Everllence

Everllence B&W
two-stroke
marine engines

Main engine auxiliary systems

Efficiency improvements

Contents

Efficiency improvements

In the design process of main engine auxiliary systems conducted by the shipyard, options are available to improve efficiency, reduce daily fuel oil consumption, and consequently lower CO_2 emissions. These options cover power efficiency improvements of electric auxiliary equipment, pumps, fans, etc., serving the main engine, but also efficiency improvements related directly to the main engine specific fuel oil consumption.

This technical paper describes each of the different relevant main engine auxiliary systems and the options available for efficiency improvements. Different solutions are mentioned for each system, some of these can be combined and the savings potential added up, while others will exclude each other depending on the selected option.

To illustrate the potential savings obtained by installing the suggested efficiency improvements, a specific engine type has been chosen and an annual operating profile has been defined. Some of the efficiency improvements may also have a beneficial impact on the common auxiliary system for other consumers. This is not accounted for in the description and calculation of the savings potential.

Note that other larger efficiency improvements, such as the waste heat recovery system, are not described in this document as these systems are described in other Everllence documents.

1. Basic data

In this document, the identified savings potential is illustrated by an annual HFO saving based on a specific main engine and a specific annual operating profile. The annual HFO savings potential is not converted to operational expenditure (OPEX) savings, as fuel oil prices differ according to bunker oil quality, location, and the global financial situation. Furthermore, the capital expenditure (CAPEX) for each suggested efficiency improvement has not been estimated, since it will differ for each contractor/supplier/shipyard.

The business case based on OPEX savings and the additional CAPEX for each individual efficiency improvement must be carefully investigated by the vessel operator. This evaluation will determine whether the investment offers a beneficial payback time and positive net present value, considering factors such as operating profile, interest rates, and vessel type and size.

Main engine

To estimate the specific savings for the suggested efficiency improvements, the engine type Everllence B&W 8G95ME-C9.5 Tier II has been chosen. The specific fuel oil consumption (SFOC) versus engine load for this engine is listed in Table 1 and depicted in Fig. 1. This engine type is installed in, for example, large container vessels operating worldwide. It is possible to obtain the computerised engine application system (CEAS) report for the Everllence B&W two-stroke engine 8G95ME-C9.5 Tier II at: www.everllence.com

$SFOC_{ISO}$ for 8G95ME-C9.5 Tier II

Engine load [%]	Engine load [kW]	Rev. [rpm]	$SFOC_{ISO}$ [g/kWh]
0	-	-	-
10	5,496	37.1	187.1
15	8,244	42.5	179.1
20	10,992	46.8	174.1
25	13,740	50.4	171.1
30	16,488	53.6	169.1
35	19,236	56.4	168.1
40	21,984	58.9	166.7
45	24,732	61.3	165.6
50	27,480	63.5	164.5
55	30,228	65.5	163.6
60	32,976	67.5	162.8
65	35,724	69.3	162.3
70	38,472	71.0	162.0
75	41,220	72.7	162.1
80	43,968	74.3	162.5
85	46,716	75.8	163.0
90	49,464	77.2	163.8
95	52,212	78.6	164.8
100	54,960	80.0	166.0

Table 1: $SFOC_{ISO}$ for 8G95ME-C9.5 Tier II

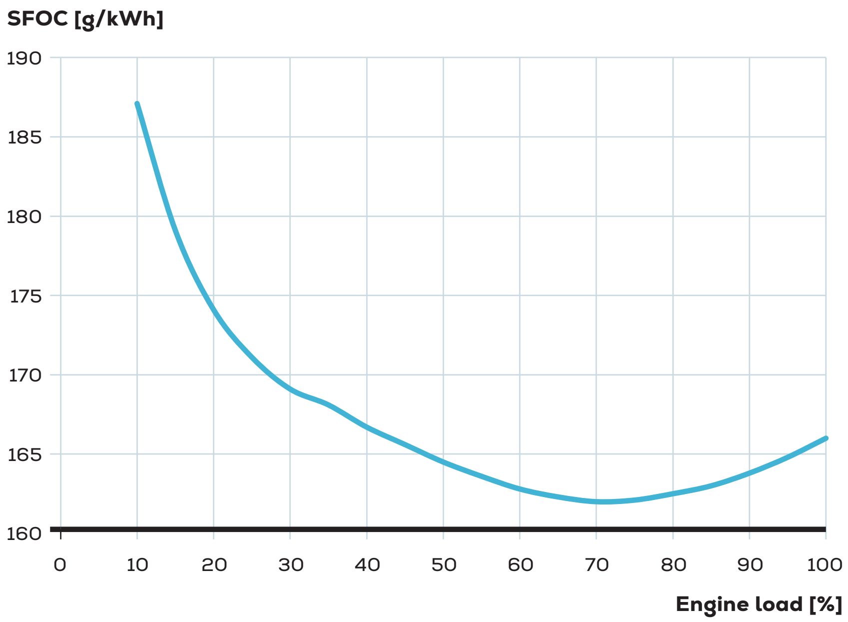


Fig. 1: $SFOC_{ISO}$ for 8G95ME-C9.5 Tier II

Annual operating profile

An annual operating profile has been estimated to calculate the main engine (ME) annual fuel oil consumption. This profile will differ according to the actual route of the vessel. However, the profile is needed to define the matrix in Table 2 to estimate the number of operating hours at different ME loads and ambient seawater (SW) temperatures (see Fig. 2). Port stays are listed as 0% engine load in Table 2.

Annual main engine fuel oil consumption

Based on the SFOC for the 8G95ME-C9.5 Tier II engine combined with the annual operating hours, it is possible to get the SFOC for each specific condition via the CEAS application. The adjusted SFOC value is multiplied by the specific engine load and operating hours (given in each matrix cell) to get the heavy fuel oil consumption (HFOC), which is entered in Table 3.

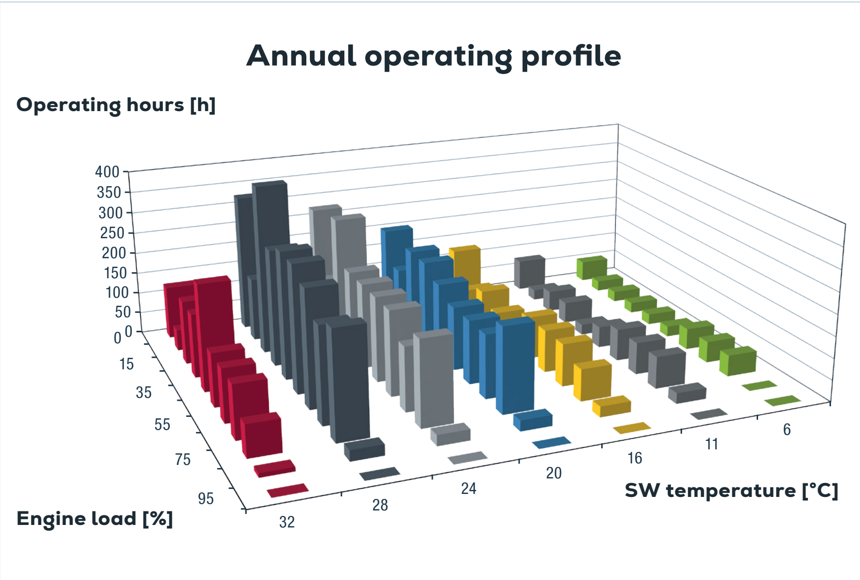


Fig. 2: Operating hours at different ME loads and ambient seawater parameters

The following assumptions are used in the HFOC calculation in Table 3:

- The engine is only operated on HFO with the lower calorific value, LCV = 40,200 kJ/kg.
- The cooling water temperatures for the scavenge air cooler follow the SW temperature +4°C, as the three-way valve setpoint is 10°C.
- The ambient air inlet temperature will follow the SW temperature with the following estimates: The deck air temperature is on average equal to the SW temperature + 3°C. Normally, the engine room temperature is considered approx. 10 to 14°C (on average 12°C) higher than the deck air temperature, which leads to an engine room temperature approx. 15°C above the SW temperature. As the combustion air to the engine is supplied by ventilation ducts placed near the turbocharger, the air inlet temperature to the turbochargers will be lower than the engine room

Annual operating profile

Ambient SW		Engine load [%] 0	1-10	11-20	21-30	31-40	41-50	51-60	61-70	71-80	81-90	91-100	Total	Perc.
temperature [°C]		0	10	15	25	35	45	55	65	75	85	95	[h]	[%]
>30	32	125	50	150	150	250	125	125	125	75	10	0	1,185	13.5
26-30	28	325	150	400	275	300	300	275	225	250	25	0	2,525	28.8
22-26	24	275	125	300	200	200	200	200	150	200	25	0	1,875	21.4
18-22	20	200	125	200	200	175	150	150	150	200	25	0	1,575	18.0
14-18	16	125	50	75	50	50	100	100	100	75	25	0	750	8.6
8-14	11	75	25	50	50	25	50	75	75	75	25	0	525	6.0
<8	6	50	25	25	25	25	25	50	50	50	0	0	325	3.7
Total	[h]	1,175	550	1,200	950	1,025	950	975	875	925	135	0	8,760	100
Percent	[%]	13.4	6.3	13.7	10.8	11.7	10.8	11.1	10.0	10.6	1.5	0.0		

Table 2: Annual operating profile at different engine loads and seawater temperatures

temperature, and probably 3 to 5°C (on average 4°C) higher than the ambient deck air temperature. The ambient air intake temperature used in the calculations is the SW temperature +7°C (i.e., 3+4°C).

From Table 3, it can be observed that the annual HFOC is estimated to 30,746 t/yr. at the described annual operating profile.

Fuel oil consumption for electrical power production

To evaluate the electricity savings for pumps, fans, etc., and convert these to an annual HFOC saving, we have used the figures listed in Table 4. The $SFOC_{AE}$ figure for producing one electrical kWh is 216.8 g/kWeh. The assumptions used in the above HFOC calculation are:

- AE(s) are only operated on HFO with the LCV = 40,200 kJ/kg.
- The $SFOC_{AE}$ is not converted from ISO conditions ($SFOC_{AE\ ISO}$) to ambient conditions, except for the LCV.
- The $SFOC_{AE}$ has been obtained at 90% auxiliary engine load.

Annual HFO consumption

Ambient conditions				Engine load										
	[°C]	[°C]	[°C]	0	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
>30	32	36	39	0	55	237	378	867	549	663	777	537	82	0
26-30	28	32	35	0	165	631	691	1,037	1,313	1,453	1,394	1,785	203	0
22-26	24	28	31	0	137	472	501	689	873	1,054	927	1,424	203	0
18-22	20	24	27	0	137	314	499	601	652	788	924	1,419	202	0
14-18	16	20	23	0	54	117	124	171	434	523	614	530	202	0
8-14	11	15	18	0	27	78	124	85	216	391	458	528	201	0
<8	6	10	13	0	27	39	62	85	108	260	304	351	0	0

Annual HFO consumption: 30,746 t/yr., ref. LCV: 40,200 kJ/kg

Table 3: Annual ME HFOC [t/yr.]

HFO SFOC for electricity production

$SFOC_{AE\ ISO}$	Generator efficiency	Power net efficiency	LCV	$SFOC_{AE}$
[g/kWh]	[%]	[%]	[kg/kJ]	[g/kWeh]
190	95	98	40,200	216.8

Table 4: HFO SFOC for electricity production

2. Engine room ventilation system

Ventilation system description

The engine room ventilation system is designed to remove radiation and convection heat from the main engine, auxiliary engines, boilers, and other components and to provide sufficient air for combustion purposes for the main engine, auxiliary engines, and fuel oil fired boiler, etc.

Fig. 3 shows an example of an engine room ventilation system, where ventilation fans blow air into the engine room via air ducts.

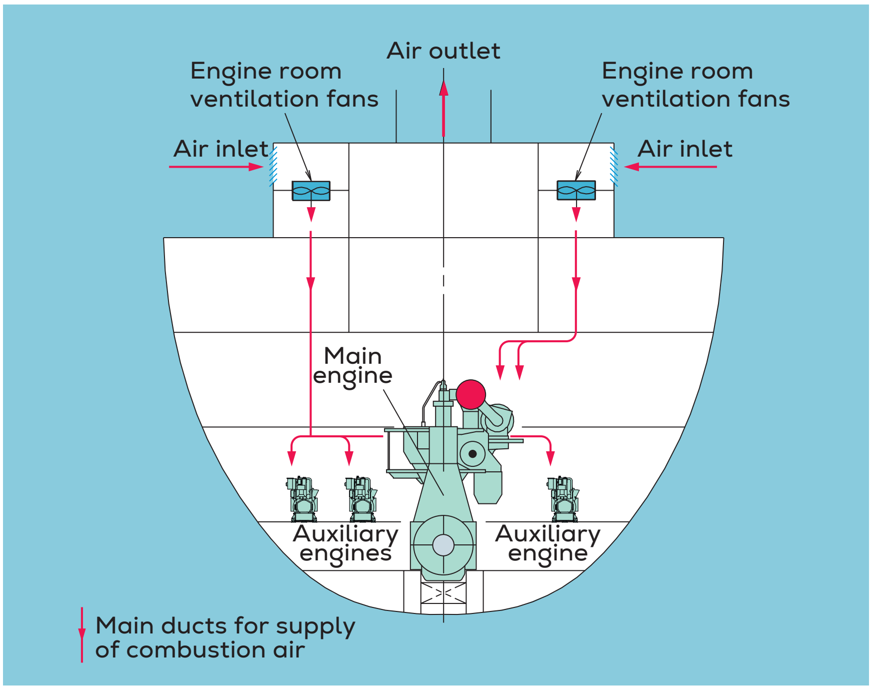


Fig. 3: General conventional ventilation system

Efficiency improvements to the ventilation system

When designing the ventilation system for the engine room, various solutions for efficiency improvements are available:

- Main engine direct air intake duct by means of a pipe duct, connecting the turbochargers with the outside.
- Automatic adjustment of the ventilation system based on the air demand of the ME.
- General ventilation design.

Main engine direct air intake
Introducing an air intake duct, connecting the turbochargers directly with the outside, will reduce the electrical power consumption of the ventilation system, as no ventilation fan is needed to supply the ME combustion air. It should be mentioned that the ventilation fan capacity still has to cover combustion air for AE(S) and the oil fired boiler, and ventilation air for removal of radiation and convection heat from the ME, AE(S), boilers, and other components. When designing such a direct air intake, special precautions must be taken. The most important design issues are the requirements to air filtration, turbocharger noise attenuation, and duct strength. For detailed design information, see our document No. 0787858-0, which is available on request.

When focusing only on ventilation air for ME combustion, it is possible to calculate the savings by removing the power needed for this purpose. The savings can be calculated using the figures in Table 5. The ME air consumption at 100% ME load is given in the CEAS document, and the pressure head and efficiency has been estimated as for a normal ventilation system installation.

Besides the power savings obtained by eliminating the fan capacity for ME combustion, the ME SFOC will also be improved by the slightly lower air intake temperature. As described in 'Annual ME fuel oil consumption' on page 6, the

Annual HFOC savings based on ME direct air intake

ME air consumption at 100% ME load, ISO		Pressure head	Fan efficiency	Motor efficiency	Ventilation fan power	Annual operation
[kg/s]	[m³/s]	[Pa]	[%]	[%]	[kWe]	[h/yr.]
119	105	700	75	85	116	7,585

Annual HFOC saving: 191 t/yr.

