THE INSTITUTE OF MARINE ENGINEERS
76 Mark Lane, London EC3R 7JN *Telephone: 01-481 8493 Telex: 886841*

TRANSACTIONS (TM)

ECONOMIC DESIGN OF COOLING WATER, FUEL AND ELECTRICAL POWER GENERATION SYSTEMS FOR MOTORSHIPS

Professor Dr-Ing G. Grossman C Eng, FIMarE

Read at 1730 on Monday 12 November 1984

ISSN 0309-3948
Trans I Mar E (TM)
Vol. 97, Paper 7 (1985)

Economic Design of Cooling Water, Fuel and Electrical Power Generation Systems for Motorships

Professor Dr-Ing. G. Grossmann
Technical University of Berlin

SYNOPSIS

It is the task of the marine engineer to provide the ship with the simplest propulsion system so as to reduce installation costs, space and maintenance demands. The design of the engine systems must also take into account the reduced size of the crew. A recent newbuilding has a technical complement of two engineers and two mechanics, with only the chief engineer from Europe. The answer to this reduced complement cannot be an increase in the number of systems or automation, which merely increases the costs of installation and maintenance. All systems have to be as simple and as reliable as possible. The reduction in the number of parts should be proportional to the reduction in manpower. At the same time the control systems should be simplified. The author discusses suggestions for simplifying the fuel, cooling water and electrical power generation systems, which will also reduce the fuel consumption.

FUEL SYSTEM SPECIFICATION

The system has to provide the main engine and two diesel generators with heavy fuel at a viscosity of up to 600 cSt/50°C and pour point of 25–30°C. It will have a gravity settling tank and a buffer tank for one day's fuel consumption; the buffer tank is supplied from the separators. For flushing purposes, the system must also be able to run on diesel oil.

The fuel oil bunkers must be heated continuously to maintain the fuel oil temperature above 45°C and the buffer tank must be kept between 70 and 90°C. The final separator preheater must bring the fuel oil temperature up to 99°C and the final heater before the diesel engines is required to bring the temperature up to 150–160°C.

Three subsystems can be recognized: a fuel service system; a fuel cleaning system, and a fuel heating system.

Fuel service system

The fuel service system is supplied with clean, heavy fuel from the buffer tank, heated to a viscosity which enables the diesel engines to use this fuel.

Take for example an MaK 8M601 engine with $P_{MCR} = 8000$ kW and $\dot{m}_{BSPEC} = 0.190$ kg/kWh (residual fuel HU = 40 900 kJ/kg, 450 cSt/50°C) and two diesel generator sets of 600 kW each with $\dot{m}_{BSPEC} = 0.210$ kg/kWh. The maximum fuel consumption per day $\dot{m}_{BD} = 39.5$ t/d. The capacity of the buffer tank (V_{DT}) must therefore be at least 40 m^3.

The service power of the main engine is $P_S = 7400$ kW and one diesel generator will run with $P_{ES} = 470$ kW. At service conditions a fuel consumption of $\dot{m}_{BS} = 1500$ kg/h is to be expected, whereas the design consumption is $\dot{m}_{BMCR} = 1646$ kg/h.

To maintain a constant pressure at the inlet to the injection pumps and to keep the pump plungers floating in fuel oil, the service pump has to deliver more oil than needed at full power and a pressure control valve recirculates the surplus.

To ensure the correct operation of the pressure-control valve, the minimum flow rate through it should not exceed more than 100% of the full power flow to the engines. This means that the fuel oil service pumps require a design capacity of 3300 kg/h. With a high viscosity fuel, the temperature at the inlet to the heater might be 150°C, the same temperature as the recirculating oil.

When the ship is running at 50% service power and 79% service speed, the recirculation increases from 1650 kg/h to 2475 kg/h. With a load of 20% on the main engine—and still a speed of 58%—the flow through the recirculation line is 2970 kg/h or 180% of the full power flow to the engines.

The recirculation line is fed back to the day tank, which is fitted with a degassing device to separate the gas from the fuel oil; 95% of the recirculating oil is led to the suction of the fuel oil service pump.

The hot fuel can cause several difficulties. Today, even a 600 cSt/50°C heavy fuel may be a mixture of 12% cracked gas oil and 88% 6000 cSt/50°C residual fuel; and the cracked gas oil can evaporate after passing through the pressure control valve into the lower-pressure circulation line. Even if evaporation does not take place in that line, it can take place in the suction line from the standpipe, in which the fluid velocity is increased and the pressure lowered possibly below the vapour pressure of the cracked gas oil.

A gas bubble will result in the fuel oil service pump losing suction and, consequently, an irregular fuel supply to the diesel engines. One way to prevent this is to fit a closed circulation system in which additional booster pumps supply the circulation pumps with make-up fuel. This is rather a complicated way of solving the gas bubble problem.

A much simpler solution is to fit a day tank with a large head above the fuel oil service pump; in this example the 40 m^3 day tank has a bottom area of 2×4 m^2, a height of 5 m and a short, large-diameter suction line, to avoid accelerating the fluid.

The recirculation line is led into the bottom of the day tank and connected to the fuel pump suction by a pipe five times the diameter of the suction line, with a minimum horizontal run of 1.0 m. At the highest point, four holes should be fitted (of the same diameter as the inside of the suction pipe) to allow gas and steam bubbles in the recirculating line to escape into the day tank. With this arrangement nearly all the recirculating fuel is fed into the suction of the fuel oil service pump.

The second danger is that the day tank will overheat during normal service owing to the fuel from the separators being at 90°C, and the oil from the recirculation line at 150°C. Under the conditions when all the recirculating fuel is returning to the suction, it will be difficult to maintain the day tank at not more than 90°C, the recommended temperature.

It is therefore recommended that a heating coil be fitted to maintain the day tank temperature at 80°C when the main engine is not running. This can be done by using liner cooling water at 85°C from the outlet of the main engine. The heating coil will also act as a cooler when the day tank temperature rises above 90°C. The inlet from the recirculation line should be just below the heating coil. To avoid the oil stratifying in the day tank, the inlet from the separator line should be as high as possible, to ensure a positive flow in the tank under all conditions.

A third problem is that at the high temperature of the recirculating fuel oil, the light, cracked gas oil and the heavy residual fuel may separate, the light part rising to the top. But if the inlet from the recirculating line, the inlet from the separators and the suction to the pump are arranged so that at least 90% of the incoming flow goes to the suction, separation of the cracked gas oil from the residual oil can be avoided.

A fourth difficulty arises from the fact that when the oil temperature at the inlet to the final heater is about 150°C, the wall temperature of the heating elements will be above 180°C. Owing to this high temperature, formation of insoluble asphaltenes may occur with some fuels (this is known to happen with steam-heated coils at 10 bar and above). This asphaltene formation cannot be prevented even if the recirculation system is kept at a higher pressure.

Owing to the build-up of insoluble asphaltenes, the filter in the fuel oil service system will become clogged. As fuel from the separators is clean, no filter should be needed in the fuel oil service system, provided that the day tank is coated internally and that the fuel oil service pipe system is made of corrosion-resistant material, and flushed and cleaned thoroughly before commissioning. The asphaltenes will then be circulated and burnt in the engines.

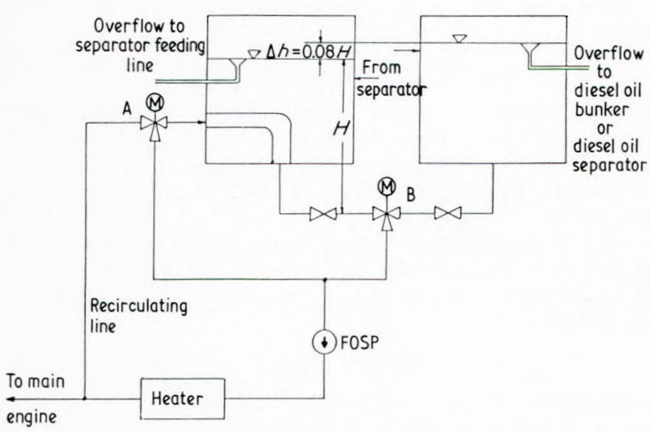


FIG. 1 Diesel oil and heavy fuel oil day tanks

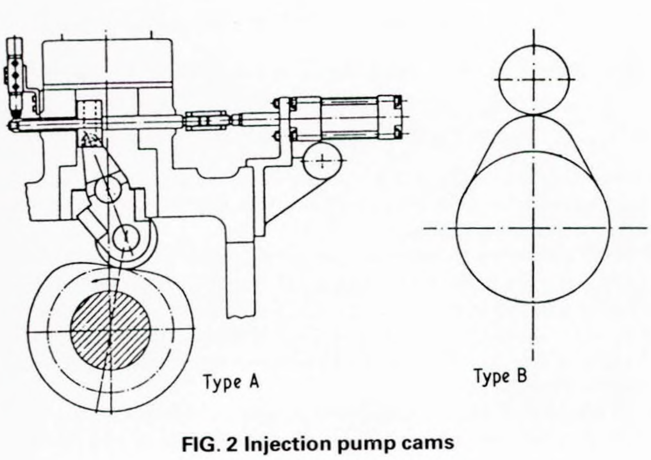


FIG. 2 Injection pump cams

To avoid asphaltenes sticking to the heating elements, the full fuel oil flow should always pass through the heater. This means that the bypass with the pressure control valve must be installed after the heater.

Diesel oil service system

When the ship has been stopped for long periods, the main engine fuel system is flushed with diesel oil to achieve an easier restart. In this case, the diesel oil line, which is connected to the suction line of the fuel oil service pump, is opened up gradually. The pump then takes a mixture of diesel and heavy fuel. As this mixture has a lower viscosity, the viscosity controller will reduce the oil temperature to the main engine.

Owing to the low temperature of the diesel oil—which may be around 15°C in winter—the diesel/heavy fuel mixture will have a temperature of about 70°C, instead of 108°C. This means that the temperature at the inlet to the final heater, which is still on heavy fuel duty, will be 110°C (instead of 150°C) and, to avoid thermal shock to the injection pumps, the temperature of the fuel must be decreased gradually. The permissible temperature gradient for injection pumps is not available, but a ship with an 8M601 engine had trouble with the injection pumps when changing over to diesel oil at service load. The temperature gradient ($d\theta/dt$) of the oil at the injection pump approximated to -25°C/min, which is on the high side.

The fuel oil service system acts as an energy storage system, collecting and releasing energy when the temperature of the make-up fuel is altered.

