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Graduation thesis

ENERGY EFFIENCY OF THE MARINE MAIN ENGINE AUXILIARY SYSTEMS

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Tämän työ tarkoituksena oli selvittää, kuinka paljon energiaa tarvitaan nesteiden pumppaamiseen laivan pääkoneen apulaitejärjestelmissä ja valita energiaa kuluttavin järjestelmä lähempään tarkasteluun. Kun yksi järjestelmä on valittu, on tarkoitus selvittää voidaanko kyseisen järjestelmän energiatehokkuutta parantaa säilyttäen kuitenkin Wärtsilän tekniset suositukset. Päämoottoriksi valittiin 9L46 moottori ja kaikki valmistajien laitteistotiedot, kuten esimerkiksi pumput, saatiin Wärtsilä Ship power:in TERPS tietokannasta. Saatujen tulosten pohjalta tehtiin taloudellisuus vertailu normaalin ja muutetun järjestelmän välillä.

Hakusanat: Energiatehokkuus, apulaitesysteemi, pumppu

Säilytyspaikka: Turun ammattikorkeakoulun kirjasto

TURKU UNIVERSITY OF APPLIED SCIENCES ABSTRACT

Degree programme: Mechanical Engineering	
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The purpose of this thesis was to study, how much energy is used in pumping fluids in each auxiliary system of marine main engine and to choose the system that is the most energy consuming. After choosing one system for closer observation the target was to study how pumping energy efficiency of that specific auxiliary system can be improved by changing the design, but still keeping the original technical limits for the system valid. 9L46 marine main engine was chosen as a reference engine for all calculations and all the component data for each part and unit of different manufacturer was obtained from the TERPS database in Wärtsilä Ship power's intranet page. Based on the calculated results a ballpark economical estimate of investment difference between a normal system and an energy efficient system was made.

Keywords: energy, efficiency, auxiliary system, pump

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PREFACE

Today it is important to pay attention to energy consumption. Using more energy efficient ways can save companies assets and the environment. Considering energy efficiency in the design phase is the best way to make savings. Although some investments can be a bit high at the beginning, they will pay it self back in the long run. This concerns also ship design from propeller to the exhaust pipe. Paying attention to the engines, auxiliary systems, piping etcetera will make difference in energy consumption. This is even more important in the current economical situation of the world. Making little savings here and there can save a reasonable amount of assets.

"Now when the world faces this long and deep depression attention must be focused on expenses. Research and development grants have usually been cut down in this situation, but not anymore. This year Wärtsilä invests more to research and development." (Ole Johansson, Wattsup, 1/2009)

This work commissioned by for Wärtsilä Finland Technical Service department as a part of engineering education at Turku University of Applied Sciences. The supervisors of this work were Seppo Rautava from Wärtsilä Technical Service and Kari Nieminen from Turku University of Applied Sciences.

The author thanks both supervisors for making this work possible and for very good co-operation. Big thanks also to Mr Rautava for arranging the opportunity to visit onboard Estraden to get a concrete view of the engine room and its components.

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1 OVERVIEW OF THE AUXILIARY SYSTEMS

1.1 Fuel oil system

The Wärtsilä 46 engine is designed for using heavy fuel oil, but it is possible to operate the engine on diesel fuel. The fuel system consists of the internal fuel system and the external fuel system. The internal system includes injection pumps built on the engine, injection valves and a pressure control valve in the outlet pipe. The design of the external fuel system may alter from installation to installation however, they should all provide well cleaned fuel in correct temperature and pressure to the engine. In the use of heavy fuel oil it is most important that the fuel is well cleaned from solid particles and water. The fuel treatment system should include at least one settling tank and two separators. The settling tank ensures fuel for minimum time required and provides efficient sludge and water rejecting effect. The preheater before the separator maintains the correct separating temperature and the feed pump feeds fuel oil to the separator.

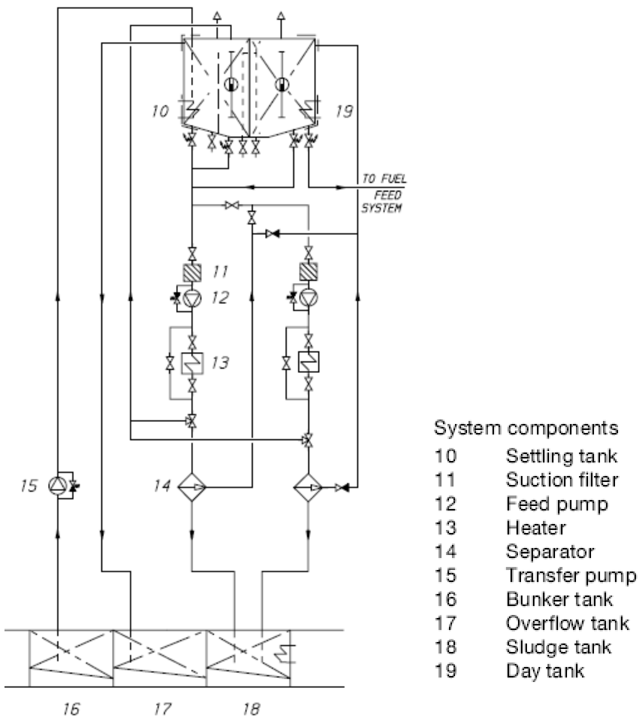


Figure 1.1 HFO separating system (Wärtsilä marine project guide 2001, 57)

Pressurized fuel feed system is needed in HFO installations. The fuel feed system can be installed as a whole fuel feed unit and usually it is delivered assembled from the supplier as a whole unit. In the fuel feed system feed pumps maintain the system pressure and suction strainers protect these pumps. The pressure control valve maintains the pressure in the de-aeration tank directing the surplus flow to the day tank. It is also recommended to use automatic back-flushing filters. Fuel consumption is measured with a consumption metre installed between the fuel feed pumps and de-aeration tank. Maintaining the correct pressure at the engine inlet is done with a circulating pump and a pressure valve; this also keeps circulating fuel in the system. The heater is dimensioned to heat fuel oil to maintain the specific viscosity for injection. A viscosimeter is installed to control the heater to maintain that specific viscosity and a thermostatic valve is installed for safety in case the viscosimeter breaks. The safety filter for fuel oil is installed near the engine.

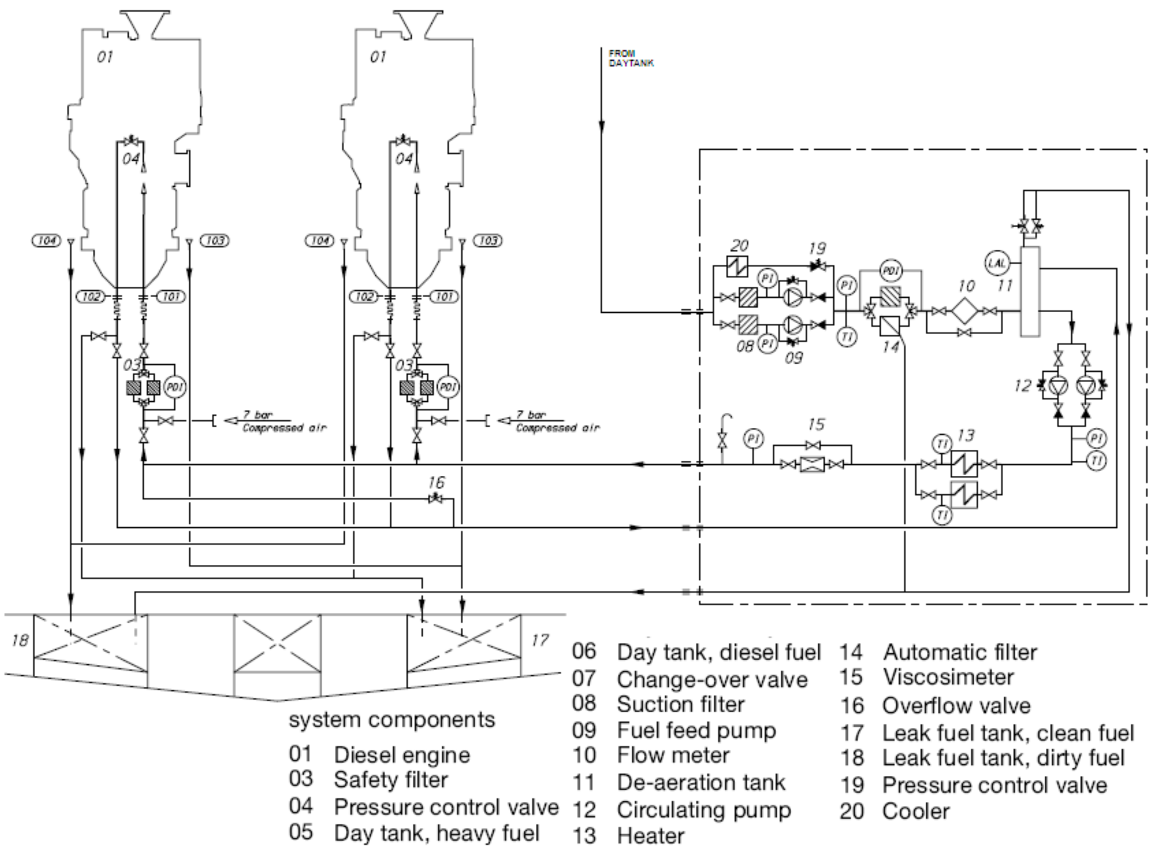


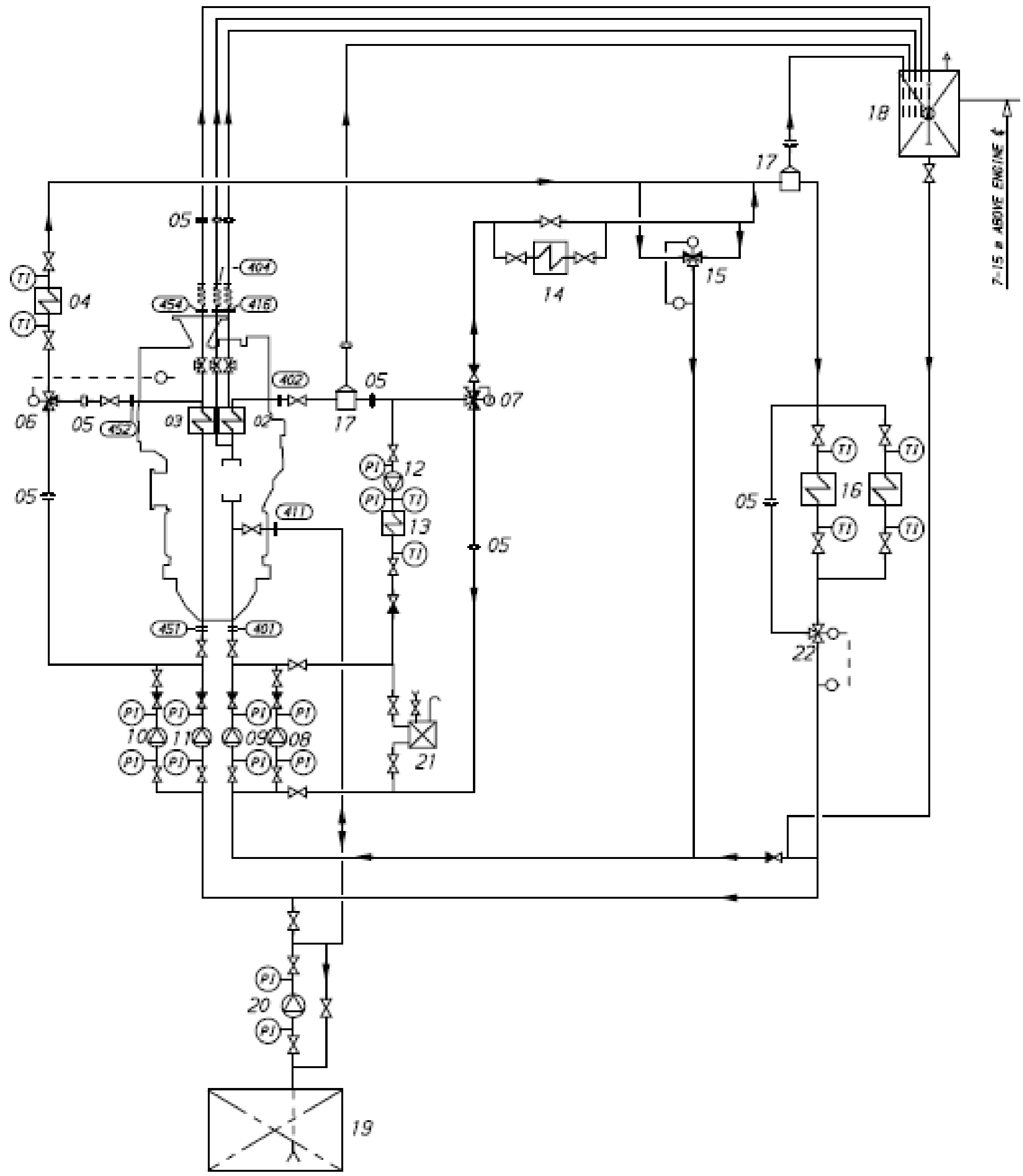
Figure 1.2 Pressurized fuel feed system (Wärtsilä marine project guide 2001, 63)

1.2 Cooling water system

Fresh water is used for cooling the jacket, charge air and lubricating oil. Fresh water can be generated by a reverse osmosis plant or evaporator onboard the vessel. The pH-value and hardness of the water should be within normal values and the chlorine and sulphate content should be as low as possible to avoid corrosion.

