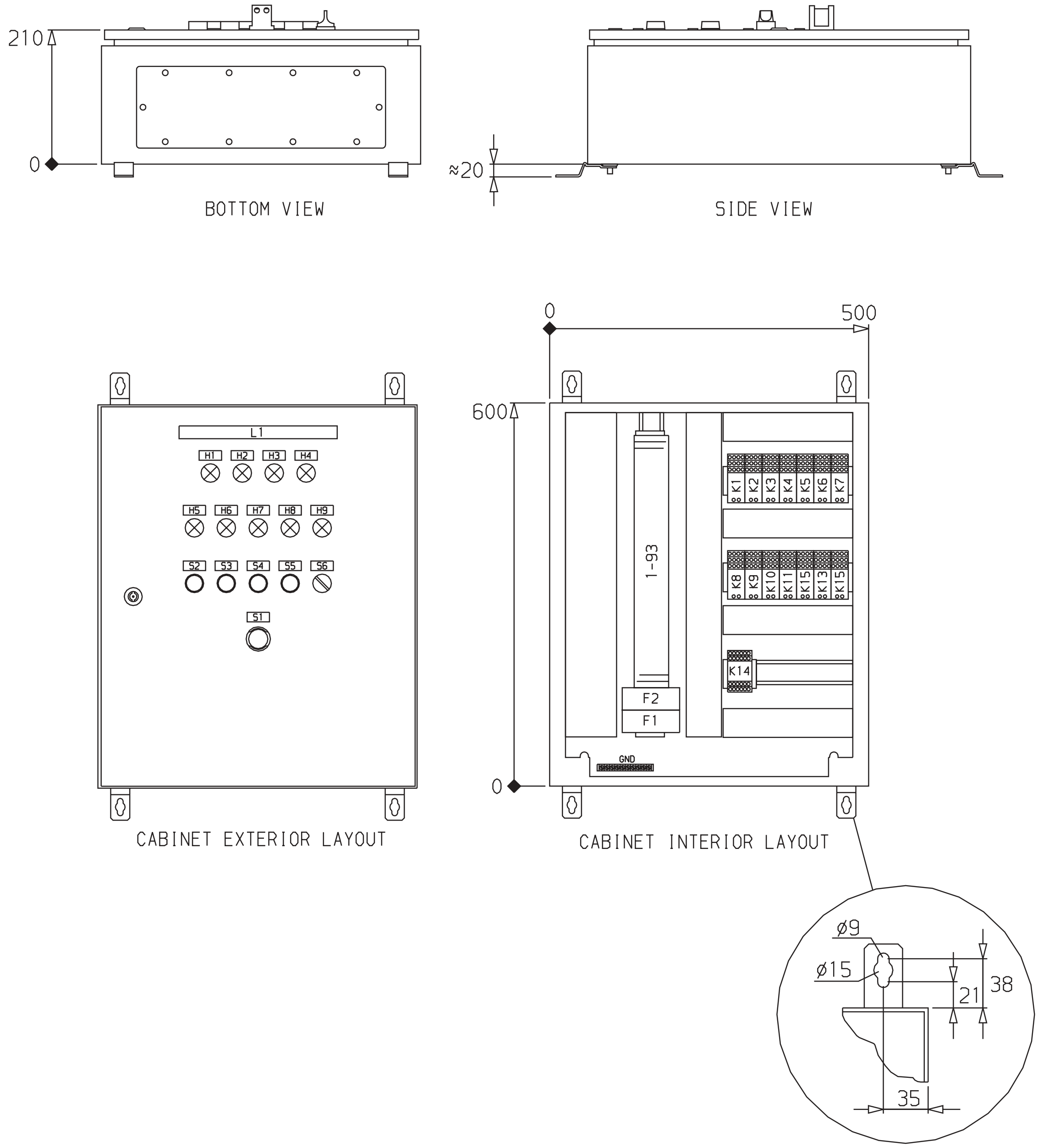
Start stop system for an auxiliary engine (4V50G3472-3d)



Start stop system for an auxiliary engine (4V50G3472-4d)



Start/stop system for a main engine (4V50G3619-1a)



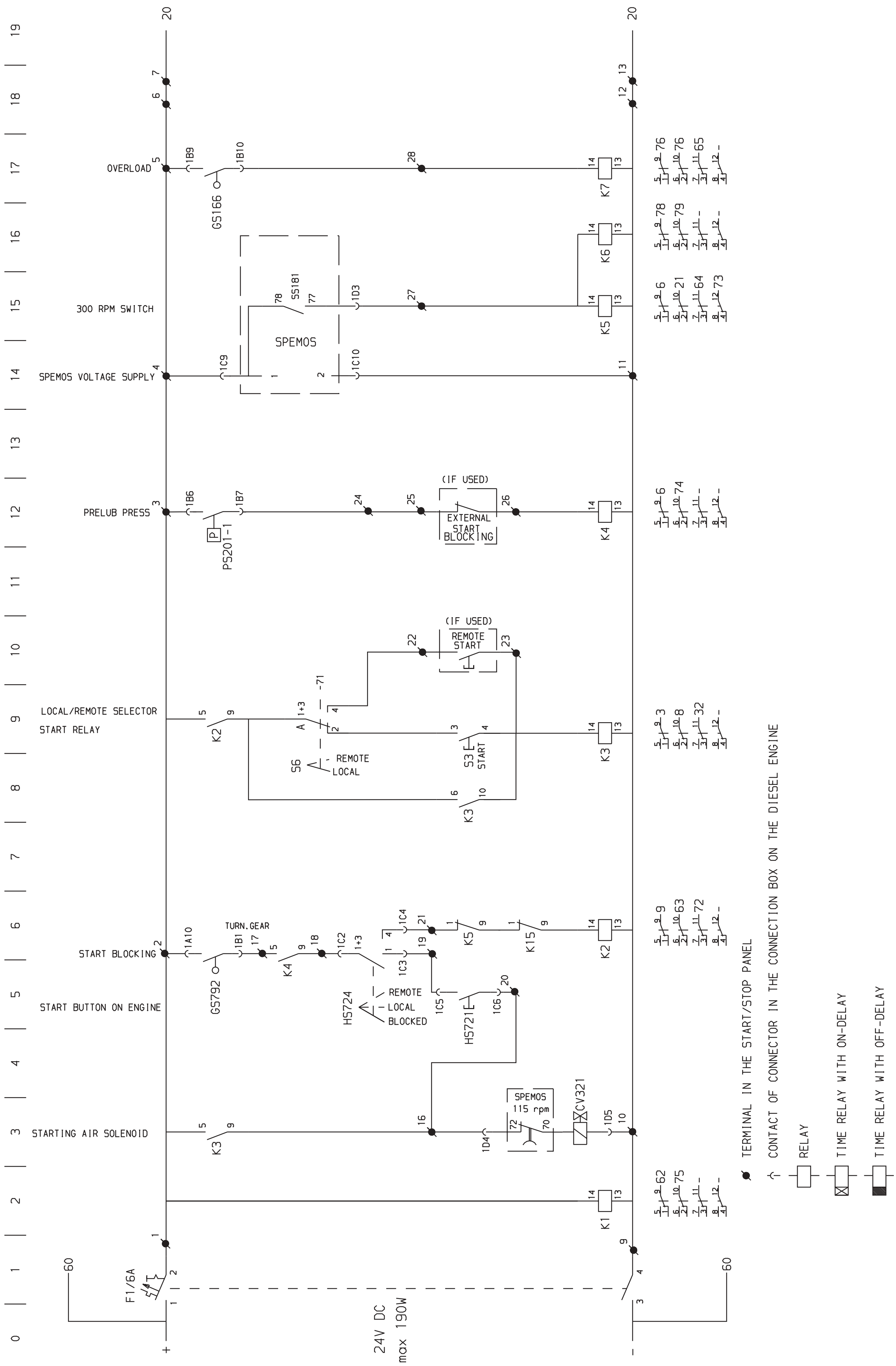
Indication lamps

H1	Power on
H2	Ready for start
H3	Engine running
H4	Overload
H5	Cooling water temperature high
H6	Lubrication oil pressure low
H7	Overspeed
H8	Reduction gear lubrication pressure low
H9	Start failure

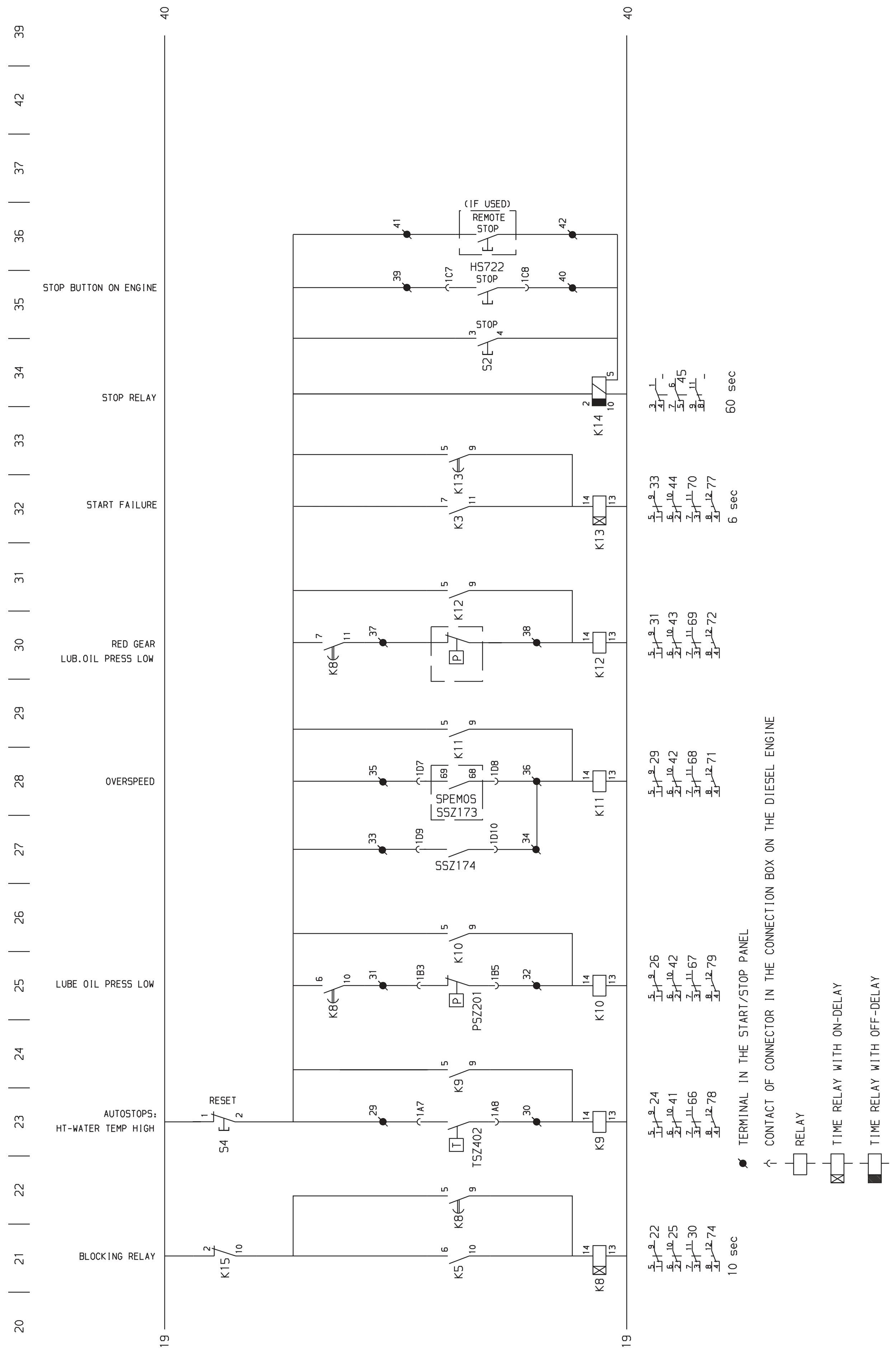
Pushbuttons and selectors

S1	Emergency stop
S2	Stop
S3	Start
S4	Reset
S5	Lamp test
S6	Local / Remote mode of start

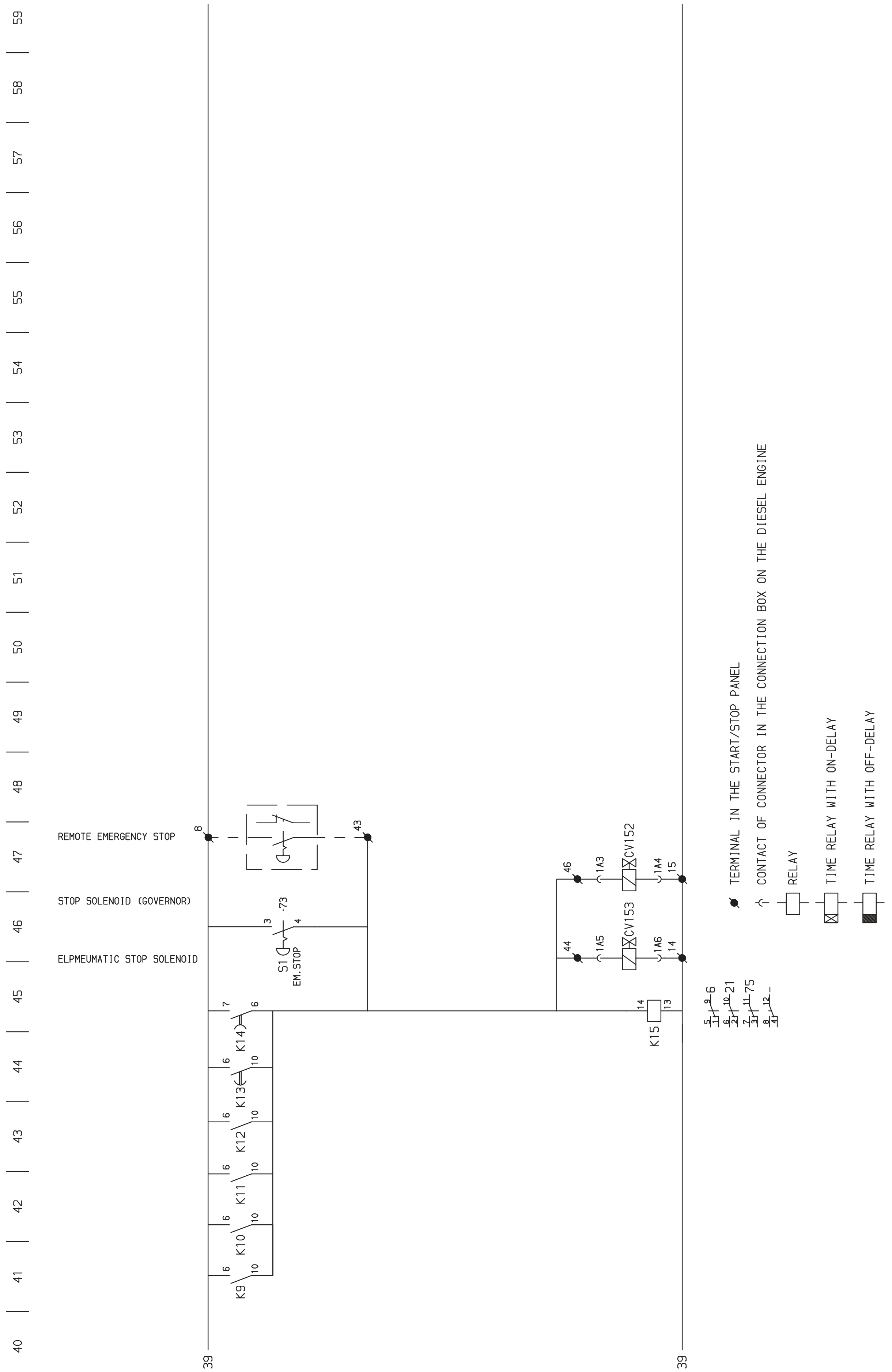
Start/stop system for a main engine (4V50G3619-2a)