The approximate temperature of the oil before the engine can be expressed by:

$$\theta_{B(t)} = \theta_{BE} + (\theta_{BAO} - \theta_{BE})e^{-t/T} \tag{1}$$

$t = 0$ when the heater is closed down and the fuel is switched over to diesel oil. The time constant T is:

$$T = [(m_{FL}c_{pFL}) + (m_{ST}\,c_{ST}) + (m_{H}c_{pH})]/(\dot{m}_{B}c_{pFL}) \tag{2}$$

where m_{FL} is the mass of fuel in pipes, valves, heaters and mixing tubes; m_{ST} is the mass of steel of pipes, valves, heaters and mixing tubes; m_H is the mass of heating fluid in heater and pipe heating; and $\dot{m}_B$ is the fuel flow at load P_M.

In Eqn (1) the time constant T is a function of the load of the main engine, represented by fuel consumption $\dot{m}_B$. With an almost constant fuel velocity in the fuel pipe, T for the pipe system is a function of the length of the piping between the suction to the fuel oil service pump and the main engine. For a motorship with an 8000 kW main engine, T was found to be 140 s at full load.

This time is too short for a system without a mixing pipe for changing over to diesel oil, as the drop in temperature can be 38°C when switching from 90°C heavy fuel in the daytank to 15°C diesel oil at full load, giving a temperature gradient ($d\theta/dt$) of 16.5°C/min, which is too fast. If the change-over to diesel oil is done gradually in, say, 4 min, the maximum temperature gradient will be 8.8°C/min, which can be accepted by the fuel pumps. To achieve this, the diesel oil day tank requires an 8% higher level of oil, to allow for the different densities of the fuels. In addition, the pipelines from both day tanks to the change-over valve should be as similar as possible, to keep the pressure losses the same. The two three-way valves (A and B in Fig. 1), with a 4 min closing time, can then be operated by one push-button to change the system to diesel oil and by another to change back to heavy fuel.

From a safety point of view, an injection pump cam of the type A in Fig. 2 is more suitable than type B. With type A, the plunger has a greater chance of seizing in the high position when the temperature decreases too fast. In this case the injection pump will not be destroyed as, after it has cooled down, it will work again. With type B a stuck plunger can be forced upwards, destroying the pump.

Fuel cleaning system

The fuel cleaning system comprises the separators, the main fuel oil filter and a settling tank. The separators should be designed for continuous service at full power. This will keep their size as small as possible and the day tank will normally be full.

With a high viscosity of 600 cSt/50°C, the recommended rating for the separators is 500% of the diesel oil rating for a single stage, and 385% of the diesel oil rating for two stages. For two-stage separation the system needs two separators, each with a nominal diesel oil capacity ($\dot{V}_{SOP}$) of 6350 litre/h. For parallel purification and continuous operation of both separators, each should have a capacity $\dot{V}_{SOS}$ of 4125 litre/h. When a third standby separator is required for security reasons, parallel operation with two smaller separators would be advisable as the most economic installation.

A separator cannot remove 100% of solid particles, so the installation of a very fine filter after the separator outlets is recommended. This filter should be installed between the separators and the heavy fuel day tank, so that the fuel oil service system is provided with the cleanest fuel available on board.

If the fuel oil service system is properly designed, there will be no new dirt and extra filters are not required as they would only become clogged by the asphaltenes. Since, with very heavy fuels, the separators may not be able to remove all the water from the fuel, a coalescer should be installed in addition to, and in series with, the filter.

To adjust the separator capacity to the fuel consumption of the engine, it is only necessary to connect the suction line of the separators to the overflow from the day tank. The separator then runs at its rated capacity and, at partial loads, obtains only part of its supply from the bunkers, the remainder being overflow from the day tank.

To avoid overloading separators with bigger dirt particles like sand and rust, suction should be taken from a tank which is designed to let these particles settle out. The settling velocity of particles or water droplets in a viscous fluid is governed by Stokes' law.

$$w_S = [D_i^2(\rho_i - \rho_o)g]/(18\ \nu_o\rho_o)$$

where D_i is the particle diameter (m); ν_o is the viscosity of the fluid (m^2/s); ρ_o is the density of the fluid (kg/m^2); and ρ_i is the density of the particle or water droplet (kg/m^3). Settling tank designs can be compared using this equation.

At given conditions of oil temperature and viscosity, a certain settling velocity will be achieved at which a particle needs a time t_s to move from the top of the tank to the bottom; t_s should be smaller than the mean time of duration, t, of the oil in the tank.

$$(t_{s1}/t_{s2}) = (H_1/H_2)(\nu_2\rho_2/\nu_1\rho_1)$$

$$t_{s1} = H_1/w_{s1} \quad \text{and} \quad t_{s2} = H_2/w_{s2}$$

where H is the height of the tank and t_s the time the oil remains in the tank. A settling tank should have the same capacity as a day tank and two tanks are needed to guarantee adequate time for dirt to settle. For the settling tank there is a viscosity of 180 cSt, $t_1 = 24$ h, $H_1 = 5$ m and $\theta = 60$°C.

The bunkers in this ship have a capacity of 200 m^3, a height H_2 of 1.8 m, a temperature θ of 50°C for a viscosity of 360 cSt, and the average time (t_2) the fuel remains in the bunker is 20 days. These figures give:

$$t_1/t_2 = 0.05 \quad \text{and} \quad t_{s1}/t_{s2} = 5.6$$

This indicates that even a shallow double-bottom fuel bunker functions much better as a settling tank than a conventional high settling tank. The fuel system can be simplified if bunkers are designed as settling tanks and the settling tanks omitted. The separators then take their oil directly from the bunkers (the suction head of the separator charging pumps has to be checked) and clean oil is discharged to the heavy fuel day tank.

Fuel heating system

Although this affects the fuel system it is part of the cooling water system. Bunker heating has two purposes:

(a) To maintain the bunkers at 50°C and the day tank at 80°C when the seawater temperature is 5°C and the air temperature is 0°C.

(b) To raise the temperature of the fuel in the bunkers to 50°C in a reasonable time after the oil temperature has fallen owing to a long period without bunker heating. It also has to raise the fuel oil temperature in the separator heaters from 50°C to 99°C and in the final heater from 80°C to 160°C maximum.

For the separator heater and the final fuel oil heater there are two alternatives; either:

1. Both heaters are heated electrically. This is the simplest solution but the capacity of the electric system has to be increased by an additional 125 kW. If 240 days at service power are assumed per year then 125 t of extra fuel are used more per year at a cost of $22 000; or
2. With the costs in (1) above, the installation of a small exhaust gas boiler for 300 kg/h of steam is economic, especially if the boiler can be incorporated in the 'harbour' boiler, with a capacity of 1400 kg/h. Combination boilers with a common water volume and natural circulation have a simple steam system and are inexpensive. The exhaust gas section is not large, as the gas is only cooled by about 20–25°C. Therefore, even with slow-speed engines, where exhaust gas temperatures of 235°C have been encountered recently, there is a reasonable temperature difference at the pinch point. There is no need for an economizer in the exhaust gas boiler section as the condensate can be fed directly into the boiler.

To minimize heating losses and the energy needed to warm the bunkers, the design of the bunkers and the warming-up sequence should be optimized.

In the equation for the transfer coefficient of a flat vertical wall:

$$\alpha_{WO} = c[(\theta_o - \theta_w)/l]^{0.25}\ \nu_o^{-0.08}\ \nu_w^{-0.17}$$

the heat transfer coefficient is proportional to the length l. This length influences the losses shown in Fig. 3, where the heat transfer coefficient (α_w) of a wall with stiffeners and webframes is reduced from 22.8 W/m^{2}K to 14.45 W/m^3 (oil ν_o = 60 cSt/50°C).

A further decrease in heating energy for the bunkers can be achieved if the warming-up sequence for each bunker is adjusted individually to allow for the cooling of the oil due to heat losses:

$$\theta_o(t) = \theta_2 - (\theta_2 - \theta_{UM})(1 - e^{-t/T_i})$$

with a time constant of each bunker

$$T_i = (m_o c_{pi})/(3.6\ k_i A_i)$$

The energy needed to warm up the fuel from θ_1 to θ_2 in t_H is then

$$\dot{Q}_{Hi} = [(\theta_2 - \theta_1)/(1 - e^{-(t_H/T_i)})] + \theta_1 - \theta_{UM})k_i A_i$$

The energy is transferred to the oil by means of the heating coil:

$$\begin{aligned}\dot{Q}_{Hi} &= \alpha_R(\theta_W - \theta_o)A_R \\ &= 3.74 L D_A^{0.75}(\theta_W - \theta_o)^{1.25}\nu_o^{-0.04}\nu_W^{-0.21}\end{aligned}$$

The reduction of $\dot{Q}_{Hi}$ will result in shorter heating coils but it is unnecessary to save energy as there is surplus energy in the liner cooling water ($\dot{Q}_{CW} = 0.2\ P_M$).

It is advisable to take the hot cooling water for the heating coils direct from the main engine. This is not dangerous even in case of leakage in the heating coils, as the cooling water system pressure is always higher than the oil pressure, owing to the height of the expansion tank. Thus water may leak into the fuel oil but not vice versa.

The cost of the heating system can be further reduced if finned tubes are used, as this reduces the cost of the heating coils and the man-hours needed for installation. The dimensions of the fin tubes must be selected very carefully, so that the fins do not become blocked when using high viscosity fuel.

If heating is not used for more than 4 days, an external heat source must be provided. With full bunkers at 45°C (100 cSt/50°C fuel) and an air temperature of 0°C, the bunkers will require 200 kW. A 3 bar steam boiler with 300 kW capacity could take care of all heating requirements for the bunker.

Conclusion for fuel system

The fuel system can be simplified by omitting the settling tanks and the mixing pipe. This makes both installation and maintenance easier, cheaper and less complex than with a pressurized fuel system.