Table 5: Annual HFOC savings based on direct air intake and the $SFOC_{AE}$ = 216.8 g/kWeh

air temperature will be approx. 4°C higher than the outside deck temperature for a conventional air intake system. Table 6 shows the annual HFOC when making the calculation with a 4°C lower air intake temperature.

A comparison of the result in Table 6 and the result in section 'Annual ME fuel oil consumption', where the annual ME HFOC was calculated to 30,746 t/yr., gives a potential saving of 24 t/yr. Table 7 shows the overall HFOC savings potential of 215 t/yr. for the specific engine type by installing a direct air intake system.

Automatic adjustment of main engine combustion air fan
An alternative option to improving the efficiency is to install an automatically controlled fan motor equipped with a variable frequency drive (VFD) operated by a differential pressure. This option can be used when a conventional ventilation system is installed, i.e., the combustion air for the ME is supplied by a ventilation duct terminated close to the turbochargers. The differential pressure between the engine room and the outside atmospheric pressure is normally 5 mmWC (overpressure), which can be the operating parameter for the VFD. Assuming that the ME combustion air fan follows the air required by the turbocharger and that the fan has a minimum rpm level at 40% of maximum rpm, the fan

Annual HFO consumption, direct air intake and lower air intake temperature

Ambient conditions				Engine load										
Ambient SW temp		**Cooling water temp.**	**Ambient air temp**	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
	[°C]	**[°C]**	**[°C]**	**0**	**10%**	**15%**	**25%**	**35%**	**45%**	**55%**	**65%**	**75%**	**85%**	**95%**
>30	32	36	35	0	55	237	378	866	548	662	776	537	82	0
30>26	28	32	31	0	165	631	690	1,036	1,312	1,452	1,393	1,784	203	0
26>22	24	28	27	0	137	472	501	688	872	1,053	926	1,423	203	0
22>18	20	24	23	0	136	313	499	600	652	787	923	1,418	202	0
18>14	16	20	19	0	54	117	124	171	433	523	613	530	201	0
14>8	11	15	14	0	27	78	124	85	216	391	458	528	201	0
<8	6	10	9	0	27	39	62	85	107	259	304	351	0	0

Annual HFO consumption: 30,722 t/yr., ref. LCV: 40,200 kJ/kg

Table 6: Annual ME HFOC [t/yr.] based on direct air intake and 4°C lower air intake temperature

Annual savings based on ME direct air intake

	[t/yr.]
ME SFOC savings 30,746 t/yr. - 30,722 t/yr.	24
Savings by removing the power needed for the fan for ME combustion air	191
Total annual consumption	**215**

Table 7: Overall HFO savings potential for Everllence B&W 8G95ME-C9.5 Tier II by installing a direct air intake system

power savings can be determined based on the engine load (see Table 8). The overall HFOC savings potential from installing a VFD for the ventilation fan motor for ME combustion air is 163 t/yr.

General ventilation design

When the ventilation system is specified in a building specification and subsequently designed, several issues may be considered to make an energy efficient system.

Duct design (conventional system)

The mechanical power needed to supply the specified air capacity directly relates to the total duct friction loss according to Equation 1.

As the duct friction loss is related to the duct air velocity in second order, it is very important that the velocity is as low as possible. Assuming that a duct air velocity of 12 m/s has been used in the previous design, giving a total duct friction loss of 700 Pa. A new enlarged cross-sectional duct providing a duct air velocity of 10 m/s will result in a total duct friction loss of 486 Pa instead, using Equation 2.

Going from a duct design with a cross-sectional area corresponding to 12 m/s to a cross-sectional area corresponding to 10 m/s will reduce the fan related HFOC from 191 t/yr. to 132 t/yr. Table 9 shows HFOC savings of 59 t/yr.

Equation 1

$$Pi = dp \times q$$

where:

Pi = ideal power consumption (W)
dp = total pressure increase in the fan (Pa)
q = air volume flow delivered by the fan (m^3/s)

Equation 2

$$dp_2 = dp_1/(V_1/V_2)^2$$

where:

dp_1 = initial pressure loss at V_1 (Pa)
V_1 = initial velocity (m/s)
dp_2 = pressure loss at V_2 (Pa)
V_2 = new velocity based on new duct cross-sectional area (m/s)

Savings potential based on a VFD-operated fan for ME combustion air

Engine load	[%]	0	1-10	11-20	21-30	31-40	41-50	51-60	61-70	71-80	81-90	91-100
Engine load, used	[%]	0	10	15	25	35	45	55	65	75	85	95
ME air consumption, ISO	[kg/s]	0	19	28	44	51	64	70	87	98	107	115
ME air consumption at 100% ME load, ISO (Rho=1.13 kg/m³)	[m³/s]	0	17	25	39	45	56	62	77	86	94	102
Needed capacity	%	0	40	40	40	40	47	52	65	72	79	86
Initial fan power	[kWe]	0	116	116	116	116	116	116	116	116	116	116
Fan power, affinity law corrected	[kWe]	0	7	7	7	7	12	16	32	44	58	73
Fan power saving	[kWe]	0	109	109	109	109	104	100	84	72	58	43
ME annual operating hours	[h]	1,175	550	1,200	950	1,025	950	975	875	925	135	0
Annual power savings	[kWeh]	0	59,950	130,800	103,550	111,725	98,800	97,500	73,500	66,600	7,830	0

Annual heavy fuel oil savings: 163 t/yr.

Table 8: Savings potential based on $SFOC_{AE}$ = 216.8 g/kWeh and a VFD-operated fan for ME combustion air

Savings potential based on low duct air velocity

ME air consumption at 100% ME load, ISO		Duct velocity	Pressure head	Fan efficiency	Motor efficiency	Ventilation fan power	Annual operation	Annual HFOC
[kg/s]	[m³/s]	[m/s]	[Pa]	[%]	[%]	[kWe]	[h/yr.]	[t/yr.]
119	105	12	700	75	85	116	7,585	191
119	105	10	486	75	85	80	7,585	132

Annual HFOC savings: 59 t/yr.

Table 9: Savings potential for low duct air velocity based on $SFOC_{AE}$ = 216.8 g/kWeh

To improve the pressure loss, it is also necessary to look at design issues, such as:

- Bends: Wide radius bends are better than narrow radius bends
- Tees: Flow tees are better than 90 degree tees
- Reducers: Reducers are better than interrupt size changes
- Outlet: Should not be an interrupt outlet.

For further information about friction loss coefficients, consult general literature about the subject.

Fan design
The fan and motor efficiencies must be specified as high as possible when specifying/purchasing fans. A new type of ventilation fan has been developed with mechanical efficiencies and motor efficiencies of up to 90%. By specifying the use of the new type, it is possible to improve the HFOC related to the ventilation fan as follows in Table 10.

Closing remark

As can be observed, several efficiency improvements are present and large savings are obtainable. It should be mentioned again that the suggested solutions cannot be compared directly. As an example, the ME direct air intake will eliminate the ME combustion fan. Therefore, it does not make sense to discuss efficiency savings in terms of VFD, duct design, and fan efficiency.

As the ventilation system serves consumers other than the ME, it is possible to adopt some of the above recommended solutions for these systems and thereby increase the savings potential.

Savings potential based on a high-efficiency ventilation fan for ME combustion air

ME air consumption at 100% ME load, ISO		Pressure head	Fan efficiency	Motor efficiency	Ventilation fan power	Annual operation	Annual HFOC
[kg/s]	[m³/s]	[Pa]	[%]	[%]	[kWe]	[h/yr.]	[t/yr.]
119	105	700	75	85	116	7,585	191
119	105	700	90	90	91	7,585	150

Annual HFOC saving: 41 t/yr.

Table 10: Savings potential when specifying a high-efficiency ventilation fan for ME combustion air based on $SFOC_{AE}$ = 216.8 g/kWeh

3. Cooling water system

Description of the cooling water system

The cooling water system serving the ME is divided into two different systems:

- Low temperature (LT) cooling water system
- Jacket cooling water (JCW) system, also known as the high temperature (HT) cooling water system.

Low temperature cooling water system
The LT cooling water system supplies cooling water for the lubricating oil, jacket water, and scavenge air coolers. The LT cooling water system can be arranged in several configurations:

- Central cooling water system, the most common system choice and the basic execution for Everllence B&W engines
- Seawater (SW) cooling system, the most simple system
- Combined cooling water system with SW-cooled scavenge air cooler, but freshwater-cooled (FW-cooled) jacket water and lubricating oil cooler.

The following efficiency improvement proposals and calculations have been made for a central cooling water system. Fig. 4 shows a simplified version of the ME LT central cooling water system.

The central cooling water system is characterised by having only one heat exchanger cooled by seawater. The other coolers, including the jacket water cooler, are cooled by the central cooling water system.

Jacket cooling water system
The JCW system supplies cooling water to cylinder liners, cylinder covers, and exhaust gas valves of the ME and heats the fuel oil drain pipes. The JCW pump draws water from the jacket water cooler outlet through the deaerating tank and delivers it to the engine. A thermostatically controlled regulating valve is located at the inlet to the jacket water cooler, or alternatively at the outlet from the cooler (see Fig. 5). The regulating valve keeps the main engine cooling water outlet at a fixed temperature level independent of the engine load.

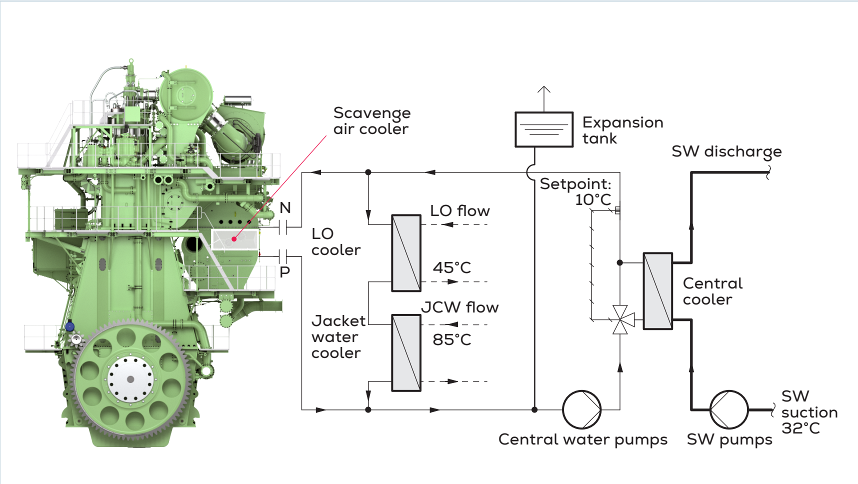


Fig. 4: Simplified ME LT central cooling water system. The required redundant pumps are not shown.

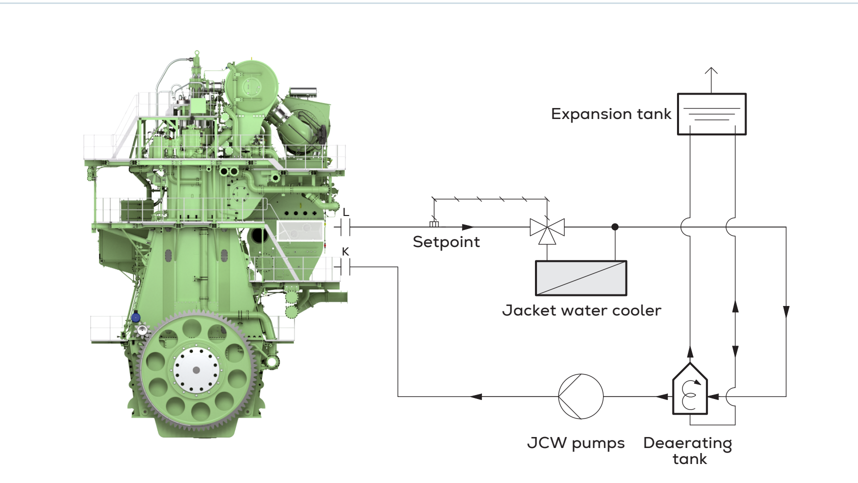


Fig. 5: Simplified ME jacket cooling water system. The required redundant pumps are not shown.

Capacities of cooling water systems

The CEAS capacities in Table 11 and Fig. 6 for the engine type 8G95ME-C9.5 Tier II have been used to calculate potential savings related to pump operation.

The figures in Table 11 are based on maximum ME load and a safety factor as well as tropical conditions. These figures should be used when designing the cooling system, i.e., the size of the coolers.

We have used pumping heads (Table 11), mechanical centrifugal pump efficiency (EFFm), EFFm = 75%, and an electric pump motor efficiency (EFFe) EFFe = 93.5% as the standard in the calculations.

- Jacket water pump: 380 m³/h at 3.0 bar – 45 kWe
- Central water pump: 1,270 m³/h at 2.5 bar – 126 kWe
- SW pump: 1,580 m³/h at 2.0 bar – 125 kWe

It is presumed that the pumps in the SW and FW systems are stopped during port stay. This is indicated as zero heat and flow at zero engine load in the following tables. This assumption is naturally only valid for a cooling water system serving only the main engine. The most commonly used cooling water system also covers other auxiliary systems, such as AEs, starting air compressors, etc. The pumps in this system can therefore not be stopped during port stays.

Data from CEAS, reproduced in Table 12, has been used for the calculation of heat dissipation as a function of ME load.

The heat dissipation is given as an ISO value, as the value varies slightly in the different conditions (tropical and low temperature conditions).

Pump	Flow capacity m³/h	Pump head Bar
Fuel oil circulation	24.1	6.0
Fuel oil supply	14.6	4.0
Jacket water	380	3.0
Central water	1,270	2.5
Sea water for central cooling	1,580	2.0
Lubrication oil	860	4.8

	Tier II		
Cooler	Flow m³/h	Central water flow m³/h	Heat dissipation kW
Scavenge air	-	740	20,620
Lubrication oil	860	530	4,270
Jacket water	380	530	7,330
Central water*)	1,580	1,270	32,220
Fuel oil circulation (MGO/MDO)	-	-	154

Table 11: Capacities obtained from CEAS

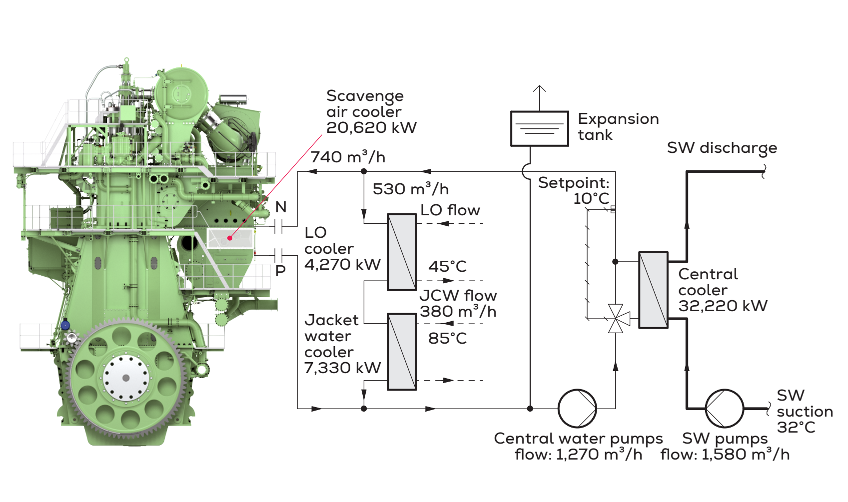


Fig. 6: Main engine LT central cooling system, including pump capacities and heat dissipations

Heat dissipation

	Engine load										
	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
Scavenge air cooler heat [kW]	0	820	1,230	2,060	3,430	5,650	8,130	10,760	13,900	16,300	19,160
Jacket water cooler heat [kW]	0	1,180	1,770	2,950	3,450	3,940	4,440	4,930	5,420	5,920	6,410
ME LO heat [kW]	0	870	1,310	2,180	2,560	2,880	3,150	3,370	3,560	3,710	3,830
ME total heat dissipation [kW]	0	2,870	4,310	7,190	9,440	12,470	15,720	19,060	22,880	25,930	29,400

Table 12: Heat dissipation as a function of ME load

Efficiency improvements to the low temperature system

The efficiency improvements to the LT system are described by evaluating the following suggestions for improvements:

- General design issues
- Variable flow for the SW pump, depending on ME load and ambient conditions (SW temperature)
- Variable flow for the SW and central cooling pumps, depending on ME load and ambient condition (SW temperature)

General design issues
When designing the LT system, the FW as well as the SW part, it is crucial that the cooling water system design and the specified components are based on the lowest possible pressure loss. The power supplied to the centrifugal pumps is computed by multiplying the pump flow capacity with the pumping head, including mechanical and electrical efficiencies (see Fig. 7). It is therefore important that:

- The pumping head is as low as possible
- The mechanical pump and electric motor efficiencies are as high as possible.