The cooling water system, like fuel system, is divided into the internal and external system. The internal system keeps the temperature low enough to limit thermal load and prevents hot corrosion in the combustion chamber at high loads. At low loads it keeps the temperature high enough to ensure complete combustion. The cooling water systems, both internal and external, are divided into low-temperature (LT) and high-temperature (HT) circuits. The internal LT-circuit includes LT-charger air cooler and lubricating oil cooler and the HT-circuit includes the cylinders and the HT-charge air cooler.

The external system is divided into the HT-circuit and LT-circuit and they can be separated from each other, but in the standard Wärtsilä marine cooling system the HT-circuit mixes to the LT-circuit before the central cooler. The fresh water central cooler in marine applications are usually plate type coolers and it can be common for several engines. The lubricating oil cooler is also usually a plate type cooler and it is cooled by LT water. Sea water pumps are electrically driven pumps and circulating pumps are can be electrically or engine driven. The HT-circuit has individual pumps on each circuit, but the LT-circuit can have individual pumps for each engine, or two or three engines in the same circuit can be supplied by the same separately installed pump. In both LT-circuit and HT-circuit there is a thermostatic valve installed. In the LT-circuit it controls the charge air temperature and in the HT-circuit the outlet temperature from the engine. The HT-circuit is heated with a preheater and before the preheater, a preheater pump is installed. The expansion tank compensates volume changes in the cooling water system. It provides static pressure for the system and ensures constant positive suction head at the circulating pump.



System components

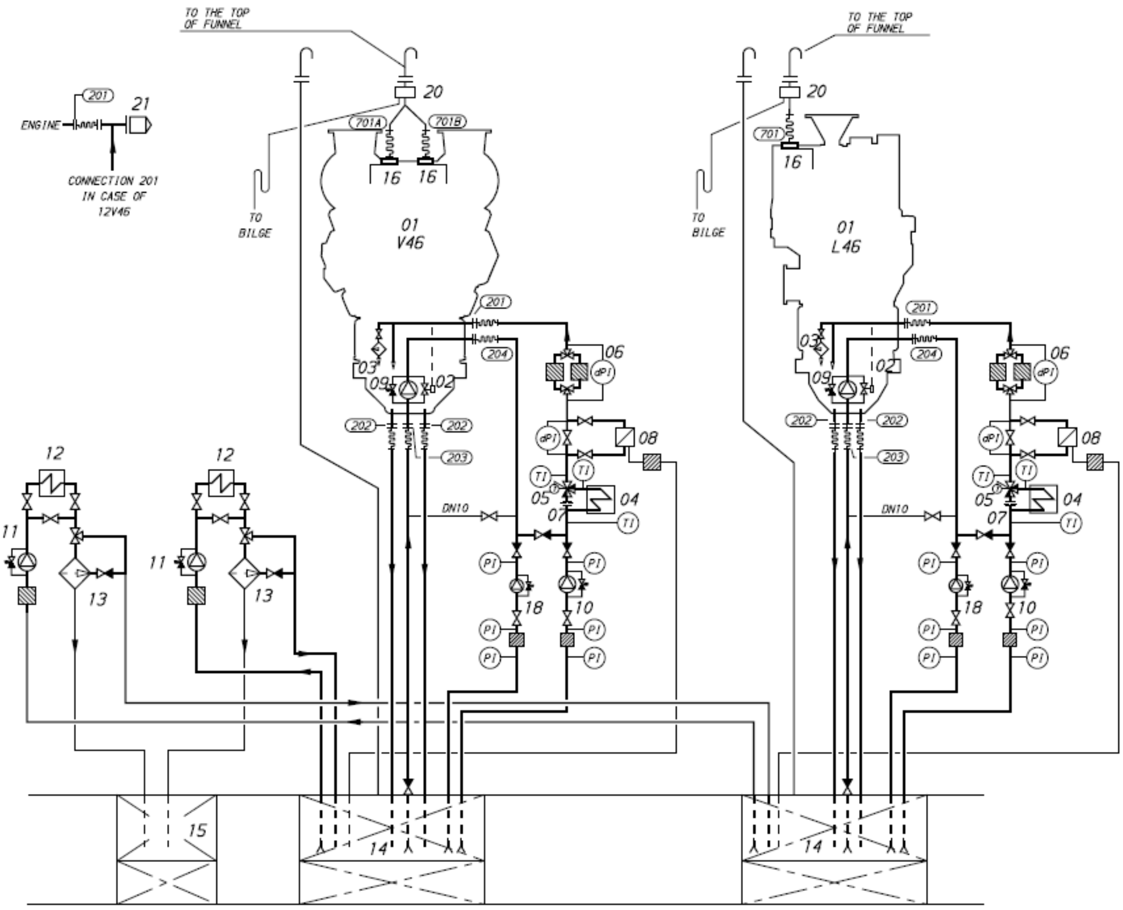
01	Diesel engine	08	HT-circulating pump	15	Thermostatic valve, 60°C
02	HT-air cooler	09	HT-stand-by pump	16	Central cooler
03	LT-air cooler	10	LT-circulating pump	17	Air separator
04	Lube oil cooler	11	LT-stand-by pump	18	Expansion tank
05	Orifice	12	Preheating pump	19	Drain tank
06	LT-thermostatic valve	13	Preheater	20	Transfer pump
07	HT-thermostatic valve, 91°C	14	Heat recovery	21	Additive dosing tank

Figure 1.3 External cooling water system (Wärtsilä marine project guide 2001, 87)

1.3 Lubricating oil system

As the previous systems the lubricating oil system is divided into the external and internal system. Side stream centrifugal filter and starting-up/running-in filter or filters are installed on the engine. If there are engine driven pumps they are located at the free end of the engine.

All the other equipment of the lubricating oil system belongs to the external system. When running with heavy fuel oil engines should have continuous centrifuging of the lubricating oil. The engine dry sump has four drain outlets and totally at least three should be used in 9L46 engine. The oil sump drains to system oil tank. The system oil tank is placed as close to the engine as possible and so that the oil is not cooled too much to maintain the proper oil temperature. The connection between the sump and the system oil tanks is flexible due to thermal expansion. Before the lubricating pumps a suction strainer is installed to protect the pumps. Also a safety valve is attached to the pump, which protects it from overpressure. A thermostatic valve ensures the desired oil temperature at the engine inlet and the pressure control valve ensures the correct pressure. Also an automatic filter and a safety filter need to installed. The lubricating oil separator is dimensioned for continuous centrifuging and each oil system has its own separator. The separator pump is either electrically driven or directly driven, and before the separator there is a preheater to heat the oil to required separating temperature. The heater can be steam or electric model. The lubricating oil temperature increases during operation so it must be cooled with the lubricating oil cooler to the required temperature.



System components

01	Diesel engine Wärtsilä L46/V46
02	Pressure control valve
03	Centrifugal oil cleaner
04	Oil cooler
05	Thermostatic valve
06	Safety filter
07	Orifice
08	Automatic filter
09	Lube oil pump, Engine driven
10	Lube oil stand-by pump
11	Separator pump
12	Heater
13	Separator
14	System oil tank
15	Sludge oil tank
16	Crankcase breather
18	Prelubricating oil pump
20	Condensate trap
21	Damper (only 12V46)

Figure 1.4 Lubricating oil system with engine driven pumps for L and V type engine (Wärtsilä marine project guide 2001, 74)

2 ENERGY CONSUMPTION OF THE PUMPS

2.1 Fuel system

In HFO separating system pumping is done with a transfer pump and a separator feed pump. The transfer pump is used for pumping fuel to the day tank and the separator feed pump feeds fuel to the separator with required flow. In the fuel feed system pumping is done with a feeder pump and a circulating pump. These pumps are used for maintaining the correct pressure in the system and they keep fuel circulating in the system.

To know how much energy is needed for pumping fuel oil, the required flow rate must be known for each pump. The required flow rate can be found in Wärtsilä's guide books or it can be calculated. When the flow rate is known, a right-sized electric motor is needed. Various pump manufacturers have determined a right-sized electric motor for each pump size and these figures can be found in manufacturers' catalogue. In this case all the pumps and the electrical motor were chosen from TERPS database.
There are not exactly right-sized pumps for each flow rate so in this case the next standard size up is selected. (Wärtsilä marine project guide 2001, 56-66)

The transfer pump is dimensioned with flow 30% over the overall fuel oil consumption. (Wärtsilä diesel power plants guide, 21)

Fuel oil consumption

$$F_{E,i} = F_{N,i} \cdot \frac{W_{S,i}}{1000} = 0.174\,\frac{kg}{kWh} \cdot 9450\,kW = 1644.3\,\frac{kg}{h} \qquad (2.1)$$

Fuel consumption = $F_{N,i} = 174\,\frac{g}{kWh}$ (Wärtsilä marine project guide 2001, 20)

Engine output power = $W_{S,i} = 9450\,kW$ (Wärtsilä marine project guide 2001, 20)

Flow rate

$$\frac{F_{E,i}}{\rho} \approx 1.6592\,\frac{m^3}{h}$$

Fuel oil consumption = $F_{E,i} = 1644.3\frac{kg}{h}$

Fuel density = $\rho = 991\frac{kg}{m^3}$ (Wärtsilä marine project guide 2001, 2)

The separator pump is dimensioned according to the flow of the separator with formula 2.3 (Wärtsilä marine project guide 2001, 56) Fuel oil density for exact temperature is calculated according to Wärtsilä's recommendations (Wärtsilä diesel power plant guide, 3) For every temperature degree rise density drops approximately 0.64kg/m³ (formula (2.2)

$$\rho = 991\frac{kg}{m^3} - ((t - 15^\circ C)\cdot 0.64\frac{kg}{m^3}) = 991\frac{kg}{m^3} - ((98^\circ C - 15^\circ C)\cdot 0.64\frac{kg}{m^3}) = 937.88\frac{kg}{m^3} \quad (2.2)$$

Desired temperature = $t = 98^\circ C$

$$Q = \frac{P\cdot b\cdot 24\,h}{\rho\cdot t} = \frac{9450\,kW\cdot 0.1978\,\frac{kg}{kWh}\cdot 24\,h}{937.88\,\frac{kg}{m^3}\cdot 23\,h} = 2.07\,\frac{m^3}{h} \qquad (2.3)$$

Max continuous rating = $P = 9450\,kW$

Fuel consumption + 15 percent margin = $b = 197.8\frac{g}{kWh}$

Fuel oil density = $\rho = 937.88\frac{kg}{m^3}$ at 98ºC

Daily separating time = $t = 23\,h$

The feeder pump is dimensioned to cover fuel oil consumption and the flow of the automatic filter. The feeder pump is located in the feeder and booster unit which comes as a whole unit. The circulating pump is dimensioned to cover fuel oil consumption with a circulating factor, which is 4 (Wärtsilä marine project guide 2001, 20). In other words four times the fuel oil consumption means that the required flow is 6.6 m³/h.

The circulating pump capacity is chosen according to Wärtsilä's recommendations (Wärtsilä marine project guide 2001, 20)

Table 2.1 Required flows

Pump	Selected flow rate (m³/h)	Required flow (m³/h)	Electric motor (kW)
Transfer pump	8.00	8.00	3.00
Separator pump	2.15	2.07	1.33
Feeder pump	2.00	1.96	3.00
Circulating pump	7.00	6.60	7.55
Total	**19.2**	**18.6**	**14.8**

The actual power consumption of the pumps is calculated using known pressures of the pumps and required liquid flow for each pump (Table 2.2). All the calculated pumps are screw pumps and for all pumps the average efficiency 60% is used. As the pump's efficiency is average it is important to use the same value in every calculation so that the results are comparable.