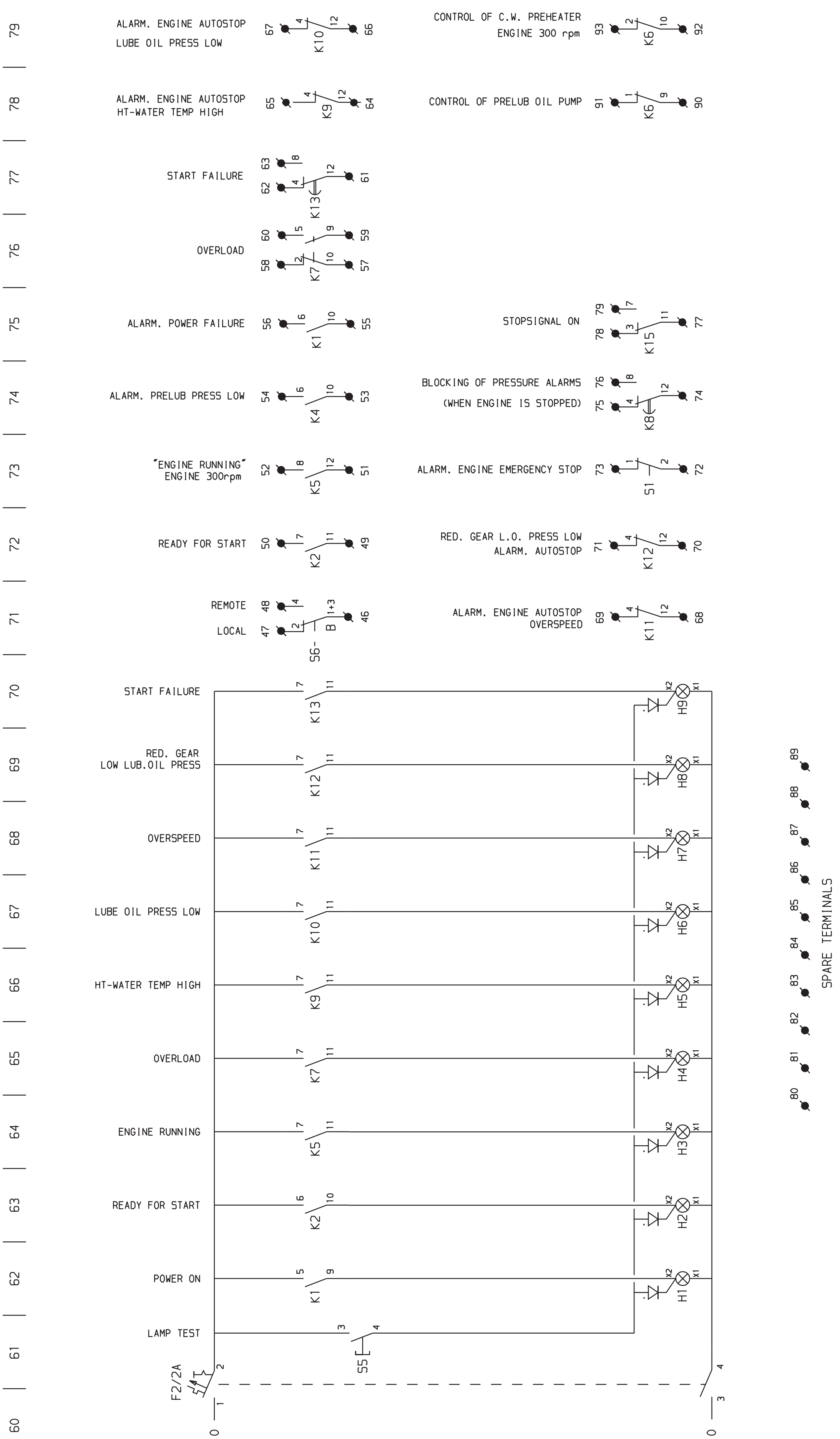
Start/stop system for a main engine (4V50G3619-3a)



Start/stop system for a main engine (4V50G3619-4a)



Start/stop system for a main engine (4V50G3619-5a)



15. Electrical power generation and management

15.1. General

The electrical concept design, either performed by the Ship Owner, Consultant, Yard or Wärtsilä as 'The Ship Power Supplier', is the basis for a co-ordination and optimisation of the electric power generation and management being supplemented by these general guidelines.

15.1.1. Definitions

The marine vessel's electric supply system is basically an alternating current (a.c.) three-phase, three–wire insulated. 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.

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 = 60 * f/p

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

Generator power definitions:

Sr = rated output, rated apparent power in kilo-volt-amperes kVA

Sr = Pr/cos φr

Pr = rated active power in kilowatts kW

cos φr = rated power factor

cos φr = Pr/Sr

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

$Pr = \eta_{GEN} * P_{DIESEL}$

Generator η_{GEN} is typically 95…97 % at full load and cos φ 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 among others (illustrated by examples):

- 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 Pr)

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:

Table 15.1.

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. Reference is also made to Chapter 2: Operating Ranges and loading capacity.

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 maximum connected electric load, which can be expected used simultaneously. The demanding operation mode is usually manoeuvring or cargo handling, while max speed at sea in a diesel-electric ship may be the actual mode.

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 φr of the generator shall be selected in accordance with the network load cos φ, which regularly 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 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 xd". The xd" is typically 15…20 (up to 25) %.

Generally a high xd" 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 xd" increases the generator size in comparison to a high xd", but the possibility to choose a specific xd" 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.

The temperature rise of the generator windings is recommended to be one class lower than the temperature class of the insulation, e.g.:

Insulation class/Temperature rise: F/B or H/F

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 xd'', xd' 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 parallelling generators in order to have a proportional load sharing.

An isochronous load sharing for parallelling generators is possible with an electronic governor. All parallelling 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. AVR (Automatic Voltage Regulator)

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 basically 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 * IN) 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 parallelling generators is provided by the AVR using parallelling 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 // propeller shaft)
- a primary PTO from a step-up gear (generator runs // 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 basically 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 ordinary 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 Rules, 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 the electric power management (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 basically controls the starting/stopping and synchronising 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 bustie 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 main system. It is recommended that means is to be provided to start an engine locally and to synchronise manually at the main switchboard in case of the PMS failure. The back-up system is recommended to be an independent operating system, hard wired and with galvanic isolation to 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 for 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
- bustie 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, and the following units to be automatically synchronised.

Precautions against failing blackout start are among others:

- 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 among others:

- 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. Parallelling of generators, load sharing

The PMS provides automatic synchronising of auxiliary diesel generators i.e. 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 parallelling generators in order to have a proportional load sharing.

But 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. parallelling 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
- synchronise the SG-section and the MSB (i.e. the auxiliary diesel engine(s) are usually synchronised to the main engine) and close the SG-section bustie breaker
- transfer load to SG by unloading the auxiliary diesel generator(s) according to unloading rate
- open the auxiliary diesel generator’s breaker(s) when unloading trip level is reached
- stop the auxiliary diesel engine(s)

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 parallelling 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 nominal power 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 synchronise 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 e.g.:

- galley consumers
- AC-compressors
- accommodation ventilation
- reduction of propulsion power

15.3.10. Special applications, e.g.: Auxiliary Propulsion Drive (APD)

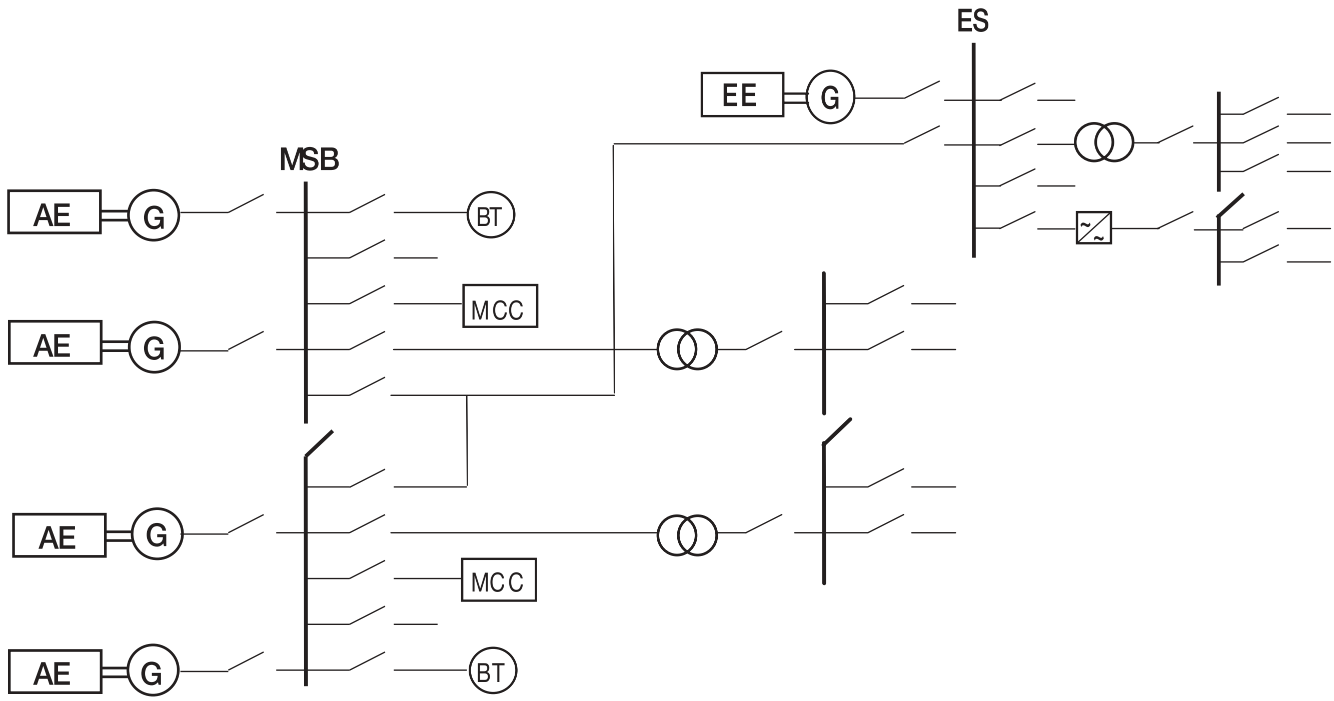
A special application providing limited redundancy with respect to increased availability of the vessel's propulsion system is the so-called Auxiliary Propulsion Drive (APD). The principle idea of this solution is that the ship can be propelled by the auxiliary generating sets, by using the shaft generator as an electric motor, in case the main engine (ME) is not available.

The benefit of the combined shaft generator and APD is, among others, an increase of safety when it is used as back-up propulsion in e. g. following operating modes:

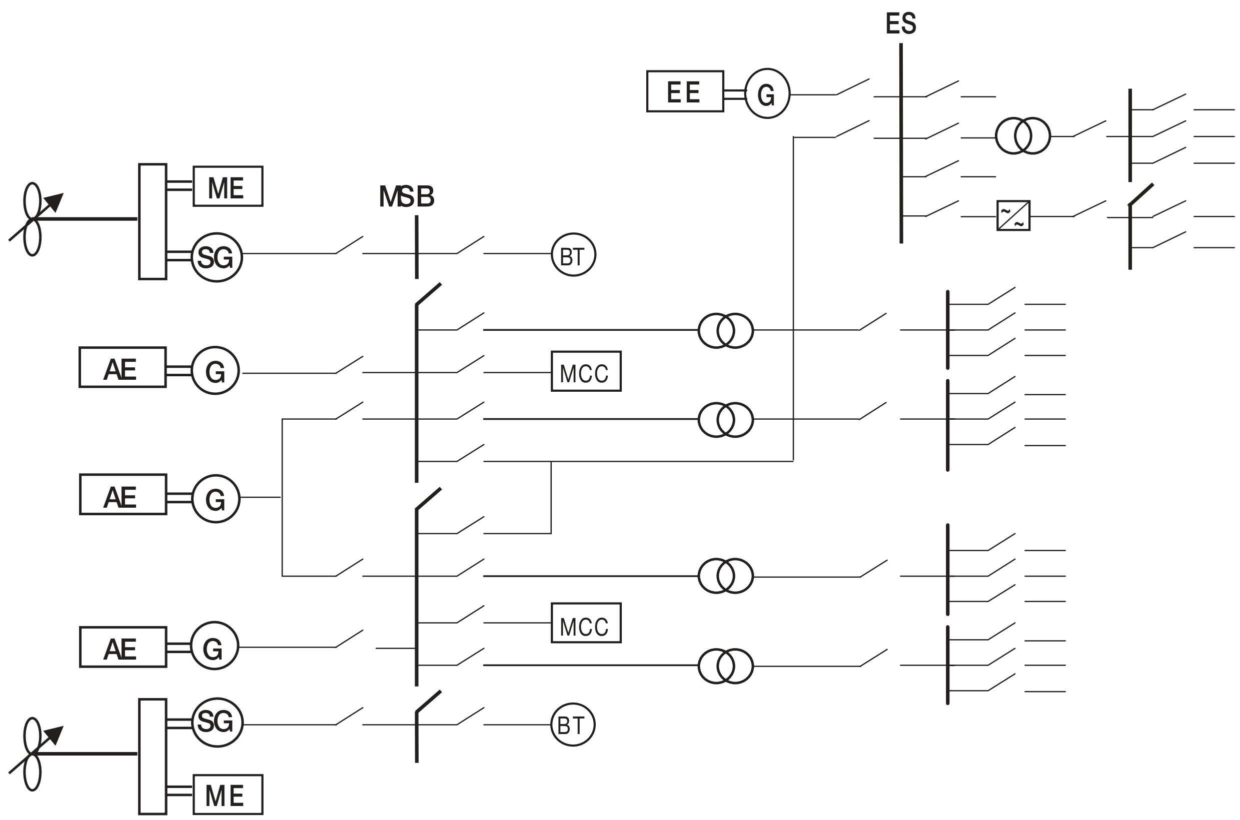
- booster mode, both ME and PTO are driving the propeller
- standby mode, ME disconnected for maintenance and APD is connected if manoeuvring is required
- emergency mode (take me home), APD is used to propel the ship if ME fails

15.4. Typical one line main diagrams

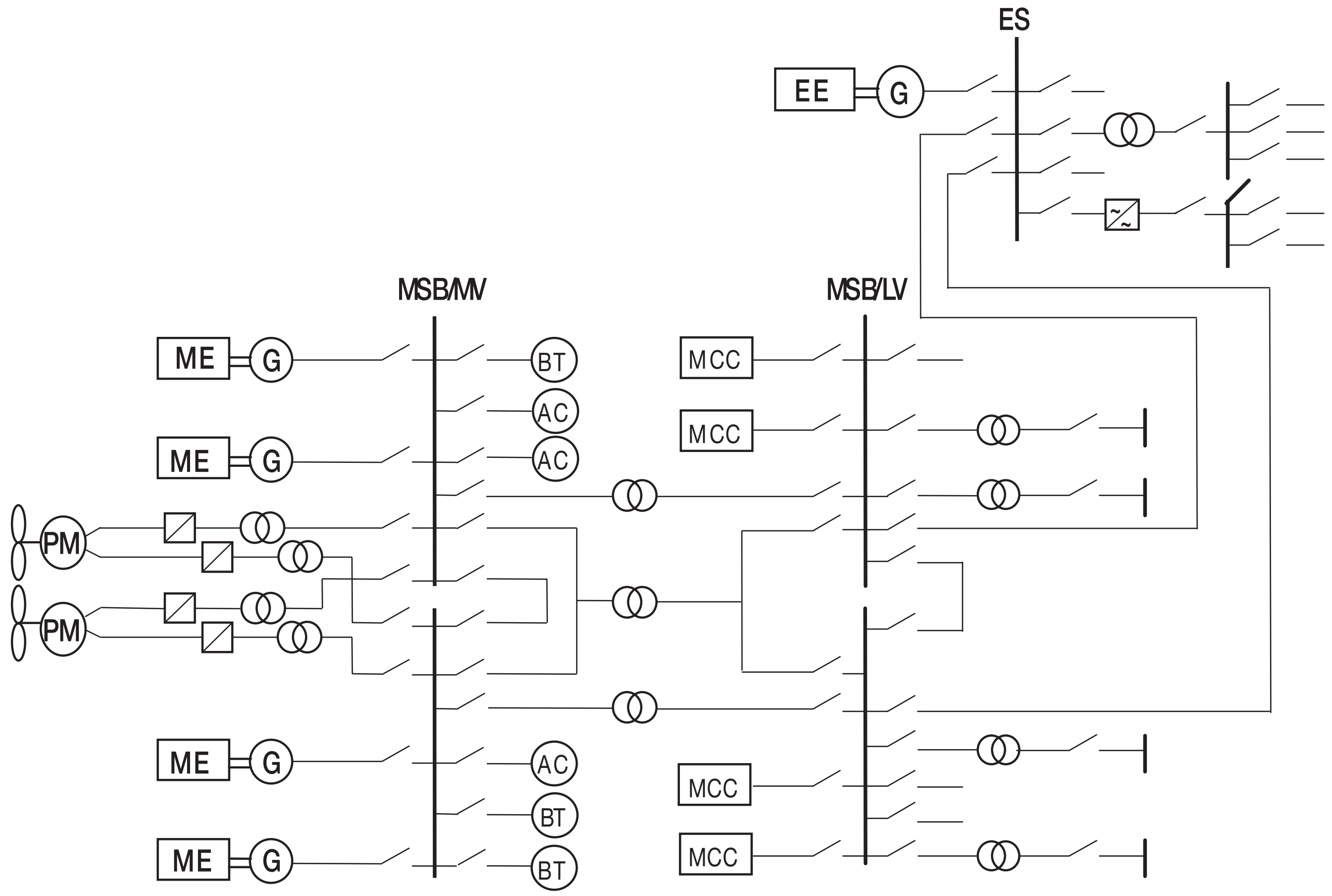
4 ADG low voltage network



3 ADG + 2 SG low voltage network



Diesel electric ship, medium voltage network



16. Foundation

16.1. General

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.