Pumping head
Some data may be extracted from CEAS and the project guide from Everllence, for example, pump pressure heads and pressure loss across coolers. The data is only to be used as a first estimate:

SW system differential pressure: 2.0 bar (central cooler, filter, pipe, and pipe components, etc.)

Central system differential pressure: 2.5 bar (central cooler, scavenge air cooler, pipe, and pipe components, etc.).

When ship designers make the detailed design of a system, all pressure loss figures for the components must be evaluated and challenged to ensure that the overall circulation pressure loss is as low as possible. By doing this, a smaller pumping head can thereby be specified compared with the pump specified by CEAS. The relevant design parameters are:

- Pipe system length/pipe diameter/wide radius bends/flow tees, etc.
- Lower differential pressure across the central cooler
- Low differential pressure across the valve components, especially the three-way regulating valve and SW filters.

By specifying and designing according to the parameters above, our best guess is that the system pressure loss can be lowered and the pumping head consequently reduced by approx. 20%:

- SW system differential pressure: 1.6 bar
- Central system differential pressure: 2.0 bar

A 20% decrease in pumping head results directly in a 20% annual HFO saving as indicated in Table 13, which shows the annual HFO consumption related to pump operation in the central cooling water system.

SW and central cooling water pumps – annual HFO consumption

	[t/yr.]
SW pump, head=2.0 bar: 125 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	205.5
SW pump, head=1.6 bar: 100 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	164.4
Annual savings by decreasing the SW system pressure loss by 20%	41
Central water pump, head=2.5 bar: 126 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	207.2
Central water pump, head=2 bar: 101 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	166.1
Annual savings by decreasing the central water system pressure loss by 20%	41
Total savings	**82**

Table 13: Cooling water pumps – annual HFO savings based on a lower differential pressure

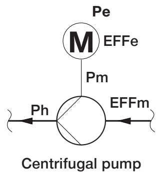
Pump efficiencies

As Fig. 7 shows, the mechanical efficiency (EFFm) is the relationship between the power supplied to the pumped fluid (Ph) and the power supplied to the pump shaft (Pm). The electric pump motor efficiency is the relation between the mechanical power supplied to the shaft (Pm) and the electrical power supplied to the electric motor (Pe).

Pump mechanical efficiency

The pump efficiency is provided by the centrifugal pump supplier. It varies depending on where the pump is working on the flow versus pressure (Q-H) curve. Fig. 8 shows examples of Q-H and Q-EFFm curves from a pump supplier. As can be observed, it is very important that the pump operates where the EFFm is at the highest possible level. This means that when the pipe system designer has determined the pipe system pressure, and therefore the pumping head (also called the specified nominal duty point), the pump supplier must provide a pump with the highest possible EFFm at the required duty point.

As an example, Fig. 9 shows Q-H and Q-EFFm curves from a pump supplier for a given pump type. Based on the pipe design and the specified nominal duty point: 1,580 m^3/h at 2.0 bar, the mechanical efficiency for this pump type can be read as EFFm = 75%.
In the past, pump suppliers did not always have a pump type, which fitted the purpose. Therefore, pump types with a very low EFFm have been installed. Examples have been seen where the maximum mechanical efficiency was 50%. Normally pump suppliers with a wide range of pump sizes are able to comply with high



Equation 3

Hydraulic power: $Ph = QH \frac{10^5}{3600}$ (kW)

Where: Q is the flow in m^3/h and H the pumping head in bar

Mechanical power: $Pm = \frac{Ph}{EFFm}$ (kW)

Where: EFFm is the mechanical efficiency

Electrical power: $Pe = \frac{Pm}{EFFe}$ (kWe)

Where: EFFe is the electric motor efficiency

Fig. 7: General pump power theory

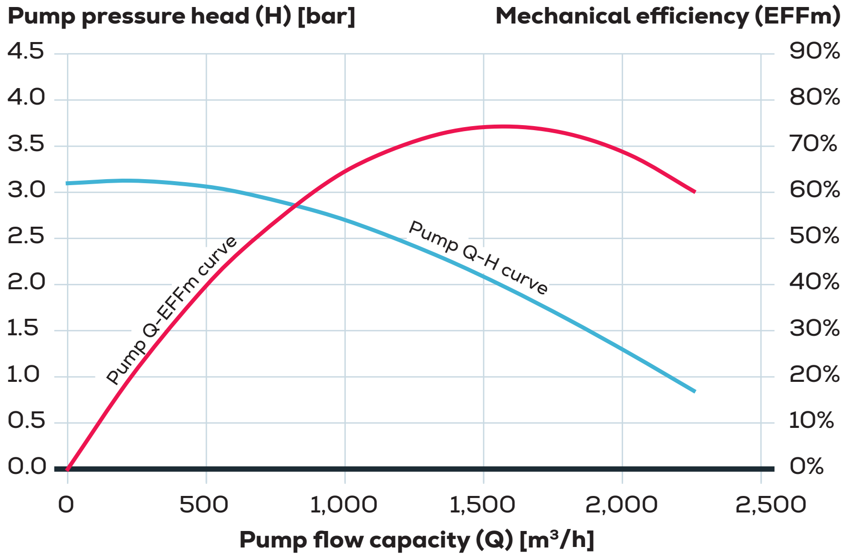


Fig. 8: Q-H and Q-EFFm curves for a centrifugal pump

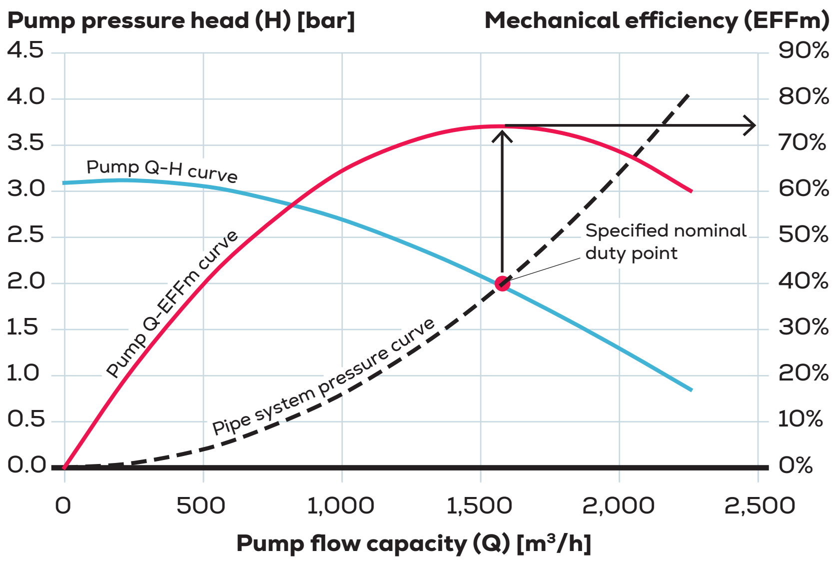


Fig. 9: Q-H and Q-EFFm curves for a centrifugal pump, pipe system pressure curve and duty point

requirements to EFFm and are even able to offer pumps with EFFm above 80%.

Table 14 gives annual savings obtained by specifying a cooling water pump with 5% higher EFFm.

Electric motor efficiency

It is important to make high demands on electric pump motor efficiency, as this is not always a matter of course. Savings can be obtained by specifying, for example, in the ship building contract that the electric motor efficiencies must be the highest possible.

The International Electrotechnical Commission (IEC) has issued an energy efficiency standard (IEC 60034-30), the so-called IE-code, classifying rotating electric machines. This code classifies how efficiently the machines convert electricity into mechanical energy.

The code is currently divided into four levels:

IE1: Base standard for efficiency
IE2: High efficiency
IE3: Premium efficiency
IE4: Super premium efficiency (not yet implemented)

The European Union (EU) has adopted the above IEC standard and issued a regulation forcing the EU industry to install electric rotating machines with at least class IE3 (or IE2 combined with variable speed drive) from January 2015. The EU regulation, reference is made to the EU Minimum Energy Performance Standard (MEPS), covers two-, four- and six-pole motors in the power range from 0.75 to 375 kW for a 50/60 Hz AC power supply. Other nations outside EU also comply with the IE classes, but the year when IE3 becomes mandatory differs.

As national regulations and thereby EU regulations do not cover vessels operating in international waters, the IE3 class can be specified from building contract to building contract to ensure that the electric motors are delivered with the highest possible efficiency, as this is not, as mentioned, a matter of course. The differences in nominal efficiencies according to power and IE class are listed in Table 15 and depicted in Fig. 10 for a 60 Hz, four-pole electric motor. Fig. 10 shows that the nominal efficiency difference between IE1 and IE3 is much larger at smaller motor output powers than higher motor output powers. Therefore, efficiency gains are greater at smaller motor powers.

By looking at the given pump sizes for the SW and central water systems, it can be seen

SW and central cooling water pumps – annual HFO consumption

	[t/yr.]
SW pump, EFFm=75%: 125 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	205.5
SW pump, EFFm=80%: 117 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	192.4
Annual savings by increasing the SW pump EFFm by 5%	**13**
Central water pump, EFFm=75%: 126 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	207.2
Central water pump, EFFm=80%: 118 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	194.0
Annual savings by increasing the central water pump EFFm by 5%	**13**
Total savings	**26**

Table 14: Cooling water pumps – annual HFO consumption savings based on 5% higher mechanical efficiency

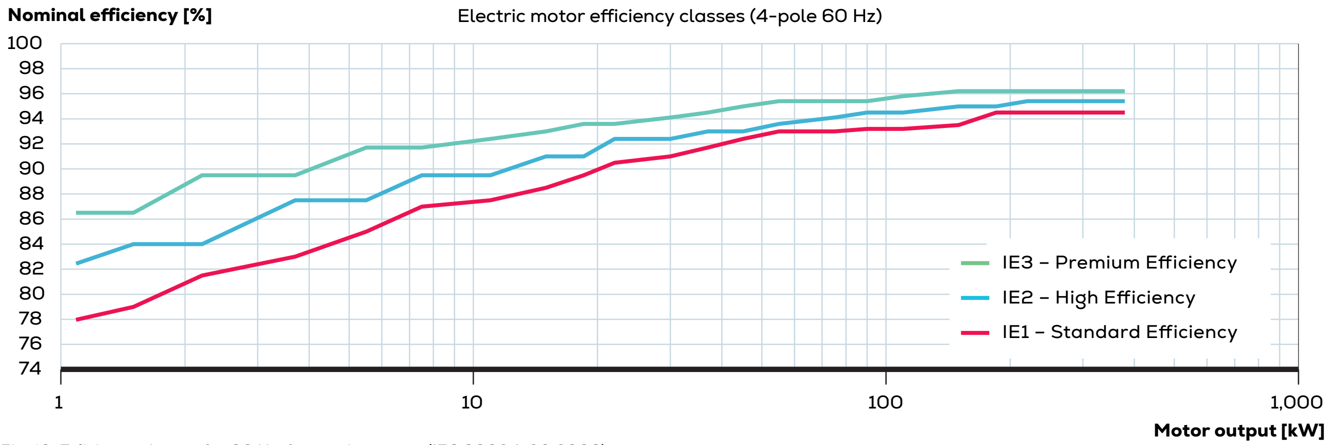


Fig. 10: Efficiency classes for 60 Hz, four-pole motors (IEC 60034-30:2008)

that a gain can be obtained by changing the class. At the chosen baseline IE1, the nominal efficiency is EFFe = 93.5% (60 Hz, four-pole motor, approx. 110 kWe) and at IE3, EFFe = 95.8%. Table 16 shows the calculated annual savings in HFO consumption by changing the SW and the central water pumps to the IE3 standard.

60 Hz

Power	IE1 – Standard Efficiency			IE2 – High Efficiency			IE3 – Premium Efficiency		
	2-pole	4-pole	6-pole	2-pole	4-pole	6-pole	2-pole	4-pole	6-pole
[kW]	[%]	[%]	[%]	[%]	[%]	[%]	[%]	[%]	[%]
0.8	77.0	78.0	73.0	75.5	82.5	80.0	77.0	85.5	82.5
1.1	78.5	79.0	75.0	82.5	84.0	85.5	84.0	86.5	87.5
1.5	81.0	81.5	77.0	84.0	84.0	86.5	85.5	86.5	88.5
2.2	81.5	83.0	78.5	85.5	87.5	87.5	86.5	89.5	89.5
3.7	84.5	85.0	83.5	87.5	87.5	87.5	88.5	89.5	89.5
5.5	86.0	87.0	85.0	88.5	89.5	89.5	89.5	91.7	91.0
7.5	87.5	87.5	86.0	89.5	89.5	89.5	90.2	91.7	91.0
11.0	87.5	88.5	89.0	90.2	91.0	90.2	91.0	92.4	91.7
15.0	88.5	89.5	89.5	90.2	91.0	90.2	91.0	93.0	91.7
18.5	89.5	90.5	90.2	91.0	92.4	91.7	91.7	93.6	93.0
22.0	89.5	91.0	91.0	91.0	92.4	91.7	91.7	93.6	93.0
30.0	90.2	91.7	91.7	91.7	93.0	93.0	92.4	94.1	94.1
37.0	91.5	92.4	91.7	92.4	93.0	93.0	93.0	94.5	94.1
45.0	91.7	93.0	91.7	93.0	93.6	93.6	93.6	95.0	94.5
55.0	92.4	93.0	92.1	93.0	94.1	93.6	93.6	95.4	94.5
75.0	93.0	93.2	93.0	93.6	94.5	94.1	94.1	95.4	95.0
90.0	93.0	93.2	93.0	94.5	94.5	94.1	95.0	95.4	95.0
110.0	93.0	93.5	94.1	94.5	95.0	95.0	95.0	95.8	95.8
150.0	94.1	94.5	94.1	95.0	95.0	95.0	95.4	96.2	95.8
185.0	94.1	94.5	94.1	95.4	95.4	95.0	95.8	96.2	95.8
220.0	94.1	94.5	94.1	95.4	95.4	95.0	95.8	96.2	95.8
250.0	94.1	94.5	94.1	95.4	95.4	95.0	95.8	96.2	95.8
300.0	94.1	94.5	94.1	95.4	95.4	95.0	95.8	96.2	95.8
330.0	94.1	94.5	94.1	95.4	95.4	95.0	95.8	96.2	95.8
375.0	94.1	94.5	94.1	95.4	95.4	95.0	95.8	96.2	95.8

Table 15: Efficiency classes: IEC 60034-30 (2008), 60 Hz

SW and central cooling water pumps – annual HFO consumption

	[t/yr.]
SW pump, EFFe=93.5%: 125 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	205.5
SW pump, EFFe=95.8%: 122 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	200.6
Annual savings by changing the SW pump EFFe from IE1 to IE3 standard	5
Central water pump, EFFe=93.5%: 126 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	207.2
Central water pump, EFFe=95,8%: 123 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	202.2
Annual savings by changing the central water pump EFFe from IE1 to IE3 standard	5
Total savings	**10**

Table 16: Cooling water pumps – annual HFO consumption savings based on 2.3% higher electric motor efficiency

VFD-operated seawater pump

There is a possibility to vary the SW pump flow based on ME heat dissipation values as a function of engine load and ambient conditions. Table 12 gives ME heat dissipation values. The SW pump flow to the central cooler can be decreased while maintaining the following:

- The cooling water temperatures for the ME scavenge air cooler, lubricating oil (LO) cooler, and jacket water cooler follow the SW temperature + 4°C, as the three-way valve setpoint is 10°C (see Fig. 11).
- The central water flow is kept constant.

