Heat Exchangers for Internal Combustion Engines

H. E. UPTON, O.B.E. (Member)*

The paper describes tubular heat exchangers used for the purpose of cooling (*a*) engine coolants; (*b*) lubricating oil; and (*c*) the pressure charge air. The performance of these three heat exchange components has an influence on the efficiency of the engine while it functions as a machine for converting the heat in the fuel into shaft horse power.

Information is given on heat distribution, the systems of which the heat exchangers form part, while the following aspects will also be discussed:—

The closed system of engine cooling.

Oil coolers, their performance with reference to thermal instability and coring in low sea water temperatures.

Automatic temperature control and valve arrangements.

Pressure charging and the effect of charge air cooling.

The paper will also examine details of design materials and general maintenance.

INTRODUCTION

The internal combustion engine is a machine designed to convert the heat content of the fuel into useful work in the form of shaft horse power. Therefore, it is proposed in this paper to discuss the three principal heat exchangers and the associated systems which materially contribute to this end.

There are, in marine applications, tubular heat exchangers for (*a*) engine coolants; (*b*) lubricating oil; and (*c*) pressure charge air. The two former are necessary because during the process of converting the heat into mechanical energy the ratio of work obtained to heat input has a limiting value; the remaining heat must be rejected to the coolants or the exhaust gas at the end of each working cycle.

Such are the combustion temperatures that the materials used in the construction of the engine would be unable to withstand for any length of time the conditions imposed, unless coolants were employed to maintain the jackets, pistons, covers, bearings, etc., at reasonable temperatures.

Pressure charge air coolers materially increase the output of an engine without increasing the average combustion temperature or the heat to the jackets, etc.

The exhaust gas boiler and other heat recovery equipment are not included because their performance should only be related to the overall fuel utilization of the vessel.

DISTRIBUTION OF HEAT

The distribution of heat in the fuel will vary according to the design characteristics of the engine, but, in general, the total heat available to heat transfer is largely independent of design; distribution will depend upon whether the engine is normally aspirated or pressure charged. Fast running engines will surrender more heat to the coolant because there is less radiating surface per pound of fuel burnt, while the heat transfer through the metal of the engine working parts is better in smaller engines, so higher coolant temperatures can be accepted.

The total percentage of heat rejected to the jackets, covers and pistons, appears to be independent of the stroke/bore ratio of the engine, but the heat rejection may vary to each of these individual components. An engine of longer stroke/bore ratio will reject more heat to the jackets but less to the covers and pistons than a "square" engine. Reducing the engine speed does not proportionately reduce the heat rejected to the coolants, for in some designs the rejection remains fairly constant over a wide speed range, a factor which affects the heat exchanger particularly in the case of engine driven circulating pumps.

Regarding temperatures, tests carried out at Manchester University showed that by raising the jacket outlet from 77 deg. F. to 185 deg. F., the fuel consumption was decreased by approximately 8 per cent, while the friction in the engine was also reduced. Tests have also shown that by raising the water temperature from 75 deg. F. to 175 deg. F. at constant engine speed, the amount of heat transfer to the cylinder walls was reduced from 26 per cent to 20 per cent of the heat in the fuel. High water temperatures in themselves are not harmful provided the corresponding metal temperatures in the liners are maintained in the region of 200/230 deg. F. and adequate lubrication is not destroyed.

Tests on high speed Diesel engines carried out by Williams and Wilson(1) show the effect of temperature on cylinder wear. J. L. Koffman and also G. B. Fox(2) show that a lower percentage of heat is rejected to the coolant as the b.m.e.p of an engine is increased. From a medium speed high duty engine at 60 b.m.e.p. it was 36 per cent of the heat in the fuel; at 120 b.m.e.p. it was 29 per cent, while for a larger but slower speed engine, at 60 b.m.e.p. the heat rejected was 25 per cent and at 100 b.m.e.p. only 19 per cent. The figures are typical of other cases analysed. These are the general factors which influence the heat dissipation, so the following table, which is given for guidance, includes typical temperature and circulation rates for the coolants.

* Director, Serck Radiators, Ltd., Birmingham.

TABLE 1

Speed range, r.p.m.	Low speed	Medium speed	High speed
	90–200	250–750	1,000–2,000
Heat distribution, B.Th.U./b.h.p./hr.			
Jackets and covers	1,100–1,500	2,000–2,300	2,500
Pistons	300–400	450–500	—
Lubricating oil	90–120	130–170	200–250
Circulation rates, gal./b.h.p./hr.			
Fresh water to jackets and covers	5·5–7·5	13–15	16
Fresh water to pistons	1·5–2·0	3·0–4·0	—
Lubricating oil	1·3–2·0	2·0–2·5	3·0–4·0
Temperatures, degs. F.			
Jackets and covers, Pistons	120/130	140/150	160/180
Lubricating oil	120/130	130/140	140/160

NOTES—1. For high speed engines, circulation rates may be higher in certain types.
2. American engines are usually run at substantially higher temperatures.
3. Doxford engines operate at 135/150 deg. F. for fresh water and 160 deg. F. from pistons.
4. Where pistons are oil cooled, circulation is two or three times the normal lubricating oil rate.

To determine the size of heat exchangers involves a study of heat transfer. However, a broad outline of approach is given later, but reference can be made to specialist books[3, 4, 5].

WATER COOLANT SYSTEMS

The cooling of jackets, covers, etc., requires careful consideration, for in the case of an engine developing, say, 5,000 shaft horse power and operating for 250 days per annum, the throughput of coolant is in the order of 1,350,000 tons annually or 250 times the fuel consumption.

In the early engines with open passageways, the use of sea water in a straight-through system was attractive, but any apparent advantages were outweighed by the attendant risk of cracked liners, jackets and covers.

Fig. 1 (Plate 1) shows the scale accumulation in a cylinder cover of an engine using direct sea water cooling, while Fig. 2 (Plate 1) illustrates the effect of sea water corrosion on a cylinder liner. The closed fresh water system of cooling with the use of a tubular heat exchanger is now widely adopted because it eliminates the risk of scale and sludge deposits; it permits the system to be operated at the best temperatures with a close temperature gradient across the engine, independent of the sea water temperature, while the introduction of thermostatic temperature control will reduce supervision and,

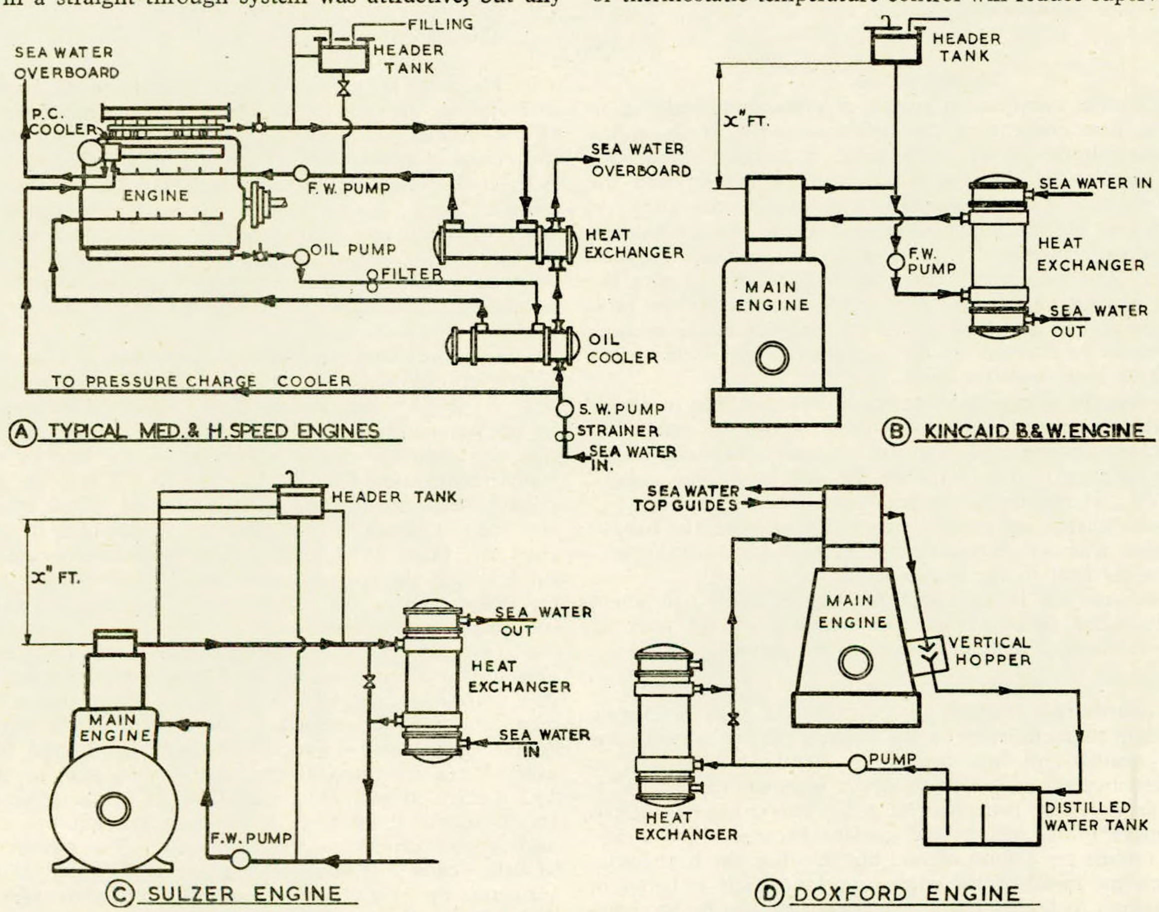


FIG 3—*Four examples of water coolant systems showing the relationship between engine, pump, heat exchanger and header tank in the system*

equally important, it should enable the engine to be operated at predetermined temperatures.

After-cooling without scale deposition, and a negligible make-up, can be included in the advantages.

In early installations the series system was common practice where the water from the engine was discharged directly into an open or covered overhead tank, a pump drawing from the tank and discharging through the heat exchanger and engine back again to the overhead tank. The disadvantage of this system is that hottest water is discharged into a large tank, with resultant aeration and vapour loss by evaporation. The system now favoured for some main propulsion sets and most of the medium and high speed installations is the parallel circuit, where a pump draws cooled water from a heat exchanger and discharges through the engine and back to the heat exchanger. A small header tank to accommodate the expansion of the water (about 10 per cent by volume) provides against casual leakage and is connected by a small bore pipe to a point nearest to the pump suction. The parallel system is suitable for installations without head room, for the tank can be sited with its water level just above the highest outlet point from the engine. Fig. 3 shows the parallel system and also how three different builders place the four principal components—engine, pump, heat exchanger and header tank—in relation to each other. A Kincaid/Burmeister and Wain system arranges the heat exchanger on the fresh water pump discharge; the water, after passing through the engine, returns to the pump suction; the header tank is connected to the pump suction on the engine outlet. Under running conditions, this system puts the highest pressure on the heat exchanger, so in the case of a tube failure, the fresh water would leak to sea; although this guards against contamination, there is a risk of losing water, so an alarm in the header tank is sometimes fitted.

A Sulzer arrangement provides for the heat exchanger on the fresh water pump suction, the engine being on the pump discharge, the water from the engine circulating back to the heat exchanger. The header tank is connected by a small bore pipe to a point nearest the pump suction. To avoid risk of surge and depression at the pump suction, the pipe should be of ample bore, of the shortest length possible and free from awkward bends and pockets. The heat exchangers should also be designed for high flow rates with baffles and tube pitch which will ensure low pressure loss over long periods between cleaning. The system is suitable for the highest temperatures as the pump deals with cooled water. In case of tube failure there would be no loss of water; on the other hand, the tank would fill and overflow, denoting the condition; there would, however, be some contamination. At all speeds the engine is under the highest pressure in the circuit.

Both these systems call for maximum height of header tank above the engine commensurate with the outlet pressures, an arrangement possible in most large marine propulsion installations. It is necessary to provide ample air escape pipes arranged with suitable slope, depending on whether the header tank is forward or aft of the engine.

The Doxford arrangement differs from the other two because the distilled water tank is below the engine, from which a pump draws and discharges through a heat exchanger to the engine. The outlets from the engine flow into a vented hopper and gravitate to the distilled water tank for recirculation. This engine is designed to operate at high temperatures, so it is desirable to place the pump as near to the tank as possible because it is drawing the hottest water. It is necessary in this arrangement to avoid aeration and on shutting down there should be sufficient capacity in the tank to contain the distilled water in the system. Under running conditions the pressure in the distilled water heat exchanger is greater than the sea water.

For high speed engines a development of the parallel system has been the introduction of the pressurized system, usually operating at between 5 and 10lb. per sq. in., where the entire system is sealed by means of a pressure cap fitted to the header tank in order to raise the operating pressures, with correspondingly high temperatures to reduce the size of all components associated with the system. As the operating temperature depends upon pressure, it is possible when operating at high temperatures for the pressure at the pump suction to fall below the corresponding temperature; for example, at 180 deg. F., a pressure of at least 1·31b. per sq. in. (3ft. head) above atmosphere is needed if cavitation is to be prevented, otherwise surge may occur, with consequent breakdown in the circulation system.

In pressurized systems, allowance must be made for expansion of the water, the separation of the steam and air, the prevention against vapour bubbles forming and the loss of water when the pressure breaks on cooling. Fig. 4 shows a combined heat exchanger and oil cooler with suitable header tank for high speed engines. The pressure cap (Fig. 5) embodies a spring loaded valve which opens when the pressure in the system exceeds the setting, while a vacuum valve is incorporated which will open after a partial vacuum is created when the engine has cooled. The cap is also used in combined units even though the temperatures are below the boiling point. Fig. 6 (Plate 1) shows a unit on a Ruston six-cylinder marine engine of 112 b.h.p. at 1,500 r.p.m.

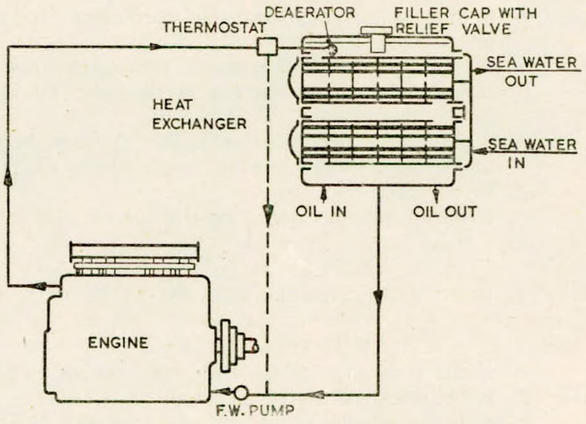


FIG. 4—*Pressurized coolant system including combined heat exchanger, oil cooler and header tank*

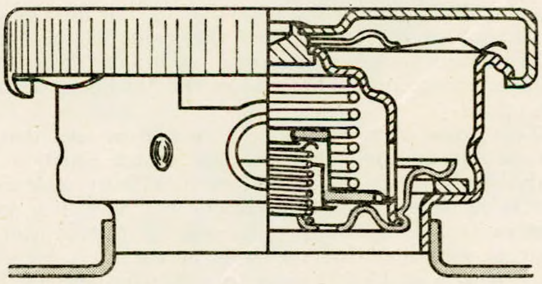


FIG. 5—*Combined pressure cap and relief valve*

THE FRESH WATER HEAT EXCHANGER

The variables which govern the design of a heat exchanger are heat dissipation, the circulation rate and the inlet temperature of the primary fluid, the circulation rate and the temperature of the sea water, the allowable pressure losses and the rate of heat transfer.

The heat transfer performance of a tubular heat exchanger may be expressed by the equation:—

$$E = U \theta A_o \qquad \text{.........................(1)}$$

where E, the rate of transfer of heat from one fluid to the other, is defined as B.Th.U./hr; U, the overall heat transfer coefficient from one fluid to the other, as B.Th.U./hr. deg. F. ft.^{2} of outside tube surface; θ, the logarithmic mean temperature difference between the two fluids, as deg. F.; and A_o the area in ft.^{2} of outside tube surface. E must be equal to the rate of rejection of heat from the engine to the jacket water cooler in order to give stable temperature conditions. The value of E, then, is dependent on the type, size and conditions of operation of the engine, and the heat exchanger must be designed to provide sufficient cooling when the engine is working under conditions of maximum heat rejection E to the jacket water. The value of E is a major factor in determining the size of a heat exchanger. The overall heat transfer coefficient for a fresh water heat exchanger can be expressed by the equation:—

$$\frac{1}{U}=\frac{1}{h_o}+r_o+r_w+r_i\left(\frac{A_o}{A_i}\right)+\frac{1}{h_i\left(\frac{A_i}{A_o}\right)}\quad\text{..........(2)}$$

where: U is the overall heat transfer coefficient (B.Th.U./hr. ft.^{2} deg. F.)

h_o is the partial heat transfer coefficient from the jacket water to the outside of the tubes (B.Th.U./hr. ft.^{2} deg. F.)

h_i is the partial heat transfer coefficient from the tubes to the sea water flowing inside them (B.Th.U./hr. ft.^{2} deg. F.)

r_o is the fouling resistance on the outside of the tubes $\left(\frac{1}{\text{B.Th.U./hr. ft.}^2\text{ deg. F.}}\right)$

r_i is the fouling resistance on the inside of the tubes $\left(\frac{1}{\text{B.Th.U./hr. ft.}^2\text{ deg. F.}}\right)$

r_w is the resistance of the tube wall $= t_w/k_w$

where: t_w is the thickness of the tube wall (ft.)

k_w is the conductivity of the tube material (B.Th.U./hr. ft. deg. F.)

A_o/A_i is the ratio of outside tube surface to inside tube surface.

Note: In equation (2), U, h_o, r_o and r_w are referred to the outside surface area (A_o) and h_i and r_i to the inside surface area (A_i).

The tube material has a sufficiently high thermal conductivity to enable the factor r_w to be ignored in the estimation of U; r_o and r_i represent the resistance to heat flow of films of scale or dirt on the outside and the inside of the tubes respectively, their values depending upon the conditions of operation, increasing with time as the thickness of the film builds up.

With reference to equation (2), it can be seen that if any of the factors on the right-hand side of the equation is considerably higher than the other factors, a fairly wide variation in the value of the remaining factors will have little influence on the value of U obtained. Ignoring for the moment r_o and r_i, that is, assuming that the walls of the tubes are perfectly clean, if h_o is considerably lower than h_i, variations in the value of h_i will have little effect upon the value of U. In this case h_o is said to be the "controlling" partial heat transfer coefficient. The converse applies if h_i is appreciably lower than h_o.