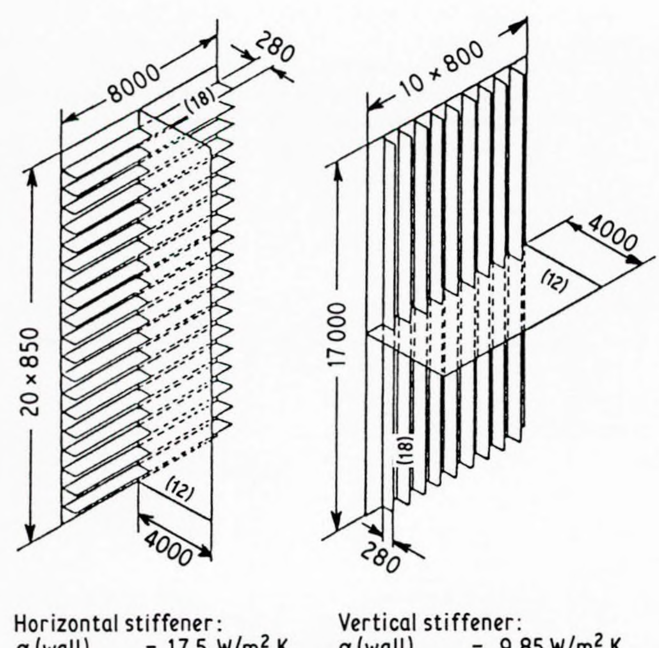
MAIN ENGINE COOLING WATER SYSTEM

One of the most important systems on board motorships is that for cooling water, which has to transfer about 65% of the main engine energy to seawater. To achieve this, several heat exchangers, pumps and pipes are used.

Specification

All cooling has to be done with fresh water and a central fresh water/seawater heat exchanger transfers the energy losses to seawater. The liners of the main engine and the diesel generators are cooled by a high-temperature cooling system. At full power the inlet temperature (θ_{ME}) is 75°C and the outlet temperature (θ_{MA}) is 85°C. θ_{ME} has to be kept constant. The air coolers are arranged so that the air can be heated to 80°C when the power of the engine falls below 40% of design load. At higher loads the air has to be cooled as low as possible.

Lube oil is also used to cool the pistons and an oil temperature (θ_{OE}) of 47°C is required at the inlet to the engines. At full power the temperature of the oil at the engine outlet must not exceed 55°C.



Horizontal stiffener:		Vertical stiffener:	
α (wall)	= 17.5 W/m² K	α (wall)	= 9.85 W/m² K
α (stiffener)	= 4.2 W/m² K	α (stiffener)	= 3.93 W/m² K
α (frame)	= 1.12 W/m² K	α (stringer)	= 0.67 W/m² K
αΣ	= 22.82 W/m² K	αΣ	= 14.45 W/m² K

FIG. 3 Influence of structure type on heat transfer coefficient

Heating of fuel bunkers, day tank, seawater evaporator and accommodation is done by cooling water from the HT system. A harbour circulating pump keeps the cooling water from the diesel generator sets circulating through the heating services. A steam water heater is provided to keep the HT cooling water at 75°C when no diesel engine is running.

The seawater system has to be as compact as possible and the seawater pump should be used only when the ship goes astern.

High temperature fresh water system

Liner cooling water system

The high temperature system includes liner cooling water for both the main and the auxiliary diesel engines and has to be designed to take care of the energy losses of the diesel engines at design conditions. The following considerations apply to all engines with lube oil cooled pistons, which are simpler systems and, for this reason alone, should be preferred.

In our example ship, an MaK 8M601 is assumed to be the main engine with a design power P_{MCR} of 8000 kW and energy losses to the liner cooling water of $\dot{Q}_{LCME} = 0.208\ P_{MCR}$. The two diesel generators are designed for a P_{EO} of 500 kW, and their energy losses ($\dot{Q}_{LCDG}$) are 0.21 P_{ME}. The system has to be designed for:

$$\dot{Q}_{LC} = \dot{Q}_{LCME} + \dot{Q}_{LCDG}$$

With $P_{MCR} = 8000$ kW and $P_{EO} = 500$ kW, $\dot{Q}_{LC}$ is 1770 kW. As the temperature difference between inlet and outlet is 10°C, a flow of

$$\dot{m}_{LC} = \dot{Q}_{LC}/(\theta_{HA} - \theta_{HH})c_p = 42.3\ \text{kg/s} = 153\ \text{m}^3/\text{h}$$

is needed. Through the main engine the flow $\dot{m}_{LCME}$ is 143 m³/h and the flow through the diesel generator ($\dot{m}_{LCDG}$) is 10 m³/h. As both DG sets need cooling water, regardless of which set is running, the overall flow required by the diesel generators ($\dot{m}_{LCDGO}$) is 20 m³/h, giving a total flow $\dot{m}_{LCO}$ of 163 m³/h (45.3 kg/s); and a temperature (θ_{MAM}), after mixing with the cool flow through the set which is not running, of 84.3°C if θ_{ME} = 75°C.

Charge air cooling

To keep the heating of the charge air at lower engine loads as simple as possible, two heat exchangers will be fitted in series. The first stage will be cooled by the liner cooling water from the main engine. As this water has a temperature of 75°C, the air can be cooled down in this stage to a temperature θ_{L1} of 95°C. The first stage of the air cooler is incorporated into the HT system.

For this medium speed diesel engine overall energy transferred by the air cooler ($\dot{Q}_{AC}$) is 0.45 P_{MCR} and the energy transferred by the first stage ($\dot{Q}_{AC1}$) is 0.27 P_{MCR} (2220 kW). (The auxiliary diesels have a single-stage air cooler.)

When the air coolers are arranged in parallel with the main engine and the cooling water is heated by 15°C, then an additional flow ($\dot{m}_{WL}$) of 35.4 kg/s is needed and the overall flow through the HT-system ($\dot{m}_{HT}$) is 80.7 kg/s (290 m³/h); and the temperature (θ_E) after the diesel engine at full load becomes 86.8°C.

From Fig. 4 it can be seen that, for $P_M < 0.4P_{MCR}$, the first stage air cooler no longer transfers energy to the cooling water, whose temperature is the same as air coming from the blower. At lower loads, the first stage of the air cooler will heat the air. This is the time—or the load—at which the cooling water flow to the second stage of the air cooler should be closed, to avoid the air being overcooled, resulting in liner corrosion. With heavy fuels and poor combustion, the SO_3 dew point occurs in the upper part of the liners, causing corrosion damage. Under tropical conditions with a 40°C air temperature at the blower inlet, the air temperature at the blower outlet will also be higher, and the second stage cooler should not therefore be cut out too soon.

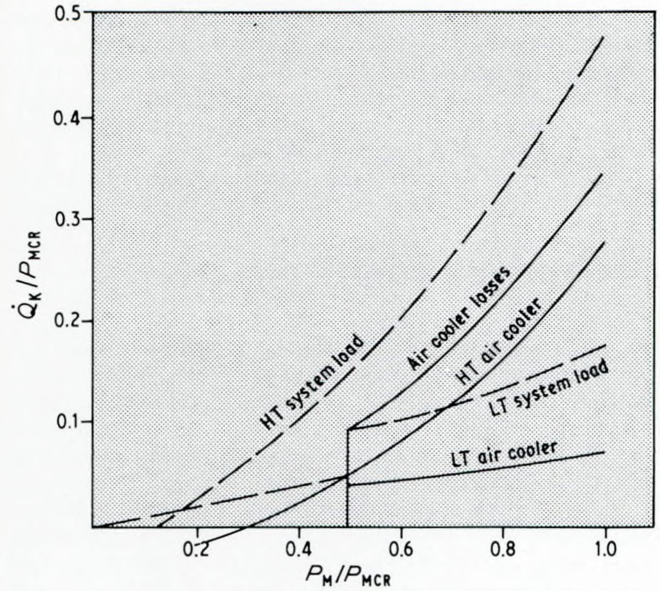


FIG. 4 Energy losses of the two-stage air cooler

Heating system

The relatively high temperature θ_{LCA} of the HT cooling water (87.6°C at MCR, 77°C at $P_M = 0.4$ MCR) means that its best use is for purposes such as bunker and day tank heating and air conditioning. In this case the cooling water should flow directly and continuously through all heating coils.

At steady steaming conditions, the energy taken from HT cooling water ($\dot{Q}_H$) is 440 kW. This value can be considered constant and the overall energy load of the HT-system at design conditions is then:

$$\dot{Q}_{HTO} = \dot{Q}_{LC} + \dot{Q}_{ACl} - \dot{Q}_H = 3550 \text{ kW}$$

The temperature θ_{HTA} of the cooling water outlet from the heating system is 85.5°C.

Low temperature cooling system

With a well-designed central heat exchanger, the fresh water outlet temperature θ_{LTO} will be 36°C for a seawater inlet temperature θ_{SE} of 32°C. For these conditions a rather large heat exchanger is needed. At design load, the lube oil and piston oil cooler transfers:

$$\dot{Q}_{PCO} = 0.108\,(P_{MCR} + P_{EO}) = 863 \text{ kW}$$

The LT air cooler of the main engine transfers:

$$\dot{Q}_{ACME} = 0.075\,P_{MCR} = 600 \text{ kW}$$

The air cooler of the diesel generator transfers:

$$\dot{Q}_{ACDG} = 0.347\,P_{EO} = 174 \text{ kW}$$

The lube oil cooler of DG transfers:

$$\dot{Q}_{LODG} = 0.11\,P_{EO} = 55 \text{ kW}$$

The gearing lube oil cooler transfers:

$$\dot{Q}_{GLO} = 0.03\,P_{MCR} = 240 \text{ kW}$$

The overall energy transferred to the LT system is

$$\dot{Q}_{LT} = 1932 \text{ kW}$$

The load of the heat exchanger is then:

$$\dot{Q}_{HE} = \dot{Q}_{HT} + \dot{Q}_{LT} = 5482 \text{ kW}$$

This requires a flow of:

$$\dot{m}_{HE} = \dot{Q}_{HE}/c_p(\theta_{HTA} - \theta_{HEA}) = 26.46 \text{ kg/s} = 95.24 \text{ m}^3\text{/h}$$

when the flow is cooled down to $\theta_{HEA} = 36$°C from $\theta_{HTA} = 85.5$°C. This is also the flow through the LT system, where it will be warmed up by $\Delta\theta_{LT}$ of 17.8°C to a temperature θ_{LTA} of 53.8°C.

The flow distribution to individual coolers must be carefully calculated; the lube oil temperature has to be kept within tight limits, and the coolers need relatively more water with a low outlet temperature than the air coolers, where the outlet temperature can be considerably higher than θ_{LTA}.