For the SW pump to deliver a variable flow, it is necessary to install a VFD for the pump motor. This VFD must be operated automatically to keep the 4°C difference between the SW and FW side of the central cooler. If the SW water temperature drops below 6°C, the three-way valve must start mixing with FW, so a minimum temperature of 10°C is maintained. An alternative to the three-way valve is to install a manual bypass valve, which must only be operated during cold climate operation. By using a manual bypass valve, the total pressure loss in the system will be lowered and thereby also the required pump power.

When operating a specific centrifugal pump with a VFD, and in that way reducing pump revolutions, the flow, pumping head, and pump power will follow the pump affinity law in Equation 4.

Taking the SW pump as an example, when reducing rpm by 50%, power is reduced from 125 kWe to 15.6 kWe as the flow decreases from 1,580 m³/h to 790 m³/h (see Fig. 12).

The SW flow required to remove the total dissipated heat at a specific engine load (see Table 17) is calculated by running a performance calculation for the central cooler for each engine load and SW temperature.

The required pump power can be determined by using

Equation 4

Flow changes: $Q_1/Q_2 = (RPM_1/RPM_2) \Leftrightarrow Q_2 = Q_1/(RPM_1/RPM_2)$

Pump head: $H_1/H_2 = (RPM_1/RPM_2)^2 \Leftrightarrow H_2 = H_1/(RPM_1/RPM_2)^2$

Pump power: $P_1/P_2 = (RPM_1/RPM_2)^3 \Leftrightarrow P_2 = P_1/(RPM_1/RPM_2)^3$

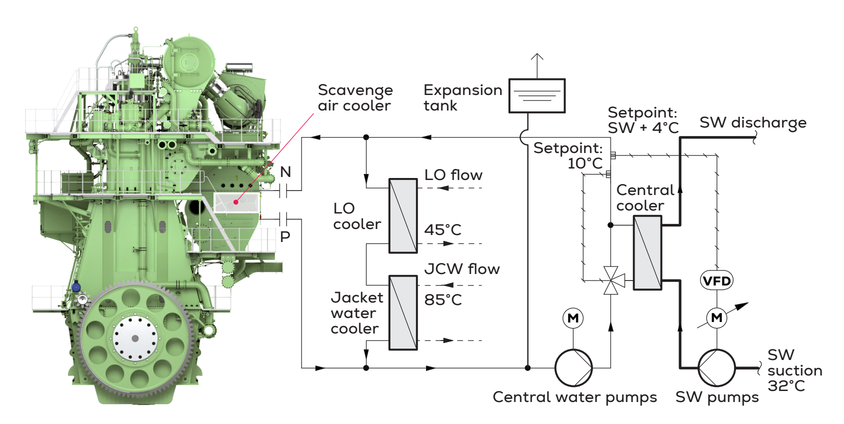


Fig. 11: VFD-operated SW system

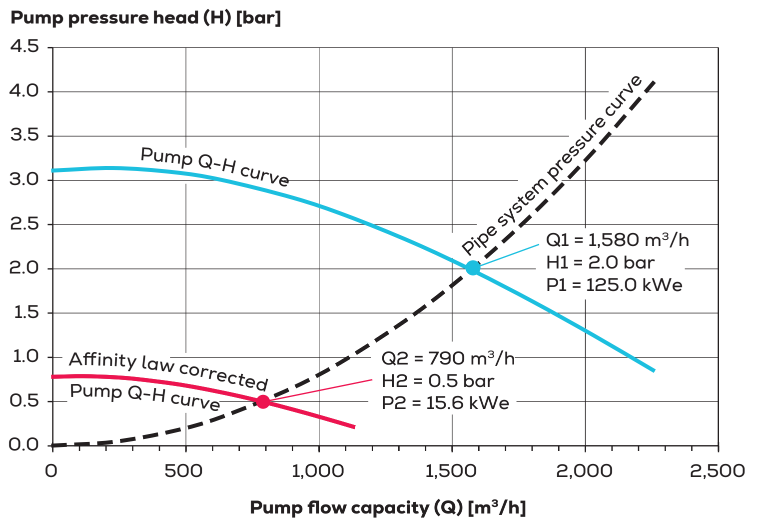


Fig. 12: The affinity law corrected SW pump Q-H curve when reducing the rpm by 50%

the pump affinity law and the pump flows in Table 17.

The calculated pump power values have been multiplied by the operating hours in Table 2 and the specific HFOC electricity production figure to get the annual HFO consumption related to operation of the SW pump (Table 18).

Table 19 gives the savings obtained by VFD-operated SW pumps.

Installing a VFD for the SW pump requires that the pump is operated at 100% rpm in small time sequences during the day to avoid cooler scaling. These 100% rpm sequences are not included in the calculation.

Required SW cooling water flow at fixed FW cooling water flow

Ambient conditions				Engine load										
Ambient SW temp.		Cooling water temp.	Ambient air temp.	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
[°C]		[°C]	[°C]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
>30	32	36	39	0	426	559	757	874	1,006	1,123	1,224	1,326	1,397	1,470
26-30	28	32	35	0	426	561	762	881	1,017	1,138	1,244	1,349	1,423	1,499
22-26	24	28	35	0	427	561	768	892	1,030	1,155	1,265	1,374	1,452	1,531
18-22	20	24	27	0	428	562	775	902	1,046	1,175	1,290	1,403	1,484	1,580
14-18	16	20	23	0	428	563	780	914	1,063	1,198	1,318	1,437	1,521	1,580
8-14	11	15	18	0	428	565	785	932	1,089	1,232	1,359	1,488	1,579	1,580
<8	6	10	13	0	429	567	791	944	1,120	1,274	1,411	1,549	1,580	1,580

Table 17: Required SW cooling water flow [m^3/h] at a fixed central cooling water flow

Annual HFO consumption for VFD-operated SW pumps (min. circulation rate: 40%)

Ambient conditions				Engine load										
Ambient SW temp.		Cooling water temp.	Ambient air temp.	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
[°C]		[°C]	[°C]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
>30	32	36	39	0	87	260	447	1,147	874	1,216	1,575	1,201	187	0
26-30	28	32	35	0	260	694	836	1,409	2,168	2,784	2,976	4,216	495	0
22-26	24	28	31	0	217	520	622	975	1,501	2,117	2,086	3,564	526	0
18-22	20	24	27	0	217	347	640	882	1,179	1,672	2,212	3,794	561	0
14-18	16	20	23	0	87	130	163	262	825	1,181	1,573	1,529	604	0
8-14	11	15	18	0	43	87	166	139	444	963	1,293	1,698	676	0
<8	6	10	13	0	43	43	85	144	241	710	965	1,277	0	0

Annual HFO consumption: 61.8 t/yr., ref. LCV: 40,200 kJ/kg

Table 18: Annual HFO consumption [kg/yr.] for VFD-operated SW pumps, based on $SFOC_{AE}$ = 216.8 g/kWeh

SW cooling pumps – annual HFO consumption

	[t/yr.]
SW pump with fixed rpm: 125 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	205.5
SW pump with VFD	61.8
Annual savings by using VFD for the SW pumps	144

Table 19: Obtained HFO savings for VFD-operated SW pumps

VFD-operated seawater and central cooling water pumps

As discussed in the previous section, huge savings can be obtained by using a VFD for the SW pump. Therefore, it is obvious that savings are also available, if it is possible to reduce the freshwater central cooling water flow by using a VFD for this pump. Unfortunately, this is not as simple as for the SW pump. The reason is that Everllence does not recommend decreasing the water amount to the scavenge air cooler as a function of the engine load and the ambient conditions. When reducing the water flow, the following implications are foreseen:

- Increased scavenge air temperature, which will lead to a reduction of the water condensation amount in certain ambient conditions. Reduced water condensation will lead to a higher humidity level in the combustion chamber, which will have a negative effect on the general cylinder condition and thereby increase cylinder wear.

- Increased scavenge air temperature, which will lead to increased fuel oil consumption.

- A too low flow may cause local boiling on the water side inside the scavenge air cooler, which may lead to cavitation in the cooling water pipes.

The stated max. scavenge air temperature values given in the technical file may be exceeded, and thereby the engine may not fulfil Tier II NO_X requirements.

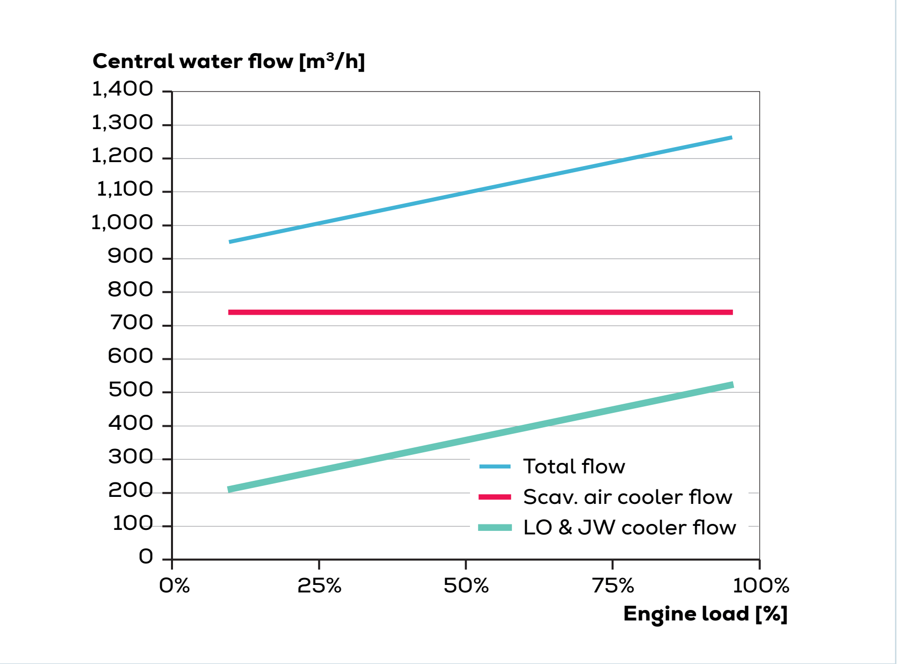


Fig. 13: Defined flow capacity for the central cooling water pump versus engine load

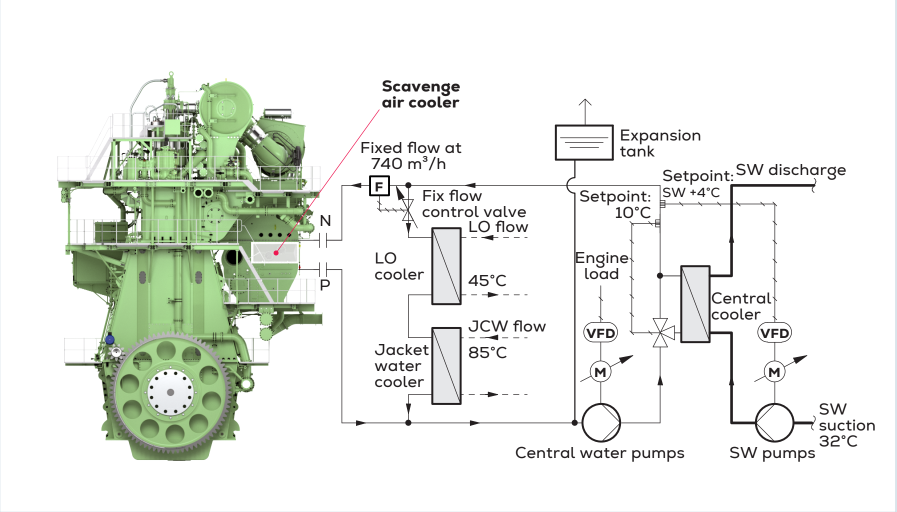


Fig. 14: VFD-operated SW pump and VFD-operated (vs engine load) central cooling water pump

On this basis, a system has to be established that maintains a constant cooling water flow to the scavenge air cooler and, at the same time, makes it possible to reduce the flow to LO and jacket water coolers (see Fig. 13).

The total flow versus engine load can be obtained by taking the approach that the water flow to LO and jacket water coolers can be determined as a function of the engine load based on the heat dissipation in the coolers, and by keeping a constant temperature difference across the coolers.

Fig. 14 shows an example of a piping diagram for the above system, including VFD operation of the SW pump.

By running a performance calculation for the central cooler for each engine load and SW temperature, the SW flow required to remove the total heat amount at the specific engine load can be calculated (see Table 20).

Using the pump affinity laws and the pump flows in Table 20, the required pump power can be determined for both the SW pump and the central cooling water pump. These pump powers have been multiplied by the operating hours

Required SW cooling water flow at different engine loads and a variable central cooling water flow

Ambient conditions				Engine load										
Ambient SW temp.		Cooling water temp.	Ambient air temp.	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
[°C]		[°C]	[°C]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
›30	32	36	39	0	366	478	668	783	913	1,036	1,150	1,268	1,361	1,458
26-30	28	32	35	0	366	479	673	791	924	1,050	1,168	1,290	1,385	1,486
22-26	24	28	35	0	367	480	679	800	936	1,067	1,180	1,314	1,414	1,518
18-22	20	24	27	0	368	481	682	810	951	1,086	1,212	1,343	1,446	1,555
14-18	16	20	23	0	369	483	685	821	968	1,108	1,239	1,375	1,483	1,580
8-14	11	15	18	0	370	485	691	831	992	1,140	1,279	1,424	1,538	1,580
‹8	6	10	13	0	371	486	698	842	1,020	1,180	1,327	1,484	1,580	1,580

Central cooling water flow according to a predefined flow vs engine load

Ambient conditions				Engine load										
Ambient SW temp.		Cooling water temp.	Ambient air temp.	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
[°C]		[°C]	[°C]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
›30	32	36	39	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255
26-30	28	32	35	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255
22-26	24	28	35	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255
18-22	20	24	27	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255
14-18	16	20	23	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255
8-14	11	15	18	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255
‹8	6	10	13	0	843	895	998	1,042	1,083	1,122	1,157	1,192	1,224	1,255

Table 20: Required SW and central cooling water flow [m^3/h]

from the annual operating profile and the specific HFO electricity production figure to calculate the HFO consumption in Table 21 for VFD-operated SW and central cooling water pump.