The general effect of the extremes in the partial heat transfer coefficient is, therefore, of this order:—

$$\frac{1}{h_o}+\frac{1}{h_i}=\frac{1}{U}$$

$$\frac{1}{300}+\frac{1}{900}=\frac{1}{225}\quad\text{........................(3)}$$

$$\frac{1}{300}+\frac{1}{1{,}800}=\frac{1}{257}$$

Therefore, by doubling the sea water partial coefficient, the overall heat transfer is increased by only 14 per cent, and as pressure loss varies as G^2 and the pumping power as G^3, where G is in lb./ft.^{2}/hr., care must be exercised in the choice of design, because on the fresh water side, i.e. outside the tubes, it is not always possible physically to make such a change in this mass flow G, particularly in engines with engine driven pumps. High values of h_o and h_i result from high velocity of the fluid; however, high velocity means that the heat transfer is expensive in terms of pressure loss; a good heat exchanger must represent a balance between theory and practice.

Further, if r_o and r_i are of the same order of value as, or lower than, $\frac{1}{h_o}$ and $\frac{1}{h_i}$, these two fouling factors will have an influence upon the heat transfer coefficient U. The higher the values of h_o and h_i, the greater will be the influence of r_o and r_i. If conditions are such that the partial heat transfer coefficients are high, greater allowance must be made for the effect of fouling resistance than if the partial coefficients are low.

Overall coefficient U can vary between 200 and 400, depending upon the conditions of performance and service.

θ, the logarithmic mean temperature difference, is dependent upon the inlet and outlet temperatures of both jacket water and sea water, and for the present purpose, can be regarded as approximately the mean of the temperature difference between the jacket water and the sea water at the two ends of the heat exchanger.

With reference to equation (1), it will be seen that if E and U remain constant, the surface A_o will be inversely proportional to the value θ. Therefore, if a heat exchanger is designed for, say, the following conditions:—

Fresh water 130-110
Sea water... .. ... 105- 85 θ = 25 deg. F.

and if it were possible to establish the fresh water temperatures 25 degrees higher, the M.T.D. would be 50 deg. F. and the unit would only be half the original size. Conversely, it shows why units designed for tropical conditions are twice the size of those designed for temperate conditions.

CONSTRUCTION

Tubular liquid/liquid exchangers, in which the primary fluid flows outside the tubes, are the types most used for marine purposes because they are the ideal shape to withstand pressure. Their form permits flexibility in design; they are reliable in operation, easy to service and overhaul, and are low in first cost.

Modern units mostly comprise a cylinder bored to receive a separate tube stack which has been turned to a predetermined diameter to register with the cylinder bore. The tube stack is allowed to expand as a unit, for normally the tubes are mechanically secured to both tube plates, one of which is fixed; the other is allowed to expand freely in a gland or expansion joint. Water boxes with covers complete the assemblies.

The heat transfer is materially affected by the baffle arrangement, for fluid, deviating from the predetermined flow path either through clearance between tube and the hole in the baffle or through bypass between baffle diameter and cylinder, contributes little to heat transfer and makes it impossible to get consistent results as between heat exchangers of similar size.

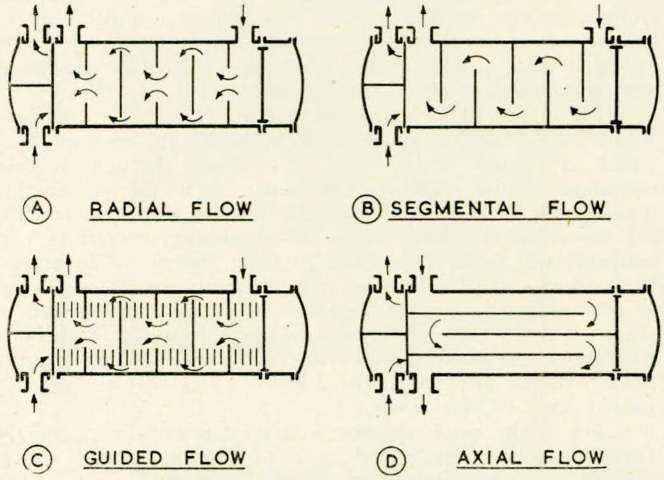


FIG. 7—*Four types of baffle arrangement used in tubular heat exchangers*

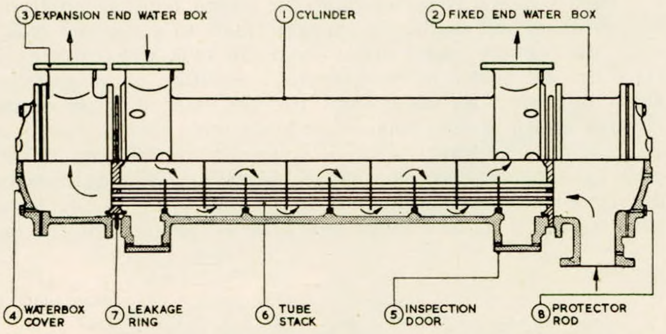


FIG. 8—*Marine type tubular heat exchanger showing distribution belts, inspection doors and removable covers in the water boxes*

Considerable thought has been given to the development of tube layout, baffle arrangement and disposition of surface. Fig. 7 shows four types of baffle arrangement in fresh water heat exchangers; the radial flow arrangement (A) ensures that the fluid flows radially inward and outward at right angles to the tubes for the greatest percentage of the swept surface, an important factor in heat transfer. The flow path is also relatively consistent between the outer and inner tube rows. It is important to minimize leakage past the large baffles; for this reason sealing rings (Fig. 8) are fitted.

An excessive clearance between baffle and cylinder, say, in the order of 0·045 inch over the normal allowable, reduces the partial heat transfer by approximately 10 per cent, but the amount of clearance between tube and baffle does not seem to have a very serious effect on heat transfer. However, it is essential to maintain close limits between tube and baffle to prevent vibration.

The segmental flow arrangement (B) demands close attention to the tube layout because the tube lanes are not of equal length across the diameter of the tube stack and at right angles to flow. Compared with the radial flow designs, less baffles are required for the same mass flow, which means there is less right angle flow, while there is greater tendency for side leakage.

Excessive clearance between the baffle and the cylinder can reduce the partial heat transfer coefficient up to 15 per cent, while 0·015 inch additional clearance in the tube holes reduces this partial by a further 10 per cent. In this arrangement there is a third source of leakage, namely, the side clearance between the outer rows of tubes and the cylinder wall, which can have up to 5 per cent effect.

The guided flow (C) is primarily used for oil coolers to achieve right angle flow and provide secondary surface. Axial flow (D), is used only in special conditions where there is considerable mass flow and very close temperatures between outlet fresh water and inlet sea water, and where it is desirable to have extremely low pressure loss.

The difficulties are in regard to control of the fluid path and provision for support; although the diagram shows an expansion type tube stack, the arrangement is usually associated with designs having two fixed tube plates. Ferrules are no longer used as a means of securing the tube to tube plate joint. Mechanical expanding enables closer tube pitch and ensures greater safety against leakage.

Where tubes are fully roller expanded into plates, the tube ends should be annealed to compensate for the effects of work hardening. When full expanding is not required to obtain a fluid tight joint, the tube should be finished to a controlled grain size and hardness without end annealing, because work hardening of the tube will be negligible. L. Baker[(6)] refers to actual cases which have occurred in practice. An additional precaution against leakage is obtained by bonding the tubes to the tube plate after the expanding operation.

In designs with two fixed end tube plates it is necessary to control the amount of roller expanding because the tubes will bow appreciably, due to about 70 per cent of the work on the tube going inwardly along the horizontal axis of the tube. A desirable feature in marine heat exchangers is to ensure adequate distribution at the entry to avoid build-up of deposit with consequent increase in pressure loss. Fig. 8 shows a typical unit with distribution belts, inspection doors and removable water boxes and covers which enables examination to be carried out without breaking the main pipe joints. The size of a heat exchanger depends upon the amount of cooling surface, the arrangement and distribution and the dimensions of the fluid paths, so the use of small tubes of close pitch is attractive.

The following table shows the effect of the tube diameter on matrix density for a constant pitch ratio of 1·25 : 1, and the relative heat transfer.

TABLE II

O.D. of tube, in.	Relative surface area	Relative heat transfer		
		Water through tubes, per cent	Water over tubes, per cent	Oil over tubes, per cent
1	1·00	100	100	100
$\frac{3}{4}$	1·33	141	150	154
$\frac{9}{16}$	1·78	199	224	237
$\frac{7}{16}$	2·28	270	318	346
$\frac{1}{4}$	4·00	528	698	800

Care must be exercised in the choice of tube and the layout, otherwise the condition can occur in a fresh water heat exchanger (Fig. 9, Plate 2) with disastrous results to the engine. Notice that on the sea water side the tube entry is quite clean.

Tubes with a pitch ratio of 1·34 : 1 are suitable for large units, and 1·25 : 1 for small heat exchangers. Tubes, $\frac{1}{4}$ inch diameter, are used for special applications where extreme compactness is the first essential in design.

Water Treatment

The original filling or the make-up for recirculated water systems should be either distilled or soft water. Should very hard water have to be used, steps must be taken to avoid the deposition of scale in the heat exchangers and the cylinders, covers, etc. Under marine conditions, elaborate control of the fresh water in the system is hardly possible, so it is probably better to depend upon treatment compounds such as the tannins, rather than to use softened water. In addition to their ability to prevent the insoluble salts forming scale, these compounds form a protective film on all surfaces in the system; they also tend to reduce the free oxygen in the water, which aggravates corrosion.

The following table gives the percentage of calcium carbonate deposited from waters of various hardness.

TABLE III

Water temperature, deg. F.		Cooling water: bicarbonate concentration, grains per gallon — Temporary hardness, degrees				
		14	28	42	56	70
104	Untreated	nil	6	23	50	—
	Treated	nil	nil	nil	nil	1
122	Untreated	nil	18	27	—	—
	Treated	nil	nil	nil	1	1
140	Untreated	4	22	32	—	—
	Treated	nil	nil	nil	$1\frac{1}{2}$	$6\frac{1}{2}$
158	Untreated	8	30	44	—	—
	Treated	nil	nil	3	9	—
176	Untreated	20	36	47	—	—
	Treated	nil	1	8	17	—

In using this table, consideration should be given to the fact that water temperatures in the cylinder head are apt to be considerably higher than the nominal figure, particularly after a shut down.

Inhibitors

The use of inhibitors in the fresh water system of marine engines has many attractions, but unfortunately this method of protection against corrosion is not without problems. Many substances reduce the general corrosion in the system, but very few do so without, at the same time, creating local areas which may be liable to attack in an even greater degree. If this latter condition occurs it is far more dangerous to the security of the system than would be a more general attack which, although removing a greater weight of metal, is slow in action.

The inhibitors under consideration may be divided into two groups—anodic and cathodic types. It has been said that the former are efficient, but need great care in application, while the latter are less effective, but safe. The difficulty when using inhibitors is to get correct concentration because weak solutions can always promote local attack; such conditions can also occur in unswept pockets, even though the inhibitor concentration in the system as a whole may be up to strength.

Inhibitors favoured for fresh water systems include sodium or potassium dichromate, but sea water leaking into the system can create a dangerously corrosive condition with particular attraction for ferrous materials. P. Jackson[7] states that, in Doxford engines, potassium dichromate plus sodium nitrite added had given satisfactory results.

The problem, however, with this form of treatment is the lack of a simple means to determine if the percentage of sodium nitrite is up to strength, whereas with potassium dichromate a rough guide is the colour of the coolant.

Soluble oils are another type of inhibitor which require care in application; slightly weak concentrations can be corrosive and they also attack natural rubbers. Synthetic rubbers, however, are immune from attack. Excessive concentration can create a jelly-like substance which will settle in pipes if the mass flow is below the critical velocity.

Soluble oils cause considerable frothing in a system which can be reduced or prevented by adding a hardening powder to maintain the *p*H value above 8. The tannin inhibitors are safe and effective.

OIL COOLING SYSTEMS

Lubricating oil functions as a lubricant and a coolant. Cooling is necessary to maintain the oil at a viscosity which will carry the bearing loads at all speeds, but within the range which will keep reasonably high temperatures without breakdown of the oil film. The viscosity should be sufficiently low, however, to reduce the friction loss and avoid generating heat which must be dissipated in a cooler; it should also be such as to ensure that the bearings are always charged with oil. Temperature control and cooling preserves the chemical stability of the oil and guards against oxidization, sludge formation and corrosion attack on bearings. Heat to oil, which may vary between 2 per cent and 8 per cent of the power developed, depends upon the load, allowable clearances, speed, and the power/weight ratio, the extent to which it is used for cooling pistons and the area of the crankcase. With trunk piston engines having a skirt projecting into the crankcase and where splash or spray lubrication is used, it is not possible to forecast the heat rejection and such a method of lubrication is inadequate at low speeds and heavy loads. F. Nixon[8] states that the heat rejection varied plus or minus 30 per cent in a batch of high speed engines. To cool adequately and lubricate, the rate of circulation is more important than clearances, while excessive pressure may cause the oil to spray out of the bearings and atomize in a heated crankcase, to oxidize and form sludge which will accumulate on the tubes of the oil cooler. Forced feed systems are most common where the oil is drawn from a sump or drain tank and pumped to the bearings through filters and cooler; oil from the bearings mixes with oil from the crankcase and drains by gravity to the drain tank for recirculation.

In dry sump systems a scavenge pump removes oil from the crankcase and discharges through filters to a reservoir tank, while the pressure pump draws from this tank and delivers oil via filter and cooler to the bearings. Aeration in oil systems has a bad effect on cooler heat transfer rates and can occur through splash in the crankcase or drain tanks; cold oil suction allows air to be drawn into the system if the viscosity is too high to allow an easy oil flow to the pump. On quick-start high-speed engines, flick pressures of 150lb. per sq. in. have been registered in oil coolers and as it is essential for the oil

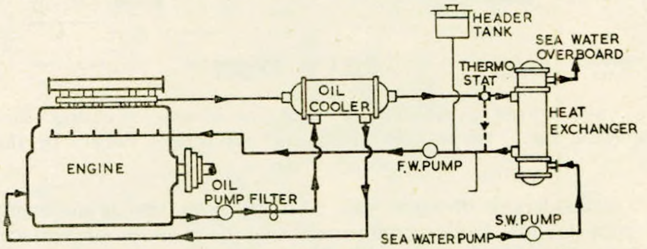


FIG. 10—*A closed system with oil cooler fitted in the engine coolant circuit*

to reach bearings at the remotest part of the engine, an automatic bypass of cooler is good practice; the oil pipes should be large enough to allow cold oil to be delivered without undue pressure loss. Where pipes have been too small, a pressure loss of 2½ to 3lb. per sq. in. per ft. of pipe has been registered. Relief valves which return oil direct to the suction before the cooler, mean a debit against cooler size and an additional risk of starved bearings under cold conditions.

As operating oil temperatures increase, and where a 20 deg. F. mean temperature difference between oil and water can be tolerated, engine-water cooled oil coolers (Fig. 10) are worth consideration, for Fig. 11 shows how quickly operating temperatures may be reached compared with an oil cooler using raw water. Another advantage is that full flow coolers can be used with low pressure loss, at the same time maintaining high rates of heat transfer over long periods, while in extremely cold climates coring and congealing conditions do not occur.

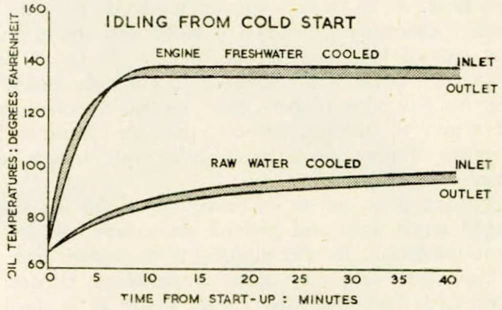


FIG. 11—*General effect on oil temperatures when engine water is circulated through the oil cooler*

THE TUBULAR OIL COOLER

The heat transfer performance can be expressed by equation (1) as before, E being the rate of heat rejection from the engine to the oil; θ is fixed by the operating temperatures and the size of the oil cooler is influenced by the tube diameter and pitch ratio.

In the case of oil coolers, the partial heat transfer coefficient between the oil flowing over the tubes and the tube wall is almost invariably "controlling" and the effect is of this order:—

$$\frac{1}{h_o}+\frac{1}{h_i}=\frac{1}{U}$$

$$\frac{1}{70}+\frac{1}{420}=\frac{1}{60} \qquad \text{......................(4)}$$

$$\frac{1}{70}+\frac{1}{840}=\frac{1}{65}$$

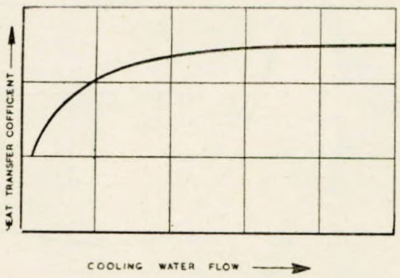


FIG. 12—*Effect of cooling water flow on the overall heat transfer in an oil cooler*

The two-fold increase in the sea water partial coefficient has only increased the overall heat transfer by 8 per cent.

The effect of this is that an increase in cooling water flow over a certain range of operation will provide only a very small increase in the value of U. This is demonstrated in the graph shown in Fig. 12. The viscosity of the oil at operating temperature has an effect on the oil side partial heat transfer coefficient, its value decreasing with increase in viscosity. The fouling factors r_o and r_i, mentioned in the fresh water heat exchanger notes, must be taken into account in the

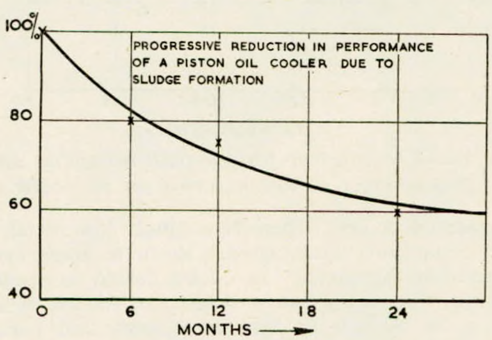


FIG. 14—*General progressive reduction in performance of a piston oil cooler, due to sludge formation*

estimation of performance, but whereas r_i will be of similar value to that in the fresh water heat exchangers, r_o for clean oil may be negligible until conditions happen to affect the cooler, as shown in Fig. 13 (Plate 2), which is a tube stack from a piston oil cooler after a period of service, while Fig. 14 shows how the general performance deteriorated due to the accumulative condition.

Practical values of the overall heat transfer coefficient U vary between 70-120, depending upon the performance and service.

STABILITY OF OIL COOLERS

Oil coolers operating in extremely cold conditions can be subject to what is known as coring, which is really a problem of instability, a condition experienced in service and on the test bed.

In a given size of oil cooler, increasing the flow of cooling fluid or reducing its temperature will, up to a point, give increased cooling of the oil. After a certain point, however, further increase in cooling fluid flow will lead to a rise in the temperature of the fluid being cooled, contrary to expectation. This phenomenon is known as coring.