The above calculation shows that the flow through the HT-bypass at normal design conditions is

$$\dot{m}_{HTBP} = \dot{m}_{HT} - \dot{m}_{LT} = 54 \text{ kg/s.}$$

Port conditions

In port only one DG set is run at a service load P_{ES} of 300 kW. The heating appliances are shut off, with the exception of the day tank. The load on the HT cooling system $\dot{Q}_{HT}$ is then 0 kW, since:

DG-liner cooling	$\dot{Q}_{LC} = 60$ kW
Line losses	$= -10$ kW
Day tank heating	$\dot{Q}_{VDT} = -20$ kW
Heat losses main engine	$\dot{Q}_{VME} = -30$ kW

On the LT system the load $\dot{Q}_{LT}$ is 70 kW, since

Oil cooler DG	$\dot{Q}_{BCP} = 30$ kW
Air cooler DG	$\dot{Q}_{AC2D} = 40$ kW

The heat exchanger load $\dot{Q}_{HE}$ is 70 kW.

Under port conditions, the flow through the cooler and the LT system becomes too low for the oil cooler of the diesel generator and to prevent high oil temperatures a small bypass of the HT temperature control valve has to be installed to guarantee a minimum flow through the LT system ($\dot{m}_{LTBP}$) of 0.6 kg/s.

If there is only very little energy transferred to the cooling water, a small harbour pump with a capacity $\dot{m}_{LCP}$ of 2.5 kg/s should be installed in parallel to the circulating pumps. This flow is sufficient to supply both DG sets with cooling water.

When the outside temperature is low, the energy losses of the diesel generator are not sufficient to maintain the HT system temperature, especially when the bunkers have to be heated. For stops exceeding 3 days in cold climates, a steam water heater of 300 kW (480 kg/h of steam, 4 bar) has to be installed in the HT bypass. 300 kW will keep the bunkers at 45°C for IF100 ($\dot{Q}_V \approx 200$ kW), the main engine ($\dot{Q}_{VME} \approx 30$ kW), the day tank temperature ($\dot{Q}_{VDT} \approx 20$ kW) and supply the ship's heating system with energy ($\dot{Q}_A \approx 40$ kW).

The LT cooling system is shown in Fig. 5.

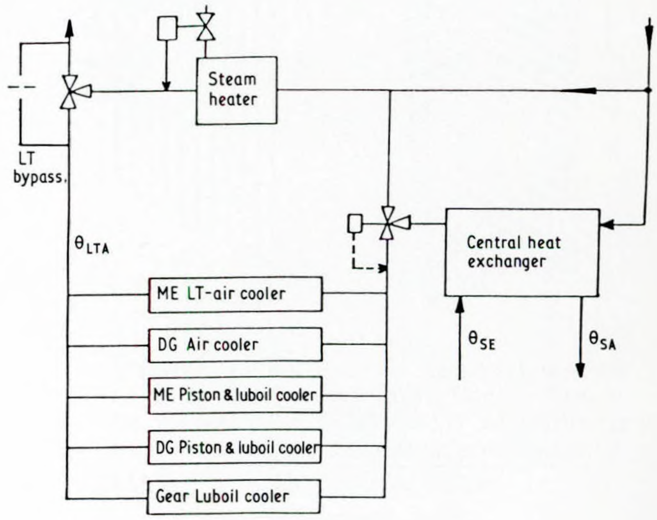


FIG. 5 LT cooling system

Overall fresh water system

The combined LT and HT cooling systems form the central fresh water cooling system with one pump, one cooler and two temperature control valves. This is a major simplification and saves some energy, as the power for one circulating pump is less than the power needed for the two pumps in the two separate and independent cooling systems. The central cooler needs a bigger area than the two independent coolers (see Fig. 6) but will not be any more expensive. The overall system costs will be no higher, and may even be lower, than the cost of the two independent systems.

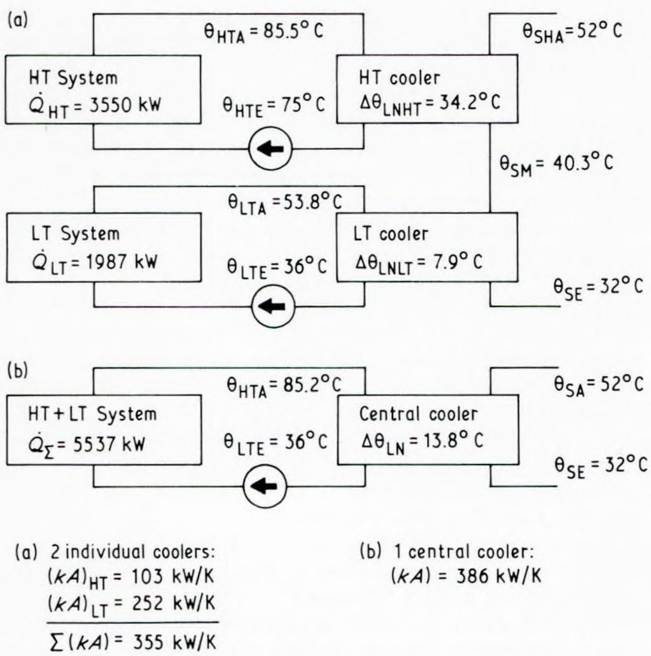


FIG. 6 Comparison of two cooler systems

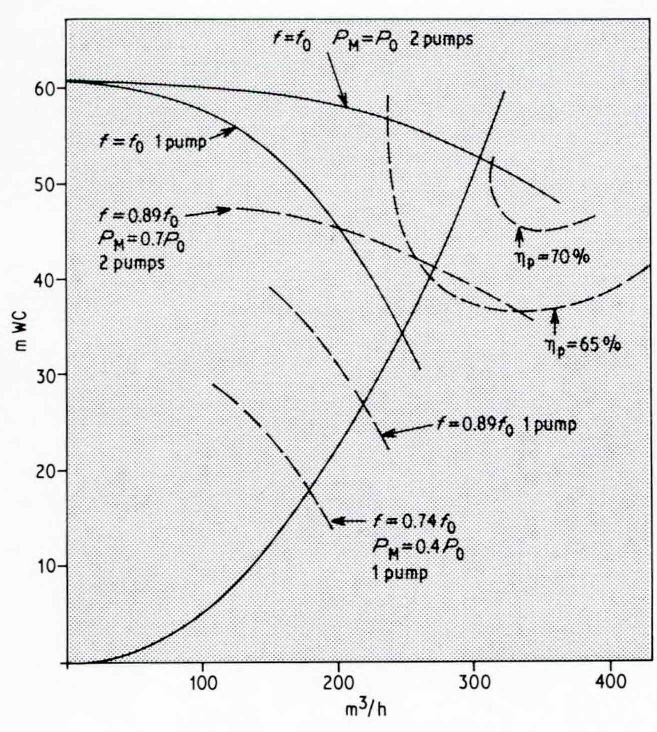


FIG. 7 Fresh water pumps: load field

With a flexible layout of fresh water circulating pumps, electric energy can be saved at partial load of the main engine and three pumps, each of 50% flow capacity, are fitted. Figure 7 shows that, at $P_M = 0.7\ P_{MCR}$ using variable frequency (as explained in the section on 'Power generation'), the frequency f of the system will be $0.89 f_{MCR}$ and the capacity $\dot{Q}$ of one pump only at this frequency is 224 m^3/h at a head H of 27 m. The required volume flow $\dot{Q}_X$ for keeping the temperature difference over the main engine constant would be 0.7 $\dot{Q}_{MCR}$ (210 m^3/h).

At this load, one pump can fulfil the requirements of the system. The electric power P_{EP} needed for this pump will be 0.4 P_{EPO} (24 kW) (where P_{EPO} is 60 kW). When the main engine is running at $P_M = 0.4\ P_{MCR}$, the frequency will reduce to 0.74 f_{MCR}. At these conditions the volume flow of the one pump in service $\dot{Q}$ will be 185 m^3/h, much more than the required flow $\dot{Q}_X$ of 130 m^3/h, and the consumed electric power P_{EP} will be 0.25 P_{EPO} (15 kW, a saving of 45 kW).

The fresh water cooling system will be run for most of the time with one pump and a second pump will be added only at powers above 80% of P_{MCR}. With this pump arrangement, the outlet temperature at the main engine is kept nearly constant, enabling a simple control of the inlet temperature.

Seawater cooling

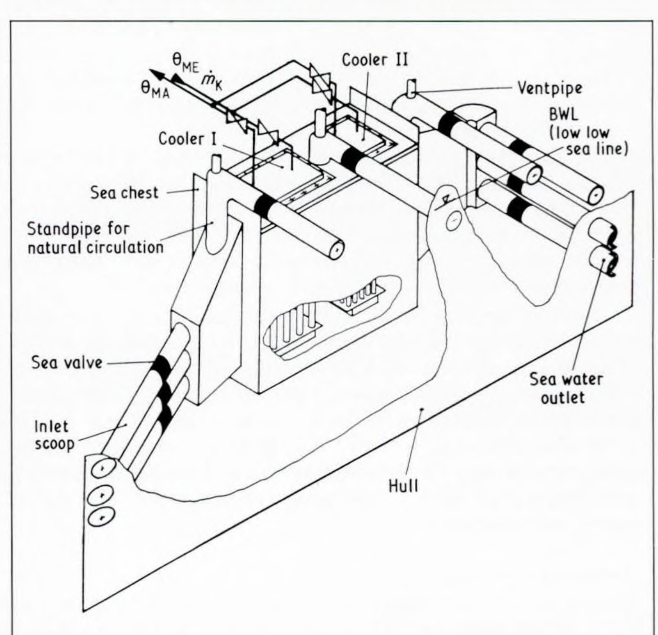
This system should also be as simple as possible, with the seawater pipes restricted to the barest minimum. Corrosion is still the biggest problem, causing much repair and maintenance work.

One solution is a sea chest cooler, which needs no seawater pipes at all; but unfortunately its heat transfer is rather low with $k \approx 600$ W/m^{2}K. A combination of sea chest cooler with scoop elements will improve the k value and considerably reduce the necessary heat exchange areas. The sea chest cooler is a normal multiple-flow tube cooler with only two shell plates. The forward and aft sides should be open, so that seawater can flow through naturally when the ship is moving (Fig. 8).

At very low ship speeds, the cooler operates with a vertical natural circulation only, taking the water in through the lowest openings and discharging it through the highest, which should be just below the light-load water level.