The total savings obtained when using a VFD for the SW and the central cooling water pumps are estimated to 241 t HFO/year, equal to 58% of the initial full consumption of 422 t/yr. (see Table 22).

Annual HFO consumption for VFD-operated SW pumps (min. circulation rate: 40%)

Ambient conditions				Engine load										
Ambient SW temp.		**Cooling water temp.**	**Ambient air temp.**	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
[°C]		**[°C]**	**[°C]**	**0%**	**10%**	**15%**	**25%**	**35%**	**45%**	**55%**	**65%**	**75%**	**85%**	**95%**
>30	32	36	39	0	87	260	307	824	654	955	1,306	1,050	173	0
26-30	28	32	35	0	260	694	576	1,020	1,626	2,187	2,463	3,687	456	0
22-26	24	28	31	0	217	520	430	703	1,127	1,669	1,693	3,117	486	0
18-22	20	24	27	0	217	347	436	639	886	1,320	1,835	3,328	519	0
14-18	16	20	23	0	87	130	110	190	623	934	1,307	1,339	560	0
8-14	11	15	18	0	43	87	113	99	335	763	1,078	1,488	625	0
<8	6	10	13	0	43	43	58	103	182	564	803	1,123	0	0

Annual HFO consumption: 50.9 t/yr., ref. LCV: 40,200 kJ/kg

Annual HFO consumption for VFD-operated central water pumps

Ambient conditions				Engine load										
Ambient SW temp.		**Cooling water temp.**	**Ambient air temp.**	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
[°C]		**[°C]**	**[°C]**	**0%**	**10%**	**15%**	**25%**	**35%**	**45%**	**55%**	**65%**	**75%**	**85%**	**95%**
>30	32	36	39	0	400	1,433	1,988	3,774	2,117	2,352	2,584	1,692	245	0
26-30	28	32	35	0	1,199	3,822	3,644	4,528	5,080	5,175	4,651	5,640	612	0
22-26	24	28	31	0	999	2,867	2,651	3,019	3,387	3,763	3,101	4,512	612	0
18-22	20	24	27	0	999	1,911	2,651	2,642	2,540	2,823	3,101	4,512	612	0
14-18	16	20	23	0	400	717	663	755	1,693	1,882	2,067	1,692	612	0
8-14	11	15	18	0	200	478	663	377	847	1,411	1,550	1,692	612	0
<8	6	10	13	0	200	239	331	377	423	941	1,034	1,128	0	0

Annual HFO consumption: 120.6 t/yr., ref. LCV: 40,200 kJ/kg

Table 21: Obtained HFO savings for VFD-operated SW and central cooling water pumps

SW and central cooling water pumps – annual HFO consumption

	[t/yr.]
SW pump with fixed rpm: 125 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	205.5
SW pump with VFD	50.9
Annual savings by using VFD for SW pumps	154.6
Central water pump with fixed rpm: 126 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	207.2
Central water pump with VFD	120.6
Annual savings by using VFD for the central water pump	86.6
Total savings	241

Table 22: Obtained HFO savings for VFD-operated SW and central cooling water pumps

Efficiency improvements to the jacket cooling water system

It is not possible to decrease the flow by using a VFD for the JCW pump during engine low load operation, because:

- By decreasing the flow there is a risk that the cooling water flow will be unevenly distributed between the cylinders and also that the cooling water is not distributed to all the cooling bores in the cylinder liner and cover.
- By reducing the flow, the JCW pumping head will also decrease according to the pump affinity law. Everllence requires an inlet pressure in the range of 3.7-5.0 barg.

As it is not possible to reduce the flow during engine low load operation, it is of utmost importance that the design of the JCW pipe system and the specified components has the lowest possible pressure loss and that the pump and pump motor are specified with the highest possible mechanical and electrical efficiencies.

Pumping head

Some data may be extracted from CEAS and the project guide from Everllence, for example, pumping heads and pressure loss across coolers. The data must only be used as a first estimate. The data given for the JCW system is 3.0 bar, which covers the pressure loss across the engine, the jacket water cooler (JWC) three-way valve, the JWC, the pressure loss in the pipe system, and pipe components.

When ship designers make the detailed design of the system, all actual pressure loss figures for the components must be evaluated and challenged to reduce the overall circulation pressure loss and to be able to specify a smaller pumping head. The relevant items are:

- Pipe system length/pipe diameter/wide radius bends/flow tees, etc.
- Lower differential pressure across the JWC
- Low differential pressure across the valve components, especially the JWC three-way regulating valve.

It has to be emphasised that the pumping head should not be used to obtain the min. inlet pressure to the engine required by Everllence by using an orifice at the outlet. The inlet pressure to the engine must be determined at the design stage by evaluating the static pressure created by the expansion tank location above the inlet and the hydrostatic pressure created by the JCW pump.

By specifying and designing according to the above parameters, it is our best guess that it is possible to lower the overall system pressure loss and thereby reduce the pumping head by approx. 10% to 2.7 bar.

Pump efficiencies

By having high requirements to mechanical and electric motor efficiencies, it is possible to lower the power consumption for the JCW pump. For a detailed explanation of the mechanical and electric motor efficiencies, see pages 16-18 and Fig. 7. By considering the following items for the JCW system, it is possible to calculate the potential savings.

- Pipe system pressure loss: 2.7 bar
- Mechanical efficiency, EFFm: 80% instead of baseline efficiency of 75%
- Motor efficiency class: IE3 instead of baseline class IE1
- The pump is only running during main engine (ME) operation

The potential savings are entered in Table 23.

Jacket cooling water pump – annual HFO consumption

	[t/yr.]
JCW pump, pumping head: 3 bar, EFFm: 75%, EFFe: 93%: 45 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	74.0
JCW pump, pumping head: 2.7 bar, EFFm: 80%, EFFe: 95%: 38 kWe × 7,585 h × 216,8 g/kWeh × 10^{-6}	62.5
Total savings	12

Table 23: Jacket cooling water pump – annual savings in HFO consumption

Closing remark
As it can be seen, the savings potential for the cooling system is big. A saving of 61%, which is equal to 299 t HFO/year, is obtainable by combining the following.

- Operate the SW pump and the central cooling water pump with VFDs as described on page 20.
- Use the optimised system pressure loss:
- SW pumping head: 1.6 bar (baseline 2 bar)
- Central cooling water pumping head: 2.0 bar (baseline 2.5 bar)
- JCW pumping head: 2.7 bar (baseline 3.0 bar)
- Use the mechanical efficiency EFFm = 80% instead of the baseline efficiency 75%
- Use the electric motor efficiency class IE3 instead of baseline class IE1.

Tables 24 and 25 provide the calculation of annual HFO consumption and total savings related to optimisation of the cooling water system.

Annual HFO consumption for VFD-operated SW pumps (min. circulation rate: 40%)

Ambient conditions				Engine load										
Ambient SW temp.		**Cooling water temp.**	**Ambient air temp.**	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
[°C]		**[°C]**	**[°C]**	**0%**	**10%**	**15%**	**25%**	**35%**	**45%**	**55%**	**65%**	**75%**	**85%**	**95%**
›30	32	36	39	0	64	191	226	607	481	703	961	773	127	0
26-30	28	32	35	0	191	511	424	751	1,197	1,610	1,813	2,714	336	0
22-26	24	28	31	0	160	383	317	518	829	1,228	1,246	2,294	357	0
18-22	20	24	27	0	160	255	321	470	652	971	1,350	2,450	382	0
14-18	16	20	23	0	64	96	81	140	459	688	962	986	412	0
8-14	11	15	18	0	32	64	83	73	247	562	793	1,095	460	0
‹8	6	10	13	0	32	32	43	75	134	415	591	826	0	0

Annual HFO consumption: 37.5 t/yr., ref. LCV: 40,200 kJ/kg

Annual HFO consumption for VFD-operated SW pumps

Ambient conditions				Engine load										
Ambient SW temp.		**Cooling water temp.**	**Ambient air temp.**	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
[°C]		**[°C]**	**[°C]**	**0%**	**10%**	**15%**	**25%**	**35%**	**45%**	**55%**	**65%**	**75%**	**85%**	**95%**
›30	32	36	39	0	292	1,047	1,451	2,755	1,546	1,717	1,887	1,235	179	0
26-30	28	32	35	0	875	2,791	2,661	3,307	3,710	3,778	3,396	4,118	447	0
22-26	24	28	31	0	729	2,093	1,935	2,204	2,473	2,748	2,264	3,294	447	0
18-22	20	24	27	0	729	1,395	1,935	1,929	1,855	2,061	2,264	3,294	447	0
14-18	16	20	23	0	292	523	484	551	1,237	1,374	1,509	1,235	447	0
8-14	11	15	18	0	146	349	484	276	618	1,030	1,132	1,235	447	0
‹8	6	10	13	0	146	174	242	276	309	687	755	824	0	0

Annual HFO consumption: 88.1 t/yr., ref. LCV: 40,200 kJ/kg

Table 24: Annual HFO consumption for VFD-operated SW and central cooling water pumps based on $SFOC_{AE}$ = 216.8 g/kWeh

Cooling water pumps – annual HFO consumption

	[t/yr.]
SW pump, fixed rpm, head=2.0 bar, EFFm=75%, EFFe=93.5%: 125 kWe × 7,585 h × 216.8 g/kWeh × 10^{-6}	205.5
Central water pump, fixed rpm, head=2.5 bar, EFFm=75%, EFFe=93.5%: 126 kWe × 7,585 h × 216.8 g/kWeh × 10^{-6}	207.2
JCW pump, fixed rpm, head=3.0 bar, EFFm=75%, EFFe=93.0%: 45 kWe × 7,585 h × 216.8 g/kWeh × 10^{-6}	74.0
Annual consumption	486.7
SW pump, VFD rpm, head=1.6 bar, EFFm=80%, EFFe=95.8%	37.5
Central water pump, VFD rpm, head=2.0 bar, EFFm=80%, EFFe=95.8%	88.1
JCW pump, fixed rpm, head=2.7 bar, EFFm=80%, EFFe=95.0%: 38 kWe × 7,585 h × 216.8 g/kWeh × 10^{-6}	62.5
Annual consumption	188.1
Total annual savings	**299**

Table 25: Overall cooling water-related annual savings in HFO consumption

4. Fuel oil system

Description of the fuel oil system

The recommended conventional fuel oil (FO) system is divided into a supply system and a circulation system. From the service tank, the supply pumps supply an amount of fuel to the circulation system equal to the ME fuel consumption. The remaining supply pump flow capacity is bypassed to the suction side again through the 4 barg self-acting pressure setting valve. The capacity of the supply pumps is based on 110% ME FO consumption, including circulation rate and safety factor.

The circulation circuit circulates fuel oil through the heater, filter, engine, venting tank, and back again to the suction side of the circulation pumps. At the engine, a self-acting pressure setting valve is installed for maintaining a constant inlet pressure independent of the ME FO consumption. The capacity of the circulation pumps is based on 110% ME FO consumption, including the circulation rate and safety factor.

Fig. 15 shows the recommended ME FO system. It should be mentioned that the arrangement of the FO system may differ. The automatic filter may, for example, be installed on the supply side, including a duplex safety filter in the circulation circuit, etc.

Efficiency improvements to main engine fuel oil systems

Efficiency improvements are also available to the FO system – to both the supply system and the circulation system – these are:

- Variable flow for the supply pump, depending on the ME FO consumption.
- 100% or 50% flow for the circulation pump, depending on whether the ME is running or not.
- Variable flow for the circulation pump, depending on the ME FO consumption combined with a sufficient circulation rate.
- The pump flow capacities for the engine type 8G95ME-C9.5 Tier II have been used for the savings calculations – these are:

Supply pump: 14.6 m³/h (estimated power consumption: 4 kWe)

Circulation pump: 24.1 m³/h (estimated power consumption: 10 kWe)

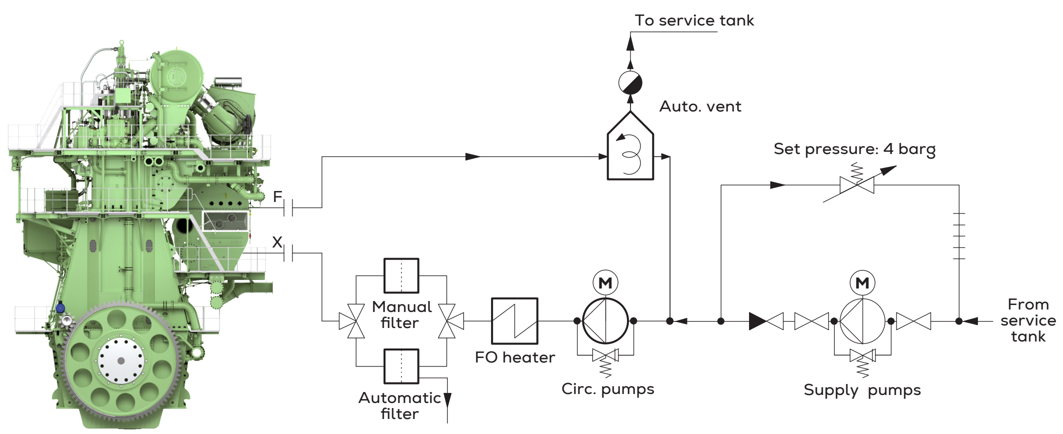


Fig. 15: Simplified main engine FO system – the required redundant pumps are not shown

Table 26 contains the ME HFOCs (m³/h) used in the calculations. In the data in Table 26, the ambient conditions have been incorporated as well as the operating profile. The engine is not operated in the following conditions at all SW temperatures: Engine load 85% at SW temp. = 6°C and engine load 95%. Therefore, zero is entered in the corresponding matrix cells.

VFD-controlled supply pump
If the FO supply pumps deliver only the needed fuel oil amount to the circulation circuit, a VFD can be installed for the supply pump electric motor. This VFD must be controlled so that a constant inlet pressure is kept to the circulation system at 4 barg. Fig. 16 shows the valve arrangement. As the minimum number of revolutions of the electric motor for the supply pump is, for example, 30% (supplier specific), the self-acting pressure setting valve is still necessary. The valve set pressure must be increased to 4.2 barg to obtain hysteresis between the control pressure signal and the set pressure for this valve.

Based on the annual operating profile and the ME load (and thereby the ME fuel oil consumption), it is possible to calculate the power needed annually for the supply pump. The required power is converted to HFO consumption based on $SFOC_{EA}$ = 216.8 g/kWeh in Table 27.

The annual savings for a VFD-operated supply pump can be determined by comparing the above result with the annual HFO consumption, using a non-VFD operated supply pump as in Table 28.