The power consumption (P_{req}) for each pump in all three systems is calculated with formula 2.4 (Wirzenius 1978, 48) For example, for the separator pump

$$P_{req} = \frac{\rho \cdot g \cdot Q \cdot h}{1000 \cdot 3600 \cdot \eta} = \frac{937.88\,\frac{kg}{m^3} \cdot 9.81\,\frac{m}{s^2} \cdot 2.07\,\frac{m^3}{h} \cdot 50\,m}{1000 \cdot 3600 \cdot 0.6} = 0.469\,kW \qquad (2.4)$$

Density $= \rho = 937.88\,\frac{kg}{m^3}$ at 98ºC

Acceleration of free fall $= g = 9.81\,\frac{m}{s^2}$

Flow $= Q = 2.07\,\frac{m^3}{h}$

Lifting height of the pump $= h = 50\,m$

Efficiency $= \eta = 0.6$

Table 2.2 Power consumption

Pump	Power consumption (kW)	Pressure (bar)
Transfer pump	1.703	4
Separator pump	0.469	5 (max)
Feeder pump	0.317	4
Circulating pump	1.106	4
Total	**3.595**	

2.2 Lubricating oil system

Pumping in the lubricating oil system is done with a lubricating oil pump, a prelubricating oil pump and a separator pump. Main lubricating oil pump is a directly driven pump or electrically driven screw type pump and it is dimensioned according to Wärtsilä's recommendations (Wärtsilä marine project guide 2001, 20). If the main lubricating oil pump is engine driven, the capacity differs a bit from the electrically driven pump, because also at low speeds engine driven pumps have to be able to cover the lubricating oil need. The prelubricating oil pump is also an electrically driven screw pump and it is dimensioned according to Wärtsilä's recommendations (Wärtsilä marine project guide 2001, 20). It is used for filling the lubricating system with lubricating oil and getting some pressure before starting. This pump is not constantly running so the pumping energy consumption must be calculated without it. The separator pump can be directly driven by the separator or separately driven by an electric motor. The flow of the separator is calculated with formula 2.5 (Wärtsilä marine project guide 2001, 70)

The separator pump is dimensioned according to the flow of the separator.

$$V = \frac{(1.2...1.5)\cdot P\cdot m}{23} = \frac{1.3\cdot 9450\ kW\cdot 5}{23} = 2670.7\frac{l}{h} \approx 2.7\frac{m^3}{h} \qquad (2.5)$$

M-value (depends which fuel is used) $m = 5\ \ for\ HFO$ (Wärtsilä recommendations)

Engine output power = $P = 9450\ kW$

Separation time = 23h/day

There is an option to choose the main lubricating oil pump to be engine driven so calculations have to be made for both cases (Table 2.3). The required flow rates are Wärtsilä's recommended values. The prelubricating oil pump is not taken into account, because it is not running at the same time with the other pumps.

Table 2.3 Electrically and engine driven LO pumps

Pump	Selected flow rate (m^3/h)	Req. flow (m^3/h)	E. motor (kW)
LO pump (electrically driven)	165.0	160.0	75.0
LO pump (engine driven)	198.0	198.0	
Separator pump	3.00	2.7	1.50
Total (electrically driven LO pump)	**168.0**	**162.7**	**76.5**
Total (engine driven LO pump)	**201.0**	**200.7**	**1.5**

The actual power consumptions of the pumps were calculated with formula 2.4 using known pressures of the pumps and required liquid flow for each pump (Table 2.4). All the calculated pumps are screw pumps and for all pumps the average efficiency 60% is used.

Table 2.4 Power consumption of LO pumps

Pump	Power consumption (kW)	Pressure (bar)
LO pump (electrically driven)	25.57	4.0
LO pump (engine driven)	31.65	4.0
Separator pump	0.16	1.5
Total (electrically driven LO pump)	**25.74**	
Total (engine driven LO pump)	**31.82**	

2.3 Cooling water system

Circulating pumps in a high temperature system and in a low temperature system are centrifugal type pumps and they can be driven by an electric motor or the engine. For both systems there is one main pump and one stand-by pump. The delivery heads for these pumps are determined considering flow resistances in the engine, pipelines and valves. Capacities for these pumps for 9L46 engine are determined by Wärtsilä's recommendations. In both systems the flow rate is $200\, m^3/h$ (Wärtsilä marine project guide 2001, 23).

The preheater pump is dimensioned according to Wärtsilä's recommendations for the required flow (Wärtsilä marine project guide 2001, 83). It depends on the quantity of the cylinders.

$$1.6\frac{m^3}{h}\cdot cyl.=14.4\frac{m^3}{h}$$

$$cyl=9$$

The transfer pump is used for emptying and filling the drain tank. Engines and coolers can be drained to the drain tank if they need maintenance.

The required sea water pump flow is normally 20-50 percent higher than fresh water flow. (Wärtsilä marine project guide 2001, 20)

The circulating pumps in the HT and LT systems are centrifugal pumps. Like in the lubricating oil system there is an optional engine driven circulating pump in both the LT and HT system.

Table 2.5 HT and LT pumps

Pump	Pump flow rate (m³/h)	Electric motor (kW)	Required flow (m³/h)
Circulating pump (HT) e. motor driven	200.0	22.0	200.0
Circulating pump (LT) e. motor driven	200.0	22.0	200.0
Circulating pump (HT) engine driven	200.0		200.0
Circulating pump (LT) engine driven	200.0		200.0
Sea water pump	320.0	37.0	300.0
Total (electrically driven pumps)	**720.0**	**81.0**	**700.0**
Total (engine driven pumps)	**720.0**	**37.0**	**700.0**

If the circulating pumps are engine driven the pressure is 0.3 bars higher than on the electrically driven pumps, because for the manufacturing reasons Wärtsilä has chosen impeller standards between 10 millimetres. In all centrifugal pumps average 60 percent efficiency is used. The actual power consumptions are calculated using formula 2.4.

Table 2.6 Power consumption of HT and LT pumps

Pump	Power consumption (kW)	Pressure (bar)
Circulating pump (HT) e. motor driven	23.148	2.5
Circulating pump (LT) e. motor driven	23.148	2.5
Circulating pump (HT) engine driven	25.926	2.8
Circulating pump (LT) engine driven	25.926	2.8
Sea water pump	19.230	1.5
Total (electrically driven pumps)	**65.527**	
Total (engine driven pumps)	**71.083**	

2.4 Choosing one system

Based on the calculations above the most energy consuming liquid pumping system is the cooling water system. Liquid flows are a bit bigger in comparison in the cooling water system than on the other two systems. The Lubricating oil system is almost as consuming as the cooling water system. The chart below shows all three systems and the differences if the pump is either electrically driven or engine driven.

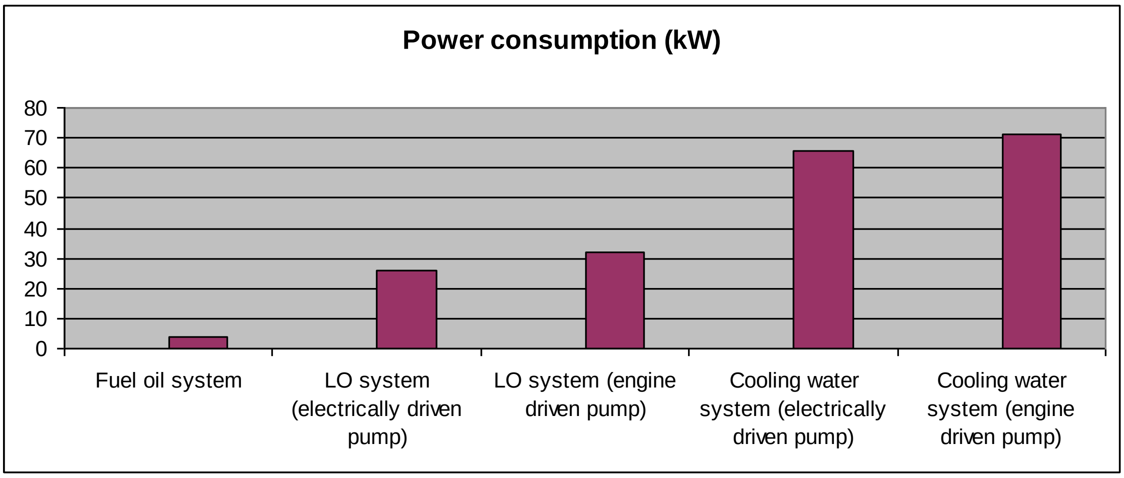


Chart 2.1 Power consumptions of different auxiliary systems

3 CLOSER OBSERVATION OF THE CHOSEN SYSTEM

3.1 HT system pressure losses

In the cooling water system the pumps are dimensioned to cover total pressure losses of the system which consists of pressure losses of the components and pipelines. Also the static pressure produced by the expansion tank is taken into account. If the pressures created by the pumps are larger than pressure losses, orifices must be used to reduce the pressure. This causes energy losses in the system, and an ideal system would be a system without the orifices.

Table 3.1 Pressure losses in the HT system (Wärtsilä marine project guide 2001, 24)

Component	Bar
Engine	0.5
Charge air cooler	0.2
Piping	0.5
Thermostatic valve x2	0.4
Total pressure loss	**1.6**
Orifice needed	**0.9**

Pressure before engine has to be between 3.2 bars and 4.8 bars (Wärtsilä marine project guide 2001, 24) Static pressure variation depends on the height the expansion tank is installed. It is between 0.7 bars and 1.5 bars (Wärtsilä marine project guide 2001, 83). It also determined that the delivery head for both the HT and LT system circulating pump is 2.5 bars (Wärtsilä marine project guide 2001, 83). Pressure losses for the pipelines is chosen to be 0.5 bars which is a rough average value, but due to the fact that pressure losses vary a lot based on the installation design, this average value was chosen. If pressure losses from system components and pipelines are 1.6 bars in total, which is 0.9 bars less than the delivery head of the pump, this has to be reduced with orifices so that the system is in balance. In this case total energy losses are 39 percent caused by the orifices.

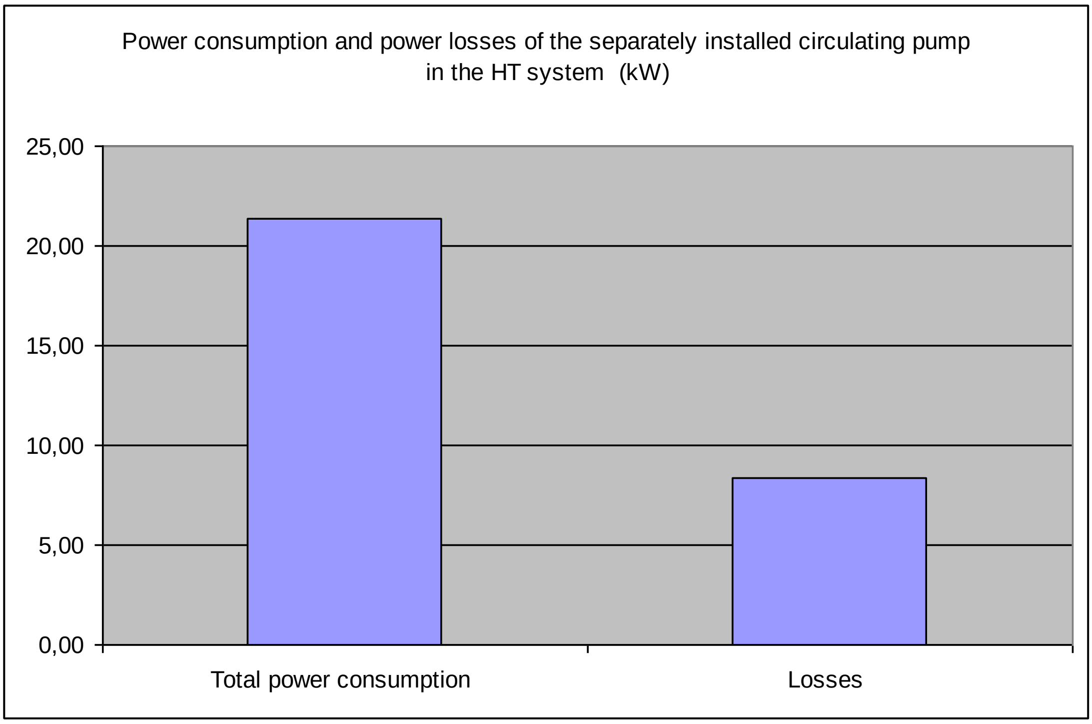


Chart 3.1 Power consumption and losses of the separately installed pump in the HT system

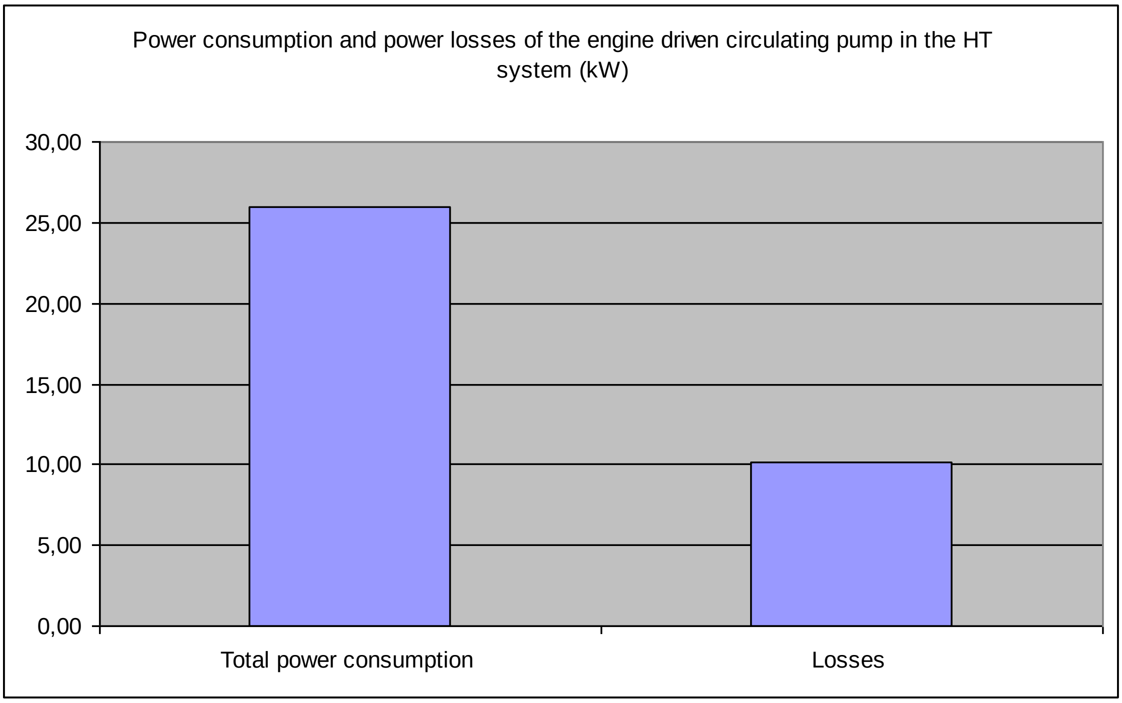


Chart 3.2 Power consumption and losses of engine driven pump in the HT system

3.2 Pressure losses in LT system

The LT system is much more balanced with given starting values as shown in the charts 3.3 and 3.4 for separately installed pump and for engine driven pump. Only 0.2 bars is needed to cover with orifices (Table 3.3 and 3.4) so there is only 8 percent losses.