Dynamic forces caused by the engine are shown in the Chapter for Vibration and noise.

16.2. Steel structure design

The system oil tank may 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 also be symmetrically located in transverse direction under the engine.

16.3. Mounting of main engines

Main engines can be either rigidly mounted on chocks, or resiliently mounted on rubber elements.

The foundation and the double bottom 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.

16.3.1. Rigid mounting

Main engines are normally rigidly mounted on the seating, either on steel or resin chocks.

The engine has 4 mounting brackets cast to the engine block. Each bracket has a threaded hole for an M16 jacking screw and two Ø22 holes for M20 holding down bolts.

The bolt closest to the flywheel at either side of the engine shall be made as a Ø23H7/m6 fitted bolt. All other bolts are clearance bolts.

The clearance bolts shall be through bolts with lock nuts at both the lower and upper ends. Ø22 holes can be drilled into the seating through the holes in the mounting brackets.

In order to avoid bending stress in the bolts and ensure that the bolts remain tight the contact face of the nut under the seating top plate shall be spotfaced.

The elongation of holding down bolts can be calculated from the formula:

$$\Delta L = 6.18 \cdot 10^{-6} F \sum_{1}^{n} \frac{L_i}{D_i^2}$$

ΔL = bolt elongation [mm]

F = tensile force 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]

Lateral supports as shown in 2V69A0236 shall be fitted against the engine block. The wedge type supports shall be lightly knocked into position when the engine is hot and secured with a tack weld. Minimum bearing surface on the wedges is 80%.

The engine can be installed on either steel or resin chocks. The chocking arrangement shall be sent to the classification society and Wärtsilä for approval.

Steel chocks

The top plates of the engine girders 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 wedge-type chocks also have an inclination of 1/100 to match the inclination of the seating. If the rider plate of the engine girder is fully horizontal, a chock is welded to each point of support. The chocks should be welded around the periphery as well as through holes drilled for this purpose at regular intervals to avoid possible relative movement in the surface layer. The welded chocks are then face-milled to an inclination of 1/100. The surfaces of the welded chocks should be large enough to fully cover the wedge-type chocks.

The supporting surface of the seating top plate should be machined so that a bearing surface of at least 75% is obtained.

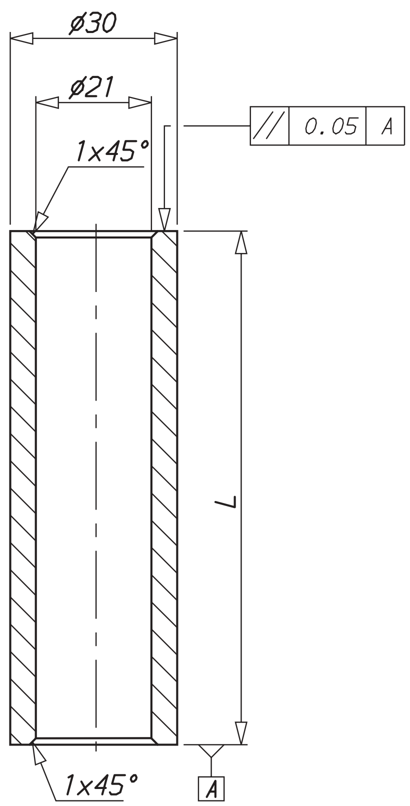
The cutout in the chocks for the clearance bolts should be about 2 mm larger than the bolt diameter. The maximum cut out area is 20%. Holes are to be drilled and reamed to the correct tolerance for the fitted bolts after that the coupling alignment has been checked and the chocks have been lightly knocked into position.

In order to assure proper fastening and to avoid bending stress in the bolts, the contact face of the nut underneath the seating top plate should be counterbored.

Holding down bolts shall be long enough to ensure an elongation $\Delta L \geq 0.25$ mm when tightened.

An effective bolt length of 160 mm (between the nuts) will ensure a sufficient elongation. It is recommended to fit distance sleeves with L ≥ 95 according to drawing 4V33F0214 under the seating top plate. M20 8.8 bolts can be used. Tightening torque 390 - 430 Nm.

Distance sleeve (4V33F0214)



Resin chocks

Installation of main engines 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 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 type, which has a type approval from the relevant classification society for a total surface pressure of 5 N/mm² (typical conservative value is $p_{tot} <$ 3.5 N/mm²).

In order to assure proper fastening and to avoid bending stress in the bolts, the contact face of the nut underneath the seating top plate should be counterbored.

If the engine is installed on resin chocks, the seating shall be as shown in 2V69A0236, except that the 1:100 inclination is not necessary.

When installing an engine on resin chocks the following issues are important:

- Sufficient elongation of the holding down bolts
- Maximum allowed surface pressure on the resin $p_{tot} = p_{static} + p_{bolt}$
- Correct tightening torque of the holding down bolts

The elongation ΔL of the holding down bolts should be:

ΔL [mm] ≥ 0.12 for a surface presure on the resin $p_{tot} \leq$ 3.5 MPa

ΔL [mm] ≥ 0.0343 x p_{tot} [MPa] for $p_{tot} >$ 3.5 MPa

The recommended dimensions of resin chocks are 140 x 410 mm. This gives gives a deadweight loading p_{static} on the resin which is presented in the table below.

Table 16.1. Total load on resin chocks

Engine	Dwt load P_{static} [Mpa]	Bolt tension load P_{bolt} [Mpa]	Total load P_{total} [Mpa]
4L20	0.33	2.9	3.23
5L20	0.37	2.9	3.27
6L20	0.40	2.9	3.30
8L20	0.50	2.9	3.40
9L20	0.58	2.9	3.48

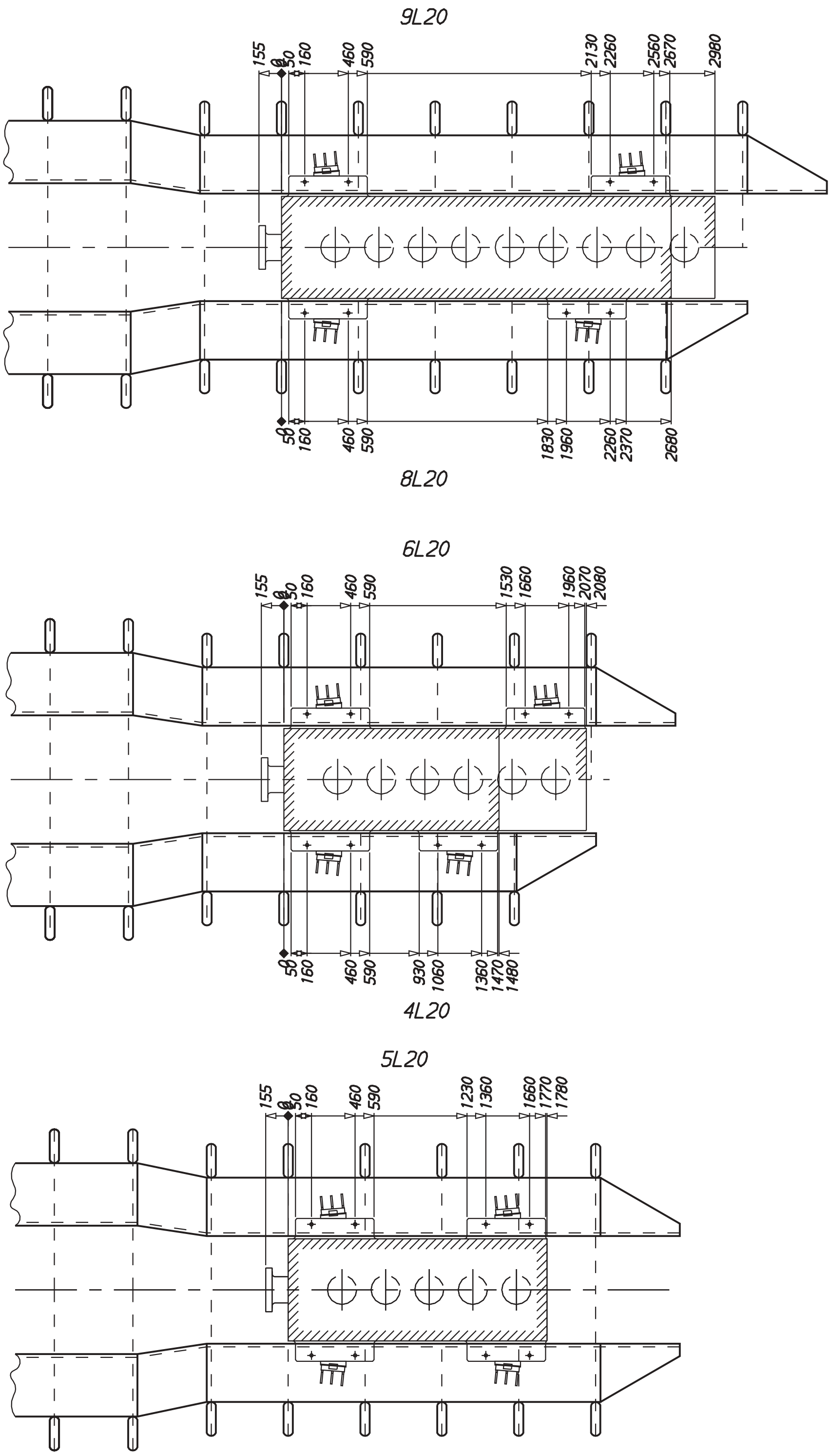
Wet engine with wet sump with standard equipment and flywheel.

Most resin types can take at least 3.5 MPa and the bolt holding down force (p_{bolt}) can be chosen to produce 3 MPa on the resin. This corresponds to a bolt tension of 83 000 N (with recommended chock dimensions) and a tightening torque of about 305 Nm tightening the bolts to 53% of yield, assuming M20 8.8 bolts.

To ensure sufficient elongation a distance sleeve according to drawing 4V33F0214 with L ≥ 45 mm shall be fitted under the seating top plate

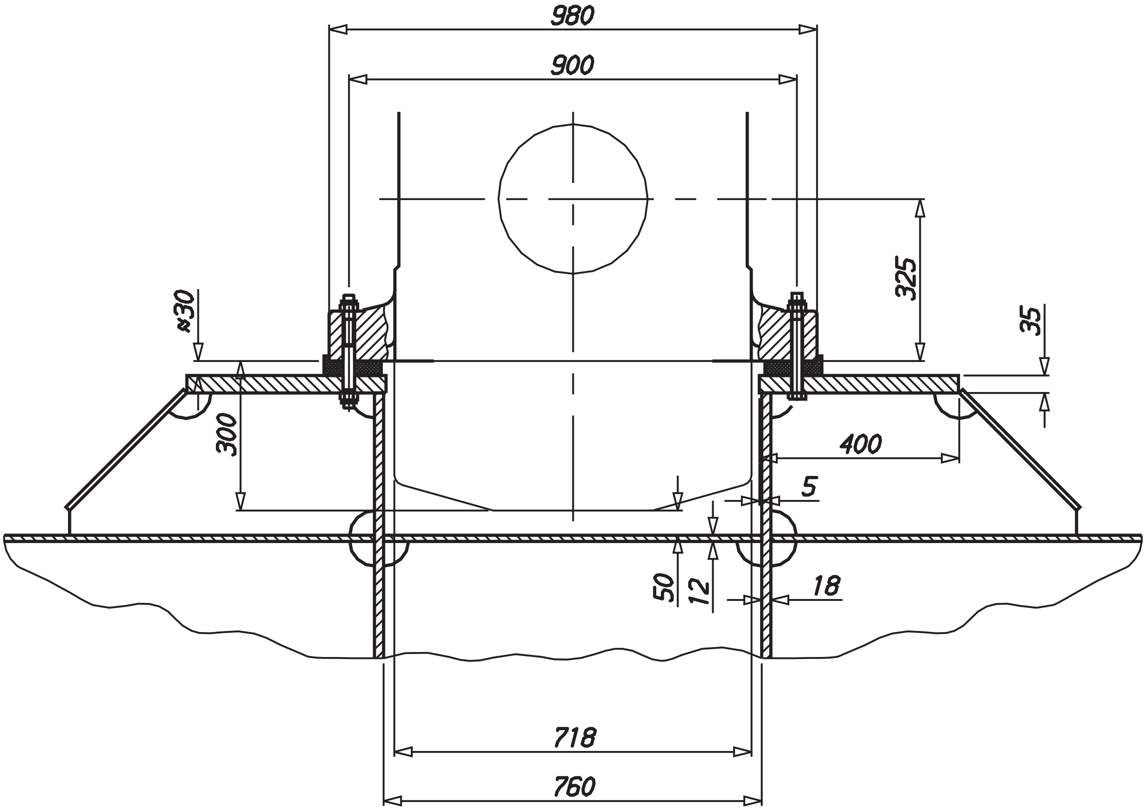
Main engine seating (2V69A0236a)