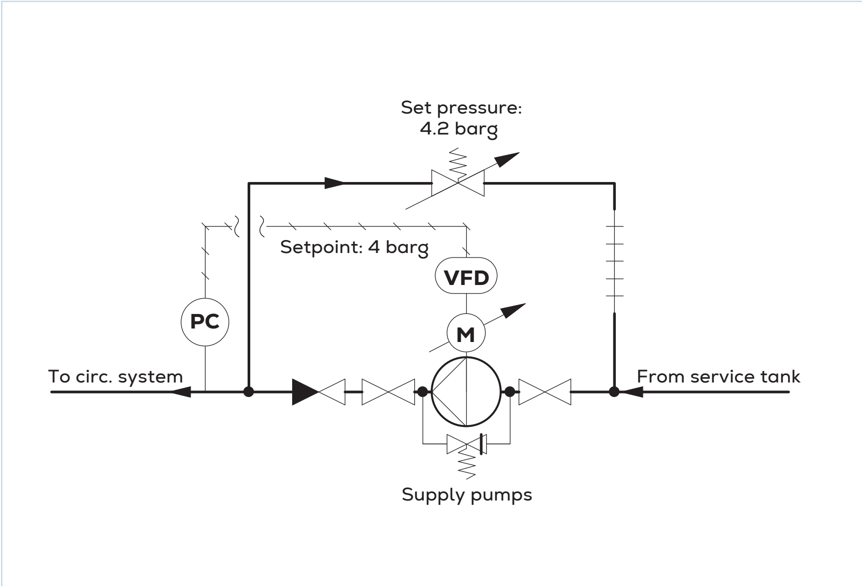


Fig. 16: Pipe arrangement and VFD-operated supply pump

Heavy fuel oil consumption

Ambient conditions				Engine load										
Ambient SW temp.		Cooling water temp.	Ambient air temp.	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
[°C]		[°C]	[°C]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
>30	32	36	39	0	1.16	1.67	2.65	3.65	4.62	5.58	6.54	7.54	8.59	0
26-30	28	32	35	0	1.16	1.66	2.64	3.64	4.61	5.56	6.52	7.52	8.57	0
22-26	24	28	31	0	1.15	1.66	2.64	3.63	4.59	5.55	6.50	7.49	8.54	0
18-22	20	24	27	0	1.15	1.65	2.63	3.61	4.58	5.53	6.48	7.47	8.51	0
14-18	16	20	23	0	1.15	1.65	2.62	3.60	4.56	5.51	6.46	7.45	8.49	0
8-14	11	15	18	0	1.14	1.64	2.61	3.59	4.55	5.49	6.43	7.42	8.45	0
<8	6	10	13	0	1.14	1.63	2.60	3.57	4.53	5.47	6.41	7.39	0	0

Ref. density: 950 kg/m³

Table 26: ME HFOC [m³/h] as a function of load versus ambient conditions, including the operating profile

100% or 50% flow operation of the circulation pump
Installing a two-speed pump (or using a VFD as two-speed control) in the circulation system combined with a simple approach to indicate ME operation, allows the flow rate to be reduced to 50% of the nominal flow when the engine is stopped. One possibility is to handle this with the “finished with engine” signal. Fig. 17 shows the proposed arrangement in detail, using a VFD for the circulation pump.

The annual savings for a 100%/50% operated circulation pump is calculated in Table 29 by comparing with the conventional circulation pump operation.

VFD-controlled circulation pump
Another possibility to obtain savings related to the operation of the FO circulation system is to operate the system as a function of the engine FO consumption. As for the supply pump, a VFD can be installed for the circulation pump electric motor. The circulation pump revolutions are verified by measuring a fixed circulation rate after the ME to ensure that the pump always keeps a minimum circulation rate independent of the ME FO consumption.

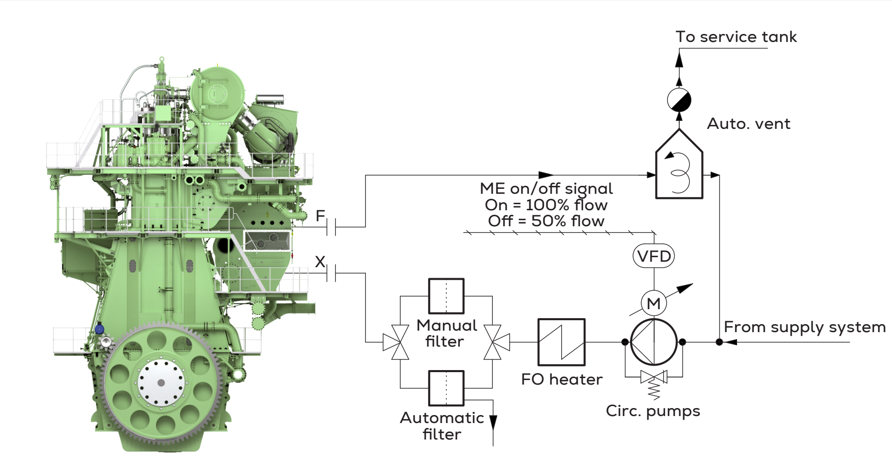


Fig. 17: FO circulation pipe diagram based on 100% flow for a running ME and 50% flow for a stopped ME

Annual HFO consumption for VFD-operated fuel oil supply pumps (min. circulation rate: 30%)

Ambient conditions				Engine load										
Ambient SW temp.		**Cooling water temp.**	**Ambient air temp.**	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
[°C]		**[°C]**	**[°C]**	**0%**	**10%**	**15%**	**25%**	**35%**	**45%**	**55%**	**65%**	**75%**	**85%**	**95%**
›30	32	36	39	32.5	13.0	39.0	39.0	65.0	34.3	41.4	48.6	33.6	5.1	0
26-30	28	32	35	84.5	39.0	104.1	71.5	78.0	82.1	90.9	87.2	111.6	12.7	0
22-26	24	28	31	71.5	32.5	78.0	52.0	52.0	54.6	65.9	57.9	89.0	12.7	0
18-22	20	24	27	52.0	32.5	52.0	52.0	45.5	40.8	49.2	57.7	88.7	12.6	0
14-18	16	20	23	32.5	13.0	19.5	13.0	13.0	27.1	32.7	38.4	33.2	12.6	0
8-14	11	15	18	19.5	6.5	13.0	13.0	6.5	13.5	24.4	28.7	33.0	12.5	0
‹8	6	10	13	13.0	6.5	6.5	6.5	6.5	6.7	16.2	19.0	21.9	0	0

Annual HFO consumption: 2.7 t/yr., ref. LCV: 40,200 kJ/kg

Table 27: Supply pump HFO consumption based on $SFOC_{AE}$ = 216.8 g/kWeh

Supply pump – annual HFO consumption

	[t/yr.]
Non-VFD operated supply pump: 4 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	7.6
VFD-operated supply pump	2.7
Savings obtained with VFD-operated supply pumps	5

Table 28: Obtained HFO savings for VFD-operated supply pumps

Fig. 18 shows the proposed arrangement in details, using a VFD and a flowmeter to determine the necessary fuel oil flow to the ME.

Based on the annual operating profile and the ME load (and thereby the ME FO consumption), it is possible to calculate the power needed for the circulation pump annually, considering a minimum circulation rate of 50%. The calculated power is converted to HFO consumption in Table 30 based on $SFOC_{EA}$ equal to 216.8 g/kWeh.

The annual savings for a VFD-operated circulation pump can be determined by comparing the above result with the annual HFO consumption when using a non-VFD operated circulation pump (see Table 31).

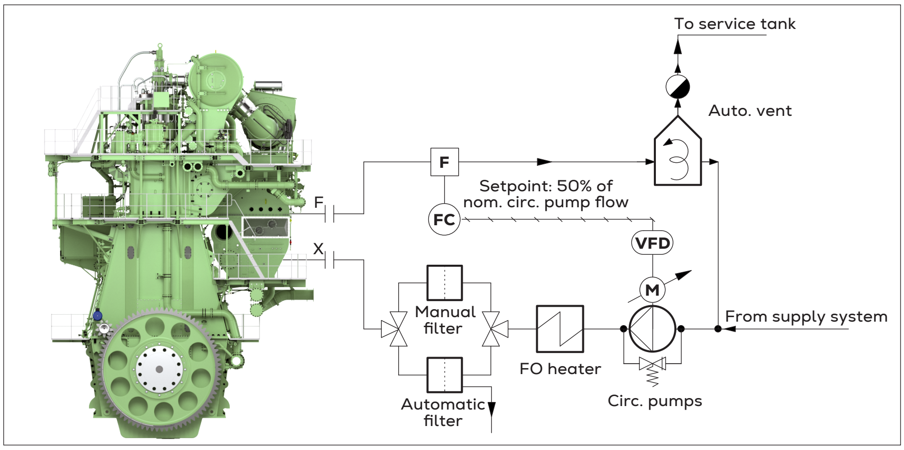


Fig. 18: FO circulation pipe diagram based on a fixed circulation flow independent of the ME consumption

Circulation pump – annual HFO consumption

	[t/yr.]
Non-flow controlled circulation pump: 10 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	19.0
100%/50% controlled circulation pump: (10 kWe × 7,585 h + 5 kWe × 1,175 h) × 216.8 g/kWeh × 10^{-6}	17.7
Savings obtained with 100%/50% operated circulation pump	1

Table 29: Obtained HFO savings for 100%/50% operated circulation pump

Annual HFO consumption for VFD-operated fuel oil circulation pumps (min. circulation rate: 50%)

Ambient conditions				Engine load										
Ambient SW temp.		Cooling water temp.	Ambient air temp.	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
[°C]		[°C]	[°C]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
>30	32	36	39	135.5	59.4	185.1	198.4	353.0	187.5	198.2	209.1	132.2	18.6	0.0
26-30	28	32	35	352.3	178.2	493.3	363.5	423.3	449.5	435.7	375.9	440.0	46.4	0.0
22-26	24	28	31	298.1	148.4	369.8	264.2	282.0	299.4	316.5	250.3	351.6	46.3	0.0
18-22	20	24	27	216.8	148.4	246.5	264.0	246.6	224.4	237.2	250.0	351.1	46.2	0.0
14-18	16	20	23	135.5	59.3	92.4	66.0	70.4	149.4	158.0	166.5	131.5	46.2	0.0
8-14	11	15	18	81.3	29.7	61.6	65.9	35.2	74.6	118.3	124.7	131.3	46.1	0.0
<8	6	10	13	54.2	29.7	30.8	32.9	35.1	37.3	78.8	83.0	87.4	0.0	0.0

Annual HFO consumption: 12.4 t/yr., ref. LCV: 40,200 kJ/kg

Table 30: Circulation pump HFO consumption based on $SFOC_{AE}$ = 216.8 g/kWeh

Circulation pump – annual HFO consumption

	[t/yr.]
Non-VFD operated circulation pump: 10 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	19.0
VFD-operated circulation pump	12.4
Savings obtained with VFD-operated circulation pumps	7

Table 31: Obtained HFO savings for VFD-operated circulation pump

Closing remark

As demonstrated, efficiency improvements are present and savings can be obtained by combining the VFD-operated supply pump (page 27) and the VFD-operated circulation pump (page 28) (see Fig. 19). Table 32 shows annual HFO savings of 45%.

The savings are rather small in HFO figures compared to other savings potentials in this paper. However, we assume that the additional cost for a VFD compared to a pump starter cabinet is very small for such small pumps, and it may therefore be a good business case.

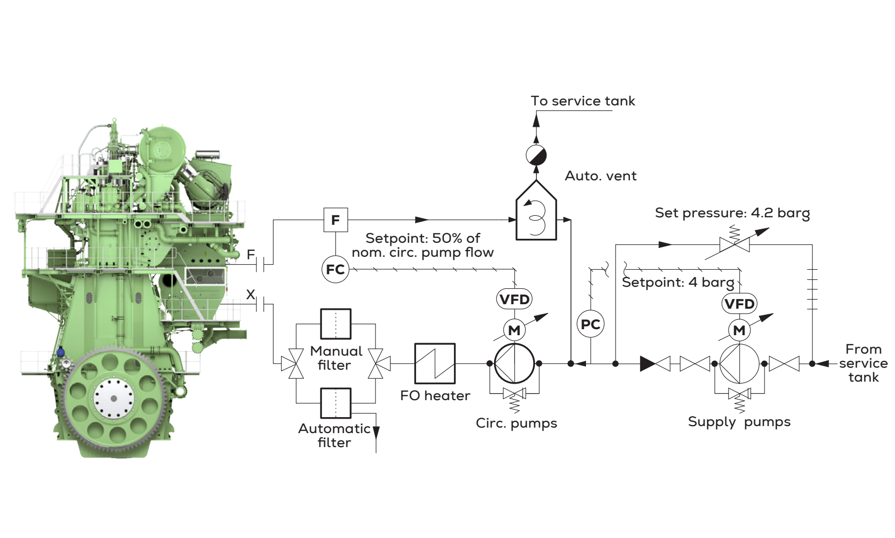


Fig. 19: VFD-operated pumps in the FO supply and circulation system

Fuel oil pumps – annual HFO consumption

	[t/yr.]
Non-VFD operated supply pump: 4 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	7.6
Non-VFD operated circulation pump: 10 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	19.0
Non-VFD operated fuel pumps	26.6
VFD-operated supply pump	2.7
VFD-operated circulation pump	12.4
VFD-operated pumps	15.1
Total annual savings	12

Table 32: Obtained HFO savings for VFD-operated pumps

5. Lubricating cooling oil

When looking at the efficiency improvements to the lubricating oil (LO) system, we have focused on two systems:

- Main LO system
- LO cleaning system

Main lubricating oil system

As Fig. 20 shows, LO is pumped from a bottom tank bythe main LO pump to the LO cooler, thermostatic valve, and through a full flow filter to the ME inlet flange. The LO system lubricates main bearings, thrust bearing, axial vibration damper, piston cooling, crosshead bearings, and crankpin bearings. It also supplies oil to the hydraulic power supply unit, the moment compensator, and the torsional vibration damper, if installed. From the engine, the oil collects in the oil pan where it is drained to the bottom tank again.

The LO pumps must supply a well-defined, load-independent inlet pressure to the ME with a LO capacity specified by Everllence. For the specific engine type: 8G95ME-C9.5 Tier II, the LO inlet pressure is normally 2.8 barg, measured 1,800 mm above the crankshaft.

Lubricating oil capacities for 8G95ME-C9.5
Calculating the savings potential, the following capacities for the engine type 8G95ME-C9.5 Tier II have been used. These are from the CEAS application and reproduced in Table 33.

As basis for our calculations, we have used a deep well centrifugal pump with a pumping head of 4.8 bar, a mechanical centrifugal pump efficiency EFFm = 75%, and an electric pump motor efficiency EFFe = 94.5%.

- LO pump: 860 m³/h at 4.8 bar
  - 162 kWe

Efficiency improvements to main engine lubricating oil system
Different efficiency improvement solutions are available when designing the ME LO system, these are:

- System layout improvements
- LO pump efficiency

As the LO pressure and thereby the flow must be constant and independent of the engine load, a VFD solution is not an option for the LO pump system.