Table 3.2 Pressure losses in the LT system (Wärtsilä marine project guide 2001, 24)

Component	Bar
Central cooler	0.6
Piping	0.5
Lubricating oil cooler	0.5
Thermostatic valve x2	0.4
Charge air cooler	0.3
Total pressure loss	**2.3**
Orifice needed	**0.2**

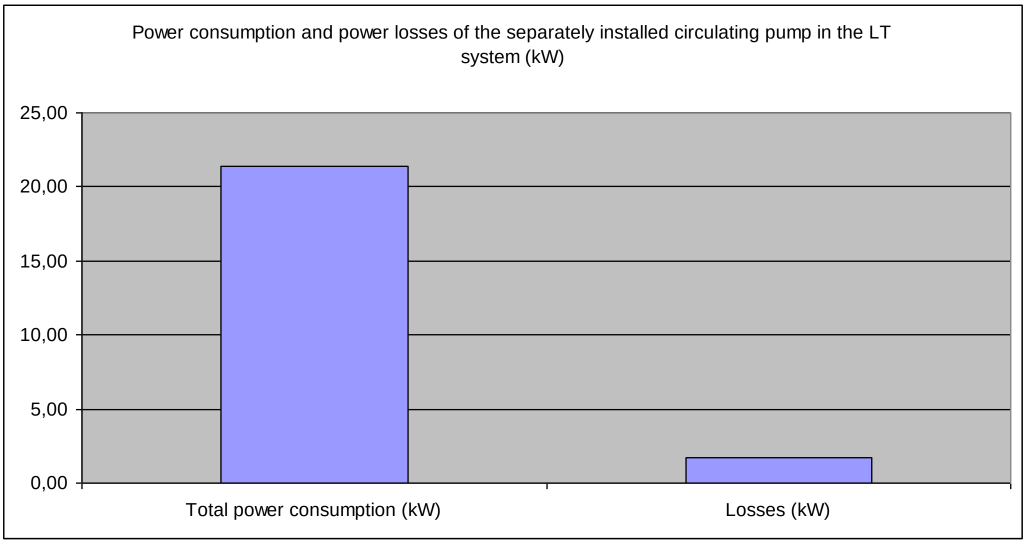


Chart 3.3 Power consumption and losses of separately installed pump in the LT system

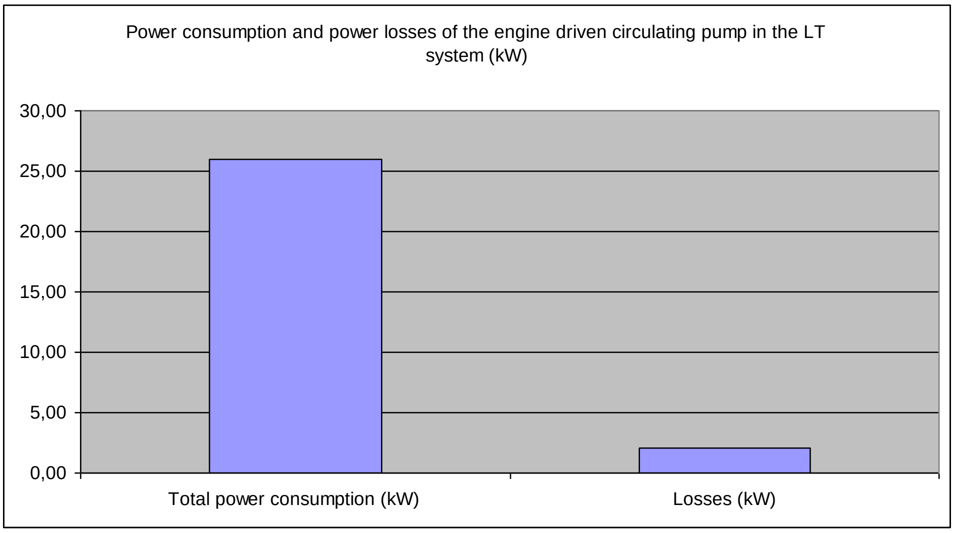


Chart 3.4 Power consumption and losses of engine driven pump in the LT system

3.3 Pump dimensioning

The backpressure of the HT system being smaller than the pressure caused by the HT circulating pump, a smaller delivery head for the pump is needed. In the figure 3.1 it is shown that the flow rate remains the same but the delivery head is decreased. This can be done by choosing a whole new pump with a smaller delivery head or adjusting the impeller on the existing pump. When the delivery head of the pump is reduced the engine inlet pressure is not according to Wärtsilä's standards anymore. This must be compensated by lifting the expansion tank to keep the right inlet pressure at the engine.

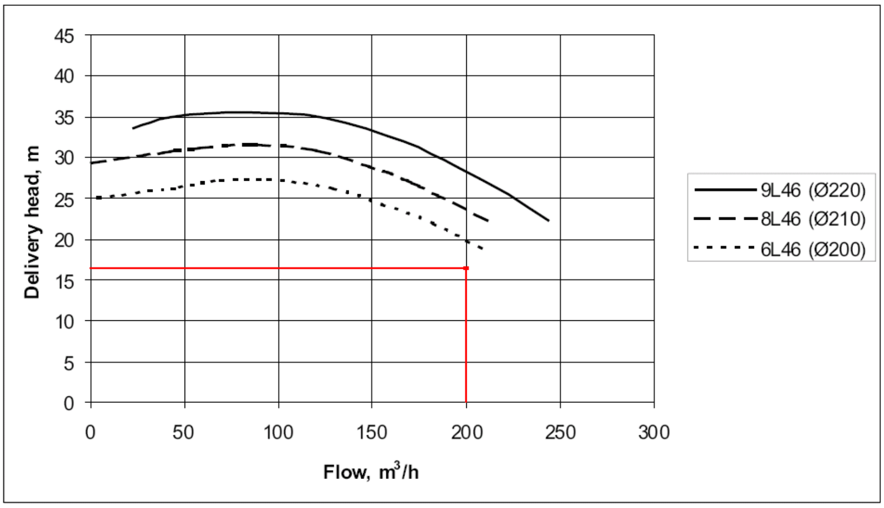


Figure 3.1 Pump curve of the HT centrifugal pump

3.4 Reduced LT flow

One way to cut down the power consumption is to alter the LT flow by reducing it. The chosen reduction was half from the original flow. The LT flow is cut down to half by simply choosing a smaller pump. Reducing the flow will affect the size of the coolers in the LT circuit. Cooling water temperatures on the inlet and outlet of the coolers will be altered.

3.4.1 LT circulating pump

The power consumption is calculated in table 3.2 with formula 2.4. Only the flow and pressure are altered. When the flow is cut down to half the back pressure is cut down to ¼ as shown below.

$$H_{normal\,flow} = \frac{v^2}{2g} = \frac{200\,\frac{m^3}{h}}{2 \cdot 9.81\,\frac{m}{s^2}} = 2038.7$$

$$H_{reduced\,flow} = \frac{v^2}{2g} = \frac{100\,\frac{m^3}{h}}{2 \cdot 9.81\,\frac{m}{s^2}} = 509.6$$

v = Flow rate ($\frac{m^3}{h}$)

g = Acceleration of free fall ($\frac{m}{s^2}$)

The back pressure on reduced flow is four times smaller than on the normal flow.

$$\frac{H_{normal\,flow}}{H_{reduced\,flow}} = \frac{2038.7}{509.6} = 4$$

Table 3.2 Power consumptions on reduced flow

Pump	Flow (m^3/h)	Pressure (bar)	Power consumption (kWh)
Circulating pump (LT) e. motor driven	100.0	0.625	2.82
Circulating pump (LT) engine driven	100.0	0.700	3.15

3.4.2 Coolers

When the flow is cut to half the lubricating oil cooler and the central cooler need to be dimensioned to fit the new flow. The effect on the second stage of the charge air cooler is not taken into account, because altering the flow has no significant effect, hence the charge air cooler is over-dimensioned.

Dimensioning of the coolers' k-values (Heat transfer coefficient) is needed. These values are dependable of so many different variables, for example, the material of the cooling plates, that for this work some average values must be used. In this work these k-values are based on the actual coolers. Knowing the heat balances (Table 3.3) for each component and flow, the temperature difference over each component can be calculated with formula 3.1 (Vihinen & Korhonen 1988, 25). When the temperature difference is known and before calculating the cooling area, the logarithmic mean temperature difference must be calculated (Formula 3.3). When the logarithmic mean temperature difference is known, the cooling area is calculated using the specific heat flow of the lubricating oil cooler, k-value and logarithmic mean temperature (Formula 3.4)

Table 3.3 Heat balances at ISO conditions (Wärtsilä marine project guide 2001, 20)

Section	Heat balance (kW)
Lubricating oil	1210
Jacket water HT-circuit	970
Charge air HT-circuit	1780
Charge air LT-circuit	1100
Radiation	360
Combinet	**5420**

3.4.2.1 Charge air cooler

Cooling water enters the charge air cooler at 38 ºC temperature (Wärtsilä marine project guide 2001, 24) and the heat balance of the charge air cooler is 1100 kW (see table 3.3). Temperatures over charge air cooler are

$$\Delta T_{reduced\,flow} = \frac{Q \cdot 1000}{q \cdot C_p \cdot \rho} = \frac{1110\,kW \cdot 1000}{100\,\frac{m^3}{h} \cdot 1.163\,\frac{kJ}{kg^\circ C} \cdot 990\,\frac{kg}{m^3}} = 9.55^\circ C \qquad (3.1)$$

$$\Delta T_{normal flow} = \frac{Q \cdot 1000}{q \cdot C_p \cdot \rho} = \frac{1110\,kW \cdot 1000}{200\,\frac{m^3}{h} \cdot 1.163\,\frac{kJ}{kg^\circ C} \cdot 990\,\frac{kg}{m^3}} = 4.77^\circ C \qquad (3.1)$$

$Q = Heat\ \ flow\,(kW)$

$q = Flow\ rate\ (\frac{m^3}{h})$

$C_p\ = Specific\ heat\ capacity\ \frac{kJ}{kg^\circ K}$

$\rho = density\ \frac{kg}{m^3}$

Table 3.4 Temperatures over charge air cooler second stage

	In (ºC)	Out (ºC)
Reduced flow 100m³/h	38	48
Normal flow 200m³/h	38	43

3.4.2.2 Lubricating oil cooler

The temperature differences over lubricating oil cooler are calculated the same as the temperature differences over charge air cooler with formula 3.1.
The cooling water outlet temperature can be calculated using the heat flow (table 3.3). The temperature difference is calculated with formula 3.1 and added to the water inlet temperature (table 3.5)

The temperature differences over lubricating oil cooler

$$\Delta T_{reduced\ flow} = \frac{Q \cdot 1000}{q \cdot C_p \cdot \rho} = \frac{1210\ kW \cdot 1000}{100\ \frac{m^3}{h} \cdot 1.163\ \frac{kJ}{kg^\circ K} \cdot 990\ \frac{kg}{m^3}} = 11^\circ C \qquad (3.1)$$

$$\Delta T_{normal\ flow} = \frac{Q \cdot 1000}{q \cdot C_p \cdot \rho} = \frac{1210\ kW \cdot 1000}{200\ \frac{m^3}{h} \cdot 1.163\ \frac{kJ}{kg^\circ K} \cdot 990\ \frac{kg}{m^3}} = 5^\circ C \qquad (3.1)$$

Table 3.5 Temperature differences over lubricating oil cooler

	In (ºC)	Out (ºC)
Reduced flow 100m³/h	48	59
Normal flow 200m³/h	43	48

Logarithmic mean temperature

The lubricating oil temperature at the lubricating oil cooler inlet is 75 ºC as it is almost the same as in the system oil tank and the outlet temperature is 63 ºC (Wärtsilä marine project guide 2001, 69).