When the ship is moving, a horizontal flow, induced by and proportional to ship's speed, is superimposed on the thermal flow. The α_i values inside the tubes are not as high as those of

FIG. 8 Sea chest cooler with scoop elements



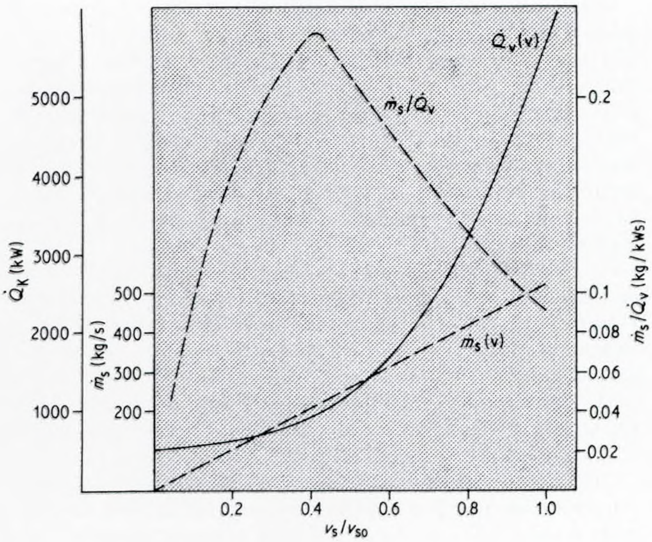


FIG. 9 Steady state load conditions of scoop sea chest cooler

the plate cooler, but the flow resistance on the outside of the tubes is very low. The sea water flow at full ship speed will therefore be rather large. This increases the temperature difference between fresh water and seawater, the α_A value, and decreases the necessary heating area.

Owing to the low flow resistance on the seawater side, the sea chest cooler can transfer more energy than a plate cooler when the ship is in port. As a tube cooler contains copper or steel alloys, the inside of the sea chest must be carefully coated to avoid severe corrosion from warm seawater.

The sea chest cooler also works satisfactorily when the ship is going astern, as the water flow induced by the velocity does not oppose the thermal flow. The sea chest is the sole seawater system; no pipes, valves or pumps are necessary, which decreases installation costs and maintenance problems.

Load characteristic of the central cooler under steady steaming conditions

The most important fact in a scoop system is that the flow of seawater through the cooler is proportional to the speed of the ship, which means that, at low ship speeds, surplus seawater is available to cool the fresh water down to θ_{LTE}. The temperature difference $\Delta\theta_{SE}$ $(=\theta_{SA} - \theta_{SE})$ will decrease with ship's speed.

Figure 9 shows the curves of the energy losses $\dot{Q}_V$, the seawater flow through the cooler $\dot{m}_s$ and the ratio $\dot{m}_s/\dot{Q}_V$ as a function of the ship's speed.

The heat transfer coefficient k, at full power and full speed, is approximately 6000 W/m^{2}K. With the ship stationary, the heat transfer coefficient decreases to about 800 W/m^{2}K.

With the decreased mean temperature difference, about 600 kW of energy can be transferred by the cooler, if the exit temperature θ_{SA} of the seawater is restricted to 52°C.

The heat transfer coefficient k is influenced very strongly by the internal heat transfer coefficient α_i of the fresh water flow, which means that the velocity of the fresh water has to be kept rather high. To achieve this at partial load, it is advisable to divide the central cooler in two, one with $A_{H1} = 0.333\,A_H$, the other one with $A_{H2} = 0.667\,A_H$. With this arrangement it is possible to reduce the flow area for $\dot{m}_H$ and the heating area A_{Hi} and adapt the cooler to the power requirements and the sea water inlet temperature θ_{SE}.

Transient conditions

During steady steaming conditions, the sea chest cooler can fulfil all demands on the system. When the ship accelerates from $v_s = 0$ to v_s, the power increase is faster than the increase of the ship's speed. The energy load on the cooling water system increases gradually from the port load $\dot{Q}_{VP}$ to $\dot{Q}_{VM}$, the losses corresponding to $(P_M + P_{ES})$:

$$\dot{Q}_{V(t)} = \dot{Q}_{VP} + (\dot{Q}_{VM} - \dot{Q}_{VP})(1 - e^{-t/T_{ME}})$$

The time constant T_{ME} of the main engine takes an important part in this load change, but it is normally not known. For this engine, it is estimated from temperature measurements to be approximately 120 s.

The change of the ship's speed is governed by the time constant T_S of the ship:

$$T_S = (m_S v_{SM})/R_M = (m_S v_{SM^2})/P_M \eta_p) = 240\ \text{s}$$

where m_S is the mass of the ship (=displacement); v_{SM} is the ship's speed at the power P_M; η_p is the propeller efficiency; and R_M is the resistance at v_{SM}.

T_s is the time in which the speed v_{SM} of the ship, running steadily at power P_M, is reduced to 0.5 v_{SM} after the engine has been stopped. On the other hand, when the engine power is increased from $P = 0$ to P_M in a step function, then the ship will move with v_{SX} of 0.5 v_{SM} after T_S seconds.

The momentary speed of the ship can be calculated by the following equation:

$$v_S = v_{SM}(1 - (1/2)^{T_s}) = v_{SO}(P_M/P_{MO})^{1/3}(1 - (1/2)^{T_s})$$

With T_{ME} and T_S the time functions of the energy losses $\dot{Q}_{V(t)}$ and the seawater flow $\dot{m}_{SC}$ through the scoop system as well as the temperature difference $\Delta\theta_{SE}$ of the water, flowing through the cooler, can be calculated.

For the example system, the main engine was started and immediately brought up to a P_M of 0.51 P_{MCR} (4080 kW). The corresponding energy losses $\dot{Q}_{VM}$ are 3110 kW and the ship's speed v_{SM} is 0.8 v_{SO} (16.3 knots).

In practice the energy losses will reach the value of $\dot{Q}_{VM}$ after a time t of 5 T_{ME}, at which time the ship has reached a speed v_S of 0.82 v_{SM} = 0.6 v_{SO} (8.96 knots).

Figure 10 shows that, even under the severe conditions of the ship's acceleration, the sea chest cooler with scoop effects can transfer all the energy losses to the sea without any increase in seawater temperature. The highest seawater temperature will occur when the ship is not moving. With 600 kW losses, the seawater temperature will increase from 32°C to 52°C, which is the normal design outlet temperature of a plate heat exchanger.

All temperatures decrease as the seawater inlet temperature decreases. For design purposes, a θ_{SE} of 32°C should really be considered only when the ship is designed for a trade route through the Red Sea and the Indonesian Archipelago. There are not many other areas with such a high seawater temperature. It is better for the system to be designed for a θ_{SE} of 28°C and an inlet temperature θ_{LTE} of 33°C into the LT system.

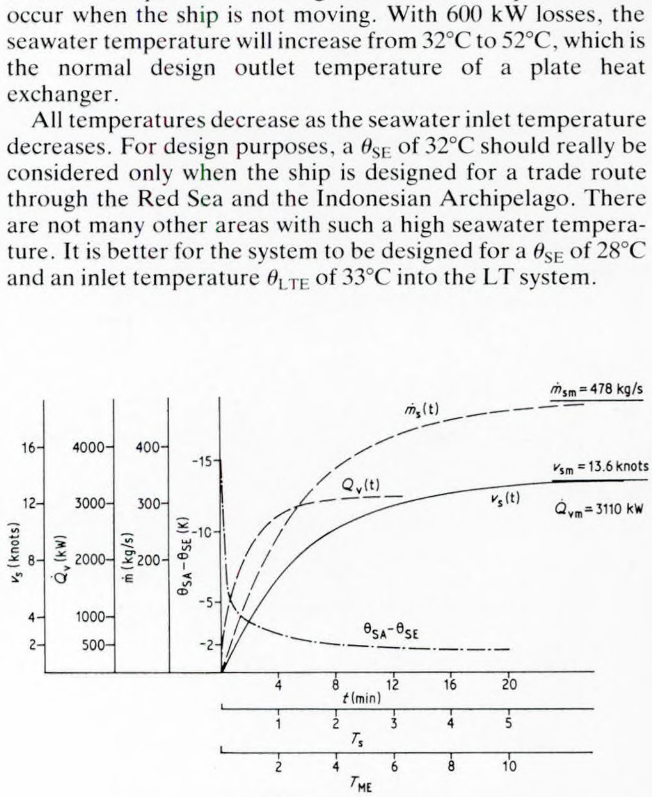


FIG. 10 Load conditions of scoop sea chest cooler during acceleration

Conclusion

The central cooling simplifies considerably the cooling system. It also saves electrical energy, as the power needed for the one remaining circulating pump is less than the power needed for two individual circulating pumps. Further energy can be saved at part loads of the main engine if three fresh water pumps, each with a flow capacity of 50% of the maximum flow, are installed. Below a P_M of 0.7 P_{MCR}, one pump is sufficient for the fresh water cooling system.

The seawater cooling system saves electric power when a scoop cooler with thermo-syphon is used. In a scoop sea chest cooler, the seawater system is practically done away with, with the advantages that there are no pipes to corrode, no electricity is required and the maintenance is minimum.

ELECTRIC POWER GENERATION

The electric power on board motorships is generated by either a diesel generator, or, under full away conditions, by a shaft-driven generator or a turbogenerator supplied with steam from an exhaust gas boiler. At full speed conditions at sea, the electric load is practically constant as, with constant frequency, the rev/min of the electric motors and their power stay constant. The electric load should be reduced as far as possible by designing energy-saving auxiliary systems.