The trouble is really the sum of two separate effects, i.e. coring, properly so-called, and parallel path instability. Dealing first of all with true coring, imagine, as shown in Fig. 15, oil passing through a cooler and, say, water passing on the other side of the heat transfer surface and acting as a cooling fluid. What happens on the water side is not of real concern (indeed, in certain applications this may be replaced by air or other fluid); the phenomenon of coring takes place entirely on

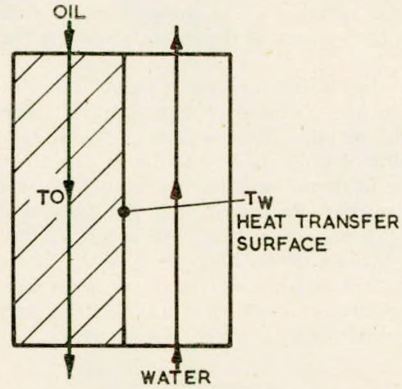


FIG. 15—*True counterflow conditions in an ideal tubular oil cooler where the liquids are separated by metal heat transfer surface*

the oil side. Imagine, therefore, the mean oil temperature in the cooler as being T_o and the metal temperature of the heat transfer surface as being T_w. The cooler is performing satisfactorily and the value of T_w is reduced either by increasing the water flow or decreasing the water temperature. If the rate of heat input to the oil system is constant, it can be shown that the mean oil temperature T_o will take up a value given by the equation: —

$$T_o = T_w + c\mu_w^{1/7} \qquad \text{.....................(5)}$$

where T_o and T_w are defined above, c is a constant depending upon the conditions in the cooler, and μ_w is the viscosity of the oil at the temperature of the metal wall T_w.

T_o and T_w in deg. F., μ_w in lb./ft. hr.

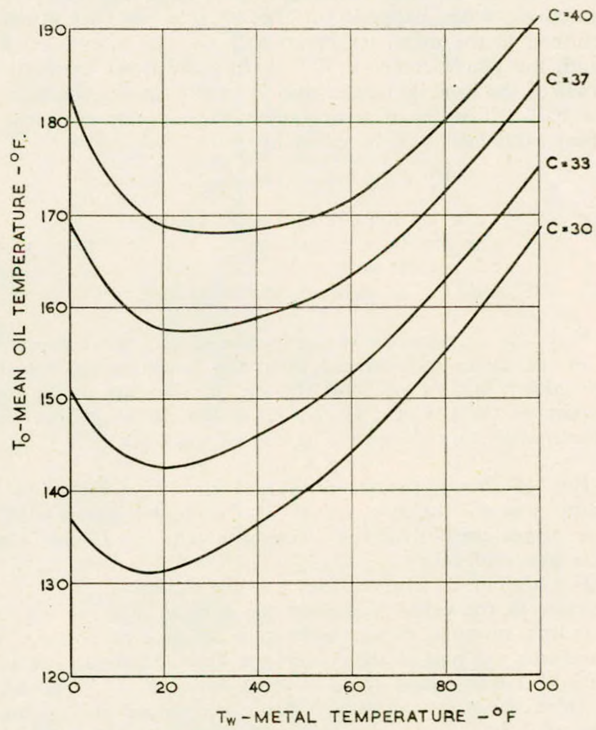


FIG. 16—*Coring curves depending upon the conditions in the cooler and viscosity of the oil at the temperature of the tube wall*

Fig. 16 shows T_o plotted against T_w for various values of c and for one particular oil.

As can be seen from these curves for each value of c, reducing T_w below a certain figure causes a rise instead of a fall in T_o. The mechanism of this effect is that, as the temperature of the metal wall is lowered, so a film of cold oil is built up on the surface of the metal, reducing the heat transfer coefficient.

At first, the reduction of the heat transfer coefficient is not as great as the increase of temperature difference between the oil and the metal. After a certain point, as shown on the curves, the film of cold oil builds up so rapidly that a further slight decrease in metal temperature leads to so great a decrease in the heat transfer coefficient that the total amount of heat transfer falls instead of rising. As a result, the oil temperature T_o rises instead of falling.

The effect is cumulative, since as soon as this happens the metal temperature may fall still further and lead to the cooler going practically entirely out of action.

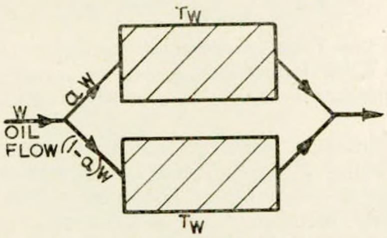


FIG. 17—*Diagram of parallel path conditions in an oil cooler*

The second effect, parallel path instability, can also be analysed mathematically. Fig. 17 shows an idealized picture of the sort of thing that happens. Imagine the cooler being divided into two equal portions as shown, and the oil flow being divided between the two. Cooling fluid passes through both halves of the heat exchanger, but again interest is concentrated on what happens on the oil side, so that attention is confined to the metal temperature T_w. The rate of oil flow through the whole cooler is W; of this, aW flows through the top half of the heat exchanger and $(1 - a)W$ passes through the lower half. The mean temperature in, say, the top half of the heat exchanger will be given by:—

$$T_o = T_1 - \gamma \frac{(T_1 - T_w)}{a^{0.67}} \quad \text{........................(6)}$$

where: T_1 is the oil inlet temperature in deg. F.
γ is a constant depending on the conditions in the cooler, and
T_o and T_w as defined above in deg. F. and W in lb./hr.

Now the oil pressure loss, q lb. per sq. in., through the top half of the cooler will depend upon the flow through this half of the cooler, viz. on aW, and also on the viscosity of the oil in this part of the cooler. It can be shown to be proportional, approximately to $a\mu$ where μ is the oil viscosity at T_o, i.e.:—

$$q \propto a\mu$$

Fig. 18 shows, for several values of $\gamma(T_1 - T_w)$, how the pressure loss in the two halves of the cooler varies with a. These curves are plotted for a constant value of T_1 and a particular grade of oil.

For high metal temperatures (i.e. low values of $\gamma(T_1 - T_w)$), a decrease in the value of a from the normal value of 0·5 (corresponding to equal flows) leads to a decrease of pressure loss through the top half of the cooler; the flow will tend to equalize itself and a will return to its normal value of 0·5. As can be seen from the curves, however, for lower values of T_w (higher values of $\gamma(T_1 - T_w)$), this is not the case. A slight accidental decrease in a, i.e. a slight decrease in the flow through the top half of the cooler, will lead to an increase in pressure loss. This increase in pressure loss will, in turn, lead to a still further decrease of flow. The whole arrangement is thus unstable and flow through the top half will decrease to a very small value. The dotted curves in Fig. 18 show the corresponding pressure losses in the lower half of the cooler.

Both these effects have been observed in actual oil cooling installations, although they tend to be very much more serious

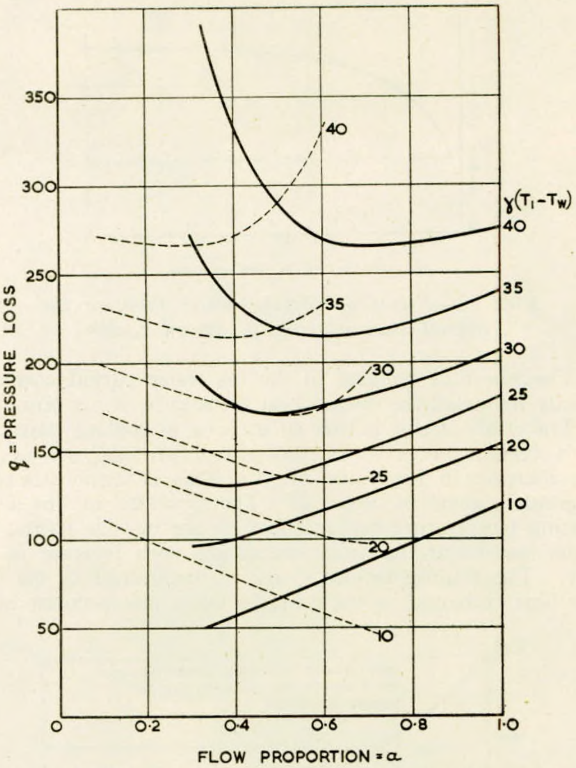


FIG. 18—*The effect of parallel-path instability on the pressure loss variation across an oil cooler*

with aircraft oil coolers, where exceedingly low metal temperatures are commonly encountered. Both, to some extent, can be countered by arranging the cooler design in special ways. For example, true coring can be delayed by the use of secondary surface on the oil side of heat exchangers, and parallel path instability, which is very troublesome indeed in aircraft oil coolers, can be countered by fitting special stabilizing devices.

CONSTRUCTION

Tubular oil coolers in marine practice are similar in construction to the fresh water heat exchangers but it is even more essential to control the clearances between tube stack, tubes, cylinder, etc.

In radial flow oil coolers, excessive clearance between baffle and cylinder incurred a 16 per cent reduction in performance.

Fig. 7(c) shows the guided flow oil cooler, designed to obtain the maximum right angle flow and at the same time provide an amount of secondary surface where the metal temperatures and the partial heat transfer coefficients permit high rating.

Tube diameters and pitch ratios for oil coolers are of the same order as those for fresh water heat exchangers.

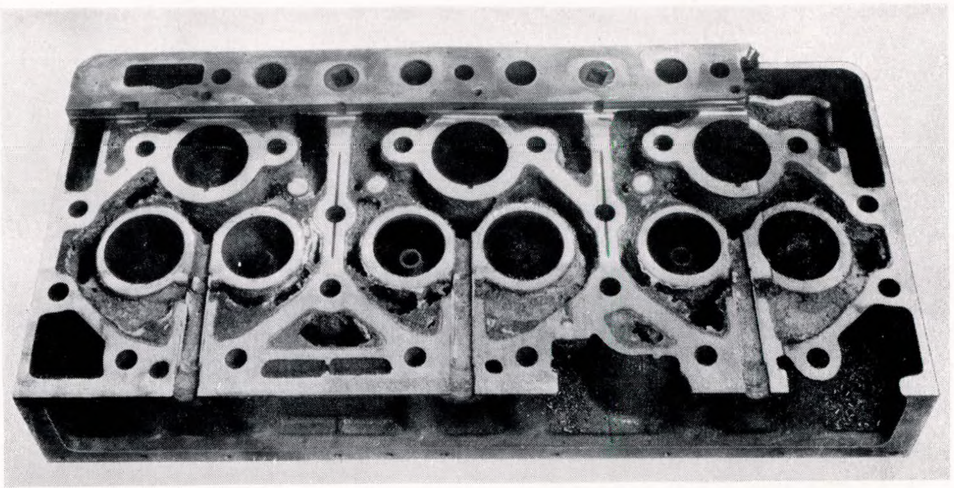
MATERIALS

Cylinders

Cast iron is the most generally used material, welded steel construction being used when low weight is necessary, but it is not suitable for cylinders of water to water heat exchangers with fixed tube stacks and where rusting may occur. Gunmetal is occasionally used because it resists corrosion and withstands underwater explosion shocks. Aluminium alloys have a limited application, but are used where extremely light weight is of advantage and non-magnetic properties are desirable. It is essential to shot blast any interior surfaces in contact with oil.

Water Boxes

Merchant practice favours cast iron, gunmetal being used on routes with very unfavourable water conditions, particularly in the East Indies, where cast iron is liable to graphitization,



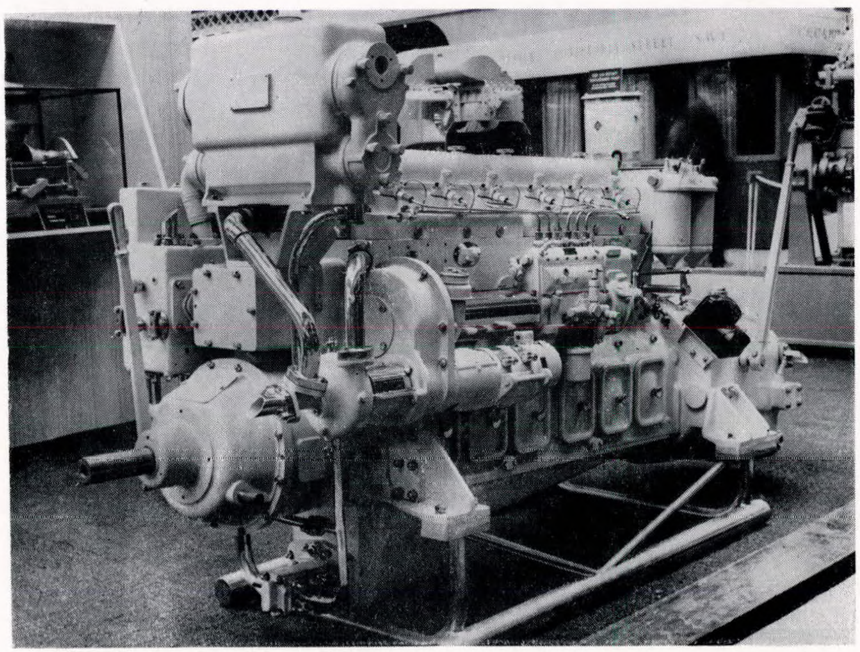
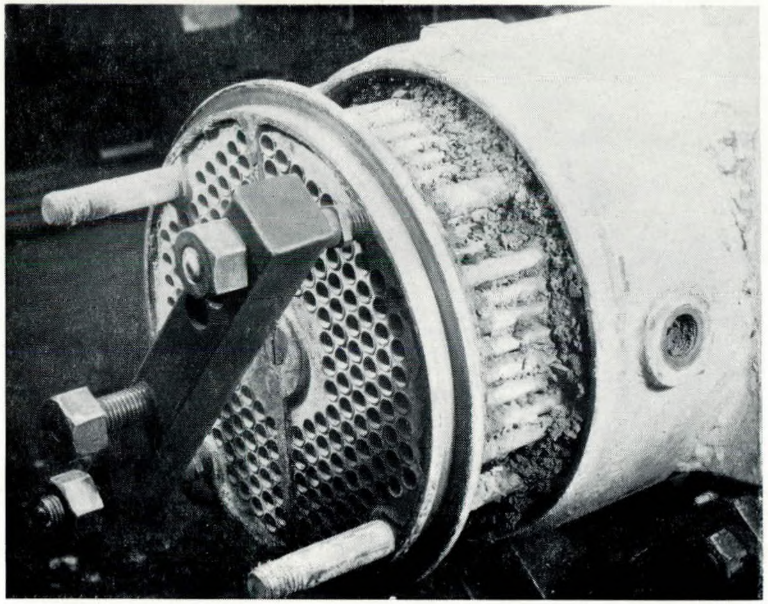


FIG. 1 *(top left)—Scale accumulation in a cylinder cover of an engine using direct sea water cooling*

FIG. 2 *(top right)—Corrosion of a cylinder liner by sea water*

FIG. 6 *(bottom left)—Combined unit mounted on Ruston marine engine*



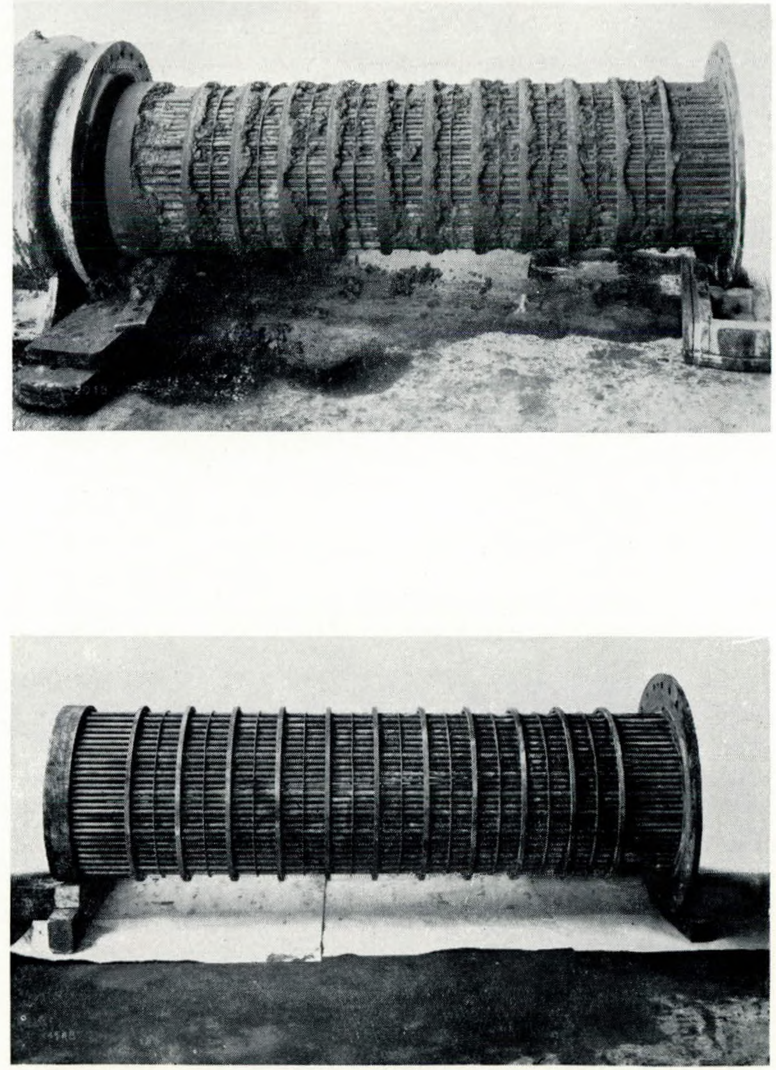


FIG. 9 *(top left)—Heat exchanger badly fouled on the engine coolant side after comparatively short service, but with sea water side in pristine condition*

FIG. 13 *(top and bottom right)—Piston oil cooler stack fouled by contaminated oil; before and after cleaning*