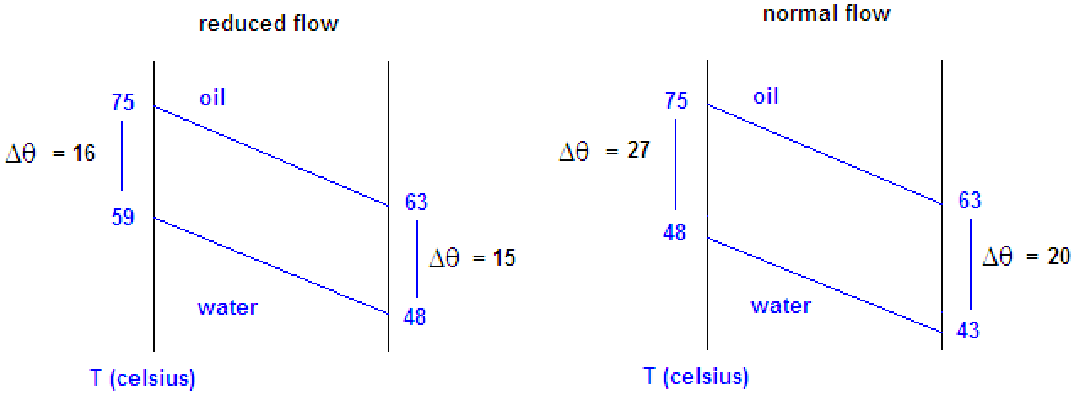


Figure 3.2 Temperature differences on the lubricating oil cooler

$$(\Delta\theta_{\ln})_{reduced\,flow} = \frac{\Delta\theta_1 - \Delta\theta_2}{\ln\frac{\Delta\theta_1}{\Delta\theta_2}} = \frac{16^\circ C - 15^\circ C}{\ln\frac{16^\circ C}{15^\circ C}} = 15.49^\circ C \quad (3.3)$$

$$(\Delta\theta_{\ln})_{normal\,flow} = \frac{\Delta\theta_1 - \Delta\theta_2}{\ln\frac{\Delta\theta_1}{\Delta\theta_2}} = \frac{27^\circ C - 20^\circ C}{\ln\frac{27^\circ C}{20^\circ C}} = 23.33^\circ C \quad (3.3)$$

$\Delta\theta_1 = Lubricating\ oil\ temp.in - Fresh\ water\ temp.out$
$\Delta\theta_2 = Lubricating\ oil\ temp.out - Fresh\ water\ temp.\ in$

Cooling area

The cooling area is calculated using the specific heat flow of the lubricating oil cooler, k-value and logarithmic mean temperature (formula 3.4)

$$A_{reduced\,flow} = \frac{Q}{k \cdot \Delta\theta_{\ln}} = \frac{1210\,kW}{0.8 \cdot 15.49\,^\circ C} = 97.64\,m^2 \qquad (3.4)$$

$$A_{normal\,flow} = \frac{Q}{k \cdot \Delta\theta_{\ln}} = \frac{1210\,kW}{0.8 \cdot 23.33\,^\circ C} = 63.83\,m^2 \qquad (3.4)$$

$Q = Heat\ flow kW$
$k = k - value$
$\Delta\theta_{\ln} = \log arithmic\ mean\ temperature$

3.4.2.3 Central cooler

Before the central cooler the HT water mixes to the LT water. The temperature difference, after the HT water is mixed to the LT water, can also be calculated with formula (3.1). The heat flow is jacket water HT circuit and charge air HT circuit combined (Table 3.3)

$$\Delta T_{reduced\,flow} = \frac{Q \cdot 1000}{q \cdot C_p \cdot \rho} = \frac{2750\,kW \cdot 1000}{100\,\frac{m^3}{h} \cdot 1.163\,\frac{kJ}{kg^{\circ}K} \cdot 990\,\frac{kg}{m^3}} = 23.88^{\circ}C \qquad (3.1)$$

$$\Delta T_{normal\,flow} = \frac{Q \cdot 1000}{q \cdot C_p \cdot \rho} = \frac{2750\,kW \cdot 1000}{200\,\frac{m^3}{h} \cdot 1.163\,\frac{kJ}{kg^{\circ}K} \cdot 990\,\frac{kg}{m^3}} = 11.94^{\circ}C \qquad (3.1)$$

$Q = Heat\ \ flow\,(kW)$

$q = Flow\ rate\ (\frac{m^3}{h})$

$C_p\ = Specific\,heat\ capacity\,\frac{kJ}{kg^{\circ}K}$

$\rho = density\,\frac{kg}{m^3}$

According to Wärtsilä’s recommendations the inlet temperature of the LT water at the engine is 38 ºC (Table 3.6).

Table 3.6 Temperature differences over central cooler

	In (ºC)	Out (ºC)
Reduced flow 100m³/h	83	38
Normal flow 200m³/h	60	38

Logarithmic mean temperature

The sea water inlet temperature to the central cooler is 32 ºC and the seawater temperature after the central cooler is 45 degrees due to Wärtsilä’s recommendations to prevent corrosion.

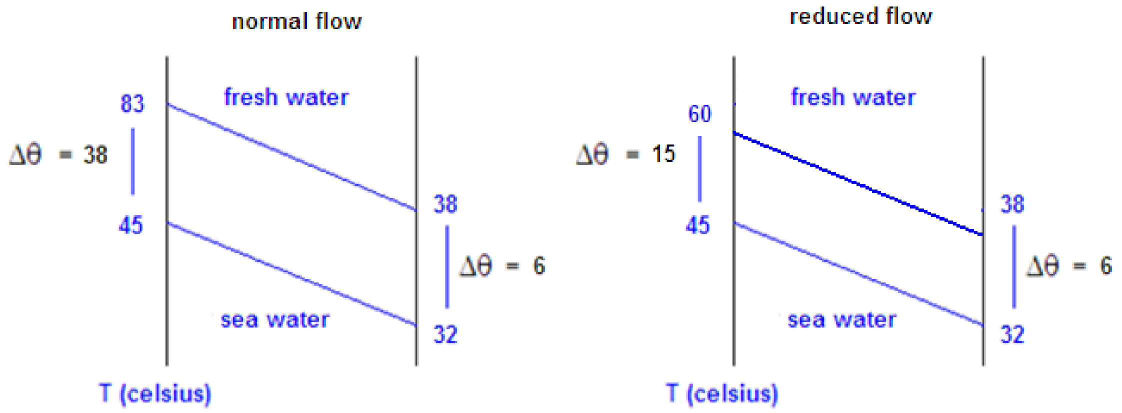


Figure 3.3 Temperature differences on the central cooler

$$(\Delta\theta_{\ln})_{reduced\ flow} = \frac{\Delta\theta_1 - \Delta\theta_2}{\ln\dfrac{\Delta\theta_1}{\Delta\theta_2}} = \frac{38^\circ C - 6^\circ C}{\ln\dfrac{38^\circ C}{6^\circ C}} = 17.33^\circ C$$

$$(\Delta\theta_{\ln})_{normal\ flow} = \frac{\Delta\theta_1 - \Delta\theta_2}{\ln\dfrac{\Delta\theta_1}{\Delta\theta_2}} = \frac{15^\circ C - 6^\circ C}{\ln\dfrac{15^\circ C}{6^\circ C}} = 10.90^\circ C$$

$$\Delta\theta_1 = Fresh\,water\,temp.in - Sea\,water\,temp.out$$

$$\Delta\theta_2 = Fresh\,water\,temp.out - Sea\,water\,temp.\ in$$

Cooling area

$$A_{reduced\ flow} = \frac{Q}{k\cdot\Delta\theta_{\ln}} = \frac{5420\ kW}{5\cdot 17.33\ ^\circ C} = 62.55\,m^2$$

$$A_{normal\,flow} = \frac{Q}{k\cdot\Delta\theta_{\ln}} = \frac{5420\ kW}{5\cdot 10.90\ ^\circ C} = 99.44\,m^2$$

$$Q = Heat\ flow kW$$

$$k = k - value$$

$$\Delta\theta_{\ln} = \log\,arithmic\ mean\,temperature$$

3.4.4 Cooling area comparison

Comparing reduced LT flow to the normal fresh water flow the temperature difference over lubricating cooler is higher. On the normal flow the cooling area needed is smaller than on the reduced flow. The central cooler cooling area is smaller on the reduced flow than on the normal flow.

Table 3.6 Cooling area

	Cooling area on reduced flow (m^2)	Cooling area on normal flow (m^2)
Lubricating oil cooler	97.64	63.88
Central cooler	62.55	99.44

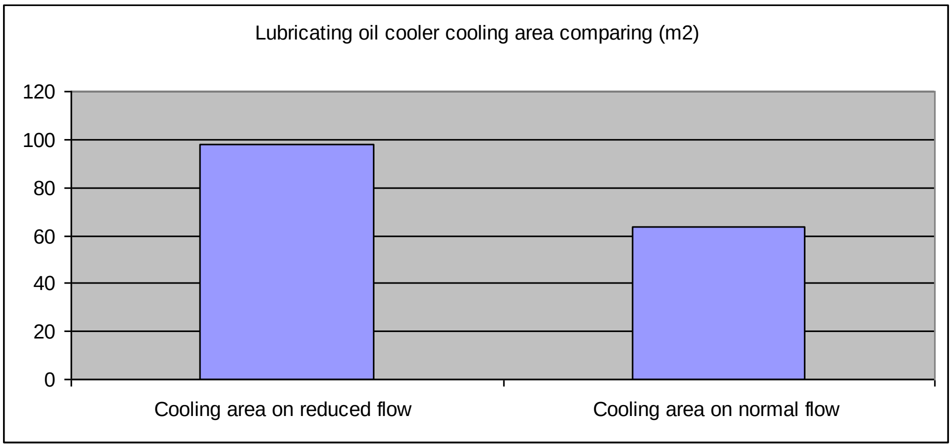


Chart 3.6 Cooling areas of the lubricating oil coolers

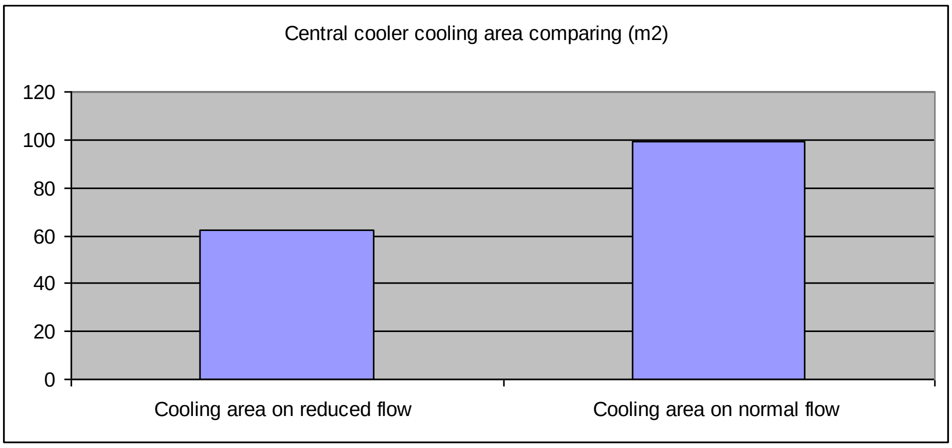


Chart 3.7 Cooling areas central coolers

3.5 Economical evaluation

3.5.1 Circulating pumps on normal flow

Energy saving for the right dimensioned pump or new impeller can be notable. Power losses are calculated in chart 3.1. below 39 percent of the produced power is wasted in the HT system.