View from above

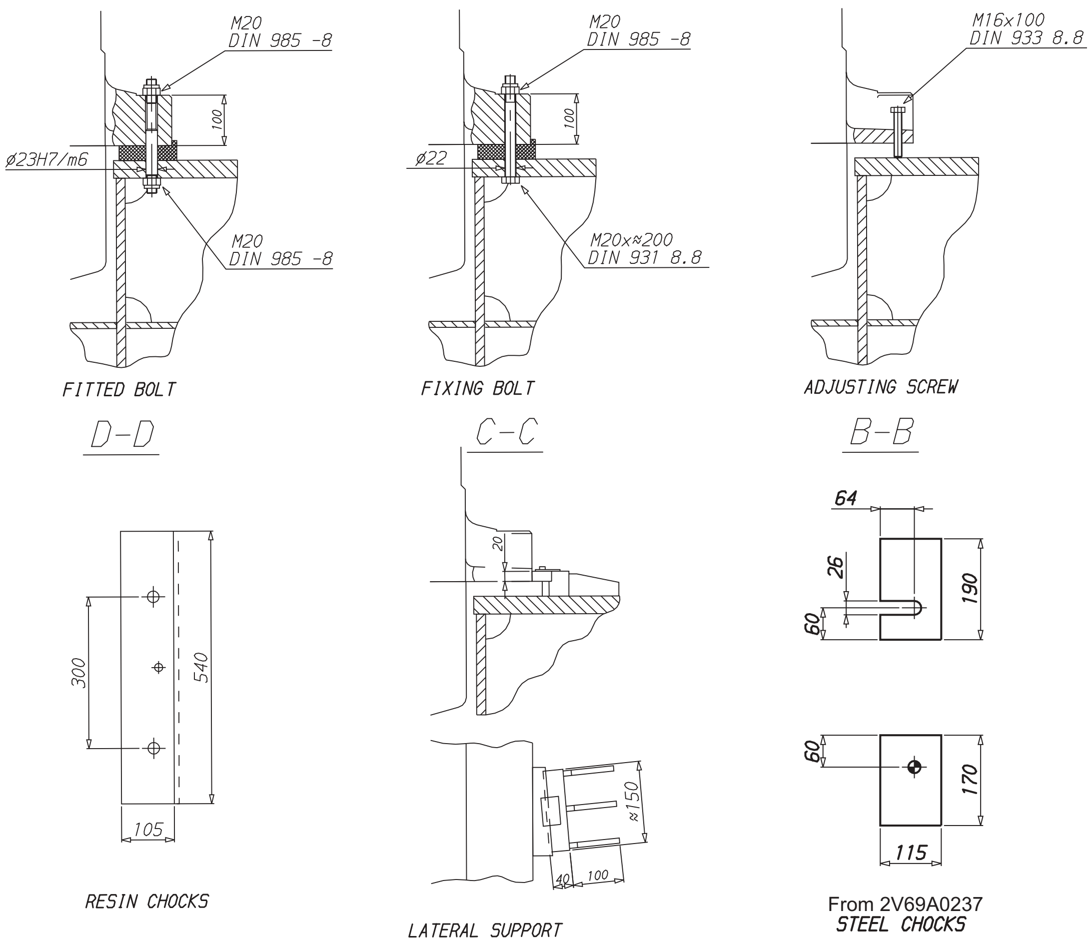


Main engine seating (2V69A0236a)

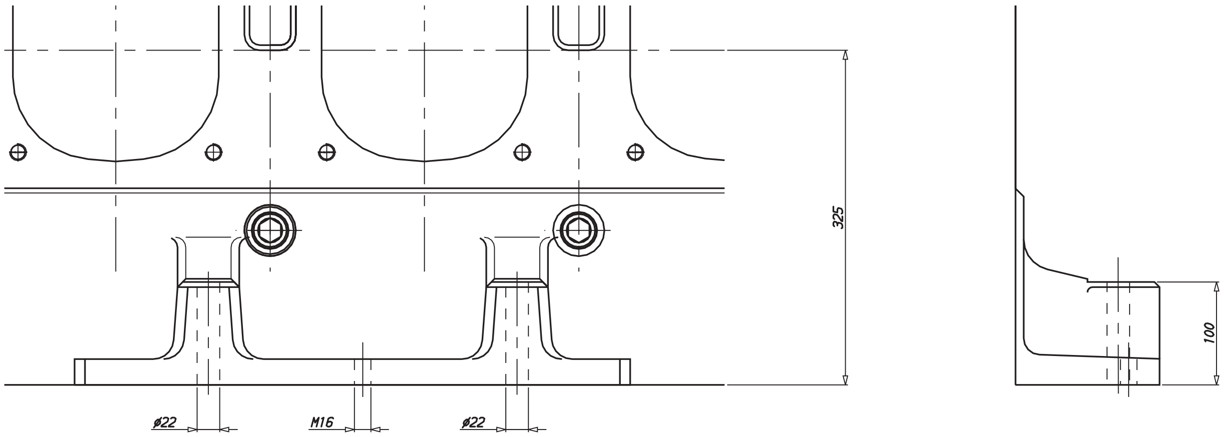
End view

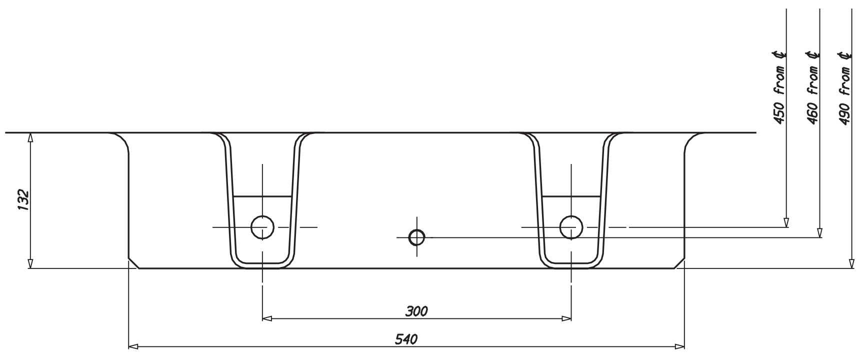


Chocking of main engines (2V69A0238b)



Mounting bracket (2V10A1836)

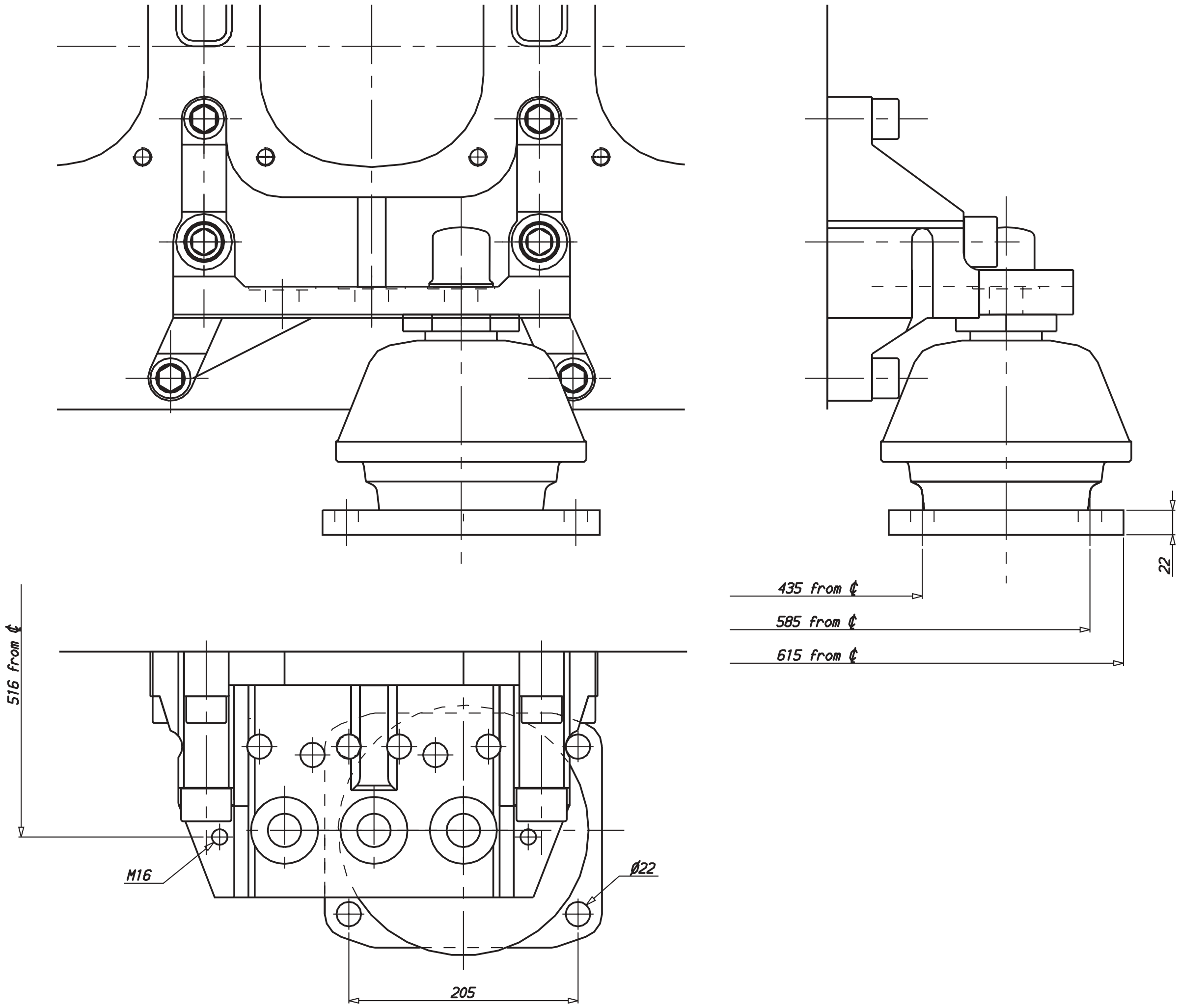




16.3.2.Resilient mounting

In order to reduce vibrations and structure borne noise, main engines may be resiliently mounted on rubber elements.

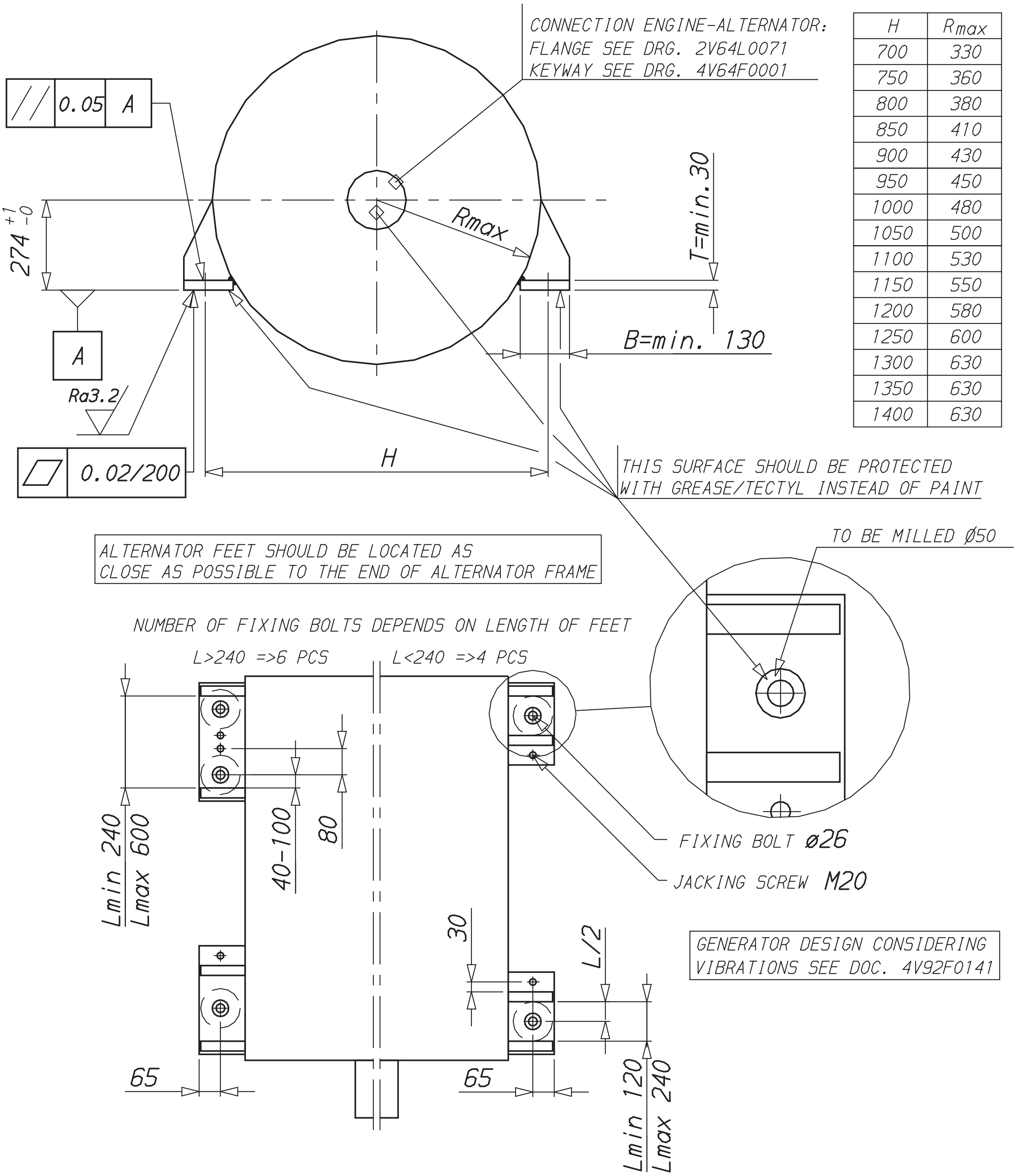
Mounting bracket (2V10A1837)



16.4. Mounting of generating sets

16.4.1. Alternator feet design

Instructions for designing the feet of the engine alternator and the distance between its holding down bolts (4V92F0134b)



H	Rmax
700	330
750	360
800	380
850	410
900	430
950	450
1000	480
1050	500
1100	530
1150	550
1200	580
1250	600
1300	630
1350	630
1400	630

16.4.2. Resilient mounting

Generating sets, comprising engine and generator mounted on a common base plate, are usually 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:

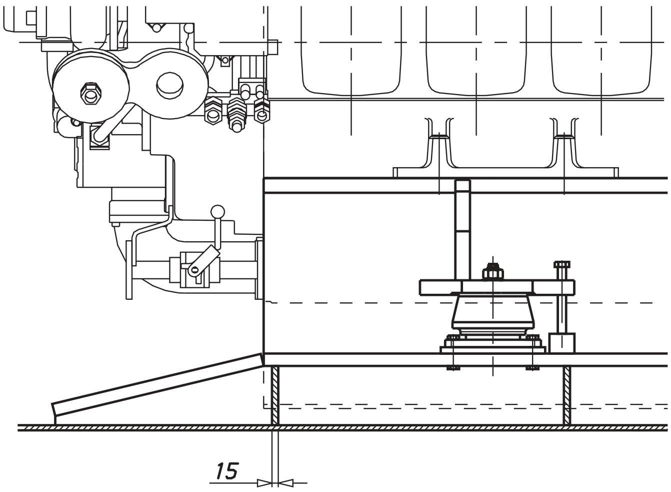
- Main engine speed [RPM] and number of cylinders
- Propeller shaft speed [RPM] and number of propeller blades

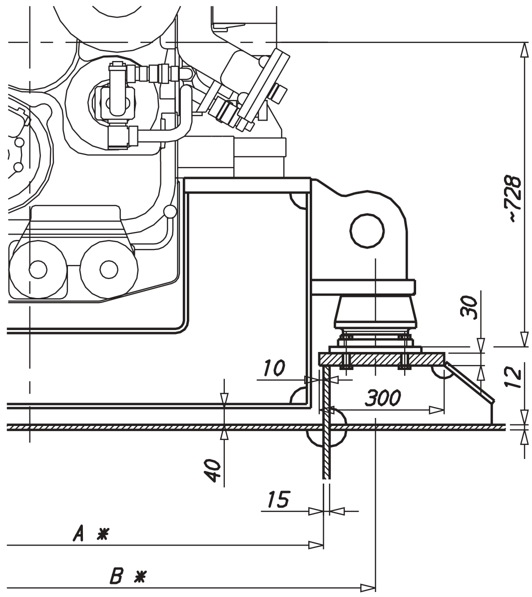
The selected number of mounts and their final position is shown in the generating set drawing.