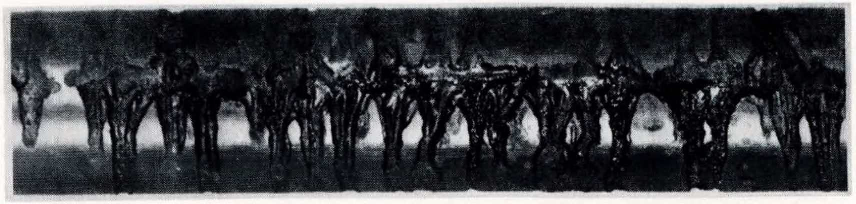
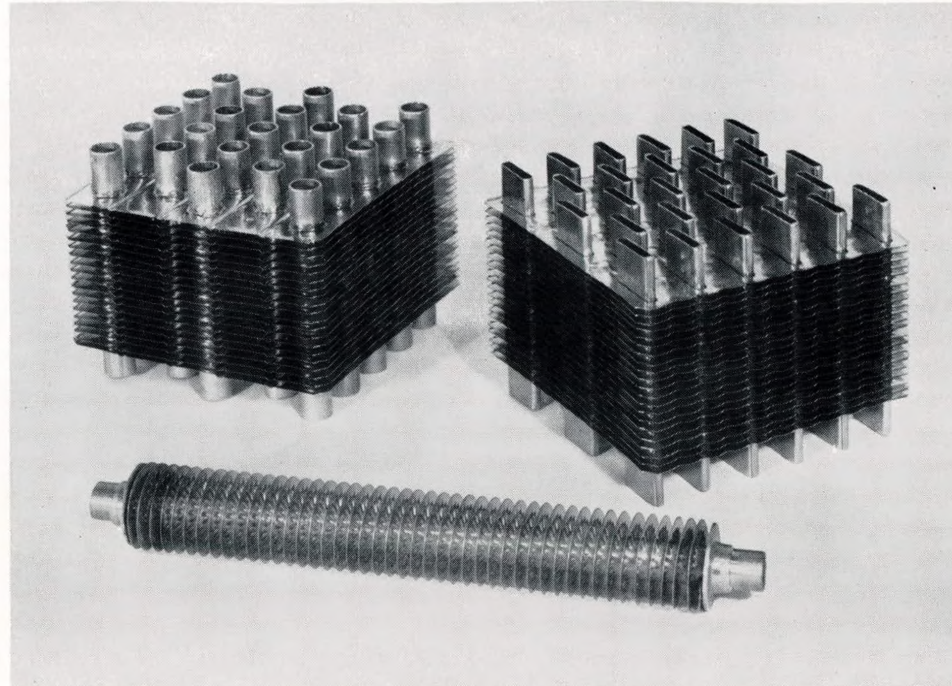


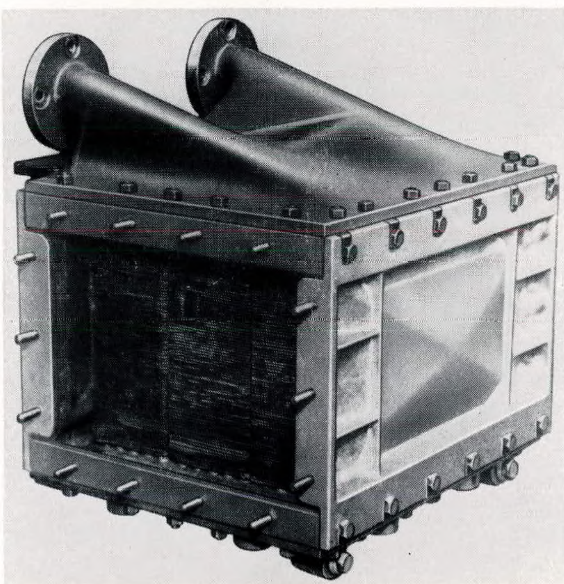
FIG. 19 *(top left)—The effect of oil corrosion on the outside of an aluminium brass tube*

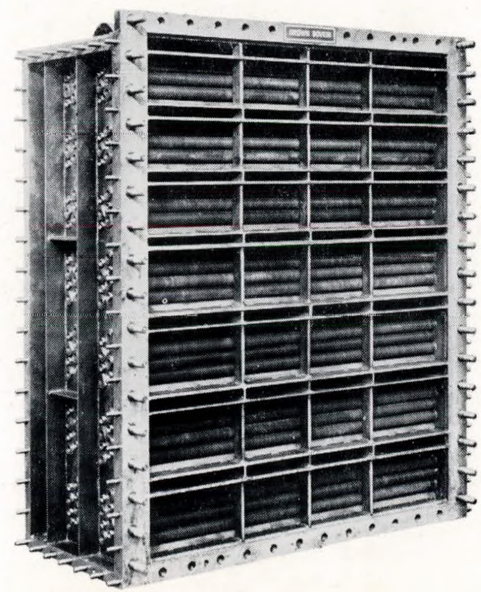
FIG. 21 *(top right)—Types of surface used in pressure charge air coolers*

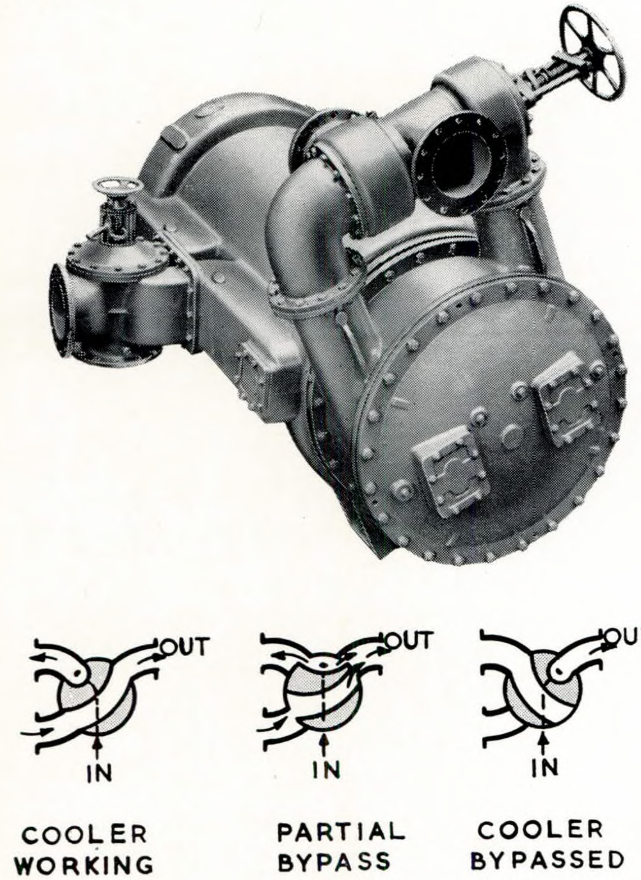
FIG. 24 *(bottom left)—Pressure charge air cooler with continuous fins*

FIG. 25 *(bottom right)—General construction of a ribbon tube Brown Boveri charge air cooler*









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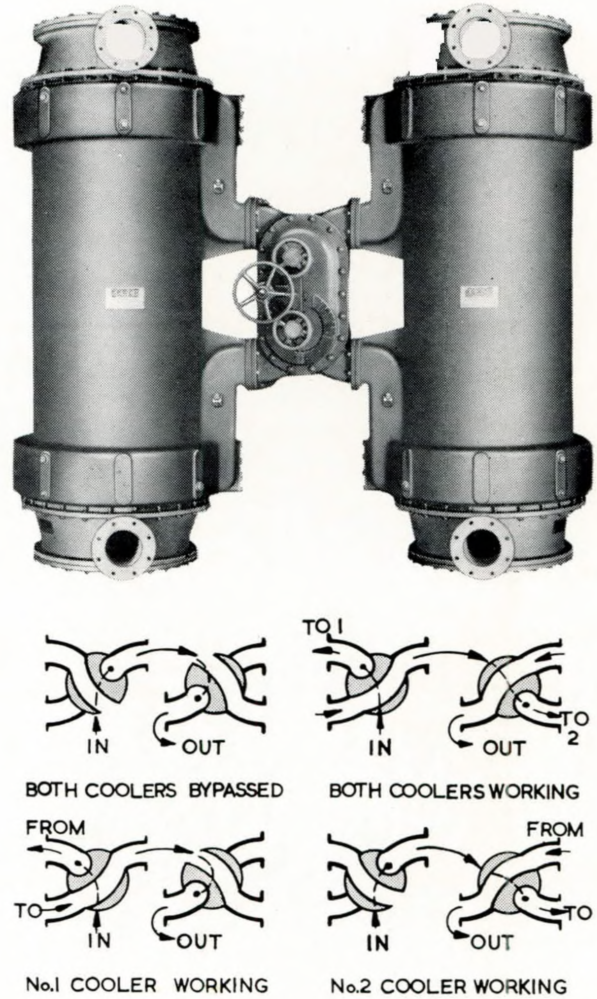

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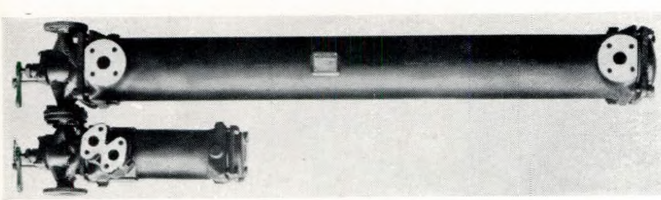

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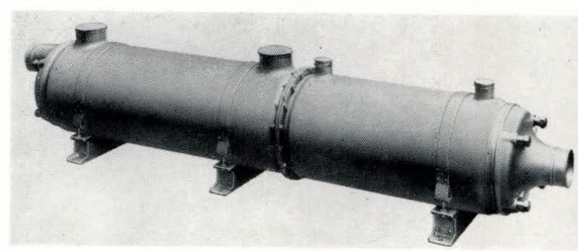

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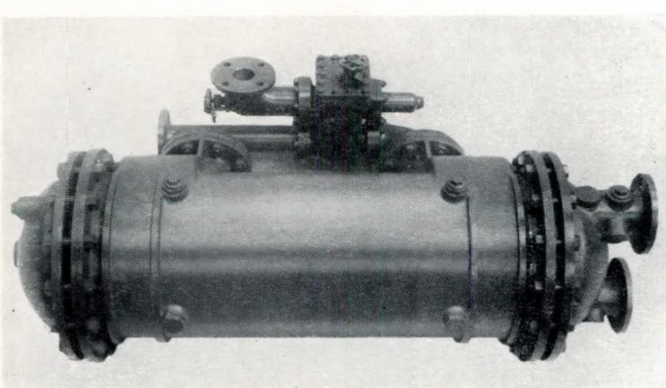

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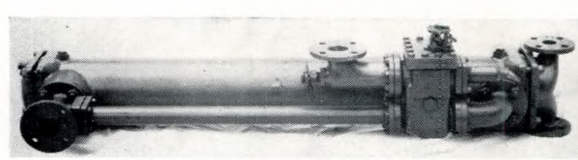

(f)

FIG. 26—*A group of marine heat exchangers fitted with valves*

a form of corrosion where the iron is converted to oxide, the graphite being left in contact; the resultant mass retains its original shape and thickness so porosity is noticed only when the attack has completely punctured the wall. Gunmetal should be thoroughly shot blasted to remove the film of graphite from the mould dressing, which is cathodic, and can have an adverse effect on heat exchanger tubes.

Tube Plates

Naval brass is used for marine heat exchanger tube plates. It can be subject to impingement attack, but it is rare provided the design of the water box is fundamentally correct. Plain 60/40 brass (Muntz metal) is occasionally specified, but there is no sound basis for such a choice. The addition of 1 per cent of tin in the naval brass is amply justified by its superior performance.

Tubes

For marine heat exchangers, aluminium brass or copper nickel are the standard tube materials. A recent development in corrosion resistant tubes is due to the work carried out by the British Non-Ferrous Metals Research Association, which has shown that the presence of a small amount of iron in copper nickel is of vital importance to its corrosion resistance and this iron content is now embodied in all relevant specifications. The table detailing the chemical composition of both aluminium brass and copper nickel to the relevant specification is given below:—

ALUMINIUM BRASS

Specification	*BS378,* per cent	*British Admiralty,* per cent
Copper	76-78	76-78
Aluminium ...	1·8-2·3	1·8-2·3
Arsenic	0·02-0·06	0·01-0·05
Total impurities ...	0·30 maximum	*0·30 maximum
Zinc	Remainder	Remainder

* of which lead must not exceed 0·03 per cent

70/30 COPPER NICKEL

Specification	*BS378,* per cent	*British Admiralty,* per cent
Nickel	30-32	30 minimum
Iron	0·4-1·0	0·4-1·0
Manganese ...	0·5-1·5	0·5-1·5
Total impurities ...	0·30 maximum	0·30 maximum
including sulphur	0·08 maximum	0·08 maximum
Copper	Remainder	Remainder

CORROSION

Corrosion can attack any part of a heat exchanger, but rarely does it affect any component other than the tubes.

The subject has been dealt with by Dr. P. T. Gilbert[9] and others in papers read before this Institute.

Oil coolers are occasionally subject to attack from the oil side and when it does occur it is most virulent. The effect can be seen in Fig. 19 (Plate 3) and it may be noted that the incidence of trouble has been higher since more main propulsion engines have been using heavy fuel oil. Efforts are being made to find an alloy which will resist this particular form of attack.

Polluted waters are stated to be the most important single factor in the failure of condenser and heat exchanger tubes at the present time, which emphasizes the need for their protection when the heat exchangers are first fitted on board and during the period the ship is being completed.

THE CHARGE AIR COOLER

The various methods of delivering air under pressure to an engine in order to increase the power is beyond the scope of this paper, but it is well established that with a moderate increase in air pressure of, say, 4·5lb. per sq. in. boost, by means of turbo-charging it is possible to get up to 50 per cent more power with little or no increase in thermal stress. No doubt in future turbo-charged four-cycle engines will operate continuously at twice the power output of normally aspirated engines of the same frame size. Unless engines are specially designed, the value of pressure charging is best when the boost is raised to a point where it is still possible to retain all the principal materials used in a normal engine and employ charge cooling to increase density. Therefore, a balance must be held between the requirements of pressure to get density and cooling to get density.

Charge cooling reduces exhaust temperatures for a given load and reduces the heat rejection per b.h.p. to the coolant.

Analysis of several engines normally aspirated and pressure charged shows an average increase in power output of 35-45 per cent with only slight increase in heat rejected to the jacket water, which means that fresh water heat exchangers need not be increased in size when the engines are changed from normally aspirated. Compressing the air in an engine cylinder raises the initial absolute temperature nearly three-fold before any of the heat in the fuel is released, so, when an engine is fitted with a pressure charger which raises the temperature of the air into the cylinder, the whole cycle temperature is raised by a corresponding amount and it is essential to cool the air, particularly when high pressure charging is used. The value of cooling the air after compression in the charger as a means of getting increased power is worth while, particularly when there is an ample supply of water available, as in marine applications.

Charge air cooling can increase the output of a pressure charged engine something like 10 per cent without raising the maximum cylinder temperatures to any appreciable degree. There is, however, little gained by using coolers for boost pressures less than 3½lb. per sq. in., for the advantage due to the reduction in air temperature is just about balanced by the back pressure of the cooler.

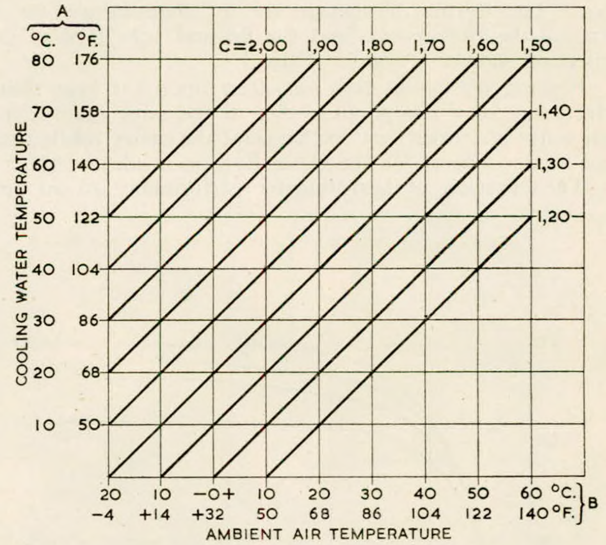

FIG. 20—*General limiting lines of economical cooling of charge air for various compressor ratios*

As a general guide, Fig. 20[10] shows the limiting lines of economical cooling of charge air for different pressure ratios (C) in the blower as a function of the cooling water temperature (A) and the ambient temperature (B). When the point of intersection between (A) and (B) is below the point of intersection for a given pressure ratio (C) then intercooling is worth while. It is possible with coolers of reasonable size and weight to cool the air to within 15 degrees of the sea water temperature. B.S.I. 649 Standards rate a normally aspirated engine on the basis of 85 deg. F. ambient and 14·45lb. per sq. in. absolute, the engine being derated 2 per cent for each 10 deg. F. rise above the standard temperature.

Conversely, cooling the air in a charge cooler would increase the power output, but not at the same rate because of the change in the turbo-charger compression pressure ratio. Tests[11] carried out on a pressure charge engine of 325 b.h.p. at 600 r.p.m. indicated that with constant exhaust temperature, the engine output was reduced 2·2 per cent for each 10 deg. F. rise above the ambient air of 85 deg. F., but when fitted with a charge cooler under similar conditions, the output decreased by only 0·8 per cent per 10 deg. F. rise, so on this basis the increase in the output of an engine would be 1·4 per cent for each 10 deg. F. of cooling, or 5·9 per cent when the air temperature is reduced in the cooler from, say, 142 deg. F. to 100 deg. F.

Tests from other sources have shown a greater increase in power when cooling through a similar temperature range, but even on a conservative basis, charge air cooling is worth while. Reduction in air temperature by refrigerated methods may eventually be used to increase the weight of the charge delivered to the engine, but as the surface metal temperature of the cooler determines the temperature limits which can be obtained without ice forming on the surfaces, temperatures less than 45-50 deg. F. should not be considered.

Pressure charge air coolers should be supplied with the coolest sea water and be of such dimensions that they can be arranged in the air manifolds adjoining the blower.

The same basic heat transfer equation (1) applies here as for other types of heat exchangers. Sea water is normally used as the coolant for charge air coolers; an important point here is that, regardless of the type of cooling surface employed, the partial heat transfer coefficient on the air side is always "controlling" to a marked extent. This calls for a high ratio of air side to water side surface area. In any unit using tubes as heat transfer surface, the desired ratio of surfaces can be obtained only by the addition of secondary surface to the outside of the tubes, the cooling water flowing through the tubes. Two forms of surface are in common use for this purpose—the ribbon tube and the fin and tube type of cooling section shown in Fig. 21 (Plate 3).

Fouling resistances have an effect upon the heat transfer performance of a charge air cooler in the same manner as in fresh water and other heat exchangers; the major fouling resistance in this case is for the water flowing inside the tubes.

The criterion of heat transfer performance in an air to water heat exchanger such as this is often taken as the "thermal ratio" defined as:—

$$\eta = \frac{(\text{Air temperature in}) - (\text{Air temperature out})}{(\text{Air temperature in}) - (\text{Water temperature in})}$$

The maximum value of η that is obtainable, assuming that the unit can be fitted with any desired value of cooling surface area, is dependent on the arrangement of water flows employed. Curves of η against specific cooling surface, defined as:—

$$\frac{\text{Cooling surface area} \times \text{overall heat transfer coefficient}}{\text{Mass air flow rate} \times \text{specific heat of air}}$$

are shown in Fig. 22 for various flow arrangements and for a ratio of air temperature drop to water temperature rise of 5 : 1, which may be taken as typical.