The calculation is based for one year usage. The average running hours for marine main engine and therefore for the circulating pump in a year is 6000. When the power losses are known, energy losses can be calculated (Table 3.8)

Table 3.7 Power losses of the HT and LT pumps

Pump	Power consumption (kW)	Losses (kW)
Circulating pump (HT) e. motor driven	22.53	8.79
Circulating pump (HT) engine driven	25.23	9.84
Circulating pump (LT) e. motor driven	22.53	1.80
Circulating pump (LT) engine driven	25.23	2.02

Table 3.8 Energy losses of the HT and LT pumps

Pump	Power consumption (kW)	Losses (kW)	Energy losses (kWh/a)
Circulating pump (HT) e. motor driven	22.53	8.79	57212
Circulating pump (HT) engine driven	25.53	9.84	59040
Circulating pump (LT) e. motor driven	22.53	1.80	10800
Circulating pump (LT) engine driven	25.23	2.02	12110

When the energy losses are known the amount of wasted heavy fuel oil can be calculated. (Table 3.9) For example, with an electrical motor driven HT circulating pump.

$$F_{N,i} \cdot Q_{wasted} = 0.174 \frac{kg}{kWh} \cdot 52712.40\, kWh = 9171\, kg \qquad (3.2)$$

Fuel consumption = $F_{N,i} = 174 \dfrac{g}{kWh}$ (Wärtsilä marine project guide 2001, 20)

Energy wasted = Q_{wasted} = 52712.40 kWh/a

Table 3.9 Wasted fuel

Pump	Fuel wasted (kg/a)
Circulating pump (HT) e. motor driven	9172
Circulating pump (LT) e. motor driven	1881
Circulating pump (HT) engine driven	10273
Circulating pump (LT) engine driven	2107

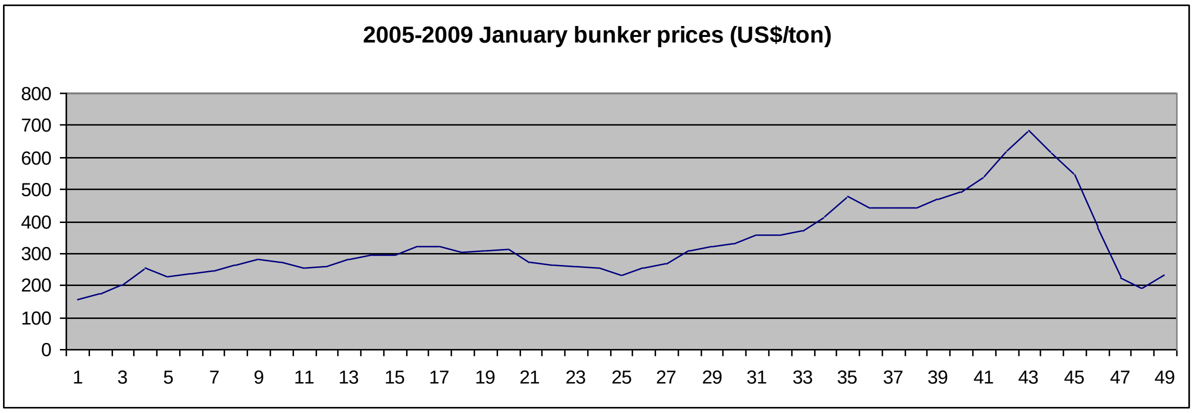


Chart 3.8 Bunker prices

As seen in chart 3.8 bunker prices have changed a lot in few years so the average sum must be calculated to get a rough idea how much money can be saved. For the calculations below an average 333 US dollars per fuel ton is used.

Table 3.10 Losses

Pump	Fuel wasted (kg/a)	US$/a
Circulating pump (HT) e. motor driven	9172	3054
Circulating pump (LT) e. motor driven	1881	627
Circulating pump (HT) engine driven	10273	3421
Circulating pump (LT) engine driven	2107	702

3.5.2 LT circulating pumps on reduced flow

On reduced and normal flow energy consumption was calculated. When the flow rate is dropped down to half, the energy consumption is dropped down (table 3.11). Both flows were compared (Chart 3.9) and energy consumption for both flows was calculated with the same 6000 running hours. The evaluation for possible savings was made when the energy consumption difference between both flows was calculated (Table 3.13). Fuel savings was calculated with formula 3.2 and savings was calculated with the same bunker average price 333 US dollars/ton mentioned in the previous paragraph.

Table 3.11 Power consumption of LT circulating pumps on reduced flow

Pump	Power consumption (kW)	Energy consumption (kWh/a)
Circulating pump (LT) e. motor driven	2.82	16895
Circulating pump (LT) engine driven	3.15	18922

Table 3.12 Energy consumption of LT circulating pumps on normal flow

Pump	Power consumption (kW)	Energy consumption (kWh/a)
Circulating pump (LT) e. motor driven	22.53	135160
Circulating pump (LT) engine driven	25.23	151379

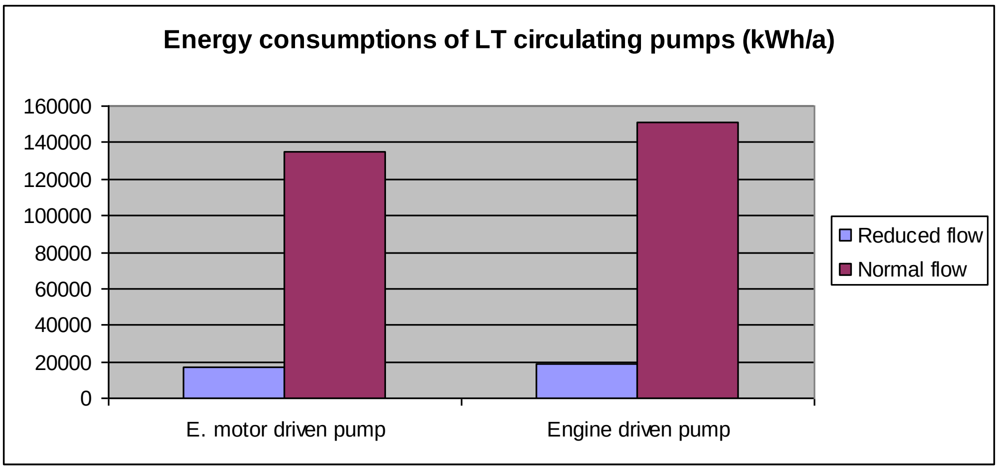


Chart 3.9 Comparing reduced and normal LT flow

Table 3.13 Results for reduced LT flow

Pump	Energy saved (kWh/a)	Fuel savings (kg/a)	Savings (US$/a)
Circulating pump (LT) e. motor driven	118265	20578	6853
Circulating pump (LT) engine driven	132457	23048	7675

3.5.3 Coolers

The cooling area of the reduced and normal flow is almost the same. As the plate material used in the coolers is different in the central coolers than the lubricating oil coolers, a direct comparison cannot be made. The plate material used in central coolers is titan and in the lubricating oil coolers it is basically stainless steel. Titan is more valuable than steel so with this LT flow reduction the cooling area of the central cooler is reduced so some savings although can be made the cooling area of the lubricating oil cooler increases.

According to Alfa laval, which is one of the Wärtsilä's cooler suppliers, the price for the titan plate (0.61m²/plate) is 101 Euros and the price for the steel plate (0.61m²/plate) is 24.7 Euros. The prices for the coolers are calculated in table 3.14 and as seen in chart 3.10 the combined prices of coolers on reduced LT flow are 4676 Euros lower.

Table 3.14 Cooler prices

	Size (m²)	Plates	Price (eur/plate)	Price (eur)
Normal flow LO cooler	63.88	104.70	24.70	2594
Normal flow central cooler	99.44	163.00	101.10	16463
Reduced flow LO cooler	97.64	160.10	24.70	3977
Reduced flow central cooler	62.55	102.50	101.00	10403

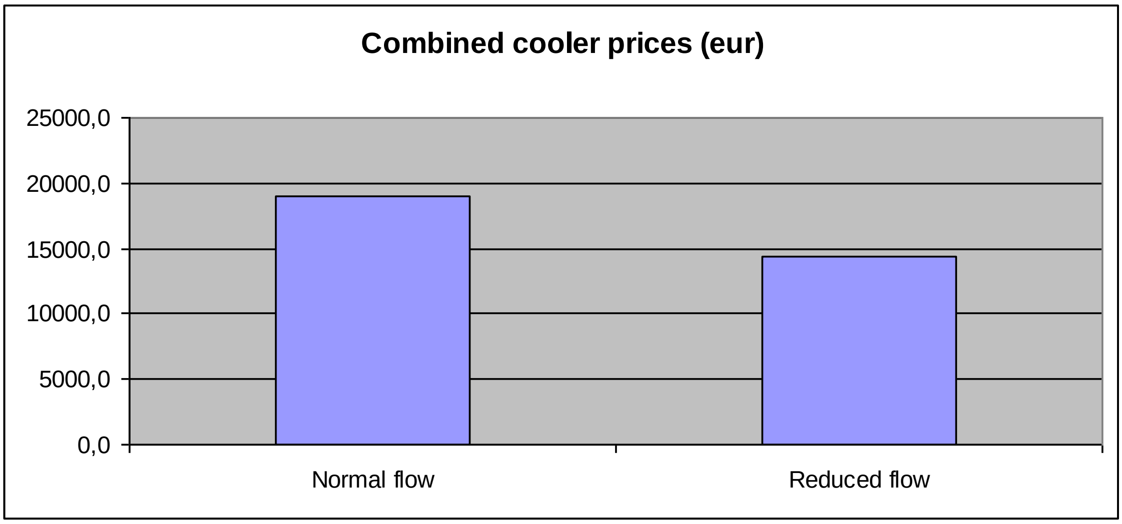


Chart 3.10 Combined cooler prices

4 CONCLUSION

Changing the design of by altering or choosing a new smaller pump or cutting the LT flow down to half can save a reasonable mount of assets. The results were a rough evaluation and the purpose was to get some idea and direction where the savings can be made and how they stand on each other. Exact results are difficult to get, because they depend on so many variables, for example, the heavy fuel oil price, which has changed a lot during the last years.

Small investments made by choosing a new pump or a smaller impeller for balancing the pressures on the HT system can pay it self back in just about one year. This depends on the price of the new pump. Of course, if this is done in the design phase, there will be no need for any extra investments. Dropping the LT flow down to half makes sufficient savings comparing to the normal LT flow. This also affects the coolers in the LT circuit. The lubricating oil cooler temperature difference between the cooling water and the lubricating oil is less on the reduced flow than on the normal flow. This causes the need for more cooling area. Reducing the flow has the exact opposite effect on the central cooler than to the lubricating oil cooler. As the plates of the central cooler are titan and the plates on the lubricating oil cooler are stainless steel this change is profitable.

5 SOURCES

1 Offer from Tage Löfman, Alfa laval

2 TERPS database

3 Vihinen, Simo; Korhonen, Unto 1988, Lämpöoppi

4 Wattsup 1/2009

5 Weekly update meetings with Seppo Rautava

6 Wirzenius, Allan 1978. Keskipakopumput

7 Wärtsilä diesel power plant guide

8 Wärtsilä marine project guide 2001

9 Wärtsilä training material by Per-Erik Andtsjö

Heavy fuel oil prices

(US$ / ton)

		Mina Al Ahmadi	Genoa	Rotterdam	Houston	Cristobal	Los Angeles	Singapore	Japan*
2005	1	183	176	156	183	199	187	180	226.4
	2	197	200	174	176	193	207	199	231.1
	3	212	228	202	200	230	231	229	264.4
	4	248	257	253	244	264	256	259	302.4
	5	256	245	227	253	264	266	257	284.8
	6	260	252	237	254	261	253	258	281.8
	7	268	266	248	248	264	257	261	293.7
	8	281	277	263	267	270	270	277	305.0
	9	311	301	281	305	306	321	312	351.5
	10	302	299	273	296	324	317	318	345.7
	11	287	272	255	266	304	297	299	331.8
	12	279	268	260	281	292	299	279	336.2
2006	1	296	298	280	297	311	315	305	353.6
	2	322	317	296	309	323	325	317	357.1
	3	331	315	294	308	331	326	329	361.3
	4	346	336	322	326	340	336	345	369.5
	5	344	339	324	330	339	356	346	376.9
	6	324	329	302	307	334	326	326	355.0
	7	333	338	310	320	332	343	337	366.1
	8	326	338	311	328	341	331	321	362.0
	9	278	295	274	282	299	296	285	336.3
	10	282	287	265	266	279	287	286	332.1
	11	270	279	259	262	277	285	267	303.0
	12	271	278	256	268	306	289	271	320.5
2007	1	269	265	233	252	267	277	276	324.0
	2	312	284	256	272	286	311	301	346.8
	3	309	298	268	270	304	324	306	349.9
	4	342	337	309	299	308	328	339	372.0
	5	346	347	324	334	361	370	341	366.7
	6	352	358	329	349	362	357	355	387.7
	7	377	387	357	359	370	388	380	419.2
	8	379	397	356	366	383	384	376	417.8
	9	402	403	371	372	382	380	387	421.4
	10	436	446	415	413	425	424	437	460.4
	11	497	504	477	486	500	504	493	550.2
	12	470	468	443	453	475	521	473	510.8
2008	1	481	469	444	464	482	478	470	524.5
	2	476	464	441	456	475	472	466	533.5
	3	499	508	468	487	508	545	489	594.3
	4	528	532	493	495	529	526	520	-
	5	586	581	543	567	578	574	585	-
	6	647	667	623	639	652	675	646	-
	7	727	725	683	717	740	727	724	-
	8	686	671	612	664	701	681	661	-
	9	601	606	545	592	629	612	604	-
	10	417	446	380	404	435	423	410	-
	11	244	267	224	246	287	258	240	-
	12	228	231	191	219	263	241	237	-
2009	1	258	272	232	249	283	265	258	-