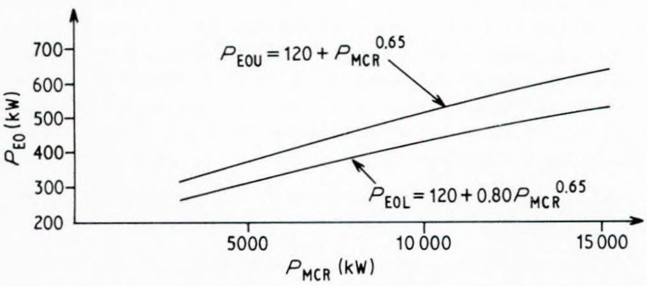


FIG. 11 Motor ships: electrical power available at full power on main engine

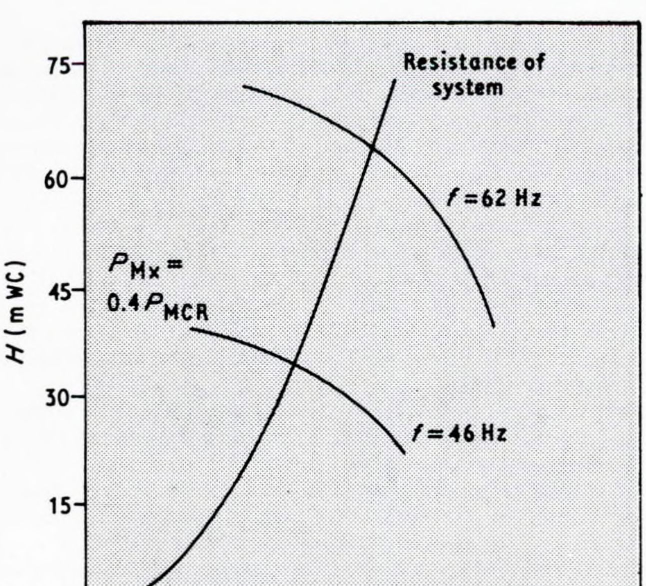


FIG. 12 Fresh water circulating pump operating on variable frequency

The electric power range at full power of the main engine can be expressed for ordinary motorships (see Fig. 11) by the equations:

$$P_{EOU} = 120 + P_{MCR}^{0.65} \quad \text{(upper range)}$$
$$P_{EOL} = 120 + 0.80\, P_{MCR}^{0.65} \quad \text{(lower range)}$$

Electrical consumers can be divided into three groups. Group 1 covers all pumps and ventilators running continuously to supply the main engine with cooling water, air and fuel. In Group 2 are all the pumps which work against a constant pressure such as the lube oil pumps of the main engine and the diesel generator sets; lighting, and electric heating for the galley. Group 3 consists of all other electrical consumers, which do not run continuously or which require a constant low-voltage supply.

Shaft-driven generator and fixed-pitch propeller

The generator should be driven by the main engine without a frequency converter. The frequency f and the voltage U are directly related to the revolution of the main engine.

$$(f_1/f_o) = (U_1/U_o) = (n_1/n_o) = \sqrt[3]{(P_1/P_o)}$$

where the index o indicates the highest power $P_o = P_{MCR}$ and where:

$$P_{min} = 0.4\, P_{MCR} \le P_1 \le P_{MCR}$$
$$n_{min} = 0.74\, n_{MCR} \le n_1 \le n_{MCR}$$

The frequency f and the voltage U will then also vary in these ranges:

$$f_{min} = 0.74 f_{MCR} \le f_1 \le f_{MCR}$$
$$U_{min} = 0.74\, U_{MCR} \le U_1 \le U_{MCR}$$

This means that with $f_o = 62$ Hz and $U_o = 456$ V:

$$46 \text{ Hz} < f_1 < 62 \text{ Hz}$$
$$336 \text{ V} < U_1 < 456 \text{ Hz}$$

within the above power range.

When the frequency and the voltage are proportional to the main engine revolutions, then the power demand of the consumers of Group 1 is related to the ratio of frequency cubed or it is equal to the ratio of the power of the main engine:

$$P_{E1x}/P_{E1o} = (f_1/f_{MCR})^3 = P_{Mx}/P_{MCR}$$

For Group 2 the electric power demand will be:

$$P_{E2x}/P_{E2o} = f_x/f_{MCR} = \sqrt[3]{(P_{Mx}/P_{MCR})}$$

The overall demand of Group 3 will remain constant. This means that all equipment requiring constant low-voltage power has to be designed for the lowest frequency and voltage. Motors not running continuously will use less power when running, but they will run longer.

According to one separator manufacturer the reduction of frequency and voltage brings no problems, as the reduction of the revolutions is accompanied by a reduction of flow. The separator itself belongs to Group 1, the electric heater of the separator to Group 2.

The behaviour of the fresh water circulating pump, belonging to Group 1, is shown in Fig. 12. One can see that the pump still delivers more water than is needed to the engine at low engine powers.

The electric system should be designed so that, at the service power of the main engine, the frequency is 60 Hz and the voltage U is 440 V. With $P_s = 0.9\, P_{MCR}$, this means $f_{MCR} = 62$ Hz, $U_{MCR} = 456$ V and $P_{min} = 0.4\, P_{MCR}$, $f_{min} = 46$ Hz, $U_{min} = 336$ V.

When the generator is directly connected to the main engine, without a frequency converter, the installation cost decreases drastically from US\$500 to about US\$60 per installed kilowatt. In addition to this, the energy losses of about 15% in the frequency converter are saved and at partial loads of the main engine the generated power is decreased, resulting in further energy savings. The actual electric power needed at partial load of the main engine can be approximately expressed by the relationship illustrated in Fig. 13.

This lower electric power demand is equivalent to a moderate fuel saving. For example, in our ship the design electric power P_{EO} is 395 kW, and the losses in the frequency converter ΔP_E amount to 60 kW. At service conditions the electric power demand P_{ES} will be 375 kW, so the saving will be 80 kW, which is equivalent to 80 t of fuel or US$13 000 for 200 days at sea.

The above electric power system provides the engine with a speed control system for all electric motors without any additional installation costs and without any additional control element. Only the signal for starting the diesel generator after a voltage failure has to be rearranged.

Shaft-driven generator and controllable-pitch propeller (CPP)

With a CP propeller, the shaft generator is usually installed without a frequency converter, as the main engine runs at constant speed. This means that the propeller runs at lower loads with a smaller pitch and an efficiency which may be 3% lower than for a fixed pitch propeller. To compensate, an increase in power of about 6% is required for a main engine load P_M of 0.4 P_{MCR}.

If the propeller is run with a constant pitch down to $P_M = 0.4\ P_{MCR}$ and $n_M = 0.74\, n_{MCR}$, the savings on the main engine are:

$$P_M = 0.4 \times 0.06\, P_{MCR} = 192 \text{ kW}$$

and on the generator, with $P_{EO} = 450$ kW:

$$P_E = 0.3\, P_{EO} = 135 \text{ kW}$$

These savings of 327 kW are equivalent to 1.45 t of fuel, or US$260/day, at $P_M = 3200$ kW.

Unlike the fixed-pitch propeller, the CP propeller permits a shaft generator to produce acceptable electric power down to a speed of 0.74 n_{MCR} of the main engine at all main engine powers. At the same time the losses due to the propeller turning with zero pitch are reduced to 40% of the losses at n_{MCR}.

Electric system with turbogenerator, operated with steam from an exhaust gas boiler

With the modern highly efficient diesel engine, these systems can be operated only with main engines of medium to high power. For a 4-stroke medium-speed engine and an exhaust

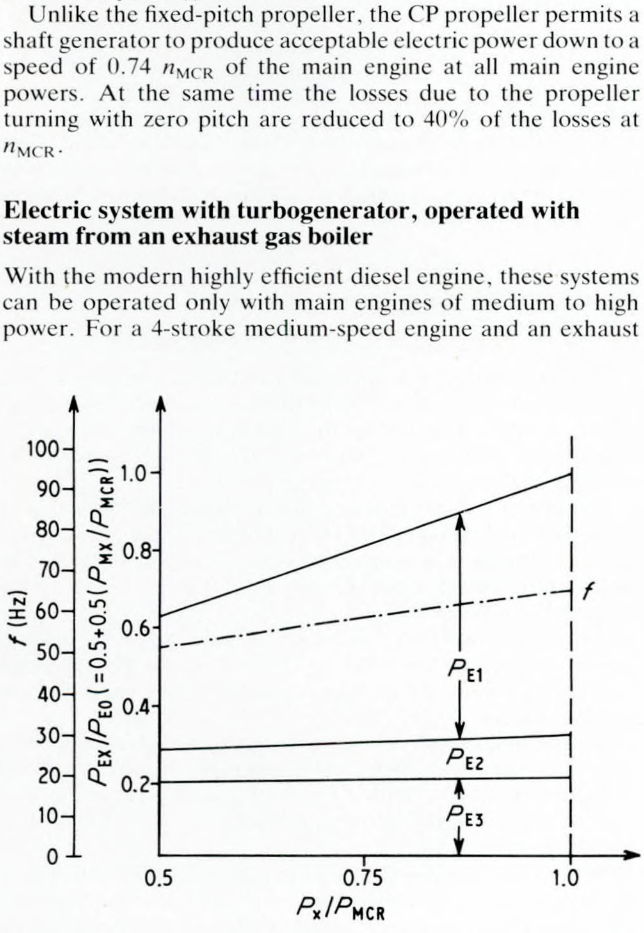


FIG. 13 Electrical power and frequency as a function of main engine power

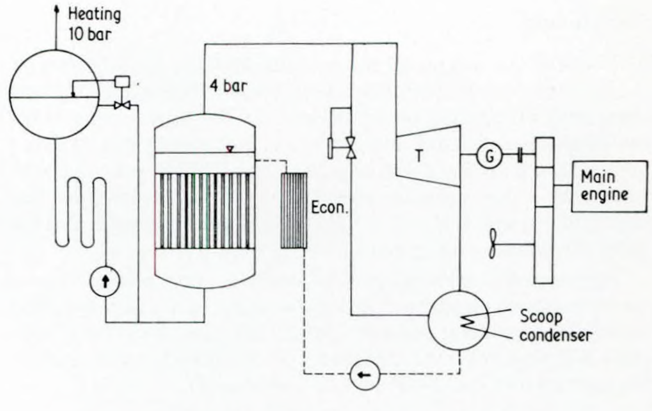


FIG. 14 Exhaust gas boiler system

gas temperature of 320°C before the boiler, the minimum load at which a turbogenerator would generate enough electric energy would be $P_M = 9000$ kW. With $P_{MCR} = 18\,000$ kW and frequency proportional to propeller rev/min, the turbogenerator could supply enough electricity down to $P_X = 0.5\ P_{MCR} = 9000$ kW. With constant frequency, the demand for electric power could be satisfied only down to $P_1 = 0.85\ P_{MCR} = 15\,300$ kW.

Conditions are worse for 2-stroke slow-speed diesels, where the minimum main engine load for a turbogenerator to generate enough electricity is about 21 000 kW. This is due mainly to the lower exhaust gas temperature of 255°C before the boiler.

If the electrical demand of the main engine auxiliaries is reduced by 16%, the minimum engine power for an exhaust gas boiler to supply all the electric power would be

$P_{O_{min}} = 6650$ kW for the 4-stroke engine, and
$P_{O_{min}} = 11\,800$ kW for the 2-stroke engine.