For counterflow, a value of 100 per cent is obtainable if sufficient cooling surface area is fitted, whereas for parallel flow only 84 per cent is possible. Single- and two-pass crossflow arrangements are those most widely used, giving intermediate maximum values of η. Fig. 23 gives the characteristics of a typical water cooled charge air cooler, showing how the mass water flow and mass air flow rates influence the thermal ratio.

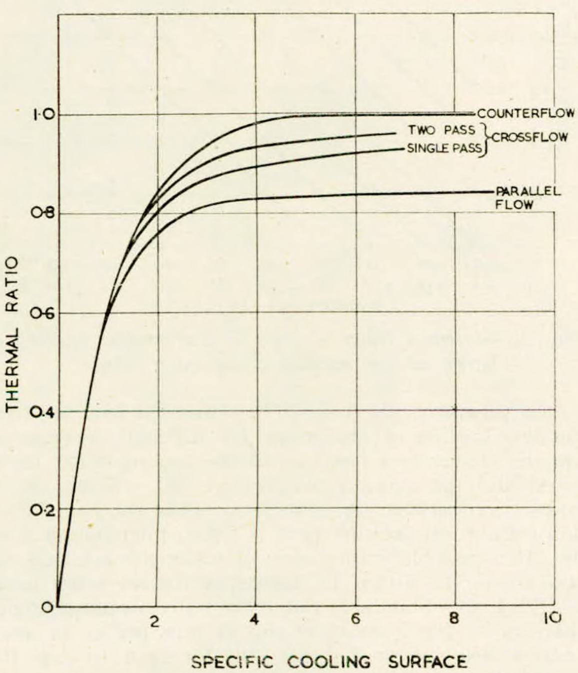

FIG. 22—*Effect of water flow arrangements on the thermal ratio of charge air coolers*

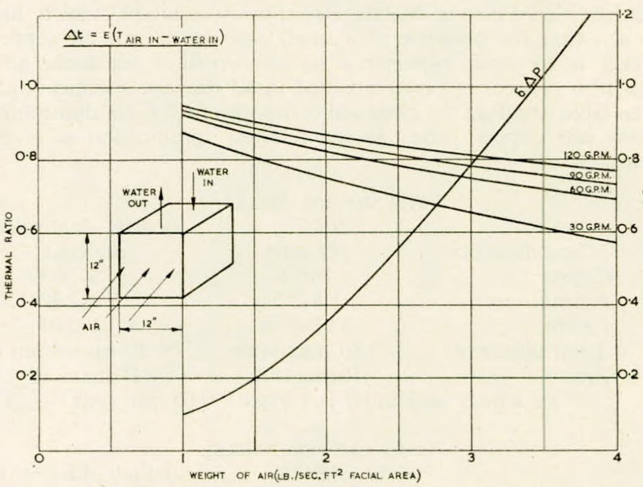

FIG. 23—*Performance characteristics of a water cooled pressure charge air cooler*

The cooling water curves which are given in g.p.m. show the influence any change in the water circulation rates will have on the thermal ratio of a cooler for a constant air flow. They also show how the thermal ratio is affected when the air flow is altered with the water circulation rate remaining constant. The air pressure loss ΔP across the cooler in lb. per sq. in. is read from the right-hand side of the curve.

The air pressure loss across a cooler must also be considered in relation to the characteristics of the compressor and the requirements of the engine and mass flows of 6,000 to 7,000lb. per hr. per sq. ft. at 0·375lb. per sq. in. pressure loss, can be accepted with fin spacings approximately eight to the inch.

CONSTRUCTION

Fin and tube is the usual form of surface used in charge air coolers although there are designs where only primary surface is adopted. All forms of secondary surface benefit from turbulence promoters on the air side and, within the range of normal secondary surface types using tubes of average form, the surface per unit volume is nearly independent of the tube diameter, hence a reasonable diameter can be accepted.

The ribbon tube cooler has individual tubes expanded into tube plates fitted with separate water boxes; as individual tubes, they are not always easy to support against vibration, while the ribbon which forms the fin is not coincident in the direction of air flow, which makes the cleaning of the air side more difficult.

With the matrix type of cooler having continuous plate fins, the surface per unit volume is increased; at the same time the continuous fins support adjacent tubes. The fin plates are straight on the leading and trailing edges to enable them to be easily cleaned.

To obtain the maximum heat transfer and as a means of protection against the effects of the salt laden charge air, the surfaces are tinned. Separate water boxes and side plates, together with inspection doors, are usually fitted in this type. The tubes are $\frac{9}{16}$ inch diameter or $\frac{7}{16}$ inch diameter, with fin spacings six or eight per inch. Fig. 24 (Plate 3) and Fig. 25 (Plate 3) show typical charge air coolers.

INSTALLATION OF TUBULAR HEAT EXCHANGERS

Heat exchangers are static pieces of equipment, frequently installed in inaccessible places. To get the best operating results the fresh water system should be vented by means of small bore pipes connected to the header tank or balance pipe. Tubular units should be installed vertically, but when placed horizontally the sea water connexions should be arranged in at the bottom and out at the top. They should be installed below the lowest water line and, when vessels are trading in muddy water, single flow coolers should be installed vertically, with sea water discharging down through the cooler. Pumps should be arranged to discharge sea water through any tubular heat exchanger. Foaming and aeration of oil is minimized by long route low velocity draining and tanks with adequate foam settling spaces. Pump suction should be carefully arranged as foaming can be caused by cavitation when the oil is highly viscous. Oil pumps should be arranged to avoid pockets where oily water may accumulate. The air separated in oil tanks should be vented to atmosphere.

OPERATIONAL

Vapour locks reduce the active surface and, in pressurized systems, failure to remove vapour and steam will create cavitation and eventually breakdown of the system. Full flow on the fresh water side must be maintained, and fresh water temperatures should be controlled by regulation of the sea water at outlet from the heat exchangers.

In oil coolers the fluid should reach near operating temperature before turning on the sea water, to prevent chilling and the build-up of a viscous film on the tube walls. The tubes of heat exchangers are vulnerable because of the widely different operating conditions encountered, so it is necessary to keep out any likely obstruction and regular attention should be given to strainers. The excessive use of snifting valves on reciprocating sea water pumps causes aeration and increases the risk of corrosion and, in certain cases, creates shock waves and high frequency, low amplitude vibrations.

MAINTENANCE

Scale, sludge, carbon deposits and marine growths cause fouling and, therefore, all heat exchangers should be regularly inspected and cleaned, particularly on the sea side; increase in temperature and pressure loss are a measure of performance and the fouling rates.

During long periods of lay-up, any deposits in the tubes should be removed and, after a general wash out, the units should be filled with clean water; under no circumstances should the unit be kept only half filled for any length of time.

For cleaning fresh water and sea water surfaces, acid base compounds are suitable, while for the oil side alkaline and phenol liquid degreasants or vapour solvents are now available.

VALVES

One function of heat exchangers is to maintain temperatures at predetermined levels, so increasing use is being made of specially designed and matched hand operated valves and automatic thermostatic valves, assembled as components of the main unit. Oil valves should have no position where they can be shut off against the oil supply to the engine.

Fig. 26 (Plate 4) shows a group of marine heat exchangers fitted with valves.

(*a*) An oil cooler with a single rotary oil valve for the purpose of controlling the oil flow, while the triple seat valve controls the sea water through the cooler and bypass.

(*b*) Two coolers connected by a Duplex rotary valve so that by means of one handwheel, each cooler or bypass can be controlled.

(*c*) A combined heat exchanger and oil cooler arranged in series with bypass valves fitted on the sea water side.

(*d*) A tandem arrangement of heat exchanger and oil cooler for high speed craft.

(*e*) A heat exchanger fitted with a self-actuating type of thermostat to give predetermined temperature control of the engine water.

(*f*) An oil cooler with a similar type of thermostat.

THERMOSTATS

Thermostatic valves maintain, within a prescribed band, constant temperatures at a particular point in a system. There are several types and the choice depends upon individual requirements, but the aim should be to provide units which are reliable in service and easy to maintain. The result will be greater efficiency of heat exchangers and elimination of the demands for supervision of the coolant systems.

The essential components of a valve are:—

(*a*) A temperature sensitive element capable of producing a mechanical or electrical signal which is dependent upon the temperature of the element.

(*b*) A device for converting the signal produced by the temperature sensitive element into a movement of the valve mechanism.

(*c*) A valve mechanism suitable for controlling the liquid concerned.

Self-actuating thermostats often have vapour- or liquid-filled bellows where the element expands or contracts with changes in temperature, the movement being transmitted directly to the valve mechanism. The disadvantage of bellows is that their operating temperature is affected by the pressure of the liquid surrounding them and, since this depends upon the flow and the pressure necessary to maintain it downstream of the element, the valve may surge when the operating temperature is reached and may result in extreme pressure fluctuations in the system.

A wax-operated thermostat has been developed in this country and the U.S.A. because it is largely insensitive to pressure changes. In operation the wax changes volume due to temperature and expands a piston, which forces a diaphragm or rod against the valve mechanism. The reaction of the force is taken by a suitable spring. Such thermostats are robust and not easily damaged.

Thermostatic valves used in large installations should have an over-riding manual control in case the temperature sensitive element fails.

Remotely operated self-acting valves have as their sensitive element a bulb containing vapour or liquid which is connected by capillary tube to the valve mechanism. The pressure in the thermostat moves the valve mechanism in one direction and a spring moves it in a reverse direction.

Externally powered valves can be fitted with various types of thermostat; one commonly used is the metal expansion stem type which is one of the simplest temperature sensitive elements; a tube of suitable metal is anchored at one end and closed at the other, expansion and contraction of which is transmitted to the actuating mechanism by an "invar" rod, which is attached to the closed end of the tube. The rod operates a servo device which depends upon a supply of compressed air, liquid under pressure, or electrical power to transmit the signal from the element to the valve.

Installation

Thermostatic valves may be installed so as to maintain a constant hot liquid temperature either at the inlet or the outlet from the engine; the position to some extent affects the control characteristics.

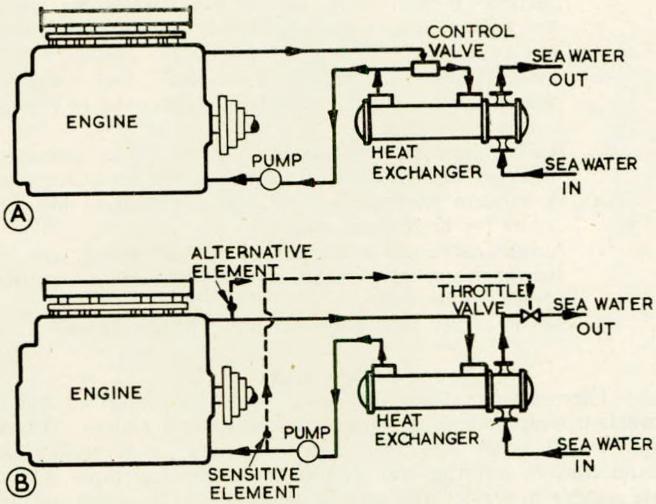

FIG. 27—*Coolant system (a) with thermostat controlling the flow of engine coolant through the heat exchanger, and (b) controlling the flow of sea water through the heat exchanger to maintain engine coolant temperature*

In all thermostatic devices, time must elapse between a correct adjustment being made by the temperature sensitive element and the resultant temperature change being sensed by the element. During this period the element will continue to call for valve adjustment so that overcorrection will occur; this will then induce a correction in the opposite direction with a tendency towards oscillation. If the element controls on the engine outlet temperature, there will be a greater time lag, with the result that there will be a greater tendency towards hunting. The effect is most noticeable if the element is very rapid in its response to temperature variations and the hot liquid has a low rate of flow. This consideration, however, does not render it unsatisfactory to have a valve controlling on the engine outlet, provided care is taken in the design to minimize this effect.

In general, there are two ways in which a thermostatic valve may be used to regulate temperature; the first is where the flow of the hot liquid through the cooler is restricted and the remainder bypassed, finally mixing with the stream which has been cooled; and, secondly, where the hot liquid continuously flows through the cooler, the valve being arranged to control the flow of cooling water.

Fig. 27(a) illustrates a typical cooling arrangement which uses the first method. Hot liquid from the engine enters the valve (which in this case is controlling on the engine outlet temperature) which directs some through the cooler and bypasses the remainder.

The two streams mix and return to the engine. Under full load conditions it is arranged that at the operating temperature the bypass port of the valve is closed, causing all the hot liquid to circulate through the cooler, and when the engine is first started the valve permits all the liquid to bypass the cooler. If the valve operates in relation to the engine inlet temperature, the mixing of cool and bypass streams usually occurs in the valve and always upstream of the temperature sensitive element.

Fig. 27(b) illustrates the type of control where the flow of water through a cooler is regulated in accordance with the temperature at any given point in the hot liquid system. This arrangement is particularly suitable when it is necessary to economize in the supply of cooling water.

Both methods have the advantage that there is always a constant flow of coolant through the jackets, which avoids the risk of hot spots and vapour formation in the upper parts of the jacket and covers. The maintenance of flow has obvious advantages for lubricating oil systems.

CONCLUSION

Tubular heat exchangers have a beneficial effect on the overall efficiency and performance of marine oil engines and, when fitted with reliable automatic controls, valves, etc., they require less supervision and make the unit independent of the human element.

Closed Water Coolant Systems

These systems provide:—

(*a*) A more even temperature gradient across the engine with less thermal stress in the components.

(*b*) The establishment of predetermined operating temperatures, independent of the sea water temperature.

(*c*) The control of higher temperatures and recirculation rates without the risk of scale formation.

(*d*) Pressure cooling at higher temperatures would mean less heat rejection, improved engine performance and a reduction in the size of heat exchangers and other components.

Oil Cooling Systems

By the use of an oil cooling system:—

(*e*) Constant viscosity of the oil for lubrication purposes is maintained.

(*f*) Predetermined temperatures can be established when the oil is used as a coolant.

(*g*) Carbonization of the oil is prevented.

Charge Air Coolers

(*h*) The output of pressure charged engines is increased by a further 5 to 10 per cent, with little or no increase in thermal stress in the engine.

(*i*) The derated engine condition is restored to normal rating, due to very high ambient temperatures.

(*j*) Extremely small space is occupied for the results obtained.

ACKNOWLEDGEMENT

Grateful acknowledgement is made by the author to his colleagues at Serck Radiators, Ltd., for their help in the preparation of these notes.

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Discussion

REAR ADMIRAL(E) W. G. COWLAND, C.B. (Member) said that the paper was most excellent for several reasons.

The first reason was that it covered ground in which all marine engineers were interested. Many people—and he was one of them—knew very little about heat transfer. Yet all engineers were dealing with it every day of their lives in boilers, condensers, and so on. As far as he was aware, this was the first paper on heat exchangers for internal combustion engines that had been read before the Institute.

The heat exchanger was a necessity in these days for internal combustion engines. One could not get away from the oil cooler: the engines heated up the oil, and it had to be cooled. With regard to the water-jacket cooler, it seemed to him that heat stresses in the engine should be kept as low as possible. One of the easiest ways of doing that, to his mind, was to pass a lot of water through the engine. If a lot of sea water was used, it was a complicated business to avoid scale if it was kept up to a reasonable temperature. Much the best way was to use fresh water with a 10, 15 or 20 deg. F. rise through the engine and cool it in a heat exchanger.

The oil cooler was a necessity, and the water cooler was becoming a necessity, hence the value of the paper. Mr. Upton had covered the practical and theoretical side extremely well. He had given those who might be installing heat exchangers sufficient theoretical knowledge to enable them to argue with him.

He had known Mr. Upton for many years, and they had had a great many ups and downs. When he was asked to open the discussion on the paper, he had anticipated being able to criticize Mr. Upton on all sorts of points. Unfortunately, he was not able to do so. The paper was exceedingly good, and there were only three points about which he wanted to ask.

First of all, he suspected that the liner shown on the screen as salt-water corroded was actually eroded.

MR. UPTON agreed that this liner was eroded.

REAR ADMIRAL COWLAND, continuing, said that, secondly, oil coring occurred in the war in aircraft engines. As far as he knew, it had not occurred yet in marine engines. He imagined a fairly easy solution if it did occur was to be found in the use of a lubricating oil of lower viscosity. It would not be difficult, as ships did not flip from the Equator to the North Pole in a few hours.

Thirdly, Mr. Upton said that charge air coolers would increase the output of a pressure charged engine by 10 per cent. That figure was, he considered, too low. The actual increase was appreciably higher.

With regard to charge cooling, he was convinced that the turbo-charged two-stroke was coming, and air coolers would be an essential part. The turbo-charged two-stroke generally required a higher boost for a given increase in power than the four-stroke and, therefore, derived greater benefit from charge cooling.

An interesting point for people to think about, though it was somewhat of a red herring, was where to put the charge cooler. When a turbo-charger was in series with a scavenge pump, did one put the charge cooler between the two or after the scavenge pump? This would cause a lot of discussion in a few years' time but he must not elaborate on it at the moment.

DR. P. T. GILBERT said he had recently presented a paper on condenser tube corrosion in Birmingham with Mr. Upton in the Chair. On that occasion, Mr. Upton had dealt with him very kindly, and he did not wish to deal in any other manner with the present admirable paper. He was singularly unqualified, in any case, to criticize a paper of this nature. There were, however, a few points on which he would like to comment.

The first concerned the section on inhibitors (page 106). The main purpose of adding inhibitors in these systems was presumably to reduce corrosion of the ferrous parts. The inhibitors mentioned, with the exception of chromate, would not be likely to have much beneficial effect on the non-ferrous parts of the system. Mr. Upton had rightly drawn attention to the danger of certain types of inhibitor, which could give rise to an increased intensity of attack if present in insufficient concentration. Dr. Gilbert was not sure that it had been made quite clear that this applied to the anodic type of inhibitor.

The effectiveness of an inhibitor depended, among other things, on the composition of the water, and one of the most important factors, as Mr. Upton had pointed out, was the chloride content. The danger of contamination with chloride was particularly serious with anodic inhibitors, since an inhibitor concentration that was adequate with the normal cooling water might be inadequate if the chloride content rose and serious local attack might then occur.

The last sentence of the section on inhibitors was of interest: "The tannin inhibitors are safe and effective". He would like Mr. Upton to elaborate on that point since, if the statement were true, the inhibitor for which corrosion workers had been looking for many years had been found. The situation was perhaps not so simple as indicated in that sentence.

The use of cast iron for water boxes raised points of interest. Such a water box would protect the non-ferrous tube plate and the tube ends by electrolytic action in the early stages of service, being itself corroded. If the water conditions were bad and the water box suffered severely from graphitization, however, the time might come when large areas of the internal surface of the water box would be coated with graphite. The position could then be reversed, and the corrosion of the tube plate and the tube ends could be accelerated because of contact with the graphitized iron. The effect would be similar to that of the film of graphite from the mould dressing sometimes left on cast gunmetal water boxes, as mentioned by Mr. Upton. This situation was not easy to deal with. One had presumably to try to maintain some protective coating on the cast iron to prevent graphitization, though this was by no means easy. Alternatively, the use of sacrificial anodes in the water box might be desirable in some cases.