Heavy fuel oil density diagram

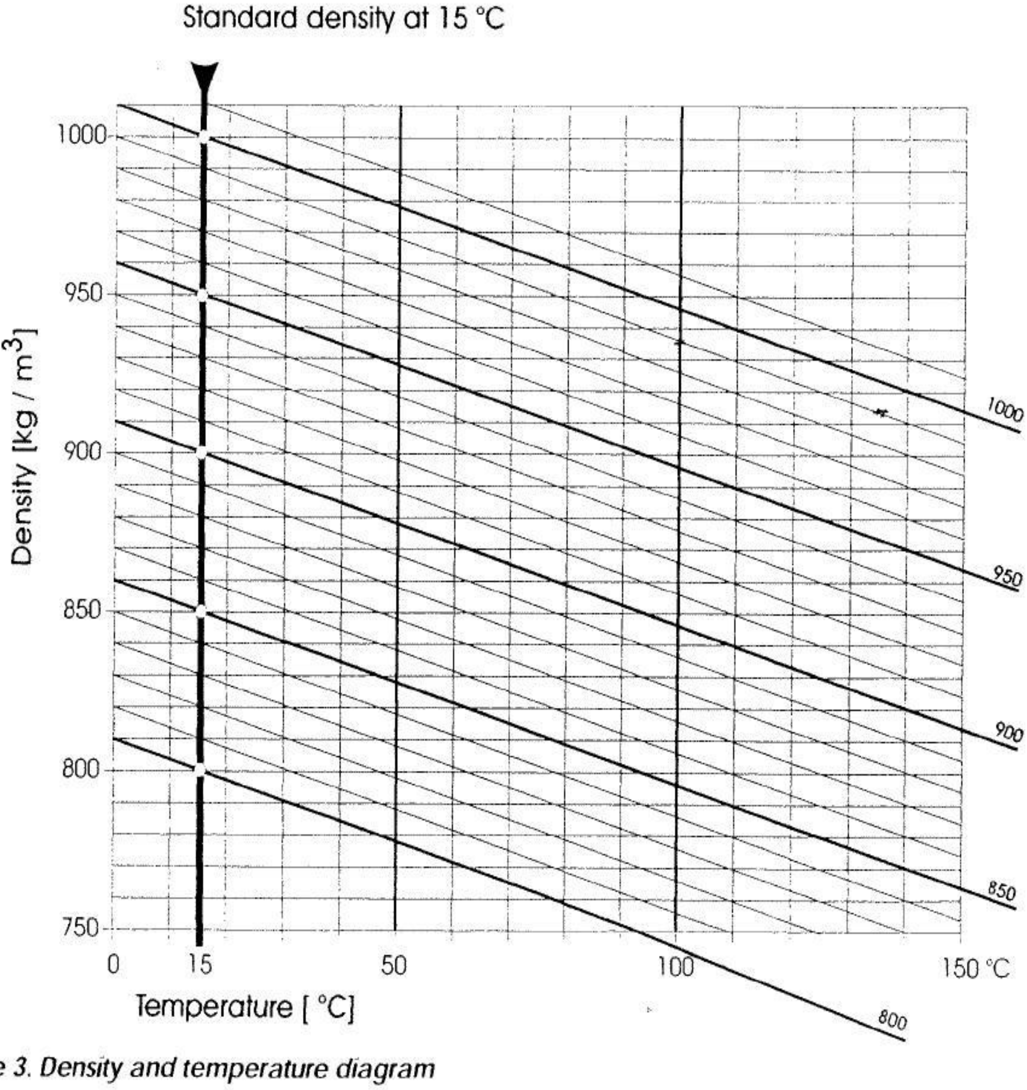
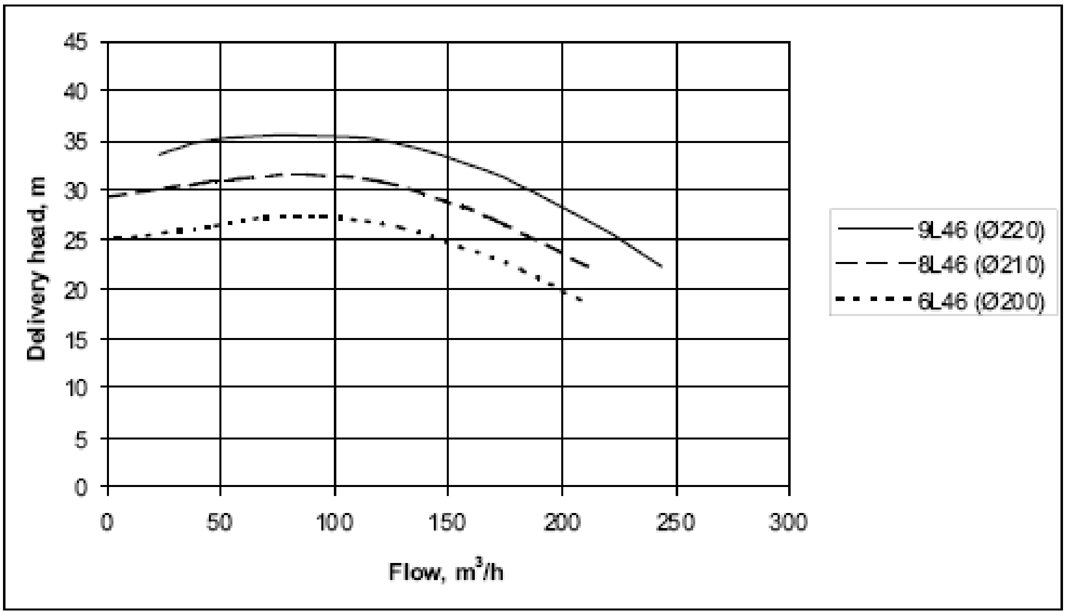


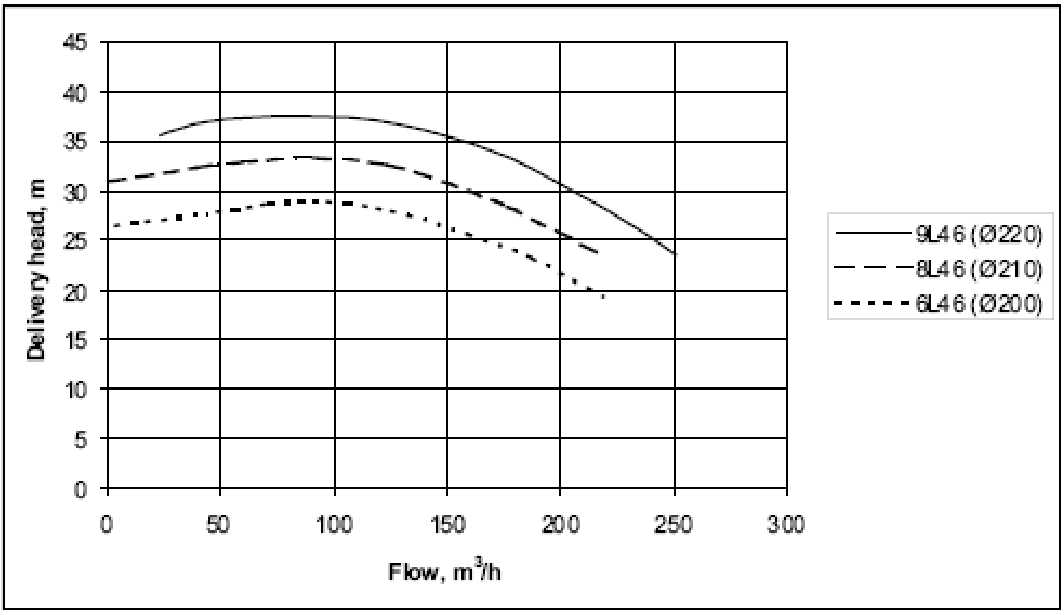
Figure 3. Density and temperature diagram

Pump curves

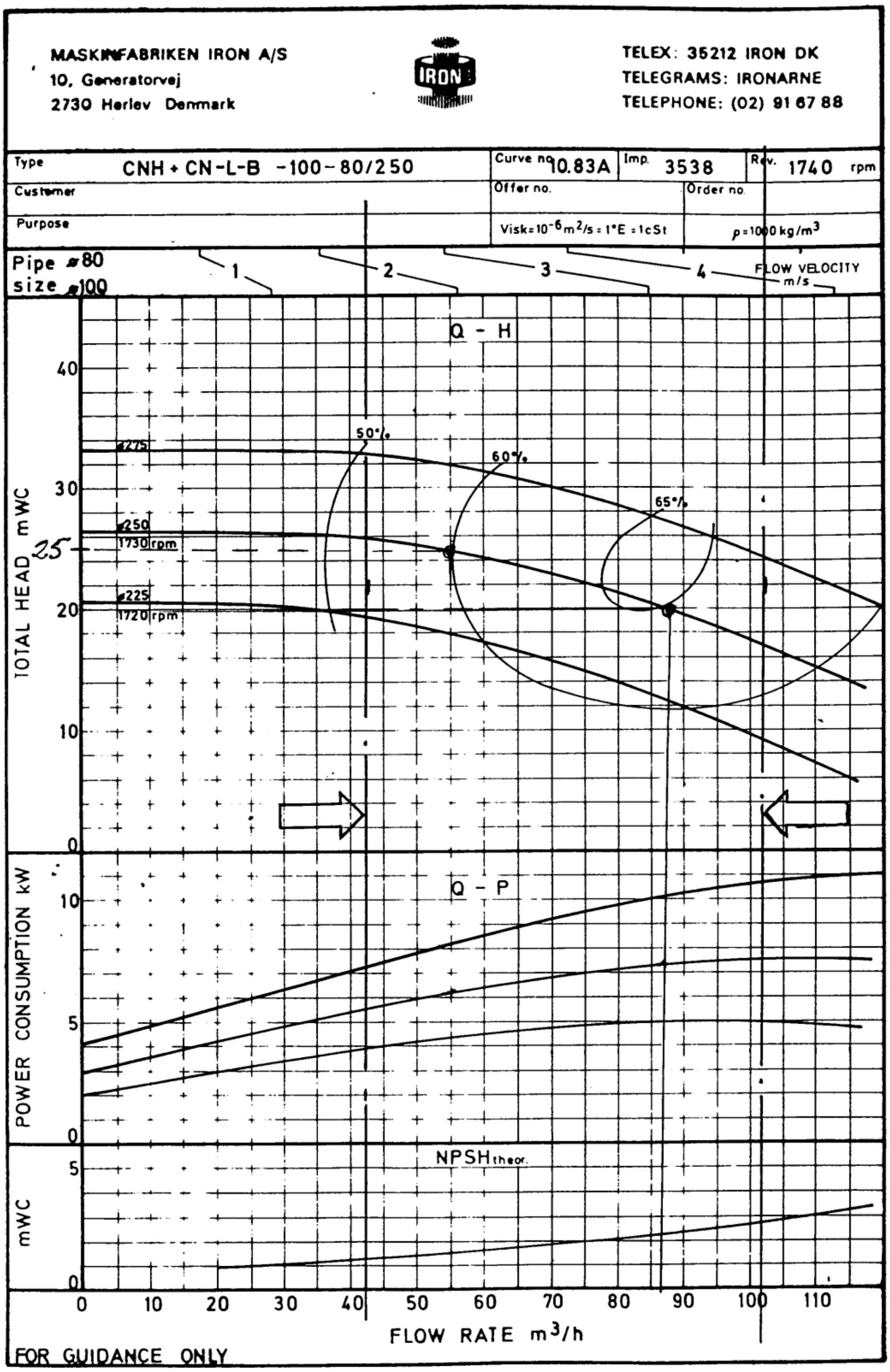
L46, HT and LT - pumps 500 rpm (based on 4V19L0332)