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 drawing below.

Recommended design of the generating set seating (3V46L0720c)



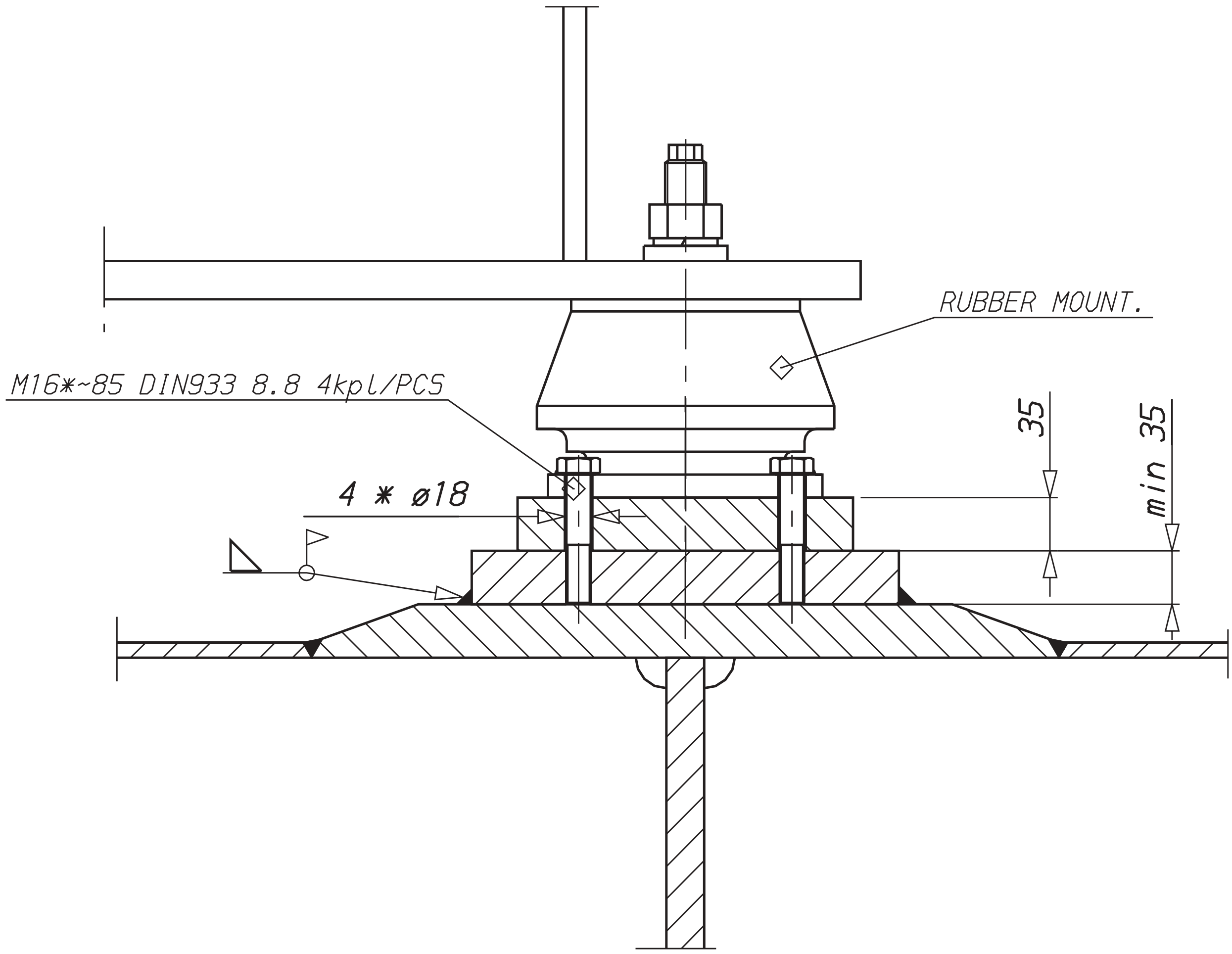


ENGINE	A *	B *
4L20	1330	1580
5L20	1330	1580
6L20	1330–1480	1580–1730
8L20	1480–1630	1730–1880
9L20	1630–1860	1880–2110

* 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.

Alternative design of seating (4V46L0296)



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.

Rubber mounts

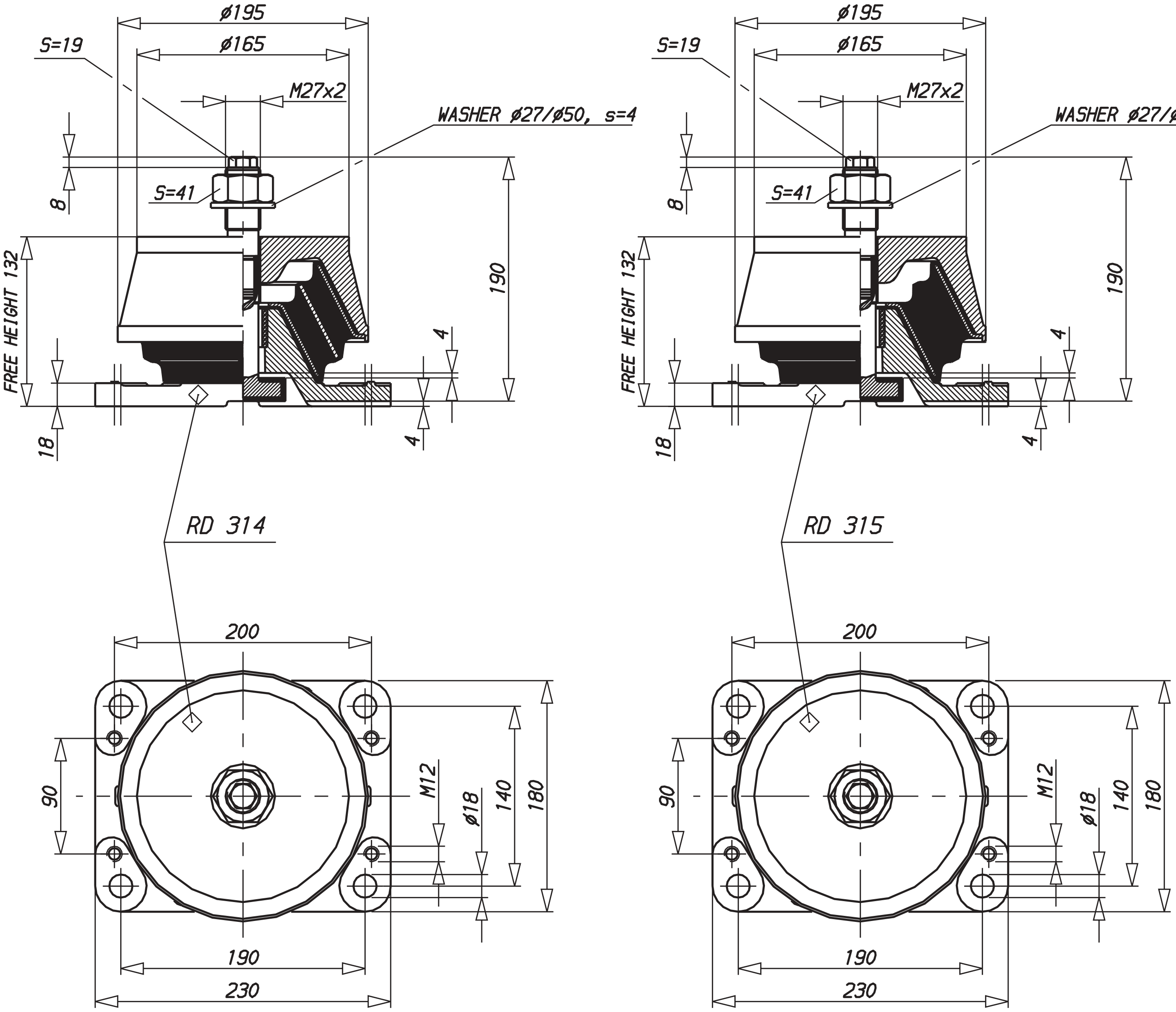
Conical mounts in natural rubber are used.

The mounts are equipped with an internal adjustable central buffer. Hence, no additional side or end buffers are required to limit the movements due to ship motions

Particularly with the alternative foundation design (see drawing 4V46L0296), care must be taken that the mounts will not come in contact with oil, oily water or fuel.

The mating surfaces of the common base plate is delivered machined, with holes for attaching the mounts which are delivered separate.

Rubber mounts (3V46L0706)



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

Installation of the generating set

A correct mounting of the generating set requires that all rubber elements are equally compressed, i.e. the load on each mount is equal. The installation procedure is:

- Remove the M27x2 nut and the washer from the mount.
- Attach each mount to the common base plate by fitting the washer on the central buffer and tighten the nut by hand (see drawing 3V46L0706).
- Lower the installation load onto the mounts, loosen the nut.
- Jacking screws are used for levelling the installation. Holes for the jacking screws are pre-drilled before delivery. M20 x 160 DIN933 8.8 jacking screws can be used. Jacking screws are supplied by the shipyard.
- Would the screw turn out to be too short, temporary chocks can be used to increase the lift.
- To avoid that the generating set weight is resting on the internal buffer instead of on the rubber it is important to check that all internal buffers can easily be turned by ap-

plying a spanner to the top hexagon (S = 19). If this is not possible, remove the installation load progressively until all buffers can be turned freely. Turn the internal buffer counter clockwise (upwards) and re-lower the installation onto the mounts. Repeat the above procedure until all buffers can be rotated freely with the full installation load applied.

- The correct deflection of the mounts is between 4 and 10 mm depending on the weight of the generating set and the selected quality of the rubber. The calculated compressed height of the mounts is shown in the generating set drawing.
- Check that the mounts are evenly compressed. The compressed height of all mounts must be within 2.0 mm. Adjustments in height shall be made using machined chocks. If shims are used the minimum thickness of a shim is 0.5 mm and only one shim per mount is permitted.
- Check that the seating of each mount is horizontal. This is done by measuring the compressed height of each mount on all sides. The difference must not exceed 0.5 mm.

Adjustments are made with wedge type chocks.

- Set the internal buffer working clearance for each mount:
- Turn the internal buffer counter clockwise (upwards) to the maximum upper position.
- Turn the internal buffer two full turns clockwise (downwards).
- Finally, tighten the nut with a torque of 300 Nm. While doing this the top hexagon must be secured with a spanner

The mounts should preferably be allowed to settle for a minimum of 48 hours, due to initial creeping, before lining up pipework, etc.

The transmission of forces emitted by the engine is 10...30% when using rubber mountings compared to rigid mounting.

16.5. Reduction gear foundations

The engine and the reduction gear must have common foundation girders.

16.6. Free end PTO driven equipment foundations

The foundation of the driven equipment must be integrated with the engine foundation.

16.7. Flexible pipe connections

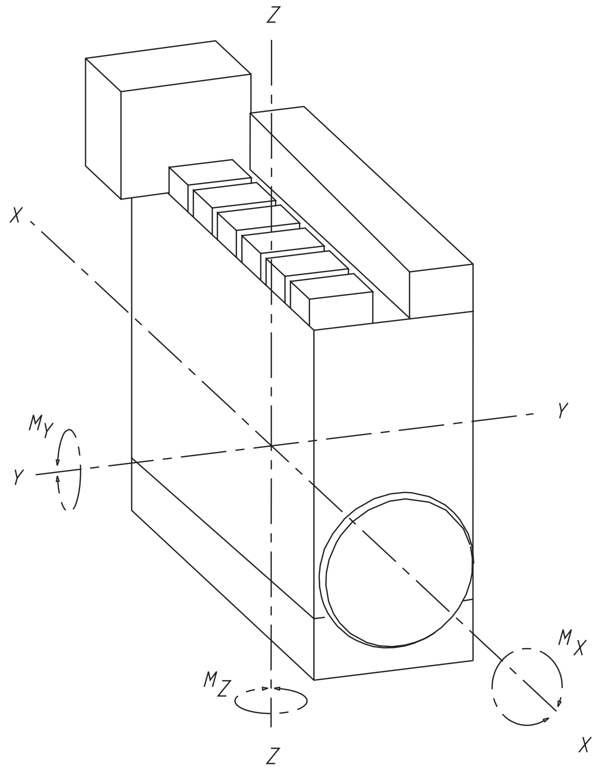
When the engine or generating set is resiliently installed, all connections must be flexible and no grating nor ladders may be fixed to the engine or generating set. When installing the flexible pipe connections, unnecessary bending or stretching should be avoided. The external pipe must be precisely 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.

Coordinate system of the external torques



17.2. External forces and couples

Table 17.1. External forces

FZ = 0, FY = 0 and FX = 0 for 5, 6 and 9 cylinder engines

Engine	Speed [rpm]	Frequency [Hz]	Fz [kN]
4L20	720	48	0.7
	750	50	0.8
	900	60	1.1
	1000	66.7	1.4
8L20	720	48	1.4
	750	50	1.5
	900	60	2.2
	1000	66.7	2.7

Table 17.2. External couples

MZ = 0, MY = 0 for 4, 6 and 8 cylinder engines

Engine	Speed [RPM]	Frequency [Hz]	MY [kNm]	MZ [kNm]
5L20	900	15	3.4	3.4
		30	21	—
	1000	16.7	4.2	4.2
		33.3	26	—
9L20	720	12	4.5	4.5
		24	3.1	—
	750	12.5	4.9	4.9
		25	3.3	—
	900	15	7	7
		30	4.8	—
	1000	16.7	8.6	8.6
		33.3	5.9	—

Table 17.3. Rolling moments

Engine	Speed	Frequency	Full load (M_X)	Zero load (M_X)	Frequency	Full load (M_X)	Zero load (M_X)
	[RPM]	[Hz]	[kNm]	[kNm]	[Hz]	[kNm]	[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
5L20	900	37.5	18	4.3	75	6.2	1.7
	1000	41.7	18	4.3	83.3	6.3	1.8
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

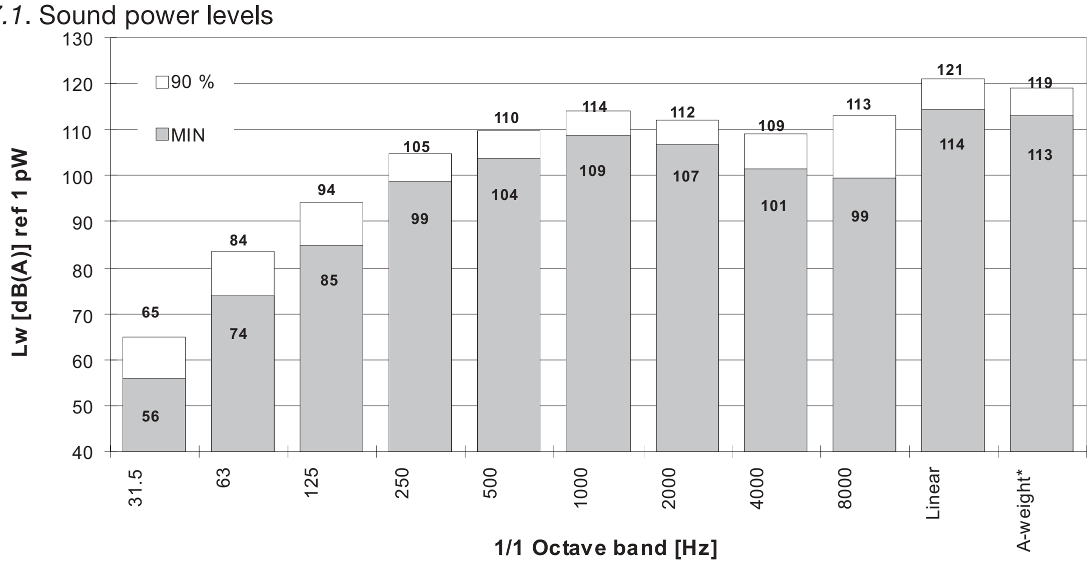
The mass-moments of inertia of the propulsion engines (including flywheel, coupling outer part and damper) Are typical as follows:

Engine	J[kgm²]
4L20	86 - 160
5L20	126 - 146
6L20	88 - 162
8L20	96 - 170
9L20	106 - 174

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 17.1. Sound power levels



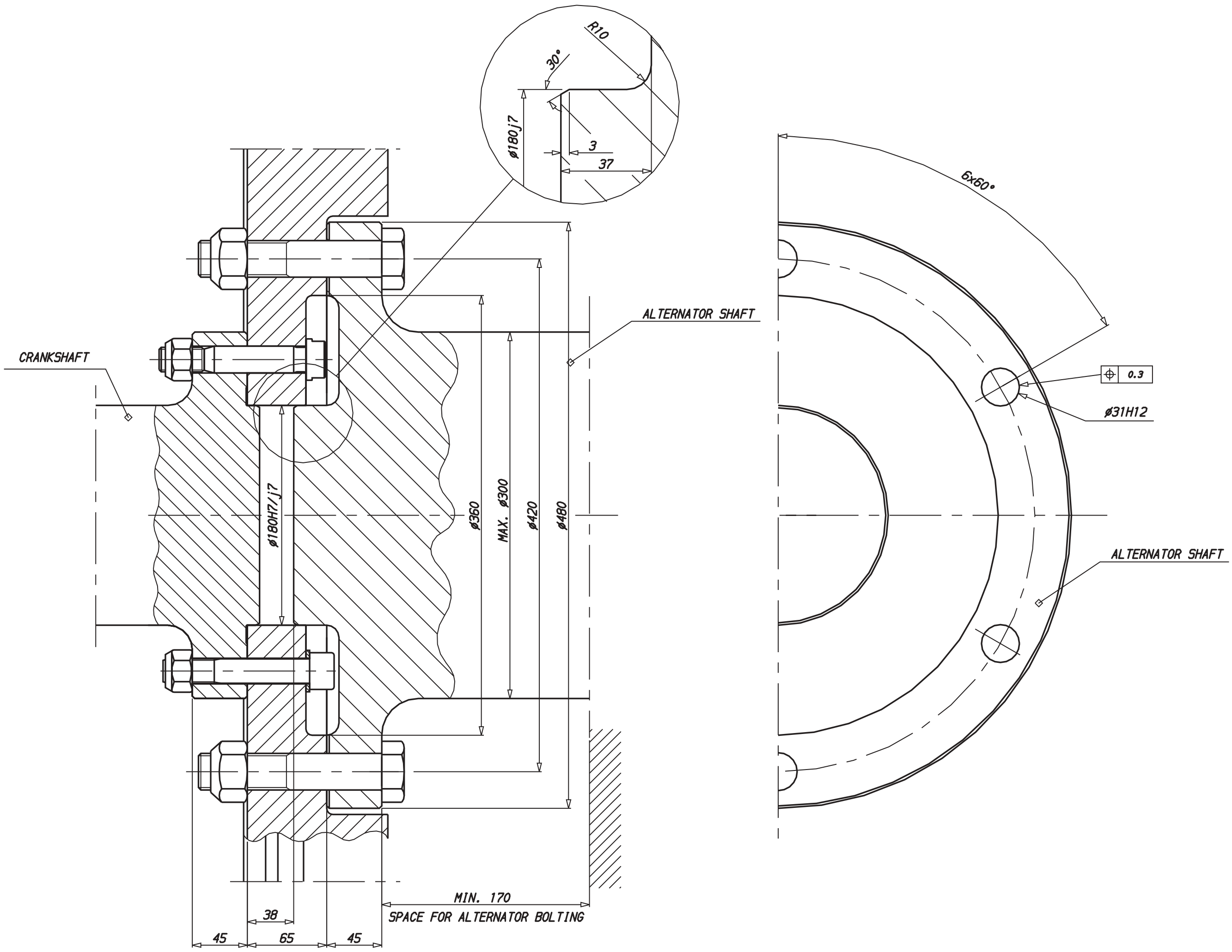
18. Power transmission

18.1. General

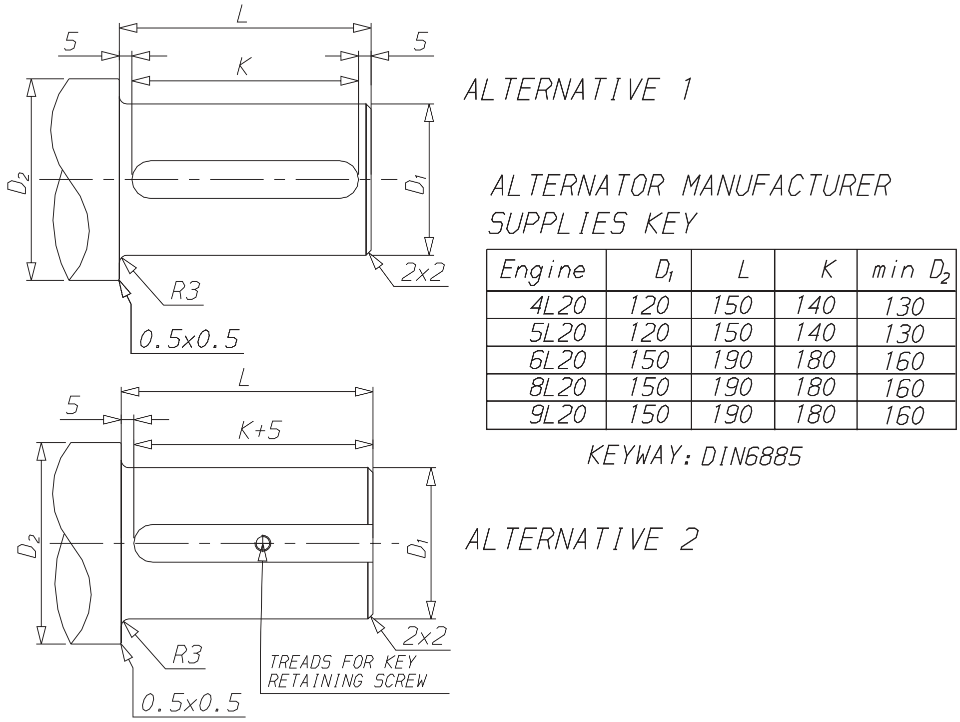
The full engine power can be taken from both ends of the engine. At the flywheel end there is always a flywheel for the management of the torsional vibration characteristics of the system and to facilitate manual turning of the engine. The flywheel creates a natural flange connection and in case needed also shaft connection can be provided. At the free end a shaft connection is provided in case a power take off is provided.

18.2. Connection to alternator

Connection engine/single bearing alternator (2V64L0071)



Connection engine/two-bearing alternator (4V64F0001)



Engine	D_1	L	K	min D_2
4L20	120	150	140	130
5L20	120	150	140	130
6L20	150	190	180	160
8L20	150	190	180	160
9L20	150	190	180	160

KEYWAY: DIN6885

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 alternator installations a flexible coupling between the engine and the generator is required.

18.4. Clutch

The clutch is required when two or more engines are connected to the same driven machinery like a reduction gear. The clutch is also required when the engine is connected to a reduction gear having a primary PTO.

Some consideration when deciding whether to have a clutch installed or not:

- In ships having more than one propeller it is possible to run the ship with just one propeller letting the other propeller(s) to windmill. This makes it possible to save the running hours of the standstill engine(s) or do maintenance on them. Anyhow for safety reasons the shaft is to be locked when working around rotating shafts in the engine.
- In case of blackout and no oil pressure the stopping of a declutched engine is so fast that the damages are minor even without gravity tank.
- The use of clutch reduces torsional stresses in elastic coupling while starting and stopping.
- The clutch creates investment and maintenance costs. It usually increases the length of the propulsion machinery.
- The clutch can lead to the loss of propulsion in case of automation or pressure problem.
- Badly adjusted clutch can cause torque peaks that cause damage to elastic coupling and reduction gear.
- Dry-friction type clutch can cause smoke formation to set off the fire alarm and sparks to ignite the oil on tank top causing engine room fire.

18.5. Shaftline locking device and brake

18.5.1. Locking device

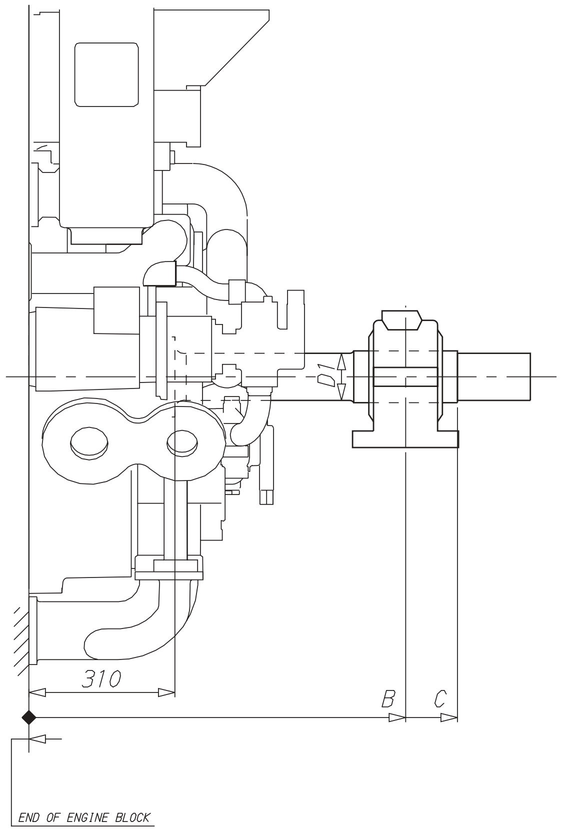
- A shaftline locking device is needed when the operation of the ship makes it possible to turn the shafting by the water flow in the propeller.

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.

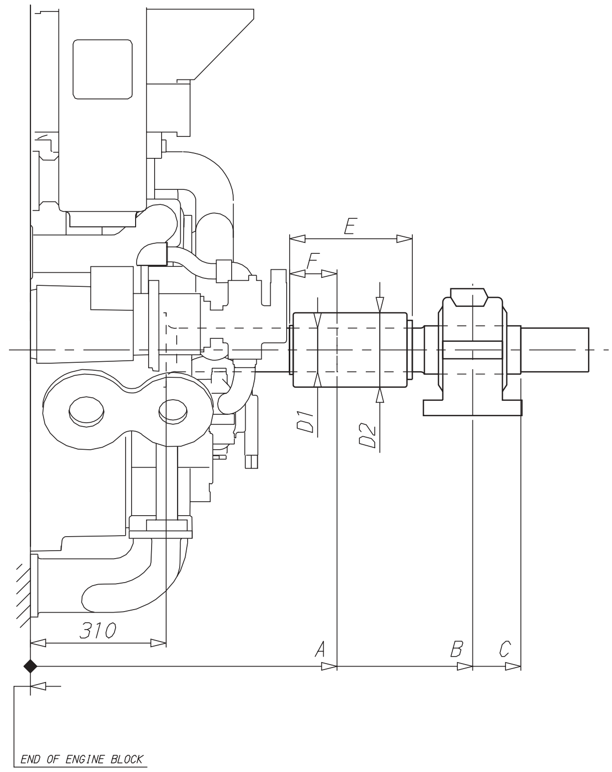
18.6. Power-take-off from the free end

Power take off at free end alternative 1 (4V62L0931)



D1	B	C
100	800	910

alternative 2 (4V62L0932)



D1	D2	A	B	C	E	F
100	170	700	1010	1120	280	108
110	185	750	1070	1180	300	118

18.7. 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 the list below.

General

- Classification
- Ice class
- Operating modes

Data of 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 number of the diagram

Data of 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/zero pitch in water
- torsional stiffness or dimensions of the shaft
- material of the shaft including tensile strength and modulus of rigidity
- drawing number of the diagram or drawing

Data of main alternator or shaft alternator

A mass-elastic diagram or an alternator shaft drawing showing:

- alternator 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 number of the diagram or drawing

Data of flexible coupling/clutch

If a certain make of flexible coupling has to be used, the following data of it must be informed:

- mass moment 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 number

18.8. Turning gear

- A manual turning tool is provided with the engine.

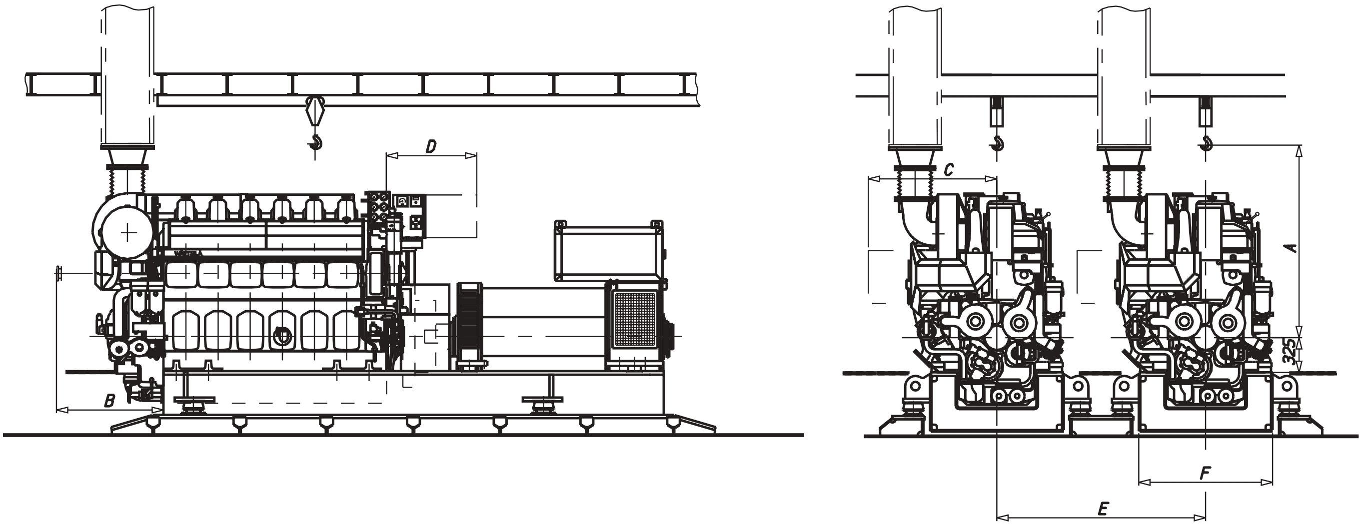
19. Engine room layout

19.1.Crankshaft distances

Minimum crankshaft distances have to be followed in order to provide sufficient space between engines for maintenance and operation.