In the section on tube plates, it might be worth mentioning that the effect of adding 1 per cent tin to the brass (to give naval brass) was to reduce the rate of dezincification. Such an addition had no effect on the resistance to impingement attack, but it materially reduced trouble due to dezincification.

In the section on tubes, Mr. Upton referred to the improvement in the performance of cupronickel tubes brought about by the incorporation of a small amount of iron. This could hardly be described as a recent development, since the beneficial effect of iron in 70/30 cupronickel had been established in this country more than twenty years ago.

Finally, he would like to comment on corrosion from the oil side, as illustrated in Fig. 19 of the paper. He had had little experience of corrosion on the oil side as opposed to corrosion on the water side of heat exchangers, but this photograph appeared somewhat similar to a specimen that had recently been examined by his colleague Mr. May.

In this particular case, evidence had been found, on removal of corrosion products, of a small crack in the tube in the corroded region. It seemed possible that water had seeped through a crack in the top of the horizontal tube and run down on both sides of the tube, becoming concentrated in the process and causing deep grooves. It was not possible to be certain about this but he thought it was worth considering since it was possible that, in some cases, damage attributed to attack by the oil might prove to be due to attack by the water from inside the tube, cracking of the tube having occurred for other reasons.

MR. W. MCCLIMONT, B.Sc. (Member) said that his remarks would be concerned mainly with the heat transfer side. He could say a great deal about this, but he would try to be brief.

The sections of the paper which dealt with the details of the design of fresh water and lubricating oil heat exchangers and charge air coolers, together with their materials and maintenance were both interesting and valuable. He ventured to suggest that the rôles played by heat exchangers for engine coolant and lubricating oil were sufficiently well appreciated to make unnecessary the rather obscure explanations of their significance which the paper contained.

In equation (2), h_i was a partial heat transfer coefficient on the basis of inside tube surface area, the term (A_i/A_o) being used to correct it to the outside tube surface area. But when the author led one through the factors which might be ignored in order to produce equation (3) the basis of h_i was changed, the coefficient then being based on outside tube surface area, the term (A_i/A_o) being dropped. Such a change would only be valid if considering an infinitely thin tube.

The author's remarks on the comparatively small effect of leakage between tube and baffle in the disc-and-doughnut arrangement of baffles were of great interest and confirmed the view that such leakage was not, as Tinker suggested, lost as far as heat transfer was concerned. The corresponding figures given for segmentally-baffled exchangers were unfortunately not absolute, and as the effect of the tube hole leakage did not appear to be linear, one could not comment on the figures.

Considering now tube bundle to cylinder wall clearances, the author mentioned that this could have up to 5 per cent effect on the partial heat transfer coefficient. Tinker*, considering possibly more extreme cases, had claimed that complete elimination of this clearance would frequently increase h_o, the outside partial coefficient, by over 40 per cent—a much greater figure.

The effect of clearance between baffle and cylinder wall in a segmentally-baffled exchanger could be expressed from the correlation of published data as:—

$$\frac{1}{\left(1+\left(\frac{A_c}{A_b}\right)\right)^2}$$

where A_c was the clearance area between the curved edge of a baffle and the shell.

A_b was the net flow area through a baffle window. This ratio tended to one as the clearance was reduced to zero.

For the reduction of 15 per cent in the value of h_o quoted in the paper, the ratio A_c/A_b would be 0·085; that was to say, the area between the curved edge of the baffle and the shell would be 8·5 per cent of the net baffle window area. Could the author say whether this agreed with his data?

Finally, he thought the author's exposition of the conditions of instability which might occur in oil coolers was very valuable, though he wished he had used the original index of the Sieder and Tate viscosity ratio, namely 0·14, rather than the index 1/7 used in equation (5).

MR. J. CALDERWOOD, M.Sc. (Vice-President), said he had had a number of remarks to make on the paper, but as time was getting short he would be brief.

He would like first to emphasize the value of the paper from two points of view. In the first place, while the owners and in particular the engineers of the larger ships were fully converted to indirect cooling, this did not apply to many of the smaller ships; and certainly in land work overseas there were many who were not converted to the use of heat exchangers. If the paper converted a few more people who at the moment would not pay the extra cost of heat exchangers, it would be very gratefully received by every engine builder. There were still many people who would not fit heat exchangers because of the extra cost, though the engine builders wanted them to do so.

Secondly, the paper was of great value as a work of reference. From that angle he must agree, though perhaps not so forcefully, with the last speaker. He did feel that Mr. Upton, largely as a result of his daily contact with the subject, had assumed that the reader was equally conversant with it and had failed to make a number of points clear.

To take one or two simple examples, he referred, in connexion with Table II, page 105, to matrix density, but he did not say what it was. He himself happened to know, but a lot of the young men who would read the paper would never have heard the term and might not follow it.

There were other points, and he hoped that in replying to the discussion the author would clear up some of them for the benefit of people who wanted to use the paper as a work of reference.

On more specific points, the author showed in Fig. 9 an almost unbelievably dirty condition on the clean water side of the cooler. He gave an example of what would happen if the sea water side was clean and the fresh water side was in this condition. It was difficult to understand how, with any control whatever of the fresh water side, this state of affairs could happen. He would like to know what were the circumstances.

On water treatment, the author had been either too brief or too lengthy. He had done well to emphasize the need for proper water treatment, but the subject was dealt with so briefly that the reader might be left with somewhat misleading ideas. Tannin inhibitors were not the complete answer, as the author seemed to suggest. He also made one very unfair statement: that under marine conditions control of the fresh water was hardly possible. It was his own experience as between marine and land conditions that on most ships the fresh water side was very well controlled. In the majority of land stations—overseas, at any rate—there might be anything in the fresh water side.

With regard to corrosion of the tubes in the oil cooler, could Mr. Upton say whether these tubes corroded on the oil side were tin plated on that side? It was a well-known fact that lubricating oil, even fresh clean oil, corroded brass, but tin was possibly immune.

This raised another point. Had Mr. Upton any experience with these materials of the effect of detergent lubricating oils? One test of the quality of detergent lubricating oil was how badly it affected copper lead bearings. If it attacked copper lead, he would expect it to attack brass tubes.

There was one section of the paper with which he disagreed considerably, and that was the first part relating to distribution of heat. Possibly this section had been too abbreviated and written too hurriedly; there were some obvious mistakes

* Tinker, T. "Shell-Side Characteristics of Shell and Tube Heat Exchangers". General Discussion on Heat Transfer. London Conference (1951), Section II. Parts I, II and III.

and generalized conclusions seemed to be drawn from one or two specific tests.

To deal with the points in detail, it was stated that fast running engines will surrender more heat to the coolant because there is less radiating surface per pound of fuel burnt. Surely the reason was not the relation between the surface and the quantity of fuel, but the relation between the surface and cylinder volume, which was higher in the smaller engines, and this increased surface volume ratio had a somewhat bigger effect than the decreased surface/fuel ratio. Turbulence was also a very important factor and it was difficult to generalize on this question.

In the next paragraph, it was stated that the stroke-bore ratio had no effect on rejected heat. It was true that the stroke-bore ratio did not have a large effect, but in general short-stroke engines did reject a somewhat higher proportion of heat to the jacket and cylinder heads than did long-stroke engines, again for the reason that with a short-stroke engine the area exposed to the hot gases during the combustion period was higher related to the volume than it was in a long-stroke engine.

The next paragraph referred to a specific test which showed greatly improved fuel consumption at higher jacket water temperature. It was the general experience that there was some improvement in consumption with higher water temperatures, but in general this was very small, and Mr. Calderwood suggested that the 8 per cent improvement recorded in this case was due to the lubricating oil becoming thinner so that the lubricating oil consumption increased very substantially and the reduction in fuel consumption was due to the lubricant burnt. He did not disagree with the principle of high water temperatures, as within reasonable limits they did undoubtedly improve conditions in the cylinder, and in fact the risk of lubrication being destroyed did not occur until a much higher temperature than that mentioned in the paragraph. He would say that so long as the temperature around the piston ring grooves and the cylinder bores, i.e. all lubricated parts, remained below 400 deg. F., the lubricating properties of the oil were not likely to be destroyed.

It was generally accepted that with higher b.m.e.p's the percentage of heat going to the cooling water was less than that at low b.m.e.p's, although the total heat was greater.

With reference to the table that concluded this part of the paper, the figures in general were correct but he felt they would be better prepared in the form of curves, as the table suggested that there was a sudden big increase in the heat going to the jackets and covers between 200 and 250 r.p.m.; also, the table did not suggest to what specific rating the figures referred. The figures appeared to relate to a normal naturally aspirated engine and would not be correct for a pressure-charged engine.

Generally speaking, the header tank in the (C) system was considerably above the water line, and in general sea water pressure from the pumps was low. The result was that actual pressure on the fresh water side would generally be higher than the pressure on the sea water side of the cooler, in spite of the fact that the pump suction was from the cooler. As a result, any leakage was likely to be from the fresh to the sea water side, although the reverse conditions might apply in small ships where the head from the tank was less.

Mr. Calderwood said he would like to refer to Fig. 10, showing an oil cooler fitted on the engine water outlet. Generally speaking, experience had shown that whilst there was no harm in having a high jacket water temperature, it was advisable for the sake of the bearings to keep the oil temperature as low as reasonably possible. Certainly an engine seemed to run more smoothly and quietly with low oil temperature. It might be that this system was intended only to apply to limited specific types of application, but this was not sufficiently explained in the paper.

Finally, Mr. Calderwood again wished to congratulate the author on an excellent and most valuable paper, dealing with a subject about which there was very little published information.

MR. M. S. CROZIER (Associate Member) said Mr. Upton had presented a paper that not only dealt with the heat exchanger but also covered a much wider field.

His own interest was on the maintenance side of heat exchanging and associated installations.

Mr. Upton had mentioned the use of compounds of tannins. He could assure him that they had proved very successful, not only for ferrous but also for non-ferrous metals. He must, however, give a word of warning about tannins. There were hundreds of species and nearly every one of them came in several varieties, so no one tannin would give the protection quoted by Mr. Upton. Great care was necessary in the selection and blending of tannins.

He would like to suggest that the water treatment engineer could save many headaches if he were consulted at the specification stage. He knew from personal experience, particularly abroad, where machinery and installations had been installed before any consideration was given to the type of water available, that there had been many complaints of poor design causing cracked heads, cooler tube corrosion, and so on. He had had to point out that they had nothing whatsoever to do with manufacture. This unfortunately led to criticism of the goods and to the remark, "Give us the work and we will finish the tools!" There were unforeseen difficulties abroad. One came across many cases where the cooling water was also used for making the tea!

Regular cleaning of heat exchange equipment was essential. A friend of his had told him the other day that when he was at sea he shut in on the discharge valve when his coolers were cleaned and as time went on, of course, every time he passed the cooler he opened the valve up a little. In due course, the cooler was badly fouled, probably because of the restriction of the flow in the first place, no doubt, and a major overhaul was necessary. A good spring cleaning gave everybody a feeling of renewed vigour, therefore, so long as they could get someone in to do the work.

MR. G. W. LASCELLES (Member) said that he had only two questions to ask Mr. Upton and they were confined to the section on thermostatic control.

He had usually found heat exchanger manufacturers very loath to discuss this point in any detail. He wondered just how reliable this equipment was now, as no heat exchanger manufacturer appeared able to speak with any assurance about it. As a superintendent engineer, he had frequently been concerned in cases where trouble had been experienced with fast-running Diesel engines. Sometimes the manufacturers expressed the opinion that they had been running too hot and sometimes that they had been running too cold. He supposed the trouble was that they had been running at all! Surely this uncertainty could be removed by the fitting of thermostatic control to the heat exchangers so that the generators would be running at a prearranged temperature prescribed by the manufacturers and to the benefit of all concerned.

The second question was, how close was co-operation in this field between the heat exchanger manufacturers and the Diesel engine manufacturers. Could not something be done in the design stage to fit the thermostatic control, as at the moment manufacturers of both heat exchangers and Diesel engines quite frequently did not even fit a thermometer pocket in the equipment, so that one could not tell how hot or cold was the cooling medium.

He would be pleased if Mr. Upton could reassure him on these two points.

Correspondence

MR. L. BAKER, D.S.C. (Vice-President) wrote that the paper by Mr. Upton was a notable contribution to the TRANSACTIONS as it tackled the question of heat exchanger equipment on a relatively narrow front. In general, apart from new designs, the practising engineer was interested primarily in the applications of equipment rather than the specialized knowledge that went into its design and manufacture. It was, nevertheless, of the greatest value to have before him the fundamental theory and practical background so that he could the more readily assess what improvements could be expected.

He disagreed with the statement that higher coolant temperatures could be accepted by fast moving engines: equally high temperatures could be accepted by slow-running engines if prejudice were overcome. Unfortunately, prejudice was strongly entrenched but it was perhaps of interest to record that most chief engineers operated low-speed engines in the Blue Funnel Fleet at 140 deg. F. for cooling water and lubricating oil, whilst several were running both low- and medium-speed engines with water temperatures of 160 deg. F. with completely satisfactory results. The lower temperatures were, however, a necessity with salt water cooling.

In Table II, Mr. Upton had shown the way to improvements in the performance of the plant and it was to be hoped that he would soon be offering smaller, lighter and cheaper coolers with ¼in. tubes!

The author's comments on water treatments and the use of inhibitors were very timely. There was no doubt that the only really satisfactory inhibitor was sodium benzoate and sodium nitrite; corrosion fatigue tests showed that this treatment in the recommended proportion was as effective as oil as an inhibitor. Oil itself gave slight improvement on air and was far better than any of the more usual inhibitors such as soluble oil, potassium bichromate, etc. The benzoate nitrite treatment had the advantage that it was equally effective with fresh water contaminated by sea water.

The use of steel or cast iron for the cylinders and water boxes was, of course, well known. It would appear, however, that the ideal material today was thin steel cylinder and a water box with a steel tube sheet, the whole being rubber coated wherever corrosion could be significant. A suitable technique existed for applying the rubber to the tube sheet after tubing had been completed and considerable experience had not brought to light a compensating disadvantage!

The author had drawn attention to the need for venting, but whilst "small" is a relative word, it was suggested that the size should not be as small as that usually adopted. One inch on the high points of the system, two inches on shipside fittings, would be very much better than the usual ½-in. tube.

MR. A. K. BRUCE considered that, while it was manifest that Mr. Upton had addressed his most interesting and informative paper to marine engineers, its contents had high significance for users of land engines, particularly where the conditions might be—as they often were—specially arduous. He, therefore, submitted the following comments suggested by a careful reading of the paper.

It was a fairly common practice in land installations to aim at a temperature rise of about 20 deg. F. through the engine jackets, with a maximum outlet temperature of, say, 135 deg. F. Since the amount of heat rejected changes with load, it was necessary to have some means of temperature control, otherwise damage could be done by overcooling the cylinder liners when the engine was lightly loaded. A lightly loaded engine with low jacket water discharge temperature was a sad sight whether at sea or ashore. It could be corrected by regulating the quantity and temperature of the inlet water. In this connexion the concurrent employment of closed-circuit engine cooling with heat exchanger removed the risk of excessive jacket deposits, a risk intensified (where the water was scale-forming) as the temperature rose.

Regarding oil cooler capacity, it was high time for rationalization in this respect, by which he meant some correspondence—in comparable engines operating under comparable conditions—between the engine rating and the size of the oil cooler, expressed in heat dissipating capacity. If the proposals of engine manufacturers were compared, it would be found that there was apt to be a wide disparity as regards declared oil cooler capacity. There was also a wide disparity in the ideas as to how the water circuit was to be arranged and in the provision made for cleaning the oil cooler. The engine user could, of course, protect himself here by insisting not only upon ample oil cooling capacity, but on such installation details as would enable him to keep the cooler clean with the minimum of labour. It happened too often that there was no way of getting at the waterside of the tubes except by breaking pipe joints. This could be avoided by fitting water box covers as shown in Fig. 8 of the paper. Another objectionable arrangement was to have the water side of the oil cooler in series with the engine jackets without there being any means for varying the quantity of jacket water unless by cutting down the water supply to the oil cooler. It should always be possible to put plenty of water through the oil cooler without overcooling the cylinder liners, the reference here being particularly to tropical conditions.

He was happy to note that the author was categorical as to the need to shot-blast the oil-wetted surfaces of cast iron oil cooler cylinders. The need for such a precaution should be manifest when one considered the havoc which could result from the passage of sand particles into the very heart of the engine, particularly where the bearing lining was of a type lacking in imbedibility. Again, they had to remember that in addition to the objections to overheated bearings—arising from whatever cause—hot oil could induce corrosion of the anti-friction lining.

In regard to pressure conditions in oil coolers, insufficient attention was paid, by many engine users, to the need for avoiding risks due to excessive oil pressures when starting up. There was no generally accepted practice, for example, as to the position of the safety (relief) valve or valves on the delivery side of the lubricating oil pump. Sometimes the relief valve was on the up-stream side of the oil cooler and sometimes it was on the down-stream side. A sound practice was to have a safety valve on the lubricating oil pump itself, this lifting only when the delivery pressure would otherwise become excessive. Any relief valve (or pressure regulating valve) was then put on the down-stream side of the oil cooler, so that the engine could get the benefit, all the time, of the heat-dissipating effect of the cooler. Incidentally, recognition of the high pressures which could be imposed on the delivery side of the lubricating oil pump brought into prominence the need for insisting upon a sufficient pressure test on the complete oil cooler before it was passed for service. Perhaps the author would, when replying to the discussion, give some guidance as to the hydraulic test pressure which should be imposed on lubricating oil coolers with cast iron cylinders.

Those who might be somewhat bewildered by the diversity now manifest in grades of fuel oil and lubricating oil, would note with some apprehension Fig. 19 (Plate 3). If trouble from such corrosion had become more marked with extension of the use of heavy fuel oil in main propulsion engines, it emphasized—among other things—the need for complete separation in such engines of the combustion chamber from the crankcase.

In conclusion, it seemed to him that there was need for closer collaboration between engine builders and the makers of oil coolers and other heat exchangers so that the engine user should be spared some at least of the disappointments which arose from inefficient apparatus and/or defective arrangements. There was no designing and manufacturing activity which was more to be desired than activity directed to increasing the security of engine operation and reducing the cost of engine maintenance. In this paper Mr. Upton had, he thought, pointed out some directions along which this activity could prove beneficent.

MR. F. G. VAN ASPEREN (Associate) thought the following points were of special interest in this interesting paper on engine auxiliaries, the right choice of which might add, to an important extent to the success of a ship's Diesel installation.