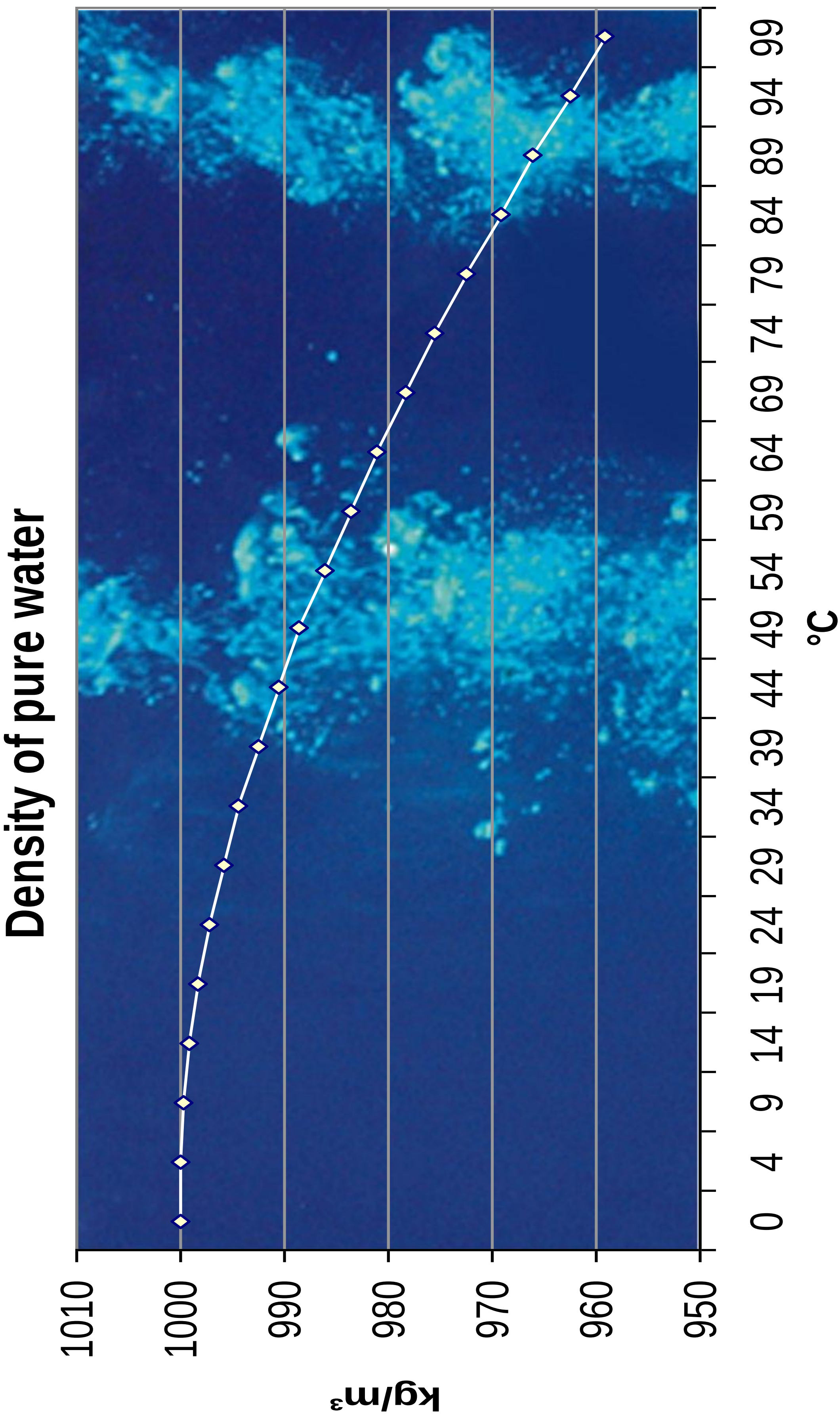
L46, HT- and LT -pumps 514 rpm (based on 4V19L0332)



Pump delivery head and power consumption chart



Water densities at different temperatures



Engine driven lubricating oil pump chart

ALTER.	DATE	MADE	APPVD.	MEMO NO.	EXPLANATION
b	25 11 97	JMO	JJL	D19808	Table (item 1) and item 6 supplemented
c	27 01 98	PHV	SHS	D20520	THREADS OF HOLES ADDED (D300 AND D250)
d	01 03 99	JMO	JJL	D25810	ITEM 7 ADDED (PAGE 2)
e	11 08 99	MSS	JJL	D27719	Added dimensions and views updated

1. TEKNISET TIEDOT: TECHNICAL DATA

Installaatio APPLICATION	Syl. luku CYL. NUMBER	Pyör. nopeus SPEED Moottori ENGINE (rpm)	Öljypumppu OIL PUMP (rpm)	Pumppu tyyppi PUMP TYPE	Pumpun pyör. suunta PUMP DIR. OF ROTATION	Tuotto FLOW (m^3/h)	Ottoteho INPUT POWER	Pos. ITEM Mat. nro. MAT. CODE
VARIABLE SPEED b a	6	300-500	668-1113	L3NG180/185	CCW	97-157	52	00188022901
	8	300-500	668-1113	L3NG200/190	CCW	116-198	65	00188022902
	9	300-500	668-1113	L3NG200/190	CCW	116-198	65	00188022902
	12	300-500	714-1187	L3NG200/240	CCW	158-263	83	00188022904
	16	300-500	714-1187	L3NG200/255	CCW	167-279	90	00188022905
VARIABLE SPEED b a	6	300-500	668-1113	L3NG180/185	CW	97-157	52	00188022907
	8	300-500	668-1113	L3NG200/190	CW	116-198	65	00188022908
	9	300-500	668-1113	L3NG200/190	CW	116-198	65	00188022908
	12	300-500	714-1187	L3NG200/240	CW	158-269	83	00188022910
	16	300-500	714-1187	L3NG200/255	CW	167-279	90	00188022911
CONSTANT SPEED b	6	500/514	1113/1144	L3NG180/175	CCW	149/153	49	00188022906
	8	500/514	1113/1144	L3NG180/175	CCW	149/153	49	00188022906
	9	500/514	1113/1144	L3NG180/185	CCW	157/162	52	00188022901
	12	500/514	1187/1220	L3NG200/190	CCW	215/221	65	00188022902
	16	500/514	1187/1220	L3NG200/240	CCW	263/272	83	00188022904
	18	500/514	1187/1220	L3NG200/255	CCW	279/287	90	00188022905

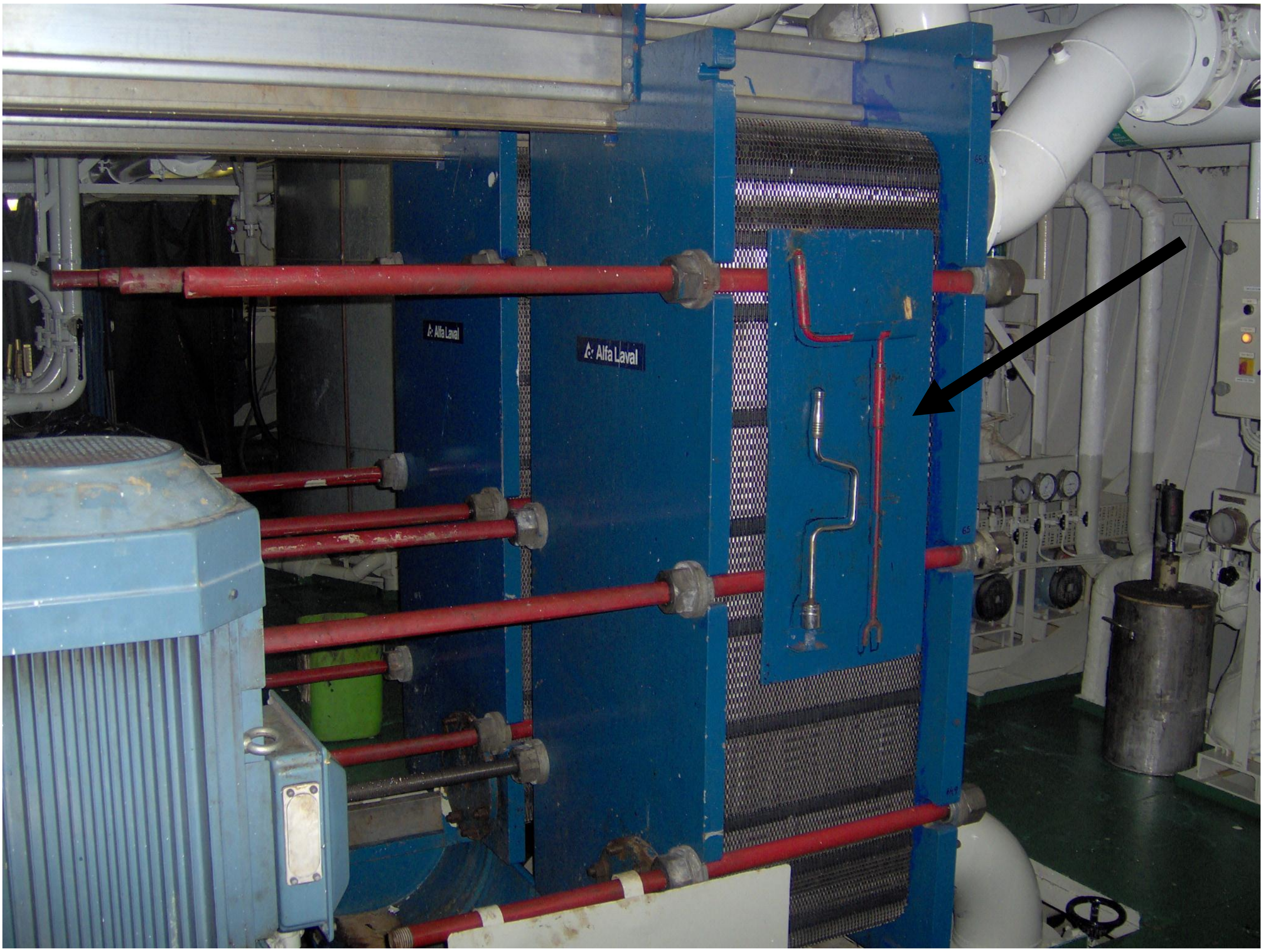
Huom! Myötäpäivään pyörivän moottorin öljypumpun pyörimissuunta on vastapäivään =CCW
NOTE! THE DIRECTION OF ROTATION FOR PUMP IS COUNTER CLOCK WISE =CCW IN CLOCK WISE ROTATING ENGINE

*) Maksimivirtauksella, öljyn viskositeetti 30 cSt, paine 8 bar
MAX. FLOW, OIL VISCOSITY 30 cSt, PRESSURE 8 BAR

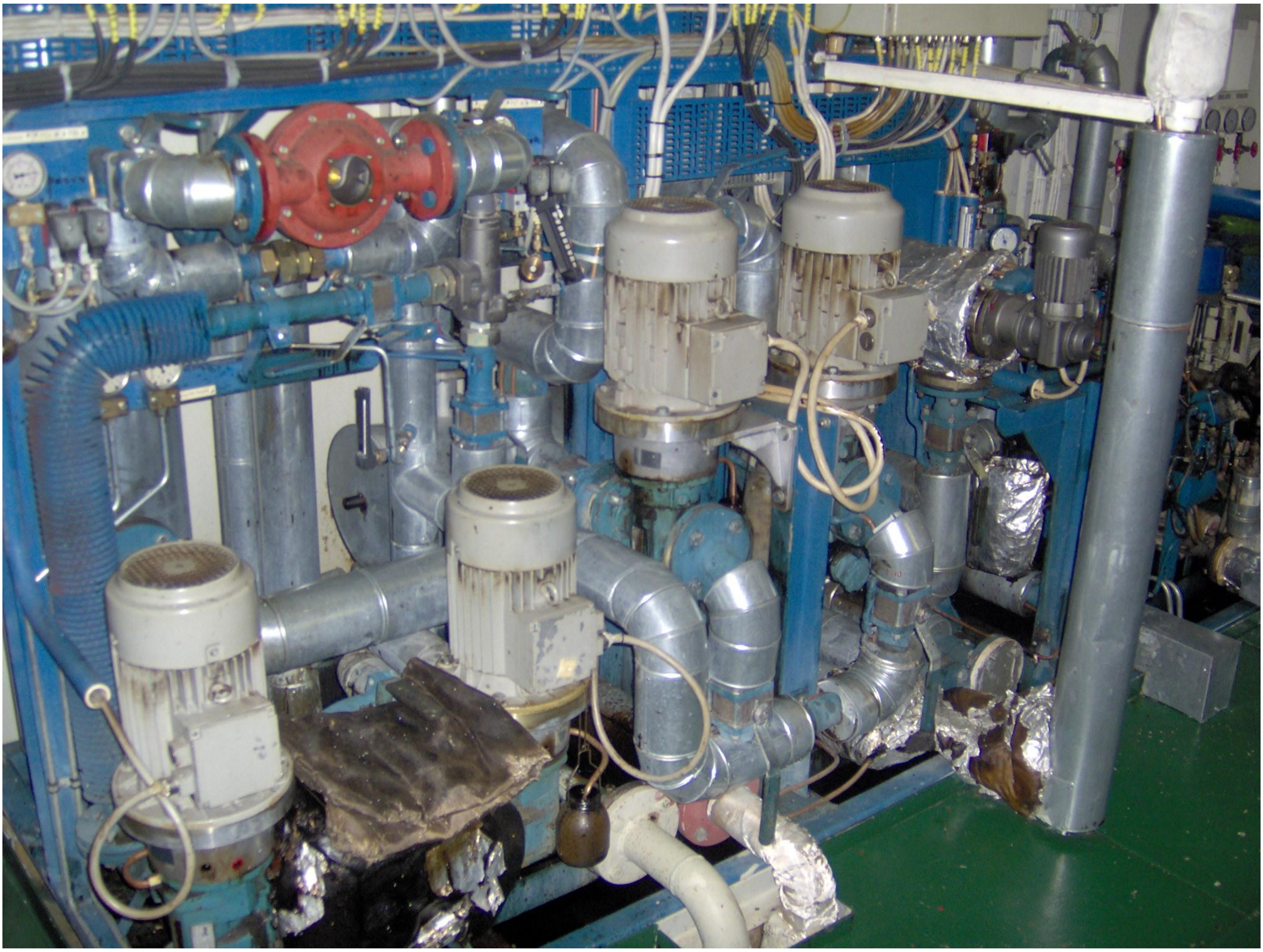
WÄRTSILÄ DIESEL Diesel Technology Turku	Wartsila Diesel International Ltd Oy TURKU- FINLAND	Voiteluöljypumppu L3NG Ruuvipumppu
Tuote	Suhde MED	

Picture from Estraden (Engine driven cooling pumps)

Picture from Estraden (Central cooler)



Picture from Estraden (Booster unit)



Picture from Estraden (Lubricating oil cooler)