Pump	Flow capacity m³/h	Pump head Bar
Fuel oil circulation	24.1	6.0
Fuel oil supply	14.6	4.0
Jacket water	380	3.0
Central water	1,270	2.5
Sea water for central cooling	1,580	2.0
Lubrication oil	860	4.8

Table 33: CEAS capacities

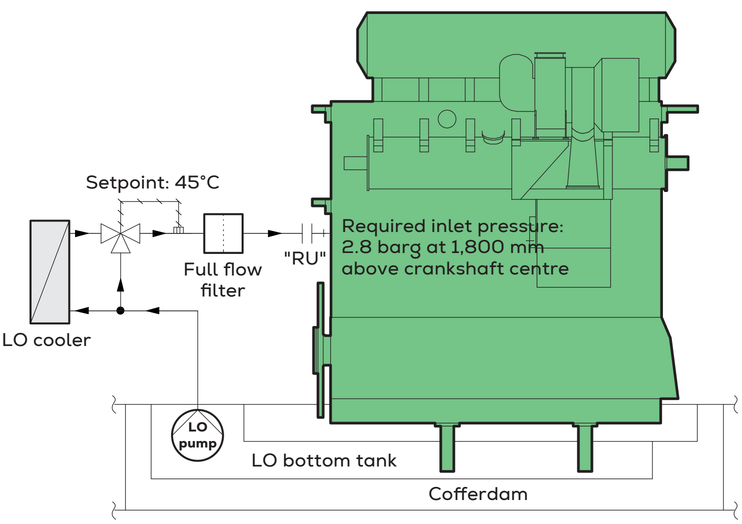


Fig. 20: Typical LO system for ME

System layout improvements

The ME LO pump must lift the oil the hydrostatic height from the LO bottom sump tank to the engine through the LO cooler, thermostatic valve, through a full flow filter, the entire pipe system, and supply 2.8 barg to the engine. In the data provided by Everllence, the LO ME pump capacity is given as well as a guidance value for the pumping head.

The guidance value for the ME LO pumping head is 4.8 bar, and it must be emphasised that this pressure head is only a guidance value based on assumptions about the hydrostatic lifting height and pressure loss in the system from the pump to the engine. The assumption for this specific engine is: 4.8-2.8 bar = 2.0 bar. When the system is designed and the pumping head specified, it is therefore very important that the LO components are well-known and specified with the lowest possible pressure loss. When the ship designers make the detailed design of the system, all the pressure loss figures for the components must be evaluated and challenged to ensure that the pressure loss is the smallest possible. Thereby a smaller pumping head can be specified.

The relevant items:

- Pipe system length/pipe diameter/wide radius bends/flow tees, etc.
- Low differential pressure across the LO cooler
- Low differential pressure across the LO filter
- Low differential pressure across the valve components, especially the LO thermostatic three-way regulating valve. Alternatively, this valve can be located at the LO cooler water side, which is fully acceptable (see Fig. 21).

As an example, if the hydrostatic lifting height, from the tank bottom to 1,800 mm above the crankshaft, and the total pressure loss in the components and pipe system is calculated to 1.5 bar at the given flow, the LO pump can be specified at 2.8+1.5 = 4.3 bar, instead of the preliminary estimate of 4.8 bar provided by Everllence.

Lubricating oil pump efficiency

By having high requirements to the mechanical and electric motor efficiencies, it is possible to lower the power consumption for the LO pump. See Fig. 7 for a detailed explanation of mechanical efficiency and electric motor efficiency.

Considering the items below for the lubricating oil system, it is possible to calculate the potential savings in Table 34.

- Pipe system pressure loss of 1.5 bar, i.e., a pumping head of 4.3 bar instead of the baseline head of 4.8 bar
- Mechanical efficiency, EFFm: 80% instead of the baseline efficiency of 75%
- Electric motor efficiency class IE3 (96.2%) instead of baseline class IE1 (94.5%)
- The pump is only running during ME operation.

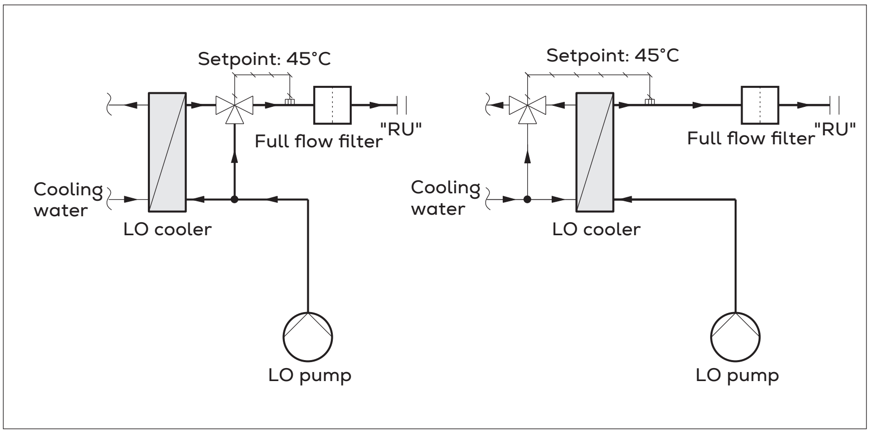


Fig. 21: Standard (left) and alternative location (right) of the thermostatic three-way regulating valve

LO pump – annual HFO consumption

	[t/yr.]
LO pump, head=4.8 bar, EFFm=75%, EFFe=94.5%: 162 kWe × 7,585 h × 216.8 g/kWeh × 10^{-6}	266.4
LO pump, head=4.3 bar, EFFm=80%, EFFe=96.2%: 133 kWe × 7,585 h × 216.8 g/kWeh × 10^{-6}	218.7
Total annual savings	48

Table 34: LO system annual HFO consumption savings

Lubricating oil cleaning system

The LO cleaning system is a continuously running system, even at stopped ME. The separator pump draws LO from the bottom tank, pumps it through the separator preheater to heat up the oil to 95°C, and eventually to the separator. The separator discharges the purified clean oil back to the LO bottom tank and discharges water/dirty oil to the sludge tank. Fig. 22 shows the standard system.

Lubricating oil cleaning capacity

The capacities in Table 35 have been used in the calculation of the savings potential for the engine type 8G95ME-C9.5 Tier II. The capacities are drawn from the CEAS application. As the basis for our calculations, we have used a positive displacement pump with a pumping head of 2.5 bar, a pump mechanical efficiency of EFFm = 60%, and a pump motor elec-

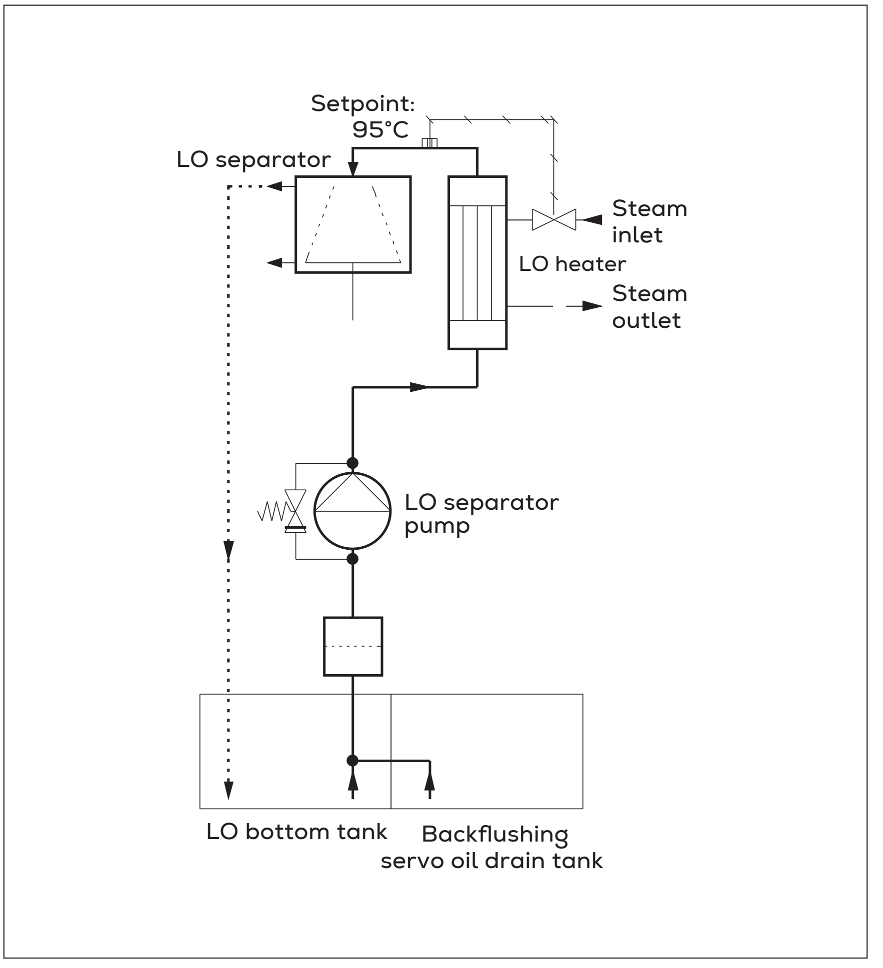


Fig. 22: Standard LO cleaning system

Lubrication oil system	
Storage tanks (2 x 3 months)	2 x 8.4 m^3
Centrifuge, 95 °C	7,470 l/h
Recommended lube oil bottom tank	52 m^3

Tables of cooler capacities (Tier II)

1	Engine load (% SMCR)	4	Scavenge air receiver temp. (°C)	7	Main lubrication oil heat (kW)
2	TC air consumption (kg/s) +/-5%	5	Scavenge air cooler heat (kW)	8	Condensed water (t/24h)
3	Scavenge air pressure (bara)	6	Jacket water cooler heat (kW) +0/-15%		

Loads below 50% are associated with larger tolerances.

1	2	3	4	5	6	7	8
ISO condition			Ambient air: 25.0 °C			Scavenge air coolant: 25.0 °C	
100	119.1	4.20	37	20,720	6,590	3,840	0.0
95	115.1	4.02	36	19,240	6,350	3,800	0.0
90	110.9	3.83	35	17,780	6,100	3,740	0.0
85	106.6	3.65	34	16,340	5,860	3,680	0.0
80	102.2	3.50	33	15,060	5,610	3,600	0.0
75	97.5	3.38	32	13,920	5,370	3,520	0.0
70	92.6	3.13	31	12,250	5,120	3,440	0.0
65	87.4	2.91	30	10,730	4,880	3,340	0.0
60	81.9	2.72	30	9,380	4,630	3,230	0.0
55	76.1	2.54	29	8,070	4,390	3,120	0.0
50	70.1	2.35	28	6,800	4,150	2,990	0.0
45	63.8	2.17	27	5,580	3,900	2,850	0.0
40	57.3	1.98	26	4,420	3,660	2,700	0.0
35	50.5	1.80	33	3,350	3,410	2,540	0.0
30	52.2	1.64	32	2,890	3,170	2,360	0.0
25	44.1	1.49	32	1,970	2,920	2,160	0.0

Table 35: Capacities drawn from CEAS

trical efficiency of EFFe = 78%.

- LO separator pump: 7.47 m^3/h at 2.5 bar
  - 1.1 kWe

Efficiency improvements to the main engine lubricating oil cleaning system
Today, steam energy on board a vessel is not necessarily a free energy source as the fuel optimised engines may not always be able to cover the steam demand from the installed auxiliaries (e.g., tank heating, fuel oil preheating, air conditioning heating, etc.). Especially for smaller engines at low load operation, this can pose a problem.

On large engine installations where a waste heat recovery system is installed, there will be an interest in utilising as much as possible of the steam produced in the exhaust gas boiler for electricity production.

Efficiency improvements leading to a reduced steam consumption are therefore very important, as a reduced steam consumption means less steam production in the oil-fired boiler, or less steam available for producing electricity in the waste heat recovery system.

By installing a heat exchanger in the cleaning loop, the heated return oil can be used to heat the inlet oil to the preheater (see Fig. 23). Thereby, it is possible to save steam energy in the LO preheater, as the temperature difference between the required 95°C and the LO inlet temperature to the preheater will be smaller. A small setback by installing a heat exchanger is that the pressure drop across the heat exchanger will require a larger LO separator pump pumping head, resulting in a higher electricity consumption.

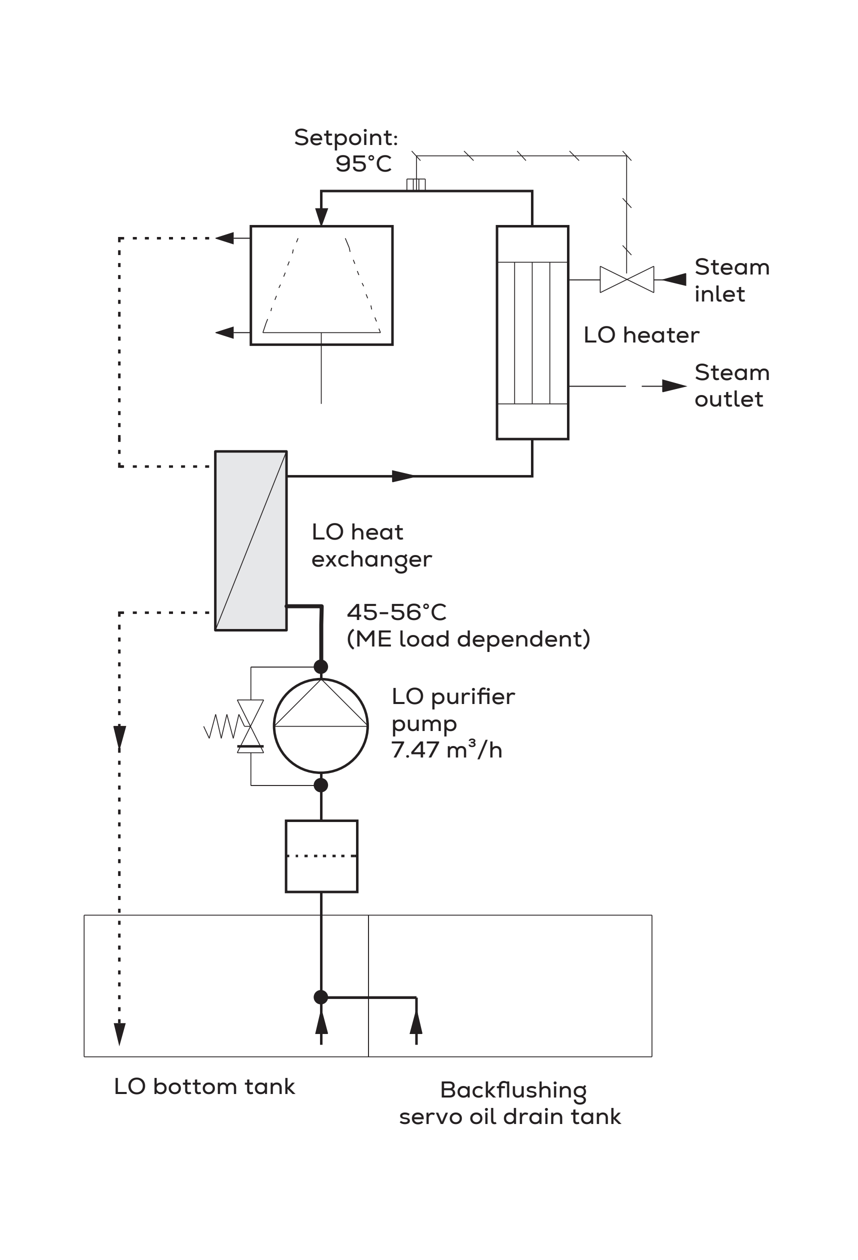


Fig. 23: Heat exchanger performance calculation model

To give a realistic picture, only the savings at low engine load are considered in the investigation of efficiency improvements. If the steam demand on board is large, beneficial savings are also achievable at high engine loads.