First of all, after long years of practical experience with the efficient performance of the type of heat exchangers described by the author, he was perhaps allowed to express his satisfaction on finally learning some values of their overall coefficients "U", both for the fresh water and the oil coolers, although the values given still allowed for some variation, depending upon the actual fluid quantities and temperatures adopted in service.

Further constructional indications given in the paper left little room for inquisitive questions as to how the problems of the production of these highly efficient fluid and charge-air coolers had been solved.

In addition to the author's statement that the total amount of heat rejected to the jackets, covers and pistons appeared to be independent of the stroke/bore ratio (S/D) of an engine, it might be stated that the specific heat load to the total surface for a given cylinder volume with a constant piston speed and a constant b.m.e.p., varied in reverse proportion to $(S/D)^{2/3}$ if the specific heat load was represented according to:—

$$T = \frac{\text{b.h.p. per cylinder}}{(\text{cylinder volume})^{2/3}} (\text{van Tyen number})$$

and even more, if the actual surfaces were exactly calculated. So, the engine with a longer stroke/bore ratio was, in this way, not only more advantageous in respect of the covers, but also as regards the jackets, which meant an additional reason for adopting high S/D ratios in uniflow scavenged engines.

In regard to Fig. 3 and the descriptions of the systems of closed fresh water cooling circuits, it might be of interest to mention that often in installations in Continental ships the header tank as indicated in (C) was not directly connected to the suction side of the feed water pump, but was simply arranged with a single wide vertical pipe above a deaerator-bend on the highest point of the main outlet of the engine; also, that the outlets from the auxiliary Diesel engines were being led to this point. In this way there was practically no water flow in the vertical pipe but all air was easily freed from the system. Also, the highest points of the injector and turbo-charger fresh water cooling systems were being led to this header tank.

The paragraph dealing with charge-air coolers was of special interest to builders of modern two-stroke turbo-charged engines. It was not quite clear whether the analysis of several engines, mentioned on page 109, only referred to four-stroke normally aspirated and pressure charged engines or also to two-stroke engines.

In addition to recent publications of measured wall temperatures and heat transfer comparisons between normally scavenged and turbo-charged two-stroke engines, some general information might be given on actual conditions observed on these types of engines.

If, say, the specific fuel consumption per b.h.p. hour decreased by 5 per cent when the power absorbed by direct driven scavenging pumps was partly or totally taken over by the exhaust-gas-driven turbo-chargers, and the percentage of heat from the fuel-input transferred to the cylinder cooling water was reduced from 16 per cent on the normally scavenged engine to 15 per cent on the turbo-charged engine, this meant that $\frac{100}{95} \times \frac{16}{15} = 112$ per cent of the non-pressure-charged output might be reached with the pressure-charged engine, before the amount of heat to the cylinder water was increased. Further increase of power, normally till maximal 35 to 40 per cent, showed a gradual increase of this heat to fresh water, and also the heat rejected in the fresh cooling water of the turbo-chargers (2·2-2·5 per cent of fuel heat input) and that in the sea water of the charge-air cooler (2·5-3 per cent) had to be taken into account.

The conditions on the side of the piston cooling (by oil) appeared to be more advantageous; as the percentage of heat rejected to the lubricating and cooling oil, under the same circumstances as above, decreased from, say, 5 to 4·2 per cent, the pressure-charged engine could have an increased power of $\frac{100}{95} \times \frac{5}{4 \cdot 2} = 125$ per cent against 100 per cent non-pressure-charged output before the oil cooler capacity should be enlarged.

These values were, of course, approximate, but they showed that, as the author already indicated in his introduction, the balance of the heat brought in by the fuel in the pressure-charged engine, above that transferred into effective power and that rejected in the coolants, must remain available in the exhaust gases for utilization, not only in the turbo-chargers, but also, after these, in the exhaust gas boiler.

Finally, it might be stressed that Mr. Upton's paper had given to those involved in the further development of economical Diesel installations much valuable information and subject for further thought.

MR. I. WANS (Member) had found the paper very interesting and full of useful information. The only point he wished to raise was in connexion with the cleaning of the oil coolers. His company had had several cases of oil coolers becoming ineffective due to the oil side of the element getting clogged with a hard black substance and this they had found in several cases impossible to remove. Possibly the author had experienced the same clogging but had been more successful in finding a method of cleaning the elements.

MR. F. SIMONDS (Associate Member) wished to express his appreciation of a very informative and stimulating paper. The paper could very easily have developed into a mathematical treatise on heat transfer; he felt that it was a tribute to Mr. Upton's experience and knowledge of the subject that this had not occurred and that the general interest had been maintained throughout. There were, however, some points on which a word of further explanation was necessary.

In discussing the distribution of heat, he did not quite appreciate the meaning of the statement that the total heat available was largely independent of design. Without becoming too deeply involved in the design characteristics of the engine, the design must anticipate this quantity. Was the meaning that, irrespective of variation in design, the total quantity of heat rejected would be in proportion to the heat potential of the fuel fed to the engine?

Dealing with the pressurized system, he was sure that Mr. Upton, if he could have extended the paper, would have brought in the possibility of utilizing the heat available in the cooling water. This aspect of economic running costs was developing for certain land installations and no doubt, as higher working temperatures became possible, in marine practice they might well see this procedure adopted. The recovery of a major portion of this heat, which represented a substantial percentage of the fuel heat content, was extremely attractive. The author's remarks on this point would be appreciated.

He would remark on the allowance for expansion of the water, which was quoted as 10 per cent; this would appear to be high but, no doubt, could be explained.

With regard to the section dealing with the mathematics of heat transfer, he would suggest that perhaps the explanations had been over-simplified, principally from the point of view that after reading this section one was left with the impression that the material and thickness of the transfer surface had no influence whatsoever on the transfer coefficient. The danger lay not in the application to the heat exchangers under discussion, which had a reasonable consistency of materials, but application to a surface dealing with liquids having low film resistances which would be influenced by a complete change of material.

Although it would have involved extending the paper, the inclusion of a brief summary of the method of determining the economic proportions of a heat exchanger would have been helpful. The figures quoted in Table II were very misleading without a complete explanation of the various factors that influenced the numerical values of the film resistances h_i and h_o; other than the effect of the water rate in oil coolers (Fig. 2) little mention was made of these factors.

The heat exchanger stack (Fig. 9) appeared to be suffering from trouble other than that of its own making and he was surprised that this point had not been made more forcibly, particularly as these accumulations could occur in less accessible points of the circuit.

The condition of the oil cooler tube (Fig. 19) had obviously been given most careful consideration but, presuming that this corrosion had occurred with the tube submerged, the only reason he could see for the linear formation was, perhaps, a form of cavitation due to high local velocities across the tube which could be caused in many ways. Maybe this had been considered and dismissed as impracticable but the possibility of cavitation within the stack warranted a few words from Mr. Upton, perhaps when discussing limiting factors.

The section on charge air coolers was comprehensive and mention had been made of ice formations but the problem of condensation of water vapour, particularly in high humidity areas, must be a limiting factor to the depression that could be exerted on the dry bulb temperature.

The possibility of coring in marine practice would occur only in extreme cases and surely could be overcome by recirculation of part of the cooling water; this would not affect the pumping costs to any great extent as the total quantity would be reasonably constant.

Although the foregoing remarks would appear to be very critical, they did not detract from the high value he placed on the paper.

MR. J. M. SMITH, B.Sc. (Member) considered Mr. Upton's paper to be an admirable survey of the problems encountered in arranging cooling for Diesel engines and that the information given would prove of great value not only in the marine field but also in other applications of internal combustion engines.

The advantages of high jacket water temperatures were rightly emphasized and there was no doubt that increasing attention was being paid to this feature by both builders and users of engines.

As regards the heat distribution figures given in Table I, the figures given for medium to high-speed engines appeared to be rather on the high side, even assuming that the figures were nominal for designing cooling equipment and not exact quantities obtained on heat balance test. Based on his company's experience, he would suggest that they could be reduced by some 10 or 15 per cent.

Water treatment was another aspect of cooling which was now receiving much overdue attention from users and Table III of the paper indicated very clearly the marked rise in deposits from untreated water as the temperature was raised. It was important, therefore, when advocating the use of high jacket water temperatures that full information on the quality of the cooling water to be used should be known in advance so that suitable treatment could be specified.

On industrial installations this problem had been particularly troublesome owing to the wide use of evaporative coolers in the jacket circuit. That the advantages of high jacket temperature were increasingly appreciated was shown by the greater number of plants which incorporated heat exchangers in the cooling system, thus allowing treated water to be used economically.

Mr. Upton made some interesting remarks on the problems of starting with cold oil and rapid warming, but the example shown in Fig. 10, of a system in which the oil cooler was in the water outlet pipe from the jackets, was not good practice. To place the oil cooler in this position meant that the oil must be permitted to reach exceptionally high temperatures or that the jackets must be restricted to a rather low temperature. The oil cooler should preferably be in the jacket inlet side of the circuit and reliance placed on the thermostat bypassing the heat exchanger to give the rapid warming-up effect desired.

The main objection to passing relatively warm water through the oil cooler was that the size and cost of the cooler rose proportionately.

In marine installations the oil coolers might be arranged in the salt water circulation system but with a bypass which enabled them to be cut in or out as required to maintain the desired oil temperature. In engines of the medium-speed class with trunk pistons, warming of the oil was hastened by contact with the pistons and cylinder jackets.

MR. B. G. HOUSEMAN wrote that in his view the way to stop scale forming in cylinder heads in particular was that the system should be a closed one. Too many people until comparatively recently made the mistake of thinking that if the outlet temperature of the cooling water did not exceed 180 deg. F. and provided lots of water was run to waste there would be no trouble with concentrations of salt and so on. In fact, however, in the cylinder head conditions comparable with boiler conditions, so far as scale and corrosion were concerned, were locally obtained. The same, of course, applied with the closed system in a small degree, but the advantage of the closed system compared with the open system was that in the latter it was far too costly to treat the cooling water if it ran to waste and the cylinder heads could only be chipped periodically or the scale removed by chemical descaling.

The introduction of the closed cooling system enabled water treatment to be done economically. This therefore assisted in selling the engine because formerly the cooling water was the last thing to be considered and cracks in cylinder heads and so on were often wrongly blamed, especially overseas, on the design or metal of the engine.

It was unfortunate, however, that some engineers in charge of marine and stationary oil engines did not give the question of cooling water the attention it really deserved, especially in new installations. In the early days engine builders recommended the use of rain water. This, of course, did not contain hardness salts to any extent, but whereas hardness deposits caused cracked cylinder heads and general overheating, soft acid supplies such as rain water (often contaminated from the engine exhaust) were always corrosive and often violently so. He had known many cases where even ordinary soft water supplies, not collected from roofs of buildings, suffered from periodic acid contamination by fumes from chimneys. This had caused the wrong thing to be blamed because the contamination occurred when the wind was only in a certain direction.

Oxidation caused by corrosion could block passages in just the same way as hardness scale, because rust occupied more bulk than the good metal. Distilled water was often used for the same purpose, namely, to stop hardness entering the system, but here again pitting and corrosion had to be guarded against and in view of the fact that oil engine cooling water treatment, stationary, marine and vehicular, might be in the hands of those who were not knowledgeable in the properties of water, he did feel the simplest and most harmless form of treatment should always be employed. This was the method by which blended tannins were employed and water treatment should be considered as part of the installation when it was new, otherwise it was too late to retrieve the reputation of the engine or the system when cracked cylinder heads and so on had occurred.

A very important point in this respect should be remembered, namely, that in boiler installations, so also in engine cooling systems, excessive alkalinities attacked non-ferrous metals. Copper, for instance, was affected fairly rapidly but the addition of suitable blended tannins inhibted this.

While it was necessary to use non-ferrous metals in engine cooling systems in general, it was important also to realize the risk of galvanic action if the wrong metals were in direct contact and in acid water conditions galvanic action was increased, thus, so far as *p*H value was concerned the cooling water should be maintained at a *p*H value of about 9 but not more. In such circumstances the disadvantages of the alkali in respect of non-ferrous metal were inhibited by the blended tannins and the mechanical action of blended tannins in dealing with hardness and fixing oxygen was improved at that *p*H value.

He recommended, however, that as bichromate of potash could of itself become corrosive if contamination of the distilled water occurred, the combination of blended tannins with a suitable alkali was the safer and better method, just as it was in steam boilers.

Again, there were some installations where external treatment should be provided if the size of the installation warranted it. This reduced the cost of after treatment and if external softening was done the after treatment could be confined to blended tannins.

Where marine engines were converted to stationary use, or *vice versa*, he strongly recommended that chemical descaling with inhibited acid solvents should be carried out by an experienced firm using the correct inhibitor for the hydrochloric or other acid, bearing in mind that dissimilar metals existed and that damage could be caused if chemical descaling with inhibited acid was indiscriminately carried out by the unskilled.

As in the case of many modern boilers, it was quite impossible to properly chip an engine cooling system manually.

There was another very important point allied to engine cooling and that was the formation of algæ or weed growth. This could quite simply be dealt with by proper water treatment without the necessity to use copper sulphate and other chemicals which could cause corrosion.

Author's Reply

The author stated he was appreciative of the complimentary remarks made by the contributors to the discussion and should the paper be considered of some value to the members, he was amply rewarded.

Admiral Cowland was thanked for opening the discussion, in which he emphasized the benefits of the closed systems of cooling for internal combustion engines and stressed the need for maintaining reasonable temperature gradients across the engine.

The attack on the liner (Fig. 2) through the action of sea water could well have been a combination of corrosion and some erosion.

He appreciated the comments of Mr. Calderwood and the manner in which he had expanded, corrected and clarified several points in connexion with heat distribution. When referring to the heat surrendered by high-speed engines, he had in mind those radiating surfaces of small engines which gave up heat to the ambient air.

Mr. van Asperen was also thanked for his remarks on heat distribution; in reply to the welcome contribution by Mr. Simonds, the heat surrendered to the coolant systems was a percentage of the fuel consumption, while it was the design characteristics of the engine which determined how this percentage was distributed between the various components—jackets, covers, pistons, etc.

The welcome contribution of Mr. J. M. Smith added emphasis to the consideration of high temperatures, while his comments on heat distribution would be noted with interest.

Mr. McClimont's view that the rôles played by heat exchangers and oil coolers were sufficiently well appreciated would not be wholly shared by those whose business it was to sell engines to owners and operators of coastal craft and the like, particularly overseas.

The contribution and advice of Mr. McClimont on heat transfer was valued, for his remarks did emphasize the great need for care in the utilization of symbols and units in heat transfer formulæ. The term (A_o/A_i) in the expression was included to cover cases where fins, or some other form of secondary surfaces, were bonded to one or both sides of the separating wall and where the areas of such surfaces could be disproportionate to each other. However, when considering marine oil coolers and heat exchangers fitted with plain tubes of normal wall thickness, the areas of the inside and outside surfaces were, for ordinary calculations, sufficiently equal to each other that the term could be regarded as unity and omitted from the equation (3). Mr. McClimont was correct in assuming that in regard to baffle clearance the cases considered were different from American practice so there was no reason to doubt the figures quoted by Tinker when applied to the cases he studied as he was possibly considering more extreme cases.

The expression Mr. McClimont used in regard to the effect of clearance between baffle and cylinder wall was of interest, but there could be other factors which would have to be considered particularly with other forms of baffle arrangement. It was essential in heat exchanger design to ensure that any baffle arrangement directed the fluid in the paths anticipated by the designer.

Regarding the indices used in the oil cooler stability calculations, it was possible that Sieder and Tate actually used 0·14 as the approximation to 1/7th (0·1428) because the difference was so small.

He agreed with Mr. Simonds that while the thickness of the tube wall had in general a negligible influence on the overall heat transfer coefficients in the type of heat exchanger under discussion, it was necessary to treat each type of heat exchange apparatus on its merits, particularly when the tube wall separated gases from fluids.

To keep the paper in bounds it was necessary to suggest specialist books on the subject of heat transfer to those who desired to interest themselves in this complex subject. To make clear a point raised by Mr. Calderwood, matrix density in plain tubular equipment was the number of tubes which could be packed into a given area of tube plate, while in secondary surface heat exchangers it was the amount of surface which could be packed into a given volume.

By reference to inhibitors, Dr. Gilbert, Mr. Calderwood and also Mr. Houseman in his welcome contribution, had all stressed the need for specialist advice on water treatment and in the choice of inhibitors. In the paper, tannin inhibitors were stated to be safe because they had no adverse effect on the metals used in the construction; they had no adverse reaction if salt water contaminated the system, while they had no reaction if, through any leakage, cooling water entered the lubricating oil. They had no corrosive or destructive effect even if used at strengths much in excess of their normal application and, therefore, excess bulk dosage had no serious effect. On the other hand, if they were used at concentrations below normal they would not lead to intensive local corrosion.

The remarks of Mr. Crozier and Mr. Smith would be noted by those interested in water systems and inhibitors, and particularly the point that care was needed when higher water temperatures were considered in association with evaporative type coolers. When higher temperatures were acceptable there could be a strong case for the use of the closed system including tubular heat exchangers, radiators or airblast coolers, for the size of this type of equipment was directly influenced by the mean temperature difference between the mediums.

In reply to Mr. Calderwood on water treatment, the remarks in the paper were meant to be in defence of marine engineers who, in the majority of cases, took considerable interest in the quality and condition of the engine cooling water, but under marine conditions it was not practicable to provide lime soda or base exchange softening plant, both of which could be made available to the engineer in charge of land installations and in any case on board ship there would be no laboratory facilities for full scale testing.

The contributors' remarks did again emphasize the need for engine users to follow the instructions of the engine builder about the quality of the water used in engine cooling systems.

In reply to Mr. Wans, phenolic base solvents had proved satisfactory in many obstinate cases of badly fouled oil coolers.

The heat exchanger (Fig. 9) to which Mr. Calderwood and Mr. Simonds referred, was undoubtedly fouled through lack of appreciation of the elementary fact that even closed water systems, including jackets, pipes, etc., should be clean and precautions taken to ensure that they were in a reasonable condition before ever putting an engine into operation, but in regard to heat exchanger design the important aspect was the danger of reducing the size of tube and the pitch ratio below that which experience dictated as suitable for a particular service.

Both Mr. Calderwood and Mr. Smith questioned the location of the oil cooler in the engine circuit; there appeared to be an increasing request for this arrangement, both here and in America, particularly for trawlers and coastal vessels. The oil cooler could be placed in the circuit, as illustrated in Fig. 10, or between the water pump discharge and the engine, depending upon the temperature difference between the oil and the coolant, which determined the size of the unit required.

In such applications smaller tubes could be employed without serious risk of fouling on the coolant side.

It was invaluable to have the views of Mr. Baker and to know he was encouraging his engine room staffs to run their engines at least somewhere near the temperatures recommended by the engine builders.

There were many heat exchangers and oil coolers available in a variety of combinations in service, fitted with ¼-in. tubes, but it would be a bold experiment to try them on large marine propulsion engines operating at great distances from a home port. The experience reported by Mr. Baker on the protection of steel in contact with sea water would be noted with interest.