19.1.1.In-line engines

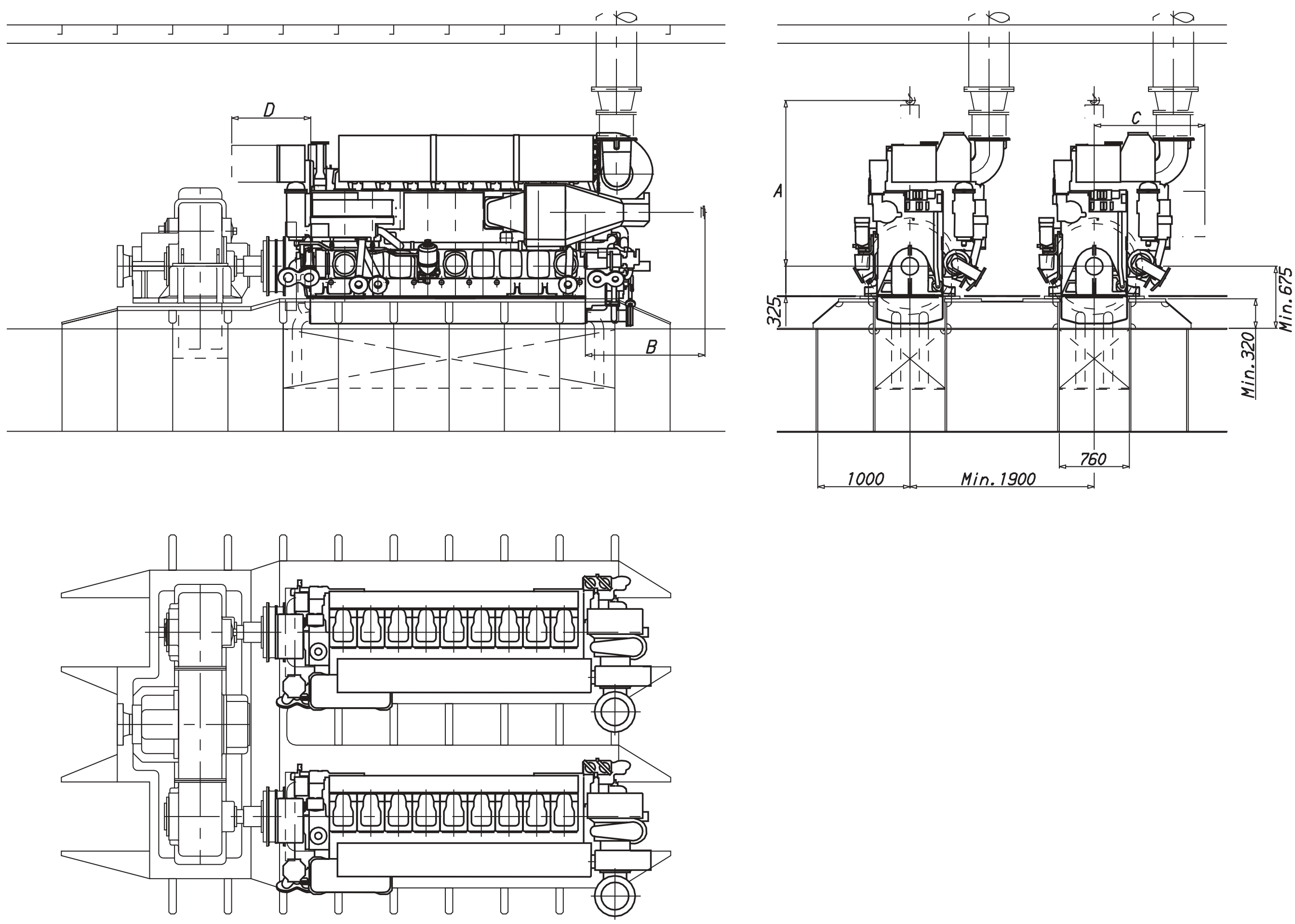
Engine room arrangement, generating sets (2V69C0278d)



ENGINE	A	B	C	D	E	F
4L20	1800	700	1200	845	1970	1270
5L20	1800	700	1200	845	1970	1270
6L20	1800	1000	1200	845	1970 / 2020	1270 / 1420
8L20	1800	1300	1200	845	2020 / 2170	1420 / 1570
9L20	1800	1300	1200	845	2170/2400	1570/1800

A = Minimum height when removing a piston
B = Camshaft overhaul distance
C = Charge air cooler overhaul distance
D = Length for the door in the connecting box, from engine block
E = Min. distance of engines dependent on common base plate
F = Width of the common base plate dependent on width of the alternator

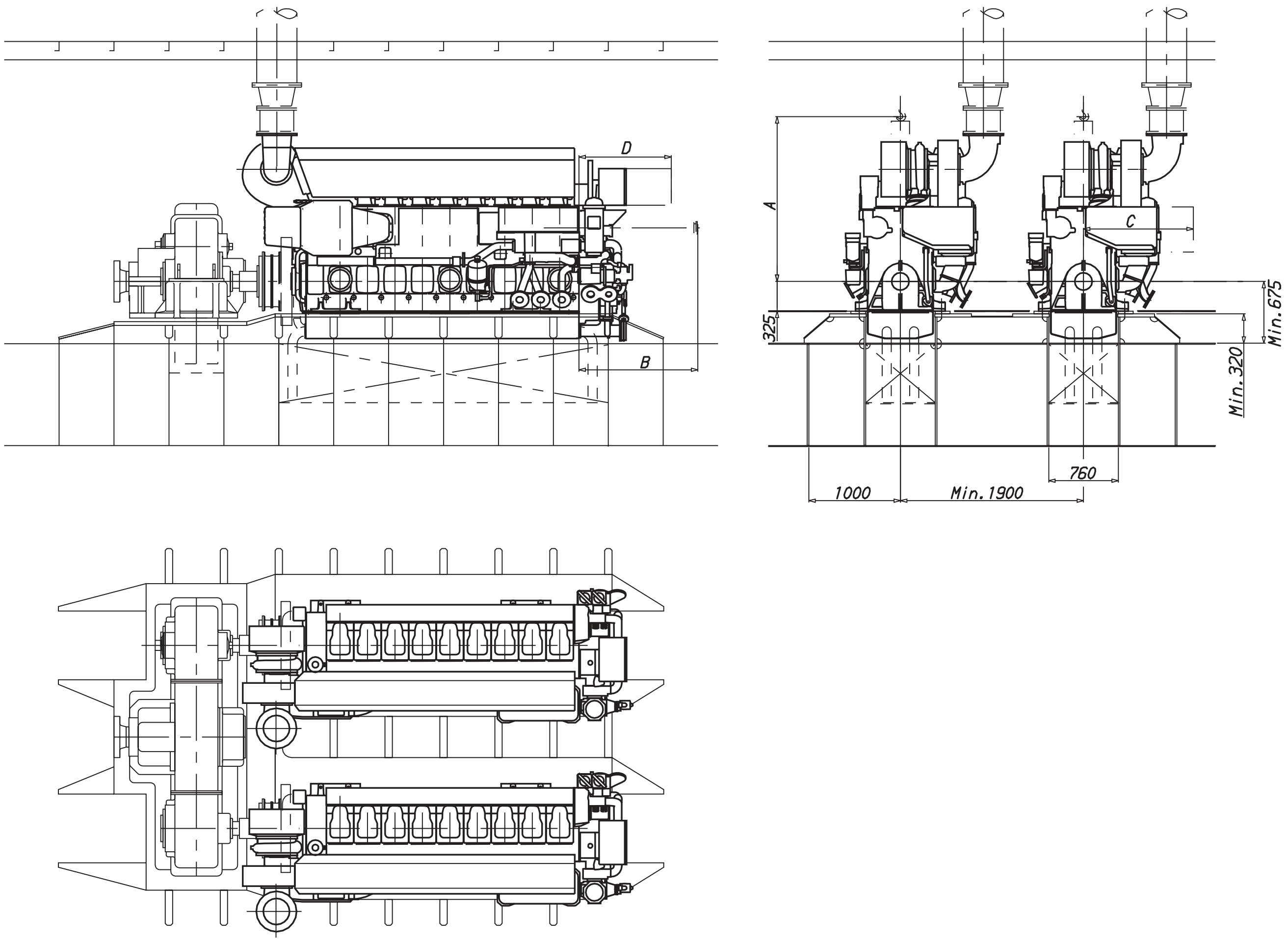
Engine room arrangement, main engines TC in free end (2V69C0275b)



Engine	A	B	C	D
4L20	1800	700	1200	845
5L20	1800	700	1200	845
6L20	1800	1000	1200	845
8L20	1800	1300	1200	845
9L20	1800	1300	1200	845

A = Minimum height when removing a piston
B = Camshaft overhaul distance
C = Charge air cooler overhaul distance
D = Length for the door on the connecting box, from engine block

Engine room arrangement, main engines TC in driving end (2V69C0276)



Engine	A	B	C	D
6L20	1800	1000	1200	1010
8L20	1800	1300	1200	1010
9L20	1800	1300	1200	1010

A = Minimum height when removing a piston
B = Camshaft overhaul distance
C = Charge air cooler overhaul distance
D = Length for the door on the connecting box, from engine block

19.1.2. Father-and-son arrangement

When connecting two engines of different type and/or size to the same reduction gear the minimum crankshaft distance has to be evaluated case by case. However, some general guidelines can be given:

- It is essential to check that all engine components can be dismounted. The most criticals are usually turbochargers and charge air coolers.
- Special care has to be taken checking the maintenance platform elevation between the engines to avoid structures that obstruct maintenance.

19.1.3. Distance from adjacent intermediate/propeller shaft

Some machinery arrangements feature an intermediate shaft or propeller shaft running adjacent to engine. To allow adequate space for engine inspections and maintenance there has to be sufficient free space between the intermediate/propeller shaft and the engine.

To enable safe working conditions the shaft has to be covered. It must be noticed that also dimensions of this cover have to be taken into account when determining the shaft distances in order to fulfil the requirement for minimum free space between the shaft and the engine.

19.2. Space requirements for maintenance

19.2.1. Working space around the engine

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. Engine room height and 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 for General data and outputs for the necessary hook heights.

19.2.3. Maintenance platforms

In order to enable efficient maintenance work on the engine, it is advised to build the maintenance platforms on recommended elevations. The width of the platforms should be at minimum 800 mm to allow adequate working space. The surface of maintenance platforms should be of non-slippery material (grating or chequer plate).

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.

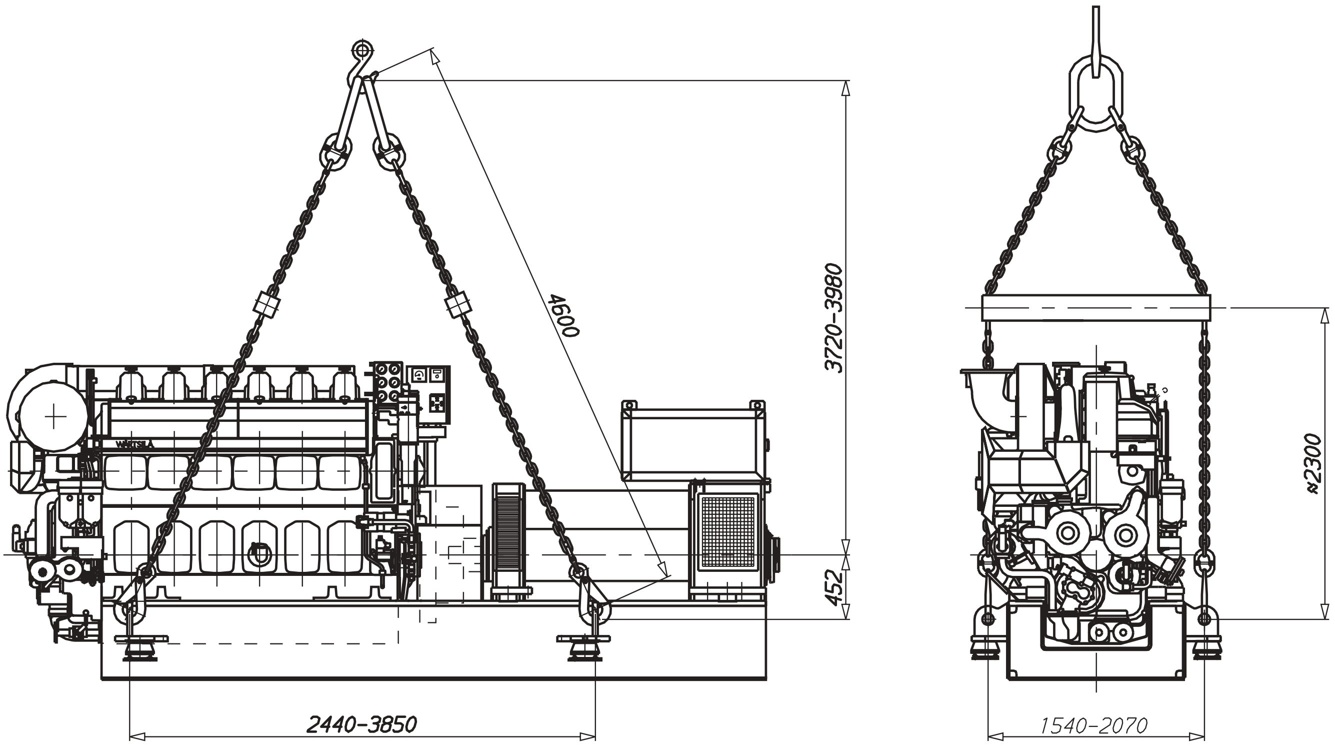
19.4. Required deck area for service work

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.

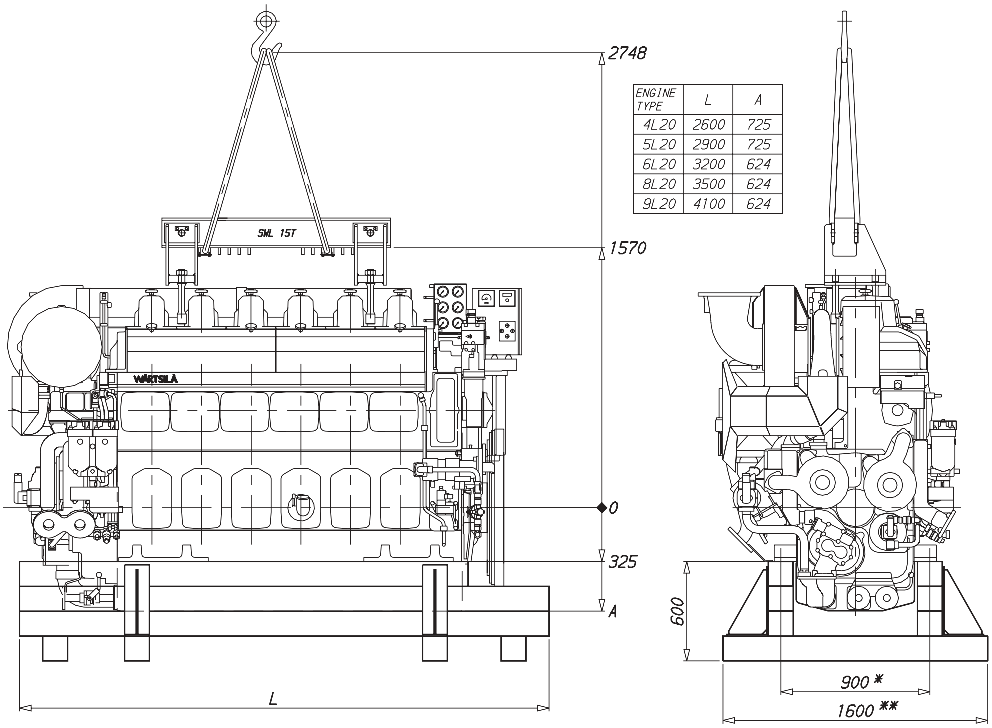
20. Transport dimensions and weights

20.1. Lifting of engines

Lifting of generating sets (3V83D0300c)



Lifting of main engines (3V83D0285b)