These considerations show the importance of the auxiliary systems when using exhaust gas boilers with turbogenerators and the power demand of the auxiliaries should be reduced as far as possible. Cooling water systems must have a low resistance and the need to correct the flow later with energy consuming orifices must be avoided.

For example, the diesel engine generators should have the same low resistance for all liner cooling water as the main engine. It also means that the exhaust gas steam system has to be simple, to keep the electrical power requirements as low as possible. An exhaust gas boiler with natural circulation, a stainless steel economizer and a scoop condenser would be the answer to this problem.

Electric power generation can be further increased in slow speed diesel engines by reducing the steam pressure to 4 bar, but a small 10 bar steam system is still required for the fuel oil heaters. Evaporation tubes would replace the superheater tubes and with a turbine internal efficiency of 65%, the water content at the turbine outlet would be below 10%, which a well-designed plant can accept.

If a turbine is then connected to the free end of a shaft generator (operating at a frequency and voltage proportional to propeller speed), all the steam will be used for power generation; at partial loads the main engine will deliver make-up energy to the generator. Parallel running of a diesel generator is not needed at steady steaming (Fig. 14).

A further reduction of the electric power demand is possible if a fresh water cooling pump is attached to the main engine, resulting in a decrease of 40 kW in our example ship. With this arrangement, the minimum main engine power for running a turbogenerator becomes 5700 kW for 4-stroke engines and 8800 kW for 2-stroke engines. Unfortunately, engine driven pumps require complicated arrangements except when fitted with CP propellers. The shaft generator system on the other hand would use about 2 tonnes of fuel per day, costing about US$70 000 per year. If an exhaust gas boiler with a turbo-generator could be bought for US$310 000, this system could be considered economically feasible even in our example ship. But, on the other hand, it adds to the complexity of the whole engine plant and cannot therefore be recommended as it would overload the drastically reduced engine-room crew.

CONCLUSION

International competiton places many, often conflicting, demands on the marine engineer. The plant he designs must be fuel-efficient; but at the same time it must be simple, reliable, maintenance-free, automated, and suitable for operation by the smallest possible crew. In addition the installation must be reasonable in price, weight and volume. There is still much room for improvement and the best overall results can be achieved only through a thoroughly integrated design.

BIBLIOGRAPHY

1. S. Akagi, 'Heat Transfer in Oil Tanks of Ships'. ISME Report No. 4 (1969).
2. J. S. C. Holt, 'Bulk Heating of Heavy Oils'. BSRA Rep. No. 334 (1961).
3. S. F. Corpe, 'The Heating of High Viscosity Oils'. BSRA Rep. No. 371 (1961).
4. A. A. J. Couchman, W. F. Dowie and W. McClimont, 'Grid Heating of High Viscosity Oil'. BSRA Rep. No. 34 (1964).
5. J. F. T. MacLaren, 'Heating High Viscosity Oils'. *British Chemical Engineer*, Vol. 10, No. 3 (1965).
6. D. J. van der Heeden and Mulder, 'Heat Transfer in Cargo Tanks'. Netherlands Ship Research Centre TNO, Report No. 67S (1965).
7. R. J. Saunders, 'Heat Losses from Oil-Tanker Cargos'. *Trans. IMarE*, Vol. 79, No. 12 (1967).
8. D. J. van der Heeden, 'Theoretical Evaluation of Heat Transfer'. Netherlands Ship Research Centre TNO, Rep. No. 86M (1966).
9. D. J. van der Heeden, 'Experimental Evaluation of Heat Transfer'. Netherlands Ship Research Centre TNO, Rep. No. 111M (1968).
10. D. J. van der Heeden, 'Guide for the Calculation of Heating Capacity'. Netherlands Ship Research Centre TNO, Rep. No. 128M (1969).
11. S. Akagi, 'Free Convective Heat Transfer in Viscous Oil'. *Trans. Jap. Soc. Mech. Eng.*, Vol. 30, No. 213 (1964).
12. J. Ulm, 'Untersuchung über den Wärmeübergang in Tankheizungsanlagen'. FDS Bericht Nr. 15, Hamburg (1970).
13. C. Christensen, 'Tankopvarming Varmetab og'. Danish Ship Research Inst., DSF, Re. (1971).
14. C. B. Lamb, 'The Heating of High Viscosity Oils by Natural Convection'. *J. Inst. Petrol.* 57, S. 1–8 (1971).
15. *Arbeitsmappe für Mineralöl-Ingenieure*, 2. Aufl., VDI-Verlag, Düsseldorf (1962).
16. *Schweres Heizöl—Eigenschaften und Handhabung*, Deutsche Shell AG Technischer Dienst, Hamburg (1979).
17. R. F. Thomas, 'Development of Marine Fuel Standards'. *Trans. IMarE (TM)*, Paper 9, Vol. 93 (1981).
18. W. Baur, 'Future Fuels for Marine Diesel Engines'. Fifth WEGEMT Conference, Paper C-1 and C-2, TUB (1981).
19. G. Grossmann, 'Vorlesungsmanuskript Schiffskraftanlagen'. TUB (1982).
20. G. Grossmann and C. Hadler, 'Bunker Heating for Fuel Oils with High Viscosity'. 5th WEGEMT 81 Conference, TUB, Paper C-4 (1981).
21. J. Gutermuth, 'Kühlsysteme und Nutzung der Kühlwasser-energie bei Schiffsmotoranlagen'. MaK Toplaterne, Nr. 50 (July 1982).
22. W. Bauer, 'Future Fuels for Marine Diesel Engine Lubrication of Modern Diesel Engines'. WEGEMT 81 Conference, TUB, Paper C1, C2 (1981).
23. R. Dien, 'Ein Vorschlag für die Beurteilung der Stoppfähigkeit von Schiffen'. *Schiff und Hafen*, Bd. 29, Heft 6 (1977).
24. G. Grossmann, 'Scoop Systems for Central Heat Exchangers for Ships'. *Hansa* 167, Heft 18.
25. Birger and Jacobson, 'Recoverable Scavenging Air and Jacket Cooling Water Heat'. MAN/B&W Report (1982).
26. MAN/B&W, Mini-Specification for L-GB/GBE Engines (1983).
27. J. E. Church, 'Total Economy: The Case for the Main Engine Driven Pump'. *MER* (April 1982).
28. W. Droste and W. Hensel, 'Energiebedarf und Möglichkeiten zur Einsparung in elektrischen Anlagen an Bord von Schiffen'.
29. G. G. Pringle, 'Economic Power Generation at Sea: The Constant-speed Shaft-driven Generator'. *Trans. IMarE (TM)*, Paper 30, Vol. 94 (1982).

Professor Dr-Ing. Günter Grossmann obtained his Dipl.-Ing. degree in 1950 at the Technical University, Braunschweig. From 1950 to 1953 he was in the R & D office of MaK at Kiel and from 1953 to 1957 was an Assistant at the Technical University, Hannover. Awarded a doctorate in 1957, he joined Howaldtswerke-Deutsche Werft where he was employed in marine engineering design, marine engineering projects, shipyard projects and R & D until 1968. Since 1969 he has been Professor of Marine Engineering at the Technical University, Berlin.

Discussion

J. K. ROBINSON (Lloyds Register of Shipping): ICE Publication 92–101 requires marine electrical equipment to function with supply frequency variations of $\pm 5\%$ and major owners have been known to operate their 60 Hz systems at 56 to 58 Hz for periods to reduce system load.

As system frequency falls, so will the speed of induction motor pump and fan drives, and hence their loads. Unfortunately their cooling, being normally by shaft-mounted fans, is also less effective and hence, to prevent overheating, the voltage is normally reduced, eg 440 V, 60 Hz auxiliaries can generally be run on 380 V, 50 Hz shore supplies.

Whilst voltage can be made proportional to frequency by fixing the generator field current (ie by selecting hand voltage regulation) this has the disadvantage that the starting of an induction motor is liable to result in system voltage collapse. Also, resistive heating, being proportional to voltage squared, and fluorescent light output, approximately proportional to $V^{1.6}$, will be less effective. Protection and instrumentation problems[1] should also be expected with the large frequency-reduction proposed by the author.

I suggest that more practical electrical system designs would employ two-speed motors for the main engine's auxiliary drives, or have split bus-bars (one variable frequency and the other fixed frequency) connected by a smaller frequency converter.

1. D. E. McPherson, 'Slow-speeding with gas-turbo-electric drives'. MER (June 1982).

J. R. WILLIAMS (BS Eng and Tech Services): The idea of effecting a major simplification in the fuel system by removing the settling tank is a good one but using the double bottom as a settling tank does carry with it a number of major drawbacks. It is self-evident that, if Stokes' Law operates in ideal conditions, then a 1.8 m tank with an oil residence time of 20 days is bound to be more effective than a much larger settling tank with an oil residence time of 24 h.

Actually, the use of a double bottom as a settling tank carries with it real problems. In service, once the head has been taken off the settling tank, the free surface will result in considerable sloshing when the ship is in the seaway. Ideal conditions for Stokes' Law to operate are, therefore, not present.

Additionally, in some ships it is common practice to shift bunkers around in order to trim the vessel as the oil is used up. Furthermore, in order to maintain the oil at a reasonably low viscosity, as would be expected in a normal settling tank, the temperature must be maintained at a high level. For a double bottom with a large area exposed to the sea, this could be wasteful of steam or other heating media.

Even in the event of the settling-out process performing as desired, the problem of disposal still remains. In a normal settling tank, accessible from the engine room, the sludge and water can be easily drained off. Within a double bottom this would be more difficult to undertake, requiring some sort of additional sludge removal pump and, hence, more complication.

Perhaps it would be better to consider the elimination of double bottoms for fuel bunkers altogether and re-design for a centralised bunkering and storage system, employing engine room cross-bunkers. This not only simplifies the whole bunkering process but could also be adapted for all the best features of a double bottom by a series of transverse separations, arranged vertically up the bunker.

Concerning the section in the paper on transient conditions, the author has discussed the relationship between the time constant of the main engine and the time constant of the ship and has shown how the two are related for a particular case. It is further shown that, by the time the maximum energy losses Q_{vm} occur, the ship has reached sufficient speed to enable the scoop to function adequately. However, the author does not indicate the displacement of the ship from which the time constant T_S is calculated. With a very large ship of fairly modest power (for example a very large bulk carrier of circa 200 000 tonnes with a medium-speed diesel engine of, say, 14 000 or 15 000 kW with low thermal inertia), quite a different time-constant to the one shown in the paper could occur. Has the author done any calculations for several ship types of large inertia in conjunction with a machinery system of low thermal inertia?