Mr. Bruce was thanked for his contribution which, coming from him as a consultant, would be carefully noted.

The position of the relief valve in any oil system was important, but while it was necessary to protect the components and pipes in an oil system, it was also essential that oil did reach the bearings under running conditions.

An oil cooler and the oil pipes should, therefore, be proportioned to pass adequate quantities of oil with reasonable pressure loss even when the oil was cold; where an engine operates over a widely fluctuating speed range and under extremely cold atmospheric conditions, a pressure relief valve was frequently fitted to guard the cooler against high flick pressures. Small oil coolers and engines operating under set load conditions were usually subject to hydraulic test pressures of 60 to 80lb. per sq. in. while larger coolers were tested at 120lb. per sq. in.; the individual tubes were subject to a test of 1,000lb. per sq. in. and normal designs of coolers and heat exchangers were such that they could be tested at much higher pressures if desired.

Admiral Cowland had stated that instability of water-cooled oil coolers could be corrected by changing the viscosity of the oil, but in the case of air-cooled engines fitted with secondary-surface air-blast oil coolers operating in Arctic conditions, coring and instability could occur even with oils less viscous than those normally used for lubrication.

As Mr. Simonds had suggested, there were several ways of preventing the coring or instability in oil coolers and so, having discussed the subject, those engineers in charge of machinery fitted with water or air cooled oil coolers operating in low temperatures, would now be made aware of the phenomena and should instability conditions occur, the symptoms would be recognizable and a solution possible.

He appreciated the contribution of Dr. Gilbert, particularly on the use of cast-iron water boxes and although he had not seen a case of tube plate corrosion attributable to graphitization of water boxes, cases had been brought to his notice where graphitization of cast-iron water boxes had occurred in vessels operating in Eastern waters and it had been necessary to change them to gunmetal.

Regarding the corrosion on the oil side of cooler tubes, he had seen tubes where oil corrosion had occurred, but without penetrating the tube wall. The presence of sea water undoubtedly accelerated this form of attack, but the cause was not primarily due to this condition.

In reply to Mr. Calderwood, an oil cooler of a pre-war ship, in which detergent oils were not used, had suffered from corrosion on the oil side and four tube materials—aluminium brass, 70/30 copper nickel, Admiralty mixture and tinned aluminium brass, were tried but all failed with equal rapidity.

The suggestion of Mr. Simonds that velocity might have had some influence on the corrosion was not supported by experience to date, where the attack had occurred indiscriminately over low and high velocity areas.

It was interesting to learn from Admiral Cowland that a 10 per cent increase in engine output was a modest claim for the installation of pressure charge air coolers.

He was grateful to Mr. van Asperen for his contribution, demonstrating the extent to which the output of a normally aspirated engine could be increased by pressure charging without the necessity of increasing the capacity of the heat exchanger and oil cooler. The systems analysed covered both two-cycle and four-cycle engines.

Mr. Lascelles was thanked for his remarks and in reply it could be stated that reliable thermostats were now more readily available and their application would be an undoubted advantage to the satisfactory running of an engine; it would then be necessary to encourage the engine room staff to accept automatically controlled temperatures and to operate the engines in accordance with the recommendations of the engine builders.

An assurance could be given that close contact did exist between the engine builder and the heat exchanger designer, otherwise it would not be possible to produce a heat exchanger of reasonable dimensions.

There was a wide choice of cooling systems and heat exchanger and oil cooler combinations available to superintendent engineers, so it was desirable and advantageous to study systems and methods of temperature control at an early stage in the development of a specification.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 9th March 1954

An Ordinary Meeting was held at the Institute on Tuesday, 9th March 1954, at 5.30 p.m., when a paper by Mr. H. E. Upton, O.B.E. (Member), entitled "Heat Exchangers for Internal Combustion Engines", was presented and discussed. Mr. Stewart Hogg (Chairman of Council) was in the Chair. Seventy-one members and visitors were present and six speakers took part in the discussion. A vote of thanks to the author was proposed by the Chairman and awarded by acclamation. The meeting ended at 7.50 p.m.

Local Sections

Calcutta

At the Annual General Meeting of the Calcutta Section held on 2nd March 1954, the following officers were elected:—

Chairman: J. Connal (Member)
Committee: H. Kidd (Member)
Y. Arakie (Member)
Capt.(E) T. B. Bose, B.Sc., I.N. (Member)
S. Kasthuri (Member)
—. Morrison
B. Hill (Member).

Arrangements for the papers programme for the coming year for both senior and junior members were discussed.

Following the business meeting, a film on "Research in Engineering", produced by Metropolitan-Vickers and Co., Ltd., was shown.

Merseyside and North Western

At the invitation of Mr. John Lamb, O.B.E. (Member), 140 members and 70 students visited the gas-turbined tanker *Auris* at Cammell Laird's shipyard, Birkenhead, on Thursday, 6th May 1954.

The gas turbine set was running light during the visit, full load conditions not being possible owing to circumstances prevailing at the time. Members were chiefly impressed by the quietness and smooth running of the plant, and by the combustion equipment. Mr. R. M. Duggan, B.A. (Associate Member), was in charge of the demonstration. He, together with the chief engineer and his staff, devoted much time to answering members' questions.

Mr. R. W. Johnson, managing director of Cammell Laird and Co., Ltd., welcomed the members and entertained them to tea in the mess before the tour of the ship. The visit of such a large number of people to the ship in the short time available was only made possible by excellent organization, both on board ship and by the staff of the company ashore.

Southern Joint Branch, I.N.A. and I.Mar.E.

On the 10th April 1954, by the kind invitation of Mr. J. S. Milne, C.B.E., about eighty members of the Branch visited the Cowes, Isle of Wight, yards of J. Samuel White and Co., Ltd. A paper was read by Mr. R. Cousland, M.I.N.A., on "Present-day Construction of Intermediate Naval Class Vessels". This was followed by an instructive visit to the vessels under construction, which included large and small types for the Royal Navy, also R.N.L.I. lifeboats. The whole visit was most interesting and tea was provided at the conclusion.

A visit to the Reserve Fleet at Portsmouth on 29th May 1954 has been arranged for members on the invitation of the Senior Officer, Reserve Fleet.

Annual General Meeting

At their Annual General Meeting, held on Tuesday, 30th March 1954, it was reported that there were now 410 registered members of the Branch. The inaugural dinner held at Portsmouth and the dinner and dance in Southampton were both sufficiently successful to justify a recommendation of the Council that these be annual events.

Papers and lectures presented during the 1953-54 session included the following:—

"Maintenance of Small Craft of Composite Construction", by Commander F. W. Matthews.
"Pametrada Research Station", by H. G. Yates, M.A.
"Stabilisers", by Sir William Wallace, C.B.E.
"The Salvage Association", by L. M. Scaife.
"The Survey of Wood Craft", by J. M. Robertson.
"Prefabrication of Welded Ships' Structure", by R. Cousland.

The average attendance at meetings was about one hundred. An interesting papers programme is being prepared for next session.

The officers of the Branch for the following year are:—

President:	Eng. Capt. G. Villar, C.B.E., R.N. (ret.)
Past President:	R. W. L. Gawn, O.B.E., D.Sc., R.C.N.C.
Vice-Presidents:	K. C. Barnaby, O.B.E., B.Sc.
	J. P. Campbell
	Eng. Capt. W. A. Graham, O.B.E., R.N.R. (Chairman of Branch Council)
	J. A. Milne, C.B.E.
Honorary Secretary:	T. W. Paradise
Honorary Treasurer:	W. J. Ayers
Senior Members of Council:	J. Boylan
	G. M. Kennedy
	J. E. King, C.B.E., R.C.N.C.
	S. Walker
Junior Members of Council:	J. R. Cousins
	B. W. Ramsey, R.C.N.C.
	M. Varvill
	A. F. Weeks, R.C.N.C.

Student Lecture

A meeting of the Junior Section was held at the Institute on Monday, 3rd May 1954 at 6.30 p.m. Mr. R. S. Brett (Associate Member) was in the Chair and seventy-three students and members attended.

A lecture entitled "The Harland and Wolff Marine Diesel Engine" was given by Mr. C. C. Pounder (Vice-President) which was greatly appreciated by all present. Mr. Pounder dealt mainly with a description of the single-acting two-stroke engine, but his audience were very fortunate in hearing of numerous examples of design problems overcome by Mr. Pounder in his long experience as chief designer to Harland and Wolff, Ltd.

The Chairman's expression of thanks to Mr. Pounder for his lecture was endorsed enthusiastically.

Membership Elections

Elected 4th May 1954

MEMBERS

William John Stuart Allen, Lieut.-Cdr.(E), R.N.
Hugh Armstrong
Arthur Champness
Duncan Clark
James Henry Eken
Ernest Redver Head, Lieut.(E), R.N.
Evan R. Jones
Walter Kean
William George Sherwood Lack
Santosh Kumar Paul
Robert Sharpe, Lieut.(E), R.N.
Arthur Webb
Henry James Woodmore

ASSOCIATE MEMBERS

Nicholas Polydoro Nicolaides
John Rorke, B.Sc.

ASSOCIATES

Robert William Abraham
Ralph Charles Baker
William Cairo Beeley
Terence William Billett
David Paul Bond
Robert Gray Caldwell
Basil Clarricoates
Victor James Cooney
Heinz Theodor Ferber
John William Harris
Robert Black Hughes
William Pirrie Leiper
Gordon Francis Lyford
Thomas Macquhae
James Manson
Joseph Christian Miller
Martin Peter Nielsen
David Shirley Pike
Walter Ross
Robert Edwin Slade
Vivian Jacob Meyer Solomon
Herbert Victor Spawls, M.M.
Norman Stanley Swindells
Alan Towse
Arthur Herbert White
John Whittle

GRADUATES

Douglas John Evans
Michael Cameron Rumsby

STUDENTS

Barnes C. J. Christopher, B.Sc.
Philip Henry Inman
Christopher Matthews
Gordon Leslie Winfield

PROBATIONER STUDENTS

Michael John Andrews
Royston Lloyd Baker
Hugh Alexander Boyd
Peter Alexander Stark
Ralph Wardley

TRANSFER FROM ASSOCIATE TO MEMBER

Vincent Richard Ballenger
John Herbert Harkness
David Rhymes Knopp
Percy Henry Martin
William Raymond Mathers
Charles Grenville Parsons, D.S.C.
Harold Frederick Ravenscroft
Royden Douglas Ronald

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Harry Allison Bell, Lieut.(E), R.C.N.
John Edward Philip Cresswell

TRANSFER FROM STUDENT TO ASSOCIATE

John Raymond Cousins

TRANSFER FROM STUDENT TO GRADUATE

John Lloyd Frederick Prickett

OBITUARY

GEORGE EDGAR (Member 2001) was born in 1877. After serving an apprenticeship to engineering he went to sea and obtained a First Class Board of Trade Certificate. He was appointed assistant works manager of the Tredegar Dry Dock, Newport, in 1903, and works manager of the Port Talbot Dry Dock Company in 1912, being promoted general manager later, a position he held until his retirement in 1945. In the 1945 New Year Honours he was awarded the M.B.E. for his services to shipping during the war.

Mr. Edgar was elected a Member of the Institute in 1908.

WILLIAM FREDERICK FLETCHER (Member 11223) was born in 1881. He served an apprenticeship with John Cockeril, Hoboken, William Dickinson and Co., Ltd., Sunderland, and the Selby Shipbuilding and Engineering Company, Selby. He was at sea for seven years and obtained a First Class Board of Trade Steam Certificate before going to Montreal to become manager and president of the Hall Engineering Works; he held this position from 1920-26, when he founded the company of W. F. Fletcher, Sons and Company, consulting engineers, in Montreal, continuing as senior partner until his death on 2nd April 1954.

Mr. Fletcher was vice-president of the Coal Carriers Corporation and surveyor to the British Corporation Classification Society; during the war he was technical director for Canada, British Ministry of War Transport for the United Kingdom. He was a member of the Institution of Naval Architects and of the North East Coast Institution of Engineers and Shipbuilders, and had been a Member of the Institute since 1947.

ROBERT GEORGE GILFILLAN (Member 12543) died suddenly on 15th April 1954. He was born in 1911 and served an apprenticeship with D. and W. Henderson and Co., Ltd., Glasgow, from 1928-33; the next eleven years were spent at sea in ships owned by Glen and Company, Hogarth and Company, Crawford and Company, Thomas Dunlop and Company, and Robert Dunlop and Company. He obtained a First Class Combined Board of Trade Certificate in 1939. In 1944 he joined J. A. Billmeir and Company (Stanhope Steamship Co., Ltd.), serving at sea for four years, then as engineer superintendent until April 1952. From that date until his death he was superintendent engineer to Maclay and McIntyre, Ltd., Glasgow.

CHARLES WILLIAM GLEED (Member 6786) was born in October 1881. He served an apprenticeship with the Stanhope Dock Company, South Shields, from 1897-98, and with Brigham and Cowan, Ltd., South Shields, from 1898-1902. For two years thereafter he sailed as third engineer of the s.s. *Eshcolbrook* and then joined Elder, Dempster and Co., Ltd., Liverpool, serving with them from October 1904 until his retirement in 1940. He obtained a First Class Board of Trade Steam Certificate and, in 1930, a First Class Motor Endorsement. From 1910 he sailed as chief engineer in steam and motor ships. Mr. Gleed died on 14th October 1952 after a short illness.

He was elected to membership of the Institute in 1931.

PETER MACFARLANE MCWILLIAMS (Member 12236) was born in 1913. He served an apprenticeship with William Beardmore and Co., Ltd., Dalmuir, from 1930-35. From 1935 until his death on 16th March 1954 he was employed in ships of the British India Steam Navigation Co., Ltd., as second engineer from 1935. He was a Member of the Institute from 1949.

WILLIAM PATTON (Member 13141) was born in 1890. He served an apprenticeship with Hawthorn, Leslie and Co., Ltd., Newcastle on Tyne, from 1906-11, and then spent three years as a junior engineer with the Indo-China Steam Navigation Co., Ltd., and the China Navigation Co., Ltd. He obtained a First Class Board of Trade Steam Certificate in 1916. From 1916-20 he was a commissioned officer in Inland Water Transport with the Royal Engineers, when he was employed as chief engineer of craft on the River Tigris. From 1922-48 he sailed, for two years as second engineer and thereafter as chief, with H. Harrison (Shipping), Ltd., Ocean Steam Ship Co., Ltd., Loyal Line, Ltd., Cardiff, Avon Steamship Co., Ltd., Bristol, W. J. Tatem, Ltd., Cardiff, P. B. Pandelis, Ltd., Andros Shipping Co., Ltd., Montreal, and Atlantic Shipping Agencies, Ltd. From 1948 until his death on 19th February 1954 he was the secretary of the Marine Engineers' Association, Ltd., in Cardiff.

Mr. Patton was elected a Member of the Institute in January 1951 and was the Honorary Secretary of the Cardiff Section from 1951 until October 1953, devoting much time and interest to the service of the Section and of the Institute.

HENRY J. RAHLVES (Member 10623) died suddenly on 27th February 1954. He was born in 1880 and from 1896-1900 he served an apprenticeship with the Joshua Hendy Machine Works, San Francisco. In 1900 he served on the Army Transport *Thomas*, obtaining a third assistant's certificate; for the following year he was junior engineer with the Oceanic Steamship Company. From 1903-11 he sailed in ships of the Standard Oil Company of California, as chief engineer from 1906, and for the next six years he supervised the construction of this company's ships. From 1917-19 he was the general superintendent of vessels under construction in the Seattle area for the United States Shipping Board, and the following year he was at Portland as superintendent of hulls for the G. M. Standifer Corporation. In 1920 he was construction inspector at New Jersey for the Standard Oil Company, and from 1921-33 he was employed as assistant manager of the marine department of Imperial Oil Co., Ltd., Toronto. From then until his retirement in 1945 Mr. Rahlves was manager of the marine departments of the Imperial Oil Company and International Petroleum Co., Ltd. After his retirement he was president of the Park Steamship Company for two years, and president of the Oriental Trade and Transport Company of Toronto. During the second World War he was in charge of Canada's only ocean tanker fleet of fourteen Canadian lake vessels and other ships in the South American coastal trade.

Mr. Rahlves was a member of the executive committee of the Shipping Federation of Canada and a member of the Royal Canadian Institute, Toronto. He had been a Member of the Institute since 1946.

JOHN C. RUSSELL (Member 7844) was born in 1891. He served an apprenticeship with McKie and Baxter, Glasgow, from 1908-13, and then went to sea, serving for a year with John Glyn and Company of Liverpool, and twelve years with the White Star Line. He obtained a First Class Board of Trade Certificate. In 1932 he was appointed mechanical superintendent of Vizagapatam Harbour Construction. Mr. Russell was elected a Member of the Institute in 1935.

GEORGE STELL (Member 10484) was born in 1896. His apprenticeship was served with John Robson (Shipley), Ltd., of Shipley, from 1912-16. From 1916-19 he served in the Royal Navy as an engine room artificer and from 1919-21 as installation engineer with the Invincible Marine Oil Engine Company of Keighley. From 1921-23 he was test shop engineer with William Beardmore and Co., Ltd., Coatbridge, and in 1924 he returned to sea, serving with Owen and Watkin Williams and the British Molasses Co., Ltd. In 1928 he joined the British Tanker Co., Ltd., and remained in their employment until his death on 25th November 1953, when he had been a chief engineer for many years. He obtained a First Class Board of Trade Motor Certificate in 1931. Mr. Stell was elected to membership of the Institute in 1945.

ARTHUR EDWIN STOCKDALE (Member 5756) was born in 1886. He served an apprenticeship with Smith's Dock Company, North Shields. Later he was works manager and director for twelve years with the Yorkshire firm of Livingstone and Cooper, shipbuilders and engineers, but at the end of the first World War he set up his own business as consulting engineer and naval architect. He became senior partner of the Liverpool firm of Hay and Smart, Ltd., consulting marine engineers and ship surveyors.

Mr. Stockdale was a fellow of the Society of Consulting Marine Engineers and Ship Surveyors and served as a member of their council from 1947-50; he was a member of the Institution of Naval Architects, and had been a Member of the Institute since 1927.

GEORGE TORRIE (Member 10816) died in India on 27th February 1953. He was born in 1899 and served a five-years' apprenticeship with Hawthorns and Co., Ltd., Leith. All his sea service was with the British India Steam Navigation Co., Ltd., and in 1936 he was appointed assistant engineer to the Mazagon Dock Company, Bombay. From 1938-41 he was assistant superintendent engineer with the British India Steam Navigation Company. After an illness he returned to sea service with the same company, sailing as chief engineer in their ships until his death.

Mr. Torrie was elected to membership of the Institute in 1946.