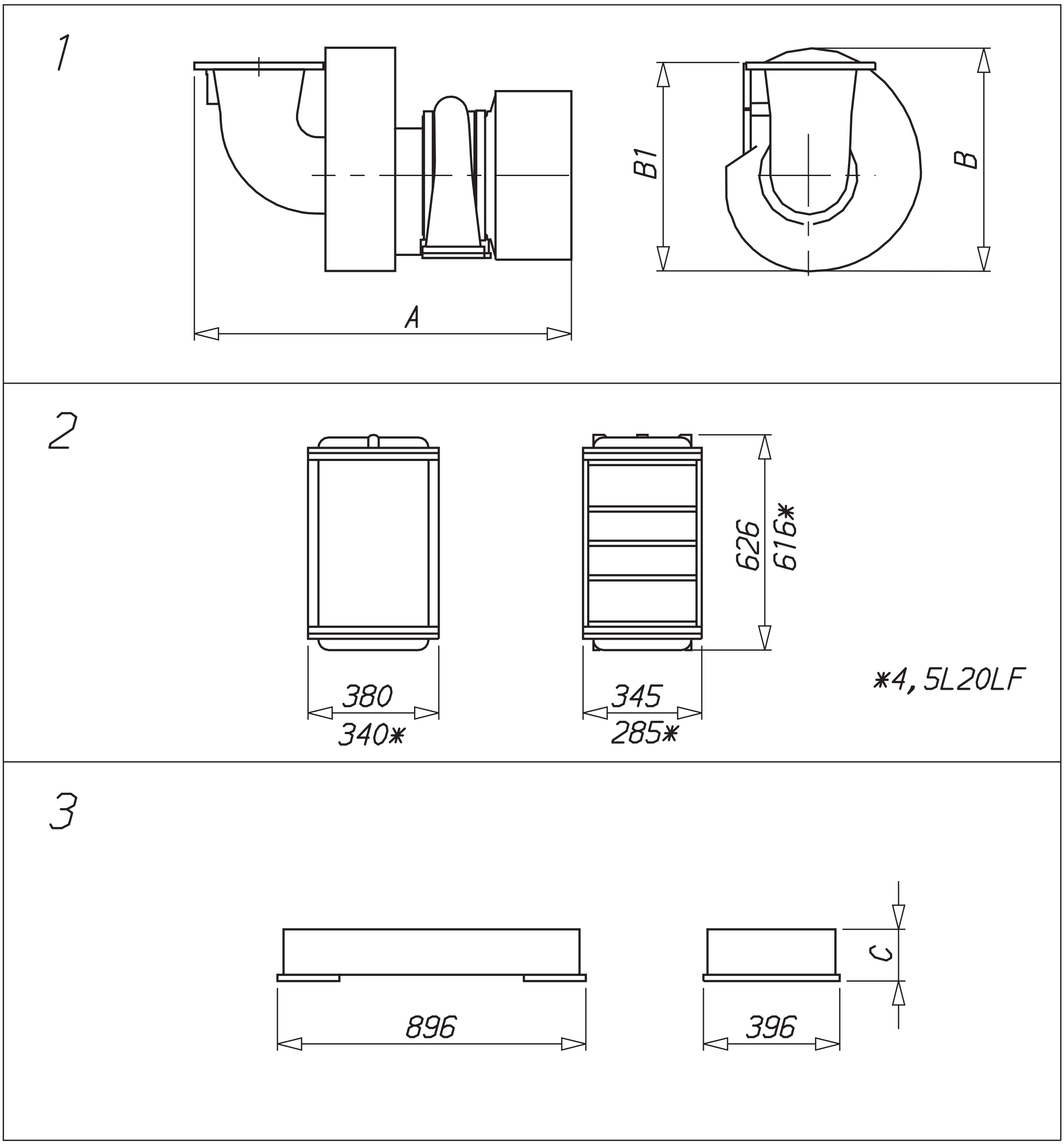


ENGINE TYPE	L	A
4L20	2600	725
5L20	2900	725
6L20	3200	624
8L20	3500	624
9L20	4100	624

*) 1020 FOR FLEXIBLE MOUNTED ENGINE
**) 1720 FOR FLEXIBLE MOUNTED ENGINE

20.2. Engine components

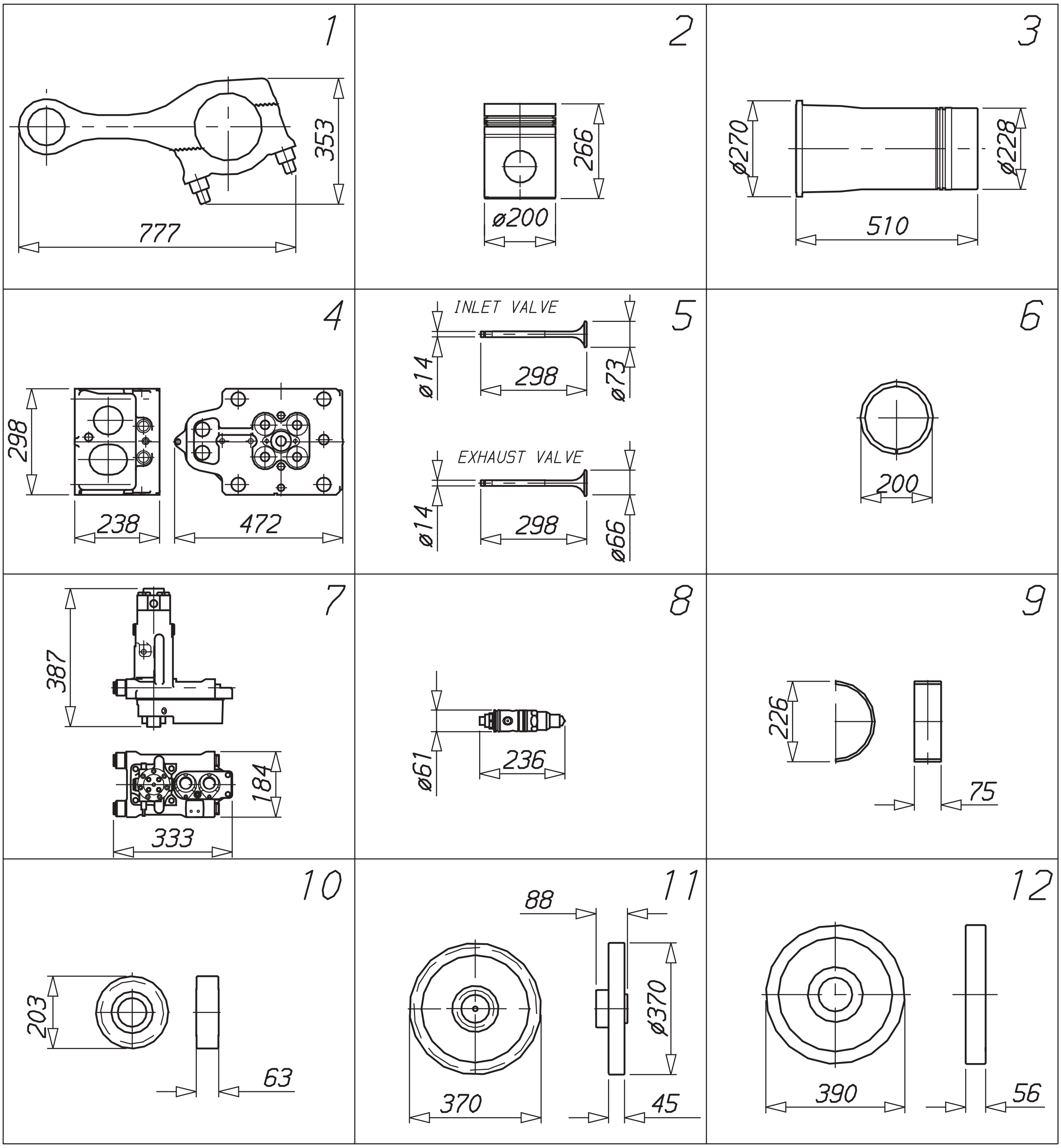
Turbocharger and cooler inserts (4V92L1282)



Engine	Weights (kg)				Dimensions (mm)			
	1 (LF)	1 (LD)	2	3	A	B (LF)	B1 (LD)	C
4L	147	-	120	78	945	576		150
5L	212	-	120	87	1097	648		168
6L	212	212	160	101	1097	636	692	196
8LPC	340	-	160	123	1339	785		247
8L2P	340	340	160	123	1339	760	797	247
9L	340	340	160	137	1339	773	813	275

1. Turbocharger
2. Charge air cooler
3. Lubricating oil cooler insert

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:

3V58E0584	4L20 Diesel alternator set LF
3V58E0580a	5L20 Diesel alternator set LF
3V58E0553b	6L20 Diesel alternator set LF
3V58E0552b	8L20 Diesel alternator set LF
3V58E0549b	9L20 Diesel alternator set LF
4V58E0541e	6L20 Diesel engine LD
4V58E0533c	8L20 Diesel engine LD
4V58E0534d	9L20 Diesel engine LD
4V58E0564	4L20 Diesel engine LF
4V58E0568a	5L20 Diesel engine LF
4V58E0560b	6L20 Diesel engine LF
4V58E0559b	8L20 Diesel engine LF
4V58E0561a	9L20 Diesel engine LF

22. ANNEX

22.1.Ship inclination angles

Inclination angles at which main and essential auxiliary machinery is to operate satisfactorily (4V92C0200a)

Classification society	Lloyd's Register of Shipping 2000	Det Norske Veritas 1999	American Bureau of Shipping 2000	Germanischer Lloyd 1998	Bureau Veritas 2000
Main and aux. engines					
Paragraph	Pt.5 Ch.1 Sec.3 Par.3.6	Pt.4 Ch.1 Sec.3 Par.B200	Pt.4 Ch.1 Sec.1 Par.7.9	Pt.1 Ch.2 Sec.1 Par.C 1.1	Pt.C Ch.1 Sec.1 Par.2.4.1
Heel to each side	15	15	15	15	15
Rolling to each side	22.5	22.5	22.5	22.5	22.5
Ship length, L	L < 100 L > 100	—	—	—	—
Trim	5 500/L	5	5	5	5
Pitching	7.5	7.5	7.5	7.5	7.5
Emergency sets					
Heel to each side	22.5*	22.5*	22.5*	22.5*	22.5*
Rolling to each side	22.5	22.5*	22.5*	22.5*	22.5*
Trim	10	10	10	10	10
Pitching	10	10	10	10	10
Electrical installation**				***	
Paragraph	Pt.6 Ch.2 Sec.1 Par.1.9	Pt.4 Ch.4 Sec.2 Par.A101	Pt.4 Ch.1 Sec.1 Par.7.9	Pt.1 Ch.3 Sec.1 Par.E 1.1	Pt.C Ch.2 Sec.2 Par.1.6.1
Heel to each side	15	15	15	15	15
Rolling to each side	22.5	22.5	22.5	22.5	22.5
Ship length, L	L < 100 L > 100	—	—	—	—
Trim	5 500/L	5	5	5	5
Pitching	7.5	10	7.5	7.5	7.5

Classification society	Russian Maritime Reg. of Shipping 1995	Polsky Rejestr Statkow 1990	Registro Italiano Navale 1999	China Classification Society 1998	Korean Register of Shipping 1995
Main and aux. engines					
Paragraph	VII-1.6	VII-1.6	Sec.C Ch.2 Par.2.1.5	Pt.III Ch.1 Sec.1.1.3.1	5.1.103
Heel to each side	15	15	15	15	15
Rolling to each side	22.5	22.5	22.5	22.5	22.5
Trim	5	5	5	5	5
Pitching	7.5	7.5	7.5	7.5	7.5
Emergency sets					
Heel to each side	22.5*	22.5*	22.5*	22.5	22.5*
Rolling to each side	22.5*	22.5*	22.5*	22.5	22.5*
Trim	10	10	10	10	10
Pitching	10	10	10	10	10

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.

*** Machinery and equipment relative to main electrical power installation require the same inclination angles as "Main and aux. engines"

22.2. Unit conversion tables

Length

Length	m	in	ft	mile	nautical mile
m	1	39.370	3.2808	6.2137e-04	5.3996e-04
in	0.0254	1	8.3333e-02	1.5783e-05	0.0000137149028078
ft	0.3048	12	1	1.8939e-04	1.6458e-04
mile	1609.3	63360	5280	1	0.86898
nautical mile	1852	72913	6076.1	1.1508	1

Values are rounded to five meaning digits where not 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

Equations are accurate.

Area

Area	square m	square inch	square foot
square m	1	1550.0	10.764
square inch	6.4516e-04	1	6.9444e-03
square foot	9.2903e-02	144	1

Values are rounded to five meaning digits where not 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

Equations are accurate.

Volume

Volume	cubic m	l (liter)	cubic inch	cubic foot	Imperial gallon	US gallon
cubic m	1	1000	61024	35.315	219.97	264.17
l (liter)	0.001	1	61.024	3.5315e-02	0.21997	0.26417
cubic inch	1.6387e-05	1.6387e-02	1	5.7870e-04	3.6047e-03	4.3290e-03
cubic foot	2.8317e-02	28.317	1728	1	6.2288	7.4805
Imperial gallon	4.5461e-03	4.5461	277.42	0.16054	1	1.2009
US gallon	3.7854e-03	3.7854	231	0.13368	0.83267	1

Values are rounded to five meaning digits where not accurate.

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

Equations are accurate but some of them are reduced in order to limit the number of decimals.

Energy

Energy	J	BTU	cal	lbf ft
J	1	9.4781e-04	0.23885	0.73756
BTU	1055.06	1	252.00	778.17
cal	4.1868	3.9683e-03	1	0.32383
lbf ft	1.35582	1.2851e-03	3.0880	1

Values are rounded to five meaning digits where not accurate.

Mass

Mass	kg	lb	oz
kg	1	2.2046	35.274
lb	0.45359	1	16
oz	0.028350	0.0625	1

Values are rounded to five meaning digits where not accurate.

Density

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

Values are rounded to five meaning digits where not accurate.

Power

Power	W	hp	US hp
W	1	0.0013596	0.0013410
hp	735.499	1	1.0136
US hp	745.7	0.98659	1

Values are rounded to five meaning digits where not accurate.

Pressure

Pressure	Pa	bar	mmWG	psi
Pa	1	0.00001	0.10197	0.00014504
bar	100000	1	10197	14.504
mmWG	9.80665	9.80665e-05	1	0.0014223
psi	6894.76	0.0689476	703.07	1

Values are rounded to five meaning digits where not accurate.

Massflow

Massflow	kg / s	lb / s
kg / s	1	2.2046
lb / s	0.45359	1

Values are rounded to five meaning digits where not accurate.

Volumeflow

Volumeflow	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	1.6667e-05	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	1.6387e-05	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	3.7133e-05
cubic ft / h	7.8658e-06	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	1.0515e-06	15.850	264.17	15.584	26930	7.4805	3600	1

Values are rounded to five meaning digits where not accurate.

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:

T = Tera = 1 000 000 000 000 times

G = Giga = 1 000 000 000 times

M = Mega = 1 000 000 times

k = kilo = 1 000 times

m = milli = divided by 1 000

μ = micro = divided by 1 000 000

n = nano = divided by 1 000 000 000

22.3. Collection of drawing symbols used in drawings

- Valve, general sign
- Non-return valve, general sign
- Automatic actuating valve
- Spring-loaded overflow valve, straight, angle
- Remote-controlled valve
- Three-way valve, general sign
- Self-contained thermostat valve
- Solenoid valve
- Pump, general sign
- Electrically driven pump
- Turbocharger
- Filter or strainer
- Automatic filter with by-pass filter
- Heater
- Separator
- Flow meter
- Viscosimeter
- Receiver
- Flexible pipe, hose
- Insulated pipe
- Insulated and heated pipe
- Water, oil and condensate separator, general sign
- Electrically driven compressor
- Tank
- Orifice
- Quick-coupling

LOCAL INSTRUMENTS

- TI 231 — Thermometer
- PI 201 — Pressure gauge
- PDI 243 — Differential pressure indicator

SENSORS

- LS 204 — Level switch
- PS 311 — Pressure switch
- PDS 113 — Differential pressure switch
- TSZ 201 — Temperature switch for control
- PSZ 201 — Pressure switch for control
- PT 301 — Pressure transmitter
- TE 201 — Temperature sensor (analogue)
- GS 792 — Limit switch
- ST 173 — Inductive transducer
- SE 518 — Magnetic transducer

22.4. 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 xx or later or other application capable of reading the files

The files are organized in folders according to the engine types.

W0204E / Bock's Office / Fram

Wärtsilä Finland Oy
P.O.Box 252
65101 Vaasa, Finland
Tel. +358 10 709 0000
Fax. +358 6 356 7188

www.wartsila.com

WÄRTSILÄ