To estimate the saving potential for such an installation, firstly the steam consumption and secondly the annual HFO consumption for a conventional main engine lubricating oil cleaning system must be known. A consumption of 0.073 kg HFO per 1 kg steam has been used.

The LO preheater steam consumption is directly related to the heat necessary to raise the LO temperature to the required 95°C. It is possible to determine the inlet temperature to the LO preheater at different engine loads by considering the CEAS data for engine heat radiation to the LO (see Table 35, column number 7). Note that only the ISO condition has been used since the difference between this condition and other ambient conditions is minor. Note also that heat radiation to the surroundings in the LO tank has not been considered. Tables 36 and 37 show the calculations and the result for a standard system.

Installing the heat exchanger and executing a heat exchanger performance calculation for

LO cleaning system – required heat

	Added heat to the LO from the engine - ISO condition according to CEAS [kW]										
Engine load	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
Radiated heat [kW]	0	990	1,390	2,160	2,540	2,850	3,120	3,340	3,520	3,680	3,800

	LO bottom tank temperature [°C]										
Engine load	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
Inlet temp. to separator preheater [°C]	45.0	47.9	49.0	51.3	52.4	53.3	54.0	54.7	55.2	55.7	56.0

	Preheater heat [kW]										
Engine load	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
LO separator preheater heat [kW]	150	141	138	131	128	125	123	121	119	118	117

Annual HFO consumption for producing steam for LO preheating purposes for the LO separator

Engine load	**0%**	**1-10%**	**11-20%**	**21-30%**	**31-40%**	**41-50%**	**51-60%**	**61-70%**	**71-80%**	**81-90%**	**91-100%**
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
HFO consumption [kg/yr.]	22,607	9,974	21,227	15,988	16,816	15,257	Not considered				

Annual HFO consumption: 102 t/yr., ref. LCV: 40,200 kJ/kg

Table 36: Annual HFO consumption for the LO cleaning system

LO preheater and LO separator pump – annual HFO consumption

	[t/yr.]
LO preheater (based on oil fired boiler steam production)	101.9
LO preheater pump, head=2.5 bar, EFFm=60%, EFFe=78%: 1.11 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	2.1
Annual HFO consumption	104

Table 37: Annual HFO consumption for LO preheater and LO separator pump

each ME load (Table 38) makes it possible to find the new inlet temperature to the LO preheater. In this way it is possible to calculate the heat needed for the LO preheater and the annual HFO consumption in Tables 38 and 39, respectively. In the calculated annual HFO consumption in Table 39, the LO separator pumping head has been increased as well as the electric motor efficiency EFFe (based on IE3 motor). Table 40 provides the savings obtained by installing a heat exchanger in the LO cleaning system.

Required heat for the LO cleaning system with heat exchanger

	LO temperature [°C]										
Engine load	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
Inlet temperature to heat exchanger [°C]	45.0	47.9	49.0	51.3	52.4	53.3	54.0	54.7	55.2	55.7	56.0
Outlet temperature from heat exchanger [°C]	64.2	66.1	66.8	68.3	69.0	69.5	70.0	70.4	70.7	71.0	71.2

	LO separator preheater heat [kW]										
Engine load	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
LO seperator preheater heat using a heat exchanger [kW]	92	87	84	80	78	76	75	74	73	72	71

LO separator inlet temperature: 95°C, LO density: 875 kg/m^3

Annual HFO consumption producing steam for LO preheating purpose using heat exchanger

Engine load	0%	1-10%	11-20%	21-30%	31-40%	41-50%	51-60%	61-70%	71-80%	81-90%	91-100%
Engine load, used [%]	0%	10%	15%	25%	35%	45%	55%	65%	75%	85%	95%
HFO consumption [kg/yr.]	13,935	6,118	13,021	9,768	10,263	9,307	Not considered				

Annual HFO consumption: 62 t/yr., ref. LCV: 40,200 kJ/kg

Table 38: Annual HFO consumption for the LO cleaning system using heat exchanger

LO preheater and LO separator pump – annual HFO consumption using heat exchanger

LO preheater (based on oil-fired boiler steam production)	62.4
LO preheater pump, head=3.5 bar, EFFm=65%, EFFe=85.5%: 1.31 kWe × 8,760 h × 216.8 g/kWeh × 10^{-6}	2.5
Annual HFO consumption	65

Table 39: Annual HFO consumption for LO preheater and LO separator pump using heat exchanger

Total obtained savings

Annual HFO consumption – standard system	104
Annual HFO consumption – using heat exchanger	65
Total annual savings	39

Table 40: Annual HFO savings using a heat exchanger

Closing remark

By implementing the efficiency improvements to the main LO system and the LO cleaning system, the savings in Table 41 can be obtained.

Total savings for the main LO system and the LO cleaning system

Main LO system annual HFO savings	48
LO cleaning system annual HFO savings	39
Total annual savings:	**87**

Table 41: Total savings for the main LO system and the LO cleaning system

6. Tier III improvements – additional efficiency

Scavenge air cooling water control

The scavenge air cooling flow required by engines equipped with a Tier III exhaust gas recirculation (EGR) system supplies both the scavenge air cooler and the EGR cooler. Everllence defines this system as shown in the CEAS report in Table 42 (example: 5G70ME-C9.5-GI, SMCR: 12,675 kW at 69.4 rpm).

Two options are mentioned: a standard and an optimised cooling system. The information in the red box describes the standard solution, and the information in the green box describes the option called optimised cooling system for EGR.

The standard system requires a constant coolant water flow when operating the vessel in both Tier II and Tier III areas. In the optimised cooling water system, the water flow differs

Capacities of pumps and coolers

Pump	Flow capacity m³/h	Pump head bar
Fuel oil circulation	6.0	6.0
Fuel oil supply	3.3	4.0
Jacket water	90	3.0
Central water	410	2.5
Sea water for central cooling	460	2.0
Lubrication oil	360	4.5

	Tier II			Tier III		
Cooler	Flow m³/h	Central water flow m³/h	Heat dissipation kW	Flow m³/h	Central water flow m³/h	Heat dissipation kW
Scavenge air	-	260	3,960	-	260	6,400
Lubrication oil	360	150	1,210	360	150	1,210
Jacket water	90	150	1,670	90	150	1,700
Central water*)	460	410	6,840	460	410	9,310
Fuel oil circulation (MGO/MDO)	-	-	29	-	-	29

Optimized cooling system for EGR

If equipped with a variable frequency drive on the central- and sea water pumps, and other optional engine functionality, the above capacities can be reduced to the below.

	Tier II		Tier III	
Cooler	Flow m³/h	Central water flow m³/h	Flow m³/h	Central water flow m³/h
Scavenge air	-	150	-	230
Central water*)	340	300	460	380

*) The "Flow" column contains the sea water flow through the central cooler.
Scavenge air heat dissipation for Tier III includes the EGR cooler heat.

All flows are stated as minimum required flows.
The pump heads stated are for guidance only, and depend on the actual pressure drop across coolers, filters, etc. in the systems. The capacities do not account for any components other than the engine itself.
Pertaining cooling water flow diagram, temperatures, viscosities and pressures for pumps and coolers, see "Engine Project Guide".

Table 42: CEAS capacities for Everllence B&W two-stroke engine type: 5G70ME-C9.5-GI

in Tier II and Tier III operation, see Fig. 24. The optimised system requires that it is possible to adjust the flow to the engine by means of, for example, VFD-operated, or two-speed operated cooling water pumps.

Furthermore, it requires installation of a control system, which controls the flow to the scavenge air cooler and the EGR cooler, respectively, depending on the Tier operating mode.

More information about the optimised system is available and can be forwarded by contacting Everllence/Marine Installation.

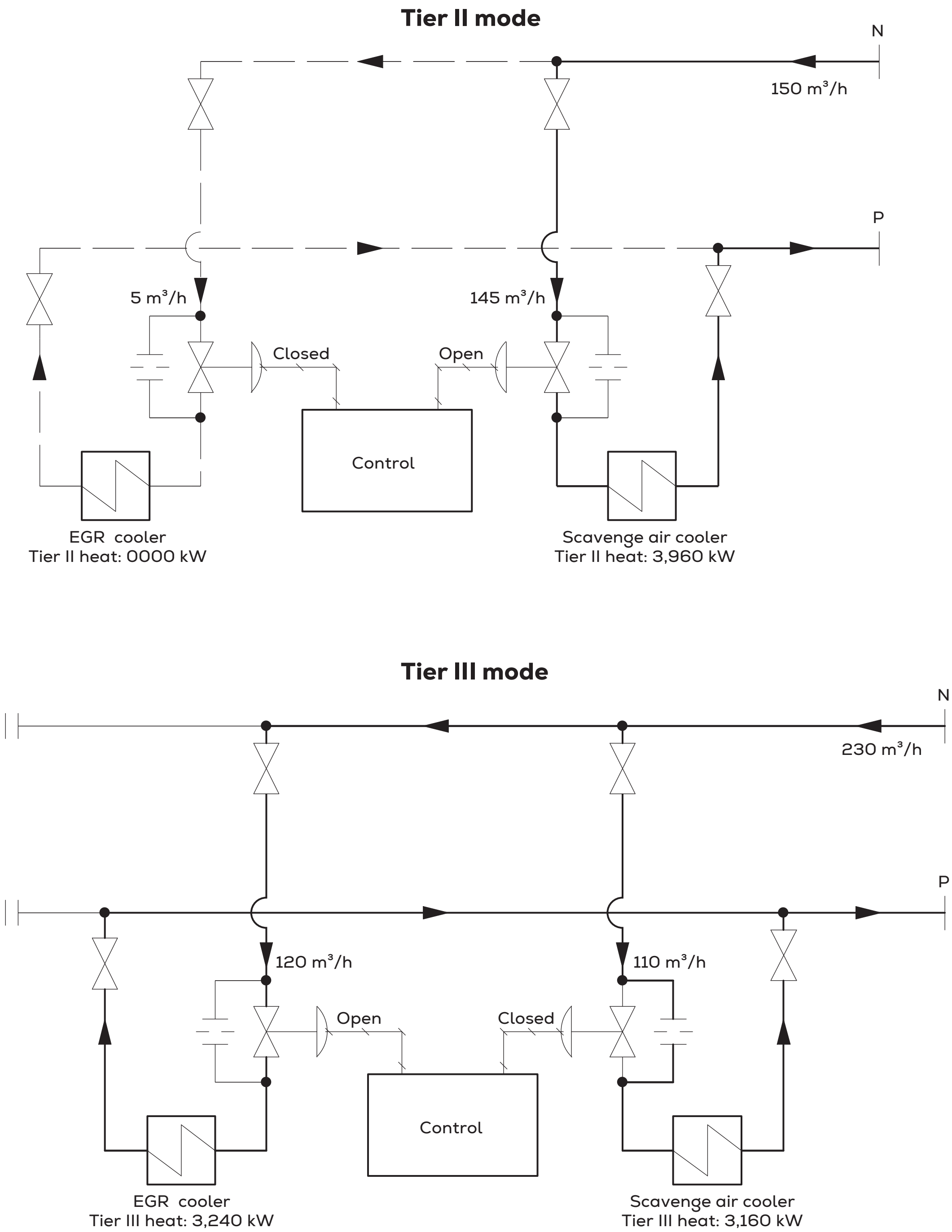


Fig. 24: Principle diagram of the internal pipe system for the optimised cooling system for EGR operation of an Everllence B&W two-stroke engine type 5G70ME-C9.5-GI

7. Summary

When choosing the most beneficial solution for each system described earlier and adding the savings, it is possible to get an idea of the total savings obtainable for the auxiliary system for an Everllence B&W two-stroke engine type: 8G95ME-C9.5 Tier II.

The following options have been chosen:

- Main engine direct air intake (page 8)
- Cooling water system (pages 24-25)
  - Operate the SW pump and central cooling water pump with VFDs
  - Use the optimised system pressure loss:
    - SW pumping head: 1.6 bar (baseline 2 bar)
    - Central water pumping head: 2.0 bar (baseline 2.5 bar)
    - JCW pumping head: 2.7 bar (baseline 3.0 bar)
  - Use a mechanical pumping efficiency EFFm = 80% instead of the baseline efficiency 75%
  - Use the electric motor efficiency class IE3 instead of baseline class IE1
- VFD-operated fuel oil supply pump and circulation pump (page 30)
- Optimised LO main system (page 37).

Table 43 shows overall savings for each system.

Fig. 25 shows OPEX savings as a function of HFO bunker price.

As mentioned before, the CAPEX for the proposed efficiency improvements are not covered in this paper, and it is therefore not possible to estimate the payback time.

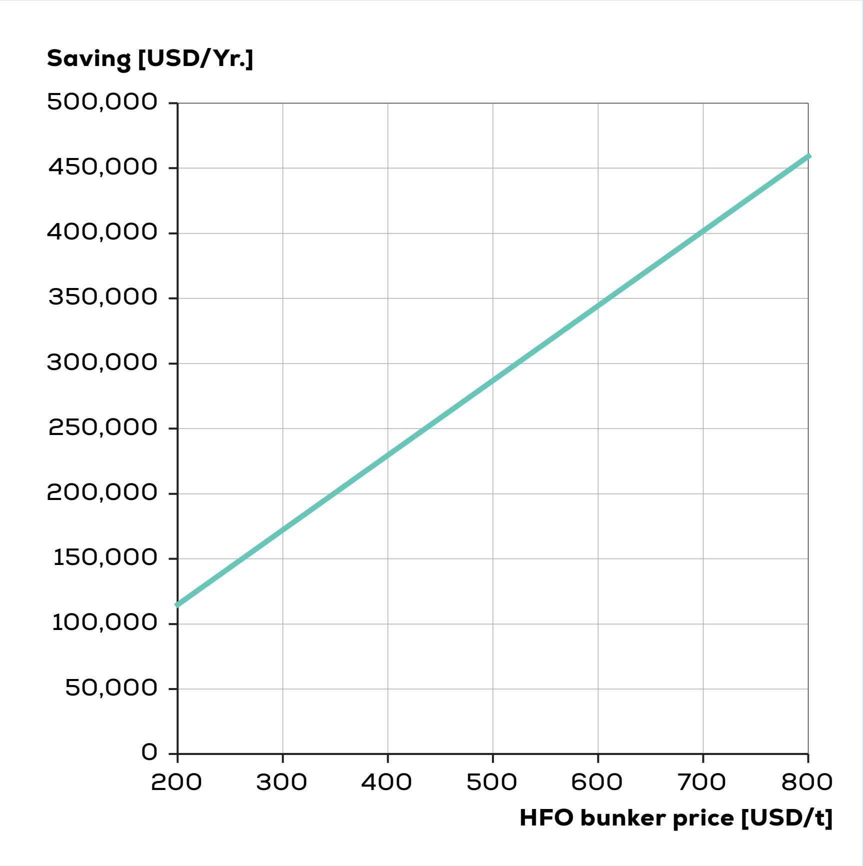


Fig. 25: OPEX saving as a function of HFO bunker price

Total savings

	[t/yr.]
Main engine direct air intake	215
Cooling water	299
Fuel oil system	12
LO system	48
Total annual savings	574

Table 43: Overall savings

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