J. C. HATCHMAN (Heat & Power Eng. Consultancy & Computing Series). On page 4 the author quotes an exhaust-gas temperature of 235°C on slow-speed engines. Could he elaborate?

I was present at the recent trials of a new-generation slow-speed engine and noted a temperature of 280°C after the turbo-charger at loads above 50% output. This was a new engine, burning marine diesel oil. Also the 'plateau effect' on exhaust temperature, some 2–4 months after entering service, is now well documented (Fig. 7[1]) (Figs 7 and 8[2]) (Figs 14 and 15[3]). The temperature increase recorded was 30–60°C. A similar increase was noted by a leading 4-stroke medium-speed engine manufacturer (p. 79, contribution to Ref. 4). Would the author agree that the heat transfer coefficient for the tank wall losses depends also upon the siting and position of the wall and the thermal resistances in the heat transfer process?

The work on the use of hot water for bunker heating is extremely valuable and it has been reported that several installations using a secondary fluid system are already in service. It is a pity not to give similar thorough consideration to the most valuable source of high-grade energy, the exhaust gas. Each kW of heat going up the stack is money disappearing into hot air. Each tonne of fuel burned needlessly by a diesel- or shaft-generator is that much less for the future. If a dual-pressure steam-cycle waste-heat recovery plant with scavenge-air feed-heating were to be used, what would be the threshold for the installations shown? Does the author believe that dual-pressure steam systems would be complex to operate and difficult to maintain and, if so, why?

Is the extra propulsion resistance caused by scoops detrimental to their application to small and medium-size merchant ships?

1. D. R. Cusdin and M. J. Virr, 'A marine fluidized bed waste heat boiler'. *Trans IMarE*. 1979, Vol. 91.
2. J. R. Owen, 'Considerations in the application of organic ranking cycle waste heat recovery system to diesel engined vessels'. *Trans IMarE(C)*, 1981, Vol. 93, Paper C76.
3. Y. Tanaka and K. Sasaki, 'Energy saving by waste heat recovery on board'. Third International Symposium on Marine Engineering, Tokyo, 3–7 Oct 1983.
4. A. J. Morton and J. C. Hatchman, 'Taking waste heat seriously'. *Trans IMarE(C)*, 1982, Vol. 94, Paper C100.

Author's Reply

I agree fully with Mr Williams's remarks on the settling system: only a full tank has the ideal operating conditions for the settling process. So, when the mean duration for an individual bunker is 5–10 days, then all the sludge and the water should be drained off before the bunker is emptied. But there we have a problem. With the high-viscosity fuels, the engineers seem no longer able to see the difference between sludge and fuel, so they only drain the water off. This can be seen on the documentation of a test run using new separators with automatic desludging. In heavy weather and when the bunker was nearly empty, the cleaning process was activated considerably more often than under normal conditions (see Fig. D1). Naturally, a central bunker system simplifies this problem. In the German edition of the paper I have already pointed in this direction.

Mr Hatchman's remarks on the heat-transfer coefficient are not quite clear. For a sea-water cooled side wall the transfer coefficient of the sea water is so much higher than the transfer coefficient on the oil side, that I do not see any influence of the

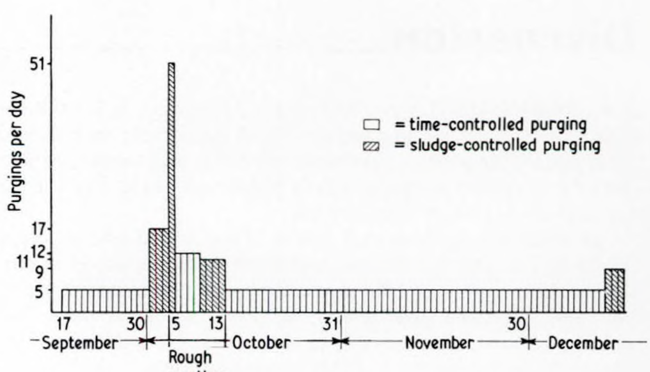


FIG. D1 Daily total purging of separators on MS 'Monte Cervantes' 17.9.83–16.12.83.

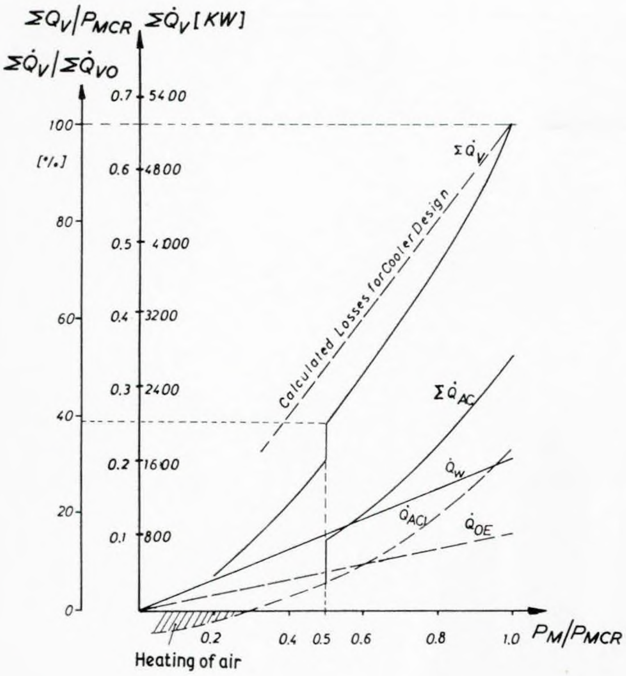


FIG. D2 Energy loss from the cooling water for a MAK 601, P_{MCR} = 8000 kW.

siting and position of the wall. The influence of the design upon the oil-side's heat transfer coefficient is shown in Fig. 3, where it is also stated that this coefficient is for IF 100. For fuels of higher viscosities, the value goes down. There is a third influence which diminishes the losses. If the ship has once carried a fuel with a rather high pour point, a wax layer will be built up on the outside wall due to the low temperature, thereby reducing the heat transfer coefficient considerably.

Mr Hatchman's remarks regarding the cooling water for bunker heating must be clarified in that here the liner cooling water is flowing directly through the heating coils. There are already several ships running with such a system. The most interesting one is a retrofit, where the ship can now run through the Baltic in winter with IF 180 instead of IF 50, owing to the heating of the bunkers with the liner cooling water.

Mr Williams' question about the ratio of time constants cannot be answered in one sentence. The time constant T_S of a ship can be calculated quite easily, but unfortunately very little is known about the time constant T_M of the main engines. The T_M = 120 s was evaluated for a MSD engine of 500 kW/cylinder. For a large bulk carrier of 200 000 t displacement and 14 knots speed, 2 × 10 cylinder units with 1000 kW/cylinder would be used as prime movers. Their time constant could be about T_M = 160 s. With a propeller efficiency of 0.6 the time constant of the ship will be T_S = 800 s, so the ratio is $T_S/T_M = 5$.

This means that after $t = T_S = 5\ T_M$ the main engine is delivering practically the full energy losses $\dot{Q}_M$ to the heat exchanger. By then the ship has reached the speed $v_M = 0.5$. When the main engine is started and brought immediately to the power $P_M = 0.5\ P_{MCR}$, the corresponding energy losses are $\dot{Q}_M = 0.4\ Q_{MO}$ (see Fig. D2). The speed of the ship at this power would be $v_M = 0.79\ v_{MO}$ and after $t = T_S$ the ship will have the speed $v_{M(t)} = 0.395\ v_{MO}$. As the sea-water flow through the scoop is proportional to the ship's speed, the ratio of the energy losses to the cooling water flow at $t = T_S$ is just the same as at full-power design conditions.

$$\dot{Q}_M/\dot{m}_K = 0.4\ \dot{Q}_{MO}/0.395\ \dot{m}_{KO} = \dot{Q}_{MO}/\dot{m}_{KO}$$

This means that the outlet temperature of the sea water at $t = T_S$ will be the same as the design outlet temperature. With $t > T_S$ the outlet temperature ϑ_{SA} will go down, when the power is not increased. As any power increase above $P_M = 0.5\ P_{MCR}$ will happen gradually with these rather big diesel engines, there should always be enough cooling water før the heat exchanger, as long as the ratio of the time constants does not exceed 5.

Owing to size limitations, each engine of the bulk carrier used as an example has to be fitted with its own scoop cooler. At the moment we see no way to design a scoop cooler with natural circulation for more than 6500 kW. (This means a main engine power of about 1000 kW.)

Mr Hatchman thinks that the extra propulsion-resistance caused by scoops might prevent their application to small and medium-size merchant ships. The extra resistance is so small that one cannot see any influence on the rudder. Anyhow, the resistance of the lip, which has a height of 20% of the boundary thickness, is not more than the resistance of an old fashioned cooling water outlet, where the cooling water leaves the ship at 90° to the wall, with a speed of up to 2.5 m/s. It is our opinion that this kind of cooler is better appreciated with small and medium size merchant ships with a power range of 2000–5000 kW and a speed range of 12–16 knots.

Mr Robinson raises several questions on the propeller proportional frequency and voltage: the first concerns the cooling of the electric motors. When these motors drive a pump or fan not working against a constant pressure, then their load—and the losses—decrease as the cube of the frequency. As the coolant flow of the shaft-mounted fan decreases in proportion to the frequency, there should be always a surplus of coolant flow at reduced frequencies with these systems.

For motors driving pumps or compressors working against a constant pressure, the torque—and therefore the current—do not decrease with the frequency and the voltage. Here the energy losses stay more or less constant and here the ventilation has to be controlled and, if necessary, improved.

We do not think that, with modern equipment, a system's voltage will collapse at low frequencies, when a big electric motor is started automatically. Tests were made by Professor Dr Droste from TU Hamburg-Harburg on a ship with a CP-propeller and a main-engine driven generator. During the test, the main engine speed, frequency and voltage were reduced to 70% of the normal values. When the low frequency was reached, all standby pumps started automatically without a voltage collapse.

Naturally the light output of the navigation lights will decrease with the square of the frequency. These lights therefore need bulbs with a higher power to ensure that their output at the lowest frequency and voltage still conforms with regulations.

In conclusion I would like to emphasize, again, that the simplest systems should be designed and put on board merchant ships. It may be necessary to redesign parts of systems to adapt them to the new philosophy.